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Project Summary

Performance Evaluation of the Braintree Electric Light Department Dry Cooling Tower

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The performance of a dry cooling tower for the 20-MW steam-electric generation portion of an 85-MW combined-cycle power plant was evaluated in a 5-year project. Under a grant to the Braintree Electric Light Department, the objectives of the demonstration were to demonstrate dry cooling tower technology at a Massachusetts seacoast site, document and optimize heat rejection performance, evaluate the effect of dry cooling tower operation on the environment, and define the effect of environmental conditions on dry cooling tower performance.

Since startup of the plant in 1977, the unit has been on-line for only about 2100 hours due to several equipment failures associated with the gas turbine which caused 17 months of forced outage. Another major reason for low utilization has been the escalating cost of fuel oil. Originally conceived as an intermediate load unit, high fuel costs have shifted it to peaking service. This decreased operation reduced the scope of the originally planned demonstration.

During 1979 and 1980 the performance of the dry cooling tower was close to design. The combined-cycle heat rate has always exceeded its design value, being about 10 percent higher in 1980. Data collected were inadequate to demonstrate the freezing and corrosion resistance of the tower's finned tubes or the noise generated by the dry cooling tower apart from the entire combined-cycle unit.

This Project Summary was developed by EPA's Industrial Environmental Research Laboratory, Research Triangle Park, NC, to announce key findings of the research project that is fully documented in a separate report of the same title (see Project Report ordering information at back).

Introduction

The purpose of this report is to present the results of a performance evaluation of the dry cooling tower serving Braintree Electric Light Department's (BELD's) combined-cycle power plant. This project was initiated with an EPA-sponsored demonstration grant in December 1975. Several areas of investigation were planned, the most important being optimization of plant performance. The plant became commercial in April 1977. Performance monitoring equipment was leased and installed between September 1976 and June 1978. During 1978 two separate failures of gas turbine equipment occurred. Another gas turbine equipment failure occurred early in 1980. The total forced outage time to date is approximately 17 months. Because of increases in the price of gas turbine fuels, the plant is currently used only for peaking service, rather than for intermediate load service as anticipated. At the time of preparation of this final report only about 2000 hours of operation have been accumulated. Thus the program envisioned at the start of this project has been greatly curtailed even though the original grant period of 4 years was extended for 1 year.

The objectives of the program were to demonstrate dry cooling tower technology and to document and optimize heat rejection performance. The effect of environmental conditions on performance and the effect of operation on the environment were to be defined. Provisions were made in plant design to utilize residual oil for fuel resulting in a performance penalty. However, only distillate oil has ever been used.

Conclusions

BELD's combined-cycle unit has not operated nearly as frequently as planned due to the substantial increase in fuel oil cost subsequent to its design and construction. There have also been substantial forced outages due to major failures of gas turbine equipment. However, with rebuilding and increased operating experience the performance of the unit has improved.

While it has not been possible to optimize the performance of the dry cooling tower, monitoring equipment is available to detect deviations in steam-turbine operation, dry cooling tower heat rejection, and combined-cycle heat rate. Ambient temperature is the most important environmental condition affecting dry tower performance and unit output. Available data are inadequate to demonstrate the freezing and corrosion resistance of the fintubes or to address the noise generated by the dry cooling tower apart from the entire combined-cycle unit. There are no data that would indicate passage of the dry tower plume in the downwind environment.

Data from approximately 350 hours of operation in 1979 and 1980 have been analyzed. The performance of the dry cooling tower is reasonably close to design when condensate temperatures are analyzed. It appears that pressure measurements are not as accurate as temperatures in this regard. However, at ambient temperatures above approximately 60°F*, a deterioration in performance is noticeable. It is possible that seasonal adjustments to the pitch angle of the fan blades (to increase air flow in the summer) is warranted to overcome this problem.

The combined-cycle heat rate has always exceeded the design value, even though it has decreased since initial operation. The problem is thought to be with the gas turbine combustor since steam cycle parameters and gas turbine generator output are reasonably close to design values. Adjustments to compressor air flow and combustion temperature may allow reduction of the heat rate.

Recirculation of hot exhaust air from the dry cooling tower to the fan inlet happens only to the bank of cells closest to the combined-cycle plant building. The causes for this recirculation are unknown but seem to be related to the diminished heat transfer of the lower portion of south-side fintubes on Cell 2A-4. Otherwise, heat transfer seems to be uniform across individual cells and the entire dry cooling tower.

Data Collection Analysis

All the performance monitoring equipment was installed by June 1978. However, minimal additional operation occurred that year. From April 1979 (when Unit 2 was returned to service following an August 14, 1978, forced outage) through October 1980 about 1200 hours of operation occurred. For that period, data have been obtained and analyzed for some 350 hours. The magnetic tape unit was found to be malfunctioning in January 1980 and only data manually obtained by plant personnel (on a schedule of approximately once per hour when the unit is running) are available.

Data on steam parameters, fuel consumption, fan operation, power production, and temperatures of air and condensate have been nearly complete. However, both air quality and meteorological data (with the exception of temperature profiles) have been sporadic and of little value.

A Fortran computer program has been written to read and analyze the performance data and to echo the input. The program computes an average condensate temperature and corresponding backpressure (based on the GEA expectation of 4°F subcooling), temperature corresponding to the measured backpressure, steam-side duty (based on measured steam flow and an assumed heat of condensation of 1000 Btu/lb), average fan inlet and fintube exhaust temperatures, localized temperature differentials on one cell, average fan power, air-side duty, theoretical backpressure (based on the GEA performance curves), heat rate, and theoretical heat rate (based on the manufacturer's design values of gas turbine output and fuel consumption vs. temperature, steam turbine output vs. steam flow, and backpressure). Tests are also performed to determine variations in local heat transfer on one cell, recirculation (with possible correlation to low-level winds), same operating power on all fans (only situation for which GEA backpressure is available), and backpressure limits (limited range of steam flow and air temperature).

Results

More than 90 percent of the data have net power values greater than 75 MW with steam flow in excess of 180,000 lb/hr and fans running at full speed. Variations in duty between steam-side cooling and air-side heating are within the range of ± 10 percent. Therefore, the primary influence on steam turbine backpressure is ambient temperature. (The average fan inlet temperature from all 10 cells is considered to be ambient temperature in the following discussion.) The gas turbine flow is essentially a constant volumetric rate, so that as ambient temperature decreases gas turbine power generation increases. Steam production in the heat recovery boiler also increases with decreasing ambient temperature and the condensing duty decreases in difficulty. Consequently backpressure goes down and steam turbine power generation increases. While generation is increasing so is fuel consumption, with the net result that heat rate changes little with ambient temperature. Both design conditions and actual data are presented in Figures 1 through 5 and are discussed below.

Backpressure

Recorded backpressure measurements (converted from the vacuum readout) at nominal full-load conditions range from 2 to 13 in. Hg absolute. The high end of this range exceeds the expected range for the dry cooling tower. The data which have been analyzed in this study are presented in Figure 1. Measured backpressure exceeds the GEA estimate by approximately 2 in. Hg in 75 percent of the comparisons. It is also apparent that the performance of the dry cooling tower

*Nonmetric units are used in this report because they remain the standard in the utility industry. Metric equivalents appear at the end of this summary for readers more familiar with that system.

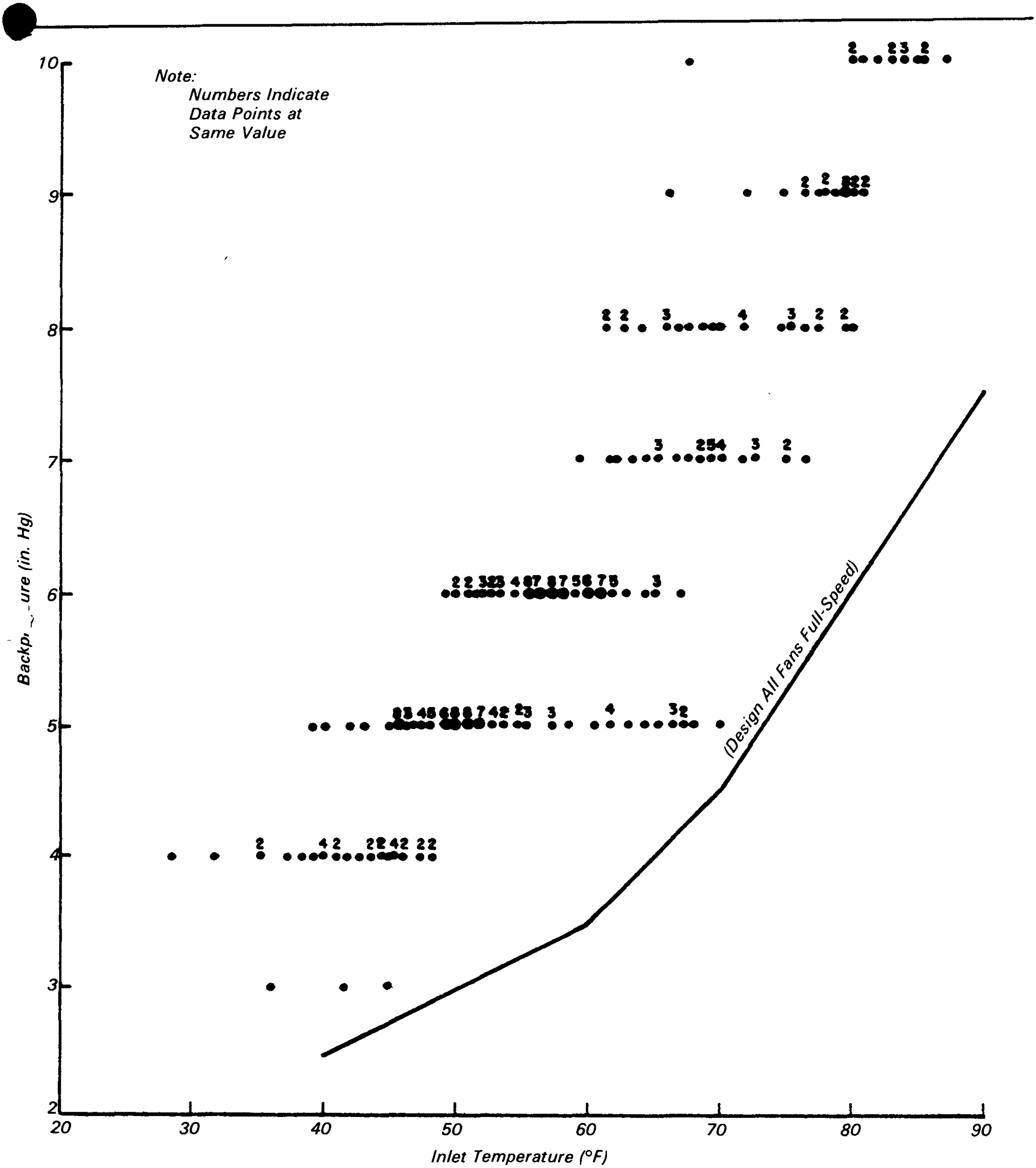

Figure 1. *Variation of measured backpressure with ambient temperature.*

deteriorates appreciably at ambient temperatures above 60°F. Only limited data were available with the fans run at half-speed and no comparisons were made for the condition.

Data from the plant operator's log indicate that the temperature of steam exhausting the turbine is very close (an average difference of approximately 2°F) to the condensate temperature. This would indicate that GEA's estimate of only 4°F subcooling is correct. Therefore, it was decided that the thermocouple measurement of condensate temperature would be a more accurate means of comparing actual and design backpressure. Figure 2 shows values approximately 1 in. Hg higher than the GEA estimate in 75 percent of the comparisons. The deviation from design values at ambient temperatures above 60°F is also less than that for the measured backpressure data.

While the average fan power is close to design, the average temperature differential across the fintubes is 58°F, compared with the design value of 53°F. This would indicate that air flow is less than design. At present the pitch angle of the fan blades is less than the design value due to the observation by BELD personnel that power consumption by the fan motors was excessive when the pitch angle was higher. However, it was also observed that backpressure was closer to design when the pitch angle was higher. It is estimated that a reduction of 1 in. Hg in backpressure would result in approximately a 1 percent increase insteam turbine generator output, or approximately 200 kW. The current fan power consumption is 300 to 400 kW. Therefore, the tradeoff would have to be analyzed with operating data to see if it is worthwhile to optimize the performance of the dry tower by increasing the fan blade angles.

Heat Rate

Of even greater concern to BELD management is the fact that the unit heat rate exceeds the design value. Operation in 1980 has been at a higher load and at a lower heat rate than 1979. Figures 3 and 4 compare actual and design values in the 2 years. As can be seen from Figures 3 and 4, the design heat rate changes little with ambient temperature. However, it is expected that heat rate increases with elapsed time since major overhaul and with decreased load. The gas turbine combustion temperature also influences heat rate (higher temperatures result in lower heat rates and less time between overhauls) and is monitored by operations personnel. Due to the expected lower efficiency for 1979 operation, it was decided to continue the heat rate analysis only for the 1980 data.

The data in Figure 4 show an average heat rate approximately 1200 Btu/kWh above the design value. The variation at a given ambient temperature is approximately 1000 Btu/kWh. No attempt has been made in this study to explain this variation. In an attempt to explain why the actual heat rate is so far above the design value, a comparison was made of actual and design power generation. As shown in Figure 5, the relationship between the two and with ambient temperature is excellent. It should be noted that data at temperatures below approximately 32°F were taken in January 1980, prior to a major equipment failure. This may account for the reduced power output at this time.

Based on the good agreement between design and actual steam flows mentioned earlier, it is thought that the steam turbine generator is performing properly. Since the combined-cycle output is also up to par, operation of the gas turbine generator is acceptable. However, the gas turbine requires more fuel than anticipated. Verification that fuel metering is correct has been obtained from periodic checks on levels in the fuel-oil storage tank. Further investigation of compressor air flow and temperatures at various points in the flow path is planned. It may also be that the machine, which is one of the first of its type, is not as efficient as the manufacturer expected.

It should also be mentioned that recent (subsequent to the data analyzed) modifications to the gas turbine combustor have reduced the heat rate by approximately 200 Btu/kWh and increased the gas turbine power output by 2 MW. This modification is only one in a series of fine-tuning steps which may enable the combined-cycle unit to lower its heat rate.

Recirculation

With regard to the question of recirculation of exhaust air from the dry cooling tower to the fan inlet, analysis of the data indicates that it happens approximately one-third of the time. The recirculation test requires that the temperature at one cell exceed the average of all 10 by more than 5°F. Typically this requirement is met when the temperature at one cell exceeds those adjacent to it by at least 10°F. In all cases the recirculated air is experienced by the A-bank of cells closest to the combined-cycle building. In most cases, the affected cell is in the middle and least accessible to ambient air.

The exhaust steam header passes beneath the middle cell and could increase the inlet air temperature to this cell. However, most of the time this is not the case. Downwash is normally expected to occur when hot exhaust air is forced down by high-velocity winds. The intent of the wind screen around the dry cooling tower is to eliminate this problem by not allowing interaction until the hot plume is consolidated near the top of the fintube bundles. Judging by the lack of recirculation along the outer bank of cells, the wind screen is effective.

Low-level winds (at the 30-ft height on the meteorological tower) were investigated to determine whether any correlation with recirculation exists. The average wind direction is bivariate with azimuth angles of approximately 50 degrees and 210 degrees, approximately perpendicular to the long axis of the dry cooling tower. The range of values in each case is approximately 100 degrees. Thus the wind direction values correspond to a downwash interpretation of the recirculation phenomenon. The average wind speed is approximately 8 mph with more than that amount of variation. By contrast, the expected velocity of hot air exhausting from the fintubes is approximately 5 mph. While the average speed is consistent with expected downwash occurrence, the wide variation discredits this interpretation.

The recirculation is related to the adjacent, higher combined-cycle building. However, a wind-related cause cannot be convincingly demonstrated. Therefore, it is planned to investigate further the flow path of hot air exhausted from the inner bank of cells.

Localized Heat Transfer

Although not originally designed for that purpose, the localized heat transfer data from one of the A-cells seem to offer some insight to recirculation occurrences. The temperature differential across the fintubes is almost always greater on the north side than on the south side. Most of the time there is also

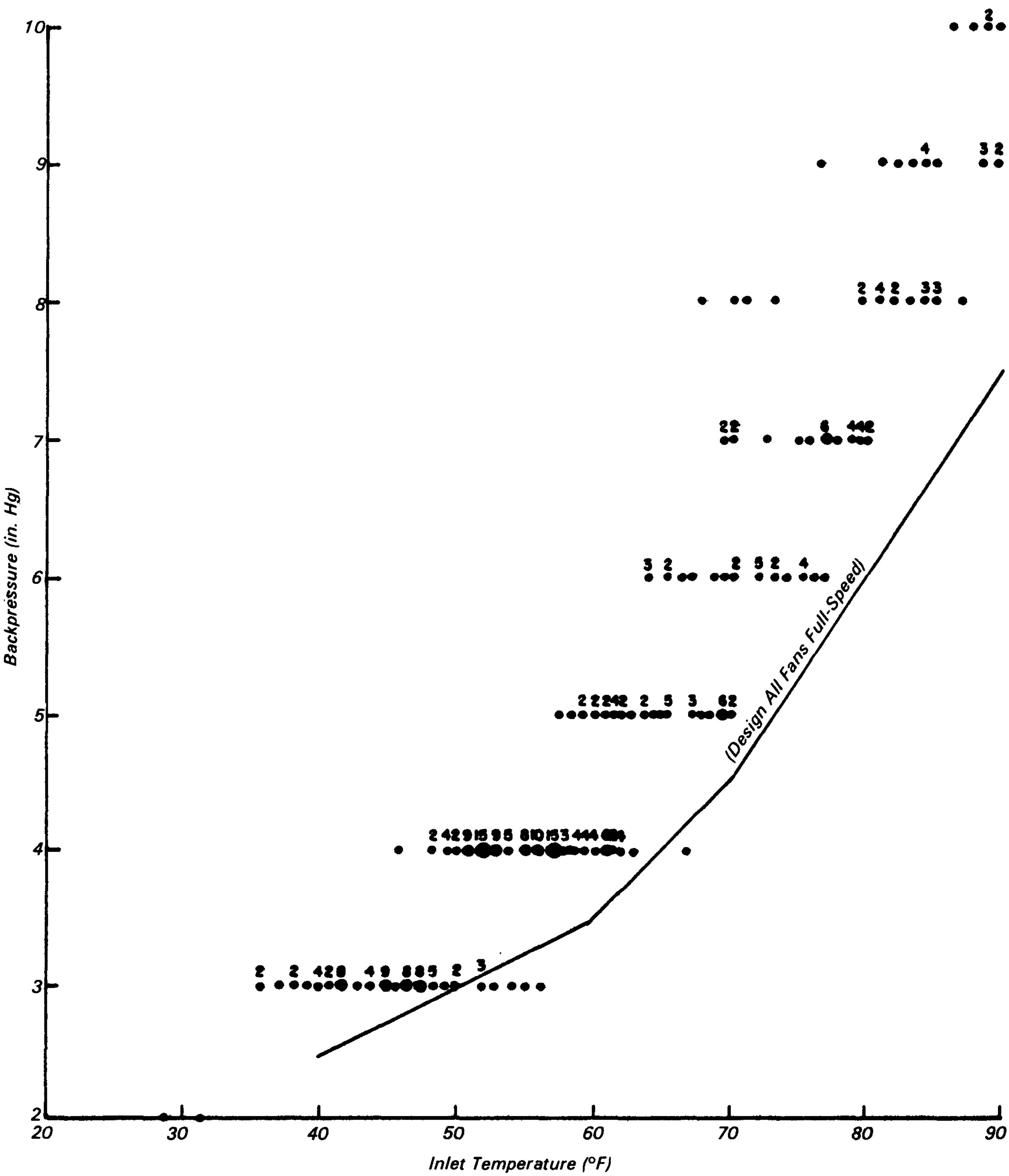

Figure 2. *Variation of condensate temperature-derived backpressure with ambient temperature.*

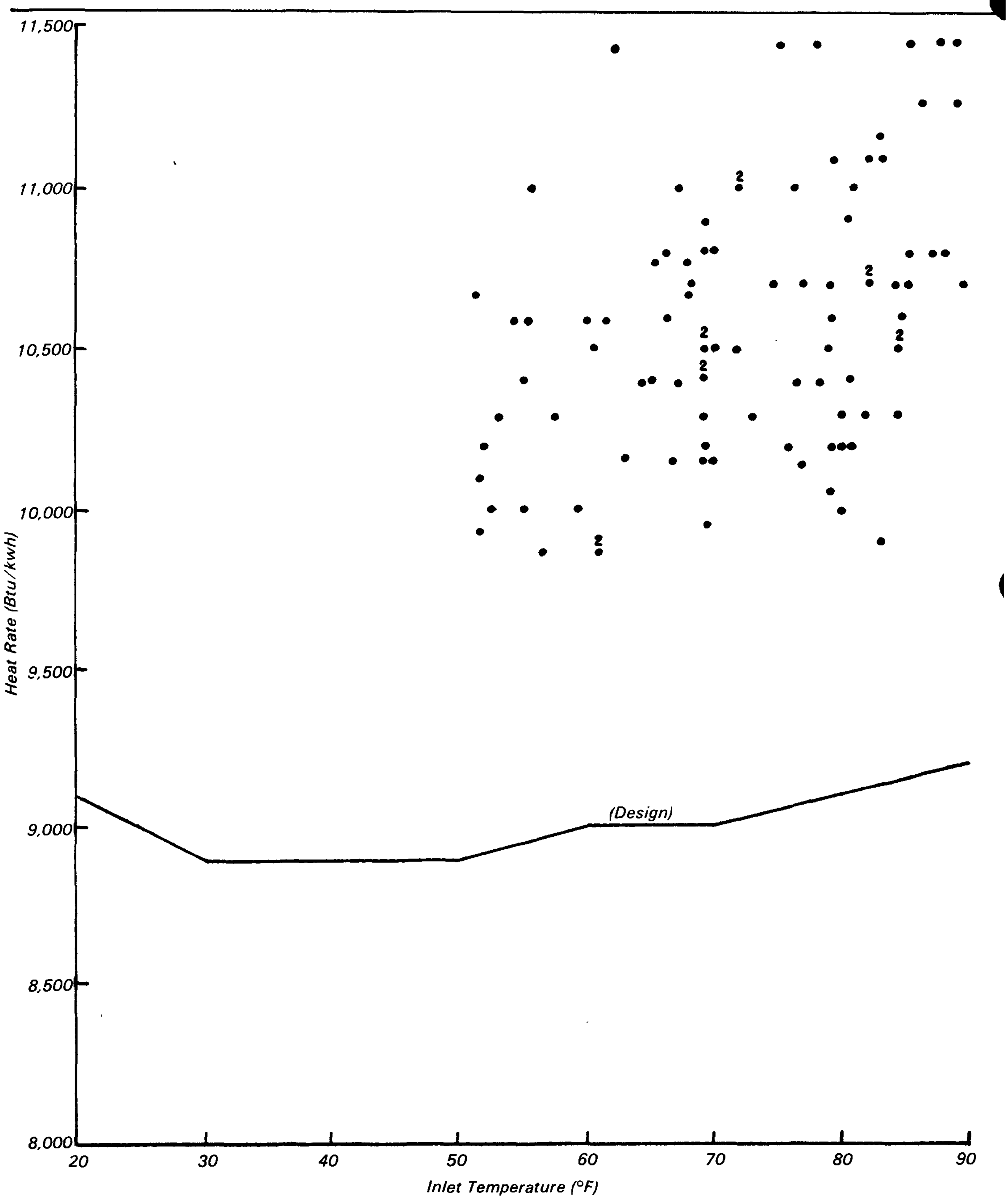

Figure 3. *Variation of heat rate with ambient temperature-1979.*

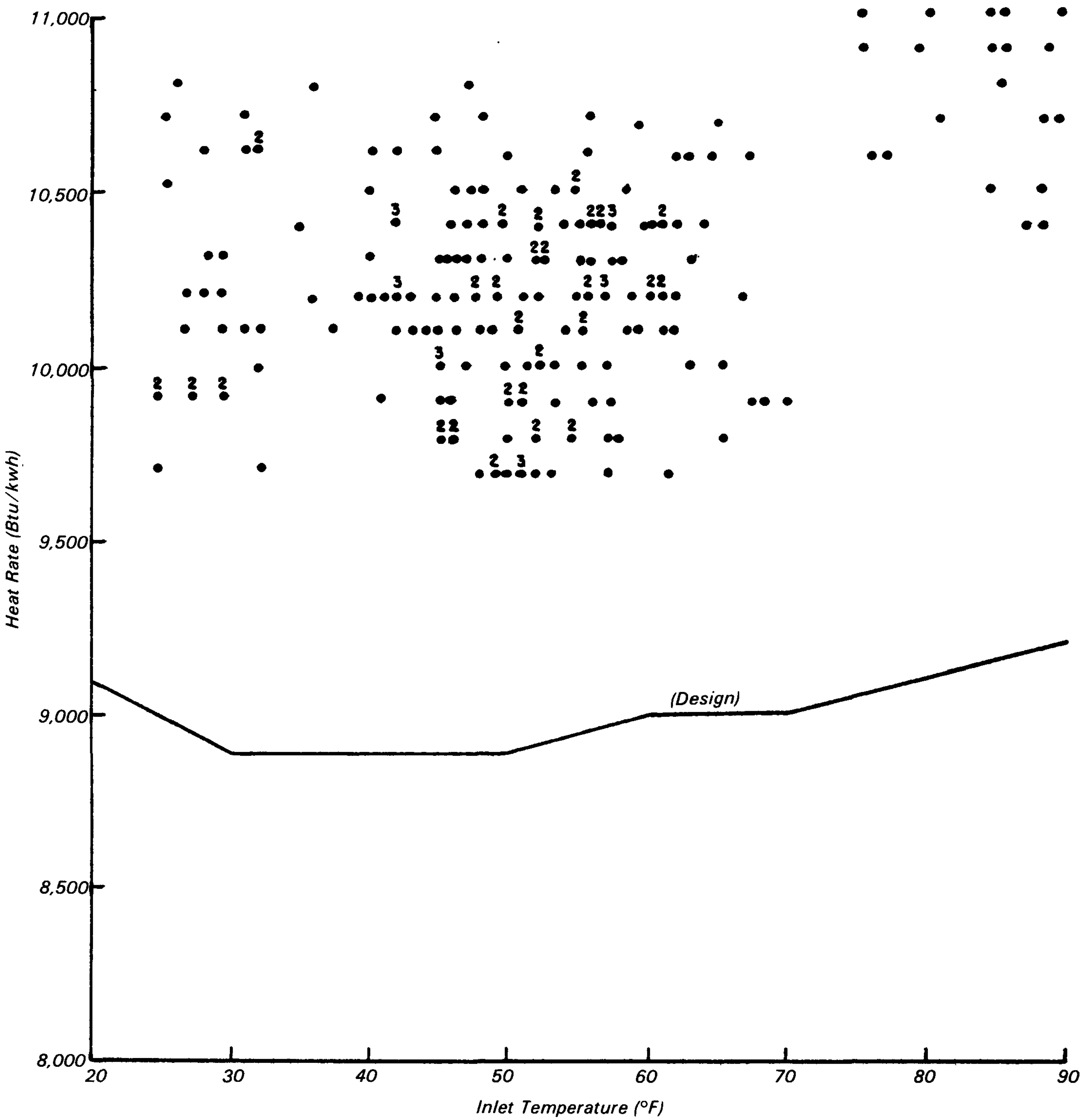

Figure 4. *Variation of heat rate with ambient temperature-1980.*

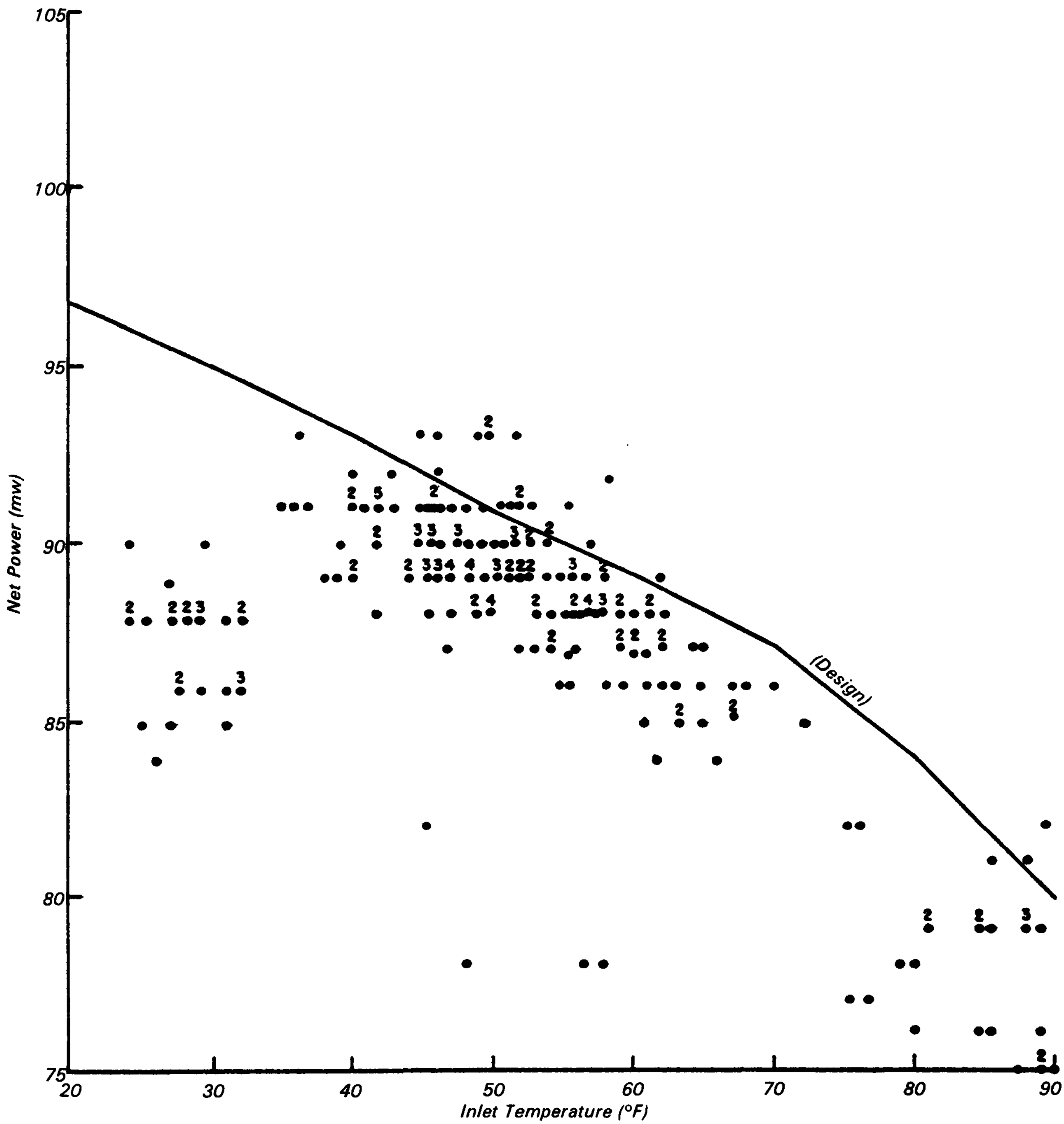

Figure 5. *Variation of power generation with ambient temperature-1980.*

local variation on the south side. Closer examination has revealed the cause. The plenum air temperature of this cell is highest at the bottom (closest portion of the fintube bundle to the combined-cycle building) and lowest at the apex. Exhaust temperatures are relatively constant leading to the variation in temperature differential. As further evidence of this phenomenon, the shroud temperatures (measured below the fan) are also highest on the building side.

The data suggest that, at least on the building side of the a A-bank of cells, heat transfer across the lower portion of the fintubes is impeded by recirculation. The exact path of this recirculation has not been determined. Otherwise, heat transfer across the fintubes appears to be uniform and in accordance with design.

Metric Conversions

Although EPA's policy is to use metric units in its publications, this document uses certain nonmetric units that remain the standard in the utility industry. Readers more familiar with metric units should use the following conversion factors:

1 in. = 2.54 cm
1 ft = 0.305 m
1 gal. = 0.0038 m^3
1 ft^3 = 0.028 m^3
1 lb = 0.45 kg
1 in. Hg = 0.033 atm
1 psi = 0.068 atm
1 mph = 0.45 m/sec
1 Btu = 252 cal
1 kWh = 860 kcal
1 hp = 0.75 kW
°C = 5/9(°F-32)
1°F (change) = 0.556°C

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The complete report, entitled "Performance Evaluation of the Braintree Electric Light Department Dry Cooling Tower," (Order No. PB 81-222 242; Cost: $8.00, subject to change) will be available only from:
National Technical Information Service
5285 Port Royal Road
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