A REDUCTION OF TURBINE EXHAUST LOSSES BY OPTIMIZING COOLING TOWER OPERATION

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ABSTRACT

Indianapolis Power & Light has been using PEPSE to reduce unit heat rate by optimizing operator controllable parameters. Turbine exhaust losses are not an operator controllable parameter, and the increase in heat rate associated with these losses was considered unrecoverable. However, PEPSE studies have shown that on units with forced draft mechanical cooling towers, the operation of the cooling tower can be modified to minimize these losses.

This paper briefly reviews the components that make up exhaust loss and how that loss affects heat rate. Model simulation and preparation for running numerous studies is discussed. Finally, the results of several studies that were conducted to determine the best achievable unit heat rate are presented.

INTRODUCTION

Indianapolis Power & Light's Unit No. 4 at our Petersburg Generating Station is a 515MWe, coal-fired unit, which started commercial operation in April 1986. The unit consists of a Combustion Engineering boiler and a General Electric turbine generator rated at 2400psig, 1005°F, 1005°F. The condenser is a two shell, dual pressure with one (1) pass per shell unit supplied by Ecolaire. Figure 1 shows the condenser arrangement for Unit No. 4. The condenser is cooled by a closed-loop circulating water system using a thirteen (13) cell mechanical draft cooling tower.

Performance tests were conducted in late 1987 on the circulating water pumps to determine actual flow rates of the pumps. The results of these tests indicated that the actual flow rate to the condenser was 217,000 GPM or eighteen percent over the original design. Table 1 shows a comparison of design to actual data for the condenser. This excess circulating water flow in addition to the reduction of air in-leakage to less than five SCFM led to operating backpressures well below the turbine design. This occurrence resulted in studies being conducted using PEPSE to determine the effects that the lower than design backpressures were having on overall system performance.

Table 1
Petersburg Unit #4 Condenser Data Comparison

PARAMETERS	DESIGN	ACTUAL
Circulating Water Flow (gpm)	177,430	217,000
Circulating Water Inlet Temp. (F)	92	92
Heat Load (BTU/Hr)	2.66×10^9 1	2.55×10^9 2
L.P. Backpressure (In of Hg)	3.35	3.29 3
H.P. Backpressure (In of Hg)	4.50	3.95 3
Tube Water Velocity (Ft/Sec)		
Low Pressure	8.70	10.6
High Pressure	7.41	9.06

Notes:
1. 5% Overpressure At VWO
2. 530 Gross Megawatts
3. Higher Than Expected Due To 80 SCFM Air In-Leakage

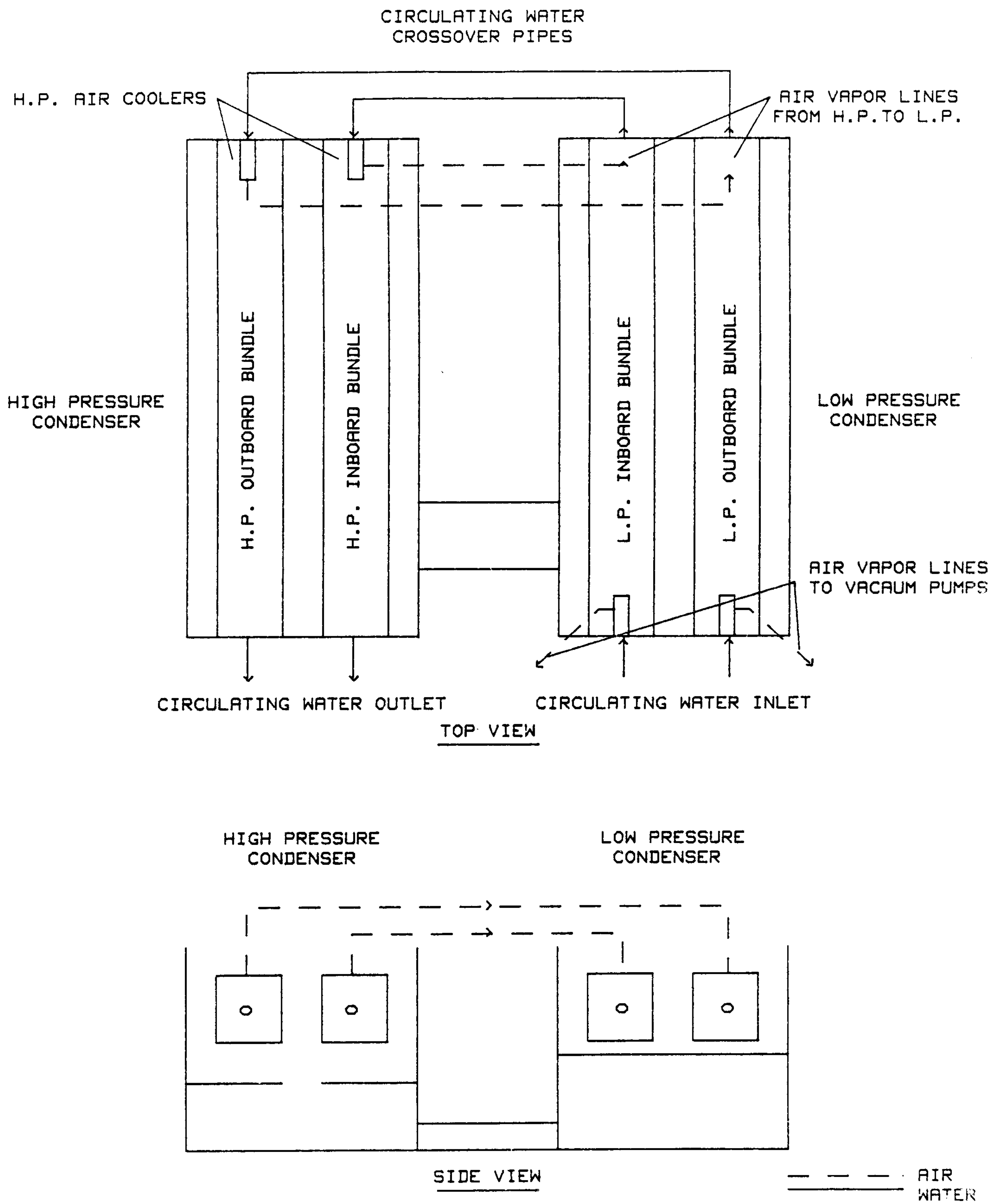


Fig. 1
Petersburg Unit #4 Condenser Arrangement

This paper is a discussion of how deviations in operating parameters from design affected unit performance by monitoring net unit heat rate. A brief review of exhaust losses and the components that effect it will be covered. A detailed description of how PEPSE was utilized to determine the optimum operating point to obtain the lowest net unit heat rate over the entire load range is given. An overview will be given of results from PEPSE runs and comparison of the output to actual unit test data. Finally, a discussion of how the output from the PEPSE study was used to help control board operators optimize unit performance.

EXHAUST LOSSES

Exhaust losses are losses which occur between the last stage of the turbine and the condenser. Exhaust loss is defined as the difference between the used energy end point and the expansion line end point. It has a greater influence on the turbine than any other single parameter. Exhaust loss is made up of four (4) component losses:

1. Actual Leaving Loss
2. Gross Hood Loss
3. Annulus Restriction Loss
4. Turnup Loss

Figure 2 shows a typical exhaust loss curve indicating the distribution of the component losses.

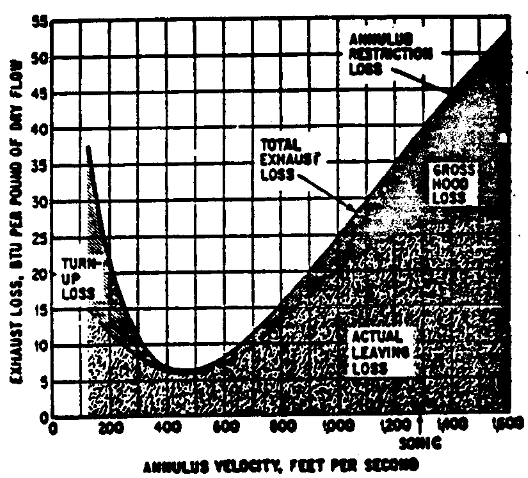


FIGURE 2

Turbine Exhaust Loss Curve Showing Distribution of Component Losses

Of the four (4) components that make up an exhaust loss curve, actual leaving loss makes up the majority of the curve. Leaving loss is the kinetic energy of the steam as it leaves the last stage. As annulus velocities increase, the next component of exhaust loss that is encountered is the gross hood loss. Hood loss is the pressure drop of the exiting steam as it passes through the exhaust hood. Once sonic velocity of the steam passing the last stage is reached, the last stage pressure ratio is constant. Any additional increase in pressure ahead of the last stage will increase the pressure following it and is unusable by the turbine. This loss is referred to a annulus-restriction loss. The final component of the exhaust loss curve is turnup losses. These losses occur when low annulus velocities are realized, which can be caused by low steam flow or high exhaust pressure.

As the descriptions of the component losses indicate, exhaust lost is a function of annulus velocity. The two operating parameters that greatly influence annulus velocity are steam flow and exhaust pressure. Petersburg Unit No. 4 is a base loaded unit for IPL and operates at constant loads for long periods of time. During these constant load periods steam flow remains relatively steady, leaving exhaust pressure as the controlling parameter for annulus velocity. There are numerous factors that can affect exhaust pressure including circulating water temperature, however, most of these factors are not operator controllable parameters. On units with cooling towers such as Unit No. 4, circulating water temperature can be controlled by putting cooling tower cells in and out of service. With this control of the circulating water temperature, control can be established of the annulus velocities.

PEPSE MODEL

Before work could begin on developing heat rate curves showing the effects of exhaust losses, a representative model of both the low and high pressure condensers was needed. In order to get a representative model of the condensers, the Heat Exchanger Institute (HEI) option was used in PEPSE to calculate the shell pressure. The data that is input for PEPSE is used in determining the heat transfer coefficient for the HEI calculation. Design data taken from the condenser specification sheets was first used in modeling the condensers. The output from this data did not adequately match test data taken from the unit.

Changes were made to the design data in PEPSE to more realistically match actual condenser performance.

Once the condensers were properly modeled, work began on setting up the model to determine the effects of exhaust losses. Figure 3 shows the last section of the low pressure turbine train and the condenser arrangement for the PEPSE model of Petersburg Unit No. 4. Changes to the LP and HP low pressure turbines had to be made. Continuance beyond minimum data for a type 7 fossil turbine is needed to include the variable LOSTYP and EXUSLY. The variable LOSTYP is a flag for PEPSE as to which moisture correction calculation is to be used. For Petersburg Unit 4, LOSTYP was set equal to 1 for use of the General Electric method of calculation. The G.E. equation in PEPSE is:

$$h_{EL} = EL\ (0.87)\ (1 - 0.01M)\ (1 - 0.0065M)$$

<u>WHERE</u>

h_{EL} = Change in expansion line end point for exhaust loss.

EL = Uncorrected exhaust loss.

M = Percent moisture.

The equation gives the difference between the expansion line end point and the used energy end point. The variable EXUSLS is the uncorrected exhaust loss which is taken from the G.E. exhaust loss curve. Figure 4 shows a G.E. exhaust loss curve. Due to the changing annulus velocity caused by varying backpressures, schedules were developed for each last stage turbine. Figure 5 shows the exhaust loss schedules that were used in the Petersburg Unit 4 PEPSE model.

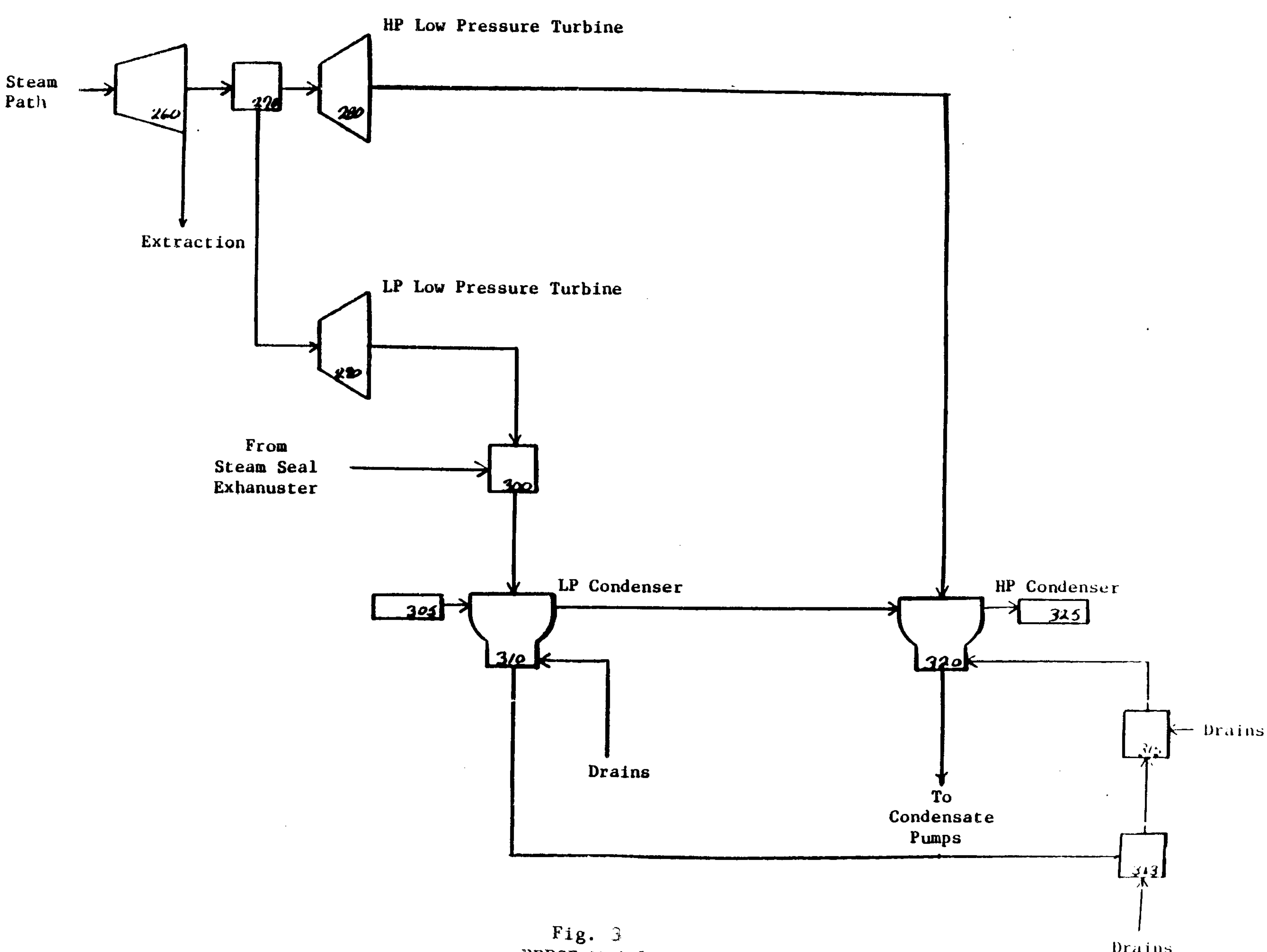


Fig. 3
PEPSE Model

EXHAUST LOSS

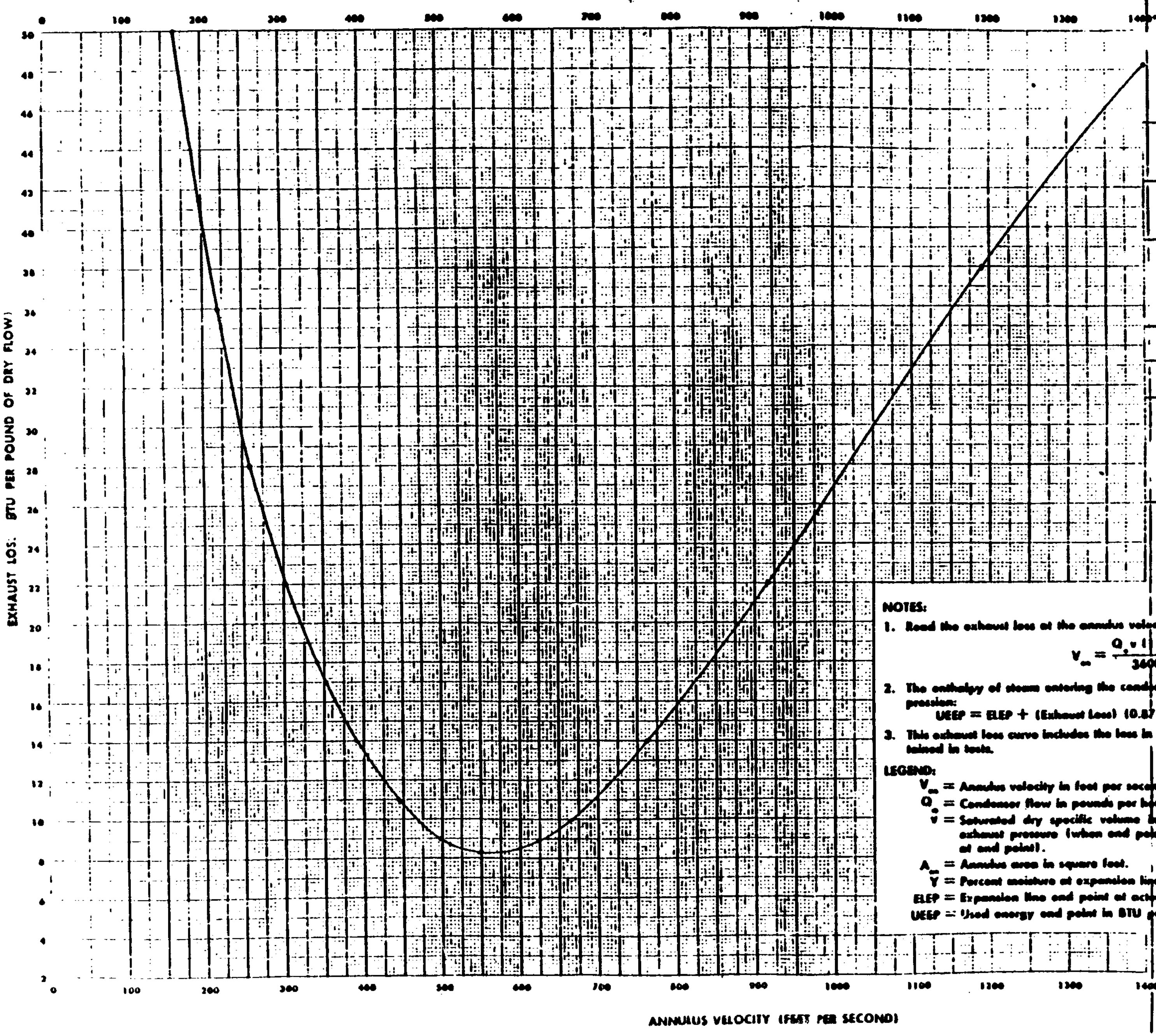
FOR STEAM TURBINE-GENERATOR UNIT

SPEED: 3600 RPM

BUCKET LENGTH (INCHES)	26
PITCH DIAMETER (INCHES)	72
LAST-STAGE ANNULUS AREA SINGLE-FLOW (SQUARE FEET)	41.1

EXHAUST PRESSURE (INCHES Hg ABS.)	SATURATED DRY SPEC. VOLUME (CUBIC FEET PER POUND)
0.5	1256.4
1.0	652.3
1.5	444.9
2.0	339.2
2.5	274.9
3.0	231.6
3.5	200.0

*For annulus velocities above 1400 ft/sec refer to Appendix III of ASME paper 62-WA 209 (GER-2007)



NOTES:

1. Read the exhaust loss at the annulus velocity obtained from the following expression.

$$V_{an} = \frac{Q_c v (1 - 0.01Y)}{3600 A_{an}}$$

2. The enthalpy of steam entering the condenser is the quanity obtained from the following expression:

   UEEP = ELEP + (Exhaust Loss) (0.87) (1 − 0.01Y) (1 − 0.0065Y)

3. This exhaust loss curve includes the loss in internal efficiency which occurs at light flows as obtained in tests.

LEGEND:

V_{an} = Annulus velocity in feet per second.

Q_c = Condenser flow in pounds per hour.

v = Saturated dry specific volume in cubic feet per pound corresponding to the actual exhaust pressure (when end point is in superheat region v = actual specific volume at end point).

A_{an} = Annulus area in square feet.

Y = Percent moisture at expansion line end point.

ELEP = Expansion line end point at actual exhaust pressure in BTU per pound

UEEP = Used energy end point in BTU per pound.

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GENERAL ELECTRIC

Fig. 4

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803200          EXHAUST LOSS VS. ANNULUS VELOCITY LP TURBINE
813200          165., 220., 260., 310., 340., 380., 445., 498., 550., 600.,
813201          668., 763., 918., 1190., 1308., 1400.
813210 0.       50., 36., 28., 22., 18., 14., 11., 9., 8.2, 8.5,
813211          10., 14., 22., 38., 44., 48.3
833200 32       EXUSLS, 280,     VEANP,  280
803300          EXHAUST LOSS VS. ANNULUS VELOCITY HP TURBINE
813300          165., 220., 260., 310., 340., 380., 445., 498., 550., 600.,
813301          668., 763., 918., 1190., 1308., 1400.
813310 0.       50., 36., 28., 22., 18., 14., 11., 9., 8.2, 8.5,
813311          10., 14., 22., 38., 44., 48.3
833300 33       EXUSLS, 290,    VEANP, 290
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FIGURE 5

EXHAUST LOSS SCHEDULE

These schedules allow PEPSE to determine the correct exhaust loss for whatever the annulus velocity is.

Other features of PEPSE that were used in determining the effects of exhaust losses included special option 3. This option fixes the generator output and calculates the required inlet flow. This option was used so that curves could be developed at a set generator output to show a comparison between circulating water inlet temperature and heat rate. Another PEPSE feature that was used was the save case option. The save case option allowed PEPSE runs over a fifty-degree circulating water temperature range with a minimum number of iterations. This saved considerable computer time.

PEPSE RESULTS AND TEST DATA COMPARISON

PEPSE runs were made at five different generator load points. The load points for this study were at 200, 300, 400, 500 and 530 gross megawatts. At each individual load point, runs were made over a range of circulating water inlet temperatures starting at fifty-degrees Fahrenheit and going up to ninety-five. Once these runs were completed, curves were developed showing the net unit heat rate versus the circulating water inlet temperature. Figure 6 shows one of the curves developed using the PEPSE output. The curve on Figure 6 indicates that once the circulating water inlet temperature drops to a certain point there is no further reduction of heat rate. The lower the circulating water temperature gets, the higher the annulus velocities become due to lower exhaust pressures. The higher annulus velocities increase the exhaust losses which eliminate any further reduction in heat rate once a certain point is reached. As circulating water temperature increases, turnup exhaust losses are encountered, thus increasing heat rate.

At some point, where the curve begins to flatten, there is a breakeven point. This breakeven point is where the gain in heat rate caused by higher circulating water temperatures is equal to the reduction in heat rate caused by lower auxiliary power usage with fewer cooling tower fans on. Since ambient conditions greatly effect cooling tower performance, the breakeven point varies. To establish a constant optimum point of operation, a value of 10 BTU/KWH above the minimum heat rate obtainable was selected. This value was selected due to its proximity of the range where the optimum points would be for different ambient conditions.

Once the optimum operating points were established at each load point, a test was conducted on the unit to indicate whether or not the PEPSE results could be validated. The set up for the test consisted of putting the boiler on manual and the turbine on standby. The procedure for the test was to shut off one cooling tower fan at a time and monitor circulating water temperature, load and exhaust pressure. With the throttle valves on the turbine fixed and boiler conditions constant, generator output would remain constant until backpressure on the unit started to raise. When the generator output started dropping off, the annulus velocity of the turbine would have moved below the optimum annulus velocity point on the exhaust loss curve and would be in the turnup exhaust loss region. Table 2 lists some of the test data taken. From the PEPSE output, the optimum operating point for 530 gross megawatts is eighty degrees Fahrenheit circulating water inlet temperature. The data illustrated in Table 2 indicates the optimum operating point to be at eighty-two degrees. The difference in optimum points is possibly due to the 10 BTU/KWH above minimum approximation point.

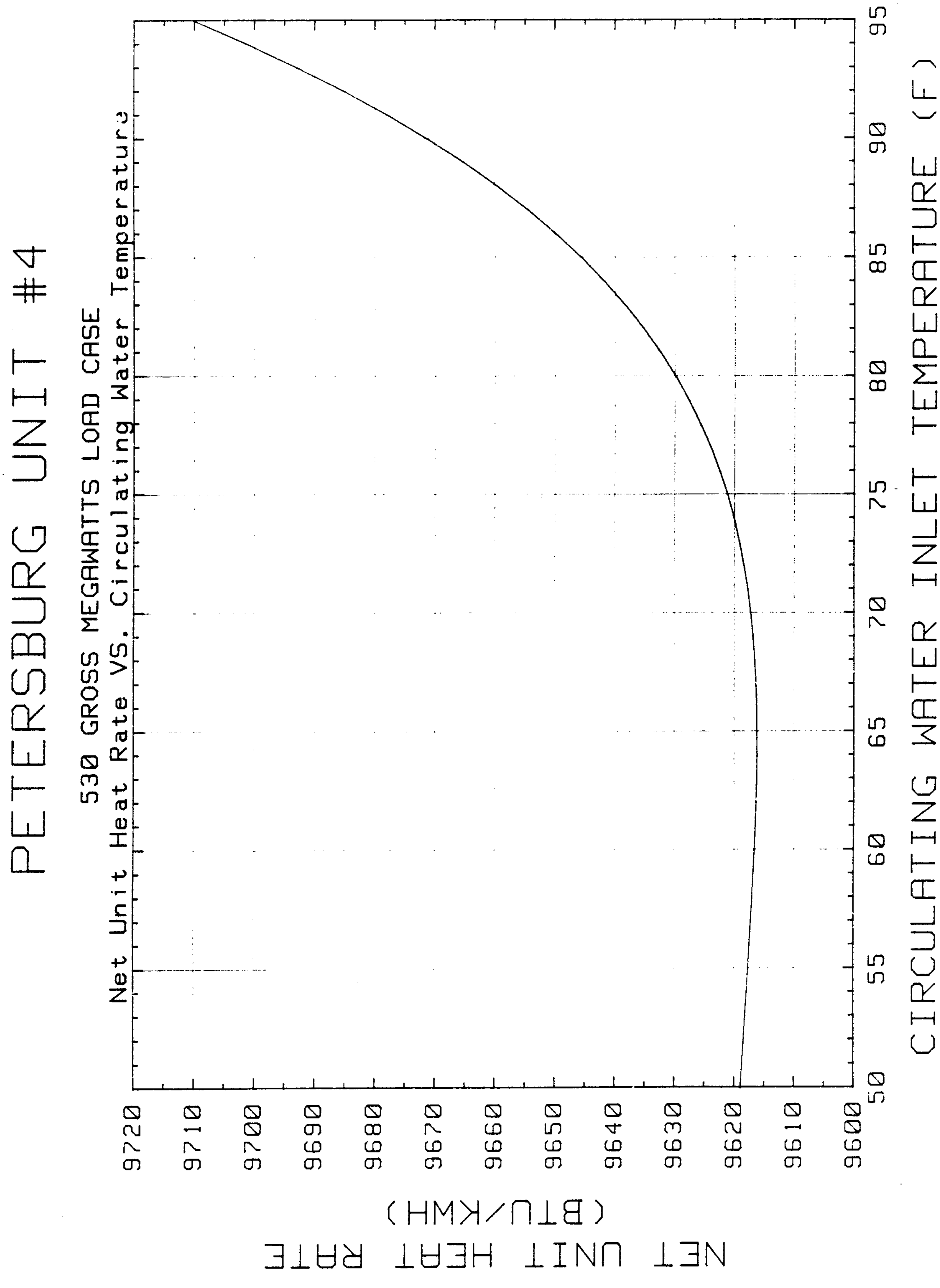
PETERSBURG UNIT #4
530 GROSS MEGAWATTS LOAD CASE
Net Unit Heat Rate VS. Circulating Water Temperature
NET UNIT HEAT RATE
(BTU/KWH)
9720
9710
9700
9690
9680
9670
9660
9650
9640
9630
9620
9610
9600
CIRCULATING WATER INLET TEMPERATURE (F)
50
55
60
65
70
75
80
85
90
95

Fig. 6

Table 2
Petersburg Unit #4 Exhaust Loss Test Data

MONITORED PARAMETER	Number Of Cooling Tower Fans On				
	13	12	11	10	9
Circ. Water Inlet Temp. (F)	77.2	79.4	80.3	82.0	84.9
L.P. Backpressure (In of Hg)	1.48	1.52	1.54	1.59	1.67
H.P. Backpressure (In of Hg)	2.08	2.17	2.19	2.23	2.31
Generator Load (MWg)	530.1	530.1	529.9	530.1	527.8

Note: Ambient Air Temperature 60F

IMPLEMENTING THE PEPSE RESULTS

The optimum operating points were determined for the five load cases that were used in the PEPSE study. From these five points, a curve was developed indicating the required circulating water inlet temperature over the operating load range. Figure 7 shows the curve that was developed for Petersburg Unit 4. The circulating water temperatures that are shown in Figure 7 are to produce the needed exhaust pressures for minimizing exhaust losses. However, there are a number of parameters that can affect condenser shell pressure other than circulating water temperature. With this in mind, a second curve was developed showing the backpressures at the optimum operating points. Figure 8 shows this curve. Due to the effects that ambient air temperatures have on cooling tower performance, it was determined that the recommended circulating water temperatures and condenser backpressures would only be useful during cooler periods of the year. Primarily during the spring, fall and winter is when the ambient air temperatures would be low enough to control the circulating water temperature for minimizing exhaust losses. If the ambient air temperature drops below thirty-four degrees, the operators are to maintain a minimum circulating water temperature of seventy-five degrees. This is to avoid icing up the cooling tower.

PETERSBURG UNIT #4

CIRCULATING WATER INLET TEMPERATURE FOR OPTIMUM HEAT RATE OPERATION

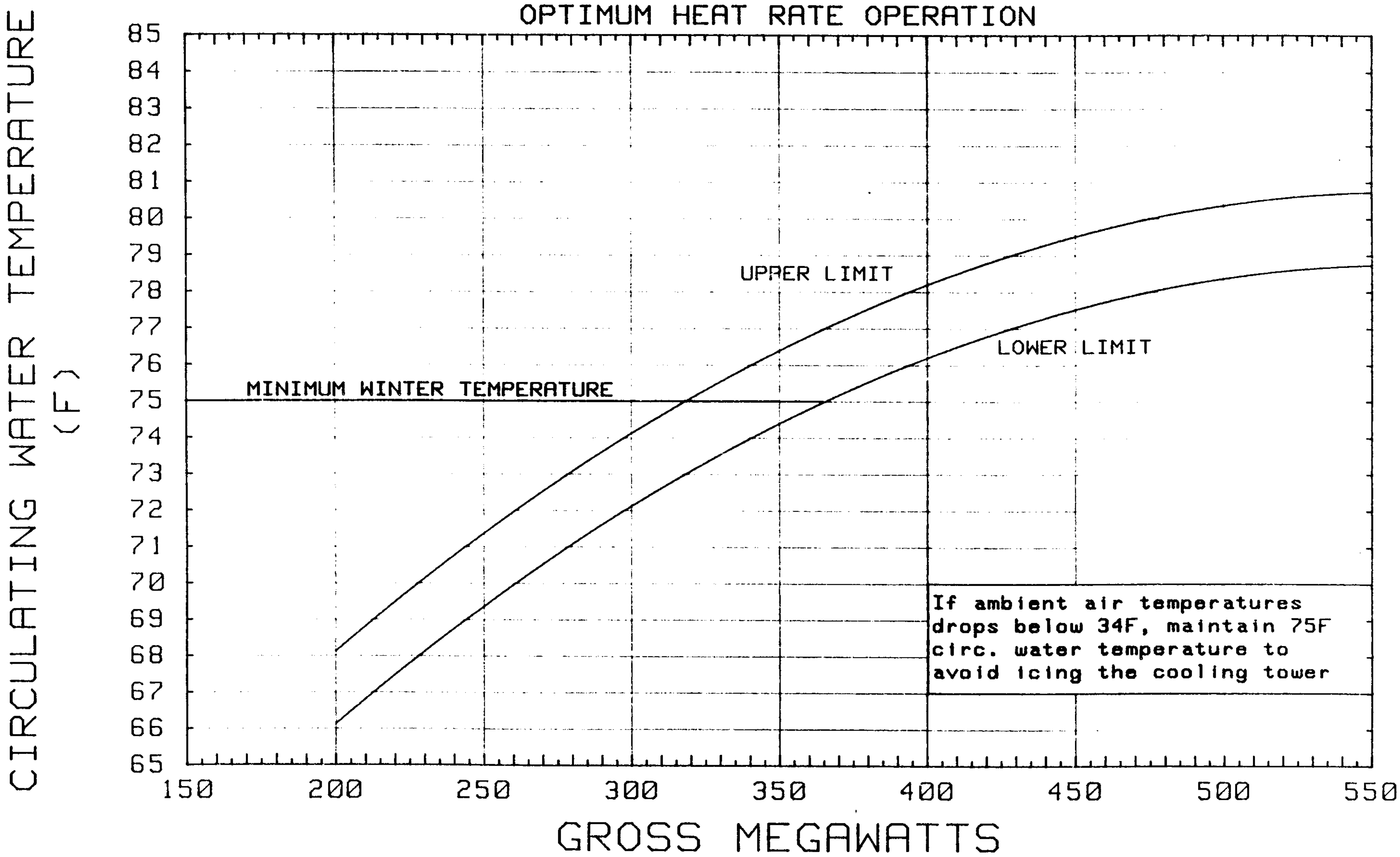

Fig. 7

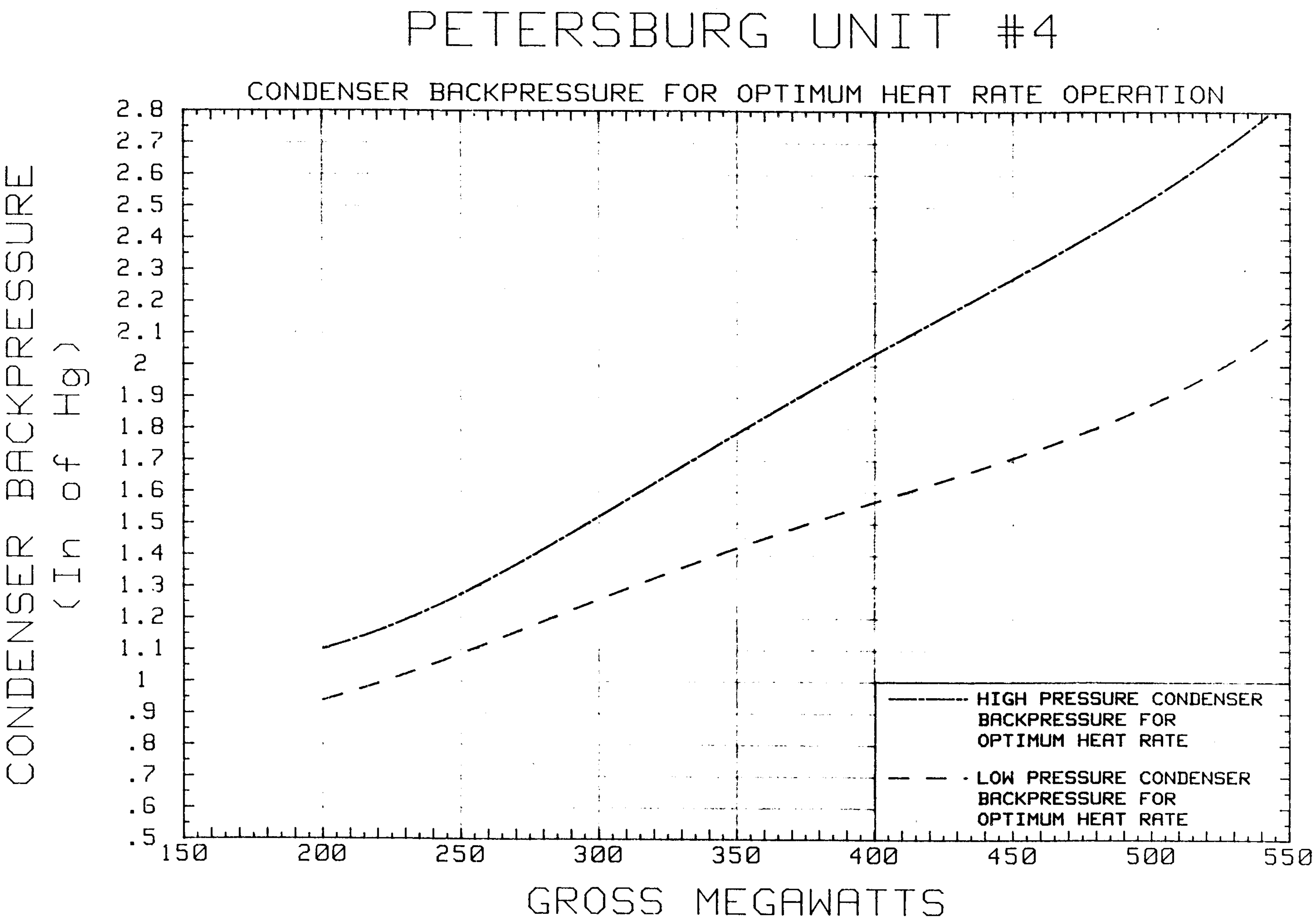
PETERSBURG UNIT #4
CONDENSER BACKPRESSURE FOR OPTIMUM HEAT RATE OPERATION
CONDENSER BACKPRESSURE (In of Hg)
2.8
2.7
2.6
2.5
2.4
2.3
2.2
2.1
2
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1
.9
.8
.7
.6
.5
150
200
250
300
350
400
450
500
550
GROSS MEGAWATTS
HIGH PRESSURE CONDENSER BACKPRESSURE FOR OPTIMUM HEAT RATE
LOW PRESSURE CONDENSER BACKPRESSURE FOR OPTIMUM HEAT RATE

Fig. 8

Operators have never been given any direction on operating the cooling tower other than keeping the backpressure as low as possible. With the PEPSE results on the effects of exhaust losses and the optimum operating points curve over the entire load range, Petersburg Unit No. 4 control board operators now know where they should be maintaining exhaust pressures. By operating along these curves, operators will be lowering heat rate by reducing exhaust losses and by lowering auxiliary power when cooling tower fans are shut off. The warmer circulating water temperatures will also help in preventing icing of the cooling tower during winter months.

SUMMARY

Condenser operation on Petersburg Unit No. 4 had improved to a point where exhaust pressures were operating well below design. This led to PEPSE studies being conducted to determine what affects the lower exhaust pressures were having on unit performance. It was found from these studies that any improvement in unit performance was negated by increased exhaust losses. Minimizing the exhaust losses can be accomplished by controlling the circulating water temperature to the condenser through cooling tower operational changes. Curves on the required circulating water temperature and condenser backpressures were generated from results of the PEPSE studies. These curves were developed to guide the operator on the proper operation of the cooling tower to improve unit performance. Tests were conducted on Unit No. 4 to confirm if the PEPSE based curves could be validated with actual unit data. The tests did confirm the PEPSE results and operators were instructed to use the PEPSE curves for operating the cooling tower.

REFERENCES

Salisbury, J. Kenneth (1950) Steam Turbines and Their Cycles. John Wiley & Sons. New York, N.Y.

Bartlett, Robert L. Steam Turbine Performance & Economics. Mc-Graw Hill

General Physics (1984) Fundamentals of Power Plant Performance for Utility Engineers. GP Courseware

Volume II PEPSE Manual: Engineering Model Description. W. Kettenacker, G. Minner, E. Hansen, P. Klink. Energy Incorporated 1988 Revision