CYCLE EFFICIENCY AND FLOWRATES IN A SUPERCRITICAL WATER REACTOR: A THERMODYNAMIC ANALYSIS OF THE EFFECTS OF FINAL FEEDWATER TEMPERATURE AND NUMBER OF FEEDWATER HEATERS

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Abstract

Cycle efficiency and flow rates are rough measures of power plant revenue and capital cost. A full thermodynamic analysis of a supercritical water reactor provides these and other parameters necessary to perform an economic analysis and optimize system design. A computer code has been developed that works with Excel to assist thermodynamic analysis of user-defined power plant designs and related operating parameters. A number of direct-cycle plant designs with steam reheat and possessing between 4 and 10 feedwater heaters were analyzed. All cycles analysed used a core coolant exit temperature of 550ºC, a HP turbine inlet pressure of 24 MPa, and turbine efficiencies of 92%. Maximum cycle efficiencies (excluding generator losses, cooling water pumping power, and station power consumption) ranged between 47.8% and 48.8%. Optimum final feedwater temperatures of 326ºC to 349ºC were observed, with variation mainly a function of extraction steam origin. Overall flow rates were observed to increase some 19% from a final feedwater temperature of 280ºC to 340ºC, while extraction steam increased 50% over the same range. It is concluded that optimum efficiency does not coincide with lowest capital cost and a detailed economic analysis is yet needed to optimize reactor design.

1. Introduction

There are many factors to consider in optimizing the design of a SuperCritical Water-cooled Reactor (SCWR), including economic restraints, design requirements, materials requirements (and how they may be affected by design), plant layout and operating conditions. It is not enough to simply build the most thermally efficient plant design, as the optimum in thermal efficiency is concurrent with the maximum in capital costs. Ideally, a series of options would be evaluated for both capital costs and projected revenue, and the option with the greatest rate of return or shortest payback period would be chosen for the final design. The starting point for such an economic evaluation is a thermodynamic analysis.

Thermodynamic analysis is here defined as the evaluation of key thermodynamic metrics for a series of plant layouts and operating conditions. The most important thermodynamic metrics are cycle efficiency and overall cycle flow rate, as these give rough measures of power output (revenue) and plant size (capital cost). However, other metrics include 2nd-law efficiency (the efficiency of conversion of available energy gained by the coolant in the core to turbine shaft power), heat exchanger duty, turbine exhaust moisture content, and others. Intrinsic in the definition of thermodynamic analysis is that care has been taken to apply the laws of thermodynamics and the principles of fluid mechanics to obtain results (through an iterative solution-finding method) that are both accurate and meaningful.

A computer program, THERMO, has been developed that works with Excel to assist thermodynamic analysis of user-defined power plant designs and related operating parameters [1]. Plant designs may consist of heat sources (core passes), steam generators, turbines, steam separators, heat exchangers, and deaerators. For each component type, the code generates smart formulae to accurately calculate the thermodynamic outcome under a wide range of input parameters using thermodynamic data for light water from FLUIDCAL (a VBA module developed by Dr. Wagner of Ruhr-Universitat Bochum, Germany using the IAPWS-95 formulation for the thermodynamic properties of water). The formulae are placed in a user-friendly spreadsheet that works as an engine for finding the thermodynamic solution for the specified design and input parameters. Separate modules vary the input parameters, solve the system of equations and store the results. In this way, accurate thermodynamic solutions to systems whose input parameters are systematically varied can be obtained, and the result is a complete parametric analysis.

The results of a thermodynamic analysis of direct-cycle SCWR designs with steam reheat are presented here. The effects of final feedwater temperature (FFT) and the number of feedwater heaters on cycle efficiency and system flow rate have been investigated for final feedwater temperatures ranging from 200 to 400ºC and utilizing from 4 to 10 feedwater heaters[1]. All cycles analysed used a core coolant exit temperature of 550ºC, a HP turbine inlet pressure of 24 MPa, and turbine efficiencies of 92%. Plant power output was calculated as turbine shaft power minus the power consumption of modelled pumps, and thus do not include generator power losses, cooling water pumping power and station power consumption. Greater details of the methodology employed and the parameters used are described in the next section and in reference [1].

2. Analysis Methodology

A significant portion of the THERMO program code is devoted to selecting the proper thermodynamic equations for the components and the system as a whole. THERMO distinguishes five types of components: core, turbine, heat exchanger, deaerator and pump.

The heat gained by the coolant within the core, Q_C, was defined for all scenarios investigated as 3200 MWth. Q_C and the available energy gained by the coolant in the core, ΔAE_C, are used in the definitions of First- and Second-Law cycle efficiencies as defined in Table 1. Note that the power output is calculated as the turbine shaft power minus the required pumping power.

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[1] Throughout this paper, the number of feedwater heaters refers to the number of heaters along a particular feedwater train (e.g., Train A) and is equal to the number of unique extraction lines from the turbines. In real systems, it is common practice to divide extraction lines among several, identically-sized feedwater heaters existing in separate trains. However, the heat balance, and thus the quantity of steam, has been calculated by assuming a single train of feedwater heaters.

As mentioned above, turbine calculations assumed 92% efficiency versus isentropic expansion and ignored terms associated with potential and kinetic energy changes as well as heat loss. When design details are known, it is important to incorporate these terms. Potential energy contributions are minimal, and even though turbines are typically designed to minimize velocity changes [2], the large degree of volumetric expansion is likely to result in a net positive change in velocity. Nonetheless, even a velocity gain of 150 m/s would amount to no more than a 2% over-prediction of the power generated by a turbine. Within the context of this analysis, where an efficiency of 92% has been assumed for all turbines, it is justifiable to ignore these terms. Note that to account for extraction lines in the turbine energy balance, a coefficient, β, was incorporated into the definition of turbine shaft work in Table 1. This coefficient effectively closes the mass balance for all steams entering and exiting the turbine and simplifies the overall computation.

Energy balances for pumps were also calculated using an assumed efficiency versus isentropic operation. An efficiency of 75% was used for the core feed pump. Other pumps, such as the condensate pump, that are often placed at low elevations, were assumed to operate at a low efficiency of 30% for simplicity. Although this number is unrealistic, the calculated power requirements for the pump are well-approximated (based on code validation to real, operating plants – see reference 1) and the need to incorporate an elevation term is eliminated, which is convenient since a detailed plant layout is well beyond the scope of the present work.

Heat exchangers are defined by the duty, Q_{HX} or rate of heat exchange, which in turn is defined by the temperature rise of the feedwater or the change in enthalpy of the steam. All heat gained by the cold stream must derive from heat lost by the hot stream, as shown in Table 1. To ensure proper heat balance within a heat exchanger, it is important to define the temperature approaches or pinch points. Figure 1 shows a simple example where two temperature approaches have been specified within a feedwater heater. Depending on the fluids, and whether (pseudo)boiling or (pseudo)condensing processes are involved in the heat exchange, defining the temperature approaches can be complicated.

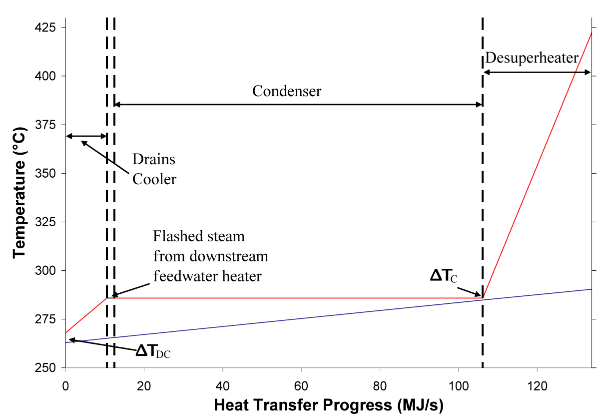


Figure 1 T-Q diagram of a typical heat exchanger with desuperheating, condensing and drains cooling sections.

Table 1 Main thermodynamic equations used in the THERMO program.

First Law Efficiency	$\eta_{1^{st}\,Law} = \dfrac{W_T - W_P}{Q_C}$	Second Law Efficiency	$\eta_{2^{nd}\,Law} = \dfrac{W_T - W_P}{\Delta AE_C}$
Pump Enthalpy Change	$\Delta h_P = \dfrac{p_{out} - p_{in}}{\rho_{avg} \cdot \eta_P}$	Pump Power Requirement	$W_P = M_\Sigma \cdot (h_{out} - h_{in}) \cdot \sum_i^{all\,streams} (\alpha_i f_i)$
Turbine Efficiency	$\eta_T = \dfrac{h_{in} - h_{out}}{h_{in} - h_s}$	Turbine Shaft Power	$W_S = (h_{in} - h_{out}) \cdot M_\Sigma \cdot \sum_i^{all\,streams} \alpha_i \beta_i f_i$
Coefficient β for extraction line A	$\beta_A = \dfrac{h_{in} - h_A}{h_{in} - h_{out}} = \dfrac{\eta_T (h_{in} - h_{s,A})}{\eta_T (h_{in} - h_{s,out})}$		
Direct Contact Unit Heat Balance	$\sum_j^{all\,inlets} \left(M_\Sigma \cdot h_{in} \sum_i^{all\,streams} (\alpha_i f_i) \right)_j = \sum_k^{all\,outlets} \left(M_\Sigma \cdot h_{out} \sum_i^{all\,streams} (\alpha_i f_i) \right)_k$		
Heat Exchanger Heat Balance	$Q_{HX} = \sum_j^{all\ hot\ inlets} \left(M_\Sigma \cdot (h_{in} - h_{out}) \cdot \sum_i^{all\,streams} (\alpha_i f_i) \right)_j = \sum_k^{all\ cold\ inlets} \left(M_\Sigma \cdot (h_{out} - h_{in}) \cdot \sum_i^{all\,streams} (\alpha_i f_i) \right)_k$		

Energy balances for direct contact units such as deaerators, steam separators and mixing chambers may deal with the saturation curve of water in some sense, which aids in finding the solution. The energy balance is very much an expression of the First Law for a net adiabatic process (see Table 1).

The THERMO program manages mass balances through two combined approaches. First, stream flow rates are approached as fractions (f_i) or percentages of the overall flow (M_Σ). Second, component flow rates are approached as linear combinations of the streams. A Boolean multiplier, α, is assigned to each stream for each component to designate whether that particular stream passes through a particular component. Naturally, the values of α may be either unity or zero, indicating "yes" the stream passes through the component or "no" the stream does not. Thus, the flow rate through component Y (M_Y) may be written:

$$M_Y = M_\Sigma \cdot \sum_{i}^{all\,streams} \alpha_i f_i \tag{1}$$

The test matrix and the base parameters used in the study are presented in Table 2 and Table 3. Figure 2 shows plant models for Test A and Test B with 10 feedwater heaters; note that the Test A model has feedwater heaters with extraction lines stemming from both the high pressure and low pressure turbines while the Test B model has feedwater heaters with extraction lines mainly stemming from the high pressure turbine. It should be noted that the Test B plant model in particular is not intended to represent a feasible design and has been arranged as indicated for the sole purpose of elucidating the effect of extraction line location on system variables.

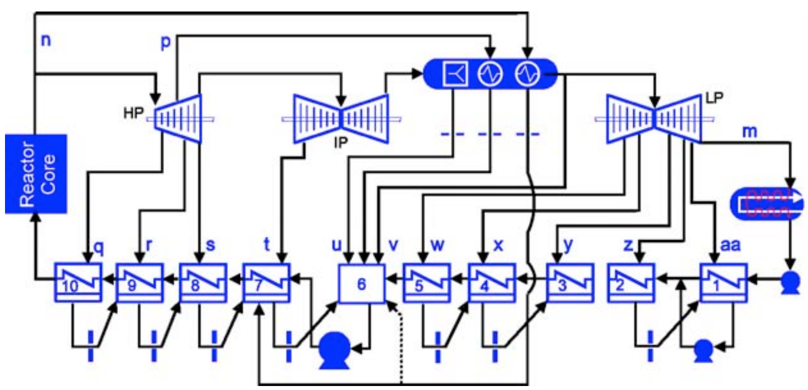
Table 2 Test matrix for evaluating the effect of the number of feedwater heaters and final feedwater temperature on system variables.

Parameter	Test A	Test B
Cycle Type	Direct	Direct
Coolant Core Exit Temperature (ºC)	550	550
1st Reheater Operating Pressure (MPa)	4	4
1st Reheater Coolant Exit Temperature (ºC)	~250	~250
1st Reheater Drains Cooler Temperature Approach (ºC)	6	6
2nd Reheater Operating Pressure (MPa)	24	24
2nd Reheater Condenser Temperature Approach (ºC)	100	100
2nd Reheater Drains Cooler Temperature Approach (ºC)	6	6
Final Feedwater Temperature (ºC)	200-400	200-400
Number of Feedwater Heaters[a]	3-10 S	4-10 B
Feedwater Heater Condenser Temperature Approach (ºC)	3	3
Feedwater Heater Drains Cooler Temperature Approach (ºC)	6	6

(a) S = Staggered around deaerator, B = Biased addition downstream of deaerator (FWH1 remains upstream of DA).

Table 3 Base parameters

Parameter	Value	Parameter	Value
Core Thermal Capacity (MWth)	3200 MWth	LP Turbine Exit Pressure	0.005 MPa
Insulation Thermal Efficiency	100%	Turbine Efficiency	92%
Extraction-Line Pressure Loss	3%	Steam Separator Exit Quality	99.5%
Pressure Loss of Other Piping	0%	Cooling Water Inlet Temperature (T_0)	10ºC
Coolant Core Exit Pressure	25 MPa	Cooling Water Outlet Temperature	22.9ºC
1st Reheater Condenser Temperature Approach	3ºC	Condensate Pump Efficiency	30%
2nd Reheater Condenser Temperature Approach	3ºC	Condensate Pump Exit Pressure	0.5 MPa
HP Turbine Inlet Pressure	24 MPa	Deaerator Operating Pressure	0.485 MPa
HP Turbine Exit Pressure	4 MPa	Booster Pump Efficiency	30%
IP Turbine Exit Pressure	0.5 MPa	Boiler Feed Pump Efficiency	75%



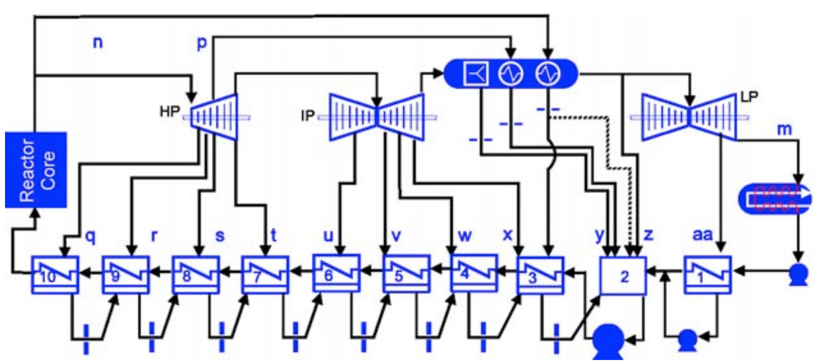


Figure 2 Plant models for Test A (top) and Test B (bottom) with 10 feedwater heaters.

3. Results and Discussion

3.1 Cycle Efficiency

The number of feedwater heaters and the origin of the turbine extraction lines feeding them had a noticeable effect on the cycle efficiency. A comparison of Test A and Test B results are shown in Figure 3. There are some common features between the two sets of results:

- There is an optimum final feedwater temperature for each case; and
- Adding an additional feedwater heater increases the overall efficiency.

A few differences are also apparent between the two sets of results:

- At low final feedwater temperatures, the number of feedwater heaters has a pronounced effect on the cycle efficiency in Test A, but not in Test B;
- The increase in efficiency obtained by increasing the number of feedwater heaters appears predictable in Test B, and more sporadic in Test A;
- The maximum efficiency for 5 feedwater heaters is greatest for Test B, while the maximum efficiency for 10 feedwater heaters is greatest for Test A.

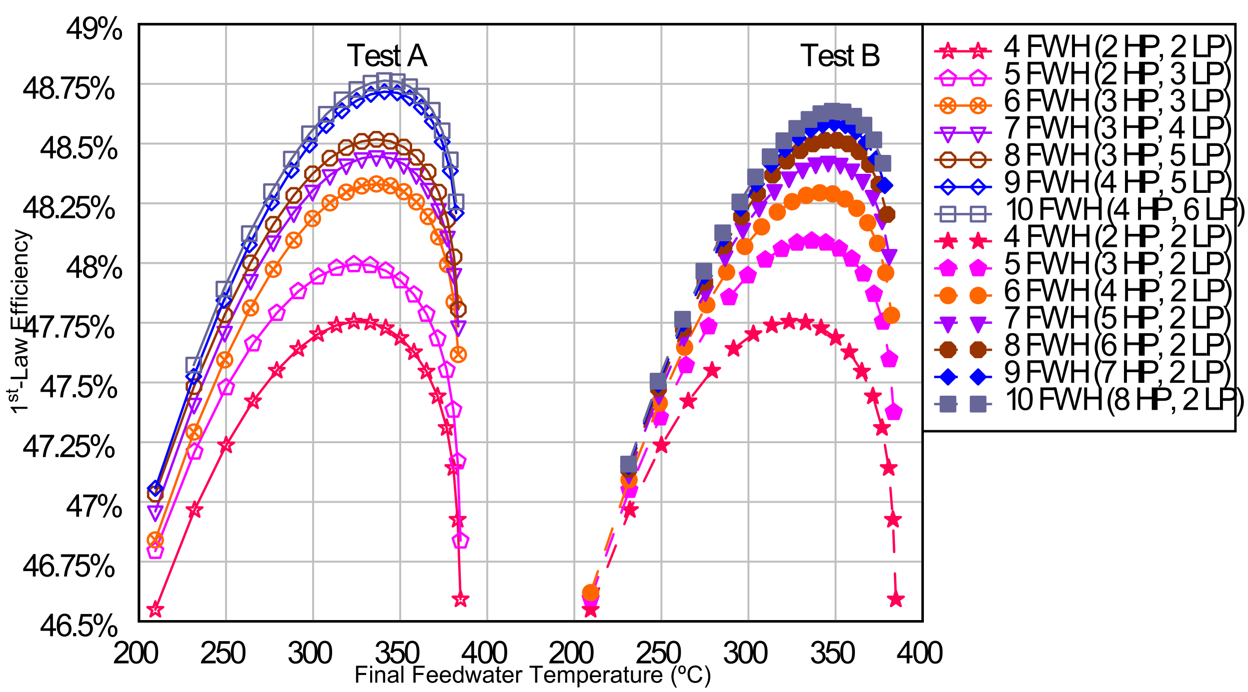


Figure 3 The effect of adding feedwater heaters to a basic 4-feedwater heater model on cycle efficiency. The origin of the extraction line (in parentheses in the legend) is seen to have an effect.

Each curve can be fit quite accurately to a rational polynomial. The optimum final feedwater temperature can be calculated by finding the zero of the derivative. The optimum final feedwater temperature and the corresponding maximum efficiencies are shown in Table 4.

Table 4 The effects of the number of feedwater heaters and the origin of turbine steam feeding them on the optimum final feedwater temperature and maximum cycle efficiency.

	Test A				Test B			
Number of Feedwater Heaters	HP	LP	Optimum Final Feedwater Temperature (ºC)	Max Cycle Efficiency	HP	LP	Optimum Final Feedwater Temperature (ºC)	Max Cycle Efficiency
4	2	2	325.9	47.75%	2	2	325.9	47.75%
5	2	3	325.9	47.99%	3	2	336.4	48.09%
6	3	3	336.4	48.33%	4	2	342.4	48.29%
7	3	4	336.4	48.45%	5	2	345.7	48.42%
8	3	5	336.4	48.52%	6	2	347.5	48.52%
9	4	5	342.4	48.72%	7	2	348.9	48.58%
10	4	6	342.4	48.76%	8	2	349.9	48.64%

It is clear from the results presented in Table 4 that the optimum final feedwater temperature is a function of the number of high pressure (HP) feedwater heaters (those fed from turbines upstream of the moisture separator reheater (MSR)). Further, it can be shown that the additions of the 3rd and 4th HP feedwater heaters result in equivalent rises in maximum cycle efficiency in both tests, irrespective of the number of LP feedwater heaters. The gain in maximum cycle efficiency observed by the addition of HP and LP feedwater heaters to the tested models is shown in Table 5.

Table 5 Observed gains in maximum cycle efficiency by the addition of HP or LP feedwater heaters as indicated.

HP Feedwater Heater Number	Gain in Maximum Efficiency	LP Feedwater Heater Number	Gain in Maximum Efficiency
3	0.34%	3	0.24%
4	0.20%	4	0.12%
5	0.13%	5	0.07%
6	0.09%	6	0.05%
7	0.07%		
8	0.05%		

It was also noted that at low final feedwater temperatures the number of feedwater heaters appears to have a noticeable effect on cycle efficiency in Test A (where LP and HP feedwater heaters were added to the base model) and not in Test B (where only HP feedwater heaters were added to the base model). A similar analysis to that shown above was conducted for a constant final feedwater temperature of 280ºC, and the results are shown in Table 6 and Table 7.

Table 6 The effects of the number of feedwater heaters and the origin of turbine steam feeding them on the efficiency of cycles with a final feedwater temperature of 280ºC.

	Test A			Test B		
Number of Feedwater Heaters	HP	LP	Cycle Efficiency with 280ºC FFT	HP	LP	Cycle Efficiency with 280ºC FFT
4	2	2	47.56%	2	2	47.56%
5	2	3	47.80%	3	2	47.77%
6	3	3	48.01%	4	2	47.88%
7	3	4	48.12%	5	2	47.94%
8	3	5	48.19%	6	2	47.99%
9	4	5	48.30%	7	2	48.03%
10	4	6	48.35%	8	2	48.05%

Table 7 Observed gains in the efficiency of cycles with a final feedwater temperature of 280ºC by the addition of HP or LP feedwater heaters as indicated.

HP Feedwater Heater Number	Gain in Efficiency with 280ºC FFT	LP Feedwater Heater Number	Gain in Efficiency with 280ºC FFT
3	0.21%	3	0.24%
4	0.11%	4	0.12%
5	0.07%	5	0.07%
6	0.05%	6	0.05%
7	0.03%		
8	0.03%		

A comparison of Table 5 and Table 7 indicates that the gain in cycle efficiency observed through the addition of a HP feedwater heater is dependent on the final feedwater temperature, while the addition of an LP feedwater heater results in a gain in efficiency that appears independent of final feedwater temperature.

3.2 Mass Flow Rate

Mass flow rate is a crude measure of plant size, as obviously the number of components and the degree of reheating and feedwater heating are large contributors as well. Nonetheless, the sizes of most plant components are directly related to the flow rate through them, with the possible exception of the core, which has particular lattice and pitch requirements. There are other factors that should be kept in mind: turbine sizes are also a function of the enthalpy change, heat exchangers are also largely a function of temperature differences, and steam separators are also a function of the outlet wetness. These factors are not negligible, but they are secondary to the flow rate, which owes its importance to the desire to maintain certain fluid velocities (typically 1-3 m/s for liquid flows in pipe, 20-50 m/s for steam flows in pipe [3], other equipment have other requirements) in order to keep head loss to a minimum. For nominally the same density at similar points between plant models, higher flow rates require larger flow areas to maintain the same flow velocity, and larger flow areas have obvious implications on pipe size, the number of heat exchanger tubes, the diameter of shells, etc.

In the systems investigated here, the core thermal capacity was kept constant at 3200 MWth, and thus the overall system flow rates were only affected by the final feedwater temperature and not the number of feedwater heaters, as shown in Figure 4. The flow rate of extraction steam was also mainly a function of the final feedwater temperature, with very little variation observed as a result of the number of feedwater heaters, as shown in Figure 5. Figure 4 and Figure 5 also show the overall flow rate and steam extraction flow rate, respectively, relative to that required at a final feedwater temperature of 280ºC. Note that the overall flow rate increases about 19% and the required steam flow rate increases about 50% between final feedwater temperatures of 280ºC and 340ºC. Over the same range, the total duty of the feedwater heater train increases about 60% (Figure 6).

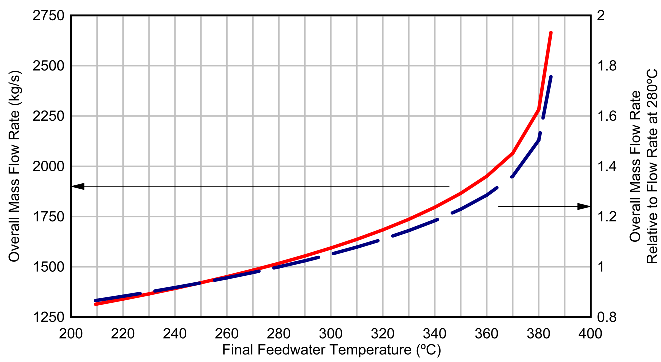


Figure 4 Effect of final feedwater temperature on overall cycle mass flow rate.

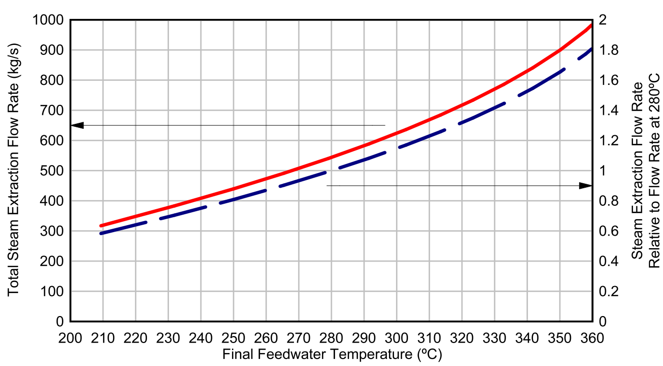


Figure 5 Effect of final feedwater temperature on the steam extraction flow rate (solid line). The effect of final feedwater temperature on steam extraction flow rate relative to that at 280ºC (dashed line).

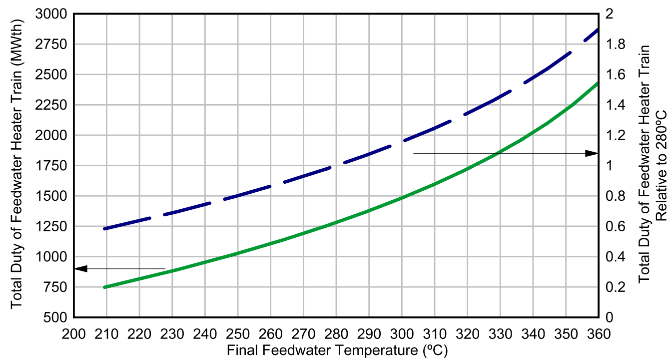


Figure 6 Effect of final feedwater temperature on the sum of the heat transfer within the feedwater heater train (solid line). The effect of final feedwater temperature on the sum of the heat transfer within the feedwater heater train relative to that at 280ºC (dashed line).

3.3 Implications on Cycle Design

The results presented above indicate how the number of feedwater heaters and the final feedwater temperature affect efficiency, a measure of economic revenue from a power plant, and flow rate, a measure of component sizes and hence capital costs. An economic evaluation would be necessary to determine the exact benefit of adding a feedwater heater or increasing the final feedwater temperature, but it is possible to discuss the general trends.

Although an optimum final feedwater temperature at which cycle efficiency reaches a maximum was shown to exist, it may be unreasonable to try to achieve it based on the effect of final feedwater temperature on overall flow rate. While an 8-feedwater heater model operating at the optimum final feedwater temperature of 342ºC (Test A) achieves 0.33% greater efficiency than one operating at a final feedwater temperature of 280ºC (Table 4 and Table 6), the hotter cycle also has about 20% greater overall flow rate and about 55% greater extraction steam flow rate (Figure 4 and Figure 5). Nonetheless, the choice of final feedwater temperature may be influenced by more practical considerations, such as the flexibility of the size of the core and the operating parameters of available turbines.

The choice of final feedwater temperature has some implications on the design of the feedwater heating train. It was shown that LP heaters increase cycle efficiency in predictable ways, and the economics surrounding the addition of a LP heater is largely independent of final feedwater temperature, making a capital cost assessment and cost-benefit analysis relatively simple. However, the same is not true of HP heaters, whose impact on cycle efficiency is tightly linked to the final feedwater temperature, which itself has an effect on flow rates and hence feedwater heater size. As well, the placement of the deaerator and its operating pressure should be functions of the final feedwater temperature and need to be considered in the design in order to avoid thermal shock to core internals under design accident conditions.

4. Conclusions

An investigation into the effects of the number of feedwater heaters and final feedwater temperature on SCWR cycle efficiency and mass flow rate was conducted by detailed simulations using THERMO, a program developed using VBA programming in MS Excel and using thermodynamic properties calculated through FLUIDCAL. It was found that an optimum final feedwater temperature may be found for a given arrangement of feedwater heaters to maximize cycle efficiency, and that this optimum temperature was a function of the number of HP feedwater heaters. However, it was also noted that maximum cycle efficiency does not correspond with minimum plant size (overall mass flow rate), and most efforts to maximize efficiency are concurrent with increases in plant size and capital cost.

5. Acknowledgments

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6. References

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