An Evaluation of Various Process Steam Models for an Electric Utility Retrofit

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ABSTRACT:

An Evaluation of Various Process Steam Models for An Electric Utility Retrofit

This paper discusses various process steam models and how PEPSE was used to evaluate their performances in a utility cycle retrofit application. Two different process steam scenerios: direct cycle extraction and indirect cycle extraction (using a heat exchanger reboiler mode) were evaluated in the cycle for 150 psia process steam application. The indirect cycle system used three different modes of application: single extraction with a single heat exchanger, a heat exchanger and drain cooler, and a dual extraction with various heat exchangers and drain coolers. Turbine ramifications were considered with retrofitting a utility turbine for this type of application.

BACKGROUND:

Commonwealth Edison Company with headquarters located in Chicago, Illinois is a large midwest utility having over 20,431 MW of generating capacity in Northern Illinois. Edison serves more than three million customers with an estimated population of eight million. The service area is 11,525 square miles which includes Chicago and 400 other incorporated communities. We have nine large nuclear units, having between 800 and 1100 MW capacity each, and three more units still under construction. Presently, Commonwealth Edison has 8,520 MWs of nuclear capacity. Those additional nuclear units will add another 3,360 MWs capacity by the end of 1988. Except for 1,331 MWs fast-start gas turbine peakers, the remaining 9,703 MWs is fossil production.

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Section I

INTRODUCTION

The process steam models presented here will function with any similar turbine design. In this particular case, Unit Seven at Crawford Generating Station, in Chicago, Illinois, was simulated to supply process steam using PEPSE. The study incorporates thermodynamic considerations. Several different process steam designs were considered leading to the conclusions and recommendations made. The study consists of the determination of the best possible thermodynamic design alternative.

When extraction steam is supplied for process steam purposes, turbine stage pressures downstream of the process extraction point are lower. This results in a lower feedwater temperature rise in each of the downstream feedheaters. More extraction steam upstream of the process extraction is needed to compensate for the lower incoming feedwater temperature.

The present approach involves using PEPSE to model Crawford Unit Seven turbine cycle along with eleven process steam models at four turbine valve point loads and process demand requirements. The discussion, together with supporting documentation drawn from the many computer runs for each cycle, draws a decision based on: thermodynamic considerations, engineering limitations, and practical comments. In the conclusion, recommendations are made for future study.

Section II

ENGINEERING CONSIDERATION

The following engineering considerations had to be addressed in the model design.

Heat Exchangers:

For a standard tube-in-shell design "reboiler" heat exchanger, the optimum design terminal temperature (TTD) should be between 40 and 80°F (22.2 and 44.4°C) to evaporate steam in the heat exchanger shell. The tube side condensing pressure used for this study was 245.0 psia (1.689 MPa). This corresponds to a 40°F (22.2°C) TTD for 150 psia steam.

Cool condensed extraction drainage should be routed to the condenser because it is the largest sink in the steam cycle. Hot extraction drainage should be returned to the deaerator. This limits increased maintenance problems caused by flashing, tube fretting, and erosion effects.

Boiler:

The boiler manufacturer suggested a five percent limit over normal feedheating needs for extra high pressure exhaust extraction. The velocity in the reheater tube banks must be maintained above a certain limit to prevent tube overheating. If beyond the limit, then the boiler would have to be redesigned with sections of the reheater tube banks removed. Such a modification would adversely affect unit efficiency. Secondly, boiler controls may not handle the temperature extremes brought about by the extra extraction. Added to these problems is the necessity to fire harder due to lower feedwater temperatures from the lower high pressure turbine extraction pressures. It may be easier to use hot reheat extraction. This may necessitate a desuperheating station in some cases.

Turbine:

The turbine manufacturer also suggests a limit on how much external extraction steam can be removed without causing damage. The main concern with a retrofit design system is the blade stress caused by increased stage expansion immediately upstream of the extraction point. As more steam is extracted, turbine stage shell pressures are reduced resulting in greater expansion across the stage. Turbine blades do more work (at a lower efficiency) resulting in excessive bucket loading. Unlike impulse stages (primarily the governing stage), the remaining stages are primarily reaction design and cannot take increased pressure drops across them. This leads to cracking at the blade (buckets) roots requiring extensive turbine repairs.

Another related problem deals with excessive force transmitted through the blades to the steam turbine rotor. Most of this extra force is transmitted as torque with some additional resulting thrust. The rotor is designed for limited torsonial stressing and may not be adequately designed to handle the increased bucket loading. This could result in premature rotor cracking and disintegration.

Operation with excessive extraction rates is not recommended since such operation encroaches upon design margins. The degree of encroachment on design margin is increased with additional extraction. Because of these concerns, one requirement which must be addressed prior to the installation of a process steam retrofit configuration is the complete analysis of all turbine components. Such a determination is beyond the scope of this study and would be best accomplished by the turbine manufacturer.

Care also must be taken to insure that the system does not increase any water induction potential probability. This is especially true when units are cross-tied together to supply steam to the same system. It is not recommended to use two units tied together at the same time to supply extraction steam. Care must also be taken when switching from one unit to the other.

Section III

PROCESS STEAM MODELS

This discussion will focus on the design and description of the different process steam models. The eleven computer models represent a wide variety of systems that can be retrofitted to an existing utility steam turbine cycle. The models vary in complexity, but achieve the same effect in the end: typically the generation of 150 psia (1.034 MPa) saturated or superheated process steam. The chief purpose behind experimenting with these models was to evaluate and determine the most acceptable choice for retrofitting Crawford Unit Seven based upon First and Second Law analyses. Before proceeding, a simplified description of the turbine cycle extraction arrangement for Crawford Unit Seven will be given. Hopefully, this will provide clarity as to how the process steam models fit into the turbine cycle arrangement.

The turbine is designed for throttle conditions at 2015 psia (13.893 MPa) and 1050°F (565.6°C) superheat, reheat at 1050°F (565.6°C) and exhaust at 1.0" Hg (3.385 kPa) absolute backpressure. The turbo-generator combination design rating is 200,000 kW(e). The turbine is a tandem compound unit, and has a triple flow, low pressure section (TCTF). Five individual turbine segments make up the tandem compound arrangement: the high pressure turbine (includes the governing stage), the reheat turbine, the intermediate pressure turbine, a single flow low pressure turbine, and a double opposed flow low pressure turbine. The sections are bolted together in the same manner listed above.

The turbine is designed with seven turbine extraction ports for feedwater heating. Three of these extractions (the first, second, and fourth) are at the exhaust ends of the first three turbine sections. The other four turbine extractions are located on the intermediate pressure or double flow low pressure turbine sections. The first three turbine extractions supply steam to the high pressure heaters above the deaerating heater, while the last three extractions supply the low pressure heaters below the deaerator. The intermediate turbine exhaust extraction supplies the deaerating heater. The feedwater can be heated to as high as 486.8°F (252.7°C) at the maximum throttle flow: 1,523,987 lbm/hr (691,281 kg/hr), with all feedwater heaters in service.* A simplified line diagram of Crawford Unit Seven is shown in Figure III-1 with an indirect process steam generation system attached.

The first four extraction points have good potential when designing and evaluating process steam systems. Because of the 150 psia (1.034 MPa) process steam requirement, the first extraction point is the only single acceptable source of steam for indirect process steam generation over the entire turbine load range. The second extraction, at the reheat turbine exhaust, is acceptable for a 150 psia (1.034 MPa) direct extraction system. The other extractions together with the first, could be utilized in a multiple extraction system, or individually for lower pressure designs.

Additionally, hot reheat steam could be used as well.

Design Alternatives

One of the purposes of the study was to compare the efficiencies of various process steam retrofit models as applied to this unit. Each model utilizes various configurations of heat exchangers, drain coolers, extraction

*Assuming no extra extraction demand.

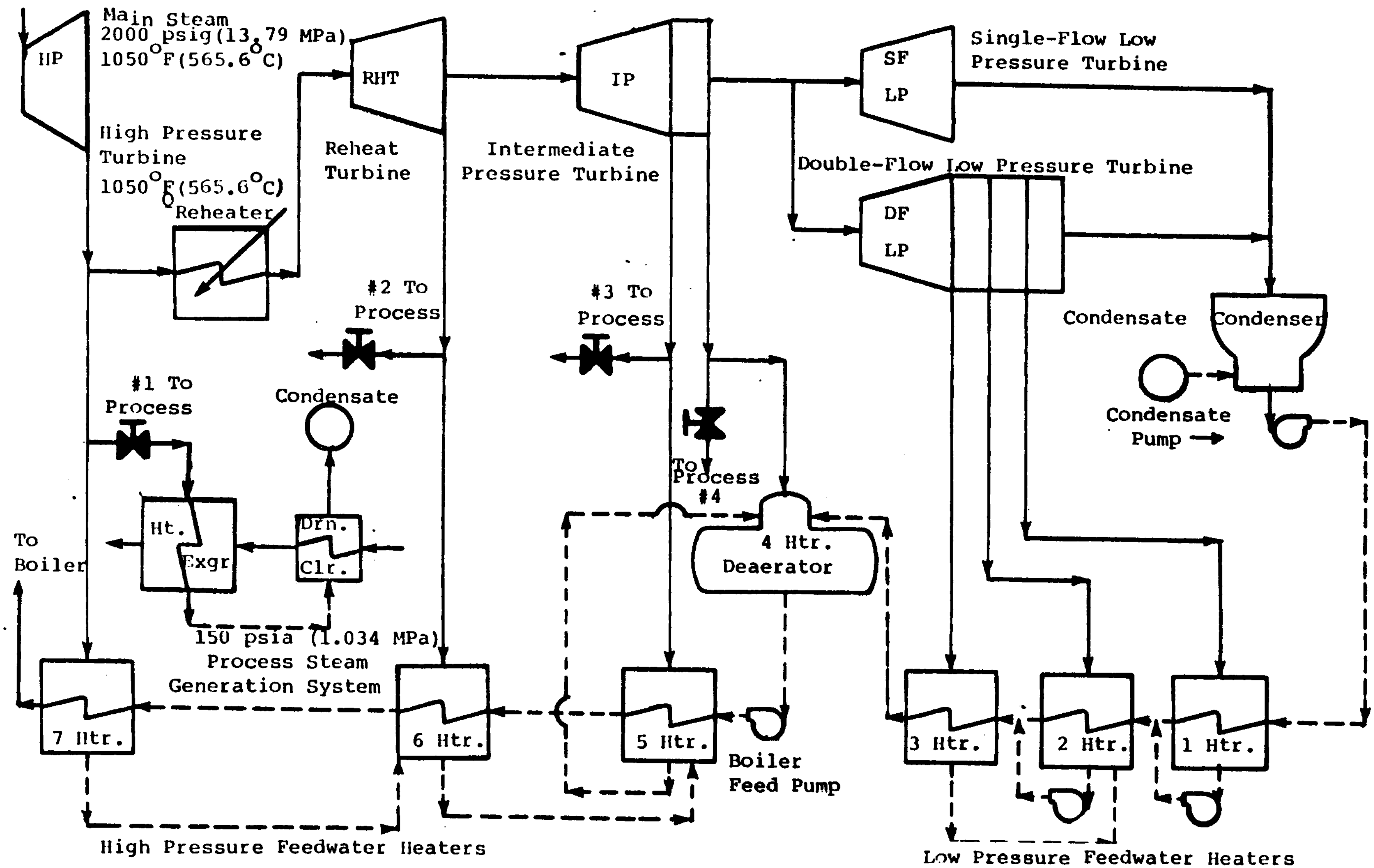

Figure III-1. Simplified Line Diagram of Crawford Unit Seven with Process Circuit Attached lines, and drainage locations hypothetically incorporated into the existing steam turbine cycle of Unit Seven at Commonwealth Edison Company's Crawford Generating Station (Chicago, Illinois). The possible model design alternatives considered by the study are classified three ways. Figure III-2 shows the logical development of the models listed in Table III-1. Refer to Figure III-1 to locate extraction points in the cycle.

The models were developed allowing for the comparison of absolute efficiencies while generation process steam at 150 psia (1.034 MPa), saturated, and at superheated 10 - 60°F (5.6 - 33.3°C) steam conditions. The Class II models proved to be the most efficient configuration requiring the lowest amount of available energy and least replacement power. Most of the modeling and analysis efforts were spent on these models. In addition the class II (single extraction models proved to be more efficient than one class III model (multiple-extraction) considered.

Before any of the eleven process steam models could be incorporated into the PEPSE model, the basic cycle was compared against vendor and acceptance test heat balances done at six turbine valve point loads. This determined whether the turbine cycle was modeled properly. The average deviation between computed and either vendor or actual test gross turbine heat rate balances is approximately 0.5%. The deviation on the gross turbine generator load was also less than 0.5%. The total system mass flow convergence was set at one pound mass.

The eleven basic process steam cycles were evaluated at four valve points. The associated generator nominal loads for Crawford Station's Unit Seven are: 163 MW(e), 200 MW(e), 220 MW(e), and 238 MW(e). Four major process steam requirements were chosen: 24,000 lbm/hr (10,886 kg/hr), 48,000 lbm/hr (21,773 lbm/hr), 96,000 lbm/hr (43,546 kg/hr), and 192,000 lbm/hr (87,091 kg/hr).

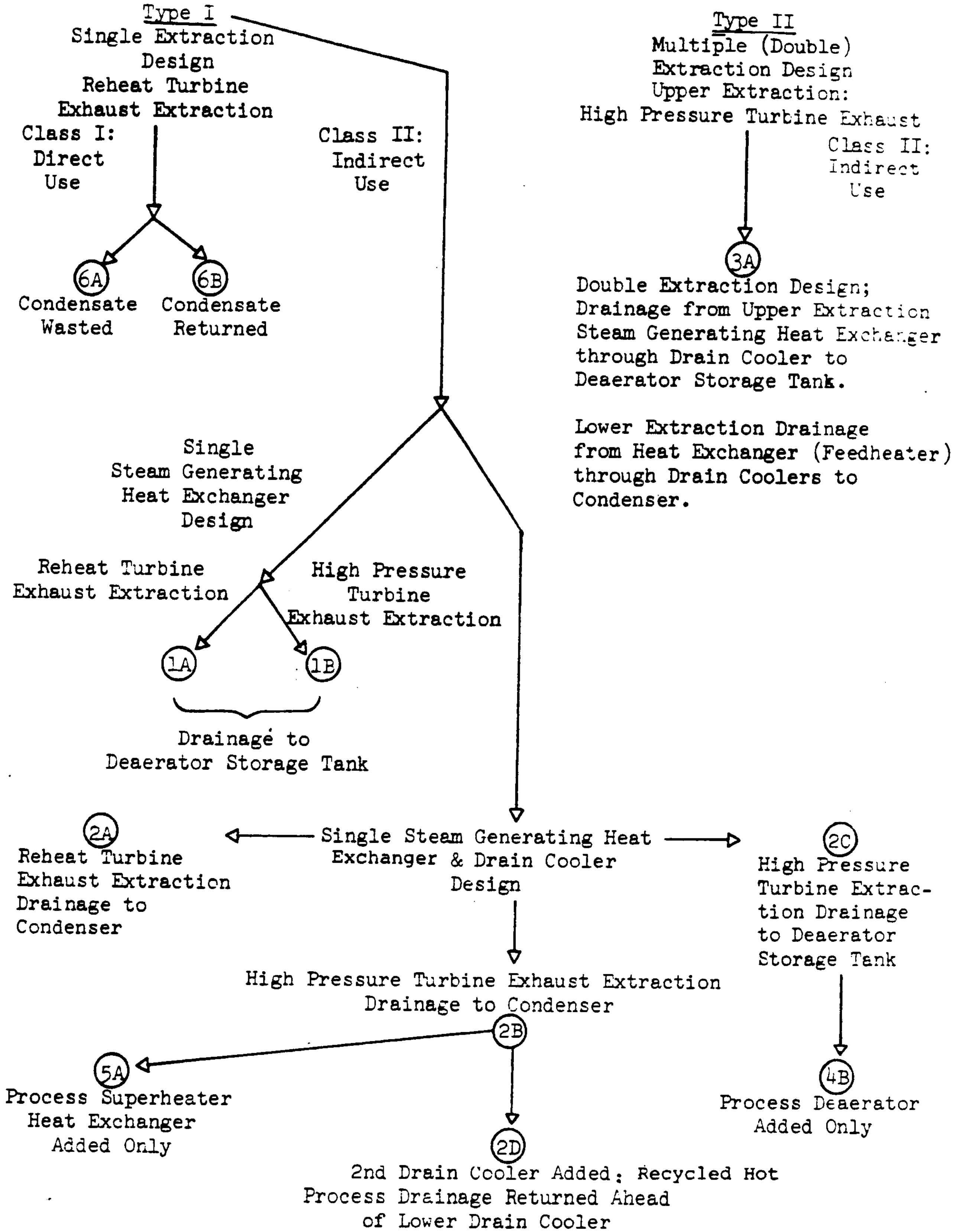

Figure III-2. Tested Process Steam Models Tree Structure

Table III-1 Short Summary of Various Process Steam Models Tested

Model or Class	Description
Class I	Direct application, extraction steam directly used as process steam.
Model 6A	Reheat turbine exhaust extraction, condensate wasted.
Model 6B	Reheat turbine exhaust extraction, condensate returned to deaerating heater.
Class II	Indirect application; extraction steam drives steam generating process heat exchangers.
Model 1A	Reheat turbine exhaust extraction without drain cooler, condensate returned to deaerating heater (model no good because available pressure is too low at most electrical generation loads).
Model 1B	Cold reheat extraction, no drain cooler, condensate returned to deaerating heater.
Model 2A	Reheat turbine extraction with drain cooler, condensate returned to deaerating heater (model no good because available pressure is too low at most electrical generation loads).
Model 2B	Cold reheat extraction, with drain cooler, condensate returned to main condenser.
Model 2C	Cold reheat extraction with drain cooler, condensate returned to deaerating heater.
Model 2D	Cold reheat extraction with two drain coolers, condensate returned to main condenser, with variations of process condensate being returned.
Model 4B	Cold reheat extraction with drain cooler and additional process deaerator, condensate returned to unit deaerator.
Model 5A	Cold reheat extraction with drain cooler and superheater, condensate returned to main condenser.
Class III	Indirect application; multiple extraction driven steam generators and feedwater heater exchangers.
Model 3A	Cold reheat and intermediate pressure turbine exhaust extractions are used in the steam generator and feedheater respectively. Drainage from cold reheat routed through a drain cooler ahead of feedheater then into the unit deaerator. Intermediate extraction routed through two cascading drain coolers behind feedheater, then into condenser.

Model Description

Model 1A is supplied with reheat turbine extraction, routing the drainage from the steam generating heat exchanger to the deaerating heater. The cold process feedwater is assumed for all models to be 45°F (7.2°C). For Models 1A and 1B condensate is fed directly into the evaporator with no preheating. Process Steam Model 1B, as shown in Figure III-4 and Table III-2 is a variation of Model 1A employing high pressure turbine exhaust steam. The model was extensively tested throughout the study.

Model 2A and 2B have an additional drain cooler component added after the steam generating heat exchanger. More heat is removed by the drain cooler from the turbine extraction drainage, after leaving the steam generating heat exchanger. Unlike Models 1A and 1B, the cold process feed is preheated before it enters the heat exchanger.* Model 2A uses reheat turbine extraction as the heat source while in Model 2B cold reheat (high pressure turbine exhaust) is utilized. Both models have extraction drainage from the drain cooler routed to the condenser. The drain cooler, with a 10°F (5.6°C) drain cooler approach, lowers the extraction steam drainage to 55°F (12.8°C). A simplified line drawing of process steam Model 2B and stream properties are shown in figure III-5 and Table III-3.

Model 2C, a variation of Models 2A and 2B, involves routing the extraction drainage from the external drain cooler to the direct-contact (deaerating heater) storage tank. However, only about half as much heat (compared to Model 2A and 2B) is transferred by the drain cooler to the process feed before

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*Using a 5°F (2.8°C) drain cooler approach (DCA) the process feed can be heated to approximately 353°F (178°C) before the water enters the steam generating heat exchanger. Good energy cascading techniques would not be followed.

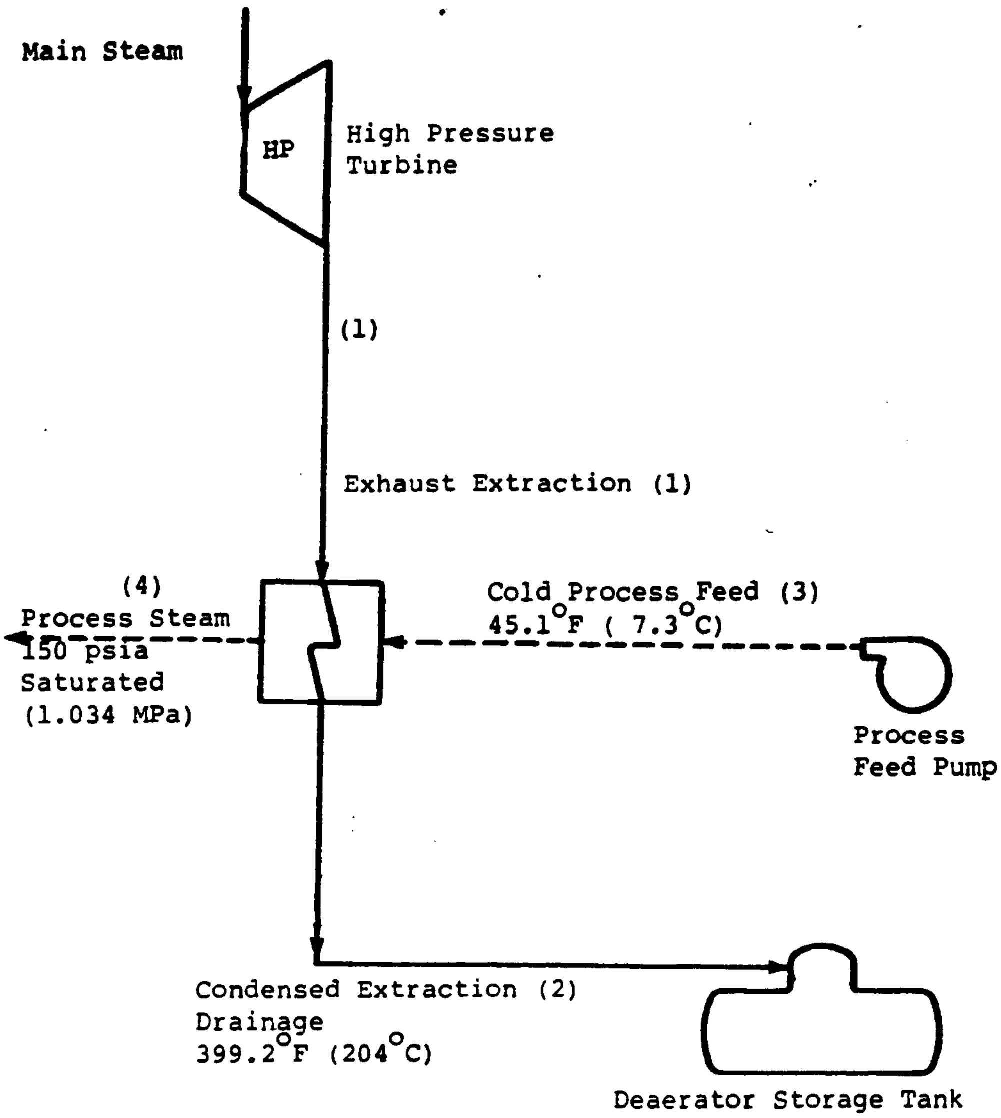

Figure III-4. Line Diagram of Process Steam Model 1B

Table III-2. Stream Properties at Standard Conditions: Model 1B

Stream Description and State Number (X)	Temperature °F (°C)	Pressure psia (MPa)	Enthalby B/lbm(kJ/kg)
Exhaust Extraction (1)	600-800 (316-427)	590-260 (4.07-1.79)	≈ 1340 (3116.8)
Condensed Drainage (2)	399.2(204)	245.0(1.689)	374.3(870.6)
Process Feed (3)	45.1(7.3)	160.0(1.103)	13.6(31.6)
Process Steam (4)	358.5(181.4)	150.0(1.034)	1194.1 (2777.5)

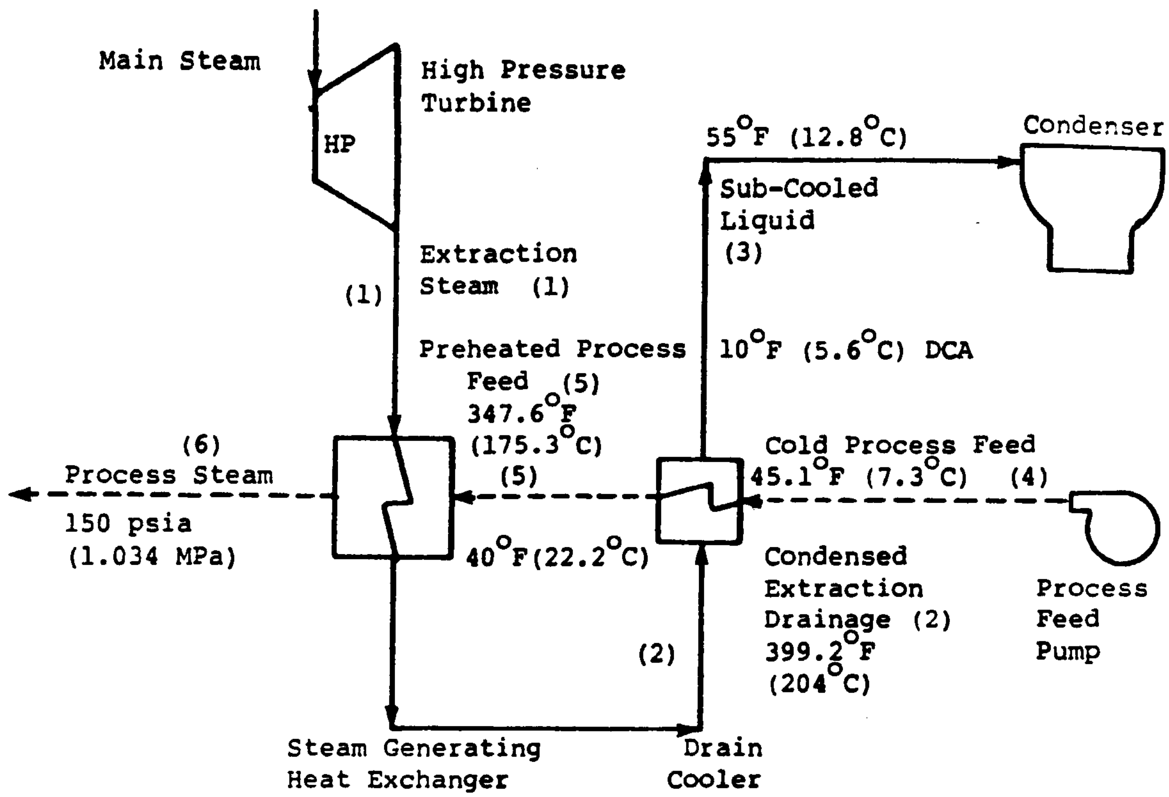

Figure III-5. Line Diagram of Process Steam Model 2B

Table III-3. Stream Properties at Standard Conditions: Model 2B

Stream Description and State Number (X)	Temperature °F (°C)	Pressure psia (MPa)	Enthalpy B/lbm(kJ/kg)
Extraction Steam (1)	600-800 (316-427)	590-260 (4.07-1.79)	≈1340 (3116.8)
Condensed Drainage (2)	399.2(204)	245.0(1.689)	374.3(870.6)
Sub-Cooled Liquid (3)	55(12.8)	30.0(0.206)	23.3(54.2)
Cold Process Feed (4)	45.1(7.3)	170.0(1.172)	13.7(31.9)
Preheated Process Feed (5)	347.6(175.3)	160.0(1.103)	319.5(743.2)
Process Steam (6)	358.5(181.4)	150.0(1.034)	1194.1 (2777.5)

entering the steam generator. A simplified line drawing of Model 2C and stream properties are shown in figure III-6 and Table III-4. A limited number of computer runs were made with Model 2C, but not pursued further.

Another variation, process steam Model 2D, involved returning a portion of the process condensate drainage back to the cycle. It employs two drain coolers as shown in Figure III-7 and Table III-5. The drain coolers are sized differently because the lower one handles the cold make-up, while the upper one carries the full process feed.

Model 3A examined possibility of the multiple turbine extraction design. The model shown in Figure III-8 and Table III-6 employs high pressure turbine extraction steam for evaporating the steam, while intermediate turbine exhaust preheats the cold condensate feed. The high pressure extraction drainage from the steam generator is routed through a drain cooler before being emptied into the deaerating heater storage tank. After leaving the feedheater the intermediate turbine exhaust extraction drainage is routed through the series drain coolers, before being emptied into the condenser.

The process feed enters the primary heat exchanger in an almost saturated liquid state. The secondary feedwater heat exchanger a has 5°F (2.8°C) terminal temperature difference. The two separate (series) drain coolers each have one-half the preheating load of the drain cooler in Model 2B and have a 100°F (56.6°C) and 10°F (5.6°) drain cooler approach.

Model 3A and Model 4B (the next one to be discussed) are by far the most complex of all the models considered in this study. As do the other models, they both employ the cascading energy design concept which attempts to associate similar available energy levels, thereby reducing the amount of lost available energy. The class III models could have been extended to as much as a quadruple extraction system. However, no more than two extractions were

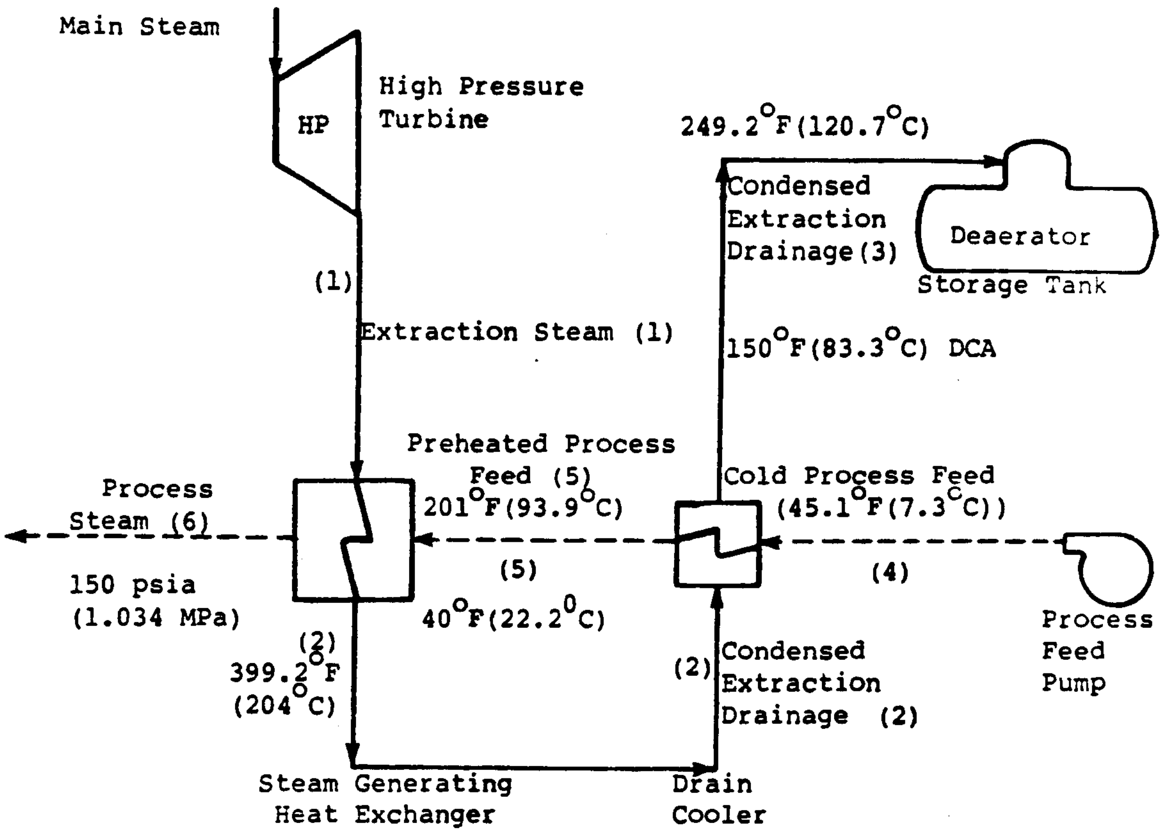

Figure III-6. Line Diagram of Process Steam Model 2C

Table III-4. Stream Properties at Standard Conditions: Model 2C

Stream Description and State Number (X)	Temperature °F (°C)	Pressure psia (MPa)	Enthalpy B/lbm (kJ/kg)
Extraction Steam (1)	600-800 (316-427)	590-260 (4.07-1.79)	≈1340 (3116.8)
Condensed Drainage (2)	399.2(204)	245.0(1.689)	374.3(870.6)
Drainage to Deaerator (3)	249.2(120.7)	80.0(0.552)	218.7(508.7)
Cold Process Feed (4)	45.1(7.3)	170.0(1.172)	13.7(31.9)
Preheated Process Feed (5)	201(93.9)	160.0(1.103)	169.2(393.6)
Process Steam (6)	358.5(181.4)	150.0(1.034)	1194 (2777.5)

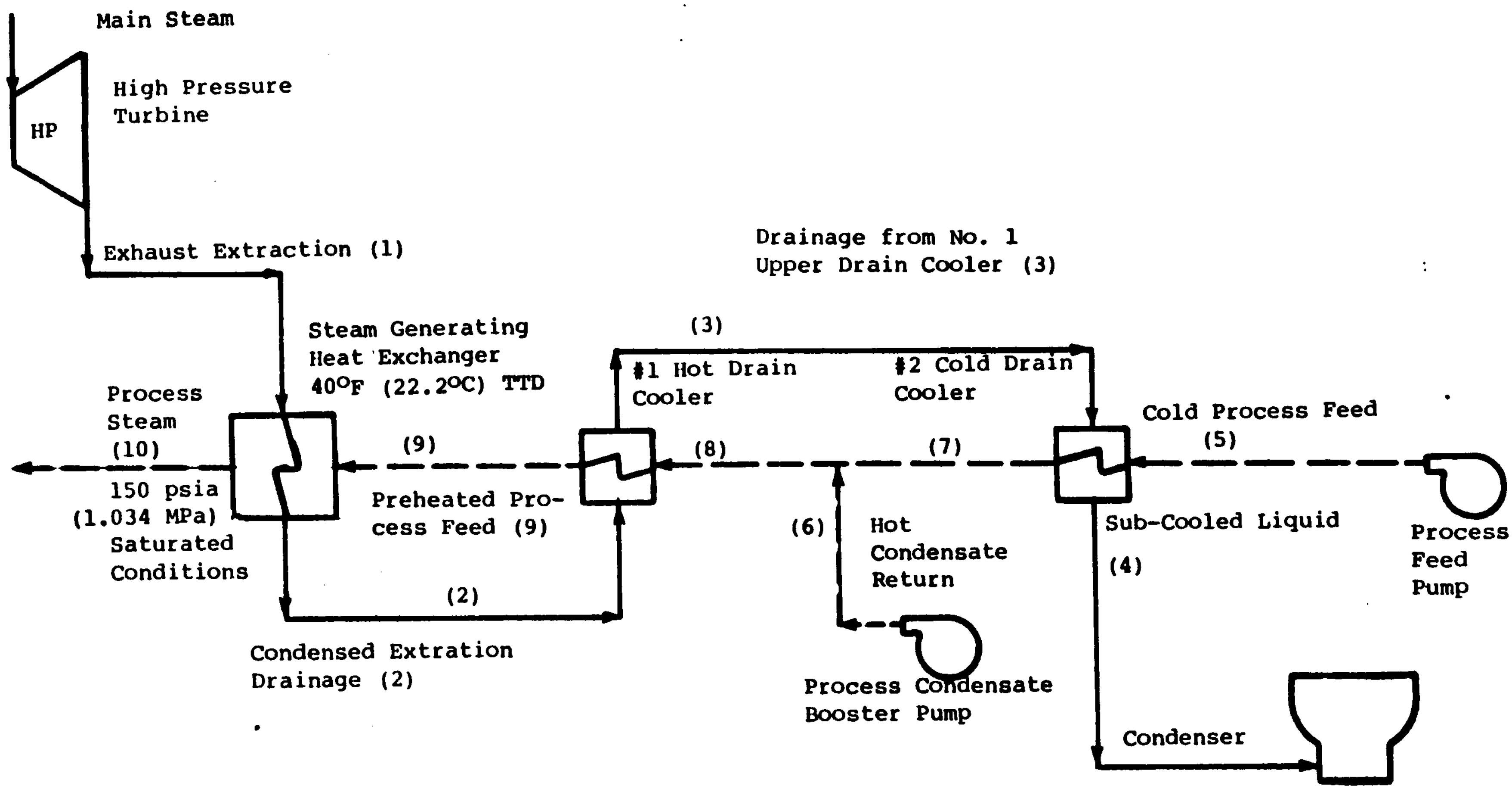

Figure III-7. Line Diagram of Process Steam Model 2D

Table III-5. Stream Properties at Standard Conditions: Model 2D

Stream Description and State Number (X)	Temperature °F (°C)	Pressure psia (MPa)	Enthalpy B/lbm (kJ/kg)
Extraction Steam (1)	600-800 (316-427)	590-260 (4.07-1.79)	≈1340 (3116.8)
Condensed Drainage (2)	399.2 (204)	245.0 (1.689)	374.3 (870.6)
Drainage from No. 1 (3) Upper Drain Cooler	142 (61.1)	80.0 (0.552)	110.2 (256.3)
Drainage from No. 2 (4) Lower Drain Cooler	55.0 (12.8)	30.0 (0.207)	23.3 (54.2)
Cold Process Feed (5)	45.1 (7.3)	180.0 (1.241)	13.8 (32.1)
Returned Process Feed (6)	132 (55.6)	170.0 (1.172)	100 (232.6)
From No. 1 Drain (7) Cooler	132 (55.6)	170.0 (1.172)	100 (232.6)
After Mixing (8)	132 (55.6)	170.0 (1.172)	100 (232.6)
Preheated Process Feed (9)	347.6 (175.3)	160.0 (1.103)	319.5 (743.2)
Process Steam (10)	358.5 (181.4)	150.0 (1.034)	1194.1 (2777.5)

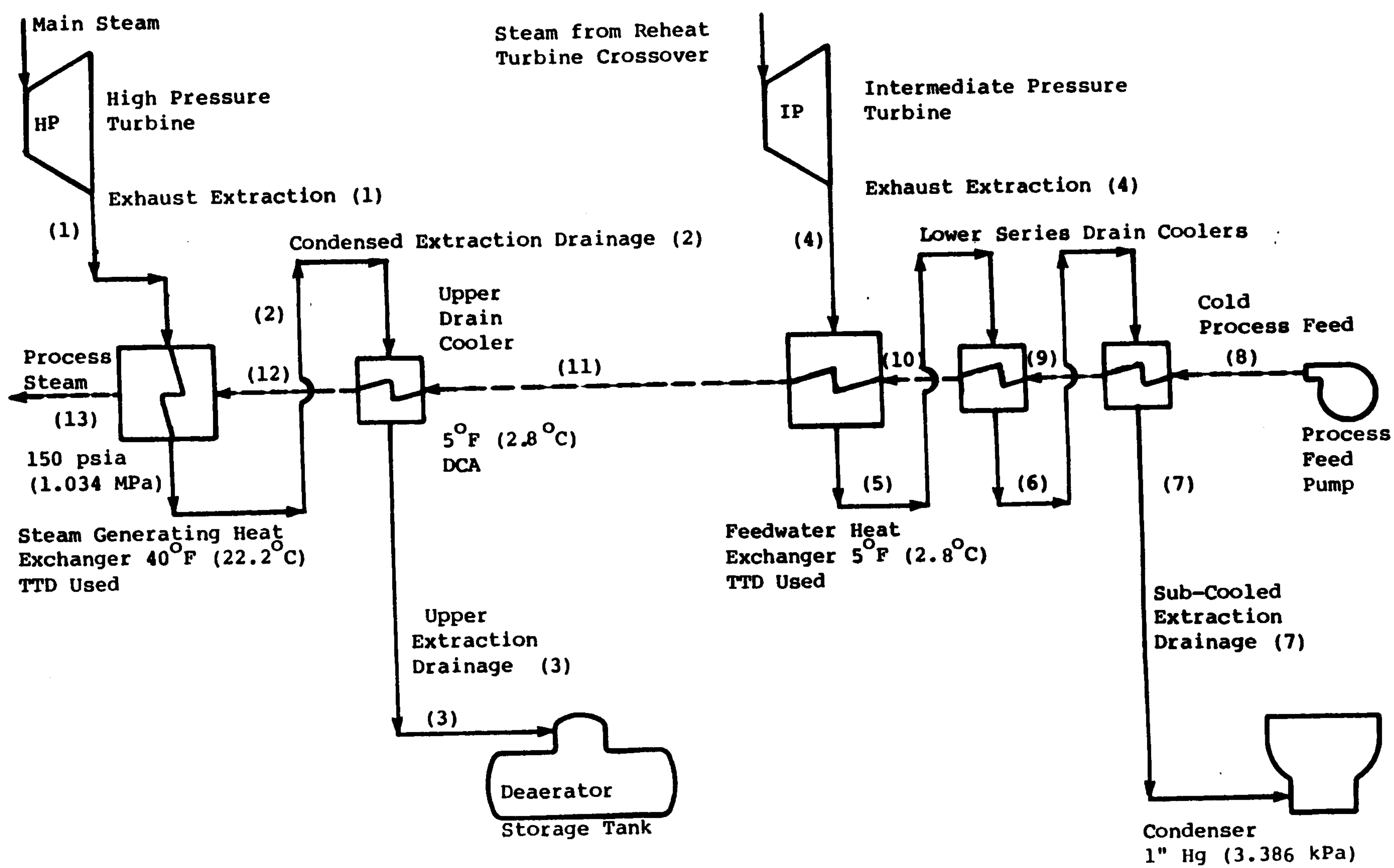

Figure III-8. Line Diagram of Process Steam Model 3A

Table III-6. Stream Properties at Standard Conditions: Model 3A

Stream Description and State Number (X)	Temperature °F (°C)	Pressure psia (MPa)	Enthalpy B/lbm (kJ/kg)
Extraction Steam (1)	600-800 (316-427)	590-260 (4.07-1.79)	≈1340 (3116.8)
Condensed Drainage (2)	399.2(204)	245.0(1.689)	374.3(870.6)
Drainage from No. 1 (3) Upper Drain Cooler	286.8(141.6)	110.0(0.758)	256.2(595.9)
Second Extraction (4) Steam	500-600 (260-316)	55-25 (0.379-0.172)	≈1291 (3003)
Condensed Drainage (5)	281.8(138.8)	55-25 (0.379-0.172)	251(584)
Drainage from No. 2 (6) Upper Series Drain Cooler	165.4(74.1)	20.0(0.138)	133.5(310.5)
Drainage from No. 3 (7) Lower Series Drain Cooler	55.1(12.8)	15.0(0.1034)	23.2(54.0)
Cold Process Feed (8)	45.1(7.3)	200.0(1.379)	13.8(32.1)
Feed from No. 3 (9) Drain Cooler	65.4(18.6)	190.0(1.310)	34.0(79.1)
Feed from No. 2 (10) Drain Cooler	87.0(30.6)	180.0(1.241)	55.5(129.1)
Feed from Feedheater (11)	276.8(136.0)	170.0(1.172)	246.1(572.4)
Feed from Upper (12) Drain Cooler	358.7(181.5)	160.0(1.103)	347.4(808.1)
Process Steam (13)	358.4(181.3)	150.0(1.034)	1194.1 (2777.5)

needed because the amount of steam required from the third and fourth extraction offer diminishing available energy returns. The extra investment would not be justified by present process steam demand requirements.

As shown in Figure III-9 and Table III-7, Model 4B is based on Model 2D discussed earlier. The difference is a direct contact heater has been inserted in between the steam generating heat exchanger and drain cooler. After the preheated process feed leaves the drain cooler, it enters into a deaerating heater supplied with process steam from the discharge line off the steam generating heat exchanger. The drainage from the process deaerator storage tank is then routed to a second process feed pump that discharges it into the steam generating heat exchanger. The turbine extraction drainage, after leaving the drain cooler, is emptied into the deaerator storage tank.* The second process feed pump must be designed with sufficient suction head to prevent boiling in the deaerator drain outlet line.

Model 5A is actually Model 2B, with a separate process steam superheater added before the steam generator. The high pressure turbine exhaust extraction serves as the heat force for the superheater. The extraction steam before entering the steam generator is routed through the superheater. From the heat exchanger shown in Figure III-10 and Table III-8, the drainage before entering the condenser, flows through the drain cooler preheating the cold incoming process feed.

For the other models studied the process steam was superheated by increasing the tube side condensing pressure. In addition, the process steam enthalpy was raised the corresponding amount without violating the 40°F (22.2°C) design constraint. In a real installation, additional heat exchanger

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*Thus with a DCA of this magnitude, the drainage was at a high enough energy level (195°F(90.6°C)) to be routed into the unit deaerator storage tank.

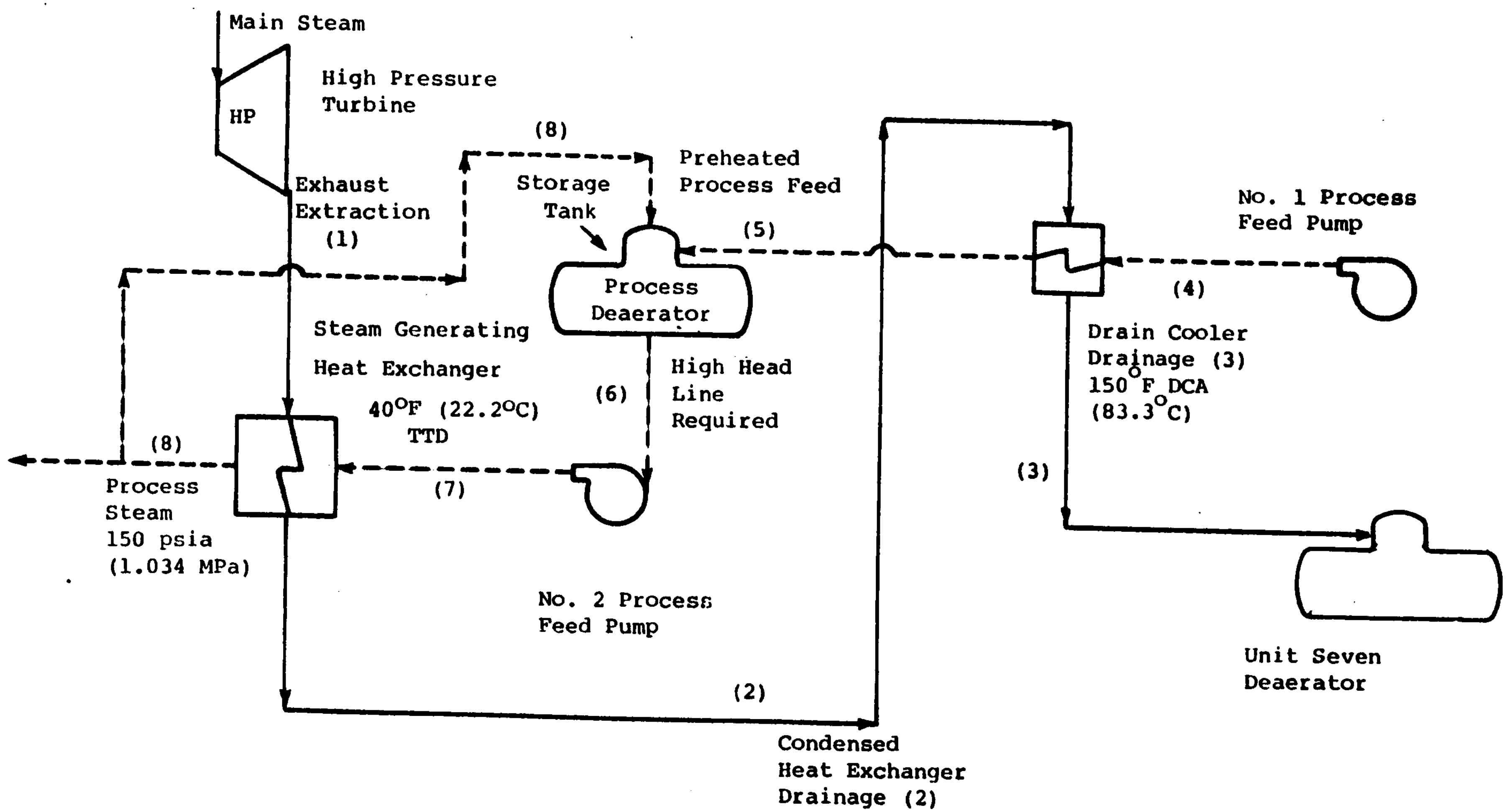

Figure III-9. Line Diagram of Process Steam Model 4B

Table III-7. Stream Properties at Standard Conditions: Model 4B

Steam Description and State Number (X)	Temperature °F (°C)	Pressure psia (MPa)	Enthalpy B/lbm (kJ/kg)
Exhaust Extraction (1) Steam	600-800 (316-427)	590-260 (4.07-1.79)	≈1340 (3116.8)
Condensed Drainage (2)	399.2 (204)	245.0 (1.689)	374.3 (870.6)
Sub-Cooled Liquid (3)	195.1 (90.6)	80.0 (0.552)	163.4 (380.1)
Cold Process Feed (4)	45.1 (7.3)	200.0 (1.379)	13.8 (32.1)
Preheated Process (5) Feed	254.1 (123.4)	180.0 (1.241)	223.1 (518.9)
Deaerator Drain (6) Inlet	358.4 (181.3)	150.0 (1.034)	330.7 (769.2)
Second Process Feed (7) Pump Discharge	358.5 (181.4)	165.0 (1.138)	330.7 (769.2)
Process Steam (8)	358.5 (181.4)	150.0 (1.034)	1194.1 (2777.5)

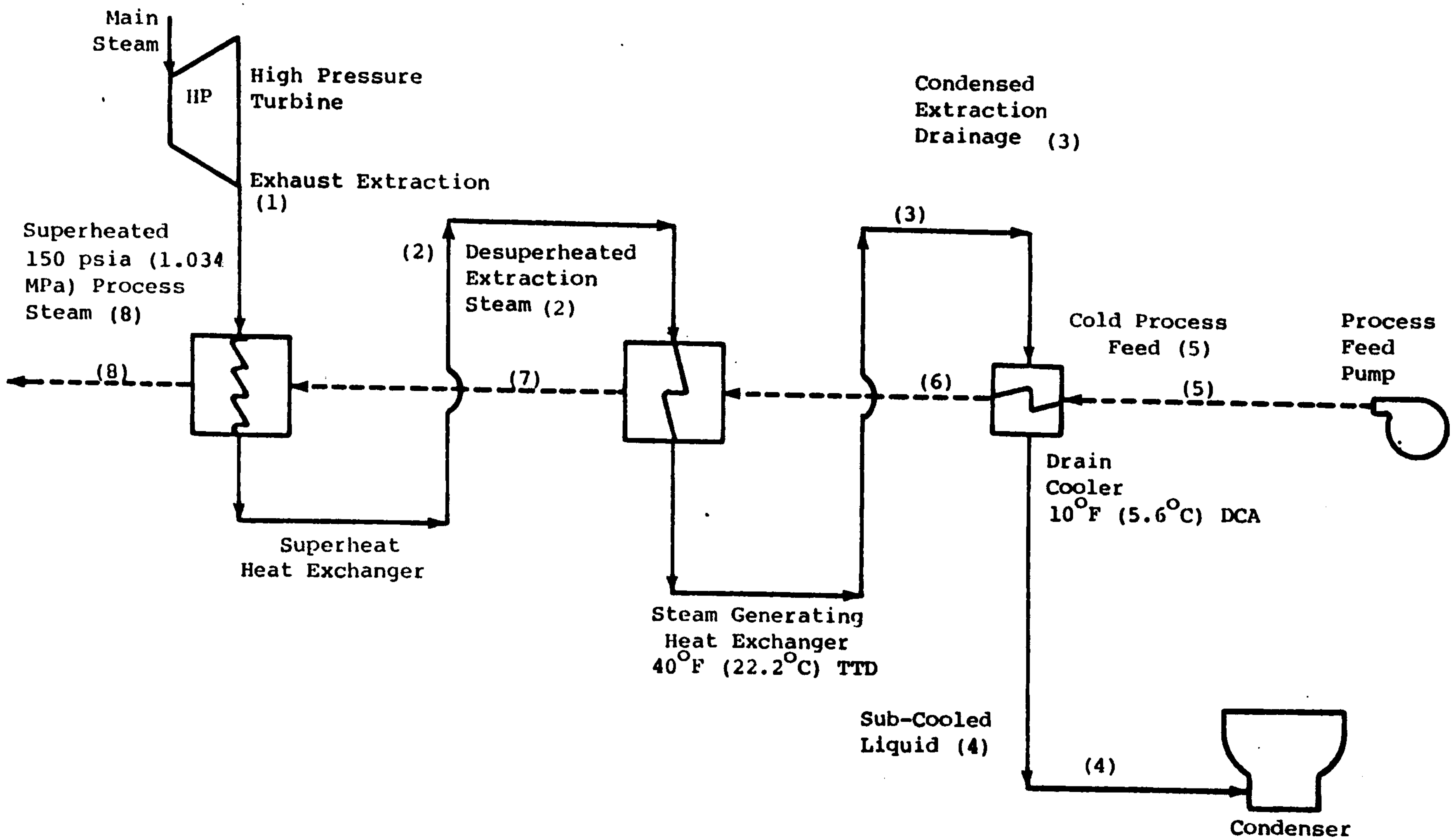

Figure III-10. Line Diagram of Process Steam Model 5A

Table III-8. Stream Properties for $50^{o}F(27.8^{o}C)$ Superheated Steam Conditions: Model 5A

Stream Description and State Number (X)	Temperature ^{o}F (^{o}C)	Pressure psia(MPa)	Enthalpy B/lbm(kJ/kg)
Exhaust Extraction (1)*	685.7(363.2)	474.1(3.269)	1351.2(3142.9)
Desuperheated Steam (2)	624.9(329.4)	426.7(2.942)	1319.8(3069.9)
Extraction Drainage (3)	399.2(204.0)	245.0(1.689)	374.3(870.6)
Sub-Cooled Liquid (4)	55.0(12.8)	30.0(0.207)	23.3(54.2)
Cold Process Feed (5)	45.1(7.3)	180.0(1.241)	13.7(31.9)
Preheated Feed (6)	357.7(180.9)	170.0(1.172)	329.9(767.3)
Process Steam (7)	363.6(184.2)	160.0(1.103)	1195.1(2779.8)
Superheated Steam (8)	408.2(209.0)	150.0(1.034)	1223.8(2846.6)

*Conditions at 1,232,714 lbm/hr (559,159 kg/hr) throttle flow and 48,000 lbm/hr (21,773 kg/hr) process demands.

surface area would be included in the design, instead of increasing the extraction condensing pressure.

Model 6A and 6B directly used turbine cycle extraction steam for process steam. For these models, reheat turbine exhaust is supplied above the third valve point (164 MW(e) gross generator load) while high pressure turbine exhaust steam is supplied for loads below the third valve point. With Model 6B the condensate is returned to the deaerating heater and wasted in Model 6A. Both models are equipped with a desuperheat-throttling station for cooling the superheated extraction steam before it enters the heat exchanger. Line diagrams for both models are shown in Figure III-11.

Assumptions

Pressure Drops Assumed for Models. Since each process steam model involves additional equipment and piping incorporated into an existing thermodynamic cycle, energy losses due to pressure drops must be assigned in order to evaluate model effectiveness and efficiency. The pressure drops assumed by the models in the study are not based upon actual lengths of pipe lines nor empirical results; but rather they are based upon standard specifications for established designs of steam turbines, heat exchanger equipment, and their associated piping. All assumed pressure drop losses across valves are to be included in the piping loss estimate. Typically a 5% extraction drop was taken in the turbine extraction lines. The extraction pressure was further dropped to 245 psi (1.689 MPa) in the heat exchanger shell. On the process side the pressure was dropped 10% across each component.

Thermodynamic Assumptions Made. In all process steam models the enthalpy for the process steam generated is assumed to be 1194.1 B/lbm (2777.5 kJ/kg) saturated process steam at 150.0 psia (1.034 MPa).

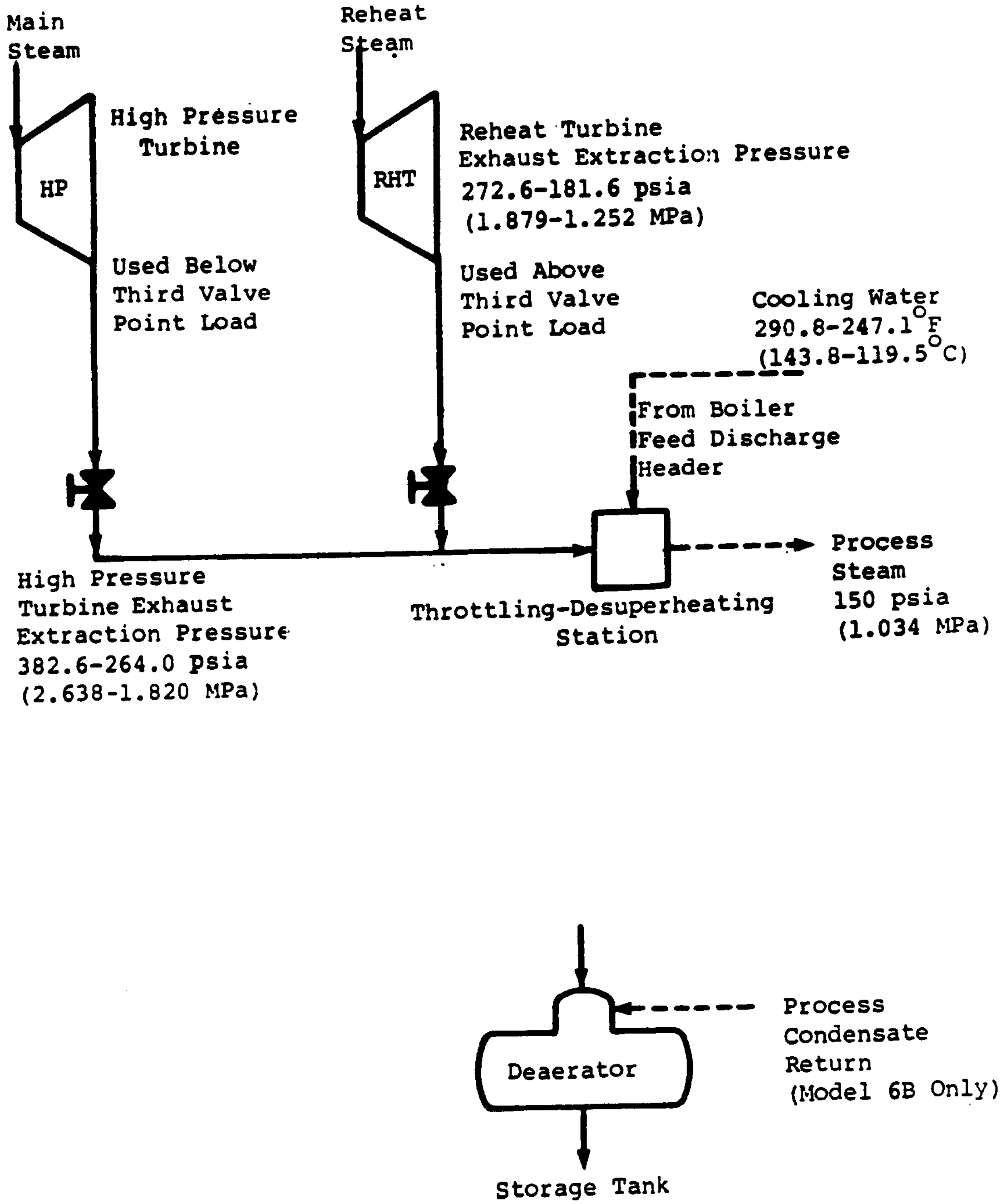

Figure III-11. Line Diagram of Process Steam Models 6A and 6B

CHAPTER IV

DISCUSSION OF THE RESULTS

The first major topic considers the changes in heat rates and turbine section efficiencies due to different throttle flow conditions. The reader will appreciate that these parameters change not only with different process requirements, and different model configurations, but from different turbine throttle flows. Care must be taken when comparisons are made between different process steam cycles to be sure they are being evaluated under the same general set of circumstances. Comparison of process steam cycles cannot be done solely at one valve point, since the results obtained would not be entirely correct. One reason why they differ at various loads is primarily due to inherent turbine design effects.

The computed output showed that the intermediate and low pressure turbine sections' efficiency for all process steam cycles increased as more high pressure exhaust extraction steam was removed from the turbine. However, the increase in efficiency by these two turbines is partially offset by the decrease in efficiency of the high pressure turbine. The increase in efficiency for the latter two turbines is created when extra high pressure extraction is removed from the turbine.

The overall turbine efficiency increases for different process flows over the entire load range. The table utilizes values for efficiencies obtained from process steam cycle 2B. The result holds true, as summarized in Table IV-1 for all three basic cycle types tested.

Table IV-1. Comparison of Total Turbine Efficiencies* for Different Process Steam Cycles: Crawford Unit Seven

Model Number and Nominal Valve Point Load MW(e)**	Total Turbine Efficiency % at Various Process Steam Demands				
	Base Case (0.0)	24.0(10.9)	48.0(21.8)	96.0(43.5)	192.0(87.1)***
1B 238.187 (6th)	84.258	84.348	84.410	84.518	84.853
2B 238.187 (6th)	84.258	84.339	84.425	84.613	84.931
3A 238.187 (6th)	84.258	84.349	84.412	84.587	84.916
1B 200.686 (4th)	84.752	84.858	85.125	———	———
2B 200.686 (4th)	84.752	84.863	84.980	85.196	85.676
3A 200.686 (4th)	84.752	84.856	84.965	85.059	85.557
1B 163.883 (3rd)	84.205	84.355	83.413	———	———
2B 163.883 (3rd)	84.205	84.539	84.704	84.709	———
3A 163.883 (3rd)	84.205	84.369	84.473	84.712	85.146

*Efficiencies given assume original turbine performance.

**Load indicated is at base case conditions with no process steam demands.

*** Notation: 1000 lbm/hr (1000 kg/hr) process steam generation rates at saturated 150 psia (1.034-MPa) conditions and 1194.1 B/lbm (2777.5 kJ/kg).

The main focus of the data in the final portion of the discussion is the evaluation of the performance of the various process steam models. Given the Crawford Unit Seven turbine cycle discussed earlier, a comparison of the performance factors can be made assuming a supply requirement for saturated process steam of 150 psi (1.034 MPa). The performance indicators used in the evaluation are efficiency, available energy, net power lost, effectiveness, and heat rate increase.

Model 5A was not considered because the cycle is strictly designed for superheat process steam applications. The other models considered are Model 1B, 2B, 3A, 4B, 6A, and 6B. Models 6A and 6B were included as references for the other models. The latter two models were not compared outright with the first four models, because of the direct versus indirect application of turbine extraction steam.

Table IV-2 summarizes the net change in megawatts lost for the six process models. When comparing all indirect cycle types, Model 2B has the best advantage. (The difference between Model 2B and Model 6B is approximately 4.0 kWh(e)/1000 lbm (8.82 kWh (e)/1000kg)).

Model 6B uses reheat turbine exhaust steam after it is desuperheated, directly as process steam, whenever the throttle flow is above the third valve point load: 968,123 lbm/hr (439,141 kg/hr). Reheat turbine extraction pressure is suitable over most of the load range in the direct process application. It has less available energy losses associated with throttling then high pressure turbine extraction steam. In addition, for this analysis only, a credit of 200 B/lbm (465.2 kJ/kg) is given to the turbine cycle for the condensate (no condensate is assumed lost) returned to the deaerating heater storage tank.

Table IV- 2 Crawford Unit Seven: Performance Indicators for Comparison Purposes; Cycle Lost kWh(e)/1000 lbm (kWh(e)/1000 kg) Process Steam Produced at Valve Point Turbine Loads

Valve Point and Nominal Load* # (MW(e))	Class II: Indirect Extraction Use				Class I: Direct Use	
Model:	1B	2B	3A	4B	6B	6A
	X (Y)	X (Y)	X (Y)	X (Y)	X (Y)	X (Y)
6th (238.116)	112.0 (246.9)	95.0 (209.4)	98.1 (216.3)	104.5 (230.4)	93.4 (205.9)	93.6 (206.3)
5th (219.970)	114.0 (251.3)	96.0 (211.6)	100.0 (220.5)	105.6 (232.8)	91.7 (202.2)	95.4 (210.3)
4th (200.686)	118.0 (260.1)	100.0 (220.5)	103.0 (227.1)	108.8 (239.9)	92.0 (202.8)	98.6 (217.4)
3rd (163.883)	82.7 (182.3)	84.0 (185.2)	96.9 (213.6)	103.1 (227.3)	82.0 (180.8)	96.2 (212.1)
Average Over Load Range:	106.7 (235.2)	93.8 (206.8)	99.5 (219.4)	103.1 (227.3)	89.8 (197.9)	96.2 (212.1)

*Nominal load indicated is at base case conditions with no process steam demands.

X = kWh(e)/1000 lbm

(Y) = kWh(e)/1000 kg

The amount of available energy consumed while generating process steam decreased with turbine load and is tabulated in Table IV-3. Model 2B uses the lowest amount of available energy outside of the direct extraction Models 6A and 6B (not shown). In contrast process steam cycle 1B consumes the highest amount of available energy over the entire range of turbine load. For all process steam cycles the amount of available energy consumed decreased with lower turbine throttle flows.

The process steam models have been categorized based upon the effectiveness of available energy utilization. Table IV-4 compares the average effectiveness of four process models over the standard turbine load range at each turbine valve point. The table shows for (except Model 1B) lower turbine valve point load the process model effectiveness increased, mainly due to smaller extraction throttling losses. Except for Model 1B, the process model effectiveness increases as the process load was increased.

The last parameter considered for comparing the effect of process steam is the net heat rate. It reflects the added cost of production (cost of additional heat input required per kilowatthour electricity generated) to produce process steam. The production cost of steam in terms of BkWh(e)/1000 lbm (W/kW(e)/1000 kg)* is tabulated for class II models in Table IV-5.

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*Saturated 150 psia (1.034 MPa) conditions at 1194.1 B/lbm (2777.5 kJ/kg.)

Table IV-3 Crawford Unit Seven: Performance Indicators for Comparison Purposes at 48,000 lbm/hr (21,773 kg/hr) Process Load Conditions; Available Energy Consumed for Process Steam Generation** by Various Cycles

Valve Point Nominal Load MW(e)	Available Energy B/1000 lbm (kJ/1000 kg) to Process Steam for Various Cycles: -1B-	-2B	-3A-	-4B-
6th(238.116)	9.958E5(2.316E6)	4.068E4(9.444E4)	5.576E5(1.297E6)	5.941E5(1.382E6)
5th(219.970)	1.009E5(2.347E5)	3.714E4(8.639E4)	5.673E5(1.320E6)	5.921E5(1.377E6)
4th(200.686)	1.008E5(2.345E5)	3.350E4(7.792E4)	5.715E5(1.335E6)	5.900E5(1.372E6)
3rd(163.883)	9.448E4(2.198E5)	1.537E4(3.575E4)	5.741E5(1.258E6)	2.916E5(6.783E5)
Average Over Load:	3.230E5(7.513E5)	3.167E4(7.366E4)	5.676E5(1.320E6)	5.170E5(1.203E6)

*Nominal load indicated is at base case conditions with no process steam demands.

**Saturated 150 psia (1.034 MPa) conditions at 1194.1 B/lbm (2777.5 kJ/kg).

Table IV- 4 Crawford Unit Seven: Performance Indicators for Comparison Purposes at 48,000 lbm/hr (21,773 kg/hr) Process Load Conditions;** Process Model Available Energy Effectiveness

Valve Point and Nominal Load Number, # (MW(e))*	Model: 1B	2B	3A	4B
	Process Cycle Portion Available Energy Effectiveness (Values Averaged Over Process Load at Valve Points)			
6th (238.116)	78.573	83.864	79.457	72.089
5th (219.970)	78.101	84.607	79.898	73.139
4th (200.686)	77.689	85.459	80.143	73.356
3rd (163.883)	77.623	80.628	80.548	74.312
Average Over Load Range:	77.980	85.140	80.012	73.224

*Nominal load indicated is at base case conditions with no process steam demands.

**Saturated 150 psia (1.034 MPa) conditions at 1194.1 B/lbm (2777.5 kJ/kg).

Table IV-5 Crawford Unit Seven: Performance Indicators for Comparison Purposes; Net Unit Heat Rate Increase per 1000 lbm (1000 kg) Process Steam Produced**

Valve Point Nominal Load* Number (MW(e))	Net Unit Heat Rate Increase for Various Models: Model 1B X (Y)	Model 2B X (Y)	Model 3A X (Y)	Model 4B X (Y)
6th(238.116)	4.428(2.860)	4.883(3.154)	5.058(3.267)	5.294(3.420)
5th(219.970)	6.058(3.913)	5.228(3.377)	5.425(3.504)	5.635(3.640)
4th(200.686)	6.576(4.248)	5.852(3.780)	6.093(3.936)	6.291(4.064)
3rd(163.883)	7.850(5.071)	6.819(4.405)	7.599(4.909)	7.505(4.848)
Average Over Load:	6.228(4.023)	5.696(3.679)	6.044(3.904)	6.183(3.994)

*Nominal load indicated is at base case conditions with no process steam demands.

X = Change in heat rate (B/kWh(e)) for each 1000 pounds per hour of process steam generated.

(Y) = Change in heat rate (W/kW(e)) for each 1000 kilograms per hour of process steam generated.

(This value represents the slope of the curve for Net Unit Heat Rate vs Process Steam Demands for each cogeneration model at various valve point loads.)

**Saturated 150 psia (1.034 MPa) conditions at 1194.1 B/lbm (2777.5 kJ/kg).

Chapter V

Summary and Conclusions

Summary

The study focussed on the performance of possible process steam-related component configurations which could be retrofitted to a utility size turbine cycle with present state of the art technology. A major portion of the work was done with a computer simulation technique which included available energy subroutines written by the author. Emphasis was placed on effective available energy utilization in the steam cycle. The design goal was to produce process steam at the lowest possible costs. Additional operating or design problems discovered were pointed out. It also mentioned how turbine shell pressures downstream of the extraction point would reduce shell pressure and result in increased feedwater heating load upstream of the extraction. As a result, the solution method used PEPSE to simulate this happening in the cycle. Eleven models were discussed and tested at four valve point loads and four extraction rates.

Section II discussed engineering limitations and how these parameters affected the study. The design limitations focussed on were: heat exchangers, boilers and turbines. A short discussion concerning the computerized simulation technique was made. The precision of this technique is within $\pm$ 0.5%. Four major valve point turbine loads were chosen for most of the analysis work, since they cover the most probable operating range of Crawford Unit Seven. One of these concerns involved the increase in water induction probability caused by potential cross-tieing units together.

Another concern cited was the recommended limitation of high pressure turbine exhaust extraction to 5% of rated boiler superheater outlet flow. Turbine blade stressing from extra extraction was also mentioned.

Section III presented a detailed description, explanation, and logical development of different process steam models used in the analysis along with assumptions made for each one. The basic Crawford Unit Seven turbine cycle was discussed along with potential process steam applications for the four extraction points. Routing condensed extraction drainage to the condenser is preferred because it is the largest sink in the steam cycle. Hot extraction drainage should be returned to the deaerator. Use of a feedheater other than the deaerator or condenser as a sink for process extraction drainage should be avoided. This limits possible flashing, tube fretting, and erosion effects which cause cycle upset and increased maintenance problems.

A summarized data presentation was made in the fourth section for various major areas of interest. These topics included effects on turbine performance when additional extraction steam is removed; and an evaluation of the different process steam cycles. The replacement power per unit process steam was determined for the process steam cycles. Model 1B was affected most in performance and stability by changes in steam generator terminal differences. It also had the lowest performance of all models considered. Model 2B proved to be the most efficient design configuration of the indirect extraction use models considered.

Possibilities for Future Studies

Before a project of this magnitude can proceed, it would be necessary to evaluate a few areas in greater detail. The safe operation of the turbines with increased extraction, is a major concern for future study. Viable alternate information exists to help evaluate turbine blade stressing without doing a complete turbine engineering study. An almost parallel set of circumstances exists when adjacent feedwater heaters are removed from service during turbine operation. Additional extraction steam requirements are made from heater(s) upstream of the inoperable heater(s).

Most major turbine manufacturers use a design criteria which guarantees maximum vendor throttle flow while accommodating extra extraction requirements associated with any one heater in the series being out of service. In addition, certain turbines have their high pressure and intermediate pressure exhaust ends specifically designed to handle extra extraction flow requirements. Operation with increased extraction is tolerated provided the work, flow, and pressure drops across the individual stages do not exceed values incurred at the maximum calculated five percent overpressure heat balance (Westinghouse, 1982). For extraction loads over this amount in most cases the turbine restriction is a reduction in maximum allowable steam flow. When the extraction rate is equivalent to one lower heater out below the extraction point as shown in Table V-1, there is no restriction. At the extraction rate above one lower adjacent feedheater out of service shown in Table V-2, the turbine throttle flow must be reduced by 10% of rated flow conditions (General Electric, 1979). The maximum throttle flow reduction is increased by another 10% (20% total) for the rate greater than two lower adjacent feedheaters taken out of service as shown in Figure V-1.

One area that needs further investigation is the 5% limit on high pressure turbine exhaust extraction recommended by the boiler manufacturer. This figure is quite conservative and could be as high as 10% of design superheater outlet flow without causing any undue damage. This depends on the condition of the tubes and would require a detailed boiler inspection to be made. Additional checks would have to be made concerning the status of the boiler controls. Using hot reheat steam instead of cold reheat may alleviate this problem.

The final system choice for the project should be based upon a set criteria for evaluation. One possible project for future analysis could determine what the final criteria and choice should be for this particular installation. Then a final recommendation could be made based upon this additional information.

CHAPTER VI

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