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Supercritical Water Reactor Cycle for Medium Power Applications

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Executive Summary

Scoping studies for a power conversion system based on a direct-cycle supercritical water reactor have been conducted. The electric power range of interest is 5-30 MWe with a design point of 20 MWe. The overall design objective is to develop a system that has minimized physical size and performs satisfactorily over a broad range of operating conditions. The design constraints are as follows:

- Net cycle thermal efficiency ≥20%
- Steam turbine outlet quality ≥90%
- Pumping power ≤2500 kW (at nominal conditions)

Three basic cycle configurations were analyzed. Listed in order of increased plant complexity, they are:

- Simple supercritical Rankine cycle
- All-supercritical Brayton cycle
- Supercritical Rankine cycle with feedwater preheating

The sensitivity of these three configurations to various parameters, such as reactor exit temperature, reactor pressure, condenser pressure, etc., was assessed. The Thermoflex software package was used for this task. The results are as follows:

- The simple supercritical Rankine cycle offers the greatest hardware simplification, but its high reactor temperature rise and reactor outlet temperature may pose serious problems from the viewpoint of thermal stresses, stability and materials in the core.
- The all-supercritical Brayton cycle is not a contender, due to its poor thermal efficiency.
- The supercritical Rankine cycle with feedwater preheating affords acceptable thermal efficiency with lower reactor temperature rise and outlet temperature.
- The use of a moisture separator improves the performance of the supercritical Rankine cycle with feedwater preheating and allows for a further reduction of the reactor outlet temperature, thus it was selected for the next step.

Preliminary engineering design of the supercritical Rankine cycle with feedwater preheating and moisture separation was performed. All major components including the turbine, feedwater heater, feedwater pump, condenser, condenser pump and pipes were modeled with realistic assumptions using the PEACE module of Thermoflex. A three-dimensional layout of the plant was also generated with the SolidEdge software. The results of the engineering design are as follows:

- The cycle achieves a net thermal efficiency of 24.13% with 350/460°C reactor inlet/outlet temperatures, ~250 bar reactor pressure and 0.75 bar condenser pressure. The steam quality at the turbine outlet is 90% and the total electric consumption of the pumps is about 2500 kWe at nominal conditions.
- The overall size of the plant is attractively compact and can be further reduced if a printed-circuit-heat-exchanger (vs shell-and-tube) design is used for the feedwater heater, which is currently the largest component by far.

Finally, an analysis of the plant performance at off-nominal conditions has revealed good robustness of the design in handling large changes of thermal power and seawater temperature.

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1. Project Objectives

The central objective of this study is to determine component sizing requirements and establish fundamental design sensitivities and limitations for supercritical water reactor (SCWR) Power Conversion Systems (PCS) at 5-30 MWe output. The investigation seeks to characterize a PCS for a direct cycle that:

- Achieves relatively high net efficiency in conversion from thermal to electrical energy
- Is compact with minimum volume and weight
- Is robust, resilient to accidents and has high long-term reliability and performance
- Shows potential for good controllability and fast response to requested power changes

2. Introduction

The SCWR concept is one of six reactor technologies selected for research and development under the Generation-IV program. SCWRs are basically light-water cooled reactors operating at higher pressure and temperature. Operation above the critical pressure of water eliminates coolant boiling, so the coolant remains a single-phase fluid (Figure 2.1), while its thermo-physical properties vary rather dramatically with temperature (Figure 2.2). If a direct-cycle system is selected, the need for steam generators, steam separators and pressurizer is eliminated, as shown in Figure 2.3. The high operating pressures and temperatures generally result in higher thermal efficiency than systems operating with traditional subcritical Rankine cycles.

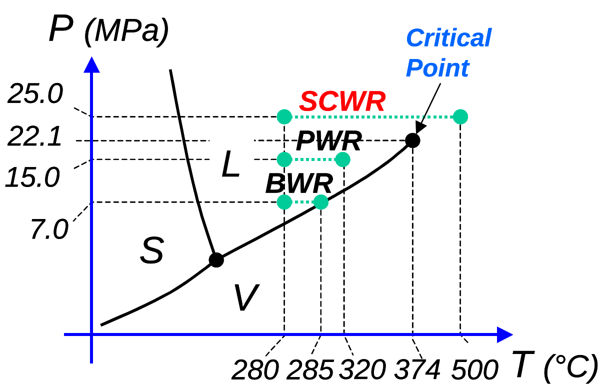


Figure 2.1 SCWR, PWR and BWR operating conditions on the phase diagram of water.

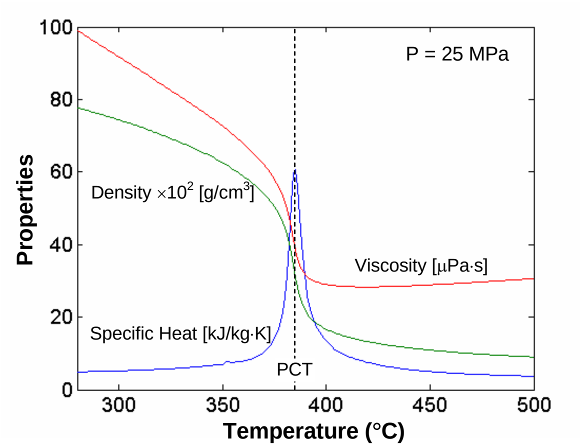


Figure 2.2 Variation of the thermo-physical properties of water at constant supercritical pressure. PCT = pseudo-critical temperature.

The Gen-IV SCWR program has been focusing on large plants with PCS rated at 1600 MWe. The interest in this work, however, is in PCS having ratings between 5 and 30 MWe in a power dense application. Scaling to these sizes may have significant effects on key design parameters, including state point optimization (temperatures, pressures and efficiencies) and turbomachinery design. For example, significant simplification may be obtained in the cycle with the elimination of the low pressure turbine and most or all feedwater heaters, thus rejecting the waste heat at higher temperature, though this would be done at the expense of the thermal efficiency. In this study, parameter trade-off and component design studies are performed in order to define the PCS options that best suit medium power, compact, load-following applications.

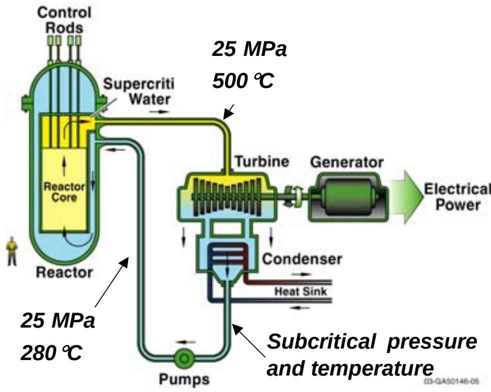


Figure 2.3 Schematic of a direct-cycle SCWR.

3. Choice and Validation of the Analytical Tools Used in this Project

To evaluate the various supercritical PCS configurations analyzed in this project, the software package Thermoflex was selected. Thermoflex is a product of Thermoflow Inc., a leading developer of thermal engineering software for the power and cogeneration industries. Thermoflex is a "bottom up" type of software which allows the user to design thermodynamic cycles with any arbitrary configurations. The graphical interface offers pre-designed components that can be chosen and placed on the screen to form the cycle. User-defined components can also be added to the system. The thermodynamic properties of water (including supercritical water) are built into the program. Thermoflex is ideally suited for scoping thermodynamic analyses. A companion software program, PEACE, provides realistic engineering estimates on the component size, layout and cost. Once the system design is fixed, Thermoflex can actually analyze the performance of the system at off-design conditions, a very useful feature since such conditions often occur during operation. Thermoflow Inc. software has been used for almost two decades by industry including recently Burns and Roe for analyzing the Gen-IV SCWR PCS[1].

To verify our ability to set the input parameters properly in Thermoflex, we first consider an ideal simple Rankine cycle (turbine + condenser + pump + steam generator) driven by dry saturated steam at 78 bar and with a condenser pressure of 0.07 bar. The theoretical (hand-calculated) thermal efficiency of such cycle is equal to 37.35%. Using Thermoflex, the thermal efficiency is also found to be 37.35%. Next, to verify Thermoflex's capability and ease for automatic multiple run analysis, we perform two parametric studies. In the first study the turbine inlet pressure is varied from 11 bar to 150 bar with the condenser pressure held constant at 0.07 bar. In the second study the condenser pressure is varied from 0.01 bar to 1.4 bar with the turbine inlet pressure held constant at 78 bar. The Thermoflex results are compared with the results from an ad-hoc MATLAB script that calculates the thermal efficiency of the cycle from the thermodynamic properties. The agreement between Thermoflex and the MATLAB script predictions is perfect, as shown in Figures 3.1 and 3.2.

[1] J. Buongiorno, P. MacDonald, *Supercritical Water Reactor (SCWR), Progress Report for the FY-03 Generation-IV R&D Activities for the Development of the SCWR in the U.S.*, INEEL/EXT-03-03-01210, Idaho National Engineering and Environmental Laboratory, September 2003.

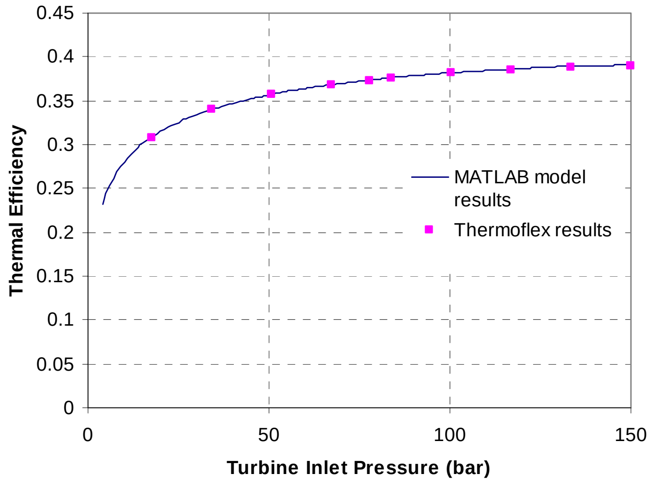


Figure 3.1 Effect of turbine inlet pressure variation on thermal efficiency of an ideal Rankine cycle.

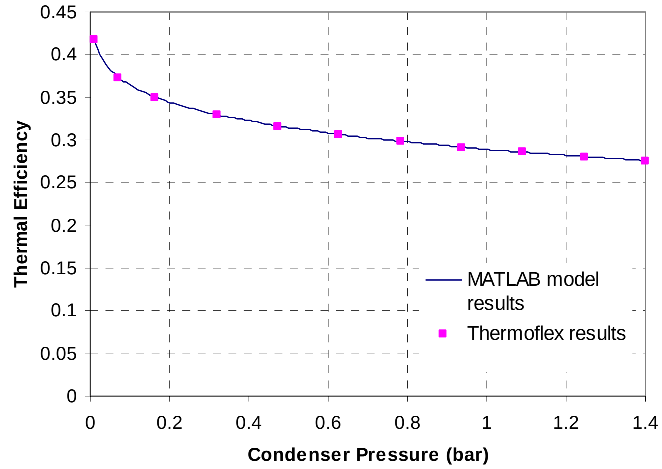


Figure 3.2 Effect of condenser pressure variation on thermal efficiency of an ideal Rankine cycle.

4. Thermodynamic Evaluation of Various Simplified Supercritical Water Cycles

The Gen-IV reference SCWR PCS has a rated power of 1600 MWe with a single-shaft turbine-generator operating at 1800 rpm. The turbine has one High Pressure (HP) section, a moisture separator-reheater and three double-flow Low Pressure (LP) sections. The feedwater is preheated in eight feedwater heaters and circulated by steam turbine-driven feedwater pumps. The waste heat is rejected in large natural draft cooling towers. This cycle was designed mainly for optimal thermal efficiency in a very large baseload land plant. Simply scaling down such complex cycle to 5-30 MWe power is deemed unattractive, because it would result in a PCS lacking the necessary simplicity, compactness and ruggedness.

Therefore, in this project we have considered three alternative and much simplified PCS configurations. These are:

1. Simple supercritical Rankine cycle
2. All-supercritical Brayton cycle
3. Supercritical Rankine cycle with feedwater preheating

The Thermoflex schematic layout and T-s diagram for these cycles are shown in Figures 4.1, 4.2 and 4.3, respectively. The thermodynamic results for these three configurations are reported in Sections 4.1, 4.2 and 4.3, respectively.

In analyzing the thermodynamic performance of the above three cycles, the following figures of merit were given particular attention:

- Cycle thermal efficiency. A high value of the thermal efficiency reduces the required reactor thermal power, thus reducing the fuel requirements and physical size of the reactor. The target net thermal efficiency is ≥20% in this study.

- Steam quality at the turbine outlet. High values of the steam quality reduce erosion concerns for turbine blades and casing. The target steam quality at the turbine outlet is ≥90%.
- Pumping power. A low pumping power reduces the pump size and facilitates fast power ramp-ups during start up or load change. Pumps rated higher than 2500 kW are considered undesirable.

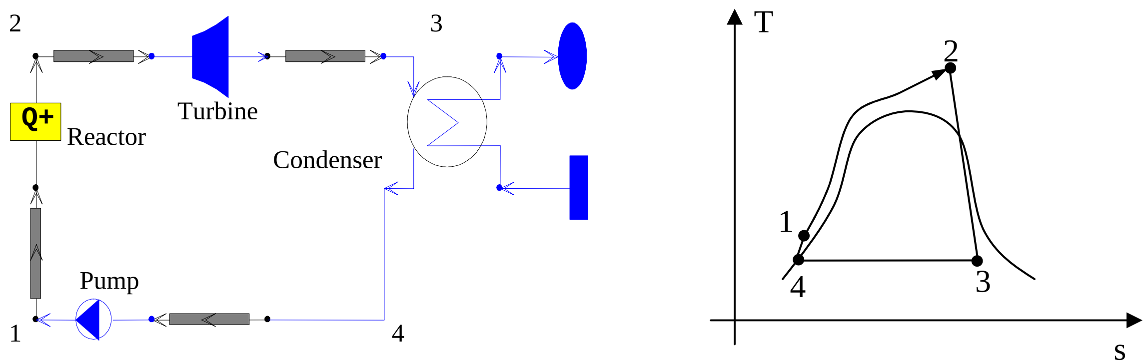


Figure 4.1 The simple supercritical Rankine cycle

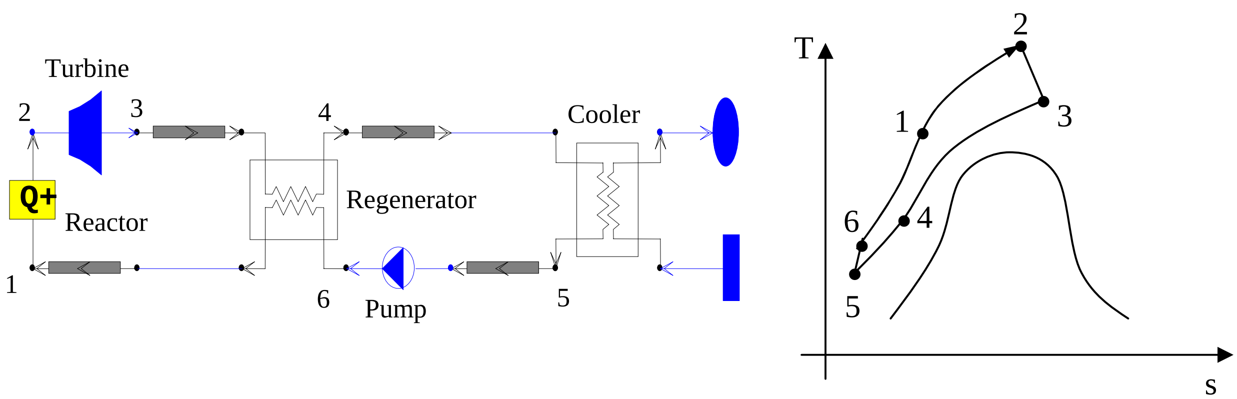


Figure 4.2 The all-supercritical Brayton cycle

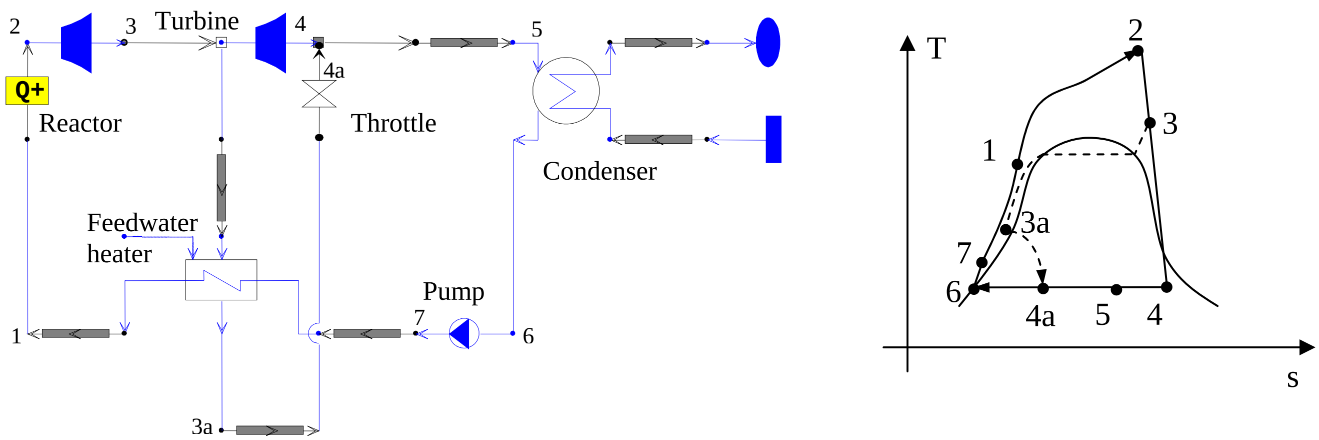


Figure 4.3 The Rankine cycle with feedwater preheating

4.1 Simple Supercritical Rankine Cycle

There are four components in this cycle, i.e., the reactor, the turbine, the condenser and the pump, connected by various pipes. We investigated the sensitivity of the cycle performance to the following three parameters: condenser pressure, reactor exit temperature (i.e., turbine inlet temperature) and reactor pressure (i.e., turbine inlet pressure). The reference values for these parameters are 64.17 bar, 500°C and 250 bar, respectively. The value of the condenser pressure is selected to give a reactor inlet temperature of about 280°C, which is the same as in the Gen-IV SCWR. The values for the turbine inlet temperature and pressure are directly drawn from the Gen-IV SCWR design. The isentropic efficiency for the turbine is assumed to be 80%, typical of small turbines. The isentropic efficiency of the pump is also assumed to be 80%, a reasonable value for centrifugal pumps. A 2% pressure loss is assumed for the reactor, while a 1% pressure loss is assumed for the pipe sections. These values are consistent with commercial plant experience. No other losses (e.g., pump mechanical efficiency, generator efficiency, etc.) are accounted for in the analysis. In all the parametric studies the net electric power is held at a constant level (20 MWe), which can be done conveniently with Thermoflex.

In the first parametric study the condenser pressure is varied from 10 to 80 bar, while the turbine inlet temperature and pressure are held fixed at their reference values. The variation of the cycle thermal efficiency, turbine exit quality and pumping power is plotted in Figures 4.4, 4.5 and 4.6, respectively. Note that the thermal efficiency rapidly decreases with increasing condenser pressure, as expected (Figure 4.4). Thermal efficiency values near the 20% target can be achieved in this cycle only at relatively low condenser pressure. On the other hand, the steam quality at the turbine outlet is above the 90% target for all values of the condenser pressure (Figure 4.5). The pumping power increases with the condenser pressure (Figure 4.6). This trend is not intuitive and thus requires some explanation. The pumping power, W_p, is proportional to the reactor mass flow rate and the pressure difference between the reactor and the condenser, ΔP. In turn, the mass flow rate is proportional to the thermal power. Since in these studies we keep the electric power constant at 20 MWe, the mass flow rate becomes inversely proportional to the thermal efficiency, η. Mathematically, $W_p \propto \Delta P/(\eta \cdot \Delta h)$, where Δh is the enthalpy rise in the reactor. Now, ΔP, η and Δh all decrease with increasing condenser pressure, but the decrease of η dominates, so the net effect is that the pumping power increases with the condenser pressure. It appears that an acceptable pumping power could be obtained if the condenser pressure is kept relatively low.

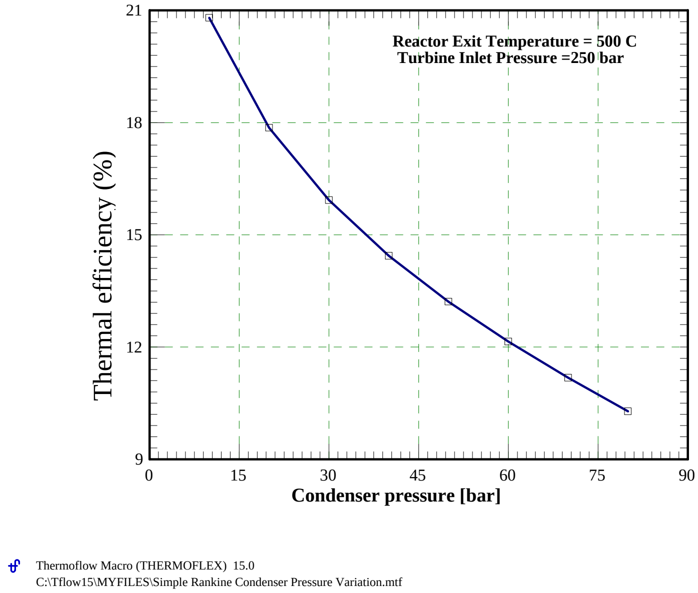


Figure 4.4 Cycle thermal efficiency vs condenser pressure for the simple supercritical Rankine cycle.

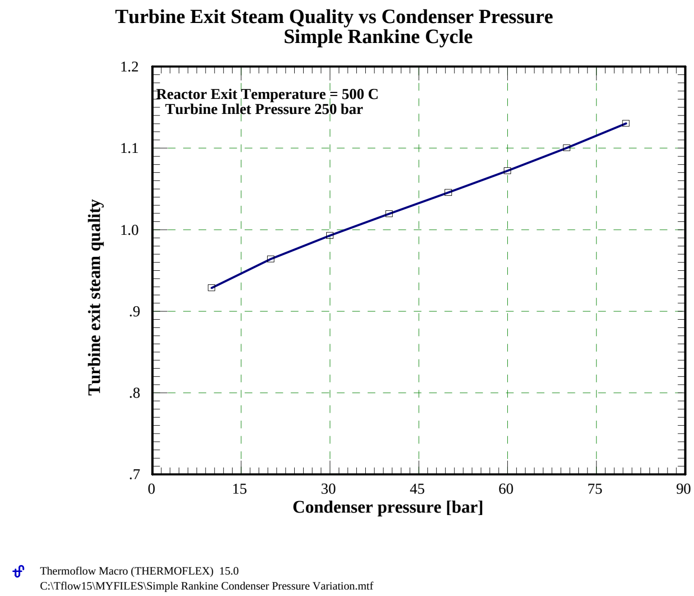


Figure 4.5 Turbine exit quality vs condenser pressure for the simple supercritical Rankine cycle.[2]

[2] The steam quality is defined as $x=(h-h_f)/h_{fg}$, where h is the enthalpy of the fluid, h_f is the saturated liquid enthalpy and h_{fg} is the enthalpy of vaporization. Therefore, x>1 indicates superheated steam.

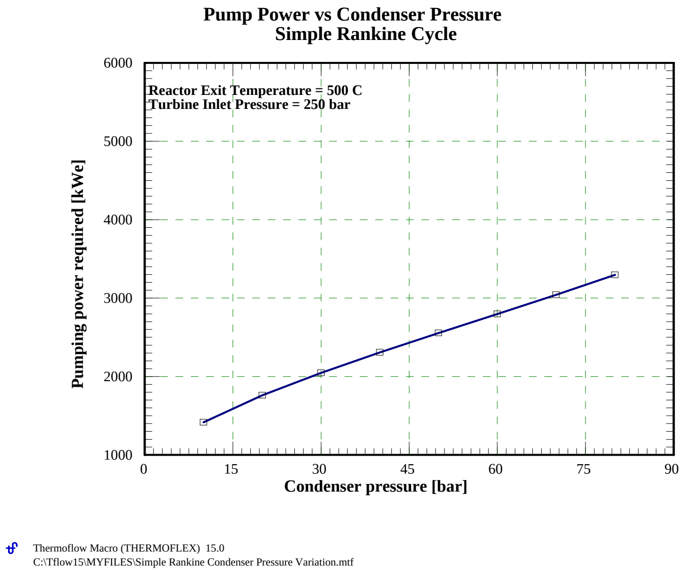


Figure 4.6 Pumping power vs condenser pressure for the simple supercritical Rankine cycle.

In the second parametric study the reactor exit temperature (turbine inlet temperature) is varied from 400 to 550°C, while the condenser pressure and the turbine inlet pressure are held fixed at their reference values. The thermal efficiency, turbine exit quality and pumping power are plotted in Figures 4.7, 4.8 and 4.9, respectively. The thermal efficiency obviously increases with the reactor exit temperature (Figure 4.7). However, the dependence does not seem very strong and the values remain well below the 20% target; moreover increasing the reactor exit temperature beyond its reference value of 500°C is questionable because of materials-related concerns in the core. On the other hand, for low reactor exit temperature the steam quality at the turbine outlet becomes unacceptably low (Figure 4.8). In this parametric study the pumping power decreases as the reactor exit temperature increases (Figure 4.9), mainly due to the increase in cycle thermal efficiency and enthalpy rise in the reactor. However, the values of the pumping power are all very high.

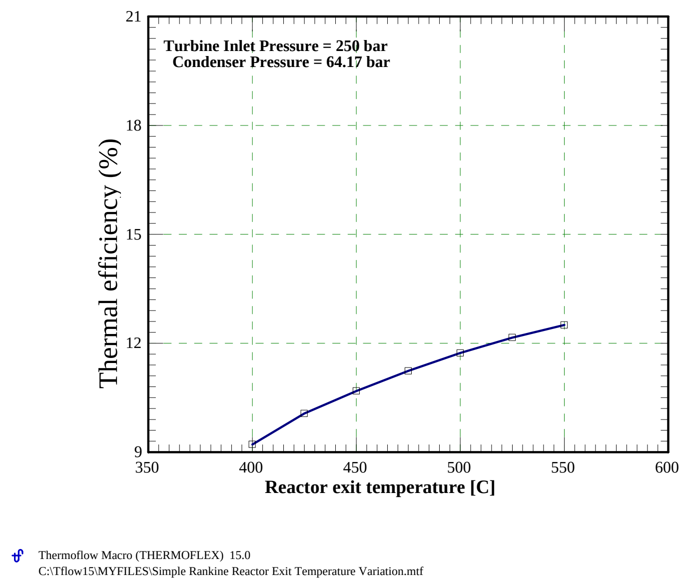


Figure 4.7 Cycle thermal efficiency vs reactor exit temperature for the simple supercritical Rankine cycle.

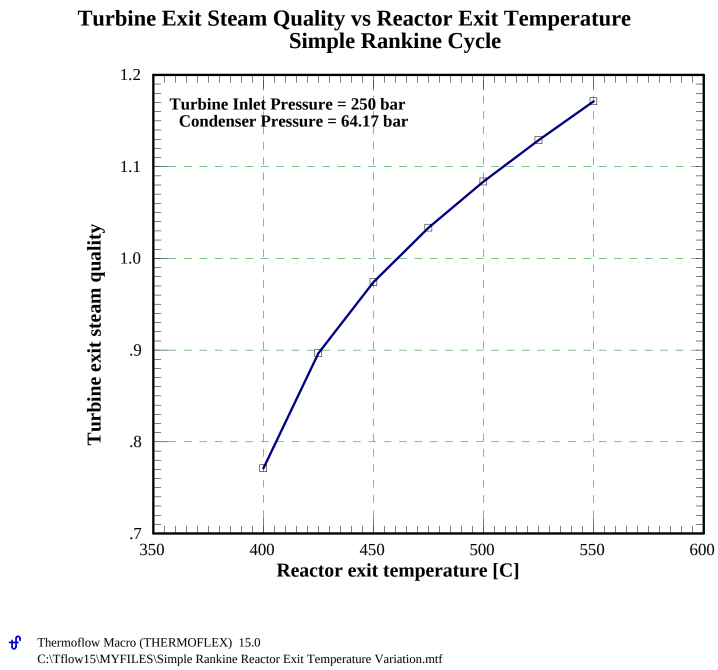
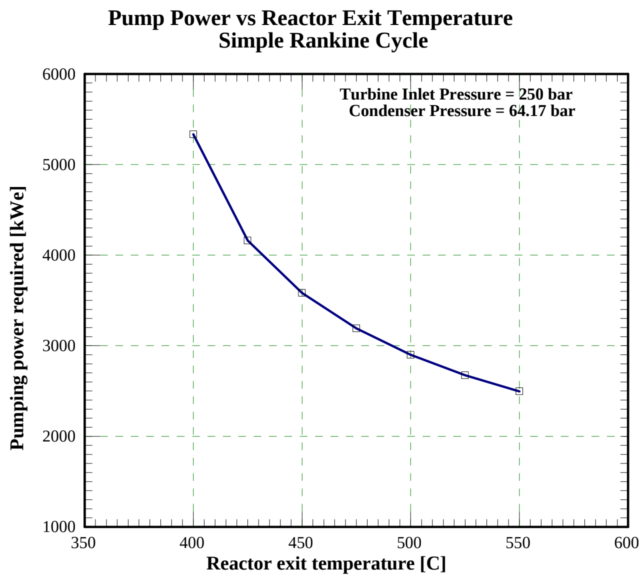


Figure 4.8 Turbine exit quality vs reactor exit temperature for the simple supercritical Rankine cycle.



Thermoflow Macro (THERMOFLEX) 15.0
C:\Tflow15\MYFILES\Simple Rankine Reactor Exit Temperature Variation.mtf

Figure 4.9. Pumping power vs reactor exit temperature for the simple supercritical Rankine cycle.

In the third parametric study the turbine inlet pressure (reactor pressure) is varied from 225 to 350 bar, while the condenser pressure and the reactor exit temperature are held fixed at their reference values. The thermal efficiency, turbine exit quality and pumping power are plotted in Figures 4.10, 4.11 and 4.12, respectively. The thermal efficiency increases slowly with the turbine inlet pressure (Figure 4.10), however, it remains well below the 20% target. The steam quality at the turbine outlet is acceptable for all turbine inlet pressures (Figure 4.11), while the pumping power is very high (Figure 4.12).

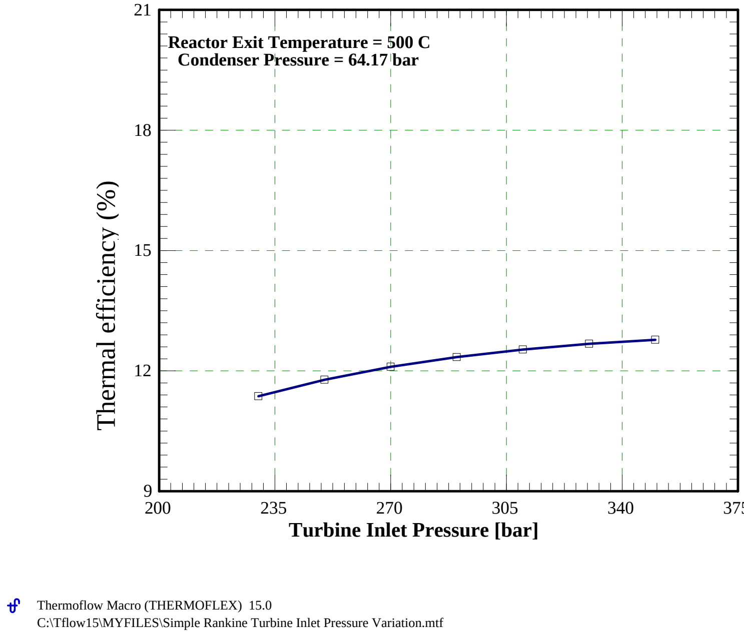


Figure 4.10. Cycle thermal efficiency vs turbine inlet pressure for the simple supercritical Rankine cycle.

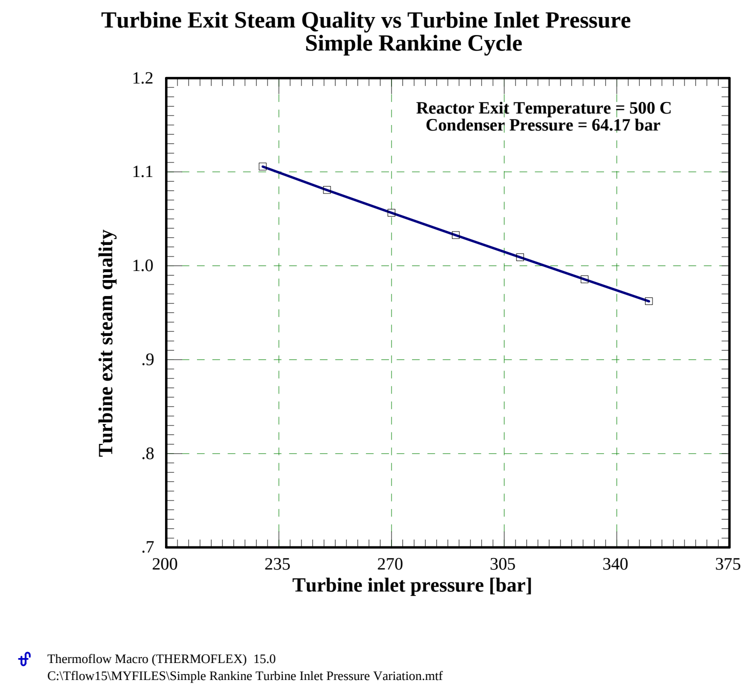


Figure 4.11. Turbine outlet quality vs turbine inlet pressure for the simple supercritical Rankine cycle.

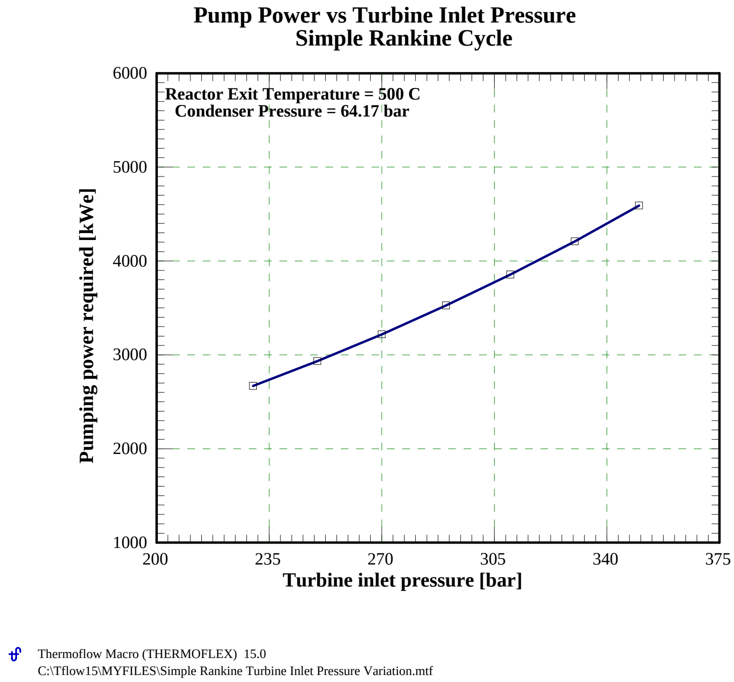


Figure 4.12. Pumping power vs turbine inlet pressure for the simple supercritical Rankine cycle.

In summary, these parametric studies show that reducing the condenser pressure to 10 bar or less is an attractive way to meet the thermal efficiency and pumping power targets with the simple supercritical Rankine cycle. Increasing the turbine inlet temperature and pressure benefits insignificantly the performance of the cycle and could actually create serious problems with materials. The drawback of reducing the condenser pressure is that the reactor inlet temperature becomes relatively low (≤180°C), thus, for given reactor exit temperature, the temperature rise and enthalpy rise in the reactor become high. This brings about a couple of problems. First, large temperature differences generate high thermal stresses in the vessel internals separating the cold and hot streams. Second, a large enthalpy rise in the core makes the system susceptible to large temperature excursions upon power-flow mismatches, which inevitably occur locally and globally in the core during normal and off-normal operation. However, these issues can be alleviated by using a recirculation which is not addressed in this report

4.2 All-Supercritical Brayton Cycle

The idea is to explore a Brayton (non-condensing) cycle with supercritical water. This cycle has five components (reactor + turbine + regenerator + cooler + pump) with connecting pipes. The reactor operates at very high supercritical pressure (380 bar). The turbine has only a very high pressure section, because the expansion is arrested at about 233 bar. Thus, this cycle operates entirely above the critical pressure of water, and no moisture is present in the turbine. The high heat content of the fluid at the turbine outlet is used for regenerative heating of the reactor feedwater coming from the pump. The cooler discharges the waste heat into the environment. The minimum temperature in the cycle is 40°C. The isentropic efficiency for the pump and turbine is 80%. The pressure loss in the reactor and pipes is assumed to be 2% and 1%, respectively. The regenerator effectiveness is assumed to be unity. No other losses are accounted for in the analysis and again the net electric power is held at a constant 20 MWe.

Figure 4.13 shows the thermal efficiency vs reactor exit temperature for this cycle. The values of the thermal efficiency are unacceptably low, even at high reactor exit temperature and pressure. This stems from the relatively low work done by the turbine in comparison with the work required by the pump. Given these results, the all-supercritical Brayton cycle is discarded from further consideration.

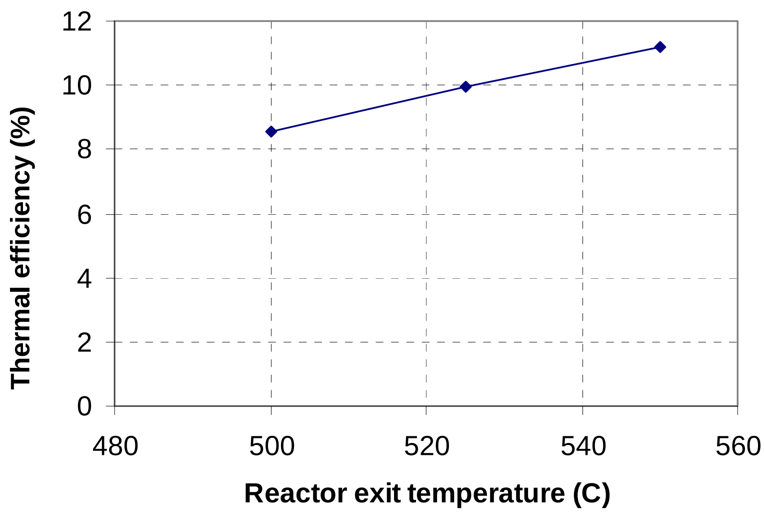


Figure 4.13. Thermal efficiency vs reactor exit temperature for the all-supercritical Brayton cycle.

4.3 Supercritical Rankine Cycle with Feedwater Preheating

There are six components in this cycle, i.e., the reactor, the turbine, the condenser, the pump, the feedwater heater and the throttle valve, connected by various pipes. Some steam is extracted from the turbine to preheat the feedwater, and then throttled and dumped into the condenser. While it adds some complication to the cycle, the feedwater heater improves the thermal efficiency and, importantly, provides flexibility in selecting the reactor inlet temperature. Note that while the schematic in Figure 4.3 shows two turbines, physically there is only one turbine in this cycle.

We investigated the sensitivity of the cycle performance to the following three parameters: extraction pressure (P_3), reactor inlet temperature and reactor outlet temperature. The reactor pressure is fixed at 250 bar. The condenser pressure is fixed at 10 bar, corresponding to a saturation temperature of about 180°C. As far as the feedwater heater is concerned, either the drain cooler approach temperature difference (T_{3a}-T_7) is set equal to 10°C or the minimum temperature difference within the heater (the so-called 'pinch point') is set equal to 5°C. Thermoflex automatically chooses the most constraining condition from case to case. The isentropic efficiency for the pump and turbine is 80%. The pressure losses in the reactor and pipes are assumed to be 2% and 1%, respectively. No other losses are accounted for and the net electric power is held at a constant 20 MWe.

The variation of the thermal efficiency vs the extraction pressure is shown in Figures 4.14, 4.15, 4.16 and 4.17, for reactor inlet temperatures of 250, 280, 300 and 350°C, respectively, and for a fixed reactor outlet temperature of 500°C. The thermal efficiency decreases as the extraction pressure increases. This is expected because a higher extraction pressure reduces the turbine work. For the feedwater heater to function properly, the extraction pressure has to be greater than the saturation pressure corresponding to the specified reactor inlet temperature, e.g., for a reactor inlet temperature of 280°C, the extraction pressure is limited to values >64.2 bar. Note that for given extraction pressure, the thermal efficiency does not appear to depend strongly on the reactor inlet temperature. This is the result of two conflicting effects. First, a higher reactor inlet temperature increases the average temperature at which heat is added to the cycle. This effect is beneficial to the thermal efficiency. Second, since only one feedwater heater is used in this cycle, a higher reactor inlet temperature also increases the fraction of steam that needs to be extracted early from the turbine. This reduces the turbine work and the thermal efficiency. These two effects appear to be of the same order of magnitude, so the net effect on thermal efficiency is small, as we said.

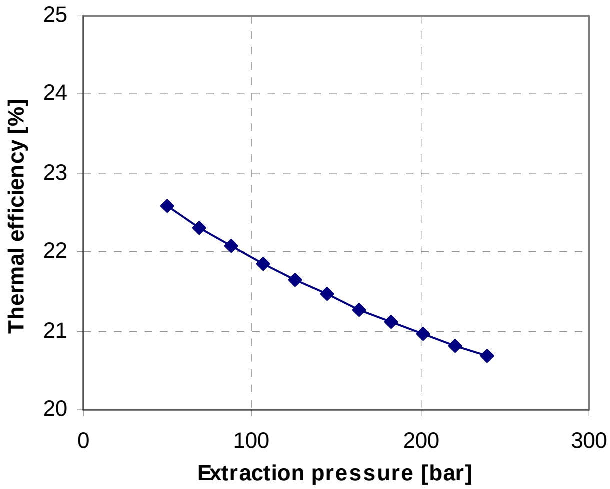


Figure 4.14. Thermal efficiency vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 250°C.

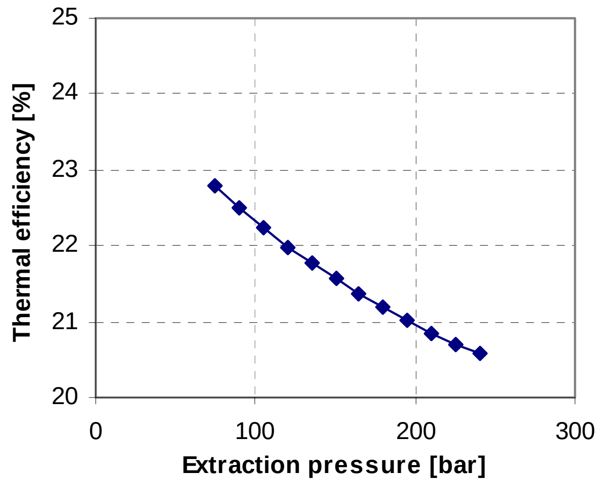


Figure 4.15. Thermal efficiency vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 280°C.

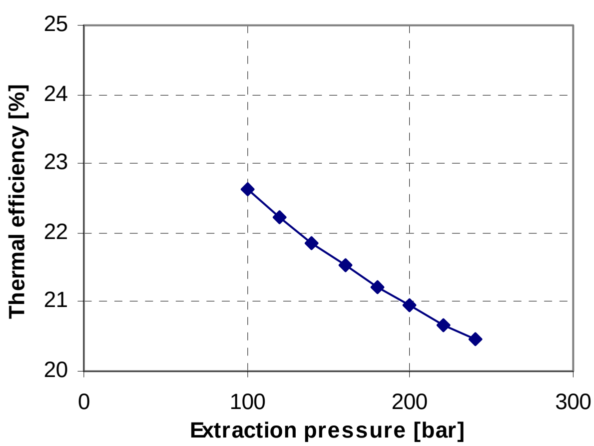


Figure 4.16. Thermal efficiency vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 300°C.

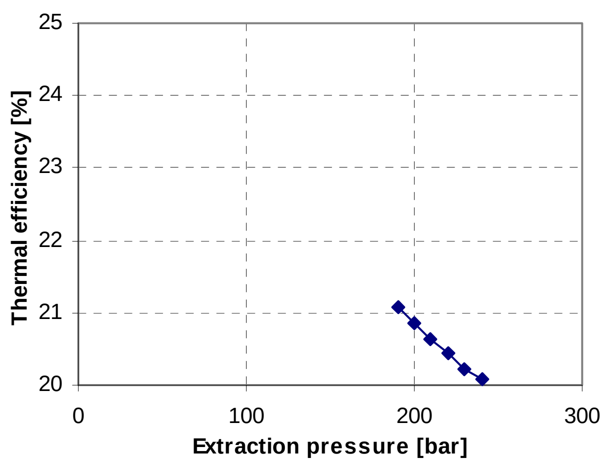


Figure 4.17. Thermal efficiency vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 350°C.

The variation of the flow fraction of extracted steam vs the extraction pressure is shown in Figures 4.18, 4.19, 4.20 and 4.21, again for reactor inlet temperatures of 250, 280, 300 and

350°C, respectively, and a fixed reactor outlet temperature of 500°C. The fraction of extracted flow decreases as the extraction pressure increases, and increases as the reactor inlet temperature increases. Both trends are expected in view of the energy needed to raise the feedwater temperature from its value at the pump outlet to its specified value at the reactor inlet.

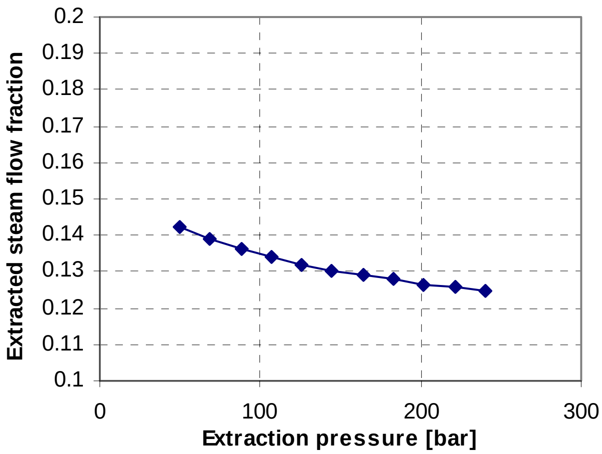


Figure 4.18. Extracted steam flow fraction vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 250°C.

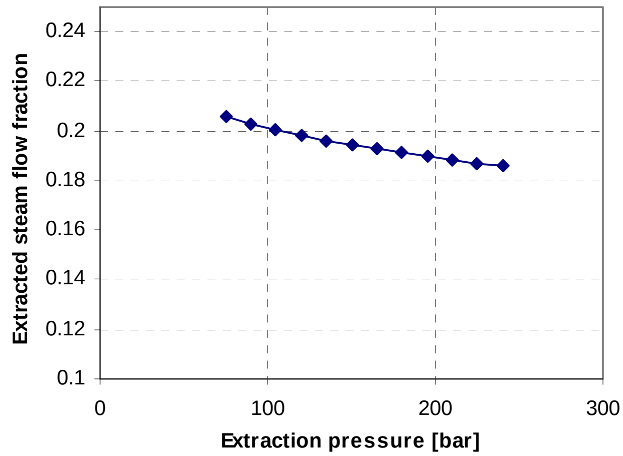


Figure 4.19. Extracted steam flow fraction vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 280°C.

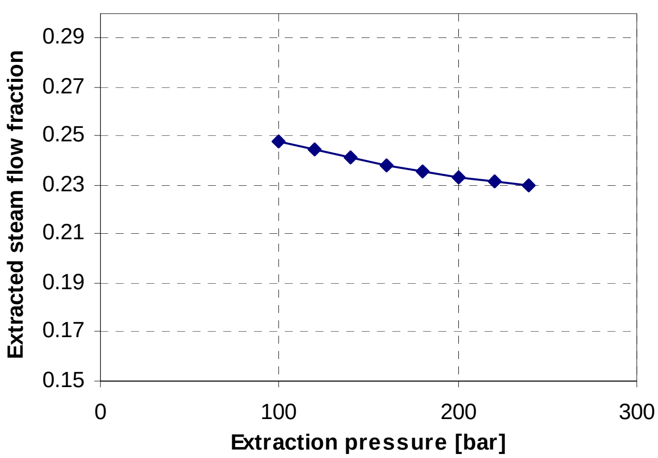


Figure 4.20. Extracted steam flow fraction vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 300°C.

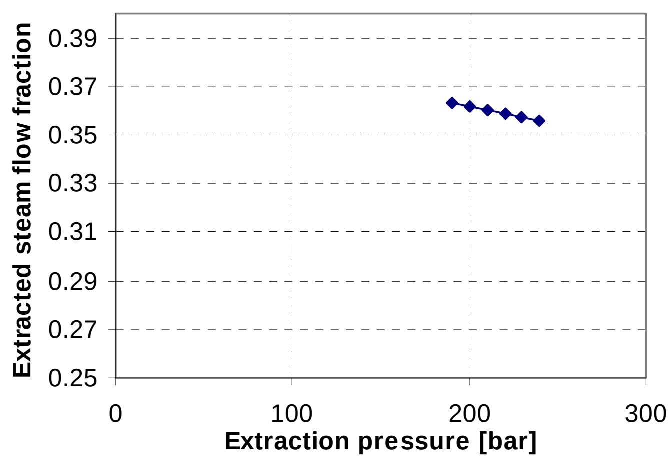


Figure 4.21. Extracted steam flow fraction vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 350°C.

The variation of the pumping power vs the extraction pressure is shown in Figures 4.22, 4.23, 4.24 and 4.25, for the reactor inlet temperatures 250, 280, 300 and 350°C, respectively, and a fixed reactor outlet temperature of 500°C. The pumping power increases weakly with extraction pressure, mostly due to the decrease in thermal efficiency. The reduction in efficiency results in higher thermal power and thus more flow is needed to maintain a constant outlet temperature. The pumping power increases also with reactor inlet temperature, because of the lower enthalpy rise in core. Note that the values of the pumping power are within the target range in all cases.

The steam quality at the turbine outlet was also monitored and found to be greater than the 90% target for all cases analyzed in this section.

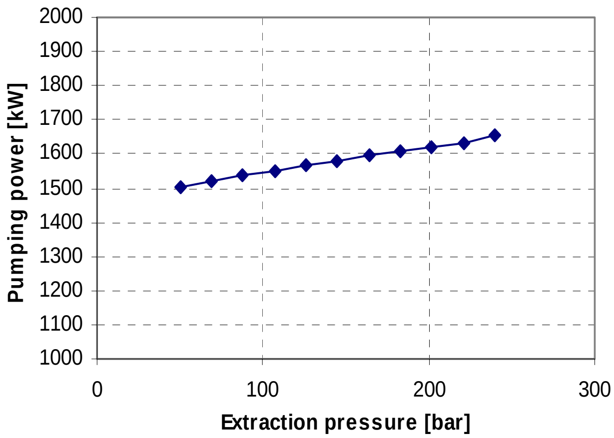


Figure 4.22. Pumping power vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 250°C.

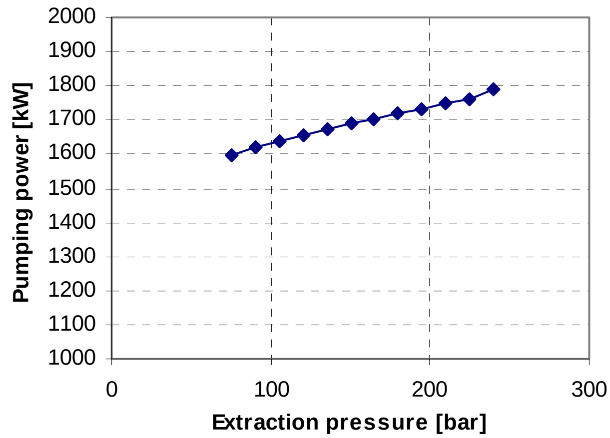


Figure 4.23. Pumping power vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 280°C.

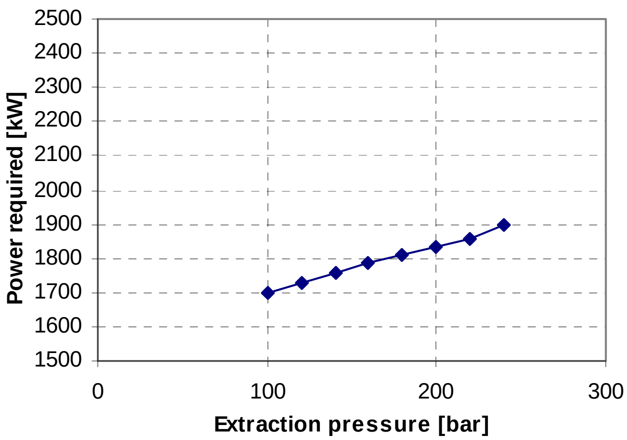


Figure 4.24. Pumping power vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 300°C.

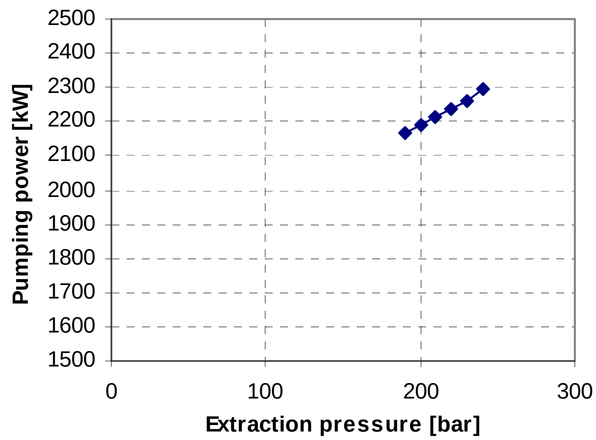


Figure 4.25. Pumping power vs extraction pressure for the supercritical Rankine cycle with feedwater preheating to 350°C.

These parametric studies show that the supercritical Rankine cycle with feedwater preheating offers an attractive combination of relatively high thermal efficiency and low pumping power. Moreover, since the reactor inlet temperature does not seem to greatly affect the thermal efficiency, we can select this parameter based on other considerations. Therefore, we select a reactor inlet temperature of 350°C, which will reduce the temperature and enthalpy rise in the reactor, a great benefit in terms of thermal and mechanical design of the core components.

In order to reduce the reactor temperature and enthalpy rise further, we also evaluate the possibility of decreasing the reactor outlet temperature. Figures 4.26, 4.27 and 4.28 show the thermal efficiency, pumping power and turbine outlet steam quality, respectively, as functions of the reactor outlet temperature, for a reactor inlet temperature of 350°C. The thermal efficiency decreases with decreasing reactor outlet temperature, but not strongly. At lower reactor outlet temperature the pumping power is also somewhat higher, mainly due to the lower enthalpy rise in the reactor. The turbine exit steam quality is lower at lower reactor outlet temperature, and a minimum 480°C is needed to meet the >90% quality target.

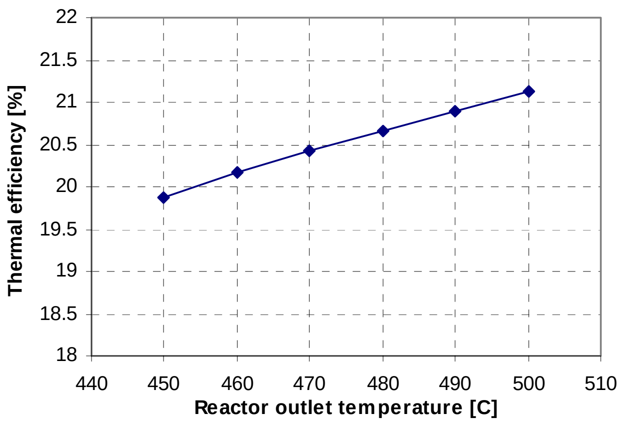


Figure 4.26. Thermal efficiency vs reactor outlet temperature for the supercritical Rankine cycle with feedwater preheating to 350°C.

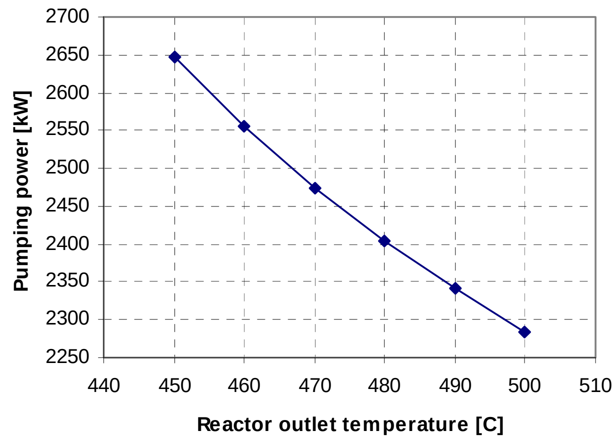


Figure 4.27. Pumping power vs reactor outlet temperature for the supercritical Rankine cycle with feedwater preheating to 350°C

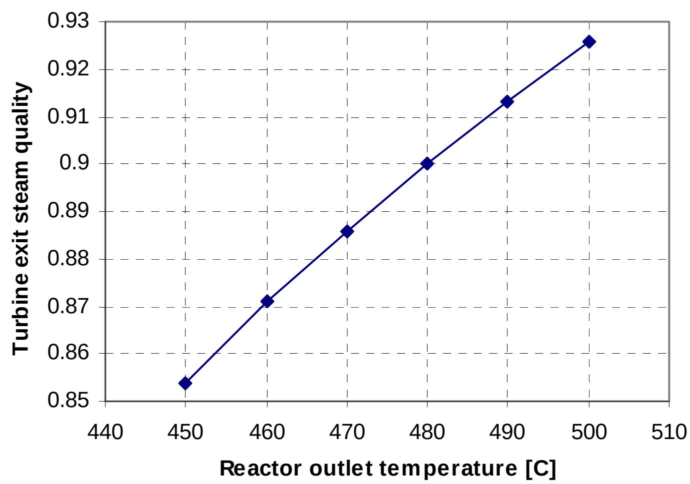


Figure 4.28. Turbine exit quality vs reactor outlet temperature for the supercritical Rankine cycle with feedwater preheating to 350°C.

To further reduce the reactor outlet temperature without taking a penalty on thermal efficiency and exit quality, a moisture separator can be used. The schematic layout of the supercritical Rankine cycle with feedwater preheating and moisture separation is shown in Figure 4.29. The moisture separator splits the turbine into a HP section and a LP section and allows for steam expansion to lower pressure, which increases the thermal efficiency, without violating the exit quality constraint. The separated moisture is throttled and discharged into the condenser. We investigated the sensitivity of the cycle performance to the following two parameters: condenser pressure and reactor outlet temperature. The same assumptions as before apply to the turbine efficiency, pump efficiency, pipe and reactor losses, reactor pressure and net electric power. In all cases the moisture separator pressure is selected to get a steam quality of 0.9 at the exit of the HP turbine. A separation efficiency of 95% was assumed for the moisture separator. The results are shown in Figures 4.30 through 4.32. For given condenser pressure, the thermal efficiency and turbine outlet quality do not depend much on the reactor outlet temperature (Figures 4.30 and 4.31, respectively). However, the pumping power strongly increases with decreasing reactor outlet temperature (Figure 4.32). A 460°C reactor outlet temperature and 4 bar condenser pressure seem a reasonable compromise between elevated thermal efficiency and acceptable pumping power and turbine outlet quality.

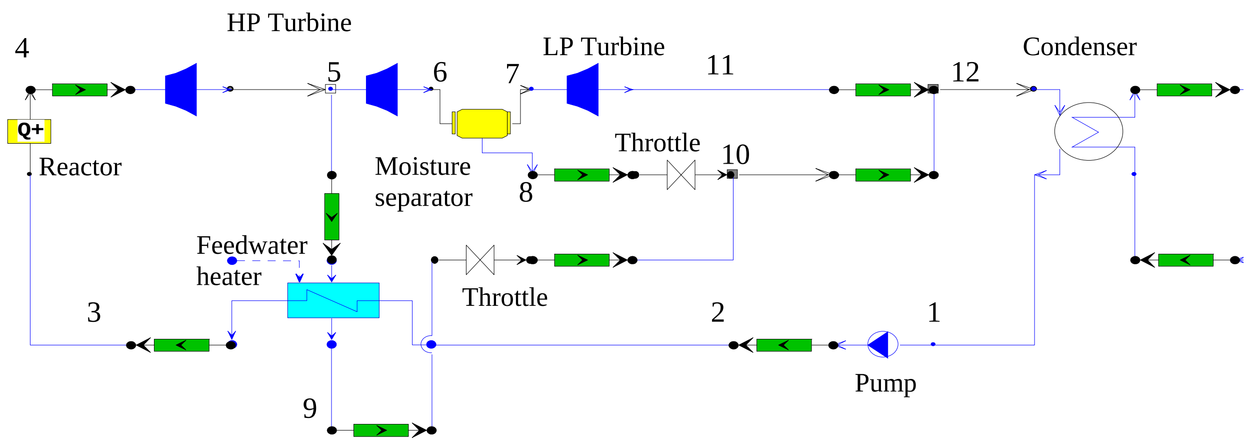


Figure 4.29 The Rankine cycle with feedwater preheating and moisture separation

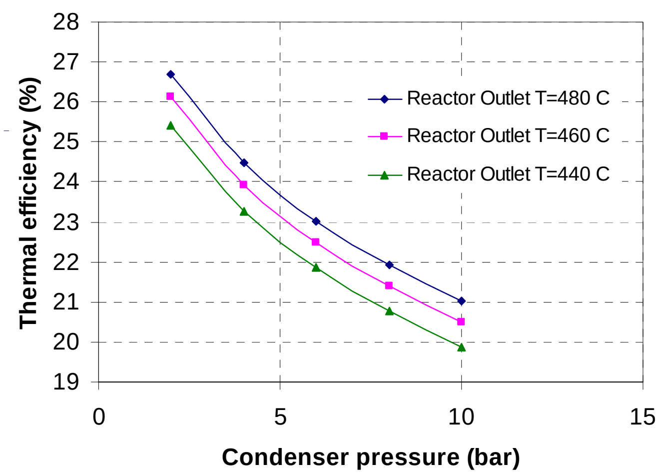


Figure 4.30. Thermal efficiency vs. pressure condenser and reactor outlet temperature for the supercritical Rankine cycle with feedwater preheating and moisture separation.

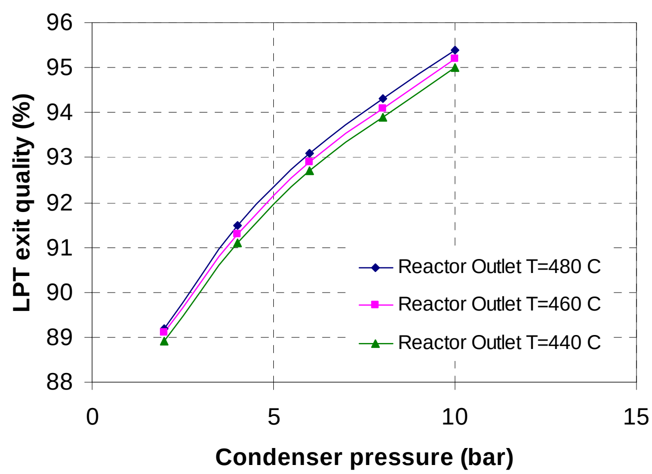


Figure 4.31. LP turbine exit quality vs. pressure condenser and reactor outlet temperature for the supercritical Rankine cycle with feedwater preheating and moisture separation.

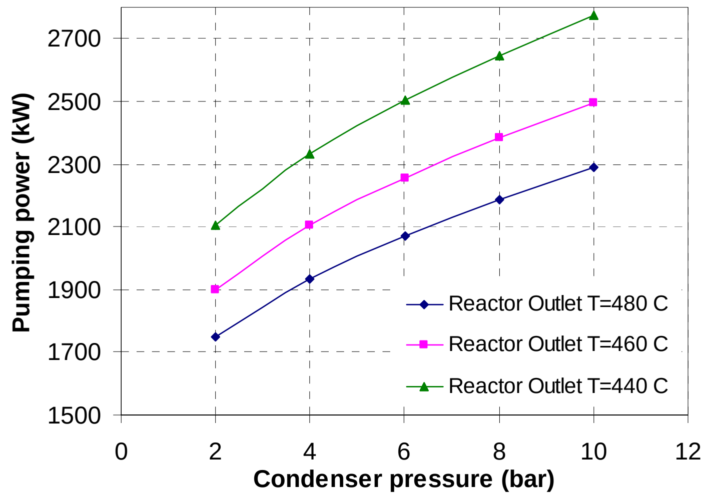


Figure 4.32. Pumping power vs. pressure condenser and reactor outlet temperature for the supercritical Rankine cycle with feedwater preheating and moisture separation.

4.4 Cycle Selection

In Table 4.1 the most attractive cases for the four cycles analyzed in the thermodynamic studies of Sections 4.1, 4.2 and 4.3 are compared. The simple supercritical Rankine cycle offers the greatest hardware simplification, but its high reactor temperature rise and reactor outlet temperature may pose serious problems from the viewpoint of thermal stresses, stability and materials in the core. The all-supercritical Brayton cycle is not a contender, due to its poor thermal efficiency. On the other hand, the supercritical Rankine cycle with feedwater preheating affords acceptable thermal efficiency with lower reactor temperature rise and outlet temperature. The supercritical Rankine cycle with feedwater preheating and moisture separation further improves the performance at even lower reactor temperature. Naturally, such improvement comes at the expense of plant simplification, as an additional component is needed (i.e., the moisture separator) and also the turbine is split into two casings. However, the lower reactor temperature rise and reactor outlet temperature are very significant advantages, so the cycle with feedwater preheating and moisture separation is selected for further analysis in this study.

Table 4.1 Comparison of supercritical Water cycles

Cycle type	Reactor inlet/outlet temp (°C)	Reactor pressure (bar)	Condenser pressure (bar)	Thermal efficiency (%)	Pumping power (kW)	Turbine exit quality
Simple supercritical Rankine	~180/500	~250	10	20.7	1500	0.93
All supercritical Brayton	~415/500	~380	n/a	8.55	6450	Supercritical steam
Supercritical Rankine with feedwater preheating	350/480	~250	10	20.7	2400	0.90
Supercritical Rankine with feedwater preheating and moisture separation	350/460	~250	4	23.99	2100	0.91

Finally, to gain additional confidence in the accuracy of the analyses discussed in this section, we analyze the supercritical Rankine cycle with feedwater preheating and moisture separation by hand. The hand-calculated thermodynamic properties of the various states are compared with the values calculated by Thermoflex for the same configuration, and the agreement is excellent, as shown in Table 4.2. The hand-calculated thermal efficiency is 23.91% vs. the Thermoflex value of 23.99%.

Table 4.2 Thermodynamic states of the supercritical Rankine cycle with feedwater preheating and moisture separation.

Point*	Temperature (°C)		Pressure (bar)		Flow Rate (kg/s)		Enthalpy (kJ/kg)		Entropy (kJ/kg·K)	
	Thermoflex	Hand	Thermoflex	Hand	Thermoflex	Hand	Thermoflex	Hand	Thermoflex	Hand
1	143.6	143.6	4.00	4.00	60.62	60.62	604.63	604.66	1.77646	1.77646
2	147.9	147.9	257.55	257.55	60.62	60.62	639.33	639.14	1.79382	1.79354
3	349.9	349.9	255.00	254.97	60.62	60.62	1622.44	1621.47	3.67522	3.67525
4	460.0	460.0	250.00	250.00	60.62	60.62	3002.08	2999.36	5.74284	5.74284
5	417.1	417.7	188.62	188.62	60.62	60.62	2949.72	2948.74	5.76060	5.76450
6	213.3	213.3	20.38	20.38	34.39	34.39	2608.89	2607.80	5.94461	5.93972
7	213.3	213.3	20.38	20.38	31.13	31.08	2787.02	2788.12	6.30905	6.31038
8	213.3	213.3	20.38	20.38	3.269	3.307	912.93	912.87	2.45536	2.45565
9	157.9	158.2	186.75	186.73	26.23	26.23	677.37	678.48	1.90098	1.90379
10	144.0	144.0	4.04	4.04	29.50	29.54	703.48	704.72	2.01560	2.01634
11	144.0	144.0	4.04	4.04	31.13	31.08	2553.29	2554.30	6.44716	6.45054
12	143.6	143.6	4.00	4.00	60.62	60.62	1653.23	1653.00	4.29528	4.29193

* Numbers refer to points in Figure 4.29.

5. Engineering Design of the Selected Power Cycle

In light of the conclusions in Section 4.4 the preliminary engineering design was performed for a supercritical Rankine cycle with feedwater preheating and moisture separation. The cycle and the associated T-s diagram are shown in Figures 5.1 and 5.2, respectively. The thermodynamic states are reported in Table 5.1. The mass flow rates in this table and in Figure 5.1 are for one power conversion system (PCS) train. The thermal and net electric power of the PCS are 43 MWt and 10.376 MWe, respectively, thus resulting in a net thermal efficiency of 24.13%.

The PEACE module in Thermoflex provides realistic engineering estimates for size and performance of some key components such as the turbo-generator, the condenser, the feedwater heater (FWH) and the pumps. A summary of the assumptions and inputs used in this study is reported in Table 5.2. The PEACE output for the turbo-generator is shown in Figures 5.3-5.7 and Tables 5.3-5.5. The PEACE output for the feedwater heater is shown in Figures 5.8 and 5.9 and Tables 5.6-5.8. The PEACE output for the condenser is shown in Figures 5.10-5.12 and Tables 5.9-5.12. The PEACE output for the feedwater pump is shown in Figure 5.13 and Tables 5.13 and 5.14. The PEACE output for the condenser pump is shown in Figure 5.14 and Tables 5.15 and 5.16. Unfortunately PEACE does not provide engineering estimates for the moisture separator, so we generated our own estimates for this component, as shown in Figures 5.15 and 5.16 and Table 5.17. The pipe size and operating conditions are reported in Table 5.18.

The reasonableness of the PEACE outputs (heat transfer coefficients, pressure drops, tube sizing, shell sizing, etc.) for the FWH, condenser and pipes was systematically verified with hand calculations. For the feedwater pump and the turbo-generator we verified the PEACE output with Flowserve Corp. and GE Energy, respectively.

The SolidEdge 3D CAD software was used to develop a first cut of the overall three dimensional layout of the plant. The reactor vessel is assumed to be 3 m high and 2 m in diameter, a reasonable extrapolation based on power scaling from a commercial reactor vessel. The control rod drive mechanisms on top of the vessel are assumed to be 1 m high. The plant layout is shown in Figures 5.18 and 5.19.

The main results of the engineering design are as follows:

1) The analysis with more realistic description of the components confirmed that a supercritical Rankine cycle with feedwater preheating and moisture separation can meet all the performance targets of this study, i.e., acceptable thermal efficiency, pumping power and turbine outlet steam quality.

2) The shell-and-tube FWH is very large, due mostly to the length of the drain cooler section. As shown in Figures 5.18 and 5.19, a reasonable PCS layout is possible even with this large FWH, however, alternatives should be explored. A possibility is to use a compact printed circuit heat exchanger (PCHE), which could reduce the size of the FWH by a factor of up to 6. PCHEs are compatible with steam and water at our conditions, according to Heatric, the world leading manufacturer of PCHEs. Another approach to reduce the FWH size somewhat is to use one FWH for both PCS trains, if this is acceptable. Both approaches should be explored in future work. In addition, potential issues with respect to inspection and leakage of the PCHE will have to be evaluated.

3) Because PEACE is intended for commercial power plants, it automatically includes a gear box in the turbo-generator if the user selects a shaft speed other than 3600, 3000, 1800 or 1500 rpm. The gear box is big. In our PCS we will use a custom-made generator rotating at high speed without the gear box. The high rotation speed will also enable a considerable reduction of the generator size. That is, a ~10 MW generator with permanent magnets and a rotation speed of 13000 rpm would shrink to 2 m length (not including the exciter) and 1 m diameter. This would be the same generator used in the supercritical CO_2 cycle also being studied at MIT.

The layout of the plant with a PCHE FWH (assumed to be 'only' 3 times shorter than the shell-and-tube FWH), without the gear box and with a full-speed generator is shown in Figures 5.20 and 5.21. Note that with these modifications the plant becomes significantly more compact.

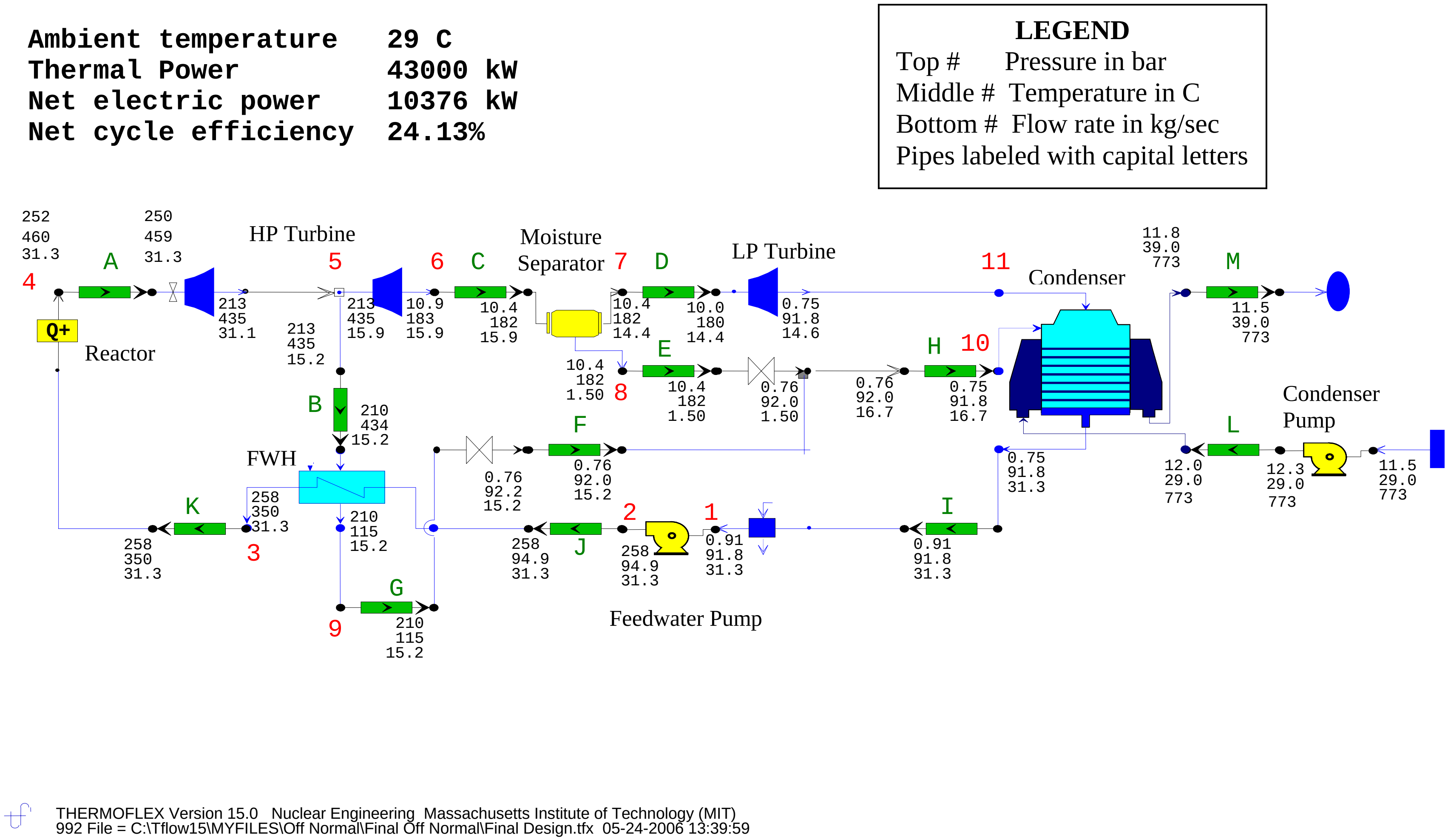


Figure 5.1 Cycle schematic. Note that between the condenser and the FW pump a makeup/blowdown component has been installed. This component ensures that any leakages (from turbines, etc.) and any numerical artifacts affecting the mass balance are accounted for and the mass balance of the coolant is maintained.

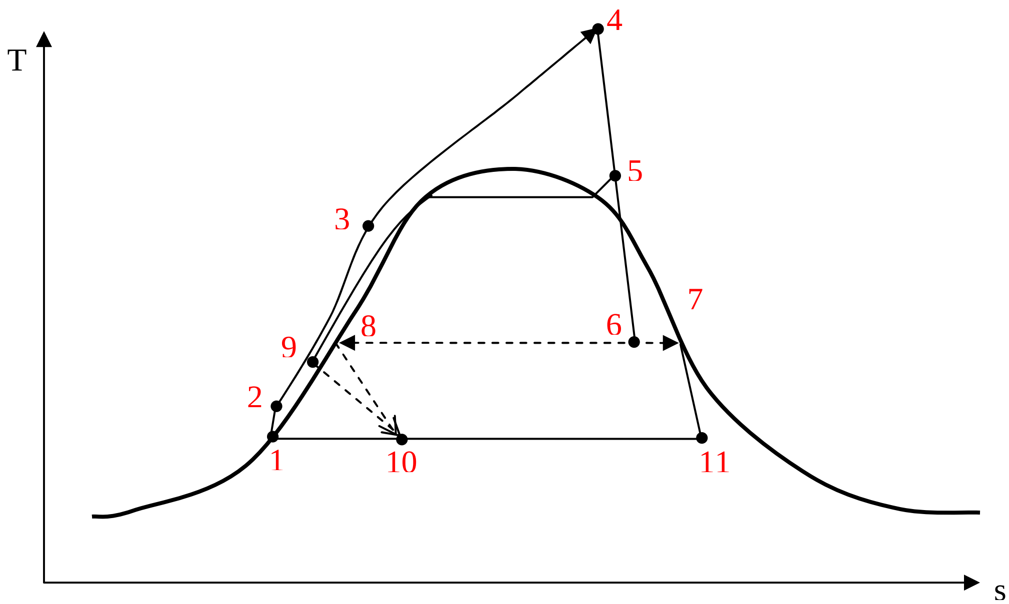


Figure 5.2 T-s diagram

Table 5.1 Thermodynamic data

Thermodynamic Point	Pressure [bar]	Temperature [C]	Quality [%]	Flow Rate [kg/s]	Enthalpy [kJ/kg]	Entropy [kJ/(C*kg)]
1	0.91	91.8	Subcooled	31.3	384.43	1.21365
2	258	94.9	Supercritical	31.3	417.22	1.23053
3	258	350	Supercritical	31.3	1621.80	3.67393
4	252	460	Supercritical	31.3	2996.77	5.73642
5	213	435	Superheated	31.1	2973.28	5.75089
6	10.9	183	89.9	15.9	2577.54	6.11274
7	10.4	182	99.5	14.4	2766.39	6.56278
8	10.4	182	0.00	1.50	770.01	2.15463
9	210	115	Subcooled	15.2	497.11	1.45597
10	0.75	91.8	6.00	16.7	521.66	1.58773
11	0.75	91.8	90.0	14.6	2433.01	6.83145

Table 5.2 Component assumptions

Component	Assumptions
Reactor	• Thermoflex Heat Adder • Pressure loss of 2% across core • Outlet Temperature (T_{out}) = 460 C
High Pressure Turbine	• Dry Step Isentropic Efficiency determined by Thermoflex • Inlet pressure (P_{in}) = 250 bar • 2 HP End leaks (0.075 kg/s each), one of which leads to the LPT crossover; the other leads to the Steam Seal Regulating (SSR) system • 1 valve stem leak (0.075 kg/s) leading to LPT crossover • 1 LP End leak (0.075 kg/s) leading to SSR system • Single flow path • Single casing
Low Pressure Turbine	• Dry Step Isentropic Efficiency determined by Thermoflex • No leaks • Single flow path • Single exhaust end • Single casing
Turbine Assembly	• Single shaft • 0.2% miscellaneous auxiliary load • Since Thermoflex forces us to assume a gear box with efficiency of 98.5%, we assume a generator efficiency of 100%
Condenser	• Condenser pressure (P_{cond}) = .75 bar • Water head to condensate outlet = 0 m • No condensate subcooling • Temperature rise of 10 C for cooling water (seawater) • Seam welded tube • Tube bundle aspect ratio of unity • Hotwell storage time of 1 minute (similar to storage time in commercial plants) • Non-condensable gas fraction in steam (1%). The non-condensable gas fraction in a BWR condenser is <0.1%, so this assumption is conservative • Heat transfer coefficient correction factor due to non-condensable gases is 0.9949 • Mechanical vacuum pump for non-condensable removal
Feed-water Heater	• Minimum pinch of 10 C • Heating steam exit T minus feedwater inlet T is 20 C • Heating steam saturation T minus feedwater outlet T is 20 C
Moisture Separator	• Separation efficiency of 95%
Feedwater Pump	• Multi-stage centrifugal pump • Variable speed (determined by Thermoflex to be 3370 RPM at 100% thermal power) • 97% mechanical efficiency • Single pump per loop • Default isentropic efficiency of 85% • 33% flow margin to enable operation at 150% flow (for off-normal conditions) • 0% head margin
Condenser Pump	• Single stage centrifugal pump • Thermoflex estimates speed (fixes at 600 RPM) • 97% mechanical efficiency • Default isentropic efficiency of 85%
Pipes	• Sized by velocity with limited pressure drop for each leg • Length estimated due to size and layout of plant components

Turbine

Our turbine is a condensing, multi-valve, multi-stage, two-casing, non-reheat, single-shaft machine. The HP section has a single automatic steam extraction that serves the feedwater heater. There is a moisture separator between the HP and LP sections. A high shaft speed (13000 rpm) was selected to reduce the turbine size. The PEACE estimates for various turbine-generator parameters, including the dry step and casing efficiencies, number of stages and physical size were discussed with engineers at GE Energy and found reasonable. However, concern was expressed about the following two items:

- the relatively low exit quality (0.9), which may cause blade erosion problems, and
- the high pressure ratio in the HP section, which may pose a challenge in the design of the HP casing because of temperature and pressure gradients.

Regarding the issue of erosion, it should be noted that steam turbines in commercial nuclear plants typically operate with exit qualities around 0.9, sometimes even lower, and the erosion issue is managed through design. That is, the moving blades have an erosion shield made of a high hardness material such as Stellite or titanium nitride. Also, the moisture is removed through suction slots in the fixed blades or drainage channels on the turbine casing.

The issue of high pressure and temperature gradients in the HP section should be addressed when the actual design of the turbine will be performed.

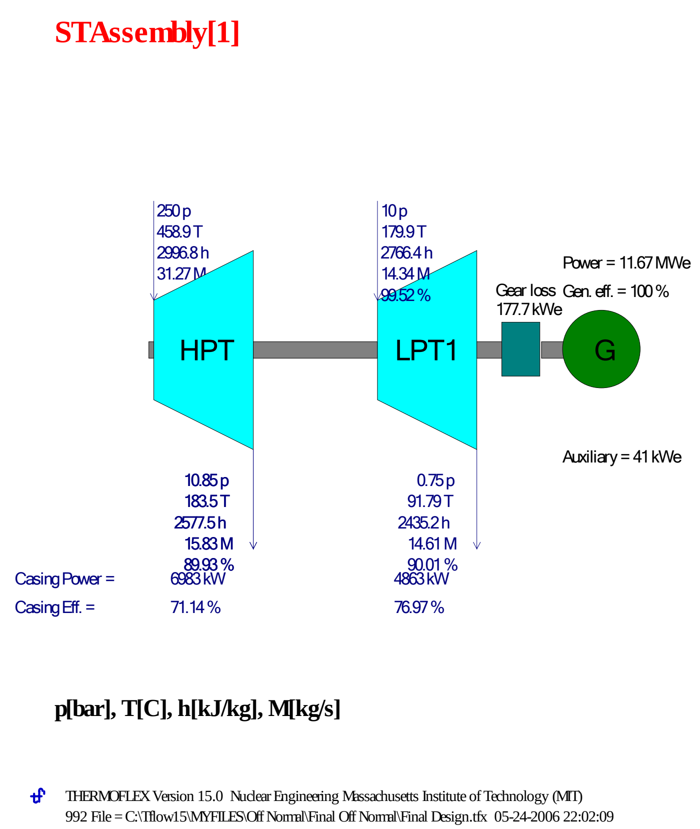


Figure 5.3 Schematic for steam turbine assembly.

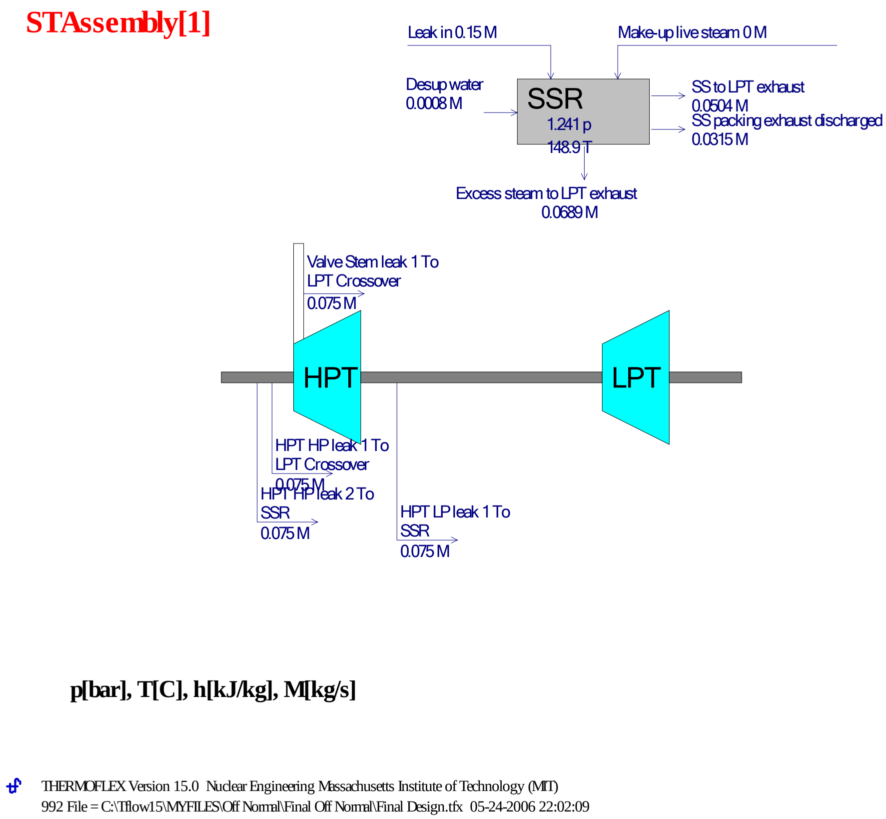
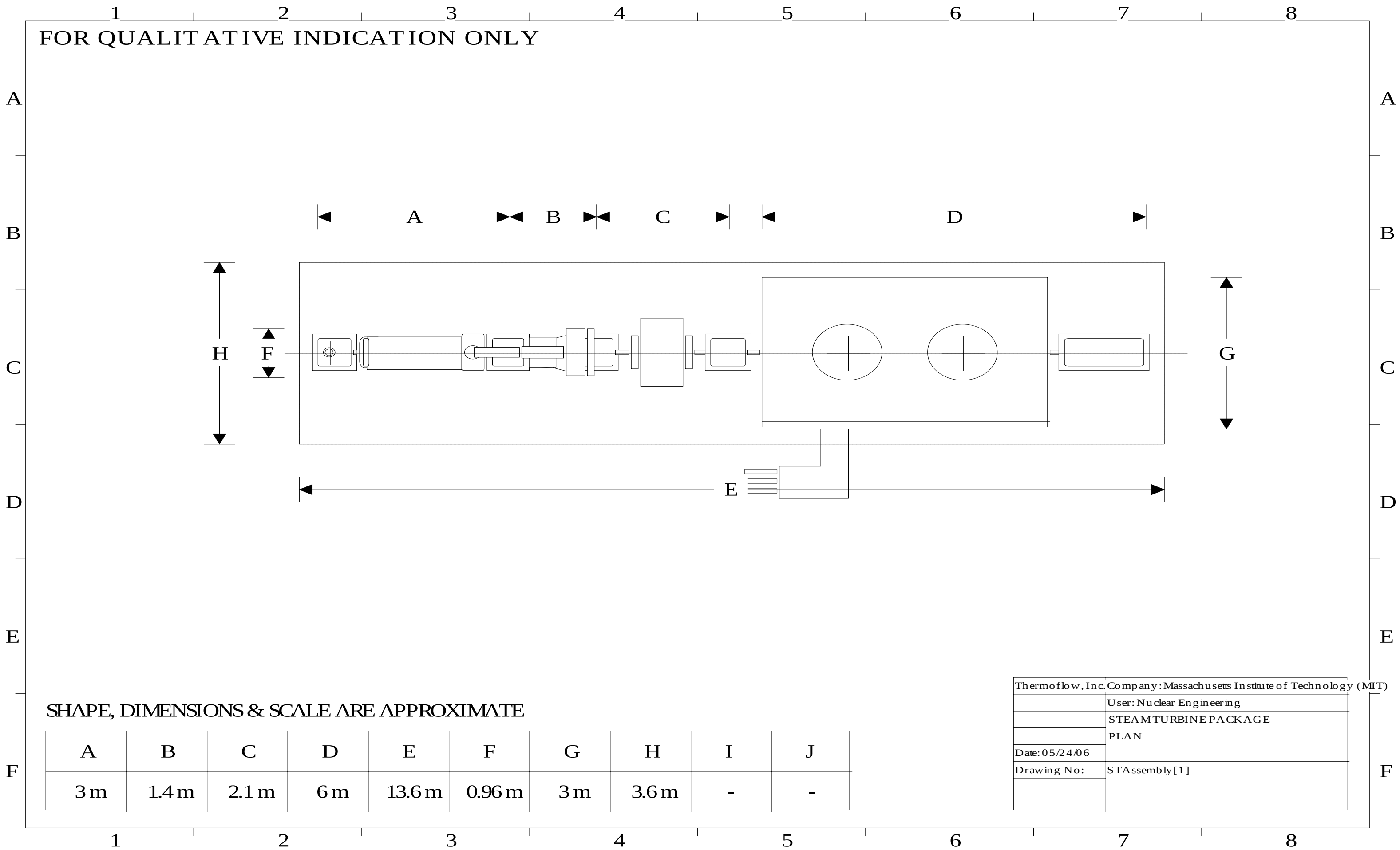


Figure 5.4 Turbine Assembly Leakage Schematic



A	B	C	D	E	F	G	H	I	J
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Figure 5.5 Top view of Steam Turbine-Generator

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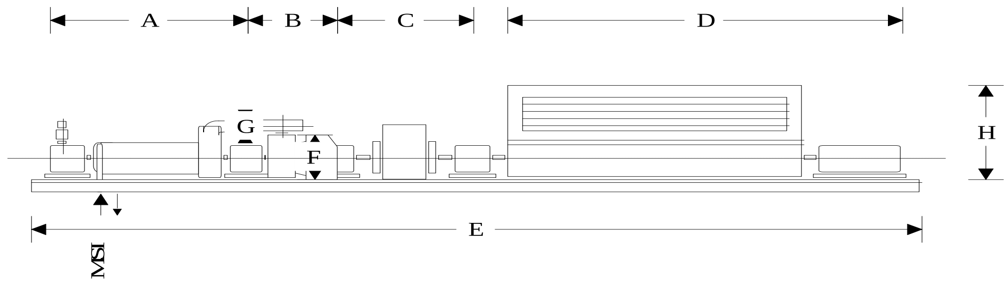

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A	B	C	D	E	F	G	H	I	J
3 m	1.4 m	2.1 m	6 m	13.6 m	0.91 m	0.23 m	1.9 m	-	-

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	User: Nuclear Engineering
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	ELEV
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Figure 5.6 Side view of Steam Turbine-Generator.

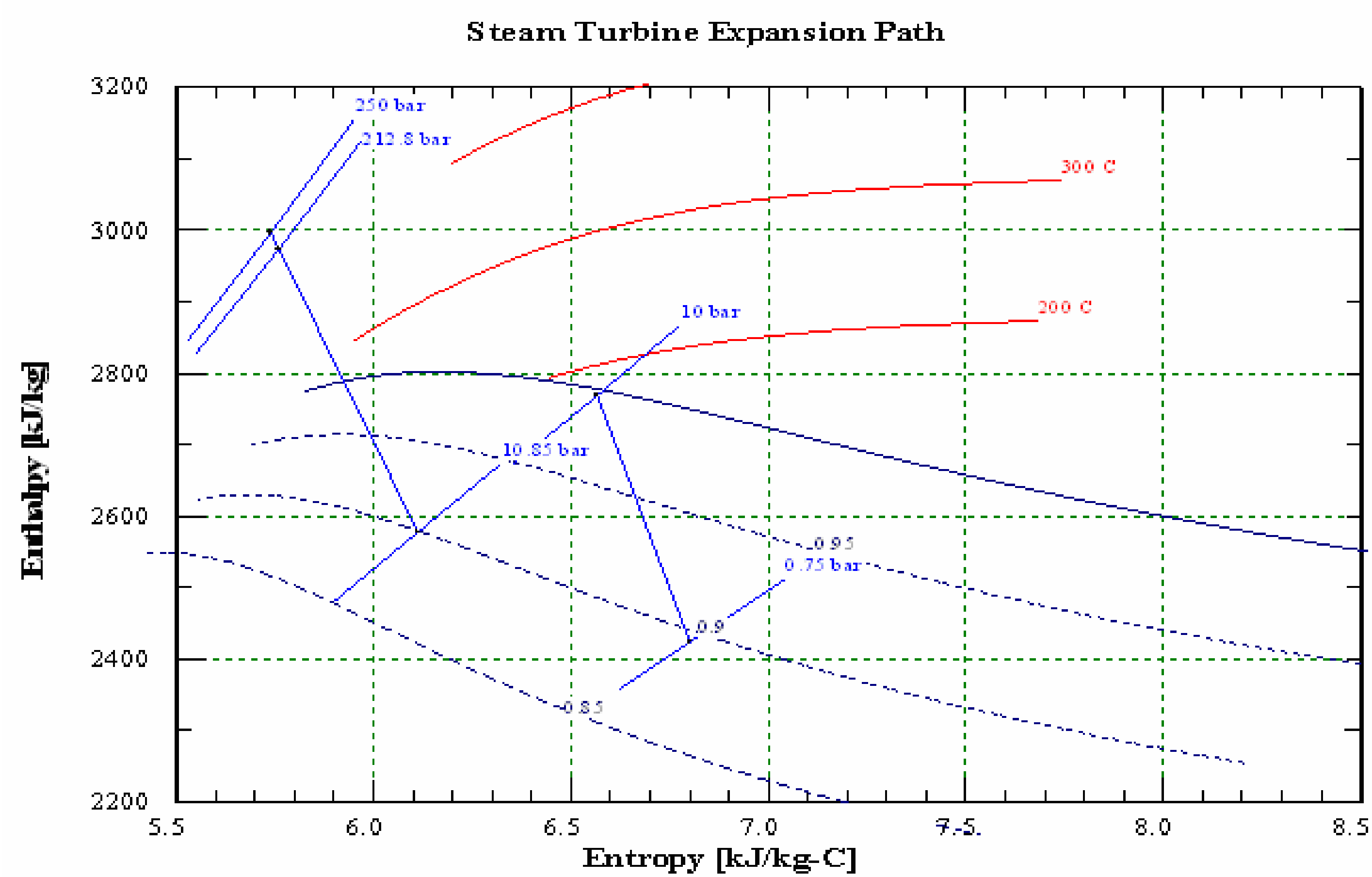


Figure 5.7 Steam Turbine Expansion Path

Table 5.3 Turbine description and physical size.

Nameplate Capacity	11.85 MVA
Steam Turbine Type	Condensing, Non-Reheat
Nameplate Throttle Pressure	250 bar
Nameplate Throttle Temperature	458.9 C
Nameplate Throttle Massflow	39.09 kg/s
Exhaust End Type	Down Draft
Number of LPT Exhaust Annuli	1
Number of Auto-Extraction/Auto-Admission Ports	1
Estimated Weights & Dimensions	
Steam Turbine Length (including Gear Box)	6.461 m
Steam Turbine Width	0.9562 m
Steam Turbine Weight (including Weight of Gear Box)	12,010 kg
Gearbox Length	2.081 m
Generator Length (Including Exciter)	6.032 m
Generator Width	2.971 m
Generator Weight	31,310 kg
Overall ST and Generator Length	12.49 m
Overall ST and Generator Width	2.971 m
Overall ST and Generator Weight	43,320 kg
Foundation Length	13.59 m
Foundation Width	3.565 m

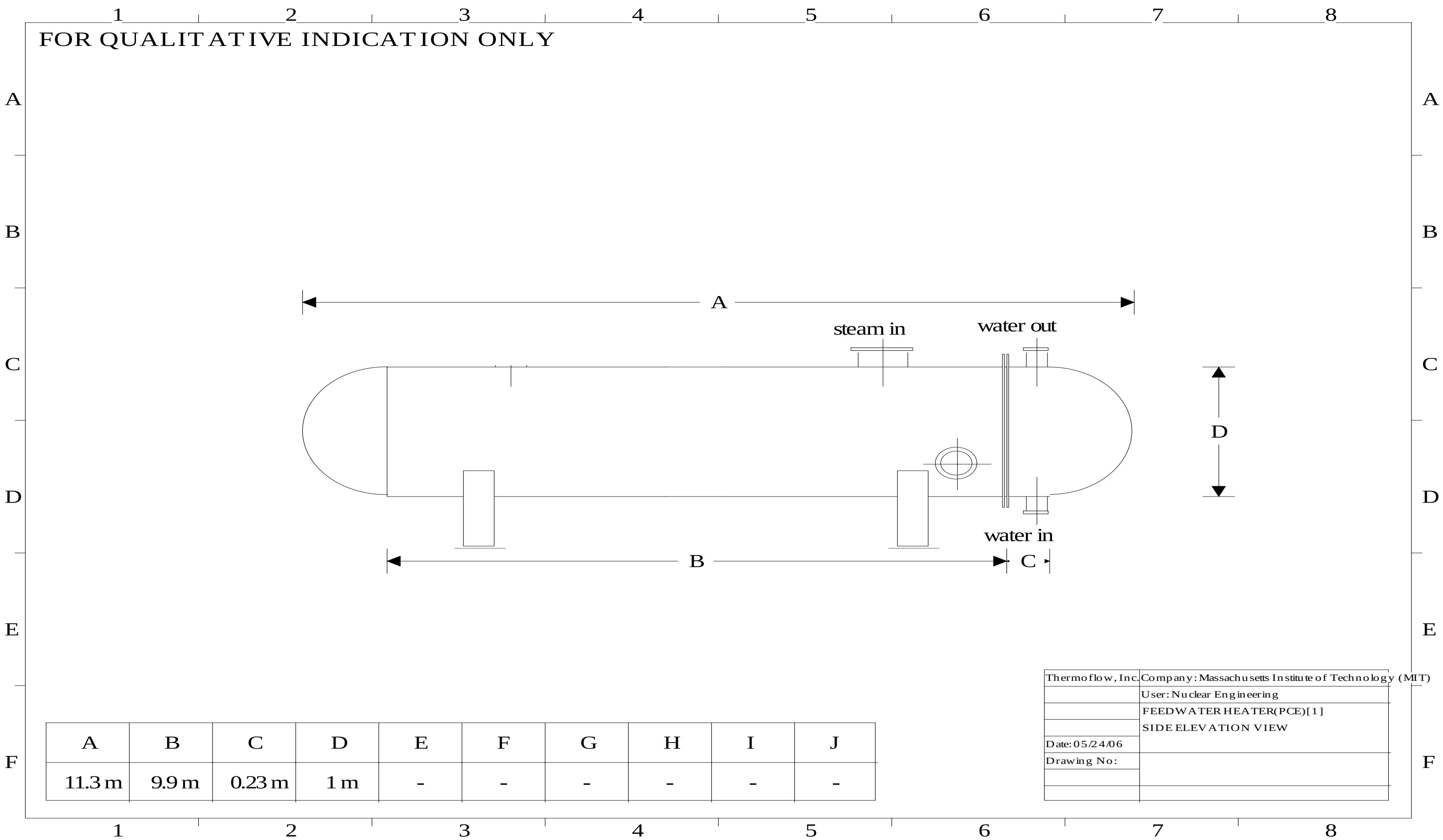
Table 5.4. Turbine thermodynamic data at nominal conditions.

Generator Power	11.67 MWe
Generator / Gear Box Efficiency*	100 % / 98.5 %
Lube Oil Pump Auxiliary	17.5 kWe
Miscellaneous Auxiliary Power	23.34 kWe
Shaft Speed	13000 RPM
Shaft Power	11.85 MWe
Gear Box Loss	177.7 kWe
Generator Loss	0 kWe
Steam Turbine Main Steam	
Mass Flow	31.27 kg/s
Pressure	250 bar
Temperature	458.9 C
Enthalpy	2996.8 kJ/kg
Casings	
HPT	
Casing Shaft Power	6983 kWe
Casing Efficiency	71.14 %
Inlet Steam	
Mass Flow	31.27 kg/s
Pressure	250 bar
Temperature	458.9 C
Enthalpy	2996.8 kJ/kg
Exit Steam	
Mass Flow	15.83 kg/s
Pressure	10.85 bar
Temperature	183.5 C
Enthalpy	2577.5 kJ/kg
Quality	89.93 %
LPT	
Casing Shaft Power	4863 kWe
Casing Efficiency	76.97 %
Inlet Steam	
Mass Flow	14.34 kg/s
Pressure	10 bar
Temperature	179.9 C
Enthalpy	2766.4 kJ/kg
Quality	99.52 %
Exit Steam	
Mass Flow	14.61 kg/s
Pressure	0.75 bar
Temperature	91.79 C
Enthalpy	2435.2 kJ/kg
Quality	90.01 %

*Since Thermoflex forces us to assume a gear box with efficiency of 98.5%, we assume a generator efficiency of 100%

Table 5.5 Turbine group data at nominal conditions.

Casing	HPT (before steam extraction)	HPT (after steam extraction)	LPT
Thermodynamic data			
Number of flow paths	1	1	1
Dry step efficiency [%]	61.43	69.94	82.79
Group efficiency [%]	61.78	71.1	78.08
Shaft speed [RPM]	13000	13000	13000
Mechanical loss [kW]	1.459	12.53	9.745
Shaft power [kW]	727.9	6255	4863
Group Inlet			
Mass flow [kg/s]	31.27	15.83	14.34
Pressure [bar]	250	212.8	10
Temperature [C]	458.9	435.4	179.9
Enthalpy [kJ/kg]	2996.8	2973.3	2766.4
Steam quality	Supercritical	Supercritical	0.9952
Pressure after inlet loss [bar]	250	212.8	10
Leakage massflow into group (net) [kg/s]	-0.225	0	0.15
Group Exit			
Mass flow [kg/s]	31.05	15.83	14.61
Pressure [bar]	212.8	10.85	0.75
Temperature [C]	435.4	183.5	91.79
Enthalpy [kJ/kg]	2973.3	2577.5	2435.2
Steam quality	Supercritical	0.8993	0.9001
Leaving loss [kJ/kg]	0	0	8.665
Leakage massflow into group (net) [kg/s]	0	0	0.1193
Blading (per flow path)			
Number of stages	2	21	6
Inlet volume flow [m^3/s]	0.2968	0.1735	2.801
Nozzle area [m^2]	0.0022	0.0005	0.0096
Exit volume flow [m^3/s]	0.3403	2.561	28.75
Annulus area [m^2]	N/A	N/A	0.1981
Annulus velocity [m/s]	N/A	N/A	145.1
Last stage pitch diameter [mm]	N/A	N/A	460.8
Last stage blade length [mm]	N/A	N/A	136.9



A	B	C	D	E	F	G	H	I	J
11.3 m	9.9 m	0.23 m	1 m	-	-	-	-	-	-

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Figure 5.8 Side view of FWH.

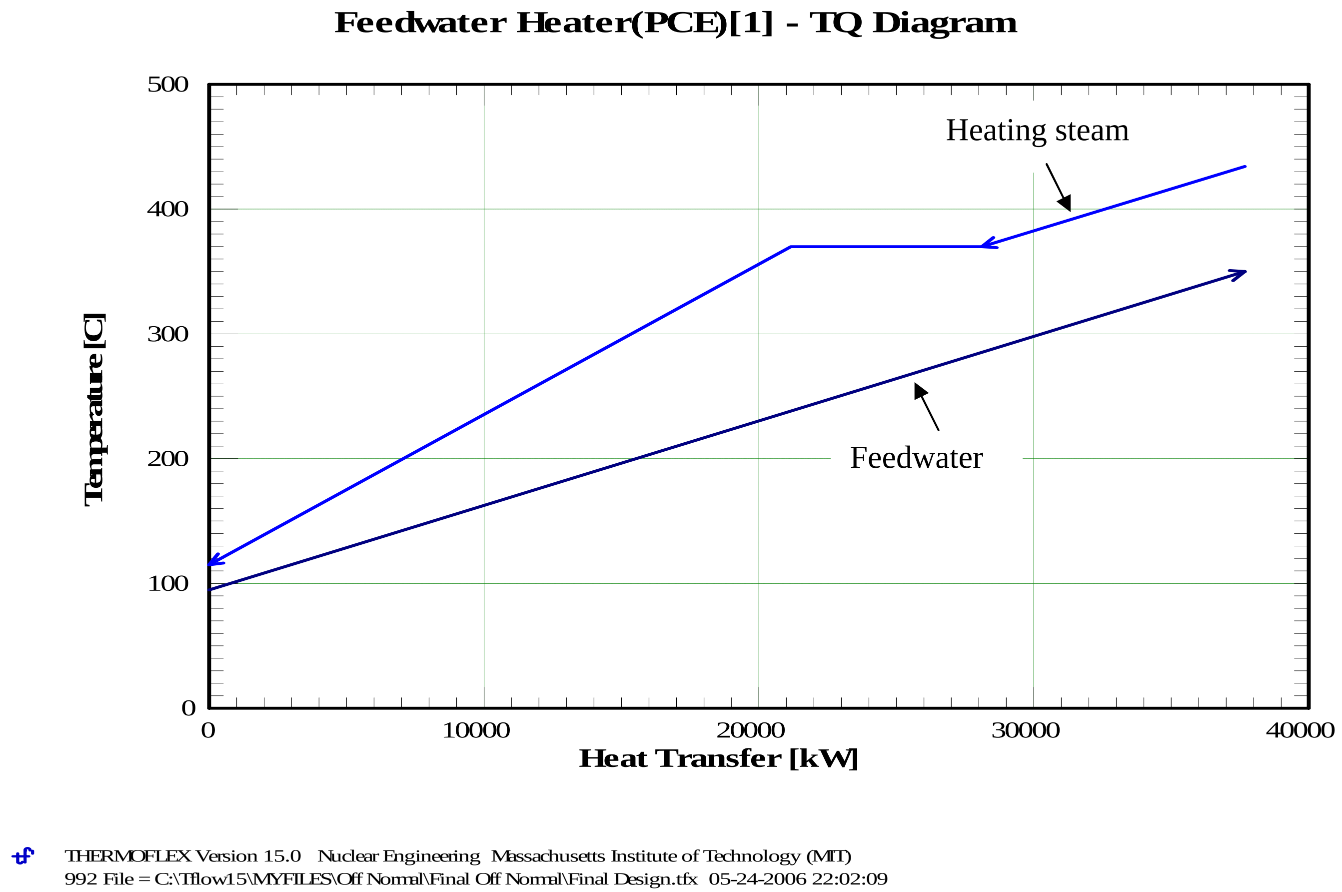


Figure 5.9 T-Q diagram for FWH.

Table 5.6 FWH Tube Physical Dimensions

Component	Dimension
Total Heat Transfer Area	296 m^2
Number of Tubes per Pass	299
Number of Passes	2
Number of Tubes in Heater	598
Tube Length per Pass	9.912 m
Tube Outside Diameter	15.88 mm
Tube Wall Thickness	2.108 mm
Tube weight, dry	4310 kg
Tube Pitch	23.02 mm

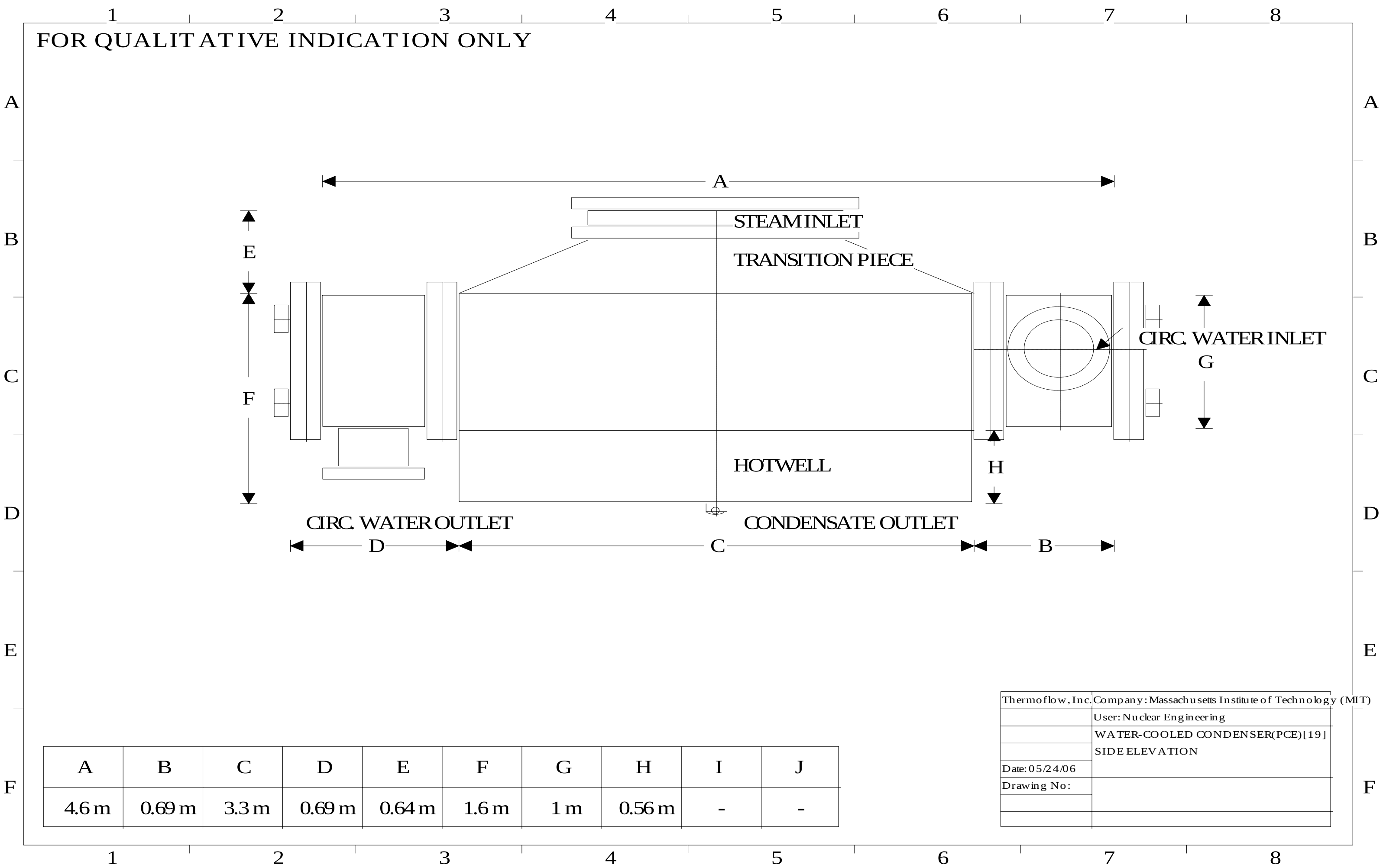
Table 5.7 FWH Shell Physical Dimensions

Component	Dimension
Shell Length	9.912 m
Shell Inner Diameter	0.8575 m
Shell Wall Thickness	88.9 mm
Number of Support Plates (baffles)	59
Tube Sheet Thickness	82.55 mm
Overall Length	11.3 m
Overall Outer Diameter	1 m
Total Dry Weight	30,140 kg
Total Operating Weight	31,000 kg

Table 5.8 FWH Heat Balance Data

Parameter	Value
Fouling Factor	$0.0581*10^{-3}$ m^2-C/W
Coolant Velocity	1.143 m/s
Coolant Reynolds Number	91,035
Coolant Pressure Drop	0.1989 bar
Desuperheating Heat Transfer Coefficient	1696.8 W/(m^2-C)
Condensing Heat Transfer Coefficient	3347 W/(m^2-C)
Drain Cooler Heat Transfer Coefficient	1725.4 W/(m^2-C)
Desuperheating LMDT*	84.45 C
Condensing LMDT*	106.5 C
Drain Cooler LMDT*	59.27 C

* LMDT = Log mean temperature difference



A	B	C	D	E	F	G	H	I	J
4.6 m	0.69 m	3.3 m	0.69 m	0.64 m	1.6 m	1 m	0.56 m	-	-

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Figure 5.10 Side view of condenser.

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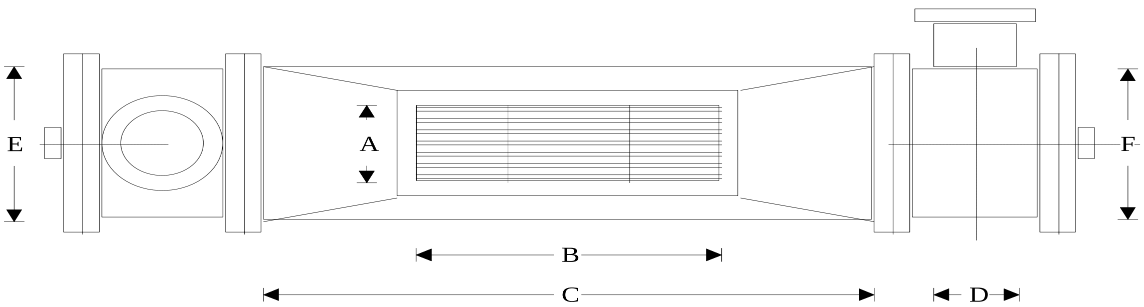

A	B	C	D	E	F	G	H	I	J
0.53 m	1.6 m	3.3 m	0.46 m	1.1 m	1 m	-	-	-	-

Thermoflow, Inc.	Company: Massachusetts Institute of Technology (MIT)
	User: Nuclear Engineering
	WATER-COOLED CONDENSER(PCE)[19]
	PLAN
Date: 05/24/06	
Drawing No:	

THERMOFLEX Version 15.0 Nuclear Engineering Massachusetts Institute of Technology (MIT)
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Figure 5.11 Top view of condenser.

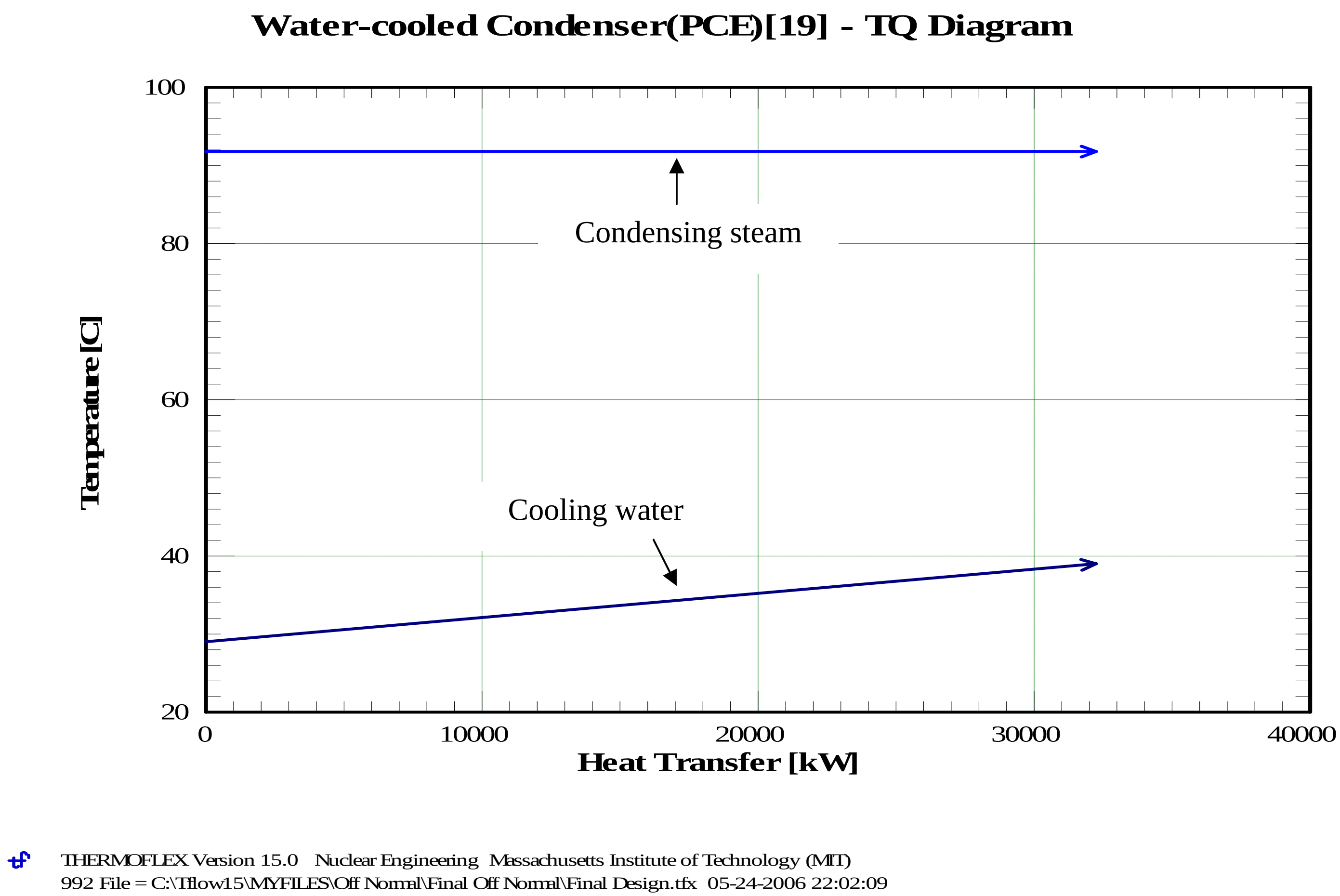

Figure 5.12 T-Q diagram for condenser.

Table 5.9 Condenser Heat Balance Data

Condenser Operating Data	Value	Units
Condenser pressure	0.75	bar
Condenser saturation temperature	91.79	C
Heat rejection	32,246	kW
Inlet Steam		
Pressure	0.75	bar
Temperature	91.79	C
Mass flow	14.61	kg/s
Enthalpy	2435.2	kJ/kg
Condensate at bottom of hotwell		
Pressure	0.75	bar
Temperature	91.79	C
Mass flow	31.32	kg/s
Enthalpy	384.4	kJ/kg
Inlet Water		
Pressure*	11.68	bar
Temperature	29.01	C
Mass flow	772.6	kg/s
Enthalpy	122.6	kJ/kg
Exit Water		
Pressure*	11.52	bar
Temperature	39.01	C
Mass flow	772.6	kg/s
Enthalpy	164.3	kJ/kg
Flash-in Stream		
Temperature	91.79	C
Mass flow	16.03	kg/s
Enthalpy	521.6	kJ/kg
Heat Transfer Data		
Fouling factor	0.0324×10^{-3}	m^2-C/W
Cleanliness factor	90.02	%
Water velocity	2.34	m/s
Water Reynolds number	75,994	
Water Prandtl number	4.934	
Water Nusselt number	349.7	
Condensing heat transfer coefficient	6947	W/m^2-C
Water-side heat transfer coefficient	9059	W/m^2-C
Overall heat transfer coefficient	2916.5	W/m^2-C
Overall UA**	559.6	kW/C
Water pressure drop in tubes and water box	0.1607	bar

* Assumes seawater at 100 m depth
** UA = (overall heat transfer coefficient) x (heat transfer area)

Table 5.10 Condenser Tube Physical Dimensions

Component	Dimension
Effective Surface Area	191.9 m^2
Number of Condenser Passes	1
Number of Tubes	735
Tube Length	3.3 m
Tube O.D.	25.4 mm
Tube I.D.	23.98 mm
Wall Thickness	0.7112 mm
Dry Weight	1060 kg
Tube Pitch	40.64 mm

Table 5.11 Condenser Shell Physical Dimensions

Component	Dimension
Shell Thickness	6.35 mm
Number of Support Plates	4
Hotwell Depth	0.56 m
Dry Weight	3380 kg
Footprint Area	4.902 m^2
Length	4.6 m
Width	1.1 m
Height	2.3 m

Table 5.12 Condenser Operating Parameters

Component	Dimension
Hotwell Volume	1.950 m^3
Operating (wet) weight	6420 kg

Feedwater Pump

Our feedwater pump is a centrifugal (radial) machine raising the feedwater pressure from the condenser pressure (<1 bar) to the reactor pressure (over 250 bar). The total head, H_{tot}, is 2622 m (or 8602 ft) and the flow is 31.3 kg/s (or 496 gpm). To select the number of stages of a centrifugal pump the following equation relating stage head, pump speed and pump flow rate can be used:

$$H = \left[\frac{N}{N_s} \sqrt{Q} \right]^{4/3} \qquad (5.1)$$

where H is the stage head (in ft), N is the actual speed (in rpm), Q is the flow rate (in gpm) and N_s is the normalized dimensionless speed. Typical values of N_s for centrifugal pumps range from 600 to 1200. If N_s is fixed (say 600), then Eq. (5.1) gives H and thus the number of stages as H_{tot}/H. Now, PEACE automatically limits the pump shaft speed to 3600 rpm. With such "low" speed the number of stages is high, i.e., 13. However if a higher speed is selected (say 6500 rpm [3]), the number of stages can be reduced down to 6. For the design of the feedwater pump we consulted with experts at Flowserve Corp., a well known manufacturer of pumps for power plants. A 6500 rpm speed is considered by the Flowserve engineers a reasonable speed that should not create vibration-related problems, in either a vertical or horizontal pump configuration. The higher speed also reduces the physical size of the pump. Therefore, the PEACE estimates for the feedwater pump size at 3600 rpm should be considered an upper bound.

Table 5.13 Feedwater Pump Operating Characteristics

Item	Value
Apparent Isentropic Efficiency	81.33 %
Shaft Speed*	3370 rpm
Hydraulic Work	1025.6 kW
Shaft Work	1070 kW
Mechanical Efficiency	95.84 %
Motor Efficiency	96.2 %
Electricity Consumption	1170.8 kWe
Mass Flow	31.27 kg/s
Pump Head	257.1 bar/2621.6 m H2O

* PEACE automatically limits the pump shaft speed to 3600 rpm. However the optimal speed may be higher, as discussed above.

Table 5.14 Feedwater Pump Physical Dimensions

Item	Dimension
Baseplate Length	5.198 m
Baseplate Width	1.342 m
Pump Weight	5510 kg
Motor Weight	4984 kg

[3] This speed is half the speed of the turbine-generator. Therefore, the pump will need a four- (vs. two-) pole electric motor, to run on the electricity generated by the turbine-generator.

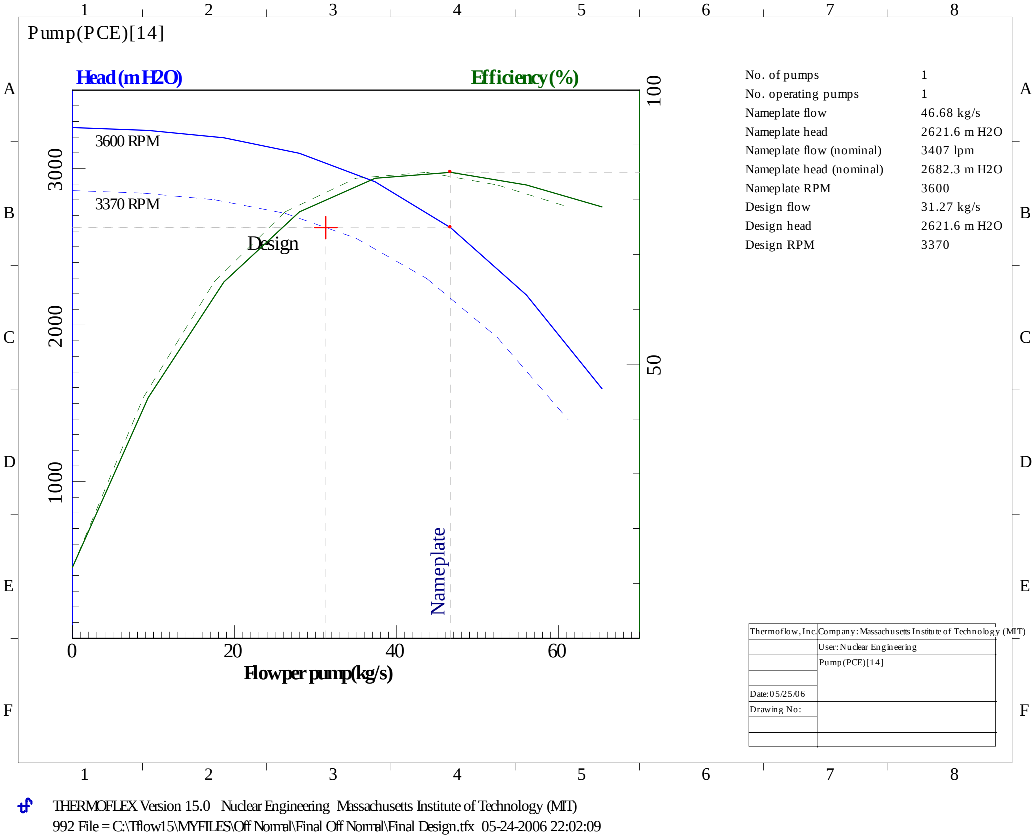

Figure 5.13 Feedwater Pump Curve.

Table 5.15 Condenser Pump Operating Characteristics

Item	Value
Apparent Isentropic Efficiency	85%
Shaft Speed	600 RPM
Hydraulic Work	74.32 kW
Shaft Work	76.62 kW
Mechanical Efficiency	97 %
Motor Efficiency	95.09 %
Electricity Consumption	80.57 kWe
Mass Flow	772.6 kg/s
Pump Head	0.8149 bar/8.31 m H2O

Table 5.16 Condenser Pump Physical Dimensions

Item	Dimension
Baseplate Length	1.935 m
Baseplate Width	0.6861 m
Pump Weight	570.3 kg
Motor Weight	430.9 kg

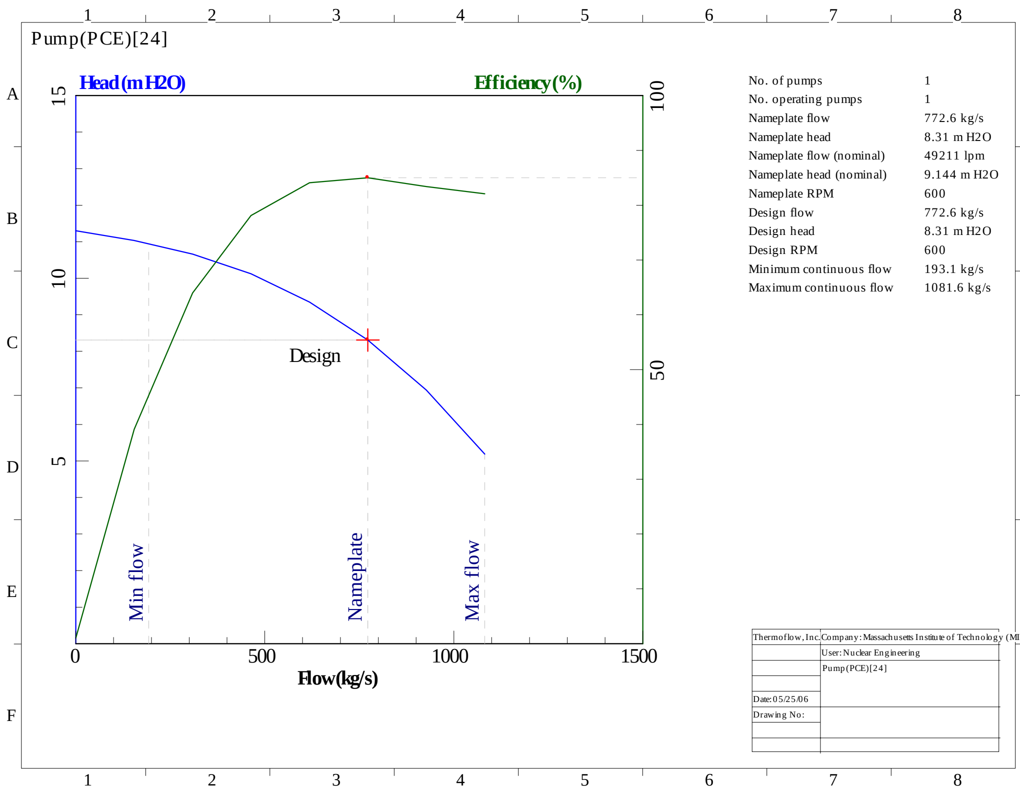

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Figure 5.14 Condenser Pump Curve.

Moisture separator

The moisture separator receives wet steam from the HPT exhaust and dries it to about 99.5% quality before it is expanded in the LPT. The separated water is throttled and discharged into the condenser. PEACE does not provide engineering estimates for the moisture separator design and size, so we have made use of information and models available in the literature. The moisture separator consists of a bank of chevron-type vertical vanes through which the steam flows. The shape of the vanes causes the steam to flow in a zig-zag path. As a result the droplets carried by the steam deposit on the chevron vanes by inertia. The liquid is drained by gravity through a drain pan at the bottom of the separator. A typical chevron separator used in a Boiling Water Reactor (BWR) steam dryer is shown in Figure 5.15 and the cross-section of a steam passage between two chevron vanes is shown in Figure 5.16. Typical dimensions for the chevron vanes are reported in Table 5.17. This basic design should work also here.

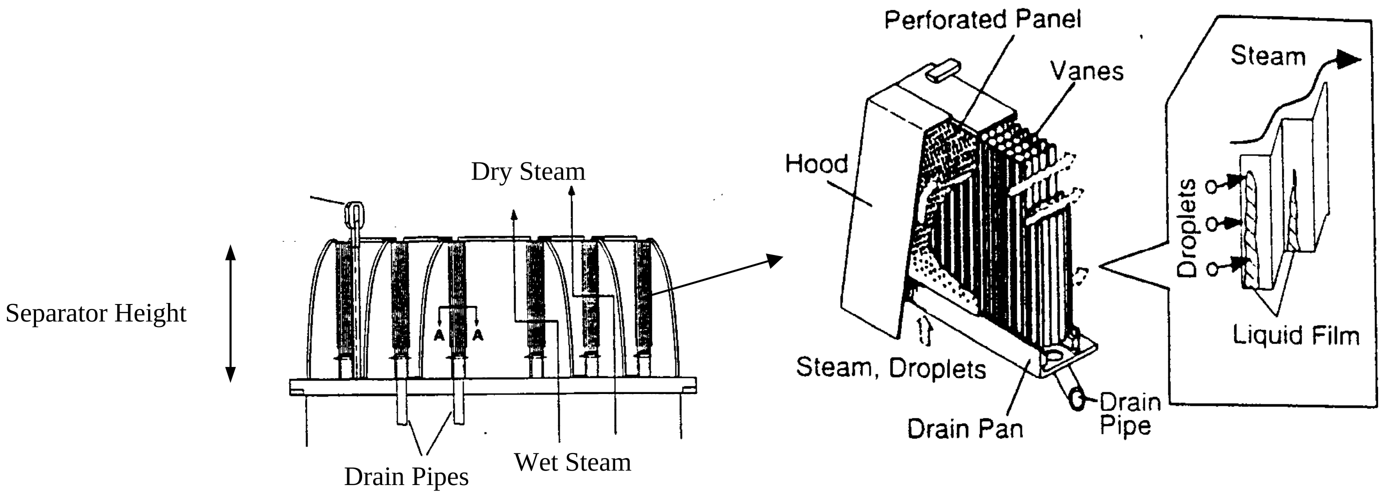

Figure 5.15 Steam separator of the chevron type.

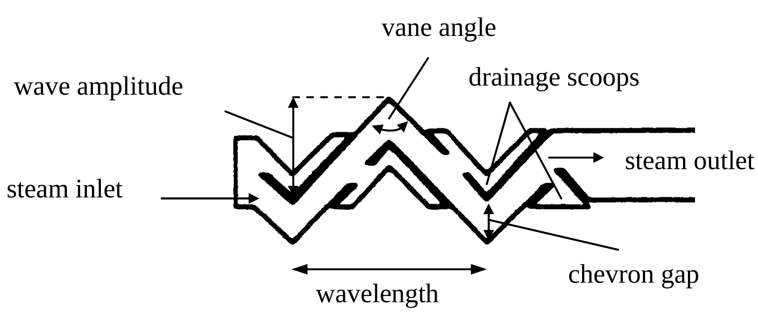

Figure 5.16 Cross-section of a chevron vane.

The separation efficiency of a chevron separator increases with the steam velocity through it. However, there exists a critical velocity above which large re-entrainment of the separated liquid occurs due to high shear stresses at the liquid/steam interface. This velocity must not be exceeded. An estimate of the critical velocity, U_{cr}, can be obtained from the Wilson's correlation for chevron separators, graphically shown in Figure 5.17, and recommended by Moore and Sieverding in their book on steam turbines and steam separation[4]. The critical velocity is a

[4] M. J. Moore, C. H. Sieverding, Two-Phase Steam Flow in Turbines and Separators, McGraw-Hill Inc. 1976.

function of the operating conditions via the fluid properties. For the conditions of interest to our moisture separator (10.4 bar) the Wilson's correlation yields a Kutateladze number, Ku, of about 1.6. The Kutateldaze number is defined as follows:

$$Ku = \frac{\rho_g^{1/2} U_{cr}}{\left[g\sigma\left(\rho_p - \rho_g\right)\right]^{1/4}} \tag{5.2}$$

where ρ_g, ρ_f, σ and g are the vapor density, liquid density, surface tension and acceleration of gravity, respectively. From Eq. (5.2) one gets a critical velocity of about 3 m/s. We conservatively assume a 2.5 m/s operating velocity and size the moisture separator accordingly. That is, the steam flow through the moisture separator is 15.9 kg/s (Point 6 in Table 5.1) and its density is 5.3 kg/m^3. Thus, the required flow area is 15.9/(5.3x2.5)=1.2 m^2. Since the height of the chevron vanes is 1.2 m, the required width of the chevron vane bank is 1.2/1.2=1 m, and the required number of chevron vanes is 1/0.01=100. Accounting for space for the drain pan, the perforated panel and the hood, we approximate the moisture separator as a box of 1.5 m height, 1.5 m width and 1 m length.

Table 5.17 Chevron vane parameters

Parameter	Value
Chevron height	1.2 m
Chevron gap	10 mm
Chevron wavelength	50 mm
Chevron wave amplitude	25 mm
Vane angle	90°

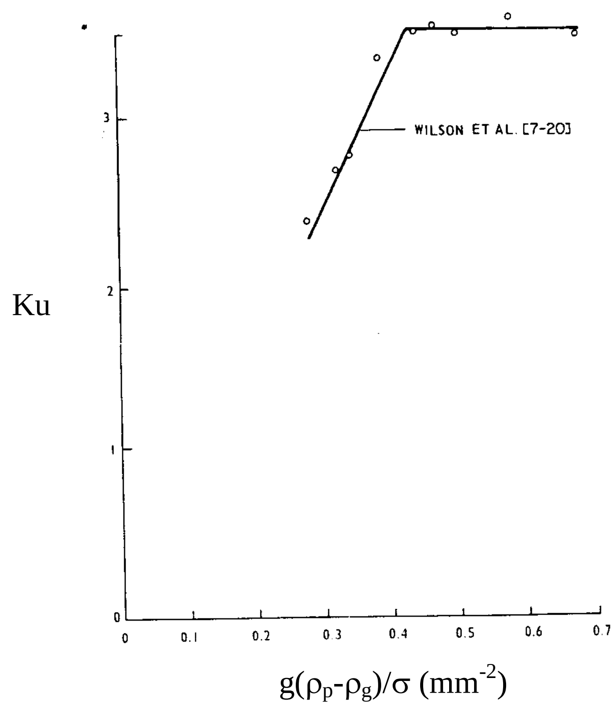

Figure 5.17 Chevron separator critical velocity

Piping

The length, number of elbows and elevations of the pipes are consistent with the layout shown in Figures 5.18-5.20. All pipes are made of carbon steel, grade A-106, except the steam line (Pipe A) is made of P22, a 2Cr-1Mo alloy steel, and the feedwater line (Pipe K) is made of stainless steel TP-304, because of their high operating temperatures and pressures. The number of additional head losses was set equal to 1.5 for each pipe, to represent the entrance and exit form losses. The pipe diameters were calculated by PEACE and verified with hand calculations based on the usual friction factor correlations.

Table 5.18 Size and operating conditions of the pipes in the system.

Pipe*	P (bar)	T (°C)	$\dot{m}$ (kg/s)	H_{out}-H_{in} (m)**	Length (m)	Number of elbows	Additional head losses	ID (mm)	Thickness (mm)	Exit velocity (m/s)
A	252.5	460	31.27	2.1	6.1	2	1.5	101.6	20.48	36.88
B	212.8	435.4	15.21	-2.5	3.4	1	1.5	69.85	12	44.02
C	10.85	183.5	15.83	0	1.8	1	1.5	202.7	8.179	92.91
D	10.39	181.6	14.34	0	1.8	1	1.5	202.7	8.179	87.25
E	10.39	181.6	1.498	-0.2	1.1	1	1.5	38.1	5.08	1.484
F	0.763	92.25	15.21	0	0.1	1	1.5	381	12.7	238.6
G	210.3	114.9	15.21	2.8	3.7	2	1.5	95.25	15.23	2.231
H	0.757	92.01	16.71	0	1.1	0	1.5	429	14.3	259.4
I	0.75	91.79	31.32	-2.2	6.2	2	1.5	154.1	7.112	1.743
J	258	94.91	31.27	1.66	2.4	1	1.5	133.4	25.31	2.3
K	257.6	349.9	31.27	0.36	2.2	1	1.5	165.1	42.74	2.325
L	12.3	29	773	0	6	0	1.5	390.6	7.925	6.471
M	11.8	39	773	0	6	0	1.5	390.6	7.925	6.494

* Pipe labels are shown in Figure 5.1

** Relative elevation of pipe outlet and inlet

Plant Layout

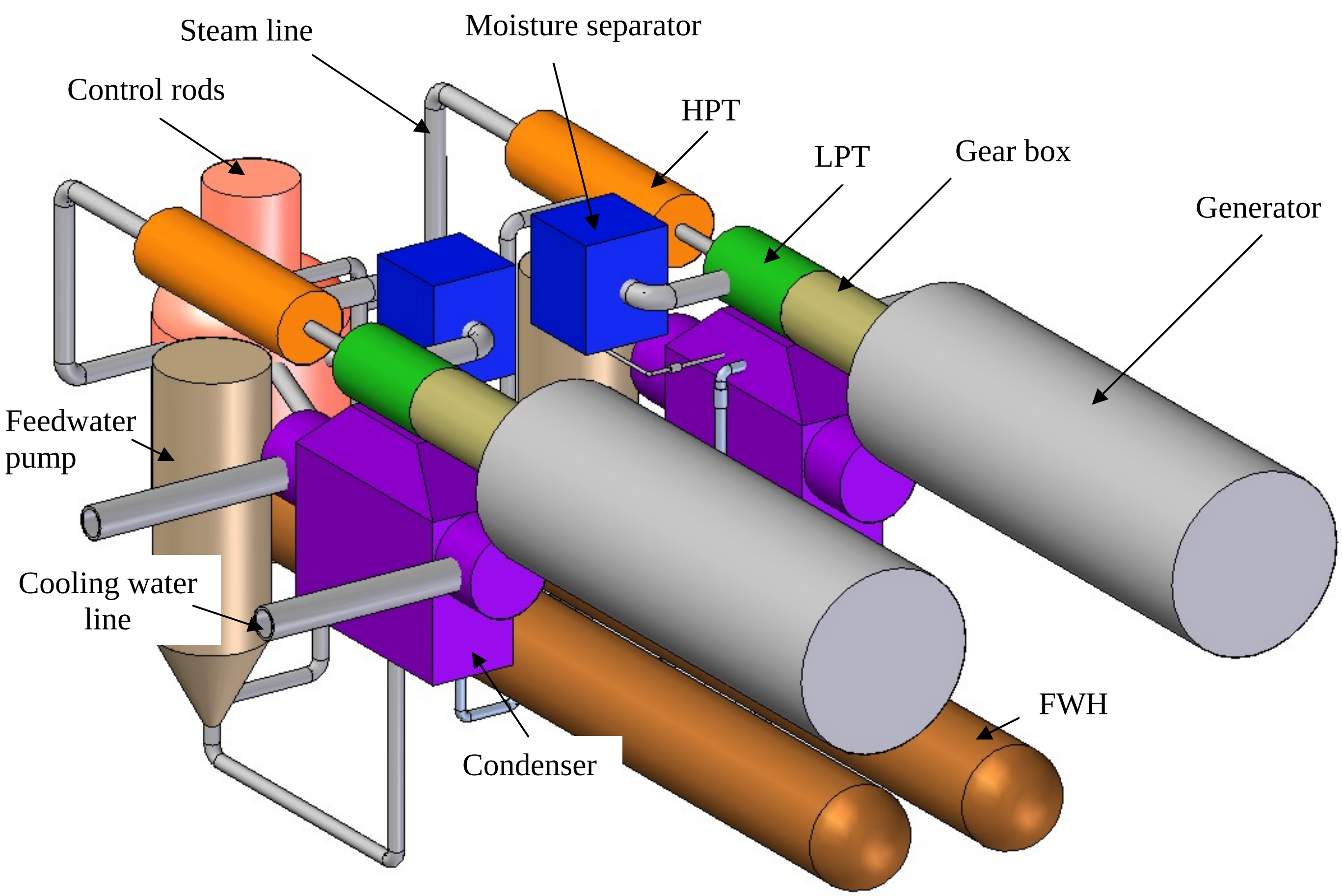

Figure 5.18 Isometric view of the plant.

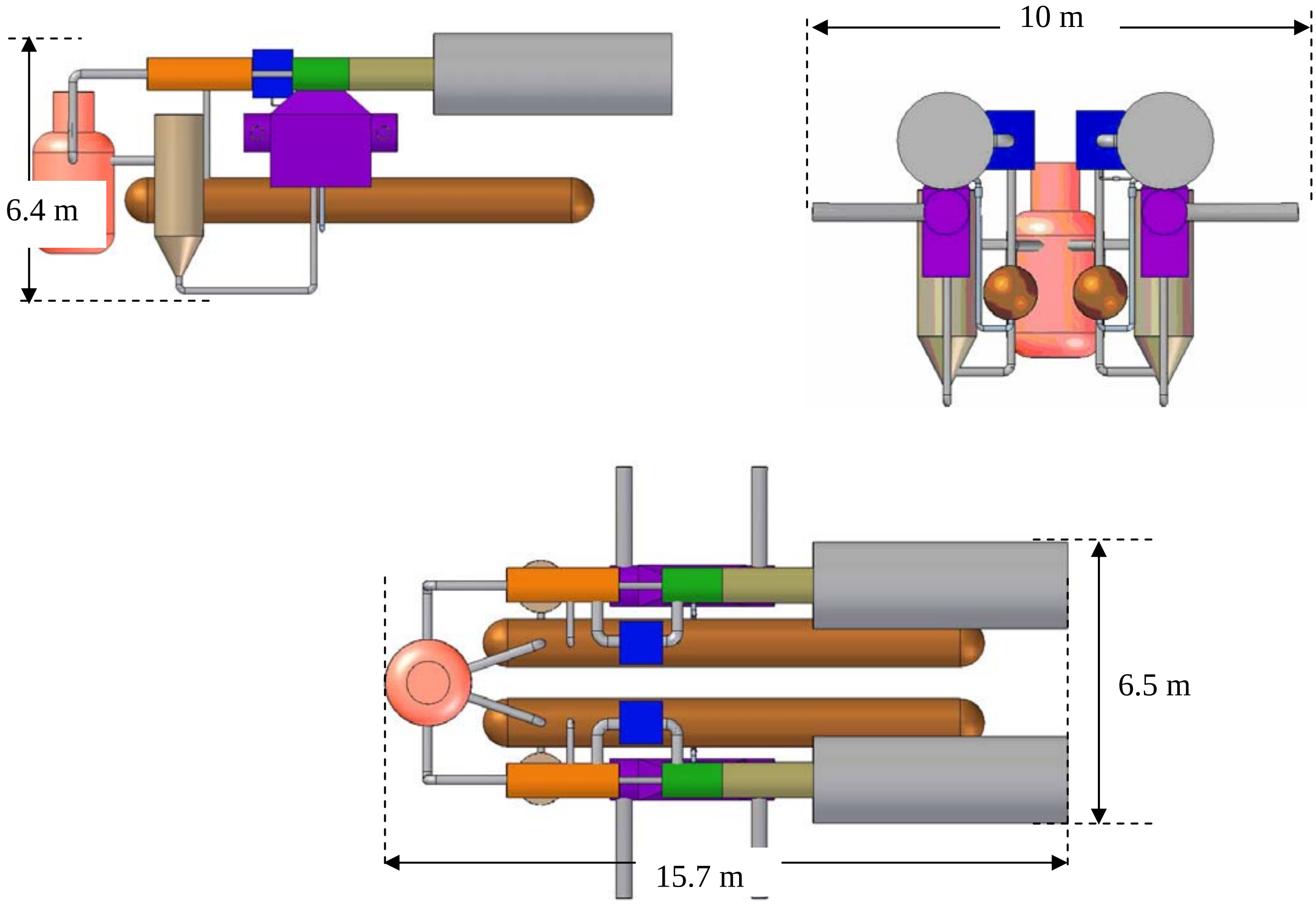

Figure 5.19 Orthogonal views of the plant.

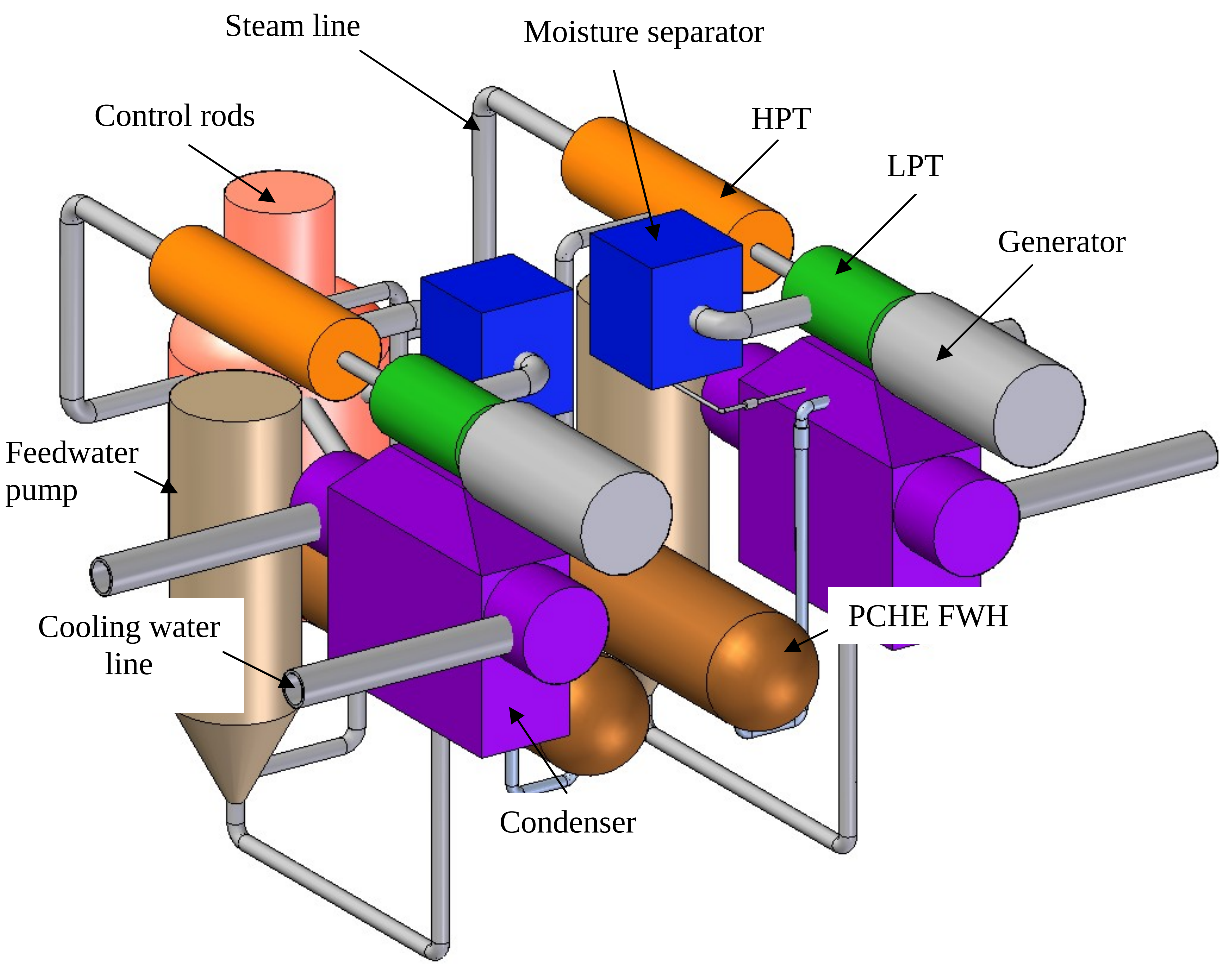

Figure 5.20 Isometric view of the plant with full speed generator, PCHE FWH and without gear box.

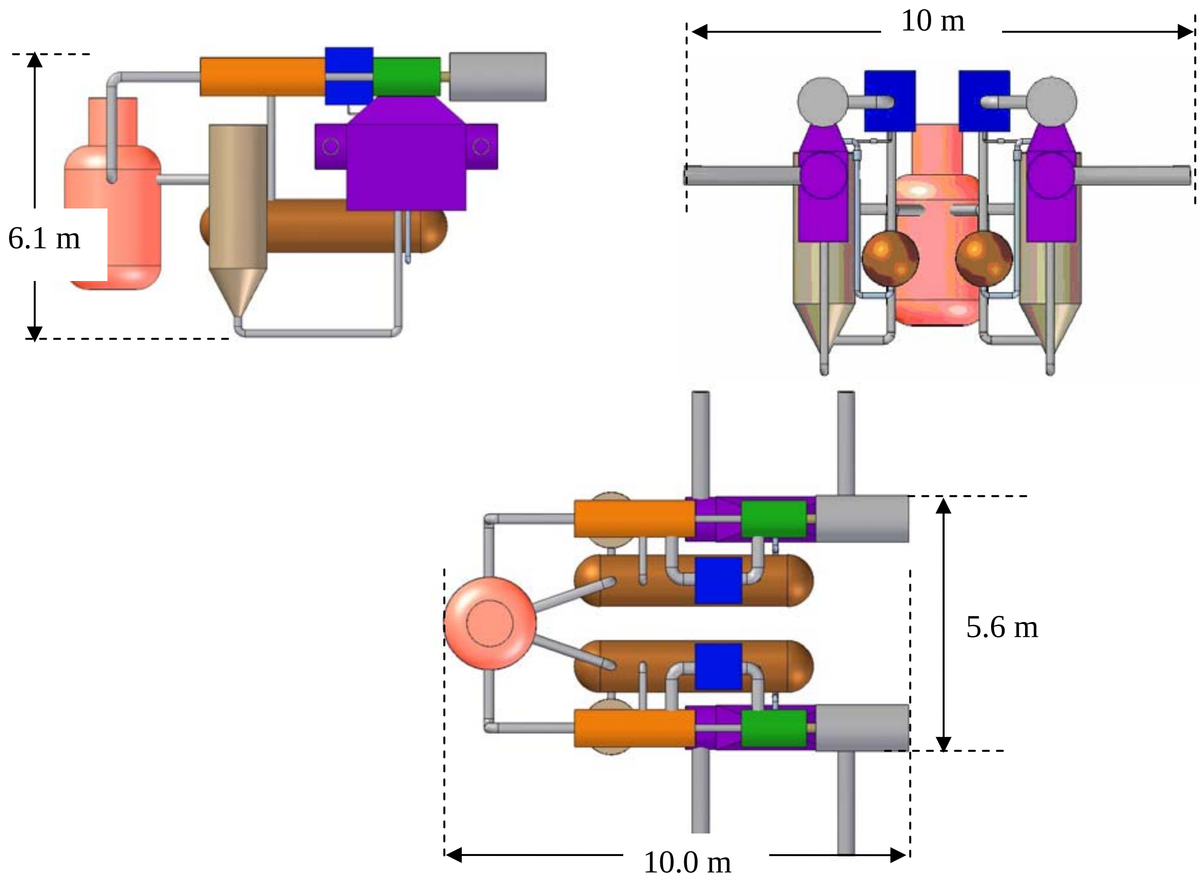

Figure 5.21 Orthogonal views of the plant with full speed generator, PCHE FWH and without gear box.

6. Analysis of the Power Cycle Performance at Off-Nominal Conditions

Once the PCS design is established, it is important to evaluate the sensitivity of the plant performance to operation at off-nominal conditions. This can be readily done with Thermoflex, which lets the user fix a plant design (such as the one presented in Section 5) and evaluate its behavior when certain parameters are changed. Using this feature of Thermoflex, we have investigated the plant performance when thermal power and seawater temperature are changed. In all cases it is assumed that the control system maintains the reactor pressure and outlet temperature at their nominal values (~250 bar and 460°C, respectively). The feedwater flow rate is adjusted proportionally to the thermal power. Therefore, the reactor inlet temperature remains constant at its nominal value of 350°C. The seawater mass flow rate is also kept constant at its nominal value of ~773 kg/s. All other parameters are allowed to change. The steam flow to the turbine is regulated with a variable-area nozzle, which in Thermoflex mimics the effect of the multi-valve admission system.

In the first study the reactor thermal power is varied between 25% and 150% of its nominal value, while the seawater temperature is kept constant at 29°C (Table 6.1). The feedwater pump electricity consumption decreases with decreasing thermal power, because the feedwater flow is proportional to thermal power. The condenser pressure drops quickly at lower than nominal thermal power, because the condenser provides more cooling than is needed. The highest cycle efficiency is achieved at the nominal conditions. This is expected since the turbine and pump efficiencies were optimized for those conditions. The cycle efficiency however remains acceptable (>20%) over a broad range of power levels (50-150%), while dropping steeply at low power. In light of this finding, the plant design should be re-optimized at a lower power level, if normal operation is expected mostly in that region.

In the second study the seawater temperature is varied from 0.01 to 39°C, while the thermal power is kept constant at 43 MW (Table 6.2). The effect of the seawater temperature change is to change the condenser pressure, however the effect on all other plant parameters is almost imperceptible.

Table 6.1 PCS parameters at different thermal power levels and fixed seawater temperature (29°C).

Thermal power (MWt-%)	Reactor T_{in}/T_{out} (°C)	Reactor P_{out} (bar)	Cycle Efficiency (%) / Net Electric Output (MWe)	HPT Exit Quality (%)	LPT Exit Quality (%)	HPT Casing Efficiency (%)	LPT Casing Efficiency (%)	Feedwater Pump Power (kWe)	Condenser Pressure (bar)
10.75 - 25%	349.9/460.0	252.47	17.53 / 1.884	93.90	91.70	50.78	71.20	574.70	0.09
21.50 - 50%	349.9/460.0	250.18	21.72 / 4.670	92.10	90.80	58.27	72.03	770.80	0.20
32.25 - 75%	349.9/460.0	250.64	23.24 / 7.495	90.90	90.10	64.25	75.81	955.90	0.40
43.00 - 100%	349.9/460.0	251.40	24.13 / 10.376	89.90	90.00	70.13	78.08	1170.80	0.75
53.75 - 125%	349.9/460.0	253.84	22.73 / 12.216	89.30	91.00	74.39	76.92	1425.40	1.40
64.50 - 150%	349.9/460.0	255.51	21.72 / 14.012	88.70	92.50	79.54	71.49	1721.50	2.61

Table 6.2 PCS parameters at different seawater temperature and fixed thermal power (43 MWt).

Seawater Temperature (°C)	Reactor T_{in}/T_{out} (°C)	Reactor P_{out} (bar)	Cycle Efficiency (%) / Net Electric Output (MWe)	HPT Exit Quality (%)	LPT Exit Quality (%)	HPT Casing Efficiency (%)	LPT Casing Efficiency (%)	Feedwater Pump Power (kWe)	Condenser Pressure (bar)
0.01	349.9/460.0	251.40	23.62 / 10.157	90.10	90.10	68.78	70.69	1159.60	0.37
7	349.9/460.0	251.40	23.62 / 10.155	90.10	89.90	68.87	72.93	1162.00	0.43
15	349.9/460.0	251.40	23.88 / 10.267	90.00	89.80	69.20	75.56	1165.10	0.53
23	349.9/460.0	251.40	23.66 / 10.174	90.00	89.90	69.29	77.48	1168.30	0.65
29	349.9/460.0	251.40	24.13 / 10.376	89.90	90.00	70.13	78.08	1170.80	0.75
31	349.9/460.0	251.40	23.53 / 10.120	90.00	90.20	69.70	78.26	1172.30	0.82
39	349.9/460.0	251.40	22.90 / 9.846	90.00	90.90	69.91	77.65	1176.40	1.03

7. Conclusions and Research Needs

Scoping studies for a compact power conversion system (PCS) based on a direct-cycle supercritical water cooled reactor have been conducted. The thermodynamic analysis suggests that a supercritical Rankine cycle with feedwater preheating and moisture separation offers the best combination of relatively high cycle efficiency, acceptable pumping power and flexibility in choosing reactor inlet and outlet temperatures. The selected operating conditions include a reactor pressure of 250 bar and reactor inlet/outlet temperatures of 350/460°C. The preliminary engineering design of such system has confirmed the compactness of the plant and the basic feasibility of the major components, i.e., turbine-generator, feedwater pump, feedwater heater, condenser and pipes. An analysis of the plant performance at off-nominal conditions also has revealed good robustness of the design in handling large changes of thermal power and seawater temperature.

The following items are recommended for future work, should the supercritical PCS option be further investigated:

- Conceptual design of a control system for the main PCS variables, e.g., reactor power, reactor inlet and outlet temperatures, reactor pressure, feedwater flow, etc. Such control system should be analyzed with respect to its capability to "handle" key operational transients. In particular a rapid power ramp up could prove problematic in a direct cycle. In the traditional indirect-cycle PCS a rapid load increase is satisfied by simply drawing more steam from the steam generator and "slowly" adjusting the feedwater flow and the reactor power to re-establish the correct level in the steam generator. This approach could be troublesome for a supercritical direct-cycle PCS, because there is not much water inventory in the reactor vessel and no level to adjust. Therefore, to accommodate the load increase, the feedwater pumps need to increase the flow quickly and the reactor needs to match that flow also quickly. A mitigating factor could be the presence of a large feedwater inventory in the feedwater heaters, from which the coolant would be drawn without the need for a very quick response from the feedwater pumps. This also needs to be quantified accurately.

- Investigation of printed circuit heat exchangers (PCHE). To reduce the physical size of the PCS further, it is recommended to explore a PCHE design for the feedwater heaters. However, basic thermal and hydraulic data for PCHEs at the conditions of interest (i.e., high-pressure high-temperature two-phase flow) are lacking. If PCHEs are to be used and optimized, these data must be obtained. Also, inspectability and maintenance of the narrow PCHE channels should be studied carefully.

- Development of a sound core design. A pre-requisite for this task is the development and verification of a reliable subchannel analysis code, to evaluate the sensitivity of the core parameters to deviations from nominal conditions. This is a key feasibility issue for supercritical water cooled reactors.

- Development of suitable materials for in-core service and coolant chemistry control strategy. As is well known, general corrosion and especially stress-corrosion cracking are major feasibility issues for the supercritical water cooled reactor. Materials used in supercritical fossil plants cannot be directly applied to a nuclear reactor because of the more aggressive environment present in the core and the more stringent corrosion limitations imposed in nuclear systems. The PCS design developed in this study should help mitigate the materials challenge somewhat, as the reactor outlet temperature has been reduced to 460°C, however the challenge remains formidable. Development of advanced alloys as well as basic radiolysis data along with the definition of a new coolant quality control strategy are probably needed to meet this challenge.

- Design of passive safety systems. Development of compact and reliable passive safety systems for a direct-cycle supercritical water cooled reactor presents various challenges including the absence of a large initial in-vessel coolant inventory to rely on, the high system pressure and the lack or incompleteness of basic data on critical flow and heat transfer at supercritical conditions.

- Improvement of the Thermoflex/PEACE software. Some improvements that would make the engineering analysis more effective are suggested for consideration by Thermoflow Inc. First, allow >3600 rpm generator speeds so that the need for a gear box is eliminated. Second, provide estimates for the length of the turbine blades of all stages, not just the last stage. Third, allow low-temperature supercritical-pressure steam at the turbine inlet, e.g., PEACE currently does not accept 400°C 25 MPa steam. Fourth, allow >3600 rpm for the speed of electric-motor-driven pumps, so that the pump size can be reduced. Fifth, add capabilities for engineering estimates of moisture separators and throttle valves. Sixth, add capabilities for automatic drawing of the T-s (or h-s) diagram for the whole cycle, not just the turbine expansion path.

Acknowledgements

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