EVALUATION OF THE FLUIDIZED BED COMBUSTION PROCESS

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VOLUME III APPENDICES

EVALUATION OF THE FLUIDIZED BED COMBUSTION PROCESS
SUMMARY REPORT

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PREFACE

The results of the evaluation of fluidized bed combustion for steam/power generation are presented in this three-volume report. The report identifies fluidized bed fuel processing systems which should meet both market requirements and air pollution abatement requirements and are likely to be cheaper than alternative, conventional systems. A program is recommended for commercializing promising processes.

This volume (Volume III) contains the detailed market survey reports, boiler design reports, pressurized boiler combined cycle power plant report, and support studies on design and operation prepared by Westinghouse and its subcontractors. The scope of the work, technical evaluation, comparisons, conclusions, and recommendations are contained in Volumes I and II.

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ELECTRIC UTILITY MARKET SURVEY

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PURPOSE

The purpose of this survey is to identify the magnitude, characteristics, and requirements of the future utility boiler market as a guide for determining design parameters in the development of the fluidized bed boiler. Its scope entails a forecast of the fossil steam market including combined cycles, a review of the availability and cost of fuels, and an economic assessment of alternative sulfur removal systems.

SUMMARY

Approximately 50% of future annual electric utility generation additions will require boilers of one form or another. The market for fossil steam generation in the United States is expected to show a constant upward trend of 4 to 5% annual growth rate through 1985, compared with a total electric utility annual growth rate of 7%. The fossil steam segment will grow mainly on the strength of increasing intermediate generation requirements as the fossil base-load market is displaced by nuclear power. Intermediate additions burning coal, natural gas, and oil will represent over 80% of the fossil steam market by 1985.

Two dominant size classes of new fossil units have emerged. The 700 to 1300 MW size class consists primarily of coal-fired units in the coal-producing areas, while the other size class, encompassing all intermediate capacity and some base-load capacity in the 400 to 500 MW range, will extend toward the smaller sizes when combined cycles come into greater use.

The advent by 1975 of the packaged combined cycle plant employing a steam turbine which utilizes the waste heat of two gas turbines is expected to precipitate the capture of half of the annual intermediate additions in the mid 1980s. Up to 90% of the larger gas turbines may be retrofitted with boilers and steam turbines. Distillate

oil, and eventually residual oil, will fuel approximately two-thirds of the combined cycle installations unless coal gasification becomes significant during the next decade.

The need for greater reliability and flexibility requires each utility to determine the operating characteristics best suited to its particular system's requirements. Operating problems appear to be centered in the boiler area due to the lack of design maturity and operating experience of larger, high-pressure, once-through boilers. For cyclic operation, utilities generally favor proven, low-pressure, drum type boilers, while the systems approach to utility economics also shows a tendency toward units with shorter expected life.

Although a growth in demand for fossil fuels will accompany the continued growth in the demand for electric energy, a variety of economic pressures will impede the ability of the coal and gas industries to meet the rising requirements for these fuels. Labor problems, new health and safety regulations, and the lack of new mine capacity will continue to plague the coal industry and limit production into the mid 1970s, even though almost unlimited reserves of this natural resource are available. The natural gas industry is faced with a diminishing gas reserve and regulated price levels that make the investment in exploratory drilling for new supplies unattractive. At the same time, the availability of adequate gas reserves that can be tapped to meet the nation's gas requirements is questionable. Oil is the only fossil fuel abundant enough to meet the electric utility demand adequately; however, its use is limited to those markets adjacent to deep waterways unless transportation by pipeline over considerable distances proves economically feasible.

In the utilization of fossil fuels for power generation, annual demand for coal will increase at a declining rate after an upward surge in the early 1970s as nuclear units are placed in operation following long delays. Coal-fired capacity installed in 1985 will equal only 60% of that added in 1970. Gas-burning steam plants, which will maintain a fairly steady rate of installation through the 1970s, will be limited primarily to gas-producing areas in the South Central United States, while areas remote to gas fields show a decreasing gas market share. As a result of the gas shortage and the difficulty of getting gas contracts, some utilities in gas-burning states are considering the installation of oil-fired plants, as well as coal, lignite, and nuclear plants. Continued shortages and increased usage of gas may ultimately result in end-use control, rationing, and restrictions on the use of gas in boilers. With the recent influx of low-cost, low-sulfur, residual oil imports along the Atlantic coast, many eastern utilities have converted their existing coal-fired plants to oil and anticipate the addition of new oil units, especially in the New England area. More than half of the annual fossil steam additions in 1985 will be oil-fired.

RECOMMENDATIONS

Analysis of fossil steam market trends suggests two courses of action as the most advantageous in the development of fluidized bed combustion in electric utility power plants.

The first recommendation is the development of a 600 MW, coal-fired fluidized bed boiler with steam conditions of approximately 2400 psig/1000°F/1000°F. The choice of capacity and steam conditions is based largely on the ability of a plant with these characteristics to operate as both intermediate and base-load generation. Such a plant would be intended for cyclical operation initially; however, with advanced technology and proven reliability it could eventually be used as base capacity. Sizes smaller than 600 MW would not be as adequate for base-load generation if load factors of 65 to 80% could be achieved, since they do not take full advantage of economies of scale. Poor performance by conventional 800 MW units installed to date casts doubt on the successful operation of larger units at this time.

Coal is recommended as the primary fuel for a number of reasons. In spite of limited coal production caused by new coal-mine health and safety regulations and an inadequate mine work force, abundant reserves of coal are available for extraction. While the coal industry is domestically controlled, the oil market is strongly influenced by foreign political policy which is subject to erratic and unpredictable fluctuations, as demonstrated by the recent unexpected reduction in world crude supply. The instability of the oil market is in contrast to the long-term contracts established between utilities and coal companies, who are no longer willing to supply coal on a short-term contract as they had in the past.

In the more distant future another important aspect of the coal versus oil question is the effect of world demand on oil availability. Energy use is increasing at a higher rate in the rest of the world than in the United States, as evidenced by the higher worldwide per capita energy consumption rate and the soaring global population. As more countries use more of their own fuels at an increasing rate, especially oil, the United States will be forced to rely more on domestic fuels, of which coal is the only one with adequate reserves to meet the future requirements of the fossil market.

The second recommendation is that the fluidized bed process be reviewed for its competitiveness with nuclear power as base load generation after initial operating experience. In this long-range activity the feasibility of larger unit sizes -- approximately 1200 MW -- should be thoroughly examined from a technical and economic standpoint. A generation cost of 7.9 mills per kwh is projected for an initial 1200-megawatt nuclear plant for operation in 1980 at a capital cost upwards of \$270 per kw and a fuel cost of 18 cents per million Btu. To provide power at a comparable cost, the fluidized bed boiler plant using coal at 45 cents per million Btu will require a capital investment of approximately \$170 per kw; for a minemouth plant with less expensive coal at 40 cents per million Btu, a capital cost of $195 per kw will be competitive with nuclear in 1980. The fuel cost is an obvious key factor in determining the competitiveness of these two types of generation. However, even with a high coal cost the possibility of attaining $170 per kw is excellent if economies of scale prove applicable to the fluid bed concept in larger units.

The likely effects of fluidized bed combustion on the fossil steam market depend on its degree of success relative other emissions control devices and on the excalation in fuel prices. There is little doubt that pollution regulations will get stricter in the future and that a workable and economical method of pollution control will be developed to satisfy these restrictions. Because of the uncertainty of the fuel picture, a precise estimate of the fluid bed boiler plant proliferation in the fossil steam market is extremely speculative. One can say, however, that if successful it will capture all of the presently predicted coal-fired additions regardless of fuel price. If the divergence in oil and coal prices is less than predicted, as now appears likely, the fluidized bed process will gain a sizable portion of the intermediate oil market, especially after 1980, when oil-burning combined cycle plants were otherwise foreseen to dominate this market.

In the fossil fuel market, this new concept of coal-fired generation will certainly have a favorable effect on the future of coal in the United States. The introduction of the fluidized bed combustion process will stimulate the demand for coal but will have little effect, if any, on the projected price of coal, since the coal demand already exceeds production. The current production shortage stems from the inability of the coal industry to attract sufficient miners to meet the demand and its understandable unwillingness to open new mines except on a long-term basis. The fluid bed boiler should help to remedy the situation on both counts. The fact that the present contract of the United Mine Workers expires in the near future, combined with the present overall fuel shortage, gives the UMW a strong bargaining position for higher wages. The combination of a better wage package and the safer conditions in new mines opened for fluidized bed plants should attract the necessary manpower in new mines to meet the future production demands. At the same time the development of power plants which burn coal exclusively may prompt utilities to establish new long-term contracts with coal companies, and may lead them to supply a portion of the capital necessary for opening new mines.

In any event, the fluidized bed combustion process will ultimately assure the continued growth of the coal industry in the United States.

FOSSIL STEAM GENERATION MARKET

Introduction

In the electric utility industry, generation units are ordered four to six years prior to scheduled operation. As a result, the market for capacity additions through 1975 can be projected with considerable accuracy. The forecast of the more speculative market from 1976-85 is based on a number of factors, including trends in annual utility peak loads as related to the U.S. Gross National Product, in the generation service mix (base, intermediate, and peaking), in the availability and price of fuel to utilities, and in power generation technology. In areas where quantification by mathematical extrapolation is not applicable, predictions were based on subjective judgment seasoned with experience. In any case, the information presented here is not to be taken as an edict of what will happen in the future, but as an indication of our best estimate of what the situation will be, based on our present knowledge.

Annual Fossil Steam Installations

The fossil steam generation market will continue to grow between 4 and 5% per year through 1985. The annual additions to capacity will average 20 to 25 million kw per year throughout the period. Figure A-1 shows the total and base load annual fossil steam additions from 1970 through 1985. The total fossil market shows an upward trend

U.S. ELECTRIC UTILITY INDUSTRY
ANNUAL FOSSIL STEAM INSTALLATIONS
1970 - 85

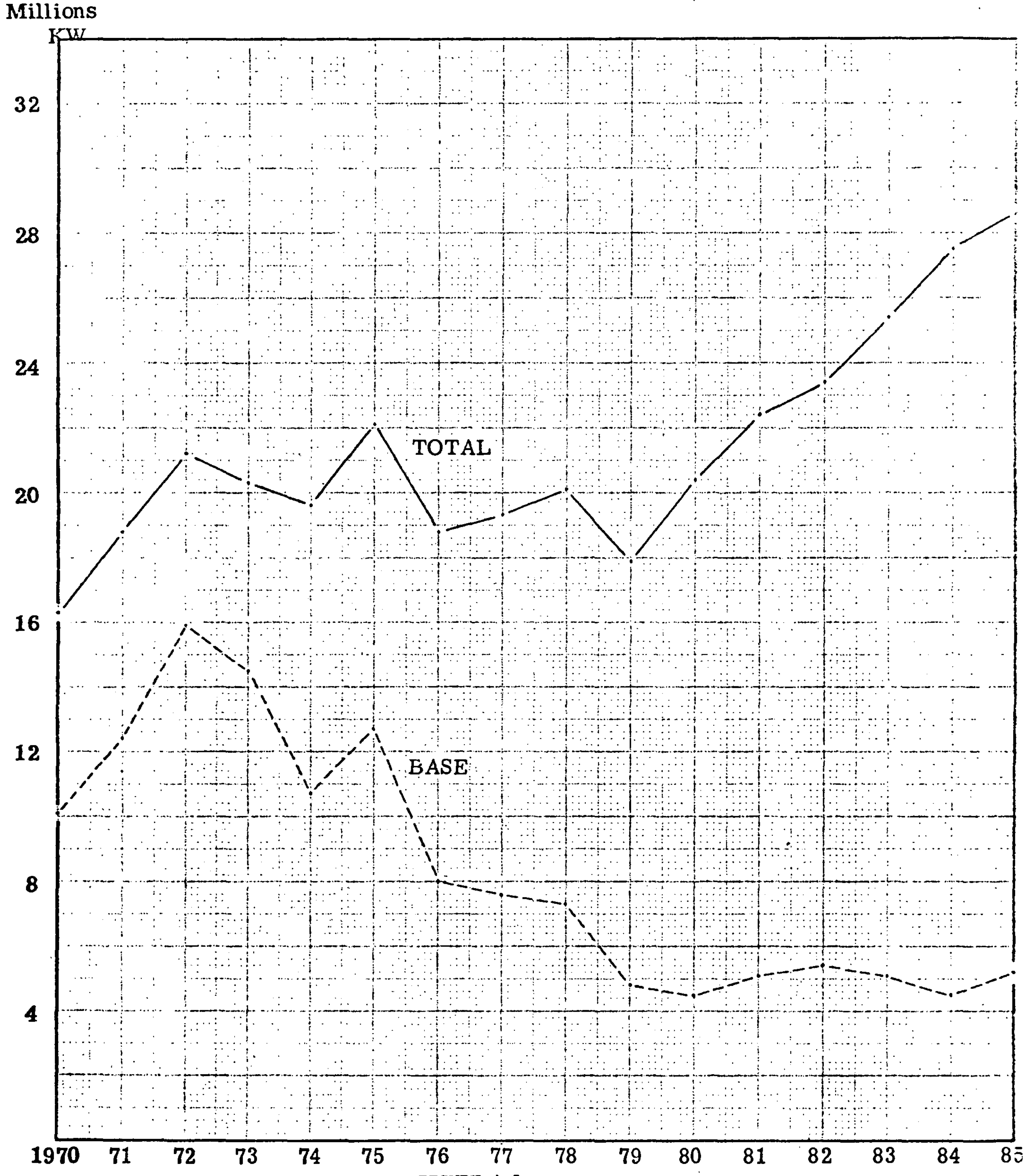


FIGURE A-1

over the sixteen-year period with an average increase in annual additions of 770 MW over the preceding year. Installations in the late 1970s level off as a consequence of overbuying by utilities in the late 1960s for installations in the early 1970s. While nuclear power is expected to supply most of the base load requirements for the nation after 1976, utilities located in coal-producing areas will add base load fossil plants representing 10 to 15% of the thermal market. In Figure A-1 these base fossil steam additions show a marked decline from 1973 to 1979 and then stabilize at approximately 5 million kw through 1985.

Cyclical operation steam plants, including possible combined cycle plants, are expected to account for 43% of the fossil steam market in 1975, 78% in 1980, and 82% in 1985. Steam cycling plants will represent 40 to 45% of the total thermal market. This new intermediate capacity has the characteristics for low or medium load factor operation and less initial cost than the more efficient base-load capacity.

Unit Sizes

Historically, the unit size distribution for fossil turbines has been fairly uniform. However, in the late 1970s and early 1980s, there will probably be two dominant size classes of fossil units. One class will be in the range of 700 to 1300 MW and will grow at the same rate as the maximum unit size. The other dominant size class will be in the 300 to 500 MW range and will be oil- and/or gas-fired, with a small portion coal-fired, at approximately 2 million KW per year.

The maximum size fossil turbine-generator has doubled every five years from the 1950s through the mid 1960s. However, it appears that the rate for the mid-1960s through 1985 will be more moderate. The maximum size of 1300 MW to be installed in 1973 is expected to remain the largest fossil unit until possibly 1983, when a 1500 MW unit can be expected. This diminishing trend toward larger sizes results primarily from the decreasing gains from economies of scale and minor technological improvements.

From the 1950s to the early 1960s, the median unit size doubled every ten years, essentially following the utility load growth. However, in the period from 1964 to 1967, the median unit size almost doubled in a three-year period. This reflects the trend to joint planning and mergers or anticipated mergers. The rate of change from 1967 through 1975 is very low, again reflecting the trend to cyclic steam units.

In the size class under 200 MW, the last few years have demonstrated an electric utility market of 1500 MW per year. This market is expected to maintain this level until the advent of combined cycles.

Combined Cycle Market Forecast

A new type of utility power plant -- the packaged combined cycle plant -- is just emerging, and in fact, is still in the developmental stage. In these plants as they are now conceived, a gas turbine exhausts into a boiler which generates steam and feeds a steam turbine. The plants are usually 50% gas turbine power and 50% steam turbine power, so they vary from combined cycle plants that have been used in the past. In most cases, additional heat must be added between the gas turbine and the waste heat boiler. The present Westinghouse combined cycle plant under development uses two 60 MW gas turbines and one 120 MW steam turbine for a total plant rating of 240 MW. These will be installed to serve as intermediate generation, which has a forecasted annual capacity factor of 30 to 50%.

The initial combined cycle plants are being designed to burn natural gas; but developments are already underway to introduce distillate oils and later residual oils as the fuel for these plants. Thus, combined cycle plants can be expected to burn two types of fuels in the future: natural gas and oil. As time progresses, oil will become the predominant combined cycle fuel. The split of the fuel will be on the order of two-thirds oil-fired and one-third natural-gas-fired.

The market for the combined cycle plant is expected to appear in about 1975 and 1976, partially because of the development time required for the equipment in these plants. Starting from a low level of 500 to 1000 MW a year, the market will increase to approximately 25 to 50% of the total intermediate generation market. The actual magnitude of the market will, of course, depend on the ultimate cost of these combined cycle plants and the economics that they hold in operation. Since these two parameters are still estimated on early data at this time, it is hard to define accurately the likely magnitude of the market.

From an initial plant size of 250 MW, combined cycle units are expected to grow to approximately 500 MW, corresponding to improved gas turbine technology and growth of gas turbine unit sizes.

The boilers used in these plants are packaged boilers; that is, shop-assembled and shipped for field erection. They are designed for high air flow and for forced circulation of water/steam. Since the firing is external to the boiler itself, there are no burners near the heat transfer surfaces. The plants presently under design use one waste heat boiler for each gas turbine. The steam conditions are in the area of 1200 to 1300 pounds pressure per square inch and 900 to 1000°F temperature.

In addition to packaged combined cycle plants, there is a market for retrofitting boilers and steam turbines to presently installed gas turbines. A great many electric utility gas turbines have been installed in the last few years, and this trend will continue for the next year or two. The 25 and 50 MW gas turbines exhaust a great deal of useful energy to the atmosphere. In the future, utilities can logically be expected to add boilers and steam turbines to these gas turbines to recover and utilize this energy. These applications will be, in all likelihood, only on those gas turbines installed at major steam plants; that is, it is not expected that waste heat boilers and steam turbines would be added to gas turbines installed at remote locations on the system. Possibly 80 or 90% of the larger gas turbines ultimately will be converted to combined cycle plants.

These retrofitted combined cycle plants can have many configurations. However, as with the packaged model, the most promising will probably be a waste heat boiler with supplementary firing feeding a steam turbine of approximately the same rating as the gas turbine. There will be cases where supplemental firing will not be applied; in these cases, the steam turbine rating is approximately one-half of the gas turbine rating. In this latter application, a header system could be installed between several gas turbine waste heat boilers to feed a single steam turbine of larger size.

Boiler Fuel Mix

The pattern in boiler fuels for future fossil steam installations will change significantly from the trend of the past decade. During the 1960s coal was the primary boiler fuel for two-thirds to three-quarters of all new fossil steam capacity; gas, for about one-fourth of new fossil capacity; and oil, for less than one-tenth of new fossil capacity. Additions in 1970 reflect this situation, with 75% of fossil capacity being coal-fired, nearly 25% gas-fired, and less than 1% oil-fired. Tables A-1 and A-2 show the response of utility buying to the changing fuel picutre in the United States. Annual coal-fired installations will drop to 63% of fossil capacity in 1975, to 34% in 1980, and to only 25% in 1985. Oil-fired additions, on the other hand, will climb to 12% of fossil additions in 1975 to over 50% in 1985, while gas-fired units will fluctuate between 23 and 31% of fossil capacity through 1985. Installations in 1980 are expected to show nearly equal proportions for each type of fossil fuel.

An explanation of the forces behind these trends is contained in the next section, Fossil Fuels for Utility Power Generation.

TABLE A-1

ANNUAL FOSSIL STEAM INSTALLATIONS
By Type of Service
1970-1985

YEAR IN SERVICE	TOTAL	BASE	INTERMEDIATE
1970	16.3	10.1	6.2
1971	18.8	12.4	6.4
1972	21.2	15.9	5.3
1973	20.3	14.5	5.8
1974	19.6	10.7	8.9
1975	22.1	12.7	9.4
1976	18.8	8.0	10.8
1977	19.3	7.6	11.7
1978	20.1	7.3	12.8
1979	17.9	4.8	13.1
1980	20.4	4.5	15.9
1981	22.4	5.1	17.3
1982	23.4	5.4	18.0
1983	25.4	5.1	20.3
1984	27.5	4.5	23.0
1985	28.6	5.2	23.4

Units: Millions KW

TABLE A-2

ANNUAL FOSSIL STEAM INSTALLATIONS
By Type of Fuel
1970-1985

YEAR IN SERVICE	TOTAL	COAL	GAS	OIL
1970	16.3	12.2	4.0	.1
1971	18.8	12.8	5.3	.7
1972	21.2	13.9	3.6	3.7
1973	20.3	11.9	5.5	2.9
1974	19.6	9.0	7.8	2.8
1975	22.1	13.9	5.6	2.6
1976	18.8	11.2	5.3	2.3
1977	19.3	10.6	5.6	3.1
1978	20.1	10.2	6.6	3.3
1979	17.9	6.7	6.0	5.2
1980	20.4	6.9	6.3	7.2
1981	22.4	7.1	6.9	8.4
1982	23.4	7.6	6.6	9.2
1983	25.4	7.1	6.5	11.8
1984	27.5	6.3	6.5	14.7
1985	28.6	7.3	6.4	14.9

Units: Millions KW

Operating Characteristics

Future operating characteristics are difficult to specify explicitly because of variations in utility needs and practices. However, several qualifications in this regard can be made.

Electric utilities are concerned about the potential difficulties of keeping generation reliability up, and fuel and maintenance costs down to tolerable levels. The operation of generating units can no longer be scheduled as a matter of simple economic choice but must rather be treated as a matter of operating flexibility of currently installed generating units seem to be centered in the boiler area rather than the turbine generator.

The low maturity factor of larger supercritical and subcritical once-through boiler installations in some instances has led utilities to favor proven drum-type boiler designs. The choice of steam pressure for these installations is a function of economics and unit size. Trends in turbine inlet pressure ranges shown in Figure A-2 exhibit a tendency of fossil steam generation toward greater use as intermediate capacity and lesser use as base capacity.

It should be kept in mind that boiler component designs for low capacity factor versus base load operation are not much different. Cycle comparisons to determine economic choice of throttle pressures for various lifetime capacity factors indicate that as the capacity factor on a system increases, higher pressure, lower heat rate cycles are justified by increased fuel cost savings. Utilities, however, cannot

U.S. ELECTRIC UTILITY INDUSTRY
ANNUAL FOSSIL STEAM INSTALLATIONS
BY TURBINE INLET PRESSURE

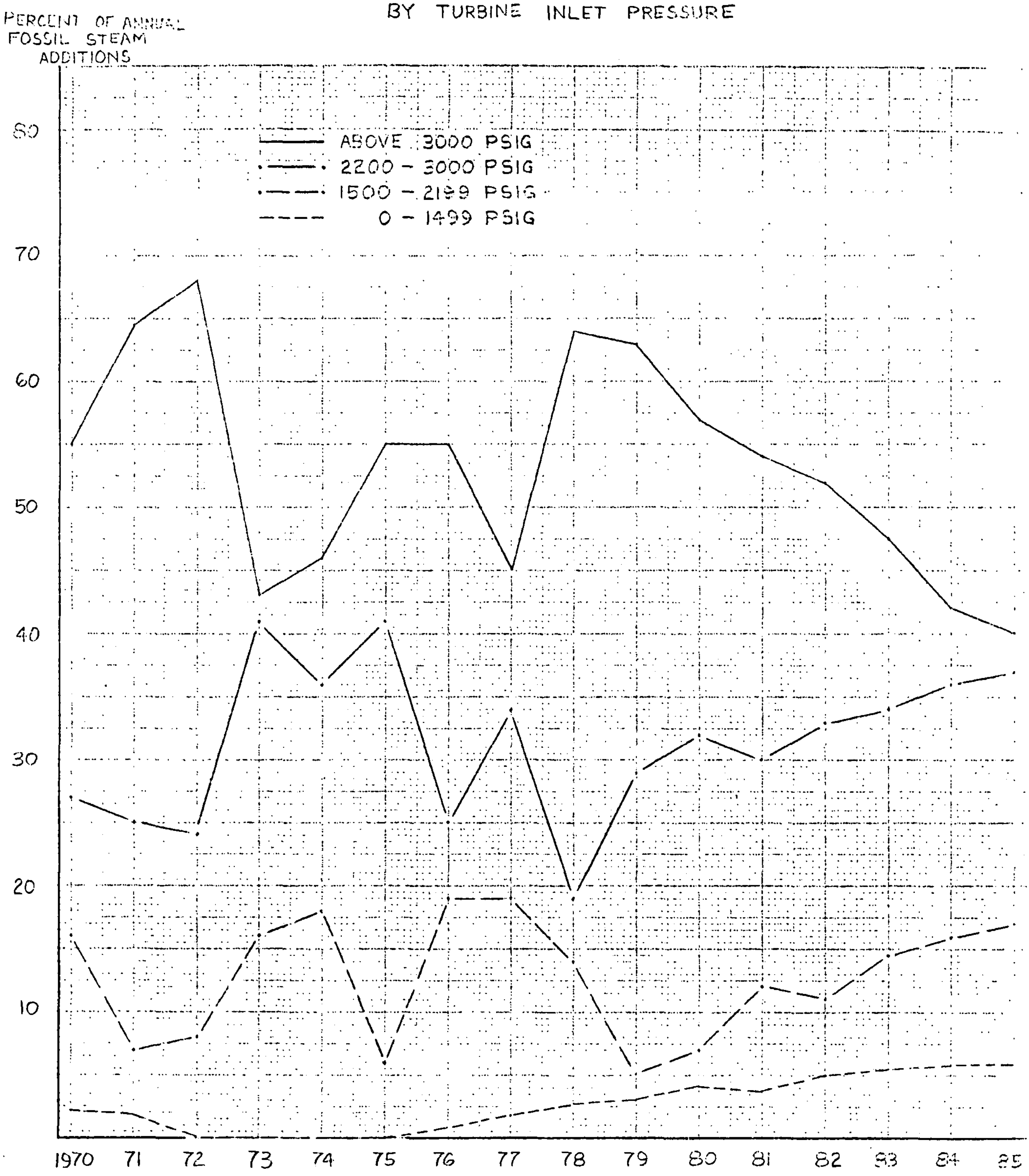


Figure A-2

establish a practical operating load factor for a specific installation. In general, the operating flexibility and particular utilization of a specific plant is determined by special operating procedures implemented by advanced control and automation systems.

Operating characteristics quoted by manufacturers -- such as cold start-up, hot restart, minimum load operating point, operating criteria (variable versus constant pressure), and rate of load change -- vary substantially among manufacturers. These performance characteristics are considered firm only after verification by utility operating departments.

Many operating problems on large boiler installations reflect the lack of design maturity and operating experience.

Expected Life of New Installations

Increasing emphasis on evaluations of plant economy and system planning has brought about a change in philosophy of many utilities toward units with a shorter expected life than units previously installed for a similar type of service. Several advantages inherent in the tendency toward shorter-life machines make it economically desirable. The system becomes more adaptable because the utility can keep its distribution of installed sizes and generation mix more closely in line with the optimum distribution corresponding to the utility's particular load characteristics and system growth. Capital costs tend to be lower for shorter-life machines. Although the capital cost for two 15-year units is more than that for one 30-year unit, the difference is usually off-set by lower operation and maintenance costs and lower fuel costs (due to better heat rate) in the second 15-year unit compared to the second fifteen years of operation of the 30-year unit. Shorter-life units also permit the utility to take earlier advantage of technological improvements.

FOSSIL FUELS FOR UTILITY POWER GENERATION

In 1968, approximately 81% of the kwh generation in the United States was produced by the use of fuel, with the balance produced primarily by hydro-electric generation and a small amount of geothermal steam generation. There are four basic electric utility fuels: coal, residual oil, natural gas, and nuclear. Figures for 1960 and a forecast of their relative position in 1970 show recent changes in percentage of capture by fuel:

RELATIVE SHARE OF ELECTRIC UTILITY STEAM GENERATION
(% of Steam Generation)

FUEL	1960	1968	1970
Coal	66.3	61.9	59.7
Gas	26.0	27.6	25.9
Oil	7.6	9.4	9.4
Nuclear	.1	1.1	5.0

Fossil Fuel Reserves

Coal Reserves

The coal reserves in the United States are tremendous; the actual coal underground approximates 1.5 trillion tons. However, not all of this is economically recoverable under present-day technology. Assuming that bituminous coal seams at least 3-1/2 feet thick are recoverable, that sub-bituminous coal seams at least 10 feet thick are recoverable, and that only 50% of the seam is economically recoverable (because of coal pillars left for roof support), we arrive at a figure of approximately 265 billion tons of known, economically recoverable coal in the United States today. Approximately 70% (based on tonnage) of these reserves are located west of the Mississippi River. Based on caloric value, approximately 45% of the reserves are located west of the Mississippi River.

Gas Reserves

At the end of 1968, the proved recoverable gas reserve still in the ground in the United States amounted to 287 trillion cubic feet. Proven reserves are those reserves in which the field has been definitely mapped and tested for quantity. In addition to proven reserves, there are potential and probable reserves which would add approximately one-thousand trillion cubic feet to the gas supply in the United States. However, these reserves are not properly identified and quantified and may not be as extensive as estimated or may not be economically recoverable. Therefore, in considering a fuel reserve supply, conservatism dictates that only the proven reserves be mentioned. These reserves can be increased, at greater expense, either by the importation of liquified natural gas or by the manufacture of gas from domestic coal supplies. Neither of these methods is economically competitive at the moment with the natural gas production in this country. However, recent long-term contracts have been signed for LNG delivery to the East Coast. The price range (40-55¢/MB) prohibits use for power generation.

Oil Reserves

Because of its ease of transportability over water, oil is truly an international fuel. Hence, the reserves for the United States, the balance of the free world, and the Communist Bloc countries, have been indicated in the following table. The proven reserves left in the United States amount to 31 billion barrels of economically productive oil. An interesting fact to note is that present technology in oil production allows the recovery of approximately 30% of the oil contained in oil sands. Hence, if we take the 85 billion barrels of oil already produced in this country plus the 31 billion barrels expected to be produced and consider it as 30% of the total oil in the country. this would leave an oil reservoir of approximately 385 billion barrels remaining in the oil-producing strata in the continental United States. Obviously, oil-producing companies have spent and are continuing to spend a significant research effort to discover methods to increase the yield of oil from the oil-bearing strata. However, in assessing fuels at the present time, we must consider only the economically and technically recoverable fuel, which leaves the proven reserves at 31 billion barrels in this country. The same reasoning applies to the proven reserves for the balance of the free world and the Communist bloc.

PROVED RECOVERABLE FOSSIL FUEL RESERVES IN 1968

	PROVEN RESERVES	ANNUAL CONSUMPTION
Coal (Tons x 10^9)		
U.S.	265	0.5
Gas (MCF x 10^9)		
U.S.	287	19.9
Oil (bbls x 10^9)		
U.S.	31	4.78
Balance free world	381	7.57
Communist bloc	59	2.08
Prudhoe Bay Field (Est)	20-30	-

Factors Affecting Future Production of Coal

Unfavorable Factors

Several major factors are a deterrent to the increased production of coal. One of the most serious obstacles facing the mining companies is the lack of manpower. Not only has productivity per man-day suffered reverses due to temporary labor unrest, but more important, the mining companies are having difficulty enticing younger miners and professional people to work in the mines. The numbers of mining engineers recently graduated from the universities have fallen short of the demand for mining engineers in the mining industry. More experienced miners prefer to remain miners or federal inspectors under the Department of Interior rather than accept jobs as mine foremen. As a result, mining companies are suffering shortages in mine management and mining engineering.

A second deterrent to production of coal is the capital cost of opening deep mines. This cost is currently running between $10 and $15 per annual ton of capacity. With the current cost of capital, the potential shortage of adequately trained personnel and wide fluctuations in the price of coal make the investment in new mines highly speculative.

A third, more recent deterrent is the strict mine safety legislation recently passed by the U.S. Congress. Not only will this increase the capital cost of the equipment in a mine, but it has the potential to reduce the output per man-day because of the additional safety precautions necessary to protect the health of the miners. This may be a significant cost increase factor in the production of coal.

A fourth area that will create serious problems in the use of coal in the utility industry are the potential air pollution requirements, particularly those relating to sulfur dioxide emissions. Most of the steam coal east of the Mississippi River has 2-1/2% or higher sulfur, which means that it cannot be burned in conventional electric utility boilers within the proposed air pollution regulations. Thus, the coal either has to be treated to remove the sulfur, or the sulfur dioxide must be removed from the flue gas. The alternatives are covered in another section of this report.

Favorable Factors

There are also several factors favorable to the potential future production of coal. Mining processes are continually being improved through mechanization. Methods of continuous mining and cutting machines, rock dusting apparatus, and gas detection systems are being improved with new technology. In addition, the application of long-wall mining techniques is finding use in certain parts of the coal fields. This technique, originally developed in Europe, has a moving cutter along the working face of the mine which may be 600 to 900 feet long. This cutter works continuously, with the cut coal conveyed to the mine face by endless belt conveyors. The productivity per man-day can be at least tripled by using this system. The cost of production is further reduced because no permanent roof supports are used, and the roof of the mine is allowed to collapse behind the mining operation. Perfections of this technique are expected to produce 200 tons per man-day of effort, versus the current average of 20 to 25 tons per man-day of effort.

Coal firms are now able to obtain long-term purchasing contracts from utility companies. These contracts enable the coal companies to obtain the capital necessary to open large deep mines. Previously the practice has been for most utilities to buy coal on a monthly or annual basis on the open market, which did not assure a specific coal producer a continuous market for his coal. With the use of long-term, life-of-mine, commitments, they can adequately plan their production, capital, and manpower to utilize the most efficient production methods.

The last -- and one of the most important -- favorable factors is the tremendous growth in the use of energy in the United States. This growth of energy requirements will tax all types of energy-producing methods, oil, gas, coal, and nuclear. Therefore, the coal industry has an assured growing market for its product over the long term.

The following figures compare the present consumption of coal with the forecasted consumption in 1980. Note that the use by utilities has a greater growth rate than the industry as a whole. Also note that the compound annual growth rate by utilities is less than the utility power production growth rate of 7.8% per year. This is the result of coal's being replaced by nuclear and oil in future utility planning.

U.S. COAL CONSUMPTION (MILLIONS OF TONS)

	1969	1980
Total U.S.	571	793
Electric Utility	310	508
E.U. % of U.S. Total	54.0%	64.0%

ANNUAL COMPOUND GROWTH RATE

	1960-69	1970-80
Total U.S.	4.0%	3.3%
Electric Utility	6.8%	5.0%

Factors Affecting Future Production of Gas

Unfavorable Factors

The use of natural gas for various heating applications has increased rapidly since World Ware II. Concurrently, the discovery of new gas reserves in the continental United States continued to increase until 1968, when the proved reserves actually decreased by 5.6 trillion cubic feet because of the continued high United States requirements.

The FPC has jurisdiction over gas use by virtue of its regulation of interstate pipelines and is able to control the prices paid for gas at the well-head. The gas industry believes that current well-head prices are too low to encourage new gas discoveries, and new gas discoveries in 1968 and 1969 indeed have occurred at a lower rate than in previous years. All of this makes the outlook for new findings of natural gas reserves in the United States unfavorable as this point in time.

There are two main sources of natural gas production: gas wells and oil wells. The economic incentive to do exploratory drilling for gas fields has been retarded by FPC policy. However, the production of gas as a by-product of oil is not ruled by the same economics. Approximately 6000 SCF of gas is produced with each barrel of crude oil production in the U.S. The production rate of crude establishes the gas production from the source.

Favorable Factors

Gas is a clean-burning fuel. It reduces or eliminates most air pollution problems in urban centers and is an excellent fuel for certain process industries -- such as the glass industry -- and as raw material in the chemical industry. In addition, gas is a very convenient fuel for residential heating and cooking and has gained a strong preference as a fuel among consumers. The industrial market accounts for 48% of gas consumption; the residential market accounts for 36%; and the utility power generation market accounts for only 16%.

Natural gas can be supplemented by the importation of liquid natural gas and by coal gasification. However, the present-day economics of these two sources of gas preclude their use in the electric utility power generation market. Hence, one can conclude that gas supplies for markets other than electric utilities will be adequately satisfied from one of several sources, while the supply of gas for power generation will be limited to natural well-head gas. The FPC must approve use of natural gas for boilers if the gas is carried in an interstate pipeline. The present policy of the FPC is to discourage the burning of natural gas under boilers unless other benefits to the public are paramount, such as reduced air pollution in California.

U.S. GAS CONSUMPTION (TRILLION SCF)

	1969	1980	1990
Total U.S.	19.9	30.0	38.5
Electric Utility	3.2	4.2	5.0
E.U. % of Total	15.2%	14%	13%

ANNUAL COMPOUND GROWTH RATE

	1960-69	1970-80	1980-90
Total U.S.	5.2%	3.2%	2.5%
Electric Utility	7.2%	2.5%	1.8%

The use of gas for power generation accounted for approximately 15% of total natural gas comsumption in 1969; the utility percentage will decline slightly through 1990. However, this forecast is subject to many factors: a national policy on energy consumption, FPC price regulation, and the proving out of the probable gas fields in the continental U.S. Any adverse action by these factors will reduce the utility gas consumption at a faster rate than any other segment of the gas market.

Factors Affecting Future Production of Residual Fuel Oil

Unfavorable Factors

The production of domestic residual oil is almost non-existent because the domestic refineries are designed to produce all of the valuable components of the crude that are available, leaving only the barest residual. This is dictated by the economics of the market place and the price of domestic crude. However, foreign refineries face a different problem in that their markets for the different petroleum products are much more limited. Hence, they make no effort to "crack" the crude into other products, and from 40 to 60 percent of the barrel of crude remains as residual. Therefore, the only significant source of residual oil for utility power generation is from foreign sources. To date the government has restricted the importation of oil, crude or otherwise, to approximately 12% of domestic production. However in District I, the Eastern Seaboard, these restrictions do not apply to residual which is imported for use as a fuel. This restriction has also been temporarily lifted in District V, the West Coast, so that low-sulfur Indonesian residual could be imported to relieve the air pollution problems in California. As of this writing, application has been made by a mid-western utility for relief so that they can convert an in-city coal plant to oil for pollution reasons. However, one of the major deterrents to the use of residual nationwide is that the cost of transportation, other than by large sea tankers, is extremely high.

Another factor which will unfavorably influence future costs is the problem of oil slicks in harbors and along the beaches. Stricter safety regulations for tanker operation coupled with international enforcement will increase the cost of sea transportation of oil. This factor cannot yet be quantified, but it will be severe.

Favorable Factors

Several factors tend to make residual an attractive fuel for utility use. First, the governments of Venezuela, Libya, Iraq, Iran and Nigeria are very aggressive in promoting the sale of their crude through the companies they have licensed to drill for the oil. Their major source of government income is the oil royalties obtained on production. Thus today, at least, there is an abundant supply of crude and/or residual from foreign sources. Secondly, the SO_2 air pollution problem can be alleviated by reducing the amount of sulfur in the fuel before combustion. This can be done either by using naturally low-sulfur African residual or by desulfurizing the high-sulfur Venezuelan residual. Both options are being exercised in the eastern utility fuel market. A third advantage is the extremely low transportation cost over water -- as low as 3.1¢/million Btu/1000 miles.

A substantial portion of generation capacity -- approximately 35% -- is required to operate in a cyclic fashion, that is, it must produce power for the daytime, weekday loads with little or no output during low load periods. Large, high-pressure, once-through coal boilers are not suited for this service. The utilities are now starting to procure lower pressure, drum type (400 MW) boilers for this duty. Oil or gas is much preferred to coal for this service because of ease of control. In addition, residual oil, with special restrictions on sulfur and vanadium, may prove an excellent fuel for gas turbine peaking service as well as combined cycle intermediate service. Residual oil has many advantages as a fuel for power generation, i.e., ease of storage, economical water transportation, abundant foreign supplies, and relative ease of start-up and shutdown of boilers.

RESIDUAL FUEL OIL CONSUMPTION
(Thousands barrels/day)

	1969	1973	1980
Total U.S.	1800	1923	2123
Electric Utility	480	600	800
E.U. % of Total	26%	34.5%	38.0%

ANNUAL COMPOUND GROWTH RATE

	1960-68	1969-80
Total U.S.	3.5%	1.5%
Electric Utility	10.0%	4.6%

The utility use of residual has been growing at a faster rate than the U.S. total residual market. In fact, the utility segment of the residual market is the only segment which shows any growth potential. Note also that the projected annual growth for utility consumption declines in 1969-80, reflecting the growth of nuclear power.

Utilization of Fossil Fuels for Power Generation

Changes in the use of coal, gas, and oil in the electric utility industry, as well as changes in the fuel situation in general, are caused by factors such as supply, demand, government regulations, pollution, nuclear power delays, fuel production, and fuel transportation costs.

The annual demand for coal by the utilities will increase through 1980, but the rate of growth will decline. The growth in coal requirements through 1976 is a result of the delays in scheduled nuclear plant additions and the need for generating capacity to meet the increasing demand for electricity in the early and mid-1970s. The addition of coal-fired plants for intermediate generation accounts for the increases in coal requirements through the end of the decade, but the commitment of nuclear plants for base-load generation will reduce the growth rate of utility coal demand.

The most significant change in the nation's fossil fuel market has occurred along the Atlantic coast. Almost unlimited amounts of low-cost, low-sulfur residual oil imports have become available to the utility companies along the East Coast. Faced with shortages in their coal supplies, rising coal prices, and legislation limiting the sulfur content of their boiler fuels in metropolitan areas, the East Coast utilities have been a very receptive market for the petroleum companies. Many have converted their existing coal-fired plants to burning imported residual oil as the primary boiler fuel, and additional fuel conversions of existing plants are scheduled. A number of eastern utilities, apparently not concerned about a non-domestic, politically-influenced energy source, are purchasing single-fuel, oil-fired 300 to 500 MW generating units. All indications are that the petroleum companies will continue to penetrate the eastern electric utility fuel market successfully.

To 1985, gas is expected to remain the primary fuel for electric generation in the gas-producing states and contiguous areas, with most of its growth confined to this area. The bulk of electric utility burning gas plants in the future will be added in Louisiana, Texas, Oklahoma, New Mexico, and Kansas, which produce 90% of the gas in the continental United States. In regions far from gas production, gas is expected to decrease its share of the market, especially after 1975. Much of the historic growth of gas use has occurred outside the area where gas is produced. It is in these regions with high fossil fuel costs that nuclear and oil power will make the largest gains, tending to reduce the overall growth of gas use in the utility fuel market. As a result of the gas shortage and the difficulty of getting gas contracts, coal, as well as oil, may penetrate outside its traditional consuming areas. This trend is already in evidence; the past few years have seen new coal-burning plants built in Florida, New Mexico, Texas, Nevada, and Kansas.

PRICE TRENDS OF FOSSIL FUELS FOR POWER GENERATION

ELECTRIC UTILITY FUEL PRICE TRENDS

It can be unequivocally stated that the days of cheap, readily available supplies of fuel for power generation are over. Gone are the days when competition in the electric utility fuel market is governed exclusively by price. While each of the fuels is basically competitive with the others in the electric utility marketplace, the major factor influencing the market price of each is different. These major factors --

coal	-	the cost of production,
oil	-	alternative market opportunities,
gas	-	government regulation, and
nuclear	-	the cost of preparation --

are, however, being mutually influenced by artificial, political and environmental constraints that are already reshaping the price, supply and demand relationships that characterized the electric utility fuel market in the 1950's and early to mid-1960's. The patterns that have been formed in the recent past, along with continuing political and environmental pressures, will affect the future utility fuel market and fuel price trends over the next 15 years.

COAL PRICE TRENDS

Coal prices in the electric utility fuel market began to climb in 1966, following almost a decade of decline. The increases in the price level for coal began over two years prior to the "energy alert" of 1969 and 1970. The increases were due primarily to improvements in coal productivity no longer being able to counter rising labor and material costs. This rising trend has continued as the labor-intensive coal industry experiences a very unstable labor climate. Labor unrest has combined with the provisions of the 1969 Mine Health and Safety Act to reduce deep-mine productivity by 15 to 20%, and conversely increase operating costs. These rising production costs plus rapidly increasing coal demands resulted in further increases in utility coal prices in 1969 and 1970, but coal industry officials state that those increases were insufficient. The continuing effects of the Mine Health and Safety Act, a new three-year contract with the UMWA, escalation not being offset by substantial improvements in productivity, and attempts by coal companies to increase their return will result in the continuing rise in electric utility coal prices.

ELECTRIC UTILITY FUEL OIL PRICE TRENDS

Historically, the fuel oil available for use in electric utility boilers was the residual waste product of the U.S. refineries, representing only 6-7% of each barrel of refined crude. Crude oil was considered too valuable to burn directly as a power plant fuel and was instead refined into high-priced petroleum products. The residual oil was functionally priced to compete with coal or gas on a limited basis. By comparison, foreign refineries permit residual yield of 40 to 55 percent, and in 1969 the availability of an apparently unlimited supply of imported, low-cost residual oil was promoted for power generation in PAD District I. Residual oil became a viable competitor in the East Coast electric utility fuel market as competition between oil companies brought a sharp reduction in the price of imported residual oil -- high-sulfur resid prices bottomed out at 28¢/MMBTU in mid-1969. However, as a result of supply and demand pressures, the price began to rise rapidly in the fall of 1969. Newly enacted air pollution regulations and tremors in the politically volatile African and Middle East oil-producing countries caused further increases in the market price of residual oil in 1970. The supply of oil for the U.S. market was reduced, tanker capacity was over-extended, and premium prices were being demanded for low (1%) sulfur characteristics. A return to reasonably stable political conditions and an agreement between the oil companies and the Organization of Petroleum Exporting Countries (OPEC) relieved the "shortage" in the world oil supply and tanker capacity, and in 1971 the price of residual oil began returning to a market level more in line with a balanced supply and demand condition. More widespread and restrictive environmental regulations, however, continue to bring a premium for low-sulfur oil, with 0.3% sulfur content becoming the most stringent limit established for boiler fuels. This highest quality oil will demand the highest premium price.

It is expected that environmental regulations limiting the sulfur content of boiler fuels will have a marked impact on the oil industry's relationship to the electric utility fuel market. High-sulfur residual fuel oils will find less demand under tightening environmental regulations, and the price will be comparable to that of high-sulfur mine-mouth coal. The lucrative market for low-sulfur oil has presented the oil refiners with the economic incentive to begin installing expensive desulfurization facilities for producing clean, heavy distillates from the high-sulfur feedstocks of the Caribbean and Middle East. The magnitude of the added cost of desulfurization is determined by the sulfur content level desired and the characteristics of the feedstock used. While the current high demand for low-sulfur fuel oil with a limited supply available is creating an artifically high differential price between high-and low-sulfur oil, the differential will become more directly related to the actual difference in desulfurization costs when the many new processing facilities begin operation and the supply of low-sulfur oil increases.

With the rapidly rising oil prices in the electric utility fuel market and the high cost of desulfurization, the oil companies have begun the unprecented marketing of crude in the U.S. directly as an electric utility boiler fuel. Continuing success of selling crude directly as a boiler fuel is dependent on the crude having a low sulfur content -- African and Indonesian crude comes closest to meeting environmental sulfur regulations -- and the price being high enough that the return on the basic crude is greater than the net return from the sale of the petroleum products refined from that crude. It is estimated that this price level on crude is just slightly less than the market price of residual oil of the same sulfur content since crude has some safety disadvantages compared to the equivalent high-quality heavy distillates. The crude has a low flash characteristic since the gasoline and naphtha have not been removed in processing.

Over the long term, the oil industry will continue to functionally price its products in the electric utility market place. Fuel oil prices will be established commensurate with the market value of alternative fossil fuels based upon their heat content, environmental characteristics, convenience, and availability. In the 1970's, low (0.3%) sulfur resid and crude will demand premium prices due to the imbalance between supply and demand for low-sulfur fossil fuels, while in the 1980's it will approach the market value of natural gas, the most desirable of the fossil boiler fuels. Fuel oil for the electric utility market with greater sulfur contents will demand less of a premium but will still command a good price due to its value as feedstock for processing into higher quality boiler fuels.

NATURAL GAS PRICE TRENDS

Natural gas has gone from being a nuisance by-product of domestic crude oil production to a position as the premiere of the electric utility boiler fuels. The Federal Power Commission, by regulating the price which gas producers can charge the interstate transmission and distribution companies and the resale price to local distributors, indirectly controls the price of interstate natural gas to the electric utility industry. Intrastate sales of gas are not subject to FPC regulation, but both inter- and intra-state transactions are mutually influenced. After many years at depressed levels, well-head gas prices have recently been increased by the FPC, and additional increases are expected. More severe price increases are being experienced for non-jurisdictional gas in the electric utility marketplace. With the shortage of deliverable gas, the FPC is also indirectly controlling the end-use of natural gas by requiring curtailment plans that provide for electric utility boiler application to be the first use of natural gas to be discontinued in the event of a supply deficiency. This presents the utilities with interruptible interstate gas supplies and rising gas prices that will reduce consumption to premium uses for peaking and intermediate cycling.

Whether directly or indirectly controlled, natural gas will be priced in accordance with its value as a premium fuel rather than on production cost. The price of natural gas paid by electric utilities is expected to increase substantially, doubling 1971 new gas prices by 1980. In spite of the sharp price increases, the quantity of natural gas available to the electric utilities will be severely constricted.

NUCLEAR FUEL COST PROJECTIONS

Each of the major categories of manufacturing costs for the nuclear fuel cycle ,

Uranium	-	$U_3 O_8$ Raw Material
Conversion	-	Converts $U_3 O_8$ to UF_6
Enrichment	-	Increases U_{235} Content in UF_6
Fabrication	-	Manufactures Fuel Assemblies
Reprocessing	-	Reprocesses the Spent Fuel Assemblies and Recovers Unused Fuel,

can be analyzed to produce a cost breakdown which can then be examined for the effects of cost escalation for the period of 1971 to 1985. Each of these fuel cycle cost factors can be apportioned into percentages for labor, material, and a firm non-variable cost. By using the past ten years for an historical base, the index for the labor and material components of each fuel cycle step was projected on a straight-line basis. In addition to mathematical cost projections, other economic factors were considered in projecting the nuclear fuel cycle costs. The supply and demand pressures on uranium prices; the volume sensitivity, automation, and learning-curve cost improvements in the fabrication process; the volume dependency of reprocessing costs; and the governmental restraints on the cost of enrichment were evaluated.

Nuclear fuel costs were calculated for a base load nuclear plant with all of the fuel cycle components escalated and the cost of enrichment at the legally established ceiling after escalation. This most conservative projection shows that nuclear fuel costs increase slightly from 20.18¢/MMBTU in 1971 to 22.10¢/MMBTU in 1985. This increase over the 15-year time period results from escalation factors in the nuclear fuel cycle beginning to overtake the cost - volume improvements in the fabrication and fuel recovery components of the cycle.

ELECTRIC UTILITY FUEL PRICE PROJECTIONS 1971-1985

STEAM COAL

Deep-Mined Mine-Mouth

YEAR	¢/MMBTU	INDEX
1968	16.7	100.00
1969	17.8	106.58
1970	21.3	127.54
1971	26.2	156.88
1972	30.0	179.64
1973	32.0	191.61
1974	33.4	200.00
1975	34.2	204.79
1976	35.3	211.37
1977	36.6	219.16
1978	37.9	226.94
1979	39.2	234.73
1980	40.6	243.11
1985	48.6	291.01

RESIDUAL OIL

Deep Water Port Contract Cargos
1968 High-Sulfur Resid Price = 100.00

	High-Sulfur Resid		Low (1%) Sulfur Resid		0.3% Sulfur Resid	
YEAR	¢/MMBTU	INDEX	¢/MMBTU	INDEX	¢/MMBTU	INDEX
1968	32.0	100.00				
1969	34.0	106.25				
1970	46.0	143.75	72.0	225.00		
1971	43.0	134.37	60.0	187.50	75.0	234.37
1972	43.5	135.93	61.0	190.62	76.0	237.50
1973	44.0	137.50	64.0	200.00	78.0	243.75
1974	44.5	139.06	67.0	209.37	82.0	256.25
1975	45.0	140.62	70.0	218.75	85.5	267.18
1976	45.5	142.18	72.0	225.00	88.5	276.56
1977	46.0	143.75	73.0	228.12	90.0	281.25
1978	46.5	145.31	74.0	231.25	91.0	284.37
1979	47.0	146.87	74.5	232.81	92.0	287.50
1980	47.5	148.43	75.0	234.37	93.0	290.62
1985	50.0	156.25	77.0	240.62	94.5	295.31

CRUDE OIL

Deep Water Port Contract Cargos

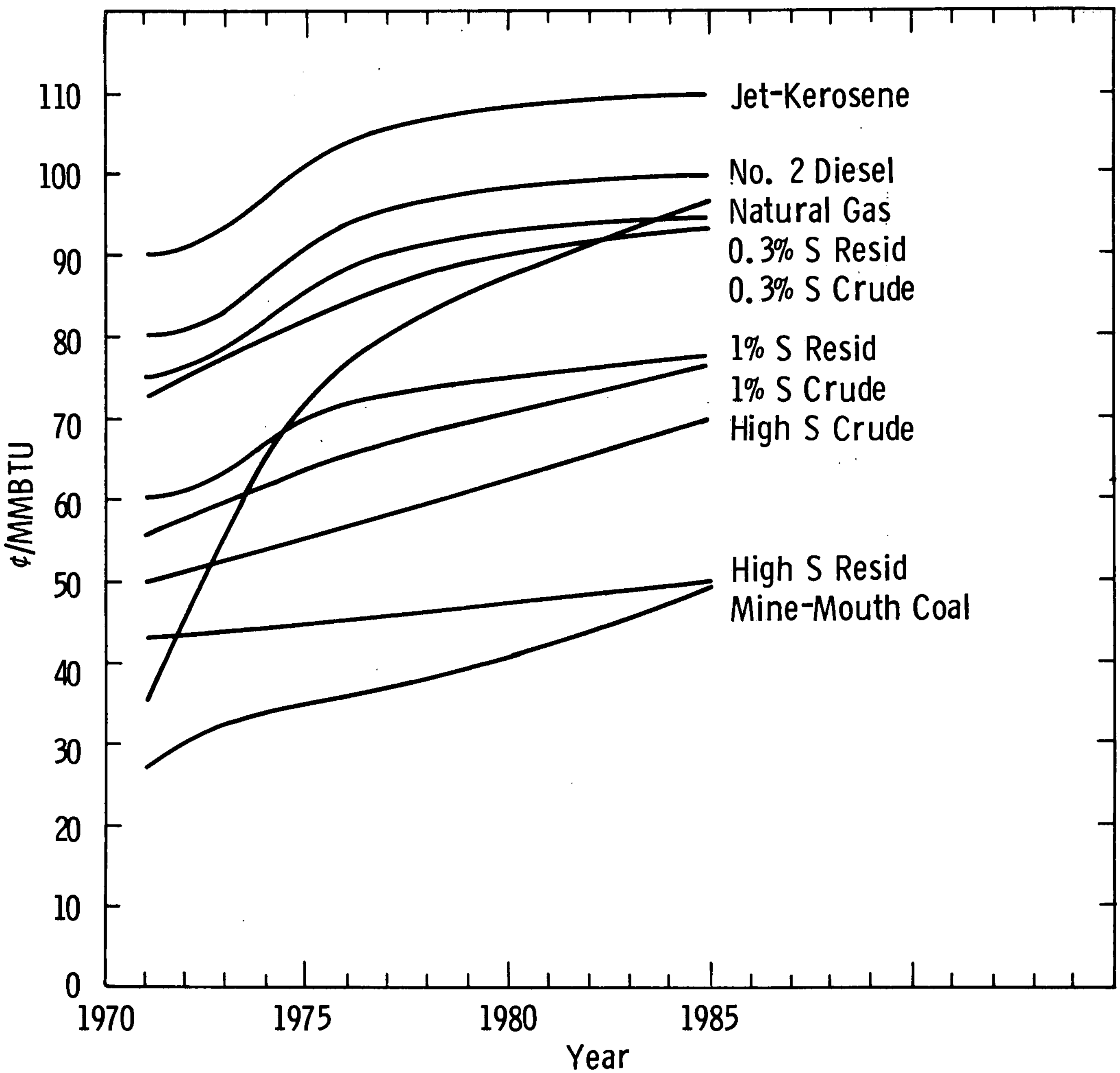
1971 High-Sulfur Crude Price = 100.00

	High-Sulfur Crude		Low (1%) Sulfur Crude		0.3% Sulfur Crude	
YEAR	¢/MMBTU	INDEX	¢/MMBTU	INDEX	¢/MMBTU	INDEX
1968						
1969						
1970						
1971	50.0	100.0	56.0	112.0	72.5	145.0
1972	51.5	103.0	58.0	116.0	75.0	150.0
1973	53.0	106.0	59.5	118.0	77.0	154.0
1974	54.5	109.0	61.5	123.0	79.5	159.0
1975	55.5	111.0	63.5	127.0	82.0	164.0
1976	57.0	114.0	65.5	131.0	84.0	168.0
1977	58.5	117.0	67.0	134.0	86.0	172.0
1978	60.0	120.0	68.5	137.0	87.5	175.0
1979	61.0	122.0	69.5	139.0	88.5	177.0
1980	62.5	125.0	70.5	141.0	89.5	179.0
1985	69.5	139.0	76.0	152.0	93.0	186.0

NATURAL GAS
Electric Utility Price

YEAR	¢/MMBTU	INDEX
1968	25.1	100.00
1969	25.4	101.19
1970	27.0	107.56
1971	35.0	139.44
1972	45.0	179.28
1973	55.0	219.12
1974	65.0	258.96
1975	72.0	286.85
1976	77.0	306.77
1977	80.0	318.72
1978	83.0	330.67
1979	85.0	338.64
1980	87.0	346.61
1985	97.0	386.45

Curve 645771-A



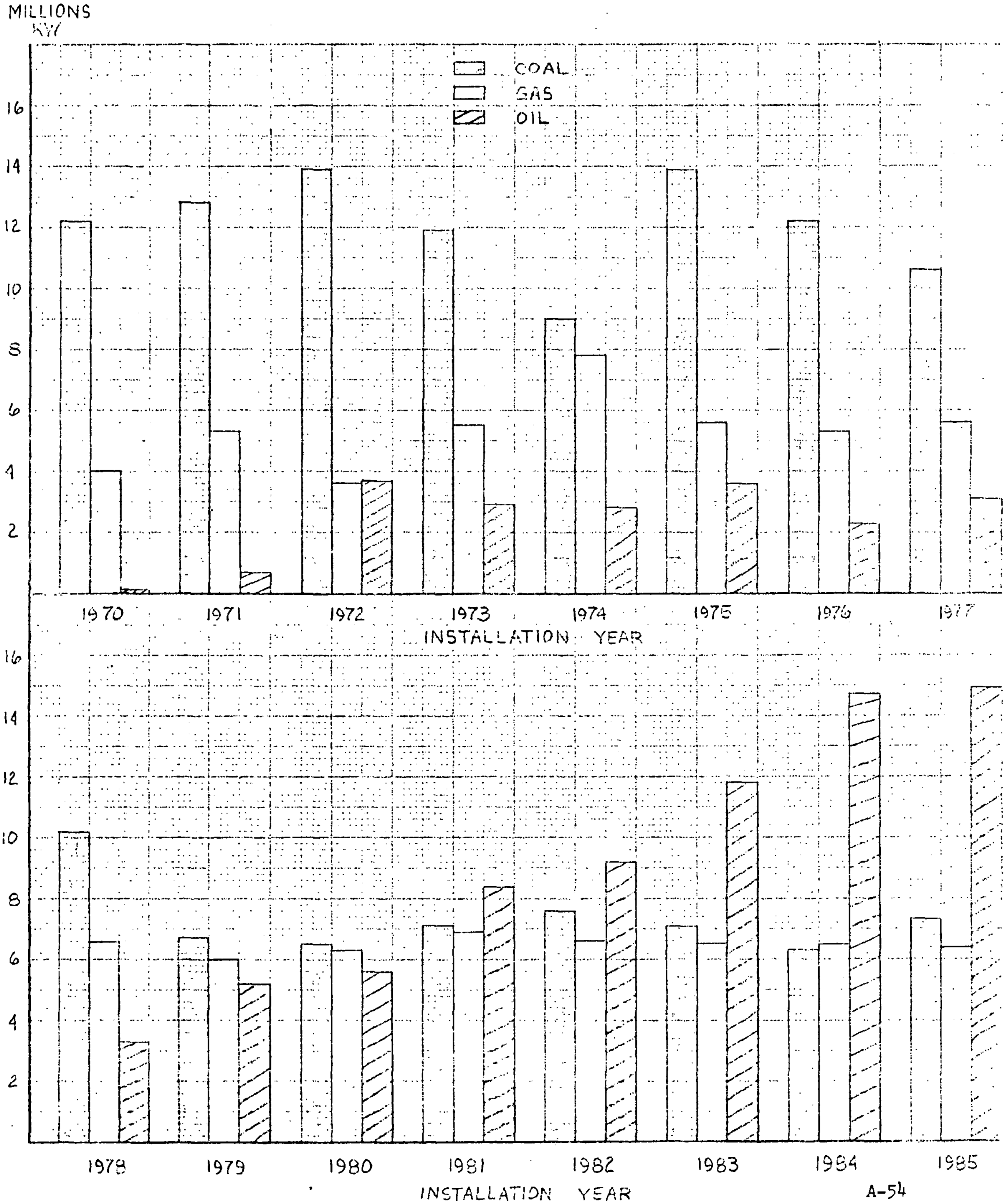
Electric utility fuel price projections 1971—1985 actual dollars

UPDATED PROJECTIONS

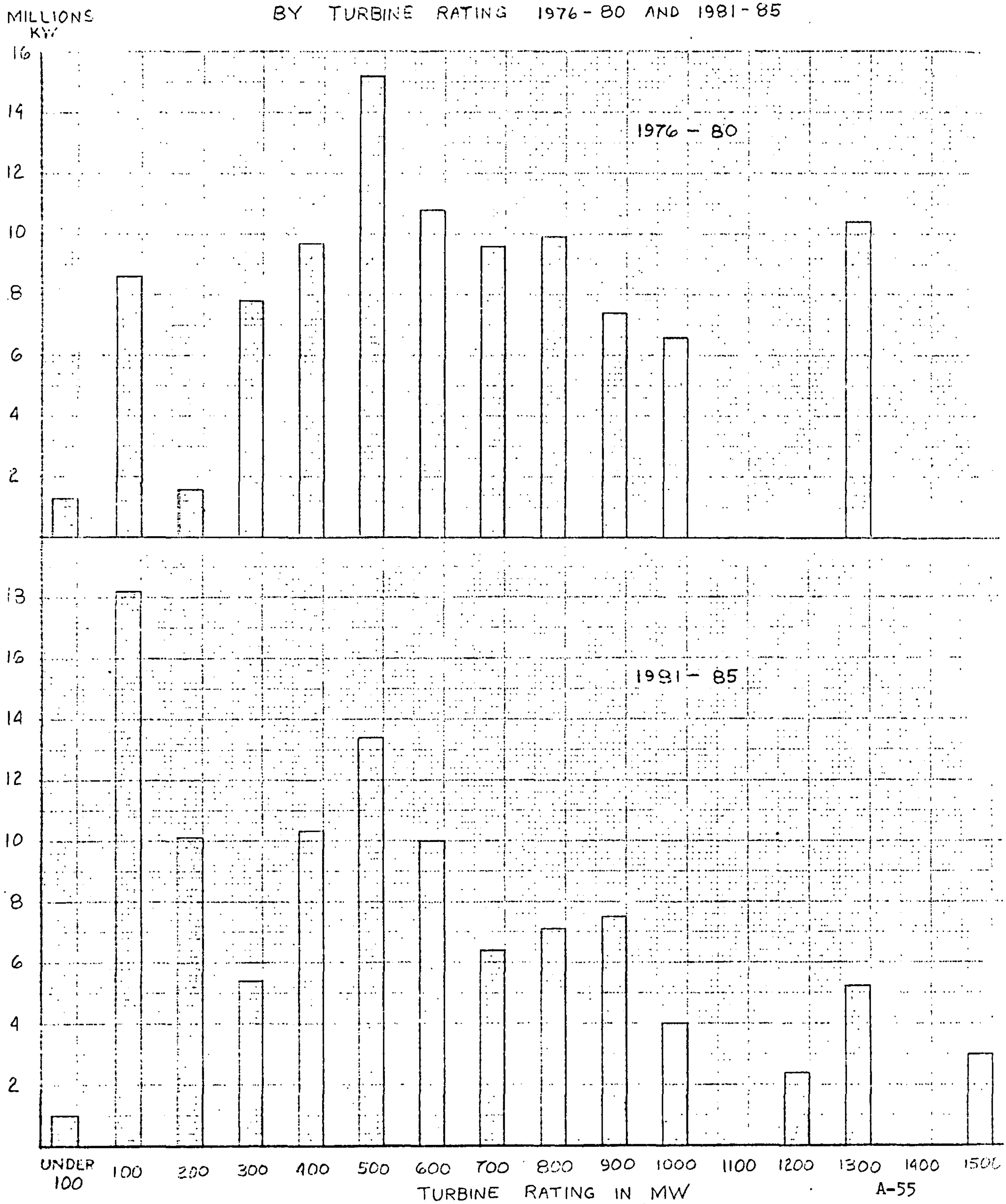
ELECTRIC UTILITY GENERATION ADDITIONS IN GW
MARKET SUMMARY 1970-1985

	1970 TOTAL	1971-1975 ADDITIONS	1975 TOTAL	1976-1980 ADDITIONS	1980 TOTAL	1981-1985 ADDITIONS	1985 TOTAL
BASE LOAD							
Coal	87.4	10.5	97.9	30.3	128.2	2.1	130.3
Oil	21.1	1.3	22.4	4.9	27.3	-	27.3
Gas	45.5	3.4	48.9	-	48.9	-	48.9
Total Fossil	154.0	15.2	169.2	35.2	204.4	2.1	206.5
Nuclear	5.3	62.1	67.4	70.1	137.5	130.9	268.4
Hydro	25.0	-	25.0	-	25.0	-	25.0
INTERMEDIATE							
Coal	51.7	30.4	82.1	45.7	127.8	73.7	201.5
Oil	10.4	8.7	19.1	15.2	34.3	5.3	39.6
Gas	21.3	12.8	34.1	7.6	41.7	-	41.7
Total Fossil	83.4	51.9	135.3	68.5	203.8	79.0	282.8
Nuclear	-		-	7.6	7.6	26.3	33.9
Hydro	23.6	2.5	26.1	2.5	28.6	2.5	31.1
PEAKING							
Inter.Comb & G.T.	21.8	30.9	52.7	11.4	64.1	37.8	101.9
Hydro	7.3	15.1	22.4	18.2	40.6	15.0	55.6
TOTAL CAPACITY	320.4	177.7	489.1	213.5	711.6	293.6	1005.2

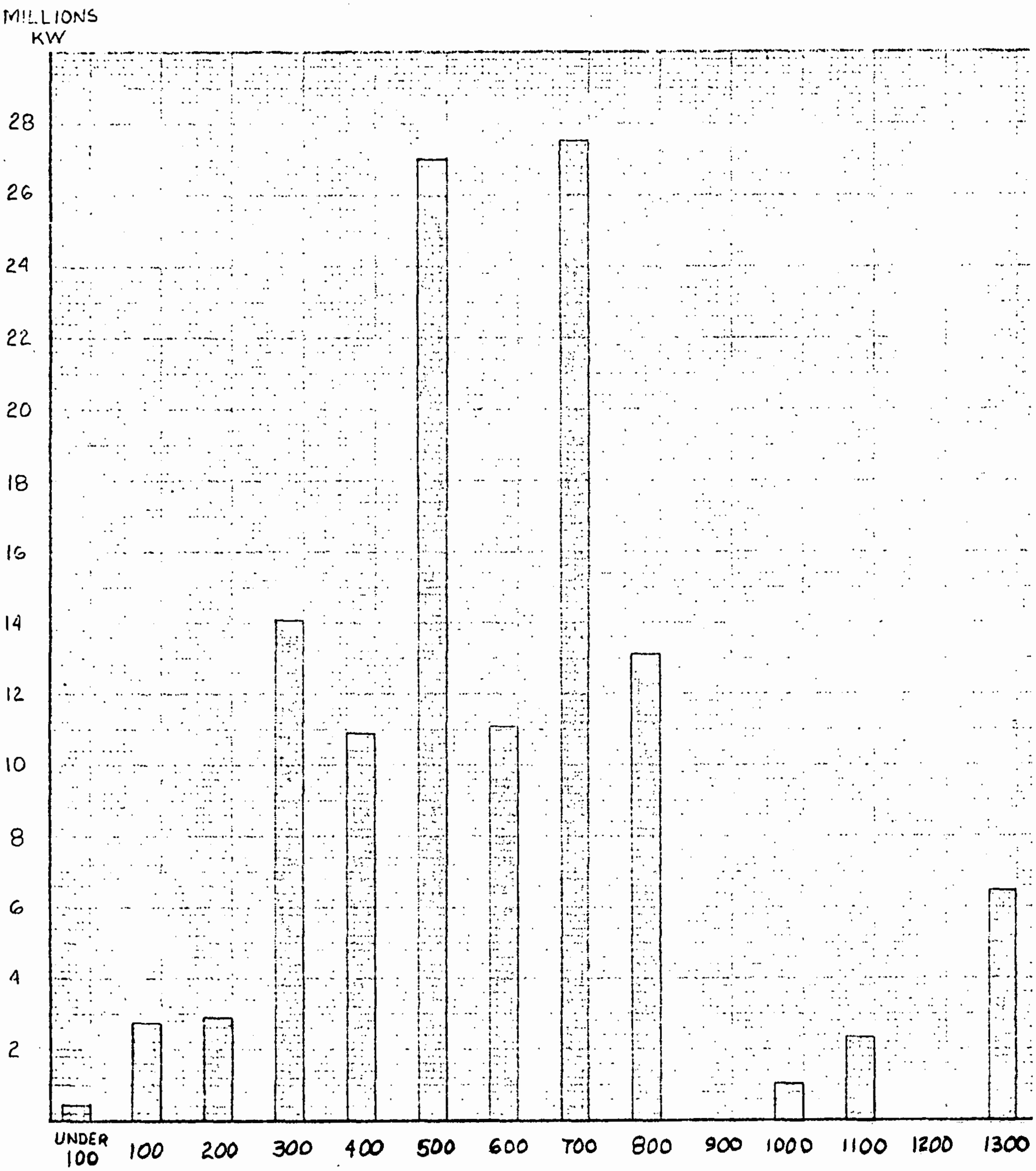
U.S. ELECTRIC UTILITY INDUSTRY
COMPARISON OF ANNUAL FOSSIL STEAM INSTALLATIONS
BY TYPES OF FUEL 1970 - 85



U.S. ELECTRIC UTILITY INDUSTRY
DISTRIBUTION OF FOSSIL STEAM ADDITIONS
BY TURBINE RATING 1976-80 AND 1981-85



U.S. ELECTRIC UTILITY INDUSTRY
DISTRIBUTION OF FOSSIL STEAM ADDITIONS
BY TURBINE RATING 1970 - 75



APPENDIX B

INDUSTRIAL BOILER MARKET SURVEY

Prepared by Erie City Energy Division
of Zurn Industries, Inc.

AUTHOR: Warren D. Schwinden

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A STUDY OF THE INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

1. INTRODUCTION. In an attempt to estimate the possible influence of fluidized-bed combustion (FBC) technology on the Industrial Stationary Watertube Boiler Market, a study was performed under NAPCA Contract CPA 70-9. The two-fold objective of the study was to first determine typical operating parameters for a prototype design of an industrial FBC unit; the other objective was to measure, in some manner, the possible areas of application for FBC technology in the industrial boiler market.

   Recognizing the state-of-the-art for this combustion method would probably remain in the developmental phase until at least 1973, any discussion of the above objectives must pertain to the post-1973 market. Hence, a forecast of the market relative to technological changes and consumption patterns would be required prior to any assessment of FBC potential.

2. CONCLUSIONS. The major results of this report are listed below.

   1. The prototype design should be in the 600 PSIG - 750 F pressure-temperature class. The output of the unit should be the maximum possible while still remaining shipable subject to current railroad restrictions. This suggests using a modular approach to unit design for minimization of field installation costs.

   2. The potential 1980 market for units with outputs exceeding 100,000 lbs. stm/hr. will be in the neighborhood of 63.5 million pounds of steam. This is the portion of the industrial boiler market which FBC technology will need to penetrate.

3. The predominant portion of the industrial boiler market is shifting from smaller to medium sized units (that is, units with outputs between 100,000 and 250,000 lbs. stm/hr.)
4. Relative to aggregate capacity sold, the industrial segment is not expanding as rapidly as the utility portion of the stationary watertube boiler market.
5. In the current market, either oil or gas are used as the primary fuel for 80 and 90 percent of the total capacity and units sold, respectively. The preferred fuel is gas which is used for 60 and 70 percent of the total capacity and units sold, respectively.
6. The use of bituminous coal as a primary fuel for units sold in the industrial boiler market has been rapidly declining. In the 1969 market, coal firing accounted for less than 3 and 2 percent of the total capacity and units sold, respectively.
7. The use of coal as a primary fuel is generally restricted to larger sized units with the preferred method of firing being either spreader stoker or pulverizer.
8. The major consumers of units with outputs exceeding 100,000 lbs. stm/hr. appears to be the chemical, paper, and petroleum industries.
9. Since it is unlikely FBC will be a competitive alternative for gas-fired units, and the region (Texas, Oklahoma, Arkansas and Louisiana) consuming the largest average unit size uses gas as the primary fuel, the domain of FBC marketability is further restricted.
10. Unless the availability of oil and gas are severely restricted by either diminishing supplies or legislation, coal does not appear to be a significant primary fuel in the future industrial boiler market.

The overall assessment of the future for FBC technology in the industrial stationary watertube boiler market, based upon the above conclusions, is not very optimistic. Although FBC will hopefully aid in the pollution control of coal fired units, it is doubtful that industrial users can be persuaded to switch from present oil or gas fired units to ones using FBC methods. To produce such a switch, some rather dramatic improvements in either initial investment or operating costs compared to existing methods would have to be offered. This appears rather unlikely for FBC units. On the other hand, since coal fired units will always have some area of application in the industrial boiler market, FBC technology should be a useful technological advancement.

3. OUTLINE OF REPORT. The remaining pages of this report contain a definition of the source, method of presentation, and limitations of the available historical statistical data. Some interpretations of these data will then be offered along with a discussion of the method used in generating the required long-term forecast.

As a means of obtaining a comparison, an additional forecast is presented using an independent approach. Both results are strikingly similar, thus yielding some degree of confidence in the forecast.

Following the interpretation of the forecast, numerous figures and appendices are presented. It must be emphasized, while reading this report, constant reference to these supplemental statistics will be required.

4. ABMA RECORDED DATA. The historical data used for this market analysis was extracted from the 1961 through 1969 annual reports of the American Boiler Manufacturers Association (ABMA). A letter authorizing the use of these data is included in Appendix I.

Perhaps a definition of the source and type of statistics recorded by the ABMA will make the data more meaningful. Each stationary watertube boiler sold by member companies (and this includes all of the major and most of the minor equipment manufacturers) is reported to the ABMA listing the following properties:

1. Purchaser's Standard Industry Classification (SIC) number.
2. Domestic or export use.
3. Domestic region (or foreign country) in which unit is installed.
4. Electric utility or industrial use.
5. Packaged or field assembled construction.
6. Steam capacity.
7. Operating pressure.
8. Outlet steam temperature.
9. Primary fuel.
10. Alternate fuel.
11. Firing method for solid fuel (if applicable).

It should be noted the above listing only includes those items which affect the intended market study. Other properties are reported, such as reheat temperatures, but this study is meant to exclude boilers intended for utility use. Also, marine boilers and hot water units are omitted.

The annual reports of the ABMA group the operating parameters (unit steam capacity in thousands of pounds of steam per hour, operating pressure in PSIG, and outlet steam temperature in degrees Fahrenheit) into discrete intervals. Likewise, the other properties are separated into distinct categories. A definition of these ABMA defined clusters is included in Appendix I.

It is important to note, however, ABMA is continually redefining these statistical groups. Consequently, the available years with consistent statistics becomes quite limited. At best, the reported industrial steam data remained consistent within the various categories for only a nine year time span. Indeed, the industrial steam classification was not tabulated in their annual reports prior to 1961. The data stated for the entire stationary watertube boiler market from 1937 through 1969, however, were useful statistics.

5. DESCRIPTIVE STATISTICS. In performing this study, the available statistics were analyzed from various viewpoints. To begin with, Appendix I includes a tabulation of the aggregate capacity sold in the entire stationary watertube boiler market from 1937 through 1969. In addition, the industrial boiler market from 1961 through 1969, and its percentage relative to the total demand, is given. The same data are shown graphically in Figure 1. Observe this graph is a semi-logarithmic plot to indicate the natural growth trends. It is thus clear the total market is expanding at a faster rate than the industrial demand. This observation is an expected one considering the growth of the utility industry during this time period. Note the utility industry steam demand is essentially the difference between the total and industrial markets.

Figure 2 is merely an extension of the total market data exhibited in Figure 1 with the exception it is shown on a rectangular grid. Notice the significant shift in the level of total demand from 1962 to 1965. Again the increasing difference between both markets is apparent.

The historical data for the clusters used by ABMA in reporting their annual statistics (that is, the ones already mentioned which are relevant to this study) are presented in Appendices II through IX. Each category covers the years for which consistent data were available and are displayed in difference table form. These tabulations represent both aggregate demand (millions of pounds of steam) and strength relative to the total demand (expressed as a percentage) for each category. And in addition, because the market can be measured with respect to either total capacity of total number of units sold, the appendices list the data accordingly.

6. INTERPRETATION OF STATISTICS. Certainly any market study has to begin by looking at the available historical data. Likewise, the statistics should be viewed from different vantage points. Thus Figures 3, 4, and 5 separate the data into broad categories to suggest trends relative to unit size. Obviously, the predominant influence in the past has been exerted by those units with outputs not exceeding 100,000 Lbs./Stm/Hr. This strength is shifting, however, to the units of medium size (that is, those units with output between 100,000 and 250,000 lbs. Stm/Hr.)

Page A-5 contains a tabulation of the average unit size for fuels, firing method, markets, and regions. Some interesting observations can be made such as the increasing unit size for both oil and gas firing. Notice these data, being average values, are influenced by the quantity of smaller sized units sold in the market place.

The need for larger sized units in the chemical, paper and petroleum industries, although intuitively evident, is substantiated by the tabulated data. Similarly, the concentration of the chemical and petroleum industries in Region 6 (Texas, Oklahoma, Arkansas, and Louisiana) is indicated by the marked increase in average unit size contrasted to the other regions.

When attempting to analyze a set of statistical data, it is always interesting to look at the corresponding cumulative distributions. Accordingly, Figure 6 displays the annual distributions for capacity (that is, unit size), pressure, and temperature.

Several comments regarding the construction of these distributions are in order. First, the reporting intervals remained consistent during the 1962 through 1969 period. The 1961 distribution (Figure 6-1) on the other hand, is included, but the different structure is evident. Indeed, this contrast is most obvious in the temperature distribution.

The second comment concerns the temperature curves. Obviously, the saturated portion of the market is not uniformly distributed throughout the saturation interval (0 to 300 degrees Fahrenheit) as indicated by the graph. Instead, the percentage level at the curve and partition line intersection relates to the saturated steam demand for that particular year.

Finally, these distributions are linear segment approximations over intervals defined in Appendix I. In fact, Appendices II, III, and IV exhibit the actual density data used to construct these cumulative distributions.

Some conclusions can be reached from reading these curves. For example, it seems in the past units with outputs not exceeding 100,000 Lbs.Stm./Hr. account for roughly 80 and 50 percent of the units and capacity sold, respectively.

When one looks at the pressure distributions, it becomes apparent 90 and 70 percent of the units and capacity sold, respectively, do not exceed 600 PSIG operating pressure. Likewise, the significance of saturated units is readily discernible from the temperature curves (approximately 75 and 50 percent of the units and capacity sold, respectively). The span between 1964 and 1967, however, illustrate the period of industrial expansion by showing greater dependence on superheated units. Note the most important superheated class is the range covering 725 to 775 degrees Fahrenheit. Certainly turbine applications cause this predominant temperature interval.

Trends exhibited by the density data (relative to percent of aggregate capacity sold) contained in Appendices II, III and IV are shown in Figures 7, 8 and 9, respectively. These trends will be used later in an attempt to anticipate technological changes.

7. LONG RANGE FORECASTING TECHNIQUES. Whenever one attempts to make a long range market forecast, he must consider both the economic and technological factors. For the prevailing economic activity will determine in some manner the level of demand, but the state of technology limits the available product species.

In forecasting economic conditions, several techniques are in current use. Perhaps the most popular among non-economists is correlating a particular market with some economic indicator. To achieve meaningful correlation, however, a rather large set of market data should be avaiable (most forecasters recommend a minimum time span of 15 years). Obtaining the necessary statistics for the indicators is no longer an insurmountable problem since several government sources publish easily accessible data. To locate long-term projections for these indicators, on the other hand, is quite a different problem. Still, some sources are available. For instance, several institutions maintain econometric models from which they make long-term projections for many of the common indicators.

Normally the market data itself, however, is the limiting factor. Indeed, this market study uses only nine known data points. Thus it would seem quite foolhardy to attempt a correlation with some type of indicator (such as the New Plant and Equipment Expenditures and Corporate Profits data published by the Council of Economic Advisors in their "Economic Indicators" periodical) especially for long range forecasting. Yet this approach would be feasible if one was only interested in short-term predictions. For then a constant surveillance of the variance between predicted and actual values could measure the validity of the correlation. This monitoring is possible since short-term predictions are, in general, repeated for each new period. In contrast, the objective of this particular market study is not to devise some repetitive forecasting procedure. Rather it is to make a single reasonably valid estimate of the potential market.

Furthermore, to use a long-range single-point estimate, as would be the case in short-term predictions, can only result in an erroneous forecast. The most one can expect to determine is some interval which will hopefully contain the future trend. Moreover, as the forecast period increases, the interval range should diverge. Note the distinction made in stating this interval should contain the expected trend value and not a single-point estimate of the actual demand. As a matter of fact, the value of the demand could well exceed the interval boundaries.

We are in essence starting with a set of nine data points originating from 1961. From this starting point, we are saying if the trend were computed after each additional year's demand was added to the existing set of data, the interval would contain the terminal point of the trend line (that is, the trend line value for the latest recorded year).

8. BETA DISTRIBUTION. If one accepts the concept of using trend interval estimates, the problem then is how to determine the interval range. This can be achieved by making optimistic and pessimistic estimates which can be used as the upper and lower bounds, respectively. If some value within the interval can be considered a most likely estimate, a statistical distribution can then be used to generate a statistically expected value. A probability measure for subinterval forecasts is then available. Observe the span of the interval is a measure of the uncertainty implicit in the forecast. The Beta distribution is the particular one used over the forecasted interval; for a discussion of its mathematical properties, see Vol. 2 of William Feller's An Introduction to Probability Theory and Its Applications.

In determining the optimistic and pessimistic bounds for the interval, one could rely on either subjective opinion or objective techniques. For a new technological method, however, subjective numbers would be hard to determine. Hence, naive methods (that is, extrapolation of historical data) were used. Assuming the greatest expansion of a market would be limited to the law of natural growth, the optimistic bound would be subject to an exponential function. Consequently, a constant ratio between consecutive years characterizes the optimistic estimate.

Since a considerably larger set of data were available for the total stationary watertube boiler market, the pessimistic constraint was generated (linearly) from these numbers. The average industrial steam portion of the total market was then assumed to be 30 percent of the total demand. Notice Figure 2 shows a significant change in demand for the total market between 1962 and 1965. Yet the trend of the raw data without any adjustments determined the pessimistic estimate. Hence, the pessimistic estimate should indeed be conservative.

For the most likely estimate, it seemed reasonable to use the current trend line for the industrial steam market. Observe a linear trend, as used for the pessimistic and most likely values, assumes a constant annual change in the magnitude of the market, as opposed to a constant proportional change for the optimistic constraint.

With the interval bounds and the most likely value estimated, a statistically expected value using the Beta distribution was calculated. Figure 10 shows this distribution for the forecasted 1980 interval with both the density and distribution curves illustrated. Notice the interval has been transformed to the zero-to-one range. This has been done for ease of calculation. In interpreting the figure, perhaps the distribution curve is most useful. For example, one can determine the median value, the probability of some subinterval estimate, and the skewness of the distribution.

9. FORECAST RESULTS. The trend interval estimates are shown in Figure 11. For each year, four points are shown on the graph: optimistic and pessimistic bounds, the most likely value, and the progression of the statistically expected values indicated by the continuous curve. A tabulation of the corresponding numbers is presented in Figure 12. As previously stated, the noticeable divergence of the interval constraints can be interpreted as a measure of the uncertainty implicit in the forecast.

After the forecast was made, preliminary data became available for the 1970 market. Therefore, the trend line for the augmented statistics was calculated as a check on the procedure used. The resulting trend line terminal point was found to be in the neighborhood of the midpoint for the subinterval bounded by the pessimistic and expected values. Thus the trend line interval forecasting procedure was not refuted.

Once the future demand for industrial steam was estimated, the changing technological requirements for the operating parameters (unit capacity, pressure, and temperature) had to be considered. To anticipate these differences, the trends within categories were calculated (see Figures 7, 8 and 9). Now the major problem in using the available data resulted from the statistics being separately reported for each property. Hence, the annual marginal distributions for capacity, pressure, and temperature were known, but not the joint distribution. Unfortunately, it is precisely the sequence of annual joint distributions which would have been most helpful in anticipating pattern changes for these properties. Observe a given joint distribution implies knowledge of the demand for each capacity-pressure-temperature combination.

Assuming the properties are statistically independent, the joint relationship is determined by the marginal distributions. If this is an erroneous assumption, however, knowledge of the marginal distributions does not provide adequate information to generate the joint profile. Indeed, the capacity-pressure-temperature properties do not exhibit statistical independence (covariance does not equal zero which is a necessary, although not sufficient, condition for statistical independence). It was decided, however, to still generate the joint distribution as an aid in projecting technological change, even though statistical independence was not characteristic of the capacity, pressure and temperature properties.

Figure 13 shows the joint capacity-pressure distribution for both the historical and future markets. It is interesting to note the increasing influence of the medium range units. Likewise, the anticipated market growth results in a shift from units with very low to very high generating capabilities.

As already noted, the industrial stationary watertube boiler market forecast was generated using an estimated trend interval procedure. An additional forecast was made using an economic indicator approach. It is important to note this method is questionable since a definite relationship is difficult to establish with such a limited set of available data. The results are merely stated here for comparison.

The instrumental variables used for the economic indicator were Corporate Profits Before Taxes and New Plant and Equipment Expenditures data as published by the Council of Economic Advisors in their periodical "Economic Indicators". The response of the indicator in relation to the actual performance of the industrial steam market is shown in Figure 14. Notice the declines for both 1967 and 1970 were anticipated by the indicator. The tabulated data displayed in Figure 14 were generated from adjusted profits and expenditures projections published in the Summer 1970 issue of the Wharton Economic Newsletter. It is interesting to note tabulated data exhibit a less rapid short-term expansion, but the 1980 estimate of 126 million pounds of steam is essentially coincident with the 127 statistically expected value for the 1980 trend interval estimate.

10. INTERPRETATION OF FORECAST. The motivation for this market forecast was to estimate the potential applications for FBC units. Limiting the region for FBC use is the general opinion that only units with outputs exceeding 100,000 Lbs. Stm./Hr. can economically apply this new combustion method. Since this will amount to approximately 50 percent of the 1980 industrial steam demand, around 63.5 million pounds of steam will be the maximum potential FBC market. Notice, however, this is almost the level of the current available market for all industrial units. Indeed, the entire 1980 industrial stationary watertube boiler market is expected to expand from the current level by an approximate 1.5 factor.

A significant shift to medium-sized units should occur. Thus the current influence which smaller sized units exert on the total market will be declining, although the actual annual quantity sold should be increasing at a slow rate.

The industrial market is not experiencing the same rapid expansion as the utility sector. But for FBC applications,the utility industry appears to offer better market potential because of the demand for larger unit size and fossil fuel usage. The competition from gas fired units would appear to severely limit FBC usage in the industrial portion of the stationary watertube boiler market.

Numerous appendices are included with this report. Thus the known history of the industrial steam market is presented to allow individual interpretation of the data.

TABLE B-1

A B M A D A T A

AVERAGE UNIT OUTPUT (IN THOUSANDS OF LBS. STM./HR.)

	CATEGORY	1962	1963	1964	1965	1966	1967	1968	1969
	COAL	66.4	80.0	107.6	116.3	74.4	95.6	128.1	105.0
	OIL	44.1	46.8	53.9	58.8	57.6	60.3	80.4	65.8
F	GAS	51.7	49.4	60.4	61.7	66.5	62.5	64.2	64.3
U	WOOD	61.5	100.0	91.7	38.5	33.3	180.0	107.7	121.0
E	BAGASSE	68.4	86.4	95.0	100.0	100.0		100.0	216.7
L	LIQUOR	137.5	212.5	291.7	268.4	282.4	175.0	233.3	300.0
	OTHER	68.8	54.0	84.5	116.7	108.1	128.2	97.5	185.4
F									
I	PULVERIZED	250.0	213.3	326.7	330.8	250.0	150.0	475.0	350.0
R	SPREADER	67.2	96.9	96.7	115.0	84.4	95.2	113.3	104.3
I	UNDERFEED	37.5	25.0	14.3	18.2	25.0	166.7	42.8	20.0
N	OVERFEED	40.0	60.0	114.3	50.0	66.7	500.0	33.3	57.1
G	OTHER	46.3	70.8	88.9	142.8	93.8	113.6	71.4	91.7
	NON-MFG			40.3	43.9	41.9	44.3	44.8	46.5
	CHEMICAL			115.2	103.3	97.4	103.5	101.0	93.8
M	PAPER			156.3	175.0	131.5	135.7	155.9	158.2
A	PETROLEUM			90.1	124.6	138.5	98.2	112.7	120.0
R	FOOD			63.2	55.9	58.0	57.7	70.2	68.4
K	METALS			77.3	131.1	92.1	96.6	78.1	178.9
E	MISC. MFG.			55.7	47.1	56.8	57.8	71.6	57.0
T	TEXTILES			54.3	37.8	44.4	48.9	51.0	56.2
	TRANSPORTATION			84.4	78.0	67.7	64.0	110.8	103.7
	WOOD			84.6	50.0	59.2	94.4	70.0	67.6
	RUBBER			36.0	37.5	59.2	46.7	55.6	50.0
	1				45.6	55.2	55.6	59.4	55.9
R	2				63.5	55.1	44.8	58.9	58.7
E	3				57.4	63.6	63.7	72.9	78.2
G	4				91.1	61.5	72.1	82.5	74.4
I	5				52.7	49.4	50.0	71.9	27.4
O	6				142.0	133.1	126.7	113.1	140.0
N	7				57.1	50.0	47.4	46.2	52.6
	8				64.6	77.8	85.2	63.6	65.8
	9				62.5	50.0	44.2	57.4	64.5

FIGURES

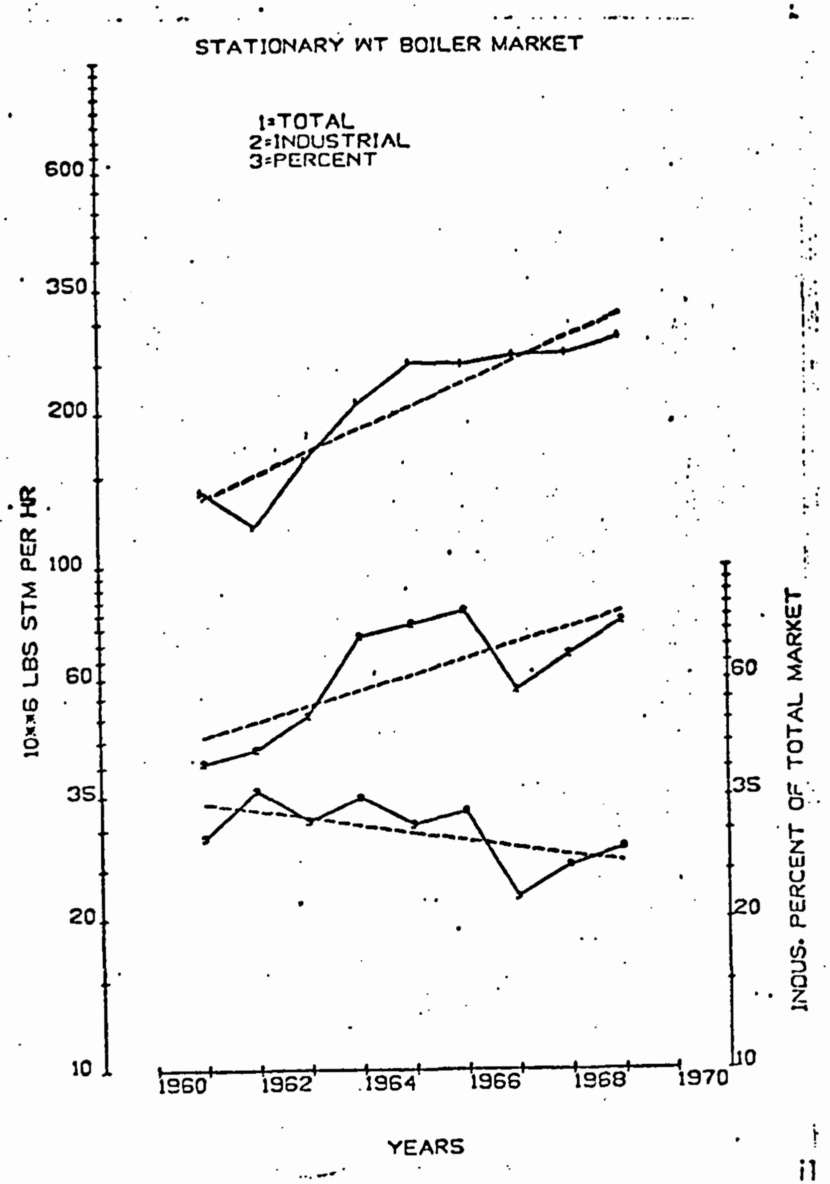


Figure 1

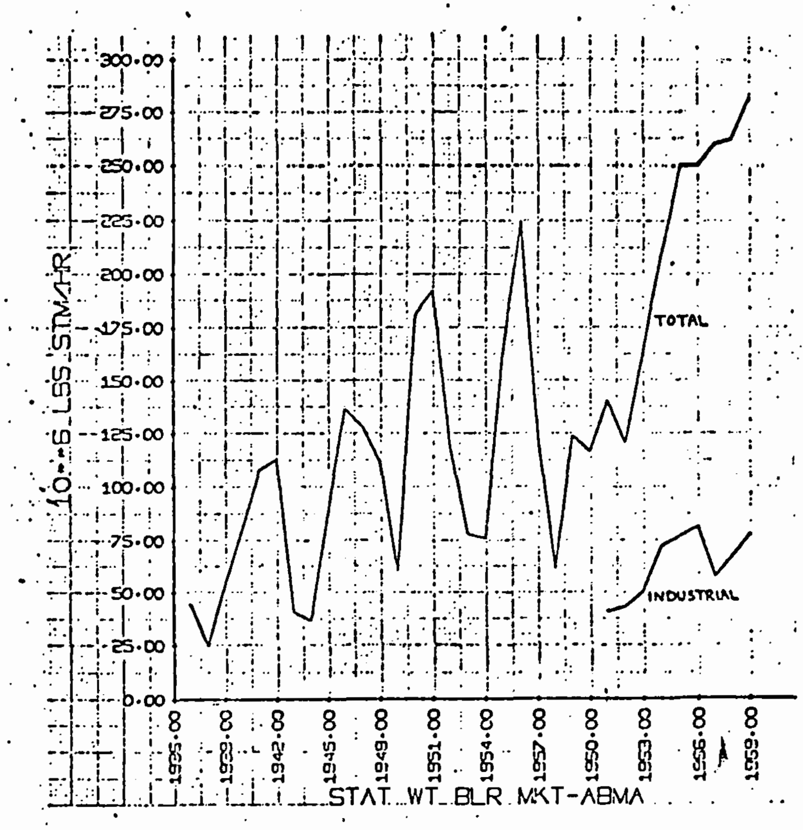


Figure 2

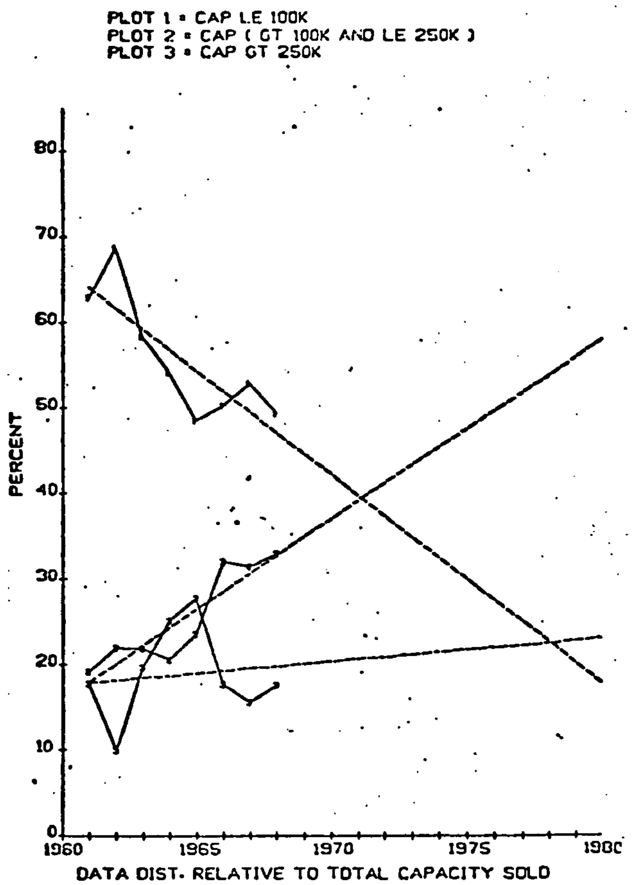


Figure 3

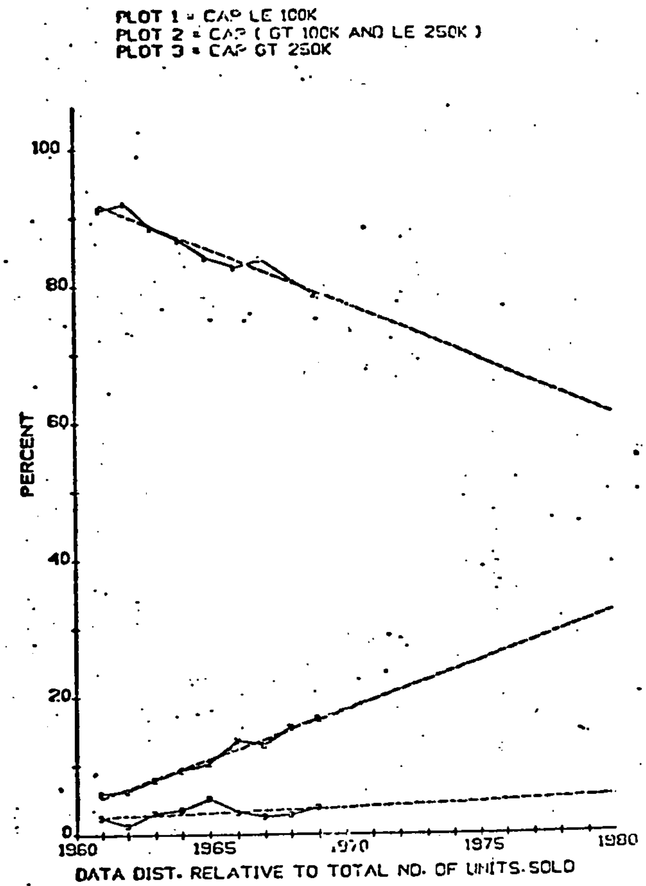


Figure 4

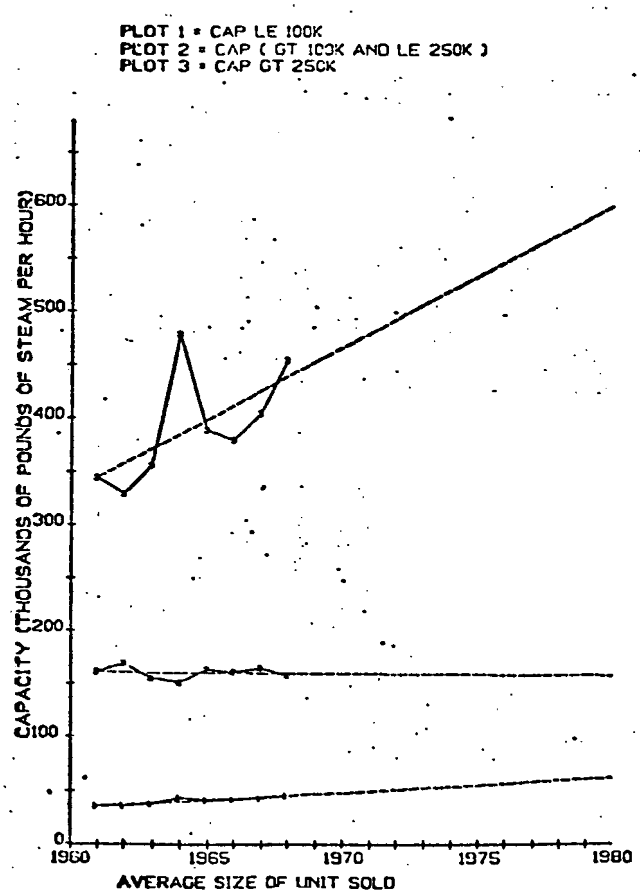


Figure 5

INDUSTRIAL STATIONARY
WATERTUBE BOILER
MARKET

1961 DISTRIBUTION

TOTAL CAPACITY
40,640,000

TOTAL NUMBER
OF UNITS
787

NOTE
SOLID LINE REPRESENTS
UNITS DISTRIBUTION

BROKEN LINE REPRESENTS
CAPACITY DISTRIBUTION

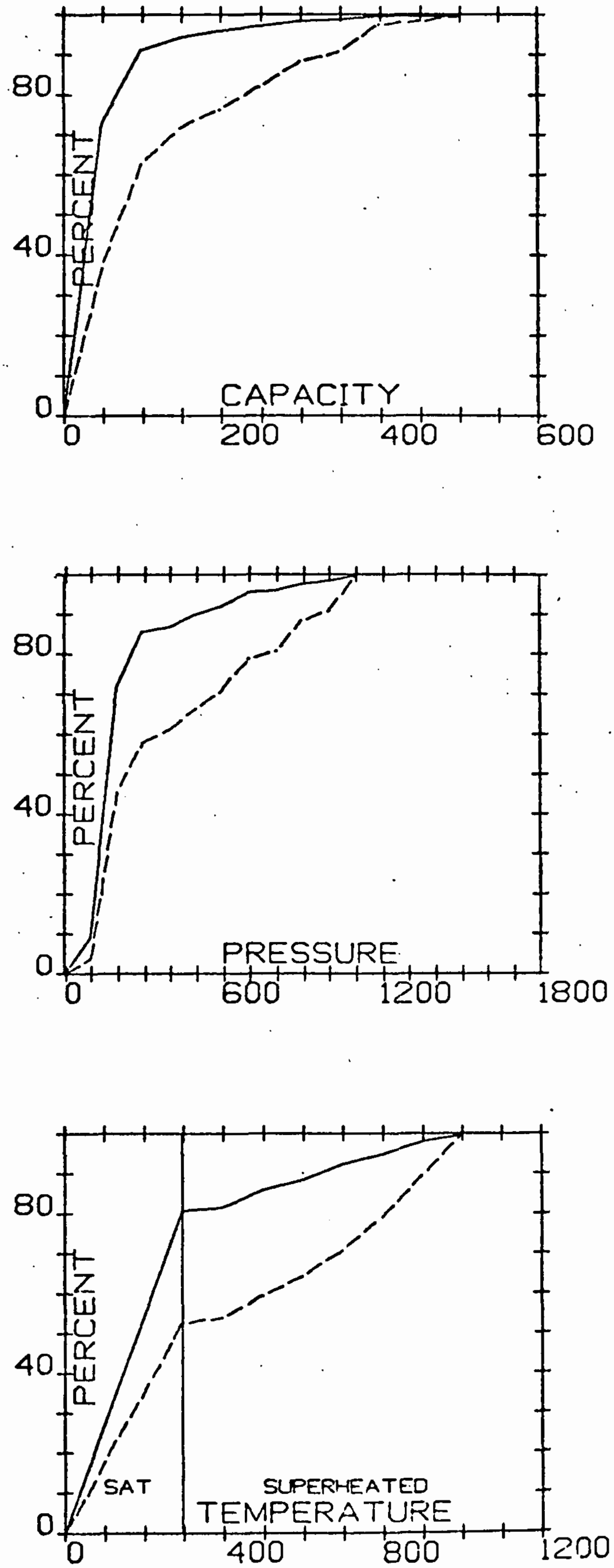


FIGURE B-1

INDUSTRIAL STATIONARY
WATERTUBE BOILER
MARKET

1962 DISTRIBUTION

TOTAL CAPACITY
43,333,000

TOTAL NUMBER
OF UNITS
877

NOTE
SOLID LINE REPRESENTS
UNITS DISTRIBUTION

BROKEN LINE REPRESENTS
CAPACITY DISTRIBUTION

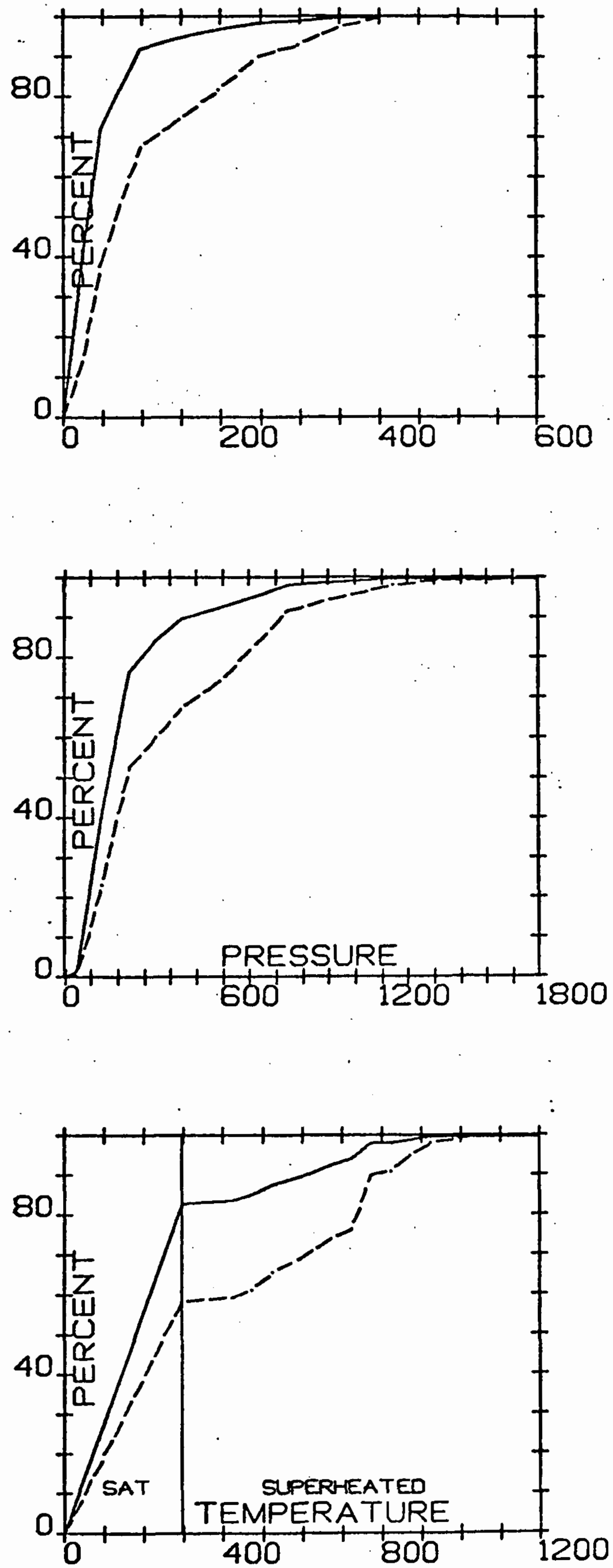


FIGURE 6-2

INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

1963 DISTRIBUTION

TOTAL CAPACITY
50,535,000

TOTAL NUMBER
OF UNITS
879

NOTE
SOLID LINE REPRESENTS UNITS DISTRIBUTION

BROKEN LINE REPRESENTS CAPACITY DISTRIBUTION

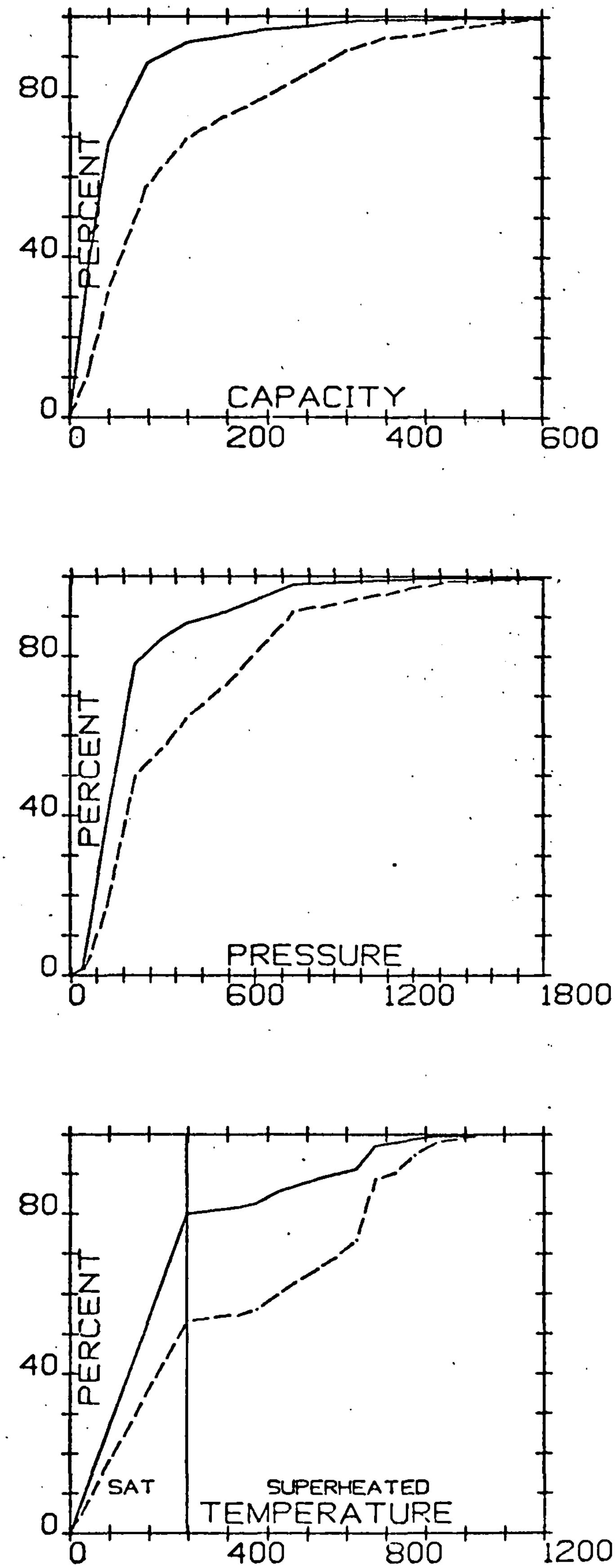


FIGURE 6-3

INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

1964 DISTRIBUTION

TOTAL CAPACITY
72,348,000

TOTAL NUMBER OF UNITS
1033

NOTE
SOLID LINE REPRESENTS UNITS DISTRIBUTION

BROKEN LINE REPRESENTS CAPACITY DISTRIBUTION

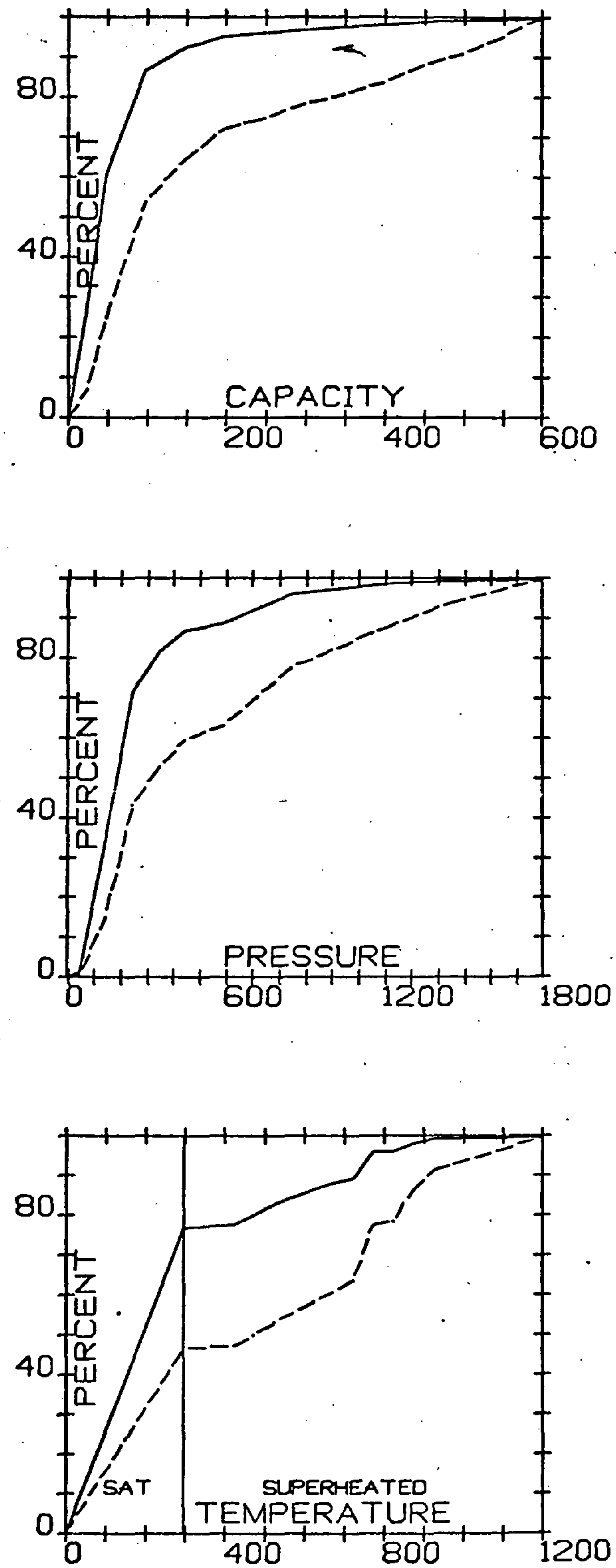


FIGURE 6-4

INDUSTRIAL STATIONARY
WATERTUBE BOILER
MARKET

1965 DISTRIBUTION

TOTAL CAPACITY
76,885,000

TOTAL NUMBER
OF UNITS
1055

NOTE
SOLID LINE REPRESENTS
UNITS DISTRIBUTION

BROKEN LINE REPRESENTS
CAPACITY DISTRIBUTION

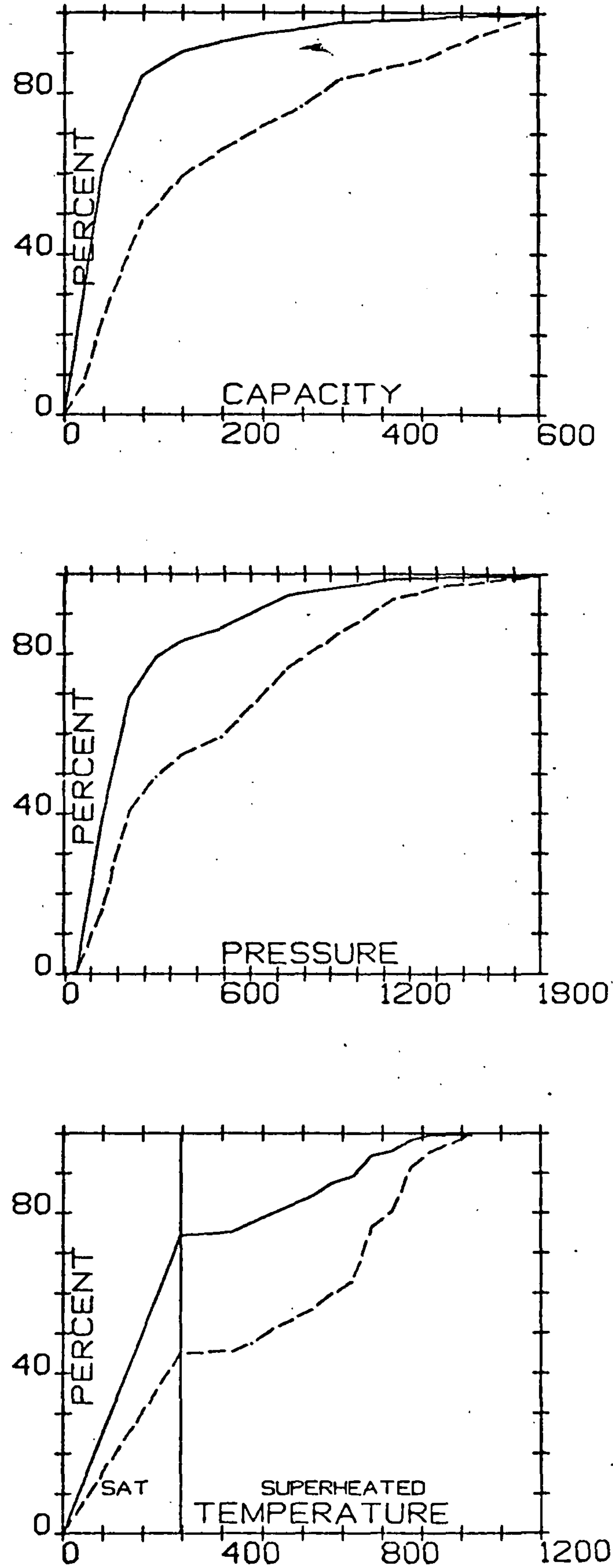


FIGURE 6-5

INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

1966 DISTRIBUTION

TOTAL CAPACITY
81,438,000

TOTAL NUMBER OF UNITS
1165

NOTE
SOLID LINE REPRESENTS UNITS DISTRIBUTION

BROKEN LINE REPRESENTS CAPACITY DISTRIBUTION

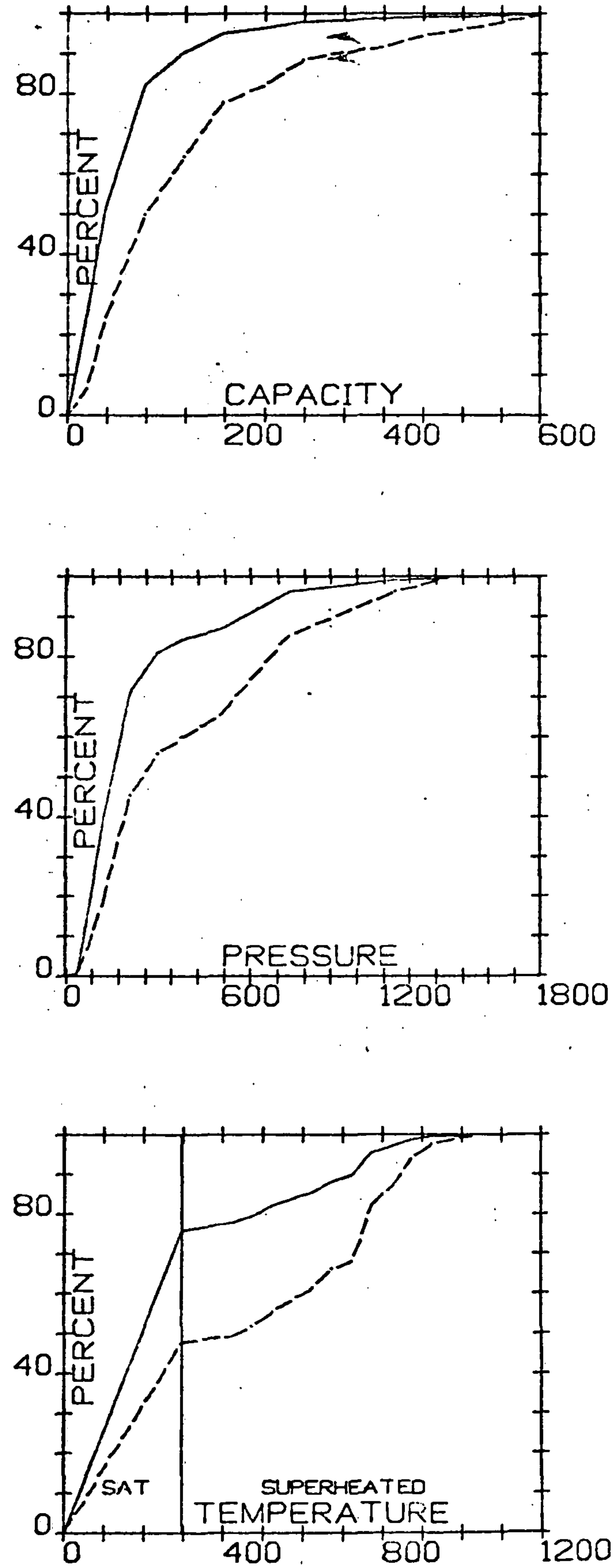


FIGURE 6-6

INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

1967 DISTRIBUTION

TOTAL CAPACITY
57,257,540

TOTAL NUMBER OF UNITS
839

NOTE
SOLID LINE REPRESENTS UNITS DISTRIBUTION

BROKEN LINE REPRESENTS CAPACITY DISTRIBUTION

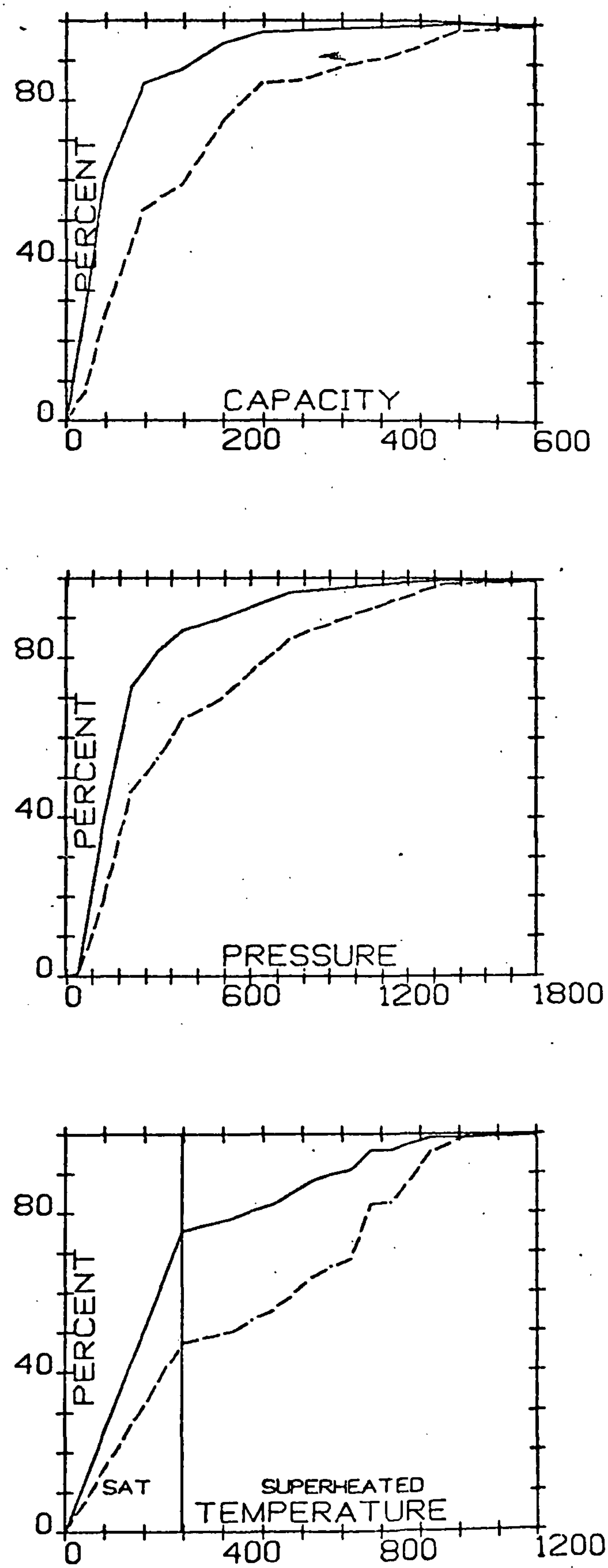


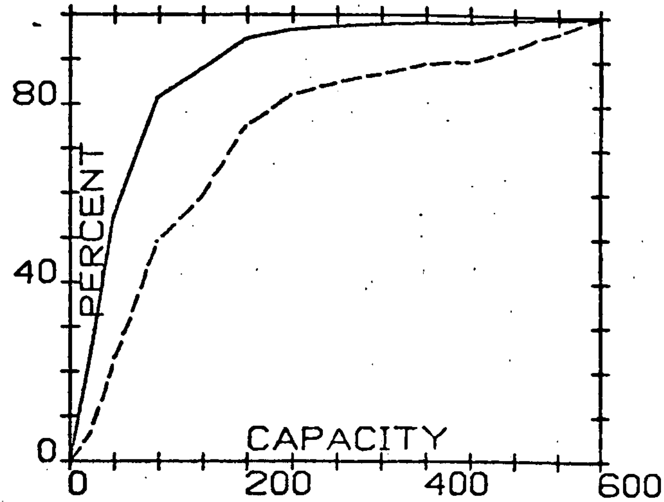
FIGURE 6-7

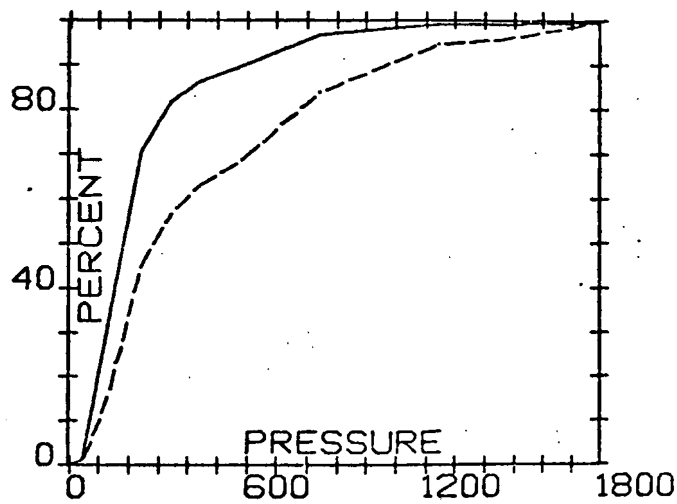
INDUSTRIAL STATIONARY
WATERTUBE BOILER
MARKET

1968 DISTRIBUTION

TOTAL CAPACITY
66,991,295

TOTAL NUMBER
OF UNITS
908





NOTE
SOLID LINE REPRESENTS
UNITS DISTRIBUTION

BROKEN LINE REPRESENTS
CAPACITY DISTRIBUTION

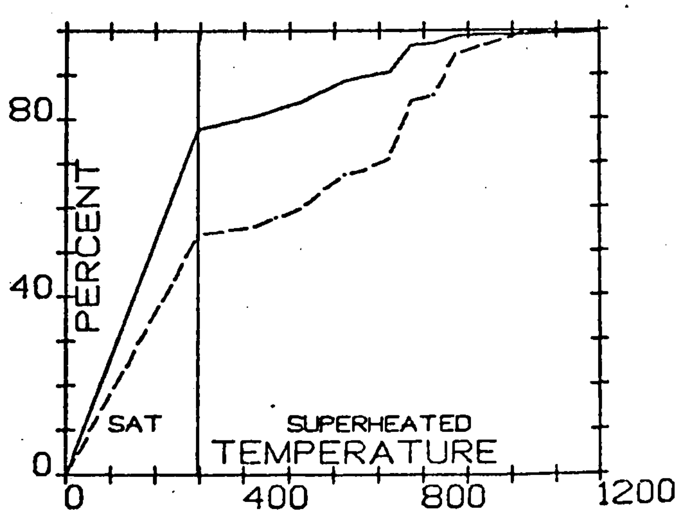


FIGURE 6-8

INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

1969 DISTRIBUTION

TOTAL CAPACITY
78,003,935

TOTAL NUMBER OF UNITS
1043

NOTE
SOLID LINE REPRESENTS UNITS DISTRIBUTION

BROKEN LINE REPRESENTS CAPACITY DISTRIBUTION

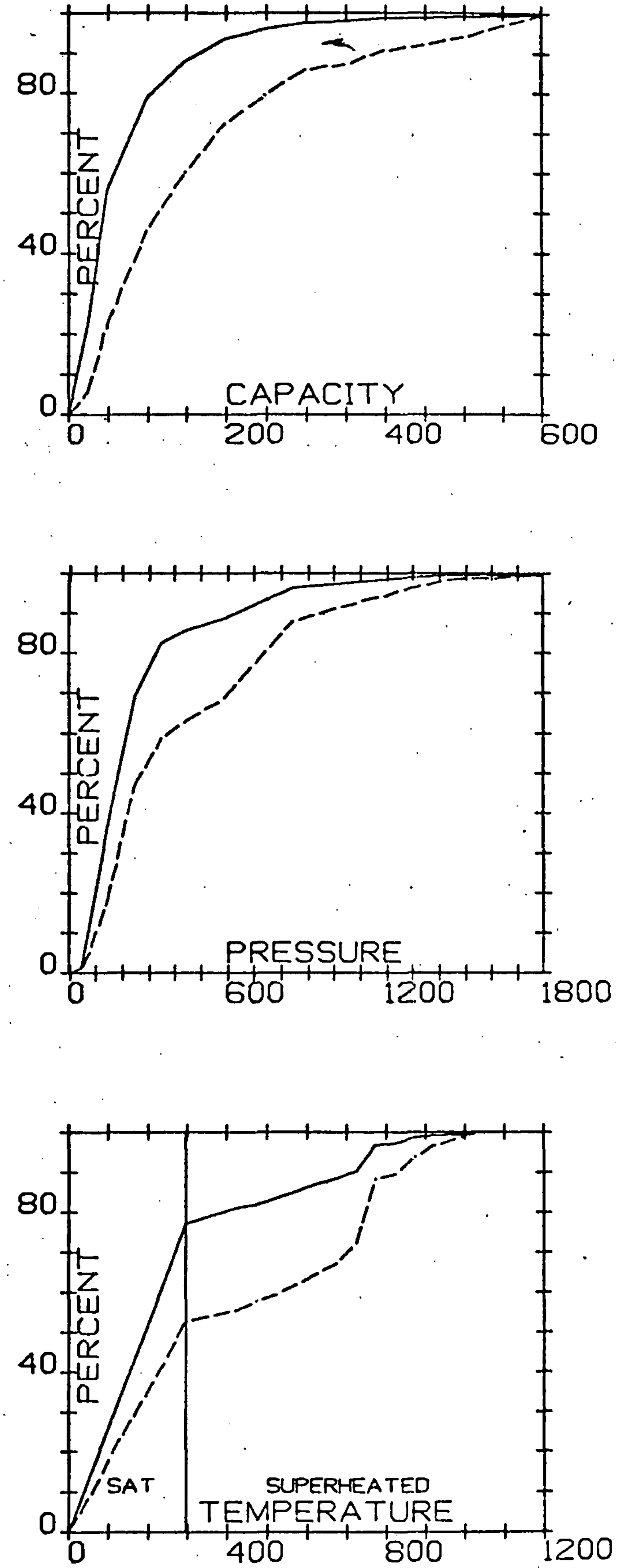


FIGURE 6-9

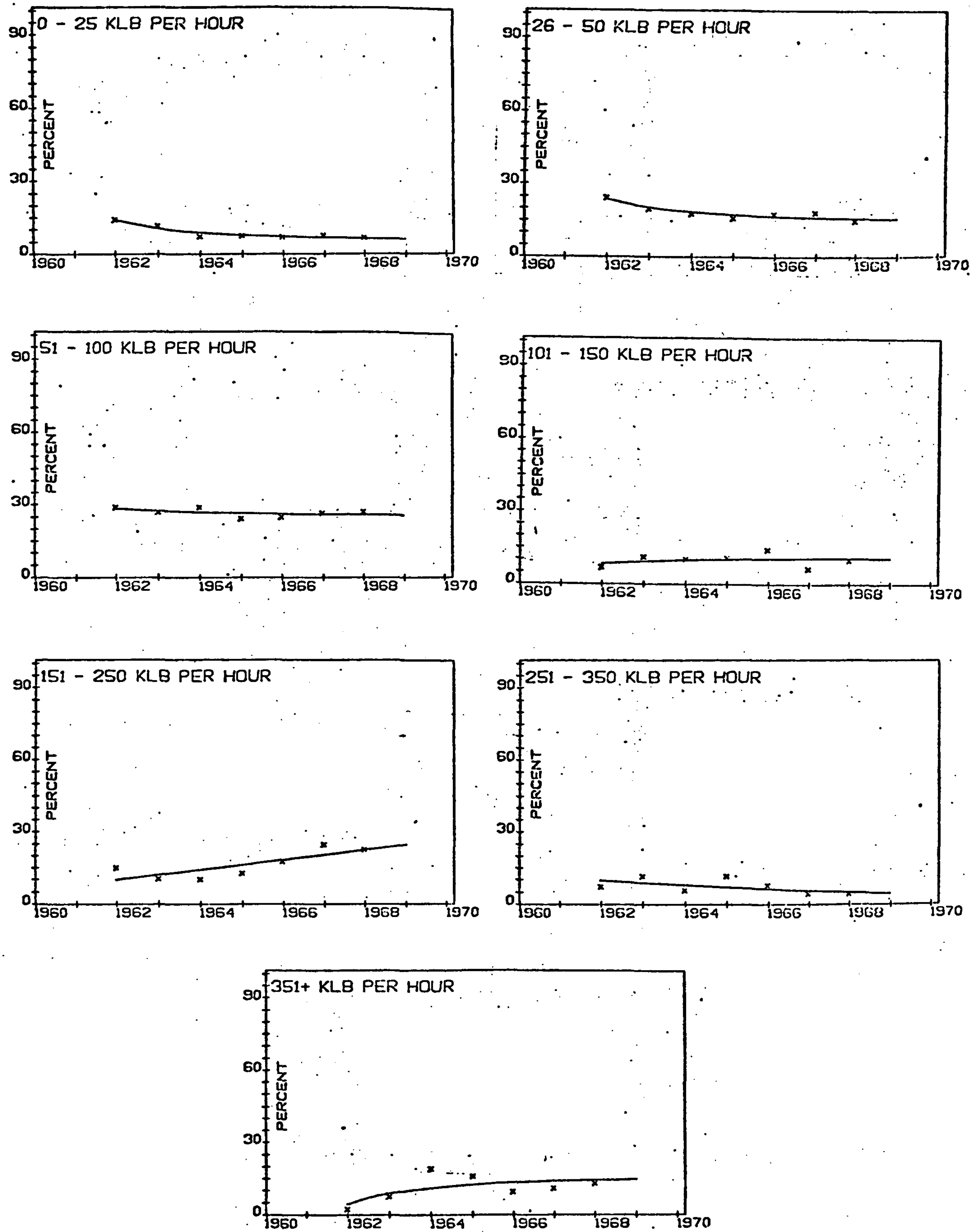

0 - 25 KLB PER HOUR
26 - 50 KLB PER HOUR
51 - 100 KLB PER HOUR
101 - 150 KLB PER HOUR
151 - 250 KLB PER HOUR
251 - 350 KLB PER HOUR
351+ KLB PER HOUR
PERCENT
0
30
60
90
1960
1962
1964
1966
1968
1970

Figure 7

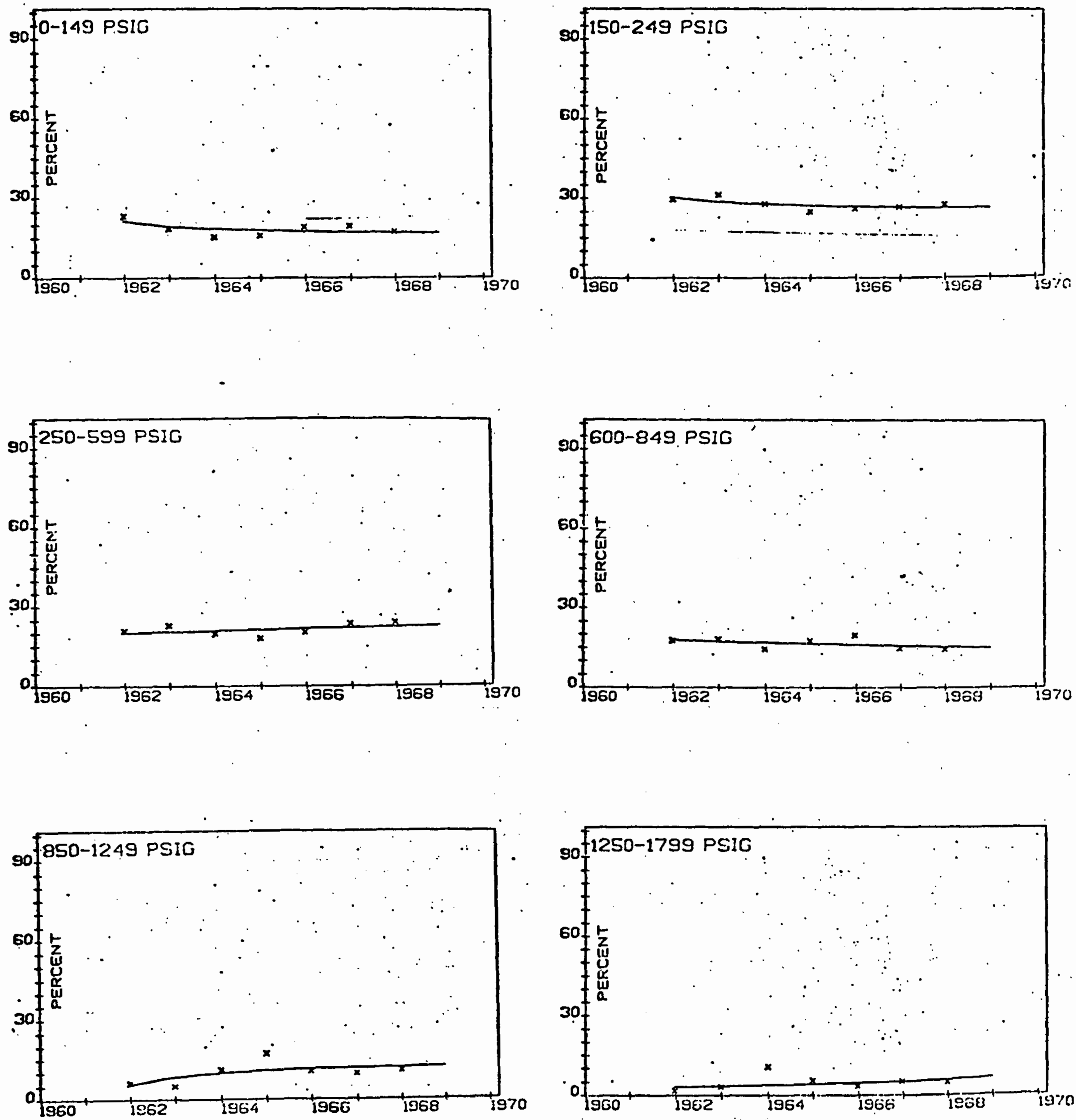

0-149 PSIG
150-249 PSIG
250-599 PSIG
600-849 PSIG
850-1249 PSIG
1250-1799 PSIG
PERCENT
0
30
60
90
1960
1962
1964
1966
1968
1970

Figure 8

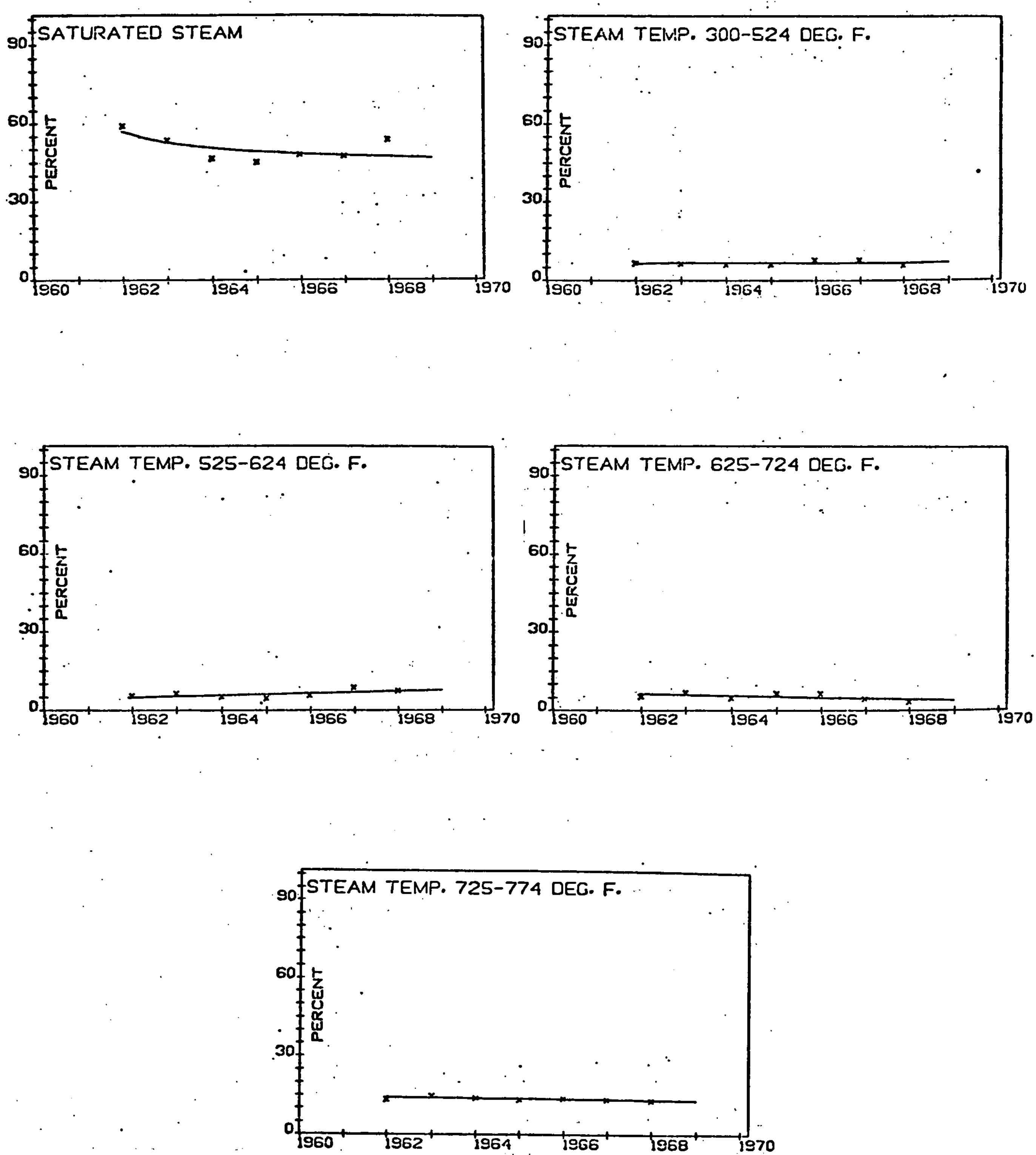

SATURATED STEAM
STEAM TEMP. 300-524 DEG. F.
STEAM TEMP. 525-624 DEG. F.
STEAM TEMP. 625-724 DEG. F.
STEAM TEMP. 725-774 DEG. F.
PERCENT
0
30
60
90
1960
1962
1964
1966
1968
1970

Figure 9-1

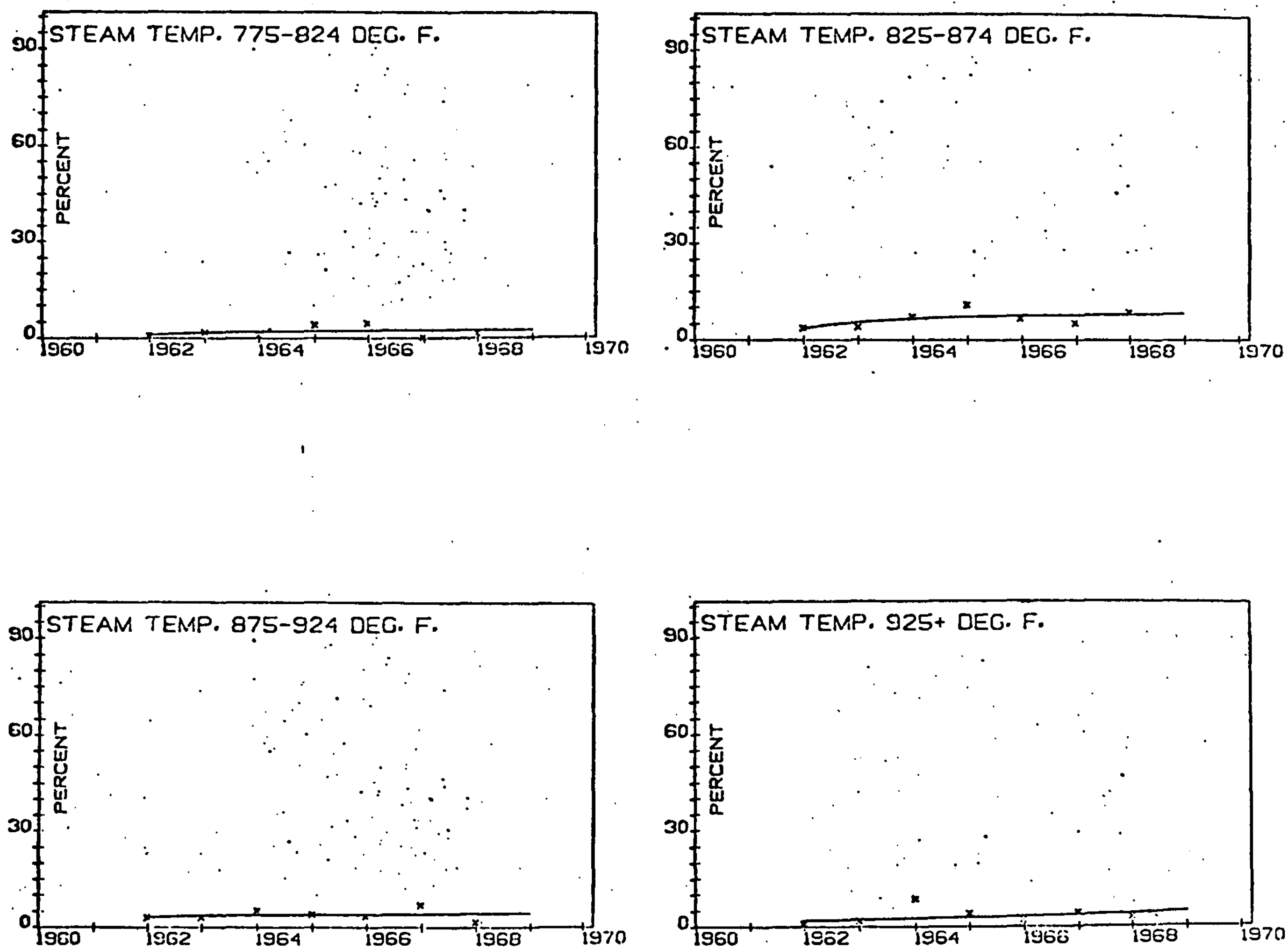

STEAM TEMP. 775-824 DEG. F.
STEAM TEMP. 825-874 DEG. F.
STEAM TEMP. 875-924 DEG. F.
STEAM TEMP. 925+ DEG. F.
PERCENT
0
30
60
90
1960
1962
1964
1966
1968
1970

Figure 9-2

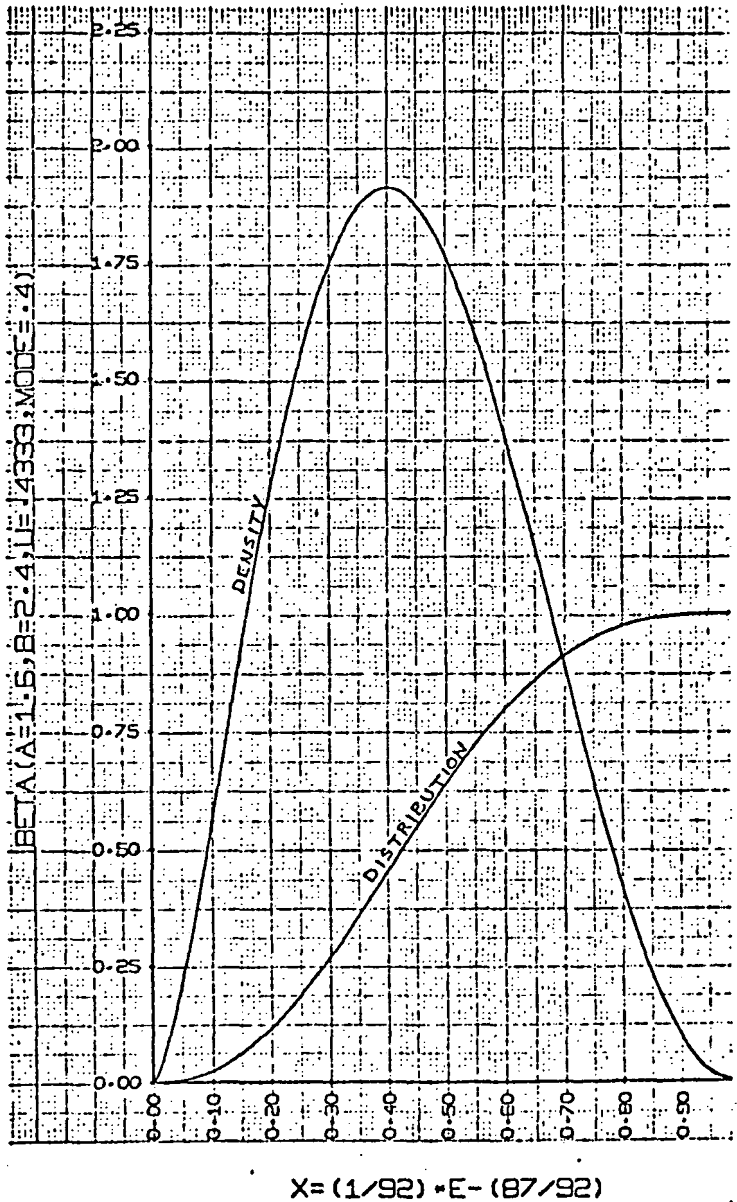


Figure 10

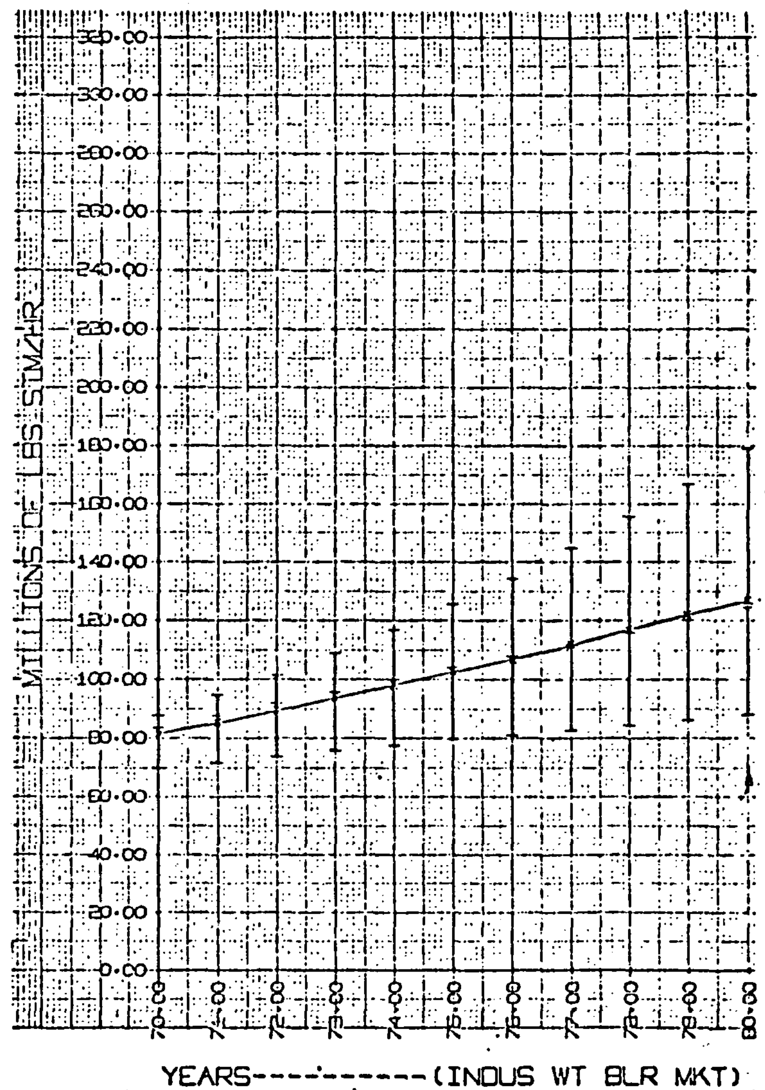


Figure 11

INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

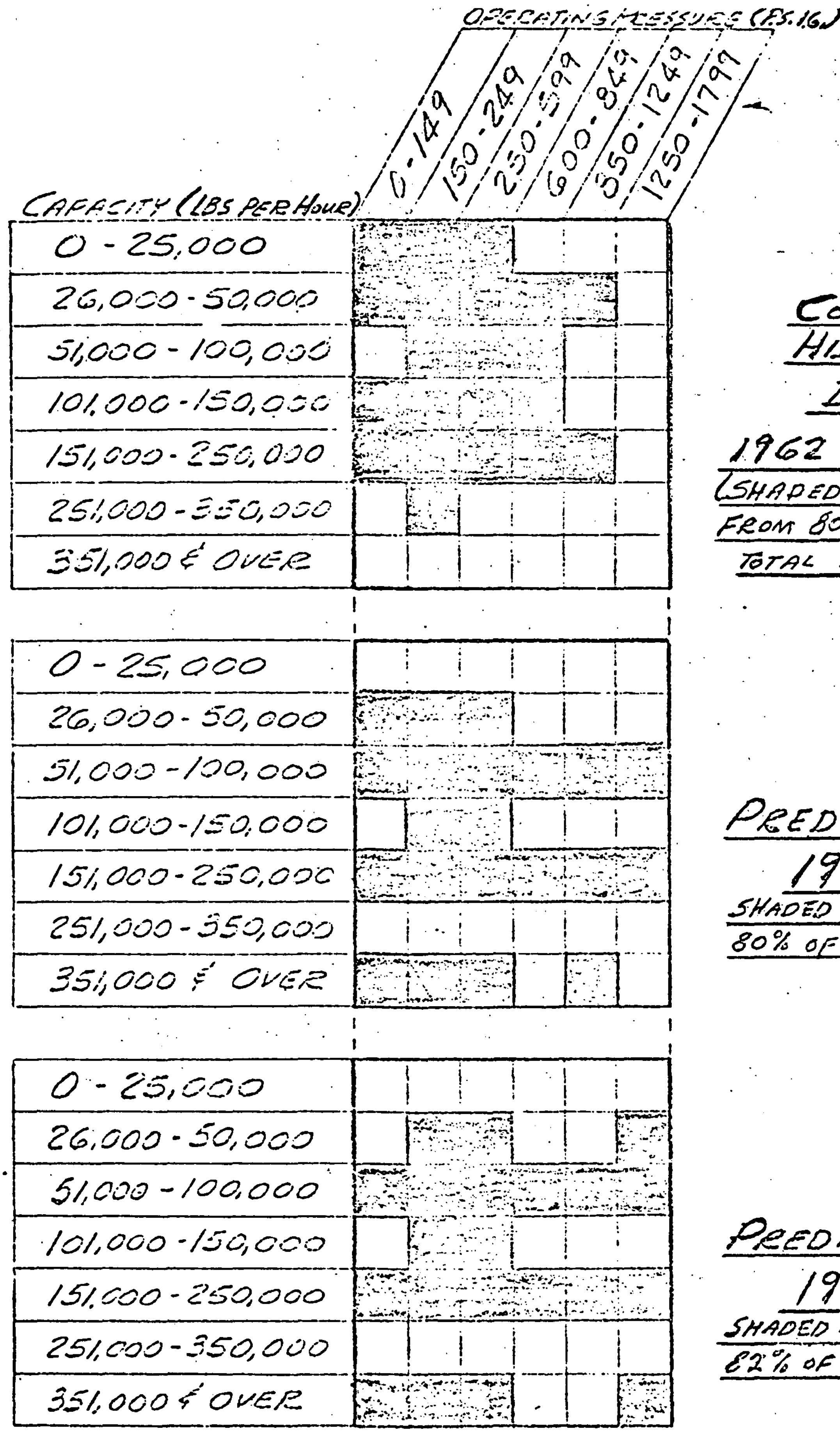
LONG-RANGE FORECAST RESULTS
USING TREND INTERVAL PROCEDURE

TREND LINE ESTIMATES

Year	Pessimistic Estimate	Most Likely Estimate	Optimistic Estimate	Statistically Expected Value
1970	70	83	87	82
1971	72	87	94	86
1972	73	91	101	90
1973	75	95	108	94
1974	77	99	116	98
1975	79	103	125	103
1976	80	107	134	107
1977	82	112	144	112
1978	84	116	155	117
1979	86	120	166	122
1980	87	124	179	127
1981	89	128	192	132
1982	91	132	206	137
1983	93	136	221	143
1984	94	140	237	149
1985	96	144	255	154

NOTE: Figures are for aggregate demand in millions of pounds of steam

Figure 12



COMBINED HISTORICAL DATA

1962 THRU 1968

(SHADED AREA INCLUDES FROM 80% TO 90% OF TOTAL MARKET)

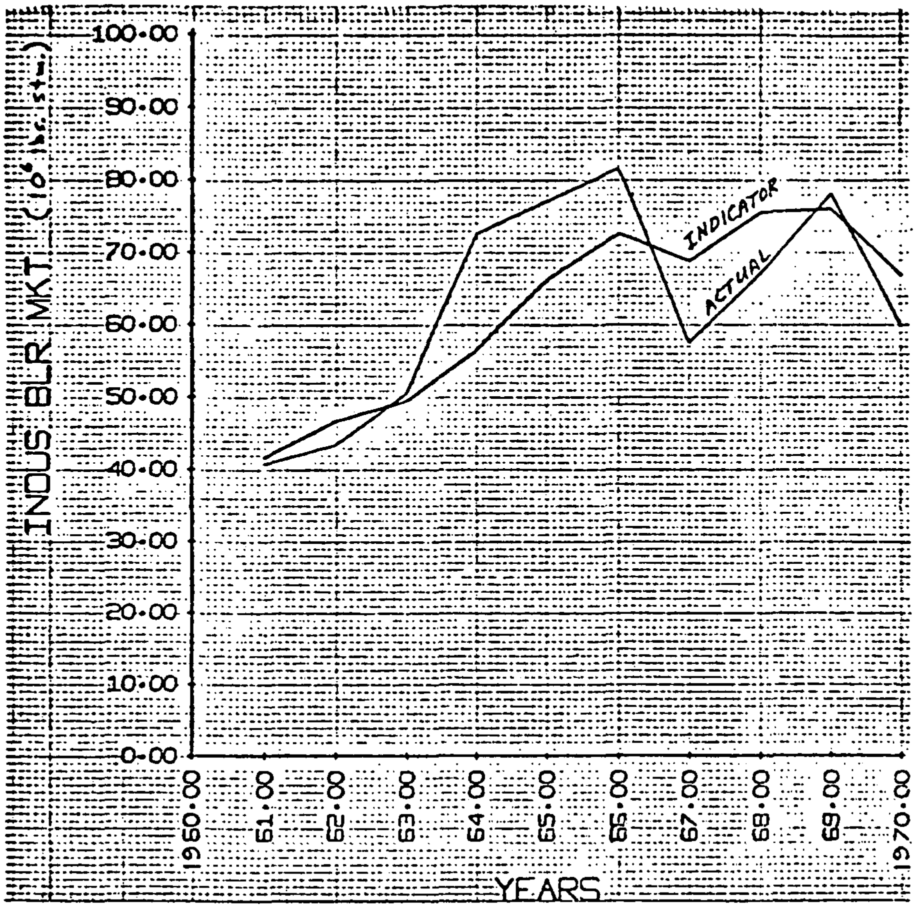
PREDICTED FOR 1975

SHADED AREA INCLUDES 80% OF TOTAL MARKET

PREDICTED FOR 1980

SHADED AREA INCLUDES 82% OF TOTAL MARKET

Figure 13



	FORECAST	
	ECONOMIC INDICATOR	TREND INTERVAL
	Predicted	Statistically Expected
1970	67	82
1971	70	86
1972	75	90
1973	81	94
1974	88	98
1975	95	103
1976	103	107
1977	110	112
1978	116	117
1979	122	122
1980	126	127

Figure 14

APPENDIX I

ABMA DATA

AB
MA

AMERICAN BOILER MANUFACTURERS ASSOCIATION

1180 RAYMOND BOULEVARD NEWARK, NEW JERSEY 07102 TEL: AREA 201: 623-8040

November 25, 1970

RECEIVED
NOV 30 1970
MAILING DEPT.
Erie City Energy Div.

Mr. R. N. Mosher
Erie City Energy Division
1422 East Avenue
Erie, Pa.

Subject: ABMA Policy - Distribution of Watertube Statistics to Federal Agencies

Dear Russ:

This letter will confirm our telephone conversation today regarding Association policy on the distribution of watertube statistics to agencies of the federal government.

It has long been a policy of the Watertube Section to make available its statistics to any agency of the federal government which has a need for this information. This policy applies, of course, to our distribution of these statistics from this office. Certainly it applies equally to the use of ABMA statistics by Watertube Section members in any work with the federal government, which requires the use of this data.

Therefore, I feel it is entirely appropriate for you to quote the Association as the source for the project you described to me over the telephone to me this morning.

Best regards.

Sincerely,

Bill Marx

William B. Marx
Manager

WBM:kw

AMERICAN BOILER MANUFACTURERS ASSOCIATION (ABMA)

STATIONARY WATERTUBE BOILER MARKET

Year	Market (1)	Industrial Market (1)	Percentage of Industrial to Total
1937	44.2		
1938	24.6		
1939	53.6		
1940	79.8		
1941	108.1		
1942	112.9		
1943	40.8		
1944	36.6		
1945	89.3		
1946	136.5		
1947	128.1		
1948	111.3		
1949	60.1		
1950	180.5		
1951	191.9		
1952	119.3		
1953	77.8		
1954	75.8		
1955	160.7		
1956	224.6		
1957	125.1		
1958	61.2		
1959	124.1		
1960	116.4		
1961	140.5	40.64	28.925
1962	120.6	43.33	35.928
1963	162.3	50.54	31.137
1964	209.0	72.35	34.616
1965	250.1	76.88	30.742
1966	249.9	81.44	32.588
1967	259.8	57.26	22.039
1968	262.1	67.00	25.559
1969	281.5	78.00	27.710

NOTE: 1. Figures are in millions of pounds of steam per hour (aggregate output sold).

AMERICAN BOILER MANUFACTURERS ASSOCIATION (ABMA)

INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

GROUPING INTERVALS DEFINED FOR ABMA DESCRIPTIVE STATISTICS

	CATEGORY					
	CAPACITY		PRESSURE		TEMPERATURE	
	GT	LTE	GTE	LT	GTE	LT
1	0	25	0	50	Saturated	
2	25	50	50	150	300	425
3	50	100	150	250	425	475
4	100	150	250	350	475	525
5	150	200	350	450	525	625
6	200	250	450	600	625	675
7	250	300	600	850	675	725
8	300	350	850	1250	725	775
9	350	400	1250	1450	775	825
10	400	450	1450		825	875
11	450	500			875	925
12	500				925	1025
13					1025	

NOTE:
1. LT denotes "Less Than"
2. LTE denotes "Less Than or Equal to"
3. GT denotes "Greater Than"
4. GTE denotes "Greater Than or Equal to"
5. ABMA data summarizes each category interval for number of units and steam output
6. Capacity is in thousands of pounds of steam per hour
7. Pressure is in PSIG.
8. Temperature is in degrees Fahrenheit

AMERICAN BOILER MANUFACTURERS ASSOCIATION (ABMA)

INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

CATEGORIES DEFINED FOR ABMA DESCRIPTIVE STATISTICS

	BASE FUEL	FIRING METHOD FOR SOLID FUELS
1.	BITUMINOUS COAL	PULVERIZED
2.	OIL	SPREADER
3.	GAS	UNDERFEED
4.	WOOD	OVERFEED
5.	BAGASSE	OTHER
6.	BLACK LIQUOR	
7.	OTHER	

NOTE: THE SAME CATEGORIES EXIST FOR ALTERNATE FUELS

	MARKET	CONSTRUCTION
1.	NON-MANUFACTURING	PACKAGED (SHOP ASSEMBLED)
2.	CHEMICAL	FIELD ERECTED
3.	PAPER	
4.	PETROLEUM	
5.	FOOD	
6.	METALS	
7.	MISCELLANEOUS MANUFACTURING	
8.	TEXTILES	
9.	TRANSPORTATION	
10.	WOOD	
11.	RUBBER	
12.	TOBACCO	

APPENDIX II

CAPACITY

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 0-25 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	6.20	43.3				14.32			
			-0.30				-2.64		
1963	5.90	50.5		-0.20		11.68		-1.58	
			-0.50		1.30		-4.21		6.13
1964	5.40	72.3		1.10		7.47		4.55	
			0.60		-1.90		0.33		-5.56
1965	6.00	76.9		-0.80		7.80		-1.01	
			-0.20		-0.40		-0.68		2.24
1966	5.80	81.4		-1.20		7.13		1.23	
			-1.40		2.80		0.55		-2.60
1967	4.40	57.3		1.60		7.68		-1.37	
			0.20		-2.00		-0.81		0.96
1968	4.60	67.0		-0.40		6.87		-0.41	
			-0.20				-1.22		
1969	4.40	78.0				5.64			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	348.00	877.0				39.68			
			-10.00				-1.23		
1963	338.00	879.0		-36.00		38.45		-8.96	
			-46.00		112.00		-10.19		21.40
1964	292.00	1033.0		76.00		28.27		12.44	
			30.00		-129.00		2.25		-19.55
1965	322.00	1055.0		-53.00		30.52		-7.11	
			-23.00		9.00		-4.86		13.95
1966	299.00	1165.0		-44.00		25.67		6.84	
			-67.00		114.00		1.99		-10.60
1967	232.00	839.0		70.00		27.65		-3.76	
			3.00		-71.00		-1.77		2.37
1968	235.00	908.0		-1.00		25.88		-1.39	
			2.00				-3.16		
1969	237.00	1043.0				22.72			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 26-50 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	10.60	43.3				24.48			
			-0.60				-4.68		
1963	10.00	50.5		3.50		19.80		2.72	
			2.90		-6.80		-1.96		-2.35
1964	12.90	72.3		-3.30		17.84		0.37	
			-0.40		5.80		-1.59		2.90
1965	12.50	76.9		2.50		16.25		3.27	
			2.10		-8.70		1.68		-4.56
1966	14.60	81.4		-6.20		17.94		-1.29	
			-4.10		9.80		0.39		-2.49
1967	10.50	57.3		3.60		18.32		-3.79	
			-0.50		0.20		-3.40		9.31
1968	10.00	67.0		3.80		14.93		5.53	
			3.30				2.13		
1969	13.30	78.0				17.05			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	287.00	877.0				32.73			
			-26.00				-3.03		
1963	261.00	879.0		105.00		29.69		6.25	
			79.00		-200.00		3.22		-11.68
1964	340.00	1033.0		-95.00		32.91		-5.42	
			-16.00		177.00		-2.20		10.39
1965	324.00	1055.0		82.00		30.71		4.97	
			66.00		-261.00		2.77		-8.19
1966	390.00	1165.0		-179.00		33.48		-3.23	
			-113.00		277.00		-0.46		-0.47
1967	277.00	839.0		98.00		33.02		-3.70	
			-15.00		-2.00		-4.16		11.89
1968	262.00	908.0		96.00		28.85		8.19	
			81.00				4.03		
1969	343.00	1043.0				32.89			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 51-100 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	12.50	43.3				28.87			
			1.20				-1.74		
1963	13.70	50.5		6.10		27.13		3.66	
			7.30		-15.50		1.92		-10.04
1964	21.00	72.3		-9.40		29.05		-6.39	
			-2.10		13.10		-4.47		11.46
1965	18.90	76.9		3.70		24.58		5.08	
			1.60		-10.50		0.61		-4.16
1966	20.50	81.4		-6.80		25.18		0.91	
			-5.20		15.10		1.52		-1.67
1967	15.30	57.3		8.30		26.70		-0.76	
			3.10		-11.80		0.76		-4.39
1968	18.40	67.0		-3.50		27.46		-5.15	
			-0.40				-4.39		
1969	18.00	78.0				23.08			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	173.00	877.0				19.73			
			8.00				0.87		
1963	181.00	879.0		76.00		20.59		4.20	
			84.00		-179.00		5.06		-11.59
1964	265.00	1033.0		-103.00		25.65		-7.40	
			-19.00		151.00		-2.34		10.02
1965	246.00	1055.0		48.00		23.32		2.62	
			29.00		-152.00		0.29		-2.68
1966	275.00	1165.0		-104.00		23.61		-0.05	
			-75.00		225.00		0.23		3.08
1967	200.00	839.0		121.00		23.84		3.02	
			46.00		-170.00		3.25		-10.07
1968	246.00	908.0		-49.00		27.09		-7.05	
			-3.00				-3.79		
1969	243.00	1043.0				23.30			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 101-150 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	2.90	43.3				6.70			
			2.80				4.59		
1963	5.70	50.5		-1.00		11.29		-5.50	
			1.80		-0.00		-0.91		6.84
1964	7.50	72.3		-1.00		10.37		1.33	
			0.80		3.50		0.42		1.70
1965	8.30	76.9		2.50		10.79		3.04	
			3.30		-13.60		3.46		-14.11
1966	11.60	81.4		-11.10		14.25		-11.08	
			-7.80		21.90		-7.62		22.21
1967	3.80	57.3		10.80		6.63		11.14	
			3.00		-8.90		3.52		-9.80
1968	6.80	67.0		1.90		10.15		1.33	
			4.90				4.85		
1969	11.70	78.0				15.00			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	24.00	877.0				2.74			
			20.00				2.27		
1963	44.00	879.0		-3.00		5.01		-1.37	
			17.00		-11.00		0.90		0.63
1964	61.00	1033.0		-14.00		5.91		-0.74	
			3.00		37.00		0.16		2.24
1965	64.00	1055.0		23.00		6.07		1.50	
			26.00		-107.00		1.66		-7.07
1966	90.00	1165.0		-84.00		7.73		-5.57	
			-58.00		167.00		-3.91		11.94
1967	32.00	839.0		83.00		3.81		6.37	
			25.00		-66.00		2.46		-5.62
1968	57.00	908.0		17.00		6.28		0.75	
			42.00				3.21		
1969	99.00	1043.0				9.49			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 151-200 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	3.10	43.3				7.16			
			-0.20				-1.42		
1963	2.90	50.5		2.80		5.74		3.28	
			2.60		-5.70		1.86		-5.99
1964	5.50	72.3		-2.90		7.61		-2.71	
			-0.30		9.30		-0.85		10.67
1965	5.20	76.9		6.40		6.76		7.97	
			6.10		-14.90		7.12		-13.44
1966	11.30	81.4		-8.50		13.88		-5.47	
			-2.40		12.70		1.65		4.26
1967	8.90	57.3		4.20		15.53		-1.21	
			1.80		-7.60		0.44		-3.53
1968	10.70	67.0		-3.40		15.97		-4.74	
			-1.60				-4.30		
1969	9.10	78.0				11.67			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	17.00	877.0				1.94			
			-1.00				-0.12		
1963	16.00	879.0		14.00		1.82		1.11	
			13.00		-29.00		0.99		-2.34
1964	29.00	1033.0		-15.00		2.81		-1.24	
			-2.00		49.00		-0.25		3.99
1965	27.00	1055.0		34.00		2.56		2.75	
			32.00		-72.00		2.51		-4.01
1966	59.00	1165.0		-38.00		5.06		-1.25	
			-6.00		54.00		1.25		0.62
1967	53.00	839.0		16.00		6.32		-0.63	
			10.00		-32.00		0.62		-1.46
1968	63.00	908.0		-16.00		6.94		-2.09	
			-6.00				-1.47		
1969	57.00	1043.0				5.47			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 201-250 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	3.50	43.3				8.08			
			-1.00				-3.13		
1963	2.50	50.5		0.40		4.95		0.81	
			-0.60		3.00		-2.32		5.00
1964	1.90	72.3		3.40		2.63		5.81	
			2.80		-7.70		3.48		-11.47
1965	4.70	76.9		-4.30		6.11		-5.66	
			-1.50		7.90		-2.18		13.16
1966	3.20	81.4		3.60		3.93		7.50	
			2.10		-6.40		5.32		-15.20
1967	5.30	57.3		-2.80		9.25		-7.70	
			-0.70		4.50		-2.38		10.40
1968	4.60	67.0		1.70		6.87		2.70	
			1.00				0.31		
1969	5.60	78.0				7.18			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	15.00	877.0				1.71			
			-4.00				-0.46		
1963	11.00	879.0		1.00		1.25		-0.02	
			-3.00		14.00		-0.48		1.62
1964	8.00	1033.0		15.00		0.77		1.60	
			12.00		-33.00		1.12		-3.41
1965	20.00	1055.0		-18.00		1.90		-1.82	
			-6.00		33.00		-0.69		4.05
1966	14.00	1165.0		15.00		1.20		2.23	
			9.00		-28.00		1.54		-4.42
1967	23.00	839.0		-13.00		2.74		-2.19	
			-4.00		22.00		-0.65		3.05
1968	19.00	908.0		9.00		2.09		0.86	
			5.00				0.21		
1969	24.00	1043.0				2.30			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 251-300 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.20	43.3				2.77			
			1.70				2.97		
1963	2.90	50.5		-2.00		5.74		-5.12	
			-0.30		3.40		-2.15		8.48
1964	2.60	72.3		1.40		3.60		3.36	
			1.10		-1.00		1.22		-3.00
1965	3.70	76.9		0.40		4.81		0.36	
			1.50		-6.60		1.58		-7.45
1966	5.20	81.4		-6.20		6.39		-7.09	
			-4.70		12.10		-5.52		14.27
1967	0.50	57.3		5.90		0.87		7.18	
			1.20		-3.60		1.66		-4.72
1968	1.70	67.0		2.30		2.54		2.46	
			3.50				4.13		
1969	5.20	78.0				6.67			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	4.00	877.0				0.46			
			6.00				0.68		
1963	10.00	879.0		-7.00		1.14		-0.95	
			-1.00		12.00		-0.27		1.58
1964	9.00	1033.0		5.00		0.87		0.63	
			4.00		-4.00		0.36		-0.68
1965	13.00	1055.0		1.00		1.23		-0.05	
			5.00		-22.00		0.31		-1.57
1966	18.00	1165.0		-21.00		1.55		-1.62	
			-16.00		41.00		-1.31		3.35
1967	2.00	839.0		20.00		0.24		1.73	
			4.00		-12.00		0.42		-1.09
1968	6.00	908.0		8.00		0.66		0.64	
			12.00				1.06		
1969	18.00	1043.0				1.73			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 301-350 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	2.00	43.3				4.62			
			1.00				1.32		
1963	3.00	50.5		-2.30		5.94		-4.91	
			-1.30		7.20		-3.59		13.04
1964	1.70	72.3		4.90		2.35		8.13	
			3.60		-12.40		4.54		-17.84
1965	5.30	76.9		-7.50		6.89		-9.71	
			-3.90		12.00		-5.17		16.66
1966	1.40	81.4		4.50		1.72		6.94	
			0.60		-5.80		1.77		-10.26
1967	2.00	57.3		-1.30		3.49		-3.32	
			-0.70		1.70		-1.55		4.21
1968	1.30	67.0		0.40		1.94		0.89	
			-0.30				-0.66		
1969	1.00	78.0				1.28			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	6.00	877.0				0.68			
			3.00				0.34		
1963	9.00	879.0		-7.00		1.02		-0.88	
			-4.00		22.00		-0.54		2.45
1964	5.00	1033.0		15.00		0.48		1.57	
			11.00		-38.00		1.03		-3.78
1965	16.00	1055.0		-23.00		1.52		-2.21	
			-12.00		37.00		-1.17		3.75
1966	4.00	1165.0		14.00		0.34		1.55	
			2.00		-18.00		0.37		-2.19
1967	6.00	839.0		-4.00		0.72		-0.65	
			-2.00		5.00		-0.27		0.77
1968	4.00	908.0		1.00		0.44		0.12	
			-1.00				-0.15		
1969	3.00	1043.0				0.29			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 351-400 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.10	43.3				2.54			
			0.50				0.63		
1963	1.60	50.5		0.20		3.17		-0.62	
			0.70		-1.30		0.01		-0.11
1964	2.30	72.3		-1.10		3.18		-0.72	
			-0.40		0.80		-0.71		0.44
1965	1.90	76.9		-0.30		2.47		-0.29	
			-0.70		0.90		-1.00		1.73
1966	1.20	81.4		0.60		1.47		1.44	
			-0.10		-0.00		0.45		-1.42
1967	1.10	57.3		0.60		1.92		0.02	
			0.50		0.00		0.47		0.58
1968	1.60	67.0		0.60		2.39		0.61	
			1.10				1.07		
1969	2.70	78.0				3.46			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	3.00	877.0				0.34			
			1.00				0.11		
1963	4.00	879.0		1.00		0.46		0.01	
			2.00		-4.00		0.13		-0.25
1964	6.00	1033.0		-3.00		0.58		-0.23	
			-1.00		2.00		-0.11		0.12
1965	5.00	1055.0		-1.00		0.47		-0.11	
			-2.00		3.00		-0.22		0.43
1966	3.00	1165.0		2.00		0.26		0.32	
			0.00		-1.00		0.10		-0.33
1967	3.00	839.0		1.00		0.36		-0.02	
			1.00		1.00		0.08		0.16
1968	4.00	908.0		2.00		0.44		0.15	
			3.00				0.23		
1969	7.00	1043.0				0.67			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 401-450 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.00	43.3				0.00			
			0.40				0.79		
1963	0.40	50.5		2.20		0.79		2.57	
			2.60		-6.10		3.36		-7.86
1964	3.00	72.3		-3.90		4.15		-5.30	
			-1.30		5.70		-1.94		7.73
1965	1.70	76.9		1.80		2.21		2.43	
			0.50		-2.80		0.49		-2.66
1966	2.20	81.4		-1.00		2.70		-0.23	
			-0.50		0.20		0.26		-2.41
1967	1.70	57.3		-0.80		2.97		-2.63	
			-1.30		2.60		-2.37		5.56
1968	0.40	67.0		1.80		0.60		2.93	
			0.50				0.56		
1969	0.90	78.0				1.15			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.00	877.0				0.00			
			1.00				0.11		
1963	1.00	879.0		5.00		0.11		0.45	
			6.00		-14.00		0.56		-1.31
1964	7.00	1033.0		-9.00		0.68		-0.86	
			-3.00		13.00		-0.30		1.21
1965	4.00	1055.0		4.00		0.38		0.35	
			1.00		-6.00		0.05		-0.35
1966	5.00	1165.0		-2.00		0.43		-0.00	
			-1.00		0.00		0.05		-0.41
1967	4.00	839.0		-2.00		0.48		-0.41	
			-3.00		6.00		-0.37		0.86
1968	1.00	908.0		4.00		0.11		0.45	
			1.00				0.08		
1969	2.00	1043.0				0.19			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 451-500 UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.00	43.3				0.00			
			0.90				1.78		
1963	0.90	50.5		0.20		1.78		-0.80	
			1.10		-0.30		0.98		0.95
1964	2.00	72.3		-0.10		2.77		0.15	
			1.00		-2.50		1.13		-3.47
1965	3.00	76.9		-2.60		3.90		-3.32	
			-1.60		5.30		-2.18		8.14
1966	1.40	81.4		2.70		1.72		4.82	
			1.10		-4.30		2.64		-8.85
1967	2.50	57.3		-1.60		4.36		-4.02	
			-0.50		1.60		-1.38		4.34
1968	2.00	67.0		0.00		2.99		0.32	
			-0.50				-1.06		
1969	1.50	78.0				1.92			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.00	877.0				0.00			
			2.00				0.23		
1963	2.00	879.0		0.00		0.23		-0.07	
			2.00		0.00		0.16		0.09
1964	4.00	1033.0		0.00		0.39		0.02	
			2.00		-5.00		0.18		-0.51
1965	6.00	1055.0		-5.00		0.57		-0.49	
			-3.00		10.00		-0.31		1.14
1966	3.00	1165.0		5.00		0.26		0.65	
			2.00		-8.00		0.34		-1.14
1967	5.00	839.0		-3.00		0.60		-0.49	
			-1.00		3.00		-0.16		0.50
1968	4.00	908.0		0.00		0.44		0.00	
			-1.00				-0.15		
1969	3.00	1043.0				0.29			

NOTE**. DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO SIZE OF UNIT 500 + UNIT CAP.

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.00	43.3				0.00			
			1.20				2.38		
1963	1.20	50.5		4.20		2.38		4.38	
			5.40		-10.40		6.75		-12.71
1964	6.60	72.3		-6.20		9.13		-6.34	
			-0.80		4.20		-1.59		6.07
1965	5.80	76.9		-2.00		7.54		-2.27	
			-2.80		2.90		-3.86		4.36
1966	3.00	81.4		0.90		3.69		2.09	
			-1.90		4.70		-1.77		4.92
1967	1.10	57.3		5.60		1.92		7.01	
			3.70		-9.50		5.24		-13.52
1968	4.80	67.0		-3.90		7.16		-6.51	
			-0.20				-1.27		
1969	4.60	78.0				5.90			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.00	877.0				0.00			
			2.00				0.23		
1963	2.00	879.0		3.00		0.23		0.22	
			5.00		-7.00		0.45		-0.59
1964	7.00	1033.0		-4.00		0.68		-0.37	
			1.00		0.00		0.08		-0.04
1965	8.00	1055.0		-4.00		0.76		-0.41	
			-3.00		4.00		-0.33		0.55
1966	5.00	1165.0		0.00		0.43		0.14	
			-3.00		8.00		-0.19		0.59
1967	2.00	839.0		8.00		0.24		0.72	
			5.00		-13.00		0.53		-1.36
1968	7.00	908.0		-5.00		0.77		-0.63	
			0.00				-0.10		
1969	7.00	1043.0				0.67			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

APPENDIX III

PRESSURE

E R I E C I T Y E N E R G Y D I V.
Z U R N I N D U S T R I E S, I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO OPERATING PRESSURE 0-49 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.20	43.3				0.46			
			0.00				-0.07		
1963	0.20	50.5		0.10		0.40		0.08	
			0.10		-0.30		0.02		-0.26
1964	0.30	72.3		-0.20		0.41		-0.17	
			-0.10		0.30		-0.15		0.31
1965	0.20	76.9		0.10		0.26		0.14	
			0.00		-0.20		-0.01		-0.20
1966	0.20	81.4		-0.10		0.25		-0.06	
			-0.10		0.30		-0.07		0.25
1967	0.10	57.3		0.20		0.17		0.20	
			0.10		-0.30		0.12		-0.36
1968	0.20	66.9		-0.10		0.30		-0.17	
			0.00				-0.04		
1969	0.20	78.0				0.26			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	11.00	877.0				1.25			
			5.00				0.57		
1963	16.00	879.0		-6.00		1.82		-0.93	
			-1.00		-1.00		-0.37		0.51
1964	15.00	1033.0		-7.00		1.45		-0.42	
			-8.00		18.00		-0.79		1.40
1965	7.00	1055.0		11.00		0.66		0.98	
			3.00		-17.00		0.19		-1.20
1966	10.00	1165.0		-6.00		0.86		-0.22	
			-3.00		13.00		-0.02		0.62
1967	7.00	839.0		7.00		0.83		0.40	
			4.00		-8.00		0.36		-0.65
1968	11.00	908.0		-1.00		1.21		-0.25	
			3.00				0.13		
1969	14.00	1043.0				1.34			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO OPERATING PRESSURE 50-149 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	9.80	43.3				22.63			
			-0.60				-4.41		
1963	9.20	50.5		2.40		18.22		1.41	
			1.80		-3.00		-3.00		2.24
1964	11.00	72.3		-0.60		15.21		3.65	
			1.20		2.80		0.65		-1.00
1965	12.20	76.9		2.20		15.86		2.65	
			3.40		-10.20		3.30		-5.92
1966	15.60	81.4		-8.00		19.16		-3.27	
			-4.60		13.20		0.03		1.38
1967	11.00	57.3		5.20		19.20		-1.89	
			0.60		-3.30		-1.86		4.49
1968	11.60	66.9		1.90		17.34		2.60	
			2.50				0.74		
1969	14.10	78.0				18.08			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	358.00	877.0				40.82			
			-17.00				-2.03		
1963	341.00	879.0		22.00		38.79		-3.27	
			5.00		26.00		-5.30		12.90
1964	346.00	1033.0		48.00		33.49		9.62	
			53.00		-41.00		4.33		-12.37
1965	399.00	1055.0		7.00		37.82		-2.75	
			60.00		-201.00		1.58		0.50
1966	459.00	1165.0		-194.00		39.40		-2.24	
			-134.00		317.00		-0.66		-1.25
1967	325.00	839.0		123.00		38.74		-3.49	
			-11.00		-59.00		-4.16		8.25
1968	314.00	908.0		64.00		34.58		4.76	
			53.00				0.61		
1969	367.00	1043.0				35.19			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO OPERATING PRS. 150-249 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	12.80	43.3				29.56			
			3.10				1.92		
1963	15.90	50.5		1.30		31.49		-5.33	
			4.40		-6.70		-3.41		5.76
1964	20.30	72.3		-5.40		28.08		0.43	
			-1.00		8.50		-2.98		3.74
1965	19.30	76.9		3.10		25.10		4.17	
			2.10		-11.20		1.19		-4.78
1966	21.40	81.4		-8.10		26.29		-0.61	
			-6.00		17.20		0.59		0.80
1967	15.40	57.3		9.10		26.88		0.19	
			3.10		-8.40		0.78		-0.03
1968	18.50	66.9		0.70		27.65		0.16	
			3.80				0.94		
1969	22.30	78.0				28.59			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	304.00	877.0				34.66			
			28.00				3.11		
1963	332.00	879.0		22.00		37.77		-3.90	
			50.00		-128.00		-0.79		-1.39
1964	382.00	1033.0		-106.00		36.98		-5.29	
			-56.00		202.00		-6.08		11.88
1965	326.00	1055.0		96.00		30.90		6.60	
			40.00		-226.00		0.52		-5.63
1966	366.00	1165.0		-130.00		31.42		0.96	
			-90.00		262.00		1.48		-0.32
1967	276.00	839.0		132.00		32.90		0.65	
			42.00		-153.00		2.13		-5.29
1968	318.00	908.0		-21.00		35.02		-4.65	
			21.00				-2.52		
1969	339.00	1043.0				32.50			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO OPERATING PRS. 250-349 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	3.10	43.3				7.16			
			0.40				-0.23		
1963	3.50	50.5		2.90		6.93		2.70	
			3.30		-6.80		2.47		-6.52
1964	6.80	72.3		-3.90		9.41		-3.82	
			-0.60		6.90		-1.34		7.66
1965	6.20	76.9		3.00		8.06		3.85	
			2.40		-9.00		2.50		-8.19
1966	8.60	81.4		-6.00		10.57		-4.34	
			-3.60		12.60		-1.84		9.41
1967	5.00	57.3		6.60		8.73		5.07	
			3.00		-8.10		3.23		-8.08
1968	8.00	66.9		-1.50		11.96		-3.01	
			1.50				0.22		
1969	9.50	78.0				12.18			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	69.00	877.0				7.87			
			-16.00				-1.84		
1963	53.00	879.0		65.00		6.03		5.68	
			49.00		-112.00		3.84		-9.54
1964	102.00	1033.0		-47.00		9.87		-3.86	
			2.00		55.00		-0.02		3.80
1965	104.00	1055.0		8.00		9.86		-0.06	
			10.00		-54.00		-0.07		-0.36
1966	114.00	1165.0		-46.00		9.79		-0.42	
			-36.00		106.00		-0.49		2.84
1967	76.00	839.0		60.00		9.30		2.43	
			24.00		-46.00		1.94		-2.17
1968	102.00	908.0		14.00		11.23		0.25	
			38.00				2.19		
1969	140.00	1043.0				13.42			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO OPERATING PRS. 350-449 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	3.10	43.3				7.16			
			0.80				0.56		
1963	3.90	50.5		0.20		7.72		-1.51	
			1.00		-1.50		-0.95		1.66
1964	4.90	72.3		-1.30		6.78		0.15	
			-0.30		0.20		-0.80		-1.40
1965	4.60	76.9		-1.10		5.98		-1.26	
			-1.40		4.90		-2.05		9.15
1966	3.20	81.4		3.80		3.93		7.89	
			2.40		-7.60		5.84		-17.23
1967	5.60	57.3		-3.80		9.77		-9.34	
			-1.40		4.50		-3.50		11.04
1968	4.20	66.9		0.70		6.28		1.70	
			-0.70				-1.79		
1969	3.50	78.0				4.49			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	47.00	877.0				5.36			
			-11.00				-1.26		
1963	36.00	879.0		27.00		4.10		2.20	
			16.00		-52.00		0.94		-4.10
1964	52.00	1033.0		-25.00		5.03		-1.90	
			-9.00		27.00		-0.96		1.87
1965	43.00	1055.0		2.00		4.08		-0.03	
			-7.00		14.00		-0.99		3.29
1966	36.00	1165.0		16.00		3.09		3.26	
			9.00		-27.00		2.27		-6.16
1967	45.00	839.0		-11.00		5.36		-2.90	
			-2.00		3.00		-0.63		1.96
1968	43.00	908.0		-8.00		4.74		-0.94	
			-10.00				-1.57		
1969	33.00	1043.0				3.16			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO OPERATING PRS. 450-599 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	2.90	43.3				6.70			
			1.30				1.62		
1963	4.20	50.5		-2.90		8.32		-6.34	
			-1.60		5.00		-4.72		11.50
1964	2.60	72.3		2.10		3.60		5.16	
			0.50		-0.80		0.44		-3.60
1965	3.10	76.9		1.30		4.03		1.55	
			1.80		-5.10		1.99		-4.50
1966	4.90	81.4		-3.80		6.02		-2.95	
			-2.00		6.90		-0.96		4.82
1967	2.90	57.3		3.10		5.06		1.88	
			1.10		-3.60		0.92		-2.88
1968	4.00	66.9		-0.50		5.98		-1.00	
			0.60				-0.08		
1969	4.60	78.0				5.90			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	24.00	877.0				2.74			
			2.00				0.22		
1963	26.00	879.0		-4.00		2.96		-0.86	
			-2.00		15.00		-0.63		2.30
1964	24.00	1033.0		11.00		2.32		1.44	
			9.00		-18.00		0.80		-2.37
1965	33.00	1055.0		-7.00		3.13		-0.93	
			2.00		-5.00		-0.12		1.03
1966	35.00	1165.0		-12.00		3.00		0.10	
			-10.00		29.00		-0.02		0.47
1967	25.00	839.0		17.00		2.98		0.57	
			7.00		-21.00		0.54		-1.28
1968	32.00	908.0		-4.00		3.52		-0.71	
			3.00				-0.17		
1969	35.00	1043.0				3.36			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO OPERATING PRS. 600-849 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	7.70	43.3				17.78			
			1.60				0.63		
1963	9.30	50.5		-0.40		18.42		-4.53	
			1.20		2.40		-3.89		11.71
1964	10.50	72.3		2.00		14.52		7.19	
			3.20		-2.80		3.29		-8.51
1965	13.70	76.9		-0.80		17.82		-1.33	
			2.40		-9.10		1.96		-5.40
1966	16.10	81.4		-9.90		19.78		-6.73	
			-7.50		18.70		-4.77		11.29
1967	8.60	57.3		8.80		15.01		4.56	
			1.30		-5.40		-0.21		-0.43
1968	9.90	66.9		3.40		14.80		4.13	
			4.70				3.92		
1969	14.60	78.0				18.72			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	49.00	877.0				5.59			
			11.00				1.24		
1963	60.00	879.0		0.00		6.83		-1.19	
			11.00		8.00		0.05		2.80
1964	71.00	1033.0		8.00		6.87		1.61	
			19.00		-12.00		1.66		-2.79
1965	90.00	1055.0		-4.00		8.53		-1.18	
			15.00		-62.00		0.48		-1.88
1966	105.00	1165.0		-66.00		9.01		-3.06	
			-51.00		124.00		-2.58		5.92
1967	54.00	839.0		58.00		6.44		2.86	
			7.00		-45.00		0.28		-2.09
1968	61.00	908.0		13.00		6.72		0.77	
			20.00				1.05		
1969	81.00	1043.0				7.77			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES. INC.

INDUSTRIAL WATERTUBE BOILERS. DISTRIBUTION AS TO OPERATING PRS. 850-1249 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	2.70	43.3				6.24			
			-0.10				-1.09		
1963	2.60	50.5		5.60		5.15		7.14	
			5.50		-5.90		6.05		-7.10
1964	8.10	72.3		-0.30		11.20		0.04	
			5.20		-9.70		6.09		-12.98
1965	13.30	76.9		-10.00		17.30		-12.94	
			-4.80		11.80		-6.85		16.95
1966	8.50	81.4		1.80		10.44		6.01	
			-3.00		2.90		-0.84		-4.00
1967	5.50	57.3		4.70		9.60		2.01	
			1.70		-7.50		1.16		-6.11
1968	7.20	66.9		-2.80		10.76		-4.11	
			-1.10				-2.94		
1969	6.10	78.0				7.82			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	12.00	877.0				1.37			
			-1.00				-0.12		
1963	11.00	879.0		20.00		1.25		1.77	
			19.00		-25.00		1.65		-2.16
1964	30.00	1033.0		-5.00		2.90		-0.39	
			14.00		-20.00		1.27		-2.22
1965	44.00	1055.0		-25.00		4.17		-2.60	
			-11.00		24.00		-1.34		3.61
1966	33.00	1165.0		-1.00		2.83		1.01	
			-12.00		12.00		-0.33		-0.98
1967	21.00	839.0		11.00		2.50		0.03	
			-1.00		-5.00		-0.30		0.47
1968	20.00	908.0		6.00		2.20		0.49	
			5.00				0.19		
1969	25.00	1043.0				2.40			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO OPERATING PRS. 1250-1449 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.50	43.3				1.15			
			0.60				1.02		
1963	1.10	50.5		1.90		2.18		1.78	
			2.50		-5.70		2.80		-6.57
1964	3.60	72.3		-3.80		4.98		-4.79	
			-1.30		5.70		-1.99		7.35
1965	2.30	76.9		1.90		2.99		2.56	
			0.60		-2.90		0.57		-2.33
1966	2.90	81.4		-1.00		3.56		0.23	
			-0.40		-0.40		0.80		-4.35
1967	2.50	57.3		-1.40		4.36		-4.12	
			-1.80		4.70		-3.32		9.21
1968	0.70	66.9		3.30		1.05		5.09	
			1.50				1.77		
1969	2.20	78.0				2.82			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	2.00	877.0				0.23			
			1.00				0.11		
1963	3.00	879.0		2.00		0.34		0.13	
			3.00		-7.00		0.24		-0.57
1964	6.00	1033.0		-5.00		0.58		-0.44	
			-2.00		10.00		-0.20		0.86
1965	4.00	1055.0		5.00		0.38		0.42	
			3.00		-9.00		0.22		-0.53
1966	7.00	1165.0		-4.00		0.60		-0.11	
			-1.00		1.00		0.11		-0.50
1967	6.00	839.0		-3.00		0.72		-0.61	
			-4.00		10.00		-0.49		1.36
1968	2.00	908.0		7.00		0.22		0.75	
			3.00				0.26		
1969	5.00	1043.0				0.48			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO OPERATING PRS. 1450-1799 PSIG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.30	43.3				0.69			
			0.30				0.50		
1963	0.60	50.5		3.50		1.19		4.40	
			3.80		-9.60		4.90		-12.65
1964	4.40	72.3		-6.10		6.09		-8.25	
			-2.30		6.30		-3.35		8.88
1965	2.10	76.9		0.20		2.73		0.62	
			-2.10		2.40		-2.73		2.98
1966	0.00	81.4		2.60		0.00		3.60	
			0.50		-0.90		0.87		-1.31
1967	0.50	57.3		1.70		0.87		2.29	
			2.20		-5.60		3.16		-8.21
1968	2.70	66.9		-3.90		4.04		-5.92	
			-1.70				-2.75		
1969	1.00	78.0				1.28			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.00	877.0				0.11			
			0.00				-0.00		
1963	1.00	879.0		4.00		0.11		0.37	
			4.00		-8.00		0.37		-0.75
1964	5.00	1033.0		-4.00		0.48		-0.38	
			0.00		-1.00		-0.01		-0.08
1965	5.00	1055.0		-5.00		0.47		-0.46	
			-5.00		12.00		-0.47		1.18
1966	0.00	1165.0		7.00		0.00		0.71	
			2.00		-6.00		0.24		-0.64
1967	2.00	839.0		1.00		0.24		0.07	
			3.00		-5.00		0.31		-0.55
1968	5.00	908.0		-4.00		0.55		-0.48	
			-1.00				-0.17		
1969	4.00	1043.0				0.38			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

APPENDIX IV

TEMPERATURE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE   SAT

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	25.50	43.3				58.89			
			1.60				-5.23		
1963	27.10	50.5		5.30		53.66		-1.41	
			6.90		-11.30		-6.64		6.40
1964	34.00	72.3		-6.00		47.03		4.99	
			0.90		9.40		-1.64		-0.58
1965	34.90	76.9		3.40		45.38		4.42	
			4.30		-19.70		2.77		-7.88
1966	39.20	81.4		-16.30		48.16		-3.46	
			-12.00		37.40		-0.69		10.86
1967	27.20	57.3		21.10		47.47		7.40	
			9.10		-25.10		6.71		-15.21
1968	36.30	67.0		-4.00		54.18		-7.61	
			5.10				-1.10		
1969	41.40	78.0				53.08			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	727.00	877.0				82.90			
			-22.00				-2.69		
1963	705.00	879.0		113.00		80.20		-0.46	
			91.00		-212.00		-3.15		1.24
1964	796.00	1033.0		-99.00		77.06		0.78	
			-8.00		209.00		-2.37		3.29
1965	788.00	1055.0		110.00		74.69		4.07	
			102.00		-496.00		1.70		-9.94
1966	890.00	1165.0		-386.00		76.39		-5.87	
			-284.00		775.00		-4.17		16.11
1967	606.00	839.0		389.00		72.23		10.24	
			105.00		-398.00		6.08		-17.25
1968	711.00	908.0		-9.00		78.30		-7.01	
			96.00				-0.93		
1969	807.00	1043.0				77.37			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E C I T Y E N E R G Y D I V.
Z U R N I N D U S T R I E S. I N C.

INDUSTRIAL WATERTUBE BOILERS. DISTRIBUTION AS TO STEAM TEMPERATURE 300-424 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.30	43.3				0.69			
			0.40				0.69		
1963	0.70	50.5		-0.70		1.39		-1.53	
			-0.30		1.00		-0.83		2.33
1964	0.40	72.3		0.30		0.55		0.80	
			0.00		0.70		-0.03		0.43
1965	0.40	76.9		1.00		0.52		1.23	
			1.00		-1.80		1.20		-1.36
1966	1.40	81.4		-0.80		1.72		-0.13	
			0.20		0.30		1.07		-1.80
1967	1.60	57.3		-0.50		2.79		-1.92	
			-0.30		1.60		-0.85		3.53
1968	1.30	67.0		1.10		1.94		1.60	
			0.80				0.75		
1969	2.10	78.0				2.69			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	8.00	877.0				0.91			
			7.00				0.79		
1963	15.00	879.0		-15.00		1.71		-1.82	
			-8.00		25.00		-1.03		3.03
1964	7.00	1033.0		10.00		0.68		1.20	
			2.00		1.00		0.18		-0.24
1965	9.00	1055.0		11.00		0.85		0.86	
			13.00		-21.00		1.04		-0.60
1966	22.00	1165.0		-10.00		1.89		0.06	
			3.00		8.00		1.09		-1.26
1967	25.00	839.0		-2.00		2.98		-1.21	
			1.00		15.00		-0.12		2.30
1968	26.00	908.0		13.00		2.86		1.09	
			14.00				0.97		
1969	40.00	1043.0				3.84			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 425-474 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.90	43.3				2.08			
			-0.20				-0.69		
1963	0.70	50.5		1.50		1.39		2.07	
			1.30		-3.20		1.38		-4.14
1964	2.00	72.3		-1.70		2.77		-2.07	
			-0.40		2.40		-0.69		3.00
1965	1.60	76.9		0.70		2.08		0.94	
			0.30		-1.20		0.25		-0.56
1966	1.90	81.4		-0.50		2.33		0.38	
			-0.20		0.40		0.63		-1.89
1967	1.70	57.3		-0.10		2.97		-1.51	
			-0.30		0.80		-0.88		2.61
1968	1.40	67.0		0.70		2.09		1.10	
			0.40				0.22		
1969	1.80	78.0				2.31			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	13.00	877.0				1.48			
			-2.00				-0.23		
1963	11.00	879.0		18.00		1.25		1.59	
			16.00		-39.00		1.36		-3.48
1964	27.00	1033.0		-21.00		2.61		-1.89	
			-5.00		21.00		-0.53		1.79
1965	22.00	1055.0		0.00		2.09		-0.10	
			-5.00		3.00		-0.63		1.05
1966	17.00	1165.0		3.00		1.46		0.95	
			-2.00		-2.00		0.33		-1.53
1967	15.00	839.0		1.00		1.79		-0.57	
			-1.00		-3.00		-0.25		0.33
1968	14.00	908.0		-2.00		1.54		-0.24	
			-3.00				-0.49		
1969	11.00	1043.0				1.05			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 475-524 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.80	43.3				4.16			
			0.10				-0.39		
1963	1.90	50.5		0.20		3.76		-0.32	
			0.30		0.00		-0.72		1.51
1964	2.20	72.3		0.20		3.04		1.19	
			0.50		-0.30		0.47		-1.36
1965	2.70	76.9		-0.10		3.51		-0.17	
			0.40		-2.20		0.30		-1.84
1966	3.10	81.4		-2.30		3.81		-2.01	
			-1.90		4.30		-1.71		3.57
1967	1.20	57.3		2.00		2.09		1.56	
			0.10		-2.00		-0.15		-1.55
1968	1.30	67.0		-0.00		1.94		0.01	
			0.10				-0.15		
1969	1.40	78.0				1.79			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	23.00	877.0				2.62			
			0.00				-0.01		
1963	23.00	879.0		3.00		2.62		-0.09	
			3.00		-4.00		-0.10		0.33
1964	26.00	1033.0		-1.00		2.52		0.24	
			2.00		3.00		0.14		-0.28
1965	28.00	1055.0		2.00		2.65		-0.04	
			4.00		-23.00		0.09		-1.01
1966	32.00	1165.0		-21.00		2.75		-1.05	
			-17.00		36.00		-0.96		1.65
1967	15.00	839.0		15.00		1.79		0.60	
			-2.00		-9.00		-0.36		-0.05
1968	13.00	908.0		6.00		1.43		0.55	
			4.00				0.20		
1969	17.00	1043.0				1.63			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 525-624 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	2.40	43.3				5.54			
			0.90				0.99		
1963	3.30	50.5		-0.30		6.53		-2.13	
			0.60		-0.50		-1.14		2.69
1964	3.90	72.3		-0.80		5.39		0.56	
			-0.20		2.10		-0.58		1.11
1965	3.70	76.9		1.30		4.61		1.67	
			1.10		-2.20		1.09		0.08
1966	4.80	81.4		-0.90		5.90		1.74	
			0.20		0.80		2.63		-5.69
1967	5.00	57.3		-0.10		8.73		-3.94	
			0.10		-1.10		-1.11		2.57
1968	5.10	67.0		-1.20		7.61		-1.37	
			-1.10				-2.48		
1969	4.00	78.0				5.13			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	25.00	877.0				2.85			
			4.00				0.45		
1963	29.00	879.0		5.00		3.30		-0.07	
			9.00		-5.00		0.38		0.47
1964	38.00	1033.0		0.00		3.68		0.40	
			9.00		-17.00		0.78		-2.28
1965	47.00	1055.0		-17.00		4.45		-1.88	
			-8.00		32.00		-1.11		5.13
1966	39.00	1165.0		15.00		3.35		3.24	
			7.00		-24.00		2.14		-6.01
1967	46.00	839.0		-9.00		5.48		-2.77	
			-2.00		4.00		-0.64		2.11
1968	44.00	908.0		-5.00		4.85		-0.66	
			-7.00				-1.30		
1969	37.00	1043.0				3.55			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 625-674 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.30	43.3				3.00			
			0.00				-0.43		
1963	1.30	50.5		0.30		2.57		0.07	
			0.30		0.70		-0.36		1.85
1964	1.60	72.3		1.00		2.21		1.92	
			1.30		-1.20		1.56		-2.33
1965	2.90	76.9		-0.20		3.77		-0.42	
			1.10		-3.30		1.14		-2.85
1966	4.00	81.4		-3.50		4.91		-3.26	
			-2.40		5.30		-2.12		4.09
1967	1.60	57.3		1.80		2.79		0.82	
			-0.60		-0.20		-1.30		1.55
1968	1.00	67.0		1.60		1.49		2.37	
			1.00				1.07		
1969	2.00	78.0				2.56			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	15.00	877.0				1.71			
			-5.00				-0.57		
1963	10.00	879.0		11.00		1.14		0.98	
			6.00		1.00		0.41		0.28
1964	16.00	1033.0		12.00		1.55		1.26	
			18.00		-30.00		1.67		-3.24
1965	34.00	1055.0		-18.00		3.22		-1.98	
			0.00		-1.00		-0.30		1.15
1966	34.00	1165.0		-19.00		2.92		-0.83	
			-19.00		33.00		-1.13		1.27
1967	15.00	839.0		14.00		1.79		0.44	
			-5.00		-6.00		-0.69		0.39
1968	10.00	908.0		8.00		1.10		0.83	
			3.00				0.15		
1969	13.00	1043.0				1.25			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 675-724 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.10	43.3				2.54			
			1.20				2.01		
1963	2.30	50.5		-1.50		4.55		-3.80	
			-0.30		2.10		-1.79		5.82
1964	2.00	72.3		0.60		2.77		2.01	
			0.30		-1.70		0.22		-3.39
1965	2.30	76.9		-1.10		2.99		-1.37	
			-0.80		1.50		-1.15		2.60
1966	1.50	81.4		0.40		1.84		1.23	
			-0.40		0.40		0.08		-0.98
1967	1.10	57.3		0.80		1.92		0.24	
			0.40		1.30		0.32		2.33
1968	1.50	67.0		2.10		2.24		2.57	
			2.50				2.89		
1969	4.00	78.0				5.13			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	15.00	877.0				1.71			
			-4.00				-0.46		
1963	11.00	879.0		5.00		1.25		0.37	
			1.00		-3.00		-0.09		-0.02
1964	12.00	1033.0		2.00		1.16		0.35	
			3.00		-4.00		0.26		-0.66
1965	15.00	1055.0		-2.00		1.42		-0.31	
			1.00		-8.00		-0.05		-0.18
1966	16.00	1165.0		-10.00		1.37		-0.49	
			-9.00		21.00		-0.54		1.19
1967	7.00	839.0		11.00		0.83		0.70	
			2.00		0.00		0.16		0.27
1968	9.00	908.0		11.00		0.99		0.96	
			13.00				1.12		
1969	22.00	1043.0				2.11			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 725-774 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	5.80	43.3				13.39			
			1.80				1.65		
1963	7.60	50.5		0.70		15.05		-2.73	
			2.50		-2.90		-1.08		3.37
1964	10.10	72.3		-2.20		13.97		0.63	
			0.30		2.80		-0.45		0.17
1965	10.40	76.9		0.60		13.52		0.80	
			0.90		-5.10		0.36		-1.61
1966	11.30	81.4		-4.50		13.88		-0.80	
			-3.60		9.20		-0.44		0.94
1967	7.70	57.3		4.70		13.44		0.14	
			1.10		-2.30		-0.30		2.60
1968	8.80	67.0		2.40		13.13		2.94	
			3.50				2.63		
1969	12.30	78.0				15.77			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	34.00	877.0				3.88			
			17.00				1.93		
1963	51.00	879.0		2.00		5.80		-0.95	
			19.00		-35.00		0.97		-1.49
1964	70.00	1033.0		-33.00		6.78		-2.44	
			-14.00		57.00		-1.47		4.27
1965	56.00	1055.0		24.00		5.31		1.83	
			10.00		-60.00		0.36		-3.08
1966	66.00	1165.0		-36.00		5.67		-1.25	
			-26.00		75.00		-0.90		3.22
1967	40.00	839.0		39.00		4.77		1.97	
			13.00		-40.00		1.07		-2.64
1968	53.00	908.0		-1.00		5.84		-0.67	
			12.00				0.40		
1969	65.00	1043.0				6.23			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 775-824 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.40	43.3				0.92			
			0.50				0.86		
1963	0.90	50.5		-0.70		1.78		-1.67	
			-0.20		3.40		-0.81		5.68
1964	0.70	72.3		2.70		0.97		4.01	
			2.50		-4.40		3.19		-6.45
1965	3.20	76.9		-1.70		4.16		-2.44	
			0.80		-2.90		0.75		-2.88
1966	4.00	81.4		-4.60		4.91		-5.32	
			-3.80		9.20		-4.56		11.03
1967	0.20	57.3		4.60		0.35		5.71	
			0.80		-5.20		1.14		-6.81
1968	1.00	67.0		-0.60		1.49		-1.10	
			0.20				0.05		
1969	1.20	76.0				1.54			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.00	877.0				0.11			
			6.00				0.68		
1963	7.00	879.0		-11.00		0.80		-1.29	
			-5.00		24.00		-0.60		2.64
1964	2.00	1033.0		13.00		0.19		1.36	
			8.00		-11.00		0.75		-1.34
1965	10.00	1055.0		2.00		0.95		0.01	
			10.00		-31.00		0.77		-2.38
1966	20.00	1165.0		-29.00		1.72		-2.37	
			-19.00		52.00		-1.60		4.40
1967	1.00	839.0		23.00		0.12		2.03	
			4.00		-27.00		0.43		-2.53
1968	5.00	908.0		-4.00		0.55		-0.50	
			0.00				-0.07		
1969	5.00	1043.0				0.48			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 825-874 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.70	43.3				3.93			
			0.40				0.23		
1963	2.10	50.5		2.80		4.16		2.94	
			3.20		-2.80		3.17		-2.39
1964	5.30	72.3		0.00		7.33		0.55	
			3.20		-6.10		3.72		-8.45
1965	8.50	76.9		-6.10		11.05		-7.90	
			-2.90		6.50		-4.17		10.60
1966	5.60	81.4		0.40		6.88		2.70	
			-2.50		5.00		-1.47		2.31
1967	3.10	57.3		5.40		5.41		5.01	
			2.90		-10.90		3.55		-13.16
1968	6.00	67.0		-5.50		8.96		-8.14	
			-2.60				-4.60		
1969	3.40	78.0				4.36			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	8.00	877.0				0.91			
			1.00				0.11		
1963	9.00	879.0		10.00		1.02		0.80	
			11.00		-9.00		0.91		-0.62
1964	20.00	1033.0		1.00		1.94		0.18	
			12.00		-28.00		1.10		-2.86
1965	32.00	1055.0		-27.00		3.03		-2.67	
			-15.00		38.00		-1.57		4.34
1966	17.00	1165.0		11.00		1.46		1.66	
			-4.00		-7.00		0.09		-1.87
1967	13.00	839.0		4.00		1.55		-0.21	
			0.00		-1.00		-0.12		0.43
1968	13.00	908.0		3.00		1.43		0.22	
			3.00				0.10		
1969	16.00	1043.0				1.53			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 875-924 -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.40	43.3				3.23			
			0.30				0.13		
1963	1.70	50.5		1.80		3.37		1.76	
			2.10		-4.50		1.89		-4.74
1964	3.80	72.3		-2.70		5.26		-2.98	
			-0.60		3.00		-1.09		3.48
1965	3.20	76.9		0.30		4.16		0.50	
			-0.30		1.20		-0.60		3.70
1966	2.90	81.4		1.50		3.56		4.19	
			1.20		-5.60		3.59		-13.15
1967	4.10	57.3		-4.10		7.16		-8.96	
			-2.90		7.80		-5.36		15.09
1968	1.20	67.0		3.70		1.79		6.14	
			0.80				0.77		
1969	2.00	78.0				2.56			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	5.00	877.0				0.57			
			1.00				0.11		
1963	6.00	879.0		5.00		0.68		0.37	
			6.00		-13.00		0.48		-1.06
1964	12.00	1033.0		-8.00		1.16		-0.69	
			-2.00		9.00		-0.21		0.73
1965	10.00	1055.0		1.00		0.95		0.04	
			-1.00		2.00		-0.18		0.68
1966	9.00	1165.0		3.00		0.77		0.71	
			2.00		-13.00		0.54		-2.23
1967	11.00	839.0		-10.00		1.31		-1.52	
			-8.00		20.00		-0.96		2.65
1968	3.00	908.0		10.00		0.33		1.13	
			2.00				0.15		
1969	5.00	1043.0				0.48			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 925-1024-- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.70	43.3				1.62			
			0.20				0.17		
1963	0.90	50.5		5.20		1.78		6.77	
			5.40		-13.80		6.93		-16.38
1964	6.30	72.3		-8.60		8.71		-11.61	
			-3.20		10.50		-4.68		14.48
1965	3.10	76.9		1.90		4.03		2.86	
			-1.30		0.00		-1.82		0.93
1966	1.80	81.4		1.90		2.21		3.80	
			0.60		-2.90		1.96		-6.98
1967	2.40	57.3		-1.00		4.19		-3.18	
			-0.40		1.80		-1.20		4.48
1968	2.00	67.0		0.80		2.99		1.30	
			0.40				0.09		
1969	2.40	78.0				3.08			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	2.00	877.0				0.23			
			0.00				-0.00		
1963	2.00	879.0		5.00		0.23		0.45	
			5.00		-13.00		0.45		-1.20
1964	7.00	1033.0		-8.00		0.68		-0.75	
			-3.00		10.00		-0.30		0.93
1965	4.00	1055.0		2.00		0.38		0.18	
			-1.00		-3.00		-0.12		-0.19
1966	3.00	1165.0		-1.00		0.26		-0.02	
			-2.00		5.00		-0.14		0.37
1967	1.00	839.0		4.00		0.12		0.35	
			2.00		-4.00		0.21		-0.41
1968	3.00	908.0		0.00		0.33		-0.06	
			2.00				0.15		
1969	5.00	1043.0				0.48			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO STEAM TEMPERATURE 1025 + -- F

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.10	43.3				0.23			
			-0.10				-0.23		
1963	0.00	50.5		0.10		0.00		0.23	
			0.00		-0.10		0.00		-0.23
1964	0.00	72.3		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1965	0.00	76.9		0.00		0.00		0.00	
			0.00		0.10		0.00		0.17
1966	0.00	81.4		0.10		0.00		0.17	
			0.10		-0.10		0.17		-0.23
1967	0.10	57.3		0.00		0.17		-0.05	
			0.10		-0.30		0.12		-0.37
1968	0.20	67.0		-0.30		0.30		-0.42	
			-0.20				-0.30		
1969	0.00	78.0				0.00			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.00	877.0				0.11			
			-1.00				-0.11		
1963	0.00	879.0		1.00		0.00		0.11	
			0.00		-1.00		0.00		-0.11
1964	0.00	1033.0		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1965	0.00	1055.0		0.00		0.00		0.00	
			0.00		4.00		0.00		0.48
1966	0.00	1165.0		4.00		0.00		0.48	
			4.00		-8.00		0.48		-0.99
1967	4.00	839.0		-4.00		0.48		-0.51	
			0.00		0.00		-0.04		0.11
1968	4.00	908.0		-4.00		0.44		-0.40	
			-4.00				-0.44		
1969	0.00	1043.0				0.00			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

APPENDIX V

BASE FUELS

ERIE CITY ENEEGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO BASE FUELS - BITUMINOUS COAL

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	7.30	43.3				16.86			
			2.70				2.94		
1963	10.00	50.5		1.40		19.80		-3.24	
			4.10		-7.50		-0.30		-0.22
1964	14.10	72.3		-6.10		19.50		-3.47	
			-2.00		1.80		-3.77		-1.37
1965	12.10	76.9		-4.30		15.73		-4.84	
			-6.30		9.10		-8.61		13.83
1966	5.80	81.4		4.80		7.13		8.99	
			-1.50		-3.50		0.38		-10.75
1967	4.30	57.3		1.30		7.50		-1.76	
			-0.20		-3.10		-1.38		-0.28
1968	4.10	67.0		-1.80		6.12		-2.04	
			-2.00				-3.43		
1969	2.10	78.0				2.69			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	110.00	877.0				12.54			
			15.00				1.68		
1963	125.00	879.0		-9.00		14.22		-3.22	
			6.00		-24.00		-1.54		1.93
1964	131.00	1033.0		-33.00		12.68		-1.28	
			-27.00		34.00		-2.82		0.95
1965	104.00	1055.0		1.00		9.86		-0.34	
			-26.00		-8.00		-3.16		2.17
1966	78.00	1165.0		-7.00		6.70		1.83	
			-33.00		27.00		-1.33		-2.34
1967	45.00	839.0		20.00		5.36		-0.51	
			-13.00		-19.00		-1.84		0.74
1968	32.00	908.0		1.00		3.52		0.23	
			-12.00				-1.61		
1969	20.00	1043.0				1.92			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO BASE FUELS - OIL

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	26.10	43.3				60.28			
			-13.80				-35.92		
1963	12.30	50.5		17.50		24.36		33.69	
			3.70		-19.80		-2.23		-30.97
1964	16.00	72.3		-2.30		22.13		2.72	
			1.40		2.10		0.50		-3.00
1965	17.40	76.9		-0.20		22.63		-0.27	
			1.20		-6.20		0.22		-2.90
1966	16.60	81.4		-8.40		22.85		-3.18	
			-7.20		19.80		-2.95		9.52
1967	11.40	57.3		11.40		19.90		6.34	
			4.20		-17.90		3.39		-15.96
1968	15.60	67.0		-6.50		23.28		-9.62	
			-2.30				-6.23		
1969	13.30	78.0				17.05			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	592.00	877.0				67.50			
			-329.00				-37.58		
1963	263.00	879.0		363.00		29.92		36.41	
			34.00		-398.00		-1.17		-35.94
1964	297.00	1033.0		-35.00		28.75		0.47	
			-1.00		63.00		-0.69		-0.11
1965	296.00	1055.0		28.00		28.06		0.36	
			27.00		-189.00		-0.33		-5.23
1966	323.00	1165.0		-161.00		27.73		-4.87	
			-134.00		300.00		-5.20		8.90
1967	189.00	839.0		139.00		22.53		4.04	
			5.00		-136.00		-1.16		-4.87
1968	194.00	908.0		3.00		21.37		-0.84	
			8.00				-2.00		
1969	202.00	1043.0				19.37			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO BASE FUELS - GAS

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	24.20	43.3				55.89			
			-4.00				-15.89		
1963	20.20	50.5		13.40		40.00		16.83	
			9.40		-17.80		0.94		-13.72
1964	29.60	72.3		-4.40		40.94		3.11	
			5.00		10.00		4.05		3.37
1965	34.60	76.9		5.60		44.99		6.48	
			10.60		-26.60		10.53		-11.81
1966	45.20	81.4		-21.00		55.53		-5.33	
			-10.40		36.20		5.20		-1.50
1967	34.80	57.3		15.20		60.73		-6.83	
			4.80		-11.80		-1.63		10.64
1968	39.60	67.0		3.40		59.10		3.81	
			8.20				2.16		
1969	47.80	78.0				61.28			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	468.00	977.0				53.36			
			-59.00				-6.83		
1963	409.00	879.0		140.00		46.53		7.74	
			81.00		-150.00		0.90		-2.90
1964	490.00	1033.0		-10.00		47.43		4.84	
			71.00		58.00		5.74		-5.38
1965	561.00	1055.0		48.00		53.18		-0.55	
			119.00		-290.00		5.19		3.37
1966	680.00	1165.0		-242.00		58.37		2.83	
			-123.00		425.00		8.02		-9.28
1967	557.00	839.0		183.00		66.39		-6.46	
			60.00		-117.00		1.56		8.18
1968	617.00	908.0		66.00		67.95		1.72	
			126.00				3.29		
1969	743.00	1043.0				71.24			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO BASE FUELS - WOOD

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	0.80	43.3				1.85			
			-0.10				-0.46		
1963	0.70	50.5		0.50		1.39		0.60	
			0.40		-1.50		0.14		-1.60
1964	1.10	72.3		-1.00		1.52		-1.01	
			-0.60		1.20		-0.67		1.35
1965	0.50	76.9		0.20		0.65		0.34	
			-0.40		1.00		-0.53		1.63
1966	0.10	81.4		1.20		0.12		1.98	
			0.80		-1.50		1.45		-2.90
1967	0.90	57.3		-0.30		1.57		-0.93	
			0.50		0.70		0.52		1.27
1968	1.40	67.0		0.40		2.09		0.34	
			0.90				0.86		
1969	2.30	78.0				2.95			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	13.00	877.0				1.48			
			-6.00				-0.69		
1963	7.00	879.0		11.00		0.80		1.05	
			5.00		-15.00		0.37		-1.35
1964	12.00	1033.0		-4.00		1.16		-0.29	
			1.00		-7.00		0.07		-0.75
1965	13.00	1055.0		-11.00		1.23		-1.05	
			-10.00		23.00		-0.97		2.36
1966	3.00	1165.0		12.00		0.26		1.31	
			2.00		-6.00		0.34		-0.82
1967	5.00	839.0		6.00		0.60		0.50	
			8.00		-8.00		0.84		-0.94
1968	13.00	908.0		-2.00		1.43		-0.45	
			6.00				0.39		
1969	19.00	1043.0				1.82			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO BASE FUELS - BAGASSE

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.30	43.3				3.00			
			0.60				0.76		
1963	1.90	50.5		-0.60		3.76		-1.89	
			0.00		-1.10		-1.13		0.66
1964	1.90	72.3		-1.70		2.63		-1.23	
			-1.70		3.40		-2.37		3.59
1965	0.20	76.9		1.70		0.26		2.35	
			0.00		-1.90		-0.01		-2.58
1966	0.20	81.4		-0.20		0.25		-0.23	
			-0.20		0.70		-0.25		0.92
1967	0.00	57.3		0.50		0.00		0.69	
			0.30		0.20		0.45		0.08
1968	0.30	67.0		0.70		0.45		0.77	
			1.00				1.22		
1969	1.30	78.0				1.67			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	19.00	877.0				2.17			
			3.00				0.34		
1963	22.00	879.0		-5.00		2.50		-0.90	
			-2.00		-11.00		-0.57		-0.28
1964	20.00	1033.0		-16.00		1.94		-1.18	
			-18.00		34.00		-1.75		2.91
1965	2.00	1055.0		18.00		0.19		1.73	
			0.00		-20.00		-0.02		-1.88
1966	2.00	1165.0		-2.00		0.17		-0.15	
			-2.00		7.00		-0.17		0.66
1967	0.00	839.0		5.00		0.00		0.50	
			3.00		-5.00		0.33		-0.59
1968	3.00	908.0		0.00		0.33		-0.09	
			3.00				0.24		
1969	6.00	1043.0				0.58			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO BASE FUELS - BLACK LIQUOR

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	1.10	43.3				2.54			
			2.30				4.19		
1963	3.40	50.5		-2.20		6.73		-6.08	
			0.10		3.70		-1.69		9.77
1964	3.50	72.3		1.50		4.84		3.68	
			1.60		-3.40		1.79		-6.21
1965	5.10	76.9		-1.90		6.63		-2.53	
			-0.30		-1.90		-0.74		-1.41
1966	4.80	81.4		-3.80		5.90		-3.94	
			-4.10		9.30		-4.66		10.53
1967	0.70	57.3		5.50		1.22		6.59	
			1.40		-5.40		1.91		-7.02
1968	2.10	67.0		0.10		3.13		-0.43	
			1.50				1.48		
1969	3.60	78.0				4.62			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	8.00	877.0				0.91			
			8.00				0.91		
1963	16.00	879.0		-12.00		1.82		-1.57	
			-4.00		23.00		-0.66		2.86
1964	12.00	1033.0		11.00		1.16		1.30	
			7.00		-20.00		0.64		-2.28
1965	19.00	1055.0		-9.00		1.80		-0.98	
			-2.00		-2.00		-0.34		0.34
1966	17.00	1165.0		-11.00		1.46		-0.64	
			-13.00		29.00		-0.98		2.14
1967	4.00	839.0		18.00		0.48		1.50	
			5.00		-20.00		0.51		-1.85
1968	9.00	908.0		-2.00		0.99		-0.36	
			3.00				0.16		
1969	12.00	1043.0				1.15			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO BASE FUELS - OTHER

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	5.30	43.3				12.24			
			-3.30				-8.28		
1963	2.00	50.5		7.30		3.96		12.62	
			4.00		-10.30		4.34		-16.15
1964	6.00	72.3		-3.00		8.30		-3.53	
			1.00		1.70		0.80		1.86
1965	7.00	76.9		-1.30		9.10		-1.68	
			-0.30		-0.10		-0.87		3.04
1966	6.70	81.4		-1.40		8.23		1.37	
			-1.70		2.00		0.50		-4.77
1967	5.00	57.3		0.60		8.73		-3.40	
			-1.10		4.20		-2.91		10.23
1968	3.90	67.0		4.80		5.82		6.83	
			3.70				3.92		
1969	7.60	78.0				9.74			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1962	77.00	877.0				8.78			
			-40.00				-4.57		
1963	37.00	879.0		74.00		4.21		7.23	
			34.00		-119.00		2.66		-11.08
1964	71.00	1033.0		-45.00		6.87		-3.85	
			-11.00		58.00		-1.19		4.67
1965	60.00	1055.0		13.00		5.69		0.82	
			2.00		-38.00		-0.37		-1.13
1966	62.00	1165.0		-25.00		5.32		-0.31	
			-23.00		49.00		-0.67		0.74
1967	39.00	839.0		24.00		4.65		0.43	
			1.00		-24.00		-0.24		-0.66
1968	40.00	908.0		0.00		4.41		-0.23	
			1.00				-0.47		
1969	41.00	1043.0				3.93			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

APPENDIX VI

ALTERNATIVE FUELS

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S, I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO ALT. FUELS - BITUMINOUS COAL

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	0.40	50.5				0.79			
			0.00				-0.24		
1964	0.40	72.3		-0.20		0.55		-0.05	
			-0.20		0.50		-0.29		0.46
1965	0.20	76.9		0.30		0.26		0.40	
			0.10		-0.60		0.11		-0.70
1966	0.30	81.4		-0.30		0.37		-0.30	
			-0.20		0.60		-0.19		0.62
1967	0.10	57.3		0.30		0.17		0.32	
			0.10		-0.50		0.12		-0.61
1968	0.20	67.0		-0.20		0.30		-0.29	
			-0.10				-0.17		
1969	0.10	78.0				0.13			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	6.00	879.0				0.68			
			-4.00				-0.49		
1964	2.00	1033.0		7.00		0.19		0.77	
			3.00		-13.00		0.28		-1.35
1965	5.00	1055.0		-6.00		0.47		-0.58	
			-3.00		9.00		-0.30		0.95
1966	2.00	1165.0		3.00		0.17		0.37	
			0.00		-3.00		0.07		-0.45
1967	2.00	839.0		0.00		0.24		-0.08	
			0.00		1.00		-0.02		0.17
1968	2.00	908.0		1.00		0.22		0.09	
			1.00				0.07		
1969	3.00	1043.0				0.29			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO ALT. FUELS - OIL

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	17.40	50.5				34.46			
			3.30				-5.82		
1964	20.70	72.3		3.30		28.63		12.69	
			6.60		-10.70		6.87		-22.51
1965	27.30	76.9		-7.40		35.50		-9.82	
			-0.80		6.00		-2.95		22.61
1966	26.50	81.4		-1.40		32.56		12.80	
			-2.20		8.90		9.85		-20.88
1967	24.30	57.3		7.50		42.41		-8.08	
			5.30		-2.40		1.77		13.41
1968	29.60	67.0		5.10		44.18		5.33	
			10.40				7.10		
1969	40.00	78.0				51.28			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	321.00	879.0				36.52			
			45.00				-1.09		
1964	366.00	1033.0		28.00		35.43		7.27	
			73.00		-41.00		6.18		-12.23
1965	439.00	1055.0		-13.00		41.61		-4.96	
			60.00		-115.00		1.22		12.28
1966	499.00	1165.0		-128.00		42.83		7.32	
			-68.00		261.00		8.54		-12.60
1967	431.00	839.0		133.00		51.37		-5.28	
			65.00		-100.00		3.25		4.35
1968	496.00	908.0		33.00		54.63		-0.93	
			96.00				2.33		
1969	594.00	1043.0				56.95			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO ALT. FUELS - GAS

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	8.10	50.5				16.04			
			1.30				-3.04		
1964	9.40	72.3		1.90		13.00		6.42	
			3.20		-5.60		3.38		-11.33
1965	12.60	76.9		-3.70		16.38		-4.90	
			-0.50		-3.40		-1.52		-0.59
1966	12.10	81.4		-7.10		14.86		-5.49	
			-7.60		15.10		-7.01		11.96
1967	4.50	57.3		8.00		7.85		6.47	
			0.40		-4.70		-0.54		-2.22
1968	4.90	67.0		3.30		7.31		4.25	
			3.70				3.71		
1969	8.60	78.0				11.03			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	101.00	879.0				11.49			
			12.00				-0.55		
1964	113.00	1033.0		-3.00		10.94		1.18	
			9.00		-7.00		0.62		-2.98
1965	122.00	1055.0		-10.00		11.56		-1.80	
			-1.00		-61.00		-1.18		-1.57
1966	121.00	1165.0		-71.00		10.39		-3.37	
			-72.00		144.00		-4.55		7.58
1967	49.00	839.0		73.00		5.84		4.21	
			1.00		-60.00		-0.33		-3.25
1968	50.00	908.0		13.00		5.51		0.96	
			14.00				0.63		
1969	64.00	1043.0				6.14			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO ALT. FUELS - WOOD

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	0.60	50.5				1.19			
			-0.40				-0.91		
1964	0.20.	72.3		1.90		0.28		2.85	
			1.50		-3.40		1.93		-4.90
1965	1.70	76.9		-1.50		2.21		-2.06	
			0.00		0.00		-0.12		0.44
1966	1.70	81.4		-1.50		2.09		-1.62	
			-1.50		3.10		-1.74		3.46
1967	0.20	57.3		1.60		0.35		1.84	
			0.10		-1.90		0.10		-2.26
1968	0.30	67.0		-0.30		0.45		-0.42	
			-0.20				-0.32		
1969	0.10	78.0				0.13			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	4.00	879.0				0.46			
			2.00				0.13		
1964	6.00	1033.0		-1.00		0.58		-0.04	
			1.00		-3.00		0.08		-0.36
1965	7.00	1055.0		-4.00		0.66		-0.40	
			-3.00		5.00		-0.32		0.62
1966	4.00	1165.0		1.00		0.34		0.22	
			-2.00		1.00		-0.10		-0.13
1967	2.00	839.0		2.00		0.24		0.09	
			0.00		-2.00		-0.02		-0.10
1968	2.00	908.0		0.00		0.22		-0.01	
			0.00				-0.03		
1969	2.00	1043.0				0.19			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO ALT. FUELS - BAGASSE

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	0.10	50.5				0.20			
			-0.10				-0.20		
1964	0.00	72.3		0.10		0.00		0.20	
			0.00		-0.10		0.00		-0.20
1965	0.00	76.9		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1966	0.00	81.4		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1967	0.00	57.3		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1968	0.00	67.0		0.00		0.00		0.00	
			0.00				0.00		
1969	0.00	78.0				0.00			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	2.00	879.0				0.23			
			-2.00				-0.23		
1964	0.00	1033.0		2.00		0.00		0.23	
			0.00		-2.00		0.00		-0.23
1965	0.00	1055.0		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1966	0.00	1165.0		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1967	0.00	839.0		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1968	0.00	908.0		0.00		0.00		0.00	
			0.00				0.00		
1969	0.00	1043.0				0.00			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO ALT. FUELS - BLACK LIQUOR

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	0.00	50.5				0.00			
			0.00				0.00		
1964	0.00	72.3		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1965	0.00	76.9		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1966	0.00	81.4		0.00		0.00		0.00	
			0.00		0.10		0.00		0.15
1967	0.00	57.3		0.10		0.00		0.15	
			0.10		-0.30		0.15		-0.45
1968	0.10	67.0		-0.20		0.15		-0.30	
			-0.10				-0.15		
1969	0.00	78.0				0.00			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	0.00	879.0				0.00			
			0.00				0.00		
1964	0.00	1033.0		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1965	0.00	1055.0		0.00		0.00		0.00	
			0.00		0.00		0.00		0.00
1966	0.00	1165.0		0.00		0.00		0.00	
			0.00		1.00		0.00		0.11
1967	0.00	839.0		1.00		0.00		0.11	
			1.00		-3.00		0.11		-0.33
1968	1.00	908.0		-2.00		0.11		-0.22	
			-1.00				-0.11		
1969	0.00	1043.0				0.00			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO ALT. FUELS - OTHER

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	2.20	50.5				4.36			
			1.80				1.18		
1964	4.00	72.3		-1.30		5.53		-0.86	
			0.50		4.70		0.32		5.01
1965	4.50	76.9		3.40		5.85		4.15	
			3.90		-13.50		4.47		-15.10
1966	8.40	81.4		-10.10		10.32		-10.95	
			-6.20		15.20		-6.48		15.23
1967	2.20	57.3		5.10		3.84		4.28	
			-1.10		-4.20		-2.20		-2.57
1968	1.10	67.0		0.90		1.64		1.71	
			-0.20				-0.49		
1969	0.90	78.0				1.15			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1963	19.00	879.0				2.16			
			14.00				1.03		
1964	33.00	1033.0		-25.00		3.19		-2.14	
			-11.00		62.00		-1.11		5.29
1965	22.00	1055.0		37.00		2.09		3.14	
			26.00		-96.00		2.03		-7.51
1966	48.00	1165.0		-59.00		4.12		-4.37	
			-33.00		87.00		-2.33		6.01
1967	15.00	839.0		28.00		1.79		1.65	
			-5.00		-24.00		-0.69		-1.20
1968	10.00	908.0		4.00		1.10		0.45	
			-1.00				-0.24		
1969	9.00	1043.0				0.86			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

APPENDIX VII

FIRING METHOD

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO FIRING METHOD - PULVERIZED

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	4.34	40.6				10.69			
			-2.34				-6.07		
1962	2.00	43.3		3.54		4.62		7.79	
			1.20		-3.04		1.72		-9.07
1963	3.20	50.5		0.50		6.34		-1.28	
			1.70		-2.80		0.44		-0.35
1964	4.90	72.3		-2.30		6.78		-1.63	
			-0.60		-0.40		-1.19		-1.55
1965	4.30	76.9		-2.70		5.59		-3.18	
			-3.30		5.30		-4.36		6.84
1966	1.00	81.4		2.60		1.23		3.66	
			-0.70		-0.30		-0.70		-0.64
1967	0.30	57.3		2.30		0.52		3.02	
			1.60		-5.10		2.32		-7.28
1968	1.90	66.9		-2.80		2.84		-4.26	
			-1.20				-1.94		
1969	0.70	78.0				0.90			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	15.00	787.0				1.91			
			-7.00				-0.99		
1962	8.00	877.0		14.00		0.91		1.79	
			7.00		-21.00		0.79		-2.84
1963	15.00	879.0		-7.00		1.71		-1.05	
			0.00		5.00		-0.25		1.08
1964	15.00	1033.0		-2.00		1.45		0.03	
			-2.00		-5.00		-0.22		-0.70
1965	13.00	1055.0		-7.00		1.23		-0.67	
			-9.00		14.00		-0.69		1.45
1966	4.00	1165.0		7.00		0.34		0.78	
			-2.00		-3.00		-0.10		-0.48
1967	2.00	839.0		4.00		0.24		0.31	
			2.00		-8.00		0.20		-0.76
1968	4.00	908.0		-4.00		0.44		-0.45	
			-2.00				-0.25		
1969	2.00	1043.0				0.19			

NOTE** DY1 = FIRST DIFFERENCE DY2 = SECOND DIFFERENCE DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO FIRING METHOD - SPREADER

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	4.99	40.6				12.29			
			-0.49				-1.90		
1962	4.50	43.3		2.29		10.39		3.98	
			1.80		-1.59		2.08		-6.37
1963	6.30	50.5		0.70		12.48		-2.39	
			2.50		-5.10		-0.30		-0.51
1964	8.80	72.3		-4.40		12.17		-2.90	
			-1.90		3.20		-3.20		1.79
1965	6.90	76.9		-1.20		8.97		-1.11	
			-3.10		2.50		-4.30		4.23
1966	3.80	81.4		1.30		4.67		3.13	
			-1.80		0.20		-1.18		-2.90
1967	2.00	57.3		1.50		3.49		0.23	
			-0.30		-0.50		-0.95		1.26
1968	1.70	66.9		1.00		2.54		1.49	
			0.70				0.54		
1969	2.40	78.0				3.08			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	47.00	787.0				5.97			
			20.00				1.67		
1962	67.00	877.0		-22.00		7.64		-1.91	
			-2.00		50.00		-0.24		3.57
1963	65.00	879.0		28.00		7.39		1.66	
			26.00		-85.00		1.41		-6.20
1964	91.00	1033.0		-57.00		8.61		-4.54	
			-31.00		73.00		-3.12		5.83
1965	60.00	1055.0		16.00		5.69		1.30	
			-15.00		-25.00		-1.82		-0.83
1966	45.00	1165.0		-9.00		3.86		0.46	
			-24.00		27.00		-1.36		0.04
1967	21.00	839.0		18.00		2.50		0.51	
			-6.00		-4.00		-0.85		0.90
1968	15.00	908.0		14.00		1.65		1.40	
			8.00				0.55		
1969	23.00	1043.0				2.21			

NOTE** DY1 = FIRST DIFFERENCE DY2 = SECOND DIFFERENCE DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO FIRING METHOD - UNDERFEED

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	1.00	40.6				2.46			
			-0.40				-1.06		
1962	0.60	43.3		0.30		1.39		0.68	
			-0.10		-0.50		-0.40		-1.00
1963	0.50	50.5		-0.20		0.99		-0.32	
			-0.30		0.50		-0.71		1.01
1964	0.20	72.3		0.30		0.28		0.70	
			0.00		-0.20		-0.02		-0.57
1965	0.20	76.9		0.10		0.26		0.13	
			0.10		0.00		0.11		0.27
1966	0.30	81.4		0.10		0.37		0.40	
			0.20		-0.50		0.50		-1.32
1967	0.50	57.3		-0.40		0.87		-0.93	
			-0.20		0.32		-0.42		0.93
1968	0.30	66.9		-0.08		0.45		0.00	
			-0.28				-0.42		
1969	0.02	78.0				0.03			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	37.00	787.0				4.70			
			-21.00				-2.88		
1962	16.00	877.0		25.00		1.82		3.33	
			4.00		-35.00		0.45		-4.70
1963	20.00	879.0		-10.00		2.28		-1.37	
			-6.00		13.00		-0.92		1.98
1964	14.00	1033.0		3.00		1.36		0.61	
			-3.00		1.00		-0.31		-0.31
1965	11.00	1055.0		4.00		1.04		0.30	
			1.00		-14.00		-0.01		-0.96
1966	12.00	1165.0		-10.00		1.03		-0.66	
			-9.00		23.00		-0.67		1.75
1967	3.00	839.0		13.00		0.36		1.09	
			4.00		-23.00		0.41		-2.17
1969	7.00	908.0		-10.00		0.77		-1.09	
			-6.00				-0.68		
1969	1.00	1043.0				0.10			

NOTE** DY1 = FIRST DIFFERENCE DY2 = SECOND DIFFERENCE DY3 = THIRD DIFFERENCE

B-114

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO FIRING METHOD - OVERFEED

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	1.76	40.6				4.33			
			-0.36				-1.10		
1962	1.40	43.3		1.06		3.23		2.03	
			0.70		-3.06		0.93		-6.00
1963	2.10	50.5		-2.00		4.16		-3.98	
			-1.30		3.70		-3.05		7.48
1964	0.80	72.3		1.70		1.11		3.51	
			0.40		-2.70		0.45		-4.78
1965	1.20	76.9		-1.00		1.56		-1.26	
			-0.60		1.50		-0.82		2.24
1966	0.60	81.4		0.50		0.74		0.96	
			-0.10		-0.80		0.14		-1.82
1967	0.50	57.3		-0.30		0.87		-0.86	
			-0.40		1.00		-0.72		1.95
1968	0.10	66.9		0.70		0.15		1.09	
			0.30				0.36		
1969	0.40	78.0				0.51			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	47.00	787.0				5.97			
			-12.00				-1.98		
1962	35.00	877.0		12.00		3.99		1.97	
			0.00		-40.00		-0.01		-5.27
1963	35.00	879.0		-28.00		3.98		-3.30	
			-28.00		73.00		-3.30		8.20
1964	7.00	1033.0		45.00		0.68		4.90	
			17.00		-77.00		1.60		-8.00
1965	24.00	1055.0		-32.00		2.27		-3.10	
			-15.00		39.00		-1.50		3.95
1966	9.00	1165.0		7.00		0.77		0.85	
			-8.00		3.00		-0.65		0.02
1967	1.00	839.0		10.00		0.12		0.86	
			2.00		-8.00		0.21		-0.74
1968	3.00	908.0		2.00		0.33		0.13	
			4.00				0.34		
1969	7.00	1043.0				0.67			

NOTE** DY1 = FIRST DIFFERENCE DY2 = SECOND DIFFERENCE DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO FIRING METHOD - OTHER

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	0.90	40.6				2.22			
			1.60				3.56		
1962	2.50	43.3		-2.40		5.77		-5.96	
			-0.80		3.10		-2.41		7.22
1963	1.70	50.5		0.70		3.37		1.25	
			-0.10		-1.20		-1.15		-1.01
1964	1.60	72.3		-0.50		2.21		0.24	
			-0.60		1.60		-0.91		1.21
1965	1.00	76.9		1.10		1.30		1.45	
			0.50		-0.60		0.54		0.52
1966	1.50	81.4		0.50		1.84		1.98	
			1.00		-3.00		2.52		-7.37
1967	2.50	57.3		-2.50		4.36		-5.39	
			-1.50		4.10		-2.87		8.17
1968	1.00	66.9		1.60		1.49		2.78	
			0.10				-0.08		
1969	1.10	78.0				1.41			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1961	25.00	787.0				3.18			
			29.00				2.98		
1962	54.00	877.0		-59.00		6.16		-6.41	
			-30.00		83.00		-3.43		8.85
1963	24.00	879.0		24.00		2.73		2.44	
			-6.00		-29.00		-0.99		-2.53
1964	18.00	1033.0		-5.00		1.74		-0.09	
			-11.00		25.00		-1.08		1.88
1965	7.00	1055.0		20.00		0.66		1.79	
			9.00		-23.00		0.71		-1.25
1966	16.00	1165.0		-3.00		1.37		0.54	
			6.00		-11.00		1.25		-2.87
1967	22.00	839.0		-14.00		2.62		-2.33	
			-8.00		20.00		-1.08		3.02
1968	14.00	908.0		6.00		1.54		0.69	
			-2.00				-0.39		
1969	12.00	1043.0				1.15			

NOTE** DY1 = FIRST DIFFERENCE DY2 = SECOND DIFFERENCE DY3 = THIRD DIFFERENCE

APPENDIX VIII

MARKETS

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - NON-MFG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	15.40	72.3				21.30			
			0.20				-1.01		
1965	15.60	76.9		1.20		20.29		1.61	
			1.40		-5.70		0.60		1.16
1966	17.00	81.4		-4.50		20.88		2.78	
			-3.10		7.40		3.37		-9.96
1967	13.90	57.3		2.90		24.26		-7.18	
			-0.20		2.40		-3.81		14.65
1968	13.70	67.0		5.30		20.45		7.47	
			5.10				3.65		
1969	18.80	78.0				24.10			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	382.00	1033.0				36.98			
			-27.00				-3.33		
1965	355.00	1055.0		78.00		33.65		4.53	
			51.00		-221.00		1.20		-3.16
1966	406.00	1165.0		-143.00		34.85		1.38	
			-92.00		227.00		2.58		-7.68
1967	314.00	839.0		84.00		37.43		-6.30	
			-8.00		22.00		-3.73		15.06
1968	306.00	908.0		106.00		33.70		8.76	
			98.00				5.03		
1969	404.00	1043.0				38.73			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - CHEMICAL

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	15.20	72.3				21.02			
			0.60				-0.48		
1965	15.80	76.9		-1.40		20.55		-1.64	
			-0.80		-1.10		-2.12		5.75
1966	15.00	81.4		-2.50		18.43		4.11	
			-3.30		4.50		1.99		-11.00
1967	11.70	57.3		2.00		20.42		-6.89	
			-1.30		3.90		-4.90		15.49
1968	10.40	67.0		5.90		15.52		8.60	
			4.60				3.71		
1969	15.00	78.0				19.23			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	132.00	1033.0				12.78			
			21.00				1.72		
1965	153.00	1055.0		-20.00		14.50		-3.01	
			1.00		-22.00		-1.28		4.54
1966	154.00	1165.0		-42.00		13.22		1.53	
			-41.00		73.00		0.25		-3.91
1967	113.00	839.0		31.00		13.47		-2.37	
			-10.00		36.00		-2.12		8.50
1968	103.00	908.0		67.00		11.34		6.12	
			57.00				4.00		
1969	160.00	1043.0				15.34			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E C I T Y E N E R G Y D I V.
Z U R N I N D U S T R I E S, I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - PAPER

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	11.10	72.3				15.35			
			2.20				1.94		
1965	13.30	76.9		-3.40		17.30		-4.37	
			-1.20		0.10		-2.43		5.20
1966	12.10	81.4		-3.30		14.86		0.83	
			-4.50		9.40		-1.60		1.24
1967	7.60	57.3		6.10		13.26		2.07	
			1.60		-4.40		0.47		-0.24
1968	9.20	67.0		1.70		13.73		1.83	
			3.30				2.29		
1969	12.50	78.0				16.03			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	71.00	1033.0				6.87			
			5.00				0.33		
1965	76.00	1055.0		11.00		7.20		0.36	
			16.00		-63.00		0.69		-2.28
1966	92.00	1165.0		-52.00		7.90		-1.92	
			-36.00		91.00		-1.22		2.96
1967	56.00	839.0		39.00		6.67		1.05	
			3.00		-22.00		-0.16		0.21
1968	59.00	908.0		17.00		6.50		1.25	
			20.00				1.08		
1969	79.00	1043.0				7.57			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - PETROLEUM

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	6.40	72.3				8.85			
			0.70				0.38		
1965	7.10	76.9		3.00		9.23		3.65	
			3.70		-12.10		4.04		-11.53
1966	10.80	81.4		-9.10		13.27		-7.88	
			-5.40		15.30		-3.84		11.55
1967	5.40	57.3		6.20		9.42		3.67	
			0.80		-5.40		-0.17		-2.76
1968	6.20	67.0		0.80		9.25		0.92	
			1.60				0.75		
1969	7.80	78.0				10.00			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	71.00	1033.0				6.87			
			-14.00				-1.47		
1965	57.00	1055.0		35.00		5.40		2.76	
			21.00		-79.00		1.29		-4.20
1966	78.00	1165.0		-44.00		6.70		-1.43	
			-23.00		67.00		-0.14		1.07
1967	55.00	839.0		23.00		6.56		-0.36	
			0.00		-13.00		-0.50		1.03
1968	55.00	908.0		10.00		6.06		0.67	
			10.00				0.17		
1969	65.00	1043.0				6.23			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - FOOD

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	7.90	72.3				10.93			
			-0.30				-1.04		
1965	7.60	76.9		0.30		9.88		0.50	
			0.00		-1.90		-0.55		1.18
1966	7.60	81.4		-1.60		9.34		1.68	
			-1.60		5.20		1.13		-1.35
1967	6.00	57.3		3.60		10.47		0.33	
			2.00		-8.20		1.47		-6.82
1968	8.00	67.0		-4.60		11.94		-6.49	
			-2.60				-5.02		
1969	5.40	78.0				6.92			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	125.00	1033.0				12.10			
			11.00				0.79		
1965	136.00	1055.0		-16.00		12.89		-2.44	
			-5.00		-6.00		-1.65		5.23
1966	131.00	1165.0		-22.00		11.24		2.80	
			-27.00		59.00		1.15		-3.79
1967	104.00	839.0		37.00		12.40		-0.99	
			10.00		-82.00		0.16		-4.15
1968	114.00	908.0		-45.00		12.56		-5.14	
			-35.00				-4.98		
1969	79.00	1043.0				7.57			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - METALS

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	3.40	72.3				4.70			
			2.50				2.97		
1965	5.90	76.9		-2.60		7.67		-3.52	
			-0.10		-0.30		-0.55		1.82
1966	5.80	81.4		-2.90		7.13		-1.69	
			-3.00		5.60		-2.24		2.78
1967	2.80	57.3		2.70		4.89		1.08	
			-0.30		-1.50		-1.16		0.70
1968	2.50	67.0		1.20		3.73		1.78	
			0.90				0.63		
1969	3.40	78.0				4.36			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	44.00	1033.0				4.26			
			1.00				0.01		
1965	45.00	1055.0		17.00		4.27		1.14	
			18.00		-69.00		1.14		-4.23
1966	63.00	1165.0		-52.00		5.41		-3.09	
			-34.00		89.00		-1.95		5.11
1967	29.00	839.0		37.00		3.46		2.02	
			3.00		-53.00		0.07		-3.79
1968	32.00	908.0		-16.00		3.52		-1.77	
			-13.00				-1.70		
1969	19.00	1043.0				1.82			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - MISC. MFG

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	3.40	72.3				4.70			
			-0.10				-0.41		
1965	3.30	76.9		1.40		4.29		1.77	
			1.30		-4.70		1.36		-4.24
1966	4.60	81.4		-3.30		5.65		-2.47	
			-2.00		8.50		-1.11		7.71
1967	2.60	57.3		5.20		4.54		5.23	
			3.20		-9.70		4.12		-12.24
1968	5.80	67.0		-4.50		8.66		-7.01	
			-1.30				-2.89		
1969	4.50	78.0				5.77			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	61.00	1033.0				5.91			
			9.00				0.73		
1965	70.00	1055.0		2.00		6.64		-0.41	
			11.00		-49.00		0.32		-1.49
1966	81.00	1165.0		-47.00		6.95		-1.91	
			-36.00		119.00		-1.59		7.05
1967	45.00	839.0		72.00		5.36		5.15	
			36.00		-110.00		3.56		-10.05
1968	81.00	908.0		-38.00		8.92		-4.90	
			-2.00				-1.35		
1969	79.00	1043.0				7.57			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - TEXTILES

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	2.50	72.3				3.46			
			0.30				0.18		
1965	2.80	76.9		0.10		3.64		0.11	
			0.40		-1.50		0.29		-0.49
1966	3.20	81.4		-1.40		3.93		-0.38	
			-1.00		2.70		-0.09		0.37
1967	2.20	57.3		1.30		3.84		-0.02	
			0.30		-0.50		-0.11		1.01
1968	2.50	67.0		0.80		3.73		0.99	
			1.10				0.88		
1969	3.60	78.0				4.62			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	46.00	1033.0				4.45			
			28.00				2.56		
1965	74.00	1055.0		-30.00		7.01		-3.40	
			-2.00		5.00		-0.83		3.41
1966	72.00	1165.0		-25.00		6.18		0.02	
			-27.00		56.00		-0.82		0.83
1967	45.00	839.0		31.00		5.36		0.85	
			4.00		-20.00		0.03		-0.14
1968	49.00	908.0		11.00		5.40		0.71	
			15.00				0.74		
1969	64.00	1043.0				6.14			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E C I T Y E N E R G Y D I V.
Z U R N I N D U S T R I E S, I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - TRANSPORTATION

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	3.80	72.3				5.26			
			-0.60				-1.09		
1965	3.20	76.9		-0.50		4.16		-0.49	
			-1.10		1.10		-1.58		2.28
1966	2.10	81.4		0.60		2.58		1.79	
			-0.50		2.40		0.21		1.32
1967	1.60	57.3		3.00		2.79		3.11	
			2.50		-6.80		3.33		-8.97
1968	4.10	67.0		-3.80		6.12		-5.86	
			-1.30				-2.53		
1969	2.80	78.0				3.59			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	45.00	1033.0				4.36			
			-4.00				-0.47		
1965	41.00	1055.0		-6.00		3.89		-0.76	
			-10.00		10.00		-1.23		2.30
1966	31.00	1165.0		4.00		2.66		1.54	
			-6.00		14.00		0.32		-0.77
1967	25.00	839.0		18.00		2.98		0.78	
			12.00		-40.00		1.10		-3.36
1968	37.00	908.0		-22.00		4.07		-2.58	
			-10.00				-1.49		
1969	27.00	1043.0				2.59			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - WOOD

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	2.20	72.3				3.04			
			-0.90				-1.35		
1965	1.30	76.9		1.20		1.69		1.63	
			0.30		-1.40		0.26		-0.90
1966	1.60	81.4		-0.20		1.97		0.73	
			0.10		0.50		1.00		-1.56
1967	1.70	57.3		0.30		2.97		-0.83	
			0.40		-0.30		0.17		0.74
1968	2.10	67.0		0.00		3.13		-0.10	
			0.40				0.07		
1969	2.50	78.0				3.21			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	26.00	1033.0				2.52			
			0.00				-0.05		
1965	26.00	1055.0		1.00		2.46		-0.09	
			1.00		-11.00		-0.15		0.07
1966	27.00	1165.0		-10.00		2.32		-0.03	
			-9.00		31.00		-0.17		1.36
1967	18.00	839.0		21.00		2.15		1.33	
			12.00		-26.00		1.16		-2.25
1968	30.00	908.0		-5.00		3.30		-0.92	
			7.00				0.24		
1969	37.00	1043.0				3.55			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

E R I E   C I T Y   E N E R G Y   D I V.
Z U R N   I N D U S T R I E S,   I N C.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION AS TO MARKETS - RUBBER

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	0.90	72.3				1.24			
			-0.30				-0.46		
1965	0.60	76.9		1.30		0.78		1.65	
			1.00		-2.50		1.19		-2.36
1966	1.60	81.4		-1.20		1.97		-0.71	
			-0.20		2.00		0.48		0.77
1967	1.40	57.3		0.80		2.44		0.06	
			0.60		-1.90		0.54		-1.67
1968	2.00	67.0		-1.10		2.99		-1.60	
			-0.50				-1.06		
1969	1.50	78.0				1.92			

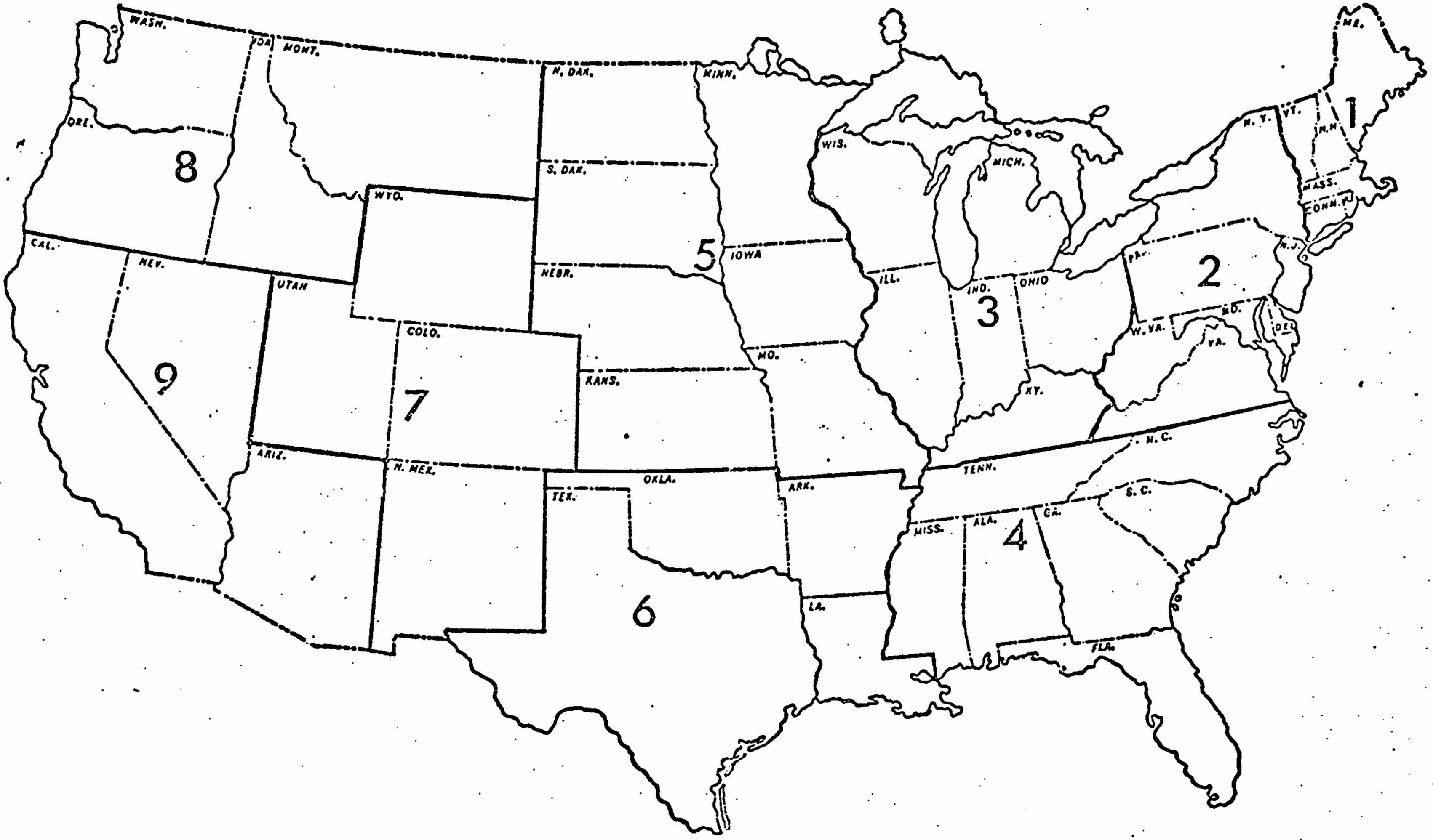
YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1964	25.00	1033.0				2.42			
			-9.00				-0.90		
1965	16.00	1055.0		20.00		1.52		1.70	
			11.00		-28.00		0.80		-1.25
1966	27.00	1165.0		-8.00		2.32		0.46	
			3.00		11.00		1.26		-1.33
1967	30.00	839.0		3.00		3.58		-0.87	
			6.00		-15.00		0.39		-0.61
1968	36.00	908.0		-12.00		3.96		-1.48	
			-6.00				-1.09		
1969	30.00	1043.0				2.88			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

APPENDIX IX

REGION

DOMESTIC SALES OF STATIONARY BOILERS
DISTRIBUTION BY REGION



ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION BY REGION - 1

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	2.10	76.9				2.73			
			1.60				1.81		
1966	3.70	81.4		-3.30		4.55		-2.87	
			-1.70		5.20		-1.06		3.72
1967	2.00	57.3		1.90		3.49		0.65	
			0.20		-0.50		-0.21		0.95
1968	2.20	67.0		1.40		3.28		1.60	
			1.60				1.59		
1969	3.80	78.0				4.67			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	46.00	1055.0				4.36			
			21.00				1.39		
1966	67.00	1165.0		-52.00		5.75		-2.85	
			-31.00		64.00		-1.46		4.10
1967	36.00	839.0		32.00		4.29		1.24	
			1.00		-2.00		-0.22		1.42
1968	37.00	908.0		30.00		4.07		2.66	
			31.00				2.44		
1969	68.00	1043.0				6.52			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION BY REGION - 2

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	15.30	76.9				19.90			
			-1.80				-3.31		
1966	13.50	81.4		-3.90		16.58		0.34	
			-5.70		13.40		-2.97		6.33
1967	7.80	57.3		9.50		13.61		6.67	
			3.80		-11.10		3.70		-10.00
1968	11.60	67.0		-1.60		17.31		-3.32	
			2.20				0.38		
1969	13.80	78.0				17.69			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	241.00	1055.0				22.84			
			4.00				-1.81		
1966	245.00	1165.0		-75.00		21.03		1.52	
			-71.00		169.00		-0.29		-0.27
1967	174.00	839.0		94.00		20.74		1.25	
			23.00		-79.00		0.96		-1.37
1968	197.00	908.0		15.00		21.70		-0.12	
			38.00				0.84		
1969	235.00	1043.0				22.53			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION BY REGION - 3

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	12.40	76.9				16.12			
			3.50				3.41		
1966	15.90	81.4		-8.70		19.53		-4.27	
			-5.20		18.00		-0.86		8.54
1967	10.70	57.3		9.30		18.67		4.28	
			4.10		-8.50		3.42		-4.52
1968	14.80	67.0		0.80		22.09		-0.25	
			4.90				3.17		
1969	19.70	78.0				25.26			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	216.00	1055.0				20.47			
			34.00				0.99		
1966	250.00	1165.0		-116.00		21.46		-2.42	
			-82.00		233.00		-1.44		6.19
1967	168.00	839.0		117.00		20.02		3.77	
			35.00		-103.00		2.33		-4.30
1968	203.00	908.0		14.00		22.36		-0.53	
			49.00				1.80		
1969	252.00	1043.0				24.16			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION BY REGION - 4

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	14.30	76.9				18.60			
			-3.60				-5.45		
1966	10.70	81.4		2.20		13.14		6.54	
			-1.40		3.10		3.09		-8.15
1967	9.30	57.3		5.30		16.23		0.39	
			3.90		-12.20		3.47		-10.48
1968	13.20	67.0		-6.90		19.70		-10.10	
			-3.00				-6.62		
1969	10.20	78.0				13.08			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	157.00	1055.0				14.88			
			17.00				0.05		
1966	174.00	1165.0		-62.00		14.94		0.39	
			-45.00		138.00		0.44		1.42
1967	129.00	839.0		76.00		15.38		1.81	
			31.00		-130.00		2.25		-8.54
1968	160.00	908.0		-54.00		17.62		-6.73	
			-23.00				-4.49		
1969	137.00	1043.0				13.14			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION BY REGION - 5

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	4.80	76.9				6.24			
			-0.30				-0.71		
1966	4.50	81.4		-1.20		5.53		0.42	
			-1.50		3.80		-0.29		0.76
1967	3.00	57.3		2.60		5.24		1.18	
			1.10		-6.10		0.88		-6.00
1968	4.10	67.0		-3.50		6.12		-4.82	
			-2.40				-3.94		
1969	1.70	78.0				2.18			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	91.00	1055.0				8.63			
			0.00				-0.81		
1966	91.00	1165.0		-31.00		7.81		0.15	
			-31.00		59.00		-0.66		-0.37
1967	60.00	839.0		28.00		7.15		-0.21	
			-3.00		-20.00		-0.87		0.75
1968	57.00	908.0		8.00		6.28		0.54	
			5.00				-0.33		
1969	62.00	1043.0				5.94			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION BY REGION - 6

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	15.20	76.9				19.77			
			1.30				0.50		
1966	16.50	81.4		-6.40		20.27		-0.88	
			-5.10		11.30		-0.37		-1.92
1967	11.40	57.3		4.90		19.90		-2.80	
			-0.20		-1.20		-3.18		6.11
1968	11.20	67.0		3.70		16.72		5.31	
			3.50				2.13		
1969	14.70	78.0				18.85			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	107.00	1055.0				10.14			
			17.00				0.50		
1966	124.00	1165.0		-51.00		10.64		-0.42	
			-34.00		94.00		0.08		0.51
1967	90.00	839.0		43.00		10.73		0.09	
			9.00		-46.00		0.18		-1.10
1968	99.00	908.0		-3.00		10.90		-1.01	
			6.00				-0.84		
1969	105.00	1043.0				10.07			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION BY REGION - 7

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	1.60	76.9				2.08			
			-0.40				-0.61		
1966	1.20	81.4		0.10		1.47		0.70	
			-0.30		-0.10		0.10		-1.47
1967	0.90	57.3		-0.00		1.57		-0.77	
			-0.30		0.70		-0.68		1.63
1968	0.60	67.0		0.70		0.90		1.06	
			0.40				0.39		
1969	1.00	78.0				1.28			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	28.00	1055.0				2.65			
			-4.00				-0.59		
1966	24.00	1165.0		-1.00		2.06		0.80	
			-5.00		0.00		0.20		-1.84
1967	19.00	839.0		-1.00		2.26		-1.04	
			-6.00		13.00		-0.83		2.26
1968	13.00	908.0		12.00		1.43		1.22	
			6.00				0.39		
1969	19.00	1043.0				1.82			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION BY REGION - 8

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	3.10	76.9				4.03			
			-0.30				-0.59		
1966	2.80	81.4		-0.20		3.44		1.17	
			-0.50		0.50		0.57		-2.62
1967	2.30	57.3		0.30		4.01		-1.45	
			-0.20		0.30		-0.88		2.40
1968	2.10	67.0		0.60		3.13		0.95	
			0.40				0.07		
1969	2.50	78.0				3.21			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	48.00	1055.0				4.55			
			-12.00				-1.46		
1966	36.00	1165.0		3.00		3.09		1.59	
			-9.00		12.00		0.13		-1.30
1967	27.00	839.0		15.00		3.22		0.29	
			6.00		-16.00		0.42		-0.70
1968	33.00	908.0		-1.00		3.63		-0.41	
			5.00				0.01		
1969	38.00	1043.0				3.64			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

ERIE CITY ENERGY DIV.
ZURN INDUSTRIES, INC.

INDUSTRIAL WATERTUBE BOILERS, DISTRIBUTION BY REGION - 9

YEAR	CAP.	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	3.50	76.9				4.55			
			0.20				-0.01		
1966	3.70	81.4		-2.00		4.55		-1.22	
			-1.80		4.60		-1.23		3.17
1967	1.90	57.3		2.60		3.32		1.94	
			0.80		-2.10		0.71		-1.56
1968	2.70	67.0		0.50		4.03		0.38	
			1.30				1.10		
1969	4.00	78.0				5.13			

YEAR	UNITS	TOTAL	DY1	DY2	DY3	PERCENT	DY1	DY2	DY3
1965	56.00	1055.0				5.31			
			18.00				1.04		
1966	74.00	1165.0		-49.00		6.35		-2.27	
			-31.00		84.00		-1.23		3.55
1967	43.00	839.0		35.00		5.13		1.28	
			4.00		-24.00		0.05		-0.56
1968	47.00	908.0		11.00		5.18		0.72	
			15.00				0.77		
1969	62.00	1043.0				5.94			

NOTE** DY1 = FIRST DIFFERENCE
DY2 = SECOND DIFFERENCE
DY3 = THIRD DIFFERENCE

APPENDIX X

1970 ABMA DATA

DOMESTIC SALES OF STATIONARY WATERTUBE GENERATORS
STATE DISTRIBUTION

Industrial Type				Utility Type		
Units	Capacity[1]	Percent Total (Ind)[2]	STATE CODE	Units	Capacity[1]	Percent Total(Util)[2]
17	1471	2.47	Alabama	0	0	.00
2	135	.23	Alaska	0	0	.00
3	580	.98	Arizona	0	0	.00
5	405	..68	Arkansas	0	0	.00
22	1181	1.99	Cal.	8	2720	1.14
16	897	1.51	Colo.	2	3487	1.46
7	444	.75	Conn.	1	4320	1.81
3	730	1.23	Del.	2	3981	1.67
1	40	.07	D. C.	0	0	.00
12	1354	2.28	Florida	4	11732	4.92
22	2532	4.26	Ga.	11	13608	5.70
2	96	.16	Hawaii	1	963	.40
2	283	.48	Idaho	0	0	.00
79	6817	11.47	Ill.	2	4349	1.82
21	935	1.57	Indiana	4	10190	4.27
8	607	1.02	Iowa	4	706	.30
7	213	.36	Kansas	6	1820	.76
17	885	1.49	Kentucky	1	1900	1.11
17	2522	4.24	La.	4	9935	4.16
10	717	1.21	Maine	0	0	.00
29	2796	4.70	Maryland	1	4600	1.93
24	1163	1.96	Mass.	3	4450	1.87
29	1474	2.48	Mich.	1	200	.08
12	916	1.54	Minn.	1	4985	2.09
4	153	.26	Miss.	1	5312	2.23
6	[illegible]	.78	Missouri	2	8430	3.53
2	560	.94	Montana	1	2520	1.06
2	175	.29	Nebraska	0	0	.00
1	40	.07	Nevada	0	0	.00
2	65	.11	N. H.	1	2992	1.25
24	1139	1.92	N. J.	1	2850	1.19
0	0	.00	N. M.	0	0	.00
61	3627	6.10	N. Y.	32	27670	11.60
31	1732	2.91	No.Carol.	1	50	.02
2	40	.07	No. Dak.	1	3075	1.29
21	1121	1.89	Ohio	2	251	.11
5	470	.79	Okla.	2	7030	2.95
14	775	1.30	Oregon	0	0	.00
62	4451	7.49	Penna.	6	27962	11.72
3	120	.20	Rd. Isl.	0	0	.00
17	1044	1.76	So. Carol.	2	200	.08
7	212	.36	So. Dak.	0	0	.00
18	1397	2.35	Tenn.	2	80	.03
56	6656	11.20	Texas	16	43435	18.21
11	576	.97	Utah	1	3300	1.38
2	80	.13	Vermont	0	0	.00
19	1365	2.30	Virginia	1	5841	2.45
22	1305	2.20	Wash.	0	0	.00
10	842	1.42	W. Virginia	3	675	.28
25	1208	2.03	Wis.	1	3800	1.59
0	0	.00	Wyoming	2	7960	3.34
6	182	.31	Puer. Ric.	0	0	.00
3	450	.76	Vir. Isles	1	420	.18
0	0	.00	Pac. Isles	0	0	.00

Note 1: Million lbs/hr
Note 2: Domestic Only

INDUSTRIAL STATIONARY WATERTUBE BOILER MARKET

1970 DISTRIBUTION

TOTAL CAPACITY
68,359,000

TOTAL NUMBER
OF UNITS
861

NOTE
SOLID LINE REPRESENTS
UNITS DISTRIBUTION

BROKEN LINE REPRESENTS
CAPACITY DISTRIBUTION

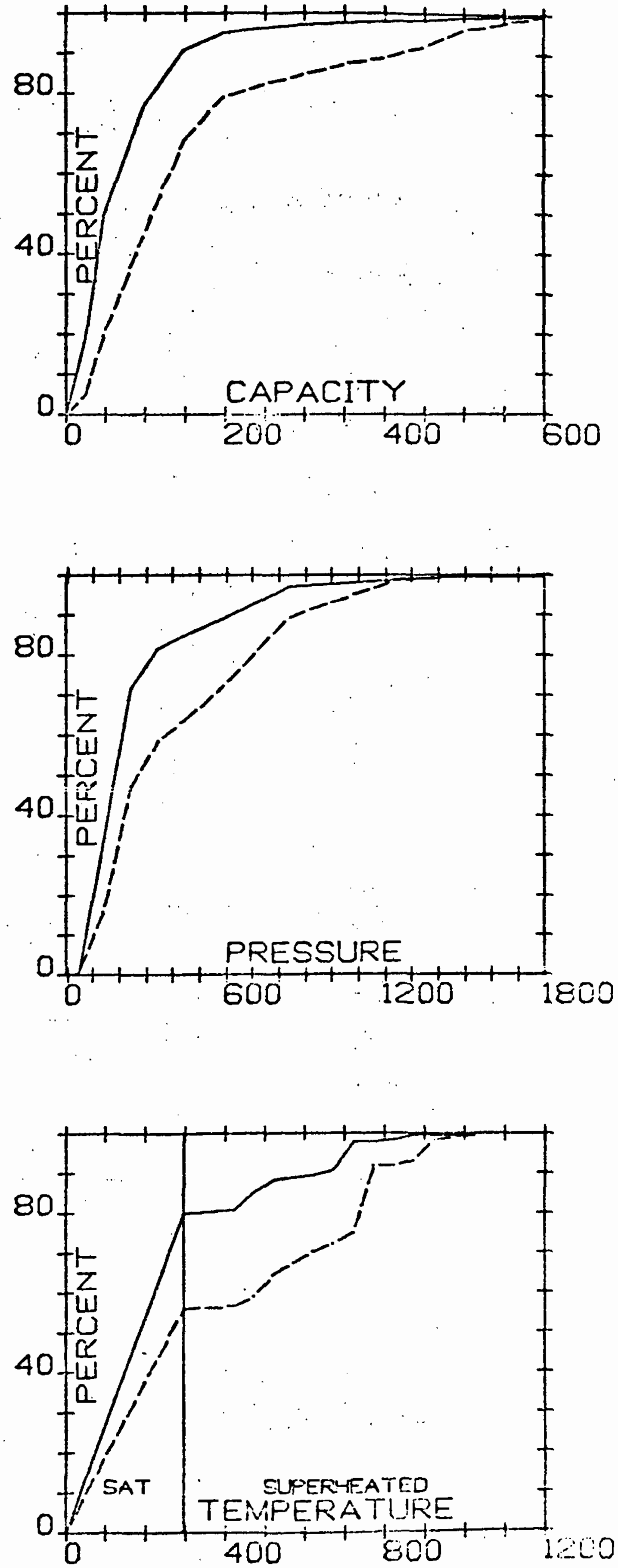
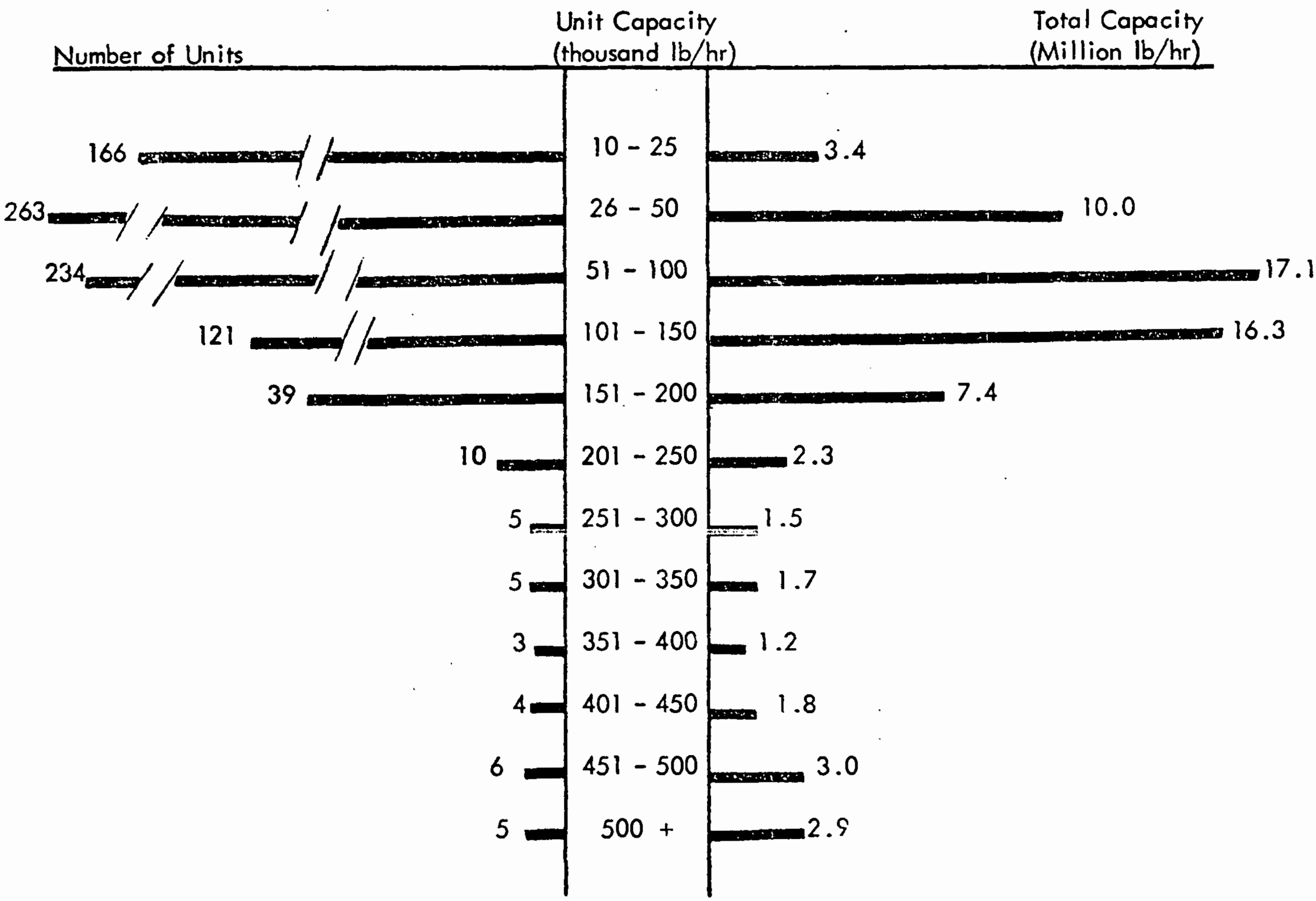


FIGURE B-1[illegible]

1970 SALES OF INDUSTRIAL TYPE WATERTUBE STEAM GENERATORS

DISTRIBUTION AS TO SIZE OF UNIT

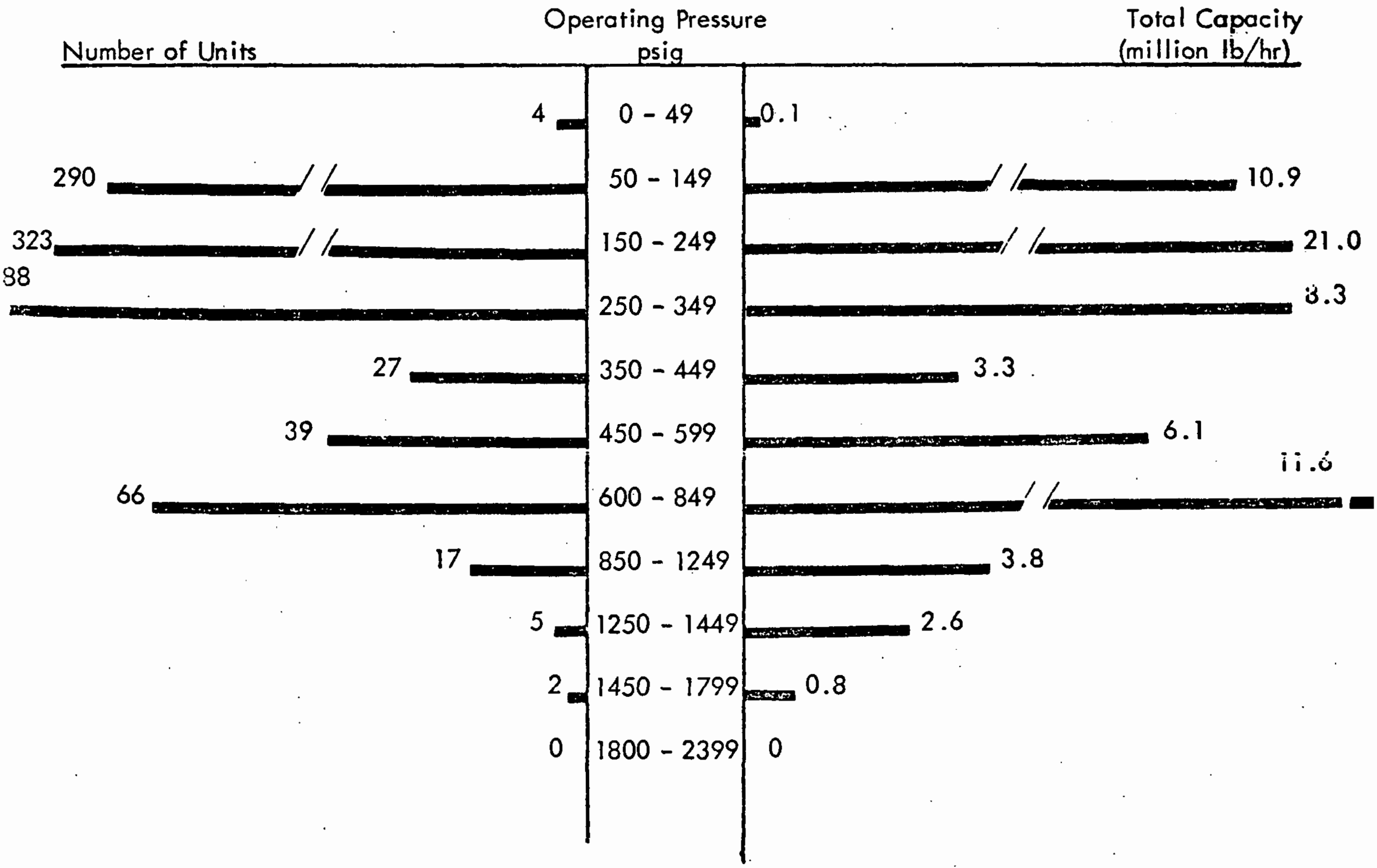


Total Number of Units - 861

Total Capacity 68,359 thousand lb/hr

1970 SALES OF INDUSTRIAL TYPE WATERTUBE STEAM GENERATORS

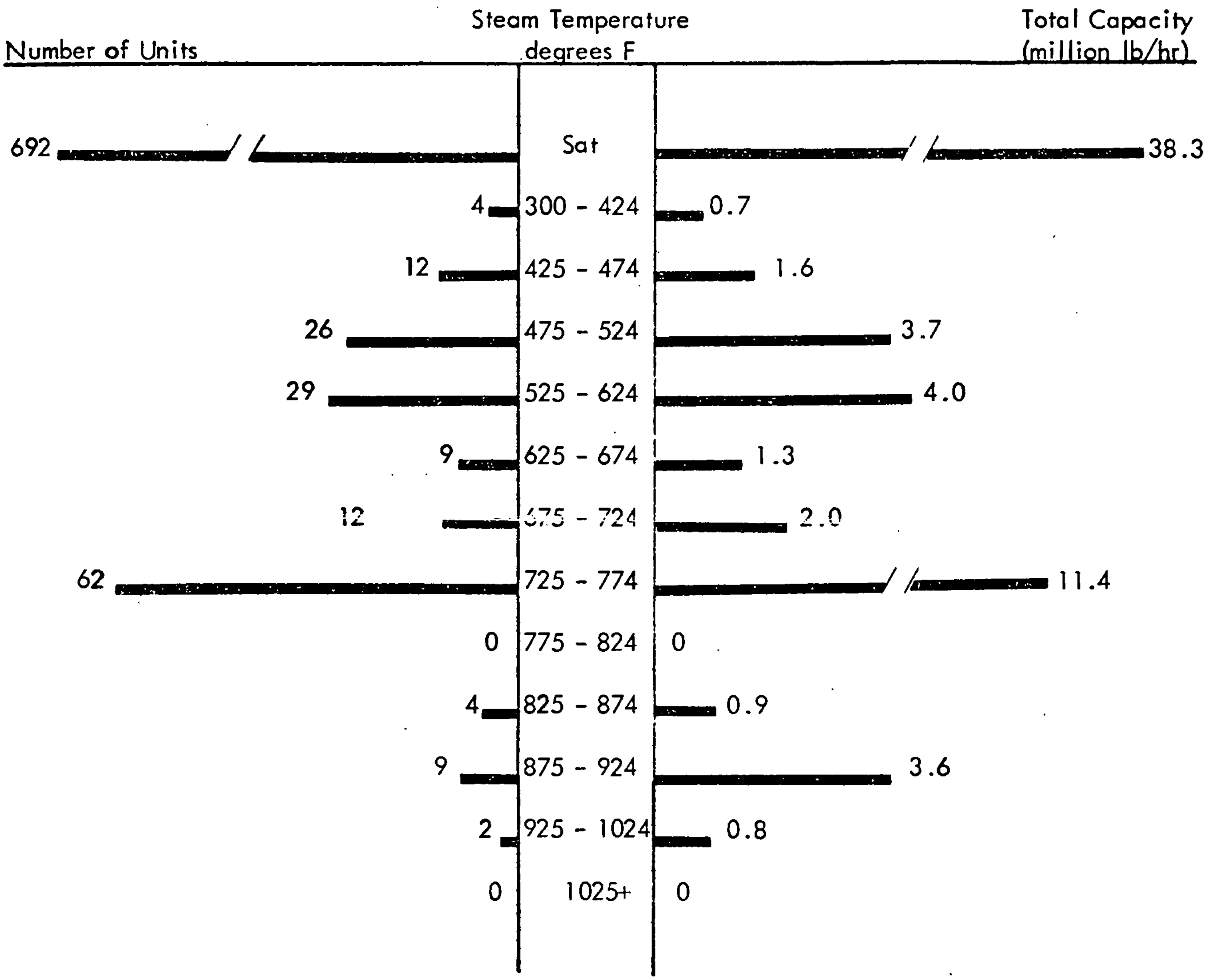
DISTRIBUTION AS TO OPERATING PRESSURE



Total Number of Units - 861

Total Capacity - 68,359 thousand lb/hr

1970 SALES OF INDUSTRIAL TYPE WATERTUBE STEAM GENERATORS
DISTRIBUTION AS TO STEAM TEMPERATURE

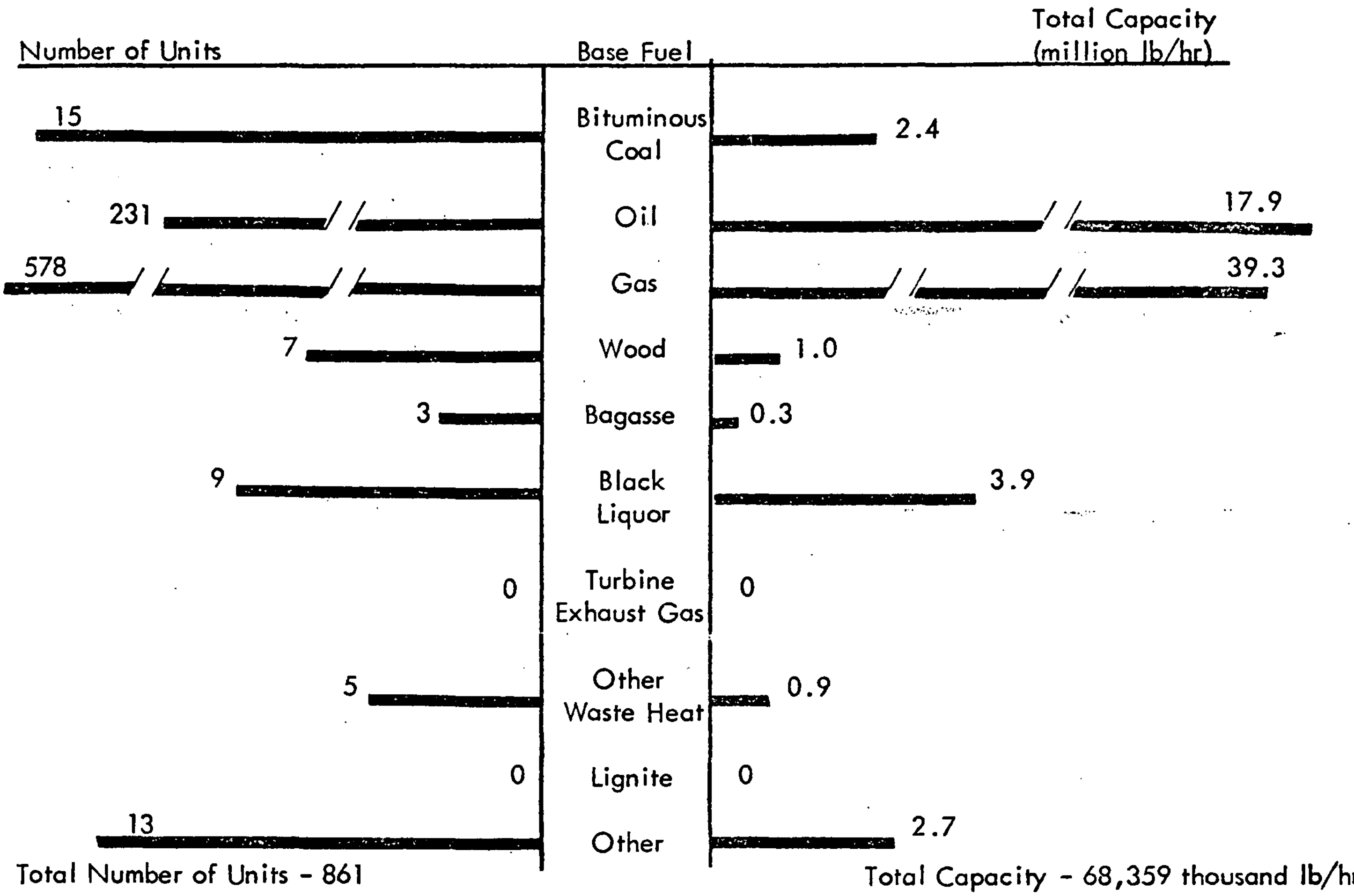


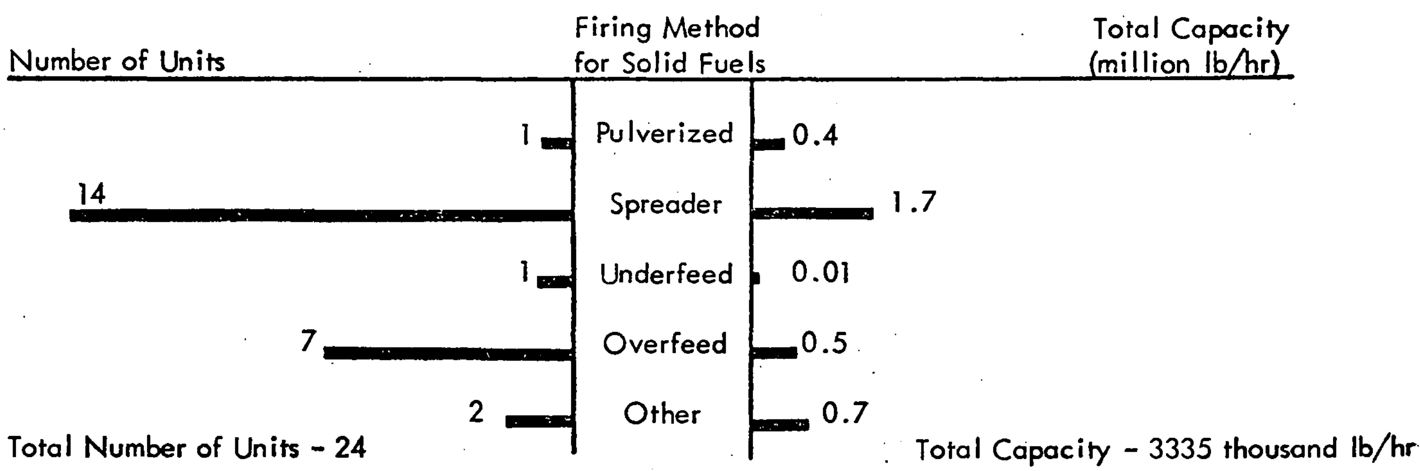
Total Number of Units - 861

Total Capacity - 68,359 thousand lb/hr

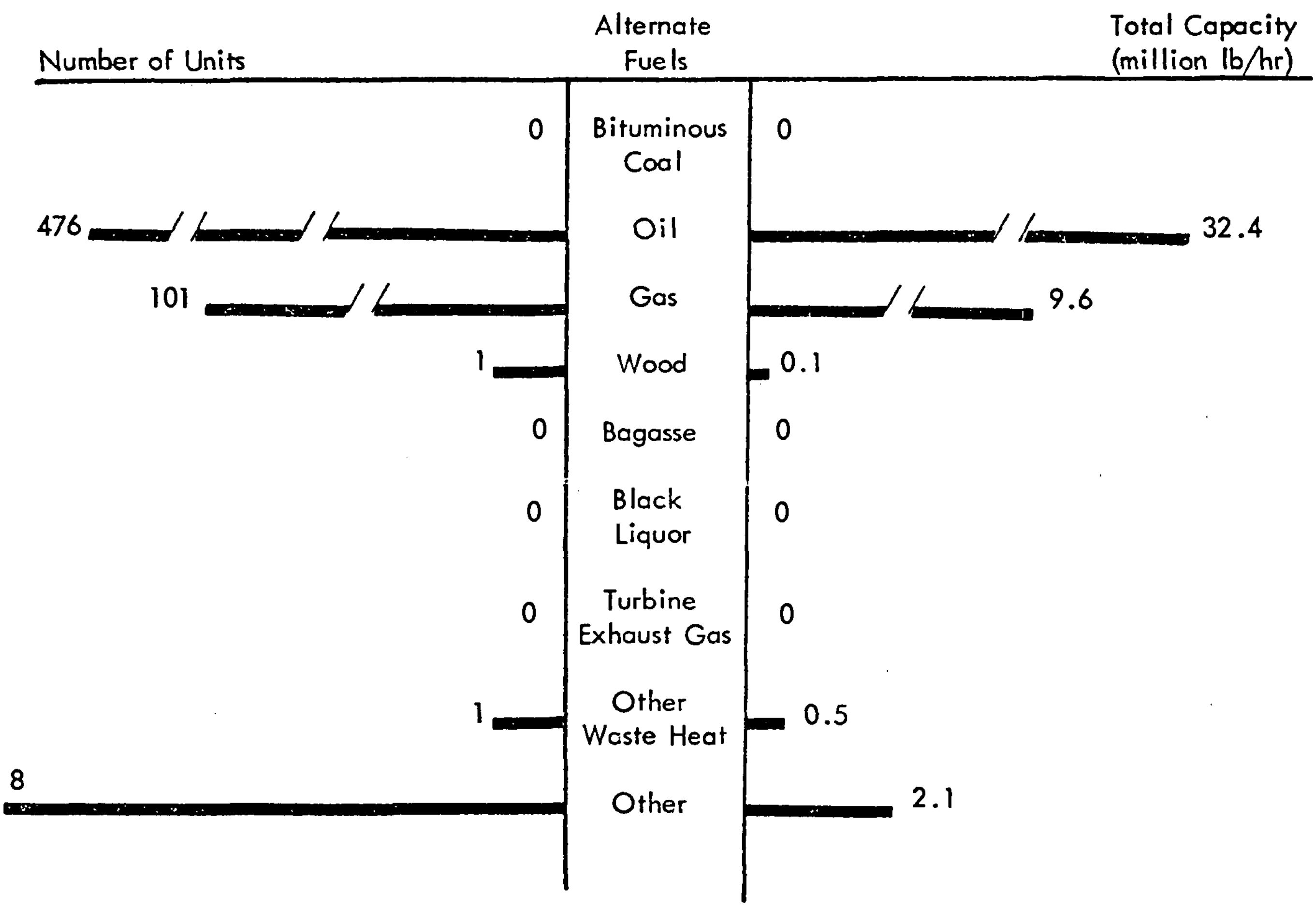
1970 SALES OF INDUSTRIAL TYPE WATERTUBE STEAM GENERATORS

DISTRIBUTION AS TO FUELS AND FIRING METHOD





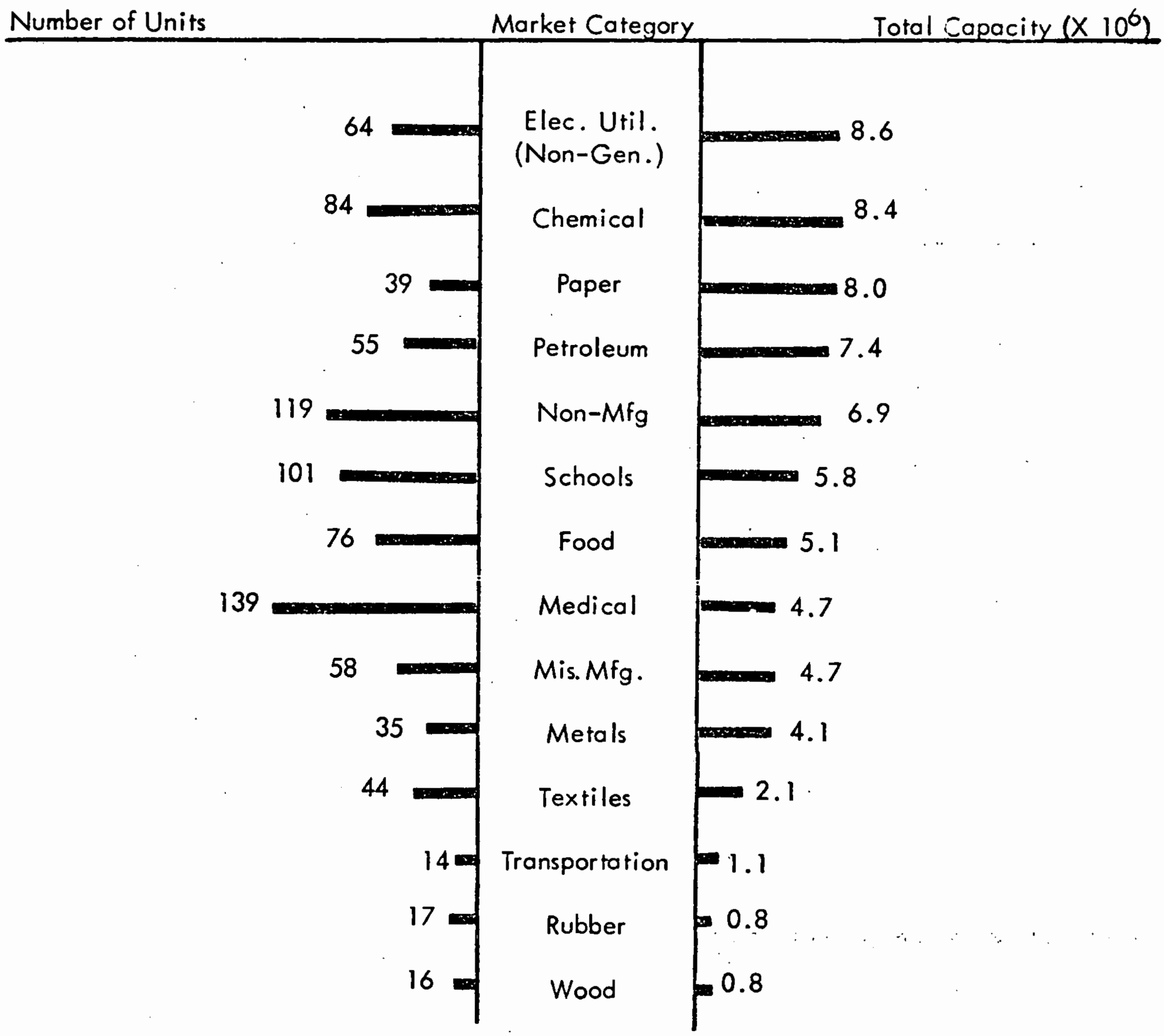
1970 SALES OF INDUSTRIAL TYPE WATERTUBE STEAM GENERATORS
DISTRIBUTION AS TO FUELS AND FIRING METHOD



Total Number of Units - 587

Total Capacity - 44,637 thousand lb/hr

1970 SALES OF INDUSTRIAL TYPE WATERTUBE STEAM GENERATORS
DISTRIBUTION AS TO MARKETS



Total Number of Units - 861
Total Capacity - 68,359 - thousand lb/hr

Note 1: This section includes all industrial type units, steam and hot water, packaged and field assembled, regardless of use.

Note 2: Schools includes schools and colleges; medical includes hosptials, medical centers and related facilities. These categories were formerly included in the Non-Mfg. group.

APPENDIX C

DEVELOPMENT OF FLUIDIZED BED COMBUSTION BOILERS

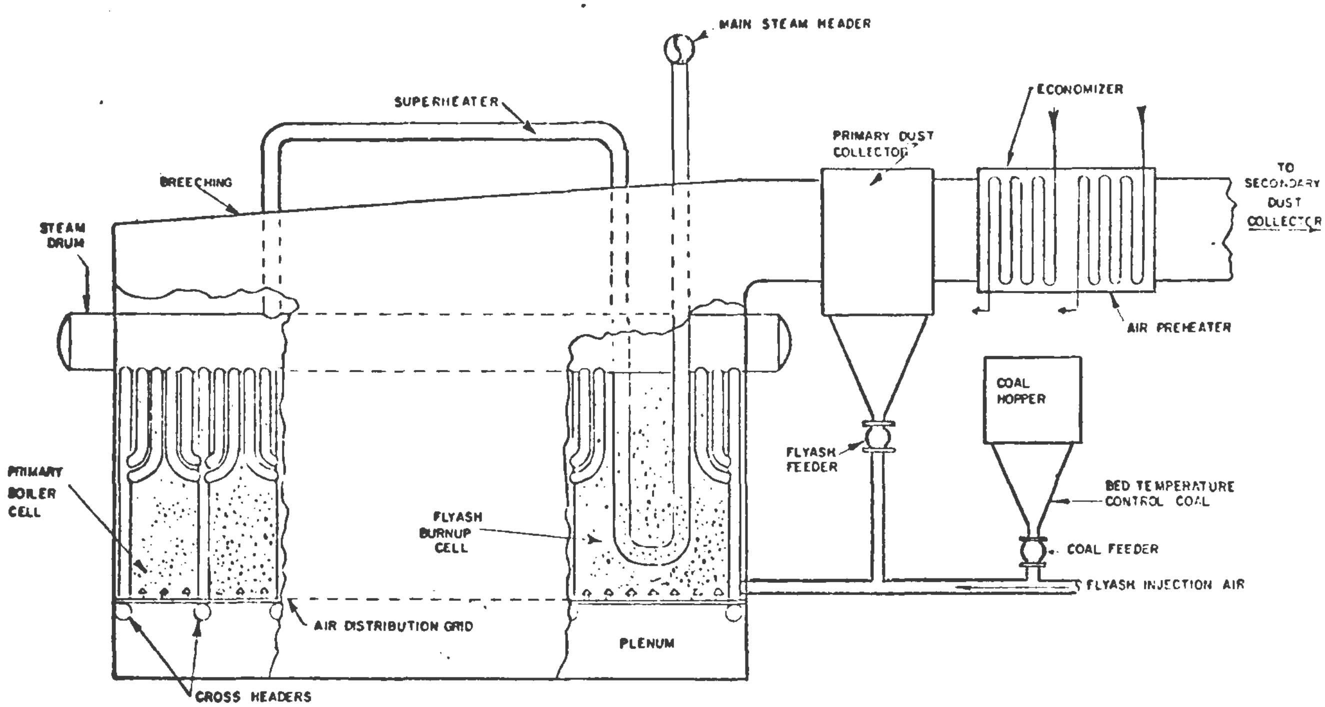
DEVELOPMENT OF FLUIDIZED BED BOILERS

Several boiler systems have been built, tested, or proposed which incorporate fluidized bed combustion. A brief listing of these systems is presented to gain an understanding of what concepts have been considered. The list is not necessarily complete and no attempt was made to present all fluidized bed combustion processes which may be applicable to fluidized bed boilers with sulfur removal.

Pope, Evans, and Robbins

A packaged industrial fluidized bed boiler of capacity 250,000 lb/hr steam has been developed and is shown schematically in Figure C-1. The design is based on modular units 18" x 12' assembled together with water walls dividing the units. High efficiency is obtained by using a carbon burn-up cell to react unburned carbon. The nominal operating conditions projected are primary bed temperature 1600°F, carbon burnup cell bed temperature 2000°F, gas velocity of 12-14 fps, bed height of 12-20 inches, and coal crushed to minus 3/4 inch. Sulfur dioxide emissions would be controlled by injecting fine limestone (perhaps 100 mesh). The modular concept has been extended to a utility boiler design concept.

A new shippable unit of capacity 300,000 lb/hr steam with modified design[9] is shown in Figure C-2. The modules or cells run parallel to the steam drum and connect to a single carbon burnup cell. Flue gas passes straight up through the convection section and economizer, both located in the freeboard. Primary superheater is arranged to serve as baffle screens above the bed. The bed operating conditions are essentially the same as the earlier design.



Overall Dimensions: 12 ft wide x 16 ft high x 40 ft long

Figure C-1 - Schematic fluidized-bed boiler incorporating a carnon burn-up cell

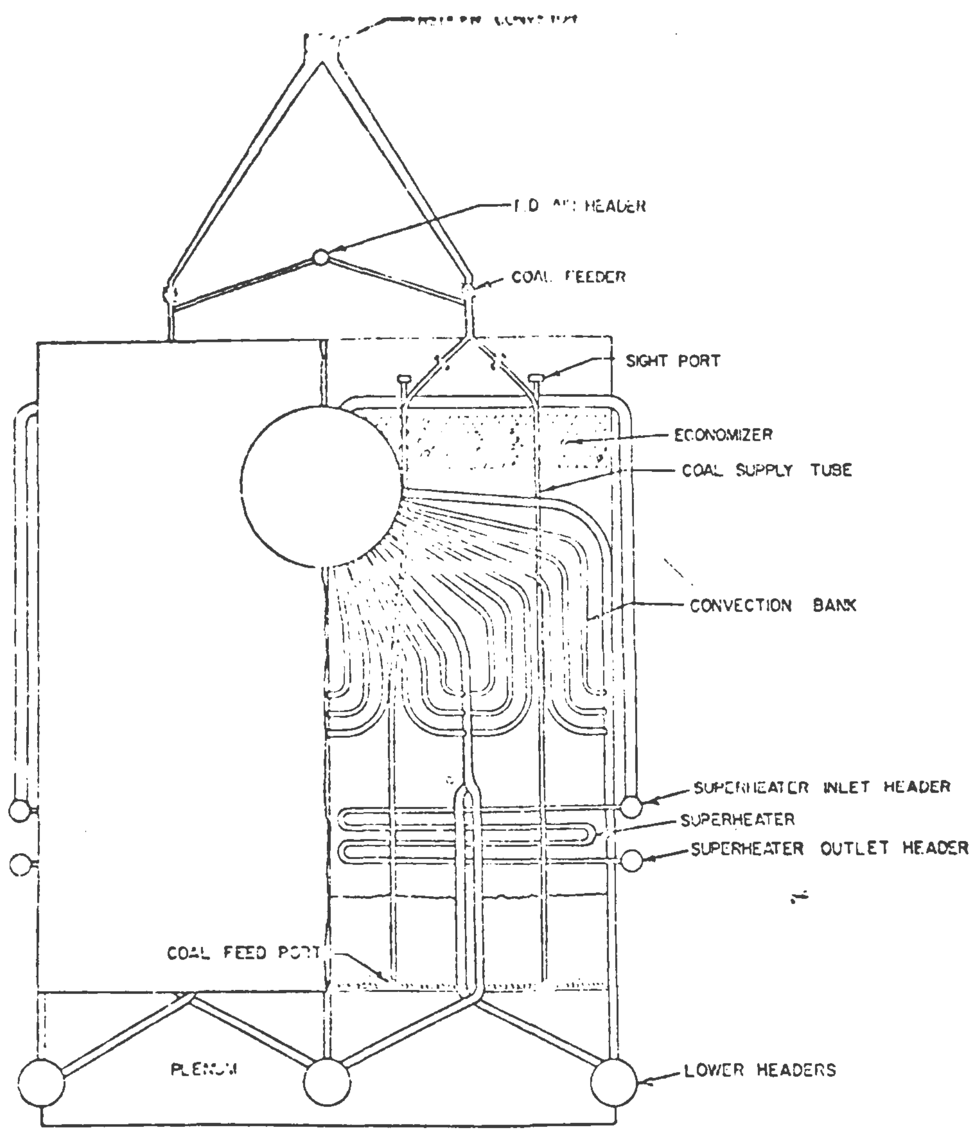


FIGURE C-2 - End View, Coal-Fired Fluidized Bed Utility Boiler, Factory-Assembled

"SO_2 acceptor process", where the SO_2 is absorbed in the primary bed and regenerated in the carbon burnup cell, was conceived. The continuous absorption-regeneration is accomplished by recirculation of limestone beds between the primary bed and carbon burnup cell through the provided openings.

Esso Research & Engineering

Esso R&E have proposed an operating concept for a fluid bed boiler with sulfur removal. The fluid bed combustor would be operated with a bed of "large" limestone particles and pulverized coal would be pneumatically fed to the bed. The ash from the coal would be entrained from the system, leaving a bed composed essentially of limestone. The reacted limestone would be removed from the bed, regenerated, and recycled back to the combustor.

Initial experiments were performed in a 3" diameter coal combustor and a 2" diameter limestone regenerator. The fluidizing velocities used were 3 ft/sec and 2 ft/sec respectively. The bed temperature was 1600°F in the combustor and 2000°F in the regenerator.

CRE-NCB

The NCB has undertaken an extensive program to provide a series of fluidized bed boiler designs. These include

- a 220,000 lb/hr, 20 MW, water tube boiler. Preece, Cardew and Rider (PCR) have produced preliminary designs for this atmospheric pressure boiler shown in Figures C-3 and C-4. The design is later changed, as shown in Figure C-5, with 1" tubes of length 60'-80' running horizontally back and forth in the direction of the larger dimension of boiler. Overall boiler dimensions are 14' x 40' x 16' with active bed area 400 ft^2. The bed is divided into three beds and a carbon burnup cell. The fluidizing velovity is 8 ft/sec and coal particle size is -1/8". The design is for installation at Grimethorpe to serve as a demonstration plant.

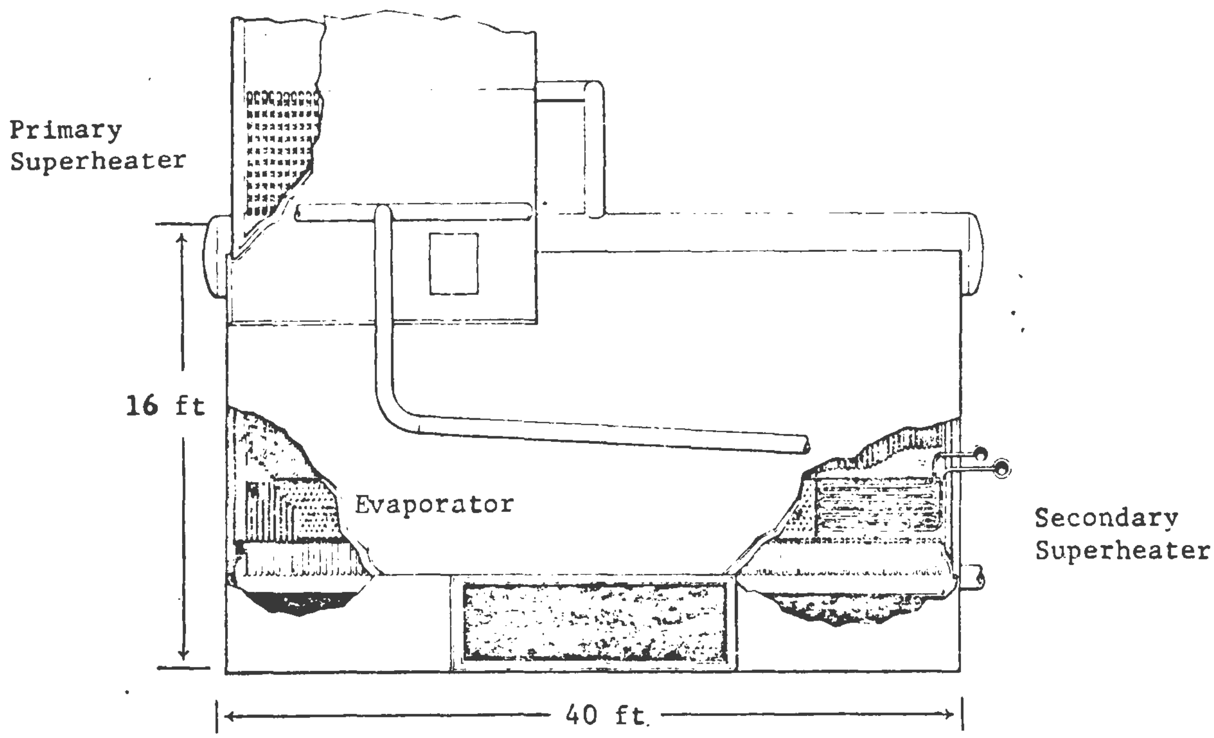


FIGURE C-3 - PCR Design of 20MW Water Tube Boiler (Side View)

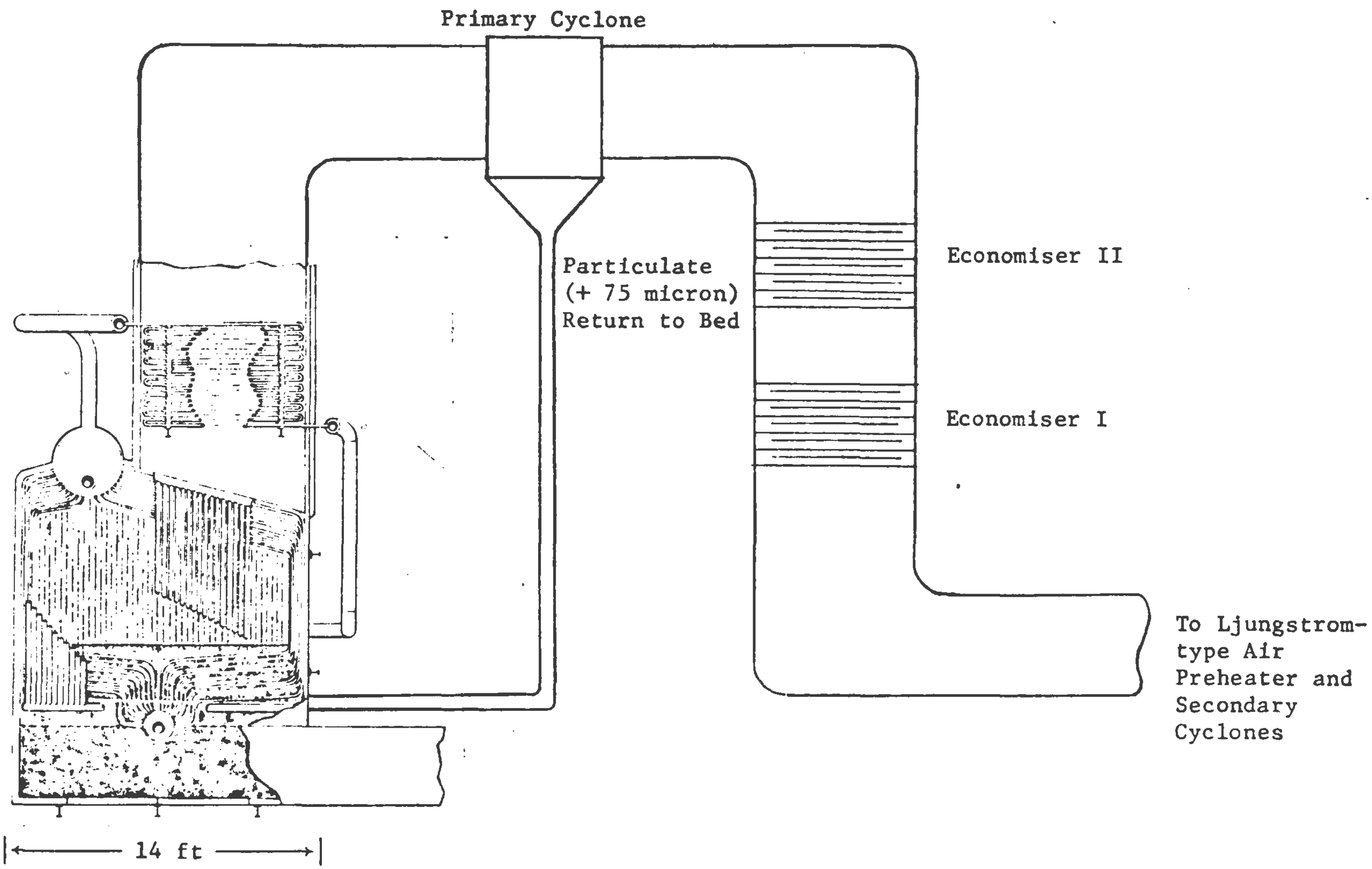


FIGURE C-4 - PCR Design of 20MW Water Tube Boiler (Front View)

Dwg. 6162A23

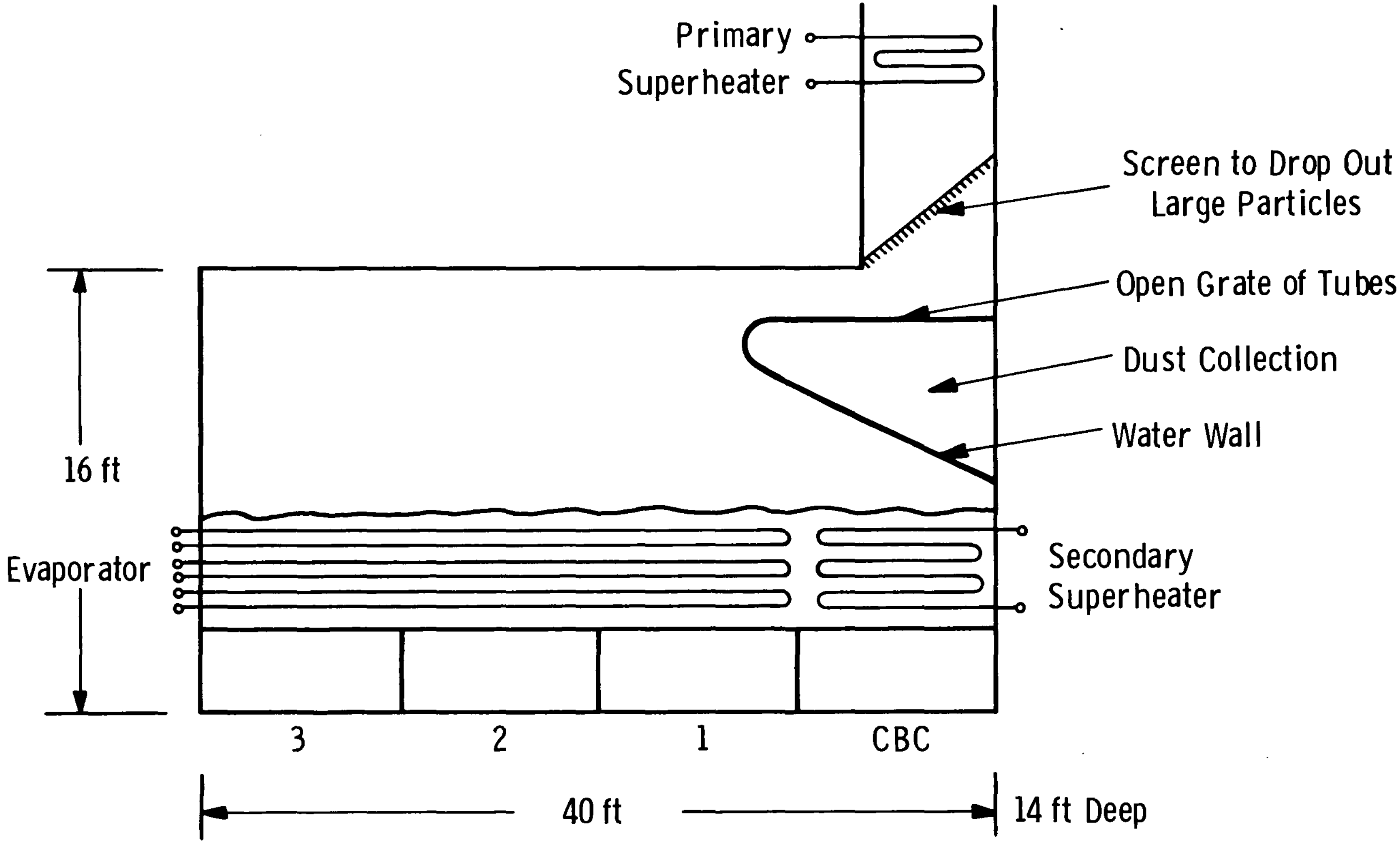
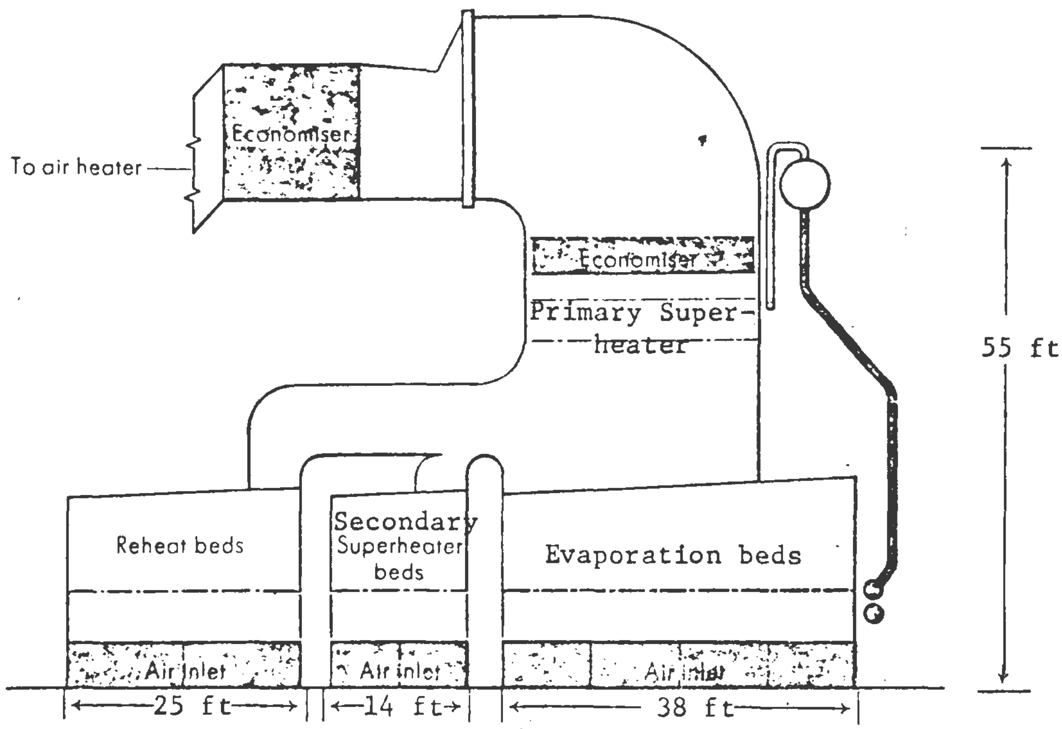


Fig. C–5 –PCR design of 20 MW atmospheric pressure boiler–Grimethorpe plant

- a 120 MW atmospheric pressure utility boiler. PCR have also carried out a design study of this utility boiler which has been sectionalized to allow shop fabrication. The conceptual design is shown in Figure C-6 with overall dimensions. The total active bed area is 2212 ft^3. The heat transfer surfaces are provided by 1.5" tubes in 3.5" spacings. The design study was carried out to uncover design and operating problems and to investigate the economics of a small-scale utility boiler.

- a 500 MW atmospheric pressure utility boiler. An early conceptual design is shown in Figure C-7[1]. The nominal operating conditions selected include a fluidizing velocity of 2 ft/sec, coal crushed to -1/16", and bed temperature near 800°C.

- a 660 MW atmospheric pressure utility boiler. The conceptual designs by International Combustion, Babcock & Wilcox, Foster Wheeler, and PCR are shown in Figures C-8 through C-11. International Combustion proposes a battery of 12 units (Figure C-8) each capable of 55 MW. Superheater and evaporator tubes are located in the fluidized bed. Babcock and Wilcox proposes a boiler containing 15 fluidized beds of 90 ft x 20 ft (Figure C-9). Air flows in parallel through each of the beds at 3 ft/sec. Beds operate at 780-900°C. Of the 15 beds, 9 are superheaters, 3 are reheaters, 2 are evaporators, 1 is combined evaporator-reheater. Foster Wheeler proposes an arrangement of 6 fluidized beds, each 66 ft wide (Figure C-10). Air flows in parallel through each of the beds at 3.5 ft/sec. Bed temperature is 850°C. The bed depth is 2.5 ft. The coal is fed to the beds at 850 points. The evaporator and primary superheater surface is provided by horizontal tubes in the beds. The top bed serves as a carbon burnup cell. Reheater and secondary superheater surfaces are located in the gas stream leaving the boiler. PCR extends its 120 MW atmospheric design concept to a 660 MW utility boiler installation. The design consists of 40' x 16' x 14' modules with 2 evaporator beds,



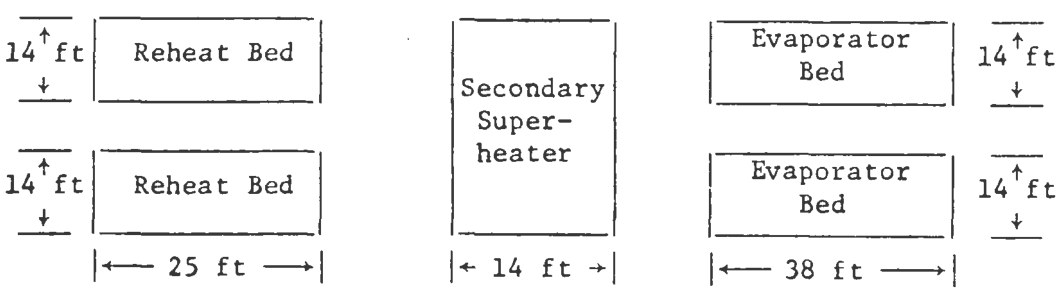


FIGURE C-6 - PCR Conceptual Design of a 120MW Atmospheric Utility Boiler

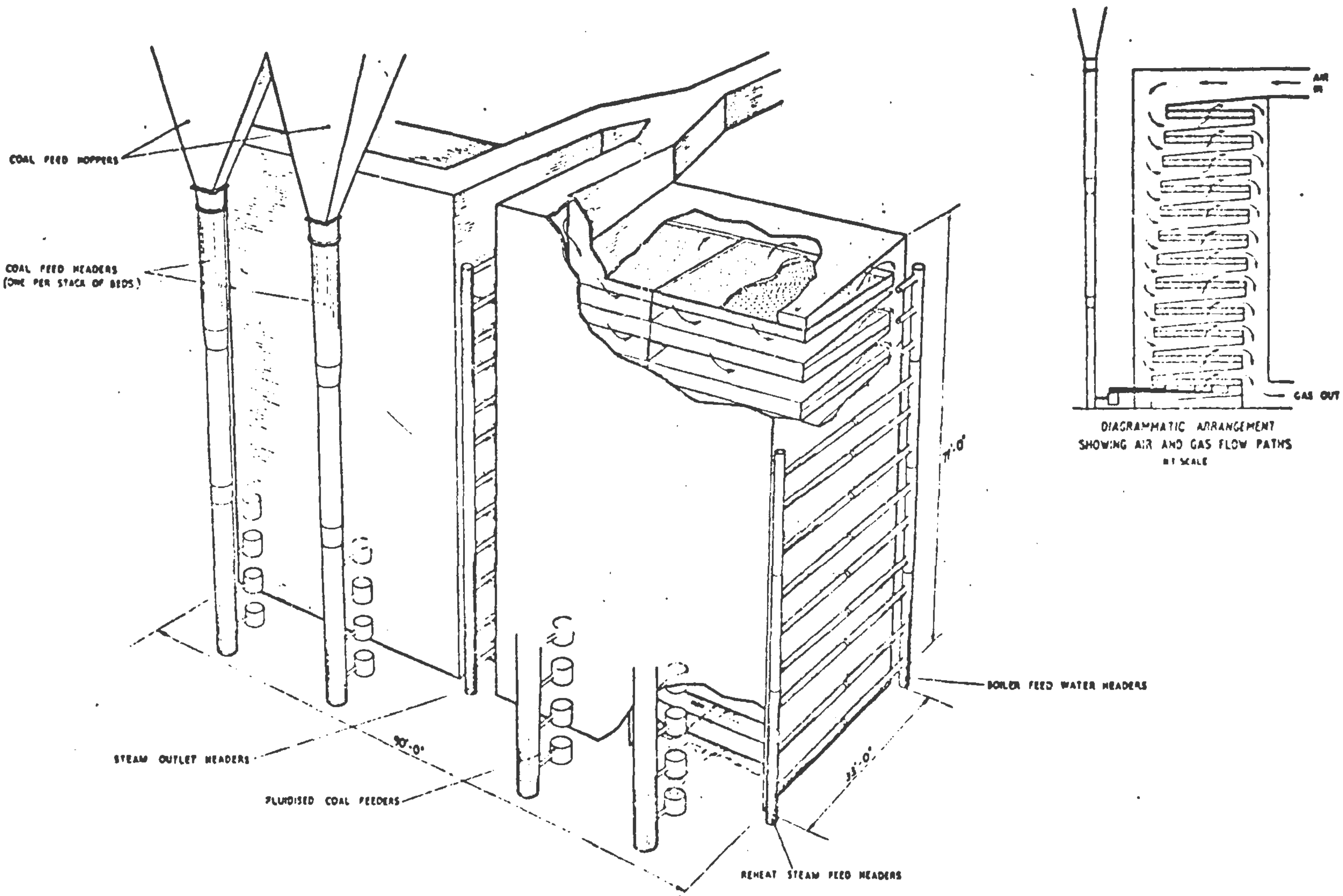


FIGURE C-7 - LOW VELOCITY, ATMOSPHERIC 500 MW FLUIDIZED BED BOILER

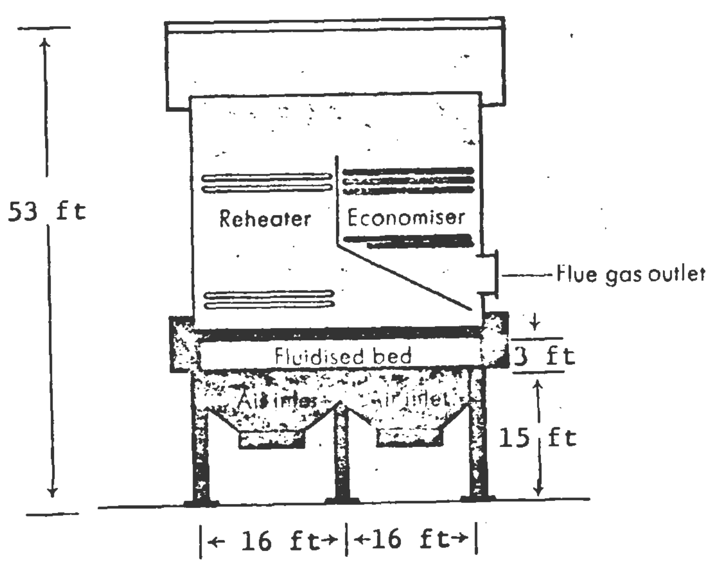


FIGURE C-8 - Conceptual Design by International Combustion for a 660MW Atmospheric Utility Boiler

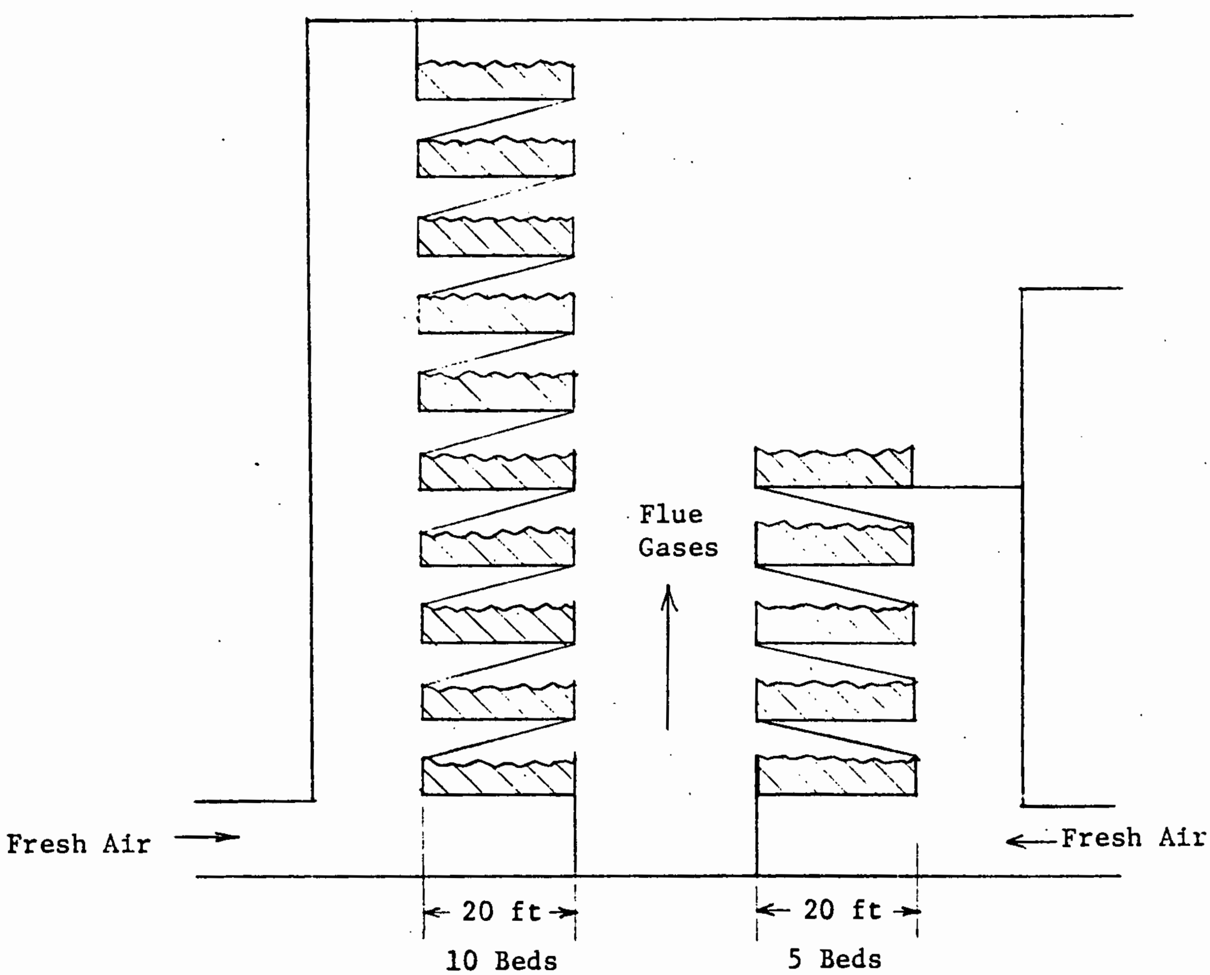


FIGURE C-9 - Conceptual Design by Babcock & Wilcox for a 660MW Atmospheric Utility Boiler

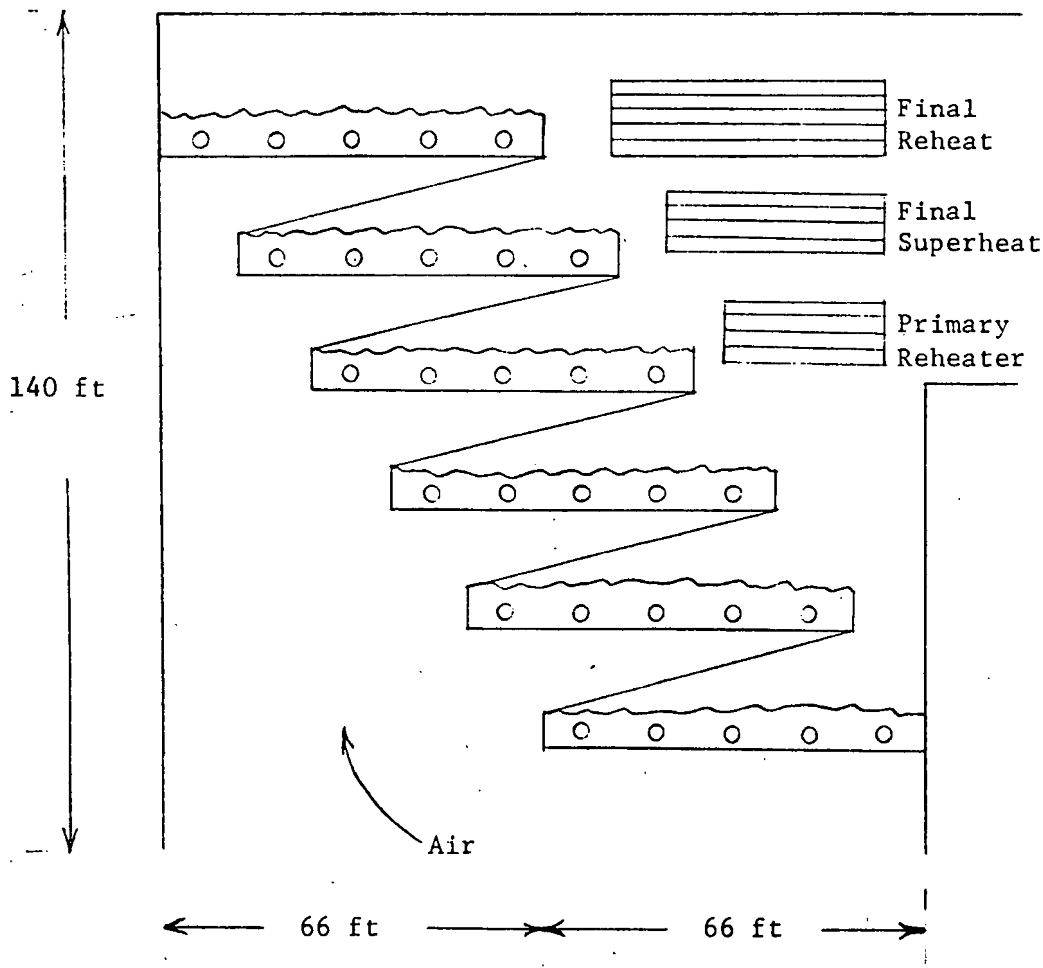


FIGURE C-10 - Conceptual Design by Foster Wheeler for
a 660MW Atmospheric Utility Boiler

2 reheater beds, and 1 superheater bed. The overall dimensions are 90' (depth) x 120' (width) x 100' (height), (Figure C-11).

BCURA Industrial Shell Boiler

BCURA is developing a fluid bed coal-fired packaged shell boiler to compete with oil-fired units at capacities up to 50,000 lb steam/hr. A prototype of the shell boiler is shown in Figure C-12. The projected operating conditions are gas velocities of 10-14 fps, bed depth of about 2 ft, and coal crushed to minus 1/4 in. The prototype is 3 ft diameter and has a capacity of 8000 lb steam/hr feeding 1000 lb coal/hr. The heat transfer tubes are set at an angle of 10° to allow for natural circulation of the water. About 50% of the heat release will be extracted in the bed.

BCURA High Pressure Boiler

BCURA proposes a high-pressure fluid bed boiler. The fluid bed is operated at 15 to 25 atm, 800°C, and the energy in the gases from the bed is recovered in a gas turbine (Figure C-13). Lower capital cost through boiler size reduction and increased cycle efficiency are projected advantages over an atmospheric fluid bed boiler. The proposed system would convert approximately 70% of the coal energy to steam. An experimental apparatus, shown in Figure C-14 has been constructed at BCURA to burn 1000 lb coal/hr at 5 atmospheres.

Conceptual design of a 140 MW pressurized boiler is shown in Figure C-15. The detailed design will be carried out by John Thompson Ltd. The boiler is contained within a pressure cylinder 14 ft in diameter and 100 ft long. Active bed area is 900 ft^2. Air is fed at 8 atm. with fluidizing velocity of 2 ft/sec. Coal is fed at 136 feed points with coal particle size - 1/16". One-inch tubes at 3" spacings are used for heat transfer surface in the bed. Gas turbine generates ∿ 12% of the total power.

Dwg. 6162A24

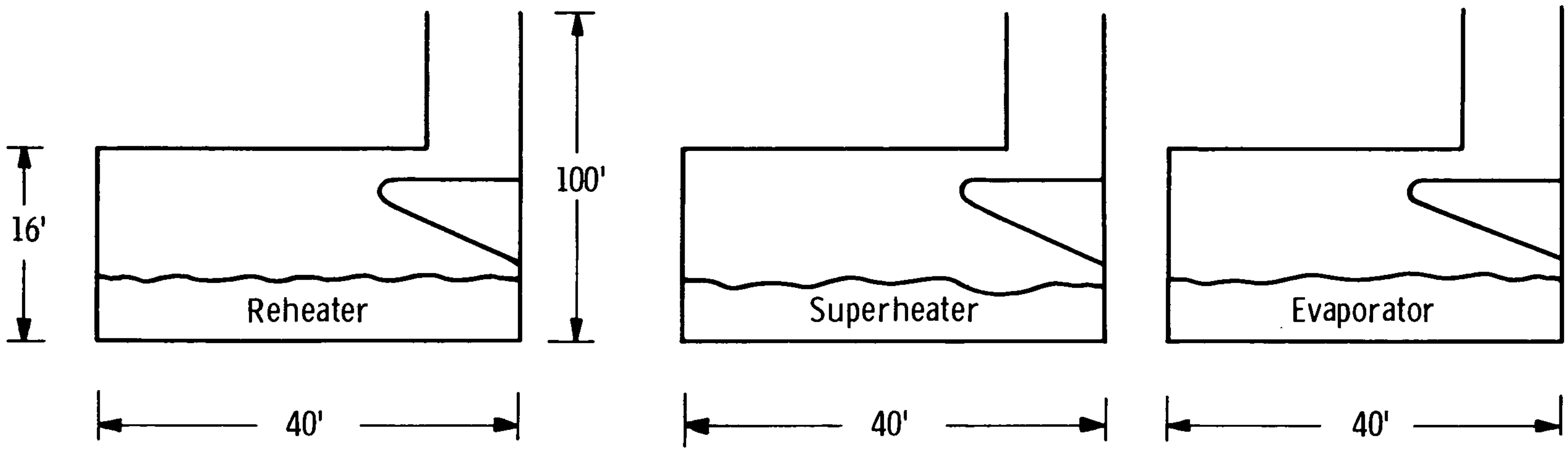


Fig. C-11 —Conceptual design by PCR for A600MW atmospheric utility boiler

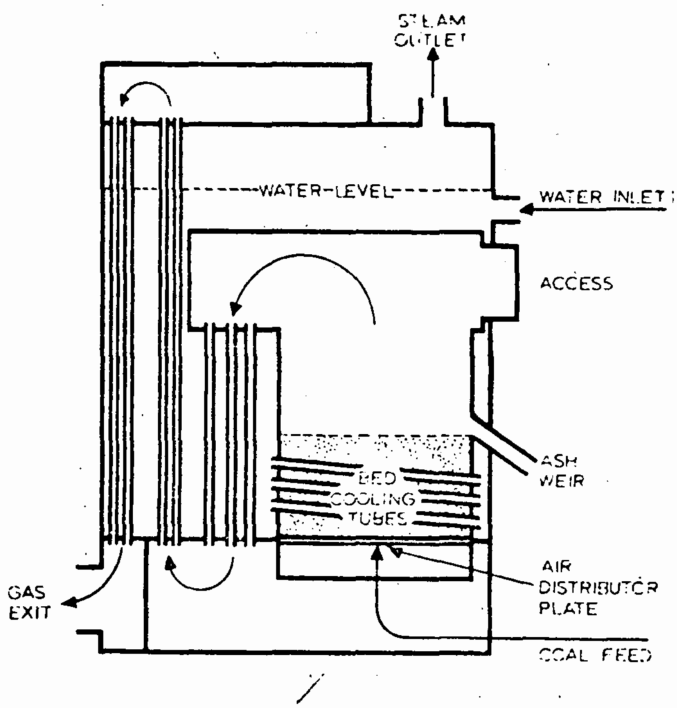


FIGURE C-12 PROTOTYPE BCURA INDUSTRIAL FLUID-BED SHELL BOILER

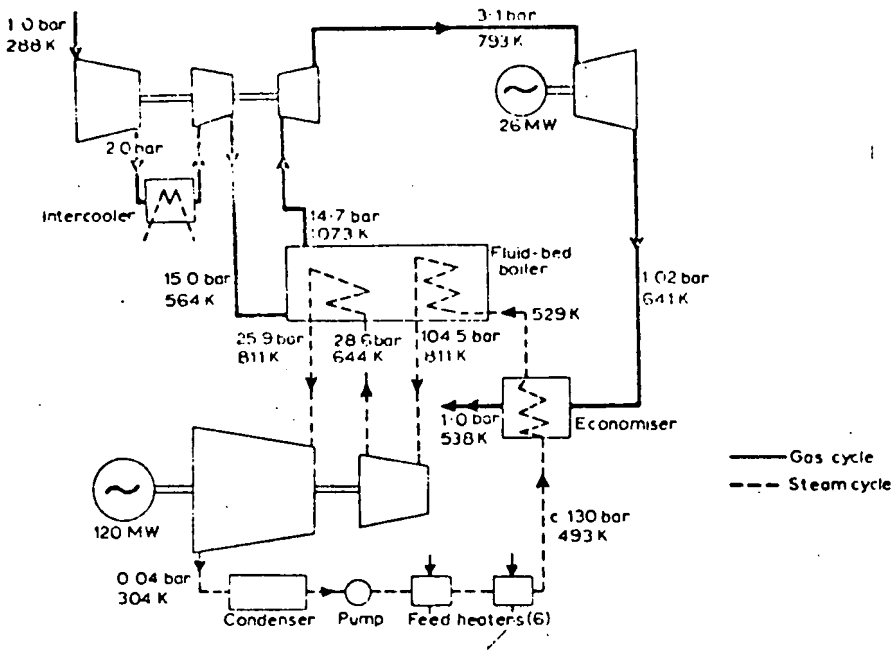


FIGURE C-13-SCHEMATIC DIAGRAM FOR COMBINED-CYCLE POWER GENERATION

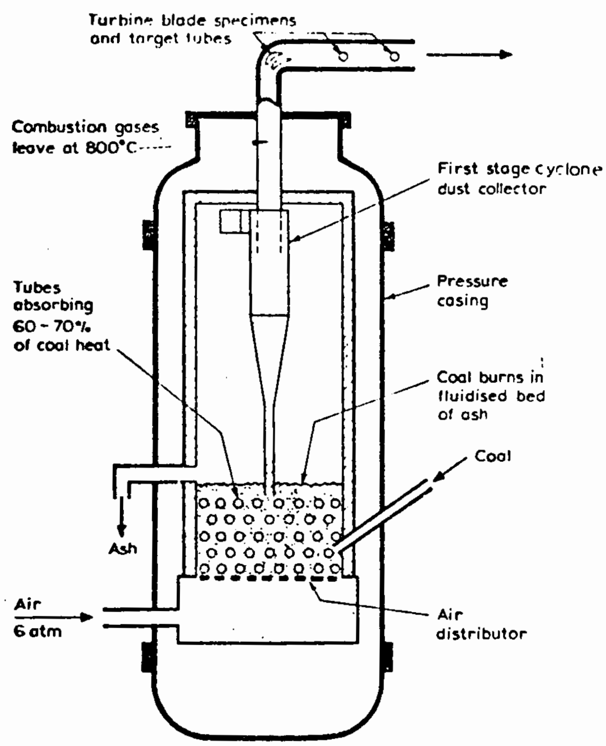


FIGURE C-14-SCHEMATIC DIAGRAM OF PRESSURIZED FLUID-BED COMBUSTOR

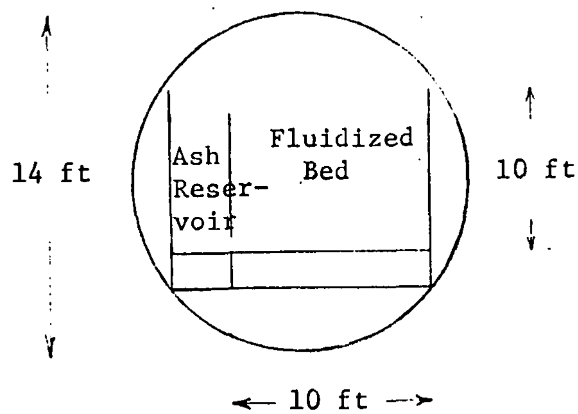


FIGURE C-15 - Conceptual Design of BCURA 140MW
High-Pressure Utility Boiler

Ignifluid Boiler

Albert Godel's Ignifluid boiler was developed by modifying the air supply system of the travelling-grate furnace to create a fluid bed on the grate[2,3]. The grate is divided into sections to control fluidization. A sketch of the design is shown in Figure C-16. The coal particle size is < 20 mm and the gas velocity is near 10 fps. The Ignifluid boiler is capable of burning a wide range of coals by using the discovery that the ash of essentially all coals agglomerates near 1000°C. The fluid bed is operated between 1000 and 1200°C. Ash particles increase in size, sink through the bed onto the grate, and are carried out of the bed on the moving grate. Partial combustion occurs in the bed and secondary air (approximately 50%) is supplied above the bed to complete the combustion. Particulates are reinjected into the bed. A 60 MW unit has been built and a 275 MW unit is projected for Northeastern Pennsylvania by UGI.

Godel's Stacked Fluid Bed Boiler

A. A. Godel has also proposed a stacked fluid bed boiler[4]. The bottom fluid bed would be an Ignifluid unit. Water walls are proposed, but no heat transfer tubes are incorporated in the first bed. A secondary fluid bed is located above the primary furnace. This bed would contain granulated refractory and heat transfer surface and would operate near 850°C. Unburned carbon particles carried by the gas from the first bed would be combusted in the second bed. Above the second bed, a third and fourth bed are provided with heat exchange tubes for steam reheat and an economizer.

Lurgi

Lurgi has a "Turbulent Layer Process" which they recommend for combustion of low-grade fuels. The heat of combustion is utilized for the production of steam. A schematic of the process is shown in Figure C-17.

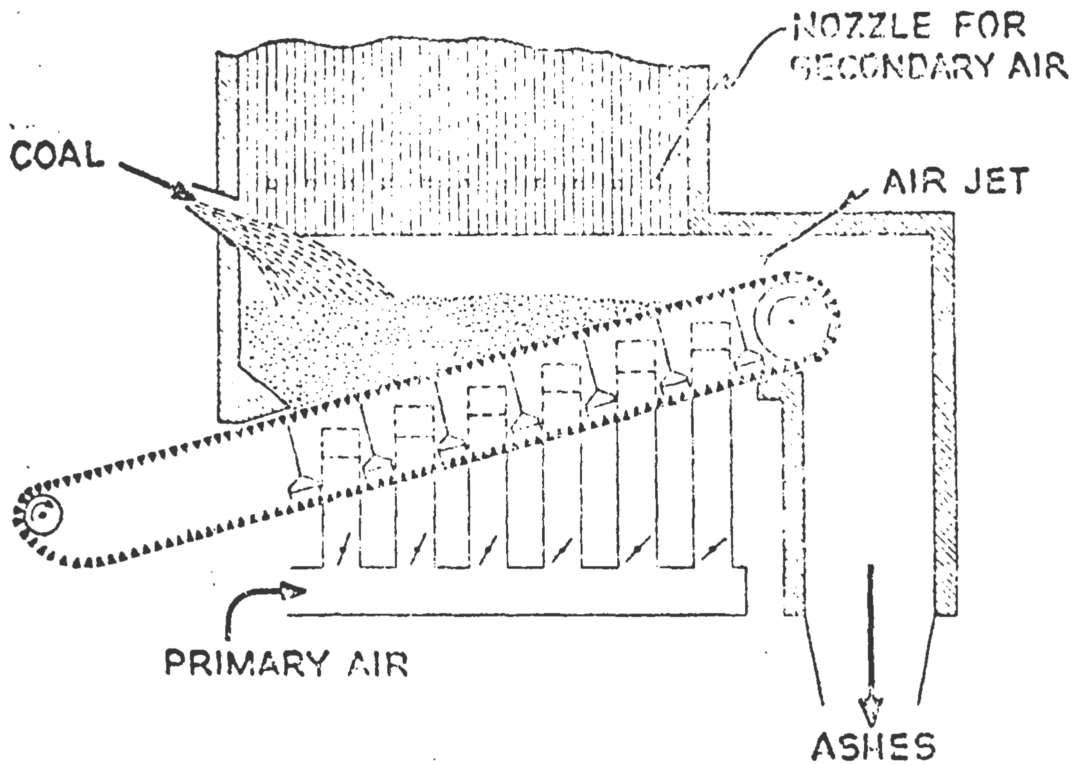


FIGURE C-16-IGNIFLUID BOILER COMBUSTION SECTION

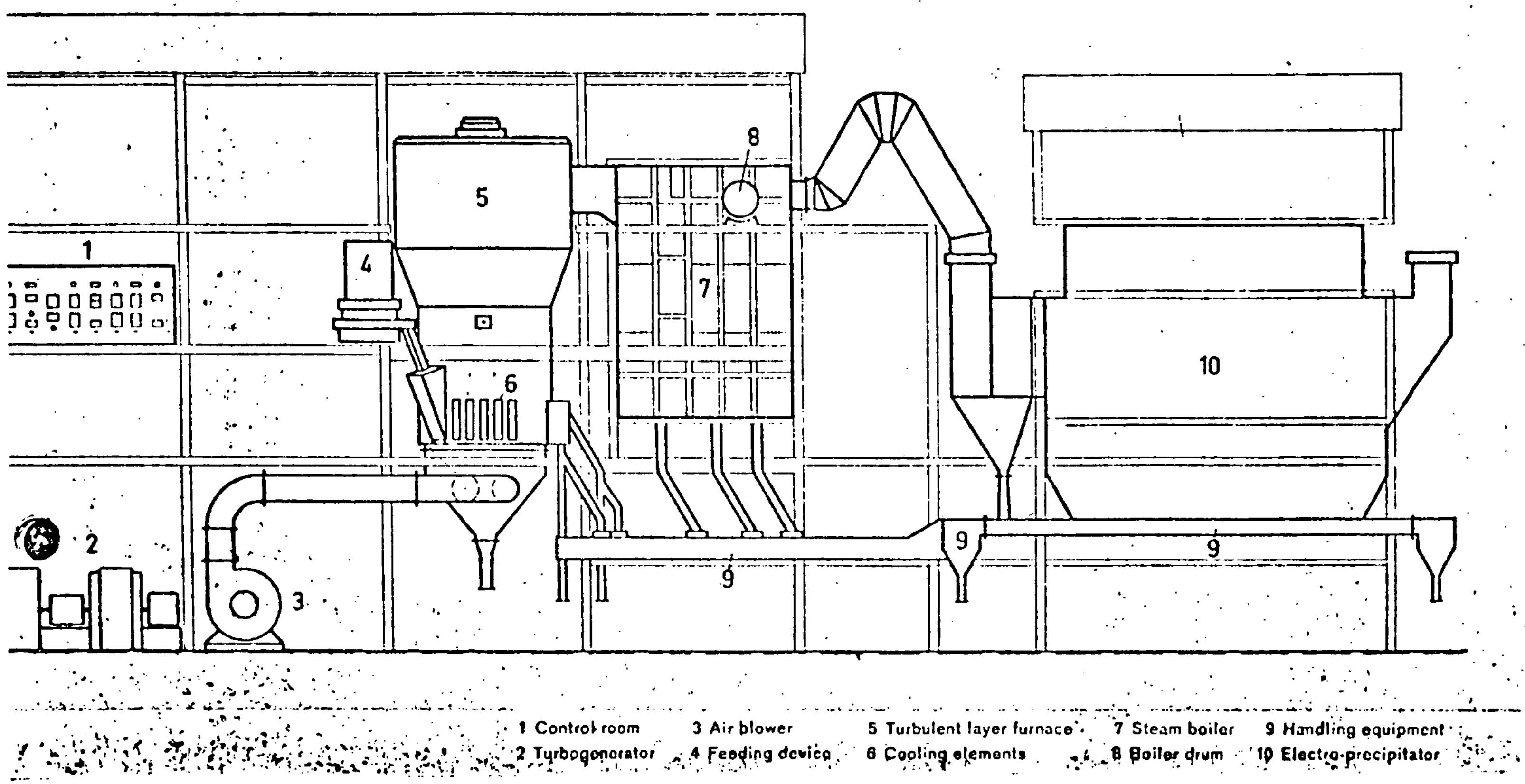


FIGURE C-17-LURGI "TURBULENT LAYER PROCESS"

The turbulent layer furnace (fluidized bed) is cylindrical with an enlarged upper section which serves as a secondary combustion chamber. The lower part of the furnace contains a grate of refractory material with openings through which the fluidizing air enters. The burned material is discharged at the bottom of the furnace.

Esso Petroleum Co., Ltd.

Esso Ltd. Research Center has considered the desulfurization of fuel oil in fluidized beds of lime particles when the combustion is complete and when it is partial. One concept being considered places a fluidized bed gasifier at the bottom of a boiler. Fuel is partially burned in the bed and combustion is completed in burners above the refractory cyclones as depicted in Figure C-18. A combined gas turbine-steam turbine plant is shown in Figure C-19.

Stratton

J. F. O. Stratton[5] presented a spouting fluid bed boiler system in 1928. Coal < 6 mm can be fed to the unit. Gas velocities range from $\sim$40 fps at the base to 10-15 fps in the furnace. The bed operated near 2000°F in order to agglomerate the ash which would be removed after falling through the grate. The cross section of a furnace and boiler system installed at a U. S. Gypsum Co. paper mill is shown in Figure C-20. The unit handled 5000 lb of crushed coal per hour.

Institute for Fuel Research, Czechoslovakia

Laboratory and semi-production units have been operated. The concept is presented in Figure C-21. Temperatures in the fluid bed were maintained between 900 and 950°C without agglomeration. Unburned combustion gases and particles from the fluid bed are burned in a second combustion space such as a cyclone furnace. Temperatures in the cyclone furnace were from 1000 to 1200°C.

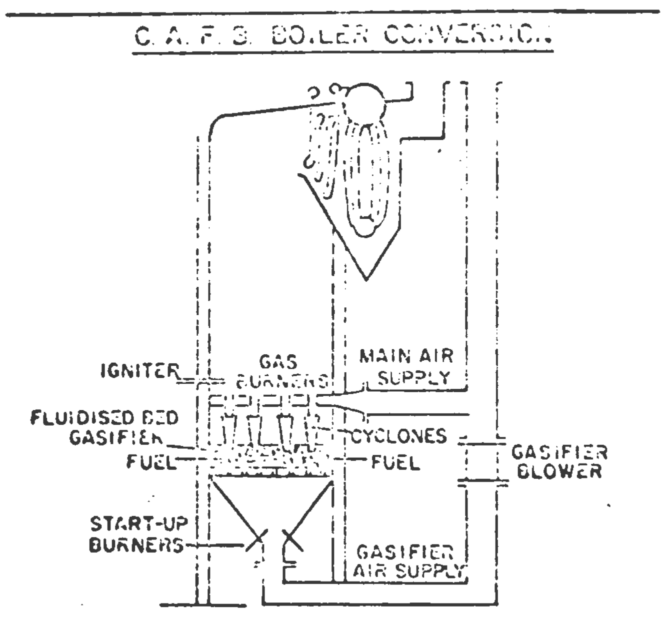


FIGURE C-18—ESSO LTD. FLUID BED BOILER FOR FUEL OIL

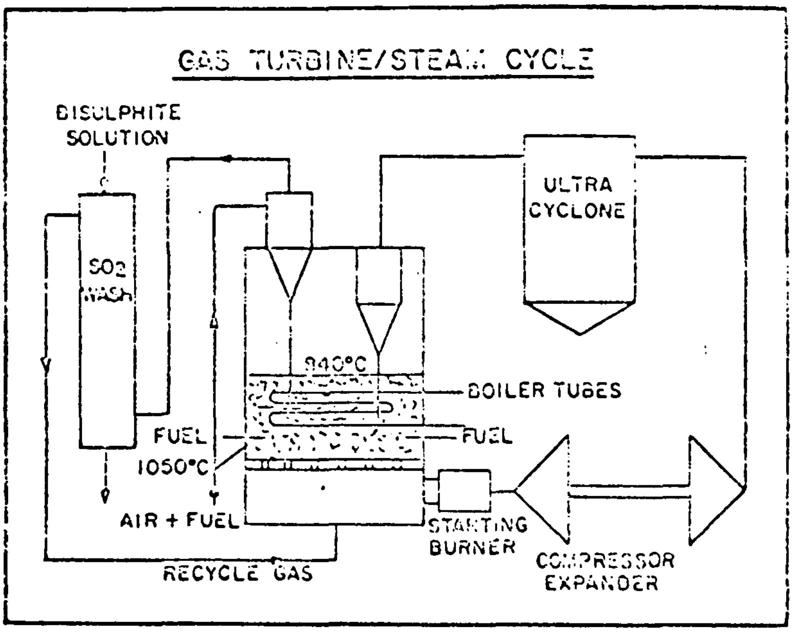


FIGURE C-19—ESSO LTD. COMBINED CYCLE FLUID BED BOILER SYSTEM

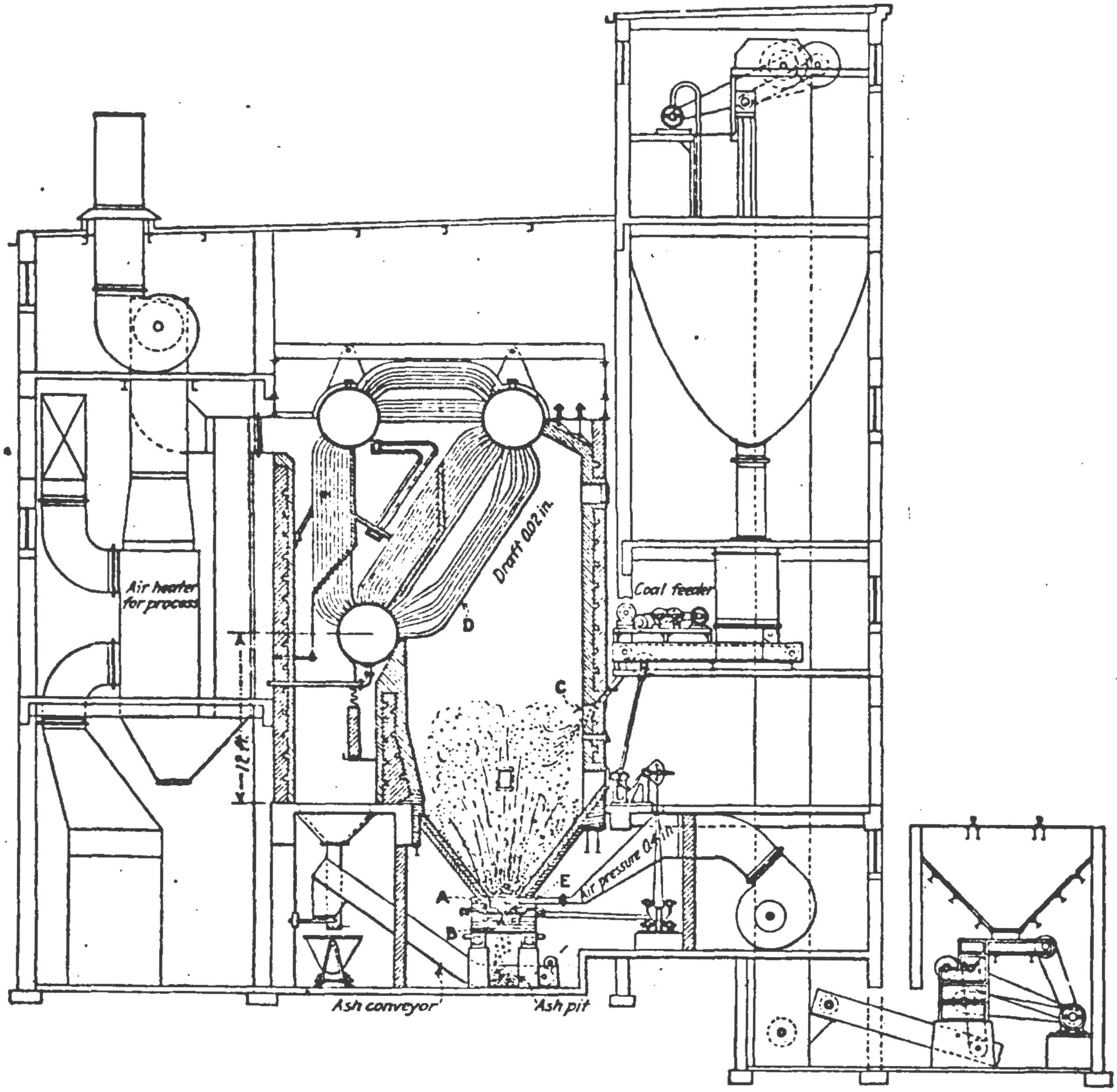


FIGURE C-20 - SPOUTED BED BOILER SYSTEM PRESENTED BY J. F. O. STRATTON

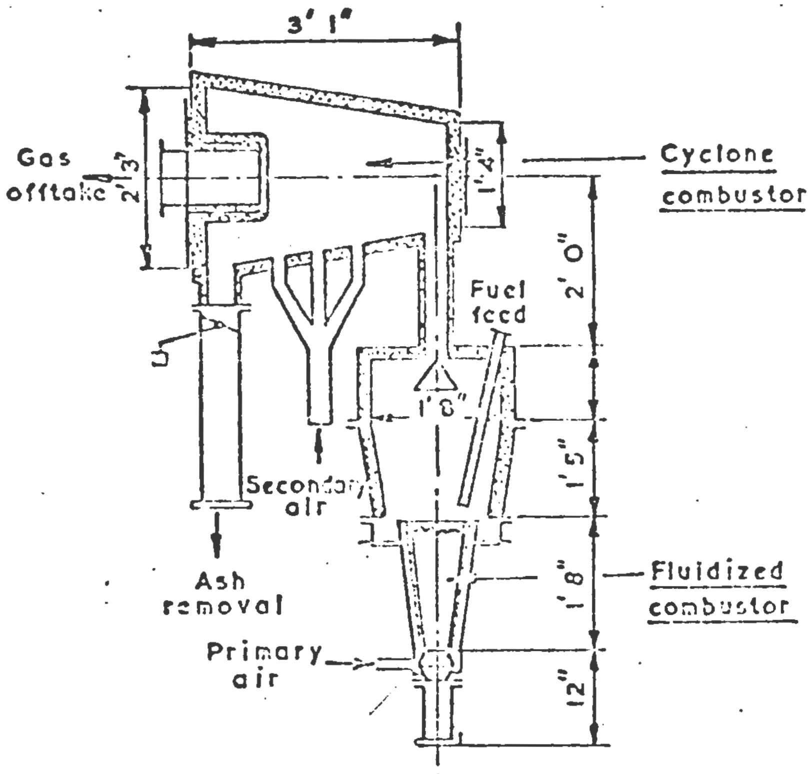


Figure C-21 - Institute of fuel research fluid bed combustion system

Moscow Power Institute

A two-stage furnace for a steam boiler was developed by the Moscow Power Institute. A fluid bed unit was combined with the combustion chamber of the steam boiler (Figure C-22). Combustible gas and particles escaping from the fluid bed would be burned in the combustion chamber. Any coarse particles would fall out of the combustion chamber and complete combustion in the ash pit.

Stouff

Stouff[6] proposed an agglomerating bed as a means for burning untreated coal fines. Coal is introduced into the cone-shaped bed at any level and the air is injected at the base of the cone. A diagram of the boiler system is shown in Figure C-23. Carryover from the bed is large, and excessive losses are avoided by recycling fines back to the bed.

Yokoyama

Okaniwa and Suzuki[7] have proposed a p.f.-dilute phase fluid bed system for burning low-grade coals available in Japan (Figure C-24). Coal sized less than 18 mesh is fired through the p.f. burners along the water walls. Particles too large to burn in the gas fall into the fluidized bed in the cone-shaped bottom of the boiler. Approximately 80% of the air is blown upwards through the base of the bed. The temperature in the fluid bed is 1550 -1650°F and the temperature at the burners is near 2000-2200°F.

Additional Concepts

Novotny[8] describes early spouting beds which were proposed for fluid bed combustion. A. A. Shershniev constructed a spouted bed system in 1927 for the combustion of peat. Two beds were located adjacent to each other such that particles could be blown from the first bed

Dwg. 861A447

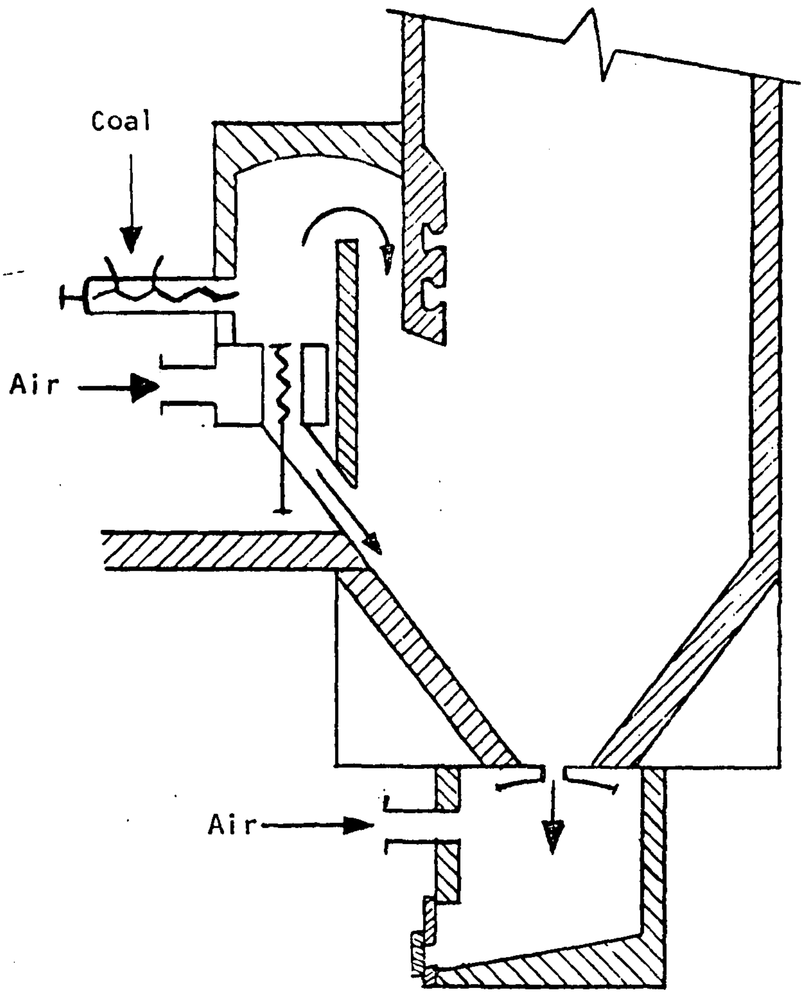


Fig. C-22- Two-stage furnace for steam boiler

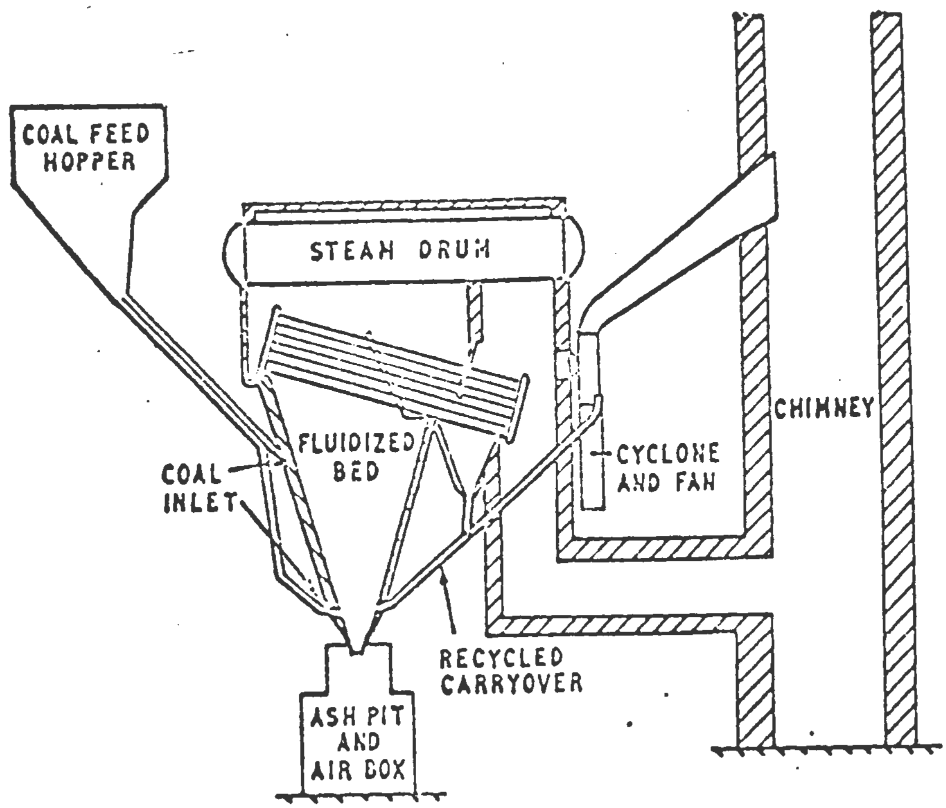


FIGURE C-23-STOUFF FLUID BED BOILER SYSTEM

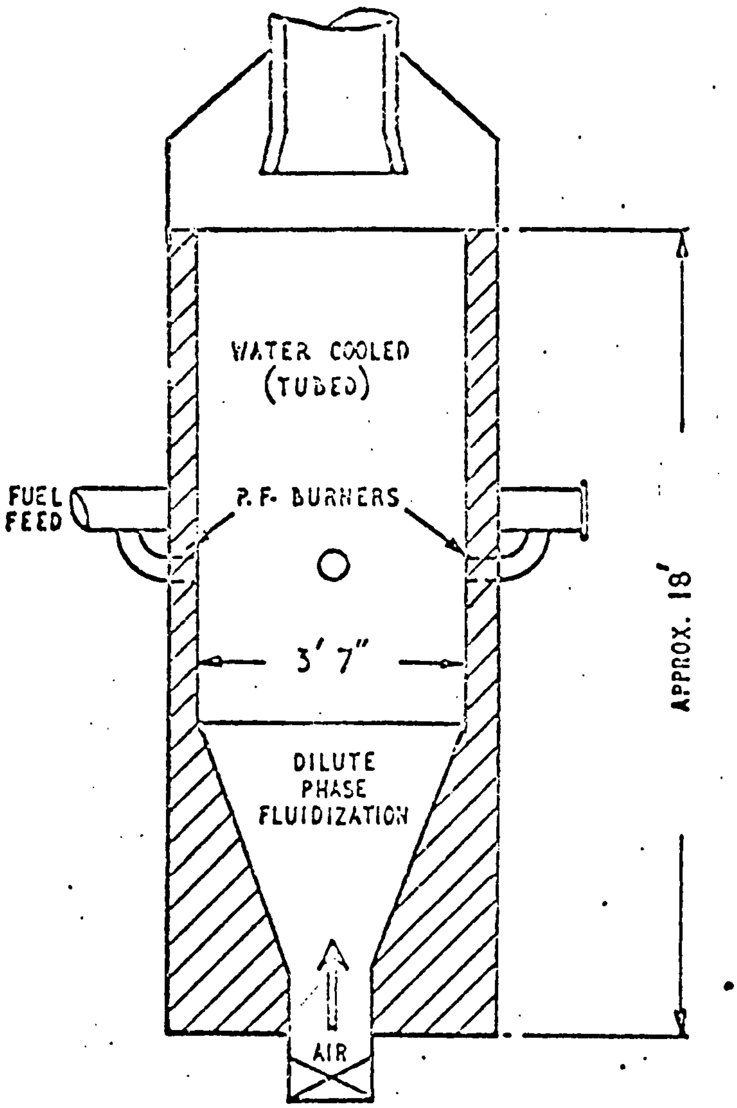


FIGURE C-24 - YOKOYAMA EXPERIMENTAL FLUID BED BOILER SYSTEM and thus the second bed helped to extend the residence time of fuel particles in the combustion chamber. Szikla-Rosinek also developed a spouting bed design. The design is complex and did not find practical application. A spouting bed design by K. Stousse is reported which attempts to solve the problem of unburned carbon by a two-stage furnace with recirculation of the fines to the fluid bed.

Several additional concepts have been proposed in recent patents. A list of patents which cover proposed fluidized bed boiler concepts is presented in Table C-1.

TABLE C-1

STATE-OF-THE-ART U. S. PATENT SEARCH: FLUID BED COMBUSTION AND COAL GASIFICATION

Patent No.	Date Patented	Title	Inventor	Assignor To
2,842,102	7/8/58	Steam Generation	H. J. Blaskowski	Combustion Eng. Inc.
2,976,853	3/28/61	Steam Generation	A. T. Hunter, R. C. Patterson, E. C. Lewis	" " "
2,983,259	5/9/61	Method & Apparatus of Steam Generation	E. C. White	" " "
2,997,031	8/22/61	Method of Heating and Generating Steam	R. C. Ulmer	" " "
3,048,153	8/7/62	Vapor Generator	R. F. Abrahamsen	" " "
3,101,697	8/27/63	Steam Generation	A. T. Hunter	" " "
3,119,378	1/28/64	Steam Generation	L. J. Marshall	" " "
3,431,892	3/11/69	Process and Apparatus for Combustion and Heat Recovery in Fluidized Beds	A. A. Godel	CIPA (in Switzerland)
3,387,590	6/11/68	System for Regulating the Total Heat Input in a Burning Fluidized Bed Heat Exchanger or Boiler	J. W. Bishop	U.S.A.
2,884,373	4/28/59	Method and Apparatus for Heating Fluids	B. E. Bailey	Esso R&E
2,884,303	4/28/59	High Temp. Burning of Carbonaceous Fuels	W. J. Metrailer	Esso R&E
2,973,251	2/28/61	Heat Transfer Apparatus	M. B. Leland, T. S. Sprague	Babcock & Wilcox
2,997,286	8/22/61	Fluid Bed Furnace and Process	G. Friese	Metallgesellschaft Aktiengesellschaft (Germany)
3,119,379	1/28/64	Apparatus for Combustion of Fuels	M. P. Sweeney	M. P. Sweeney
3,355,249	11/28/67	Producing Hydrogen and Power	A. M. Squires	A. M. Squires
3,276,203	10/4/66	Top Heat Power Cycles	A. M. Squires	A. M. Squires

TABLE C-1, continued

Patent No.	Date Patented	Title	Inventor	Assignor To
Re 24,328	6/11/57	Conversion of Hydrocarbonaceous Fuels into Synthesis Gas	L. J. Montclair, L. P. Gaucher	The Texas Company
3,004,839	10/17/61	Gasification of Carbonaceous Fuels	E. L. Tornquist	Northern Ill. Gas Co.
2,607,666	8/19/52	Apparatus for Treating Carbonaceous Solids	H. Z. Martin	Standard Oil
2,665,200	1/5/54	Process for the Gasification of Solid Carbonaceous Materials	M. Kwauk	HRI
2,671,015	3/2/54	Gasification of Carbonaceous Materials	R. J. Morley	ICI (England)
2,674,524	4/6/54	Process for the Preparation of CO & H_2	P. W. Garbo	HRI
2,686,113	8/10/54	Process of Promoting Chemical Reactions	W. W. Odell	W. W. Odell
2,689,787	9/21/54	Volatile Fuel Production and Apparatus Therefor	H. J. Ogorzaly, C. W. Tyson	Standard Oil
2,694,623	11/16/54	Process for Enrichment of Water Gas	A. B. Welty, S. B. Sweetser	Standard Oil
2,700,592	1/25/55	Method of Carrying Out Endothermic Reactions Under Fluidizing Conditions	T. D. Heath	Dorr Company
2,700,599	1/25/55	Gasification of Solid Carbonaceous Materials	J. C. Kalbach	HRI
2.705,672	4/5/55	Manufacture of Water Gas	E. Gorin	Consolidation Coal Co.
2,729,552	1/3/56	Process of Contacting Gasiform Carbonaceous Solids	K. J. Nelson, E. J. Gornowski	Esso R&E
2,741,549	4/10/56	Conversion of Carbonaceous Solids into Volatile Products	F. R. Russell	Esso R&E
2,776,879	1/8/57	Gasification of Solid Carbonaceous Fuel	W. Grumz	HRI
2,794,725	6/4/57	Manufacture of Gas Mixtures Containing CO and H_2	W. G. Scharmann	Esso R&E

TABLE C-1, continued

Patent No.	Date Patented	Title	Inventor	Assignor To
2,866,696	12/30/58	Process for the Gasification of Granulated Fluidized Bed of Carbonaceous Material, Over Moving, Sloping, Horizontal, Continuous Grate	A. Godel	CIPA
2,868,631	1/13/59	Gasification Process	H. N. Woebcke	HRI
2,906,608	9/29/59	Apparatus for Dense Phase Fluidization	L. J. R. Jequier Van de Putte	Inventors
2,911,293	11/3/59	Production of Gas	P. S. Viles	Esso R&E
2,985,515	5/23/61	Fluidized Solids Contacting System	D. L. McKinley	Union Carbide Corp.
3,086,853	4/23/63	Method of Gasifying Combustible Material in a Fluidized Bed	A. R. L. Brandberg	Swedish Co.
3,311,460	3/28/67	Method for Gasification of Carbonaceous Materials	H. H. Stotter, G. B. Farkas, P. C. Keith	HRI
3,322,521	5/30/67	Process & Apparatus for the Gasification of Ash-Containing Fuel	R. G. Cockerham	The Gas Council (England)
2,440,940	5/4/48	Gas Producer	A. L. Galusha	McDowell-Wellman
3,454,382	7/8/69	Two-Stage Type Gas Producer	G. M. Hamilton	McDowell-Wellman
3,433,859	3/18/69	Process for the Preparation of Hardened, Dense Heat Transfer Medium	T. E. Barr	McDowell-Wellman
3,226,212	12/28/65	Apparatus for the Production of Combustible Gases From Solid Carbonaceous Materials	T. E. Barr	McDowell-Wellman
3,034,776	5/15/62	Rotary Furnace	Hannenberger, et al.	Lurgi

REFERENCES

1. Physics Departmental Memorandum No: 152, National Coal Board, Coal Research Establishment.

2. Godel, A. A.; French Patent No. 1,092,540.

3. Proceedings of the First International Conference on Fluidized Bed Combustion, NAPCA, Oxford, Ohio (Nov. 1968).

4. Godel, A. A.; U. S. Patent 3,431,892.

5. Stratton, J. F. O.; Power 68, 486 (Sept. 1928).

6. Stouff, M. L.; Paper presented at the "Journees de la Combustion des Combustibles et pulverises", (1957).

7. Keiji, O. and J. Suzuki; J. Fuel Soc., Japan, 38, 429 (1959).

8. Novotny, P.; Monograph No. 14, Institute For Fuel Research.

9. Ehrlich, S., E. B. Robison, J. S. Gordon, J. W. Bishop,; paper presented at AIChE 69th National Meeting, Cincinnati, Ohio (1971).

APPENDIX D

INDUSTRIAL BOILER DESIGN REPORT

Prepared by Erie City Energy Division,
Zurn Industries, Inc.

Authors

Robert V. Seibel
Richard E. Winschel

STUDY OF FLUIDIZED BED COMBUSTION

- - INDUSTRIAL STEAM GENERATOR APPLICATION - -

INDEX

Abstract:

Fluidized Bed Combustion studies were continued by applying latest state of the art knowledge to develop a useful industrial steam generator. A study of the market (reported previously) pointed out the useful capacity, steam pressure, and, steam temperature. A commercial design was developed from an understanding of user's requisites and economic factors. Fuel handling and steam generator design was held constant in the development of two different systems to control atmospheric emission. A sample proposal in response to an assumed specification was developed. It was concluded that the cost factors appear competitive for coal burning applications when compared to present techniques.

Introduction:

Fluidized Bed Combustion techniques have been studied for some time both in the United States and in foreign countries. The investigations were not, however, extensions of fluidized bed processing which has had a long history of commercial chemical processing applications. At the outset, the studies in the United States were to develop smaller size, less expensive coal fired steam generators. The public feeling about environmental factors lead to the study of control of pollution control. Now controlling the emission of combustion products such as sulfur oxides and oxides of nitrogen appears to be the more valid reason for studying fluidized bed combustion techniques in the United States.

Pilot plant and laboratory studies formed the base of the development. No commercial sized units for pollution control were or are readily available since investment costs are prohibitive. Industrial steam generator users rely on high availability, low maintenance, and proven designs for new or replacement units. Capital investment for "non-profit making" equipment, that is not fully proven and without nearly full size pilot units is not done. Whether further development would prove fruitful was a question left to be answered by someone other than the users. Government authorities sponsored this and other studies where the worth of further pilot studies is being developed by using the background data already available and by looking into the future at steam generator markets.

This study, only a part of a much broader one, was then to produce a commercial design for the industrial steam generator markets. Economic factors would be developed and, also, as areas of further development would be made more clear, development costs could be outlined and pollution control effects could be pointed out.

Erie City Energy Division of Zurn Industries was authorized as a sub-contractor to Westinghouse to utilize information sources about the steam generator markets and to develop markets prediction as to a useful size steam generator for future markets. From the market information, a commercial design for a steam generator would be developed, using a procedure whereby the state of art would be defined by Westinghouse and integrated into commercial designs. Economic considerations such as capital costs, optimum design, and operating costs would be developed parallel with the technical design. Work would be done by Erie City Marketing Computer and Engineering Forces.

The following is a report on the work that was done to produce a steam generator design that tests the market and the sellability of a Fluidized Bed Combustion Steam Generator for industrial application.

It is assumed that throughout this report the reader has an intimate knowledge of fludized bed technique and terminology.

GENERAL DISCUSSION

As the work with fluidized bed combustion steam generators progressed two systems began to emerge. Both systems were designed to consume coal at the same combustion conditions, but each was different from the other in regard to the control of atmospheric emission of sulfur oxide gas. The steam generator, nevertheless, remained the same in both cases. It is important to bring this out at this point since references will be made to each system throughout the report.

One system utilizes lime, calcium oxide, to absorb sulfur dioxide formed in the combustion reaction as soon as the sulfur component in the fuel is oxidized. This is the "Dry solids" system. The term comes from the fact that the reacted calcium oxide is taken out of contact with the flue gas in a dry state as a powder. The other system, designed to have equivalent performance in the control of sulfur emissions, is called the wet scrubber system. In this system limestone is calcined in the fluidized bed and is mixed with water to form a slurry which contacts the flue gas. The limestone slurry has an affinity for sulfur oxide gases and they are removed from the flue gas. Although in both systems, the same steam generator is used, there are fundamental differences in the auxiliary equipment needed for the system. These differences are discussed in more detail later.

The market survey and study of the technical aspects points to a steam generation capacity of 250,000#/hr. of steam at a steam pressure of 600 PSIG, and steam temperature of 750° F. There appears to be a growing demand for units like this and in 1980 that demand will approach 40% of the industrial steam generator market. Refer to Appendix XII. These steam conditions of temperature and pressure imply a steam generator where steam will be used to generate power in a turbo generator. The steam after expansion to a lower pressure, normally 150 PSIG, is made available to the process applications such as heating. In other cases, when processing power requirements are high, the auxiliary equipment drivers may be turbines exhausting to the process steam header. With those conditions in mind and using high sulfur coal as the basic fuel, work was centered around finding the most economical arrangement of steam generator components, and the auxiliary equipment needed to complement the steam generator system. All significant components were subjected to individual study. These are as follows:

- Fuel System
- Limestone Feed System
- Steam Generator, including the superheater and saturated sections
- Economizer
- Primary ash collector
- Ash handling systems
- Controls and instrumentation
- Electrostatic precipitator
- Wet scrubber and Limestone Slurry Dewatering System

In addition to this, the system considerations such as heat and material balances were developed. Cost data were developed as the design was produced.

This report will be divided according to the system components mentioned above. Under each section will be a discussion of the design parameters, goals for the designs and what appeared to be the best arrangement according to the combination of "State of the Art" and present economic factors. Appendices covering specialized areas of study are attached.

FUEL SYSTEM

In recounting the design work on the whole project, the most difficult problems that must be solved in developing Fluidized Bed Combustion steam generators is that involved with materials handling. This includes the preparation, the handling, and the injection of coal and limestone.

In all fuel feeding systems, several factors are vital and must be accounted for. These are:

- Control of the fuel feed rate.
- Constant and preferably negligible time transients from inlet to the discharge of the system.
- Wear resistance and easy maintenance.
- Negligible power requirements.

All of the above must be solved in the design for Fluidized Bed Combustion Steam Generators.

Some of the problems presented by the fluidized bed combustion are well known in utility practice. For example, overcoming the pressure interface of 40 to 50 inches water gauge to allow injection of coal into the fluidized bed combustion chamber. Direct firing of pulverized coal and both the indirect and direct-fired cyclone furnaces utilize the head of coal feeding from the bunker to prevent significant loss of pressurizing air admitted to the fuel feeding equipment. Also, handling coal crushed to 4 mesh size as with indirect fired cyclone furnaces, in bunkers, bulk conveyors, and rate feeders has been well developed through experience in utility practice. Commercial specialized equipment to do this job is readily available. In fact, rate feeders have been developed that lend themselves to precise gravimetric rate control. Refer to Appendix IX.

The one relatively unique problem presented by the present thinking about fuel feeding is the multiplicity of fuel feed points. For example, in thinking of an atmospheric fluidized bed boiler for 250,000#/hr. steam capacity, approximately 400 sq. ft. of grid area will be required. If fuel injection must be made at the rate of one point for 10 square feet of grid area, there would have to be 40 separate feed points. This does not in itself present a problem since fuel piping and bed injection techniques are not particularly restrictive. The most difficult problem in this, is the requirement of dividing one stream of the fuel into 40 uniform divisions. (It would be quickly seen that there would be little relief if 40 fuel rate feeders were considered not only because of cost, but also, since exact parallel control that is required would be an impractical case.)

To include present practice and available equipment with a suggestion to handle new areas certain assumptions were made:

- Coal would be prepared in a bulk-type crusher upstream of the bunker in an indirect system, 100% less than 1/4" sizing was anticipated.
- One fuel rate feeder would be used.
- The air pressure interface would be made at the bunker with sealing air fed to the fuel rate-feeder.
- There would be four sections of the bed arranged so that one section could be shut off, independently.
- There would be eight injection points in each of the four bed sections. Air would transport the coal underneath the grid to the injection point.
- The coal injector nozzles would be removable for repair and replacement even with the boiler remaining in service. (But with that bed section shut down.)

To develop these ideas, arrangement drawings were made of the coal handling and boiler system. In the fuel system, after delivery to the plant, the coal is crushed at the rate of 2 to 3 times the firing rate and passed to a single surge bunker feeding the boiler. Bunker capacity would be set to handle several hours firing to allow efficient crusher and conveyor operation, and, to allow adequate periods of maintenance for the coal crusher and handling equipment without interrupting boiler operation.

From the bunker, coal is fed to the system using a single belt-type, gravimetric controlled rate feeder. From the rate feeder, the coal enters a two stage splitting system. In the first stage, the single coal stream is geometrically divided into four streams each of which can be shutoff for load control or for downstream maintenance. Each of the four streams would feed a separate section of the bed.

In the second stage, coal is split into several more streams, In the final design, for a maximum of 10 sq. ft. per injector point, there would be 8 streams leaving the second stage splitter.

From the second stage splitter coal drops into an air-swept horizontal pipe which carries the coal underneath the grid to the injection point. For the second stage splitter or distributor, it was proposed to have a vibrating table feeder to distribute the coal stream from the first stage splitter over the width of a symmetrical geometric shape with eight outlets.

At the injection point location, a special grid casting receives the nozzle at the end of the air transport pipe.

The nozzle is arranged so that a compressed air aspirator allows the removal of individual nozzles while the steam generator remains in service. This can be done by reducing the steam generator load to where one bed section may be taken out of service.

There was some concern about wear and a jet effect from injecting the coal air stream into the bed. The first nozzles proposed utilized a vane principal to direct the coal air steam normal to the axis of the nozzle which was horizontal. It was thought that wear and jet effects could reduce the effectiveness of this design. The final design proposes to have the nozzles inclined at a 30° slope into the fluidized bed chamber with a "straight-in" nozzle design.

Several study drawings were made of the coal feeding system (see drawing reference list in Appendix X.) Drawing FBB-28, in the figures section, pertains to the final design and gives an outline of the fuel system including the component parts.

LIMESTONE FEEDER

As noted on the drawing FBB-28 the limestone feeder was selected as a variable amplitude vibrating table feeder. The limestone hopper is a pressure seal type similar to the coal bunker. The limestone will discharge at the discharge end of the coal rate feeder where it mixes with the coal flowing to the fluidized bed.

In both the dry solids and the wet systems, the limestone feed rate is not changed instantaneously with the fuel feed rate. Instead, limestone feed is adjusted to optimize sulfur oxide gas removal. In both systems, either dry solids or wet scrubber, an increased limestone feed rate will reduce the sulfur dioxide concentration in the stack gas.

STEAM GENERATOR

The steam generator design is more easily visualized if it is considered to be of four components:

- The fluidized bed combustion chamber.
- The carbon burn up cell.
- The superheater.
- The convection pass.

(For reference the four steam generator components are shown on drawings FBB-19 and 20 attached).

STEAM GENERATOR - FLUIDIZED BED COMBUSTION CHAMBER

At the outset of the study, two different fluid bed chamber designs looked attractive. Each was developed to preliminary sizing where there would be equivalent bed temperatures for the 250,000#/hr. steam generator. The two were differentiated by the arrangement of heating surface submerged in the fluid bed. They were: a vertical tube design; and, a horizontal tube design. (The horizontal tube configuration is indicated on the figures attached. Study drawings FBB-1, 3, 4 and 6 presenting an outline of the vertical tube configuration. These study drawings are not presented in this report.)

Although the functional characteristics of the vertical tube design were questioned, not enough direct information was available to eliminate it as a valid concept. On the other hand, experimental data have been gathered on the horizontal tube designs and there is confidence that full scale performance would be predictable.

In that the two designs had different construction it was decided to submit them to pricing competition. Each of the designs for the fluidized bed chamber provided for shop assembly of two separate fundamental components - - the fluid bed chamber and the boiler convection zone. In both cases, the two parts would be mated in the field with a minimum of butt-welded pipe connections.

For the price competition for each design, the fluid bed chamber costs were compared on an equal basis:

- Insulation and lagging were not included.
- The fluid bed grid was not made a part of the cost.
- Air flow chambers and material handling systems were not considered.
- Field assembly costs were considered to be equal in both cases and were not included.
- Assembly costs to produce sub-assembly of tube arrangements were included.

On this basis, it was revealed that the horizontal tube design would have a very much lower selling price.

Price data showed the following comparative figures on the basis above:

	Equivalent Selling Price:	
	Vertical Tube Design	Horizontal Tube Design
Material	1.46	1.0
Labor (fabrication & assembly)	2.29	1.0
Total selling price (weighted as to the total cost of labor & material)	2.04	1.0

(The portion of the total steam generator included in the above was found to be about 20% of the total price.) Weights of material for each design were found to be equivalent with 117,000 pounds for the vertical tube and 102,000 pounds for the horizontal tube design. The number of welds required for the vertical tube was found to be 2,900 and only 770 for the horizontal tube design. The latter factor, proved to be the cause of the cost differential.

It was concluded by this analysis, that the horizontal submerged tube design would offer the better economic choice even though the above data do not reflect all of the compensating differences between the two designs. The horizontal tube design requires a recirculating pump to provide adequate water flow to cool the tubes submerged in the bed. Also, since there is no freeboard heating surface in the horizontal tube design, there would be more surface required in the boiler convection pass.

From the study of prices, then, it is felt that the vertical tube design should be largely eliminated from our work in preparing data concerning an atmospheric fluidized bed combustion steam generator design. A factor in this decision was plainly the fact that development work was concentrated on the horizontal tube designs.

As the bed design emerged utilizing the horizontal tube design, the design parameters of the bed were brought into better focus. To summarize, the design factors are:

- Grid heat release rate - - 1 x 10^6 Btu per hr. per sq. ft.
- Bed Depth - - 30 in (expanded)
- Bed temperature - - 1650 F in fuel bed - - 2000° F in carbon Burnup cell
- Heat transfer coefficient - - 50 BTU/hr. sq. ft. F
- Free volume space underneath the submerged surface
- 10% excess combustion air flow through fuel bed
- 50% excess combustion air flow through the carbon burnup cell

-Application of a particle splash screen

-Tube arrangements that minimize lane effects

To meet these criteria it appeared that the submerged horizontal tube surface must be composed of small diameter (1") tubes. Also, forced circulation to cool the submerged surface will be required. All materials of construction for the saturated heat absorbing surfaces and seals are carbon steel types.

Erie City Energy Division drawing FBB-19 attached, shows an arrangement of submerged surface, and particle splash screen using 1" diameter tubing. The spacing shown is to allow straight tubes where the 1" tubes penetrate the combustion chamber wall. The number of submerged tubes and splash screen tubes shown in FBB-19 are the maximum obtainable. Using parameters now assumed valid, there is a slight excess heat absorbing surface. As more becomes known about the bed heat transfer, the surface may be adjusted to provide operation as desired.

The fluidizing grid design and selection is a subject that still has not been well defined. The "State of the Art" designs all utilize nozzles which direct the fluidizing air either parallel to or against the grid plate. While these are not particularly restrictive mechanical problems, it is felt that castings with cored holes would be more economical and would provide for longer service life.

In working toward a commercial design some casting designs were developed and are available for review (refer to the study drawing listing in Appendix X.) Even though nozzles may be required, it was contemplated they would be applied to castings which would be contoured to closely fit the fluidized bed floor tubes. These castings would be bolted providing cooling in addition to the affect of the fluidizing air flow.

One facet of the fuel injection point spacing shown on FBB-20 in the fuel burning grid is the clearance from one transport pipe to the next. The arrangement shown is to allow all nozzles to enter from one side of steam generator.

STEAM GENERATOR - CARBON BURN UP CELL

On drawing FBB-20 attached, there is a sectional view through each component of the steam generator. In the case of the carbon burn up cell, it should be noted that the gas is collected at one end of the combustion chamber and directed through the carbon burn up cell where carbon rich ash is injected and burned with a combustion air stream that is part of the combustion air stream. Some heat transfer surface is installed in the carbon burn up cell to control the bed temperature.

The location of the carbon burn up cell whether up stream or downstream of the primary dust collector is a matter of much discussion, and review. In the case of the designs presented, here it was felt that the boiler design should take precedence. It is seen upon close review that the bed temperature control is part of the boiler steam generating circuits. To place the refiring system upstream of the primary collector requires the separation of the boiler circuits. This could very likely lead to having a third boiler component and would at least lead to higher equipment costs.

There is, of course, the likelihood of insufficient oxidation of unburned carbon in the arrangement selected which is a "once through" system. It is thought that the arrangement selected is also better since recycling of ash is not a feature. It must be realized that comprehensive data do not exist to outline the carbon burn-up cell location and operation. At this time it is felt that both higher fluid bed temperatures and higher relative air flows are returned to achieve the desired results. Bed temperatures are thought to be 1900 to 2000°F. with 50% excess combustion air supplied to the carbon refiring system.

The need for turndown led to sectionalizing the carbon burn up cell into four sections which would generally be operated parallel to the fuel burning bed sections. Ash would be air transported to the carbon burn-up cell and injected similar to the fuel. Combustion air and ash injection would be operated in parallel for each carbon burn-up cell segment. In addition, equipment and instrumentation would be installed to bias air flow to allow temperature control. Increased air flow will lower the bed temperature.

STEAM GENERATOR - SUPERHEATER

For the capacity, steam pressure and temperature conditions selected, the superheater heat absorption represents only 15% of the heat output in the steam. It should be noted that this is less than the single bed section capacity of 25% of the total fuel burning capacity. This fact and that economical design does not allow fuel modulation of a single bed section, led to the decision to utilize a "conventional" superheater design.

In the utility Fluidized Bed Combustion Steam Generator, a unit of the fuel burning bed can be large enough to have individual control of firing rate and, thus, be devoting the possible heat absorption to superheating, a constant final steam temperature can be obtained. Superheater tube excessive metal temperature protection and start-up problems are easily handled when an entire bed unit can be used for superheating. The industrial steam generator case is more restrictive because of size and heat input.

To follow conventional superheater design for the industrial boiler application, no superheater surface is included in the bed. It is arranged so that the flue gas from the fuel burning beds is gathered together and passed through the superheater and "boiler bank" or convection pass.

It is anticipated that the superheated steam temperature would be "uncontrolled". This is to say that it is designed for full load conditions with no means to "trim" or control steam temperature. There would be no need for special means to protect against exceeding the superheater tube metal temperature capability since there would be little or no disparity between steam flow and heat absorption.

The superheater would be arranged with vertical headers outside the outer wall of the convection zone. Tubes would be arranged to have counterflow steam to flue gas with the tubes passing back and forth normal to gas flow. Drawing FBB20 attached, depicts this arrangement of superheater surface.

To balance the factors of heat absorbing surface, physical limitations and pressure drop, 2-1/2" O.D. tubing was chosen. A vertical spacing normal to gas flow of 6-1/2" was used. A parallel or back spacing of 4" was used. The two basic design parameters which are the full load superheater pressure drop (27 PSI) and also the metal temperature consideration, will allow the use of carbon steel tubing.

The flue gas temperature levels in the superheater section dictate that the largest component of heat absorption is convection with only a slight radiant characteristics. In such an arrangement of superheater, it is commonly seen that the steam temperature is directly related to load. Part of this effect is the increase of the initial gas temperature. Advantageously, the fluidized bed steam generator offers some means to control this characteristic over the load range. By selecting bed sections either closer to, (or farther away,) from the superheater, it is possible to affect the superheater inlet gas temperature and to adjust the steam temperature accordingly.

In regard to turndown, if excess air is used for bed temperature control, the steam temperature will be effected. If excess air is increased, the steam temperature will increase. A de-superheater in the boiler system to reject heat from the stream will be needed in this case. Unlike utility boiler practice, however, the steam temperature trimming system would not be designed to extend the control range of steam temperature but only to adjust the unbalanced or transient operation.

The convectional type superheater would: reduce boiler design pressure since some means to protect superheater tube metals if they were located in the bed is not needed; it would provide for easy startup; and it would allow greater safety of the materials at lower loads. In light of these factors, it is the better choice for the industrial steam generator applications.

STEAM GENERATOR - CONVECTION PASS

To recover heat from the flue gas to a level where only one heat trap is needed, the flue gases leaving the superheater are passed through the convective section of the steam generator. In convection zone, 2" tubes are arranged on 4-1/2" spaces. They are part of the steam generating heat absorption.

STEAM GENERATOR - WATER CIRCULATION SYSTEM

In the steam generator, a circulating pump is applied to supply flow through the heating surface that is submerged in the fluid bed. The walls of the fluidized bed combustion chamber and all other surfaces are cooled by "natural circulation" water recirculation. Drawing FBB-32 attached shows a schematic view of the circulation system. No valves are used to parallel flow through various circuits.

Circulation requirements and parallel path pressure drops are adjusted to maintain safe operation under all operation conditions. For the proposed design, this has been studied by introducing elements of pressure drop, such as, "underdrilling" the header entries for tubes, selecting pipe sizes to introduce or to eliminate pressure drop influences. For example, in "underdrilling" a header the diameter of the hole that is drilled through the counter bore that receives the tube in the header is less than the internal diameter of the tube. An orifice results.

Since the system resistance is not great, pipe sizing to insure equal distribution must be done carefully. Also, piping links must create a good distribution system.

ECONOMIZER

A study of heat traps was made comparing tubular and regenerative air heaters and, bare tube and extended surface tube economizers. By cost comparisons it would found that the extended surface economizer was found to be most economical. (The results are inline with the present steam generator equipment purchasing practices.)

Although some work on Fluidized Bed Combustion alludes to the necessity of two heat traps, it developed through Erie City work that there is sufficient convection surface in the steam generating system to obtain high thermal efficiency. As a result of this finding one heat trap is the economic choice. See Appendix XI for a discussion of the heat trap application.

PRIMARY ASH COLLECTOR:

To obtain the maximum thermal efficiency by reducing unburned combustibles to low ends it is necessary to refire carbon rich ash that is elutriated from the fuel burning bed sections. The system chosen to do this is a cluster of cyclone dust collectors downstream of the fuel burning fluidized bed sections. The cyclones are arranged to handle all flue gas from the bed. They are refractory lined and internally insulated. Collection efficiency will be 90% of the ash, carried out of the fuel burning beds. Sectionalizing dampers will be required to maintain collection at reduced loads.

Ash removed from the gas stream is transported to the carbon burn-up cell where in another fluid bed the carbon is refired at relatively high rates of combustion air and at bed temperatures higher than the fuel burning sections.

It is anticipated that the amount of ash will be related to load. Because of this the carbon burn-up is compartmented to allow shut down of sections for turndown. Shown on the schematic drawings FBB-30 and 31, hoppers are linked in series to the ash transport system allowing independent operation of the dust collector sections and carbon burnup cell.

ASH HANDLING SYSTEMS:

There are two fundamental ash handling systems. One for carbon rich ash receiving ash and carbon from the primary cyclone collector and the other transferring ash from the bed sections and from an ash pickup point underneath the superheater. In the case of the dry solids system, the electrostatic precipitator ash is also discharged to the ash system.

In all cases a pneumatic ash handling system is contemplated. The system would be pressurized and the ash feeders would be lock hopper types to feed across the pressure interface. The lock hopper feeders offer an advantage in that feed rate can be adjusted to compensate for operational adjustments.

Lock Hoppers and ash feeders are available and are proven equipment. See Appendix IX for information.

CONTROLS AND INSTRUMENTATION:

There are three basic divisions within the overall control system for the fluidized bed combustion boiler. The combustion controls, the feedwater flow control and steam temperature control. The basic principles of the control system are the same for the fluidized bed combustion unit as for any sub-critical recirculating type boiler. Nomenclature and the detailed control loops are slightly different in the case of the Fluidized bed system, however.

The following will outline the functional aspects of the three different divisions of the control system (Reference drawing FBB-27).

INSTRUMENTS & CONTROLS - COMBUSTION CONTROL SYSTEM

The fuel burning fluidized bed is sectionalized to allow shutting down one or more sections for turndown and for on-line maintenance purposes. The air flow to each section may be modulated independently of other sections. Fuel flow, however, is split equally to each operating section and may not be biased to one bed section.

Combustion air is fed to a common plenum by the forced draft fan with the plenum air pressure being maintained by the inlet vanes. Combustion air flow to each bed section and to each section of the carbon burn-up cell is controlled by individual dampers. Flow to each section is measured and totalized.

Bed temperature is measured in each bed and carbon burn-up cell section. Bed temperature is a factor in the air flow control and also is a limit in starting up and shutting down individual bed sections.

Steam header pressure is the load sensitive control element used to initiate fuel feed.

In reference to the functional control schematic, it is seen that the plenum chamber pressure transmitter signal is compared to a summed signal originating from other control functions representing air pressure demand. The output signal from the error computer is sent to a proportional plus reset controller to a hand-auto station and to the forced draft fan vane positioner.

The fuel feed signal is developed by comparing the header pressure signal to a set point. The error signal is sent through a proportional plus reset controller to the boiler master hand-auto station to a multiplier receiving a signal from the fuel air ratio setter to the fuel feeder hand-auto station to the fuel feeder speed controller.

Multiple control loops are required for the air flow control system. One loop is shown on the schematic drawing and is typical for all bed and carbon burn up cell sections.

In an air flow loop the air flow signal is compared to a demand signal from the master and from bed temperature. The master signal is modified by a multiplier indicating the number of bed sections in service. (The multiplier is required to indicate the extent of fuel splitting--to cite an example, for the same fuel feed at 50% load 2 or 3 bed sections may receive the fuel. A different air flow relationship to master signal is required in each case). The bed temperature signal is converted to pneumatic and compared to a set point. The error is sent to a proportional plus reset controller where the output is alarmed and sent to a summing relay along with the master signal.

When there is an error between the air demand and actual air flow the error signal is sent to the proportional plus reset controller, the air flow hand-auto station, and to the damper drive positioner.

For alarm purposes, if there is a significant difference between the master and air flow signal, an alarm will be signalled.

For bed starting up a sub-loop including a transfer switch and time delay volume chamber is used to control the rate of air flow introduction and bed fluidization. The system schematic shows that the output to the final error computer is limited by a rate system initiated from the transfer switch.

The two demand signals for the plenum chamber pressure control system are developed from the master signal air flow multiplier and the output to the damper drives. To compensate quickly for a change in the number of operating beds, the master air flow multiplier output enters the summing relay. Also, the highest air flow damper signal is sent through a proportional reset controller to the same summing relay.

By using the highest damper signal sufficient plenum pressure is obtained to have sufficient air flow capability to each bed section.

FEEDWATER FLOW CONTROL SYSTEM

The feedwater flow control or steam drum level control is a three element type. Feedwater and steam flow signals are compared. If a difference occurs, a signal is sent to a rate response system and to the feedwater flow control. The steam drum level is compared to a set point. If a difference exists, a signal is set through a summing relay to the feedwater control system. The signal from the flow and from the level is compared and sent to a proportional reset controller to the feedwater hand-auto station and then on to the flow control valve positioner.

STEAM TEMPERATURE CONTROL SYSTEM:

In light of varying bed operating conditions and extending the load range of constant superheater steam temperature a spray attemperator will be used.

The principal signal will be the steam flow in a two element system. The outlet steam temperature signal is converted to pneumatic and compared to a set point. The error signal is sent through a proportional plus reset controller and summed with the steam flow signal. The combined signal is passed through the attemperator hand-auto station to the spray flow control valve.

ELECTROSTATIC PRECIPITATOR:

In the dry solids system particulare emission is controlled by an electrostatic precipitator. It is designed for low sulfur gas conditions and will have greater than 99% collection efficiency. See Appendix IX for information.

WET SCRUBBER-SLURRY DEWATERING:

In the wet scrubber system, the contacting stages scrub the flue gas free from particulates. At the same time, sulfur oxide gases are absorbed by the limestone slurry passed through the scrubber. See Appendix IX for more information.

The wet scrubber also requires a slurry dewatering system. This includes a thickener and a vacuum filter and a material handling system. Drawing FBB-36 presents a schematic view of the equipment required.

DISCUSSION OF THE TWO SYSTEMS

The arrangement drawing, FBB-23, is useful in establishing the floor space and volume required for the dry solids system. It should be noted that all components shown are on bottom supported from the grade elevation.

Drawing FBB-34 shows the heat and material balance for the dry solids system. Appendix VI presents the relationship used to establish the material balance.

DRY SOLIDS SYSTEM:

At the outset of this study of fluidized bed combustion application, the principal means to control sulfur dioxide emission was thought to be one where the fuel was burned in an environment of calcined limestone. As soon as the sulfur was oxidized it would be absorbed by sulfation of the limestone. At the bed temperature of 1650° F. this reaction was feasible although the kinetics require several times the stoichimetric amount of limestone. In fact, at the rate of 6 times the amount of sulfur in the coal, the amount of limestone required is nearly equivalent to the coal feed.

This amount of ash, for the purposes of this study, is said to be "once-through". That is, fresh limestone is fed continuously to control the sulfur emission. After it has reacted with sulfur in the bed and has been removed from the system, it is discarded. Some work is contemplated to develop regeneration techniques but the effects of limestone regeneration were not considered in this study.

For discussion purposes, three drawings attached describe the dry solids system. Drawing FBB-30 shows a flow diagram pointing out the components parts and a schematic system for auxiliary equipment. Drawing FBB-23 presents an arrangement of the equipment showing front, side and plan views.

Drawing FBB-34 presents a heat and material balance for the dry solids system.

To describe the various components and their function it is seen that all steam generator components and auxiliary equipment are shown on FBB-30. On this drawing it is seen that coal is crushed to the firing size, transferred to a surge hopper where it is metered into the fluid bed chamber along with the limestone which is rate fed into the system. Combustion air is forced through the fluidized bed grid to burn the coal. Flue gas passes through the primary dust collector. The flue gas is then cooled with superheater, convection pass and economizer prior to final particulate removal in the electrostatic precipitator. Carbon rich ash removed in the primary cyclones is burned in the carbon burn-up cell. Limestone which has collected sulfur from the coal is extracted from the bed and sent to the ash silo. This same ash handling system receives ash from the hoppers under the superheater and from the electrostatic precipitator. All components required are shown on this drawing including valves, dampers, etc.

WET SCRUBBER SYSTEM

As work progressed in this study, it became apparent that the control of and the feeding and material handling problems with the dry solids system could be somewhat alleviated if there was more efficient use of the limestone feed. To do this, it is proposed to calcine the limestone in the bed and then transfer it and make a lime slurry for a stack gas scrubber. This system requires 20% more than the stoichimetric amount of limestone.

Enough energy would be expended in the sulfur gas absorption system to remove particulates along with the absorption of sulfur gases.

Drawing FBB-31 presents a schematic flow diagram for the wet scrubber system showing the component parts. Since the fuel feeding system and the steam generator remains the same in both systems, only that equipment for the wet scrubber is shown in arrangement view on drawing FBB-29. In addition to the contacting stages of the wet scrubber FBB-29, drawing FBB-36 shows the components for the slurry dewatering system. A heat and material balance diagram for the wet scrubber system is shown on drawing FBB-35.

ECONOMIC FACTORS OF THE TWO SYSTEMS

It was brought out that two different systems apparently could emerge from further development work. The systems could be made to have equivalent stack emissions. If this is true, what is the relative merit of one versus the other?

Comparisons were made as to the cost and power requirements. In both these comparisons, the dry solids system is the better. See Appendix I for cost comparison. Appendix VIII shows power requirements.

The heat and material balance indicate a favorable efficiency for wet scrubber system. However, in reviewing capital costs, it is seen that the dry solids system is estimated to have appreciable less first cost. The dry solids system cost appears to be \$6.10 per pound per hour of steam generating capacity. The wet scrubber system is estimated to be \$7.40 or about 20% more. Moreover, the operating power requirements (a factor in how much energy conversion is available for use) greatly favors the dry solids system.

For the dry solids system, it is estimated that the auxiliary power represents 0.86% of the output; for the wet scrubber 1.21%.

In totalizing the higher efficiency and higher operating costs for the wet scrubber system and comparing this to the lower first cost but lower efficiency of the dry solids system, it appears that the operating cost of the wet scrubber would be about \$45,000 per year less. This then converts to about a 7 year payoff of the higher capital cost.

To the industrial steam generator user, the dry solids system would be favored as the economic choice and in a very important factor of flexibility of operation. The coal that was chosen is a restrictive case because of the high sulfur values. It is likely that sulfur values will vary. If this happens, and if the variation is in the direction of lesser sulfur, the amount of limestone used will be reduced and the economic factors become even more favorable. It is soon seen that the dry solids system has the capability of reacting to any sulfur variation in the fuel at the minimum expense.

An important consideration for the dry solids system is the stigma against visible stack discharge. The wet scrubber system will produce flue gas saturated with water vapor. A constant plume will be observed at the stack unless reheating is introduced adding another negative economic factor. For example, reheating the stack 100° F. will require nearly 10 million BTU per hour which more than offsets the higher efficiency savings of the wet scrubber system. In this case, that could be no recovery of the additional first cost, due to the lesser fuel flow.

Tabulating this information shows the following:

	DRY SOLIDS SYSTEM	WET SCRUBBER
System Efficiency, as pounds of coal burned per hour (Not including stack gas reheating)	26,650	25,900
Capital Costs, dollars/pound of steam generated	6.10	7.40
Auxiliary Power Requirements, as % of output	0.86	1.21
Material Handling:		
Limestone, flow, pounds/hr.	21,450	4,200
Ash Flow from system, pounds/hour	17,300	16,050
Stack Condition (without reheating)	Colorless	Saturated vapor plume
Stack Gas Reheating Heat flow, BTU/Hr.	Not required	10×10^6

In taking the above factors into account, it appears that the Dry Solids System is the better choice, economically. The one drawback not yet rationalized is the regeneration of the spent line and its effect of limestone consumption. If regeneration is developed and used, there would be another factor favoring the Dry Solids System.

CONCLUSIONS:

From the study of Fluidized Bed Combustion Steam Generator design and market factors for industrial applications, it is concluded that:

1. A useful industrial steam generator for future markets will be one to generate 250,000#/Hr. of steam at 600 PSIG and 750° F. (See Appendix XII).

2. In designing a unit to meet the future markets, complete shop assembly cannot be achieved because of configuration and size requirements. Modular construction is feasible where components are assembled in shippable modules which are joined at the site.

3. Design problem areas are not serious enough to warrant fundamental or basic research. All problems noted are of a nature where pilot size units will allow development of the design criteria. Problem areas are:

    a. Fuel handling - stream splitting - and coal injection - distribution within bed - introduction of fuel at lighting off

    b. Fluidizing grid design - cooling factors and air flow requirements - erosion

    c. Heat transfer relationships of the submerged surface in the bed - design parameters - relationship to erosion - turndown - correlation to variables for unique design

    d. Heat transfer and reaction kinetics in the carbon burnup cell - design parameters

    e. Convection zone - erosion factors - ash deposition - cleanability

    f. Ash handling system - operating temperature factors - effective feed rate control - oxidation of carbon rich ash

4. Steam generator and auxiliary design can be along the lines of current practice where units are self-supporting off grade minimizing structural components and installation time.

5. The dry solids system design provides greater flexibility in economy since it can adjust to actual fuel criteria easier than the wet scrubber system. For example, the limestone feed rate is more easily adjusted to sulfur in the coal.

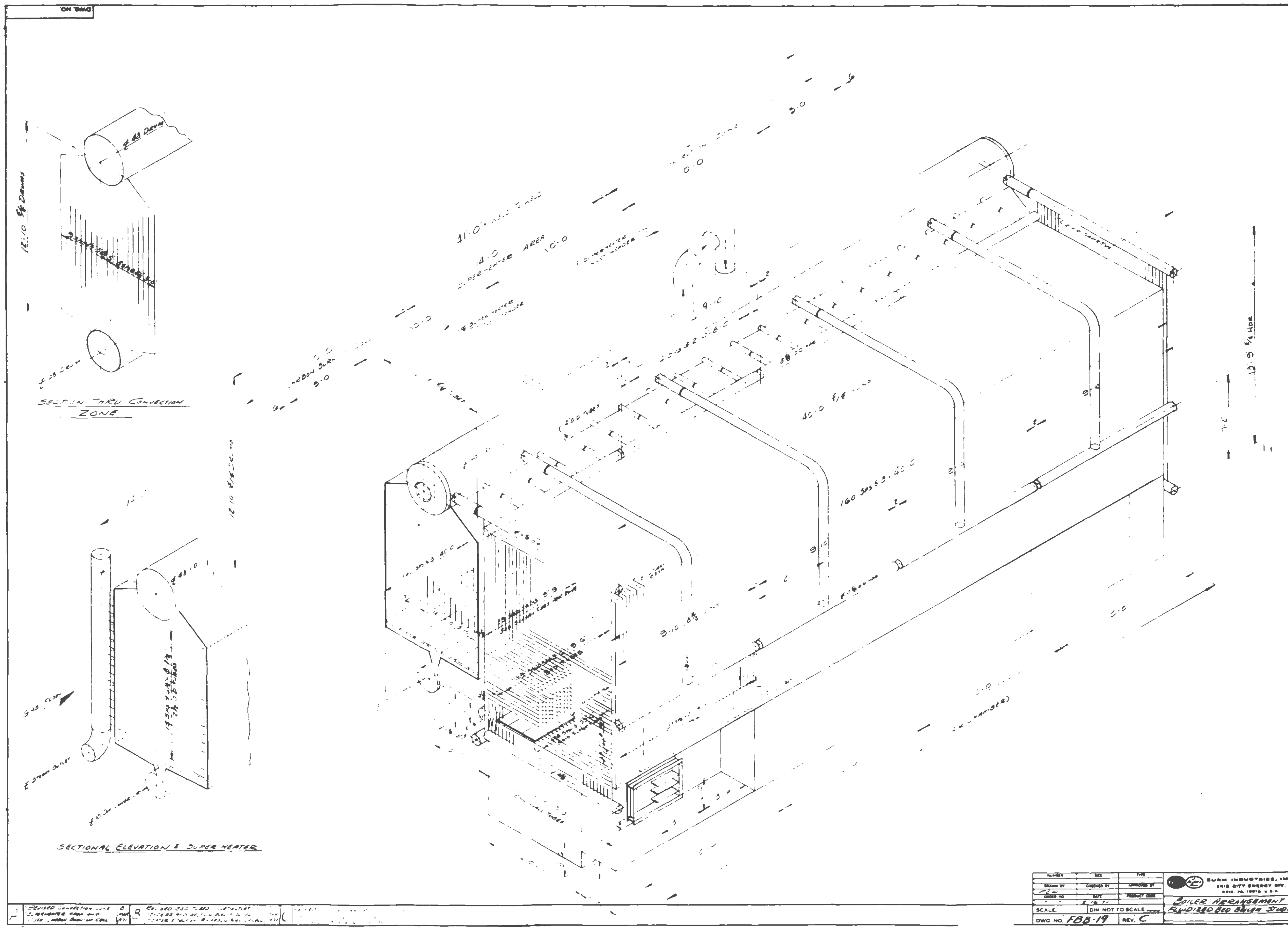
6. Operation of the Fluidized Bed Combustion Steam Generators is more complicated and exacting than present coal firing systems. Safety of operation, however, should not be a problem and no more severe than present techniques.

CONCLUSIONS (CONT.)

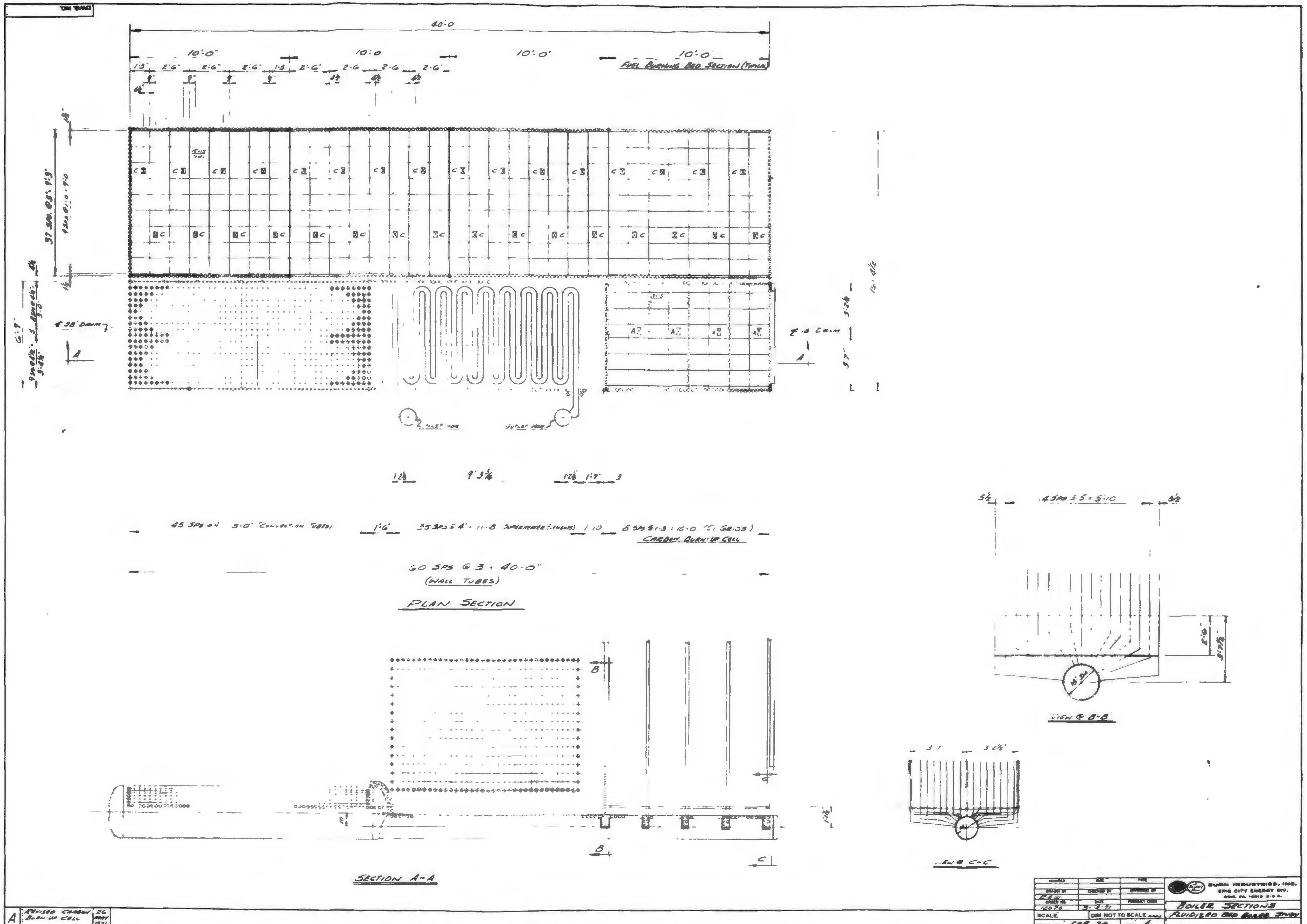
7. Industrial steam generator users will not respond favorably to Fluidized Bed Combustion units until there is an operating unit utilizing all the equipment and application rules that are to be used on the proposed system.

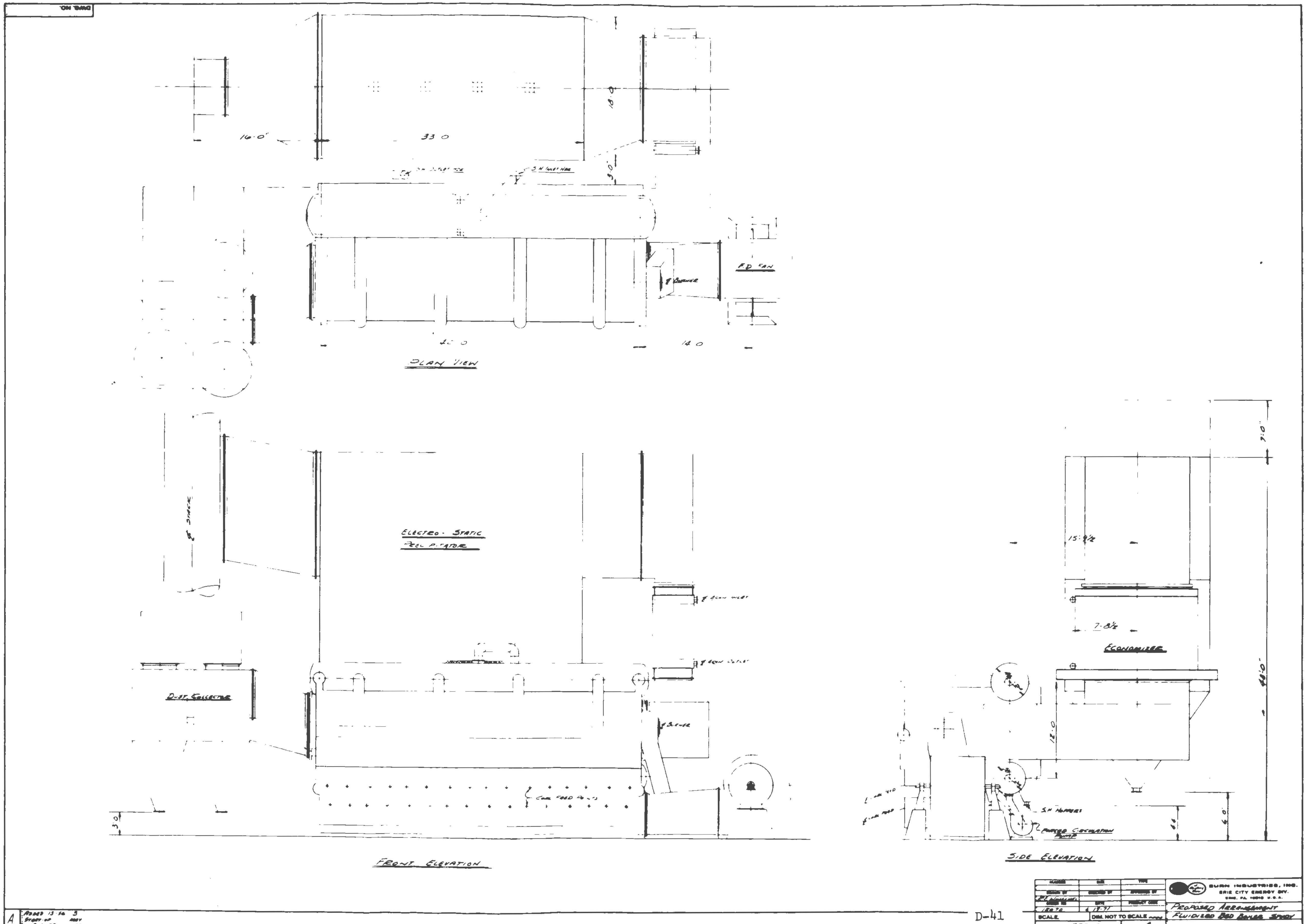
8. Present technology is sufficient to proceed to build a unit to test the fundamentals and to develop problem areas. Boiler manufacturers will not respond to support development costs without industry wide or without government support. Market possibilities are not sufficient to support the necessary funding on a unilateral commercial basis.

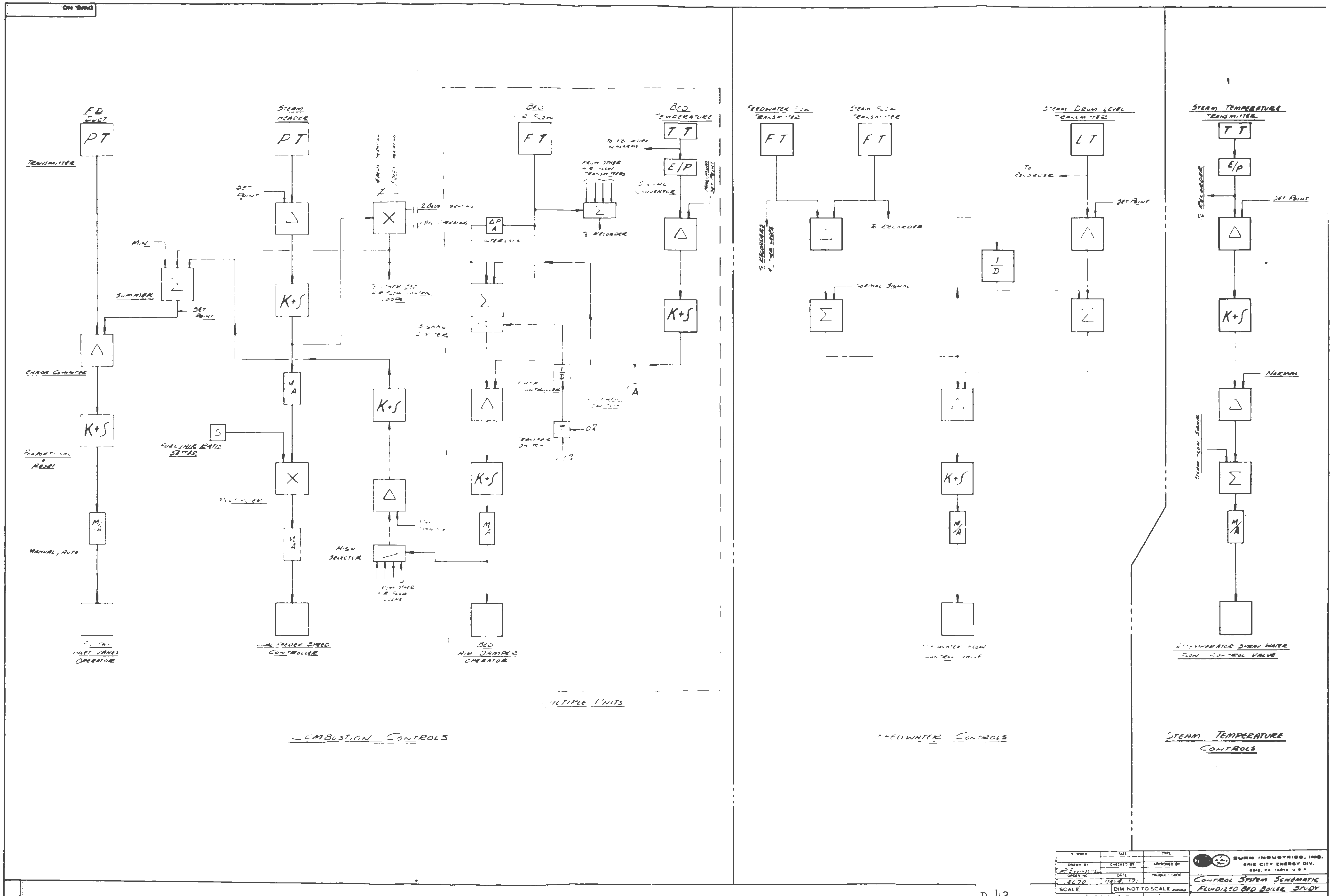
INDEX TO FIGURES

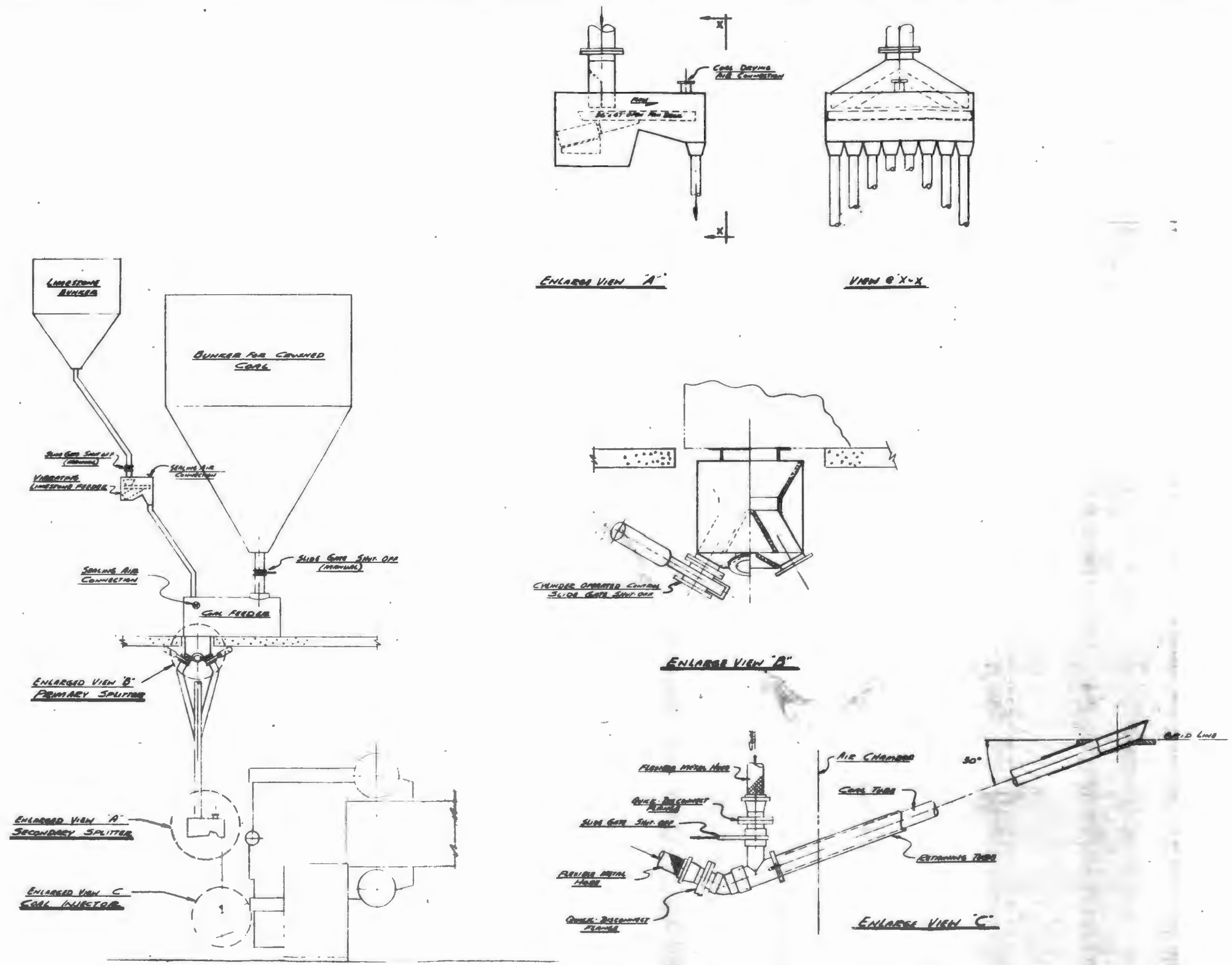


SECTION THRU CONVECTION ZONE
SECTIONAL ELEVATION & SUPER HEATER
160 SQS @ 3'-4.0
BURN INDUSTRIES, INC.
ERIE CITY ENERGY DIV.
SCALE
DIM NOT TO SCALE
DWG NO. FBB-19
REV. C
BOILER ARRANGEMENT
FLUIDIZED BED BOILER STUDY


40'-0"
10'-0"
10'-0"
10'-0"
10'-0"
FUEL BURNING BED SECTION (TYPICAL)
PLAN SECTION
60 SPS @ 3 = 40'-0"
(WALL TUBES)
CARBON BURN-UP CELL
OUTLET HDR.
SECTION A-A
VIEW @ B-B
VIEW @ C-C
REVISED CARBON BURN-UP CELL
26 MAY 1971
ZURN INDUSTRIES, INC.
ERIE CITY ENERGY DIV.
BOILER SECTIONS
FLUIDIZED BED BOILER STUDY
SCALE
DIM NOT TO SCALE
DWG NO. FBB-20
REV. A


16'-0"
33'-0
10'-0
5'-0
S.H. Inlet Hdr
S.H. Outlet Hdr
F.D. Fan
Burner
14'-0
Plan View
Electro-Static
Precipitator
Stack
Dust Collector
Burner
Coal Feed Points
Econ. Inlet
Econ. Outlet
3'-0
Front Elevation
7'-0
15'-9½
7'-8½
Economizer
12'-0
44'-0
S.H. Headers
Forced Circulation Pump
6'-0
Side Elevation
Burn Industries, Inc.
Erie City Energy Div.
Proposed Arrangement
Fluidized Bed Boiler Study
Scale
Dim. Not to Scale
DWG. NO. FBB-23
REV. A
A Added 13 SQ. 3 Start-up Burner
DWG. NO.


F.D. Duct
PT
Transmitter
Summer
Min
Set Point
Error Computer
K+∫
Manual, Auto
Inlet Vanes Operator
Steam Header
Set Point
Multiplier
High Selector
Coal Feeder Speed Controller
Bed Air Flow
FT
Bed Temperature
TT
E/P
Interlock
To Recorder
Bed Air Damper Operator
Multiple Units
Combustion Controls
Feedwater Flow Transmitter
Steam Flow Transmitter
To Recorder
Steam Drum Level Transmitter
LT
Set Point
Normal Signal
Feedwater Controls
Steam Temperature Transmitter
TT
E/P
To Recorder
Set Point
Normal
Desuperheater Spray Water Flow Control Valve
Steam Temperature Controls
Burn Industries, Inc.
Erie City Energy Div.
Erie, PA
Control System Schematic
Fluidized Bed Boiler Study
Dim not to scale
DWG. NO. FBB-27

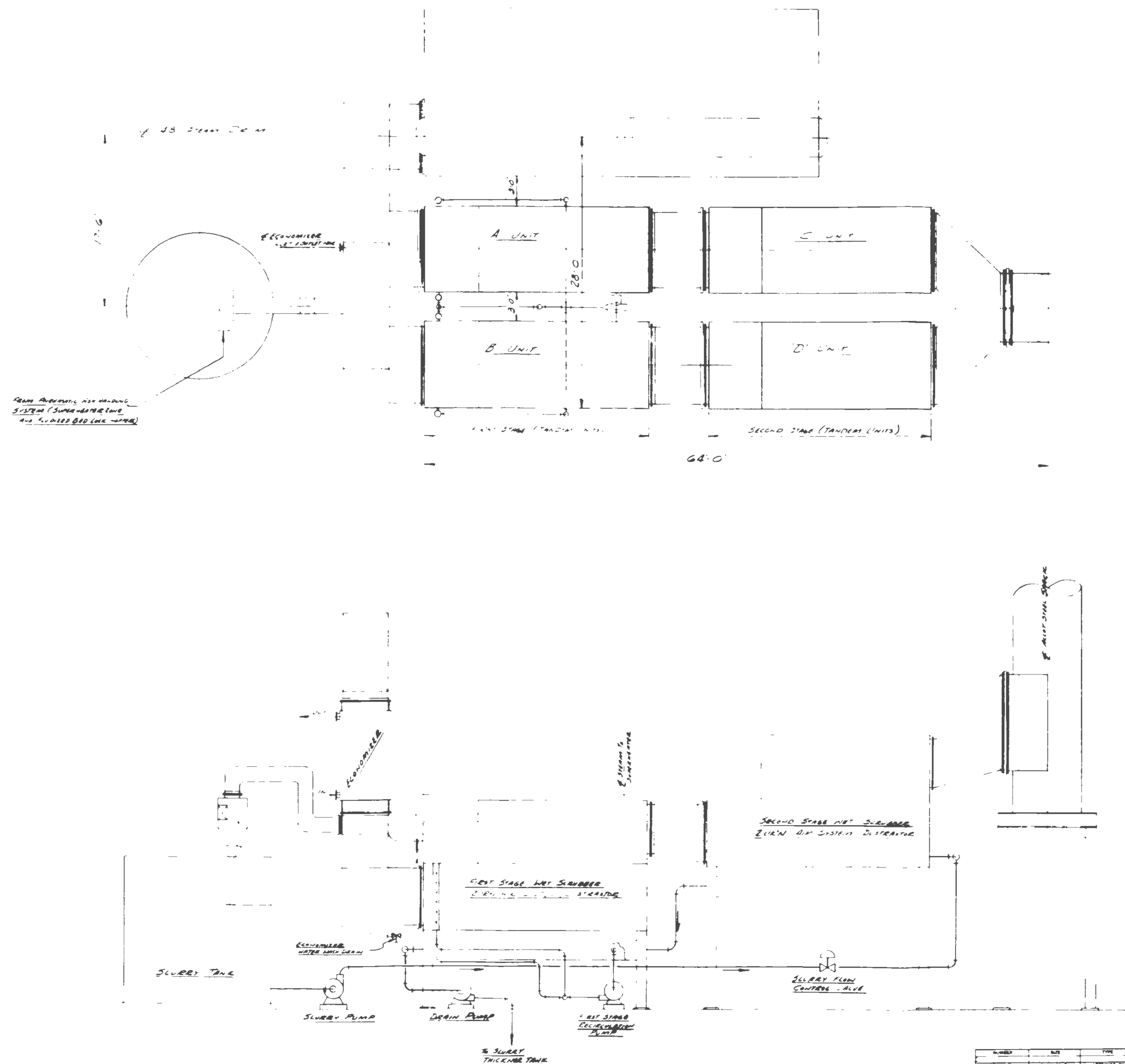


A	Added Limestone Bunker & Vibrating Feeder	26 Nov 1971

Drawn by: R.E. Hunsickel	Checked by	Approved by
Order No.: 18676	Date	Product Code
Scale	Dim not to scale	
Dwg. No. FBB-28	Rev. A	

Zurn Industries, Inc.
Erie City Energy Div.
Erie, PA. U.S.A.

Fuel System Development
Fluidized Bed Boiler Study



A	ADDED RECIRCULATION PUMP & PIPING	1971

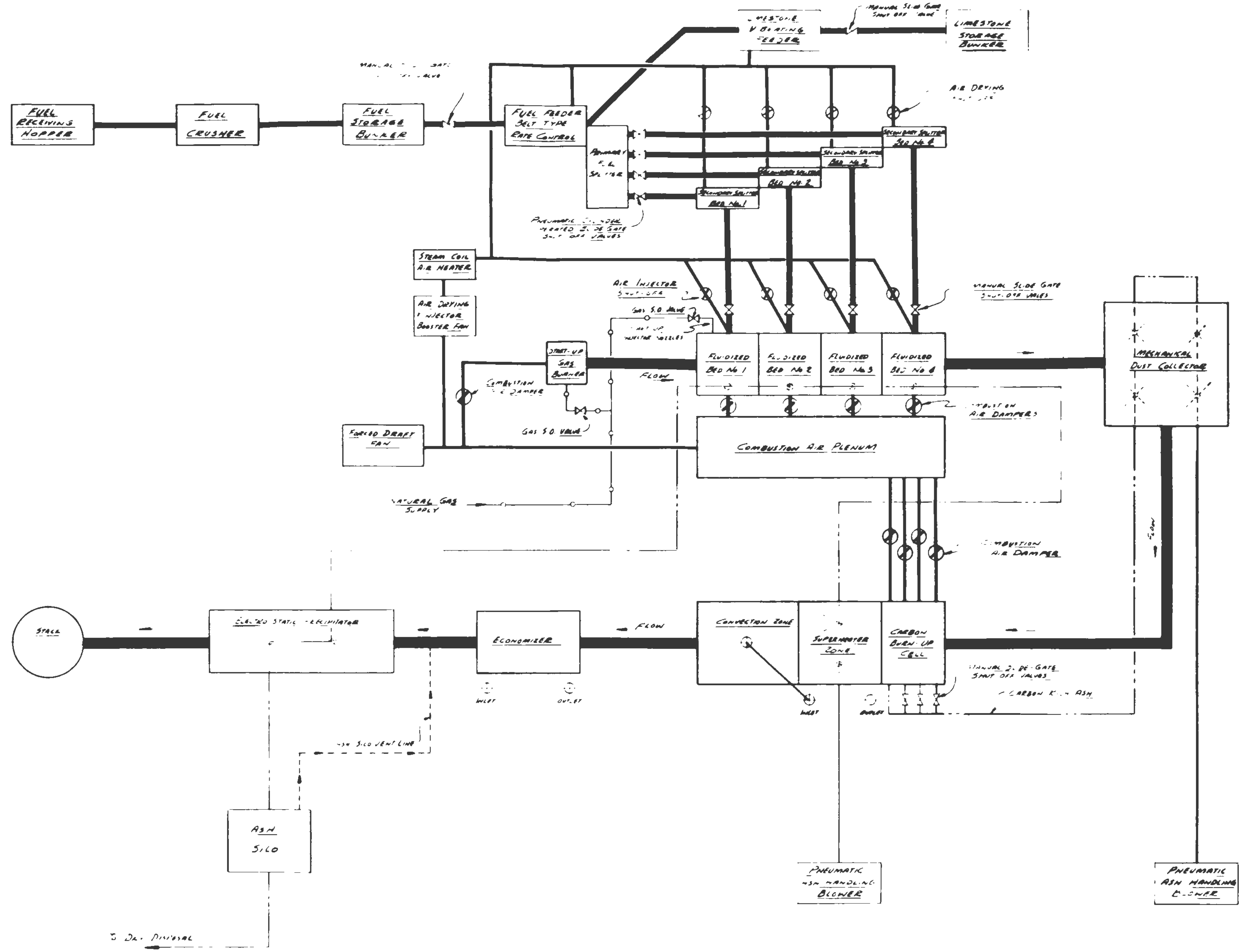
ZURN INDUSTRIES, INC.
ERIE CITY ENERGY DIV.
ERIE, PA U.S.A.

NUMBER	SIZE	TYPE
DRAWN BY	CHECKED BY	APPROVED BY
R.E. Hinkle		
ORDER NO.	DATE	PRODUCT CODE
18070	2, 1971	

WET SCRUBBER ARRANGEMENT
FLUIDIZED BED BOILER STUDY

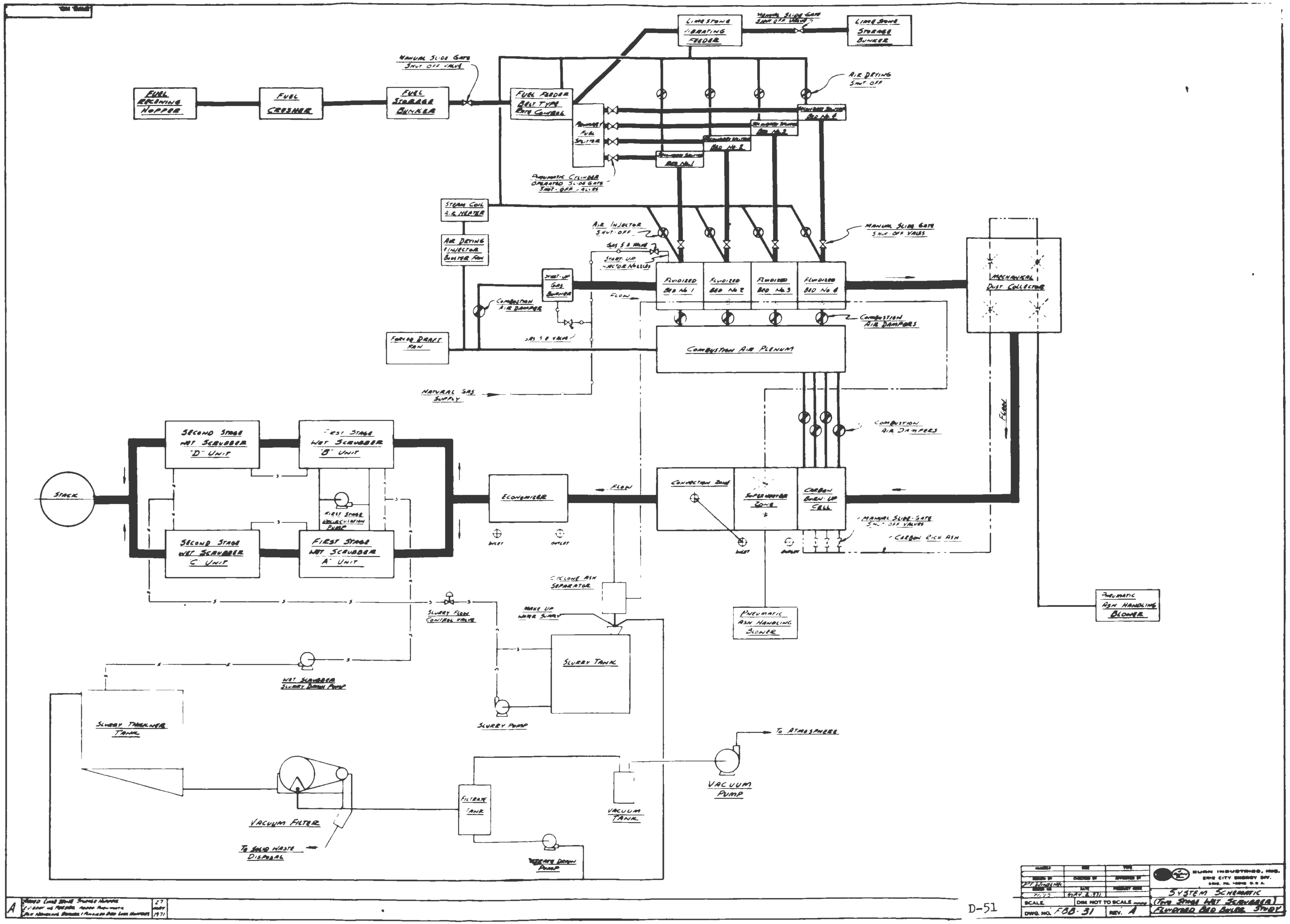
SCALE DIM NOT TO SCALE

DWG NO. FBB-29 REV.

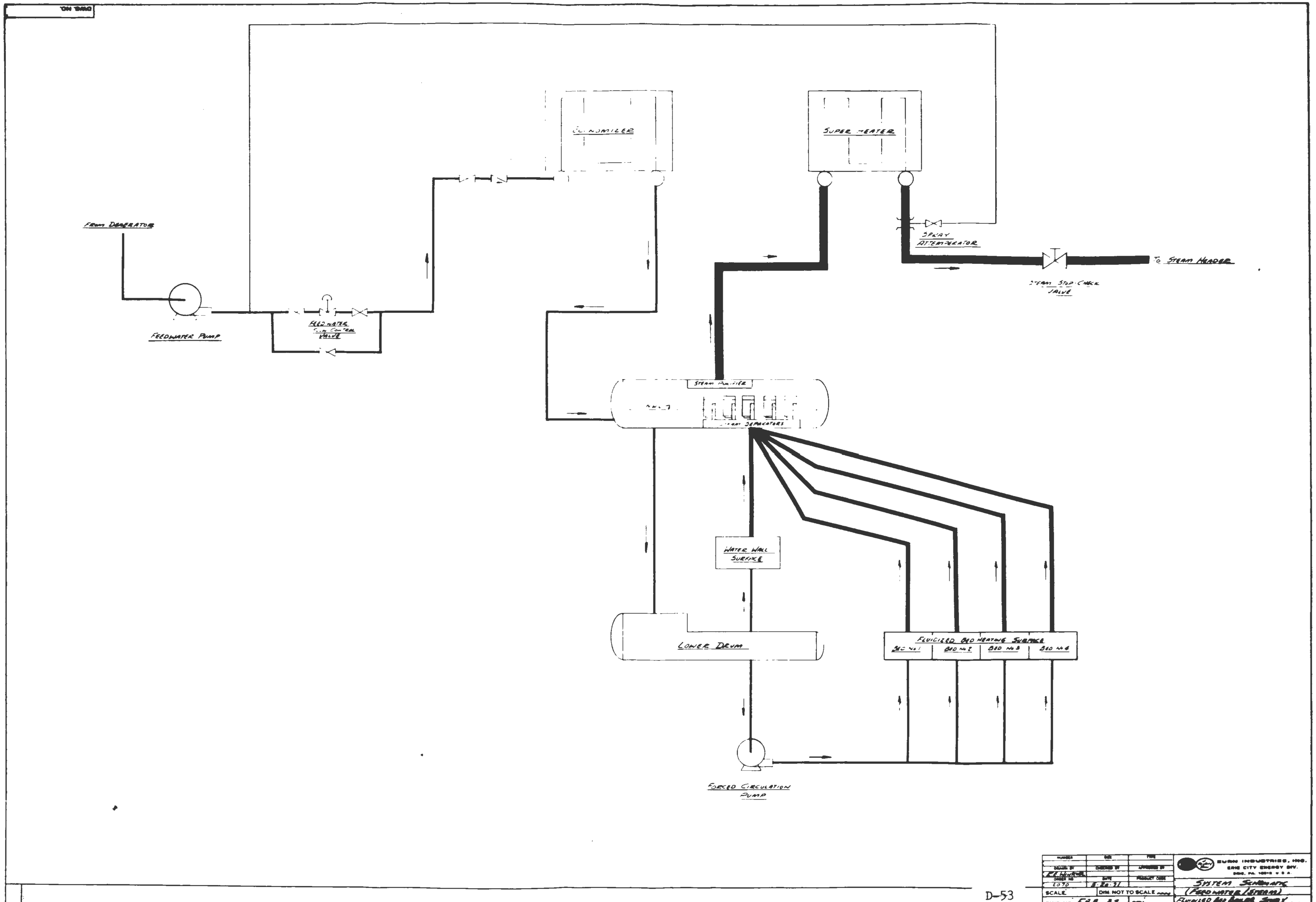


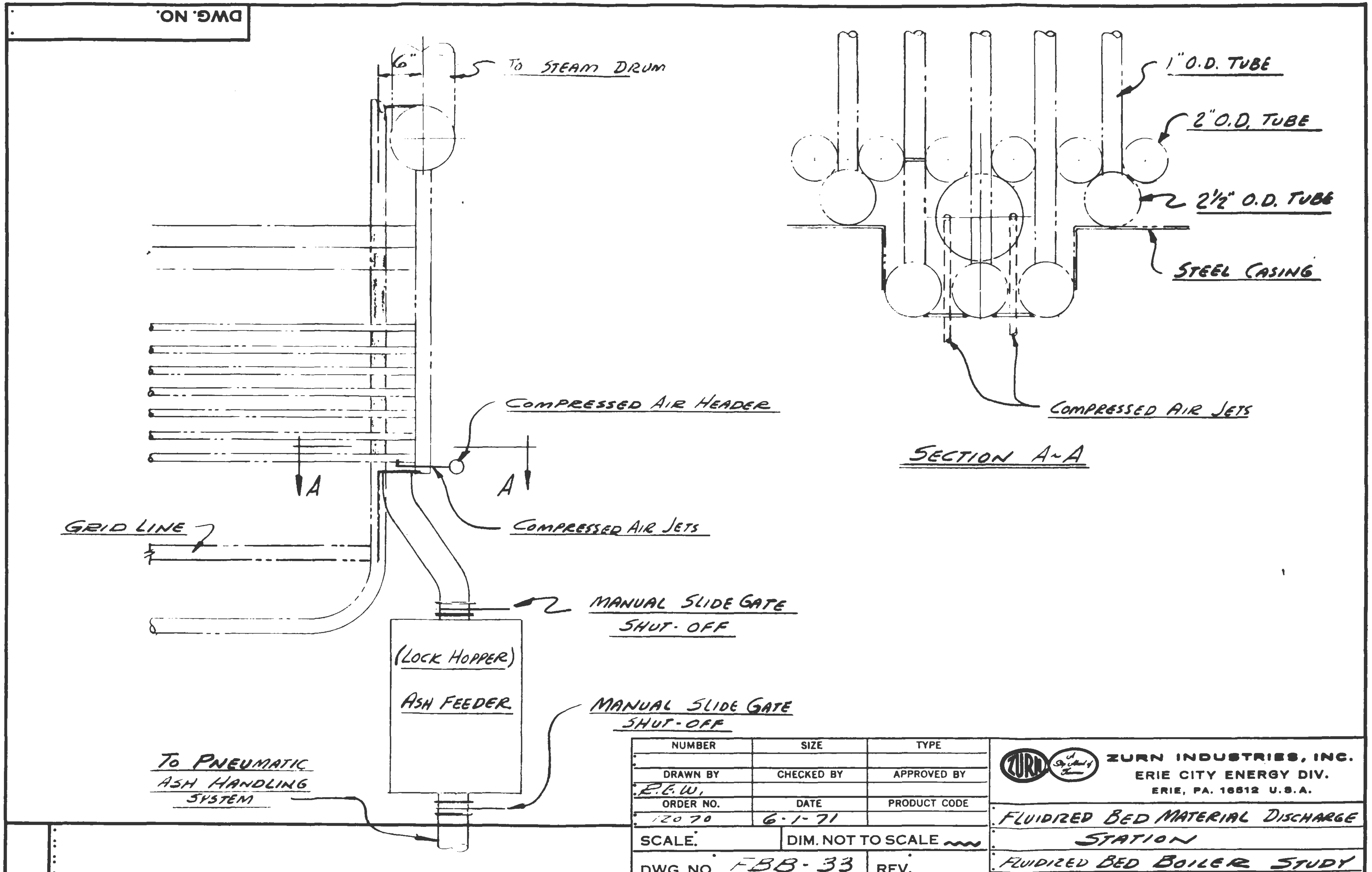
BURN INDUSTRIES, INC. ERIE CITY ENERGY DIV. ERIE PA U.S.A.	
SCALE	DIM NOT TO SCALE
DWG NO. 50	REV. A

System Schematic
(Electro-Static Precipitator)
Fluidized Bed Boiler Study



Fuel Receiving Hopper
Fuel Crusher
Fuel Storage Bunker
Manual Slide Gate Shut Off Valve
Fuel Feeder Belt Type Rate Control
Pneumatic Fuel Splitter
Limestone Vibrating Feeder
Manual Slide Gate Shut Off Valve
Limestone Storage Bunker
Air Drying Shut Off
Bed No. 1
Bed No. 2
Bed No. 3
Bed No. 4
Pneumatic Cylinder Operated Slide Gate Shut-Off Valves
Steam Coil Air Heater
Air Drying Injector Booster Fan
Air Injector Shut-Off
Gas S.O. Valve
Start-Up Injector Nozzles
Manual Slide Gate Shut Off Valves
Start-Up Gas Burner
Fluidized Bed No 1
Fluidized Bed No 2
Fluidized Bed No 3
Fluidized Bed No 4
Mechanical Dust Collector
Combustion Air Damper
Combustion Air Dampers
Flow
Gas S.O. Valve
Forced Draft Fan
Combustion Air Plenum
Natural Gas Supply
Combustion Air Dampers
Second Stage Wet Scrubber "D" Unit
First Stage Wet Scrubber "B" Unit
Stack
First Stage Recirculation Pump
Economizer
Flow
Convection Zone
Superheater Zone
Carbon Burn-Up Cell
Second Stage Wet Scrubber "C" Unit
First Stage Wet Scrubber "A" Unit
Inlet
Outlet
Manual Slide Gate Shut Off Valves
Carbon Rich Ash
Cyclone Ash Separator
Slurry Flow Control Valve
Make Up Water Supply
Pneumatic Ash Handling Blower
Pneumatic Ash Handling Blower
Slurry Tank
Wet Scrubber Slurry Drain Pump
Slurry Thickener Tank
Slurry Pump
To Atmosphere
Vacuum Pump
Vacuum Filter
Filtrate Tank
Vacuum Tank
To Solid Waste Disposal
Filtrate Drain Pump
Zurn Industries, Inc.
Erie City Energy Div.
Scale
Dim Not to Scale
System Schematic
(Two Stage Wet Scrubber)
Fluidized Bed Boiler Study
Dwg. No. FBB-31
Rev. A


DWG. NO.
Economizer
Super Heater
From Deaerator
Feedwater Pump
Feedwater Flow Control Valve
Spray Attemperator
Steam Stop-Check Valve
To Steam Header
Steam Purifier
Steam Separators
Water Wall Surface
Lower Drum
Fluidized Bed Heating Surface
Bed No. 1
Bed No. 2
Bed No. 3
Bed No. 4
Forced Circulation Pump
Zurn Industries, Inc.
Erie City Energy Div.
Drawn by
Checked by
Approved by
Order No.
2070
Date
Product Code
Scale
Dim. Not to Scale
DWG. NO. FBB-32
Rev.
System Schematic
(Feedwater/Steam)
Fluidized Bed Boiler Study

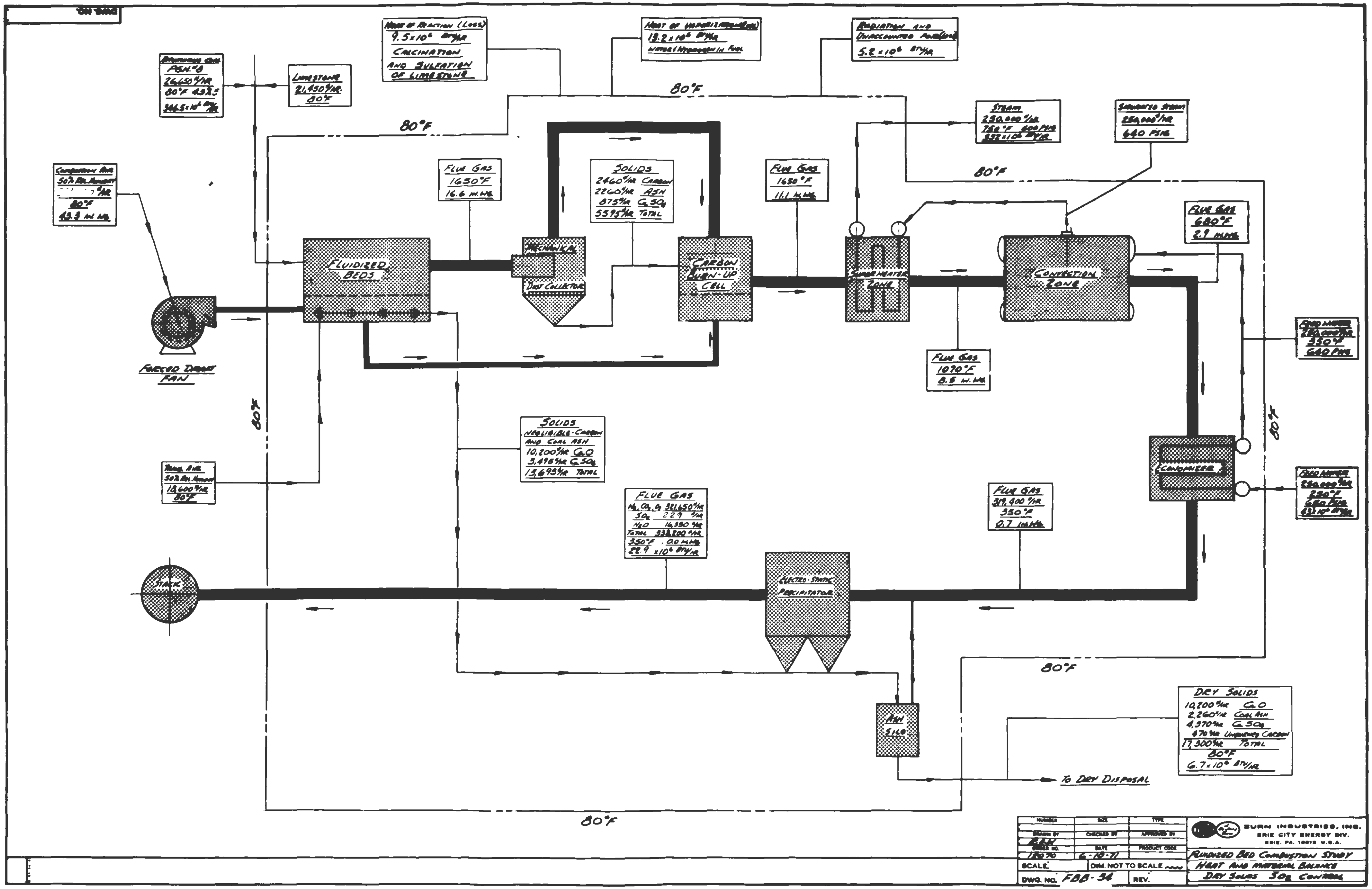


NUMBER	SIZE	TYPE
DRAWN BY	CHECKED BY	APPROVED BY
R.E.W.		
ORDER NO.	DATE	PRODUCT CODE
12070	6-1-71	
SCALE.	DIM. NOT TO SCALE	
DWG. NO. FBB-33	REV.	

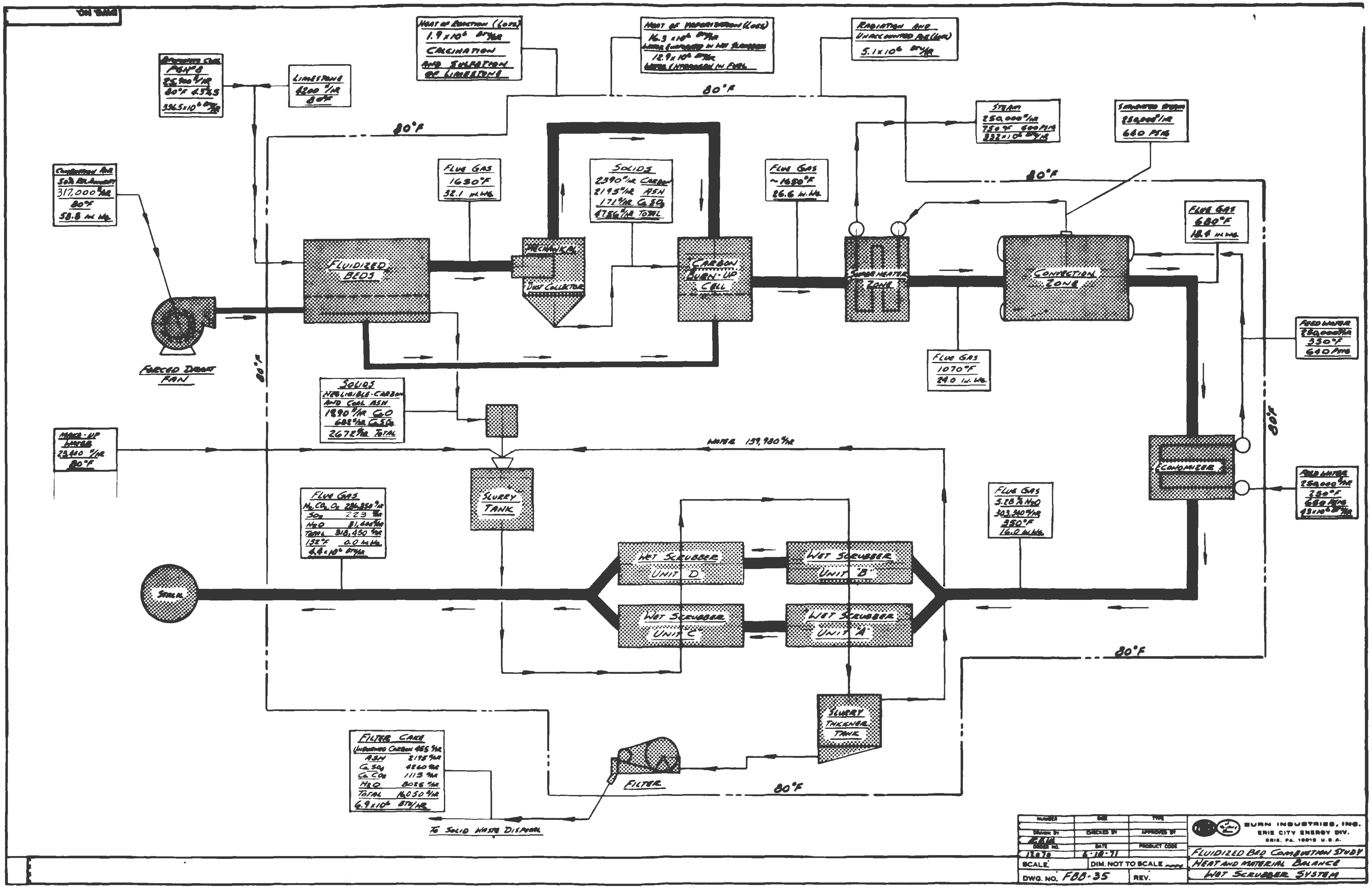
ZURN INDUSTRIES, INC.
ERIE CITY ENERGY DIV.
ERIE, PA. 16512 U.S.A.

FLUIDIZED BED MATERIAL DISCHARGE STATION

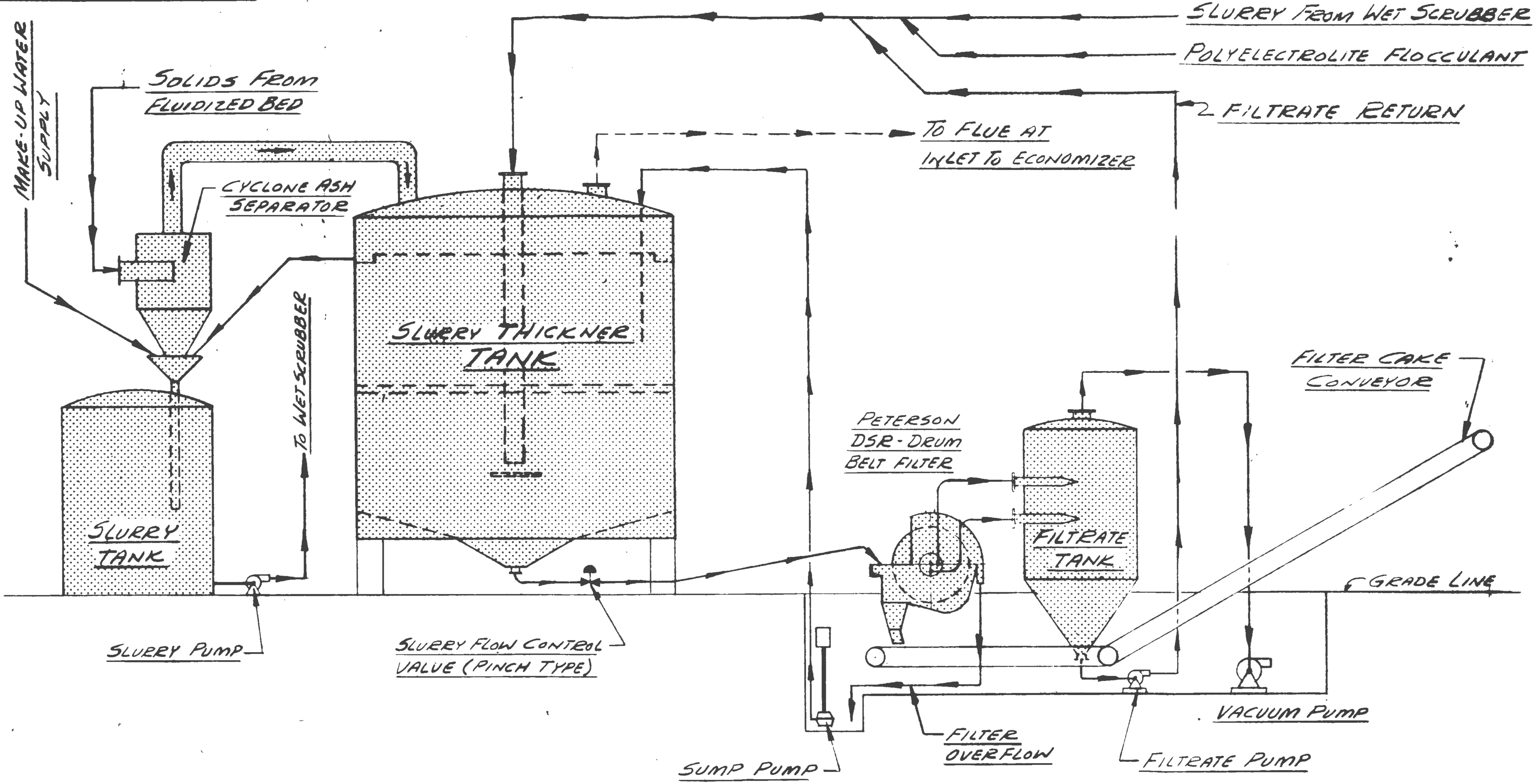
FLUIDIZED BED BOILER STUDY



PSH #8
26650#/HR
80°F
346.5×10^6 BTU/HR
LIMESTONE
21,450#/HR
80°F
HEAT OF REACTION (LOSS)
9.5×10^6 BTU/HR
CALCINATION AND SULFATION OF LIMESTONE
HEAT OF VAPORIZATION (LOSS)
19.2×10^6 BTU/HR
WATER/HYDROGEN IN FUEL
RADIATION AND UNACCOUNTED FOR (LOSS)
5.2×10^6 BTU/HR
STEAM
250,000#/HR
750°F 600 PSIG
SATURATED STEAM
250,000#/HR
640 PSIG
COMBUSTION AIR
50% REL. HUMIDITY
80°F
43.3 IN. WG
FLUE GAS
1650°F
16.6 IN. WG
SOLIDS
2460#/HR CARBON
2260#/HR ASH
875#/HR CaSO4
5595#/HR TOTAL
FLUE GAS
1650°F
11.1 IN. WG
FLUE GAS
680°F
2.9 IN. WG
FLUIDIZED BEDS
MECHANICAL DUST COLLECTOR
CARBON BURN-UP CELL
SUPERHEATER ZONE
CONVECTION ZONE
FORCED DRAFT FAN
FEED WATER
250,000#/HR
350°F
640 PSIG
FLUE GAS
1070°F
8.5 IN. WG
SOLIDS
NEGLIGIBLE CARBON AND COAL ASH
10,200#/HR CaO
3,495#/HR CaSO4
13,695#/HR TOTAL
TRANS. AIR
50% REL. HUMIDITY
18,600#/HR
80°F
ECONOMIZER
FEED WATER
250,000#/HR
280°F
680 PSIG
FLUE GAS
N2, CO2, O2 321,650#/HR
SO2 229 #/HR
H2O 16,350 #/HR
TOTAL 338,200 #/HR
350°F 0.0 IN. WG
22.9 ×10^6 BTU/HR
FLUE GAS
319,400 #/HR
350°F
0.7 IN. WG
STACK
ELECTRO-STATIC PRECIPITATOR
ASH SILO
80°F
DRY SOLIDS
10,200#/HR CaO
2,260#/HR COAL ASH
4,370#/HR CaSO4
470#/HR UNBURNED CARBON
17,300#/HR TOTAL
80°F
6.7×10^6 BTU/HR
TO DRY DISPOSAL
ZURN INDUSTRIES, INC.
ERIE CITY ENERGY DIV.
ERIE, PA. 16512 U.S.A.
FLUIDIZED BED COMBUSTION STUDY
HEAT AND MATERIAL BALANCE
DRY SOLIDS SO2 CONTROL
6-10-71
SCALE:
DIM. NOT TO SCALE
DWG. NO. FBB-34
REV.



HEAT OF REACTION (LOSS)
1.9 x 10^6 BTU/HR
CALCINATION AND SULFATION OF LIMESTONE
HEAT OF VAPORIZATION (LOSS)
16.3 x 10^6 BTU/HR
WATER EVAPORATED IN WET SCRUBBERS
12.9 x 10^6 BTU/HR
WATER (HYDROGEN IN FUEL)
RADIATION AND UNACCOUNTED FOR (LOSS)
5.1 x 10^6 BTU/HR
LIMESTONE
6200 #/HR
80°F
PSH #8
25,900 #/HR
80°F 4.5% S
80°F
STEAM
250,000 #/HR
750°F 600 PSIG
SATURATED STEAM
640 PSIG
COMBUSTION AIR
317,000 #/HR
80°F
58.8 IN. WG.
FLUE GAS
1630°F
32.1 IN. WG.
SOLIDS
2390 #/HR CARBON
2195 #/HR ASH
171 #/HR CaSO4
4756 #/HR TOTAL
FLUE GAS
~1650°F
26.6 IN. WG.
FLUE GAS
680°F
18.4 IN. WG.
FLUIDIZED BEDS
MECHANICAL DUST COLLECTOR
CARBON BURN-UP CELL
SUPERHEATER ZONE
CONVECTION ZONE
FEED WATER
330°F
640 PSIG
FORCED DRAFT FAN
FLUE GAS
1070°F
24.0 IN. WG.
SOLIDS
NEGLIGIBLE CARBON AND COAL ASH
1890 #/HR CaO
682 #/HR CaSO4
2672 #/HR TOTAL
MAKE-UP WATER
80°F
WATER 159,980 #/HR
ECONOMIZER
FLUE GAS
SO2 223 #/HR
TOTAL 318,450 #/HR
132°F 0.0 IN. WG.
SLURRY TANK
FLUE GAS
303,340 #/HR
350°F
16.0 IN. WG.
FEED WATER
250°F
STACK
WET SCRUBBER UNIT "D"
WET SCRUBBER UNIT "B"
WET SCRUBBER UNIT "C"
WET SCRUBBER UNIT "A"
80°F
SLURRY THICKENER TANK
FILTER CAKE
ASH 2195 #/HR
CaSO4 4860 #/HR
CaCO3 1115 #/HR
H2O 8026 #/HR
6.9 x 10^6 BTU/HR
FILTER
TO SOLID WASTE DISPOSAL
ZURN INDUSTRIES, INC.
ERIE CITY ENERGY DIV.
ERIE, PA. 16512 U.S.A.
6-18-71
SCALE
DIM. NOT TO SCALE
DWG. NO. FBB-35
REV.
FLUIDIZED BED COMBUSTION STUDY
HEAT AND MATERIAL BALANCE
WET SCRUBBER SYSTEM



NUMBER	SIZE	TYPE
DRAWN BY	CHECKED BY	APPROVED BY
R.E. WINSCHEL		
ORDER NO.	DATE	PRODUCT CODE
12070	6-25-1971	

SCALE. DIM. NOT TO SCALE

DWG. NO. FBB-36 REV.

ZURN INDUSTRIES, INC.
ERIE CITY ENERGY DIV.
ERIE, PA. 16512 U.S.A.

WET SCRUBBER - SLURRY DEWATERING SYSTEM - FLUIDIZED BED BOILER STUDY

INDEX APPENDICES

APPENDIX I

COMPARISON OF INSTALLED COSTS*

FOR EQUIVALENT CAPACITY, FIELD ERECTED,

STEAM GENERATORS FIRING

DIFFERENT FUELS

Steam Generator (See detailed description)	"A"	"B"	"C"	"D"
Fuel & Firing Technique	Gas or Oil firing	Coal fired, spreader stoker	Coal fired, fluidized bed combustion.	
SO_2 emission control	Low sulfur fuel	Not furnished	Dry solids system	Wet Scrubber system
Installed Cost for boiler system, $	610,000**	1,375,000	1,524,000	1,852,000
Installed cost for SO_2 control system, add, $	- - -	520,000	- - -	- - -
Installed cost total for equivalent emission to atmosphere, $	610,000	1,895,000	1,524,000	1,852,000
Cost per unit Steam generation, $/#/hr. for equivalent stack emission	2.44	7.57	6.10	7.40

* Costs are based on experience with similar steam generating units escalated to current prices. Estimated factory and installation prices are used for FBC Steam Generators.

** If the steam generator is shop assembled, the installed cost will be reduced 16%.

DESCRIPTION OF THE FOUR DIFFERENT STEAM GENERATORS USED IN CAPITAL COST AND OPERATING COST COMPARISONS

Steam Generator "A":

Erie City Energy Division, Field Erected, "Keystone" Steam Generator

Operating Conditions:
- Steam Flow - 250,000#/hr.
- Steam Pressure - 600 psig
- Steam Temperature - 750° F.
- Feedwater Temperature 250F
- Stack Temperature - 300F
- Boiler Efficiency - 82.2%
- System resistance - 11.8 in wg.
- (Pressurized operation)

Fuels: Natural Gas - #2 Oil

Terminal connections are: inlets to: economizer, fuel train and stack
outlets from: superheater

Equipment Furnished:

- Burners - ECED circular burners
- Boiler - bottom supported - no preassembly.
- Superheater
- Economizer - extended surface
- Flues, Ducts and Dampers (not including stack)
- Supporting Steel
- Soot Blowers
- Combustion and Feedwater Control Systems
- Refractory and Insulation
- Piping for: boiler fuel system, connection between economizer and boiler and trim piping

Services Furnished:

- Erection supervision
- Erection labor
- Start-Up Service

STEAM GENERATOR DESCRIPTIONS (CONTINUED)

Steam Generator "B"

Erie City Energy Division, Field Erected "Cross Drum" Steam Generator

Operating Conditions:
- Steam Flow - 250,000
- Steam Pressure - 600 psig
- Steam Temperature - 750F
- Feedwater Temperature -250F
- Stack temperature - 400F
- Boiler Efficiency - 85%
- System air resistance draft loss - 12.4 in wg. (Balanced draft)

Fuels burned: Bituminous coal (Sized to no more than 30% through 4 mesh screen)

Terminals are: outlets from: ash handling system and superheater
inlets to: economizer, stoker feeders and stack

Equipment Furnished:

- Forced draft and induced draft fans and drives
- Stoker - Detroit roto grate - spreader stoker
- Boiler - no preassembly - top supported
- Superheater
- Economizer - bare tube
- Boiler columns and top grid steel
- Soot blowers
- Combustion and feedwater controls
- Refractory and insulation
- Mechanical collector
- Electrostatic precipitator
- Flues, ducts and dampers (not including stack.)
- Ash handling system
- Piping for: connection between economizer and boiler and trim piping

Services Furnished:

- Erection supervision
- Erection labor
- Start-up service

STEAM GENERATOR DESCRIPTIONS (Continued)

Steam Generator "C"

Erie City Energy Division, Modular, Fluidized Bed Combustion, Steam Generator using Dry Solids SO_2 Control.

Operating Conditions:

Steam Flow - 250,000#/hr.
Steam Pressure - 600 psig
Steam Temperature - 750F
Feedwater Temperature - 250F
Stack Temperature - 350F
Boiler Efficiency - 85.4%
System Resistance - 43.4 in wg.
(pressurized operation)

Fuel: Bituminous coal - - 100% through 4 mesh

Terminals are: inlets to: economizer, coal feeder, limestone feeder and stack
outlets from: ash silo, superheater

Equipment Furnished:

Forced draft fan and drive
Coal feeding system including rate feeder, splitter and injectors
Limestone feeder
Boiler - bottom supported pre-assembled into two modules
Superheater
Economizer - extended surface
Supports for mechanical collector economizer and precipitator
Combustion and feedwater controls
Refractory and insulation
Mechanical collector
Electro static precipitator
Flue, ducts and dampers (not including stack)
Ash handling system and silo
Piping for: Boiler fuel system, between economizer and boiler and trim piping.

Services Furnished:

Erection Supervision
Erection Labor

STEAM GENERATOR DESCRIPTION (Continued)

Steam Generator "D"

Erie City Energy Division Modular, Fluidized Bed Combustion Steam Generator using a wet scrubber - particulate and SO_2 removal system

Operating Conditions:

Steam flow - 250,000#/hr.
Steam pressure - 600 psig
Steam temperature - 750F
Feedwater temperature - 250F
Stack temperature - 132F (Saturated with water vapor)
Boiler efficiency - 85.8%
System resistance - 58.8 in wg.
(Pressurized operation)

Fuel: Bituminous coal - 100% through 4 mesh

Terminals are: inlets to: economizer, coal feeder, linestone feeder and stack
outlets from: vacuum filter and superheater

Equipment Furnished:

Forced draft fan and drive
Coal feeding system including rate feeder, splitters and injectors
Limestone feeder
Boiler - bottom supported (pre-assembled into two modules)
Superheater
Economizer - extended surface
Supports for mechanical collector, economizer and precipitator
Combustion, feedwater and scrubber controls
Refractory and insulation
Mechanical collector
2 stage wet scrubber with slurry tank and pumps
Flues, ducts and dampers (not including stack)
Pneumatic transport ash handling system
Slurry and ash dewatering system

Piping for:

Boiler fuel system, between economizer and boiler trim piping and scrubber system convections.

Services Furnished:

Erection supervision
Erection labor

COMPARISON OF APPROXIMATE INSTALLED COSTS FOR

EQUIVALENT CAPACITY, FLUIDIZED BED COMBUSTION STEAM GENERATORS,

(SHIPPED AS TWO SHOP ASSEMBLED MODULES)

WET SCRUBBER SYSTEM

COMPONENT	APPROXIMATE EQUIPMENT COST $	APPROXIMATE INSTALLATION COST $	TOTAL $
Crusher	18,000	Not Included	18,000
Fuel system(not including bunkers, and system upstream of bunkers)	74,000	25,000	99,000
Forced Draft Fan	54,000	5,000	59,000
Fuel System Sealing Fan & Heater System	12,000	10,000	22,000
Steam Generator System Recirculating Pump & Economizer	753,000	95,000	848,000
Ash System for Slurry System for Carbon Burnup	100,000	50,000	150,000
Mechanical Dust Collector	30,000	15,000	45,000
Wet Scrubber System	300,000	100,000	400,000
Controls	61,000	30,000	91,000
Dewatering	80,000	40,000	120,000
Wet Scrubber System Approximate Cost	1,482,000	370,000	1,852,000

DRY SOLIDS SYSTEM

CHANGE THE WET SCRUBBER SYSTEM BY:

REMOVING:

	APPROXIMATE EQUIPMENT COST $	APPROXIMATE INSTALLATION COST $	TOTAL $
Forced Draft Fan	54,000	5,000	59,000
Ash System	100,000	50,000	150,000
Wet Scrubber	300,000	100,000	400,000
Dewatering	80,000	40,000	120,000
	534,000	195,000	729,000

AND, BY ADDING:

Forced Draft Fan	35,000	5,000	40,000
Ash Handling System	170,000	85,000	255,000
Electric Static Precipitator	76,000	30,000	106,000
	281,000	120,000	401,000
Adjust Wet Scrubber Price to obtain Dry Solids Price	(-) 253,000	(-) 75,000	(-) 328,000
Dry Solids System Approximate Costs	1,229,000	295,000	1,524,000

COMPARISON APPROXIMATE INSTALLED COSTS FOR EQUIVALENT
FLUIDIZED BED COMBUSTION STEAM GENERATORS

		Approximate Equipment and Installation Costs	Relative Cost
A)	For shop assembly into two modules joined together on site	$848,000	1.0
B)	For field assembly of fluidized bed section - shop assembly of convection zone. (design parameters as in A above)	$973,000	1.15
C)	For field assembly of fluidized bed section - shop assembly of convection zone (bed velocity designed to 8 ft per sec rather than 15 ft/sec)	$1,470,000	1.74

APPENDIX 2

WESTINGHOUSE RESEARCH & DEVELOPMENT CENTER

PITTSBURGH, PENNSYLVANIA

CONTROLS AND INSTRUMENTS FOR A FLUIDIZED BED BOILER

DETAILED TABULATION OF EQUIPMENT

CID PROPOSAL 41-71-04-209

Quantity	Description	Reference
Item I - Combustion Controls		
Drawing SK-1		
Group A - Equipment for Flush Panel Mounting		
8	Manual Control Stations 1 - F.D. Fan Inlet Vanes 1 - Coal Feeder 1 - Boiler Master 4 - Fluidized Bed Air Damper 1 - Carbon Cell Air Damper	PB-129-331
1	Fuel-Air Ratio Adjuster	
Group B - Equipment for Rear Panel or Rack Mounting		
23	Ratio Totalizers 1 - Duct Pressure Set Point Computer 4 - Fluidized Bed Air Flow Set Point Computers 4 - Fluidized Bed Air Flow Controllers 1 - Fluidized Bed Damper Opening Limiter 4 - Bed Temperature Limiters 3 - Multiplying Relays 1 - Carbon Cell Temperature Controller 1 - Carbon Cell Air Flow Set Point Computer 1 - Carbon Cell Air Flow Controller 4 - Air Flow Summators	
4	High Signal Selectors	
8	Solenoid Valves	
8	Needle Check Valves	
8	Needle Valves	
5	Volume Chambers	
4	Differential Pressure Switches	
5	Bed Temperature Transmitters	
5	Electro-Pneumatic Converters	
5	Pressure Switches	

Quantity	Description	Reference
1	Set of Relay Logic	
	Necessary Air Supply Accessories	

Group C - Equipment for Field Mounting

15	Bed Temperature Thermocouples	
1	F.D. Duct Pressure Controller - CAM Loaded D-33 Regulator	
5	Air Flow Trnasmitters - Type D-33 Flow Signal Transmitters	
1	Steam Header Pressure Controller - Type "F" Master Sender	
1	F.D. Fan Inlet Vanes Operator - 6" x 10" Pneumatic Power Positioner w/Dust Cover & Manual Operator	
5	Bed Air Damper Operators - 4" x 5" Power Positioners w/Dust Cover, Manual Operator & Limit Switches	
6	Simple Linkage Struts	
	Necessary Air Supply Accessories	

Quantity	Description	Reference
Item II - Feedwater Controls		
Drawing SK-2		
Group A - Equipment for Flush Panel Mounting		
1	Manual Control Station	
Group B - Equipment for Rear Panel or Rack Mounting		
3	Ratio Totalizers 1 - Drum Level Corrector 1 - Feedwater Flow Computer 1 - Feedwater Flow Controller	
2	Pressure Switches - Mercoid Model DA-33 for High & Low Drum Level	
Group C - Equipment for Field Mounting		
1	Steam Flow Transmitter - Ring Balance Model 3008-R80-HT-IG	
1	12" Chrome Moly, Weld-In Type Flow Nozzle w/Holding Ring & Pins	
1	Feedwater Flow Transmitter - Ring Balance Model 3008-R88-HT-IG	
1	4" Stainless Steel Weld-In Type Flow Nozzle w/Holding Ring & Pins	
1	Drum Level Transmitter - Type FRB-R80 w/One Charge of Sealing Fluid, Equalizing Manifold Condensate Reservoir and Suppression Weight	
1	Feedwater Control Valve - Fisher Type 667ED w/4" - 600# Flanged Carbon Steel Body and Positioner	
	Necessary Air Supply Accessories	

Quantity	Description	Reference
Item III - Steam Temperature Controls		
Drawing SK-3		
Group A - Equipment for Flush Panel Mounting		
1	Manual Control Station	
Group B - Equipment for Rear Panel or Rack Mounting		
3	Ratio Totalizers	
	1 - Characterizing Relay 1 - Steam Temperature Error Computer 1 - Steam Temperature Controller	
1	Temperature Transmitter	
1	Electro-Pneumatic Converter	
Group C - Equipment for Field Mounting		
1	Thermocouple Assembly	
1	Spray Control Valve - Fisher Type 667DBQ w/2" - 1500# Carbon Steel Body and Valve Positioner	

Quantity	Description	Reference
Item IV - Panel Instruments		
3	Miniature Pneumatic Strip Chart Recorders 1 - Steam Flow - Air Flow 1 - Steam Pressure - Steam Temperature 1 - Feedwater Flow - Drum Level	
1	Miniature Electronic One Pen Strip Chart Recorder - To Record Oxygen	
1	Miniature Electronic Three Pen Strip Chart Recorder - To Record Bed Temperature w/Alarms	
1	Miniature Electronic Two Pen Strip Chart Recorder - To Record Bed Temperature w/Alarms	
2	Integrator Counters 1 - Steam Flow 1 - Feedwater Flow	
2	Pressure Gauges - 8-1/2" - Ashcroft Type 1377A 1 - Steam Pressure 1 - Feedwater Pressure	
1	Three Point Draft Gauge - To Indicate The Following: A - F.D. Duct Pressure B - Furnace Pressure C - Boiler Outlet Pressure	
1	15 Point Annunciator - Scam De Line, Sequence AF Arranged 3 High X 5 Wide w/ Flasher, Alarm Horn & Two Pushbuttons - For The Following Functions: 1 - Drum Level "Hi" 1 - Drum Level "Lo" 5 - Bed Temperature "Hi" 5 - Bed Temperature "Lo" 1 - Steam Pressure "Hi" 1 - Steam Temperature "Hi" 1 - Feedwater Pressure "Lo"	

Quantity	Description	Reference
Item V - Miscellaneous Equipment		
2	Pressure Switches - Mercoid Model DA-33 1 - Steam Pressure 1 - Feedwater Pressure	
2	Pressure Transmitters 1 - Steam Header 1 - Feedwater Pressure	
1	Oxygen Analyzer - Westinghouse "Probe In Stack" Type w/Probe, Shield, Temperature Controller & Amplifier	
	Air Sets	

Quantity	Description	Reference
Item VI - Instrument Panel		
1	Control Panel Assembly to be 7'-0" High, 3'-8" Wide & 2'-0" Deep Construction: Panel to be U Shaped of 3/16" Steel Plate Suitably Braced to Form a Rigid Structure. To be fitted w/Top and Rear Covers and Rear Access Doors of 1/8" Steel Plate. Finish: Hammertone Gray Piping & Wiring: Terminal Piping & Wiring (#14 AWG Thermoplastic Stranded) for panel mounted equipment included in this quotation all piping and wiring performed in an "open" shop.	

Quantity	Description	Reference

Item VII - Relay Rack

1	Relay Rack Assembly to be 7'-6" high, 7'-0" long and 2'-0" deep.	
	Construction: Panel to be U Shaped of 3/16" steel plate suitably braced to form a rigid structure. To be fitted with top and rear covers and rear access doors of 1/8" steel plate.	
	Finish: Hammertone Gray	
	Piping & Wiring: Terminal Piping & Wiring (#14 AWG thermoplastic stranded) for panel mounted equipment included in this quotation all piping and wiring performed in an "open" shop.	

APPENDIX 3

FLUIDIZED BED COMBUSTION STEAM GENERATOR

OPERATING PROCEDURES

Description of Unit

For the steam generator and auxiliary equipment for either the dry solids or the wet scrubber system, the steam generator operating procedures are much the same and are generally described below. Flue gas cleanup equipment will have slightly different operating procedures and those special procedures are included.

There are four principal modes of operation: Starting up; On-line operation; Shutting down; and, Emergency operation.

Starting Up

As with any steam generator, the starting up procedures must provide adequate protection for the superheater tube sections and for the steam generator water cooled circuits. To do this, the principal fuel, coal, is not fired until the steam generator is raised to the steam system pressure and is in service. Natural gas is used to raise the boiler pressure and to establish a low rate of steam generation. Natural gas is fired through a circular burner at one end of the fluidized bed combustion chamber. The start up burning rating is only a fraction of the steam generator capacity.

Firing Natural Gas

Prior to firing the startup burner; the boiler is filled to the normal water level in the steam drum and the boiler water recirculation pump is started. The normal water amount of bed material (limestone) is transferred into the fluid bed chamber. The ash conveying equipment is put into operation as soon as fluidization begins. Because of condensation at low firing rates when the equipment is cool, ash must not be allowed to reside in any hoppers. Sufficient coal and limestone should be transferred to the bunker to form a pressure seal at the rate feeder inlets prior to fluidizing.

Prior to firing the dry solids system, the electro-static precipitator fields should be energized and rapper operation begun.

Prior to firing with the wet scrubber system, scrubbers, the slurry tank and the thickner should be filled to normal levels. All transfer pumps, and the scrubber inlet spray recirculating pumps should be started.

Prior to firing in either system, the sealing air fan should be started and the forced draft fan should be started and dampers opened to deliver air to the start-up burner only.

The start up burner is lit and the firing rate should be modulated to hold the gas temperature entering the superheater below 1000F and/or to have a rate of pressure raising not exceeding 100F saturation temperature change per hour.

After the steam generator has reached steam system pressure and is delivering steam to the system, the start up burner firing rate should be increased to maximum.

Firing Coal

When the start-up burner is firing at the maximum rate coal firing may be begun but prior to firing coal, the sealing air steam coil air heater should be put into service the speed control circuits of the coal feeder and the other material feeders should be made ready to operate. The combustion air control damper is opened for the bed section to be started to achieve fluidization and then set to the minimum firing rate air flow. A bed temperature of 800°F must be obtained to insure ignition of the coal feed. If sufficient bed temperature is not obtained from the start-up burner, additional fuel gas is injected beneath the bed through the coal injector nozzles.

When the bed section fluidized bed temperature is adequate, coal feed is begun at the minimum firing rate to one bed section. As firing is stabilized, the start-up burner may be shut down.

If greater steam flows are required, more fuel bed sections should be put into service by fluidizing the adjacent bed section, heating to 800°F and then beginning the coal feed. Certain operating parameter permissives and rate controllers are included in the control system to provide safety and uniformity in idle bed starting up conditions.

As bed sections are fluidized while beginning to fire coal, some bed material will begin to be elutriated and will appear at the carbon-burn-up cell and the ash disposal points. In either the dry solids or the wet scrubber system, lime stone feed should begin with coal firing to maintain the bed level.

Carbon-burn-up cell operation, also, should begin with coal firing. The first carbon-burn-up cell section should be started up by injecting air and by opening the cyclone dust collector ash line to that section. As more bed sections are put into service, with greater coal firing, additional sections of the carbon-burn-up cell should be started.

As firing conditions become stabilized operation of the sulfur gas emission control system must be optimized. In the dry solids system, the limestone feed rate should be adjusted to achieve the desired stack gas SO_2 concentration. As the limestone feed rate is increased, more bed material may have to be extracted from the bed by increasing the feed rate of ash feeders extracting bed material. The fluid bed pressure drop is a measure of the bed level. In the wet scrubber system, the limestone feed rate should be increased to decrease the sulfur gas concentration of the stack. By doing this, more lime can be transferred to the slurry tank increasing the slurry flow through the scrubber.

On Line Operation

As the steam generator is operated to meet steam system requirements, several parameters should be monitored. These are:

Excess air - to obtain the most efficient operation, the excess air should be controlled to as low level as possible without increasing the amount of unburned coal or the amount of combustibles in the flue gas.

Bed Temperature - Bed temperatures in the four active beds and in each carbon burn up cell section can be used to detect uneven combustion rates within the bed. The exact cause of uneven bed temperatures requires analysis of all operating factors.

Flue Gas Temperatures Leaving the Boiler and Leaving the Economizer - Flue gas temperatures are a guide to the effectiveness of the heat transfer surface. The need to operate soot blowers may be indicated by rising flue gas temperatures.

Superheater Outlet Temperature - In the proposed design there will be terminal spray attemperator to control the steam temperature. Steam temperature leaving the superheater and leaving the attemperator should be observed.

Feedwater-Steam Flow - The steam and feedwater flow should be in a definite relationship. If this varies, the reason should be explained. Normally feedwater flow will exceed the steam flow due to blowdown. Any change from normal would indicate a malfunction of the system such as a tube leak.

Grid Air Resistances - For all fluidized bed sections including the fuel beds and carbon burn-up sections, monitoring the grid air flow resistance is a guide to the proper distribution of fuel and air to the beds. An increase could indicate pluggage. Bed and grid air resistance is a guide to the proper operation of the ash extraction and ash handling equipment.

Coal Feeder and Fuel System Temperature - Sealing and transport air is preheated to prevent condensation and to enhance drying in the coal system. Too low temperatures should be avoided and high temperatures could indicate coal "hanging up" and oxidizing in the system.

On Line Maintenance:

Fuel System - Normal on line maintenance of the fuel system will be greasing the coal feeder and other moving parts and the replacement of wear parts such as the coal injection nozzles.

In regard to the coal injection nozzles, compressed air aspirators allow the nozzles to be removed from the bed when one section is shut down. The primary coal splitter is adjusted to allow the desired one of the four discharge pipes to be shut-off. Coal flow to that section and also air flow is shut off. Nozzle isolating valves are closed to isolate air flow and the coal and air lines are removed. Compressed air is opened on the nozzle aspirator and a nozzle is removed and replaced by a spare. The procedure is repeated for each of the eight nozzles in the bed section. After nozzle replacement, the bed section can then be returned to service. Replacement of eight nozzles with spare nozzles will take less than 4 hours.

Water and Steam System - The services of a competent feedwater consultant should be obtained to recommend and to supervise a comprehensive program of internal and external feedwater treatment. Prior to initial operation, the boiler should be chemically cleaned internally. Routine internal cleaning may be required after internals of operation. During operation, boiler water analysis and feedwater quality must be checked frequently.

Ash Handling System - The Fluidized Bed combustion steam generator requires continuous operation of the ash handling systems. Routine lubrication and checking of the ash feeders, blower, valves, and electrical system are requisites to good operation.

Instruments and Controls - Instrument lines should be blown out frequently, water columns blown down, and instrument and control cleaning should be done routinely.

Shutting Down:

Removing one bed from service - For load reductions or for removing the unit from operation, one fuel burning bed section should be shut down at a time.

To shut down a bed section, the fuel to a bed section should be shut off by closing the valve at the primary splitter. Simultaneous compensation should be made in the fuel and air flows to the other bed sections to maintain stable steam generation.

The bed temperature will drop upon the loss of fuel input. When the bed temperature is less than 1000F, the combustion air flow to that bed section should be shut down. At the same time, the operation of the ash feeder for that bed section should be stopped. Operation of the carbon-burn-up cell should be monitored to observe whether adjustment of the number ash injection nozzles should be made.

If the steam generator is to be taken out of service; starting up procedures should be reversed. The coal bunker level should be lowered and the fuel beds should be removed from service, leaving only the bed section nearest the start-up burner operating. The start up burner should be ignited prior to shutting down the last bed section. When coal feed is shut off and as the bed material cools, it may be expedient to transfer it to the ash handling system.

In the case of the dry solids system, the beds should be fluidized and the bed ash feeders started. Eventually all bed materials will be extracted to the ash silo. The start-up burner should be kept in service at this time to maintain the steam generator well above flue gas condensation temperatures.

The bed ash feeders should be maintained in service in the wet scrubber system to empty the bed. In wet scrubber system, the bed will be lost to the slurry tank and thickner. (The ash dewatering system may not be shut down until lime and lime-sulfur precipitates have been reduced to low levels.)

When the bed material has been removed, the startup burner may be shut off removing the steam generator from service. The steam generator may be cooled by continuing to operate the forced draft fan and the recirculating pump. Venting steam will also help reduce the boiler pressure. The cooling rate should be limited so that there are no undue thermal stresses. A cooling rate of 100F per hour, the same as the heating rate should be followed.

The recirculating pump should be kept running until the steam generator is cooled to below 200F and is ready for draining. As long as the recirculating pump is operating, the steam drum water level should be maintained.

Emergency Operations

There are several principal emergency conditions where definite operating steps should be followed to minimize the chance of damage:

Loss of combustion air flow -

A loss of combustion air flow should trip all fuel feed to the steam generator including the secondary splitter vibrator. The sealing air fan should remain in service along with the ash system blowers. The electrical sequencing system for ash feeders should be tripped. The boiler recirculation pump should remain in service and the water level in the steam drum should be maintained. The superheater vent should be opened as soon as the boiler pressure drops below the line pressure.

Loss of Boiler Recirculation Pump -

Loss of the boiler recirculation requires that the fuel air flow be stopped immediately. The seal air fan should continue to operate along with ash handling and the gas cleanup equipment. The steam drum water level should be maintained. As the boiler pressure drops below the line pressure, the superheater vents should be opened.

It is thought that there is enough thermal circulation capability to protect the submerged tubes if the fuel feed were tripped and residual combustibles are burned up.

Loss of Fuel Feed -

The loss of fuel feed is not in itself an emergency condition since the forced draft fan and other equipment may continue to operate. Danger exists when the condition is corrected and when the fuel flow is re-established. If the bed temperature were less than the normal ignition conditions, a normal, section by section, start up should be undertaken. Care in this case should be taken to prevent superheater damage due to "over firing" upon restarting.

Loss of Steam Drum Water Level -

The sudden loss of steam drum water level is usually due to the loss of the feedwater system but it could be caused by a tube failure. In any case, fuel feed should be stopped immediately and the steam generator cooled as rapidly as possible. The forced draft fan should be kept running. The boiler recirculating pump operation should be watched closely. At any indication of cavitation in the recirculating pump, such as rapidly varying pump motor current, the pump should be shut down.

To restore the steam generator to operation after the drum level has dropped to an unknown point, it should be taken out of service. Drum level should be restored by feeding water at a low rate (5 to 10% of the maximum steam flow). The steam generator can then be restarted utilizing the normal procedure where natural gas is fired prior to firing coal.

Tube Leak -

If there is an unexplained disparity where feedwater flow exceeds steam flow, a tube leak may have occurred. Tube leaks are likely to cause extensive damage if operation continues. The steam generator should be shut down as soon as possible if a leak is detected.

Any tube leak should be considered a serious situation. Minor leaks may not force an immediate outage, however, a tube failure may necessitate tripping the steam generator because of the inability to maintain the steam drum water level.

Loss of Bed Fluidization -

If there is a sudden rise or a sudden loss of forced draft fan pressure, it may indicate the loss of fluidization over the grid. Pluggage may be responsible for the rise in pressure losses (increasing forced draft fan pressure) or a blow through in a bed section could cause a loss of fluidization and less resistance. Both conditions should be treated as an emergency and the unit should be shut down until the situation can be explained and corrected.

A loss of fluidization due to bed behavior will leave the grid exposed to high temperatures because of continued burning. Also, where bed temperature control is lose, there may be clinkering and pluggage of the grid.

APPENDIX 4

STEAM GENERATING SYSTEMS • ENERGY RECOVERY SYSTEMS
POLLUTION CONTROL SYSTEMS
1422 EAST AVENUE
ERIE, PA., U. S. A. 16503 • TELEPHONE (814) 452-6421

PROPOSAL-CONTRACT

NO. DATE

TO. Sample Proposal

For and in consideration of the hereinafter named amount,

The ERIE CITY ENERGY DIV., ZURN INDUSTRIES, INC. (hereinafter called the COMPANY) proposes to furnish to the PURCHASER, in accordance with the following specifications and general conditions, the machinery, material and/or equipment (hereinafter referred to as "equipment") and/or services described below:

A steam generator system utilizing fluidized bed combustion and one of two different techniques to control stack emissions.

In either case, the steam generator will be designed to produce 250,000 pounds of steam per hour at 750F and 600 psig when supplied with 250 F feedwater. Fuels will be bituminous coal, with natural gas for starting up only.

The steam generator will be adapted to utilize either:

Dry limestone solids absorption to control sulfur dioxide emission and an electrostatic precipitator to control particulate emission, or,

Wet limestone slurry scrubbing to control both sulfur dioxide and particulate emission.

		ECED Drawings
Reference Drawings:	Dry Solids system:	FBB-23 and FBB-30
	Wet scrubber:	FBB-29 and FBB-31

Proposal
Form 101B
2M-8-69

ERIE CITY—PROPOSAL—CONTRACT

SAMPLE PROPOSAL

INDEX TO EQUIPMENT DATA AND DESCRIPTIONS

Issue No. 1
April 15, 1971

ERIE CITY ENERGY DIVISION
ZURN INDUSTRIES INC.

ERIE, PENNSYLVANIA

STEAM TRIM LIST

FOR

FLUIDIZED BED BOILER PROPOSAL

ITEM	QUANTITY	EQUIPMENT
1	1	Safety valve - Consolidated #1811 set @ 665# 600# R.F. flanged inlet, 2-1/2" - 150# R.F. flanged outlet. Suitable for outdoor installation. For superheater.
2	1	Safety valve - Consolidated #1811 set @ 700# 600# R.F. flanged inlet, 2-1/2" - 150# R.F. flanged outlet. Suitable for outdoor installation. For boiler.
3	1	Safety valve - Consolidated #1811 set @ 710# 600# R.F. flanged inlet, 3" - 150# R.F. flanged outlet. Suitable for outdoor installation. For boiler
4	1	Water column - Reliance WM-900-C8-EA16R with high and low whistle alarm and fuel cutout 1 ph, 60 cy., 110 V.
5	2	Sets gage valves - Reliance SG-860 with chains and pulls.
6	2	Mica protected flat glass insert - 12-1/2" visibility Reliance FG-909
7	2	Direct vision hood and illuminator assy. - Reliance FG-890, each consisting of one FG-90 illuminator and one FG-890 direct vision hood. 1 ph., 60 cy., 110 V.
8	1	Aux. low water cutoff - Reliance EA-100S levalarm 1 ph., 60 cy., 110 V.
9	1	Steam gauge - Ashcroft #1079-D, 0-1500# range, 6" dial 1/2" male back conn. flush mounted.
10	1	Feedwater regulating valve - 3" Fisher 600# Std. R. F. flanged ends.
11	6	Soot blowers - Diamond Model G9B vertical type, manually operated.
12	1	Superheater thermometer, Duro industrial thermometer 9" scale, 200° - 950° F. range, straight form with 304 stainless steel well.

ITEM	QUANTITY	EQUIPMENT
13	1	Superheater test well - Duro, stainless steel type 304
14	1	Water column drain valve - 3/4" Edward #848
15	2	Water gauge drain valve - 3/8" Edward #838
16	1	Steam gauge shut-off valve, 1/2" Edward #152
17	1	Steam gauge test conn. - 1/2" Edward #848
18	2	Drum level transmitter shut-off valves - 1" Edward #848
19	2	Drum level transmitter drain valves - 1/2" Edward #848
20	1	Blow-off unit - 2" Edward Tandem #641-643 - 600# R.F. flanged
21	1	Chemical feed shut-off valve 3/4" Edward #848
22	1	Cont. B. D. shut off valve - 1" Edward #848
23	1	Superheater drain valve - 1" Edward #848
24	1	Superheater vent and drain valve - 1-1/2" Edward #848
25	1	Superheater supply pipe vent valve - 1" Edward #848
26	2	Soot blower drain valve - 3/4" Edward #848
27	1	Soot blower main header shut off valve - 2" Powell #6033 600# R. F. flanged with chain and guide
28	1	Feedwater stop valve, 4" Powell fig. 6031 600# R.F. flanged
29	1	Feedwater check valve - 4" Powell figure 6061 600# R.F. flanged
30	2	Feedwater regulator isolation valves - 3" Powell 6003, 600# R.F. flanged
31	1	Feedwater regulator by-pass valve - 4" Powell 6031, 600# R. F. flanged.

APPENDIX #5

OHIO PITTSBURGH NO. 8 SEAM COAL

(Source of data: USBM, Pittsburgh, Pa.)

SAMPLE: Run of Mine - As Received

PROXIMATE ANALYSIS (wt %):

Moisture	3.3
Volatile Matter	39.5
Fixed Carbon	48.7
Ash	8.5
	100.0

ULTIMATE ANALYSIS (wt %):

Moisture	3.3
C	71.2
H	5.0
O	6.4
N	1.3
S	4.3
Ash	8.5
	100.0

GROSS HEATING VALUE: 13000 Btu/lb.

NET HEATING VALUE: 12500 Btu/lb.

ASH ANALYSIS (wt %):

SiO_2	45.3
Al_2O_3	21.2
Fe_2O_3	27.3
TiO_2	1.0
P_2O_5	0.11
CaO	1.9
MgO	0.6
Na_2O	0.2
K_2O	1.8
SO_3	0.7
	100.1

FUSIBILITY OF ASH:

Initial Deformation Temperature	2080°F
Softening Temperature	2230°F
Fluid Temperature	2420°F

PARTICLE DENSITY:

Coal - - $\sim$1.4 gm/cc
Ash - - $\sim$2.8 gm/cc

GRINDABILITY (Hardgrove): 50 - 60

FREE SWELLING INDEX 5-5.5

APPENDIX #6

DRY SOLIDS SYSTEM - MATERIAL BALANCE RELATIONSHIPS

Unburned carbon loss:

Assume:

- 90% of carbon to primary collector is refired in carbon burnup cell.
- 90% combustion efficiency in carbon burn up cell

Therefore:

$$(1-0.81)\ \frac{0.0712}{10,000} \times 14,300 = \frac{193 \text{ Btu loss}}{10,000 \text{ Btu input}}$$

Particulate Carryover to Primary Collector:

Assume:

- 13% of carbon fired is passed on to primary collector
- All ash in fuel is passed on to primary collector
- 3% of limestone feed as calcium sulfate

Therefore:

$$0.13 \times 0.712 \times \frac{10,000}{13,000} = 0.0712\ \#\ \text{carbon}/10,000\ \text{Btu input}$$

$$0.085 \times \frac{10,000}{13,000} = 0.0652\ \#\ \text{ash}/10,000\ \text{Btu in}$$

$$0.03 \times 18.75 \times 0.0331 \times \frac{136}{100} = \underline{0.0253\#}\ CaSO_4/10,000\ \text{Btu in}$$

Dust to primary collector = 0.1617# solids/10,000 Btu in.

Limestone feed to fluidized bed:

Assume 6 times stoichimetric amount

or

$$6 \times \frac{100}{32} \times \frac{4.3}{100} \times \frac{10,000}{13,000} \times \frac{1\,*}{0.97} = \begin{array}{l}\textbf{0.64\# Limestone/10,000 BTU In} \\ 0.62\#\ CaCO_3/10,000\ \text{Btu In.}\end{array}$$

Limestone Reactions in fluidized bed (based on sulfur in fuel):

Calcium sulfate produced:

$$0.9 \times \frac{136}{32} = 3.82$$

Calcium oxide produced:

$$(6.0-0.9) \times \frac{56}{32} = 8.92$$

Carbon dioxide:

$$6 \times \frac{44}{32} = 8.25$$

Sulfur dioxide unreacted

$$0.1 \times \frac{64}{32} = 0.2$$

To find the reaction products as listed above, multiply the factors by the sulfur in the fuel for the fuel used in design fuel, there is 0.033#/10,000 BTU Input (0.033#/10,000 Btu).

*** Limestone contains 97% CALCIUM CARBONATE for simplification.**

Solids extracted from the Bed:

Assume:

Carbon is negligible

Calcium sulfate that is not elutriated to primary collector

Calcium oxide from limestone.

Therefore:

$8.92 \times 0.0331 = 0.295\#$ Calcium oxide/10,000 Btu Input

$$\left(3.82 - 0.03 \times 18.75 \times \frac{136}{100}\right) \times 0.0331 = 0.100\# \frac{\text{calcium sulfate}}{\text{10,000 Btu Input}}$$

Heat Loss due to chemical reactions:

Sulfation of lime (exothermic)

$$3.82 \times \frac{3 \times 10^6 \text{ Btu}}{\text{ton calcium sulfate}} \times \frac{0.033}{10{,}000} = 190 \text{ Btu/10,000 Btu Input}$$

Calcination of limestone (endothermic)

$$18.75 \times \frac{1.5 \times 10^6 \text{ Btu}}{\text{ton calcium carbonate}} \times \frac{0.033}{10{,}000} = 465 \text{ Btu/10,000 Btu In.}$$

Heat loss --------------------------- = 275 Btu/10,000 Btu In.

WET SCRUBBER SYSTEM - MATERIAL BALANCE RELATIONSHIPS

Limestone Reactions in Fluidized Bed (Based on sulfur in fuel):

Calcium sulfate produced:

$$0.9 \times \frac{1.2}{6} \times \frac{136}{32} = 0.765$$

Calcium oxide produced

$$\left(1.2 - 0.9 \frac{1.2}{6.0}\right) \frac{56}{32} = 1.785$$

Carbon dioxide produced

$$1.2 \times \frac{44}{32} = 1.65$$

To find weight flows of the reaction products multiply the factors abcve by the sulfur in the fuel. For the design fuel there is 0.033#/10,000 BTU input.

Limestone Feed:

Assume 1.2 x stoichiometric

Therefore:

$$1.2 \times \frac{100}{32} \times \frac{0.033}{10,000} = 0.124\#/10,000 \text{ Btu Input}$$

Carryover to Primary Collector:

Assume:

13% of Carbon fired

All ash in fuel

3% of limestone feed as calcium sulfate

Therefore:

$$0.13 \times 0.712 \times \frac{10,000}{13,000} = 0.0712 \text{ \# carbon/10,000 Btu in.}$$

$$0.085 \times \frac{10,000}{13,000} = 0.0652 \text{ \# ash/10,000 Btu in.}$$

$$0.03 \times 0.124 \times \frac{136}{100} = 0.0505 \text{ \# calcium sulfate/10,000 Btu in.}$$

Solids to Slurry Tank

Assume:

Carbon is negligible

All calcium oxide unreacted in bed

Calcium sulfate not elutriated to primary collector

Heat loss due to chemical reactions:

Sulfation of calcium oxide (exothermic)

$$0.765 \times \frac{0.033}{10,000} \times \frac{3 \times 10^6 \text{ Btu}}{2000\text{\# calcium sulfate}} = \frac{38 \text{ Btu}}{10,000 \text{ Btu Input}}$$

Calcination of calcium carbonate (endothermic)

$$3.75 \times \frac{0.033}{10,000} \times \frac{1.5 \times 10^6 \text{ Btu}}{2000\text{\# calcium oxide}} = \frac{93 \text{ Btu}}{10,000 \text{ Btu input}}$$

Heat Loss = 55 Btu/10,000 Btu input

Dewatered Slurry to Disposal:

Unburned carbon

$$\frac{193.}{10,000} \times \frac{1}{14,300} = \quad 0.0135\#/10,000 \text{ Btu In.}$$

Coal ash:	0.0652#/10,000 Btu In.
Calcium sulfate:	0.126#/10,000 Btu In.
Calcium Carbonate:	0.0331#/10,000 Btu In.
Total (dry)	0.2378#/10,000 Btu In.

APPENDIX #7

FLUIDIZED BED COMBUSTION STUDY
INDUSTRIAL STEAM GENERATOR DESIGN
STUDY OF TURNDOWN PARAMTERS

One of the fluidized bed design features will be the requirement for sectionalized beds to facilitate turndown. Sectionalizing the bed to allow partial shutdown appears to be a requisite because of the narrow range of possible bed operating temperatures. A range of 1400 to 1650 F. is anticipated at this time. It is believed that below 1400 F, the combustion in the bed will become unstable. Above 1650 F, limestone absorption of sulfur gases may be reduced.

Sectionalizing the bed is a feasible way to handle the turndown requirements, however, an increased number of sections will complicate fuel feeding systems, air ducting, and the combustion control systems. Other means may have to be used to control or to smooth bed temperature fluctuations. Adjusting the expanded bed depth by transferring bed material in and out of the fluid bed chamber is one method. Adjusting the excess air in the chamber is another method.

In a bed arrangement where the particle splash screen is said to help control bed temperature, a prediction of how expanded bed depth relates to bed temperature is not available. As a result, at this time, it is not reasonable to predict how effective lowering bed depth will be.

If a constant heat transfer coefficient is assumed, it is possible to develop a relationship of excess air and reduced heat input to bed temperature.

On Figure #1, curve #1 represents the effect of load reduction on bed temperature with a constant heat transfer coefficient. As is seen 1400 F. is obtained at about 70% load.

Curve #2 on figure #1 represents a means of turndown minimizing bed temperature fluctuations without passing through a minimum of 70% load on one bed section. A minimum load of 17.5% overall is indicated. The scheme shows that as the unit load is reduced one section at a time is taken out of service. It assumes uniform loading of the operating sections.

Curve #3 on figure #1 is another demonstration of the range of bed section loading over the unit load range. In this example, 4 sections are proposed with one section having been partitioned to two halves which can operate separately. A minimum rating of 70% is adhered to; but slightly more than 100% rating is seen to make a continuous operation sequence from the minimum of 17.5% unit load.

From these data it becomes apparent that sectionalizing alone is a difficult way to produce turndown.

Figure #2 shows two effects of increasing excess air at a constant load. The bed temperature is reduced by the "cooling" effect of increased air flow. On the other hand, the volume flow increases significantly. The simplifying assumption of a constant heat transfer coefficient in this case does not seem valid. Because the bed particle size-consist will change with volume flow, it appears likely that the heat transfer coefficient will vary. It may be, for example, that the negative slope of the bed temperature curve would increase in a more negative direction to produce a sharper temperature change.

Drawn from other experience, it is felt it is not a practical consideration to program the combustion control to have a load indexed step change flue gas analysis. Therefore, bed temperature control by means of excess air control, is not practical.

Conclusions at this point in the study of fluidized bed combustion steam generator turndown would be:

> Sectionalizing, alone, is not adequate to provide smooth control over the unit load range.
>
> Excess air control is not a practical means to smooth bed temperature variations in sectionalizing.
>
> Expanded bed depth change appears to be a valid way to smooth bed temperature control but the effects cannot be predicted sufficiently at this time.

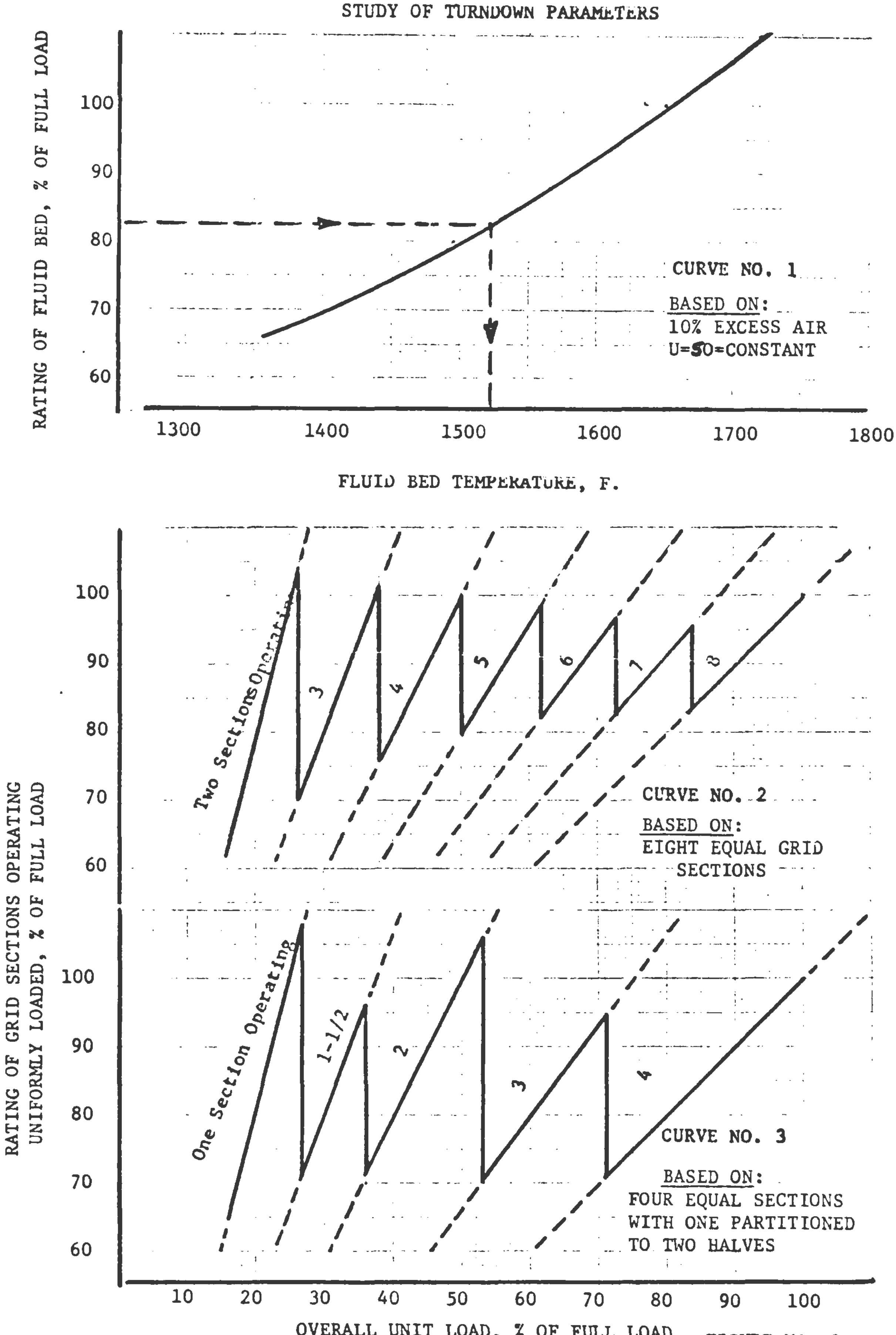


FIGURE NO. 1

ECED
ZURN IND
1-71

FLUID BED COMBUSTION
-- TURN DOWN PARAMETERS --

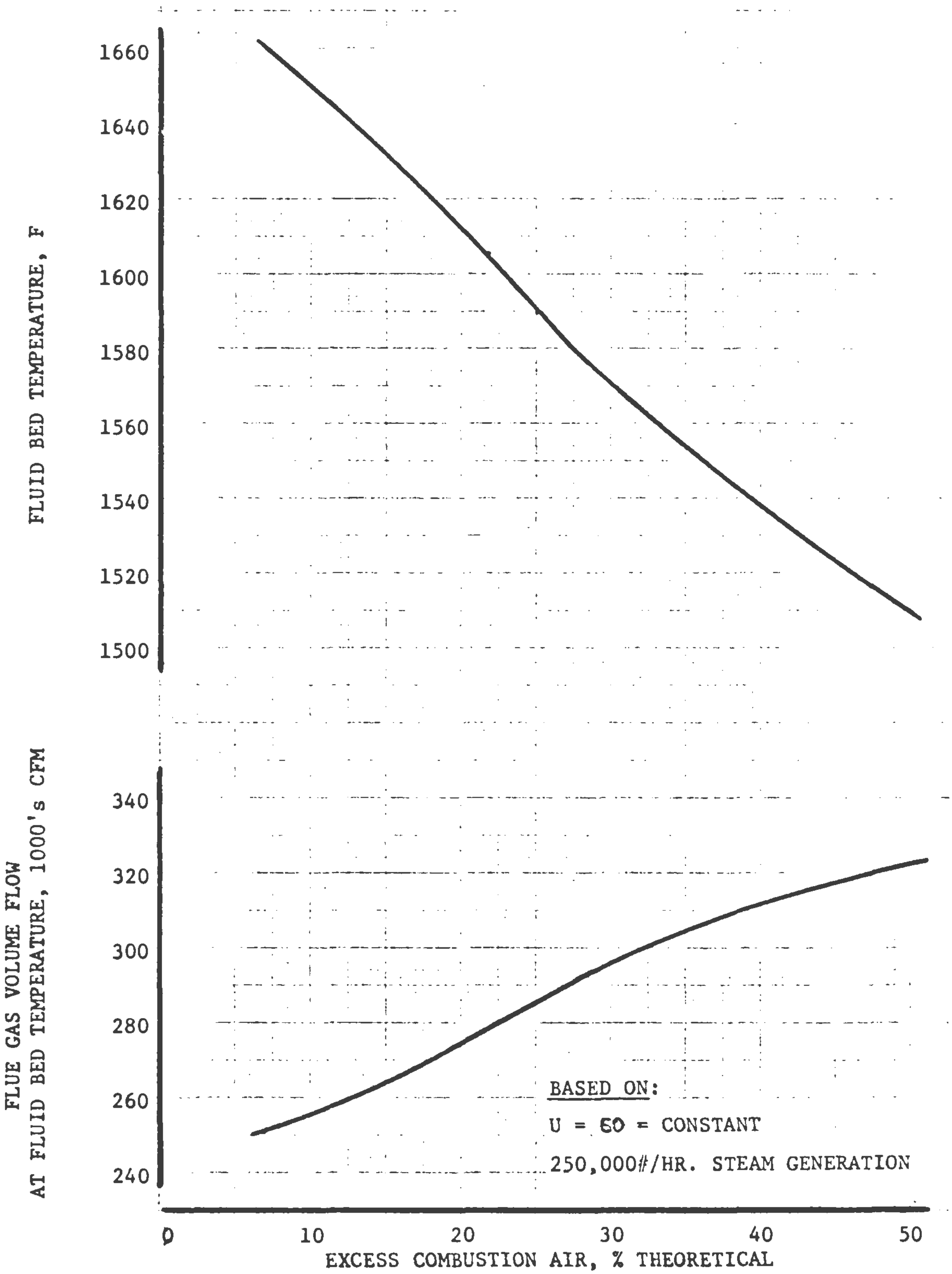


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FIGURE NO. 2

APPENDIX #8

SUMMARY OF DRAFT LOSSES

COMPONENT	AIR RESISTANCE DRY SOLIDS, SYSTEM	OR	DRAFT LOSS, IN. WG WET SCRUBBER, SYSTEM
Forced Draft Ducts	0.2		0.2
Fluidized Bed Chamber:			
Section Dampers	0.5		0.5
Perforated Plate	1.0		1.0
Grid	5.0		5.0
Bed	20.0		20.0
Mechanical Collector	5.0		5.0
Carbon Burn-Up Cell (Thru Flow)	0.5		0.5
Superheater	2.6		2.6
Convection Section	5.6		5.6
Economizer	2.0		2.0
Economizer Flues	0.2		0.2
Precipitator	0.5		-
Precipitator Flues	0.2		-
Wet Scrubber	-		16.0
Scrubber Flues	-		0.2
TOTAL	43.3		58.8

COMPARISON OF POWER REQUIREMENTS *

FOR EQUIVALENT CAPACITY, FIELD ERECTED,

STEAM GENERATORS FIRING

DIFFERENT FUELS

Steam Generator (See detailed description) in Appendix I	"A" Gas or Oil Fired	"B" Coal Fired spreader stoker	"C" Coal Fired Bed Dry Solids System	"D" Fluidized Combustion Wet Scrubber
Forced Draft Fan, HP	326	84	700	950
Induced Draft Fan, HP	-	341	-	-
Recirculating Pump, HP	-	-	200	200
Coal Crusher, HP	-	-	60	60
Slurry Pumps, HP	-	-	-	150
Total Power Requirements, HP	326	425	960	1360
Power Requirements, % of output **	0.29	0.37	0.86	1.21

* Only major operating horsepower requirements are shown for the different systems.

** Where a horsepower hour is 2544 Btu and output is 289 million Btu per hour.

REVIEW OF STEAM GENERATORS

RECIRCULATING PUMP CHARACTERISTICS

To point out the recirculating pump requirements and the system characteristics, graphical solutions were used. For each section of the recirculating pump system including suction, discharge and distribution piping upstream of the heating surface and the collecting and connecting lines to the steam drum were calculated. In this a constant heat absorption for submerged tubing, which is the "worst" or highest case was assumed. Curves were developed showing resistance to flow versus flow for steaming and non-steaming tubes within the bed sections. This data was then combined to show resistance to flow versus overall flow for five different cases from out of service conditions to full load (four sections operating).

The intersections of a "supposed" pump characteristic with these curves determines the operating points.

It may be developed that a non-steaming section will act as a bypass of the steaming section. The lowest flow in a steaming section will occur when three sections are not fired and one bed section is.

The operating point is determined by the intersections outlined above and can be used to determine flow through the non-steaming and steaming sections. Total flow and head can be used to find the pump power.

Curve #1 shows the plots of the system resistance and the pump curve selected.

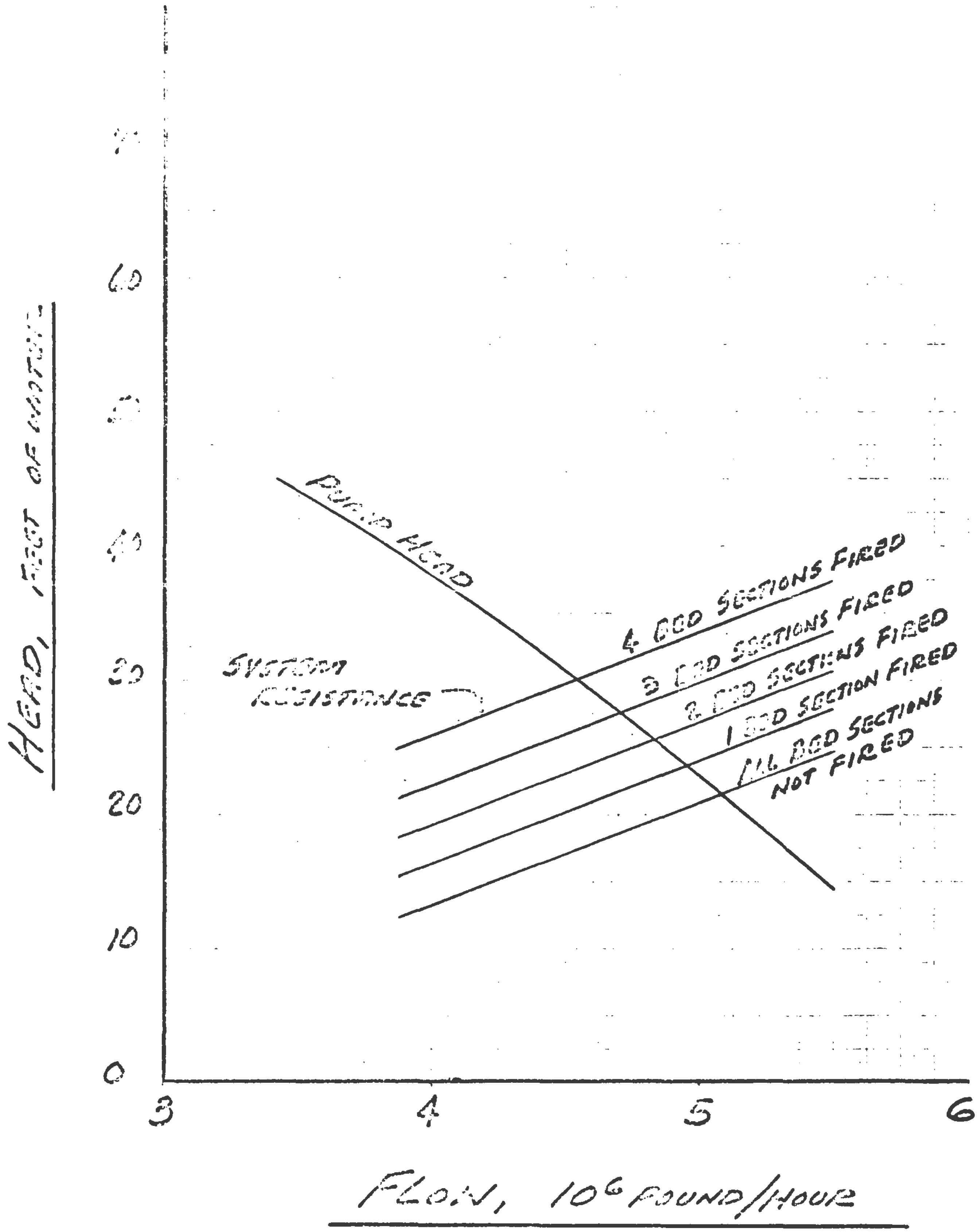
From the intersection of the pump curve, it is noted that:

CASE	PRESSURE DROP PUMP HEAD, FT WATER	TOTAL FLOW, GPH	PUMP HORSEPOWER
All sections out	21.5	12,900	137.5
One section fired	23.5	12,600	147
Two sections fired	25.5	12,250	155
Three sections fired	27.5	12,000	163
Four sections fired	30.0	11,600	172

Using this in relation to the resistance of a steaming circuit, it is developed that:

CASE	FLOW/TUBE, #/HR. (Unfired Section)	STEAM QUALITY LVG. BED, % STM BY WEIGHT
One section fired	2450	5.70
Two sections fired	2600	5.3
Three sections fired	2750	4.9
Four sections fired	3000	4.3

It is concluded by this analysis that valving or modulating flow control is not necessary to insure adequate circulation of tubes in the fluidized bed. The remainder of the steam generator is cooled by natural circulation. An overall circulation rate of 15 to 1 for the natural circulation sections is obtainable and would be valid design point.



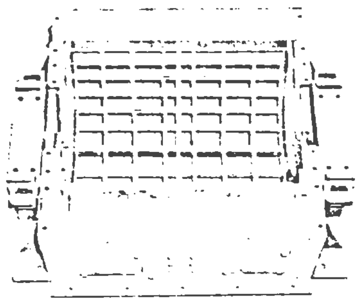
HEAD, FEET OF WATER
FLOW, 10^6 POUND/HOUR
PUMP HEAD
SYSTEM RESISTANCE
4 BED SECTIONS FIRED
3 BED SECTIONS FIRED
2 BED SECTIONS FIRED
1 BED SECTION FIRED
ALL BED SECTIONS NOT FIRED
0
10
20
30
40
60
3
4
5
6

APPENDIX IX

JEFFREY

Type AB All Steel Heavy Duty Hammermills

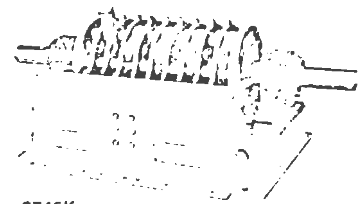
DIMENSIONS SPECIFICATIONS WEIGHTS



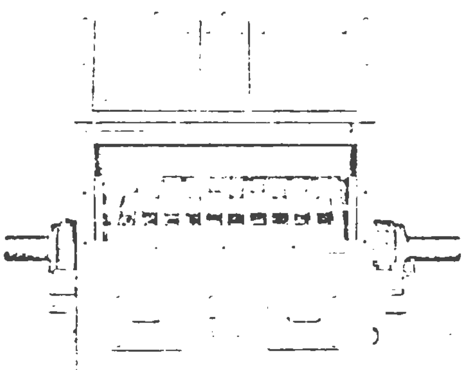
2816K

A rigid all steel base provides the mounting for the welded reinforced screen bars below the rotor.

View of the base showing the heavy duty rotor mounted on an alloy steel shaft and supported by spherical roller bearings.

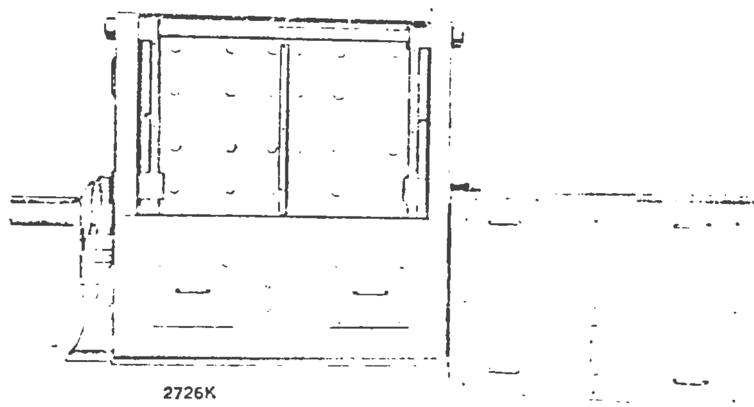


2746K



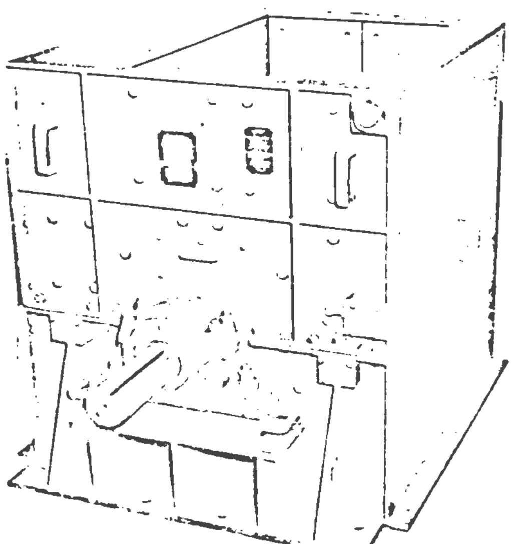
2776K

A hinged cover provides easy access to the metal trap.



2726K

View of rear side of pivoted breaker plate with cover removed.

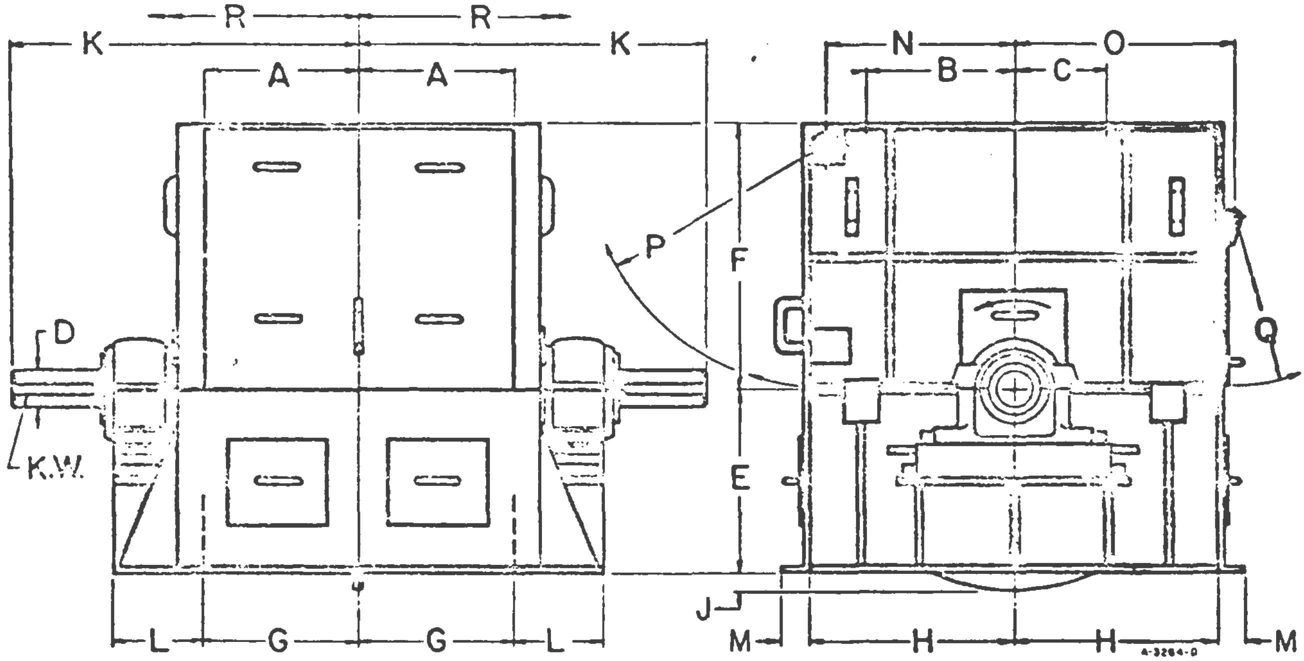


2756K

'ype 45AB Crusher. This rugged single direction unit incorporates the improvements and experience of over 73 years of designing, manufacturing and marketing Hammermills.

2786K

View looking into top of Hammermill showing rotor and liners. Liners are drilled and tapped to allow for safe and simple removal.



Unit Size	Dimensions in Inches																		Weight in Lbs.
	Feed Opening			Shaft															
	A	B	C	Dia. D	Keyseat K.W.	E	F	G	H	J	K	L	M	N	O	P	Q	R★	
44AB	16¾	18½	11½	4½	1¼ x ⅝	21	30	17¾	23½	2	41¾	10¼	3	21¾	25¼	27½	20½	60	8,000
45AB	20¾	18½	11½	4½	1¼ x ⅝•	21	30	21¾	23½	2	45¾	10¼	3	21¾	25¼	27½	20½	68	9,100
55AB	21⅝	20½	14⅜	5¾	1½ x ¾	28	35	22⅞	30½		53	13⅛	3	26½	32¾	32	33	70	15,000
56AB	26¾	20½	14⅜	5¾	1½ x ¾	28	35	22⅞	30½		58	13	3	26½	32¾	32	33	86	17,000
57AB	32	20½	14⅜	6¾	1½ x ¾	28	35¼	31¾	30½		68¼	17¼	3¼	26½	33¼	32	24	101	23,000
58AB	37¼	20½	14⅜	6¾	1½ x ¾	28	35¼	37	30½		73½	17¼	3¼	26½	33¼	32	24	112	26,000

★Space required to remove hammer pins (one side only).

SPECIFICATIONS:

Type AB Hammermills are of extra heavy construction. They are designed to handle large, heavy feed at high capacity rates and to produce acceptable end products for today's complex markets.

Units are of all steel rigid box construction.

Units are furnished with an integral metal trap and self-aligning roller bearings.

Replaceable liners are drilled and tapped, with bolts inserted from outside of unit, to prevent wearing away of bolt heads or nuts.

STOCK EQUIPMENT COMPANY
731 HANNA BUILDING - (216) 621-3054
CLEVELAND, OHIO 44115

REPLY TO:
PITTSBURGH OFFICE
3730 POPLAR AVE.
CASTLE SHANNON BOROUGH
PITTSBURGH, PA. 15234
(412) 343-8599

June 2, 1971

Erie City Iron Works
1500-1520 East Ave.
Erie, Pennsylvania

Attention: Mr. R. Winchell

SUBJECT: S-E-Co. Gravimetric Feeder for
Unknown Customer
S-E-Co. Ref. No. E-4656

Gentlemen:

In reply to your recent quotation request, we propose the following:

<u>ITEM 1</u>

1 S-E-Co. Gravimetric Feeder.

A detailed description of the Gravimetric feeder follows:

General description and principle of operation are described in the attached Bulletin No. 198.

The advantages to be gained through the utilization of Gravimetric Feeders are outlined in the attached reprint of "Combustion Control Utilizing Gravimetric Feeding of Coal" by Ralph M. Hardgrove.

Capacity of each feeder is 29,000 lbs. per hour of coal weighing approximately 50 lbs. per cu. ft. This is based on the capacity requirements stated in your inquiry.

The distance from center line of inlet to center line of discharge of feeder outlet hopper is 7'0". Inlet opening is 24" inside diameter.

The S-E-Co. Gravimetric Feeder is designed and built to discharge exactly 100 lbs. of coal per revolution of the head shaft. Head shaft speed and correspondingly the coal feed rate, are determined by an accurately controllable, variable speed drive.

Feeder body is designed to withstand a static internal pressure of 50 psi in accordance with the explosion requirements of the National Fire Prevention Association Code 60. All portions of the feeder in contact with the normal flow of coal are stainless steel or rubber. Bull's-eyes and lights are incorporated into the feeder body for inspection of the interior. Each bull's-eye and light has a special fitting for washing the interior glass surface to keep it clean. One separate window washer mounted on a cart is provided for the entire group of feeders as a portable source of water for the cleaning operation.

The rubber feed belt will be three ply, molded construction to keep sifting dust and dripping water from the feeder to an absolute minimum. The belt design features a rubber curb along each side to confine moisture and dust. It has a V-belt guide section on the inside to insure accurate tracking of the belt. Maximum operating temperature of the belt is 175°F.

The feed belt is removable through the access door provided at either end of the feeder. The belt will be carried on closely spaced idlers with anti-friction bearings. The tail pulley is mounted in conventional take-ups which include protected adjusting screws which are also used for tracking the belt. The adjustment is made from outside the housing.

The feeder is driven through a specially designed helical and worm gear reducer by a 1-1/2 H.P. magnetic slip clutch variable speed motor.

The variable feed rate is accomplished by the use of the magnetic slip clutch with its transistor control. This controller includes a tachometer feedback network and a DC power supply for the field of the magnetic slip clutch. It provides a close speed control with rapid and accurate response to the combustion control signal. A minimum speed adjustment is provided such that if the input signals fall below a certain point, then the feeder will operate at the set minimum speed.

The feeder is equipped with a drag type cleanout conveyor to remove dust accumulations from the bottom of the feeder body through the outlet. The cleanout conveyor consists of a specially designed malleable iron chain with stainless steel pins, manufactured by the Stock Equipment Company. Cleanout conveyor runs on 1/4" thick stainless steel pan. A separate 1/4 H.P. motor and gear reducer are employed to drive the cleanout conveyor. As the amount of the dust accumulation is extremely small, a timer is furnished to operate the cleanout conveyor for a preset period of time, usually 5 minutes for each hour of feeder running time.

For calibration purposes, 2 twenty-five pound field test weights are furnished.

All bearings are equipped for pressure lubrication. Where shafts pass through pressure shell, suitable seals against the internal air pressure

Continued....

are provided. To prevent dust from entering the weigh lever compartment, this compartment is pressurized.

Purchaser may provide a source of clean cooling air and sealing air at suitable pressure, which may be taken off at the discharge of the forced draft fans. The CFM required are slightly more than the loss through the seal leg up into the downspout, so that the flow of air and consequently, the dust at the outlet will be toward the feeder discharge and into the mill.

A separate free standing electrical equipment cabinet is provided for each feeder. Construction is NEMA 12. It contains circuit breaker, motor starters for the main feeder drive and the cleanout conveyor drive, control voltage transformer, relays, timer for the control of the clean-out conveyor, and the variable speed control for the magnetic slip clutch.

The feeders are provided with a conduit entry box having terminals for making the necessary electrical connection to external power supply and control wires. Feeder internal wiring, from motors and illuminating lights to their terminal blocks, is by S-E-Co. Each feeder has an integral control panel with a "Calibrate-Local Run-Off-Remote" selector switch and "Run-Wait" Pilot Lights. A large tachometer indicator to indicate rate of coal feed in pounds per hour and total coal integrator or counter to indicate the pounds of coal fed over a period of time, are furnished as separate items for mounting on purchaser's boiler control panel.

Each unit is equipped with pulsers which will give one pulse for every 100 lbs. of coal fed to both an integrator and a data logger. In the event purchaser desires to have additional pulses for data logging purposes for smaller units of coal, such pulses can be provided.

Two S-E-Co. Paddle Type Coal Stoppage Alarm Switches are furnished as an integral part of each feeder.

One paddle switch is installed at the discharge of the feeder to stop the feeder in the event flooding occurs in the hopper or the downspout to the mill. Stopping the feeder reduces the amount of coal within the system which must be removed in order to clear the stoppage, and it also prevents damage to the feed belts.

The second paddle switch is located over the feed belt to give indication of loss of coal flow to the feeder. This switch is also used in the internal circuitry of the feeder to prevent accidental improper calibration.

Continued....

Features of the S-E-Co. Gravimetric Feeder are:

1.) Assures reliable coal flow because the coal is brought down to the belt through a flow passage 24" in diameter. This large inlet provides good coal flow conditions even with fine, wet, sticky coal. The 24" width of coal stream is maintained through the entire feeder with no obstructions projecting into the coal stream at any point.

2.) Has a very uniform rate of feed because the rubber belt discharges coal at a fairly smooth rate over the head pulley.

3.) Has proper coal flow conditions to make accurate weighing possible. Accurate weighing can be achieved only if rapid, sporadic fluctuations in the effective coal density resulting from unevenness of the coal stream are avoided. The S-E-Co. Gravimetric Feeder provides the requisite high and very nearly constant filling of the discharge area because the co-efficient of friction between the rubber belt and the coal is large, because the width of coal on the belt is 24" and nowhere restricted and because the coal depth is approximately 4-1/2".

4.) Is highly reliable because it employs a heavy duty, molded endless belt, features rugged construction throughout and is designed with careful attention to every detail.

5.) Requires low maintenance because the endless belt lasts many years barring severe accident, because all surfaces normally in contact with the flow of coal are made of stainless steel or rubber, and because the unit is designed for easy and quick dis-assembly and re-assembly.

6.) Has variable speed magnetic slip clutch drive which gives very close feed rate control with a rapid and accurate response to the combustion control signal.

APPROXIMATE WEIGHT-----------------7,425 lbs.

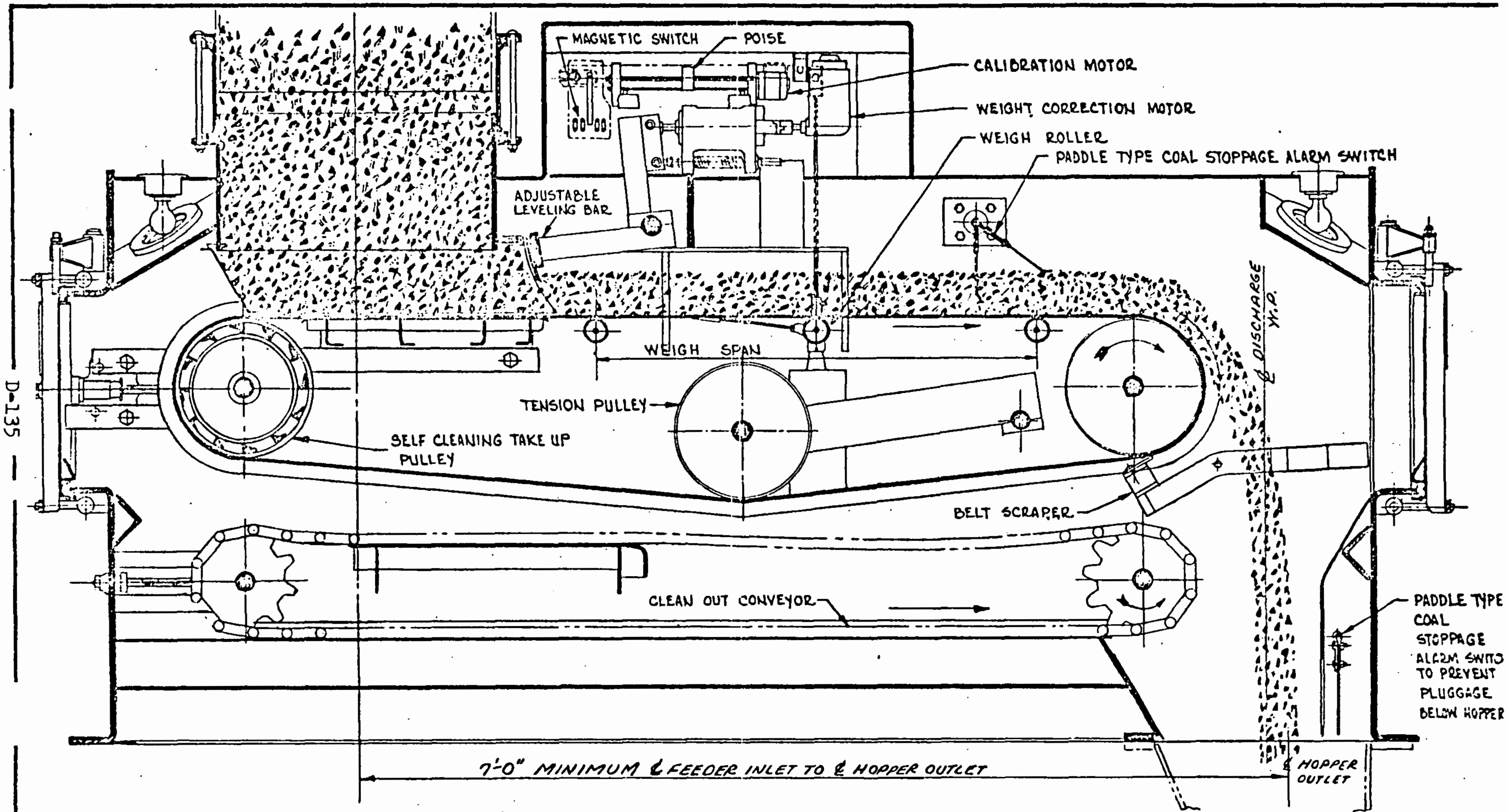
As additional equipment, we offer:

ITEM 2

1 Electric Tachometer Generator.

The tachometer generator is suggested to provide a feedback signal from the feeder to combustion control.

Continued....



D-135

S-E-Co. GRAVIMETRIC FEEDER

STOCK EQUIPMENT CO
CLEVELAND OHIO

LB 3380

REVISED 2-16-71
REVISED 1-6-65

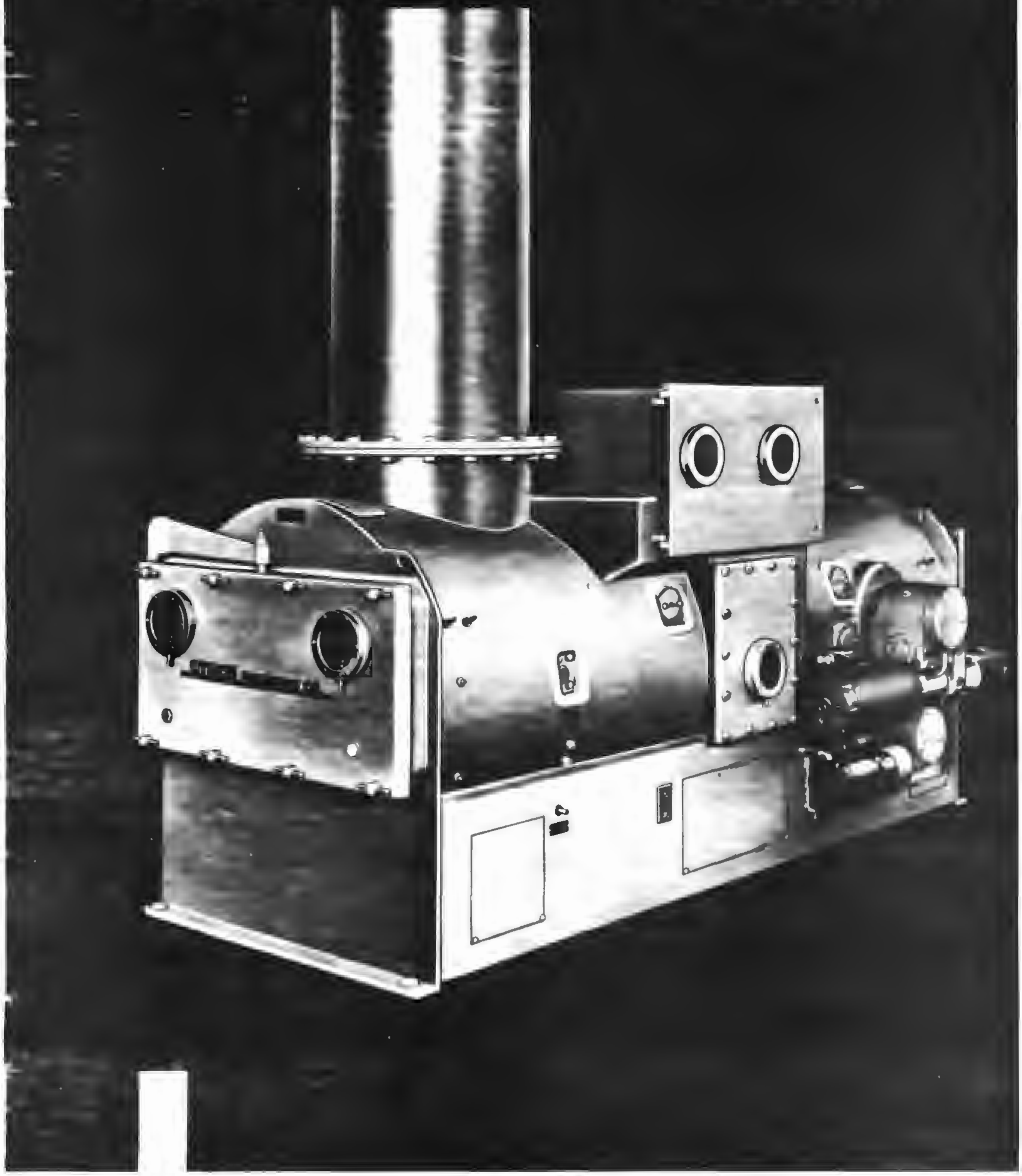
The generator is a General Electric D.C. Unit, Model GE 5BC 46AB 1590, 50 V/1000 RPM steady state, long time drift with 1.0 megohm resistance load, .05%/24 hour period.

The generator would be mounted on the belt drive reducer, and driven at motor speed.

APPROXIMATE WEIGHT-------------------35 lbs.

F. O. B. point, terms, delivery and other conditions of sale are as outlined in Form 260, which is page 6 of this proposal.

S·E·Co. GRAVIMETRIC FEEDER



STOCK EQUIPMENT CO.

HANNA BLDG. CLEVELAND, O.

D-137

THE STORY OF S-E-Co. GRAVIMETRIC FEEDER

S-E-Co. GRAVIMETRIC FEEDERS are designed to feed coal by weight, instead of by volume, to pulverizers and cyclone burners. Since feeding by weight is more accurate, these feeders enable more precise control of fuel-air ratio. They also make it possible to divide the total coal flow equally among the several firing units of a single boiler.

The S-E-Co. Gravimetric Feeder is designed to discharge exactly 100 pounds of coal for each turn of the head shaft. Head shaft speed is equivalent, therefore, to the rate of coal feed and can be expressed as pounds of coal per minute or pounds of coal per hour as desired. Total turns of the feeder head shaft times 100 equals total pounds of coal fed during any given period. Instruments can be provided to indicate coal flow feed rate and total coal burned.

The feeder body is a cylindrical housing designed to meet the National Fire Protection Association Code No. 60 for withstanding an explosion pressure of 50 p.s.i. Normal internal working pressure is slightly above or below atmospheric.

Each feeder is driven by a variable speed motor. The control for that motor balances the speed of its output shaft against an electrical voltage furnished by the combustion control. This assures a coal feed rate that corresponds exactly to the requirements of the combustion control.

Fig. 1 below shows cross-section of entire feeder.

THEORY OF DESIGN. The operating theory of S-E-Co. Gravimetric Feeders follows:

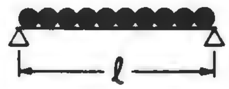


Fig. 2

Fig. 2 represents a beam resting on two supports at a distance "ℓ" apart. If ten units of weight were uniformly spaced along the length of the beam, then the load carried at each support would be five units.

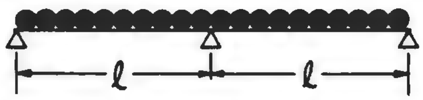


Fig. 3

If the beam were made twice as long, as shown in Fig. 3, and three supports provided at a distance "ℓ" apart, and if ten units of weight were applied to each span, then the amount of load carried by the outside supports would be five units, but the central support would carry ten units of weight. This is true provided that the supports are either in a dead level plane or the beam is non-continuous or flexible over the center support.

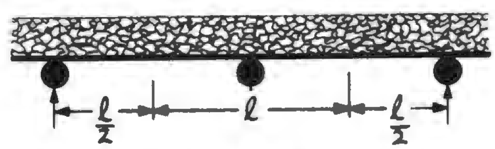
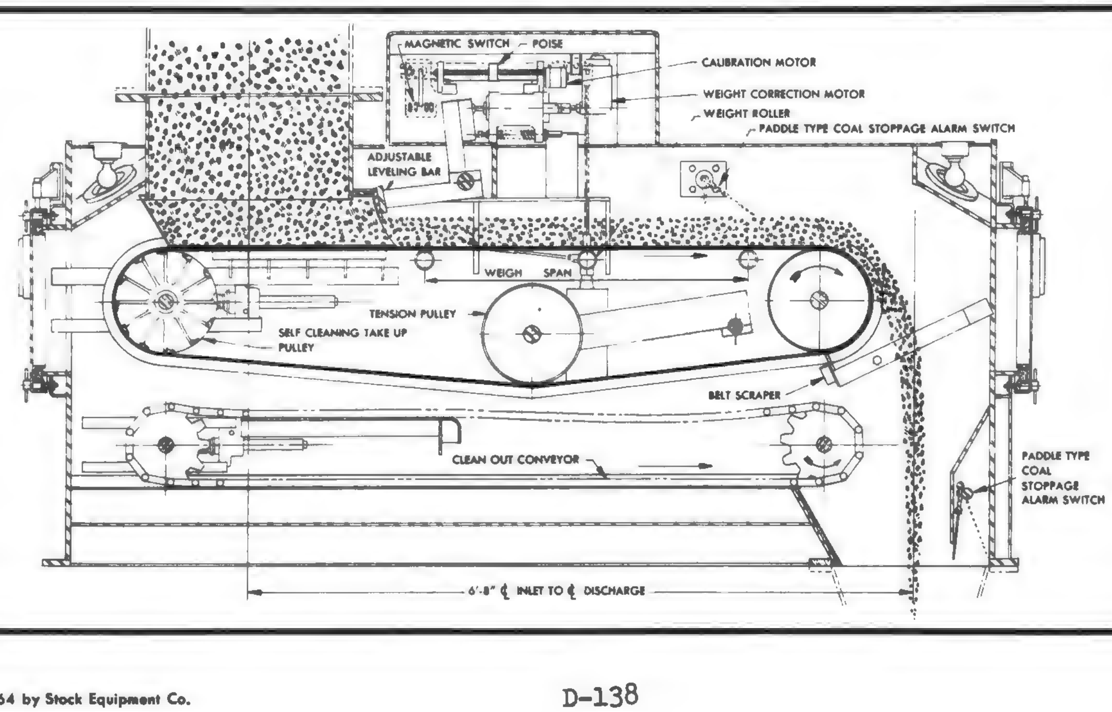


Fig. 4

Now if a short length of belt conveyor is supported by three rollers in a level plane, as shown in Fig. 4, and

Fig. 1



if the belt is loaded uniformly with coal, then the center roller will carry one half the load and the end rollers will carry one quarter of the load each.

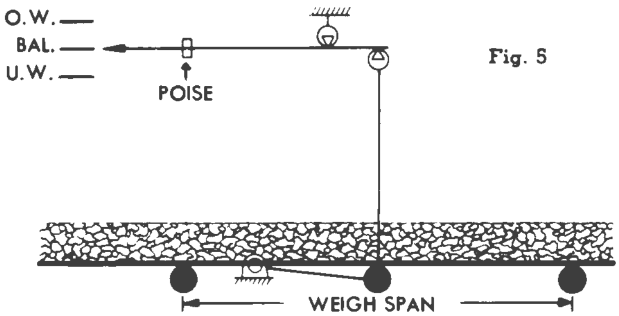


Fig. 5

The load on the weighing roller can be balanced with a simple lever scale such as diagramed in Fig. 5. The poise can be moved out or in so that the lever system comes to balance, and the actual weight on the weighing roller determined. If for any one setting of the poise the weight on the weighing roller is greater than the correct amount, then the weigh lever will indicate "overweight." Conversely, if it is less, the weigh lever will indicate "underweight."

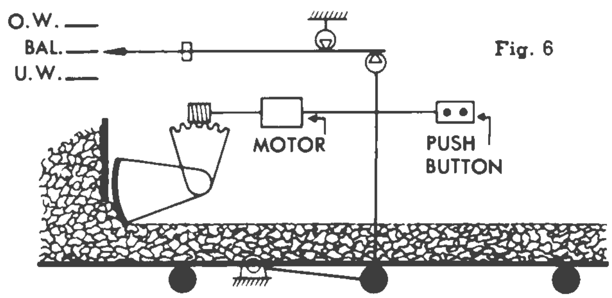


Fig. 6

The weight on the weigh lever should always be constant. Therefore, it is necessary to vary the depth of the coal on the belt as the weight per cubic foot of the coal varies. This can be accomplished rather simply as shown in Fig. 6. The leveling bar is operated through a gear reducer so that it can be raised or lowered by means of a motor. In the event of an "underweight" condition, the operator pushes a button to operate the motor and raise the leveling bar. For "overweight" the operator pushes the button to lower the leveling bar and thus reduce the weight.

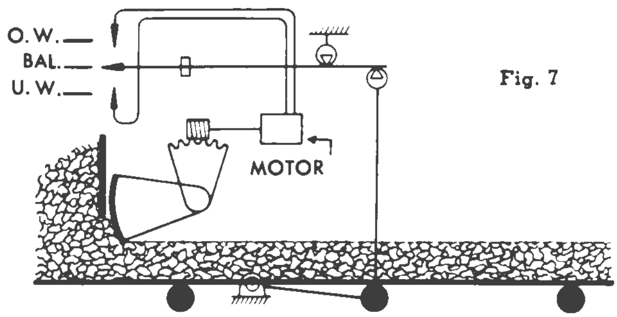


Fig. 7

However, since the weight per cubic foot of coal is continually changing, the operator would have to watch the balance indicator at all times. The job can be done automatically by means of "overweight" and "underweight" contacts to control the leveling bar motor. This is all shown in Fig. 7.

In the S-E-Co. Gravimetric Feeder, the length of the weigh span is equivalent to the belt advance for one turn of the head pulley. The normal weight of coal on the belt in the weigh span is equal to 100 pounds. Therefore, the weight on the weighing roller is normally 50 pounds.

Should this weight be greater than 50 pounds, the "overweight" contact closes, operating the motor driven leveling bar to decrease the volume of coal and thus bring the weight down to 50 pounds. On the other hand, if the weight is less than 50 pounds, then the "underweight" contact is made and the leveling bar motor will operate to increase the volume of coal and thereby bring the weight back up to 50 pounds.

CALIBRATION

Calibration of the S-E-Co. Gravimetric Feeder is very simple. The poise or movable weight on the weigh lever is moved out or in by means of the calibration motor and a screw. A selector switch allows the operation of this motor to be controlled by the "overweight" and "underweight" contacts.

If a 50 pound test weight is hung on the weighing roller (25 pounds at each end) and if the unit is operated for a short period without coal on the belt, the calibration motor will be actuated through the "overweight" and "underweight" contacts so as to move the poise to a position of balance. This simple method of calibration is indicated in Fig. 8. No tare adjustment is required because the method of calibration automatically takes into consideration the tare weight of the belt.

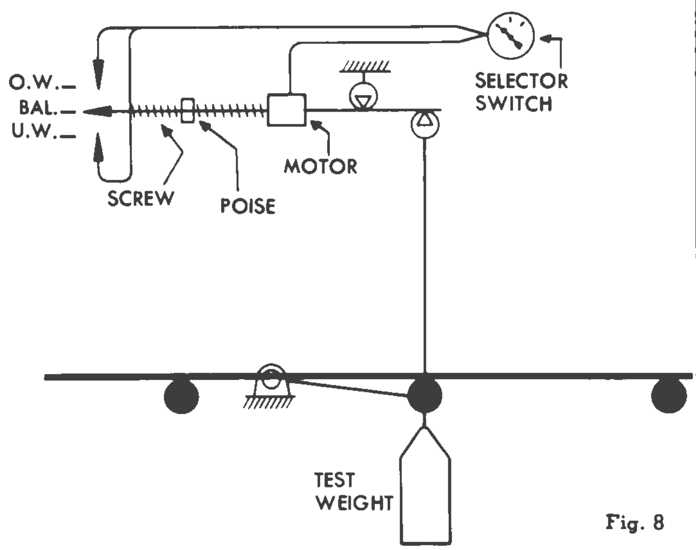
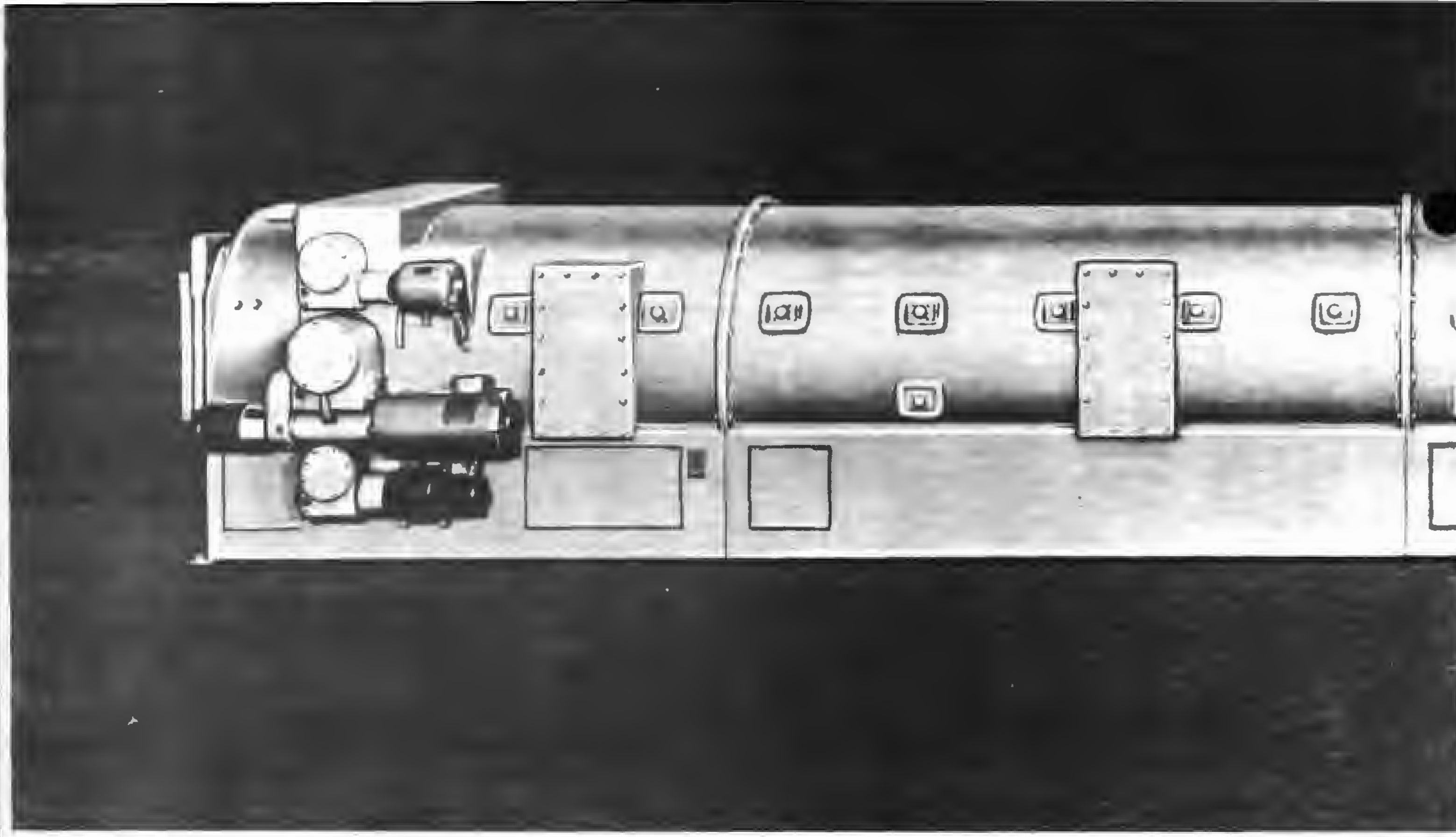
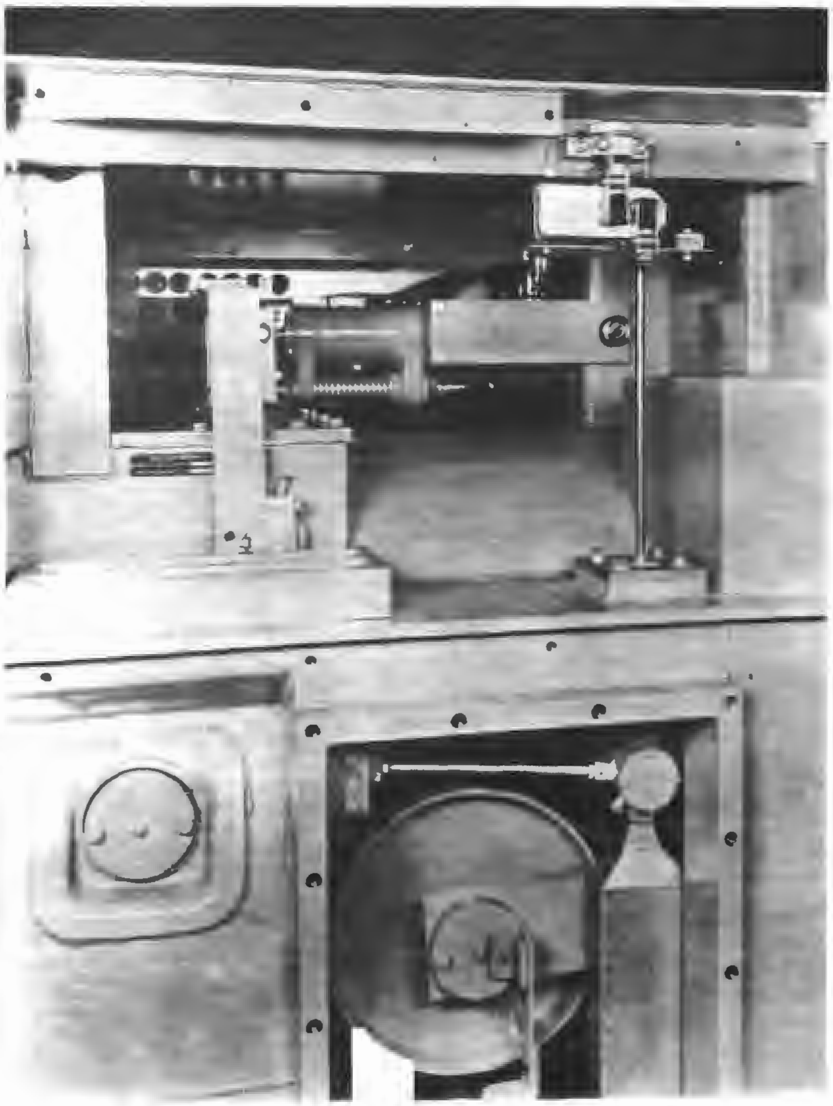


Fig. 8





S-E-Co. LONG CENTER GRAVIMETRIC FEEDER

Shown at the left in the above picture are three drives. The upper drive is for the distributor, which is standard for cyclone burner applications. The distributor assures uniform discharge of coal off the feed belt when the coal is fine and wet.

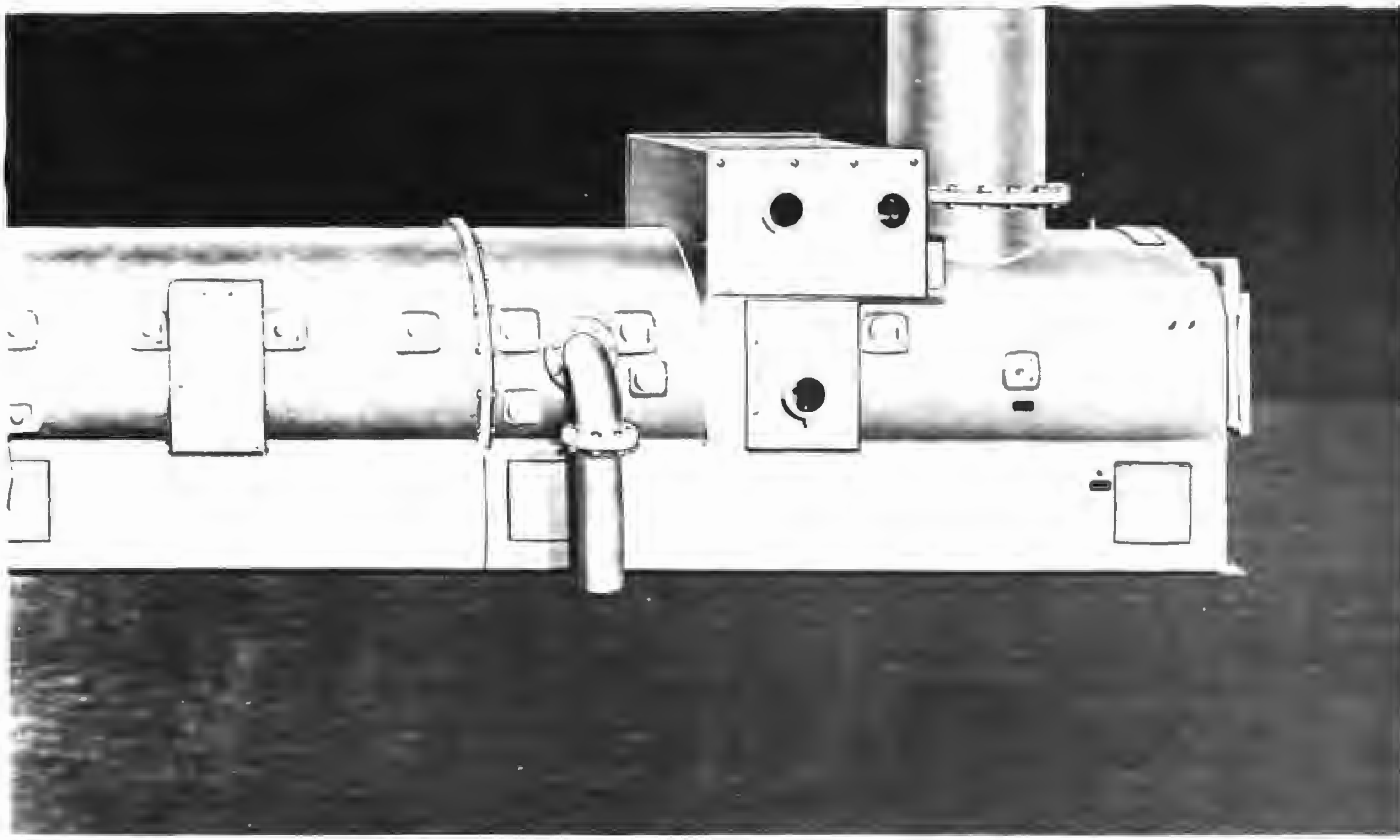
In the middle is the variable speed feeder drive, with an auxiliary tachometer generator installed at the extreme left. A centrifugal switch to signal machine operation can also be installed here. The auxiliary tachometer generator and centrifugal switch are extra price items.

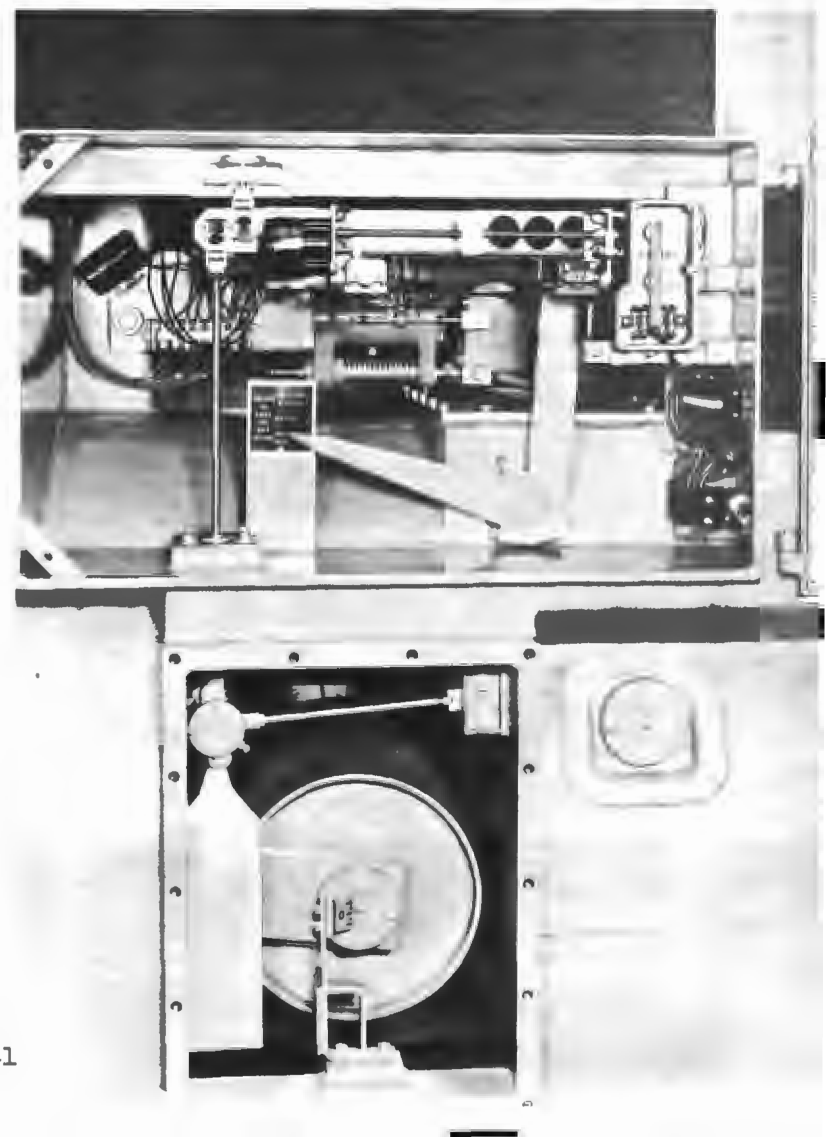
The lower drive is for the cleanout conveyor.

At the right is located the coal inlet, weighing compartment with bull's-eyes, and purge air connection.

S-E-Co. GRAVIMETRIC FEEDER

At the left is the weigh compartment with right side doors open. The lower door shows tension roll complete with height gauge and weighing roller with test weight attached thereto. The upper door shows weigh lever, leveling bar actuator and bail. Box for storing test weights is also shown.





TYPES—SIZES—CAPACITIES

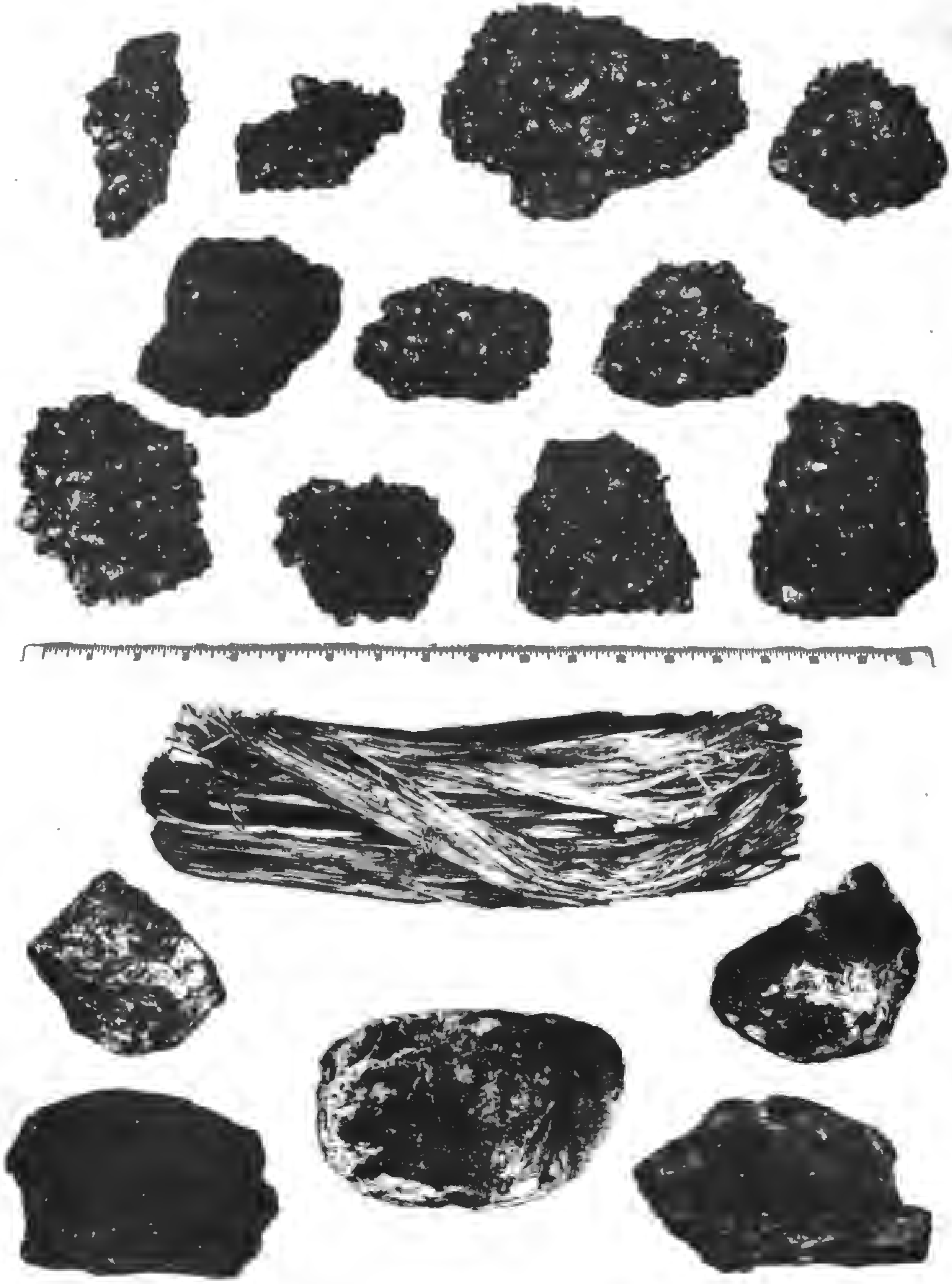
Stock Equipment Company produces a complete line of gravimetric feeders. The standard S-E-Co. Gravimetric Feeder has a length of 6′-8″ between center line of inlet and center line of discharge. The standard feeder is illustrated on the cover of this bulletin. Longer center line lengths can be furnished as required by specific applications.

S-E-Co. Gravimetric Feeders are built for any desired capacity up to 100 tons per hour. The capacity, which is determined by the maximum head shaft rpm, is matched to purchaser's requirements.

The above feeders are built for a 24″ width of coal on the belt.

S-E-Co. GRAVIMETRIC FEEDER

At the right is the weigh compartment with left side doors open. The lower door shows left end of tension roll. The upper door shows leveling bar actuator, leveling bar height gauge, weigh lever, weight calibration motor, poise, and magnetic "overweight" or "underweight" mercury switch assemblies.



Frozen Coal—Pyrites—Wood

S-E-Co. Gravimetric Feeders have an extraordinary ability to pass occasional pieces of frozen coal or foreign objects without difficulty. The full 24" width of coal on the belt is just one of the design features contributing to this ability. The endless belt is another. At the same time these feeders are also able to handle fine, wet, sticky coal with no trouble. The photographs above show lumps of frozen coal, balls of pyrites, and a piece of wood that have successfully passed through S-E-Co. Feeders.

CHANGING BELT ON 6'-8" GRAVIMETRIC FEEDER

10:00 a.m. Ready to start belt change with new belt and tools available to three men.

10:15 a.m. Tension pulley being removed. Belt scraper removed.

10:29 a.m. Weigh roller removed. Head pulley being removed.

10:36 a.m. Weigh span rollers being removed.

10:53 a.m. Self cleaning tail pulley removed.

11:00 a.m. Slide pan and belt being removed.

11:07 a.m. New belt ready to install.

11:21 a.m. New belt and slide pan in place. Tail pulley being installed.

11:40 a.m. Weigh span rollers being installed.

11:53 a.m. Head pulley installed, tension pulley being installed.

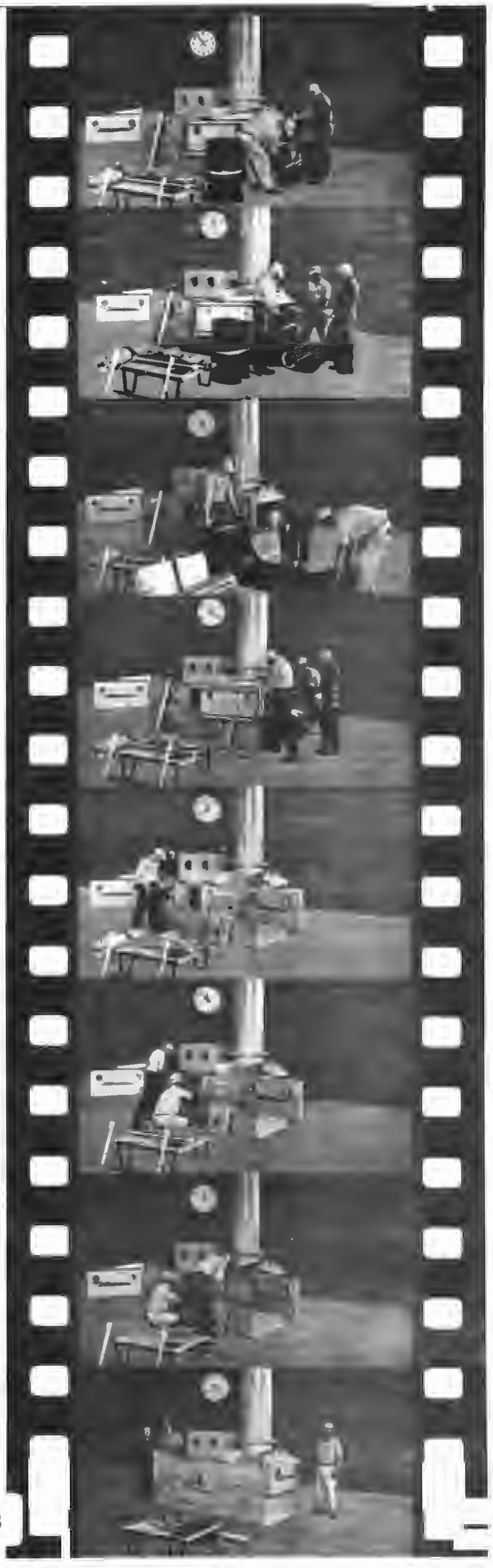
12:08 p.m. Weigh roller being installed. Tail pulley being adjusted.

12:21 p.m. Belt installation complete.

For an experienced crew with a new feeder

TOTAL TIME 2 HOURS, 21 MINUTES

An inexperienced crew will require somewhat greater time than that indicated, especially if the feeder has been in service.



FEATURES

BELT

The S-E-Co. endless rubber belt has an integrally molded high curb on each side to retain coal on the belt and to eliminate the need of metal side skirts or troughing of the belt over the weighing section. The belt also has an integrally molded V section which engages a V groove in the pulleys. Three plies of duck with a 1/8" top cover and 1/16" bottom cover are used in the belt.

SKIRT BARS AND INLET

At the stainless steel inlet to the belt, vertical stainless steel skirts are employed for coal inlet control. These skirts have a clear horizontal spacing of 24" and can be adjusted vertically for proper belt clearance.

PULLEYS AND SUPPORT ROLLERS

Head and tail pulleys are heavy castings, turned flat and having a V groove for the belt to assure positive tracking. The tail pulley is of self-cleaning construction, eliminating the need of a belt scraper on the inside of the belt. Supporting rollers are designed to allow a small amount of adjustment in the horizontal plane to assist with tracking. All bearings are protected against the high humidity and dusty atmosphere existing inside the feeders by suitable grease seals and also have a grease flushing feature for increased protection.

BELT SCRAPERS

A scraper is provided for the outside surface of the belt. It is located above the feeder discharge so that any coal removed falls out the discharge opening and does not accumulate.

CLEANOUT CONVEYOR

S-E-Co. Gravimetric Feeders are furnished with cleanout conveyors to remove accumulated dust from the bottom of the housing. This conveyor employs special malleable iron flights with stainless steel pins. The bottom strand operates in a stainless steel trough which prevents corrosion damage to the bottom of the feeder housing. The conveyor needs to be operated only a few minutes each hour.

DRIVE

S-E-Co. Gravimetric Feeders are driven by an AC induction motor through a magnetic slip clutch to a specially designed helical and worm gear reducer. The speed control unit receives a feed back signal from a tachometer generator on the clutch output shaft to provide accurate feed rate regulation. Since the S-E-Co. Gravimetric Feeder feeds the coal by carrying it, the motor power required is unusually low, typically 1 to 1½ H.P. In addition, all bearings are anti-friction. Because no excess horsepower is required to overcome friction, the wear of feeder parts is at a minimum.

BODY

The cylindrical feeder housing is constructed of 1/4" thick mild steel plate. Steel plate 3/4" thick is used where there are large openings in the housing and where machine work is required. The entire feeder or head end of long center designs is machined as a unit on a horizontal boring mill to assure accurate alignment of the head shaft and ease of realignment upon reassembly. Bull's-eyes and lights are provided at both inlet and discharge ends to enable visual inspection of the interior while in operation. The lights are wired internally, the leads being brought to junction box underneath the main drive motor.

ALARMS

Two paddle type coal alarm switches are provided. One, located over the feed belt, gives positive indication of loss of coal flow to the feeder. The other is located near the discharge to stop the feeder in the event of a coal pluggage at the mill or cyclone burner.

SERVICE

Because the S-E-Co. Gravimetric Feeder uses an endless rubber belt, very little operating attention is required. The belt take-ups and adjustments for tracking the belt are operated from outside the machine. All greasing is also accomplished from the outside.

OPTIONAL ACCESSORIES

- Pneumatic Speed Transmitter
- Electric Tachometer Generator
- Coal Feed Rate Indicators and Recorders
- Slow Speed Centrifugal Switch
- Pulser for Remote Indication of Weight of Coal Feed
- Temperature Safety Switch
- Coal Valves and Coal Scales
- Downspouts and Hoppers
- Plastic Metallic Gasket for Flange Connections

STOCK EQUIPMENT COMPANY
731 HANNA BLDG. • CLEVELAND, OHIO 44115

JEFFREY

Medium Capacity Feeders
NO. 3DHS

APPLICATION – This rugged, compact, medium capacity feeder is used primarily for feeding material to belts, screens, elevators, grinding mills, batching and continuous scales, and for proportioning various materials.

CAPACITY – Varies with the materials, and conditions under which they are handled. When feeding silica sand or similar materials weighing 100 cu. ft., this feeder will handle up to 175 TPH when operated with the pan on a 10° down-slope. Feeder capacities are proportional to the width of the deck.

CONTROL – When the feeder is operated from either 50 or 60 hertz alternating current, an SCRM rectifier set is furnished with the feeder. Capacity may be instantly varied from near zero to maximum by simply turning the calibrated control pot on the SCRM rectifier. Where desired, the feeder may be controlled, and automatically regulated, from a 10-50 ma DC signal.

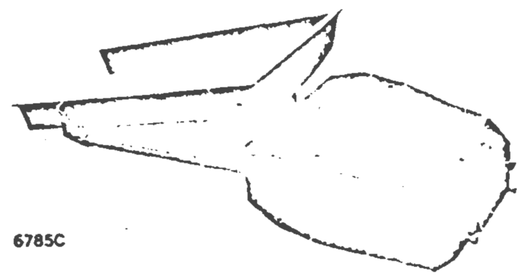
POWER UNIT MOUNTING – The feeder is normally furnished with the power unit mounted below deck, but can be arranged with the power unit above deck.

DECKS – This feeder can be furnished with open pan decks in sizes listed below. Tubular or spreader decks may also be furnished. Standard construction is of mild steel, but decks of other materials can be furnished.

CURRENT REQUIRED – 240, 480, or 580 volts; 25, 50, or 60 hertz.

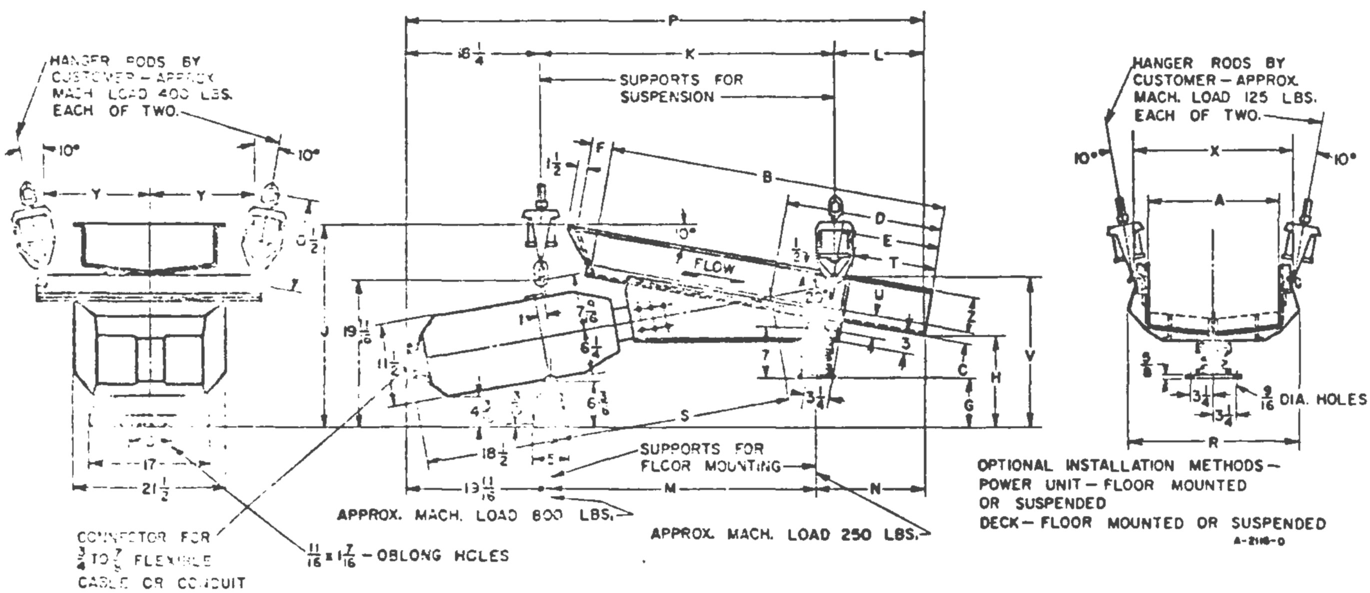
POWER CONSUMPTION – 998 watts.

WEIGHTS – Feeder only, uncrated – 1175 lbs.
Feeder with control, crated – 1500 lbs.



6785C

No. 3DTHS feeder with 24" wide x 48" long open pan deck.



OPTIONAL INSTALLATION METHODS –
POWER UNIT – FLOOR MOUNTED OR SUSPENDED
DECK – FLOOR MOUNTED OR SUSPENDED
A-2116-0

Dimensions in Inches																					
Width A	Length B	C	D	E	F	G	H	J	K	L	M	N	P	R	S	T	U	V	X	Y	Z
18	60	0	30	15	2¾	8 1/16	7¼	29 11/16	45	14⅞	42¼	16¼	77 1/16	24½	28⅜	11 11/16	1½	20	22	15	6
24	**48**	0	23½	18¼	3½	4 15/16	13⅛	28⅝	36 5/16	18	37½	14⅜	71 9/16	31	30⅛	11	1¼	22 5/16	30¼	15	6
30	36	2¼	15½	11	2⅞	6¼	13¼	27½	38	9½	33¼	12½	66	35½	32⅞	9	4	21¼	34	8	5
30	42	0	22¼	14	3	9 3/16	13½	27⅞	33⅞	13 11/16	34¼	15⅜	69 15/16	34⅜	29¼	12	2¼	22¼	35⅝	15	7

Dimensions shown are approximate and should not be used for construction purposes. **Ask for certified prints.**

Norblo

Dust Control Systems

BUELL ENGINEERING COMPANY, INC.

253 NORTH FOURTH STREET · LEBANON, PENNSYLVANIA 17042
AREA CODE 717 272-2001 CABLE BUELENC. LEBANON, PA.
TELEX 842-332

April 21, 1971

Erie City Energy Division
Zurn Industries
Erie, Pennsylvania 16500

Attention: Mr. Bob Seibel
Project Engineer

Subject: BUELL Proposal No: 71L-19073
Electrostatic Precipitator
Fluid Bed Boiler

Gentlemen:

Attached is our proposal for an electrostatic precipitator in accordance with our telephone conversation. Due to the fact that this project is a proposal for a new steam generating process for the NAPCA of the United States Federal Government, our price as quoted will apply on a one-time only basis. It is our desire to participate in this project and our price reflects this desire. We trust that you will treat this matter in this light.

The drawing that is included with this proposal is of a precipitator that has one more gas passage than that quoted. The dimensions shown for the width of the precipitator should be reduced by 9".

Should you have any questions or comments concerning our offering, do not hesitate to contact us.

Very truly yours,

BUELL ENGINEERING COMPANY, INC.

L. F. Rettenmaier

L. F. Rettenmaier
Sales Engineer
Lebanon District

LFR:sw (2)

Encl:

Suddenly Our Business is Everybody's Business

Buell Norblo

Dust Control Systems

BUELL ENGINEERING COMPANY, INC.

253 NORTH FOURTH STREET · LEBANON, PENNSYLVANIA 17042
AREA CODE 717 272-2001 CABLE BUELENC, LEBANON, PA.
TELEX 842-332

April 21, 1971
BUELL Proposal No: 71L-19073
Subject: Electrostatic Precipitator
Fluid Bed Boiler

Erie City Energy Division
Zurn Industries
Erie, Pennsylvania 16500

Attention: Mr. Bob Seibel
Project Engineer

Gentlemen:

BUELL ENGINEERING COMPANY, INC. proposes to furnish, in accordance with the conditions outlined under NORMAL OPERATING CONDITIONS, dust collection equipment and certain specific auxiliaries as hereinafter itemized and described. The items of equipment and services are listed. Details will be in accordance with referenced drawings, specifications, data sheets, interpretations and conditions. Numbered items constitute the limits of this proposal.

NORMAL OPERATING CONDITIONS

Application	Fluid Bed Boiler
Gas	Flue
Boiler Capacity - #Gas/Hr.	298,000
Boiler Capacity - #Steam/Hr.	250,000
Quantity - ACFM	97,500
Temperature - °F.	350
Moisture - % by Volume	5.3
Coal Analysis	Proximate
Ash	8.5
Sulphur	4.3

NORMAL OPERATING CONDITIONS (Cont'd)

Dust Grains/CF - Inlet at Conditions	Assume 2 to 4
Desired Efficiency	99.5
Mechanical Design Conditions	
Pressure - Inches W.G.	$\pm$ 15
Temperature - °F.	400

	Volts	Phase	Cycle
Required Power Supply	440	3	60

EQUIPMENT AND SERVICES OFFERED

Item 1 - One (1) BUELL Model BA1.1x23K333-2P Modular Electrostatic Precipitator, arranged as a one (1) chamber, one (1) cell per chamber wide, nine (9) bus section long unit. Electrically, the precipitator is three (3) fields. The precipitator, therefore, contains nine (9) independent, isolatable bus sections. The precipitator is also equipped with Safety Key Interlocks to prevent access during energization. For energization three (3) High Voltage, Silicon Rectifier, Electrical Sets are furnished. The model number of the sets is SIPP-SCR-35-11-45, each rated at 35 KVA, 550 MA, 45 KV. The High Voltage is distributed as full wave. Each set is complete with ANACOMP controls and high voltage bus-ducts to the extent as shown on referenced drawings.

Item 2 - Four (4) Sets of Contract and Customer Data.

Item 3 - Services of an Operating and/or Erection Supervisor.

DELIVERIES

Drawings for approval will be submitted four (4) to six (6) weeks after order. Equipment deliveries will initiate six (6) to seven (7) months after approval of drawings and settlement of details.

REFERENCED DRAWINGS

Precipitator NL-874

REFERENCED SPECIFICATIONS

General Conditions	GC-1
Warranties and Guarantees	SC-1

MODULAR PRECIPITATOR	SPECIFICATION BOOK
Modular Electric Precipitator	Section I
Safety Key Interlock System	Section I
Field Engineering & Contractual Data	Section I
High Voltage Power Supply	Section II
ANACOMP - Automatic Voltage Control	Section II
Vibrator Intensity Control	Section II

PERFORMANCE

Based upon the conditions outlined under NORMAL OPERATING CONDITIONS and with those instructions as issued by our operating supervisor, the draft loss through the Electrostatic Precipitator, when measured as a difference of static pressure between the inlet and outlet flanges of the collector, is guaranteed not to exceed 0.3"W.G. Based on a maximum of 15% of unburned combustibles in the fly ash entering the precipitator. The collector is guaranteed to collect 99.5% by weight of the suspended solids contained in the gases entering the collector. An outlet loading of .02 grains per cubic foot or less shall constitute fulfillment of our guarantee.

EQUIPMENT AND SERVICES NOT FURNISHED

The following equipment and services are specifically not furnished under the terms of this proposal:

Connecting Flues
Fans, Fan Motors and Drives
Dampers
Supporting Steel, Groutings and Foundations
Railings, Ladders and Walkways
Dust Disposal System
Hopper Valves and Vibrators
Thermal Insulation and Clips
Flashings
Power Supply to Control Cabinets
Low Voltage Wiring
Permanent Lighting
Safety Key Interlocks on Precipitator Bolted Access Openings
Safety Key Interlocks on Precipitator Bolted Hopper Doors
Grounding Leads and Subterranean Plant Grounds
Erection, Welding Rod and Field Painting

PRICES

Item 1 & 2...$ 76,000.00

(SEVENTY SIX THOUSAND DOLLARS)

Item 3...$ 100.00

(ONE HUNDRED DOLLARS) Per Day Per Man for every day away from Lebanon, Pennsylvania plus living and traveling expenses.

TERMS

The equipment prices are F.O.B. place of manufacture, suitably prepared for domestic shipment.

The terms of payment on the equipment are as follows: 10% upon acceptance of Approval Drawings; 85% upon shipment; and 5% on acceptance of equipment, but in no case later than 120 days from date of final shipment.

The above equipment prices are firm provided order is placed within sixty (60) days and shipment of equipment is within one (1) year of date of proposal.

This offer is subject to acceptance within sixty (60) days and the attached GENERAL CONDITIONS, GC-1 and WARRANTIES AND GUARANTEES, SC-1.

Very truly yours,

BUELL ENGINEERING COMPANY, INC.

L. F. Rettenmaier

L. F. Rettenmaier
Sales Engineer
Lebanon District

LFR:sw (2)

BUELL ENGINEERING COMPANY, INC.

BID EVALUATION FORM

for

ELECTROSTATIC PRECIPITATORS

FLUID BED BOILER

PROPOSAL NO. 71L-19073 **DATE:** April 21, 1971

OPERATING AND PERFORMANCE DATA: (See Proposal for Complete Qualifications)	
Volume CFM @ operating Conditions	97,500
Temperature — °F. @ operating conditions	300
Inlet loading - grains/cf @ operating conditions	2-4
Dust Bulk Density	---
Guaranteed Efficiency — Percent	99.5
Guaranteed Outlet Loading — gr./act. cu. ft.	.02
Pressure Drop Across Precipitator including gas distribution devices	0.3
Gas Velocity — ft./sec.	3.14
Treatment Time-Seconds	8.6
PRECIPITATOR ARRANGEMENT , MODEL NO: BA1.1x23K333-2P	
Number of Precipitators	1
Chambers (number) /Precipitator	1
Fields (number and length) /Precipitator	3 @ 9'
Cells (number) /Precipitator	1
Bus sections (number) /Precipitator	9
Casing Material and Thickness (inches)	3/16" Mild Steel
Casing Design Pressure (" W.C.) (Check one: Positive ☒ or Negative ☒)	15
Number of Hoppers/Precipitator	2
Hopper Material and Thickness (inches)	3/16" Mild Steel
Minimum Hopper Valley Angle	55
Total Hopper Capacity (cubic feet)/Precipitator	3,547
Hopper Accessories (list each separately)	---
Insulator Compartment Material and Thickness (inches)	3/16" Mild Steel
Penthouse Material and Thickness (inches)	---
Number of Insulator Compartments/Precipitator	12
Surface Area (sq. ft.)/Precipitator	
Roof	510
Shell	3,050
Hoppers	1,390
Others	---

PROPOSAL NO. 71L-19073 **DATE:** April 21, 1971

Precipitator Internal Gas Distribution Devices	BUELL Vertical Channels
Types	Adjustable
Quantity and Location/Precipitator	25 @ Inlet
Material and Thickness	12 Gauge Mild Steel
Number and Type Rappers	---
Number, Type and Size of Access Doors/Precipitator	
Roof	---
Shell	4 - 1'-8" x 3'-0"
Hopper	2 - 1'-8" Dia.
Insulator Compartments	3 - 1'-8" x 3'-0"
Other	Ducts - 2 - 1'-8" x 3'-0"

COLLECTING SYSTEM — PER PRECIPITATOR

Number of Gas Passages	23
Spacing of Gas Passages (inches)	9
Collecting Surface Material and Thickness	18 Gauge Mild Steel
Collecting Surface Effective Length (feet)	27
Collecting Surface Effective Height (feet)	30
Total Collecting Surface Area	37,260
Maximum Collecting Surface Area Rapped at any Instant (square feet)	2,050
Type Collecting Surface Rappers	BUELL Electro-Magnetic
Number of Collecting Surface Rappers per Precipitator	18

HIGH VOLTAGE SYSTEM

Discharge Electrode — Type, Material and Thickness (inches)	Hi Carbon - Straight Wire .105 Dia.
Total Lineal Feet of Effective Discharge Electrode per Precipitator	24,840
Type Discharge Electrode Rappers	BUELL Electro-Magnetic
Total Number of Discharge Electrode Rappers per Precipitator	9

HIGH VOLTAGE ELECTRICAL SET

Type Transformer Rectifier	Silicon Diode
Number Transformer Rectifiers	3
Size Transformer Rectifier	35 KVA
Voltage Rating KV (DC) Avg. (For pure resistive loads)	45
Current Rating Milliamps (DC) Avg. (For pure resistive loads)	550
Number of Transformer Rectifier Control Cabinets	3
Construction of Transformer Rectifier Control Cabinets NEMA	Nema III & V V & XII

PROPOSAL NO. 71L-19073 **DATE:** April 21, 1971

Transformer Rectifier Insulation Fluid	Oil
Wave Form of High Voltage	Full
Number and Type High Voltage Switches	3 - Air Switch
Key Interlocks (Yes —No)	Yes
Control Cabinets	Yes
Transformer Rectifiers	Yes
Access Doors	Yes
Type Transformer Rectifier Controls	Solid State Anacomp
Maximum Ambient Temperature for Transformer Rectifier — °C	40
Maximum Ambient Temperature for Transformer Rectifier Control Cabinets — °C	40
Power Consumption KVA/Precipitator	
1. Transformer-rectifier	105
2. Rappers	2
3. Insulator heaters and blowers	
4. Hopper heaters	
5. Lights	
6. Other	
Total	107
Total connected load KVA/Precipitator	122
Power Distribution	
Individual breakers each control cabinet	Yes
Central distribution panel	No

OTHER AUXILIARY EQUIPMENT OR SERVICES

Heat Insulation — Type & Thickness	---
Weather Enclosure — (Roof and/or Hopper, type)	---
Inlet and outlet Transitions — Material and Thickness	---
Access Facilities — Type & Location	---
Insulator Compartment Blower System (number)	---
Model Study	---
Startup—Training—Testing Supervision	---
Erection Supervision	---

WEIGHTS

Total Precipitator Weight Including Electrical Equipment but Excluding Dust Load	199,000

DRAWINGS:

Layout	---
Precipitator	---
Other	---

Dust Control Systems

BUELL ENGINEERING COMPANY, INC.

Buell Norblo

253 NORTH FOURTH STREET · LEBANON, PENNSYLVANIA 17042
AREA CODE 717 272-2001 CABLE BUELENC, LEBANON, PA.
TELEX 842-332

May 13, 1971

Erie City Energy Division
Zurn Industries
Erie, Pennsylvania 16500

Attention: Mr. Bob Seibel
Project Engineer

Subject: BUELL Proposal No: 71L-19073
Electrostatic Precipitator
Fluid Bed Boiler

Gentlemen:

We advise you as follows concerning the several questions raised during our telephone conversation.

The effect on the performance of the precipitator in regard to the sulphur carried over in the limestone and fly ash will, in our opinion, have little or no influence on performance.

Precipitator efficiency versus particle size is a consideration in precipitator design, but is not as important when the particle size range is evenly distributed. When there is a high concentration on low and sub-micron particles this must be allowed for in the design.

When precipitators are designed for a fly ash application, without any other introduced dust, i.e. limestone, the norm is to guarantee performance on the basis of a maximum of 15% unburned combustibles in the ash. As the amount of unburned combustibles increases over the 15%, it is our practice to also change our sizing criteria to allow for the higher carbon content of the ash. On this application, it is difficult to predict this effect because of the mixture of dust presented to the precipitator. Frankly, we do not know this effect as we have no experience in this application.

Suddenly Our Business is Everybody's Business

We feel we have conservatively sized the precipitator that we have quoted and have considered it more on the basis of the limestone content than the fly ash which we feel, as stated before, is the more difficult dust to separate. The real question is whether the combination of limestone and fly ash presents conditions which may be better or worse. We have always felt that the more conservative approach was the preferred one.

Very truly yours,

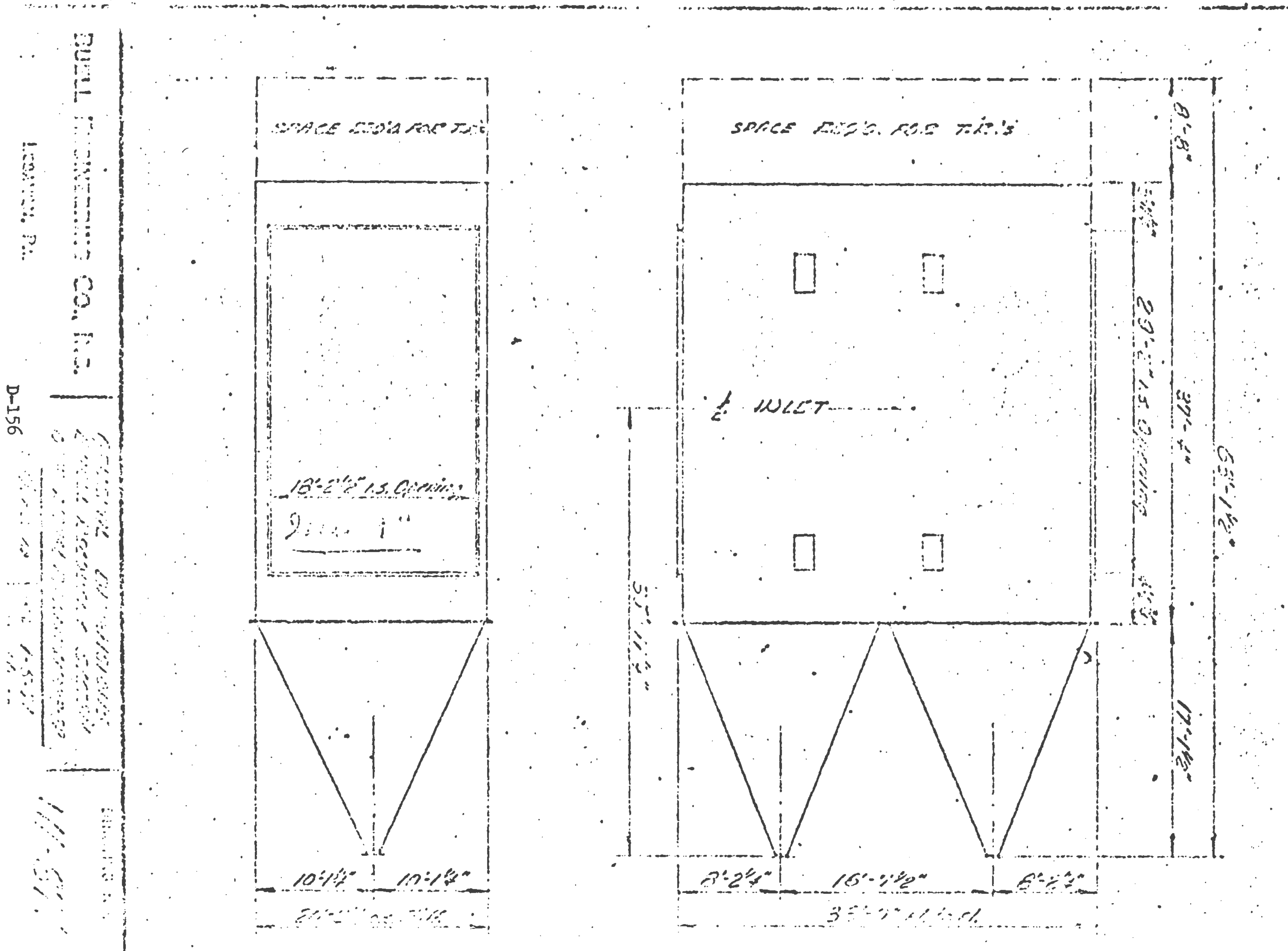
BUELL ENGINEERING COMPANY, INC.

L. F. Rettenmaier

L. F. Rettenmaier
Sales Engineer
Lebanon District

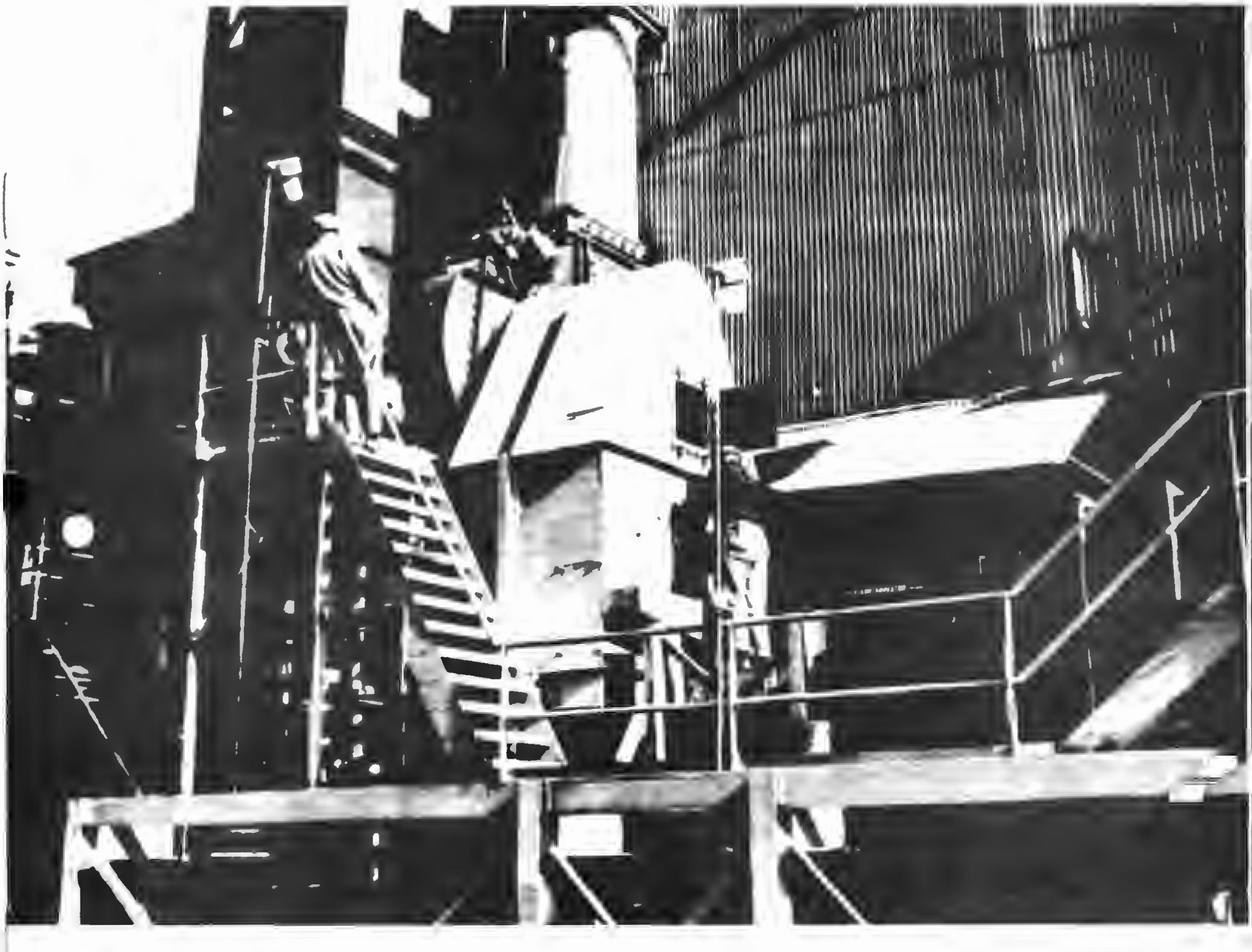
LFR:sw

cc: New York Office

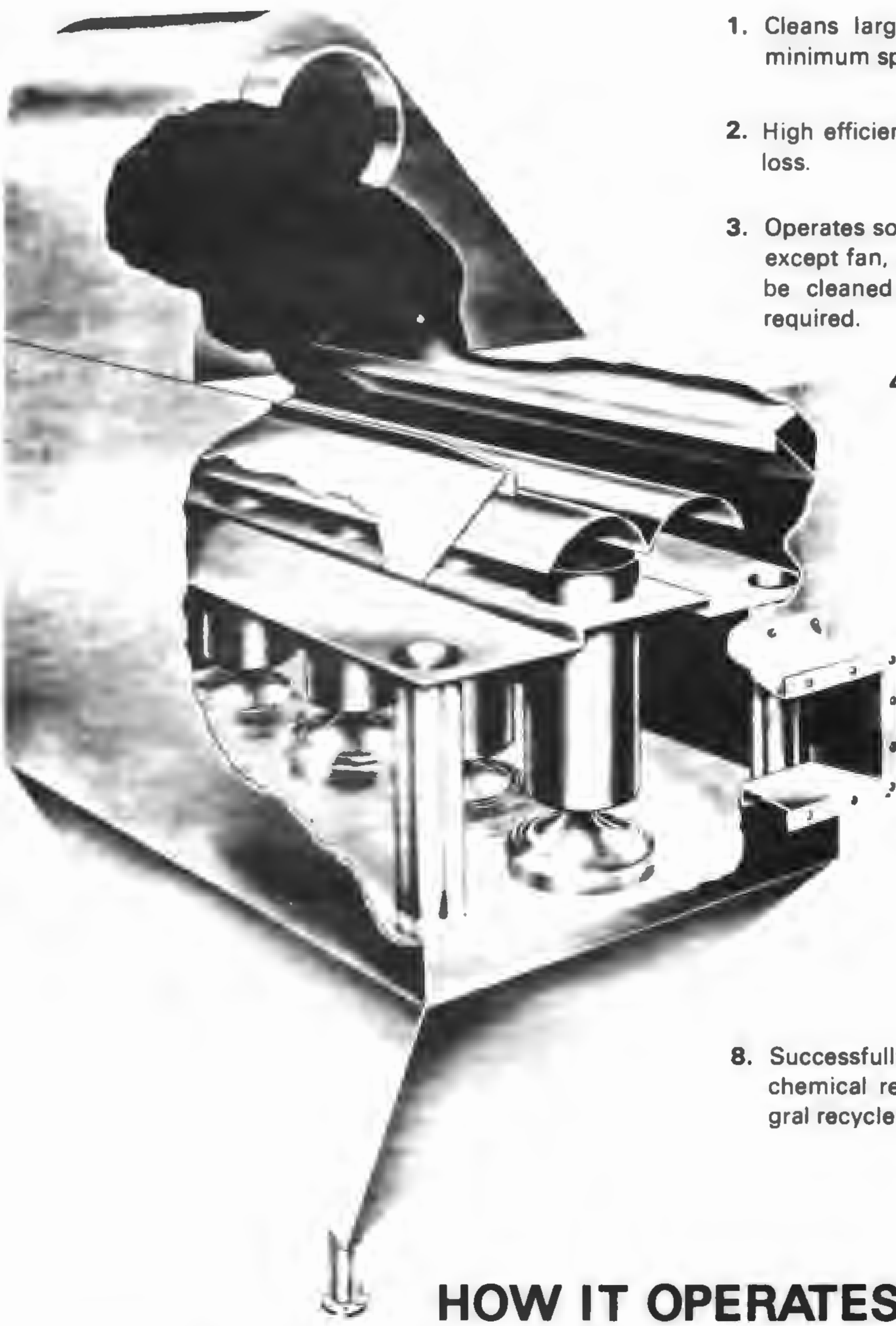

D-156
INLET
18'-2½" I.S. Opening
10'-1¼"
10'-1¼"
8'-2¼"
16'-4½"
8'-8"
17'-1½"

A NEW CONCEPT IN AIR POLLUTION CONTROL . . .

DUSTRAXTORS®



DUSTRAXTOR . . . . UNIQUE



1. Cleans large volumes of gas . . . yet requires minimum space.

2. High efficiency dust removal with moderate draft loss.

3. Operates solely on air movement. No moving parts except fan, no spray nozzles to plug, no packing to be cleaned or replaced, and no recycle pumps required.

4. Recirculating water system with very little water discharge required.

5. Available with stainless steel, fiberglass and rubber-lined collector elements for high temperature and corrosive applications.

6. Custom design. Your exact requirements in a variety of sizes and arrangements.

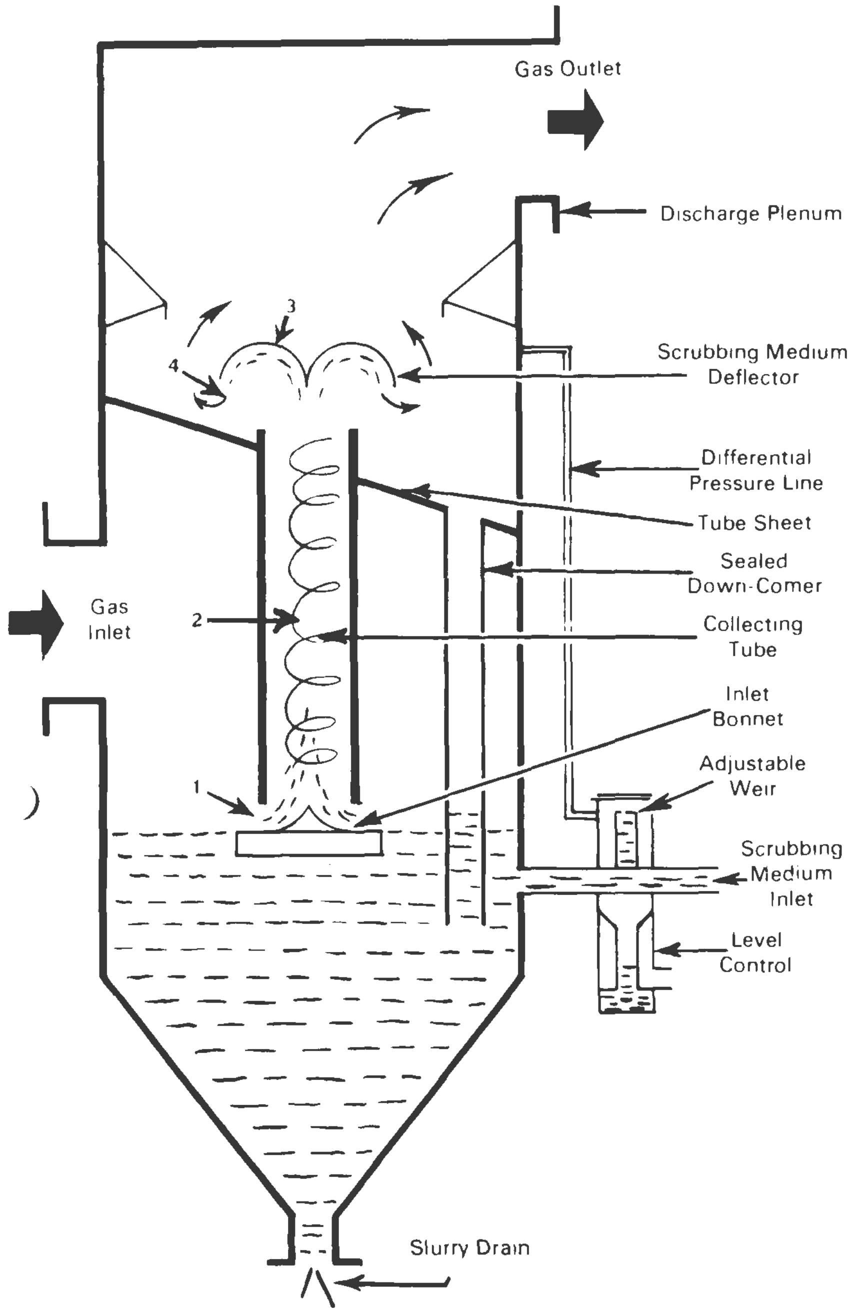
7. Fully shop-assembled for easy installation.

8. Successfully removes gaseous contaminants . . . chemical reactants can be added directly to integral recycle hopper.

HOW IT OPERATES

Dust-laden gases enter the unit and sweep over the surface of the scrubbing medium and underneath the base of collecting tubes which are suspended from an overhead tube sheet. The velocity of the gas passing beneath the tubes and across the inlet bonnets inspirates the scrubbing medium from the surface of the bonnets upward into the collecting tubes. Each inlet bonnet imparts a cyclonic, spinning action to the gas mixture. The shearing action of the gas atomizes the water into a dense spray as the gases continue up the tubes. As the gases are discharged from the tubes, they are directed against a curved deflector which acts as an entrainment separator by forcing the liquid downward onto the tube

OPERATING PRINCIPLE



1 Impaction of the dust particles into the scrubbing medium as the high velocity gases pass through the slot between the inlet bonnets and the collecting tubes.

2 Cyclonic spinning and mixing of gas and liquid droplets within the collecting tube.

3 Impingement of gases and water upon the flooded surface of the scrubbing medium deflector.

4 Scrubbing of the gas as it passes through a curtain of liquid being discharged from the scrubbing medium deflector.

sheet. The cleaned gases flow around the deflector and continue upward through the collector discharge plenum and the clean gas outlet. The scrubbing medium is returned to the hopper through a sealed downcomer. The Zurn "Dustraxtor" operates equally well with induced draft and forced draft systems. The fan may be placed on the top of the unit or remotely located. Capacities range from 1,200 to 72,000 cubic feet of saturated gas per minute in single unit construction. Multiple units are used for larger gas volume requirements.

DUSTRAXTOR . . . A NEW TOOL

TYPICAL DUSTRAXTOR in a coal mining operation used for removing coal dust from coal crushers, screens, and conveyor system transfer points. This factory-assembled unit has a solids ejector type hopper. The induced draft fan and the stack are mounted on top of the Dustraxtor.

Zurn Industries, Inc. has emerged in recent years as a "Total Environmental Control" company dedicated to solving the complex problems of air, land, water, and noise pollution, and in the process, preserving nature's ecological balance. By continually expanding its scientific — engineering — technological base, Zurn has created a team of technical experts capable of analyzing and researching your specific problems . . . then designing and installing the proper control system to assure proper emission quality. Whether your control program is related to industrial, municipal, or urban complexes, Zurn total environmental control capabilities, utilizing our proven systems concept, are your best assurance of cleaner, clearer, more healthful environment for years to come.

The combination of the facilities, products and manpower of Clarage, Swartwout and Fly Ash Arrestor has brought together in the Zurn Air Systems Division a broad range of air pollution control capabilities. The Zurn "Total System" concept implies a thorough analysis and understanding of each air pollution problem and a complete solution based on custom-fabricated components properly engineered and sized to do the job efficiently. Zurn Air Systems manufactures all the elements of the Dustraxtor wet scrubber system — fan, damper, ducts, and structural supports, stack and complete shop-assembled scrubber units.

The efficiency of a Dustraxtor scrubbing system is dependent on actual inlet conditions and on the pressure drop across the scrubbing unit. The main influencing factors are: dust loading in the gas, dust particle size, and the wetting characteristics of the dust. In general, based on a variety of existing installations, a Dustraxtor, when correctly applied to an air pollution control problem will provide efficiencies in excess of 95 percent. Dustraxtor systems have also been highly successful in the collection of fumes, atomized mists, and the absorption of soluble gases. Exceptional results have been obtained in the absorption of sulphur dioxide (SO_2) by adding chemical reactants such as sodium and calcium to the scrubbing medium. The Dustraxtor's integral hopper and automatic recycling action provide the necessary time and contact for the absorption process to take place.

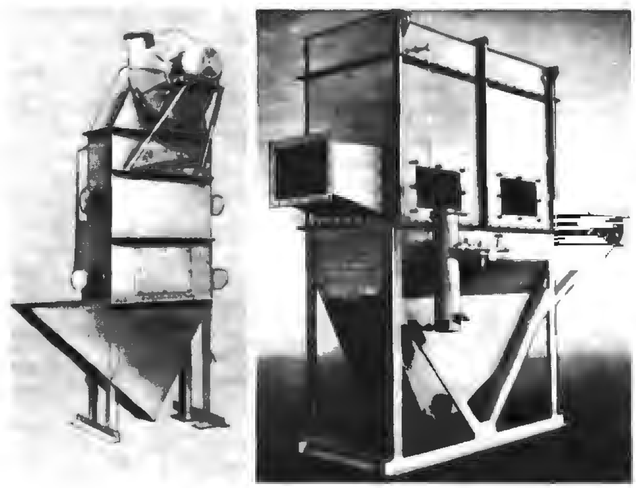
Zurn Air Systems engineers are continually developing new applications for this unique air pollution control system. Dustraxtors in series are economically utilized to achieve very high collection efficiencies, and Dustraxtors in conjunction with other types of systems are solving many serious air cleaning problems.

FOR AIR POLLUTION CONTROL

Zurn Dustraxtor medium-energy wet scrubber systems are essentially self-contained; and, to the extent practical for shipment, fully factory-assembled. The only moving part required for effective operation is a fan. This fan can be positioned in the system either before the scrubber or after with equal efficiency. Provision is conveniently made on top of the Dustraxtor for placement of an induced draft fan as shown in the photo on page 4. However, the fan may also be placed in a remote location without affecting operating efficiency.

In keeping with the "Total System" concept Zurn Air Systems has developed and manufactured an induced draft fan that is ideally suited for Dustraxtor applications. On smaller units that are suitably sized for shipment the inclusion of this fan makes available completely "packaged" systems. The special fan as illustrated above is the Zurn Air Systems Model 3400RB which is a radial-blade design for large volumes and medium pressures. It is available in all sizes and widths necessary to complement the Dustraxtor system. For higher temperature and corrosive applications special materials and coatings are available.

Where circumstances warrant in unusual air pollution control situations, Zurn Air Systems has available test models (see below) which can be temporarily installed to determine actual Dustraxtor efficiency. Zurn Air Systems as part of their "Total System" concept will pretest inlet and exit conditions for proper application and sizing of a Dustraxtor system.



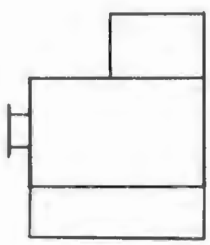
COKE AND COAL: Handling, processing, coal pulverizers, discharge following dry collectors, dryers, belt transfers.
ORE PROCESSING: Crushers, screens, transfer points and dryers.
KILNS: Lime kilns, rotary dryers.
FERTILIZER: Dryers, mixers, baggers, transfers, conveyors, coolers, screens and pulverizers.
PAINT: Pigment mixers, spray chambers.
OIL MIST: Machining, sprays, asphalt felt coating.
ACID MIST: Pickling and plating.
CLAY: Rotary Dryers, Baggers.
LINT: Buffing, paper slitters, doctor blades.
FOUNDRY: Sandblast rooms, shotblast, electric melting furnaces, shakeouts, core ovens, sand handling.
RUBBER: Roll grinding, mixers, tire grinding and buffing talc dust.
ROCK PRODUCTS AND MINING: Asphalt plants, aggregate dryers.
INCINERATORS: Spray cooling chamber must precede Dustraxtor.
STEAM GENERATION SYSTEMS: Power boilers, recovery boilers or furnaces (Salt Cake).
GASEOUS CONTAMINANTS: SO_2, NO_2, etc. . . . chemical reactants are added to scrubbing medium.

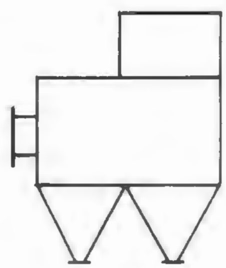
D-161

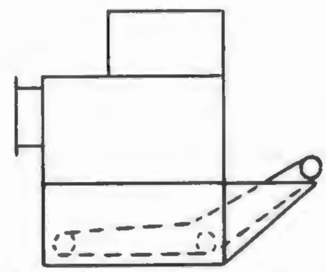
DUSTRAXTOR SPECIAL DESIGN

Dustraxtors are of heavy-duty construction in shop-assembled units. Standard materials of construction include mild steel, stainless steel, and a variety of linings. Where corrosive conditions or high temperatures exist, recommendations will be made by Zurn Air Systems. Fans, dampers, stacks, support structures or other special accessories necessary for the "Total System" will be included.

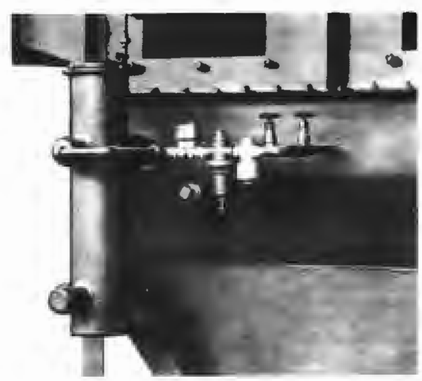
Three basic arrangements are illustrated below. The selection depends on the availability of water, method of liquid disposal and the characteristics of the dust when wet. In cases where soluble dust or gases are collected, concentrations may be raised to a level sufficient to return to process. Flat-bottom arrangements are for periodic cleanout. The hopper provides for a continuous drain out of the system. Solids ejectors are discussed on page 7.


FLAT

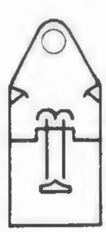

HOPPER

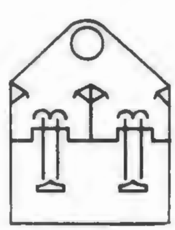

FLAT WITH SOLIDS EJECTOR

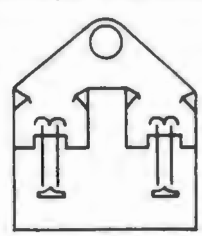
The liquid level within the Dustraxtor is controlled externally by a sealed adjustable weir as shown at the right (For a detailed view see page 3). The weir is connected to both the inlet and discharge plenum chambers and thus senses the pressure drop across the unit. The level of the scrubbing medium determines the pressure drop across the unit and the adjustable weir determines the level of the scrubbing medium. Clean liquid entering the unit passes beneath the overflow weir, assuring that only clean water will be discharged from the weir.



Dustraxtor capacities range from 1,200 to 72,000 saturated cubic feet of gas per minute in single unit construction. The capacity is determined by the number of collecting tubes within the unit. A fundamental design parameter is a volumetric capacity of 1,200 CFM per tube, saturated.

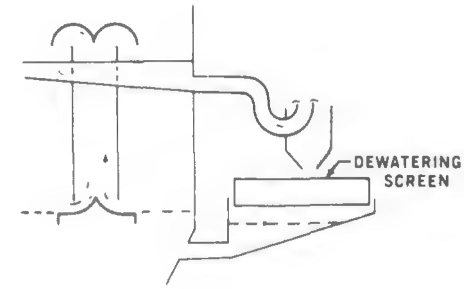

SINGLE WIDTH

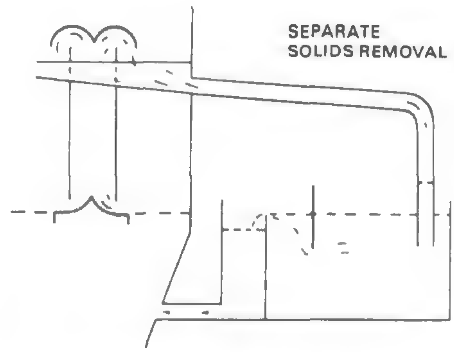

DOUBLE WIDTH (Short Length)


DOUBLE WIDTH (Long Length)

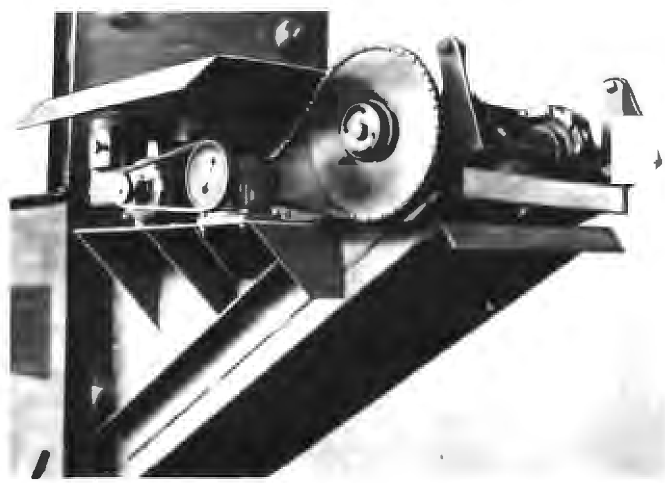
FEATURES AND APPLICATIONS

Special construction features may be provided where unusual conditions exist such as floating dusts, lint, or heavy, fast-settling dusts. By recirculating the liquid externally by gravity through weirs or dewatering screens the collected material may be removed, examined, or reprocessed without disturbing the operation of the collector. See illustration to the right.





The solids ejector as shown to the right is a mechanical drag assembly for the removal of collected dust. It is used in connection with the flat-bottom arrangement described on page 6. The mechanical drag assembly prevents any caking of dust within the hopper that might otherwise clog a hopper-type bottom. The dust is scooped out of the hopper by an endless drag chain for disposal outside the Dustraxtor system. In the illustration on page 4 the collected coal dust is dumped on a conveyor system and taken to a separate operation for useful by-product applications.



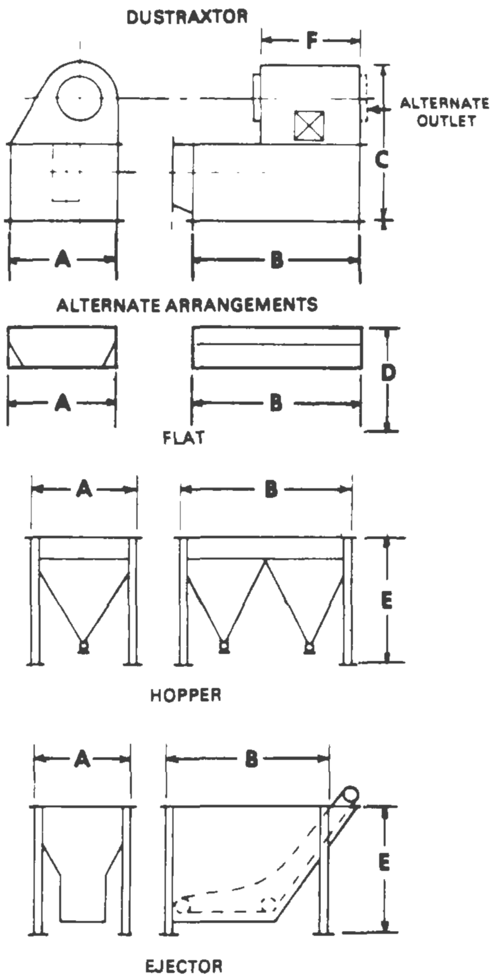
DUSTRAXTOR ACCEPTANCE

Here are some of the companies utilizing Dustraxtors for effective air pollution control:

American Cast Iron Pipe Co.
Armour Agricultural Chemical Co.
Armstrong Rubber Co.
Cannon Mills, Inc.
Coastal Chemical Co.
Continental Can Co.
Hershey Foods Corporation
Monsanto Chemical Co.
Molybdenum Corporation of America
Minnesota Mining & Mfg. Co.
National Phosphate Corporation
Pollock Paper Company
Reynolds Metal Co.
Sears, Roebuck & Company
Stauffer Chemical Company
Tennessee Valley Authority
U.S. Pipe & Foundry Co.
United States Steel Corporation
Vermiculite of Hawaii
Vulcan Materials Co.
West Virginia Pulp and Paper Co.
Woodward Iron Company

DUSTRAXTOR SPECIFICATIONS

APPROXIMATE DIMENSIONS (Not for Construction) — Dimensions Shown in Inches



SIZE	FAN	CFM	A	B	C	D	E	F
1	21.0B	1,200	36¼	13¼	101¾	42	60	13
2	4.0	2,400	36¼	26¼	101¾	42	60	26¼
3	4.5	3,600	36¼	39¼	105¾	42	60	39
4	5.0	4,800	36¼	52¼	109¾	50	60	39
5	5.5	6,000	36¼	65¼	112½	56	60	42
6	5.5	7,200	36¼	78¼	112½	60	60	44
8	5.5	9,600	72¼	52¼	112½	60	72	46
10	6.0	12,000	72¼	65¼	116¼	60	72	48
12	6.0	14,400	72¼	78¼	116¼	60	72	56
14	6.5	16,800	96¼	91¼	119¾	72	72	65
16	6.5	19,200	96¼	104¼	119¾	72	84	75
18	7.0	21,600	96¼	117¼	123	84	84	85
20	7.0	24,000	96¼	130¼	123	96	84	95
22	8.0	26,400	96¼	143¼	130⅝	96	84	105
24	8.0	28,800	96¼	156¼	130⅝	60	84	115
26	8.5	31,200	100¼	182¼	134¼	60	84	125
28	8.5	33,600	100¼	182¼	134¼	72	84	136
30	9.0	36,000	100¼	195¼	137⅝	72	96	145
32	9.0	38,400	100¼	208¼	137⅝	79	96	153
34	9.5	40,800	104	221¼	143 15/16	79	96	165
36	10.0	43,200	104	234¼	148⅜	84	96	180
38	10.0	45,600	104	247¼	148⅜	84	96	188
40	11.0	48,000	106	260¼	160⅝	96	96	197
42	11.0	50,400	106	273¼	160⅝	96	96	207

MEMBER
IGCI
INDUSTRIAL GAS CLEANING INSTITUTE INC

AMCA
MEMBER

Member Air Moving and Conditioning Association Inc

Zurn Air Systems was created by combining the resources of three well-known companies in the air moving, air controlling and air cleaning fields: Clarage Fan Company of Kalamazoo, Michigan; Swartwout, Inc. of Kokomo, Indiana; and Fly Ash Arrestor Corporation of Birmingham, Alabama. The new integrated division provides a one-source engineering and supply center for all air quality control, air moving and handling needs, coupled with supplemental benefits of enhanced dependability, broader selectivity and on-time delivery over a wide range of product capabilities. To request additional information merely contact any of the three locations indicated below.

ZURN — A Step Ahead of Tomorrow

ZURN INDUSTRIES, INC.
AIR SYSTEMS, DIV.

ONE CLARAGE PLACE KALAMAZOO, MI. 49001 (616) 349-1541	1000 N. TOUBY PIKE KOKOMO, IN. 46901 (317) 459-5151	BOX 1883 BIRMINGHAM, AL. 35201 (205) 252-2181

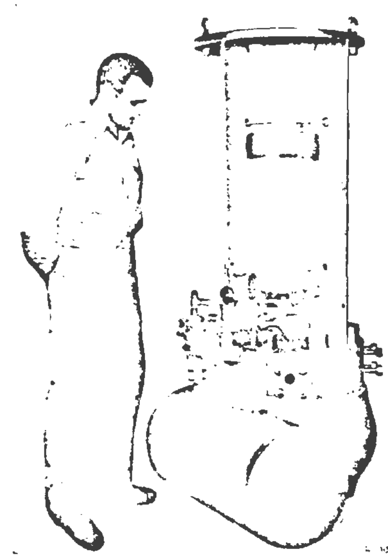
BULLETIN WS 6200-300 7.5M 2/70

Class VEM
Boiler Circulating Pumps

Ingersoll-Rand Centrifugal Pumps

INGERSOLL-RAND Class VEM pumps are vertical, single-stage units with single-suction impeller and double-volute casing. The entire assembly, including pump and driver, is arranged for mounting on the boiler piping, the suction and discharge nozzles being welded in position. This arrangement permits free movement of the pump when operating at high temperatures . . . prevents stress and strain on both the boiler piping and pump.

Pressures: to 2900 psi.
Heads: 100 to 175 ft.
Capacities: 4000 to 14,000 gpm.
Sizes: 10, 12, 14, 16 in.
Temperatures: up to and above 650°F.



VEM Pump less driver.

APPLICATION

Class VEM pumps are designed for use with high-pressure controlled circulation boilers that require an efficient, rugged pump for this severe service. VEM units are so constructed that they can be dismantled and re-assembled in the field with a minimum of down time.

GENERAL CONSTRUCTION

Casing is made of cast carbon steel to ASME specs. Its double-volute design eliminates radial thrust at off-peak capacities. Stainless inlay is provided at the gasket fit. Welding neck nozzles of sufficient size provide reasonable velocities.

Casing Cover is of forged carbon steel with stainless inlay at gasket fits. It is drilled for the necessary injection, bleed-off and cooling water connections.

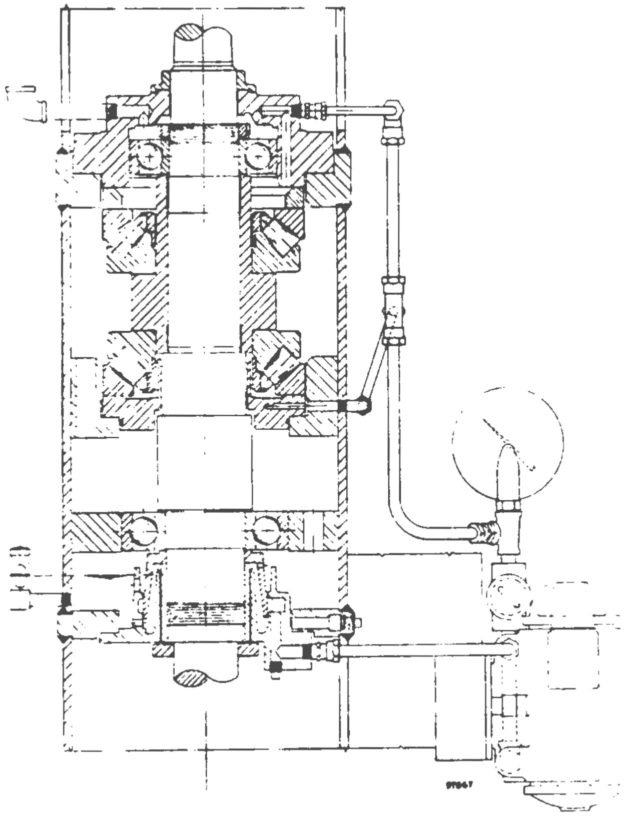
Pressure Bolting of heat treated alloy steel is accessible for easy removal with an impact wrench. Removal of pressure bolting nuts allows complete pumping element to be removed from the casing without disturbing suction and discharge piping or basic pump alignment.

Stuffing Box Extension of one piece cast chrome-moly steel, provides adequate packing depth with integral cooling jacket. The smothering type gland is made of stainless steel.

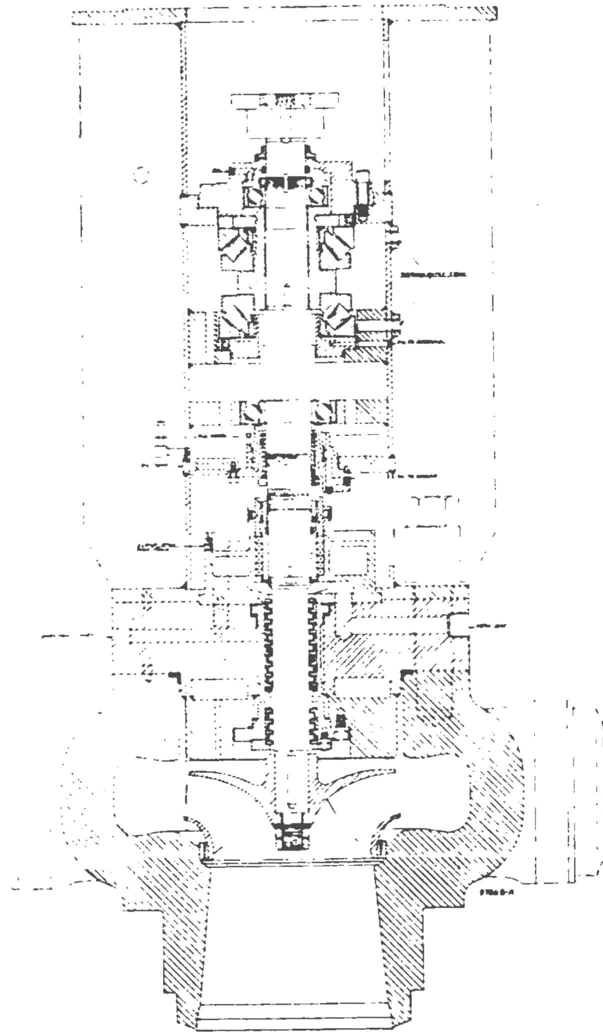
Shaft is made of heat-treated high-strength, corrosion-resistant material. It is of the rigid cantilever type operating below first critical speed, entirely supported by external bearings. Alignment is not influenced by casing distortion due to temperature or pressures.

Impeller of chrome-moly steel is of the single-suction type. Renewable impeller and casing wearing rings of adequate clearance are provided. Impeller is keyed to the shaft and secured by stainless steel lock nuts. All hydraulic passages are smoothly finished.

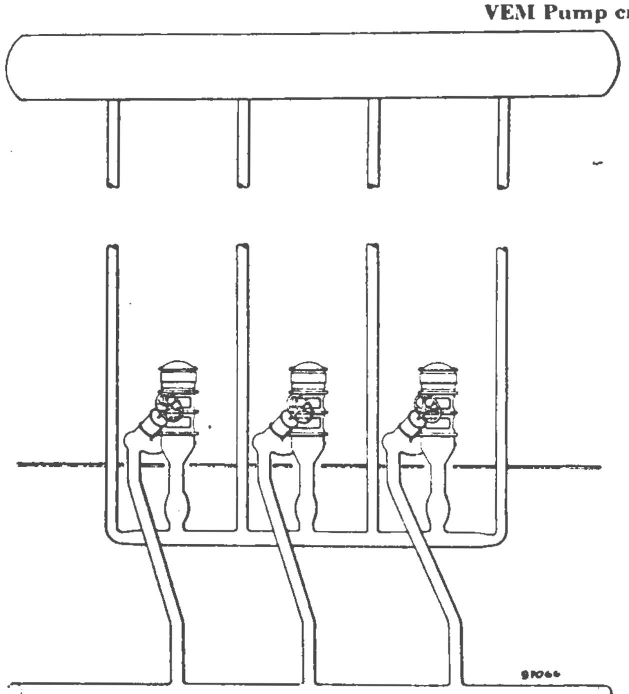
Sealing Arrangement - Boiler feed water injection is used to prevent high temperature boiler water from entering the pressure breakdown device. A throttle bushing and sleeve is used to reduce to a minimum the amount of injection water entering pump. In addition, a floating seal pressure breakdown device is located between the injection point and the stuffing box, thus keeping the quantity of injection water required to a minimum.



VEM lubrication system.



VEM Pump cross-section.

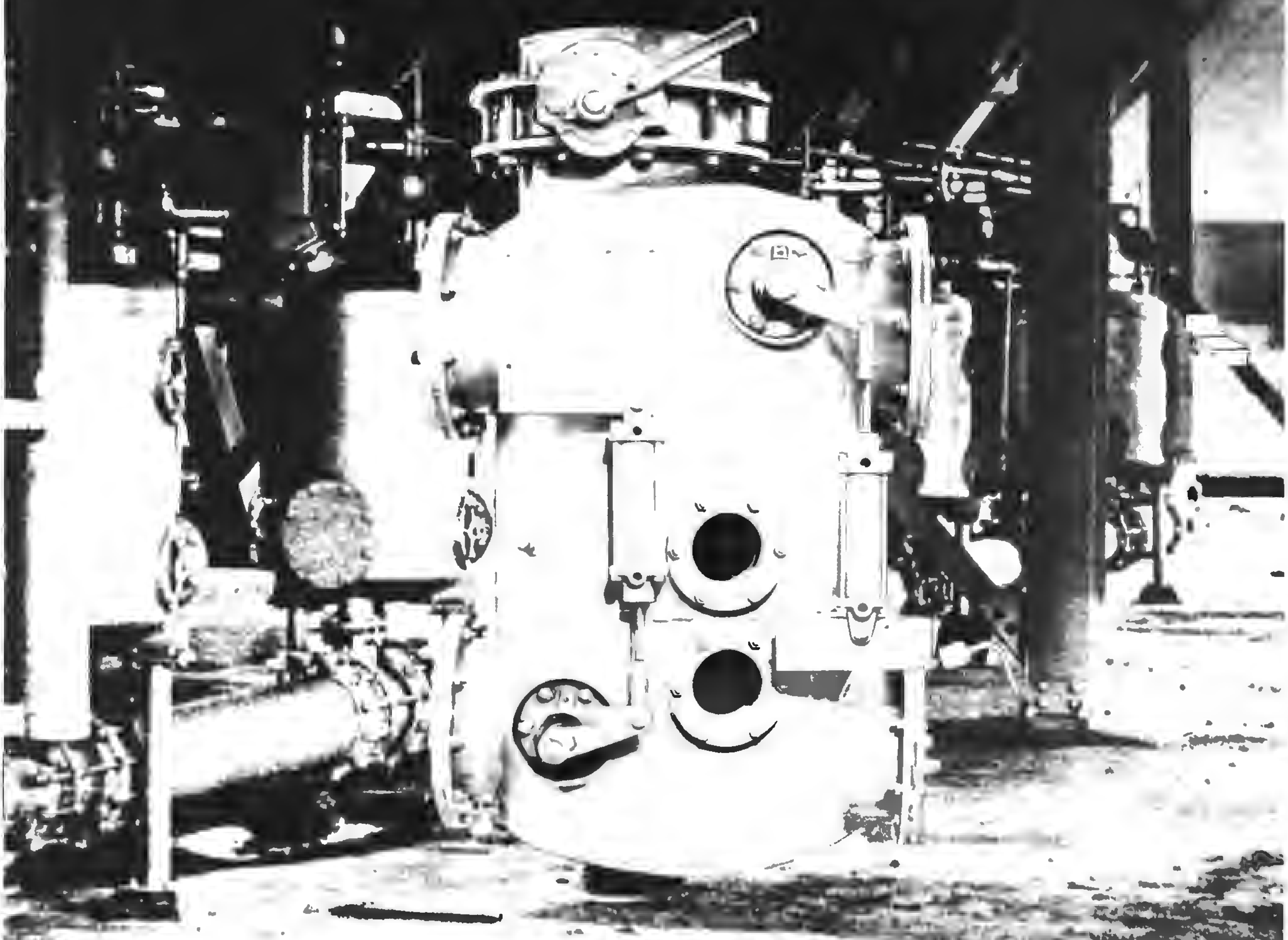


Typical VEM installation.

UNITED

NUVA FEEDER®

POSITIVE PRESSURE PNEUMATIC CONVEYOR



NUVA FEEDER covered by following patents:
U.S.—2,9[illegible],417 Canada—[illegible]
Great Britain—[illegible],7[illegible] Japan—[illegible]

Represented by
Watson Power Equipment Co.
[illegible] WILSON MILLS ROAD
CLEVELAND, OHIO 44143
TELEPHONE 216/449-3555

UNITED CONVEYOR CORPORATION
6505 NORTH RIDGE BOULEVARD, CHICAGO, ILLINOIS 60626 U.S.A.
PHONE (AREA 312) 761-4100

4NF-68A

D-167

UNITED

DESIGN AND PRINCIPLE OF OPERATION

The NUVA FEEDER System consists essentially of four basic units—

1. **The pressure NUVA FEEDER attached directly to the dust collector hopper which automatically feeds the dry, free-flowing fly ash into the conveyor system.**

2. **The conveyor pipe line with required fittings that connects to the outlet of the pressure feeder and extends to the final delivery point.**

3. **A positive displacement motor-driven blower to create the air flow in the conveyor pipe line.**

4. **Venting of storage bin. This is accomplished with a bag filter or vent returning to precipitator. The latter results in a closed circuit installation.**

PRESSURE NUVA FEEDER

The principle of operation of a suction conveyor and a pressure conveyor is essentially the same.

In both cases, higher pressure at the inlet end of a transport pipe conveys air and material through the pipe to the lower pressure area.

In the suction conveyor, a vacuum exhauster reduces the air pressure at the discharge end. In a pressure conveyor, a pressure blower builds up the pressure at the inlet end.

The inherent advantages of the pressure conveyor are—

The conveying medium passes through the blower as clean air before it reaches the transport pipe where the material is introduced. This eliminates the problem of erosion on the pressure blower.

The pressure system by compressing the air uses a denser medium with which to convey the material. This considerably improves the efficiency and capacity of the pressure system.

The power available to a vacuum conveyor is limited to atmospheric pressure. The power available for a pressure system is unlimited except for practical considerations.

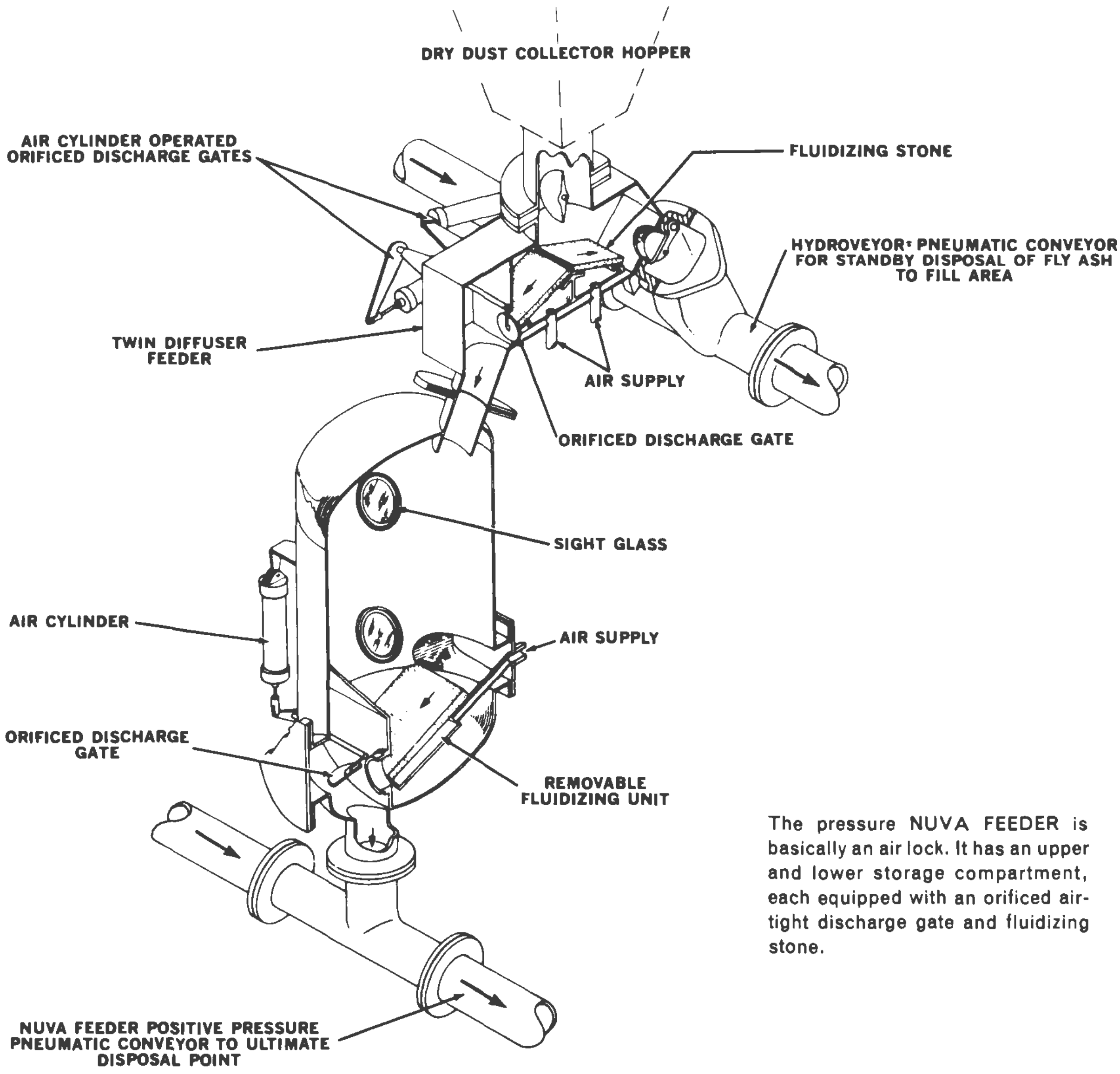
It is a single stage system that conveys the dry fly ash directly from the hopper and delivers it to its disposal point without any intermediate transfer point.

The chief problem in the development of a pressure conveyor for handling exceedingly abrasive material is the design of the unit which will feed this material against the conveying pressure into the transport line.

Rotary vane feeders, sometimes called air lock feeders, have been extensively tried in various forms and all have failed because of the impossibility of maintaining close clearances so the unit will function.

Pressure tanks which are in sequence filled with material, sealed off and blown with high air pressure into the conveyor line are in use. These units take large headroom, high pressure storage vessels and operate at exceedingly high velocities through certain sections of the system. This causes excessive wear and what is commonly called slugging shocks which make the conveyor line difficult to support.

Screw type feeders are used where a compacting screw solidifies the material to the point where the pressure in the conveyor will not blow back through the screw. These are successful with certain fine materials but are subject to high maintenance if exceedingly abrasive material is handled. The additional power used by the compacting screw is excessive, amounting to a fourth to a half of the total power used. Compressed air at relatively high pressure is also required. As a result, these conveyors are inefficient as far as power consumption is concerned. (continued on page 6)



The pressure NUVA FEEDER is basically an air lock. It has an upper and lower storage compartment, each equipped with an orificed air-tight discharge gate and fluidizing stone.

Figure 3.

The operation of the NUVA FEEDER is as follows:

With the upper discharge gate open and the lower gate closed, material flows into the storage chamber for approximately one-half of a cycle.

For the other one-half of the operating cycle, the upper gate is closed, the lower gate opened, and material is discharged into the conveyor line.

The cycle automatically keeps repeating. A continuous flow of material into the conveyor line can be accomplished by the use of two or more feeders.

UNITED

DESIGN AND PRINCIPLE OF OPERATION

(continued from page 4)

The United Conveyor Corporation has been working on this problem for many years. The following requirements were realized:

For efficient operation, material must be delivered to the conveyor at a relatively accurate rate and means must be provided to control this rate. This allows the power requirement to be accurately calculated and permits the conveyor to operate within these power calculations.

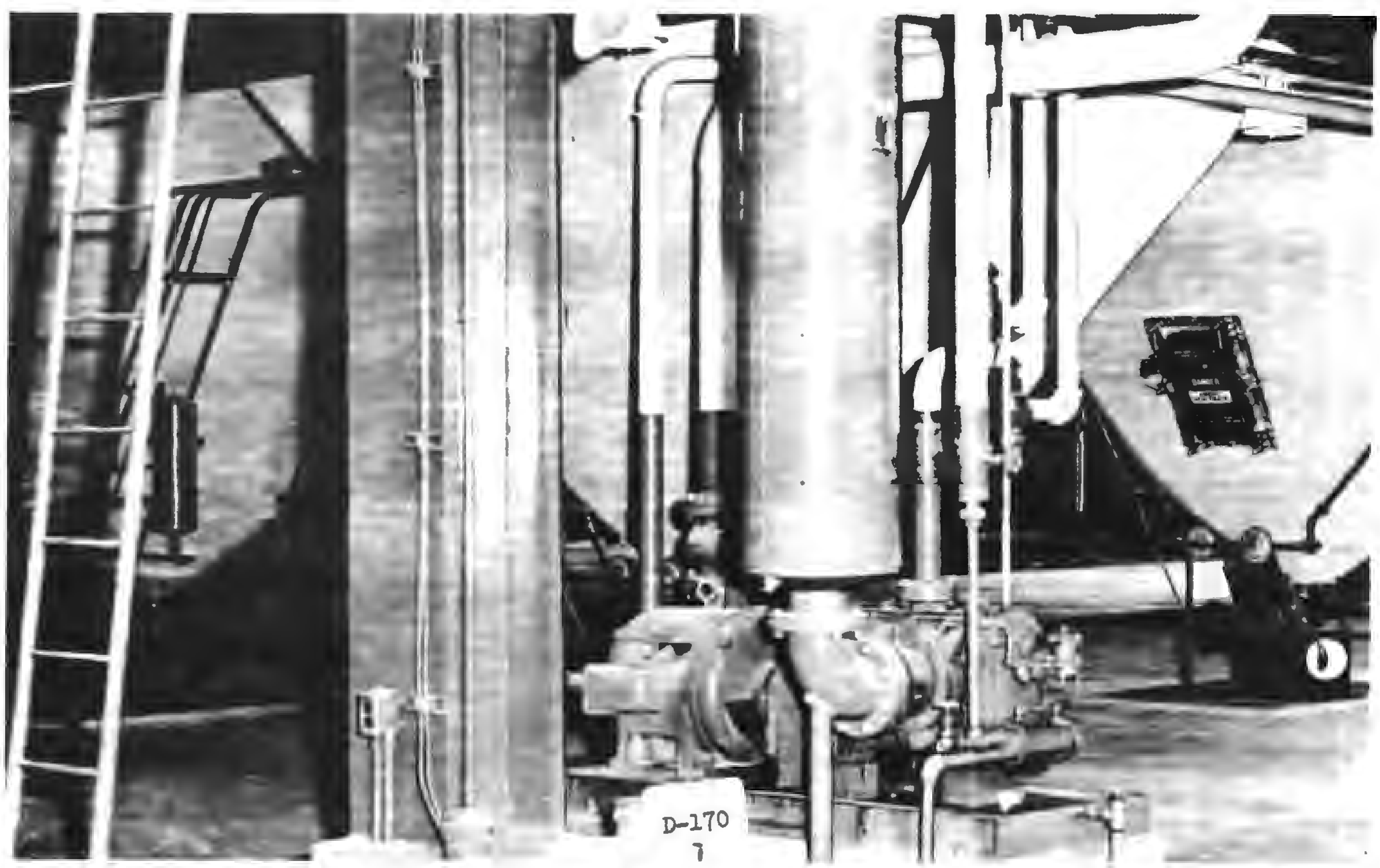
In order to obtain a smooth and nearly uniform rate of feeding, more than one pressure feeder is used. This provides for a half cycle of conveying and a half cycle for charging of the pressure feeders so that in all cases full capacity will be fed to the conveyor line.

The development of recent years of the fluidizing of finely divided material so that it will flow more like a fluid than like a solid, has revealed that a fine material properly fluidized will discharge from the opening at an exceedingly uniform rate. This allows us to make a feeder by the use of small orifice openings which will not avalanche and flood the charging hopper or the conveyor line. This is the basic discovery that has made the United pressure conveyor both unique and practical.

Another advantage of this method is that in using a small orifice we also are able to use a small gate to control the feed. This small gate is easy to maintain airtight compared to the large diameter openings required in conventional types of air locks.

In addition to using this means of controlling the feed, the openings are designed a little larger than required for maximum feed and a full load control is added which operates from the pressure in the transport line. Under given conditions, if the amount of material fed to the transport line is increased, the required pressure to move this through the conveyor line is increased. Taking advantage of this fact, our NUVA FEEDERS are arranged so that if the pressure exceeds a predetermined pressure, the feeding gate on the unit delivering material to the line is momentarily closed until the pressure drops to the desired full load capacity.

These are the conceptions which have enabled us to develop a compact unit to give uniform and high capacity delivery to a transport line at an exceedingly economical use of power.



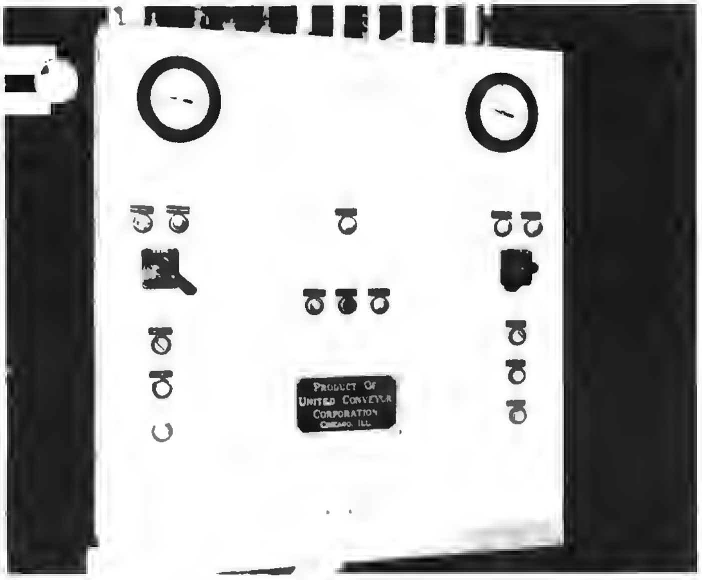


Figure 4.

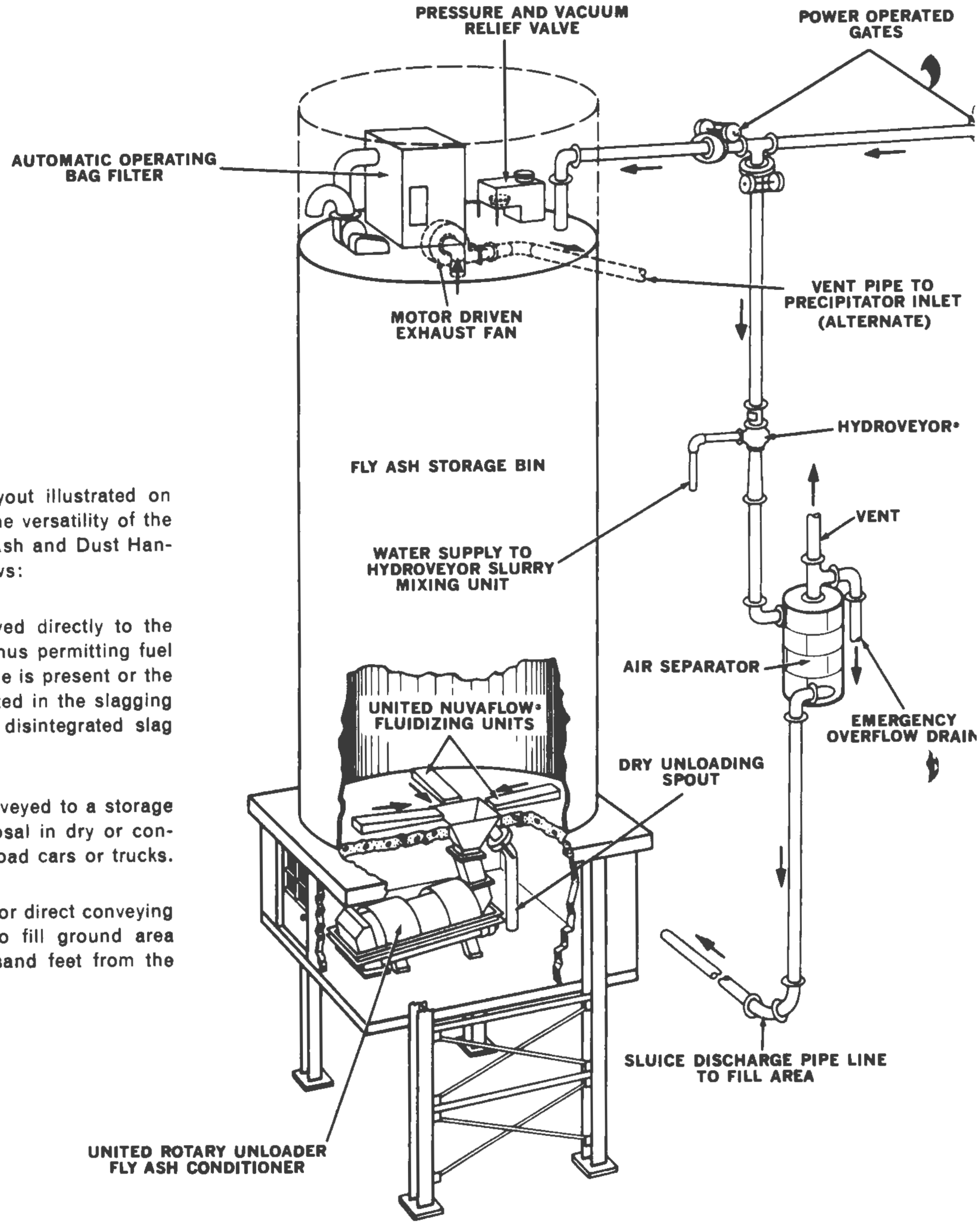
Fully controlled conveying obtained through the UNITED NUVATROL® cabinet designed for automatic and remote control operation of the NUVA FEEDER System.

NUVA FEEDER Fly Ash Conveyor System serving precipitator hoppers. Note the motor driven blower supplying the air flow to the conveyor system.

Figure 5.

Figure 6.

Fly ashes are conveyed directly and in a single operation from dust collector hoppers to an elevated storage bin located in the yard of the Power Station. Dry unloading is provided for sale of fly ash. Alternatively, a UNITED Rotary Unloader is used for conditioning fly ash hauled away from the plant by trucks without creating a dust nuisance.



The diagrammatic layout illustrated on these pages shows the versatility of the NUVA FEEDER Fly Ash and Dust Handling System as follows:

Fly ashes are conveyed directly to the furnace for refiring, thus permitting fuel recovery if combustible is present or the fly ashes can be melted in the slagging furnace so that only disintegrated slag is produced.

Fly ashes can be conveyed to a storage bin for ultimate disposal in dry or conditioned state to railroad cars or trucks.

System is adaptable for direct conveying of slurried fly ash to fill ground area located several thousand feet from the power station.

CONVEYOR PIPE LINE

DURITE® wear resisting alloy fittings with extra heavy wearing backs to withstand concentrated wear that occurs at the bends are used throughout the system. NUVALOY® wear resisting alloy is used in the conveyor pipe between the various fittings.

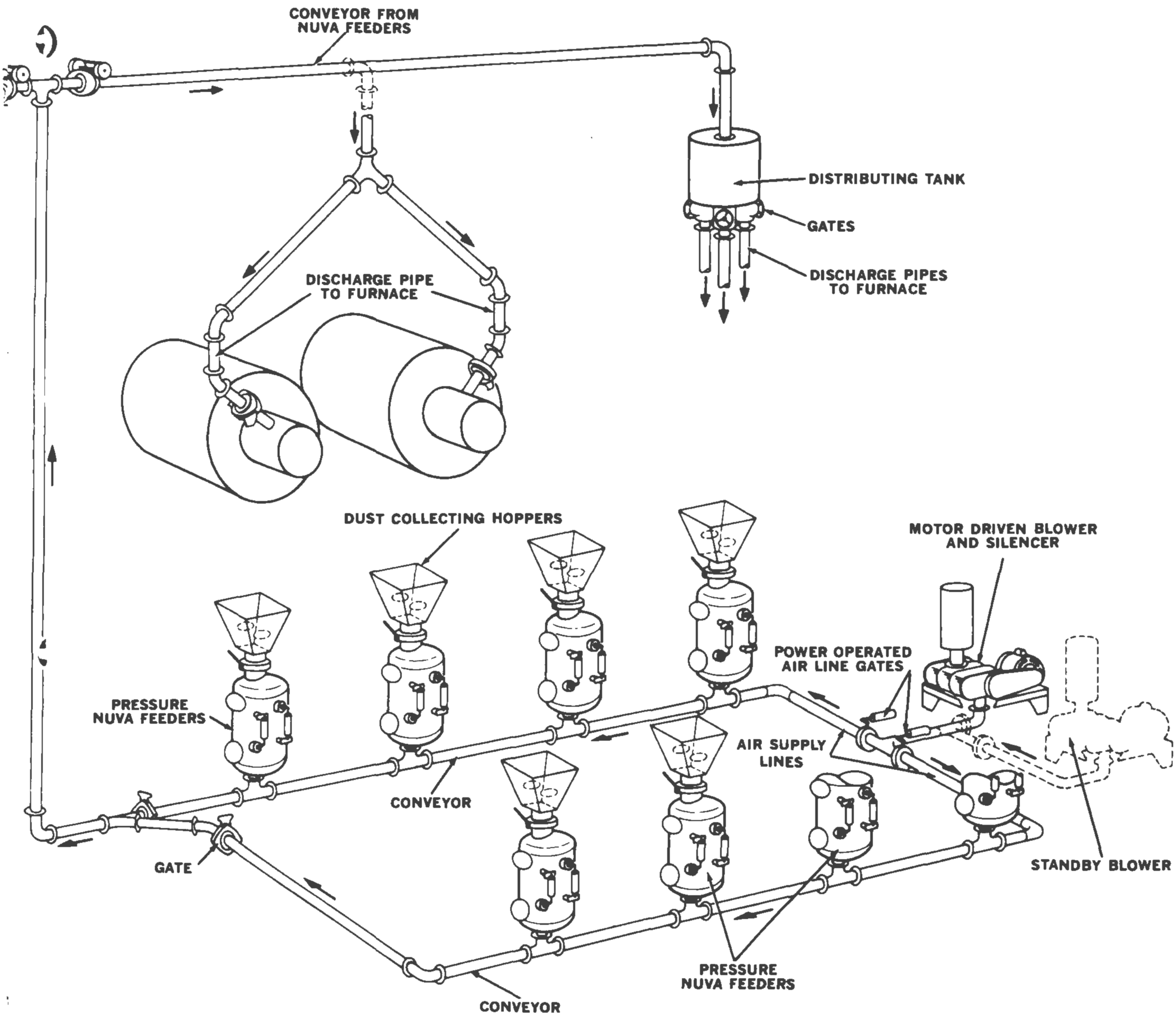


Figure 7.

POSITIVE DISPLACEMENT BLOWER

The machinery requirements for the NUVA FEEDER positive pressure system are relatively simple. The entire system is operated by a positive displacement blower located near the dust collector hoppers which supplies the conveying air as well as the fluidizing air for the feeders.

UNITED

Basic Components for UNITED NUVA
Engineered for the Heaviest

Many basic and specially designed components have been developed by UNITED engineers for pneumatic and hydraulic conveyors handling fly ash, dust and similar abrasive materials. Illustrated on pages 10 and 11 are a few of the components used in several thousand installations furnished by

Figure 8.
Bag filter

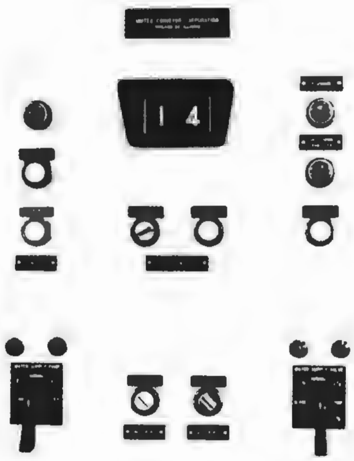


Figure 9.
Custom engineered NUVATROL control cabinet designed for automatic and/or remote control operation

Figure 10.
NUVALOY and DURITE wear-resisting alloy pipe with standard flange couplings

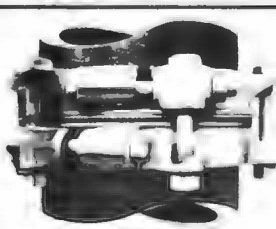


Figure 11.
NUVALOY pipe with high pressure flange couplings

Figure 12.
45° elbow with extra heavy removable wearback and handhole. 15°, 22½° and 30° elbows also available

Figure 13.
45° long radius solid elbow—designed extra heavy in area of concentrated wear. 15°, 22½° and 30° elbows also available

Figure 14.
Totally enclosed conveyor gate, manually or air power operated

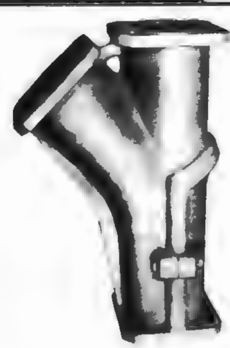


Figure 15.
45° lateral with extra heavy removable wearback

Figure 16.
45° solid type lateral—designed extra heavy in area of concentrated wear

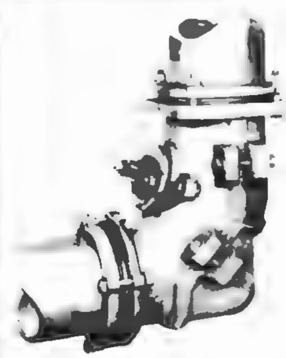


Figure 17.
90° elbow with extra heavy removable wearbacks and handhole

FEEDER Pneumatic Conveyor Duty Requirements

UNITED in over 48 years of experience in this field.
All fittings are DURITE iron castings. DURITE is a special wear-resisting, uniformly dense and close-grained alloy metal particularly suitable for handling abrasive materials.

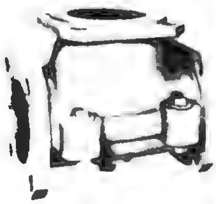


Figure 18.

Tee with extra heavy removable wearback

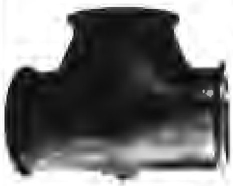


Figure 19.

Tee—solid type—designed extra heavy in area of concentrated wear

Figure 20.

Handhole section

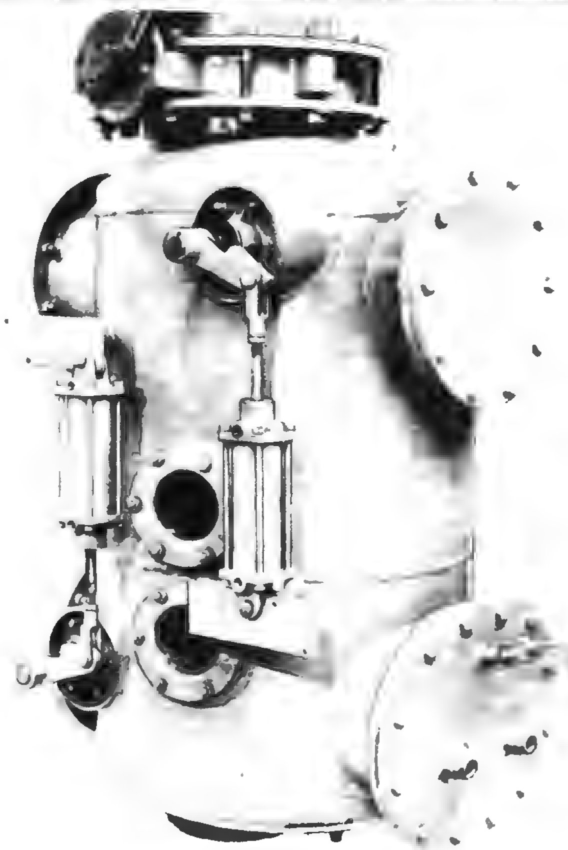


Figure 21.

UNITED NUVA FEEDER

Figure 22.

Motor-driven blower to supply conveying air to NUVA FEEDER system

Figure 23.

Expansion joint

Figure 24.

90° long radius solid elbow—designed extra heavy in area of concentrated wear

Figure 25.

HYDROVEYOR air handling unit mixes fly ash and water for discharge

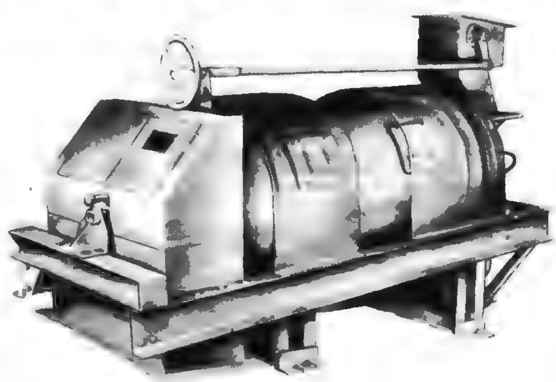


Figure 26.

Rotary unloader for moistening and conditioning dust and fly asn for discharge to trucks or railroad cars

Figure 27.
NUVA FEEDER pneumatic fly ash conveyor pipeline as it comes from the generating station

DELIVERING DRY FLY ASH ONE AND ONE-HALF MILES BY PIPE LINE TO TURN A WASTE PRODUCT INTO A PROFIT

In 1961, a well known cement company decided to build a cement product plant about 1½ miles away from an electric generating station. The availability of fly ash was one of the factors influencing the selection of this site by the cement company. They could use fly ash as a substitute argillaceous component of the raw mix. The power company had the fly ash, a waste product, being sluiced to fill areas which were rapidly filling up.

PROBLEM

How to get approximately 400 tons a day of dry fly ash from the power plant to a bin on the cement company property 1½ miles away?

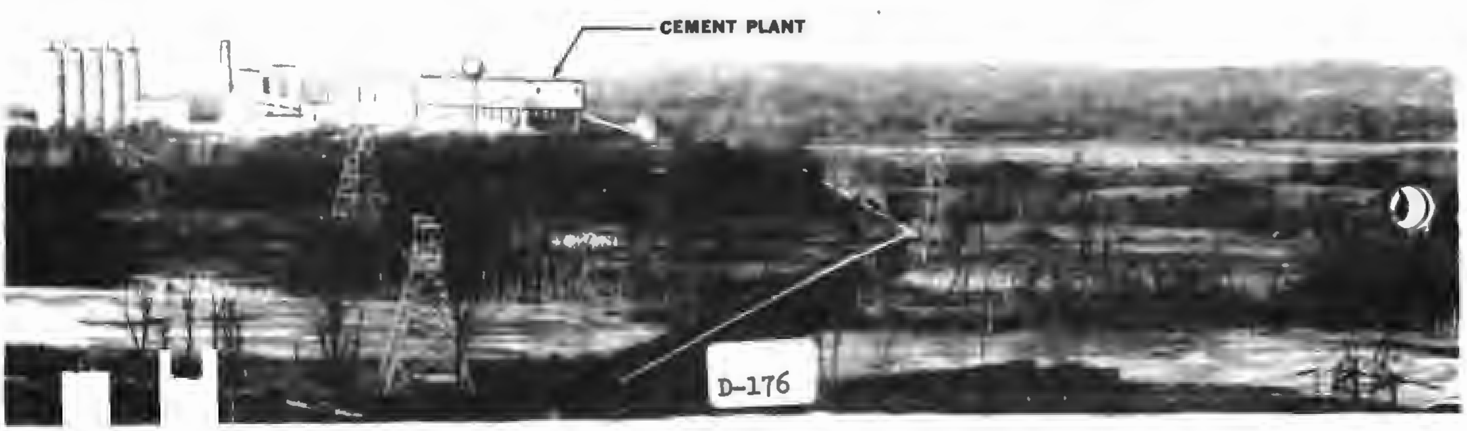
SOLUTION

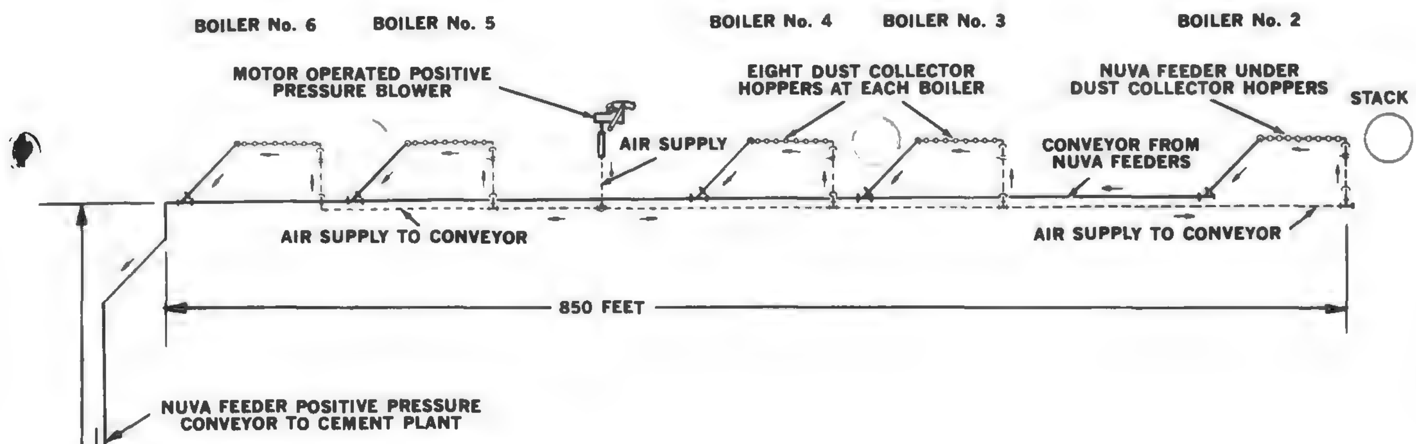
Convey it directly from the dust collectors to the storage bin in a single-stage pneumatic pipeline system illustrated on pages 12, 13 and 14.

Initially, several different schemes were investigated, all involving rehandling the material once or twice. A conventional approach was to convey from the dust collectors to an intermediate storage or transfer bin with a relatively short pneumatic system and then, either truck the material from that point to the cement plant or install a second pneumatic system. Trucking involves manpower and a problem of transferring material from trucks to bin at the cement plant. This expensive equipment, with weatherproof roads, requires maintenance which had to be considered in addition to providing means for eliminating dust nuisance at each transfer point. A two-stage pneumatic system meant extra equipment and extra expense.

No one, to our knowledge, had ever conveyed fly ash or similar material such a great distance in a single-stage pipeline system before, but based on experience with several long systems in successful operation, UNITED knew it could be done. When the idea was presented, the cement and power companies readily accepted it because of the obvious operating advantages and savings.

Figure 28.
1½ mile long NUVA FEEDER fly ash conveyor from power plant dust collector hoppers.
Note the cement plant in the distant background.





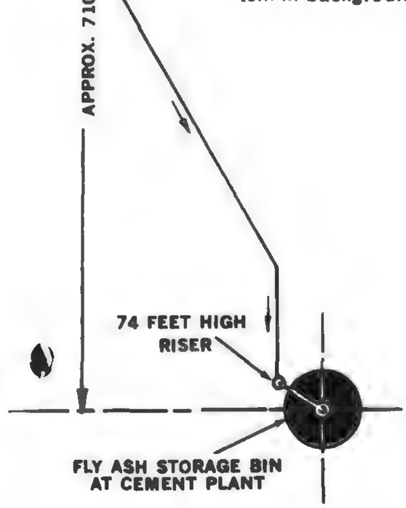


Figure 29.

The principles employed in the design of the highly successful NUVA FEEDER pressure feeder were used by UNITED engineers to develop a high pressure feeder required for long distance conveying. Once the method of feeding from dust collector hoppers at negative pressure to the conveyor line under relatively high pressure was perfected, only the selection of proper size pipe line and positive pressure blower remained.

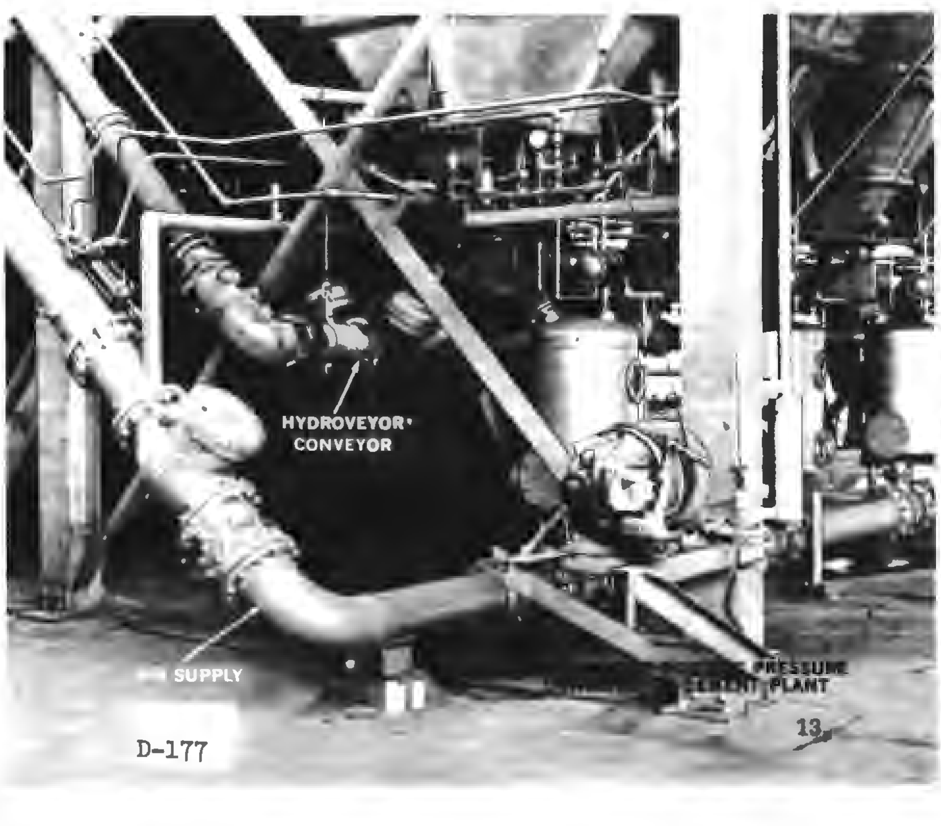
The diagrammatic layout shows the system employed. A pressure NUVA FEEDER is installed under each of the dust collector hoppers of the boilers being served. The fly ash conveyor pipeline connects each group of feeders through the main header leading to the cement plant.

The existing HYDROVEYOR pneumatic fly ash conveyors originally installed for conveying fly ash to the fill area were left in place for use any time the cement plant does not require all of the fly ash being produced by the boilers. Special by-pass gates, illustrated in figures 3,30 and 31, at each dust collector hopper permit discharge of fly ash into either the pressurized transport system for delivery to the cement plant or into a combination pneumatic vacuum and sluice type HYDROVEYOR system to the fill area.

The capacity of this system is approximately 35 tons of fly ash per hour with a power consumption of approximately 13 KW per ton.

Figure 30.

NUVA FEEDER pneumatic fly ash conveyor with HYDROVEYOR system in background



UNITED

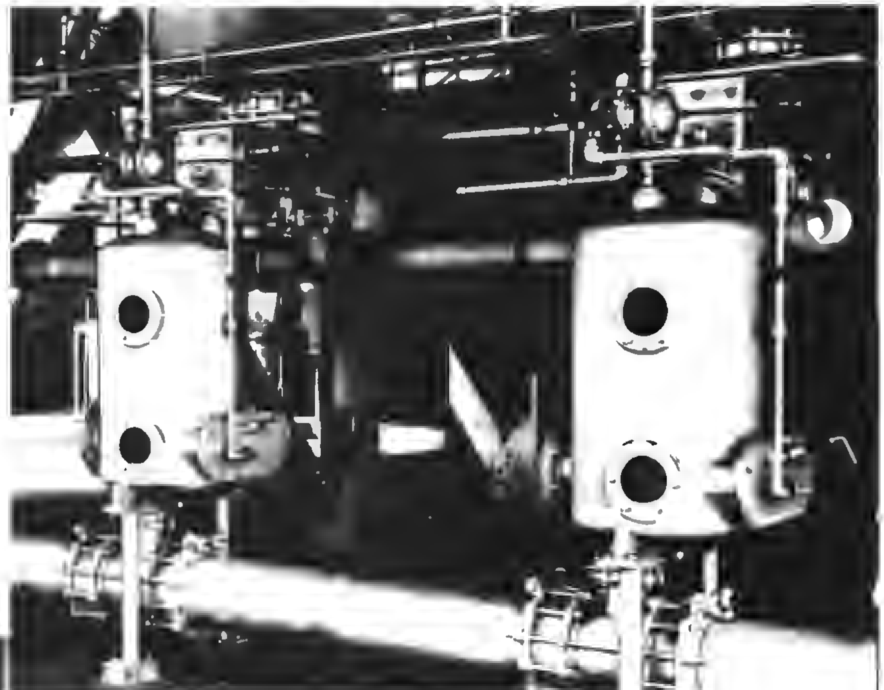


Figure 31.

Two of the forty NUVA FEEDERS installed in this plant

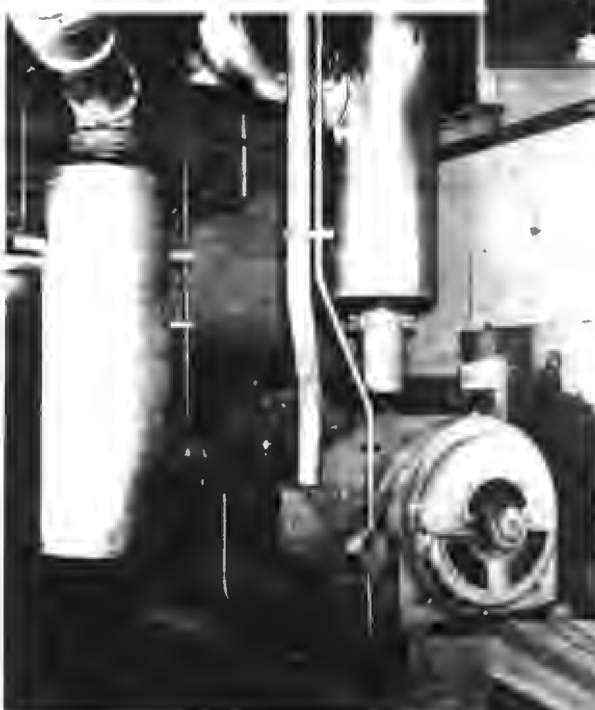


Figure 32.

Typical arrangement of positive displacement blower with silencer supplying conveying air to NUVA FEEDER pneumatic conveyor

Figure 33.

Discharge end of the fly ash conveyor at the cement plant fly ash storage bin

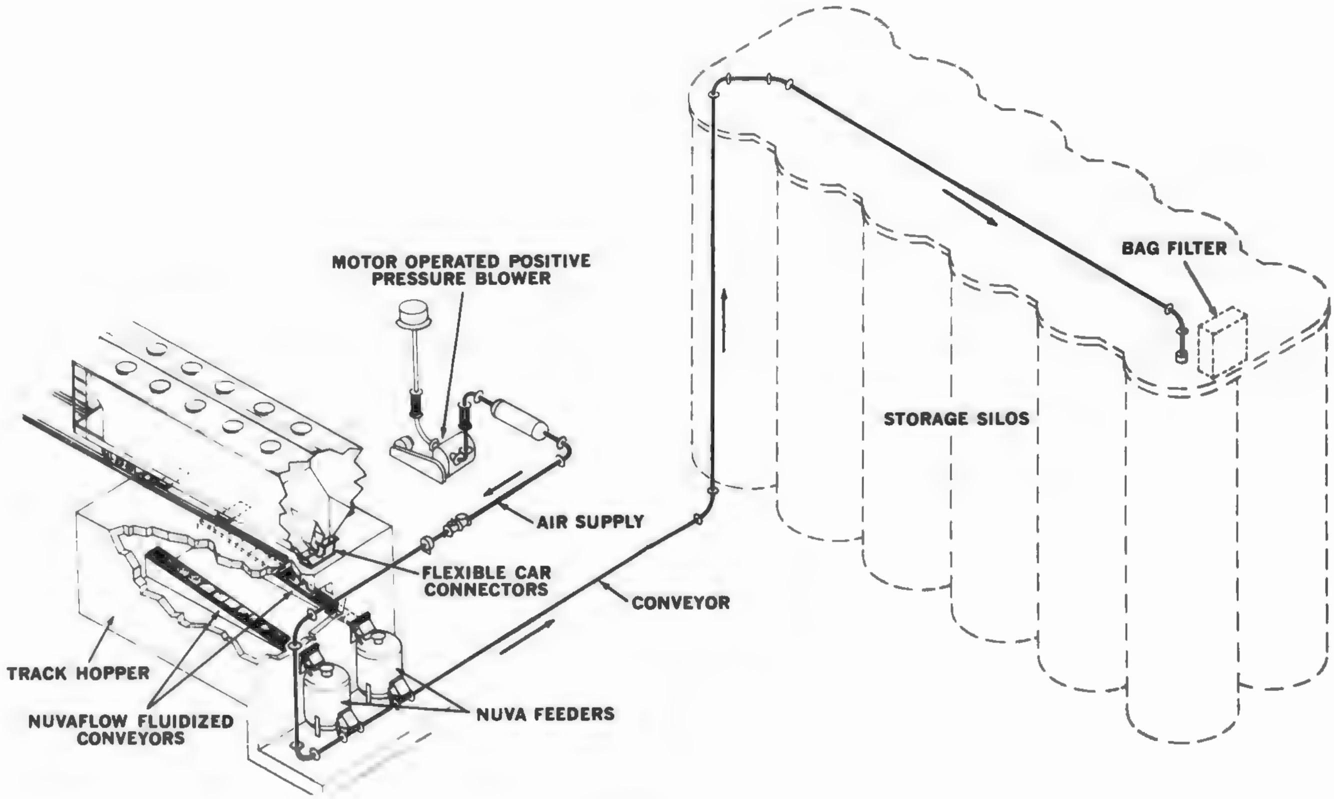


Figure 34.

The system illustrated above, furnished for a large midwestern cement company, is used to unload fly ash from hopper railroad cars and convey it to a storage bin at the rate of 120 tons per hour.

Conveyor utilizes two NUVA FEEDERS which alternately feed material into the conveyor pipe line resulting in a continuous flow of material to the bin.

Two NUVA FEEDERS furnished for plant illustrated above, to give 120 tons per hour capacity.

Figure 35.

UNITED

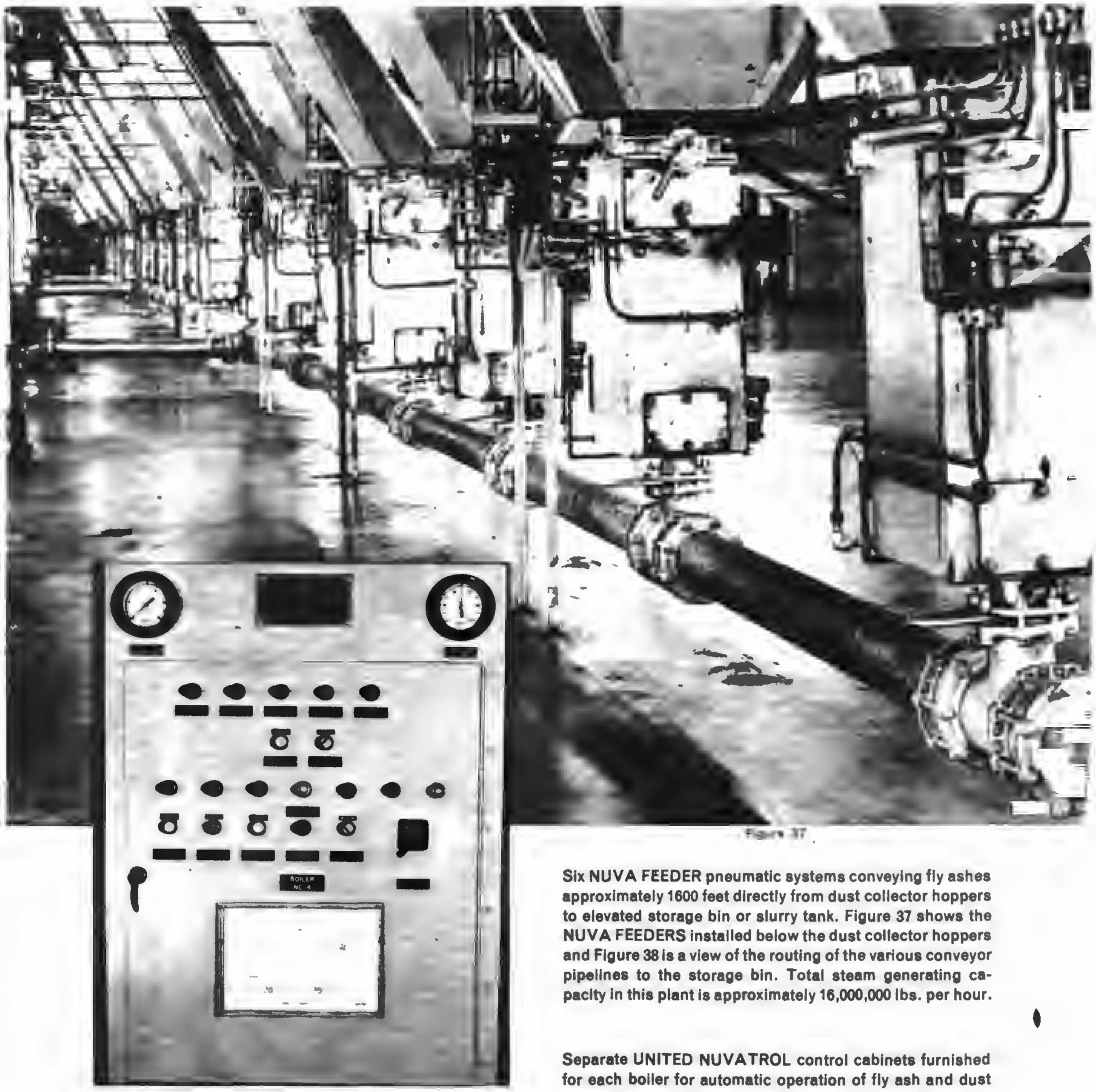


Figure 37

Six NUVA FEEDER pneumatic systems conveying fly ashes approximately 1600 feet directly from dust collector hoppers to elevated storage bin or slurry tank. Figure 37 shows the NUVA FEEDERS installed below the dust collector hoppers and Figure 38 is a view of the routing of the various conveyor pipelines to the storage bin. Total steam generating capacity in this plant is approximately 16,000,000 lbs. per hour.

Figure 36.

Separate UNITED NUVATROL control cabinets furnished for each boiler for automatic operation of fly ash and dust removal system

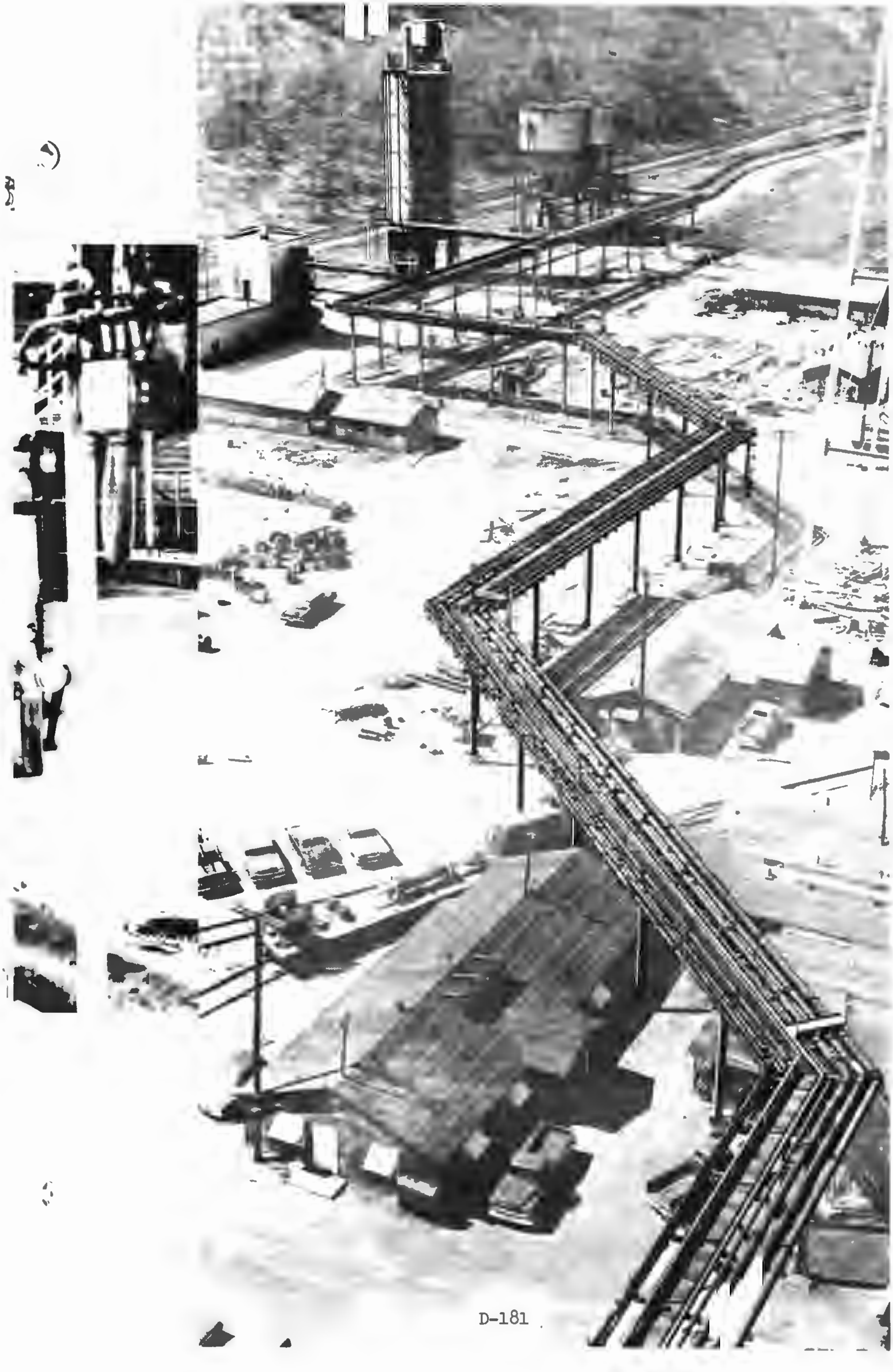


Figure 38.

UNITED

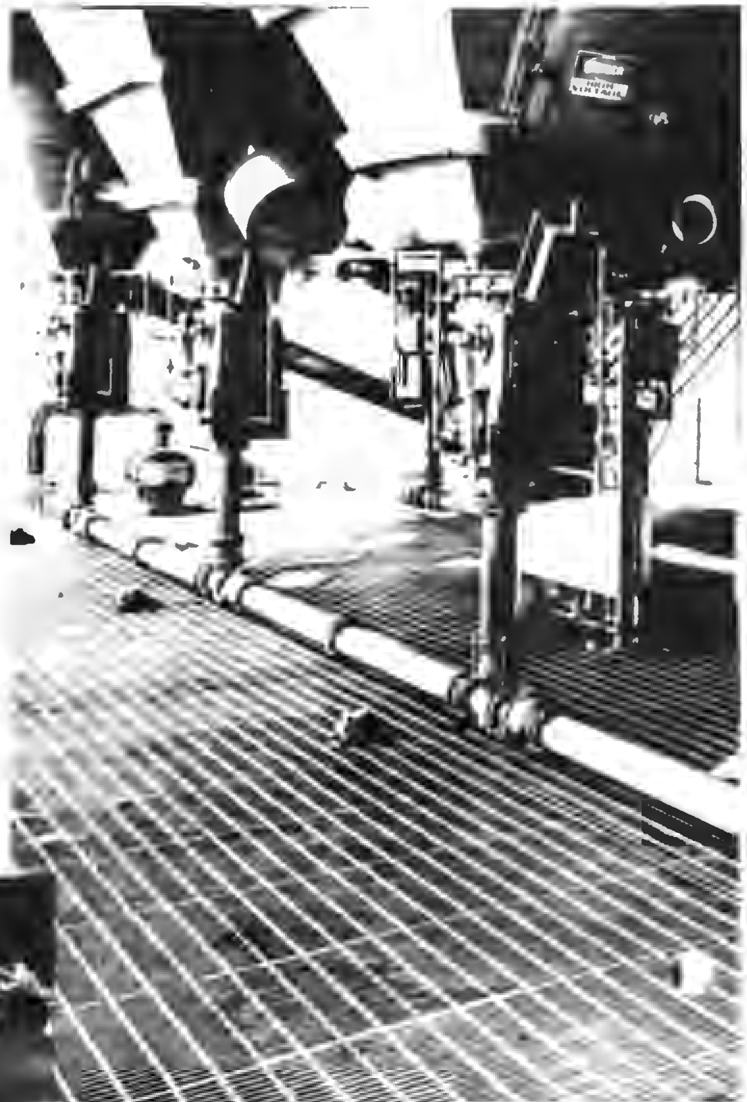


Figure 39.

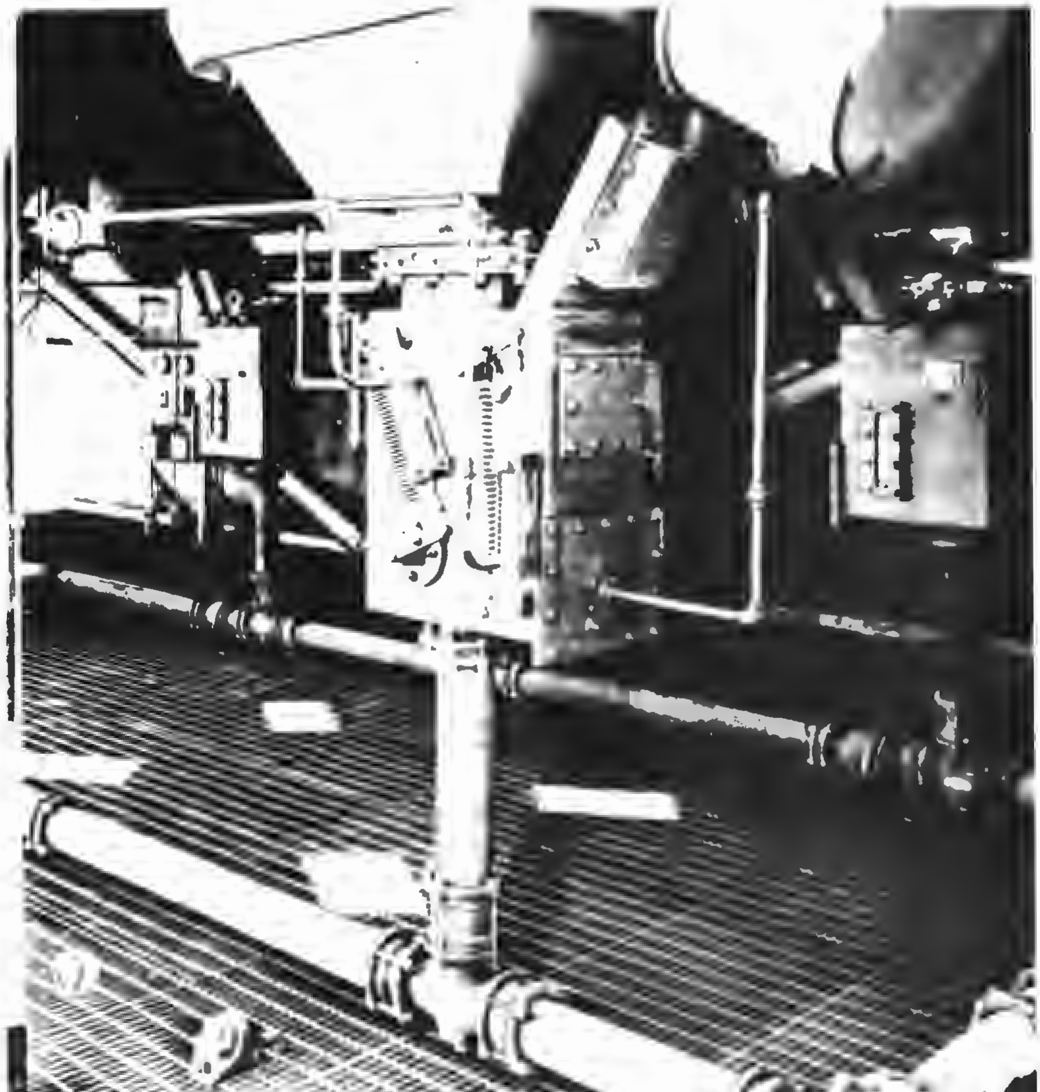
This NUVA FEEDER System is one of six pneumatic conveyors furnished for a large Central Station Power Plant with a total capacity of approximately 8,000,000 lbs. of steam per hour.

Figure 40.

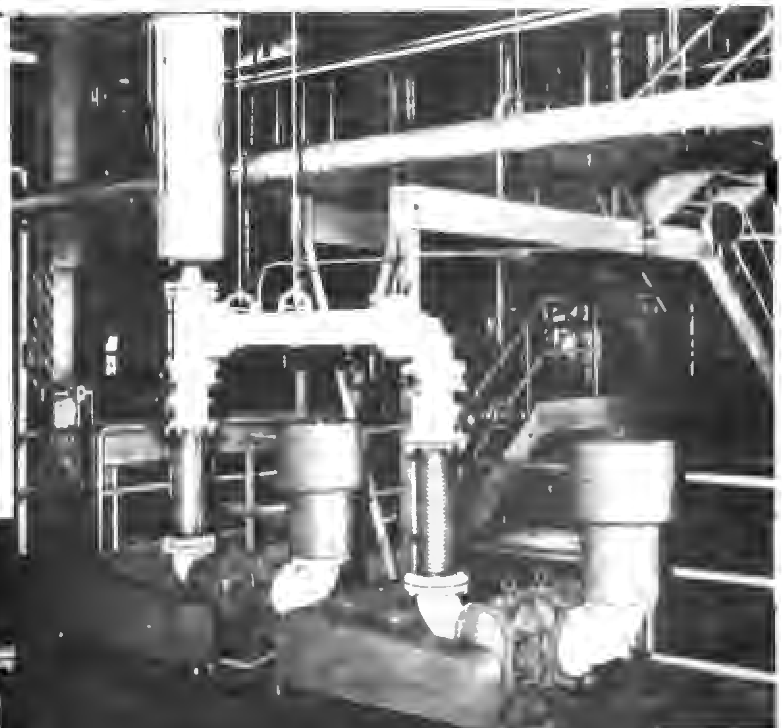
Close-up view of NUVA FEEDER air lock with motor-driven blower supplying the air to the conveying system.

Figure 41.

Typical arrangement of NUVA FEEDER conveyor, Figure 41, distributing fly ash to three points for refiring into the furnace.



UNITED



Motor-driven blower with standby blower supplying conveying air to NUVA FEEDER System.

Figure 42.

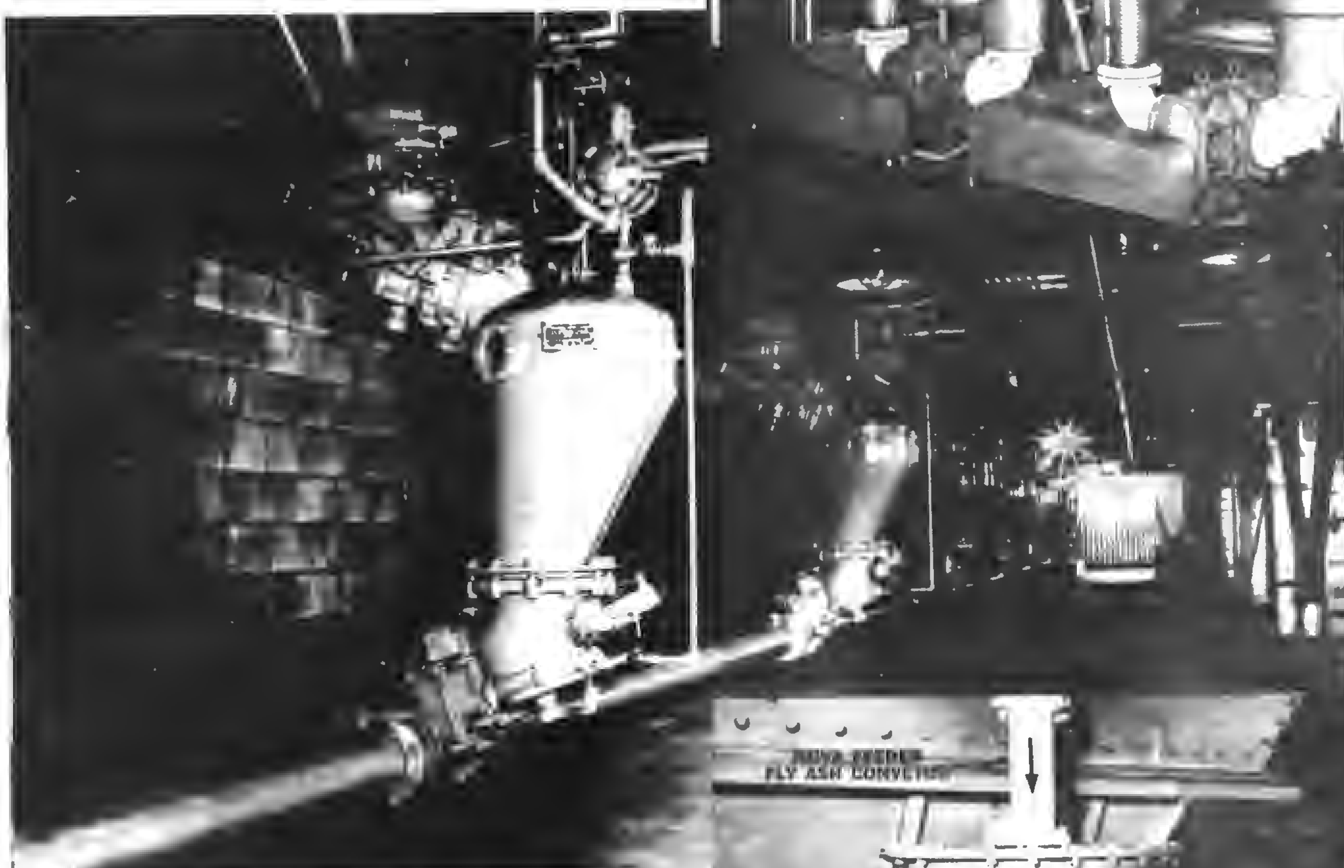


Figure 43.

Typical installation of NUVA FEEDERS connected directly to the fly ash collector hoppers.

Figure 44.

NUVA FEEDER positive pressure pneumatic conveyor discharges to distributing tank for refiring fly ash to furnace burners, or coal feed pipelines

Figure 45.

Two NUVA FEEDER conveyors discharge to a 1650 ton fly ash storage bin

Figure 46.

Automatic operating bag filter type dust collector located on roof of storage bin.

UNITED

In this central station NUVA FEEDER pneumatic systems serve two steam generators with a total capacity of 8,000,000 pounds per hour.

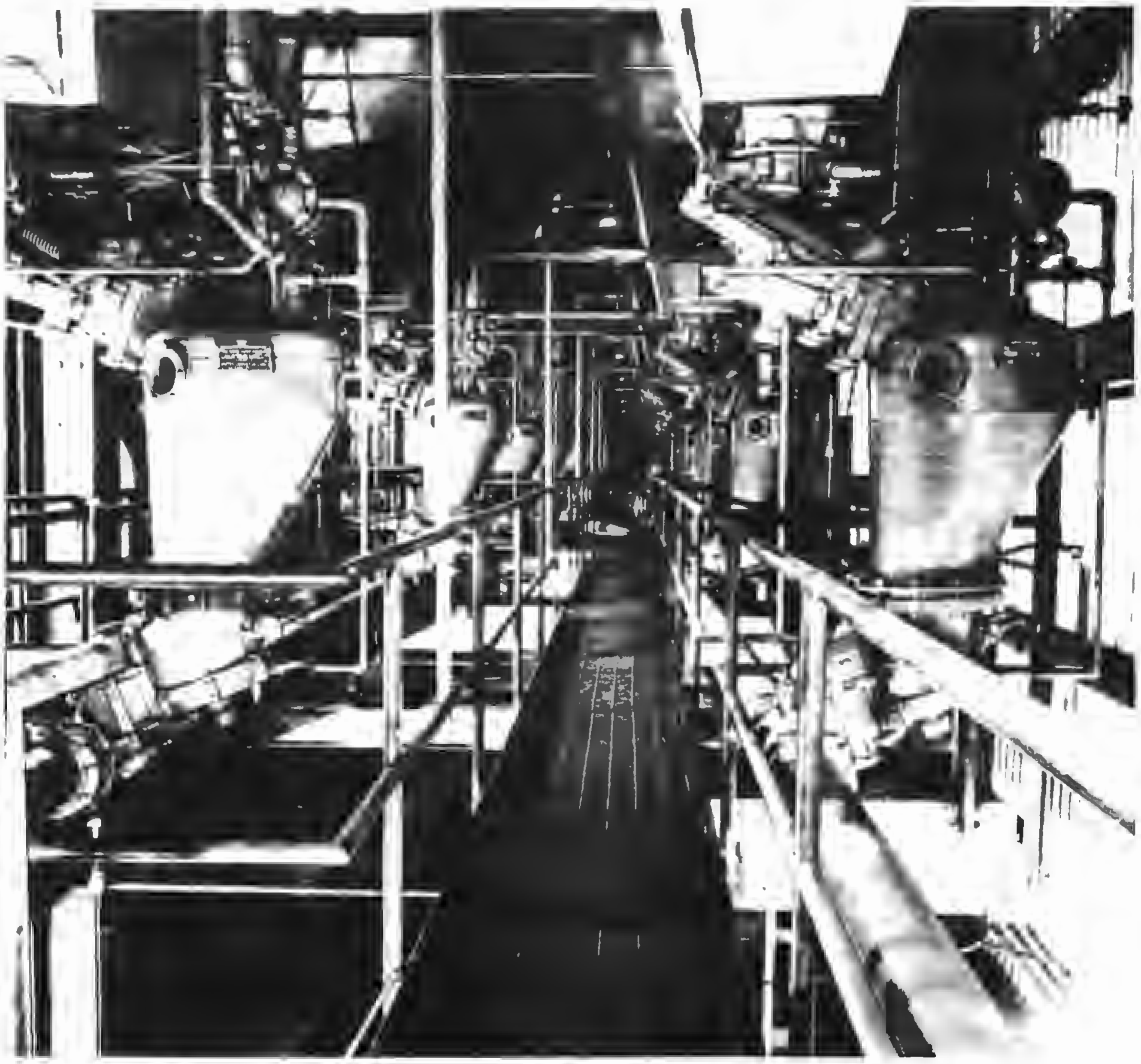


Figure 47.

NUVA FEEDER systems installed under electrostatic precipitators operate at a total capacity of 120 tons of dry fly ash per hour.

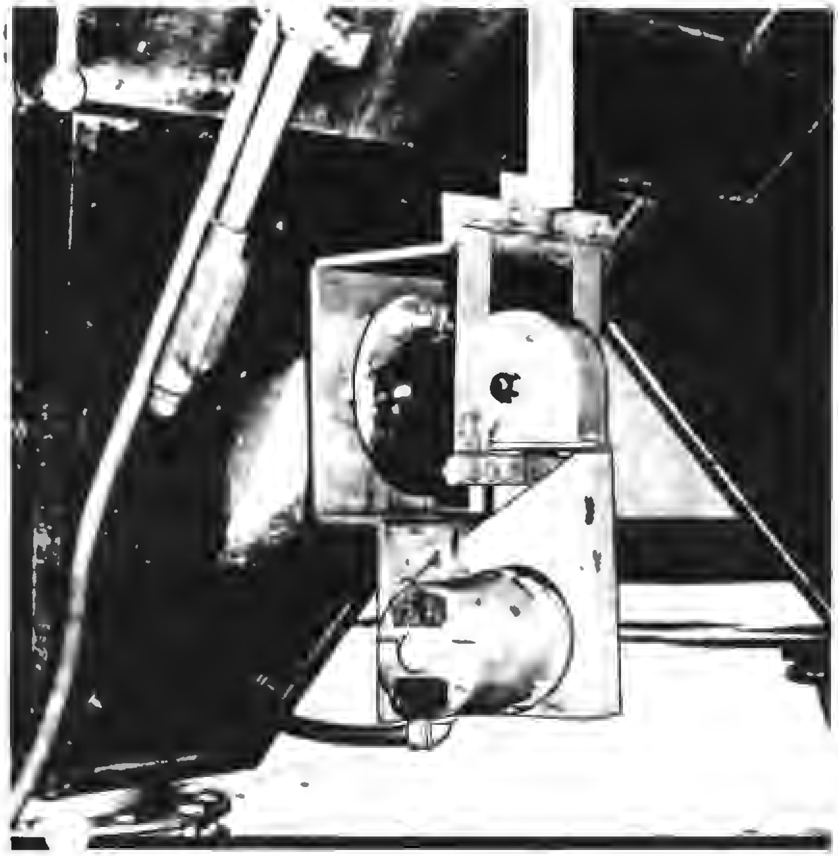


Figure 48.

UNITED DISCHARGE EQUIPMENT FOR DRY FLY ASH COLLECTOR HOPPERS

Frequently, dry fly ash is of such character that it arches and hangs up in the collector hoppers. To obtain maximum capacity for the pneumatic conveyor, UNITED engineers developed two methods to assist in overcoming this difficulty.

Upper photo, Figure 48, on this page shows the UNITED patented vibrating discharger. The vibrating mechanism located outside the hopper is connected to a vertically hung plate inside the dust collector hopper. The vibrating plate breaks down arching and results in a continuous gravity flow of dry fly ash to the conveyor system.

An alternate design is illustrated in the model photo, Figure 49. UNITED fluidizing units located inside the dust collector hoppers introduce low pressure compressed air in sufficient quantity to aerate the dry fly ash, thus causing dry material to flow freely and by gravity.

The operation of the above equipment is automatically controlled by the UNITED NUVATROL, generally a part of the conveying system.

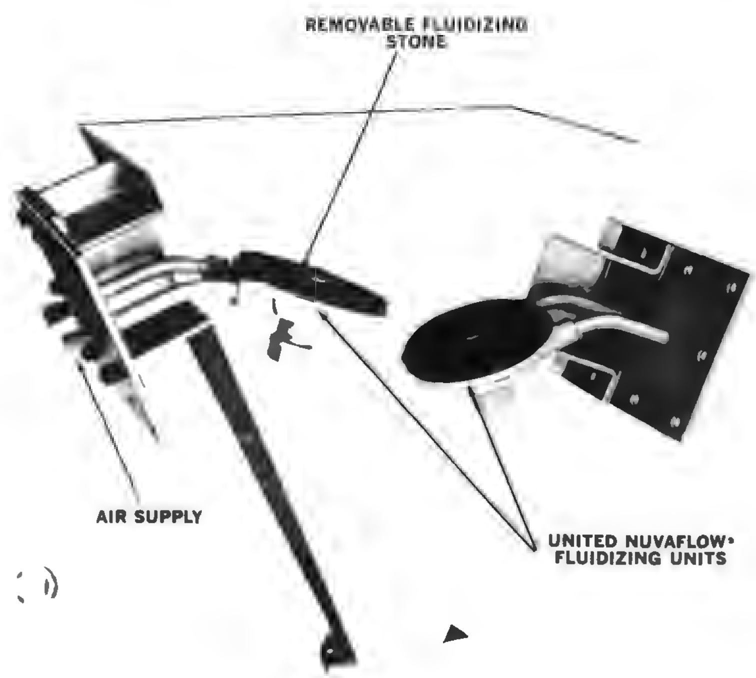


Figure 49.

UNITED

UNITED RESEARCH AND DEVELOPMENT FACILITIES

UNITED POLICY—to meet the needs of industry. A never-ending program of research and constant development of new equipment to give a high degree of efficiency and economy. Full scale equipment installed in our laboratory (partially pictured on this page) is used extensively to obtain engineering data for developing a wide range of conveyor systems, components and their application.

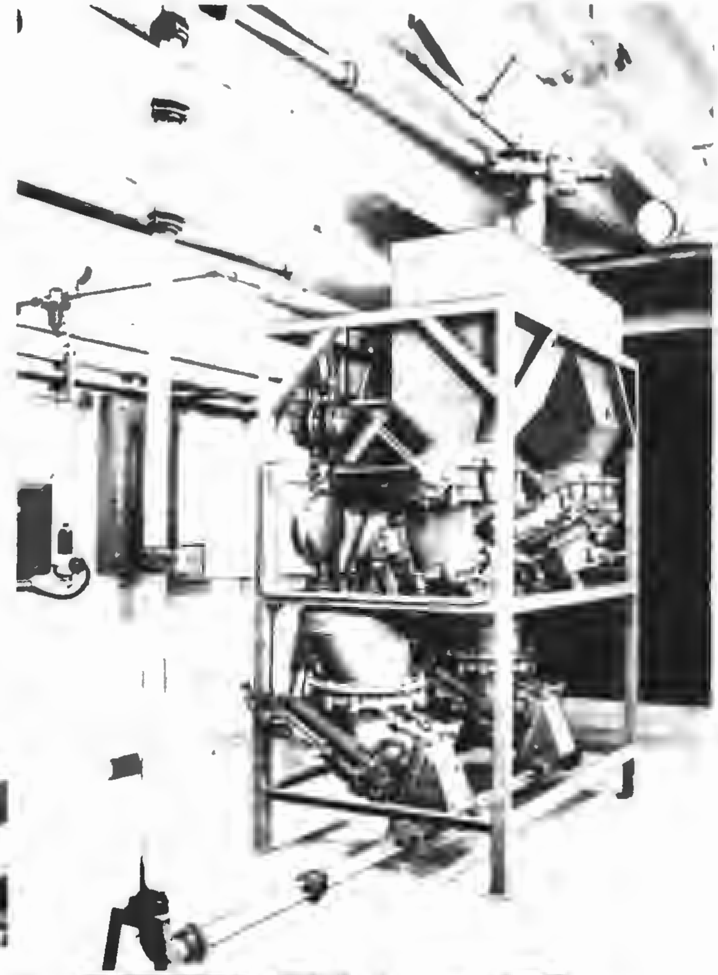
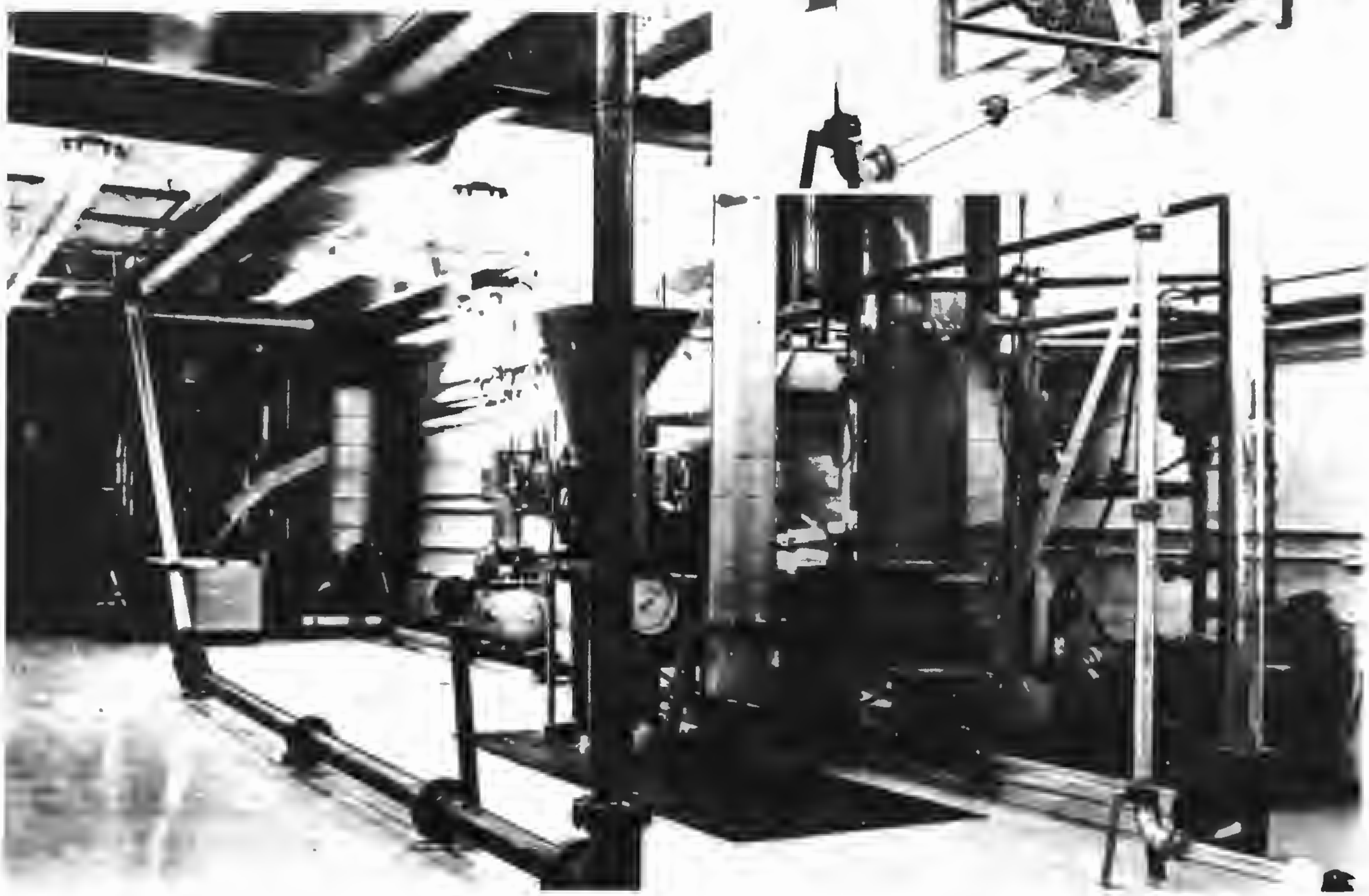


Figure 50.

Figure 51.



UNITED CONVEYOR CORPORATION

6505 NORTH RIDGE BOULEVARD, CHICAGO, ILLINOIS 60626 U.S.A.

PHONE (AREA 312) 761-4100

Nuva Feeder Bulletin No. 4NF-68A

Printed in U.S.A.

APPENDIX X

LIST OF ALL STUDY DRAWINGS

Drawing No.	Title
FBB-26	Prime Fuel Splitter
FBB-27	Control System Schematic
FBB-28	Fuel System Development
FBB-29	Wet Scrubber Arrangement
FBB-30	System Schematic (Electro-static precipitator)
FBB-31	System Schematic (Two Stage Wet Scrubber)
FBB-32	System Schematic (Feedwater/Steam)
FBB-33	Fluidized Bed Material Discharge Station
FBB-34	Heat and Material Balance - Dry Solids System
FBB-35	Heat and Material Balance - Wet Scrubber System
FBB-36	Wet Scrubber Slurry - Dewatering System

APPENDIX XI

STUDY OF HEAT TRAP - DESIGN & COST

In order to compliment the design work and boiler pricing data we obtained heat trap price data for our proposed 250,000#/Hr. design.

With the design of the boiler and convection pass, we were able to obtain sufficient boiler heating surface to reduce the heat trap heat load to about 25 million BTU/Hr. in order to achieve a stack gas temperature of 350° F.

In determining the relative costs of different heat traps, four (4) types were defined and considered. The selections are as follows:

BARE TUBE ECONOMIZER

A bare tube economizer would be a convection type unit where there are 2" tubes at 3-1/2" spaces perpendicular to flow and 4" spaces parallel to flow. Flue gas and water flow are arranged for counter flow. An inlet gas velocity of 100 ft./sec. was selected to peak the overall heat transfer coefficient. For an inlet gas temperature of 672° F. and an outlet of 350" F. and a 100° F. water temperature rise from 250° to 350° F., it was found that 7400 square feet of heating surface was required.

EXTENDED SURFACE ECONOMIZER

An extended surface economizer considers a convection type unit similar to the bare tube economizer, with multiple parallel circuits arranged for counter flow gas to water. The gas side heating surface, however, is increased by applying continuous spiral fins 0.135" thick, 3/4" high at a pitch to have 2 fins per inch lineal run of tubing. It is common to have pipe sizes used on commercially available finned tubing. In this case, 1-1/2" pipe was considered and the tubes are arranged in a 4-1/2" square pitch.

At the temperature conditions, as the first case above, 12,700 square feet of gas side surface was found necessary. This surface included fin surface.

TUBULAR AIR HEATER

The tubular air heater that was considered is a counter flow type, with gas inside the tubes, air outside. 2-1/2" tubes on a 3" triangular pitch was considered. An outlet air temperature of 515° F. was found using 120° F. inlet air temperature and the 24.9 million BTU/Hr. heat load. The higher air inlet temperature was a corrosion consideration, and would be a result of forced draft fan compression and a steam coil air heater.

Under the above conditions, 32,600 square feet of heating surface was found necessary.

REGENERATIVE AIR HEATER

The operating conditions were submitted to Air Preheater Company for estimation of a regenerative air heater, rotating heating element type, using counterflow and patented heating surface configuration. A vertical shaft type rotor was specified - a cleanability consideration. Predicted performance was in line with the above conditions.

Outlet gas temperature corrected for air leakage was somewhat lower. Approximately 10% leakage from the high pressure air side to lower pressure gas side was predicted.

For the four cases above, the comparitive estimated prices were developed - maximum shop assembly was assumed.

	ECONOMIZERS		AIR HEATERS	
	BARE TUBE	EXTENDED SURFACE	TUBULAR	REGENERATIVE
Estimated cost	60,000	25,000	37,000	32,000
Comparison of cost - %	240	100	150	130
Comparison of effect on total cost - %	115	100	105	103

APPENDIX XII

SUMMARY OF INDUSTRIAL STEAM GENERATOR

MARKET PROJECTIONS

Table #1 is an outline of technical classifications in capacity, steam pressure and steam temperature that was used to determine data in order to extrapolate and predict future markets.

Figure #1 is an outline showing the fraction of the market for different classifications of capacity projected for 1975 and for 1980.

Figure #2 is an outline showing the fraction of the market for different classifications of operating pressure.

Figure #3 shows the market fractions for different steam temperatures.

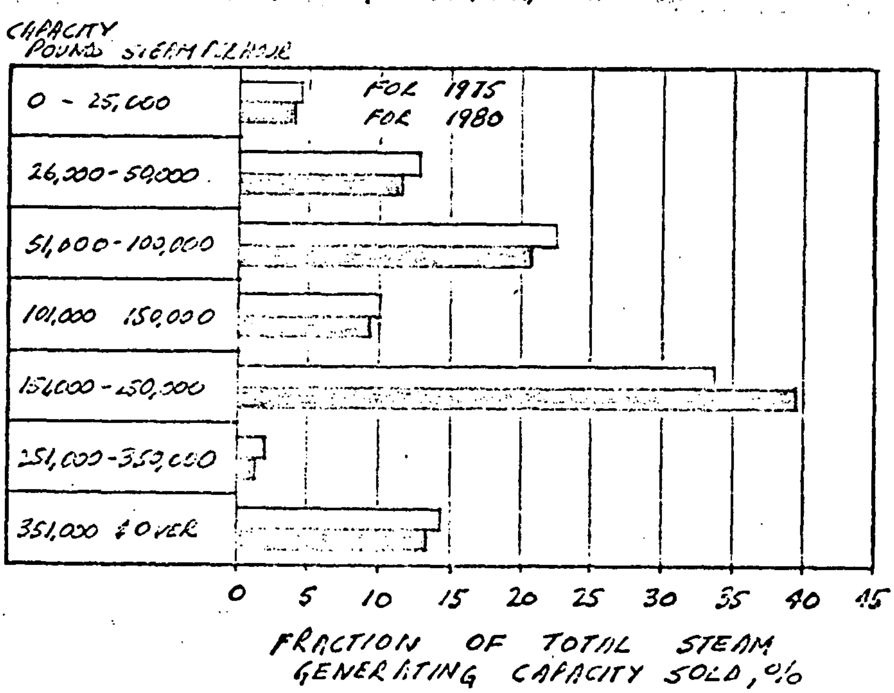
CLASSIFICATION OF DESIGN PARAMETERS
INDUSTRIAL BOILER DATA

TABLE 1

CAPACITY, POUNDS OF STEAM PER HOUR	OPERATING PRESSURE, PSIG	SUPERHEATED STEAM TEMPERATURE, F
0 - 25,000	0 - 149	SATURATED
26,000 - 50,000	150 - 249	300 - 524
51,000 - 100,000	250 - 599	525 - 624
101,000 - 150,000	600 - 849	625 - 724
151,000 - 250,000	850 - 1249	725 - 774
251,000 - 350,000	1250 - 1799	775 - 824
350,000 & OVER		825 - 874
		875 - 924
		925 & OVER

PROJECTED DISTRIBUTION WITHIN
FUTURE INDUSTRIAL BOILER MARKETS--
STEAM GENERATING CAPACITY

FIGURE 1



ECED
ZURN

PROJECTED DISTRIBUTION WITHIN
FUTURE INDUSTRIAL BOILER MARKETS

OPERATING PRESSURE

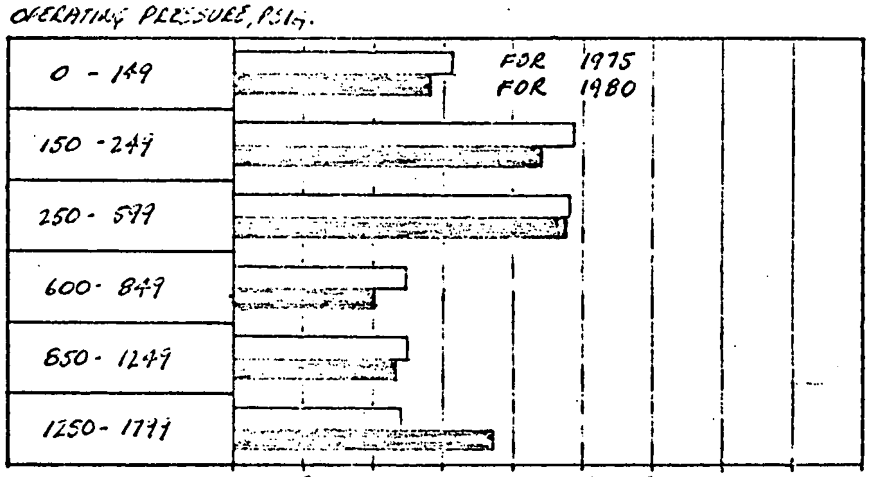


FIGURE 2

PROJECTED DISTRIBUTION WITHIN FUTURE
INDUSTRIAL BOILER MARKETS

SUPERHEATED STEAM TEMPERATURE

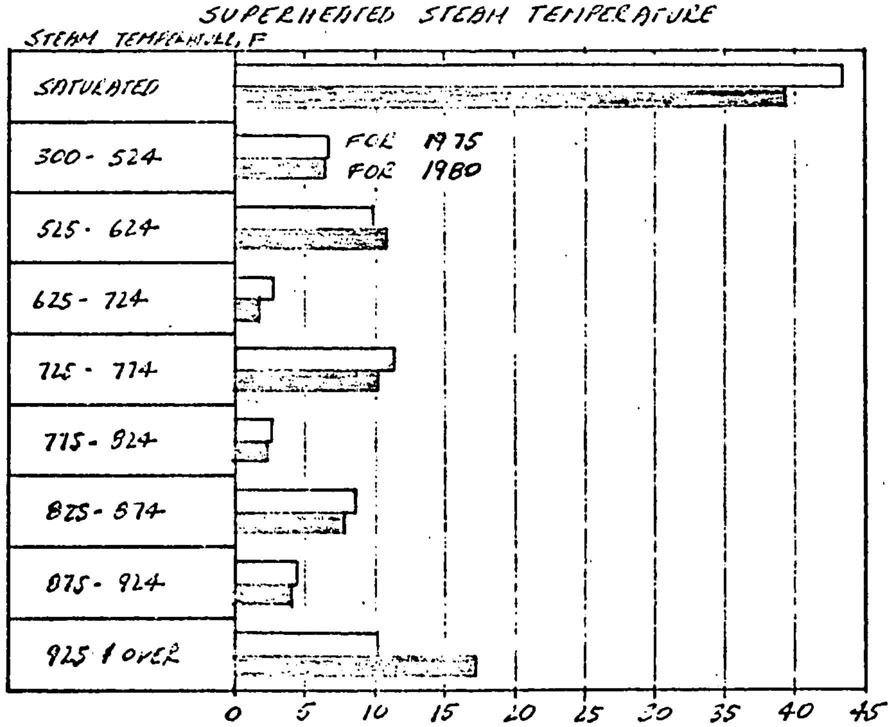


FIGURE 3

ECED
ZURN

APPENDIX XIII

SHIPPING CONFIGURATIONS

To develop optimum costs, it is necessary to shop assemble as large modules as possible to minimize field costs. An important advantage is also the field assembly time which is much less when shop assembly is contemplated.

For each large module shipped by rail individual rail clearances are developed for the most favorable routing to the destination. In the case of a very large Erie City Keystone Boiler for the Tennessee Valley Authority, page #1, represents the railroad response to a shipping clearance request. Figure #1 is a sectional dimension view of the clearance that must be adhered to.

*

PENN CENTRAL

1324 West Third Street
Cleveland, Ohio 44113
January 31, 1969

File: 9-EC-9820

Mr. C. W. Sherman
Traffic Manager
Erie City Iron Works
Erie, Pennsylvania

Dear Mr. Sherman:

Please refer to your January 7th request for rail clearance of a Shop Assembled Boiler to the Tennessee Valley Authority, Cumberland City, Tennessee.

We have cleared the handling of a dimensional shipment from Erie, Pennsylvania to the above destination routed: PC/N - Toledo - N&W - Blomington - GM&O - Humbold - L&N. The shipment, net weight of 125,000 pounds, loaded on a 70 Ton 53 Foot 6 Inch flat, must not have a combined center of gravity exceeding 98 Inches. The loaded dimensions must not exceed:-

Width	Height, ATR
8 Ft. 0 In.	19 Ft. 9 In.
9 Ft. 9 In.	18 Ft. 6 In.
12 Ft. 2 In.	17 Ft. 3 In.
13 Ft. 0 In.	15 Ft. 9 In.
13 Ft. 0 In.	4 Ft. 3 In.
10 Ft. 6 In.	3 Ft. 5 In.

Yours truly,
PENN CENTRAL COMPANY

B. L. Strohl

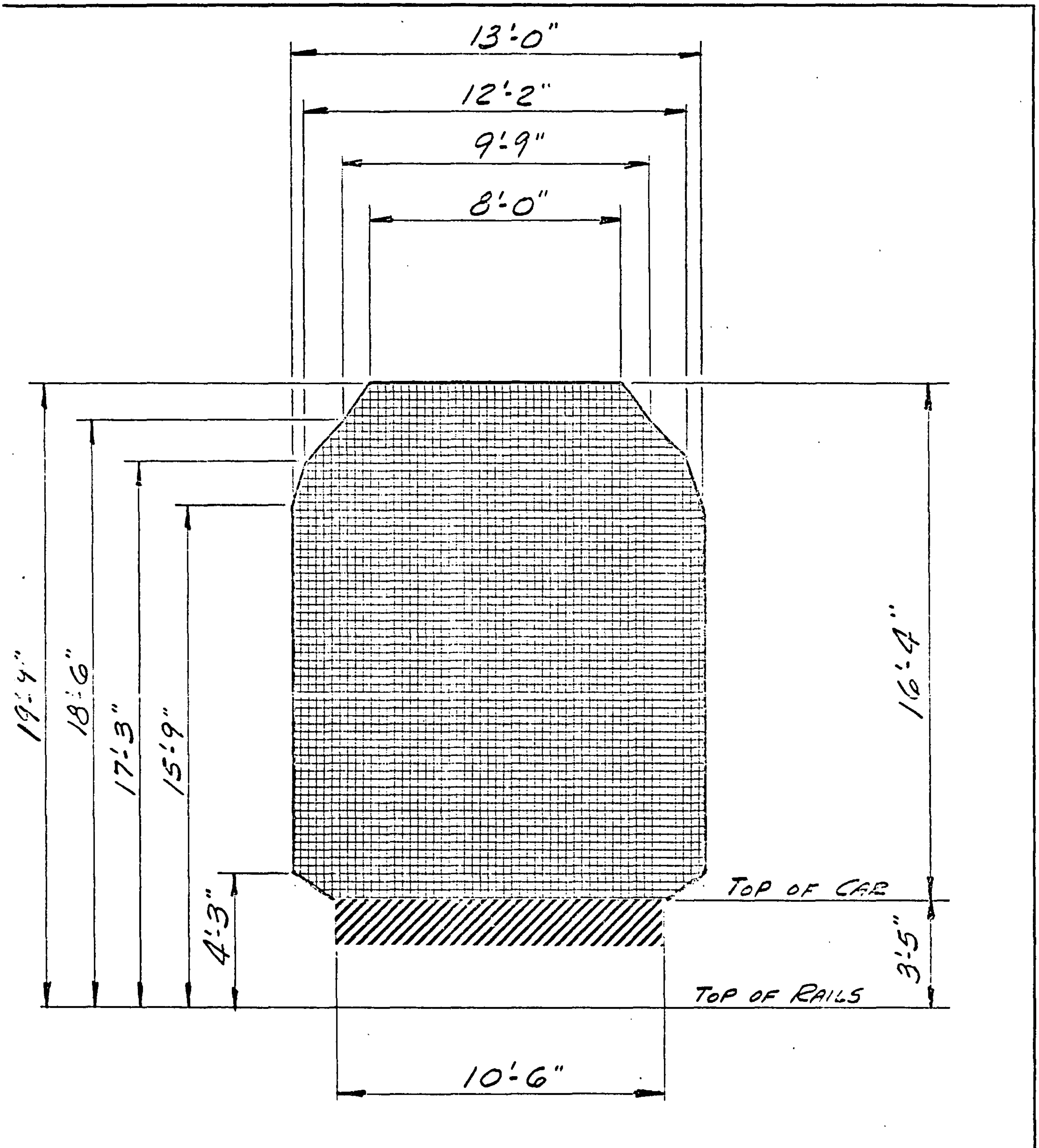
B. L. Strohl
General Superintendent-Transportation

rdg

cc: Freight Agent
Penn Central Company
Erie, Pennsylvania

Copy To C Crane
1/19/70

707/69 D-206



ZURN INDUSTRIES, INC.
ERIE CITY ENERGY DIV.
ERIE, PA. 16512 U.S.A.

SECTIONAL ELEVATION OF RAIL CLEARANCE

FLUIDIZED BED BOILER STUDY

D-207

APPENDIX E

TURNDOWN TECHNIQUES FOR ATMOSPHERIC FLUIDIZED BED BOILERS

ABSTRACT

A comprehensive analysis of applicable turn-down techniques for an atmospheric fluidized bed boiler is described. The techniques for altering the bed temperature, the heat transfer coefficient, and the heat transfer surface in the bed are discussed. The effectiveness of different techniques are evaluated. The bed operating conditions, such as bed temperature, excess air, coal feed rate, before and after turn-down are presented and compared.

The most economical and convenient turn-down technique is that of changing air and coal feed simultaneously and maintaining the excess air constant. To increase the turn-down ratio, the fluid bed can be shut down by sections.

A comprehensive analysis of applicable turn-down techniques for an atmospheric fluidized bed boiler is described. Although the analysis of the industrial fluidized bed boiler is used as illustration here, the analysis is general and applicable to both utility and industrial fluid bed boilers.

Load change in a fluidized bed boiler constitutes two additive parts: the contribution from the heat transfer in the bed and that in the convection section, including the heat traps. The change of heat transfer in the convection section is caused primarily by change in the heat transfer coefficient due to change in gas flow rate and gas temperature and can be easily evaluated once the flow rate and temperature are known. The change of heat transfer in the bed, however, involves three main parameters: the heat transfer surface in the bed, the heat transfer coefficient, and the bed temperature, assuming the water-side conditions are the same. Techniques which are capable of changing one variable or a combination of them will be effective in turn-down. Since the magnitude of the contribution from the convection section depends on different boiler designs, this general turn-down discussion will be confined to change of heat transfer in fluidized beds alone. The effect of the convection section is additive and can be determined once the design of the boiler is finalized.

The applicability of different turn-down techniques is evaluated by turning the bed from its original operating conditions at 1650°F bed temperature and 10% excess air down to 1400°F bed temperature and the appropriate percentage of excess air.

TECHNIQUES FOR ALTERING THE BED TEMPERATURE

The techniques available for altering the bed temperature and achieving the turn-down effect are:

(a) decreasing the air preheat temperature,

(b) decreasing both air and coal feed rate and maintaining the same percentage of excess air,

(c) increasing percent excess air by decreasing the coal feed and maintaining air flow constant,

(d) increasing percent excess air by increasing air feed and keeping coal feed constant,

(e) combination of (a) and (b); (a) and (c); (a) and (d),

(f) recirculation of flue gas.

The percentage of total heat input transferred in the bed for every pound of coal burned is plotted against the bed temperature in Figure E-1 for different degrees of air preheat and at different percentages of excess air based on the adiabatic flame temperature of the flue gas, assuming the heat capacity of the flue gas to be constant in the temperature range considered. We will call these lines the equilibrium lines at their respective operating conditions. Hence if the bed temperature and air and coal input conditions are known, the total heat transferred in the bed can be determined from Figure E-1. The percentage turn-down achievable for each technique can thus be quantitatively determined. The procedure is illustrated by considering the operating conditions before turn-down to be 10% excess air and the bed temperature 1650°F. Normal operating conditions other than these can be analyzed in the same way.

Curve 641624-B

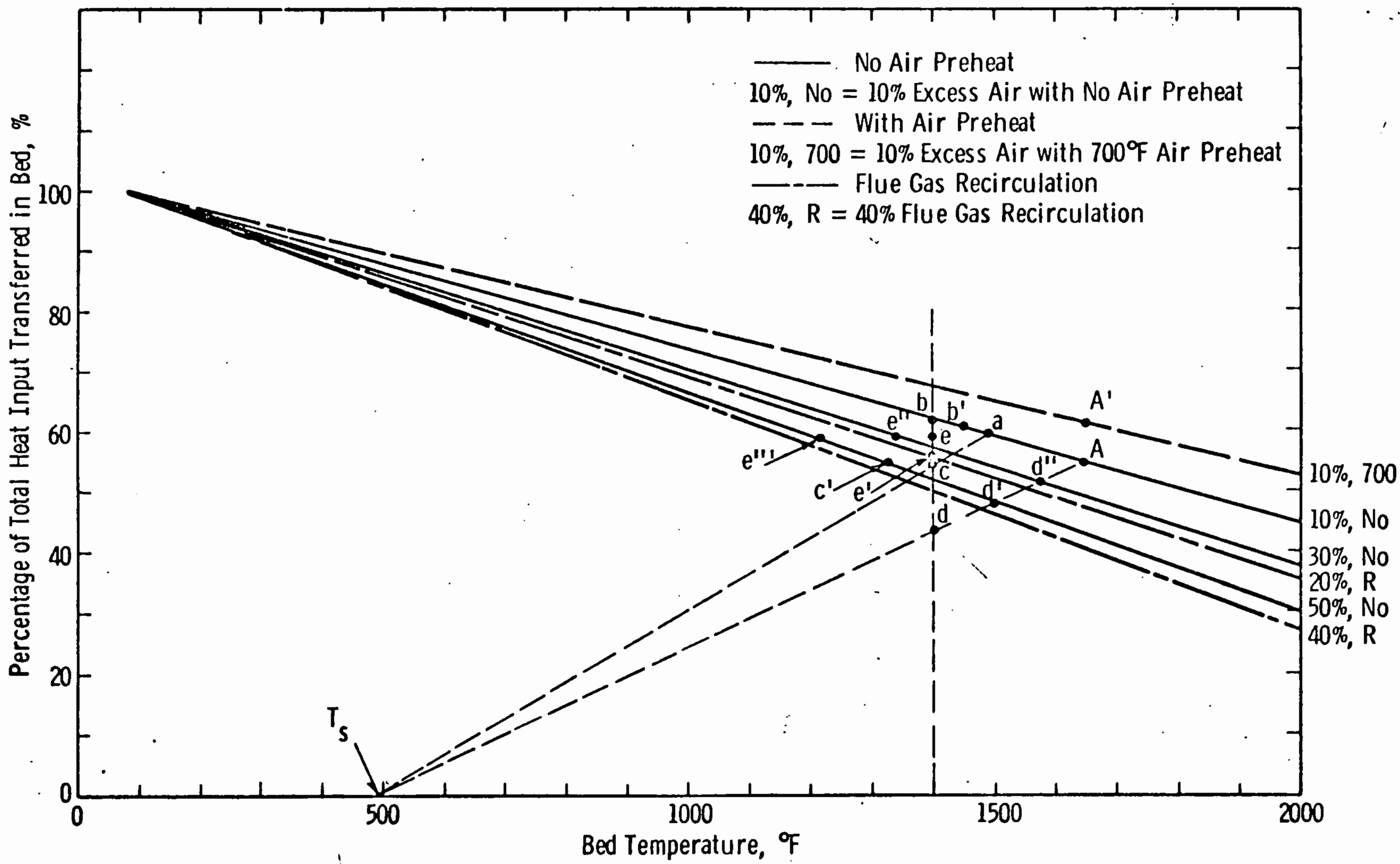


Fig. E -1—Effectiveness in turn-down by different turn-down techniques

(a) Turn-down by Decreasing the Air Preheat Temperature

With air preheated and with the fluidized bed serving as an evaporator, the heat transferred in the bed before turndown is

$$h_b S_b (T_b - T_s) = [H_c \frac{F}{f_s (1 + X_s)} + \overline{C_p} F(T_A - T_B)] y_b \tag{1}$$

where

h_b = heat transfer coefficient in the bed before turn-down, Btu/ft^2-hr-°F

S_b = total heat transfer area in the bed before turn-down, ft^2

T_b = bed temperature before turn-down°F

T_s = temperature of the saturated steam, °F

H_c = heat of combustion of fuel, Btu/lb

F = total air flow rate, lb/hr

X_s = excess air before turn-down, %

f_s = stoichiometric air requirement for each pound of fuel burned

$\overline{C_p}$ = average heat capacity of air, Btu/lb-°F

T_A = air preheat temperature, °F

T_B = base temperature, °F

y_b = percentage of total heat input transferred in the bed before turn-down, %

The heat transferred in the bed after decreasing the air-preheat temperature from T_A to T_B is

$$h_a S_a (T - T_s) = H_c \frac{F}{f_s (1 + X_s)} y \tag{2}$$

where

h_a = heat transfer coefficient in the bed after turn-down, Btu/ft^2-hr-°F

S_a = total heat transfer area in the bed after turn-down, ft^2

T = bed temperature after turn-down, °F

y = percentage of total heat input transferred in the bed after turn-down, %.

In the case of the atmospheric industrial boiler, the following conditions are assumed:

T_b = 1650°F ; T_s = 489°F ; $\underline{X_s}$ = 10%

T_A = 700°F ; T_B = 80°F ; C_p = 0.24 Btu/lb-°F

H_c = 12500 Btu/lb ; f_s = 10.04 lb air/lb coal burned.

The total heat input transferred in the bed before turndown, y_b, can be read off from Figure E-1 to be 61%.

Dividing Eq. (2) by Eq. (1), we have

$$y = 0.69 \cdot \frac{h_a S_a}{h_b S_b} \cdot \frac{(T - T_s)}{(1650 - T_s)} \; . \tag{3}$$

For the time being, assume $h_a = h_b$ and bed depth is constant; i.e., $S_a = S_b$. Equation (3) is a straight line passing ($y = 0$, $T - T_s$) ($y = 0.69$, $T = 1650$), and we shall call this line an operating line. The total reduction of heat transfer in the bed by decreasing the air preheat temperature from 700°F (point A' in Figure E-1) to no air preheat (point a obtained from Eq. (3)) is 14% calculated from Eqs. (1) and (2). After turn-down the new bed temperature is 1490°F, from Figure E-1. If the bed temperature is to be decreased to 1400°F after turn-down, y can be obtained from Eq. (7) (see Section (e)) to be 0.59 (point e in Figure E-1). This gives an air input condition of 20% excess air without preheat from Figure E-1. In other words, 1490°F is the lowest bed temperature obtainable under the specified condition. Any attempt to lower the bed temperature further requires other techniques such as increasing the excess air with decreasing coal feed.

(b) Turn-down by Decreasing Both the Air and Coal Feed Rate and Maintaining the Same Percentage of Excess Air

Operating condition before turn-down:

bed temperature: 1650°F

air preheat: none

excess air: 10%

percentage heat transferred in the bed (from Figure E-1): 55%

Operating line:

$$y = 0.55 \cdot \frac{h_a S_a}{h_b S_b} \cdot \frac{F_b}{F_a} \cdot \frac{(T - T_s)}{(1650 - T_s)} \tag{4}$$

where

F_a, F_b = total air flow rate after and before turn-down respectively.

Example: (assuming again $h_a = h_b$; $S_a = S_b$)

If air flow rate and the corresponding coal flow rate is to be decreased by 25% after turn-down, then Eq. (4) becomes

$$y = 0.55 \times \frac{1}{0.75} \times \frac{(T - T_s)}{(1650 - T_s)} ,$$

a straight line passing ($y = 0$, $T = T_s$) and ($y = 0.733$, $T = 1650°F$). The intersection with the equilibrium line of 10% excess air with no air preheat is point b' in Figure E-1. The turn-down achieved is

$$\frac{0.55 - 0.75 \times 0.61}{0.55} = 16.82\%$$

and the bed temperature after turn-down is 1450°F.

If the bed temperature is to be reduced to 1400°F after turn-down, Eq. (4) becomes

$$y = 0.55 \cdot \frac{F_b}{F_a} \cdot \frac{(1400 - 489)}{(1650 - 489)} \cdot \qquad (5)$$

From Figure E-1 $\quad y = 0.622$

Combining the two equations we have

$$F_b/F_a = 1.44 \qquad \text{or} \qquad F_a/F_b = 0.695 .$$

Since the percentage excess air is constant before and after turn-down the total coal and air feed needed to bring the bed temperature from 1650°F down to 1400°F, a reduction of heat transfer in the bed of 21.5%, is 69.5% of their original feed.

The maximum percentage turn-down achievable by this technique is determined by the lowest bed temperature allowable based on considerations of coal combustion efficiency and SO_2 removal efficiency by limestone or dolomite.

(c) Turn-down by Increasing Percent Excess Air by Decreasing the Coal Feed and Maintaining Air Flow Constant

Operating condition before turn-down:

bed temperature: 1650°F

air preheat: none

excess air: 10%

percent heat transfer in the bed (from Figure E-1): 55%

Operating line:

$$y = 0.55 \cdot \frac{(1 + x)}{1.10} \cdot \frac{h_a S_a}{h_b S_b} \cdot \frac{(T - T_s)}{(1650 - T_s)} \qquad (6)$$

where

x = percent excess air after turn-down.

Examples: (assuming $h_a = h_b$; $S_a = S_b$)

If $x = 0.5$, i.e., after turn-down the bed is operated at 50% excess air, then the operating line becomes

$$y = 0.75 \frac{(T - T_s)}{(1650 - T_s)}$$

a straight line passing ($y = 0$, $T = T_s$) and ($y = 0.75$, $T = 1650°F$). The operating line intersects with the equilibrium line for 50% excess air and no air preheat at c'. The turn-down obtained is about

$$1 - \frac{1.10}{1.50} = 0.267 = 26.7\%.$$

The bed temperature after turn-down is 1330°F, from Figure E-1.

If the bed temperature after turn-down is 1400°F, Eq. (6) becomes

$$y = 0.55 \cdot \frac{(1 + x)}{1.1} \cdot \frac{(1400 - 489)}{(1650 - 489)} . \tag{7}$$

For every x there is a corresponding y. The correct x and y can be found by trial and error between Eq. (7) and Figure E-1. Here the excess air required after turn-down is found to be 40%, and the coal feed is 78.5% of the original rate.

The maximum percentage turn-down achievable by this scheme depends also on the lowest allowable bed temperature after turn-down.

(d) Turn-down by Increasing Percent Excess Air by Increasing of Air Feed and Keeping the Coal Feed Constant

Operating condition before turn-down:

Same as outlined in (c).

Operating line:

$$y = 0.55 \cdot \frac{h_a S_a}{h_b S_b} \cdot \frac{(T - T_s)}{(1650 - T_s)} \quad . \tag{8}$$

Examples : the points of intersection after turn-down are d" and d' for their respective operating conditions. The percentage turn-down and the bed temperature after turn-down are 6.4%, 1580°F; 11.8%, 1510°F respectively.

If the temperature after turn-down is 1400°F, Eq. (8) becomes

$$y = 0.55 \cdot \frac{(1400 - 489)}{(1650 - 489)} = 0.432 .$$

From Figure E-1, the excess air required after turn-down is found to be ∿ 75% by extrapolation. The maximum percentage turn-down obtainable through this scheme depends on the maximum fluidizing velocity allowable after turn-down based on the solid elutriation consideration.

(e) Turn-down Through Combination of (a) & (b); (a) & (c), (a) & (d)

Considering here only the case of a combination of (a) & (c), the operating lines for cases of (a) & (b); (a) & (d) can be obtained through the same analysis.

Operating conditions before turn-down:

bed temperature: 1650°F

air preheat: 700°F

excess air: 10%

percentage heat transferred in the bed (from Figure E-1): 61%

Operating line:

$$y = 0.61 \cdot \left[\frac{(1 + x)}{(1.10)} + 0.12(1 + x)\right] \cdot \frac{h_a S_a}{h_b S_b} \cdot \frac{(T - T_s)}{(1650 - T_s)} \quad . \tag{9}$$

Examples: the point of intersection after turn-down are e'' and e''' for their respective operating conditions. The percentage turn-down and the bed temperature after turn-down are 26.4%, 1335°F; 37.4%, 1215°F respectively.

The final operating conditions for turn-down to 1400°F bed temperature was found to be 20% excess air for the (a) & (c) combination scheme and 35% excess air for the (a) & (d) combination scheme.

(f) Turn-down Through Recirculation of Flue Gas

The equilibrium lines for 20% and 40% of total flue gas circulation were also included in Figure E-1, assuming the flue gas temperature to be 350°F. Flue gas recirculation has the combined effects of air preheat (due to the sensible heat carried by the flue gas) and increasing excess air and keeping the coal feed constant (due to the increased flow caused by flue gas recirculation). The operating line is similar to the one for turn-down through scheme (d). Combination of this scheme with techniques (b), (c), and (d) can also be obtained by the same analysis.

The turn-down techniques available, their corresponding percentage turn-down, and their final operating conditions by decreasing the bed temperature from 1650°F to 1400°F are summarized in Table E-1 assuming $h_a = h_b$ and $S_a = S_b$. Note that mode (a) turn-down alone cannot achieve bed temperature lower than 1490°F for the chosen conditions. Mode (b) offers the most efficient means of turn-down because the lowest coal feed is needed and the lowest flue gas heat loss is experienced. It should be therefore employed as the primary scheme for turn-down whenever possible.

If the fluidized bed serves as a superheater or reheater instead of an evaporator, the operating line for each turn-down technique can be obtained through the same analysis. These are presented in Table E-2.

TABLE E-1

EFFECTIVENESS OF TURN-DOWN BY DIFFERENT TECHNIQUES*

(Fluidized Bed Serves As An Evaporator)

Conditions Before & After Turn-down (Refer to Fig. E-1)	Mode of Turn-down	Bed Temperature After Turn-down (°F)	Degree of Turn-down in Bed (%)	Coal Feed (Arbitrary Unit) Before Turn-down	Coal Feed (Arbitrary Unit) After Turn-down	Excess Air (%) Before Turn-down	Excess Air (%) After Turn-down
A' → a	(a)	1490	14.0	1.0	1.0	10	10
A → b	(b)	1400**	21.5	1.0	0.695	10	10
A → c	(c)	1400	21.5	1.0	0.785	10	40
A → d	(d)	1400	21.5	1.0	1.0	10	75
A' → e	(a) & (c) combination	1400	21.5	1.0	0.915	10	20
A' → e'	(a) & (d) combination	1400	21.5	1.0	1.0	10	35
A → d	(d) & (f) combination	1400	21.5	1.0	1.0	10	60% Flue Gas Recirculation

* Operating conditions before turn-down are 10% excess air, 1650°F bed temperature, without air preheat or with 700°F air preheat wherever is applicable.

**1400°F is by no means the lowest bed temperature allowable. It is used here just as an example for purpose of illustration.

TABLE E-2

EQUATIONS FOR OPERATING LINE OF DIFFERENT TURN-DOWN TECHNIQUES WHEN FLUIDIZED BED SERVES AS A SUPERHEATER OR REHEATER**

Mode of Turn-Down	Operating Line	R_b/R_a
(a)	$y = 0.69 \cdot G$*	$0.69/y$
(b)	$y = 0.55 \cdot \frac{F_b}{F_a} \cdot G$	$0.55\ F_b/y \cdot F_a$
(c)	$y = 0.55 \cdot \frac{(1 + x)}{1.10} \cdot G$	$0.55\ (1 + x)/1.10\ y$
(d)	$y = 0.55 \cdot G$	$0.55/y$
(a)&(b) combination	$y = 0.69 \frac{F_b}{F_a} \cdot G$	$0.69\ F_b/y \cdot F_a$
(a)&(c) combination	$y = 0.61\ [\frac{(1 + x)}{1.10} + 0.12(1 + x)]G$	$0.61[\frac{(1 + x)}{1.10} + 0.12(1 + x)]/y$
(a)&(d) combination	$y = 0.69 \cdot G$	$0.69/y$

$$*G = \frac{h_a S_a}{h_b S_b} \cdot \frac{\ln \frac{(1650-T_1)}{(1650-T_2)}}{\ln \frac{(T-T_1)}{(T-T_2)}}$$

**Operating conditions before turn-down are 10% excess air, 1650°F bed temperature, without air preheat or with 700°F air preheat wherever applicable.

Where

T_1 = inlet temperature of steam into the superheater or reheater

T_2 = outlet temperature of steam out from the superheater or reheater

R_b, R_a = steam flow rate in lb/hr before and after turn-down respectively.

When the fluidized bed serves multiple function, i.e., for both evaporator and superheater or reheater, the operating lines for techniques (b), (c), and (d) were evaluated and shown to be:

Technique (b)

$$y = 0.55 \cdot \frac{h_a}{h_b} \cdot \frac{F_b}{F_a} \cdot \left[\frac{x_e(T_b - T_s) + x_s \dfrac{(T_2-T_1)}{\ln \dfrac{(T_b-T_1)}{(T_b-T_2)}}}{x_e(1650-T_s) + x_s \dfrac{(T_2-T_1)}{\ln \dfrac{(1650-T_1)}{(1650-T_2)}}} \right] \tag{10}$$

Technique (c)

$$y = 0.55 \cdot \frac{(1+x)}{1.10} \cdot \frac{h_a}{h_b} \cdot \left[\frac{x_e(T_b - T_s) + x_s \dfrac{(T_2-T_1)}{\ln \dfrac{(T_b-T_1)}{(T_b-T_2)}}}{x_e(1650-T_s) + x_s \dfrac{(T_2-T_1)}{\ln \dfrac{(1650-T_1)}{(1650-T_2)}}} \right] \tag{11}$$

Technique (d)

$$y = 0.55 \cdot \frac{h_a}{h_b} \cdot \left[\frac{x_e(T_b-T_s) + x_s \dfrac{(T_2-T_1)}{\ln \dfrac{(T_b-T_1)}{(T_b-T_2)}}}{x_e(1650-T_s) + x_s \dfrac{(T_2-T_1)}{\ln \dfrac{(1650-T_1)}{(1650-T_2)}}} \right] \tag{12}$$

assuming the total heat transfer surface is constant, i.e., the bed depth is constant, where

x_e, x_s = fraction of total heat transfer surface in the bed for evaporation and superheating functions respectively.

The operating lines for other turn-down techniques can be readily obtained through the same analysis.

TECHNIQUES FOR ALTERING THE HEAT TRANSFER COEFFICIENTS AND/OR HEAT TRANSFER SURFACE

The analysis presented above and the operating lines obtained (Eqs. (3) through (12)) are general since both the change of heat transfer coefficient and heat transfer surface before and after turn-down are considered, although the illustrations are based on $h_a = h_b$ and $S_a = S_b$. In fact, it is almost impossible just to change one of the three parameters without altering the other two, since the heat transfer coefficient depends on both the bed temperature and the fluidizing velocity. A change of gas and/or fuel flow rate will automatically change the heat transfer coefficient in addition to the bed temperature. So most of the techniques discussed can still be applied here with the same operating lines derived, but with $h_a \neq h_b$ and $S_a \neq S_b$.

(g) Turn-down by Slumping Part of the Bed or Completely Shutting One Section of the Bed

This is actually technique (b) carried out to the extreme. When air and fuel feed rate decrease more and more, the bed becomes de-fluidized and slumped, and both the heat transfer coefficient and total heat transfer surface in the bed decrease until the complete section is finally shut off. To find out the percentage turn-down achievable by this scheme, the operating line for technique (b) (Eq. (4) is applicable.

(h) Turn-down Changing the Bed Depth Through Adjustable Bed Weir Height

This will change the bed depth and thus change the total heat transfer surface in the bed. However, due to the high heat transfer coefficient for the surface just above the fluidized bed, this turn-down scheme applied alone may not give the effective turn-down desired. Nevertheless, this scheme can be applied in conjunction with techniques (a) through (f) described above. The operating lines derived (Eq. (3) through (12)) are still applicable in calculating the percentage turn-down.

APPENDIX F

DYNAMICS OF ATMOSPHERIC FLUIDIZED BED BOILER

ABSTRACT

The dynamic behavior of atmospheric fluidized bed boilers was analyzed mathematically to give some insight into the design and operational problems which might be encountered.

Response time for ignition from a neighboring bed was quantitatively estimated using the atmospheric industrial fluidized bed boiler design as an illustration. Burners for start-up and restart during load swing were also sized, based on a load response rate of 10%/min. To achieve this response rate, the total thermal capacity of the burners is close to 50% of total boiler capacity. Methods for start-up and restart were also reviewed and discussed.

Rate of bed temperature change and residual steam generation in a fluidized bed after shut-down were also mathematically analyzed. Three cases were studied: turning off the bed by 1) stopping the coal and air feed simultaneously; 2) stopping only the coal feed and keeping the bed fluidized; 3) stopping the coal feed and maintaining the fluidization with solid recirculation from the neighboring operating bed.

DYNAMICS OF ATMOSPHERIC FLUIDIZED BED BOILER

Knowledge of the dynamic change of bed temperature during start-up and shut-down of fluidized beds is important for sizing the burners and devising the light-off system and control scheme. With a requirement of turning the beds on and off to provide load swing, the fast response of the bed temperature change becomes more critical. In the absence of actual pilot plant data, the dynamic behavior of fluidized beds was analyzed mathematically to give some insight into the problems which might be encountered. The analysis for the atmospheric industrial fluidized bed boiler will be used as an illustration.

RESPONSE TIME FOR IGNITION FROM NEIGHBORING BED

There are several methods to ignite the idle beds either for cold start-up, hot start-up, or load swing. The two most common methods are: 1) providing an individual burner for each bed; 2) providing a burner for only one bed and igniting the other beds by solid circulation from the operating bed. To use the first method, large burners have to be provided for each bed to accommodate the fast load swing because the burners are estimated to have only approximately 30%-50% efficiency (see discussion in next section). Hence the latter method is always preferred if the bed ignition can be accomplished fast enough for the desired load change.

To analyze the latter method, now consider two beds: Bed 1 is operating at full load with a bed temperature of 1650°F, and Bed 2 is in idle with a bed temperature of either 80°F (for cold start-up) or 489°F (for hot start-up). At time t = 0, the idle bed will be fluidized at a minimum fluidizing velocity and be heated by solid circulation from the neighboring operating bed. At the instance of fluidization, the bed temperature of Bed 2 will rise due to the hot solid circulated from Bed 1, and the bed temperature of Bed 1 will drop due to heat loss to Bed 2, unless the firing rate is increased at the same time. Mathematically, the bed temperature change in both beds can be expressed as

Bed 1:

$$H - h_1A(T_1 - T_s) - F_1Cp_a(T_1 - T_B) - WCp_s(T_1 - T_2) = W_BCp_s\frac{dT_1}{dt} \quad (1)$$

Bed 2:

$$WCp_s(T_1 - T_2) - h_2A(T_2 - T_S) - F_2Cp_a(T_2 - T_B) = W_BCp_s\frac{dT_2}{dt} \quad (2)$$

where

A = total heat transfer surface in the bed, ft^2
Cp_a = heat capacity of air, Btu/lb-mole-°F
Cp_s = heat capacity of bed material, Btu/lb-°F
F_1,F_2 = total air flow rate in Beds 1 and 2, lb-mole/hr
H = total heat supplied to Bed 1 by burning coal, Btu/hr
h_1,h_2 = heat transfer coefficients in Beds 1 and 2, Btu/hr-ft^2-°F
T_1,T_2 = bed temperature of Beds 1 and 2, °F
T_s = saturated water temperature, °F
T_B = base temperature, °F
W_B = total weight of bed materials, lb
W = rate of solid circulation, lb/hr

Equations (1) and (2) are coupled. To solve the equations, knowledge of the relationship between T_1 and T_2 is necessary. Fortunately, in the actual operation it is advantageous to keep the operating bed temperature, T_1, constant to assure stability of operation, to facilitate control, and to shorten the light-off time of Bed 2 by maintaining the possible maximum bed temperature difference between the two beds. With the operating bed temperature, T_1, kept constant, Eq. (1) becomes

$$H = h_1A(T_1 - T_s) + F_1Cp_a(T_1 - T_B) + WCp_s(T_1 - T_2) \tag{3}$$

and Eq. (2) can be integrated to give

$$t = \frac{W_BCp_s}{k_2} \cdot \ln \frac{(k_1 - k_2T_o)}{(k_1 - k_2T_2)} \tag{4}$$

$$T_2 = \frac{k_1}{k_2} + (T_o - \frac{k_1}{k_2}) \exp (- \frac{k_2t}{W_BCp_s}) \tag{5}$$

where

$$k_1 = WCp_sT_1 + h_2AT_s + F_2Cp_aT_B \tag{6}$$

$$k_2 = WCp_s + h_2A + F_2Cp_a \tag{7}$$

T_o = the bed temperature of Bed 2, T_2, at the instance of fluidization ($t = 0$).

When Bed 1 is operating alone, the total heat input through coal feeding to maintain a constant bed temperature, T_1, is

$$H_o = h_1A(T_1 - T_s) + F_oCp_a(T_1 - T_B). \tag{8}$$

At the instance of starting solid recirculation, the firing rate of Bed 1 has to be increased as shown in Eq. (3) to maintain the constant bed temperature. The additional heat required is expressed as

$$\Delta H = H - H_o = (F_1 - F_o)\, Cp_a(T_1 - T_B) + WCp_s(T_1 - T_2). \qquad (9)$$

Substituting Eq. (5) into Eq. (9) we have

$$\Delta H = (F_1 - F_o)Cp_a(T_1 - T_B) + WCp_s\{T_1 - \frac{k_1}{k_2} + [\frac{k_1}{k_2} - T_o]\exp(-\frac{k_2 t}{W_B Cp_s})\}. \qquad (10)$$

Physically, $\frac{k_1}{k_2}$ is always larger than T_o. When $\frac{k_1}{k_2} = T_o$, Eq. (10) becomes time independent and gives a solution at equilibrium condition where

$$WCp_s(T_1 - T_o) = h_2A(T_o - T_s) + F_2Cp_a(T_o - T_B). \qquad (11)$$

Theoretically, Bed 2 can be ignited in as short a time as desired by increasing simultaneously the firing rate in Bed 1 and the solid circulation rate. In actual practice, an increase of firing rate through increase of coal input requires a proportional increase of air. With the bed area constant, an increase of firing rate necessitates an increase of fluidizing velocity which in turn determines the carbon carry-over and CO loss. With the maximum firing rate restricted, the solid circulation rate has to be controlled, i.e., the opening between the two beds has to be determined, in order to maintain the bed temperature, T_1, constant. Note that ΔH is largest at $t = 0$ and decreases progressively when the bed temperature, T_2, increases. Thus the design should be carried out based on ΔH at $t = 0$.

For example, to provide a 10%/min load swing capability for the industrial fluidized bed boiler, a boiler module which constitutes 25% of the total load has to be brought to full operation in 2.5 minutes. This represents the most pessimistic case and thus the minimum time limit. Assuming the bed temperature required for self-sustained combustion in a fluidized bed is 700-800°F, the time required to increase the bed temperature from 800°F to 1650°F is estimated to be ∿2.5 minutes from PER and British experiences.[1,2] That gives no time at all to heat the bed from 489°F to 800°F. In actual operation, however, we can fire the operating beds at >1650°F, say 1700°F or 1800°F, to meet part of the load swing requirement. In this way we can relax the time constant from 2.5 minutes to about 5 minutes or even longer. Assume now we have 2.5 minutes to heat the bed from 489°F to 700 °F (case 1) or 800°F (case 2) before coal injection into the bed. The solid circulation rate required from the operating bed at 1650°F can be found by trial and error from Eq. 4. to be $W = 12.0 \times 10^4$ lb/hr. Assuming solid circulation rate of 1×10^4 lb/hr ft^2,[3,4] an opening between adjacent beds of 12 ft^2 is required. The amount of increase in the coal feeding rate required can be found from Eq. 10 and is presented in Figure F-1 for both case 1 and case 2. At the moment Bed 2 is fluidized, a step increase of coal feed to Bed 1 to 145% (case 1) and 168% (case 2) of the original coal rate is required to maintain the bed temperature constant. Increasing the bed temperature of Bed 2 during this heat-up period will decrease the coal feed requirement to Bed 1 as shown in Figure F-1. Increase in coal feed necessitates increase in air flow proportionally. During this heating-up period, Bed 1 is thus required to operate at up to 18 ft/sec (case 1) and 21 ft/sec (case 2) for a period of 2.5 minutes. Increase in carbon carry over due to increase in fluidizing velocity during this period is probably small, although more experimental studies must be performed to confirm it. The bed temperature change and the steam production of Bed 2 during this period is also presented in Figure F-2 (here the gas-side heat transfer coefficient in the minimally fluidized bed is taken to be $h_2 = 20$ Btu/ft^2-hr-°F).

The effectiveness of using solid recirculation between beds for hot restart as described requires further experimental verification. The foreseeable problems are:

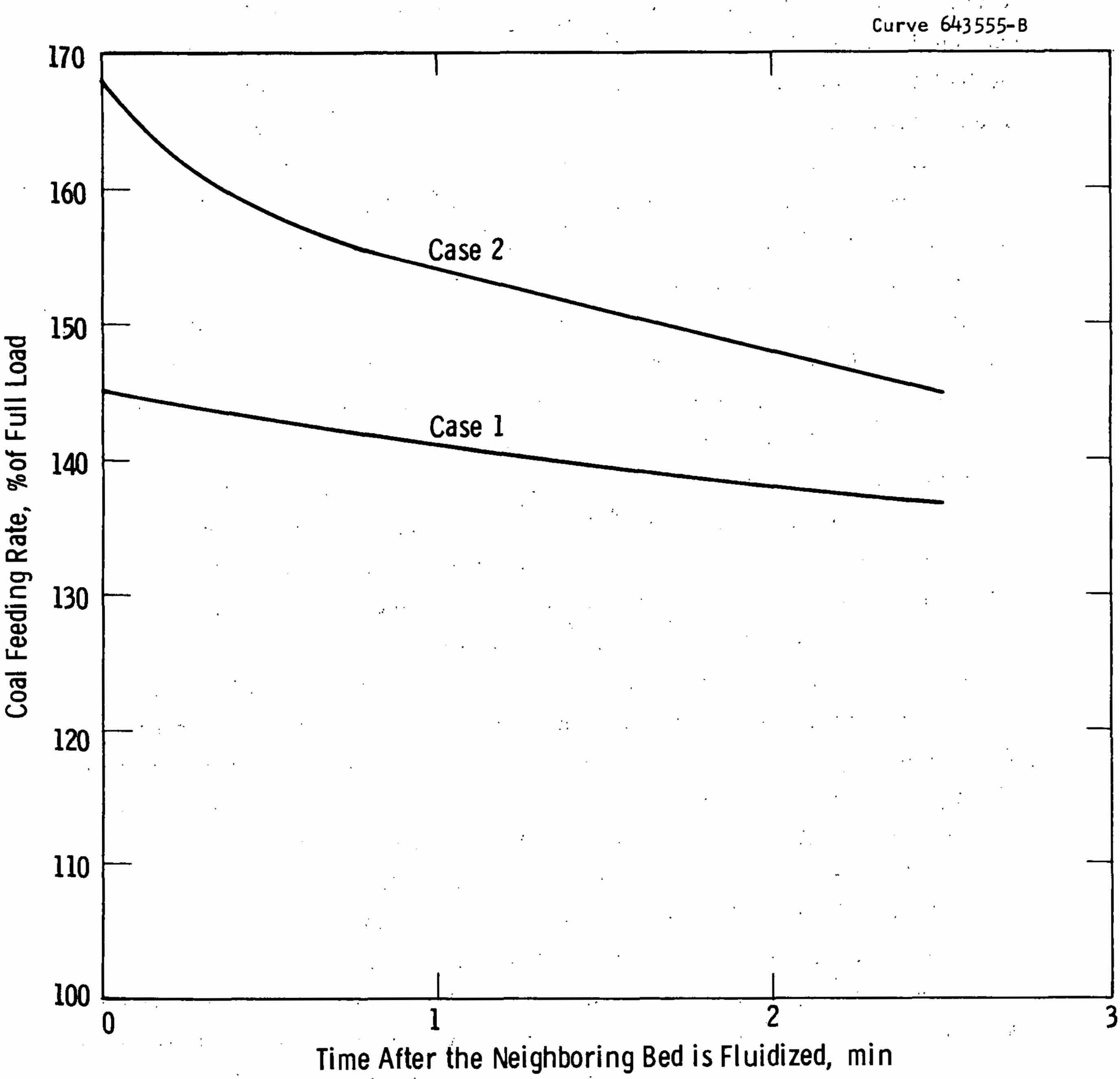


Fig. F-1 —Coal requirement in bed 1 during ignition of the neighboring bed

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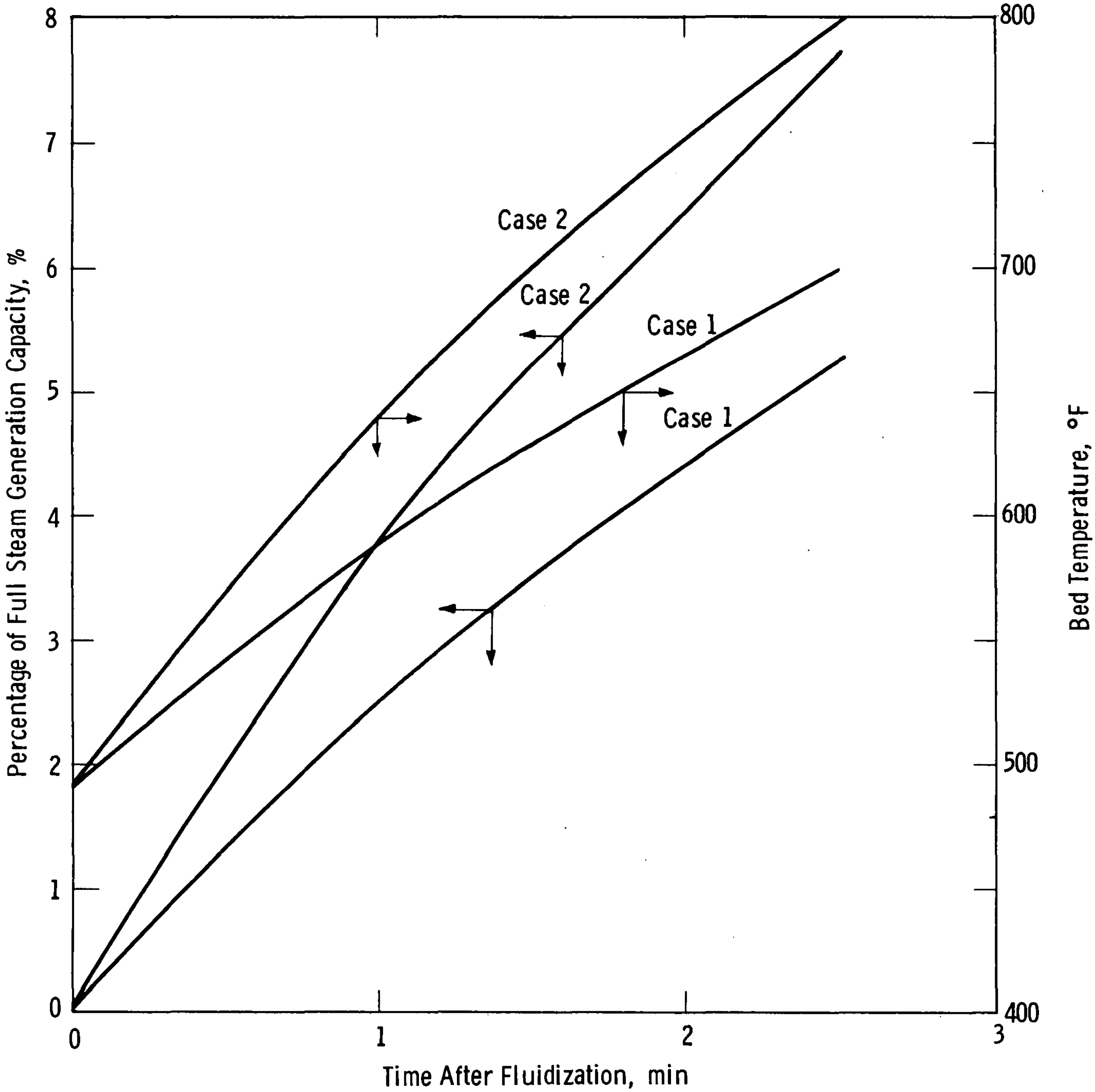

Fig. F-2—Rate of bed temperature change & steam generation in bed 2 during heating-up period

1) Stability of bed temperature. At the moment of fluidizing Bed 2, the stability of bed temperature in Bed 1 is of concern. Will the bed temperature fluctuation be short-lived or persistent and last over the period of restart? Will this temperature fluctuation be detrimental to overall control stability?

2) Temperature gradient. An extreme temperature gradient may exist in both beds during restart, especially in Bed 2 which is only minimally fluidized. For small beds, the problem is not serious. However, when the bed area becomes larger, the problem may increase in dimensions. What are the size limitations beyond which this scheme will no longer provide the beneficial and possible way for restart?

Sizing the Burners for Start-up and Restart During Load Swing

Sizing the burners for initial start-up is less important than sizing the burners for restart, because there is no stringent time limit for initial start-up of a boiler. However, there is a strict time response requirement for restart of a bed to meet the designed rate of load response of a boiler. The size of the burner required to meet this load response can be estimated from the following energy balance equation in the bed:

$$H - hA(T - Ts) - FCp_a(T - Ta) = W_B Cp_s \frac{dT}{dt}. \qquad (12)$$

Note that no solid circulation from the neighboring bed is assumed.

One idled module of the boiler system is assumed to be kept hot at all times by recirculation of saturated water through the immersed tubes from the steam drum, in the case of an atmospheric industrial boiler, or from a flash tank at $\sim$1000 psi (saturated water temperature 545°F), in the case of an atmospheric utility boiler. When restarting an idle bed, only minimum fluidizing velocity is provided ($\sim$1 ft/sec) to minimize the heat transferred to the immersed tubes (assume $h = 20$ Btu/ft^2-hr-°F).

Integrating Eq. 12 gives

$$H = \frac{FCp_a Ta \{[Ex] -1\} + FCp_a \{T - Ts[Ex]\} -hA(Ts - T)}{1 - [Ex]} \tag{13}$$

where

$$Ex = \exp \left[- \frac{t(hA + FCp_a)}{W_B Cp_s} \right].$$

From Eq. 13 the theoretical heat requirement to heat the bed to a specified temperature in a finite time limit can be estimated. The heat required to heat the bed to 800°F for both atmospheric industrial and utility boilers is presented in Figure F-3 for different response times. For example, to heat the bed of the industrial boiler from 489°F to 800°F in 2.5 minutes, corresponding to 10%/min load response, the theoretical heat requirement is 29.0×10^6 Btu/hr, from Figure F-3. However, the burners may have low efficiencies. From analysis of the data collected by PER[1] and the British[2] using Eq. 13, the actual burner efficiency during start-up is found to be about 30% to 50% depending on the position of burners in the bed. Assuming the burner has 50% efficiency, then a burner of size of 58×10^6 Btu/hr is required for each individual bed to give the stated load response. The burners required (assuming only three are needed) are thus almost equivalent to 50% of the total boiler capacity. Hence, it would be more economical to use solid circulation between beds for restart purposes. Similar analysis can also be done for the atmospheric utility boiler.

Curve 643552-B

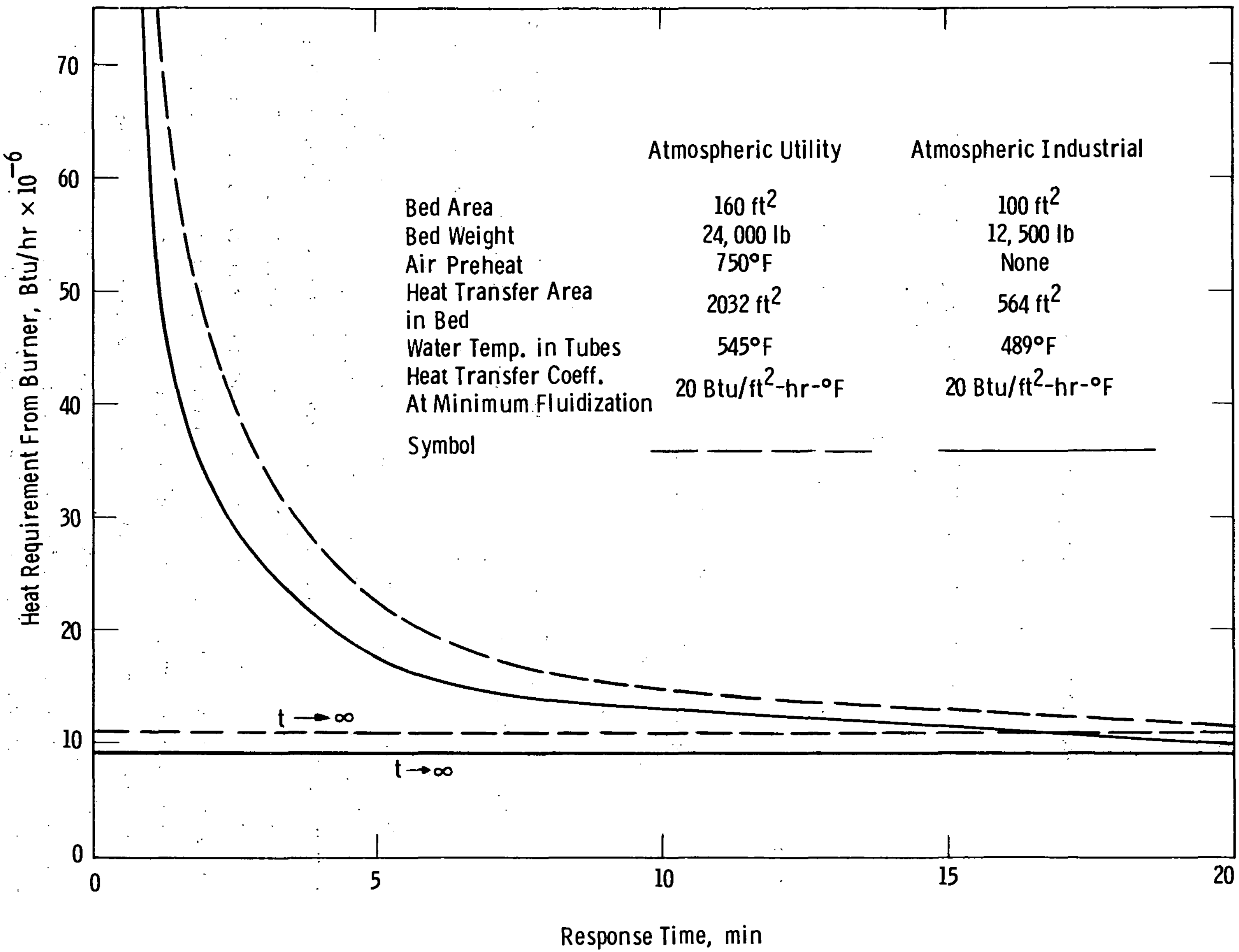

Fig. F-3 —Theoretical heat requirement in heating the bed to 800°F

The single factor which is responsible for the large burners required for start-up and restart is the heat transfer from the bed to the immersed water tubes in the bed. Any means which can reduce this heat transfer during start up will be helpful, e.g., keeping the water temperature in the tube essentially equal to the bed temperature by close looping the water circulation circuitry during start-up.

Alternative methods for start-up and restart besides the solid circulation scheme discussed in the last section are summarized below:

A. <u>Methods for Restart</u>

(1) Use Gas Burners:

Gas burners can be directed on the bed surface to heat up the bed. Disadvantages are a) low burner efficiency, b) tube screen immediately above the bed has to be removed.

Gas burners can be put between the tube bundles and distributor plate. Disadvantages are a) space limitation - some tubes may have to be removed from the bed, b) there may be some danger of impinging high-temperature gas flame on the water tubes.

External gas burners -- combust the gas in a separate burner and inject the high-temperature gas into the bed. Disadvantages are a) the arrangement might be more expensive, b) low efficiency - if the high-temperature gas cannot be distributed through the distributor plate because of temperature limitation.

(2) Solids Handling:

Heat up initially using shallow bed in order to reduce burner size. Control of bed level may be difficult.

Utilize high-temperature regenerated limestone to assist in heating the bed. This requires flexibility in the solids flow control system for the regenerator and may require a high-temperature surge tank.

B. Methods for Initial Start Up

Methods for initial start up are less restricted because the response time limitation is relaxed. In addition to the methods discussed above, some other methods can also be used.

(1) Use preheat air and auxiliary burner. Disadvantages are a) separate burners outside boiler are needed for preheating the air, b) long response time.

(2) Pass gas into the bed (through separate pipes or coal feeding port) and ignite above the bed. Disadvantages are a) low burner efficiency, b) tube screen immediately above the bed has to be removed, c) danger of having high-temperature gas flame around water tubes in the bed.

Rate of Bed Temperature Change and Residual Steam Generation in a Fluidized Bed after Shut-down

An operating fluidized bed can be turned off by (1) stopping the coal and air feed simultaneously or (2) stopping only the coal feed and still keeping the bed fluidized. In both of these cases, the bed eventually will drop off to the equilibrium ambient temperature and cease generation of steam. In the case where solid circulation from the neighboring operating bed is possible, the residual steam generation will never cease but only level off after equilibrium conditions are reached. These three cases are analyzed below.

Case (1)

When a fluidized bed is turned off for turn-down purposes by stopping the coal and air feed, the bed becomes defluidized immediately; however, the water in the tubes is kept running to prevent the overheating of the tubes in the bed and thus continuously draws heat from the bed. Before the bed temperature is cooled down to the water temperature in the tube, the turned-off bed will continue to generate steam at a rate dependent on the cooling rate of the bed temperature. To estimate the rate of bed temperature change, the following model is employed, as shown in Figure F-4. In the industrial boiler, 1-inch water tubes are staggered in the bed with 6-inch horizontal spacing and 3-inch vertical spacing. If imaginary cylinders of 3-inch diameter are slipped over each tube, as shown in Figure F-4, analysis of the bed volume encompassed by a single cylinder would be a reasonable approximation for the behavior of the total bed.

Energy balance within a single cylinder results in the following partial differential equation

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} = \frac{\rho Cp}{k} \cdot \frac{\partial T}{\partial t} \qquad (14)$$

with boundary conditions

$r = R_i$	$T = T_s$	for $t \geq o$
$r = R_o$	$\frac{\partial T}{\partial r} = o$	for $t > o$
$t = o$	$T = To$	for $R_i < r \leq R_o$

where

- r: radial coordinate
- R_i: outside radius of water tube
- R_o: radius of the imaginary cylinder
- t; time variable; $t=o$ is the time the bed is turned off
- T_s: saturated water temperature
- T_o: bed temperature during shut-down
- ρ: density of the bed
- Cp: heat capacity of the bed
- k: heat conductivity of the bed.

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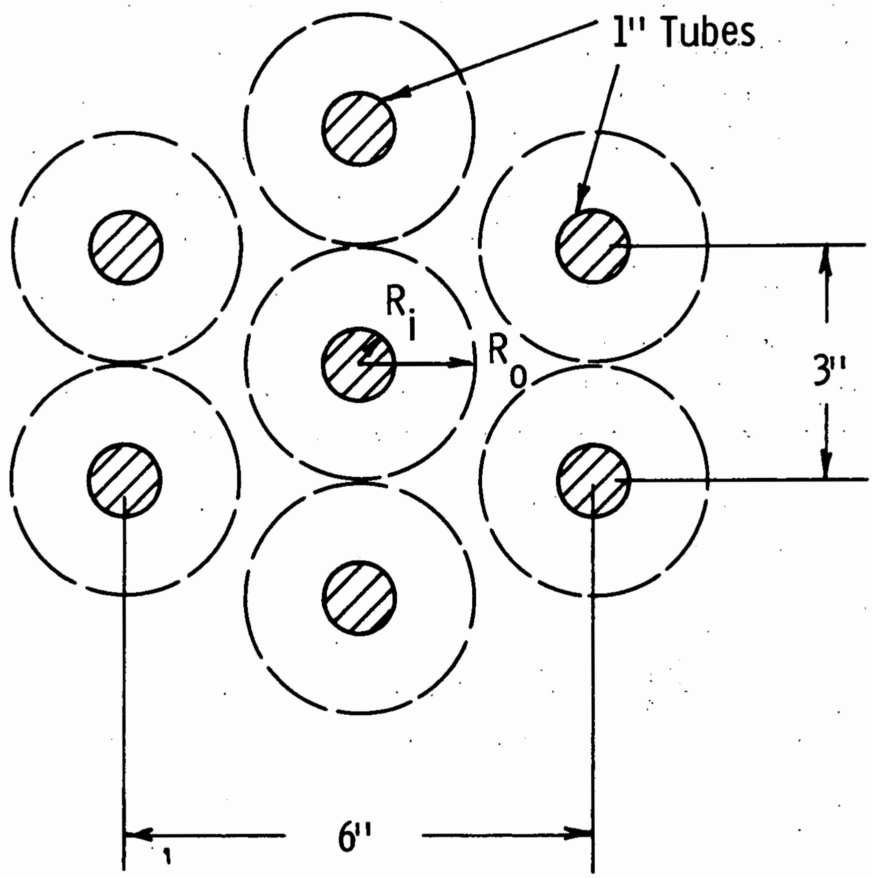

Fig. F-4—Model for estimating dynamics of Industrial Fluidized Bed Boiler

The solution of Eqs. (14) and (15) is

$$\frac{T - T_s}{T_o - T_s} = \pi \sum_{n=1}^{n=\infty} \frac{[J_o(\frac{R_i}{R_o}\alpha_n)\, Y_o(\frac{r}{R_o}\alpha_n) - J_o(\frac{r}{R_o}\alpha_n)\, Y_o(\frac{R_i}{R_o}\alpha_n)]}{[\frac{J_o(\frac{R_i}{R_o}\, _n)}{J_1(\alpha_n)}]^2 - 1} \cdot \exp[-\frac{k\alpha_n^2 t}{\rho C_p R_o^2}] \tag{16}$$

With α_n's the roots of

$$J_1(\alpha_n)\, Y_o(\frac{R_i}{R_o}\alpha_n) - J_o(\frac{R_i}{R_o}\alpha_n)\, Y_1(\alpha_n) = 0. \tag{17}$$

Here the J_n's and Y_n's are the Bessel Function of 1st and 2nd kind. The first two roots, α_1 and α_2, of Eq. (17) were determined graphically to be 1.877 and 6.3 respectively. The temperature change with time at $r = R_o$ was calculated and plotted in Figure F-5 for the industrial boiler where T_s = 489°F; T_o = 1650°F; R_i = 0.5"; R_o = 1.5"; ρ = 45 lb/ft^3; Cp = 0.24 $\frac{Btu}{lb\text{-}°F}$; k = 0.377 Btu/hr-ft-°F.

The value of heat conductivity, k, used here is obtained from fitting actual PER experimental data by assuming that the model of heat transfer from a semi-infinite slab to surroundings at uniform temperature is applicable. It is interesting to note that the k value used here is approximately the average heat conductivity of air and limestone. From Figure F-5 the bed temperature at $r = R_o$ is found to decrease from 1650°F to half of its value 825°F in 11.5 minutes and to 617°F in 20 minutes.

The average temperature change can be evaluated by defining

$$\overline{\Delta T} = \overline{(T - T_s)} = \frac{\int_{R_i}^{R_o} 2\pi r \cdot \Delta T \cdot dr}{\pi(R_o^2 - R_i^2)} \; ; \; \Delta T = T - T_s . \tag{18}$$

Curve 642036-B

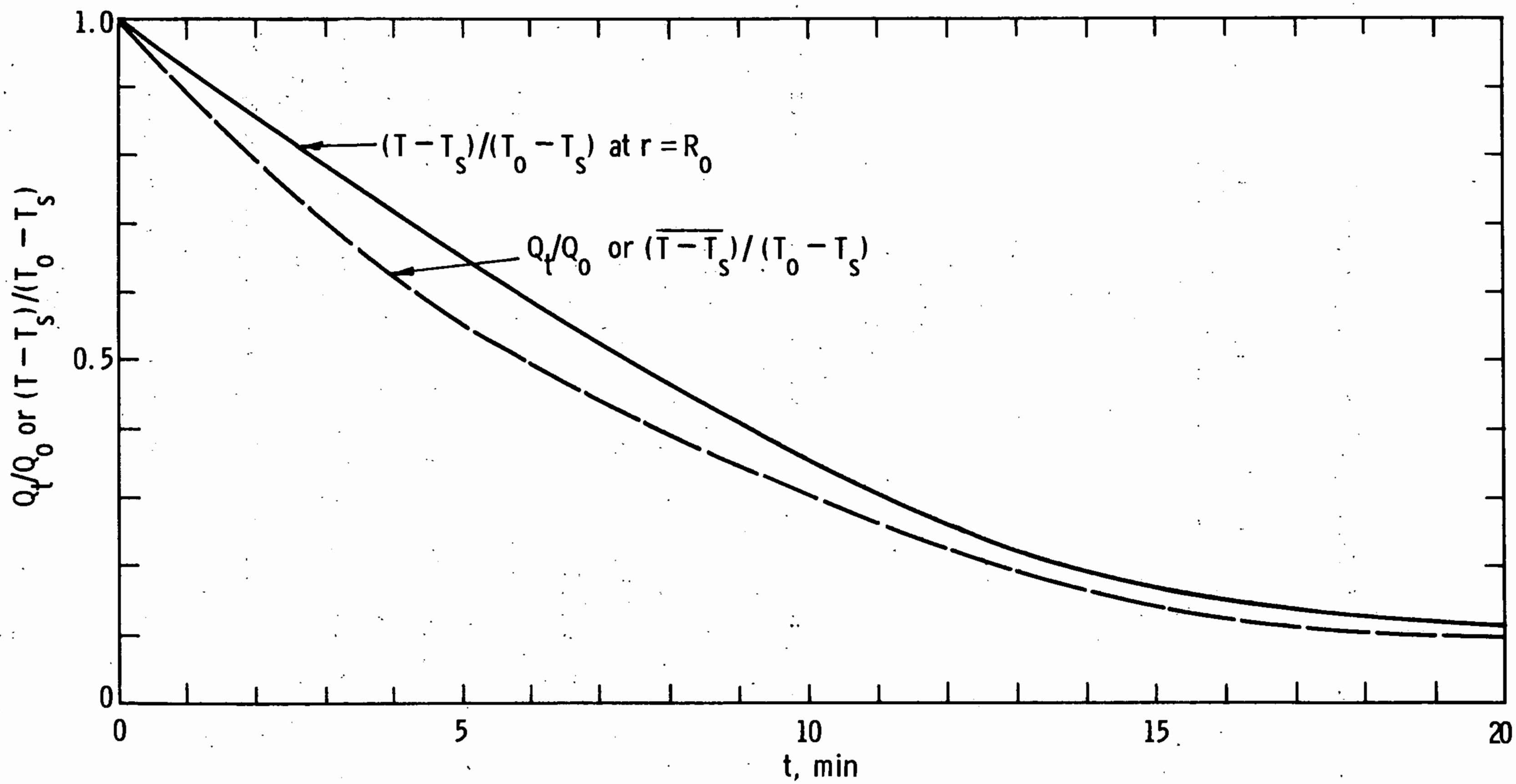

Fig. F-5 —Change of bed temperature & bed sensible heat after shut-down

Substituting Eq. (16) into (18) and integrating, we have

$$\frac{\overline{\Delta T}}{T_o - T_s} = \tag{19}$$

$$\frac{2\pi(\frac{R_i}{R_o})}{(1 - \frac{R_i^2}{R_o^2})} \sum_{n=1}^{n=\infty} \cdot \frac{[Y_o(\frac{R_i}{R_o}\alpha_n)J_1(\frac{R_i}{R_o}\alpha_n) - J_o(\frac{R_i}{R_o}\alpha_n)Y_1(\frac{R_i}{R_o}\alpha_n)]}{\alpha_n\{[\frac{J_o(\frac{R_i}{R_o}\alpha_n)}{J_1(\alpha_n)}]^2 - 1\}} \exp(-\frac{k\alpha_n^2}{\rho C p R_o^2}t).$$

Total sensible heat available from the bed material at t = o is

$$Q_o = V \rho Cp(T_o - T_s) \tag{20}$$

and total sensible heat remained in the bed at time t after shut-off is

$$Q_t = V \rho Cp \overline{\Delta T} \tag{21}$$

or

$$\frac{Q_t}{Q_o} = \frac{\overline{\Delta T}}{(T_o - T_s)} . \tag{22}$$

Equations (19) and (22) were also calculated and plotted for the industrial boiler in Figure F-5. The result shows that the overall sensible heat of the bed material is reduced by half in 7 minutes. The average bed temperature as defined in Eq. (18) is also plotted as Curve 1 in Figure F-6. When the heat conductivity is assumed to be 0.2 Btu/ft-hr-°F instead of 0.377 Btu/ft-hr-°F, the time required to cool the bed to a given temperature is shown in Figure F-8. The effect of heat conductivity is not overwhelming.

Curve 643557-B

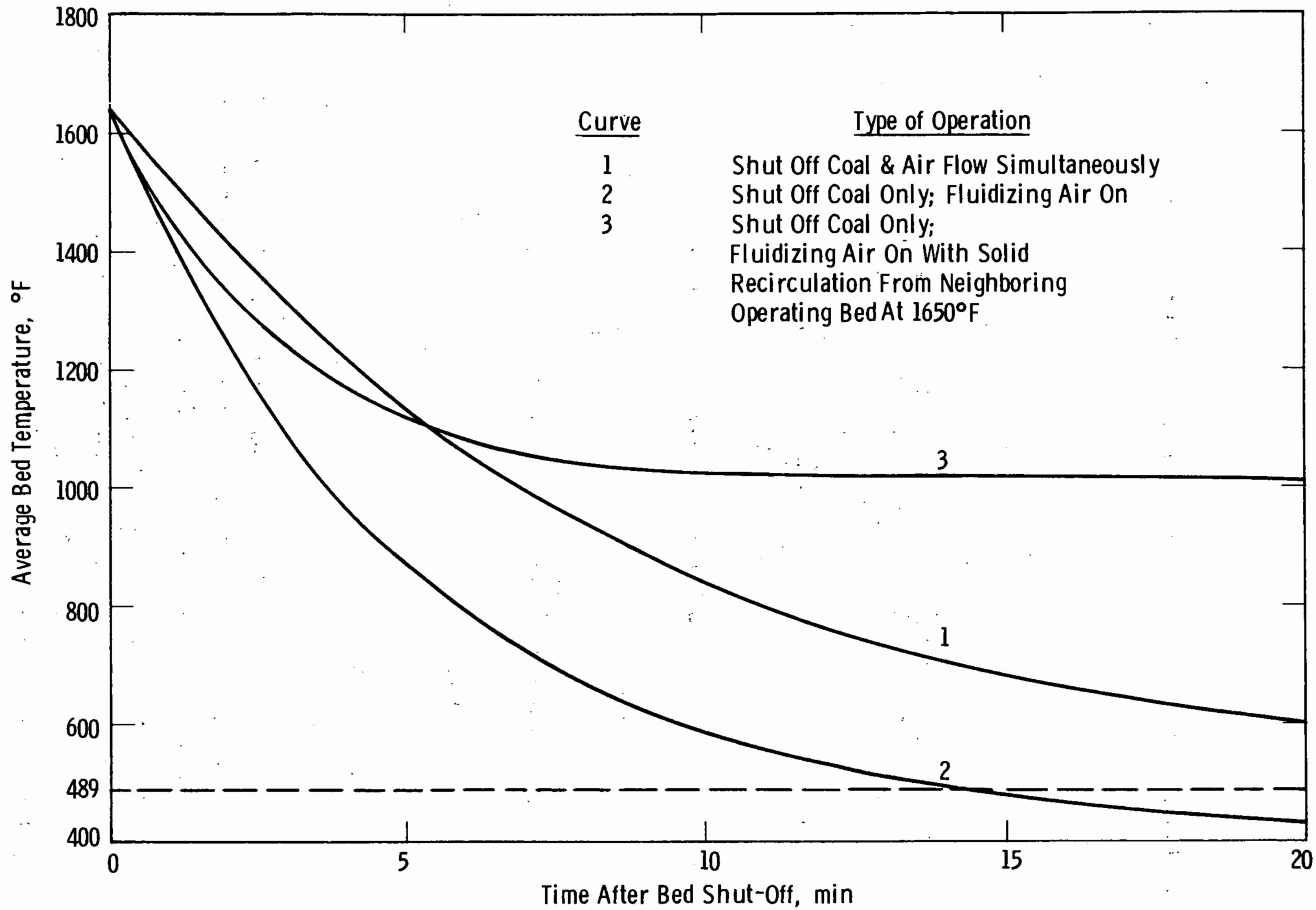

Fig. F-6 —Change of bed temperature after shut-down

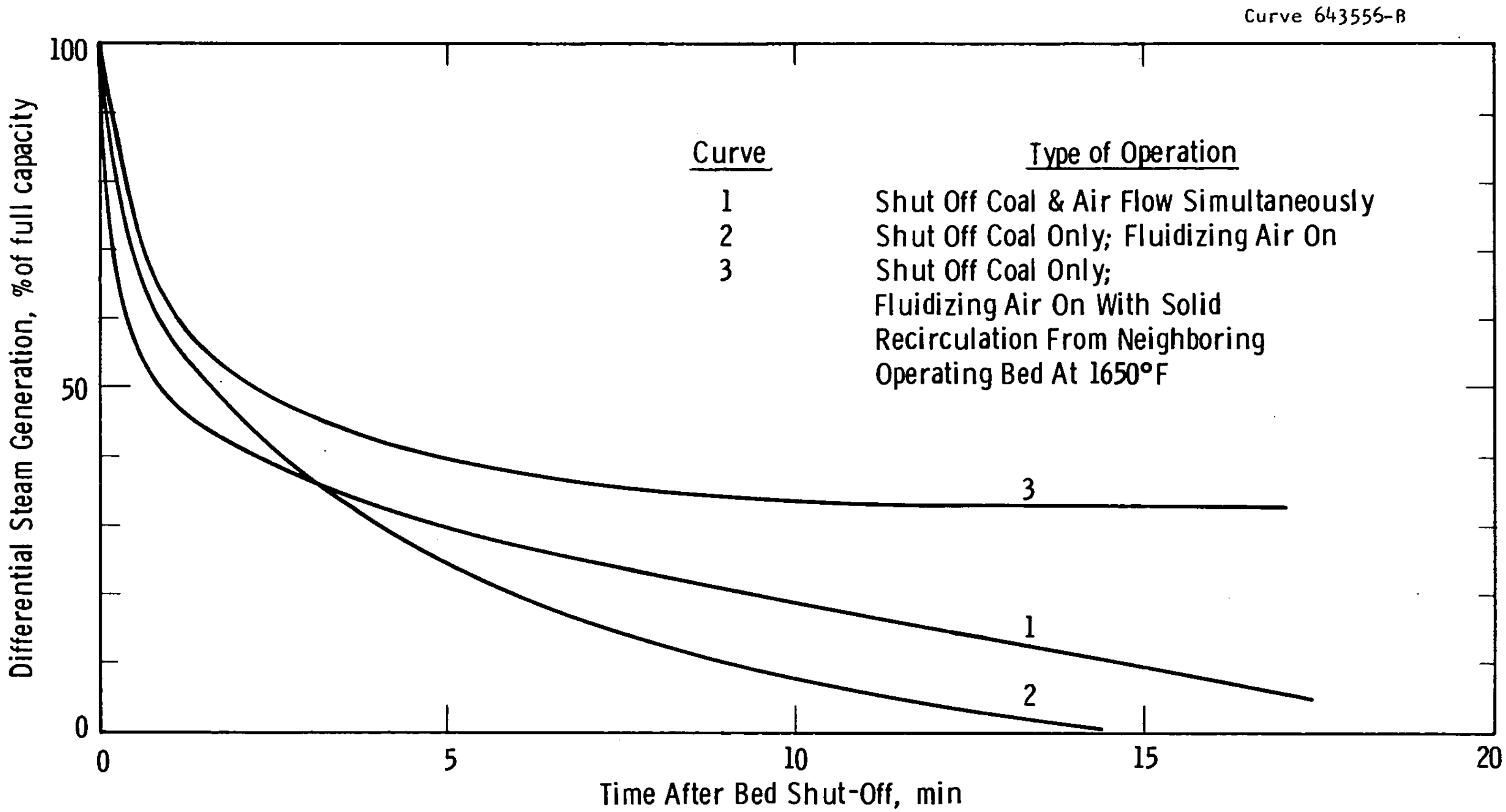

Fig. F-7 —Residual steam generation after bed shut-off

Curve 643553-B

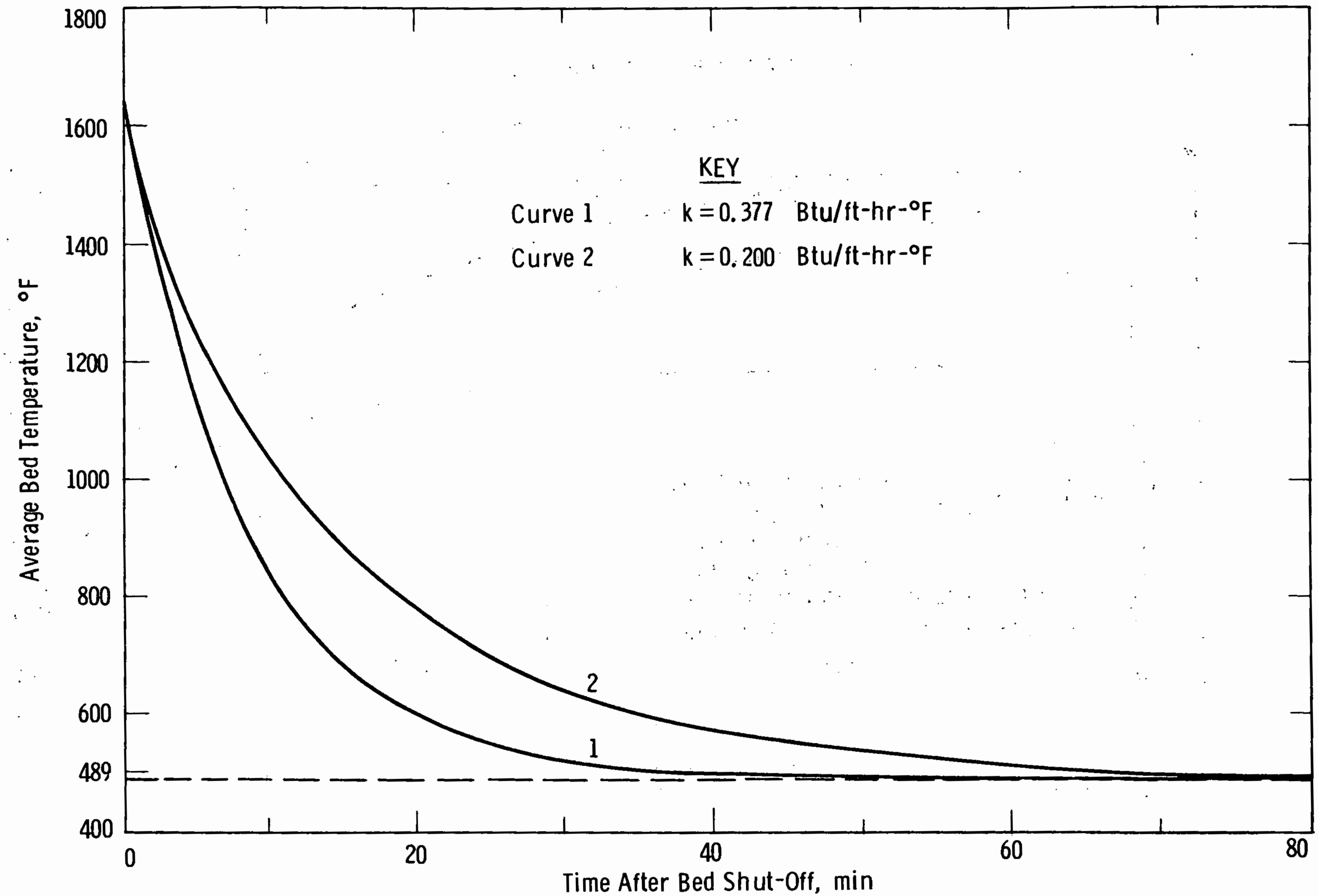

Fig. F-8—Change of bed temperature after bed is slumped

Note that the equations derived are general and can be applied to other fluidized bed configurations with different tube arrangements, bed material, and steam conditions for both atmospheric or pressurized boiler designs.

Case (2)

When the operating fluidized bed is turned off by stopping the coal feed only, the bed remains fully fluidized. The rate of bed temperature change can be expressed by the following equation:

$$hA(T - T_s) + F\,Cp_a(T - T_a) = -\,W_B Cp_s \frac{dT}{dt}\,. \tag{23}$$

Integrating Eq. (23) gives

$$T = \frac{k_3}{k_4} + [T_o - \frac{k_3}{k_4}]\ \exp[-\frac{k_4 t}{W_B Cp_s}] \tag{24}$$

where

$$k_3 = hAT_s + FCp_a T_a \tag{25}$$

$$k_4 = hA + FCp_a. \tag{26}$$

The bed temperature change after shut-down expressed by Eq. (24) is evaluated for the industrial boiler and shown as Curve 2 in Figure F-6. The corresponding residual steam generation rate is also shown as Curve 2 in Figure F-7.

Case (3)

When solid circulation from the neighboring operating bed at 1650°F is possible, the bed temperature change after shut-down can be expressed as in Eq. (5). Both the rate of bed temperature change and residual steam generation are shown in Figures F-6 and F-7 as Curve 3.

It is interesting to note that although the average bed temperature in Case 1 decreases less rapidly compared to that in Case 2 (Figure F-6), the reduction in residual steam generation is actually larger in the first 3 minutes after bed shut-down, as shown in Figure F-7. The reason is that in Case 1 the average bed temperature results from integration as defined in Eq. (18) while the total heat transferred from the bed to the water tubes depends on the temperature gradient at $r = R_i$. For an industrial boiler generating steam at a rate of 250,000 lb/hr, average steam generation in each bed at full load is 1042 lb/min. When cooled down from 1650°F to 489°F the total sensible heat of bed material is capable of generating 4800 lb of steam. The cooling rate of bed temperature determines the rate of residual steam generation after bed shut-off. At the instance of defluidization, the temperature of the solid layer immediately outside the water tubes drops off rapidly; however, the rate of heat transfer levels off after the first few minutes because of slow heat conduction through solid layers of bed material. After bed shut-off the steam generation capacity of the bed is halved in one minute. The further reduction in steam generation is more gradual; after 5 minutes, the bed is still generating 30% of full steam capacity.

Theoretically, at $t = \infty$ the bed temperature will be in equilibrium with the water tube temperature, i.e., the saturated water temperature, 489°F. Thus, it is logical to keep the saturated water continuously running in the tubes to maintain the bed temperature essentially at 489°F for fast light-off of the bed during load swing. For an expected long idle period of the bed, the scheme is, of course, not justifiable because of the pumping cost.

The same scheme (running the saturated water in the tubes) can also be used to heat up the bed for cold start-up. The rate of bed temperature change was calculated for the industrial boiler case as shown in Figure F-9. Without fluidizing, the bed can be heated up from 80°F to 440°F in 20 minutes.

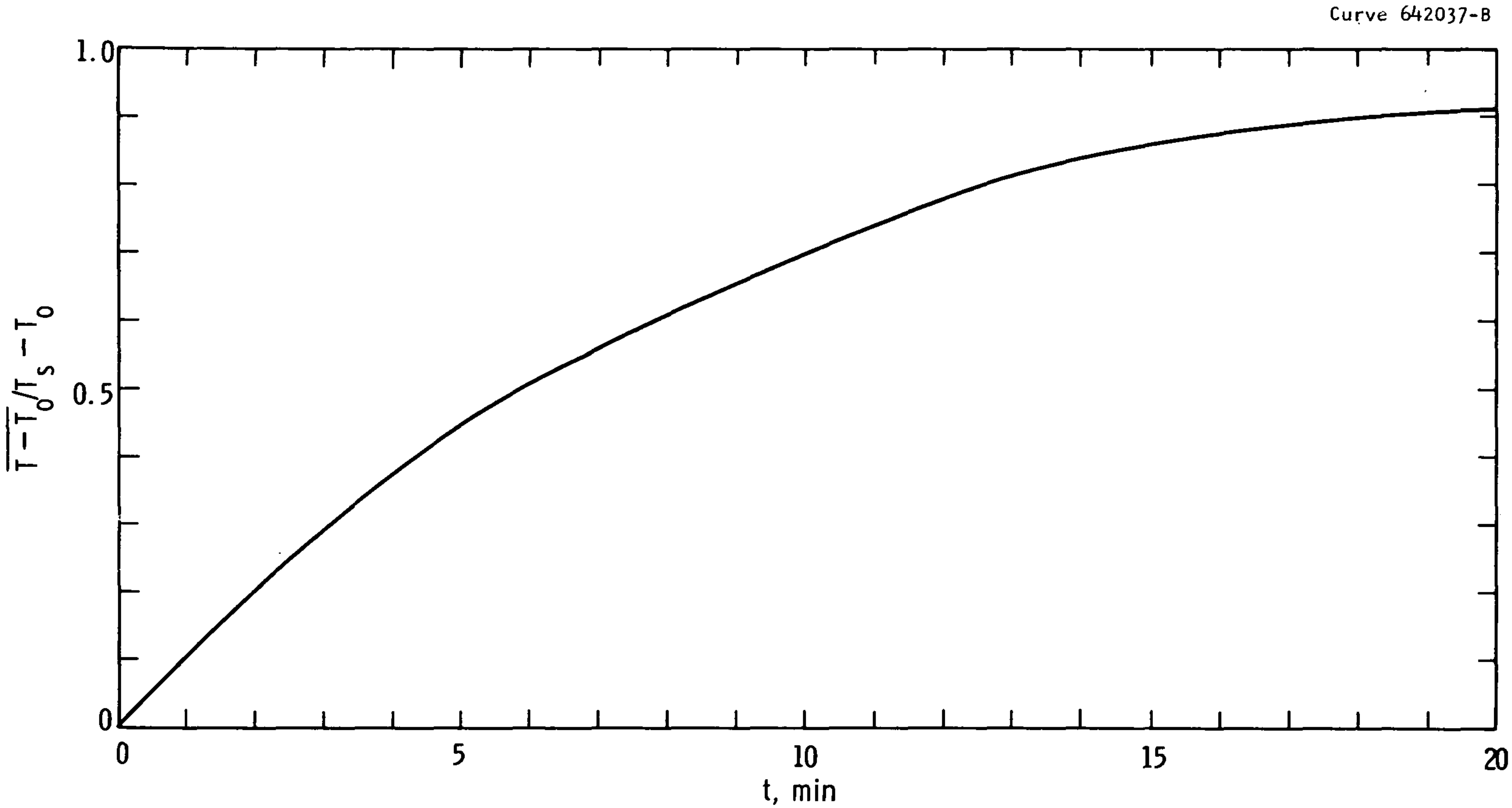

Fig. F- 9—Change of bed temperature during cold start by passing saturated water through heat transfer tubes in the bed

In Case 2, the steam generation decreases to zero after 14.5 minutes. From there on, heat transfer reverses the direction from water tubes at 489°F to the bed for heating the incoming fluidizing air at 80°F. The bed temperature finally levels off at 408°F.

For Case 3, the final bed temperature is 1011°F with a residual steam generation rate of 32% of full capacity due to solid recirculation from the neighboring operating bed at 1650°F.

To provide a 10%/min load change during turn-down, a bed of 25% total boiler capacity has to be reduced from 100% load to 0% load in 2.5 minutes. Clearly the response of the three methods considered is not fast enough to provide the load reduction (Figure F-7). However, by shutting down the designated bed before it reaches the lowest allowable operating bed temperature, further reduction of temperature in the remaining operating beds can reduce the load further and attain the load response required.

Highley, of the National Coal Board, England, has analyzed the same problem[5] of rates of cooling the slumped and fluidized beds of ash by steam tubes (equivalent to Case 1 and Case 2 in our discussion here). Although his analysis of Case 2 shows results consistent with those of the present analysis, his result in Case 1 gives about 10 times longer for the time required to cool the bed temperature to a designated temperature. In other words, his calculation of cooling rates is under-estimated by a factor of 10 if his data are used in Eqs. (16) and (19). The discrepancy comes primarily from his assumption that the tubes are surrounded by an infinite bed, while the present analysis takes into consideration the geometrical limitations of different tube arrangements in the bed, and thus is inherently a better model in predicting the cooling rate of a slumped bed. The general analytical solutions in close form as in Eqs. (16), (17), and (19) are more accurate and convenient to use.

REFERENCES

1. "Development of Coal-Fired Fluidized-Bed Boilers," Interim Report to Office of Coal Research, U. S. Dept. of Interior, by Pope, Evans and Robbins (1969).

2. Smale, A. W., "Investigation of Start-Up Techniques for Fluidized Combustion Using a One Foot Fluidized Bed Combustor," Fluidized Combustion Section Rept. No. 41, Coal Research Establishment, National Coal Board, England (1969).

3. "Study of Characterization and Control of Air Pollutants from a Fluidized-Bed Combustion Unit," Monthly Progress Rept. No. 9 (June 1970), prepared under Contract No. CPA 70-10 for NAPCA by Pope, Evans and Robbins.

4. Ehrlich, S., Robison, E. B., Gordon, J. S., and Bishop, J. W., "The Fluid Bed Boiler and Its Applicability to Pollution Control," Paper No. 8, Presented in AIChE 69th National Meeting, Cincinnati, Ohio (May 1971).

5. Highley, J., "Rates of Cooling of Slumped and Fluidized Beds of Ash by Steam Tubes," Fluidized Combustion Section Rept. No. 13, Coal Research Establishment, National Coal Board, England (1968).

APPENDIX G

OPTIMIZATION OF HEAT TRAP SYSTEM COST

ABSTRACT

Two methods, one graphic and one analytical, were presented for determining the optimum air preheat temperature and for optimizing the cost of the heat trap system, including economizer and air preheater.

The calculation is sensitive to the relative unit cost of economizer and air preheater. For a 300 MW atmospheric design with operating conditions similar to those outlined in the 14th Monthly Progress Report, the optimum air preheat temperature (which minimizes the total cost of the heat trap system) was found to be 560°F.

OPTIMIZATION OF HEAT TRAP SYSTEM

For considerations of economy, means of recovering part of the flue gas energy before the stack are necessary in a steam generating unit. To accomplish this, either the economizer or the air preheater or both are usually used. When both the economizer and the air preheater are employed, there is an optimum distribution of heat load between economizer and air preheater which minimizes the total cost of the heat recovery apparatuses. In the case of a fluidized bed boiler, additional heat transfer surface has to be provided in the bed to transfer the sensible heat carried by the preheated air to maintain a constant bed temperature. To be rigorous, the cost of this additional surface required in the bed has to be included in the determination of the optimum air preheat temperature.

Consider the heat recovery system as shown in Figure G-1, where:

T_b = flue gas inlet temperature to economizer

T_i = flue gas outlet temperature from economizer or inlet temperature to air preheater

T_o = flue gas outlet temperature from air preheater (stack gas temperature)

T_{ai}, T_{ao} = air inlet and outlet temperature to air preheater

T_{wi}, T_{wo} = water inlet and outlet temperature to economizer

W_f, W_a, W_w = mass flow rate of flue gas, air, and water respectively

$\overline{C_{pf}}, \overline{C_{pa}}, \overline{C_{pw}}$ = average heat capacity over the temperature range considered for flue gas, air, and water respectively.

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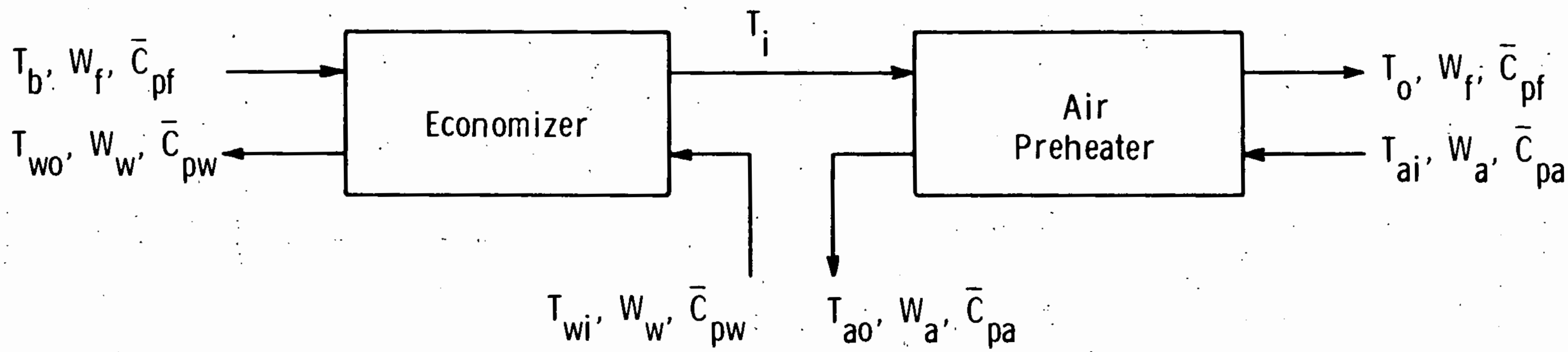

Fig. G-1—Flue gas heat recovery system

In the actual design, the temperatures T_b, T_o, T_{ai}, and T_{wi} are fixed, and the air preheat temperature, T_{ao}, is to be found by optimization. The unknown temperatures T_i and T_{wo} can be expressed in terms of a single variable, T_{ao}, by following energy balances.

Heat balance at air preheater:

$$W_f \cdot \overline{C_{pf}} \cdot (T_i - T_o) = W_a \cdot \overline{C_{pa}} \cdot (T_{ao} - T_{ai}) \tag{1}$$

or

$$T_i = T_o + \frac{W_a}{W_f} \cdot \frac{\overline{C_{pa}}}{\overline{C_{pf}}} (T_{ao} - T_{ai}) \tag{2}$$

Heat balance at economizer:

$$W_f \cdot \overline{C_{pf}} \cdot (T_b - T_i) = W_w \cdot \overline{C_{pw}} \cdot (T_{wo} - T_{wi}) \tag{3}$$

or

$$T_{wo} = T_{wi} + \frac{W_f}{W_w} \cdot \frac{\overline{C_{pf}}}{\overline{C_{pw}}} (T_b - T_o) - \frac{W_a}{W_w} \cdot \frac{\overline{C_{pa}}}{\overline{C_{pw}}} (T_{ao} - T_{ai}) \tag{4}$$

Heat load at air preheater:

$$W_f \cdot \overline{C_{pf}} \cdot (T_i - T_o) = W_a \cdot \overline{C_{pa}} \cdot (T_{ao} - T_{ai}) \tag{1}$$

Heat load at economizer:

$$W_f \cdot \overline{C_{pf}} \cdot (T_b - T_i) = W_f \cdot \overline{C_{pf}} (T_b - T_o) - W_a \cdot \overline{C_{pa}} \cdot (T_{ao} - T_{ai}) \tag{5}$$

LMTD (logarithmic mean temperature difference) in air preheater:

$$\frac{(T_i - T_{ao}) - (T_o - T_{ai})}{\ell n \dfrac{(T_i - T_{ao})}{(T_o - T_{ai})}} = \frac{(T_{ao} - T_{ai}) \left(\dfrac{W_a}{W_f} \cdot \dfrac{\overline{C_{pa}}}{\overline{C_{pf}}} - 1\right)}{\ell n \dfrac{(T_i - T_{ao})}{(T_o - T_{ai})}} \tag{6}$$

LMTD in economizer:

$$\frac{(T_b - T_{wo}) - (T_i - T_{wi})}{\ell n \dfrac{(T_b - T_{wo})}{(T_i - T_{wi})}}$$

$$= \frac{\left(\dfrac{1}{W_f \cdot \overline{C_{pf}}} - \dfrac{1}{W_w \cdot \overline{C_{pw}}}\right) \left[W_f \cdot \overline{C_{pf}} (T_b - T_o) - W_a \overline{C_{pa}} (T_{ao} - T_{ai})\right]}{\ell n \dfrac{(T_b - T_{wo})}{(T_i - T_{wi})}} \tag{7}$$

The total cost of the heat recovery system, after considerable simplification, can be expressed as:

$$C_T = C_A \cdot \frac{\ell n \dfrac{(T_i - T_{ao})}{(T_o - T_{ai})}}{h_A \cdot \left(\dfrac{1}{W_f \cdot \overline{C_{pf}}} - \dfrac{1}{W_a \cdot \overline{C_{pa}}}\right)} + C_E \cdot \frac{\ell n \dfrac{(T_b - T_{wo})}{(T_i - T_{wi})}}{h_E \cdot \left(\dfrac{1}{W_f \overline{C_{pf}}} - \dfrac{1}{W_w \cdot \overline{C_{pw}}}\right)}$$

$$+ C_B \cdot \frac{W_a \cdot C_{pa} \cdot (T_{ao} - T_{ai})}{h_B \cdot (T_B - T_S)} \tag{8}$$

where

C_A, C_E, C_B = the unit cost of heat transfer surface in air preheater, economizer, and bed, respectively.

h_A, h_E, h_B = heat transfer coefficient in air preheater, economizer, and bed respectively

T_B = bed temperature

T_S = saturated water temperature.

The optimum air preheat temperature, T_{ao}, can be found by setting

$$\frac{dC_T}{dT_{ao}} = 0. \tag{9}$$

Since the heat transfer coefficient and LMTD in the bed are much higher than in the air preheater, while the unit cost of heat transfer surface is slightly higher, the cost of additional heat transfer surface required in the bed for absorbing the sensible heat of the preheated air is estimated to be about 10% of the cost of the air preheater. Hence, for mathematical convenience the third term in Equation (8) was neglected and its cost was combined with the cost of the air preheater by increasing the unit cost of the air preheater 10%. Substituting Equations (2) and (4) into Equation (8) and carrying out the mathematical manipulation for Equation (9), we found that the optimum air preheat temperature can be parametrically expressed in the following equation:

$(T_{ao} - T_{ai}) =$

$$\frac{(T_b - T_{wi})}{2} \left[\frac{W_w}{W_a} \cdot \frac{\overline{C_{pw}}}{\overline{C_{pa}}} \left(\frac{h_A C_E}{h_E C_A} - 1 \right) + \frac{W_f}{W_a} \frac{\overline{C_{pf}}}{\overline{C_{pa}}} \left(1 - \frac{h_A C_E}{h_E C_A} \cdot \frac{W_w}{W_a} \frac{\overline{C_{pw}}}{\overline{C_{pa}}} \right) \right.$$

$$\left. - 2 \frac{W_f}{W_a} \cdot \frac{\overline{C_{pf}}}{\overline{C_{pa}}} \frac{(T_o - T_{wi})}{(T_b - T_{wi})} \right]$$

$$+ \frac{(T_b - T_{wi})}{2} \left\{ \left[\frac{W_w}{W_a} \frac{\overline{C_{pw}}}{\overline{C_{pa}}} \left(1 - \frac{h_A C_E}{h_E C_A} \right) + \frac{W_f}{W_a} \frac{\overline{C_{pf}}}{\overline{C_{pa}}} \left(\frac{h_A C_E}{h_E C_A} \cdot \frac{W_w}{W_a} \frac{\overline{C_{pw}}}{\overline{C_{pa}}} - 1 \right) \right]^2 \right.$$

$$\left. + 4 \frac{h_A C_E}{h_E C_A} \frac{W_f}{W_a} \frac{\overline{C_{pf}}}{\overline{C_{pa}}} \frac{W_w}{W_a} \frac{\overline{C_{pw}}}{\overline{C_{pa}}} \left[\frac{(T_{wi} - T_{ai})}{(T_b - T_{wi})} + \frac{W_f}{W_a} \frac{\overline{C_{pf}}}{\overline{C_{pa}}} \frac{(T_o - T_{wi})}{(T_b - T_{wi})} \right] \right\}^{1/2} . \tag{10}$$

Using

$C_E = \$3.50/ft^2$; $C_A = \$1.5/ft^2$;

$h_E = 20$ Btu/ft^2-hr-°F ; $h_A = 10$ Btu/ft^2-hr-°F ;

and the 600 MW atmospheric power plant design described in the Seventh Monthly Progress Report, we found the optimum air preheat temperature to be 433°F. With other design and operating conditions, the optimum air preheat temperature can be readily determined by Equation (10).

Curve 642086-B

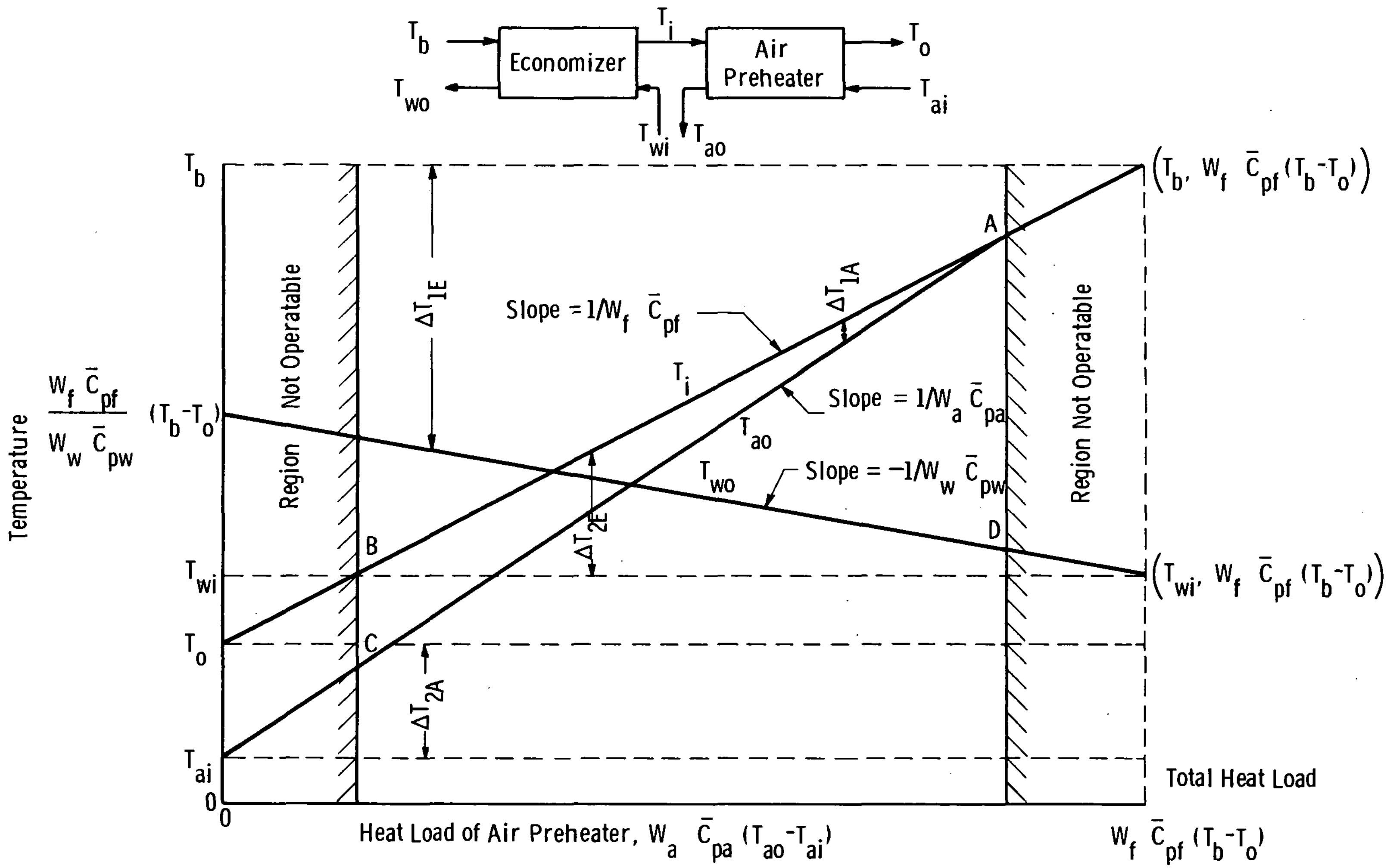

Fig. G-2 -Temperature difference & heat load distribution for air preheater & economizer

A graphic method can be used instead of the analytical method described above to find the optimum air preheat temperature. The total cost of the economizer, air preheater, and the additional heat transfer surface in the bed can be plotted against the air preheat temperature, T_{ao}; the air preheat temperature which gives the lowest total cost is the optimum one. To facilitate the calculation, the temperature change of T_i, T_{ao}, and T_{wo} were plotted at different air preheater heat loads in Figure G-2. The total heat transfer surface required for the economizer and air preheater can be readily calculated from the temperature differences shown where

$$\Delta T_{1A} = T_i - T_{ao}; \quad \Delta T_{2A} = T_o - T_{ai}$$

$$\Delta T_{1E} = T_b - T_{wo}; \quad \Delta T_{2E} = T_i - T_{wi}.$$

Physically impossible situations where $T_{wi} > T_i$ (to the left of point B in Figure G-2) and $T_{ao} > T_i$ (to the right of point A) are also shown in Figure G-2. The graph can easily be constructed for an actual case, for example the 600 MW atmospheric design outlined in the Seventh Monthly Progress Report, as shown in Figure G-3. The graphic method gives an optimum air preheat temperature of ~450°F (Figure G-4) compared to 433°F from the analytical method. The resulting cost of the air preheater is comparable to the cost cited for utility boilers in the Tenth Monthly Progress Report.

For the 300 MW atmospheric design presented in the 14th Monthly Progress Report, the optimum air preheat temperature was found to be 560°F, by the analytical method (Equation (10)).

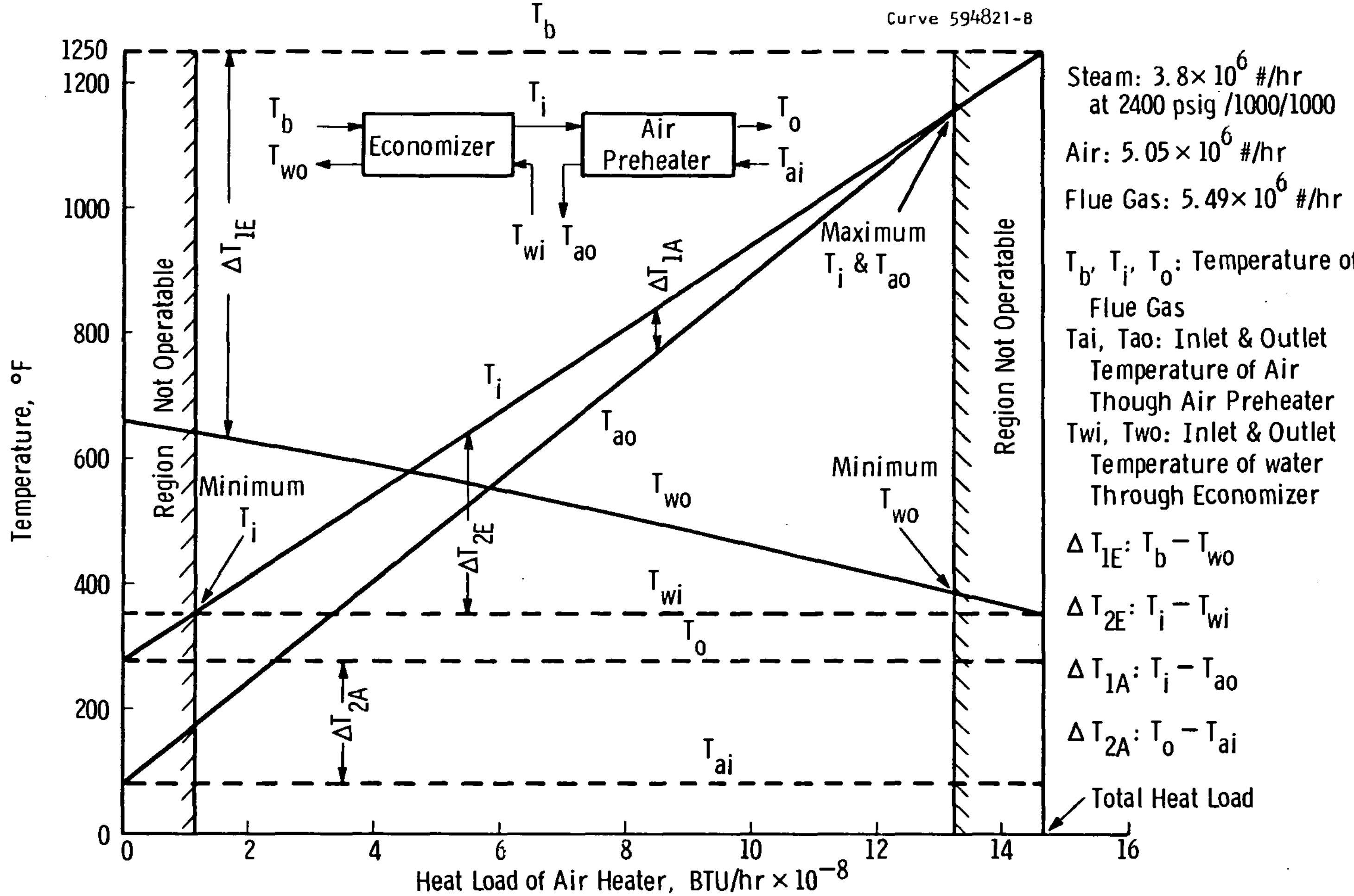

Fig. G-3—Temperature difference & heat load distribution for air preheater & economizer

Curve 642087-B

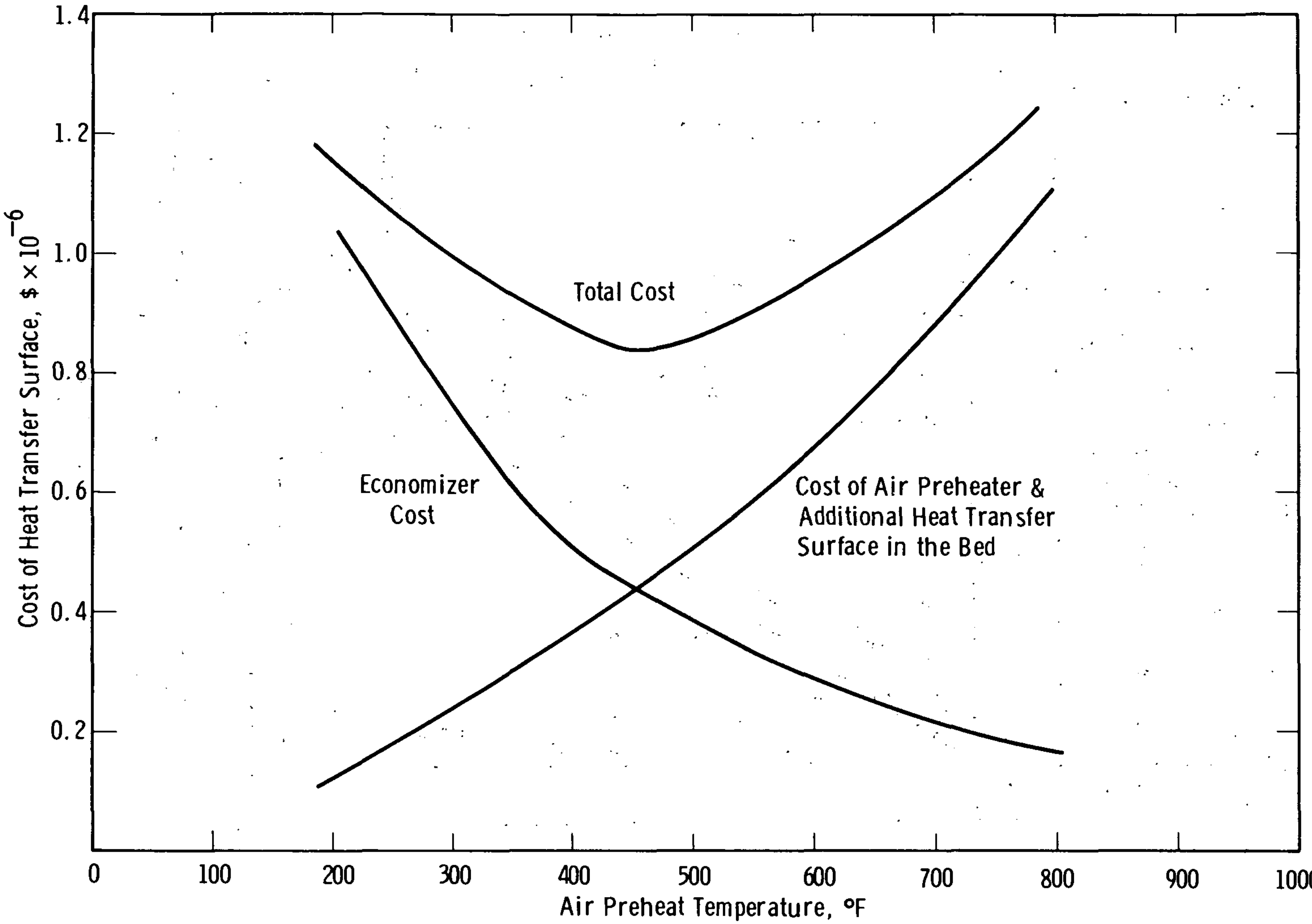

Fig. G-4—Determination of Optimum air preheat temperature

APPENDIX H

PRESSURIZED BOILER DESIGN REPORT

Prepared by Foster Wheeler Corporation,
John Blizard Research Center

Authors: R.W. Bryers
J.D. Shenker
R.J. Zoschak

ABSTRACT

A conceptual design was developed for a pressurized fluidized bed steam generator to be used in a 318 MW combined cycle utility power plant. The investigation includes a study of design and cost for the steam generator, coal handling and feeding equipment, and particulate removal equipment. The cost estimates are also extrapolated to a 636 MW capacity.

SUMMARY

The preferred concept for the pressurized steam generator is a once-through steam circuit with a four module boiler. The pressure vessels are vertically oriented with four stacked fluidized beds and one carbon burn-up cell in each module. The heat transfer surface is arranged such that each of the fluidized beds mainly provides the steam duty for only one steam generating function (i.e., superheating, reheating, or pre-evaporation). This arrangement simplifies operation at reduced loads and during start-up.

Wherever possible the benefits to fluidized bed combustion that result from a pressurized system are exploited. The ability of a pressurized system to operate economically with relatively large pressure drops was taken advantage of in the design of the fluidized beds. The relatively deep beds that are possible in the pressurized system greatly simplify the coal feeding system and allow a large amount of the heat transfer surface to be located in the fluidized beds where the heat transfer coefficient is high.

Load reductions are accomplished by lowering the temperature of the fluidized beds. Lowering the bed depth was also considered, but this type of system did not result in operational benefits and required more complex equipment. Fuel/air ratios at the various loads are determined by gas turbine inlet requirements and this results in high excess air values at reduced loads. The excess air values, however, do have the beneficial effect of maintaining

the design efficiency of the particulate removal equipment over a wide range of operation.

The mechanical design of the steam generator provides for maximum use of existing shop technology. The steam pressure parts are all of standard design, and the arrangement of the pressure parts, although different from conventional units, does not present any major difficulties. The pressure vessel is used as a support system for the steam pressure parts besides serving as a vessel to contain the gas pressure. The design of the combustion air and flue gas circuits is such that the pressure vessels is cooled by the combustion air and does not require internal insulation.

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1.0 INTRODUCTION

As subcontractor to Westinghouse on Public Health Service Contract CPA 70-9 to evaluate the fluidized bed combustion process, it was Foster Wheeler's responsibility to develop conceptual designs of two utility boilers at the 300 and 600 MW level according to specifications provided by Westinghouse. At the onset of the program it was decided that development of a pressurized boiler as part of a combined cycle and an atmospheric boiler at the 300 MW level would probably be the most worthwhile approach. As work developed and a better feel for the boiler market was observed, it was found necessary to extrapolate the data to 600 MW to provide meaningful results.

This section of the report which appears in two volumes discusses the pressurized boiler design and cost estimated at the 318 MW level. The data is extrapolated to 636 MW to make meaningful comparisons with conventional equipment. The nominal plant sizes of 300 MW and 600 MW are used in the text. The report includes a presentation of specifications provided by Westinghouse resulting from a state-of-the-art review and discussion of boiler design, performance, operation, cost and trade-offs in design where they apply. Auxiliary equipment such as coal handling, coal feeding, and particulate removal were developed and cost estimated through the courtesy and cooperational efforts of numerous equipment manufacturers. The results are also presented.

Concepts investigated and discarded are briefly reviewed and recommendations for future consideration are made.

2. SPECIFICATIONS

The plant specifications were supplied by the Westinghouse Electric Corporation.

2.1 Fuel Specifications

The fuel selected for the fluidized bed steam generator is Ohio Pittsburg No. 8 Seam Coal having the specifications listed in Table 2.1. This coal will be received at the plant by unit train. The size of the coal as received is 1-1/2" x 0. This sizing is specified due to coal flow dryer requirements, but no cost penalty is paid for this sizing. The size distributions of the received coal and the coal fed to the steam generators are shown in Figure 2.1.

2.2 Fluidized Bed Boiler Specifications

2.2.1 Bed Characteristics The range of allowable superficial velocities can be seen in Figure 2.2. The allowable superficial velocities are based on the average particle size in the beds. Table 2.2 lists the bed temperature, gas side heat transfer coefficients, and the combustion efficiencies for the primary beds and the carbon burn-up cell.

2.2.2 Sorbent Specifications The sorbent is BCR 1337 dolomite. The chemical composition and flow rates for the dolomite are listed in Table 2.3. The recycled sorbent ranges in size from 500 to 5000 microns with an average size of 2500 microns. The make-up sorbent is -1/4".

2.3 System Specifications

2.3.1 Power Cycle A schematic diagram of the combined cycle plant is shown in Figure 2.3. Also indicated on this drawing are the parts of the cycle that Foster Wheeler Corporation investigated.

TABLE 2.1

OHIO PITTSBURGH NO. 8 SEAM COAL

(Source of data: USBM, Pittsburgh, Pa.)

SAMPLE: Run of Mine -

PROXIMATE ANALYSIS (wt %):

Moisture	3.3 *
Volatile Matter	39.5
Fixed Carbon	48.7
Ash	8.5
	100.0

ULTIMATE ANALYSIS (wt %):
(includes moisture)

C	71.2	
H	5.4	
O	9.3	
N	1.3	
S	4.3	(~60% organic 40% pyritic)
Ash	8.5	
	100.0	

GROSS HEATING VALUE: 13000 Btu/lb

NET HEATING VALUE: 12500 Btu/lb

ASH ANALYSIS (wt %):

SiO_2	45.3
Al_2O_3	21.2
Fe_2O_3	27.3
TiO_2	1.0
P_2O_5	0.11
CaO	1.9
MgO	0.6
Na_2O	0.2
K_2O	1.8
SO_3	0.7
	100.1

FUSIBILITY OF ASH:

Initial Deformation Temperature	-- 2080°F
Softening Temperature	-- 2230°F
Fluid Temperature	-- 2420°F

PARTICLE DENSITY: Coal -- ~1.4 gm/cc
Ash -- ~2.8 gm/cc

GRINDABILITY (Hardgrove): 50-60

FREE SWELLING INDEX: 5-5.5

COST: $5.50-5.75/ton for ~4% S coal, 13,000 Btu/lb
($6.00/ton cost projected by end of 1970)

*Possible Pick-up in Storage and Handling 6.7%
Giving Maximum Total Moisture as Received of 10%

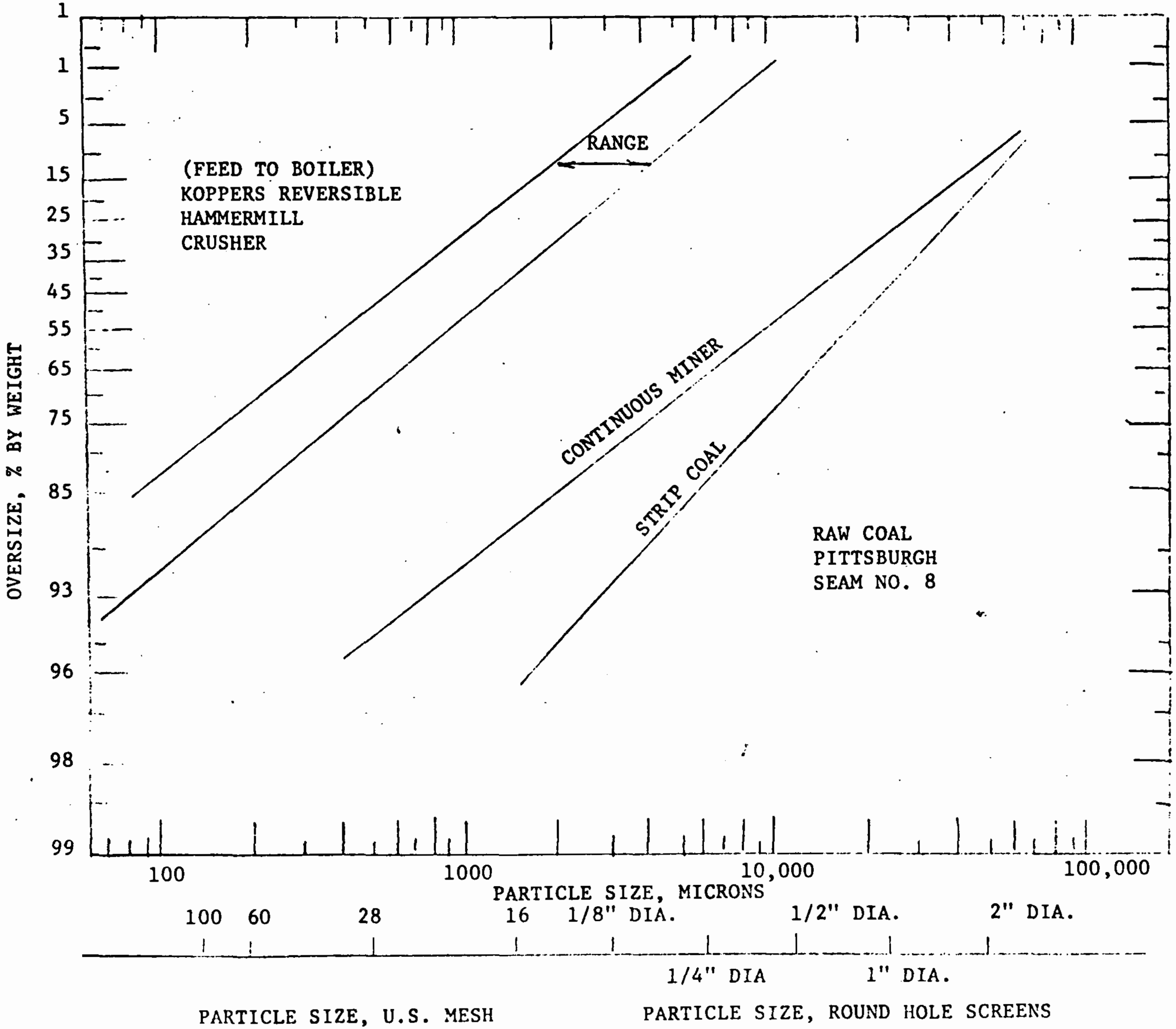
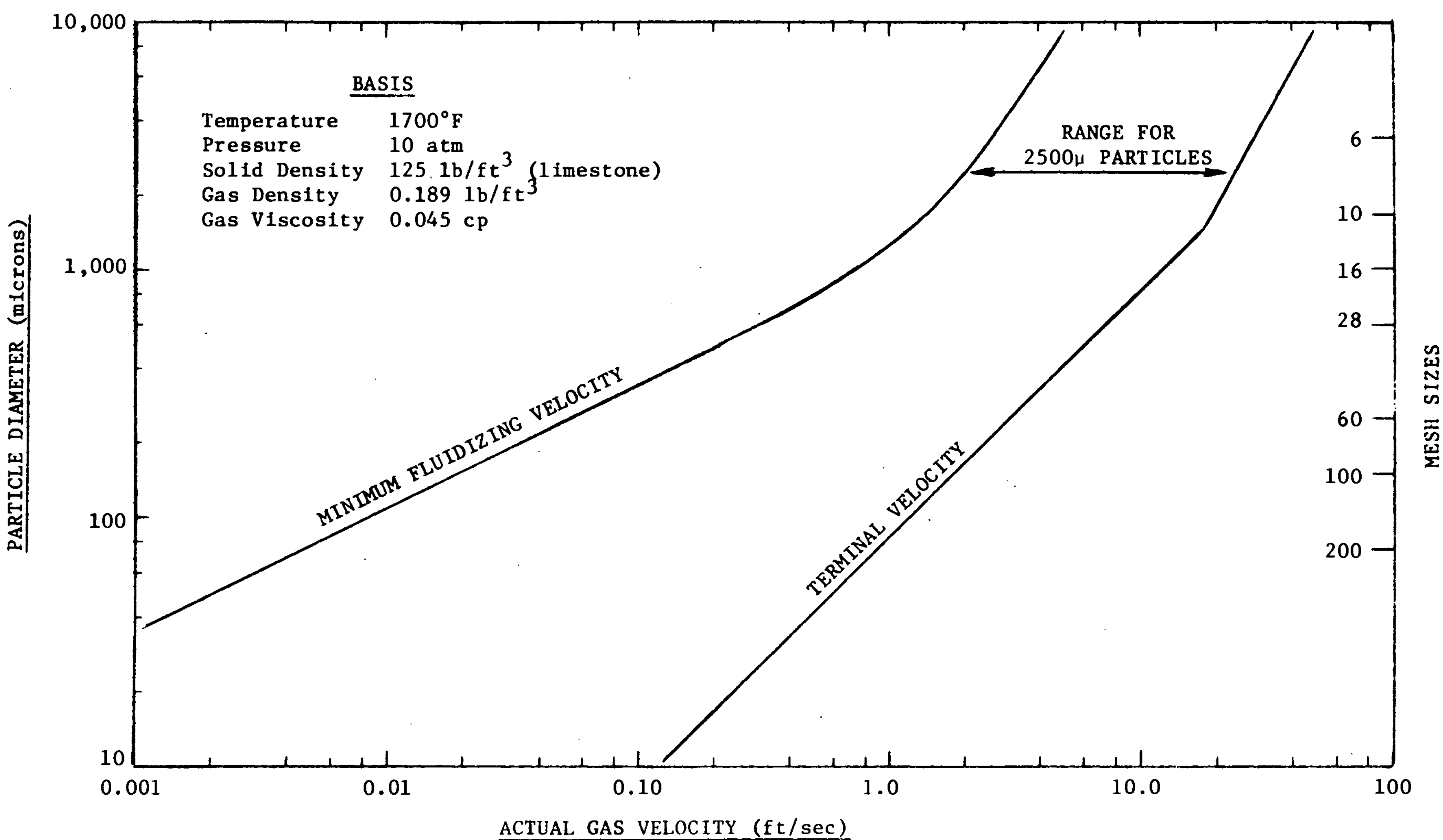


FIGURE 2.1 DISTRIBUTION OF COAL SIZE AS RECEIVED AND AFTER CRUSHING



BASIS
Temperature 1700°F
Pressure 10 atm
Solid Density 125 lb/ft3 (limestone)
Gas Density 0.189 lb/ft3
Gas Viscosity 0.045 cp
RANGE FOR
2500μ PARTICLES
MINIMUM FLUIDIZING VELOCITY
TERMINAL VELOCITY
PARTICLE DIAMETER (microns)
10,000
1,000
100
10
0.001
0.01
0.10
1.0
10.0
100
ACTUAL GAS VELOCITY (ft/sec)
MESH SIZES
6
10
16
28
60
100
200

FIGURE 2.2

TABLE 2.2

BED PARAMETERS AT DESIGN CONDITIONS

PRIMARY BEDS

Temperature -	1750°F
Gas Side Heat Transfer Coefficient in Bed -	50 Btu/hr-ft^2-°F
Gas Side Heat Transfer Coefficient in Transition Zone -	40 Btu/hr-ft^2-°F
Elutriated Carbon -	6%
CO -	<0.5% in flue gas
Excess Air -	10%
Superficial Velocity -	5.6 - 9.1 ft/sec

CARBON BURN-UP CELL

Temperature -	2000°F
Gas Side Heat Transfer Coefficient in Bed -	50 Btu/hr-ft^2-°F
Gas Side Heat Transfer Coefficient in Transition Zone -	40 Btu/hr-ft^2-°F
Combustion Efficiency -	90%
Excess Air -	79%
Superficial Velocity -	5.7 ft/sec

TABLE 2.3

ABSORBENT SPECIFICATIONS

BCR 1337 Dolomite

Component	% wt As Received
SiO_2	0.78
Al_2O_3	0.15
Fe_2O_3	0.25
MgO	45.0
CaO	53.
TiO_2	0.02
SrO	<0.03
Na_2O	<0.02
K_2O	<0.1
MnO_2	<0.03

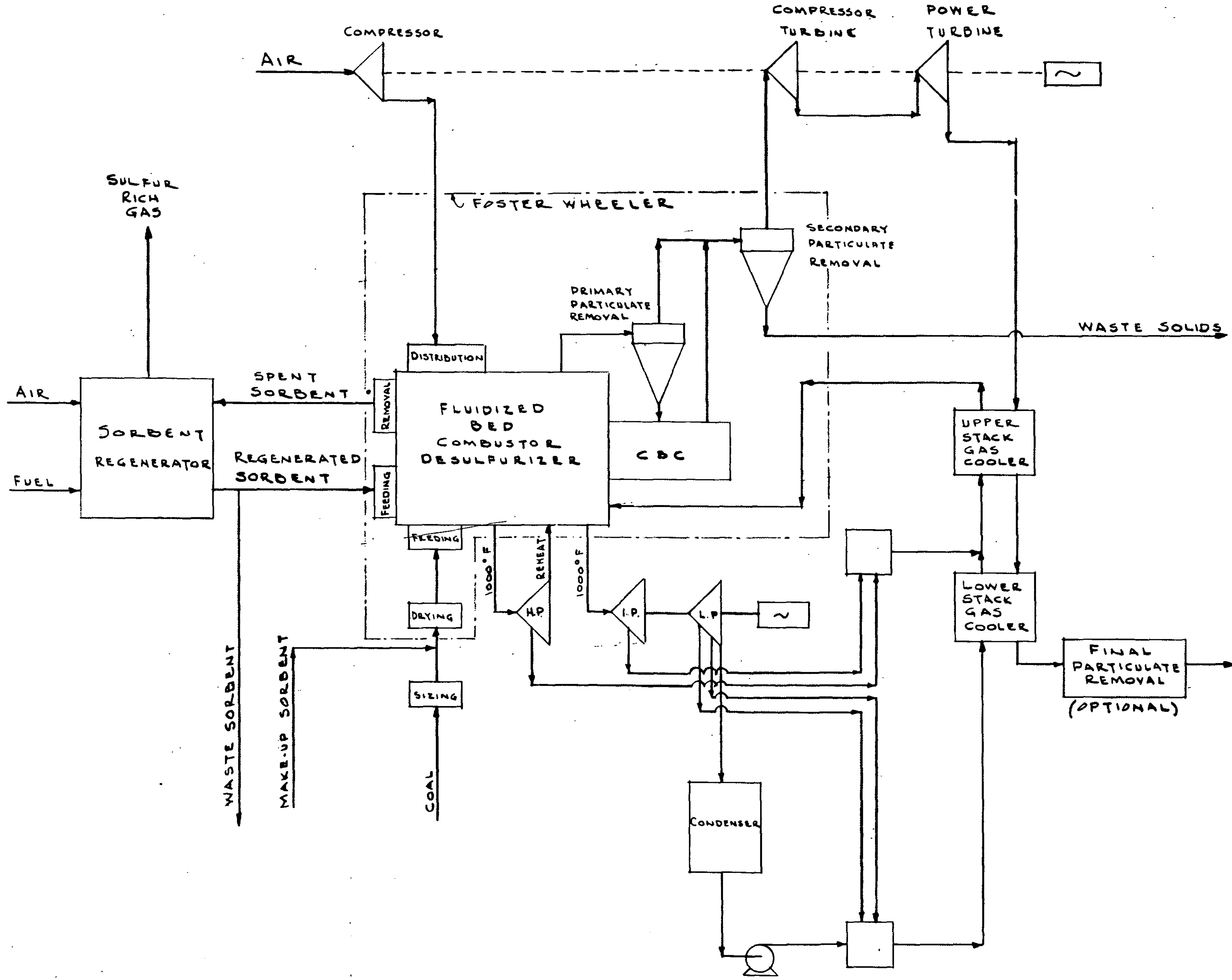


FIG. 2.3

SCHEMATIC DIAGRAM
PLANT POWER CYCLE

DRAWN BY:	JR	5/10/71	SCALE: — =1'–0"
CHECKED BY:			RD-713-115A
APPROVED BY:			

REVISION

ORDER NO. O-59-1192

This report deals primarily with the equipment in these areas.

2.3.2 Plant Capacity The plant capacity is tabulated as a function of gas turbine inlet temperature and steam cycle load fraction in Table 2.4.

2.3.3 Cycle Conditions The steam cycle conditions are tabulated in Table 2.5 as a function of steam cycle load fraction. The only parameter in this table that is also a function of gas turbine inlet temperature is the feedwater inlet temperature. The values of this parameter listed in the table correspond to the expected gas turbine inlet temperatures at the specified steam load fractions.

The gas cycle conditions are listed in Table 2.6 as a function of steam cycle load fraction and gas turbine inlet temperature.

2.3.4 Particulate Removal Requirements The first cyclone stage is required to collect 90% by weight of the particulate in the gas from the primary beds. The expected size distribution can be seen in Figure 2.4. The second stage cyclone system is required to collect 97% by weight of the particulate of the combined gas stream. This gas stream consists of the gas cleaned by the first cyclone stage and the gas from the carbon burn-up cell. It is also required of the second cyclone stage that all particles greater than five microns be collected.

2.3.5 Sulfur Removal Requirements It is specified that the flue gas contain less than 5% of the sulfur in the coal.

2.3.6 Turn Down Requirements It is required that the combined cycle plant have a turn down ratio of four-to-one. It is also required that the response rate to load changes be at least 5% of load per minute.

TABLE 2.4

PLANT CAPACITY [1]

300 MW COMBINED CYCLE

Steam Cycle Load Fraction (Ref.)	1.02	1.00	0.8	0.6	0.4
Turbine Inlet Temp. - °F	1600				
Plant Power - MW	317.7	313.7	271.7	223.1	162.8
Steam Power - MW	269.2	265.3	223.5	175.1	114.7
Plant Heat Rate - Btu/kw-hr	8974	8967	8869	8846	8952
Fuel-Air Ratio	0.0919	0.0907	0.0777	0.0636	0.0470
Turbine Inlet Temp. - °F	1500				
Plant Power - MW	309.8	305.8	263.7	215.3	155.8
Steam Power - MW	268.8	264.8	222.9	174.4	114.7
Plant Heat Rate - Btu/kw-hr	9091	9080	8999	8998	9165
Fuel-Air Ratio	0.0908	0.0895	0.0765	0.0625	0.0460
Turbine Inlet Temp. - °F	1400				
Plant Power - MW	301.8	297.8	255.8	207.5	148.7
Steam Power - MW	268.1	264.1	222.1	173.6	114.7
Plant Heat Rate - Btu/kw-hr	9211	9202	9134	9161	9392
Fuel-Air Ratio	0.0896	0.0884	0.0753	0.0613	0.0451

[1] Reference conditions for steam cycle are from Hammond Station, Unit #4, Georgia Power and Light Company.

TABLE 2.5

STEAM CYCLE CONDITIONS [1]

Steam Cycle Load Fraction (Ref.)	1.02	1.00	0.8	0.6	0.4
Inlet Feedwater Temp. - °F	578	576	554	540	536
Main Steam Outlet Temp. - °F	1,000	1,000	1,000	1,000	1,000
Main Steam Outlet Press. - psig	2,500	2,486	2,454	2,434	2,410
Main Steam Flow - M lb/hr	1,727	1,693	1,354	1,016	677
Reheat Inlet Temp. - °F	650	632	592	560	530
Reheat Inlet Press. - psig	600	587	465	345	223
Reheat Outlet Temp. - °F	1,000	1,000	1,000	1,000	1,000
Reheat Outlet Press. - psig	580	568	450	333	217
Reheat Flow - M lb/hr	1,644	1,614	1,308	998	677

[1] Reference conditions are from Hammond Station, Unit #4, Georgia Power and Light Company.

TABLE 2.6

GAS CYCLE CONDITIONS

Steam Load Fraction (Ref.)	1.02	1.00	0.8	0.6	0.4
Turbine Inlet Temp. - °F	1600				
Air Compressor Outlet Temp. - °F	636	636	632	628	623
Air Flow* - lb/sec	650	650	650	650	650
Fuel-Air Ratio	0.0919	0.0907	0.0776	0.0636	0.0470
Heat to Gas Turbine - M Btu/hr	1,423	1,420	1,385	1,348	1,304
Turbine Inlet Temp. - °F	1500				
Air Compressor Outlet Temp. - °F	628	628	624	620	615
Air Flow* - lb/sec	650	650	650	650	650
Fuel-Air Ratio	0.0908	0.0895	0.0765	0.0624	0.0461
Heat to Gas Turbine - M Btu/hr	1,284	1,281	1,249	1,215	1,175
Turbine Inlet Temp. - °F	1400				
Air Compressor Outlet Temp. - °F	620	619	616	611	608
Air Flow* - lb/sec	650	650	650	650	650
Fuel-Air Ratio	0.0896	0.0884	0.0753	0.0613	0.0451
Heat to Gas Turbine - M Btu/hr	1,148	1,145	1,116	1,084	1,049

* Value given is gas turbine compressor air flow. Five percent of compressor air flow is used for turbine blade cooling and by-passes boiler.

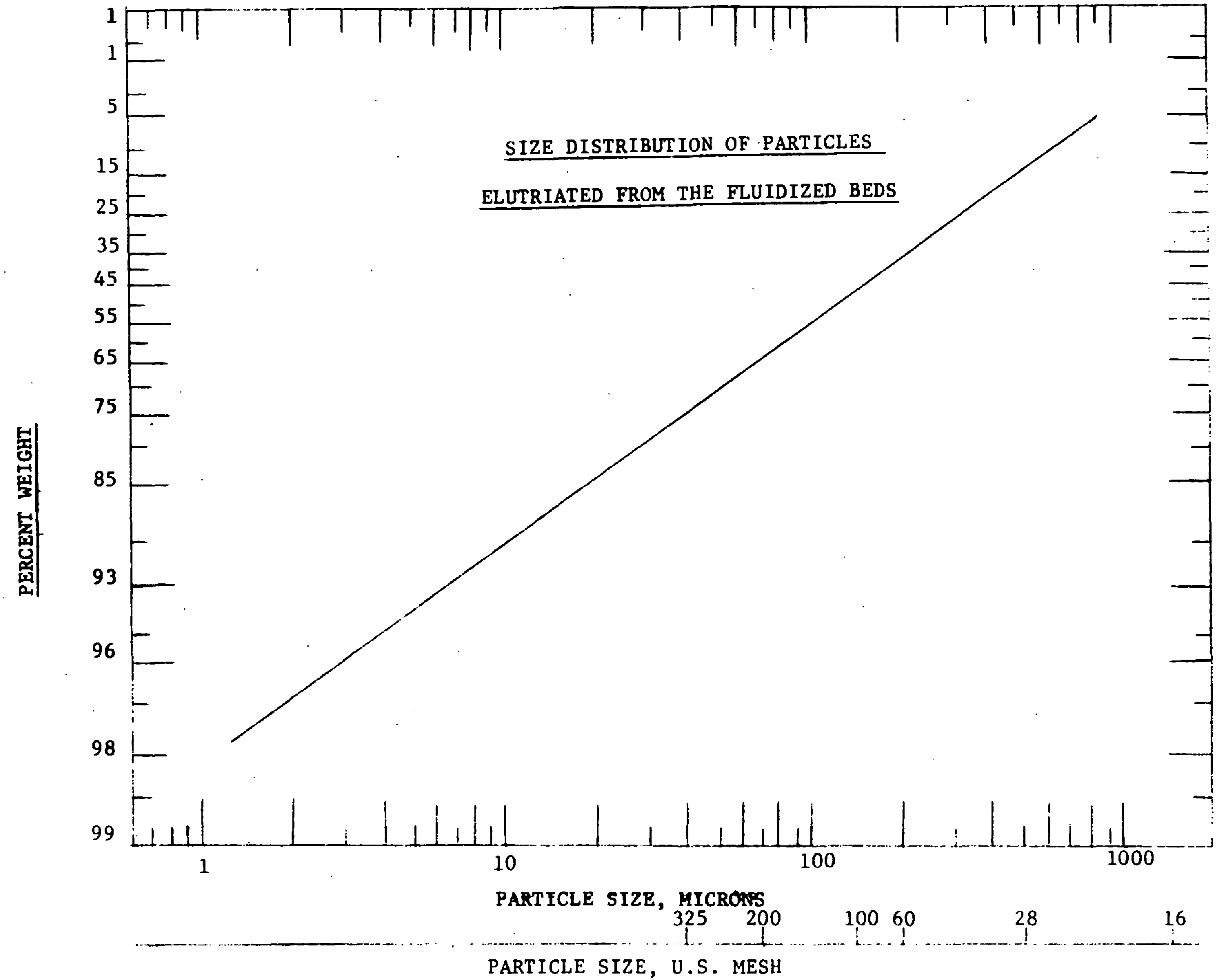

SIZE DISTRIBUTION OF PARTICLES
ELUTRIATED FROM THE FLUIDIZED BEDS
PERCENT WEIGHT
1
1
5
15
25
35
45
55
65
75
85
93
96
98
99
1
10
100
1000
PARTICLE SIZE, MICRONS
325
200
100
60
28
16
PARTICLE SIZE, U.S. MESH

FIGURE 2.4

3. BOILER DESIGN CONCEPTS

3.1 Selection of System Parameters

There are two general areas that have to be investigated to arrive at a design for the 300 MW, pressurized steam generator. The first area to be considered is the selection of cycle conditions. The determination of steam cycle parameters and gas pressure are the major categories in this area of design. The choice of steam cycle is basically between the subcritical and supercritical cycles. An investigation of steam generator capital costs indicates that the subcritical boiler is more economical for all size once-through steam generators. This is mainly due to the fact that subcritical steam conditions result in thinner pressure part walls and lower tube metal temperatures. Also natural circulation steam generators are more economical then supercritical once-through steam generators up to 700 MW. Thermodynamic and cost analyses of 10 atmosphere combined cycle plants with 1600°F gas turbine inlet temperature done by Westinghouse have also shown that there is no advantage in using supercritical steam conditions at 1000°F steam temperature. For these reasons subcritical steam conditions were used for the detailed design.

Since a subcritical cycle was chosen, it was necessary to determine whether a natural circulation or a once-through steam generator circuit should be used. Although the steam conditions specified in Section 2 are often used with natural circulation systems in conventional steam generators, the physical restrictions of a pressurized, fluidized bed design make this system impractical. These physical restrictions mainly affect the location of evaporator surface and the steam drum. The once-through system, on the other hand, allows more freedom in the arrangement of the evaporating tube surface and does not require a steam drum. Therefore, it was decided that a once-through system would be better for the pressurized steam generator application.

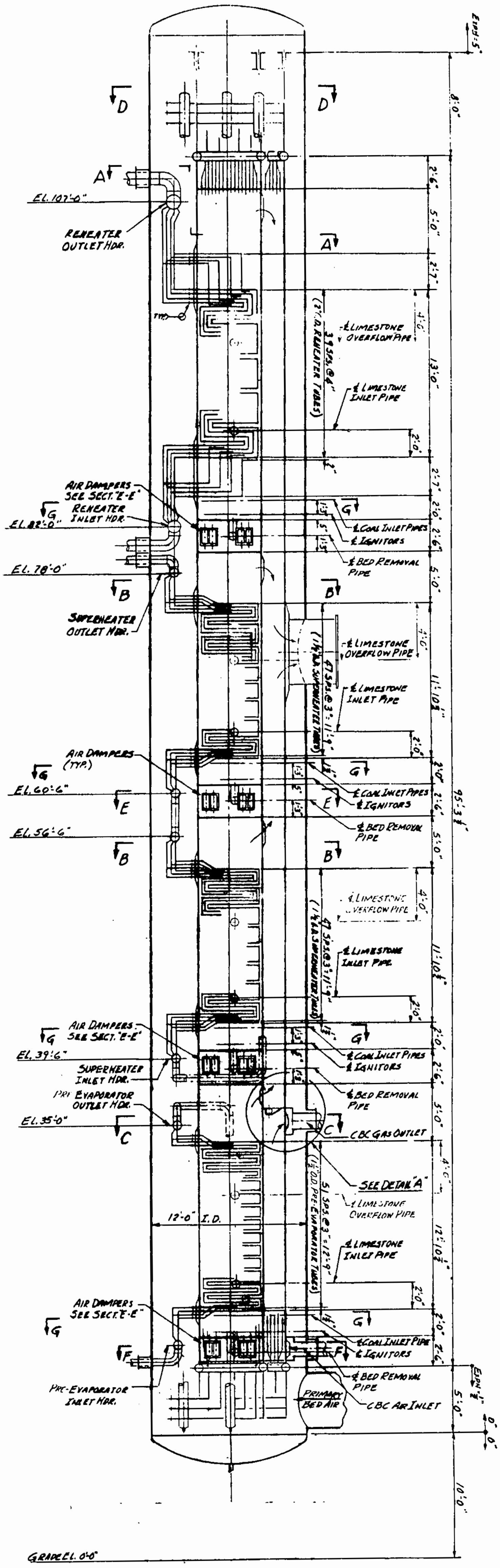
The most important gas side parameter to be determined is the gas pressure. This parameter primarily affects the pressure vessel cost, but it also influences the selection of the number of boiler modules. An analysis was performed on the effects of gas pressure on pressure vessel costs. This analysis showed that although increasing the system pressure does decrease the required vessel diameter for a given gas flow rate, this effect becomes less significant as the pressure level increases. Above 10 atmospheres it appeared that very little cost saving would be realized by increasing the pressure level. It was also anticipated that the cost of auxiliary equipment such as coal and sorbent feeding equipment would be considerably greater at higher pressures. Consideration of these factors resulted in the choice of a 10 atmosphere gas pressure. The selection of system parameters is discussed in more detail in Appendix 1.

3.2 Selection of Steam Generator Design Concept

Once the steam cycle conditions were selected and the gas pressure was determined, the second area of design had to be investigated. This area of design pertains to the layout of the steam generator. The arrangement of steam pressure parts, the size and shape of the fluidized beds, the arrangement of the beds in the vessel, and the orientation of the vessel are all aspects of the design that have to be considered. These design areas are all interrelated, and affect many aspects of the process. For example, the selection of bed shape can place restrictions on the arrangement of steam pressure parts. If beds of annular cross section are chosen, it becomes impossible to use horizontal, serpentine tube banks in the beds. Besides affecting the steam flow, the type of tube bank in the bed affects the fluidization parameters of the bed such as temperature distribution and particle mixing. From this brief example the complexity of choosing an optimum design can be seen. Also due to the state-of-the-art of fluidized bed design, many of the decisions have to be based on engineering judgment rather than firm experimental data. A detailed discussion of how the final steam generator design was selected is presented in Appendix 1.

The final design is shown in Figures 3.1, 3.2, 3.3 and 3.4. Four of these modules are used for a 300 MW combined cycle plant. This design consists of a vertically oriented pressure vessel with four, stacked primary beds and one carbon burn-up cell. The fluidized bed cells are rectangular in cross section and are about 20 ft. high. The actual bed depths are around 12 ft. in the fluidized state. The 12 ft. beds take maximum advantage of the high heat transfer coefficient in the fluidized beds without causing unrealistic superficial velocities or gas pressure drops. Horizontal, serpentine tube banks are used in the beds. It is felt that this arrangement will give even fluidization of the beds. This type of tube bank design also allows for conventional fabrication techniques and is adaptable to a wide range of tube spacings.

At first glance it may appear that using beds of rectangular cross section in a cylindrical vessel is an inefficient use of space. It can be seen in Figures 3.1, 3.2 and 3.3, however, that the void areas between the pressure vessel shell and the finned tube welded wall enclosure are used quite extensively for transport piping and accessory equipment. Also, the flat enclosure wall panels used to form the beds have the advantage of being fabricated with automatic equipment. It is not certain that other wall shapes will be adaptable to fabrication with existing automatic equipment.

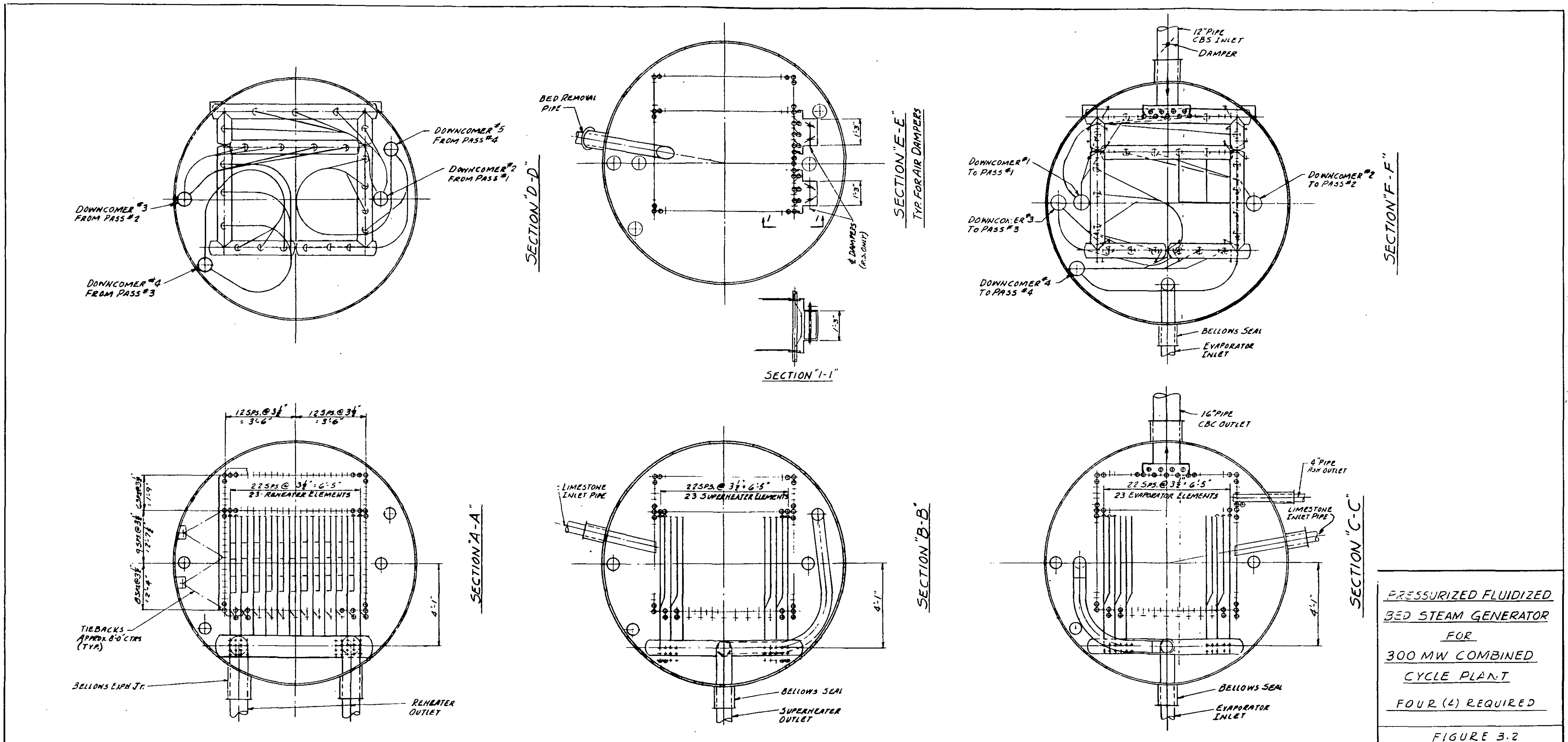


ELEVATION

PRESSURIZED FLUIDIZED BED STEAM GENERATOR FOR 300 MW COMBINED CYCLE PLANT

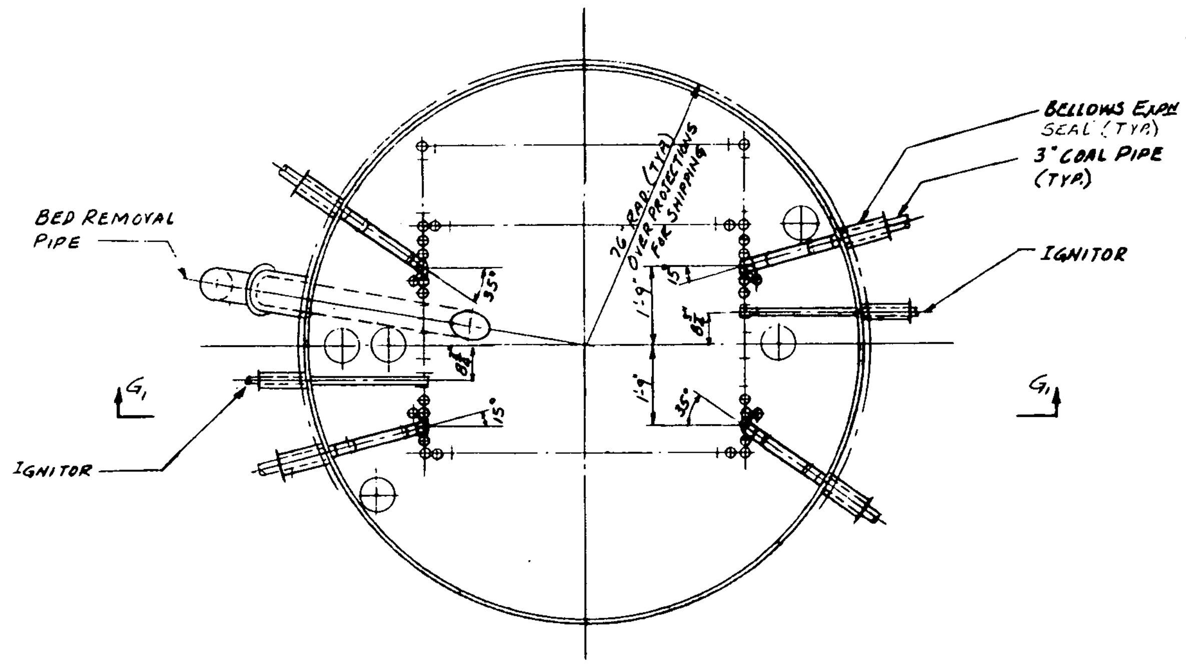
FOUR (4) REQUIRED

FIGURE 3.1

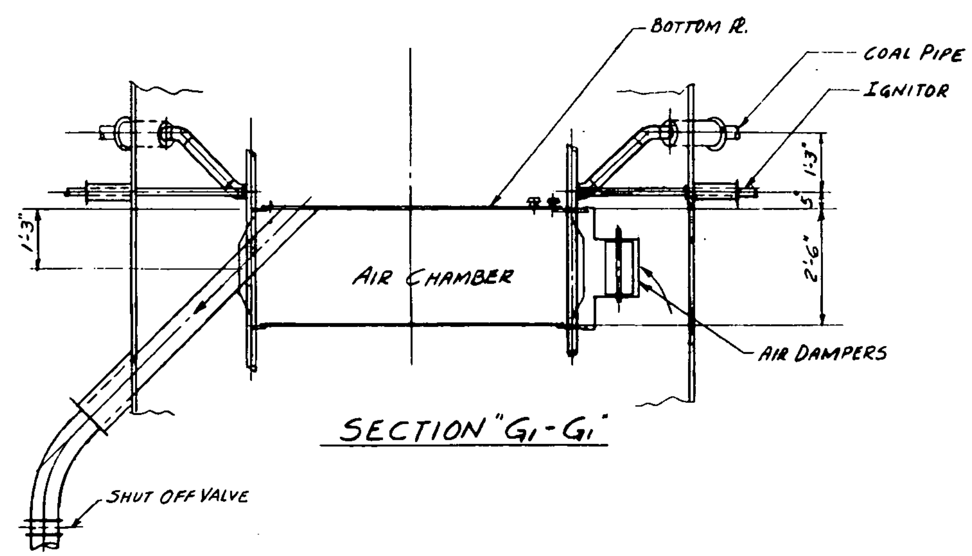


BED REMOVAL PIPE
DOWNCOMER #5 FROM PASS #4
DOWNCOMER #2 FROM PASS #1
DOWNCOMER #3 FROM PASS #2
DOWNCOMER #4 FROM PASS #3
SECTION "D-D"
SECTION "E-E" TYP. FOR AIR DAMPERS
℄ DAMPERS (P.J. ONLY)
SECTION "1-1"
12" PIPE CBS INLET
DAMPER
DOWNCOMER #1 TO PASS #1
DOWNCOMER #3 TO PASS #3
DOWNCOMER #4 TO PASS #4
DOWNCOMER #2 TO PASS #2
BELLOWS SEAL
EVAPORATOR INLET
SECTION "F-F"
12 SPS.@ 3½" = 3'-6"
22 SPS.@ 3½" = 6'-5"
23 REHEATER ELEMENTS
TIEBACKS APPROX. 8'-0" CTRS (TYP.)
BELLOWS EXP'N JT.
REHEATER OUTLET
SECTION "A-A"
LIMESTONE INLET PIPE
23 SUPERHEATER ELEMENTS
BELLOWS SEAL
SUPERHEATER OUTLET
SECTION "B-B"
16" PIPE CBC OUTLET
4" PIPE ASH OUTLET
23 EVAPORATOR ELEMENTS
LIMESTONE INLET PIPE
EVAPORATOR INLET
SECTION "C-C"
4'-1"
PRESSURIZED FLUIDIZED BED STEAM GENERATOR FOR 300 MW COMBINED CYCLE PLANT
FOUR (4) REQUIRED

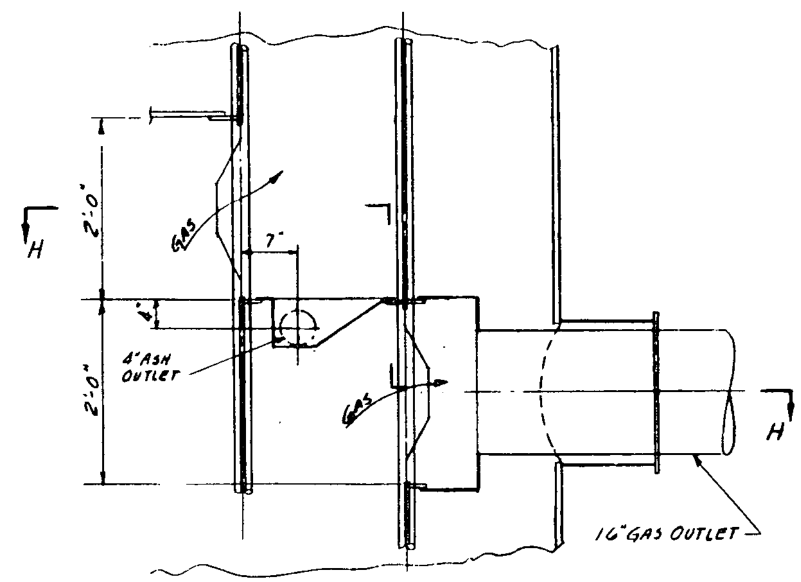
FIGURE 3.2



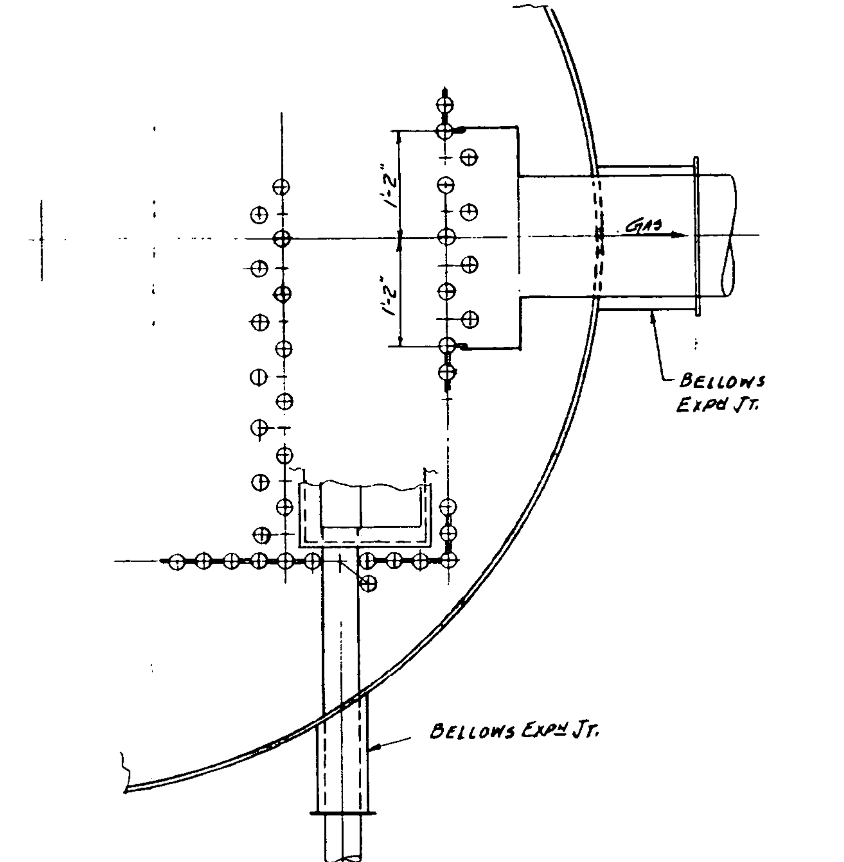
SECTION "G-G"



SECTION "G1-G1"



DETAIL "A"

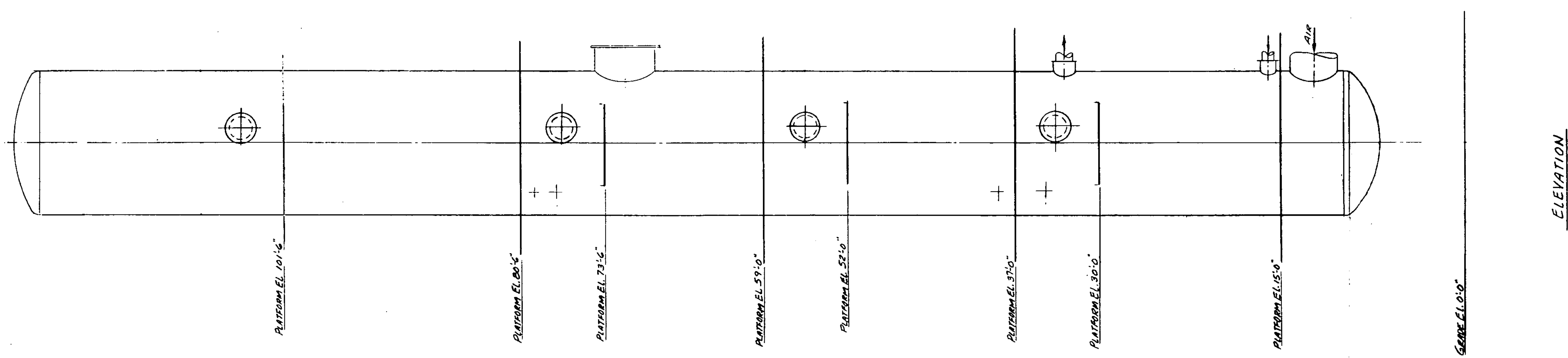


SECTION "H-H"

PRESSURIZED FLUIDIZED
BED STEAM GENERATOR
FOR
300 MW COMBINED
CYCLE PLANT

FOUR (4) REQUIRED

FIGURE 3.3



PRESSURIZED FLUIDIZED BED STEAM GENERATOR FOR 300 MW COMBINED CYCLE PLANT

FOUR (4) REQUIRED

FIGURE 3.4

4. DETAILED BOILER DESIGN

4.1 Circulation through the Boiler

Detailed drawings of the boiler are shown in Figures 3.1, 3.2, 3.3 and 3.4. Simplified drawings showing the steam side and gas side flow circuits are shown in Figure 4.1. The steam circuits and gas circuits are really an integrated whole as far as the design and operation of the steam generator is concerned, but they will be discussed separately here for the sake of clarity.

The steam circuits are of once-through design. The location of the tube surface is different from that of a conventional once-through unit, but the concepts and steam side design criteria used in this steam generator are typical of conventional units. In the design presented in this report feedwater enters the steam generator near the bottom of the module. The feedwater then flows upward through the lowest tube bank. This tube bank has been labeled a pre-evaporator. Most of the bank contains sub-cooled water, but steam starts to form in the uppermost loops of the bank. At the top of the pre-evaporator tube bank the water steam mixture is collected in a header and then flows through a transfer pipe to one of the lower wall headers.

The tube walls are divided into four separate circuits, and the water steam mixture flows through each of the circuits in succession. After each pass through a wall circuit, the mixture is transported to the bottom of the next circuit through unheated downcomers. This results in upward flow in all heated circuits that contain a water-steam mixture. Upward flow is highly desirable,

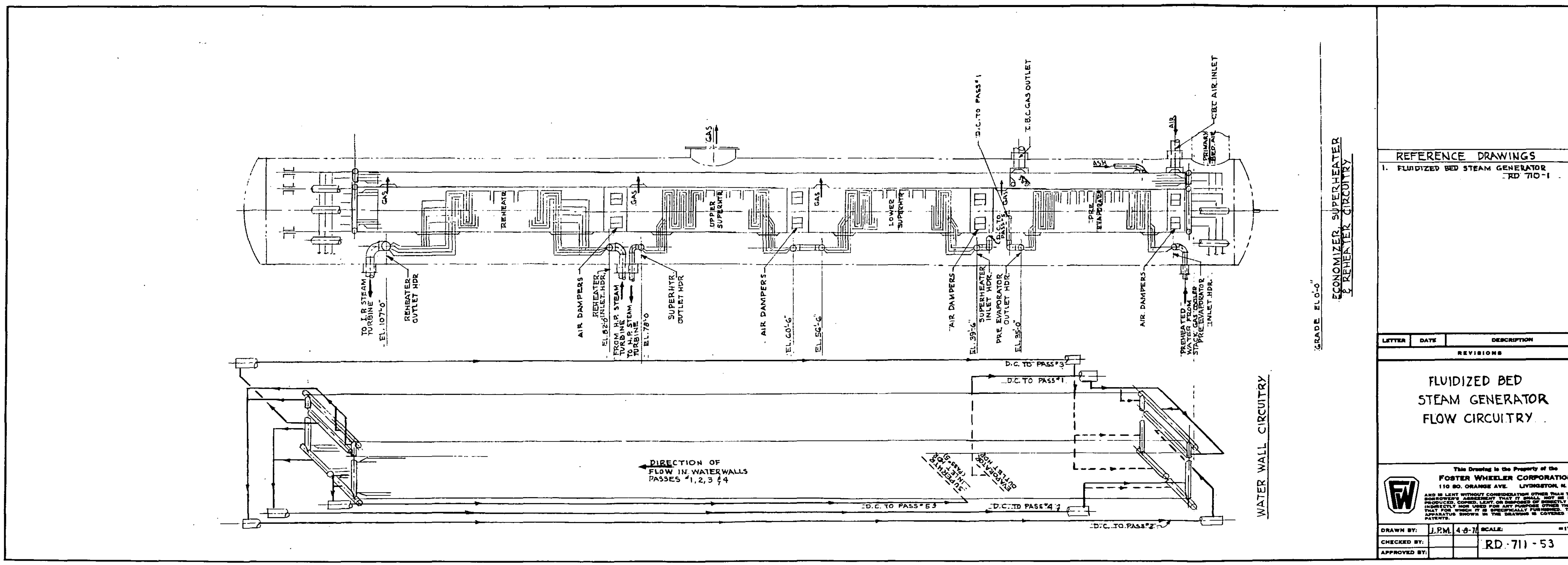

REFERENCE DRAWINGS
1. FLUIDIZED BED STEAM GENERATOR RD 710-1
FLUIDIZED BED STEAM GENERATOR FLOW CIRCUITRY
FOSTER WHEELER CORPORATION
110 SO. ORANGE AVE. LIVINGSTON, N. J.
DRAWN BY: J.P.M.
RD-711-53
ECONOMIZER, SUPERHEATER & REHEATER CIRCUITRY
WATER WALL CIRCUITRY
DIRECTION OF FLOW IN WATERWALLS PASSES #1, 2, 3 & 4
REHEATER
UPPER SUPERHTR
LOWER SUPERHTR
PRE EVAPORATOR
AIR DAMPERS
GAS
GRADE EL 0'-0"

Fig. 4.1

because it prevents bouyant forces from separating the water and steam.

After passing through the last wall circuit, the now almost saturated steam enters the first superheater bank. The superheater duty is accomplished in the two series connected superheater banks. The steam then leaves the vessel and is transported to the high pressure turbine. Steam from the discharge of the high pressure turbine returns to the boiler module at the bottom of the top tube bank to be reheated. The steam leaving this bank is at the final reheat conditions and is fed to the intermediate pressure turbine after being combined with the reheat steam from the other modules.

The steam side circulation is more complex during boiler start-up than in the description presented above. Under the conditions of start-up, the steam side circulation is strongly interrelated with gas side ignition problems and plant steam cycle requirements. This type of steam generator operation is discussed in Section 5 of the report.

The air and gas flow system can be seen in Figure 4.1. The combustion air for the primary beds enters the pressure vessel and fills in the void volume between the steam generator tube walls and the pressure vessel shell. This volume acts as a manifold to distribute the air among the four primary fluidized beds. The flow of air from the void volume to the air plenum chambers located under each of the beds is regulated by control dampers. The air then flows from the plenum chambers through distribution plates to the fluidized bed cells.

In the four cells coal is combusted in a bed of fluidized dolomite. The tube surface in the fluidized beds and the transition zones absorbs approximately 80% of the steam cycle duty. This situation makes efficient use of the high bed-to-tube heat transfer coefficient that is expected in the fluidized bed. The dolomite particles in the bed react with oxides of sulfur formed during the combustion of the coal and remove this pollutant from the gas stream. The actual fluidized bed extends to 2 ft. below the top of the tube bank. The top 2 ft. of the tube bank and the tube free space above the bed serve as a disengaging zone for particles elutriated from the bed.

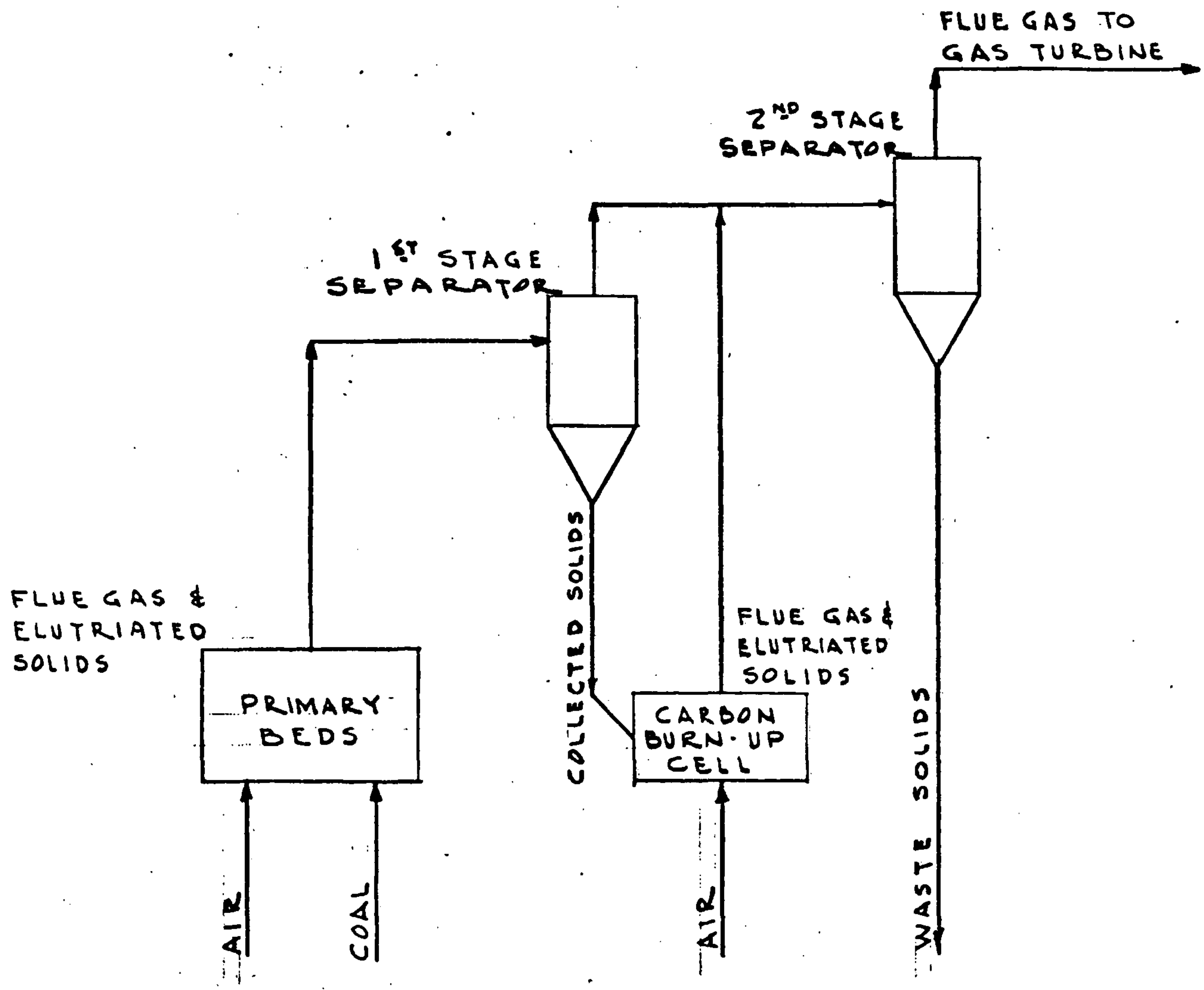
The flue gases leave the primary fluidized bed cells through tube screens that are located near the top of the cells. Tube screens are openings in the enclosure walls that are formed by offsetting some of the tubes in the wall. The gases then combine in a gas passage that, like the cells, is formed by tube walls. Convection heat transfer between the flue gases and the tube walls takes place in this passage. This zone is the last heat transfer surface in the gas circuit. Near the middle of the gas passage an insulated duct carries the gases from this passage, through the shell wall and to the first cyclone stage.

The carbon burn-up cell has an air-gas circuit that is separate from that of the primary fluidized beds. Combustion air for the carbon burn-up cell enters the pressure vessel through a duct located above the primary bed air inlet duct. This duct connects directly to the air plenum chamber under the carbon burn-up cell. The flue gases from the carbon burn-up cell are also segregated from the primary bed flue gases in the module. These gases do reunite with the flue gases from the primary beds after the primary bed flue gases are cleaned by the first cyclone stage. A schematic diagram of the flue gas flow streams exterior to the vessel can be seen in Figure 4.2

4.2 Energy and Mass Balances

The energy and mass balances were calculated for design load and 70% plant load. These calculations are based on the specifications listed in Section 2. The steam duty requirements at different loads were taken from Table 2.5. For the design load the bed temperatures were fixed at 1750°F and the boiler surface was designed for this temperature. At reduced loads the required bed temperatures were calculated from the steam duty requirement. Fuel/air ratios were fixed by the overall plant design for each load. Once steam duty requirements, bed temperature, and fuel/air ratio are established, mass flow rates and velocities in the steam generator can be calculated. Tables A2-1 and A2-2 of Appendix 2 list the flow parameters for each bed. The mass balances and flow circuits for air, flue gases and solids are shown in Figures A2-1 and A2-2 for each module.

The energy balances were also calculated for each bed. Once the bed temperature was calculated, the bed was considered as a control volume and the heat inputs and losses were determined on a pound of fuel basis. The difference between the heat inputs and the heat losses is the energy per pound of fuel that is transferred to the steam. The air sensible heat input and the flue gas loss from the cells varied with the air/fuel ratio and the bed temperature. The rest of the heat inputs and losses were assumed



SCHEMATIC OF PARTICULATE COLLECTION SYSTEM	ORDER NO.	0-59-1192
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	CHECKED BY:	
	APPROVED BY:	
	SCALE:	= 1'-0"
	RD-714-123	

C. B. CO., NO. 44-512

FIG. 4.2

to be invariant with load on a pound of fuel basis. The inputs from the combustion of coal and the Ca-SO_2 reaction were 13,000 Btu/lb coal and 325 Btu/lb coal respectively. The radiation, latent heat of H_2O, solids sensible heat and unaccounted for losses were combined into one heat loss term of 797 Btu/lb coal. Six percent carbon carryover was assumed for the beds. This resulted in a carbon loss of 624 Btu/lb coal from the primary beds. Heat transfer above the beds and in the gas passage were calculated to determine the flue gas temperature decrease in these passages and hence the final flue gas energy loss from the steam generator.

The carbon burn-up cell is designed to be maintained at 2000°F entirely by the combustion of carbon returned from the first particulate collection system. The fuel input to this cell will therefore decrease as the load of the steam generator is decreased. When the fuel input to the primary beds is reduced, the bed temperature drops. In the carbon burn-up cell, however, it was felt that a temperature drop at reduced loads should be avoided because it would probably reduce the efficiency of the carbon burn-up process. For this reason the carbon burn-up cell was designed for 80% excess air at rated load. As the steam generator load is reduced the excess air to this cell is lowered to keep the bed temperature at 2000°F. At 70% load approximately 20% excess air is required for a bed temperature of 2000°F. If the excess air is lower than 20%, the carbon burn-up cell may lose efficiency due to insufficient excess air, so at loads less than 70% the bed level or temperature will have to be lowered. Flow diagrams showing the energy inputs and losses in the cycle can be found in Figures A2-3 and A2-4 of Appendix 2.

Steam generator efficiency calculations are misleading in a combined cycle plant, because the high flue gas temperature leaving the steam generator is not really a loss. Table A2-3 of Appendix 2 lists the expected steam generator losses based on a flue gas temperature of 275°F. This is not a true representation of the system, but it gives a means of comparing the steam generator with conventional units.

4.3 Tube Details

The layout of the tube walls and tube banks can be seen in Figures 3.1, 3.2 and 3.3. The tube enclosure walls, tube banks and the support systems for them are of conventional design. In fact, at first glance the pressure parts very closely resemble the convection pass of a conventional unit. The operation of the unit is, of course, very different from the operation of a conventional steam generator, but the fabrication of the pressure parts requires no new machines or new shop technology.

The tube enclosure walls are of monowall construction. That is, they are panels made up of straight, vertical tubes with metal fins welded between the tubes. The enclosure walls perform other important functions besides providing heat absorption surface for the evaporation duty. They also serve as partition walls to form various air and gas passages. Steam or water cooled walls are very desirable for this purpose, because they do not require as much maintenance as insulated walls and are not as expensive as walls made of high temperature alloy. Another function of vessel. The coal is transported pneumatically through these lines directly to the coal inlet nozzles at the fluidized bed. The flow through these lines is continuous and they are calibrated with respect to pressure drop so that the coal flow can be measured.

4.5 Sorbent Feeding

There are two areas of design in the sorbent feeding system. One area pertains to the location and number of sorbent inlets and outlets in the fluidized beds. This aspect of the system is discussed in Appendix 4. The other area of design is the transporting system that circulates the sorbent between the steam generator system and the regeneration system. This part of the system is presented in Appendix 5.

Since the sorbent average residence time is about one hour, only one inlet pipe and one outlet pipe are provided in each bed. The inlet pipe is located about 4 ft. above the air distributor plate. The outlet pipe is located about 4 ft. below the top of the tube bank. Fresh sorbent is added to the system to make up for the stone that is expected to be elutriated from the beds and for the spent stone removed in the regeneration system. This sorbent will be fed through the coal feeding system. No special equipment was required for this, but the equipment required for the coal feeding was sized considering this additional flow. Two types of systems were considered for the circulation of sorbent between the steam generator and the sorbent regenerator. One system that could be used (and which has been selected for the plant by Westinghouse) would be a lock hopper system similar to the coal feeding system designed by Petrocarb. With this type of system the possibilities for regenerator locations and operating pressures are quite extensive. An alternate system investigated by Foster Wheeler would be one which uses static pressure heads of the sorbent to achieve pressure sealing between different parts of the circulation system. A schematic of this type of system is shown in Figure A5-32 of Appendix 5. Slide valves are used to provide some means of control over the flow rate. These valves also separate the dense phase transport from the dilute phase transport. The operation of this system is described in Appendix 5.

4.6 Particulate Removal

The particulate collection system consists of two stages of particulate removal equipment. The flow of gases and solids in this system is shown schematically in Figure 4.2. The first stage of separators is used to minimize the unburned carbon energy loss by removing 90% of carryover. Particles collected in this stage are fed to the carbon burn-up cell through dip legs. In the carbon burn-up cell the excess air and bed temperature are kept at relatively high levels to enhance the combustion of the carbon in the returned solids. In this manner the carbon energy loss of the steam generator can be reduced without maintaining high excess air and temperature levels in the primary beds. The first separator stage consists of four Duclone type cyclones manufactured by the Ducon Company. These cyclones are housed in a pressure vessel located adjacent to the boiler module. The arrangement of these cyclones in the pressure vessel can be seen in Figure A4-4 of Appendix 4. The location of this pressure vessel and the secondary cyclones can be seen in Figure A4-5 of vessel. The coal is transported pneumatically through these lines directly to the coal inlet nozzles at the fluidized bed. The flow through these lines is continuous and they are calibrated with respect to pressure drop so that the coal flow can be measured.

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Appendix 4. A discussion of alternate arrangements for the first separator stage and the expected collection efficiency can also be found in this appendix.

The secondary particulate removal stage consists of two model 1,800 separators manufactured by the Aerodyne Development Corporation. This separator can be seen in Figure A5-35 of Appendix 5. The flue gas from the carbon burn-up cell enters the upper chamber of the first stage separator pressure vessel. Here this gas stream mixes with the cleaned gas from first stage separators and the combined stream leaves the pressure vessel through two gas pipes. Each of these gas pipes enters one of the secondary separators. The layout of this equipment is shown in Figure A4-5 of Appendix 4.

The secondary separators were designed for 97% collection efficiency. Basically these separators depend on the cyclonic action of two gas streams to effect particulate removal from the gases. In the design shown here both of the gas streams are dirty gas. After entering the separator, the flue gas from the steam generator is divided into two streams. One stream enters the bottom of the inner chamber and rotates as it flows upward. The other stream enters at the wall of the inner chamber near the top of the chamber and flows downward with a rotational motion. The combined effect of these two streams provides good efficiency over a range of particle sizes that extends down to 5 microns. The operation and efficiency of this separator is discussed in more detail in Appendix 5. After leaving the secondary separators, the gases go directly to the gas turbine.

4.7 Carbon Burn-up Cell

The location of the carbon burn-up cell can be seen in Figure 3.1. This cell is formed by the same tube walls that form the gas passage and there are no tubes in this fluidized bed. Although having tubes in this bed would probably provide a more even fluidization, there are several problems involved that excluded the use of tubes in the bed. One factor was the need to keep the bed height approximately the same as the other fluidized beds. This keeps the pressure drops across the beds approximately equal and thereby simplifies the gas handling aspect of the design. It was determined that placing tubes in the bed would require a lower bed height in order to maintain a high temperature in the carbon burn-up cell. This is due to the fact that the carbon input to the bed is fixed by the operation of the other beds, and it is not desirable to use a supplementary fuel in this bed. Also, the possibly unstable operation of this bed would increase the possibility of tube burn-out if steam circuits were located in this bed.

The solids that are collected in the first separator stage are fed to the carbon burn-up cell by four dip legs. This stream of solids contains particles of sorbent, ash and carbon. All of the sorbent and ash from the primary bed is also expected to be elutriated from the carbon burn-up cell. The carbon burn-up cell has a sorbent circulation system which can maintain the bed height in the cell so that any particles fed to the bed that are not elutriated will not accumulate in the bed. This cell is also supplied with an ignitor to accomplish start-up of the bed and to provide supplementary heat if it is needed.

5. OPERATION

5.1 General Problems

The steam flow circuits of the boilers presented in this report are very similar to those of typical once-through units. The modular boiler design and the requirements of fluidized bed combustion affect the operation of the steam cycle, but the basic philosophy of once-through operation is not changed. Overall steam generator operation depends both on steam side and gas side requirements and limitations. Certain requirements of fluidized bed combustion make the operation of the boiler more difficult than conventionally fired boilers, but there are also some aspects of boiler control that benefit from the fluidized bed design. One characteristic of fluidized bed combustion is that there are limitations on the amount of combustion air that can be fed to each bed. The superficial velocity in the beds must be greater than the minimum fluidizing velocity but less than the terminal velocity for the particles in the bed. The range of velocities that are allowable can be seen in Figure 2.2 of Section 2. The limitations on air flow place restrictions on firing rate, but other parameters such as bed temperature and air/fuel ratio can be adjusted to extend the range of operation.

Besides limitations imposed by the fluidization requirements of the bed, there are also restrictions on operation that are caused by the heat transfer characteristics of the bed. One characteristic is that the mixing action of the fluidized bed uniformly distributes any heat added to the bed. In other words, there are only small temperature gradients in the bed. Also the bed-to-tube heat transfer coefficient remains nearly constant during turn down. The overall heat transfer coefficient is reduced slightly at reduced loads due to a decrease in the steam side heat transfer coefficient, but the major effect of reducing the heat input during turn down will be a reduction in bed temperature. There are, however, limits to how low the bed temperature can be. During normal operation the sulfur removal process requires that the bed temperature be above 1400°F. During start-up operation, the uniformity of the temperature in the bed and the high heat transfer coefficient make ignition of the coal a problem. The ignitors must supply enough heat to overcome the heat transfer to the boiler tubes and heat the bed up to the ignition point of the coal.

In some aspects of steam generator operation the fluidized bed design presented in this report has a greater flexibility than conventional units. The separation of steam generator functions (i.e., superheating, reheating, etc.) into separately fired beds is highly desirable, since the firing rate and hence heat input, to each part of the water/steam circuitry can be separately and positively controlled.

There are several reasons why it is important to have control over the distribution of heat to the water/steam circuitry. First, the tube banks are designed for specific locations in the circuit. If at a certain load the distirbution of heat transfer is such that water is flowing in tubes designed for steam, or vice versa, pressure drop and/or tube metal temperatures can be seriously affected. Also during start-up of a unit, certain tube banks must be protected from hot gases during these time periods or excessive tube metal temperatures will result.

In conventional units all the fuel is combusted in one furnace so the distribution of heat transfer to the water/steam circuitry must be accomplished in a more indirect manner than in the fluidized bed design presented in this report. Two methods are used concurrently to control the distribution of heat transfer in conventional units. One method is to separate the convection pass of the boiler into two parallel passes. By means of dampers located in the relatively cool zones of the passes the gas flow distribution between the two passes can be controlled. The other means of control that is used is spraying feedwater into the steam. If at a certain load the heat transfer characteristics of the boiler are such that too much heat is being absorbed in the superheater elements and not enough in the economizer and evaporator sections, feedwater is bypassed around the economizer and evaporator sections and sprayed into the superheater circuit. The spray nozzles are carefully designed so that the feedwater will mix rapidly with the steam and evaporate. This type of operation insures that the tube banks will have the phase of steam or water flow that they were designed for. Figure 5.1 shows a typical layout of the parallel gas passes.

There are several problems involved with the conventional means of distributing the heat that can be overcome by separately fired beds. It is impractical to have more than two parallel gas passes in a conventional unit, but the stacked bed design gives four areas of direct heat input control. The separate bed arrangement also provides a better means of keeping hot gases away from certain tubes during start-up, because in reality, control dampers do not provide 100% shut off and so it is not possible

PARALLEL PASS WITH DAMPER CONTROL

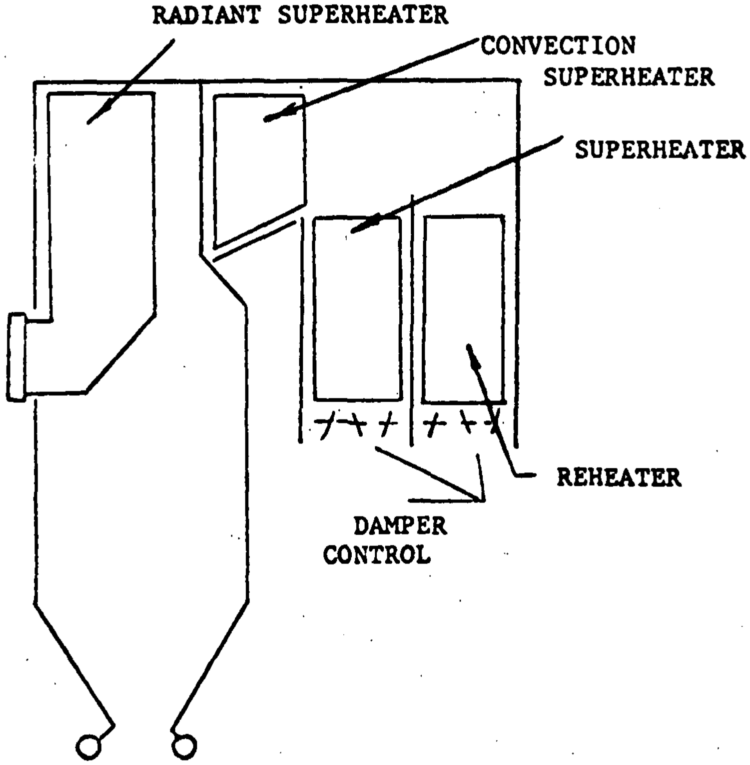


FIGURE 5.1

to completely stop the flow of hot gases to one of the two parallel convection passes in a conventional unit. However, leakage of the control dampers in the fluidized bed design just results in relatively cool air flowing through the bed.

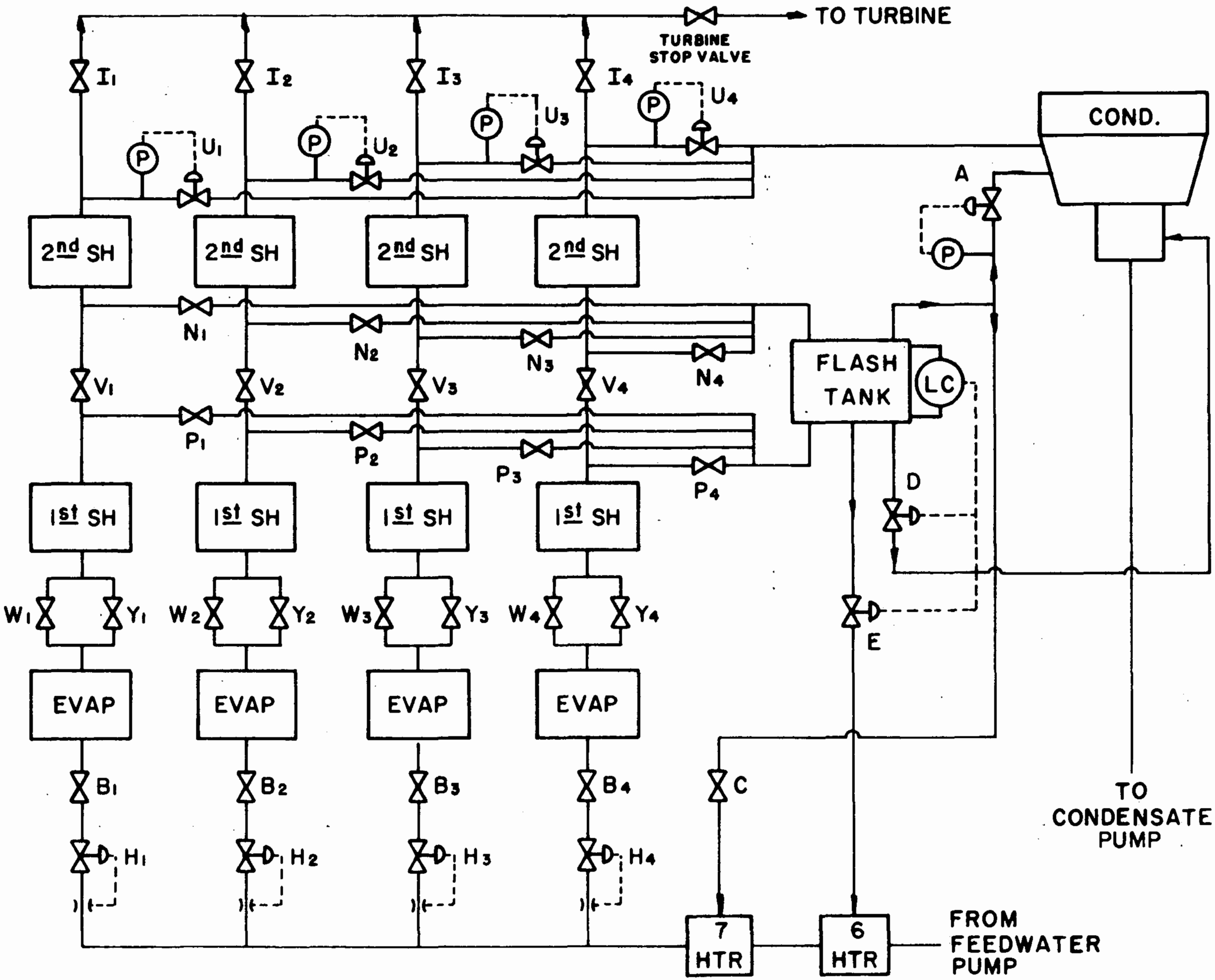
It is not meant to be implied that conventional means of distributing heat to the steam cycle are inadequate, because these systems are used successfully. It should just be kept in mind that the obvious operational problems associated with the fluidized bed process may be balanced by certain favorable operational features of the process.

5.2 Start-Up

Start-up of the plant is accomplished by putting one module into operation at a time. There are two possible start-up situations. One situation is initial start-up of a cold module. This type of start-up would be required for the first module to be put into operation. The other situation would be start-up of a module with one or more modules already in operation. This would occur during plant start-up and during load changes.

The boiler start-up system is shown in Figure 5.2. This is basically a conventional once-through start-up system that is modified for a modular steam generator design. To start-up the initial steam generator module the B, W, P and D valves are opened. Water flow is then started through the pre-evaporator bed, wall tubes and first superheater bed, from which it flows to the flash tank and then to the condenser. Next the pre-evaporator and first superheater beds are ignited.

The bed conditions that will be necessary to ignite the

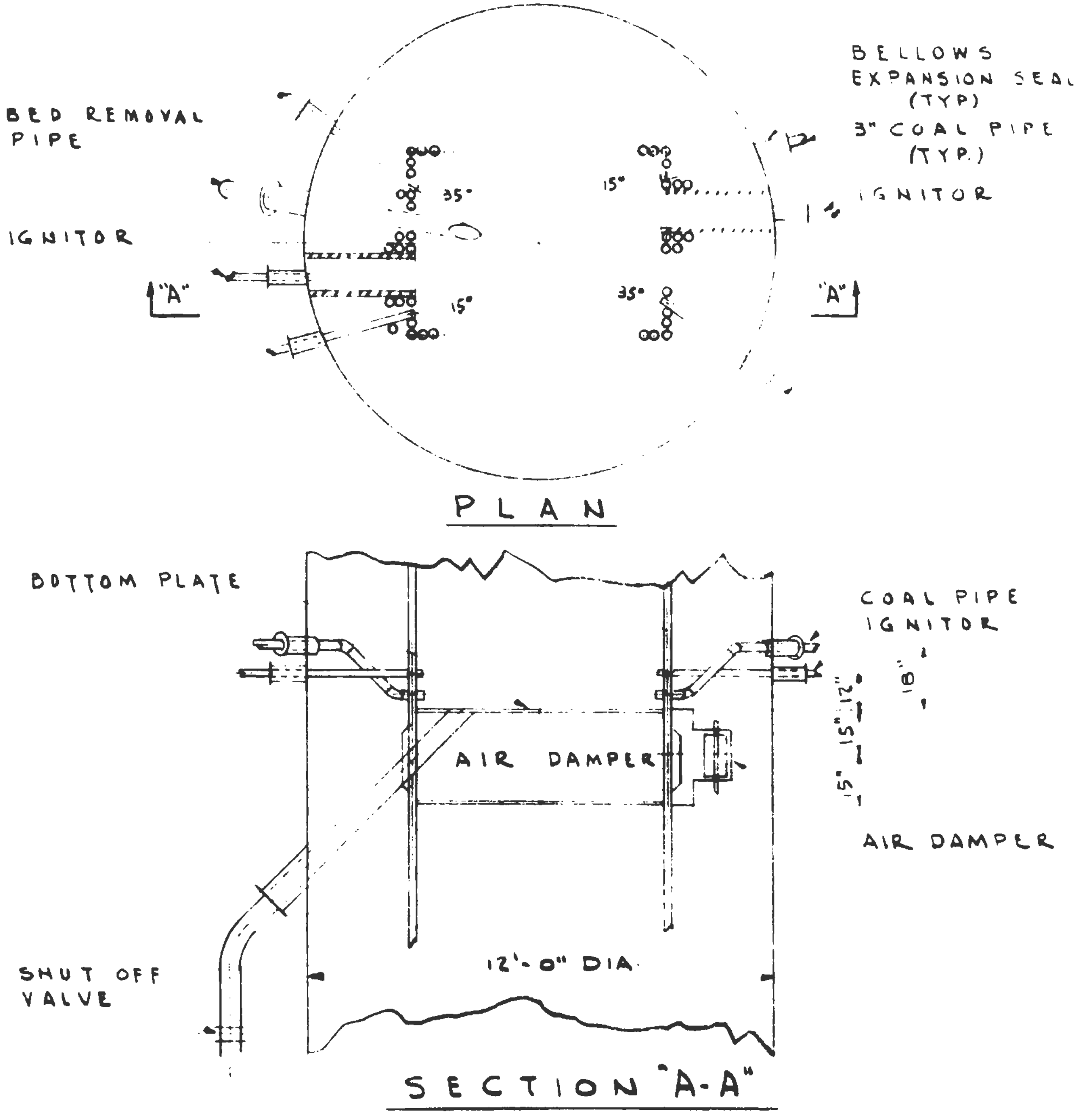


BOILER START-UP SYSTEM

FIGURE 5.2

bed will have to be determined from pilot plant tests. For this design it was assumed that during cold start-up the water temperature in both tube banks will be 80°F and that a bed temperature of 750°F will be necessary for the ignition of coal. The ignitors were assumed to have a capacity of 25 x 10^6 Btu/hr. Under these conditions it was estimated that a 3 ft. high bed would be the maximum that could be brought up to the ignition temperature. It was assumed in these calculations, that the ignitor combustion efficiency of the system shown in Figure 3.3 was 70%. If this combustion efficiency is not achieved, the ignitor system could be modified as shown in Figure 5.3 or 5.4. The system in Figure 5.3 uses a combustion chamber to provide a hot zone that will give the ignitor fuel a better chance to combust before it is exposed to the heat sink affects of the fluidized beds. Another approach would be to put more ignitors into the beds. This could be done as shown in Figure 5.4. Another possibility rather than modifying the system would be to maintain only a very small flow in the tubes during start-up. If the flow were kept small enough, the water temperature would increase and reduce the heat loss to the tubes. Once coal ignition is established the bed height can be increased as rapidly as the dolomite feed system will allow. Assumming a feed rate of 200 lb/min, it would take about one hour to raise the pre-evaporator and first superheater beds to full capacity.

As heat is added to the system, valve D is closed and the heat recovery valve E is opened. This mode of operation allows a more rapid heating of the water than would be possible with



ALTERNATE IGNITOR SCHEME "A"	ORDER NO.	0-59-1192	
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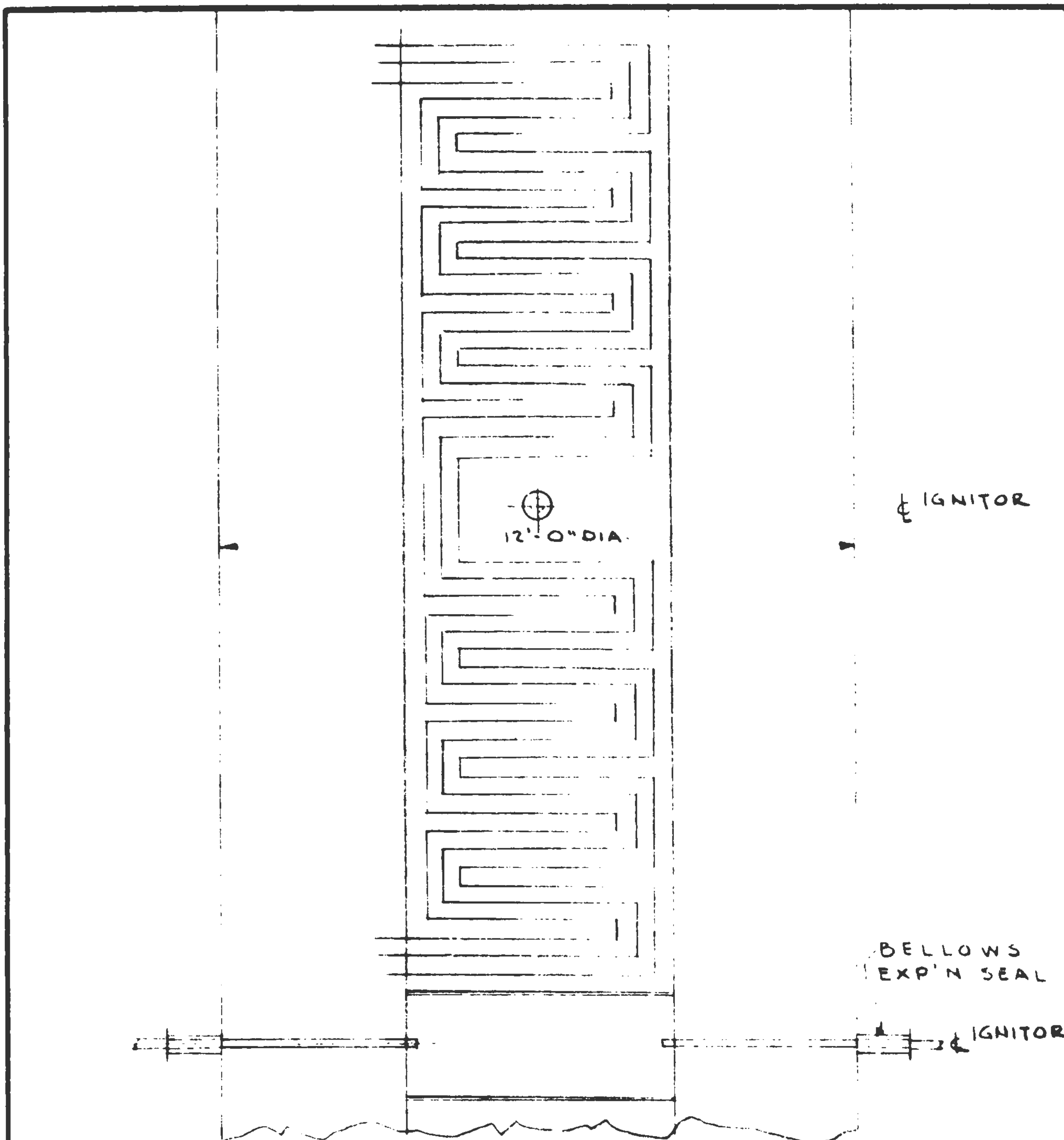
FIG. 5.3



ALTERNATE IGNITOR
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FIG. 5.4

circulation through the condenser. As heat is added to the circuit a pressure of 2600 psig is maintained in the pre-evaporator and evaporator circuits by throttling the W valve. The pressure in the first superheater and flash tank is kept at 600 psig by controlling the A, D and E valves. A steam-water level forms in the flash tank, and the N and I valves are opened to allow this steam to warm the second superheater tubes and the subsequent steam lines. Next the second superheater bed is ignited. Igniting this bed will be considerably easier than the first two beds because the tubes will be heated by the entering steam. It is anticipated that this bed can be ignited with a full capacity of dolomite.

When the steam enthalpy is sufficient, the turbine can be warmed and rolled. The flash tank pressure is then raised to 1000 psig and the steam flow is increased until the load is about 10% of full plant load. At this point the turbine throttle pressure is raised to the operating value by closing the P and N valves and opening the V and Y valves. Subsequent increases in load are controlled by the turbine governor and accomplished by increased steam flow.

Start-up of additional modules whether during plant start-up or load changes is very similar to the start-up of the initial module, but there are some important differences. First the feedwater temperature will be greater than 500°F instead of the 80°F temperature that is present in the cold start. With this reduced heat loss to the water circuit, the pre-evaporator and first superheater beds need not be ignited at a low dolomite capacity. This speeds up the start-up process which is important for load changes.

In this start-up situation the flash tank system is used in the same manner as before, but the I valve cannot be opened until the steam temperature and pressure match the existing steam turbine conditions. Until these conditions are reached, the U valve will be open and the steam will be sent to the condenser or a high pressure feedwater heater. In a conventional once-through steam generator plant it takes 6-7 hours to reach full load from a cold start. Steam turbine considerations are the main factor in this time schedule so the extra time required to ignite the fluidized beds should not be a problem.

5.3 Load Control

A procedure has been developed to change the plant power output to meet system load demands. It is required that the plant be capable of a 5% load change per minute and turn down to 25% of full load. If the plant is operating at full load and it is desired to reduce the load, the output of each of the modules will be reduced simultaneously until 75% load is reached. At this point one module will be removed from service, and the load of the remaining modules will be increased to maintain 75% plant load. The output of the remaining three modules can then be lowered simultaneously until 50% plant load is reached. At 50% plant load the three individual modules would be operating at 67% of their full load. At this point one of the modules would be taken out of service, and the remaining two modules would be brought up to full load. The same procedure will be used as the load is decreased further. The individual module turn down capability is 50%. With a four module plant this will provide a smooth load change curve with the above turn down procedure. The 50% module turn down limit is dictated by the sulfur removal process. Since load is reduced by lowering the bed temperature, the low temperature limit of the sulfur removal process also limits the turn down capability of the module.

The steam generator parameters for each module at reduced load can be seen in Tables 2.4 and 2.5 of Section 2 and Tables A2-1 and A2-2 of Appendix 2. The steam side conditions are based on the Hammond #4 Station of Georgia Power and Light Company. The feedwater inlet temperatures to the steam generator modules differs from the Hammond #4 temperatures because of the stack gas cooler in the combined cycle plant. The gas side parameters are a function of the heat requirements of the steam cycle and the flow requirements of the gas turbine. The bed levels are not changed during turn down. As the feedwater flow rate is reduced, the fuel input to the various beds is reduced. This lowers the bed temperature, and thereby reduces the heat to steam. The air/fuel ratio is increased during turn down. This is necessary both for the fluidization of the bed and for the efficient use of the gas turbine. The fuel/air ratios for different loads are given in Table 2.4 of Section 2. The resulting bed superficial velocities can be found in Tables A2-1 and A2-2. The gas turbine compressor air rate is constant so as fuel rate (load) changes the air/fuel ratio must change also. Fortunately, the superficial velocity remains fairly constant with load turn down. The temperatures of individual beds within a module can be different at low loads. This is due partially to changes in heat transfer characteristics of the steam generator at low loads and partially due to changes in the steam cycle conditions. These changes in the steam cycle conditions include the proportion of reheat flow and the inlet feedwater temperature.

5.4 Shut-Down

The normal shut-down of a module will be accomplished by basically the reverse procedure of start-up. Again there are two possible situations. One condition would be the shut-down of one module while the other modules are in operation. The other situation would be the shut-down of the final module. In the first case the module load would be reduced until it is at 50% of the full module steam flow. At this point the I valve would be closed and the U valve would open to permit flow to the condenser. Fuel flow would be stopped, but fluidizing air flow will be maintained in the beds in order to purge and cool the beds. When the bed has cooled sufficiently to indicated that it is purged, the air flow can be stopped. If it were desired to keep the module in a standby condition, feedwater flow would be continued through the pre-evaporator bed, evaporator walls and first superheater bank. This flow would be achieved by closing the V valve and opening the P valve. Feedwater would then flow through these tube circuits and maintain the bed at approximately 500°F. If it is not required that the module be kept in a standby condition, feedwater flow will be stopped by closing the B valve.

The last module to be removed from service would be shut down in a different manner. When the last module is operating at full load, the steam turbine will be at 25% load. At this point the pressure reducing valve W will be throttled and the

P valve will be controlled in order to ramp the turbine throttle pressure down to 1000 psi. At this point the steam turbine would be removed from service and the I valve would be closed. From this point on the shut-down operation would be the same as with the previously removed modules.

Several possibilities exist that would require emergency shut-down of the steam generators. One possibility would be the loss of load on a gas turbine which would cause a run-away condition. If this occurred, the fuel input would be automatically stopped and a relief valve would open to divert the flow from the gas turbine. As the gas turbine slows down, the air flow through the beds should still be sufficient to purge any volatiles from the bed. On the steam side, the I valve would be closed to protect the steam turbine.

Emergency shut-down due to steam side problems would be accomplished in the same manner as in conventional units. The components of the steam system (i.e., condenser, flash tank, feedwater heaters, etc.) are designed on the same basis as a plant of the same steam flow as the total of the four modules. This is necessary because only one steam turbine is used for the four module plant. One problem of emergency shut-down is that the fluidized beds will contain a greater amount of heat than conventional boilers due to the specific heat of the sorbent. This same situation, however, does occur to some degree in conventional units where a highly fouling fuel is used. In these units ash attached to the tubes also has a specific heat that must be dissipated. In either case, if a circulation of feedwater flow is maintained, no problem exists. Even if circulation is stopped on the steam side, however, the combined effects of air flow through the beds and blow down on the steam side should prevent any damage to the tubes.

A tube failure in the steam generator would also require removing a module from service. This event, however, would not require an emergency shut-down. In fact, it would probably take a relatively long period of time to determine that a tube rupture had occurred. An indication of a tube rupture would be a loss of make-up water. When this is noticed, the steam generator would be shut-down in the normal manner.

5.5 Performance

The performance of the steam generator is summarized in Table 5.1. The values tabulated are for the total 300 MW plant. Additional information on performance can be found in Section 2 of the text and Appendix 2.

TABLE 5.1

STEAM GENERATOR PERFORMANCE SUMMARY

(FOR TOTAL PLANT)

	DESIGN LOAD	70% PLANT LOAD
Fuel flow - M lb/hr	215	154
Air flow - M lb/hr	2,506	2,506
Flue gas flow - M lb/hr	2,705	2,650
Feedwater inlet temperature - °F	578	547
Feedwater inlet pressure - psig	2,824	2,650
Main steam outlet temperature - °F	1,000	1,000
Main steam outlet pressure - psig	2,500	2,444
Main steam flow - M lb/hr	1,727	1,140
Reheat inlet temperature - °F	650	584
Reheat inlet pressure - psig	600	405
Reheat outlet temperature - °f	1,000	1,000
Reheat outlet pressure - psig	580	391
Reheat flow - M lb/hr	1,644	1,090
Flue gas exit temperature - °F	1,650	1,400
Flue gas pressure drop - psig	4.8	4.8
Fuel/air ratio	0.0856	0.0615

6. STEAM GENERATOR COSTS

6.1 Capital Costs

Cost estimates were obtained for the following pieces of equipment in the 300 MW plant described in this report. The cost estimates include engineering, material, fabrication, profit and overhead. Shipping costs to the plant location are not included since a plant site was not specified.

6.1.1 Steam Generator Modules Details of these modules are shown in Figures 3.1, 3.2, 3.3 and 3.4. Four modules are required for a 300 MW plant. The estimated cost includes piping in the vessel up to and including the penetrations of the pressure vessel shell.

6.1.2 First Stage Separators and Gas Piping Details of this system can be seen in Figures A4-4 and A4-5. Included in this cost estimate are the first stage separators, the separator pressure vessel and all of the gas piping from the steam generator outlet to the secondary separator inlets. The cost for this equipment was estimated for two cases. In one case the gas piping from the steam generator to the first stage separator is lined with hard refractory, but the first stage separator pressure vessel and the gas piping from it are lined with stainless steel. In the other case hard refractory without an alloy liner was used throughout.

6.1.3 Second Stage Separators The second stage separators can be seen in Figures A4-5 and A5-35. The cost for these cyclones includes the pressure vessel and internals. The pressure vessel is lined with insulating refractory and an alloy shroud.

6.1.4 Coal Handling System Details of this system can be seen in Figures A5-3 and A5-4. Costs for this system include coal conveying, drying, crushing, and storage equipment from the coal train discharge up to but not including the surge bin of the coal pressurizing system. Structural steel, instrumentation, controls and erection are included in this cost estimate.

6.1.5 Coal Pressurizing and Feeding System This system can be seen in Figures A5-3 and A5-4. The cost estimates include the storage bins, storage injectors, primary injectors, and the piping and valves between these vessels. Cost estimates are also included for the coal feed lines from the primary injectors to the beds and for instrumentation and control of the entire system.

Table 6.1 lists the cost estimates for the equipment specified.

6.2 Maintenance and Operating Costs

A quantitative estimate of maintenance and operating costs is impossible to determine at this stage of development. As far as the maintenance of the steam generator modules is concerned, it is probable that certain repairs will be more time consuming than in conventional units. Specifically, the additional problems in the repair of tube leaks are discussed in Appendix 3. The significant effect on maintenance cost, however, will depend more on the frequency that tube ruptures occur than on the increase in the cost of repairing them. The steam generator design presented in this report may be less prone to tube ruptures than conventional

TABLE 6.1

CAPITAL COSTS FOR 300 MW COMBINED CYCLE PLANT*

(Shop Assembled)

1.	Steam Generators	$3,856,000
2.	Steam Generators - Erection	$ 500,000
3.	First Stage Separators and Gas Piping	
	With stainless liner	$2,110,000
	With hard refractory liner	$1,500,000
4.	Second Stage Separators	$1,992,000
5.	Coal Handling System	$2,500,000
6.	Coal Pressurizing and Feeding System	$2,400,000

* The extent of equipment in each item is specified in the text.

units, because the heat absorption rates in the steam and water circuits will be more even. A major cause of tube ruptures in conventional units is the uneven heat absorption rates in the furnace. Since the fluidized beds should have very even heat absorption rates, the possibilities of a tube overheating may be reduced, but it is nearly impossible to estimate the frequency of tube ruptures even in conventional units. Another advantage of the fluidized bed steam generator is that soot blowers and pulverizers are not required. These items, which are necessary in conventional coal fired plants, require a large amount of maintenance. Other maintenance of the steam generator such as repairing insulation, ignitors and dampers should be no more frequent or expensive than in conventional units.

The high temperature flue gas piping is not found in conventional steam generators, but experience with this type of piping in process applications has shown that practically no maintenance is required.

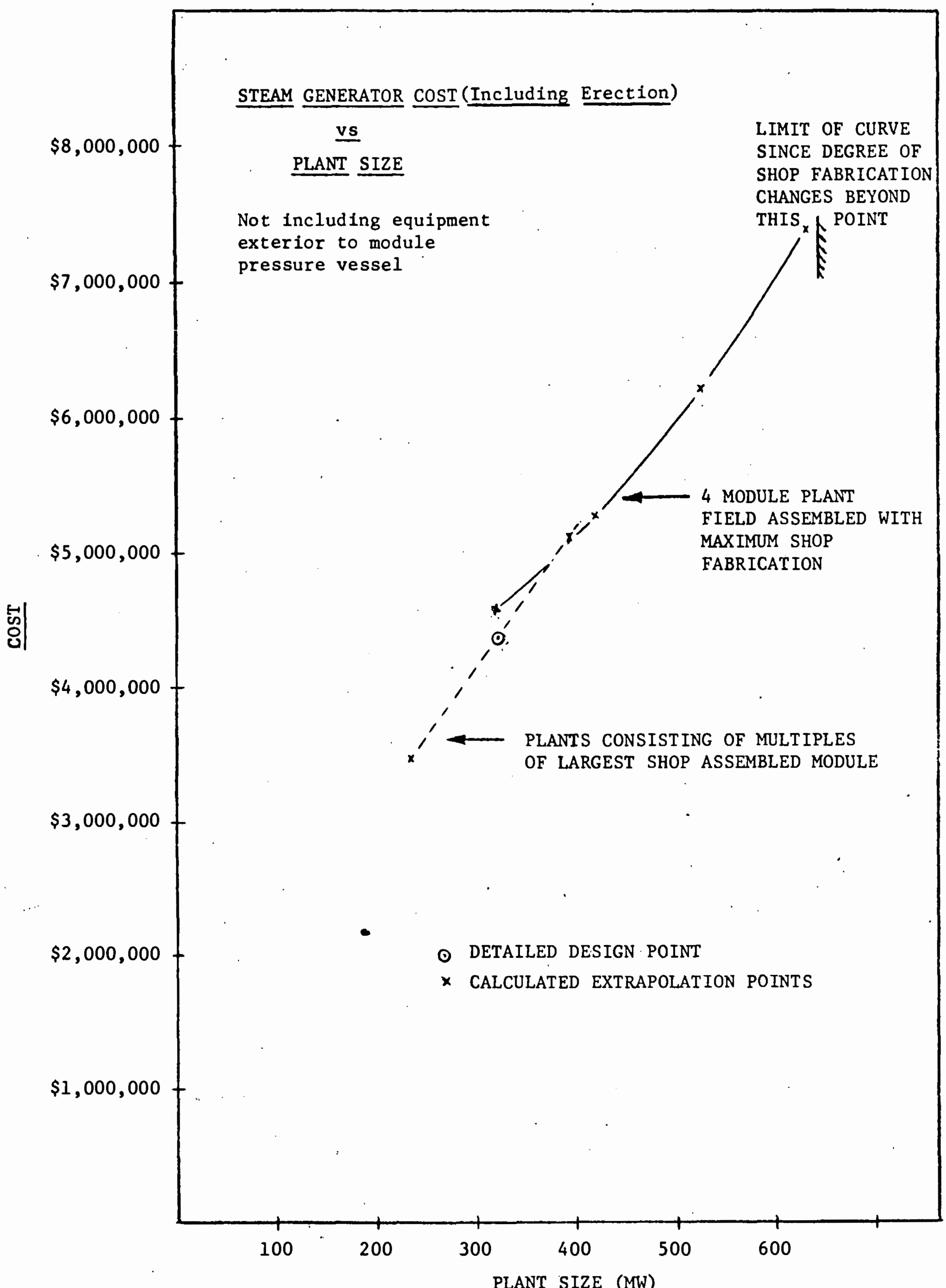
6.3 Extrapolation of Capital Costs to 600 MW

The capital cost estimates presented in Section 6.1 for a 300 MW plant were extrapolated to obtain cost estimates for a 600 MW plant. These estimates are rough, however, because detailed designs of the equipment were not developed for the 600 MW plant. Instead, the equipment was scaled up assuming that the four-module arrangement will be retained as the plant size is increased to 600 MW. The selection of the number of boiler modules is discussed in Appendix 1. Basically, it is felt that the added complexity of plant operation with more than four modules and the additional costs of auxiliary equipment offset the benefits of a fully shop assembled steam generator module.

Figure 6.1 shows the expected steam generator costs for different size plants. One curve assumes that four modules are used in each plant, but the module size is increased. The other curve assumes that the maximum size shop assembled module is used, but the number of these modules is increased. It can be seen from these curves that a five-module shop assembled plant is approximately the same in cost as a four-module field erected plant. If the costs of auxiliary equipment such as coal handling and feeding equipment were included, however, the cost of the five module plant would exceed that of the four module plant. This is due to the additional number of vessels, hoppers, and controls that are required as the number of modules is increased.

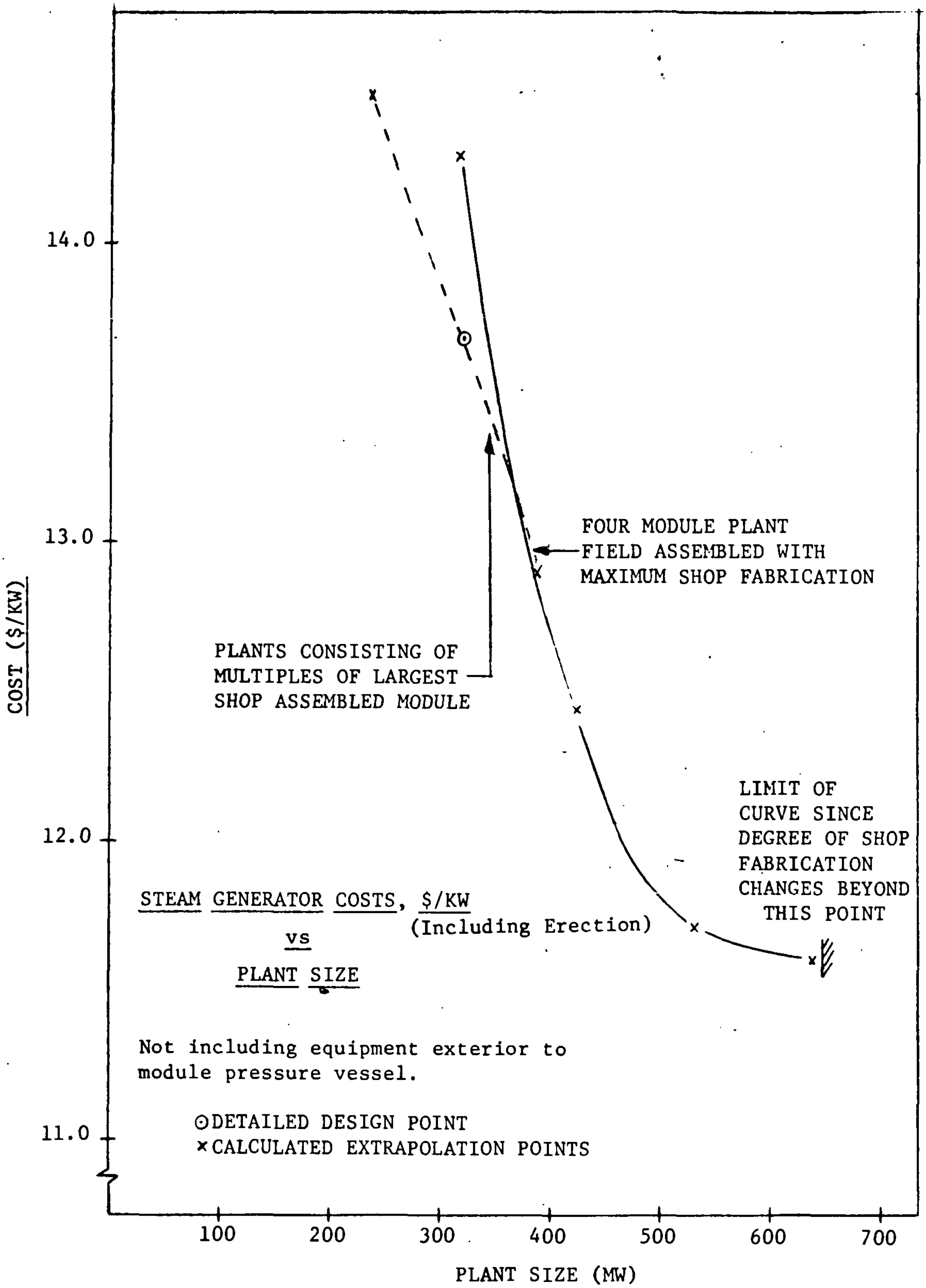
Over the range of plant sizes shown in Figure 6.1, the benefits of shop assembly can still be taken advantage of to some extent, even with four-module plants. The resulting sizes are such that individual fluidized bed cells could be assembled in the shop including the enclosure walls, tube banks, headers and dampers. These units could then be shipped to the plant site for final field assembly of the steam generators. This work would consist of welding the cells together and assembling the pressure vessel. In conventional field assembled steam generators the enclosure wall panels, tube banks and headers have to be shipped separately so there is a larger scope of field work.

Figure 6.2 shows the cost of the extrapolated steam generators on a dollars per kilowatt basis. It can be seen that the dollars per kilwatt cost of the four-module plant levels off at



STEAM GENERATOR COST (Including Erection)
vs
PLANT SIZE
Not including equipment
exterior to module
pressure vessel
LIMIT OF CURVE
SINCE DEGREE OF
SHOP FABRICATION
CHANGES BEYOND
THIS POINT
4 MODULE PLANT
FIELD ASSEMBLED WITH
MAXIMUM SHOP
FABRICATION
PLANTS CONSISTING OF MULTIPLES
OF LARGEST SHOP ASSEMBLED MODULE
DETAILED DESIGN POINT
CALCULATED EXTRAPOLATION POINTS
$8,000,000
$7,000,000
$6,000,000
$5,000,000
$4,000,000
$3,000,000
$2,000,000
$1,000,000
COST
100
200
300
400
500
600
PLANT SIZE (MW)

Figure 6.1



COST ($/KW)
14.0
13.0
12.0
11.0
FOUR MODULE PLANT
FIELD ASSEMBLED WITH
MAXIMUM SHOP FABRICATION
PLANTS CONSISTING OF
MULTIPLES OF LARGEST
SHOP ASSEMBLED MODULE
LIMIT OF
CURVE SINCE
DEGREE OF SHOP
FABRICATION
CHANGES BEYOND
THIS POINT
STEAM GENERATOR COSTS, $/KW
(Including Erection)
vs
PLANT SIZE
Not including equipment exterior to
module pressure vessel.
⊙DETAILED DESIGN POINT
×CALCULATED EXTRAPOLATION POINTS
100
200
300
400
500
600
700
PLANT SIZE (MW)

Figure 6.2

about 600 MW. This is mainly because pressure vessel costs rise at an increasing rate as plant, and hence module, size is increased. Beyond this point it might be beneficial to increase the number of modules, or to change the basic design in some manner. In order to investigate this, however, another detailed design would have to be developed for the steam generator and the auxiliary equipment. The reason for this is that the cost of considerable extrapolation has already been used in arriving at the five-module plant, and the compounded errors that would result from further extrapolation to larger size modules would make the results of quentionable value.

The estimated costs for the steam generator and auxiliary equipment for a 600 MW, four-module plant are tabulated in Table 6.2. Except for the structural steel and platform cost, which was not included in the 300 MW plant cost, the scope of equipment in Table 6.2 is the same as that in Table 6.1. It should be kept in mind, however, that these costs are not based on a detailed design, but on an extrapolation of the 300 MW plant. Figure 6.3 indicates the proportionate cost of the different items that make up total steam generator cost and how these cost items change as the plant size is extrapolated.

TABLE 6.2

CAPITAL COSTS FOR A 600 MW COMBINED CYCLE PLANT*

1.	Steam Generators	$6,669,000
2.	Steam Generators - Erection	$ 725,000
3.	First Stage Separators and Gas Piping	
	With Stainless Lining	$3,560,000
	With Hard Refractory Lining	$2,560,000
4.	Second Stage Separators	$3,984,000
5.	Coal Handling System	$2,750,000
6.	Coal Pressurizing and Feeding System	$3,400,000
7.	Structural Steel and Platforms for Steam Generators and First Stage Separators	$ 631,000
8.	Structural Steel and Platforms for Steam Generators and First Stage Separators - Erection	$ 303,000

* The extent of equipment in each item is specified in the text.

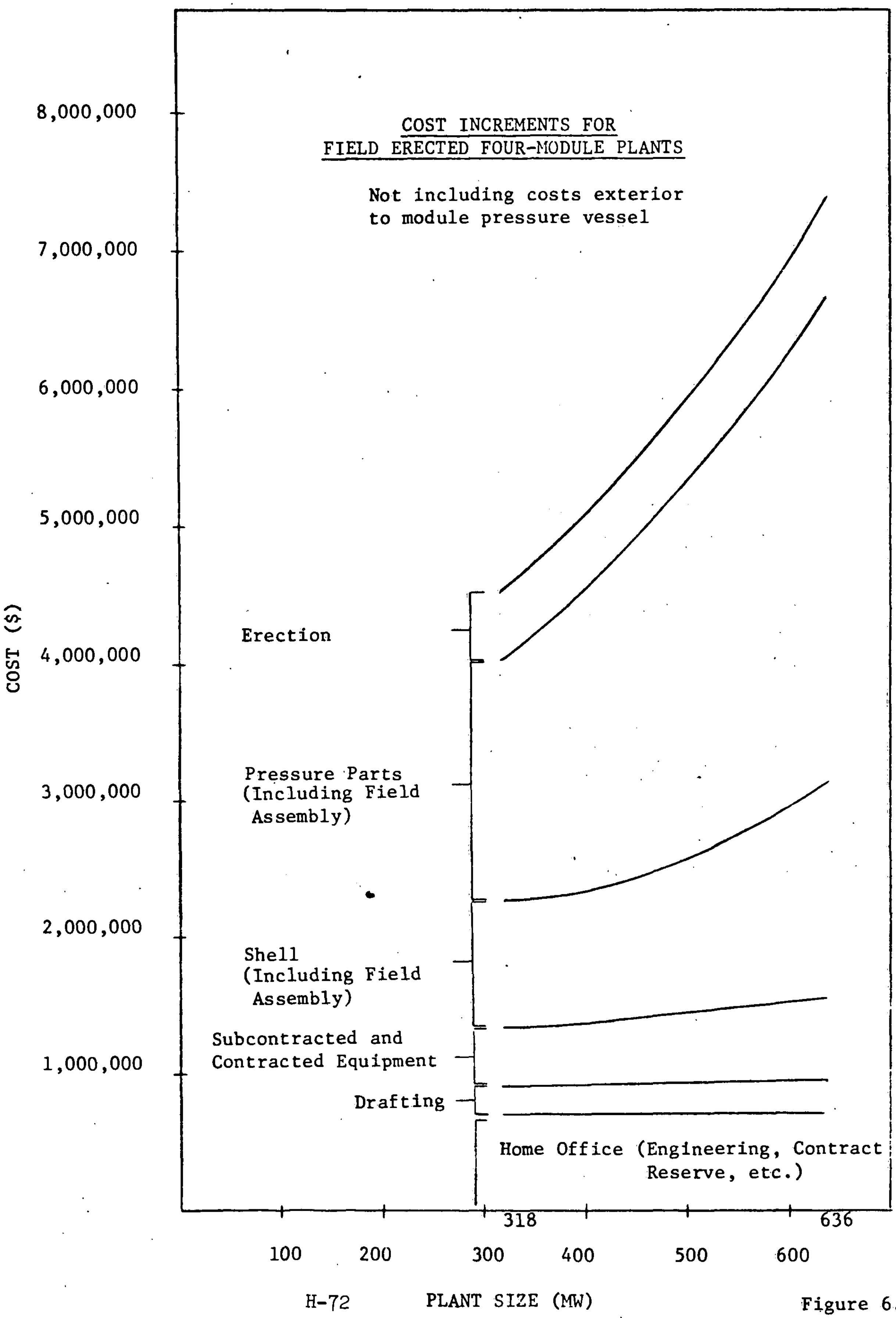

COST INCREMENTS FOR
FIELD ERECTED FOUR-MODULE PLANTS
Not including costs exterior
to module pressure vessel
8,000,000
7,000,000
6,000,000
5,000,000
4,000,000
3,000,000
2,000,000
1,000,000
COST ($)
Erection
Pressure Parts
(Including Field
Assembly)
Shell
(Including Field
Assembly)
Subcontracted and
Contracted Equipment
Drafting
Home Office (Engineering, Contract
Reserve, etc.)
318
636
100
200
300
400
500
600
PLANT SIZE (MW)

Figure 6.3

7. DEVELOPMENT REQUIREMENTS

One of the objectives of this study is to determine what areas in the plant design require further efforts in research or development. Broad areas to be considered are the fluidized bed characteristics, mechanical design problems, and safety aspects.

The steam generator design that was developed in this study was based on certain bed characteristics. Some bed characteristics were quantitative in nature such as the heat transfer coefficient and the amount of particulate elutriation. Others were qualitative such as the uniformity of bed temperature and the absence of fouling in the bed. The quantitative assumptions need to be more accurately determined by experiments on beds that are similar to the beds in the design. The qualitative assumptions should be validated over the entire range of possible operation of the beds. All of these characteristics influence the design of the steam generator. The uniformity of the bed temperature and the value of the heat transfer coefficient influence the selection of tube materials and determine the required amount of tube surface. Fouling in the bed would influence the design of the tube bundles, because it would be very difficult to clean ash deposits off of tubes with the close tube spacings that are desirable for good fluidization.

It would also be desirable to optimize the coal feed nozzles and the air distribution nozzles. This type of an optimization can only be done experimentally. Aspects of the air nozzles that require optimization are the number of nozzles and the shape of the nozzles. The effects of the number and location of the coal nozzles on carbon burn-out and bed temperature distribution should also be determined.

Certain characteristics of the gas flow after the fluidized beds also require further development. The height and heat transfer coefficient of the transition zone above the bed should be more accurately determined. Both of these parameters of the transition zone could be strongly dependent on bed dimensions, tube orientation and fuel firing rate. There is also a possibility of a gas temperature rise above the bed. This would be due to a certain amount of combustion after the bed. The extent of this combustion should be evaluated for the type of fluidized beds shown in this design so that convection heat transfer surface could be located after the beds if it is required.

There are certain mechanical aspects of the design that need further development. The possibility of extreme tube vibration in the beds could influence the selection of tube diameters and the shape of the beds. A more detailed analysis of the shell penetrations then was possible in the scope of this project should be undertaken for all possible operating conditions.

Safety aspects of the system should be analyzed further. Certain problems relating to safety will require feedback from experimental tests of the fluidized beds. The carbon carryover from the beds and extent of combustion after the beds will affect the possibilities of fire and explosion in the system. Other safety aspects will primarily be concerned with the control system. The possibility of a pressure build-up in the vessel can easily be taken care of with safety valves. A leak in the pressure vessel due to other causes, however, would be more difficult to handle. The reaction force caused by a gas leak of this type could put a stress on the vessel and also the hot escaping gas could injure personnel. Some means of automatically determining this type of a leak and depressurizing the vessel should be developed and incorporated into the control system.

APPENDIX 1

CANDIDATE CONCEPTS

A1.1 Determination of the Steam Cycle and Gas Pressure

The selection of a fluidized bed combined cycle power plant design requires the examination of both the steam side and the gas side of the power cycle. The steam side parameters are somewhat fixed by the present market for large, high pressure steam generators. Basically, the choice is between two cycle conditions, subcritical and supercritical. Two steam cycles were chosen as being typical and suitable as a guideline for the fluidized bed study. The Homer City Station, Unit 2, of the Pennsylvania Electric Company was used as a basis for the supercritical cycle, and the Hammond Station, Unit 4, of the Georgia Power and Light Company was used for the subcritical cycle. The parameters of both of these units are listed in Table A1-1. The subcritical cycle results in lower boiler costs because of thinner pressure part tube walls and lower tube metal temperatures. Thermodynamic and cost analyses of 10 atmosphere combined cycle plants done by Westinghouse have also shown that there is no advantage in using supercritical steam conditions at 1000°F final steam temperature.

The type of steam flow circuit in the boiler also has to be determined. Both natural circulation and once-through circuits were considered. Although natural circulation systems have characteristics that are beneficial in subcritical steam generators, the physical restrictions of a pressurized fluidized bed boiler

TABLE A1-1

TYPICAL SUBCRITICAL AND SUPERCRITICAL STEAM CYCLES

	SUBCRITICAL	SUPERCRITICAL
Unit Capacity, MW	500	660
Steam Flow, lb/hr	3,326,000	4,620,000
Steam Pressure, psig	2,486	3,800
Steam Temperature, °F	1,000	1,005
Reheat Steam Flow, lb/hr	3,206,500	3,873,000
Reheat Inlet Pressure, psig	452	630
Reheat Inlet Temperature, °F	607	588
Reheat Outlet Pressure, psig	581	605
Reheat Outlet Temperature, °F	1,000	1,005
Feedwater Inlet Temperature	465	543
Drum Pressure, psig	2,530	-

make the use of a natural circulation system impractical. A major problem with this type of circuit is the need for a large steam drum. This drum would have to be placed on the top of the steam generator pressure vessel and many downcomers and risers would have to penetrate the pressure vessel shell.

The gas side parameter that has the greatest effect on steam generator cost is the gas pressure. It is desirable to relate the gas side design to the steam side design so that the various curves based on gas flow will have more meaning in terms of boiler size. This can be accomplished by fixing certain parameters. If the furnace exit temperature, excess air, heating value of fuel, and steam cycle conditions are established, a relationship between steam flow and gas flow can be established. This relationship is shown in Figure A1-1 for the subcritical cycle.

Figure A1-2 is a plot of vessel diameter versus gas flow with gas pressure and velocity as parameters. The velocities that this graph is based on are not the only possible gas velocities, but they serve as a reasonable basis for the study. It is obvious from this graph that the rate of reduction in vessel diameter becomes smaller as the pressure is increased. At any gas flow rate the vessel diameter can be changed by changing the number of vessels in the plant. The reduction in vessel diameter as a function of the number of vessels can be seen in Figure A1-3. It can be seen that with more than 4 or 5 vessels the effect of increasing the number of vessels on the vessel diameter becomes less significant.

GAS FLOW VERSUS STEAM FLOW FOR COST STUDY

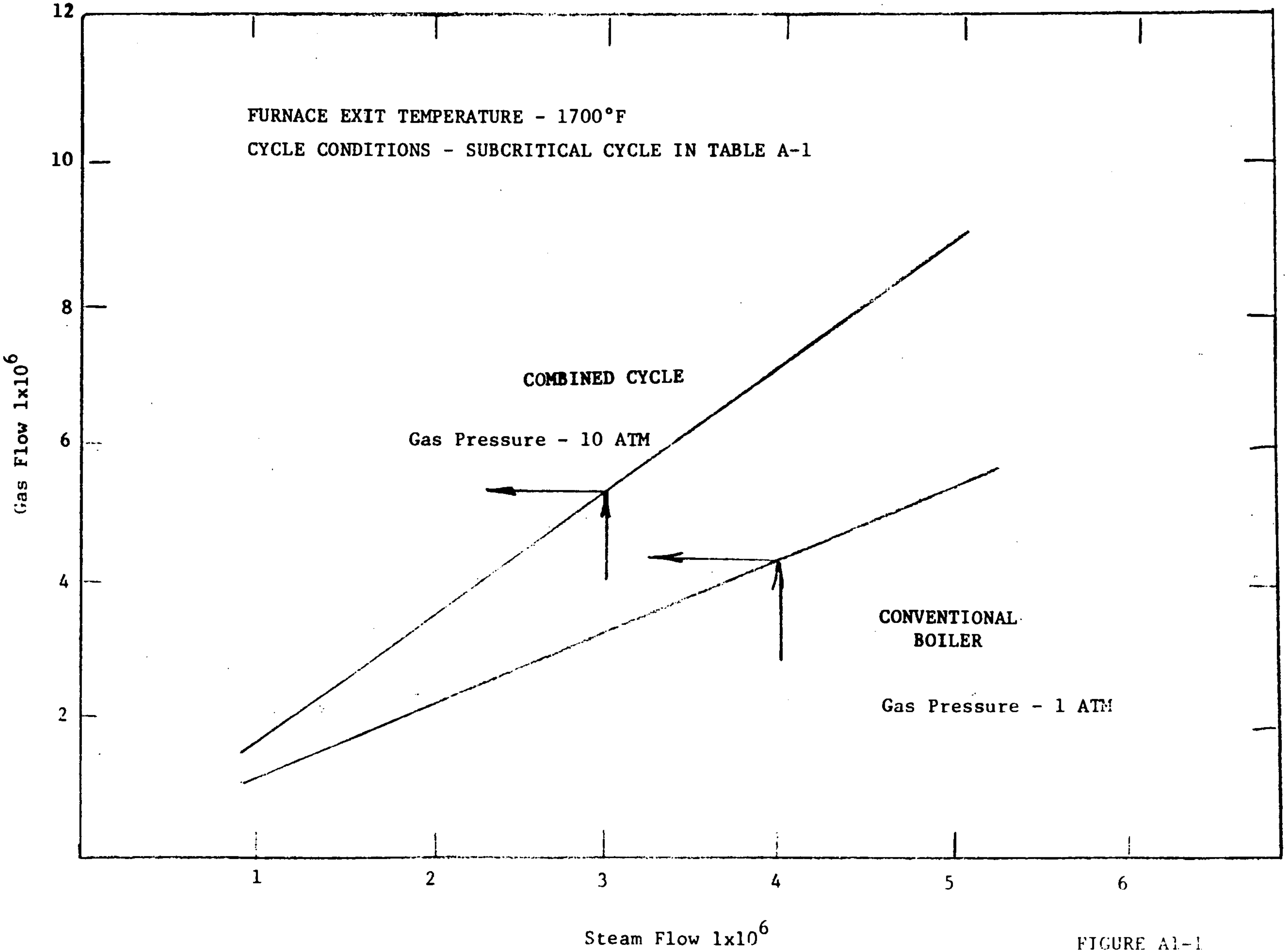


FIGURE A1-1

VESSEL DIAMETER VERSUS GAS FLOW

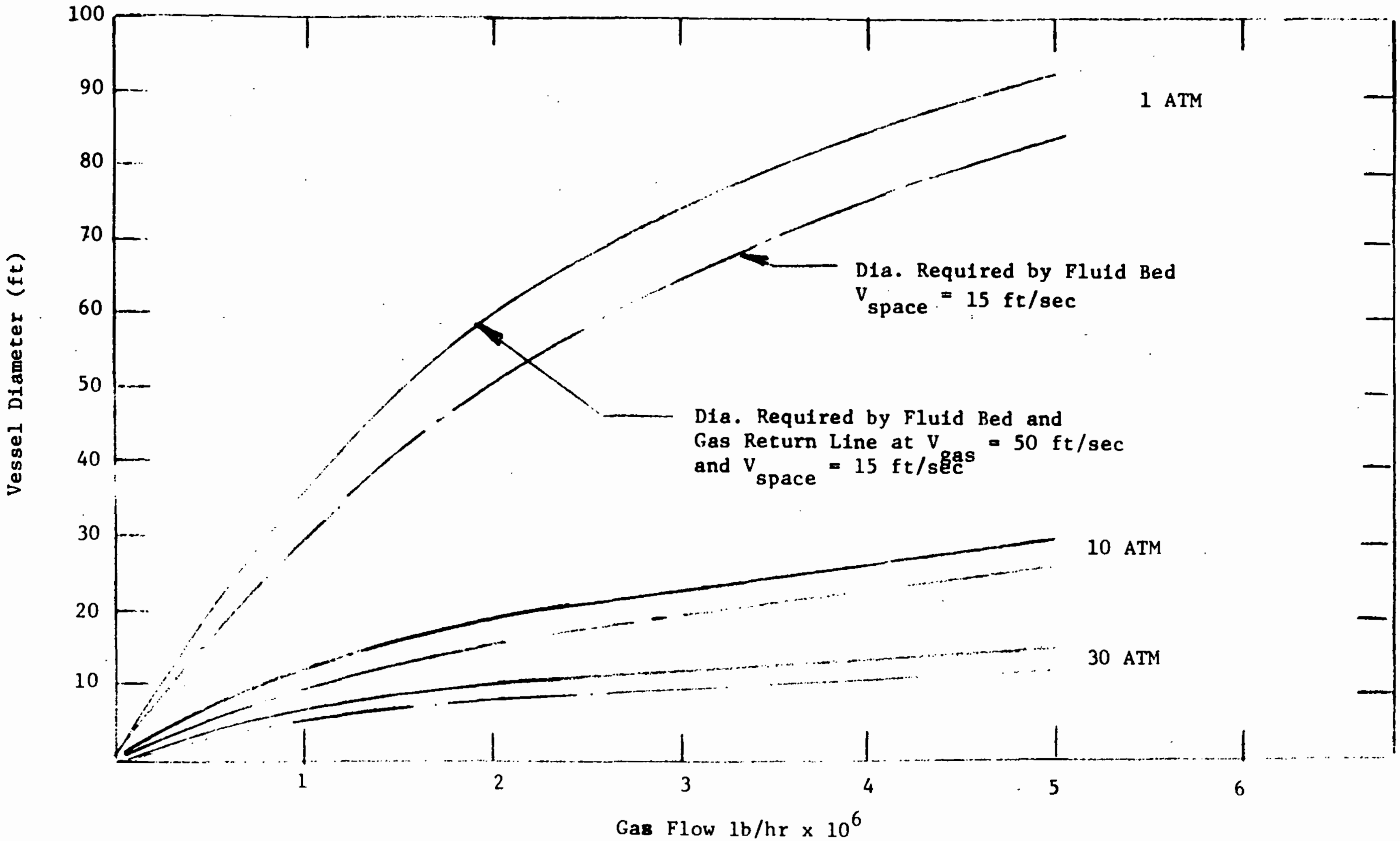


FIGURE A1-2

EFFECT OF THE NUMBER OF VESSELS SELECTED

ON VESSEL SIZE

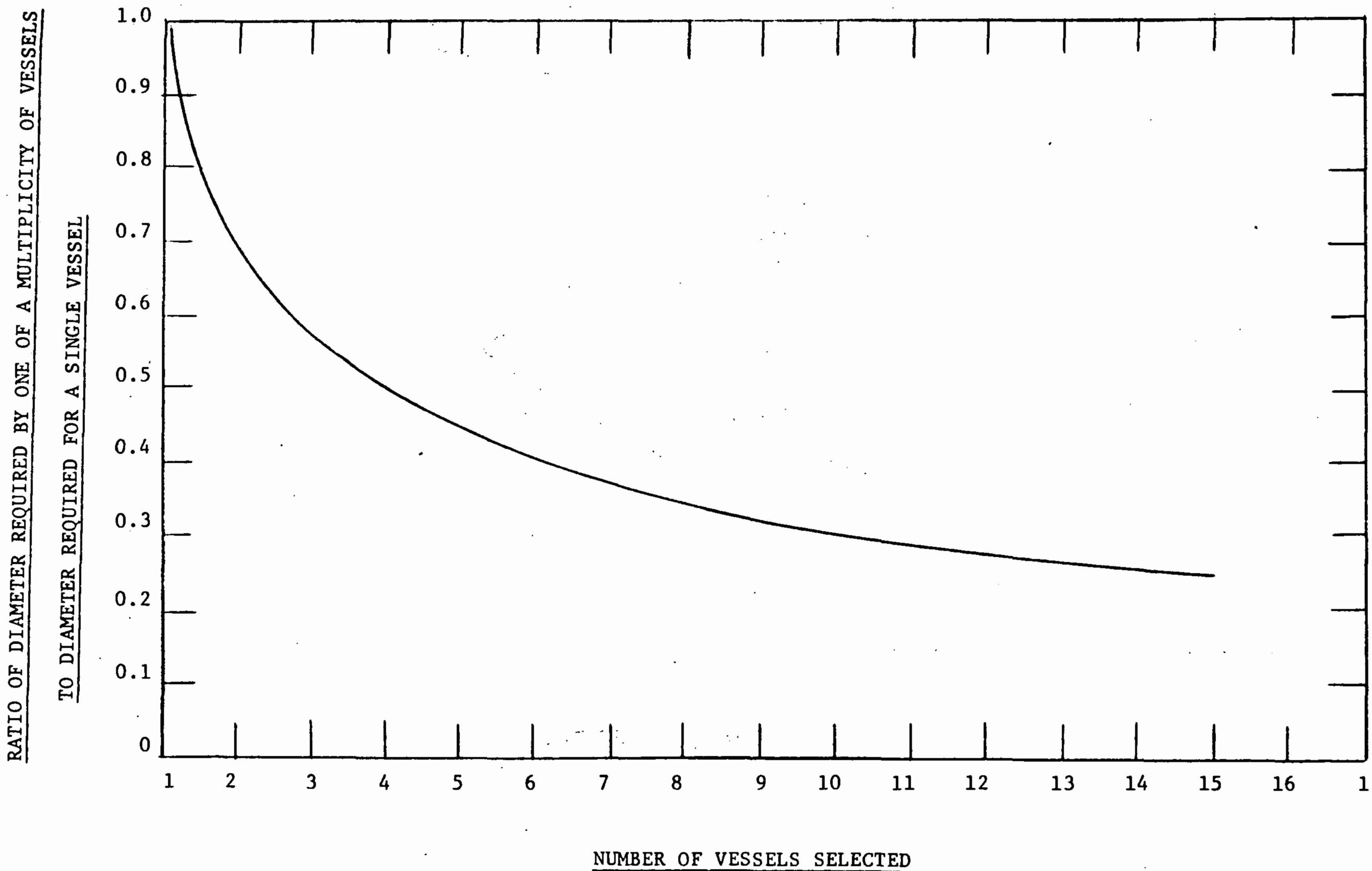
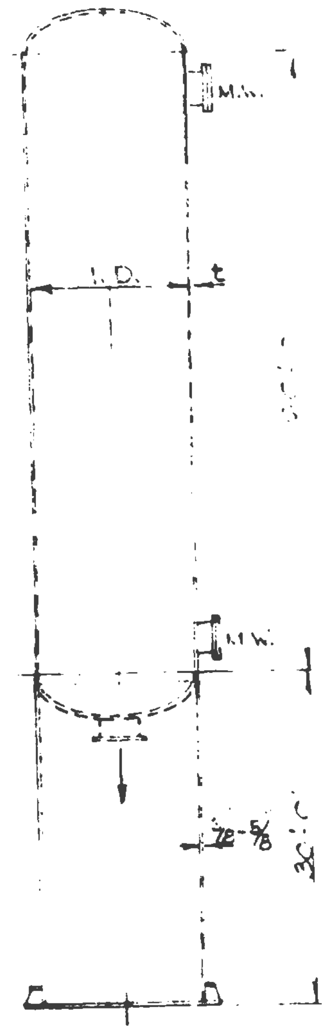


FIGURE A1-3

It should also be pointed out that other aspects of steam generator design are adversely affected with more than 4 or 5 modules. These problems are in the areas of control and auxiliary equipment, and it is not possible to quantitatively assess these affects without detailed studies. It is relatively simple, however, to quantitatively determine the effects of gas pressure, vessel diameter and vessel height on cost of the pressure vessel. This can be seen in Figures A1-4, A1-5 and A1-6. Figure A1-4 shows the type of vessel used in the cost study. It should be pointed out that if the curves are used to evaluate a multi-vessel system, the gas flow or steam flow per vessel should be used. Figure A1-5 shows the effect of gas pressure on the vessel cost for different vessel diameters. Figure A1-6 summarizes the pressure vessel cost study. This graph shows the relationship between cost and gas mass flow. This graph takes into account both changes in vessel height and diameter as the gas flow and pressure are changed. This is necessary because the tube spacings in the beds are independent of the system gas pressure. Therefore, even though for a given gas velocity the vessel diameter will decrease with increasing pressure, the vessel height must be increased to maintain the required heat transfer surface area. This curve is based on the same gas velocities as Figure A1-2 and the type of vessel shown in Figure A1-4. It is obvious from the plots in Figure A1-6 that the pressure vessel cost increases with the system pressure for a given size plant. The actual costs are only applicable for the specific gas velocities that were assumed, but the trend will be the same for any gas velocity. The difference



COMPONENTS

1 - 24" Gas Outlet
4 - 12" Air Inlets
4 - 6" Coal Inlets
2 - 24" Manways
2 - Heads
1 - Shell
1 - Skirt
1 - Base

SCHEMATIC DIAGRAM OF VESSEL USED IN COST STUDY

FIGURE A-4

VESSEL DIAMETER VERSUS COST

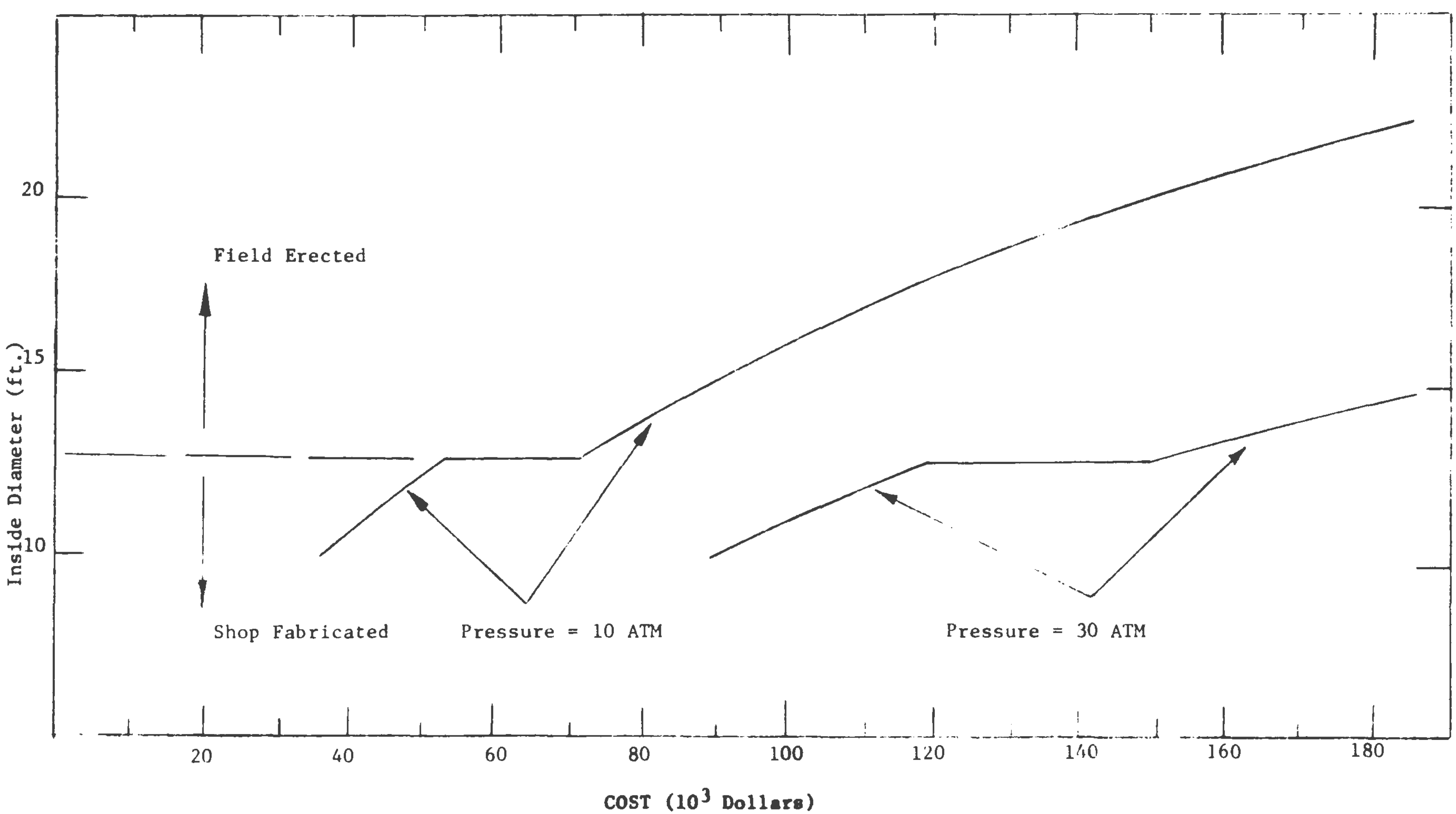


FIGURE A1-5

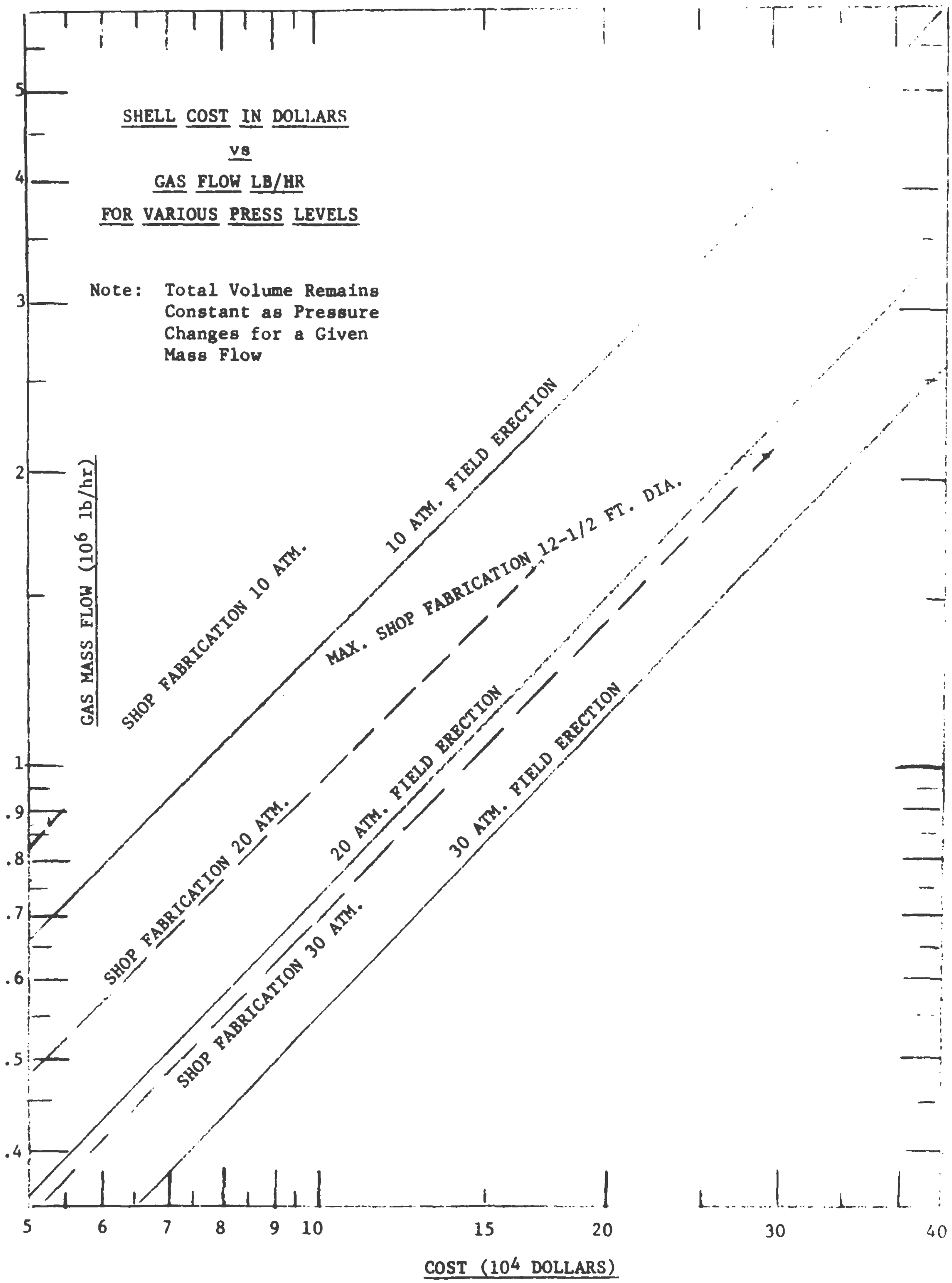

SHELL COST IN DOLLARS
vs
GAS FLOW LB/HR
FOR VARIOUS PRESS LEVELS
Note: Total Volume Remains Constant as Pressure Changes for a Given Mass Flow
GAS MASS FLOW (10^6 lb/hr)
COST (10^4 DOLLARS)
SHOP FABRICATION 10 ATM.
10 ATM. FIELD ERECTION
MAX. SHOP FABRICATION 12-1/2 FT. DIA.
SHOP FABRICATION 20 ATM.
20 ATM. FIELD ERECTION
SHOP FABRICATION 30 ATM.
30 ATM. FIELD ERECTION
5
4
3
2
1
.9
.8
.7
.6
.5
.4
5
6
7
8
9
10
15
20
30
40

FIGURE A1-6

in shell costs coupled with the anticipated increase in auxiliary equipment at pressure levels above 10 atmospheres indicates that ten atmospheres is probably the optimum pressure level for assumed gas turbine conditions.

A1.2 Determination of Optimum Design

Many factors have to be considered in choosing an optimum design for a fluidized bed steam generator. The requirements of fluidization, gas side boiler design and steam side boiler design have to be considered in evaluating any design. Besides the technical aspects of the design, the economic aspects of the different designs must be considered.

As a starting point the fluidized bed parameters can be divided into four broad categories:

1. Bed cross-sectional shapes
2. Bed heights
3. Arrangements of tubes in beds
4. Arrangement of beds in the vessel.

Each of these categories will be discussed separately, and the effects of the different possibilities in each category on steam side boiler design, gas side boiler design and economics will be evaluated.

A1.2.1 Bed cross-sectional shapes Beds of annular and rectangular cross section were considered feasible for the pressurized design. Just considering the shape of the bed per se neither design seems vastly superior to the other. At first glance, the annular shape seems to provide a more efficient use of space in a cylindrical pressure vessel. However, if space is required between the tube walls and the shell for piping or equipment, fully using vessel space by having an annular bed is no longer a benefit. The fabrication of annular beds may be more costly than rectangular beds due to more difficult tube wall fabrication. None of the above reasons are sufficient for choosing one bed shape as superior, but these considerations coupled with the tube arrangement restrictions of each shape influenced the final choice.

A1.2.2 Bed heights Three ranges of bed heights were considered. Shallow beds of less than 5 ft., medium beds of 5 ft. to 15 ft., and deep beds greater than 15 ft. were considered for the design. Shallow beds were ruled out for several reasons. First, shallow beds require many more coal injection points than deeper beds. In the pressurized design this multiplicity of coal injection points is a problem for two reasons. First, space is at a premium inside the pressure vessel, and the coal injection lines could take up a significant amount of space in the vessel. Also the increased number of vessel penetrations caused by the coal pipes makes the expansion problem more serious. Shallow beds also require more stacked beds than a deep bed design. The extra tube banks that result from increase in the number of beds require more headers, transport piping, and accessories such as air dampers and ignitors. Shallow beds do have the advantage of having a lower pressure drop than deeper beds, but in a pressurized-combined cycle design the pressure drop across the bed is not the most important design consideration. A more important consideration is to have most of the heat transfer surface in the uniform temperature fluidized bed. For a given upper limit on bed temperature, this achieves the highest possible gas turbine inlet temperature because any heat transfer outside the bed will lower the gas temperature. It is obvious that this situation can be more easily attained with deeper beds.

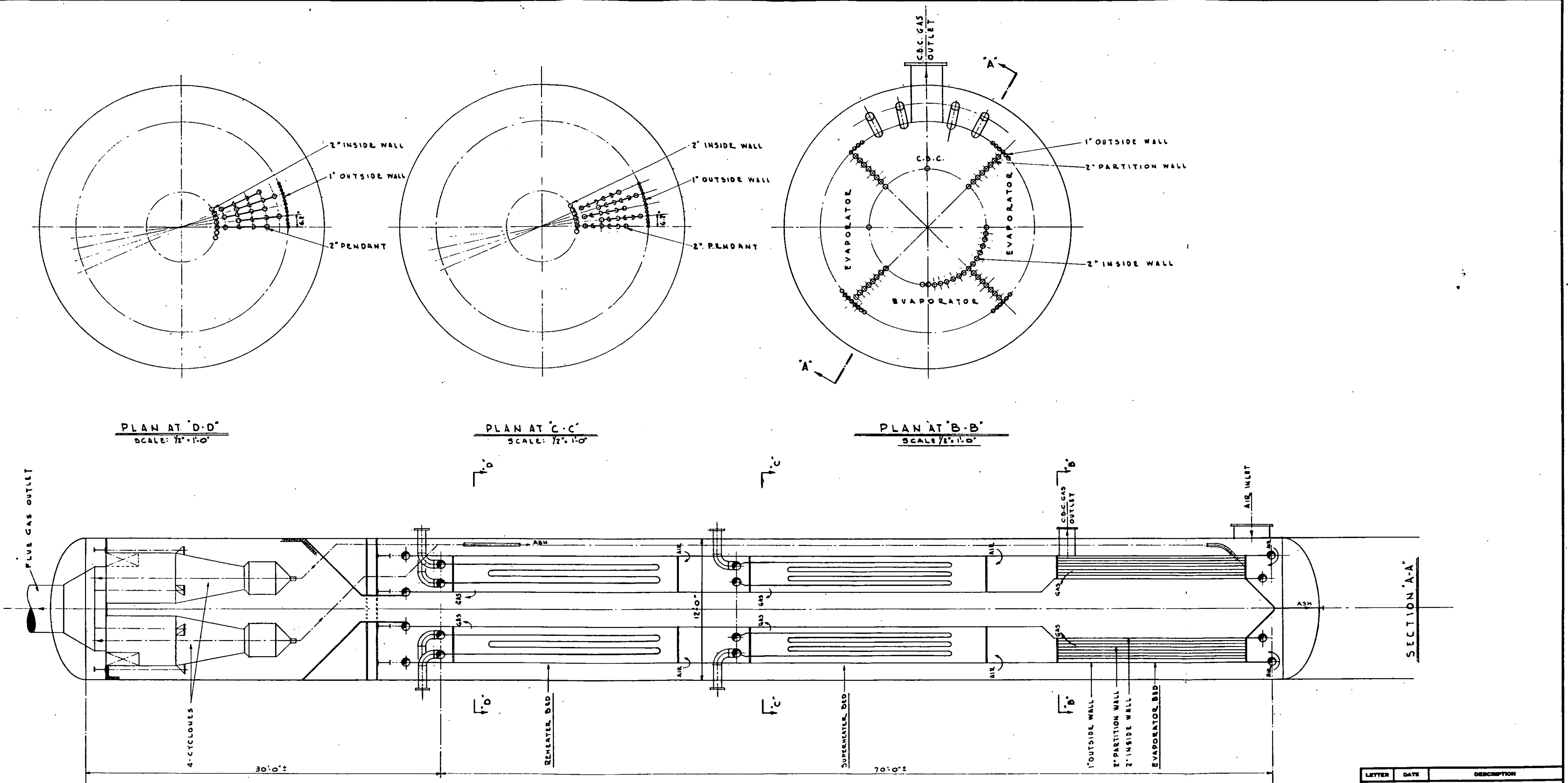
Beds deeper than about 15 ft. also have many problems associated with them. Certain problems have to do with how the beds can be located in the pressure vessels, and how the different boiler functions (superheat, reheat, etc.) can be distributed among the steam generator modules. These problems will be discussed later. Certain problems, however, result just from the properties of a very deep bed. The deeper the bed, the more difficult the ignition problem becomes. This is because more dolomite and heat transfer surface must be brought up to the ignition temperature of the coal. There is also a problem that results from air to fuel ratio and fluidization velocity requirements. The velocity requirements for fluidization can be seen in Figure 2.2. This curve gives a minimum and a maximum air mass flow rate for a given bed temperature. The fuel input requirement is a function of the amount of heat transfer surface that is located in the bed. As the bed height is increased the fuel requirement increases, and the air flow requirement increases. If the cross-sectional area of the bed is restricted a bed height will be reached where the air flow requirement will yield a superficial velocity that is greater than the maximum allowable superficial velocity. The range of bed cross-sectional areas that can be achieved is restricted in the pressurized deisgn due to pressure vessel size limitations. Also, although pressure drop across the bed is not critical in the pressurized design, it is an energy loss, and should be weighed against any benefits of deep beds.

Bed depths at the high end of the middle range seem best suited for the design requirements of the pressurized steam generator. These depths coupled with bed cross-sectional areas that can be reasonably fit into a pressure vessel result in allowable superficial velocities. It is felt that the simplicity that is derived by having fewer beds of this depth rather than more shallow beds overrides the disadvantage of increased pressure drop.

A1.2.3 Arrangement of tubes in the bed Four types of tube bundles were considered for the steam generator. These were helical tube bundles, horizontal serpentine tube banks, straight vertical tubes and vertical pendant tube elements. At first glance it seems that helical tube bundles in annular beds is a good design because of the efficient use of vessel space that is achieved. However, this design has many inherent problems in steam side design, construct and maintenance. In designing a tube bundle the mass flow rates in the tubes must be within a certain range of values so that the pressure drop will be acceptable and the distribution of the flow among the tubes of a bank should be as even as possible. In order to fulfill the first design criterion, it is required to have some flexibility in the number of tubes that can be located in a type of tube bundle. Also, in order to have equal flow distribution and heat absorption in all the tubes of the bank the tube lengths must be approximately equal. It is difficult to fulfill both of these design requirements with a helical tube bundle. In order to get equal tube lengths in the bundle the number of tubes and the pitch of the tubes at a given radius from the center of the bundle are restricted. This combined with the limits of bed cross-sectional area and tube sizes provides severe restrictions on the steam flow rates tha can be achieved. It is also felt that the construction of the tube bundles though not impossible, would be difficult and costly.

Another type of tube arrangement that could be used with an annular bed is vertical pendant elements. A design using this type of bed is shown in Figure A1-7. This design provides more flexibility in the steam side design than helical tube bundles, but the fluidization characteristics of the bed may not be very good. Because of problems with supporting and locating the headers, the only practical arrangement of the pendant elements in an annular bed is a radial layout. This presents problems, however, because this layout results in uneven spacing of the tubes in the bed. This characteristic of the bed may result in channeling of the flow, spouting of the bed, or the development of hot spots in the bed.

Straight vertical tubes could be placed in either annular or rectangular beds and there is a lot of flexibility in the tube spacings that can be achieved. The main problem with vertical tubes is the large amount of headers and transfer pipes that are required. The number of individual tubes required to fill a bed is much greater with straight vertical tubes than with any of the other configurations. This can readily be seen in Figure A1-8. Arranging these tubes in one pass would result in a very small steam side velocity in the tubes. To divide the tubes into a sufficient number of circuits requires a large number of headers and



LETTER	DATE	DESCRIPTION
REVISIONS		

ANNULAR BED -
VERTICAL TUBE
FLUIDIZED BED
STEAM GENERATOR

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FIG. A1-7

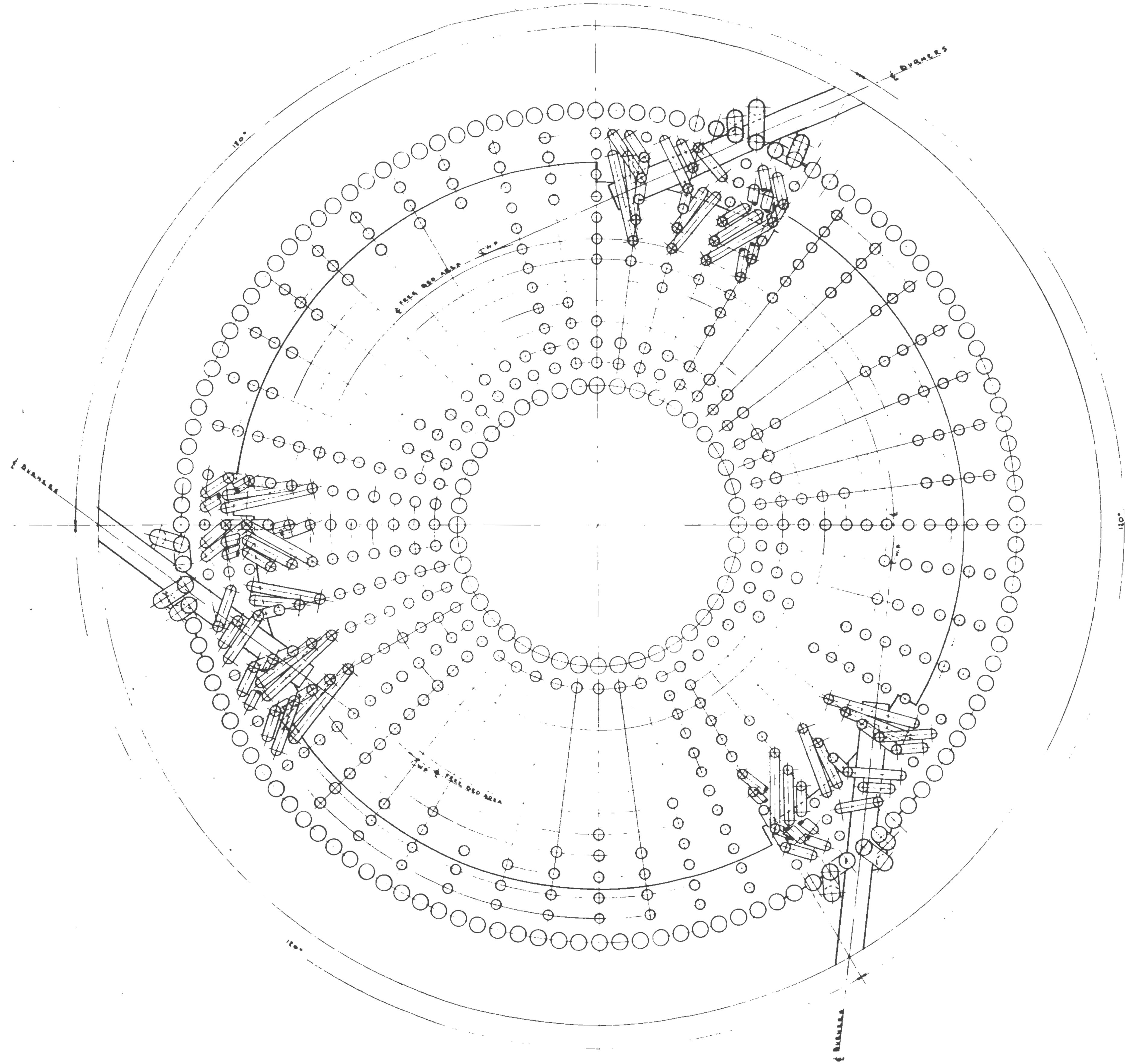

FIG. A1-8

NOTES

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REVISIONS

LAYOUT OF VERTICAL TUBES IN AN ANNULAR BED

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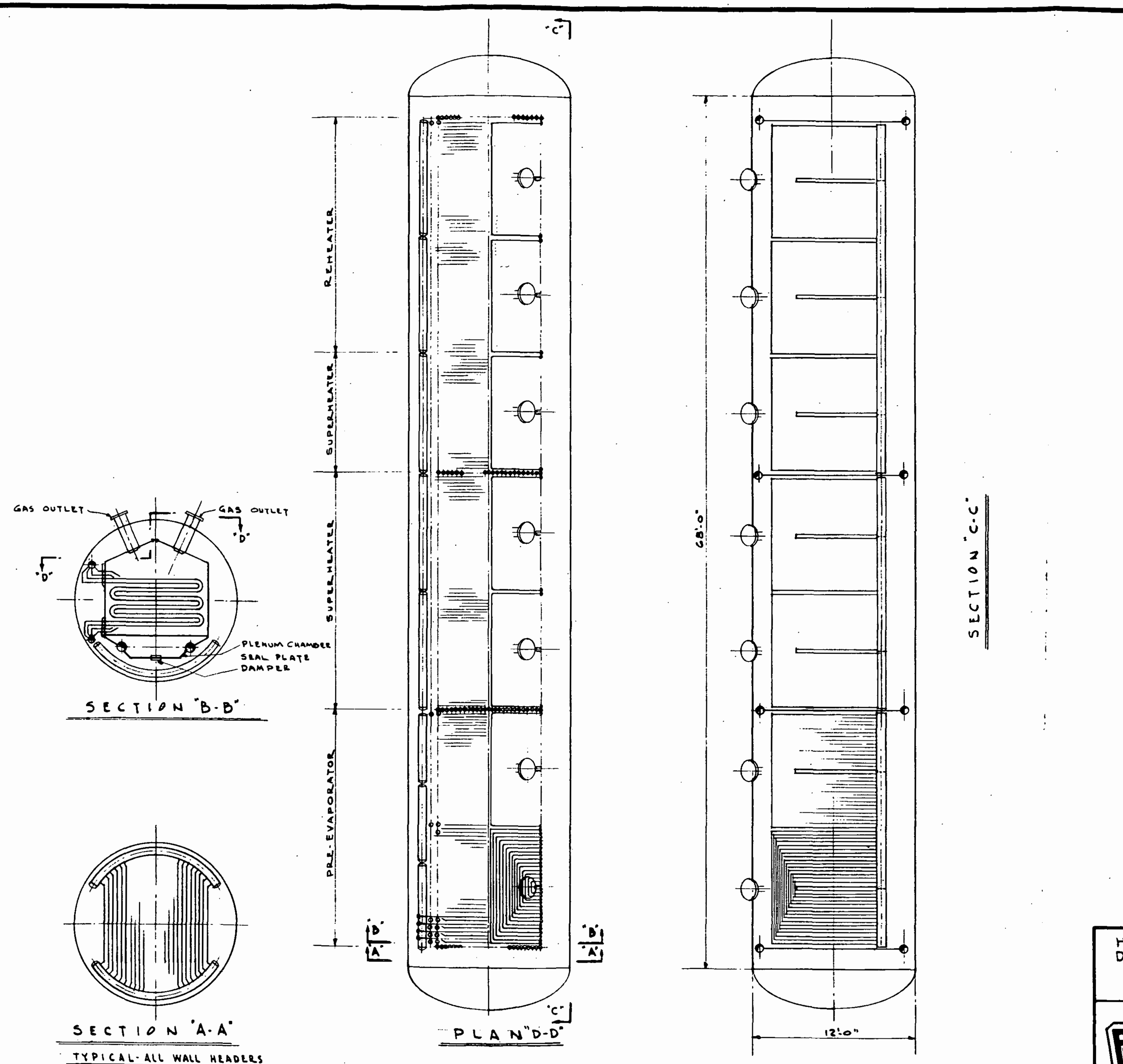
large number of transport pipes.

A1.2.4 Number of vessels and arrangement of beds in the vessel It was determined that a 12 ft. diameter vessel would be the optimum size for a 300 MW plant. This diameter vessel could be shipped in long sections by rail. The maximum shipping length would, of course, depend on the exact location of the proposed plant, but for most locations a 12 ft. diameter vessel could be shipped as long as 120 feet. With a 12 ft. diameter vessel having a length in the range of 120 ft., a 300 MW plant requires four modules. Besides the economic benefits of a plant consisting of shop assembled modules, there are alos operational benefits that result from a modular design. One beneficial aspect of a modular plant is that the turn-down ratio required of each module is less than would be required from a single steam generator unit. This mode of operation is discussed in more detail in Section 5 of the report. Also, plant availability increases with the number of modules. A drawback to a modular plant is that as the number of modules in a plant increases, the problem of distributing the fuel, air and feedwater flows between the modules becomes more complex. Both of these apsects of operation have to be considered in determining the optimum number of modules in a plant.

A four module plant is a good compromise between the two criteria discussed above. The resulting module turn-down requirements are such that turn-down can be accomplished by reducing the bed temperature. If a greater turn-down ratio were required, the bed depth would have to be lowered to achieve turndown. Lowering the bed depth to change load would require a more complex dolomite recirculation system and would probably result in a slower load response time. The problem of distribution of the various flows between the boiler modules is not unrealistic in a four module plant, so a four module plant is a good design from an operational point of view. At 300 MW a four module plant also has the economic benefit of shop fabrication and assembly since the resulting modules are 12 ft. in diameter and approximately 120 ft. long.

The arrangement of the fluidized beds in the vessel is another aspect of the design that must be analyzed. Basically there are two possibilities. One would be a vertically oriented vessel with stacked fluidized beds and the other would be a horizontally oriented vessel with the fluidized beds located adjacent to each other. A horizontal arrangement is shown in Figure A1-9. There are several drawbacks to a horizontal arrangement. One problem is that bed height is severely limited. The steam generator in Figure A1-9 was designed with a 12 ft. diameter vessel to allow shop fabrication and assembly. The resulting bed height was only 4 ft. and even with this low a bed height the particle disengaging zone is not as deep as it should be. Another problem with the horizontal design is the location of evaporation surface. Upward flow in vertical tubes is the preferred design for evaporation circuits. This type of an arrangement prevents problems which arise from water and steam separation in the evaporation circuit. In a once-through steam generator evaporator tubes can be oriented in other positions, but the mass flow in the tubes must be maintained at a high rate. Also, walls of bent tubes are more expensive to fabricate. In the horizontal steam generator the



HORIZONTAL CONCEPTUAL DESIGN FOR A PRESSURIZED FLUIDIZED BED STEAM GENERATOR

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FIG. A1-9

0-59-1192

walls are relatively short, and so straight upward flow evaporator sections require many headers and downcomers. As can be seen in Figure A1-9, the arrangement of the beds in a horizontal vessel does not leave much room for the location of headers and downcomers. Because of these space limitations the tube walls shown in this design would probably have to be used in spite of the drawbacks discussed previously. The flue gas passages in the horizontal steam generator are also a problem. If the gases leave the vessel at one location, there will be gas flow across the top of some of the beds. This type of a flow is very likely to cause uneven fluidization of the bed. To overcome this it was decided to have separate gas ducts leaving each bed. This design will require a manifold system exterior to the vessel.

A vertically oriented pressure vessel eliminates many of the problems that are encountered in the horizontal arrangement. The bed heights are not restricted by the orientation of the vessel and the resulting shape of the tube walls is more reasonable for use as evaporating surface. Also the space that exists between the fluidized bed cells and the pressure vessel shell in the vertical design is more adaptable to transfer piping.

After analyzing all the aspects of steam generator design and operation presented in this appendix, the detailed design for a 300 MW plant was developed. This design is shown in Figures 3.1, 3.2, 3.3 and 3.4 of the main report. A vertically oriented pressure vessel is used with vertically stacked beds. This layout allows the use of optimum bed heights and tube orientations. Four of these modules are used in a 300 MW plant which is the number that resulted as optimum from the pressure vessel study and the operational analysis. This design is discussed in detail in the remaining appendices.

APPENDIX 2

ENERGY AND MASS BALANCES

The energy and mass balances for the steam generators are summarized in the tables and figures of this appendix. The assumptions used and the methods of calculation are discussed in Section 4.2 of the text.

TABLE A2-1

FLUIDIZED BED PARAMETERS

DESIGN LOAD

(1 MODULE)

	Fuel Flow (lb/hr)	Air Flow (lb/hr)	Flue Gas (lb/hr)	Superficial Velocity	Bed Temperature
Pre Evaporator	17,680	192,100	206,200	9.0	1,750
1st Superheater	13,100	142,700	153,700	6.8	1,750
2nd Superheater	10,740	116,900	125,900	5.6	1,750
Reheater	12,390	134,800	145,200	6.4	1,750
CBC	6,494[1]	42,600	44,700	6.1	2,000

GAS TEMPERATURE LEAVING BOILER[2] - 1650°F

1 31.7% Carbon
63.3% Ash
5.0% Dolomite

2 Includes CBC Gas

TABLE A2-2

FLUIDIZED BED PARAMETERS

70% PLANT LOAD

(1 MODULE)

	Fuel Flow (lb/hr)	Air Flow (lb/hr)	Flue Gas (lb/hr)	Superficial Velocity	Bed Temperature
Pre Evaporator	12,600	199,200	209,200	8.7	1,470
1st Superheater	8,950	141,200	148,600	6.3	1,470
2nd Superheater	7,370	116,300	122,400	5.2	1,470
Reheater	9,680	152,200	160,800	7.0	1,530
CBC	4,650[1]	20,200	21,700	3.2	2,000

GAS TEMPERATURE LEAVING BOILER[2] - 1400°F

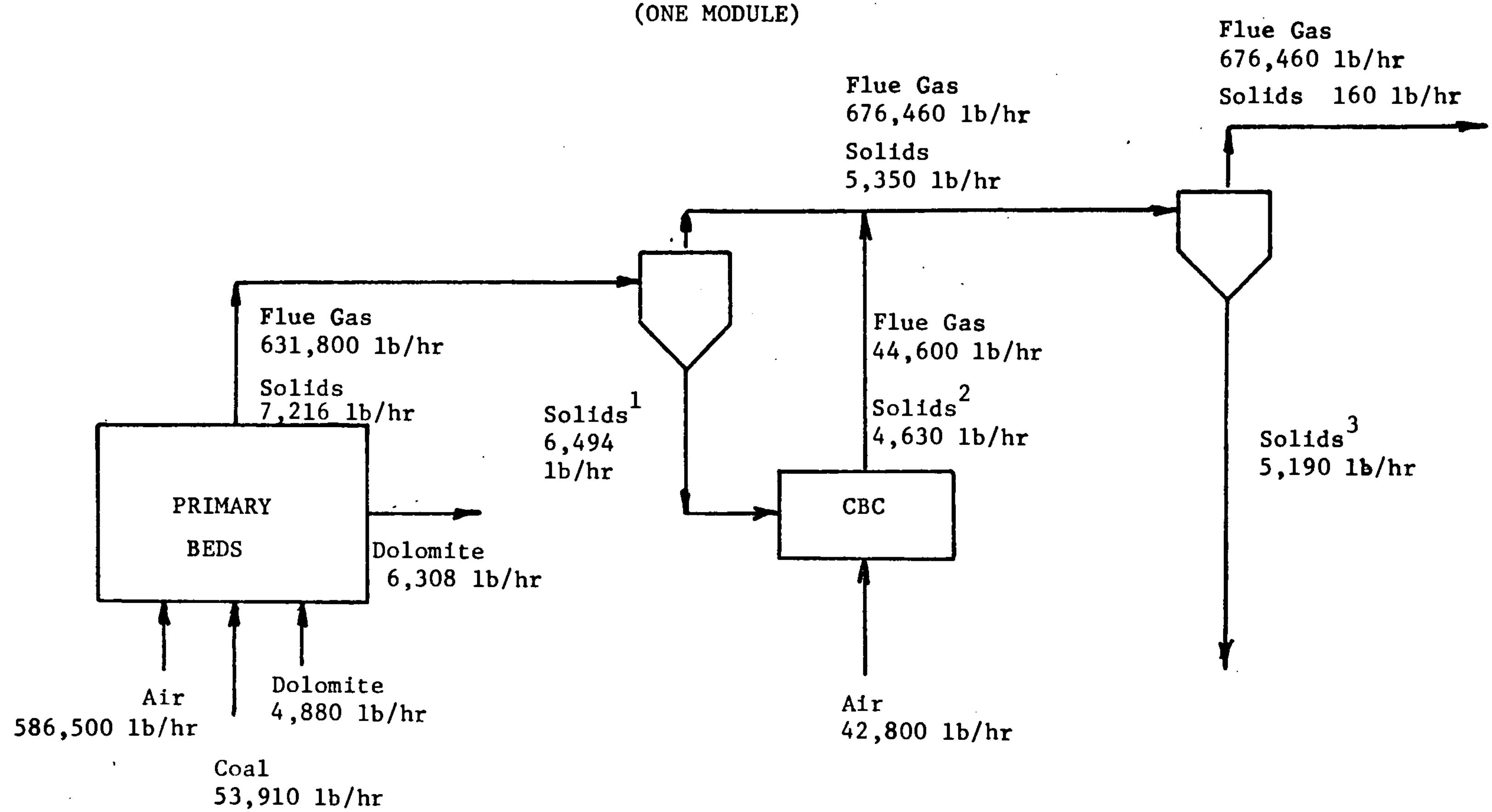
1 31.2% Carbon
62.1% Ash
6.7% Dolomite

2 Includes CBC Gas

MASS BALANCE

DESIGN LOAD

(ONE MODULE)



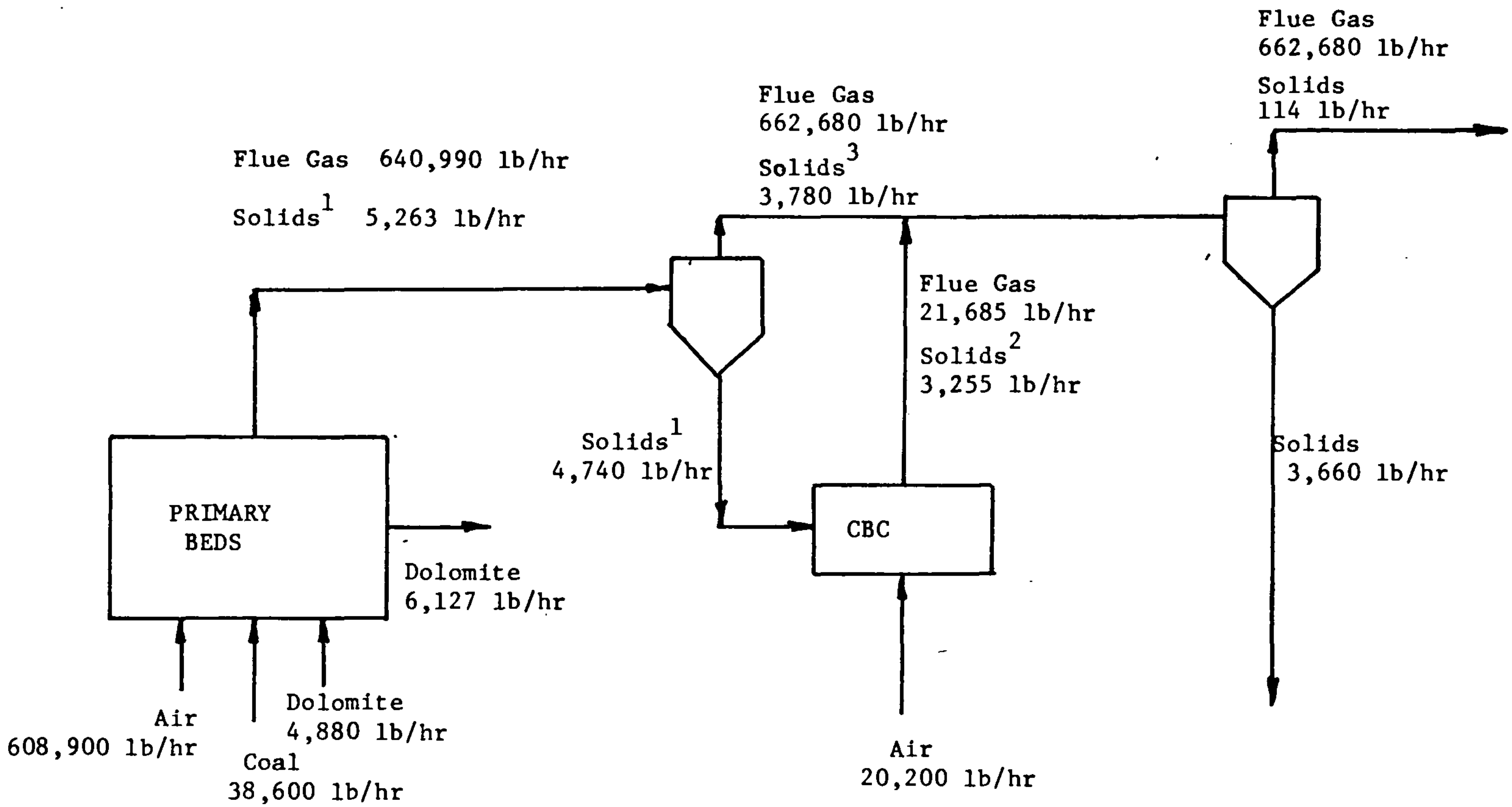
	1		2		3
Carbon	31.9%	Carbon	4.5%	Carbon	8.2%
Ash	63.4%	Ash	89.0%	Ash	85.8%
Dolomite	4.6%	Dolomite	6.5%	Dolomite	6.0%

FIGURE A2-1

MASS BALANCE

70% PLANT LOAD

(ONE MODULE)



[1] Carbon	31.2%	[2] Carbon	4.3%	[3] Carbon	7.0%
Ash	62.1%	Ash	86.4%	Ash	83.9%
Dolomite	6.7%	Dolomite	9.3%	Dolomite	9.1%

FIGURE A2-2

ENERGY BALANCE

DESIGN LOAD

(ONE MODULE)

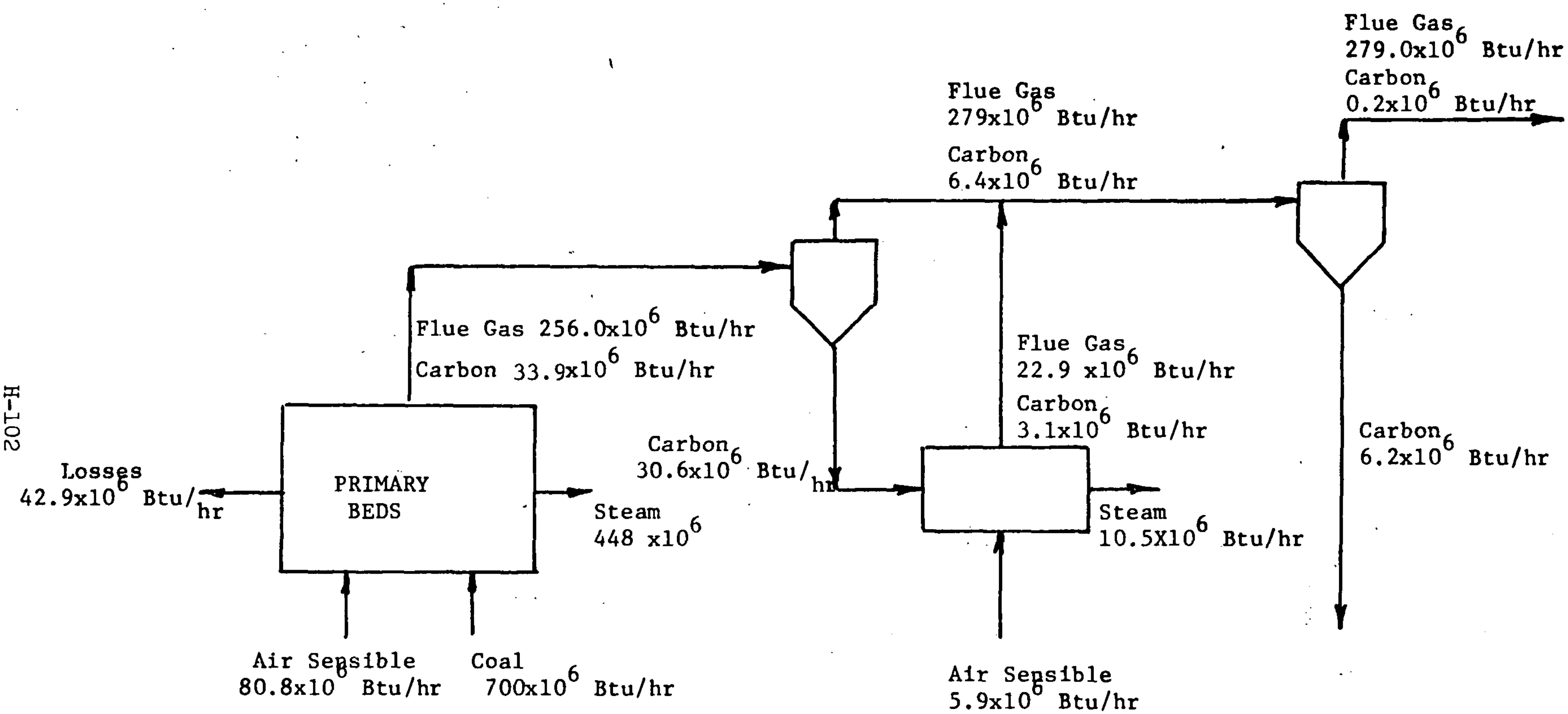


FIGURE A2-3

ENERGY BALANCE

70% PLANT LOAD

(ONE MODULE)

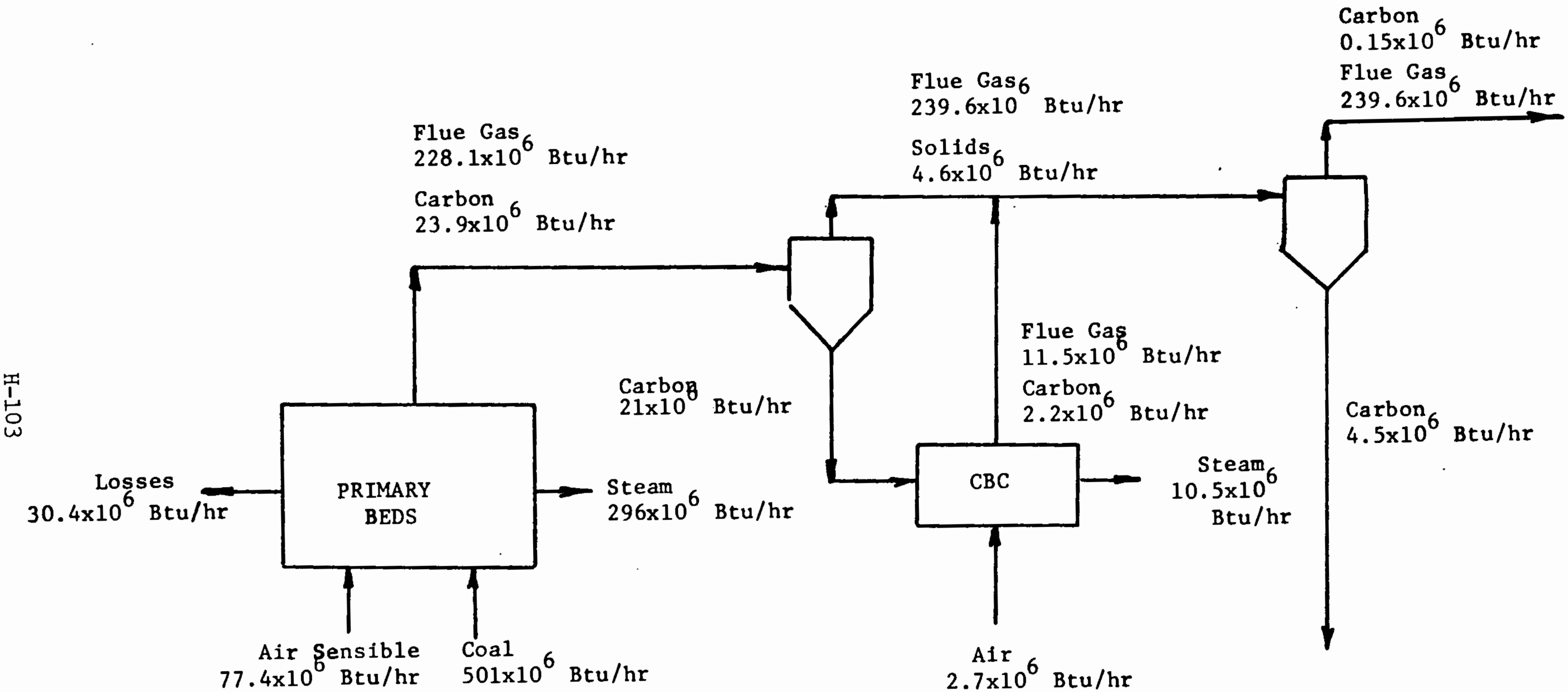


FIGURE A2-4

TABLE A2-3

STEAM GENERATOR LOSSES BASED ON

275°F EXIT GAS TEMPERATURE

Dry Gas		3.88%
Hydrogen and Moisture in Fuel		4.14%
Moisture in Air		0.08%
Unburned Combustible		1.51%
Radiation		0.15%
Sensible Heat of Solids		0.11%
Unaccounted for Losses and Manufacturers Margin		1.50%
	TOTAL	11.37%

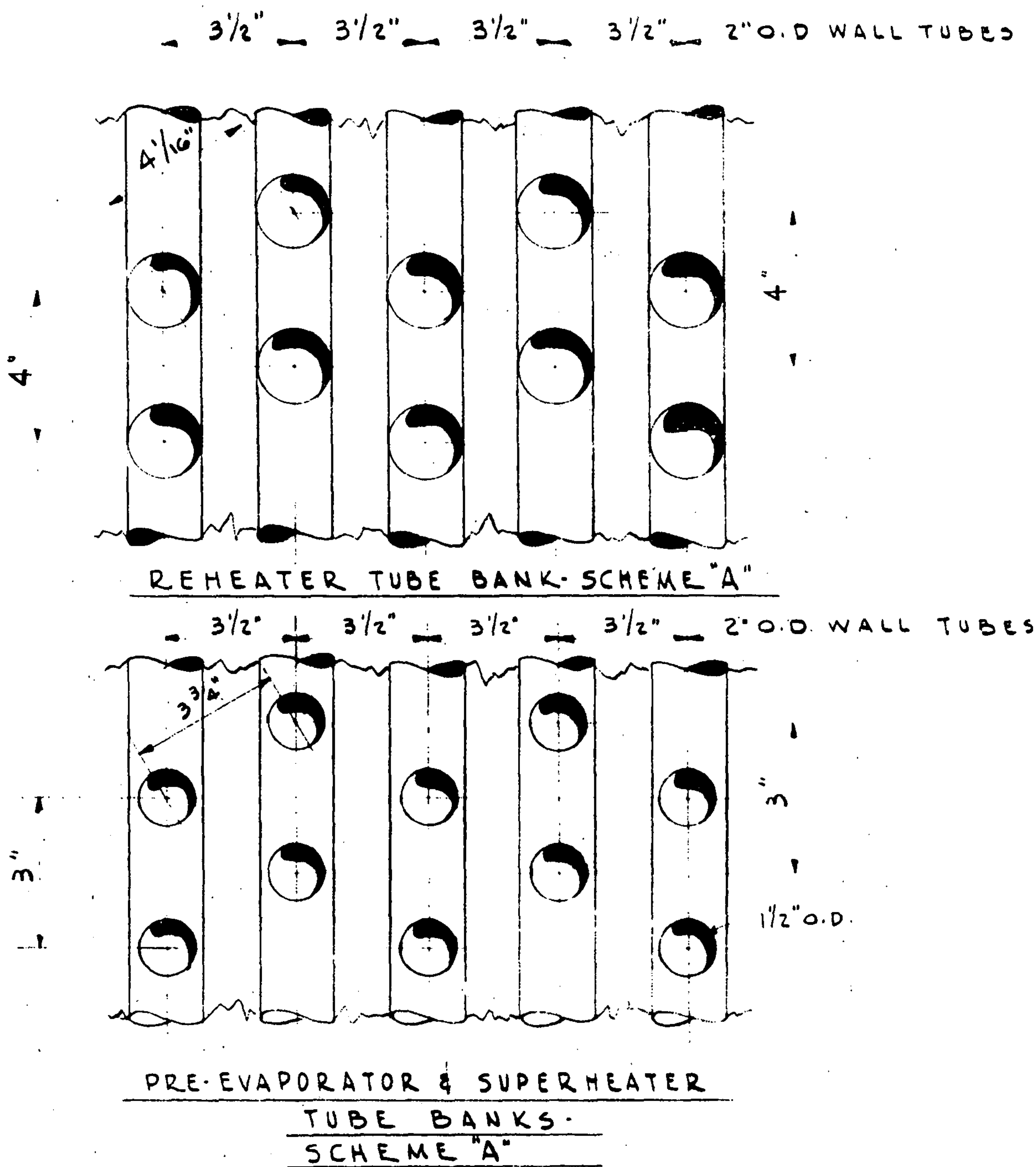
APPENDIX 3

TUBE DESIGN INFORMATION

A3.1 Sizing and Arrangement

One factor that contributed to the choice of a horizontal tube stacked bed design as optimum was that the tube arrangements and tube support systems allow for conventional fabrication methods. Structurally the arrangement of pressure parts is very similar to the convection pass of conventional boiler. The tube walls are supported at the upper headers and allowed to expand downward during operation. The walls in turn support the tube bundles. The tubes in the tube bundles can be bent on bending machines, and the tube walls can be welded into panels by automatic equipment.

The tube bundles consist of serpentine elements spaced across the width of the fluidized bed. The configuration of the tubes in the bed is an important design consideration from the standpoint of fluidization. There is still much debate about what tube configuration gives the best fluidization, but the design presented in this report has the benefit of being quite flexible in this regard. By changing the tube spacing on the tube walls and the bend radii of the serpentine elements, the configuration of the tubes in the bed could be changed without changing the basic design. The present tube configurations are shown in Figure A3-1. Other possible tube layouts are shown in Figures A3-2 and A3-3. The cost difference between the tube banks in the present design and the alternative



TUBE CONFIGURATIONS IN THE FLUIDIZED BEDS	ORDER NO	0-59-1192	
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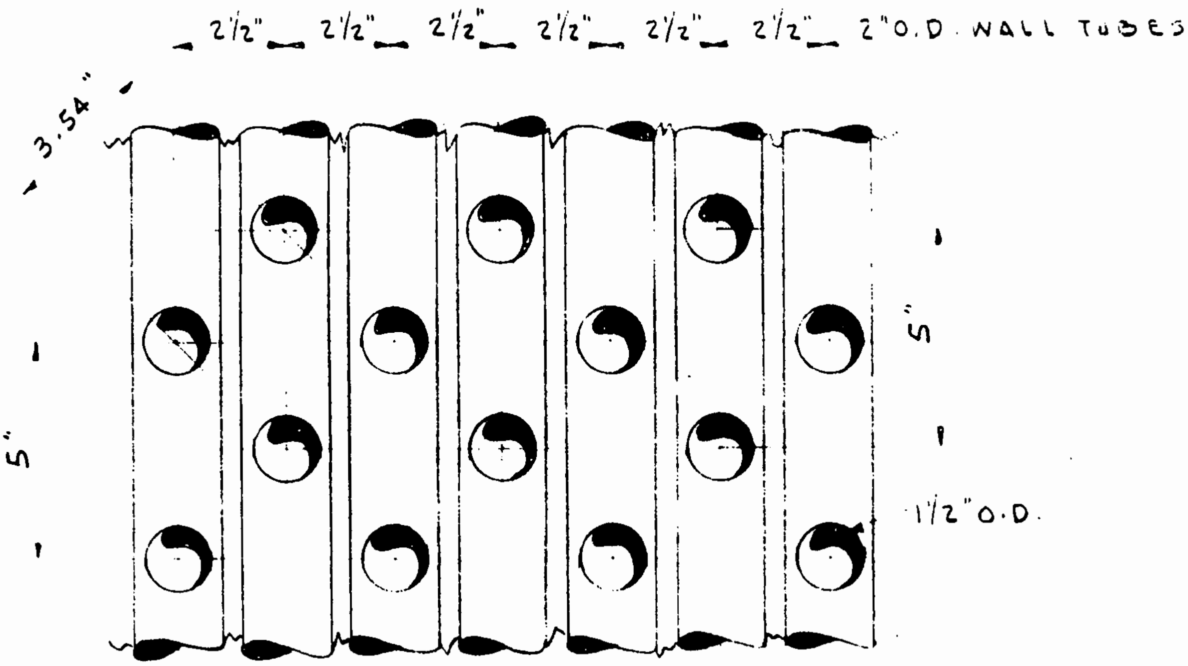
FIG A3-1



PRE-EVAPORATOR & SUPERHEATER
TUBE BANKS
SCHEME "B"

TUBE CONFIGURATIONS IN THE FLUIDIZED BEDS

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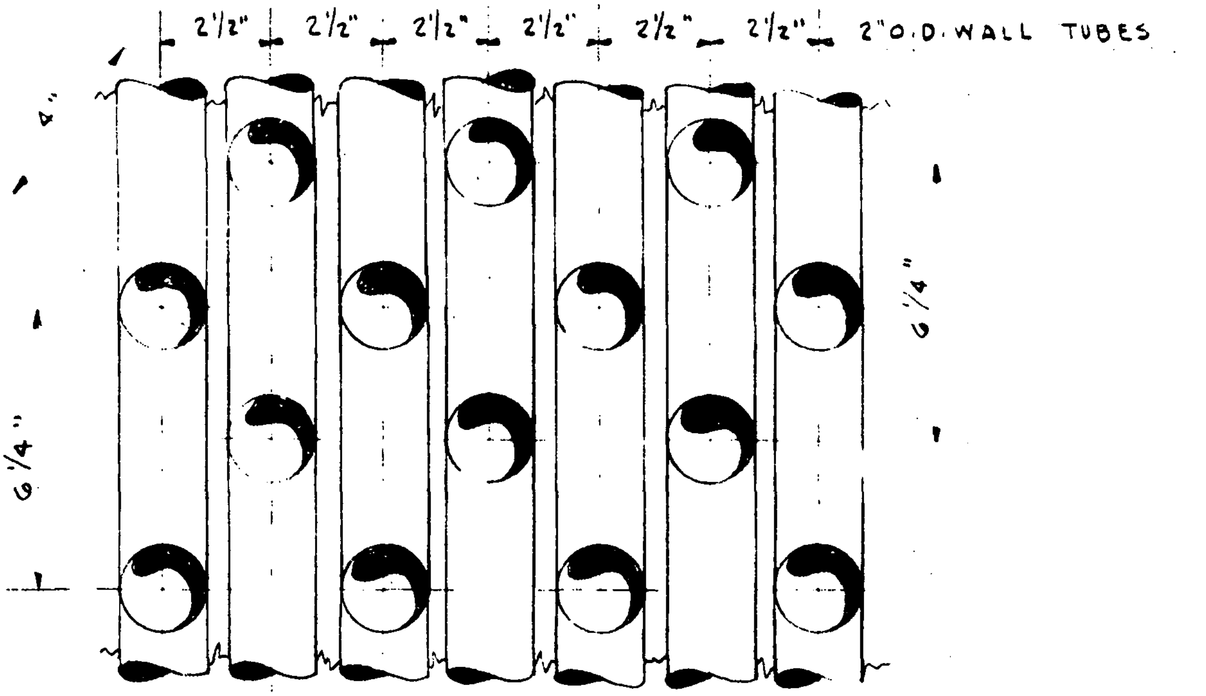
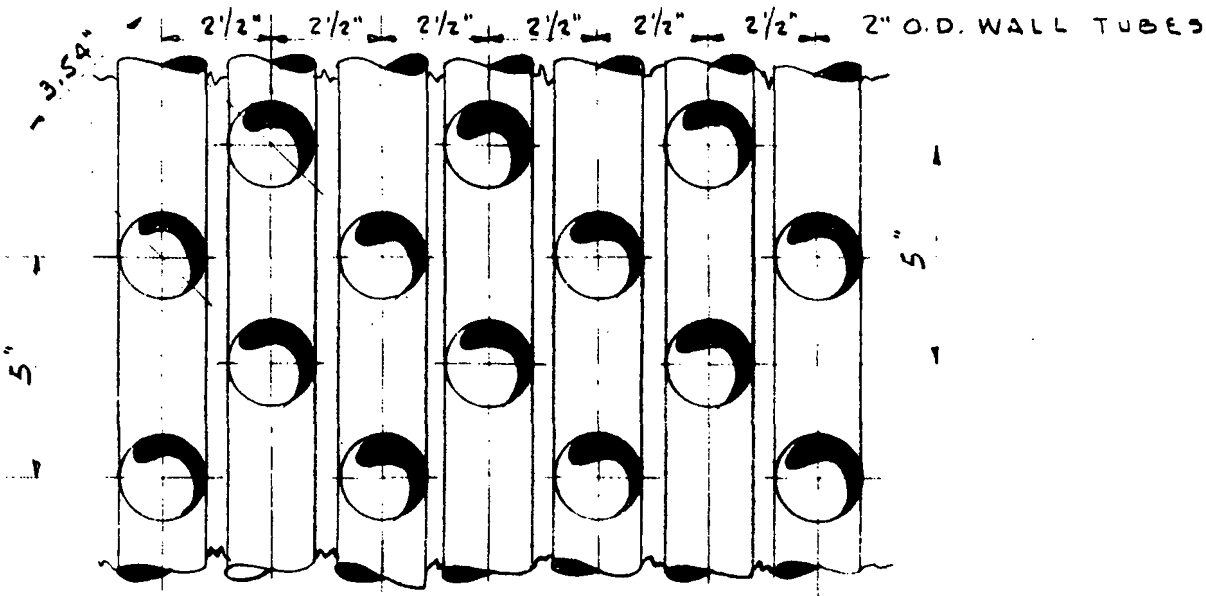
FIG A3-2



REHEATER TUBE BANK - SCHEME "B"



REHEATER TUBE BANK - SCHEME "C"

TUBE CONFIGURATIONS IN
THE FLUIDIZED BEDS

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FIG A3-3

arrangements would be negligible.

All of the tube bundles consist of 23 serpentine elements spaced across the bed. The pre-evaporator and superheater tubes are 1-1/2" O.D. The reheater tubes are 2" O.D. From the standpoint of fluidization, economy of space could be achieved by using the smallest diameter tubes possible, but both steam flow requirements and structural problems had to be considered in choosing the tube sizes. For example, if the diameter of the reheater tubes was reduced, steam flow requirements would dictate an increase in the number of tubes in each loop. It was felt that in a bed of these dimensions more than four tubes in a loop would be impractical. The choice of tube size for the pre-evaporator and superheat beds was mostly structural. It was felt that the tubes in these beds should not be reduced below 1-1/2" O.D. due to the possibility of tube vibration in the beds. A listing of tubing and header specifications is presented in Table A3-1. Also included in this list is the maximum mean metal temperature for the various tubes.

In each fluidized bed a bed-to-tube heat transfer coefficient of 50 Btu/hr-ft^2-°F was used for the bed proper. A zone extending two feet above the bed was considered to be a transition zone between fluidized bed heat transfer and gas convection. The mechanism of heat transfer in this zone would mainly be particles that rise above the relatively dense phase of the bed, transfer heat, and then fall back into the bed. Here, a bed-to-tube heat transfer coefficient of 40 Btu/hr-ft^2-°F was assumed and the zone was considered to be at the same temperature as the bed. The overall

TABLE A3-1

TUBE AND HEADER SPECIFICATIONS

TUBES

	SIZE	MATERIAL	DESIGN MEAN METAL TEMPERATURE
Water Walls	2" O.D. x 0.280" M.W.	SA-213-T22	975
Pre-evaporator	1-1/2" O.D. x 0.150" M.W.	SA-210-A1	732
Lower Superheater	1-1/2" O.D. x 0.165" M.W.	SA-213-T2	900
Upper Superheater (loops 1-7)	1-1/2" O.D. x 0.318" M.W.	SA-213-T22	1058
Upper Superheater (loop 8)	1-1/2" O.D. x 0.238" M.W.	SA-213-TP304H	1150
Reheater	2" O.D. x 0.186" M.W.	SA-213-T22	1122

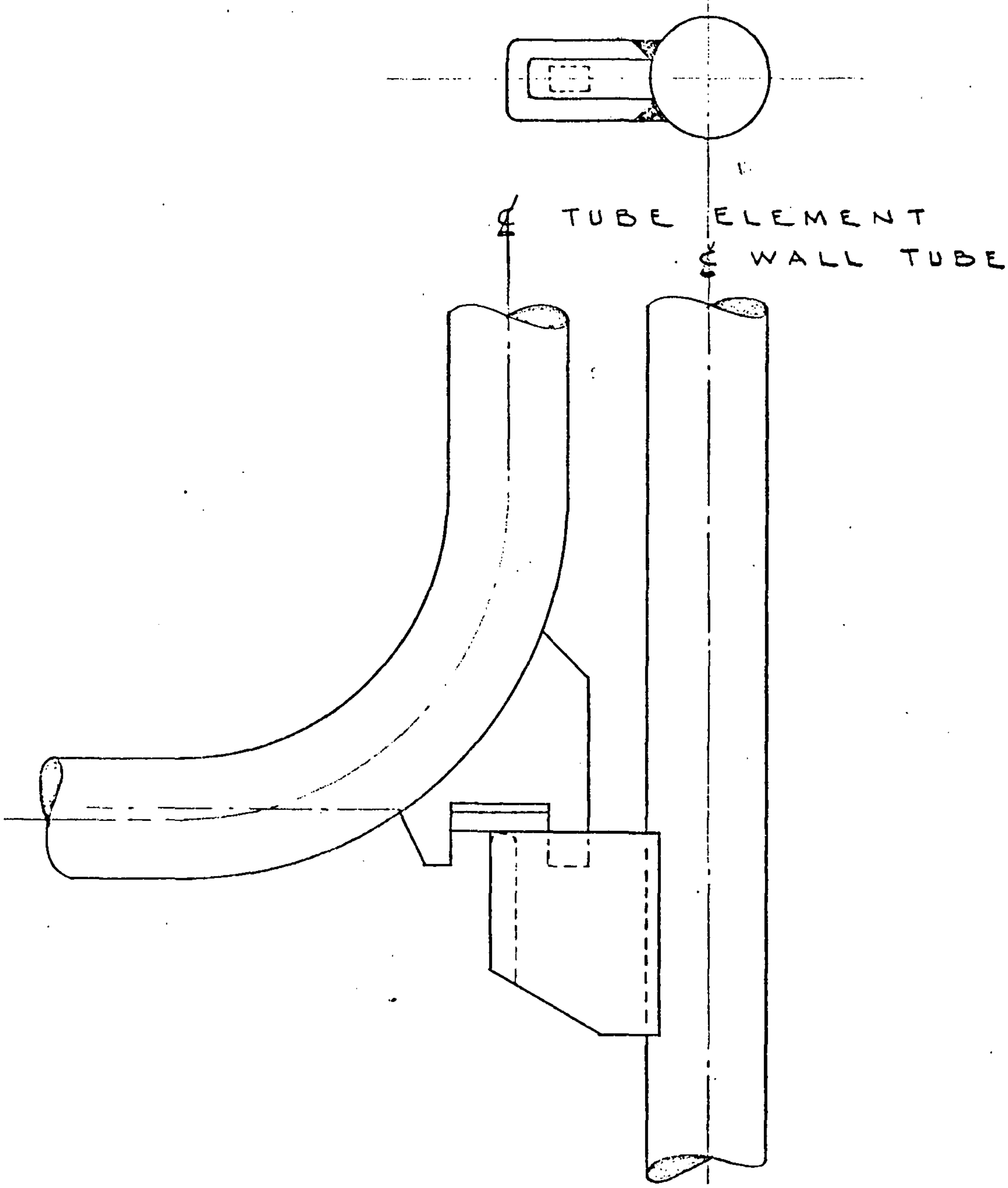
HEADERS

	SIZE	MATERIAL
Downcomers	10-3/4" O.D. x 1.00" A.W.	SA-106-C
Water Wall Headers	6-5/8" O.D. x 1.00" A.W.	SA-106-C
Pre-evaporator Inlet	8-5/8" O.D. x 1.255" A.W.	SA-106-C
Pre-evaporator Outlet	16" O.D. x 2.03" A.W.	SA-106-C
Lower Superheater Inlet	6-5/8" O.D. x 0.864" A.W.	SA-106-C
Lower Superheater Outlet	6-5/8" O.D. x 0.925" A.W.	SA-213-P11
Upper Superheater Inlet	6-5/8" O.D. x 0.925" A.W.	SA-213-P11
Upper Superheater Outlet	8-5/8" O.D. x 2.00" A.W.	SA-335-P22

heat transfer coefficients varied throughout the circuits, but the evaporator, superheater and reheater averaged 47, 45 and 43 Btu/hr-ft^2-°F, respectively. These heat transfer coefficients were chosen as conservatively low values. A low heat transfer results in conservatively large surface requirements, but this will not necessarily be the most expensive case. An increase in the heat transfer coefficient or the possibility of very uneven bed temperature could result in the need for more high grade steel in the uppermost loops of the superheater and reheater tube bundles. In the zones of the steam generator that were not in the fluidized bed, standard formulas for convection and radiation were used.

A3.2 Mechanical Design

As mentioned previously, the layout of this steam generator allows for conventional fabrication and support of pressure parts. A general layout of the steam generator is shown in Figure 3.1. The tube walls are supported by structural steel located at the top of the vessel. This is accomplished by hangers that connect the upper wall headers and the structural steel. There will be a relatively large gas pressure drop across the fluidized beds. Since the horizontal span of the tube walls is short, the tendency of the walls to bend is much less than on a conventional unit. To be safe, however, a tieback support system was used to further stiffen the walls (see Figure 3.2). The serpentine tube elements are supported by the wall tubes. A detail of this support system is shown in Figure A3-4.



TUBE ELEMENT SUPPORT ASSEMBLY	ORDER NO.	0-59-1192	
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FIG A3-4

All headers and downcomers are located in areas of the vessel that are not exposed to the hot gases. The tubes of the tube banks penetrate the finned tube enclosure walls at the top and bottom of the banks, and connect to the headers in the gas pass. The penetrations of the tube walls are sealed to prevent air leakage into the beds. The headers are supported both by the tubes they are attached to and by the shell. This can be observed in Figure 3.2. The combined flexibility of the expansion joint connection to the shell and the length of tubing from the header into the bed allows for differential expansion of the tube walls and the shell. In areas where the steam temperature is significantly higher than the combustion air temperature the headers are insulated.

Throughout the design of the steam generator, compromises had to be made between economy of design and ease of maintenance. In the resulting design it is felt that all normal repairs could be made without a major disassembly of the steam generator. Manholes provide access to the interior of the shell at several locations as shown in Figure 3.4. There is sufficient space between the tube walls and the shell to permit repair of the pipes and headers located in this zone. There are also access doors in the enclosure walls above each tube bank and in the roof plate of each cell. These two doors provide access to the upper bed areas, air plenum chambers and lower bed areas.

The repair of tube leaks in the tube banks is slightly more difficult in this design than in conventional units. In the conventional units access spaces are provided at certain locations in the interior of the tube bundles. A typical layout of the conventional design is shown in Figure A3-5. This type of design allows

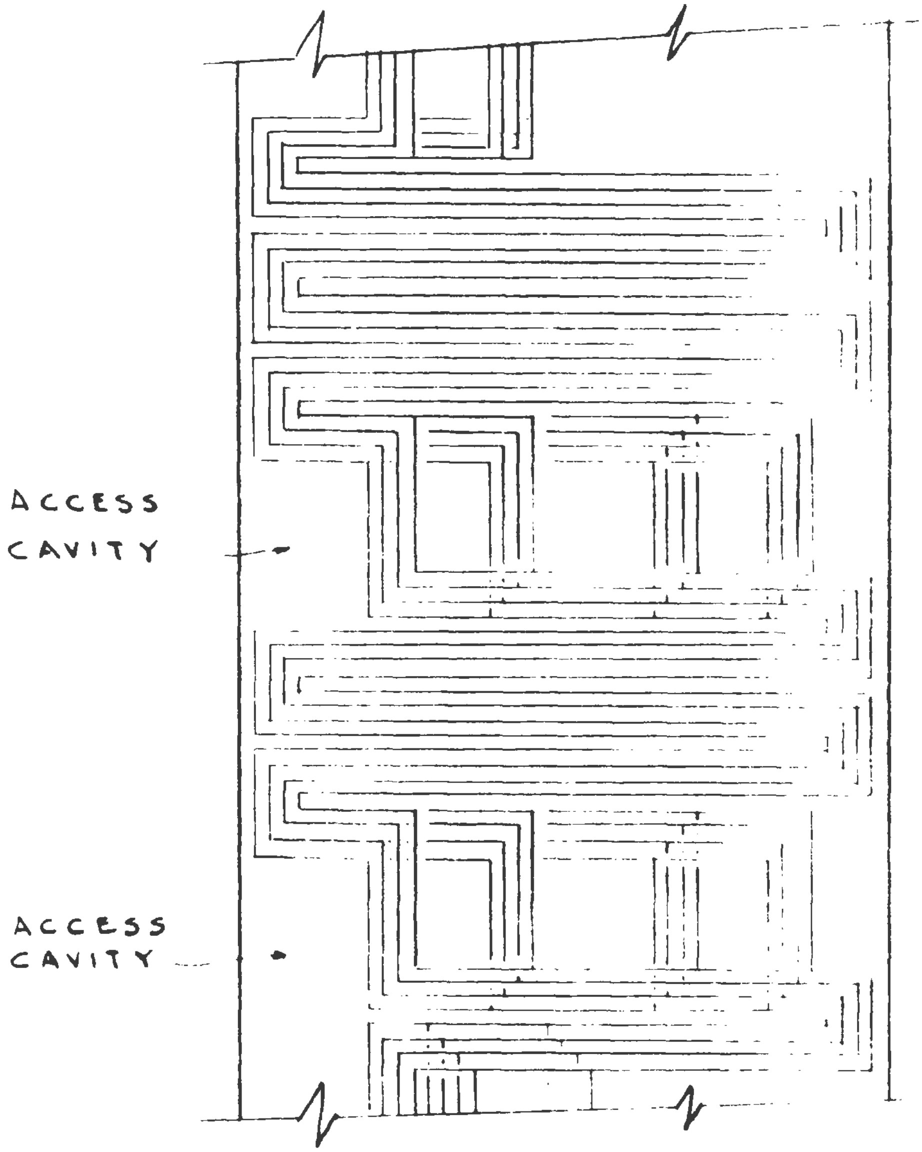

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FIG. A3-5

for the removal of a section of tubing by cutting the tube at the nearest access locations. The conventional tube bank design is undesirable in a fluidized bed because the tube-free areas that are provided for access could cause hot zones in the bed and uneven fluidization. The repair of a tube leak in a fluidized bed tube bank is technically no different from a repair in a conventional unit, but it would be more time consuming, because the entire length of the tube element would have to be removed. The tube element would be pulled out of the top of the tube bank in as many sections as the overhead space requires. The tube leak would be repaired and the sections of the tube element would be rewelded as it is lowered back into the bundle.

A leak in the tube walls would also require the removal of a section of tubing that is as long as the tube bundle. This is because the elevation of a wall tube leak in this area would be difficult to determine. Again this causes no technically unsurmountable problems.

Replacement of entire tube banks or headers would require a major disassembly of the module, but the need for this type of maintenance is unlikely. All anticipated normal maintenance appears to be feasible. Many repairs are made more difficult by the economical use of vessel space and configuration requirements for fluidization, but the maintenance requirements do not seem to be a major drawback of this system.

APPENDIX 4

MODULE DESIGN

A4.1 Pressure Vessel

The steam generator is housed in a 12 ft. diameter pressure vessel shown in Figure 3.4. This diameter was chosen because it is the maximum diameter that can be shipped in long sections. Manholes are provided at various locations on the vessel to allow access to the pressure parts and accessory equipment located inside the vessel. The shell itself is 1 in. thick and is made of SA 515-70 steel. The outside of the vessel is insulated with 4 in. of batt insulation, and the insulation is protected by an aluminum sheet. Besides maintaining the system gas pressure, the vessel supports the steam generator pressure parts. This is done with structural steel that is attached to the shell near the top of the vessel. The location of this structural steel can be seen in Figure 3.1.

A4.2 Air Piping

The module air piping can be seen in Figures 3.1 and 3.2. Air for the primary fluidized beds enters the pressure vessel through a 42 in. diameter pipe located near the bottom of the module. Air entering through this pipe fills the space between the steam generator tube walls and the pressure vessel shell. This space serves as a manifold to distribute the combustion air between the four primary fluidized beds. The distribution was accomplished in this manner for several reasons. One benefit of having the combustion air fill this entire area is that it will cool the pressure vessel and the structural steel that supports the boiler pressure parts. In fact, a small amount of this air will be injected at the top of the vessel in order to prevent hot air zones from forming there. Also, since the combustion air is at a higher pressure than the fluidized beds and the subsequent gas passages, any gas leaks that develop in the finned tube enclosure walls will result in air leakage into the high temperature gas zones. The possibility of leakage in the opposite direction would be dangerous because an increase in pressure vessel temperature could cause failure of the vessel.

There are other possible means of accomplishing air distribution between the fluidized beds, but they have serious drawbacks. One possibility would be putting air ducts inside the pressure vessel. This scheme would result in a very crowded module with maintenance of internal equipment very difficult if not impossible. External ducts with separate inlets to each bed would be another possibility, but this would require many penetrations of the pressure vessel. It is desirable to have as few penetrations of the pressure vessel as possible to minimize thermal expansion problems between the external piping and the pressure vessel.

The carbon burn-up cell combustion air is kept separate from the primary bed air. A 12 in. diameter pipe penetrates the pressure vessel just above the primary air inlet pipe, and connects directly to the carbon burn-up cell plenum chamber. A control damper is located in this air line. The separation of the primary bed air supply and the carbon burn-up cell air supply gives more flexibility in establishing different conditions in the carbon burn-up cell. The air flow to the carbon burn-up cell affects bed temperature, excess air, and superficial velocity. These parameters are quite critical for good carbon burnout so it was felt that good control of the air supplied to this bed was highly desirable.

A4.3 Plenum Chamber and Distributor Plate Design

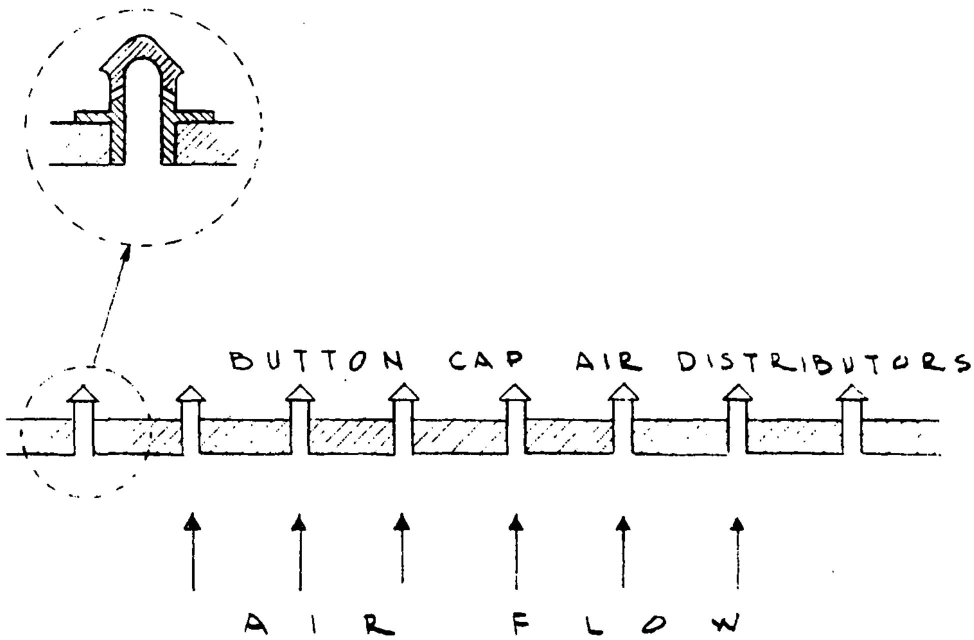
Each fluidized bed has an air plenum chamber underneath it. Figure 3.3 shows a typical air plenum chamber. The location of the plenum chambers in the module can be seen in Figure 3.1. The side walls of the chambers are formed by the same enclosure walls that form the fluidized bed cells. Air enters the plenum chambers through control dampers that are shown in Figures 3.1 and 3.3. These dampers are located over screen openings in the enclosure walls. It was felt that this would be easier to construct than providing openings in the enclosure walls that would be large enough to contain a damper. The dampers are controlled by drives that are external to the pressure vessel. The dampers on each of the plenum chambers are controlled separately so that the relative distribution of air between the beds can be varied. Although these dampers are basically control dampers, they should provide sufficient sealing in the fully closed position to essentially stop the air flow to any bed.

The top and bottom walls of the plenum chamber consist of steel plates. The top plate is the fluidized bed air distributor plate. The function of this plate is to provide an air flow pattern that is conductive to good fluidization of the bed. Basically the air distributor plate is a flat plate with air nozzles spaced across the surface. The design of the nozzles, the number of nozzles and size of the nozzles will have to be determined by pilot plant tests. Air distribution plates that are now in use will not necessarily be effective in the deep, high pressure beds of this design. For the purpose of including a cost estimate for the distributor plate, "button" nozzles of the type used by Pope, Evans and Robbins were assumed. These nozzles are shown in Figure A4-1. In order to allow for differential thermal expansion between the distributor plate and the enclosure walls, the distributor plate is not welded to the enclosure walls. Instead, a scalloped bar is welded to the walls around the perimeter of the air plenum chamber and the plate sits on the rim formed by these bars. This construction can be seen in Figure A4-2.

The bottom plates of the air plenum chambers are solid plates that seal the plenum chamber off from the gas passage of the next lowest cell. These plates are supported off of the wall tubes in the same manner as the distributor plates. The side of the bottom plate that is in contact with the hot gases is insulated to prevent warpage of the plate.

A4.4 Coal Feeding

The design and location of the coal feed nozzles can be seen in Figure 3.3. Since the fluidized beds in this design are relatively deep, it was felt that four coal feed points spaced around the perimeter of the bed would be sufficient to give an even distribution of coal in the bed. The nozzles are directed into the bed directions that tend to provide equal distribution of the coal across the cross sectional area of the bed. Also a 2 ft. high tube-free zone is provided above the air distributor plate to mitigate



CROSS SECTION OF AIR DISTRIBUTOR PLATE WITH BUTTON CAP AIR DISTRIBUTORS

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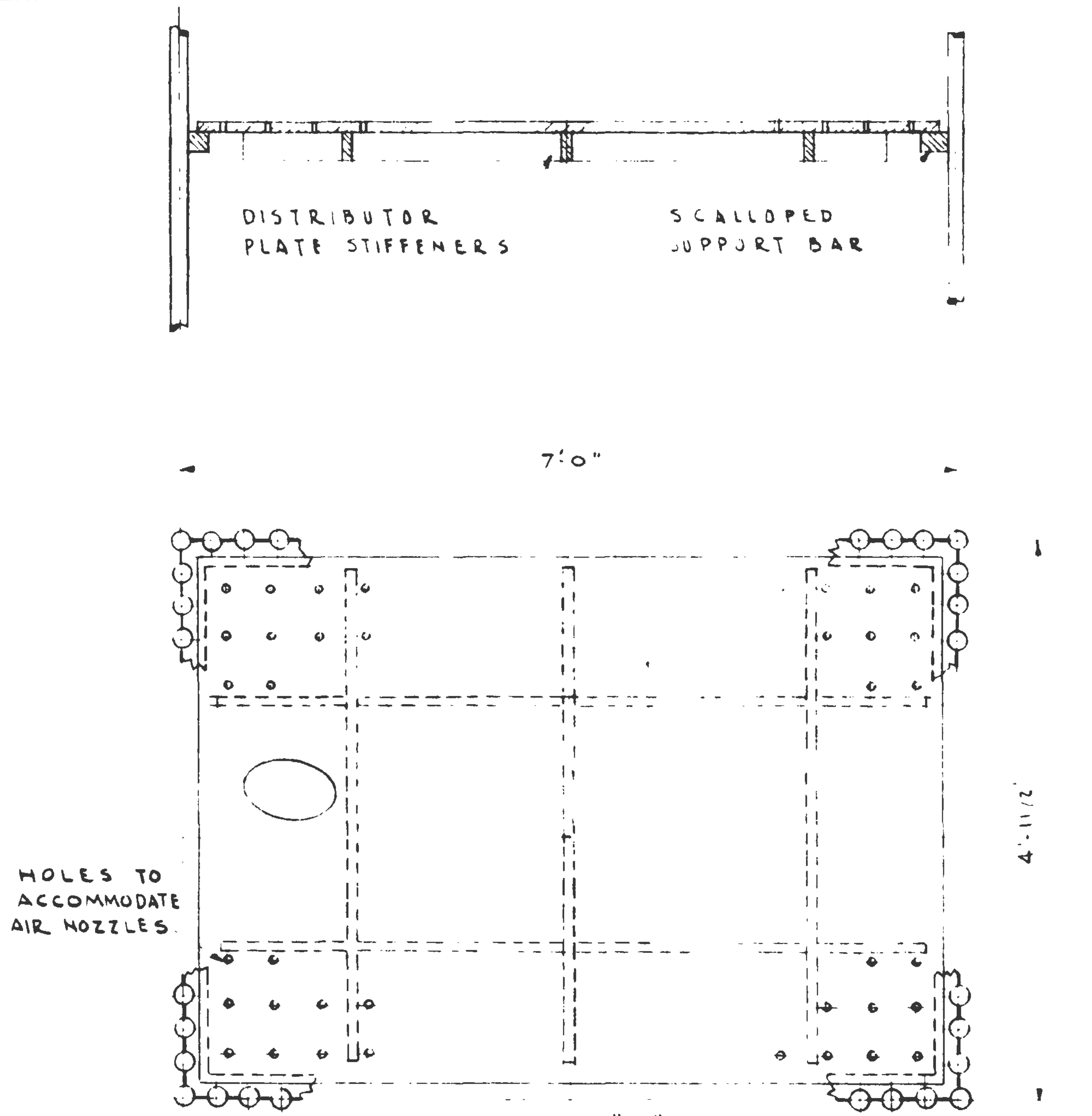
FIG. A 4-1



DETAIL "B"
BOTTOM PLATE (TYP.)

AIR DISTRIBUTOR PLATE SUPPORT

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FIG A4.2

the effect of possible flow channeling in the upper zones of the bed. The coal is transported pneumatically through the feed pipes and nozzles.

The problem of differential thermal expansion between the enclosure walls and the pressure vessel affects the coal feed pipes in the same way as it affects the gas and steam piping, so flexible penetration seals were used on the pressure vessel shell.

A4.5 Sorbent Feeding and Circulation

The sorbent feeding and circulation system interior to the module is relatively simple. Since the modules are designed to operate at constant bed depths, there is a certain flexibility in the location of the inlet and outlet pipes of the dolomite circulation system. Also the circulated dolomite is a small portion of the total bed capacity, so only one inlet and one outlet pipe are provided. These pipes are separated as much as possible, however, to prevent any possible bypassing of the regenerated inlet stone directly to the outlet pipe. The dolomite circulation inlet pipe is located 4 ft. above the air distribution plate. The dolomite circulacion outlet pipe is located 4 ft. below the top of the tube bank. The location of these pipes can be seen in Figures 3.1 and 3.2.

In normal operation the bed height will be kept constant. During start-up, however, some beds will have to be filled during operation. This is accomplished by feeding dolomite into the bed through the dolomite circulation inlet pipe. The coal feeding system also has the capability of feeding dolomite. The dolomite fed in this manner, however, will only be used to replenish the stone that is elutriated and spent from the bed, because the dolomite feeding capability of this system is not sufficient for circulation. It is estimated that 0.5% of the bed dolomite will be elutriated per hour, and the coal feed system is designed to make-up this amount of stone and the spent stone. An outlet pipe is provided at the very bottom of the bed. This pipe is normally completely closed off, but it can be used to empty the bed when necessary. Sorbent taken from the bed in this manner will be stored in a surge vessel.

A4.6 Steam Piping

As with the air and gas ducts, it was desired to keep the penetrations of the shell by steam piping down to a minimum. To achieve this as much steam transfer piping as possible was located inside the pressure vessel. These pipes are situated in the free space between the steam generator enclosure walls and the pressure vessel shell. Figures 3.1 and 3.2 show the layout of the transfer piping. The only transfer pipes that penetrate the vessel are the feedwater inlet, superheater outlet, reheater inlet and reheater outlet. These pipes are attached to the pressure vessel with bellows type seals to allow for differential expansion of the shell and the headers.

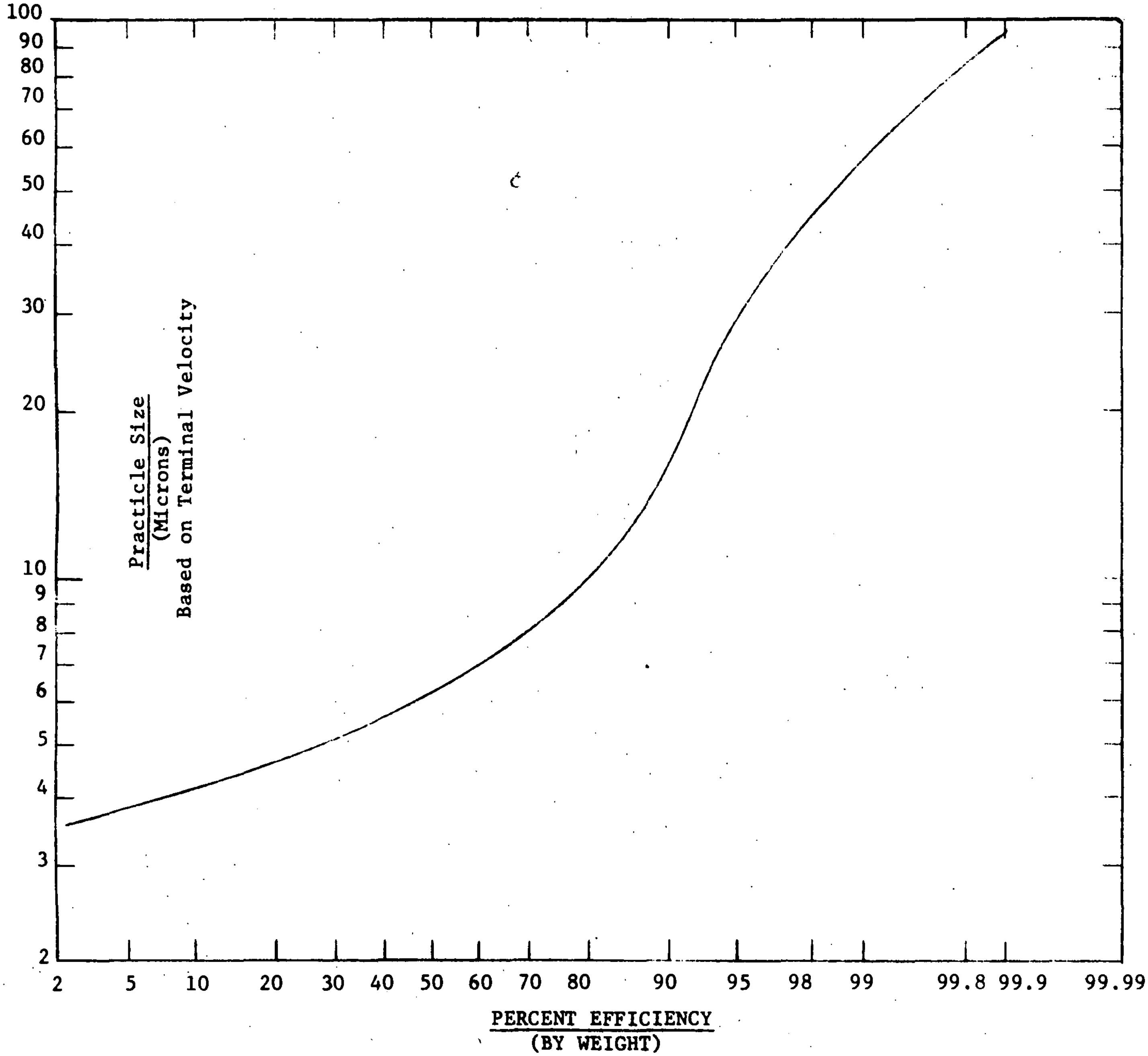
A4.7 Particulate Removal

A schematic flow diagram of the particulate removal system is shown in Figure 4.1. The first separator stage cleans the flue gases from the primary beds and the solids collected in this stage are returned to the carbon burn-up cell. The purpose of this first particulate removal stage is to increase the efficiency of the steam generator by minimizing the unburned carbon loss. It was assumed that 5% of the bed sorbent per hour, 6% of the carbon and all of the ash in the coal would be elutriated from the primary fluidized beds. The size distribution of these particles can be seen in Figure 2.3. The first separator stage returns 90-95% of these solids to the carbon burn-up cell where optimum conditions for the combustion of carbon are maintained. The gas cleaned by the first separator stage and the gas from the carbon burn-up cell unite before the second particulate removal stage. The purpose of the second stage is to clean the gas sufficiently to prevent damage to the gas turbines.

The efficiency specification of the first particulate removal stage is somewhat flexible. Increasing the efficiency of the cyclones also increases the pressure drop, and the gain in combustion efficiency that results from high collection efficiency must be weighed against the loss of cycle efficiency that results from an increased pressure drop. Four Ducon Company cyclones of the Duclone type, size 4-355 VM 81-/150 were chosen for the first stage collection. These cyclones are operated in parallel and provide a collection efficiency of 90.2% by weight with a 0.7 psi pressure drop. A fractional efficiency curve for this cyclone is shown in Figure A4-3. It is felt that these conditions will give optimum plant efficiency.

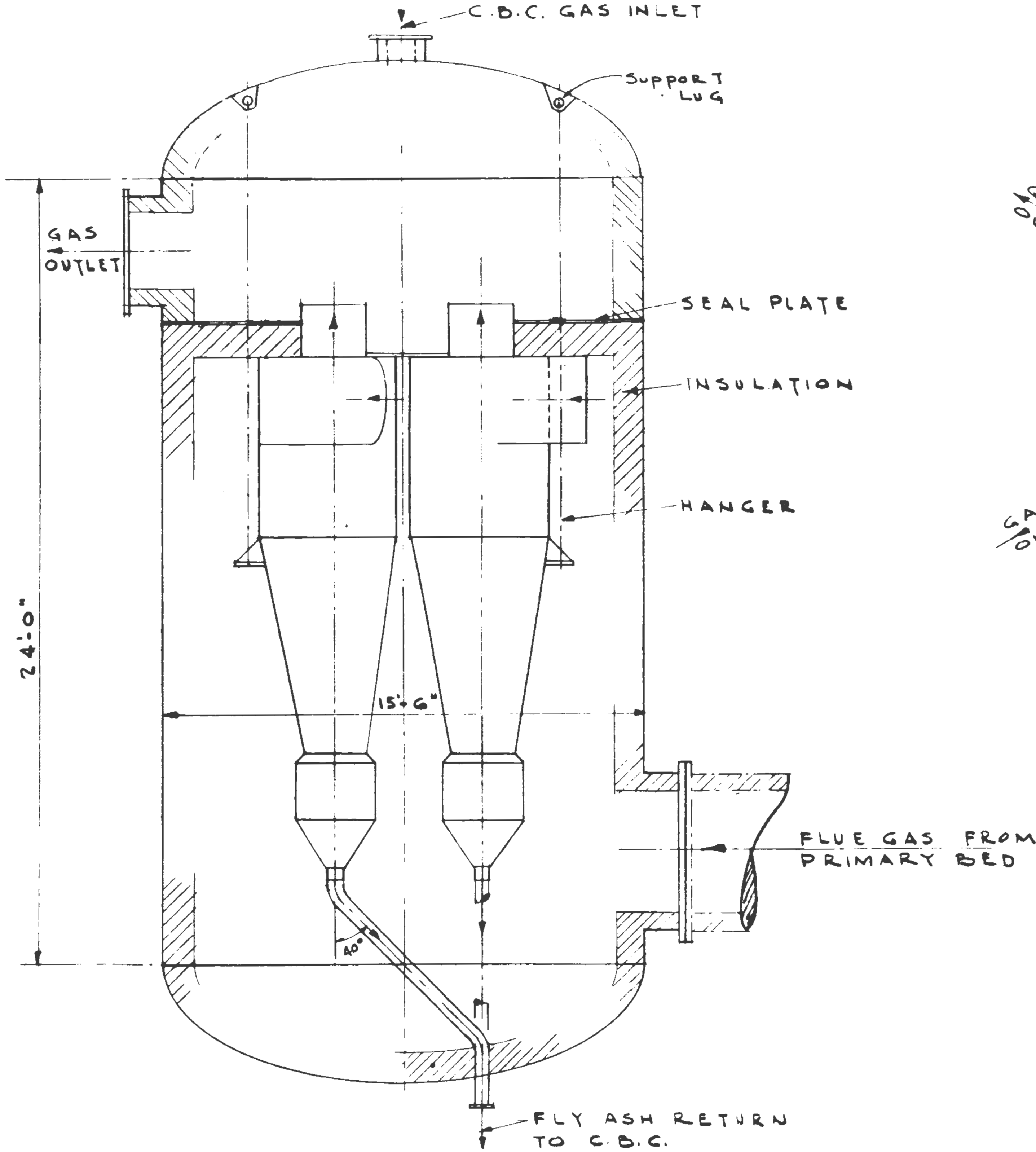
The arrangement of these cyclones in the pressure vessel is shown in Figure A4-4. Figure A4-5 shows the location of the cyclone pressure vessel in relation to the steam generator. In the arrangement shown here the cyclones are exposed to the hot flue gases both internally and externally. This requires that the cyclones be made of relatively expensive steel alloys. Two

ESTIMATED FRACTIONAL EFFICIENCY CURVE[1]
FOR
DUCON CO. SIZE 335 VM 810 CYCLONE
WITH OUTLET AREA/INLET AREA TO 1.0



1. Ducon Co. Curve #33A

FIGURE A4-3



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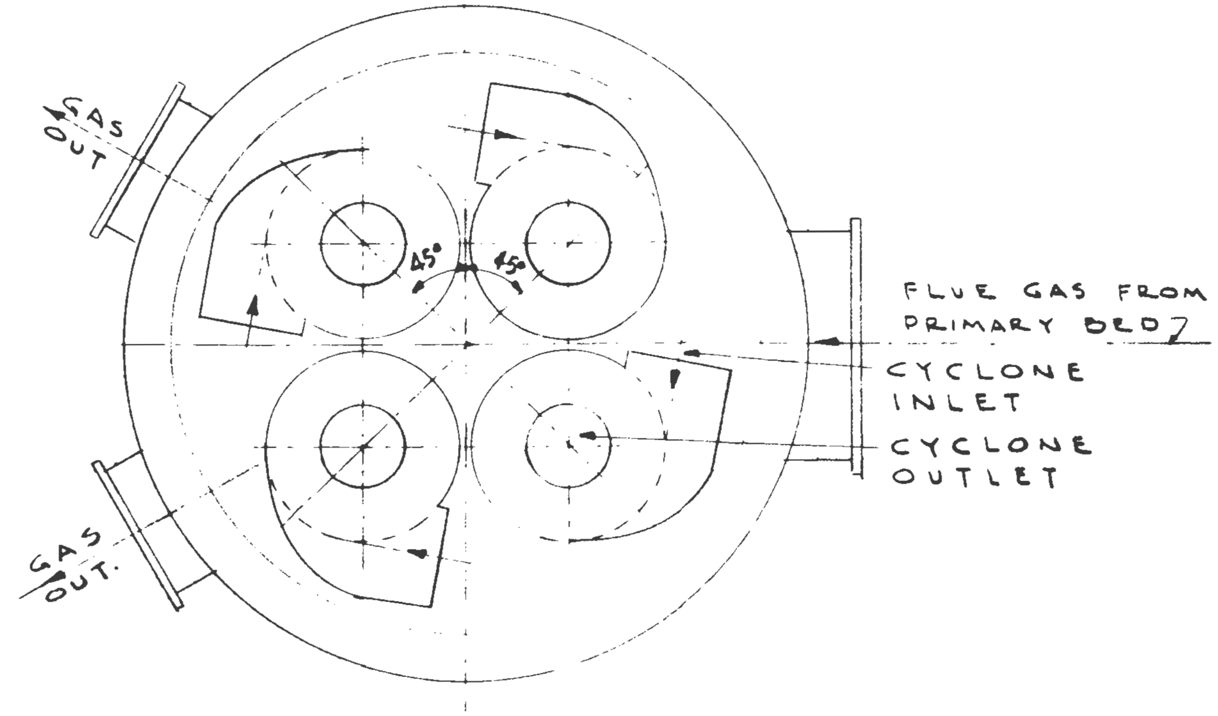

PLAN

1ST STAGE CYCLONE SYSTEM

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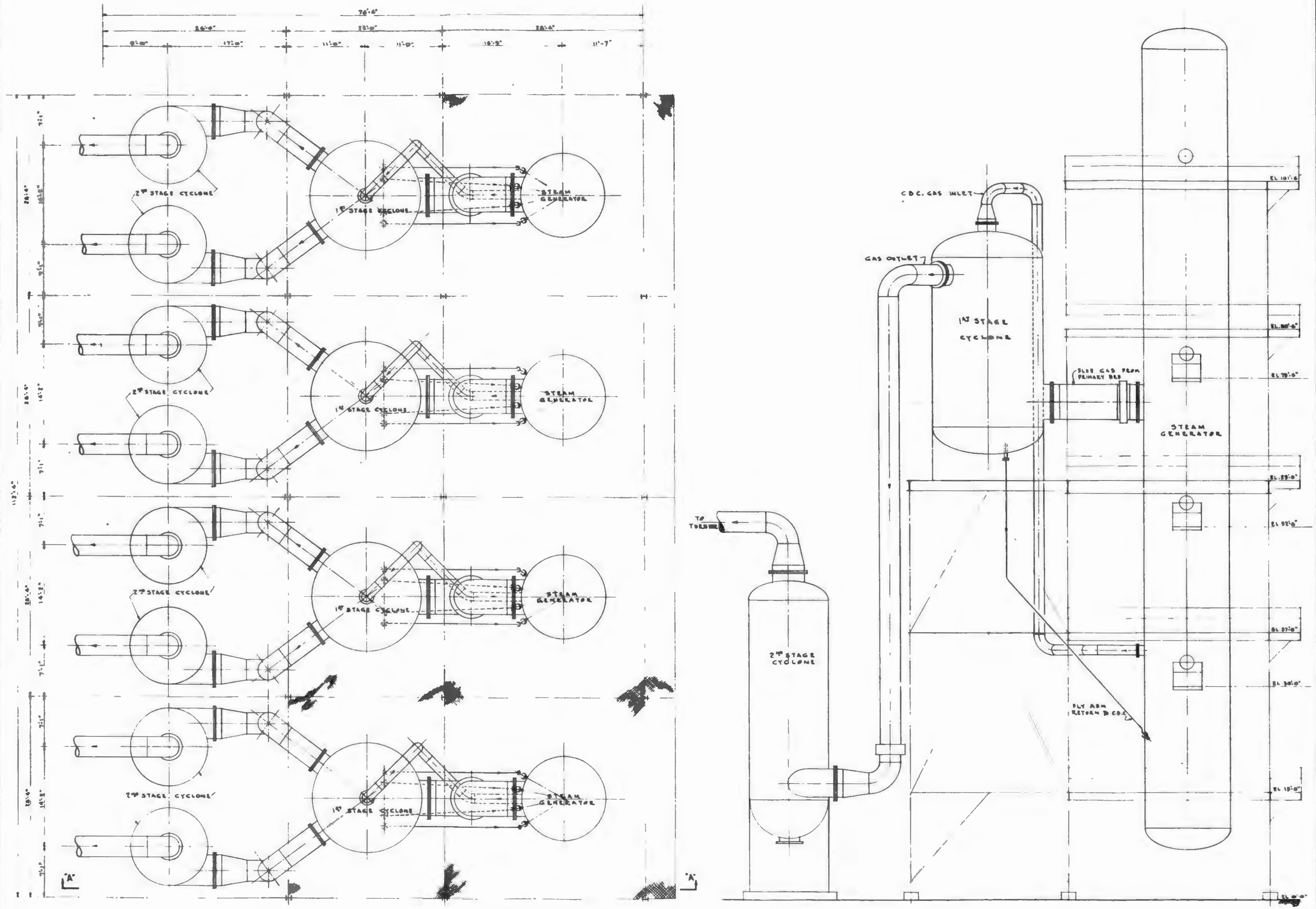
FIG. A4-4



PLAN

SECTION "A-A"

FIG. A4-5

NOTES

REVISIONS

PLOT PLAN

STEAM GENERATORS AND ACCESSORY EQUIPMENT FOR A 300 MW COMBINED CYCLE PLANT

SCALE 3/32" = 1'-0"

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other arrangements are possible. One would be to place cyclone forms in a pressure vessel and fill the void areas with refractory. This would do away with the need for high temperature alloy in the cyclones and in pressure vessel lining material. A drawback of this system is that a separate plenum chamber must be supplied to ensure even gas distribution to the cyclones. The design shown in this report conveniently uses the cyclone pressure vessel as a plenum chamber. Another problem with refractory formed cyclones is that if wear occurs, repair of cyclones is impossible. The second alternate cyclone arrangement would be to line the interior of the cyclones with refractory and design them to withstand the system pressure. In this case also, a separate plenum chamber would have to be built, and the refractory lining requires a relatively large amount of maintenance.

In the design presented in this report, the flue gas from the carbon burn-up cell enters the top chamber of the first stage separator pressure vessel. Here it combines with the gases cleaned by these cyclones. Two pipes transport this combined stream from the chamber to the two secondary separators. These two secondary cyclones operate in parallel and their purpose is to clean the gas sufficiently to protect the gas turbine. This particulate collection system is discussed in detail in Appendix 5.

APPENDIX 5

STEAM GENERATOR SUB-SYSTEM DESIGN

A5.1 Steam Generator Module Arrangement

The suggested plot plan for the steam generator modules and the particulate removal equipment in the 300 MW plant is shown in Figure A4-5. An in-line arrangement was chosen for the steam generator modules. A square layout was considered, but realistic location of the large coal feeding and particulate removal equipment was impossible with a square module layout. The location of the steam generators and auxiliary equipment with respect to the total plant layout is shown in Figure A5-1.

Each steam generator module has a coal pressurizing and feeding system and a particulate removal system. The limiting factor in the module spacing is the layout of the particulate removal equipment. A minimum spacing between the particulate removal systems results in a module spacing of approximately 28 ft. This spacing should be adequate for all normal maintenance and repairs. The coal pressurizing and feeding equipment is located in two groups. Another possible layout would be one in which each coal feeding system would be located on the centerline of the steam generator module it serves. Placing this equipment in two compact groups, however, requires less structural steel and platform.

Platforms for access to the steam generator modules are located as shown in Figure A4-5. Full platforms are located at the

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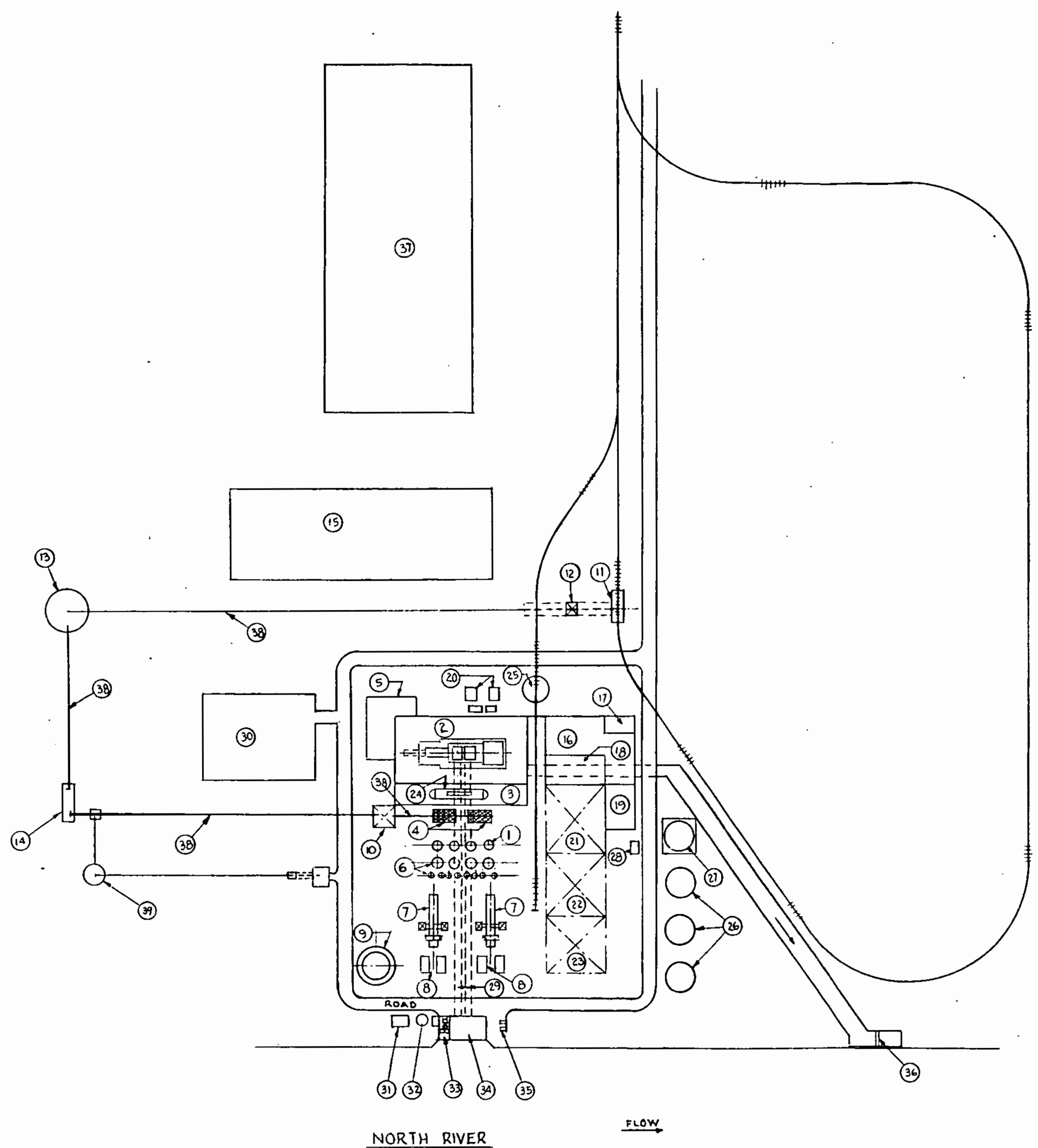

SITE PLOT PLAN

0 50 100 200 300

SCALE

(1) BOILERS	(21) CLAUS PLANT
(2) TURBINE BAY	(22) CO_2 ABSORBERS
(3) HEATER BAY	(23) REGENERATORS
(4) STORAGE BINS	(24) DEAERATOR
(5) ADMINISTRATION BLDG.	(25) ASH SILO
(6) PARTICULATE REMOVAL EQUIPT.	(26) WATER STORAGE TANKS
(7) GAS TURBINE GENERATORS	(27) LIGHT OIL TANK
(8) STACK GAS COOLERS	(28) LIGHT OIL UNLOADING PUMP
(9) STACK	(29) CIRCULATING WATER INTAKE PIPES
(10) SURGE BIN	(30) PARKING AREA
(11) RECEIVING HOPPER	(31) GASES (STORAGE)
(12) RECLAIM HOPPER	(32) FILTERED & FIRE SERVICE WATER TANK
(13) SILO (12,000 TONS)	(33) SERVICE WATER PUMP HOUSE
(14) DRYER	(34) CIRCULATING WATER INTAKE STRUCTURE
(15) DEAD STORAGE (50,000 TONS)	(35) CHLORINATION EQUIPTMENT
(16) WATER TREATMENT	(36) VARIABLE WEIRS
(17) DIESEL GENERATOR ROOM	(37) SWITCH YARD
(18) CONTROL ROOM	(38) CONVEYORS
(19) MACHINE SHOP	(39) STONE STORAGE
(20) TRANSFORMERS	

NOTES

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300 M.W. COMBINED CYCLE PLANT SITE PLOT PLAN

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SQUAD LEADER			
SECTION HEAD			
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FIGURE A5-1

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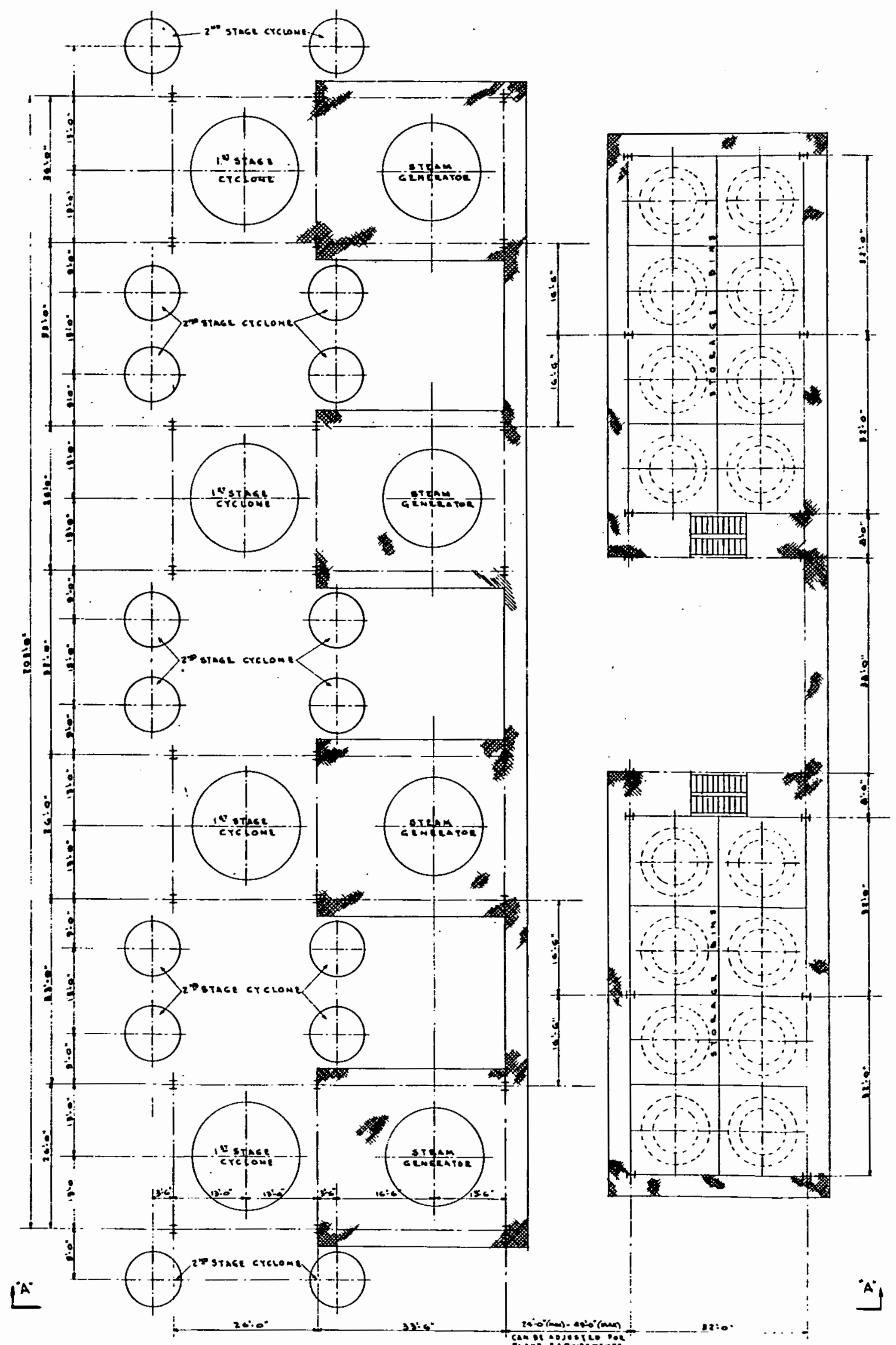
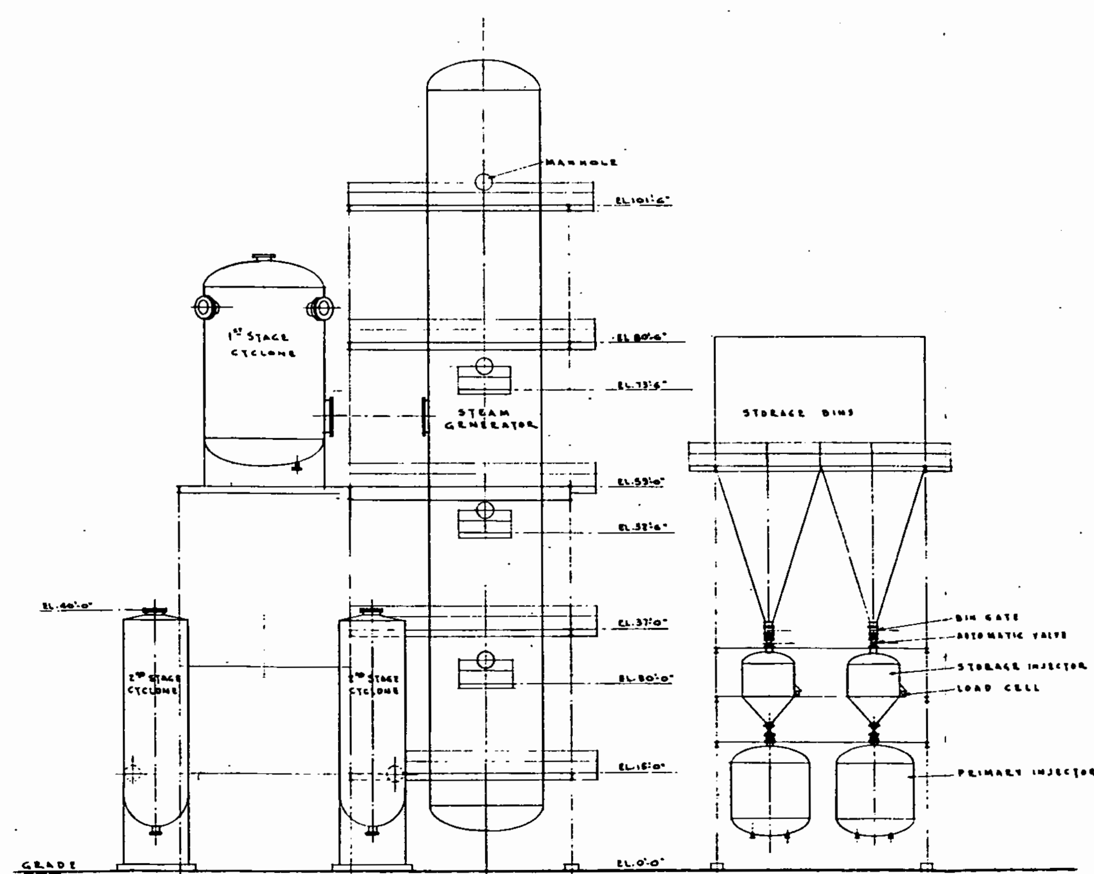
elevations of the ignitors, damper drives and coal feed pipes because these are items that require frequent access. These platforms extend for the full length of the four modules so that an elevator is practical. Small platforms are located below the main platforms to provide access to the three lower manholes. A full platform is located near the top manhole.

Access must also be provided for the particulate removal system and the coal pressurizing and feeding system. At several elevations the steam generator platform steel extends to the first stage cyclones to provide access to this equipment. It was felt that permanent ladders would provide sufficient access to the secondary cyclones. Platforms are specified at three elevations in the coal pressurizing and feeding system to provide access to the valves and storage bin.

The arrangement of the steam generators and auxiliary equipment for the 600 MW plant is shown in Figure A5-2. The size of this equipment is approximate since a detailed design was not done on this plant. Basically the layout is the same as the 300 MW plant, but the vessel diameters are increased. The exception to this is the secondary cyclones. Better efficiency was attained by using four of these cyclones instead of two larger ones.



PLAN



SECTION "A-A"

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PLOT PLAN
STEAM GENERATORS AND ACCESSORY EQUIPMENT FOR A 600 MW COMBINED CYCLE PLANT

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Figure A5-2

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A5.2 Coal Handling System and Limestone Make-Up

Coal handling for the pressurized boiler includes receiving the coal, storage, transportation to the surge bins of the injector, drying, sizing and crushing. McNally Pittsburg developed the coal handling plant and a price including erection.

A5.2.1 Assumptions The coal handling plant was designed on the basis of the following assumptions:

The coal to be used is Ohio Pittsburgh Seam No. 8 described in detail in Table 2.1 of the text.

The size of the coal is 1 1/2" x 0" as delivered. It was initially assumed that the coal would be sized to 5" x 0". However, this size coal is much too large for drying in a fluid bed dryer which appears to be the most economical drying method at the present time. According to the Bureau of Mines at Bruceton, there is no penalty paid for the 1 1/2" x 0" coal as it is a standard size commonly requested. Under these circumstances it was felt justified to revise the specifications. In addition to meeting the technical requirements, there appears to be a savings in cost of power required for crushing the smaller coal size.

Size distribution of coal crushed to 1 1/2" x 0" at the mine appears in Figure 2.1 of the body of the report.

Moisture as mined, is 3.3% total and 2.1% inherent. The difference is surface moisture.

Moisture as-delivered is 10% total. This is about 6.7% in excess of the as-mined coal and represents a typical moisture pick-up during handling and storage in an eastern steam generating plant location.

Moisture required at the boiler feed system is 3.0% total maximum. Higher moisture content result in agglomeration in the surge bins and feed lines.

Size required at the boiler feed system is 1/4" x 0". The Petrocarb pressurized feed system can handle 1/4" x 0" coal, however, 1/8" x 0" is preferred. With the larger sizes the risk of erosion is greater. Unfortunately, mills and/or crushers equipped with classifiers to size the effluent coal stream to 1/8" x 0" are not readily available. A screening operation which would scalp sizes greater than 1/8" and return them for further grinding could be developed, but at great expense. The 1/4" x 0" coal size is a compromise that might warrant further investigation in the future.

The Hardgrove grindability index is 60.

The burning rates which affect hopper sizes, conveyor speeds, etc., were set at 113 tons per hour; 2,712 tons per day and 18,984 tons per week. Allowance was made for handling of sorbent feed or sorbent make-up as it was not certain at what point it would be added to the coal stream.

Storage of the coal is divided into two categories, active storage to handle short term contingencies such as weekend outages for maintenance and repair and dead storage to handle long term outages due to short strikes, delays in shipment, etc. Active storage was set at three days or 8,136 tons. The storage capacity of the silo was rounded off to 12,000 tons.

Dead storage was set at two weeks or 37,968 tons. It was rounded off to 50,000 tons of open coal pits.

The values were selected as typical based on the recommendation of McNally Pittsburg.

Dead storage requirements are vulnerable to considerable change according to individual customers' requirements. Changes should only affect operating cost and land requirements. The additional cost of coal handling should be marginal.

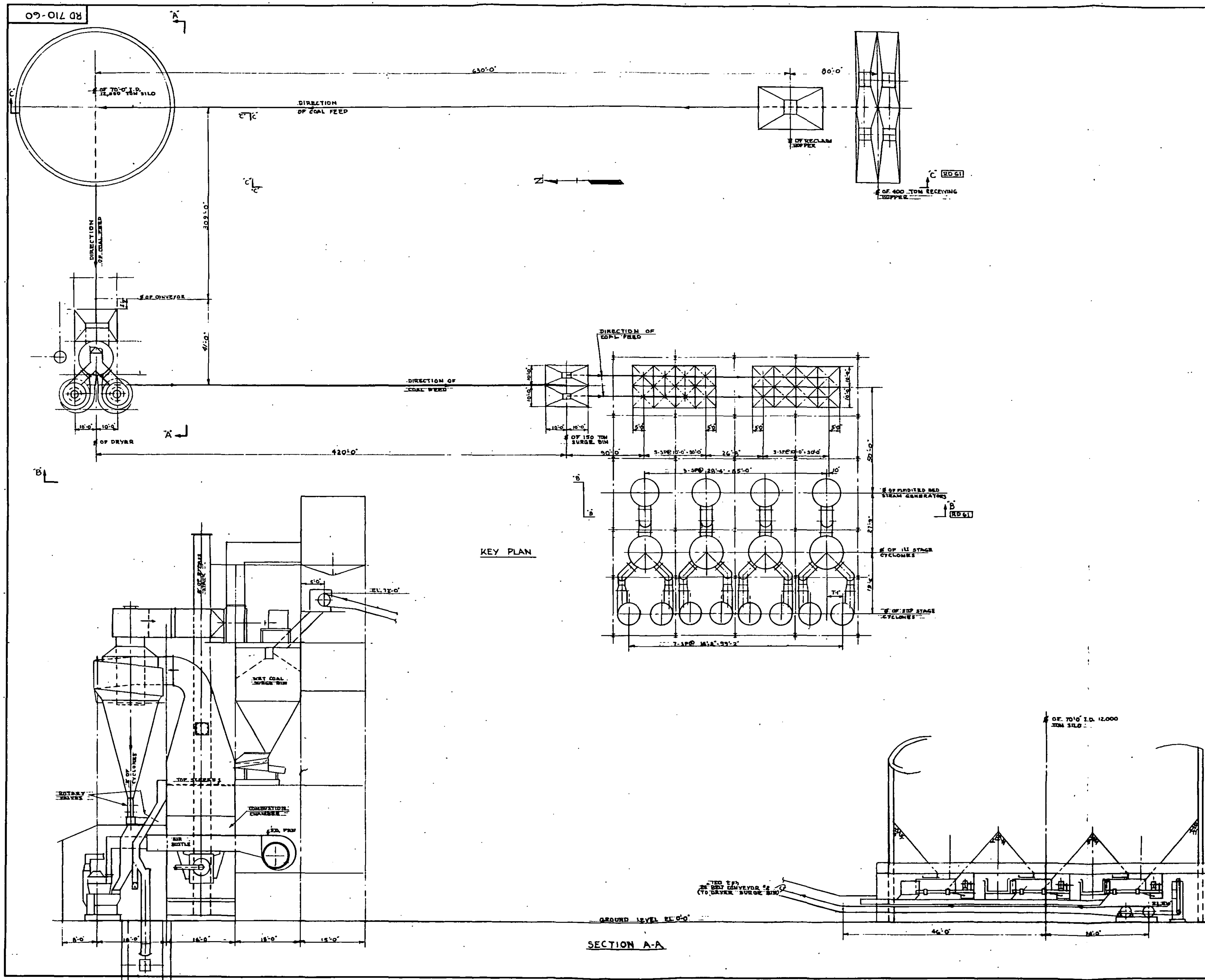
It was assumed that the coal would be delivered by 10,000 ton unit trains at a rate of 2 per week. The unloading rate is 2,000 tons per hour and the train would require 5 hours to deliver its load.

The dead storage pile would require 49 weeks to fill to a capacity of 50,000 tons.

A5.2.2 Scope The system begins with the unloading of bottom dump rail cars in unit train delivery and continues through active and dead storage, thermal drying, crushing and delivery to the coal feeder surge silos by Petrocarb ahead of the boiler firing system.

The rail car receiving hopper will have a minimum of 400 tons capacity to provide storage volume for controlling unloading of moving cars. Details of the receiving hopper and other components are illustrated in Figures A5-3 and A5-4. Their relationship to other components in the coal handling system is illustrated in Figure A5-5. Their relationship to the entire power plant facilities is shown in Figure A5-1.

The coal discharge from the receiving hopper will be controlled by four reciprocating feeders, F1-A, F1-B, F1-C and F1-D, (Fig. A5-5)



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REVISIONS

300 M.W. PRESSURIZED FLUIDIZED BED STEAM GENERATOR CONVEYOR SYSTEM

SHT. 1 OF 2

RD 710-60

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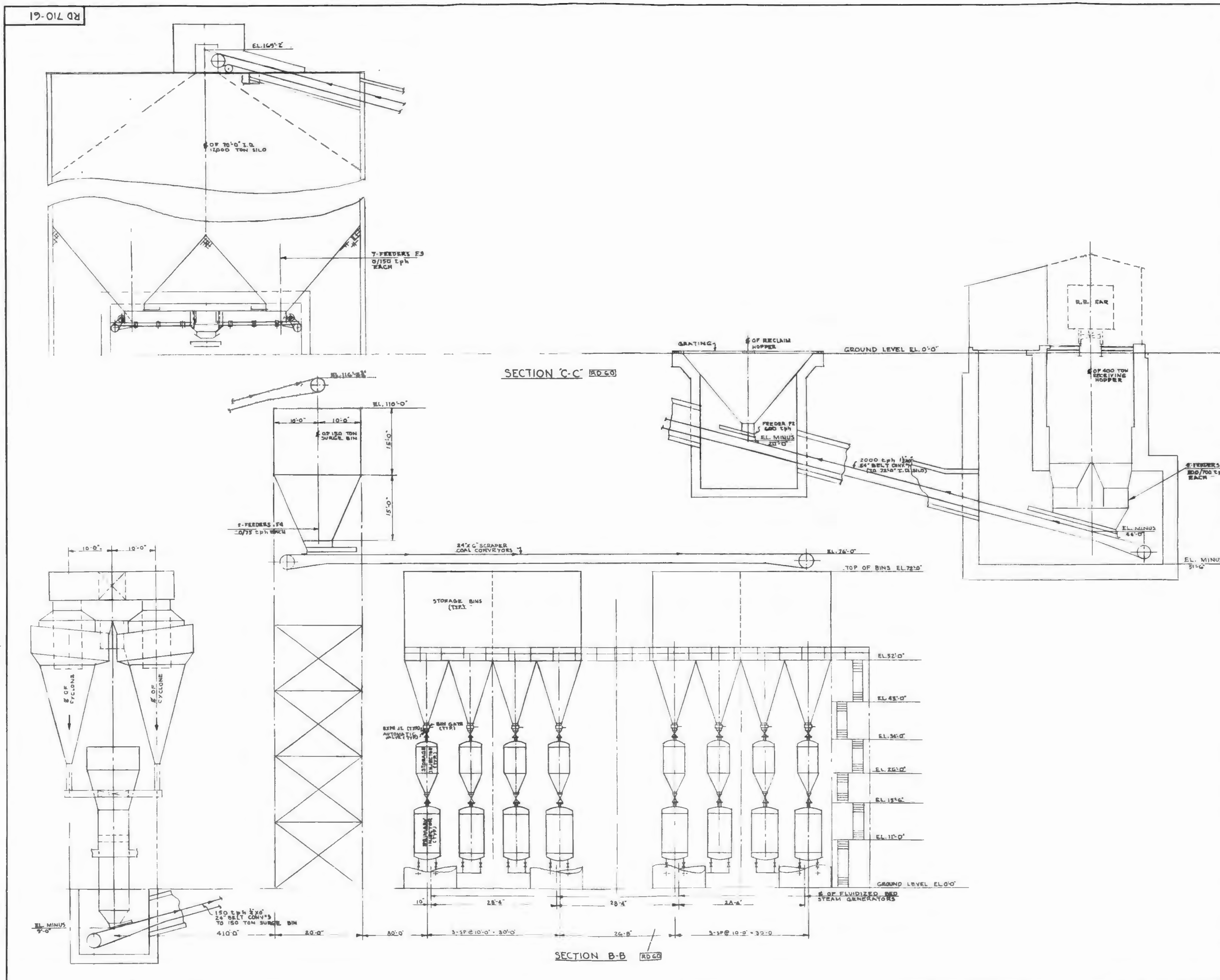

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300 M.W. PRESSURIZED FLUIDIZED BED STEAM GENERATOR CONVEYOR SYSTEM

SHT. 2 OF 2

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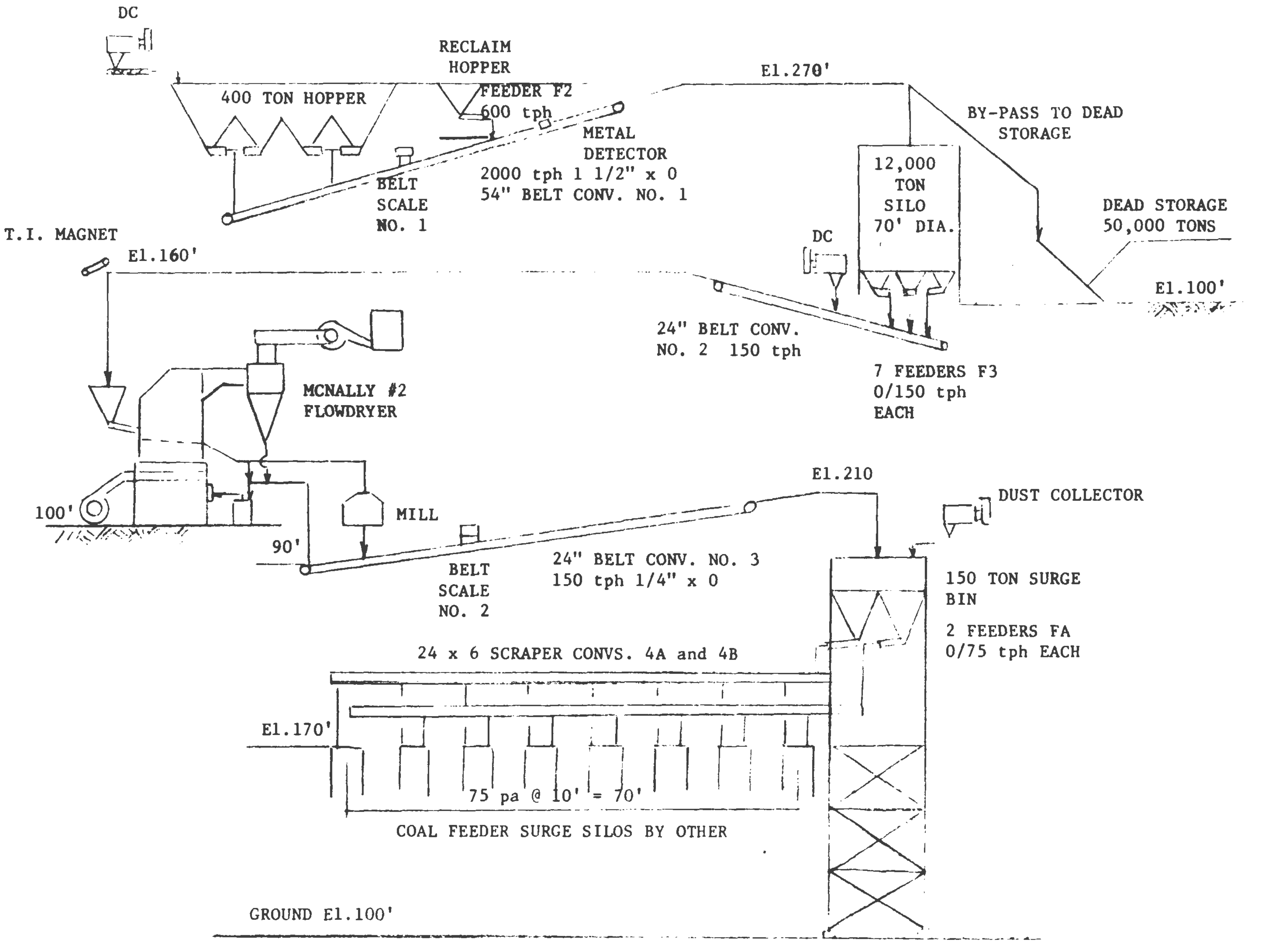


FIGURE A5-5 SCHEMATIC DIAGRAM, COAL HANDLING PLANT each feeder nominally rated at 500 tph but capable of delivering 700 tph if one feeder is out of service.

A 54" wide inclined belt conveyor No. 1 will transport the coal at the rate of 2000 tph to the silo. Conveyor No. 1 will be equipped with a tramp metal detector and a belt scale for weighing and recording coal quantities delivered to the plant site.

A 12,000 ton silo will nominally contain an entire unit train delivery without exposure to wind blown dust and provide active storage for plant feed without the use of mobile equipment. Excess coal will overflow the silo to an initial pile for placing in permanent storage. See Figure A5-5.

During extended periods of no-train delivery, coal will be recovered from permanent storage through a reclaim hopper and Feeder No. F2 delivering to Conveyor No. 1 for refilling the silo.

Coal will be fed from the silo by seven Feeders No. F3A to F3G, each rated a maximum of 150 tph and operating in timed sequence for uniform drawdown of the silo, onto a 24" wide belt Conveyor No. 2. Conveyor No. 2 will be fitted with a tramp iron magnet before delivery to the coal dryer. Details of the magnets and dust recovery equipment have not been illustrated.

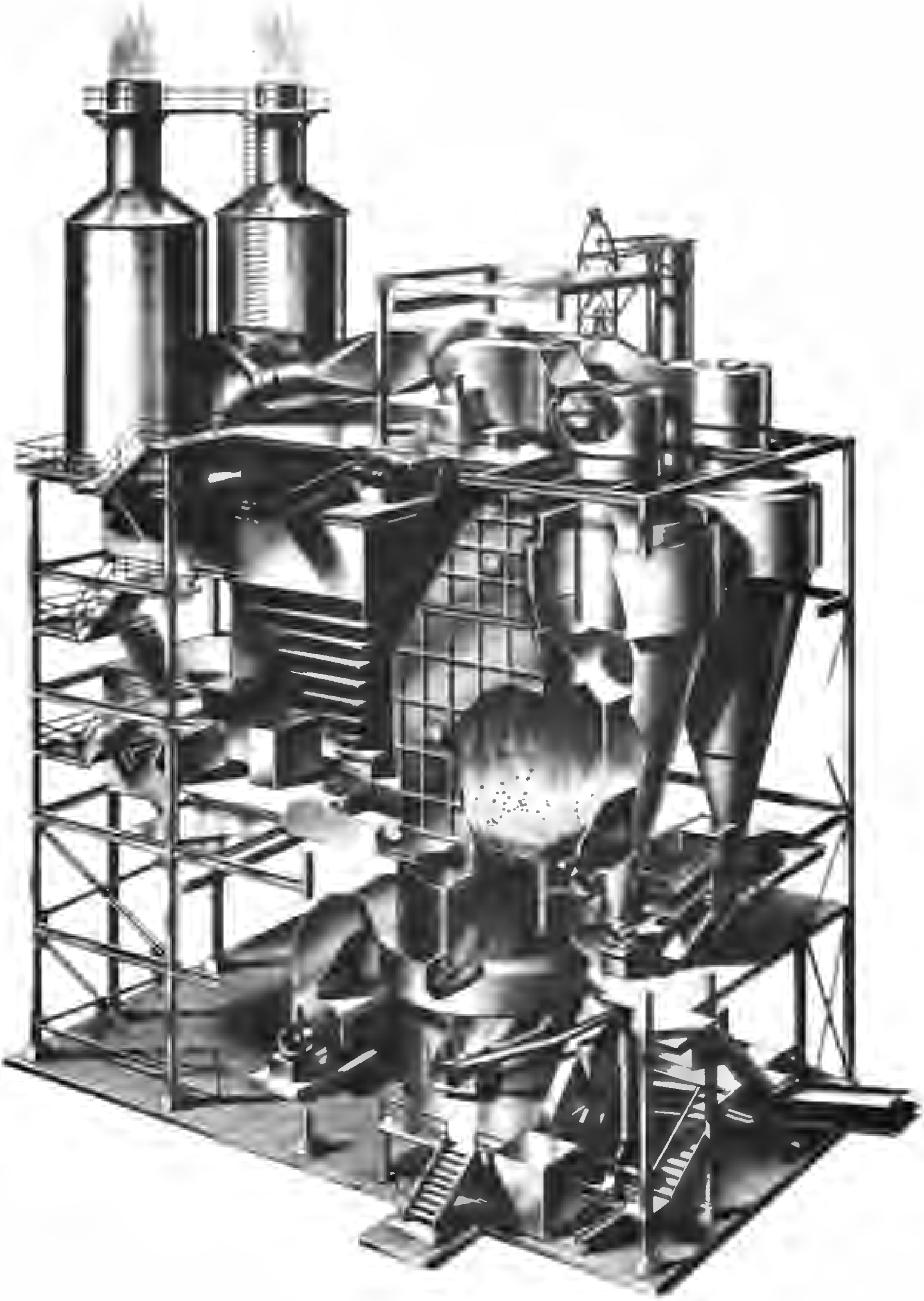
The McNally Flowdyrer will be sized to evaporate 12 tons per hour of water from a feed of 150 tph (dry basis) of 1 1/2" x 0" coal and delivery a product containing 3% total moisture. The dryer will be fueled by pulverized dried coal and will be fitted with a cyclone dust collector to capture the coarse dusts and a high energy scrubber to reduce exhaust dust to acceptable limits.

The Flowdryer releases measureable quantities of particulate matter and sulfur, probably in the form of H_2S. The latter depends, to a large extent, upon the percent pyrites present in the fuel. To overcome these problems, a high-energy wet scrubber has been included, in recent years, as part of the Flowdryer package. No data are available on the exact quantities of sulfur and particulate matter emitted from the unit. However, it would seem reasonable to conclude that, if necessary, emission could be further reduced by conventional equipment at a modest price. If this were not found to be the case, an alternate means of drying coal would have to be devised. Effluent from the scrubber will be disposed to the power plant ash pond. The Flowdryer is illustrated in Figure A5-6 and Figure A5-7.

Coal from the dryer will be crushed to 1/4" x 0" size in a reversible hammermill and, along with the coarse dust from the dryer cyclone, delivered via 24" belt Conveyor No. 3 to the plant surge bin. Conveyor No. 3 will be fitted with belt scale No. 2 to weigh and record coal quantities to the boilers.

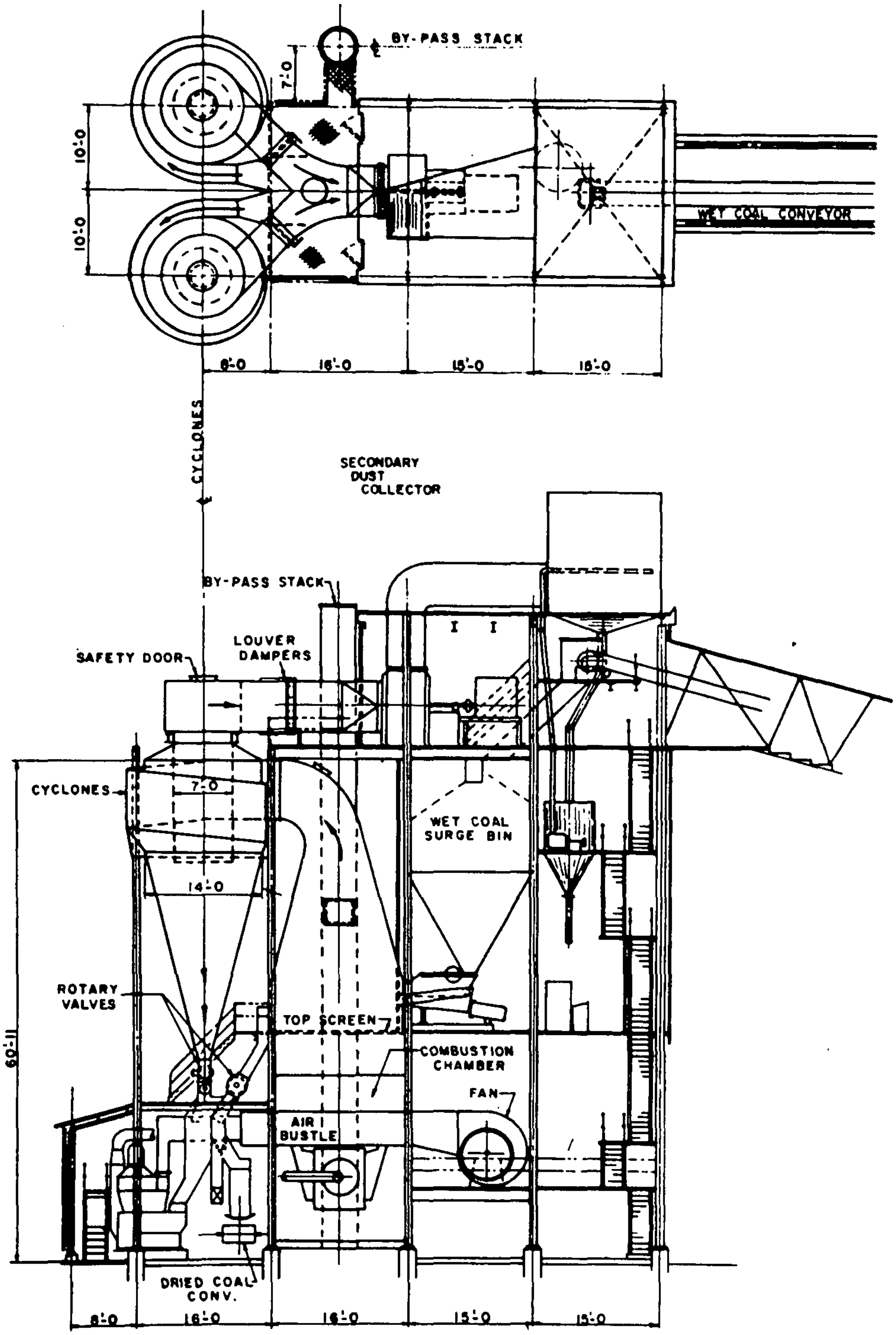
From the 150 ton capacity plant surge bin the coal flow will be split into two streams each served by a 0 to 75 tph vibrating Feeder F4A and F4B to its associated coal feeder surge silo group through scraper Conveyors No. 4A and 4B.

The silo feeding system will be provided with automatic sequential controls with provisions for operator to overide to accommodate unusual operating conditions. The plant surge bin



McNally Flow Dryer

Fig. A5-6



TYPICAL INSTALLATION DIAGRAM OF A McNALLY FLOWDRYER

FIGURE A5-7

level will control the feed to the dryer and in turn the dryer surge bin will regulate the silo withdrawal feeders.

A5.2.3 <u>Equipment</u> The coal handling system will consist of the following equipment, materials and services:

1. Receiving Hopper
2. Feeders No. F1A, F1B, F1C and F1D
3. Chutes
4. Conveyor No. 1
5. Metal Detector
6. Belt Scale No. 1
7. Reclaim Hopper
8. Shut Off Gate
9. Feeder No. F2
10. Silo
11. Feeders No. F3A and F3G
12. Conveyor No. 2
13. Tramp Iron Magnet
14. McNally Flowdryer
15. Mill
16. Conveyor No. 3
17. Belt Scale No. 2
18. Surge Bin
19. Feeders No. F4A and F4B
20. Conveyors No. 4A and 4B
21. Dust Collecting Equipment
22. Ventilating and Heating
23. Control Equipment
24. Motors and Controls

25. Structures

26. Foundations

27. Electric Wiring

28. Erection

A 5.2.4 <u>Cost</u> McNally Pittsburg indicates the cost for the system described should be about $2,500,000. The flow dryer represents about $750,000. This includes an additional 24" belt conveyor to accommodate the additional handling required by the thermal dryer. The complete system contributes about $8.3/KW to the overall cost at the 300 MW level.

McNally Pittsburg indicates for an increase in capacity to handle a burn rate of 226 tons per hour, the conveyor sizes should stay substantially the same; however, their horsepower would increase and the size of the McNally Flowdryer would have to be increased. The probable additional cost for increasing the system might amount to $250,000 for a total of about $2,750,000. This represents about $4.5 KW at the 600 MW level.

McNally Pittsburg reports the following cost analysis for the flow dryer:

TABLE A5-1

Operations	Cents/ton
Operating Labor	.955
Fuel (Start-up Oil and Coal)	3.475
Power	2.140
Miscellaneous (Operating Supplies)	.105
Total Operating Costs	6.675
Maintenance	
Maintenance Labor	1.366
Maintenance Supplies	.332
Total Maintenance Costs	1.698
Total Cost per ton of Dried Coal	8.373

The operating cost of a McNally Flowdryer is generally about 10 cents per ton. The figures on the previous page are based on actual costs at a typical installation over a five month period.

Johnson and Auth (1)* and Lyons and Richardson (2) report that operating costs of drying ranges from 8 to 18 cents per ton of dried coal produced with the majority of dryers having costs of approximately 10 cents per ton.

On a basis of per ton of water removed, operating costs range from \$1.00 to \$3.00 with the majority of the dryers having costs of approximately \$1.50 per ton of water removed.

The Bureau of Mines at Bruceton, Pennsylvania (3) report that drying costs at the mine ran about 15 cents per ton of dry coal or \$2.00 per ton of water removed.

Capital charges based on the cost of the dryer furnace, the dryer, the duct work, exhaust fan and exhaust stack, for a plant having one shift operation for a 200 day year, assuming complete depreciation in 10 year range from 4 to 12 1/2 cents per ton of dried coal, or 60 to 188 cents per ton of water removed. The capital charges for the McNally Flowdryer come to about 7.15¢/ton of dried coal.

Johnson and Auth (1) indicate that maintenance, a major item representing from one-third to one-half of the total operating costs of drying, could be reduced in many cases by (1) coating exhaust ducts, fan housing and exhaust stack to reduce corrosion; (2) the use of corrosion and fly ash-resistant fan impellers; (3) the use of stainless steel for exposed steel surfaces such as

*(1) A. J. Johnson and G. M. Auth, "Fuels and Combustion Handbook", McGraw-Hill, New York, 1951.

(2) Lyons and Richardson, AIME Tech. Pub., 2399, August, 1948.

(3) Communication with Bureau of Mines, Bruceton, Pennsylvania.

screens; (4) adequate controls to prevent overheating of furnace dryer inlet ducts; and (5) a well-planned maintenance program.

The coal handling plant proposed differs from conventional practice by virtue of the coal drying requirements and the McNally Flowdryer. The costs are comparable as shown in Table A5-2

TABLE A5-2 COMPARISON OF CAPITAL COST OF COAL HANDLING EQUIPMENT

Plant Size (MW)	100(4)	200(4)	230(4)	246(4)	327(4)	300 Fluid Bed	600(5)	600 Fluid Bed
Coal Handling and Storage	1010*	1720	1860	1925	2240	2,500	3,122	2,750
$/KW	10.10	8.6	8.1	7.9	6.85	8.35	5.20	4.55
Fuel Burning	760	1295	1400	1450	1685	1800	†	2800
$/KW	7.60	7.4	6.1	5.9	5.15	6.00		4.66

* Values in $1000
† Not reported

Fuel burning equipment generally includes bunkers, feeders, exhauster mills, burners, conduits and controls. They have been included here to be certain appropriate comparisons are made. Although the drying operations would normally take place in the mill under coal burning equipment, this operation has been included as part of the coal handling operation because of its relocation in the sequence of operation.

(4) T. V. Rallo and G. Guarraia, "Technology and Economics for Domestic Boiler and Power Plant Designs" F.W. Report, June 1969, Fluidized Bed Combustion, Monthly Progress Report, January 1970

(5) S. J. Jack, Power Systems Planning-Fuels and Energy, Westinghouse, July 1970.

A 5.2.5 Sorbent Make-Up Storage and Feed The sorbent feed make-up system which consists of a receiving hopper, enclosed storage silo and discharge gallery is illustrated in Figure A5-8. Orientation with respect to the coal handling system is shown in Figure A5-1. This equipment was priced by McNally Pittsburg.

.5.1 Assumptions and feed conditions The sorbent will have the characteristics outlined below.

Table A5-3 (6)

SO_2 Sorbent Analysis Used for Preliminary Fluidized Bed Boiler Analyses Weight/Percent as Received

Component	Dolomite BCR 1337
SiO_2	0.78
Al_2O_3	0.15
Fe_2O_3	0.25
MgO	45.0
CaO	53.0
TiO_2	0.02
Si_2O	<0.03
Na_2O	<0.02
K_2O	<0.1
MnO_2	<0.03

(6) R. W. Coutant, J. S. McNulty, R. E. Barrett, J. J. Carson, R. Fischer and E. H. Lougher, "Investigation of the Reactivity of Limestone and Dolomite for Capturing SO_2 from Flue Gas", August 1968, Battelle Memorial Institute.

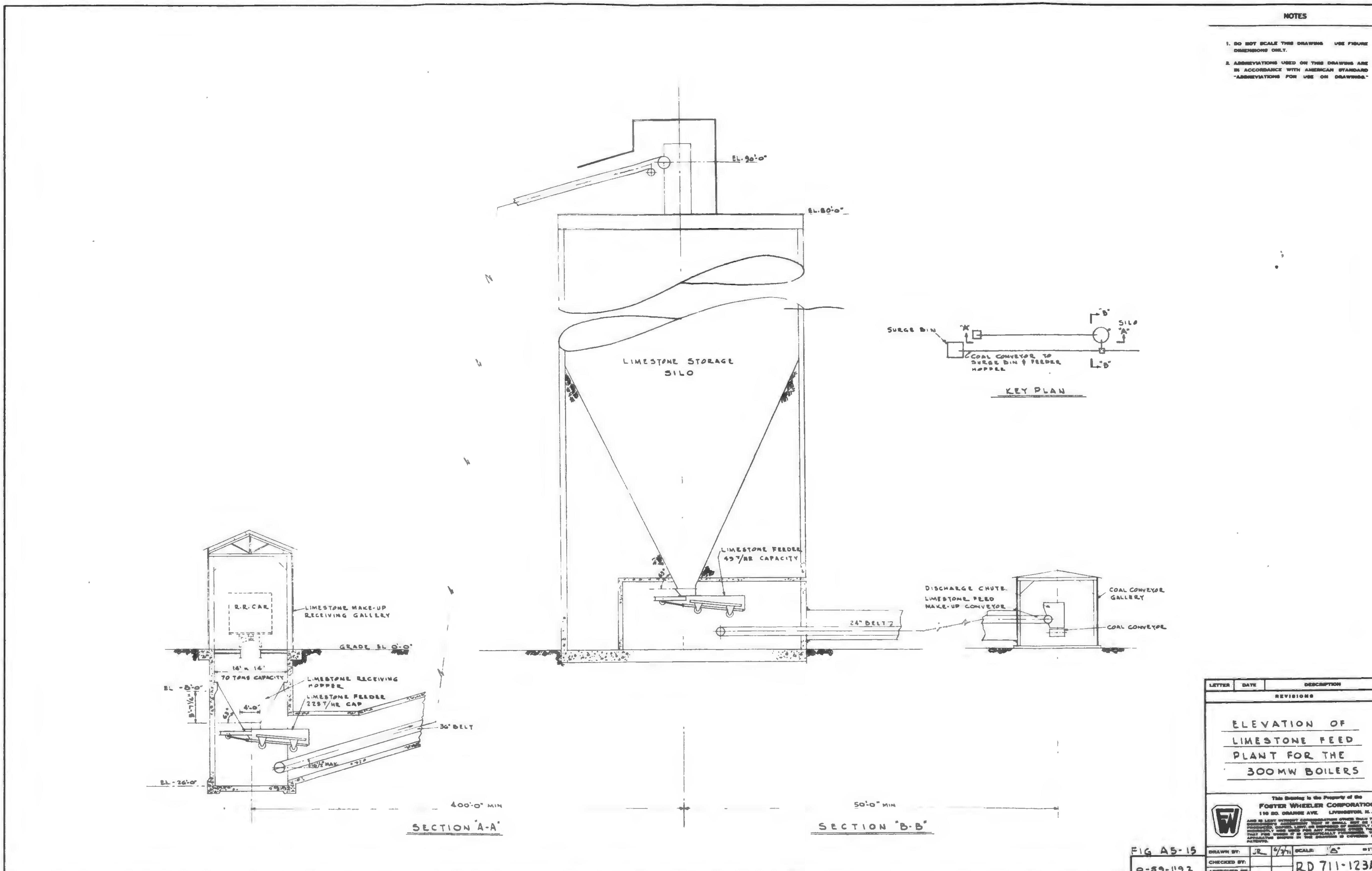

NOTES
1. DO NOT SCALE THIS DRAWING USE FIGURE DIMENSIONS ONLY.
2. ABBREVIATIONS USED ON THIS DRAWING ARE IN ACCORDANCE WITH AMERICAN STANDARD "ABBREVIATIONS FOR USE ON DRAWINGS."
EL. 90'-0"
EL. 80'-0"
LIMESTONE STORAGE SILO
LIMESTONE FEEDER 45 T/HR CAPACITY
SURGE BIN
SILO "A"
"B"
COAL CONVEYOR TO SURGE BIN & FEEDER HOPPER
KEY PLAN
R.R. CAR
LIMESTONE MAKE-UP RECEIVING GALLERY
GRADE EL 0'-0"
14' x 14'
70 TONS CAPACITY
LIMESTONE RECEIVING HOPPER
LIMESTONE FEEDER 225 T/HR CAP
EL -8'-0"
4'-0"
36" BELT
16 1/2° MAX.
EL -26'-0"
400'-0" MIN
SECTION "A-A"
DISCHARGE CHUTE
LIMESTONE FEED MAKE-UP CONVEYOR
24" BELT
COAL CONVEYOR GALLERY
COAL CONVEYOR
50'-0" MIN
SECTION "B-B"
LETTER
DATE
DESCRIPTION
REVISIONS
ELEVATION OF LIMESTONE FEED PLANT FOR THE 300MW BOILERS
This Drawing is the Property of the FOSTER WHEELER CORPORATION
110 SO. ORANGE AVE. LIVINGSTON, N.J.
DRAWN BY: JR
SCALE:
CHECKED BY:
APPROVED BY:
RD 711-123A
FIG A5-15
0-59-1192

Fig. A5-8

The bed material size will range from 1000-5000 microns with an average diameter of 2500 microns.

Feed requirements were based on the assumption that 90% of the SO_2 would be removed with six times stoichiometric feed ratio.

The heat generated would be 3×10^6 Btu/ton of $CaSO_4$ produced by the reaction of SO_2 with CaO at 1900°F.

The recycle rate of the sorbent will be based on the assumption that 3% of the sorbent is elutriated from the bed and 10% of the sorbent is ejected during regeneration.

The sorbent will be received by unit train in covered car with 100 ton capacity pre-crushed and dried to 1-2% moisture. Material with such specifications is not uncommon. Covered storage is provided. The material would be fed to the coal conveyor after the coal drying and crushing operation just prior to the surge bin for the feed injection system.

Storage offers several options. The incoming material could be stored either in a storage silo or in a combination of silo and pile storage. The silo is essential in any event, even if pile storage were selected the silo would need to serve the same purpose as bunkers in the coal feeding operation, that of giving some leeway in scheduling the workers who reclaim from the storage pile and move coal to the bunkers.

At feed rates of about 200 tons/day to 400 tons/day for the 300 MW system and 600 MW system respectively, the single silo system handling "live" and "dead" storage appears to be preferred over the pile-silo system. A silo of about 2000 tons capacity should handle the system requirements. The direct feed system eliminates the need for reclaiming hoppers, reclaiming equipment and operators, limestone dryers and limestone crushers. It should simplify the delivery of limestone by reducing the frequency of delivery and at the same time handle interruptions in operations that might result from delays in delivery. TVA indicates truck capacities would run about 20 tons per car load (5). Rail capacities are running about 100 tons per car load.

.5.2 Scope The sorbent will be received in covered rail cars with truck unloading as an option. The material will be discharged at the receiving hopper which has a capacity of 100 tons. Its feeder capacity and conveyor belt were selected for 20 minute unloading time for a 100 ton car. The storage silo holds 2000 tons of limestone equivalent to two weeks' supply. The limestone silo differs from the coal silo in that the hopper slope should be increased to about 63° to handle the small particle size. Since the stone is pre-crushed to the size required, it can be discharged on to the coal conveyor at grade level just after the Flowdryer.

(5) ____ "Sulfur Oxide Removal from Power Plant Stack Gas; Conceptual Design and Cost Study Sorption by Limestone or Lime Dry Process", Tennessee Valley Authority, prepared for National Air Pollution Control PB 178972, 1968.

Investment Capital investment for the 300 MW and 600 MW limestone handling plant are listed in Table A5-4. Comparisons are made with data reported for a dry limestone injection system excluding pulverizing and injecting equipment and attendent power plant modifications. Engineering contractors' fees and contingency allowance have been prorated to compensate for the exclusions. Data on the fluid bed limestone handling plant was provided by McNally Pittsburg.

TABLE A5-4

SUMMARY OF ESTIMATED FIXED INVESTMENT REQUIREMENTS

	McNally 300(MW)	Pittsburg 600	(5) Dry Limestone 200	Injection 1000
1) Yard Improvements			25,000	50,000
2) Limestone Storage and Handling Facilities				
2(a) Concrete Foundations and Conveyor Tunnel			80,000	180,000
2(b) Receiving Hopper Storage Silo Conveyor Conveyor Supports and Bridges, and Powerhouse Storage			71,000	200,000
Sub-total			176,000	430,000
3) Engineering Design*			13,000	23,500
4) Contractor Fees and Overhead*			19,300	34,000
5) Contingency Allowance			13,000	33,000
Total	325,000	325,000	221,300	520,000

* Proportion from Original Estimate to Include Only that Equipment Listed.

A5.3 Coal Feeding

The coal feeding system includes all the process equipment from the surge bins adjacent to the boiler to the inlet coal feed points at the boiler shell. This includes surge bins, lock hoppers, fuel injectors and pneumatic transport lines. A state-of-the-art review indicated the localized fluidized bed feeder and lock hopper system proposed by Petrocarb best fit the system feed requirements of pressurizing to 10 atm., and multipoint injection. Petrocarb developed and priced a system for four module boilers operation at 300 and 600 MW levels.

A5.3.1 Assumptions The feed system selected must be capable of transporting coal from the Ohio Pittsburgh No. 8 Seam sized to 1/4" x 0" with a portion of limestone sized from 1000 to 5000 microns. The material would be dried to insure non-agglomerating and free flowing conditions.

The system must be capable of transporting the feed about 150 feet to elevations of about 100 feet. A total of 80 feed lines are required serving four boilers each containing five beds. Each feed line should be capable of turn down of 30%.

A5.3.2 Petrocarb System

.2.1 Description The Petrocarb System of coal feeding was recommended by all of the people consulted on coal feeding (7) (8). Petrocarb also felt pressurized fluid bed combustion was a

(7) Communication with the Bureau of Mines at Morgantown, West Virginia.

(8) Communication with the Bituminous Coal Research at Pittsburgh, Pennsylvania.

proper and interesting application of their system.

The Petrocarb System simply consists of a surge bin which receives the coal and dampens fluctuations in flow rate, a lock hopper which pressurizes the coal, a localized fluidizing fuel injector or feeder which feeds and distributes the coal, and several dense phase pneumatic transport lines which convey the coal. In the local fluidization hopper fluidization takes place only in the solids in the vicinity of the outlets to the conveying tube. In a conventional fluidizing hopper the fluidization air would have to be recycled and repressurized. In the local fluidizing hopper some power is saved as the fluidizing air also becomes a portion of the pneumatic transport medium. Petrocarb has combined these components in what they call the Petrocarb Mark IV Coal Injection System illustrated in Figure A5-9 and A5-10. The system is normally recommended for blast furnace operation. In this case, it was modified to handle the higher pressures.

The primary injector is the heart of the system. It has the function of maintaining a continuous feed of coal to the furnace at a designated rate, and distributing the coal among the working tuyeres in proportion to the blast air passing through each tuyere.

Above the primary injector is the storage injector, which automatically replenishes the coal injected from the primary injector without interrupting or disturbing the injection process.

Above the storage injector is a surge bin containing a supply of coal for ready transfer through the storage injector to the primary injector. Prepared coal is fed continuously to this

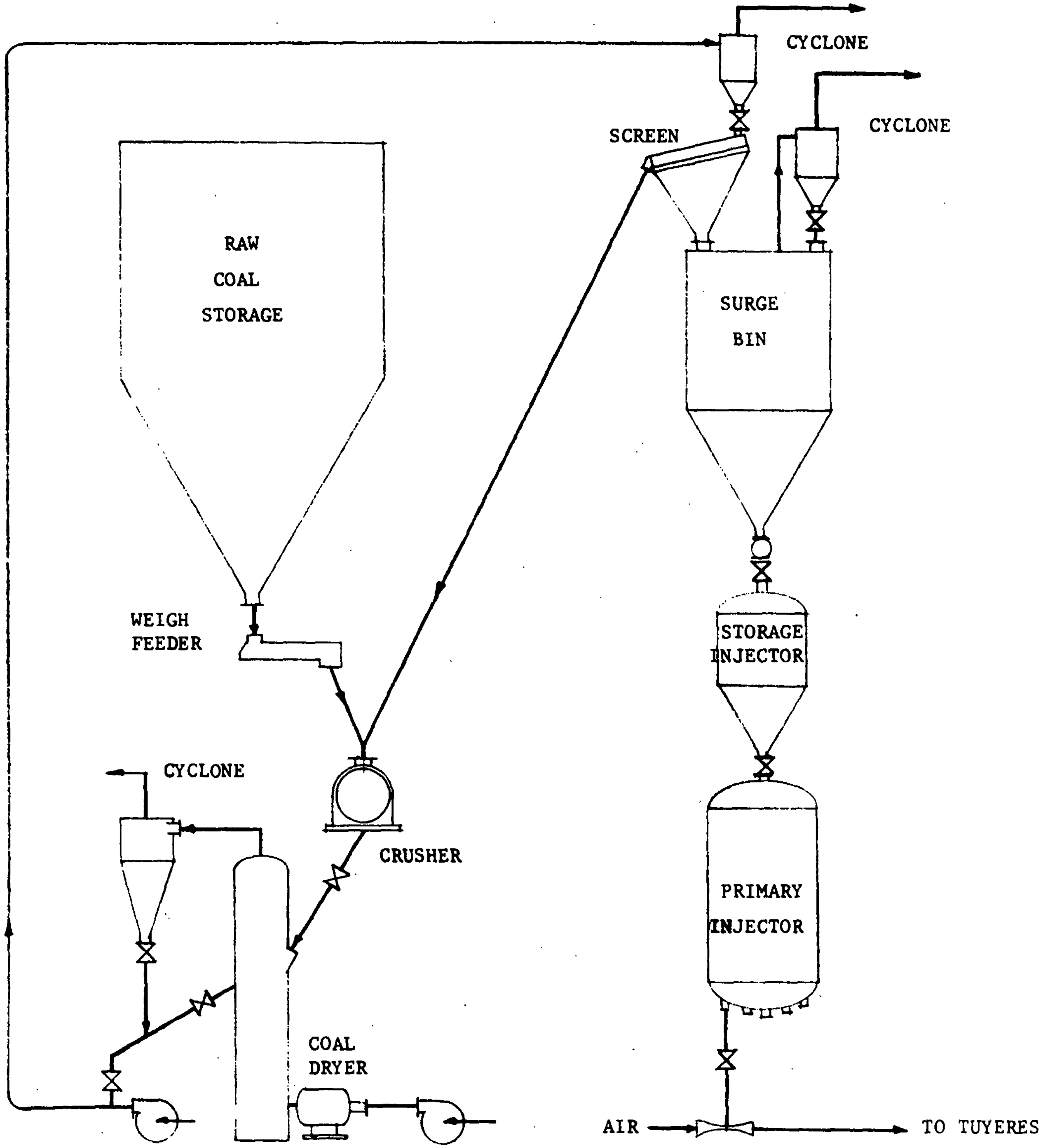


FIGURE A5-9 COAL HANDLING AND INJECTION SYSTEM

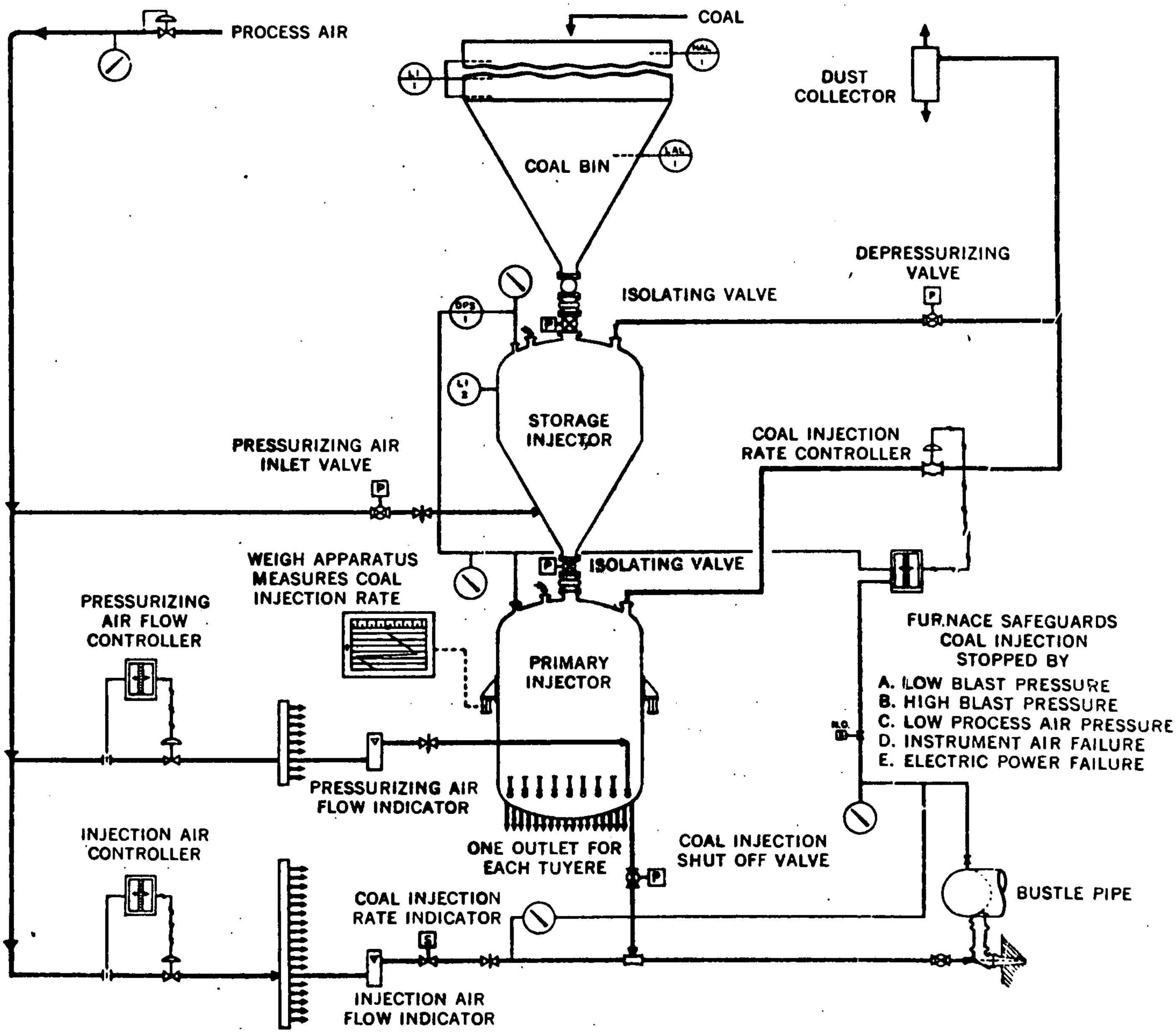


FIGURE A5-10 PETROCARB PRESSURIZED COAL FEEDING SYSTEM bin, which is sized to provide a ready supply for the storage injector. The means by which the surge bin is automatically kept supplied depends on local conditions. Frequently the Petrocarb pneumatic transport system is recommended. In this case, the McNally Handling Plant provides the continuous feed of coal to the surge bin. Regardless of the system used, means for maintaining an adequate supply of coal in the bin must be employed.

.2.2 Instruments and controls The instruments and controls of the Petrocarb Mark IV System, alarms and indicator lights are mounted on a semigraphic, mimic control panel showing the complete installation from the storage bins to the furnace tuyeres, it is incorporated in the control panel.

"Instrument alarm annuciators, pressure gauges and valve position indicating lights, mounted on the face of the panel enable the operator to determine at a glance the cyclic position of the automated system. The rate of flow is determined by the different pressure between the furnace and the primary injector, and once the plant has been calibrated, the rate of coal injection can be varied by altering the setting of the differential pressure controller which thereafter maintains a constant differential pressure between the hot furnace and the injector. Thus, if the pressure in the furnace increases, the pressure in the primary injector is automatically increased in the same proportion, and the reverse compensation takes place if the pressure in the hot blast main drops." (9)

(9) E. M. Summers, "Engineering and Design Considerations of Coal Injection", AIME Ironmaking Procedures, Vol. 23, pp. 69-96, 1964.

"The automatic cycle is controlled by sequence timers which trip relays to energize the operative circuits, and a cycle is started when the weight of the coal in the primary injector vessel falls to 2000 lb., thus producing a low level signal from this vessel. On receipt of this signal, the sequence automatically continues in the following order:

a. The ball valve between the primary and storage injector vessels and the pressurizing valve in the storage injector vessel open, while the pressure equalizing valve between the two vessels closes, coal then flows from the storage into the primary injector vessel.

b. A low level signal from the storage injector closes the ball valve and the pressurizing valve, and the vessel is vented to atmosphere.

c. The vent closes when the pressure in the storage vessel reaches atmospheric.

d. The inlet ball valve on the top of the storage injector vessel opens, together with a vent valve fitted with a cyclone dust extractor and rotary valve.

e. Coal flows from the surge bin into the storage injector.

f. A high level signal showing that 4000 lb. of coal have been transferred into the storage injector closes the ball valve and vent valve on top of the storage injector. The vessel is, then, pressurized to the same pressure as the primary injector.

g. The pressure equalizing valve opens leaving the storage injector ready for the next cycle.

"At the completion of each full cycle the timers return to their original position. The entire cycle takes 20 minutes to complete, and the storage vessel then stands full and pressurized until the primary injector calls for coal."

"Several interlocks and switches are included in the instrumentation of the plant to insure safe operation, and an alarm system comes into operation in any one of the following circumstances:

1. Low solids flow to any of the tuyeres
2. Low solids flow to storage injector
3. High and low level in the storage hopper
4. Plant air failure
5. High level in the surge bin
6. High and low level in the storage injector
7. Low level in the primary injector
8. Failure of fluidizing air to primary injector.

In addition to sounding an alarm, the coal feed from the primary injector is cut off in any one of the following abnormal conditions:

1. Excessive high or low pressure in the hot blast main
2. Plant air failure
3. Instrument air failure."

"Other special devices in the instrumentation are as follows:

a. An interlock is provided which prevents the ball valve between the primary and storage injectors opening until both vessels are at the same pressure.

b. When the pressure in the furnace falls, the differential pressure controller automatically drops the pressure in the primary injector vessel. To avoid the consequential large scale discharge of air from the primary injector vessel at times when the furnace pressure has been deliberately reduced, a special arrangement comes into operation. A solenoid valve in the line from the furnace closes when the pressure falls to below a prescribed level and another solenoid valve in the instrument air line opens and this now maintains a differential with the primary injector through the differential pressure controller."

"When operating normally, the injection plant is fully automatic from the storage bunkers onward. However, from time to time, it is necessary to break the automatic sequence and adopt manual control. The changeover from automatic to manual control is done very simply by altering the switches on the panel to the desired positions. In doing this, the manual cycle must follow the same steps as the automatic cycle. The timers which control the automatic cycle will return to their "home" position and remain there until the plant reverts to normal automatic working."

.2.3 Scope For the specific application of the fluidized bed steam generator consisting of four modules each containing four coal fed beds with four independent coal injection points, Petrocarb proposed a system with a feed rate turn down ratio of 30%. This will be provided as 30% of the total rate to each bed. Four points of injection are to be provided for each bed. Each of the points will require a direct pipe run, supplied by Purchaser, from a common primary injector. The feed rate through these lines would be field calibrated to provide equal flow, with only a small deviation allowed, to each of the four feed points.

Each of the sixteen (16) injection systems will include two vessels. The primary injector to be maintained at a preset and controlled feed rate at all times. The second vessel, a storage injector, will periodically replenish the material supply in the primary unit, operating automatically in lock-hopper manner. The feed rates for the four fluid beds are not so diverse as to dictate disparate vessel sizing. Therefore, for the sake of economy all storage injectors are sized alike as are the primary injectors, six and eight feet in diameter respectively.

In addition to the sixty four (64) injectors complete with their proprietary internals the stated estimate includes the following equipment:

- Complete lot of process and specialty valves. Valves involved in the automated cycling operations will be either electric motor or air cylinder actuated ball valves.
- Sixty four proprietary injection solids/air mix assemblies, one for each of the feed points.
- All required flex hoses and expansion joints.
- Vessel relief valves.
- Complete instrumentation and controls for functional operation and vital indications.
- A semi-graphic panel with enclosed internally piped and wired cubicle for each of the four injection systems serving each of the four boilers.

The estimate also includes the following design and services:

- Schematic piping and wiring diagrams.
- Design criteria and specifications for design of structures to support the vessels on their force cell mounts.
- Consultation in design and approval of fabrication and installation of injection lines from primary injectors to fluid bed feed points.
- Bills of Materials.
- Equipment Manuals.
- Operating Instructions.
- Start-up assistance, one engineer for a period of thirty (30) days. Subsistence and local transportation to be for the account of the Purchaser.

Compressed air requirements for each group of four injection units serving a boiler is estimated to be 8400 scfm at 200 psig.

Oil free air having a dew point sufficiently low that moisture will not condense in the injection systems after expansion from the 200 psig regulated pressure is essential.

Excluded from this estimate are the following:

- Air compressors and related equipment.
- Coal bins and bin gates.
- Structural steel.
- Piping, standard valves and fittings.
- Electrical wiring and conduit.

.2.4 Cost The cost of the fuel injection system for four field erected vessels including calibration of coal feed lines for a 300 MW plant is $1,800,000. The same plant with eight outlets supplying coal to supplementary units such as the regenerator or simply increasing the number of bed feed points is $2,000,00.

Extrapolating the system to a 600 MW level the price for a four outlet feed system is $2,800,000 and an eight outlet feed system is $3,000,000. The 600 MW feed system would use the same number of vessels lock hoppers and bins. The cross-sectional area of the vessels would simply increase in proportion to the feed ratio.

A5.3.3 Literature survey The art of pressurizing coal and feeding or pumping at elevated pressures is not clearly defined. This area of technology has received only sporadic attention within the last 10 to 20 years as interest has been directed at such coal burning processes as the coal fired gas turbine, the pressurized steam generator and pressurized coal gasification. Selection of the appropriate coal feeding system for the pressurized boiler thus required reviewing the literature and soliciting information from various investigators.

In the various processes investigated the solid fuel has been pulverized, crushed or sized. Loose-packed bulk densities range from 20 lb/ft^3 for sized coke, 30 lb/ft^3 for pulverized coal, 40 lb/ft^3 for sized coal, 55 lb/ft^3 for crushed coal, 100 lb/ft^3 for stone and up to 200 lb/ft^3 for rich ores. (14)

(14) M. N. Aref, "Pressurization of Granular Solid Fuels".

Capacities run from 2 tons/hr for small gas turbine plants to 100 tons/hr or higher for big gasification plants. Delivery pressures range from 5 to 1000 atm. and powdered fuel size, for satisfactory pumping varies from 80 percent through 200 mesh for pulverized coal down to 1/8 x 0 or 1/4 x 0 for crushed coal.

Solid fuel pumping has been broken down into two basic groups; cyclic action and continuous action. Some systems are hard to define as they may contain components that fall into either group. Each category includes the following fuel feeding systems:

1. Cyclic action group:

   Simple lock hopper, lock hopper with moving wall, positive displacement ram pump, solid extension, slurry pump.

2. Continuous action group:

   Positive-displacement gear pump, multi-stage centrifugal compression with air, peristaltic-displacement (rubber) pump, fluid bed fuel injector, jet pump fuel injector.

.3.1 <u>Cyclic action group</u>

.1.1 <u>Simple lock hopper</u> The lock hopper system is probably the most common and well established of all the methods of pressurizing granular solids. The method employs as shown diagrammatically in Figure A5-11, two high pressure closed hoppers connected in series to an atmospheric hopper. The sequence of third cycle is as follows:

1) Valve 1 is opened. The solid is introduced by gravity at atmospheric pressure into the lock hopper, Valve 2 being closed.

2) Valve 1 is closed. Auxiliary pressurizing air is applied through Valve 3 to the lock hopper until it reaches the same pres-

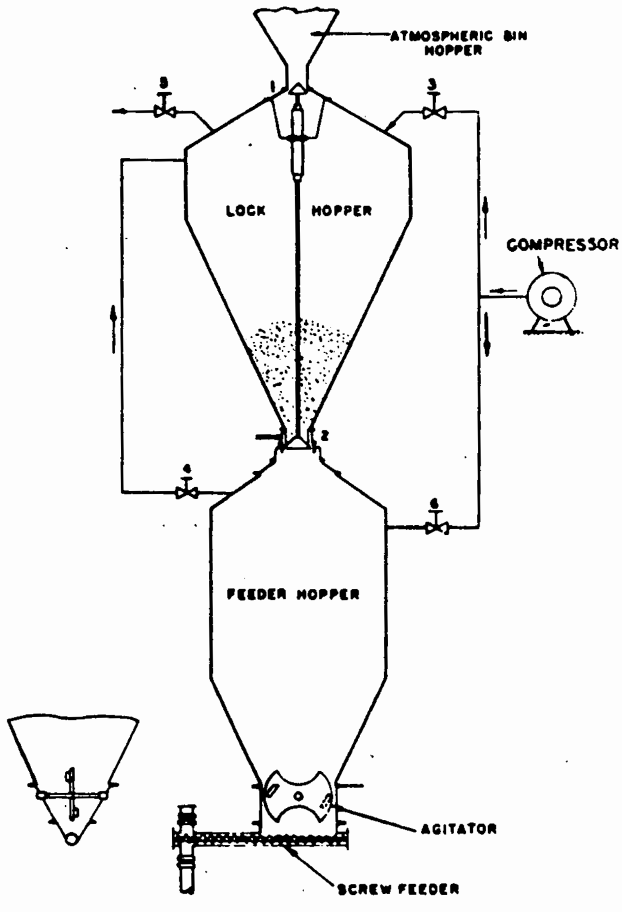


FIGURE A5-11 TYPICAL LOCK HOPPER ARRANGEMENT (14) sure as that of the feeder hopper.

3) Valve 2 is opened. Solid is then displaced by gravity into the pressurized feeder hopper at system pressure. Valve 4 serves to equalize pressure in both hoppers.

4) Valves 2, 3 and 4 are closed. Valve 5 is open to vent.

In each cycle a certain volume of pressurized air must be vented to the atmosphere. This volume of vented air equals the difference between the volume of lock hopper and any dead space inside of it as for example the operating valves and rods. The energy required to pressurize this particular volume of air added to the working pressure of the system is wasted.

The pressurizing air displaces the whole bulk volume of coal, thus an equal volume of air per cycle has to be vented. This represents a power loss for each cycle according to the volume and pressure of air involved. The minimum power loss assuming polytropic compression is given in hp-hr/ton of coal by:

$$W_{min} = CRT \text{ Ln } r/\eta_{isoth} S \tag{1}$$

$$S = [(\rho_{fb}/M_a r)-1]/[1-(\rho_{fb}/\rho_{ac})]$$

Where:

C = Constant equals 1.01×10^{-3}

S = Coal/Air Ratio by Weight

ρ_{fb} = Bulk Density

ρ_{ac} = Bulk Density

M_a = Specific Weight per Cubic Foot of Air

T = Temperature Level

r = Compression Ratio

R = Gas Constant

W = Minimum Work

η_{isoth} = Efficiency due to isothermal compression

The weight ratio of coal to air, S, at varying compression ratios is given by equation 1. The variation of S with the compression ratio r at different bulk densities of the coal-air mixture as a parameter is shown in Figure A5-12. Assuming a constant isothermal efficiency for multistage compression constant at .90, the minimum power loss in venting the pressurized air from the lock hopper per ton of coal is as shown in Figure A5-13 for different final bulk densities of coal-air mixture.

The lock hopper system is a means of pressurizing coal. As already indicated, pressurizing is done at considerable power loss in venting and pressurizing the lock hoppers and feeder hoppers. The complete system requires feeders at the inlet and exit which control the flow rate of coal. Stone feeders, table feeders and screw feeders have been used with the lock hopper system to regulate the flow of coal from the pressurizing vessel. (15, 16, 17, 18, 19, 20). They are expensive and subject to wear and are limited

15) J. D. Spencer, T. J. Joyce, and J. H. Faber, "Pneumatic Transportation of Solios", Proceedings: Institute of Gas Technology - Bureau of Mines Symposium, Morgantown, West Virginia, October 1965, U. S. Bureau of Mines IC 8314.

16) W. J. Morley and B. Parmington, "Preliminary Investigations into the Downdraft Combustion of Granulated Coal in Air-Cooled Equipment", Mechanical Engineering Technical Memorandum 303, Department of Supply, Australian Defense Scientific Service, Melbourne, Australia, August 1967.

17) W. J. Morley, "Pressurized Combustion Pot Tests of a High Volatile Bituminous Coal", Mechanical Engineering Note 270, Department of Supply, Australian Defense Scientific Service, Melbourne, Australia, July 1965.

18) J. P. McGee, J. Smith, R. W. Cargill and D. C. Strimbeck, "Bureau of Mines Coal-Fired Gas Turbine Research Project", Redesign and Assembly of Turbine, United States Bureau of Mines Rl 5958, 1962.

19) J. Smith, R. W. Cargill, D. C. Strimbeck, W. M. Nabors, and J. P. McGee, "Bureau of Mines Coal-Fired Gas Turbine Research Project", United States Bureau of Mines 1R 6920, 1967.

20) W. R. Huft, J. H. Holden, L. F. Willmott, and G. R. Strimbeck, "A Pilot-Scale Fluidized Coal Feeder Utilizing Zone Fluidization", United States Bureau of Mines 1R 6488, 1964.

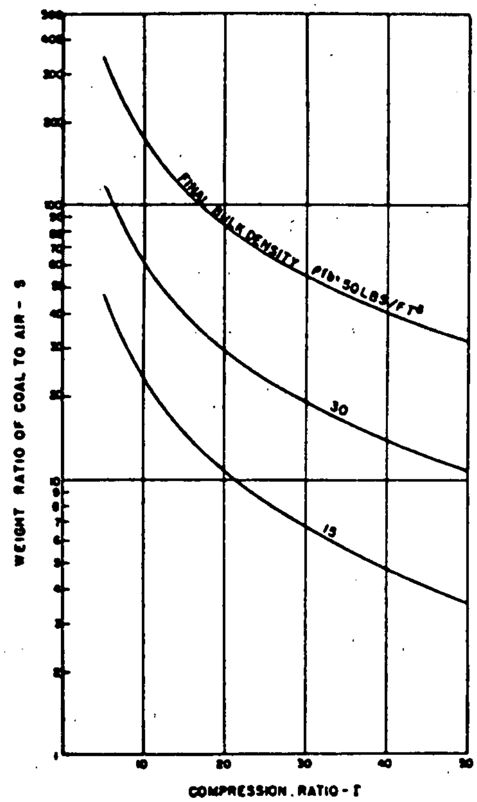


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FIGURE A5-12 VARIATION OF COAL/AIR RATIO WITH COMPRESSION RATIOS AT VARIOUS BULK DENSITIES

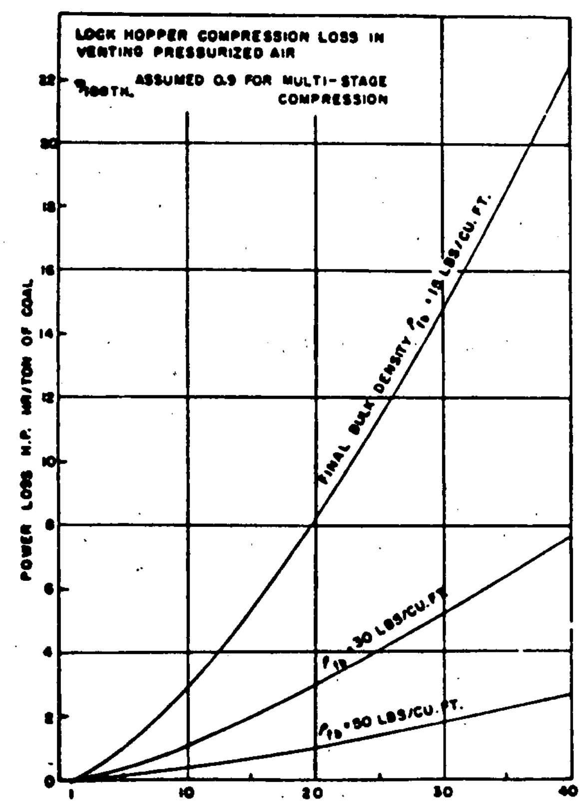

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FIGURE A5-13 POWER REQUIREMENTS FOR DIFFERENT COMPRESSION RATIOS AND FINAL BULK DENSITIES with regard to distribution. Only one feed point per lock hopper is available. Feed to the lock hoppers has been accomplished pneumatically as illustrated in Figure A5-14 or by such mechanical devices as the rotary valve or screw pump as illustrated in Figures A5-15.

The conventional lock hopper system is reliable and well established with regard to experience. However, it requires a large capital outlay for feeders and controls, it consumes substantial quantities of power per ton of feed, and is subject to considerable wear in the feed system.

.1.2 Modified lock hopper system Conservation of energy lost during pressurizing and venting operations is one obvious source of cost reduction in the lock hopper system. This might be simply accomplished by placing lock hoppers in series. Figure A5-16 illustrates the ratio by which the operating pressure is reduced in proportion to the pressurizing level for 2, 3, and 4 equal hoppers pressurized in series. It is noticed that a considerable reduction in pressure to be vented can be obtained. If the hoppers are of equal volume, the pressure of air to be vented after pressurizing two, three, or four hoppers approaches 0.5, 0.34, and 0.25 of that of one hopper, respectively as the bulk density of the mixture is reduced. Figure A5-17 shows what power savings can be realized for a coal with a bulk density of 30 lb/ft^3.

Other modifications to the lock hopper system that have been proposed to reduce power consumption include lock hoppers in parallel and cascading of pressurizing air through a system of lock hoppers operating in parallel and serving a multiplicity

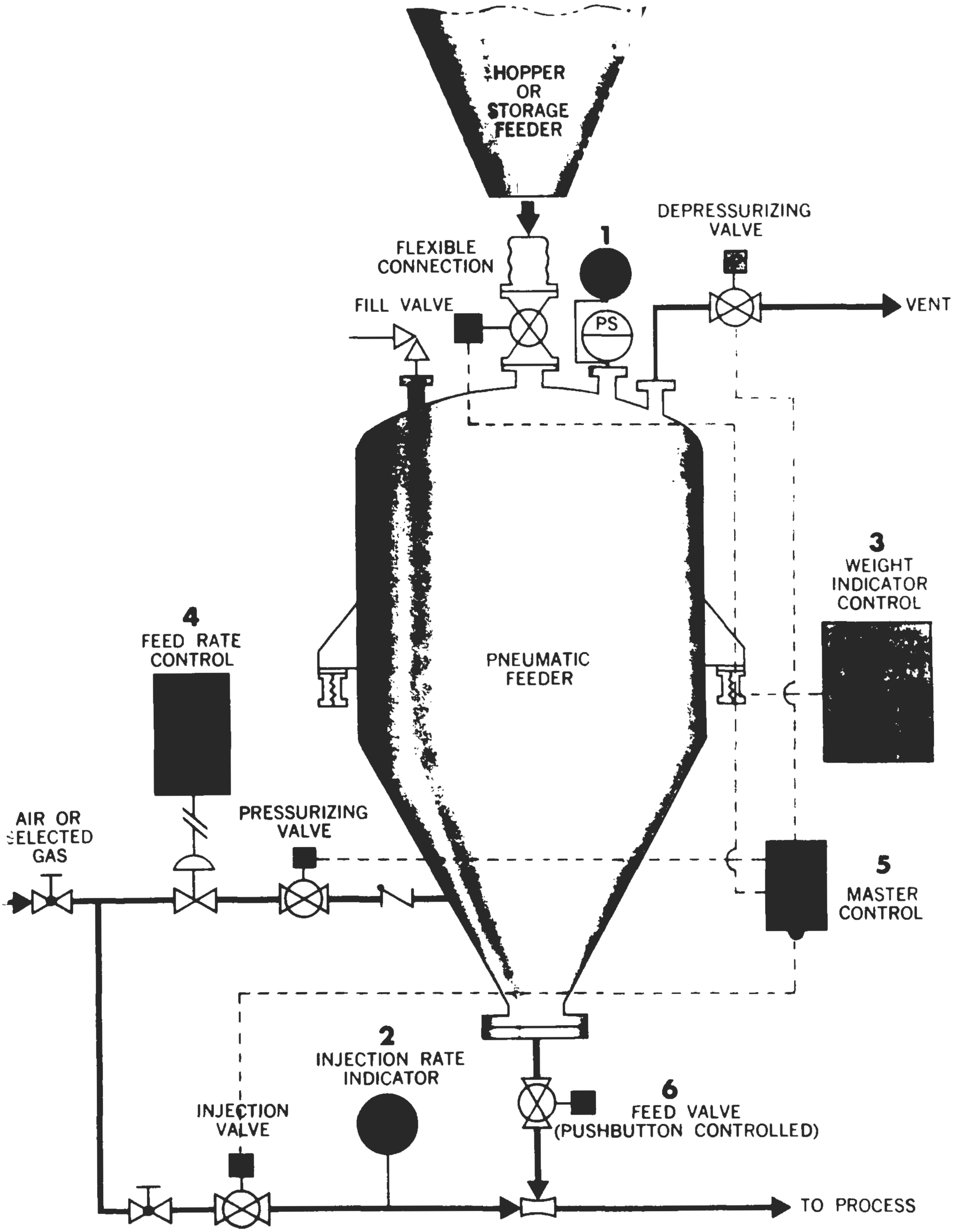

FIGURE A5-14 PNEUMATIC FEEDING OF COAL

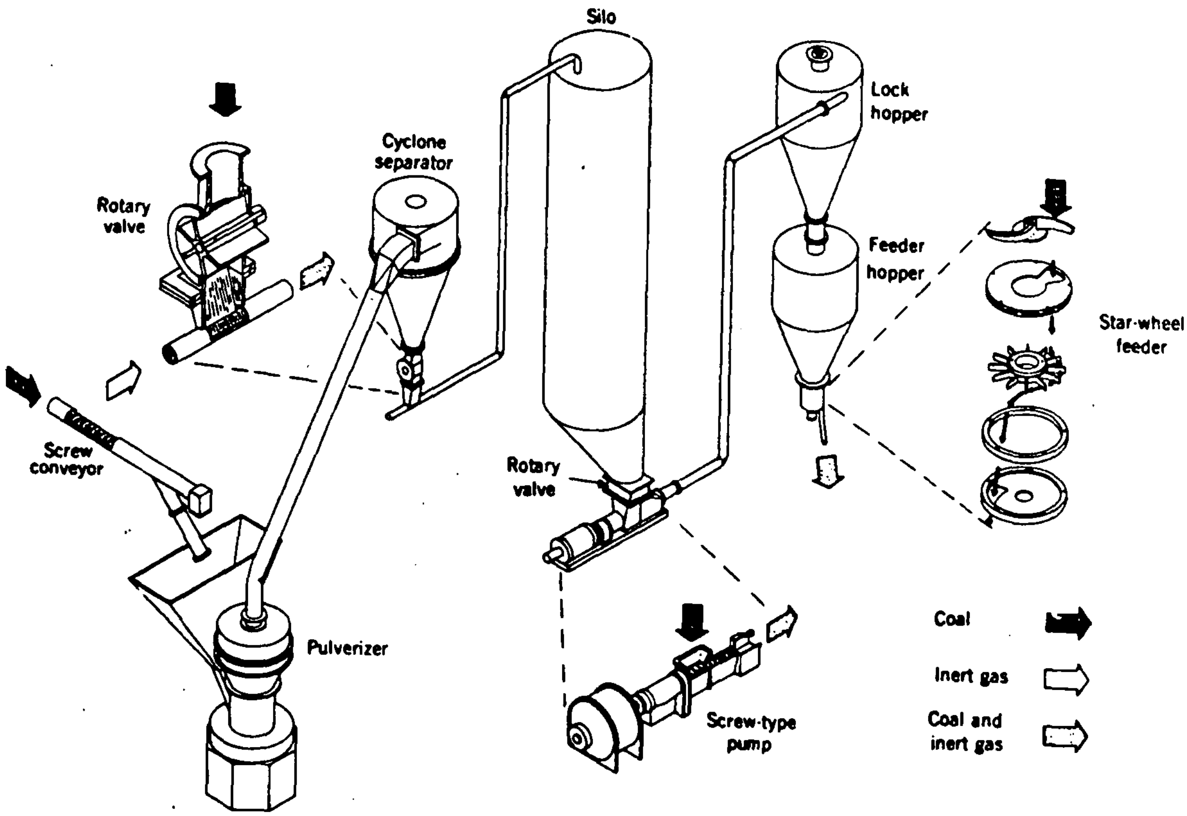

FIGURE A5-15 TYPICAL LOCK HOPPER FEED SYSTEM USED TO FEED COAL UNDER PRESSURE TO GAS TURBINES

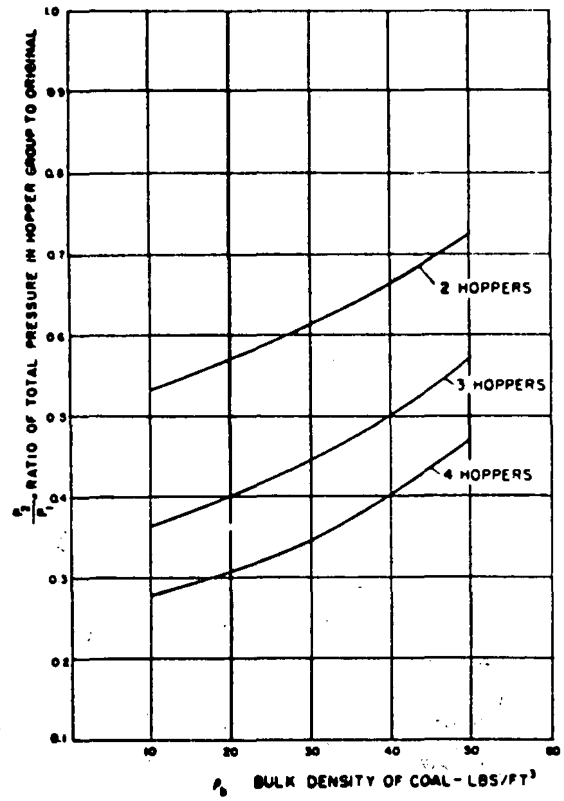

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FIGURE A5-16 REDUCTION IN OPERATING PRESSURE BY PLACING HOPPERS IN SERIES (14)

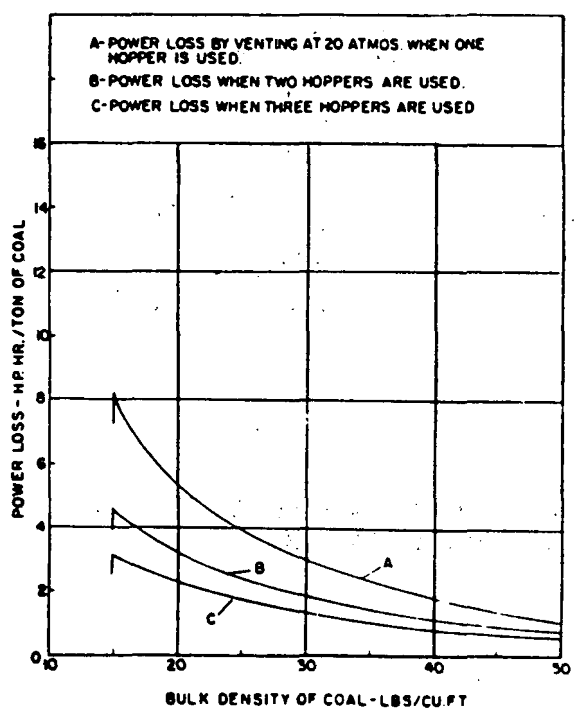

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FIGURE A5-17 REDUCTION IN POWER USING HOPPERS IN SERIES (14) of feed points. Savings are achieved which might not be as great as lock hoppers in series and at the expense of more complicated controls. Bituminous Coal Research uses two lock hoppers in parallel feeding, a pressurized piston common to both, and a feed vessel for pressurizing to 1000 atm.(27) Petrocarb (28) proposes cascading of pressurizing air when using a multiplicity of lock hopper systems to feed large numbers of feed points.

In any case the power savings must be realized at the expense of an additional capital outlay in lock hoppers and controls. Neither time nor scope of effort permitted optimizing the lock hopper system for the pressurized fluid bed boiler. Attempts were made to obtain a price for a simple single stage system. Lurgi, one of the prime vendors of lock hopper systems, would not quote a price or provide a conceptual design as they felt it would give away portions of their proprietary position.

.1.3 Lock hopper with moving wall A positive displacement or moving wall lock hopper is another attractive method for reducing power loss by venting.

In place of a pressurizing tank the principle of a moving wall uses a continuous positive displacement pressurizer which conveys into a stream of high pressure air or feeding vessel whatever quantity of pulverized or crushed coal it may receive from the feeder(14, 29). Dr. Donath proposed a system illustrated in Figure A5-18.

27) Communication with Dr. Glenn, Bituminous Coal Research.
28) Communication with Mr. Rienjtes of Petrocarb, New York City.
29) E. E. Donath, "Progress Achieved in Development of a Piston Coal Feeder", Proceedings of Gas Technology - Bureau of Mines Symposium, Morgantown, W. Va., October 19-20, 1965.

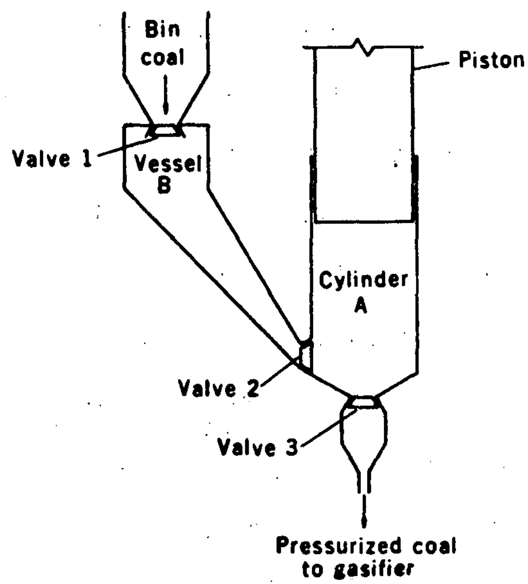


FIGURE A5-18 SCHEMATIC DIAGRAM OF DR. DONATH'S PRINCIPLE OF THE CONTINUOUS POSITIVE DISPLACEMENT PRESSURIZER EMPLOYED AT BCR (29)

Coal from the bin flows through valve 1 into the auxiliary feed vessel B. While in a fluidized state it is blown with inert gas through valve 2 into cylinder A while the piston moves upwards. The admission ports for the gas are not shown in Figure A5-18 When the piston reaches the top position valve 2 closes and cylinder A is pressurized with product gas at the feeding pressure. Then valve 3 opens and the coal, suspended in high-pressure gas is pushed by the descending piston into the gasifier.

Power requirements for the piston feeder can be calculated. Dr. Donath compares them with other feeders in Table A5-5 .

TABLE A5-5 POWER REQUIREMENTS FOR FEEDING COAL TO A 1000 PSI GASIFIER

Types of Feeder	Kwh per MM Btu in Gas	Density of Coal Change lb/ft^3
Piston	0.2	35
Slurry Pump	0.2	35
Lock Hopper Gas Compression	1.4	42

The reduction in power requirements would not be expected to be as great at lower pressures, say 10 atm. Calculations, however, have not been made to verify this. The pump has not been put to practice. There is no information on such characteristics as capacity, limitations, wear, or economics. It is definitely limited to single feed point applications.

Bituminous Coal Research in cooperation with Petrocarb has put together a system using two lock hoppers in parallel feeding to a rubber "boot" or rubberized piston which in turn, feeds to a single feeder vessel. The system essentially puts to practice the Donath system at the pilot plant level. Data on economics and maintenance are not yet known.

.1.4 Rotary pumps using the moving ball principle

Two similar types of solid-fuel pumps working on this method includes the rotary pumps designed by Yellott (30) in the U.S.A. and by the Incandescent Heat Company in England (31). These two pumps illustrated in Figure A5-19, are designed for applications where the work pressure need not exceed 10 atm. The speed of the rotary disc carrying coal to the high pressure zone can vary up to 60 rpm and coal flow rates can vary up to 2.5 tons/hr.

The practical problems center around:

1) Disc wear due to abrasion caused by entrained, fine particles of coal during its traverse between sealed pads or covers from atmospheric hoppers to high pressure air duct.

2) Perfect sealing between high-pressure zones, while maintaining at the same time efficient operation of the sealing medium upon surfaces in contact.

3) Avoiding venting of pressurizing air from the pockets before reaching the atmospheric hopper each cycle. This can be overcome by momentarily closing the pockets after the coal discharge by means of mechanical or hydraulic thrust if the pressure to be regained is not exceedingly high.

30) J. I. Yellott, "Gas Turbine Power Plant Solid-Fuel Feeding Machine", United States Patent No. 2652687, September 1955.

31) G. Waller, "Rotary and Ram Pumps for Solid Fuels", Journal of the Oil Engine and Gas Turbine, May 1953, pp. 28-30.

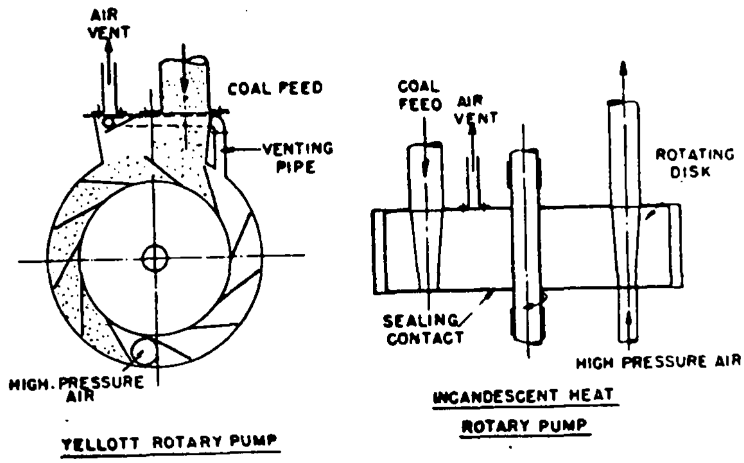


FIGURE A5-19 ROTARY PUMPS PROPOSED BY YELLOTT AND INCANDESCENT HEAT COMPANY (14)

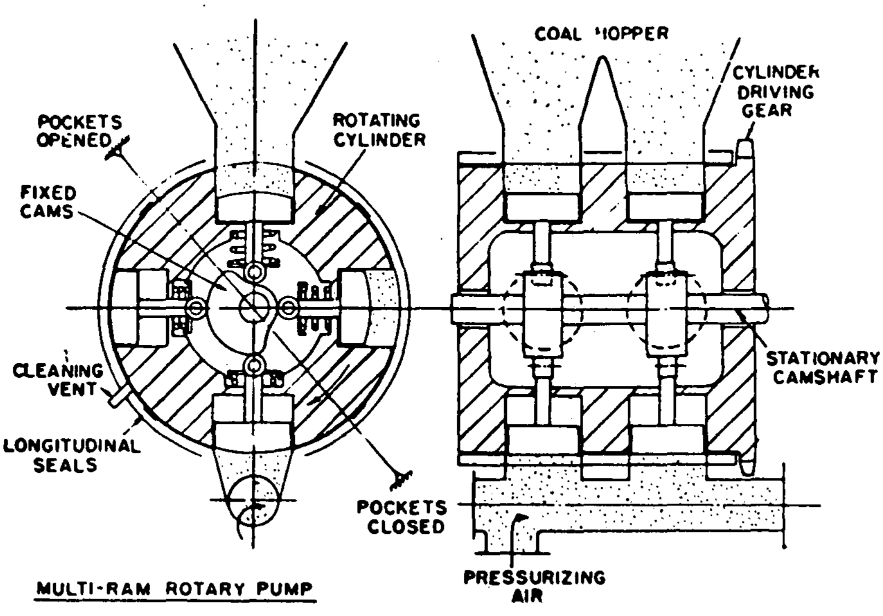


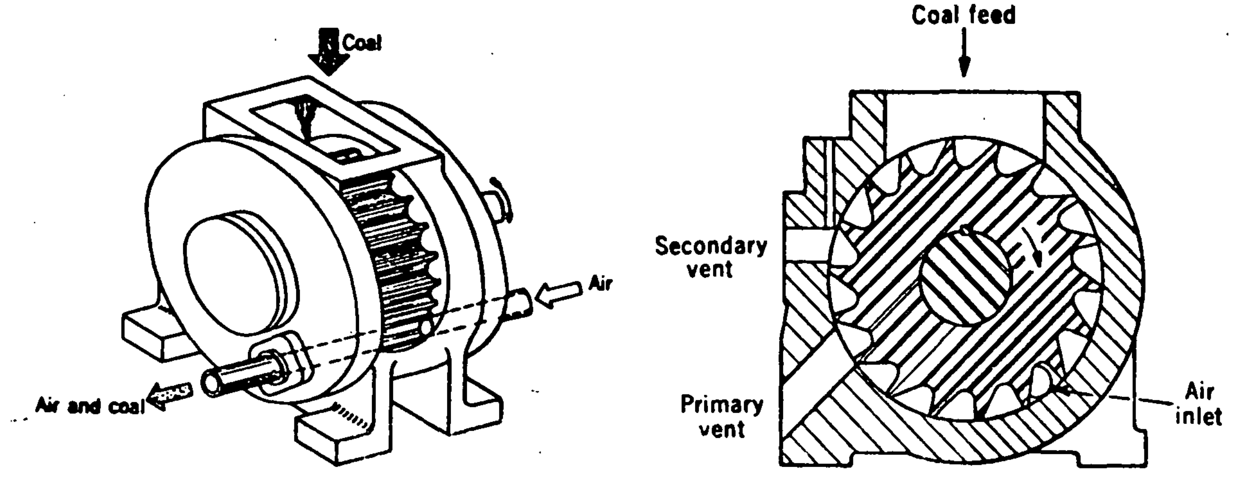
FIGURE A5-20 MULTI-RAM ROTARY PUMP (14)

As a modification of these pumps Dr. M. N. Aref proposed a multi-ram rotary pump illustrated in Figure A5-22. The pump essentially consists of a rotary cylinder containing two sets of four spring loaded pistons. The pistons are diametrically opposed and are actuated by a cam which remains fixed on a stationary shaft. On the hopper side the piston retracts forming a cavity allowing coal to flow into the vacated space. As the cylinder rotates, the pocket is sealed. At 180° from the start or hopper position, the cam forces the piston to compress at the time when the seal is broken forcing the coal into a pressurized gas stream. The pressurized coal is then transported away with the pressurized air. The cylinder continues to rotate and the pocket is once again sealed. Just prior to returning to the initial position on the hopper side after almost 360° of rotation has been completed the pocked is vented. This depressurizes the pocket and purges it at the same time. The pocket is ready to repeat the cycle.

The moving wall principle requires less power. However, it is subject to wear. Nothing is known about such characteristics as maintenance, capacity or overall economics. Care must be taken during the short period of compression that briquetting of the coal does not take place. The pump may be more readily adaptable to multi-feed point distribution by using several rotating cylinders on a common shaft. This would require further investigations.

.1.5 Koppers coal pump Koppers developed a "gear" type of coal pump that also operates on the moving wall principle. It is illustrated in Figures A5-21 and A5-22.

In the Koppers system, coal arriving at the injection plant



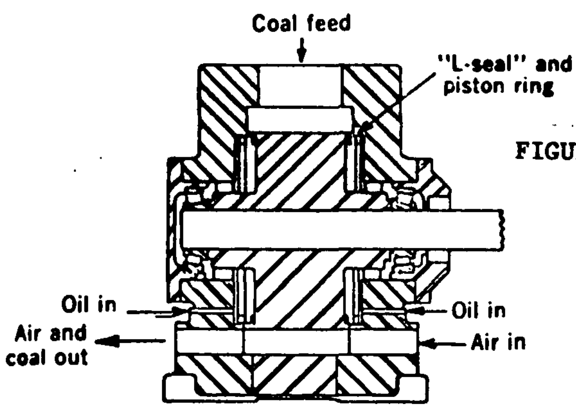
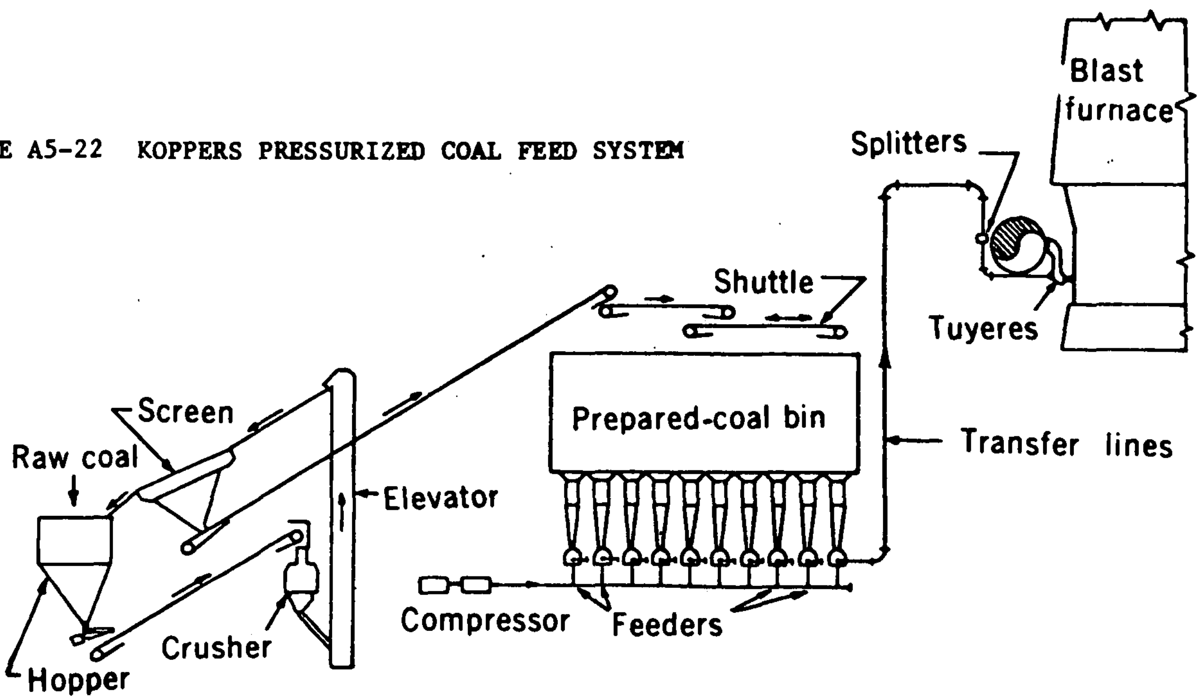


FIGURE A5-21 KOPPERS GEAR TYPE PRESSURIZED COAL PUMP

FIGURE A5-22 KOPPERS PRESSURIZED COAL FEED SYSTEM



from the mine or from storage is dumped into a truck hopper from which it is carried by belt to an impact crusher. The crushed coal is transferred by the bucket elevator from the crusher to a vibrating screen which scalps the oversize coal. The oversize material is returned to the hopper. The coal passing the screen is fed by belts to a reversible shuttle-conveyor which deposits the prepared coal evenly in the rectangular bin. The coal in the bin is maintained at atmospheric pressure.

Underneath the bin are ten BCR patented "Easy-Flo" bin discharge sections, each supplying a feeder, in order to ensure continuity of feed to feeders, particularly with wet coal. The bin is mounted on load cells so that the weight of coal in the bin can be determined periodically to obtain a check on the feed rate.

Compressed air is supplied to all the feeders through a common header from which parallel branch lines flow to the individual units. The coal-air mixture leaving the feeders flows through transfer lines to the splitters located near the bustle pipe. After the splitters the branch lines lead to alternate tuyeres. The main transfer lines vary in length from about 250 to 300 feet, including about 40 to 45 feet of vertical rise.

The coal injection system was designed to supply the furnace with as much as 600 tons per day of coal. The system handles the minus 3/16-inch or minus 1/8-inch coal with the fines included and requires no special preparation of the coal with respect to particle size distribution. The moisture content of the "as-received" coal usually averages between 3 and 4 percent and on occasions, coals with a moisture content up to 8 percent have been used. No problems have been encountered in feeding coals of the higher moisture contents from the atmospheric storage bin.

Instrumentation provides for complete automatic control with remote adjustment from a panel board located in a centralized control room. Air pressures and air flows are automatically regulated to follow changes in furnace pressure. Feeder speeds can be varied either simultaneously or individually. The control system provides for automatic shut-down and start-up of the feeders during cast periods or whenever checking of the furnace is required. Controls also provide for automatic shut-down of the injection system in the event of electrical power failure or a failure of auxiliary equipment.

If, for any reason, such as slag build-up at a tuyere, an individual line is plugged between the splitter and the tuyere, the respective feeder is shut down automatically; no plugging of the main transfer line occurs. This is an inherent feature of the system wherein one feeder supplies two tuyeres. Another advantage of this arrangement is that not more than two tuyeres need be without coal at any one time or for more than a few minutes. The piping arrangement at the lances is such that, if the lance-piping plugs, unplugging may be accomplished in a matter of minutes.

Koppers claims no major difficulties have been encountered with the equipment or the operation of the coal injection system.

Start-up operations were completely free of difficulties customarily associated with initial start-ups. Performance continues to be excellent.

Koppers claims the following advantages.

1. "No special equipment is required, such as pressurized bins, lock-hoppers, etc. Materials are delivered from a storage bin at atmospheric pressure to the blast furnace or other vessels operating at high pressures by means of the high-pressure feeder.

2. Virtually all types of solid carbon, such as bituminous or anthracite coals, lignites or chars, can be used.

3. No special preparation of the coals with respect to size consist is required. Simple crushing to minus 3/16-inch or minus 1/8-inch is all that is necessary. Coarser or finer sizes may be handled if desired.

4. No drying of the coals is required. Coals with moisture contents up to 9 percent have been used in the pilot plant.

5. The system is flexible. Large turn-down ratios can be built in without sacrificing accuracy of feed rate.

6. The system is completely automatic. Intermittent shut-down and start-up during cast periods or furnace checking are controlled automatically.

7. The system is adaptable to injecting other types of solid materials, into vessels operating at atmospheric or high pressures." (32)

32) ______, "Operating Characteristics of Koppers High Pressure Solids Injection System", Brochure, Koppers Company, Inc., Engineers and Construction Division, Pittsburgh 19, Pennsylvania.

The Koppers system is not without liabilities. Koppers is reluctant to design for pressures in excess of 100 psi and they are reluctant to handle stone. The pump is subject to wear and it does not easily handle the multiple feed point problem. Koppers would not quote on the system as specified.

.1.6 Screw feed using moving wall principle The Fuller-Kinyon Pump is not a rotary pump. However, it does use the moving wall principle in the form of a screw feeder which forces the coal into a pressurized chamber. The pump is illustrated in Figure A5-23.

Robert F. Loomis (33) reports that the Fuller-Kinyon system by virtue of its high material-to-air ratio makes safe handling of pulverized coal a reality. The total system air is only 0.0014 percent of the air necessary for combustion.

Basically the pump consists of a high-speed screw with a gradully reducing pitch section at the delivery end. The material being conveyed is advanced by the screw from the hopper section into a short barrel section where it is compressed to form a seal against blow back. The material is then discharged into a mixing chamber where compressed air is introduced through a series of nozzles or jets. The air mixes with and fluidizes the material and conveys it as a relatively dense mixture through the conveying line. Air requirements are relatively low as compared with those of most other pressure systems. Pick-up velocity is usually about 1,600 to 2000 ft/min with material-to-air ratios of 40 or more to 1 on a weight basis. Maximum air pressures are usually limited

33) Robert F. Loomis, "Pipeline Transport of Dry Solids", Proceedings, Institute of Gas Technology - Bureau of Mines Symposium, Morgantown, W. Va., October 19-20, 1965.

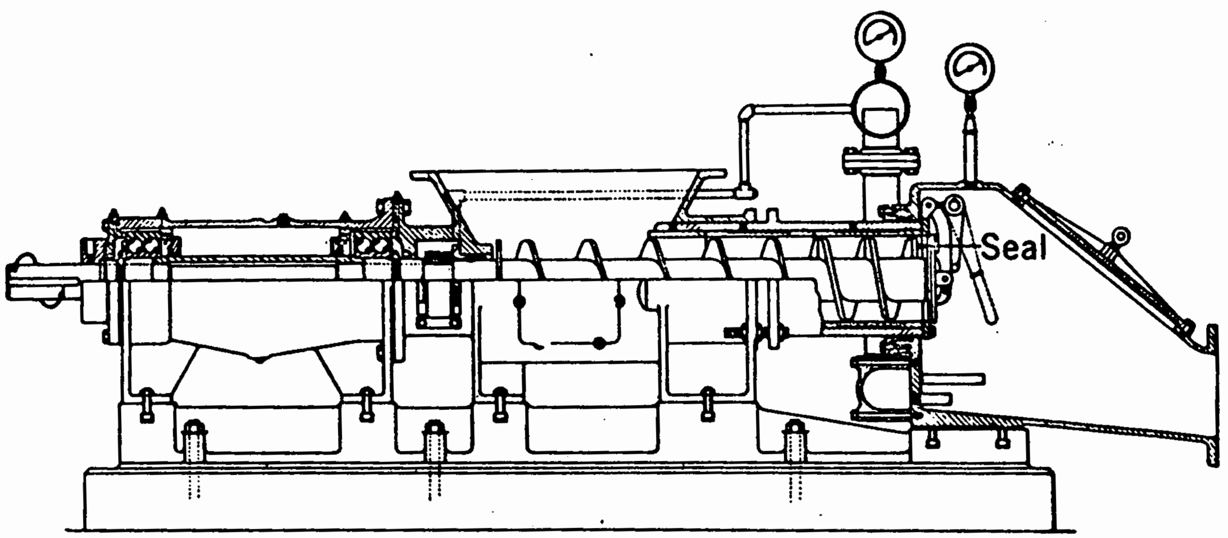


FIGURE A5-23 FULLER-KINYON SCREW FEED PUMP (33) to 35 to 40 psig keeping within the limits of a single stage rotary compressor. The pump is built in a wide range of sizes with capacities from 100 to 5000 Cfh.

Distribution and compression ratios are the two limiting factors for this system.

.1.7 Positive displacement ram pump This type of pump has proved satisfactory in operation to pressurize solid fuel at higher pressures than 10 atm. Figure A5-24 depicts a diagrammatic arrangement of a ram pump and its cycle. It operates by drawing air-laden solid-fuel dust into a cylinder through a non-return valve by means of the suction strokes of a reciprocating ram, which is connected to a conventional crank chain assembly. The ram is usually hollow and cooled during operation. Situated at a mid-point in the cylinder wall is a cam operated, non-return valve which admits high pressure air to the cylinder. At the start of the compression strokes, the tappet valve opens to admit air having a pressure at least equal to that of the system being supplied with fuel. Throughout most of the compression stroke the contents of the cylinder are discharged into the receiving system through a non-return outlet valve.

Unfortunately the main drawback of a ram pump is that it suffers from a limited low filling density of the solid fuel in order to avoid briquetting. Careful control of initial or filling density must be assured, particularly at higher pressures to avoid damage to the pump. This requires using a sensitive quantity governer for the solid fuel charge. The relationship

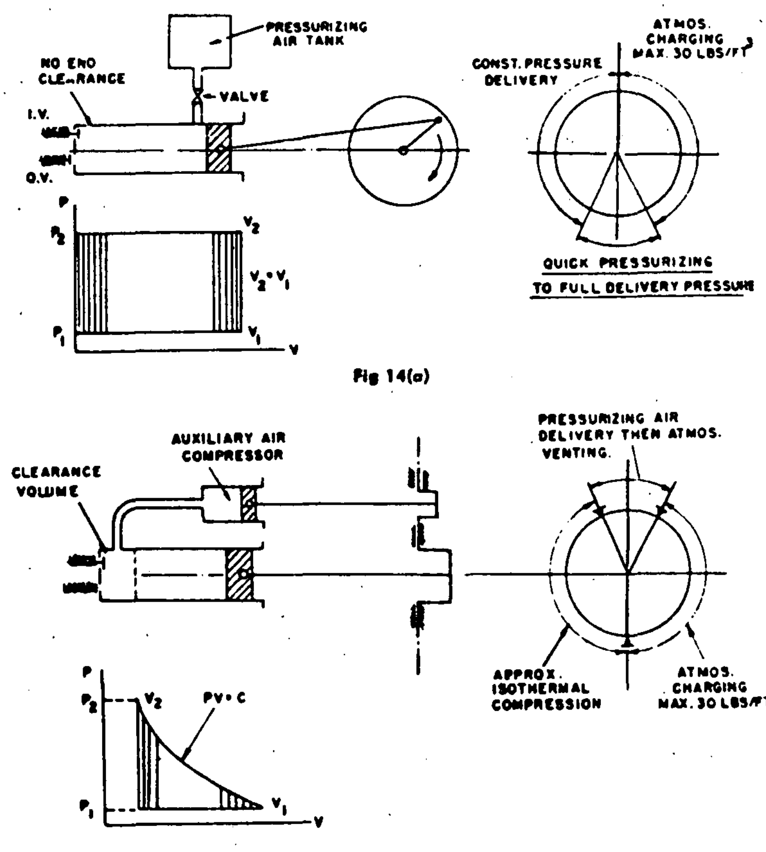


FIGURE A5-24 DIAGRAMMATIC ARRANGEMENT OF A RAM PUMP AND ITS CYCLE (14) between initial and final solid-air mixtures versus the compression ratio is illustrated in Figure A5-25. It has been found that briquetting of the solid fuel inside the cylinder may occur at 20 atm if the final mixture density exceed 30 lb/ft^3 which corresponds to about 2.5 lb/ft^3 initial density as illustrated in Figure A5-25. Consequently, a very low solid-fuel output per unit volume of pumps may be expected which is greatly reduced by the rise in temperature. This may be compromised by increasing the pump speed or number of cylinders in the cycle.

The power requirements for the positive displacement ram are about the same as a lock hopper with a moving wall and a final bulk density of about 30 lb/ft^3. This is illustrated in Figure A5-26 and represents the maximum practical limit for 20 atm. before briquetting. In some cases additional cylinder volume clearance is included to avoid damages due to briquetting. Scavaging air is used at the end of compression to vent the cylinder to atmosphere. In this case the power requirements must increase over a simple lock hopper system.

.1.8 Solid Extrusion H. Koppers (34) has shown that a coal plug 20 inches long can be extruded by means of a 2 in.-diameter plunger coaxially located in a 2.5 inch diameter tube at a rate of 120 strokes per minute and are extension pressure of 700 atm. and satisfactorily sealed against 30 atm. gas pressure. The plug can be repulverized very easily in a simple air-jet micronizer. This system is illustrated in Figure A5-27.

Substantial briquetting occurs, however, when the greater part of air in the voids is removed by the extension pressure.

34) H. Koppers, "Improvement in or Relating to the Introduction and Removal of Granular Solid Materials into or from Closed Chambers at Increased Pressures", GmbH Co., Germany, British Patent No. 699747, Appl. Date, October 1950.

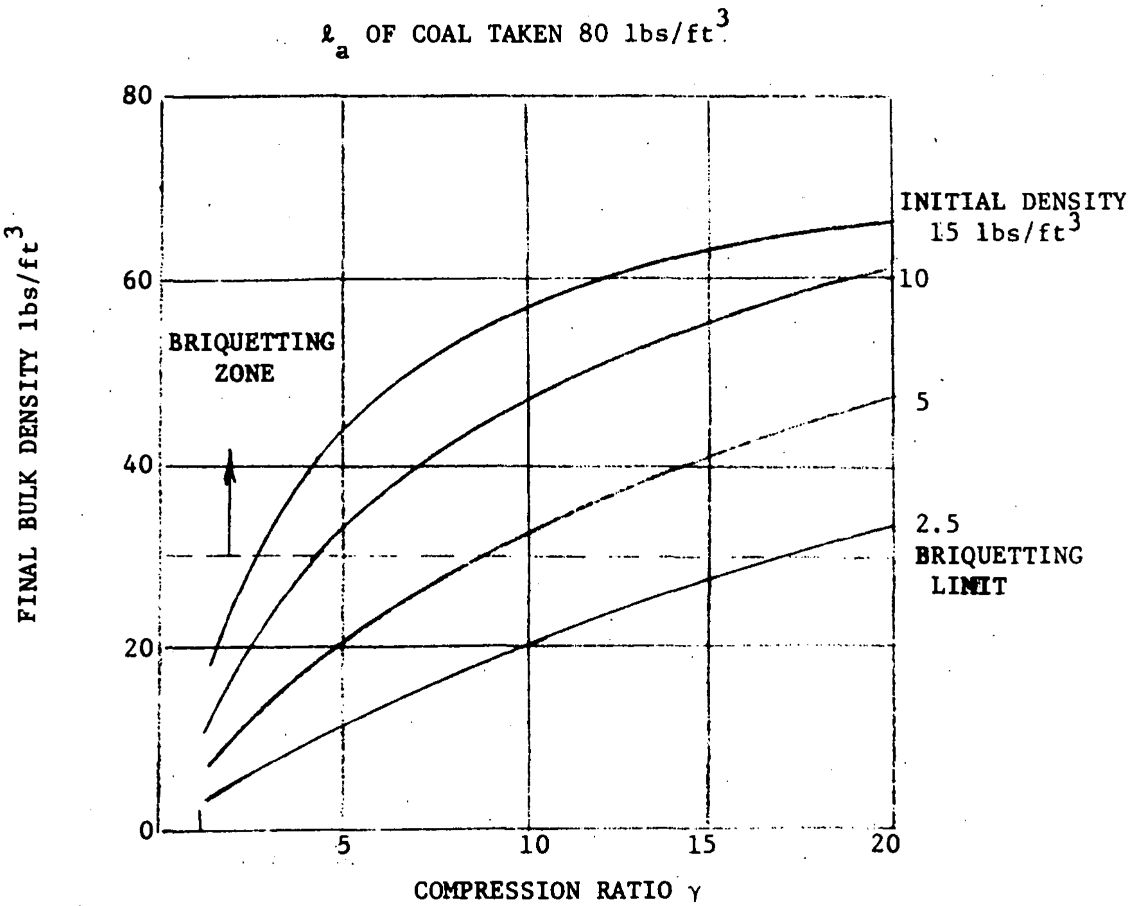


FIGURE A5-25 RELATIONSHIP BETWEEN INITIAL AND FINAL AIR MIXTURES VERSUS COMPRESSION RATIO AND BRIQUETTING LIMIT

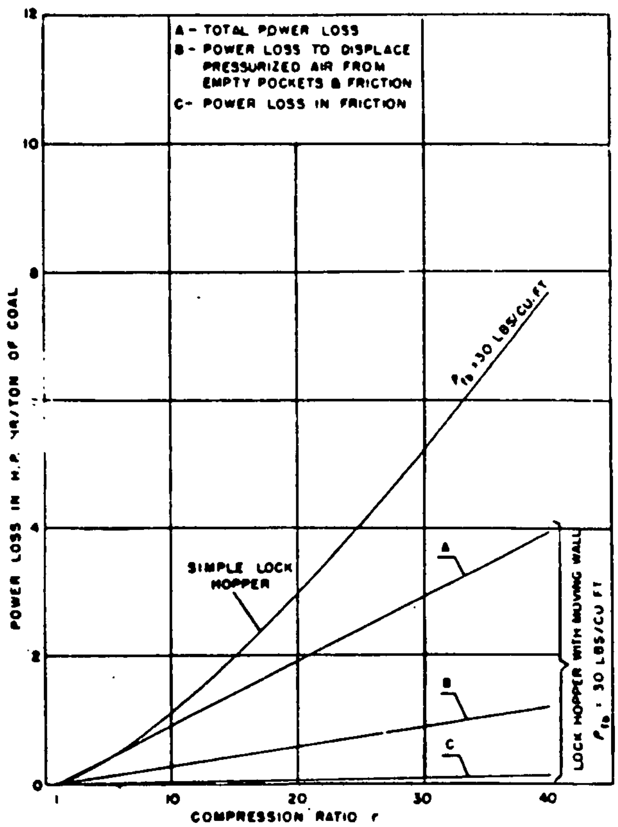
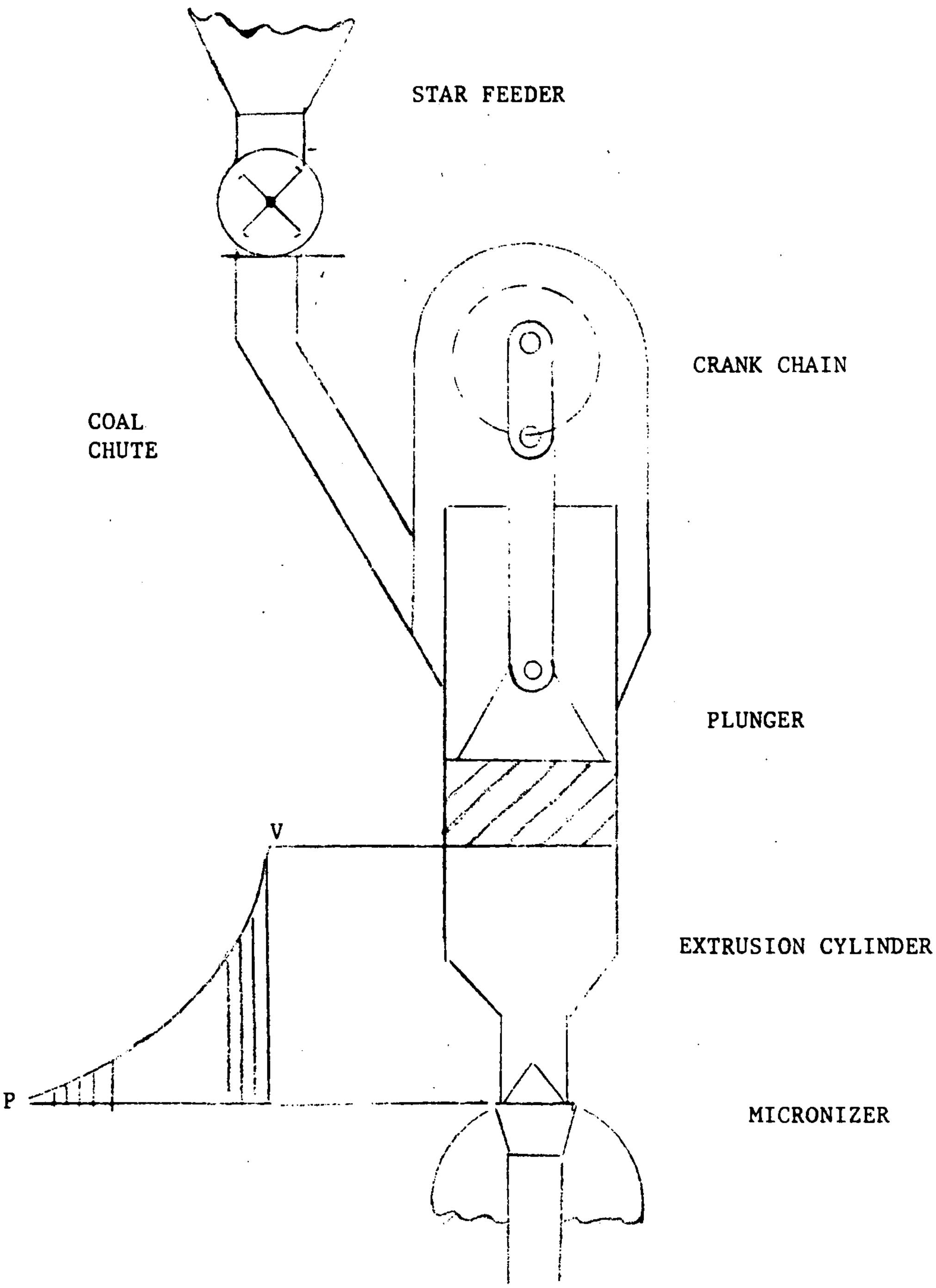


FIGURE A5-26 POWER REQUIREMENTS FOR SIMPLE LOCK HOPPER AND LOCK HOPPER WITH MOVING WALL

FIGURE A5-27 H. KOPPERS COAL SOLID EXTRUSION



This air escapes past the cylinder wall when the plug advances therein under the action of the plunger. A small portion of air in the voids remains in the form of more or less small compressed bubbles and this air, together with any pressurized air from the high pressure chamber which may have penetrated into the plug, tends to expand during the return stroke of the plunger. This tends to press coal particles against the cylinder wall thus increasing the sealing pressure of the plug, particularly in that constricted passage nearest the high pressure chamber.

Sufficient compression by the plunger is therefore required to:

1) Maintain high friction-bearing pressure between coal plug and cylinder wall in order to hold it in place against the pressurized air from below particularly on the return stroke of the plunger.

2) Achieve satisfactory sealing pressure of plug, depending on type of coal and its fineness.

It is possible to reduce the friction pressure which contributes to excessive wear of the cylinder, if the diameter of the plug is decreased to a practical minimum while the plunger stroke and speed are correspondingly increased. Although this procedure undoubtedly reduces the plunger loading and may increase the sealing efficiency of the plug, yet it has to be determined experimentally whether or not it significantly reduces wear.

Minimum power loss is substantially higher than other systems proposed at the lower compression ratios. Solid extrusion is more adaptible to the very high pressure levels. It is not certain that it has been reduced to practice at this point.

.1.9 Slurry pump Hydraulic pressurizing of coal-water slurries by lock hoppers with moving walls or ram pumps, reduces the power to almost that required for positive displacement of solids alone but this gain is swamped by the thermal effect of increased moisture content - after dewatering and by the serious complication of pressure drying even if the residual moisture can be used effectively in the system. See Figure A5-28 .

Usually coal-water slurry contains from 37 to 45 percent water by weight, depending on size of the coal particles and type of coal.

The plunger may be considered saturated with water when the bulk density of its mixture approaches the absolute density of coal, which often occurs when the water content reaches approximately 40 percent by weight. Oversaturation, however, takes place when the water content exceeds that limit and consequently the water, unless prevented, starts to drain out of the slurry mixture.

Power requirements for pressurizing coal slurries with various percent water are shown in Figure A5-29.

.3.2 Continuous action group

.2.1 Position displacement seal pump In this type of pump, positive displacement can be approximated and leakage reduced by meshing several gears together, three to four gears being sufficient. This is decided according to whether or not a maximum sealing distance is obtained due to a central zone of intermediate

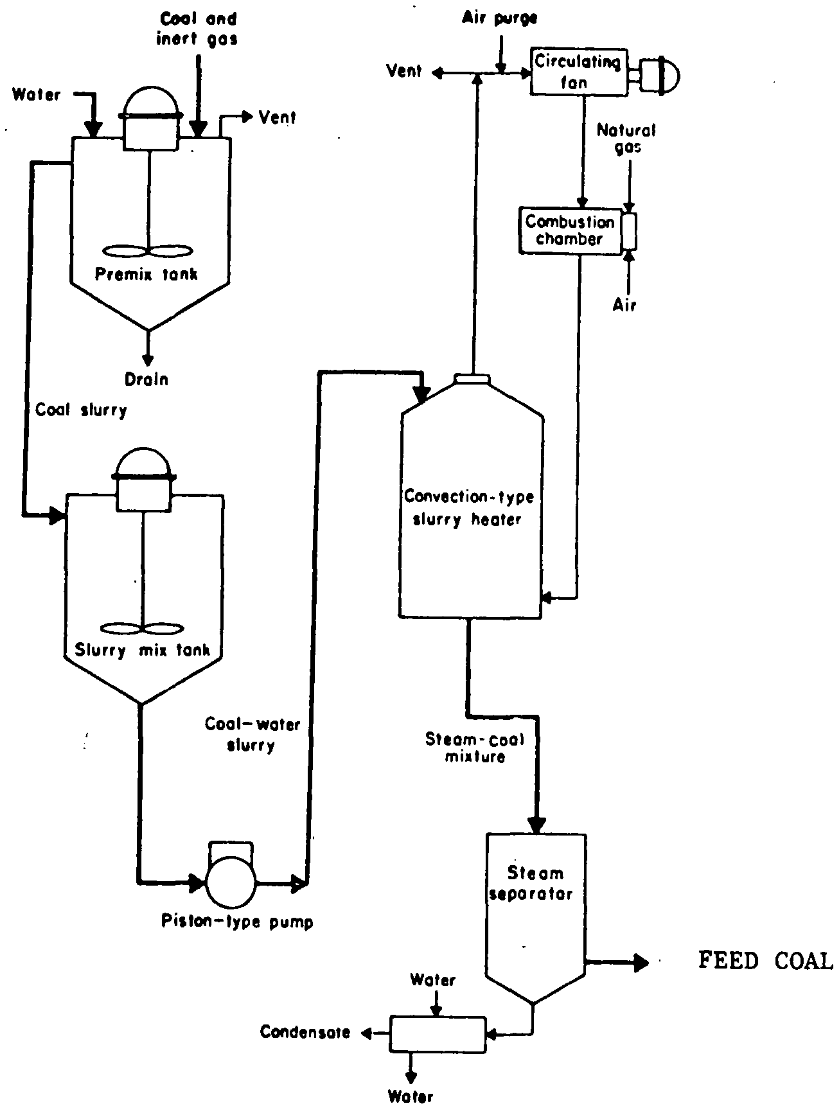


FIGURE A5-28 SCHEMATIC FLOW DIAGRAM OF A COAL SLURRY FEED SYSTEM (35)

(35) L. F. Willmott, K. D. Plants, W. R. Huff and J. H. Holden, "Gasification of Bituminous Coal with Oxygen in a Pilot Plant Equipped for Slurry Feeding", U. S. Bureau of Mines RI 6117, 1962.

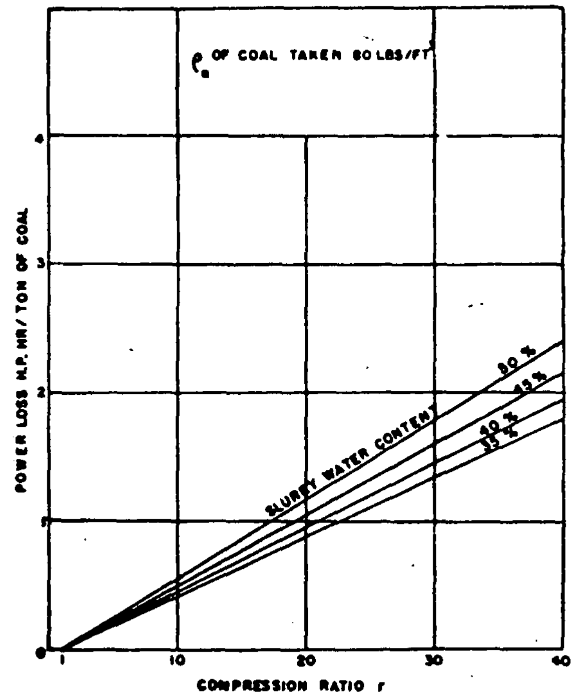


FIGURE A5-29 POWER REQUIREMENTS FOR PRESSURIZING COAL SLURRIES (14)

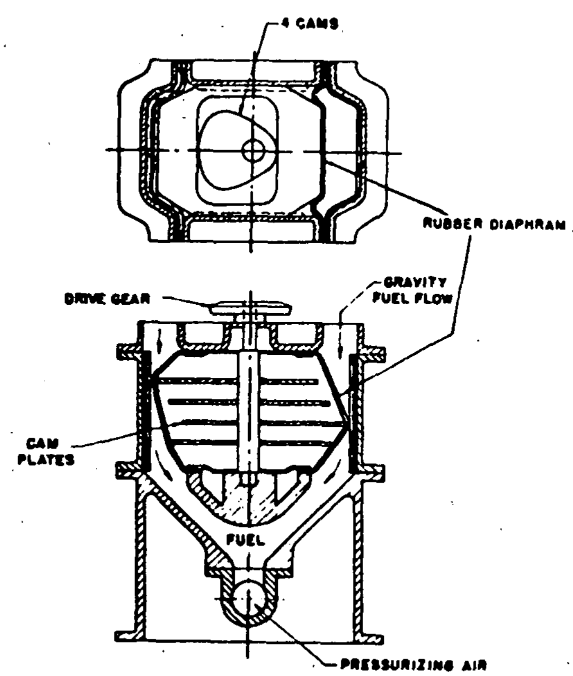


FIGURE A5-30 PERISTALTIC OR RUBBER PUMP (14) pressure. Undoubtedly this is a practical variant of the Yellott design to eliminate venting but in this case, it is probably the wear of the rotary disc seals which limits the speed and capacity rather than rate of filling and discharging. (14)

.2.2 <u>Multistage centrifugal compression with air</u> A small rubber-lined multistage shrouded centrifugal compressor offers another possible method for pressurizing finely pulverized materials with an intake density below 2.3 lb/ft^3 for 20 atm. delivery. This, however, is the limit over which briquetting of the solid might occur and cause damage of the impeller vanes. This pump has the advantage of keeping the solids fully entrained and their cooling effect at pressurizing will contribute something to efficiency. Nevertheless, excessive wear of the casing is liable to occur, particularly at higher impeller speeds and solids concentrations. Obviously this limits the practicality of the pump. (14)

.2.3 <u>Peristaltic or rubber pump</u> A peristaltic pump based on the principle of squeezing rubber ducts by a set of cams mounted on the same shaft provides a promising method for pressurizing solid fuels in the low pressure range up to 6 or 7 atm. In this case, wear, leakage, and air venting are eliminated while the pump offers a steady continuous flow path for the coal. Even so, it has the disadvantage of a pressure limitation with the unsupported side of the rubber duct. This is illustrated in Figure A5-30 where it has been established that collapsing the rubber at intermediate points along its length is more feasible than by a longitudinal roller system. (14)

.2.4 Jet pumping (35) The jet pump fluidizing feeder was developed to produce a uniform solids delivery rate when feeding against a constant pressure. It produces the desired uniformity mainly because it has no moving parts to suffer wear from abrasion. A jet pump employing a high velocity, inert motivating gas to entrain the particles of solids and propel them from atmospheric pressure to a higher pressure should also give a steady feed rate.

The jet pumping of solids functions on the theory that when a high pressure fluid is discharged from a nozzle of a jet the stream of high velocity fluid acts as a pump for moving a low-pressure surrounding fluid at the periphery of the nozzle. When the pumping process occurs there are two principle mechanisms taking place. (1) Acceleration of the particles of the surrounding fluid, which are relatively at rest, by the impact with the high-velocity escaping nozzle fluid, (2) entrainment of the surrounding fluid particles by viscous friction around the pumping of the nozzle of the jet.

Fluidized solids behave as a non-homogeneous fluid. When the non-homogeneous fluid surrounds the jet, the air molecules associated with the solid particles can be effectively pumped. However, the larger and heavier solid particles tend to lag thus reducing the effective work produced by the jet stream by consuming much of the energy. It is the high transfer of energy to these larger solid particles, serving to increase their velocity and moving them against pressure which reduces

35) T. E. Corrigan, M. J. Dean, and D. Denton, "Feeding Pneumatic Conveyors", British Chemical Engineering, March 1969, Vol. 14 No. 3.

the ability of the jet pump to move the non-homogeneous fluid effectively.

This method of pumping is not practical for conveying large quantities of fluidized solids into a higher pressure zone. The pressure range against which solids can be pumped is too small and this method would not be economical because of the large volume of high-pressure motivating gas requirements. It is believed that the design of the pump could be improved so as to increase the efficiency slightly but that this increase in efficiency would not produce a sufficiently high solids-to-air ratio to be an economical method of transporting solids.

2.5 Local fluidization In the local fluidization hopper fluidization takes place only in the vicinity of the outlet to the solids conveying tube. See Figure A5-31. The fluidizing gas for the most part is also used to convey the material from the hopper. When used in conjunction with a lock hopper system, the total concept offers the capability of pressurizing over a wide pressure range with a minimum of maintenance and power requirements. In addition it offers the flexibility of feeding to a large number of feed points from a single feed vessel using the simplest of feed devices. Except for the lock hopper valving there are no moving parts. At high pressure levels dense phase transport can be used to further minimize the total gas or air requirements. The system makes use of practical, available, equipment and has been put to practice by the Petrocarb Corporation.

A5.3.4 Fire and explosion in the pneumatic transport of coal A primary concern in the pneumatic transport of coal is the conditions which lead to fire and explosion. While this

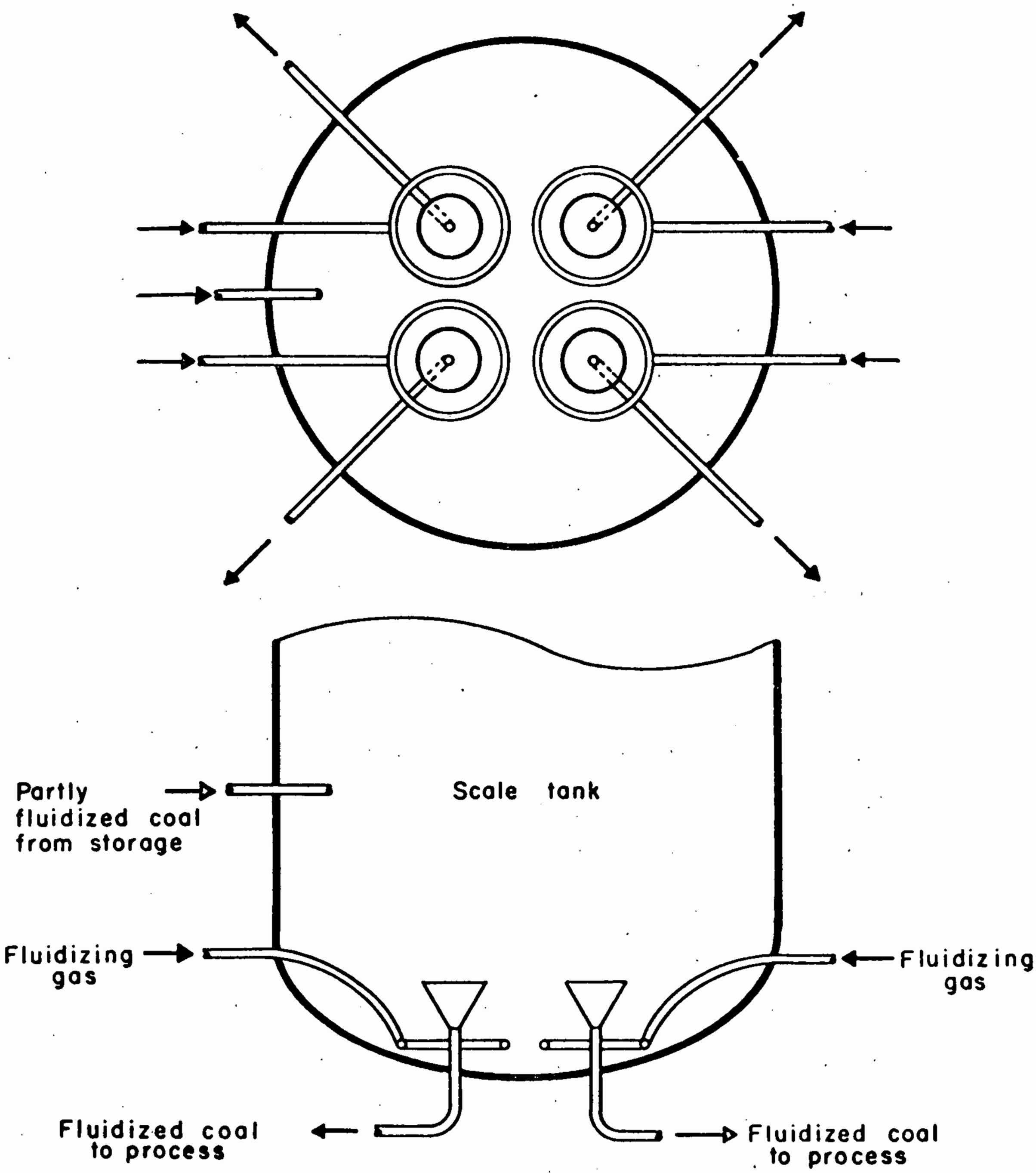


FIGURE A5-31 SCHEMATIC ILLUSTRATION OF LOCAL FLUIDIZED FEEDING OPERATION (37)

(37) W. R. Huff, J. H. Holden, L. F. Willmott, G. P. Strimbeck, "A Pilot-Scale Fluidized-Coal Feeder Utilizing Zone Fluidization", U. S. Bureau of Mines, RI 6488, 1964.

problem is less with the crushed coal which is to be used in the fluidized bed systems than with the more conventional pulverized coal, an evaluation of the problem was thought to be in order especially for the high pressure system.

Discussions were held with personnel at the Bureau of Mines at Bruceton and Morgantown and at Bituminous Coal Research to get the benefit of their experience in this area. The consensus among these groups was that crushed coal with a low fines content could be transported with air at atmospheric pressure with no explosion hazard and that there would probably be no problem at pressures up to 10 atmospheres. At substantially higher pressure an inert gas would be required.

The primary consideration in the prevention of explosions and fires in mixtures of coal and air is the ignition mechanism. As temperatures approach the autoignition level any coal which deposits in the transport duct is a potential source of ignition. The key to preventing fires is the prevention of saltation or deposition of coal particles in the duct.

The effect of pressure on ignition of coal deposits depends upon the rate of heating. In situations with slow heating rates, the ignition of coal is zero order with respect to the partial pressure of oxygen. The rate limiting factor is the thermal decomposition of the coal surface. With higher temperatures and heating rates which result in thermal cracking and the release of volatiles, partial pressure of oxygen is important. However, even under these conditions, residence time is the controlling factor. If the coal particles are kept in motion the deposition of fines is prevented and fires can be avoided even at elevated pressures. Therefore, the most important effect of pressure is in the area of two-phase flow.

Deposition can be prevented by the use of high transport velocities and proper design of bends. For atmospheric pressures, velocities of 60 ft/sec are recommended. Higher velocities should be used in high pressure systems to provide a margin of safety.

The general conditions which lead to fires and explosions in coal-air mixtures are as follows:

Particle size - <100 mesh

Air temperature ->600°F to 700°F (500°F is safe)

Concentration of fines - 50 mg/liter of the particles which are less than 100 mesh

Type of coal - high volatile, e.g., Pittsburgh seam coal and highly reactive, e.g. lignite

Pipe diameter - large diameter ducts have a low quenching effect (stay under 6 inches - the smaller the better)

Pipe temperature - insulated pipe reduces the heat losses from material deposited in the pipe and promotes fires.

When deposits of coal dust are formed anywhere in the duct, localized ignition occurs if the rate of heat generation exceeds the rate of heat dissipation. The system design should be made to promote the dissipation of heat from any deposits which might occur. Do not insulate the pipes at points where deposition is likely to occur.

Private communication with A. A. Orning, Bureau of Mines, Bruceton, Pennsylvania.

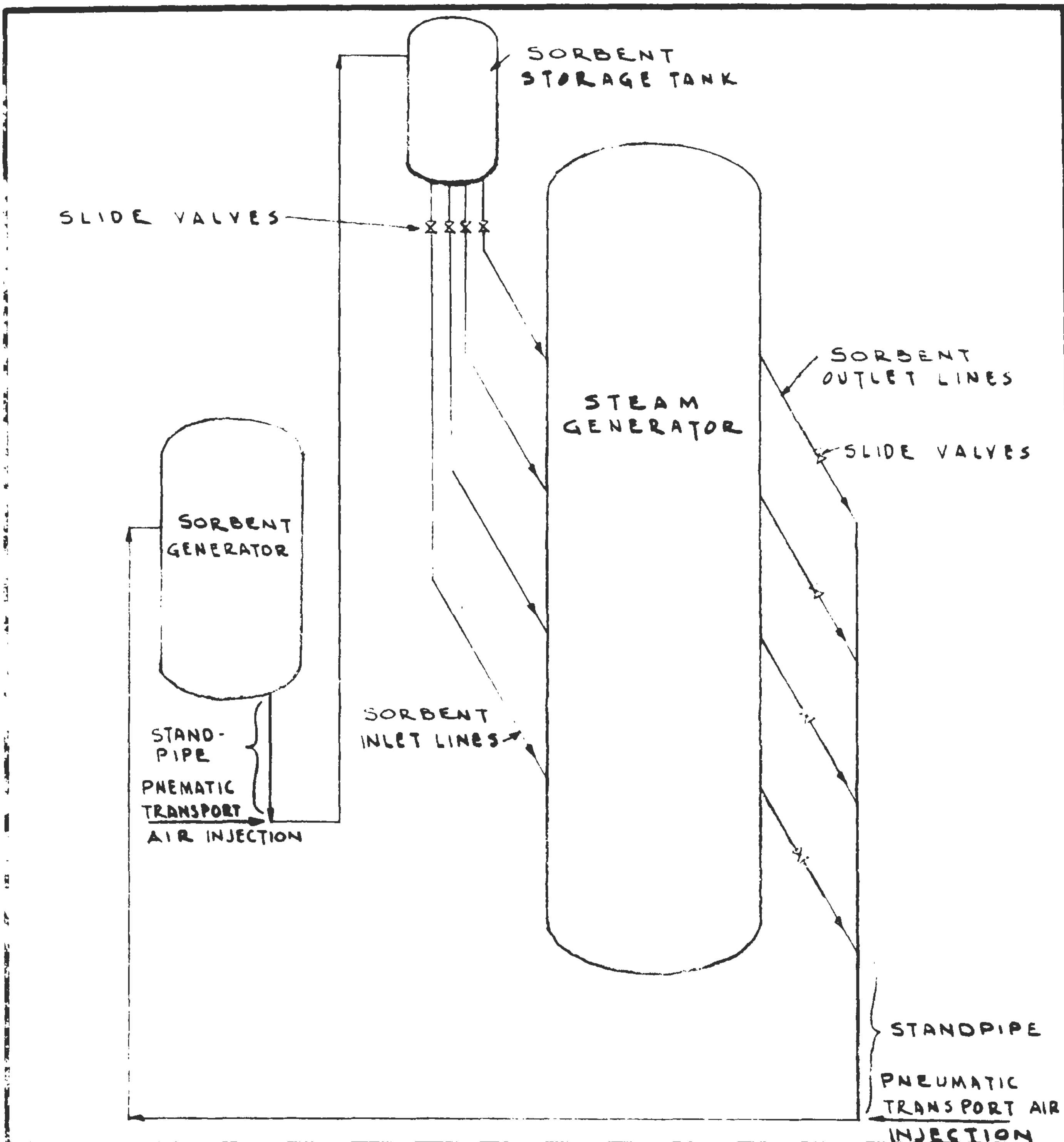
A5.3.5 Conclusions The problems confronting most solid pressurizers are (a) power loss due to venting of pressurizing air, (b) wear of surfaces in contact with solid, (c) effective and reliable sealing, (d) capital cost and capacity limitations of device and (e) pressure limitations. Practicality and state-of-the-art development are also pertinent to short term evaluation or immediate applicability of equipment. Local fluidization in combination with a simple lock hopper or staged lock hopper appears to be the only device that is presently marketable which minimizes all the problems confronting the solid pressurizer in the pressure range of 10 atm. It gives the best combination of pressurizing feeding and distribution characteristics of all the devices reviewed. It is on this basis that the Petrocarb system is recommended for the 10 atm. pressurized fluid bed operation.

A5.4 Sorbent Circulation System

A possible method of circulating the sorbent is shown schematically in Figure A5-32. This system relies on columns of sorbent to seal pressure differences and pneumatic transport to move the solids. A pipe from each of the beds penetrates the pressure vessel shell and travels downward either vertically or with a relatively steep slope. Each pipe has a slide valve located at a specific point in the line. Above the valve the pipe is designed for dense phase flow and the pressure head of this column of absorbent allows the pressure in the dilute phase transport line to be greater than the pressure in the beds.

Piping below the slide valves is designed for dilute phase transport. The pipes can be combined into a larger pipe if economics dictate this. Below the lowest bed, transport air is injected into the line, and the sorbent is transported pneumatically to the sorbent regenerator. The same principle of a standpipe and air injection system is used to transport sorbent from the regenerator to the sorbent storage tank. From this tank gravity flow regulated by slide valves is used to feed the sorbent back to the fluidized beds in the steam generator.

This circulation system uses a minimum of mechanical equipment and requires no pressurizing equipment such as lock hoppers. It does, however, place restrictions on the location of the sorbent regenerator. The elevations of the fluidized beds, sorbent regenerator and sorbent storage tank are determined by a pressure balance around the loop. THe height of the dense phase standpipes must be balanced with the pressure drop and gravity



SCHEMATIC OF SORBENT CIRCULATION SYSTEM

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L. B. CO., NO. 44-812

FIG. AS-32

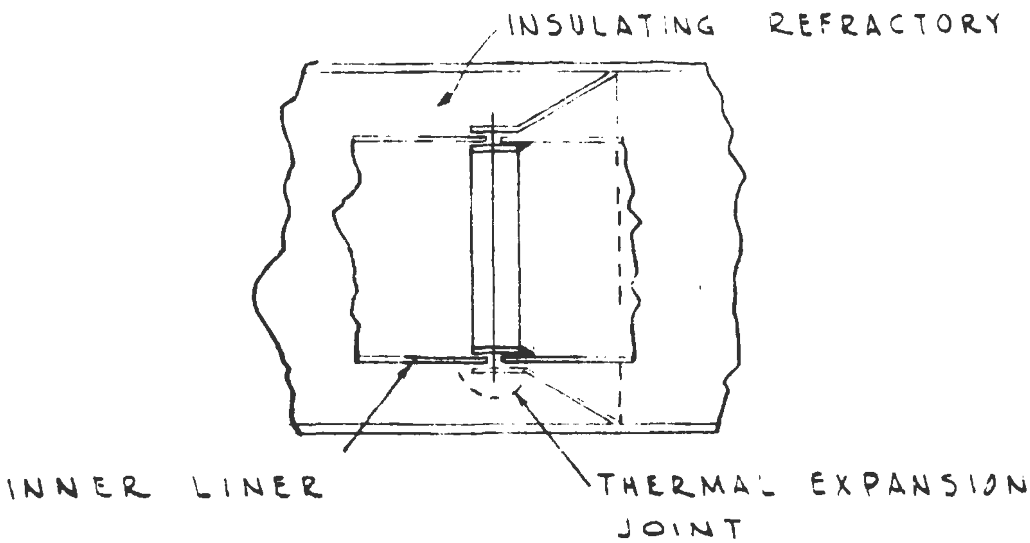
potential losses. This restricts the possible equipment locations and also limits the range of circulation rate that can be achieved. The high temperature slide valves are used to control the rate of flow, but the pressure drop across these valves is limited to about 3 psi. This could restrict the range of flow rates that can be achieved.

An alternate approach to the system described above would be a lock hopper system similar to the coal feeding system described in this report. This type of a system would allow more flexibility in the location of the regenerator. Also the relative pressure of the steam generator and sorbent regenerator could be varied considerably. Westinghouse has included this type of a system in the overall plant design.

A5.5 Gas Piping

The layout of the flue gas piping system up to the secondary cyclone inlet can be seen in Figure A4-5. The gas pipe connecting the steam generator and the first cyclone stage is a 74 in. O.D. pipe with an inner liner of a 3 in. thickness of hard formed refractory and a 4 in. thickness of insulation. In order to keep the cyclones as close to the steam generator as possible, both the cyclone pressure vessel and the steam generator are supported at the same elevation. This support arrangement allows the use of a straight gas pipe between the two vessels. The expansion joint in this line is only effected by the radial growth of the two vessels and the thermal expansion of the line itself. The gas outlet pipe from the carbon burn-up cell to the first stage cyclone pressure vessel is the same type of a pipe as the main gas outlet. This pipe, however, is 30 in. O.D.

Two possible designs were considered for the first stage cyclone pressure vessel and the gas lines from it to the second stage cyclones. In one design the first stage cyclone pressure vessel and the gas lines were lined with a stainless steel shroud. This type of a gas pipe design can be seen in Figure A5-33. The inner shroud is used as a safety precaution to prevent any refractory from breaking off and being carried to the gas turbine. The other design that was considered uses a hard refractory as the inner liner instead of stainless steel. A considerable cost savings could be realized by using this type of a gas pipe as can be seen in Tables 6.1 and 6.2, but a further investigation of the desirability of using this type of pipe is needed. Besides the economic considerations, customer acceptance from a safety point of



HOT GAS PIPING

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RD 704-122		

C. B. CO., NO. 44-512

FIG. A5-35

view must also be considered.

A5.6 Second Stage Particulate Removal

For the 300 MW plant each steam generator module requires a secondary particulate removal system capable of handling 62,400 CFM of flue gas with a temperature of 1600°F and a pressure of 145 psia. The particulate rate is 5,397 lb/hr. The particulate removal system selected uses two Aerodyne Development Corporation high pressure, Model 18,000, Type "S", collectors. These separators were designed for 97% collection efficiency by weight with effectively all particles larger than 5 microns removed from the stream. A fractional efficiency curve is shown in Figure A5-34. The detailed design of the separator can be seen in Figure A5-35.

The flue gas is cleaned by the centrifical effects of two rotating gas streams. The motion of the gas streams are shown pictorially in Figure A5-36. One gas stream enters at the bottom of the inner cylinder in an axial direction. This stream is given a rotational motion by guide vanes at the bottom of the inner cylinder. The second gas stream enters the inner cylinder tangentially near the top and spirals downward along the walls of the cylinder. There is a baffle plate near the bottom of the inner cylinder that fills most of the space between the inlet duct and the cylinder wall. There is, however, a 1-3/4 in. gap between the baffle plate and the cylinder wall. The downward flow of gas hits the baffle plate and is then reflected upward. The particles, however, fall through the gap and are collected in the chamber below.

The separator shown in Figure A5-35 uses dirty flue gas for both of the gas streams. Another method of operation would

FRACTIONAL EFFICIENCY OF AERODYNE

TYPE "S" COLLECTOR

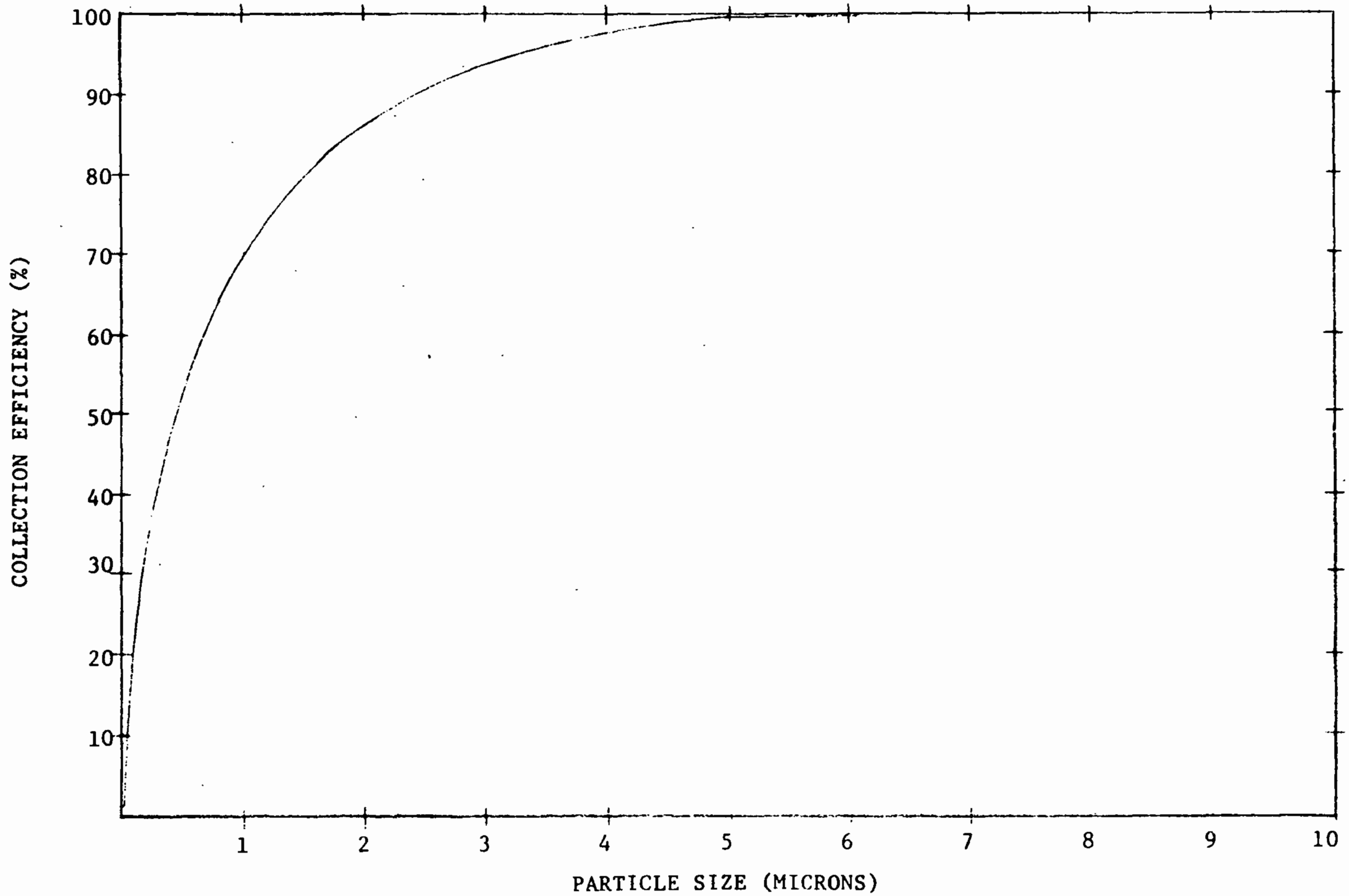

FIGURE A5-34

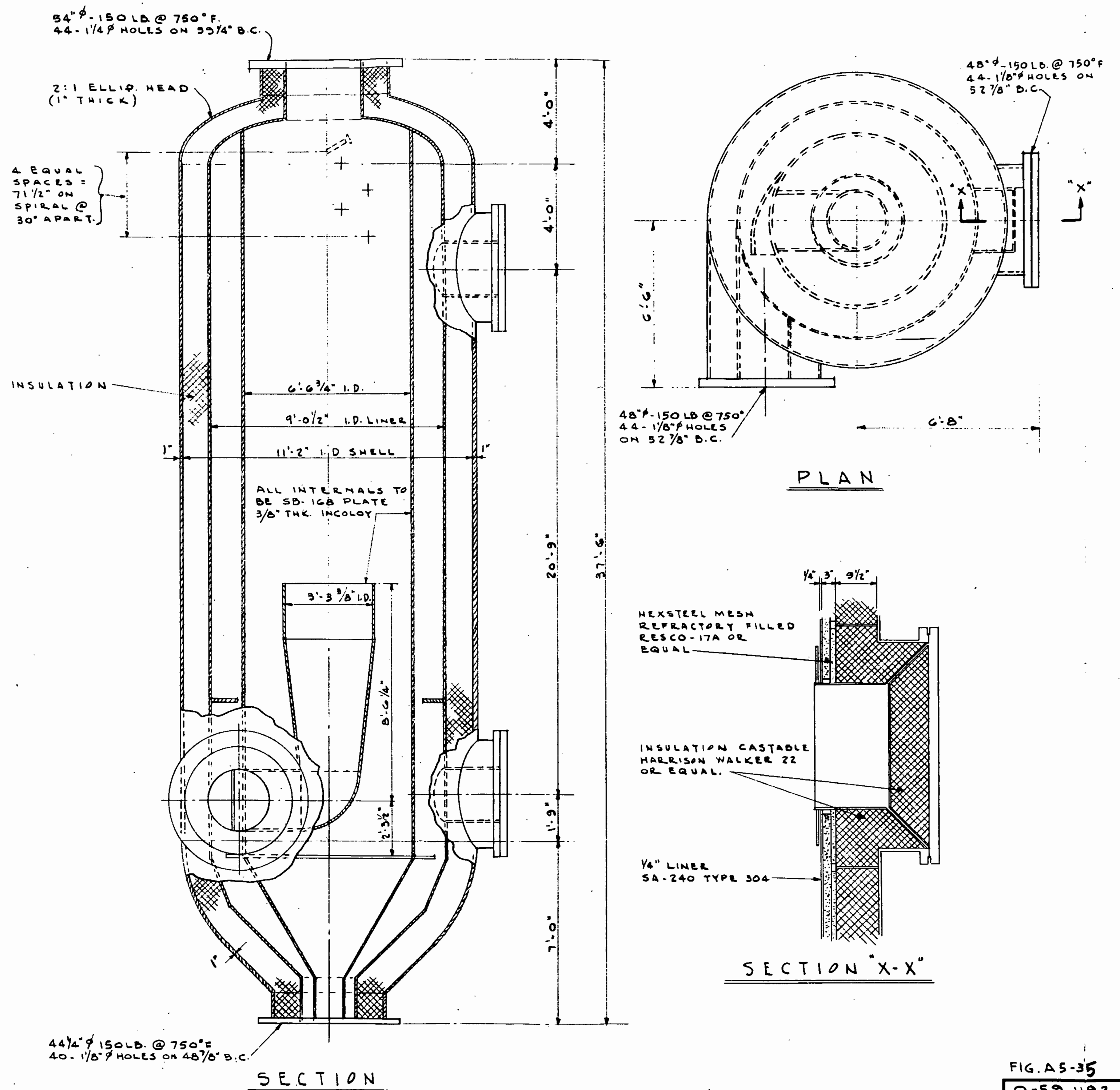

FIG. A5-35

0-59-1192

GENERAL NOTES

1. VESSEL LINING TO BE IN ACCORDANCE WITH F.W. STANDARD # 10A3
2. MAIN VESSEL TO BE IN ACCORDANCE WITH ASME CODE SECT VIII DIV. 1. DESIGN: 147 PSI - 650°F SA-285-C
3. INTERNALS TO BE DESIGNED FOR 1700°F.
4. THIS DESIGN IS A MODIFICATION OF DESIGN SHOWN ON AERODYNE DRAWING # SF-18000-420.

AERODYNE SEPARATOR IN PRESSURE VESSEL

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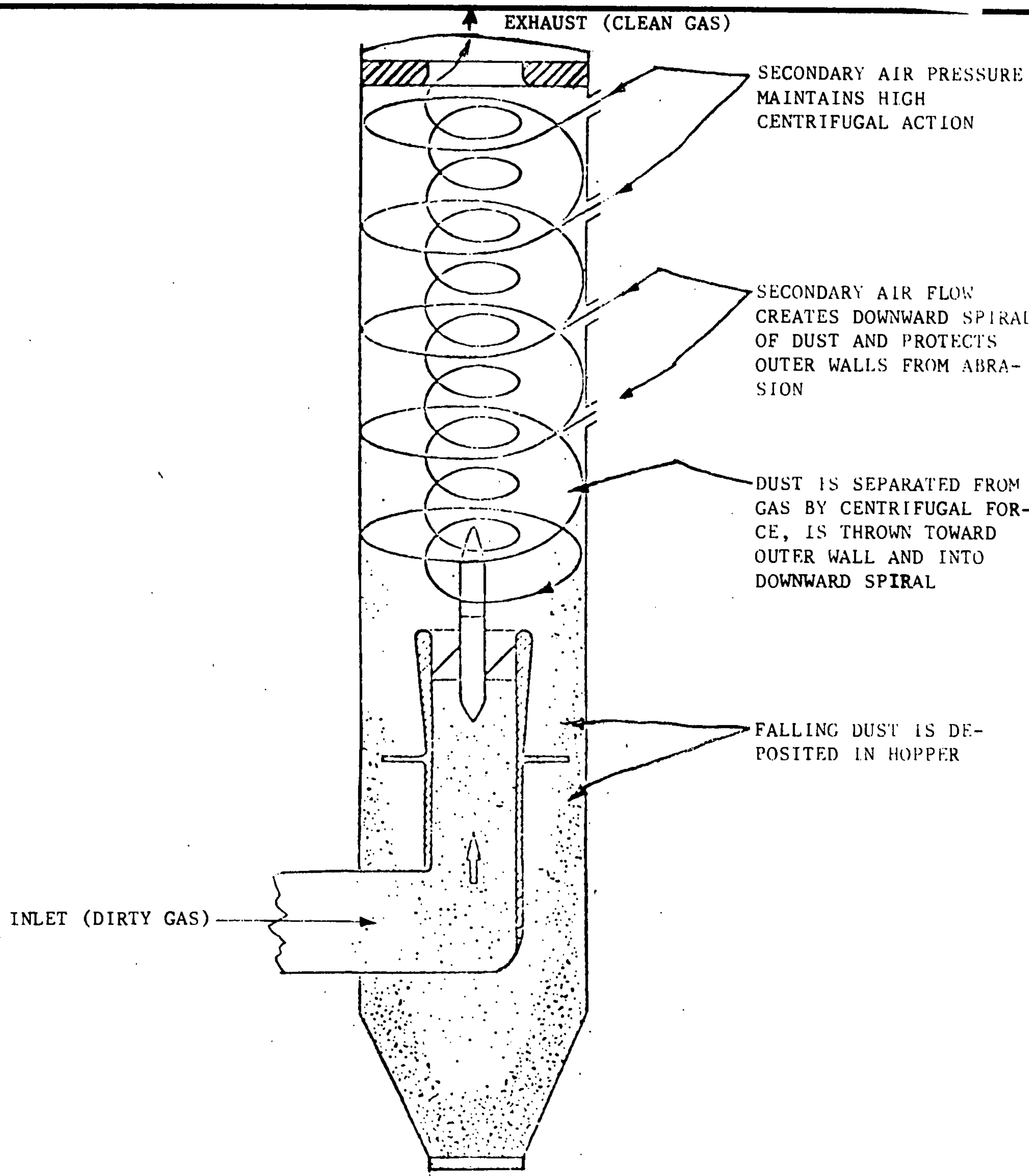
APPROVED BY:				

FORM 259-71C



OPERATION OF AERODYNE PARTICULATE SEPARATOR

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FIGURE A5-36

be to use clean gas for the second stream, but using dirty gas for both streams allows a greater throughput of dirty flue gas. The distribution of the flue gas into the two streams is achieved by the sizing of the inlet orifice for the inner flow stream and the sizing of the nozzles for the tangential flow stream. This system does not allow for adjustments during operation, but the absence of valves or dampers increases the reliability of the system.

APPENDIX I

SORBENT REGENERATION/SULFUR RECOVERY SYSTEMS DESIGN AND COST

ABSTRACT

Flow diagrams, material balances, equipment design, and cost estimates for the regeneration and sulfur recovery plants are presented. The regenerator process coupled to the pressurized boiler employs the two-step conversion of $CaSO_4$ to $CaCO_3$ while producing H_2S for sulfur recovery in a Claus plant. The regenerator process coupled to the atmospheric boiler employs the one step conversion of $CaSO_4$ to CaO while driving off SO_2 for subsequent sulfur recovery as sulfur or sulfuric acid.

1. High-Pressure Regenerator: Reduction-Steam CO_2 Oxidation Producing H_2S

Flow diagram and material balance information for this process is given in Figure I-1 and Table I-1. Figure I-2 shows a plant layout for this regeneration scheme. This information was presented in Volume II.

Table I-2 decribes the operation of the dolomite surge tank.

Cost estimates for the reducer vessel, the H_2S generator vessel, the gas producer vessel, and the particulate collector pressure vessel are based on the preliminary sketches shown in Figures I-3 to I-6.

Included in this section is a description of the solids feed system by Petrocarb; a description of the Claus plant by Ford, Bacon and Davis; a description of the CO_2 scrubbing system by Benfield Corporation; and a cost estimate for process vessels prepared by Westinghouse Heat Transfer Division.

2. Atmospheric Pressure Regenerator: Direct Reduction Producing SO_2

Flow diagram and material balance information for this process are given in Figure I-7. Figure I-8 shows a plant layout for this regeneration scheme. This information was presented in Volume II.

Wherever possible cost data from the pressurized regenerator design was scaled to estimate the cost of this process. Figure I-9 is a sketch of the regenerator vessel used to estimate its cost. Process vessel costs were scaled from the Westinghouse Heat Transfer Division estimate for a high-pressure regenerator. These costs are given in Table I-3. Sulfuric acid plant cost information from Monsanto and Parsons is included in this section as is a summary of sulfur recovery cost information obtained from the Allied reports.

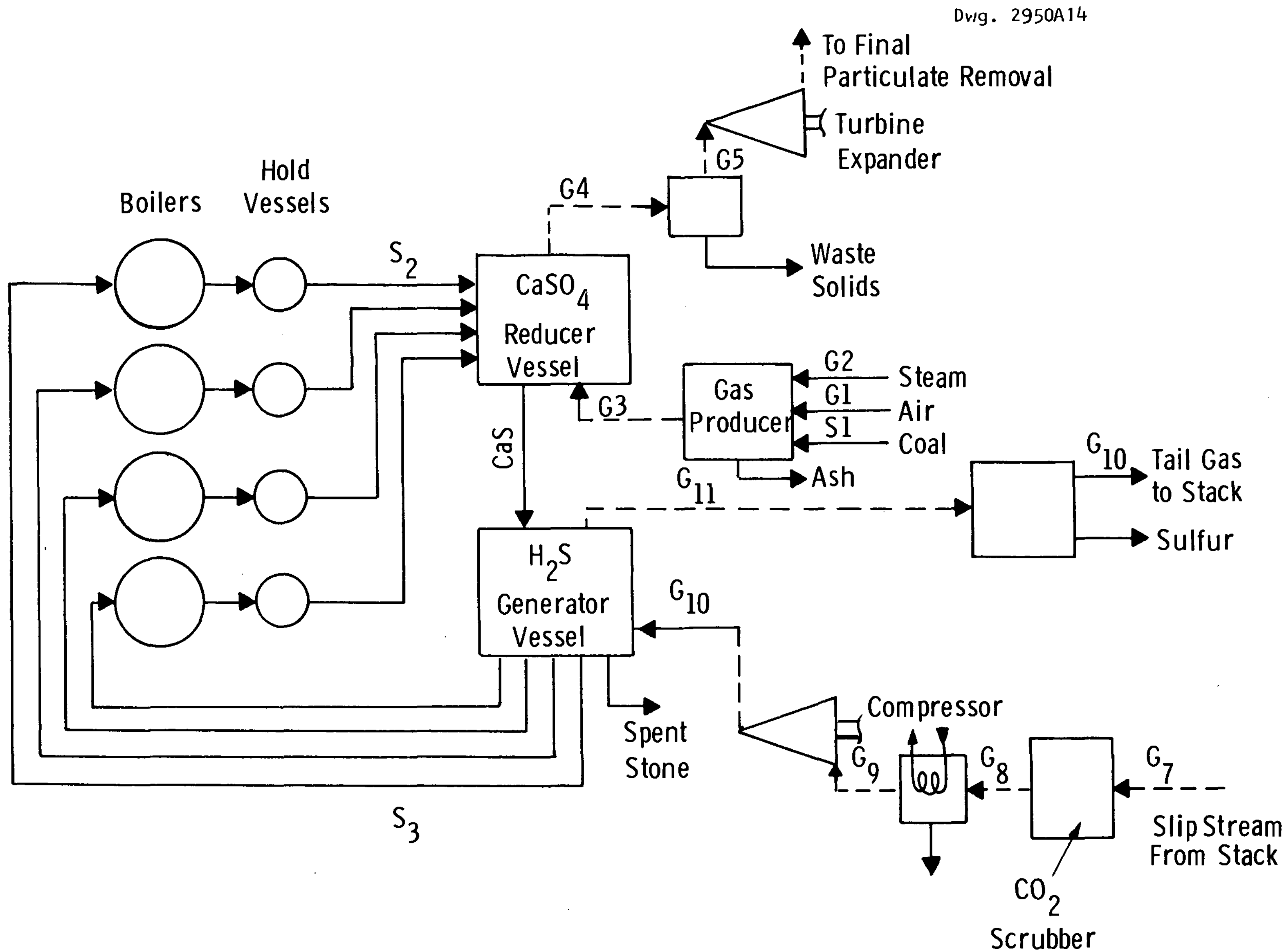


Fig. I-1—Pressurized regeneration System—Flow Diagram

TABLE I-1

HIGH PRESSURE REGENERATION BALANCE

Stream	T (°F)	P (psia)	lb/hr	lb mole/hr	Mole % H_2	H_2O	CO	CO_2	N_2	O_2	H_2S
G1	640	150	183,000	6,300	--	--	--	--	79.1	20.9	--
G2	640	150	9,000	500	--	100	--	--	--	--	--
G3	1500	135	221,000	8,400	8.0	8.4	16.3	7.4	59.4	--	0.5
G4	1500	120	182,000	6,800	1.0	19.8	0.3	5.5	73.4	--	--
G5	1500	116	182,000	6,800	1.0	19.8	0.3	5.5	73.4	--	--
G6		∿18	182,000	6,800	1.0	19.8	0.3	5.5	73.4	--	--
G7	230	135	595,000	20,000	--	8.6	0.2	16.0	73.4	1.7	--
G8	220	19.7	266,000	10,000	--	68.8	--	30.9	0.2	--	--
G9	212	19.0	179,000	6,500	--	63.7	--	36.3	--	--	--
G10	>212	180	179,000	6,500	--	63.7	--	36.3	--	--	--
G11	1100	165	117,000	4,900	--	73.9	--	16.1	--	--	10.0
S1 (coal)	Amb	135	33,500								
S2 (spent dolomite)	1500	135	400,000								
S3 (regenerated dolomite)	1100	180	450,000								

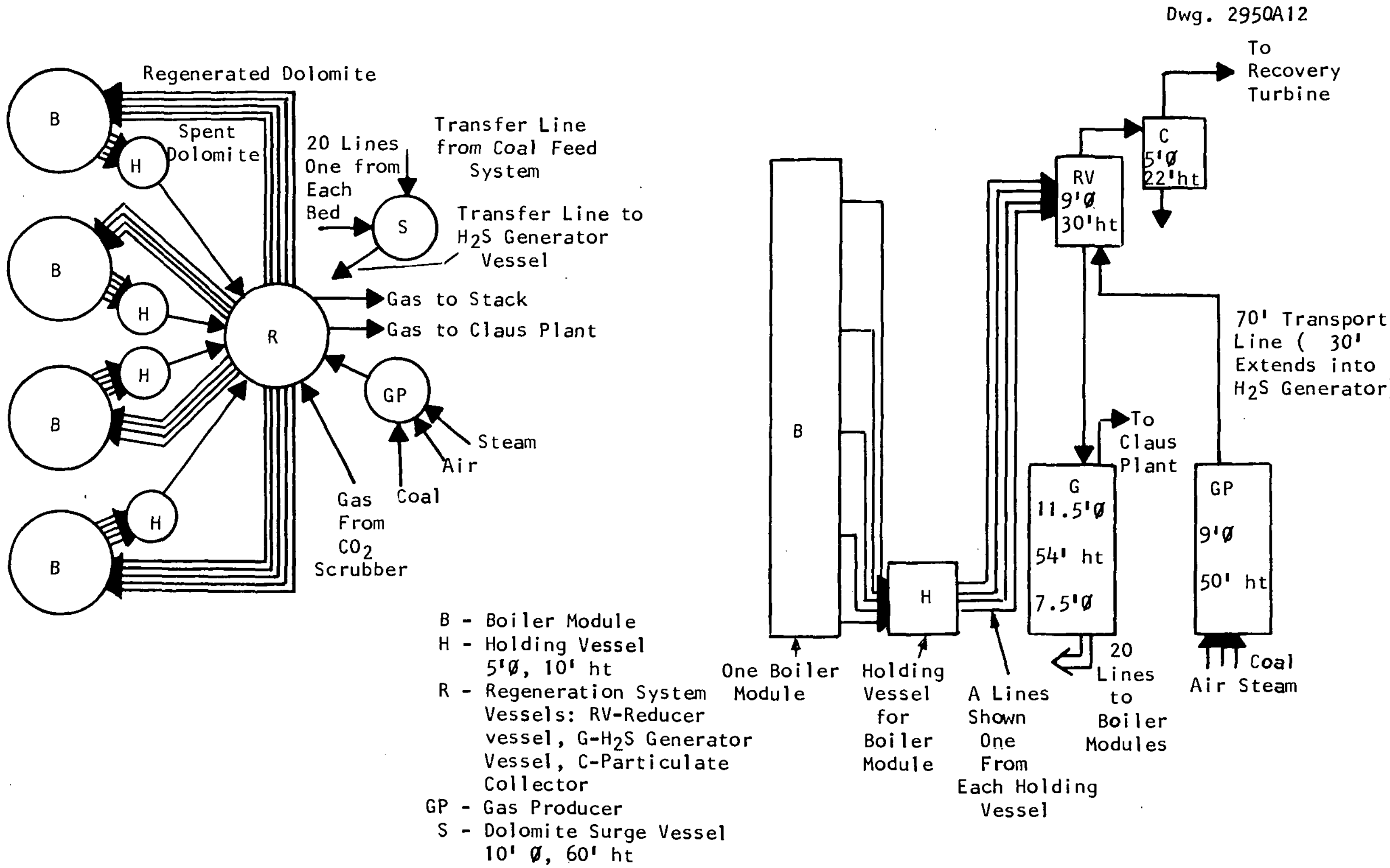


Fig. I-2 — Pressurized regenerator System: Plant Layout

TABLE I-2

DOLOMITE SURGE TANK UTILIZATION

Start up

- Surge tank gas turbine assumed operating to supply transport air.

- Use coal feed system to fill the fluid beds.

  - 2nd superheater and rehcated beds are filled to operating depth.

  - pre-evaporator and 1st superheater bed are filled to $\sim$ 4 ft.

- Use coal feed system to fill surge tank with stone required to fill the partially filled beds, for make-up, or for start-up of regeneration vessels. Stone could be transported to the regenerator vessels directly from the beds.

- After the partially filled beds are ignited, stone is added from the H_2S generator vessel, which is supplied by the surge tank.

Shut down

- The surge tank is designed to receive all of the stone from two boiler modules (10 beds). This is provided for maintenance. The stone is removed from each bed via a pipe through the distribution plate to the surge tank. The surge tank is below grade to provide gravity feed of the solids from the beds.

Normal operation

- The surge tank could provide back-up dolomite in case of problems with the dolomite feed system or poor performance which could be overcome by using the surge tank.

Dwg. 2950A20

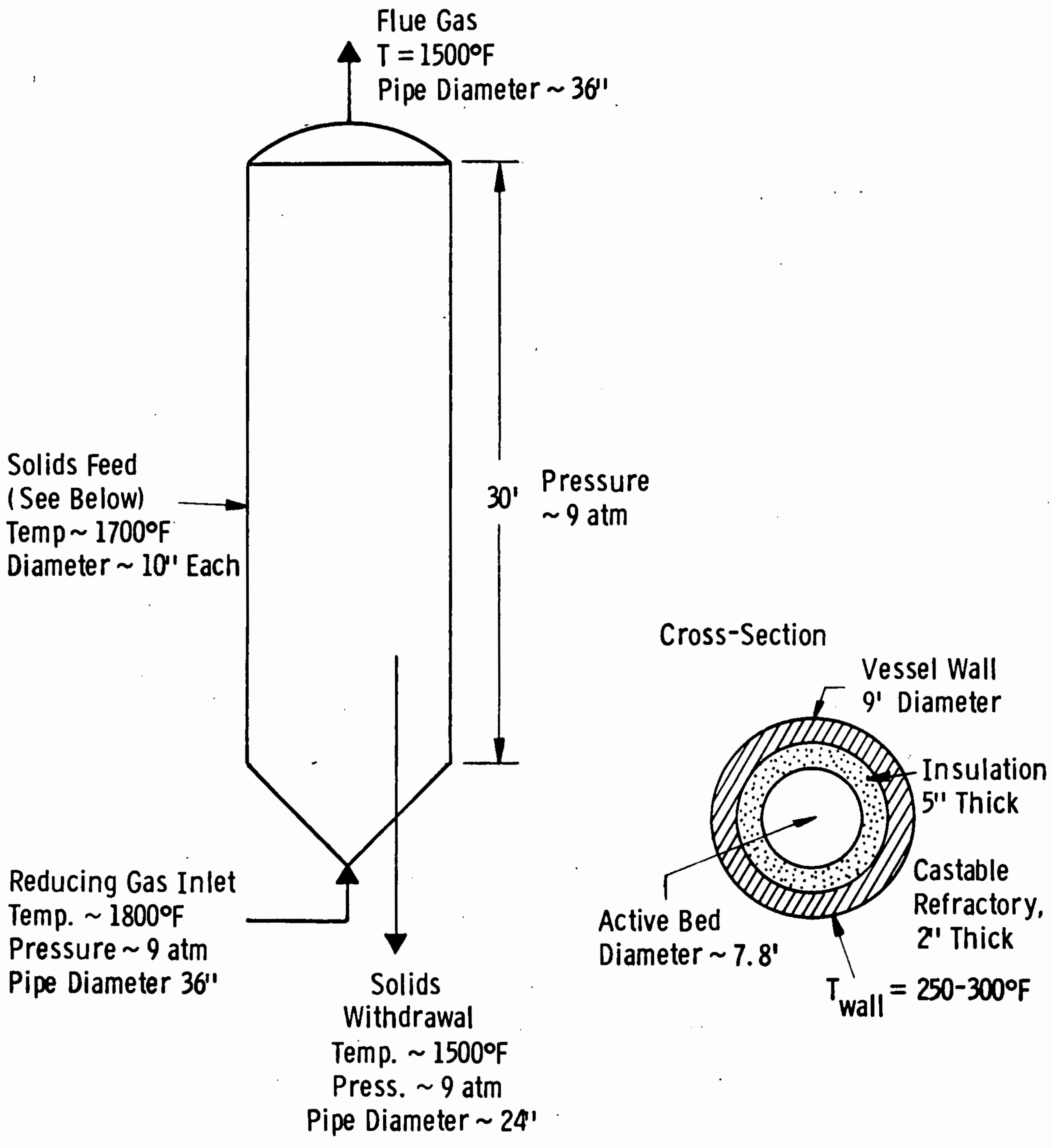


Fig. I-3—Reducer Vessel

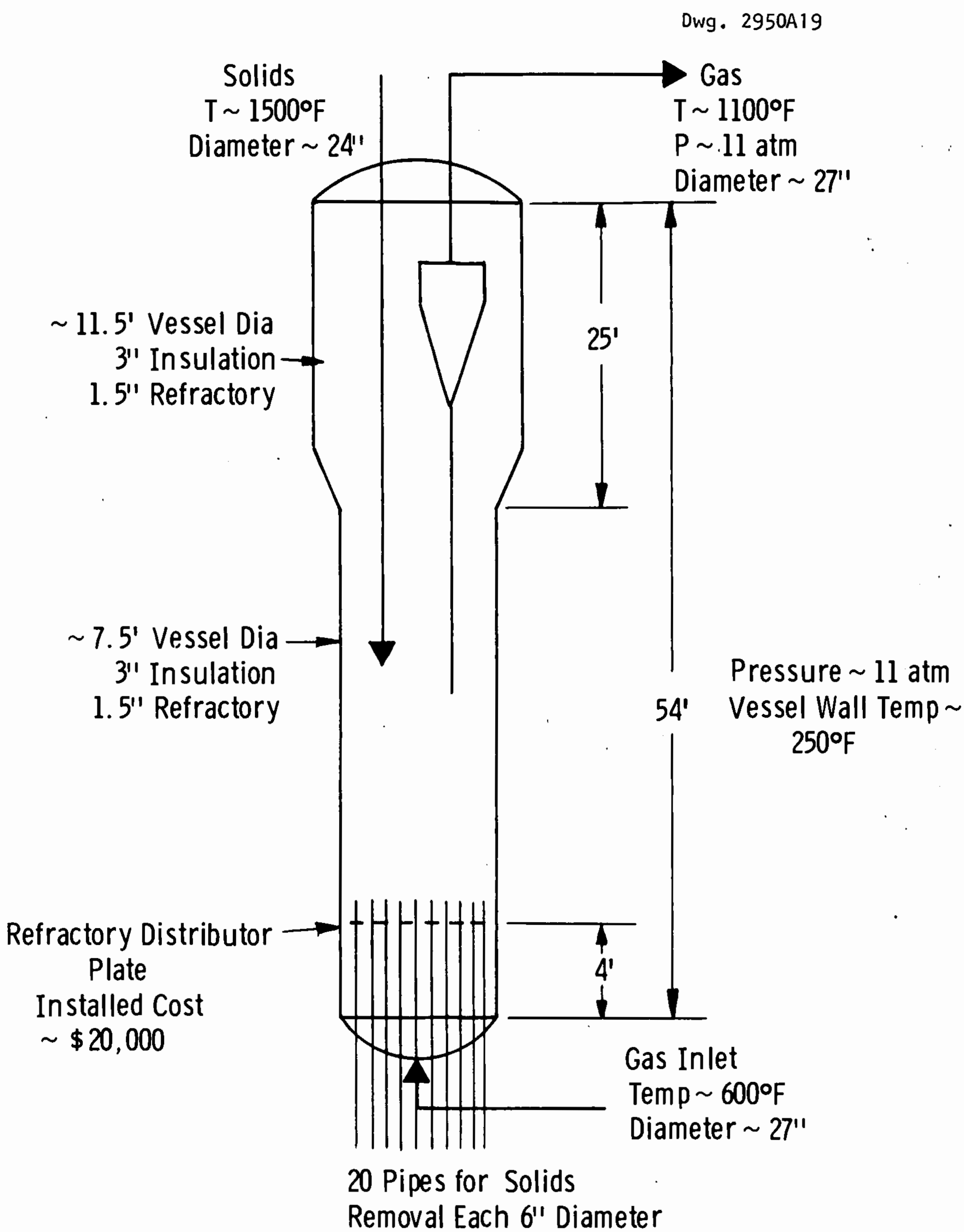


Fig. I-4 — H_2S Generator Vessel

Dwg. 2950A17

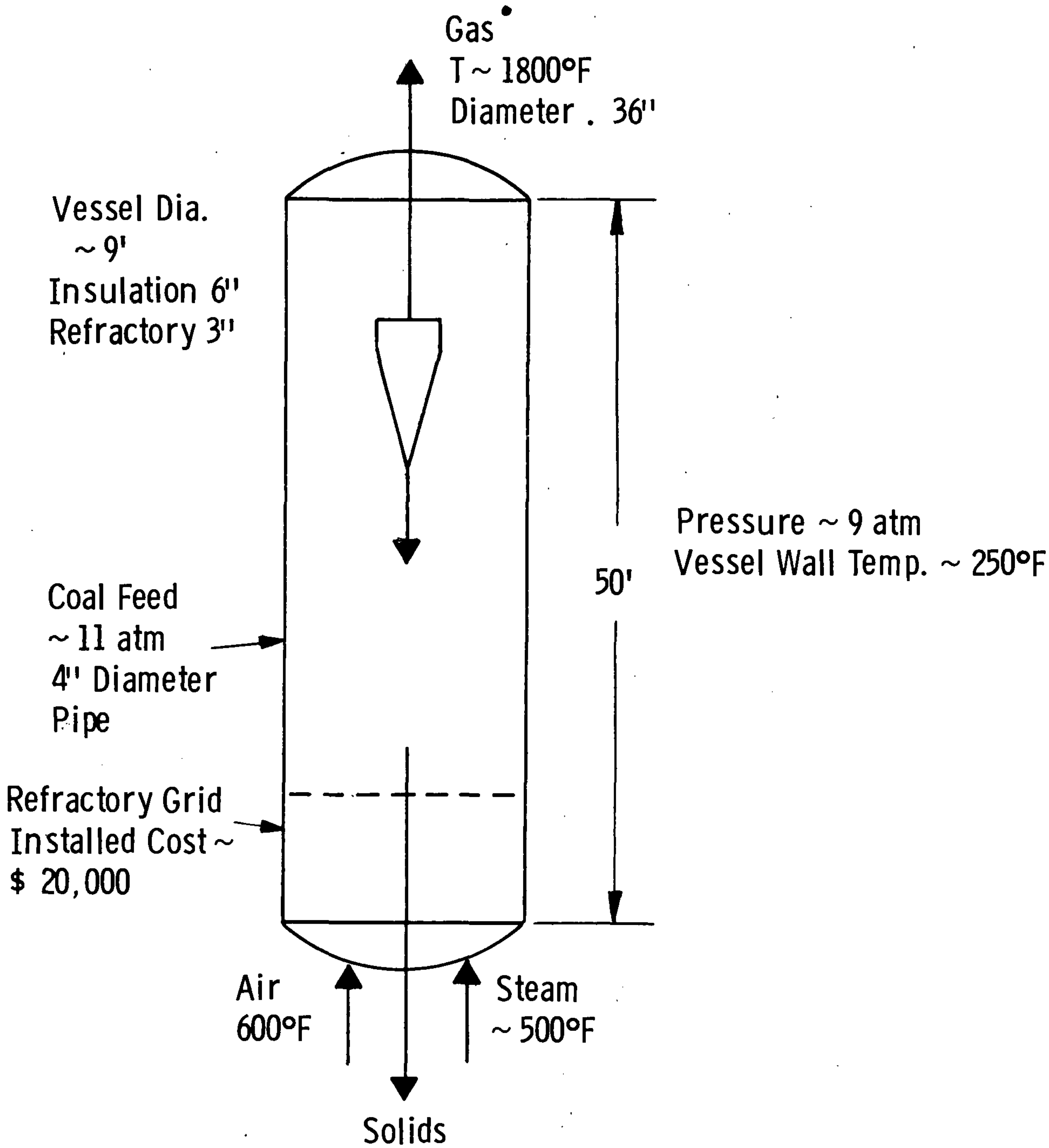


Fig. I-5—Gas Producer

Dwg. 2950A18

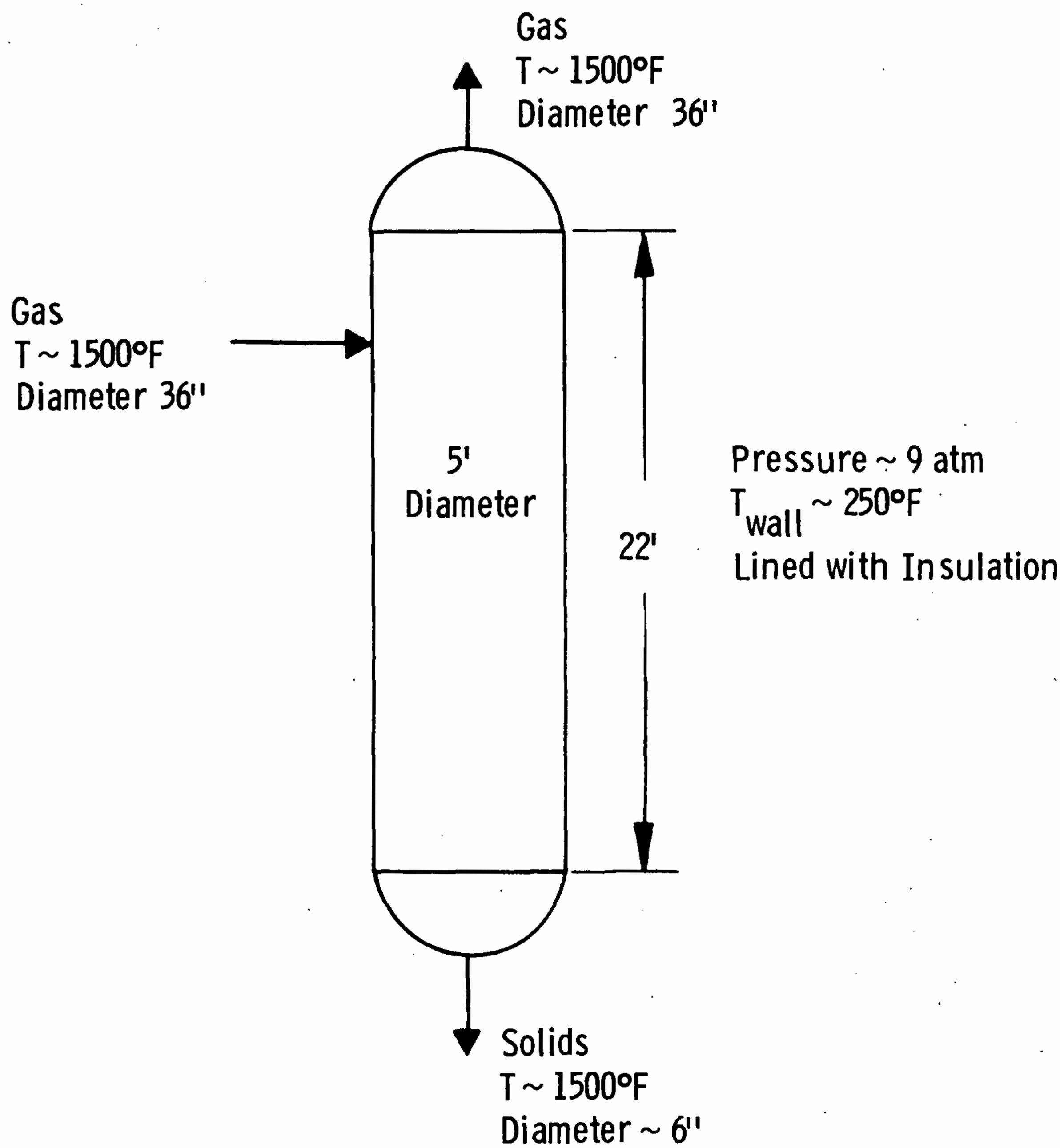
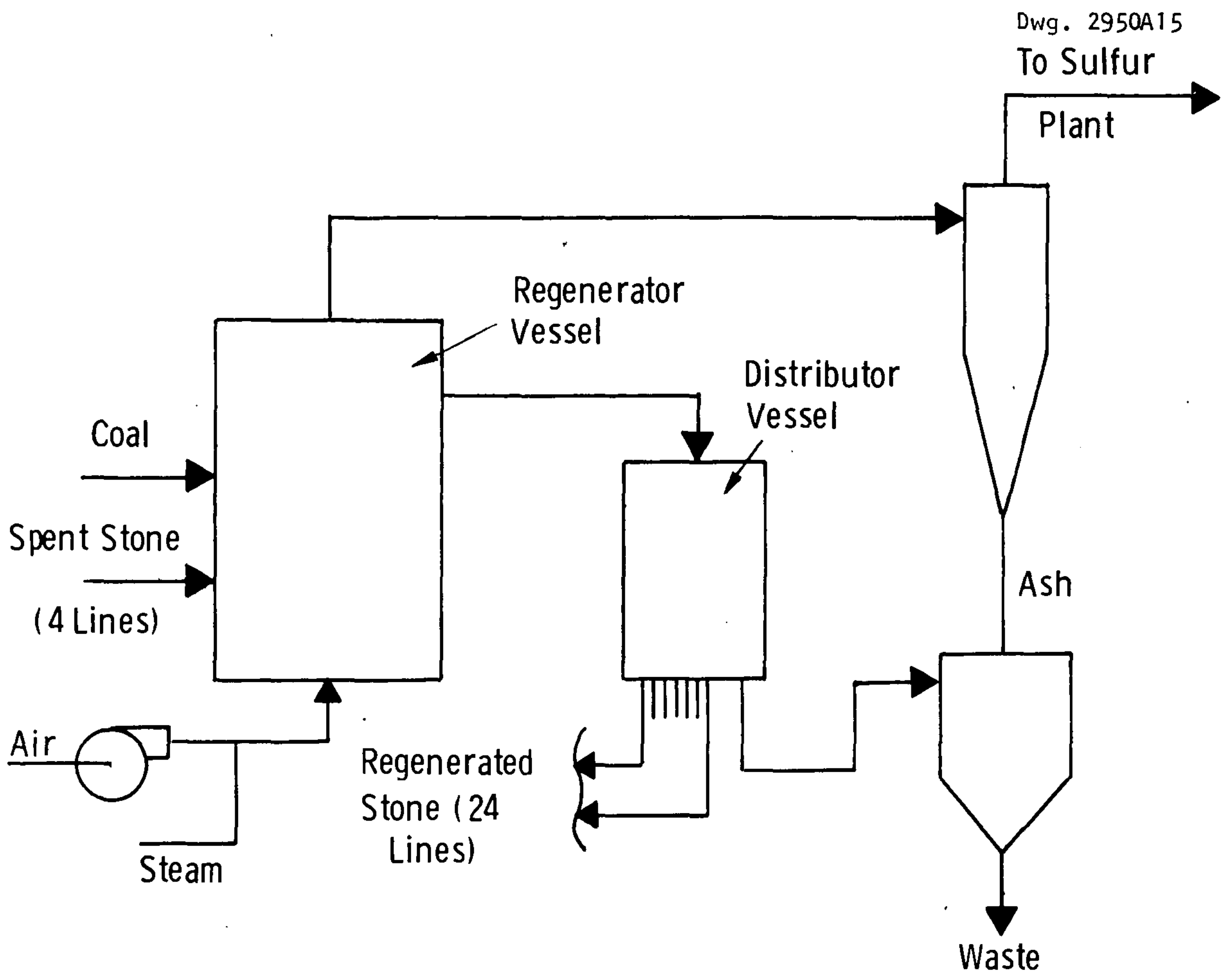


Fig. I-6—Pressure vessel for particulate collector



Spent Stone: 56.6 tph = 113,000 lb/hr, 37.8 wt % $CaSO_4$, 1500°F

Regenerated Stone 43.5 tph = 87,000 lb/hr, 1.33 wt % $CaSO_4$, 2000°F

Waste Stone: ~ 9000 lb/hr

Air: 2800 lb mole/hr

Coal: 5.6 tph

Ash: 0.48 tps

Sulfur Gas: 3500 lb mole/hr 2000°F

SO_2: 6.3% (mole %)	N_2: 61.1%
H_2O: 13.3%	CO: 0.3%
CO_2: 18.9%	H_2: 0.1%

Adiabatic Heat: −9 MMBTU/hr

Fig. I-7 —Atmospheric pressure regeneration flow diagram

Dwg. 2950A13

Atmospheric-Pressure 300 MW Boiler System

1. Boiler Module..........16' x 16' ea
2. Regenerator Vessel.....See Sketch
3. Distributor Vessel.....12' Ø x 12' ht.
4. Hold Drums.............5' Ø x 5' ht.
5. Cyclone ....
6. Waste Stone Hopper.....8' Ø x 12' ht.
7. Air Blower.............

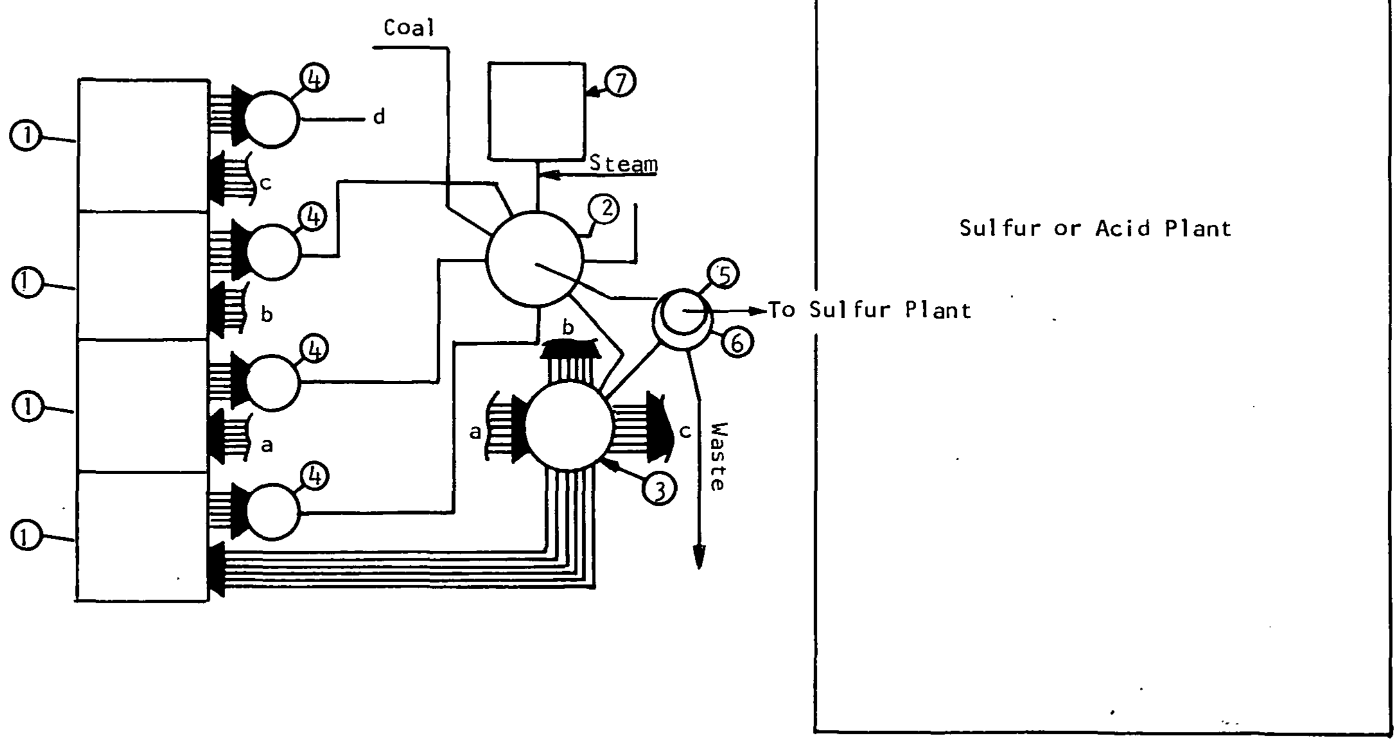


Fig. I-8 — Atmospheric pressure regenerator plant layout

Dwg. 2950A16

Bottom : 9' I.d. 6" Refractory; 6" Insulation
Top : 12' I.d. 6" Refractory; 6" Insulation

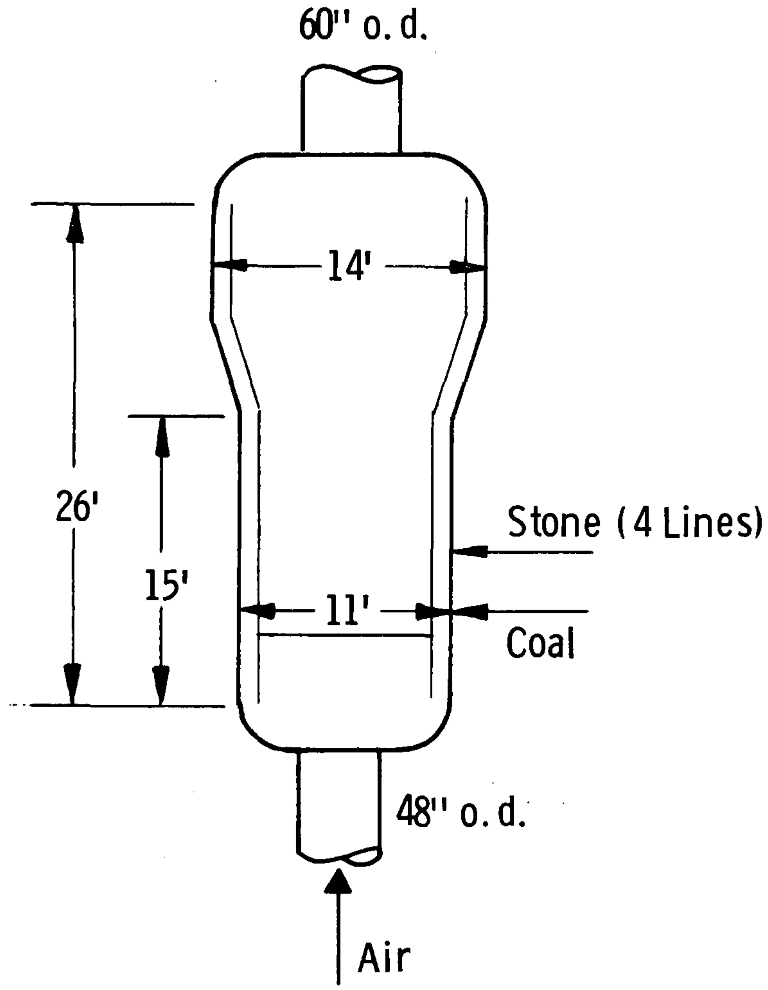


Fig. I-9 — Regenerator Vessel

TABLE I-3

PROCESS VESSEL COSTS - SO_2 RECOVERY PROCESS (200 MW)

	F.O.B.	Erected
Regenerator vessel	$115,000	$485,000
Distributor vessel	68,000	290,000
Hold drums (4)	41,000	175,000
Waste stone hopper	21,000	89,000
Cyclone		
Air blower		

PETROCARB, INC. 250 BROADWAY, NEW YORK, N. Y. 10007

CABLE ADDRESS CADOCARB NEW YORK

TELEPHONE (212 267-8510

April 30, 1971

Westinghouse Electric Corporation
Research and Development Center
Beulah Road
Pittsburgh, Pennsylvania 15235

Attention: Dr. D. L. Keairns, Senior Engineer
Chemical Engineering Research

Gentlemen:

We are pleased to supply "order of magnitude" cost estimates for two additional solids feeding systems required for the 600 M.W. pressurized fluid bed boiler system. These are referred to as A (Solids handling for the dolomite regeneration system) and B (Coal feed to the gas producer).

Our concept for System A has been described to you verbally but will be summarized briefly herein. We propose that solids will be removed from each of the fluid beds in the boiler through down-legs that are kept full of solids. A level control impulse from each bed will initiate a regulated flow of solids from the down-legs by means of a fluidizing jet of air. Solids will thus flow into a Petrocarb High Temperature Injector from each group of four fluid beds in a boiler module. There will thus be four Petrocarb High Temperature Injectors — one associated with each of the four boiler modules. The Injector will continuously feed solids to the upper regeneration vessel. The rate of solids flow will be controlled by two rate control functions, a) the pressure in the regenerator - which will vary as directed by a gamma ray level control signal indicating the level in the Injector- and b) the volume of transport air. You are to provide means for varying the pressure in the regenerator to follow a linear signal from our Injector. The pressure should be capable of being varied from approximately 8 to 10 atmospheres based on 10 atmospheres in the boiler. We will provide necessary transport air controls.

The Petrocarb High Temperature Injector will be a refractory lined pressure vessel approximately six feet outside diameter and approximately ten feet overall height. Approximately three foot clearance should be provided below the vessel for additional fittings.

ET NO. 2 FROM **PETROCARB, INC.** TO Westinghouse Electric Corporation
Pittsburgh, Pennsylvania 15235
April 30, 1971

The second phase of the System is based on our concept that you will provide an outlet pipe with flange under the lower regenerator vessel for each of the sixteen streams of regenerated dolomite. To these outlet flanges - tentatively six inch pipe size - will be attached Petrocarb Feeder Assemblies which will include a shut-off valve capable of handling 1100°F solids. The equipment will be alloy steel applicable for the service. It will be necessary for the regenerator to modulate in pressure between the range of approximately 10 to 12 atmospheres to control the level of solids in the fluid bed and thus control the rate of solids discharged. Secondary air controls will be provided by Petrocarb to further adjust the flow of solids to the fluid beds in the boiler. Transport piping is included in the estimated costs on the assumption that there will be 2500 linear feet of piping between the regenerator vessel and the fluid beds in the boilers.

The lump sum estimated price for the dolomite recirculation system is Five Hundred Thousand Dollars ($500,000.00), exclusive of erection costs.

The air required to serve the dolomite handling system would be approximately 50,000 SCFM at 175 psig.

System B (Coal feed to the gas producer) will be capable of feeding prepared coal at a controlled rate of 11 TPH to a single point in the gas producer. The arrangement will encompass a typical Petrocarb system including a Surge Bin, Storage Injector and Primary Injector generally as covered in our letter of February 18, 1971 to Foster Wheeler. The cost of this system would be approximately One Hundred Twenty Five Thousand Dollars ($125,000.00) assuming that it would be procured as part of the package of coal feeding systems supplying the boilers.

This coal feeding system would require approximately 2,000 SCFM of air compressed to 175 lbs/in^2. Incidentally I wish to call your attention to the fact that the overall air requirements required for feeding coal to the boilers may be reduced by about 25% in the final analysis if the instantaneous demands are scheduled by a logic system to avoid pyramiding of peak loads. Some adjustment of cycle time and equipment sizing might be affected, but this should not influence equipment costs at this time.

Enclosed is a copy of a letter to Foster Wheeler which confirms the costs for changes in the coal feeding system and the cost of a solids feed system to the carbon burn-up cell

We are looking forward to working with you on further phases of the project.

Very truly yours,

PETROCARB, INC.

Reintjes

Harold Reintjes
President

HR:jml

encls.(1)
-Letter dtd. 4/30/71 to Foster Wheeler

PETROCARB, INC. 250 BROADWAY, NEW YORK, N. Y. 10007

CABLE ADDRESS "CABOCARB" NEW YORK

TELEPHONE (212) 267-8510

April 30, 1971

Foster Wheeler Corporation
John Blizard Research Center
12 Peach Tree Hill Road
Livingston, New Jersey 07039

Attention: Mr. Richard W. Bryers, Research Associate
Subject: Coal Injection Systems for Pressurized-Fluidized Boilers
Petrocarb Project E-430

Gentlemen:

At the request of Dr. Dale L. Keairns of Westinghouse we are pleased to supply the following additional information on the approximate costs of coal feeding systems for the pressurized fluid bed boilers:

	300 M.W. System	600 M.W. System
Four Outlet Coal Feeders	$1,800,000.00	$2,800,000.00
Eight Outlet Coal Feeders	$2,000,000.00	$3,000,000.00

Also Dr. Keairns requested that we supply some information on a system to feed 850° solids to sixteen carbon burn-up cells associated with the boiler modules. The solids were stated as being approximately 100 lbs/cu ft. bulk density and that the maximum rate was 10 tons/hr. Obviously this unit must be designed for the abnormally high temperature of 850°F but this can be accommodated. Other than this, the system would be quite similar to the other systems described in earlier correspondence.

A cost of approximately $250,000.00 has been estimated for this system on the same basis as previously quoted.

The peak load air demand would be approximately 2,000 SCFM compressed to 200 psig.

Please let us know if you have any questions concerning this matter.

Very truly yours,

PETROCARB, INC.

Harold Reintjes
President

HR:jml
cc: Dr. D. L. Keairns, Senior Engineer
Westinghouse Electric Corporation
Research and Development Center
Beulah Road, Pittsburgh, Pa.

Ford, Bacon & Davis Texas

Incorporated

ENGINEERS — CONSTRUCTORS

P. O. Box 38209 • Dallas, Texas 75238

HAROLD B. WARNICK
Vice President

2908 National Drive
Garland, Texas
214/278-8121

April 30, 1971

Westinghouse Electric Corporation
Research and Development Center
Beulah Road
Pittsburgh, Pennsylvania 15235

Attention: Mr. L. M. Handman
Chemical Engineering Research

Gentlemen:

In accordance with your letter of 12 February 1971, we have estimated the cost of a Claus type sulfur recovery unit. This unit will feed about 2000 moles/hr of acid gas having about 14% H_2S and 43% CO_2. The cost estimate for this plant has been previously reported to you via telephone by our Chief Process Engineer, Mr. David Parnell, and this letter serves to confirm these numbers.

The feed gas as reported in your letter has as much as 43% water and is available at 1000°F and 130 psig. The presence of this much water is detrimental to the Claus reaction, thus we would recommend that the acid gas be cooled to about 120°F prior to feeding the Claus sulfur plant. The estimated installed cost for the sulfur unit including the suggested feed gas cooler is $1,360,000. This includes a two catalytic reactor Claus plant with a tail gas incinerator-stack combination for oxidizing all the sulfur bearing compounds to SO_2 before atmospheric discharge via the stack.

If the feed gas cooler is not installed, the added water processed by the sulfur plant increases the equipment size. It is estimated the plant would cost $1,410,000 installed without the feed gas cooler. An additional disadvantage of this unit would be the reduction in overall sulfur recovery caused by the excess water present in the system. Installation of the plant without the feed gas cooler is not recommended.

For either of the above plants, a plot area of at least 4000 sq. ft. would be required. There are essentially no tall vessels in the sulfur unit except the incinerator-stack. A summary of the estimated utilities is shown on the attached table.

Recently, as you know, considerable emphasis has been placed on sulfur dioxide (SO_2) emissions from various industrial complexes. Lately, various pollution authorities have tended to require three catalytic reactors on all newly installed Claus type sulfur recovery units. Some states will not permit installation of a two catalytic reactor sulfur recovery plant. The third catalytic reactor would increase the overall sulfur recovery from about 92% to about 94%. This results in a considerable reduction in the amount of sulfur dioxide emitted to the atmosphere. To add a third reactor would cost an estimated $60,000 additional for the sulfur recovery unit.

It is unlikely that the increased sulfur production will pay for the added costs of the third reactor. However, we have found it often necessary to include a third catalytic reactor in the sulfur plant design to satisfy the local, state or federal pollution codes. Frequently, addition of the third reactor has permitted a reduction in the required stack height because of less SO_2 being discharged to the atmosphere. This phenomenon tends to reduce the effective costs of adding a third reactor.

Addition of the third reactor includes the following equipment: (1) third reactor, (2) an additional sulfur condensation pass in the condenser, (3) feed preheater for the third catalytic reactor (reheat exchanger) and (4) an air blower with a higher differential head. The revised blower is required because of the increased pressure drop through the unit caused by the addition of the third catalytic reactor. We feel the third reactor should be seriously considered based on the recent emphasis on reducing SO_2 atmospheric emissions, which is expected to continue.

The tail gas from the sulfur recovery unit is fed to an incinerator where all sulfur bearing compounds are oxidized to SO_2. The incinerator operates at about 1200°F with 25% excess air. The resulting flue gases from a two reactor plant will contain about 10,000 to 12,000 ppm SO_2. From the three reactor plant the SO_2 concentration in the flue gases will be about 7000-8000 ppm.

It should be mentioned that these above prices are budget-type numbers, accurate only within ± 10% to 15%. Also, they are applicable only at the present time (30-60 days). If the proposed plants are to be installed at later dates, a minimum inflation factor of about 0.6% per month (7.2% per year) should be used in arriving at future estimates. For a change in the

Westinghouse Electric Corporation
Attention: Mr. L. M. Handman
April 30, 1971

Ford, Bacon & Davis Texas
Incorporated
DALLAS, TEXAS

size of the sulfur plant, the ratio of the sulfur capacity raised to the 0.6 power can be used to arrive at an approximate estimate for the installed cost of the revised plant.

All of our sulfur recovery plants have been designed under license from Pan American Petroleum Corporation. Presently, we have designed more sulfur recovery units than any other Pan American licensee. Our experience provides extensive know-how for design and operation of sulfur recovery units. This particular process was developed in 1956 and is a modern day improvement to the classic Claus process. It involves patented equipment and design features combining process functions into one or two pieces of equipment such as oxidizing H_2S, cooling the gases, condensing sulfur and separating the liquid sulfur from non-condensed gases. These design concepts have provided a better operating plant, installed and maintained at lower cost than was possible with the older style Claus units. The attached article reprint describing various sulfur plant design concepts provides additional information on this process.

Ford, Bacon & Davis Texas has consistently made efforts to improve our design by obtaining actual operating data from on-stream units. This has resulted in a continued optimization and improvement in the type plant we design and build. Attached is a current list of sulfur plants we have designed and/or constructed.

We hope this information will be helpful to you. We appreciate your continued interest in our services and if we can be of further assistance, please contact us.

Very truly yours,

FORD, BACON & DAVIS TEXAS

Harold B. Warnick

Harold B. Warnick

DCP:jr
Attachments

April 30, 1971

TABLE I

UTILITY SUMMARY

Power		KW
Air Blower		250
Sulfur Pump (1)		20
Boiler Feed Water Pump		15
Lighting and Controls		15
	Total	300
Fuel Gas, SCFM		
Acid Gas Burner (2)		300
Incinerator		200
Instrument Air, SCFM		8
Boiler Feed Water, GPM (3)		50
Steam, lbs/hr		
Production		24,000
Consumption		1,200
Cooling Water, GPM		3,200 (4)

(1) Not for Continuous Operation
(2) For Startup Only
(3) Based on 5% blowdown, rate measured @ 60°F
(4) Based on 20°F rise

Benfield CORPORATION • 666 WASHINGTON RD., PITTSBURGH, PA. 15228

February 23, 1971

Westinghouse Electric Corporation
Research and Development Center
Beulah Road
Pittsburgh, Pa. 15235

Attention: Mr. L. M. Handman

Subject: Benfield design for flue gas scrubbing

Reference: Your letter of February 9, 1971

Gentlemen:

We have studied the process conditions outlined in your referenced letter and offer the following information on a Benfield unit to satisfy those conditions:

1. Feed gas (flue gas) flow rate to the absorber would be 3,480,000 ncfh or 9,694 lb. mols/hr and this would be at approximately 135 psia and 110° C.

2. CO_2 produced would be 1500+ lb. mols/hr at 19.7 psia and approximately 105° C. The CO_2 out of regenerator would be mixed with about 60,000 lbs/hr of water vapor and there would be approximately 15 lb. mols of other gases - mostly nitrogen.

3. A $\pm$ 15% estimate of the total investment cost of the Benfield plant would be $1,300,000.

4. Approximately 535 hp would be required for solution circulation.

5. Regeneration energy requirement would be approximately 80.2 MM BTU/hr.

6. Solution cooler duty would be approximately 53 MM BTU/hr.

Very truly yours,

THE BENFIELD CORPORATION

Robert D. Karns

Robert D. Karns
Engineering Sales Manager

RDK:em

CABLE ADDRESS: BENFIELD PITTSBURGH • PHONE: 412-341-4569 TELEX: 086-732

From HEAT TRANSFER DIVISIO
WIN
Date : May 5, 1971
Subject F.B.C.O.M. Project
Regenerator Vessels
Study

Westinghouse R&D Center, Pittsburgh
Energy Systems Research Bldg. 501

Mr. D. Keairns

cc: Messrs. S. Lemezis
R. Giardina

This is to confirm the prices given to you by phone on May 3, 1971.

Based on the preliminary information given to R. Giardina, our estimates of the 1971 price for the pressure vessels are as follows:

One (1) Cyclone Pressure Vessel	$52,500
One (1) Reducer Vessel	$110,000
One (1) Gas Producer Vessel	$147,500
One (1) H_2S Generator Vessel	$210,000

These estimates include only the shells and insulation. Internal structures, vessel supports, support skirts, and accessories are not included.

No attempt has been made to ascertain a realistic shipping date. If an indication is desired, please contact me.

The prices given above are for reference and study purposes only and are not to be considered as an offer to sell at these levels.

G. F. Pierson

G. F. Pierson

GFP/nmm

Monsanto
Enviro-Chem
Systems Inc.
10 SOUTH RIVERSIDE PLAZA / CHICAGO, ILLINOIS 60606 / (312) 782-5041

June 24, 1971

Westinghouse Electric Corporation
Research and Development Center
Beulah Road
Pittsburgh, Pennsylvania 15235

Attention: Mr. L. M. Handman
Chemical Engineering Research

Subject: 250 T/D Sulfuric Acid Plant

Gentlemen:

In answer to your letter of May 7, your study of a limestone-base sulfur dioxide recovery process, we are submitting the following information.

As you requested, we have included order of magnitude pricing and operating data on an interpass absorption sulfuric acid plant based on the information as follows:

Flow Rate	3500 pound moles per hour
Pressure	Atmospheric
Temperature	400 - 500°F.
Dust Loading	0.5 gr/scf (maximum)

Composition — Mole %

SO_2	6.3%
H_2O	13.3%
CO_2	18.9%
N_2	61.1%
CO	0.3%
H_2	0.1%

Continued . . .

Our order of magnitude cost for an interpass absorption sulfuric acid plant with a rated capacity of 250 tons per day (100% H_2SO_4 basis) on a turnkey basis is estimated to be $3,100,000.

The above price does not include:

Product acid storage
Water cooling tower
Paving
Piling
Sales or use tax

The plot required would be approximately 125' by 200'.

Operating utilities are as follows:

Power	1100 HP
Other electrical	70 KVA
Natural Gas	0 - 500 scfm
Instrument air -	30-40 scfm
Process water -	20 GPM
85°F. Cooling water -	3000 GPM

The estimated yearly downtime is 15 days per year. Estimate annual maintenance cost will average 4-1/2% of installed cost.

The plant will handle a turndown of 50% of the feed gas rate keeping the mole % constant.

The process is relatively insensitive to fluctuations of SO_2 concentrations of plus or minus 1%. The conversion efficiency usually increases as SO_2 concentration decreases, assuming O_2:SO_2 ratio increases.

Monsanto Enviro-Chem will guarantee 99.5% conversion of SO_2 at the converter to SO_3. The stack gas will contain 500 ppm or less SO_2.

Feed gas characteristics shown above:

Temperature	400-500°F.
Pressure	Atmospheric
Dust Loading	0.5 gr/scf maximum.

Continued . . .

Westinghouse Electric Corporation
Pittsburgh, Pennsylvania 15235
Mr. L. M. Handman

June 24, 1971
Page Three

We have included a typical "Description of Process" for an interpass absorption sulfuric acid plant.

If we can be of further assistance, do not hesitate to contact us.

Very truly yours,

MONSANTO ENVIRO-CHEM SYSTEMS, INC.

H C Heinemann

H. C. Heinemann, Sales Director
Chemical Process Division

HCH:slc
Attachment

6/28/71

B. DESCRIPTION OF PROCESS

A general description of the process and plant is as follows:

The process consists of three (3) principal steps:

1. Purification, cooling and drying of the sulfur dioxide gas from the copper converters.

2. Conversion of sulfur dioxide gas to sulfur trioxide gas.

3. Absorption of the sulfur trioxide gas in sulfuric acid.

The copper converter gas contains dust, metallic fume, acid mist, water vapor, and other impurities varying with the composition of ore, converter operation, gas cooling in the converter gas handling and cleaning system, and efficiency of dust removal.

The gas purification steps consist of gas scrubbing, gas cooling and acid mist removal. The gas scrubbing and gas cooling are accomplished in a combination unit.

Most of the dust, metallic fume and some acid mist are removed from the gas by scrubbing with acidic liquor in the lower open-type humidifying section. The gas flows upward through a spray of weak acid. The gas is cooled by evaporation of water from the weak acid. The weak acid is recirculated without cooling.

The gas then passes through the gas cooling section, which is located on top of the humidifying section. Cooling is achieved by circulating cooled weak acid over the packed section. The weak acid is cooled by cooling tower water in shell and tube heat exchangers. A purge stream of weak acid is delivered to battery limits.

B. DESCRIPTION OF PROCESS (Cont'd)

The process gas then flows through an electrostatic precipitator. In the precipitator, the acid mist and those solids not previously removed in the humidifying and gas cooling tower are removed.

The removal of the acid mist and solids is accomplished by passing the process gas between pairs of electrodes. An electrical charge is imparted on the particle of acid mist or solid and the particle is deposited on the electrically grounded collecting tube. The acid mist serves to wash the solids collected to the bottom chamber of the precipitator. Liquid effluent from the mist precipitator flows to the humidifying and gas cooling system.

The process gas and dilution air then pass through the packed drying tower countercurrent to a flow of 93% H_2SO_4 to remove water vapor contained in the gas. Sufficient acid is circulated over the tower so that the water vapor removed does not significantly reduce the strength of the acid in a single pass over the tower. Entrained acid in the gas is removed by a mist eliminator pad installed in the top section of this tower.

Gas from the drying tower is compressed in the main compressors and forced through the remainder of the plant.

The clean dry gas now passes through the shell sides of two cold heat exchangers arranged in series, and the hot heat exchanger where it is heated.

From the hot heat exchanger, the gas flows to the first converter pass where the sulfur dioxide is partially converted to sulfur trioxide in the presence of Monsanto Enviro-Chem's vanadium pentoxide catalyst. The conversion reaction consumes oxygen and produces heat.

B. DESCRIPTION OF PROCESS (Cont'd)

Gas leaving the first converter pass flows in parallel to the tube sides of the hot heat exchanger and the hot interpass heat exchanger, where it is cooled by the SO_2 gas passing through the shell sides, and then flows to the second converter pass.

In the second converter pass, additional conversion of sulfur dioxide to sulfur trioxide takes place accompanied by the generation of additional heat.

Hot gas leaving the second converter pass is then cooled by passing through the cold interpass heat exchangers arranged in series. The gas then goes to the interpass absorption tower where the SO_3 is absorbed. After the absorption has been completed, the process gas and acid mist pass through a high efficiency Brink Mist Eliminator. The gas then passes back through the cold interpass heat exchangers and the hot interpass heat exchanger before entering the third converter pass. On leaving the third converter pass, the gas flows through the tube side of the cold heat exchangers where it is cooled.

The converter preheater is used to bring the catalyst in the converter system up to the temperature required for the catalytic conversion of SO_2 to SO_3 if the converter temperatures have decreased due to a prolonged shutdown. The preheater will also be used to heat part of the incoming SO_2 gas stream as the gas strength drops below 5.3% SO_2 or the gas volume exceeds 117,106 SCFM (dry basis).

After leaving the tube side of the cold heat exchangers, the SO_3 gas passes to the final absorbing tower. The gas passes through the packed absorbing tower counter current to a flow of 98-99% H_2SO_4 which absorbs that SO_3 contained in the process gas. Sufficient acid is circulated over the tower so that the SO_3 absorbed does not significantly increase the strength of the acid in a single pass over the tower.

B. DESCRIPTION OF PROCESS (Cont'd)

After the absorption has been completed, the process gas passes through a Brink Mist Eliminator. The acid mist particles present in the gas are almost completely removed by the mist eliminator.

The gas now passes to the atmosphere through the exit stack.

The strong acid circulating systems used to dry the SO_2 gas from the purification system and to absorb the SO_3 gas from the converter system are separate systems, each with a pump tank, circulating pump, coolers and piping system. They are interconnected to permit control of drying and absorbing acid concentrations.

The acid circulated over the drying tower is weakened by the water vapor removed from the SO_2 gas, while the acid circulated over the absorbing tower is strengthened by the absorption of the SO_3 gas. To counteract these strength changes, absorber acid is pumped to the drying acid pump tank and drying acid is allowed to underflow to the absorbing acid pump tank.

If sufficient water to control the strength of the absorbing acid is not available from the drying acid, additional dilution water is added directly to the absorbing acid pump tanks.

The dilution of the drying acid by the water vapor from the process gas and the addition of 98-99% acid to the drying acid pump tank raises the temperature of the acid. To reduce the temperature, the drying acid is passed through coolers where cooling tower water removes the heat. A side stream of 93%

B. DESCRIPTION OF PROCESS (Cont'd)

acid is taken from the drying tower pump tank, equivalent to the quantity of acid produced and flows through the 93% product cooler to storage.

In the interpass absorbing tower, the SO_3 does not combine directly with water, but must be combined indirectly by absorbing it in sulfuric acid where the SO_3 reacts with water in the acid. The temperature of the 98-99% H_2SO_4 circulated over the interpass absorbing tower increases due to the heat of formation and the sensible heat of the gas stream entering the tower. Acid from the bottom of the interpass absorbing tower is passed through coolers and returned to the top of the tower. A side stream of 98-99% acid is taken from the interpass absorbing tower pump tank and flows to the drying tower pump tank.

Sufficient water and 93% drying tower acid are admitted to the interpass absorption tower system to make up for quantity of 98% acid withdrawn from the system, and to control the strength of acid circulated over the interpass tower between 98-99%.

In the final absorbing tower, SO_3 in the gas stream reacts with water in the 98-99% circulating acid. The temperature of the stong acid circulated over the final absorbing tower increases due to the heat of formation and the sensible heat of the gas stream entering the tower. Acid from the bottom of the final absorbing tower is passed through coolers and returned to the top of the tower. A side stream of 98-99% acid is taken from the final absorbing tower pump tank, equivalent to the quantity of acid produced and flows through the 98% product cooler to storage. Sufficient 93% acid from product stripper is admitted to the final absorbing tower system to control the strength of acid circulated over the tower between 98-99%.

B. DESCRIPTION OF PROCESS (Cont'd)

The plant water balance is based on an incoming gas strength of 5.3% SO_2 entering the drying tower. When the incoming gas strength drops below 5.3% SO_2 for an extended period of time, it will be necessary to bring 98% acid back from storage to hold drying acid strength at 93%. When the incoming gas strength rises above 5.3% SO_2, surplus 98% acid will be returned to storage.

The Ralph M. Parsons Company

Engineers • Constructors

617 WEST SEVENTH STREET, LOS ANGELES, CALIFORNIA 90017

July 20, 1971

Mr. L. M. Handman
Westinghouse Electric Corporation
Research & Development Center
Beulah Road
Pittsburgh, Pennsylvania 15235

SUBJECT Your Letters of June 4 and June 7, 1971
Concerning Conversion of SO_2 or H_2S to
Sulfuric Acid or Elemental Sulfur

Dear Mr. Handman:

On June 28, 1970 we sent you a letter stating that a plant for the manufacture of sulfuric acid being fed with Stream I of your letter of June 4, 1971 would cost $5.5 to $6 million installed. Approximately half of this capital cost is required in the gas cleaning section of the plant.

The operating cost of such a plant for utilities and operating personnel would be approximately $4.00 per ton of acid produced.

With respect to the manufacture of elemental sulfur from stack gases, I am enclosing a copy of a memo from our Mr. D. K. Beavon, Director of Process Operations. His memo discusses in some detail possible process configurations that might be used to supplement your process schemes. The fifth paragraph of his memo may be of special interest to you.

It is not possible for us to give you a capital cost estimate or an operating cost estimate without doing considerable process design work. Further, before proceeding with process design and estimating, we should have agreement from you that our process scheme is compatible. Since the development of such a process scheme involves a considerable number of manhours, we feel we should be paid for this work. We estimate such an effort would cost approximately $8,000.00 plus whatever travel expenses might be involved. We believe that such an effort might be of considerable value to your project.

We will be pleased to discuss such an undertaking with you in more detail.

I am enclosing our latest brochures dealing with Sulfuric Acid, and Gas Processing Plants and Sulfur Recovery Units. As you will see by the enclosed brochures we have had substantial experience in the field of gas treating and sulfur recovery.

Very truly yours,

THE RALPH M. PARSONS COMPANY

Dudley Field

Dudley Field

DF:jp
Enclosures
As above

THE RALPH M. PARSONS COMPANY

INTEROFFICE CORRESPONDENCE — Date July 2, 1971

To Dudley Field (2)

From D. K. Beavon

SUBJECT Westinghouse Electric Corporation
SO_2 Abatement Process for Boiler Plants

I have reviewed the information sent you by Mr. Handman of Westinghouse R&D and offer the following comments.

When the sulfur is produced in the form of SO_2, there is another option available beside making sulfuric acid, which is the Bureau of Mines' sodium Citrate Absorption process. We have followed the development of this process with keen interest and it appears to me that it is the best one under development for scrubbing out SO_2 and producing elemental sulfur. The gas would need to be pretreated by cooling and washing to remove solids as it would be ahead of a sulfuric acid plant. The SO_2 is then washed out in a countercurrent tower using sodium citrate solution; the rinse solution is reacted with H_2S to form elemental sulfur, which is centrifuged, washed and melted.

The H_2S needed in the process is produced by hydrogenating part of the sulfur product (we have worked out a good design for the H_2S production plant in connection with an inquiry by Sherritt Gordon).

I am inclined to favor the alternate procedure of producing an H_2S-rich gas, with eventual conversion of H_2S to sulfur by the Claus process. I recognize that it costs more to produce the H_2S-rich gas and a complete economic study might change my initial feeling that this is the way to go.

If the H_2S -rich gas is produced, I would recommend a flow scheme different than that described, which produces CO_2 for the H_2S generator by treating a flue gas with MEA or hot carbonate. I think it would be preferable to treat the gas from the H_2S generator at 10 atmospheres pressure with Purisol to extract both H_2S and CO_2 together, then regenerate the Purisol in two stages; the first stage would produce H_2S-rich feed gas for the Claus unit, and the second stage would produce CO_2 for recycling to the H_2S generator. This scheme would be more economical, I think, than the one Westinghouse proposes and I believe that it has a very substantial operating advantage in making the H_2S concentration in the Claus feed controllable at a level which would assure good operation of the Claus unit, regardless of variations in the sulfur content of the fuel.

I note that the written description states that the H_2S concentration of the Claus feed gas is 23% (dry basis) while Table I Stream G11 seems to indicate about 38% on a dry basis. We have found the Claus unit operation begins to be tricky at a concentration around 30% and it would be very desirable to have a flow scheme which can reliably produce the H_2S in a a controlled concentration preferably substantially higher than 30%.

I suggest that Mr. Handman investigate rather carefully the formation of sulfur compounds not listed in his tabulations which are almost surely present in substantial concentrations. In the production of the SO_2-rich gas, it would appear to me that some COS and some CS_2 would be formed. Similarily in the production of H_2S, where the formation of COS is acknowledged, some CS_2 is probably present also.

It is apparent that a firm recommendation on a flow scheme would have to be based on a fairly extensive study which would require considerable process design work and cost estimating, which we have not done. My opinion is that a promising flow scheme would involve the production of H_2S-rich gas, with subsequent Purisol treatment and a Claus plant.

D. K. Beavon

D. K. Beavon

DKB/ju

CC: L. W. Dailey
W. M. Parsons
F. C. Riesenfeld
O. C. Roddey

SUMMARY SULFUR PLANT COST DATA

		Fixed Capital MM $	Operating Cost $/NTD
100 NTD:	DMA Plant	3.2	27
	Asarco	1.2	17
	Total	4.4	44
200 NTD:	DMA Plant	4.8	20
	Asarco	1.8	13
	Total	6.6	33

Basis: 6% SO_2 in feed gas to DMA absorber.

Cost information scaled from Allied Chemicals reports with 10%/year inflation.

APPENDIX J

PRESSURIZED BOILER COMBINED CYCLE PLANT REPORT

Prepared by United Engineers and Constructors Inc.,
a Subsidiary of Raytheon Company

Authors

M. Casapis
E. Berman
W. Craig
J. Crowley

united engineers & constructors inc. a subsidiary of Raytheon Company

I N D E X

635 MW FLUIDIZED BED

BOILER COMBINED CYCLE PLANT

1. DESCRIPTION OF OPERATIONS

PRESSURIZED FLUID BED BOILER

DESCRIPTION OF OPERATIONS

1. GENERAL

The modules of the once-through pressurized fluid bed boiler consist of four individual beds, filled to the required level with particulate matter through which air flows. The pressure drop Δ p across the packed bed of particulate increases with increasing air flow. Fluidization of the bed occurs when the Δ p (lb/in^2) equals the per unit weight loading of the bed (particulate weight/bed area). When the critical air velocity is reached and the bed fluidized, the particulate matter behaves like a fluid and not as a solid mass.

2. CONTROL SYSTEMS

The control systems for operation of the pressurized fluid bed boiler are designed to perform the following functions:

(1) Maintain steam header pressure to the turbine at the desired value by regulation of the turbine valve control.

(2) Maintain the steam header temperature to the turbine at the desired value by regulation of the fuel input to the boilers.

(3) The boiler feedwater flow is set by the load demand from the dispatcher.

A. Combustion Control

The temperature in the steam header to the turbine is directed to a Boiler Master Steam Temperature Controller and compared to an internal set point (automatically set and adjusted to plant demand requirements). As the header temperature decreases/increases, the output signals increases/decreases and are directed as an internal set point to the individual Boiler Bed Steam Temperature Controllers. The temperature at the superheater exit of the individual module is compared to the internal set point. As the header temperature decreases/increases, the output signal increases/decreases. This signal is directed to the individual Bed Fuel Stations which provide:

(1) A means for allowing the operator to adjust the individual bed output so that it represents a greater or lesser share of the boiler load and the reheat bed output to maintain a constant reheat temperature by permitting the operator to add or subtract a manually adjusted bias to the automatic set point signal transmitted by the Master Steam Pressure Controller.

(2) Interruption of the automatic set point signal and substitution of a manually adjusted set point signal.

The output signal of the Bed Fuel Station is directed to the Bed Fuel Controller. Signals from the Bed Fuel Controller are directed to the Petro Carb. control system for feeding of coal to the bed and to the Bed Coal Feeder Transport Air Valve Controller.

B. Steam Flow Control

The basic control parameter for Steam Flow Control from the boiler is Feedwater Control to the boiler.

Steam flow is measured by the Steam Recorder. The recorder senses the differential pressure generated by the orifice in the steam line, extracts the square root and transmits a signal inversely proportional to the measured flow. The output is directed to the Boiler Feedwater Valve Position Controller and compared to an internal set point (set by the dispatcher to meet plant demand requirements). When the steam flow decreases/increases, the output signal from the Boiler Feedwater Valve Position Controller is directed through the Boiler Feedwater Valve Station to the Valve Power Positioner which assumes a position relative to the control signal, opening/closing the Boiler Feedwater Valve, increasing/decreasing feedwater flow to the boiler.

C. Dolomite Regeneration and Make-Up Control

The basic control parameter for recycling regenerated dolomite back to the beds and adding make-up dolomite to the coal feed is performed by an SO_2 analyzer located in the flue gas piping leaving the boiler. The flue gas is monitored for percent of SO_2. When the SO_2 content exceeds an internal set point value, an alarm is activated to notify the operator that the rate of recycle of regenerated dolomite and/or the rate of addition of make-up dolomite should be increased. In a like manner the SO_2 monitor could be equipped with a low level alarm to prevent excessive use of dolomite.

2. START-UP SYSTEM

PRESSURIZED FLUID BED BOILER

START-UP SYSTEM

1. GENERAL

The start-up and placing in operation of a steam turbine generator requires low pressure and low temperature steam. Since the once-through pressurized fluid bed boiler cannot operate at low pressure, a special start-up system must be provided. The capacity of the start-up system should be capable of bringing the steam turbine generator up to and handling 10% load. Included in the start-up system are all the necessary valves, controls and a flash tank which is utilized for providing the low pressure required.

2. FLASH TANK OPERATION

During initial firing of the boiler either from a cold start or a hot restart, Valve A is modulated to maintain a feedwater rate of approximately 25%, Valve C is modulated to maintain required pressure in the evaporator bed. Since the outlet from the boiler has a high percentage of water, the flash tank will be flooded and the pressure low because Valve 2 is open. During firing, the water warms and steam flashes across valves C and D and accumulates in the flash tank. The level of water goes down causing Valve 2 to close. Closing Valve 2 increases the flash tank pressure, following the saturated water temperature into the tank. The flash tank provides the storage facility where the high water is separated from the fluid entering the tank and settles to the bottom. The steam raises to the top of the tank.

The flash tank level control regulates the opening of valves 1 and 2 putting steam and water to the deaerator. If the deaerator cannot absorb all the water and steam, the excess is passed through Valve 1. When the flash tank pressure reaches approximately 400 psi, Valve G is opened and steam admitted to the second superheater bed. Valve I is opened and the steam lines to the turbine warmed. At 600 psi and 100°F superheat the turbine stop valve is opened and the turbine rolled and brought up to speed. At 900 psi the turbine is synchronized and load is picked up. After the turbine is synchronized the steam temperature to the turbine is approximately 700°F. Steam flow to the reheater bed is established, Valve J is closed and Valve K is opened.

3. INCREASING PRESSURE, TEMPERATURE AND LOADING

Now that a complete steam cycle has been established the firing rate of the boiler may be increased to provide the higher temperature and pressure required to sustain operations. The governor valve of the main turbine is set at a fixed position 25%. Valve F is opened and valves E and G are closed, passing all the feedwater into the turbine as steam through the second superheater bed. The feedwater flow rate is maintained at about 25% and the firing rate increased to approximately 100% increasing temperature and pressure. When the pressure reaches 2400 psi, Valve C which has been regulating the furnace will be wide open. The boiler is now up to full pressure and temperature.

3. EQUIPMENT LIST

united engineers
& constructors inc.
a subsidiary of Raytheon Company

635 MW FLUIDIZED BED BOILER

COMBINED CYCLE PLANT

Account No.

TURBINE GENERATOR & AUXILIARIES

a)	Turbine	538 MW turbine; tandem compound four flow exhaust; 25 inch last stage blades, rating at 1-1/2 inch Hg A; 3600 rpm. Turbine throttle conditions 2400 psig at 1000°F and reheat conditions 500 psia.
b)	Generator	597, 778 KVA; 3600 rpm, 3 phase, 22kv, hydrogen cooled at 75 psig.
c)	Exciter	Director connected rotating rectifier brushless exciter.
d)	Lubricating Oil Conditioning Equipment	One continuous by-pass lube oil conditioner. 2000 GPH capacity; with clean and dirty oil storage tanks, circulation pump and transfer and clean-up pump.

HEAT REJECTION SYSTEMS

a)	Circulating	Two vertical motor driven pumps each to deliver 162,000 gpm @ 21 ft. TDH, 89% efficiency, 250 rpm, 965 BHP, with 1250 hp, 600 rpm, 4160 volts, 3 phase, 60 cycle.

Account No.

b) Traveling Screens — Two 12 ft. wide by 43 ft. high screens, traveling at 10 fpm. Each screen has 49 baskets, 24 inches high and 8 ft. long, covered with .080 dia. Type 304 stainless steel wire cloth with 3/8 gpm at a velocity of 1.0 fps to a circulating water pump and is cleaned by a spray system. One 4 ft. wide screen with same construction that passes 10,000 gpm to service water pumps at a velocity of 1.00 fps.

c) Chlorination System — Storage tanks, pumps, control devices and diffusers provided to inject liquid sodium hypochlorite into the river water as it leaves the traveling screens to prevent the formation of algae and slime in the circulating and service water systems.

d) Variable Weir — A variable weir maintains a constant discharge channel water level to allow the recovery of a substantial portion of the circulating pumps discharge head by virtue of the syphon effect. This is accomplished by the automatic positioning of two nested submerged movable gates, installed in the discharge channel.

Account No.

CONDENSING SYSTEMS

a) Condensers — Two main surface condensers single stage, single pass with fabricated steel water boxes and steel shell. Each with condensing surface of 132,000 square feet, 10,100 tubes, 1 inch, 22 BWG, 50 ft. long A-249 welded Type 304 stainless steel. 162,000 gpm cooling water required at 57°F with 90% tube cleanliness factor, 18.48°F temperature rise, tube velocity 8 ft./sec. and 3.1" Hg absolute exhaust pressure.

b) Condenser Air Removal Equipment — Two Condenser Vacuum Pumps cast iron construction, fitted with a steel shaft and directly connected through a Fasts gear type flexible coupling to a 125 hp motor. Rated holding capacity at 1.5" Hg A is 24 scfm and hogging capacity at 15" Hg A is 800 scfm. 125 hp induction motor, 600 rpm, 480 volt, 60 cycles.

c) Condensate Pumps — Two condensate pumps rated at 3,100 gpm, 350 ft. TDH, 86% efficiency at 1185 rpm, driven by 400 hp, 1200 rpm, 4160 volts, 3 phase, 60 cycle.

FEEDWATER HEATING SYSTEM

a) Feedwater Heaters — Two 1,400 sq. ft. horizontal feedwater heaters with 5°F terminal difference each transferring

Account No.

a) (Cont'd) No. 1 Low Pressure Heaters	14.20×10^6 Btu/hr; 50 psig shell design, 750 psig tube design. Each with 26" ID steel shell, steel tube channel, plates and baffles; A-249 - Tp 304 stainless steel tubes 3/4" OD, No. 22 BWG with effective length of 30 ft.
No. 2 Low Pressure Heaters	Two 5,860 sq. ft. horizontal feedwater heaters with 10°F approach and 5°F terminal difference each transferring 55.4×10^6 Btu/hr; 50 psig shell design, 750 psig tube design. Each with 39 inch ID steel shell, steel tube channel, plates and baffles; A-249 - Tp 304 stainless steel tubes 3/4" OD, No. 22 BWG with effective length of 30 ft.
No. 3 Low Pressure Heaters	Two 4,910 sq. ft. horizontal feedwater heaters with 10°F approach and 5°F terminal difference each transferring 49.1×10^6 Btu/hr; 50 psig shell design, 750 psig tube design. Each with 38 inch ID steel shell, steel tube channel, plates and baffles; A-249 - Tp 304 stainless steel tubes 3/4" OD, No. 18 BWG with effective length of 30 ft.
No. 4 Low Pressure Heaters	Two 6,900 sq. ft. horizontal feedwater heaters with 10°F approach and 5°F terminal difference each transferring 64.1×10^6 Btu/hr; 100 psig

Account No.

a) (Cont'd)	shell design, 700 psig tube design. Each with 32 inch ID steel shell, steel tube channels, plates and baffles; A-249 - Tp 304 stainless steel tubes, 3/4" OD No. 22 BWG with effective length of 30 ft.
No. 5 Low Pressure Deaerator-Heater	One full size, horizontal, 8 ft. dia. and 35 ft. long deaerator; 12 ft. dia. and 60 ft. long storage tank. Feedwater leaving deaerator: 2,317,487 #/hr. @-358°F. Condensate entering deaerator 2,197,050 #/hr. @ 289°F. Extraction steam entering deaerator: 121,000 #/hr. @ 1365.0 Btu/# and 148 psia. Design pressure 250 psig to 29 inches vacuum.
No. 6 High Pressure Heaters	Two 8,200 sq. ft. horizontal feedwater heaters with 10°F approach and 0°F terminal difference, each transferring 63.5 x 10^6 Btu/hr; 450 psig shell design, 4000 psig tube design. Each with 40 inch ID steel shell, steel tube channels, plates and baffles; A-249 - Tp 304 stainless steel 3/4" OD, No. 14 BWG tubes with effective length of 30 ft.
No. 7 High Pressure Heaters	Two 6,530 sq. ft. horizontal feedwater heaters with 10°F approach and 0°F terminal difference, each transferring 79.6 x 10^6 Btu/hr; 800 psig

Account No.

a) (Cont'd) shell design and 4000 psig tube design. Each with 40 inch ID steel shell, steel tube channels, plates and baffles; A-249 - Tp 304 stainless steel 3/4" OD, No. 14 BWG tubes with effective length of 30 ft.

Stack Gas Coolers

Two (2) Stack Gas Coolers arranged in parallel with the high pressure heaters No. 6 and 7. Each stack gas cooler consists of three modules. The design of the modules is as follows:

Module #1 - Shell: 3/4" Carbon Steel Plate, insulated with 2-1/2" thermosbestos block, lagged with .060" thick carbon steel lagging; headers = six (67, each 10" long (4 internal, 2 external 8" sch. 120 A106 Gr. C;

Modules 2 & 3 Shell: 3/4" C.S. plate insulated with 2-1/2" thermosbestos block lagged with .060" thick Carbon Steel.

Headers: Twelve (12) headers each 10" long (8 internal, 4 external) 8" sch. 120 A106 Gr. C

Account No.

a) (Cont'd)

Tubes: 2.0" OD, 30' long, 0.259 min. wall A192

Fins: 1.0" high, .0478" thick, 0.156" reg. width.

b) Boiler Feed Pumps & Drive Turbines

Pump - two one-half size horizontal, high speed, multistage, double case barrel type, centrifugal pumps each delivering 1,750,000 #/hr. at 7350 ft. TDH with a suction temperature of 358°F.

Drive Turbine - each pump directly driven by a turbine rated at 7,730 hp and 5,350 rpm. Turbine operates at normal loads with cross-over steam and steam direct from steam generators during start-up and reduced load operation.

c) Pumps

Two horizontal condensate booster pumps each delivering 3200 gpm at 462' TDH with a suction temperature of 101.4°F. Each driven by 600 hp, 4160 V, three phase, 60 cycle, 3550 rpm motor.

d) Condensate Transfer Pumps

One condensate transfer pump delivers 1500 gpm at 400' TDH and 70°F temperature. Driven by a 250 hp, 3550 rpm, 480 V, three phase 60 cycle motor.

Account No.

e)	Stack Gas Cooler Boiler Feed Pumps	Two Stack Gas Cooler boiler feed pumps each driven by a 3000 hp motor to deliver 650,000 #/hr of feedwater at 7350 ft. TDH with suction temperature of 226°F.

STEAM GENERATORS & AUXILIARIES

a)	Steam Generators	Four, once through, single reheat type, pressurized fluidized bed boiler modules, to supply steam to the turbine generator unit. Each designed for continuous operation at 875,000 #/hr, 2400 psig and 1000°F at superheater outlet and 800,000 #/hr, 580 psia at the reheater outlet; maximum allowable working pressure of 3000 psig. The units are arranged for firing bituminous coal having a higher heating value of 13,000 Btu/#.

COAL HANDLING AND FEEDING SYSTEM

a)	System Description	The coal is transported from the mine in specially constructed cars assembled into a unit train comprised of high capacity, semi-automatic bottom-dump hopper cars providing 10,000 tons each shipment; two shipments per week. The rail car receiving hopper will have a minimum of 400 tons capacity to provide storage

Account No.

a) (Cont'd)

volume for controlling unloading of moving cars. The coal discharge from the receiving hopper will be controlled by four reciprocating feeders, F1-A, F1-B & F1-C, F1-D, each feeder nominally rated at 500 tph but capable of delivering 700 tph if one feeder is out of service.

A 54" wide inclined belt conveyor No. 1 will transport the coal at a rate of 2000 tph to the silo.

Conveyor No. 1 will be equipped with a tramp metal detector and belt scale for weighing and recording coal quantities delivered to the plant site.

A 12,000 ton silo will nominally contain an entire unit train delivery without exposure to wind blown dust and provide active storage for plant feed without the use of mobile equipment.

Excess coal will overflow the silo to an initial pile for placing in permanent storage.

During extended periods of no-train delivery, coal will be recovered from permanent storage through a reclaim hopper and Feeder No. F2 delivering to Conveyor No. 1 for refilling the silo.

Account No.

a) (Cont'd)

Coal will be fed from the silo by seven Feeders No. F3A to F3G, each rated a maximum of 150 tph and operating in timed sequence for uniform drawndown of the silo, onto a 24" wide belt Conveyor No. 2.

Conveyor No. 2 will be fitted with a tramp iron magnet before delivery to the coal dryer. The McNally Flowdryer is sized to evaporate 12 tons per hour of water from a feed of 150 tph (dry basis) of 1-1/2" X 0 coal and deliver a product containing 3% total moisture. The dryer will be fueled by pulverized dried coal and will be fitted with a cyclone dust collector to capture the coarse dusts and a high energy scrubber to reduce exhaust dust to acceptable limits. Effluent from the scrubber will be disposed to the pcwer plant ash pond.

Coal from the dryer will be crushed to 1/4" X 0 size in a reversible hammer mill and, along with the coarse dust from the dryer cyclone, will be delivered via 24" belt Conveyor No. 3 to the plant surge bin. Conveyor No. 3 will be fitted with belt scale No. 2 to weigh and record coal quantities to the boilers.

Account No.

a) (Cont'd) From the 150 ton capacity plant surge bin the coal flow will be split into two streams each served by a 0 to 75 tph vibrating Feeder F4-A and F4-B to its associated coal feeder surge silo group through scraper Conveyors No. 4A and 4B.

The silo fueling system will be provided with automatic sequential controls with provisions for operator to override to accommodate unusual operating conditions. The plant surge bin level will control the feed to the dryer and in turn the dryer surge bin will regulate the silo withdrawal feeders.

b) Equipment

a) The coal handling system will consist of the following equipment, material and services:

1. Receiving Hopper
2. Feeders No. F1A, F1B, F1C & F1D
3. Chutes
4. Conveyor No. 1
5. Metal Detector
6. Belt Scale No. 1
7. Reclaim Hopper
8. Shut Off Gate
9. Feeder No. F2

Account No.

b) (Cont'd)

a)

10. Silo
11. Feeders No. F3A-F3G
12. Conveyor No. 2
13. Tramp Iron Magnet
14. McNally Flowdryer
15. Mill
16. Conveyor No. 3
17. Belt Scale No. 2
18. Surge Bin
19. Feeders No. F4A & F4B
20. Conveyors No. 4A & 4B
21. Dust Collecting Equipment
22. Ventilating & Heating Equipment
23. Control Equipment
24. Motors & Controls
25. Structures
26. Foundations
27. Electric Wiring
28. Erection

b) The associated coal feeder surge silo group will consist of the following:

1. Screens
2. Cyclones

Account No.

b) (Cont'd)

b) 3. Surge bins

4. Storage injectors

5. Primary injector

6. Booster air compressor

CLOSED WATER COOLING SYSTEM

The closed cooling water system provides water to the sample coils, pump glands and motor bearings, oil coolers, turbine generator oil and hydrogen coolers, boiler feed pump turbine oil coolers, air compressor coolers, instrument and station air after-coolers, condenser vacuum pumps, air conditioners, etc. It includes three pumps @ 4000 gpm capacity with a total head of 50 psi, each driven by a 200 hp, 1800 rpm, 480 volt, 3 phase, 60 cycle motor, two horizontal single pass heat exchangers 8000 sq. ft. transferring 28 x 10^6 Btu/hr., 150 psig shell and tube design pressure.

WATER TREATMENT SYSTEM

a) Raw Water Treating System

Water is supplied to the raw water treating system by the low pressure water pumps. After softening and coagulation (by reactivators) and filtration (by gravity filters) the water is pumped to the treated water storage tank.

Account No.

a) (Cont'd)

The raw water treating system is sized to handle a capacity of 500 gpm and includes the following components:

1) One reactivator flow split box.
2) Two reactivators with integral cleanwells (each 250 gpm capacity and 15,000 gallons clearwell capacity).
3) One ferric sulphate feeder.
4) One ferric sulphate solution pump.
5) Two lime feeders.
6) Two lime solution pumps.
7) One chemical solution flow split box.
8) Two gravity filters of 250 gpm capacity each, 10 feet dia. x 15 feet high.
9) One local control panel with annunciator, flow recorder, indicator, timers, control switches, level indicator gauge, etc.
10) Two treated water pumps each of 250 GPM capacity and a total head of 40 psi driven by a 7.5 hp motor, 1770 rpm, 3 phase, 60 cycle, 480 volts.
11) 120,000 gallon carbon steel treated water storage tank, 27 feet dia. x 30 feet high.

Account No.

b) Demineralizer System

A header from the treated water storage tank supplies the demineralizer feed pumps and in turn supply the demineralizers and other services requiring treated water. The demineralizer system includes the following components:

1) Two demineralizer feed pumps each with a capacity of 250 gpm and a total head of 91 psi, driven by a 20 hp, 3550 rpm, 3 phase, 60 cycle, 480 volt motor.

2) One cation exchanger having a capacity of 100 gpm - size: 54" dia. x 8'-0" high.

3) One anion exchanger having a capacity of 160 gpm - size: 54" dia. x 8'-0" high.

4) Two 60 gph acid pumps each driven by a 1/2 hp, 1750 rpm motor.

5) Two 32 gph caustic pumps each driven by 1/2 hp, 1750 rpm motor.

6) Two caustic heat exchangers to heat 8.2 gpm of 4% caustic solution from 40°F to 110°F utilizing 400 lbs./hr of 20 psig steam.

7) Two 50 gpm recycle pumps with a total head of 50 psi driven by a 2 hp, 3600 rpm motor.

8) Controls and instrumentation including a local panel with a service selector, power switch, "Regen. Test" pushbuttons, "Alarm

Account

Account No.

b) (Cont'd)

Silence" pushbuttons for annunciators, and local "start-stop" motor pushbutton stations, etc.

9) One stainless steel basic demineralized water storage tank having a capacity of 250,000 gallons - size 34 ft. dia. x 39 ft. height.

c) Polishing System

The polishing system consists of the following equipment:

1) One 250,000 gallon stainless steel polished deionized water storage tank - size: 34' dia. x 39' height.
2) One precoat tank to prepare a precoat slurry, with agitator.
3) Dust collector to remove resin fines from the air.
4) Three precoat type filter deionizer tanks containing nylon wool filter elements.
5) Precoat pump for transferring precoat material to the filter deionizer tanks.
6) Blower for backwashing filter elements.
7) Control panel for precoat system and filter deionizer tanks.
8) Associated valves.

Account No.

c) (Cont'd) — 9) Control, interlocks & alarms.

10) One motor driven air compressor.

FIRE PROTECTION SYSTEM

The fire protection system is designed to conform to the guide lines set up by NEPIA.

The fire line headers are designed as a loop system to permit water flow in either direction. Sectionalizing valves throughout the system are located to permit isolating damaged sections of header.

The fire protection system includes a network of fire service water piping, a fire service water storage tank, a diesel engine driven fire pump, fire protection systems for fuel oil system, generator hydrogen seal oil unit, turbine oil reservoir and transformers, and hydrants and hose reels located at various places throughout the plant and yard.

a) Fire Service Water Tank — One combination filtered and fire-service water storage tank (above ground) having a capacity of 200,000 gallons.

b) Diesel Engine Driven Fire Pump — The fire pump has a capacity of 2000 gpm at a total head of 150 psi driven by a 250 hp,

Account No.

b)	(Cont'd)	1750 rpm diesel engine. The engine is equipped with a hot start engine preheater, fuel system, muffler, 350 gallon fuel tank, batteries and battery chargers, and combined manual-automatic controller.
c)	Motor Driven Fire Pumps	Two motor driven fire pumps each having 2000 gpm capacity at 150 psi total head driven by a 250 hp, 1750 motor.
d)	Motor Driven "Jockey" Pump	One jockey pump having a capacity of 100 gpm. This pump starts when the system pressure drops to 135 psig.
e)	Fire Protection Systems	Water spray, automatically operated and remote-manual with automatic detection. Fire protection includes automatic water spray systems with automatic fire detection, detection and alarm panels and necessary piping, fittings, flanges and valves.
f)	Hydrants and Hose Reels	The fire hydrants have a 5" main valve opening. The hose reel assemblies are continuous flow units provided with two 50' lengths of 1-1/2" hose.

OTHER MECHANICAL EQUIPMENT

a)	Gas Turbine Generators	Two Gas Turbine Generators each rated at 56,556 KW at 3600 rpm.

Account No.

a)	(Cont'd)	The generators are supplied with pressurized flue gas connections and a starting motor rated at 800 hp.
b)	Emergency Gas Turbine Generator	One 22 MW emergency gas turbine generator.
c)	Emergency Diesel Generators	Two emergency diesel generators capable of providing the power for one set of safeguards equipment. Each generator is rated at 350 KW at 1800 rpm. The diesel generators are supplied with a 2000 gallon underground day tank, fuel pumps, lube oil system, engine cooling system and starting system.
d)	Service Water Pumps	Two vertical service water pumps each to deliver 5000 gpm at 220 ft. TDH; 85% efficiency, driven by a 350 hp, 60 cycle, 3 phase, 480 volt motor at 1770 rpm. Each with steel column and discharge tee, chrome steel shaft, cast iron bowls, bronze impellers and bearings.
e)	Instrument Air Compressor	One horizontal, motor driven single-stage, double acting, non-lubricated compressor, including intake filter and silencer, after-cooler, air dryers, dual control, receiver and 60 hp motor with V-belt drive; rated 215 scfm at 100 psig discharge pressure.

Account No.

f) Station Air Compressors — Two angle type, motor-driven, two-stage, double acting, lubricated compressors including intake filter, 100 hp motors, inter-cooler and after-cooler, dual control and receiver; rated 320 scfm at 125 psig discharge pressure.

g) Booster Air Compressors — Two booster air compressors each rated 13,000 cfm at 50 psig total head; suction pressure: 150 psig; suction temperature: 636°F.

h) Crane — Turbine Building - one (1) 60 ton overhead traveling crane with 20 ton auxiliary hook. Main hook 80 ft. lift, auxiliary hook 80 ft. lift.

Main hoist speed 4 fpm, auxiliary hoist speed 15 fpm, bridge speed 75 fpm, trolley speed 50 fpm.

OTHER MECHANICAL EQUIPMENT

CO_2 Scrubber — One CO_2 scrubber rated 7,000,000 ncfh of flue gas at 135 psia and 230°F. The CO_2 scrubber consists of two 12 ft. diameter and 100 ft. high columns and one hot carbonate regenerator 23 ft. diameter by 100 ft. high producing a gas mixture of 1,080,000 ndfh CO_2 and 2,390,000 ncfh steam at 19.7 psia and 220°F; one reboiler requiring 160 MM Btu/hr; horsepower required for the various equipment is 1100.

Account No.

Claus Plant

One "package" Claus plant (manufactured by Ford Bacon and Davis, Texas) to produce 200 tons per day sulfur.

Gas from the dolomite regenerator system (H_2S. generator vessel) at the rate of 117,000 #/hr is expanded to 15 psig and cooled to condense out water vapor. The process involves combustion of the H_2S to SO_2 and subsequent reaction between the remaining H_2S and the produced SO_2 to form sulfur and water.

Dolomite Regenerator System

One dolomite regenerator system consisting of the following:

Reducer vessel: 9' O.D. x 30' high.

Gas Producer vessel: 9' O.D. x 50' high.

H_2S Generator Vessel:

bottom portion = 7-1/2' O.D. x 25' high.

top portion = 11-1/2' O.D. x 25' high.

Cyclone Pressure Vessel: 5' O.D. x 22' high.

Description:

The dolomite regenerator system receives sulfated dolomite ([$CaSO_4$ + MgO] and [Cao + MgO]) from the pressurized fluid bed boiler modules and converts ~ 90% of the sulfated stone to half calcined dolomite ([$CaCO_3$ + MgO]) for recycle back

Account No.

Dolomite Regenerator System

to the boiler modules. A hydrogen sulfide rich gas is produced and sent to a Claus Plant where elemental sulfur is produced.

Spent dolomite is pneumatically conveyed to the regenerator system from four separate FBB discharge holding vessels (~ 50 tph each).

One holding vessel serves all 5 beds in a boiler module.

Spent stone first enters a reducer vessel in which the calcium sulfate, reacted with a reducing gas, is converted to calcium sulfide. The reducer vessel, 9' O.D. x 30' high, is a fluid bed reactor operating at 1500°F and 8 atms. A portion of the calcium oxide carried with the spent dolomite is carbonated to half-calcined dolomite in the reducer vessel. Ten percent of the stone charged to the regenerator system is carried overhead from the vessels or removed from the reducer vessel and sent to waste. Top gas from the reducer vessel passes through a turbine expander, which drives a compressor to compress the gas to the H_2S generator vessel.

Account No.

Dolomite Regenerator System

Reducing gas for the reducer vessel is made in the gas producer vessel, a 9' O.D. x 50' high fluid bed gasifier operating at 1800°F and 9 atm. Approximately 55% stoichiometric air containing 27 gr/scf of moisture is used to combust 17 tph coal, producing 221,000 #/hr reducing gas containing 8.0% H_2 and 16.3% CO. (% by volume.)

Reduced stone, 400,000 #/hr of [CaS + MgO], [$CaCO_3$ + MgO] and [CaO + MgO], is gravity fed from the reducer vessel to the H_2S generator vessel through a dip leg pressure seal. The H_2S generator vessel is also a fluid bed reactor with an expansion section on top enclosing a solid separation cyclone. The bottom portion of this vessel is 7-1/2' O.D. x 25' high and the top expansion section is 11-1/2' O.D. x 25' high. The H_2S generator operates at 1100°F, 12 atms and produces a 10% H_2S gas stream by volume which is sent to the Claus Plant. Twenty, 20, separate solids removal ports are provided to pneumatically transport regenerated dolomite back to each FBB module. Feed gas for the H_2S generator, 180,000 #/hr, 64% H_2O, 36% CO_2 (% by volume) is made in the CO_2 scrubber system. This gas reacts with the reduced stone to produce the H_2S rich gas and the regenerated stone [$CaCO_3$ + MgO].

EQUIPMENT LIST - ELECTRICAL

& INSTRUMENTATION

Account No.

MAIN POWER TRANSFORMERS

One - 600 mVA transformer, oil immersed, 55°C rise, 3 phase, 60 Hz, FOA, 22 kV-345 kV, delta-wye, 1050 kV - H.V. BIL, no-load tap-changer, four - 2-1/2% H.V. taps, liquid level guage, dial type liquid thermometer, vacuum pressure guage, tank pressure relief device, tank grounding provisions, valves, three 312 kV lightning arresters, six-bushing C.T.'s, one winding temperature relay, three-L.V. bushing enclosures, one sudden pressure relay.

One 115 mVA transformer, oil immersed 55°C rise, 3 phase, 60 Hz, FOA, 22 kV-138 kV, delta-wye, 550 kV - H.V. BIL, no-load tap-changer, four - 2-1/2% H.V. taps, liquid level guage, dial type liquid thermometer, vacuum pressure guage, tank pressure relief device, tank grounding provisions, valves, three 144 kV lightning arresters, six-bushing C.T.'s, one winding temperature relay, three L.V. bushing enclosures one sudden pressure relay.

MAIN GENERATOR BUS

An isolated phase bus is run between the main generator line terminals and the low voltage bushings of the main 600 mVA step-up power transformer. The isolated phase bus is metalclad,

Account No.

MAIN GENERATOR BUS (Cont'd)

forced cooled, continuous housing, 3 phase, 60 cycle, 23 kV, 150 kV basic impulse level, 17,000 ampere, 65°C temperature rise with forced air cooling.

An isolated phase bus is run between the gas turbine driven generators and the low voltage busings of the main 115 mVA step-up power transformer. The isolated phase bus is metal-clad, forced cooled, continuous housing, 3 phase, 60 cycle, 23 kV, 150 kV basic impulse level 3750 Amperes from each generator to connecting tap, and 7500 Amperes from tap to transformer, 65°C temperature rise with forced air cooling.

NEUTRAL TRANSFORMER

Each generator neutral is grounded through a single phase transformer rated at 20,000 volts on the primary and 240 volts on the secondary. The secondary of the transformer is connected to a resistor bank.

STATION START-UP TRANSFORMERS

One 20 mVA transformer, oil immersed, 55°C rise, 3 phase, 60 Hz, FOA, 138 kV - 6.9 kV, delta-wye, 550 kV - H.V. BIL, no-load tap-changer, four - 2-1/2% H.V. taps, liquid level gauge, dial type liquid thermometer, vacuum pressure gauge, tank pressure relief device, tank grounding provisions,

Account No.

STATION START-UP TRANSFORMERS (Cont'd)

valves, three 144 kV lightning arresters, six-bushing C.T.'s, one winding temperature relay, three-L.V. bushing enclosures, one sudden pressure relay.

One 20 mVA transformer, oil immersed, 55°C rise, 3 phase, 60 Hz, FOA, 22 kV - 6.9 kV, delta-wye, 150 kV - H.V. BIL, no-load tap-changer, four - 2-1/2% H.V. taps, liquid level gauge, dial type liquid thermometer, vacuum pressure gauge, tank pressure relief device, tank grounding provisions, valves, winding temperature relay, H.V. bushing enclosures, L.V. cable entrance, one sudden pressure relay.

AUXILIARY POWER TRANSFORMER

One 20 mVA transformer, oil immersed 55°C rise, 3 phase, 60 Hz, FOA, 22 kV - 6.9 kV, delta-wye, 150 kV - H.V. BIL, no load tap-changer, four 2-1/2% H.V. taps, liquid level gauge, dial type liquid thermometer, vacuum pressure gauge, tank pressure relief device, tank grounding provisions, valves, winding temperature relay, H.V. bushing enclosures, L.V. cable entrance, one sudden pressure relay.

4160 VOLT INDOOR METALCLAD SWITCHGEAR

The 4160 volt, 3 phase, 60 Hz indoor metalclad switchgear is arranged into two sections each consisting of:

Three - 2000 ampere, incoming line circuit breaker units

Account No.

4160 VOLT INDOOR METALCLAD SWITCHGEAR (Cont'd)

each with voltmeter, ammeter, ammeter switch, breaker control switch with indicating lights, two drawout type potential transformers, six current transformers and relaying.

Six 1200 ampere, motor feeder circuit breaker units each with ammeter, ammeter switch, breaker control switch with indicating lights, six current transformers and relaying. Two units include three motor differential relays.

Ten 1200 ampere, transformer feeder circuit breaker units each with ammeter, ammeter switch, breaker control switch with indicating lights, six current transformers and relaying.

STATION SERVICE TRANSFORMERS

Ten 1500/2000 kVA transformers, dry type, 3 phase, 60 Hz, AA/FA, 6900 volt - 480 volt, delta-wye, 35 kV H.V. BIL, 10 kV L.V. BIL, with two 2-1/2% taps AN and BN.

480 VOLT METALCLAD SWITCHGEAR

The 480 volt metalclad switchgear is rated 600 volt, with 2000 ampere busses arranged in sections. Each section is supplied thru a 2000 ampere, 75,000 ampere i.c. incoming line circuit breaker from one of the 6900 volt - 480 volt Station Service Transformers. All feeder circuit breakers have an interrupting capacity of 50,000 amperes.

Account No.

MOTOR CONTROL CENTERS

Eleven 480 volt motor control centers are located at various points of electrical load concentration. Each motor control center compartment is provided with circuit breaker, motor starter, overload relays, control power transformers and auxiliary relays as required by the individual auxiliaries. Reversing starters are equipped with electrical and mechanical interlocks.

MAIN CONTROL ROOM EQUIPMENT

The main control room contains the following supervisory and control panels for control and operation of the steam generators, gas turbine generators and boilers. The panels are complete with required instrumentation and control devices:

Main Control Benchboard

Steam Turbine & Generator Control Panels

Gas Turbine & Generator Control Panels

Start-up Turbine & Generator Control Panels

Boiler Control Panels

Coal Handling Control Panel

Supervisory Control Board

BOILER INSTRUMENTATION & CONTROL

EQUIPMENT LIST

SERVICE	INSTRUMENT	QUANTITY	COMMODITY
Main Steam Output	Recorder	1/Boiler	Steam - Flow Steam - Pressure Steam - Temperature
Reheat Steam Flow (Input & Output to Reheat Bed)	Recorder	2/Boiler	Steam - Flow Steam - Pressure Steam - Temperature Integrated - Steam Flow
Boiler Feedwater Consumption	Recorder	1/Boiler	Water - Flow Water - Pressure Water - Temperature
Boiler Combustion Air Flow	Recorder	1/Boiler	Air - Flow Air - Pressure Air - Temperature
Bed Combustion Air Flow	Recorder	1/Bed	Comb. Air Flow Coal FDR Transport Air Flow Reg. Dolomite Transport Air Flow
Bed Fuel	Recorder	1/Bed	Coal Flow Rec. Dolomite Flow Fuel Oil Flow
Bed. Pres. & Temp.	Recorder	1/Bed	Bed - Diff. Press. Bed - Height Bed - Temp. (Lower Zone) Bed - Temp. (Upper Zone)
Flue Gas Output	Recorder	1/Boiler 1/CBC	Flue Gas - Flow Flue Gas - Pressure Flue Gas - Temperature
SO_2 Content	Analyzer	1/Boiler	SO_2
Bed. Temperature	Monitor	1/Bed	Bed Temp. - 9 Thermocouples Tube Temp. - 9 Rtd's Shell Temp. - 6 Rtd's

SERVICE	INSTRUMENT	QUANTITY	COMMODITY
Main Steam	Controller	1/Boiler	Steam Pressure
Bed Steam	Controller	1/Bed	Steam Pressure
	Load Station	1/Bed	
Boiler Combustion Air	Transmitter	1/Boiler	Combustion Air Flow
	Controller	1/Boiler	
	Control Valve	1/Boiler	
Bed Combustion Air	Transmitter	1/Bed	Combustion Air Flow
	Controller	1/Bed	
	Damper Station	1/Bed	
	Damper Positioner	1/Bed	
Bed Fuel	Transmitter	1/Bed	Coal Flow
	Transmitter	1/Bed	Dolomite Reg. Flow
	Controller	1/Bed	Coal Feeder
	Control Valve	1/Bed	Coal Transport Air
Main Steam	Recorder - Transmitter	1/Boiler	Steam Flow
	Valve Station	1/Boiler	Feedwater Control
	Control Valve	1/Boiler	

Account No.

COMPUTER

One Westinghouse PRODAC 250 process computer system.

STATION BATTERIES

Two station batteries rated 125 V d-c, 60 cell, lead acid type 480 ampere hour at 8 hour rate.

BATTERY CHARGER

Two static rectifier type battery chargers rated at 15 kW each, 100 d-c amp output, 140 V d-c output.

Account No.

POWER INVERTERS

Two 10 kW static type power inverters for instrument control busses.

DISTRIBUTION PANELS

Two 125 V d-c power panels 200 ampere main bus with branch circuit breakers rated at 10,000 amperes i.c.

Three 120/208 volt, 3 phase, 4 wire instrument panels each with mechanically interlocked 100 ampere main breakers and 30 branch circuit breakers.

Ten 120/208 volt, 3 phase, 4 wire lighting panelboards. Two 480 volt, 3 phase, 3 wire power distribution panelboards.

LOCAL CONTROL STATIONS

Approximately 50 miscellaneous local control stations consisting of essentially NEMA 1 enclosures containing: pushbuttons, control switches, control relays, motor starters, terminal blocks, etc.

PLANT COMMUNICATIONS SYSTEM

One plant communication system complete with speakers, handsets, amplifiers and communications console located in the control room.

Account No.

LIGHTNING PROTECTION MAST

Lightning protection masts complete with air terminal and baseplates.

CATHODIC PROTECTION SYSTEM

Cathodic protection is provided for underground piping and other vulnerable plant equipment in the vicinity of the river.

4. ALTERNATE HEAT REJECTION SYSTEMS

FLUIDIZED BED BOILER COMBINED CYCLE PLANT
ALTERNATE HEAT REJECTION SYSTEMS

1. INTRODUCTION

This report presents the design and economic evaluations for the utilization of natural draft and induced draft cooling towers in lieu of once-through river water cooling for condenser circulating water for the 635 MW Fluidized Bed Boiler Combined Cycle Plant.

It is recognized that optimum conditions have not been realized for each of the alternate systems. The use of cooling towers in lieu of a once-through system would require a complete re-evaluation of the turbine steam, condensing and heat rejection systems. The cold water from the cooling tower will be at a higher temperature than the river water used for once-through cooling. This will result in a higher condenser pressure, affecting the station heat balance.

The site land area is sufficient to incorporate either type of cooling tower within the established boundaries.

The accompanying equipment list and cost estimate show only those accounts affected by the substitution of cooling towers for once-through river water. All other accounts remain as shown in the basic reports.

No evaluations of maintenance or operating costs for induced draft towers are included.

2. COOLING TOWERS

The natural draft towers consist of a concrete hyperbolic shell supported by a foundation ring on soil. The internal fill consists of asbestos-cement board sheets supported by a concrete structure on spread footings.

Induced draft towers consist of a structural framework and fill support system fabricated from treated lumber supported on the cooled water basin. Fill material is fire retardant polyvinylchloride. Fan, gear and motor are supported on a structural steel framework.

The cooled water basin is constructed of reinforced concrete with an overflow sump for removal of blowdown and for draining the tower basin for silt removal. The basin floor is sloped to the de-silting sump mentioned above which is fitted with a sludge valve to allow for removal of silt.

All towers are provided with drift eliminators to reduce entrainment losses to not more than 0.2% of the circulating water flow. Also provided are access stairs and platforms, lightning protection system and a de-icing system.

Cooling tower design conditions assumed for this report were 15° approach, 70° wet bulb and 50% relative humidity. Quantity and resulting heat rise of cooling water are approximately the same as the basic plant with once-through cooling. A greater heat rise would reduce the quantity of water required and permit a more efficient cooling tower design, but the condenser operating pressure would be increased.

3. CONDENSERS

When a cooling tower is used for heat rejection, the condenser will operate at a higher pressure than with a once-through circulating water system using river water. The higher initial temperature of the cooling water to the condenser and the increased temperature rise result in a higher condensing temperature which corresponds to the operating pressure of the condenser.

A condenser pressure of 2.5" Hg. has been assumed. Condensers are two pass design with 46 ft. long tubes. The material selection is identical to the basic plants, and the number of tubes has been adjusted to maintain a water velocity of 8 ft. per sec.

ALTERNATE HEAT REJECTION SYSTEMS

EQUIPMENT LIST - MECHANICAL

POWER PLANT

Account No.

HEAT REJECTION SYSTEMS

a. Circulating Water Pumps	Four horizontal centrifugal pumps, motor driven, each to deliver 80,000 gpm at 80 feet TDH; 90% efficiency; 590 rpm; 1800 BHP. Each with 2000 hp, 6900 volt horizontal induction motor, 3 phase, 60 cycle, 600 rpm.
b. Make-up Pumps	Two vertical sump-type pumps, each to deliver 5000 gpm at 20 ft. TDH, 80% efficiency, driven by 50 hp, 440 volt motor at 1200 rpm. Pumps located in intake structure similar to service water pumps.
Cooling Tower	a. One hyperbolic natural draft cooling tower designed for 70° wet bulb, 15° approach, 18.7° range, 50% R. H. and 320,000 gpm total water flow. Overall dimensions 390 ft. diameter at base, 400 ft. high. or

Account No.

Cooling Tower (con't)	b. Two induced draft cooling tower units, each with 8 cells, designed for 70° wet bulb, 15° approach, 18.7° range and 320,000 gpm total water flow. Unit dimensions: 288 ft. long, 75 ft. wide and 65 ft. high. Total fan hp, 3200.
Traveling Screens	Two screens, one to pass 10,000 gpm to Service Water Pumps and one to pass 10,000 gpm to Circulating Water Make-up Pumps.
Variable Weir	Not required

CONDENSING SYSTEMS

Condensers	Two single stage, two pass surface condensers with fabricated steel water boxes and steel shell. Each with condensing surface of 220,000 sq. ft., 18,300 one inch, 22 BWG, 46 ft. long #304 stainless steel tubes (9150 each pass). Condensers to operate at pressure of 2.5 with 320,000 gpm water from cooling tower (tube velocity, 8 ft/sec).

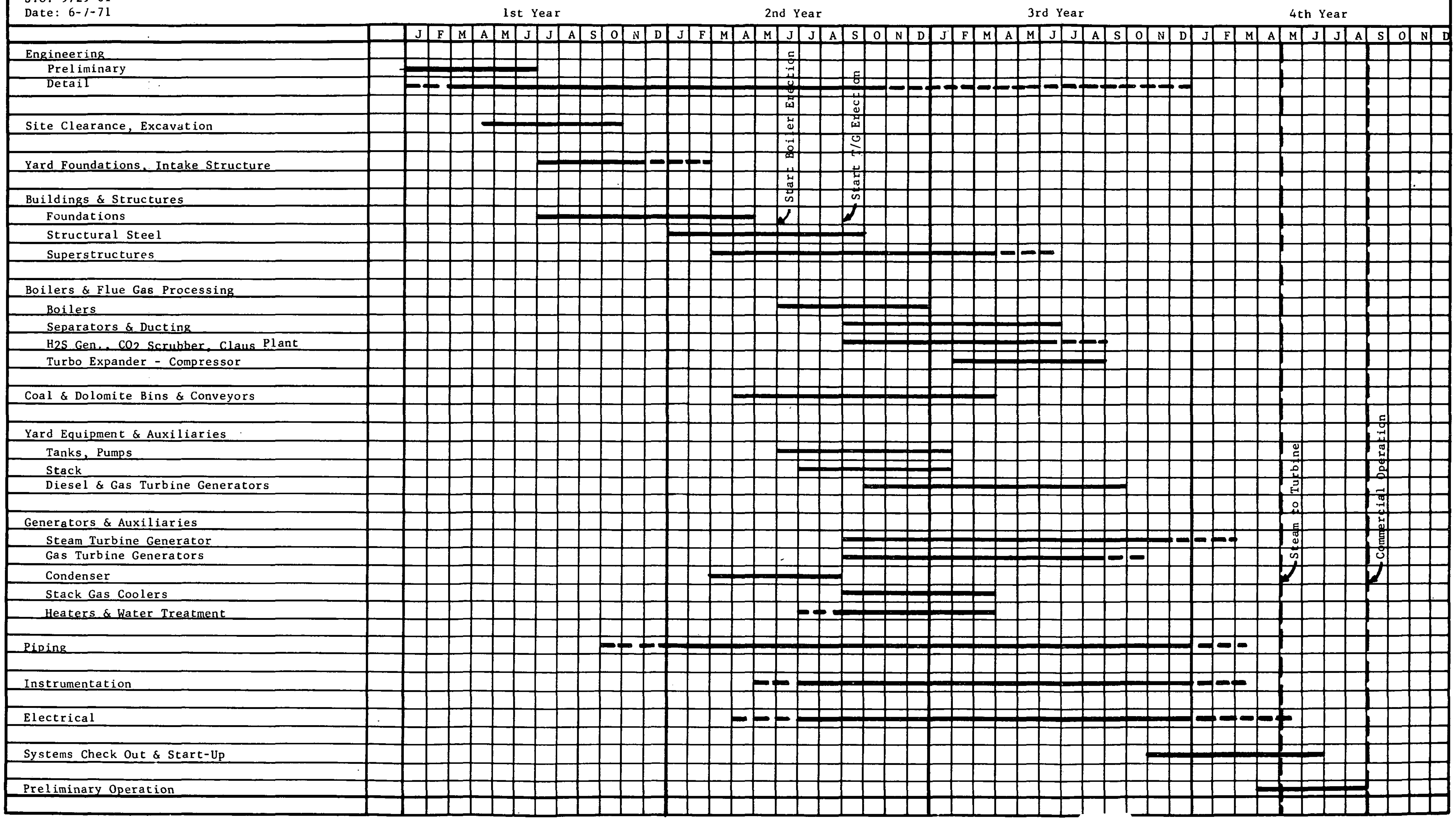
5. CONSTRUCTION SCHEDULE

united engineers & constructors inc.

635 MW Fluidized Bed Boiler
Combined Cycle Plant

Proposed Project Schedule

J.O. 9729-01
Date: 6-/-71



6. COST SUMMARY

635 Mw Fluidized Bed
Boiler Combined Cycle Plant

June 19, 1971

SUMMARY

Account No.	Labor	Material	Subcontract	Total
10. Land and Land Rights				"Not Included"
11. Structures and Improvements				
Yard Work	$ 900,000	$ 550,000	$	$ 1,450,000
Boiler Module Area	840,000	900,000		1,740,000
Turbine Room and Heater Bay	2,465,000	1,885,000		4,350,000
Control Building	420,000	400,000		820,000
Services Building	1,675,000	1,235,000		2,910,000
Administration Building	383,000	407,000		790,000
Intake and Discharge Structures	850,000	470,000		1,320,000
Total Structures and Improvements, -	7,533,000	5,847,000		13,380,000
12. Boiler Plant Equipment				
Steam Generating Equipment	750,000	7,500,000		8,250,000
Draft System	3,030,000	8,180,000		11,210,000
Coal Fuel and Equipment	341,000	709,000	5,900,000	6,950,000
Ash and Dust Handling System	175,000	175,000		350,000
Dolomite Regeneration System	1,769,000	2,361,000	3,920,000	8,050,000
Other Mechanical Equipment	705,000	8,695,000		9,400,000
Instruments and Controls	695,000	1,275,000		1,970,000
Miscellaneous Suspense Items	500,000	100,000		600,000
Total Boiler Plant Equipment, -	7,965,000	28,995,000	9,820,000	46,780,000

SUMMARY (Continued)

Account No.	Labor	Material	Subcontract	Total
14. Turbine Plant Equipment				
Turbine-Generator Equipment	$ 828,000	$11,042,000		$11,870,000
Circulating Water System	1,336,000	1,274,000		2,610,000
Condensing Systems	950,000	2,520,000		3,470,000
Feedwater System	1,285,000	5,070,000		6,355,000
Other Turbine Plant Equipment	1,409,000	1,531,000		2,940,000
Instruments and Controls	200,000	585,000		785,000
Total Turbine Plant Equipment, -	6,008,000	22,022,000		28,030,000
15. Electric Plant Equipment				
Switchgear	135,000	695,000		830,000
Station Service Equipment	410,000	2,800,000		3,210,000
Switchboards	121,000	459,000		580,000
Protective Equipment	150,000	90,000		240,000
Electrical Structures and Wiring Containers	1,196,000	414,000		1,610,000
Power and Control Wiring	1,931,000	869,000		2,800,000
Total Electric Plant Equipment, -	3,943,000	5,327,000		9,270,000
16. Miscellaneous Plant Equipment				
Transportation and Lifting Equipment	50,000	190,000		240,000
Air, Water and Steam Service Systems	757,000	703,000		1,460,000
Communications Equipment	85,000	55,000		140,000
Furnishing and Fixtures	55,000	365,000		420,000
Total Miscellaneous Plant Equipment, -	947,000	1,313,000		2,260,000

SUMMARY (Continued)

Account No.	Labor	Material	Subcontract	Total
53. Station Equipment - Transmission				
Main Power Transformers	$ 51,000	$ 779,000	$9,820,000	$ 830,000
Subtotal, -	26,447,000	64,283,000	9,820,000	100,550,000
91. Undistributed Costs				
Engineering Construction - Management and Field Supervision	2,800,000	7,110,000		9,910,000
Temporary Facilities	1,670,000	750,000		2,420,000
Construction Equipment	375,000	3,665,000		4,040,000
Construction Services	380,000	1,250,000		1,630,000
Total Undistributed Costs, -	5,225,000	12,775,000		18,000,000
Subtotal, -	31,672,000	77,058,000	9,820,000	118,550,000
Other Plant Costs (Unclassified)				
Operator Training		100,000		100,000
Spare Parts		750,000		750,000
Owner's General Office and Administrative Cost		1,000,000		1,000,000
Total Other Plant Costs (Unclassified), -		1,850,000		1,850,000
Subtotal, -	31,672,000	78,908,000	9,820,000	120,400,000
Normal Contingency				7,300,000
Subtotal, -				127,700,000
Escalation 7½% per year (1975 Operation)				23,900,000
Interest during Construction 7½% per year (3-1/2 year schedule)				19,900,000
Total Estimate, -				$171,500,000

635 Mw Fluidized Bed
Boiler Combined Cycle Plant

June 19, 1971

SUMMARY

Account No.	Labor	Material	Subcontract	Total
Alternate I Induced Draft Power				
Net Additional Costs	$825,000	$1,155,000	$1,200,000	$3,180,000
Undistributed Costs				No Change
Other Plant Costs				No Change
Normal Contingency				180,000
Escalation 7-1/2% per year (1975 Operation)				620,000
Interest during Construction 7-1/2% per year (3-1/2 year schedule)				520,000
Total Net Additional Cost Alternate I, -				$4,500,000
Alternate II Natural Draft Tower				
Net Additional Costs	300,000	1,000,000	3,600,000	4,900,000
Undistributed Costs				No Change
Other Plant Costs				No Change
Normal Contingency				300,000
Escalation 7-1/2% per year (1975 Operation)				980,000
Interest during Construction 7-1/2% per year (3-1/2 year schedule)				820,000
Total Net Additional Costs Alternate II, -				$7,000,000

7. PLATES

635 MW FLUIDIZED BED BOILER COMBINED CYCLE PLANT

MECHANICAL

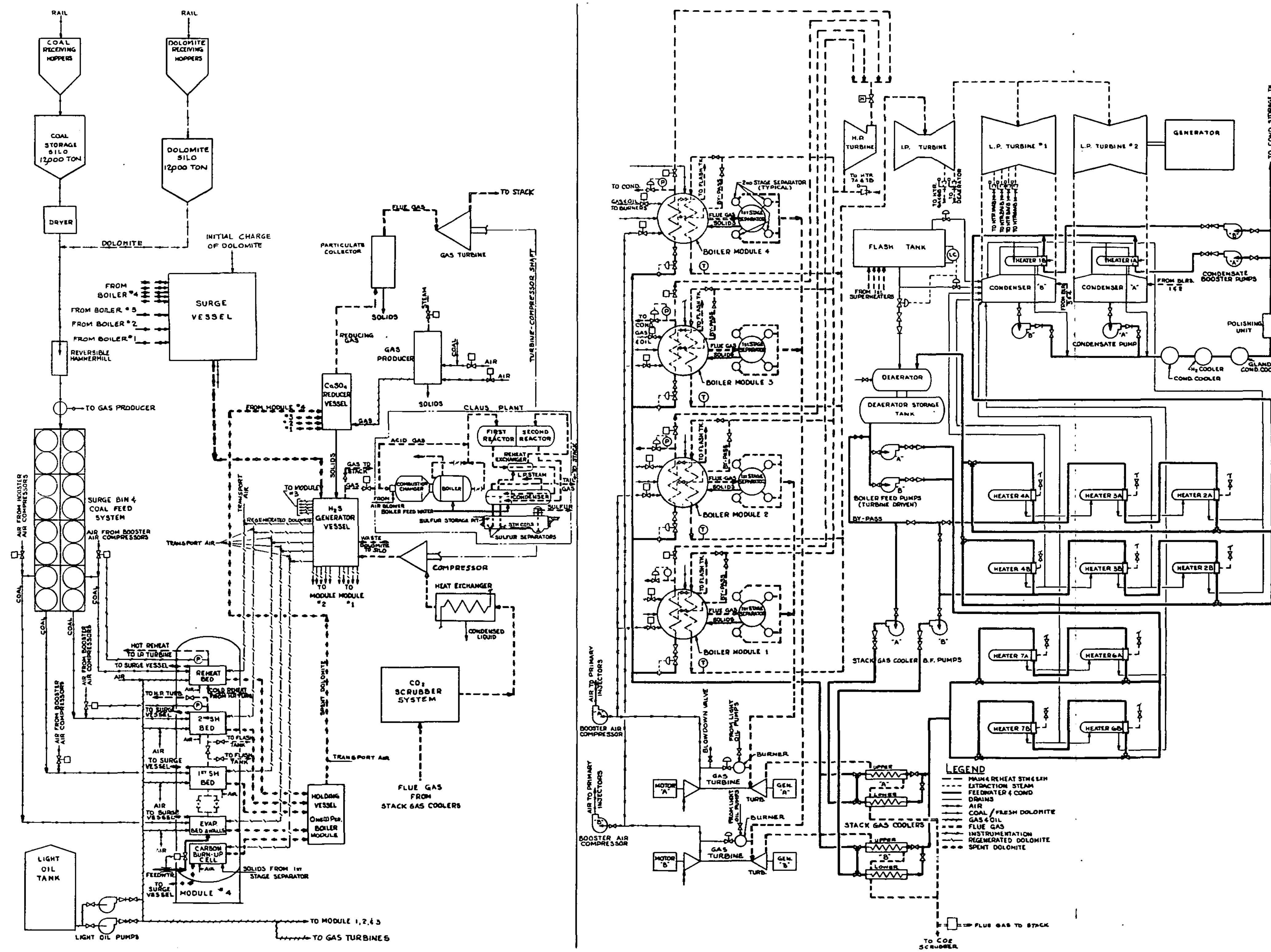
PLATE NO.	TITLE
I	Composite Flow Diagram
II	Performance Diagram
III	Site Plot Plan
IV	Plan @ Grade Floor - Elevation - 18'-0"
V	Plan @ Operating Floor - Elevation - 58'-0"
VI	Mezzanine Floor Plan
VII	Elevation
VIII	Flue Gas from Boiler to Gas Turbine - Isometric
IX	Plan - Gas Piping from Separators (2-2nd Stage Separators)
X	Elevation - Gas Piping from Separators (2-2nd Stage Separator)

ELECTRICAL

XI	Single Line Diagram
XII	Start-up Sequence Diagram - Sheet No. 1
XIII	Start-up Sequence Diagram - Sheet No. 2
XIV	Start-up Sequence Diagram - Sheet No. 3

STRUCTURAL

XV	Steam Generator & Storage Bins - Concrete Foundations
XVI	Steam Generator & Storage Bins - Structural Framing
XVII	Steam Turbine Foundations - Plans & Sections
XVIII	Steam Turbine Foundations - Cross Sections



635 MW FLUIDIZED BED
BOILER COMBINED CYCLE PLANT
COMPOSITE FLOW DIAGRAM

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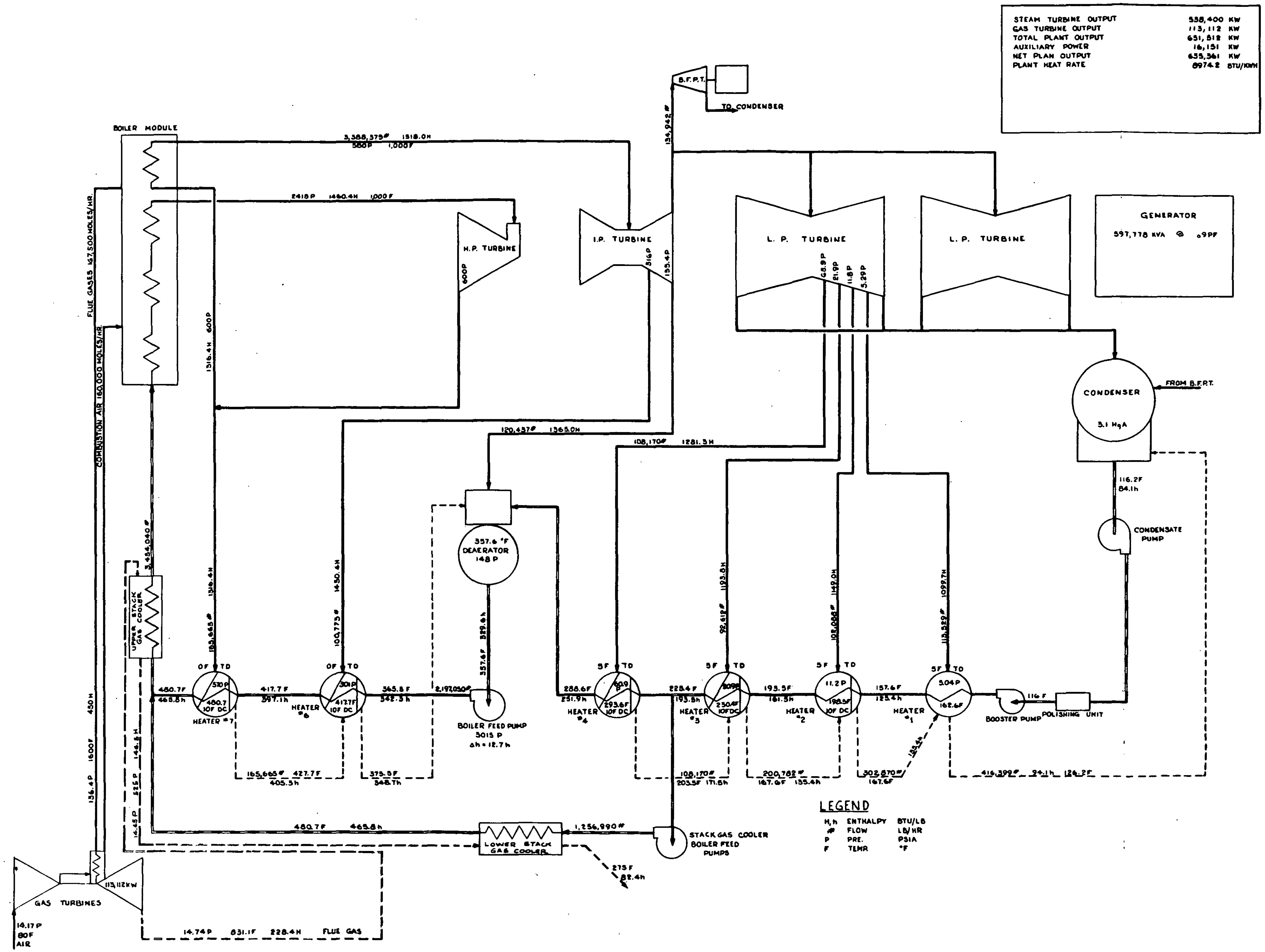
PLATE I



635MW FLUIDIZED BED
BOILER COMBINED CYCLE PLANT
PERFORMANCE DIAGRAM
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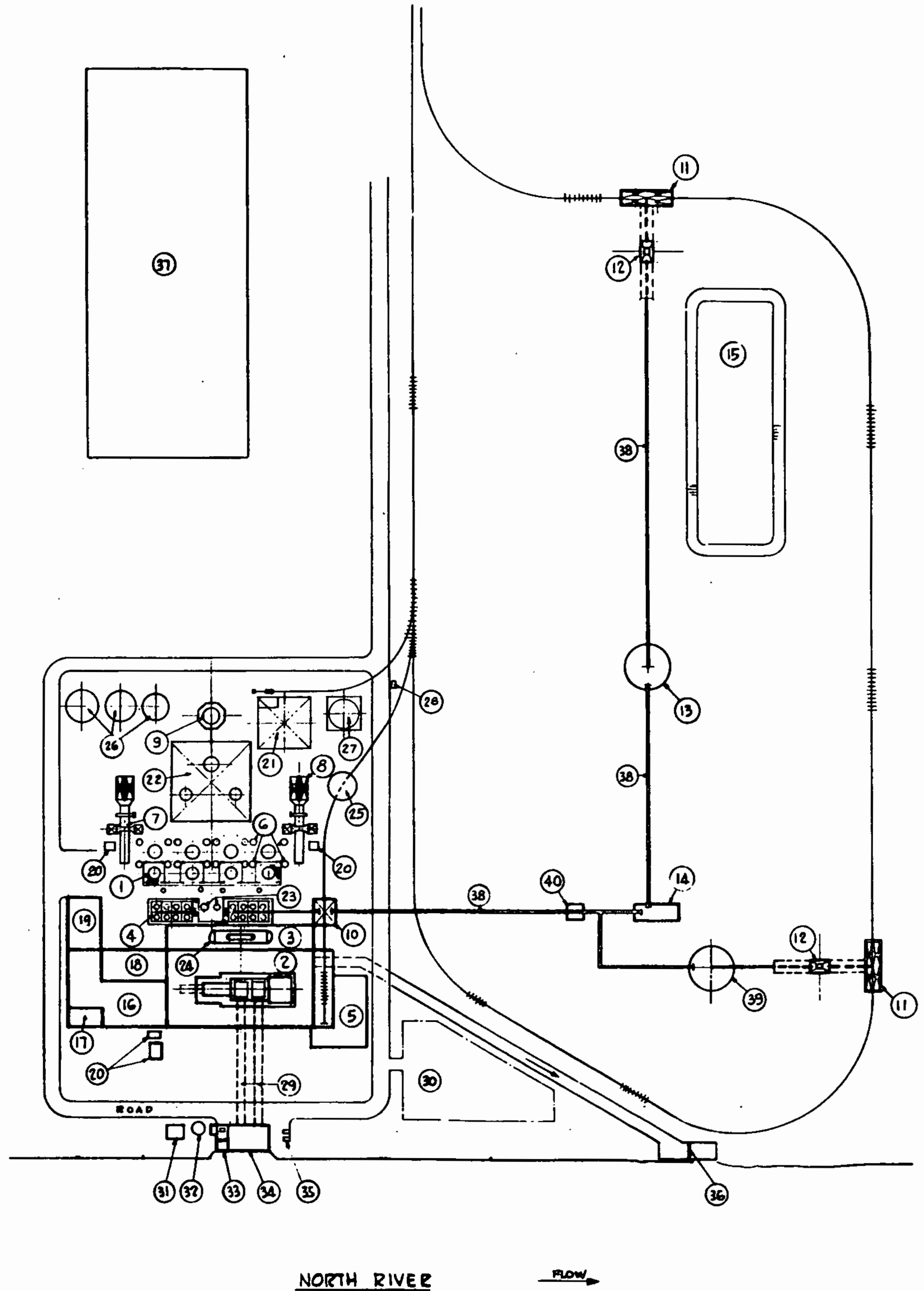
PLATE II



1. BOILERS
2. TURBINE BAY
3. HEATER BAY
4. STORAGE BINS
5. ADMINISTRATION BLDG.
6. PARTICULATE REMOVAL EQUIPT
7. GAS TURBINE GENERATORS
8. STACK GAS COOLERS
9. STACK
10. 150 TON SURGE BIN
11. RECEIVING HOPPER
12. RECLAIM HOPPER
13. SILO (12,000 TONS)
14. DRYER
15. DEAD STORAGE (50,000 TONS)
16. WATER TREATMENT
17. DIESEL GENERATOR ROOM
18. CONTROL ROOM
19. MACHINE SHOP
20. TRANSFORMERS
21. CLAUS PLANT & SULFUR LOADING
22. CO_2 ABSORBERS
23. REGENERATORS & SURGE VESSEL
24. DEAERATOR
25. ASH & DOLOMITE SILO
26. WATER STORAGE TANKS
27. LIGHT OIL TANK
28. LIGHT OIL UNLOADING PUMP
29. CIRCULATING WATER INTAKE PIPES
30. PARKING AREA
31. GASES (STORAGE)
32. FILTERED & FIRE SERVICE WATER TANK
33. SERVICE WATER PUMP HOUSE
34. CIRCULATING WATER INTAKE STRUCTURE
35. CHLORINATION EQUIPMENT
36. VARIABLE WEIRS
37. SWITCH YARD
38. CONVEYORS
39. DOLOMITE SILO (12,000 TONS)
40. MILL

SITE PLOT PLAN

0 50 100 200 300

SCALE

635 MW FLUIDIZED BED BOILER COMBINED CYCLE PLANT

SITE PLOT PLAN

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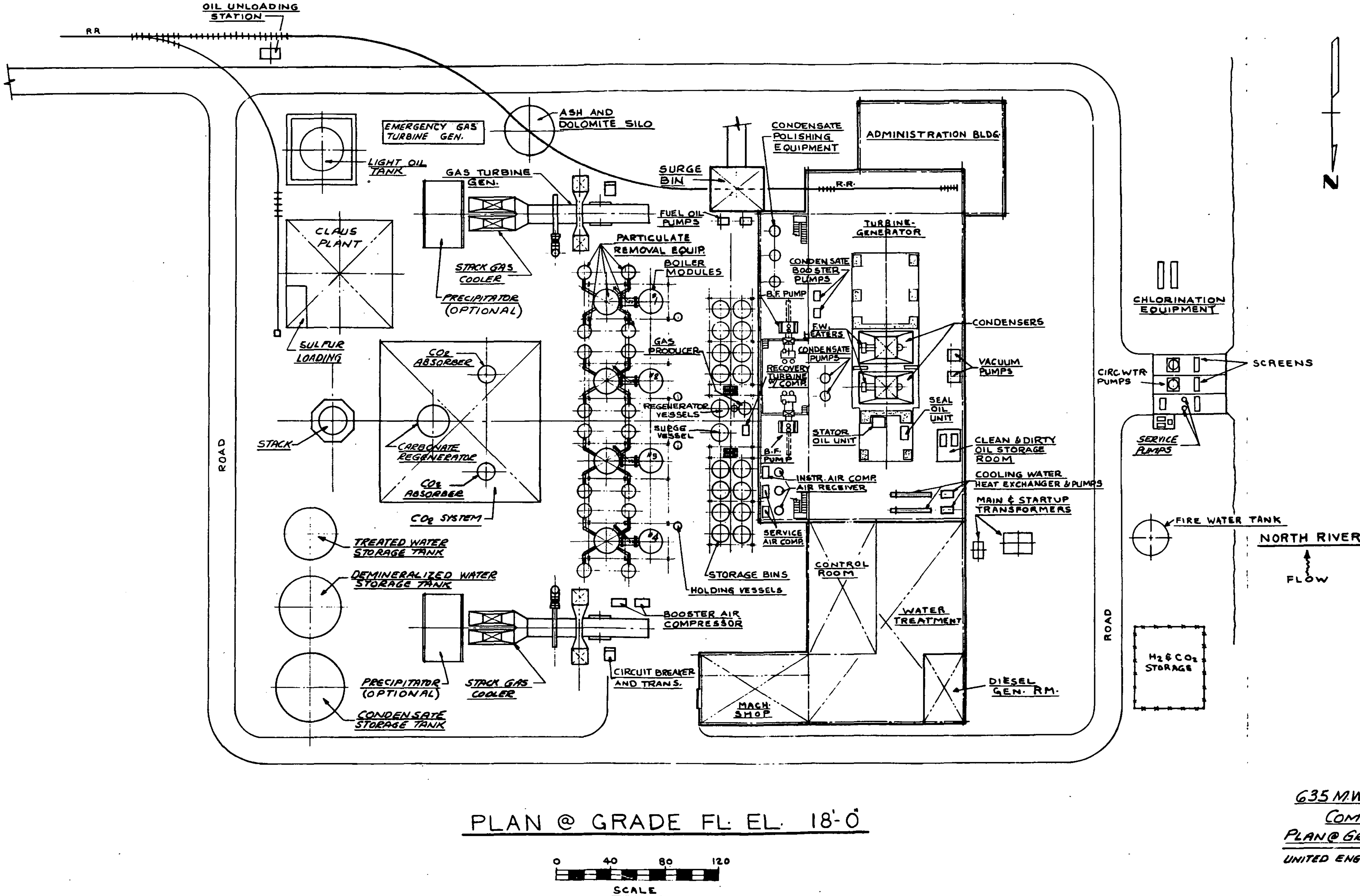
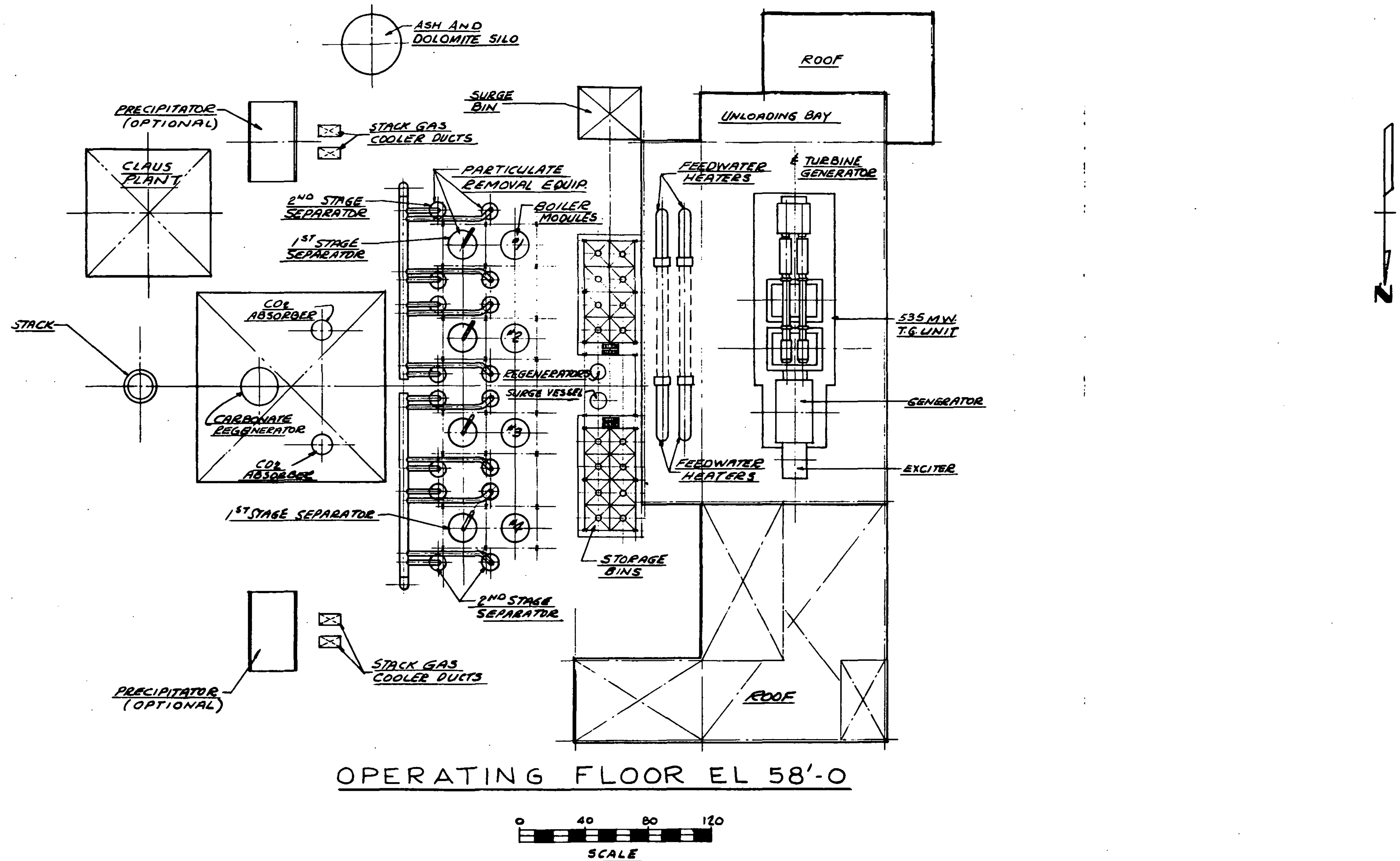
PLATE III



PLAN @ GRADE FL. EL. 18'-0"

635 M.W. FLUIDIZED BED BOILER
COMBINED CYCLE PLANT
PLAN @ GRADE FLOOR - ELEVATION 18'0"
UNITED ENGINEERS & CONSTR. INC. PHILA., PENNA.

PLATE IV

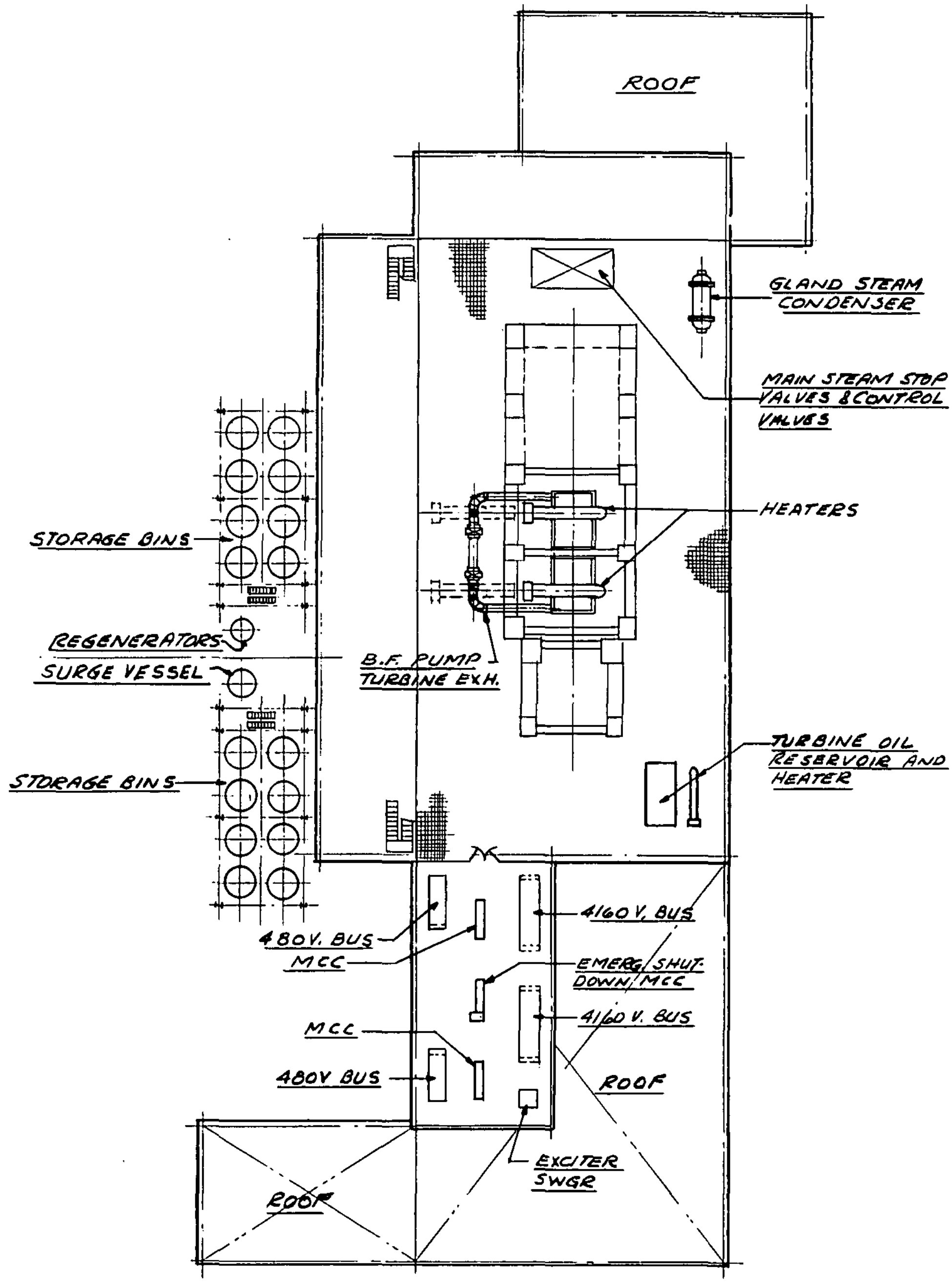


OPERATING FLOOR EL 58'-0

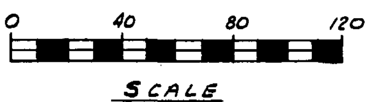
635 M.W. FLUIDIZED BED BOILER
COMBINED CYCLE PLANT
OPERATING FLOOR PLAN ELEV. 58'-0"

UNITED ENGINEERS & CONSTR. INC. PHILA., PENNA.

PLATE V



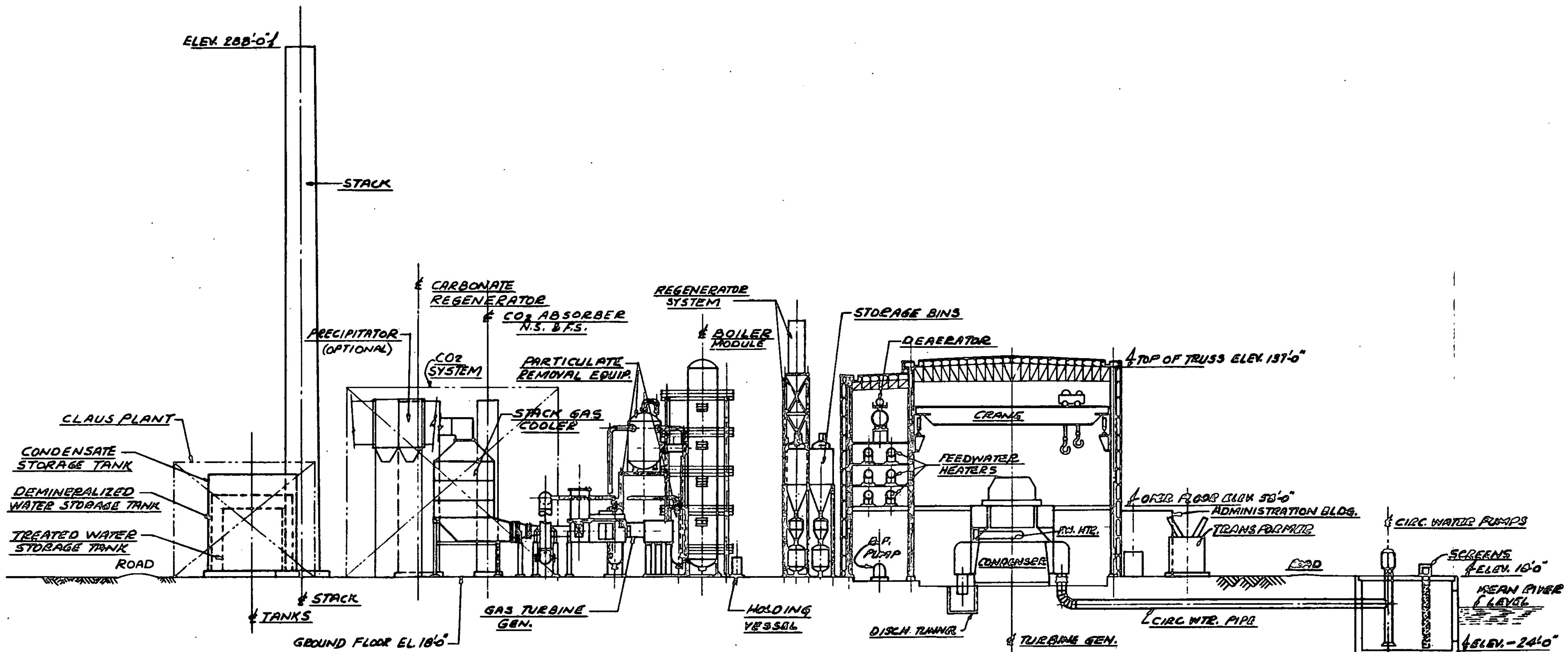
MEZZANINE FLOOR EL. 43'-0"



635 M.W. FLUIDIZED BED BOILER
COMBINED CYCLE PLANT
MEZZANINE FLOOR PLAN

UNITED ENGINEERS & CONSTR. INC PHILA, PENNA.

PLATE VI



ELEVATION

0 40 80 120

SCALE

635 M.W. FLUIDIZED BED BOILER
COMBINE CYCLE PLANT
ELEVATION

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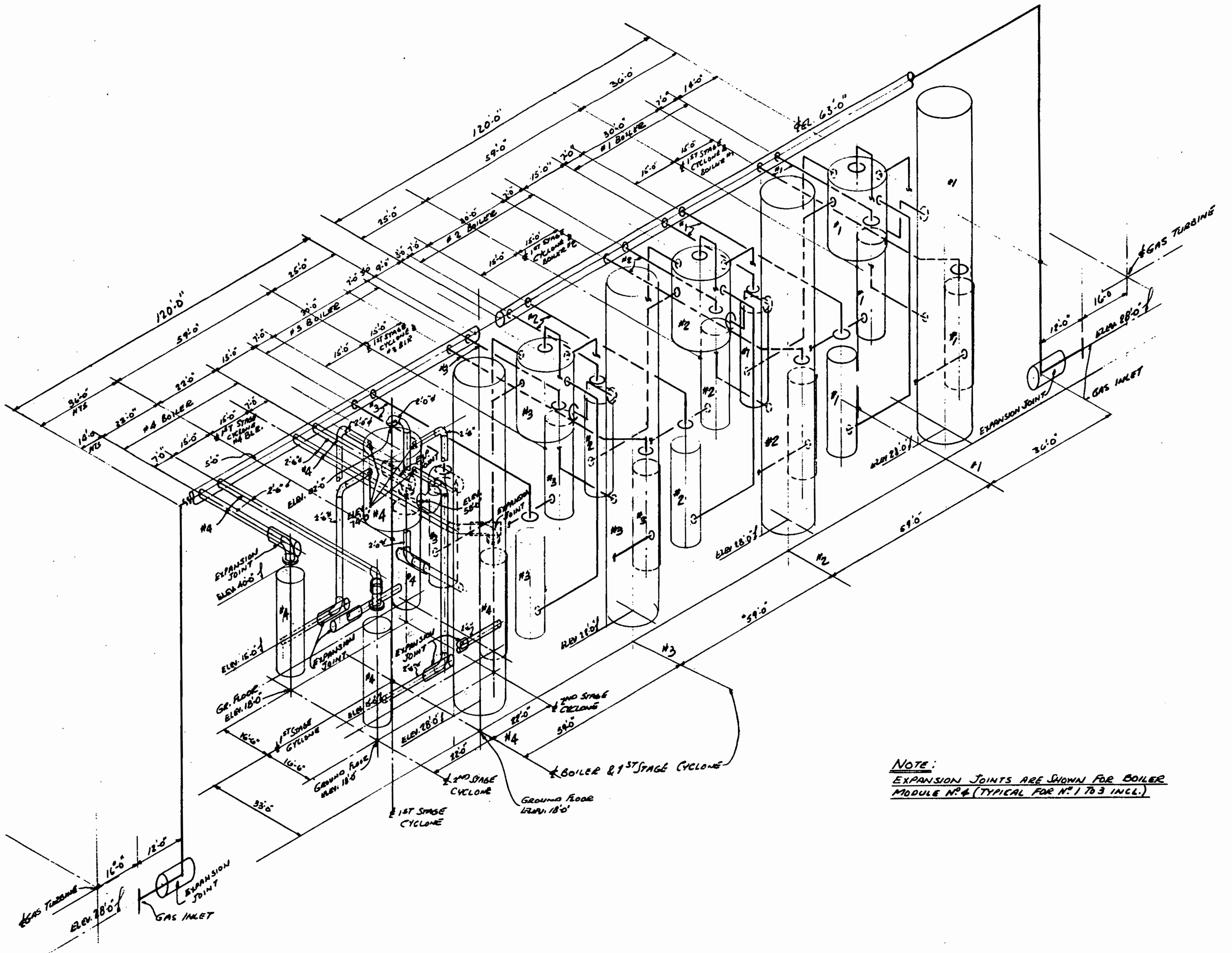
PLATE VII

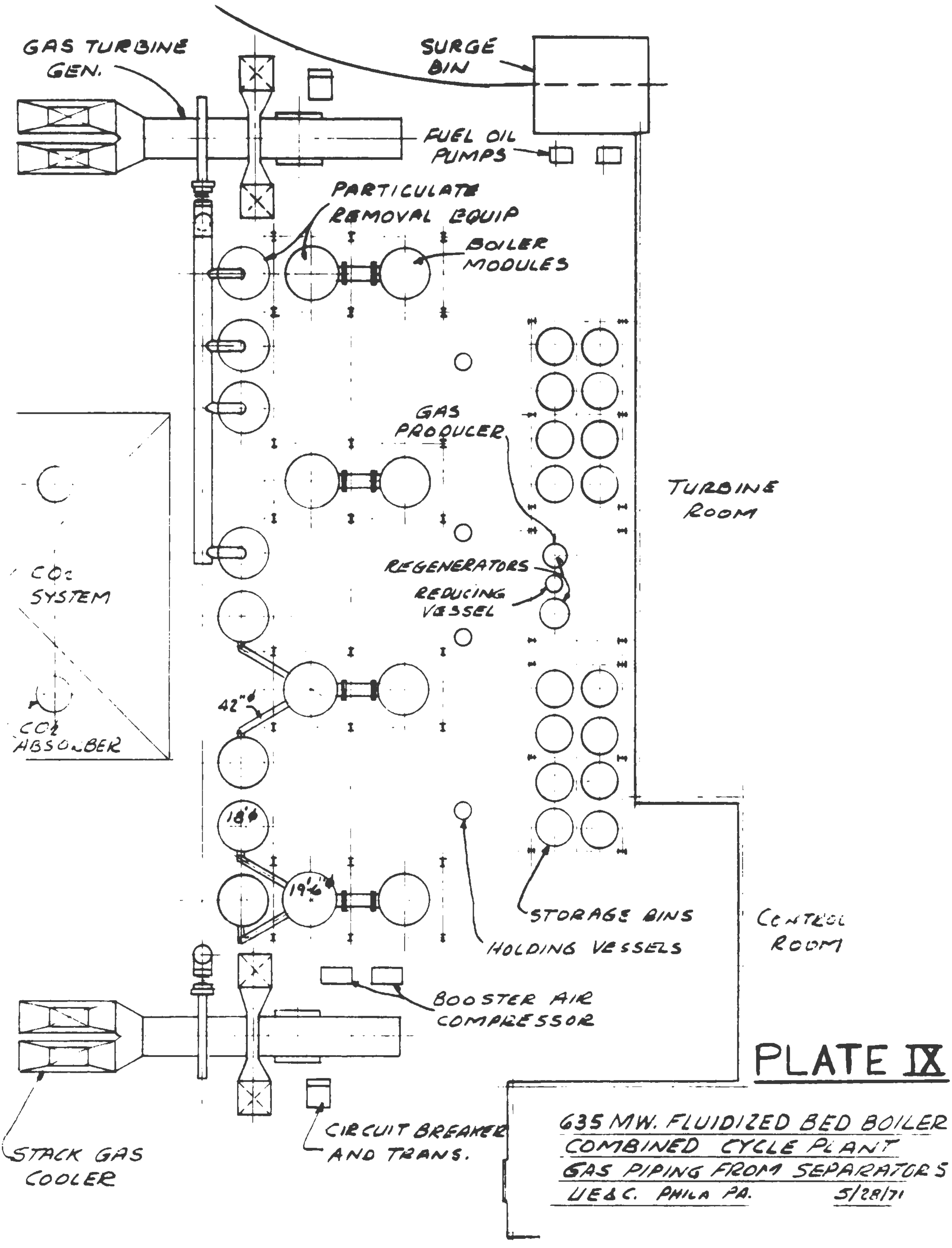


635 M.W. FLUIDIZED BED BOILER COMBINED CYCLE PLANT

FLUE GAS FROM BOILER TO GAS TURBINE · ISOMETRIC

UE&C. PHILA

PLATE VIII


GAS TURBINE GEN.
SURGE BIN
FUEL OIL PUMPS
PARTICULATE REMOVAL EQUIP
BOILER MODULES
GAS PRODUCER
TURBINE ROOM
REGENERATORS
REDUCING VESSEL
CO2 SYSTEM
42"ϕ
CO2 ABSORBER
18'ϕ
19'6"ϕ
STORAGE BINS
HOLDING VESSELS
CONTROL ROOM
BOOSTER AIR COMPRESSOR
PLATE IX
STACK GAS COOLER
CIRCUIT BREAKER AND TRANS.
635 MW. FLUIDIZED BED BOILER
COMBINED CYCLE PLANT
GAS PIPING FROM SEPARATORS
UE&C. PHILA PA.
5/28/71

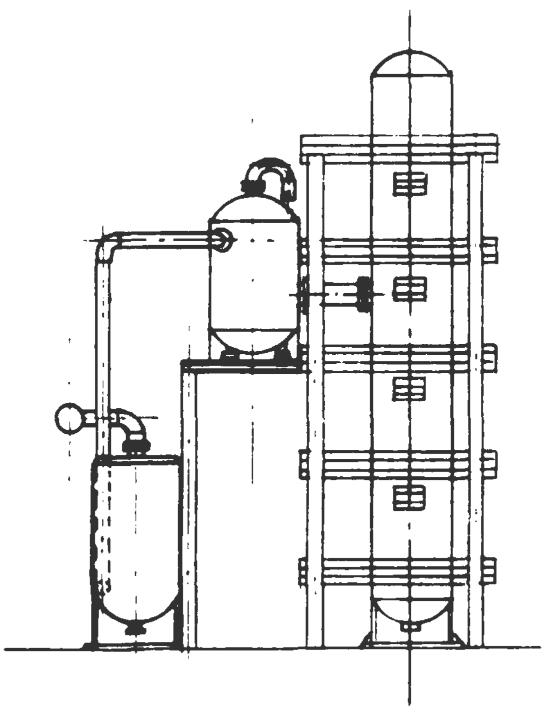
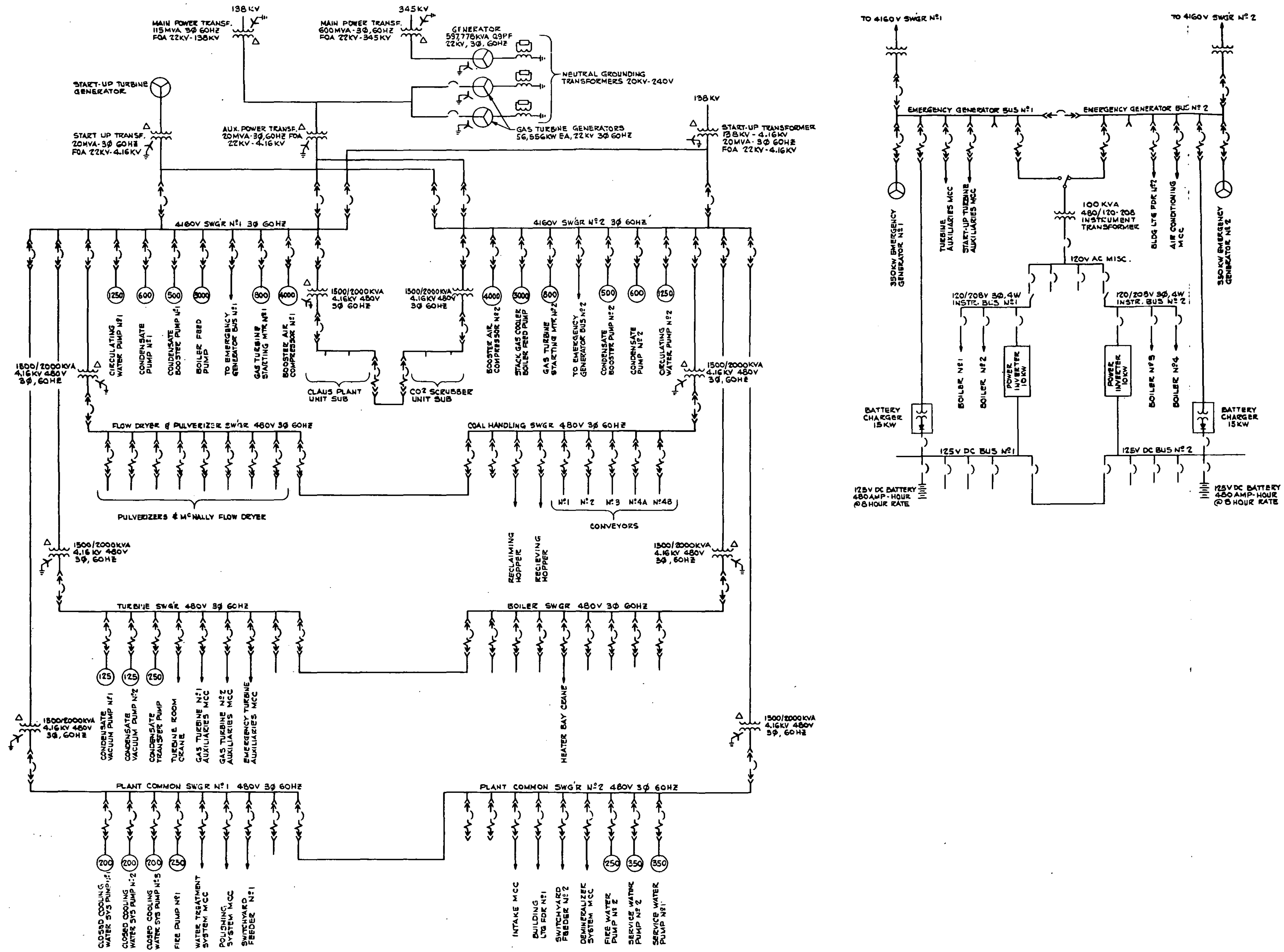


PLATE X

635 M.W. FLUIDIZED BED BOILER
COMBINED CYCLE PLANT
GAS PIPING FROM SEPARATORS
ELEVATION 6/7/71

UE&C PHILA.



635MW FLUIDIZED BED
BOILER COMBINED CYCLE PLANT
ELECTRICAL
SINGLE LINE DIAGRAM

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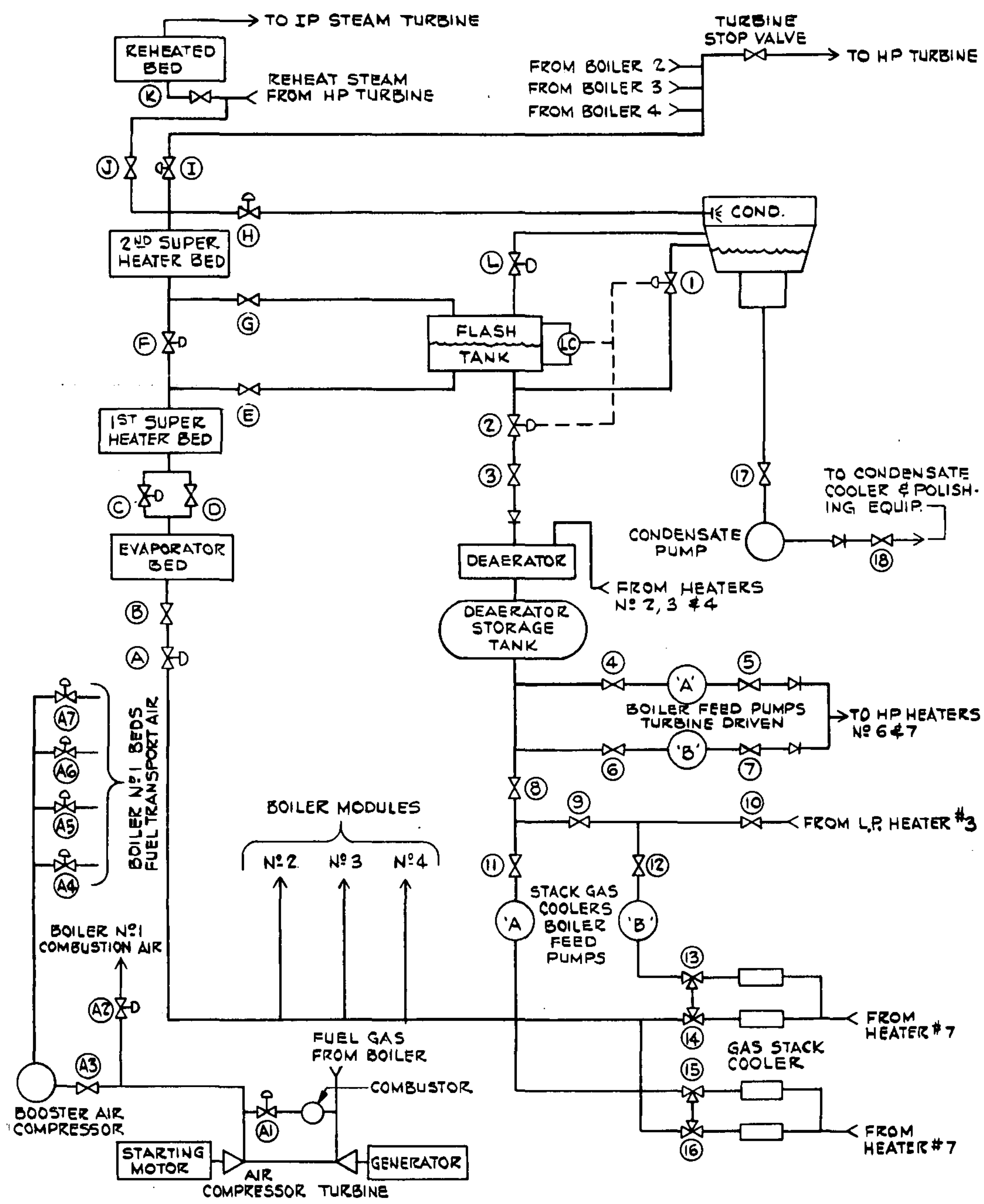
PLATE XI



GAS TURBINE

BOILER START-UP SEQUENCE DIAGRAM
(FLOW DIAGRAM)

VALVE POSITIONING - COLD BOILER START-UP

BOILER VALVES		FEEDWATER SYSTEM VALVES				AIR VALVES	
Nº	POSITION	Nº	POSITION	Nº	POSITION	Nº	POSITION
A	OPEN	1	CLOSED	10	OPEN	A1	CLOSED
B	OPEN	2	CLOSED	11	OPEN	A2	CLOSED
C	OPEN	3	OPEN	12	OPEN	A3	CLOSED
D	OPEN THROTTLED	4	OPEN	13	BY PASS-OPEN	A4	CLOSED
E	OPEN	5	OPEN	14	BY PASS-OPEN	A5	CLOSED
F	CLOSED	6	OPEN	15	BY PASS-OPEN	A6	CLOSED
G	CLOSED	7	OPEN	16	BY PASS-OPEN	A7	CLOSED
H	CLOSED	8	OPEN	17	OPEN		
I	CLOSED	9	OPEN	18	OPEN		
J	CLOSED						
K	CLOSED						
L	CLOSED						

NOTE: IT HAS BEEN ASSUMED THAT FOR A COLD START-UP SEQUENCE THAT

1. THE PLANT WATER SYSTEM HAS BEEN PLACED IN OPERATION AND DEMINERALIZED WATER HAS BEEN USED TO FILL THE CONDENSER
2. THE SURGE TANK HAS BEEN FILLED WITH DOLOMITE
3. THE PLANT COAL AND DOLOMITE FEEDING SYSTEMS HAVE BEEN PLACED IN OPERATION AND FUEL IS AVAILABLE AT THE STORAGE BINS AND INJECTORS FOR TRANSPORT TO THE BED.

VALVE POSITIONING - HOT BOILER RESTART

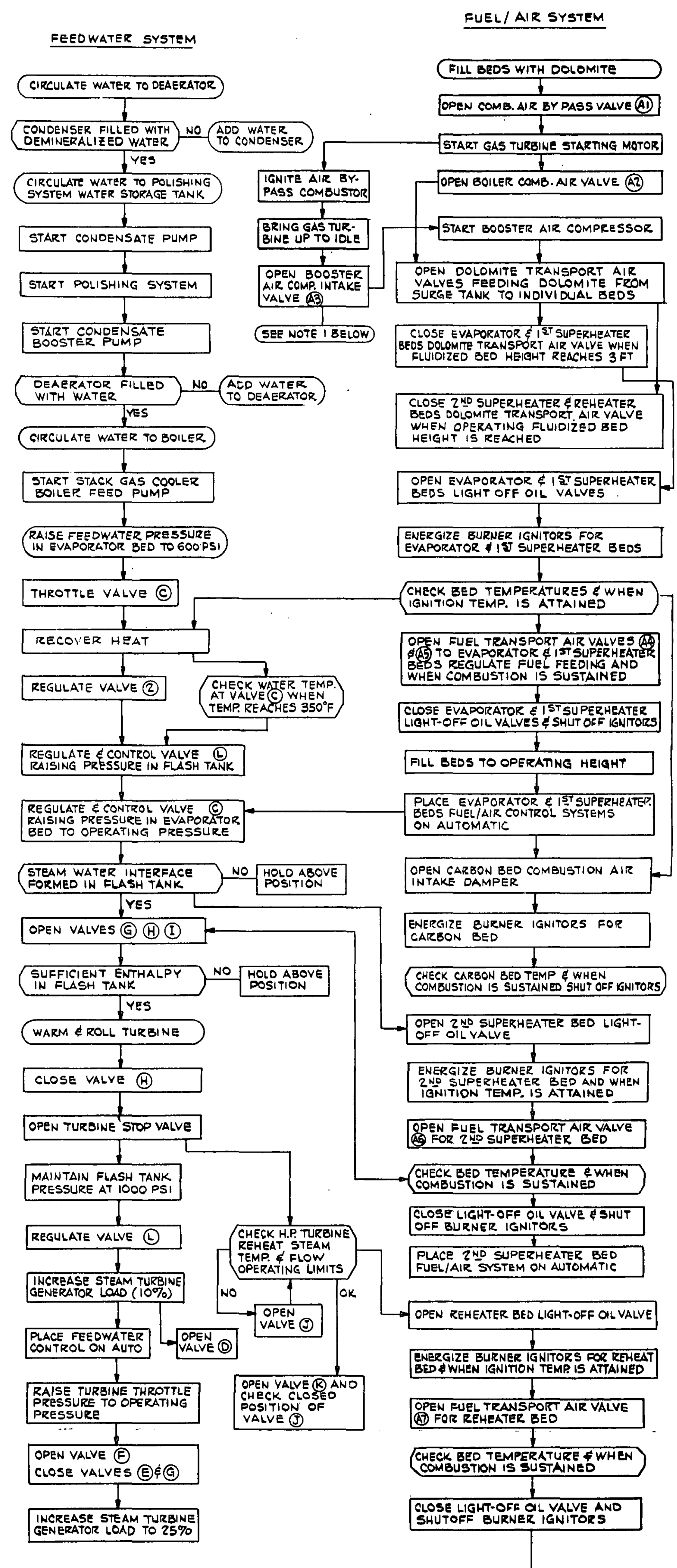
BOILER VALVES		FEEDWATER SYSTEM-VALVES				AIR VALVES	
Nº	POSITION	Nº	POSITION	Nº	POSITION	Nº	POSITION
A	OPEN-THROTTLED	1	CLOSED	10	OPEN	A1	CLOSED
B	OPEN	2	CLOSED	11	OPEN	A2	CLOSED
C	OPEN-THROTTLED	3	OPEN	12	OPEN	A3	OPEN
D	OPEN-THROTTLED	4	OPEN	13	OPEN TO COOLER	A4	CLOSED
E	OPEN	5	OPEN	14	OPEN FROM COOLER	A5	CLOSED
F	CLOSED	6	OPEN	15	OPEN TO COOLER	A6	CLOSED
G	CLOSED	7	OPEN	16	OPEN FROM COOLER	A7	CLOSED
H	OPEN	8	OPEN	17	OPEN		
I	CLOSED	9	OPEN	18	OPEN		
J	CLOSED						
K	CLOSED						
L	CLOSED						

NOTE: IT HAS BEEN ASSUMED THAT FOR A HOT RESTART SEQUENCE THAT
1. THE BEDS ARE FILLED WITH DOLOMITE 2. AT LEAST ONE BOILER IS IN OPERATION

START-UP SEQUENCE DIAGRAM - SHEET Nº 1

635 MW FLUIDIZED BED BOILER
COMBINED CYCLE PLANT
START-UP SEQUENCE DIAGRAM
SHEET Nº 1
UNITED ENGINEERS & CONSTR. INC
PHILA., PENNA.

PLATE XII



NOTES & SYMBOLS

1. IGNITOR FOR AIR BY-PASS COMBUSTOR AUTOMATICALLY SHUTS OFF WHEN FLUE GAS TEMPERATURE FROM BOILER IS WITHIN TURBINE OPERATING LIMITS

DENOTES ACTION TO BE TAKEN

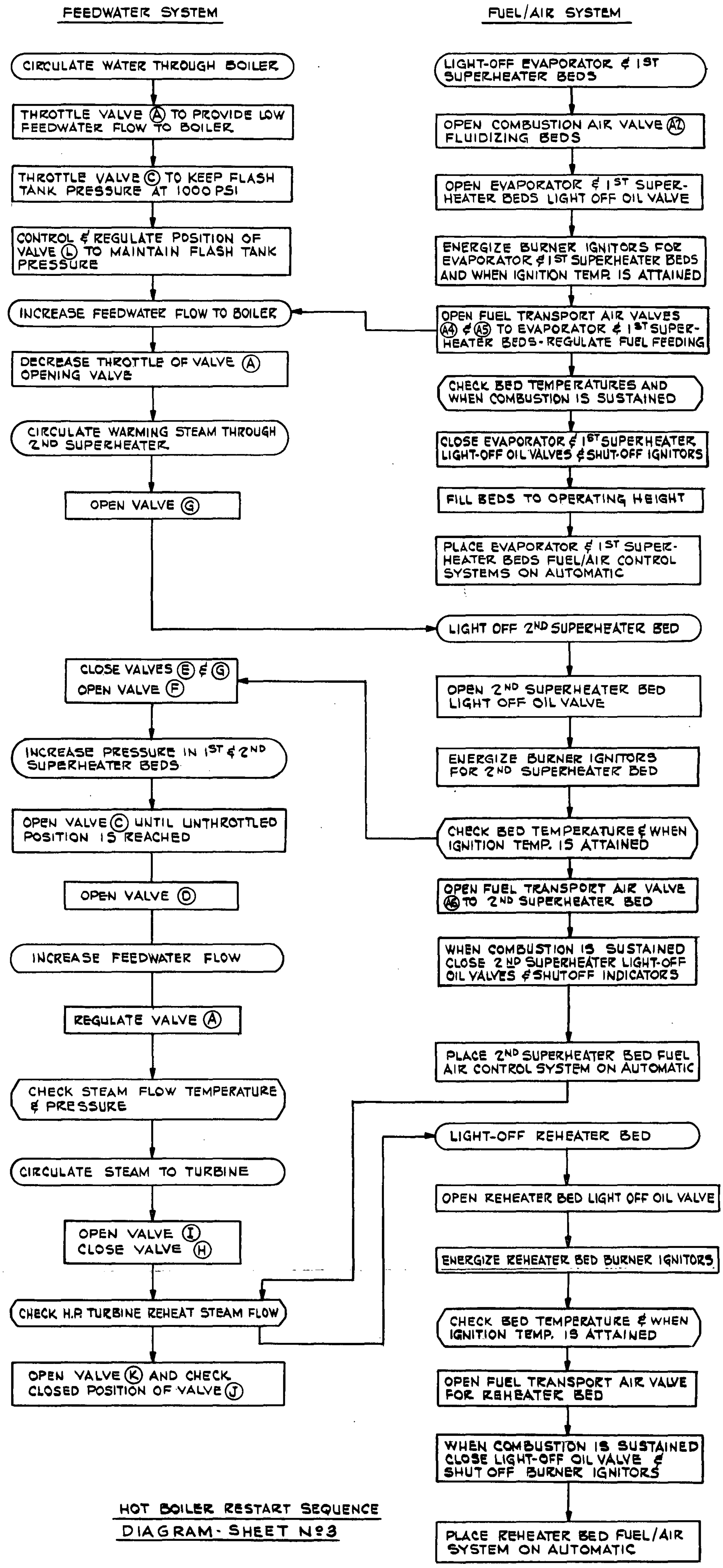
DENOTES OPERATION TO BE PREFORMED

DENOTES CHECK TO BE MADE BEFORE NEXT ACTION OR OPERATION

COLD BOILER START-UP SEQUENCE DIAGRAM - SHEET No 2

635 MW FLUIDIZED BED BOILER COMBINED CYCLE PLANT
START-UP SEQUENCE DIAGRAM
SHEET No 2
UNITED ENGINEERS & CONSTR. INC
PHILA. PENNA.

PLATE XIII



HOT BOILER RESTART SEQUENCE
DIAGRAM - SHEET No 3

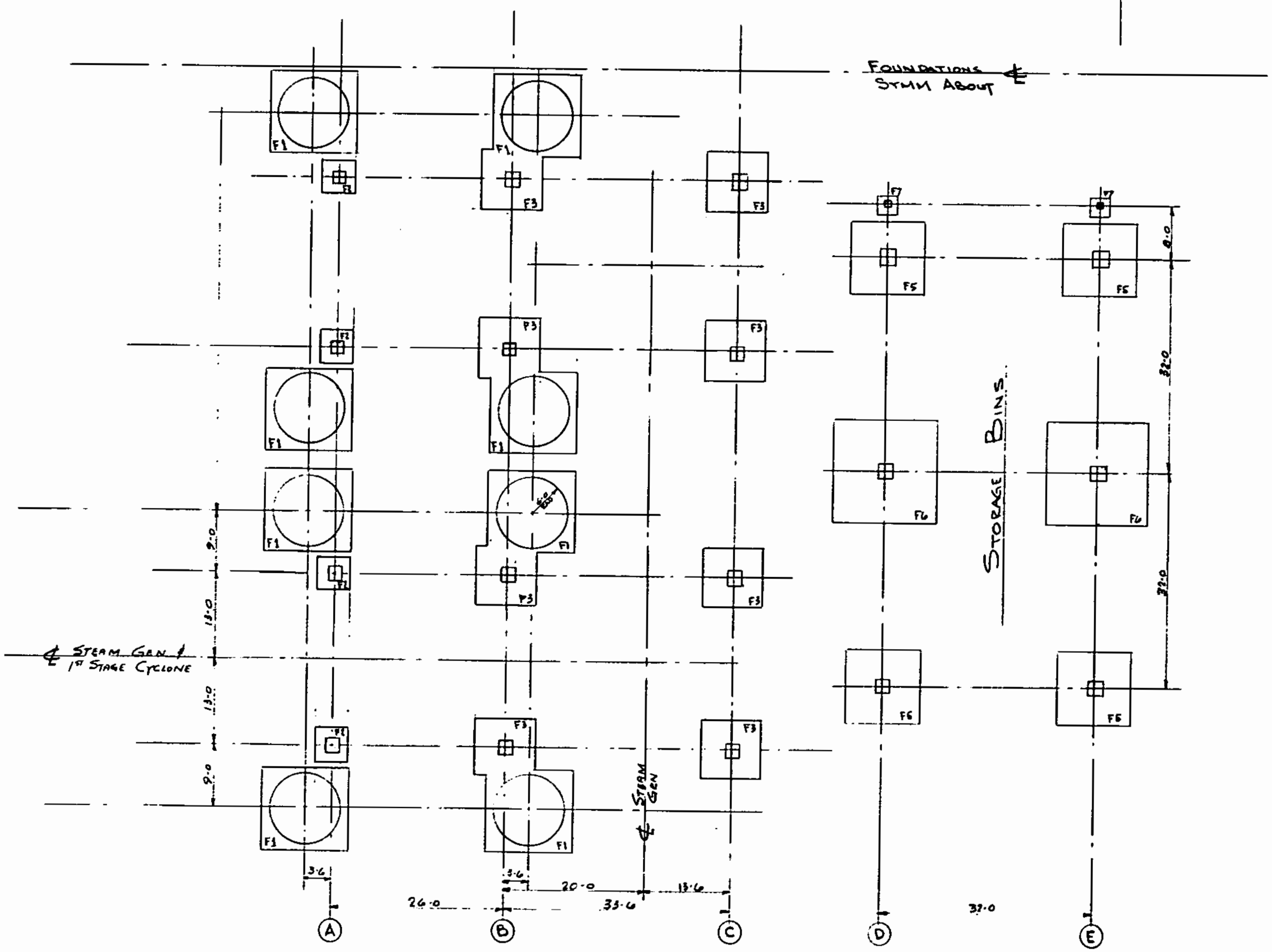
635 MW FLUIDIZED BED BOILER
COMBINED CYCLE PLANT
START-UP SEQUENCE DIAGRAM
SHEET No 3
UNITED ENGINEERS & CONSTR. INC
PHILA., PENNA.

PLATE XIV

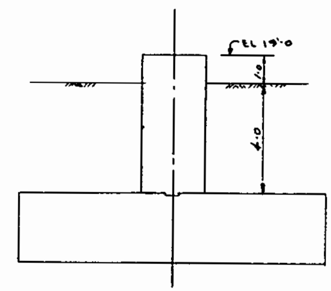
Foundation Schedule

Ftg No	Total No	Footing Size	Footing Conc	Footing Forms	Pier Size	Pier Conc	Pier Forms	Total Conc CY	Total Form Sqft
F1	16	13·0 × 13·0 × 24"	12.5	104	12'·0"⌀ × 5·0	7	113	312	3472
F2	8	5·0 × 5·0 × 18"	1.4	30	2·0 × 2·0 × 5·0	.74	40	17.1	560
F3	16	9·0 × 9·0 × 24"	6	72	2·0 × 2·0 × 5·0	.74	40	107.8	2272
F4	—								
F5	8	11·0 × 11·0 × 30"	11.2	110	2·3 × 2·3 × 5·0	.94	45	97	1240
F6	4	15·6 × 15·6 × 45"	33.4	232	2·3 × 2·3 × 5·0	.94	45	137.4	1108
F7	4	3·6 × 3·6 × 12"	.45	14	1·3 × 1·3 × 5·0	.29	25	3	156
Gas Turbine Foundation								1000	2850
								1674.3	11,658

3000 PSI · 28 Day Conc.
6000 PSF Allowable Soil Bearing



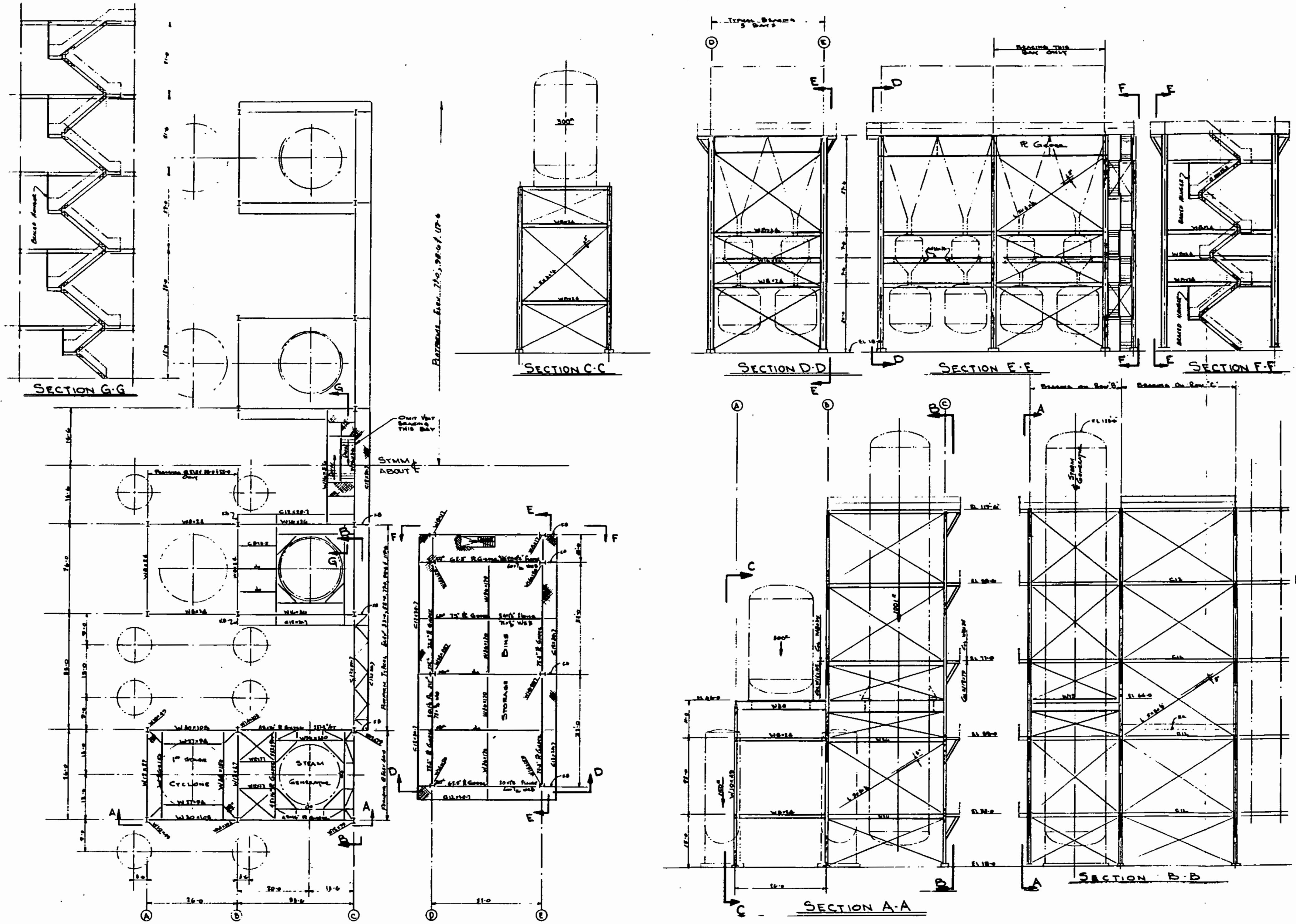
FOUNDATION PLAN



Typical Footing

635 MW FLUIDIZED BED
BOILER COMBINED CYCLE PLANT
STEAM GEN. & STORAGE BINS
CONCRETE FOUNDATIONS
United Engineers & Constructors Inc.

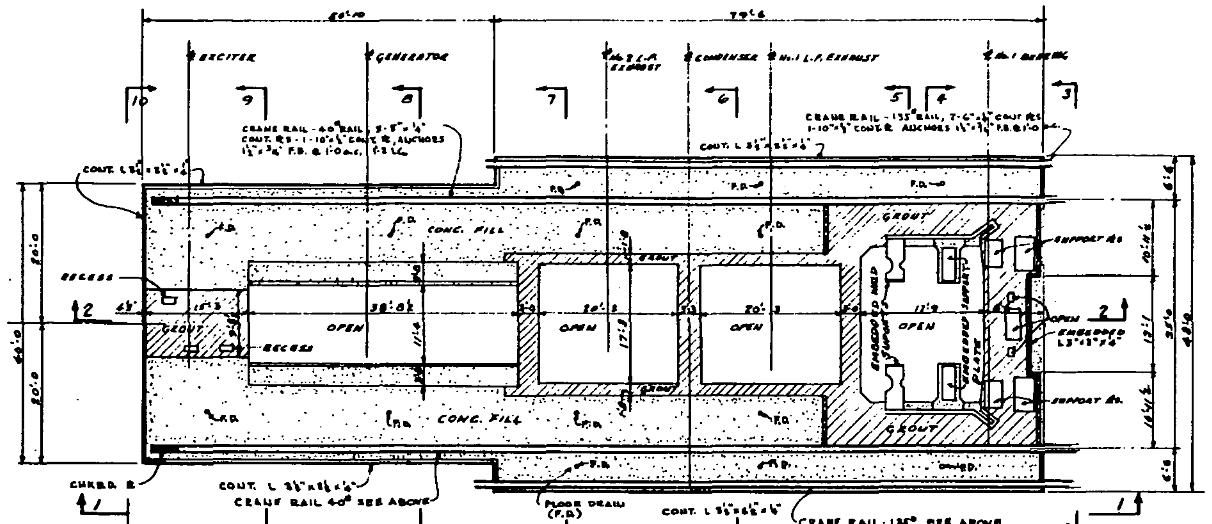
PLATE XV



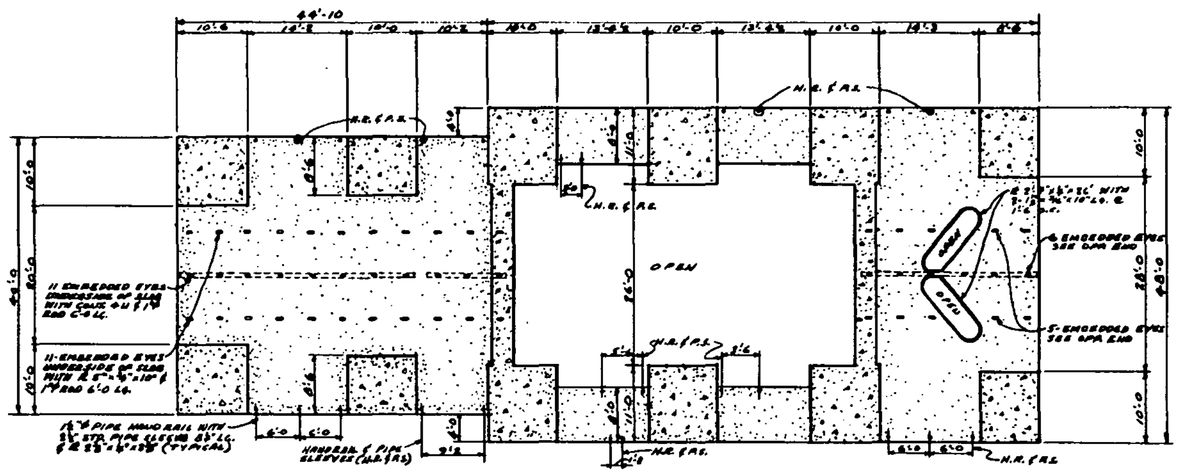
635 MW FLUIDIZED BED
BOILER COMBINED CYCLE PLANT
STEAM GEN. & STORAGE BINS
STRUCTURAL FRAMING

United Engineers & Constructors Inc.

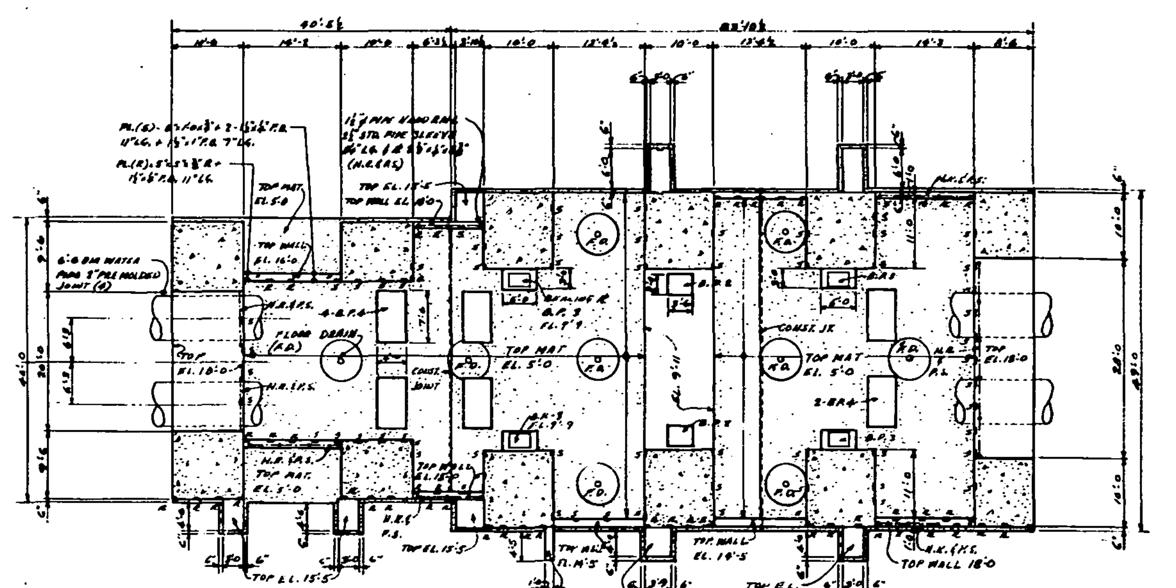
PLATE XVI



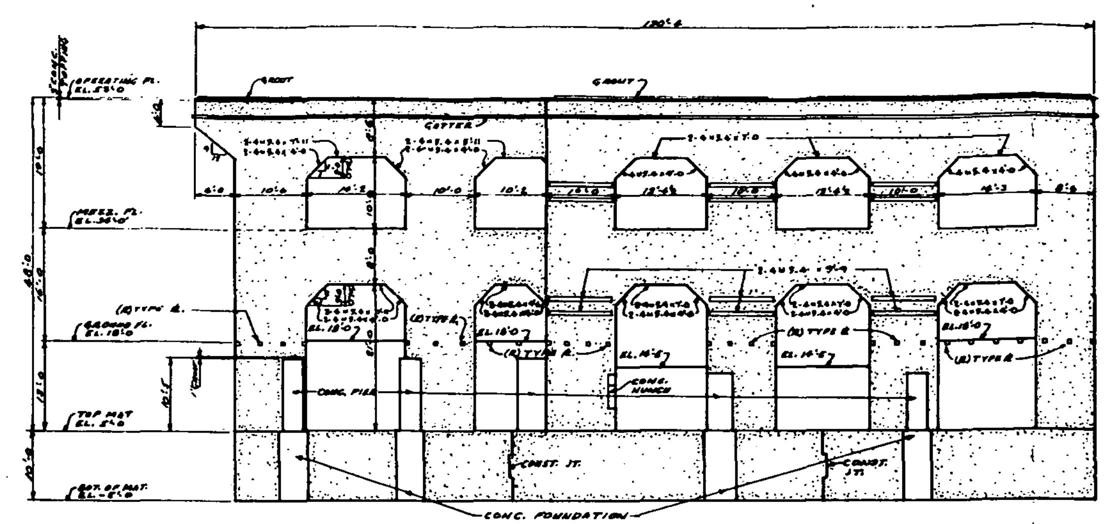
OPERATING FLOOR
FIN. FL. ELEV. = 55'-0



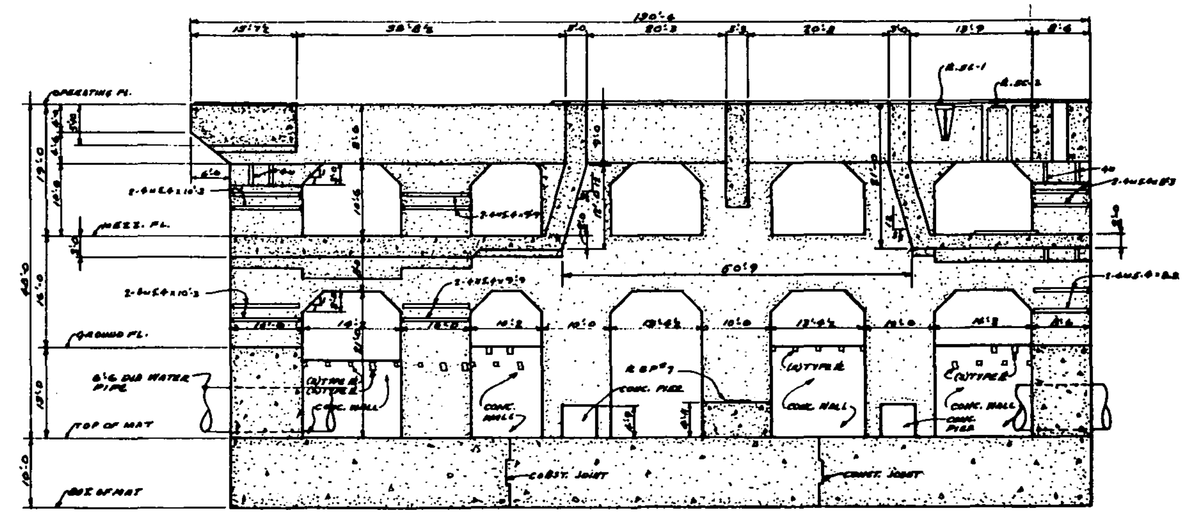
MEZZANINE FLOOR SECTIONAL PLAN
FIN. FL. ELEV. 34'-0



FOUNDATION PLAN
TOP OF MAT. ELEV. 5'-0



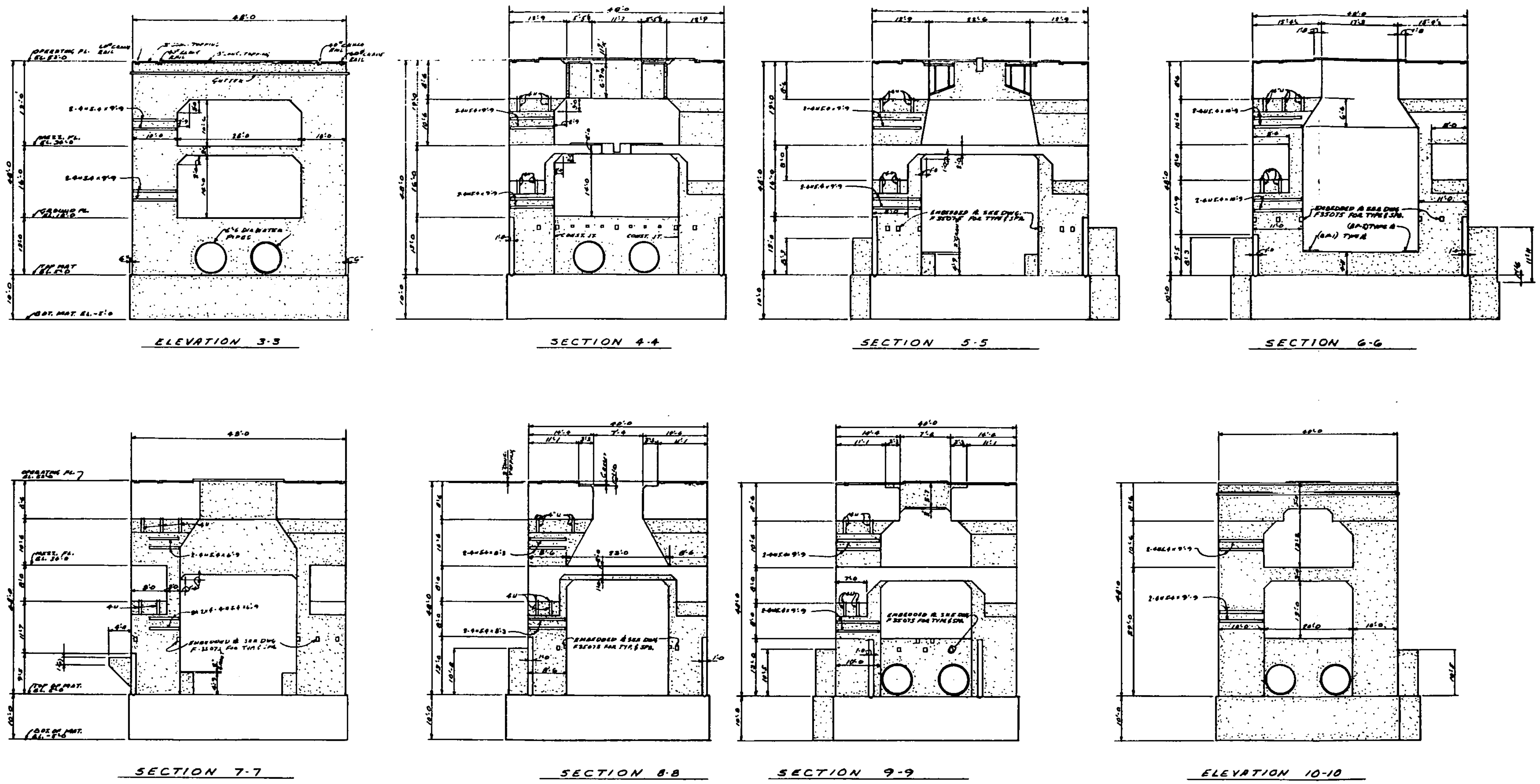
ELEVATION 1-1



SECTION 2-2

635 MW FLUIDIZED BED
BOILER COMBINED CYCLE PLANT
STEAM TURBINE FOUNDATION
PLANS & SECTIONS
United Engineers

PLATE XVII



635 MW FLUIDIZED BED
BOILER COMBINED CYCLE PLANT
STEAM TURBINE FOUNDATION
CROSS SECTIONS

united engineers

PLATE XVIII

APPENDIX K

ATMOSPHERIC-PRESSURE BOILER DESIGN REPORT

Prepared by: Foster Wheeler Corporation

Authors: R. W. Bryers
J. D. Shenker

ABSTRACT

A proposal type design of 280 MW atmospheric-pressure fluidized-bed boiler was prepared after reviewing several alternate arrangements. The preferred arrangement, a once-through, sub-critical pressure steam generator, consists of four modules each containing six vertically stacked beds including a crabon burn-up cell. The investigation included a study of the boiler design and operation, the selection of coal handling and feeding equipment, the selection of particulate removal equipment, and cost estimation of the boiler system with auxiliary equipment. The cost estimates were extrapolated to 300 MW and 600 MW level and compared with conventional boilers of similar size. It is concluded that an atmospheric pressure fluidized bed boiler would generate steam and provide adequate control of particulates, at a capital cost of boiler plant equipment 10% lower than a conventional coal-fired plant.

SUMMARY

The preferred concept for the atmospheric fluidized bed steam generator is a subcritical, once-through, four-module boiler with six vertically-stacked beds including a carbon burn-up cell. The once-through system permits more freedom to the designer in locating and orientating heating surface, which is compatible with the fluid bed concept. With overall heat transfer coefficients of btu/hr-ft^2-°F at bed temperatures of 1600°F, there are no particular problems with high tube metal surface temperatures because even though the average heat flux is greater in the fluidized bed steam generator, the maximum heat flux is much lower than the peak heat flux rate encountered in the furnaces of conventional steam generators.

The preferred concept was selected over other arrangements, including a compartmentalized, vertically-stacked bed arrangement and a horizontal, tandem bed arrangement, for its simplicity in construction and adaptability to the desired mode of operation.

The vertical arrangement greatly simplifies water circuitry and eases construction by reducing the need for headers, downcomers, and inter-connecting pipes to a minimum. This arrangement also simplifies the distribution of air to and gas from the steam generator. Plenum chambers used for providing and distributing air to the bed also serve the purpose of isolating adjacent beds.

The vertically-stacked, modular construction using six beds including the carbon burn-up cell minimizes operating problems. By assigning separate beds to carry out the functions of reheating,

superheating and evaporating, individual beds may be started up in sequence thus avoiding the risk of overheating uncooled surface. Turn down ratios of 4 to 1 are achieved with four-module construction by simply shutting down individual modules as required. To maintain continuity in plant load as the modular load is reduced and individual modules are shut down each module must have a turn down capability of 25% at 75% of plant load, 33% at 50% of plant load and 50% at 25% plant load. Load reduction on any given module is accomplished by dropping the firing rate at constant excess air with a fixed bed height. The bed temperature drops to about 1325-1375°F at 75% full load, 1300-1350°F at 50% load, and 1144-1223°F at 25% load. The turn down capability of any one individual module of 50% required for continuous load reduction actually extends the load reduction capability of the plant to 8 to 1.

Particulate removal from the flue gas is achieved by means of a mechanical dust collector operating in series with an electrostatic precipitator. One mechanical dust collector serves the main fluid bed cells and discharges its particulate effluent to the carbon burn-up cell. The second mechanical dust collector serves the carbon burn-up cell and discharges its particulate effluent to the ash removal system. The total particulate removal system reduces the dust loading of the flue gas to 0.01 grains per standard cubic foot. The system costs $5.00/KW. There is no direct cost comparison with a conventional plant equipped with a wet scrubber for pollution control.

Tubular air preheaters were selected over regenerative air heaters to minimize excessive gas losses associated with the higher flue gas pressure in the fluid bed boiler. This results in a net increase in capital cost of the air preheater of $1/KW.

Coal injection is accomplished by dilute phase transport of 1/4" X 0" coal from a fluid bed dryer-distributor. Coal pulverizing equipment is not required by the fluid bed boiler system. This results in a net savings of $2.08/KW in coal handling and feeding equipment.

The surface requirements of this boiler have been reduced by about 37% in comparison to a conventional boiler. The total volume has been cut in half. This results in a reduction in cost for the steam generator of about 20%. The reduction in cost of the entire boiler plant equipment including the steam generator and accessories such as particulate removal, coal handling, etc. is 9% or $6.94/KW.

Construction time has been reduced from 17 to 14 months with each module requiring about 10 months. This savings in time appears as a reduction in cost of erection and interest paid during construction.

The operating availability should increase considerably with modular construction. Outages need only affect the module in question and result in partial load rather than full load reduction.

The availability of this boiler should also increase by virtue of its inherent capability to handle low grade fuels. At 1600°F bed temperatures the high temperature fouling and corrosion attributed to the alkalis should be supressed. Furnace slagging should be no problem, as this bed temperature is sufficiently below the softening and initial deformation temperatures of the most basic ash to avoid the presence of a liquid phase. High ash coals should not present a problem as the concentration of the ash in the bed at any given time is small.

In conclusion, this conceptual design study has shown, based on the design data used, that the atmospheric fluid bed boiler is a workable concept offering an opportunity for a net reduction in capital cost of 9% over a conventional pulverized coal-fired steam generating plant equipped with pollution abatement equipment. The concept also shows a reduction in construction time which should be reflected in a savings in interest during construction and ultimately total plant investment. The fluid bed boiler should be capable of handling fuels which are troublesome for the present conventional pulverized coal plant. This should result in an improvement in boiler availability.

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APPENDICES

1.0 INTRODUCTION

As subcontractor to Westinghouse on Public Health Service Contract CPA 70-9 to evaluate the fluidized bed combustion process, it was Foster Wheeler's responsibility to prepare proposal type designs of utility boilers at the 300 and 600 MW level according to specifications provided by Westinghouse. At the onset of the program it was decided that design of a pressurized boiler as part of a combined cycle and of an atmospheric boiler as part of a conventional cycle, both at the 300 MW level, would probably be the most worthwhile approach. As work developed and a better assessment of the boiler market was obtained, it was decided to extrapolate the designs and cost data to 600 MW to provide more meaningful results.

This second volume of the report, which appears in two volumes, discusses the atmospheric boiler designed and cost estimated at the 280 MW level nominally referred to as the 300 MW boiler. The data is extrapolated to 300 and 600 MW's to make more meaningful comparisons with conventional equipment. The report includes a presentation of specifications provided by Westinghouse resulting from a state-of-the-art review and discussions of boiler design, performance, operation, cost and trade offs in design where they apply. Auxiliary equipment such as coal handling, coal feeding, heat recovery and particulate removal were selected and cost estimated through the courtest and cooperation of numerous equipment manufacturers. This information is given in appendices.

Concepts investigated and discarded are briefly reviewed and recommendations for future consideration are made.

2. SPECIFICATIONS

2.1 Fuel Specifications

The fuel selected for the atmospheric fluidized bed steam generator was Ohio Pittsburgh No. 8 Seam coal with the detailed specifications listed in Table 2.1. The coal is sized to 1 1/2" x 0". This is a standard commercial size and no penalty is paid for the specification. Typical size disbributions are illustrated in Figure 2.1. The coal will be received by unit train.

2.2 Sorbent Specifications

The sorbent selected for the atmospheric fluidized bed boiler was BCR 1359 Limestone with chemical composition and flow rates as prescribed in Table 2.2. The dolomite will be received pre-crushed to -1/4 inch size in covered rail cars. Truck unloading will be offered as an alternate. The recycled sorbent ranges in size from 1000 to 5000 microns with a mean size of 2500 microns.

The feed requirements are based on the assumption that 90% of the SO_2 is removed with six times the stoichiometric feed rate of regenerated stone plus make-up. The recycle and feed make-up rates were based on the assumption that 3% of the limestone would be elutriated with the fly ash and 10% would be rejected in the regeneration. The stone after regeneration enters the boiler at 1900°F and has a heat of reaction of 3×10^6 btu/ton of $CaSO_4$ produced.

2.3 Fluidized Bed Specifications

A fluid bed superficial space velocity of 10-15 ft/sec at 1600°F was specified. The allowable velocity was based on the average particle size of the bed. It was selected sufficiently below.

TABLE 2.1
SPECIFICATIONS OF
OHIO PITTSBURGH NO. 8 SEAM COAL

(Source of data: USBM, Pittsburgh, Pa.)

SAMPLE: Run of Mine -

PROXIMATE ANALYSIS (wt %):

Moisture	3.3*
Volatile Matter	39.5
Fixed Carbon	48.7
Ash	8.5
	100.0

ULTIMATE ANALYSIS (wt %)
(includes moisture)

C	71.2	
H	5.4	
O	9.3	
N	1.3	
S	4.3	(∿60% organic, 40% pyritic)
Ash	8.5	

GROSS HEATING VALUE: 13,000 Btu/lb

NET HEATING VALUE: 12,500 Btu/lb

ASH ANALYSIS (wt %):

SiO_2	45.3
Al_2O_3	21.2
Fe_2O_3	27.3
TiO_2	1.0
P_2O_5	0.11
CaO	1.9
MgO	0.6
Na_2O	0.2
K_2O	1.8
SO_3	0.7
	100.1

FUSIBILITY OF ASH: Reducing Atmosphere

Initial Deformation Temperature -- 2080°F
Softening Temperature -- 2230°F
Fluid Temperature -- 2420°F

PARTICLE DENSITY: Coal -- ∿1.4 gm/cc
Ash -- ∿2.8 gm/cc

GRINDABILITY (Hardgrove): 50-60

FREE SWELLING INDEX: 5-5.5

COST: \$5.50-5.75/ton for ∿4% S coal, 13,000 Btu/lb
(\$6.00/ton cost projected by end of 1970)

*Possible Pick-up in Storage and Handling 6.7%
Giving Maximum Total Moisture as Received of 10%

TABLE 2.2

SORBENT SPECIFICATIONS

BCR 1359 DOLOMITE

COMPONENT	% WEIGHT AS RECEIVED
SiO_2	0.78
Al_2O_3	0.15
Fe_2O_3	0.25
MgO	45.0
CaO	53.0
TiO_2	0.02
SrO	0.03
Na_2O	<0.02
K_2O	<0.1
MnO_2	<0.03

Sorbent to Boiler from Regenerator (600 MW)	83.0 tons/hr
Feed Make-Up Sorbent	7.1 tons/hr

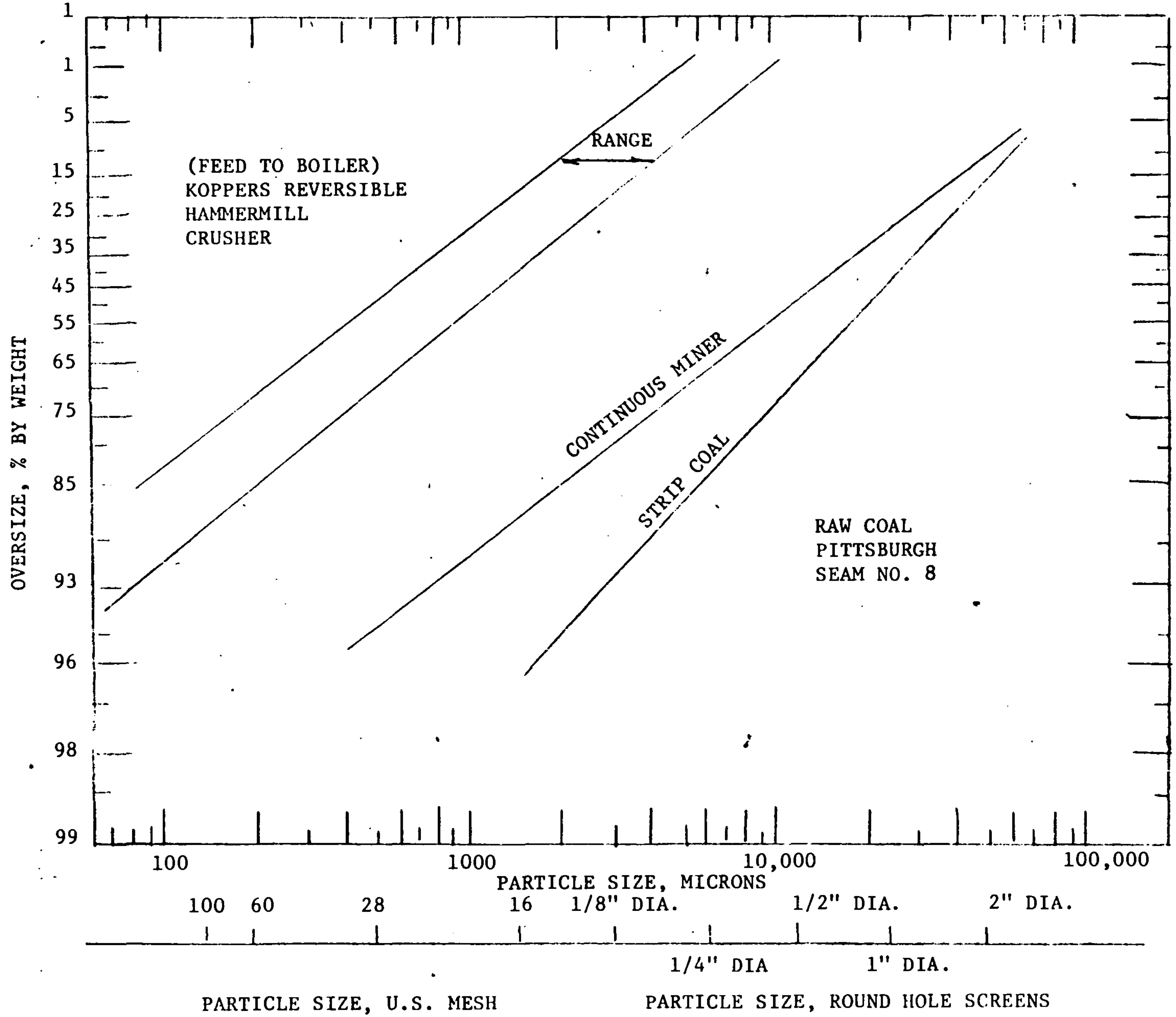


FIGURE 2.1 DISTRIBUTION OF COAL SIZE AS RECEIVED AND AFTER CRUSHING

the terminal velocity to prevent loss of the bed and sufficiently above the minimum fluidizing velocity to insure fluidization of the bed. The ranges of fluidizing and terminal velocities with particle size is illustrated in Figure 212. The material elutriated from the bed consists of 33.75% ash, 59.45% Coal and 6.8% limestone with a size distribution as illustrated in Figure 2.3.

The coal combustion efficiency in the primary beds is assumed to be 87%. 10% excess air is specified. The particle size of the feed required is 1/4" x 0" with a distribution as illustrated in Figure 2.1.

The carbon burn up cell operates at ≈1900°F with sufficient excess air to maintain the temperature level. No heat transfer surface is installed in the bed. The superficial fluidizing velocity is 9.2 ft/sec. The fuel to the bed contains 59.45% carbon. The remaining portion is ash and limestone. Material elutriated from the bed has a size distribution as shown in Figure 2.4. It consists of 79.5% ash, 18.7% coal and 1.8% limestone.

Air preheat temperature would be in the vicinity of about 700°F. Heat losses from the boiler would include 0.18% radiation loss, unaccounted for and manufacturer's margin 1.50%, unburned combustible 2.39%, dry flue gas loss 6.50%, and total moisture loss of 5.15%. The boiler efficiency would be 84.78%.

The stack gas temperature was set at 340°F.

Two stages of particulate removal equipment are required, each with an efficiency of 96%. The grain loading of the stack gas emissions should not exceed 0.01 grains per standard cubic foot.

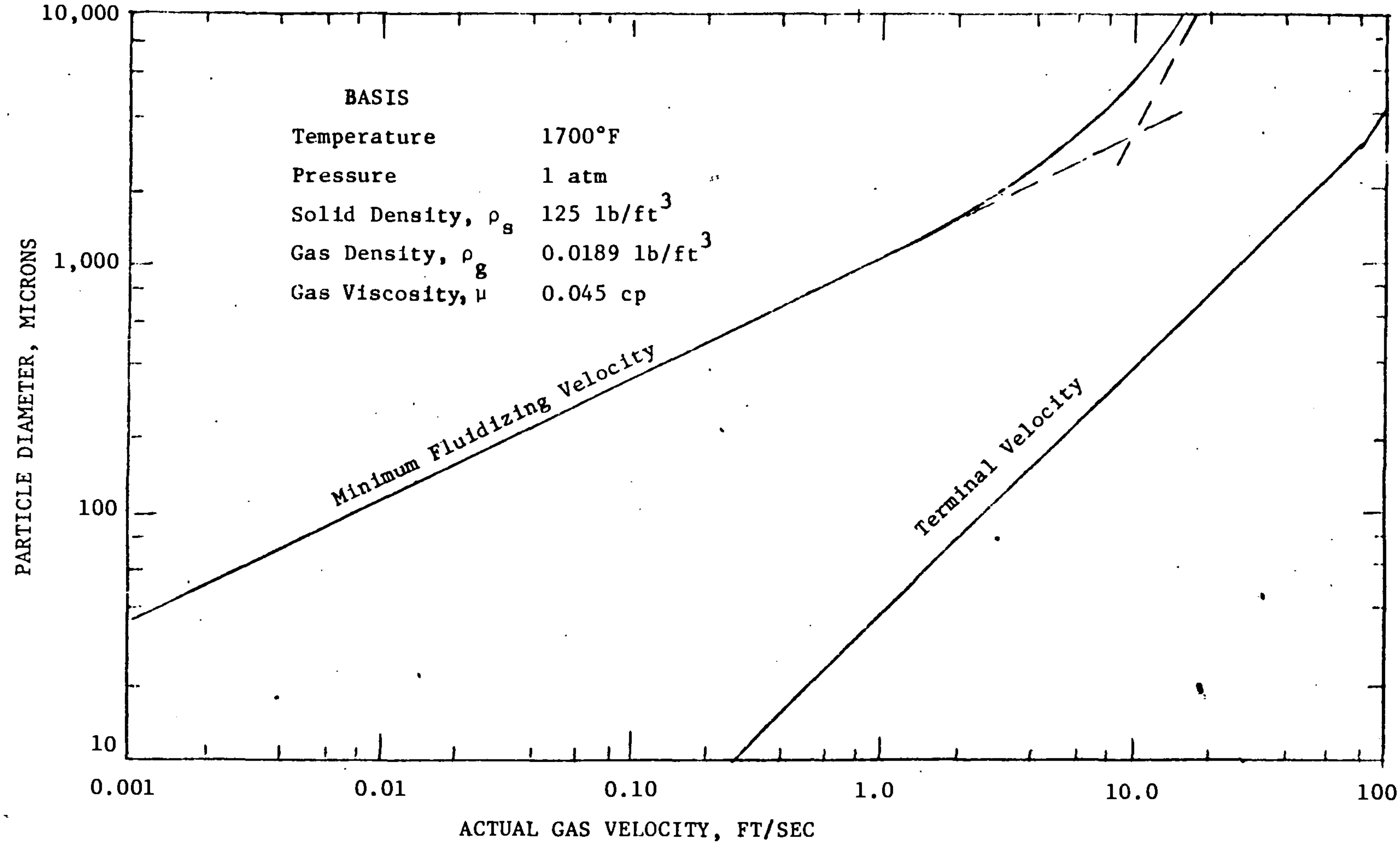


FIGURE 2.2 FLUIDIZING VELOCITY AND TERMINAL VELOCITY FOR 1 ATM SYSTEM

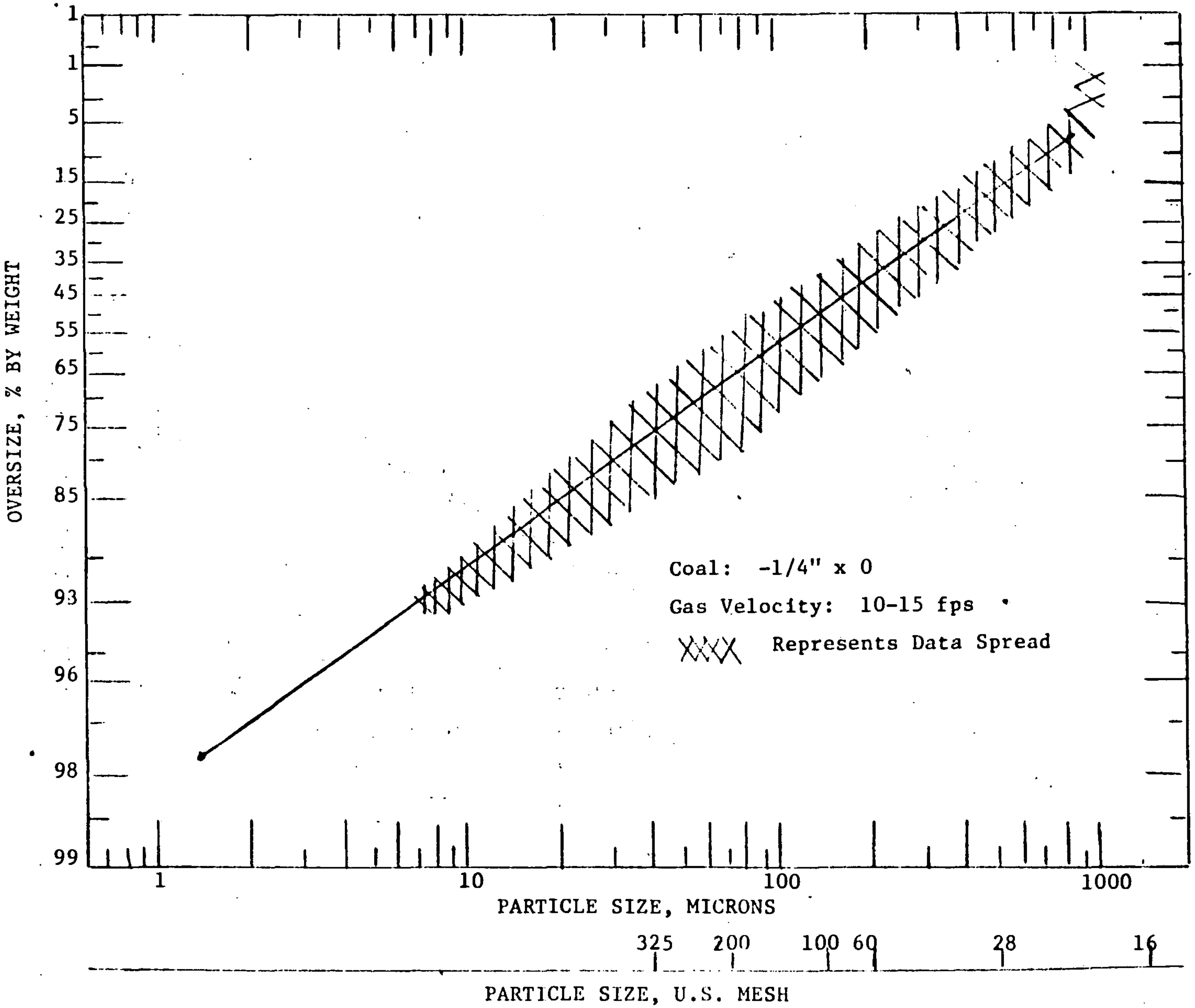


FIGURE 2.3 PROJECTED PARTICLE SIZE DISTRIBUTION FOR MATERIAL ELUTRIATED FROM THE FLUIDIZED BED COMBUSTOR.

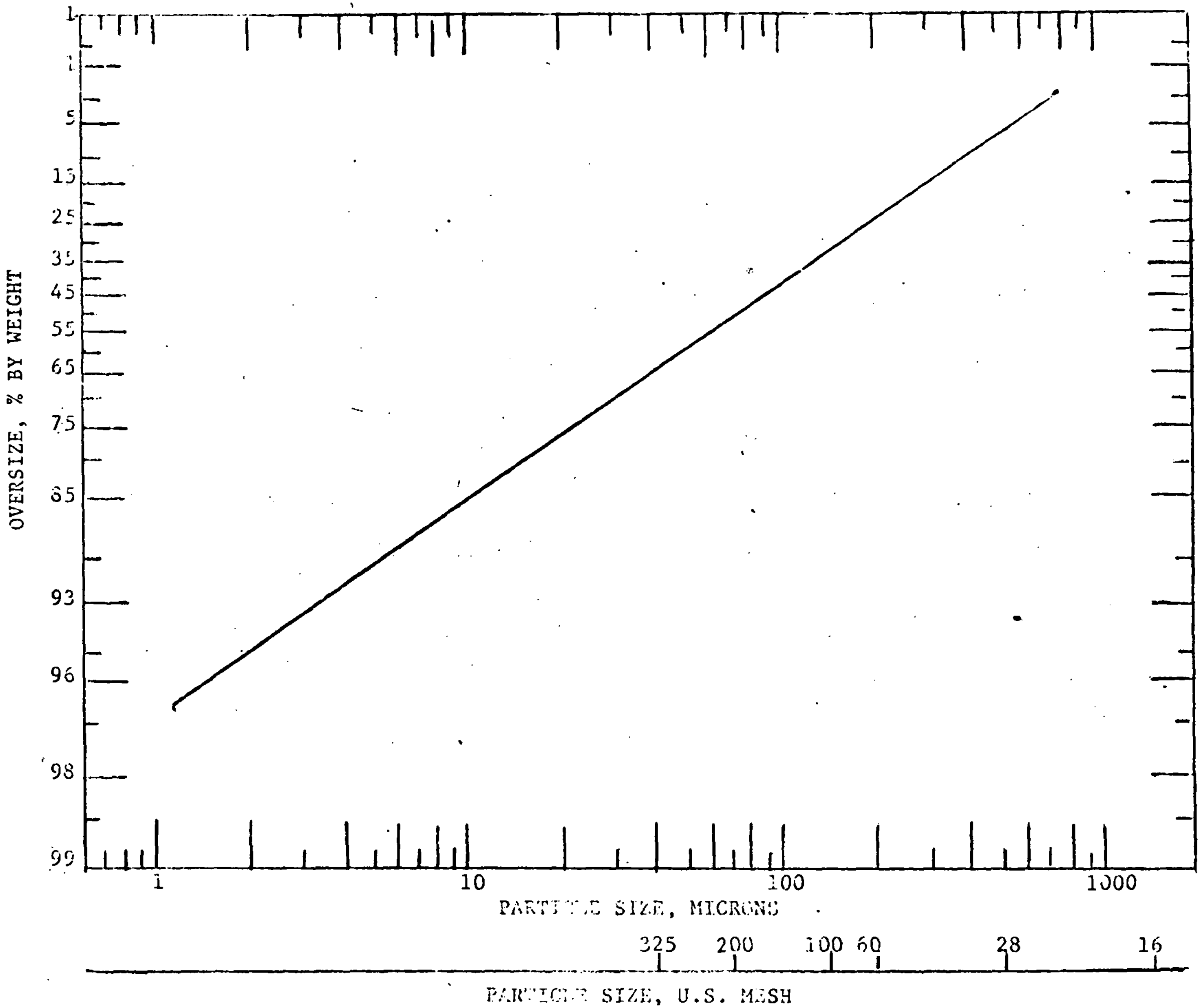


FIGURE 2.4 PROJECTED PARTICLE SIZE DISTRIBUTION FOR MATERIAL ELUTRIATED FROM THE CARBON BURN-UP CELL

No specifications were set for bed depth. It was felt that pressure drop requirements would limit it to about 2 1/2 feet. An overall coefficient of heat transfer of 50 was selected based on the experience of the British and U. S. work. A heat transfer coefficient of 40 was assumed in the 3 foot space above the bed due to combustion of volatile matter and partial elutriation of larger particles. The heat absorbed in this zone is credited to the bed. Tube spacing and size was left open to the designer. Fluidized-bed combustion data indicate that corrosion, erosion and slagging problems either do not exist or can be overcome.

2.4 System Specifications

The capacity of the proposal type design was specified as 280 MW with the intent of extrapolating costs to the 300 and 600 MW level. Steam cycle conditions selected were the same as Georgia Power and Light's Hammond No. 4 conditions. They are tabulated in Table 2.3. The preliminary design was to include coal handling plant, fuel injection system, steam generator, fans, heat recovery system and particulate removal system.

The plant turn down ratio was specified as 4 to 1 with a response rate to load changes of at least 5% of load per minute.

TABLE 2.3

STEAM CYCLE CONDITIONS

300 MW ATM. FLUIDIZED BED BOILER

Primary Steam Flow	1.9×10^6 lb/hr
Steam Pressure, Superheater Out	2400 psig
Reheater In	601 psig
Reheater Out	581 psig
Steam Temperature, Superheater Out	1000°F
Reheater In	650°F
Reheater Out	1000°F
Steam Flow (Reheater)	1.6×10^6 lb/hr
Boiler Feed Water Temperature	480°F

3. BOILER DESIGN CONCEPT

3.1 General Considerations

A 2400 psig once-through unit with cycle conditions similar to Hammond No. 4 was considered as the basis for design. The once-through unit was selected for its inherent flexibility with regard to steam heating circuitry, thereby offering a greater freedom in surface arrangement and location. The once-through boiler offers an improvement in cycle efficiency and eliminates the need for a costly steam drum. The improvement in cycle efficiency over the natural circulation boiler is due to higher operating temperatures clearly illustrated by the diagram of Figure 3.1.

The subcritical cycle offers several advantages over the supercritical cycle. These include lower tube metal temperatures associated with lower steam temperatures (See Figure 3.1), some improvements in mean temperature difference between the gas and the steam and less severe operating conditions during turn down. In addition, lower alloy steels can be used. Although the once-through subcritical units are frequently plagued with departure from nucleate boiling causing burnout of heating tubes, the problem should be minimized in the fluid bed boiler where steam mass flows approach 1,500,000 lb/hr-ft^2 and heat flux approach 50,000 Btu/hr-ft^2 at full load operation. This is illustrated in Figures 3.2, 3.3, and 3.4.

Physically, the boiler consists of four modules, each containing five fluid beds and a carbon burn-up cell. This arrangement should simplify start-up, shut-down and partial load operation. It also facilitates turning down 4 to 1 by cutting one boiler

TEMPERATURE-ENTHALPY DIAGRAM
FOR STEAM AND WATER

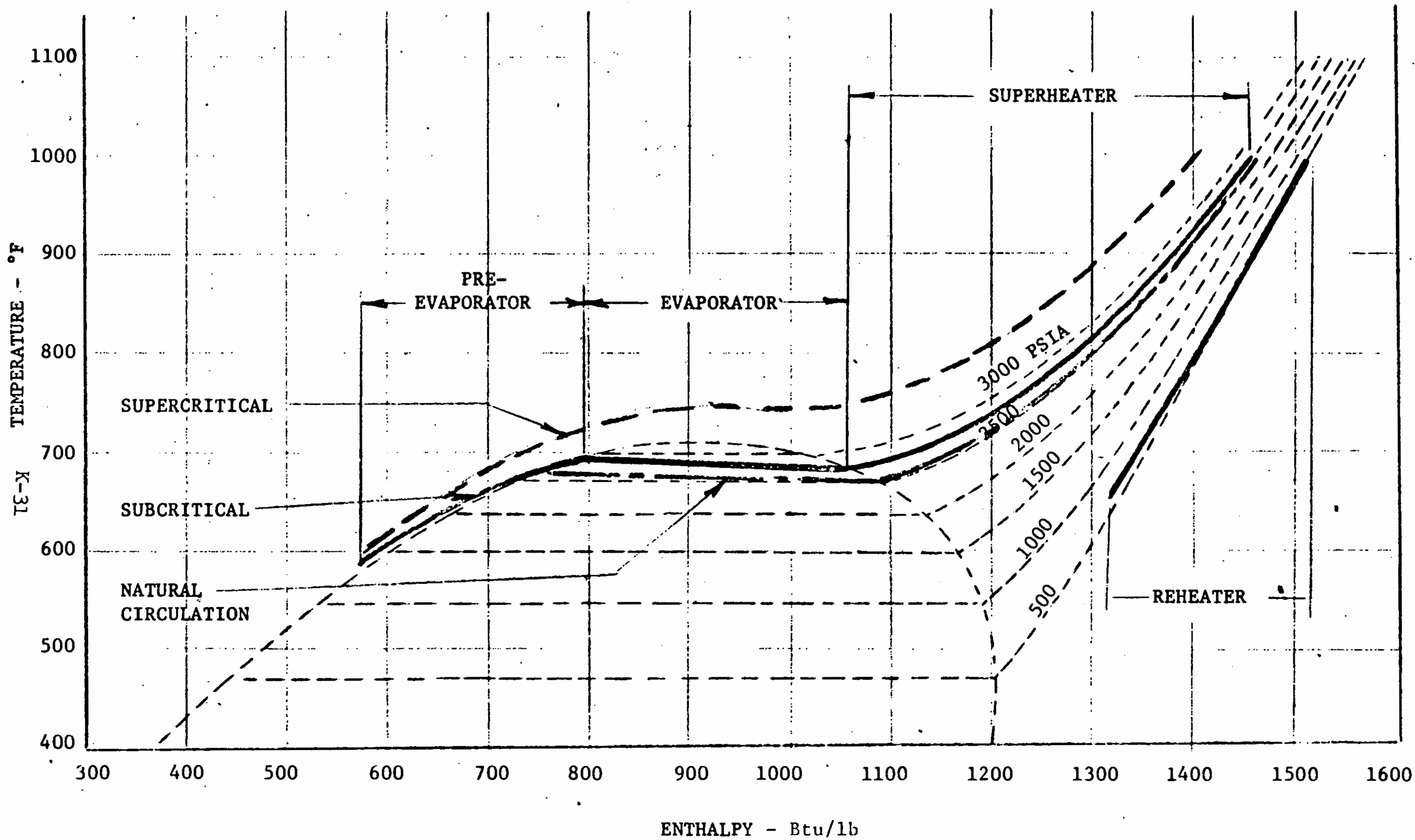


FIGURE 3.1

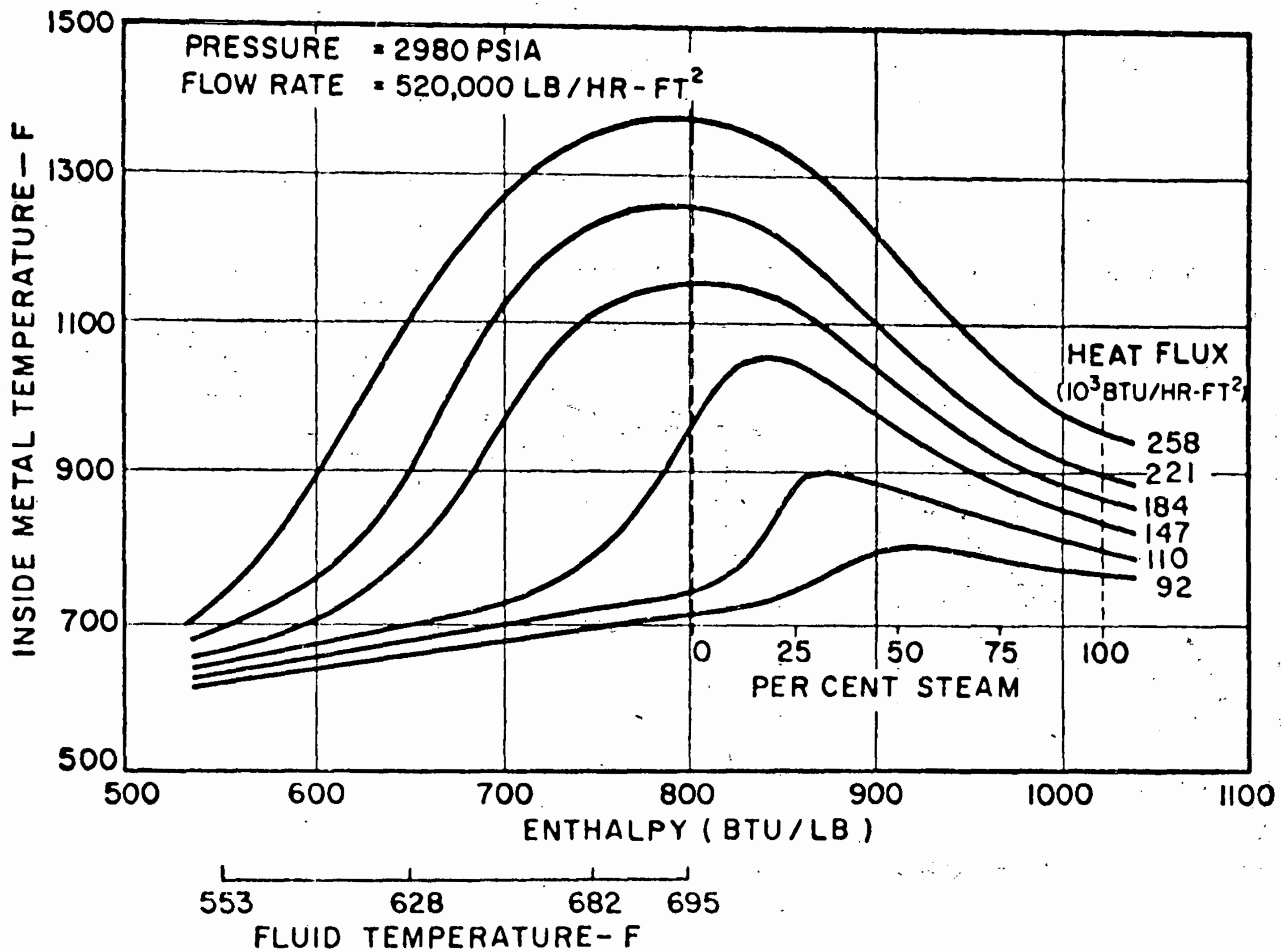


FIGURE 3.2 EFFECT OF CHANGE IN HEAT FLUX ON BURN-OUT AND DEPARTURE FROM NUCLEATE BOILING POINTS (REF: 1)

(1) Glenn R. Fryling, "Combustion Engineering", Combustion Engineering Incorporated, 1966.

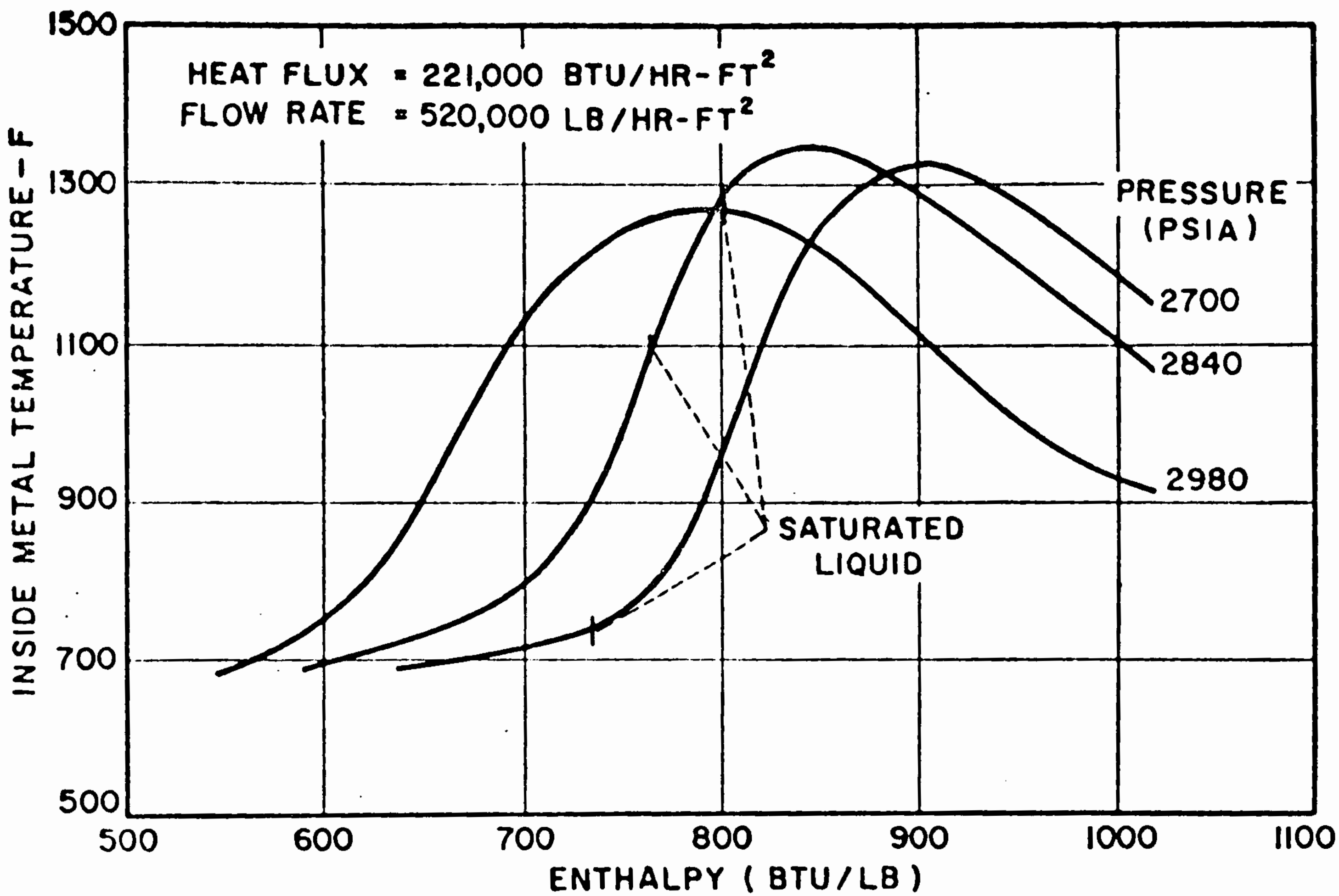


FIGURE 3.3 EFFECT OF CHANGE IN PRESSURE ON BURN-OUT AND DEPARTURE FROM NUCLEATE BOILING POINTS (REF: 1)

(1) Glenn R. Fryling, "Combustion Engineering", Combustion Engineering Incorporated, 1966.

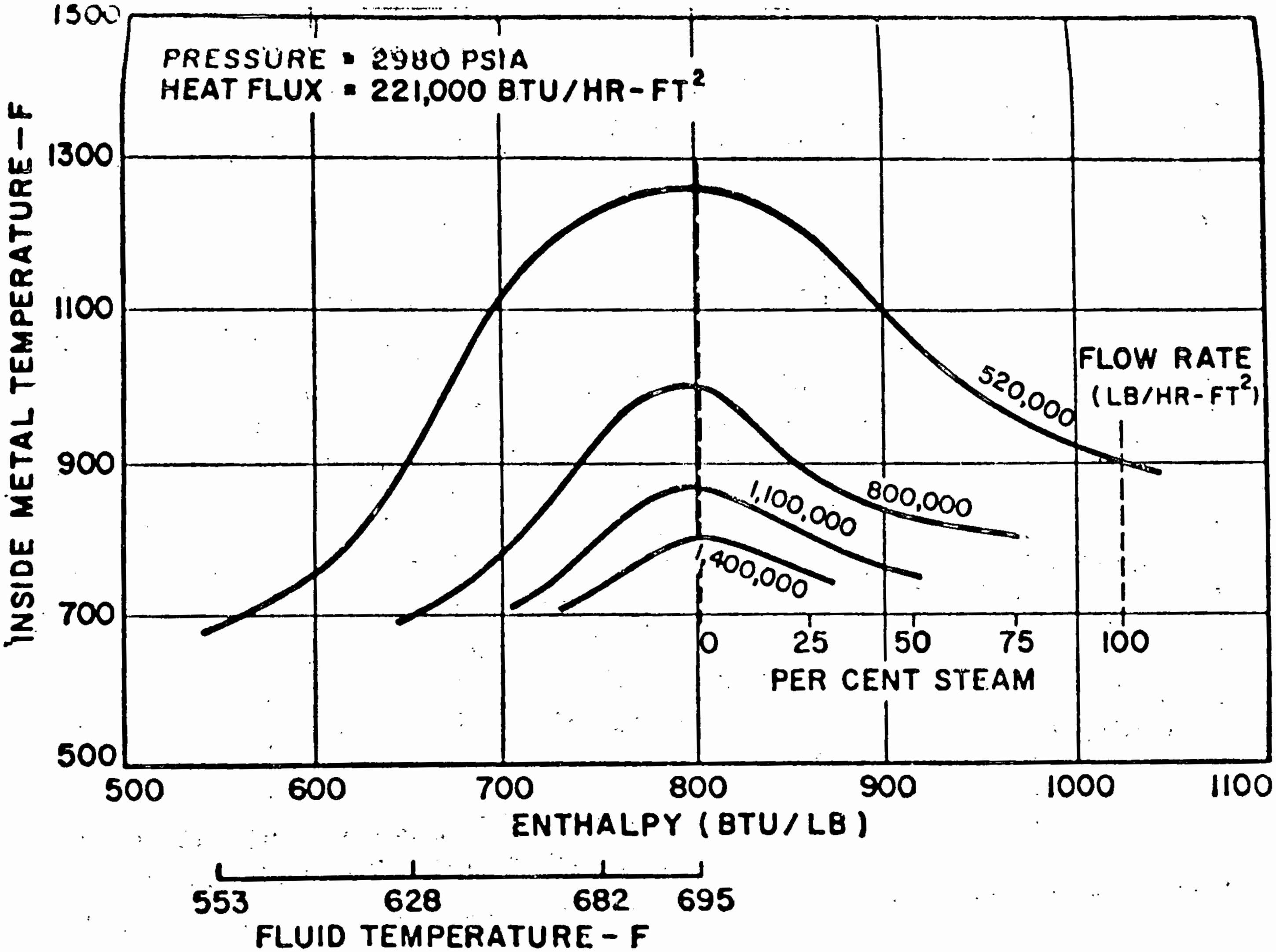


FIGURE 3.4 EFFECT OF CHANGE IN FLOW ON BURN-OUT AND DEPARTURE FROM NUCLEATE BOILING POINTS (REF: 1)

(1) Glenn R. Fryling, "Combustion Engineering", Combustion Engineering Incorporated, 1966.

module out of service. Two of the beds are assigned to evaporating functions, two to superheating and one to reheating. The enclosure walls are comprised of evaporating tubes. Preheating of feedwater normally assigned to the economizer is accomplished in the convection pass. The schematic in Figure 3.5 illustrates the concept as it might look in a vertical arrangement. Actually several geometric orientations were considered.

The fluid beds for a four module construction are nearly square, with dimensions of 2 1/2' x 13' x 12'. The depth is presently limited by pressure drop. Future concepts incorporating "high-pressure" fans may justify the use of deep beds. This approach represents a significant deviation from normal practice and is beyond the scope of the present study. Rectangular or square beds are used to facilitate installation of conventional banks of heat exchange surface, making use of presently available fabricating techniques. The approximately square shape is a compromise between steam side pressure drop and mass flow requirements, and shipping tolerances.

Three basic bed arrangements were considered:

1) Vertically stacked beds in a square array.

2) Vertically stacked beds in separate modules in an in-line arrangement.

3) Horizontal tandem arrangement with four modules in parallel.

In the first arrangement the five beds in each module are stacked one above the other. Air is introduced to the beds by way of a common duct on one side of the module. Gas is withdrawn on the opposite side in a similar manner. By arranging the four

NOTE: FBC - FLUID BED CELLS
CBC - CARBON BURN-UP CELLS

BEDS CONTAIN SUPERHEATING, REHEATING AND EVAPORATING SURFACE

CONVECTION SURFACE - FEEDWATER HEATING SURFACE

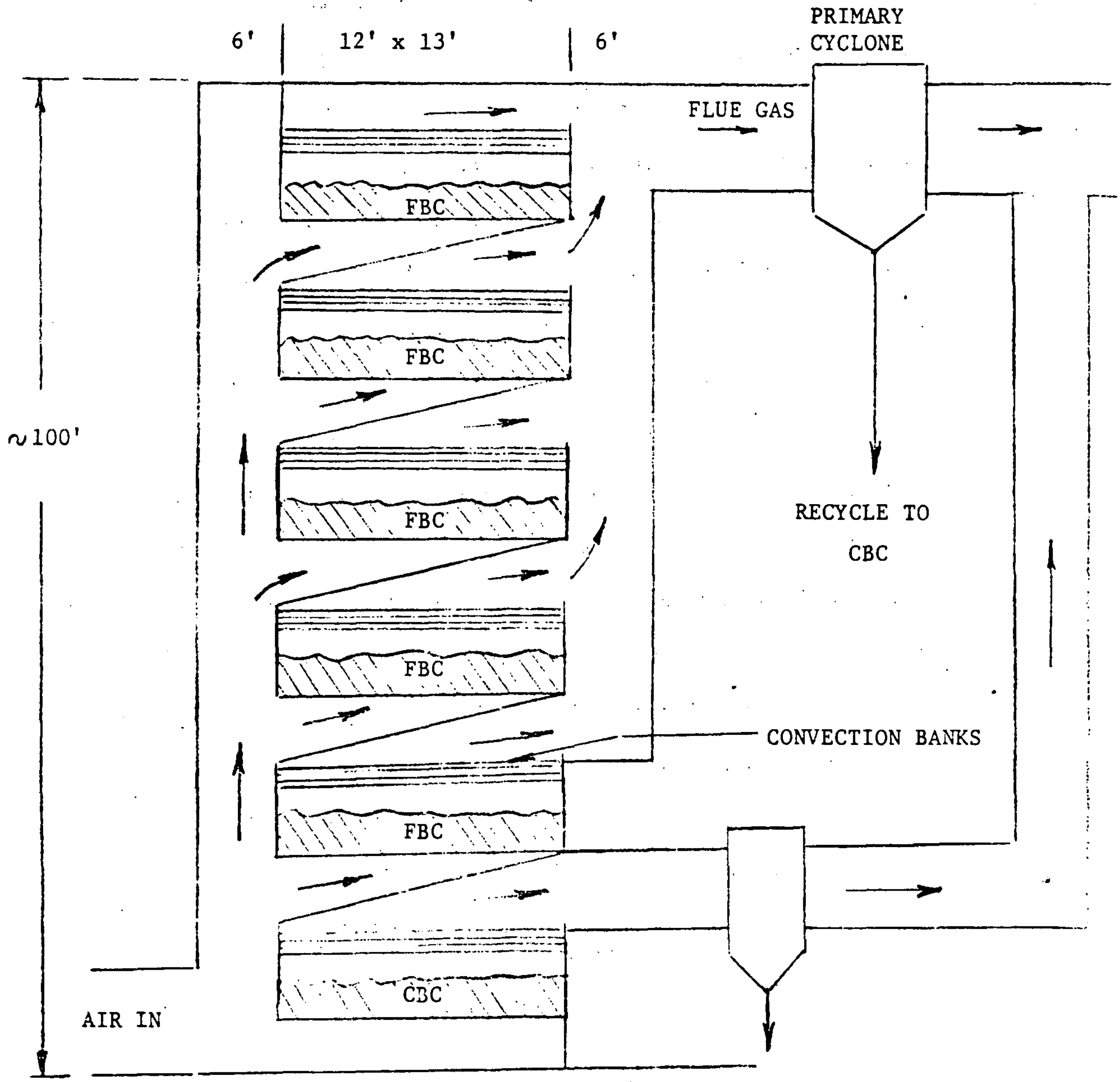


FIGURE 3.5 CONCEPTUAL ARRANGEMENT OF ATMOSPHERIC UTILITY BOILER (1 MODULE OF 4)

modules in a square pattern with each module sharing two adjacent sides with two other modules it was hoped some savings in duct work and boiler structure might be achieved. This arrangement, however, was dismissed for several reasons.

1) Sharing common walls or partitions offered little savings in construction costs and no solution was forthcoming on how to isolate a shut down module.

2) Shared partition walls can present thermal expansion problems and circulation problems.

3) It is almost impossible to arrange auxiliary equipment symmetrically about the boiler without introducing complicated duct work and piping.

The horizontal arrangement is illustrated in Figure A1-2 of the appendix. The horizontal arrangement hopefully would reduce structural steel requirements and facilitate shop fabrication. After a brief but careful evaluation of this arrangement it was dismissed for the following reasons.

1) The number of headers and downcomers required is almost quadrupled. This introduces a very complex, costly piping system.

2) Downcomers and headers could be reduced by introducing horizontal tubing in the water walls. This may not create any unusual problems for a single phase fluid. However, in the evaporating zones the possibility of the separation of phases which may result in tube failure is greatly enhanced unless some basic modification is made to the system such as turbulators, forced recirculation, etc. Expansion and circuitry problems may also exist due to the differences in heat flux in and above the bed.

3) The horizontal unit virtually dictates the use of a regenerative air heater with its high leakage loss at the higher pressures associated with the fluid bed. In the vertical arrangement a vertical tubular air heater is compatable with the stacked fluid beds and there is no air leakage.

4) The head-room below the mechanical dust collectors is small, making it impossible to install a suitable particulate handling system for the carbon burn-up cell fuel feed injector without losing some of the advantage of reduced height.

5) The clearances on the beds violate shipping tolerances, necessitating field erection.

6) Isolating plenum chambers below bed grids for the installation of dampers is complicated. Large areas may have to be sealed off to accommodate air control dampers.

The vertically stacked bed in an in-line arrangement as shown in Figures 3.5 and A1-1 appears to have overcome most of the liabilities cited for the other two systems and, therefore, was selected as the concept to develop.

The accessories accompanying the fluid bed steam generator include a conventional coal handling system, a modified fuel injection system, heat recovery equipment, fans and particulate removal equipment.

In the basic concept the coal handling system includes receiving, crushing, conveying and storage of coal.

The fuel injection system includes pressurizing coal to furnace pressure, drying, distributing and transporting to the fluid beds.

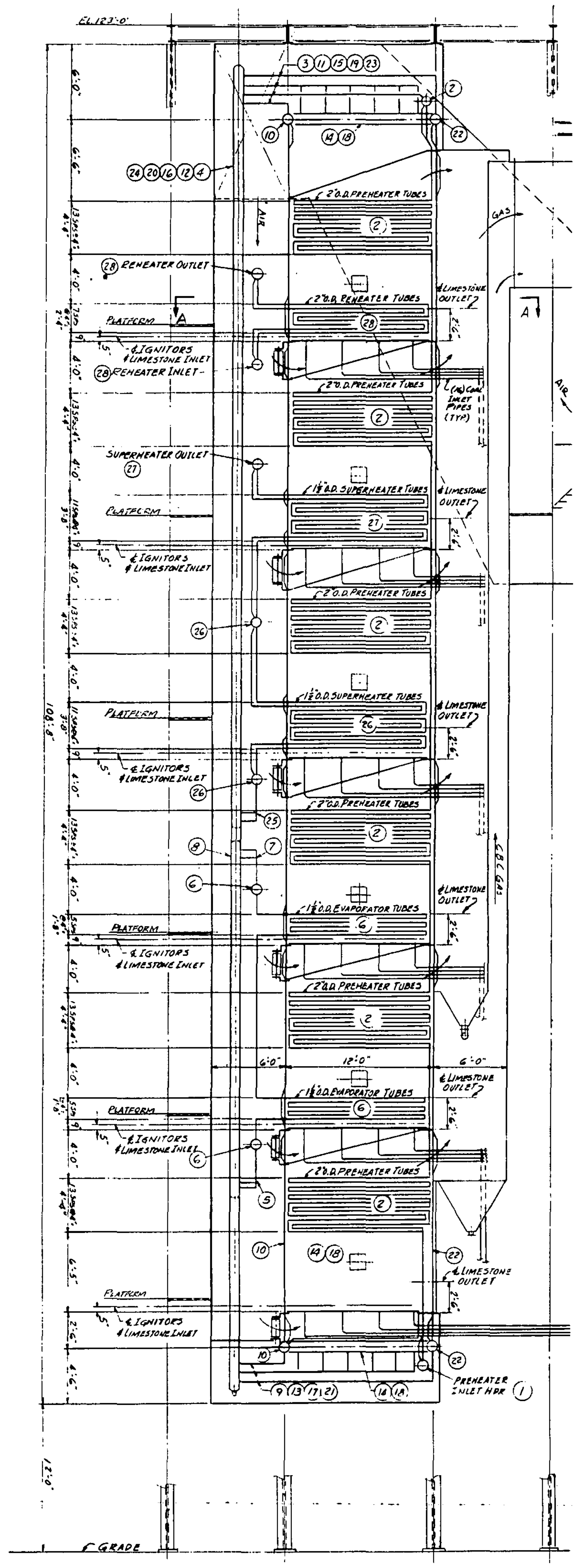
The heat recovery equipment is required for the exchange of heat from the flue gas to the incoming air to reduce thermal losses to the stack, to provide heat for drying the coal and to insure the beds are not quenched by cold air resulting in the loss of ignition.

The particulate recovery equipment is divided into two stages. The first stage recovers elutriated material rich in carbon from the fluid beds in order that it may be returned to the carbon burn-up cell. It also collects ash elutriated from the carbon burn-up cell which is removed from the system. The two processes are structurally integrated but functionally segregated, thereby preventing contamination of either stream. A mechanical dust collector is proposed for the first stage of particulate recovery. The second stage of recovery is strictly flue gas clean-up. An electrostatic precipitator is proposed for this job.

3.2 Preferred Steam Generator Concept

The vertically stacked bed arrangement selected as the preferred concept is illustrated in Figures 3.6, 3.7 and 3.8. The steam generator consists of four separate modules, each part of four completely separate but parallel processes from the coal handling plant to the electrostatic precipitators.

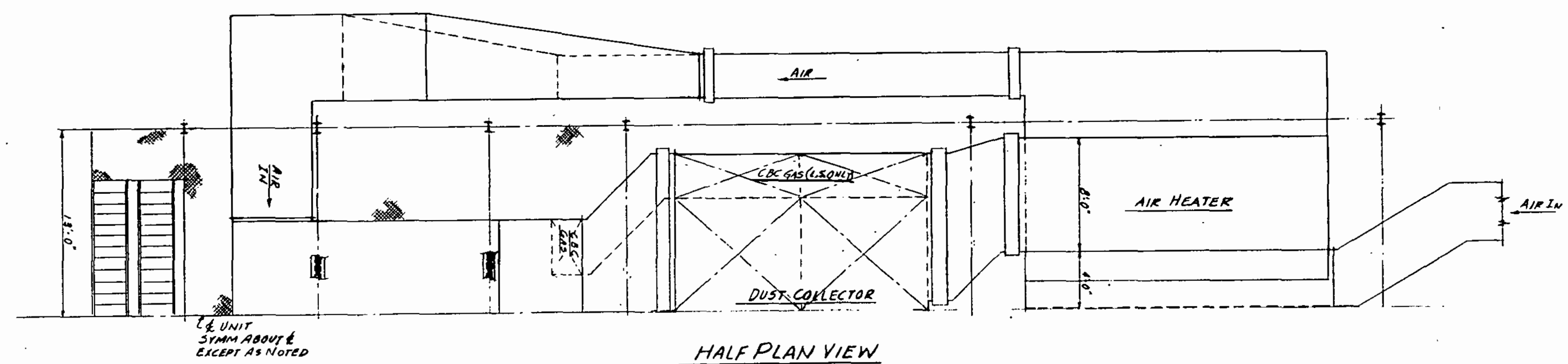
The steam generator contains five fluidized beds and a carbon burn-up cell stacked one above the other. The lowest bed is the carbon burn-up cell which receives elutriated material from the other fluidized beds collected by the mechanical dust collector. Directly above the carbon burn-up cell are two evaporating beds, two superheater beds, and finally the reheater bed. The fluid



ELEVATION

VERT. ARRGT. OF PRESS. PARTS
ATMOSPHERIC FLUIDIZED
BED STEAM GENERATOR
FOR
300 MW PLANT
FOUR (4) REQUIRED

FIGURE 3.6

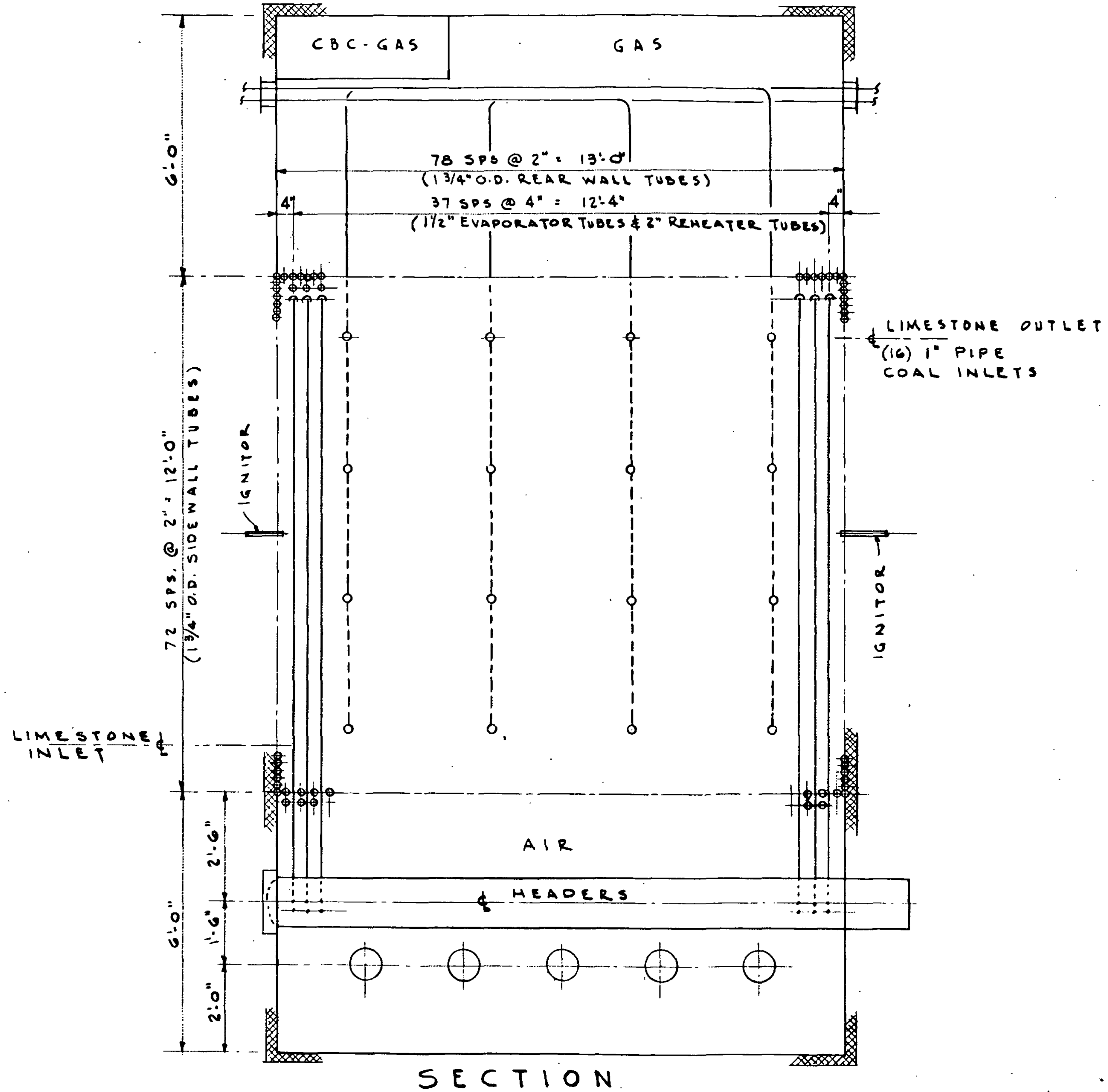


HALF PLAN VIEW

PLAN VIEW
ATMOSPHERIC FLUIDIZED
BED STEAM GENERATOR
FOR
300 MW PLANT

FOUR (4) REQUIRED

FIGURE 3.7



ATMOSPHERIC FLUID BED BOILER

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Fig. 3.8

beds are enclosed by welded fin tube walls which carry evaporating water. Each fluid bed cell contains a convection bank which cools the hot flue gases before they leave the steam generator and screens out some of the large particulate material elutriated from the bed.

Air from the air preheater passes through two ducts, one on each side of the boiler as shown in the half plan view in Figure 3.7, to a common air duct which runs the full height of the boiler. Portions of the air feed into each plenum chamber located below the fluid beds. The quantity is regulated by dampers located in the air duct. The air then passes through the distribution plate supporting each bed and into the bed itself. A separate air stream fed directly from the forced draft fan feeds cool air to the CBC to regulate the temperature of the bed by control of the excess air introduced.

Separate gas streams leave the fluidized beds and the carbon burn-up cell, thereby separating the carbon rich ash from the spent ash. The CBC duct is located inside the flue gas duct on the back side of the boiler and extends over its full length. The gas is withdrawn at the top of the duct where it is fed to the dust collector.

Coal is pneumatically transported through sixteen separate feed lines to each bed as a dilute phase of dried coal at 250°F. Distribution and drying is accomplished in a fluid bed injector constituting part of the coal feeding system. The coal feed lines enter the gas duct from either side of the boiler, pass through the flue gas exit opening in the water wall of the preceding cell and then up into the plenum chamber and through the grid plate. Penetrations through the duct work are made on opposite sides of the boiler to which the coal will be fed. This should provide sufficient length of feed line to take up any differential in expansion of duct work and steam pressure parts. The coal is injected into the bed in the horizontal direction in an area free of tube bundles. One coal injector is provided for each ten square feet of fluid bed surface.

For steam generators 300 MW and larger the boilers will be field erected. Beds are 12' x 13' in size to permit shipment of bundles. The overall size of the boiler modules is too large for shipment in assembled sections. Furthermore, the cost of reinforcing the boiler, which is designed to hang from structural steel, for shipping purposes would nullify some of the advantage of shop assembly over field erection.

Each module requires about 10 months to construct. Offsetting construction of each module by one month to minimize total labor force required and to allow free flow of labor and materials from one module to the next would result in a total field erection elapsed time of 14 months. This represents about 18% savings in erection time over a conventional boiler of similar capacity.

3.3 Future Concepts

In the course of developing the concept selected new ideas for improvement were generated requiring less conservative design practice but hopefully resulting in an improved cost picture.

The study indicated that pressure drop across the bed and number of coal feed points required are two key problem areas with the atmospheric fluidized bed boiler. By increasing bed depth and pressure drop the number of coal feed pipes and feed injection points could be greatly reduced. This could result in a smaller bed cross-sectional area which might ultimately result in a shop-assembled, sectionalized unit. If the number of boiler penetrations required for the coal and limestone feed are reduced, the designer might consider replacing the structural steel with a self-supporting cylindrical vessel completely enclosing the pressure parts.

This approach would not only simplify shipping; it would also reduce field erection time, eliminate the need for flues and duct work, and minimize structural steel required. Since the air and gas temperatures are nearly the same during normal operation and start-up, expansion problems, sealing problems and shell distortion should be minimal. With a small pressure differential between the air and gas side (40-86 in. w.g.) sealing the two streams should not present a problem. The "pressure" vessel or boiler shell can also be insulated on the inside rather than outside to minimize heat losses and maintain a cool vessel.

Other consideration should be given to installation of the tubular air heater in the convection pass above the water or steam cooled heat recovery surface, thus providing a much more compact boiler and minimizing duct work.

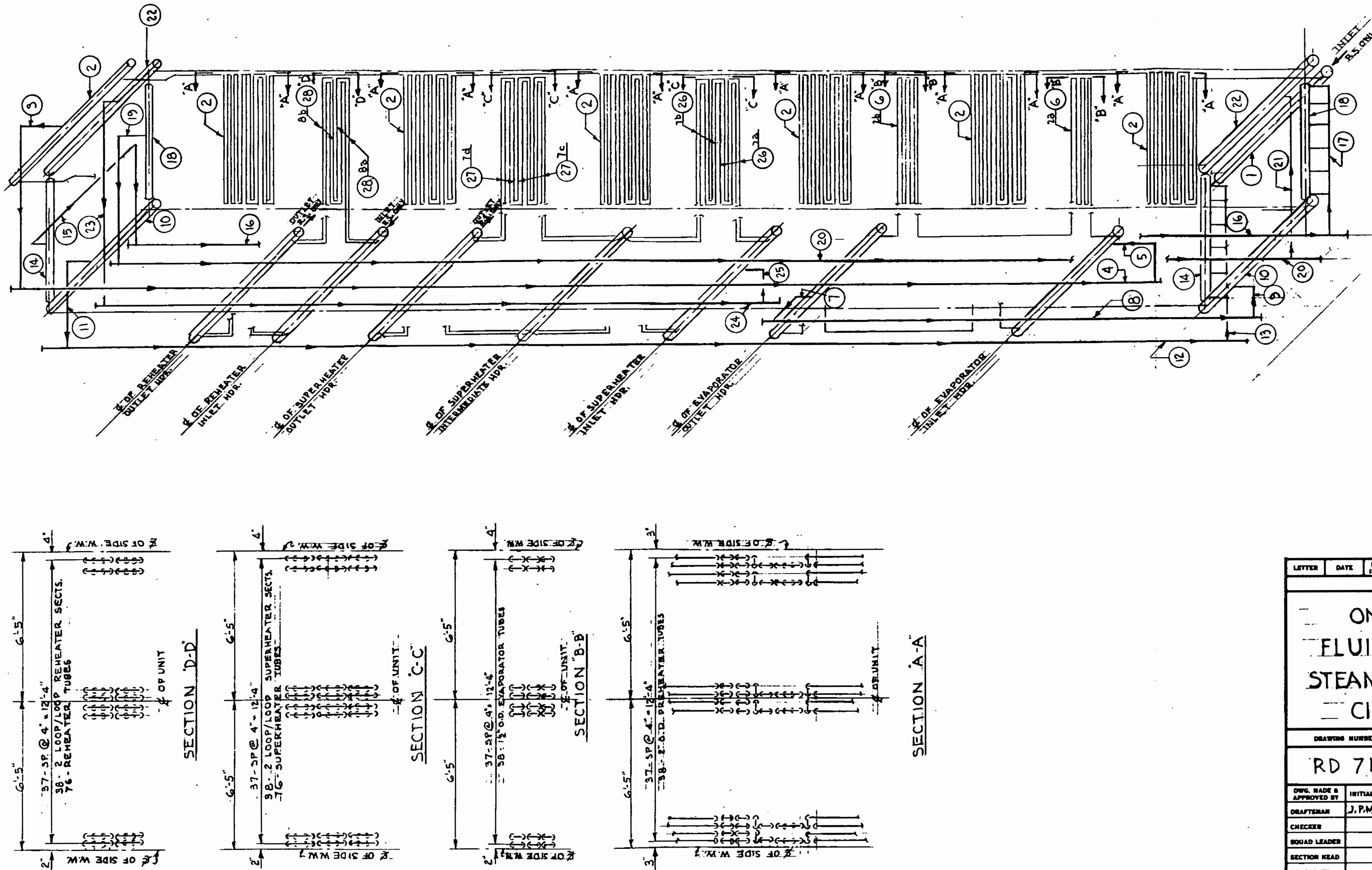
4. DETAILED BOILER DESIGN

4.1 Circulation of Steam

The heating of the water as it passes through the boiler is clearly illustrated in the temperature-enthalpy diagram in Figure 3.1. Water entering the boiler at 480°F is heated to the evaporation temperature of about 690°F in the pre-evaporator. Then the water and steam remain at approximately this temperature until all the water is evaporated. The steam is then superheated to a temperature of 1000°F at a pressure of about 2,400 psig. Some of the energy in the steam is released in the high pressure turbine after which it is returned to the boiler for re-heating to 1000°F at 581 psig. These same steps are followed in the once-through boiler in a simple, continuous path which is only interrupted at appropriate points with mixing vessels or headers to insure good distribution and uniform heating of the steam.

This system is applied to the fluid bed boiler by breaking up the heating functions into individual beds and tube bundles or enclosure walls. To minimize the surface requirements the largest mean temperature difference is maintained when feasible. To do this the preheating or feedwater heating is accomplished in the convective passes. The first stage of evaporation takes place in the tube bundles. It is completed in the enclosure walls. Superheating and reheating takes place in tube bundles located in the beds.

The circuitry may be simply followed on the isometric flow diagram illustrated in Figure 4.1. Water at 480°F enters the feedwater inlet header at point (1). It proceeds through a series of banks of preheater tubes (2) located in each convection pass of each fluid



LETTER	DATE	DR. INIT.	DESCRIPTION
REVISIONS			

ONCE THRU FLUIDIZED BED STEAM GENERATOR CIRCUITRY

DRAWING NUMBER	SCALE:	= 1'-0"
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DWG. MADE & APPROVED BY	INITIAL	DATE	CONT. NO. 0-59-1192
DRAFTSMAN	J.P.M.	4-29-71	ORDER NUMBERS
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SQUAD LEADER			
SECTION HEAD			
CONT. SUPV.			

bed including the CBC. The bundles are all connected in series. The water leaves the outlet header (2) at the top of the boiler and proceeds through some feeder tubes (3) to a downcomer (4) which carries it to the first stage of evaporation (5). Evaporation takes place in two separate bundles (6) located in two adjacent beds. The bundles are connected in series without the benefit of an intermediate mixing header. The partially evaporated steam leaves the bundles through feeder tubes (7) to the downcomer (8) which feeds the front wall header (10) by a second series of feeder tubes (9).

The mixture passes up the entire evaporating wall (10). At the top of the boiler the mixture is collected in header (10), and fed by means of a series of feeder tubes (11) to a downcomer (12) which carries the mixture to the side wall header (14). All four walls are connected in series as described for the front wall. At header (22) the evaporated water is collected and passed through a series of feeders (23) to a downcomer (24) which carries the steam to the first bank of superheater tubes (26) located in a fluid bed.

Superheating is completed in a second bank (27) connected in series with the first and located in a separate bed. The connecting tubes are interrupted by a mixing header. The steam leaves the second bundle and the steam generator through a single-ended header (27).

Reheat steam enters the boiler at the header feeding bundle (28) and leaves the boiler from the header collecting steam at the opposite end of the bundle. Both headers are single ended. The bundle occupies one fluid bed.

4.2 Air and Gas Circuitry

Pressurized firing was selected over a balanced draft system for the fluidized bed boiler. In conventional practice selecting one over the other is very much dependent on the personal preference of the customer. Pressurized firing offers the advantage that a single set of fans handle only clean, low-temperature air and thus have reduced maintenance. Controls are also simplified. It is readily adaptable to the new furnace wall construction which is gas tight. With pressurized firing plant maintenance frequently goes up. It also requires a pressure-tight boiler and air or steam lancing or furnace probing becomes more difficult. In the case of the fluid bed a gas tight enclosure must be used to contain the bed and accomodate the relatively high gas side pressure differentials that are proposed. There is no need for lancing or probing in the fluid bed boiler. Therefore, pressurization seems a natural choice.

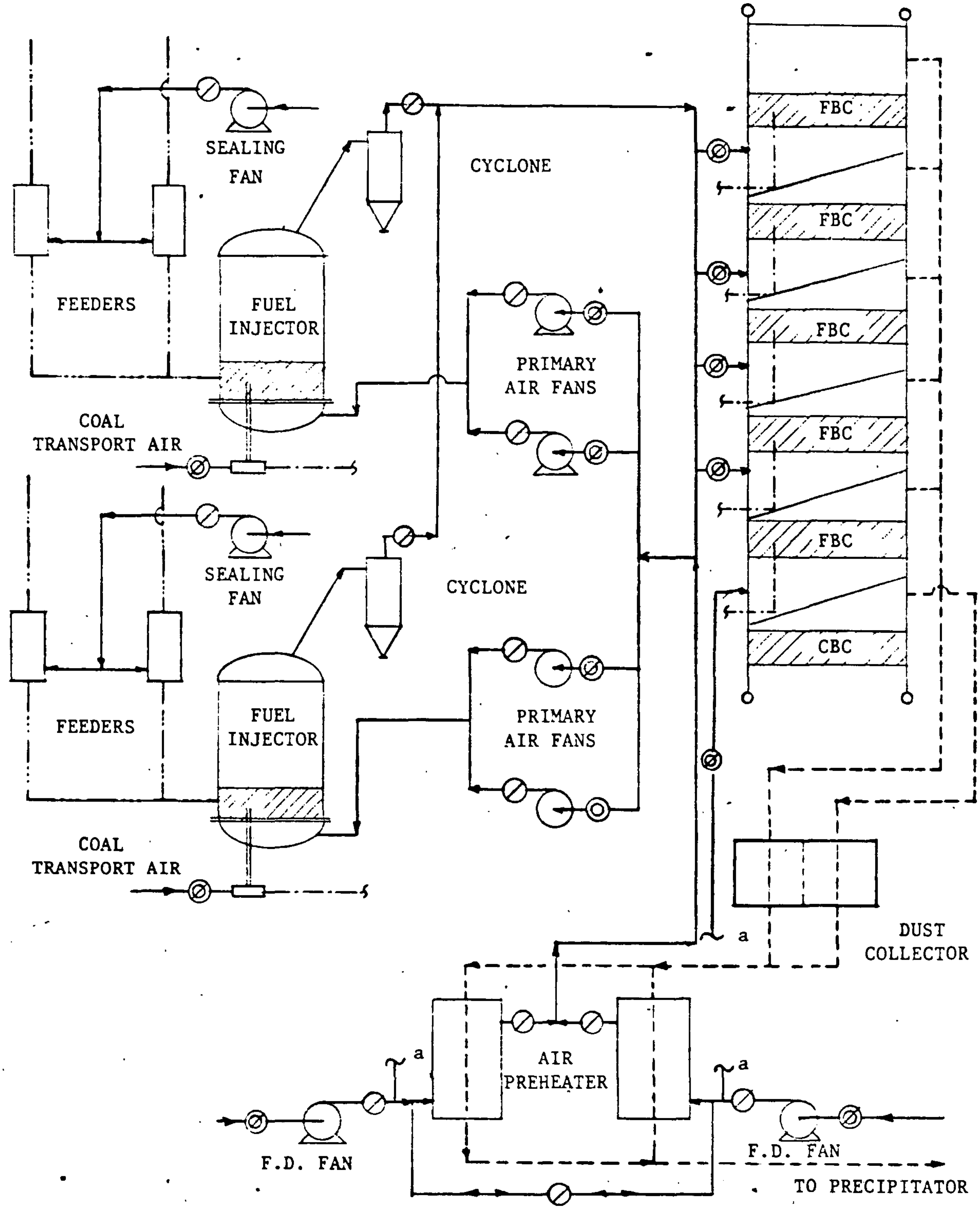
The air and gas flow circuitry for the fluid bed boiler is illustrated in Figure 4.2. Air enters the system at three points, the forced draft fan, the coal-air transport lines and the sealing air to the coal feeders. The latter two sources are minor and may be dismissed from further discussion. The primary source of air in conventional practice is referred to as the secondary air stream, as it supports combustion. To avoid confusion the nomenclature will not be changed.

The secondary air enters the system at the forced draft fan. It is heated in the air preheater to 735°F and passed on to the air ducts of the boiler where it is split into five streams feeding each one of the fluid beds. A separate cool air stream is

FIGURE 4.2

AIR FLOW DIAGRAM

(ONE MODULE OF FOUR)



LEGEND

COAL FLOW —··—

AIR FLOW ———

AIR AND COAL —·—

GAS – – –

taken off after the forced draft fan and before the air preheater and is feed to the CBC. This is done to maintain the bed temperature at 1900°F, since the CBC does not have submerged cooling surface.

A second stream of air, referred to as the primary air, is drawn off after the air preheater. Again, this is a practice carried over from conventional boiler design. This hot air is pumped to a higher pressure by primary air fans and fed to the fuel injector. At the fuel injector it passes up through a grid plate and through a bed of fluidized coal where it acts as a drying and fluidizing agent for the coal. The air, cooled to 250°F, leaves the fuel injector laden with coal fines which are subsequently removed by a Stairmand type single cyclone. The clean air is returned to the secondary air stream.

The combustion gas leaving the beds is withdrawn from the boiler through a common gas duct and passed through a mechanical dust collector where it joins a gas stream from the CBC which has undergone a similar process. The united streams pass through the air preheater and electrostatic precipitator before leaving by way of the stack.

4.3 Energy and Mass Balance

The energy and mass balances were calculated for full load and 75,50 and 25% of full load. The calculations were made on the basis of data specified in Section 2 by Westinghouse. Partial load data was calculated for reduced bed temperatures on the basis of steam duty and selected air-to-coal ratios. The bed temperautre is allowed to drop to avoid overheating of the tubes but not to a level which would risk unstable combustion or impair sulfur dioxide sorption.

The overall energy and mass balance appears in Figure 4.3 and 4.4, respectively. Partial load operation is tabulated in the appendix along with energy balances for individual beds and carbon burn-up cells.

The energy balance for individual beds was made by considering the bed a control volume and balancing the energy input and energy losses. The difference between the two or the unbalance is the heat transferred to the steam in the bed. The input. energy includes the sensible heat of the air, coal and limestone at 80°, and the heats of reaction of coal and limestone. The output or losses, includes the overall steam generator loss, and the heat loss to the convection bank. At partial load the energy input and energy losses (excluding some of the fixed losses affecting boiler efficiency) all change due to the decrease in air and coal input at constant excess air.

At full and partial load the unburned carbon loss with elutriated material accounts for a 13 percent loss in heating input tc the fluid beds.

The carbon burn-up cell is designed to be maintained at 1900°F entirely by the combustion of carbon elutriated from the fluid beds and collected by the dust collector. This amounts to 11.6% of the carbon feed to the steam generator. Bed temperature is controlled by introducing cool excess air from the forced draft fan. A portion of the heat is removed by the enclosure walls. The flue gas leaving the bed is cooled by a feed preheating convection bank. At reduced loads the CBC temperature is held constant to avoid a loss in combustion efficiency. This is accomplished by reducing the excess air requirements.

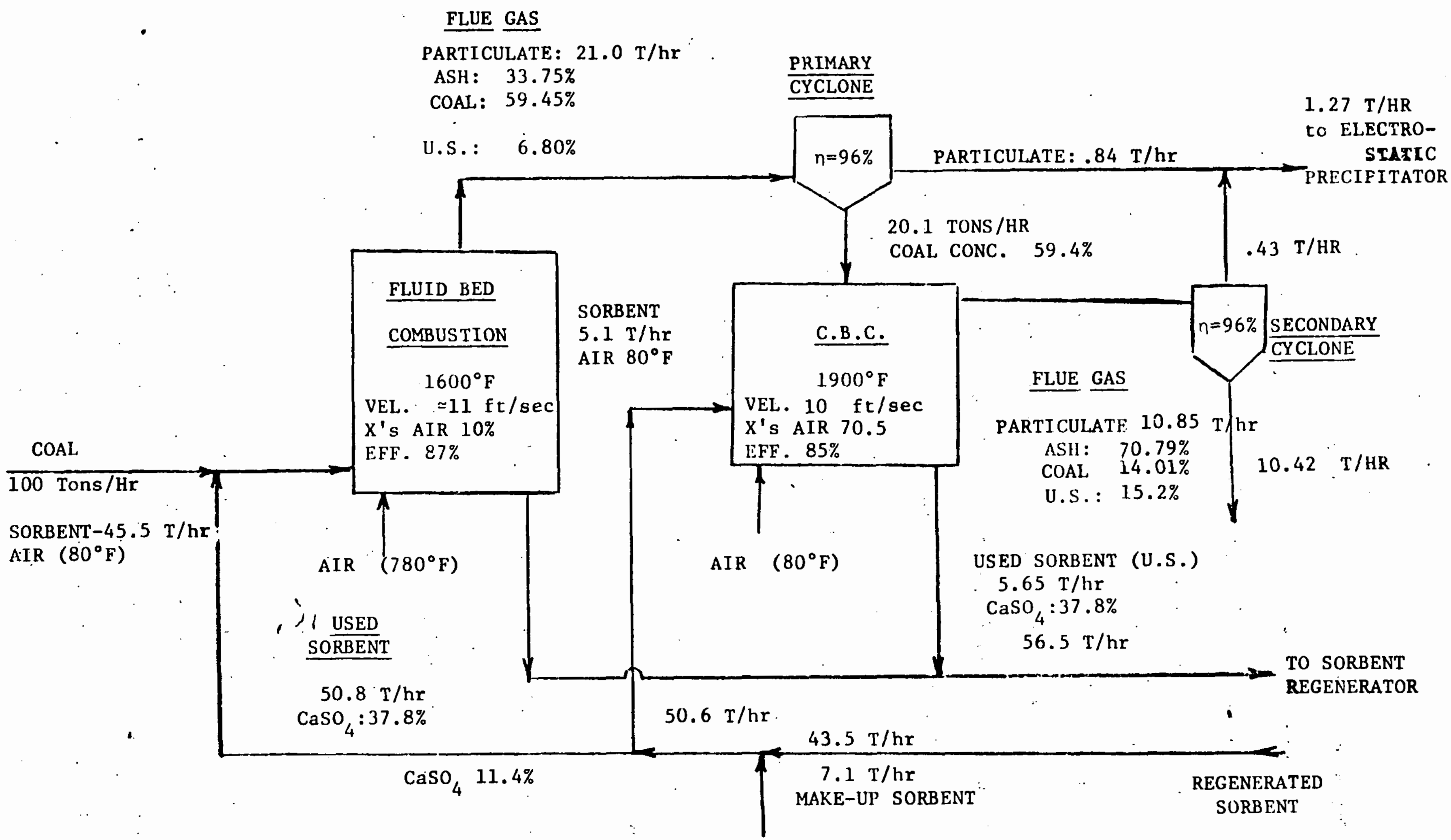


FIGURE 4.3 OVERALL MATERIAL BALANCE - FULL LOAD

(FOUR MODULES)

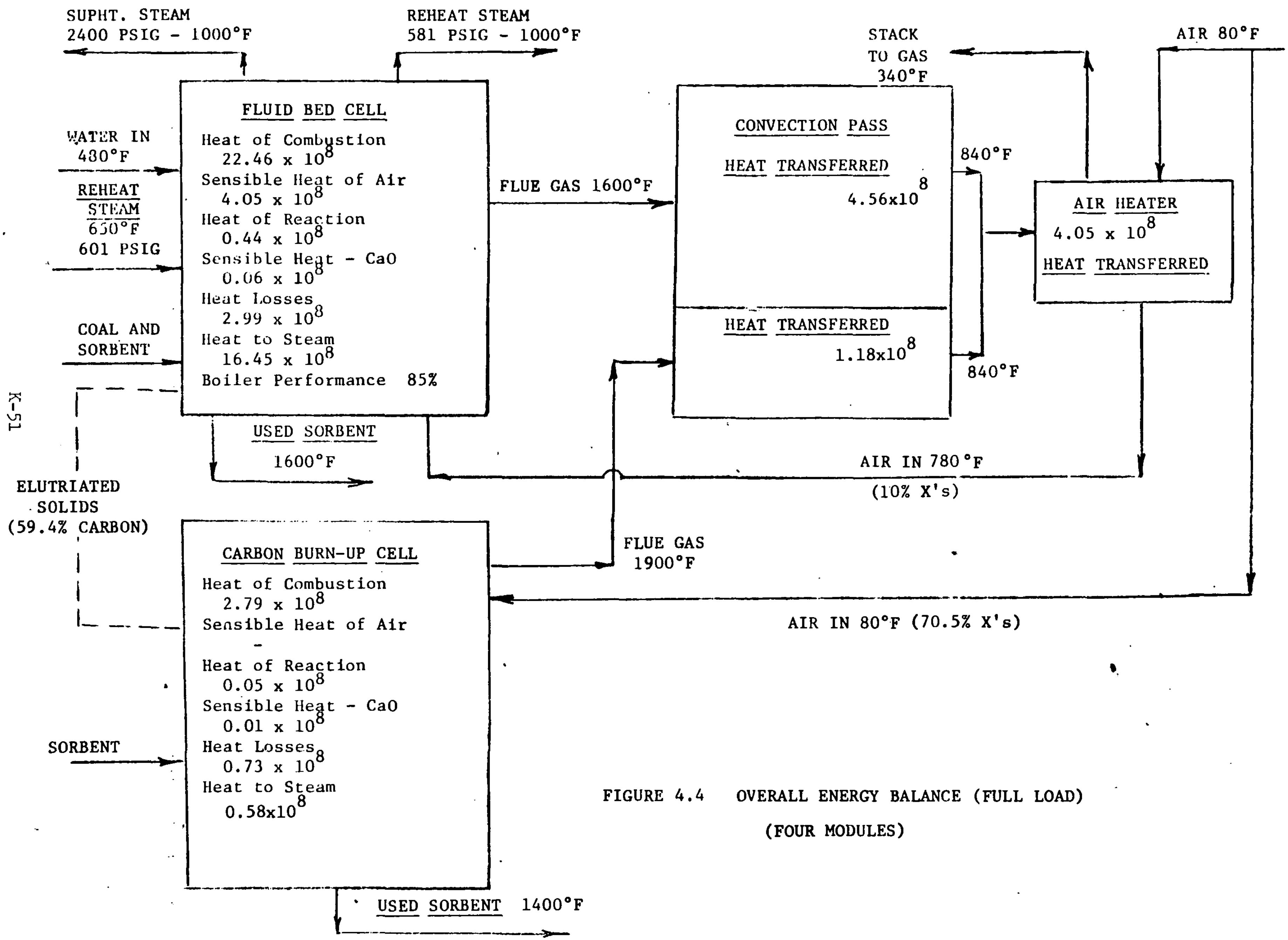


FIGURE 4.4 OVERALL ENERGY BALANCE (FULL LOAD)
(FOUR MODULES)

4.4 Tube Details

Enclosure walls are fin tube construction, also frequently referred to as membrane walls. The walls are made of panels of tubes connected by metal fins welded to the tubes. The panel walls are common construction used in pressurized furnaces of large capacity steam generators. The panel walls are fabricated by automatic procedures presently available.

The tube walls serve several functions. They act as heat absorption surface for evaporating water as well as a partition to separate gas and air streams. Where air must be admitted to the fluid bed the fins are simply discontinued and the tubes are bent out of line to provide a suitable opening.

The walls are relatively inexpensive and require little maintenance. They also serve as a conventional support system for the horizontal tube banks, base plates, roof plates and many of the air control dampers. The walls are supported off of structural steel by hinged rods connected to the upper wall headers.

The tube banks consist of standard serpentine tube elements constructed of 1 1/2" and 2" O.D. tubes as commonly used in the convection passes of large boilers. The number of tubes in the bundle is dictated by the steam side pressure drop. The arrangement of loop-in-loop or single loop tubes is a compromise worked out by the designer between numbers of tubes required, surface required and geometry or the volume the bundle is to occupy.

The tubes have a minimum bending radius of 1 1/2 times the tube diameter which more or less limits the longitudinal pitch of the tube bundle. In conventional boilers the lateral pitch is fixed by erosion, heat transfer and gas-side pressure drop considerations. In the case of tubes in the fluid bed the lateral pitch is dictated by the tube penetration spacing of the "Monowall" enclosure.

The tube bundles are fabricated on automatic tube bending machines at an average rate of 1.4 bends per minute. After the individual tubes are bent they are assembled into elements and finally into tube banks ready for shipment.

Individual coils or loops forming elements of the bundle are connected by welded fins near the bends of the tube. At this point the tubes in the lower half of the bundle are slightly offset to minimize the fin required. The tube bundles are supported off the finned water wall enclosure tubes.

4.5 Air Preheater Surface

Air preheater surface serves several purposes, 1) it minimizes hot gas losses and thereby improves plant performance, 2) it provides hot air to the furnace to avoid excessive quenching of the bed temperature and, 3) it provides hot air for drying the coal.

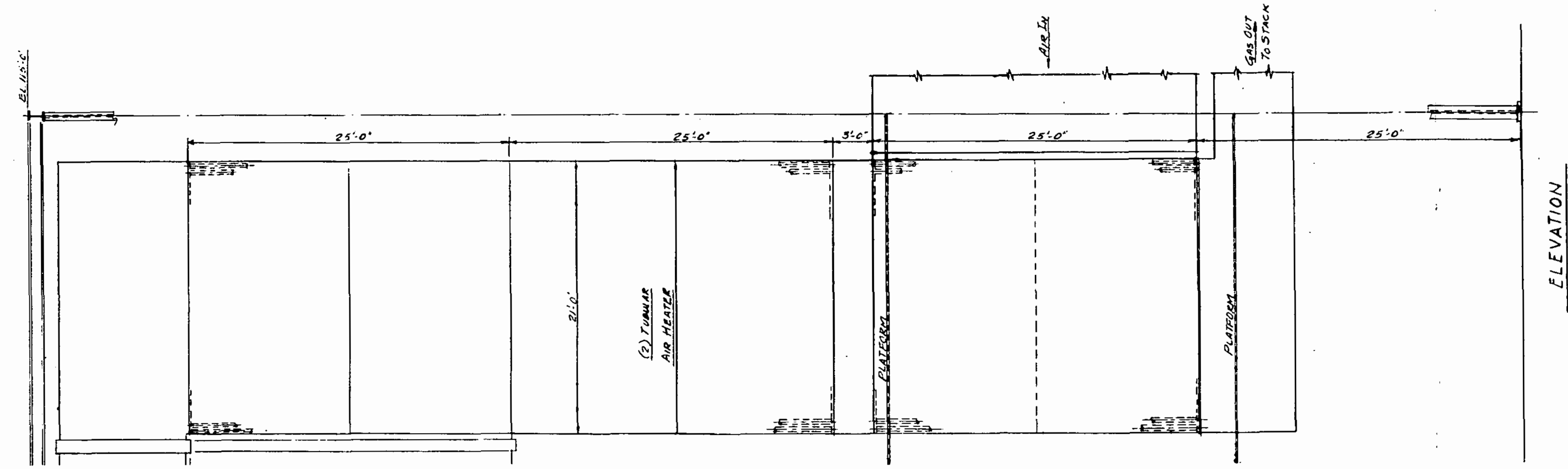
Several types of heat exchangers are available for this service. These include the regenerative air preheater and the tubular air preheater. The regenerative air preheater simply consists of a large rotor containing buckets of corrugated surface. As these buckets are rotated they are alternately heated and cooled by exposure to the hot gas and cool air stream respectively. Air leakage is inherent to this system at the seals between the hot gas and cool air stream. In low pressure systems this does not amount to much. In the atmospheric fluidized bed boiler substantial pressure differential must exist between the inlet air and outlet gas stream by virtue of the large pressure drop across the bed. To minimize the system pressure drop without effectively reducing bed depth, the pressure drop through the air heater and other auxiliaries must be as low as possible. This requires large auxiliary equipment which increases the size of the seals and compounds the air leakage problem.

The tubular air heater, as the name implies, is simply a tubular heat exchanger with the gas inside the tubes and the air making about 3 passes over the outside of the tubes. It is illustrated in Figures 4.5 and 4.6. In this type of heater, there is no air leakage. The cost of conductance, i.e., the cost of surface divided by the heat transfer coefficient is slightly higher for the tubular heat exchanger and, therefore, it is not as frequently used as the regenerative air heater.

Both units were designed and priced and the tubular air heater described in detail in the appendix was selected as the preferred heat exchanger.

4.6 Particulate Removal

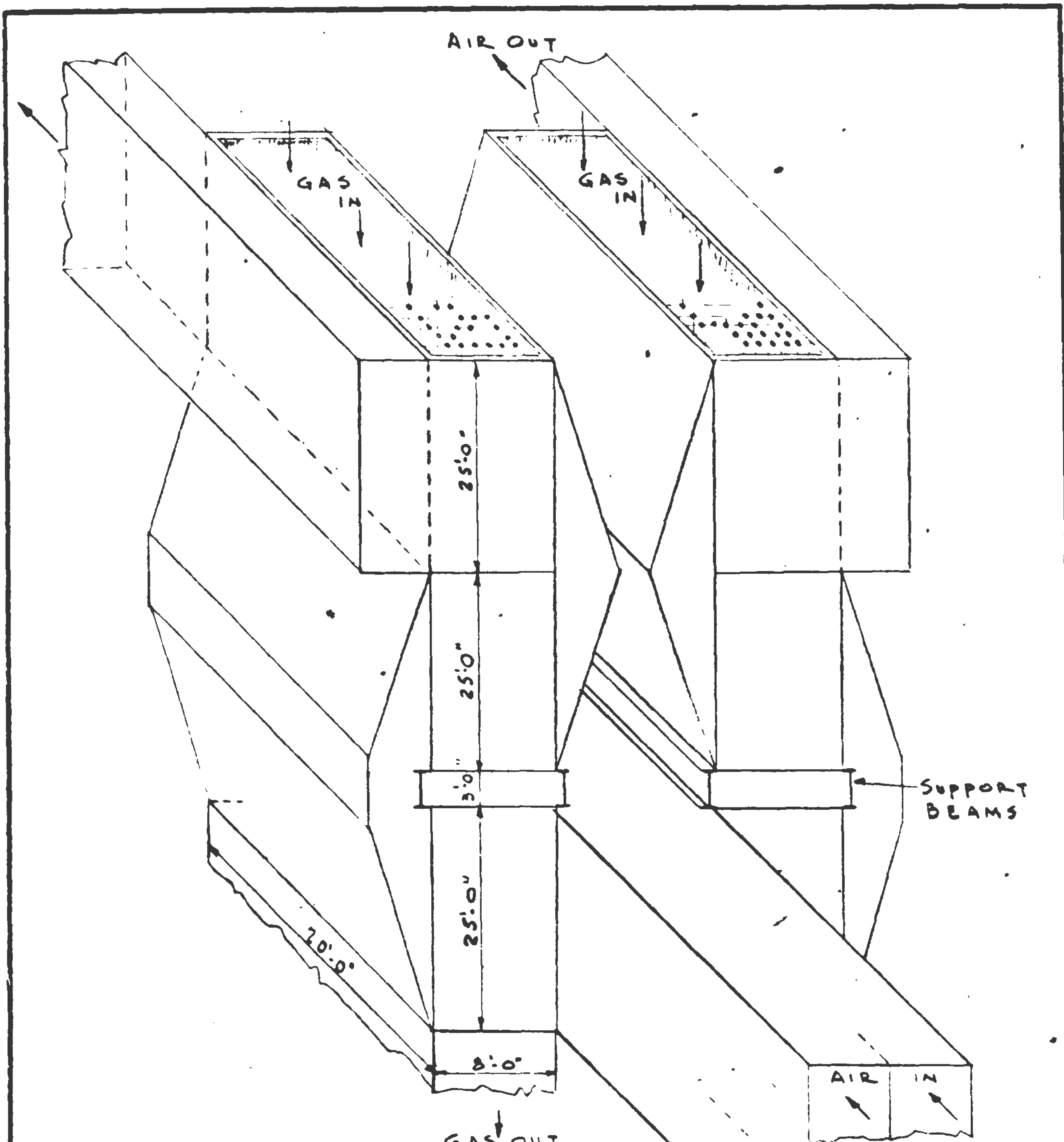
A two-stage particulate removal system is proposed, using a mechanical dust collector in the first stage to retrieve the carbon from the fluid bed and ash from the CBC and an electrostatic



AIR HEATER
ATMOSPHERIC FLUIDIZED
BED STEAM GENERATOR
FOR
300 MW PLANT

FOUR (4) REQUIRED

FIGURE 4.5



TUBULAR AIR HEATER

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SCALE:	= 1' - 0"

RD 714-112

FIGURE 4.6

precipitator in the second stage to clean the flue gas to .01 grains per standard cubic foot.

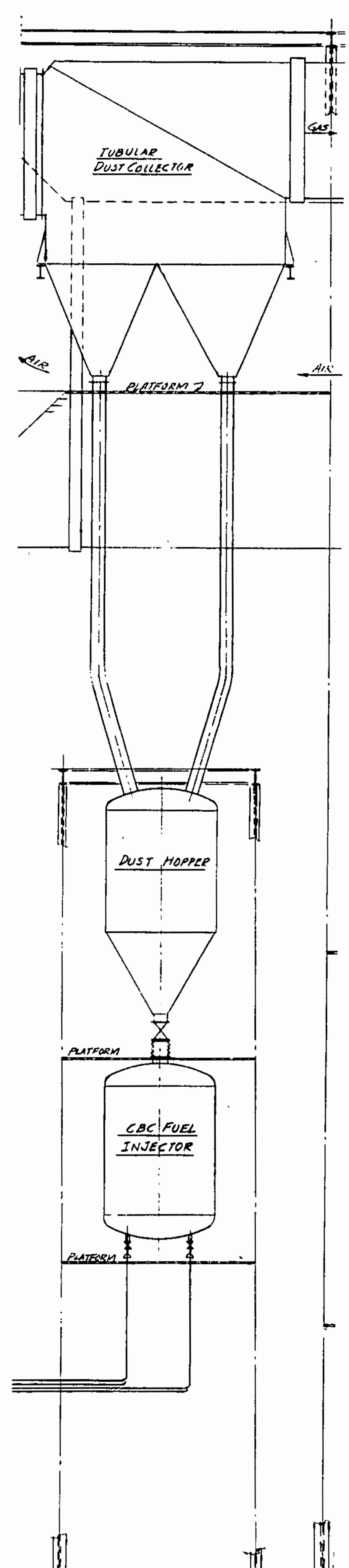
The gas laden with elutriated material rich in carbon enters the multiclone cyclone dust collector illustrated in Figure 4.7. Ninety-six percent of the material is removed, collected and passed on to a Petrocarb fluid bed injector which sends it to the CBC as a fuel. The collected material contains about 59.45 coal.

A portion of the dust collector has been partitioned off to receive flue gas from the CBC laden with ash and spent sorbent. Ninety six percent of this material is collected and withdrawn from the system as waste material.

Multiclone cyclones in theory are more effective collectors of small particles than larger cyclones using the same gas velocities. In practice, this imporved efficiency is reduced because of the increased short circuiting that takes place in small cyclones operating in parallel. The one important asset of the multiple cyclone arrangement is the efficient use of space at high capacity in contrast to the large volumes and high head-room required by the single Stairmand type of dust collector.

An electrostatic precipitator is used to clean the effluent gas from the mechanical dust collector. This is not an uncommon practice. Electrostatic precipitators are often combined with mechanical dust collectors in cases where gases have exceptionally high dust concentrations.

One electrostatic precipitator is used for all four boiler modules as the most economical approach to the problem.



ELEVATION

DUST COLLECTOR
ATMOSPHERIC FLUIDIZED
BED STEAM GENERATOR
FOR
300 MW PLANT
FOUR (4) REQUIRED

FIGURE 4.7

4.7 Coal Handling

The coal handling plant was designed to process 1 1/2" x 0 coal. The handling process illustrated in Figure 4.8 includes receiving, storage, crushing and conveying of coal to the feed injection system.

Coal arrives at the plant twice a week by unit train and is discharged into the receiving hopper. Unloading requires about 5 hours. The coal is conveyed to a 12,000 ton silo which acts as a buffer between the batch-type receiving operation and the continuous feeding operation. It also handles live storage to compensate for short term interruptions in delivery.

Live storage is supplemented with a 50,000 ton open pit dead storage system including reclamation equipment to provide for longer term interruptions in coal delivery of up to four weeks duration

The coal from the silo is fed to a reversible hammermill crusher where it is ground to 1/4" x 0 with a size distribution as shown in Figure 2.1, Section 2.1.

The crushed coal is then fed to a 150 ton surge hopper feeding a second series of surge hoppers serving individual feed injectors to the boiler.

4.8 Coal Feeding

A direct fired coal feed system illustrated in Figure 4.9 was selected to dry, distribute and transport crushed coal to the boiler. Generally, one of four systems is used to distributing pulverized coal to steam generators. These include:

1) Bin system

2) Direct firing system

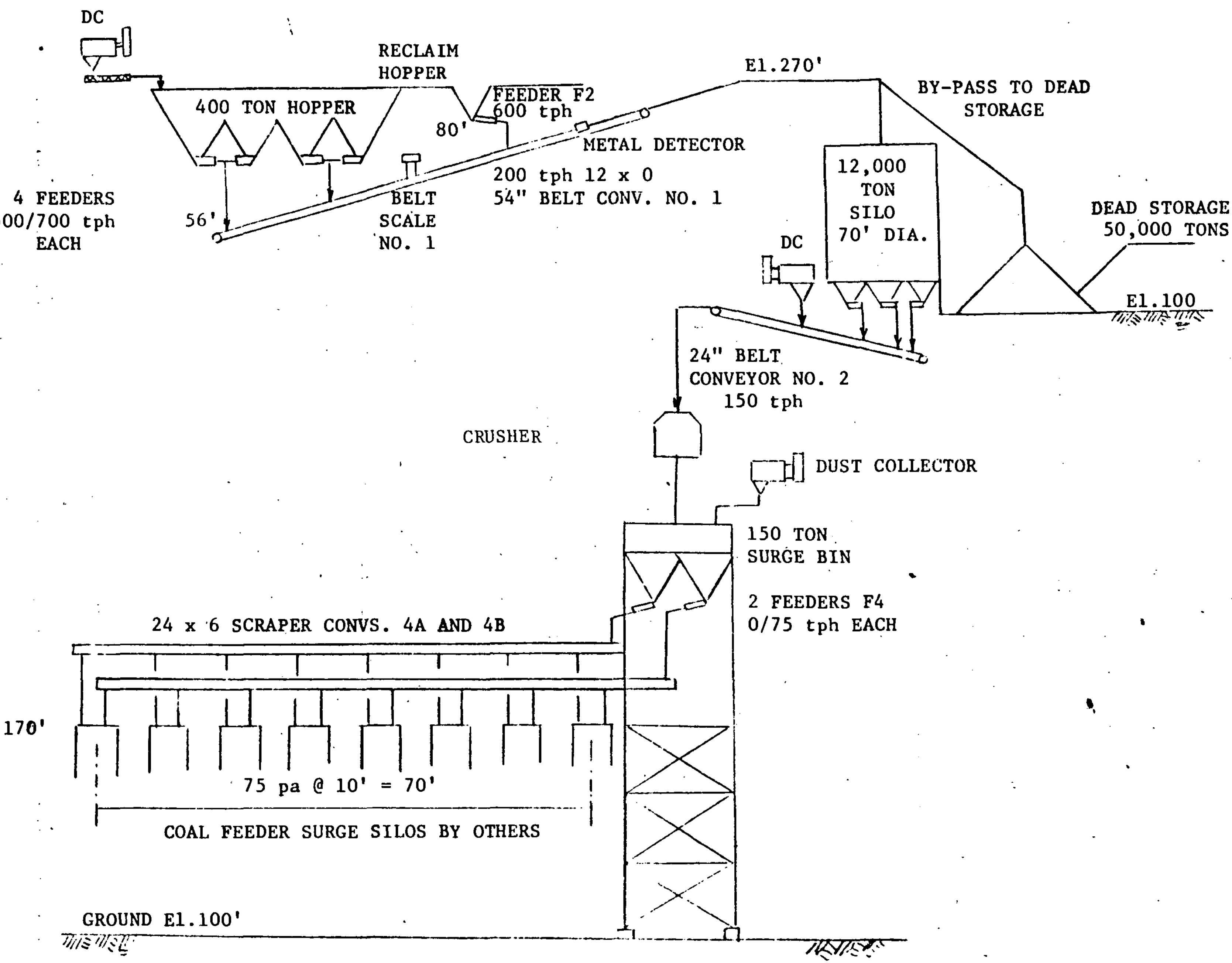
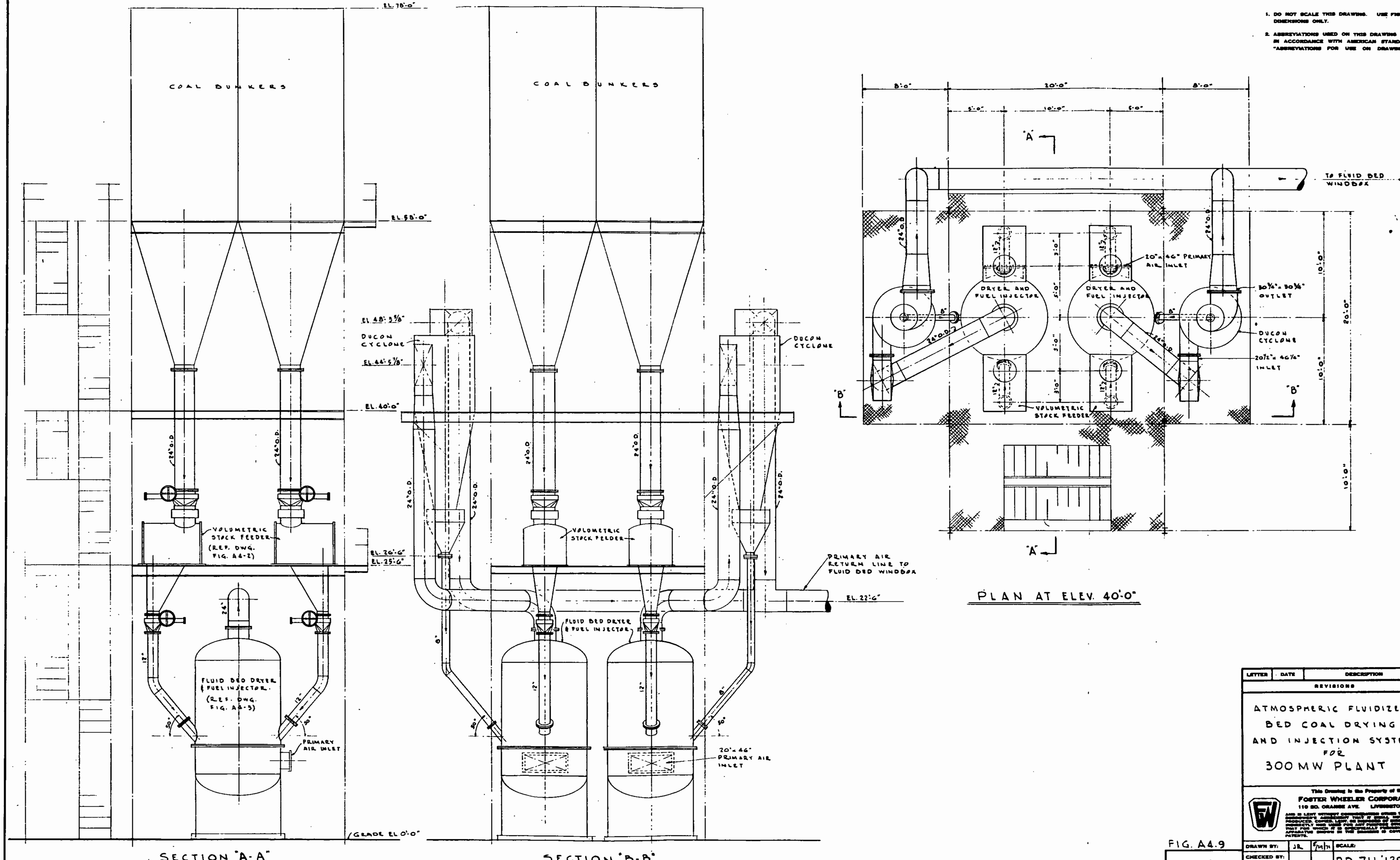


FIGURE 4.8 COAL HANDLING PLANT

NOTES

1. DO NOT SCALE THIS DRAWING. USE FIGURE DIMENSIONS ONLY.
2. ABBREVIATIONS USED ON THIS DRAWING ARE IN ACCORDANCE WITH AMERICAN STANDARD "ABBREVIATIONS FOR USE ON DRAWINGS."



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ATMOSPHERIC FLUIDIZED BED COAL DRYING AND INJECTION SYSTEM FOR 300MW PLANT

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CHECKED BY:			RD-711-129
APPROVED BY:			

FIG. A4.9

0-59-1192

3) Central pulverized-coal circulating system

4) Direct firing circulating system

They are illustrated in Figures 4.10, 4.11 4.12 and 4.13.

The direct firing system is simple and efficient in operation and well adapted to automatic control. Practically all pulverized coal installations within the last 25 years have used the direct firing system (2).

The system normally consists of receiving bunkers, feeders, pulverizers, transporting conduit burners and primary or transport air fans. The functions of the equipment is to pulverize dry and feed coal to the boiler. Adaptions of the system to a fluid bed boiler require several modifications. The modified system must dry, pressurize, distribute and feed coal to a multitude of injection points far exceeding the number of burners used in a conventional boiler of comparable size. The key factors here are multipoint distribution and pressurizing to 45" H_2O.

The modified system includes a receiving bunker, a surge bin, volumetric feeder, fluid bed dryer and distributor dust collector, primary air fan and coal-air transport lines. The coal flows from the surge bunker through a downspout which acts as a pressure seal to a volumetric feeder. The feeder regulates

(2) Allen J. Johnson and George H. Auth, "Fuels and Combustion Handbook", McGraw-Hill Book Company, Inc., New York, 1951.

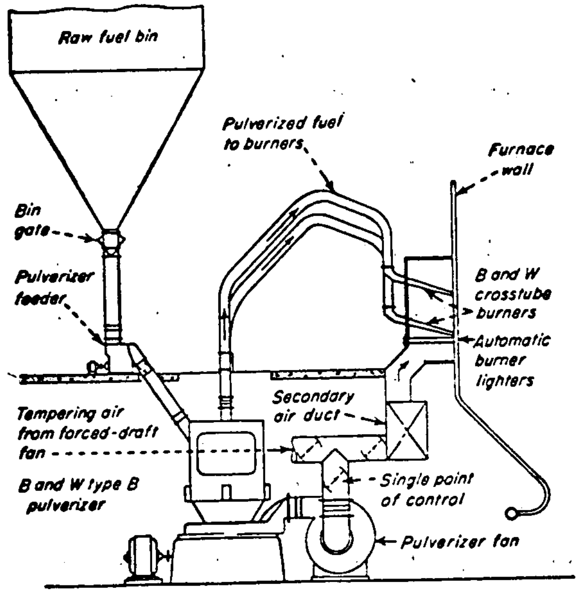


FIGURE 4.10 DIRECT FIRING SYSTEM (The Babcock & Wilcox Co.)
REF. (2)

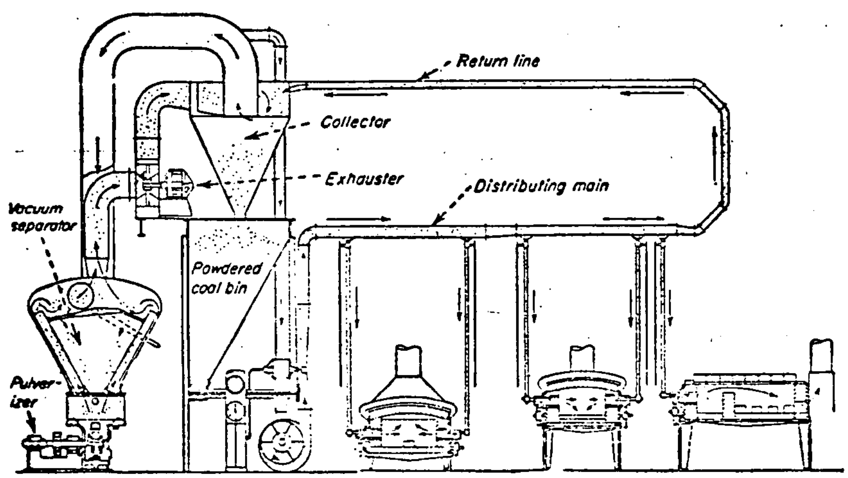


FIGURE 4.11 CENTRAL PULVERIZED-COAL CIRCULATING SYSTEM.
(Babcock & Wilcox Bull. No. 3-392) REF. (2)

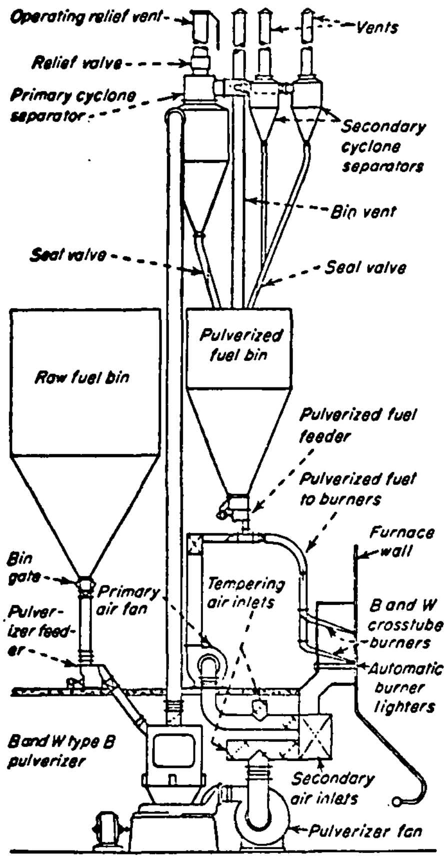


FIGURE 4.12 BIN SYSTEM (*The Babcock & Wilcox Co.*)
REF. (2)

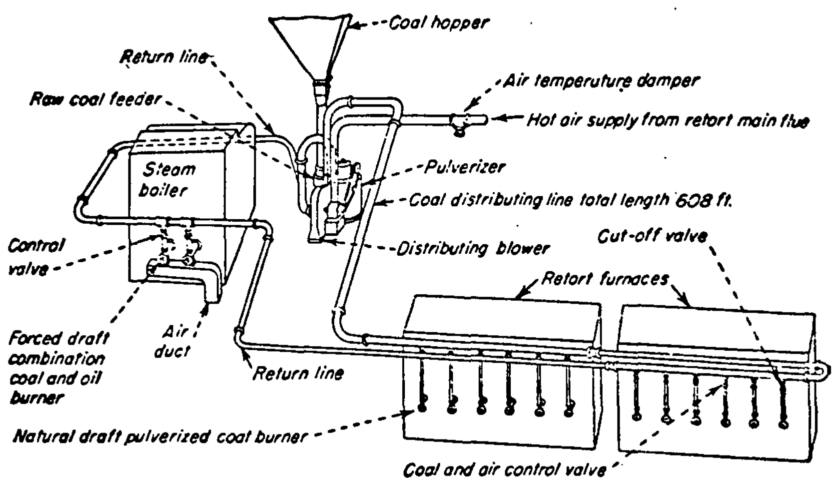


FIGURE 4.13 DIRECT-FIRED CIRCULATING SYSTEM. FIRING CHEMICAL RETORT FURNACES AND STEAM BOILERS AS INSTALLED IN 1944 AT STAUFFER CHEMICAL CO., BENTONVILLE, VA. CAPACITY 23,000,000 Btu PER HOUR. LENGTH OF LINE 608 FT. (*The Babcock & Wilcox Bull. No. 3-392*) REF. (2)

the coal feed to the system by means of a wire and variable speed belt. The coal flows by gravity from the feeder to a fluid bed dryer and distributor. After careful consideration of numerous possible means of flow splitting and distribution within the boiler shell and out the fluid bed with its inherent homogeneous distribution characteristic appear to be the most reliable means of distribution to a large number of feed points. Its desirability was further enhanced by its ability to dry coal in the same operation eliminating the need of a separate coal dryer and enclosed feed conveyors. The fluid bed dryer is illustrated in Figure A4-7 of the appendix. Its accessory equipment includes a primary air fan providing hot pressurizer air for fluidizing and drying and dust collectors for removal of elutriated material from the cooled air.

One inch feed lines transport the coal from the bed to 2" feed lines. The coal is pneumatically conveyed to the boiler in a dilute phase.

4.9 Sorbent Feeding

Sorbent feeding falls into two categories; feed make-up and recycle of regenerated sorbent. Provisions were made in each of the beds for one injection point to handle the regenerated material and one withdrawl point for removing spent limestone. The regeneration cycle and equipment was designed by Westinghouse and are discussed in their portion of the final report.

The feed make-up system was included as part of the boiler accessories and is described in Figure A4-4 of the appendix. It simply consists of a receiving hopper capable of handling limestone delivered in covered rail cars or trucks, a storage silo for dead and live storage and pertinent conveying belts. The limestone is discharged on to the coal conveyor just before the surge hopper to the boiler feed system. Although the limestone feed is -1/4 as received at the plant the hopper bin angles and conveyor belt were designed to handle the fine material (1000-5000 μ in diameter) at little if any extra cost. Conveyor elevations had to be minimized and hopper slopes had to be increased to 63°.

4.10 Carbon Burn-Up Cell

The carbon burn-up cell is a sixth fluid bed located at the bottom of the boiler. Its function is to burn-up the material elutriated from the fluid beds and collected by the mechanical dust collector. This material contains about 60% carbon with a reactivity somewhat lower than the raw coal feed. The bed is operated at 1900° without submerged tube surface to insure complete combustion. High excess air (about 50%) at ambient temperature is used to control bed temperature.

Material elutriated from the main fluid beds and captured by the mechanical dust collectors is fed to the CBC by a Petrocarb fuel injection system. This is illustrated in Figure 4.7. The system includes a collector vessel receiving the partially spent ash from the dust collector hoppers. The ash flows freely from the dust collector minimizing its residence time and exposure to hot gases and thereby avoiding the risk of fire. If necessary the atmosphere in the collector can be maintained inert. From the collector vessel the material flows to the fuel injector at periodic intervals. By means of localized fluidization the particulate matter is fed from the fuel injector vessel through 1 1/2" lines to 16 individual feed injection points in the CBC.

5.0 OPERATION AND PERFORMANCE

5.1 Operating Procedures

5.1.1 General Operating Characteristics Operation of the steam generator might best be discussed by considering it as a simple heat exchanger containing an exothermic reaction. Its behavior then is dependent upon the operating characteristics of the two flow streams, the hot gasses and steam, and their interactions.

The steam side of the heat exchanger is a once-through flow circuit in which water is heated, evaporated and superheated. There is little difference between it and a conventional once-through boiler and, therefore, its mode of operation is basically the same.

The gas side flow path is decidedly different from a conventional steam generator. Therefore, an understanding of the behavior of the fluid bed and its limitations and restraints are essential to an understanding of the operating characteristics of the boiler.

Combustion takes place in the bed at a nearly uniform temperature, as a result of the excellent mixing of particulate material. These are somewhat idealized conditions. However, they are reasonable assumptions for developing a preliminary design. Gas side heat transfer coefficient is nearly constant regardless of bed temperature or gas mass flow. The overall heat transfer coefficient is reduced slightly at reduced loads due to a decrease in the steam side heat transfer. Credit may also be taken for a high heat transfer coefficient in the volume immediately above the bed about 3' high due to partial elutriation of large particles which fall back into the bed and the combustion of some volatiles from the coal. For any given bed there are essentially two limits which define its size 1) heat transfer and 2) pressure drop.

Once fluidized, the pressure drop through the bed remains essentially constant until the terminal velocity for the bed particles is exceeded and the bed is partially elutriated. This is illustrated in Figure 5.1. The gas velocities then are limited to a range bounded by the fluidizing velocity and terminal velocity as shown in Figure 2.2.

Pressure drop through the bed and distributor plate is high. Therefore, the bed depth is limited to about 30 inches at atmospheric pressure. Deeper beds require optimization of the trade-off of capital investment in the shell structure and feed points versus the increase in capital cost and operating cost of larger fans.

The maximum bed temperature is 1700 - 1800° F based on characteristics of the ash and sulfur recovery considerations. 1600°F was selected as the design temperature for the atmospheric boiler. 1300°F is set as a lower limit for continuous operation based on sulfur recovery limitations and stable combustion.

The fluid bed appears to be restricted to a large flat "pancake" type furnace constructions or combustion zone unless stacked as selected in the conceptual design. At temperatures of 1600°F it is essential that most of the steam vapot superheating surface be installed in the bed. To avoid overheating during start-up, the stacked beds were assigned individual heating functions permitting independent light off in a sequence that would be compatible with steam generation. This arrangement has the added feature of separate control of firing rates for each heating function, i.e., reheating, superheating, evaporating, and feed heating.

Since the rates of transfer of heat in the bed and in the

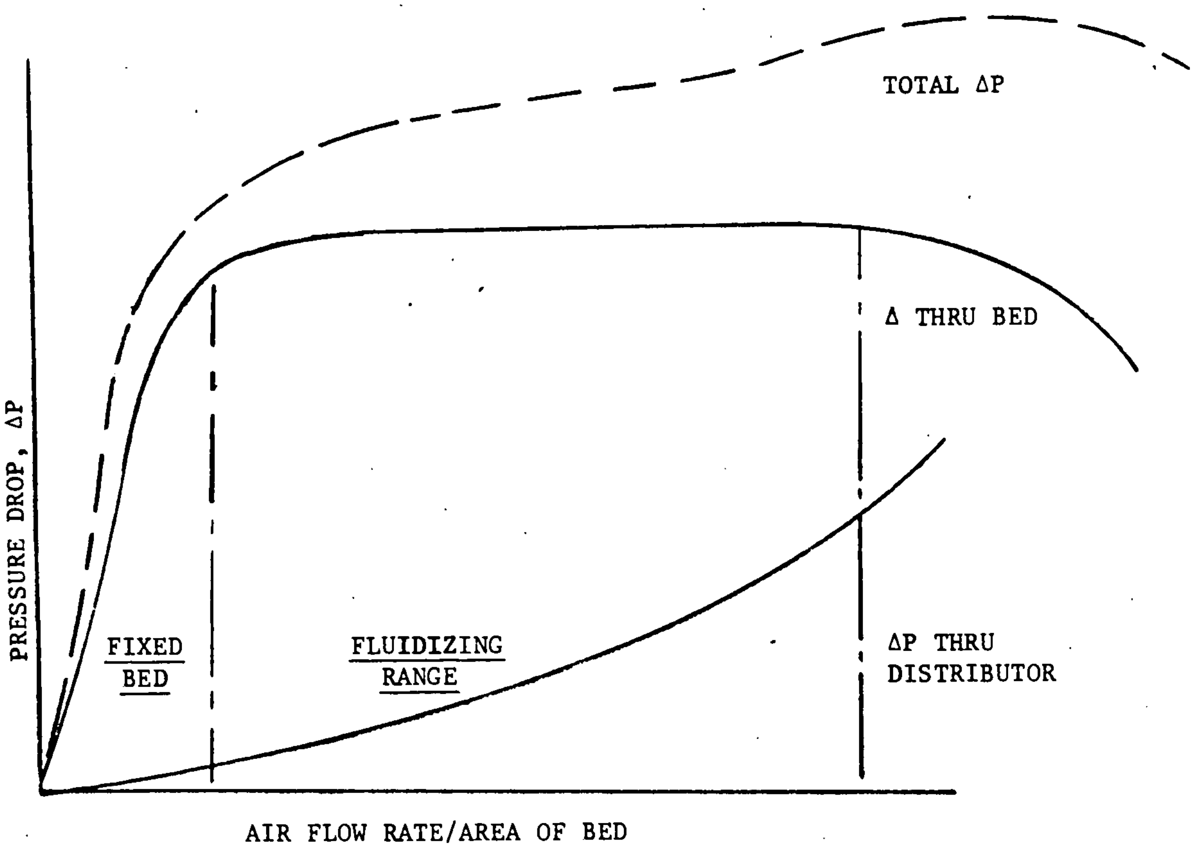


FIGURE 5.1 PRESSURE DROP OVER THE DISTRIBUTOR AND BED OF A FLUIDIZED COMBUSTION BOILER

volume immediately above the bed are high and nearly the same, load cannot be effectively controlled by raising and lowering bed height. Needless to say, the mechanical problems of doing this may also be quite cumbersome. Modular construction appears to be the logical means of meeting a 4-to-1 turn down. This arrangement also reduces construction time and should minimize total outage.

Modular construction also minimizes an otherwise disturbing metal temperature problem. The transfer of heat in the bed is governed by the simple relationship described by Equation (1).

$$q = US \cdot \Delta T_{mean} \qquad (1)$$

Where:

$$\Delta T_{mean} = \frac{T_1 - T_2}{\mathrm{Ln}\dfrac{(T_B - T_1)}{(T_B - T_2)}}$$

q = Duty Btu/hr
S = Surface Ft^2
U = Overall Heat Transfer
T_B = Bed Temperature °F
T_1 = Temperature Steam In °F
T_2 = Temperature Steam Out °F

The equation indicates that a drop in load reflected in Equation (1) as a decrease in heat flux q/S must be accompanied by a reduction in mean temperature difference. U is virtually independent of load. With a constant bed temperature and reduced S the tube metal surface temperature must increase. This may have serious consequences. Higher alloy steels are required. In addition the tube life may be shorter and the risk of burnout exists.

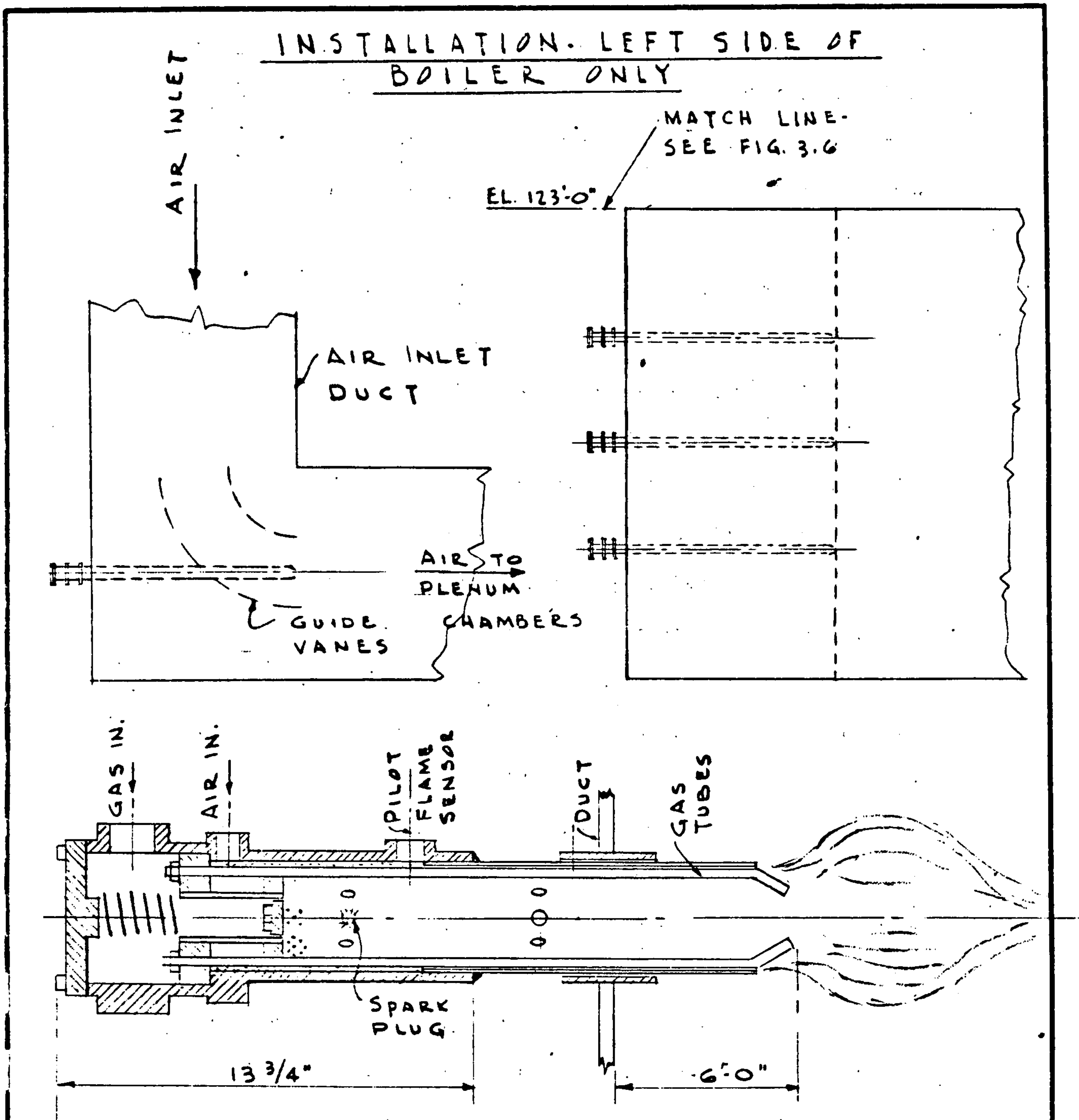
By going to modular construction any individual module need only be turned down 50% This load reduction in the module may be achieved by reduction in bed temperatures of about 200°F. Stable combustion is maintained and sulfur recovery is assured. Turn down to levels of 4-to-1 are achieved by cutting out individual modules.

5.1.2 Ignition Ignition of the fluidized bed is broken down into three basic steps 1) warm-up of the bed and pressure parts, 2) heating of the bed to ignition temperatures and 3) coal injection at ignition. Ignition of coal in the bed should take place once the bed temperature exceeds 850°F.

Warming up of the beds and pressure parts is accomplished by six 6.0×10^6 Btu/hr ignitors located in the air ducts upstream of the steam generator but down stream of the air heater as shown in Figure 5.2. This insures rapid heating of pressure parts in the immediate vicinity of the fluid bed in the most efficient manner. Heat not absorbed by the bed waterwalls or heating bundles is partially recovered in the heat recovery equipment and recirculator. Warming up the heat recovery equipment is of secondary interest and will probably not be completed until after the coal is injected into the bed and ignited.

Air flow may be increased as the pressure parts and bed temperature rise. Once the pressure parts approach 650°F, the air flow can be increased to fluidize the bed.

Once the bed is fluidized the second set of ignitors located immediately above the bed surface at an acute angle with the waterwalls can be turned on. There function is to heat the



WARM-UP IGNITOR
ATMOSPHERIC FLUIDIZED BED BOILER

ORDER NO.	0-59-1192
DRAWN BY:	JR 7/12/..
CHECKED BY:	
APPROVED BY:	
SCALE:	= 1' - 0"

RD-714-159

C. B. CO., NO. 44-512

FIG. 5.2

bed to ignition temperature. If it is found that total fluidization of the bed dilutes the heating process and prolongs the final warm-up stage to ignition, Local airports using compressed air from the coal feed system can be used to locally fluidize the bed along one wall.

The secondary ignitors are shorter versions of the warm-up ignitors. They are rated at 2×10^6 Btu/hr since over bed ignitors are reported to be only 70% efficient in transmitting heat to the bed.

When the total or local bed temperature, as the case may be, reaches 850°F the coal can be injected into the bed. It may be that a separate coal injection port will be required at the top of the bed in the vicinity of ignition for light off purposes only.

The carbon burn-up cell will receive less reactive coal so its ignition point will be higher. The bed should be heated to 735°F before it receives any fuel. This cell should be equipped with a gas burner capable of on or off operation based on bed temperature. This will insure combustion of the elutriated material and account for variation in rates of carryover.

A review of the literature indicated of the four possible arrangements of ignitors, i.e., 1) external burner, 2) internal submerged tunnel burner, 3) tunnel burner above the bed, and 4) open flame above the bed. The tunnel burner above the bed was selected as the best alternative.

5.1.3 Start-Up Start-up from a cold condition is achieved by placing one module in operation at a time. Starting up the second, third and fourth modules differs slightly from the first in that feed water circulating through the tubes will be 500° rather than 80°. The start-up system for the boiler is the same as that for a conventional once-through unit and appears in Figure 5.3

Valves B, W. P and D are initially open. Water flow is then started through the pre-evaporator bed wall tubes and first superheater bed. After the first superheater bed the water flows to the flash tank and then to the condenser. Next the pre-evaporator and first superheater beds are ignited.

As heat is added to the system valve D is closed and the heat recovery valve E is opened. This mode of operation allows a more rapid heating of the water than would be possible with circulation through the condenser. As heat is added to the circuit a pressure of 2900 psig is maintained in the pre-evaporator and evaporator circuits by throttling the W valve. The pressure in the first superheater and flash tank is kept at 600 psig by controlling the A, D and E valves. A steam-water level forms in the flash tank and the N and I valves are opened to allow this steam to warm the second superheater tubes and subsequent steam lines. Next the second superheater bed is ignited.

When the steam enthalpy is sufficient the turbine can be warmed and rolled. The flash tank pressure is then raised to 1000 psig and the steam flow is increased until the load is about 10% of full plant load or 40% module load. At this point the turbine throttle pressure is raised to the operating value

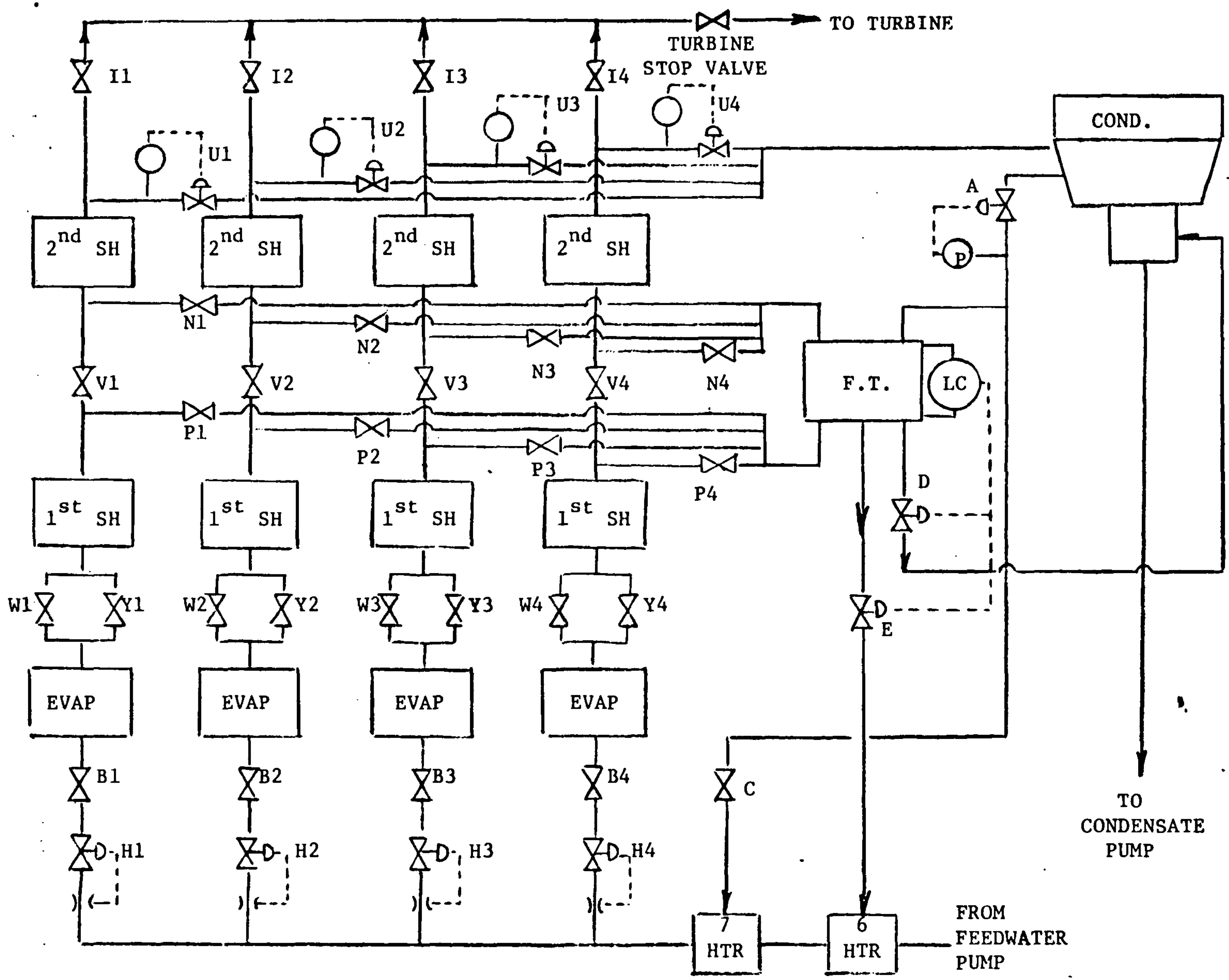


FIGURE 5.3

BOILER START-UP SYSTEM

by closing the P and N valves and opening the V and Y valves. Subsequent increases in load are controlled by the turbine governor and accomplished by increased steam flow.

Start-up of additional modules is very similar. There are a few differences, however. First the feed water temperature as already mentioned will be greater than 500°F, rather than 80°F as it is during the first module cold start. Initial feed heating should be considerably quicker. With the second, third and fourth modules the flash tank is used in the same manner as before but the I valve cannot be opened until the steam temperature and pressure match the existing steam turbine conditions. Until these conditions are reached the U valve will be open and the steam will be sent to the condenser or a high pressure feed water heater.

5.1.4 Load Control There are numerous turn down techniques which can be employed to achieve the percentage turn down desirable, however, the turn down technique preferred must be the one based on the consideration of ease of control, stability and reliability of operation, degree of turn down desirable and rate of response necessary. Turn down for the atmosphere utility boiler should be 4-to-1 at a load swing rate of response of 5% per minute.

In Section 5.1.1 it was explained that turn down would be achieved by using modular construction with constant bed heights. Reduction in load from full plant capacity would be achieved by reducing load on all four modules simultaneously until 75% plant load is achieved. At this point one module would be removed from service and the load on the remaining three modules would be returned to maximum capacity. Further reduction would be achieved in a similar fashion until two remaining modules are at 50% capacity. The option, then, is left to remove one of the boilers and return the last module to full capacity or to continue to operate both modules at 50% load. The selection of the appropriate alternative would depend on length of time at reduced load and operating characteristics of the sulfur removal system or boiler in general. Reduction in load is illustrated in Figure 5.4.

Turning down of individual modules to 75%, 67%, or 50% of full load may be achieved in one of two ways.

(1) Turn down by decreasing fuel and air input and maintaining excess air and bed depth constant.

(2) Turn down by decreasing fuel input and maintaining air input and bed depth constant.

For the same bed temperature mode 1 offers the greatest reduction in load. With a reduction in air flow along with coal flow the gas mass flow through the convection section is reduced significantly resulting in a substantially lower convective head transfer coefficient. Since the gas temperature follows the bed temperature, the convection section duty is proportionate to heat transfer coefficient. In addition to the greater flexibility in operation less fuel and lower limestone recirculation rates are required and much less heat loss through the flue gas is realized.

5.1.5 Shut-down Normal shut-down is accomplished in approximately the reverse manner to start-up. The first module load is reduced until it is 50% of full module steam flow. At

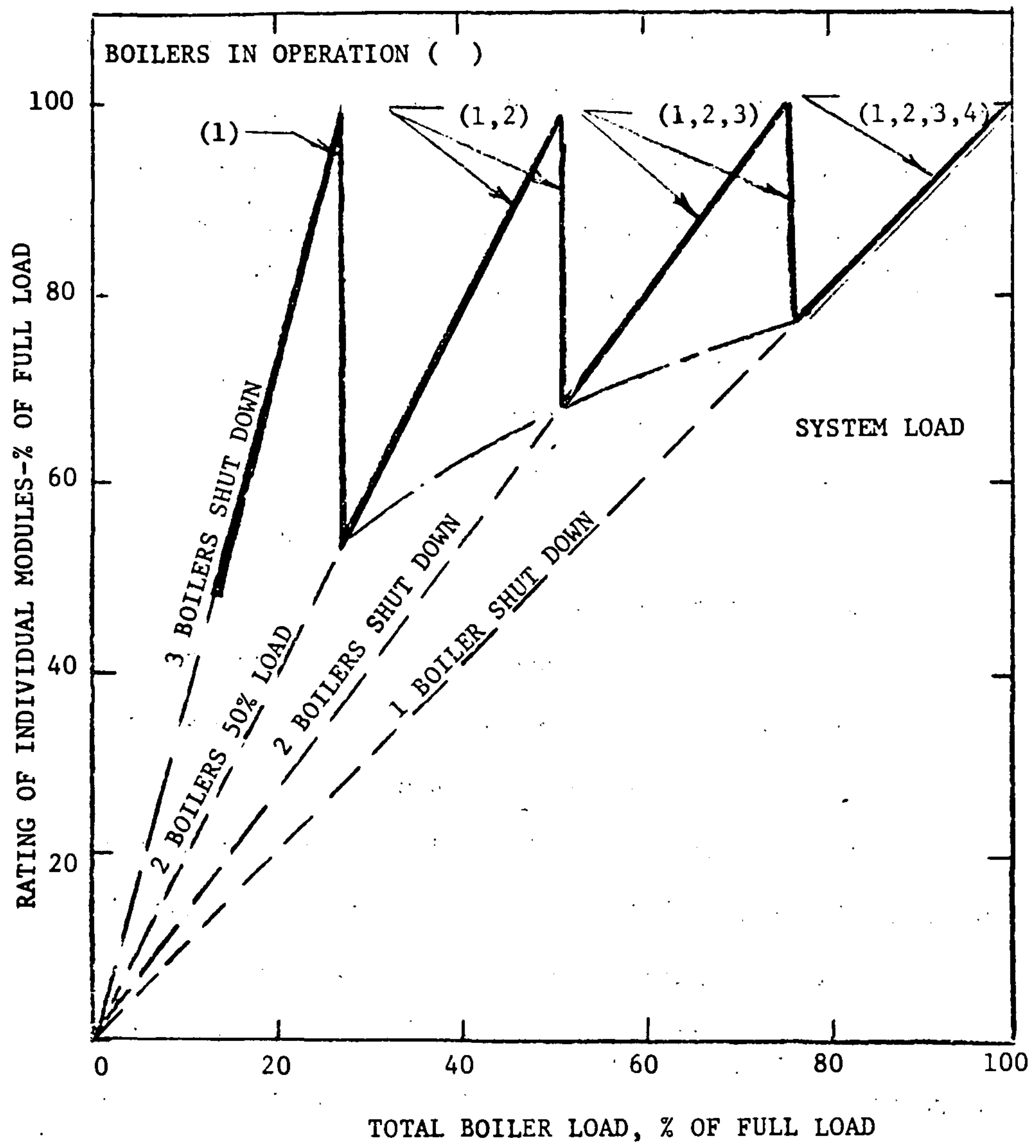


FIGURE 5.4 BOILER LOAD REDUCTION

this point the I valve would be closed and the U valve would be open to permit flow to the condenser. Fuel flow would be stopped but fluidizing air would be maintained in the beds in order to purge and cool the beds. When the bed is sufficiently cooled the air flow can be stopped. If it is desirable to keep the module in a standby condition, feedwater flow would be continued through the pre-evaporator bed, evaporator walls and first superheater bank. This flow would be achieved by closing the V valve and opening the P valve. Feedwater would then flow through these tube circuits and maintain the bed at approximately 500°F. If it is not required that the module be kept in a stand-by condition feedwater flow will be stopped by closing the B valve.

The last module is removed from service by throttling the pressure reducing valve W and controlling the P valve in order to reduce the turbine throttle pressure linearly with load down to 1000 psi. At this point the steam turbine would be removed from service and the I valve would be closed. From this point on the shut-down operation would be the same as with the previously removed modules.

5.1.6 Emergency Shut-Down Emergency shut-down due to steam side problems is accomplished in the same manner as in conventional units. Loss of turbine, trip of fans or loss of pump should automatically cut off the fuel supply. Loss of fans, fuel supply or pump automatically trip the turbine and the feed pump if the case applies. Although the unit contains large quantities of ash in the bed which has a reasonably high thermal inertia or stored heat capacity the bed temperatures are low by conventional standards. With only a 5% fuel charge in the bed at any given time the bed temperature should not experience much of a rise before it starts to decay. Tube metal and stored water should dissipate the heat when the bed temperatures are the highest. This may result in popping of safety valves for a short period of time. This, however, is not certain. Once the fans are cut off and the stored air is dissipated, the bed will slump and heat transfer from the limestone to the tube surface should be greatly diminished.

5.2 Performance Characteristics

The unit performance is summarized in Table 5.1.

TABLE 5.1

300 MW ATMOSPHERIC UTILITY BOILER
PLANT PERFORMANCE

Losses	Percent
Dry Gas Loss (Stack Temp. 340°F)	6.50
Loss due to Hydrogen & Moisture in Coal	5.05
Loss due to Moisture in Air	.10
Radiation Loss	.18
Incomplete Combustion	2.39
Manufactures Margin & Unaccounted for	1.50
	15.22
Boiler Performance	84.78

Radiation, moisture, and manufacturers losses are the same as for a conventional boiler. Dry gas is slightly lower than most units due to the low stack gas temperature. Incomplete combustion must be higher due to the elutriation of fines from the FBC and CBC.

Pressure drop at full load is summarized in Table 5.2. Pressure drop at lower loads should be only slightly less as the bed differential essentially remains constant.

TABLE 5.2

Summary of Gas Side Pressure Drop

Ducts	1.43
Distributor Plate	10.8
Bed	27.5
Convection Bank	0.1
Dust Collector	3.0
Air Heater Air Side	3.0
Gas Side	1.5
Electrostatic Precipitator	3.0
TOTAL	50.33

TABLE 5.3

Summary of Steam Side Pressure Drop

	PSI
Feed Heating, Convection Bank	125
Evaporator Bundle 1, Evaporator Bundle 2	96
Water Walls Pass 1	41
Water Walls Pass 2	55
Water Walls Pass 3	55
Water Walls Pass 4	51
Superheater Bank 1, Superheater Bank 2	142
TOTAL	565
Reheater	17.30

6.0 BOILER COSTS

6.1 Capital Cost

The capital cost of the 300 MW atmospheric boiler and assessory equipment is summarized in Table 6.1. All cost data was developed to proposal accuracy by commercial estimators. Quotations for standard equipment and assessories were obtained form qualified vendors. Where possible the costs have been broken down to show the cost of erection. Erection cost for the assessories include field assembly of the equipment. Erection cost for structural steel was all assigned to the steam generator.

Steam generator cost includes headers, tubes, insulation, ignitors, platforms, structural steel, flues and ducts, seals, coal feed piping, tie backs and buckstays, springs, thermocouples, etc. Cost distribution of major components of the steam generator cost have been broken down and appear in Figure 6.1. This data is also extrapolated to 600 MW with reasonable accuracy to illustrate the effect of capacity on component cost.

The total capital cost has also been broken down by individual components and extrapolated to 600 MW. This is illustrated in Figure 6.2 and Table 6.2.

To complete the comparison and provide a measure for evaluation the cost of the fluid bed has been compared with several cost estimates for 600 MW boilers in Table 6.3.

The figures quoted in these tables reflect the estimated cost of the design developed in the report and show areas where efforts may best be directed to further reduce cost. The data indicates substantial reduction has been achieved in steam generator fab-

TABLE 6.1

SUMMARY OF COST
PRINCIPLE EQUIPMENT
300 MW ATMOSPHERIC FLUID BED BOILER

1)	Steam Generator	$5,500,000	
	Erection	3,320,000	$ 8,820,000
2)	Dust Collector	116,000	
	Erection	13,400	8,949,400
3)	Precipitator	614,000	
	Erection & Foundation (3)	645,600	10,209,000
4)	Air Heater	1,110,000	
	Erection	272,000	11,591,000
5)	Fans	200,000	
	Erection & Foundations (3)	220,000	12,031,000
6)	Fuel Injection (FBC)	1,780,000	
	Fuel Injection (CBC)	500,000	
	Erection (3)	1,020,000	15,331,000
7)	Coal Handling (2)	1,750,000	17,081,010
8)	Limestone	325,000	
		TOTAL	$17,406,000

(1) Cost extrapolated from 280 MW Boiler

(2) Complete cost including erection and foundations

(3) Cost provided by United Engineers

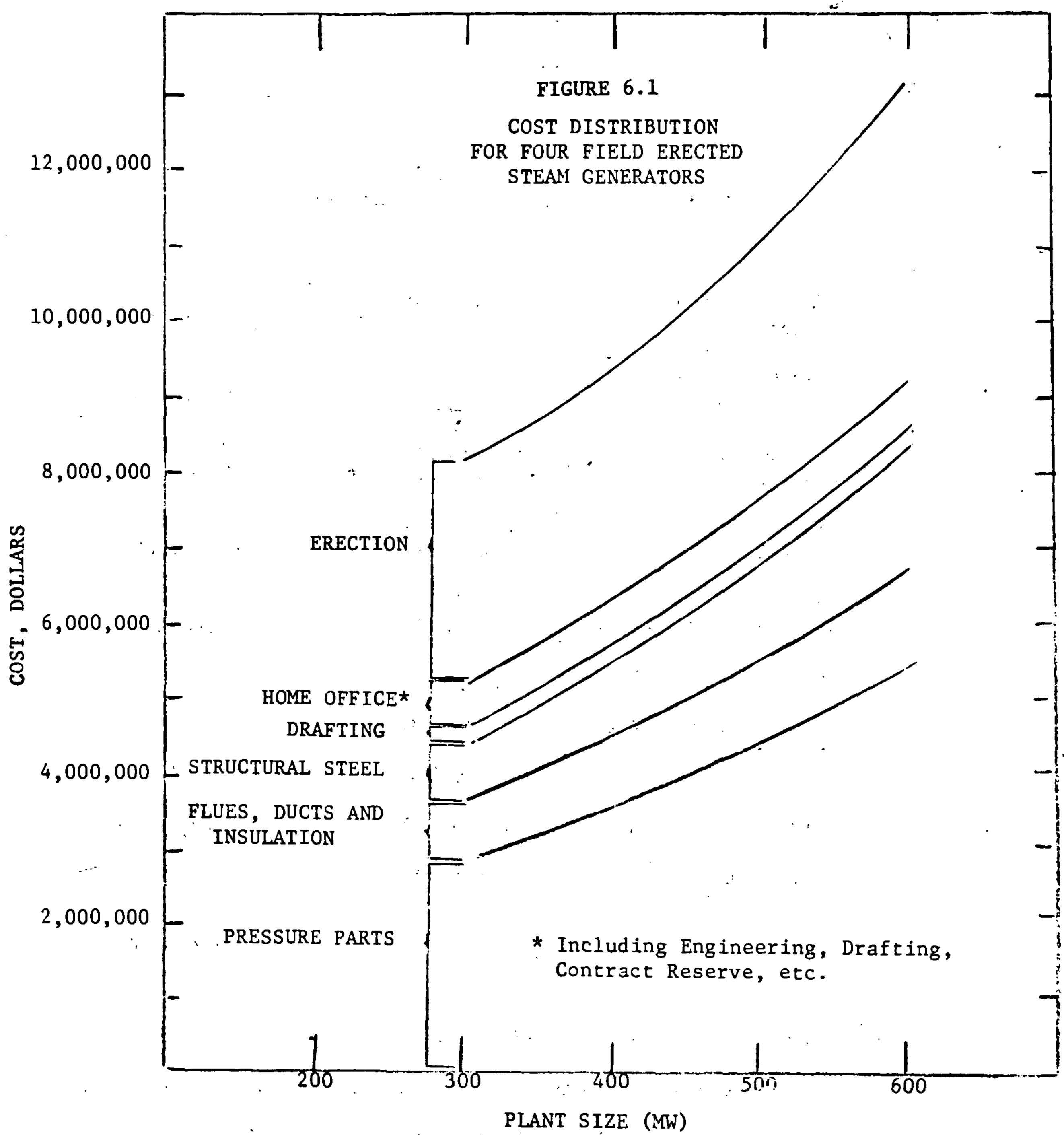


FIGURE 6.1
COST DISTRIBUTION
FOR FOUR FIELD ERECTED
STEAM GENERATORS
12,000,000
10,000,000
8,000,000
6,000,000
4,000,000
2,000,000
COST, DOLLARS
ERECTION
HOME OFFICE*
DRAFTING
STRUCTURAL STEEL
FLUES, DUCTS AND
INSULATION
PRESSURE PARTS
* Including Engineering, Drafting,
Contract Reserve, etc.
200
300
400
500
600
PLANT SIZE (MW)

TABLE 6.2

SUMMARY OF COST
PRINCIPLE EQUIPMENT
600 MW ATMOSPHERIC FLUID BED BOILER[1]

1)	Steam Generator	$10,100,000	
	Erection	4,200,000	$111,300,000
2)	Dust Connector	232,000	
	Erection	26,800	14,558,800
3)	Precipitator	1,228,000	
	Erection and Foundation	1,291,200 (3)	17,078,000
4)	Air Heater	2,220,000	
	Erection	544,000	19,842,000
5)	Fans	400,000	
	Erection and Foundation	440,000 (3)	20,682,000
6)	Fuel Injection (FBC)	3,560,000	
	Fuel Injection (CBC)	550,000	
	Erection	2,040,000 (3)	26,832,000
7)	Coal Handling (2)	2,000,000	28,832,000
8)	Limestone Feeding	325,000	
		TOTAL	$ 29,157,000

(1) Cost extrapolated from 300 MW level

(2) Complete cost including erection and foundations

(3) Cost provided by United Engineers

FIGURE 6.2

DISTRIBUTION OF COST OF
STEAM GENERATOR AND ACCESSORIES

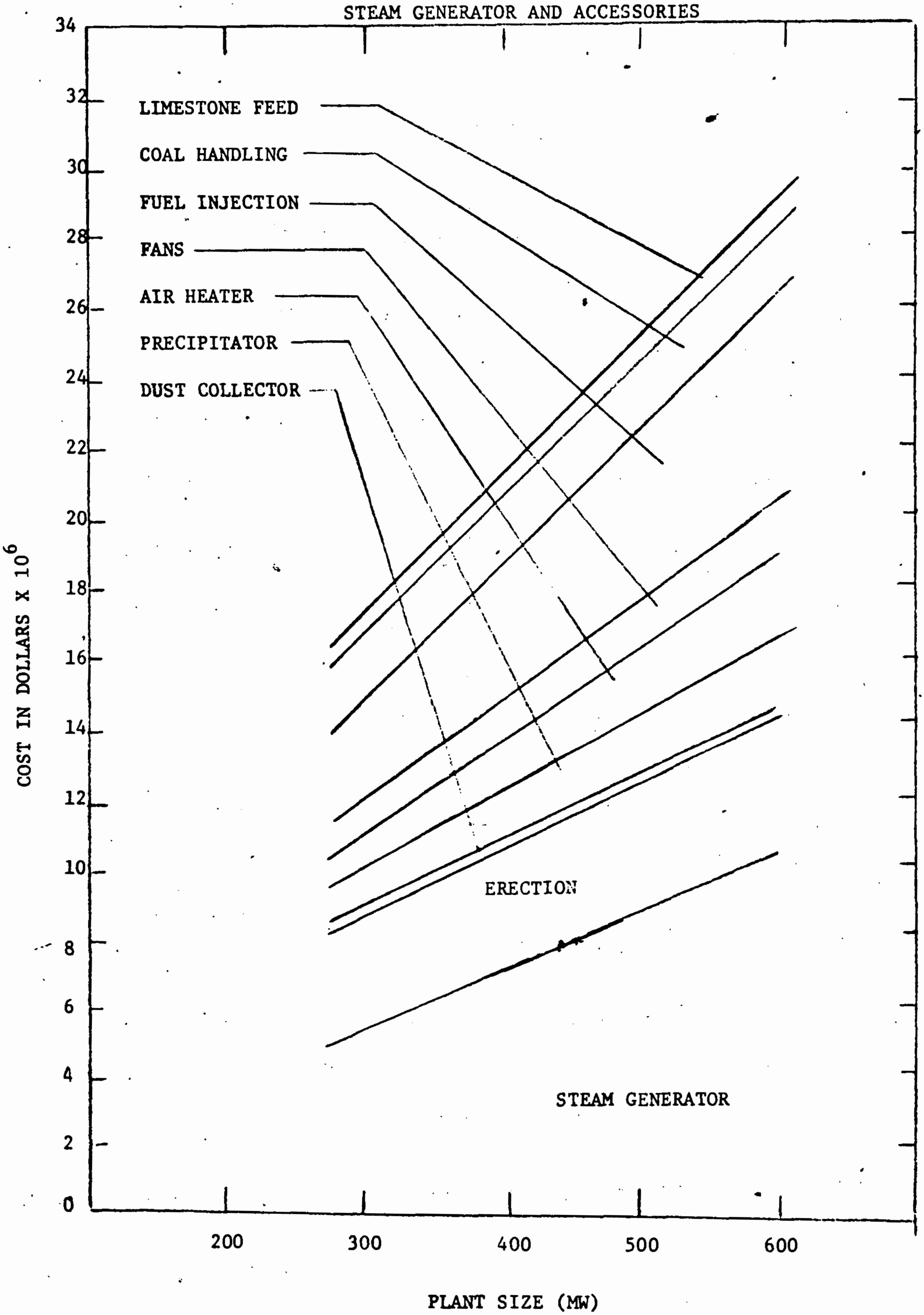


TABLE 6.3

COMPARISON OF COST OF A FLUID BED BOILER
WITH CONVENTIONAL BOILERS

	BOILER PLANT EQUIPMENT	600 MW CONVENTIONAL BOILER		600 MW FLUID BED BOILER	
		DOLLARS	$/KW	DOLLARS	$/KW
	TOTAL	44,707	74.49	40,735	67.83
1)	Steam Generator and Support	17,800	29.8 (5)	14,300	23.8
2)	Draft System				
	(a) Fans	582	0.96 (5)	840	1.42
	(b) Particulate Removal	------	(1).	2,830	4.71
	(c) Draft Flues and Ducts	1,490	2.47 (5)	940	1.37 (2)
	(d) Stack and Foundation	270	0.45 (3.5)	270	0.115 (5)
3)	Air Heater	1,860	3.10	2,640	0.45
4)	Coal Fuel Equipment	7,850	12.91 (5)	6,550	10.90
5)	Ash and Dust Handling Systems	1,520	2.54 (5)	1,520	2.54 (5)
6)	Stack Gas Cleaning	11,300	18.9 (5)		--------
7)	Regeneration System		--------	2,360	3.97 (5)
8)	Sulfur Recovery System		--------	6,200	10.3 (5)
9)	Instruments and Controls	1,500 (4)	2.50 (5)	1,750	2.92 (5)
10)	Miscellaneous Equipment	535	0.89 (5)	535	0.89 (5)

(1) No particulate control assumed beyond wet scrubber.

(2) Ducts incorporated with steam generator; cost between air preheaters, precipitator, and stack obtained from United Engineers cost estimate for ducting at the same conditions for the pressurized plant.

(3) May be greater since no NO_x control assumed. Stack height assumed for all plants of 280 ft. based on emissions projected from presurized fluid bed boiler combined cycle plant.

(4) Instrumentation for stack gas cleaning, regeneration, and sulfur recovery systems is included with the respective systems.

(5) Data provided by United Engineers.

ricating cost. However, there appears to be room for improvement by reducing structural steel, cost of flues and ducts, and erection time. A reduction in the cost of the former must also result in a reduction in erection cost. As indicated in other sections this might best be achieved by going to a self-supporting boiler enclosed in a wrap around shell.

6.2 Maintenance, Supplies and Operating Cost

Maintenance, supplies and operating cost are most frequently lumped together. In keeping with convention, they are lumped together here.

Maintenance, costs, etc., are virtually impossible to estimate for untried equipment. With proven systems the costs vary from year to year, station to station and fuel to fuel. It seems reasonable then to project available maintenance and operating cost and indicate what affect the fluidized boiler might have on it. Figure 6.3 illustrates maintenance and operating cost as a function of plant capacity for oil fired, gas fired and coal fired boilers (5, 6).

Maintenance, operating and supply costs for the fluid bed boiler should be less than those indicated as the system no longer requires the services of mills and soot blowers. Tube fouling has been minimized which should reduce outage and downtime for cleaning operation. The combustion temperatures are much lower than conventional units and as long as erosion is not found to be

(5) G. Guarria and T. V. Rallo, "Technology for Domestic Boiler and Power Plant Designs", Foster Wheeler Corporation, June 12, 1969.

(6) F. L. Roisson, A. J. Giramonti, G. P. Lewis, G. Gruber, "Technological and Economic Feasibility of Advanced Power Cycles and Methods of Producing Non-Polluting Fuels for Utility Power Stations", UARL Report J-970855-13, December 1970.

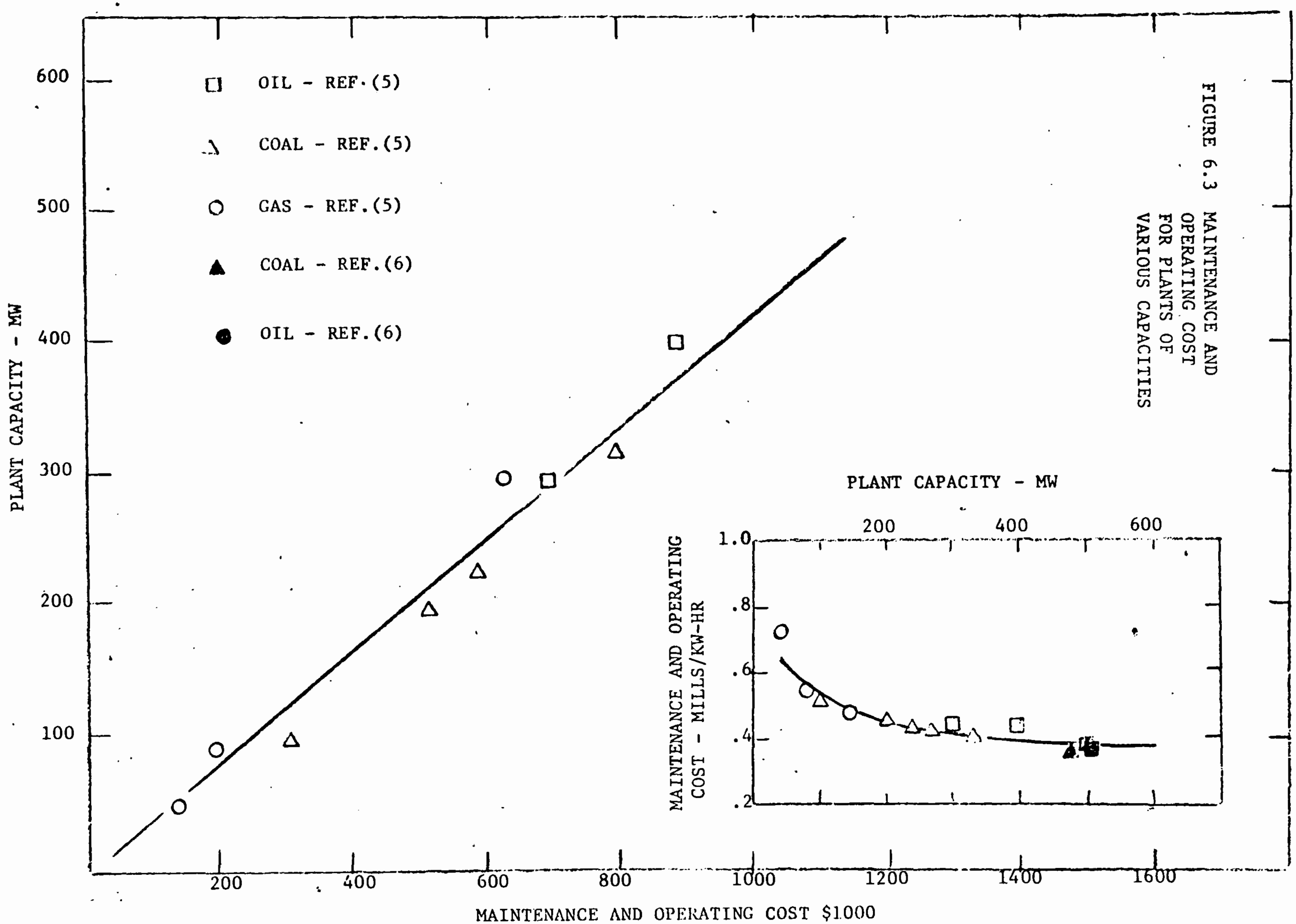


FIGURE 6.3 MAINTENANCE AND OPERATING COST FOR PLANTS OF VARIOUS CAPACITIES

a problem. The operating conditions to which tube surface is exposed is much less severe.

Coal feeding and drying may present some maintenence problems in the development stages. It would not be expected, however, that these costs would be projected on to established power plants.

6.3 Operating Costs

Operating costs in this case include fuel cost, limestone feed cost and power requirements as distinguished from supplies, salaries, wages, etc.

Westinghouse indicated the coal should cost $6/ton. No penalty is paid for crushing to 1 1/2 x 0 at the mines.

Mills exhaustors, soot blowers and motor drives in the air preheater have been eliminated in the fluid bed steam generating plant. Fan power has increased and may increase further if it is deemed worthwhile to go to a deep bed. Power requirements for transmitting the air pneumatically have increased somewhat.

7.0 DEVELOPMENT REQUIREMENTS

One of the purposes of going through the exercise of developing a conceptual design for the fluid boiler is to pinpoint areas requiring development work for further exploration. The steam generator and coal handling plant development requirements are discussed separately.

7.1 Steam Generator

Development requirements on the steam generator center around the bed. For the most part the information could be obtained or the data confirmed on a single fuel scale fluid bed cell. The comments that follow do not concern themselves with air pollution or limestone as they are treated elsewhere by Westinghouse.

Some of the basic areas of operation should be verified. These include the overall heat transfer coefficient and its relationship to other bed parameters, i.e., gas velocity, tube spacing, etc. The absence of erosion, agglomeration of ash in the bed, or fouling of surface where the bed should be confirmed. Distribution of coal and air in the bed should be looked into with every effort being made to reduce coal distribution feed points. Improvements in the latter area could open new areas for cost reduction in solids handling as well as boiler construction. Carryover must be confirmed and evaluated with regard to feed distribution, gas velocity, bed depth and coal feed size. Modes of ignition and ignition temperature for large beds must be proven. Temperatures distribution in the bed and potential inbalances must be evaluated to economically and safely select tube materials. Turn down of the bed at constant bed depths and response time with a change in load should be investigated. Control of air to four beds operating in parallel should be investigated for instabilities during operation.

7.2 Coal Handling

The coal handling system proposed consists of a fluid bed dryer-distributor and attendent equipment. Controls should be developed for the system and a detailed design should be made. The dryer-distributor should be tested on a pilot plant scale to confirm its ability to dry, distribute and regulate the flow of crushed coal. The ability to pneumatically convey 1/4 x 0 coal reliably and maintenance free should be demonstrated and design parameters should be confirmed.

APPENDIX A1

A1. CANDIDATE CONCEPTS

Three basic concepts were considered in making the final selection.

1) Vertically stacked beds in a square array.

2) Vertically stacked beds in four separate modules in an in-line arrangement.

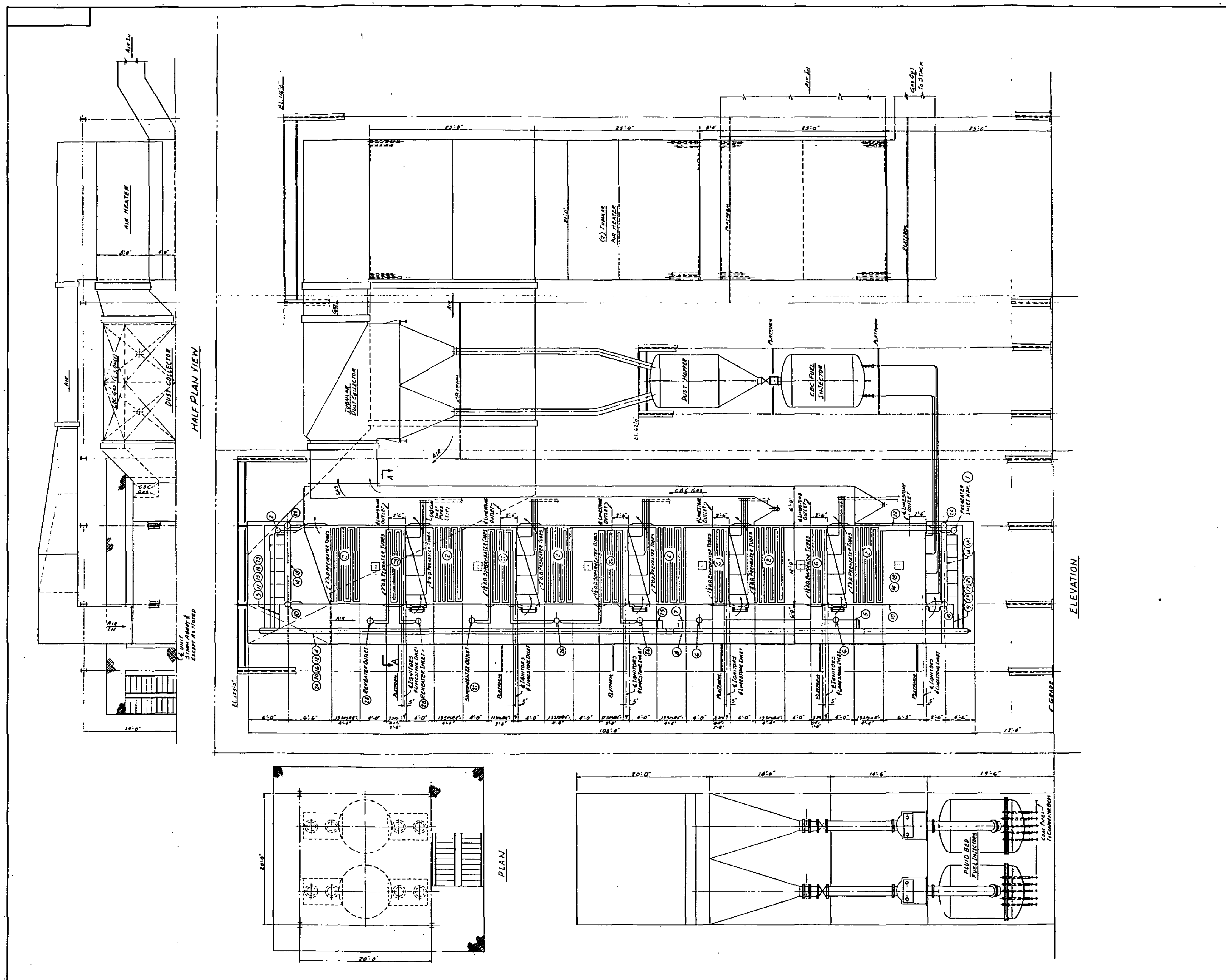
3) Horizontal tandem arrangement with four modules in parallel.

By initially assuming modular construction with a single bed for each heating function many other concepts were automatically considered and eliminated. This, of course, would include single bed arrangements, single module construction, etc. The virtues of the five bed, four module construction during start-up, operation, etc., are adequately discussed elsewhere and need not be reviewed here.

A1.1 Preferred Concept

The vertically stacked arrangement was selected as the preferred arrangement. It is illustrated in Figure A1-1 primarily as a reference for comparison with the horizontal arrangement and discussion of future concepts.

The preferred concept was selected as it conveniently adapted the once-through fluid circuitry to the multi-bed modular construction concept of fluid bed combustion with a minimal need for interconnecting pipes, headers and downcomers. This arrangement also simplifies the flow pattern and distribution of air to and gas from the steam generator. Plenum chambers used for providing and distributing air to the bed also serve the purpose of isolating adjacent beds. The arrangement adapts quite readily to



NOTES

1. DO NOT SCALE THIS DRAWING. USE FIGURED DIMENSIONS ONLY.
2. ABBREVIATIONS USED ON THIS DRAWING ARE IN ACCORDANCE WITH AMERICAN STANDARD "ABBREVIATIONS FOR USE ON DRAWINGS."
3. FLUE & DUCT TO BE STIFFENED FOR 45" OF WATER.
4. TIEBACKS & HORIZONTAL BUCKSTAYS SPACED ON APPROX 8'-0" CTRS.

LETTER	DATE	DR. INIT.	DESCRIPTION
A	5-14-71	H.S.	ADDED STEEL & PLATFORMS

REVISIONS

ATMOSPHERIC FLUIDIZED BED STEAM GENERATOR

RD-710-87

CONT. NO. 0-59-1192

DRAFTSMAN H.S.

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DWG. NO. RD-710-87

auxiliary equipment with the minimum amount of interconnecting ducts or pipes.

Orientation of the beds and the number required have little if any affect on the coal handling plant. The number of feed points required is primarily a function of the ability to diffuse the coal in the bed and thus is dependent on the cross-sectional area of the bed. Once the bed depth is set nothing can be done to simplify the coal handling process unless there is something unique about the system that complicates or simplifies the coal transport lines.

Convection surface consisting of preheater tubes or economizer tubes was placed above the bed for several reasons. First of all, this arrangement makes effective use of the boiler cross-sectional area. Secondly, by inserting the convection surface above the bed the gases are cooled down in a zone designed to handle hot gases, thereby eliminating special high temperature convection pass enclosures. This arrangement also avoids thermal losses due to mixing of hot gases at different temperatures as the gases from the different beds rejoin as one stream. Individual heat exchangers are the most effective way to exchange heat from the flue gas to the stream. Finally the convective pass acts as a particle screen returning large particles elutriated from the bed to the bed.

A cylindrical shell was considered as an enclosure for the pressure parts. It was felt that it would reduce cost by simplifying the enclosure. Being that the boiler would be self supporting the cylindrical shell would also eliminate the need for expensive structural steel. Erection time should also be reduced. The drawbacks to this arrangement were the large shell that would be required to house shallow beds at atmospheric pressure and the large number of penetrations required by the coal handling systems, the limestone recirculation system and the steam and water system.

A1.2 Horizontal Concept

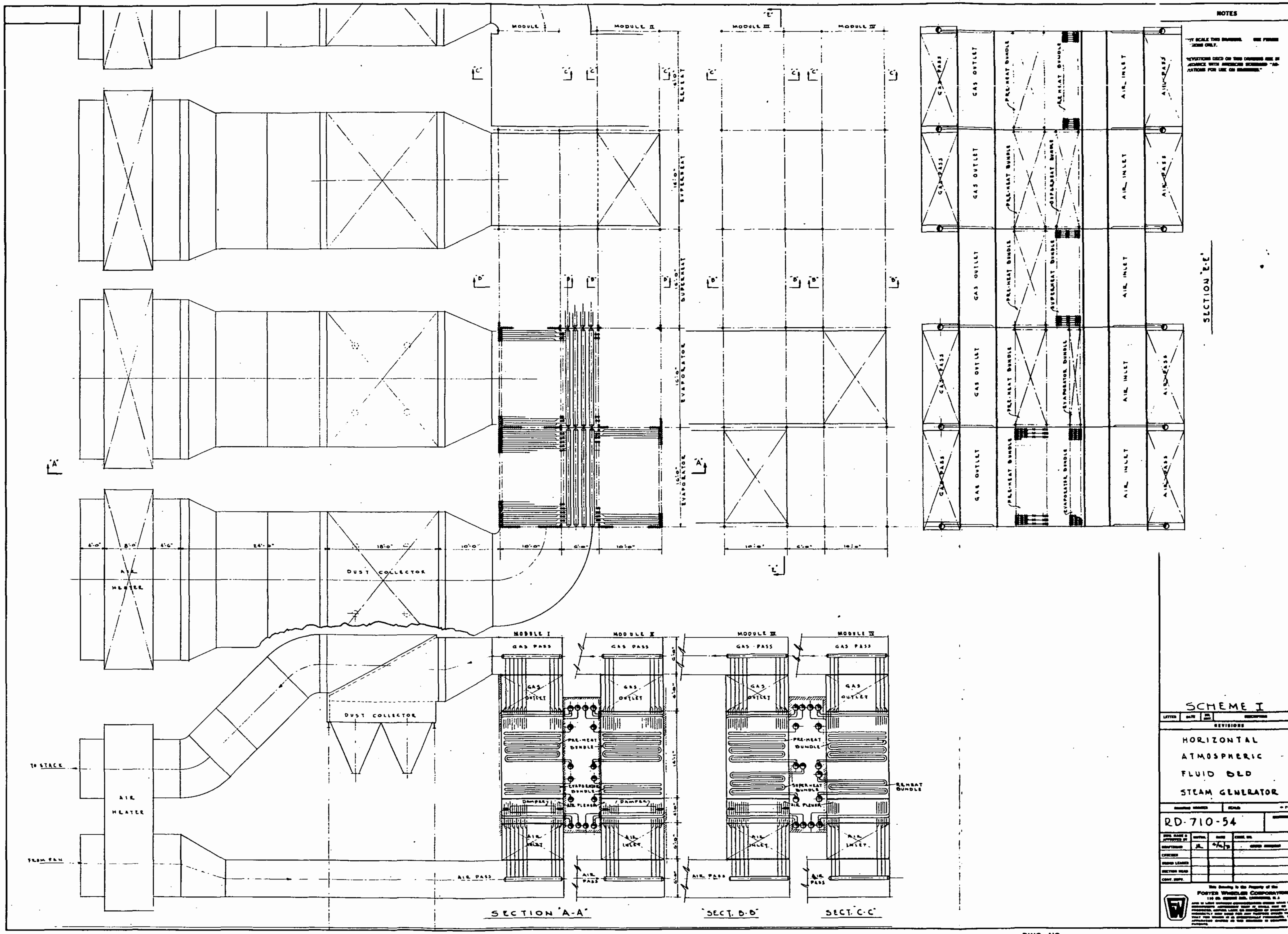
The multi-bed four module concept was laid out horizontally by placing the beds side by side in a tandem arrangment as shown in Figure A1-2. The beds had the same physical dimensions as the vertically stacked concept. This arrangement should minimize the need of structural steel. It was also thought it would offer greater potential for shop fabrication.

Whatever advantages may have been gained in time, material and labor appears to have been lost in complicating the fluid circuitry with additional headers, feeders and downcomers. The continuous pattern of the water wall tubes in the vertical arrangement is now broken with headers and downcomers required to transfer the fluid from chamber to chamber.

A portion of the savings in structural steel, of course, must be consumed in the form of supports in the horizontal plane. Instead of the boiler acting as a column supported from the top, it must be treated as a beam supported along its length. Expansion must be absorbed in the horizontal direction and the water walls must be designed to be self-supporting.

Water wall dividers separating beds into various heating functions present expansion problems and are subject to warpage due to a difference in heat absorption rates on opposite sides of the wall during normal operation as well as start-up.

Duct work has not been simplified. Separate enclosures


NOTES
MODULE I
MODULE II
MODULE III
MODULE IV
SUPERHEAT
EVAPORATOR
AIR HEATER
DUST COLLECTOR
GAS OUTLET
PRE-HEAT BUNDLE
REHEAT BUNDLE
SUPERHEAT BUNDLE
EVAPORATOR BUNDLE
AIR INLET
AIR PASS
GAS PASS
SECTION "E-E"
TO STACK
FROM FAN
DAMPERS
AIR PLENUM
REHEAT BUNDLE
SECTION "A-A"
SECT. B-B
SECT. C-C
SCHEME I
REVISIONS
HORIZONTAL
ATMOSPHERIC
FLUID BED
STEAM GENERATOR
RD-710-54
FOSTER WHEELER CORPORATION

DWG. NO.

are required to house the headers. The gas pass and air pass are complicated by the fact that dimensions of the fluid bed and the auxiliary equipment are not compatible.

Regenerative air preheaters must be used with their inherent high air leakage at the higher system pressure drop.

Head room below the dust collectors whether a single cyclone or multiple cyclone arrangement is limited. This complicates collection of the elutriated material from the FBC which must be returned to the CBC.

Each fluid bed requires its own lower and upper enclosure. There is no sharing of plenum chambers as in the vertical arrangement. Large areas may have to be sealed off to accommodate air control dampers.

The increased complexity of the fluid circuit is probably the major objection to the horizontal arrangement. Alternate arrangements may be considered which reduce the number of headers required. This is not accomplished, however, without a price. For example, the tube bundles may be connected such that the elements run in a horizontal direction and the fluid flows along the length of the bed rather than from the bottom up. The vertical water walls may also be arranged in a horizontal direction. In this arrangement there is greater chance of liquid vapor phase separation during evaporation which increases the risk of tube burn-out. Expansion problems are created between the tubes exposed to the bed and tubes exposed to the hot gases above the bed. Withdrawal of tubes through the water walls may become complicated.

For the many reasons cited the horizontal concept was not selected.

A1.3 Future Concepts and Potential Modifications

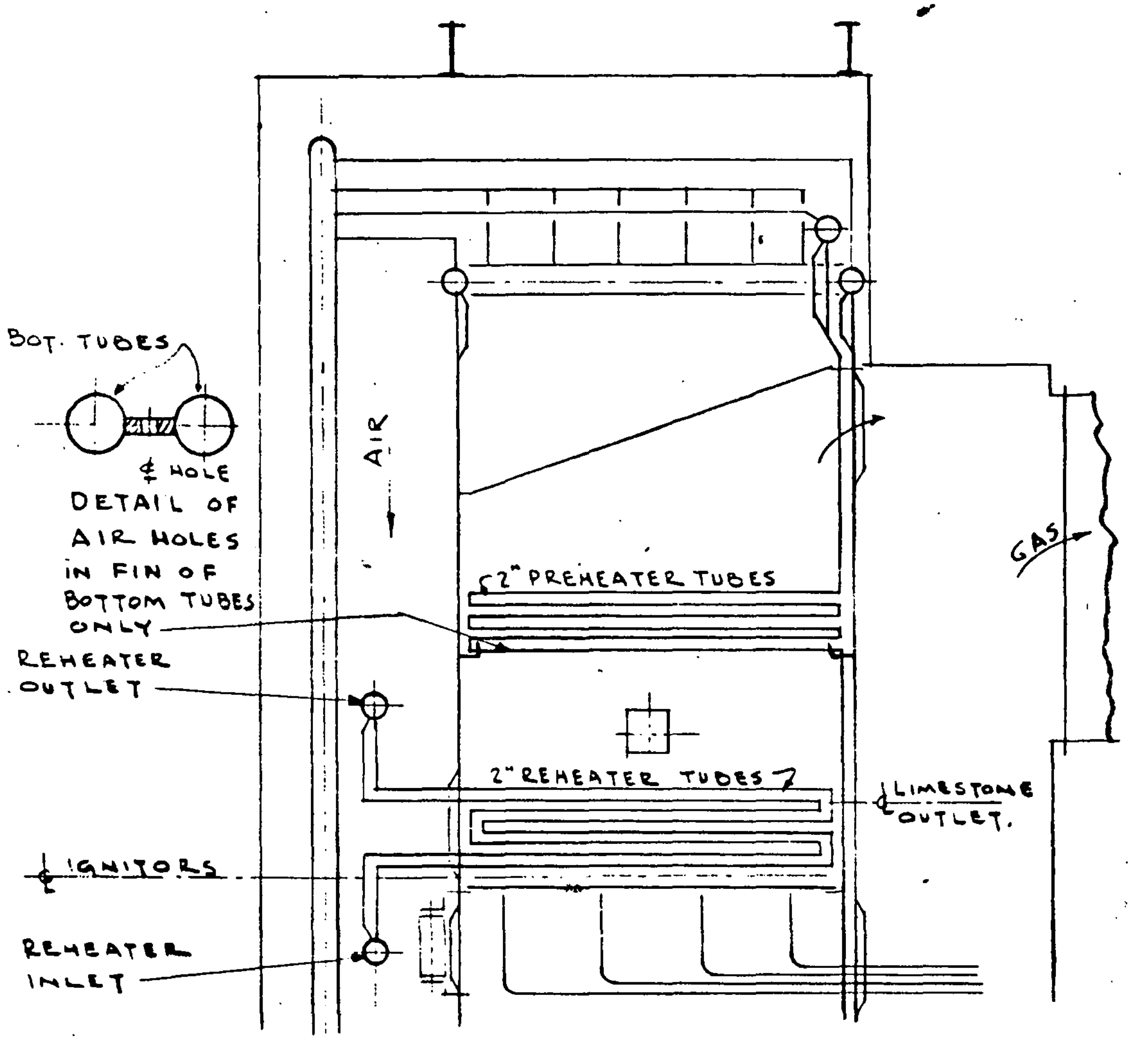
Developing the preferred concept revealed other areas for potential improvement of the design. Neither the scope of work nor time permits investigation and evaluation of all avenues of approach. Furthermore, some of the ideas fall out of the range of conventional practice, making technical and economic evaluation nearly impossible. The potentials of several of the unexplored areas appear worth noting for further consideration.

The areas worth mentioning include:

1) Installation of a second bed in series with the first which is not supporting a combustion reaction but serves the purpose of improving heat transfer coefficients in the preheater or convection zone.

2) The use of a cylindrical shell with a "deep" bed.

In the first concept it is proposed that a second bed be installed in series with the first such that it would contain the preheater or convection tubes. The fluid bed might be modified as shown in Figure A1-3. Fan power would have to be increased substantially to overcome bed expansion and pressure drop through the distribution plate. If the bed depth were approximately the same depth as the primary bed the pressure drop of the system would be a little short of being doubled. Beds of this depth should contain the entire preheater bundle once the heat transfer coefficient was taken into consideration.



TYPICAL ATM. FLUID BED CELL WITH PREHEAT TUBES SUBMERGED IN SECOND FLUID BED

ORDER NO.	0-59-1192
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APPROVED BY:	
SCALE:	= 1' - 0"
RD 714-118	

C. B. CO., NO. 44-312

FIG. A1-3

A distribution plant could be formed by the lower row of the tube bundle by using modified fin tubes.

Assuming the surface was reduced by a factor of two, roughly speaking, one might expect a 20% reduction in the cost of pressure parts which corresponds to a total reduction in the cost of the steam generator of 6.5% or about $3.00/KW. No credit is taken for a reduction in boiler height and no additional charge is made for handling the aggregate in the bed. If the fan powers were doubled there would be about $1.6/KW increase in cost that must be charged to the system. Operating costs have not been taken into consideration.

This concept may be subject to wear problems on the distribution plate as it must be designed to handle dust laden material.

The concept, however, appears worthy of more detailed analysis.

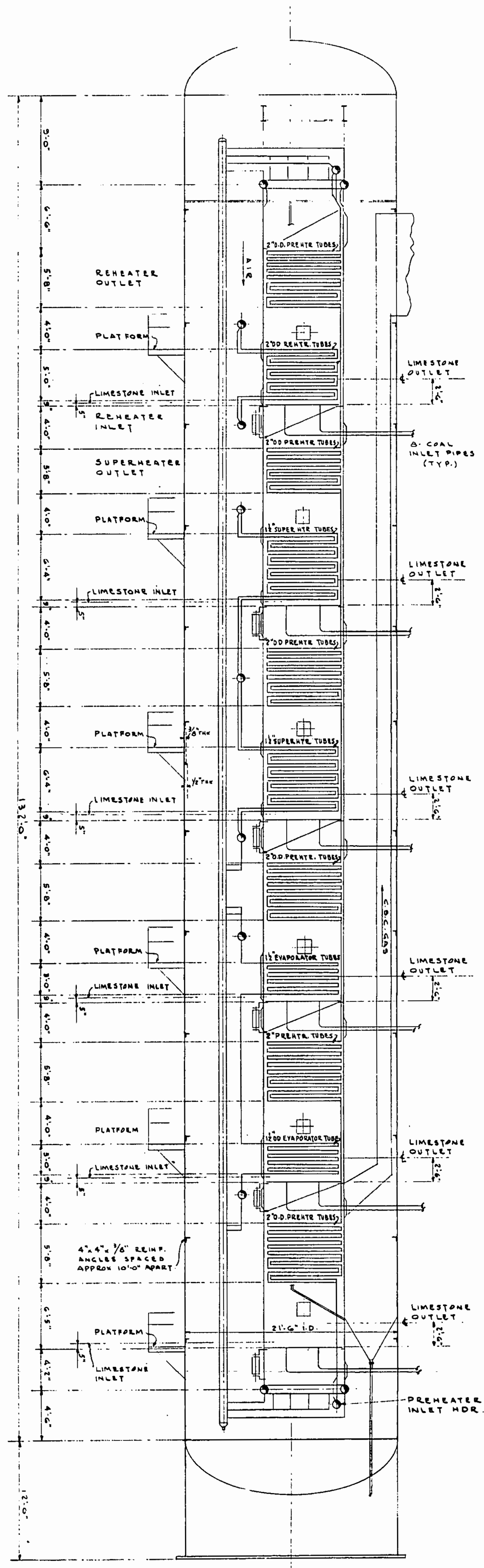
The second concept proposes increasing the bed depth by a factor of two and enclosing the boiler in a self-supporting cylindrical shell as illustrated in Figures A1-4 and A1-5. It is not absolutely necessary to increase the fan power. However, by doing so one reduces the cross-sectional area of the bed and thus reduces thenumber of coal feed points. This simplifies the shell design and makes it more attractive by reducing the number of penetration and reducing the diameter. The height must be increased but by only 30 feet.

Introducing a self-supporting shell eliminates a large quantity of structural steel, reduces erection time and simplifies boiler enclosure. Some structural steel must be attached to the shell for access. Additional grating could be simply installed inside the vessel either for maintenance either in a temporary or permanent basis.

Since the gas and air temperatures are nearly the same, there should be no warpage of the shell due to a difference in expansion. Sealing of the air and gas passages at the low pressure differentials could be handled by a simple plate steel weld or curtain on either side of the boiler. Expansion problems might be resolved by removing steam lines either from the top or bottom of the shell through a sliding sleeve. The coal and limestone transport lines might be removed at the appropriate elevations by taking up the expansion through moment arms as shown in Figure A1-5. By offsetting the penetrations in the coal position the relative movement between fixed positions in the shell and pressure parts may go from +3 inches to minus 3 inches or a total of 6 inches.

The shell as proposed would be 1/2 inches thick in the lower half and 3/8 inches thick in the upper half with 4" x 4" x 3/8" stiffeners located on the inside on 10 ft. centers. The shell costs about $75,000 per module. Unfortunately this represents little savings over ducts and structural steel material cost for the shell diameter proposed. However, a savings of $100,000 per module may be achieved in field erection cost. This represents a savings of $1.33/KW. No credit was taken for maximum shop fabrication of pressure parts at the smaller diameter. No credit was taken for reduction in coal feed points at the smaller diameter. No liability was assigned for the increase in operating and capital cost of the fan. At 300 MW the capital cost should increase by $.42/KW.

This approach appears to merit further study and consideration.



ELEVATION

VERT. ARRGT. OF PRESS. PARTS
ATMOSPHERIC FLUIDIZED
DEEP BED STM. GENERATOR
WITH CIRCULAR SHELL

FOR

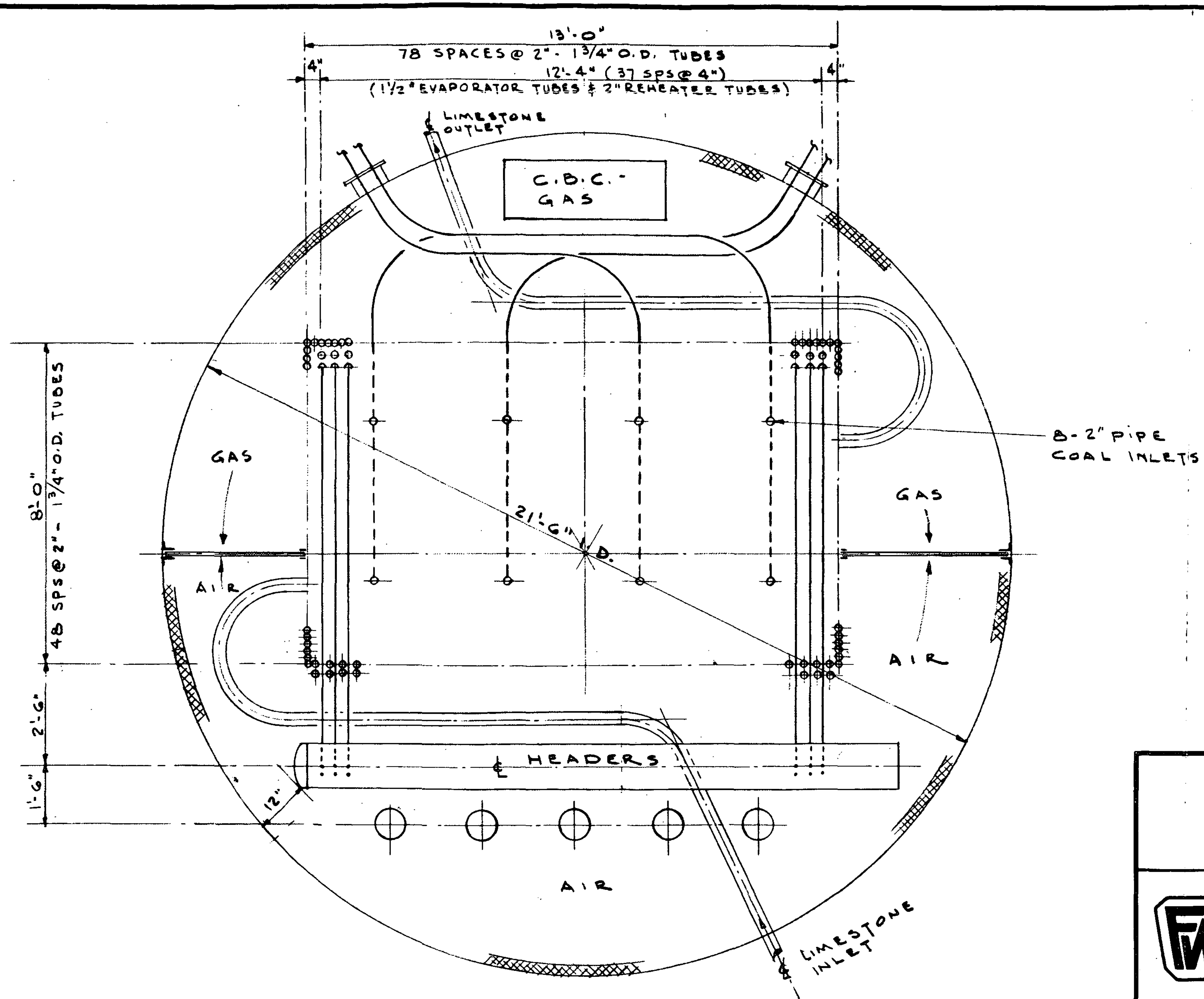
300 MW PLANT

FOUR (4) REQUIRED

SCALE 3/16" = 1'-0"
DRN. BY JR. 7/7/71

RD-711-155

A1.11



SECTION

ATMOSPHERIC FLUID BED BOILER

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APPENDIX A2

A2. ENERGY AND MASS BALANCE

The partial load energy and mass balance are summarized and illustrated in Tables A2-1 through A2-5.

TABLE A2-1

FLUIDIZED BED PARAMETERS

DESIGN LOAD

(ONE MODULE)

	FUEL FLOW LB/HR	AIR FLOW LB/HR	FLUE GAS LB/HR	SUPERFICIAL VEL. FT/SEC.	BED TEMPERATURE °F
REHEATER	11,600 (2)	0.126×10^6	0.137×10^6	10.9	1600 (3)
SUPERHEATER I	11,700 (2)	0.126	0.138	11.0	1600 (3)
SUPERHEATER II	11,700 (2)	0.126	0.138	11.0	1600 (3)
EVAPORATOR I	7,000 (2)	0.076	0.083	7.85	1600 (3)
EVAPORATOR II	7,000 (2)	0.076	0.083	7.85	1600 (3)
CBC	10,300 (1)	0.103 (4)	0.108	10.7	1900 (3)

(1) 59.45% Coal
33.75% Ash
6.80% Sorbent

(2) Fuel as Fired

(3) Specified by Westinghouse

(4) 70.5% X's Air Based on Coal Flow to CBC

TABLE A2-2

FLUIDIZED BED PARAMETERS

75% BOILER LOAD

(ONE MODULE)

	FUEL FLOW LB/HR	AIR FLOW LB/HR	FLUE GAS LB/HR	SUPERFICIAL VEL. FT/SEC	BED TEMPERATURE °F
REHEATER	8,630 (2)	0.0935×10^6	0.102×10^6	8.10	1425
SUPERHEATER I	8,790 (2)	0.095	0.104	7.91	1390
SUPERHEATER II	8,790 (2)	0.095	0.104	8.29	1433
EVAPORATOR	5,250 (2)	0.057	0.062	4.74	1350
EVAPORATOR II	5,250	0.057	0.062	4.74	1350
CBC	7,670 (1)	0.076 (4)	0.081	6.10	1900 (3)

(1) 59.45% Coal
33.75% Ash
6.80% Sorbent

(2) Fuel as Fired

(3) Specified by Westinghouse

(4) 47% X's Air Based on Coal Flow to CBC

TABLE A2-3

FLUIDIZED BED PARAMETERS

67% BOILER LOAD

(ONE MODULE)

	FUEL FLOW LB/HR	AIR FLOW LB/HR	FLUE GAS LB/HR	SUPERFICIAL VEL. FT/SEC	BED TEMPERATURE °F
REHEATER	7,760 (2)	0.084×10^6	0.092×10^6	7.00	1350
SUPERHEATER I	7,840 (2)	0.085	0.093	6.94	1310
SUPERHEATER II	7,840 (2)	0.085	0.093	7.10	1360
EVAPORATOR I	4,690 (2)	0.051	0.055	4.05	1299
EVAPORATOR II	4,690 (2)	0.051	0.055	4.05	1299
CBC	6,700 (1)	0.069	0.073	7.27	1900 (3)

(1) 57.45% Coal
33.75% Ash
6.80% Sorbent

(2) Fuel as Fired

(3) Specified by Westinghouse

(4) 43.5% X's Air Based on Coal Flow to CBC

TABLE A2-4

FLUIDIZED BED PARAMETERS

50% BOILER LOAD

(ONE MODULE)

	FUEL FLOW LB/HR	AIR FLOW LB/HR	FLUE GAS LB/HR	SUPERFICIAL VEL. FT/SEC	BED TEMPERA-TURE °F
REHEATER	5,800 (2)	0.063×10^6	0.068×10^6	4.80	1223
SUPERHEATER I	5,050 (2)	0.064	0.069	4.79	1170
SUPERHEATER II	5,850 (2)	0.064	0.069	4.70	1218
EVAPORATOR I	3,500 (2)	0.038	0.041	2.79	1144
EVAPORATOR II	3,500 (2)	0.038	0.041	2.79	1144
CBC	5,150 (1)	0.052 (4)	0.054	5.40	1900 (3)

(1) 59.45% Coal
33.75% Ash
6.80% Sorbent

(2) Fuel as Fired

(3) Specified by Westinghouse

(4) 21.4% X's Air Based on Coal Flow to CBC

TABLE A2-5

SUMMARY OF ENERGY AND MASS BALANCE
PARTIAL BOILER LOAD OPERATION
(ONE MODULE)

LOAD	100%	75%	67%	50%
Coal Flow (Total) lb/hr	49.5×10^3	37.2	32.8	24.5
Air Flow (Total) lb/hr	$.602 \times 10^6$	.474	.425	.319
Gas Flow (Total) lb/hr	$.658 \times 10^6$	.515	.461	.342
Reheater				
Bed Temp.	1600	1425	1350	1223
Conv. Exit Temp.	840	780	764	739
Heat-Bundle btu/hr	80×10^6	60.5	54.8	42.2
Walls btu/hr	9	6.9	6.1	4.8
Conv. btu/hr	26.8	18.3	14.7	9.1
Superheater I				
Bed Temp.	1600	1396	1310	1170
Conv. Exit Temp.	840	764	745	715
Heat-Bundle	85.5	65	58.8	44.7
Walls	9	6.9	6.1	4.8
Conv.	27.4	18.0	14.4	8.7
Superheater II				
Bed Temp.	1600	1433	1360	1218
Conv. Exit Temp.	840	764	745	708
Heat-Bundle	85.5	65	58.8	44.7
Walls	9	6.9	6.1	4.8
Conv.	27.4	18.5	15.5	9.7
Evaporator I				
Bed Temp.	1600	1350	1277	1144
Conv. Exit Temp.	840	750	726	687
Heat-Bundle	48	37.6	33.2	25.7
Walls	9	6.9	6.4	4.8
Conv.	16.3	10.7	8.6	5.4
Evaporator II				
Bed Temp.	1600	1350	1229	1144
Conv. Exit Temp.	840	740	717	668
Heat-Bundle	48	37.6	33.2	25.7
Walls	9	6.9	6.1	4.8
Conv.	16.3	12.2	11.1	8.4
Carbon Burn-Up Cell				
Bed Temp.	1900	1900	1900	1900
Conv. Exit Temp.	840	815	803	777
Heat-Walls	14.7	14.7	14.7	14.7
Conv.	29.6	21.2	18.2	11.7
Air Heater				
Mean Temp. Gas In	840	769	751	719
Temp. Gas Out	340	803	291	275
Air Temp. Out	780	720	700	685
FBC				
X's Air	10%	10%	10%	10%
CBC				
X's Air	70.5%	47%	43.5%	21.4%
Cyclone, Eff. η	96%	95%	95%	94%

APPENDIX A3

A3. OVERALL BOILER DESIGN

A3.1 Tube Design Information

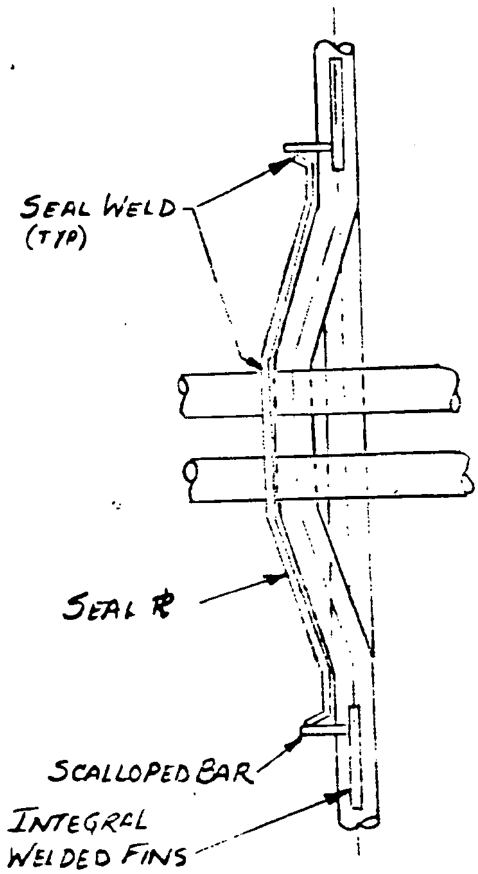
A3.1.1 Bundle Arrangement Horizontal tube bundles are used in all beds for evaporating superheating and reheating the water and steam. This arrangement makes use of conventional bundle fabrication techniques and maximizes the quantity of surface submerged in the bed. Long lengths of tubes to and from headers outside of the bed are virtually eliminated and return bends on each loop may be kept submerged. By using shallow to moderately deep beds the number of bends required are minimized. By maintaining 12 to 13 feet lengths for each loop separation of phases in the evaporating section should be reduced and the risk of tube burn-out eliminated. With the horizontal arrangement all tube bundles are drainable. This should simplify start up and minimize maintenance.

All bundles were constructed in a rectangular pattern and the tubes are on a square or rectangular pitch. All the tubes submerged in the bed have a 4" horizontal pitch and a 4" vertical pitch. The horizontal pitch was dictated by the spacings on the wall tubes forming the bed enclosure. This arrangement also simplified maintenance problems and should minimize the possibility of erosion. The vertical pitch was dictated by the minimum bending radius of the tubes. Bends with radii smaller than one diameter thin out the wall on the large diameter side of the bend subjecting the tube to rupture. The fluidizing characteristics of the 4" x 4" pitch was the subject of much controversy. Consideration should be given to the compromise that might be required between fabricating techniques, economics and fluidizing characteristics of the bed before modifying the present bundle design. The influence of bed design on fluidizing characteristics should be investigated experimentally to be certain any changes are truly justified.

Water walls forming the enclosure of the beds are of the wetted fin construction. They are constructed of 1 3/4" tubes on 2" centers requiring a minimum fin of 1/4". The spacing selected is the minimum that could be used with this type of construction. Smaller tubes would result in excessive pressure drop on the steam side.

Preheater bundles are arranged horizontally in the convection pass above the bed. Once again a rectangular pitch is used, this time to minimize gas side pressure loss. The first two loops have a vertical pitch of 4" and a horizontal pitch of 8". The wide open spacing provides for some subcooling of elutriated particles at low gas velocities before entering the more tightly spaced gas pass. The loop and loop construction of the lower bundle becomes a single loop construction in the upper bundle by simply displacing the inner loop. This is illustrated in Figure 4.1. The remaining portion of the bundle is constructed on a 4" x 4" square pitch.

Preheater tube bundles are connected one FBC to another with no intermediate header. All other bundles are connected to headers located adjacent to the tube walls in the cooler air ducts. The inlet and exit tubes from each bundle pass through the water walls by bending alternate water wall tubes off center in the immediate area of the penetration. The enclosure is sealed as illustrated in Figure A3-1.



DETAIL OF TUBE PENETRATION (TYPICAL)

ORDER NO.	0-59-1102
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APPROVED BY:	
SCALE:	= 1'-0"

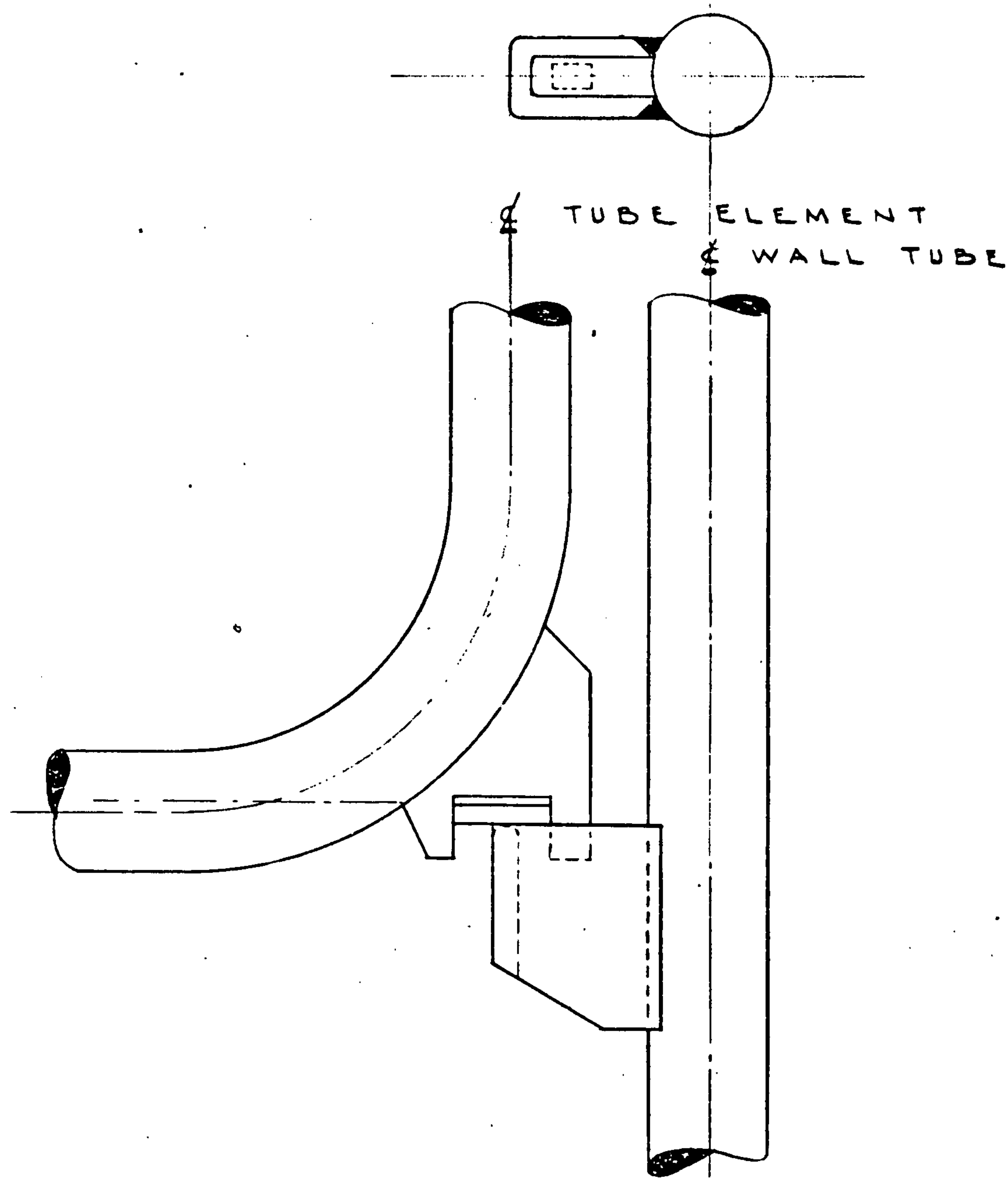
FIGURE A3-1

Bundles may be shop fabricated and shipped in a unit and installed in the field. Headers with tube stubs welded in the shop are attached to the bundles in the field. This type of construction limits the number of tubes contained in any loop to about three which is no problem in the bundle arrangement selected for the atmospheric unit. Very deep beds with a nearly square pattern would normally require a greater number of tubes to provide sufficient free area in the steam side to avoid high pressure drop. This would require more bed depth over and above that normally required by the tube bundle to offset the tubes in groups of two or three so that the bundle can penetrate the water wall enclosure and header stubs can be welded in the field. Although this is no problem here it is mentioned as a precaution when extrapolating the present design to larger capacities and possibly greater bed depths.

The width of the beds are set at about 13 feet. This is a reasonable limit to set for unsupported tube lengths. Extrapolation to larger capacity boilers should be done by simply expanding the bed width.

Tubes are supported as shown in Figure A3-2. Tubes that form the inner loop of a loop in loop construction are supported by off setting the outer loop tube just prior to the bend so as to reduce the distance between centerlines of adjacent tubes. The two loops are then connected by short welded fin connectors about 2 or 3 inches in length.

A3.1.2 <u>Tube Size</u> Two inch O.D. tubes are used in the preheater sections and reheater bundle. Superheater and evaporator tubes are 1 1/2" O.D. tubes. Enclosure wall tubes are 1 3/4" O.D.



TUBE ELEMENT
SUPPORT ASSEMBLY

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SCALE:	= 1'-0"

0-59-714-39

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Selection of tube size is a compromise between many factors. The final selection is not arrived at explicitly but must be achieved by trial and error techniques using a good deal of judgment and experience along the way. The final solution, therefore, may not be the optimum. However, alternate solutions at this point should not significantly influence the overall cost of the boiler.

Surface requirements, geometric limitations of the bundle and steam side pressure drop or mass flow per unit area are prime variables in selecting tube size. Bundle geometry fixes the tube orientation and influences the gas side flow parameters and number of tubes per loop. Surface requirements fix tube length and total number of loops required. Steam mass flow rates and pressure drop dictate the cross-sectional area and number of tubes required.

Once the bed geometry is fixed the problem becomes one of selecting the number of tubes required to provide sufficient heating surface. Connecting the tubes in adjacent rows in single loop or loop-in-loop fashion depends on the number of tubes required and thus the mass flow velocity and pressure drop requirements. A small pressure drop is essential to establish good distribution. However, the pressure drop must not be so large that the pumping requirements become excessive. There are obvious discontinuities in selecting the number of tubes and number of rows to match steam flow conditions. The mere step from single loop to loop-in-loop decreases the tube length by a factor of two and doubles the steam mass flow area. This problem may be partially resolved by selecting different tube sizes and altering the tube pitch slightly. Thus tube size selection offers an additional degree of freedom in bundle design.

Tube size is also dependent on pressure requirements, material selection and tube wall temperature. Large diameter, low alloy steel tubes used in high pressure service must be thick. There outside surface temperatures, therefore, must be high. Smaller diameter tubes made of high alloy steel might be considerably thinner with some reduction in tube metal surface temperature. This is clearly shown by the ASME Code formulation for selecting tube metal thickness.

$$T = \frac{PD}{2S+P} + 0.005D$$

T = Thickness
P = Pressure
D = Diameter
S = Allowable Stress

Selection of tube size based on simple mechanical design is a compromise made between size and material selection to give the most economical solution.

At high pressures small diameter tubes with thin walls and low metal surface temperatures would appear to offer the best approach to bundle design. Limitations, however, are imposed by process design. As the tube diameter decreases the number of tubes required to handle the steam flow increases inversely with the change in diameter of the tube to second power. Diameter in this case must refer to the free flow area. When the diameter is expressed in terms of the outside tube diameter tube thickness must be taken into consideration. The increase in number of tubes required becomes even more pronounce as the outside tube diameter decreases. Taking into consideration the increase in number of tubes required the length of the tube decreases in direct proportion to the decrease in outside diameter. This means that the depth of the bundle must increase considerably unless the gas side dynamics are adjusted to accommodate the new tube size. With the reduction in length and little change in fluid dynamics on the steam side the pressure drop is greatly reduced. Ultimately compromises may have to be made between numbers of tubes and steam side pressure drop which upsets the fluid dynamics and heat transfer on the steam side.

Installation of the bundle becomes complicated by the change in number of tubes and spacing. Tubes must be bent out of line to penetrate the water wall enclosure. If more than three rows of tubes are required to connect the headers and the bundles they will have to be separated into groups or legs to make welding to the header stubs feasible. This atuomatically increases the bed size. An increase in the number of tubes must also increase the man hours to fabricate the bundle or assemble it in the field which might mitigate any savings in material cost.

All these factors were taken into consideration in arriving at the tube sizes specified for the fluid bed boiler.

A3.1.3 <u>Tube Temperature and Material Selection</u> The metal temperatures and material selection for the tubes and headers along with sizes and thickness are summarized in Figure A3-1. The data is coded so that the information may be related to the circuitry illustrated in Figure 4.1. In arriving at tube metal

ONCE-THRU FLUID BED STEAM GENERATOR CIRCUIT DESIGN SUMMARY (cont'd)

KEY	LOCATION	DESIGN PRESSURE (PSI)	DESIGN TEMP. (F)	TUBES				RISERS AND DOWNCOMERS									
				NO. OF TUBES	OUTSIDE DIA. (IN.)	NOMINAL THICKNESS (IN.)	MATERIAL	CONNECTIONS OR DOWNCOMERS	OUTSIDE DIA. (IN.)	NOMINAL THICKNESS (IN.)	MATERIAL	OUTSIDE DIA. (IN.)	NOMINAL THICKNESS (IN.) IN. HEADERS	MATERIAL	OUTSIDE DIA. (IN.)	NOMINAL THICKNESS (IN.) OUT. HEADERS	MATERIAL
17	FEEDERS TO PASS 5	3010	700					6	3	.300mw	SA-210						
18	PASS 5 SIDE WALL	3010	975	71	1-3/4	.284mw	SA-213 T-22					8-5/8	1.13aw	SA-106C	8-5/8	1.13aw	SA-106C
19	RISERS FROM PASS 5	3010	700					6	3	.300mw	SA-210						
20	DOWNCOMER TO PASS 6 REAR WALL	3010	700					1	8-5/8	SCH XX	SA-106C						
21	FEEDERS TO PASS 6	3010	700					6	3	.300mw	SA-210						
22	PASS 6 REAR WALL	3010	975	76	1-3/4	.284mw	SA-213 T-22					8-5/8	1.13aw	SA-106C	8-5/8	1.13aw	SA-106C
23	RISERS FROM PASS 6	2760	700					8	3	.280mw	SA-210						
24	DOWNCOMER TO PASS 7 SUPERHEATER	2760	700					1	8-5/8	SCH 140	SA-106C						
25	FEEDERS TO SUPERHEATER	2760	700					8	3	.280mw	SA-210						
26	PASS 7a SUPERHEATER RUNS 1, 2, 3	2760	875	76	1-1/2	.150mw	SA-213 T-2					8-5/8	SCH 160	SA-106C			
26	PASS 7b SUPERHEATER RUNS 4, 5, 6	2760	930	76	1-1/2	.220mw	SA-213 T-2					8-5/8	SCH 160	SA-106C			
27	SUPERHEATER RUNS 7, 8, 9, 10	2760	1075	76	1-1/2	.280mw	SA-213 T-22							*			
27	SUPERHEATER RUNS 11, 12	2760	1175	76	1-1/2	.238mw	TP-304H					10-3/4	1.50aw	SA-335 P-22			
28	REHEATER INLET LOOP	700	-	76	2	.165mw	SA-213 T-22					12-3/4	SCH ST-405	SA-106C			
48	REHEATER OUTLET LOOP	700	-	76	2	.165mw	TP-304H					14	SCH XS	SA-335 P-22			

Fig. A3-1

ONCE-THRU FLUID BED STEAM GENERATOR CIRCUIT DESIGN SUMMARY

KEY	LOCATION	DESIGN PRESSURE (PSI)	DESIGN TEMP. (F)	TUBES				RISERS AND DOWNCOMERS									
				NO. OF TUBES	OUTSIDE DIA. (IN.)	NOMINAL THICKNESS (IN.)	MATERIAL	NO. OF CONNECTIONS OR DOWNCOMERS	OUTSIDE DIA. (IN.)	NOMINAL THICKNESS (IN.)	MATERIAL	OUTSIDE DIA. (IN.)	NOMINAL THICKNESS (IN.) IN. HEADERS	MATERIAL	OUTSIDE DIA. (IN.)	NOMINAL THICKNESS (IN.) OUT. HEADERS	MATERIAL
1	FEED PIPE PASS I INLET	3260	495					1	8-5/8	1.04 aw	SA-106C						
2	PASS I PREHEATER	3260	710	38	2	.220mw	SA-210A					8-5/8	1.04aw	SA-106C	8-5/8	1.10aw	SA-106C
3	RISERS FROM PASS I	3260	700					6	3	.300mw	SA-210						
4	DOWNCOMER TO PASS 2 (EVAP)	3110	700					1	6-5/8	SCH 160	SA-106C						
5	FEEDERS TO PASS 2	3110	700					6	3	.300mw	SA-210						
6	PASS 2a (EVAP) PASS 2b (EVAP)	3110 3110	INLET 850 OUTLET 890	38 38	1-1/2 1-1/2	.165mw .180mw	T-2 T-2		8-5/8 8-5/8	1.00aw 1.00aw	SA-106C DES. T=700 SA-106C DES. T=700						
7	RISERS FROM PASS 2	3010	700					6	3	.300mw	SA-210						
8	DOWNCOMER TO FRONT WALL PASS 3	3010	700					1	6-5/8	SCH 160	SA-106C						
9	FEEDERS TO PASS 3	3010	700					6	3	.300mw	SA-210						
10	PASS 3 FRONT WALL	3010	975	76	1-3/4	.284mw	SA-213 T-22		8-5/8	1.13aw	SA-106C	8-5/8	1.13aw	SA-106C	8-5/8	1.13aw	SA-106C
11	RISERS FROM PASS 3	3010	700					6	3	.300mw	SA-210						
12	DOWNCOMER TO PASS 4 SIDE WALL	3010	700					1	8-5/8	SCH XX	SA-106C						
13	FEEDERS TO SIDE WALL	3010	700					6	3	.300mw	SA-210						
14	PASS 4 (SIDE WALL)	3010	975	71	1-3/4	.284mw	SA-213 T-22					8-5/8	1.13aw	SA-106C	8-5/8	1.13aw	SA-106C
15	RISERS FROM SIDE WALL	3010	700					6	3	.300mw	SA-210						
16	DOWNCOMER TO PASS 5 (SIDE WALL)	3010	700					1	8-5/8	SCH XX	SA-106C						

temperature it was assumed that no unbalance in operating conditions existed on the gas side and 20% unbalance may exist on the steam side. It was also assumed that no corrosion or erosion problems existed in the bed on the gas side. In several cases, material selection was based on economic factors alone. It was felt that it would be cheaper to use a high alloy steel throughout the bundle rather than use two grades of steel.

A3.1.4 Overall Heat Transfer Coefficient In each fluidized bed a bed-to-tube heat transfer coefficient of 50 Btu/hr-ft^2-°F was used for the bed proper. A zone extending two feet above the bed was considered to be a transition zone between fluidized bed heat transfer and gas convection. The mechanism of heat transfer in this zone would mainly be particles that rise above the relatively dense phase of the bed transfer heat and then fall back into the bed. A bed-to-tube heat transfer coefficient of 40 Btu/hr-ft^2-°F was assumed and the zone was considered to be at the same temperature as the bed. The overall heat transfer coefficients varied throughout the cycle depending on the liquid side heating function. Overall heat transfer coefficients for the evaporator, superheater and reheater ran about 47, 45 and 43 Btu/hr-ft^2-°F, respectively.

In the evaporator the water side coefficient varies according to the mode of heat transfer taking place, nucleate boiling, departure from nucleate boiling or film boiling. Numerous correlations exist. Each manufacturer has his own preference and generally uses data developed in his own laboratory which he keeps confidential. There correlations are cited as typical examples. These are not necessarily used by Foster Wheeler or any other boiler manufacturer.

Nucleate (8)

$$h = S(0.00122)\,\frac{k_1^{0.79} c_1^{0.45} \rho_1^{0.49} g_e^{0.25} \Delta T^{0.24} \Delta p^{0.75}}{\sigma^{0.5} \mu_1^{0.29} H_{fg}^{0.24} \rho_v^{0.24}} \quad *$$

$$+F(0.023)(Re)_1^{0.8}(Pr)_1^{0.4}\,\frac{k_t}{D_e}$$

DNB (Departure from Nucleate Boiling) (8)

DNB (low flow)

$$\frac{q''_{crit}}{10^6} = 0.00633 H_{fy}\, d^{-0.1} \left(\frac{G}{10^6}\right)^{0.51} (1-\chi_e)$$

(high flow)

$$\frac{q''_{crit}}{10^6} = \frac{A + 1/4\,C d (G \times 10^{-6})(H_1 - H_{in})}{(1+CL)}$$

Film Boiling (8)

$$\frac{h_c D_e}{k_{w,v}} = 0.005 \left(\frac{D_e V_m \rho_{w,v}}{\mu_{w,v}}\right) \left(\frac{c_p \mu}{k}\right)_{w,v}^{1/2}$$

In the cases where superheated steam is being heated rather than boiling water the following correlation may be used.

$$\frac{U_{ot} D_t}{k} = 0.023 \left\{\frac{G D_t}{\mu}\right\}^{0.8} \left\{\frac{c p \mu}{k}\right\}^{0.4} \left\{\frac{T_b}{T_f}\right\}^{0.8}$$

(8) L. S. Tong, "Boiling Heat Transfer and Two-Phase Flow", John Wiley and Sons, Inc., New York, 1965.

* Nomenclature appears at the end of this section.

A3.1.5 Distribution of Surface The distribution of surface to the waterwalls, superheater, reheater, etc., is summarized in Table A3-2.

A3.2 Mechanical Design

A3.2.1 Duct Work The duct work connects the steam generator with the dust collector, air preheater, electrostatic precipitator, fans and air to the coal injection system and provides a means of transporting air and gas to and from the steam generator. The air and gas streams are illustrated in Figure 4.2. The duct work also runs the full length of the front and back of the steam generator providing a windbox for the distribution of hot air and a means of collecting hot gases. A seperate duct as contained in the flue at the gas exit from the steam generator to segregate and carry off hot gases from the CBC to the dust collector. This makes it possible to segregate ash from the FBC and carbon rich ash from the CBC.

The duct work is rectangular in shape and constructed of gauge steel lagged with 3" of batt insulation suitable for outdoor exposure. The duct work contains expansion points and dampers for control of gas flow where it is felt necessary.

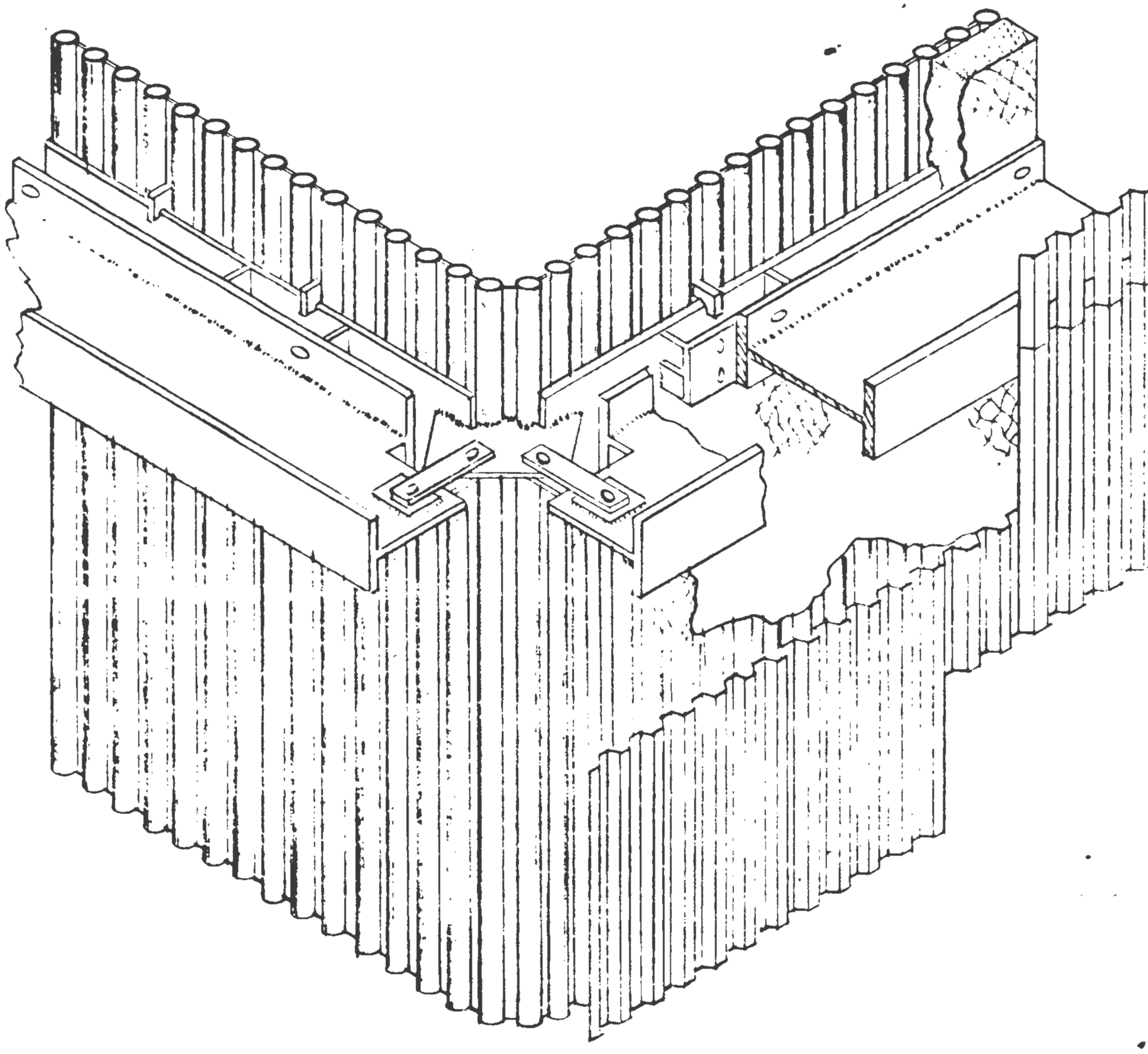
Sides of the steam generator not protected by a windbox or flue are enclosed with insulation and corrogated sheathing as illustrated in Figure A3-3.

A3.2.2 Plenum Chamber and Distribution Plate The plenum chamber isolates one fluid bed from the other, provides a means of distributing air and fuel in the bed and supports the bed aggregate. The plenum chamber is illustrated in Figure A3-4

TABLE A3-2

COMPARISON OF HEAT TRANSFER SURFACE AREA

FUNCTION	ATM. 300 MW DESIGN $1.9x10^6$ lb Steam/hr 2400 psig/1000°F/1000°F		ATM. 600 MW SCALE-UP $3.8x10^6$ lb steam/hr 2400 psig/1000°F/1000°F	CONVENTIONAL BOILER (F.W. HAMMOND NO. 4) $3.6x10^6$ lb steam/hr 2486 psig/1000°F/1000°F
Evaporator	$2.64x10^4$	-FBC-	$5.28x10^4$	$2.97x10^4$
Superheater I	.800	-FBC-	1.75	$8.16x10^4$
II	.950	-FBC-	1.75	
Reheater	.825	-FBC-	1.650	$11.30x10^4$
Economizer	5.72	-CONV. PASS-	11.44	$15.36x10^4$
TOTAL	10.93×10^4		21.86×10^4	$37.79x10^4$
Air Preheater	TUBULAR $104x10^4$		$208x10^4$	REGENERATIVE $53.0x10^4$



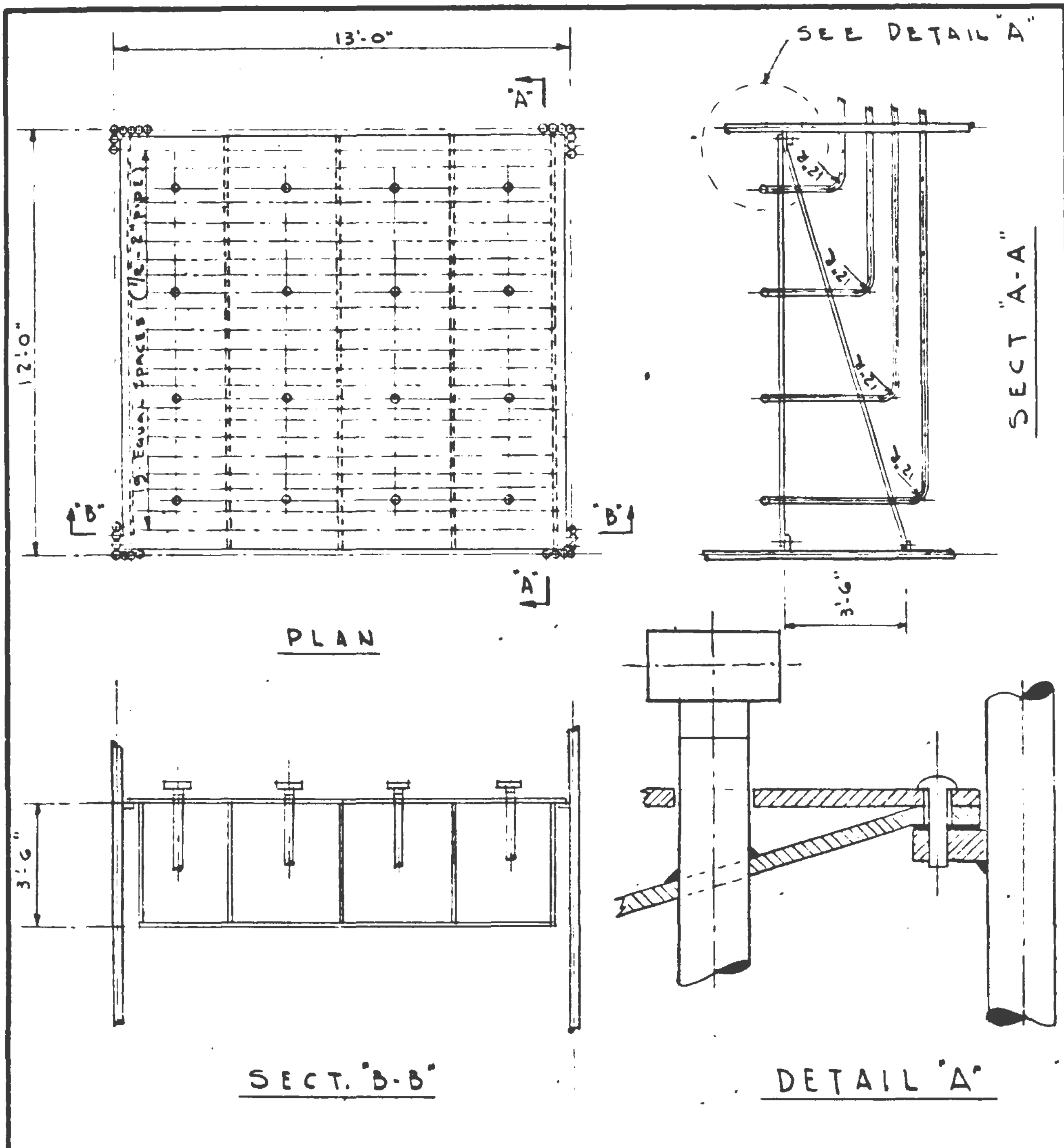
FIN-TUBE WALL CONSTRUCTION

ORDER NO.	0-59-1192
DRAWN BY:	JR 7/8/71
CHECKED BY:	
APPROVED BY:	
SCALE:	= 1' - 0"

RD-714-136

C. B. CO., NO. 44-813

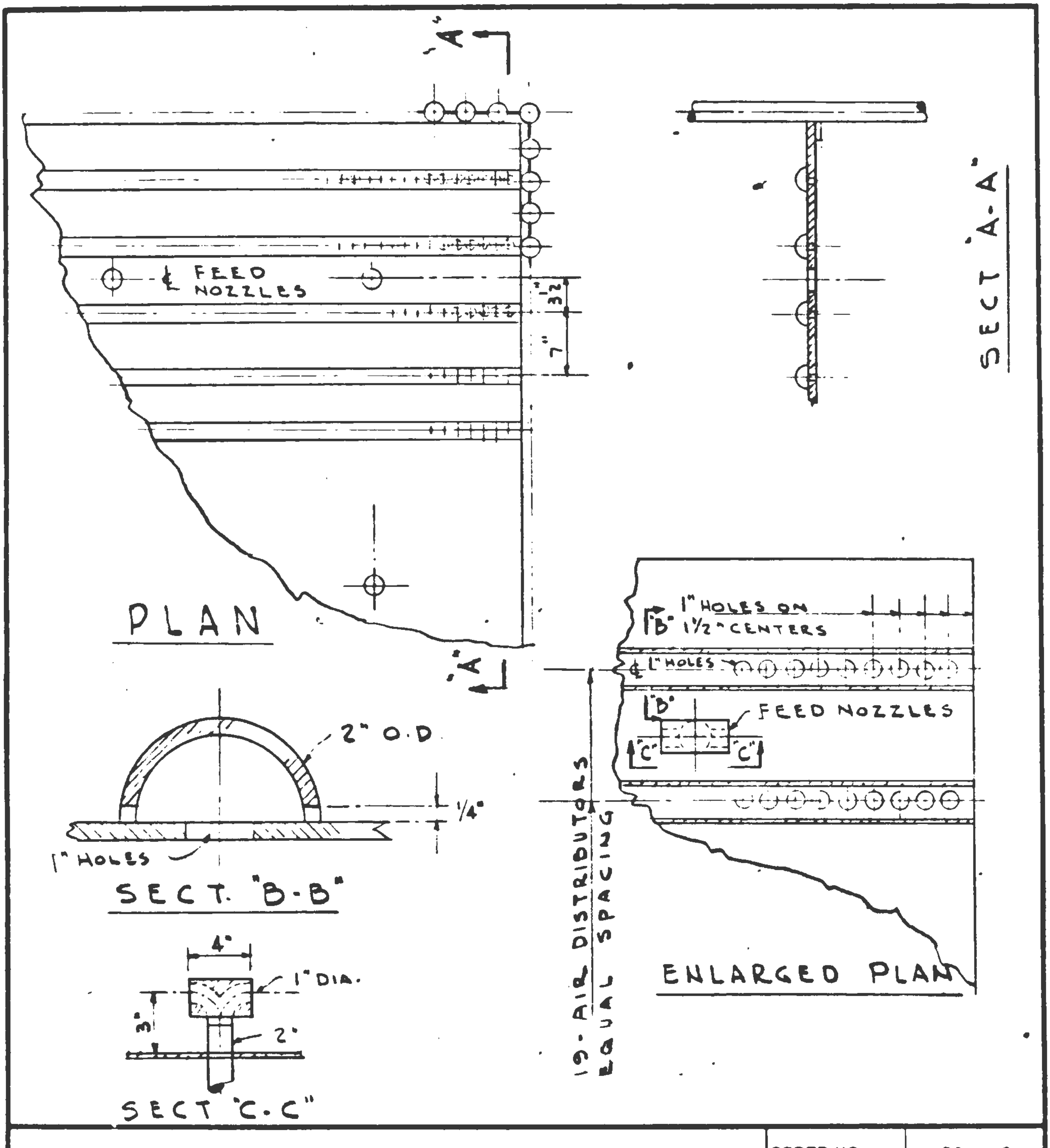
FIG. A3-3



PLENUM CHAMBER DETAIL

ORDER NO.	0-59-1192
DRAWN BY:	JR 5/19
CHECKED BY:	
APPROVED BY:	
SCALE:	= 1' - 0"

RD-714-120A



DISTRIBUTOR PIPE DETAIL	ORDER NO.	0-59-1192
	DRAWN BY:	JR 5/19
	CHECKED BY:	
	APPROVED BY:	
	SCALE:	= 1'-0"
		RD-714-119A

C. B. CO., NO. 44-812

FIG. A3-5

and A3-5.

It consists of two steel plates separated by wedge shaped stiffeners welded to the lower plate. The two stiffeners closest to the water walls also act as enclosure walls separating the air stream to one fluid bed from the gas stream from the adjacent fluid bed. The lower plate to which the stiffeners are attached is supported by lips resting on scallop bars welded to the water wall enclosure. The upper plate or distributor rests on top of the stiffeners and scallop bars.

Coal feed lines penetrate the rear water wall enclosure of the gas off take points where the tubes have already been separated to provide for gas departing from the fluid bed. They make a right angle turn and penetrate the lower plate forming one side of the plenum chamber. They are seal welded at the point of penetration. They proceed up through the distribution plate to a coal nozzle which distributes the coal in the bed in the horizontal plan in opposite directions.

The coal is distributed in the bed in a mixing zone just below the tubes. Sixteen feed points are used to insure diffusion of the feed in the bed. This provides about one feed point for each ten square feet of bed surface as recommended by the British.

Air enters the plenum chamber through dampers on one side of the boiler. The stiffeners act as guide vanes to insure good distribution of air in the plenum chamber. The air then passes up through the one inch holes in the grid plate aligned in rows on 7" centers. The holes are drilled on 1 1/2" centerlines. Once through the holes the air is enclosed in a slotted channel constructed from tubular material or angle iron. This makes it possible to reduce the air over the distribution plate as it enters the bed, sweeping the plate clean of particluate matter and cooling it at the same time. The tubular distributor should also prevent the back flow of particulate matter during shut-down. The angle of repose between the slot and the hole in the distributor plate is too small to allow the flow of small particles back into the plenum chamber.

The pressure drop through the distributor is estimated at .4 times the pressure drop through the bed.

Surfaces not swept clear with air are curved to prevent collection and agglomeration of ash particles.

A3.2.3 <u>Maintenance</u> Throughout the design of the steam generator compromises have been made between economy of design and ease of maintenance. Manholes have been provided for access to windbox, duct work, flues and crawl space above each bed. Sufficient room has been provided in each bed for removal of individual tube element from the convection pass or bed. Access to the plenum chamber may be had through the dampers.

Repair of tube leaks in the bundle would simply require cutting the element at either end of the bundle and dropping it into the crawl space for maintenance. Complete replacement of an element would require cutting in sections for removal and welding in sections for installation. Providing erosion does not become a problem in the bed, the maintenance on the tube surface should be minimal. Fouling of tubes supposedly is eliminated in fluid bed operation.

Replacement of entire tube banks or headers would require a major disassembly of the module. The need for this type of maintenance is unlikely. All anticipated normal maintenance appears to be feasible.

Elimination of soot blowers, mills and exhausters from the list of boiler assessories should also reduce the maintenance required by the boiler. Constructing the boiler in four module units should limit the complete down time for maintenance by isolating the module requiring repair. By operating three modules at 8 to 9% overload the entire system could be kept at full load during repair periods.

A3.2.4 <u>Expansion</u> The pressure parts are supported from the top allowing the unit to expand downwards. This is accomplished by hangers that connect the upper wall headers and the structural steel. All penetrations made through the side of the boiler move freely with the wall. Duct work, tube wall and air and gas, to and from the boiler, are at about the same temperature, thereby, minimizing expansion problems between various component parts.

Coal feed lines enter the boiler through the duct work in the wall opposite from the side of the bed it is feeding. This should provide a sufficient length of unsupported tubing to absorb any differences in expansion between the fixed support points.

TABLE A3- 3

NOMENCLATURE

h,U	-	Heat Transfer Coefficient	Btu/hr-ft^2-°F
K	-	Thermal Conductivity	Btu/hr-ft-°F
g	-	Acceleration of Gravity	ft/hr^2
Cp	-	Specific Heat at Constant Pressure	Btu/lb-°F
ℓ	-	Density	lb/ft^3
μ	-	Viscosity	lb/ft-hr
δ	-	Surface Tension	lb/ft
T	-	Temperature	°F
H	-	Enthalpy	Btu/lb
Hfg	-	Latent Heat of Vaporization	Btu/lb
G	-	Mass Velocity	lb/hr-ft^2
g"	-	Heat Flux	Btu/hr-ft^2
V	-	Velocity	ft/hr
X	-	Quality	
D	-	Hydraulic Diameter	ft
L	-	Length	ft
Pr	-	Prandtl Number	Cμ/k
Re	-	Reynolds Number	DρG/μ
S	-		$(\Delta To/\Delta T)^{0.59}$
F	-	Reynolds Number Factor	$(Re/Re_L)^{0.8}$

<u>Subscripts</u>

c	-	Core Condition
f	-	Film Condition
e	-	Exit Condition
w	-	Wam Condition
d	-	Saturated Liquid
ν	-	Saturated Vapor

APPENDIX A4

A4. ACCESSORY EQUIPMENT

A4.1 Coal Handling System and Limestone Make-Up

Coal handling for the atmospheric boiler includes receiving the coal, storage, transportation to the surge bins of the injector, sizing and crushing. McNally Pittsburg developed the coal handling plant and a price including erection.

A4.1.1 Assumptions The coal handling plant was designed on the basis of the following assumptions:

The coal to be used is Ohio Pittsburgh Seam No. 8 described in detail in Table 2.1 of the text.

The size of the coal is 1 1/2" x 0" as delivered. It was initially assumed that the coal would be sized to 5" x 0". According to the Bureau of Mines at Bruceton, there is no penalty paid for the 1 1/2" x 0" coal as it is a standard size commonly requested. Under these circumstances it was felt justified to revise the specifications. In addition to meeting the technical requirements, there appears to be a savings in cost of power required for crushing the smaller coal size.

Size distribution of coal crushed to 1 1/2" x 0" at the mine appears in Figure 2.1 of the body of the report.

Moisture as mined, is 3.3% total and 2.1% inherent. The difference is surface moisture.

Moisture as-delivered is 10% total. This is about 6.7% in excess of the as-mined coal and represents a typical moisture pick-up during handling and storage in an eastern steam generating plant location.

Moisture required at the boiler feed system is a maximum of 3.0% total moisture. Higher moisture contents result in agglomeration in the surge bins and feed lines.

Size required at the boiler feed system is 1/4" x 0".

The Hardgrove grindability index is 60.

The burning rates which affect hopper sizes, conveyor speeds, etc., were set at 113 tons per hour; 2,712 tons per day and 18,984 tons per week. Allowance was made for handling of limestone feed or limestone make-up, as it was not certain at what point it would be added to the coal stream.

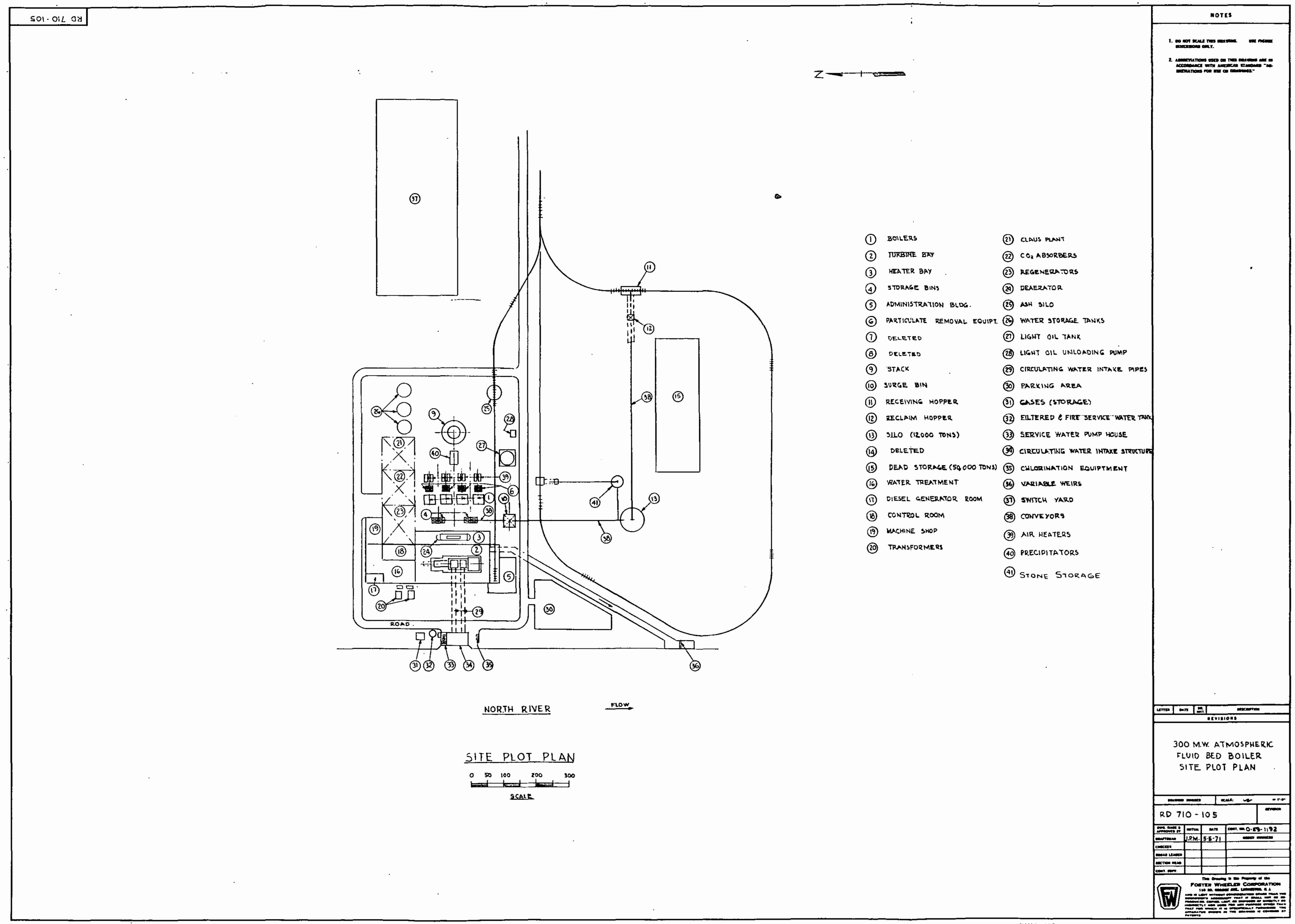
Storage of the coal is divided into two categories, active storage to handle short term contingencies, such as weekend outages for maintenance and repair, and dead storage to handle long term outages due to short strikes, delays in shipment, etc. Active storage was set at three days or 8,136 tons. The storage capacity of the silo was rounded off to 12,000 tons.

Dead storage was set at two weeks or 37,968 tons. It was rounded off to 50,000 tons of open coal pits.

The values were selected as typical based on the recommendation of McNally.

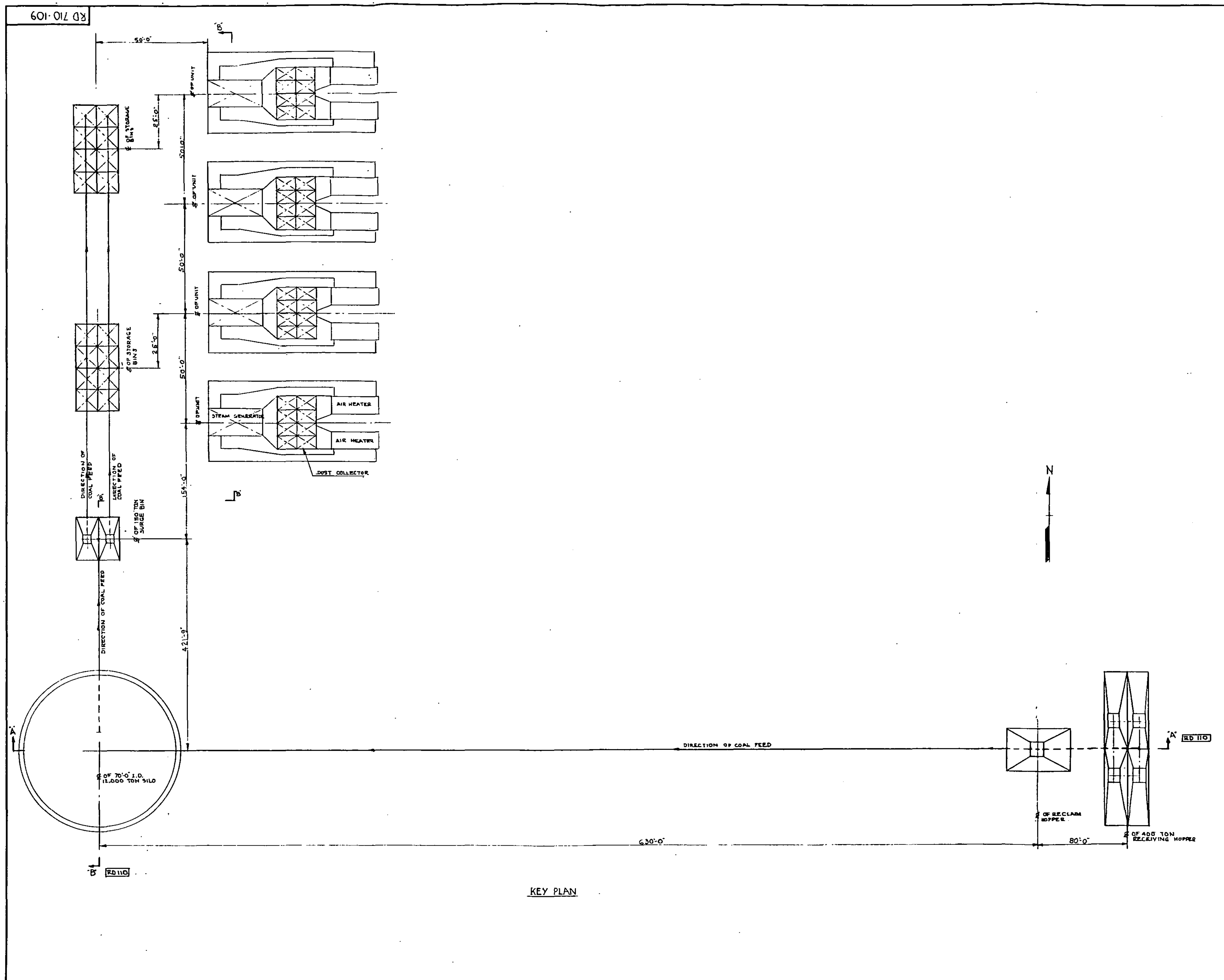
Dead storage requirements are vulnerable to considerable change according to individual customer's requirements. Changes should only affect operating cost and land requirements. The cost of coal handling should be marginal.

It was assumed that the coal would be delivered by 10,000 ton unit trains at a rate of 2 per week. The unloading rate is



DWG. NO. RD 710-105

Fig. A4-1



NOTES

1. DO NOT SCALE THIS DRAWING. USE FIGURE DIMENSIONS ONLY.
2. ABBREVIATIONS USED ON THIS DRAWING ARE IN ACCORDANCE WITH AMERICAN STANDARD "ABBREVIATIONS FOR USE ON DRAWINGS."

REFERENCE DWGS.

1-300 M.W. ATMOSPHERIC FLUIDIZED BED STEAM GENERATOR CONVEYOR SYSTEM SHT. 2 OF 2 RD710-110

REVISIONS

300 M.W. ATMOSPHERIC FLUIDIZED BED STEAM GENERATOR CONVEYOR SYSTEM

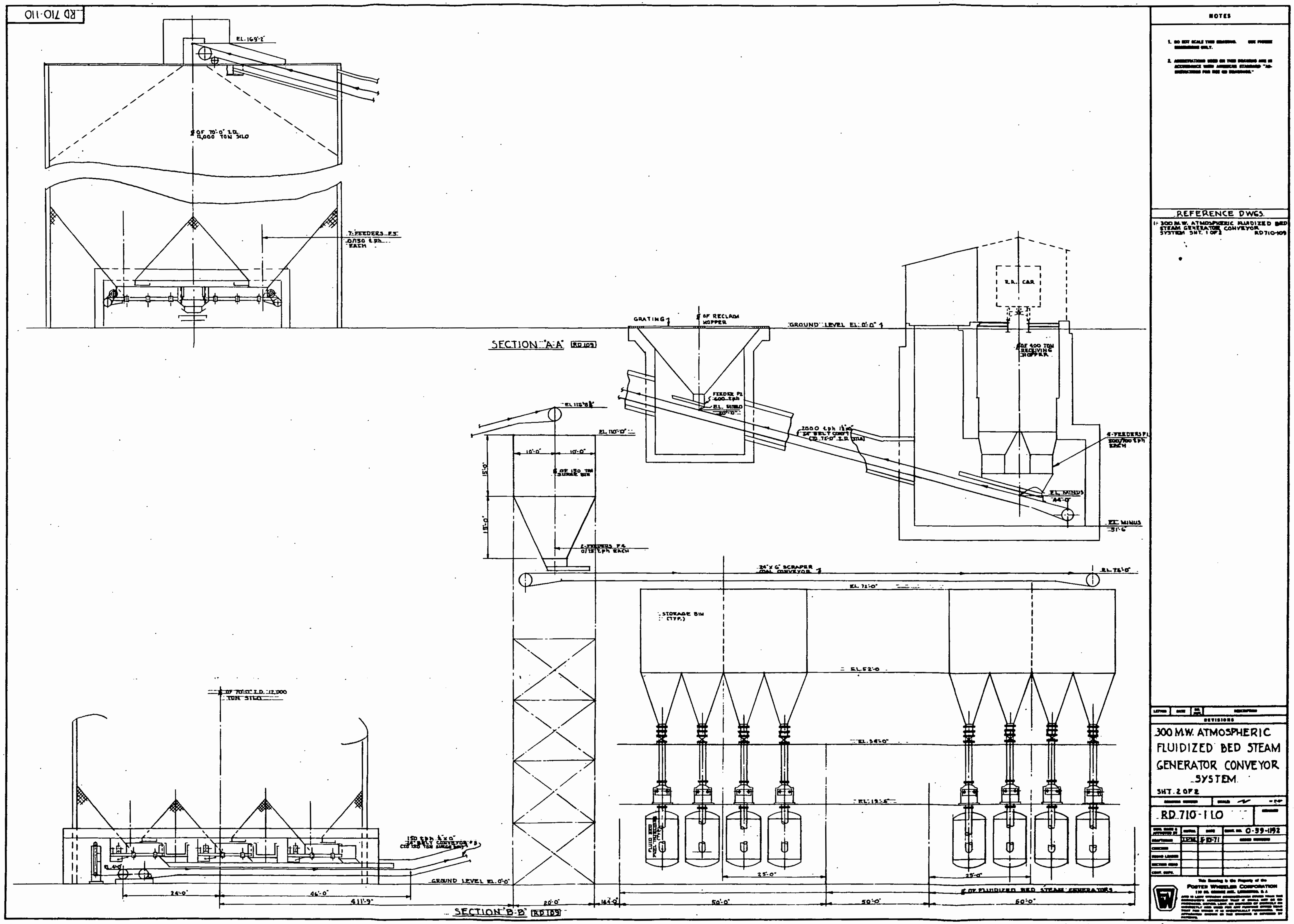
SHT. 1 OF 2

RD 710-109

CONT. NO. 0-59-1192

DRAFTSMAN J.P.M. 5/6/71

FOSTER WHEELER CORPORATION


RD 710-110
EL. 169'-2"
7-FEEDERS F5
SECTION "A-A" RD 109
GRATING
GROUND LEVEL EL. 0'-0"
R.R. CAR
EL. 110'-0"
10'-0"
10'-0"
15'-0"
15'-0"
2-FEEDERS F4
24" x 6" SCRAPER COAL CONVEYOR
EL. 72'-0"
STORAGE BIN (TYP.)
EL. 52'-0"
25'-0"
25'-0"
50'-0"
50'-0"
60'-0"
GROUND LEVEL EL. 0'-0"
24'-0"
46'-0"
20'-0"
SECTION "B-B" RD 109
NOTES
REFERENCE DWGS.
REVISIONS
300 M.W. ATMOSPHERIC FLUIDIZED BED STEAM GENERATOR CONVEYOR SYSTEM
SHT. 2 OF 2
RD 710-110
0-39-1192
FOSTER WHEELER CORPORATION

2,000 tons per hour and the train would require 5 hours to deliver its load.

The dead storage would require 49 weeks to complete to a capacity of 50,000 tons.

A4.1.2 Scope The system begins with the unloading of bottom dump rail cars in unit train delivery and continues through active and dead storage, crushing and delivery to the coal feeder surge silos provided by other vendors ahead of the boiler firing system.

The rail car receiving hopper will have a minimum of 400 tons capacity to provide storage volume for controlling unloading of moving cars. Details of the receiving hopper and other components are illustrated in Figures A4-2 and A4-3. Their relationship to other components in the coal handling system is illustrated in Figure 4.8. Their relationship to the entire power plant facilities is shown in Figure A4-1.

The coal discharge from the receiving hopper will be controlled by four reciprocating feeders, F1-A, F1-B, F1-C and F1-D, each feeder nominally rated at 500 tph but capable of delivering 700 tph if one feeder is out of service.

A 54" wide inclined belt conveyor No. 1 will transport the coal at the rate of 2000 tph to the silo. Conveyor No. 1 will be equipped with a tramp metal detector and a belt scale for weighing and recording coal quantities delivered to the plant site.

A 12,000 ton silo will nominally contain an entire unit train delivery without exposure to wind blown dust and provide active storage for plant feed without the use of mobile equipment. Excess coal will overflow the silo to an initial pile for placing in permanent storage. See Figure 4.8.

During extended periods of no-train delivery, coal will be recovered from permanent storage through a reclaim hopper and Feeder No. F2 delivering to Conveyor No. 1 for refilling the silo.

Coal will be fed from the silo by seven feeders No. F3A to F3G, each rated a maximum of 150 tph and operating in timed sequence for uniform drawdown of the silo, onto a 24" wide belt conveyor No. 2. Conveyor No. 2 will be fitted with a tramp iron magnet before delivery to the surge bin. Details of the magnets and dust recovery equipment have not been illustrated.

Coal from the silo will be crushed to 1/4" x 0" size in a reversible hammermill and, along with the coarse dust from the dryer cyclone, delivered via 24" belt conveyor No. 3 to the plant surge bin. Conveyor No. 3 will be fitted with belt scale No. 2 to weigh and record coal quantities to the boilers.

From the 150 ton capacity plant surge bin the coal flow will be split into two streams each served by a 0 to 75 tph vibrating Feeder F4A and F4B to its associated coal feeder surge silo group through scraper conveyors No. 4A and 4B.

The silo filling system will be provided with automatic sequential controls with provisions for operator to override to accommodate unusual operating conditions. The plant surge bin level will control the feed to the dryer and in turn the dryer surge bin will regulate the silo withdrawal feeders.

A4.1.3 Equipment The coal handling system will consist of the following equipment, materials and services:

1. Receiving Hopper
2. Feeders No. F1A, F1B, F1C and F1D
3. Chutes
4. Conveyor No. 1
5. Metal Detector
6. Belt Scale No. 1
7. Reclaim Hopper
8. Shut-Off Gate
9. Feeder No. F2
10. Silo
11. Feeders No. F3A and F3G
12. Conveyor No. 2
13. Tramp Iron Magnet
14. Mill
15. Surge Bin
16. Feeders No. F4A and F4B
17. Conveyors No. 4A and 4B
18. Dust Collecting Equipment
19. Ventilating and Heating
20. Control Equipment
21. Motors and Controls
22. Structures
23. Foundations
24. Electric Wiring
25. Erection

A4.1.4 <u>Cost</u> McNally Pittsburg indicates the cost for the system described should be about $1,750,000. The complete system contributes about $5.8/KW to the overall cost at the 300 MW level.

McNally Pittsburg indicates that for an increase in capacity to handle a burn rate of 226 tons per hour, the conveyor sizes should stay substantially the same; however, their horsepower would also increase. The probable additional cost for increasing the system might amount to $250,000 or about $2,000,000. This represents about $3.3/KW at the 600 MW level.

The costs are comparable as shown in Table A4-1

TABLE A4-1 COMPARISON OF CAPITAL
COST OF COAL HANDLING EUQIPMENT

Plant Size (MW)	100	200	230	246	327	300 Fluid Bed	600	600 Fluid Bed
Coal Handling and Storage	1010*	1720	1860	1925	2240	1750	3122	2000
$/KW	10.10	8.6	8.1	7.9	6.85	5.80	5.20	3.30
Fuel Burning	760	1295	1400	1450	1685	1,147	†	2,294
$/KW	7.60	7.4	6.1	5.9	5.15	3.8	-	3.8

* Values in $1000
† Not reported

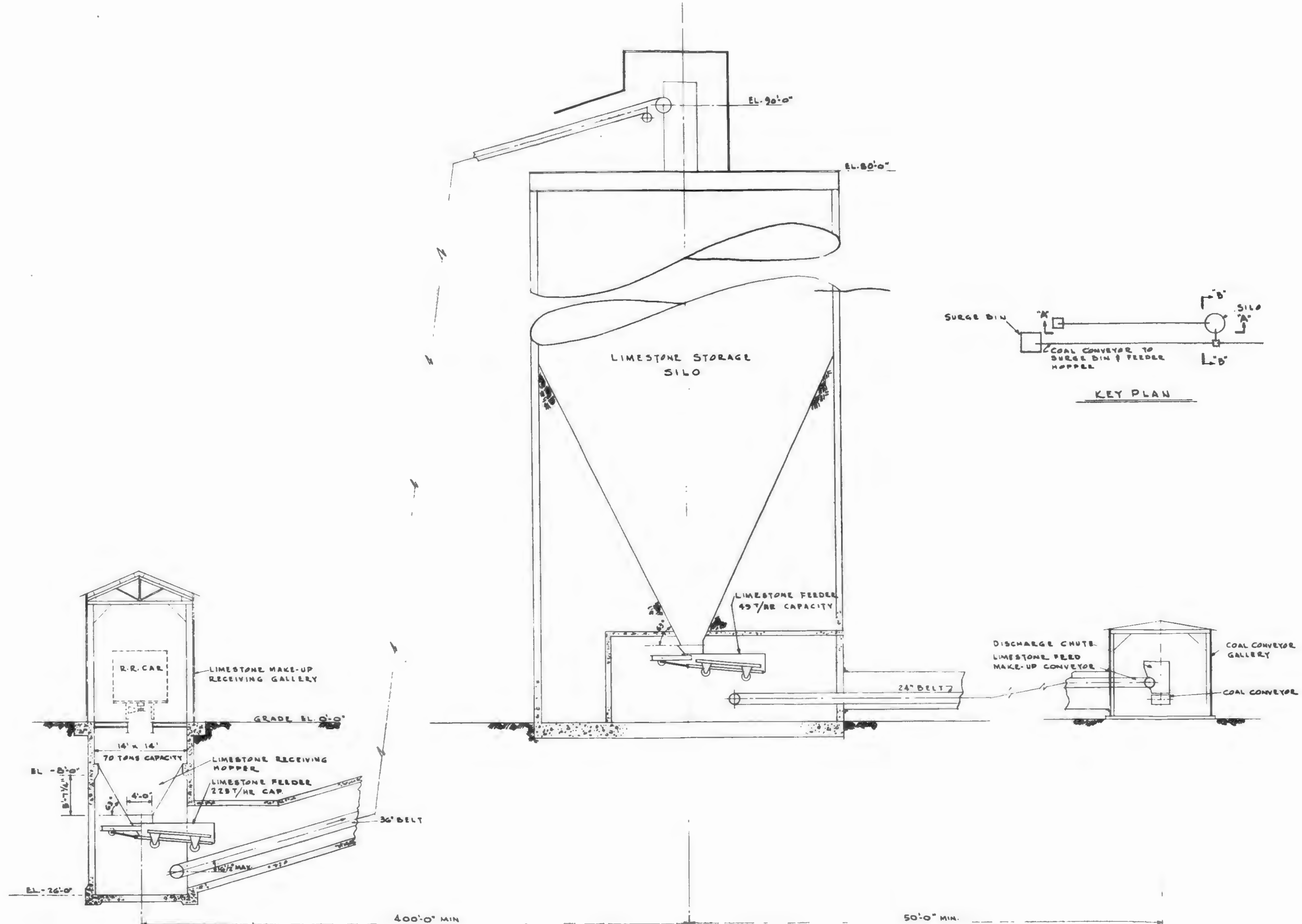
Fuel burning equipment generally includes bunkers, feeders, exhauster mills, burner, conduits and controls. They have been included here to be certain appropriate comparisons are made. The drying operations would normally take place in the mill under coal burning equipment, this operation has been included as part of the fuel burning operation since drying and distribution are handled together.

A4.1.5 Limestone Make-Up Storage and Feed The limestone feed make-up system which consists of a receiving hopper, enclosed storage silo and discharge gallary is illustrated in Figure A4-4. Orientation with respect to the coal handling system is shown in Figure A4-1.

The limestone plant is the same as that reported for the pressurized boiler and need not be repeated here.

NOTES

1. DO NOT SCALE THIS DRAWING. USE FIGURE DIMENSIONS ONLY.
2. ABBREVIATIONS USED ON THIS DRAWING ARE IN ACCORDANCE WITH AMERICAN STANDARD "ABBREVIATIONS FOR USE ON DRAWINGS."



SECTION "A-A"

SECTION "B-B"

LETTER	DATE	DESCRIPTION
REVISIONS		

ELEVATION OF LIMESTONE FEED PLANT FOR THE 300MW BOILERS

This Drawing is the Property of the
FOSTER WHEELER CORPORATION
110 SO. ORANGE AVE. LIVINGSTON, N.J.

FIG A5-15

0-59-1192

DRAWN BY:	JR	6/7/71	SCALE: 1/8" =1'-0"	
CHECKED BY:			RD 711-123A	
APPROVED BY:				

A4.2 Coal Feeding

The fuel injection system for the atmospheric boiler is similar to the pressurized boiler system in many ways. Although the furnace pressure is much reduced, the problem of pumping to an elevated pressure still exists. Instead of dealing with 10 atm. we must now deal with about 45 in. w.g. The large pressure drop across the bed pressurizes the feed system at the tuyeres or burners. Fortunately, lock hoppers are not required. Feeders in "conventional" coal plants are capable of sealing against pressures of about 60 in. w.g.

The coal feeding in the atmospheric boiler is complicated by distribution requirements of coal in the bed. According to experimental efforts by the British (13). one coal injector is needed for each 10 square feet of bed surface. This means about 16 injectors are required for each bed.

In developing a concept to handle the problems of pressure and distribution, consideration was given to flow dividers, mani-

(13) D. H. Archer, et al, "Evaluation of Fluidized Bed Combustion Process", Sixth Monthly Progress Report to NAPCA Contract No. CPA 70-9, 1970.

folds and other means of splitting the coal flow internally and externally to the boiler. There did not appear to be any method of splitting the flow, once the coal was introduced to the transport lines, that did not contain areas of high risk and uncertainty. Distribution to a multi-point injection system from a fluid bed seemed to be the only reasonably accurate approach. This concept is proposed.

A4.2.1 Scope The injection system chosen consists of a coal surge bunker volumetric feeder, sealing air fan, fluidized bed coal dryer and feed injection lines. The complete system is illustrated in Figure 4.9 of the text. The surge bunker is compartmentized and sized in the same manner as proposed by Petrocarb. This will discharge into a conventional Stock Volumetric feeder illustrated in Figure A4-5. The feeder will, more or less, act as a lock hopper and serve two functions, regulation of the flow of coal and provide an air seal for the bunker. It is fed sealing air at 300 scfm and 45 in. w.g. by a sealing air fan.

Coal flows from the feeder by gravity into the fluid bed feed injector vessel. In this vessel hot air from the secondary air ducts pressurized by high temperature primary air fans fluidize the coal and dries it. The cooled air with elutriated material passes out of the fluid bed dryer to the Ducon cyclones, illustrated in Figure A4-6, where it is cleaned prior to being returned to the secondary air duct. Residue from the cyclones is returned to the bed.

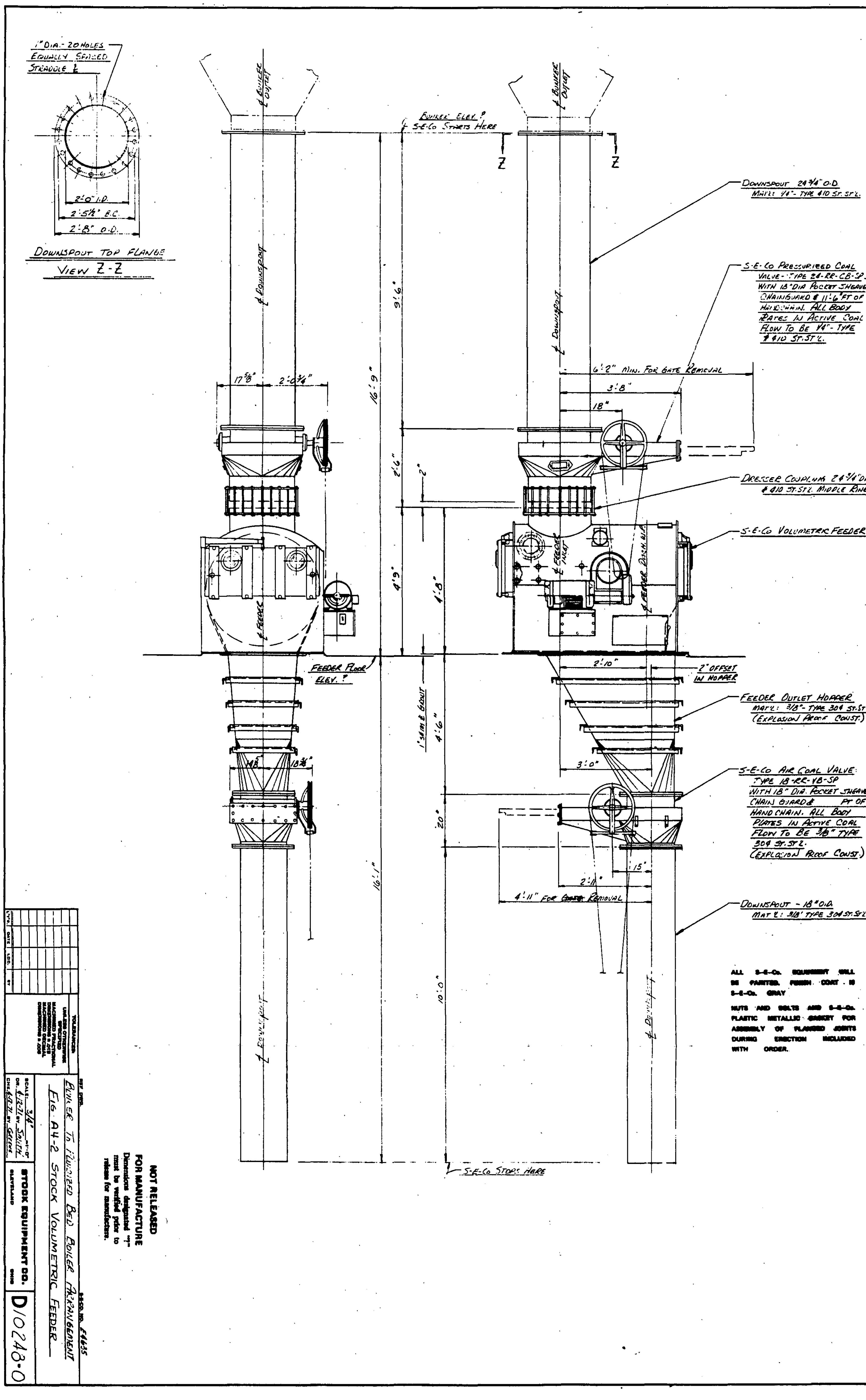

1" DIA. - 20 HOLES EQUALLY SPACED STRADDLE ℄
2'-0" I.D.
2'-5¼" B.C.
2'-8" O.D.
DOWNSPOUT TOP FLANGE
VIEW Z-Z
℄ BOILER OUTLET
BOILER ELEV.
S-E-Co STARTS HERE
DOWNSPOUT 24¾" O.D. MAT'L: ¼" - TYPE 410 ST. ST'L.
S-E-Co PRESSURIZED COAL VALVE - TYPE 24-RR-CB-SP. WITH 18" DIA POCKET SHEAVE CHAINGUARD & 11'-6" FT OF HANDCHAIN. ALL BODY PLATES IN ACTIVE COAL FLOW TO BE ¼" - TYPE #410 ST. ST'L.
6'-2" MIN. FOR GATE REMOVAL
3'-8"
18"
℄ DOWNSPOUT
9'-6"
16'-9"
2'-6"
2"
4'-9"
4'-8"
DRESSER COUPLING 24¾" DIA. #410 ST. ST'L. MIDDLE RING
S-E-Co VOLUMETRIC FEEDER
℄ FEEDER INLET
℄ FEEDER DISCH.
℄ FEEDER
FEEDER FLOOR ELEV.
2'-10"
2" OFFSET IN HOPPER
FEEDER OUTLET HOPPER MAT'L: 3/8" - TYPE 304 ST. ST'L. (EXPLOSION PROOF CONST.)
1'-5" MIN & GOUT
4'-6"
3'-0"
S-E-Co AIR COAL VALVE TYPE 18-RR-YB-SP WITH 18" DIA. POCKET SHEAVE CHAIN GUARD & FT OF HAND CHAIN. ALL BODY PLATES IN ACTIVE COAL FLOW TO BE 3/8" TYPE 304 ST. ST'L. (EXPLOSION PROOF CONST.)
20"
15"
2'-11"
4'-11" FOR GATE REMOVAL
DOWNSPOUT - 18" O.D. MAT'L: 3/8" TYPE 304 ST. ST'L.
16'-1"
10'-0"
ALL S-E-Co. EQUIPMENT WILL BE PAINTED. FINISH COAT IS S-E-Co. GRAY
NUTS AND BOLTS AND S-E-Co. PLASTIC METALLIC GASKET FOR ASSEMBLY OF FLANGED JOINTS DURING ERECTION INCLUDED WITH ORDER.
S-E-Co STOPS HERE
NOT RELEASED FOR MANUFACTURE
Dimensions designated "†" must be verified prior to release for manufacture.
BOILER TO FLUIDIZED BED BOILER ARRANGEMENT
FIG A4-2 STOCK VOLUMETRIC FEEDER
STOCK EQUIPMENT CO. CLEVELAND OHIO
D10248-0

Fig. A 4-5

FIGURE A4-6 DUCON CYCLONE - ATM. BOILER FUEL INJECTION SYSTEM

The fluidized material is removed from the vessel illustrated in Figure A4-7 through 40 - 1" distribution lines penetrating the bottom of the vessel and passing up through the grid plate. Each one of these lines connects to a 2" coal feed injection pipe which carries coal to various feed points in the stream generator fluid beds. The coal in the 1" withdrawal lines is fluidized and thus acts as a pressure seal between the fuel injector and the steam generator.

Coal is fed through the 2" line as a dilute phase at an air-to-coal ratio of about 0.35 by weight. Dilute phase was chosen to minimize the pressure requirements to pump the mixture to elevations of about 100 feet. The pipes were sized to avoid saltation of the coal and minimize pressure drop. Calculations were made of the saltation velocity for various pipe sizes and temperature levels according to Zenz (9) as recommended by W. C. Yang (11). These are illustrated in Figures A4-8, A4-9, A4-10 and A4-11. The correlations proposed by Huff and Holden were used to calculate the pressure drop (10).

(9) Frederick A. Zenz, "Fluidization and Fluid-Particle Systems", Reinhold Chemical Engineering Series, Chapman and Hall Ltd., London, 1960.

(10) William R. Huff and John M. Holden, "Pressure Drop in the Pneumatic Transport of Coal", Proceedings, Institute of Gas Technology, Bureau of Mines Symposium, Morgantown, West Virginia, October 19-20, 1965, 1C 8314.

(11) Communication, W. C. Yang, Westinghouse

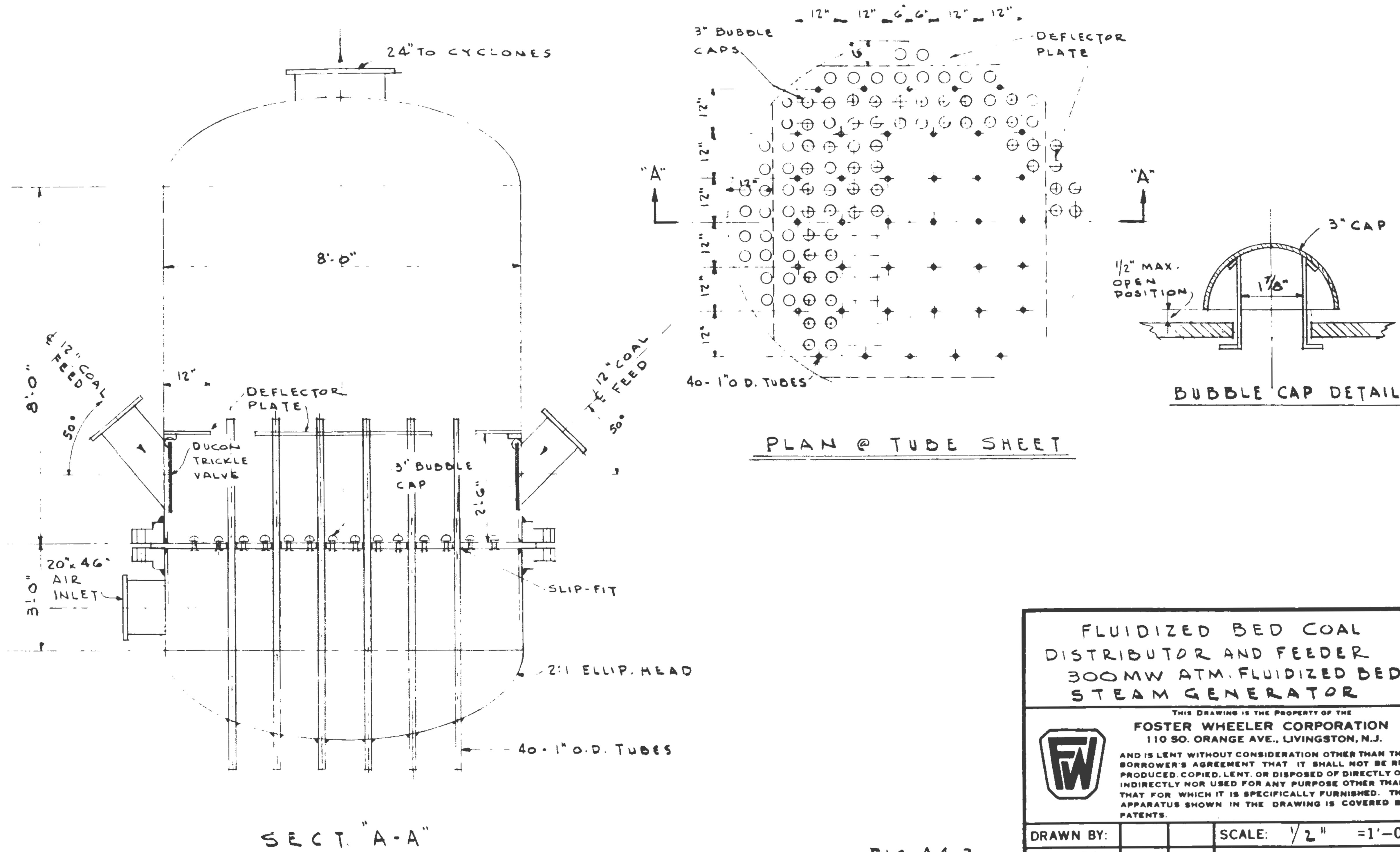


FIG. A4-3

FLUIDIZED BED COAL DISTRIBUTOR AND FEEDER
300 MW ATM. FLUIDIZED BED STEAM GENERATOR

DRAWN BY:		SCALE: 1/2" =1'-0"
CHECKED BY:		RD 713-127A
APPROVED BY:		

REVISION

ORDER NO. O-59-1197

FIGURE A4-8 PNEUMATIC CONVEYABILITY OF ANGULAR COAL PARTICLES IN DILUTE PHASE AT 1 ATM AND TEMPERATURES OF 100-500°F

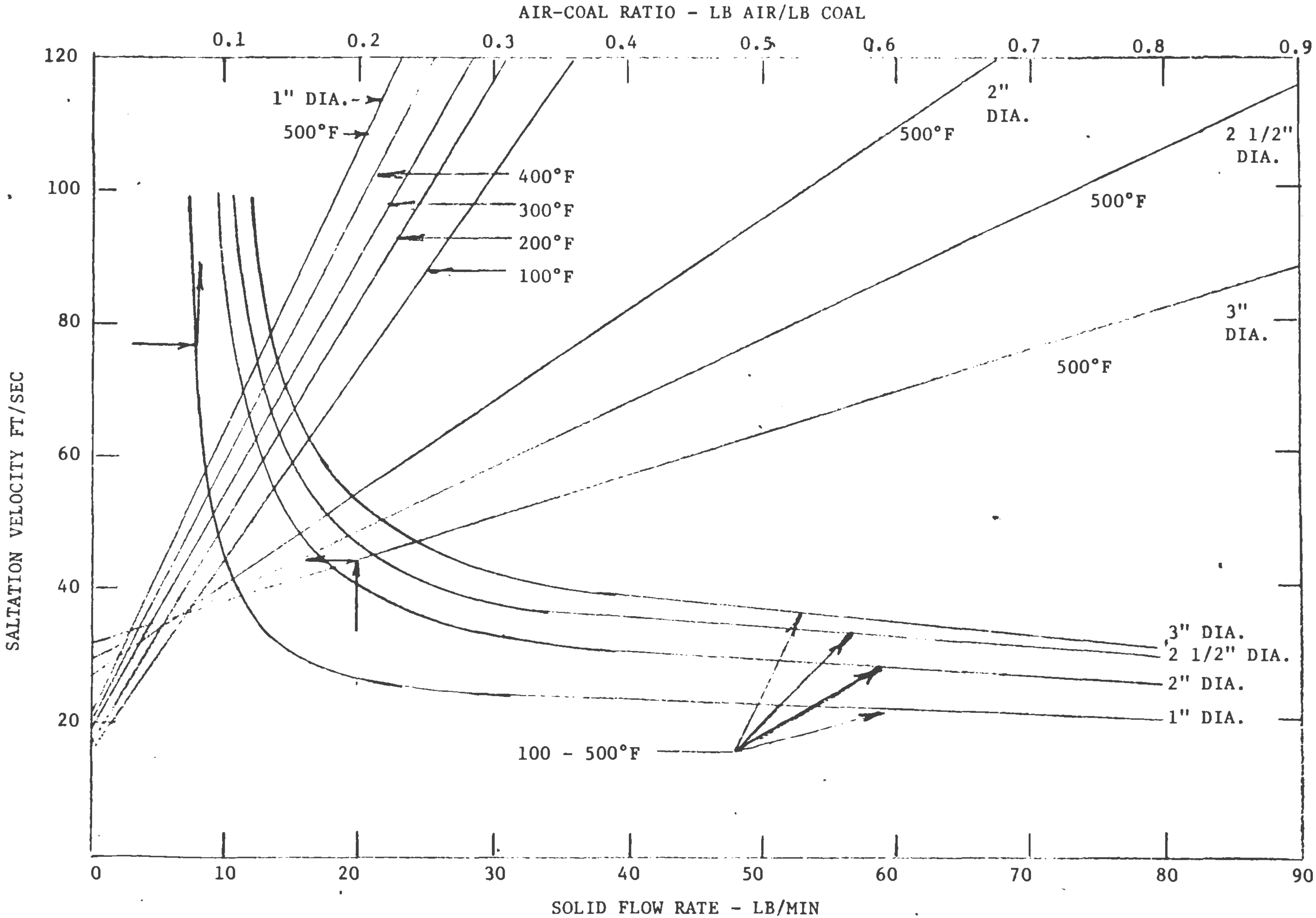


FIGURE A4-9 PNEUMATIC CONVEYABILITY OF ANGULAR COAL PARTICLES IN DILUTE PHASE AT 1 ATM AND TEMPERATURES OF 100-500°F

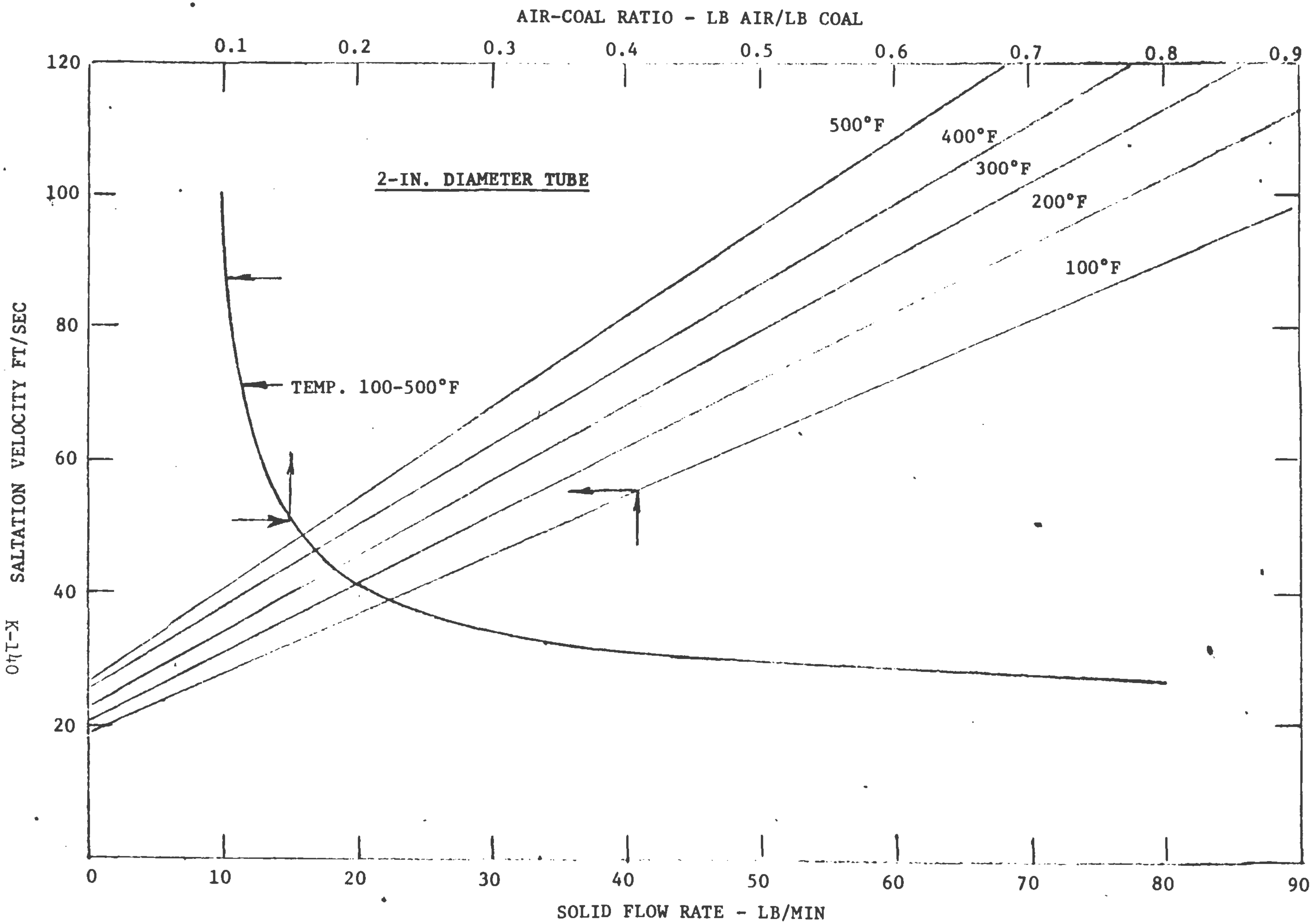


FIGURE A4-10 PNEUMATIC CONVEYABILITY OF ANGULAR COAL PARTICLES IN DILUTE PHASE AT 1 ATM AND TEMPERATURES OF 100-500°F

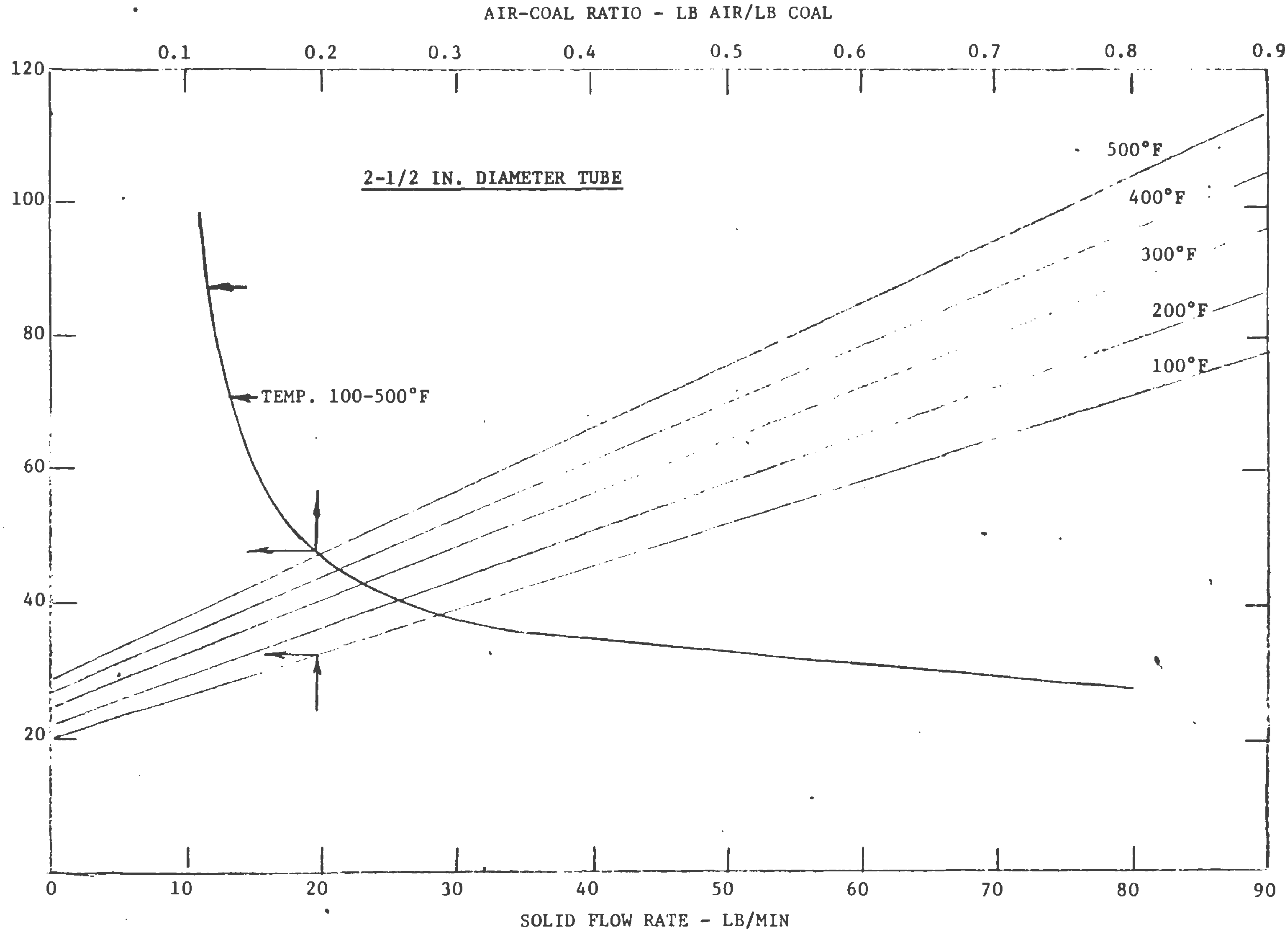
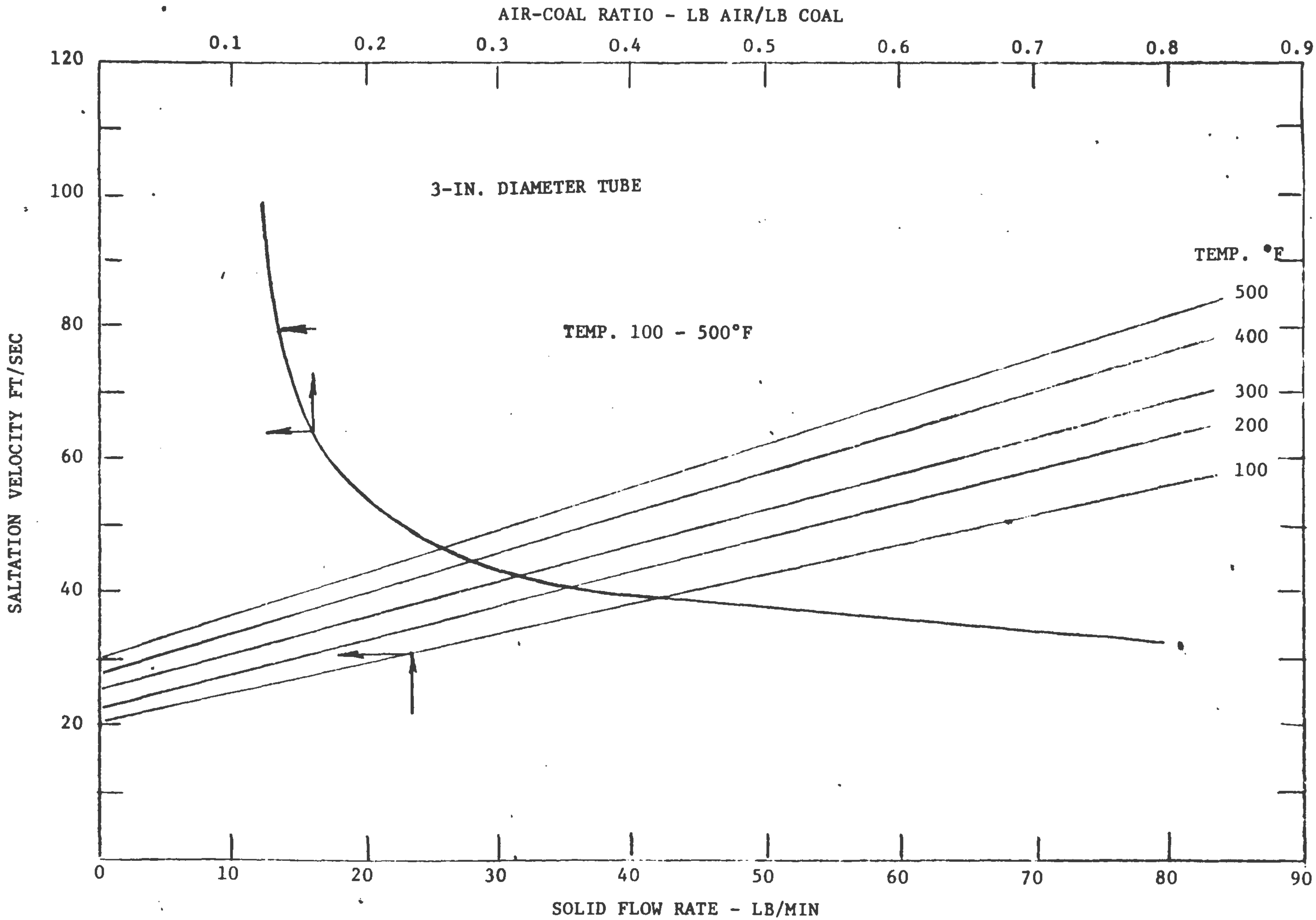


FIGURE A4-11 PNEUMATIC CONVEYABILITY OF ANGULAR COAL PARTICLES IN DILUTE PHASE AT 1 ATM AND TEMPERATURES OF 100-500°F



K-142

A4.2.2 Equipment The equipment in the coal feeding process includes:

1) Surge Bin
2) First Downspout
3) Volumetric Feeder
4) Second Downspout
5) Fuel Injector
6) Ducon Cyclones
7) Feed Injection Lines
8) Sealing Air Fan
9) Primary Air Fan

Volumetric feeder and connections include downspouts from the surge bin hopper and to the fluid injector, volumetric feeder, feeder hopper, mechanical and electrical positioners and electric tachometer generator.

Downspouts from the surge bin are 24-3/4" O.D., 9'6" long, and are constructed of Type 410 stainless steel. Inlet and outlet are fitted with 3/4" thick mild steel flanges. Stainless steel is usually selected for these fittings to resist corrosion due to the presence of sulfur in the coal and to prevent coal from adhering to the inside of the pipe.

Stock Equipment Company Type 24-RR-CB-SP coal gate valves are installed at inlet and outlet openings and are 24" in diameter. The inlet is fitted with a 3/4" thick mild steel flange and the outlet is a collar sized for a Dresser Coupling connection. The body is of pressurized construction. Body plates in active coal flow are 1/4" thick Type stainless steel. The valve is equipped with a pocket sheave hand chain and chain guard.

Style 38 Dresser Couplings connect the 24-3/4" O.D. valve outlet with the feeder inlet. The coupling middle ring is 3/8" thick, Type 410 stainless steel.

The Stock Equipment Company volumetric feeder has a capacity at maximum motor speed of 22,700 lbs/hr. Two feeders were selected for each fluid bed injector with sufficient over capacity to provide continuous flow at partial load with one feeder out of operation.

Distance between the centerline of the feeder inlet to centerline of the feeder hopper discharge is 3'-0". The inlet is 24 3/4" O.D. formed for used with a Dresser Coupling. All portions of the feeder in contact with coal flow are either rubber or Type 410 stainless steel. The feeder body is explosion proof construction, rated at 50 psi.

Accessories furnished with each feeder are as follows:

Paddle alarm at the inlet over belt.

Paddle alarm at the discharge.

Stock Equipment Company solid state speed controller in separate enclosure.

Variable speed belt drive motor with reducer.

Tachometer indicator reading in pounds of coal per hour.

Purge air inlet connection.

The feeder outlet hopper is 4'-6" high, constructed of 3/8" thick Type 304 stainless steel and reinforced for 50 psi explosion proof rating. The inlet is sized to fit the feeder and the outlet was originally sized to 18" diameter as appearing in Figure A4-5. This has since been changed to 12".

The feeder discharge valves are Type 18-RR-VB-SP 18" (12") diameter fitted with 3/4" thick mild steel flanges. Body is explosion proof construction with a 50 psi rating. Body plates in active coal flow are 3/8" thick Type 304 stainless steel. The valve is equipped with a pocket sheave, hand chain and chain guard.

The downspouts are 18" O.D. (12") by 10'-0" long, constructed of 3/8" thick Type 304 stainless steel. The inlet is fitted with a 3/4" thick mild steel flange and the outlet is plain.

Mechanical and electrical position indicators are optional. The position indicator can be used on either or both valves. The indicator consists of a large pointer and the words "open - closed" which gives visual indication of the valve gate position. It also includes two micro-switches mounted in the housing to control indicating lights at remote locations.

As an added option Stock Equipment Company includes a General Electric D.D. Auxiliary Tachometer Generator to be mounted on the feeder drive motor for feedback to the combustion controls system or integrated into the coal feed flow controls.

Sealing air is provided at the volumetric feeder by a Westinghouse size 2724-7 1/2 Turbo Blower sealing air fan with a capacity of 300 cfm and static pressure discharges of 43" H_2O at 80°F. The blower comes equipped with a TFC 7 1/2 HP - 3 phase 60 cycle motor. An alternate fan can be provided to give a slightly higher pressure level of 51.8" H_2O at about twice the price. In the alternate case a 20 HP motor would be required .

The fluid bed dryer and distributor is illustrated in Figure A4-7. It simply consists of a cylindrical vessel enclosed at either end with ASME dished heads. The vessel is constructed of 3/8" rolled and formed 304 stainless steel. The lower head is flanged to accommodate a grid plate and provide access to the internals of the vessel. Coal is fed to the vessel at bed level through two downspouts which are sealed with Ducon trickle valves. Recycle coal from the cylones also arrives at the bed level through a single 8" connection or dipleg. Carbon steel 1" coal withdrawal lines penetrate the lower dished head where they are seal welded to the grid plate at a simple air seal. A rectangular air inlet is located in the skirt of the dished head which allows hot air from the primary air fan to enter the vessel and then pass up through 3" bubble caps in the grid plate. The cooled air leaves the vessel at the top through a 24" O.D. line where it is transported to the Ducon cyclone.

The bed was designed at 2-1/2 ft. depth to dry the coal at about 250° exit temperature. Air inlet temperature is 735°F. The superficial bed velocity is 4 ft/sec. It was selected to handle 1/4" x 0" coal. The air leaving the bed has an estimated saturated loading of coal dust of 173 grains per standard cubic ft. with a size distribution as illustrated in Figure A4-12. Sizing was accomplished according to procedures recommended by Zenz and Othmer (9). Deflector plates have been installed to insure mixing and avoid by passing of air around the tube bundle.

(9) Fredrick A. Zenz and Donald F. Othmer, "Fluidization and Fluid-Patricle Systems", Reinhold Chemical Engineering Series, Reinhold Publishing Corporation, New York, 1960.

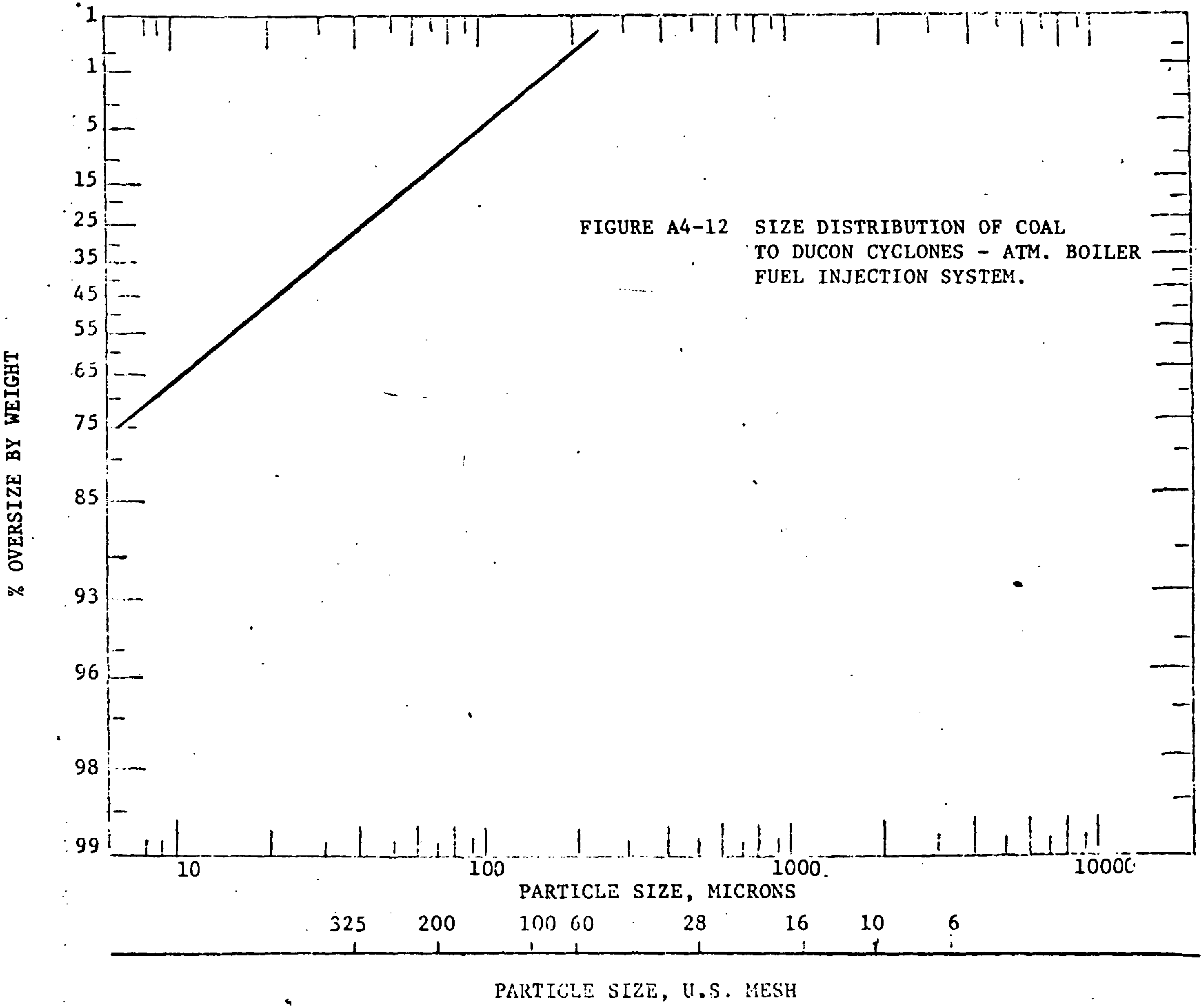


FIGURE A4-12 SIZE DISTRIBUTION OF COAL TO DUCON CYCLONES - ATM. BOILER FUEL INJECTION SYSTEM.

Air and elutriated material leave the vessel at 250°F at a rate of 26,000 lb/hr and enter a Ducon Model 660 VM 700-150 Stairmand type cyclone constructed of 1/4" mild steel. The anticipated pressure drop is 5.3" H_2O. It has an efficiency of 90%. The cyclone is illustrated in Figures 4.9 and A4-6.

Feed lines withdrawing coal are 1" I.D. and transport lines are 2" I.D. They are constructed of mild steel schedule 80 pipe. The lines should be installed such that minimum tube bend radii are at least 12 diameters to minimize erosion. The transport medium is dilute phase at 0.35 pounds of air per pound of coal and gas velocities of 30 ft/sec.

Two fuel injector vessels are used to serve the five beds. Each injector contains about 40 feed points. This is a conceptual feed injector and experimentation would have to be carried out to be certain of the optimum number of feed points.

One Westinghouse primary air fan serves each fuel injector. The capacities are too low for the pressure drops required to provide individual primary air fan for each injector vessel. The fan has a capacity of 34,000 lb/hr at a pressure of 45" H_2O.

A4.2.3 Cost The total cost of the system is broken down as illustrated in Table A4-2.

TABLE A4-2

COST BREAKDOWN OF COAL FEEDING SYSTEM

1)	Bunkers	$690,000
2)	Volumetric feeders	269,214
3)	Cyclones	84,000
4)	Fuel Injectors	220,786
5)	Transport lines	16,000
6)	Controls	500,000
	TOTAL	$1,780,000

The estimates do not include a price for controls. Development of a control system is beyond the scope of effort of the contract. The system would probably be similar but less expensive than the Petrocarb system since weighing is included in the price of the volumetric feeders and no lock hopper controls are required. For estimating purposes a figure of $500,000 would probably be conservative.

A4.2.4 Carbon burn-up cell feeder The carbon burn-up cell feeder has been included here as an adjunct to the section on coal feeding. It was not mentioned in the scope of work in this section and certainly it does not warrant a section by itself.

The carbon elutriated from the fluidized beds contain about 13% of the original carbon fired. Details of the exact material balance are reported in the body of the report. The elutriated material must be recovered at the dust collector and fed to the carbon burn-up cell as a fuel. Collection takes place at 840°F. To avoid fires the residence time in the dust collector must be minimized.

A modified Petrocarb system was selected to handle the fuel to the carbon burn-up cell (CBC) at 840°F. The elutriated material passes on through the dust collector hopper to a collecting vessel. A level is maintained in this vessel to regulate control of the fuel from the Petrocarb fuel injector. The complete system is illustrated in Figure 4.7 of the body of the report. The collected material periodically flows from the collector vessel into the localized fluid bed fuel injector from which it is pumped through 16 separate lines to the CBC.

Once again 16 feed points were felt necessary for adequate distribution of fuel to the larger flat beds encountered at atmospheric pressure. Petrocarb indicated service air for transporting the fuel would be approximately 2000 scfm compressed to 200 psig.

One system as described above would be required for each module. Consideration was given to one injection module for all four boiler modules. The problems of collecting the residue, integrating operation between modules and injecting to 64 feed points were too great to consider the concept further.

Petrocarb estimates the cost of the system to be $500,000.

A4.3 Patriculate Removal

A two stage particulate removal system is used to clean the flue gases of carbon sorbent and ash elutriated from the fluidized beds and carbon burn-up cell (CBC) to 0.01 grains per standard cubic foot. The first stage is a multi-cyclone mechanical dust collector illustrated in Figure A4-13. The second stage consists of a Research Cottrell electrostatic precipitator illustrated in Figure A4-20.

The first stage of separation is divided into two streams, one stream from the fluidized bed cells and the second stream from the CBC. In this way elutriated material from the fluid beds can be segregated from the ash released in theCBC, so that it can be used as a fuel in the latter.

The flow diagram in Figure A4-14 illustrates the distribution of flue gas and particulate matter in the various tanges of separation. Size distribution of particulate material elutriated from the beds is illustrated in Figure A4-15, along with the size distribution of residue in the flue gas. The sizing of elutriated material was specified by Westinghouse based on data provided by NAPCA contractors.

A4.3.1 Mechanical Dust Collector A hot multiclone cyclone dust collector was selected for the first stage of particulate removal. The dust collector was designed for high temperature operation, 840°F, to avoid exposing the heat recovery surface to heavily dust laden flue gases. Multiclone cyclones in theory are more affective collectors of small particles than larger cyclones using the same gas velocities. In practice, this improved efficiency is reduced because of the increased short cir-

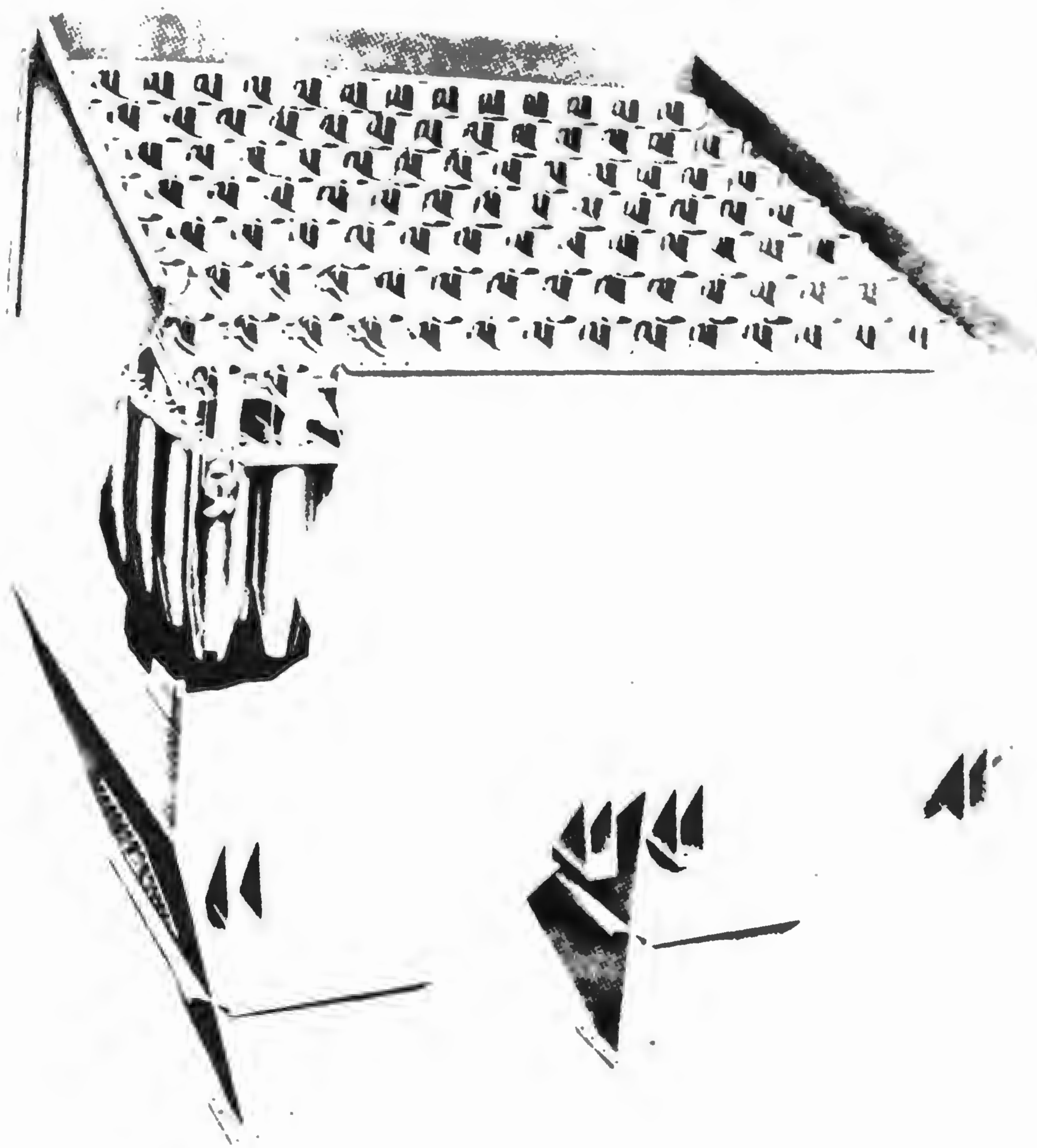


FIGURE A4-13 PARALLEL ARRANGEMENT OF MULTICYCLONES

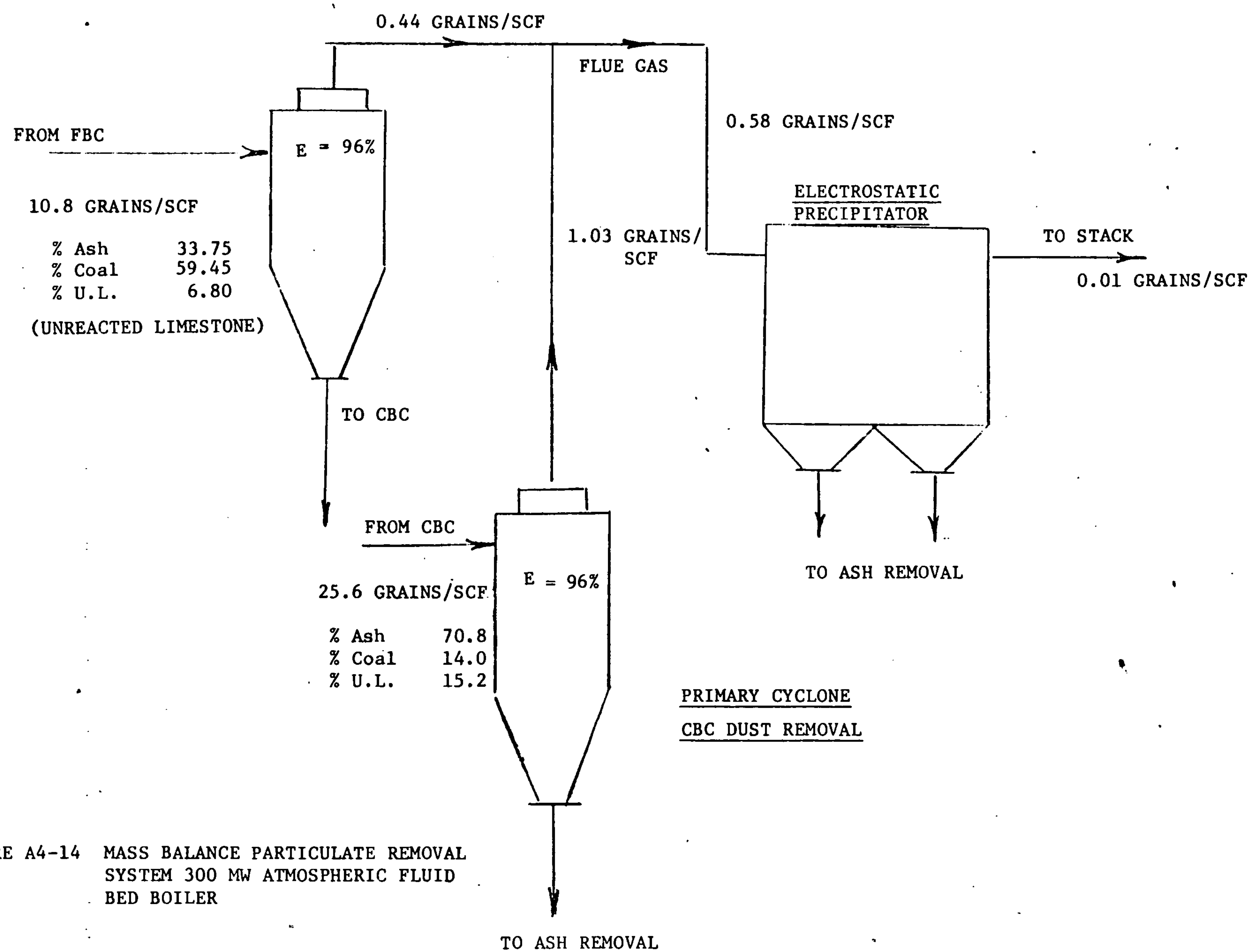


FIGURE A4-14 MASS BALANCE PARTICULATE REMOVAL SYSTEM 300 MW ATMOSPHERIC FLUID BED BOILER

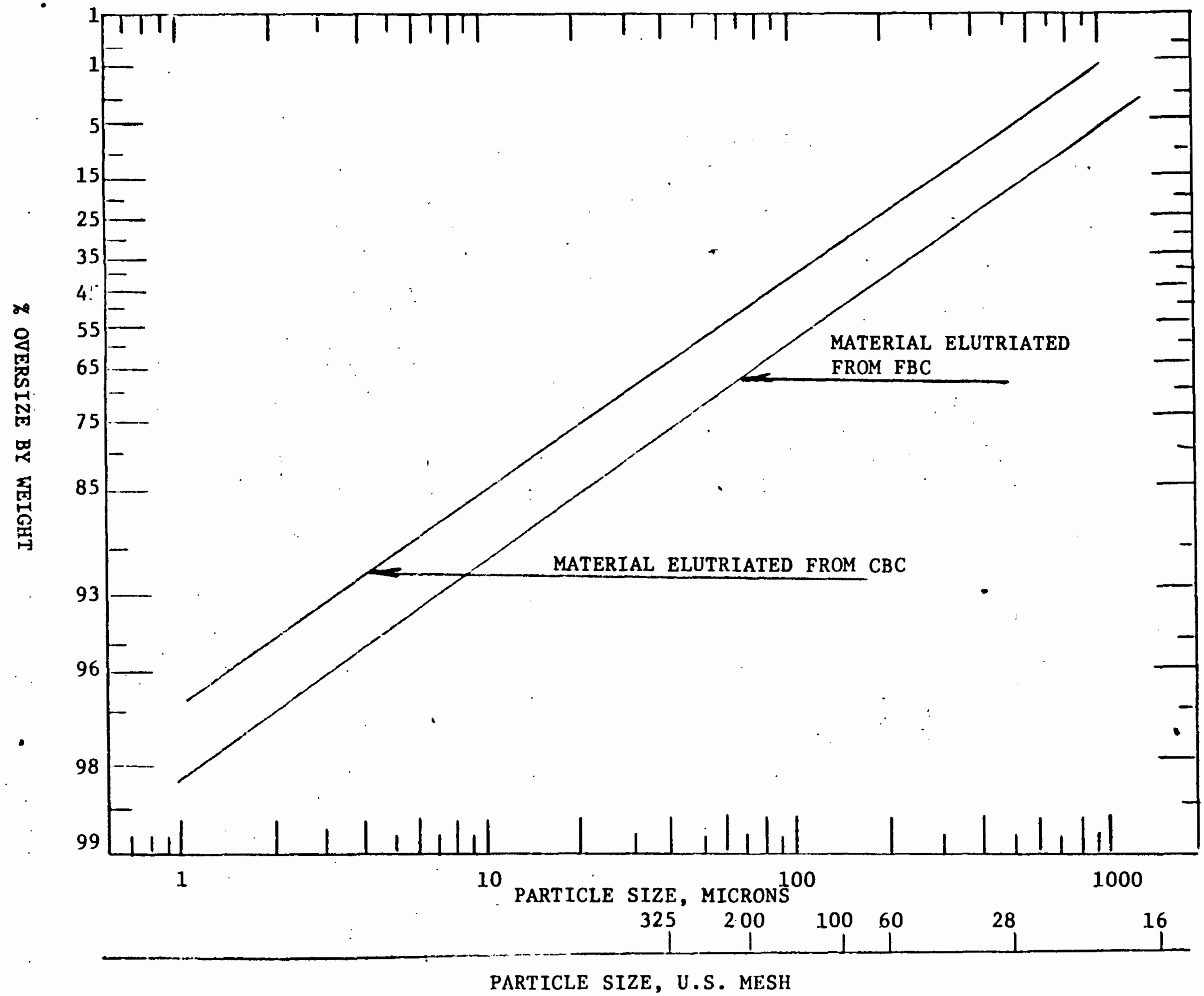


FIGURE A4-15 DISTRIBUTION OF PARTICULATE MATTER IN FLUE GAS cuiting that takes place in small cyclones operating in parallel. One worker, in comparing high efficiency cyclones with multiple small cyclone designs has shown that the multiple cyclone arrangement is only marginally better than the large cyclone (10) (11). The one important asset of the multiple cyclone arrangement is the efficient use of space at high capacity in contrast to the large volumes and high head room required by the single Stairmand type of dust collector.

The Air Correction Division of Union Oil Products designed and priced the mechanical dust collector to operate at 840°F with 96% efficiency on flue gas containing 10.8 grains per standard cubic foot with a particle size distribution as shown in Figure A4-15. The size distribution of dust leaving the mechanical collector for the precipitator is illustrated in Figure A4-16. A single dust collector is used for each module capable of handling 610,000 lb/hr of flue gas with a turn down of 70% efficiency. The pressure loss through the dust collector is 3" H_2O. The guaranteed efficiency for various Bahco dust particle size is illustrated in Figure A4-17. Dimensions of the unit are shown in Figure A4-18.

The collector is equipped with while iron tubes having an average Brinell hardness of 450 and special heat resistant tube gaskets. Each tube is 10 1/2" nominal diameter with 0.25" nominal wall. The tubes are equipped with directional guides at the inlet openings. The outlet tubes are 6 5/8" O.D. #12 guage wall standard tubing. The lower tube sheet is constructed from 1/11"

(10) C. J. Stairmand, R. M. Kelsey, Chem and Ind. 1324, (1955)

(11) W. Strauss, "Industrial Gas Cleaning", Pergamon Press, 1966.

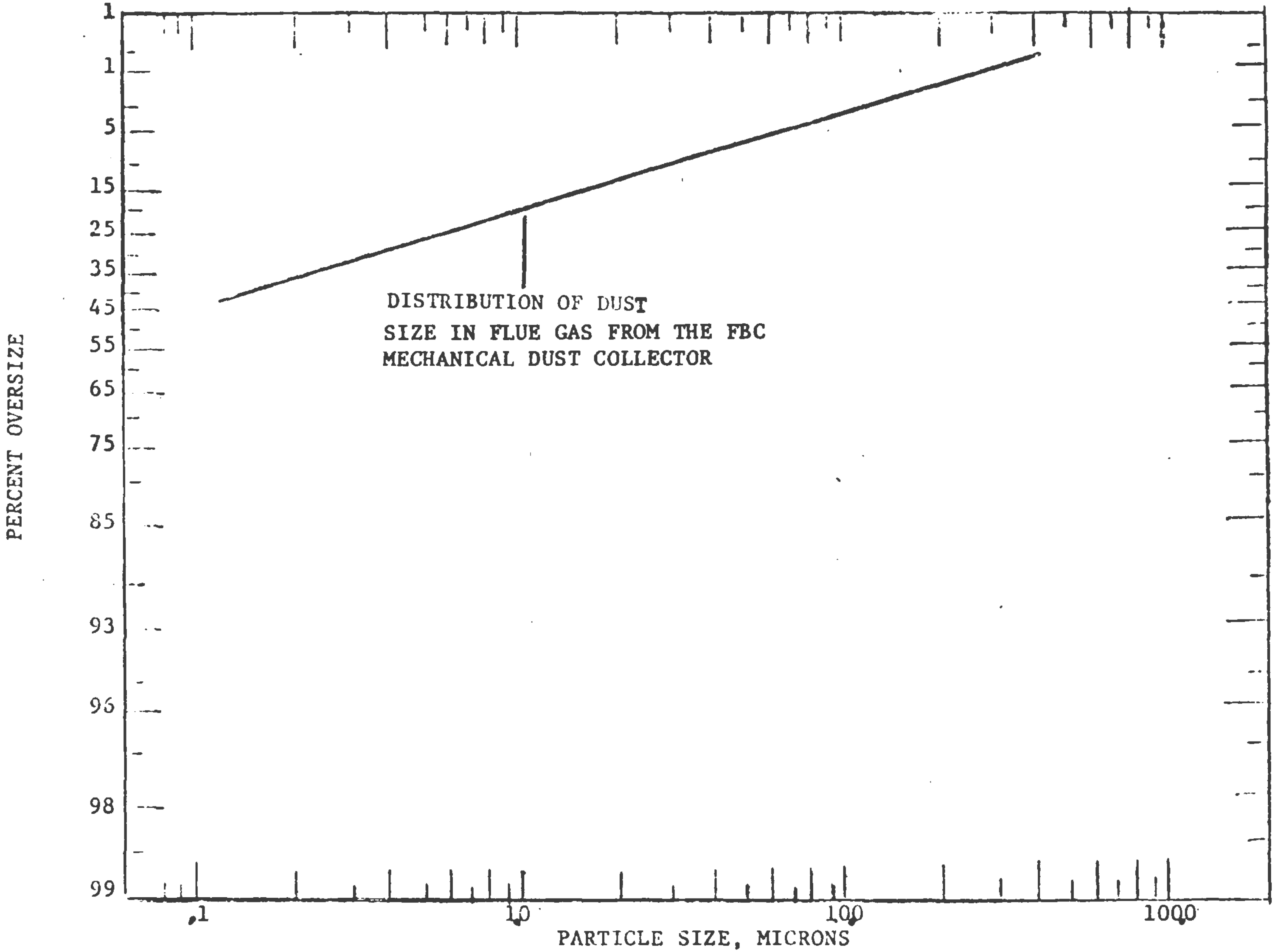
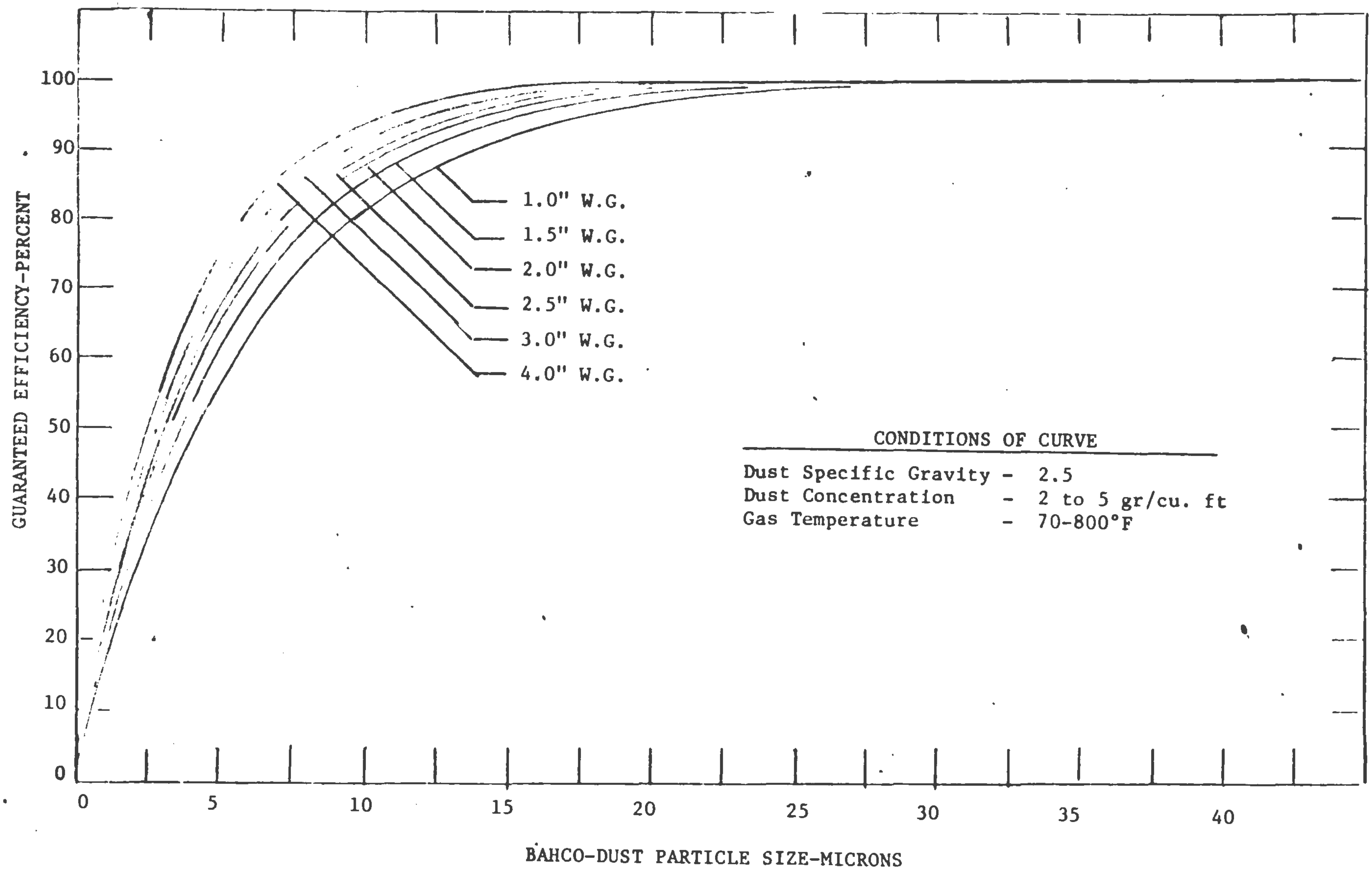


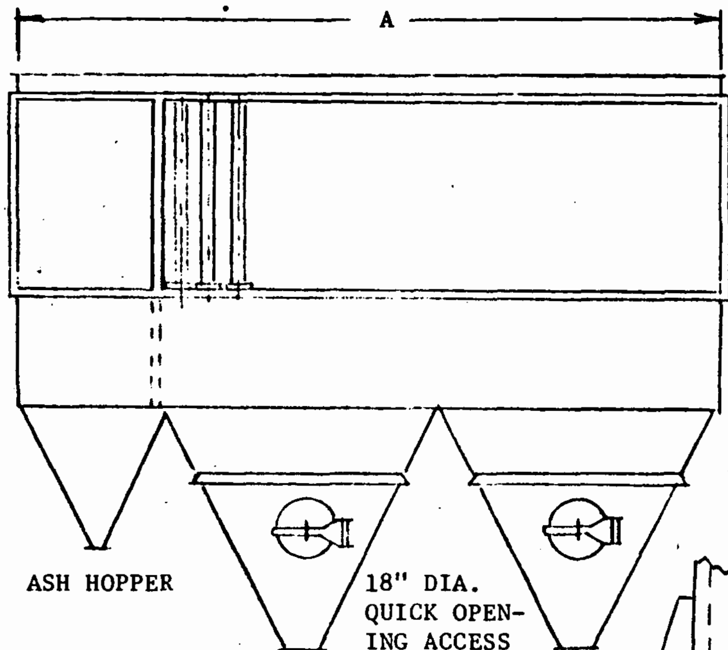
FIGURE A4-16 SIZE DISTRIBUTION OF PARTICULATE MATTER FROM THE MECHANICAL DUST COLLECTOR



K-157

FIGURE A4-17 MICRON EFFICIENCY CURVES TUBULAR DUST COLLECTOR DESIGN 106A

K-158

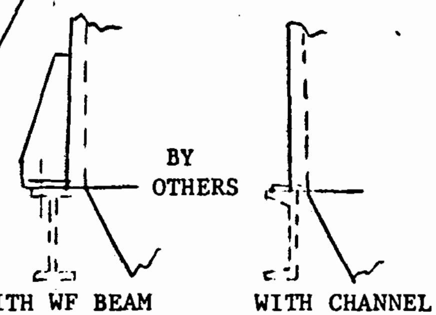


CBC HOPPERS

NOTE:

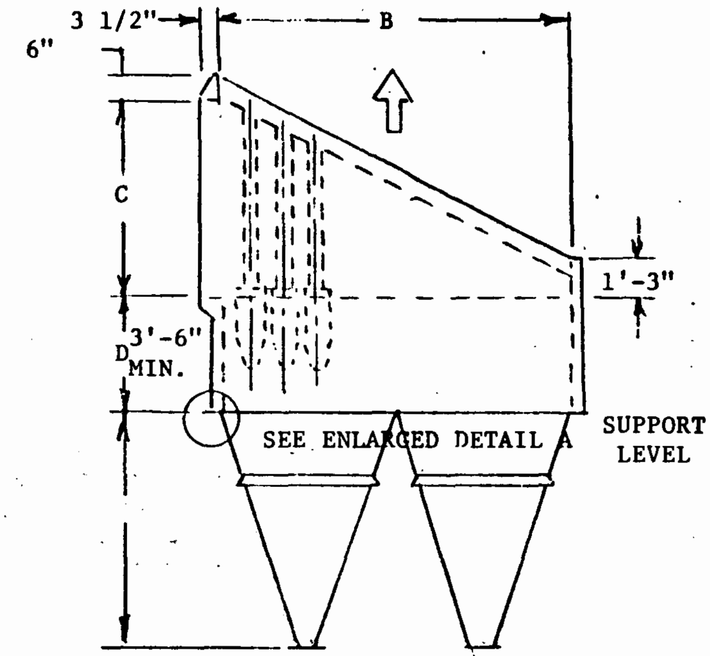
STD. HOPPER FLANGES
MATCH 8" 150 LB.
ASA DRILLING

END ELEVATION



DETAIL A

METHODS OF SUPPORT



SIDE ELEVATION

DESIGNATION DESIGN	DIMENSIONS (FEET AND INCHES)					WEIGHT @ 55#/cu.ft.		DUST CAP.
	A	B	C	D	E	WITHOUT DUST LOAD	WITH DUST LOAD	CU. FT.
106AWHS #17-357	22-0	17-8	10-3	3-6	8-1	66,000	127,050	1100

FIGURE A4-18 MODIFIED U.O.P. SERIES 100 TUBULAR DUST COLLECTOR
DIMENSION SKETCH
SIDE INLET - TOP OUTLET

steel plate. The upper tube sheet is 3/16" steel plate. The envelopes are made of 12 guage steel plate. Casing and hoppers are constructed from 3/16" steel plate suitably reinforced to withstand design conditions. Hoppers are designed for a minimum of 55° hopper valley angle. Normally the dust collector comes with 4 hoppers, two rows of two each. To accommodate the split gas stream and to segregate the material elutriated from the fluid beds from that elutriated from the CBC, six hoppers will be required as shown in Figure 3.7.

Each unit weighs 50,300 lb, costs $27,000, and requires 315 man hours to erect. At $10/hr mean cost rate for field erection the total cost of erection amounts to $3150. Other material, equipment and structural steel are included in the boiler cost.

A4.3.2 <u>Electrostatic Precipitator</u> An electrostatic precipitator is used to clean the effluent gas from the mechanical dust collector. This is not an uncommon practice. Electorstatic precipitators are often combined with mechanical dust collectors in cases where gases have exceptionally high dust concentrations or need conditioning or both (11).

The gas to the electrostatic precipitators is a blend of streams from the barbon burn up cell and the fluid bed combustion cells. It has an estimated grain loading of about 0.58 (see A4-32)grains per standard cubic foot. Size distribution of particulate material appears in Figure A4-19. The gas flow is 2,440,000 lb/hr (see A2.6) at 275°F. A residual gas loading of 0.01 grains per standard cubic foot is required.

The particulate matter consists of 79.5% ash, 18.7% coal, and 1.8% limestone.

For this particular application Research Cottrell selected one precipitator consisting of four units serving all four boiler modules. Each unit contains 45 ducts, 9" duct spacing with 30' high collecting plates and 36' of treatment length (12' + 12' + 12'). Each precipitator is energized by size 70 KV_p, 1500 on a silicon transformer rectifiers complete with automatic voltage controls. Details are illustrated in Figures A4-20 and A4-21.

The selling price of the precipitator is $571,000. Erection requires 16,869 boiler maker man-hours and 6,720 electrician man-hours. Assuming a mean cost of $10/hr for both, the cost of erection is $235,000. Detailed cost of erection cannot be made without an exact site location. Wages and subsistence varies considerably from union hall to union hall and the subsistence is not only dependent upon the union and geographic location but the local distance from the hiring hall.

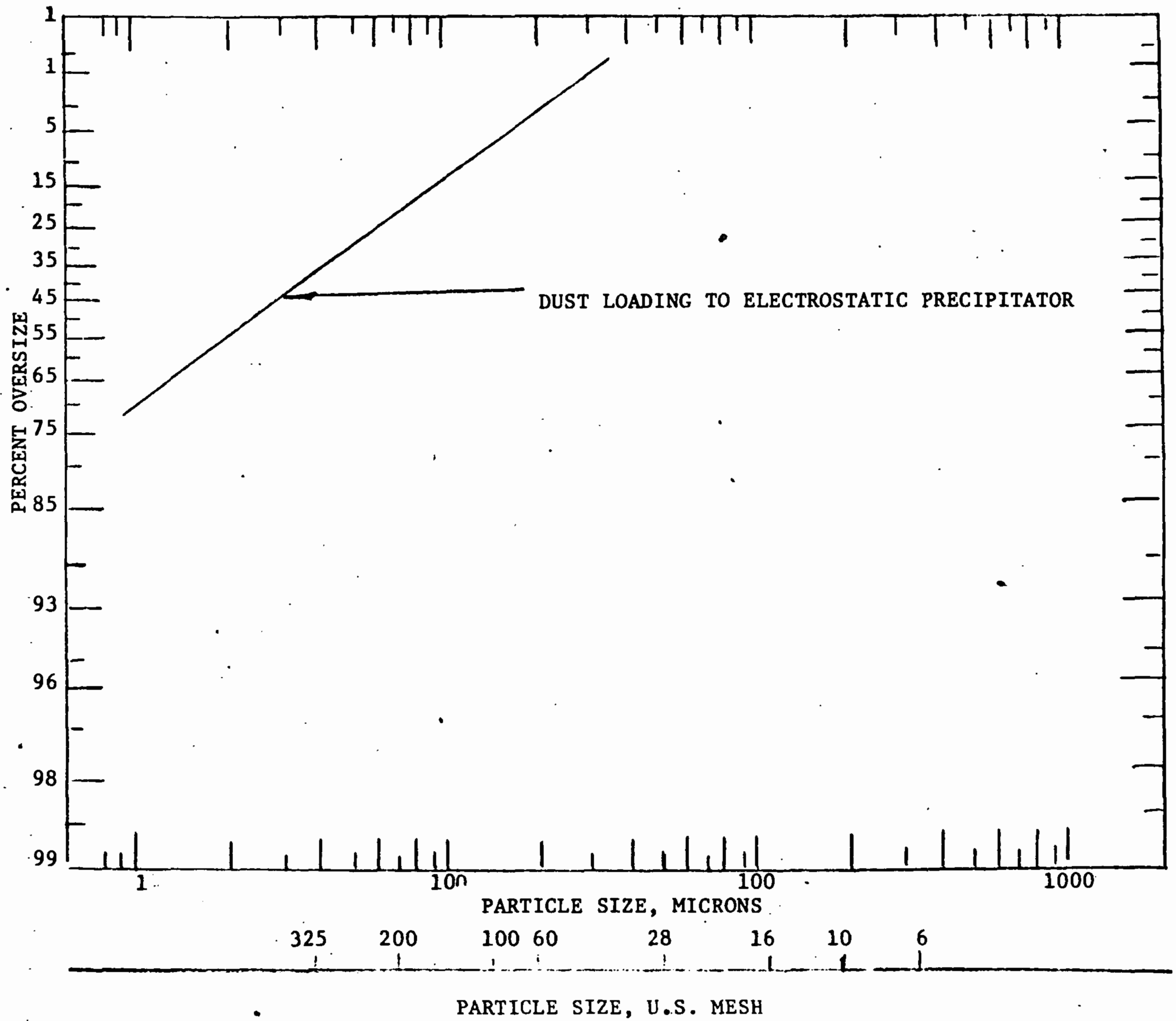


FIGURE A4-19 SIZED DISTRIBUTION OF PARTICULATE MATERIAL TO ELECTROSTATIC PRECIPITATOR

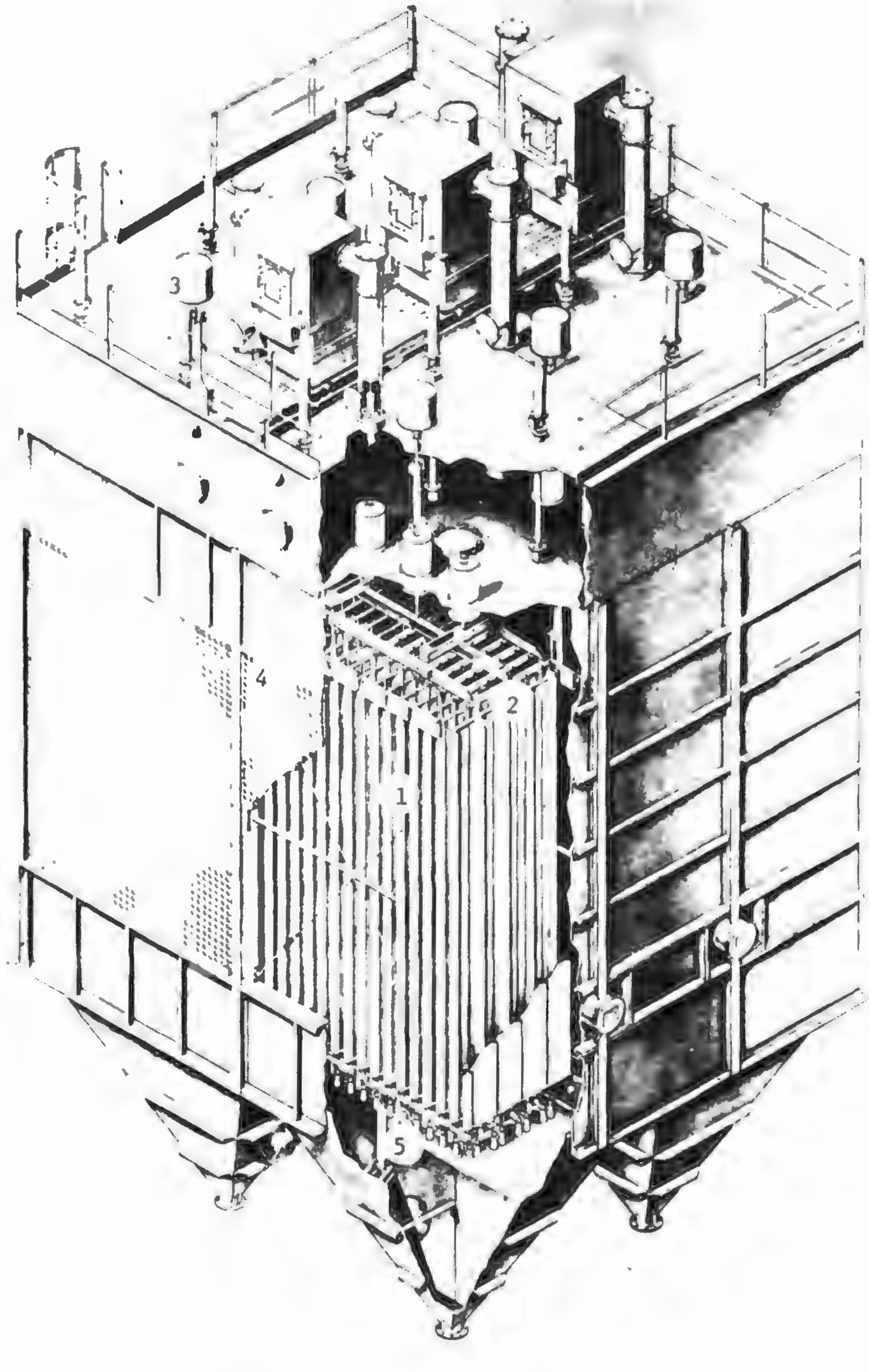
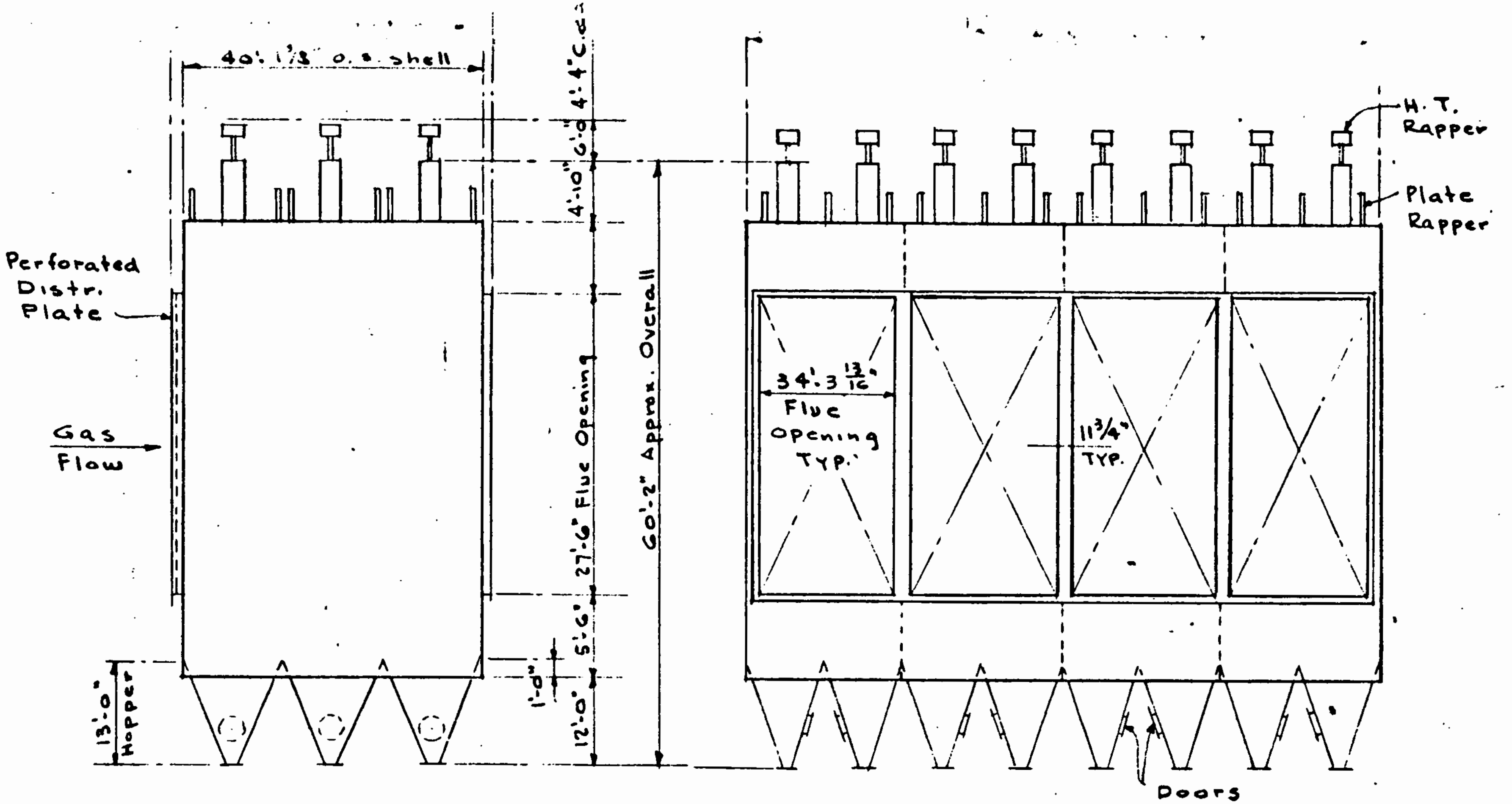


FIGURE A4-20 DESIGN FEATURES OF ELECTROSTATIC PRECIPITATOR



NOTE: Dimensions & Construction Details are subject to change.

RESEARCH-COTTRELL INC. BOUND BROOK, N.J.	
OPZEL PLATE PPTR 4U.45 D 9"x 30'·0"x 36'·0" (12'·0" PLS)	
DRN. JR DATE 5/25/71 APP. CHK.	DWG. RD714-

FIGURE M-21

A4.4 Heat Recovery Equipment

A4.4.1 Introduction Heat recovery equipment in this application will refer to the air preheater which exchanges heat from the effluent flue gas stream to the incoming combustion air. The heat exchanged is energy that is recirculated on the gas side of the boiler. The energy has no direct effect on plant performance. The quantity circulated, however, establishes the mean temperature level at which heat is exchanged between the hot gases and steam. In the case of a fluid bed combustion process where the bed temperature is fixed, the quantity of heat recirculated determines what what percent of the overall energy exchange between gas and steam takes place in the fluid bed. The heat recovery equipment increases the mean temperature difference between gas and steam and thereby reduces the total surface required to transfer energy to the steam. It does so, of course, at the expense of additional heat recovery surface. This suggests that there is a trade-off of steam generating surface and heat recovery surface which requires optimization. The end result of the study is the selection of an operating temperature level for the heat recovery equipment.

The air preheater indirectly affects the plant performance. The outlet gas temperature is limited by the dewpoint of sulfuric acid. If this limit is exceeded cold end corrosion can be expected. In a fluid bed boiler process where 90% of the sulfur is captured during combustion the permissible flue gas exit temperature can be quite low. Westinghouse recommended 275°F. Under these circumstances the plant performance should be improved by reducing the heat loss in the flue gas. This also reduces the mean temperature difference of the air preheater for a given inlet temperature and hence, increases the surface requirements. Trade-offs of surface and plant performance can be made. However, variations in either are probably small and of minor importance.

Large quantities of heat recovery surface can result in increased draft loss and hence increased fan power in a system which already has a large system resistance. Trade-offs in this case are equipment, size, quantity of surface, capital invested in fans, and power requirements.

Another facet in air heater performance is the service it provides in heating air for drying coal. Moisture requirements if sufficiently large can dictate the air heaters duty.

Generally speaking it would be undesirable to drop the temperatures of the air to the bed below 500°F for fear of quenching ignition in the bed. In addition the temperature of the transport air for coal should not go below 250° for fear of allowing condensation of moisture to take place. This could cause pluggage of the coal transport line and/or corrosion. The influence of these temperature levels on equipment selection must be determined during the optimization study.

Two types of heat exchangers are commonly considered for air preheating, 1) the Ljungstrom air preheater or regenerative air preheater and 2) the tubular heat exchanger. Other versions of these two types are available but not widely used.

A4.4.2 Description of the Tubular Air Heater The tubular air heater as the name implies, is simply a tubular heat exchanger which is normally arranged for vertical gas flow through the tubes. The unit being proposed is illustrated in Figure 4.6. Air flows horizontally across the tubes which are generally in staggered relationship. Air passes back and forth over the heating surface three times to approach counterflow. Tube sheets are provided at top and bottom through which the tubes pass and to which tubes are attached. Structural support is provided by members which are attached to the top of the upper tube sheet or located below the lower tube sheet.

The air heater is divided into three passes. The lower pass is constructed separately to allow for replacement of tubes in the event of cold end corrosion.

The tubular heat exchanger has the unique advantage of no air leakage and thus a minimum contribution to fan power requirements.

A4.4.3 <u>Description of the Regenerative Air Heater</u> The Ljungstrom air preheater operates on the regenerative principle in which rotating baskets are alternately heated and cooled by the flue gas and air streams, respectively. The air preheater assembly consists of a housing, divided into two end or outside compartments and one middle compartment in which the heating surface contained in a slowly moving rotor is installed. The outside compartments are divided by partitions which confine the hot gas to one side of the apparatus, while the air to be heated is on the other side. For each revolution of the rotor there is a complete cycle of heat exchange in which heat from the hot gas is continually absorbed by the regenerative method by the heating surface, and then given up as the rotor moves into the path of the air to be heated.

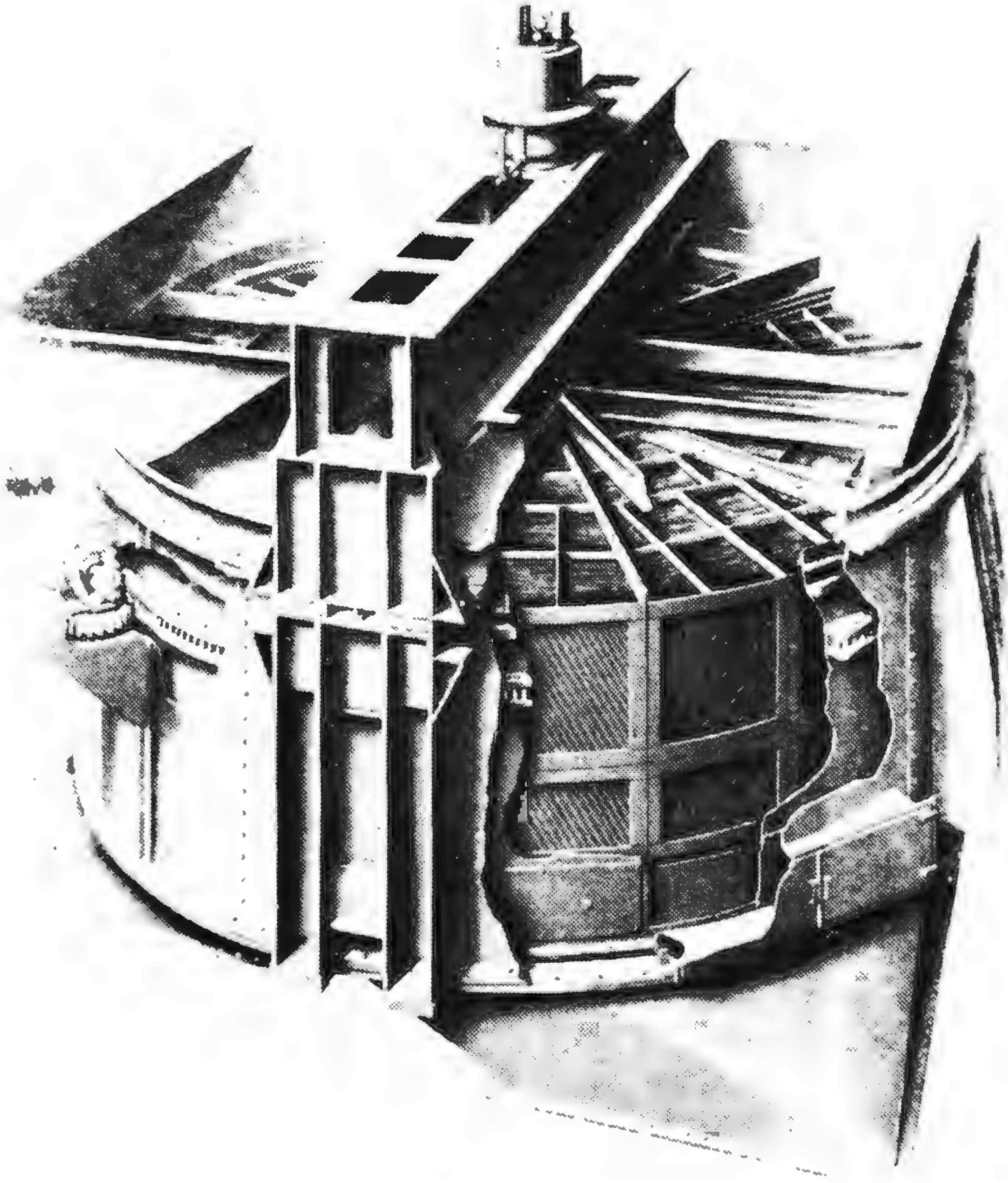


FIGURE A4-23 LJUNGSTROM REGENERATIVE AIR PREHEATER WITH VERTICAL FLOW

A typical unit of the vertical flow design is illustrated in Figure A4-22. The air for the combustion enters at the lower left side, passes upward through the heating surface and discharging at the upper left side into the hot air ducts. The hot flue gases enter at the upper right side, flows continuously downward through the heating surface into the lower right chamber, counter flow to the air, and is then exhausted to the stack. Upward flow of gas and downward flow of air are permissible if required. Horizontal flow design may also be furnished with the flow passages either side by side, or with one stream passage vertically over the other depending on the layout requirements.

Both vertical flow and horizontal flow designed units are used with gas temperatures up to but not exceeding 1000°F. For gas temperatures in excess of 1000°F special designs are available.

The rotor which carries the heating surface contains sector-shaped cells into which the heating surface is fitted. The heating surface is divided into a number of groups or baskets which can be easily inserted in the rotor. The surface is made of combinations of flat or formed thin sheets. The formed sheets may be corrugated, notched or undulated with the ribs forming longitudinal passages of the most desirable contours for the predetermined spacing. The design and arrangement of the surface provides only point contact between adjacent plates. The gas and air flow are turbulent, but at the same time the flow path through the rotor is smooth and offers low resistance. The combined pressure drop through the air side and gas side generally runs about 10" H_2O. As an approximate rule it may be stated that one inch of standard regenerative air heater surface will recover about as much heat as a two-foot length of tubular air heater surface offering equivalent resistance to gas and air flow.

The rotor motor is driven through reduction gears to a pinion shaft and then, finally, through a pinion gear to a pin rack mounted on the periphery. The rotor turns from approximately one to three rpm and the actual power required to drive it varies from 1/2 to 5 hp depending on the preheater size.

Air leakage into the gas stream is divided into three categories:

1) Leakage into the gas chamber resulting from entrainment in the rotor passages.

2) Leakage at the periphery of the rotor through the clearance space between the rotor and the housing and then into the gas passage.

3) The third source of leakage is across the radial seals into the gas passage. Although the air leakage in a conventional preheater may be a small percentage of the air provided by the forced draft fan, a significant loss might be expected in the fluid bed application where the air-to-gas pressure differential is large. This is compounded by the fact that larger diameter rotors must be used to minimize the resistance in the air and gas passages. In this case, more surface area is required and the sealing area is greater.

A4.4.4 Selection of Equipment and Costs Selections were made for the regenerative air preheater and the tubular air heater in order to evaluate the applicability of both concepts to the fluidized bed boiler. Two selections were made for the regenerative air heater to determine the influence of draft loss on air leakage. In all cases the units were designed on the basis of total gas flow of 610,000 lb/hr at an inlet temperature of 840° and outlet temperature of 275°F. One heater was selected for each boiler module. The gas temperature was selected to meet coal drying requirements and to permit placing the greater portion of feed preheating surface in the convection gas pass zone. A crude optimization of convective heat transfer surface requirements indicated the duty would be nearly evenly split between the air preheater and economizer. Subsequently an optimization study was run to evaluate the selection.

The inlet air temperature was set at 80°F with an approximate outlet temperature of 735°F. Total pressure drop desired was estimated at about 4" H_2O.

The performance and specifications for all three air heaters are summarized in Table A4-3. The tubular air heater was selected over the regenerative air heater on the basis of air leakage considerations. Air leakage amounting to almost 20% could be completely eliminated at an incremental cost of capital investment of only 15%.

Both the tubular and regenerative air heater require field erection. The regenerative air heater is generally shipped in sections and assembled in the field. The units under consideration would require 4300 man-hours, excluding duct work, which amounts to about $43,000. The tubular heat exchanger must be completely field erected. Consideration was given to maximum shop fabrication. It was the opinion of the manufacturer that no real saving in shop assembly would be achieved since so much additional stiffening would

TABLE A4-3

AIR PREHEATER PERFORMANCE

	TUBULAR		REGENERATIVE LOW PRESS. DROP	REGENERATIVE HIGH PRES. DROP
Gas Flow (per unit)	610,000	lb/hr	610,000	610,000
Air Flow	560,000	lb/hr	560,000	560,000
Gas Temp. In °F	840	°F	840	840
Gas Temp. Out °F	274	°F	274	273
Air Temp. In °F	80	°F	80	80
Air Temp. Out °F	750	°F	739	741
ΔP In H_2O	4.5	In H_2O	4.05	8.7
Air Side	3.0	In H_2O	1.6	3.5
Gas Side	1.5	In H_2O	2.45	5.2
Surface	260,000	Ft^2	152,600	122,300
Weight	986,000	Lb	469,000	349,000
Air Leakage	-	Lb/hr	118,000	105,000
Cost (4 Units)	$1,040,000		$899,000	$697,000

Gas Composition

Constituents	% Mole	% Wt.
CO_2	14.76	22.0
H_2O	8.71	5.25
N_2	74.46	71
SO_2	330 ppm	330 ppm
O_2	1.79	1.75

Gas Ash Loading 1.03 grains/scfm

be needed or individual assemblies just for shipping purposes. It was estimated that 8,360 field hours would be required for field erection or about $83,600 per heater.

A4.4.5 Optimization Study An optimization study of the distribution of heat recovery surface in the convection heat transfer zone was made to determine what trade-offs could be made between steam heating surface and air preheating surface. In this particular case heat recovery surface refers to all surface subjected to convective heat transfer for all surfaces not submerged in the fluid bed. It excludes the convective heat transfer surface above the beds which is normally considered an integral part of the bed enclosure. The trade-offs are evaluated in terms of the parameters that might affect the optimum distribution of the surface. These include the cost of surface, the heat transfer coefficient, the water temperature to the boiler, the stack gas exit temperature and the fluid bed temperature.

Two banks of surfaces must be installed in the flue gas between the fixed temperatures of 1600°F and 275°F for the exchange of heat to the inlet feedwater and the air to the combustion zone. These banks may be represented as blocks of heat exchanged as illustrated in Figure A4-23. As an initial condition it may be assumed that the air inlet temperature is fixed at 80°F and the water inlet temperature is fixed at 400°F. Outlet temperatures of the air and water are dependent upon the duty to each heat exchange bank. The duty in turn is fixed by the prime variable in the study, the gas exit temperature from the high temperature bank which is also the gas inlet temperature to the low temperature bank. Variations in air and water outlet temperature determine

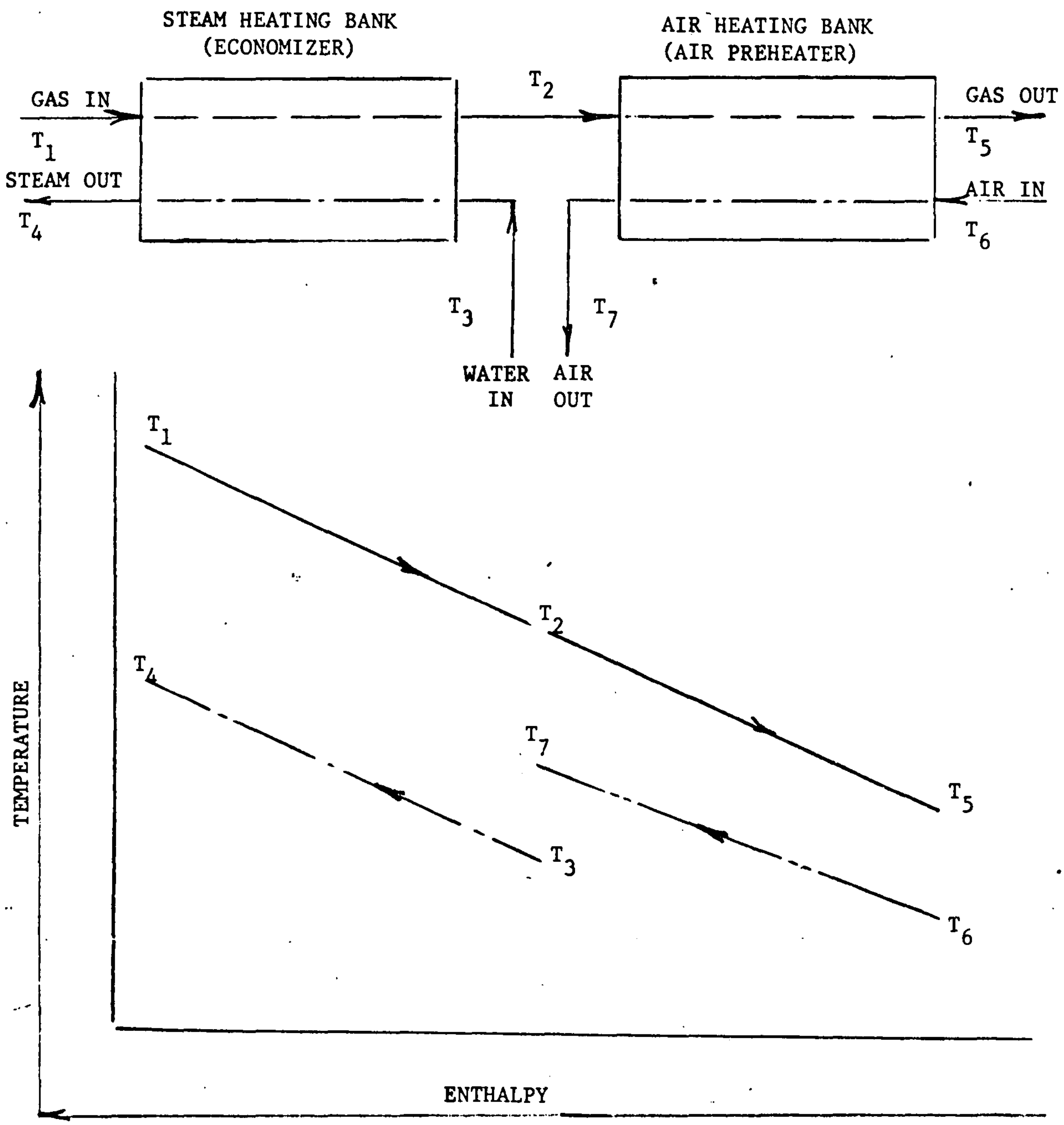


FIGURE A4-23 TEMPERATURE DISTRIBUTION HEAT RECOVERY EQUIPMENT the quantity of additional surface which must be installed or removed from the bed to handle the increase or decrease in heat released with the change in combustion air inlet temperature. It should be pointed out that the bed temperature is fixed and the duty required to generate steam is constant. The bed temperature acts more or less, like a fulcrum about which the bed and convection surfaces are balanced. At high flue gas exit temperatures to the air heater, feedwater heating surface or economizer surface must be placed in the bed. At low gas exit to the air preheater the surface must be placed in the convection heat transfer zone or heat recovery area.

Optimizing requires selecting the flue gas temperature to the air preheater which gives the minimum total cost of economizer surface in the heat recovery area, air preheater surface and feedwater heating surface assigned to the fluid bed for a given set of costs of surface and heat transfer coefficient assigned to the respective surfaces. Westinghouse (12) investigated cost of air preheating surfaces and economizer surfaces and found that economizer surface ran about $\$3/ft^2$ with heat transfer coefficients ranging from 10 to 20 $Btu/hr\text{-}ft^2\text{-}°F$. For this analysis it might be assumed that surface would cost the same but have heat transfer coefficients of about 5 $Btu\text{-}hr\text{-}ft^2\text{-}°F$. Regenerative air preheater surface should run about $\$1.2/ft^2$ for exchangers with total installed surface exceeding 20,000 square feet with

(12) D. H. Archer, et. al, "Evaluation of the Fluidized Bed Combustion Process", Tenth Monthly Progress Report to NAPCA Contract No. CPA70-9, 1970.

heat transfer coefficients in the vicinity of 5 Btu/hr-ft^2-°F. The cost of tubular heat exchanger surface runs slightly higher than regenerative air heater surface. Riley estimated a figure of \$1.5/ft^2 as a very general figure without reference to size. Overall heat transfer coefficients can be expected to range between 3-5 Btu/hr-ft^2-°F.(13)

Data obtained from vendors quotation indicate it is fair to assume economizer costs of about \$3/ft^2, tubular air heater cost of \$ 1/ft^2 and regenerative air heater cost of \$1.47/ft^2. For the purpose of the analysis the exact costs are unimportant, as a design point will be selected based on some reasonably assumed values. The unit cost of conductance which is the unit cost of the surface divided by the heat transfer coefficient will be assumed a parameter and its effect on the optimum flue gas exit temperature will be evaluated.

As indicated earlier the total cost of heat recovery surface may be written as follows:

$$C_T = C_e + C_a + C_{fb} \qquad (1)$$

Where:

C_T = Total cost, dollars

C_e = Cost of economizer surface, dollars

C_a = Cost of air preheater surface, dollars

C_{fb} = Cost of feedwater heating surface assigned to the fluid bed, dollars

(13) D. H. Archer, et. al., "Evaluation of the Fluidized Bed Combustion Process", Sixth Monthly Progress Report to NAPCA, Contract No. CPA70-9, 1970.

The cost of the individual components must be expressed in terms of the cost of conductance and heat exchanger performance. Each heat exchanger bank's performance must be expressed in terms of the limiting temperature of the fluid stream. These limits are either constant or they can be expressed in terms of the flue gas exit temperature, the primary variable with which we are concerned and the one variable that divides the duty to the two heater banks. Recognizing these facts, equation 1 can be written in terms of the flue gas exit temperature with a few simple substitutions. Thus:

$$C_T = C'_e S_e + C'_a S_a + C'_{fb} S_{fb} \tag{2}$$

Where:

C'_e , C'_a , C'_{fb} = Unit cost dollars/sq. ft.

and

S_e, S_a, S_{fb} = surface required, sq. ft.

Surface is a function of heater performance based on simple heat transfer equations.

ECONOMIZER PERFORMANCE

$$Q = U_e S_e \frac{(T_1-T_3) - (T_2-T_4)}{\mathrm{Ln}\left[\frac{T_1-T_3}{T_2-T_4}\right]} = W_g C_{pg} (T_1-T_2) \qquad (3)*$$

* Nomenclature appears at the end of this section.

$$W_g C_{pg}(T_1-T_2) = W_s C_{ps}(T_3-T_4) \tag{4}$$

Simplifying Eq. 3 and Sub. Eq. 4:

$$S_e U_e\left[\frac{1}{W_g C_{pg}} - \frac{1}{W_s C_{ps}}\right] = \mathrm{Ln}\left[\frac{(T_1-T_3)}{(T_2-T_4)}\right] \tag{5}$$

Air Preheater

$$Q = U_a S_a \frac{\left[(T_2-T_7) - (T_5-T_6)\right]}{\mathrm{Ln}\left[\frac{(T_2-T_7)}{(T_5-T_6)}\right]} = W_g C_p(T_2-T_5) \tag{6}$$

$$W_g C_{pg}(T_2-T_5) = W_a C_{pa}\ (T_7-T_6) \tag{7}$$

Similarly

$$S_A U_A\left[\frac{1}{W_g C_{pg}} - \frac{1}{W_s C_{ps}}\right] = \mathrm{Ln}\left[\frac{(T_4-T_7)}{(T_5-T_6)}\right] \tag{8}$$

Fluid Bed Surface

$$Q = U_{pb} S_{fb} \frac{\left[(T_{1600}-T_{s1}) - (T_{1600}-T_{s2})\right]}{\mathrm{Ln}\left[\frac{T_{1600}-T_{s1}}{T_{1600}-T_{s2}}\right]} \tag{9}$$

Simplifying:

$$S_b = \frac{W_g C_p(T_2-T_4)}{U_b\left[T_{1600}-\frac{(T_2-T_4)}{2}\right]} \tag{10}$$

Since two of the four limiting temperatures of the fluid stream in each heat bank are constant one of the remaining temperatures can be expressed in terms of the one remaining variable common to both heat exchanger bank T_2.

Equations (5) and (8) indicate that the surface of each heat exchanger can be expressed in terms of the flue gas exit temperature T_2. Without going through the physical exercise, equaling equations (5), (8) and (10) can be substituted back into the original cost equation. The total cost, C_T, then, can be differentiated with respect to the flue gas exit temperatures T_2. Thus:

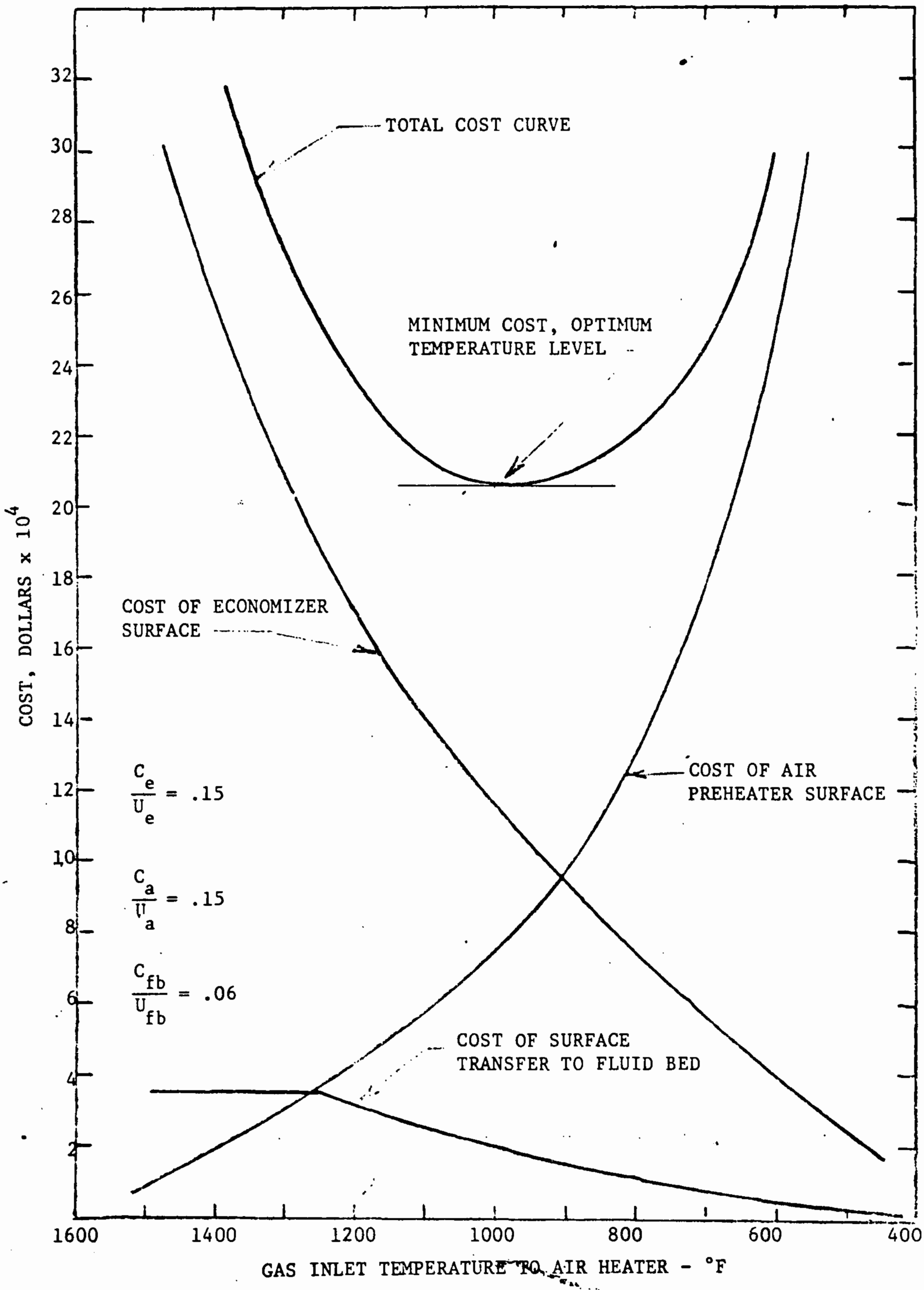
$$dC_t = \frac{C_a}{U_a} \frac{\partial(S_a)}{\partial(T_2)} dT_2 + \frac{C_e}{U_e} \frac{\partial(S_e)}{\partial(T_2)} dT_2 + \frac{C_{fb}}{U_{fb}} \frac{\partial(S_{fb})}{\partial(T_2)} dT_2 \quad (11)$$

In equation 11 the heat transfer coefficients have been factored out as they are constant.

Setting $\frac{dC_T}{d_{T2}} = 0$ and solving the equation for T_2 an expression is developed for the optimum flue gas exit temperature from the steam heating bank. This must then be substituted back into equation (2) to obtain the expression for the minimum total cost of surface. From equation (11) it is noted that the equation can be expanded to include numerous other components of cost that may have been considered similar and thus neglected by simply expressing them in terms of T_2 and adding on.

The final expression for the minimum total cost and optimum operating temperatures are very complicated and require considerable algebra to manipulate and simplify. It is much simpler to plot the cost curves, for the economizer, air preheater and fluid bed surface, and graphically add them together. The result is an optimum curve for a single set of parameters, i.e., constant T_1, T_4, T_5, T_6, U_A, U_E, U_{FB}, C_A, C_E, C_{FB}, C_{PG}, C_{PA}, C_{PS}, W_G and W_A. This is illustrated in Figure A4-24.

FIGURE A4-24 OPTIMIZATION OF ECONOMIZER
AND AIR HEATER SURFACE IN THE
HEAT RECOVERY SECTION OF THE BOILER



Heat transfer coefficients and cost per unit surface are parameters that can be combined to form unit cost of conductance which simplifies and enhances the analysis. Each one of these parameters may now be carefully examined to determine what affect they have on the optimum temperature level and minimum cost of total surface. Figures A4-25, A4-26, A4-27, A4-28 and A4-29 show the affect of the cost of conductance of the air heater economizer and fluid bed surfaces, and the feedwater inlet and the gas exit from the air heater on the minimum operating temperature and minimum cost of conductance.

Numerous conclusions can be drawn from these curves. First of all the optimum operating temperature nearly splits the duty between heat exchanger banks. This is not unexpected if it is realized that the cost of conductance of the two banks are nearly the same. Differences in duty are primarily dependent on the differences in mean-temperature. As the cost of conductance of the air heater increases the optimum operating temperature (T_2) drops and the tendency is to eliminate the less desirable heat transfer surface. The same is true of the economizer except that the operating temperature (T_2) must rise. Economizer in this study loosely refers to all convection surfaces including some evaporating or superheating surface for those cases in which the heat available exceeds the duty required by the economizer or feedwater heating section.

For the design conditions it is noted that the optimum operating temperature is slightly above the temperature originally selected. The effect of the deviation in cost does not warrant a change. Furthermore, other technical considerations such as change in material requirements, etc., may limit the selection to an off-optimum choice.

Improvements in cost of conductance would lower the flue gas exit temperature from the economizer and encourage more use of economizer surface. This affect might be achieved by installing a non-heating fluid bed in the convection pass in which the heat recovery surface would be installed. Such a step would have to be further optimized by including a charge for fan power requirements and capital investment. As indicated earlier this would require adding two additional cost factors to equation 1 or 11, cost of energy, and cost of capital investment.

The cost of conductance of the surface that must be placed in the fluid bed at the high flue gas temperatures to the air preheater has only a small effect on the optimum operating cost of surface. An increase in cost of conductance lowers the flue gas temperature to the air preheater and encourages returning steam heating surface to the convection passes.

Feedwater temperature to the boiler contributes little to the total minimum cost of surface, however, it has a marked affect on the flue gas exit temperature from the "economizer" bank. As the water temperature drops the total cost of surface rises and the optimum operating temperature drops substantially. A drop in fluid bed temperature or decrease in the top temperature limit of the flue gas to the economizer section has a similar effect. In an earlier communication with W. C. Yang (14) a flue gas inlet temperature of 1250 and a feedwater inlet temperature of 350 was selected which resulted in an optimum operating temperature of about 740°F. All the other variables were approximately as selected in this study.

An increase in flue gas exit temperatures from the air preheater reduces the total cost of surface substantially and raises the optimum gas inlet temperature level slightly for the design parameters.

The parametric study provides a quick accurate way of selecting and evaluating the operating conditions for the total heat recovery equipment. It also gives insight as to the effects of future improvements or changes in selection of equipment. Each parameter in this study was evaluated separately. Once the optimum equation is derived and simplified, the influence of parameters on each other at the optimum point can be studied giving families of curves representing different optimum conditions for various parametric valves.

Studies similar to this have also been made by R. Ecabert and L. Silberring (15, 16).

(15) R. Ecabert and L. Silberring, "Optimizing the Stack Gas Temperature and Air Heating on Steam Generators", Sulzer Technical Review 2, 1969.

(16) R. Ecabert and L. Silberring, "Optimizing the Stack Gas Temperature and Air Heating Surface on Steam Generators", Combustion Engineering, 1970.

FIGURE A4-25 EFFECT OF COST OF CONDUCTANCE OF THE AIR PREHEATER ON MINIMUM TOTAL COST AND OPTIMUM TEMPERATURE LEVEL

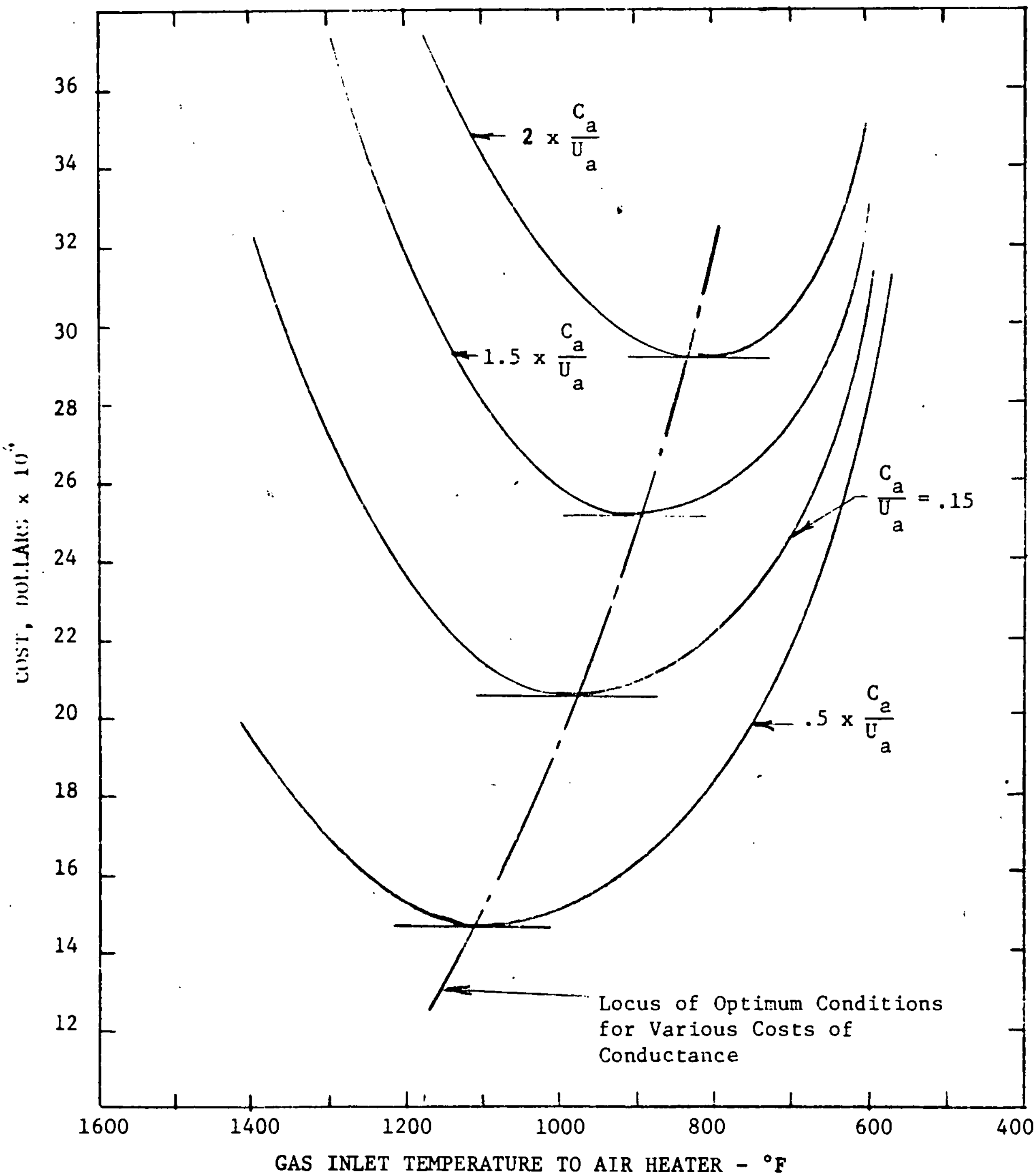


FIGURE A4-26 EFFECT OF COST OF CONDUCTANCE OF THE ECONOMIZER ON MINIMUM TOTAL COST AND OPTIMUM TEMPERATURE LEVEL

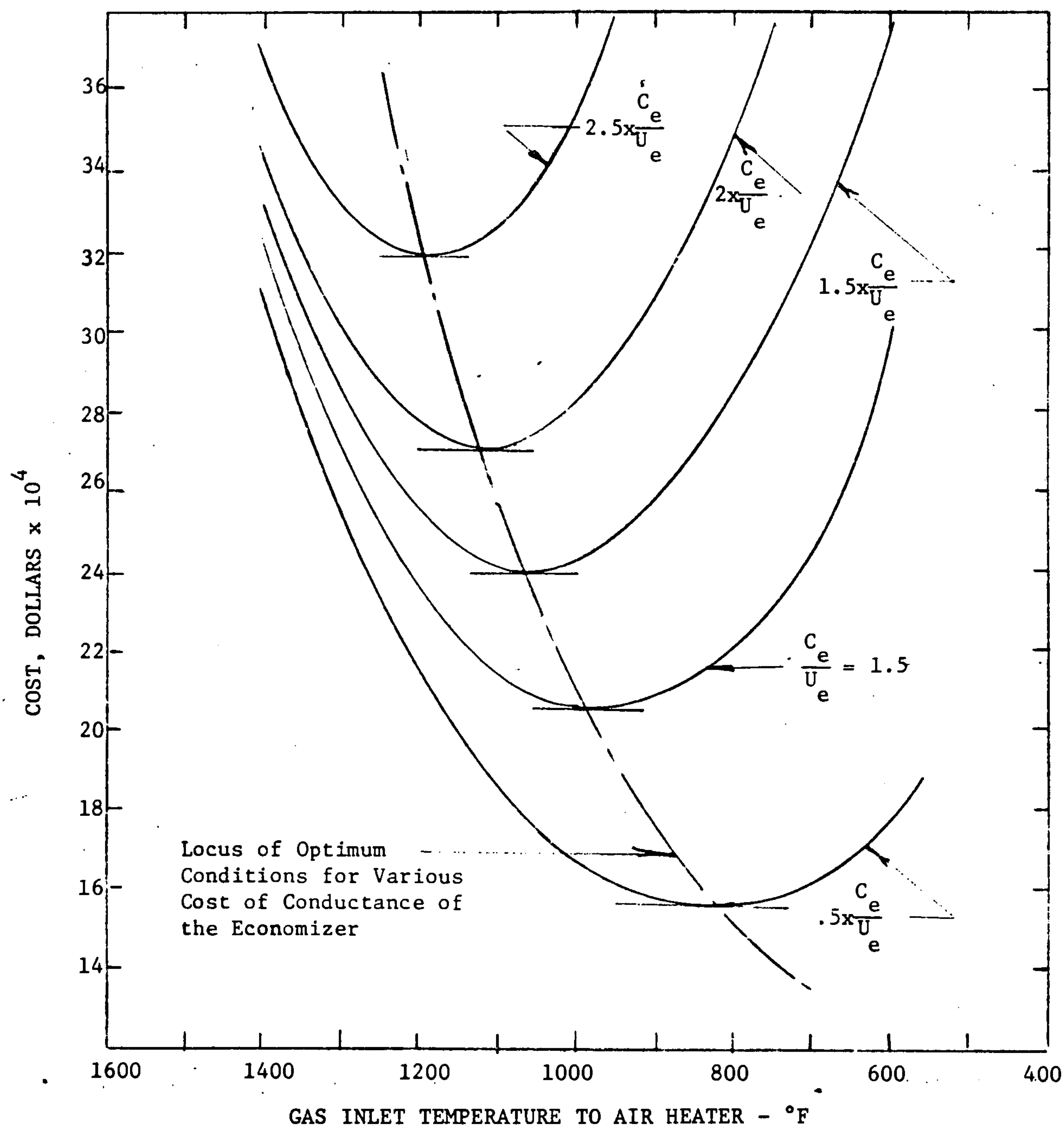


FIGURE A4-27 EFFECT OF FLUE GAS
EXIT TEMPERATURE FROM THE AIR PREHEATER ON
THE MINIMUM TOTAL COST AND OPTIMUM TEMPERATURE LEVEL

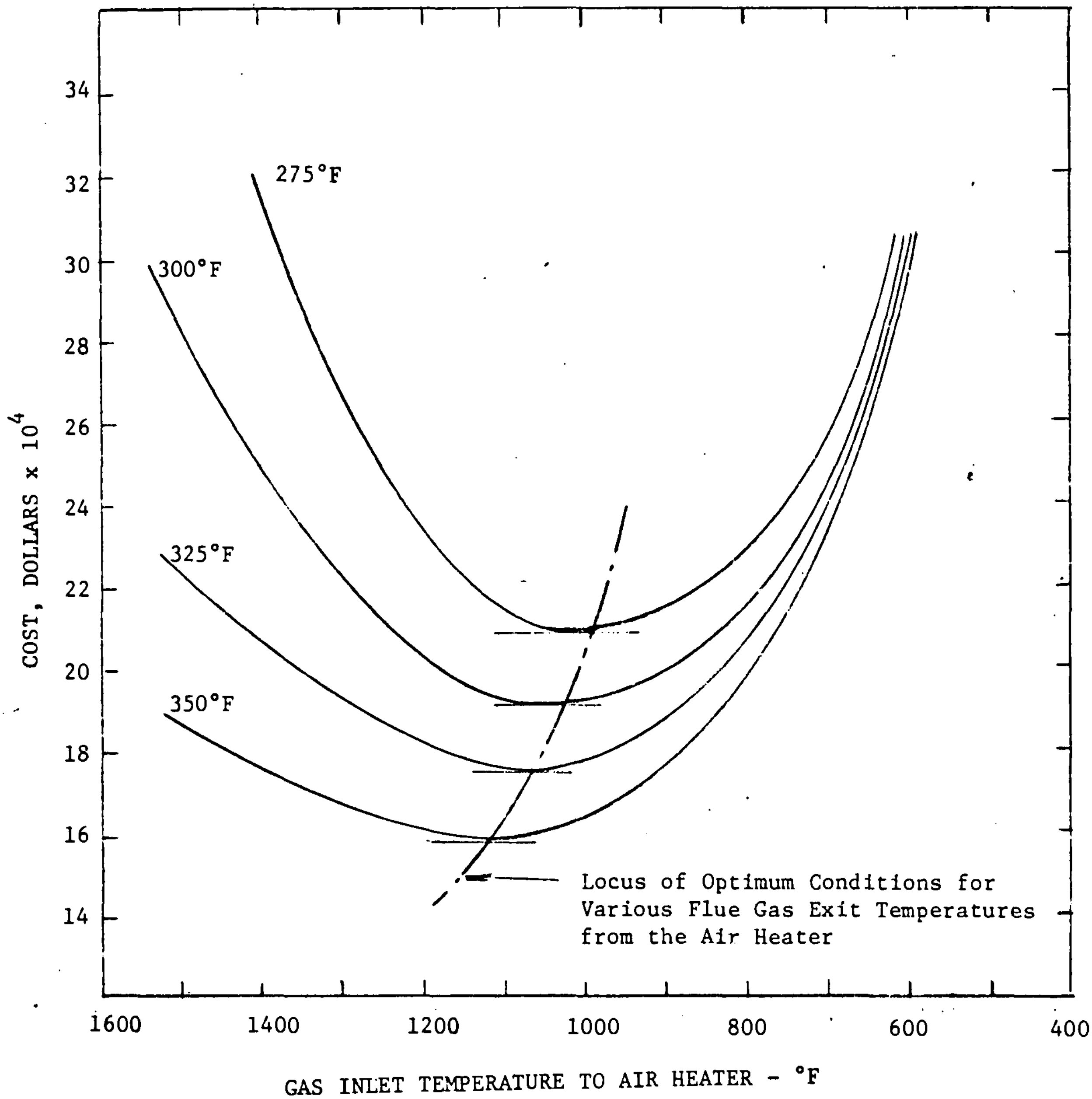


FIGURE A4-28 EFFECT OF COST OF CONDUCTANCE OF SURFACE TRANSFERRED TO THE FLUID BED ON THE MINIMUM COST OF SURFACE AND OPTIMUM OPERATING CONDITIONS

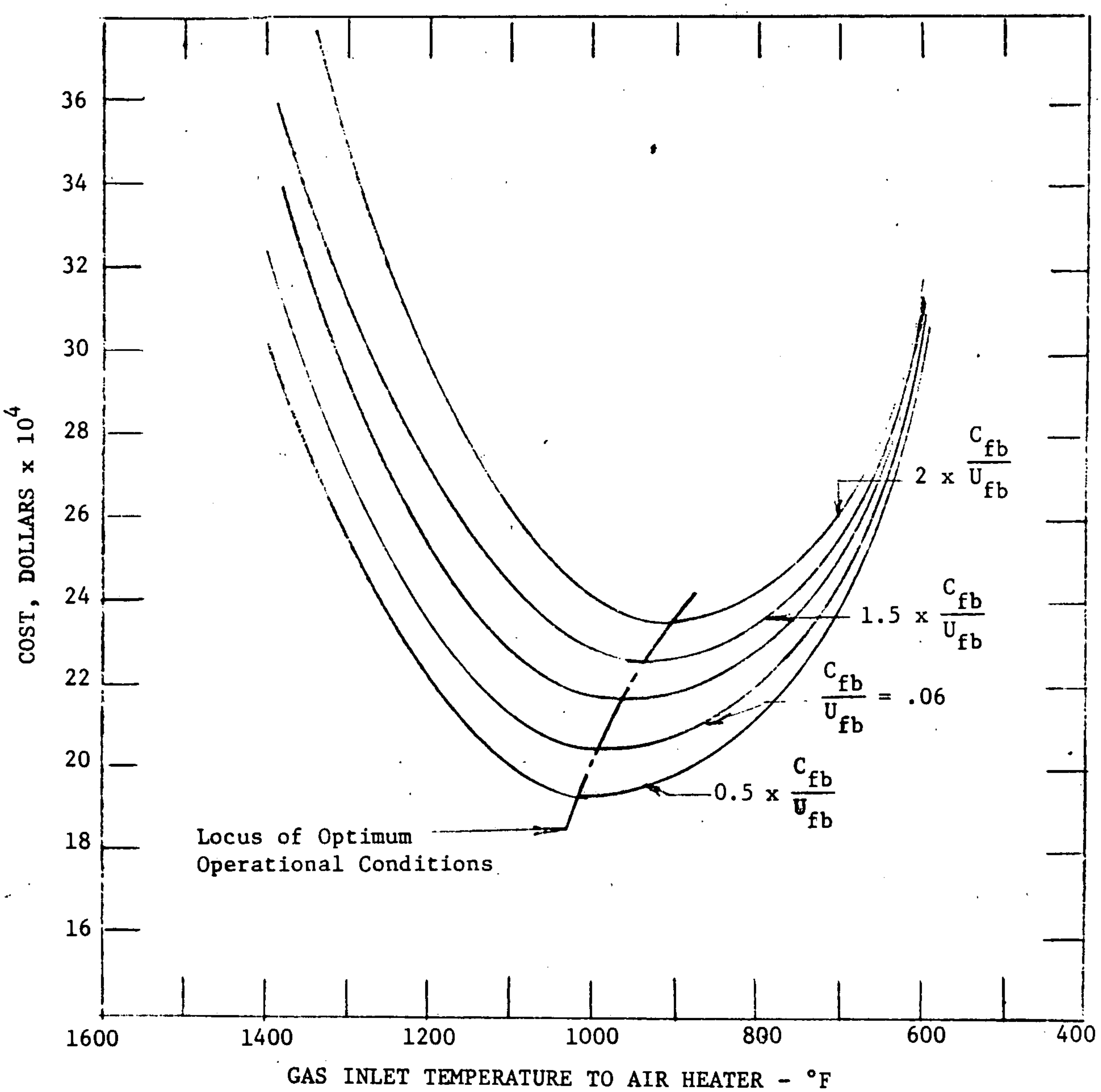
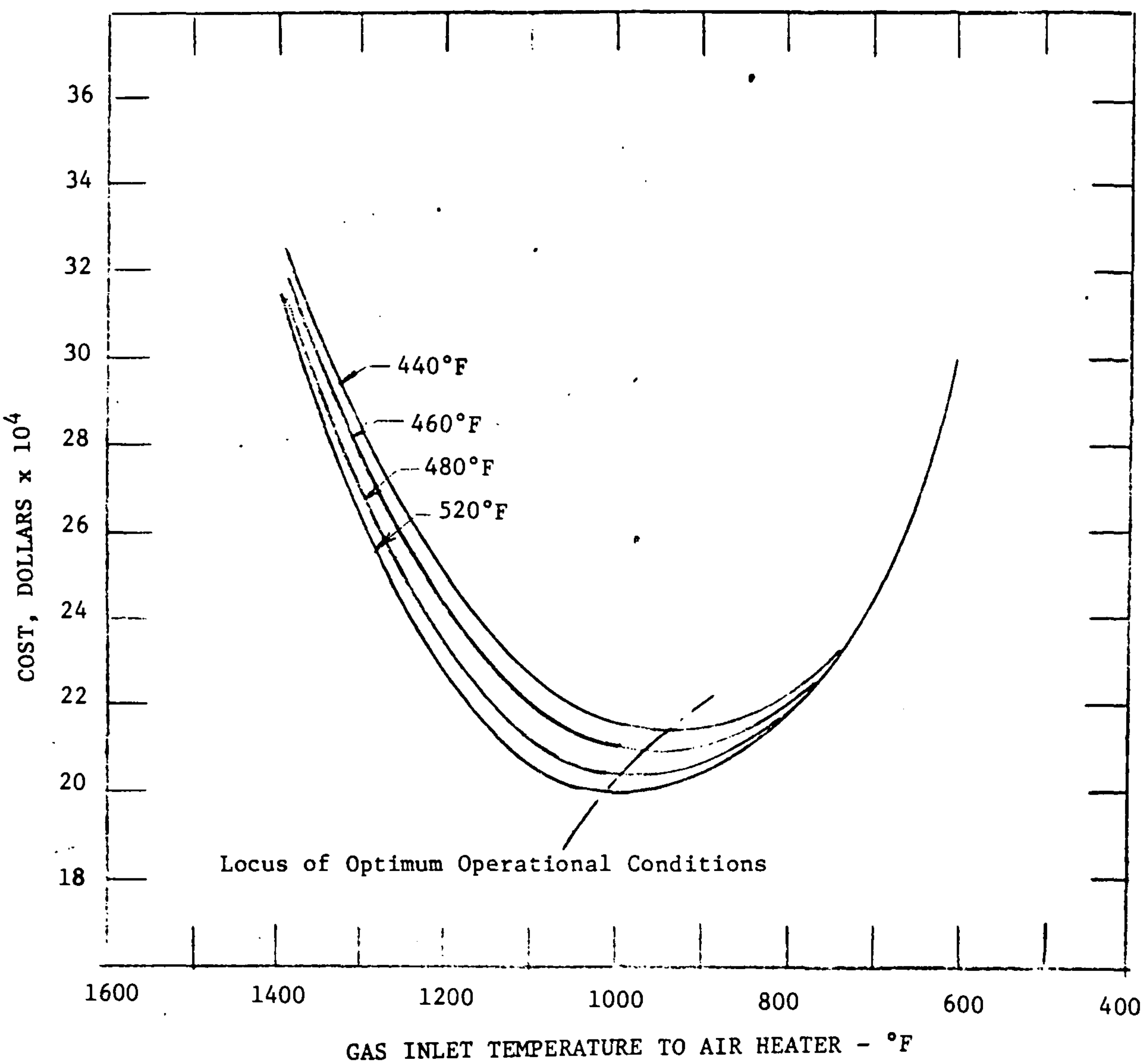


FIGURE A4-29 EFFECT OF FEEDWATER INLET TEMPERATURE ON MINIMUM COST OF SURFACE AND OPTIMUM TEMPERATURE LEVEL



NOMENCLATURE

C_T	-	Total Cost	\$
C_e	-	Cost of Economizer Surface	\$
C_a	-	Cost of Air Preheater Surface	\$
C_{fb}	-	Cost of Fluid Bed Surface	\$
C'_e	-	Unit Cost of Economizer Surface	$\$/Ft^2$
C'_a	-	Unit Cost of Air Preheater Surface	$\$/Ft^2$
C'_{fb}	-	Unit Cost of Fluid Bed Surface	$\$/Ft^2$
S_e	-	Economizer Surface	Ft^2
S_a	-	Air Preheater Surface	Ft^2
S_{fb}	-	Fluid Bed Surface	Ft^2
Q	-	Heat Transferred	Btu/hr
U_e	-	Overall Heat Transfer Coef., Economizer	Btu/hr-ft^2-°F
U_a	-	Overall Heat Transfer Coef., Air Preheater	Btu/hr-ft^2-°F
U_{fb}	-	Overall Heat Transfer Coef., Fluid Bed	Btu/hr-ft^2-°F
W_g	-	Gas Mass Flow	lb/hr
W_s	-	Steam Mass Flow	lb/hr
W_a	-	Air Mass Flow	lb/hr
C_{pg}	-	Specific Heat Gas	Btu/lb-°F
C_{ps}	-	Specific Heat Steam	Btu/lb-°F
C_{pa}	-	Specific Heat Air	Btu/lb-°F
T_1	-	Temperature, Gas In - Economizer	°F
T_2	-	Temperature, Gas Out - Economizer	°F
T_3	-	Temperature, Water In - Economizer	°F
T_4	-	Temperature, Water Out - Economizer	°F
T_5	-	Temperature, Gas Out - Air Preheater	°F
T_6	-	Temperature, Air In - Air Preheater	°F
T_7	-	Temperature, Air Out - Air Preheater	°F

APPENDIX L

BOILER BURNER FOR LOW BTU GAS

ABSTRACT

Burner design and engineering vendors were contacted to prepare a preliminary design and cost estimate and to assess the development requirements for a boiler burner for desulfurized, low Btu gas from the atmospheric pressure fluidized bed oil gasification system. A burner for this gas is technically feasible, and burners could be designed for a demonstration plant. Rough cost projections were made. Correspondence with Process Combustion Corporation and Bloom Engineering Company on a boiler burner design for this application is presented.

PROCESS COMBUSTION CORPORATION

South Hills Village, Suite 300-A • Pittsburgh, Pennsylvania 15241 • (412) 561-6200

12 July 1971

WESTINGHOUSE ELECTRIC CORPORATION
Chemical Engineering Research
and Development Center
(Building 501-1W63)
Beulah Road, Churchill Borough
Pittsburgh, Pennsylvania 15235

Attn: Mr. Dale L. Keairns

Gentlemen:

Ref: ACH-0571-20
PCC/Bloom Boiler Burner
for Desulfurized Gas

Following your recent visit to the Bloom Engineering Company Laboratory, and our mutual discussions with Jim Johns, we are pleased to confirm the points covered.

1. Our experience on boiler burner designs ranges up to 100 MM BTU/hour when using lean coke oven gas. There is no reason why the larger heat releases should not be realized, providing that the combustion space available is adequate.
2. Since we have had the necessary experience in lean gas, we do not envisage having any development work involving your gasified fuel.
3. The gases, at 1600°F, will have no effect on existing gas tube materials employed, and we do not feel that the small quantities of H2S contained in the gases will have substantial corrosion effects.
4. If the burners are vertically upshot fired, and utilizing an open nozzle gas gun, the carry-over of limestone particles from the fluid bed should not have any build-up effects within the tube. However, this is something which only will be brought out by experience, and if we find a problem in the actual operating plants, we are confident that there exists satisfactory solutions within our present technology.
5. Taking as an example the 100 MM BTU/hour unit, we estimate the price of the burner and port block to be approximately $12,000.
6. Operational maintenance on the burners will be very low, since they contain no moving parts, comprising only cleansing of the ports due to limestone blockage, if this occurs.

- continued -

7. Turndown in excess of 5:1 with good stability should be achievable on this gas.

We hope that you have been able to visit an existing boiler burner site in the locality, and that this together with your visits to our plant has given you a good insight of our application engineering.

If you require further information for the second phase of your development work, we would be most happy to assist.

Sincerely,

Vincent C. Grimshaw

Vincent C. Grimshaw
Sales Engineer

cc: Mr. Richard Newby - Westinghouse Electric Corporation
Mr. James E. Johns - Bloom Engineering Co.

PROCESS COMBUSTION CORPORATION
South Hills Village, Suite 300-A • Pittsburgh, Pennsylvania 15241 • (412) 561-6200

19 July 1971

Mr. Dale L. Keairns
Westinghouse Electric Corporation
Chemical Engineering Research
and Development Center
(Building 501-1W63)
Beulah Road, Churchill Borough
Pittsburgh, Pennsylvania 15235

Dear Dale:

Ref: ACH-051-20
PCC/Bloom Boiler Burner
for Desulfurized Gas

I am pleased to enclose the information requested during our telephone conversation last Friday, 16 July 1971, and confirm the points of our discussion.

Regarding the budget price of $12,000 given in my letter of 12 July 1971, this would be applicable to an established design for a commercial plant where the necessary engineering parameters were established, and possibly several burners involved. For special applications, where engineering and drawing time could only be charged against that specific job, it would be necessary to increase the base price accordingly. This situation could well occur on pilot plant designs, for instance.

The enclosed Bloom drawing C-2209-17 shows the overall layout of a 108 MM BTU/hour combination fired Boiler Burner. This will give you a sense of proportion of the unit, but it is not necessarily analagous to your own situation, and could well vary considerably from the designs which we would propose for your purpose.

Typically, we would expect to use approximately 6 inches w.c. on both the gas and air sides of the burner for a standard unit, but increasing the pressure drop as higher energy flames and smaller combustion volumes came into the picture.

-continued-

I shall be making a request to the Bloom Engineers for a brief evaluation of the feasibility and overall concept of a similarly fired 250 MM BTU/hour unit. This will be processed during the next few weeks, and I shall write to you again when the material is available.

In the meantime, if you require further assistance from us, please feel free to call.

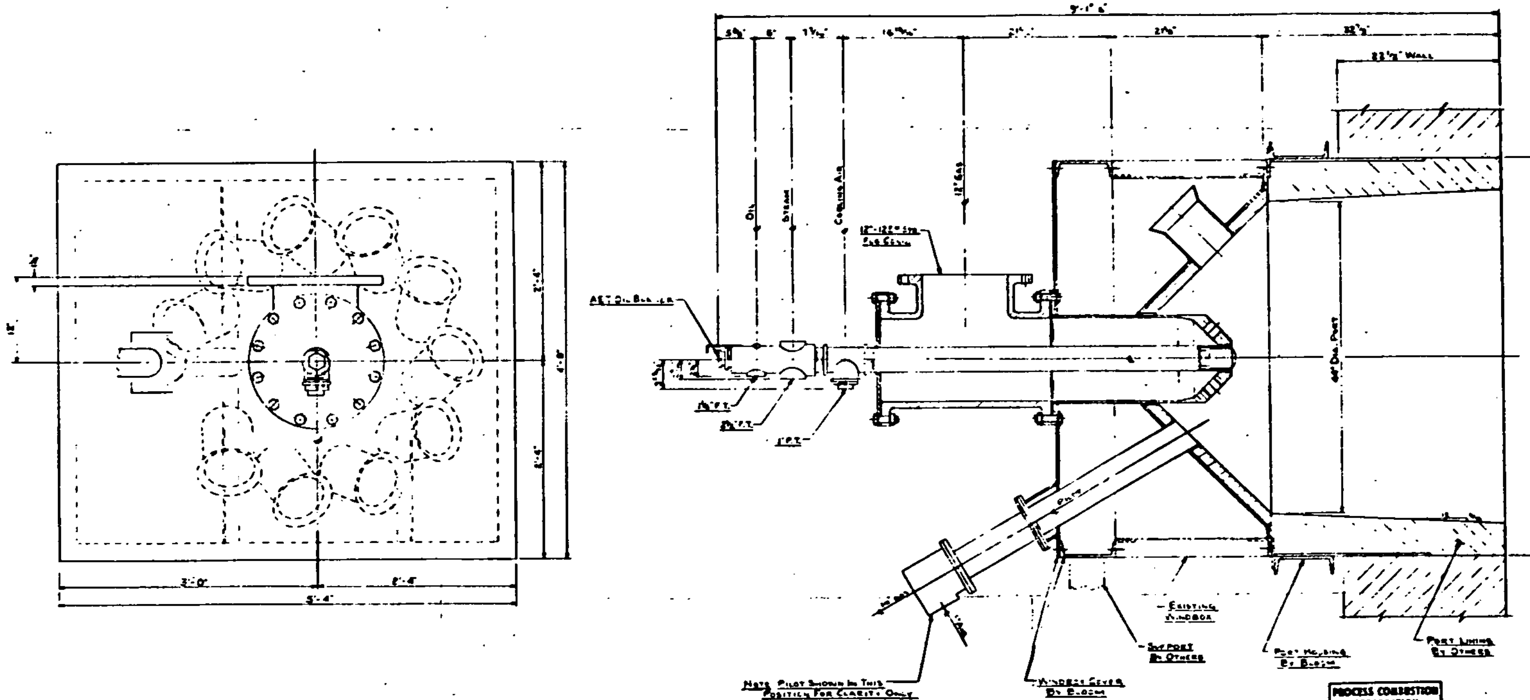
Sincerely,

Vince Grimshaw

Vincent C. Grimshaw
Sales Engineer

dp
Enclosure-Drawing C-2209-17

C-2209-117



PROCESS COMBUSTION CORPORATION
JUL 21 1971
WESTINGHOUSE

PRELIMINARY DRAWING

INTERNAL COPY

THIS DWG APPLIES TO ORDER

BLOOM ENG'R CO. INC.

BLOOM
ENGINEERING COMPANY, INC.
PITTSBURGH, PA.

C-2209-117

L-7

PROCESS COMBUSTION CORPORATION

South Hills Village, Suite 300-A • Pittsburgh, Pennsylvania 15241 • (412) 561-6200

August 11, 1971

Mr. Dale L. Keairns
Westinghouse Electric Corporation
Chemical Engineering Research & Development Center
Building 501-1W63
Beulah Road, Churchill Borough
Pittsburgh, Pennsylvania 15235

Ref: ACH-0571-20
PCC/Bloom Boiler Burner for
Desulfurized Gas

Dear Dale:

We have now received from Bloom a brief evaluation of the possibility of designing and manufacturing boiler burners to handle heat releases in the order of 250 MM BTU/hour.

As you will appreciate, we have only looked at this concept in an overall conceptual sense, and therefore, at this point in time, we could not relate specific design data.

We believe it is totally feasible to design a burner of this type, and we would certainly be prepared to undertake a design contract when the time is right and positive commercial undertakings have been placed by your clients.

Projecting from our present data, we would estimate the port diameter to be approximately 70 inches and the port length 48 inches to achieve good flame stability, and an allowance in the boiler floor or side walls should be allowed to take holes off this dimension.

We hope that you are making good progress with your design studies, and look forward to working with you further during the coming phases of your work.

Sincerely,

Vince Grimshaw.

Vincent C. Grimshaw
Sales Engineer

cc: Mr. Richard Newby

cannot provide [illegible]
until they begin the unit
$20,000 – 25,000 projected

APPENDIX M

GAS TURBINE CORROSION, EROSION, AND FOULING

TABLE OF CONTENTS

A. INTRODUCTION

At one time the gas turbine was thought to be capable of handling the combustion product of any fuel from natural gas to solid waste. This has not proven to be the case. Many ashy type materials in the fuel and even in the combustion air have been found to be corrosive at high temperatures, and those not corrosive may be erosive or may build heavy deposits in the hot regions which tend to choke the turbine and change the blade profile. Solids in the combustion air are sometimes collected in the compressor and are released in chunks, passing through the combustor and eventually reaching the turbine.

Normally combustion takes place at a temperature in excess of 3000°F in a gas turbine combustor. At this temperature chemical reactions are accelerated and the ashy material may become liquid. Although secondary air is admitted to lower the temperature of the combustion products, the ash particles may be molten or at least tacky when they reach the turbine. If so, the ash particles will deposit on the turbine surfaces, which may cause corrosion of the blade material and/or heavy bonded deposits. Even if completely dry, it may be erosive, having solidified into dense glassy beads.

In the pressurized fluid bed system, combustion takes place at a much lower temperature; and it seems likely that dry ashy material carried from the bed (and through the separation) will arrive at the turbine in a dry form unlikely to corrode directly or form bonded deposits. Furthermore, photomicrographs of such ash, made by BCURA using their pressurized fluid bed boiler, showed that the ash was in the form of delicate disks, which should break up upon entering the turbine with no erosion. Their tests also confirmed that the ash (at least from British coals) was not corrosive or deposit-forming.

Production of dry ash has become more difficult over the years of gas turbine development because turbine inlet temperatures continue to rise. At one time gas turbines operated at 1200°F. Now they have passed the 2000°F temperature. At the same time it has been necessary to develop new alloys with improved strength characteristics at high temperature; and this has been done at the expense of corrosion resistance. Present-day alloys have a reduced chromium content and are considerably less corrosion-resistant than those in use ten years ago.

Erosion has not been a problem in Westinghouse turbines with gaseous and liquid fuels even though the additives used with liquid fuels are sometimes abrasive and damage fuel nozzles. Erosion has been a problem, however, in experimental, direct-fired, simple cycle gas turbines burning pulverized coal. Severe erosion resulted at the LDC Dunkirk test stand several years ago even though separators were used. Subsequently the LDC turbine was set up at the Bureau of Mines, Morgantown Laboratory. Redesign and improved separators increased the projected life of the turbine. Projected permissible dust loading was less than 1 gr/100 SCF. This operation will be described in more detail in the review section.

Similarly deposits have not been a problem in Westinghouse turbines mainly because a good grade of fuel is usually specified. However, when residual oil was burned in a test stand under gas turbine conditions, heavy deposits frequently formed. This has been confirmed by other gas turbine manufacturers based on field experience. Brown coal has been burned in an open cycle gas turbine at the Aeronautical Research Laboratories in Australia. It was reported that even with the characteristically low ash content (2%) of their coal and with gas cleaning, heavy deposits formed in a few hours, requiring special blade cleaning procedures. This work will be described in more detail in the review section.

B. REVIEW OF GAS TURBINE OPERATING EXPERIENCE

Pressurized Fluidized Bed Boiler Operation[1,2]

The British National Coal Board has operated a pressurized, pilot-scale fluidized bed combustor at BCURA to obtain process data and to assess the suitability of the combustion gases for gas turbines. A diagram of the combustor and test passage is shown in Figure M-1. The combustor is a refractory vessel of approximately 48" x 24" rectangular cross-section and 10' high. The combustor is housed in a cylindrical steel pressure shell.

The base of the combustor is closed off by a horizontal bubble cap air distributor plate. The fluidizing air, which includes the corrosion tube cooling air, is preheated to 390 - 540°F in tubes immersed in the bed.

During normal operation the fluidized bed contains approximately 1800 lb of coal ash at a fluidizing velocity of 1.8 to 2.0 ft/s; the expanded depth is about 44 inches. Cooling surfaces, consisting of 65 hairpin loops, are immersed in the bed.

A bank of four rows of horizontal, uncooled, heat-resisting, steel tubes, 1" O.D., spaced on a nominal 3" triangular pitch, is installed at the top of the bed as a baffle system to minimize splashing.

The combustion products leave the shaft via a 10" diameter recirculation cyclone suspended from the top closure plate. The fines are returned via a dipleg which terminates approximately 3" above the air distributor plate.

Combustion products from the recirculation cyclone exit pass through 10" diameter primary and secondary cyclones arranged in series. Fines from both cyclones are transported out of the pressure vessel.

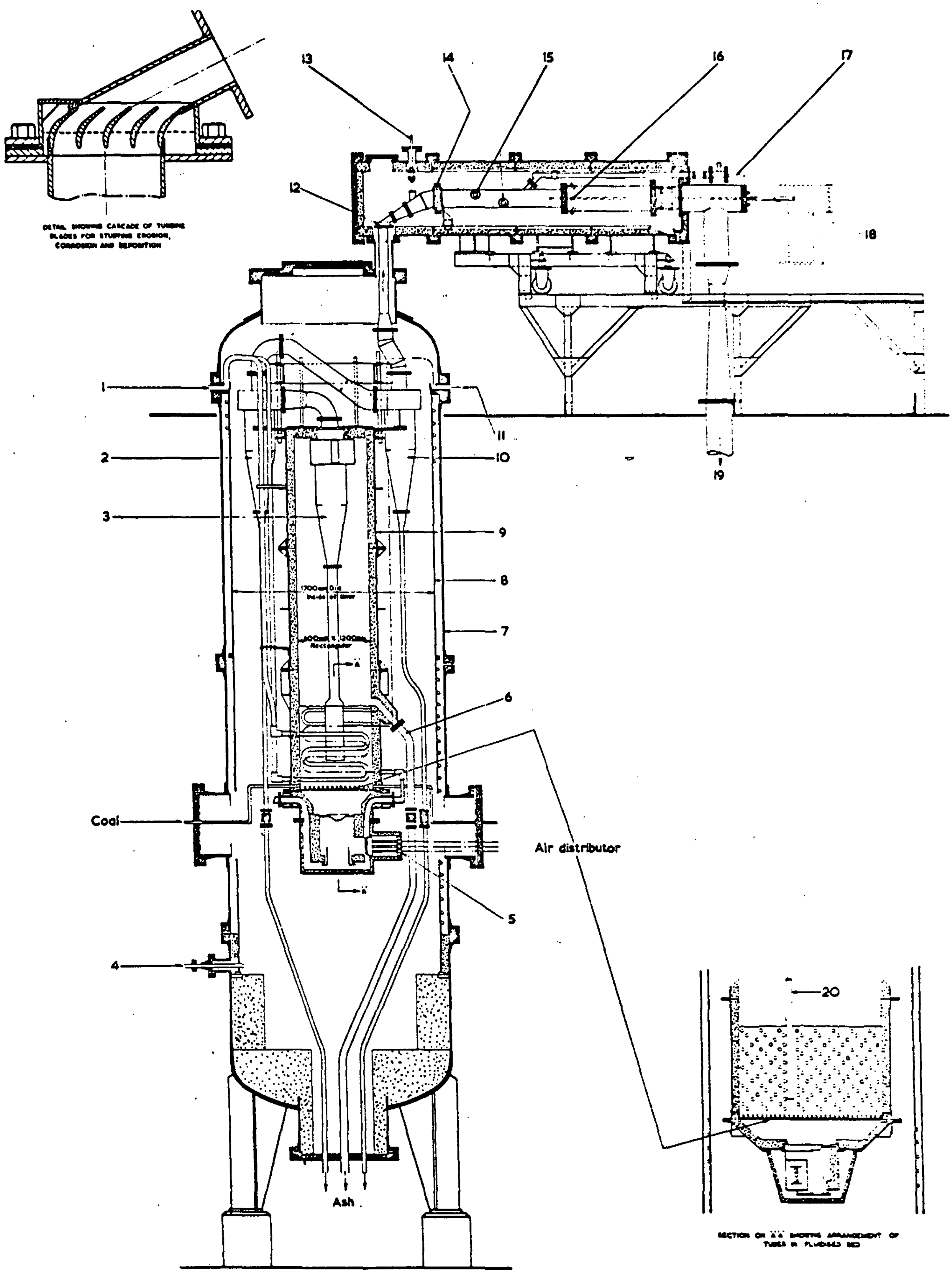


FIG. M-1 The 48in x 24in pressurised combustor

Key to Figure M-1

1	Water inlets and outlets
2	First-stage cyclone
3	Recirculation cyclone
4	Balancing air supply
5	Gas burners for initial heating of bed
6	Ash offtake from bed
7	Pressure casing
8	Water-cooled liner
9	Combustor casing
10	Second-stage cyclone
11	Air intake
12	Cascade
13	'Alkali' and NO_x sampling probes
14	Flow correction baffle
15	Cooled tube for studying erosion, corrosion and deposition
16	Water sprays
17	Air-cooled tube with corrosion specimens for studying condensation of alkalis
18	Dust sample collection and gas analysis
19	To pressure let-down and stack
20	Dip-leg of recirculation cyclone

Note: Air distributor plate is of bubble cap design with 465 caps on a 1 1/2" square pitch

Gases from the secondary cyclone pass through an egg crate type flow straightener before entering the cascade section. The cascade has been made from an inlet nozzle segment of a Rolls Royce Proteus gas turbine engine and consists of five aerofoil blades. The cascade assembly, the measuring duct which follows and the subsequent quench section, are mounted in a separate 2 ft diameter by 9 ft long horizontal pressure vessel.

Data from the pressurized unit have been obtained at the operating conditions summarized in Table M-1. Typical results from the unit using a U.S. coal and dolomite are summarized in Table M-2. Slight deposits were formed on the turbine blade cascade. The deposits can be easily removed. No signs of erosion have been observed. Dust loading of the gases has generally been less than 0.15 gr/SCF with 80 - 90 wt % less than 20 micrometers and 30 - 40% less than 5 micrometers. Most of the particles greater than $\sim$10μm are in the form of "platelets". Thus, it has been very difficult to obtain a realistic size distribution of the ash.

Oil Refinery Experience

Gas turbines have been used on expanders on catalytic cracking units, and problems due to erosion and after-burning have occurred. The two phenomena, erosion and after-burning, are described in a paper[3] by Ingersoll Rand. Erosion, caused by minute particles of catalyst dust entrained in the flue gas, is considered to be more serious. Relative velocity to the rotor blades was 1400 - 1500 ft/sec.

The particle size range was not stated, but it was observed that some particles simply polish whereas larger ones erode. It was also noted that material sometimes collected on one row and let go, eroding the next row. Two fixes were proposed: a) Use three stages of separation. Without separation, serious erosion occurred in 800 hours. Three separators gave satisfactory operation for four or five years. b) Flame-spraying erosion-resistant ceramic coatings on the blades is a promising possibility.

TABLE M-1

PRESSURIZED FLUID BED PILOT SCALE

Operating Conditions

Pressure	up to 6 atm
Bed Temperature	1500°F
Turbine Blade Cascade Temperature	1435°F
Bed Depth	3.75 ft.
Fluidizing Velocity	1.8 - 2 fps
Heat Transfer Surface	
Tube Diameter	1 in.
Pitch	3 in.
Bank Height	2.5 ft.
Sulfur Removal	>95%
NO_X Emission	<150 ppm
Ca/S Ratio	2

TABLE M-2

PRESSURIZED FLUID BED PILOT SCALE RESULTS

Test Series	2
Coal	Humphrey No. 7 (2.8%S)
Sulfur Absorber	Dolomite 1337
Feed Ca/S Ratio	2.04 - 2.09
Duration of Test Period, Hr. (Total)	55
Bed Temperature, °F	1456 - 1465
Pressure, atm	5
S Retention, %	94.4 - 96.8
SO_2 Emission, ppm	72 - 107
NO_X Emission, ppm	73 - 126
Na_2O Emission, ppm	0.3 - 0.5
K_2O Emission, ppm	0.1 - 0.2
Cl Emission, ppm	50
Exhaust Dust	
Dust Loading, gr/SCF	0.07 - 0.16
Composition of alkalis, clorine, carbon, and sulfur	
Carbon	2.3 - 4.0
Hydrogen	0.1 - 0.2
Sulfur	3.3 - 4.4
Chlorine	0.1 - 0.3
CaO	9.0 - 10.5
MgO	5.2 - 6.5
Na_2O	0.7 - 0.8
K_2O	1.6 - 1.8
Median Diameter μm	6 - 8

Aircraft Experience

The problem of dirt removal for small gas turbine engines was considered by the Donaldson Company.[4] These engines are of a size used in helicopters, wheeled and tracked vehicles, and air cushion vehicles. The paper contains a very complete set of 78 references. In general it is oriented toward the use of Donaldson Air Cleaners (which is to be expected since the two authors are both associated with Donaldson). The following items are noted:

1) Experience is shown, both field and laboratory. It is concluded that air cleaners of some sort are necessary and that they can do a satisfactory job of protecting the turbine.

2) Typical dust concentrations around various types of vehicles are shown. For example helicopters may ingest a concentration of 6 to 30 mg/cu ft. of particles 30 - 170 microns. Hovercraft raise concentrations of 140 - 200 mg of 100-micron particles. At the other end of the scale are highway trucks, with 0.004 to 0.1 mg/cu ft. of 0 - 20 micron size.

3) Erosion depends on dust particle size, impact velocity, concentration, particle hardness, angle of impingement, and properties of material being eroded.

4) Laboratory tests are made with five dust specifications:

AC Fine	-	39%	0-5 Micron
AC Coarse	-	30%	40-80 Micron
BS 1701	-	33%	80-152 Micron
Mil-E-5007 c	-	36%	200-400 Micron
Ottawa Sand	-	70%	297-595 Micron

5) Based on literature and experience, erosion has never occurred on a gas turbine fitted with a filter 99% efficient on AC coarse dust.

6) Three basic air cleaner types are offered:

a. Inertial separators
b. Barrier filters
c. Two-stage, i.e., inertial separators followed by high-efficiency barriers.

7) 90% efficiency filters extend engine life by 10 to 20 times compared to no problem with 99 + % barrier filters.

8) Particles smaller than 1 to 2 microns pass through a system without impacting, or deposit only by diffusion.

9) Particles greater than 10 microns probably impact on blade surfaces.

10) A well-designed cyclone should be capable of separating nearly all the particles above 10 to 12 microns.

<u>Blast Furnace Gas Experience</u>

Experience on four gas turbines of 7500 KW rating, which had accumulated a total of 10 years operating time, was reported at the Gas Turbine Conference in Zurich.[5] The turbines were run for the first six months on a blast furnace gas with a normal dust level of 4 to 10 mgr/Nm^3. Later on, an additional electrostatic filter dropped the dust level to 1.2 mgr/Nm^3. Finally, because of lowered gas rating values, it was necessary to enrich the fuel with gas-oil or naphtha.

The first effect was a build up of deposit at such a rate that the power dropped 16% in 4000 hours and 27% in 7000 hours. However, with cleaning, sand blasting and metallic brushing during periodic overhauls, power was always restored.

Some corrosion was also evident, although it was not as bad as that observed in other turbines in Germany burning unenriched blast furnace gas. The first-stage stator and rotor blades of Nimonic 80A (similar to Inconel X) had an expected life of 100,000 hours based on stress considerations. However, it was shown that because of corrosion, the first stage would probably have to be replaced after 30,000 to 40,000 hours, having operated at 700 - 720°C. Cleaning would be required every 7000 to 8000 hours to restore power. If alkaline sulfates are present more severe attack would be expected.

Westinghouse built and operated a blast furnace gas burning gas turbine at the U.S. Steel plant in Gary several years ago. This was the forerunner of the Westinghouse 301 machine but used a large external combustor, i.e., six feet in diameter. The blast furnace gas compressor was the axial flow compressor from the W-21 machine. The gas contained at most 15 grains/1000 ft^3 mostly below 1 μ. With an electrostatic precipitator dust loading dropped to 6 grains per 1000 cu ft. The air rate was 38 lb/sec with 154 lb/sec extracted. This left 164 lb/sec for combustion. The fuel (saturated and 86 BTU/ft^3) was equivalent to 40 lb/sec. The machine ran regeneratively from 850°F to 1350°F. The machine ran at 1350°F turbine inlet temperature with recuperation to 850°F.

The dust loading in the flue gas was: $6(\frac{40}{204}) = 1.18$ gr/1000 ft^3. This contrasts to 140 gr/1000 ft^3 in the BCURA unit. 16000 hours of operation were obtained on blast furnace gas without problem. A light dusty film of red deposit -- probably an iron compound -- was found on the turbine blades. It was not detrimental to the performance of the unit. There was a problem with the gas compressor. The dust bonded badly, perhaps because it was saturated with water. Failure finally occurred, thought to be due to stress corrosion cracking.

Coal-Burning Gas Turbine Experience

Starting in 1951, the Locomotive Development Committee pioneered in the development of an open-cycle, coal-burning, gas turbine which it ran for some 4000 hours in Dunkirk, New York. Various types of separators were tried, but erosion was very severe and the blades were replaced three times during the program, which finally ended in 1959.

The turbine was reassembled in Morgantown, West Virginia, in 1959, and the work[6] was continued. Some redesign was necessary. The blade profiles were changed and the number of stages reduced from six to five. Armored strips of titanium carbide were inserted near the blade platform. Attempts were made to spread out the flow of particle-bearing gases. Nearly 2000 hours of operation were carried out. It was estimated that the stator had a useful life of 5000 to 7500 hours, and the rotor blades, 20,000 to 30,000 hours. Dust loading entering the turbine was 12 grains/100 SCF with sizes and concentrations:

above 20 micron -	2.4%
above 10 micron -	9.7%
above 5 micron -	26.5%
above 2 micron -	100.0%

It was concluded that efforts should be made to reduce the size and concentration of the ash particles.

Following work on the LDC turbine, a test facility was built for further investigations of the effect of dust loading on turbine blade erosion.[7] This facility consisted of a turbosupercharger coupled with a natural-gas-fired combustor. Fly ash obtained from a pulverized coal utility boiler was injected into the combustion gases to give a particulate loading of 1 gr/100 SCF. Erosion experienced in this turbine was much greater than had been projected for this loading from the LDC turbine tests. However, the design velocities in this turbine are unrealistically high for utility gas turbines and the size of the fly ash used was 94% by weight greater than 5 μ. Because of these two factors, the results of the later tests at the Bureau of Mines are inconclusive.

In Australia, the Commonwealth Department of National Development and the Joint Coal Board have for some years sponsored a program for burning pulverized coal in an open cycle Ruston and Hornsby TA gas turbine.[8] The turbine has been operated in two forms, without cleaning and with cleaning using a multicyclone ash separator. The combustor was interchangeable with the standard oil-burning unit, but was modified to provide a larger residence time for burning coal particles.

The fuel burned was a brown coal pulverized to 80% through a 300 mesh sieve. The ash content was low by U.S. standards -- 1.5 to 2.5%. Residence time in the combustion zone was 100 milliseconds. Turbine inlet temperature was 650°C. The ash composition was:

SiO_2	0.07 - 0.82%
Al_2O_3	0.94 - 0.12
Fe_2O_3	1.75 - 0.23
CaO	0.37 - 0.08
MgO	0.51 - 0.14
Na_2O	0.19 - 0.05
K_2O	0.02 - 0.01
TiO_2	0.07 - 0.01
S	0.41 - 0.22
Cl	0.16 - 0.04
Total Ash	2.5 - 1.5

It was estimated that the ash contained 50% unburned coal with a dust loading of 9.4×10^{-4} lb/lb of gas leaving the combustor and 3.4×10^{-4} lb/lb of gas leaving the separator. Average particle size leaving the combustor was 37 microns and that from the separator, 5 microns. All particles in excess of 23 microns were removed.

Although it was implied that erosion could be a problem, this study was devoted mainly to deposit buildup. Two types of deposit were evident, a dense, sintered material near the leading edge of the blade, and a light, powdery material on the convex face. The powdery material was easily removed. Underneath the dense deposit the blade surface was shiny, with no indication of chemical attack. Analyses of the deposits showed oxides of silicon, magnesium, and iron. It was concluded that initially calcium sulfate was the bonding agent and that over a period of time the level of sodium sulfate, also a bonding agent, built up. Final bonding took place in as little as 30 minutes, and sodium sulfate had migrated to the metal surface in two hours.

It was concluded that severe ash deposition would occur from burning brown coal either with or without a separator. The rate of build up depended upon the concentration of sodium, and incandescent coal particles were found to accelerate bonding by producing locally high temperatures in the vicinity of the blade. Gas velocity and temperature also had a measurable effect. Deposits built up to an objectionable degree in about 20 hours. No way to inhibit the formation of the bonding sulfate could be suggested. Consequently a method of cleaning was developed, adding water (under reduced load) to dissolve soluble materials and milled apricot shells to abrade away the more heavily bonded deposits.

In later paper[9], the Australian coal-burning unit indicated that erosion had in fact occurred in the Ruston and Hornsby turbine using brown coal but that a high-pressure turbine blade life of several thousand hours could be achieved by using the ash separator. The erosion of the low-pressure turbine blades was much less, and a blade life of many thousands of hours was predicted.

C. EROSION IN GAS TURBINES

Erosion of gas turbine blades is a function of:

1. Dust particle size
2. Dust particle physical properties
3. Dust particle shape
4. Dust particle concentration-weight per unit volume
5. Impact velocity
6. Impact angles
7. Turbine material physical properties.

Finnie[10] has shown by analysis and experiment that the erosion from a ductile metal by a given particle is proportional to $V^2 f(\alpha)/p$, where V is velocity, α is angle, and p is plastic flow stress. The flow stress is proportional to hardness. The maximum erosion occurs at angles between 10 and 20 degrees for angular particles and at about 30 degrees for spherical particles.

For brittle materials such as glass, porcelain, and hardened steel the weight loss was found to be proportional to the velocity to a power varying from 3 to 6.[10] The maximum rate of erosion of brittle materials occurs at near 90°.

Martlew[11] has investigated the impingement of ash particles on turbine blading by calculation and test for turbine inlet conditions of 120 ft/sec and 75°F. Figures M-2 and M-3 show the calculated trajectories of 4 and 16 μ particles. Figures M-4 and M-5 show the resultant velocities and angles of impact with the concave blade surface for the two particle sizes. These show that the smaller particles have higher impact velocities but less critical impact angles then the larger particles. The angle of impact is nearly constant over the blade chord distance, but the velocity increases with distance from

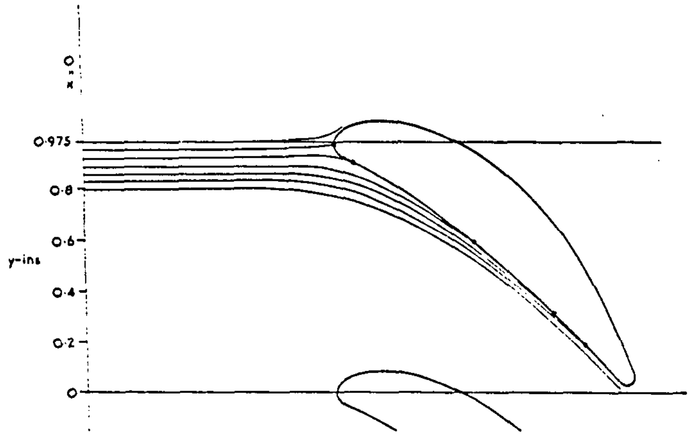


Fig. M-2 - Trajectories of 4 μ particles[11]

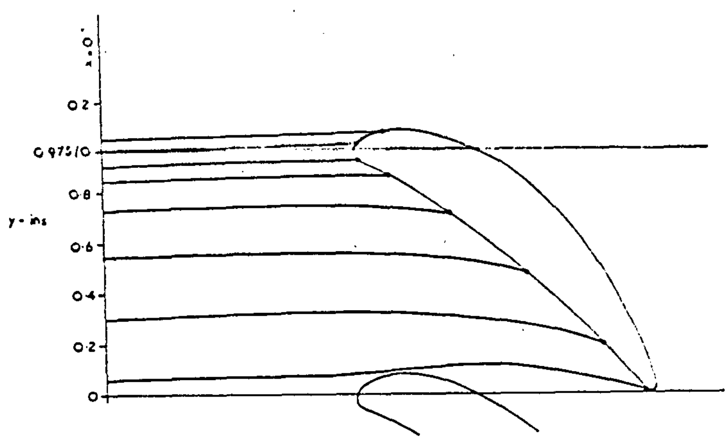


Fig. M-3 - Trajectories of 16 μ particles[11]

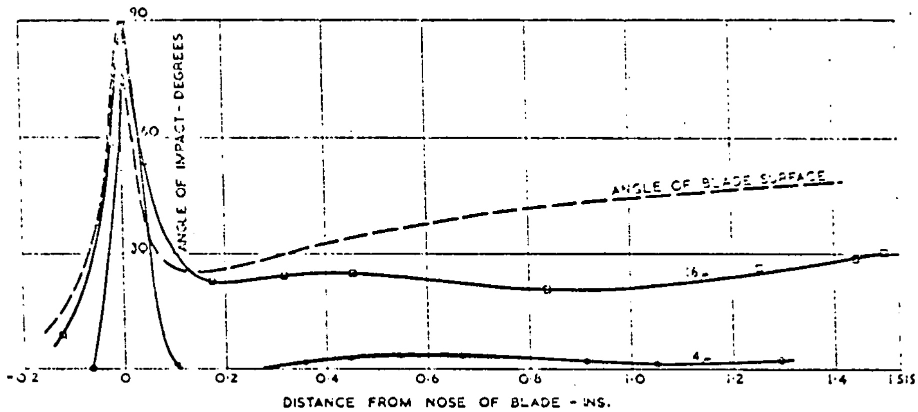


Fig. M-4 - Angle of impact (11)

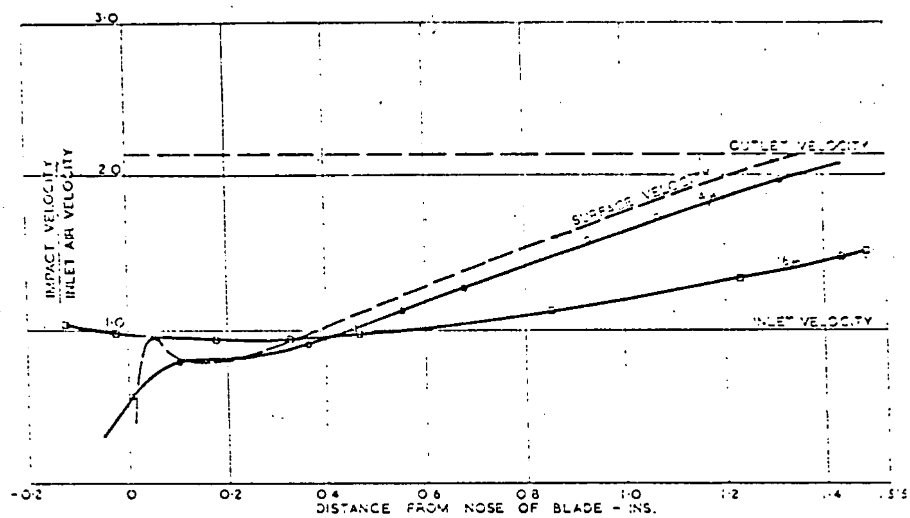


Fig. M-5 - Velocity of impact (11)

the nose of blade. This indicates that the erosive effect of particles in this size range is greatest at the blade trailing edge.

For the purpose of determining the minimum particle size which will impinge on the concave surface of a turbine blade, Martlew has introduced the impact number, I.[11,12] Under conditions where Stokes law applies, I is defined by the following groups of dimensionless numbers.

$$I = K \left(\frac{\delta_p}{\delta_g}\right) \left(\frac{Y}{L}\right)^2 \left(\frac{L\delta_g V}{\mu_g}\right)$$

where δ_p = particle density

δ_g = gas density

Y = particle radius

L = characteristic length (blade chord)

V characteristic velocity (relative)

μ_g = gas viscosity

It is assumed that a particle which has an impact number less than the critical value for a circular cylinder* of radius equal to the nose radius of the blade, cannot impinge on the nose or any other part of the blade. The blading configuration which was used in Martlew's analysis had a minimum particle diameter for impingement of 0.7 microns.

Catch efficiency is defined by Martlew[11] as the ratio of actual impact rate to the maximum possible rate contained in the band of gas which the blade intercepts in the approach velocity pattern. Figure M-6 shows the variation of catch efficiency with particle size for the turbine blade used in Martlew's study.

For flow around a bend where the radius is large compared to the passage width, the radial velocity, assuming Stokes law applies, is

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* This value is that at which impaction on the cylinder is incipient.

Curve 646940-A

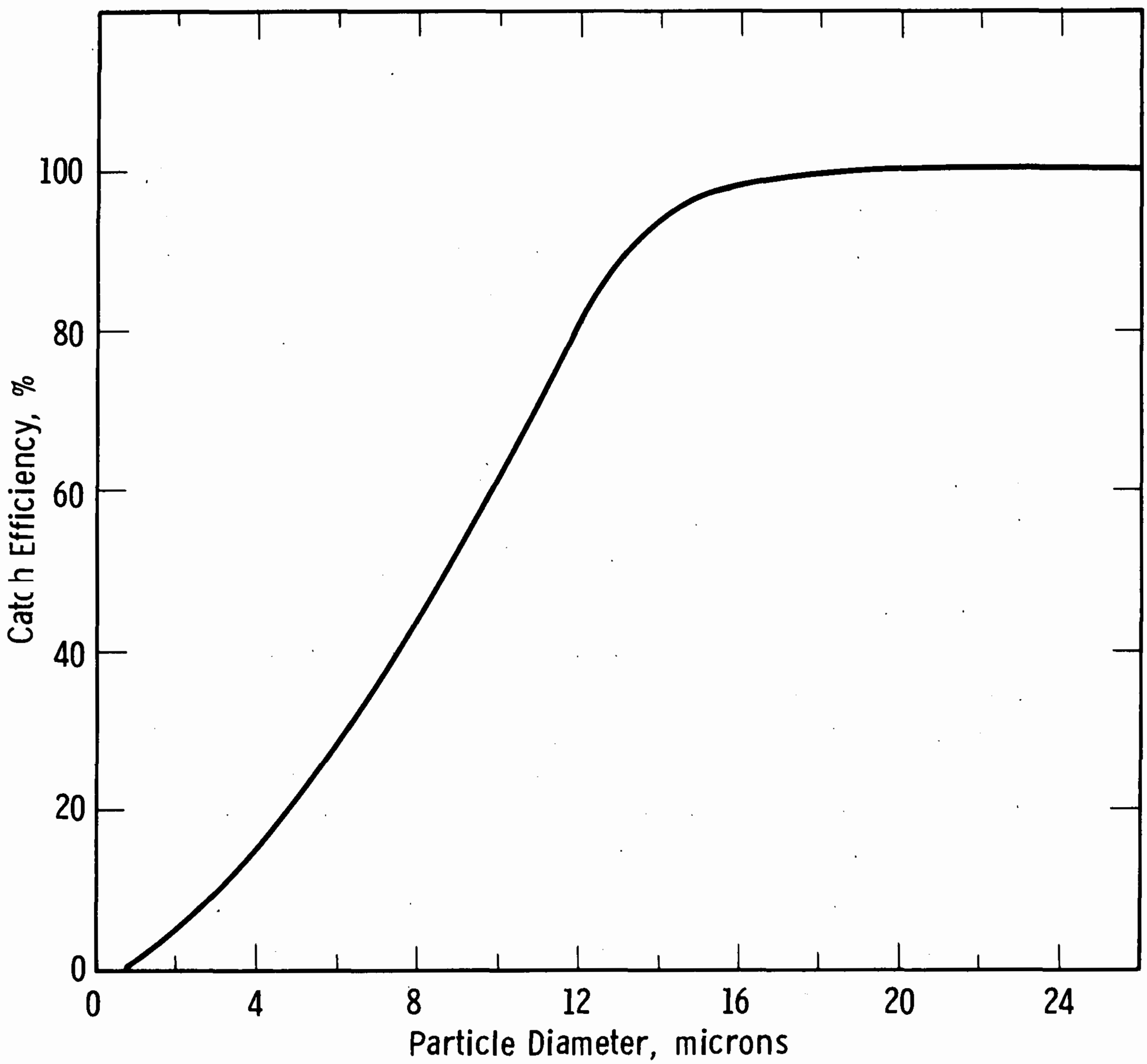

Fig. M-6 — Catch efficiency of ash particles on turbine blades (11) approximately proportional to V^2/R. Hence, the angle of impact would be proportional to V/R. For impact angles up to about 20°, the erosive effect is roughly proportional to the angle. Therefore, for a given ductile metal and particle size, the erosive effect is proportion to $V^2 \times \frac{V}{R} = \frac{V^3}{R}$.

The number of particles entering the passage between two turbine blades is proportional to the product of the volumetric particle concentration and the gas inlet velocity, i.e., CV. Combining this with the previous expression indicates that the rate of erosion of a turbine blade for a given particle size consist is

$$\frac{V^3}{R} \times CV = \frac{CV^4}{R}$$

Wisdom[12] cites experimental data which indicate that the erosion rate is proportional to the fifth power of the velocity.

Using Martlew's trajectory and catch efficiency data, and the particle erosion data obtained by Finnie, Dygert and Bjorklund[13] estimated the life of turbine blading for two condition sets. It was assumed that the useful life of turbine blading ends when 10% of the total blade weight is removed. For one case where a high-efficiency separator was used, the computed blade life was 14,000 hrs. In a 4000-hr test no blade wear was discernable, whereas a 3% blade weight loss was predicted for this period.

Without the high-efficiency separator, the computed blade life was 1400 hrs. A 650-hr test at these conditions indicated at least an order of magnitude correlation, with the computed value being conservative.

Fisher and Davis[14] conducted laboratory experiments on erosion using fly ash samples from ten different bituminous coals. The fly ash particles were introduced into a heated air stream in the quantities required to simulate the exhaust from pulverized (coal combustion) and tests were conducted with and without Aerotec dust separators. The test specimens were discs, rectangular pieces of metal, small turbine blades, and standard stress-rupture test specimens which were kept at 1350°F and 15,000 psi stress. Tests were made mainly in the velocity range between 500 and 850 ft/sec.

It was found that erosion when using the dust separator was only ten percent of that with the raw ash for the same weight concentration of fly ash per unit volume of gas. The Aerotec separator removed about 90 percent of the raw ash. Therefore, erosion with the separator was only about 1% of that with raw ash. A persistent deposit of ash particles was found on the targets when operating with the ash separator. In general, it was found that deposit formation was associated with the impingement of ash particles smaller than about 10 microns, while large particles tended to remove deposits and produce erosion of the metals. The amount of deposit increased with increasing temperature. The ash particles tended to sinter into a strongly adhering layer even at temperatures as low as 800°F.

Dust Erosion in Small Gas Turbines

In 1960 the Southwest Research Institute made a study of the problem of atmospheric dust erosion of small gas turbines under the sponsorship of the Corps of Engineers.[16,17] Tests were made to determine the effect of particle size and dust concentration. The results of these investigations are shown in Figure M-7. The gas turbine life varies inversely as the product of the maximum particle size and the total dust concentration. It was concluded that all particles greater than 2 μ would have to be removed. The engine life levels shown are probably not applicable to large axial flow rates, but the basic relations probably apply.

The Army Material Command sponsored an investigation at Southwest Research Institute on the basic mechanisms of dust erosions.[17] The following conclusions were made:

Curve 646941-A

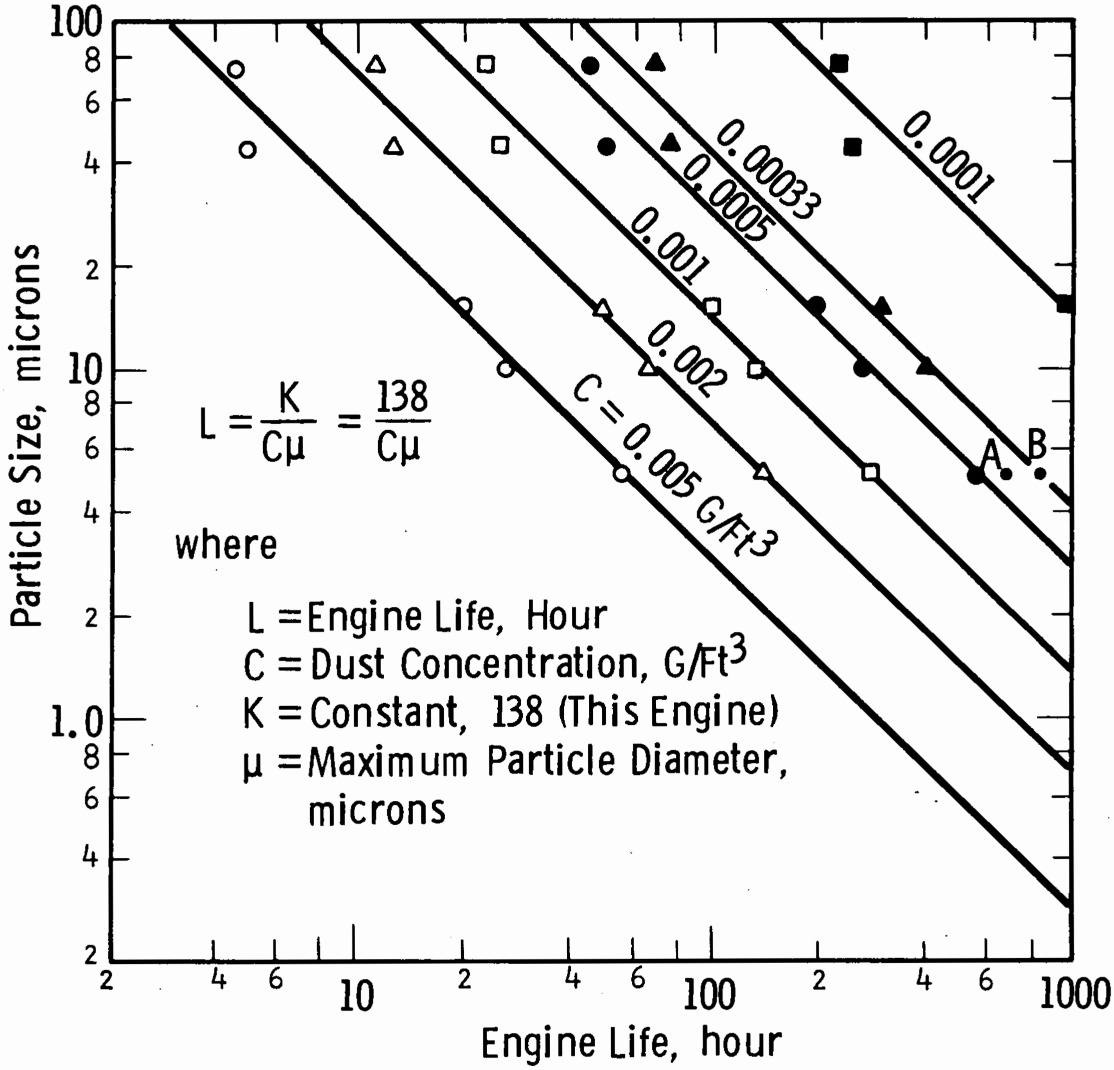

Fig. M-7 — Particle size vs usable safe engine life (max. impeller wt. loss of 21 gr.) (16)

1. Erosion by airborne dust is primarily a process of plastic displacement of material by the impacting dust particle, and the effects of dust particle velocity and angle of impact upon erosion rates may be described adequately by equations defining the trajectory of the particle through the material.

2. The relationship between dust concentration and erosion loss is not a linear one, the average erosion loss per impact being greater at low dust concentrations than at the higher concentrations.

3. Erosion rates are a strong function of dust-particle size, the correlation being best described by consideration of the average erosion loss per particle impact.

4. The effect of material properties on dust-erosion loss must include consideration of local temperatures resulting from particle impacts, as well as the relationship of depth of cut to the mean distance between crystal imperfections.

In conclusion, the principal parameters in the erosion of turbine blades by a solid particle are: the kinetic energy of the particle, the angle of impact, and the physical properties of both materials.

REFERENCES

1. Hoy, H. R., Paper presented at Second International Conference on Fluidized Combustion, Hueston Woods, Ohio, October 1970.

2. National Coal Board, reports prepared for APCO, August and November 1970, February 1971.

3. Stettenberg, L. M., "Minimizing Erosion and Afterburn in the Power Recovery Gas Turbine," Oil and Gas Journal, October 19, 1970.

4. Mund, M. G. and H. Guhne, "Gas Turbines--Dust-Air Cleaner: Experience and Trends," A.S.M.E. Paper 70-GT-104.

5. Jaumotte, A. L. and J. Hustin, "Experience Gained from a Ten-Year Operation of a Gas Turbine Working With Blast Furnace Gas," A.S.M.E. Paper 66-GT-97.

6. "U.S. Bureau of Mines Coal-Fired Gas Turbine Research Project," U.S. Bureau of Mines Report RI6920-1967.

7. "Turbine Blade Wear by Coal Ash in Working Fluid at 1200°F," U.S. Bureau of Mines Report RI7255-1969.

8. Morley, W. J. and J. C. Wisdom, "Brown Coal Ash Deposition in the Open-Cycle Gas Turbine," Journal of the Institute of Fuel, May 1964, pp. 187-200.

9. Atkin, M. L., "Australian Coal-Burning Unit," Gas Turbine International, September-October 1969, pp. 32-36.

10. Finnie, Iain, "Erosion by Solid Particles in a Fluid Stream," Symposium on Erosion and Cavitation, ASTM Spec. Tech. Pub. No. 307, 1962, p. 70.

11. Martlew, L., "The Distribution of Impacted Particles of Various Sizes on the Blades of a Turbine Cascade", Proc. Conf. BCURA, 1960, E. G. Richardson, Editor, Pergamon Press, London, 1960, p. 104.

12. Wisdom, J. C., "The Prospects of the Coal Burning Gas Turbine for Electric Power Generation," World Power Conf., Sept. 13-17, 1964, Switzerland, Paper No. 8.

13. Dygert, J. C. and G. S. Bjorklund, "Integration of the Power Recovery Gas Turbine with Fluid Bed Processes", ASME, Paper No. 59-A-231.

14. Fisher, M. A. and E. F. Davis, "Studies of Fly Ash Erosion", ASME Paper No. 48-A-53, December, 1948.

15. Report on Trip to U.S. Bureau of Mines, Morgantown, W. Va., January 5, 1965, Hamm, Wolfe, and Young.

16. "Dust Erosion Parameters for a Gas Turbine", Petro/Chem Engineering, December 1962, p. 198.

17. Horton, J. H., "Environmental Factors in Engine Design for Military Applications", ASME Paper 65-CTP-1, December 28, 1964.

APPENDIX

CALCULATED FREEZING POINT CURVES

Westinghouse has built a micro-fusion apparatus which proved to be useful as a rapid and convenient means for evaluating fuel additives and can be used to evaluate the carryover from a fluidized bed system. However, early attempts to find a satisfactory additive for a fuel with a high sodium concentration showed that what was really required was a fairly complete set of phase diagrams of binary mixtures for the different ash constituents and various proposed additives. As may be seen from the melting point diagram for Na_2SO_4 - $MgSO_4$ mixtures in Figure M-7 in the text, when these two sulphates were mixed with a magnesium-sodium ratio of 0.5 the temperature of the complete melting point was lower than that of either of the pure compounds. In a few instances such diagrams were found in the literature,* but for many of the more promising additives no diagrams were available. However, using thermochemical data it is possible to calculate ideal melting point curves for binary mixtures which have been of some use in selecting ranges of fuel-additive compositions which should result in solid ash products at turbine operating temperatures.

Theoretical melting point curves for binary systems can be computed from the thermodynamic properties of the components. Such a computation is based on the lowering of the freezing point of a solution as known amounts of solute are added to the solvent. Rossini** considers the case of equilibrium between a pure solid substance and an ideal liquid solution containing the pure solid in dissolved form with the following simplifying assumptions:

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*Ginsberg, A. S., Proceedings Polytechnic Institute of St. Petersburg, Vol. 6, p. 493 (1906).

**Rossini, F. D., "Chemical Thermodynamics," John Wiley and Sons, Inc. 1950.

(a) The mixtures are true binaries

(b) They behave as ideal solutions over the entire range of composition

(c) The solutes are completely soluble in the molten state and completely insoluble in the solid state

(d) They form no intermediate compounds.

The mole fraction of one component, N_A, will vary with the solution melting point temperature, T, and may be found from either of the two expressions:

$$\ln N_A = -\frac{\Delta H^*_{MA}}{R}\left[\frac{1}{T} - \frac{1}{T^*_A}\right]$$

or

$$\ln N_A = -\frac{\Delta H^*_{MA}}{R}\left[\frac{1}{T} - \frac{1}{T^*_A}\right] + \frac{(C^*_A) - \langle C^*_A \rangle}{R}\left[\ln\frac{T}{T^*_A} + \left(\frac{T^*_A - T}{T}\right)\right]$$

In these expressions the symbols are defined as follows:

N_A = Mole fraction of the pure substance, A.

ΔH^*_{MA} = Heat of fusion of pure substance, A, at T^*_A, cal./mole.

R = Universal gas constant, cal./mole °C.

T^*_A = Fusion temperature of pure substance, A, °K.

T = Temperature of complete melting of solution, °K.

(C^*_A) = Specific heat at constant pressure of molten pure substance, A, at the fusion point, cal./mole °C.

$\langle C^*_A \rangle$ = Specific heat at constant pressure of pure substance, A, in the solid state at T^*_A, cal./mole °C. This may be expressed in terms of the temperature as $\langle C_A \rangle = a + bT + cT^2 + e/T^2$, where a, b, c and e are constants characteristic of the substance.

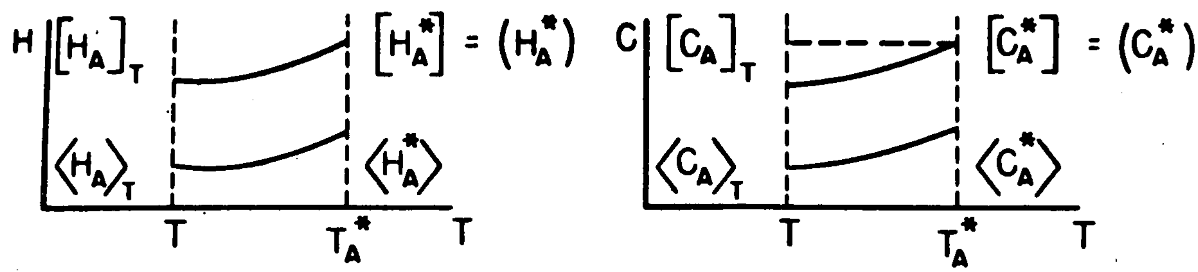
( ) = Indicates the liquid state.

$\langle\ \rangle$ = Indicates the solid state.

[ ] = Indicates dissolved substance.

The first of these expressions assumes that the heat of fusion is constant for all concentrations of "A" and is equal to the value at the melting point of the pure component as shown in Figure M-8, Assumption I.

DWG. 189A922



Assumption I

$$\Delta H_{MA} = \text{Constant} = \Delta H_{MA}^*$$

or

$$[H_A]_T - \langle H_A \rangle_T = (H_A^*) - \langle H_A^* \rangle$$

Assumption II

$$[C_A]_T - \langle C_A \rangle_T = \text{Constant} = (C_A^*) - \langle C_A^* \rangle$$

Fig. M-8

The second includes a correction which takes into account the difference in specific heats between the solid and liquid phases of the pure solvent as shown in Assumption II.

Further refinement is possible in the form of a third assumption that the specific heat of "A" in the dissolved state is constant over the range of composition (dashed line, Assumption II) while the specific heat in the undissolved solid state may be expressed in terms of the temperature. This results in a more complicated form of expression than II but produces insignificant changes in the calculated values of N_A.

The first of the above expressions is intended to be used only where the solution approaches pure "A" in composition. A further extension of the range of composition is possible using the second expression, but even in this case its application to actual fuel ashes would yield approximate values at best, since such ash mixtures may not form ideal solutions. In one case where experimental data were available, for the Na_2SO_4 - $MgSO_4$ system, good agreement was found between such data and calculated values obtained by means of the second equation, as shown in Figure M-7. It was assumed that no intermediate compounds form. If such compounds do exist, the experimental curve departs from calculated values as shown in the 0.45 to 0.7 range of N_A. Assuming that these expressions are valid, it is apparent that on a mole basis, the lowering of the fusion point of substance "A" depends on the characteristics of "A" and the quantity of added substance "B". When the fusion point lowerings of two substances are calculated, and the curves plotted from opposite sides of the diagram, they will intersect at a point shown in the curve. This will represent the lowest temperature at which any liquid can appear and is defined as the eutectic temperature. In the above formulas, all values are available except the specific heat of the melted material, (C_A^*), which has seldom been determined for the high melting point compounds such as MgO. Also, the specific heat of pure solid $\langle C_A^* \rangle$ may be in error, since the range of temperatures covered by the equation do not extend to the fusion points of the compounds.

An estimate of (C_A^*) may be obtained as follows: Using the first equation, the melting point diagram may be plotted for the compound in question. From published phase diagrams,[*,Δ] several binaries may generally be found which contain the compound in question as one of the components. Eutectic temperatures and the corresponding concentrations from such diagrams may be plotted on the calculated diagram. In general, these points will lie below the calculated curves, indicating the necessity for the specific heat correction. Eutectics should be chosen for relatively high concentrations to lessen the possibility of encountering an intermediate compound, and no eutectic containing such a compound should be used. Several points are desirable in order to minimize the errors prevalent in any physical test. A segment of the curve most nearly satisfying these points is then drawn from the fusion point of pure compound and a convenient value of molar concentration and corresponding temperature selected from this segment. These values may then be inserted in the second equation and a value for (C_A^*) computed. With this value for (C_A^*), the second equation may now be evaluated throughout the required temperature range.

The calculation of specific heats by such a method implies that the mixtures of inorganic salts for which freezing point data were available are ideal solutions, and it should be noted that such calculated values are only first approximations. However, the incorporation of the experimental points may partially compensate for any departure from ideal solutions and variation in solubilities. Here again the correction must be considered only approximate since these characteristics probably vary throughout the range of composition. This procedure, incidentally, might be proposed as a method of obtaining approximate specific heats at the fusion points of high melting point compounds where physical measurements would be extremely difficult. Only observed melting points of mixtures of the compounds are necessary.

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* International Critical Tables.

Δ Hall, F. P., Insley, H., "Phase Diagrams for Ceramists," American Ceramic Society, Inc., 1947.

An uncorrected calculation for magnesium oxide obtained using the first equation is shown in Figure M-9, and three experimental points are also shown. From the broken curve fitting these points, a value of $N_A = 0.5$ at 2190°C was taken, and using the second equation, $(C_A^*) = 18.86$ cal./mole°C was obtained. A continuation of the sample calculation using the second equation with the value of (C_A^*) obtained above is also plotted on the curve. The calculated curve for magnesium sulphate is also shown. Here, no correction was made for the change in specific heat, since the several test points fell exactly on the curve.

Figure M-10 shows composites of all the oxides and sulphates examined, together with sodium sulphate. The intersection of each of these curves plotted from the right-hand side of the diagram with the sulphate plotted from the left represents the theoretical eutectic point. From such curves, it appears that variations in the amount of an additive should have little effect on the sintering or deposit-forming characteristics of an ash, since this probably depends on the eutectic temperature which, in turn, is dependent on the composition. At most, an additive can only raise the sinter point to a temperature which approaches the fusion temperature of the corrosive compound. This is evident from the intersections for aluminum and magnesium oxides. However, the quantity of eutectic material may be so small that no liquid is evident when the ash reaches the eutectic temperature. Of course, if compounds containing elements from the ash and additives are formed, these calculated curves give no indication as to the actual curves.

Ash fusion tests have shown magnesium additives to be quite ineffective in raising the sintering temperature whereas aluminum produces noticeable increases, indicating the possible formation of solid solution. In practice, both aluminum and silicon additives have shown less tendency to produce deposits than magnesium. Briefly, it may be concluded, on the basis of the theoretical data, that magnesium and calcium would be excellent additives if they were present as oxides.

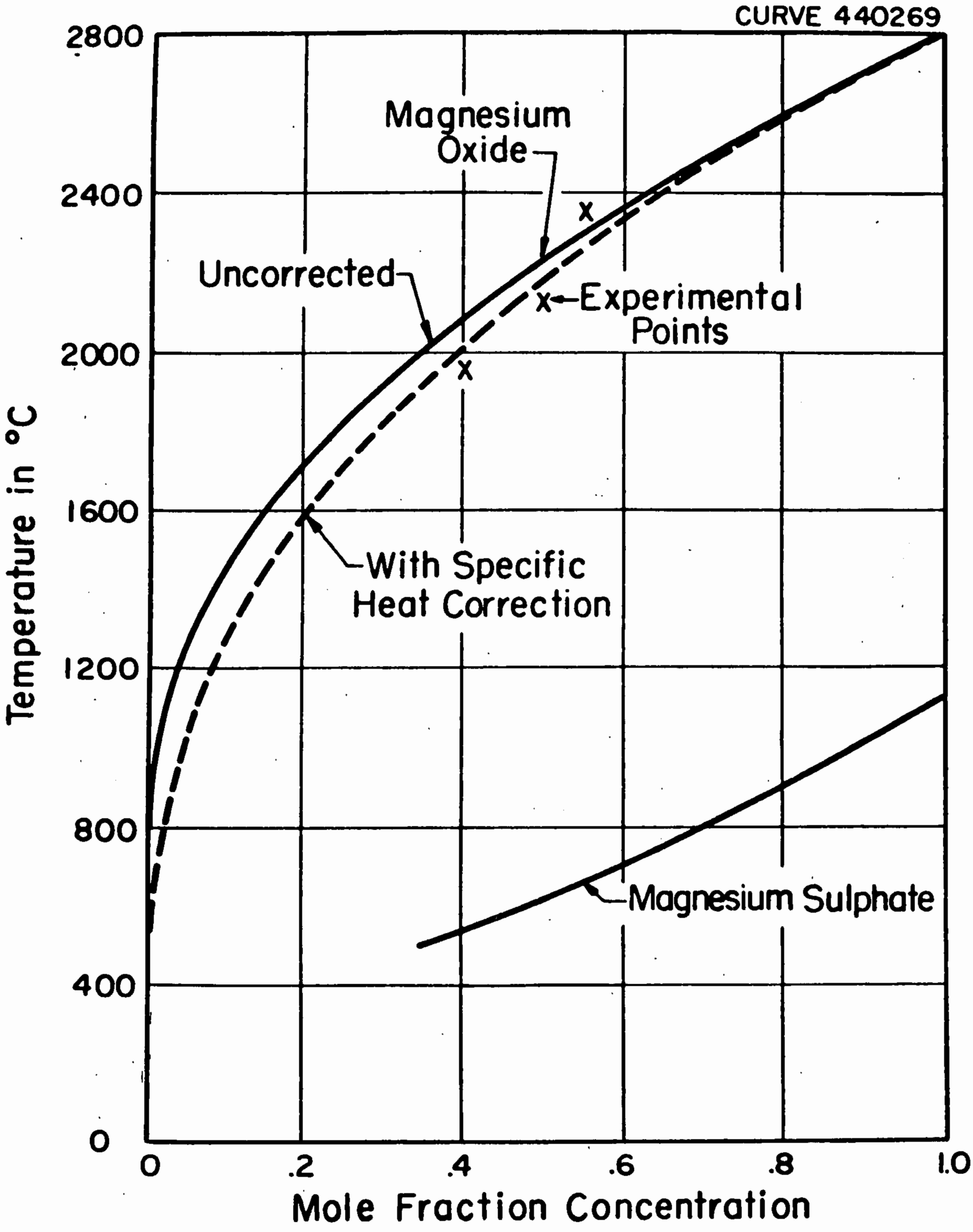


Fig. M-9—Calculated freezing points of magnesium compounds

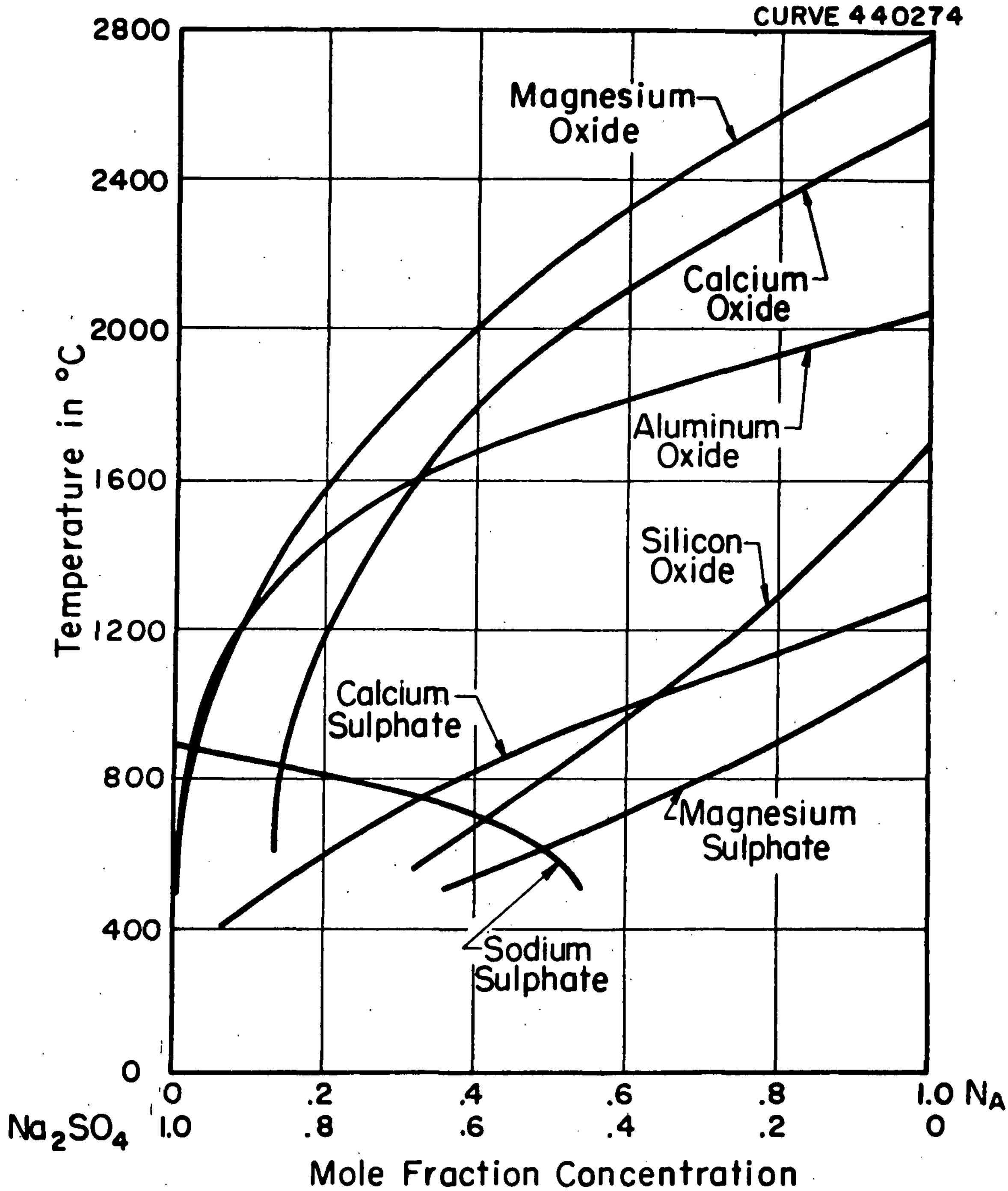


Fig. M-10—Calculated freezing point diagrams with sodium sulphate

However, as sulphates, they are undesirable. Although silicon oxide has a lower melting point than either magnesium or calcium oxide, it would appear to have an advantage in not forming a sulphate. Aluminum is, perhaps, the best compromise. While its oxide has a somewhat lower melting point than that of either magnesium or calcium, it can be shown that its sulphate, although stable at some conditions, reverts to the oxide at a relatively low temperature. A calculated curve for the magnesium-calcium sulfate system is shown in Figure M-11. This indicates a eutectic temperature slightly less than 800°C. But it must be emphasized that this curve is not confirmed. It has not been located in the literature, and no fusion test has yet been run in our laboratory.

Curve 646942-A

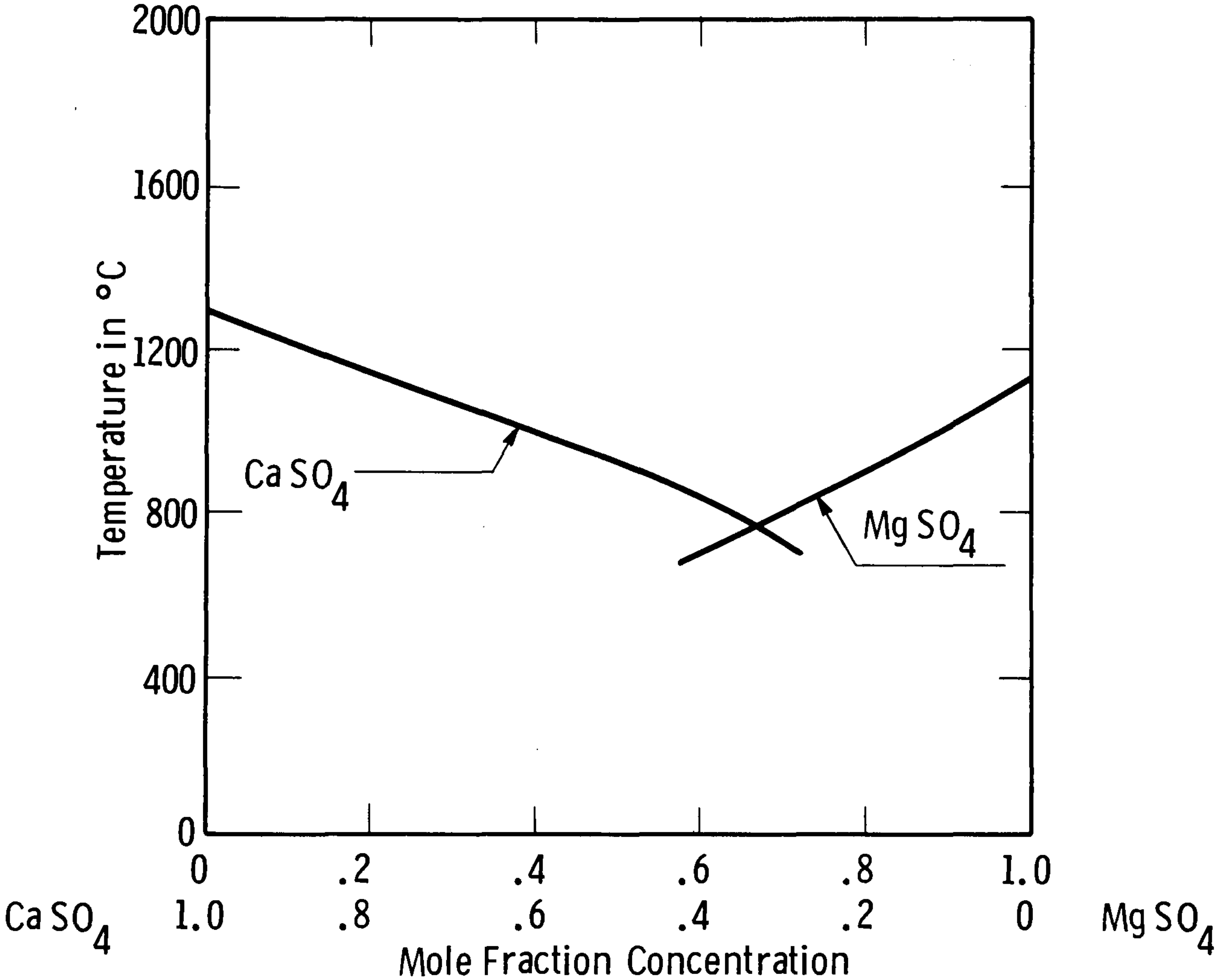


Fig. M-11 Calculated freezing point diagram for magnesium and calcium sulfates

APPENDIX N

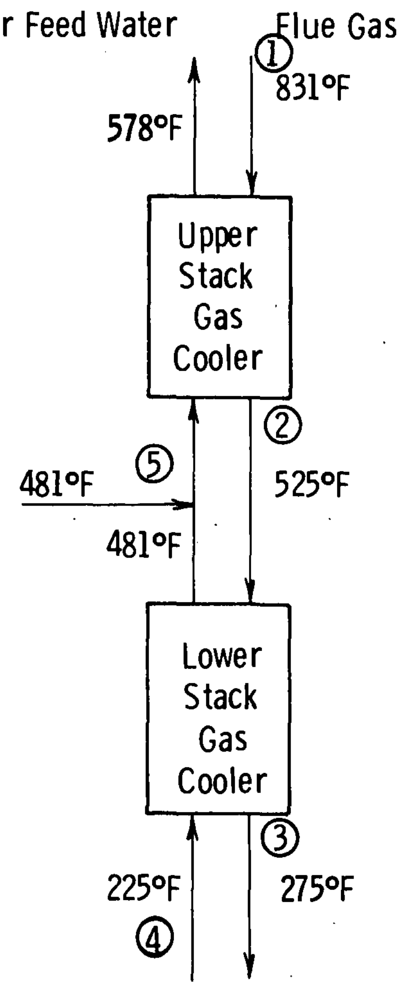
STACK GAS COOLER DESIGN

ABSTRACT

Design specifications for the stack gas coolers that recover heat from the exhaust gas from the gas turbine are summarized in Figure N-1. Boiler feedwater is heated to 578°F in the heat recovery system. Two quotations which were received for the stack gas coolers are presented. The Westinghouse Heat Transfer Division submitted a design and cost. The design is modular, and the cost for both a 318 MW and 635 MW plant is $4.1/kw, which includes transition ducting with the gas turbines. Struthers Nuclear also submitted a quotation. Their estimate is $3.7/kw for the 635 MW plant and does not include ducts and connection with the gas turbines.

Dwg. 6169A81

Fig. N-1—Design specifications for stack gas coolers



Flue Gas Composition:

	Mole%
N_2	74.4
CO_2	14.5
H_2O	8.5
O_2	2.4
CO	0.2
SO_2	~ 160 ppm
NO_x	< 400 ppm
	100.0%

Solids:
< 0.15 gr/SCF

Ash	79.7 wt %
Carbon	6.6
$CaO/CaSO_4$	13.7
	100.0

	319 MW Case I	635 MW Case II
Gas Flow:	707 lb/sec	1414 lb/sec
Water Flow:		
4	628,500 lb/sec	1,257,000 lb/hr
5	1,727,000 lb/hr	3,454,000 lb/hr
Gas Side Pressure Drop:	(State 3 ~ Atmospheric)	
$P_1 - P_2$	0.29 psi	0.29 psi
$P_2 - P_3$	0.28 psi	0.28 psi
Water Side Pressure:	— ~ 2800 psi	———

From : HEAT TRANSFER DIVISION
WIN : 325-2044
Date : April 23, 1971
Subject : Stack Gas Coolers for P. F. B. Power System

Pittsburgh R&D Center
Mr. Dale L. Keairns

Design of 319 MW Stack Gas Cooler

Exhaust gases from the W-501 B gas turbine enter the equipment as follows:

1. Inlet transition duct -- Exhaust gases expand from 12' x 12' Gas Turbine exhaust flange to 22' wide x 14' high in a length of 10'.

   Ductwork is ¼" C.S. braced and ribbed. It is insulated with a 2½" block and lagged with 16 gage C.S.

2. The gases then enter an inlet plenum which turns the gas from the horizontal to the vertical direction. Turning vanes, formed from incoloy (approx. 11% Cr) are used to evenly distribute the GT exhaust.

3. Following the inlet plenum the gases enter the heat exchanger sections.

On the following page is a sketch depicting its design.

4. The exhaust gases then pass through the transition ductwork which is not insulated. The unit is completely supported with structural steel able to withstand nominal wind loadings.

Total estimating price for 1971 delivery .... $1,300,000.
f.o.b. Lester.

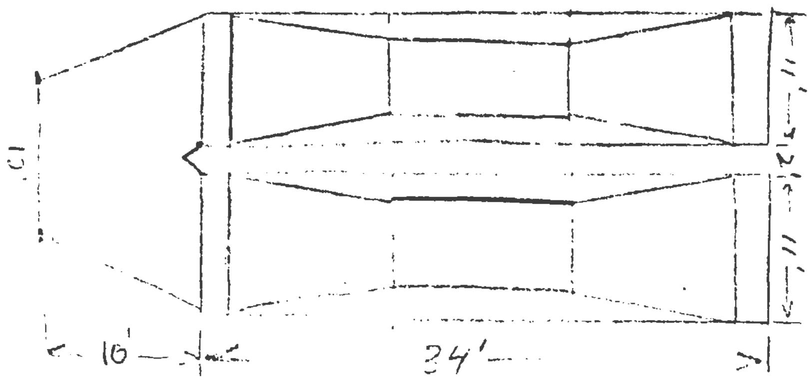
S C Jamison
S. C. Jamison

SCJ/nmm

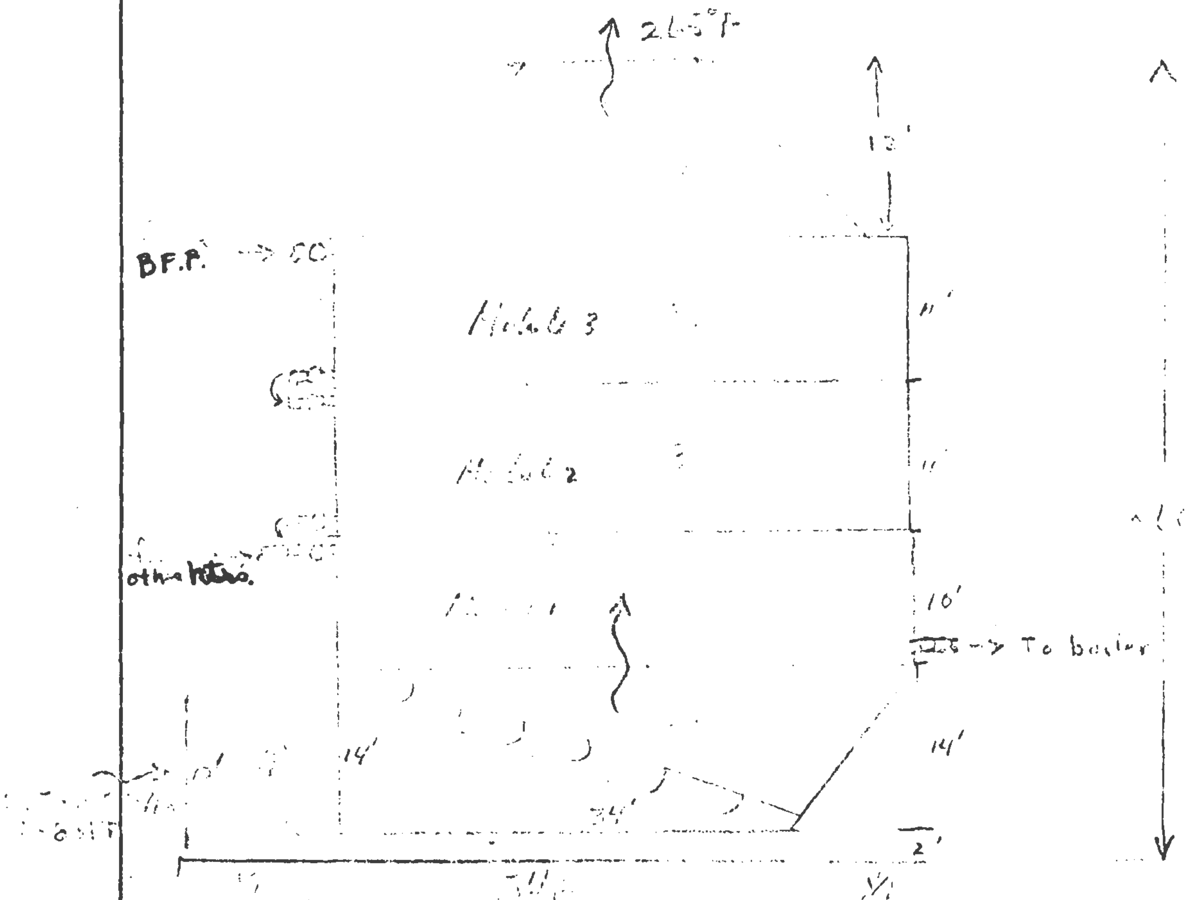
WESTINGHOUSE FORM 2444D

WESTINGHOUSE ELECTRIC CORPORATION

319 MW



- Unit has two external headers (shown below) which join module to module
- Internal headers connect to tubes (see next pg.)



NOTE - STRUCTURAL STEEL NOT SHOWN

WESTINGHOUSE ELECTRIC CORPORATION

Design Data per 319 MW. plant.

Modules #1

shells 3/4" C.S plate
insulated with 2 1/2" Thermobestos block - C.S lagging .060" thk.

headers, 6, each 10' long (4 internal, 2 external)
8" sch 120 A106 Gr C.

18 rows vertical
40 rows horizontal (21 rows long 1/module)

tubes 3.0" OD, .313 min wall, A210-A1, 30' long
fins 1.0" high, .0475" thick, .156" sq width.

Modules #2 & 3

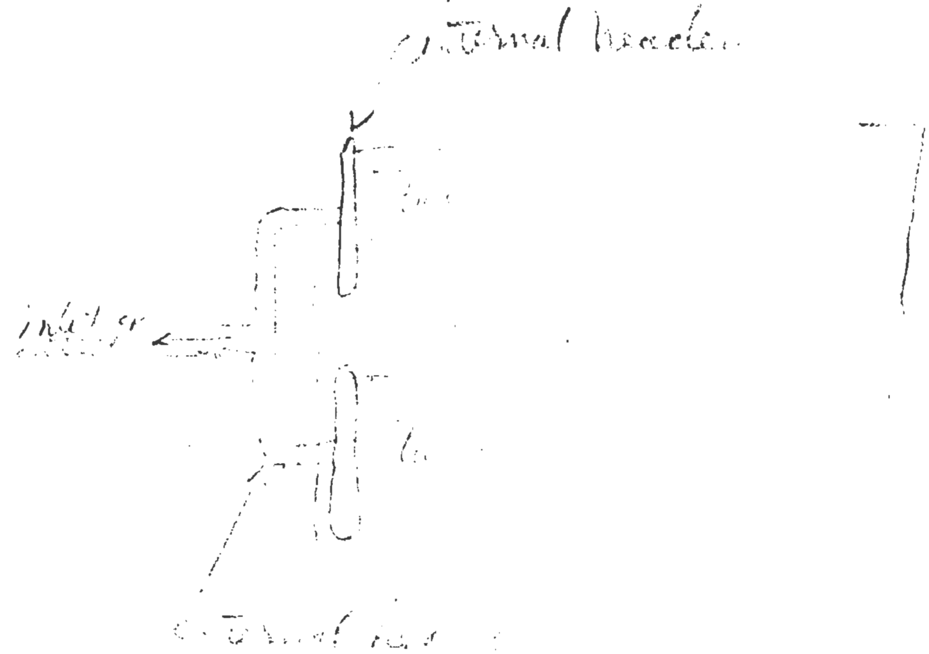
shells 3/4" C.S. plate
insulated with 2 1/2" Thermobestos block
lagged with .060 thick C.S.

Headers 12, each 10' long (8 internal, 4 external)
8" Sch 120 A 106 GR C.

53 Rows vertical
52 Rows horizontal (26 rows long/module)

tubes 30' long, 2.0" OD, .259 min wall A192
fins 1.0" high, .0478 thick, .156 sq width

Price of $1,300,000. does not include any control, valves or protective systems header arrang.



MEMORANDUM
WESTINGHOUSE FORM 2478 K

TO Pittsburg R&D Center (LOCATION) (DEPT.) Dale Keairns (NAME) 4/2/71 (DATE)

SUBJECT STACK GAS COOLERS ~ 319 MW

ATTACHED IS A MARKED UP DWG SHOWING THE APPROX. DIMEN. FOR THE SUBJECT UNIT.

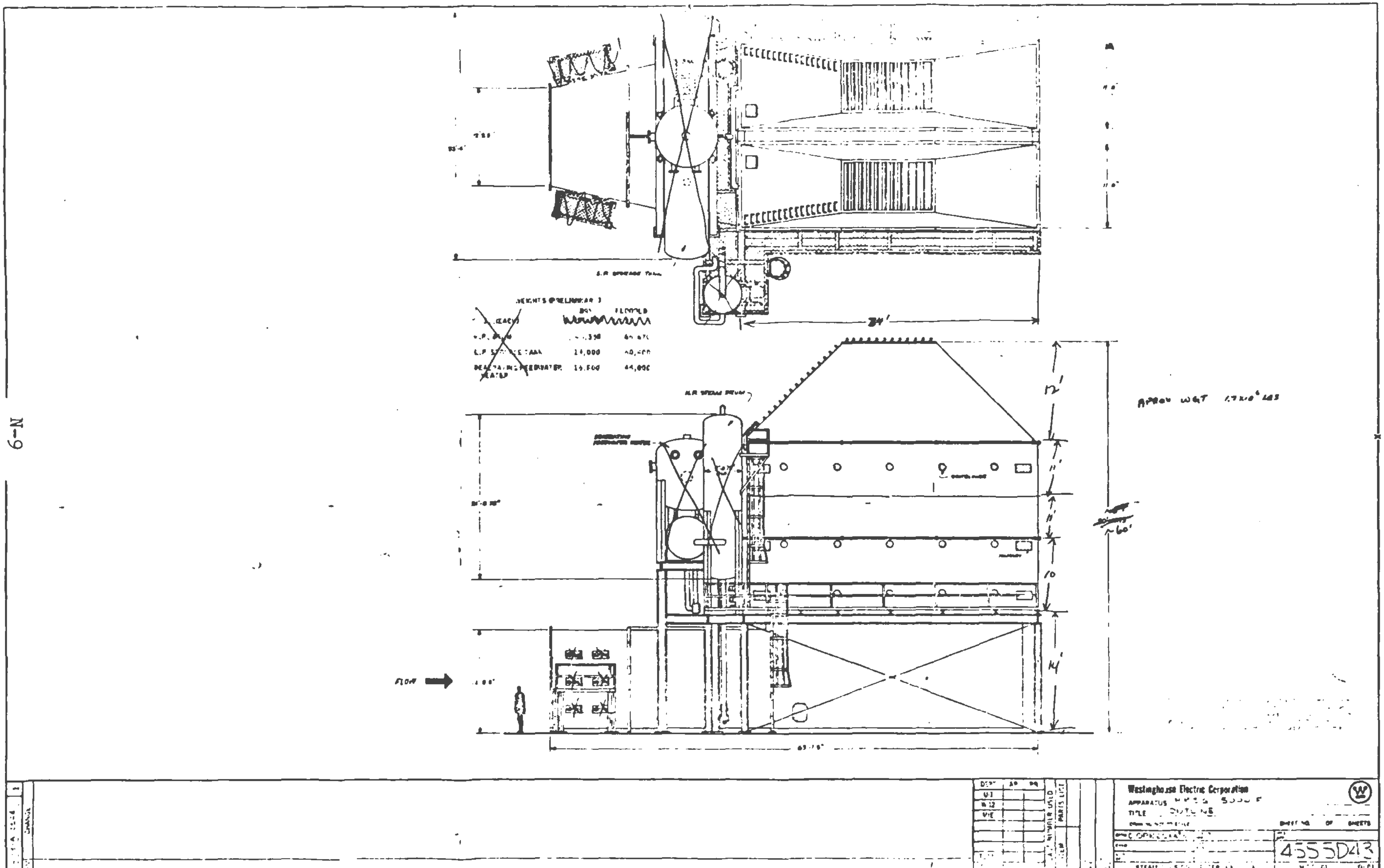
NOTICE THAT I HAVE CROSSED OUT CERTAIN ITEMS — THESE DO NOT APPLY TO THE DESIGN

I HAVE SENT UPDATED INFO TO COST EST. AND WILL SEND YOU PRICES AND SCOPE OF MATERIAL WHEN AVAILABLE

— DESIGN IS THREE MODULES HIGH X TWO (2) MODULES WIDE (501 GT)

— DESIGN FOR 251 GT WOULD BE THREE MODULES HIGH ONLY

FROM HTD SALES (LOCATION) LESTER, PA (DEPT.) N-8 (NAME) S. JAMISON 2044



34'
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FLOW
Westinghouse Electric Corporation
APPARATUS
TITLE
SHEET NO. OF SHEETS
4555D43

Westinghouse Electric Corporation
Research and Development Center
Beulah Road
Pittsburgh, Pennsylvania 15235

RE: Our Proposal 3492-W3

Struthers

Struthers Nuclear & Process Company
division of STRUTHERS WELLS CORPORATION

P. O. BOX 8 • WARREN, PENNSYLVANIA 16365 • 814/726 1000

February 11, 1971

Westinghouse Electric Corporation
Research and Development Center
Beulah Road
Pittsburgh, Pennsylvania 15235

ATTENTION: Mr. Dale L. Keairns
Senior Engineer
Chemical Engineering Research

SUBJECT: Stack Gas Coolers for
Pressurized Fluid Bed Power System
Our Proposal 3492-W3

Gentlemen:

We are pleased to submit our engineering estimate for the equipment as required on the above subject.

The waterside of this equipment would be built in accordance with Section VIII of the ASME Pressure Vessel Code and would be so stamped. The gas side would be code constructed but would not be stamped.

For this service we propose to furnish serpentine coil finned tube bundles. We believe the coils should be arranged for down flow of the flue gas, and soot blowers would probably want to be installed. The dust loading of the gas as given (0.15 grains/SCF) would indicate that some 715 #/hour of dust would be passing through the units. We cannot predict what amount of this dust would be retained by the finned tubes, if any, and would suggest a small scale experimental model be used to determine the affinity of the dust for the tubes. We, therefore, have not included any soot blowers or related dust handling or collecting equipment in our proposal.

<u>Design 319 MW Case I</u>
<u>Upper Stack Gas Cooler</u>

We designate this coil as our 1.5" VC 140-10-25. This serpentine coil would be made up using 1½" I.P.S. Sch. 80 SA-106B pipes, spaced on 4½" triangular pitch, using long radius return bends. The coil face would be 140 tubes wide and each would be finned for a distance of 10 feet. The coil would be 25 rows deep. Each tube would contain 5 - ½" high x .06" thick fins per inch. The inlet and outlet header size would be 17-3/8" O.D. x 12-7/8" I.D. x 53 ft. long. Each header would contain 3 - 1500# R.F. feedwater nozzles. One end of each header would be capped and the other would contain a special high pressure end closure. Each of the 140 - 1½" I.P.S. pipes would be connected to the headers thus providing 140 parallel passes on the waterside. The coil would be completely drainable with this arrangement. One tube support plate would be provided at each end of the finned section.

The coil would be enclosed within a #10 Ga steel casing, suitably stiffened to withstand a design pressure of 12" W.C. The casing and coil would be provided with structural steel supports designed to carry its own weight. The casing would be internally insulated with one inch of block insulation. The insulation would be protected from gas erosion by a #18 Ga carbon steel liner.

The unit would be painted with one coat of our standard shop primer.

<u>Design 319 MW Case I</u>
<u>Lower Stack Gas Cooler</u>

Our designation for this coil would be a 1.5 VC 140-14-59. It differs from the upper stack gas cooler only in the finned length of the pipes and the number of rows deep. This coil would use 14'-0" finned length with 59 rows deep.

The inlet and outlet headers size would be 12-5/8" O.D. x 9-1/4" I.D. x 53 ft. long. Each header would contain 3 - 6" 1500# R.F. inlet and outlet nozzles.

One half of the 140 tubes would be connected to the headers, thus providing 70 parallel passes on the tubeside. The coil would be completely drainable.

All other remarks as given under the Upper Stack Gas Cooler would apply to this unit also.

The coil design as quoted exceeds your allowable gas pressure drop by 2.0" W.C.

Design 635 MW Case II
Upper Stack Gas Cooler

We designate this coil as our 1½ VC 140-14-36. This serpentine coil would be made up using 1½" I.P.S. Sch. 80, SA-106-B pipes spaced on 6" triangular pitch using extra long radius return bends. The coil face would be 140 tubes wide and each would be finned for a distance of 14 feet. The coil would be 36 rows deep. Each tube would contain 5 - ½" high x .06" thick fins per inch. The inlet and outlet header size would be 25-1/4" O.D. x 19-1/8" I.D. x 70'-0" long. Each header would contain 3 - 12" x 1500# R.F. inlet and outlet nozzles.

Two rows of tubes would be connected to each header which in effect produces 280 parallel passes. The coil would be completely drainable. Three tube support plates would be provided for the coil; one at each end and one in the center of the finned section.

Casing, insulation and structural would be provided as described under Case I.

The coil design as quoted exceeds your allowable gas pressure drop by 1.5" W.C.

Design 635 MW Case II
Lower Stack Gas Cooler

Our designation for this coil would be a 1.5 VC 140-20-92. It differs from the upper stack gas cooler only in the finned length and number of rows deep. This coil would use 20'-0" finned length with 92 rows deep.

The inlet and outlet header size would be 17-1/8" O.D. x 12-7/8" I.D. x 70'-0" long. Each header would contain 3 - 8" 1500# R.F. inlet and outlet nozzles.

Two rows of tubes would be connected to each header which in effect produces 280 parallel passes on the waterside. The coil would be completely drainable. Three tube support plates would be provided for the coil; one at each end and one in the center of the finned section.

Casing, insulation and structural would be provided as described under Case I.

The coil design as quoted exceeds your allowable gas pressure drop by 2.0" W.C.

<u>Budget Price</u>
<u>319 MW Case I</u>

For one upper and one lower stack gas cooler as described, F.O.B. Warren, Pennsylvania, $880,000.

Estimated shipping weight of both coils 1,250,000 pounds.

<u>Budget Price</u>
<u>635 MW Case II</u>

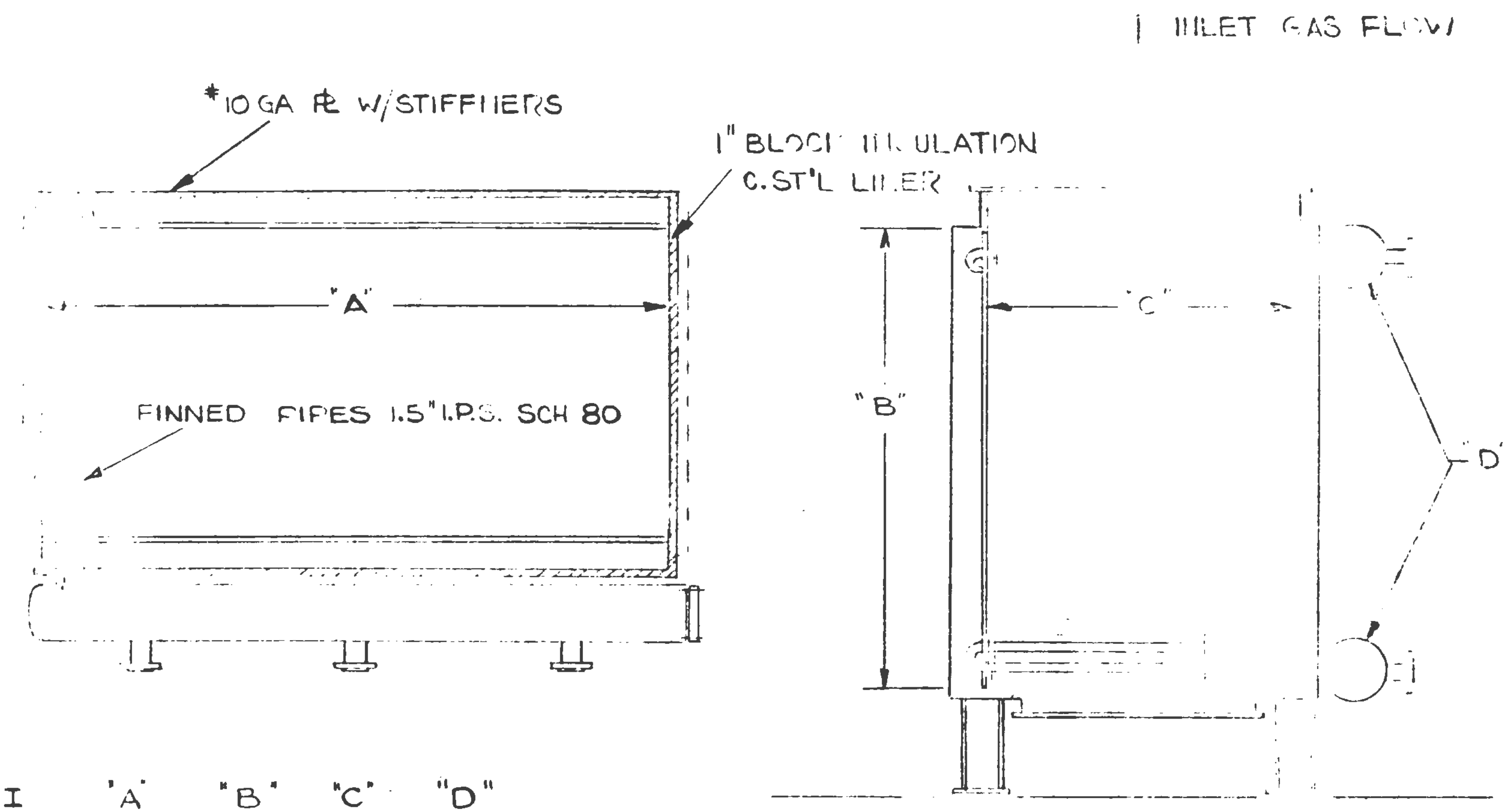
For one upper and one lower stack gas cooler as described, F.O.B. Warren, Pennsylvania, $2,370,000.

Estimated shipping weight of both coils 3,250,000 pounds.

Struthers would require that progress payments be made on any order of this magnitude.

We would estimate delivery of this equipment could be made in 40 to 50 weeks after receipt of the approved setting plan drawing to be submitted 4 weeks after receipt of the formal order.

We thank you for this opportunity to quote on your requirements and trust that we may serve you further in this connection.



N-16

CASE I	"A"	"B"	"C"	"D"
UPPER	53'	12'	10'	$17\frac{3}{8}$" O.D.
LOWER	53'	23'	14'	$12\frac{5}{8}$" O.D.
CASE II				
UPPER	70'	18'	14'	$25\frac{1}{2}$" O.D.
LOWER	70'	48'	20'	17[illegible]" O.D.

STRUTHERS

STRUTHERS NUCLEAR & PROCESS CO.
WARREN, PENNSYLVANIA
DIVISION OF STRUTHERS WELLS CORP.

DRAWN:		SCALE:	HIGH PRESSURE FEEDWATER EC[illegible]	DRAWING & CONTRACT NO.	REV.
CHECKED:		NONE		[illegible]	
APPROVED:					

Terms of Sale

1. CONTRACT Seller's proposal previous to acceptance is subject to change or withdrawal without notice. Purchaser's order, accepting with or without specific modifications the descriptions, specifications, terms and conditions given in Seller's proposal, when approved and accepted on the standard form of Seller's acceptance by an authorized employee of the Seller at one of its plants constitutes the entire contract, which then may be modified only by written agreement, approved by such an authorized employee and no other promises, agreements or understandings shall be binding on the Seller.

2. TAXES If any sales or use taxes or other taxes, whether now in effect or hereafter enacted applicable to the equipment or any performance in regard thereto are imposed, such taxes shall be for the Purchaser's Account.

3. PAYMENT (a) Payments are to be made in current funds of the United States to Seller at Seller's plant.

(b) Payments for each invoice covering partial shipments shall become due in accordance with the quoted terms of payment.

(c) If Seller shall be delayed in starting, manufacture, shipment or other phase of the contract by action or inaction of Purchaser, payment shall be due as if Seller had shipped the equipment as called for in the contract, and when so delayed Seller may store the equipment for Purchaser's account and risk.

4. GENERAL PROVISIONS (a) Right of possession to goods to secure the payment of the purchase price shall remain in Seller until all payments therefor hereunder shall have been fully made. Upon request Purchaser agrees to do all acts necessary to perfect and maintain such right to the Seller.

(b) Seller shall be excused for reasonable delay in performance or for non-performance due to any cause beyond its control, including but not limited to shortages of materials or manpower.

(c) If Purchaser desires to inspect goods for workmanship and material, inspection and acceptance must be made before shipment, unless otherwise agreed in writing.

(d) Shipments and deliveries under this agreement shall at all times be subject to approval of Seller's Credit Department.

(e) Receipt of goods by Purchaser without objection shall constitute a waiver of any and all claims for delay.

(f) Unless otherwise agreed in writing Purchaser and Carrier are responsible for goods lost or damaged in transit.

5. QUALITY, PERFORMANCE AND LIMITATION OF LIABILITY (a) Seller warrants the equipment against defects in material and workmanship, under normal use and service, for a period of one year after date of shipment of the equipment: Seller's obligation under this Warranty being limited, however, to furnishing or repairing, without charge, F.O.B., its Plants, a similar part to replace any part of its own equipment which not more than one year after date of shipment of the equipment is proven to have been defective at the time it was shipped, provided Purchaser has given Seller immediate written notice upon discovering such defect. Seller shall have the option of requiring the return of the defective material, to establish the claim. Seller shall not be held responsible for work done, apparatus furnished or repairs made by others unless done with Seller's written approval.

(b) Seller assumes no responsibility for the effects of corrosion or erosion.

(c) Seller shall not be liable for any special, indirect or consequential damages resulting in any manner from the furnishing of the equipment. In special cases Seller's liability may be further defined previous to the acceptance of the order by mutual agreement in writing between the parties hereto.

6. CANCELLATION Cancellation of orders accepted by the Seller can be made only with the Seller's consent. Should cancellation be accepted by the Seller, the Purchaser shall pay the full purchase price for items completed. On items not completed, a charge will be made for incurred material and labor costs together with material handling, manufacturing, sales, engineering and administrative overhead plus a reasonable percentage of profit. Purchaser shall also pay in full the cost of all special dies, tools, patterns and fixtures, to all of which at all times possession and title remain in Seller unless otherwise expressly provided. Seller may, at its option, accept cancellation on a no-charge basis, retaining in its possession any production material acquired for processing the cancelled order.

7. GOVERNING LAW The entering into, construction, interpretation, performance and discharge of this contract will be governed by the Laws of Pennsylvania.

PRICE: As noted on page 4.

TERMS: 30 Days Net

DELIVERY: F.O.B. WARREN, PA.

Very truly yours,

STRUTHERS WELLS CORPORATION

R. R. James

R. R. James, Sales Engineer
Special Products Department

RRJ:par
(Original + 2)

CC: H. A. Backstrom
E. W. Eschborn

FORM 60.2