THERMODYNAMIC ANALYSIS OF A SUPERCRITICAL WATER REACTOR

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Abstract

A thermodynamic model has been developed for a hypothetical design of a Supercritical Water Reactor, with emphasis on Canadian design criteria. The model solves for cycle efficiency, mass flows and physical conditions throughout the plant based on input parameters of operating pressures and efficiencies of components. The model includes eight feedwater heaters, three feedwater pumps, a deaerator, a condenser, the core, three turbines and two reheaters. To perform the calculations, Microsoft Excel was used in conjunction with FLUIDCAL-IAPWS95 and VBA code. The calculations show that a thermal efficiency of 47.5% can be achieved with a core outlet temperature of 625°C.

1. Introduction

The Generation IV International Forum (GIF) has set four goals for Gen IV nuclear energy systems: namely, that fuel cycles be *sustainable*, *safety* and *security* be paramount and *economic advantages* clear [1]. Six nuclear energy systems were selected by GIF for further research and development[1], of which the Supercritical-Water-Cooled Reactor (SCWR) System is one. Each one has its own advantages, but one of the most highly ranked economically is the SCWR [1], for which the CANDU pressure-tube design is particularly well suited.

The CANDU-SCWR will undoubtedly pose many design challenges for Canadian engineers and scientists. With anticipated core exit temperatures around 625°C and pressures around 25 MPa, the limits of material strength and corrosion resistance will be tested. As one particular example, the corrosion rate of Zircaloy-4 at core temperatures is prohibitive [3]. This will require untraditional fuel sheathing with a higher neutron absorption cross-section. While this will alter core neutronics, the sheathing itself will experience neutron damage in addition to the low strength and fast corrosion kinetics exhibited at these high temperatures.

Nonetheless, the CANDU-SCWR has many design benefits. The high pressures are easily accommodated in the CANDU-style pressure-tubes, unlike the excessive thickness of the PWR-SCWR pressure-vessel. The direct cycle will eliminate steam generators and thereby simplify plant design. The high thermodynamic efficiency of the supercritical Rankine cycle reduces coolant and cooling water flow rates, core size and fuel

[1] The other five nuclear energy systems selected by GIF are (in alphabetical order): the Gas-Cooled Fast Reactor System (GFR), the Lead-Cooled Fast Reactor System (LFR), the Molten Salt Reactor System (MSR), the Sodium-Cooled Fast Reactor System(SFR) and the Very High Temperature Reactor System (VHTR) [1].

consumption. The use of enriched fuel will increase burnup, reducing waste. The use of high temperatures will allow for flexible hydrogen co-generation, granting CANDU-SCWR systems the ability to meet peak energy demands, not just base load [2]. The CANDU-SCWR will also benefit from Canada's expertise in water reactor safety and design.

The design of a nuclear reactor is an engineering tug-of-war, pulling toward efficiency, restricted by what is reasonable. One anchor is a thermodynamic model based on reasonable parameters. The other is an economic model based on a preliminary design. Together, they give insight into which components are essential and which can be modified or removed. This paper describes the development of one anchor, the thermodynamic model for a SCWR.

2. Methodology

The thermodynamic model was based upon a specific plant model, or proposed layout of components. The plant model was designed as a hybrid of a supercritical fossil power plant and a CANDU (see section 2.1). For each component in the plant model, parameters were chosen to assign each a reasonable duty. What is considered reasonable here should not be misconstrued as feasible or viable. Some of the heat exchangers may be prohibitively expensive and some of the turbines may have fatally large extraction lines. However, most of the parameters were chosen to resemble a supercritical fossil power plant [4] to give credibility to the numbers. In order to show some trends, unreasonable parameters were evaluated. Thought was nonetheless given to limiting the number of feedwater heaters, restricting the pressure drops across turbines, and bracketing the moisture content of the low pressure turbine exhaust. Mass and energy balances were performed on each component using accurate thermodynamic data from FLUIDCAL. The entire model was solved using iterative procedures built into Excel's Solver.

2.1. Plant Model

The plant model was a hybrid of a supercritical fossil power plant and a CANDU nuclear plant. It resembled a supercritical fossil plant in its feedwater heater train, which included three pumps and a deaerator. It resembled a CANDU plant in that it had a reheater section fed by steam. As with most energy systems, it included a high pressure, an intermediate pressure, and a low pressure turbine, as well as turbine extraction streams and a condenser. A schematic of the plant model is shown below in Figure 1.

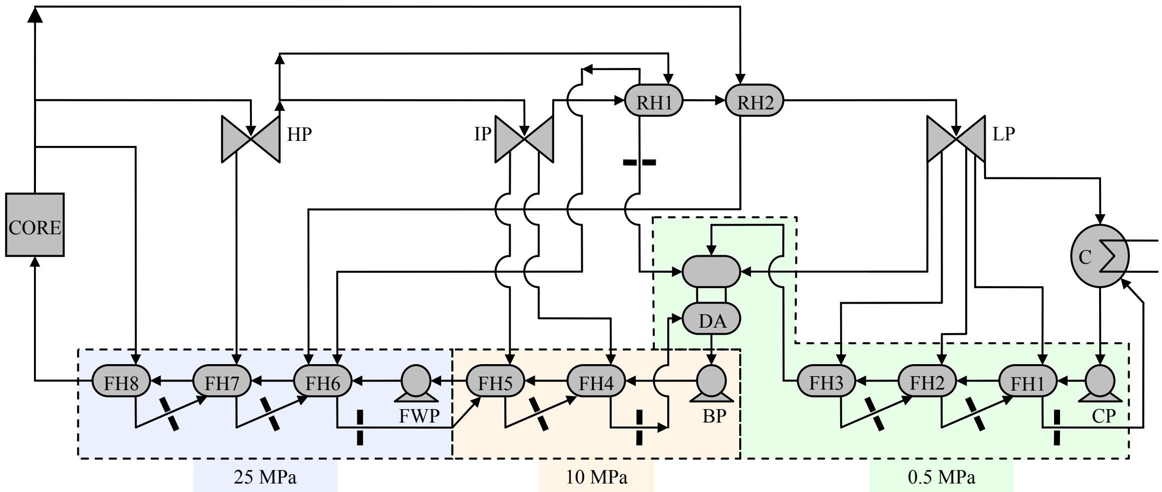

Figure 1. Plant model for SCWR. Abbreviations: HP = high pressure turbine, IP = intermediate pressure turbine, LP = low pressure turbine, RH = reheater, C = condenser, CP = condensate pump, BP = booster pump, FWP = feedwater pump, FH = feedwater heater, DA = deaerator

2.2. Component Parameters

Components in an energy system may be broken down into two broad categories: (1) turbines and pumps and (2) heat exchangers. The parameters required for turbines and pumps are inlet pressure, outlet pressure and efficiency. These parameters remained largely unchanged for each scenario analysed and are listed as such in Table 1 below. Conversely, the parameters required for heat exchangers are mixed and depend on the requirements of the heat exchanger. Furthermore, a range of parameter values was used in many instances. Parameters for heat exchangers include pressures of hot (extraction) streams, condenser temperature approach, drains cooler temperature approach, and cold stream exit temperature. The parameters for heat exchangers are listed in Table 2 below.

Table 1. Parameters for turbines and pumps.

Component	**Inlet pressure**	**Outlet Pressure**	**Efficiency**[a]
HP Turbine	25 MPa	5 MPa	85%
IP Turbine	5 MPa	1 MPa	85%
LP Turbine	1 MPa	0.005 MPa	85%
Condensate Pump	0.005 MPa	0.5 MPa	70%
Booster Pump	0.5 MPa	10 MPa	70%
Feedwater Pump	10 MPa	25 MPa	70%

[a] Conservatively adapted from [4].

Table 2. Parameters for heat exchangers.

	Hot Stream		Temp. Approach (°C)		Cold Stream Temp. (°C)	
Component	**P (MPa)**	***a* (%)**[a]	**ΔT_C**[b]	**ΔT_{DC}**[b]	**T_{in}**	**T_{out}**
Reactor Core						475-625
Reheater 1	5		1	5		
Reheater 2	10-25[c]		1	5		300-450
Condenser	0.005				10	23
Feedwater Heater 1	0.025		1	5		
Feedwater Heater 2	0.09		1	5		
Feedwater Heater 3	0.2		1	5		
Deaerator	0.5					151.83
Feedwater Heater 4	1		1	5		
Feedwater Heater 5	2		1	5		
Feedwater Heater 6	5		1	5		
Feedwater Heater 7	7-11		1	5		
Feedwater Heater 8	25	0-20				

[a] *a* represents that stream's share of the overall system flow; e.g., (1000 kg/s)(20%) = 200 kg/s.
[b] C = condenser section; DC = drains cooler section
[c] Reheater 2 operated in one of two modes: supercritical or superheated steam. If supercritical water was used as the heating medium, the pressure was 25 MPa, with ΔT_{DC} = 5°C and a set reheat temperature. If superheated steam was used, the pressure varied from 10 MPa to 15 MPa, with ΔT_C = 1°C and again ΔT_{DC} = 5°C.

2.3. Mass and energy balances

Mass and energy balances for turbines and pumps are simple compared to heat exchangers, which may require sectional division (desuperheater, condenser, drains cooler) and an iterative solution. The mass and energy balances are presented below, grouped according to the type of balance needed. They are followed by an overall balance and a definition of the thermodynamic cycle efficiency.

2.3.1. Mass and energy balances for turbines

The efficiency of a turbine is the ratio of the actual specific enthalpy change to the isentropic specific enthalpy change. Assuming the same efficiency applies to each stage of turbine blades, we write:

$$h_a \text{ or } h_{out} = h_{in} - \eta\left(h_{in} - h_{\hat{s}}\right) \tag{1}$$

where h_a is the specific enthalpy of an extraction stream and h_{out} is the specific enthalpy of the exhaust stream. The power produced by a turbine is therefore

$$\dot{W}_T = \sum m_a \left(h_{in} - h_a\right) + \left(M - \sum m_a\right)\left(h_{in} - h_{out}\right) \tag{2}$$

where m_a is the mass flow rate of an extraction stream and M is the mass flow rate of steam entering the turbine.

2.3.2. Mass and energy balances for pumps

The efficiency of a pump is the ratio of fluid work to electrical work. The electrical consumption of a pump is therefore:

$$\dot{W}_p = \frac{M}{\eta}\left(\frac{\Delta P}{\rho_{avg}}\right) \tag{3}$$

where M is the mass flow rate entering the pump, ΔP is the increase in pressure across the pump, and ρ_{avg} is the average density of the fluid.

In a well insulated pump, most of the energy not used in pressurizing the fluid is converted to heat energy within the fluid. Thus,

$$h_{out} \approx h_{in} + \frac{\dot{W}_p}{M} \tag{4}$$

2.3.3. Mass and energy balances for drains cooler sections of heat exchangers

The drains cooler acts as the entrance for the cold stream and an exit for the hot stream. The condensed fluid passes into the drains cooler and becomes subcooled before leaving the heat exchanger. The temperature difference between the exiting hot stream and the entering cold stream is the drains cooler temperature approach, ΔT_{DC}, as shown in Figure 2. For a given ΔT_{DC}, the temperature and enthalpy of the exiting hot stream may be known. Therefore, the rate of heat transfer within a drains cooler section is:

$$\dot{Q}_{DC} = M_{hot}\left(h_{hot,in} - h_{hot,out}\right) = M_{cold}\left(h_{cold,out} - h_{cold,in}\right) \tag{5}$$

2.3.4. Mass and energy balances for condenser sections of heat exchangers

Within the condenser section of a heat exchanger, saturated steam condenses to saturated water. The saturated steam comes from an extraction stream, after passing through the desuperheater, or from an upstream drains cooler, after flashing through a throttle. The smallest temperature difference between the cold stream and the hot stream is the condenser temperature approach, ΔT_C, which occurs at the end of the condenser section as shown in Figure 2. For a given ΔT_C, the temperature and enthalpy of the exiting cold stream may be known. The heat transfer rate within a condenser section is:

$$\dot{Q}_C = M_{hot}\Delta h_{fg} + m_{UDC}x\Delta h_{fg} = M_{cold}\left(h_{cold,out} - h_{cold,in}\right) \quad (6)$$

where Δh_{fg} is the enthalpy of condensation, m_{UDC} is the mass flow from the upstream drains cooler, and x is the quality of that flow after flashing.

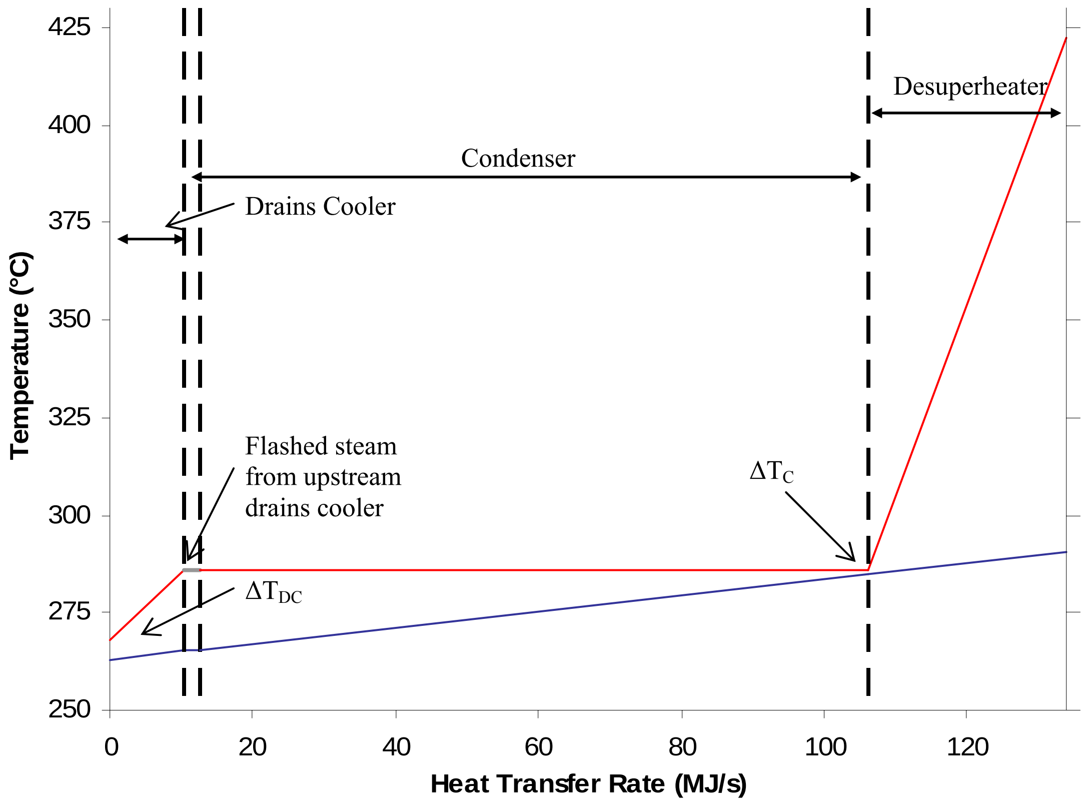

Figure 2. Typical T-Q diagram illustrating heat exchanger sections and temperature approaches. The hot stream is shown in red on top, and the cold stream is shown in blue on bottom.

2.3.5. Mass and energy balances for desuperheater sections of heat exchangers

The purpose of a desuperheater is to cool superheated steam within a confined space. In this way, the cold stream may be heated above the temperature of the condensing steam before it exits the heat exchanger. The desuperheater also acts to minimize the temperature difference between the hot and cold streams, thereby maximizing the effectiveness of available energy transfer. As shown in Figure 2, the hot stream exits the desuperheater at the saturation temperature, while the cold stream enters a few degrees below it. The mass and energy balance on a desuperheater section is given by an equation analogous to (5).

2.3.6. Mass and energy balance for the condenser

The main condenser operates like the condenser section of any heat exchanger, with the notable exception that the cold stream is cooling water. An average inlet temperature of

10°C was assumed for the cooling water, with an average outlet temperature of 23°C. Equation (6) applies, where the mass flow rate of cooling water is the unknown.

2.3.7. Mass and energy balance for the deaerator

The deaerator is a large direct contact heat exchanger, with many inlets and one outlet. It acts to remove dissolved air by providing a water-vapour interface. The exiting stream is saturated water at the pressure of the deaerator. It immediately passes to a booster pump situated below the deaerator. The column of water above the pump pressurizes the water, making it slightly subcooled. This is done to avoid cavitation.

The deaerator may be balanced a number of ways. Perhaps the easiest way is to vary the flow of the extraction line from the low pressure turbine until the enthalpy of the exiting stream is equal to the enthalpy of the saturated liquid at the given pressure. The specific enthalpy of the exiting stream may be calculated as follows:

$$h_{out} = \frac{\sum m_{in} h_{in}}{M_{out}} \quad (7)$$

where m_{in} is the mass flow rate of an entering stream with specific enthalpy h_{in}, and M_{out} is the mass flow rate leaving the deaerator.

2.3.8. Mass and energy balance for the core

The core was a flexible component in this thermodynamic model. Its duty was a function of the feedwater temperature and the desired core outlet temperature. Thus,

$$\dot{Q}_{Core} = M\left(h_{out} - h_{in}\right) \quad (8)$$

2.3.9. Overall mass and energy balance and definition of cycle efficiency

The net power output of the cycle is given by:

$$\dot{W}_{net} = \dot{W}_{HP} + \dot{W}_{IP} + \dot{W}_{LP} - \dot{W}_{Condensate} - \dot{W}_{Booster} - \dot{W}_{Feedwater} \quad (9)$$

where the first three terms are the power outputs of the turbines and the latter three are the power requirements of the pumps.

The thermodynamic cycle efficiency is defined as:

$$\eta_{net} = \frac{\dot{W}_{net}}{\dot{Q}_{core}} \quad (10)$$

3. Results and Discussion

The results that follow are for the plant model presented, using a base set of parameters and result from varying one parameter at a time. Each result was normalized to 1000 MW by adjusting system flow rate.

3.1. Feedwater heater 8 flow

For this scenario, the base case was as presented in Table 2, with the following specifications:

Reactor core outlet temperature:	625°C
Reheater 2 pressure:	25 MPa
Reheat temperature:	400°C
Feedwater heater 7 pressure:	7 MPa
Feedwater heater 8 flow:	0 %

The base case produced 1000 MW at a cycle efficiency of 46.9% using a system flow of 934 kg/s. Increasing the flow through feedwater heater 8 had a detrimental, albeit small effect on the cycle efficiency and system flow rate, as shown in Table 3.

Table 3. Effect of feedwater heater 8 operation using supercritical water on system flow and cycle efficiency.

Flow through FH8 (% of system flow)	Cycle Efficiency	System Flow Rate (kg/s)
0%	46.92%	934
1%	46.90%	943
2%	46.88%	953
5%	46.83%	983
10%	46.74%	1039

It can be seen that a supercritical feedwater heater in this position is not thermodynamically effective. If the feedwater entered at a higher temperature, the available energy transfer would be more effective, but may not alter the negative result. A lower-pressure feedwater heater in this place may be more favourable.

Nonetheless, the positioning of a supercritical feedwater heater at the end of the feedwater train is strategic. Transition through the critical point results in a dramatic drop in dielectric constant and presumably corrosion product solubility. A high temperature feedwater heater may be used to transition through the critical point and retain precipitating solids. At steady state, it may be possible to avoid crud accumulation on the fuel rods. To perform this task, a supercritical feed or high-pressure extraction line would be needed.

3.2. Reheat temperature

For this scenario, the same base case was used as above. The reheat temperature was varied from 300.10°C to 450°C. (Reheater 1 uses superheated steam at 5 MPa. When reheater 2 is shut off, a 300.10°C reheat temperature results.) The results are shown in Table 4 below. It can be seen that any flow through reheater 2 has a slight detrimental effect on cycle efficiency and system flow rate. This is due to the ineffectiveness caused by large temperature differences within the heat exchanger, as can be seen in Figure 3.

Table 4. Effect of reheat temperature on cycle efficiency, system flow, and LP turbine exhaust moisture content.

Reheat Temperature (°C)	Cycle Efficiency	Moisture Content of LP Turbine Exhaust	System Flow Rate (kg/s)
300.10	47.30%	10.6%	926
325.00	47.16%	9.3%	929
350.00	47.05%	8.0%	931
375.00	46.97%	6.8%	933
400.00	46.92%	5.6%	934
425.00	46.88%	4.4%	934
450.00	46.86%	3.3%	935

Table 4 also shows that the LP turbine exhaust increases beyond 10% moisture when reheat temperature is below 325°C. It is generally desirable to maintain the LP turbine exhaust below 10% moisture to avoid excessive erosion of the turbine blades. Therefore, despite its effect on cycle efficiency, reheater 2 must be used. However, it should operate at a lower pressure, using an extraction stream from the high pressure turbine.

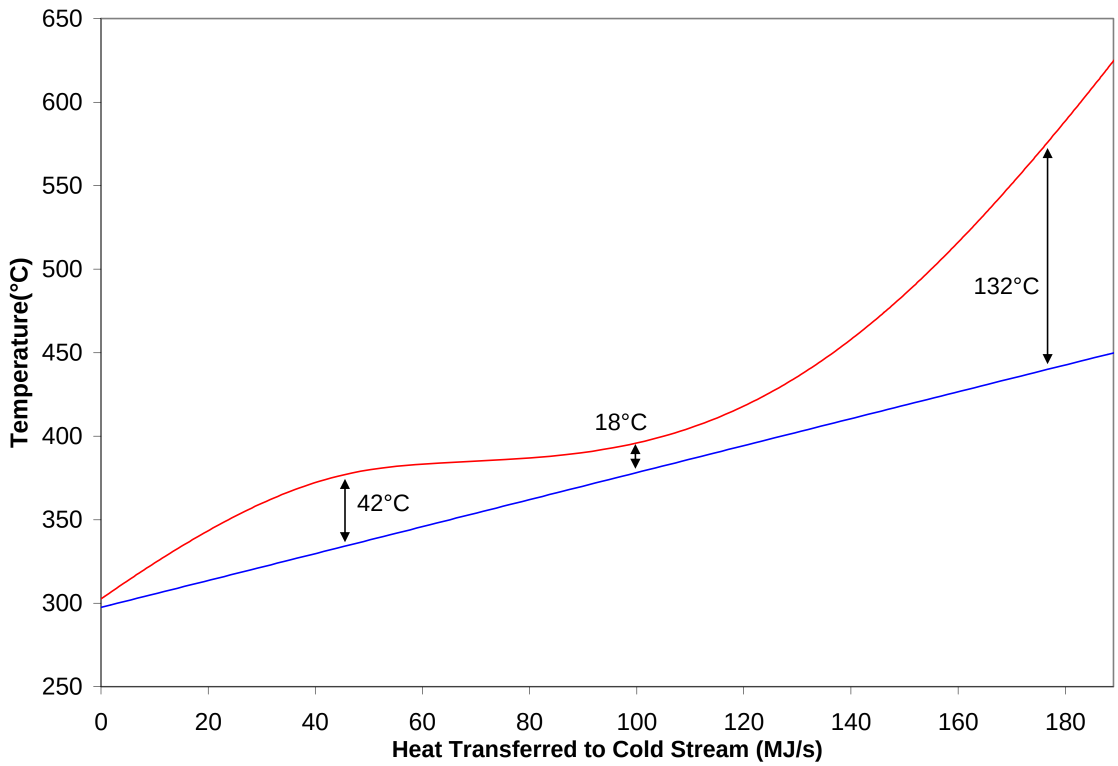

Figure 3 Temperature differences within reheater 2 using supercritical water as a heating fluid. Reheat temperature is 450°C.

3.3. Reheat pressure

For this scenario, the base case was as presented in Table 2, with the following specifications:

Reactor core outlet temperature:	625°C
Reheater 2 pressure:	10 MPa
Reheater 2 ΔT_C:	1°C
Reheater 2 ΔT_{DC}:	5°C
Feedwater heater 7 pressure:	7 MPa
Feedwater heater 8 flow:	0 %

The base case produced 1000 MW at a cycle efficiency of 47.2% using a system flow of 929 kg/s. At this pressure, the reheat temperature is about 315°C, regardless of core outlet temperature. Increasing the pressure of the extraction line used by reheater 2 reduced turbine exhaust moisture and cycle efficiency, as shown in Table 5 below. Note that the base case of 10 MPa kept the LP turbine exhaust at 9.8% moisture.

Table 5. Effect of reheater 2 pressure on LP turbine exhaust and cycle efficiency.

Reheater 2 Pressure (MPa)	Cycle Efficiency	Moisture Content of LP Turbine Exhaust	System Flow Rate (kg/s)
10	47.2%	9.8%	929
11	47.1%	9.3%	931
12	47.0%	8.7%	932
15	46.8%	7.1%	935

3.4. Core outlet temperature

The major factor affecting thermodynamic efficiency and system flow was found to be the core outlet temperature. To show this, two base cases were used: the first used a pressure of 7 MPa in feedwater heater 7 and the second used 10 MPa. All other parameters were the same as the base case used in the above section. The 7 MPa base case produced 1000 MW at a cycle efficiency of 47.2% using a system flow of 929 kg/s, as quoted above. The 10 MPa base case produced 1000 MW at a cycle efficiency of 47.5% using a system flow of 1006 kg/s. The core outlet temperature was decreased from 625°C to 475°C. The results are presented in Figure 4 below. Note that while the 10 MPa scenario results in improved plant efficiency over the 7 MPa scenario, it also results in an increased system flow rate—an indication that there is a trade-off between cycle efficiency and plant size.

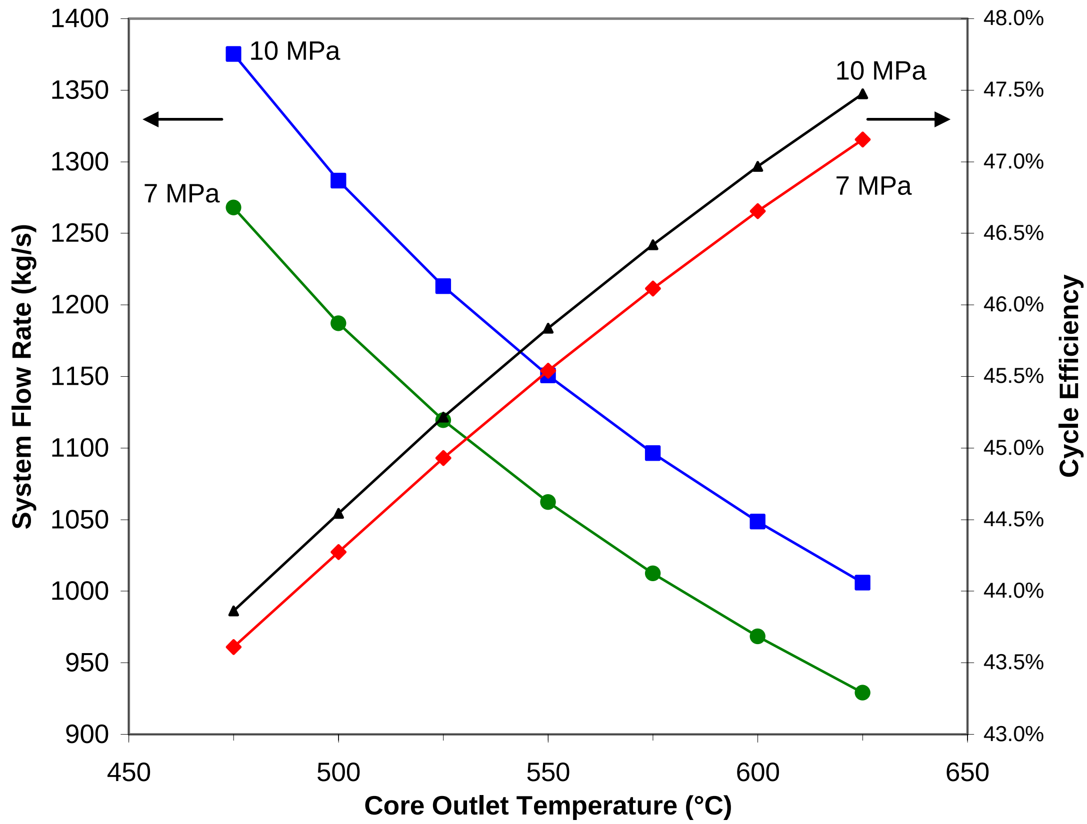

Figure 4. Effect of core outlet temperature and feedwater heater 7 pressure on efficiency and overall mass flow rate. LP turbine exhaust was kept at 9.8% moisture.

3.5. Comparison to an operating 950 MW class CANDU and the ACR

The results of this analysis have been compared to a 950 MW class CANDU and the ACR-1000 in Table 6 below. The figures show an approximate 1/3 reduction in core and plant size over the 950 MW CANDU.

Table 6. Comparison of the CANDU-SCWR with the CANDU 9 and ACR-1000.

	SCWR	950 MW CANDU[a]	ACR-1000[b]
Net Plant Power (MWe)	1000	1121	1100
Cycle Efficiency	47.4%	34.4%	37%
Reactor Core Power (MWth)[c]	2250	3400	3400
Core Pressure (MPa)	25	10	13
HP Turbine Pressure	25	5	7
Primary Coolant Flow Rate (kg/s)	960	13 500	Not Available
Secondary Coolant Flow Rate (kg/s)	—	1540	Not Available

[a] 950 MW class CANDU data from [5].
[b] Net plant power from AECL website. Efficiency and pressures from [6].
[c] Calculated by the author and rounded. Incorporates ~150 MWth lost to the moderator.

4. Conclusions

A supercritical feedwater heater placed at the end of the feedwater train was shown to be ineffective. However, this does not preclude the use of a lower-pressure feedwater heater in this position. Further analysis is needed in this area. Furthermore, a supercritical feedwater heater would become more effective at higher temperatures. It is doubtful that cycle efficiency would be greatly affected by such a feedwater heater, but it may be useful as a crud-precipitating chamber.

The reheater was shown to have a large impact on LP turbine exhaust, but was detrimental to cycle efficiency. A reheat temperature of around 315°C using steam at 10 MPa was shown to give an optimum cycle efficiency while maintaining moisture at the LP turbine exhaust below 10%.

It was shown that the operating pressure of the last feedwater heater (#7) had a dramatic effect on cycle efficiency and system flow rate. An increase from 7 MPa to 10 MPa resulted in an increase in cycle efficiency of 0.3% and an increase in system flow of 70-100 kg/s, depending on core outlet temperature.

Core outlet temperature was shown to be the major influence on cycle efficiency and system flow rate. At 625°C, the cycle efficiency averaged 47.4% and the system flow rate averaged 960 kg/s. A marked difference was observed at 475°C, where the cycle efficiency averaged 43.7% and the system flow rate averaged 1300 kg/s.

5. References

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