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FEASIBILITY OF A HEAT AND EMISSION LOSS PREVENTION SYSTEM FOR AREA SOURCE FURNACES

Industrial Environmental Research Laboratory
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FEASIBILITY OF A HEAT AND EMISSION LOSS PREVENTION SYSTEM FOR AREA SOURCE FURNACES

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FOREWORD

This report presents the results of a study to determine the feasibility of candidate concepts for simultaneous heat and air pollutant emission recovery from the combustion products of domestic-size furnaces. The study was performed by the Aerotherm Division of Acurex Corporation for the U.S. Environmental Protection Agency.

Aerotherm extends its appreciation to Mr. Kenneth M. Brown for his valuable assistance with the detailed heat transfer analysis.

The EPA Task Officers were Mr. D. B. Henschel and Mr. W. B. Steen. The Aerotherm Program Manager was Dr. Larry W. Anderson. The study was performed during the months of June through September 1974.

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SECTION 1

SUMMARY

Residential space heating represents 11.0 percent of the total U.S. energy consumption. Thus there is a strong incentive to reduce energy consumption in the residential section particularly for space heating. There are a great number of studies being conducted through the National Bureau of Standards, NSF, American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE), American Gas Association, public utilities and others on the overall problem of energy use reduction in residential heating and cooling. These studies have shown that between 9 and 28 percent of the heating load could be recovered by heat recovery of the furnace flue and shut off of the flue during nonuse periods. This represents from 3.6 to 11 percent of the total energy consumption of the house and from 1 to 3 percent of total U.S. energy consumption. Other significant energy conservation techniques include reduction of air infiltration and increased insulation in wall ceilings and floors. However, none of these studies dealt directly with the development of a heat recovery device.

The first step to develop a heat and emissions recovery system (HELPS) is to determine the distribution of heating systems across the U.S., their efficiency, their design and limitations on their design. U.S. census data for 1970 show that the central forced air, gas-fired furnace is the principle heating method with oil and gas-fired hot water system and oil-fired central air systems ranking 2, 5, and 4, respectively. In the Northeast section of the country, oil-fired hot water systems are the most prevalent. However, in the last ten years, gas-fired central air furnaces accounted for 29 to 46 percent of the heating systems sold nationwide.

The gas-fired central air furnace typically has a steady state efficiency of about 80 percent with oil-fired units either a few percentage points below or nearly equal to the 80 percent figure. The gas-fired system uses an aspirated burner and relies on the draft created by the buoyancy forces in the flue to draw sufficient combustion and excess air through the furnace. This results in very low pressures (.05" W.C.) in the combustion chamber, low heat transfer coefficients and a low pressure drop through the unit. Higher efficiencies could be achieved with lower flue gas temperatures. At 80 percent efficiency, the temperature of the flue gas exiting the furnace is such that condensation is avoided in the stack and sufficient buoyancy is created to discharge the flue from the stack.

Concepts for higher efficiency furnaces can be retrofit systems or systems applicable only to new installations. Retrofit concepts are important because of the large number of existing units that require attention. Concepts that can be applied only to new installations deserve attention because they offer greater potential in fuel, emission, and cost savings. Research aimed at investigating both new and retrofit systems is ongoing.

Research on higher efficiency furnaces is being conducted by the furnace manufacturers, oil companies and by the American Gas Association (AGA). The Amana Company recently announced a new furnace design concept (Heat Transfer Module) which utilizes a forced air surface combustion burner which transfers heat to an intermediary fluid. Slightly higher efficiencies are claimed (5-8 percent) while flue gas temperatures are maintained high enough to avoid condensation.

AGA over the last 20 years has explored a number of unconventional concepts for new installations. The most significant and one on which they are still working is the submerged pulsed combustion concept. Efficiencies as high as 95 percent are claimed for this concept which relies on condensation of the water in the flue.

Two commercially available flue heat recovery devices have been identified for retrofit on residential heating systems. A heat pipe device manufactured by Isothermics, Inc., of New Jersey can recover from 5,000 to 12,500 Btu/hr or 5 to 12-1/2 percent of a typical 100,000 Btu/hr furnace, depending on flue gas temperature. A flue gas-to-air tubular exchanger is available from the Dolin Manufacturing Company. Their unit recovers only from 3000 to 5300 Btu/hr. Other retrofit devices consist of off-cycle flue dampers to prevent heat loss through the flue pipe.

For the retrofit market, external modifications were determined to be the most promising path. For these retrofit schemes, the heat in the flue may be exchanged with one of the following mediums:

- Combustion air (air preheat)
- Fuel (fuel preheat)
- Recirculating house air/water exiting from furnace
- Independent air/water stream
- An intermediary fluid or solid which eventually exchanges heat with one of the other mediums ("runaround system").

It was determined that the last four schemes were potentially practical. The latter two are represented by current devices on the market so were not considered further although improvements were identified.

Common to all retrofit designs are the following problem areas:

- Pressure drop
- Removal of condensate (if condensing system)
- Corrosion and materials selection
- Thermal stresses and cycling fatigue.

It was determined that in order to make any significant improvement on efficiency, a compact heat exchanger design and induced draft fan (to compensate for increased system pressure drop) would be required. This heat exchanger and the associated duct work would be manufactured from a low grade stainless steel to avoid corrosion problems. Up to 30,000 Btu/hr could be recovered from a high temperature flue if the temperature was reduced to the condensation point so that the latent heat of vaporization as well as the sensible heat could be recovered.

Detailed analysis was undertaken on a plate fin and fin-tube design for exchange with the exit air or water, or inlet air or water and flue for both condensing and noncondensing systems. A simple computer program was written to calculate the performance of any potential surface configuration. From these calculations, a noncondensing plate fin exchanger with dimensions of 1.2" x 7.1" x 15" was determined to be an attractive arrangement. A larger unit was designed for a condensing heat exchange surface. These units would easily fit over the air inlet or outlets of the furnace in much the same manner as a filter.

Bids were solicited from a number of heat exchanger manufacturers and a broad spectrum of cost were returned. From these data, cost estimates were made for each scheme for the heat exchanger, required ductwork, insulation, fan, and installation. From the performance curves, the fuel cost savings per year was estimated and the payback period determined. The payback period takes into account the time value of money and assumes that a seven percent rate of return is available. The calculations show that in terms of payback, the flue damper is the best buy. It also reveals that the most promising devices are those that are applied to oil-fired furnaces with high flue gas temperature.

It was also shown that if a 50 percent fuel savings could be achieved with a new furnace design incorporating higher efficiency heat exchanger, closure of the flue during non-use periods and furnace modulation, an additional

$500 to $1000 could be invested as initial capital investment with a payback in five to ten years.. However, although the large scale economics appear attractive, formidable social and institutional obstacles would act to prevent the widespread use of either the retrofit or new furnace designs.

The potential market is, however, rather large if one assumes 20 percent of the units in the Northeast and Northcentral portion of the U.S. are converted. This assumption would indicate a total conversion of 6.33 million units and a potential fuel savings by 1982 of 3.02 percent of the U.S. space heating energy consumption estimate.

In addition to the energy savings aspects of the proposed concepts, the HELPS study also reviewed the potential for reducing emissions from domestic-size furnaces. Emissions from gas and oil-fired units account for 1-10 percent of the nation's total emissions. Control strategies for these sources include fuel substitution, combustion process modification and post-combustion control. Fuel substitution has been incorporated for quite a few years with the switch from coal to oil and to gas with a subsequent decrease in emissions, especially particulates. However, as gas becomes scarce, there may be a trend back to the other fuels or electricity. Most of the combustion control strategies can be applied to oil-fired units with excess air level adjustment the primary control for gas-fired units. Research is currently underway by EPA and AGA on low emission, high efficiency oil and gas burners, respectively. In all, the feasibility of most of the post-combustion control devices for residential heating application is unknown.

The results of the present study lead to the following recommendations for future work of a general nature and for additional studies of a retrofit scheme:

1. Obtain actual operating data on furnaces by measurement
2. Study British and European design practice where fuel prices have been historically higher
3. Develop a furnace performance code and compare with 1 above
4. Monitor commercial and legal progress of retrofit devices and test these units
5. Refine the design of a condensing flue gas heat recovery device.

However, the most fruitful approach would probably be to explore further the possibility of a new high efficiency furnace design.

Recommended components in the development of improved furnace designs include Recommendations 1 and 3 above, followed by tasks to:

6. Predict efficiency of proposed modulation schemes by analytical techniques; identify cycles to optimize operation; modify cycle to see how to improve efficiency

7. Analytically predict the effect on heat transfer due to modulating output

8. Explore methods of modulation

   - Vary gas flow rate
   - Use multiple burners in various combinations
   - Heat capacitance/hot water systems

9. Develop heat load detector and furnace controller

10. Modify an existing furnace for modulated mode

11. Design a new furnace using either submerged combustion or catalytic combustion concepts in conjunction with the modulated concept

12. Build and test a prototype unit as described in Recommendation 11

13. Relate the progress of this design and development effort to:

   - Commercial and research developments in alternative heat pump/all electric systems (including solar-assist components)
   - Other integrated energy management system programs, including total-energy and MIUS
   - Household energy conservation programs which will be affecting thermal load data for new construction

With regard to emission control devices two specific areas of study are recommended:

14. Further define the real contribution of residential emissions to environmental air quality

15. Further investigate post combustion control devices, if required

16. Define the potential of the modulated furnace concept (Recommendations 6-11) to reduce the high peak emission associated with start-up and shutdown periods.

SECTION 2

INTRODUCTION

This report presents the results of a study to determine the feasibility of candidate concepts for simultaneous heat and air pollutant emission recovery from the combustion products of domestic-size furnaces. The major objectives of the study, separated into report Sections, were as follows:

Definition of Problem and Existing Technology (Section 3)

This Section includes statistics on energy consumption and emissions associated with domestic furnaces, a review of furnace design practice, and descriptions of commercially available heat recovery devices.

Identification of Alternate Heat Recovery Designs (Section 4)

For typical operating conditions, the most promising concepts identified in the preliminary phases were characterized in detail, designed, and optimized, both for retrofit applications and for new installations.

Detailed Analysis of Retrofit Schemes (Section 5)

Two of the most attractive options for retrofit flue gas heat recovery devices were analyzed in greater depth.

System Cost (Section 6)

Capital and operating costs were estimated in this phase for the systems designed for retrofit and new installations. Possible impacts of mass production were considered.

Control of Air Pollutant Emissions From Residential Heating Equipment (Section 7)

Past, present, and future combustion and post-combustion control methods were reviewed.

Potential Market (Section 8)

This final task evaluated the potential of energy recovery devices considering overall economics (payback periods) and the total size of the potential markets. The possible impact of such devices on the U.S. energy situation was estimated.

Conclusions and Recommendations (Section 9)

Further work that is required to complete the analysis of the costs and benefits of the HELPS concept is outlined.

In all, the report serves to clarify the energy and environmental impact of the residential heating sector, as well as to analyze the feasibility of several solutions proposed to lessen this impact.

SECTION 3

DEFINITION OF PROBLEM AND EXISTING TECHNOLOGY

As background to this study a brief survey was made of other work being done in this field, current furnace design practice, and current heat recovery devices on the market.

3.1 ENERGY CONSUMPTION AND METHODS OF ENERGY SAVINGS

Table 1 shows the pattern of basic energy consumption in the U.S. for 1968 (Reference 1). This reveals that 11 percent of the total energy consumption went to residential space heating and 6.9 percent went to commercial space heating. The total residential and commercial energy usages are 19.2 percent and 14.4 percent respectively. Thus, there is a strong incentive to reduce energy consumption in the residential and commercial sectors and particularly for space heating.

Table 2 lists a number of government agencies, universities, and trade associations with ongoing programs in various aspects of energy conservation in the residential and commercial sector. Entries in the table were compiled from an Aerotherm survey conducted under this task from testimony presented at recent hearings on Conservation and Efficient Use of Energy before the Committee on Government Operations and Science and Astronautics of the House of Representatives (Reference 2) and from a detailed inventory of current energy research and development activities compiled by the Oak Ridge National Laboratory (Reference 3)*. Table 2 cites only programs which

- Have a direct relevance to this study and which have been personally investigated in detail in this program, or
- Provide a representative sampling of the total research picture in residential space heating

*This document was obtained late in the course of this brief study, and not all apparently relevant leads have as yet been pursued in detail.

TABLE 1

ENERGY CONSUMPTION IN THE U.S. BY END USE 1968
(TRILLIONS OF Btu AND PERCENT PER YEAR)
(Reference 1)

	Consumption	Percent of National Total
Residential		
Space Heating	6,675	11.0
Water Heating	1,736	2.9
Cooking	637	1.1
Clothes Drying	208	0.3
Refrigeration	692	1.1
Air Conditioning	427	.7
Other	1,241	2.1
Total	11,616	19.2
Commercial		
Space Heating	4,182	6.9
Water Heating	653	1.1
Cooking	139	.2
Refrigeration	670	1.1
Air Conditioning	1,113	1.8
Feedstock	984	1.6
Other	1,025	1.7
Total	8,766	14.4
Industrial	24,960	41.2
Transportation	15,184	25.2
National Total	60,526	100.0

TABLE 2

STUDIES IN CONSERVATION OF ENERGY FROM
RESIDENTIAL AND COMMERCIAL BUILDING

Contacted	Agency/Association/Institution	Nature of Studies	Reference(s)
X	U. S. Department of Commerce Bureau of Standards - Center for Building Technology	• Computer analysis of building thermal response • Importance of insulation • Fenestration and Infiltration • Equipment and System Options • Heat pumps • Heat Exchangers • Underground heat distribution systems • Field demonstration projects using energy conservation projects • Total energy systems • Publication of relevant energy conservation information • Workshops on energy conservation • Recommendation of standards	4, 5, 6, 7, 8
X	U. S. Department of Housing and Urban Development - Office of the Assistant Secretary for Policy Development and Research (Support from NSF-RANN and EPA) (Hittman Associates)	• Identify means for obtaining greater efficiencies in the consumption of residential power • Quantify total energy balance in typical residence • Indentify technical innovation to reduce residential energy utilization • Determine the total annual energy savings possible for several modified versions of the characteristic house incorporating energy conservation modifications • Evaluate heating loads in old residential structures	9, 10, 11, 12, 13, 14
X	- Building Technology and Site Operations (California State Polytechnic University)	• Research, evaluation of a system of natural air conditioning with solar energy as the power source	2, 17

TABLE 2 (Continued)

STUDIES IN CONSERVATION OF ENERGY FROM
RESIDENTIAL AND COMMERCIAL BUILDING

Contacted	Agency/Association/Institution	Nature of Studies	Reference(s)
X	National Science Foundation/RANN - Center for Environmental Studies, Princeton University	• Research project entitled Energy Conservation in Housing - Statistically study of a large multiple family dwelling to determine distribution of energy utilization - Development of data acquisition system - Surveillance of the construction of townhouses and apartments - Interview and discussion with developers to develop ideas for improvement for energy utilization	15
X	- Oak Ridge National Laboratory U.S. Atomic Energy Commission	• Estimate potential monetary and energy savings through the use of additional thermal insulation in residential construction	16
		• Develop inventory of current energy research and development	2
	- Colorado State University, Solar Energy Applications Laboratory	• Experimental development of a full operational residential heating and cooling system based on solar energy	2
	- Texas A&M, Texas Engineering Experiment Station	• Development of a compressed-film floating deck solar water heater	2
	- University of Wisconsin, Solar Energy Laboratory	• Simulation study of solar heating and cooling for the United States	2
	- California Institute of Tech., Environmental Quality Lab.	• Solar/thermal technologies for buildings - technical, economic and institutional aspects • Project SAGE - a project to investigate the commercial feasibility of solar assisted gas energy for water heating in new apartments	2

TABLE 2 (Continued)

STUDIES IN CONSERVATION OF ENERGY FROM RESIDENTIAL AND COMMERCIAL BUILDING

Contacted	Agency/Association/Institution	Nature of Studies	Reference(s)
X	- American Society of Heating Refrigerating and Air Conditioning Engineers (ASHRAE)	• Preparation and publication of a ASHRAE Guide chapter on application of solar energy for heating and cooling of buildings	2
	- University of Florida	• Formulation of a data base for the analysis, evaluation and selection of a low temperature solar powered air conditioning system	2
	- Carnegie-Mellon University	• A selective total energy system for a residential complex	2
X	American Gas Association		
	- Institute of Gas Technology	• Detailed energy usage profile in the Canton/test homes • An assessment of selected heat pump systems	18
	- Allied Chemical Corporation	• Absorption heat pump program • New working fluids for Rankine cycle residential air conditioning system • New heat pump working fluids	18
	- K. Johnson Co.	• Thermal refrigeration - direct conversion of heat energy into refrigeration	18
	- Ohio University	• Free piston stirling engine driven - gas fired air conditioner	
	- Dynatech Corporation	• Basic research in heat transfer	18
X	- A.G.A. Laboratories	• Design of domestic appliances to use gas at elevated pressures • Development of a central heating furnace with ultra high efficiency (submerged-pulsed combustion device)	18 18, 19, 20, 21, 22

TABLE 2 (Continued)

STUDIES IN CONSERVATION OF ENERGY FROM
RESIDENTIAL AND COMMERCIAL BUILDING

Contacted	Agency/Association/Institution	Nature of Studies	Reference(s)
	- Battelle Columbus Laboratories	• Investigation of thermal fatigue of heat exchangers in domestic gas furnaces	18
		• Indoor air quality control device	2
	- Thermo-Electron Corporation	• New developments in gas fired appliances; application of Rankine cycle engines to residential gas air conditioning	2
	- AiResearch Manufacturing Co.	• Development of a heat actuated space conditioning unit for commercial applications - utilization of a Brayton cycle engine	2
	Pope, Evans and Robbins	• Economic advantage of central heating and cooling systems	23
	University of Oklahoma Research Institute/Public Service Co. of Tulsa	• A comparitive study of energy usuage	23
X	American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE)		
	- Edison Electric Research Corp. (York Research Corp.)	• Measurement of the performance of the ASHRAE infrared space heating system	2
		• Determination of specific heats of building materials	
	- University of Florida	• Determination of shading coefficients for reflective coated glasses with draperies	2
X	- ASHRAE Research and Technical Committee, Task Group on Energy Conservation	• A task group on energy conservation has been established to evaluate the manner in which basic energy sources can be conserved through the more effective utilization of energy in residential commercial institutional and industrial buildings. Conducts symposia, publishes papers by members, e.g. reference	24, 25

TABLE 2 (Continued)

STUDIES IN CONSERVATION OF ENERGY FROM
RESIDENTIAL AND COMMERCIAL BUILDUNG

Contacted	Agency/Association/Institution	Nature of Studies	Reference(s)
		25 Symposium on Heat Recovery. Update ASHRAE Guide and data book to reflect latest technology on conservation of energy. Provides component index of technical articles published by the Society and presented in the Journals.	
X	- ASHRAE Standard 90-P Committee	• Has taken the NBS document on design and evaluation criteria for energy conservation mentioned above and drafted an ASHRAE Interim Standard on Energy Conservation for use by the National Conference of States on Building Codes.	5, 26, 27
	Long Island Lighting Co.	• Electric energy utilization study of the generic environmental impact of electric versus oil space heating, i.e. what effect electric heat will have on air quality	2
	Stevens Institute of Technology	• Determine efficiency fuel oil burners in the field and difference between makes. • Determine method of maintaining efficiency	2
X	Pennsylvania Power and Light Co. - Franklin Institute Res. Labs.	• Assist in the design of energy conserving home incorporating features such as heat pumps, waste heat recovery, supplementary solar heating and thermal energy storage • Construction and testing of the above house	2
	Consolidated Natural Gas Service Co. (Thermo-Electron Corp.)	• Development of a compact liquid heater for industrial and commercial heat transfer applications	2
	Oklahoma Gas and Electric Co.	• Development of a computer oriented system for determining building energy requirements and an economic analysis of their requirement	2

TABLE 2 (Concluded)

STUDIES IN CONSERVATION OF ENERGY FROM
RESIDENTIAL AND COMMERCIAL BUILDING

Contacted	Agency/Association/Institution	Nature of Studies	Reference(s)
X	Southern California Gas Co.	• Similar to above; development and distribution of E cube computer program to determine energy consumption of a building. Also sponsored by A.G.A.	28
	- Thermo-Electron Corporation	• Application of heat pipe technology to appliances	2, 29
	Environmental Protection Agency		
X	- Contractor	• Conducted study of various strategies for reducing national energy demand	30
X	- Rocketdyne	• Development of optimum oil-fired burner and furnace package	39
X	- Aerotherm/Acurex	• HELPS	

There are many more programs dealing with solar heating and typical energy utilization of family dwellings.

A large number of these studies are concerned with the overall reduction of energy for both heating and cooling through

- Better and more insulation
- Lower indoor temperature levels for heating
- Higher indoor temperature levels for cooling
- Setback of temperature at night
- Closing off of unused rooms
- Reduced lighting
- Addition of insulating glass or storm windows
- Better window caulking
- Insulation of heating ducts
- Improved maintenance of heating equipment
- Better management of doors, windows, and drapes
- Installation of automatic pilots
- Adjustment of ventilation systems
- Minimized use of portable electric heaters by improving main heating system
- Replacement of inefficient heating systems with systems of higher efficiency
- Systems modification for zone control
- Means to transfer heat from the center of a large building to the cool periphery
- Automatic door closures
- Installation of heat recovery devices (References 6, 10, 30)

For example, the National Bureau of Standards estimated that the winter heating loads could be reduced by as much as 40 percent if all these options were incorporated (Reference 6). To recover heat from stacks Reference 6 mentions heat pipes, runaround circuits and automatic stack dampers. For ventilation systems Reference 6 recommends runaround circuits, thermal wheels, heat pipes, heat pumps, and other heat exchange circuits. However, no details were given of these systems.

The Hittman Report on residential energy consumption (Reference 10) came to the following conclusions:

- Infiltrated air was the greatest load component for both heating and cooling
- Conductive losses thorugh the walls and windows ranked second and third, respectively
- Losses through ceiling, floors and doors were relatively small.

Table 3 shows the estimated annual energy savings by incorporation of various energy conservation modifications as presented by Siedel, et al. (Reference 30). This shows that in terms of saving energy, heat recovery of the furnace flue represents the largest single contributor. This figure of 28 percent, however, includes recovery of the latent heat of vaporizations and incorporation of a furnace flue shutoff device. A more detailed breakdown of these savings is available from the Hittman Report and is shown in Table 4.

This shows that a noncondensing heat recovery device can save about 9 percent of the heating load and only 3.6 percent of the total energy consumption of the house.

Research by the Oak Ridge National Laboratory shows even greater potential savings from increased insulation (Reference 16). They estimate that a gas heated home could realize energy savings as high as 30 to 50 percent at no economic penalty through increased insulation in floor, walls, and ceiling.

For a more detailed discussion of the various strategies for energy reductions, the reader is referred to the EPA document on Energy Conservations Strategies (Reference 30) and the NBS document on Technical Options for Energy Conservation in Buildings (Reference 6) as well as the Hittman-HUD reports (References 14-16).

Energy conservation strategies discussed by the American Society for Heating Refrigeration and Air Conditioning Engineers (ASHRAE) (References 24, 25) have been primarily oriented towards commercial and institutional buildings. However, their recent development of draft interim standards, ASHRAE Standard 90P (Reference 27), for energy conservation in new buildings (the implementation of recommendations by the NBS) includes all building and equipment sizes. This standard is under open review to be completed September 1974. These new standards cover the following areas:

TABLE 3

SAVINGS FROM MODIFICATION OF CHARACTERISTIC DESIGN
(Reference 30)

Load or MOD	Winter Load		Summer Load	
	Percent Saved	MBTU Saved	Percent Saved	MBTU Saved
Furnace Reference Load	100.0	101.4		
Air Conditioner Load*			100.0	10.8
Furnace Recovery	27.9	28.3		
High Performance Unit**			33.3	3.6
Furnace Pilot Elimination***	3.4	3.5		
Open Air Cycle			8.3	1.0
Storm Windows	15.8	16.0	8.1	.9
25% Window Area Reduction	19.1	19.4	9.5	1.0
Cinder Block Insulation	7.1	7.2	.6	.1
High Capacity Wall	2.6	2.6	.8	.1
Sealed Furnace Air Supply	4.6	4.7		
Sealed Hot Water Air Supply	1.7	1.7	1.9	.2
Clothes Dryer Recovery	2.4	2.4		
Double Door Design	1.6	1.5	3.8	.4
Revolving Door Design	2.6	2.7	6.9	.7
Ducted Oven Design			1.9	.2
Ducted Refrigerator			7.0	.8
Attic Ventilation			.5	.1

* - Based on 8.0 Btu/watt-hr performance
** - Based on 12.0 Btu/watt-hr performance
*** - Based on a 1000 Btu/hr pilot light

TABLE 4

BREAKDOWN OF ENERGY SAVINGS FOR FURNACE RECOVERY AND FLUE DAMPER
(Reference 10)

"Characteristic House Energy Consumption"
(Therms)

	a		b		c
	Heating	Cooling	Total Heating & Cooling	Lights & Appliances	Total
Annual Consumption	1044	388	1432	1174	2606

	Energy Therms	Percent of a	Percent of b	Percent of c
Heat Recovery Device Noncondensing	94	9	6.6	3.6
Heat Recovery Device Condensing	~ 194	18.6	13.5	7.4
Furnace Flue Closure	~ 100	9.6	7.0	3.8
Heat Recovery & Furnace Flue Closure Noncondensing	194	18.6	13.5	7.4
Heat Recovery & Furnace Flue Closure Condensing	294	28.1	20.5	11.28

- Insulation
- Procedures for determining heating or cooling system capacity
- Temperature control, single zone and multiple zone
- Humidity control
- Controlled temperature setback during non use periods
- Simultaneous heating and cooling
- Reheat
- Illumination
- Energy recovery (air exhaust-makeup systems only)
- Air infiltration
- Procedures for HVAC zoning
- Duct air leakages
- Automatic ignition system
- Recommended equipment efficiencies (efficiencies should comply with current American National Standards Institute (ANSI) for heating equipment; to be discussed in Section 3.4).
- Prevention of off cycle air flow heat loss

The later criterion recommends that all combustion heating equipment with greater than 250,000 Btu/hr input should be equipped with devices, such as automatic vent dampers, to minimize air flow through the combustion chamber during off cycles. These standards will significantly affect the energy consumption of new residential construction through the areas of better insulattion and limitations of air infiltration. It is estimated that through these two areas alone from 30 to 50 percent of the heating load could be saved depending on location of total number of infiltration points. An overview of the Energy Conservation Standard recently appeared in the ASHRAE July 1974 issue (Reference 26).

Although the quantitative results of these various studies differ according to differences in assumptions, the results agree that improvements in furnace efficiency (or, alternatively, flue gas energy recovery) can provide significant energy savings in a major energy consumption area.

3.2 FURNACE TYPES

The first step in designing a heat recovery device is to determine the types and distributions of furnaces across the U.S. ASHRAE, AGA, GAMA, furnace manufacturers and other trade associations consistently referred us to the 1970 U.S. Census information as the most detailed breakdown available.* Table 5 from Reference 31 presents the 1970 U.S. Census information on the distribution of residential heating systems for the entire U.S. This information is divided into heating system type, fuel, owned or rented, percentage of total population of furnaces and the ranking.** Unfortunately, there was not a good breakdown on definitions of "other heating equipment", which may include room heaters with flue, room heaters without flue, fireplaces, stoves or portable heaters. This table shows that the principal heating method for the U.S. as a whole is the gas fired warm air furnace (31.6 percent; this includes both gravity type and forced air systems). Oil fired hot water or steam ranks second with 12.29% of the heating systems. There are, however, considerable variations in the mix of heating system types across the country. Figures 1 and 2, also from the U.S. Census data, show the gross distribution of heating equipment types and housing fuels for four regions of the country. These figures show that in the Northeastern section of the country, 56.2 percent of the homes are steam or hot water heated and that 54.4 percent of all the furnaces are oil fired. By contrast, 41.1 percent of the homes in the West are warm air furnaces and 71.1 percent are gas fired.

Table 6 from data published by the Gas Appliance Manufacturers Association (GAMA) show that the trend in the last ten years is toward gas fired forced air systems, which account for 46.2 percent of the furnaces sold in 1973 (Reference 32). This is further substantiated by U.S. Census data shown in Table 7 (Reference 31), which indicates that for houses built prior to 1939 almost 30 percent of the warm air furnaces are oil fired compared to approximately 15 percent for houses built during the period 1960-1970. These data show conditions at the time of the 1970 Census which no doubt include a considerable number of replacement units. The older more inefficient furnace are most amenable to retrofit devices. Unfortunately, one cannot determine directly the furnace age from these statistics. These tables and figures thus indicate that both gas and oil fired central warm air and hot water furnaces should

*Some large manufacturers indicated they had similar information but that it was considered proprietary.

**The two categories, "built in electric unit" and "none", appearing in the U.S. census tables of Reference 31 have been omitted from Table 5.

TABLE 5

RESIDENTIAL HEATING SYSTEM IN THE U.S. 1970 CENSUS
(Reference 31)

	Owner	Rented	Total	%	Rank
Steam or Hot Water					
Gas	2,415,455	2,694,746	5,110,201	8.78	5.
Fuel Oil	3,687,913	3,463,636	7,151,549	12.29	2.
Coal or Coke	183,293	407,821	591,114	1.02	
Wood					
Electricity					
Bottled, Tank, or LP Gas	84,758	75,334	160,092	0.28	
Other Fuel	32,205	175,181	207,366	0.36	
Warm Air Furnace					
Gas	13,968,429	4,389,501	18,357,930	31.56	1.
Fuel Oil	4,746,515	1,284,546	6,031,061	10.37	4.
Coal or Coke	428,130	155,518	583,648	1.00	
Wood					
Electricity					
Bottled, Tank, or LP Gas	966,673	224,906	1,191,579	2.04	9.
Other Fuel	24,162	20,686	44,848	0.08	
Floor Wall or Pipeless					
Gas	2,672,364	1,858,511	4,530,875	7.79	6.
Fuel Oil	414,983	156,663	571,646	0.98	
Coal	40,637	20,644	61,281	0.10	
Wood					
Electricity					
Bottled, Tank, or LP Gas	273,788	103,834	377,622	0.65	
Other Fuel	3,447	3,673	7,120	0.01	
Other Heating Equipment					
Gas	3,213,424	3,801,315	7,014,739	12.06	3.
Fuel Oil	1,650,175	1,069,039	2,719,214	4.67	7.
Coal	320,499	264,410	584,909	1.00	
Wood	438,330	355,578	793,908	1.36	10.
Electricity					
Bottled, Tank, or LP Gas	1,406,157	670,498	2,076,655	3.57	8.
Other Fuel	3,683	3,249	6,932	0.01	
			58,174,289	99.98	

	Northeast	Northcentral	Southwest	West
Steam or Hot Water	56.2%	16.0%	5.9%	5.1%
Warm-air Furnace	31.6%	64.5%	34.9%	41.1%
Built-in Electric Units	2.6%	2.3%	7.4%	8.8%
Floor, Wall, or Pipeless Furnace	1.7%	4.3%	10.8%	22.1%
Room Heaters with Flue	5.3%	10.2%	15.7%	14.2%
Room Heaters without Flue	1.1%	1.3%	14.8%	2.8%
Fireplaces, Stoves, or Portable Heaters	1.3%	1.4%	0:6%	2.2%
None	0.1%	0.0%	0.6%	2.2%

= 5%

Figure 1. Heating Equipment, by Regions: 1970-Percent (All Occupied Units) (Reference 31)

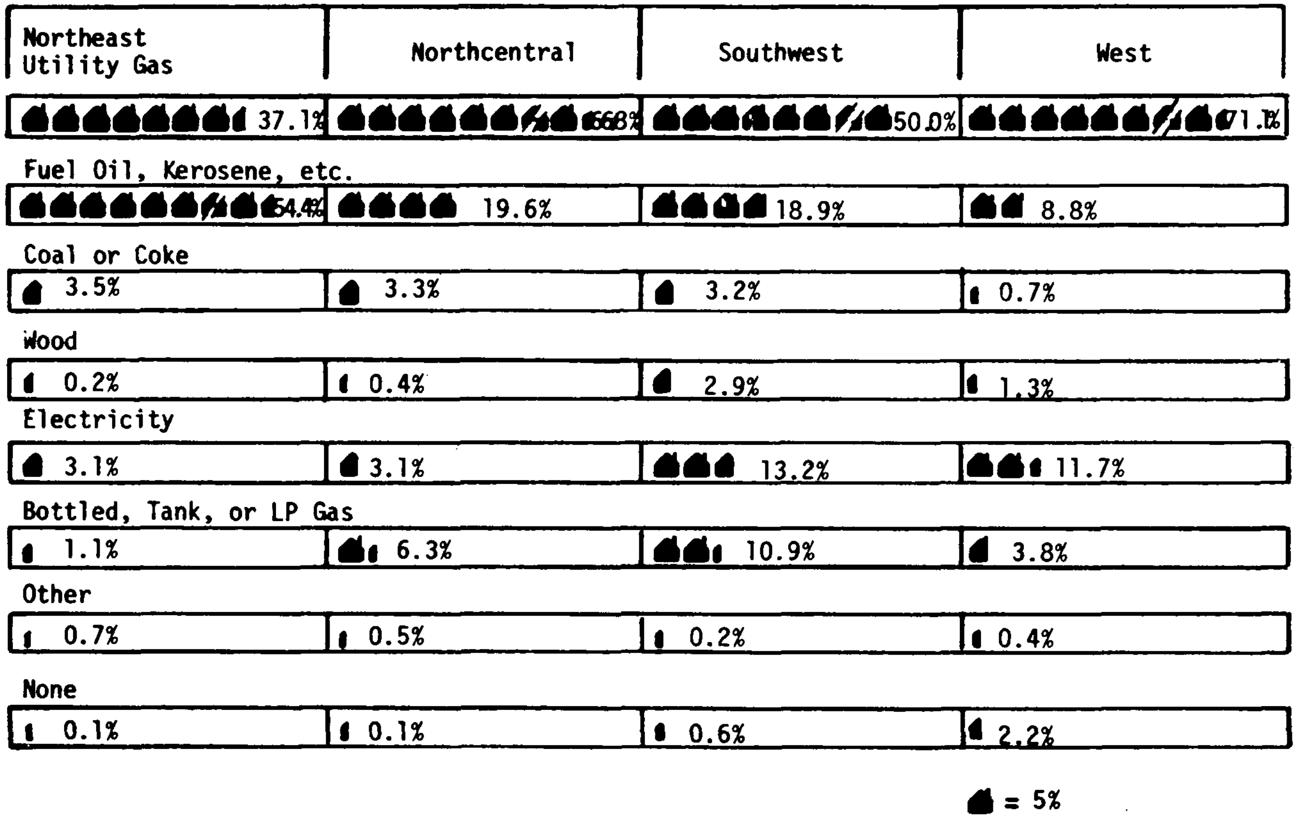


Figure 2. House Heating Fuel, by Regions: 1970 Percent (All Occupied Units) (Reference 31)

TABLE 6

FURNACE SYSTEMS SOLD (GAMA)
(Reference 32)

	1973		1963	
	#	%	#	%
Gas Fired Forced Air	2,261,600	46.2	1,394,300	29.4
Floor - Gas	42,900	0.9	77,200	1.6
Vented Wall Furnace - Gas	719,660	14.7	448,100	9.4
Oil Fired Cent.	734,000	15.0	520,200	11.0
Oil Fired Floor & Wall	N/A		30,300	0.6
Gas Fired Direct Heating	708,300	14.5	1,323,800	27.9
Oil Fired Direct	197,600	4.0	417,000	8.8
Coal & Wood Direct	228,200	4.7	537,400	11.3
Totals	4,892,260	100%	4,748,300	100%

TABLE 7

(Reference 31)

Year Structure Built for Owner and Renter Occupied Housing Units by Heating Equipment 1970

Household Head—All Races

[Data based on sample, see text. For meaning of symbols, see text]

United States

HEATING EQUIPMENT BY HOUSE HEATING FUEL	Owner occupied: Total	Owner occupied: 1960 to March 1970	Owner occupied: 1950 to 1959	Owner occupied: 1940 to 1949	Owner occupied: 1939 or earlier	Renter occupied: Total	Renter occupied: 1960 to March 1970	Renter occupied: 1950 to 1959	Renter occupied: 1940 to 1949	Renter occupied: 1939 or earlier
Steam or hot water	**6 403 624**	**1 130 071**	**1 240 413**	**601 621**	**3 431 519**	**6 816 718**	**1 014 859**	**696 233**	**782 369**	**4 323 257**
Utility gas	2 415 455	505 486	372 917	200 555	1 336 497	2 694 746	559 174	267 402	297 964	1 570 206
Fuel oil, kerosene, etc	3 687 913	582 870	836 556	379 325	1 889 162	3 463 636	399 179	379 730	403 094	2 281 633
Coal or coke	183 293	6 295	13 609	11 813	151 576	407 821	11 408	17 840	44 394	334 179
Wood	–	–	–	–	–	–	–	–	–	–
Electricity	–	–	–	–	–	–	–	–	–	–
Bottled, tank, or LP gas	84 758	30 033	12 474	6 659	35 592	75 334	13 510	9 011	11 408	41 405
Other fuel	32 205	5 387	4 857	3 269	18 692	175 181	31 588	22 250	25 509	95 834
None	–	–	–	–	–	–	–	–	–	–
Warm-air furnace	**20 884 609**	**6 988 059**	**5 875 640**	**2 158 617**	**5 862 293**	**6 616 735**	**2 282 206**	**1 127 940**	**705 771**	**2 500 818**
Utility gas	13 968 429	4 921 180	4 155 148	1 411 688	3 480 413	4 389 501	1 527 538	794 339	489 389	1 578 235
Fuel oil, kerosene, etc	4 746 515	1 058 415	1 365 369	586 325	1 736 406	1 284 546	243 604	225 578	156 177	659 187
Coal or coke	428 130	16 676	45 265	52 093	314 096	155 518	4 307	8 126	15 064	128 021
Wood	–	–	–	–	–	–	–	–	–	–
Electricity	750 700	504 785	143 320	41 875	60 720	541 578	422 922	62 401	21 145	35 110
Bottled, tank, or LP gas	966 673	481 897	160 884	63 335	260 557	224 906	76 950	34 505	21 546	91 905
Other fuel	24 162	5 106	5 654	3 301	10 101	20 686	6 885	2 991	2 450	8 360
None	–	–	–	–	–	–	–	–	–	–
Floor, wall, or pipeless furnace	**3 406 219**	**463 420**	**1 164 639**	**725 396**	**1 052 764**	**2 143 325**	**464 856**	**596 720**	**442 833**	**638 916**
Utility gas	2 672 364	284 025	950 491	622 917	814 931	1 858 511	410 865	531 263	394 156	522 227
Fuel oil, kerosene, etc	414 983	99 701	141 154	60 527	113 601	156 663	29 343	41 509	28 025	57 786
Coal or coke	40 637	2 093	4 411	5 509	28 624	20 644	991	1 252	2 524	15 877
Wood	–	–	–	–	–	–	–	–	–	–
Electricity	–	–	–	–	–	–	–	–	–	–
Bottled, tank, or LP gas	274 788	76 939	67 757	35 838	94 254	103 834	22 563	22 003	17 543	41 725
Other fuel	3 447	662	826	605	1 354	3 673	1 094	693	585	1 301
None	–	–	–	–	–	–	–	–	–	–
Other heating equipment	**7 241 305**	**1 051 219**	**1 511 323**	**1 242 996**	**3 435 767**	**6 306 899**	**634 368**	**955 817**	**1 144 942**	**3 571 772**
Utility gas	3 213 424	354 334	690 623	611 441	1 557 021	3 801 315	396 548	591 460	722 745	2 090 562
Fuel oil, kerosene, etc	1 650 175	248 853	341 817	235 636	823 869	1 069 039	88 790	156 262	174 148	649 839
Coal or coke	320 499	27 860	40 105	49 799	202 735	264 410	13 288	23 427	37 786	189 909
Wood	438 330	67 875	67 457	72 549	230 449	355 578	24 136	35 542	53 855	242 045
Electricity	209 037	48 968	64 887	39 804	55 378	142 810	24 654	33 283	30 483	54 390
Bottled, tank, or LP gas	1 406 157	302 735	305 513	233 203	564 706	670 498	86 747	115 333	125 290	343 128
Other fuel	3 683	594	916	564	1 609	3 249	205	510	635	1 899
None	–	–	–	–	–	–	–	–	–	–
TOTALS	37 935 757	9 632 769	9 792 015	4 728 630	13 782 343	21 883 677	4 396 289	3 376 710	3 075 915	11 034 763
%/TOTAL	100	25.4	25.8	12.5	36.3	100	20.1	15.4	14.1	50.4

be considered. Since noncentral systems are so widely distributed in types, they will be eliminated from further discussion.

3.3 FURNACE DESIGN PRACTICE

With the furnace designs to be considered now determined, the next step is to define a characteristic design for each of these furnace categories.

3.3.1 Central Air Gas Fired

Figures 3 and 4 show some typical gas fired forced central air systems which account for the single largest category.* There are generally three forced air types: upflow, downflow, and horizontal. There is little variation among manufacturers in the overall layout. Typically for an upflow unit, the house circulating air enters the bottom or side of the unit, flows through the main circulating fan, then to the heat exchanger and usually out the top to the distribution ducting. The gas burners are naturally aspirated types consisting of three to four venturies, with distribution pipes consisting of rows of small orifices. The primary air is drawn into the venturi by the gas presssure and premixes with the gas prior to ignition. The primary air/fuel ratio can be controlled through small shutters at each venturi. Secondary air from the furnace room enters around the burners to quench the peak flame temperature and allow complete combustion. The flue products then proceed upward through a parallel flow heat exchanger and exit into a common plenum. Units always include draft diverters which dilute the flue gas stream and prevent downdrafts from blowing out the pilot. Typical operating conditions are tabulated in Table 8.

Several comments concerning this data of Table 8 are appropriate. First, the wide range in excess air levels is due to variations in installations and meterological conditions. Second, the combustion chamber pressures are quite low due to the naturally aspirated burners. Consequently, the heat transfer area is quite large on the flue gas side to prevent excessive pressure drop. This results in fairly low heat transfer coefficient on the hot side and consequently fairly large surface areas. The furnace relies on the stack to create sufficient draft to draw the secondary air through the furnace. Furnace manufacturers frequently claim that reduction in the stack

*In the past there were also a considerable number of gravity or natural draft gas and oil fired central air systems. These have fallen out of favor due to the rather large ducts furnace volume required and higher cost.

TABLE 8

TYPICAL OPERATING CONDITIONS*
Assume 100,000 Btu/hr

Recirculating Air Flowrate (SCFM): 800-1200 SCFM
Excess Air: 20% - 500%
Flue Exit Diameter: 3-1/2" - 5"
Heat Exchange Area: ~ 30 ft^2 - 35 ft^2
Overall Heat Transfer Coefficient: 2 Btu/hr-ft^2°F - 3 Btu/hr-ft^2°F
Exit Flue Gas Temperature (Before Draft Diverter): 450 - 650
Draft Diverter Dilution Air Flow Percent of Flue: 20% - 50%
Combustion Chamber Pressure: ±0.2 "H_2O
Temperature Rise on Air Side: 70°F - 75°F
Overall Steady State Efficiency: 75% - 80%

*Data is compiled from discussion with manufacturers and from References 33, 34, 35, 36, and 37.
**New installation must meet ANSI standards of 75 percent.

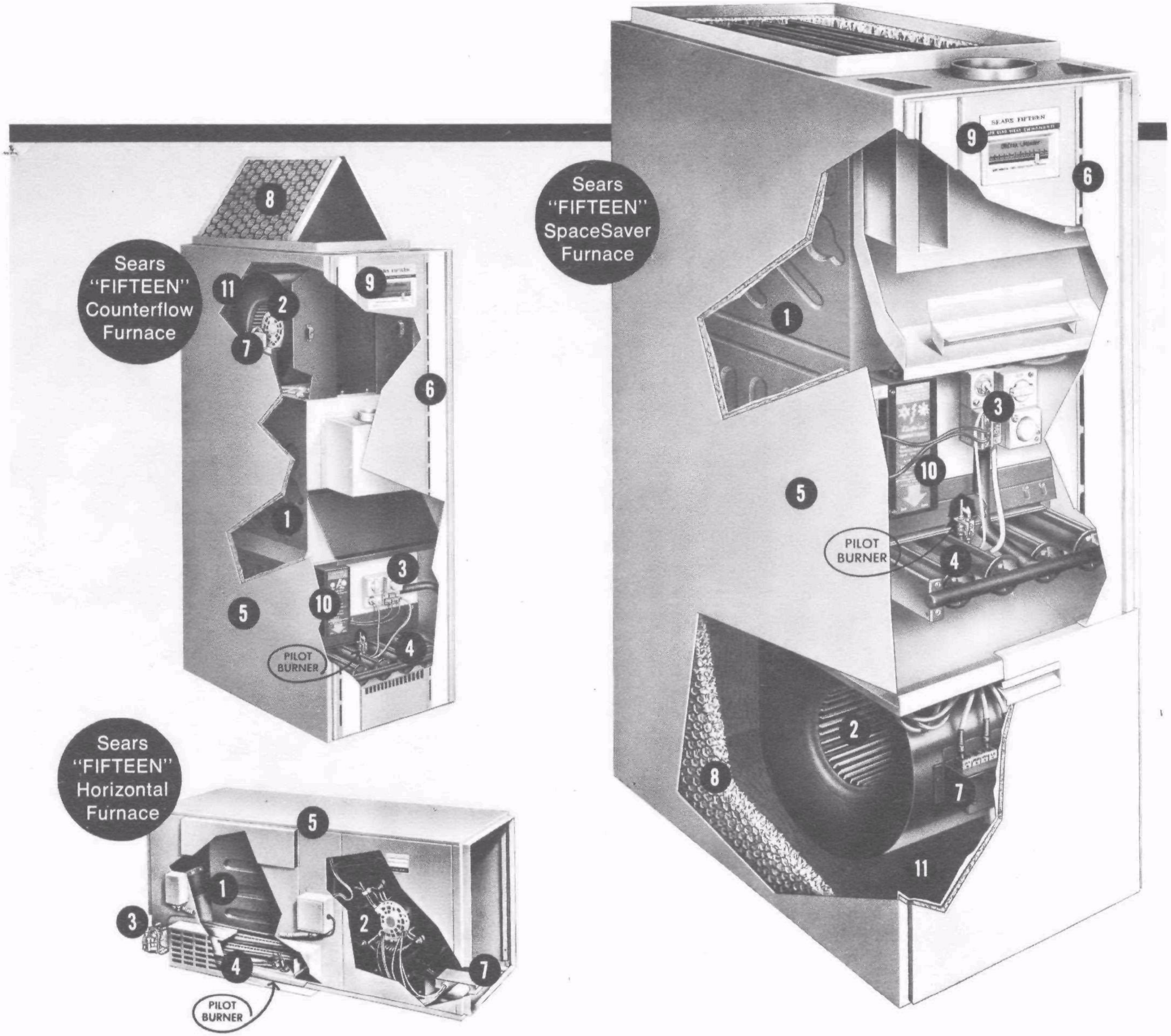
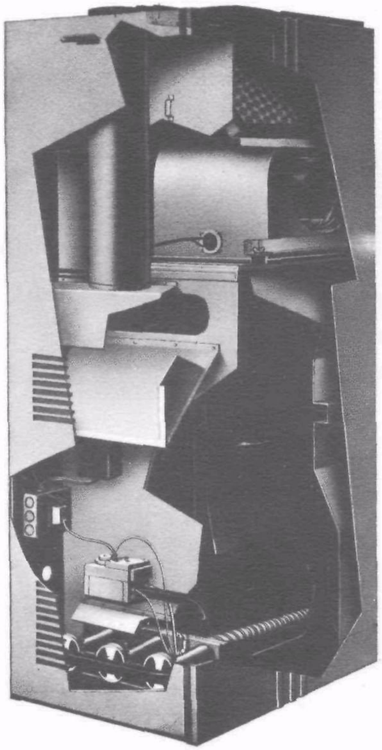
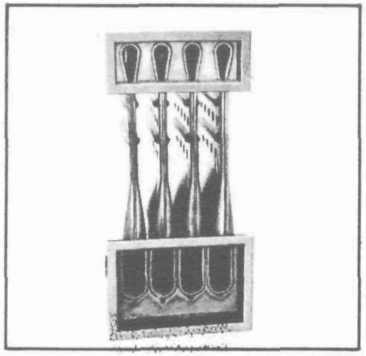


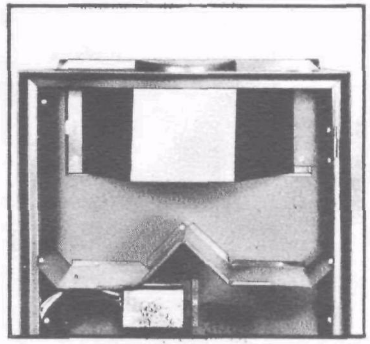
Figure 3. Gas Fired-Forced Air Furnaces
(Courtesy Sears Roebuck & Co.)



UNIVERSAL SLOTTED PORT STEEL BURNER
Burns Natural And Liquefied Petroleum Gases. Stainless Steel Head.



HEAT EXCHANGER



UNIQUE PATENTED DRAFT DIVERTER

Figure 4. Gas Fired Forced Air Downflow Furnace and Details of Components (Courtesy Coleman Co. and Westinghouse)

temperature below 300°F will result in insufficient draft for the burners and cause condensation in the stack. Simple calculations reveal these statements to be approximately correct (see the appendix for calculations). The shape of the heat exchanger in Figure 4 should also be noted. Notice that the base of the unit near the burners is flared to prevent flame impingement on the carbon steel heat exchange surface. It is possible that if insufficient secondary air is drawn in around this surface, it could result in excessive temperatures.

The general design method for these furnaces was characterized through discussions with 19 furnace manufacturers and through the presentations in the Gas Engineers Handbook, ASHRAE Handbook of Fundamentals, ASHRAE Guide and Data book and the Handbook of Air Conditioning, Heating and Ventilating (References 33-36). In summary:

- The procedure is largely empirical and relies on testing and modifications of experimental units to meet AGA/ANSI standards (see Sections 3.3.5).
- Efficiencies are checked on the basis of exit temperature and CO_2 level using charts published by the AGA or ASHRAE (see Section 3.3.5. on efficiency calculations).
- An overall heat transfer coefficient of 2.7 Btu/hr-ft^2°F is recommended by ASHRAE.
- Empirical correlation are available for stack sizing (see References 36, 38).

3.3.2 Central Air-Oil Fired

Figure 5 shows several typical oil-fired forced air furnaces. From the outside, these furnaces look very similar to the gas-fired units and are now being made with overall volumes not much greater than the gas-fired units. There is a considerable internal difference. The heat is usually supplied by a gun type oil burner as pictured in Figure 6. This unit consists of a combustion air blower, motor, damper, fuel pump, ignition system, main air tube and swirlers, and fuel nozzle(s). The fuel flow rate is controlled by the orifice size in the oil nozzle, the total air flow rate by the blower and damper. The proper air/fuel ratio is adjusted using the damper until minimum CO and smoke levels are achieved.

The burner is mounted in a refractory or refractory felt lined combustion chamber which is cooled by the house circulating air. From the combustion chamber the flue gases pass through a gas to air heat exchanger and finally out the stack. Generally there is no draft diverter in the flue because there is no pilot flame and a forced draft system is used. The burner blower

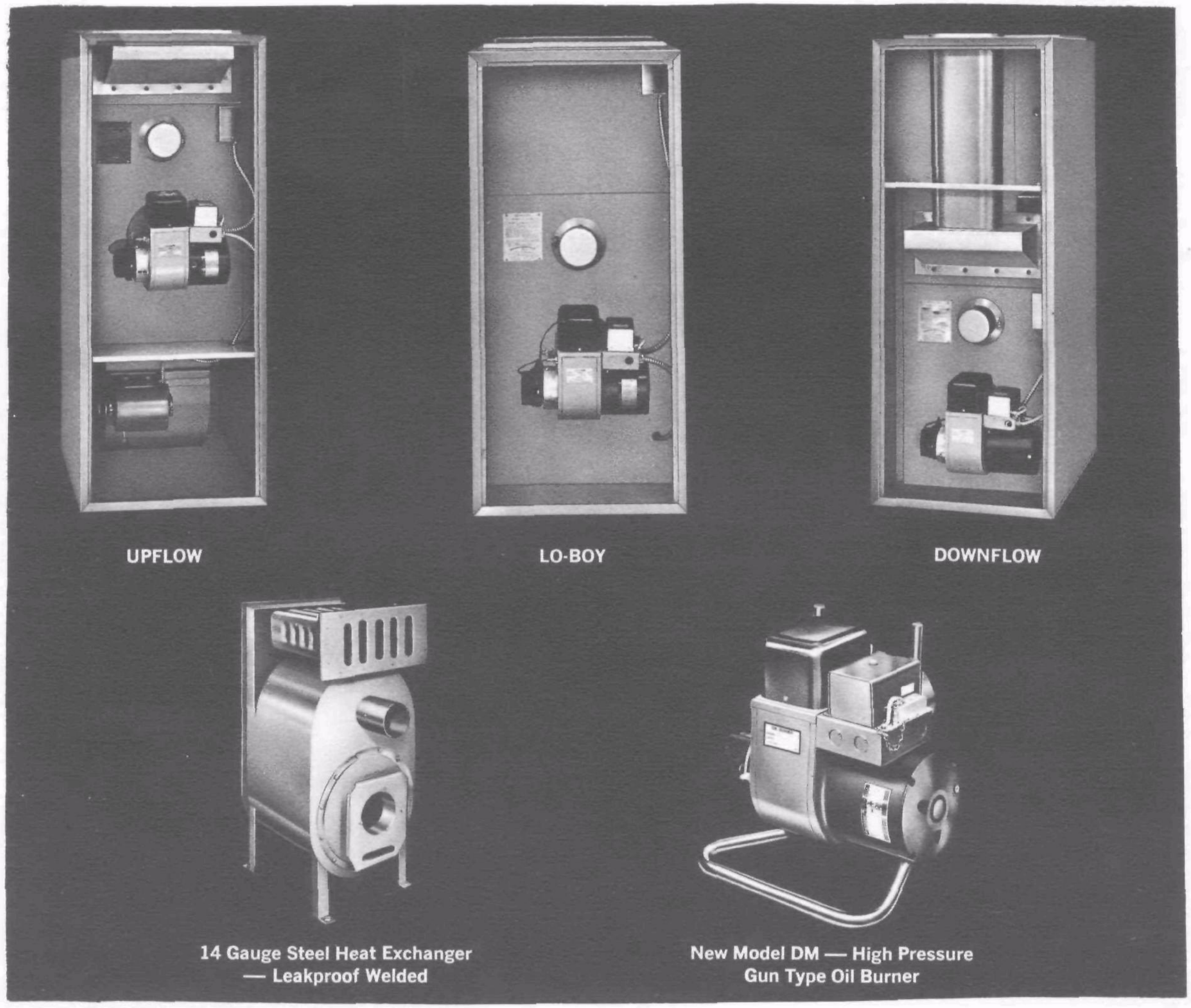


Figure 5. Oil Fired Central Air Furnaces
(Courtesy Day & Night Co.)

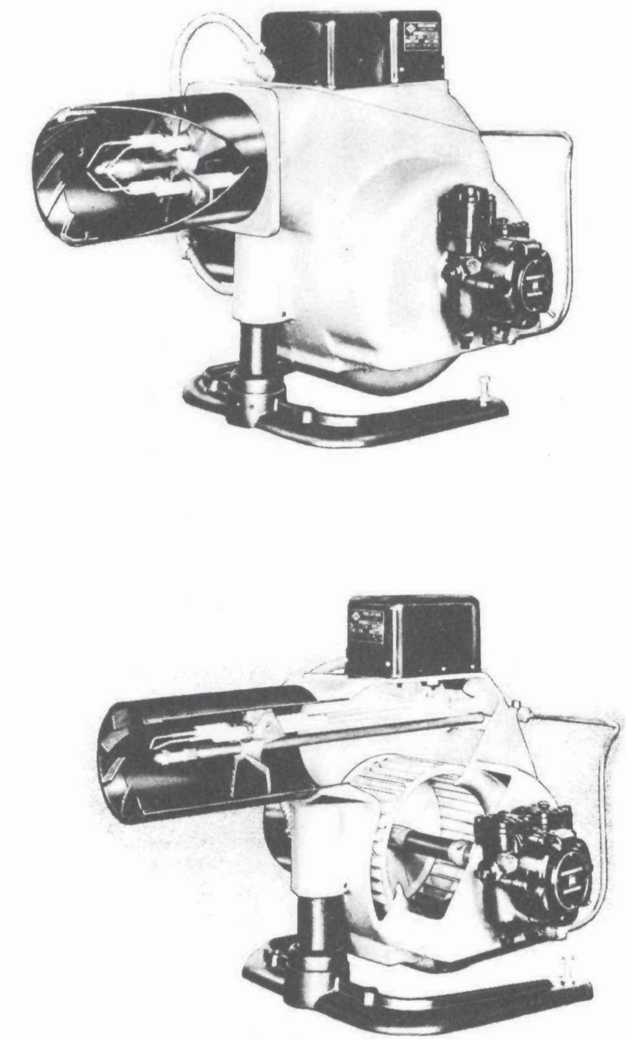


Figure 6. Typical Oil Burners
(Courtesy Johnson Oil Burner Co.)

may supply the full pressure to exhaust the flue from the stack or it may rely partially on the bouyancy forces downstream of the furnace. Table 9 lists typical operating condition.

Again the design procedure is quite empirical and relies on prototype development, experience, and testing.

Efficiency and standards for construction are set by Underwriters Laboratory and ANSI.

3.3.3 Hot Water-Gas Fired

Figure 7 shows a typical gas fired-hot water or hydronic heater. New units are usually considerably more expensive than forced air systems and therefore are now being made in the larger size ranges of 130,000 Btu/hr and higher. However, old units are still quite prevalent in the Northeastern and Northcentral regions of the U.S. The gas systems on the new units are quite similar to the central air furnace utilizing the naturally aspirated burners. Typical operating and design conditions for both oil and gas systems are given in Table 10.

Heat transfer area is governed by the gas side transfer coefficient which is once again limited by the pressure drop requirements of the naturally aspirated burners. Further details on the design procedures of these units were not investigated.

3.3.4 Hot Water-Oil Fired

Figure 8 shows the oil fired-hot water system. Again, although quite prevalant in older homes in the Northeast and Northcentral States, they are now quite costly and are limited to the larger size ranges. The design approaches that of larger hot water boilers and these units are similar in some respects to the oil fired central air designs in that they employ a refractory lined combustion chamber followed by the heat exchange surface. However, no draft diverter is required. It has been reported that poor matching of burner and combustion chamber requirements occurs between the burner manufacturer and furnace manufacturer (Reference 39). Typically, considerable leakage of secondary air into the firebox can occur, lowering flame temperatures. In many cases this is necessary to prevent damage to the combustion chamber but leakage probably results in overall lower furnace efficiency.

TABLE 9

OIL FIRED CENTRAL AIR SYSTEMS

Typical Operating Conditions
Recirculating Air Flow: 800 -1200 SCFM
Excess Air: 10% - 100%
Flue Exit Diameter: 5" - 7"
Heat Exchange Area: 20 ft^2 - 30 ft^2
Overall Heat Transfer Coefficient: 2 Btu/hr-ft^2°F - 3 Btu/hr-ft^2°F
Exit Flue Gas Temperature: 500°F - 900°F (Older Units)
Combustion Chamber Pressure: .05 "H_2O - 0.2 "H_2O
Temperature Rise on Air Side: 75°F - 80°F
Overall Steady State Efficiency: 70% - 75%

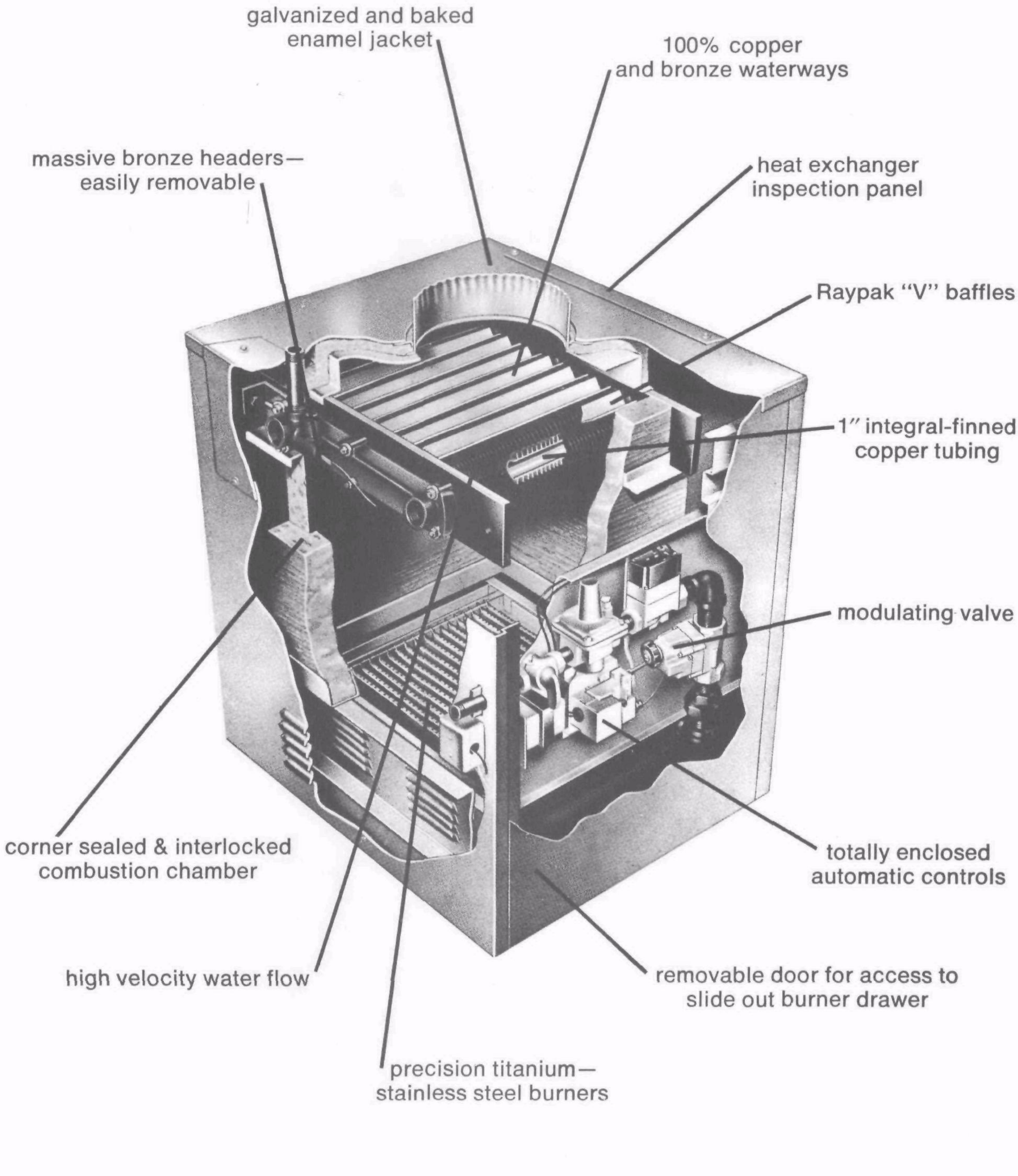


Figure 7. Gas Fired Hydronic Boiler (courtesy of the Raypack Co.)

TABLE 10

OIL AND GAS FIRED-HOT WATER
TYPICAL OPERATING CONDITIONS OF DESIGN PRACTICE
100,000 Btu/HR INPUT

	Gas	Oil
Recirculating Water Flows:	3-15	3-15
Excess Air (percent):	20-500	30-100
Flue Exit Diameter:	3 1/2-5	3 1/2-5
Exit Flue Gas Temp (Upstream Draft Diverter):	500-600	400-600
Exit Flue Gas Temp (Downstream Draft Diverter):	300-400	-
Combustion Chamber Pressure:	.05-2	0.2
Temperature Rise on Water Side:	10-40	10-40
Overall Steady State Efficiency:	75-80	70-75

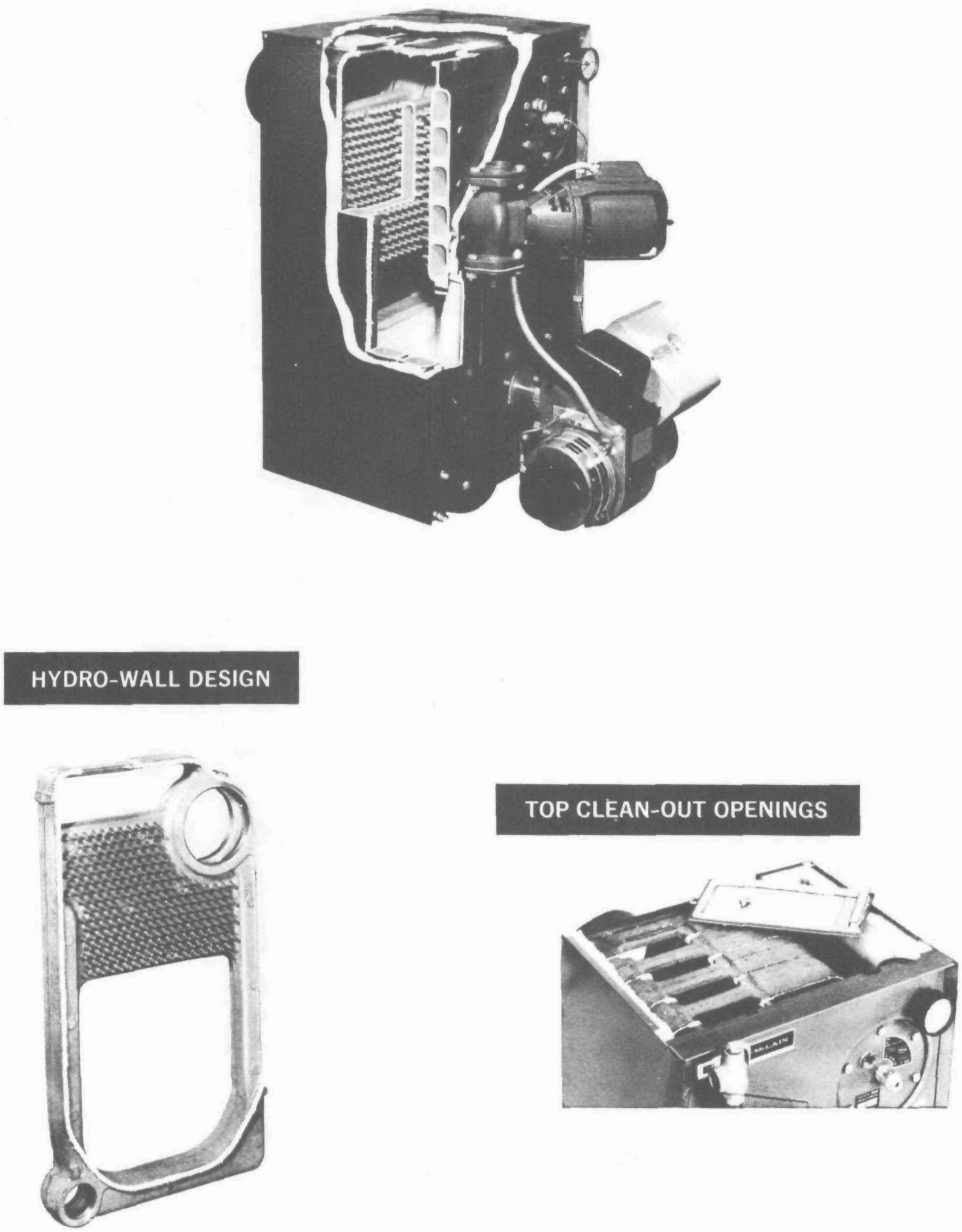


Figure 8. Oil Fired-Hydronic System
(Courtesy of Weil-McLain Co.)

3.3.5 Furnace Efficiencies and Standards

As stated in the previous sections, furnaces must meet certain standards sanctioned by the AGA through ANSI or by U.L. to carry their seals.

These institutes are composed of members from trade association and government agencies, and insurance organizations. Each standard is developed and updated by a committee composed of representatives of concerned groups. Each standard itself is a 30- to 40-page document which covers nearly every aspect of the system from the efficiency to motors, wiring, sheet metal, burners, and safety systems. Table 11 gives a brief listing of the various ANSI and U.L. standards pertinent to this study and the recommended efficiencies. It will be noticed that the efficiency for gas fired forced air central furnaces is 75 percent. However, standard practice in the industry is currently about 80 percent.

These efficiencies are determined, however, under controlled steady state conditions. Actual efficiencies in the home will depend on the given installation (closet, basement, garage), air flow to the furnace, wind and atmospheric conditions, effects of cycling to satisfy heating load, pilot usage, standby losses, condition of air filter and air settings on the furnace. In addition, the furnace efficiencies will usually degrade with time due to accumulations of soot, nozzle fouling, filter plugging, fan belt slippage, and similar degradation effects.

In 1966 the AGA laboratories conducted performance tests on identical gas and oil fired hot water boilers. They attempted to quantify the "service efficiency" to take into account these losses. Table 12 shows the results of these tests (Reference 40). These data show the service efficiencies to be 1 to 8 percentage points below the thermal efficiency depending on cycle on/off time. In general, the data showed the gas furnace to be slightly more efficient than the oil units while not deteriorating with age. Another interesting study in 1970 by Stricker (Reference 41) attempted to measure the efficiencies of two forced air furnaces in the "as found" condition in the field over a six-month period. Figures 9 and 10 from this study show the typical temperature rise on the air side during a cycle. Figure 11 shows the resulting average energy balance for the oil fired and gas fired furnaces. The seasonal efficiencies were calculated to be 61 percent for oil and 62 percent for gas. The authors warn that these values cannot be considered typical and that a large number of furnaces and structures must be studied to obtain typical values of seasonal efficiency.

TABLE 11

LISTING OF STANDARDS FOR HVAC EQUIPMENT WHICH INCLUDE EFFICIENCY

ANSI Z31.11 - 1974

Gas-Fired Room Heaters

Volume I (Z21.11.1) Vented Room Heaters, Greater than 20,000 Btu/hr - Efficiency 70 percent based on total volume of gas; Less than 20,000 Btu/hr - Efficiency 65 percent.

Volume II (Z21.11.2) Unvented Room Heaters - 1974 - Performance - No number for efficiency.

ANSI Z21.13 - 1972

Gas-Fired Steam and Hot Water Boilers

Shall have a thermal efficiency of 75 percent based on total heating value of the fuel.

ANSI Z21.34 - 1971

Gas-Fired Duct Furnaces

Thermal efficiency of 75 percent based on total heating value of fuel.

ANSI Z21.43 - 1968

Unvented Gas-Fired Infrared Radiant Heaters
Radiant efficiency of at least 35 percent.

ANSI Z21.44 - 1973

Gas-Fired Gravity and Fan Type Sealed Combustion System Wall Furnaces

Gravity type - Efficiency 70 percent based on total heating value of fuel.

Fan type - Efficiency 75 percent based on total heating value of fuel.

ANSI Z21.47 - 1973

Gas-Fired Gravity and Forced Air Central Furnaces

Gravity type - Thermal efficiency 70 percent based on total heat value of fuel.

Fan type - Thermal efficiency 75 percent based on total heating value of fuel.

ANSI Z 21.48 - 1973

Gas-Fired Gravity and Fan Type Floor Furnaces

Gravity type - Efficiency 65 percent based on total heating value of fuel.

TABLE 11 (Concluded)

Fan type - Efficiency 70 percent based on total heating value of fuel.

ANSI Z21.51 - 1971

Vented Gas-Fired Infrared Radiant Heaters

Efficiency - shall have a radiant efficiency of at least 35 percent.

ANSI Z21.52 - 1971

Gas-Fired Single Firebox Boilers

Steam and hot water boilers shall have efficiency 75 percent based on total heating value of fuel.

ANSI Z21.53 - 1967

Gas-Fired Heavy-Duty Forced Air Heaters

Shall have thermal efficiency of not less than 75 percent based on total heating value of fuel.

ANSI Z91.1 - 1972

Oil-Powered Central Furnaces

Performance - Output capacity in Btu/hr 0.75 times input.

TABLE 12

THERMAL AND SERVICE EFFICIENCIES OF RESIDENTIAL HOT WATER BOILERS
(Boiler A Differs From Most Contemporary Oil-Fired Equipment)
(Reference 40)

Boiler and Condition	Thermal Eff., %	Standby Loss % of Input	Service Eff. % ("On/Off" Time, Min.)		
			20/10	15/15	10/20
Boiler A (Oil)					
- New	76.2	2.02	75.50	74.80	71.30
- After 6 mos.	74.8	----	74.10	73.30	70.00
- After 10.5 mos.	73.43	----	72.87	72.06	68.78
Boiler B (Oil)					
- New	72.60	2.40	71.50	70.00	67.10
- After 6 mos.	71.50	----	70.30	68.80	66.00
Boiler C (Gas					
- New	76.80	3.00	75.30	73.70	68.40
Boiler D (Gas)					
- New	76.78	3.14	75.90	74.20	70.40
- After 6 mos.	76.78	----	75.90	74.20	70.40

NOTE: Electric consumption of oil boiler accessories included in input.

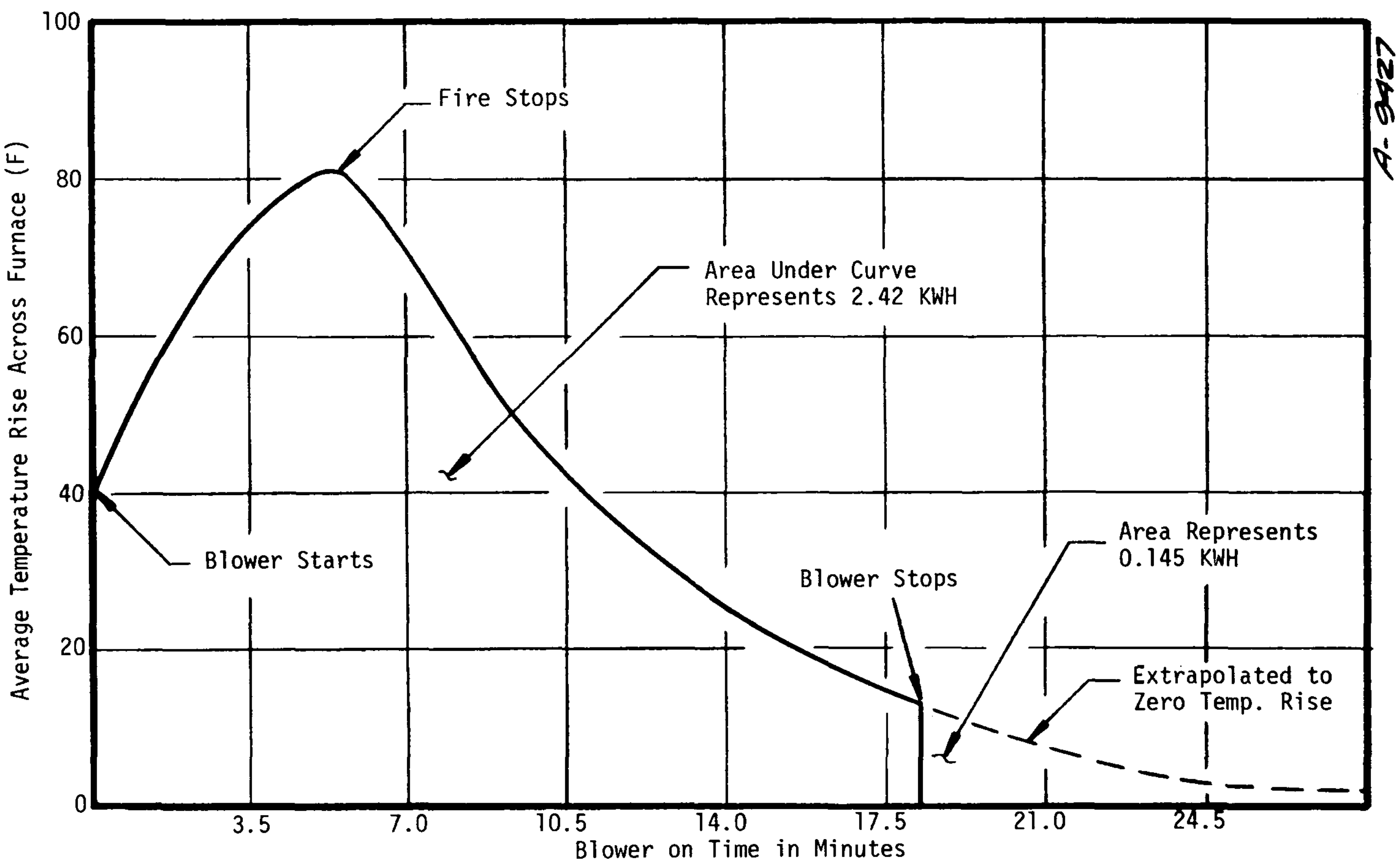


Figure 9. Temperature Rise Across Oil Furnace During a Typical Normal Cycle (Reference 41)

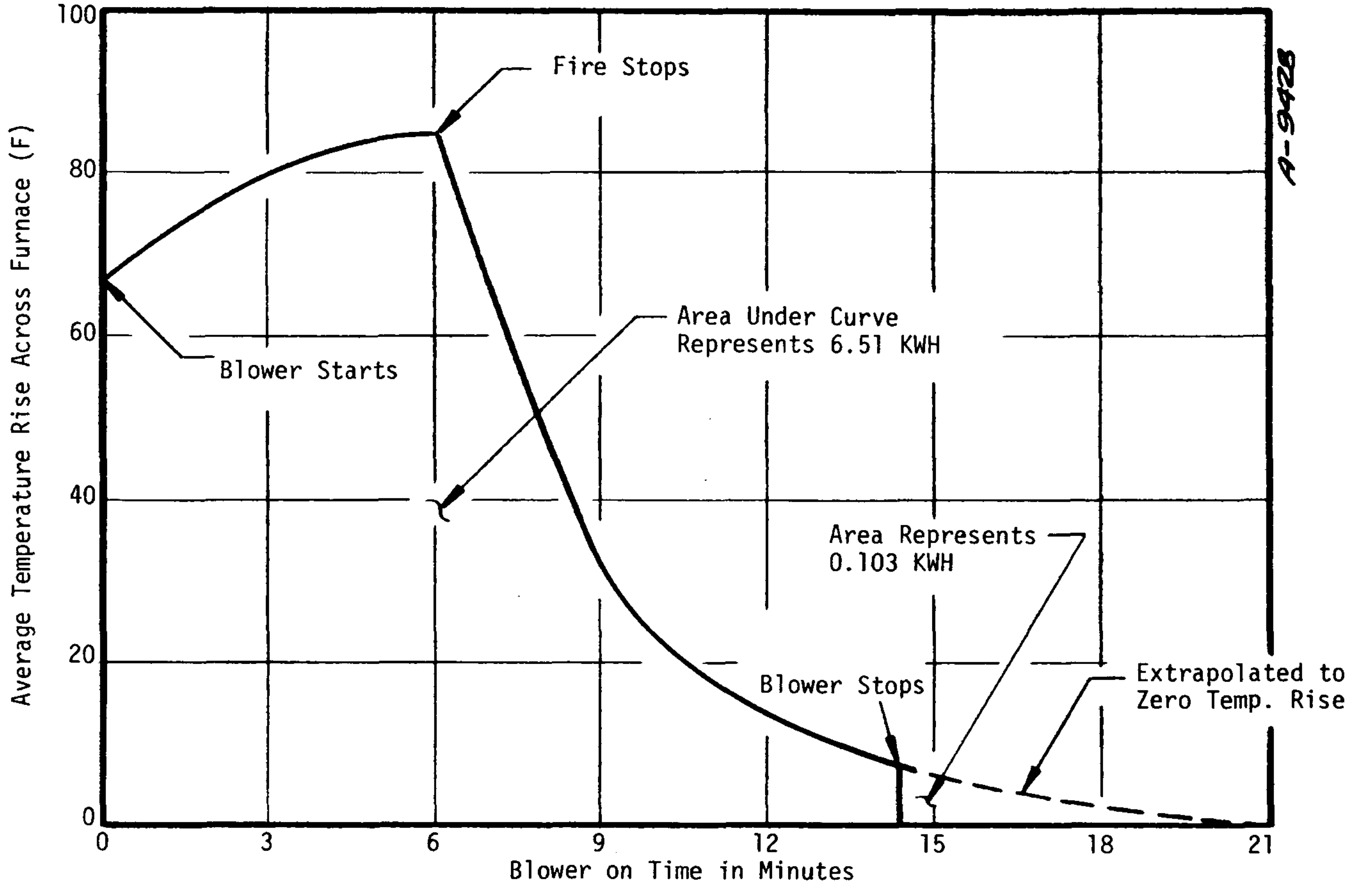
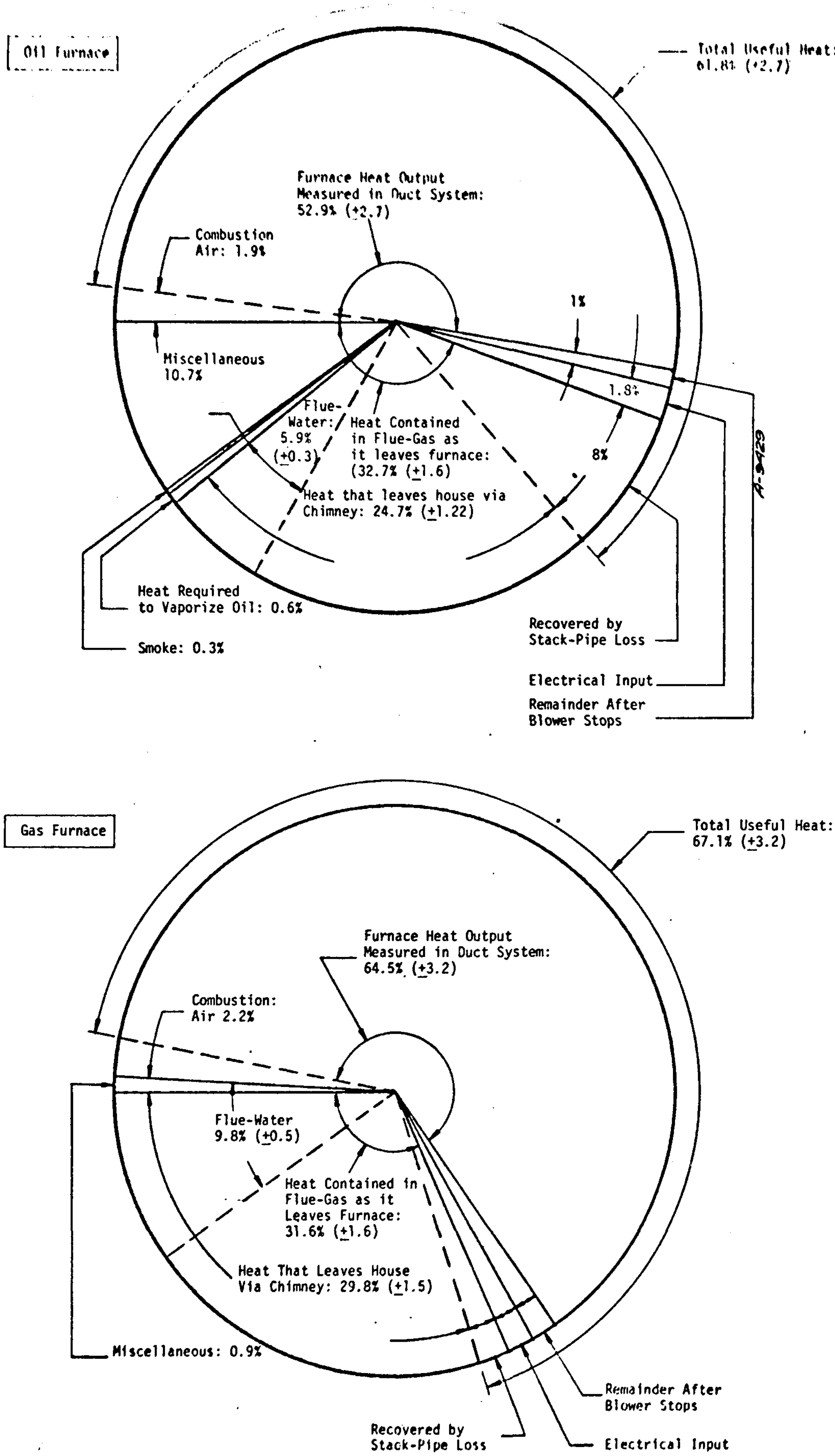


Figure 10. Temperature Rise Across Gas Furnace During A Typical Normal Cycle (Reference 41)



Total Accounted Energy: 99.1% (±3.6)

Figure 11. Energy Balance for a Particular Cycle of Oil Furnace (Top) and of Gas Furnace (Bottom). (Reference 41)

A significantly lower value was reported by Dunning, Loeary, and Thumbower as part of the Westinghouse Energy Utilization Project (Reference 42). Their analysis determines the furnace efficiency of gas furnace to be 47 percent based on an hour by hour computer study of the heat losses and gains of a characteristic house. A furnace efficiency of 42 percent was reported for the equivalent house heated with an oil furnace. Their analysis takes into account the heat provided by a variety of other heat sources, such as appliances, occupants, and the sun, and the heat required to warm to room temperature that outside air infiltrating to replace air going up the flue.

Similar efficiencies were reported by Kurylko (Reference 43) in a study of 1260 oil fired home heating plants in the Boston area. He found thermal efficiences as low as 48 percent, although 15 percent of the oil burners were found operating close to optimum with efficiencies above 80 percent.

These significantly lower overall efficiencies may be partially explained by referring back to Figures 9 and 10 which show the typical temperature rise on the air side. Notice that the fire is extinquished at the peak temperature rise on the air side. It is at this point the peak ("certified") efficiency is probably achieved. For the remainder of the cycle the air blower remains on but the flue gas temperatures and flowrates have dropped. Perhaps more important and what is not shown on this curve is that the burner has been firing for several seconds prior to the blower on cycle. This again means that there is a significant period of time when there is no flow (or very little flow) on the air side and therefore inefficient heat exchange is occurring. Of course, some of this heat is stored as heat capacitance in the furnace structure to be eventually recovered but much also escapes out the stack under inefficient conditions.

An oversized furnace will in addition accentuate the cycling phenomena. Although the methods outlined by ASHRAE for determining the heat load in a structure [see Chapter 21, ASHRAE Handbook of Fundamentals (Reference 34) or Chapter 2 of Strock, Handbook of Air Conditioning, Heating and Ventilating (Reference 36) are quite detailed, many residential furnaces are grossly oversized to assure a wide margin of safety. It might be possible to design a modulated furnace that would avoid these severe inefficient transients and control steady state heat output closely to the required heat demand.

Any study of a modulated furnace must explore whether cycling allowed use of lower cost materials by avoiding higher steady state temperatures. On the other hand, the problem associated with thermal fatigue due to cycling may not be as severe with a modulated system. This area warrants further study.

In addition using flue shut off dampers during off cycles to prevent exfiltration of air through the stack may actually result in greater saving than reported by Hittman (Reference 10). The argument is made that low efficiency heat exchange would be prevented that would normally allow low temperature gases to pass out of the furnace. The Save-Fuel Corporation which manufactures the Vent-O-Matic damper (see Section 3.5.3) claims savings of 20 to 30 percent of the space heating requirement.

In any case, during the steady state portion of the cycle (if one really exists), from 20 to 30 percent of the input energy escapes from the stack. Addition of a heat recovery device or a higher efficiency furnace (steady state) would not only improve the steady state efficiency but quite likely the overall efficiency.

3.4 HIGH EFFICIENCY FURNACE AND HEAT RECOVERY DEVICE RESEARCH

During the course of the study portion of this task a number of manufacturers or associations were contacted which had conducted or are conducting studies of high efficiency furnaces or heat recovery devices.* For example, Chevron Research are presently doing a cost effectiveness evaluation on commercially available heat recovery devices. This study will lead to the development of improved devices. However, all this information is proprietary. Other large manufacturers, such as GE, indicated that they have built or are designing higher efficiency furnaces (> 90 percent).

Electric Furnace Man (EFM) has developed a hydronic furnace with 89.5 percent efficiency (Reference 44). A recent development is the Heat Transfer Module developed by AMANA (Reference 44). A solution of ethylene glycol and water is heated in the "Heat Transfer Module" (HTM) and then passed through the air conditioning coils to transfer the heat to the room air. The HTM and furnace is pictured in Figures 12a and 12b. The natural gas fuel burns on a central cylinder and passes readily through a porous matrix carrying the heat transfer liquid. One unique feature of this unit is that the gas is ignited with spark ignition rather than a pilot flame. In addition a forced draft

*Most research, however, has been in the area of heat pumps rather than in improvements of conventional systems. The Westinghouse Utilization project (Reference 42) showed that there was overall cost benefit for heat pumps over gas and oil system in the majority of the locations studied. Since the subject of heat pumps, heat pump-solar energy systems, and total residential energy conservation projects utilizing heat pumps is covered in detail in other references (Reference 2), no further discussion will be given here. Similarly, considerable research is being conducted in the academic community as was indicated in Table 1, particularly in the area of solar energy research. However, no specific academic programs were found relating to a high efficiency furnace or heat recovery device.

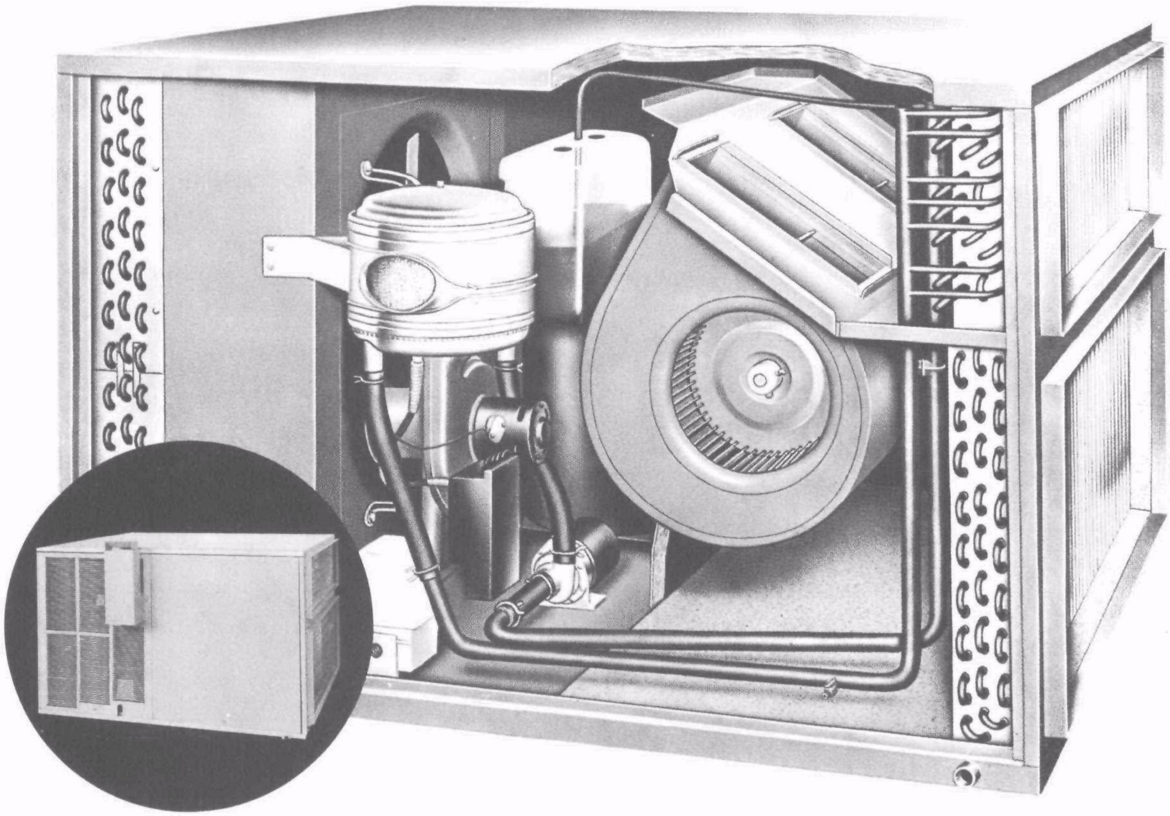
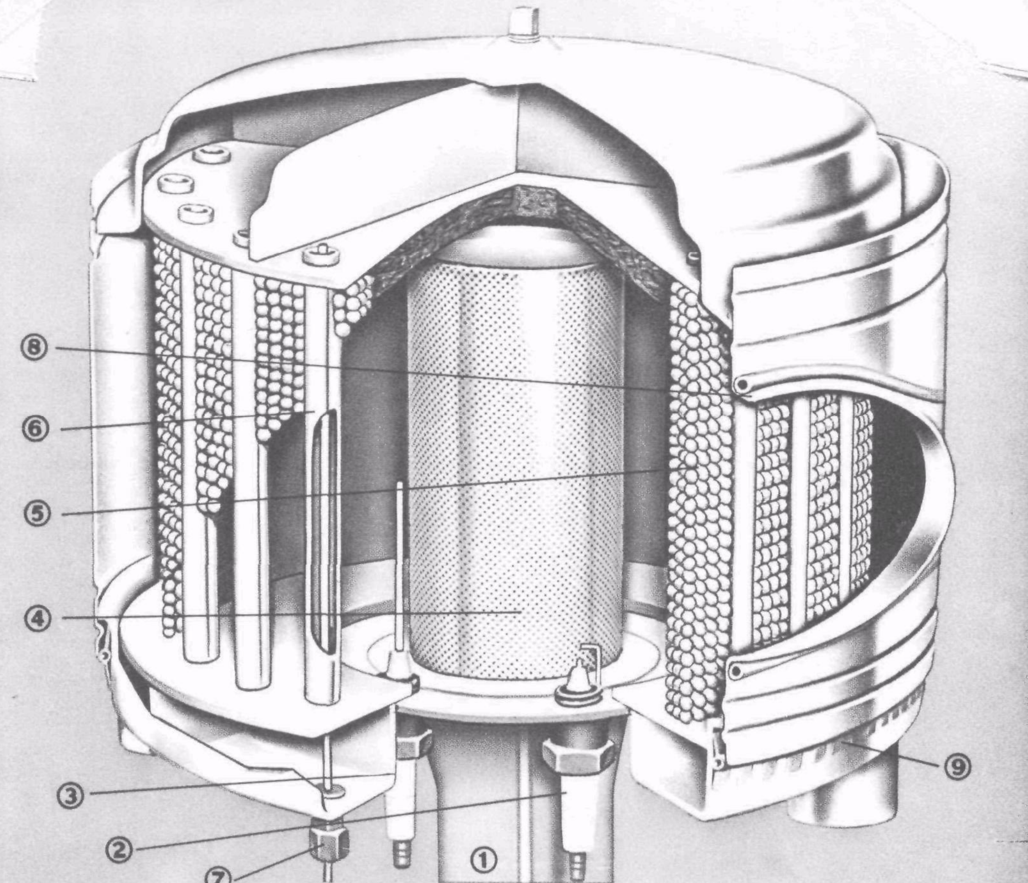


Figure 12a. AMANA Electric/Gas Heating Cooling Unit Using the HTM*



The first major breakthrough in heating technology in years.

The Exclusive Amana® HTM*

It's only 9" high and 9" in diameter. It's the most advanced and compact heating system available today.

Here's how it works:

(1) Gas (natural or propane) is pulled from the gas valve by the combustion blower into the HTM* Heat Exchanger burner. The burner is only 4" high x 2" in diameter.

(2) A spark plug ignites the gas and air mixture in the burner. No wasteful, bothersome pilot light. Wind does not affect its operation.

(3) A flame probe monitors the burner to give proof of combustion. If combustion doesn't take place, the flame probe will close the gas valve within 15 seconds.

(4) The stainless steel burner provides 9000 tiny flames which produce extremely hot flue gases.

(5) These gases passing through the porous matrix create high turbulence to produce rapid heat transfer. Exclusive porous matrix is made up of thousands of copper coated steel balls fused together to perform the function of heat exchange.

(6) The solution (50% water and 50% ethylene glycol to prevent freezing) carrying tubes embedded in the matrix pick up the heat and transfer the hot solution, moving at 4.7 feet per second, through the HTM* Heat Exchanger.

(7) A limit control monitors the temperature of the solution. And, if it rises above design temperatures, it shuts down the system.

(8) You get 7% - 10% more usable heat than industry standards (depending on the firing rate) from the gas burned because of less heat loss through the flue.

(9) The heater keeps the temperature of the HTM* Heat Exchanger above surrounding temperature at all times, so it remains dry and untarnished by atmospheric moisture.

Figure 12b. AMANA Heat Transfer Module

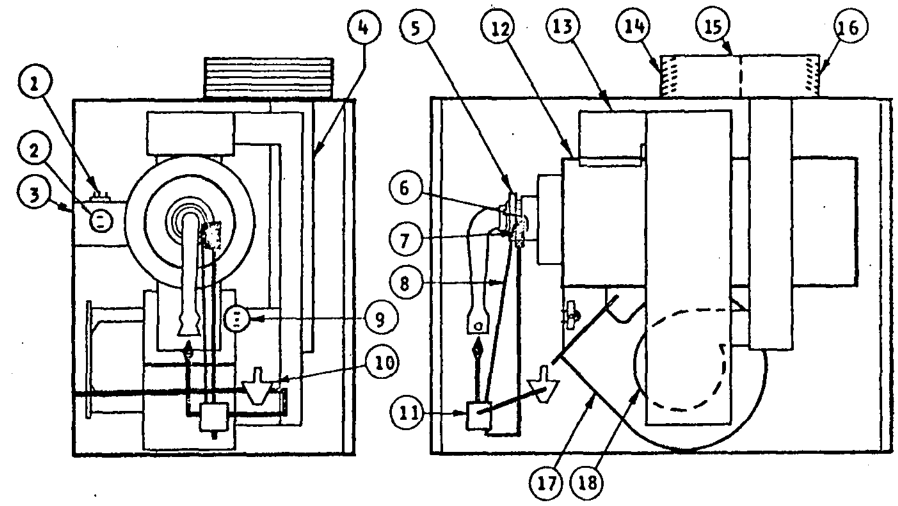
combustion air fan is used to overcome the pressure drop of the combustion chamber and heat exchange surfaces. No draft diverter is required. Overall efficiencies, however, range between 75-89 percent depending on load.

The American Gas Association has been conducting studies for the past 10-15 years on improving heat exchange performance and evaluating new or different ways to heat the home. DeWerth and Smith (Reference 21) studied the effects of flame aeration, quantity and direction of recirculating air flow, internal flue baffling and heat exchanger materials of construction on heat transfer efficiency. Experimental application of the developed optimum heat transfer conditions produced on 18.5 percent increase in element loading. Since this data are now about 13 years old, it is quite likely most of the results have been incorporated into present designs.

DeWerth, Schaab, and Hellstern (Reference 19) studies several new ways of heating localized areas around the home. Seven experimental units were developed which operate at an efficiency of 80 percent or more. Many of the prototype units can be scaled to sizes and capacities of central heating systems. The following designs were investigated:

- Baseboard Designs
  - Direct-Fired Tube-In-Tube Baseboard Convector
  - Liquid Filler Baseboard Heater
  - U Tube Direct-Fired Baseboard Convector
- Outdoor Installation
  - High Temperatures, High Velocity - Forced Air Baseboard Heater
- Radiant Types
  - "Infra-Vector" Heater
  - Radiant Bathroom Heater
  - Metallic Radiant Heater

Of particular interest is that three of the designs have "power" exhaust systems necessitated by higher pressure drop designs. One of these designs, the high velocity, high temperature forced air baseboard heater, is shown in Figure 13. It is possible that this design could be scaled to central heating system size.



1. SINGLE POLE DOUBLE THROW FAN LIMIT CONTROL
2. HIGH TEMPERATURE LIMIT CONTROL
3. HOT AIR FEED TO ROOM
4. DUCT FROM FLUE EXHAUST BLOWER TO OUTSIDE
5. BURNER AND VENTURI
6. PILOT SWITCH
7. PILOT
8. BLEED TUBE FOR VALVE
9. PRESSURE SWITCH - (VALVE CONTROL)
10. PRESSURE REGULATOR
11. 110 VAC GAS VALVE
12. HEAT EXCHANGER
13. FLUE DUCT TO FLUE EXHAUST BLOWER
14. INLET AIR SUPPLY OPENING
15. VENT CAP FOR BALANCED FLUE SYSTEM
16. FLUE PRODUCT'S EXHAUST OPENING
17. CIRCULATING AIR BLOWER
18. FLUE EXHAUST BLOWER

Figure 13a. The High-Velocity, High-Temperature, Forced-Air Baseboard Heater System

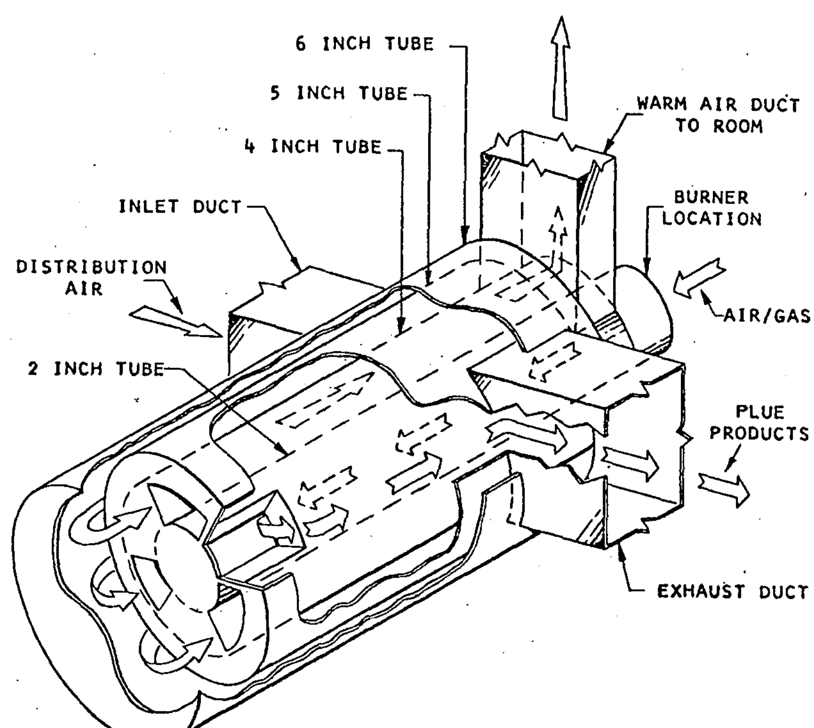


Figure 13b. Section View of the Heat Exchanger and Air Flow

In a later AGA study Griffiths and Niedzwiechi (Reference 20) investigated several novel high efficiency concepts for central heating systems. Figures 14 through 23 show the following schemes considered:

- Ceramic heat exchanger forced air furnace
- Four pulsed combustion furnaces
- Warm air heating panel
- Warm water heating panel
- Direct force wall panel
- Snorkel-vented dual wall furnace with bypass arrangement
- Forced air furnace with integral blower heat exchanger

All of these furnaces were constructed on an experimental basis and found to operate with efficiencies of 75-85 percent or higher. Considerable experimental data is reported in Bulletin 101 on each of these designs. To our knowledge none of these designs has found a commercial marketplace nor has further research been carried out except for the pulsed combustion concept. This bulletin also reports the results of an experimental investigation which increased the heat exchange area of a conventional gas fired central heating furnace. The draft hood of the furnace was removed and combustion products were passed through a secondary heat exchanger. An exhaust blower was placed at the outlet of the secondary heat exchanger to vent the combustion products. The flue losses were reduced from 22.5 percent to 12 percent with the same mass flow through the unit. The modified unit was also tested under natural draft conditions and found to be unstable during the first five minutes of operation. Steady condensation of water vapor took place during these tests. The report points out the following precautions if a condensing system is used:

- Corrosion resistant materials would be required, particularly if trace amounts sulfur are present in the exhaust gas stream.
- All ducting would have to have watertight joints and seams.
- A forced draft system might be needed.

AGA is continuing to do research on the pulsed combustion concept. The latest Research and Development Bulletin (Reference 18) of AGA describes the work under way:

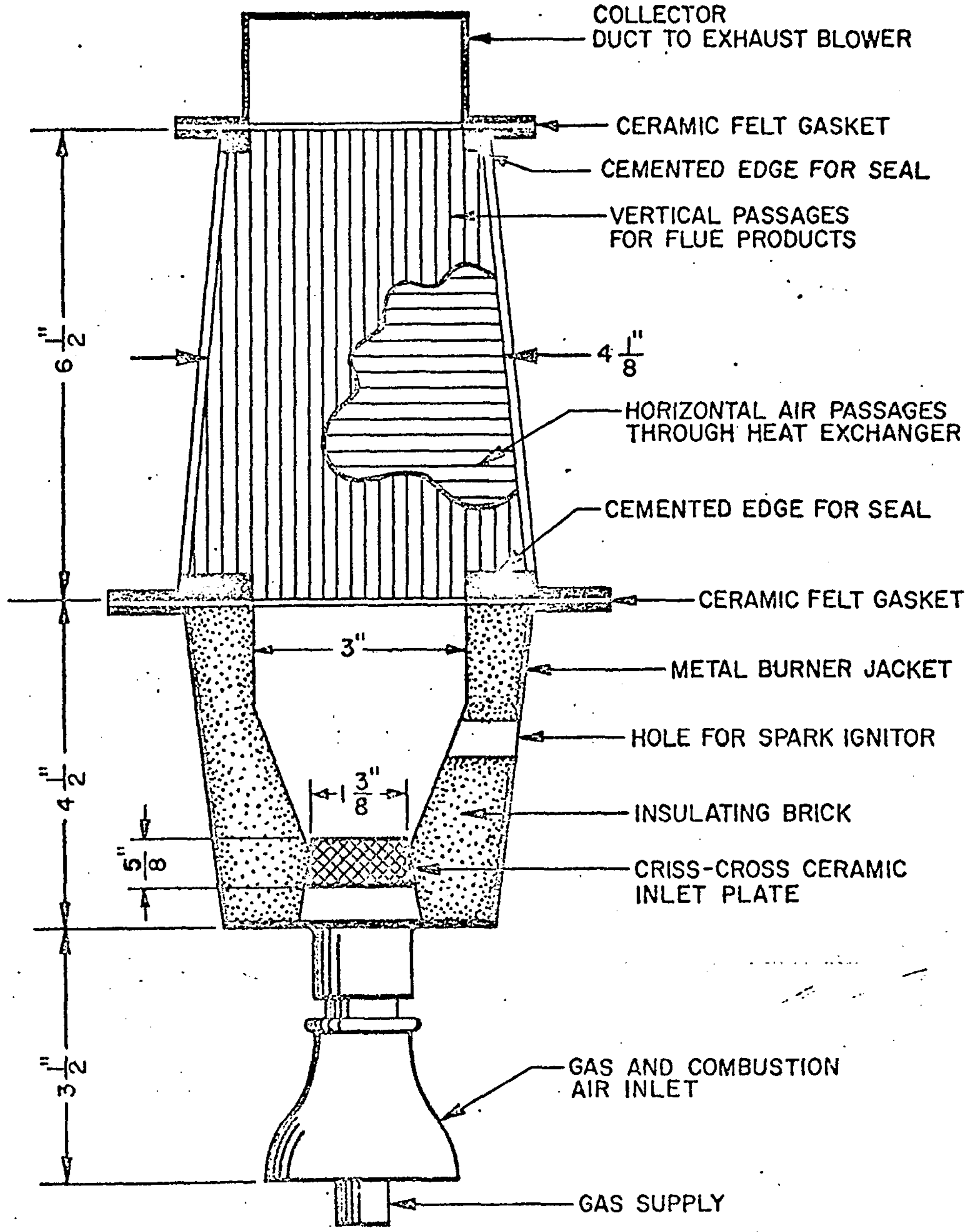


Figure 14a. Cross Section of Ceramic Heat Exchanger-Burner Unit

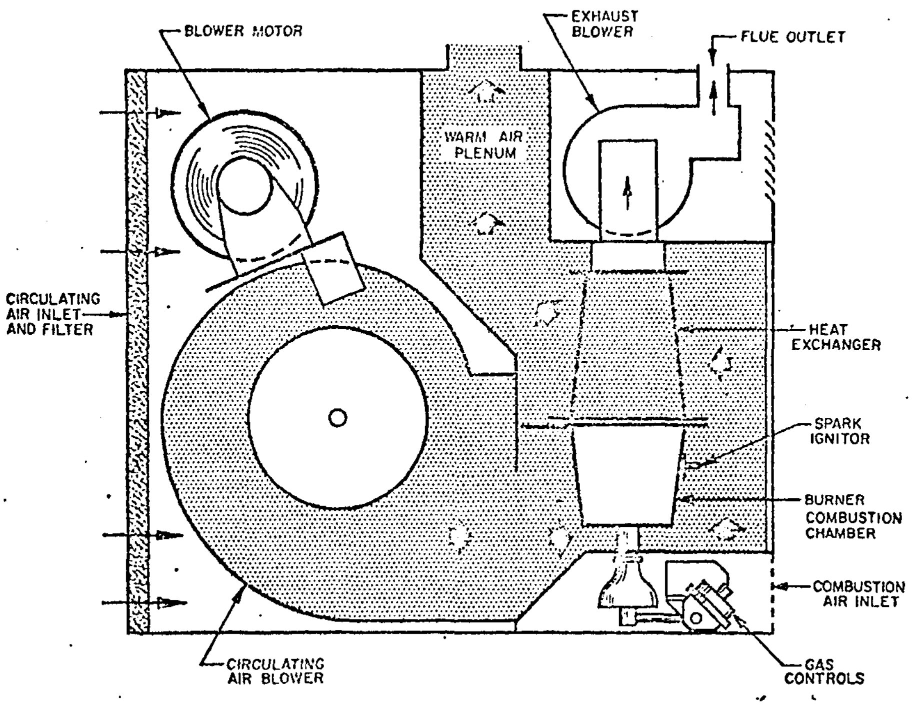


Figure 14b. Cross Section of Ceramic Heat Exchanger Force-Air Furnace

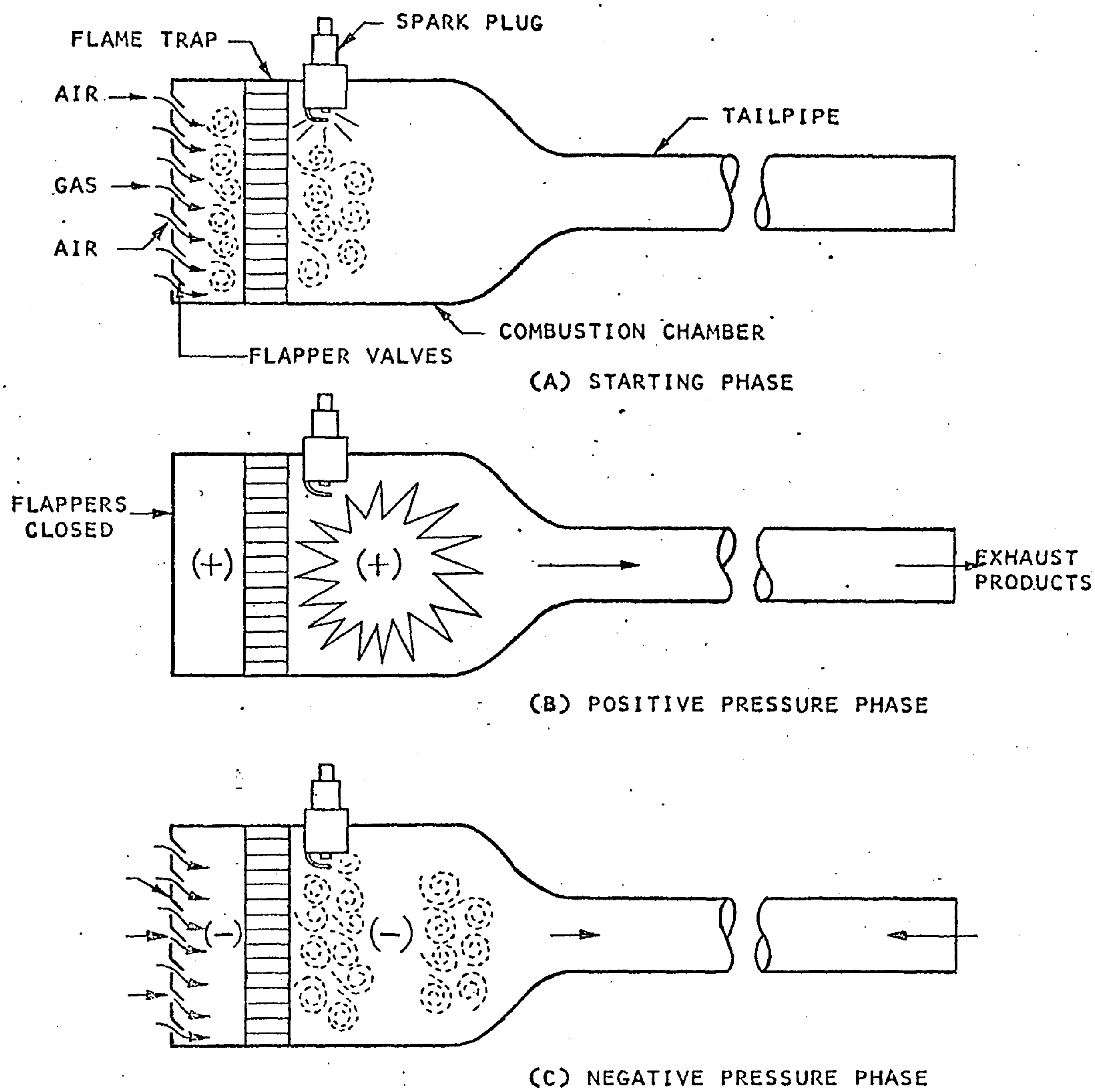


Figure 15. Schematic Illustration of the Pulse Combustion Process

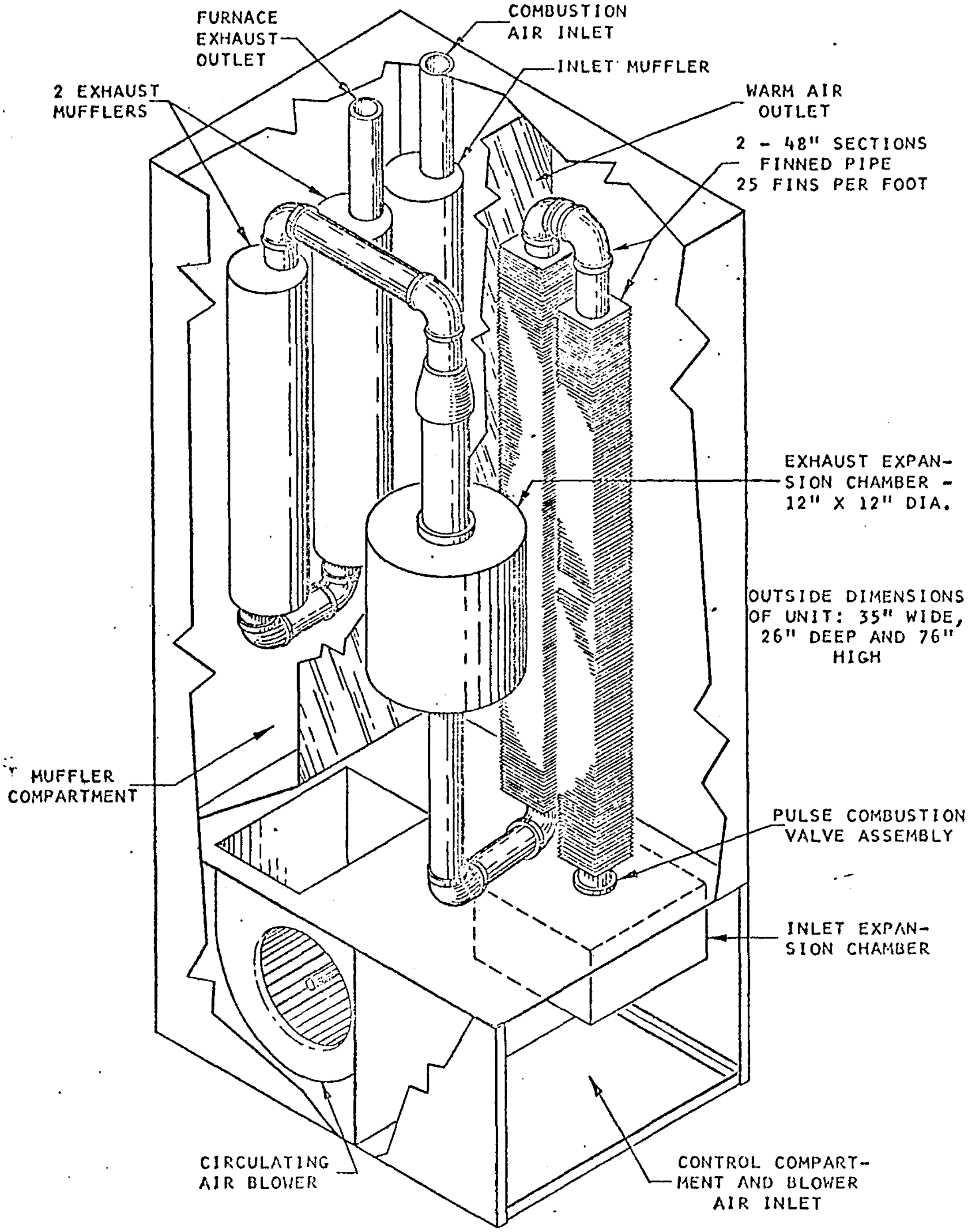


Figure 16. An Experimental "Finned Tube" Pulse Combustion Furnace

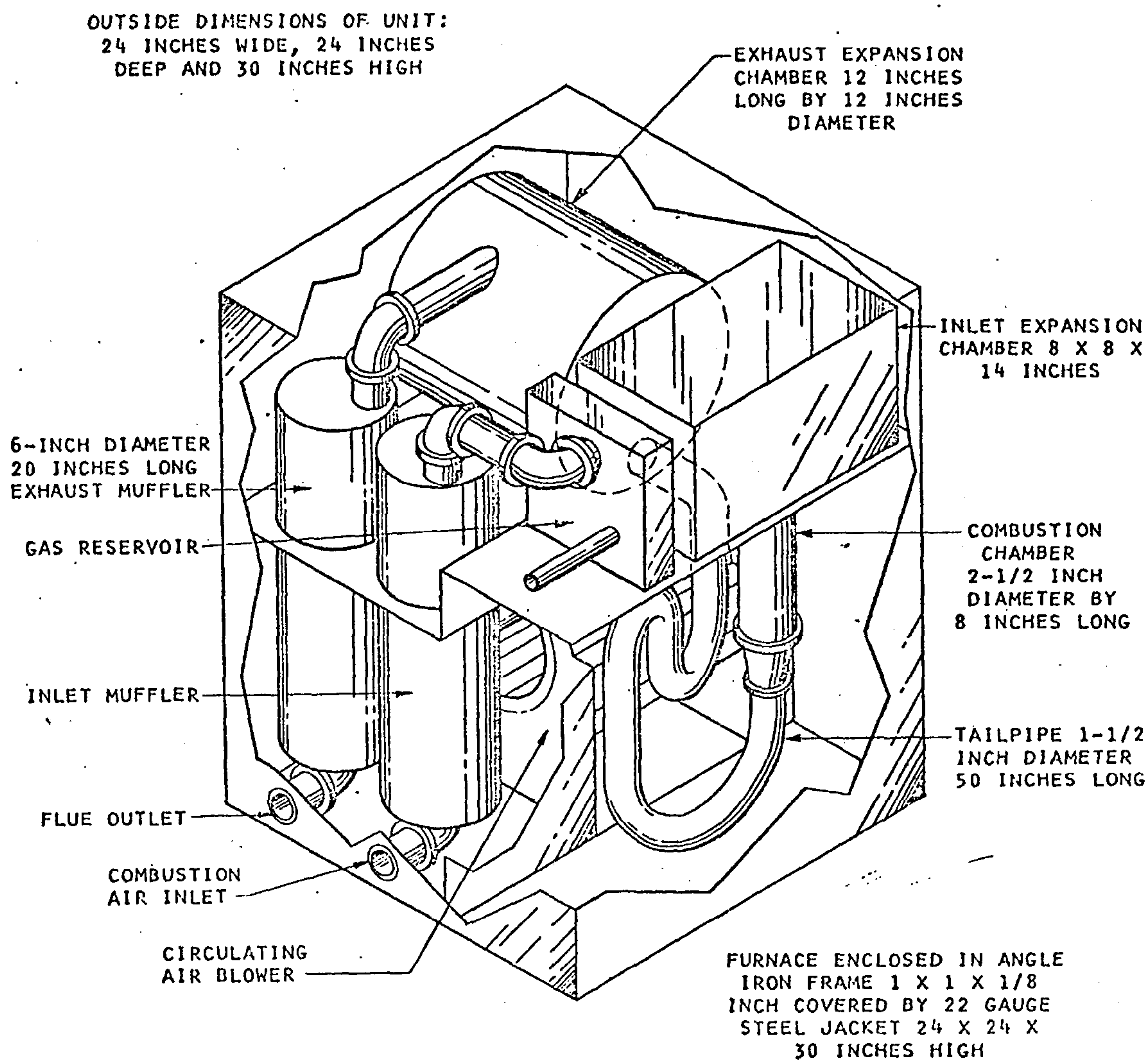


Figure 17. An Experimental, Lo-Boy
Pulse Combustion Furnace

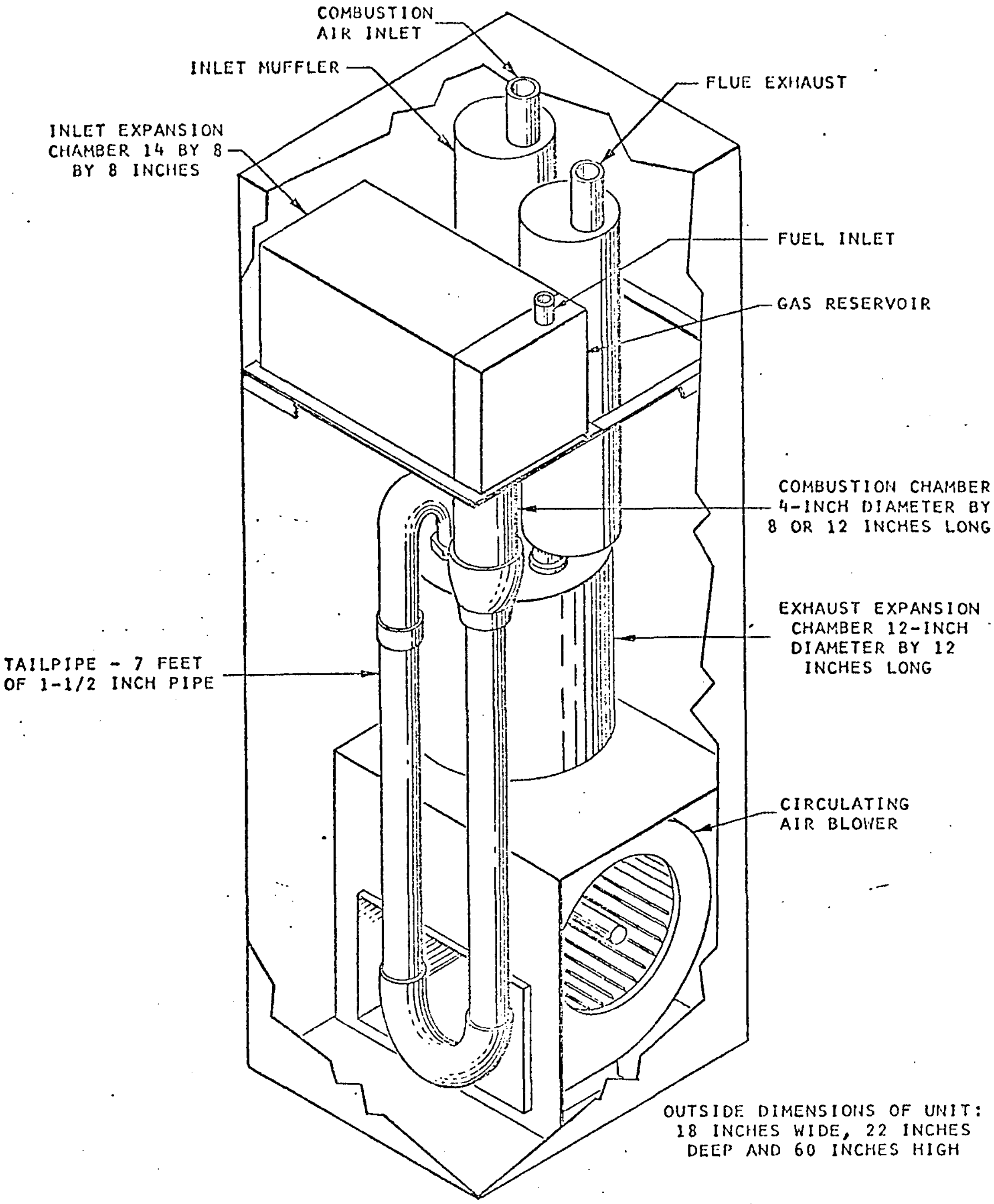


Figure 18. An Experimental, Hi-Boy Pulse Combustion Furnace

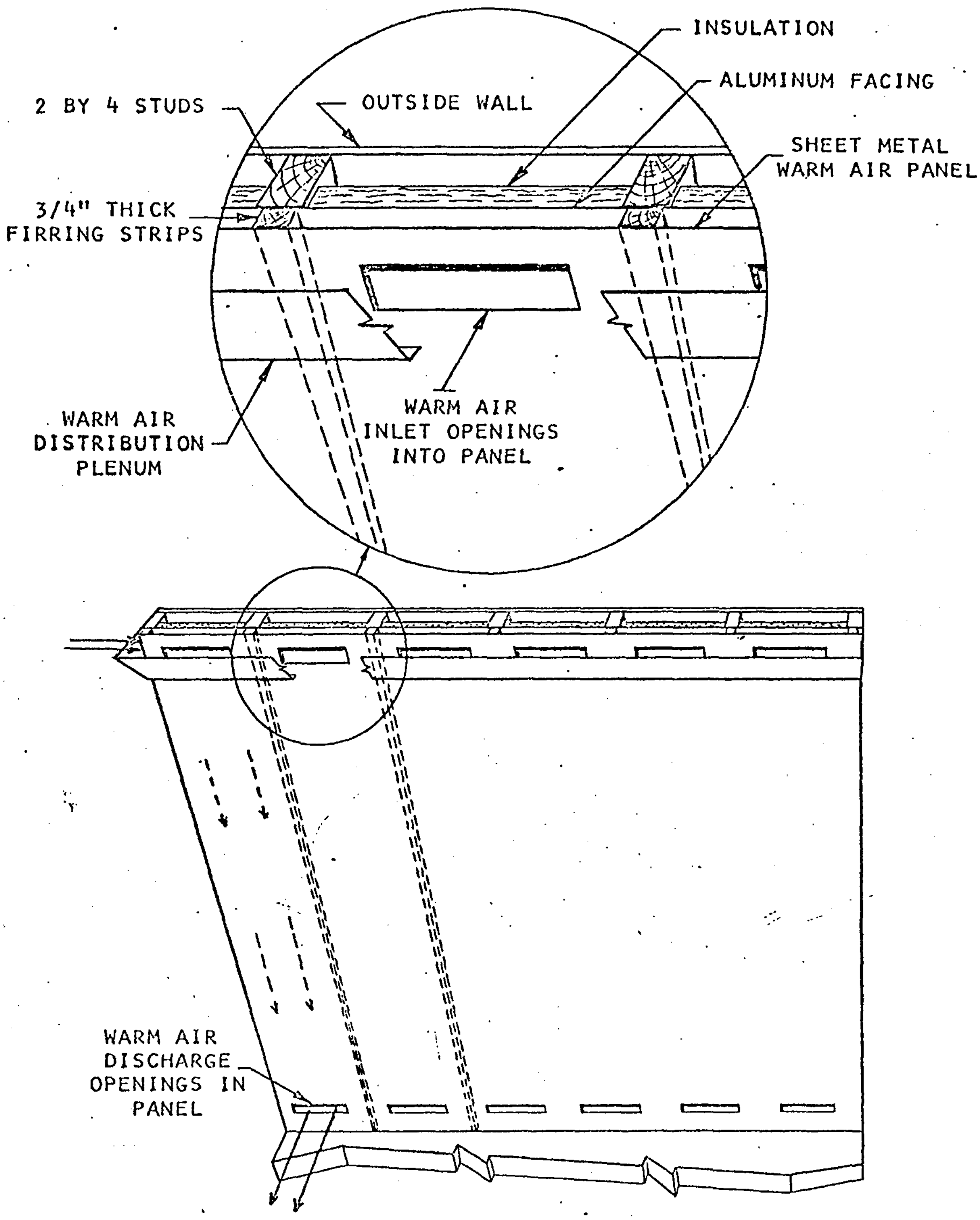


Figure 19. An Experimental, Warm Air Heating Panel System

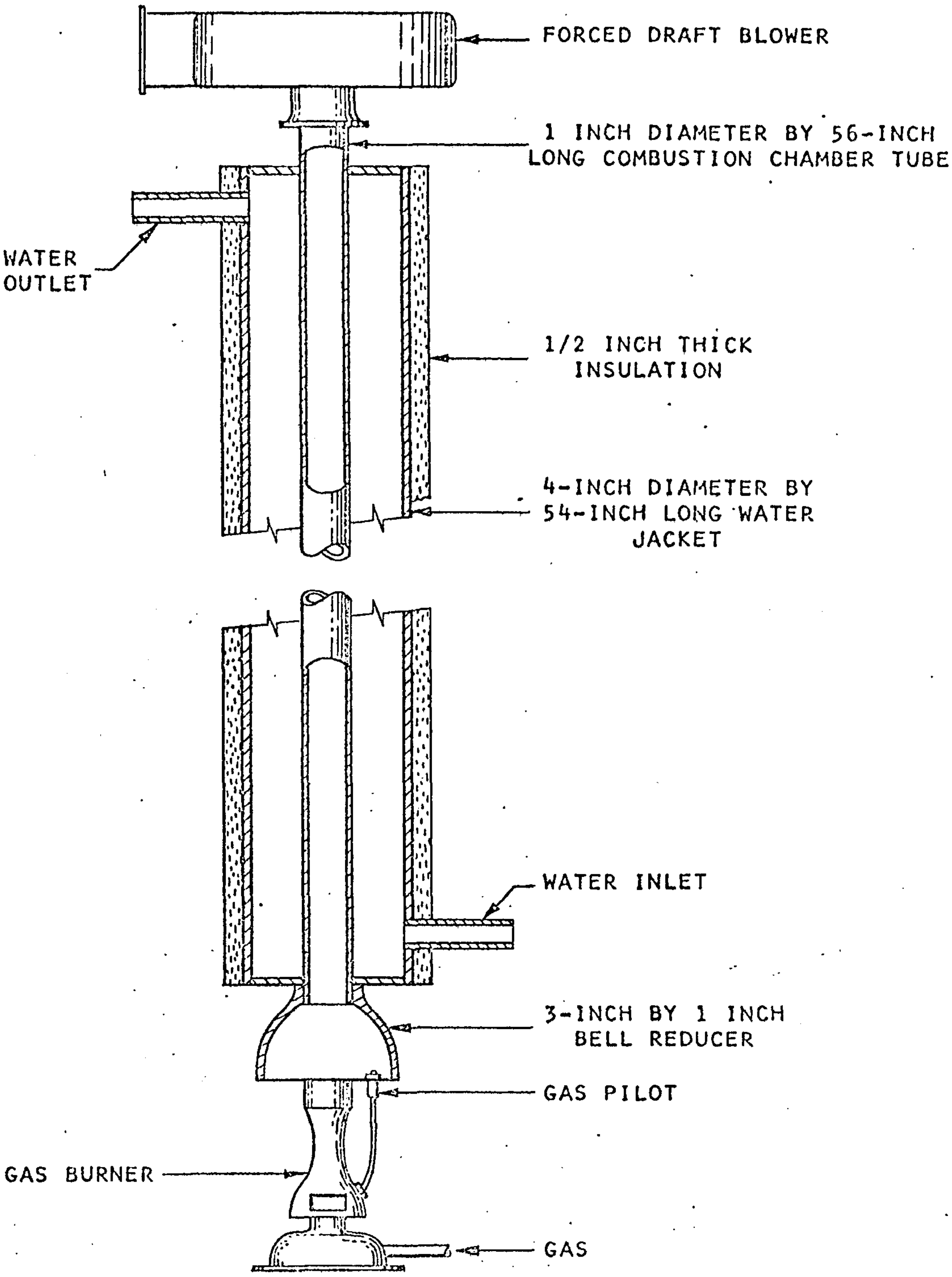


Figure 20. An Experimental Heat Exchanger for a Warm Water Heating Panel Application

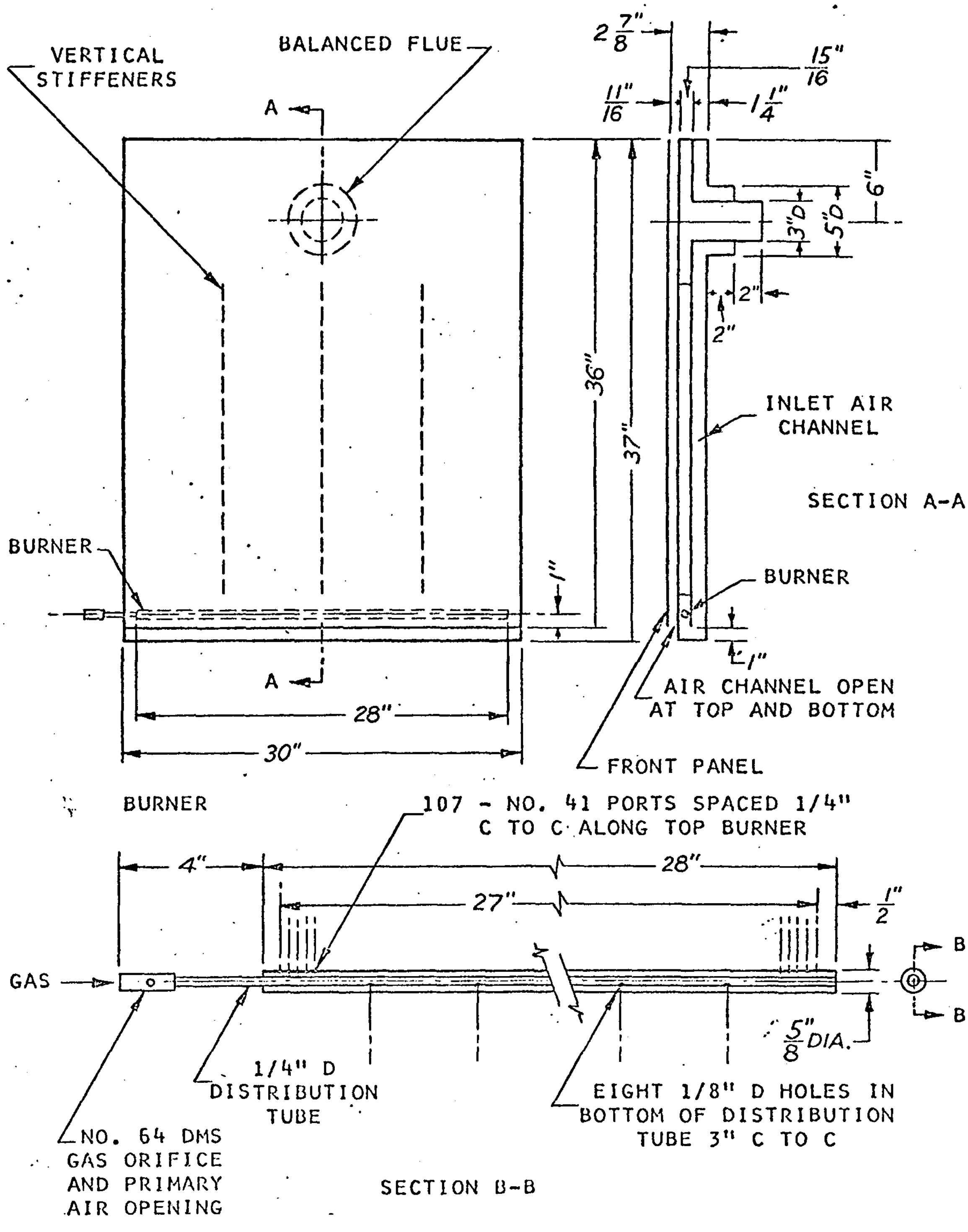


Figure 21. Experimental Direct-Fired Wall Panel

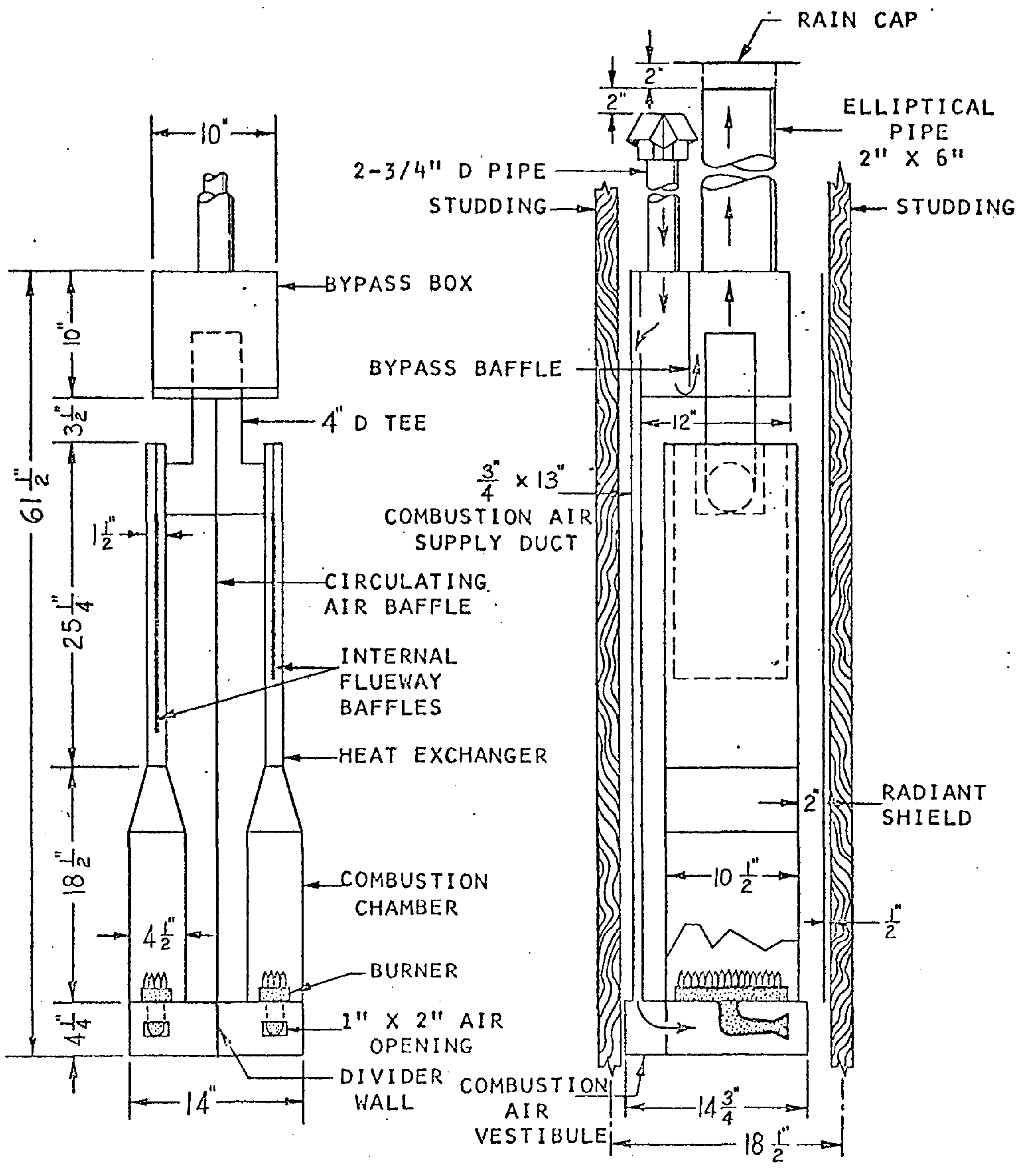


Figure 22. Experimental Snorkel-Vented Dual Wall Furnace with Bypass Arrangement

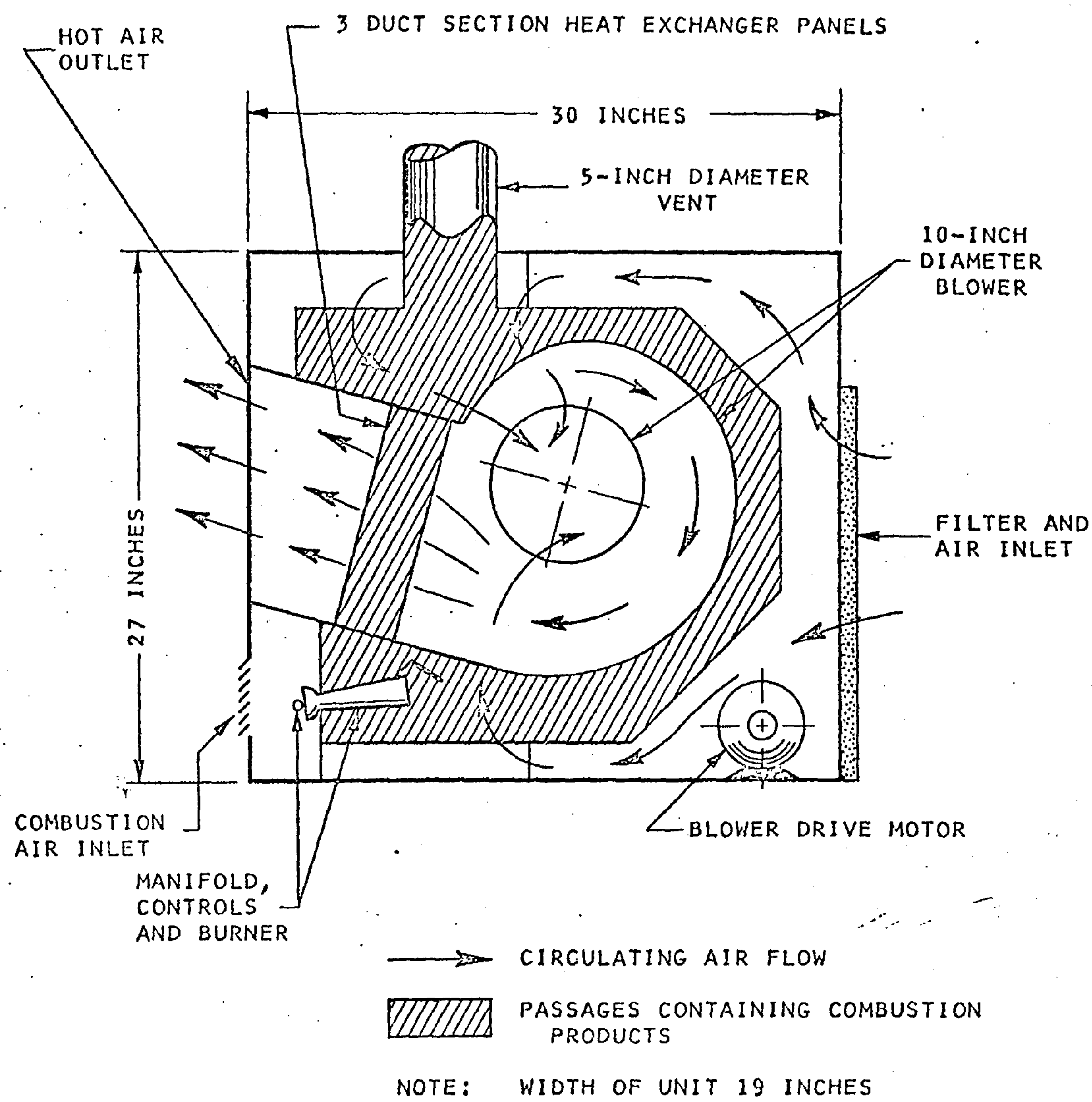


Figure 23. Forced-Air Furnace with Integral Blower-Heat Exchanger

> The burner is immersed in a small nonpressurized water tank, transferring most of the heat input through its walls to the water. The burner's combustion products exit under water in the upper part of the tank, using pressures developed by the pulse combustion process to vent these gases. The heated water is then circulated through a finned water-to-air heat exchanger to heat circulating room air. Flue gas temperatures of 100°F with CO_2 concentrations of 9 to 10 percent are achieved with this arrangement, providing efficiencies of 90 to 95 percent, based on flue losses. At the same time, bubbling flue gases through the water effectively muffles noise of operation at the exhaust to a very low level. Further work is under way to minimize operating noise at the inlet and consideration is being given to air heat exchanger design, materials of construction and arrangements of components to provide compactness.

The latest developments on this concept are considered to be proprietary, however, and no recent reports have been published.

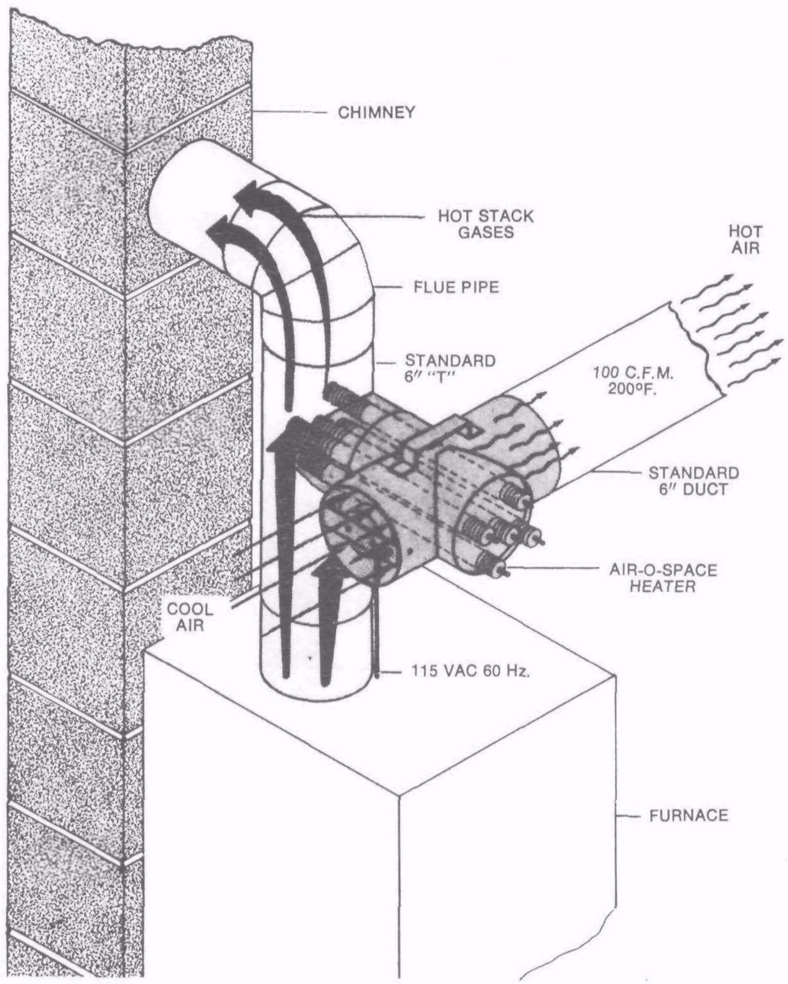
The only other research efforts on heat recovery devices found in the literature were another AGA study by Potter and Wail (Reference 22) and an NBS document by Garrivier on the Use of Air to Air Heat Exchange Systems (Reference 46). These papers deal primarily with either large, higher temperature recuperator systems or with the recovery of ventilated air from commercial building. Many of these heat recovery systems for commercial buildings are currently available on the market.

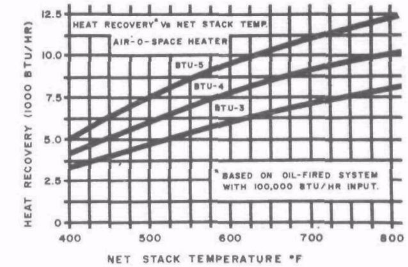
3.5 HEAT RECOVERY DEVICES COMMERCIALLY AVAILABLE

Discussions with furnace manufacture personnel at the Center for Building Technology at NBS, review of the ASHRAE Journal and other HVAC journals, as well as such popular publications as Popular Mechanics and Popular Science, and follow up of other leads have revealed only two devices presently on the market for residential application: a heat pipe device marketed by Isothermics of Augusta, New Jersey and a flue gas to air heat exchanger manufactured by Dolin of Brooklyn, New York. Descriptive literature on each of these devices will be found in the appendix. The paragraphs below provide brief descriptions.

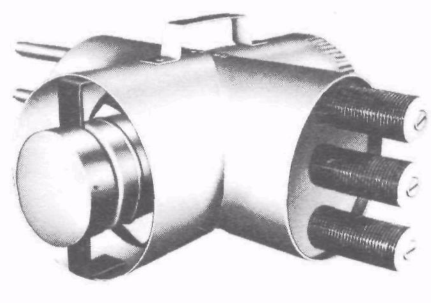
3.5.1 Isothermics Air-O-Space Heater

The Isothermics Air-O Space Heater utilizes 3-5 finned heat pipes projecting into the flue (see Figure 24). A separate 100 CFM blower delivers air across the opposite end of the heat pipe to recover the energy. Figure 24 also shows the potential heat recovery as a function of stack temperature.





Space heater *uses series of heat pipes to extract heat from hot flue gases exhausted from oil-fired home furnace. Recovered heat can be used to heat the basement or directed to the supply duct. Fan blows cool air over condenser end of heat pipes which are isolated from ends in flue by a metal barrier. Fan plugs into 115-v a-c outlet and is controlled by a thermostat in supply duct. Heat output can be adjusted by moving pipes in or out of flue. Air-O-Space Heaters can recover from 5000 to 15,000 btuh, depending on number of pipes in unit (3, 4 or 5), and stack temperature. Unit weighs 15 lb and installs in standard 6-in. tee in about 15 min.*



AIR-O-SPACE HEATER

Figure 24. Isothermics - Air-O-Space Heater

The normal operating range is for flues in the 500°F to 800°F range. In fact Isothermics does not recommend using the unit if the flue is below 500°F for two reasons:

1. The potential heat recovered is below 5000-7500 Btu/hr, or less than 5 to 7.5 percent improvement in efficiency for 100,000 Btu/hr furnace.
2. The exit gas temperatures downstream of the unit will become low enough to cause condensation.

They also recommend that the unit be cleaned periodically and removed during the nonheating season. The device sells for around $100 plus installation.

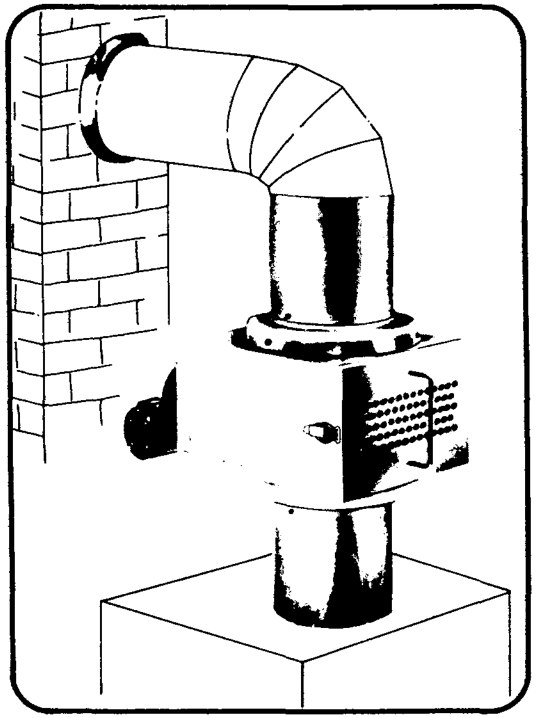
Isothermics indicated they have other domestic units in the prototype stage, but all information is proprietary.

3.5.2 Dolin Metal Products - Heat Reclaimer

The Dolin Heat Reclaimer is a simple flue gas to air heat exchanger made up of 52 3/8" base steel tubes on a 5/8" triangular pitch (see Figure 25). Cooling air provided by a separate blower flows through the tubes while flue gases are directed over the tubes. Unfortunately, wide gaps on either side of the heat exchanger surface on the flue gas side allow nearly half the flow to bypass the unit. This is regarded as a "safety" feature in case the tube bundle becomes plugged. Dolin claims 19,800 Btu/hr heat recovery at a flue temperature of 800°F, although they have no test data to substantiate this claim. A simple analysis reveals a more likely figure of 5300 Btu/hr at 800°F. Figure 26 shows the more detailed results of this analysis which may be found in the appendix. As in the case of the Isothermic Unit, Dolin stipulates that the unit should not be used with flue temperature below 500°F and that the unit should be inspected and cleaned periodically. This unit sells for $130 plus installation and optional duct and register kit.

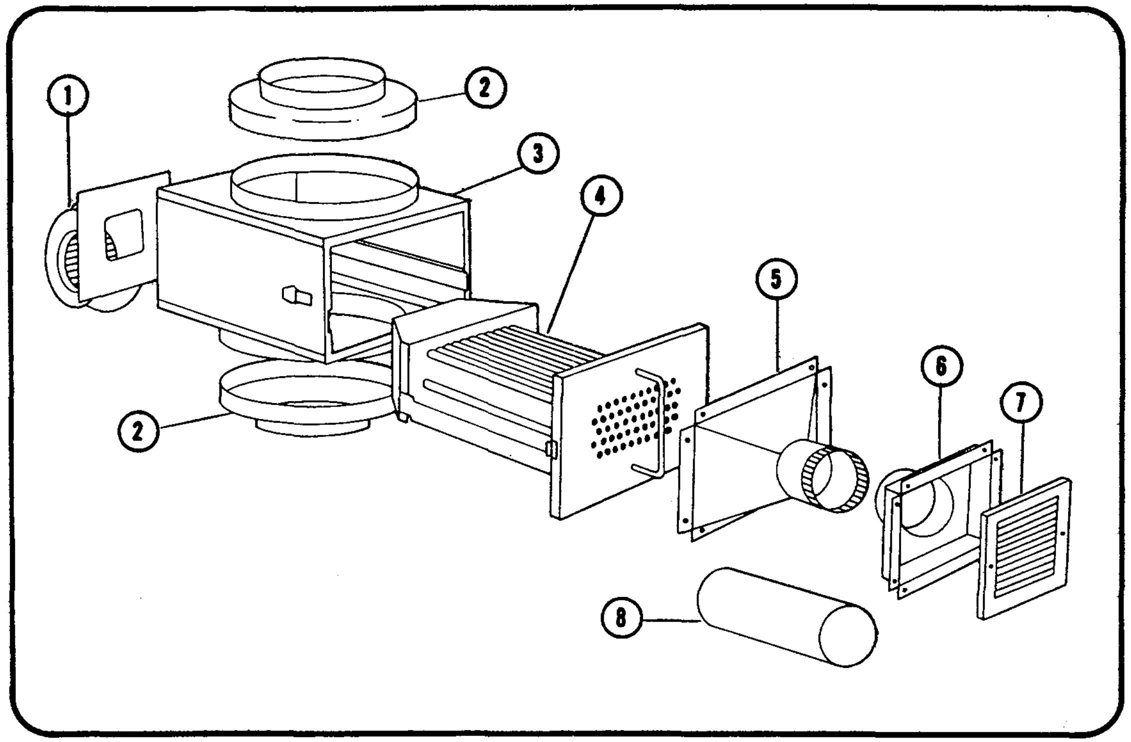
3.5.3 Other Manufacturers and Devices

Several other manufacturers were contacted who indicated they manufactured heat recovery devices. Table 13 gives the companies contact and the type of equipment they manufacture. As can be seen from the table, none offers equipment in the residential furnace size range.



Heating engineers have designed a simple, relatively inexpensive unit that reclaims the wasted heat from oil and coal furnaces that is normally lost up the chimney. It's called . . .

The Dolin
HEAT-RECLAIMER



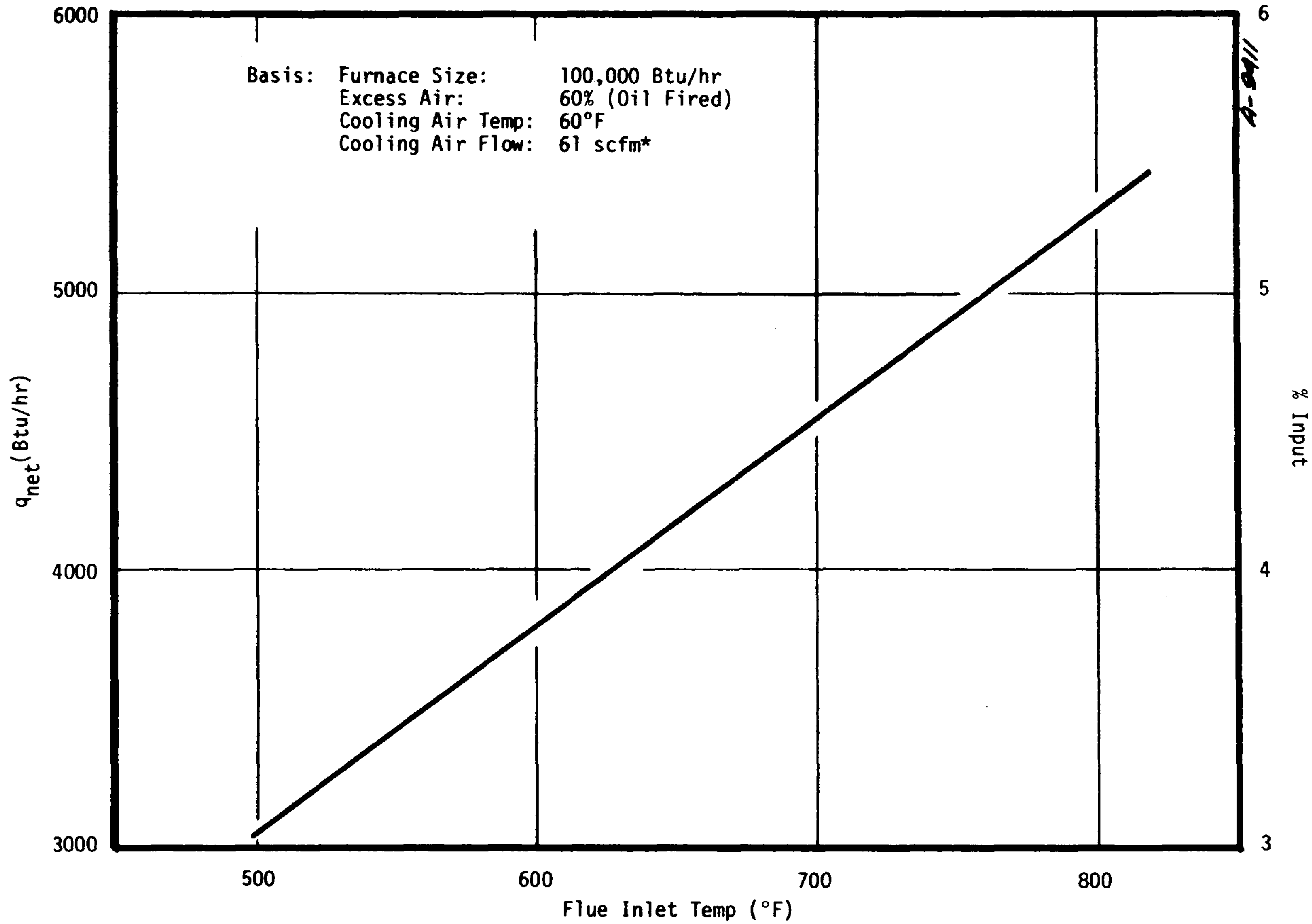
DOLIN HEAT-RECLAIMER
Suggested Retail Price: **$130.00**
(Installation not incl. See your dealer.)

1. Motor and Blower
2. Flue Pipe Adapter (Specify flue pipe diameter: 5″ to 10″.)
3. Reclaimer Housing
4. Reclaimer Core

OPTIONAL DUCT AND REGISTER KIT
Suggested Retail Price: **$15.00**

5. Transition Piece
6. Register Box
7. Register Grill
8. 4″ Duct (supplied and installed by local dealer.)

Figure 25. Dolin-Heat Reclaimer



*See attached curve of resulting operating point of Blower

Figure 26. Dolin Heat Reclaimer Performance

TABLE 13

HEAT RECOVERY DEVICE MANUFACTURERS

Company	Product
Wing Co., Division of Aero Flow Dynamics, Inc. Linden, New Jersey	Rotary air to air heat exchangers - applied to ventilated air
Hughes Tool Co. Torrance, California	Manufactures line of heat pipe to be applied by others.
Heat Recovery Corp. Kearny, New Jersey	Manufactures large heat exchangers for industrial and commercial applications.
Heat Pipe Corp. of America Whippany, New Jersey	Building prototype unit; no product yet. Referred to Q-Dot.
Industrial Air, Inc. Ameilia, Ohio	Manufacturers large heat exchangers for industrial and commercial applications.
Q-Dot Corp. Dallas, Texas	Heat pipe devices for commercial buildings; heat recovery of ventilated air.

In England, a common method of heat recovery uses concentric flue and combustion air supply piping. In this country, the combustion air supply is normally taken from the house air and/or from vents below the house. No further information has been obtained to date on the concentric flue design approach. Section 4.1.5 discusses this general scheme.

Two manufacturers of automatic flue dampers were contacted for information.

Save-Fuel Corporation

The Vent-O-Matic Device manufactured by the Save Fuel Corporation claims to save between 20 to 30 percent of the space heating requirements. The vent is automatically opened electrically prior to the burner ignition and interlocked with the heating applicance gas valve to prevent ignition if the vent is not open. The unit was initially approved by the AGA and the Canadian Gas Association as well as by numerous city public works departments.

However, AGA recently withdrew their approval for application to old furnace systems and the matter is under study by the Federal Power Commission, Federal Trade Commission and Senate Antitrust Committee.

It has not been applied and tested on oil fired units however. The installed cost of this unit is around $100. A bulletin on the device will be found in the appendix.

Werner Diermayer

Werner Diermayer offers a thermally actuated vent damper for about $7-$15 in Europe. This unit utilizes bimetal quadrants as the closure plates. AGA approval has not been granted, although the unit has been used for over 40 years in Europe. A bulletin describing the device will be found in the appendix.

Not found!

SECTION 4

IDENTIFICATION OF ALTERNATE HEAT RECOVERY DESIGNS

Section 3.5 above identified the current state of the art in furnace and heat recovery design. Earlier sections indicate that a potential of 20 to 35 percent of the input energy is available for recovery. Depending on the flue gas temperature, 30 to 50 percent of this energy is from the latent heat of vaporization of the water in the products of combustion. Thus to realize a significant heat recovery, particularly with the higher efficiency, lower exhaust temperature furnaces that are found in a large number of homes, a condensing system will be required.

Because of the different engineering problems involved, it is useful to consider designs in two separate categories:

- Retrofit schemes for existing furnaces
- New furnace design

Each of these general approaches will be discussed in the following sections.

4.1 RETROFIT SCHEMES

In a retrofit approach the furnace may be regarded either as a black box with a hot flue gas stream exiting from the unit or a system subject to internal modifications. In the first approach we concentrate on add-on devices, outside of the original envelope of the furnace, for exchanging heat between the exit flue gas stream and any of several other streams of interest, such as the intake air. Section 4.1.1 presents various alternatives in this general class. The second general approach, discussed briefly in Section 4.1.3, contemplates modificaitons much as heat transfer improvements, within the original furnace envelope.

4.1.1 Black Box Approach

4.1.1.1 General Discussion

In this approach a hot flue gas stream is available to exchange heat with some other medium. The other medium may be one or more of the following:

1. Combustion air

2. Fuel

3. Recirculating house air/water exiting from furnace

4. Recirculating house air/water returning to the furnace

5. Independent air/water stream

6. An intermediary fluid or solid which eventually exchanges heat with one of the other mediums (trade jargon denotes systems of this type as "runaround" systems)

Alternatively, one might also consider converting that energy directly into work or electrical energy. The latter two ideas will be further discussed in Section 4.3, NOVEL APPROACHES. The Isothermics device utilizes Methods 5 and 6 by using the fluid in the heat pipe to exchange heat with an independent air supply. The Dolin device uses Method 5 to exchange an independent air supply directly with the flue.

The advantages and disadvantages of each of these approaches are tabulated and ranked subjectively according to probability of success in Table 14. Although the Method 6 "runaround" circuit shows a fairly high potential, the cost is probably greater, except for the heat pipe approach, than for schemes 3, 4, and 5.

Rotating heat capacitance systems represent another offshoot of Method 6. The rotary metal or ceramic wheel used on gas turbine recuperators or ventilation ducts of commercial buildings is an example of this approach. Rotating wheels that allow leakage between streams could not be allowed with a flue-gas-to-house-air system. However, it may be possible to mechanically rotate a solid between streams with adequate sealing. The chief disadvantages with solid rotation are as follows:

- Mechanical drive system reliability and noise
- Sealing, leakage, and conveyor problems
- Limitation on heat uptake rate by transient conduction effects

Since the probability of success of this scheme seems fairly remote and since the heat pipe represents an inexpensive and simpler solution, this approach was not further explored.

TABLE 14

"BLACK BOX" HEAT EXCHANGER SCHEMES

Scheme Number	Exchange Flue Heat With	Advantages	Disadvantages	Rank*
1	Combustion Air	• Equal mass flow rates • Room air (low temperature) available for heat exchange • Low temperature duct work	• Preheated combustion air may produce excessive furnace metal temperatures • Require extensive duct work • May not be compatible with a naturally aspirated gas burner or oil burner	5
2	Fuel	• Compact heat exchanger • Easy retrofit	• Heat capacity of oil or gas is fraction of available within temperature restrictions	6
3 A	Hot Air from furnace	• Flue stream and air stream are parallel and adjacent in up flow design • Possibly can get by without auxillary air fan	• Potential for heat recovery limited to noncondensing case due to ~ 140 air temperature • Will decrease main air flow rate	4
W	Hot Water from furnace	• Simple piping • Does not require auxillary pump	• Potential for heat recovery limited to noncondensing case due to 180 water temperature • May decrease total water flow rate	
4 A	Cold Air to furnace	• Potential for condensing heat exchange • May not require auxillary fan	• Required extensive duct work reventing • May decrease main air flow rate	3
W	Cold Water to furnace	• Simple piping • Potential for condensing heat exchange • Does not require auxillary pump	• May decrease main water flow • May not be compatible with a hydronic system	
5	Independent air/water stream	• No change to main air ducting • Can choose air/water flow for optimum heat exchanger design • Does not affect main air/ water flow rates • Probably least expensive scheme • Can run duct to spare room, garage, basement, etc. or back into main air stream if pressures compatible • Potential for condensing heat exchange	• Requires auxillary fan or pump • In some case would require extensive separate duct work or piping	1

TABLE 14 (Concluded)

"BLACK BOX" HEAT EXCHANGER SCHEMES

Scheme Number	Exchange Flue Heat With	Advantages	Disadvantages	Rank*
6	Intermediary Fluid ("Runaround Circuit")	• Least possibility for leakage of gas/air streams • Simple manifolding - can run streams in parallel • Ideal for heat pipe • Potential for heat sink remote from heat source	• Except for heat pipe, requires auxillary "runaround circuit" and equipment (pump, valves piping, etc.)	2

*1 = best
6 = worst

A more elaborate version of Scheme 5 (or Scheme 6) would have a heat pipe unit or heat exchanger transfer heat to a central energy recovery water loop to store energy in a hot water system. All energy using appliances should be tied into this scheme to achieve best results. This scheme, while it may offer some advantages, seems unlikely to contribute an economical retrofit scheme.

Since the usual versions of Schemes 5 and 6 are already represented on the marketplace, no further detailed analyses were done on these. However, the potential for improving on these devices still exists. Table 15 lists some possibilities.

In general the disadvantages of Method 1 outweigh the advantages resulting in a fairly low ranking. However, as was mentioned earlier the British commonly use this system in the form of concentric inlet and outlet ducts for the flue gas and combustion air. Furnaces are designed to accommodate the preheated combustion air and higher flame temperatures.

Scheme 2 can be dismissed because the heat capacity of the fuel within the temperature limitation of the flue gas is a small fraction of the potential heat recoverable.

4.1.1.2 Design Problem Areas

Problems common to all retrofit designs will be discussed in more detail in the following paragraphs:

Pressure Drop

It can be shown by conventional heat exchanger design techniques that to increase the system efficiency by only 7 percent using existing furnace heat surface types (heat transfer coefficient approximately 2.7 Btu/hr-ft^2°F), thereby incurring minimum pressure drop, would require nearly double the surface area. A preferable solution uses more compact heat exchange surface (high surface to volume ratios), higher heat transfer coefficients and consequently higher pressure drops. Tables 8-10 in Section 3 indicate that the available pressure exhausting the products of combustion is quite limited, on the order of 0.05-0.2 inches H_2O. Thus it seems quite likely that to decrease the flue temperature below 500°F an induced draft fan will be required. The feasibility of this with a naturally aspirated gas fired burner has been shown by DeWerth, et al. (Reference 19) in a novel wall furnace (see Figure 13 in Section 3).

TABLE 15

IMPROVEMENTS TO EXISTING DEVICES

Scheme Number	Present Manufacturer	Improvement	Problems
5	Dolin	• Close off bypass area • Use fin tubes • Increase heat transfer area to achieve condensing heat exchanger • Use plate-fin heat exchanger	• Satisfying local city safety codes • Provision for condensate • Change ducting materials to avoid corrosion problems • May require induced draft fan once the bouyancy is reduced and pressure drop increased
6	Isothermics	• Increase heat transfer area to achieve condensing heat exchanger	• Provision for condensate • Change ducting materials to avoid corrosion problems • May require induced draft fan once the bouyancy is reduced and pressure drop increased

Condensate

If temperatures anywhere in the system (including surface temperatures) get below the dewpoint (approximately 123°F, depending on fuel, excess air and relative humidity (see Figure 27), condensation will begin. If the flue is cooled to 100°F approximately 2 to 2-1/2 qts/hr will be condensed for natural gas combustion with 60 percent excess air. Provision must then be made for collecting and draining off this water. Careful installation of the device is required as well to prevent condensate from flowing back into the furnace.

Corrosion

As the dewpoint is reached on any metal surface and condensate begins to flow, the potential of corrosion exists due to the sulfur in the fuel. The problem will no doubt be more severe for oil and coal fired furnaces than for gas fired. Most furnaces are manufactured from carbon steel and the flues from galvanized steel or aluminized steel. The choice of materials for the heat exchange surface and ducting will depend on the temperature level and whether there is a condensing condition. It may be possible to use different materials in different temperature regimes. Table 16 lists some potential materials of construction, maximum temperature, relative cost, resistance, thermal conductivity (important for finned surfaces), and potential application. It should also be kept in mind that once the bulk exhaust temperature gets below 300°F, the probability of condensation on cold metal surfaces is fairly high; therefore, any heat recovery device applied to flue temperatures below 500°F will probably require a change of ducting material.

Thermal Stress and Cycling

Thermal stresses and thermal cycling are common problems with all furnace components which must be taken into account in the design. Temperature gradients, particularly in cross flow heat exchangers, can easily produce destructive thermal stress in an improper design. In addition, many commercial systems suffer from failures due to fatigue cracking causes by cyclic stresses. Overcoming these problems usually requires extensive experience and/or rather sophisticated analysis. AGA has a program devoted to studying the problem of cycling fatigue as this remains the principle failure mode of the main furnace heat exchanger.

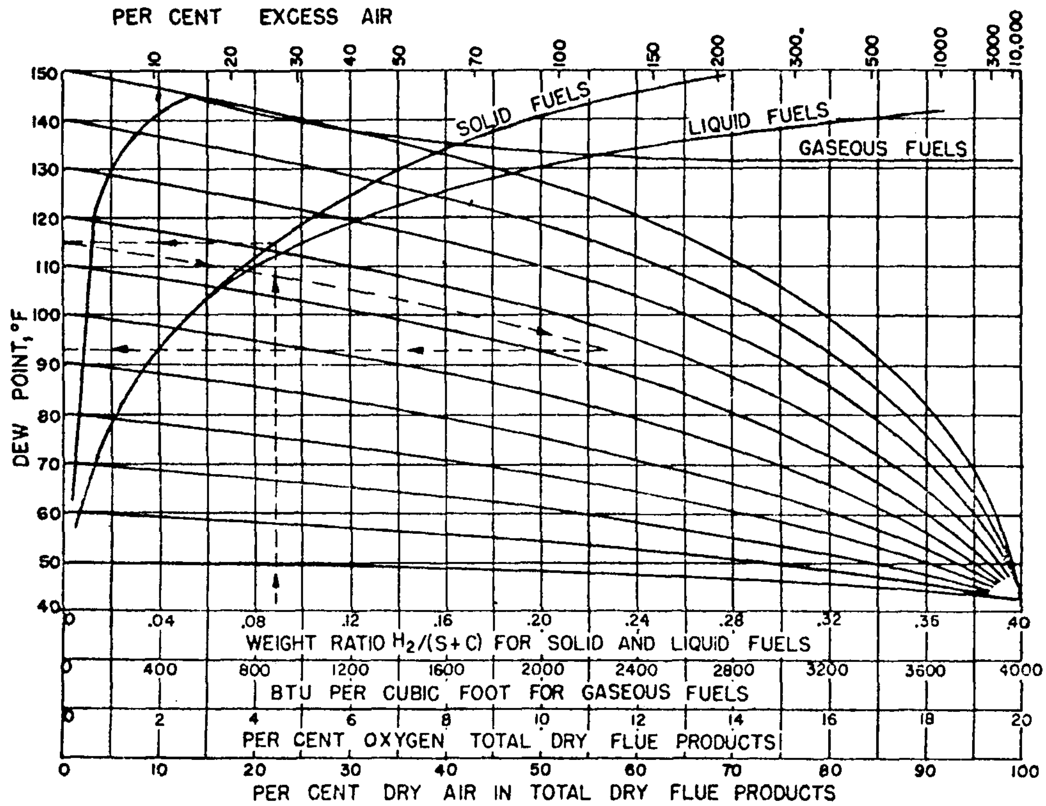


Figure 27. Theoretical Dew Points of the Combustion Products of Industrial Fuels

TABLE 16

POTENTIAL MATERIALS OF CONSTRUCTION

Material	Maximum Temperature Range (°F)	Corrosion Resistance	Thermal Conductivity Btu/hr-ft°F	Cost $/#	Potential Application
Carbon Steel	800-900	Poor	26	.33-.42	Furnace H.E. surface
Aluminized Steel	1180	Fair	~ 26	.9-1.0	Duct - noncondensing H.E. - noncondensing
Copper	500	Fair	220		Low Temperature H.E.
Aluminum (3003)	400	Fair	116	.74	Low temperature but noncondensing H.E. ducts
Stainless Steel (409)	1100	Excellent	8.0	1.05	All parts
Stainless Steel (301)	1470	Excellent	9.4	1.15-1.20	All parts
Galvanized Steel	550	Fair to Poor	26	.24	Ducts - noncondensing
Porcelain Coated Steel	1100	Excellent	26		Ducts - condensing possibly H.E.
Ceramic Plasma Sprayed Steel	1100	Excellent	26		Ducts - condensing possibly H.E., poor fabricability
Plastics	250	Excellent	.05-.2		Ducts - low temperature H.E. - low temperature

Potential Heat Recovery

Figure 28 shows the potential heat recovery as a function of flue temperature upstream and downstream of the device, for a 100,000 Btu/hr furnace operating on natural gas at 60 percent excess air. From this curve it can be seen that with a low upstream flue gas temperature one must go into the condensing regime to achieve significant percentage heat recovery. If the furance is fairly inefficient and the upstream flue temperatures are high (800°F), significant recovery can be made in the noncondensing regime.

4.1.1.3 Conclusions

All the retrofit external modification schemes share some common problems summarized briefly in Section 4.1.1.2. In addition, each of the methods has particular difficulties, as discussed in Section 4.1.1.1 and summarized in Table 14. Scheme 2 is definitely not feasible. Scheme 1, although sound in concept, does not appear practical as a retrofit scheme for the geometries of common U.S. furnaces. Schemes 3, 4, and 5 appear to survive this first examination as candidates for retrofit schemes; more detailed analyses of these concepts will be described in Section 5.

4.1.2 Internal Modifications

It may be possible to design a device such as additional surface which would be inserted as a retrofit unit into a furnace heat exchanger to improve the heat transfer performance. This would have to be done within the pressure limitations of the burners, unless an induced draft fan could be added. Only sensible heat could be recovered since condensation within an existing furnace could not be tolerated. Care would also have to be taken so that furnace wall temperatures would not become excessive. Access to the heat exchanger surface would be a problem in many cases and installation costs would usually be high. For these reasons this avenue was not further pursued at this time.

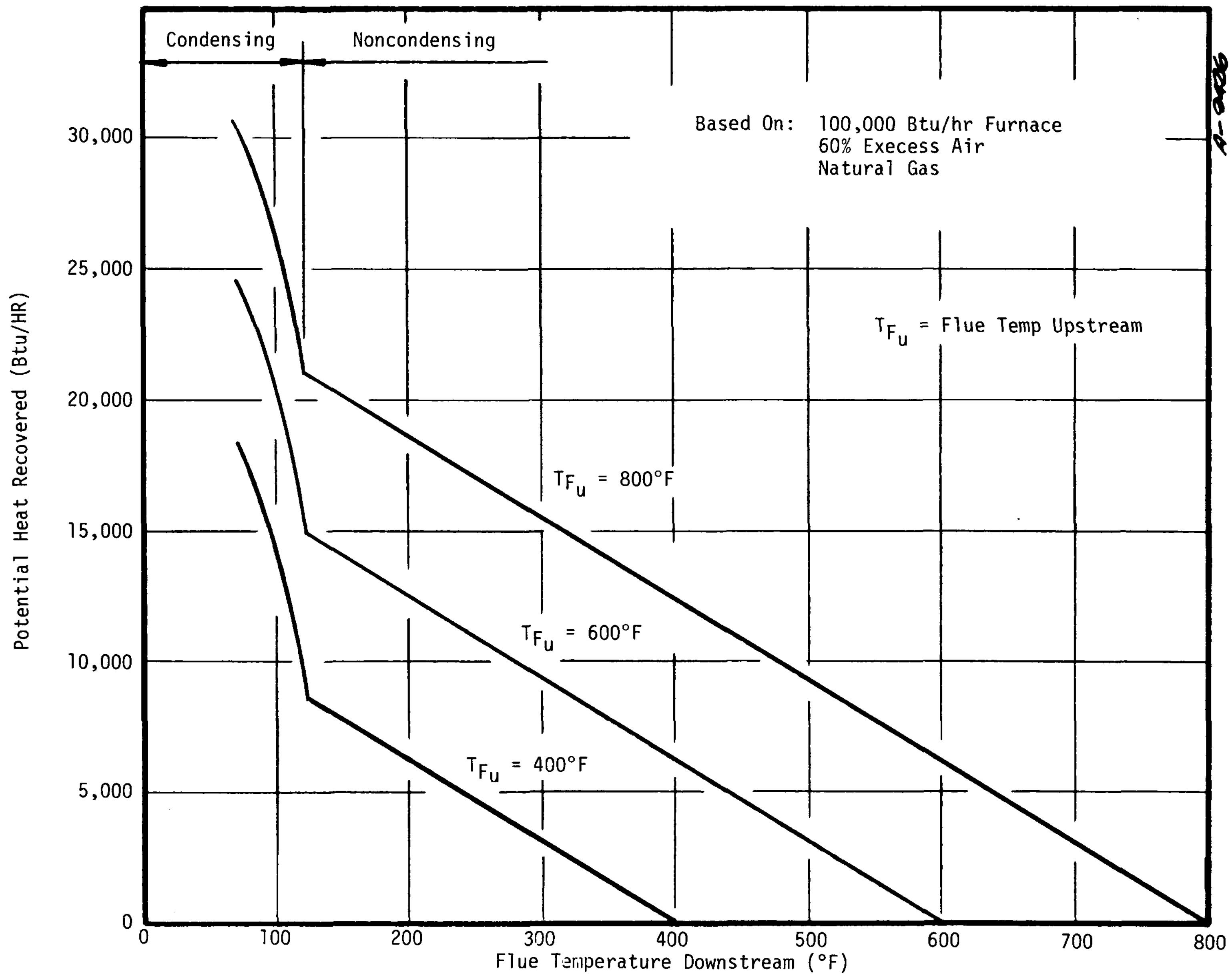


Figure 28 . Potential Heat Recovery

4.2 NEW FURNACE DESIGNS

Two general clarifications of new furnace design approaches can be made.

- Increase Heat Transfer Surface
- Novel Approaches

These are treated separately in the subsections below.

4.2.1 Increased Heat Transfer Surface

As pointed out in the previous section, to increase the efficiency of the furnace using the existing heat transfer surface configuration would require excessive volumne. In addition, to increase the efficiency beyond what is currently being achieved, 80 percent, will require provision for condensation in the stack or for a full condensing furnace. The latter approach seems to be the most advantageous. To achieve within a reasonable volume the heat transfer surface required for a condensing heat exchanger will probably require a so-called "compact" heat exchanger with associated higher pressure drops. Therefore, either a "power burner" (forced draft) or an induced draft fan will be required.

We see no serious impediments to the design of such a furnace. Careful consideration will have to be given to materials, as mentioned in the previous section, as well as to cost. Since several large furnace manufacturers indicated they had such designs in the prototype stage, no further analysis was performed on the concept at this time.

4.2.2 Novel Approaches

Several novel designs have been identified as having potential. They include the AGA pulsed combustion concept, the ceramic honeycomb and heat exchanger, a catalytic central house heater, a submerged combustion heater, and the AMANA heat transfer module approach. The first and last of these were discussed in Section 3.4. The first, the pulsed combustion development, is just now being costed by AGA and leading manufacturers. No data are yet available on this unit. The last is a commercially available system.

A number of other novel ideas as well as the catalytic and submerged combustion concepts will be discussed in the following subsections.

4.2.2.1 Catalytic Combustion

The potential for low emissions and high heat transfer in catalytic unit rates (References 47, 48) suggest the possibility of applying the concept to residential heating systems. Several investigations have in fact pursued this possibility in various forms. The so-called "surface infra-red panels", in which fuel and air burn on the surface of a porous refractory panel, are used to supply local heat in several space heating applications such as semi-outdoor restaurant terraces. The similar catalytic "tent heater" provides another example of local space heating. The application of such devices to space heating when burning a hydrocarbon fuel is limited by CO and hydrocarbon emissions; safety requires adequate venting. Proposed schemes involving hydrogen as a fuel do not have stringent venting requirements and have many attractive design possibilities.

In a central heating concept, surface or catalytic combustion is of chief interest because it may provide a very low NO_x combustion unit. Reference 48 surveyed the current research and development scene and explored several potential applications, all of which promised low NO_x performance. Several experimental systems have demonstrated excellent combustion and emission characteristics.

Because surface combustion allows high combustion rates per unit area, proper design incorporating this concept can reduce equipment size over that of conventional equipment, and often can achieve dramatically lowered costs. In addition, if the system is designed to have low mass, the heat output of a surface combustion unit can be modulated with great exactness and speed. Section 3.3.5 discusses the possible benefits of a modulation or "turn down" capability.

The small size and high heat transfer rate capability of a surface combustion unit may make it unsuitable for use with warm air heating because it typically is difficult to provide a high enough heat transfer coefficient on the air side to accept the high heat fluxes without forcing heat exchange wall temperatures too high. The approach should be much better suited to a water scheme. Table 17 summarizes some of the other potential disadvantages of the surface combustion approach, and some of the outstanding question areas.

Despite the possible disadvantages and the question areas, surface combustion deserves careful study for house heating application. The subject was not pursued, however, under this program because EPA is currently sponsoring a separate feasibility review of surface combustion applied to home heating, industrial furnaces, and similar applications, following up the critical survey

TABLE 17

ASPECTS OF CATALYTIC OR SURFACE COMBUSTION FOR CENTRAL UNITS

Probable Advantages	Possible Disadvantages	Question Areas
1. Clean combustion: low particulate, low hydrocarbon, low NO_x	1. May not be suitable for direct transfer to air	1. Most experiments to date show higher hydrocarbon emissions than desired
2. Compact heat transfer area; implied low cost	2. Unless carefully de-designed for low mass, will not be suitable for short "on" periods and cannot be modulated	2. Design problems for liquid fuels are difficult and need further study
3. Modulation capability if properly designed	3. Preferred low temperature operation requires a catalyst, with associated problems of cost, lifetime, and maintenance	
	4. Cold starts are difficult	

of Reference 48. Should this feasibility study indicate definite potential in the home heating area, a detailed design and cost analysis task is recommended.

4.2.2.2 Submerged Combustion

High heat transfer rates and the potential for emission prevention suggest that a submerged combustion might be a feasible scheme for a high efficiency low emission furnace. In fact, the AGA pulsed combustion furnace is just such a device. However, it is not clear that the submerged combustion concept can be applied to distillate oil burners and conventional powered gas burners as well as this novel gas burner. Applications of submerged combustion are known in the petrochemical industry for combustion of "waste" fuels where scrubbing of the exhaust products is required. Table 18 lists some of the advantages and disadvantages of this approach. The problems of light off and stable combustion are probably the most severe. Figure 29 shows one possible solution, placing the burner above the water but forcing the products of combustion through the water. The upper part of this immersion tube may have to be refractory or ceramic lined. Continuous water treatment is also required to avoid acid conditions in the tank over a period of time. The burner will be required to develop several inches of water pressure to overcome the immersion depth. As with the catalytic combustion system, there could be a conflict between the heat capacity and the cyclic nature of home furnaces. Unless heat output rate can be modulated to match the load, the time response of this approach may be inadequately slow. Further studies are required to size the tank, determine the optimum size and type of heat transfer surface burner configuration, and to determine the dynamic response of the system. Many of the answers will only be attainable through development of an experimental prototype model.

4.2.2.3 Thermoelectric Device

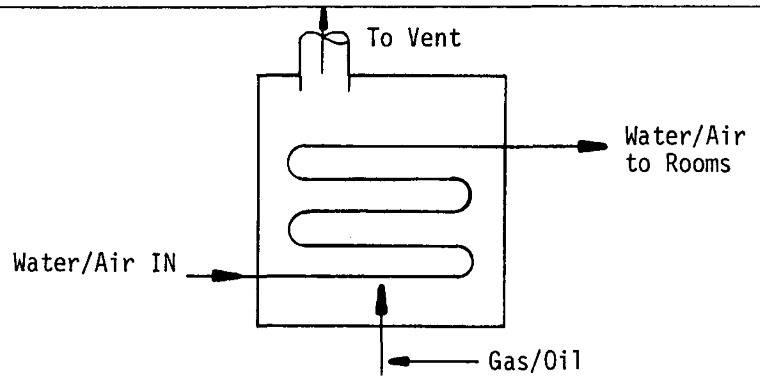
It is possible to convert the flue gas thermal energy directly into electrical energy. Thermopiles have been applied to similar problems for a number of years. The problems associated with this concept are noted below:

- Very low conversion efficiency at low temperatures
- No common home use for the power developed
- High cost; expensive material

For these reasons this concept was not further studied.

TABLE 18

SUBMERGED COMBUSTION



Advantages	Disadvantages
• Clean stack gases • High heat transfer rates • Possibility of direct exchange with circulating water	• Potential start up problems • Corrosion and water treatement problems • High pressure drop, particulary for gas • Unstable combustion • Possibly expensive • Heat capacitance problems

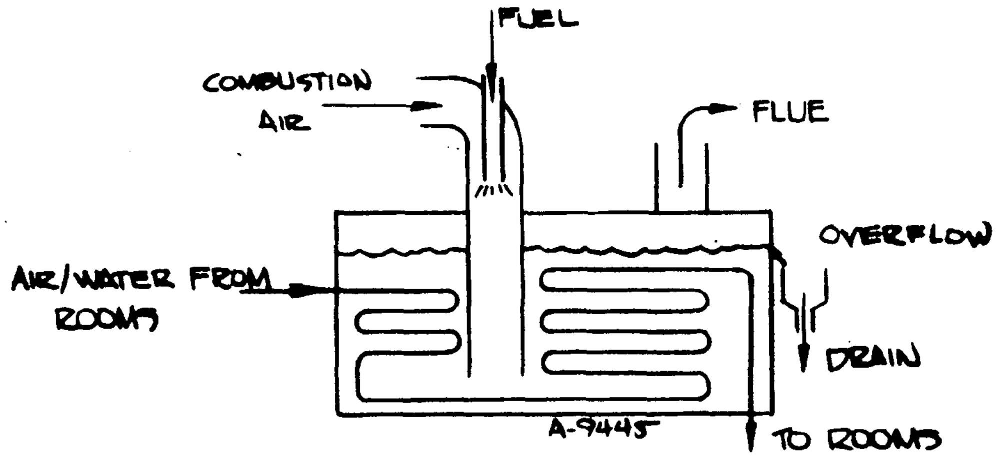


Figure 29. Burner Arrangement - Submerged Combustion

4.2.2.4 Other Concepts

Most other concepts for improvements in ventilation heating systems demand much more elaborate changes and in fact impact the entire energy management of the building. These were outside the scope of this program, but deserve (and are receiving) careful study as evergy conservation becomes more and more important:

- Heat pumping systems, with utilization of main power unit waste heat
- Total energy systems, for single family units, for large buildings, and for building complexes or residential districts; various main power units
- Solar-assisted systems of all types

SECTION 5

DETAILED ANALYSIS OF RETROFIT SCHEMES

One of the prime objectives of the current study was to conduct a detailed design and cost examination of retrofit flue gas energy recovery units. These seemed to offer reasonably attractive promise, particularly in the face of fuel shortages and increased fuel prices. In addition, various and scattered commercial ventures to produce and market very simple versions for such units with limited goals suggested that the qeneral concept is sound and competitive.

Section 4.1.1 above reviewed the alternative approaches to flue gas recovery and identified two general approaches to flue gas energy as most promising:

- Direct exchange with air/water exiting from the furnace (Scheme 3 of Section 4.1.1)
- Direct exchange with air/water entering the furnace (Scheme 4 of Section 4.1.1)

These schemes are to be considered for gas and oil fired burners over a variety of temperature conditions (or furnace inefficiencies) and for air and water systems. In addition we will consider a noncondensing and a condensing heat exchanger in each case.

Details of the four potential schemes to be analyzed are given in Table 19, which includes the potential heat recovery rate, the therms saved per year and potential savings for typical West Coast and East Coast cities.

In this table we follow the numbering system of Section 4.1.1, and append an A or W to indicate air or water respectively as the heated fluid. Case 3A represents an approach of exchanging heat with the exit air from the furnace. This is ideal because a larger number of these are parallel upflow enabling a simple heat exchanger to be placed at the air outlet side in a crossflow arrangement. This approach is represented by Case No. 1 and is limited to a noncondensing heat exchanger because the exit air to be heated is already higher than the flue gas dewpoint. Scheme 3W represents the same approach with water as a working fluid. This scheme is marginal since it has the lowest potential in total energy savings.

TABLE 19

POTENTIAL SCHEMES TO BE ANALYZED

Sch No.	Fuel	Heated Medium	Forced or Natural Draft Burner	Exit Flue Temp.	Heated Medium Temp.	Scheme	Condensing Heat Exchange	Potential* Heat Recovery BTU/Hr	Therms/ Year** Saved	~ $ Savings*** / Year SF	$ Saving/Year**** Mass.
3A	Gas	Air	N	300-500	140	Exchange flue gas heat with exit air from existing furnace	No	4223-	44-	5-20	10/40
	Oil	Air	F	500-800	140			17419	181		
2 4A,C	Gas	Air	N	300-500	70	Exchange flue gas heat with furnace inlet air	Yes	13493-	140-	15-32	30-64
	Oil	Air	F	500-800	70			27797	289		
4A,N							No	6000-19000	62-198	7-22	14-44
3W	Gas	H_2O	N	300-500	180	Exchange flue gas heat with boiler outlet water	No	3167-	33-	4-19	8-38
	Oil	H_2O	F	500-800	180			16364	170		
4W	Gas	H_2O	N	300-500	70	Exchange flue gas heat with return water or fresh feed water	Yes	13493-	140-	15-32	30-64
	Oil	H_2O	F	500-800				27797	289		
4W,N							No	6000-19000	62-198	7-22	14-44

*Based on 100,000 BTU/Hr heat input to the furnace; 60% excess air and 85% efficient heat retrofit heat exchanger.

**Based on nominal 1040 therms/year normal for heating.

***Based on $.11/therm San Francisco Bay Area cost.

****Based on $.22/therm for oil.

Cases 3 and 4 allow a choice between noncondensing and condensing approaches. The noncondensing approach, in which the flue gas is cooled only to 200°F - 250°F to prevent condensation in the flue, is reasonably attractive for the most inefficient furnaces, i.e., those with exhaust temperatures of 500°F - 800°F; however, Table 20 shows only condensing approaches to Case 4 because noncondensing schemes are already being calculated commercially, apparently with indifferent success.

To achieve a relatively compact arrangement, plate fin and fin tube designs as shown in Figures 30a and 30b were considered; both designs were considered to see if one held an economic advantage over the other.

Hand calculations proved to be time consuming in evaluating different passage heights, shapes and overall dimensions. Therefore a simple computer program "PERF" which utilizes the heat exchanger performance calculation technique of Kays and London (Reference 49) was written. Given the input information listed in Column A of Table 20, the program calculates the performance as listed in Column B. A listing of this program may be found in the appendix.

It will be noted that the code does not perform a final design choice but simply runs out a series of designs from which a design may be selected, by interpolation if necessary, to achieve a specified performance.

For a condensing type heat exchanger it was necessary to use the Log Mean Temperature Difference (LMTD) technique in conjunction with an approach detailed by McAdams (Reference 50, pages 355-356) to determine the heat transfer of condensing vapors in a noncondensable gas. Details of this technique can also be found in the appendix.

Note that the determination of heat exchanger size require simultaneous solution of heat transfer and pressure drop requirements for the hot and cold streams. Since the open area and path length, the factors which determine heat transfer and pressure drop, for one side are a function of the opposite side, the sizing procedure is often a tedious iterative procedure, even with the aid of the PERF code. The fact that the air flowrate is approximately 40 times the flue gas flowrate for Schemes 3A and 4A compounds the basic problem. The design procedure for Scheme 5A is somewhat simplified since the cooling air flowrate can be an independent variable.

A typical blower performance curve is shown in Figure 31. Assume that a typical operating point will be 1150 CFM and 0.4 inch H_2O, as shown in Figure 30. Increases in the system pressure drop of 0.1 and 0.2 inch H_2O will cause the flow to drop to 1090 CFM and 1025 CFM, respectively, which should not affect the overall performance of the main furnace. Therefore, if we limit the pressure drop to tenths of inches of water we should not cause a serious deterioration in the furnace performance.

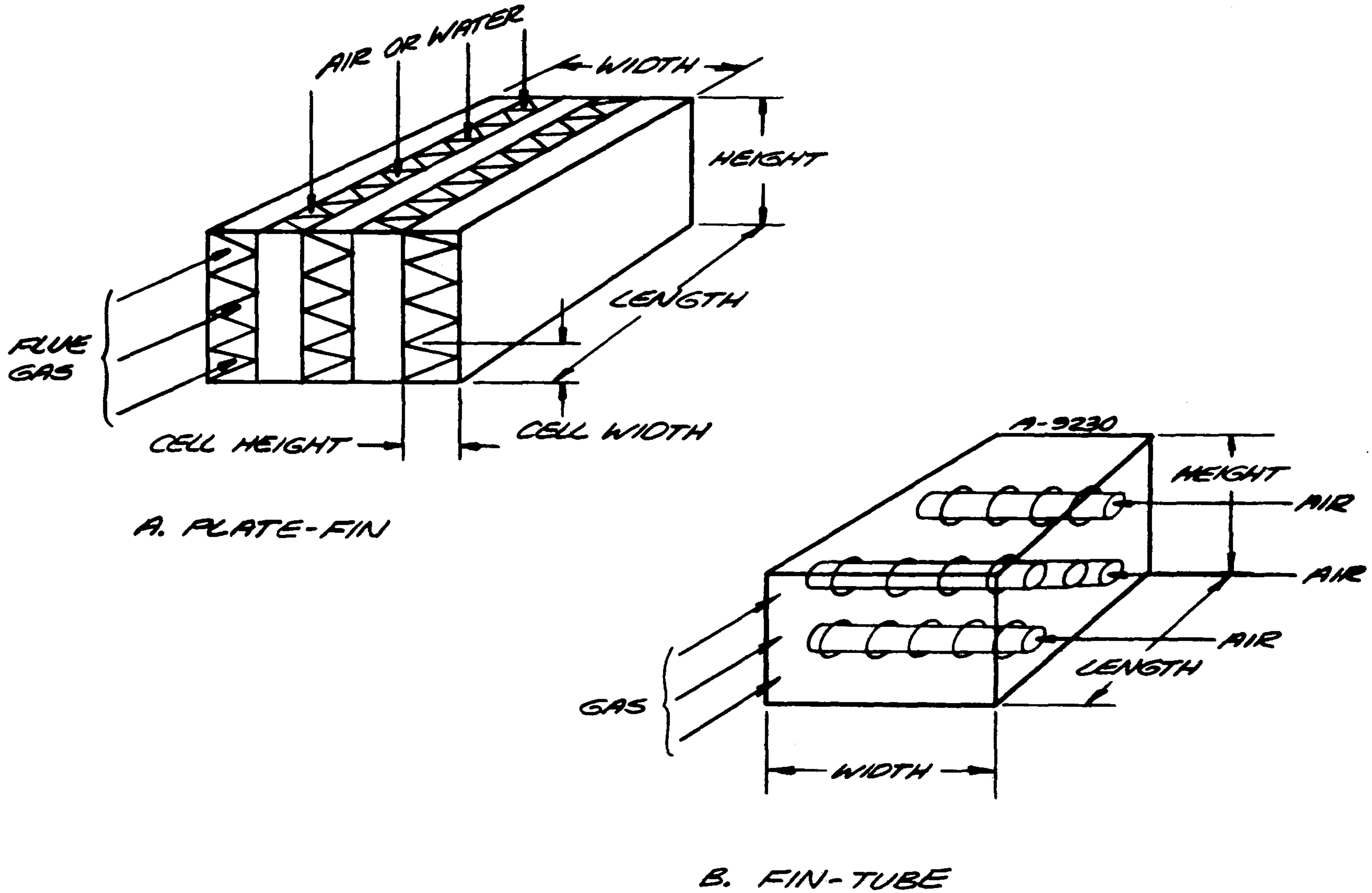


Figure 30. Compact Heat Exchanger Designs

TABLE 20

HEAT EXCHANGER PERFORMANCE PROGRAM (PERF)

A Input	B Output
Overall Dimensions	All input information
Fin Dimensions	Cold Side Reynolds Number Hot Side Reynolds Number
Hydraulic Diameter, Cold Side	Cold Side Heat Transfer Coeffecient Hot Side Heat Transfer Coefficient
Hydraulic Diameter, Hot Side	Overall Heat Transfer Coefficient, U
Frontal Area	Pressure Drop Each Side
Properties of Fluids	NTU's (see the Appendix)
Properties of Heat Exchange Materials of Construction	Overall Effectiveness, ε
Inlet Temperature, Cold Side	Efficiency Coefficient = Vol/ε
Inlet Temperature, Hot Side	Outlet Temperature, Hot Side
Mass Flowrate, Cold Side	Outlet Temperature, Cold Side
Mass Flowrate, Hot Side	
Heat Exchanger "j" and "f" Factor as Function of Reynolds Number	

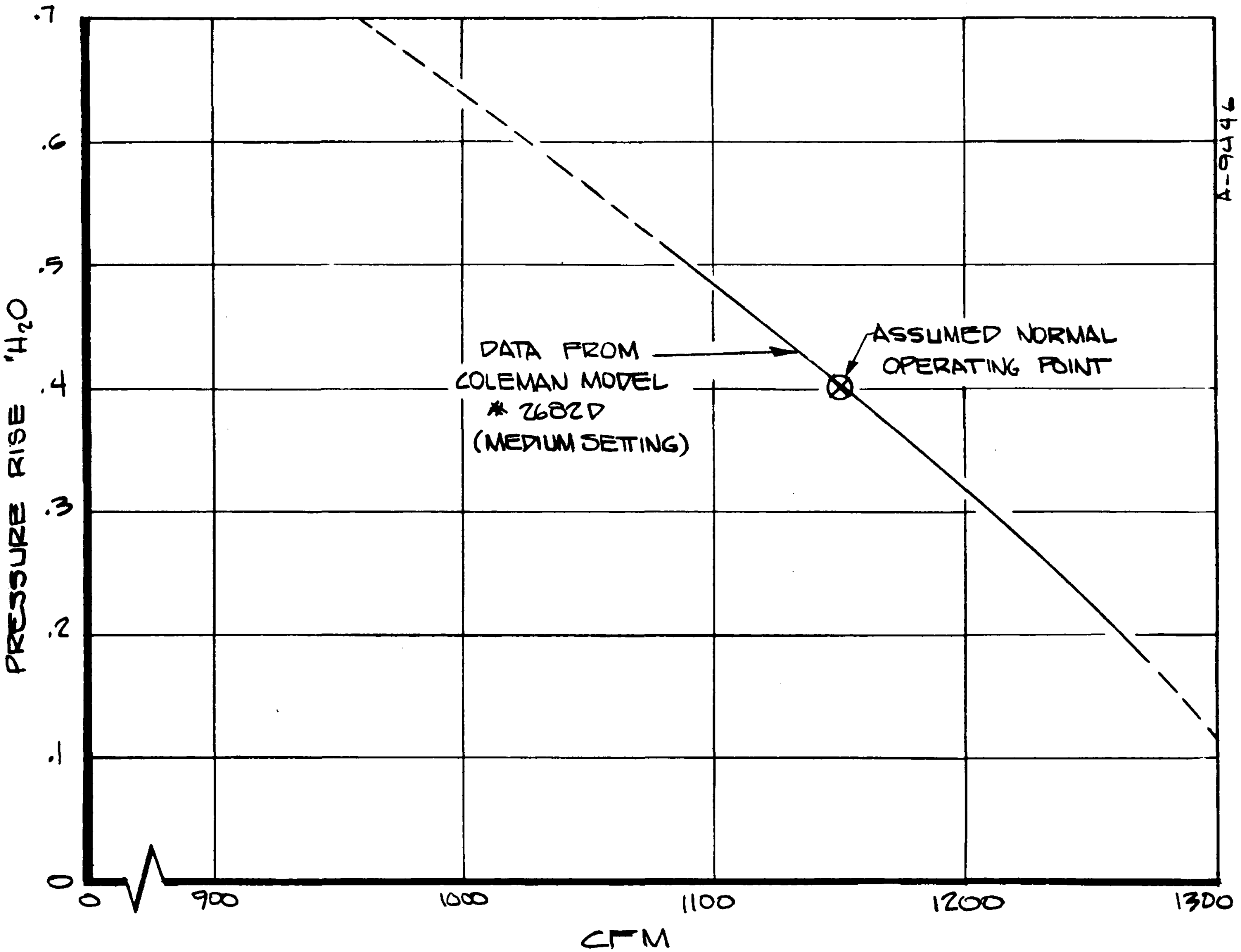


Figure 31. Typical Blower Performance Curve

Pressure drop on the flue gas side will be determined by the selection of an induced draft fan. Pressure drops as high as several inches of H_2O may be tolerated with a fairly inexpensive blower (less then $20). If an induced draft fan is not used we are limited to approximately .02" for gas systems and .05" for oil fired furnaces.

5.1 SCHEMES 3A AND 4A

A series of runs was made using PERF looking at various plate-fin and tube fin arrangements and overall dimensions with the objective of achieving the highes heat transfer within the smallest volume and within certain pressure drop limitations. For an air system the pressure drop limitation on the air side is controlled by the recirculation blower and system pressure drop.

The results of the PERF runs for the noncondensing heat exchanger and hand calculations for the condensing heat exchangers are summarized in Table 21 for the plate fin configuration. Similarly, Table 22 tabulates the results for a fin tube configuration for Schemes 3A and 4A. Figures 32 and 33 illustrate the concept and provide more detailed data on the surfaces chosen. Several surfaces were studied for which heat transfer data were available from Kays and London (Reference 49). The surfaces shown in Figures 32 and 33 were chosen on the basis of maximum heat transfer surface per unit volume within the pressure drop constraints. Numerous other plate fin surfaces such as louvered fins, strip fins, wavy fins, pin fins and perforated fins were not considered, and similarly many other fin tube designs could not be explored within the scope of this project. For these reasons the chosen surfaces therefore do not necessarily represent the optimum configuration of all possible surfaces or even for these surfaces but should represent a typical heat exchanger that might be built.

Notice that the pressure drop for the finned tube approach on the gas side may be low enough to be adaptable to an oil fired furnace without using an induced draft fan.

Figures 34 and 35 show the performance in graphical form for various inlet temperatures for the noncondensing heat exchangers. Figures 36 and 37 show the performance of the condensing plate fin and fin-tube heat exchangers respectively. These performance curves are only approximate since constant effectiveness has been assumed.* This assumption does not take into account the change in heat transfer coefficient with temperature.

*This is not true for the condensing portion of the condensing heat exchanger. The overall effectiveness actually increases with $T_{F_{IN}}$.

TABLE 21

PLATE FIN HEAT EXCHANGERS - FLUE GAS TO AIR

	Noncondensing		Condensing	
Scheme No	(3A),(4A-W)		(4A-C)	
Width (in)	7.137		15.0	
Height (in)	1.17		1.508	
Length (in)	15.00		20.00	
	Cold	Hot	Cold	Hot
No. of Passes	1	1	1	1
Hydraulic Diameter (ft)	.01259	.0094	.01259	.0094
Fin Height (in)	.544	.249	.544	.249
Cell Width (in)	.195	.168	.195	.168
Reynolds No.	2646	1234	882	480
Mass Flow Rate (lb/min)	75	2.07	75	2.07
Noncondensing Heat Transfer Coef. (Btu/hr-ft²°F)	11.69	10.47	7.02	7.11
Condensing Heat Transfer Coef. (Btu/hr-ft²°F)	-	-	-	~27
Total Heat Transfer Area (ft²)	14.45	8.97	53.3	32.45
Pressure Drop ("H_2O)	.123	1.72	.0187	.82
Inlet Temp (°F)	140	400	70	400
Outlet Temp (°F)	147	181	77	100
Overall Effectiveness	.84		.78	
Total Heat Transferred (Btu/hr)	6785.		14621	
Vol/E - Effectiveness	149.		471.	

TABLE 22

FIN TUBE HEAT EXCHANGER - FLUE GAS TO AIR

	Noncondensing 3A, 4A-N		Condensing 4A-C	
Scheme No.	1		2	
Width (in)	4		4	
Height (in)	17.6		23.4	
Length (in)	12.4		17.6	
	Cold	Hot	Cold	Hot
Fluid	Air	Flue Gas	Air	Flue Gas
No. of Passes	1	3	1	4
Hydraulic Diameter (in)	.964	.23	.350	.154
Tube and Fin Diameter (in)	-	1.737	-	.975
Fin Pitch (Fins/in)	-	8.8	-	8.72
Tubes/Frontal Area	45	3	484	6
Total Number of Tubes	45	45	484	484
Flowrate (lbs/min)	75	2.07	75	2.07
Reynolds Number	33853	598	9070	328
Noncondensing Heat Transfer Coefficient (Btu/hr-ft^2°F)	13.8	8.0	17.3	7.64
Condensing Heat Transfer Coefficient (Btu/hr-ft^2°F)	-	-	-	~27
Total Heat Transfer Area (ft^2)	4.02	46.1	14.8	155
Pressure Drop ("H_2O)	.263	.075	.051	.052
Inlet Temperature (Also See Figure 35)	140	400	70	400
Outlet Temperature (Also See Figure 35)	146	206	102	99
Overall Effectiveness	.74		.815	
Total Heat Transferred Btu/hr	6008		15351	
Vol/ε - Effectiveness Factor	1173		3068	

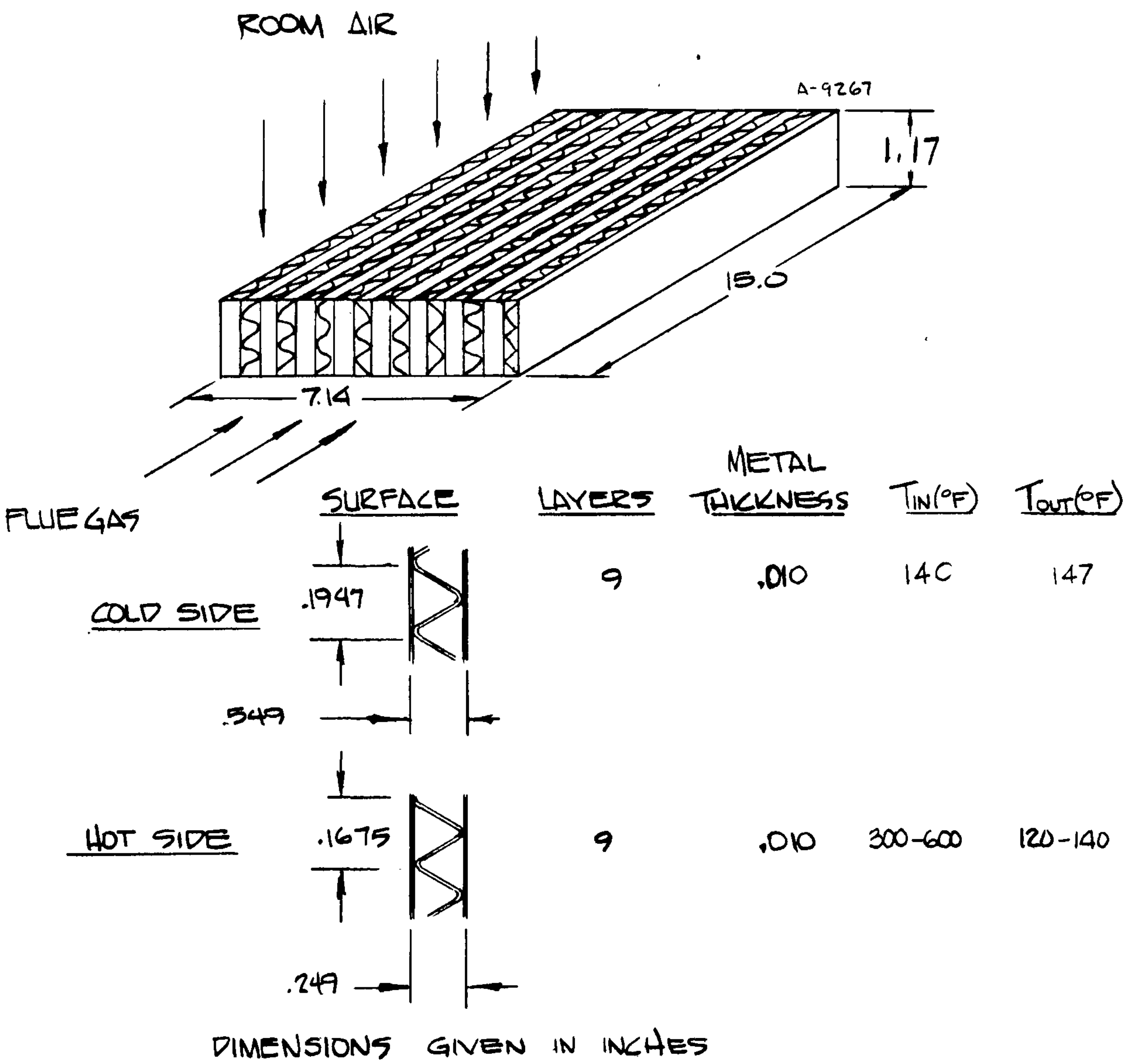
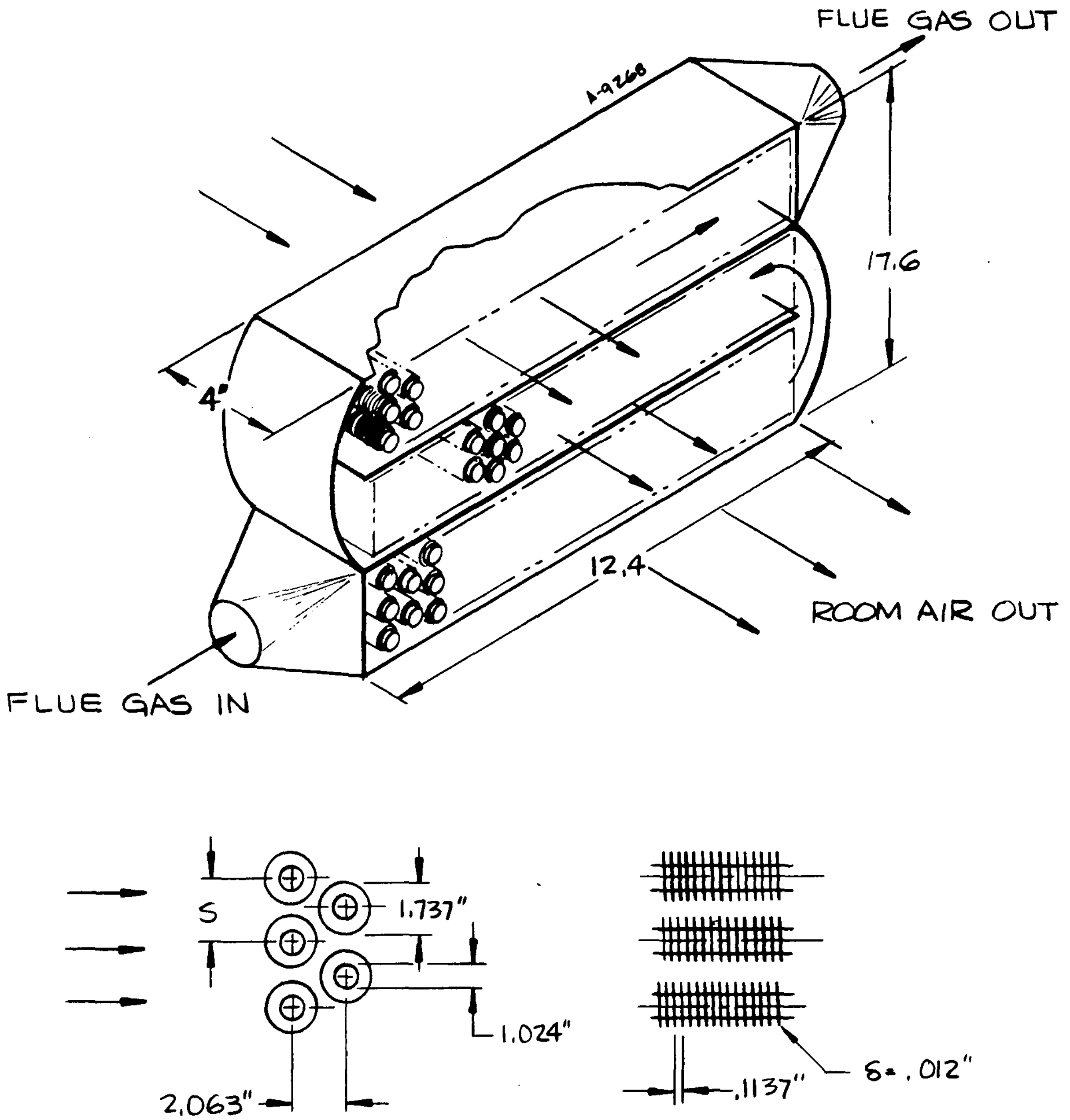


Figure 32. Plate-Fin Exchanger
Single Pass Each Side Pressure
Atmosphere Both Sides



$T_{C_{IN}} = 140\ °F \qquad T_{C_{OUT}} = 147\ °F$

$T_{H_{IN}} = 300-600°F \quad T_{H_{OUT}} = 125-140°\ F$

TUBE WALL THICKNESS = .030 , NO. OF TUBES = 45

Figure 33. Fin Tube Design
3 passes hot side
1 pass cold side
Pressure ~ atmosphere both sides

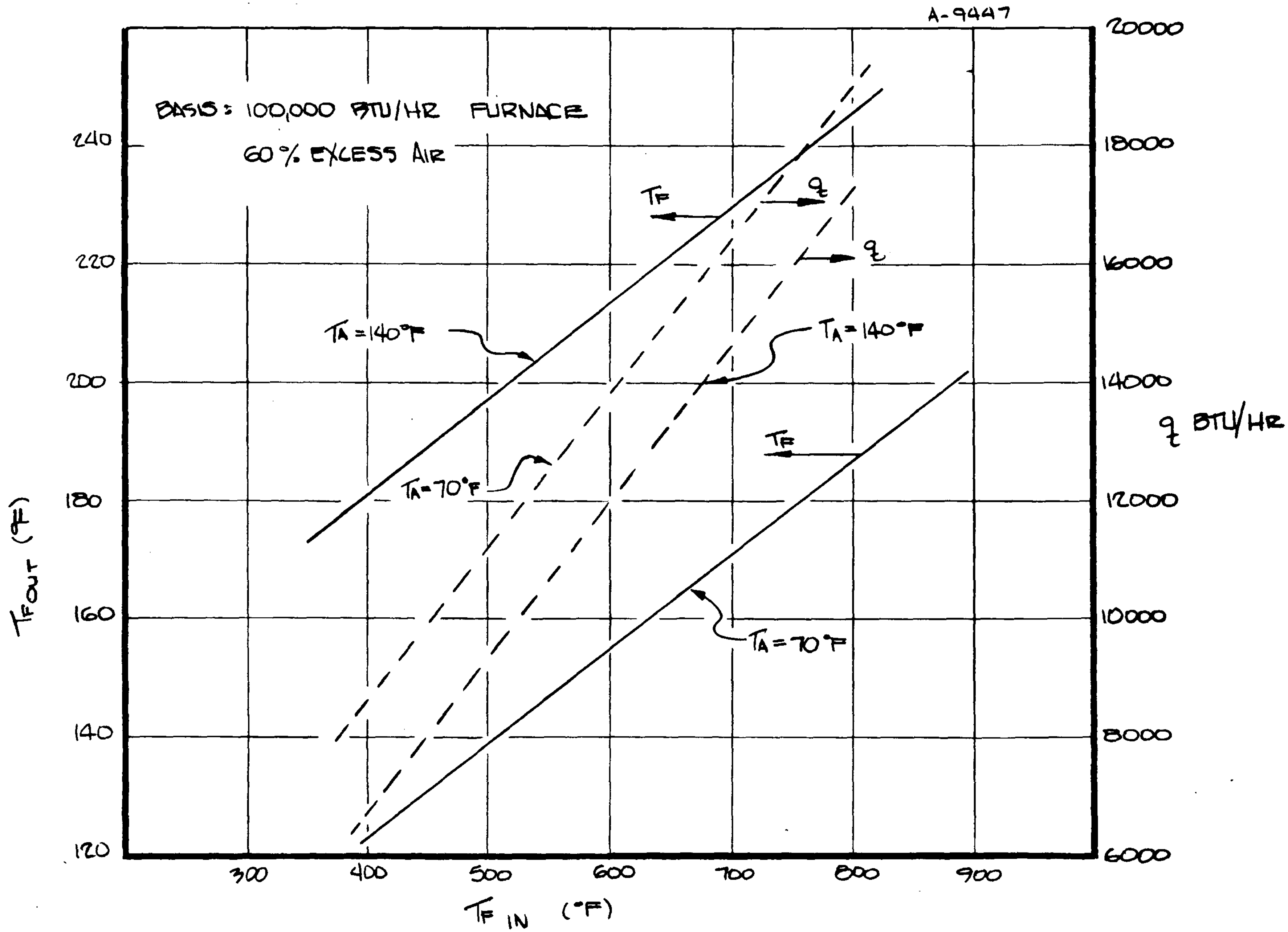


Figure 34. Performance of the Noncondensing Plate Fin Flue Gas to Air Heat Exchanger

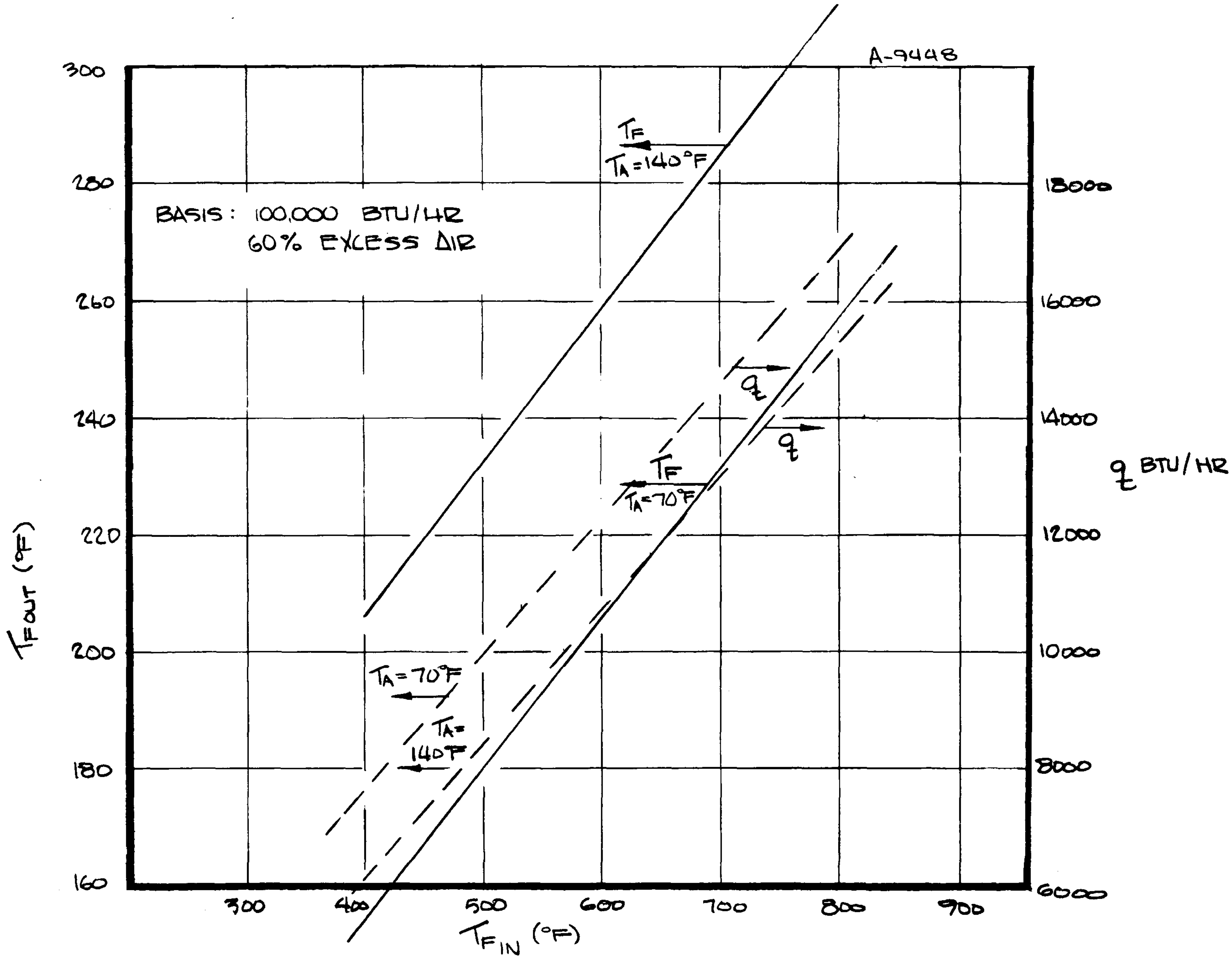


Figure 35. Performance of the Noncondensing Fin-Tube Flue Gas to Air Heat Exchanger

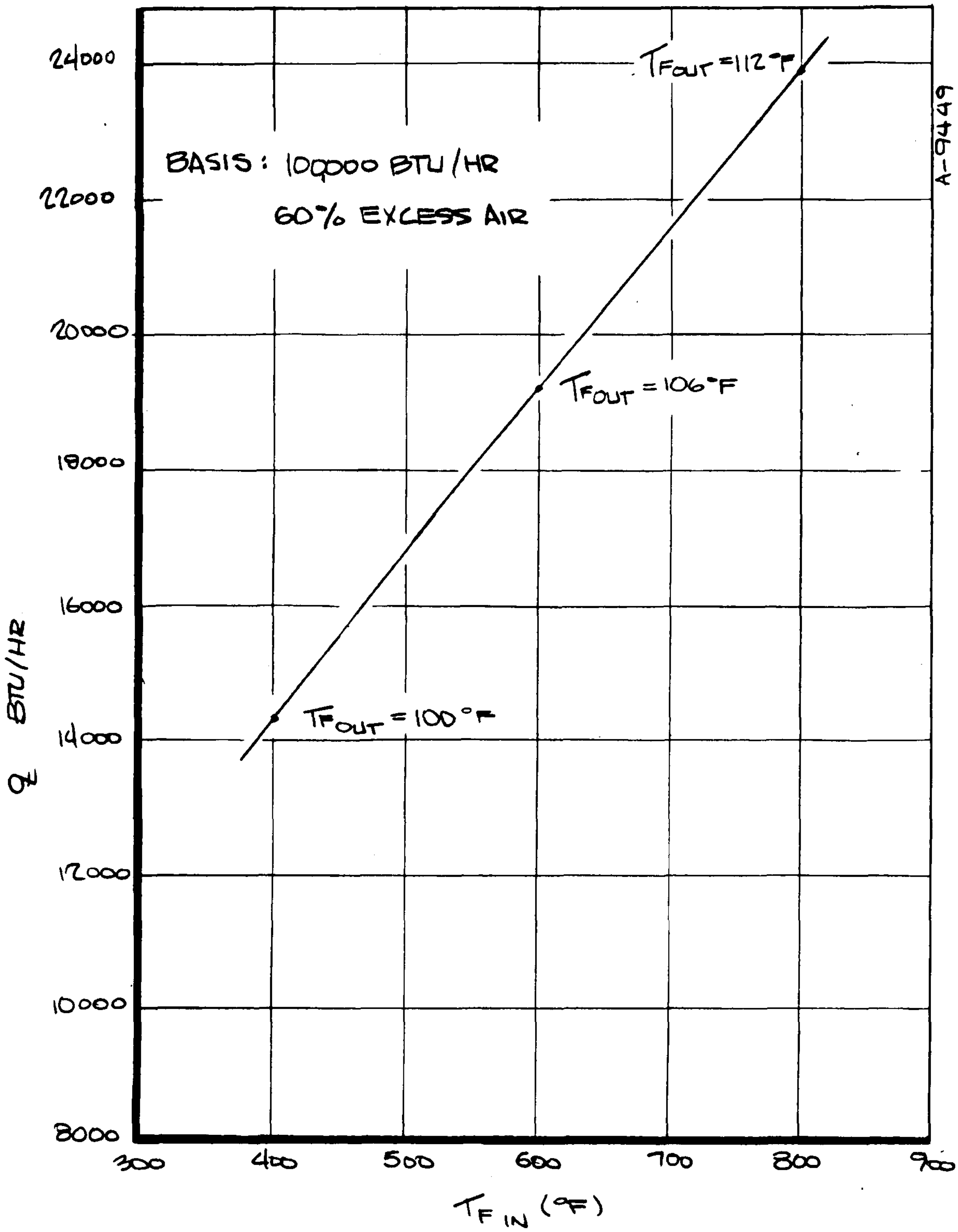


Figure 36. Performance of the Condensing Plate Fin Flue Gas to Air Heat Exchanger

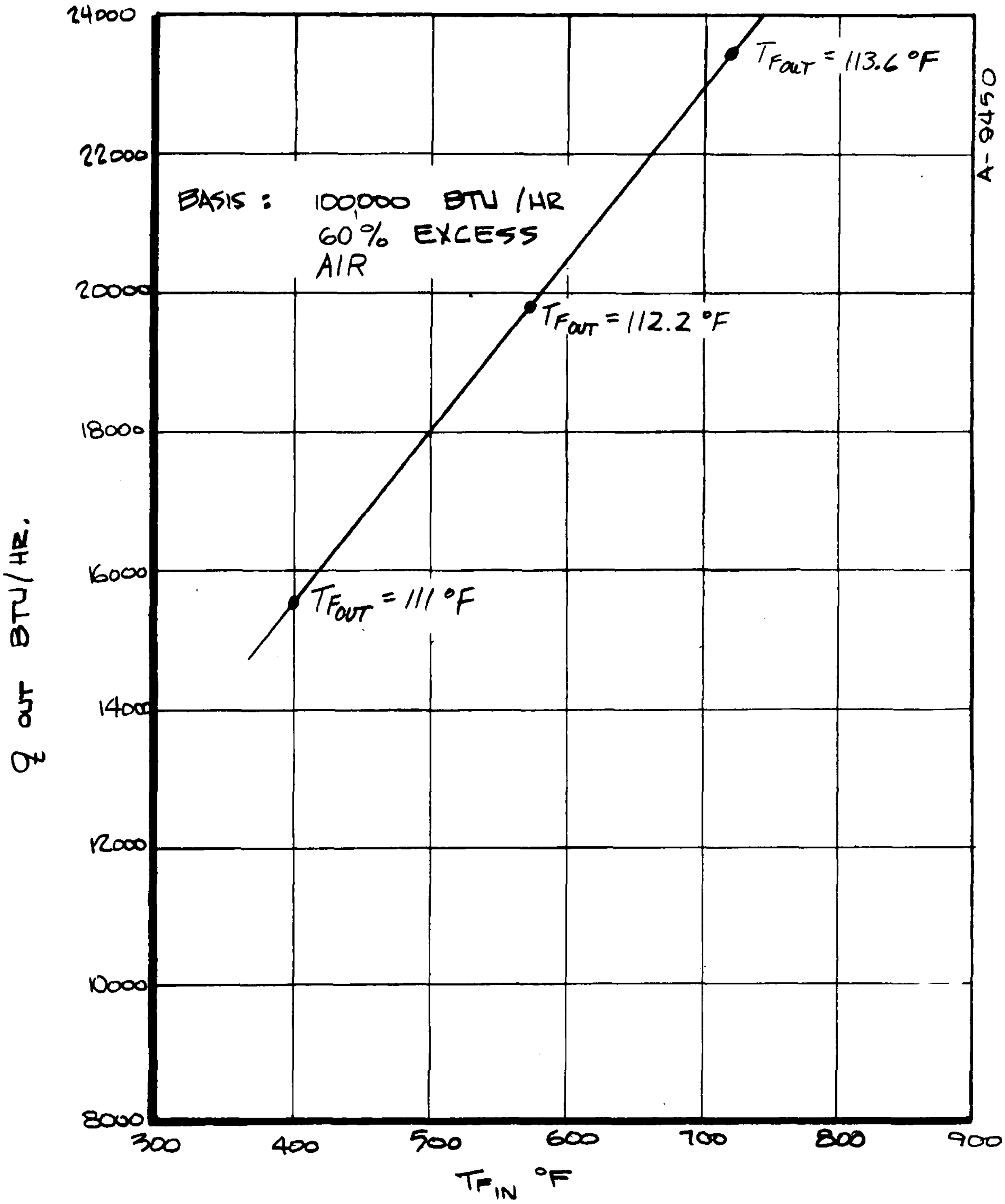


Figure 37. Performance of the Condensing Fin-Tube Flue Gas to Air Heat Exchanger

It is of interest to note that it takes approximately the same heat transfer surface area to go from 400°F to 123°F, noncondensing as it does to go from 123°F to 100°F condensing. The greater than triple surface area required is accounted for by the lower heat transfer coefficient in the noncondensing regime due to greater open flow area on the air and gas sides.

The physical size and shape of these units are such that they could be easily placed over the air inlet or outlet of the furnace in much the same fashion as a filter. For applications to a gas fired furnace the draft diverter would have to be closed off and the sheet metal and duct work to the heat exchange insulated.

For costing purposes a preliminary selection of an induced draft fan was made. The requirements for this unit are tabulated in Table 23.

TABLE 23

INDUCED DRAFT FAN REQUIREMENTS

Flow (cfm)	30-40
ΔP inch H_2O	.02-3
Temperature	120°F - 300°F

Preliminary investigation show that a vacuum cleaner type blower running at 3450 RPM (normal RPM and pressure is 10,000 RPM and 80 inch SP) would produce about the right flow and pressure. This blower is readily available from the leading manufacturers for under $20.

5.2 SCHEME 4W

Only a fin-tube design was determined for Scheme 4W, exchange with a cold water stream. Figure 38 shows the chosen configuration for noncondensing unit. Flue gases pass over a bank of forty finned tubes in a single mass and water flows through a manifold to feed four parallel paths with ten passes each. The high number of passes are necessary to keep the heat transfer coefficient on the water side relatively high (~100 Btu/hr-ft^2°F). Figure 39 shows the chosen configuration of a condensing unit. In this case flue gases pass over a bank of 128 finned tubes in a single pass and water flows through a manifold to feed eight parallel paths with sixteen passes each. Table 24 tabulates the design information and base performance calculations for the noncondensing and condensing unit. The frontal area on the flue gas side for

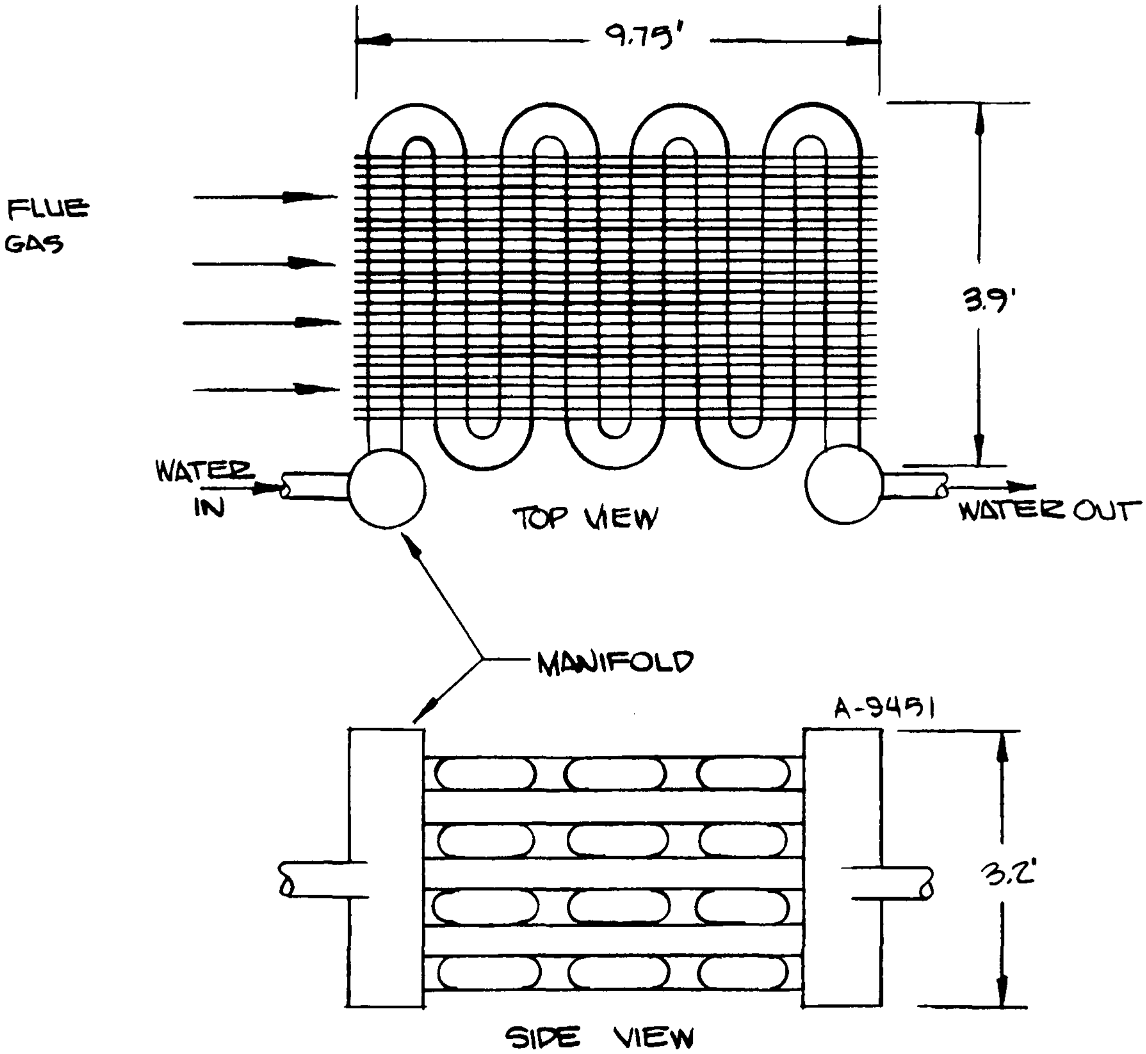


Figure 38. Noncondensing Flue Gas to Water Fin Tube Heat Exchanger (Scheme 4W-N)

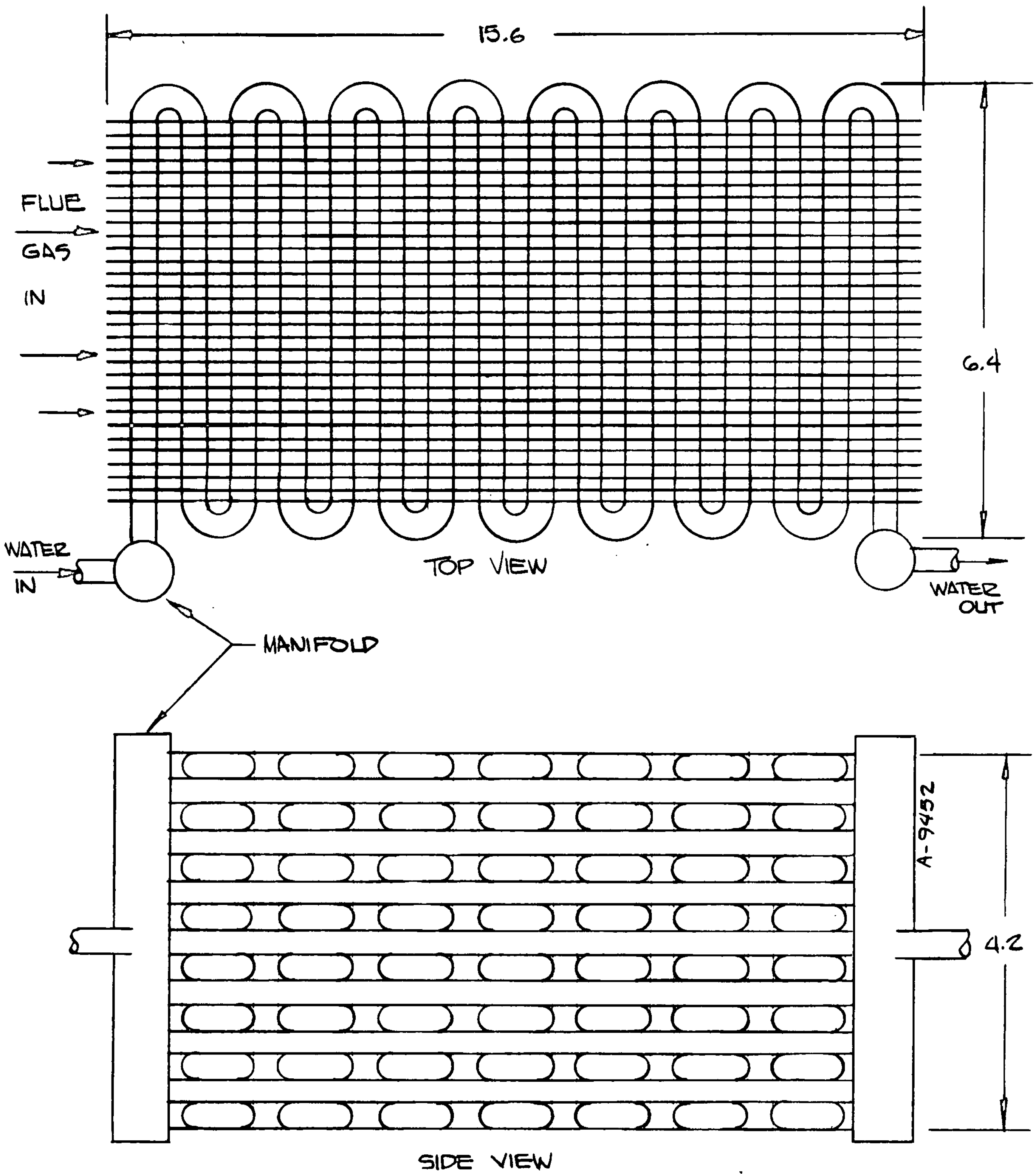


Figure 39. Condensing - Flue Gas to Water Fin Tube Heat Exchanger (Scheme 4W-C)

TABLE 24

FIN TUBE HEAT EXCHANGER - FLUE GAS TO WATER

	Noncondensing		Condensing	
Scheme No.	4W-N		4W-C	
Width (in)	3.9		6.4	
Height (in)	3.2		4.2	
Length (in)	9.75		15.2	
	Cold	Hot	Cold	Hot
Fluid	Water	Flue Gas	Water	Flue Gas
No. of Passes	10	1	16	1
Hydraulic Diameter (in)	.350	.154	.350	.154
Tube and Fin Diameter (in)	-	.92	-	.92
Fin Pitch (Fins/in)	-	8.72	-	8.72
Tubes/Frontal Area	4	4	8	8
Total Number of Tubes	40	40	128	128
Flowrate (gal/min, lbs/min)	.5	2.11	1.00	2.11
Reynolds Number	1154	2490	1257	287.4
Noncondensing Heat Transfer Coefficient (Btu/hr-ft^2°F)	100	7.94	75.9	10.67
Condensing Heat Transfer Coefficient (Btu/hr-ft^2°F)	-	-	-	42
Total Heat Transfer Area	1.2	11.48	4.1	39
Pressure Drop ("H_2O)	~.27	.21	.15	.46
Inlet Temperature } Base Condition Also See Figure 3	65	500	65	500
Outlet Temperature } Base Condition Also See Figure 3	113	113	96.5	112
Effectiveness	.87		.73	
Total Heat Transferred Btu/hr Base Condition	11932		15758	
Vol/ε - Effectiveness Coefficient	139.9		511.3	

the condensing heat exchanger was increased to keep the pressure drop down and maintain reasonable dimensions. As a result the heat transfer coefficient in the noncondensing region is decreased and the overall volume is slightly larger than if the frontal area had been kept the same as the noncondensing heat exchanger.

The condensing heat transfer is considerably lower than the plate fin case for the air to flue gas exchanger. This is due primarily to the lower mass flux (lbs/hr-ft^2) on the flue gas side due to the greater open area. Consequently the heat transfer area for the condensing heat exchanger is over triple the area for the noncondensing heat exchanger.

SECTION 6

SYSTEM COST

This section presents the potential savings per year in fuel cost, the estimated costs for the retrofit schemes designed in Section 5, the relative costs of new furnace design, costs of conventional designs, and a summary of the costs of retrofit devices on the market.

6.1 POTENTIAL SAVING AND PAYBACK PERIOD

The potential savings for a retrofit application will depend on the flue gas temperature (for a given excess air level), the cost of fuel, and the efficiency of the device or exit gas temperature. Fuel costs vary from location to location, depending on fuel type (oil or gas), availability, and the current political situation around the world. Table 25 gives an indication of the cost of natural gas in a number of cities as of May 1974 (Reference 51).

The trade journal, Fuel Oil and Oil Heat, indicates that a September 1974 fuel oil price is \$.36/gal which is about \$.27/therm (Reference 52).

Figure 40 shows the potential dollar savings per year as a function of flue gas temperature and exit temperature. This curve is based on 100,000 Btu/hr furnace operating at 60 percent excess air, fuel cost of \$0.2/therm and is analogous to Figure 26 in Section 3. The assumption is made here that the ratio of the potential heat saved to the heat input to the furnace is the same as the therms saved per year to the total therms required per year. That is,

$$\dot{q}_S/\dot{q}_F = T_S/T_T$$

where

$\dot{q}_S$ = Heat Recovered Btu/hr (from Figure 1)

$\dot{q}_F$ = 100,000 Btu/hr

T_S = Therms saved per year

T_T = 1040 therms/year [from Hittmann Report (Reference 10)] for space heating

TABLE 25

FUEL COST
$/THERM
(Reference 51)

City	Natural Gas May 1974
San Francisco	.110
Los Angeles	.135
San Diego	.150
Baltimore	.176
Boston	.335
Chicago	.142
Houston	.134
Washington	.165
Seattle	.223
N.Y.C	.261

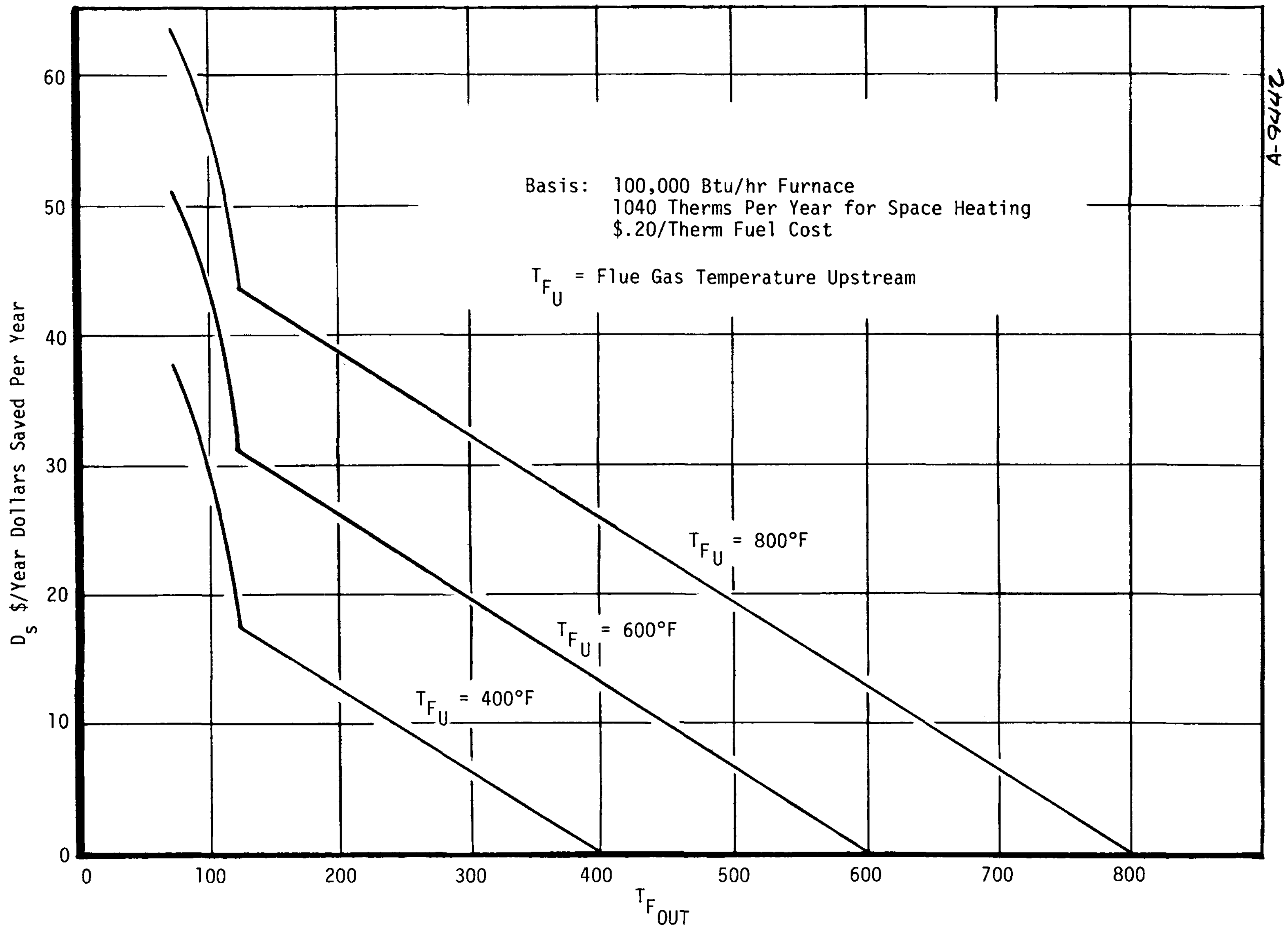


Figure 40. Potential Saving as a Function of Flue Temperature for Retrofit Heat Recovery Device

Then the dollars saved per year is equal to $D_{s_o} = .20\ T_s$

$$D_{s_o} = .00208\ \dot{q}_s$$

If higher fuel costs are experienced then this number may be multiplied by the ratio of fuel prices or

$$D_s = 5 \cdot C \cdot D_{s_o}$$

where D_{so} = Fuel savings in \$/year based on Figure 41

C = Cost of fuel \$/therm

If in addition a flue damper is used, an additional 100 (Reference 10) to 250 (Reference 53) therms per year may be saved or from \$20 to \$50 per year. This can in many cases represent as much savings potential as a heat recovery device.

Using Figure 40 as a basis, a curve was generated for the payback period versus equipment cost and heat recovered. This curve, Figure 41 is based on a 7 percent interest rate using the present worth technique (Reference 54) as outlined below.

$$D_{s_o} = I\ (crf - 7\% - n)$$

where I = Investment in dollars

D_{s_o} = Fuel savings per year

$(crf - 7\% - n)$ = Capitol Recovery Factor @ 7% for n years [see Table E-14, Grant and Ireson (Reference 54)]

$$= \frac{i(1 + i)^n}{(1 + i)^n - 1} \qquad i = .07$$

This curve also assumes the fuel costs per therm remains constant over the years. Actually, of course, the payback period will be shorter if the fuel costs increase with time. As an example, it can be shown that the dollars saved in fuel cost is then equal to:

$$D_s = \underbrace{I(crf - 7\% - n)}_{\text{Term A}} - \underbrace{\Delta D_s(gf - 7\% - n)}_{\text{Term B}}$$

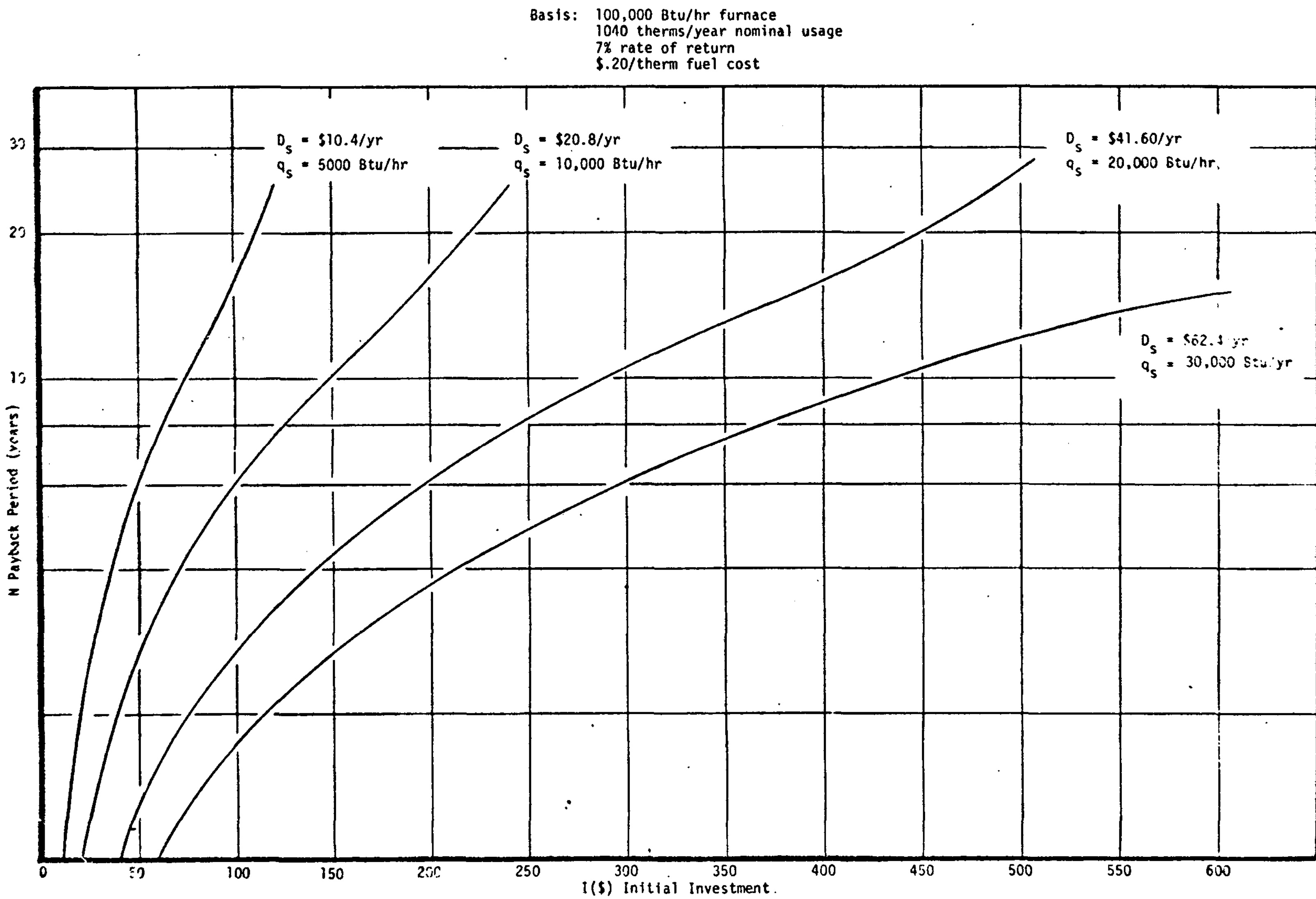


Figure 41. Payback Period as a Function of Heat Saved and Initial Investment

where

ΔD_s = additional increase in fuel cost each year

(gf - 7% - n = factor to convert a Gradient Series to a Equivalent Uniform Annual Series at 7% for n years. (See Table E-23, Reference 54)

If we let

$I = \$200$

$D_s = \$42.4/\text{year}$

$\Delta D_s = \$4.24/\text{year}$ (10% increase)

Then the following tabulation may be computed from the tables of Reference 54.

n	Term a	Term b	D_s
4	59.0	6.02	52.97
5	48.6	7.89	40.71
6	41.8	9.75	32.05

Interpolating yields a payback in 4.8 years as opposed to 6 years as given by Figure 41.

6.2 COST OF RETROFIT UNITS

Estimates were obtained from ten heat exchanger fabricators for typical plate-fin and fin-tube designs (Scheme 3A). Prices for production of 1000 units and 10,000 units per year were requested. Both designs were based on using a low grade stainless steel throughout. Table 26 summarizes the range of prices obtained.

TABLE 26

HEAT EXCHANGER COSTS

	1000 Units/Year	10,000 Units/Year
Plate Fin (Figure 32)	\$50 - \$400	\$24 - \$250
Fin Tube (Figure 33)	\$100 - \$600	\$50 - \$500

From these data estimates were then made for each scheme including factors such as an induced draft fan, ducting changes, insulation, and installation costs. These data are summarized along with the payback period based on Figure 41 in Table 27. the Dolin, Isothermics and Vent-O-Matic devices are also included for comparative purposes.

This table shows that in terms of payback the flue damper is the best buy. It also reveals that the most promising devices are those that are applied to oil fired furnaces because: (1) The oil fired furnaces are initially less efficient (higher flue gas temperatures), (2) It takes more hardware to convert a gas furnace (replacement of the draft diverter) and (3) The cost of oil is higher. It should be remembered that the estimated costs are rather speculative and could be considerably higher. Also if one assumes a higher interest reate (for example the interest rate to secure a loan (approximately 14 percent) rather than a nominal rate of return from a savings and loan), the payback period will increase in all cases. It should also be remembered that as the price of fuel increases so will the price to manufacture these items.

6.3 NEW FURNACE DESIGNS

At this time none of the new furnace schemes has been designed in enough detail to allow an accurate estimatation of costs. However, we should be able to set a reasonable upper boundary on system costs to be economically feasible.

Assume for the sake of argument that the following improvements to the heating system produce the indicated savings in energy consumption (compared with the fuel consumption for the unidentified case):

Improvement	Assumed Savings (%)
Improved furnace efficiency	10
Furnace modulation	25
Flue damper	15
Total	50

Yielding a total reduction in energy consumption of 50%. At an assumed fuel cost of \$.20/therm this would amount to $D_s = (0.2) \times (.50) \times (1040) = \$104/\text{year}$.

Next we must assume some value for an assumed lifetime of the furnace. Ideally, the furnace should last for 15-25 years. On the other hand, a typical average residence time for a homeowner is more on the order of 5 years.

TABLE 27

SYSTEM COSTS AND PAYBACK PERIOD

Scheme Number	3A	3A	4A-C	4A-C	4W-C	4W-C	Dolin	Isothermics	Ventomatic
Exchange Heat W/	Air Out	Air Out	Air In	Air In	Water	Water	Air	Air	-
Fuel	Gas	Oil[1]	Gas	Oil[1]	Gas	Oil[1]	Oil[1]	Oil[1]	Gas
Heat Exchanger[2]	100	100	150	150	300	300	130.00	100	-
Ducting & Misc.[3]	15.00	10.00	30	20	15	10	15.00	15	-
Insulation	7.50	5.00	10	7.50	7.50	5.00	-	-	-
Induced, Draft Fan	15.00	15.00	15	15.00	15.00	15.00	-	-	-
Installation[4]	40.00	32.00	100	92.00	72.00	64.00	24.00	16	-
Total Cost	177.5	162.00	305	284.5	409.5	394.0	169.00	131	87
Heat Saved (Btu/hr)[5]	4,223-12,000	2,400-17,400	11,800-16,800	16,800-24,000	13,308-17,718	17,718-26,396	3,000-5,300	5,000-12,500	104 → 390[6]
Fuel Cost Saving Dollar/yr	9-25	26-49	25-35	47-67	28-37	50-74	8-15	14-35	21-78
Payback Period (Years)	10 → ∞	3.75-8.5	13.5-27	5.5-7.5	22-45	6.5-11.5	22 → 100	4.5-15	1.2-5

[1]For oil systems the fuel price is based on \$.27/therm (\$.35/gal)
[2]For plate fin heat exchanger except for water/gas heat exchange which is fin-tube; costs estimated from data of Table 26.
[3]Includes closing off integral draft diverter and installation of in-duct draft diverter for gas systems; Dolin and Isothermics, assumes heat is routed into main air system.
[4]Installation costs will vary depending on labor rate; assumption here is \$8/hr.
[5]See Table for assumptions and supporting data
[6]For Ventomatic, heat saved is expressed in therms/year (Reference 55).

It seems unlikely that a homeowner would thus be willing to pay for a furnace lifetime longer than, say, 7-10 years unless he was able to be assured of a return on his investment when he sold the house. For the moment let the lifetime or payback period be a variable.

In addition, assume that fuel costs will increase uniformly at 10 percent of the first year at an assumed price of $.2/therm. Then there will be an additional $10.40/year savings added each year. The initial investment delta will then be equal to:

$$\Delta I = \frac{104. + 10.4(gf - 7\% - n)}{(crf - 7\% - n)}$$

with (gf - 7% - n) and (crf - 7% - n) as defined in Section 6.2. Figure 42 presents the results of this expansion for n's from 2 to 25 years. This shows that if a homeowner accepts a 25-year lifetime or payback period, he should be willing to invest up to $2250 to achieve the assumed improvement in efficiency. However, because the historical philosophy of buying has been one of lowest initial cost, it may be quite difficult to convince the public to purchase on the basis of a long term total investment.

In fact, however, the assumed 50 percent savings can probably be achieved for an additional $500 to $1000 with the new designs. Figure 42 shows the payout could be achieved within 5 to 10 years, or perhaps sooner if fuel prices increase at a faster pace than assumed.

As a support example, the Westinghouse Utilization Project (Reference 56) showed that in most U.S. cities the total annual cost of a heat pump, more costly than conventional equipment, was lower than either a gas or oil system, for typical conditions. Table 28 presents this cost advantage for a number of U.S. cities.

6.4 COST OF CONVENTIONAL DESIGNS

There seems to be a wide spread in furnace costs for a typical 100,000 Btu/hr furnace. Gas systems vary anywhere from $200 to $600 depending on quality, lifetime guarantee, sophistication of controls, and ability of the fan to produce the high air flow rates required for air conditioning. All AGA approved furnaces guarantee the 80 percent steady state efficiency. New oil fired units are selling for $500 to $1200 with efficiency of 75-80 percent. Heat pump systems are currently selling for around $1600 and total heating and air conditioning systems sell for $1100 to $1600.

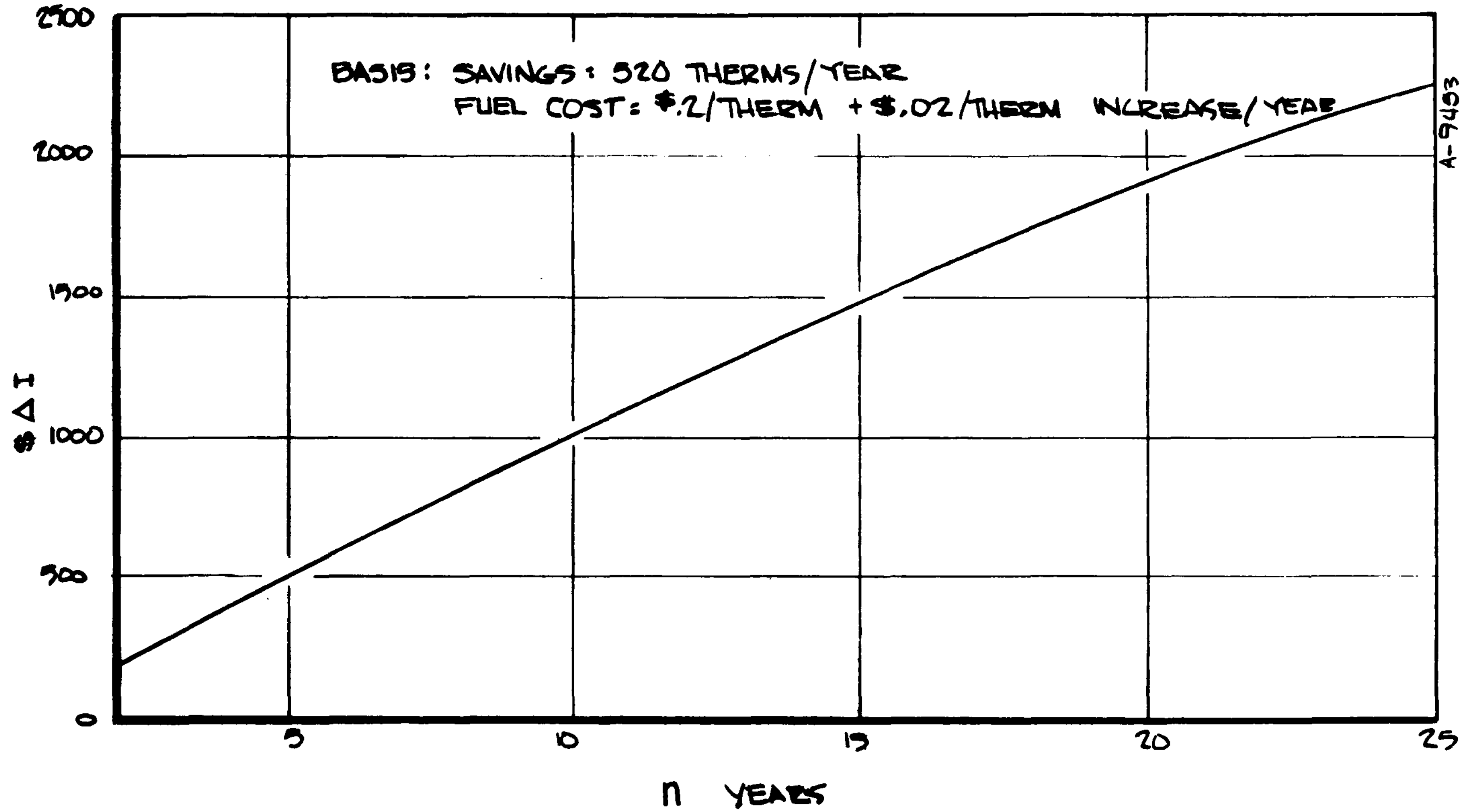


Figure 42. Initial Investment Increment vs. Lifetime or Payout Period

TABLE 28

ANNUAL SAVINGS TO HOMEOWNER USING A
HEAT PUMP VS. EQUIVALENT COMFORT-CONDITIONING SYSTEMS
(Reference 56)

	Fuel Rates(10)#		Annual Fuel Savings - $		Total Annual Savings - $	
	Electric (¢/KWH)	Gas (¢/Therm)	Heat Pump Vs. Gas	Heat Pump Vs. Oil*	Heat Pump Vs. Gas	Heat Pump Vs. Oil*
Minneapolis	1.45	10.5	-39	85	-60	111
Milwaukee	1.56	12.9	40	92	19	118
Pittsburgh	1.59	11.9	15	93	-6	119
Oklahoma City	1.00	7.7	23	111	2	137
Atlanta	1.30	8.9	24	84	3	110
Sacramento	.88	9.036	43	98	22	124
Tampa	1.85	12.0	11	19	-10	45

#Includes all adjustment and surcharges except fuel adjustment.

*Since oil price is increasing rapidly, a fixed value of $.20/gallon was used here.

However, of the total furnace system sold during a year a large percentage are for new homes and particularly for tract developments. In these situations the lowest initial cost system is undoubtedly preferred. (Manufacturers have stated that a $2 increase in manufacturing cost is a significant factor). The price of tract furnace systems may be considerably lower than the above retail prices. Convincing tract developers to utilize a more expensive furnace may be a more formidable task than convincing the public.

6.5 CONCLUSIONS

The preliminary estimates of this section indicate that the amount of savings provided by a new high efficiency furnace system design appear attractive, with payback periods in the 5- to 10-year range, and at costs which are roughly double the cost of a current system. However, although the large scale economics appear attractive, formidable social and institutional obstacles would act to prevent the widespread use of such new furnace designs. New furnaces are sold to a public which is extremely conscious of initial capital cost and relatively unaware of energy management and engineering. This is particularly true in the new residence market, which accounts for a very appreciable portion of furnace sales. Heat furnace design has settled into a nominal efficiency range of 70 to 80 percent reflecting a conjunction of market forces which have integrated the typical effects of initial costs, fuel costs, home turnover rate, and purchase behavior in buying new and used homes. However, fuel prices and inherent rates tend to counteract, and the apparent economic optinum efficiency has shifted upward only slightly. Other reasons, such as long term conservation of valuable fuels, make the desired efficiency somewhat higher still, but it will be very difficult to integrate such new factors into the market system. It may be necessary to consider federal efficiency standards.

SECTION 7

CONTROL OF AIR POLLUTANT EMISSIONS FROM RESIDENTIAL HEATING EQUIPMENT

7.1 INTRODUCTION

The preceding sections of the present study have dwelt on the possible methods for increasing the efficiency of existing residential furnaces. It was determined that a number of controlling variables exist for this purpose, such as modifying heat exchanger design and reducing the frequency of cyclical operation. Likewise, there exists a set of options for the control of air pollutant emissions from gas-and oil-fired heating systems. This section is concerned with defining the most important of these emission reduction techniques.

Distillate oil and natural gas heaters account for 90 percent of the residential heating units in the U.S. (Reference 57). The actual contribution of fuel combustion in this equipment to the total national air pollutant loading (summation of mass of particulate, SO_x, NO_x, CO, and unburned hydrocarbons) is unclear. One investigator estimated that air pollution from the combination of residential and commercial heating sources constitutes about 10 percent of the national total (Reference 58). This figure, however, may be outdated; the source was published in 1964. Fuel usage and emission factors were the basis for this estimate.

Another data source, the National Emission Data System (NEDS), does not compile information for sources that emit less than 25 tons/year of any pollutant. This limitation causes the exclusion of residential heating from consideration. From all indications, however, their contribution to the nation's total emissions probably falls in the 1-10 percent range. In any case, their impact was deemed sufficiently significant that EPA has instigated an emission reduction R&D program (see Section 7.3).

Typical emission levels are presented in Table 29. The test comparisons between three gas furnaces and three oil furnaces indicate that the level of gaseous pollutant emissions from the gas burners is comparable to those from equivalent size oil burners, with particulate being a problem only from the latter. Compared to other sources of air pollution, the levels of

TABLE 29

EMISSIONS FROM NATURAL GAS- AND OIL-FIRED BURNERS
(Reference 57)

Burner Manufacturer	Stoichiometric Ratio	NO, g/10^6 cal input	HC, g/10^6 cal input	CO, g/10^6 cal input	
Gas-fired:					
Williamson furnace	1.20	0.084	0.0007	0.022	
Bryant boiler	1.40	0.115	0.0014	0.099	
Bryant furnace	1.60	0.112	0.0075	0.032	
		0.104	0.0032	0.051	Average Emissions
Oil-fired:					
Union (Pure)	1.20	0.115	0.0055	0.046	
ABC Mite	1.38	0.071	0.0055	0.046	
ABC Standard (Model 45)	1.53	0.102	0.0055	0.046	
		0.096	0.0055	0.046	Average Emissions

HC and CO from these properly adjusted furnaces are very low (Reference 57).

The remainder of this section will review the major options available for combustion or post-combustion control of emissions from gas- and oil-fired domestic furnaces. The principal recently-completed or current research programs aimed at reducing these emissions are described as well.

7.2 CONTROL STRATEGIES FOR EMISSIONS FROM RESIDENTIAL HEATING SYSTEMS

The three major options for reducing pollutant emissions from combustion sources include fuel substitution, combustion process modification, or post-combustion control. Fuel switching (to gas) would logically be applied only to oil-fired furnaces, mainly for particulate abatement. However, the associated equipment conversion required by such a move would be economically unfeasible. This leaves combustion and post-combustion control as the most viable means for achieving emission reduction goals. These are discussed separately in the following subsections.

7.2.1 Combustion Process Modification

Reducing emissions evolved by flames by modifying the combustion process itself has, in recent years, gained popularity as a NO_x control strategy for industrial and utility boilers. The same principles can be applied on a much smaller scale but with equal validity to residential heating systems. As with the larger scale combustion equipment, care must be taken to avoid emissions trade-off problems, that is, increasing one pollutant species due to a control aimed at reducing another.

The important combustion control emission reduction strategies include:

- Excess air level adjustment
- Modification of combustion chamber design
- Installation of flame retention burners
- Ignition system state-of-repair
- Regular service and maintenance

Descriptions of these control options appear in Table 30, with comments describing the relevant effect of each option.

TABLE 30

COMBUSTION CONTROL STRATEGIES FOR
REDUCING AIR POLLUTANTS FROM
RESIDENTIAL HEATING EQUIPMENT

Control Strategy	Impacted Pollutant Emission	Comments	References
Excess air level adjustment	NO	• As excess air is increased, CO HC, and smoke pass through a minimum, but NO passes through a maximum • Optimum pollutant and thermal efficiency level occurs at a stoichiometric ratio greater than one (~1.6 for oil burners).	59, 57, 60
Combustion chamber design	NO CO HC Smoke/particulate	• Combustion chamber design affording long residence time at high temperature minimizes smoke, particulates, CO, HC, but may increase NO • Refractory-lined chamber affords better combustion and lower emissions	57, 60, 61
Flame retention burners	Smoke	• Oil burners with retention-type end cones give superior emission and efficiency performance	57, 60
Ignition system condition	NO	• No effect on HC and smoke • Minimal effect on NO	57, 60
Service and maintenance	NO CO HC Smoke/particulate	• Equipment state-of-repair very important for providing breadth for reducing emissions by other methods • Oil nozzles should be changed each season • Air filter should be replaced regularly	59, 57, 60

7.2.2 Post-Combustion Control

The second alternative available for reducing undesirable furnace emissions is the treatment of the combustion product gases, usually by a retrofit device installed in the flue. Most of the candidate devices are designed to abate a single pollatant, and combinations of them may be used for multiple pollutant species reduction.

The principle types of flue gas treatment (FGT) devices for possible application to home heaters include the following:

- Fabric filters
- Wet scrubbers
- Wet, packed beds
- Activated charcoal filter
- Cyclone
- Combinations of devices

Table 31 describes these devices in greater detail.

In all, the feasibility of most of these devices for residential heating applications is unknown. The success attained so far by combustion process modification may negate possible reduction benefits afforded by the less well-defined FGT devices.

7.3 PRINCIPAL RECENT OR CURRENT RESIDENTIAL EMISSION REDUCTION R&D EFFORTS

As mentioned in Section 7.1 of this report, there is a significant amount of industry and government-sponsored research and development effort currently being expended in the area of reducing emissions from residential heating equipment. These activities are described in Table 32. Information on level of funding was not available.

7.4 CONCLUSIONS

The following general conclusions have been reached concerning emission control of residential furnaces:

- The contribution of residential furnace emissions to the nation's totals for each type of pollutant should be more closely defined. This can be done by obtaining updated emission factors and furnace inventories, as well as fuel usage figures. When the actual contribution is known, a more valid decision base will be available for R&D program planning efforts.

TABLE 31

POST COMBUSTION CONTROL (FLUE GAS TREATMENT) STRATEGIES FOR AIR POLLUTANTS FROM RESIDENTIAL HEATING EQUIPMENT

Control Strategy	Impacted Pollutant Emission	Comments	References
Fabric Filter	Smoke/particulate	• Most high efficiency filters are not designed for high stack temperatures • "Absolute" - type filter (to 500°F) cause excessive Δp (~1"W.C.)	62, 63, 64
Wet Collectors	Smoke/particulate	• Some industrial uses, but feasibility for residential furnaces yet to be determined • Small units used in electronic industry may be applicable	65, 66, 63
Wet, packed beds	Smoke/particulates HC	• Efficient particulate and odor removal method • Applicability for residential use not yet determined	67, 68
Activated charcoal filter	HC	• Used frequently to remove odors and solvent aerosols • Max. temperature = 100°F; may be practical for furnace flue gas treatment if stack temperature can be reduced	63, 69, 70
Electrostatic precipitator	Smoke/particulate	• ESP's designed for commercial/residential air filtering not usuable at high temperatures found in stacks (plates warp) • Particle agglomeration on plates possibly a problem • No known ESP specifically designed for residential furnace use	71, 72, 73, 64, 74
Cyclone	Particulates	• Feasibility for residential use unknown	66, 75, 71
Combinations of filtration, ESP, activated charcoal	Particulates HC	• Three step system • Roughing filter • ESP for fine particulate • Activated charcoal for HC • System this elaborate probably not warranted	70

TABLE 32

PRINCIPAL RECENT OR CURRENT RESIDENTIAL EMISSION REDUCTION R&D EFFORTS

Agency and/or Organiation	Investigated Pollutant(s)	Nature of Study	References
EPA	NO_x CO HC Smoke/particulates	• Define major emission level control variables • Recommend emission minimization methods through combustion controls • Status: Continuing	57, 60
Battelle	NO_x CO HC Smoke/particulates	• Field investigation of emission from oil-fired equipment • Found serious emission trade-off difficulties if tuning to low smoke level performed without monitoring instrumentation • Made comparisons of measured emissions with EPA emission factors • Status: final report submitted June, 1973	59
Walden		• Study of Air Pollution from Intermediate Size Fossil-Fuel Combustion Equipment • Status: Final report submitted July, 1971	61
American Gas Association (sponsor) :			
• Institute of Gas Technology (IGT)	NO_x	• Study of fundamental mechanisms of the formation and suppression of NO_x in natural gas combustion • Not strictly concerned with residential heating units, but results will be relevant • Status: ongoing	18
• The Research Corporation of New England (TRC)	NO_x SO_x CO HC Smoke/particulates	• Measuring the environmental impact of domestic gas-fired heating systems • Emissions being related to ambient air quality • Status: ongoing (through 1973-74 heating season)	76

- Combustion process modification is clearly the most viable short-term option available for emission reduction.
- For abating such pollutants as particulates emitted during furnace start-up, the application of some of the candidate FGT devices may be warranted (i.e., high temperature fabric filters).
- Acquisition of detailed cost information on FGT devices, in addition to continued applied research on devices feasible for use on domestic furnaces, are required before the cost/effectiveness of such devices can be fully assessed.

SECTION 8

POTENTIAL MARKET

The potential market for either a retrofit device or a new high efficiency home furnace will depend in general on the following factors:

- New home building
- Fuel availability
- Fuel price
- Local ordinances
- AGA or similar approval
- Proof of economic viability
- National interest - fuel saving, emission prevention
- Advertising and consumer education
- Climate
- Furnace age and condition

Proof of economic viability includes overcoming the lowest initial price economic philosophy of private homeowners or tract developers as discussed in Section 6 on system costs. Ordinances and utility companies in many communities will presently not allow any device in the flue which could cause blockage. Approval of a heat recovery device which provided a safety vent in case of blockage would probably be fairly easy especially if AGA or U.L. approval was obtained. However, AGA approval is not always easy to obtain. For example, the Vent-O-Matic device was initially approved by AGA, then rescinded and only given approval for installation on new furnaces. The whole matter was brought to the attention of the Federal Power Commission, Federal Trade Commission and Senate Anti-Trust Committee and is currently under investigation. The issue is expected to be resolved by October 1974 (Reference 55). The Diermayer vent damper devices was not given approval by AGA and therefore is not being sold at this time in the U.S., although it has been in use in Europe for over 40 years.

Fuel price and availability may either be a force in favor or against sales of such a device. For example, on the West Coast, the winters are fairly mild, so far gas is plentiful and remains relatively cheap compared with prices in other parts of the country.

The potential market for new furnaces will depend on the housing building market or for replacement units the number of old furnaces which need replacing. Table 6 showed that in 1973 over 2 million gas fired furnaces were sold as well as over 700,000 units, each of oil fired central air, vented wall gas furnaces and gas fired direct heating equipment. A total of 4,892,260 of all type units were sold in 1973, adding and/or replacing part of the 58 million heating systems in the U.S. (1970). New units in 1973 therefore accounted for about 8.4 percent of the 1970 census of heating systems.

From the U.S. Census data and the GAMA data on total furnace systems sold, it was determined that approximately 36 percent of the furnaces sold during 1963 were for new housing, and the reminder were replacement units. Obviously this number will vary from year to year but using this figure, it indicates that about 5 percent of the furnace population will be replaced each year.

8.1 POTENTIAL MARKET ESTIMATE

If we assume that 10 percent of the new and replacement units could be high efficiency units, this represents a market of about 500,000 units per year.

Table 33 presents the results of an estimate of the market for a heat recovery device. It was assumed that only central air and central water or steam systems could be retrofited and that the units converted in the Southern and Western regions would only be on the order of 1 percent. Twenty percent conversion was assumed in the Northeast and North Central regions. This shows a potential of over 6 million units.

8.2 NATIONAL REDUCTION IN FUEL CONSUMPTION

The assumption was made that 500,000 new high efficiency units are sold each year and that the 6 million retrofit devices are sold according to the schedule in Table 34 with 500,000 units each year thereafter.

In addition we assumed that the new high efficiency units use 50 percent less fuel per year, that the retrofit devices decreased the older unit's fuel consumption by 20 percent and the nominal fuel usage was 1040 therms/year. From these assumptions and the data in Table 34 the total

TABLE 33

ESTIMATE OF HOT WATER AND CENTRAL AIR
FURNACE INSTALLATION FOR 1974
BY REGION (MILLIONS)

	NE	NC	S	W	All
Total Units	15.0	15.9	8.95	5.79	45.64
Assumed Converted Percent	20%	20%	1%	1%	13.87
Converted Total Units	3.0	3.18	.09	.06	6.33

NE = Northeast
NC = North-central
S = South
W = West

TABLE 34

RETROFIT SCHEDULE

Year	Retrofit Units Sold	Accumulated Retrofit Units Sold	Accumulated New Units Sold
1	2.0×10^6	2.0×10^6	0.5×10^6
2	1.0×10^6	3.0×10^6	1.0×10^6
3	0.5×10^6	3.5×10^6	1.5 x 1
4	0.5×10^6	4.0×10^6	2.0 x 1
5	0.5×10^6	4.5×10^6	2.5 x 1
6	0.5×10^6	5.0×10^6	3.0 x 1
7	0.5×10^6	5.5×10^6	3.5 x 1
8	0.5×10^6	6.0×10^6	4.0 x 1
9	•	•	
10	•	•	
•	•	•	
•	•	•	

national energy savings as a function of a year were calculated. These savings and the split between gas and oil are presented in Figure 43. The split between fuels is based on the 1970 census data of the distribution of furnaces according to fuel. Figure 44 and 45 show the annual gas saving in ft^3/year and oil saving in barrels/year.

If these products were available for production in 1975, they could affect the total U.S. energy consumption and total space heating energy consumption as tabulated in Table 35. This table indicates that after 8 years residential space heating energy consumption would be reduced by 3 percent. This represents about .33 percent of the national total.

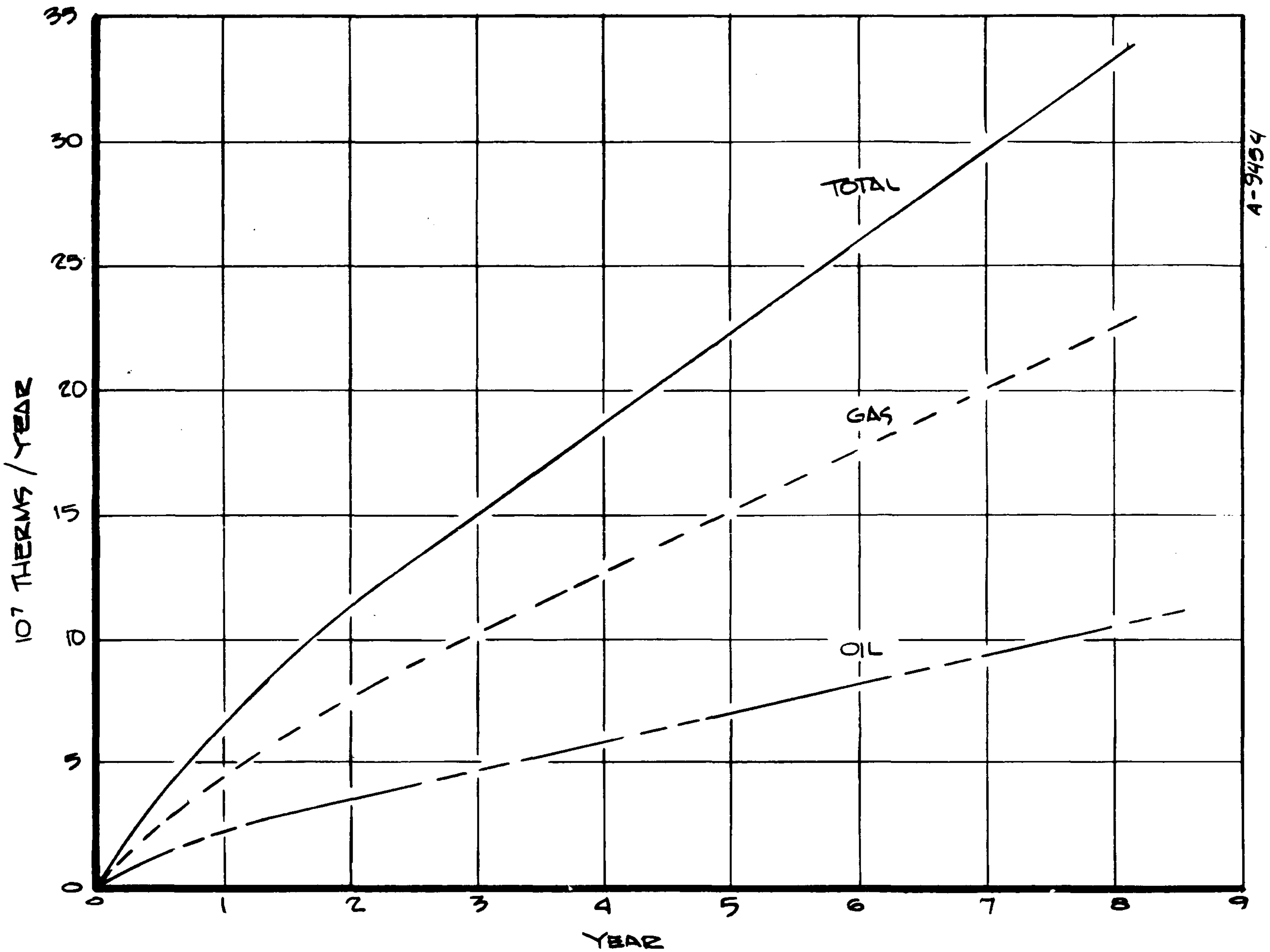


Figure 43. Residential Fuel Saving Per Year
Retrofits Units
New High Efficiency Furnaces

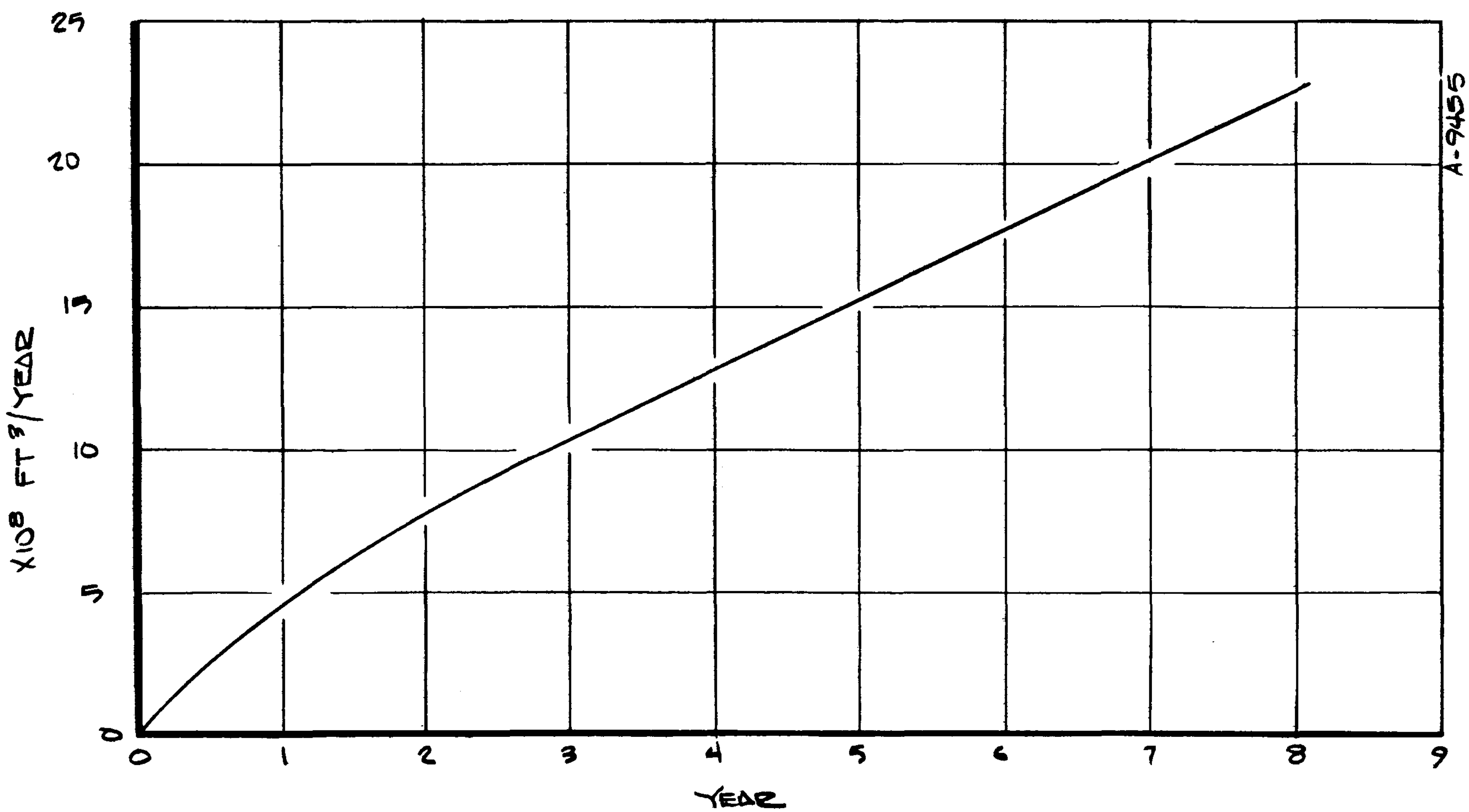


Figure 44. Savings Per Year
Retrofit Units
New High Efficiency Furnaces

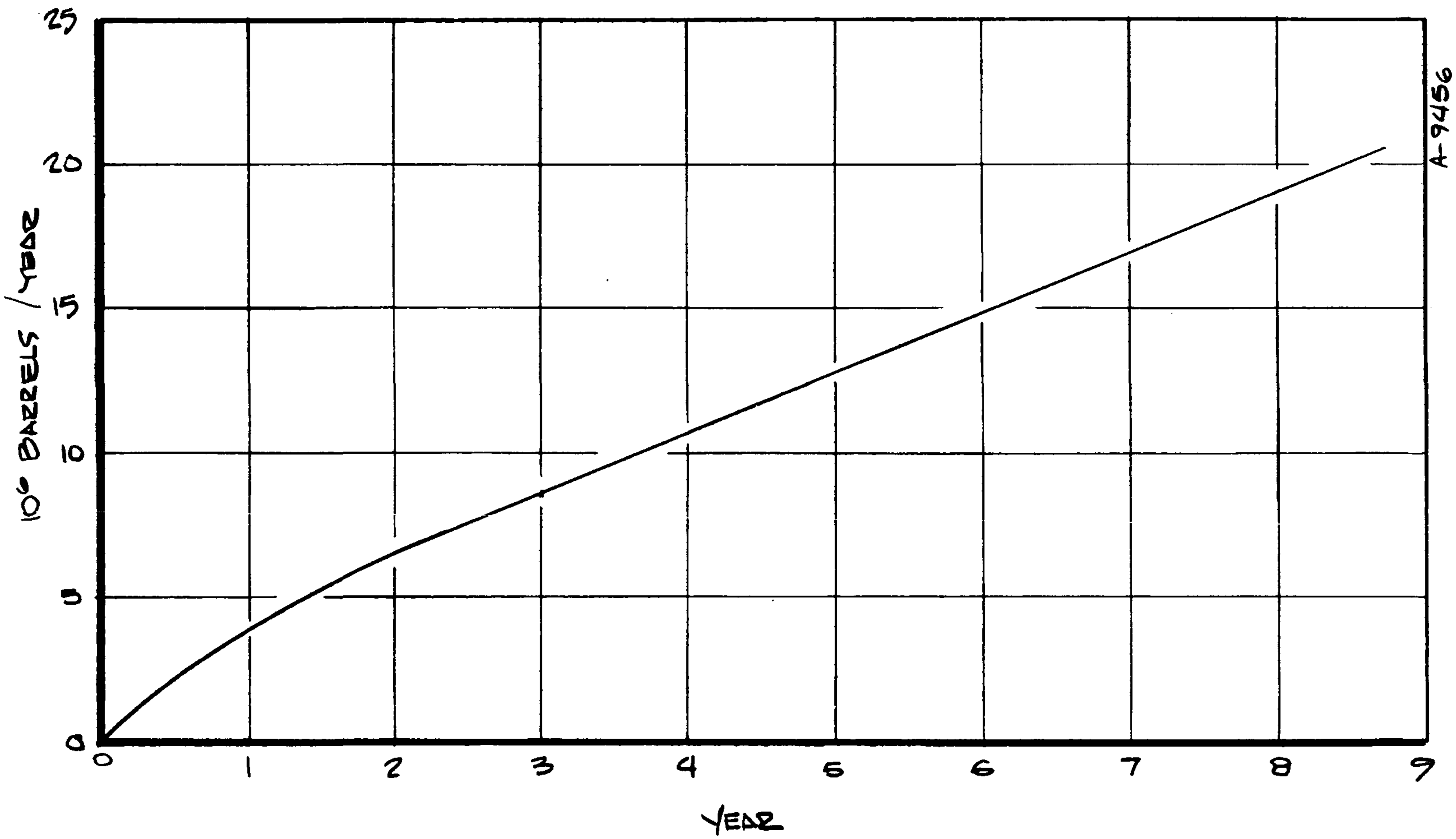


Figure 45. Oil Saving Per Year
Retrofit Unit
New High Efficiency Furnaces

TABLE 35

ENERGY SAVINGS

Years	n^{th} Year	Predicted Total Annual Energy Consumption U.S. (Reference 77)	Space Heating	Savings	Percent of Space Heating	Percent of Total
1974	0	78 x 10	8.6 x 10	0	0	0
1975	1	80 x 10	8.8	.07	0.8	.088
1976	2	83 x 10	9.1	.11	1.2	.132
1977	3	86 x 10	9.46	.15	1.58	.174
1978	4	89 x 10	9.8	.19	1.94	.213
1979	5	91 x 10	10.0	.22	2.20	.242
1980	6	94 x 10	10.3	.26	2.52	.277
1981	7	96.5 x 10	10.6	.30	2.83	.312
1982	8	99.0 x 10	10.9	.33	3.02	.332

SECTION 9

CONCLUSIONS AND RECOMMENDATIONS

9.1 CONCLUSIONS

This study has produced the following conclusions:

- Very little is known about the actual operating efficiency of residential heating units. This efficiency is influenced by hardware design and the on-off cycle (which in turn depends on many factors: ambient conditions, thermal load, thermostat setting), aerodynamic conditions, and by condition, adjustment, and maintenance factors. Nominal steady state efficiencies typically range from 70 to 80 percent, but actual operating efficiencies may range from 40 to 75 percent; lack of this kind of information hampers the evaluation of all proposed improvements.
- A heat recovery device to recover the sensible heat can be economically manufactured and sold if the flue temperatures are high enough (>500°F to avoid condensation). This concept is represented by the Isothermic and Dolin device.
- A heat recovery device to recover the sensible and latent heat will be difficult to accomplish within the current economic framework.
- Considerable energy (up to 30 percent of the heating load) may be conserved if the supply could be closely matched to the load by modulating and closer matching of the furnace to the structure of heating load.
- Considerable energy (up to 20 percent) could be saved by closing of the flue during burner off periods.
- The load could be greatly reduced by greater use of insulation and prevention of air infiltration.
- A high efficiency furnace for new installation could probably be designed to achieve an overall efficiency greater than 80 percent within the current economic structure.

9.2 RECOMMENDATIONS

The present study has been of limited scope and amounts to a preliminary survey activity. All conclusions noted above could benefit from more detail and a more through testing of alternative assumptions, where data were lacking or uncertain. With these concepts in mind, the following recommendations regarding the direction and emphasis of future work are presented.

9.2.1 General Recommendations

1. Obtain actual operating data by measurements on typical furnaces in typical installations. Define achieved efficiencies in detail and relate to operating conditions, particularly to on/off cycle times and modulation aspects. This information will provide the detailed data base for all future work.

2. Study British and European design practice and current research activities. Information from these high fuel cost areas will form a valuable supplement to this study.

3. Survey state of the art in furnace performance prediction techniques, especially those which can account for cyclic effects. Obtain and develop a suitable predictive code, and verify it against typical data from Recommendation 1.

9.2.2 Retrofit Device Recommendations

4. Continue to monitor commercial and legal progress of retrofit vent dampers and of noncondensing flue gas devices. Test typical devices to determine actual performance and define possible improvements. Coordinate with Recommendation 2 curve.

5. Refine the design exercise on condensing flue gas devices of Section 5 and 6 to further define the apparently marginal attractiveness of these systems.

9.3.3 New Design Recommendations

Probably the most fruitful work would be to explain in much further detail than has been possible here the possibility of a high efficiency modulated furnace concept. Although commercial research of a proprietary nature continues in this area, the apparent progress does not seem commensurate with the promise.

Recommended components in this activity include Recommendations 1 and 3 above, followed by tasks to:

6. Predict efficiency of proposed modulation schemes by analytical techniques; identify cycles to optimize operating parameters data. Modify cycle to see how to improve efficiency.
7. Analytically predict the effect on heat transfer due to modulating output.
8. Explore methods of modulation
   - Vary gas flow rate
   - Use multiple burners in various combinations
   - Heat capacitance/hot water systems
9. Develop heat load detector and furnace controller (see Section 6)
10. Modify an existing furnace for modulated mode
11. Design a complete new furnace using either the submerged combustion or catalytic combustion in conjunction with the modulated concept
12. Build a prototype unit and test
13. Relate the progress of this design and development effort to:
    - Commercial and research developments in alternative heat pump/ all electric systems (including solar-assist components)
    - Other integrated energy management system programs, including total-energy and MIVS
    - Household energy conservation programs which will be affecting thermal load data for new construction

9.3.4 Emissions

With regard to emission control devices three specific areas of study are recommended:

14. Determine the real contribution of residential emissions to environmental air quality
15. Further investigate post combustion control devices, if required
16. Define the potential of the modulated furnace concept (Recommendations 6-11) to reduce the high peak emission associated with start-up and shutdown periods.

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APPENDIX A

FLUE LOSS CHARTS

Figure A-1 presents a flue loss nomogram from the AGA Laboraties for butane, propane; coke oven and natural gases. A typical line is shown for natural gas at 60 percent excess air, 20 percent flue losses and 335°F temperature difference or about 405°F exit temperature.

Figure A-2 shows a similar nomograph from the Gas Engineers Handbook[33] for determining a gas appliance performance. Table A-1 lists the procedures for using this chart. A typical example might be to determine the overal heat transfer coefficient knowing the CO_2 level, temperature rise, and heat exchanger loading (the fourth problem in the table). The applicable appliance classes referred to in Table A-1 are as follows:

1. Forced air furnaces:

Parallel flow		updraft
Counterflow	with	downdraft
Crossflow		steel or cast iron

2. Gravity furnaces; floor furnaces; room, space, and recessed heaters
3. Water heaters and boilers
4. Deep fat fryers
5. Gas range ovens (for input - CO_2 relationships and flue losses)

Figure A-3 shows similar curves for heat loss in flue gas for fuel oils. To use this chart either enter with the excess air or CO_2 level and proceed to find the other using the excess air curve. From that intersection move vertically upward or downward on the excess air line to the flue temperature line. Then proceed horizontally to read the flue loss.

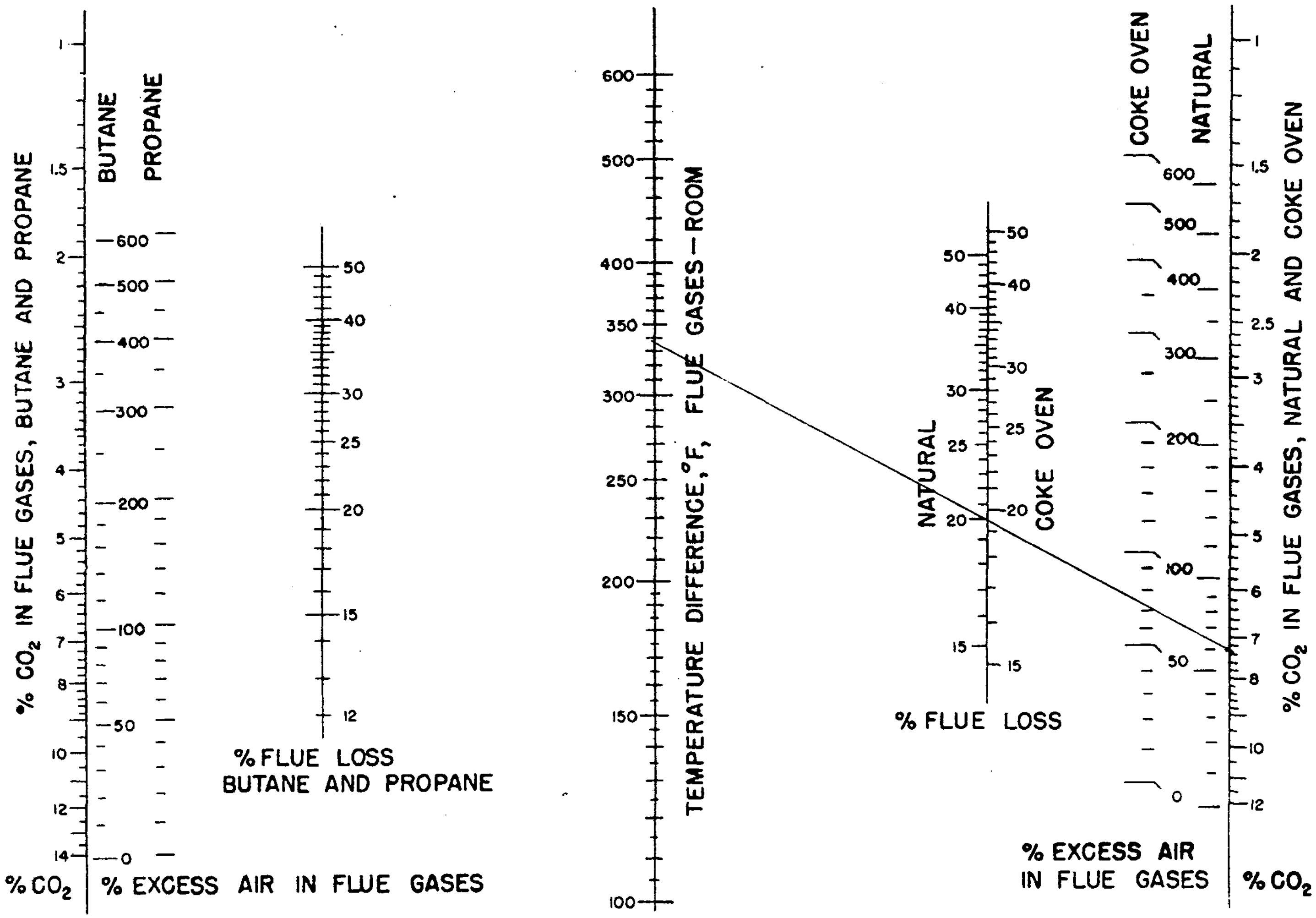


Figure A-1. Alignment Chart for Calculation of Flue Losses for Butane, Propane, and Coke Oven and Natural Gases. (Adapted from A.G.A. Laboratories' Flue Loss Charts.)

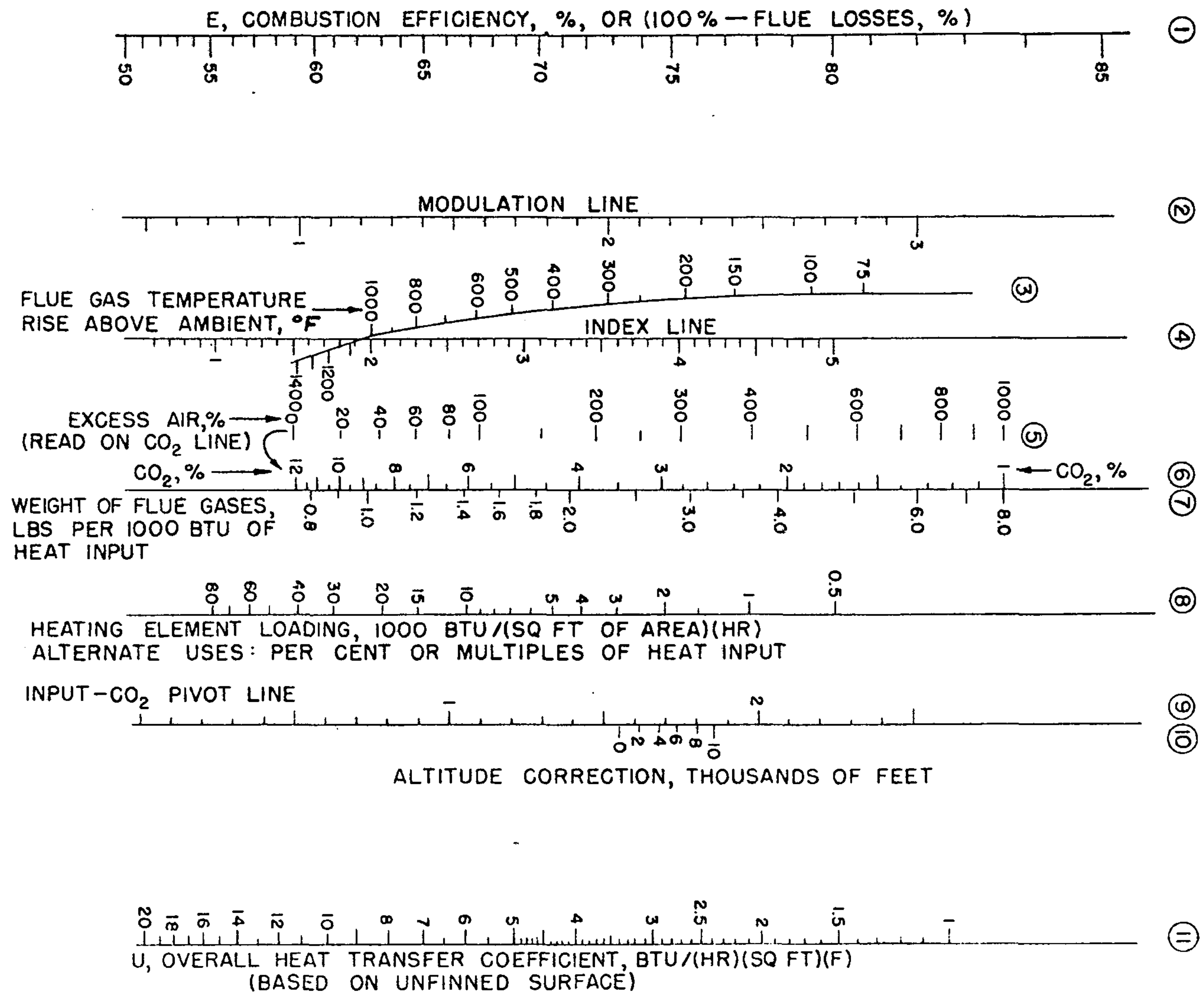


Figure A-2. Gas Appliance Performance Chart. (Data obtained in part from A.G.A. Laboratories Res. Bull. 63, 1951.)

TABLE A-1.

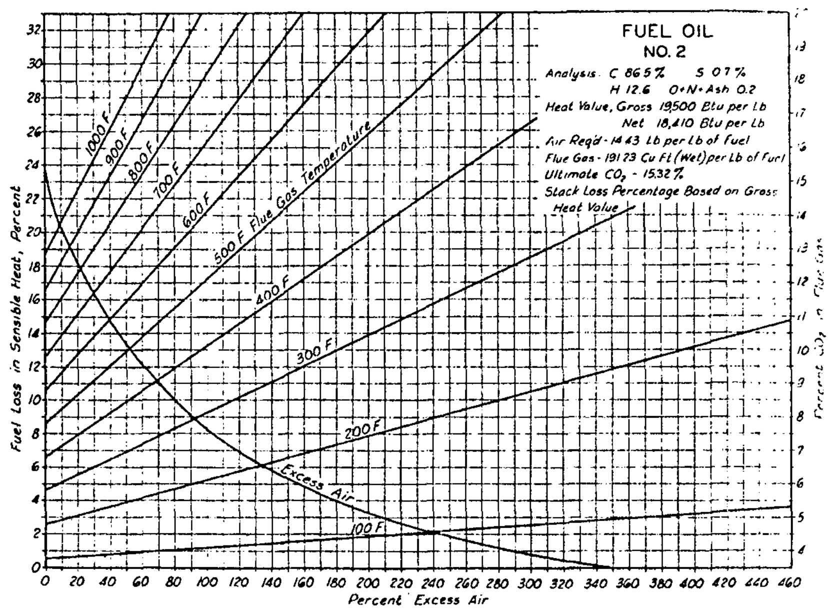
APPLICATIONS OF THE GAS APPLIANCE PERFORMANCE CHART (see Figure A-2)

Problems involving	Known factors	Find	Procedure	Additional steps or remarks	Answer	Applicable appliance classes*	Applicable fuel gases†
Efficiency, CO_2, flue gas temperature	Efficiency (1), CO_2 (6)	Flue gas temp rise (3)	(1) (6)→(3)			1, 2, 3, 4, 5	N, T, M
	Flue gas temp rise (3), CO_2 (6)	Combustion efficiency (1)	(3) (6)→(1)				
	Efficiency (1), flue gas temp rise (3)	CO_2 (6)	(1) (3)→(6)				
Efficiency, CO_2, overall heat transfer coefficient, area, heat input	CO_2 = 8% (6), flue gas temp rise = 300 F (3), loading = 3000 Btu/sq ft-hr (8)	*U*: overall heat transfer coefficient (11)	(6) (3)→(4) (3.27) ‖ (4) (8)→(11)	Variations of this procedure permit solving for flue temperature, CO_2, etc.	U = 2.75 Btu/hr-sq ft-°F	1, 2, 3, 4	N, T, M
	Estimated U = 6.0 (11), input = 72,000 Btu/hr, area = 4 sq ft, loading = 72,000/4 = 18,000 (8), operating CO_2 = 7% (6)	*E*: predicted combustion efficiency (1)	(11) (8)→(4) (1.82) ‖ (4) (6)→(1)		E = 52.5%		
Input and CO_2 (no changes in appliance geometry)	CO_2 = 8.5% (6), input = 80,000 Btu/hr, i.e., 8.0 on (8) scale	CO_2 (6) at: (a) 100,000 input, (b) 50,000 input	(6) (8)→(9) (1.47) ‖ (9) $(8)_a$→$(6)_a$ ‖ (9) $(8)_b$→$(6)_b$	Applies from 40 to 125% of normal input to all appliances with enclosed flames	CO_2: (a) 11%, (b) 5.4%	1, 2, 3, 4, 5	N, T, M C, B, P
Efficiency variations for excess air changes at constant heat input	Input = 60,000 Btu/hr (8), CO_2 = 9% (6), E = 70% (1)	Efficiency at 6% CO_2 and 60,000 Btu/hr input	(1) (6)→(4) (2.34) ‖ (4) (6)′→(1)′	Total air may be changed by flue outlet or combustion air inlet baffles, or by different draft hood	E = 61%	1, 2, 3, 4	N, T, M
Effect of altitude changes on heat input and CO_2 in flue products	Input = 50,000 Btu/hr, i.e., 5.0 on (8) scale, at sea level (10), CO_2 value not reqd.	Input at 6000 ft (10)′ to obtain same CO_2 as that at sea level (without altering appliance)	(10) (8)→(6) (5.8) ‖ (6) (10)′→(8)′	Altitude changes follow rule of 4% reduction in input or in weight flow/1000 ft alt. increase‡	Input = 39,000 Btu/hr	1, 2, 3, 4, 5	N, T, M C, B, P
	CO_2 = 7.5% (6) at sea level (10)	CO_2 at same input and 4000 ft alt. (10) (without altering appliance)	(10) (6)→(8) (6.5) ‖ (8) (10)′→(6)′	Value on (8) acts as pivot only	CO_2 = 9.0%		
	CO_2 = 7.5% (6), input = 50,000 Btu/hr, i.e., 5.0 on (6)′ scale, at sea level (10)	CO_2 at 30,000 Btu/hr input, i.e., 3.0 on (8)″ scale, and 5000 ft alt. (10)′	(a) CO_2 (6)′ at 5000 ft and 50,000 input: (10) (6)→(8) (6.5) ‖ (8) (10)′→(6)′ (9.4)	(b) CO_2 (6)″ at 30,000 input: (6)′ (8)′→(9) (1.9) ‖ (9) (8)″→(6)″	CO_2 = 5.5%		
Weight of flue products	Input = 150,000 Btu/hr, CO_2 = 7.9% (6)	Flue gas rate	(6)→(7) (1.12 lb/1000 Btu)		(1.12 × 115,000)/1000 = 129 lb/hr	1, 2, 3, 4, 5	N
Effect of modulated heat input on appliance efficiency (from 50 to 125% of normal heat input)	Input = 170,000 Btu/hr, i.e., 17 on (8) scale, CO_2 = 7% (6), efficiency = 80% (1)	Efficiency (1)′ at 120,000 Btu/hr, i.e., 12 on (8)′ scale (without altering appliance)	(6) (8)→(9) (0.78) ‖ (9) (8)′→(6)′ CO_2 at 120,000 input = 4.8%	(1) (6)→(2) (2.2) ‖ (2) (6)′→(1)	E = 78.3%	1, 2, 3, 4	N, T, M
		Efficiency (1)″ at 220,000 Btu/hr, i.e., 22 on (8)″ scale	Using points (9) and (2) as before: CO_2 at 220,000 input = 9.2% and (2) (6)″→(1)″	For precise work, the modulation point should be found by operating at two different heat inputs	E = 81.3%		
Temperature of flue gases after dilution at draft hood	Flue gas temp rise = 450°F (3) CO_2 = 8% (6) (before dilution)	Temperature at 3% CO_2 (6)′ after dilution with room air	(6) (3)→(1) (77.5) ‖ (1) (6)′→(3)′	Assume no heat loss at draft hood; thus, E is constant	Temp = 190°F above room	1, 2, 3, 4	N, T, M C, B, P
	Flue gas temp = 800 F at 8.0% CO_2 (6) in recessed heater. Dilution air is heated to 300 F, CO_2 after dilution is 2.5% (6)′	Gas temperature downstream of draft hood	Temp rise entering hood = 800 − 300 or 500°F (3) above dilution air temp (6) (3)→(1) (76.0) ‖ (1) (6)′→(3)′	Dilution air temperature is treated as ambient air temperature for this calculation	Temp = 175°F above 300, or 475 F		

* See text.

† *Code for fuel gases:* N = *natural;* T = *manufactured, 525 Btu, 0.42 sp gr;* M = *mixture, natural and manufactured;* C = *coke oven;* B = butane and butane-air; P = propane and propane-air.

‡ This rule as subsequently revised states that appliances do not have to be derated at elevations up to 2000 ft; for elevations above 2000 tf, ratings should be reduced at the rate of 4 per cent for each 1000 ft above sea level.



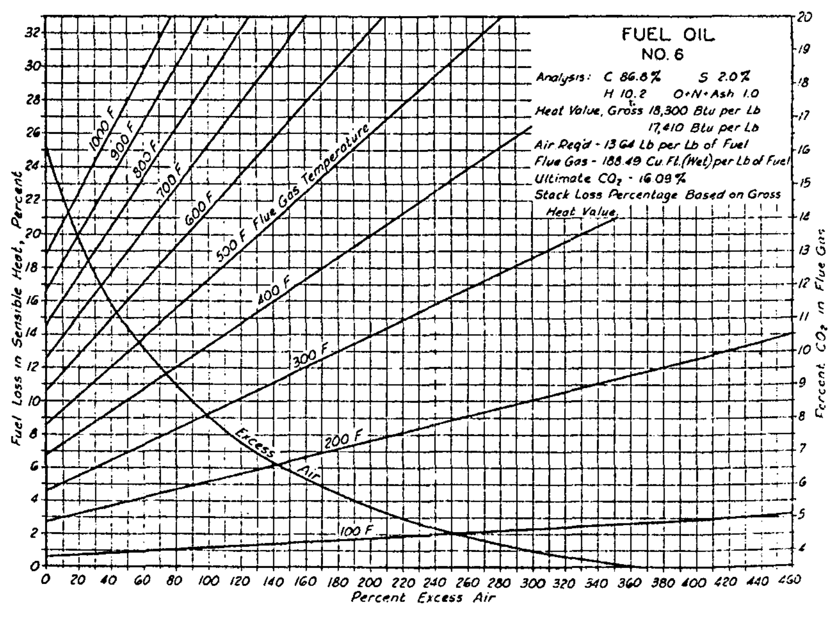


Figure A-3. Heat Loss in Flue Gas-Fuel Oils From Strock & Koral (36) Chapter 6, Fuels & Combustion

DOLIN HEAT TRANSFER ANALYSIS

Specifications:

Tubes: 52 - 3/8 OD x 20 ga. Bare Steel Tubes

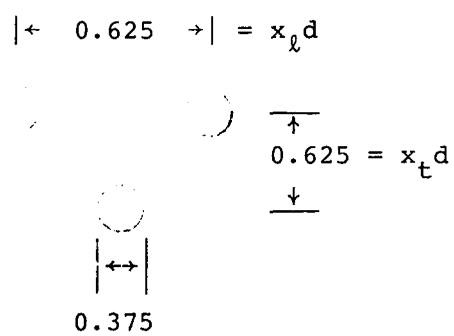
Spacing: 5/8 x 5/8 Triangular Pitch
5 Tube High x 11 Tubes wide

Overall Length: 10-3/8" long

Width: Tubes + 2" each side for bypass

Blower: Grainger 2C610:Nominal 140 CFM

Analysis:



$$x_\ell = 0.625/0.375 = 1.667$$

$$x_t = 0.625/0.375 = 1.667$$

<u>Outside Heat Transfer Coefficient</u>

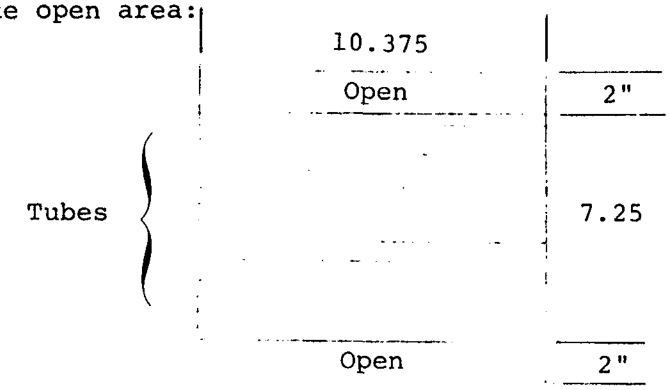
From Kays and London (Reference 53) p. 127

$$N_{ST}\, N_{Pr}^{2/3} = 0.38 N_R^{-0.4} \qquad f = 0.40 N_R^{-0.18}$$

Assume the flue gases are from oil fired furnace using 60 percent excess air. Then,

$$\dot{M}_f = 2.11 \text{ lbs/min}$$

Calculate open area:



$$\text{Side Area} \simeq 40 \text{ in}^2$$

$$\sigma = 0.4 \ \frac{\text{Free Flow Area}}{\text{Frontal Area}} \quad \text{Heat Exchanger}$$

$$\text{Free Flow Area} = (10.375)(7.250)(0.4) - 30.09$$

Now assume the flow is proportional to the open area:

$$\text{Total Open Area} = 30.09 + 40 = 70.09$$

Then,

$$\dot{M}_e = \frac{30.09}{70.09} \, 2.11 = 0.906 \text{ lbs/min}$$

$$G = \frac{0.906}{30.09} \, 144 = 4.33 \text{ lbs/min-ft}^2$$

$$\text{Heat Transfer Area} = 635.58 \text{ in}^2 = 4.414 \text{ ft}^2$$

$$D_u = 0.0547 \text{ ft}$$

$$N_R = 245$$

$$N_{ST} \, N_{Pr}^{2/3} = 0.38 \, (245)^{-0.4} \approx 0.0421 = \frac{h_o}{GCp} \, (0.7)^{2/3}$$

$$Cp \simeq 0.25$$

$$h_o = 3.468 \text{ Btu/hr-ft}^2 \, {}^\circ\text{F}$$

Inside Heat Transfer Coefficient

Figure A-4 shows the fan curve and the system curve for the air flow through the tubes. This reveals that the actual flow is about 61 CFM. Using this number the following calculations were made to determine the inside heat transfer coefficient:

$$\dot{M} = (61)(0.075) = 4.575 \text{ lbs/min}$$

$$A_c = \frac{\pi(0.305)^2}{4}\,\frac{(52)}{144} = 0.264 \text{ ft}^2$$

$$G = \frac{4.575}{0.0264} = 173.3 \text{ lbs/min-ft}$$

$$N_R = \frac{(0.305)(173.3)}{(12)(1.2 \times 10^{-5})60} = 6117.5$$

$$N_u = 0.022(6117.5)^{0.8}(0.8073) = 18.998 = \frac{hd}{k}$$

$$h = \frac{(18.998)(0.015)12}{0.305} = 11.21$$

Overall Heat Transfer Coefficient

Then the overall heat transfer coefficient:

$$\frac{1}{u} = \frac{1}{3.468} + \frac{1}{11.21\left(\frac{0.305}{0.375}\right)}$$

$$U \approx 2.5 \text{ Btu/hr-ft}^2\,{}^\circ\text{F}$$

Then to determine the overall efficiency using the Kays and London (53) technique:

$$NTU = \frac{AU}{C_{min}}$$

$$C_n = (0.9058)(0.25) = 0.226 = C_{min}$$

$$C_c = (4.575)(0.24) = 1.098$$

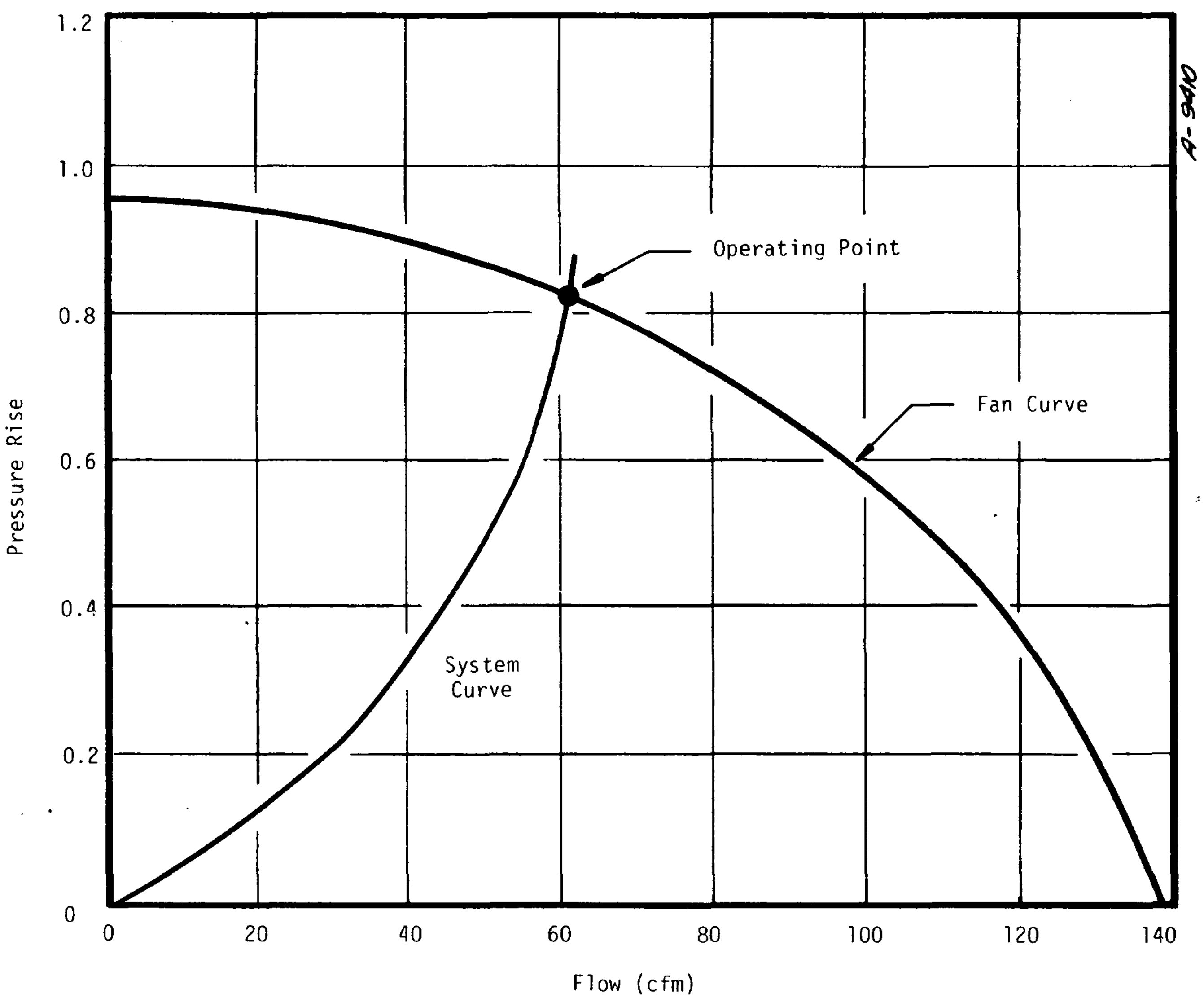


Figure A-4. Blower Performance Curve
(Crainger 2C610 Blower)

$$C_{min}/C_{max} = 0.206$$

$$NTU = \frac{(2.5)(4.414)}{(0.226)(60)}$$

$$NTU = 0.8138$$

$$\varepsilon \approx 1 - e^{-NTU} = 1 - e^{-0.8128}$$

$$\boxed{\varepsilon = 0.5568}$$

Table A-2 summarizes the flue temperature heat absorbed and percent savings of the total furnace input. The energy consumed by the blower has been subtracted from the total. Figure A-5 shows this data in graphical form.

TABLE A-2

DOLIN HEAT RECLAIMER PERFORMANCE

Let's assume $T_{c_{in}}$ = 60°F

Case	T_{h_1}	T_{h_2}	$T_{h_{2c}}$	T_{c_o}	q(Btu/hr)	q_M	q_{net}	% Input
1	500	255	395	110	3322	-273	3049	3.05
2	600	299	470	122	4077	-273	3804	3.80
3	700	344	547	133	4827	-273	4554	4.55
4	800	388	623	145	5586	-273	5313	5.31
						Due to Blower Motor		

Definitions

$T_{c_{in}}$ = Cooling air temp in (°F)

T_{c_o} = Cooling air temp out (°F)

T_{h_1} = Flue gas temp in (°F)

T_{h_2} = Flue gas temp out which goes through tube bundle (°F)

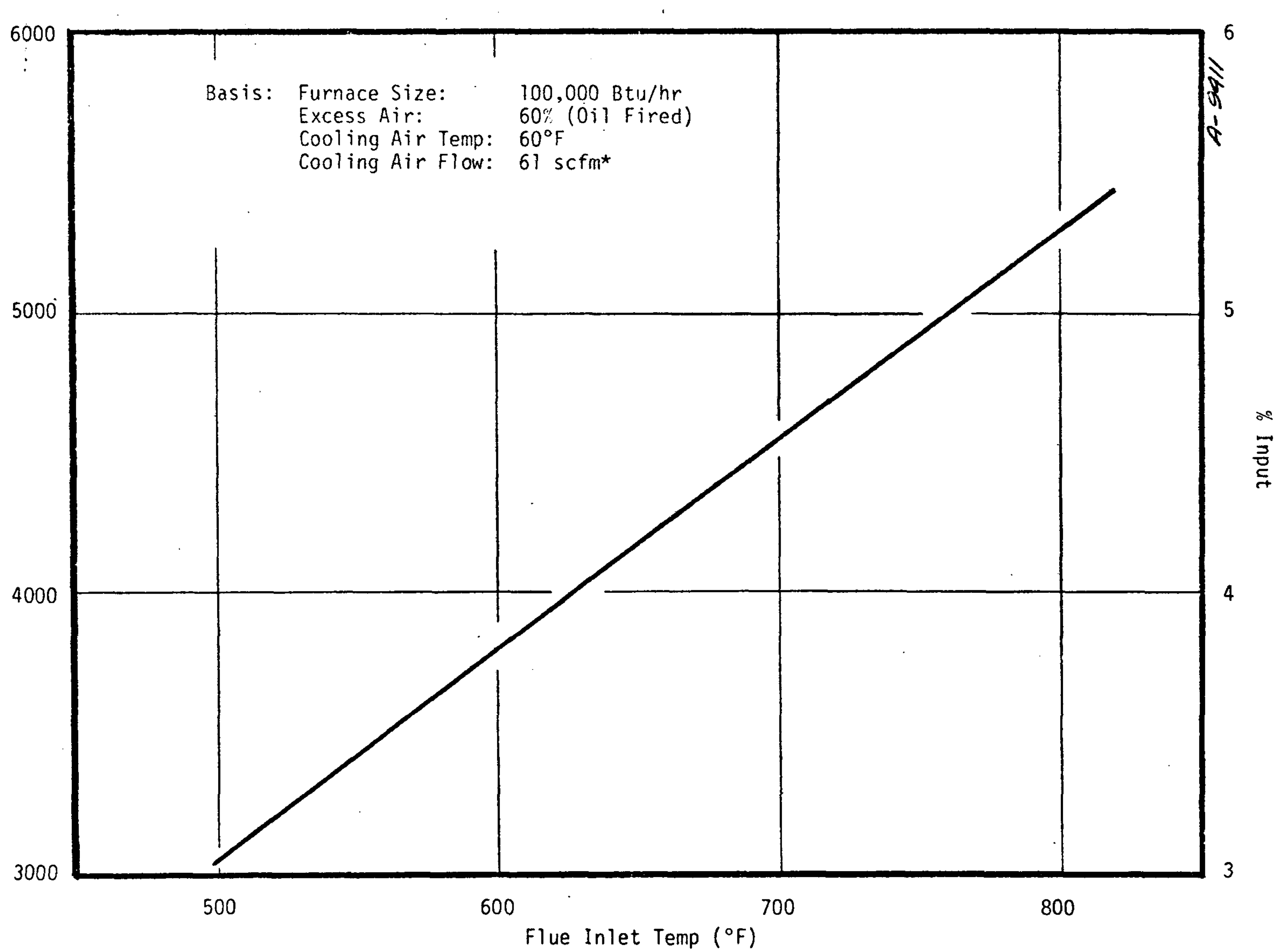
$T_{h_{2c}}$ = Combined flue gas temperature out

q = Heat recovered by device

q_M = Energy used to drive air blower

q_{net} = Net heat saved $q_{net} = q - q_M$

% Input = q_{net}/100,000 Btu/hr



*See attached curve of resulting operating point of Blower

Figure A-5. Dolin Heat Reclaimer Performance

HEAT TRANSFER ANALYSIS

The performance of noncondensible heat exchangers was determined using the technique of London and Kays (49). A simple computer program was written in order to facilitate rapid screening of potential designs. A listing of this program together with an input format is included in this appendix. The program is applicable to plate-fin and fin tube designs in single pass or multipass crossflow arrangements. In addition C_{min}/C_{max} as defined below is assumed to be much less than 0.25.

Condensible heat exchangers were analyzed using a combination of hand calculation based on a technique by McAdams (50) and the above computer program to determine the performance of the noncondensible portion as well as the pressure drop.

<u>Noncondensible - NTU Heat Exchanger Analysis</u>

The efficiency of a crossflow heat exchanger with $C_{min}/C_{max} << 0.25$ is approximately equal to the following expression:

$$\varepsilon \approx 1 - e^{-NTU}$$

where

$$NTU = \frac{AU}{C_{min}}$$

$C_{min} = W\ C_p$ for either the hot or cold side.

U = overall conductance fo heat transfer

$$= \frac{1}{\eta_{o,h}\ h_n} + \frac{a}{A_w/A_h k} + \frac{1}{A_c/A_h\ \eta_{o,c}\ h_c}$$

A = surface area on which U is based (ft^2)

η_o = fin efficiency

The effectiveness is related to the temperatures by the following expression:

$$\varepsilon = \frac{q}{q_{max}} = \frac{C_h\left(t_{h,in} - t_{h,out}\right)}{C_{min}\left(t_{h,in} - t_{c,in}\right)} = \frac{C_c\left(t_{c,out} - t_{c,in}\right)}{C_{min}\left(t_{h,in} - t_{c,in}\right)}$$

The reader is referred to Kays and London (49) for specific definition of the above nomenclature. The computer program calculates the overall conductance from given performance data for a specific surface (correlations of St $Pr^{2/3}$ vs. Reynolds number), the flows, temperatures, and overall configuration. In addition the pressure drop is calculated from the following expression:

$$\Delta P = \frac{G^2}{2g_c} \nu_1 \left[\underbrace{(K_c + 1 - \sigma^2)}_{\substack{\text{Entrance}\\ \text{Effect}}} + \underbrace{2\left(\frac{\nu_2}{\nu_1} - 1\right)}_{\substack{\text{Flow}\\ \text{Acceleration}}} + \underbrace{f\,\frac{A}{A_c}\,\frac{\nu_m}{\nu_1}}_{\substack{\text{Core}\\ \text{Friction}}} - \underbrace{(1 - \sigma^2 - K_e)\,\frac{\nu_2}{\nu_1}}_{\substack{\text{Exit}\\ \text{Effect}}} \right]$$

The friction factor versus Reynolds Number data for the surface must be provided to the program. Table A-3 lists the required input information and Figure A-6 shows a typical output listing. Figure A-7 is a listing of the program.

<u>Condensible Heat Exchanger Performance</u>

The condensing heat exchangers were analyzed using the Log-Mean-Temperature Difference (LMTD) approach. The principal expression relating heat transfer to the LMTD is as follows:

$$q = U\,A\,F_1\ (\text{LMTD})$$

where

F_1 = correction factor for crossflow and/or multipass arrangement (see Figure H4.2 of Fraas and Ozisik (78))

U = overall conductance for heat transfer (provides for a condensing coefficient)

A = surface area on which U is based

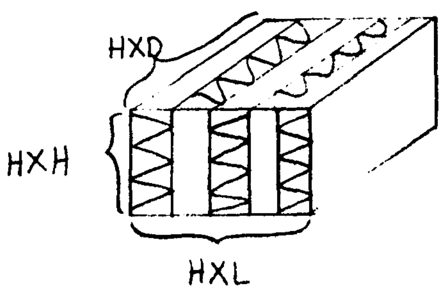
$$\text{LMTD} = \frac{\Delta t_o - \Delta t_L}{\ln \Delta t_o / \Delta t_L}$$

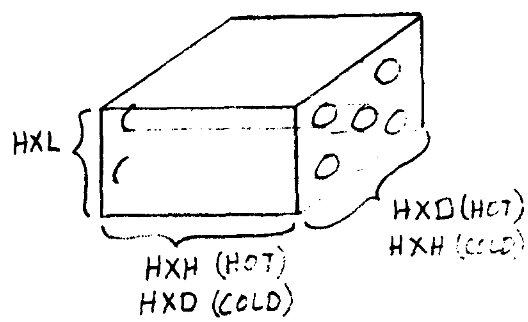
TABLE A-3

INPUT DATA FOR PERF

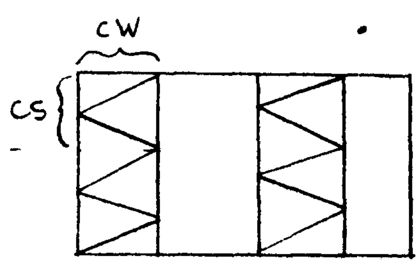
NAMELIST/HX

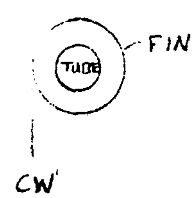
HXH	HX length perpendicular to flow	inches
HXD	HX length parallel to flow	inches
HXL	HX length direction of cells	





CS	Cell size or tube diam. or tube + Fin Diam.	inches
CW	Cell width or fin length on finned tube	inches





NFC	Number of fins per cell	
FT	Fin thickness	inches
NS	Number of segments per side #tubes	
XHR	Hydralic radius	feet
AFAT	Fin area/total area	
XPTC	Plate thickness	inches
ACTOVC	Total heat transfer area/volume between plates - cold side N.A. to tubes	Ft^2/Ft^3

TABLE A-3 (continued)

AHTOVH	Total heat transfer are/volume between plates - hot side	Ft^2/Ft^3
XKM	Thermal conductivity of HX material	Btu/hr-ft-°F
IDIAG	Flag for diagnostic output 1 = Diagnostic output 0 = No diagnostic output	
NTUBE	Total number of tube in bank and # tubes in first bank facing gas	
SIG	Free flow area/frontal area	
COND	Thermal conductivity of air	
NAMELIST/FLOW		
XMDOT	Mass flow rate	Lb/min
TIH	Inlet temp of hot side	°F
TIC	Inlet temp of cold side	°F
NAMELIST/HXTYPE		
IFLAG	0 - Plain plate fin surface 1 - tube bank	
DIAMTI	Tube diameter (inside) - only used if IFLAG = 1	inches
DIAMTO	Tube diameter - (outside)	inches
DTH	Distance between tube center - bank side	inches
DTC	Distance between tube centers - tube side	inches
NAMELIST/CHECK		
RHO	Density - value for air and gas	Lb/Ft^3
XMLL	Viscosity	Lb/Ft-sec
CP	Heat capacity	Btu/lb-°F
XPR	Prandtl Number	
NAMELIST/HC		
IA	1 - air side HX characteristic table	
IB	2 - gas side HX table (This enables user to determine effect of changing fluid flowing thru each side of HX.)	

TABLE A-3 (concluded)

NAMELIST/REN	
REYN	Reynolds Number - ascending order
NPT	Number of Reynolds Number pts
NAMELIST/HT2	
HTRC1	j factor - HX side 1 - match to corresponding Reynolds Number ($j = (h/GC_p) N_{Pr}{}^{2/3}$)
NAMELIST/ERIC1	
FRF1	f: friction factor -HX side 2
NAMELIST/HT2 NAMELIST/ERIC3	Same as above for HX side 2
NAMELIST/CONFIG	
ICONFG	Number of configurations to be run

Notes: Tube Bank Crossflow HX

1. Tube bank size input must be the open area of the first pass only.
2. For each configuration each NAMELIST having changed values must be input again

NAMELIST 1	
NAMELIST 2	Config. 1
NAMELIST 3	
NAMELIST 2	Config. 2 - uses namelist values from Config. 1 for NAMELIST 1 and 3 and new values input in NAMELIST 2.

   a. The read statement for each NAMELIST with changed values must be inside the 5000 DO loop - see program listing line 35, 3b.

NAMELIST 1	
NAMELIST 2	
NAMELIST 3	
NAMELIST 4	
Do 5000	
NAMELIST 5	These will be reread for configuration 2 through
NAMELIST 6	N with new values.

3. The NAMELIST input must be in the same order as the read statements - program lines 25 through 36.

HEAT EXCHANGER PERFORMANCE CODE

CONFIGURATION 5	COLD SIDE	HOT SIDE
HEAT EXCHANGER LENGTH - PARALLEL TO FLOW - INCHES	1.173	15.000
HEAT EXCHANGER LENGTH - PERPENDICULAR TO FLOW - INCHES	15.000	1.173
HEAT EXCHANGER LENGTH - DIRECTION OF CELLS - INCHES	7.137	7.137
SINGLE CELL HEIGHT - INCHES	.195	.168
SINGLE CELL WIDTH - INCHES	.544	.249
FLOW RATE - LB/MIN	75.000	2.070
NUMBER HX SEGMENTS	9	9

	REYNOLDS NUMBER	HEAT TR. COEF (BTU/FT**2 HR F)	HEAT TRANSFER AREA (FT**2)	PRESSURE DROP (IN H2O)	INLET TEMP (F)	OUTLET TEMP (F)
COLD SIDE	2646.8531	11.6940	14.4537	.1226	140.0000	146.5342
HOT SIDE	1234.5896	10.4731	8.9677	1.7211	400.0000	181.4632

U (BTU/FT**2 HR F)	NTU	EFFICIENCY	QDOT (BTU/HR)
6.6109	1.8359	.8405	7056.9897

TERMS 1 AND 3 OF 1/U EQUATION

TERM 1	TERM 3
.09640	.05487

EFFICIENCY COEFFICIENT

149.4011

Figure A-6. Typical Output Listing

```
@ FOR PERF,PERF                                                        28 AUG 74   00:00:00.321
DATE, TIME, LEVEL OF OUTPUT ELEMENT: 28 AUG 74  14:00(00)
FORTRAN V:  ISO VERSION 2.9

   MAIN PROGRAM

   STORAGE USED (BLOCK, NAME, LENGTH)

         0001    *CODE    001267
         0000    *DATA    001332
         0002    *BLANK   000000


   EXTERNAL REFERENCES (BLOCK, NAME)

         0003    TBLP
         0004    NRNLS
         0005    NEXP6$
         0006    TANH
         0007    EXP
         0010    NWDU$
         0011    NIO2$
         0012    NSTOP$


   STORAGE ASSIGNMENT FOR VARIABLES (BLOCK, TYPE, RELATIVE LOCATION, NAME)

    0001     000332 10L        0000     000661 1000F      0000     000657 1200F      0000     000663 1400F      0000     000760 1450F
    0000     001023 1460F      0001     000366 15L        0001     001175 150L       0000     001072 1510F      0001     001212 160L
    0000     001177 1600F      0000     001226 1610F      0000     001247 1620F      0001     000120 165G       0001     000130 173G
    0001     001137 2000L      0001     000602 304G       0001     000620 310L       0001     000625 320L       0001     000631 330L
    0001     000677 337G       0001     001066 417G       0001     000404 50L        0001     000422 55L        0001     000446 58L
    0001     000450 59L        0001     000522 60L        0001     000167 600L       0001     000220 610L       0001     000210 620L
    0001     000543 65L        0001     000763 80L        0001     000767 85L        0001     000272 9L         0000 R   000355 A
    0000 R   000415 AC         0000 R   000417 ACTOAH     0000 R   000370 ACTOVC     0000 R   000061 AFAT       0000 R   000414 AH
    0000 R   000371 AHTOVH     0000 R   000115 AREA       0000 R   000071 AREAA      0000 R   000420 AVGAW      0000 R   000357 B
    0000 R   000067 C          0000 R   000025 CA         0000 R   000426 CAIR       0000 R   000427 CGAS       0000 R   000431 CMAX
    0000 R   000430 CMIN       0000 R   000375 COND       0000 R   000047 CP         0000 R   000011 CS         0000 R   000017 CW
    0000 R   000363 D          0000 R   000433 DCOEF      0000 R   000111 DELTAP     0000 R   000401 DIAMTI     0000 R   000402 DIAMTC
    0000 R   000105 DP         0000 R   000404 DTC        0000 R   000403 DTH        0000 R   000365 E          0000 R   000432 EFF
    0000 R   000055 ETAF       0000 R   000113 ETAO       0000 R   000361 F          0000 R   000015 FA         0000 R   000107 FF
    0000 R   000255 FRF1       0000 R   000313 FRF2       0000 R   000023 FT         0000 R   000045 HTC        0000 R   000043 HTN
    0000 R   000161 HTRC1      0000 R   000217 HTRC2      0000 R   000005 HXD        0000 R   000003 HXH        0000 R   000007 HXL
    0000 I   000411 I          0000 I   000405 IA         0000 I   000406 IB         0000 I   000367 ICONFG     0000 I   000373 IDIAG
    0000 I   000400 IFLAG      0000 I   000437 MN         0000 R   000000 MSPV       0000 I   000425 N          0000 I   000013 NC
    0000 I   000021 NFC        0000 I   000410 NHXCON     0000 I   000407 NPT        0000 I   000027 NS         0000 I   000412 NTP
    0000 R   000002 NTU        0000 I   000351 NTUBE      0000 I   000413 NUSLT      0000 R   000435 QDOT       0000 R   000123 REYN
    0000 R   000073 RHO        0000 R   000374 SIG        0000 R   000103 SIGMA      0000 R   000077 SPECVL     0000 R   000353 TA
    0000 R   000421 TERM1      0000 R   000422 TERM2      0000 R   000117 TI         0000 R   000377 TIC        0000 R   000376 TIH
    0000 R   000121 TO         0000 R   000436 TOC        0000 R   000434 TOH        0000 R   000424 U          0000 R   000423 UINV
    0000 R   000031 XG         0000 R   000037 XHR        0000 R   000372 XKM        0000 R   000057 XL         0000 R   000053 XM
    0000 R   000033 XMDOT      0000 R   000041 XMU        0000 R   000416 XPASS      0000 R   000051 XPR        0000 R   000063 XPT
    0000 R   000065 XPTC       0000 R   000035 XRE
```

Figure A-7. Program PERF Listing

```
00101        REAL MSPV,NTU
00103        DIMENSION HXH(2),HXD(2),HXL(2),CS(2),NC(2),FA(2),CW(2),NFC(2),
00103       1FT(2),CA(2),NS(2),XG(2),XMDOT(2),XRE(2),XHR(2),XMU(2),HTN(2),
00103       2HTC(2),CP(2),XPR(2),XM(2),ETAF(2),XL(2),AFAT(2),XPT(2),XPTC(2),
00103       3C(2),AREAA(2),RHO(2,2),SPECVL(2,2),SIGMA(2),MSPV(2),DP(2),FF(2),
00103       4DELTAP(2),ETAO(2),AREA(2),TI(2),TO(2)
00104        DIMENSION REYN(30),HTRC1(30),HTRC2(30),FRF1(30),FRF2(30)
00105        DIMENSION NTUBE(2),TA(2)
00106        DIMENSION A(2),B(2),F(2),D(2),E(2)
00107        NAMELIST/CONFIG/ICONFG
00110        NAMELIST/HX/CS,CW,NFC,FT,XHR,AFAT,XPTC,ACTOVC,AHTOVH,XKM,TDIAG,
00110       *                SIG,COND
00111        NAMELIST/VAR/HXH,HXD,HXL,NS,NTUBE
00112        NAMELIST/FLOW/XMDOT,TIH,TIC
00113        NAMELIST/HXTYPE/IFLAG,DIAMTI,DIAMTO,DTH,DTC
00114        NAMELIST/CHECK/RHO,XMU,CP,XPR
00115        NAMELIST/HC/IA,IB
00116        NAMELIST/REN/REYN,NPT
00117        NAMELIST/HT1/HTRC1
00120        NAMELIST/FRIC1/FRF1
00121        NAMELIST/HT2/HTRC2
00122        NAMELIST/FRIC2/FRF2
00122  C
00122  C     READ INPUT
00122  C
00123        READ(5,HX)
00126        READ(5,CONFIG)
00131        READ(5,HXTYPE)
00134        READ(5,CHECK)
00137        READ(5,HC)
00142        READ(5,REN)
00145        READ(5,HT1)
00150        READ(5,FRIC1)
00153        READ(5,HT2)
00156        READ(5,FRIC2)
00161        READ(5,FLOW)
00164        DO 5000 NHXCON=1,ICONFG
00167        READ(5,VAR)
00167  C
00167  C     DETERMINE MASS FLOW INTO HEAT EXCHANGER
00167  C
00172        DO 100 I=1,2
00175        IF(IFLAG.EQ.1) GO TO 600
00177        NC(I)=HXH(I)/CS(I)
00200        FA(I)=CW(I)*CS(I)-NFC(I)*CW(I)*FT(I)
00201        CA(I)=FA(I)*NC(I)*NS(I)
00202        GO TO 610
00203    600 IF(I.EQ.2) GO TO 620
00205        NC(I)=NTUBE(I)
00206        NS(I)=NC(I)
00207        FA(I)=HXL(I)*HXH(I)
00210        TA(I)=(3.1415*(DIAMTI**2))/4.
00211        CA(I)=NC(I)*TA(I)
00212        GO TO 610
00213    620 NC(I)=NTUBE(I)
00214        NS(I)=NC(I)
00215        FA(I)=HXL(I)*HXH(I)
00216        CA(I)=FA(I)*SIG
00217    610 XG(I)=(XMDOT(I)*144.)/CA(I)
00220        XRE(I)=(XHR(I)*XG(I))/(XMU(I)*60.)
```

Figure A-7. Continued

```
00220      C
00220      C      TABLE LOOK-UP
00220      C
00221             IF(I.EQ.2) GO TO 10
00223             IF(IFLAG.EQ.1)GO TO 9
00225             CALL TBLP(XRE(IA),REYN,HTN(IA),HTRC1,NTP,0)
00226             CALL TBLP(XRE(IA),REYN,FF(IA),FRF1,NTP,1)
00227             GO TO 10
00230           9 IF(XRE(1).LT.2000.) FF(1)=64./XRE(1)
00232             IF(XRE(1).GT.2000.) FF(1)=.184/(XRE(1)**.2)
00234             NUSLT=.022*(XRE(1)**.8)*(XPR(1)**.667)
00235          10 IF(I.EQ.1) GO TO 15
00237             CALL TBLP(XRE(IB),REYN,HTN(IB),HTRC2,NTP,0)
00240             CALL TBLP(XRE(IB),REYN,FF(IB),FRF2,NTP,1)
00241          15 CONTINUE
00241      C
00241      C      DETERMINE HEAT TRANSFER COEFFICIENT
00241      C
00242             IF(IFLAG.EQ.0) GO TO 50
00244             IF(I.EQ.2)GO TO 50
00246             HTC(1)=(NUSLT*COND*12.)/DIAMTI
00247             GO TO 55
00250          50 HTC(I)=(HTN(I)*XG(I)*CP(I)*60.)/XPR(I)**.667
00250      C
00250      C      DETERMINE U
00250      C
00251          55 XM(I)=((2.*HTC(I)*12.)/(XKM*FT(I)))**.5
00252             IF(IFLAG.EQ.1) GO TO 58
00254             XL(I)=CW(I)/(2.*12.)
00255             GO TO 59
00256          58 XL(I)=(((DTH+DTC)/2.)-DIAMTO)/(2.*12.)
00257          59 ETAF(I)=TANH(XM(I)*XL(I))/(XM(I)*XL(I))
00260             ETAO(I)=1.-AFAT(I)*(1.-ETAF(I))
00261             XPT(I)=XPTC(I)/12.
00262         100 CONTINUE
00264             IF(IFLAG.EQ.1) GO TO 60
00266             AH=AHTOVH*NS(2)*(HXH(2)*HXD(2)*CW(2))/1728.
00267             AC=ACTOVC*NS(1)*(HXH(1)*HXD(1)*CW(1))/1728.
00270             GO TO 65
00271          60 AC=3.1415*DIAMTO*HXD(1)*NC(1)/144.
00272             XPASS=HXL(1)/HXL(2)
00273             AH=AHTOVH*(HXH(2)*HXD(2)*HXL(2)*XPASS)/1728.
00274          65 CONTINUE
00275             ACTOAH=AC/AH
00276             AVGAH=(AH+AC)/2.
00277             TERM1=(1./(ETAO(2)*HTC(2)))
00300             TERM2=(1./((AC/AH)*ETAO(1)*HTC(1)))
00301             UINV=TERM1+(XPT(2)/((AVGAH/AH)*XKM))+TERM2
00302             U=1./UINV
00302      C
00302      C      DETERMINE CMIN AND CMAX
00302      C
00303             DO 300 N=1,2
00306             C(N)=XMDOT(N)*CP(N)*60.
00307         300 CONTINUE
00311             CAIR=C(1)
00312             CGAS=C(2)
00313             IF(CAIR-CGAS) 310,310,320
00316         310 CMIN=CAIR
00317             CMAX=CGAS
```

Figure A-7. Continued

```
00320          GO TO 330
00321      320 CMIN=CGAS
00322          CMAX=CAIR
00323      330 CONTINUE
00323    C
00323    C     DETERMINE NUMBER OF TRANSFER UNITS AND EFFICIENCY
00323    C
00324          NTU=AH*U/CMIN
00325          EFF=1.-EXP(-NTU)
00326          DCOEF=(HXD(1)*HXH(1)*HXL(1))/EFF
00326    C
00326    C     DETERMINE IN/OUT TEMPERATURES
00326    C
00327          TOH=TIH-EFF*(TIH-TIC)
00330          QDOT=XMDOT(2)*CP(2)*(TIH-TOH)*60.
00331          TOC=TIC+(QDOT/(XMDOT(1)*CP(1)*60.))
00331    C
00331    C     DETERMINE PRESSURE DROP
00331    C
00332          SPECVL(1,1)=13.33
00333          SPECVL(1,2)=13.65
00334          SPECVL(2,1)=21.74
00335          SPECVL(2,2)=14.08
00336          DO 420 MN=1,2
00341          IF(IFLAG.EQ.1.AND.MN.EQ.1) XHR(1)=2.*XHR(1)
00343          AREAA(MN)=HXD(MN)*4./(XHR(MN)*12.)
00344          IF(IFLAG.EQ.1.AND.MN.EQ.1) XHR(1)=XHR(1)/2.
00346          IF(IFLAG.EQ.1) GO TO 80
00350          SIGMA(MN)=NS(MN)*NC(MN)*FA(MN)/(HXH(MN)*HXL(MN))
00351          GO TO 85
00352       80 SIGMA(1)=CA(1)/FA(1)
00353          SIGMA(2)=SIG
00354       85 CONTINUE
00355          MSPV(MN)=(SPECVL(MN,1)+SPECVL(MN,2))/2.
00356          A(MN)=((XG(MN)**2)*SPECVL(MN,1))/(2.*32.2*3600.)
00357          B(MN)=(1.+SIGMA(MN))
00360          F(MN)=((SPECVL(MN,2))/(SPECVL(MN,1)))-1.
00361          D(MN)=(FF(MN)*AREAA(MN))
00362          E(MN)=MSPV(MN)/SPECVL(MN,1)
00363          DP(MN)=A(MN)*((B(MN)*F(MN))+(D(MN)*E(MN)))
00364          DELTAP(MN)=DP(MN)/(144.*.036125)
00365      420 CONTINUE
00367          AREA(1)=AC
00370          AREA(2)=AH
00371          TI(1)=TIC
00372          TI(2)=TIH
00373          TO(1)=TOC
00374          TO(2)=TOH
00374    C
00374    C     PRINT OUTPUT
00374    C
00375          IF(IDIAG.EQ.0)GO TO 2000
00377          WRITE(6,1200)AH,AC,U,EFF,TOH,TOC,      ACTOAH,CMIN,CMAX,QDOT
00377         *,AVGAW,NTU
00415     1200 FORMAT(1H1,13F9.3)
00416          DO 1100 I=1,2
00421          WRITE(6,1000)HXH(I),HXD(I),HXL(I),      FA(I),CA(I),XG(I),XRE(I),
00421         *HTC(I),DP(I),DELTAP(I),NC(I)
00436          WRITE(6,1000) XM(I),XL(I),ETAF(I),ETAO(I),AREAA(I),SIGMA(I),
00436         *MSPV(I),HTN(I),FF(I)
```

Figure A-7. Continued

```
00451   181*        WRITE(6,1000) A(I),B(I),F(I),D(I),E(I)
00460   182*   1000 FORMAT(10F10.4,I5)
00461   183*   1100 CONTINUE
00463   184*   2000 CONTINUE
00464   185*        WRITE(6,1400) NHXCON,HXD(1),HXD(2),HXH(1),HXH(2),HXL(1),HXL(2)
00475   186*   1400 FORMAT(1H1,48X,'HEAT EXCHANGER PERFORMANCE CODE',//,55X,
00475   187*       1'CONFIGURATION',1X,I2,14X,'COLD SIDE',5X,'HOT SIDE',//,24X,
00475   188*       2'HEAT EXCHANGER LENGTH - PARALLEL TO FLOW - INCHES',10X,F9.3,
00475   189*       35X,F9.3,//,24X,
00475   190*       4'HEAT EXCHANGER LENGTH - PERPENDICULAR TO FLOW - INCHES',5X,
00475   191*       5F9.3,5X,F9.3,//,24X,
00475   192*       6'HEAT EXCHANGER LENGTH - DIRECTION OF CELLS - INCHES',8X,F9.3,5X,
00475   193*       7F9.3,//,52X)
00476   194*        IF(IFLAG.EQ.1) GO TO 150
00500   195*        WRITE(6,1450)CS(1),CS(2),CW(1),CW(2),XMDOT(1),XMDOT(2),NS(1),NS(2)
00512   196*   1450 FORMAT(52X,'SINGLE CELL HEIGHT - INCHES',4X,F9.3,5X,F9.3,//52X,
00512   197*       9'SINGLE CELL WIDTH - INCHES',5X,F9.3,5X,F9.3,//,52X,
00512   198*       *'FLOW RATE - LB/MIN',13X,F9.3,5X,F9.3,//,52X,
00512   199*       1'NUMBER HX SEGMENTS',17X,I2,12X,I2,///)
00513   200*        GO TO 160
00514   201*    150 WRITE(6,1460)CS(1),CS(2),CW(1),CW(2),XMDOT(1),XMDOT(2),NS(1),NS(2)
00526   202*   1460 FORMAT(52X,'TUBE DIAMETER - INCHES',9X,F9.3,//52X,
00526   203*       1         'TUBE+FIN DIAMETER - INCHES',19X,F9.3,//52X,
00526   204*       2         'FIN LENGTH - INCHES',12X,F9.3,5X,F9.3,//,52X,
00526   205*       3         'FLOW RATE - LB/MIN',13X,F9.3,5X,F9.3,//,52X,
00526   206*       4         'NUMBER HX TUBES',21X,I2,12X,I2,///)
00527   207*    160 CONTINUE
00530   208*        WRITE(6,1510)XRE(1),HTC(1),AREA(1),DELTAP(1),TI(1),TO(1),
00530   209*       *XRE(2),HTC(2),AREA(2),DELTAP(2),TI(2),TO(2)
00546   210*   1510 FORMAT(31X,'REYNOLDS',2X,'HEAT TR.',4X,'HEAT TRANSFER',5X,
00546   211*       1'PRESSURE',3X,'INLET',5X,'OUTLET',/,32X,
00546   212*       2'NUMBER',5X,'COEF',10X,'AREA',11X,'DROP',6X,'TEMP',7X,'TEMP',/40X,
00546   213*       3'(BTU/FT**2',5X,'(FT**2)',9X,'(IN H2O)',4X,'(F)',7X,'(F)',5X,/42X,
00546   214*       4'HR F)' ,//22X,'COLD',3X,2F10.4,5X,F10.4,5X,3F10.4,/22X,'SIDE',//,
00546   215*       522X,'HOT',4X,2F10.4,5X,F10.4,5X,3F10.4,/,22X,'SIDE',///)
00547   216*        WRITE(6,1600)U,NTU,EFF,QDOT
00555   217*   1600 FORMAT(44X,'U',8X,'NTU',4X,'EFFICIENCY',3X,'QDOT',/,40X,
00555   218*       1 '(BTU/FT**2',21X,'(BTU/HR)',/,42X,'HR F)',//,38X,3F10.4,2X,F10.4)
00556   219*        WRITE(6,1610)TERM1,TERM2
00562   220*   1610 FORMAT(///,46X,'TERMS 1 AND 3 OF 1/U EQUATION',//52X,'TERM 1',
00562   221*       *5X,'TERM 3',//,48X,F10.5,2X,F10.5)
00563   222*        WRITE(6,1620) DCOEF
00566   223*   1620 FORMAT(///,50X,'EFFICIENCY COEFFICIENT',//,61X,F10.4)
00567   224*   5000 CONTINUE
00571   225*    800 STOP
00572   226*        END
```

* CROSS REFERENCE BY SEQUENCE NUMBER *

```
NAMES------
A      :   0106   0356   0363   0451
AC     :   0267   0271   0275   0276   0300   0367   0377
ACTOAH:    0275   0377
ACTOVC:    0110   0267
AFAT   :   0103   0110   0260
AH     :   0266   0273   0275   0276   0300   0301   0324   0370   0377
AHTOVH:    0110   0266   0273
AREA   :   0103   0367   0370   0530
AREAA  :   0103   0343   0361   0436
```

Figure A-7. Continued

```
            DLB,289777,1,50                                                          28 AUG 74   14:00:43   PAGE   12

# FOR TBLP,TBLP                                                                      28 AUG 74   00:00:03.784
DATE, TIME, LEVEL OF OUTPUT ELEMENT: 28 AUG 74  14:00(00)
FORTRAN V:  ISD VERSION 2.9

   SUBROUTINE TBLP       ENTRY POINT 000101

   STORAGE USED (BLOCK, NAME, LENGTH)

         0001    *CODE    000123
         0000    *DATA    000024
         0002    *BLANK   000000


   EXTERNAL REFERENCES (BLOCK, NAME)

         0003    NERR3$


   STORAGE ASSIGNMENT FOR VARIABLES (BLOCK, TYPE, RELATIVE LOCATION, NAME)

    0001    000032 1L          0001   000041 2L          0001    000056 20L        0001    000022 4L          0001    000044 8L
    0000 I 000000 I            0000 R 000001 RX


00101      1*          SUBROUTINE TBLP(X,XT,A,AT,NX,INT)
00103      2*
00103      3*    C     SINGLE INTERPOLATION WITH EXTRAPOLATION.
00103      4*    C         INT=0, FIND FACTOR RX FIRST.  INT=1, RX HAS BEEN FOUND PREVIOUSLY.
00103      5*    C
00103      6*          DIMENSION XT(1),AT(1)
00104      7*          IF(INT.GT.0)GO TO 20
00106      8*       10 I=2
00107      9*        4 IF(XT(I)-X) 1,2,8
00112     10*        1 I=I+1
00113     11*          IF(I.EQ.NX)GO TO 8
00115     12*          GO TO 4
00116     13*        2 RX=1.0
00117     14*          GO TO 20
00120     15*        8 RX=(X-XT(I-1))/(XT(I)-XT(I-1))
00121     16*       20 A=RX*(AT(I)-AT(I-1))+AT(I-1)
00122     17*          RETURN
00123     18*          END

                                        * CROSS REFERENCE BY SEQUENCE NUMBER *

 NAMES-------
 A       :   0101    0121
 AT      :   0101    0103    0121
 I       :   0106    0107    0112    0113    0120    0121
 INT     :   0101    0104
 NX      :   0101    0113
 RX      :   0116    0120    0121
 TBLP    :   0101
 X       :   0101    0107    0120
```

Figure A-7. Continued

```
 @ ELT RNDK,1,740828, 50400

          $HX CS=.377,.1675,CW=.47,.249,NFC=2,2,FT=.006,.006,XHR=.02016,.009396,
              AFAT=.719,.769,XPTC=.006,.006,ACTOVC=188.,AHTOVH=393.,XKH=120.,
              IDIAG=1  $END
          $CONFIG ICONFG=2   $END
          $HXTYPE IFLAG=0  $END
          $CHECK RHO=.075,.0733,.046,.071,XMU=.000013,.0000155,CP=.24,.26,
               XPH=.72,.7  $END
          $HC IA=1,IB=2  $END
          $REN REYN=300.,400.,500.,600.,800.,1000.,1200.,1500.,2000.,3000.,4000.,
                    5000.,6000.,7000.,8000.,9000.,10000.,NPT=17  $END
          $HT1 HTRC1=.013,.011,.0095,.0085,.0073,.0065,.0061,.0057,.0055,.0052,
                     .0048,.0045,.0043,.004,.00397,.0038,.0037  $END
          $FRIC1 FRF1=.057,.047,.037,.03,.023,.019,.017,.0146,.013,.011,.00978,
                      .00913,.0087,.0082,.0081,.0078,.0076  $END
          $HT2 HTRC2=.0122,.0098,.0083,.0073,.0059,.005,.0044,.0038,.0033,.0032,
                     .0033,.0032,.00322,.0032,.0031,.003,.00294  $END
          $FRIC2 FRF2=.0535,.0411,.0336,.0285,.022,.0181,.0159,.0135,.01145,
                      .0096,.0089,.0086,.0082,.0078,.0076,.0073,.0072  $END
          $FLOW XMDOT=75.,2.07,TIH=500.,TIC=70.  $END
          $VAR HXH=20.0,1.508,HXD=1.508,20.0,HXL=15.0,15.,NS=18,18  $END
          $VAR HXH=20.,2.33,HXD=2.33,20.,HXL=10.,10.,NS=13,13  $END

 END CUR,    ISD VERSION 2.14
```

Figure A-7. Concluded

Δt_o = temperature difference between streams at the inlet (greatest temperature difference)

Δt_L = temperature difference between streams at the exit (least temperature difference)

The overall conductance is calculated as before for the noncondensible portion and a heat transfer area determined to achieve an effectiveness to reach the dewpoint.

The condensing heat transfer coefficient was estimated using a procedure outlined by McAdams (50) in Chapter 13, Part IV. The situtation is one of a condensible in an otherwise noncondensing medium. This produces a heat transfer coefficient different from either a pure condensing stream or pure forced convection. The procedure is as follows:

A partial resistance $1/U^1$ is calculated which consists of only the resistances on the cold side, the dirt deposit, the tube wall and the condensate film:

$$\frac{1}{U^1} = \frac{1}{h_L A_L/A_v} + \frac{1}{h_d A_w/A_v} + \frac{1}{k_w A_w/A_v} + \frac{1}{h_m}$$

The overall heat transferred balance from the cold side to the gas and condensing stream is then

$$\dot{q}'' = U^1(t_i - t_L) = \underbrace{h_G\ (t_\nu - t_i)}_{\text{sensible heat}} + \underbrace{\lambda[K_G(P_\nu - P_i)]}_{\text{latent heat}}$$

where

h_G = forced convection heat transfer coefficient on the gas side (Btu/hr-ft^2°F)

λ = latent heat of condensation Btu/lbm

K_G = mass transfer coefficient through gas film (lbs vapor condensed/hr-ft^2)

t_L = temperature of cold stream

t_ν = vapor temperature

t_i = interface temperature between condensed layer and gas

P_i = vapor pressure of water at t_i

P_ν = vapor pressure of water at t_ν

One assumes a value of t_i and substitutes the vapor pressure and repeats the procedure until the equation balances. The total area required is then determined by graphically or otherwise integrating the following expression:

$$A = \int_o^q \frac{dq}{U^1 (t_i - t_L)}$$

This must be done stepwise proceeding through the heat exchanger. A rough guess (and this is what was done) on the resulting overall heat transfer coefficient to any location may be found by using the following expression:

$$\bar{h} = \frac{\dot{q}''}{(t_\nu - t_L)}$$

By looking at this value of $\bar{h}$ at the inlet and outlet conditions an average value may be chosen to estimate the overall area required for the condensing portion of the exchanger. Mass transcoefficients can be determined from the expression:

$$\frac{K_G P_{nm}}{G} \left(\frac{\mu_\nu}{\rho_\nu D_\nu}\right)^{2/3} = \frac{0.023}{(N_R)^{0.2}}$$

where

$\frac{\mu_\nu}{\rho_\nu D_\nu} \approx$ Schmidt Number $\approx$ 0.6 for water vapor in air

P_{nm} = log mean vapor pressure of the noncondensible portion at t_i and t_ν

N_R = Reynolds number of the vapor

Pressure drops were calculated using the computer program as before but using the total heat transfer volume.

TECHNICAL REPORT DATA
(Please read Instructions on the reverse before completing)

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15. SUPPLEMENTARY NOTES Task officer for this report is W.B. Steen, Mail Drop 61, Ext 2825.

16. ABSTRACT The report gives results of a brief study to determine the feasibility of candidate concepts for simultaneous heat and air pollutant recovery from the exhaust of domestic-size furnaces. Among the concepts investigated were improved heat exchanger design, vent dampers and heat pipes, and post-combustion emission control devices such as filters and wet scrubbers.

17. KEY WORDS AND DOCUMENT ANALYSIS

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Air Pollution	Heat Exchangers	Air Pollution Control	13B
Heat Recovery	Vents	Stationary Sources	20M,13A
Combustion	Damping	Domestic Heating	21B
Furnaces	Pipes (Tubes)	Vent Dampers	13K
Flue Gases	Filters	Wet Scrubbers	
Exhaust Systems	Scrubbers		07A

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