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ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
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Master

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PORST:
A COMPUTER CODE TO ANALYZE THE
PERFORMANCE OF RETROFITTED STEAM TURBINES

by
C. Lee and I.T. Hwang
Energy and Environmental Systems Division

September 1980

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by

C. Lee and I.T. Hwang

ABSTRACT

The computer code PORST was developed to analyze the performance of a retrofitted steam turbine that is converted from a single generating to a cogenerating unit for purposes of district heating. Two retrofit schemes are considered: one converts a condensing turbine to a back-pressure unit; the other allows the crossover extraction of steam between turbine cylinders. The code can analyze the performance of a turbine operating at: (1) valve-wide-open condition before retrofit, (2) partial load before retrofit, (3) valve-wide-open after retrofit, and (4) partial load after retrofit.

1 INTRODUCTION

The computer code PORST was developed at Argonne National Laboratory to analyze the performance of a retrofitted steam turbine that is converted from an electricity-generating, single-purpose unit to an electrical/thermal cogenerating unit. Thermal energy generated will be used for purposes of district heating. The district heating services may include space heating, process heat, domestic water heating, and (through conversion devices) space cooling and refrigeration.

The code considers two categories of retrofit schemes. In the first category, a condensing turbine is converted to a back-pressure turbine so that the exhaust steam is of higher grade and can be utilized. Such a modification would involve the removal, through machining, of both turbine wheels and diaphragms in the condensing end of the machine; this results in a permanent reduction of electrical generating capacity. Although this type of conversion allows efficient use of the heat content of exhaust steam, it does not have flexibility in following a wide variation in the electrical-to-thermal load ratio. This is because the ratio of thermal-to-electrical outputs of a back-pressure turbine stays approximately constant.

The second category of retrofit schemes provides a passage in the crossover line, especially between the intermediate and low-pressure turbines, to extract steam for use. The crossover extraction retrofit scheme permits higher degrees of operational flexibility and can avoid a permanent loss of electrical generating capacity.

The PORST code analyzes the performance of a steam turbine at:

- (1) valve-wide-open condition before retrofit;
- (2) partial-load operation before retrofit;
- (3) valve-wide-open operation after retrofit; and
- (4) partial-load operation after retrofit.

For the condensing-to-back-pressure conversion, the back pressure of exhaust steam after retrofit should be specified according to the condition of steam required for district heating. The relation between electrical and thermal outputs with respect to input heat added to the throttle steam then is determined. For crossover extraction retrofit, the upper limit of the thermal energy generation is dictated by a minimum condensing flow requirement for the low-pressure (LP) turbine. This minimum flow of steam (~ 10% of valve-wide-open throttle flow) is required to avoid overheat of the LP stages and should be maintained at any permitted level of turbine operation. In general, if both condensing flow and throttle flow are specified, the code will calculate the extraction steam flow available for district heating and the electrical output for that particular mode of operation.

The methodology used for the analysis and the capability of the code is discussed in Sec. 2, which includes flow charts that show the program logic. Section 3 describes: (1) the input data requirement and instruction and (2) outputs of the program. Section 4 describes some modifications and improvements that will be made in the next fiscal year, and Sec. 5 presents a sample program that illustrates the utilization of the code on a complicated and sophisticated steam turbine. An appendix is included to show the formulas used in the analysis.

2 DESCRIPTION OF THE PORST CODE

2.1 GENERAL DESCRIPTION

Because the turbine considered for retrofit is an existing unit, the proprietor should possess a vendor-supplied heat balance diagram at the valve-wide-open operating condition. This diagram serves as a basis for the analysis. In general, the heat balance diagram provides pre-retrofit data on:

- (1) flowrates and conditions of throttle, extraction, leakage, and feedwater flow streams;
- (2) electric power output; and
- (3) heat rate at design condition.

Before proceeding to the off-design analysis and the retrofit cases, the following items obtained from the heat balance diagram are entered into the code:

- (1) the throttle flowrate and flow conditions;
- (2) extraction conditions;
- (3) leakage flowrates and conditions; and
- (4) heater characteristics, such as terminal temperature differences and/or drain cooler approach.

From these inputs, the code calculates:

- (1) extraction flowrates;
- (2) electrical output; and
- (3) heat input.

The purpose of this analysis is to verify that the input setup is sound. The outputs are compared to those listed on the manufacturer's heat balance diagram. Causes of any significant inconsistency should be found and measures taken to correct any error before proceeding to the next step. This preliminary run, if valid, will also yield parameters that are required for the analysis to be followed. The parameters include packing constants for steam leakages from valve stems and shaft, first-stage efficiency, and slope of the expansion line on a Mollier diagram at the design point. Design point characteristics would be used to obtain these off-design conditions.

2.2 DESIGN POINT ANALYSIS BEFORE RETROFIT

For the design point analysis, data of the following categories are required from the vendor's heat balance diagram at valve-wide-open conditions:

- (1) throttle steam flowrate and conditions;
- (2) governing stage condition and characteristics;

- (3) condenser condition;
- (4) feedwater extraction steam flow conditions;
- (5) stem and shaft leakage, steam flowrates and conditions;
- (6) feedwater heater characteristics;
- (7) reheat conditions, if any;
- (8) auxiliary turbine characteristics and extraction steam condition; and
- (9) exhaust, mechanical, and generation energy loss data.

The design point calculation, i.e., that of electric output and steam extraction flowrates, is straight-forward. Figure 2.1 shows the flow diagram of the design point analysis. The flowrate of steam extracted for feedwater heating can be obtained by performing a heat balance on each feedwater heater. The extraction flow for the auxiliary turbine (to drive the feedwater pump) is calculated based on the feedwater flow, extraction condition, and power requirements of the pump. Based on the extraction flowrates and corresponding enthalpy drops, the mechanical shaft power generated can be obtained. The gross output then is adjusted by the exhaust loss of the last stage of turbine and the generator loss (including mechanical loss) of the electrical generators. The generator loss caused by windage, core loss, I^2R losses, and mechanical bearing losses is expressed as a function of kVA output of the generator. Exhaust loss for a steam turbine occurs in two forms:

- (1) the loss of kinetic energy of steam leaving the last stage and not recoverable; and
- (2) the hood loss resulting from the pressure drop of steam passing through the exhaust hood.

Heat requirements for the cycle are determined by multiplying throttle flow and enthalpy difference between throttle steam and boiler feedwater and -- if there is reheat -- adding the reheat input. Fuel consumption is the heat to the cycle divided by the boiler efficiency.

This run will yield packing constants for leakages, governing stage efficiency, and the slope of the expansion curve on a Mollier diagram at the design point required for partial-load analysis. Packing constants obtained at the design load will be used to find the leakage flows at partial loads according to the prevailing pressures and specific volumes. The governing stage efficiency at the design point will be used to find efficiency at partial-load operation. This calculation is required if the measured variation of governing stage efficiency vs. the ratio of bucket velocity to isentropic expansion velocity is not available. The expansion lines on the Mollier diagram for partial loads are paralled to the design point expansion line. This characteristic enables the enthalpies to be defined at various points in the course of steam expansion.

Fig. 2.1 Flow Diagram at the Design Point

2.3 PARTIAL-LOAD ANALYSIS BEFORE RETROFIT

The analysis of partial-load performance of a steam turbine in the PORST code uses the parameters calculated in the design point analysis. These parameters include:

- (1) packing constants for leakage flow calculation;
- (2) governing stage efficiency to be adjusted for partial-load use; and
- (3) slope of the expansion line on the Mollier diagram, which is the same for partial-load operation as for design-point operation.

In the present version of the code, a governing stage (first stage) shell pressure (smaller than that at the design point) should be specified first. This value is then used to obtain the available energy and efficiency of the governing stage, based on those at valve-wide-open conditions. Enthalpy of steam at the first-stage shell is then calculated, followed by calculation of the throttle steam flow. The next step is to determine the flowrate and flow conditions of extracted steam and leak steam. This calculation involves an iteration process that first assumes that flowrate ratios of all pertinent steam flows at partial load with respect to those at the design point are the same as the throttle flowrate ratio. The iteration process involves:

- (1) obtaining flow conditions such as pressure, temperature, enthalpy, and entropy according to the flowrates assumed -- or obtained from a previous round of iteration;
- (2) performing a heat balance on each feedwater heater to obtain new extraction flowrates and using packing constraints to calculate new leakage flowrates;
- (3) establishing new flow conditions; and
- (4) comparing new flow conditions with those from the previous round.

This process continues until flow condition differences are within 0.1% of the previous round.

After all the flowrates and flow conditions are established, the electricity output can be obtained by following the same procedures in the design point analysis. The flow diagram of the partial-load analysis is shown in Fig. 2.2. The formula used in this analysis are presented in the Appendix.

2.4 ANALYSIS AFTER TURBINE RETROFIT

The analysis for the performance of a steam turbine after retrofit is similar to that for partial-load analysis before retrofit, with the addition of the following considerations: for a condensing-to-back-pressure conversion, the pressure of exhaust steam at the valve-wide-open condition after retrofit

Fig. 2.2 Flow Diagram for Partial Load Analysis

should be specified, and the condensing pressure in the before-retrofit analysis would be replaced by this back pressure for after-retrofit analysis. Depending on the district heating requirement and design, this back pressure may be changed at partial-load performance in two cases:

- (1) steam is not condensed in a heat exchanger but is piped to the consuming site, and
- (2) a hot water exchanger is used to condense exhaust steam for district heating, and the inlet temperature of district heating hot water cannot be constant.

For the first case, the exhaust steam pressure would be dictated by the throttle flow. In the second, it is determined by the inlet temperature of the district heating hot water.

For crossover extraction retrofit, the desired extraction flowrate at the crossover point may be specified. This extraction rate should follow the minimum flow requirement to the low-pressure section of the unit. If the desired extraction is not given, the code would calculate the maximum flow available.

Currently, the coding of this analysis is not yet completed; it will be continued in fiscal year 1981.

3 INPUT DATA REQUIREMENTS AND DESCRIPTION OF OUTPUT

3.1 INPUT DATA REQUIREMENTS

The input data requirements include two main categories: the turbine cycle characteristics and the feedwater configuration. Those used for this report are listed in Table 3.1 in the order in which they are entered. Definitions of input variables are shown in Table 3.2.

3.2 DESCRIPTION OF OUTPUT

Outputs of the code provide reports on the performance of the turbine, feedwater heater, and condenser. The report on the turbine includes gross and net electricity outputs and heat rates. The feedwater performance summary provides flowrates and flow conditions of streams taking part in heat exchange and the heat transferred. Outputs for the condenser include exhaust flowrate and condensate, condensate characteristics, and heat transferred.

Table 3.1. Input Data Requirements

Card Number	Input	Format
1	NX, NLK, NXC, NRH, NXBR1, NXBR2 IDSGN, IPARL, IROW, NCL	20I4
2	ITYP(I), I=1, NX	20I4
3	IDRAN(I), I=1, NX	20I4
4	LD(I), I=1, NX	20I4
5	PTHU, HTHU, FTHU, PGSD, HGSD	8F10.0
6	PELEP, HELEP	8F12.2
7	PBP, PAP	8F10.0
8 ^a	PXI(I), I=1, NX	8F10.0
9 ^a	HXI(I), I=1, NX	8F10.0
10	KAUXI, KAUXO, HAUXID, PAUXID, PWAUX	2I2, 6X, 3F10.0
11	PFO(I), I=1, NX	8F10.0
12	APH(I), I=1, NX	8F10.0
13	TD(I), I=1, NX	8F10.0
14	DC(I), I=1, NX	8F10.0
15	BFP(I), I=1, NX	8F10.0
16	HBR1, HAR1, HBR2, HAR2, PBR1, PBR2	8F10.0
17	NPTS	20I4
18	ANVEL(I), I=1, NPTS	8F10.0
19	EXLOSS(I), I=1, NPTS	8F10.0
20	NP, PF	I4, 2F10.0
21	GROSS(I), I=1, NP	8F10.0
22	CLOSS(I), I=1, NP	8F10.0
23	I, J, KIK(I,J), XL(I, J), HXL(I, J)	3I2, 4X, 2F10.0
24	PGSP, WRPM, DPCH, END, ANAREA	8F10.0
25	BFNX, BFPEFF, BFT	8F10.0

^a Card #8 and 9 should be replaced by the following 3 cards, if IDSGN=1,

XSSP(I), I=1, NX	8F10.4
PXC(I), I=1, NXC	8F10.0
HXC(I), I=1, NXC	8F10.0

Table 3.2. Definition of Input Variable

Variable Name	Definition	Comment
NX	Number of feedwater heaters	Max. = 20
NLK	Number of leakages	
NXC	Number of points for expansion line specification	
NRH	Number of reheaters	Max. = 2
NXBR1	Number of heaters before reheater 1	
NXBR2	Number of heaters before reheater 2	
IDSGN	Extraction condition specification	1-if relation between extraction pressure and enthalpy w.r.t. flow to the following stage is known 0-if enthalpies and pressures of extractions are known
IPARL	Flag for the extent of analysis	1-partial load analysis desired 0-design load analysis only
IROW	Number of rows of nozzle-buckets system for governing stage	
NCL	Number of leakages to condenser	
ITYP (I)	Type of I th feedwater heater	0-for gland steam condenser 1-contact heater 2-flash heater 3-drain cooler heater 4-pump heater 5-water to water heater negative, if heater is related to auxiliary turbine. Only one auxiliary turbine is allowed.
IDRAN (I)	Heater number from which cascade condensate exits and enters feedwater heater I	N-cascade from N th upstream heater 0-no cascade
LD (I)	Heater number where drain of I th heater goes	0-for gland steam condenser or water-water heater N-drain of I th heater entering N th downstream heater

Table 3.2 Definition of Input Variable (Cont'd.)

Variable Name	Definition	Comment
HTHD	Throttle enthalpy of steam at valve-wide-open condition	Btu/lb
FTHD	Throttle flow at valve-wide-open	lb/hr
PGSD	First stage shell pressure at valve-wide-open	psia
HGSD	First stage shell enthalpy at valve-wide-open	Btu/lb
PELEP	Pressure of expansion line end point at valve-wide-open	psia
HELEP	Enthalpy of expansion line end point at valve-wide-open	Btu/lb
PBP	Pressure before auxiliary turbine driven boiler feed pump	psia
PAP	Pressure after auxiliary turbine driven boiler feed pump	psia
PXI(I)	Pressure of extraction for I th heater	psia, leave blank for gland steam condenser and water-water heater
HXI(I)	Enthalpy of extraction for I th heater	Btu/lb; for gland steam condenser, enter condenser exit enthalpy
KAUXI	Identification of source of steam for auxiliary turbine	1-if steam comes from front of reheater 1 2-if steam comes between reheater 1 and 2 3-if steam comes from behind reheater 2
KAUXO	Identification of destination of steam for auxiliary turbine	Same as KAUXI
HAUXID	Enthalpy of steam inlet condition for auxiliary turbine at design point	Btu/lb
PAUXID	Pressure of steam inlet condition for auxiliary turbine at design point	psia

Table 3.2. Definition of Input Variable (Cont'd.)

Variable Name	Definition	Comment
TD(I)	Terminal difference for I^{th} heater	$^{\circ}\text{F}$
DC(I)	Drain cooler approach for I^{th} heater	$^{\circ}\text{F}$, 0 for heaters other than drain cooler heater
BFP(I)	Boiler feed pump enthalpy increase after I^{th} heater	Btu/lb
HBR1	Enthalpy of steam before re-heater 1	Btu/lb
HAR1	Enthalpy of steam after re-heater 1	Btu/lb
HBR2	Enthalpy of steam before re-heater 2	Btu/lb
HAR2	Enthalpy of steam after reheater 2	Btu/lb
PBR1	Pressure of steam before reheater 1	psia
PBR2	Pressure of steam before reheater 2	psia
NPTS	Number of points with which the exhaust losses will be defined w.r.t. last stage annulus velocity	
ANVEL(I)	The I^{th} annulus velocity where exhaust loss is defined	ft/sec
EXLOSS(I)	Exhaust loss for I^{th} annulus velocity	
NP	Number of points with which the generator loss curve w.r.t. generator output will be defined	Btu/lb
PF	Power factor	
GROSS(I)	The I^{th} gross output where generator loss is defined	kVa
GLOSS(I)	Generator loss for I^{th} gross output	kW
I	Heater number where leakages enter	$\text{NX}+1$ for leakages going to condenser
J	Index of leakages to the same heater	$J=1, 2, \dots, N$, where N is the number of leakages entering heater I
KLK(I,J)	Identification of source of leakage for J^{th} leakage entering I^{th} heater	1-if leakage before re-heater 1 2-if leakage between re-heater 1 and 2 3-if leakage after re-heater 2

Table 3.2. Definition of Input Variable (Cont'd.)

Variable Name	Definition	Comment
XL(I,J)	Mass flow rate for J th leakage entering I th heater	lb/hr
HXL(I,J)	Enthalpy for J th leakage entering I th heater	Btu/lb
PGSP	Partial load governing stage pressure	psia
WRPM	Shaft rpm	revolutions per minute
DPCH	Pitch diameter of first stage	in
END	Number of exhaust ends for steam	
ANAREA	Annulus area per end	ft ²
BFNX	Enthalpy increase of feedwater due to hot well pump (condensate pump)	Btu/lb
BFPEFF	Efficiency of hot well pump	
BFT	Temperature change caused by hot well pump	°F
PTHD	Throttle pressure of steam at valve-wide-open condition	psia
PWAUX	Power required to drive auxiliary turbine	hp
PFO(I)	Pressure of feedwater at exit of I th heater	psia
APH(I)	Extra heat input or output to I th heater	Positive for heat output; negative for heat input, Btu/hr

4 MODIFICATIONS

Certain modifications could improve the capability of the code and fine-tune the analysis. In the current version of the code, it is assumed that the throttle steam is at the superheated state. However, steam turbines used in most of today's nuclear power stations admit steam at the saturation state. During the expansion process, excessive moisture developed should be removed to avoid severe erosion. This can be done by adjusting the enthalpy of steam after moisture removal according to the moisture content and the effectiveness of the moisture-removal device. Also, pressure drops through steam and hot water piping are assumed to be zero. This assumption can be modified.

The off-design analysis is based on a specification of first-stage shell pressure. The specification is cumbersome and can be altered to allow a specification of throttle flow to the turbine. This alteration involves a minor iteration to find the proper first-stage shell pressure.

The auxiliary turbine analysis needs some additional work to avoid duplication. The current input data are printed in the output without any narrative explanation. This should be modified to make the turbine better understood.

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5 SAMPLE PROBLEMS

The turbine used in the sample problem is a General Electric unit operated at the Philadelphia Electric Company's Eddystone Station. The unit, rated at 385 MW, has a supercritical pressure (SP) turbine, a high-pressure (HP) turbine; two intermediate-pressure (IP) turbines, and two low-pressure (LP) turbines. Each LP turbine has three exhaust ends, i.e., the whole turbine has six exhaust ends. There is one gland steam condenser and eight feedwater heaters, among which seven are drain cooler heaters and one is a contact heater (deaerator). The vendor's heat balance diagram is shown in Fig. 5.1. The input data are shown in Table 5.1. Tables 5.2 - 5.7 present the outputs of the sample problem. The results calculated by using PORST for the design point have excellent agreement with the data provided in the vendor's heat balance diagram.

Table 5.1 Sample Problem Input Data

10	27	1	2	1	2	0	1	1	1
3	3	-3	3	1	2	3	2	0	5
0	1	1	1	1	0	1	1	0	2
1	1	1	1	0	1	1	2	0	0
3500.	1461.0		2178000.	0.		0.0		1439.7	
0.491	1018.5								
2850.	4470.								
1024.3	525.9	331.5	174.5	77.	36.9	14.9	5.16		
0.0	0.0								
1338.3	1467.3	1261.	1481.2	1383.7	1308.	1230.	1150.9		
396.7	131.5								
1	2	1338.3	1045.0	7800.0					
2240.	2240.	2240.	2240.	340.	340.	340.	340.		
340.	340.	340.							
0.0	0.0	0.0	0.0	0.0	0.0	0.0	33500000.		
0.0	0.0								
-5.0	-5.0	0.0	-5.0	0.0	0.0	0.0	3.0		
0.0	0.0								
10.0	10.0	10.0	10.0	0.0	0.0	10.0	0.0		
0.0	10.0								
4.4	0.0	0.0	0.0	8.1	0.0	0.0	0.0		
0.0	0.0	0.0							
1338.9	1534.8	1397.4	1551.2	940.6	314.				
21									
128.	150.	175.	200.	250.	300.	350.	400.		
450.	500.	550.	600.	650.	700.	800.	900.		
1000.	1100.	1200.	1300.	1400.					
64.09	56.15	47.58	40.55	29.75	22.48	17.28	13.55		
10.82	8.85	8.30	8.5	9.5	11.25	15.82	21.13		
26.82	32.82	38.5	43.8	48.3					
11	1.0								
0.0	50000.	100000.	150000.	200000.	250000.	300000.	350000.		
400000.	450000.	500000.							
1850.	2000.	2200.	2550.	2950.	3500.	4150.	4400.		
4470.	4520.	4540.							
5	1	3	3017.	1419.8					
5	2	3	4748.	1507.7					
5	3	1	8102.	1336.5					
5	4	1	10498.	1336.5					
5	5	2	4400.	1410.8					
5	6	2	5336.	1410.8					
8	1	1	421.3	1461.					
8	2	1	980.0	1336.5					
8	3	1	754.	1336.5					
8	4	3	1227.7	1419.8					
8	5	2	797.1	1410.8					
8	6	2	657.1	1410.8					
8	7	3	1227.7	1419.8					
9	1	1	215.4	1461.					
9	2	1	501.	1336.5					
9	3	1	385.5	1336.5					
9	4	3	627.6	1419.8					
9	5	2	407.5	1410.8					
9	6	2	335.9	1410.8					
9	7	3	627.6	1419.8					
11	1	1	215.4	1461.0					
11	2	1	501.	1336.5					
11	3	1	385.5	1336.5					
11	4	3	627.6	1419.8					
11	5	2	407.5	1410.8					
11	6	2	335.9	1410.8					
11	7	3	627.6	1419.8					
1385.6579	3600.	65.5.	6.0	41.1					
1.0	.70	0.8							

Table 5.2 Sample Program Output: Design Point, Condenser Performance Summary

ENTHALPY AT EXPANSION LINE END POINT	BTU/LB	1018.5000
PRESSURE AT EXPANSION LINE END POINT	PSIA	0.4910
CONDENSER ENTRANCE FLOW	1000 LB/HR	1305.9128
CONDENSATE FLOW	1000 LB/HR	1586.6337
TEMPERATURE	DEG F	79.0300
ENTHALPY OF CONDENSATE	BTU/LB	47.0684
HEAT TRANSFERRED IN CONDENSER	10**6 BTU/HR	1272.6818

Table 5.3 Sample Program Output: Design Point, Turbine Performance Summary

GROSS INTERNAL GENERATOR OUTPUT	KW	377462.3304
EXHAUST LOSS	KW	7862.9378
GENERATOR LOSS	KW	4421.0553
AUXILIARY TURBINE POWER	KW	4559.9437
NET INTERNAL GENERATOR OUTPUT	KW	360618.4436
FLOW TO AUXILIARY TURBINE	1000 LB/HR	201.3189
TERMINAL HEAT RATE	BTU/KW-HR	6833.9277
NET HEAT RATE	BTU/KW-HR	7070.3694

Table 5.4 Sample Program Output; Design Point. Feedwater Heater Performance Summary

HEATER NO.	TERM. DIFF. DEG F	HEAT TRANSFERRED 10**6 BTU/HR	PRESSURE PSIA	FEEDWATER		TEMP. IN DEG F	TEMP. OUT DEG F	ENTHALPY IN BTU/LB	ENTHALPY OUT BTU/LB
				ENTRANCE FLOW 1000 LB/HR	EXIT FLOW 1000 LB/HR				
1	-5.0	193.12626	2240.0000	2178.0000	2178.0000	477.2403	552.5021	461.6196	550.2910
2	-5.0	121.26939	2240.0000	2178.0000	2178.0000	426.6043	477.2403	405.9404	461.6196
3	0.0	118.13084	2240.0000	2178.0000	2178.0000	375.5368	426.6043	351.7021	405.9404
4	-5.0	139.76845	2240.0000	2178.0000	2178.0000	313.4307	375.5368	237.5293	351.7021
5	0.0	75.51845	340.0000	1586.6337	2178.0000	262.4182	309.4112	231.8327	279.4293
6	0.0	79.71592	340.0000	1586.6337	1586.6337	212.6975	262.4182	181.5905	231.8327
7	0.0	82.90400	340.0000	1586.6337	1586.6337	160.5730	212.6975	129.3390	181.5905
8	3.0	105.50350	340.0000	1586.6337	1586.6337	93.9263	160.5730	62.8406	129.3390
9	0.0	3.10046	340.0000	1586.6337	1586.6337	91.9654	93.9263	60.8865	62.8406
10	0.0	20.33775	340.0000	1586.6337	1586.6337	79.8300	91.9654	48.0684	60.8865

HEATER NO.	EXTRACTION FLOW 1000 LB/HR	LEAKAGE FLOW 1000 LB/HR	PRESSURE PSIA	TEMPERATURE DEG F	EXTRACTION ENTHALPY BTU/LB	LEAKAGE ENTHALPY BTU/LB
1	223.1668	0.0	1024.3000	720.8493	1338.3000	0.0
2	103.0358	0.0	525.9000	903.0744	1467.3000	0.0
3	110.8700	0.0	331.5000	511.4533	1261.0000	0.0
4	93.6217	0.0	174.5000	906.4139	1481.2000	0.0
5	16.8051	43.8660	77.0000	706.1506	1383.7000	1401.4451
6	74.0317	0.0	36.9000	546.8554	1302.0000	0.0
7	69.6718	0.0	14.9000	379.0490	1230.0000	0.0
8	127.8520	6.0649	5.1600	205.1525	1150.9000	1396.6879
9	0.0	3.1005	308.1796	0.0	0.0	1396.6871
10	0.0	0.0	0.0	0.0	0.0	0.0

Table 5.4 Sample Program Output: Design Point, Feedwater Heater Performance Summary (Cont'd)

<u>DRAIN</u>						
HEATER NO.		FLOW 1000 LB/HR		TEMPERATURE DEG F	ENTHALPY BTU/LB	
1		223.1658		487.2403	472.9102	
2		326.2036		436.6043	415.2448	
3		437.0735		385.5368	359.5059	
4		530.6952		323.4807	294.0191	
5		0.0		309.4112	279.4253	
6		74.0317		262.4182	231.2193	
7		143.7035		170.5730	138.5482	
8		277.6204		163.5730	131.5356	
9		3.1005		419.8206	396.7000	
10		277.6204		89.8300	57.8485	

<u>CASCADE</u>						
HEATER NO.	SOURCE HEATER	FLOW 1000 LB/HR	PRESSURE PSIA	TEMPERATURE DEG F	ENTHALPY BTU/LB	HEAT TRANSFERRED 10**6 BTU/HR
1	0	0.0	0.0	0.0	0.0	0.0
2	1	223.1668	1024.3000	487.2403	472.9102	12.8690
3	2	326.2036	525.9000	436.6043	415.2448	18.1822
4	3	437.0735	331.5000	385.5368	359.5059	28.6225
5	4	530.6952	174.5000	323.4807	294.0191	7.7427
6	0	0.0	0.0	0.0	0.0	0.0
7	6	74.0317	36.9000	262.4182	231.2193	6.8606
8	7	143.7035	14.9000	170.5730	138.5482	1.0077
9	0	0.0	0.0	0.0	0.0	0.0
10	8	277.6204	5.1600	163.5730	131.5356	20.4570

Table 5.5 Sample Program Output: Partial Performance, Condenser Performance Summary

ENTHALPY AT EXPANSION LINE END POINT	BTU/LB	1018.5000
PRESSURE AT EXPANSION LINE END POINT	PSIA	0.4910
CONDENSER ENTRANCE FLOW	1000 LB/HR	843.6227
CONDENSATE FLOW	1000 LB/HR	1010.3655
TEMPERATURE	DEG F	79.0300
ENTHALPY OF CONDENSATE	BTU/LB	47.0684
HEAT TRANSFERRED IN CONDENSER	10**6 BTU/HR	822.3699

Table 5.6 Sample Program Output: Partial Performance, Turbine Performance Summary

GROSS INTERNAL GENERATOR OUTPUT	KW	236259.4070
EXHAUST LOSS	KW	2075.7899
GENERATOR LOSS	KW	3297.8928
AUXILIARY TURBINE POWER	KW	2557.0025
NET INTERNAL GENERATOR OUTPUT	KW	228328.7218
FLOW TO AUXILIARY TURBINE	1000 LB/HR	112.8902
TERMINAL HEAT RATE	BTU/KW-HR	6964.3138
NET HEAT RATE	BTU/KW-HR	7193.0492

Table 5.7 Sample Program Output: Partial Performance, Feedwater Heater Performance Summary

<u>FEEDWATER</u>									
HEATER NO.	TERM. DIFF. DEG F	HEAT TRANSFERRED 10**6 BTU/HR	PRESSURE PSIA	ENTRANCE FLOW 1000 LB/HR	EXIT FLOW 1000 LB/HR	TEMP. IN DEG F	TEMP. OUT DEG F	ENTHALPY IN BTU/LB	ENTHALPY OUT BTU/LB
1	-5.0	90.30114	2240.0000	1306.8000	1306.8000	430.9866	493.2605	410.6736	479.7745
2	-5.0	63.93240	2240.0000	1306.8000	1306.8000	385.1061	439.9866	361.7507	410.6736
3	0.0	61.04690	2240.0000	1306.8000	1306.8000	340.2639	385.1061	315.0359	361.7507
4	-5.0	75.66915	2240.0000	1306.8000	1306.8000	283.6125	340.2639	257.1317	315.0359
5	0.0	43.34391	340.0000	1010.3655	1306.8000	237.0474	279.8629	206.1325	249.0317
6	0.0	46.52075	340.0000	1010.3655	1010.3655	191.2869	237.0474	160.0890	206.1325
7	0.0	49.01826	340.0000	1010.3655	1010.3655	142.7911	191.2869	111.5736	160.0890
8	3.0	51.94963	340.0000	1010.3655	1010.3655	91.2333	142.7911	60.1569	111.5736
9	0.0	3.10046	340.0000	1010.3655	1010.3655	88.1539	91.2333	57.0882	60.1569
10	0.0	9.11339	340.0000	1010.3655	1010.3655	79.8300	88.1539	48.0684	57.0882

<u>STEAM</u>						
HEATER NO.	EXTRACTION FLOW 1000 LB/HR	LEAKAGE FLOW 1000 LB/HR	PRESSURE PSIA	TEMPERATURE DEG F	EXTRACTION ENTHALPY BTU/LB	LEAKAGE ENTHALPY BTU/LB
1	100.5165	0.0	611.5597	545.7009	1318.5000	0.0
2	52.9344	0.0	329.3218	913.0099	1482.0060	0.0
3	56.6433	0.0	208.0226	503.9509	1270.3709	0.0
4	51.8506	0.0	110.7123	907.3790	1483.8633	0.0
5	6.5308	27.9589	49.0890	708.9214	1336.4705	1401.4497
6	42.0313	0.0	23.6638	550.8671	1310.9566	0.0
7	40.8703	0.0	9.5996	383.4461	1232.8637	0.0
8	76.7558	3.9344	3.3474	209.6937	1153.6775	1396.6831
9	0.0	3.1005	308.1796	0.0	0.0	1396.6871
10	0.0	0.0	0.0	0.0	0.0	0.0

Table 5.7 Sample Program Output: Partial Performance, Feedwater Heater Performance Summary (Cont'd)

DRAIN

HEATER NO.	FLOW 1000 LB/HR	TEMPERATURE DEG F	ENTHALPY BTU/LB
1	100.5165	440.9866	420.1285
2	153.4509	395.1061	369.8028
3	210.0943	350.2639	322.0352
4	261.9448	293.6125	263.1334
5	0.0	279.8629	249.0317
6	42.0813	237.0474	205.4593
7	82.9521	152.7911	120.7427
8	163.6423	145.7911	113.7404
9	3.1005	419.8206	396.7000
10	163.6423	89.8300	57.8485

CASCADE

HEATER NO.	SOURCE HEATER	FLOW 1000 LB/HR	PRESSURE PSIA	TEMPERATURE DEG F	ENTHALPY BTU/LB	HEAT TRANSFERRED 10**6 BTU/HR
1	0	0.0	0.0	0.0	0.0	0.0
2	1	100.5165	611.5597	440.9866	420.1285	5.0586
3	2	153.4509	329.3218	395.1061	369.8028	7.3300
4	3	210.0943	208.0226	350.2639	322.0352	12.3739
5	4	261.9448	110.7123	293.6125	263.1334	3.6952
6	0	0.0	0.0	0.0	0.0	0.0
7	6	42.0813	23.6638	237.0474	205.4593	3.5650
8	7	82.9521	9.5996	152.7911	120.7427	0.5309
9	0	0.0	0.0	0.0	0.0	0.0
10	8	163.6423	3.3474	145.7911	113.7404	9.1463

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APPENDIX

The formulas used in the analysis are summarized below.

(1) Exhaust loss of steam is expressed as a function of its annulus velocity at last stage and can be calculated as:

$$V_{an} = \frac{M_c V_c (1-m)}{3600 A_{an}} \quad (1)$$

where:

M_c = exhaust flow per end lb/hr;

V_c = saturated dry specific volume of steam at exhaust pressure
ft /lb;

m = moisture content at expansion-line end point; and

A_{an} = annulus area, ft

(2) The relation between leakage flowrate and flow condition is calculated as:

$$M_{\ell} = C \sqrt{\frac{P_i}{V_i}} \quad (2)$$

where:

M = leakage flowrate, lb/hr;

P_i = initial pressure, psia; and

V_i = initial specific volume, ft/lb.

(3) The relation between the first-stage shell available energy at partial load and that at design point is calculated as:

$$(AE)_1 = (AE)_{1d} \times \frac{1-(P_1/P_o)^{0.3/1.3}}{1-(P_{1d}/P_o)^{0.3/1.3}} \quad (3)$$

where:

$(AE)_1$ = available energy of 1st stage at partial load, Btu/lb;

$(AE)_{1d}$ = available energy of 1st stage at design point, Btu/lb;

P_1 = first-stage shell pressure at partial load, psia;

P_{1d} = first-stage shell pressure at design point, psia;

P_o = steam pressure ahead of the first-stage nozzle, psia; and

1.3 = ration of specific heats, C_p/C_v , of superheated steam.

(4) *Isentropic steam expansion velocity of the first stage is calculated as:*

$$V_o = 223.9 \sqrt{(AE)_1} \quad (4)$$

where:

V_o = isentropic expansion velocity, ft/sec; and

$(AE)_1$ = first stage shell available energy, Btu/lb.

(5) *Bucket velocity of the first stage is calculated as:*

$$W = \frac{\pi D f}{720} \quad (5)$$

where:

w = bucket velocity of first stage, ft/sec;

D = average pitch diameter of first-stage bucket, in.; and

f = wheel velocity, rpm.

(6) *The relation between first-stage efficiency at partial load and that at design point, required if a measured relation of efficiency vs. velocity ratio is not available, is calculated as follows:*

$$\eta = \eta_d \times 4 \left[\frac{W}{V_o} - \left(\frac{W}{V_o} \right)^2 \right] \text{ for 1-row governing stage} \quad (6a)$$

$$\eta = \eta_d \times \frac{8W}{V_o} \left(1 - \frac{2W}{V_o} \right) \text{ for 2-row governing stage} \quad (6b)$$

where:

η = efficiency at partial load,

η_d = efficiency at design point; and

$\frac{W}{V_o}$ = ratio of bucket velocity to isentropic expansion velocity of steam.

(7) Throttle steam flow, as a function of first-stage shell condition, is calculated as:

$$M_1 = M_{1d} \times \frac{P_1}{P_{1d}} \times \sqrt{\frac{T_{1d}}{T_1}} \quad (7)$$

where:

M_1 = partial load throttle flow, lb/hr;

M_{1d} = design throttle flow, lb/hr;

P_{1d} = first-stage shell pressure at design flow, psia;

P_1 = first-stage shell pressure at partial flow, psia;

T_{1d} = absolute temperature of steam at first-stage shell and design point, °R; and

T_1 = absolute temperature of steam at first-stage shell and partial load, °R.

(8) Pressure of extraction steam and exhaust steam for noncondensing cylinders as a function of steam flowrate going to the following stage or exhausted is calculated as:

$$P = P_d \times \frac{M}{M_d} \times \sqrt{\frac{T}{T_d}} \quad (8)$$

where:

P = pressure of extraction or exhaust steam at partial load, psia;

P_d = pressure of extraction or exhaust steam at design point, psia;

M = flowrate going to following stage or exhausted at partial load, lb/hr;

M_d = flowrate going to the following stage or exhausted at design point, lb/hr;

T = temperature of steam at partial load, °R; and

T_d = temperature of steam at design point, °R.

(9) *The relation between the expansion line on the Mollier diagram at partial load and that at design point, i.e., the slopes of expansion lines are the same.*

$$\frac{\Delta h}{\Delta s} = \left(\frac{\Delta h}{\Delta s} \right)_d \quad (9)$$

where:

$$\frac{\Delta h}{\Delta s} = \text{slope of expansion line at partial load, } ^\circ\text{F; and}$$

$$\left(\frac{\Delta h}{\Delta s} \right)_d = \text{slope of expansion line at design point, } ^\circ\text{F.}$$

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