Variable Pressure Operation: PEPSE® Computer Modeling Assessment

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ABSTRACT

Power plants designed for baseload service increasingly are being required to operate in a cycling mode due to overcapacity problems. Improved cycle capability due to conversion to Variable Pressure Operation (VPO) results in performance benefits and drawbacks. To evaluate these changes, simulation calculations were carried out for Niagara Mohawk Power Corporation's Dunkirk Steam Station Unit 1 and 2 combined turbine and T-fired, CE 100 MW (coal-fired) boiler models using version 57 of PEPSE computer program. Furthermore, limited studies were done with gas and oil fuels to simulate conditions on Albany Station's CE boilers (converted from coal to gas/oil) design similar to Dunkirk Steam Station Units 1 & 2. Runs were also made on the 200 MW Cross-compound turbine cycles to evaluate general conclusions in the paper. Model development, technical background and effects of VPO on the boiler, turbine and balance of plant are discussed.

INTRODUCTION

The emergence of unregulated generators in Niagara Mohawk's service territory has created a glut of capacity and over-generation, at times. Because these unregulated generators are generally base loaded, Niagara Mohawk is required to respond to system load demands with fewer units on line. This requires greater daily load changes. During periods of low demand, fossil generation units are often producing power that is not needed and is sold at a loss. By reducing minimum loads, the losses associated with these periods will be reduced.

As the relationship between system supply and total demand continues to change at Niagara Mohawk, the flexibility represented by units that can economically operate at extreme low loads will provide NMPC with significant operating and competitive advantages. In order for these units to continue profitable operation, they must be able to reduce load during periods of low demand and respond to periods of high demand.

To perform the analysis of this reduced load operation process NMPC developed one combined turbine and boiler cycle simulation model, using V57 of the PEPSE Computer Program to evaluate variable pressure operation (VPO). This model represents NMPC's Dunkirk Steam Station Units 1 & 2 turbine and T-fired CE 100 MW boiler. The results from the model, unless otherwise specified, were extrapolated for NMPC's remaining 100 MW and 200 MW coal fired units. Furthermore, limited studies were done with gas and oil fuel to simulate conditions on Albany's CE boilers and runs were made on the 200 MW turbine cycles to evaluate general conclusions in this paper, as to their applicability to our units.

Model Development

Figure 1 shows the PEPSE model constructed for the turbine and boiler cycle model. The combined model was developed with all major boiler and turbine components, making extensive use of the design mode input options. One of the major tasks was to account for the radiant heat transfer to the convective stage secondary superheater. (This had significant affects on the furnace exit gas temperature). In order to accomplish this, the number of tubes of the secondary SH which were primarily radiant heat transfer surfaces had to be calculated. The quantity of heat penetration was established by the analytical method[4]. The effective area of these tubes was based on FEGT given at that load. Correction for fin efficiencies was also made to properly account for all fined tube surfaces for the economizer.

The model was "tuned" to design data using PEPSE controls. Four design cases were used for tuning and validation. The validation was completed by running the model at valve points which matched the design parameters, and design performance of the model was matched with the tuning factors. All tuning factors (heat transfer coefficient multipliers) were well within an acceptable range of 0.85 to 1.15. Convective heat transfer coefficients particularly for convective stages within the gas stream (not wall tubes) had values in the range of 10 to 17 BTU/HR-FT^2-F. Combined form loss and friction factor multiplier were also within reasonable limits to account for pressure drop throughout the boiler. Input/ output and boiler loss methods efficiencies agreed within 0.2% (to insure a fuel flow consistent with the model, because fuel flow strongly affects FEGT). Air preheater efficiencies and x-ratios also matched with design values within 0.15%.

The tuning factors were then scheduled against main steam flow or as appropriate to yield a model which was "self-correcting" for all loads. VPO study was performed at the lowest 4 valve points, (on 8 valve machines) where SH/RH temperature generally start to fall off under CPO using PEPSE Option No. 1. Minimum air flow of 25% (required by NFPA) of total air flow at full load conditions was kept as a control limit.

Modeling Limitations:

1: Model cannot predict flame stability.
2: Model cannot simulate the effect of burner tilt angles, at present.
3: Model cannot simulate ramp rate (dynamic) effects.
4: Model cannot simulate deaerator 0_2 removal effectiveness.

VARIABLE PRESSURE OPERATION STUDY

Variable Pressure Operation

NMPC fossil units currently operate in the Constant Pressure Operating (CPO) mode. That is the boiler pressure is held constant and load is controlled by modulating the turbine control valves. Under VPO, turbine control valve position is held constant while the boiler pressure is varied as a function of the desired unit output. There are two modes of VPO, pure and hybrid. Under pure VPO, the unit output is controlled entirely by varying boiler steam pressure. Turbine control valves are held at or near their full open position and modulated very little. Under hybrid VPO, initial load reduction from full load to part load is accomplished by closing some of the turbine control valves (in sequential mode) while maintaining full throttle pressure. Further load reduction to low loads is accomplished by reducing boiler steam pressure while holding control valve position. The hybrid operating mode is the operating mode normally used with units of NMPC's design.

Figure D12-11.ATB defines and compares the concept of constant pressure (CPO) and variable pressure operation (VPO) at Dunkirk Units 1 & 2 and could be applied for the other units, too.

Main Steam Temperature

One of the major advantages of VPO is that it produces higher steam temperatures at all loads below the constant pressure guaranteed steam temperature control range (GSTCR). The heat of vaporization and specific heat of steam changes with reduced pressure. The model shows that these changes in steam properties lead to an improvement in VPO main steam temperature. Figure D12-16.ATB shows that at CPO the temperature starts dropping at approximately 68% (4th valve point) of load while during VPO the superheat temperature starts improving - relative to CPO - at approximately the same load.

Reheat Steam Temperature

Reheat steam temperature droop for CPO and VPO is shown in figure D12-17.ATB for the Dunkirk boiler. It should be noted that at CPO the RH temperature change starts dropping at approximately 68% of load whereas under VPO RH temperatures start improving -relative to CPO- at approximately the same load.

Reheat pressure is a function of load. This function is the same whether or not the unit is operated in constant or variable pressure mode. Hence the specific heat of reheat steam (at the target reheat temperature) is not changed by VPO. Increasing cold reheat steam temperature (the result of increased SH temperature), however, tends to directly raise the hot reheat temperature. In this model it was found that at 4 VWO, 68% load (approximately 68 MW), reducing throttle pressure from 1494 psia (CPO) to 600 psia (VPO) reduces reheat duty by over 65% (4.5% of total load). Fig D12-04.ATB plots reheat duty as a percent of total load vs. gross generation.

Main Steam Heat Distribution

The overall effect of pressure on the distribution of main steam heat pickup per pound of steam at various final feed water temperatures under VPO can be seen in figure D12-10.ATB. The graph is presented in fractional terms so that the shift in relative heat duty of various boiler components is more apparent.

The breakdown in figure D12-10.ATB shows that the percent of heat required per pound of water to raise the temperature to saturation from the economizer outlet to the drum circuit increases due to lower fluid heat capacity. Heat of vaporization per pound of water in waterwall circuits grows significantly due to reduced pressure which decreases the heat required. Heat required per pound of steam to raise temperature from saturation to 1000°F from drum to SH outlet circuit almost remains constant and does not change the percent of heat to main steam significantly.

Economizer Steaming

A reasonable differential between the saturation and measured economizer outlet temperatures should be maintained. The OEM suggests the measured economizer outlet temperature be maintained at least 20°F to 50°F below the calculated saturation temperature.

Under VPO, drum pressure and its corresponding saturation temperature drops, whereas under CPO, saturation temperature in the drum essentially remains unchanged with load. See figure D12-02.ATB for the comparison of both modes of operation. Steady state heat transfer in the economizer is almost unchanged by VPO. Figure D12-01.ATB shows the economizer temperature profile at design excess air in a steady state heat transfer condition under VPO. It appears that in the case of our 100 MW coal fired units there is approximately 90°F difference at the economizer outlet between the feed water out temperature and saturation temperature at 4th valve point.

In the case of gas-fired units, we should expect slightly higher economizer out temperatures, but this temperature is still significantly lower than saturation temperature.

However, during load increases, VPO requires substantial over-firing which can raise economizer heat transfer over that of CPO load changes. Increasing the economizer heat transfer will increase feed water out temperature, given the same flows. Under VPO, saturation temperature is lowered and water leaving the economizer is, therefore, also less sub-cooled because of the reduced sensible heat required for boiling at low pressures. Steam contained in water leaving the economizer can lead to economizer flow imbalances which can result in overheating. Hence it is easier to cause economizer steaming under over-firing with VPO.

A study was performed to evaluate cycle performance when burning oil at very high excess air and fuel flow higher than design at 1st valve point with sliding pressure from 1465 psia to 1000 psia to simulate adverse operating conditions and the effect of the NFPA excess air requirements. The results of the modeling indicated that subcooling at the economizer out is reduced by about 50 degrees at a given valve point under VPO, compared to CPO. These results agreed with the trends noted at Albany Station units, and confirmed that the above-design excess air was a primary cause of the reduction in subcooling.

SH & RH Spray System

Load increases under VPO generally require over-firing, which if uncontrolled, produces elevated steam temperatures. The level of over-firing required is dependent on the load ramp rate. The fact that VPO produces higher main and reheat steam temperatures at steady state part load operation makes steam temperature control during load increases even more demanding.

On drum units, control of steam temperatures during rapid VPO load increase may require more attemperation than available on units originally designed for base load operation. This might require replacement of lines, valves, or sprays in the attemperation systems, unless ramp rates are controlled to avoid this situation.

High levels of spray flow into a single attemperating unit should be avoided, since during a load increase, it is possible to spray so much that a saturated condition arises. Spraying to saturation must be avoided. It can lead to water droplet carry-over to the turbine or cause slugs of liquid to block circuits in the superheater, leading to tube burn out.

The boiler's tendency to produce higher steam temperatures under VPO will generally require higher levels of de-superheating spray under steady state operation. However, under hybrid VPO, pressure often starts sliding at or below the boiler's normal temperature control range. Therefore, the steady state VPO de-superheating water flow requirement will generally not exceed design levels, which are usually based on full load needs.

The supply of reheat attemperation might have to be changed. The reheat attemperating water supply pressure will drop as BF pump pressure drops. The reheat steam pressure is affected only by load and not the boiler feed system pressure. Therefore the possibility of inadequate pressure to supply attemperating water at lower pressure exists.

Experience at some of NMPC's units has shown that de-superheating sprays are not required. At all fixed loads modeled, down to 20% MCR, PEPSE modelling indicated that the CE 100-MW boilers do not require any spray flow, at design excess air levels, so this appears not to be a problem. Given the ability to fire the RH furnace separately on the 200-MW units (twin-furnace design), these units are expected to be more flexible rather than less. Testing will be used to determine the need for the de-superheaters before extensive investment is made refurbishing equipment that may not be needed.

Governing Stage Exhaust Temperature

It can be seen from the figure D12-03.ATB that governing stage exhaust temperature remains almost constant as flow is reduced at a given valve point with VPO, due to reduced control-valve throttling and the corresponding steam property changes at lower initial pressure. This results in more superheat in all turbine sections. At low loads, the drop-off in SH and RH temperature tends to reduce this SH slightly, but the drop-off in these temperatures will be less than the rated pressure at the same load.

Higher governing stage exit temperature leads to higher cold-reheat temperature, which directly reduces the hot reheat temperature drop.

Due to this higher temperature, if the turbine is shut down under VPO the metal temperature can be kept higher which this constant temperature causes less thermal stress due to smaller temperature changes vs. load for an equivalent load change under CPO. Also, the VPO allows more flexibility to maintain the turbine metal temperatures on shutdown. This can help at the time of the next morning's start-up, by allowing a more rapid match of steam and metal temperature, reducing start-up times. (Note: By closing valves partially, shortly before taking a unit off-line, turbine metal temperatures can be "adjusted" downward to match boiler capabilities to increase steam temperature on start-up. This can further optimize start-up times, if necessary).

HP Turbine Efficiency

The beneficial effect of VPO can be seen by noting the difference in overall HP turbine efficiency between the constant and variable pressure cases at 4th valve point. Figure D12-09.ATB compares the results under constant and sliding pressure operation against percent load. It shows that the HP turbine overall efficiency stays relatively constant (as a function of the constant throttle flow ratio used for VPO). The benefit due to constant higher HP efficiency could also be interpreted similarly for the 3rd, 2nd, or 1st valve points, relative to the 1st valve point.

HP Turbine Available Energy

The major disadvantage of VPO is that it has the lowest amount of energy per pound of steam available for conversion to shaft horsepower. Figure H78-04.ATB shows a comparison of available energy profile at 3rd valve point under CPO and VPO. It is clear that under CPO the highest energy available is at approximately 40 MW. But this disadvantage is cancelled out by the lower feed pump horsepower, HP turbine thermal stress, and higher HP turbine efficiency, and governing stage exit temperature is considered.

BFP Power Consumption

Boiler feed pump power consumption at low load can be significantly reduced by employing VPO. In fact, decreasing boiler feed pump power consumption is one of the largest factors providing heat rate improvement under VPO. The power savings are made of two components. First, required pumping power is reduced due to low discharge head. Second, a variable speed pump, if used, becomes more efficient at low loads when part pressure is required because it rotates more slowly. Figure H78-01.ATB shows power consumption under VPO for Huntley Unit 67 at three VWO. This shows that VPO reduces boiler feed pump power consumption by 44% @ 40 MW. More than 80% of that reduction is due to reduced head requirements.

Plants similar to NMPC's larger units with steam driven boiler feed pumps use extraction steam to supply the boiler feed pump turbine. Since VPO does not affect pressure at extraction points, the usual supply of steam should be adequate for normal operation. At very low loads under CPO, the steam is usually switched from extraction steam to higher pressure source, because extraction pressure is too low to achieve the required feed water head. Under VPO, main steam pressure drops off and may be too low to support boiler feed pump turbine operation.

Modeling of the 200-MW units indicated that there would be insufficient steam flow to the deaerator when main steam pressure drops below 1000 psia (low pump head, low steam demand). This is due to the cycle design, which utilizes an extraction turbine for the BFP drain, the exhaust of which feeds the deaerator. Throttling of the BF pump's discharge to increase pressure (which raises pump head and reduces benefits), and thus raise steam demand was simulated in the model to indicate the amount of pressure increase required and ascertain the final heat rate effect.

At loads as low as 38 MW, if the BFP discharge pressure is 200-300 psi above the throttle pressure, BFP turbine steam demand is sufficient to run the deaerator at its pressure control point can be maintained without excessive flow through the condenser dump valve. Under VPO, moreover, the use of primary SH "booster" steam should not be necessary at low loads.

Extraction Steam System

Under VPO, steam temperatures downstream of the HP turbine governing stage are higher than at comparable loads at full pressure. A concern was raised that the downstream extraction temperatures under VPO could approach or exceed piping and/or FWH design limits; the model was used to evaluate this issue. Figure H78-02.ATB compares 7th stage feed water heater extraction temperature profile at constant and variable pressure operation at 3 VWO for Huntley Unit 67. We should expect similar extraction temperature profiles for the other units, too, for heaters taking superheated steam. It indicates that VPO would not cause steam temperatures to exceed extraction steam system design limits, at least not at steady load conditions.

Feed Water Heaters

The quantity of condensate and feed water flowing through the heaters is unchanged by VPO. Higher steam temperatures and unchanged water flow combine to cause an imbalance that can affect heater performance. Figure H78-03.ATB indicates that heat duty of the condensing section reduces due to occurrence of de-superheating in this section with VPO. Model showed that this desuperheating in the condensing section is not significant enough to upset feedwater and its effectiveness.

Depending on the chosen VPO pressure ramp, steam temperature in the high pressure turbine can actually be higher at part load than at full load. The temperature increase over the full load level is usually small. However, design temperatures for feed water heaters and extraction piping on units designed for base load, CPO were often set very close to expected full load temperature. Exceeding heater design temperatures during prolonged periods of part load operation can lead to premature piping or heater failure.

Problems Encountered

At very low loads modelled, PEPSE could not handle the reverse flow phenomenon which can occur in the lowest-pressure extraction line. In order to run our models at these low loads, we are attempting to address to his problem through the use of conditional OPVB's and/or FORTRAN subroutines, but have not completed work at present. (Suggestions are welcome).

Conclusions

This paper has presented an analysis of variable pressure operational considerations using PEPSE. It provides a source for utility engineers to assess the potential benefits of VPO retrofit for a specific unit and determine the applicability of specific retrofit options. Advantages of VPO can include improved heat rate, decreased turbine stress, higher steam temperature at partial load, reduced boiler feed pump power consumption, and decreased minimum operating load. Reduced HP turbine available energy is a potential disadvantage, but of minor importance, given the advantages. This work also validated the VPO study done by EPRI.

Summary

Several PEPSE® cases were run to evaluate VPO effects on NMPC units. A combined turbine and boiler cycle model is essential to perform simulation under variable pressure operation without an extreme amount of external computation and iteration between models. The PEPSE models developed for this study have proven to be an accurate tool when compared to boiler vendor and EPRI predictions. Because of this, the Dunkirk 1 & 2 model will be used with confidence to generate correction curves for this unit, which will ultimately affect the thermal-performance-related databases used by operators to optimize heat rate and by staff to formulate fuel purchasing forecasts and dispatch incremental heat rate curves.

References:

1) PEPSE Manual: Users Input Description, Volume 1, Version 57; Haliburton NUS Corporation; 1993

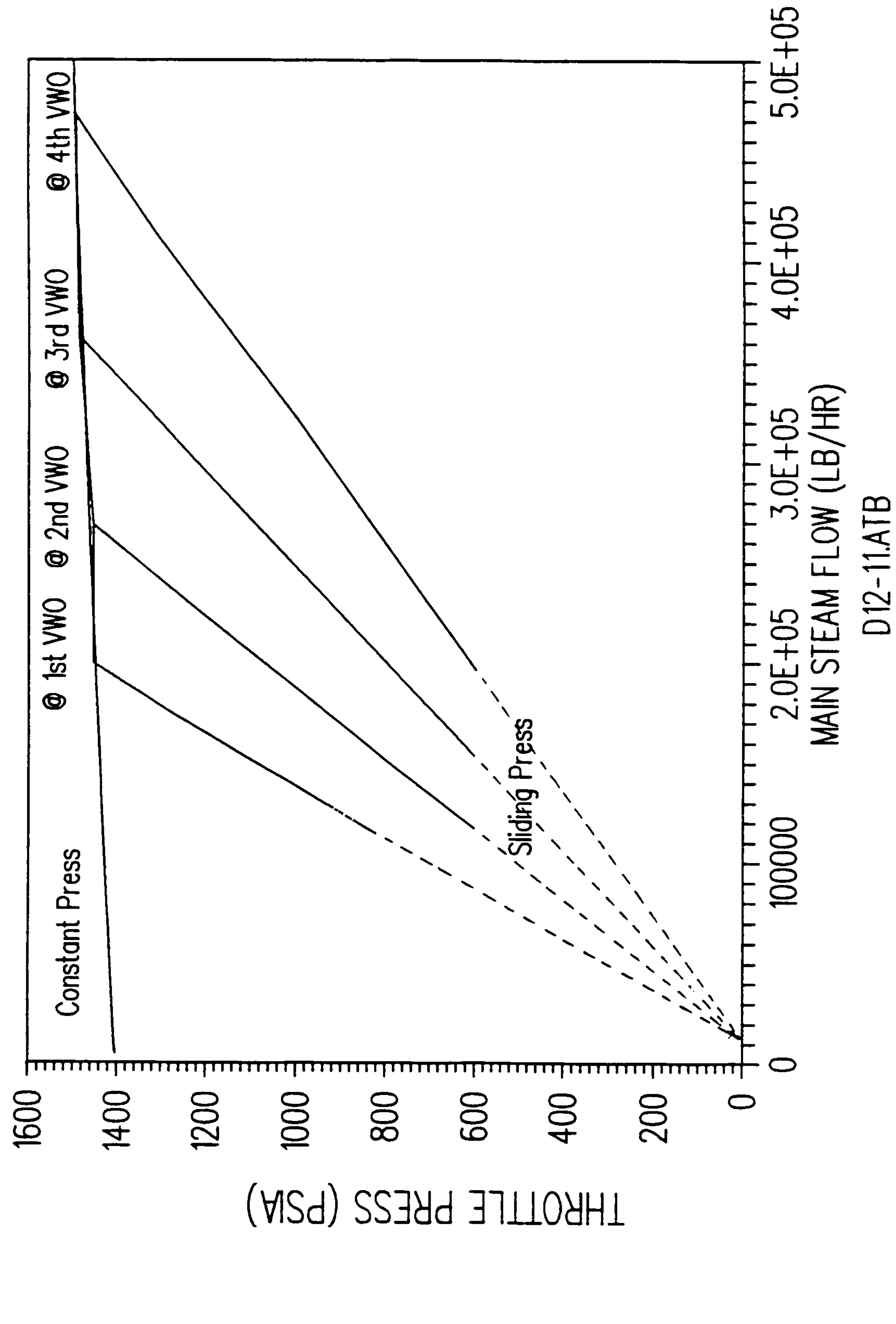
2) PEPSE Manual: Engineering Model Description, Volume II, Version 57, Haliburton NUS Corporation; 1993

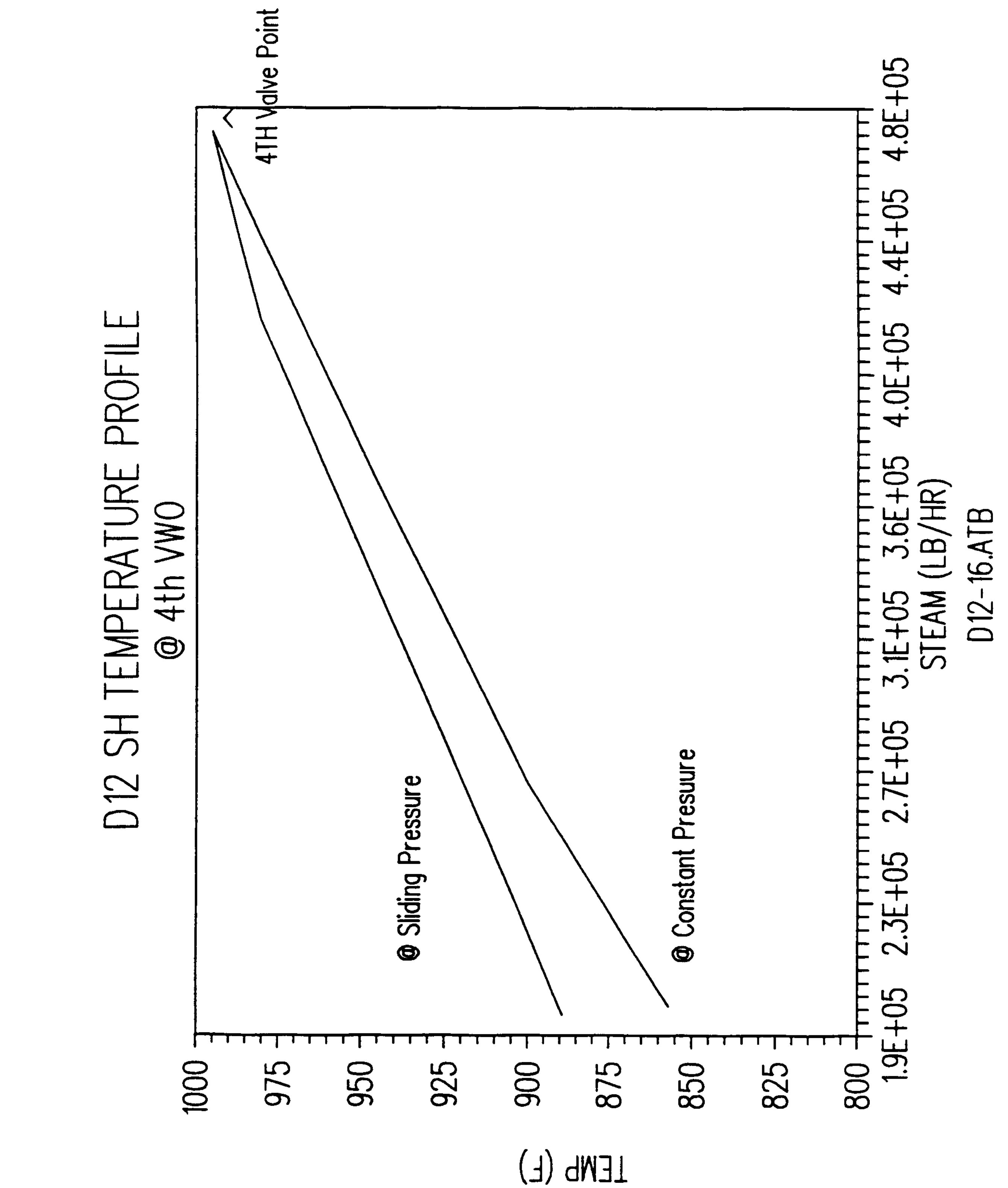
3) "Sliding Pressure Analysis"; Gerald Weber Proceedings of the 1992 Performance Software User's Group Meeting; Haliburton NUS Corporation

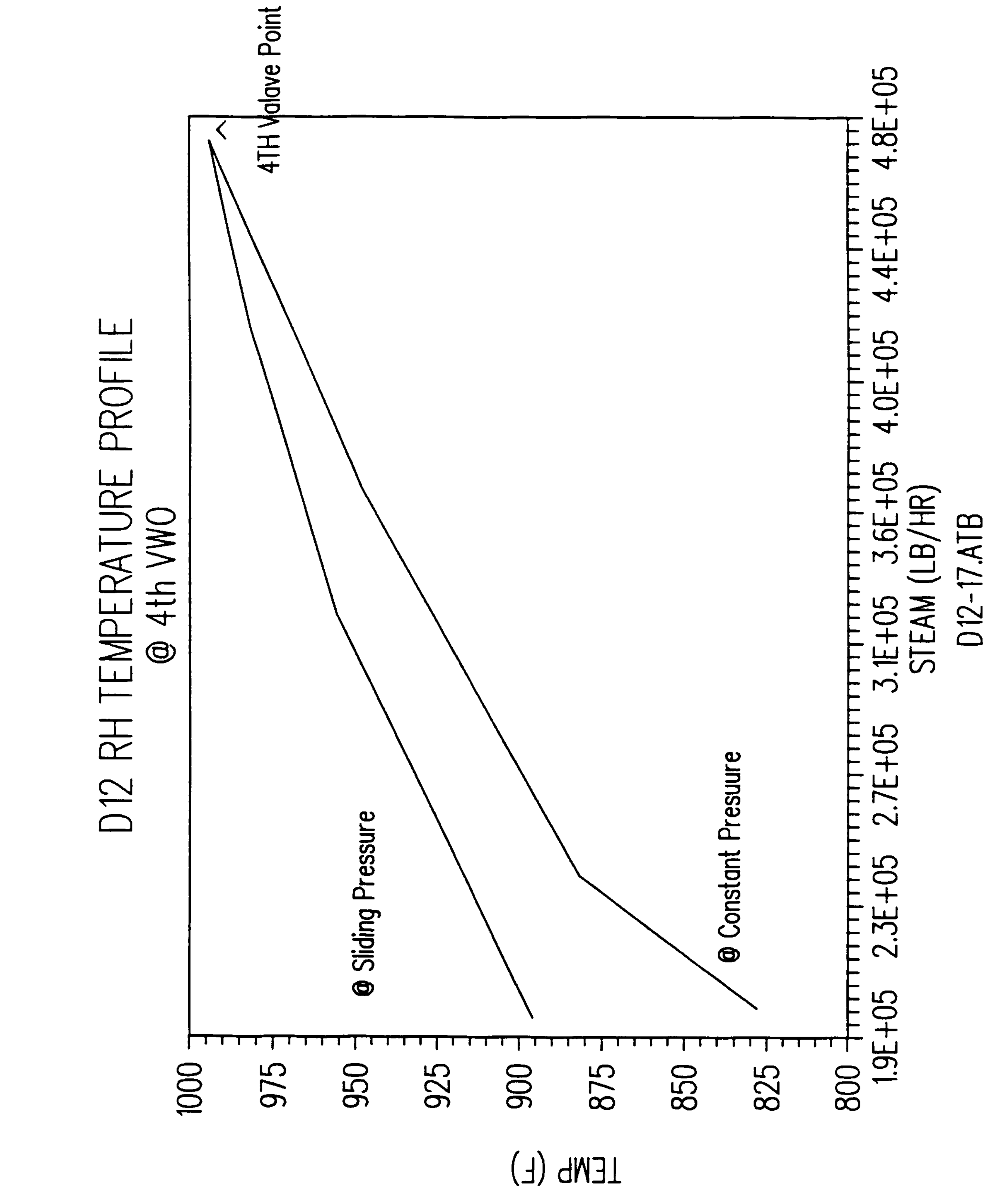
4) Steam Its Generation and Use, Babcock & Wilcox, 39th Edition, 1978

APPENDIX

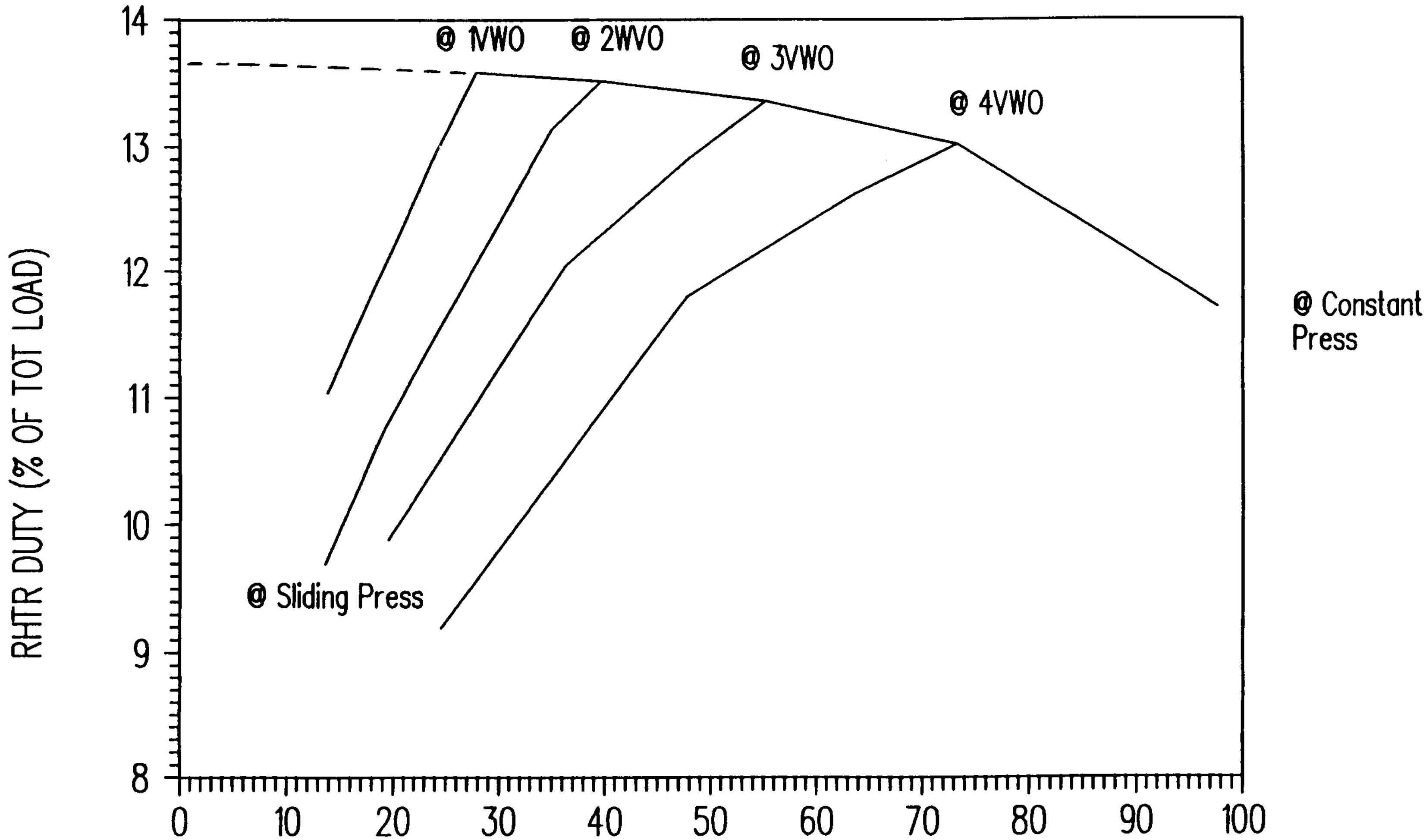
D12- DEFINITION OF VPO




D12 SH TEMPERATURE PROFILE
@ 4th VWO
4TH Valve Point
@ Sliding Pressure
@ Constant Presuure
TEMP (F)
1000
975
950
925
900
875
850
825
800
1.9E+05
2.3E+05
2.7E+05
3.1E+05
3.6E+05
4.0E+05
4.4E+05
4.8E+05
STEAM (LB/HR)
D12-16.ATB

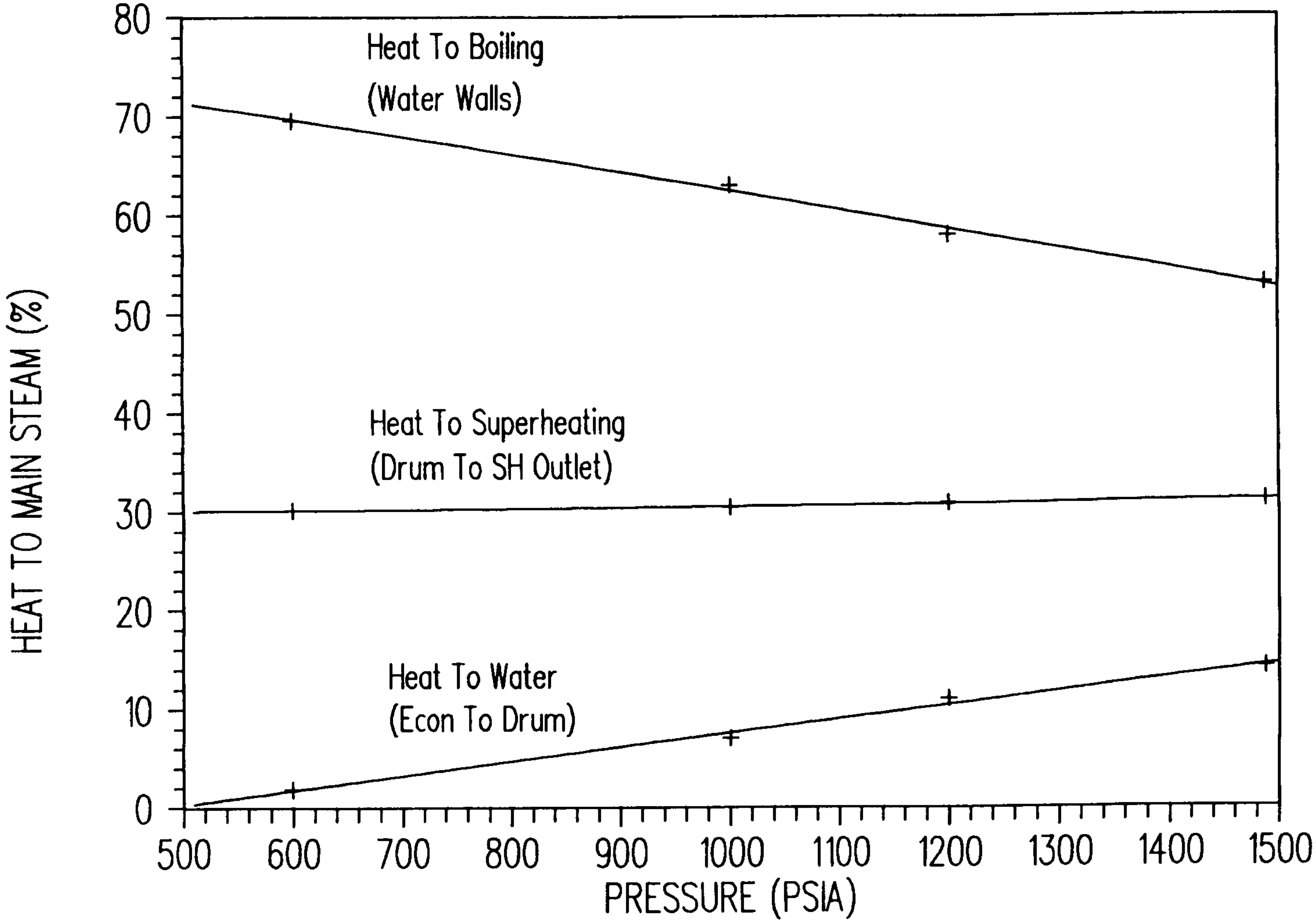

D12 RH TEMPERATURE PROFILE
@ 4th VWO
TEMP (F)
1000
975
950
925
900
875
850
825
800
1.9E+05
2.3E+05
2.7E+05
3.1E+05
3.6E+05
4.0E+05
4.4E+05
4.8E+05
STEAM (LB/HR)
D12-17.ATB
@ Sliding Pressure
@ Constant Presuure
4TH Valave Point

D12 REHEATER DUTY PROFILE



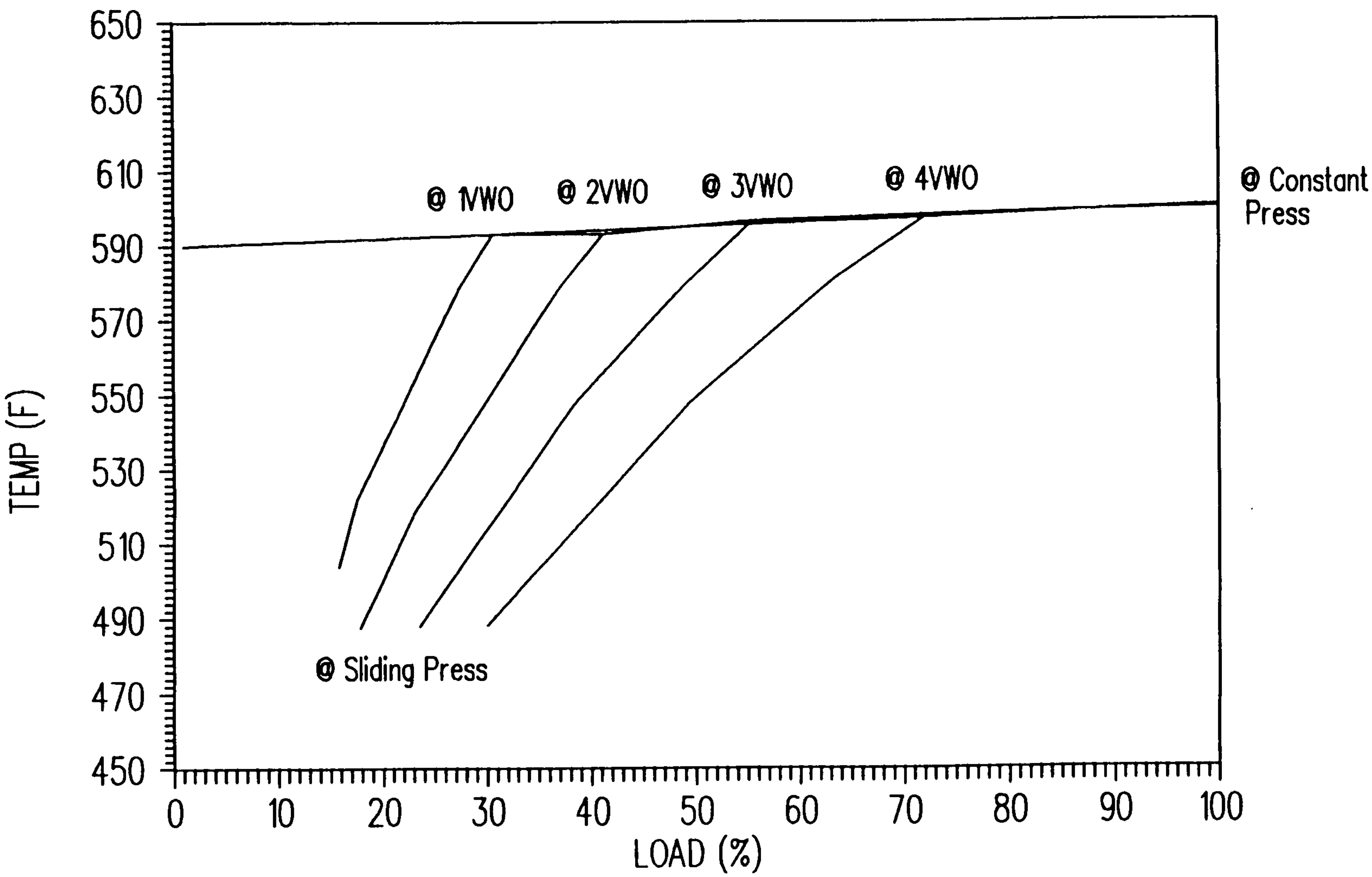
D12-04.ATB

D12 MAIN STEAM HEAT DUTY PROFILE
@ VARIABLE FW IN TEMPERATURE

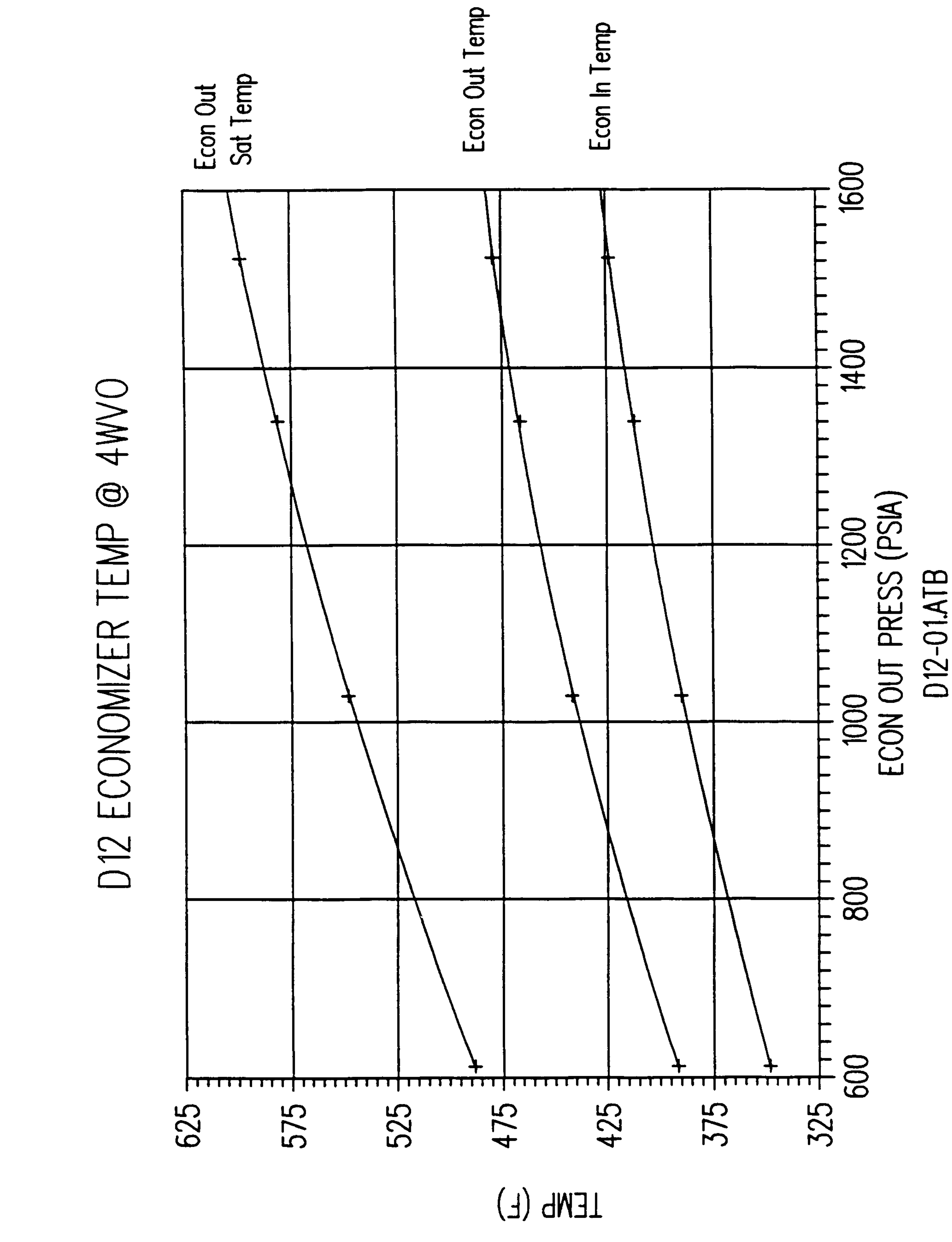


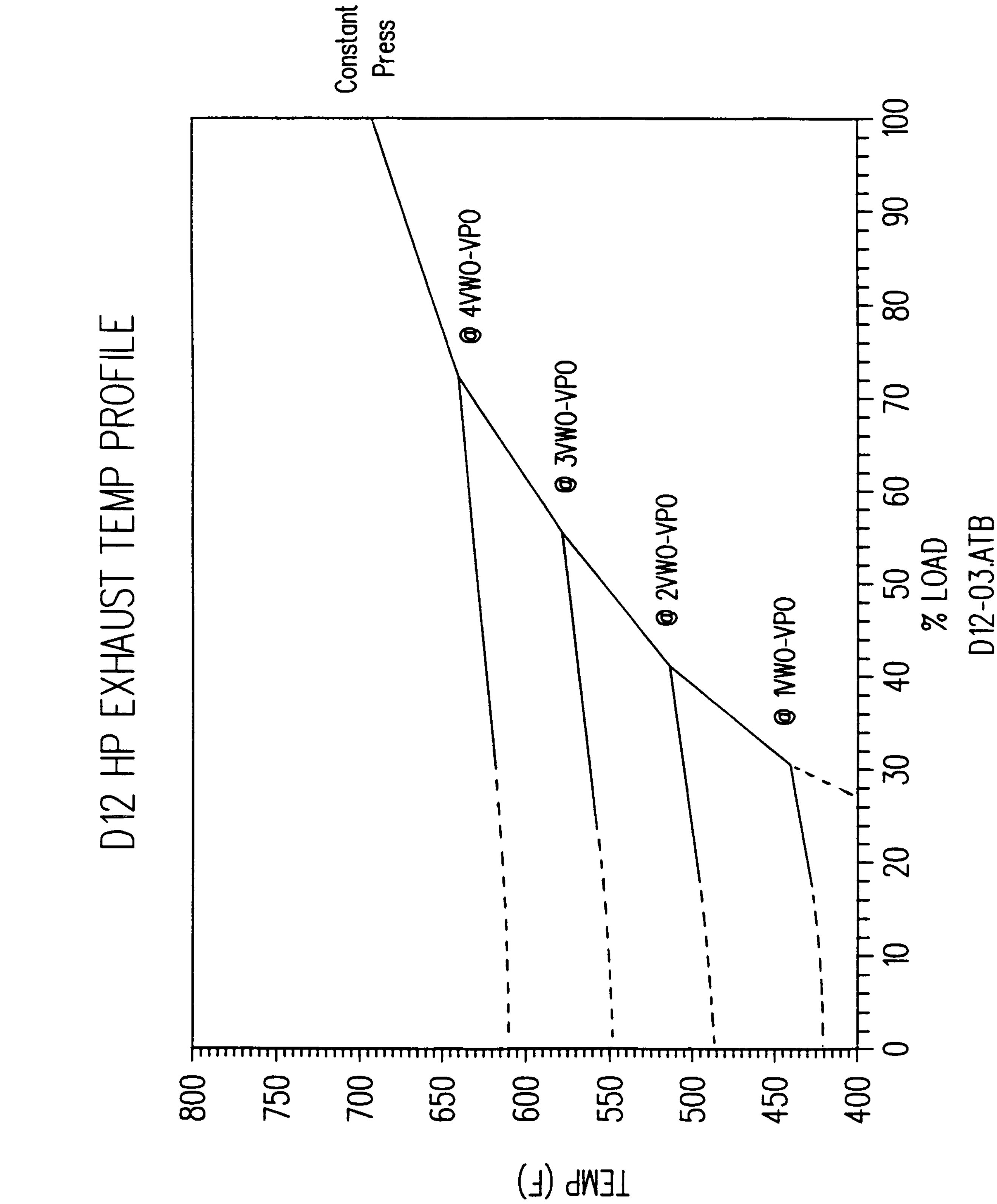
D12-10.ATB

D12 ECON OUT SATURATION TEMP PROFILE

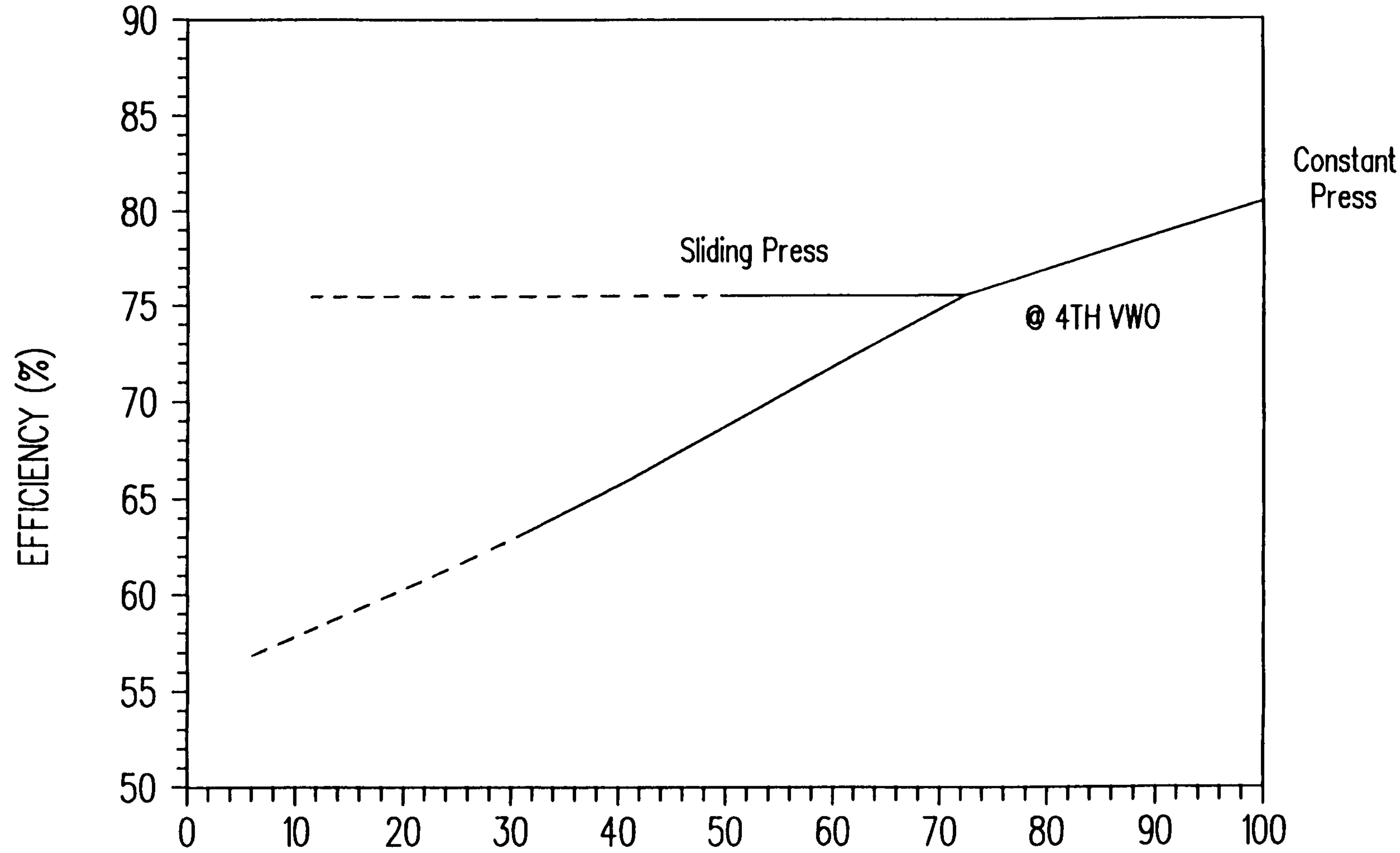


D12-02.ATB

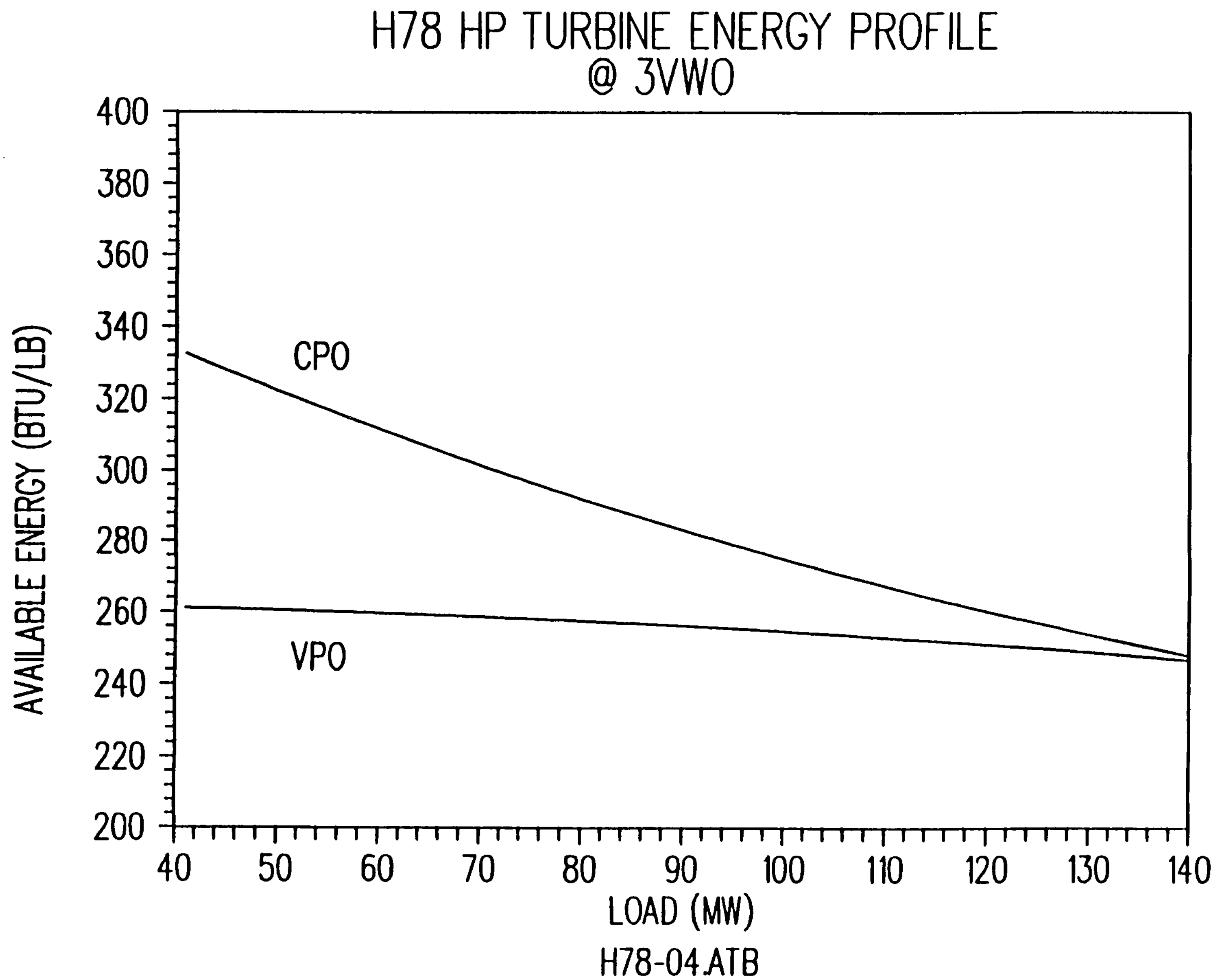

D12 ECONOMIZER TEMP @ 4WVO
TEMP (F)
625
575
525
475
425
375
325
600
800
1000
1200
1400
1600
ECON OUT PRESS (PSIA)
D12-01ATB
Econ Out
Sat Temp
Econ Out Temp
Econ In Temp

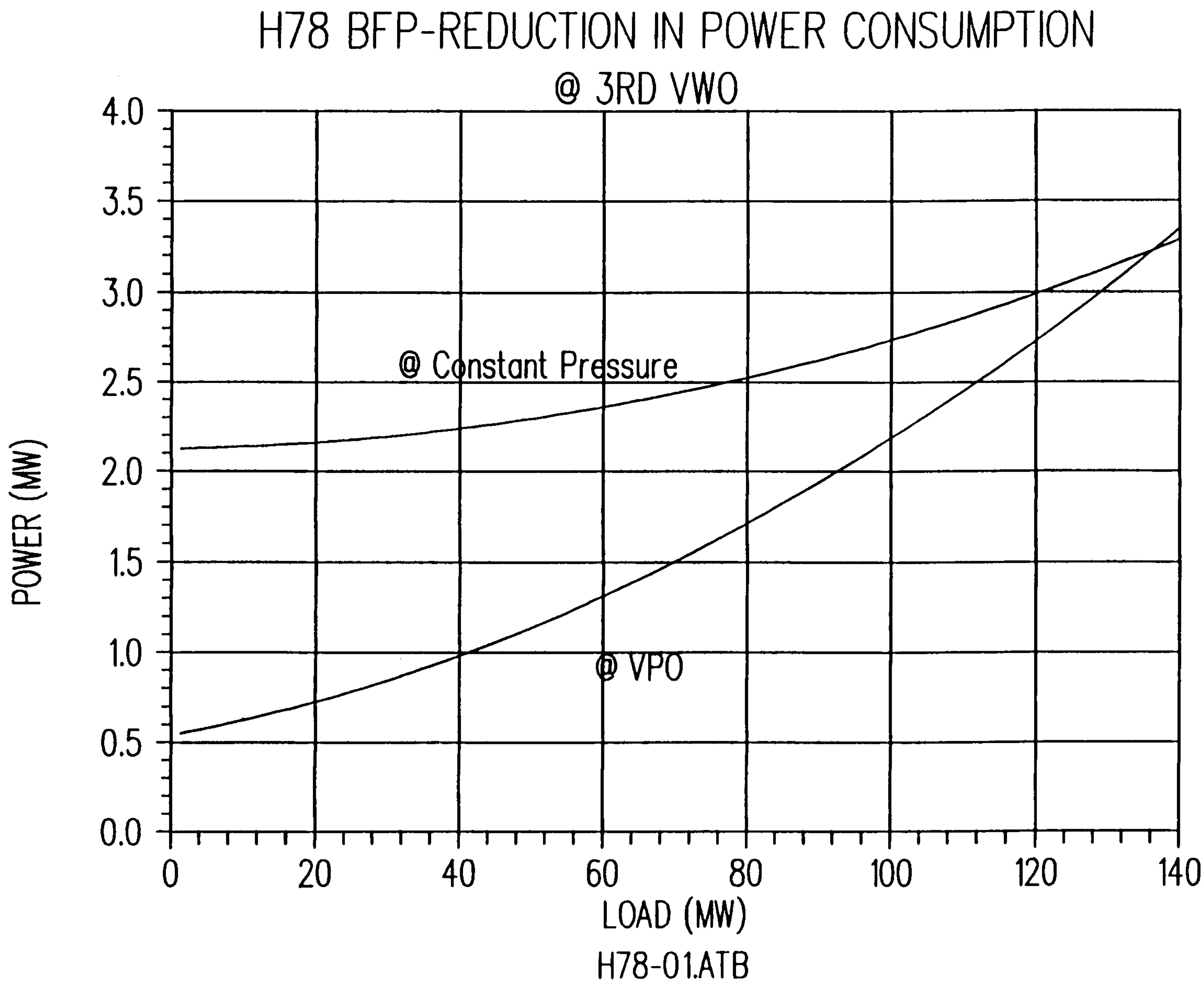

D12 HP EXHAUST TEMP PROFILE
TEMP (F)
800
750
700
650
600
550
500
450
400
0
10
20
30
40
50
60
70
80
90
100
% LOAD
D12-03.ATB
Constant Press
@ 4VWO-VPO
@ 3VWO-VPO
@ 2VWO-VPO
@ 1VWO-VPO

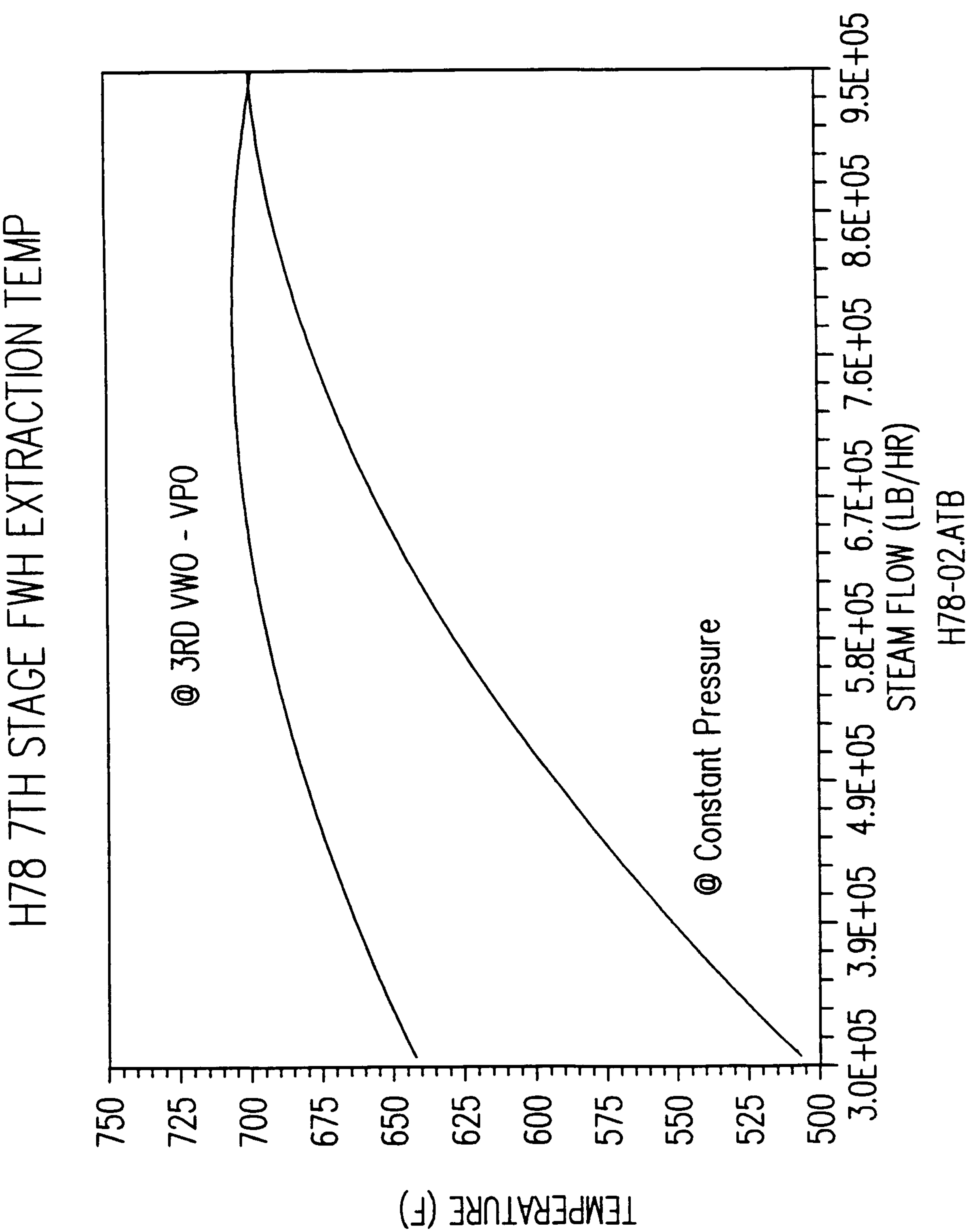
D12 HP TURBINE EFFICIENCY PROFILE

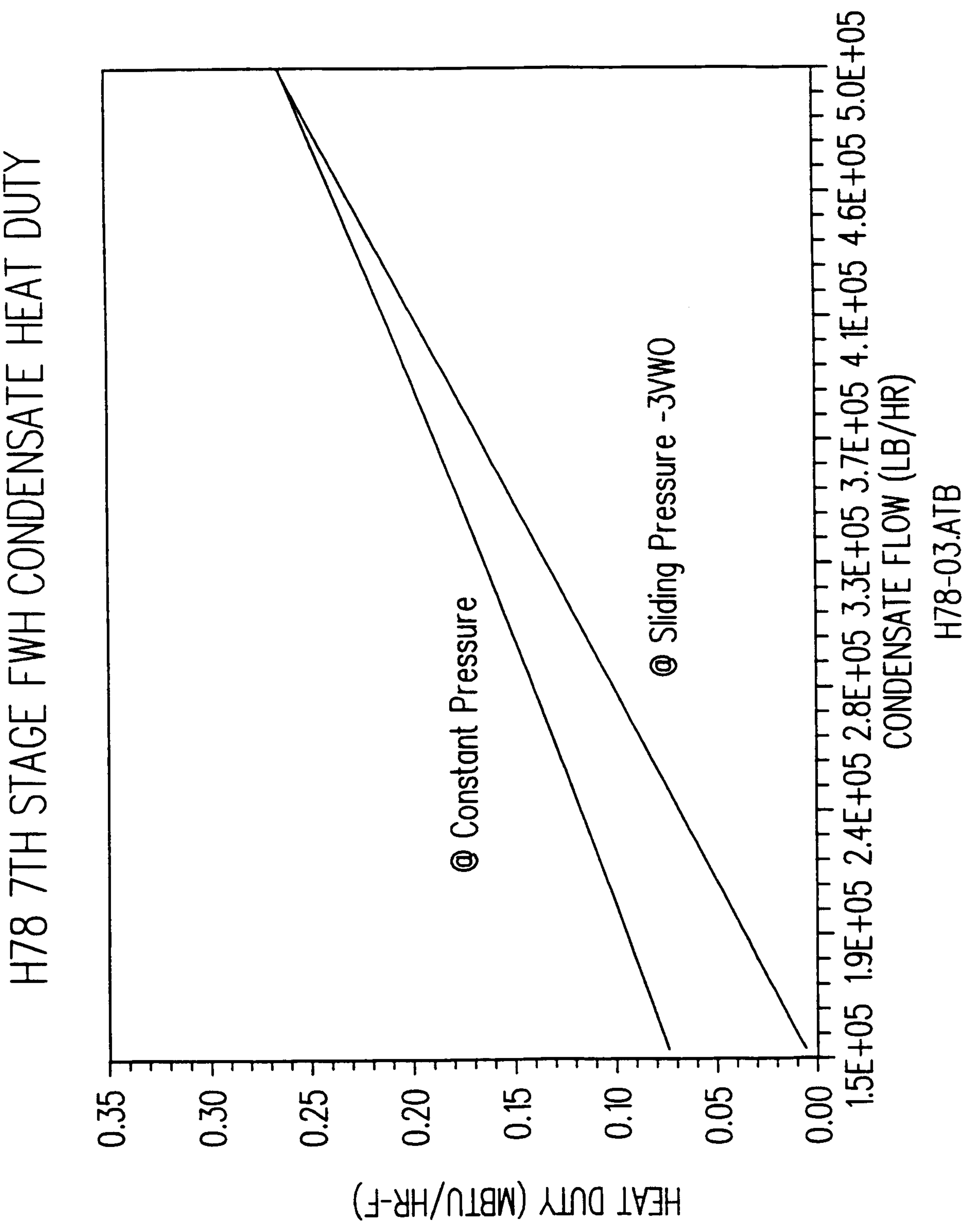


D12-09.ATB


H78 HP TURBINE ENERGY PROFILE
@ 3VWO
CPO
VPO
AVAILABLE ENERGY (BTU/LB)
400
380
360
340
320
300
280
260
240
220
200
40
50
60
70
80
90
100
110
120
130
140
LOAD (MW)
H78-04.ATB


H78 BFP-REDUCTION IN POWER CONSUMPTION
@ 3RD VWO
@ Constant Pressure
@ VPO
POWER (MW)
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
20
40
60
80
100
120
140
LOAD (MW)
H78-01.ATB


H78 7TH STAGE FWH EXTRACTION TEMP
@ 3RD VWO - VPO
@ Constant Pressure
TEMPERATURE (F)
750
725
700
675
650
625
600
575
550
525
500
3.0E+05
3.9E+05
4.9E+05
5.8E+05
6.7E+05
7.6E+05
8.6E+05
9.5E+05
STEAM FLOW (LB/HR)
H78-02.ATB


H78 7TH STAGE FWH CONDENSATE HEAT DUTY
HEAT DUTY (MBTU/HR-F)
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0.00
@ Constant Pressure
@ Sliding Pressure -3VWO
1.5E+05 1.9E+05 2.4E+05 2.8E+05 3.3E+05 3.7E+05 4.1E+05 4.6E+05 5.0E+05
CONDENSATE FLOW (LB/HR)
H78-03.ATB

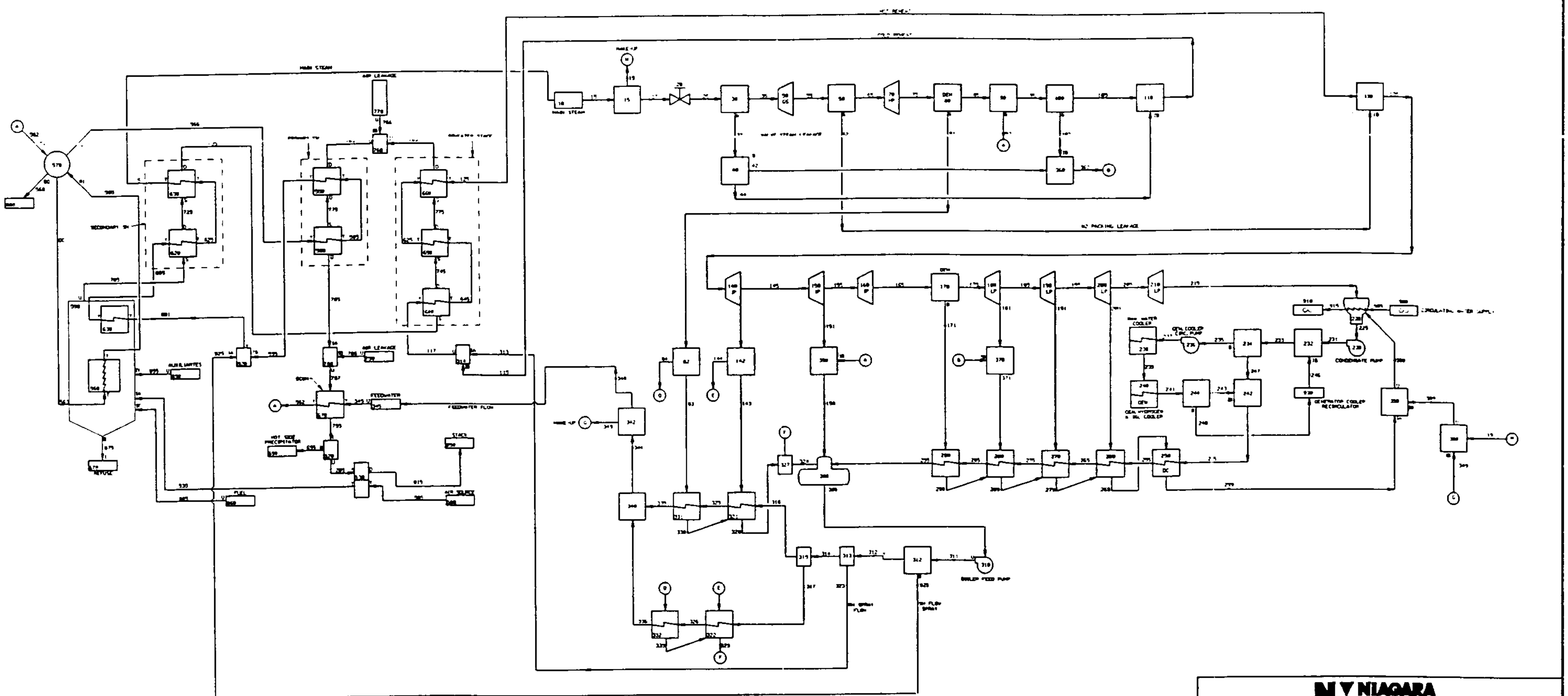


Figure 1

NIAGARA MOHAWK

NIAGARA MOHAWK POWER CORPORATION

DUNKIRK UNIT 1 & 2

TURBINE & BOILER CYCLE MODEL

PEPSE SCHEMATIC

DES.	DR.	CK.	DATE 3/31/94	SCALE NONE
APPROVED		APPROVED		INDEX
APPROVED		APPROVED		NO.

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