PERFORMANCE EVALUATION OF A NUCLEAR STATION WITH PEPSE

By

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Presented at PEPSE User's Group Meeting
November 1-2, 1984 New Orleans, LA

ABSTRACT

In January 1984, Vepco's Performance Tests and Results Analysis group began a performance evaluation of the secondary plant of a PWR nuclear station located in Surry, Va. in an effort to identify the correct secondary side flow which should be used to calculate reactor power. Also addressed in this study was secondary plant performance characteristics which affect load. To complete this study, a PEPSE computer model was used in sensitivity analysis and in the simulation of the plant with test data. The results of the study on Unit 2 showed that the steam flow measurement was a better indication of the system mass flow and that approximately 18 MW of output could be gained by using steam flow to calculate reactor power.

SECTION 1

INTRODUCTION

In early August 1983, the Performance Tests and Results Analysis (PT&RA) group at Vepco was asked to evaluate the maximum dependable capability of our two nuclear generating units in Surry, Virginia. The results of this preliminary study showed that: 1) the design unit rating at the licensed core power level with the new steam generators was not known; 2) even with a conservative estimate of the design rating, approximately 30 MW of output was unaccounted for from actual operating history; and 3) the measured steam flow plus the measured blowdown flow was typically 1 to 2 percent below measured feedwater flow.

Because of these problems, a two phase study was started in November 1982. Phase 1 was to determine the correct flow to be used in the reactor calorimetric calculation. Phase 2 was to identify the effect on plant performance for various secondary plant performance characteristics.

The two units at Surry are identical with Westinghouse turbine - generators and pressurized water reactors. The original design conditions were: throttle pressure - 733 psia, SGU outlet quality - 99.75 percent; reheater TTD - 25°F and moisture removal effectiveness of the MSRs - 98 percent. There are two trains of six feedwater heaters. The tandum compond turbine has five stages of moisture removal and two double flow L.P. turbines each of which exhaust to a separate condenser. The condensers have titanium tubes and gravity feed circulating water.

A flash evaporator is located in the condensate system prior to the lowest feedwater heater. Steam can be supplied by either the extraction line or auxiliary steam from the H.P. turbine exhaust. Furthermore, an auxiliary condenser in the flash evaporator can use bearing cooling water to boost distillate output.

In 1980, the steam generators on Unit 2 were replaced with a new design by Westinghouse. Later in 1981, those on Unit 1 were also replaced. The new design together with other cycle modifications increased the throttle pressure to 850 psia with no change in the guaranteed SGU outlet quality.

In the first attempt to complete phase 1 by a consulting engineering firm, analysis of test results showed that steam flow was a good prediction of the actual system mass flow. However, their model ignored the operation of the flash evaporator, which during the test was run with auxiliary steam and bearing cooling water. Further analysis by PEPSE showed that the auxiliary steam load could be as great as the uncertainty between steam flow and feedwater flow. Therefore, Vepco's PT&RA group redefined the objectives of phase 2 to calculate the true system mass flow.

To complete phase 2 we first developed a PEPSE design model to calculate a realistic baseline output for the units. Next sensitivity studies were run on the model to find the effect on load of secondary plant performance. Finally a simulation of a performance test on Unit 2 was made to find the best approximation of the actual system mass flow and the effect on load for each component operating off design. This report is a brief review of each of these steps.

Section 2

SURRY DESIGN MODEL

The Surry PEPSE model begins and ends at the steam generator. There are 179 components, 261 streams in the model shown in figures 1A and 1B. A connection between the beginning and end of the model is made with a schedule of SGU pressure drop supplied by Westinghouse, so the PEPSE model is of the entire cycle.

For the turbines, GE procedures, were used to solve for the stage shell conditions. The GE models were chosen because accurate predictions of off design operation was desired from the sensitivity studies. Unfortunately, the GE model cannot exactly match the Westinghouse H.P. turbine conditions. Every attempt to match the conditions at the first stage, the first point extraction and the section exhaust resulted in a stage efficiency greater than 100 percent for the first H.P. blade group. The GE predicted turbine exhaust enthalpy is only 1.0 Btu/lb higher than the Westinghouse value; therefore, the difference has been ignored. The L.P. turbines were matched as closely as possible using schedules for exhaust loss, efficiency multipliers and shape factors.

The two feedwater heater trains have all design feedwater heaters modeled after vendor specifications. The two condensers use titanium heat transfer coefficients and match the HEI predictions of performance. All four MSRs are modeled with original performance data.

A new PEPSE heat balance is shown in Figure 2.

Section 3

SENSITIVITY STUDIES

Once the Surry design model was finished and checked against the original Westinghouse calculations, sensitivity studies were run to find the effect on load for secondary plant performance parameters. Each of these sensitivity studies was run at 100 percent licensed reactor power (2441 MW_t). This technique differs greatly from the standard industry practice of performing sensitivity studies at a constant control valve setting. However, since the unit in question is operated at a constant power level regardless of the secondary performance, this technique should be more applicable to performance test analysis.

One PEPSE study which showed a considerable difference from the Westinghouse predictions was the throttle pressure study. Westinghouse predicted an 11 MW loss in power for a 10 psi drop in throttle pressure. At a constant core power, the drop in load is only 0.9 MW. The difference in these two predictions can be explained by claculating the change in throttle flow at a constant valve setting with equation (1) and then the change in load for that change in flow.

$$\dot{m} = c \left(\frac{P}{v} \right)^{1/2} \tag{1}$$

Where: $\dot{m} \equiv$ mass flow (lb/hr)
$c \equiv$ flow coefficient
$P \equiv$ pressure (psia)
$v \equiv$ specific volume (ft^3/lb)

The sensitivity studies run on Surry were:

1) Throttle pressure
2) MSR moisture removal effectiveness
3) Reheater TTD
4) Reheater pressure drop
5) Steam generator quality
6) Circulating water temperature
7) First point heater TTD
8) Second point heater TTD
9) Flash evaporator performance
10) L.P. drain pump out of service
11) MSR tube leaks

Section 4

PERFORMANCE TEST ANALYSIS

Introduction

As mentioned in the introduction to this report, the calculated steam flow is generally 1.8 percent below the feedwater flow when the most accurate calculations of both flows are used[1]. Since the total uncertainty in the measurements can account for no more than about 1 percent, one of the two flows, or both, are in error by at least 0.8 percent.

To find the true flow, the PEPSE model was used to simulate the plant with actual data. The chief difficulty with this technique is that the majority of points which ultimately determine the required mass flow to produce a given output are immeasurable. For example, the thermodynamic states of the steam which are necessary in the calculation of turbine power cannot be determined without very elaborate techniques. Therefore, the PEPSE analysis could only generate the best approximation for the plant.

Procedure

To calculate the "true" cycle mass flow, test data on Unit 2 from June 28, 1984, was entered into a PEPSE performance model. In gathering this data as many unknown flows and immeasurable data points as possible were removed from the cycle. The test procedure called for isolation of the flash evaporator, auxiliary steam, and the Unit 2 distilled water tanks. In this set-up the unit not tested supplied auxiliary steam including the steam for the air ejector.

[1] In this report, for simplicity, when steam and feedwater flows are compared, the measured blowdown flow has been accounted for.

Because of measurement difficulties the PEPSE model was run with the following assumptions:

1) The turbine operated as designed with the given steam conditions that were measured. General Electric procedures, slightly modified for low pressure turbine efficiency were used to calculate turbine stage end points.

2) The moisture removal effectiveness of C MSR was calculated to yield the measured tube side drain flow regardless of the system mass flow.

3) The moisture removal effectiveness of A, B and D MSRs were set equal to one another and calculated to yield the design turbine first stage shell flow coefficient. This was necessary because the measured tube side flow of these MSRs was unreliable. Since design turbine efficiency was assumed, the flow coefficient must also be design in order for assumption (1) to be correct; therefore, this calculation had only one solution. The effectiveness of these MSRs was calculated regardless of the system mass flow.

4) Due to a suspected error in the measurement of the HP exhaust pressure, design pressure drops were calculated for the second point extraction lines and the exhaust pressure was back calculated.

5) No MSR tube leaks were modeled. However, tube leaks in the B MSR were suspected.

6) Design extraction pressure drops for those not measured were calculated from the equivalent lengths of pipe.

7) The change in tank and storage levels was used to calculate the net cycle makeup and losses. Auxiliary steam drips returned to Unit 2 from Unit 1 were ignored. The location of the excess losses over design was chosen as main steam for the most conservative estimate of system mass flow requirements.

8) The main steam quality leaving the steam generator was taken from actual test data which showed it to be 99.94 percent.

With these assumptions there were two uncertainties to account for. First, the third point extraction temperatures were about 30°F above design. Ironically, the third point extraction is the only point which can be measured to calculate a turbine section efficiency. To analyze the effect of this uncertainty on the system mass flow, two alternate models were analyzed. The first assumed design turbine efficiency. The second assumed that the turbine seals were leaking in both A and B LP turbines enough to raise the third point extraction temperature to the measured values[2].

Second, a leak detection device showed that one main steam dump valve and two main steam drain lines were leaking steam to the condenser. Again, two cases were chosen to find the effect on system mass flow of this leak. The first case assumed no leak and the required mass flow was calculated to yield the measured output. The second case assumed that the measured feedwater flow was correct and a leak was calculated to yield the measured output.

The four cases just described are reviewed in Table 1.

Table 1 PEPSE Performance Cases

Case No.	LP Turbine By-Pass A&B	Main Steam Leak To Condenser	System Mass Flow
1	No	No	Calculated
2	Yes	No	Calculated
3	No	Yes	Measured Feedwater
4	Yes	Yes	Measured Feedwater

The results of these four runs were then analyzed. Cases which yielded unrealistic or unreasonable values were discarded. Of the remaining cases, a compromise of the differences was used to create a "most reasonable" heat balance.

[2]Documented cases of turbine seal leakage have shown that, in some cases, the extraction temperature can be up to 80°F higher than the calculated turbine shell temperature.

Results

The results of the four computer simulations are presented in Table 2. This table shows the calculated steam and feedwater flows and the results of values calculated from the seven assumptions.

For case 1 in Table 2 the effectiveness of A, B and D MSRs was 94.6 percent. Considering that the design value was 98 percent, the effectiveness of "C" MSR was 93.6 percent, and tube leaks were expected in the B MSR, case 1 was eliminated from the possible cases.

Table 3 is a comparison of the actual turbine stage pressures to the design pressures. In this table, if the unit had been modeled correctly according to assumption 1, the corrected and design pressures should be equal. The corrected pressures were calculated with equation 2 which was derived from the flow equation for a compressible fluid through a turbine [1]. The design pressures were found at the actual flow to the turbine bowl. No corrections were necessary for the first stage pressure.

$$P_c = P_o \frac{\sqrt{v_a}}{\sqrt{v_d}} \tag{2}$$

where: $P_c \equiv$ corrected pressure (psia)

$P_o \equiv$ observed pressure (psia)

$v_a \equiv$ actual specific volume (ft^3/lb)

$v_d \equiv$ design specific volume (ft^3/lb) calculated at the actual bowl flow or flow to intercept valve.

Notice in Table 3 that case 1 has the greatest difference from the design case and can be eliminated on these grounds.

Table 2 PEPSE Heat Balance Results
Test 6-28-84

Case	1	2	3	4
Steam flow (lb/hr)	10,259,793	10,464,954	10,643,429	10,643,429
Feedwater flow (lb/hr)	10,305,114	10,510,275	10,688,750	10,688,750
Throttle flow (lb/hr)	9,531,021	9,539,337	9,535,419	9,535,026
Main steam leak (lb/hr)	0.0	0.0	288,069	272,083
HP exhaust press. (psia)	196.7	197.0	197.2	197.2
A,B,D MSR effectiveness (%)	94.6	86.1	82.2	80.2
C MSR effectiveness (%)	93.6	93.4	93.2	93.2
A MSR TTD (°F)	46.7	46.6	46.5	46.5
B MSR TTD (°F)	85.7	85.6	85.5	85.5
C MSR TTD (°F)	31.4	31.3	31.2	31.2
D MSR TTD (°F)	37.8	37.6	37.6	37.6
3rd point ext. temp. A (°F)	332.6	351.5	332.6	351.2
3rd point ext. temp. B (°F)	354.6	384.0	354.6	384.1
LP htr. drip pump flow A (lb/hr)	433,036	440,857	451,503	449,283
LP htr. drip pump flow B (lb/hr)	424,719	431,456	443,038	439,726
Steam leak to atm. (lb/hr)	27,668	27,670	28,070	28,070

Measured steam flow = 10,450,160 lb/hr

Measured feedwater flow = 10,688,750 lb/hr

Table 3 Turbine Corrected Pressures
Test 6-28-84

Case	1	2	3	4
First stage pressure (psia)	532.6	532.6	532.6	532.6
1st point extraction (psia)	403.3	403.3	403.3	403.3
Corrected (psia)	398.6	397.9	397.8	397.6
Corrected-design (psia)	-3.1	-2.1	-2.1	-2.1
LP "A" bowl pressure (psia)	172.5	172.5	172.5	172.5
Corrected (psia)	164.5	166.1	166.2	166.9
Corrected-design (psia)	-8.0	-6.0	-7.9	-5.8
LP "B" bowl pressure (psia)	174.3	174.3	174.3	174.3
Corrected (psia)	172.2	172.4	172.4	172.7
Corrected-design (psia)	-1.1	1.4	-1.0	1.5

From a comparison of the three remaining cases and a review of the test instrumentation, the apparent turbine seal leakage seemed to be a fact. Therefore, case 3 was eliminated due to a lack of seal leakage in the model. From all the data, case 2 and case 4 are almost indistinguishable from one another. What remained was to determine if the calculated main steam leak to the condenser was reasonable for case 4.

Since the leak was from main steam to the condenser, the leaking steam reached sonic velocity through the leaking hole. Also since the leak was around the seats of valves, the shape of the areas may have been annular. Theoretical calculations for flow through turbine shaft seals [2], which are also annular and sonic, showed that the required gap around the entire valve seat for the main steam dump valve was between 0.32 in. and 0.52 in. for case 4. For this flow (272,000 lb/hr) the total area had to be between 8.0 sq. in. and 13.0 sq. in. Thus, if the leak had been through a hole, similar to an orifice with a poor flow coefficient, its diameter would have to be between 3.3 in. and 4.2 in.

As a double check of these calculations the ideal flow equations were used to find the throat area of a converging diverging nozzle with sonic flow [3]. From these equations the area for an isentropic process for a water steam mixture at equilibrium is 6.8 sq. in. Since this area is less than that previously calculated, the areas between 8.0 sq. in. and 13.0 sq. in. should be reasonably correct.

In the view of this reporter, areas and flows of this magnitude, through closed valves are unreasonably high, particularly since a conservative choice of the location of excess cycle losses was chosen as the main steam line. The total main steam leakage of case 4 was almost 3 percent of the measured steam flow and costs about 22 MW of output. A more reasonable, and conservative, estimate of this leakage is 1 percent.[3] The required leakage area for a flow of this size is between 2.8 sq. in. and 4.5 sq. in. which is still rather large for a closed valve.

[3]A typical design value for steam, leakage at a fossil plant, where the pressure is almost three times that at Surry, is 0.5 percent. Actual test values range between 0.7 percent and 1.1 percent.

Since case 2 used a flow almost equal to the calculated steam flow, steam flow was assumed to be correct. PEPSE was then programmed to calculate the required main steam leak to the condenser given the steam flow to produce the measured generation. Also, the LP turbine seal leakage was calculated for the A and B turbines. The results of this run are shown in Table 4. A heat balance drawing of the test conditions is shown in figure 3.

Notice in Table 4 that the MSR effectiveness for all MSRs are nearly the same. This is a reasonable finding since all the MSRs are the same age and exposed to the same environment. The lower value for A, B and D may be conservative and allow for the effect of tube leaks on the effectiveness. Notice also that the main steam leak to the condenser is about 130,270 lb/hr or 1.2 percent of steam flow. Since the calculated value is greater than the estimated value, allowance for MSR tube leaks is accounted for.

Table 4 PEPSE Best Simulation of Test on 6-28-84

Steam flow (lb/hr)	10,451,059
Feedwater flow (lb/hr)	10,495,480
Throttle flow (lb/hr)	9,536,768
Steam leakage to condenser (lb/Hr	129,610
HP exhaust press. (psia)	197.0
A,B,D MSR effectiveness	90.1
C MSR effectiveness	93.4
MSR TTD (°F) A	46.6
B	85.6
C	31.3
D	37.7
3rd point ext. temp. (°F) A	352.1
B	384.5
Leak to atmosphere (lb/hr)	27670.
Error in 1st point ext. (psia)	-2.1
Error in "A" LP bowl press. (psia)	-6.0
Error in "B" LP bowl press. (psia)	1.4
"A" LP heater drip pump flow (lb/hr)	440,043
"B" LP heater drip pump flow (lb/hr)	430,615

Summary

In conclusion to this section, a snapshot of historical data was analyzed. A snapshot of an instant in time can only show relative values of the various parameters, and not the truth of either one. Determination of truth is better done by trending more than two parameters over time. The extra parameters monitored can serve as a double check against the two primary ones and help establish the truth of one or the other in the case that there is a change over time.

On August 10, 1981 a new flow calculation program was completed and tested on Unit 2. This program calculated feedwater flow and steam flow. The results show that the ratio of steam flow plus blowdown to feedwater flow was 1.00434. That is steam flow was 0.434 percent higher than feedwater flow.

Obviously, something has occurred to one or both the sets of flow venturies since 1981. However, without a double check we must rely on the PEPSE simulation of the June 28, 1984 test to determine the best approximation of flow. Thus, based on all the available data Surry was recommended to use steam flow as the basis for their reactor heat input. The estimated increase in output expected from this recommendation was 17.8 MW. On September 12, 1984, Surry accepted this and other recommendations and began software changes to make them workable. Completion is expected by December 1, 1984.

Section 5

CORRECTIONS TO DESIGN OUTPUT

From the sensitivity studies on the Surry units a list of controllable deviations from design load was compiled. The results of the sensitivity runs were compared to the best representation of the June 28 test and deviations from design load were calculated. The results of these calculations are shown in Table 5. Here the three main areas of losses are: 1) system mass flow 17.8 MW, 2) steam leakage 11.3 MW, and 3) the MSRs 14.5 MW. The total losses from design load at the licensed power was 44.3 MW. The three main areas account for 88 percent of all the losses in load. Below is a short description of the main deviations and actions recommended to eliminate or reduce them and other minor deviations.

System Mass Flow

The test reactor core power was 2386 MW_t or 97.8 percent of licensed power. From the actual steam rate (the amount of steam required to produce 1 MW) the difference in core power from 2441 MW_t costs 17.8 MW of electrical output.

From preliminary calculations on Unit 1 the difference in steam and feedwater flows may not be as dramatic as on Unit 2. However, since the units are so similar, the same potential exists to gain output power by using steam flow as the basis of reactor power. This will be particularly true in the future.

Steam Leakage to the Condenser

This loss can be classified as a catchall which explains the difference between the theoretical and actual generation. The 9.6 MW may actually be comprised of 1) actual leakage to the condenser, 2) MSR tube leaks, and 3) turbine efficiency. Without further information[4] the allocation of the possible losses to these three causes can be completely arbitrary. One estimate, based on experience with fossil units is:

[4] At the time of this report a test is planned to help identify the steam leakage to the condenser and atmosphere.

1) Steam leakage to condenser 7.6 MW
2) MSR tube leaks 0-2.0 MW
3) Turbine efficiency 0-2.0 MW

From these estimates the steam leakage to the condenser was about 105,000 lb/hr and the reheater tube leaks amounted to 65,000 lb/hr.

Moisture Separator Reheaters

Including the above comments, the total loss due to the reheater pressure drop, TTD, effectiveness, and tube leaks is about 16 MW. When the MSRs are replaced these losses should be recovered. In fact another 7 MW should be seen due to a decrease in both the design TTD and pressure drop.

Minor Losses

Of the minor losses listed in Table 5, many seem rather insignificant in relation to the three major losses just discussed. However, several of these losses require very little, if any, capital expenditures and only a few man-hours to investigate. One such loss is that due to the first point heater drain cooler approach (DCA). From test data on both units the drain temperatures on these heaters are often close to the saturation temperature of the heater shell pressures. This means that the drain cooler section may be by-passed allowing steam to blow through the heater.

From experiments with vertical heaters the DCA can be lowered by as much as 50°F by increasing heater level only two inches. Therefore, tests should be conducted on each heater to measure the heater level effects on the DCA. From the results of these tests, optimum heater levels should be set and periodically checked.

Another loss which can be reduced is that due to excess make-up. To reduce or eliminate the loss of these 2 MW, a thorough search for the cycle water losses must be done. Obviously, this may require considerable man-hours over a month's time. However, the corrective action may require very little capital expense for a possible savings of 2 MW.

Finally, the possible existence of turbine seal leakage should be investigated. First a check of the present RTDs in the third point extraction lines should be made with a thermocouple or RTD of a known accuracy. If the temperatures measured by the standard RTDs can be shown to be correct, then there is good evidence of turbine seal deterioration in the first section of the LP turbines. In this case preparations should be made to inspect and repair the seals at the next opportunity. Experience with fossil plants show that these seals can deteriorate at an exponential rate and lead to poor turbine efficiencies.

Another reason for a high third point extraction temperature is actual blade damage in the first rows of the turbine. From fossil experience, blade damage can go hand-in-hand with seal leakage problems. At Surry, there is the possibility that small pieces of metal from the deteriorating reheaters and moisture separators have passed through the LP turbines. If a positive check of the temperatures measured during the test, and the pressures predicted by the PEPSE simulation is made, then there should be serious consideration of an inspection of the LP turbines.

Table 5 Surry Unit 2 Corrections To Load From Test 6-28-84

Description	ΔLoad (MW)
1) System mass flow	-17.8
2) Steam pressure	2.7
3) SGU quality	0.2
4) Reheater TTD	-7.2
5) Reheater pressure drop	-5.8
6) MSR effectiveness	-3.1
7) Back pressure correction	-1.8
8) 1st point heater DCA	-0.7
9) 1st point heater TTD	0.2
10) Cycle steam losses	-2.1
11) Steam leakge to condenser	-9.6
12) LP turbine seal leakage	-1.6
Total accounted for deviation	-46.6
Predicted load	781.7
Actual load	780
Unaccounted for deviation	-1.7
Unaccounted for percent of actual	0.218%

REFERENCES

[1] K. C. Cotton, P. Schofield; "Analysis of Changes in the Performance Characteristics of Steam Turbines"; ASME, Nov. 3, 1970

[2] J. K. Salisbury; Steam Turbines and Their Cycles; Robert E. Krieger Publishing Co.; Huntington, N.Y.; 1950

[3] Thermodynamic and Transport Properties of Steam; ASME, 1967

FIGURES

The figures are attached in pairs. To see the entire figures match 1A to 1B, 1C to 1D and 2A to 2B.

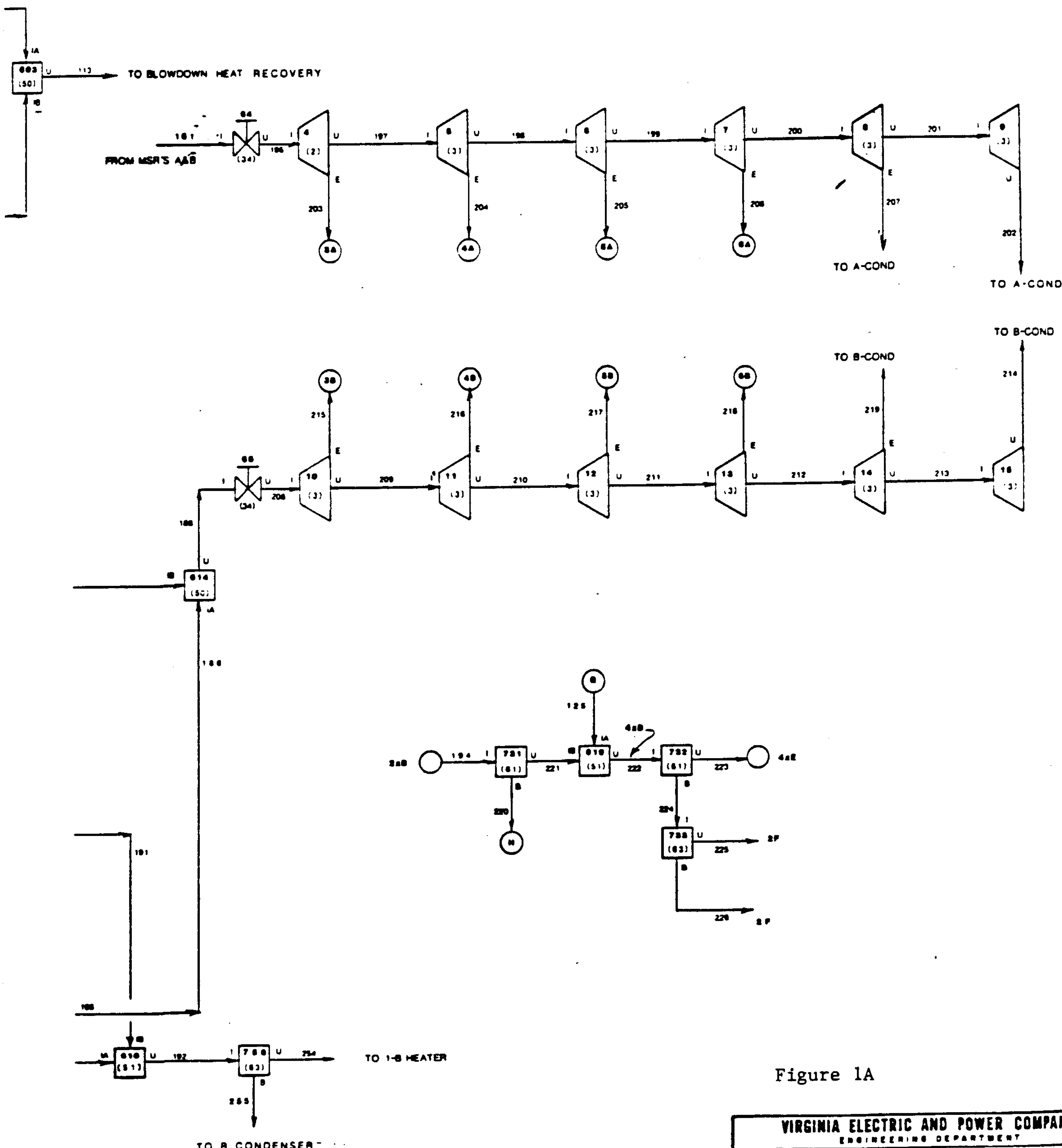


Figure 1A

VIRGINIA ELECTRIC AND POWER COMPANY
ENGINEERING DEPARTMENT
SURRY UNITS 1&2
PEPSE MODEL

	NAME	DATE	PROJECT NO	SHEET
DRAWN	CEW	9-20-84		1 OF
CHECKED			SCALE	
INSPECTED				
CORRECT			REVISIONS	
APPROVED				

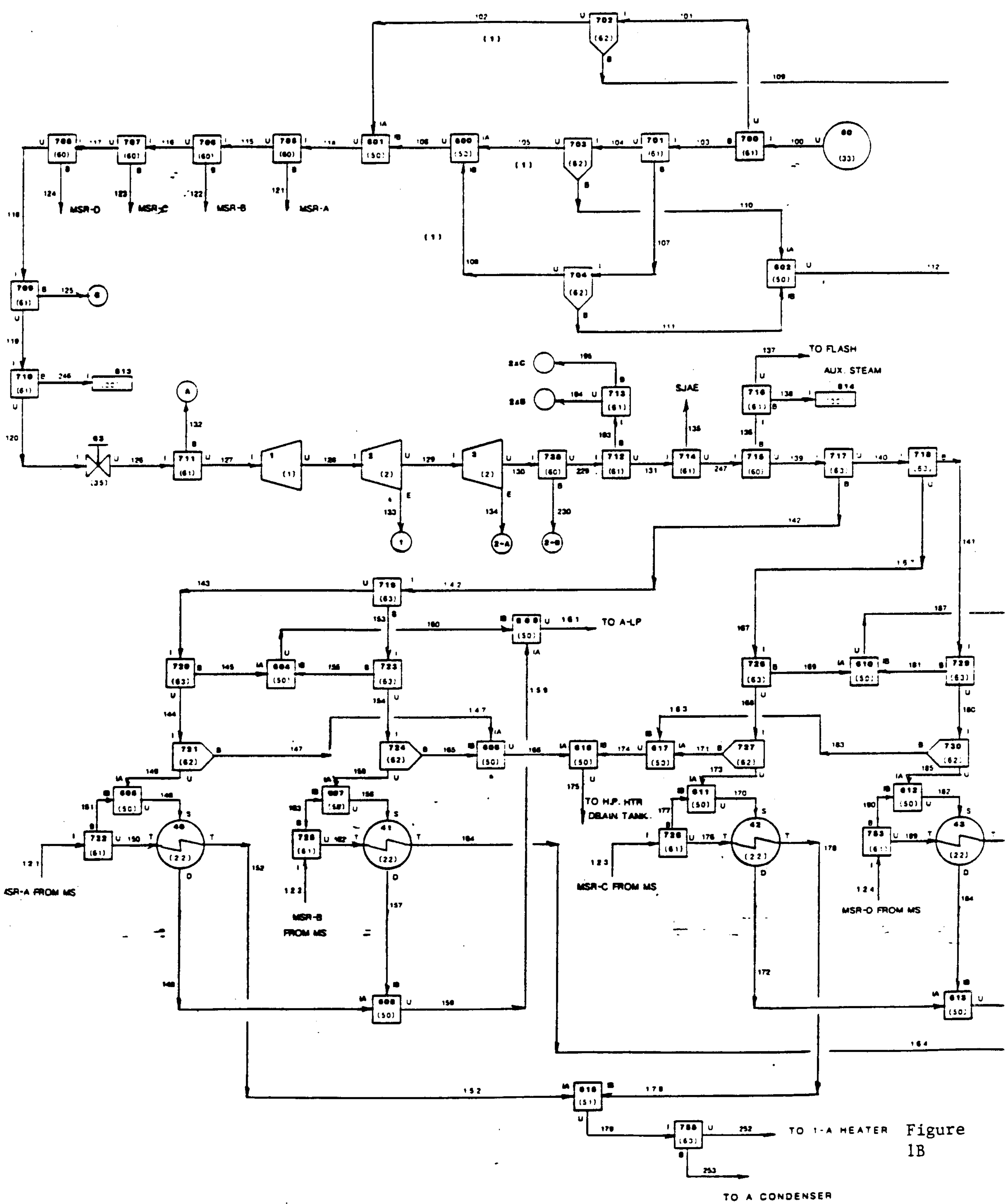

TO FLASH
AUX. STEAM
SJAE
MSR-D
MSR-C
MSR-B
MSR-A
TO A-LP
TO H.P. HTR DRAIN TANK
MSR-C FROM MS
MSR-D FROM MS
MSR-B FROM MS
TO 1-A HEATER
TO A CONDENSER

Figure 1B

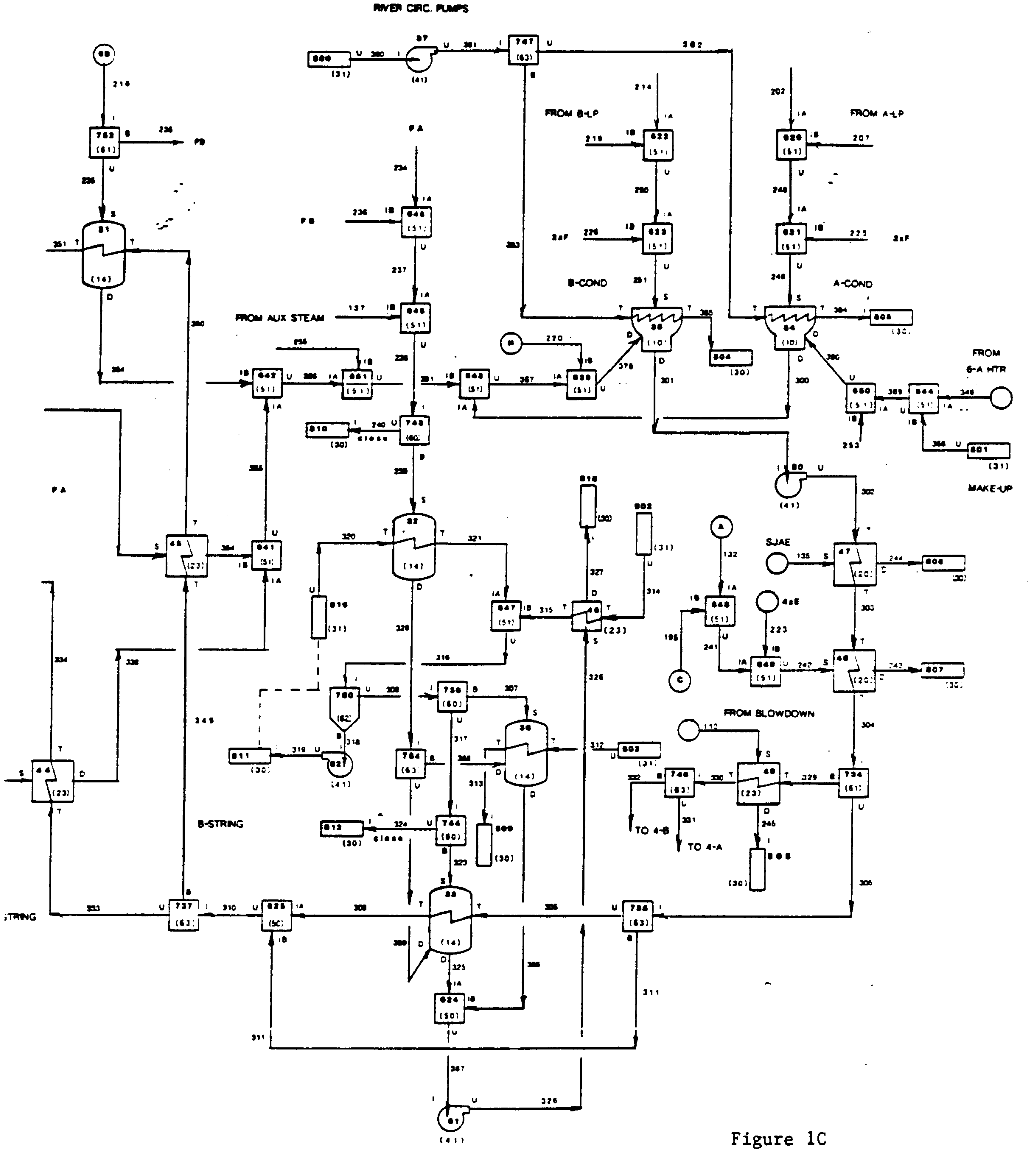


Figure 1C

VIRGINIA ELECTRIC AND POWER COMPANY
ENGINEERING DEPARTMENT

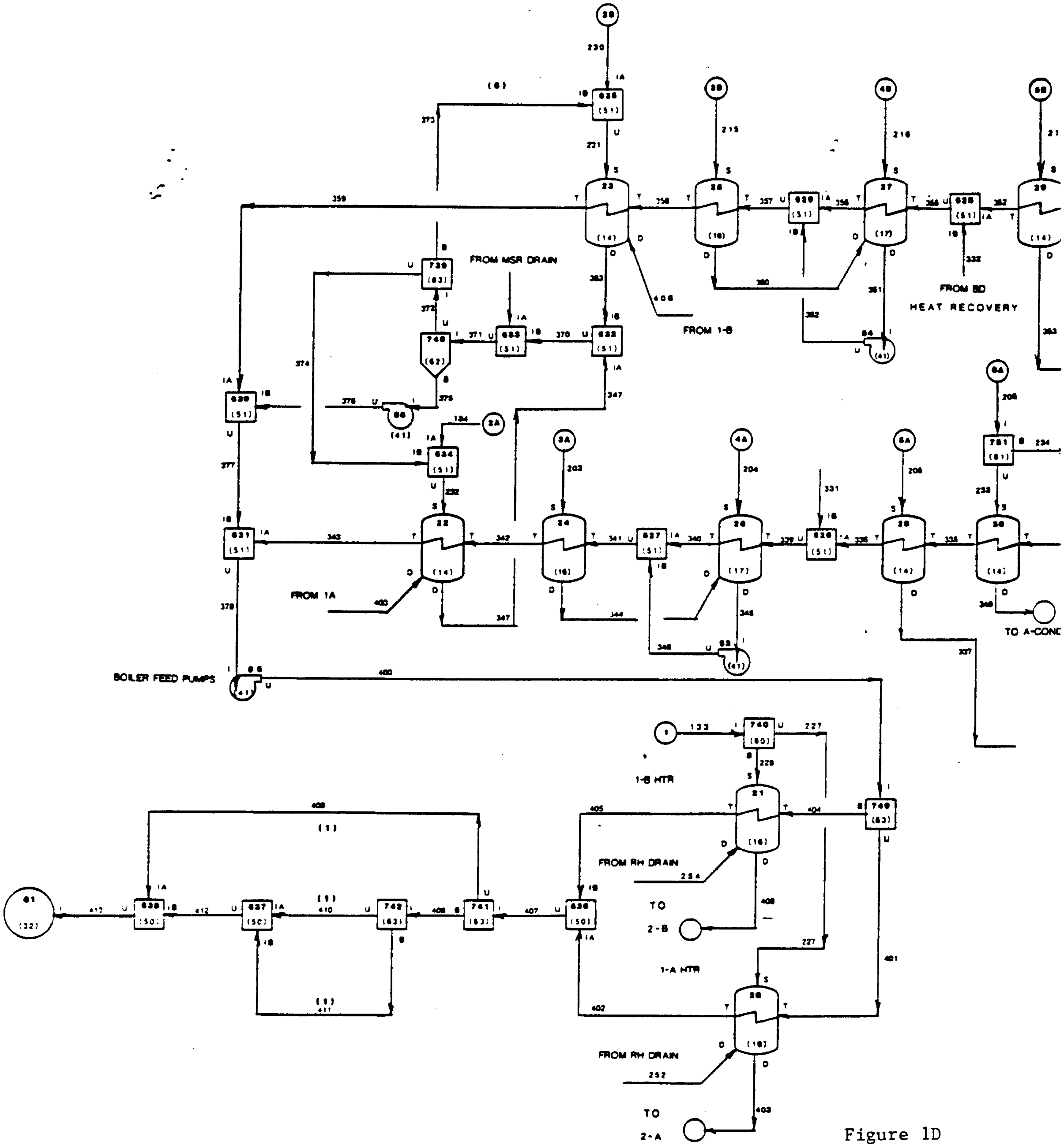
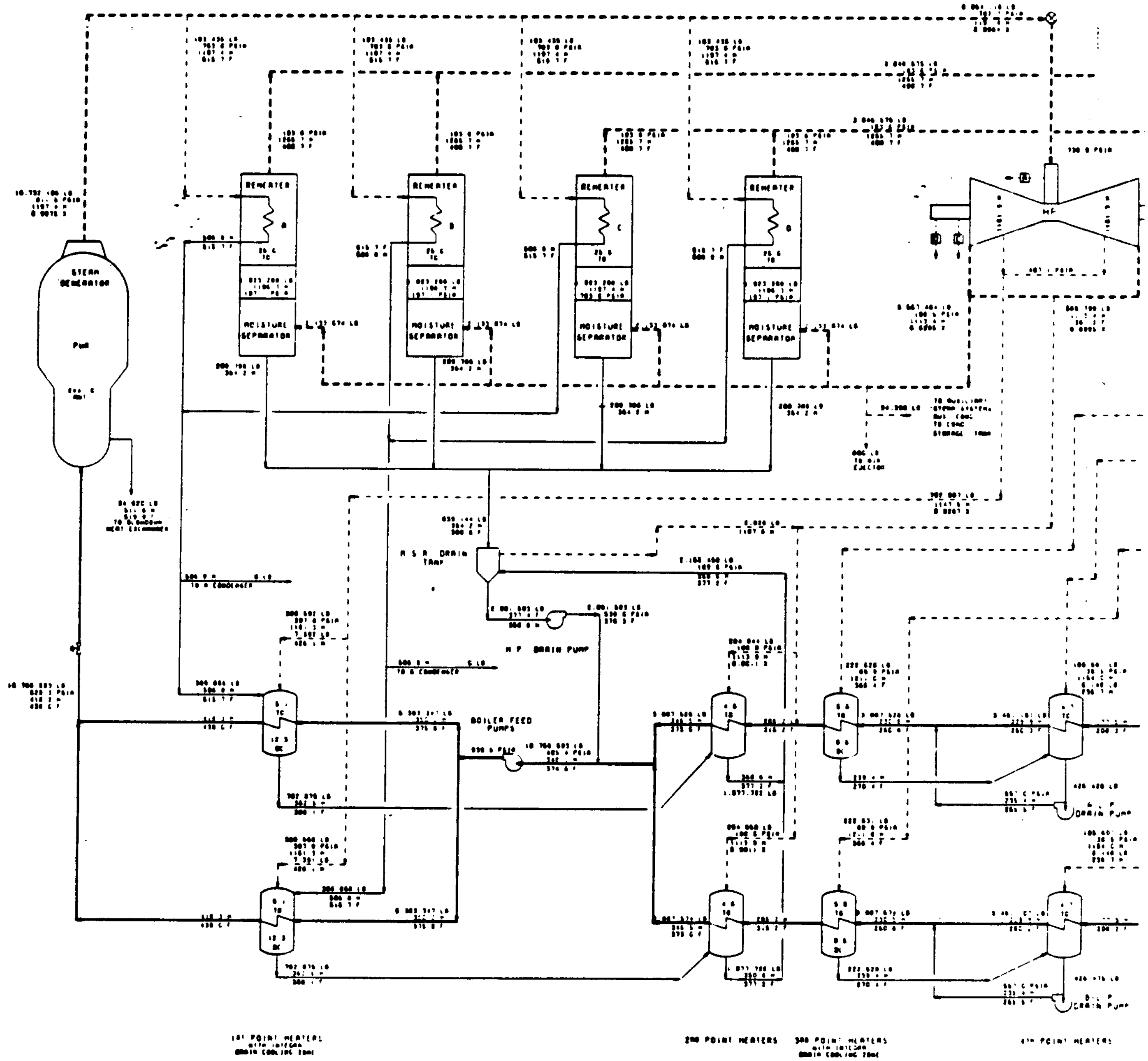


Figure 1D



BASIS OF HEAT BALANCE CALCULATIONS

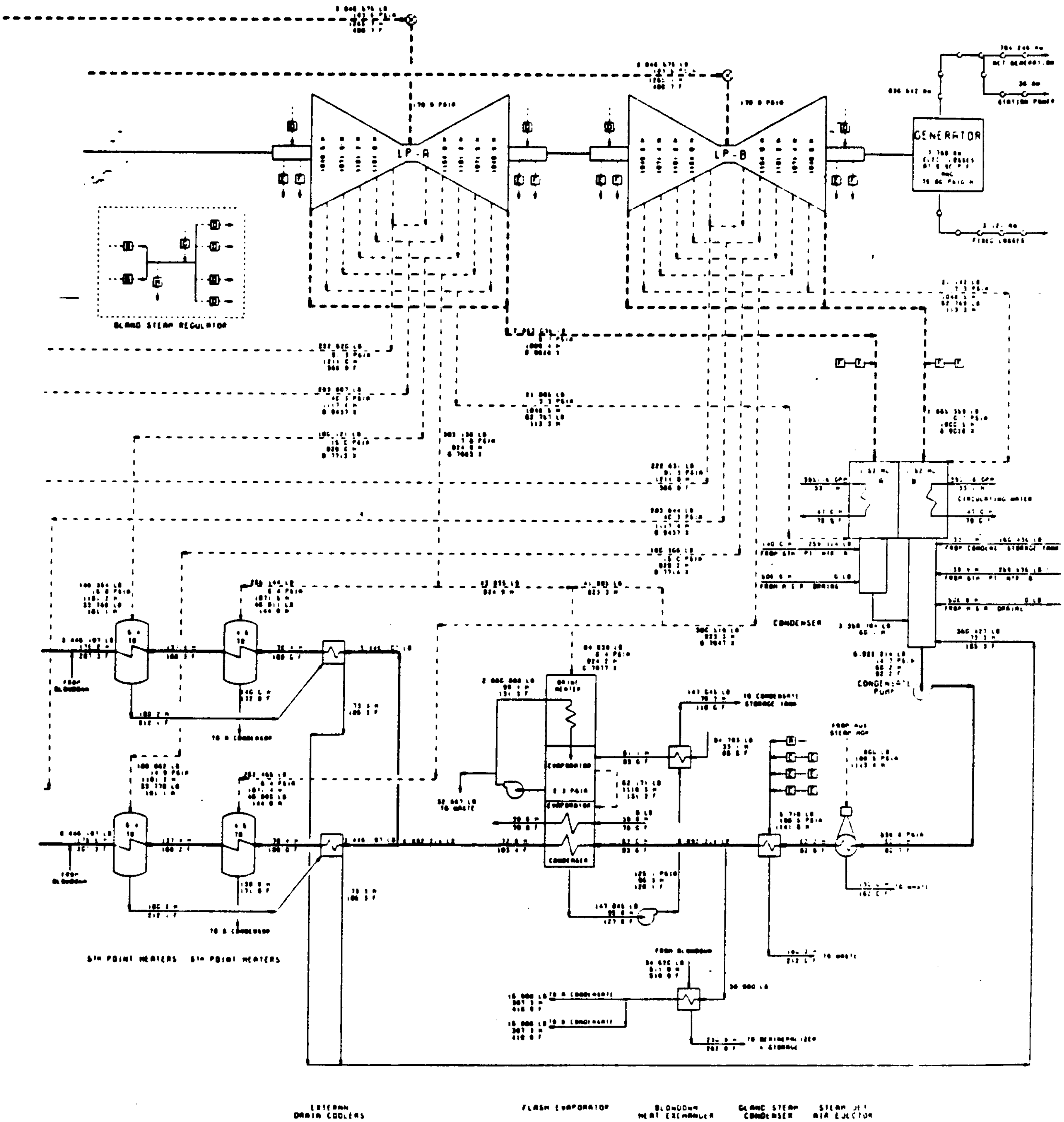
GROSS TURBINE HEAT RATE =

NET TURBINE HEAT RATE =

STATION HEAT RATE = 10487 BTU/KW

Figure 2B

822,571 KW TURBINE GENERATOR - TC4F-44" LSB



10072 BTU/KWH

* 10532 BTU/KWH

Figure 2A

VIRGINIA ELECTRIC AND POWER COMPANY

SURRY 1 & 2
HEAT BALANCE DIAGRAM
2441.0 MW LOAD

DRAWING NO 11448-FM-13 J

SHEET NO 1 OF 5