

THE DEVELOPMENT OF A STEADY STATE RANKINE CYCLE OPTIMIZATION ALGORITHM WITHIN
ASPEN HYSYS

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AUTHORIZATION TO SUBMIT THESIS

This thesis of Stephen G. Hancock, submitted for the degree of Master of Science with a Major in Nuclear Engineering and titled “The Development of a Steady State Rankine Cycle Optimization Algorithm within Aspen HYSYS,” has been reviewed in final form. Permission, as indicated by the signatures and dates below is now granted to submit final copies for the College of Graduate Studies for approval.

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ABSTRACT

The economic competitiveness of the energy market drives thermal power plant designers and operators to maximize the thermal efficiency of their power production cycles. The steam Rankine cycle is common to all nuclear power plants in the United States. A Rankine cycle is modeled in Aspen HYSYS and additional logic is used in the model to develop an optimization algorithm which will determine the expected thermal efficiency of the complete cycle based on user input parameters. The results are compared to work previously performed via manual optimization and show that the optimization algorithm performs better in most cases while requiring considerably less time to solve. The user is instructed on good practices for improving the efficiency further and suggested applications of the algorithm are provided.

ACKNOWLEDGEMENTS

Through this process I learned time management and self motivation for completing an independent project. I also struggled with the availability of my academic and research advisors on this work and I learned to speak up when a timely issue needed to be resolved.

I would like to thank those who have taught me and advised me through my master's program to help me finish this thesis and obtain the degree. Specifically my major professor, Dr. Christensen, for helping me pin point my research focus as a graduate student and also Dr. McKellar for providing the background work and scope for this thesis as well as additional instruction on Aspen HYSYS. Additionally, I would like to thank Dr. Boardman for his current support of my education and his future support as I continue my research and education.

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DEDICATION

Dedicated to my wife, Roxana, who always believes in me and continues to push me outside of my
comfort zone to be successful in my professional endeavors,

And to my parents for providing the motivation and support to attend graduate school.

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LIST OF ACRONYMS

BWR	Boiling Water Reactor
DCA	Drain Cooler Approach
DOF	Degrees of Freedom
GUI	Graphical User Interface
HTGR	High Temperature Gas Reactor
HP	High Pressure
IP	Intermediate Pressure
LMFR	Liquid Metal Fast Reactor
LP	Low Pressure
LWR	Light Water Reactor
NPP	Nuclear Power Plant
PWR	Pressurized Water Reactor
TTD	Terminal Temperature Difference

CHAPTER 1: INTRODUCTION

1.1 BACKGROUND

The energy sector in the United States is currently driven by economic factors. Manufacturing, construction, and operating costs are often cited as the motivation for pursuing a specific energy source [1]. The use of fracking in oil drilling has allowed for cheap access to natural gas, driving the cost down, and making it a more viable fuel option for electricity production than conventional base load electricity sources such as coal and nuclear power plants [2].

Nuclear reactors and coal fired power plants make up the majority of the base load power plants in the United States. In 2016 coal-fired power plants produced 30.4% and Nuclear Power Plants (NPPs) generated 19.8% of all utility scale electricity in the United States [3]. NPPs can produce a large amount of consistent electricity regardless of the time of day and weather [4]. They also operate most economically when generating constant power due to the high fixed and low variable operating costs [5].

Currently, NPPs in the United States are struggling economically due to the high availability and low cost of natural gas power plants [6, 7]. In many cases, this struggle has forced the early closure of NPPs which results in the loss of base load electricity for the energy sector, and creates other financial concerns for the surrounding communities [8]. Since the main reasons for early retirement of current NPPs are almost entirely financial, increasing the economic viability of new NPPs would allow these to reach the end of their licensed lifetime prior to shutting down [6, 7].

1.2 SCOPE OF WORK

There are currently 99 operating NPPs in the United States. Of these, two-thirds are Pressurized Water Reactors (PWRs), and the remaining are BWRs [9]. Each of these designs has its benefits and trade-offs. For example, the PWR designs generally have a higher thermal efficiency than the BWR due to the higher core operating temperature, 326.8°C for the PWR vs. 286.1°C for the BWR [10]. The trade-off required for this increased efficiency is the manufacturing cost of the reactor pressure vessel, and other reactor systems. To produce the higher temperature water, the PWR must operate at a pressure of 15.51 MPa compared to the 7.14 MPa required for the BWR [10].

There are other NPP designs which have been investigated over the years using different nuclear reactor designs. Table 1.1 shows some examples of these reactor designs with the primary coolant conditions [11, 10]. The High Temperature Gas Reactor (HTGR) and the Liquid Metal Fast Reactor (LMFR) designs are included in the Generation IV reactor designs discussed by the Generation IV International Forum [12]. These designs are encouraged and supported internationally for future development and commercial construction. The parameters shown in Table 1.1 are all United States designs except for the

Table 1.1: Different nuclear reactor types and selected thermal hydraulic parameters

Reactor Type	BWR	PWR	HTGR	LMFR
Manufacturer	General Electric	Westinghouse	General Atomic	Novatome
Primary Coolant	Water	Water	Helium	Sodium
Thermal Efficiency	32.0%	33.7%	38.6%	41.3%
Coolant Outlet Temperature	286.1°C	326.8°C	741.0°C	545.0°C
Reactor System Pressure	7.14 MPa	15.51 MPa	5.0 MPa	0.1 MPa

LMFR design which was a Swiss design and operated as the Superphenix-1 reactor.

The general trend presented by the different reactor designs is an increase in thermal efficiency as the coolant outlet temperature increases. This trend is not followed when comparing different coolants, such as the HTGR which uses helium as the primary coolant. Although helium has a higher heat capacity than water and sodium, the coolant is a gas and therefore has a lower volumetric heat capacity [10]. The outlet temperature of the BWR is the saturated temperature at the system pressure which forces the upper limit of thermal efficiency. The reactor pressure vessel produces the steam which is used in the Rankine cycle for power generation. The PWR steam generator uses the heat from the primary coolant to generate the steam for the Rankine cycle in the secondary loop. Depending on the steam generator design, which is discussed later, the steam used in the Rankine cycle is either saturated or superheated by just over 25°C [11].

Increasing the coolant outlet temperature allows for more superheat in the steam. Generation IV nuclear reactor designs have higher coolant outlet temperatures which theoretically allow for a power cycle with a higher thermal efficiency. This is one economic factor which can be improved to allow NPPs to continue to be competitive in the current energy market. Any marginal increase in the thermal efficiency of the generation system would improve the economics of the plant significantly. For example, the Palo Verde Nuclear Generating Station in Arizona has three PWRs which each produce around 1300 MWe from almost 4000 MWth [13]. Each hundredth of a percent (0.01%) increase in thermal efficiency for these reactors provides 400 kW of additional electricity from each reactor to be used on site or dispatched to customers. If the average price of electricity for customers is 10 ¢/kWh, then there is potential for an additional \$1.05M increase in revenue per year for the entire plant.

Since any fractional increase in thermal efficiency of a power plant can impact the economic stability of the plant to such a large degree, the purpose of this thesis is to develop a model which will accurately estimate the optimum thermal efficiency of a Rankine cycle given the estimated superheated steam temperature and the cooling water availability temperature. The cooling water temperature specified in the model is connected to a forced draft cooling tower; therefore, the cooling water temperature can be found by using the average ambient conditions in the expected area and the estimated temperature of the source water. This work has already been completed by Dr. Michael McKellar at several different

steam conditions. However, the time commitment undertaken by Dr. McKellar to achieve optimal cycle operation is significant and the results are limited to specific temperatures and pressures. To improve on this work, the model will automatically estimate the optimum operating conditions based on basic user inputs. The Rankine cycle is modeled using the Aspen HYSYS process modeling software with an academic license purchased by the University of Idaho for the 2017-2018 academic year.

1.3 THESIS OUTLINE

Chapter 2 discusses the overall Rankine cycle operation including all unit operations used in the model. Different designs and types of each system are discussed for a thorough understanding of how each operates. The thermodynamic behavior and equations is also developed for understanding of how Aspen HYSYS performs its calculations. Additionally, Aspen HYSYS is briefly introduced.

Chapter 3 presents the methodology used to enhance the capabilities of Aspen HYSYS beyond the basic modules simulation. This enhancement uses the customization techniques discussed in Chapter 2 as well as additional operations to assign set points for different parameters. The assumptions and set points used are discussed with explanations. The process by which the pressure is found is also developed.

Chapter 4 compares the results of the updated model and its optimization with the optimization work performed by Dr. Michael McKellar on the same model in HYSYS. This establishes an overall performance criteria for the model. Further analysis is also performed to analyze how the efficiency may vary when ambient conditions are changed and to evaluate how these changes might affect the other systems in the model.

Chapter 5 summarizes the topics explored by the thesis and the development of the optimization algorithm and accompanying results. The conclusions are stated and the potential for future applications are explored.

The Appendix contains two parts. The first part shows the process report generated by Aspen HYSYS for the modeled system. This includes the process flow diagram and all pertinent streams and unit operations with the set points required for the model to be recreated. The second part of the appendix contains the user variables used in the model to automate calculations to achieve the recommended operating pressure.

CHAPTER 2: THEORY

2.1 RANKINE CYCLE

The Rankine steam cycle is the most widely used power cycle for the production of base load electricity [11]. Large base load thermal power plants, such as coal and nuclear, use the Rankine steam cycle to generate electricity [14]. Many natural gas power plants use a combined cycle where the primary power generation is accomplished through the use of an air Brayton cycle and the exhaust air is then used to generate steam for a coupled Rankine cycle [15]. The Rankine cycle in a power plant is generally a complex system with many turbine stages and heat exchangers used to achieve a high thermal efficiency [16].

In its simplest form, the Rankine cycle consists of four steps: compression, heat addition, expansion, and heat rejection. In the ideal Rankine cycle the pressure changes occur through adiabatic expansion and compression in the turbine and the pump. The heat addition and heat rejection are constant pressure processes in the boiler and the condenser [16]. This cycle is a derivative of the Carnot cycle, or the ideal heat engine. Two-phase processes such as the Rankine cycle are typically represented on a phase diagram, in this case a temperature-entropy (T-S) diagram. The dome encompasses the two-phase region with the liquid phase on the left outside of the dome and the vapor phase on the right outside. The apex of the dome is the fluid's critical point.

Figures 2.1 and 2.2 show the process diagram for the Carnot cycle and its accompanying T-S diagram.

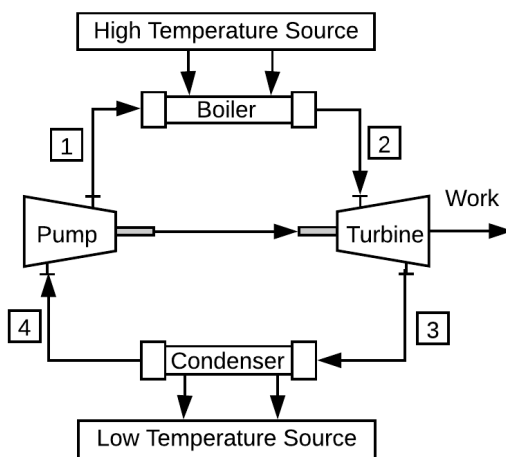


Figure 2.1: Process diagram of the Rankine cycle

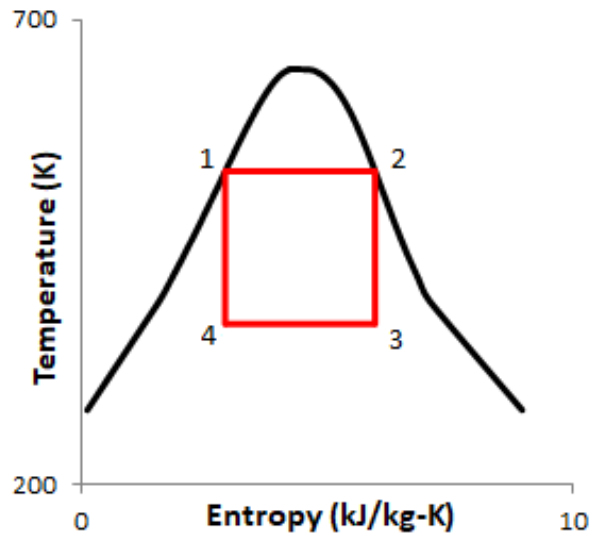


Figure 2.2: Temperature-Entropy diagram of the Rankine cycle

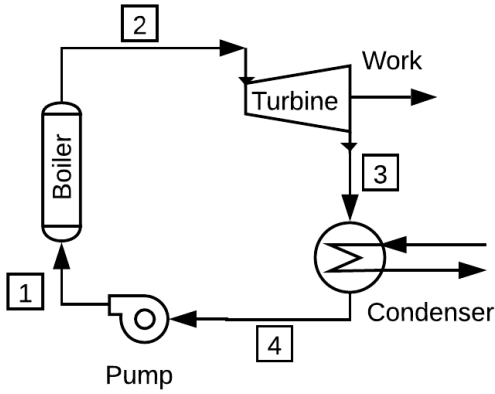


Figure 2.3: Process diagram of the basic Rankine cycle

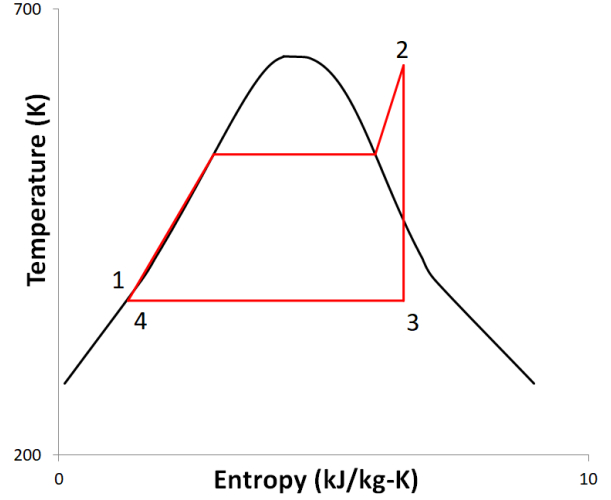


Figure 2.4: Temperature-Entropy diagram of the basic Rankine cycle

Figures 2.3 and 2.4 show the same diagrams for the basic Rankine cycle. The main difference introduced by the Rankine cycle is the complete condensation of the working fluid to the saturated or subcooled liquid state, as can be seen in the T-S diagrams. Therefore, a liquid pump replaces the compressor in the Carnot cycle. Since the liquid heating in the Rankine cycle is non-isentropic, shown in the T-S diagram, the Rankine cycle has a lower efficiency than the Carnot cycle [16]. The efficiency is represented by the relative areas within the red cycle shown in Figures 2.2 and 2.4 divided by the sum of that area and the total area underneath the line from point 3 to 4 if the graph reached 0 K. The main impact on efficiency is the wider range of entropy covered by the Rankine cycle.

For a thermal cycle, the efficiency is defined according to Equation 2.1. The end result for the Carnot cycle efficiency is shown in equation 2.2 [16].

$$\eta_{cy} = \frac{W_s}{Q_{in}} \quad (2.1)$$

$$\eta_{cy} = \frac{Q_h - Q_l}{Q_h} \Rightarrow 1 - \frac{T_l}{T_h} \quad (2.2)$$

The Carnot cycle efficiency represents the maximum theoretical efficiency for converting heat into mechanical work. The Rankine cycle efficiency will always be less than the Carnot efficiency and is found by following Equation 2.1 with the W_s being the net work output of a system and Q the total heat input.

Since the Rankine cycle is a derivative of the Carnot cycle, there are only three options for increasing thermal efficiency: decreasing T_l , increasing T_h , or both [14]. In an operating power plant the condensing temperature is set by the available heat sink which is based on ambient conditions. The high temperature

can only be adjusted to a certain degree because of mechanical and structural limitations on the materials available. Other techniques are available for increasing thermal efficiency in addition to superheating the steam which include regeneration and reheating [14, 16, 10].

The steam expansion process can be broken into several stages and small amounts of the high temperature or two-phase steam can be extracted from an expansion stage and used for regenerative feed water heating [14]. In general there are two types of feed water heaters, the contact heater and the surface heater, which are developed in a later section [16]. The small amount of mass flow lost from the extraction causes a decrease in generated work in successive expansion stages; however, the feed water heating generates a higher temperature feed water supply to the steam generator. This allows for a higher steam flow rate at the same thermal power rating. A basic regeneration process and its accompanying T-S diagram are shown in Figures 2.5 and 2.6. The 3a split in the T-S diagram represents the state point of 3a and 3b. The b path follows the dotted line across the two-phase dome to preheat the water prior to entering the boiler. The actual position of point 4b is slightly subcooled liquid at a temperature slightly above that of point 6. The outlet of the condenser is the segment beneath points 5 and 6 but then the mixture of points 4b and the condenser outlet bring the system to the conditions at point 5.

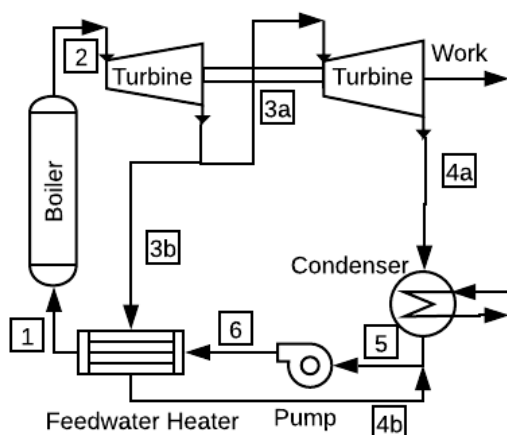


Figure 2.5: Process diagram of the regenerated Rankine cycle

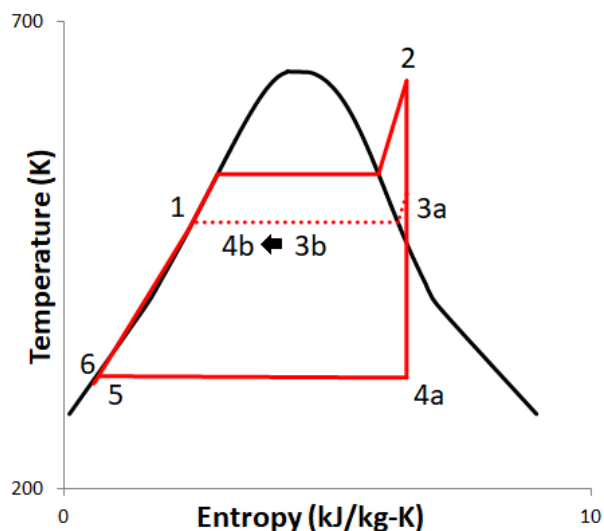


Figure 2.6: Temperature-Entropy diagram of the regenerated Rankine cycle

Now that the expansion process has been split into stages, the main steam line can also be sent back to the steam generator for reheating to increase the temperature after partial expansion [10]. For maximum effect, this process typically occurs following the first stage of expansion in the HP turbine. This allows the steam to be theoretically reheated to the maximum temperature at a relatively high pressure before continuing the expansion process in the lower pressure turbines [10]. The reheating process may

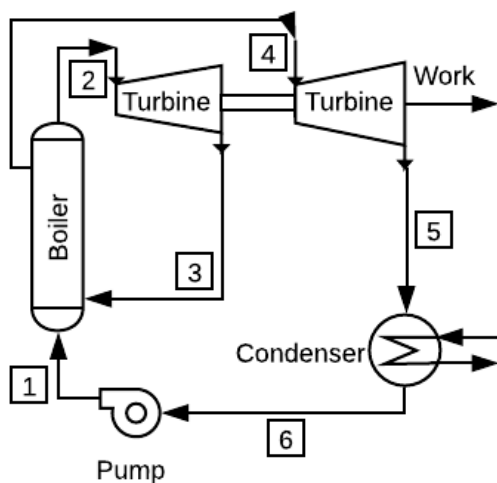


Figure 2.7: Process diagram of the reheated Rankine cycle

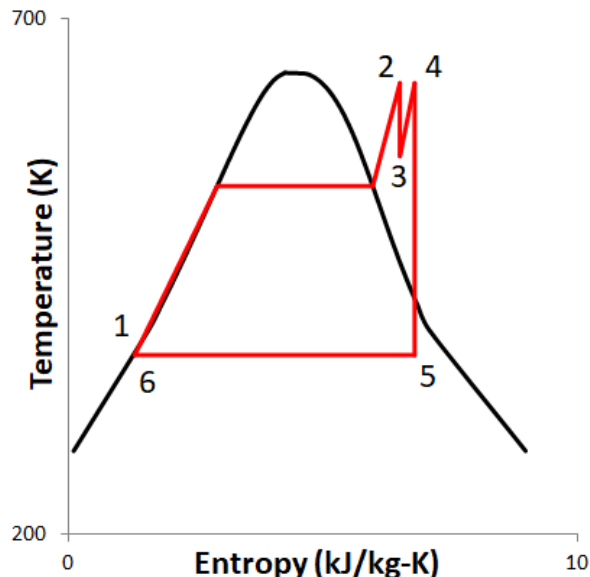


Figure 2.8: Temperature-Entropy diagram of the reheated Rankine cycle

not largely increase the thermal efficiency in all cases, but it does reduce the moisture content leaving the turbine stages at lower pressures, which allows for additional expansion which may otherwise be impractical due to moisture limits on steam turbines [16]. The reheating process and T-S diagrams are shown in Figures 2.7 and 2.8.

The Rankine cycle modeled in this Thesis includes all processes for increasing thermal efficiency: superheating, reheating, and regeneration. The complete process diagram and T-S diagram are shown in Chapter 3.

Through the development of the completed cycle, the system pressure becomes another parameter that must be varied to accommodate the increasing temperature. The boiling pressure is mainly tied to the expansion process from the higher temperature to the low temperature and is a side effect of the regeneration process. Steam pressure also affects the capital cost of the system, which is another condition to ultimately optimize; however, this optimization focuses on the thermal efficiency of the system, disregarding the capital cost.

The design and theory for the steam Rankine cycle has been investigated extensively over several decades. It is necessary to note that current research and development for the Rankine cycle deals with the organic Rankine cycle. The overall process is the same as the steam Rankine cycle, the difference is in the working fluid. Fluids which have lower evaporation temperatures, such as refrigerants or hydrocarbons, may be used in the place of steam to generate electricity when the available temperature is not high enough to warrant the use of a steam cycle [17]. An example of this is geothermal energy production, where the

available temperature typically falls in the range of 100°C-220°C. Water is used as the primary coolant for the extraction of heat, but is then used to vaporize the secondary organic fluid for use in the organic Rankine cycle [18]. Another application of the organic Rankine cycle is in solar thermal systems where several mirrors are used to direct the solar heat to a concentrated point [17]. The available temperature for power generation in some cases is not consistently high enough for the steam cycle. A major concern with the organic Rankine cycle is the capital cost for the power plant; however, when a steam cycle is not efficient enough to be profitable, it may be worth the investment to increase power production from lower temperature sources [18].

2.2 UNIT OPERATIONS

A complete Rankine cycle is a complex system full of different turbomachinery, pipes, heat exchangers, valves, control systems, electricity generator and much more. Each part is integral to the controlled operation of such a large system at a constant power level. The model presented here is a base model of the performance of a complete system without all of the necessary parts. The model is made up of the main shell of the system containing only the large unit operations which form the thermodynamic calculations. These will provide an estimate of the total power production from the system while excluding the equipment and electronics required for operation and control, which is outside the scope of this thesis. The unit operations under consideration for the calculations and analysis are heat exchangers, turbines, pumps, feed water heaters, and the cooling tower.

2.2.1 HEAT EXCHANGERS

The heat exchangers discussed here refer to the steam generator, reheater, and condenser of the system. The feed water heaters will be explored in a later section. In all heat exchangers the fundamental equation for heat exchanger performance is shown in Equation 2.3

$$\dot{Q} = \dot{m}\Delta H \quad (2.3)$$

\dot{Q} is the total heat rate, \dot{m} is the mass flow rate of the transfer fluid, and ΔH is the enthalpy change which occurs in the fluid. The most accurate approximation for the change in enthalpy in single phase heat transfer is the specific heat of the fluid at constant pressure multiplied by the temperature change ($c_p\Delta T$). When going through a phase change, the enthalpy change is based on the heat of vaporization of the fluid at the operating pressure of the heat exchanger. The use of the constant pressure specific heat and/or heat of vaporization works because the heat exchanger is generally a constant pressure operation when compared to other thermodynamic operations in the Rankine cycle. There are further correlations and equations which are used to describe the geometry, size, and flow pattern of the heat transfer fluids. These designs specifications correlate with Equation 2.3 through the overall heat transfer coefficient (UA)

and the logarithmic mean temperature difference as shown in Equation 2.4. The temperature differences shown in Equation 2.5 are the approach temperatures (hot and cold fluid temperature difference) on either side of the heat exchanger.

$$\dot{Q} = UA\Delta T_{lm} \quad (2.4)$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (2.5)$$

For all heat exchangers in this thesis, Equation 2.3 is sufficient since the geometric design and configuration of the heat exchangers is not necessary for the model.

All of the heat exchangers in the Rankine cycle modeled in this thesis, with exception to the reheater, undergo a phase change in either the hot or cold fluid. This is significant because phase change heat transfer is remarkably effective in heat exchangers due to typically large heats of vaporization [19]. Due to the high effectiveness of boiling heat transfer, the heat exchangers in the Rankine cycle are able to be designed to have a low minimum approach temperature. That is, the temperature difference between the hot and the cold fluids can be rather small without the need of abnormally large heat exchangers. Although the design of the heat exchanger is not being considered in this thesis, this supports several assumptions made for the heat exchanger operation.

Steam Generators This section focuses primarily on the steam generators typically used in NPPs currently operating in the United States. Additionally, steam generators suggested for Generation IV reactor designs are also discussed; however, in more minor detail. Aside from the steam generator, the remainder of the Rankine cycle is general across all thermal power plants, therefore other unit operations are not explored in as much detail as the steam generator.

The steam generator serves as the boiler in a NPP. In most cases, the hot primary fluid transfers heat to the secondary fluid in the steam generator to generate the steam used in the Rankine cycle [11]. There are advanced reactor designs which have an intermediate fluid and loop between the primary coolant loop and the steam generator. This is necessary if there is an immediate need to separate the primary coolant, which may contain radioactive particles, from the water in the steam loop. One example of this design is the Sodium Fast Reactor, since sodium reacts violently with water [10]. All commercial reactors in the United States are Light Water Reactors (LWRs), therefore, the primary coolant is always water. The PWR design requires two cooling loops with the steam generator as the heat sink for the primary coolant. These steam generators are limited to generating saturated or slightly superheated steam due to the operating temperature and pressure of the power cycle. The reactor pressure vessel of a BWR serves as the steam generator since the primary coolant is used as the working fluid in the Rankine cycle [11].

Operating PWRs in the United States use two different steam generators. Reactors manufactured by Westinghouse Electric and Combustion Engineering use a U-tube steam generator. The U-tube steam generator system is made up of a one shell pass, two tube pass heat exchanger and a moisture separator and steam dryer, shown in Figure 2.9. The primary coolant flows inside the tubes entering and exiting at the bottom of the heat exchanger. The secondary fluid feed water enters the heat exchanger towards the bottom and flows up on the outside of the tubes as it gains energy to produce wet steam. The water level of the secondary fluid is maintained above the tubes and below the moisture separator. The moisture is removed from the steam and recycled into the secondary inlet stream as the steam is dried to a desired moisture content of 0.45% or less [11]. The result is a dried saturated steam for use in the Rankine cycle.

The Babcock & Wilcox reactor design uses a once through shell and tube steam generator, shown in Figure 2.10 [11, 14]. Like the U-tube steam generator, the primary coolant flows inside the tubes but enters at the top of the exchanger and exits the bottom. The feed water enters at the side of the exchanger and flows down through the annulus and then upward through the generator. At the top the secondary fluid is directed down the annulus and out the side of the exchanger at a location slightly higher than the feed water inlet. In the steam generator, the secondary fluid is converted to steam and then slightly superheated with 50°F to 60°F (27.77°C to 33.33°C) of superheat [11, 14]. The advantage of the superheated steam in the Rankine cycle raises the thermal efficiency of Babcock & Wilcox designs to 34.5% compared to the Westinghouse Electric PWR design at 33.7% [14, 10].

The General Electric BWR reactor design uses the reactor core as its steam generator, shown in Figure 2.11. The BWR is used to boil the primary coolant which enters the vessel at the bottom and flows up through the fuel assemblies. The two-phase mixture leaves the top of the fuel assemblies with a steam quality of 14% maintaining the fuel rods entirely submerged in water. This mixture then enters three stages of moisture separation followed by the steam dryer which dries the steam to a quality of at least 99.6% [11]. The BWR operates at a lower temperature than the PWR and as a result has a thermal efficiency of 32.0% [10]. At the loss of thermal efficiency, there are two benefits from this design. First, the lower operating pressure which decreases capital cost and overall safety concern, and second, the heat transfer in the core is primarily due to latent heat instead of sensible heat which allows for a lower mass flow rate of the coolant which results in a decrease in pumping costs [11].

Other reactor types use different steam generator designs. The Fort Saint Vrain reactor in Colorado, which no longer operates, was a HTGR designed by General Atomic. This reactor used a single pass helical coil steam generator, shown in Figure 2.12 [20]. The helium entered the steam generator at the bottom and flowed up around the coil exiting at the top. The feed water also flowed from the bottom of the steam generator and exited as steam at the top [11]. Due to the high temperature of Helium entering the steam generator (741°C), the steam generator was capable of producing superheated steam at 513°C and 17.3 MPa which produces a thermal efficiency of 38.6% [10].

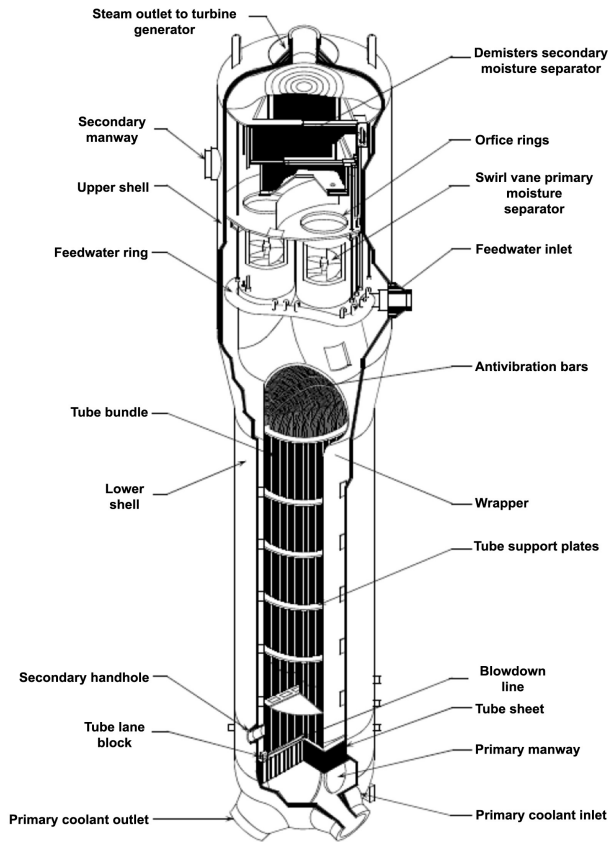


Figure 2.9: Westinghouse u-tube steam generator [23]

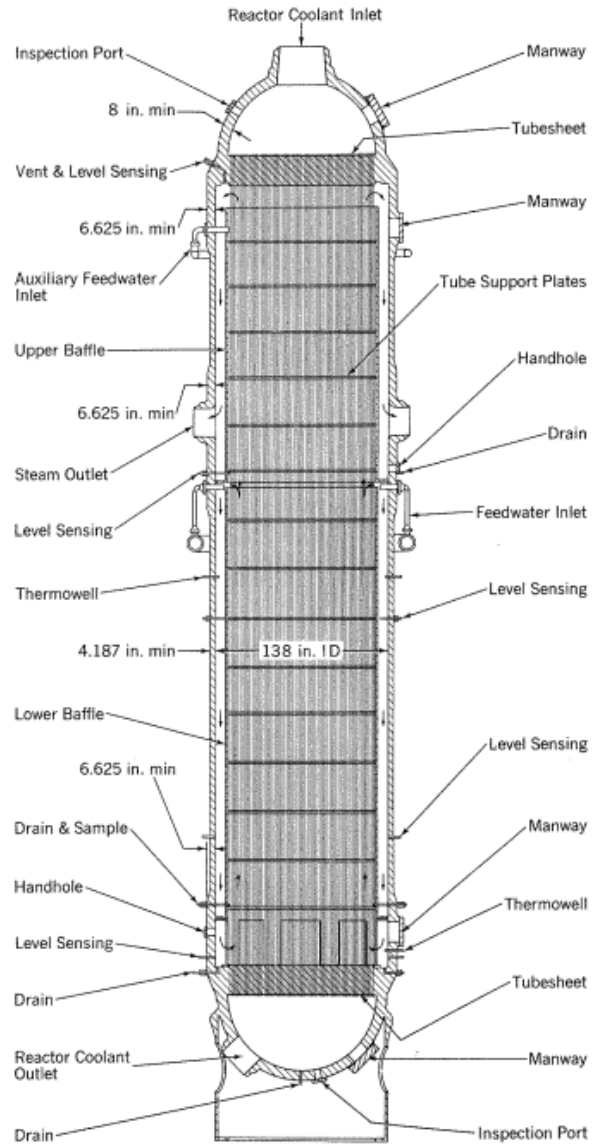


Figure 2.10: Babcock & Wilcox once through steam generator [14]

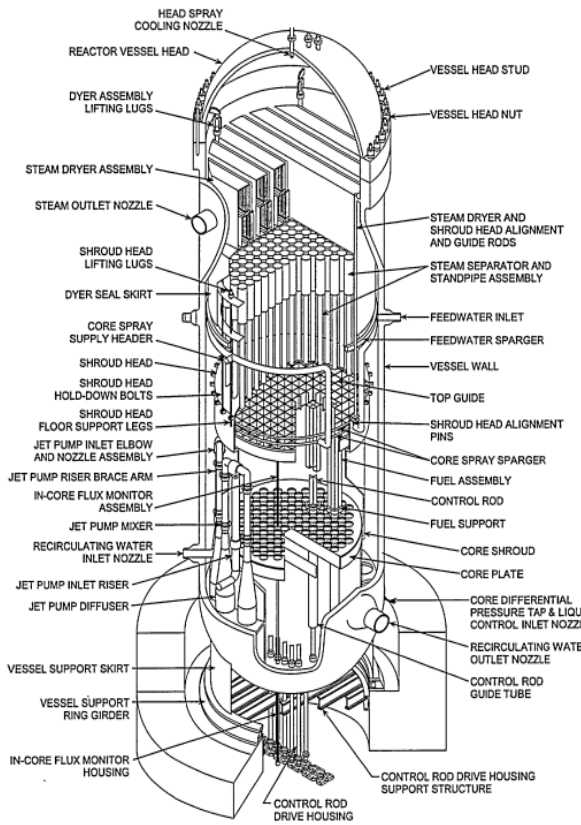


Figure 2.11: General Electric BWR/4 reactor pressure vessel diagram [24]

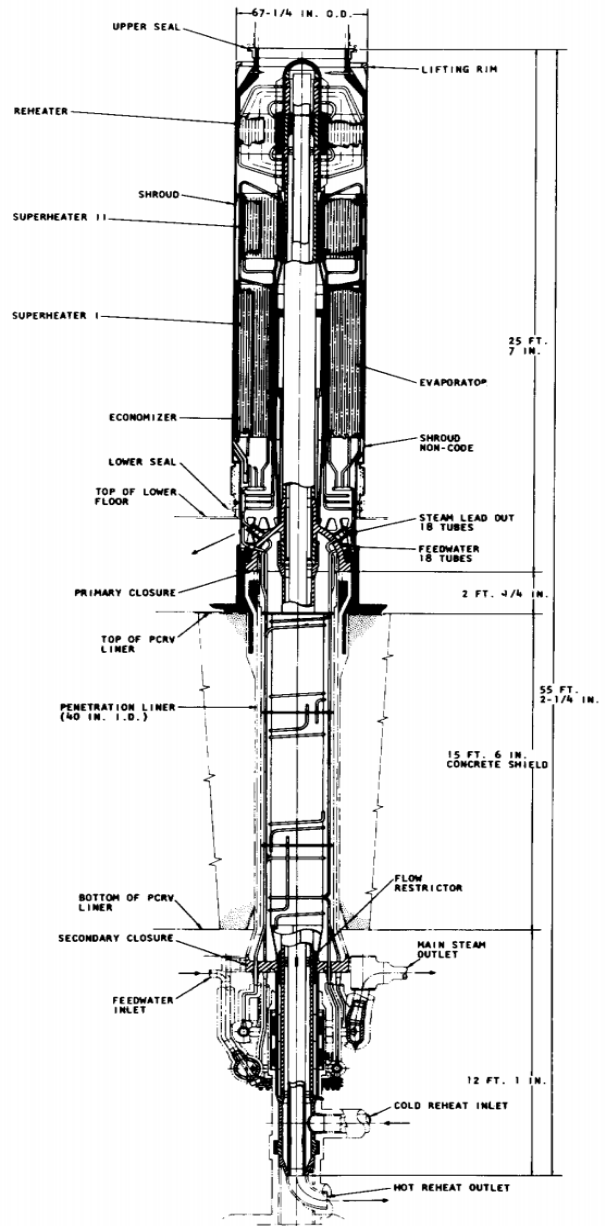


Figure 2.12: General Atomic helical coil steam generator from Ft. St. Vrain station [25]

Novel reactor designs typically use different steam generator designs which are more effective or more applicable to the system. The International Reactor Innovative and Secure design and the Integral Inherently-Safe Light Water Reactor design are two medium and small sized LWRs with integral reactor designs [21, 22]. The integral design means that all primary systems, steam generator, primary coolant pumps, and pressurizers, are built inside of the reactor pressure vessel [21]. The International Reactor Innovative and Secure uses a once through helical coil tube bundle design with the secondary fluid inside the tubes. There are eight steam generator modules located inside the reactor vessel on the outside of the core barrel. The primary coolant flows from the top to the bottom of the vessel and the feed water enters at the bottom of the coil and flows upwards [21]. The Integral Inherently-Safe Light Water Reactor uses a micro-channel heat exchanger design for the steam generators located within the reactor pressure vessel [22].

An example of a non LWR novel design is the High-Temperature-Reactor Pebble-Bed Module which is a demonstration plant in China. This reactor uses a vertical steam generator, which is separate from the reactor vessel, with 19 separate helical tube assemblies. Each assembly has 5 layers and include 35 helical tubes [26]. Although effective in design, this steam generator is the most difficult piece of equipment to manufacture for this reactor system [27].

Reheaters Since current operating NPPs in the United States are LWRs which produce slightly superheated steam at best, the reheater is essential to power plant operation. In fully reheated systems the steam returns to the high temperature source after being expanded through the HP turbine. The reheating process either brings the lower pressure steam back to the high pressure steam temperature or to a temperature slightly below, depending on the design [14]. This effectively reduces the moisture content of the steam allowing for additional expansion and in turn, more power production. However, a typical LWR steam generator does not have excess heat for a reheat stream. The reheat stage in LWR systems produce partially reheated steam by combining the HP turbine outlet stream with some bleed steam from the steam generator while also undergoing a moisture separation stage. This partial reheat, however small, provides an increased thermal efficiency to the system allowing LWR systems to be competitive with other energy sources.

A nuclear system that was designed with full reheat is the HTGR design by General Atomic. Since the steam is capable of being superheated substantially, the general design for the system contains a reheater section in the steam generator [11]. A diagram of this reheater/steam generator system is shown in Figure 2.12. As the helium exits the core, it first flows through the reheater section of the steam generator and then up through the primary steam generator [11]. This design was used in the Fort Saint Vrain reactor which helped to bring the thermal efficiency of the plant to about 39% [20].

Condensers Since steam cycles have low thermal efficiencies, the excess heat generated must be released from the system. This is accomplished in the condenser, which returns the steam or two-phase mixture to a completely liquid state with a few degrees of subcooling. Most of the heat that is released is latent heat with room for some sensible heat. Cooling water from some source is used in the condenser to reject the heat from the system [14]. This is accomplished either by direct cooling from a body of water (a lake, river, or the ocean) with a once-through process or by the use of a cooling tower which will be explained later in this chapter. At atmospheric conditions this steam to liquid phase change occurs at 100°C which is significantly higher than most available heat sinks. To further increase thermal efficiency of the cycle, the steam is expanded even further to vacuum conditions at which the saturated temperature is relatively close to the heat sink temperature. In addition to higher thermal efficiency, this has the effect of less temperature increase in the heat sink. Operating at vacuum conditions also causes the condenser to be more expensive and larger by design [10].

There are three main types of condensers: water-cooled surface condensers, water-cooled contact condensers, and air-cooled surface condensers. The most common for power plant applications is the water-cooled surface condenser [16]. This condenser is a shell and tube heat exchanger with the cooling water circulating through the tubes and the steam condensing on the outside of the tubes. These can either be single-pass or double-pass, with the double-pass being generally more economical in a cooling tower application and the single-pass being a better choice with direct cooling [16].

The condensing steam causes a pool of condensate to form at the bottom of the condenser. This requires a condensate pump to move the feed water back to the steam generator. There is also air that can accumulate in the steam space of the condenser because of the vacuum conditions. This air is generally removed using a steam-jet air ejector, although mechanical vacuum pumps also work [16]. The jet air ejector is preferred over the vacuum pumps since it does not require power, but works based on natural forces.

2.2.2 TURBINES

The turbine is the most important operation in the Rankine cycle because it is used to convert the energy stored in the steam to mechanical work. This mechanical work is then used to generate electricity by directly spinning the electrical generator. Two types of steam turbines are typically used simultaneously in power generation: impulse and reaction [16]. In an impulse turbine, the fluid's energy is converted to kinetic energy by flowing through a nozzle. This energy is then used to spin moving blades and convert the energy into mechanical energy [28]. This is often compared with a water wheel because the steam jets flow against the moving turbine blades. In a true impulse turbine all of the pressure drop occurs in the nozzle to generate the kinetic energy [29]. A reaction turbines consist of various types of blades which control and direct the flow onto the runner blades which are the rotating element used to

generate the mechanical energy [28]. There is a continuous pressure drop throughout the turbine as the steam passes through each stage of stationary blade nozzles [29]. Figure 2.13 shows a representation of both types of turbine stages and their blade configurations. The diagram on the far right of Figure 2.13 shows a combination of both impulse and reaction stages in succession.

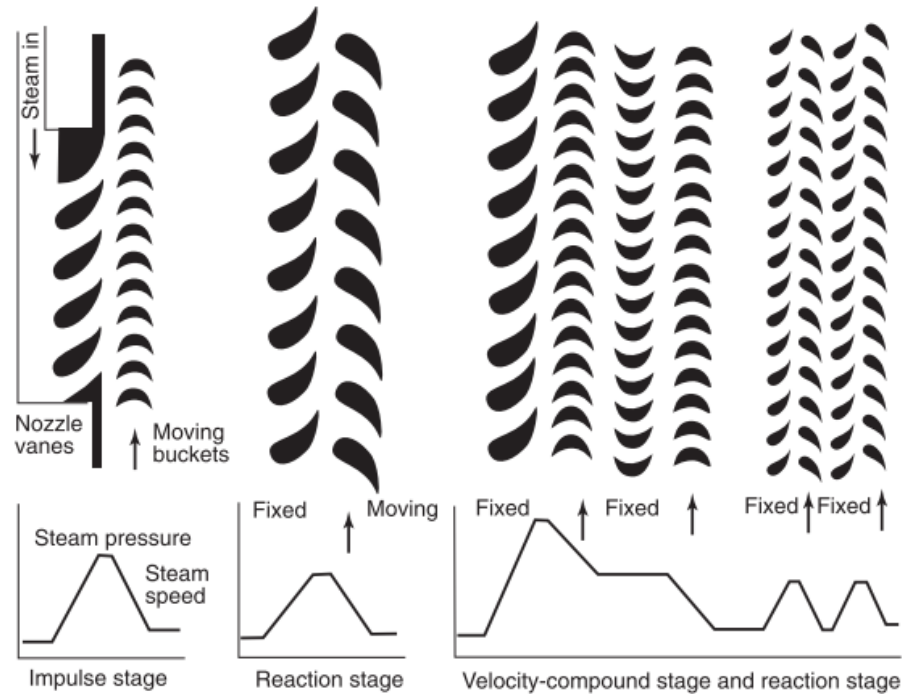


Figure 2.13: Simple impulse and reaction turbine stages [29]

If the enthalpy of the working fluid is known, the power generated by the turbine can be estimated using an equation similar to the heat exchanger equation, Equation 2.6. \dot{W}_s is the shaft work on the turbine and η is the isentropic efficiency of the turbine. The operation of the turbine is not reversible and therefore there are thermal losses of the fluid in the turbine during the expansion. The efficiency is generally described as the total work in divided by the total work out.

$$\dot{W}_s = \eta \dot{m} \Delta H \quad (2.6)$$

If the enthalpy of the fluid is unknown, the work generated by the turbine can be estimated using the first and second laws of thermodynamics. This approximation is not as straight forward as for the heat exchanger since we are going through a pressure change operation. Four assumptions are used to generate the approximation: steady state operation, steam is treated as an ideal gas, there is no heat transfer, and there is compressible work. The first law of thermodynamics states that energy can neither be created nor destroyed in an isolated system. The second law states that the entropy of an isolated

system will always increase. These are represented mathematically in the following equations:

$$\delta\dot{Q} = dH + \delta\dot{W} \quad (2.7)$$

$$\delta q < T ds \quad (2.8)$$

For an ideal gas:

$$c_p = \frac{dh}{dT}, \rho = \frac{P}{RT}, R = c_p - c_v \quad (2.9)$$

Combining these equations and assuming that the system is reversible and adiabatic gives:

$$\left(\frac{T}{T_{in}}\right) = \left(\frac{P}{P_{in}}\right)^{\frac{\gamma-1}{\gamma}}, \gamma = \frac{c_p}{c_v} \text{ and } \delta w = -\frac{dP}{\rho} \quad (2.10)$$

$$w_{ideal} = \int_{P_{in}}^{P_{out}} -RT_{in} \left(\frac{P}{P_{in}}\right)^{\frac{\gamma-2}{\gamma}} dP = RT_{in} \left(\frac{\gamma}{\gamma-1}\right) \left[1 - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{\gamma-1}{\gamma}}\right] \quad (2.11)$$

Equation 2.11 is the approximation for the change in enthalpy of the fluid as it is expanded through the turbine, which can be substituted into Equation 2.6.

When operating a turbine, the inlet should be superheated steam or saturated steam and the exiting stream must have a steam quality above 85% as minimum limit [29]. Moisture in the steam is extremely erosive at the high rotational speeds of the turbine [29]. The later stages in a turbine may be designed to handle a two-phase mixture as an inlet [14]. This is most common in the condensing turbine which exhausts steam at less than atmospheric pressure, shown in Figure 2.15. This is typically the LP turbine in the power plant. The HP turbine can be coupled with the IP turbine as a reheat steam turbine. In this case the HP turbine exhaust is returned to the boiler or steam generator to be reheated, as discussed above, and then returned to the IP turbine for further expansion [29]. Figure 2.14 shows several different configurations of turbine stages and generators based on capacity of the power plant.

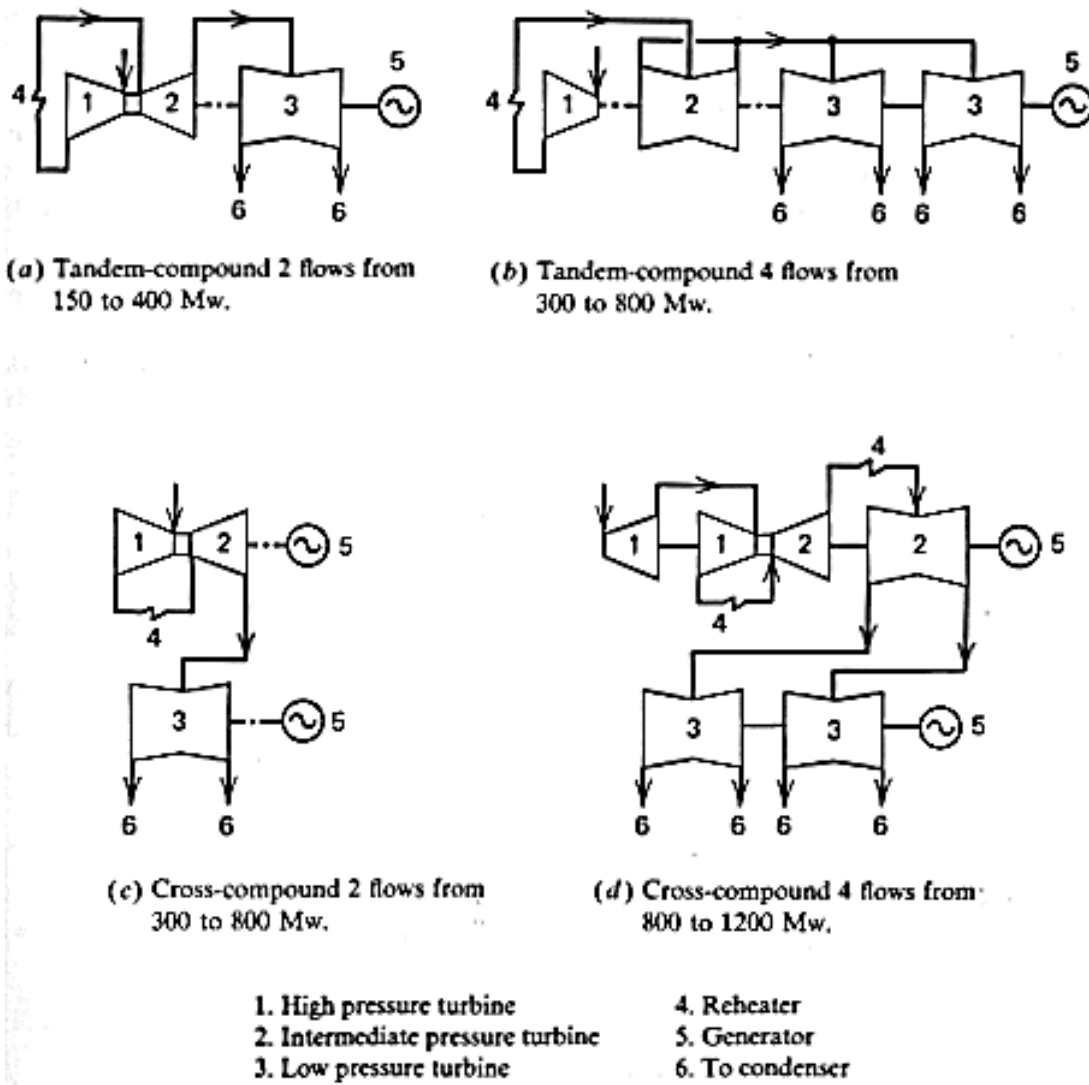


Figure 2.14: Examples of different turbine stage arrangements and shaft configurations [16]

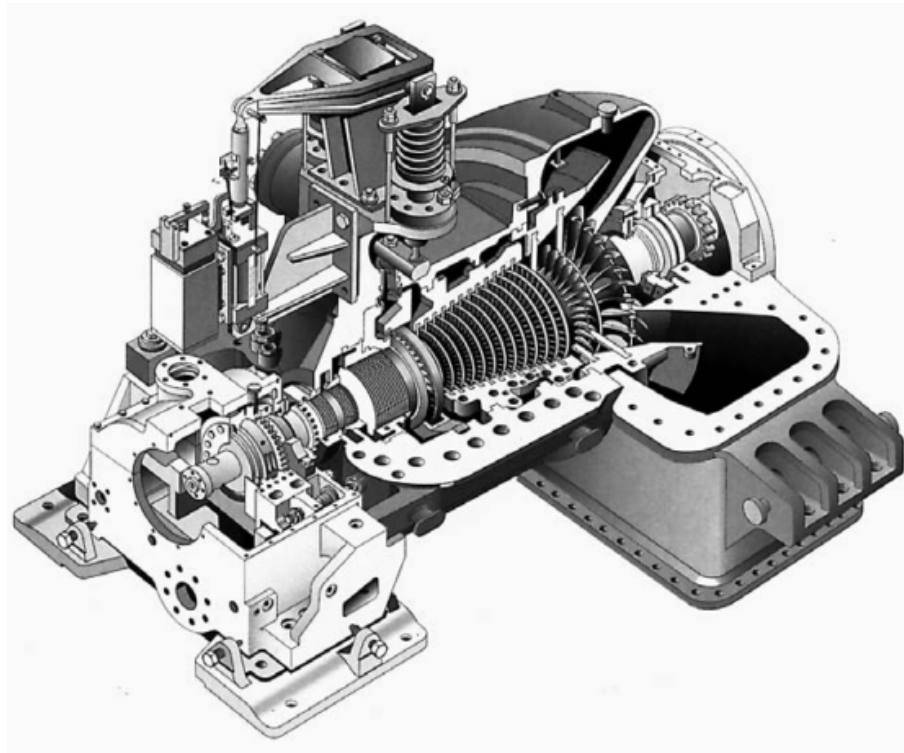


Figure 2.15: Condensing steam turbine for approximately 65-MW output [29]

2.2.3 TURBO GENERATORS

The mechanical work generated by the steam turbines is used to generate electricity through a turbo electric generator. There are several different configurations for steam turbines, shown in figure 2.14, and in all of these configurations there are several turbine stages on the same shaft as the generator [16]. The most common turbo generator is the hydrogen cooled generator which uses hydrogen gas to cool parts of the generator. The main source of heat is in the production of electricity through the conductors [30]. Hydrogen is advantageous to air due to the better thermophysical properties of the gas. Once hydrogen could be isolated correctly, it was determined to be the best option for cooling generators [30].

Most turbo generators produce alternating current electricity at 50 Hz or 60 Hz [31]. This requires the rotational speed of the generator to be at 3000 rpm or 3600 rpm. In the United States, turbo generators use drive shafts that are spinning at 3600 rpm or 1800 rpm which results in 60 Hz electricity [16]. Gear reducers are used in the case of 1800 rpm. The typical mechanical efficiency of the a hydrogen cooled generator is from 98.5 to 99% [16].

The generator works by electromagnetic induction using the magnetism to make electricity [29]. The spinning shaft spins the coil contained in the rotor of the generator between the poles of a magnet. As the rotor passes through the magnetic force lines, an electric current is generated which flows through

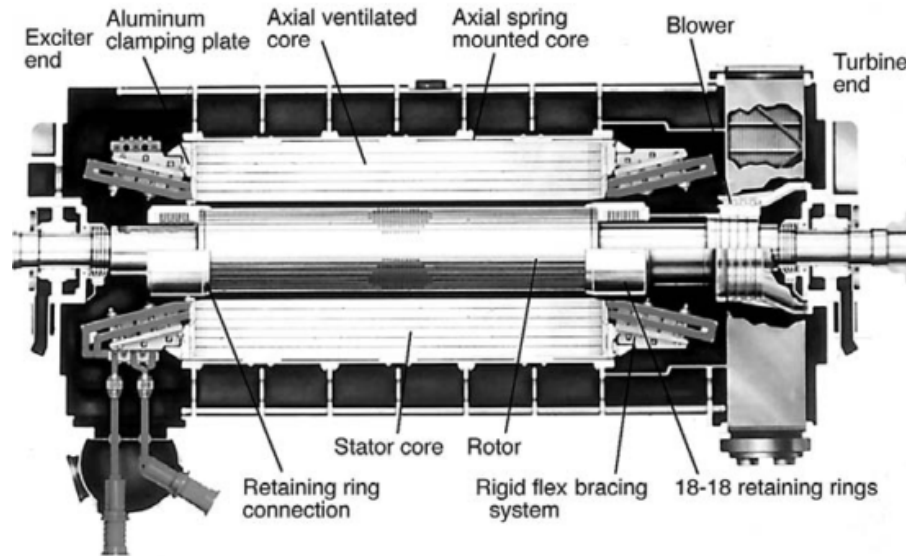


Figure 2.16: Cross-section of a typical hydrogen cooled electric generator [29]

the coil and towards the transformer for transmission [29]. Each generator contains the following major components: frame, stator core and winding, rotor and winding, bearings, and cooling system. A cross section of a hydrogen cooled generator is shown in Figure 2.16.

2.2.4 PUMPS

The pumps are used in the system to bring the condensate and feed water back to the steam generator pressure. Pumps are used in different areas throughout the feed water heating or regeneration process. The condensate pump is used immediately following the condenser and generally operates at standard pressure head and flow capacity. The boiler pump is the final pump to bring the feed water back to the steam generator pressure and usually has a high pressure rise over a standard capacity [16].

Pumps are typically divided into two major categories: dynamic and displacement pumps [29]. A dynamic pump adds power continuously to the fluid and a displacement pump adds the power periodically as needed. Each category envelopes several different pump designs, many of these are frequently used in steam power plants for various applications [29]. These include, injector or jet pumps, reciprocating, rotary, and centrifugal pumps. Each pump has its advantage in specific applications [29]. A simple diagram of a centrifugal pump is shown in Figure 2.13.

The inlet to a pump must be a subcooled liquid. It is important to know the design of the pump to ensure that the absolute inlet pressure is higher than the net positive suction head as defined by the pump performance curve [11]. As the liquid enters the pump it goes through a nozzle which causes a slight pressure drop in the liquid. If that pressure drop brings the liquid pressure below the vapor pressure it will cause vapor bubbles to form which can damage the pump blades, this is called cavitation [11].

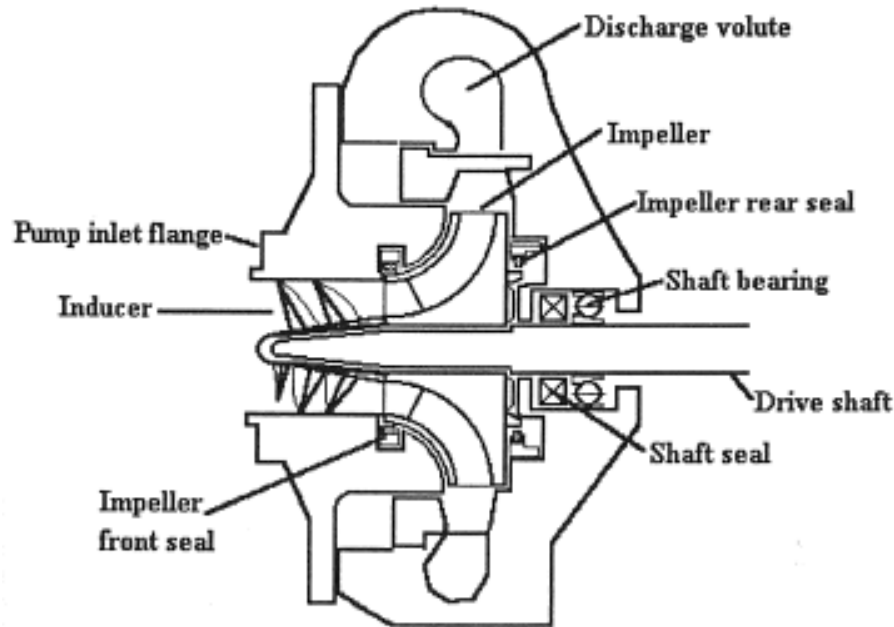


Figure 2.17: Simplified schematic of a centrifugal pump stage [32]

The operating equation for the pump is that same as for a turbine shown in Equation 2.6; however, since liquids are incompressible the approximation for the change in enthalpy is much simpler. Also the sign of the shaft work is different since a pump consumes mechanical energy and a turbine produces mechanical energy. The change in enthalpy for a pump must be divided by the efficiency since there are losses as the mechanical work of the pump is working on the fluid.

$$\delta w = -v dP \quad (2.12)$$

$$w_{ideal} = -v \int_{P_{in}}^{P_{out}} dP = -v(P_{out} - P_{in}) \quad (2.13)$$

2.2.5 FEED WATER HEATERS

Feed water heaters are heat exchangers and function thermodynamically the same as other heat exchangers. These are separated from the heat exchanger section to distinguish the added importance of these in the power cycle for increased efficiency. These are used in the regeneration process to preheat the feed water prior to entering the steam generator. Steam is pulled from different turbine stages and used to heat the feed water [14]. There are two types of feed water heaters: surface heaters and contact heaters.

In the surface heaters, the feed water and the extracted steam are unmixed and the latent heat of

the steam is utilized to heat up the feed water. The feed water is on the tube side of the heat exchanger and the steam enters at the top and condenses on the tubes, much like the condenser. The condensate is typically collected at the bottom in a drain and can either be pumped back into the feed water or be used in other surface heaters by cascading the drain water and mixing it with extracted steam at a lower pressure [16]. These two surface heater system designs are shown in Figure 2.18 below.

The cascade is driven by the pressure differences in each of the surface heaters since the steam is extracted from a lower pressure turbine stage. Although these surface heaters are heat exchangers, they can also be described based on the temperature differences below known as the Terminal Temperature Difference (TTD) and the Drain Cooler Approach (DCA) [16].

$$TTD = T_{sat} - T_{out} \quad (2.14)$$

$$DCA = T_{drain} - T_{in} \quad (2.15)$$

The appropriate values for the TTD range from -3°F to 10°F (-1.67°C to 5.56°C) and for the DCA values range from 10°F to 20°F (5.56°C to 11.11°F) [16]. The ranges give small approach temperatures on both sides of the heat exchanger; this is possible without too large of heat exchanger because the heat transfer is causing condensation in the steam which supplies a much larger heat transfer coefficient. The range on the TTD allows for the outlet feedwater temperature to be higher than the saturated steam temperature. This can happen if the extracted steam is still superheated, and the actual approach temperature remains at or above 10°F (5.56°C).

The contact heater is typically referred to as the deaerator [16]. In the contact heater the feed water is mixed with the extracted steam as well as the cascading stream from any higher pressure surface heaters. The end result on heat transfer is the thermal equilibrium of the mixture. This is modeled through a simple mass and energy balance on the inlet and outlet streams. The deaerator is used to remove any air from the system in the case that there is any remaining after the condenser. Air is driven out of the

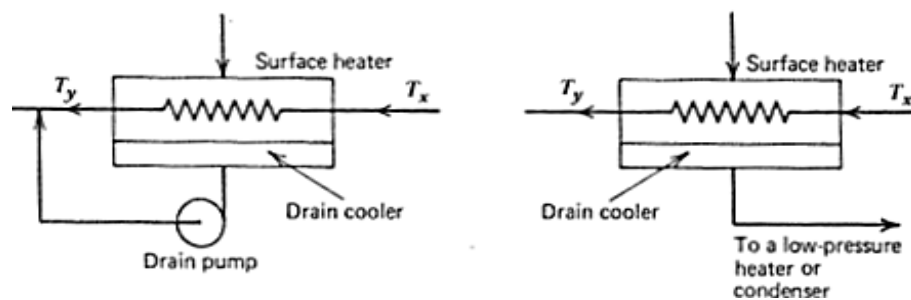


Figure 2.18: Diagram of different surface heater types [16]

deaerator by a positive pressure differential pump instead of the jet ejector required in the condenser. The desired condition of mixture is a saturated liquid or slightly subcooled since it will be pumped out of the container as described in the previous section.

2.2.6 COOLING TOWERS

In the case of a condenser which does not use a once-through cooling process, a cooling tower is required to expel the excess heat. Cooling towers are used in many industries, but one of the largest of these is electricity generation. For a 2000 MW power plant, cooling water at more than $60 \text{ m}^3/\text{sec}$ is required for correct condenser operation [33]. There are multiple classifications of cooling towers that range from a simple heat exchanger to a natural draft contact cooling tower. The main function of the cooling tower is heat rejection to the environment through heat transfer with the air.

There are both dry and wet cooling towers. A dry cooling tower has the water and air separated or unmixed through the cooling process and a wet tower has the water and air come in contact within the cooling tower. Overall, the wet cooling tower is more common among all applications [33]. The cooling tower can also be classified as natural draft or mechanical draft. A natural draft cooling tower requires a tall structure to make use of natural flow. As the air is warmed up in the cooling tower, the natural buoyancy or the chimney effect causes the air to flow up the tower and out the top which in turn pulls air into the tower at the bottom [16]. The mechanical draft tower does not need to be as tall because a mechanically driven fan either pushes or pulls air through the tower depending on the location of the fan [16]. In both cases louvers are used to direct the air as it enters the bottom of the tower in order to produce an even distribution of air through the tower cross-section for even cooling [16]. There has also been work on hybrid towers which mix wet and dry tower designs and natural and mechanical draft designs to produce the most efficient system.

The flow direction of the air is always either vertically up the tower, or horizontally across the tower in the case of smaller applications [33]. In the wet tower the water always flows down over the packing which provides even distribution of the water over the air flowing upwards. In the dry tower the packing is replaced with the heat exchanger portion of the cooling tower. In either case the performance of the packing is the most crucial part of the cooling tower operation because the processes that take place in the packing define the cooling that the tower is capable of providing [33].

Natural draft cooling towers, large concrete hyperboloid structures, are typically correlated with NPPs; however, the type of cooling tower used at a NPP depends on the location and the environmental conditions. The Palo Verde Generation Station in Arizona uses mechanical draft towers in their cooling system and the Diablo Canyon Generating Station in California uses a once-through cooling system with ocean water [13, 34]. In each case a choice was made as to what system would be most effective based on available cooling sources.

In a wet cooling tower design water is removed from the system through evaporation in the cooling tower, and then replaced by the water source. It is estimated that about 3-4 liters of water are required to be replaced, due to evaporation and cooling water purity, for every kilowatt-hour generated by the power plant [33]. The relative humidity of the available air plays an important role in this evaporation process. To ensure the purity of the cooling water, the system also undergoes blow down which is a process in which waste water containing built up minerals is released back to the source continually [35].

The cooling tower of a power plant is important to the overall function of the plant in many ways. Because the tower is the plant's thermal connection to the environment it is important to design the correct cooling tower for the location of the plant. Additionally, if the cooling water cannot fully condense the steam in the plant, the system cannot function correctly.

2.3 ASPEN HYSYS

The Rankine cycle is modeled using the Aspen HYSYS software. Aspen HYSYS is a thermal fluids modeling software within the Aspen ONE software set. HYSYS is the energy industry's leading process simulation software with applications in upstream production, gas treating and processing, refining and safety analyses. The simulator contains multiple unit operations and user tools to obtain optimal design. The unit operations include pumps, compressors, turbines, valves, pipes, chemical reactors, chemical separators, phase separators, and other useful process modeling modules. The user tools include, optimization, case studies, energy efficiency optimization and economics analyses [36].

2.3.1 UNIT OPERATIONS

The unit operations discussed in the previous section are mostly defaults built into HYSYS with the exception of the cooling tower, which will be explained in detail in Chapter 3. The equations used by HYSYS to calculate the results for these operations are explained here. In most cases these are either the same or similar to the equations developed above. All of the information comes from the operations guide to Aspen HYSYS [37].

Heat Exchangers The different heat exchanger modules used for this model are the shell and tube heat exchanger module and the heater modules. These all function according to Equation 2.3 where the enthalpy for the inlet and outlet conditions are calculated according the fluid package specified for the model. In the case of the heater module, Q is specified and in the case of the shell and tube module, Q is equal to the opposite fluid conditions according to Equation 2.16.

$$\dot{m}_{hot}(H_{in} - H_{out})_{hot} = \dot{m}_{cold}(H_{out} - H_{in})_{cold} \quad (2.16)$$

If the design of the heat exchanger is explored, a correction factor is added to Equation 2.4 to specify a

different flow configuration in the heat exchanger.

Turbines The calculation of the turbine work is the most different between HYSYS and previously shown. That being said, the equation is similar in most aspects to Equation 2.6, and is as follows:

$$W_{ideal} = F_1(MW) \left(\frac{n}{n-1} \right) CF \left(\frac{P_1}{\rho_1} \right) \left[\left(\frac{P_2}{P_1} \right)^{\left(\frac{n-1}{n} \right)} \right] \quad (2.17)$$

where:

F_1 = molar flow rate of the inlet stream

MW = molecular weight of the gas

n = volume exponent

CF = correction factor

P_1 = pressure of the inlet stream

P_2 = pressure of the exit stream

ρ_1 = density of the inlet stream

The equations for the volume exponent and the correction factor are:

$$n = \frac{\ln\left(\frac{P_2}{P_1}\right)}{\ln\left(\frac{\rho_2'}{\rho_1}\right)} \quad (2.18)$$

$$CF = \frac{h_2' - h_1}{\left(\frac{n}{n-1}\right) \left(\frac{P_2}{\rho_2'} - \frac{P_1}{\rho_1}\right)} \quad (2.19)$$

The inlet stream is flashed isentropically to the outlet pressure to find the values for ρ_2' and h_2' .

Equation 2.17 works for vapor expansion and compression processes in HYSYS and represents the ideal work of expansion and compression. The result is then multiplied by the adiabatic efficiency (or divided in the case of compression), to find the actual work. The remaining work goes towards the increase in the outlet temperature of the unit. The main difference between Equation 2.17 and Equation 2.6 is the use of the volume exponent in the place of the heat capacity ratio.

Turbo Generators HYSYS has no module to represent a turbo generator. Additionally, the turbo generator efficiency is extraordinarily high. Therefore, the operating efficiency of the generator is neglected in the analysis and the total mechanical power produced by the turbines is used when determining the thermal efficiency of the system.

Pumps The equation to calculate pump power is identical to Equation 2.13 using the density of the fluid in the place of the specific volume (fluid density is the inverse of specific volume).

$$Power_{ideal} = \frac{(P_{out} - P_{in})Flow}{Density} \quad (2.20)$$

2.3.2 CUSTOMIZATION

In addition to typical unit operations and user tools, HYSYS also allows for customization in the simulations. These options allow the user to push the limits of what processes HYSYS is capable of simulating and make it more suitable for specific applications. The customization options for HYSYS include embedded spreadsheets, user variables, custom unit operations, property packages, and kinetic reactions. Embedded spreadsheets and user variables contribute to the automation of HYSYS which allows a process to be programmatically run according to the user specifications. The custom unit operations and other packages become a part of the simulation as built in objects to extend the use of HYSYS beyond the basic modules. The information for all options except the embedded spreadsheet comes from the HYSYS Customization Guide [38]. The information for the embedded spreadsheet is included in the Operations Guide [37].

Embedded Spreadsheets The spreadsheet is an included HYSYS module which applies the functionality of spreadsheet programs to the flow sheet modeling. The spreadsheet is capable of importing and exporting process variables from the flow sheet and has access to almost all possible variables. Custom calculations may be performed on flow sheet variables to calculate any number of desired variables. For example, the dynamic pressure drop of a heat exchanger may be calculated within the spreadsheet since the specified pressure drop within the heat exchanger operation remains constant regardless of flow [37]. Complex mathematical formulas and other arithmetic, logarithmic, and trigonometric functions can be used in the spreadsheet. There is also the option for logical programming such as boolean logic [37].

User Variables User variables help increase overall functionality of a HYSYS flow sheet. These can interact automatically with streams and unit operations to create additional user specified variables which are indistinguishable from standard HYSYS variables [38]. A basic example of a user variable is the dew point temperature. This can be added to all material streams and could be used in additional logical operations, such as spreadsheets, for a temperature specification based on the dew point such as degrees of superheat.

The user variable is added in process streams and unit operations using the user variable page under the worksheet tab for process streams or the user variable tab for unit operations. A user variable can also be added to the entire flow sheet or case by selecting the user variable option on the "Customize"

ribbon [38]. There are multiple options for the type of variable which need to be specified. The subroutine written in Visual Basic can be run either before or after the solver for process streams and global user variables or dynamically within unit operations for a transient model [38]. The user variable can either be enabled or disabled in any stream or unit operation that matches the one it was written in. To interact with this feature, use the green check mark to enable the use of the check box in the user variable tab for each stream or unit operation.

User Unit Operations The final customization option deals with user unit operations built both within HYSYS and externally. The internal user unit operation uses only the Visual Basic coding language with a set template and Graphical User Interface (GUI). The GUI contains the connections for feeds, products, and energy streams, the code for the operation, and the worksheet which provides the inlet and outlet properties. The user unit operation can also have user variables. This operation is accessed by selecting the flow sheet and pressing "F12" to open the UnitOps window. Select User Unit Op under the User Ops category, this will create the module that can be manipulated. The coded portion of the unit operation contains options for three subroutines: Initialize, Execute, and StatusQuery. The Initialize and Execute subroutines bring in the required stream information and perform the unit operations. The Status Query gives the error messages and required information. Since HYSYS operates on a Degrees of Freedom (DOF) basis, the missing information required for the calculation may be displayed in the status box as coded in the Status Query subroutine.

Externally built unit operations can be coded in either Visual Basic or C++. The correct syntax must be employed for each coding language in order for the system to interact with HYSYS correctly. The advantage of an externally built unit operation is the use of a customizable GUI. Using the "viewed" application included in the HYSYS install allows for a GUI to be built more appropriately for the unit operation being created. This is imported into HYSYS together with the code as an extension. All other capabilities are similar between the internal and external user unit operations.

Features Used The model developed for this analysis makes use of the embedded spreadsheet and user variables for automation purposes. Since the Rankine cycle uses mainly basic unit operations there is no need to design any new modules to be used with the model. However, the embedded spreadsheet is used widely for operating set points as well as extra calculations. User variables are also used to automate some otherwise manual inputs and to help ensure physically real results.

2.3.3 SOLVER

HYSYS can function both as a steady state model and a transient model. In the steady state mode, the simulation is constantly solving unless the solver is put on hold. The unit operations in use in the system will use the thermodynamic equations as described above for the different unit operations. The

thermophysical properties are taken from the HYSYS materials database and the selected fluid package for multi-phase interactions.

The progression of modeling would begin with the steady state model for the desired system. Once the steady state system has been designed and solved, the user can begin the transient applications. Using reference points and desired inputs and outputs, each of the unit operations must be sized in order for stream specifications to be eliminated. Performance curves must be added to the turbomachinery and geometries and vessel sizes are needed for tanks and heat exchangers. Once all unit operations are sized and calculating successfully, the model can be placed in transient mode, which is interactive for the user to change conditions as needed and see how the system will respond to the changes. Logic controllers and data recording systems are also available for process control and data analysis during and following any transient run.

The combination of steady state and transient operating modes allows the designer to fully understand how a system will perform under all levels of interaction. This project aims to determine the optimal steady state configuration for a specific Rankine cycle design based on the unit operations involved and the maximum and minimum steam temperatures in the cycle as specified. The logical transition would be to implement the optimal system as a transient model and measure how weather data affects cycle performance and analyze any potential issues or abnormalities during operation that could then have adverse affects on the nuclear reactor coupled to the power production system.

CHAPTER 3: METHOD AND PROCEDURE

3.1 MODEL OVERVIEW

The project is carried out through modeling the complete Rankine cycle and implementing design parameters and assumptions directly into the simulation to allow for the optimization of the Rankine cycle design. The Rankine cycle modeled in this analysis is a superheated steam Rankine cycle with reheat. There is 1 HP turbine stage, 2 IP turbine stages, and 5 LP turbine stages, all in series. The reheater is between the HP turbine and the first IP turbine stage. Steam or a water-steam mixture is pulled from each of the first 7 turbine stages for use in the feed water heating system. There are 2 feed water trains each with 3 heaters, and a deaerator connecting the feed water trains. Three pumps are used to pump the condensate back to the superheating pressure. The condenser is a water cooled condenser connected to a forced draft cooling tower. Figure 3.1 shows the diagram of the cycle modeled. A natural draft cooling tower is shown in the diagram in the place of the forced draft cooling tower. Figure 3.2 shows the T-S diagram for the full cycle in Figure 3.1.

The base Rankine cycle used for designing the model is attached to a 100 MW reactor, with the steam generator contributing 90.13 MW and the reheater the remaining 9.87 MW. The turbine stages and pumps with pressure ratios and efficiencies are detailed in Table 3.1. In Figure 3.1, the pumps are in order from the condenser to the steam generator according to Table 3.1: the condensate pump follows the condenser, the booster pump follows the deaerator, and the boiler pump follows the fourth surface feed water heater. Feed water heaters are numbered starting with number one at the condensate pump and number six just before the steam generator.

The solid red lines in Figure 3.2 show the main portion of the cycle through all of the feed water heaters, pumps, turbines, and other heat exchangers. The dotted lines represented the use of the feed water heaters in the regeneration process for the preheating. The main heating in the steam generator and reheater is carried out from the highest temperature dotted line (representing feed water heater six) to the first peak in the superheated steam region for the steam generator, and the second peak for the reheater.

3.2 HYSYS MODEL

The purpose of the HYSYS model is to allow for cycle optimization by automating the physical restraints on the calculation to force the model to calculate real results from two or three main inputs. The main inputs into the system are the high steam temperature and the cooling water availability temperature (which is a function of ambient conditions). An additional main input may be the steam generator operating pressure. In the case of two inputs, the pressure is solved for by the system. Adjusting

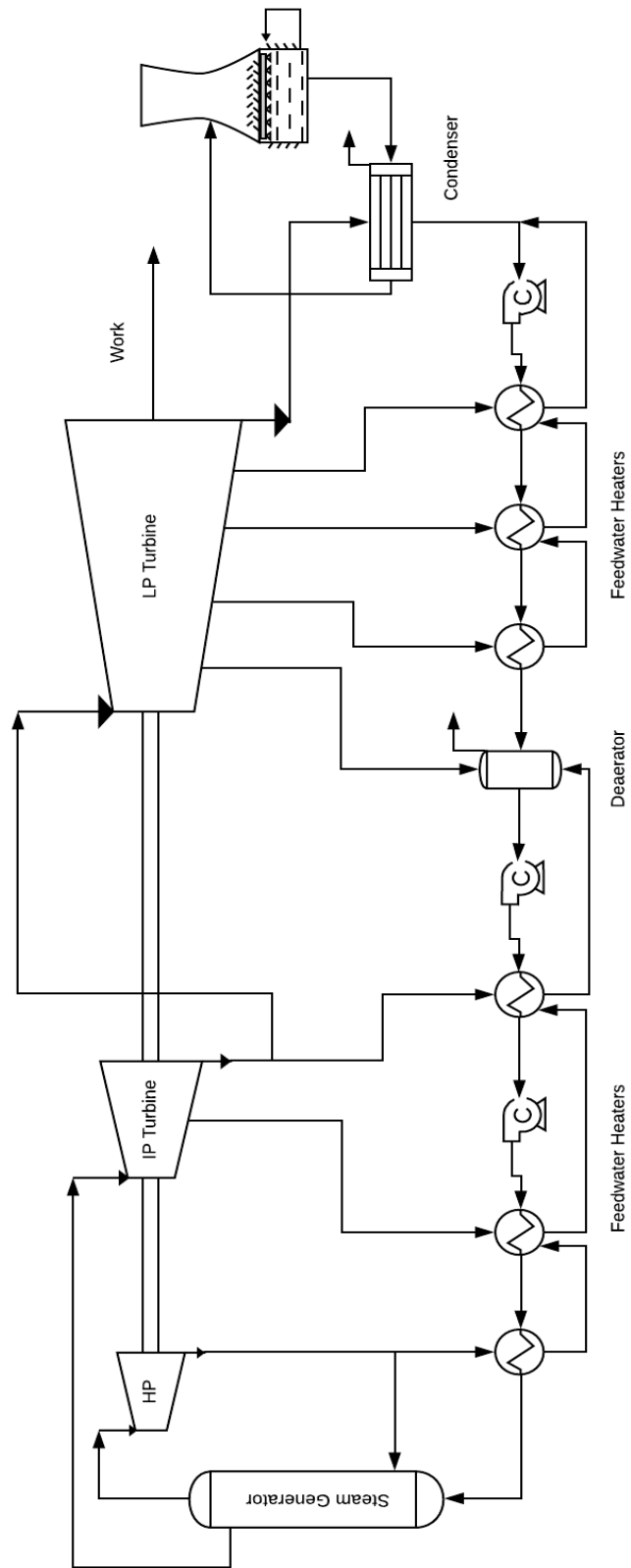


Figure 3.1: Diagram of full Rankine cycle model

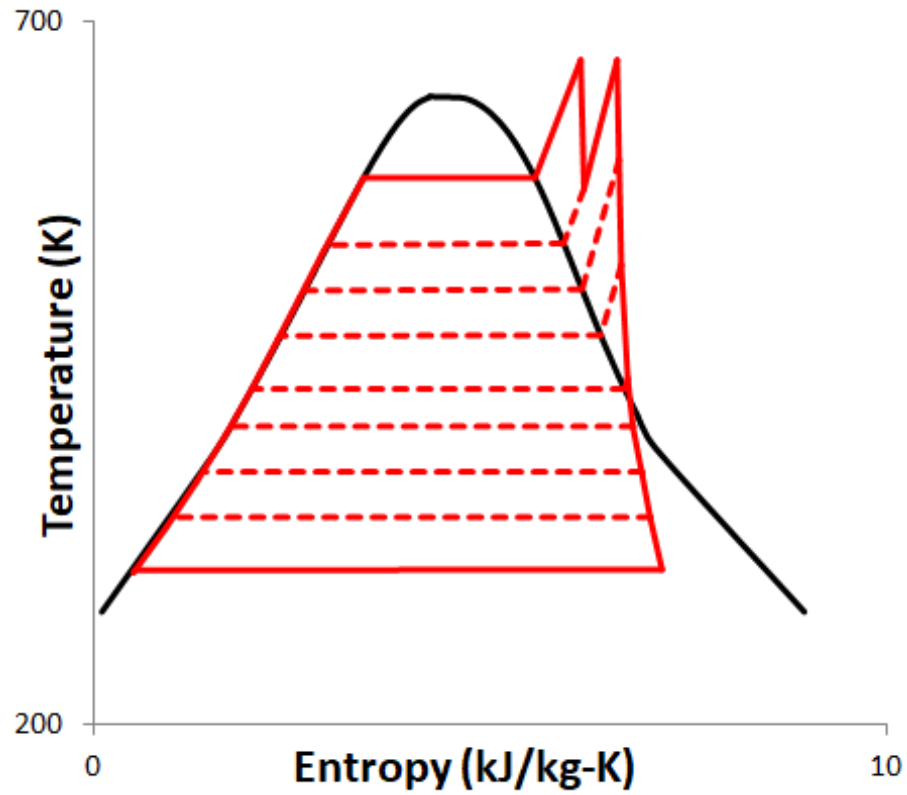


Figure 3.2: Temperature-Entropy diagram of the entire Rankine cycle to be modeled

Table 3.1: Base Model Turbine Stages with Pressure Ratios and Efficiencies

Turbine Stage	Pressure Ratio	Efficiency
HP Turbine	0.5167	85%
IP Turbine Stage 1	0.6451	90%
IP Turbine Stage 2	0.5697	90%
LP Turbine Stage 1	0.4430	80%
LP Turbine Stage 2	0.5005	80%
LP Turbine Stage 3	0.3817	75%
LP Turbine Stage 4	0.3050	80%
LP Turbine Stage 5	0.2055	80%
Condensate Pump	90.55	75%
Booster Pump	3.710	75%
Boiler Pump	3.545	75%

a few aspects of the model allow the pressure to be specified. Optional inputs into the model include the outlet pressures at each turbine stage and the desired TTD and DCA temperature differences; however, the model contains defaults for these values.

In this case plain water is used with the National Bureau of Standards (NBS) steam fluid package. The cooling tower uses a mixture of nitrogen, oxygen, and water with phase properties calculated by the Peng-Robinson fluid package. For plain water, the steam tables provided by NBS are considered to be highly accurate and are approved by the Office of Standard Reference Data [39]. The Peng-Robinson Equation of State is used for the air-water mixture in the cooling tower because this equation of state is one of the most widely used and accepted equations for thermodynamic calculations of phase equilibrium [40].

3.2.1 THERMAL INPUT

The original intent of the modeled Rankine cycle was to provide different system parameters when coupled to a PWR as a heat source. The thermal input to the model is generated using two heater unit operations where the thermal load is simply specified or calculated based on the mass flow rate of water and the desired change in enthalpy. The model uses a separate heater for the steam generator and the reheater where the total thermal input sums to be 100 MW. Since the model uses thermodynamic equations to solve all unit operations in steady state mode, an increase in the thermal input would simply result in an increase in the system mass flow rate and all other values would remain the same, including the thermal efficiency.

The thermal input of the model determines the total mass flow rate of the system. The steam generator is between the final feed water heater and the HP turbine and the reheater is between the HP turbine and the first stage of the IP turbine. The hot steam temperature is specified at the outlet of the steam generator and carried to the outlet of the reheater for consistency. The thermal input of the steam generator is adjusted so that the sum of the two heater unit operations is equal to the reactor thermal power under consideration. Since the thermal input is modeled using only heater unit operations, the source could be a heat source other than a nuclear reactor; however, for this thesis it is appropriate to assume the heat source is a nuclear reactor.

3.2.2 FEED WATER HEATERS

The feed water heaters are complex operations to balance in the Rankine cycle. The steam inlet conditions are dependent on the turbine outlet pressure, the feed water inlet conditions are based on another feed water heater, and the entire energy balance between the streams is dependent on the amount of steam bled from the corresponding turbine stage. Therefore, any adjustment to these values for any one feed water heater will affect the balance of the rest of the system.

It is necessary that the performance of each heater remain physically possible during the automated calculation. Since heat is transferred from the hot stream to the cold stream, the cold stream cannot be at a higher temperature than the hot stream at any point in the feed water heater. However, the simulation allows this to happen at times and it is known as temperature cross. This is particularly important during phase change heat transfer. An example of temperature cross in a heat exchanger operation is shown in Figure 3.3. The inlet and outlet temperatures of each stream may not immediately show that temperature cross has occurred in the heat exchanger, but boiling and condensation are constant temperature processes and more attention is required. A solution to this phenomenon is shown in Figure 3.4, where the pressure of the steam has been adjusted to increase the condensing temperature and prevent temperature cross.

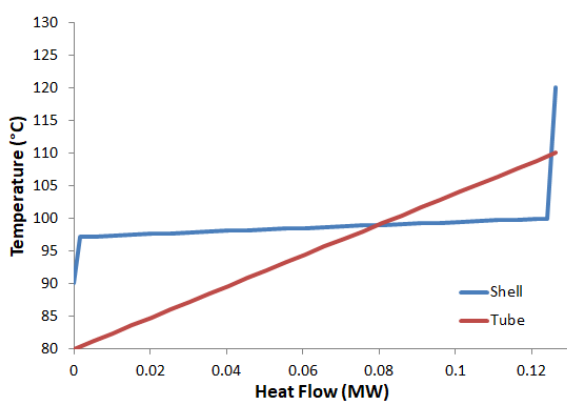


Figure 3.3: Heat exchanger with temperature cross

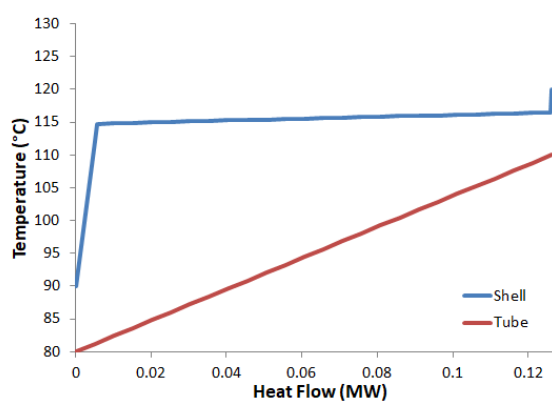


Figure 3.4: Heat exchanger without temperature cross

The model provided for the project specifies the mass flow of steam or two-phase water to be diverted from each turbine stage to the corresponding feed water heater. This method requires significant manual manipulation of multiple different factors within the model, including, turbine stage pressure, feed water temperatures in and out, and drain temperature. Each train must be solved individually as well since they are typically treated as one system.

The TTD and DCA methods are used in the upgraded model for each of the surface feed water heaters. The outlet of the deaerator is set as a saturated liquid at the corresponding turbine stage pressure. The DCA and the TTD values are all set at 5.556°C (10°F) resulting in an optimal temperature difference on both sides of the heater. Creating a methodology that will force the feed water heater to have the specified TTD and DCA as the approach temperatures on each side ensures that the physics of the system are maintained without further manipulation. Since the outlet feed water temperature is specified based on the saturated steam temperature, temperature cross such as shown in Figure 3.3 is impossible.

These specifications for the surface heaters are implemented in HYSYS through the use of an embedded spreadsheet for each surface heater in the simulation. Starting with the last feed water heater, the trains

can be solved systematically down the line from the steam generator to the condenser. The mass flow rate of the feed water in the high temperature train is the total mass flow rate of the system. The feed water outlet temperature of the first heater is automatically known from the conditions of the HP turbine and the specified TTD. Likewise, the feed water inlet temperature is known from the first IP turbine stage and the TTD. Using the feed water inlet temperature, the drain temperature is determined using the DCA. Using this methodology within HYSYS will result in only one DOF for the heat exchanger unit operation calculation, the diverted steam mass flow rate.

This method will work for the highest temperature heater in each of the two trains since there is one inlet and one outlet on each side of the heat exchanger. When the drain begins to cascade to the next heater HYSYS is unable to perform this calculation on its own due to the required mixing of the drain cascade stream and the steam or two-phase stream prior to entering the heat exchanger. In this case, HYSYS will not solve for the steam flow coming from the turbine because that is not technically an inlet to the heat exchanger. Additionally, since the mixing of the drain cascade and the steam will most likely result in a two-phase mixture, the mixing unit operation is unable to return the information from the heater since the enthalpy is not based on only temperature and pressure, but the steam quality of the inlet. For these heat exchangers, the steam flow rate can be determined by solving the heat exchanger energy balance,

$$\dot{m}_{steam}h_{steam} + \dot{m}_{cascade}h_{cascade} = Duty + \dot{m}_{drain}h_{drain} \quad (3.1)$$

for the steam mass flow rate,

$$\dot{m}_{steam} = \frac{Duty + \dot{m}_{cascade}(h_{drain} - h_{cascade})}{h_{steam} - h_{drain}} \quad (3.2)$$

where the duty is simply calculated from the feedwater side of the exchanger.

$$Duty = \dot{m}_{FW}(h_{FW_{out}} - h_{FW_{in}}) \quad (3.3)$$

The values are brought into a spreadsheet in order to perform the calculation. The resulting mass flow rate for steam is exported to the diverted steam stream. However, in order for this to work and for HYSYS to not think that the heat exchanger is over specified, a "dummy" stream must carry the temperature and the pressure of the heat exchanger drain, otherwise the entire exchanger has been specified and HYSYS won't solve anything. By creating the "dummy" stream to carry the enthalpy for the drain that we need to solve the energy balance, HYSYS is tricked into believing that it is solving the balance by solving for the drain temperature, and the solver completes its task. In reality, the temperature was known and the spreadsheet energy balance is used to force HYSYS to give the already specified temperature result in

exchange for the steam flow rate from the turbine stage.

The deaerator between the two trains of surface heaters is a contact heater. All of the streams mix at this location. The deaerator is modeled using a stream mixer unit operation where all of the inlet and outlet streams are at the same pressure. The outlet of the deaerator is specified as a saturated liquid as there is usually a pump immediately following this contact heater since the liquid collects in a pool at the bottom of the tank. Using a saturated liquid will always accurately solve the system regardless of pressure. Since all streams are mixed together and there is no required heat exchanger unit operation, HYSYS is able to calculate the required steam flow from the turbine stage given the saturated condition on the outlet. This is accomplished by the mass and energy balance on the mixing operation.

3.2.3 PRESSURE

The pressure limits of the optimal system are determined by the hot steam temperature and the ambient conditions surrounding the plant. The base model had a hot steam temperature of 350°C, which corresponds to a high pressure of around 8 MPa. The ambient conditions of the model gave a cooling water inlet temperature to the condenser of 25°C. The lowest turbine outlet pressure was set as 7.584 kPa which gives a saturated water temperature of 40.13°C. This design gives the condenser room for subcooling of the water before it mixes at the bottom of the cycle with the cascade stream from the lower temperature heater train.

Keeping with this design, the upgraded model bases the lowest turbine stage outlet pressure on the expected ambient conditions of the system. The pressure is set so that the saturated temperature is 12.25°C greater than the inlet cooling water temperature. This allows the average ambient conditions to change and see how that change will affect the overall efficiency of the Rankine cycle. This is accomplished in HYSYS by generating another "dummy" stream to simulate a temperature increase of 12.25°C. The saturated pressure is found by using a vapor fraction specification; the result is communicated to the turbine outlet stream. The 12.25°C set point is based on the information in Table 3.2 [11].

The problem description of the project calls for superheated steam temperatures in a range from 300°C to 600°C. As the hot steam temperature increases, the process pressure must increase as well in order for the system to be able to converge. The steam generator outlet temperature and pressure for some reactor

Table 3.2: Recommended Exhaust Pressure for Various Cooling Water Temperatures [11]

Cooling Water Temperature (°F)	Exhaust Pressure (inHg Abs.)	ΔT (°C)
60	1.0	10.51
70	1.5	12.05
80	2.0	11.72
85	2.5	13.16
90	3.0	13.91
95	3.5	14.18

Table 3.3: Different Reactor Designs' Steam Generator Temperature and Pressure

Reactor Type	Ave Outlet Temperature (°C)	Ave Outlet Pressure (MPa)	Reference
PHWR ¹	260	4.70	[10]
PWR	285	6.89	[10]
BWR	286.1	7.14	[10]
BWR	286.1	7.17	[11]
BWR	296.7	6.21	[11]
BWR	298.3	6.21	[14]
PWR	289.3	8.76	[11]
BWR	315.6	7.24	[14]
PWR	316.4	7.31	[11]
BWR	332.8	8.51	[14]
CRBPR ²	485.0	10.69	[11]
SFBR ³	490.0	17.70	[10]
HTGR	510.0	17.24	[11]
HTGR	513.0	17.30	[10]
AGR ⁴	543.0	17.00	[10]

1. Pressurized Heavy Water Reactor, 2. Clinch River Breeder Reactor, 3. Sodium Fast Breeder Reactor, 4. Advanced Gas Reactor

designs are shown in Table 3.3. These are mostly BWRs and PWRs since these are the reactors currently in operation in the United States. However, some advanced reactor designs are included to show higher temperature operations.

The values in Table 3.3 are plotted in Figures 3.5 and 3.6 with pressure as a function of temperature. The black line is the linear fit and the red lines show the 95% confidence interval of the linear y-intercept. There is a major outlier in Figure 3.5; this is the Clinch River Breeder Reactor Project. This was a demonstration plant which was never constructed [41].

Since the plant was a demonstration plant it would in theory not be operating at the desired pressure; also, since it was never built it can be excluded from the data. Excluding the Clinch River Plant gives the results in Figure 3.6. The error on the y-intercept of the linear fit is significantly less with the exclusion of the Clinch River Reactor. The data show that pressure increases with an increase in temperature. A higher pressure gives a higher thermal efficiency. This pressure should be limited by the maximum interval since the data represent actual reactor designs. However, there is a trade-off between thermal efficiency and capital cost from a higher operating pressure.

Fossil fuel power plants typically operate at a higher pressure than NPPs [14]. This operating pressure is determined by the plant designer, as it is in the NPP. Due to regulations and costs, the NPPs operate at a lower pressure than fossil fuel plants. Overall the losses are balanced due to the inefficiencies of the combustion process compared to the fission process.

The pressure drop across each of the heat exchangers is set to be 2% of the inlet temperature [42, 43]. This includes both fluids in all surface heat exchangers, the steam generator and reheater, and both sides of the condenser. Including an automatic pressure drop specification allows the model to scale

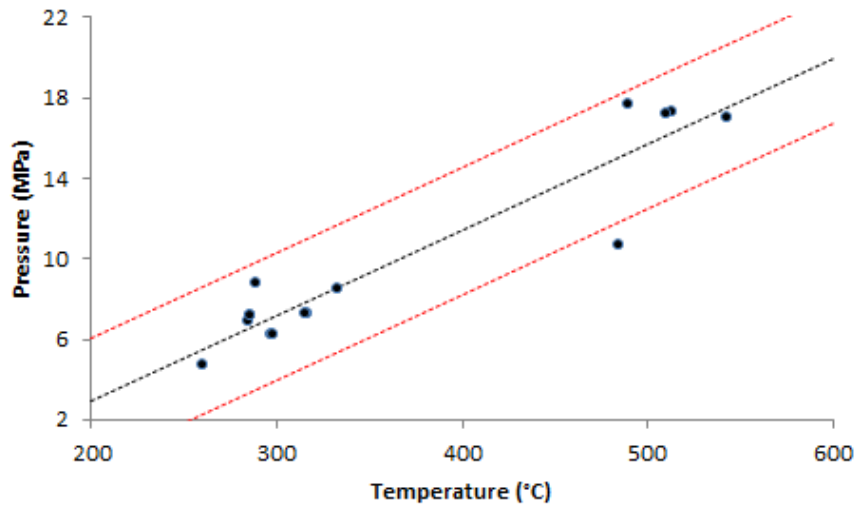


Figure 3.5: Selected steam generator outlet temperatures and pressures

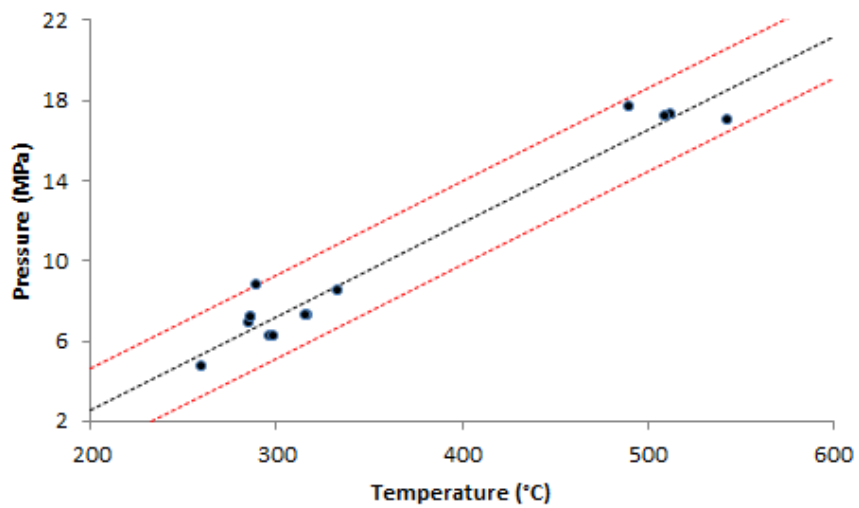


Figure 3.6: Selected steam generator outlet temperatures and pressures (outlier removed)

appropriately and never produce a negative absolute pressure anywhere in the system. No other pressure drops are specified in the system other than the expansion that occurs in the turbine stages. Pressure drop in the heat exchangers will have a minor effect on the thermal efficiency of the system. The losses in thermal efficiency associated with these pressure drops is an increase in pumping power. Overall, feed water pumping power is a minor parasitic loss, making up around 1.5% of the power produced. The reason for this is the change in fluid enthalpy for a liquid is insignificant compared to that of the vapor. Figure 3.7 shows the T-S diagram for water with several isobars. The pressures shown range from 1 kPa to the critical pressure, 22.064 MPa.

For all of the pressures, the temperatures and entropies on the liquid side of the curve are overlapping each other to the point that they cannot be distinguished. For illustration of what this means, the isotherm for 200°C plotted on a Pressure-Enthalpy (P-H) diagram of water is shown in Figure 3.8. This will be true at all temperatures and pressures when both diagrams are considered. This shows that with a large increase in the pressure of an isothermal system the enthalpy does not change. Liquid compression is an isothermal process when compared to vapor compressions. Therefore the pressure drop associated with the heat exchangers in the system has a negligible affect on the overall thermal efficiency of the system.

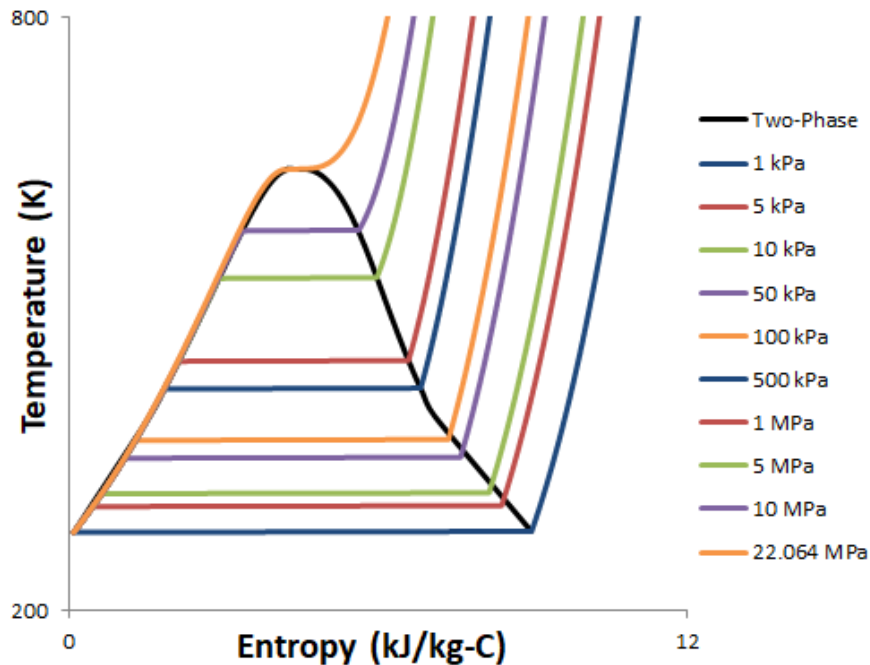


Figure 3.7: Different pressure lines represented on an T-S diagram of water

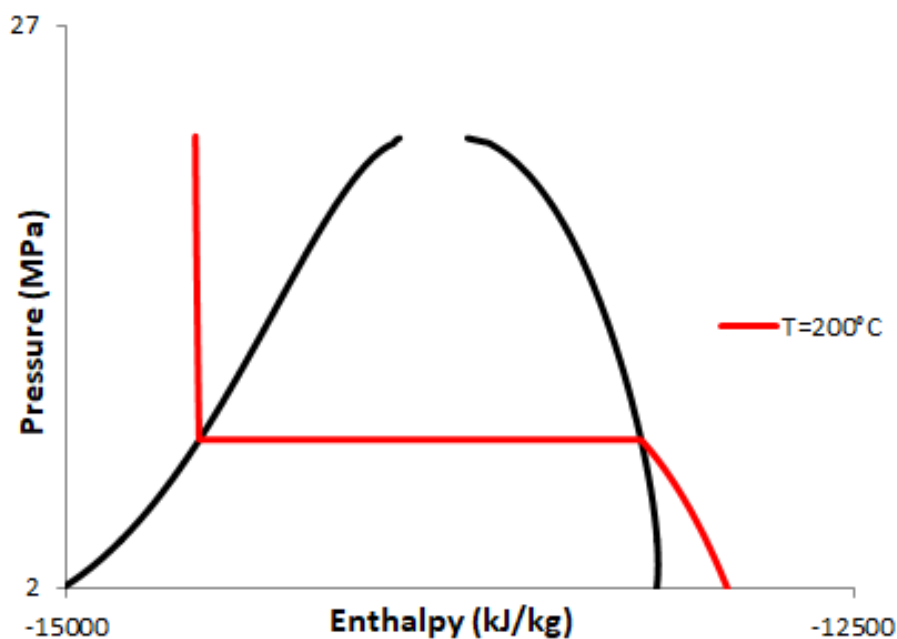


Figure 3.8: Isotherm of 200°C water on a pressure-enthalpy curve

3.2.4 COOLING TOWER

The condensing pressure of the system is determined by the cooling water temperature available which is in turn controlled by the ambient conditions. The condenser is modeled as a heat exchanger and the cooling water is restored through the use of a cooling tower. The cooling tower module used is a forced draft cooling tower since the air flow is generated by a fan. The HYSYS model of the cooling tower represents what is occurring in the cooling tower thermodynamically and therefore does not resemble the physical system. The cooling tower itself is combined into a sub-flow sheet within HYSYS for cleaner look and analysis since it represents a different system entirely. The overall model of the cooling tower is based on work previously done at the Idaho National Laboratory for simulating wet cooling towers [44]. Figure 3.9 shows the diagram of the modeled cooling tower.

The air used in the cooling tower is moist air with input options of air temperature, pressure, and relative humidity. The moist air stream is a mixture of a dry air stream containing 79% N_2 and 21% O_2 by mole and a water vapor stream (100% H_2O). The water content of the air is calculated based on the moisture ratio of the air through the use of an embedded spreadsheet and reference streams. The partial pressure of the water in the air is calculated based on the relative humidity specification and the reference stream containing the saturated water pressure at the air temperature. This is used to calculate the dry air pressure and the moisture ratio.

A moist air mass flow rate is required to define the dry air flow rate and then a mixing unit operation

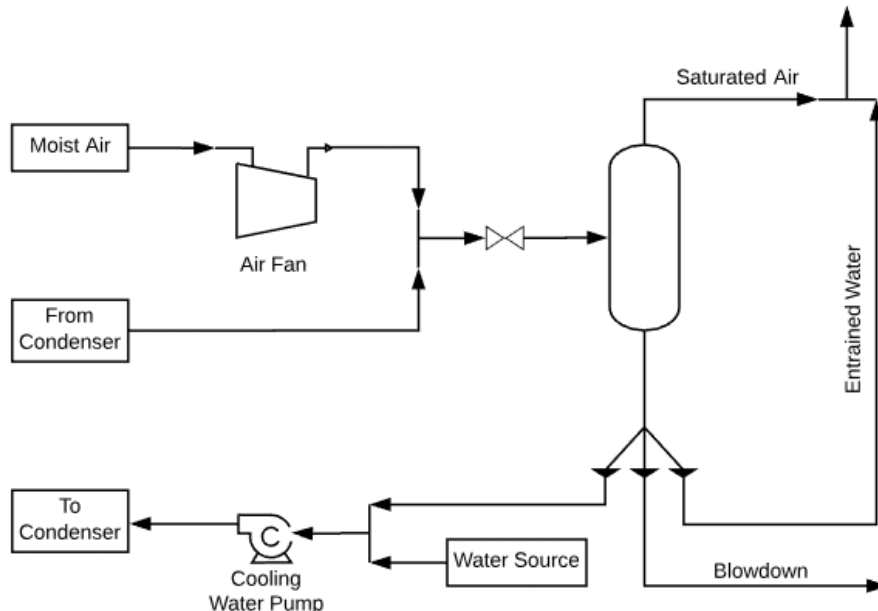


Figure 3.9: Schematic of modeled cooling tower

is used to mix the dry air stream and the water vapor stream. The moist air flow rate is defined and a dry air flow rate is calculated in the spreadsheet. This causes the mixing unit operation to calculate the water vapor required to make up the difference. The outlet of the mixing contains the correct mole fraction of each component in the moist air. As a reference the dew point temperature of the air is calculated using a composition balance and saturated condition input into the balanced stream.

The moist air is brought to a pressure of 101.473 kPa, which is slightly higher than ambient. The cooling water return pressure matches the air pressure following the fan. The cooling water and the air are mixed and then brought back to atmospheric pressure together by flashing through a valve. A phase separator is then used to separate the two-phases generated by the mixing and the flash. Some of the cooling water is vaporized by the flash and leaves the system with the air. There is also water that is leaving the top of the tower entrained in the air. This is assumed to be 0.1% of the total cooling water mass flow rate. Additionally, the blow down, which is the water returned to the source, is assumed to be 20% of the total cooling water mass flow rate [44].

The cooling water which was lost through evaporation, entrainment, and blow down is replaced by the make-up water from the source. The make-up water should be colder than the cooling water inlet to the condenser to bring the water from the tower back to the design temperature. In the case of the base model, the cooling water temperature is 25°C and the make-up water source is at 20°C. Once the make-up water has been mixed with the water from the tower pool, it is pumped backed into the system.

In order for the cooling tower to solve correctly, the air mass flow rate into the tower is adjusted so

that the temperature of the water at the cooling water pump outlet matches the design cooling water temperature for the system. For HYSYS, the adjust operation is explained in a later section.

The parasitic losses involved with the cooling tower are the power required for the fan and the power required for the pump. Typically, the pump power will not vary with a change in ambient conditions. On the other hand, the fan power will vary as ambient conditions change, since fans are rated based on the volumetric flow rate capacity.

3.3 ADDITIONAL LOGIC

3.3.1 RECYCLE

There are still a few logical operations that the model requires in order to be completely solved and capable of iterations. The first is a recycle stream. The recycle unit operation creates a duplicate stream of the inlet with the same specifications. If the model were to break and stop solving this duplicated stream will continue supplying the previously solved result. A benefit of the recycle stream is for generating iterations in the model that will allow for HYSYS to converge on an expected or design result without having to make multiple guesses. When using a recycle stream in HYSYS it is optimal to only have one recycle stream per process or loop. In this case the recycle stream is inserted at the end of the low temperature feed water heater drain cascade between the pressure drop valve and the mixing unit operation which combines it with the condensate flow. This results in only one unknown in the system, the temperature inlet to the condensate pump. By specifying this temperature, the process can solve with the assistance of the recycle stream iterations.

3.3.2 ADJUST

Upon inspection the converged solution for the steam pulled from the turbine stage for feed water heater one does not match the calculated design value from the spreadsheet calculation. An adjust unit operation is required to change the input temperature to the condensate pump so that the feed water heater performs as desired. The temperature is adjusted so that the HYSYS calculated value matches the spreadsheet calculation with no offset. The adjust unit operation requires a tolerance, step size, and maximum iterations. The minimum and maximum adjusted values are optional but in most cases required so that the adjusting does not get out of control. Since we are dealing with the condensate pump, the maximum temperature that we would want in the input is either the saturated temperature or just below so that there is no two-phase flow in the pump. In order for the physical system to make sense, the flow from the condenser is the lower temperature flow and the cascading flow is the higher temperature flow entering the mixing operation. Therefore, the minimum temperature set point should be based on the condenser outlet temperature.

3.3.3 CALCULATION ORDER

This logical change in the system streamlines the convergence of the simulation. HYSYS performs calculations of all unit operations based on the calculation order. The default value for all main flow sheet operations and spreadsheet calculations is 500. The default value for any sub-flow sheet is 2500, and the default for adjust and recycle operations is 3500. This causes the system to solve systematically, so that everything in the main flow sheet finishes solving before the sub-flow sheet begins. Then the whole simulation finishes before any changes are made for convergence or value matching from the adjust and recycle operations. In most basic process simulations this calculation scheme works well; however, with this particular simulation it causes large delays in convergence at each test value. For example, the sub-flow sheet triggers the adjust and recycle unit operations. But each time a change is made in the main-flow sheet using the default calculation values, the sub-flow sheet must also change its values to match. In this model, the cooling tower solution has no effect on the Rankine cycle since the cooling water temperature is specified. Therefore, decreasing the calculation values on the main flow sheet adjust and recycle streams to 2000 instead of 3500 will cause them to finish solving before the sub-flow sheet attempts to solve. This greatly decreases the calculation time of the system, since the cooling tower only solves once at the end.

3.3.4 USER VARIABLES

Since the model should be integral, meaning that it doesn't require any other outside influence to solve once the optimization process has begun, the minimum and maximum adjusted temperature values for the condensate pump inlet should also change based on system operation. Unlike most inputs and outputs the minimum and maximum temperature specifications of the adjust operation cannot communicate with an embedded spreadsheet. They can, however, be changed by using a user variable within HYSYS. The first instinct is to set the minimum temperature as the condenser outlet temperature. This creates problems because of the internal logic of the adjust variable. Since the minimum value is changing each calculation it struggles to arrive at the correct temperature value. An alternate solution is to set the minimum temperature based on the cooling water availability and ultimately the ambient conditions. The minimum temperature is set as 8°C higher than the available cooling water. A temperature difference of 5.556°C is the theoretical best heat transfer that can occur, and increasing that difference to 8°C allows for some amount of mixing to occur with the cascading stream. The maximum temperature is set as the saturated temperature of the condensate pump inlet stream minus 0.001°C. This small decrease in temperatures insures that the inlet is a sub-cooled liquid. If the saturated temperature were set, then any amount of two-phase liquid might be calculated and will cause the calculation to delay significantly. The codes used for the user variables are included in Appendix B.

As the hot steam temperature of the system changes, the inlet temperature to the condensate pump changes if the system remains at the same pressure. As the high temperature increases, the pump inlet temperature decreases and vice versa. This pushes the adjusted temperature towards the maximum and minimum limits which are set by the previous user variable discussed. When these limits are reached in HYSYS, a dialog box appears informing the user that either the maximum or minimum value has been reached and requires user interaction before the solver will continue. The purpose of this model is to achieve the approximate pressure conditions while requiring no additional user interaction following the initial set-up; therefore, it becomes necessary to eliminate these dialog boxes.

This is possible with the use of user variables. This is accomplished using multiple subroutines, one following the operation and one prior to the operation. The post execute subroutine checks whether or not the maximum or minimum limit has been reached. If so, the adjust is ignored in order to suppress the dialog box. If no other action is taken the process will continue to reach this limit. Therefore, the temperature is set back within the appropriate range and the pressure is adjust accordingly. A minimum limit being reached requires the system pressure to increase and a maximum limit reached requires the system pressure to decrease. Additionally, once ignored, the adjust will remain ignored until the user specifies otherwise. This is corrected with the use of a pre execute user subroutine. This subroutine tells the adjust to never be ignored prior to execution. In most cases, this will change nothing; however, when the adjust has been ignored due to one of the limits being reached, it will remove the ignored condition of the adjust and continue solving normally.

Consequently, this is the process by which the operating pressure of the system is obtained. Limits are set for the maximum and minimum pump inlet temperatures and the pressure can be adjust when these limits are reached. When the temperature reaches the minimum, the pressure is increased by a factor of 1.01, until the minimum limit is no longer reached. During this process there will be a moment when the pressure is such that the system can solve with an inlet temperature inside of the acceptable range. A study was performed on a smaller version of the Rankine cycle model which has 3 turbine stages and a single pump following the condenser. The final turbine stage outlet pressure is set by the cooling water temperature as discussed previously. The study manipulated the other three pressure set points for the outlet pressures of the condensate pump, HP turbine, and the IP turbine. The thermal efficiency was measured as each point was adjusted within possible values at the specific hot temperature set point. Ten operating points were evaluated for each of the three pressures resulting in a total of 1000 data points. Figure 3.10 shows how increasing the pump outlet pressure affects the system.

Since the thermal efficiency always increases with an increase in the high pressure of the system, the other two pressure effects on the thermal efficiency were plotted together in Figure 3.11. From the data shown in Figure 3.11, there was a maximum thermal efficiency for the two turbine outlet pressures. Therefore, it is most beneficial to achieve the maximum possible operating pressure for the system. As

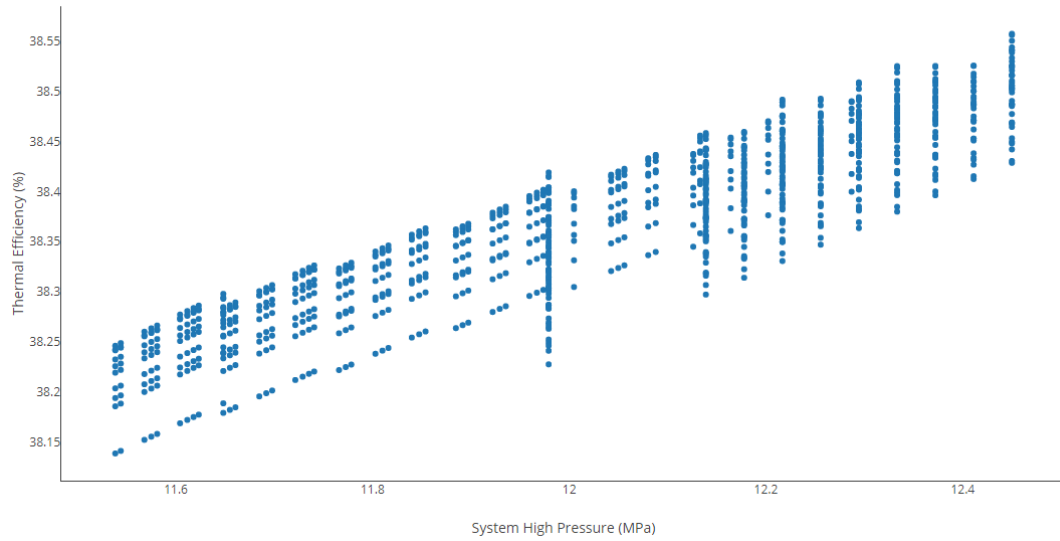


Figure 3.10: Effect of high pressure on thermal efficiency of the cycle

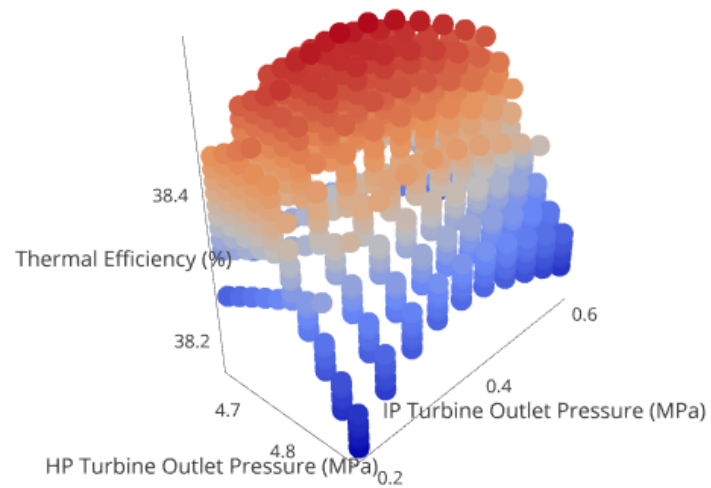


Figure 3.11: HP and IP turbine outlet pressure effect on thermal efficiency from small model

the pressure of the system is adjusted by the user variable, the system will eventually solve within the specified condensate pump inlet temperatures. However, the maximum pressure is desired to achieve highest thermal efficiency, as shown in Figure 3.10. Therefore, it is necessary for the user variable to be of the enumeration type. This allows the variable to change values when the code calls for it. In this case, there is a starting condition for the variable, and the subroutine will repeat the same process until that condition is reached. The condition is changed once the maximum temperature restriction on the adjust operation is reached. This requires the pressure to increase even when the system solves, which is taken care of by a third option in the code. This option tells the pump outlet pressure to increase even if the process is solved, unless the value of the enumeration has already been changed by the maximum temperature being reached. This allows the system to reach a high optimal operating pressure without any user interaction to test a number of pressures.

Due to the complexity of the model, it is difficult for the model to converge while maintaining similar pressure ratios in each of the turbine stages as the system pressure changes. The limits on the condensate pump inlet give a small window of operation and the overall system responds differently at different temperatures and pressures. The recommended optimal design of any Rankine cycle systems calls for a relatively similar increase in feed water temperature across each of the feed water heaters. This approach addresses the issue of controlling turbine stage pressures by taking an approach based on feed water heater optimization. In turn this determines which pressures to operate each of the turbines based on the TTD of the feed water heaters.

This is another user variable generated to determine each of the middle turbine outlet stages. This excludes the HP turbine and final stage of the LP turbine. The final stage of the LP turbine has already been discussed with reference to the cooling water availability temperature. This method of designating turbine outlet pressures based on feed water temperature increase requires at least one set point in the system. The HP turbine is set to have an operating pressure ratio of 0.5. This is consistent with the model provided by Dr. McKellar operating at various steam temperatures. This pressure ratio gives the outlet pressure and in turn the saturated steam temperature for the corresponding feed water heater inlet. This allows the steam generator inlet temperature to be known by the system. At the low pressure end of the cycle the condensate pump outlet temperature is known from the specified inlet temperature associated with the adjusted variable.

The temperature rise across each of the feed water heaters is determined by dividing this temperature difference by the total number of feed water heaters, in this case seven. Each heater outlet temperature is now known and can be used to find the corresponding turbine outlet pressure. This is done by performing a temperature-vapor fraction flash calculation of the fluid, water, to a temperature equal to the feed water outlet temperature plus the corresponding feed water heater's TTD. The fluid is flashed to the specified temperature and a vapor fraction of 1.0, which will give the pressure required for the specification. That

pressure is then exported to each of the six turbine stages which require a pressure specification. By automating this process in the user variable, each time the system has a pressure increase, all of the turbine stage outlet pressures will be updated for the new temperature difference across the feed water heaters.

Once the pressure has been reached, there are many options for proceeding to the optimum solution. The user can either leave the result as is or use predetermined turbine outlet pressures to fine tune the system. Another option is for the user to keep playing with the pressures and increase the system pressure until the predetermined absolute maximum pressure is reached, this will cause the thermal efficiency to increase, in most cases. The final result after these pressure adjustments can be used to predict the thermal efficiency of the cycle. The analysis portion of this thesis, presented in Chapter 4 uses the raw result from the automated process to compare with Dr. McKellar's models, and experiments with manually adjusting the other turbine outlet pressures. Additionally, if the turbine outlet pressures are not known but the steam generator pressure is known, the model can be used to find the thermal efficiency using the different assumptions made in this thesis. This requires the recycle stream to be removed from the system and the steam flow rate result from the first feed water heater calculation to be exported to the stream. The condensate inlet temperature must be set at the desired value since the adjust operation will now always be solved. This adjustment to the model also allows for increased solving speed once the steam generator pressure has been determined.

3.4 SUMMARY OF ASSUMPTIONS AND SET POINTS

With the information included in this chapter, the model is ready to be solved using only the steam generator outlet temperature, and cooling water availability temperature as user inputs. Additionally, the steam pressure may be specified as well if the proper adjustments have been made to the model. The following is a recap of all of the adjustments and set points used to automate the model:

- The total thermal power from both the steam generator and the reheater is 100 MW. The power applied in the steam generator is adjusted so that the sum of the two is 100 MW.
- The reheater outlet temperature is set to be equal to the steam generator outlet temperature.
- The efficiencies used in all of the turbine stages and pumps is outlined in Table 3.4.
- The outlet pressure of the final LP turbine stage is set so that the saturated temperature of the steam is 12.25°C higher than the cooling water temperature.
- The pressure ratio for the HP turbine is set at 0.5 so that the user variable can solve the remaining turbine outlet pressures. The remaining pressures are determined by the user variable such that the feed water temperature change across all heaters is the same.

Table 3.4: All turbomachinery and their specified isentropic efficiencies

Operation	Efficiency	Operation	Efficiency
HP Turbine	85%	LP Turbine Stage 5	80%
IP Turbine Stage 1	90%	Condensate Pump	75%
IP Turbine Stage 2	90%	Booster Pump	75%
LP Turbine Stage 1	80%	Boiler Pump	75%
LP Turbine Stage 2	80%	Cooling Tower Fan	75%
LP Turbine Stage 3	75%	Cooling Water Pump	75%
LP Turbine Stage 4	80%		

- The TTD and DCA for all of the surface feed water heaters is set as 5.556°C (10°F).
- The mass flow rate used from each turbine stage for the feed water heaters is calculated in a spreadsheet based on the energy balance around the heater and is exported to the correct stream.
- The outlet conditions of the deaerator is set as a saturated liquid at the corresponding pressure. The pressure is determined by the turbine stage that the steam is pulled from.
- The cascading streams from the feed water heaters are expanded through a valve to match the pressure of the steam pulled from the turbine.
- All heat exchanger pressure drops are set as 2% of the inlet stream pressure. This applies to the steam generator, reheater, condenser, and all surface feed water heaters.
- The deaerator pressure and the lower temperature feed water heater pressure drops determine the condensate pump outlet pressure.
- The boiler pump outlet pressure is defined by the steam generator outlet pressure and the corresponding heat exchanger pressure drops.
- The boiler pump pressure ratio is defined as 3.545 which determines the booster pump outlet pressure. This was chosen to maintain the feed water sub-cooled in the higher temperature feed water heaters. This is also the pressure ratio for the boiler pump shown in Table 3.1
- The bottom adjust minimum and maximum limits are set with the minimum as 8°C higher than the cooling water temperature and the maximum as just below the saturated temperature.
- The current cooling tower set points are: 20°C air with 50% relative humidity at atmospheric pressure. The water source is also 20°C.

CHAPTER 4: ANALYSIS

4.1 INITIAL RESULTS

The results of the model developed in Chapter 3 are compared to optimized results prepared by Dr. Michael McKellar for steam power conversion cycles and their integration with nuclear reactors [42]. Table 4.1 shows the results used in the report and Table 4.2 shows the results for this optimization algorithm. The argument that the model produces the optimal thermal efficiency is valid even though the actual thermal efficiency may be increased with further user interaction. The argument is valid since the logic used and developed in Chapter 3 is based on the physical constraints of the system. The entire Rankine cycle may be viewed as a black box with specific inlets and outlets. In this case, the inlet conditions include the maximum temperature and the thermal load. The outlet is the cooling water availability temperature. These constraints were placed on the system with all unit operations held to physical constraints on operation. All operation within the Rankine cycle black box are thermodynamically and physically sound making the result of the simulation, the optimum solution.

The starting position for the code to be run for the comparison was with the steam generator outlet temperature at 300°C with a pressure of 4 MPa. The HP turbine pressure ratio was set at 0.5 and left there for the duration of the calculations. Overall, the model required around 30 minutes to complete

Table 4.1: Data from the models created by Dr. Michael McKellar, this data was used to show optimal performance of power cycles [42]

Steam Temperature (°C)	Steam Pressure (MPa)	Thermal Efficiency
300	5.282	34.16%
350	7.683	36.65%
400	12.01	39.36%
450	15.00	41.13%
500	19.21	43.01%
550	21.13	44.34%
600	24.00	45.91%

Table 4.2: Data from my model using only optimization algorithm discussed in Chapter 3 with HP turbine pressure ratio set at 0.5

Steam Temperature (°C)	Steam Pressure (MPa)	Thermal Efficiency
300	7.122	35.87%
350	9.041	37.67%
400	11.14	39.40%
450	13.73	41.15%
500	16.75	42.83%
550	20.03	44.39%
600	23.95	45.91%

all of the calculations shown in Table 4.2. The model was not modified beyond the automated tasks. This represents a significant time save for arriving at the optimized solution. Further interaction with the model would undoubtedly increase the performance of the model; however, this was not the purpose of these results.

Figure 4.1 shows the temperature and pressure of the steam generator outlet at each temperature for each model. These values are plotted along with the generated temperature and pressure operational data from Chapter 3. At 600°C, neither model could converge at a pressure within the limit. The critical temperature and pressure of water is 374.14°C and 22.06 MPa [45]. 600°C is well beyond the critical temperature of water. With an operating pressure of 24 MPa, which is above the critical pressure of water, the 600°C cycles are supercritical steam cycles. Supercritical cycles are common in fossil fuel power plants which commonly operate as high as 1000°F (537.78°C) and 3675 psig (25.4 MPa) [46]. In the case of the 600°C operating temperature, the pressure was limited to 24 MPa in order to keep the recommended operating pressure [42, 43]. The HP turbine pressure ratio had to be adjust for this result to 0.511 to find the maximum condensate pump inlet temperature.

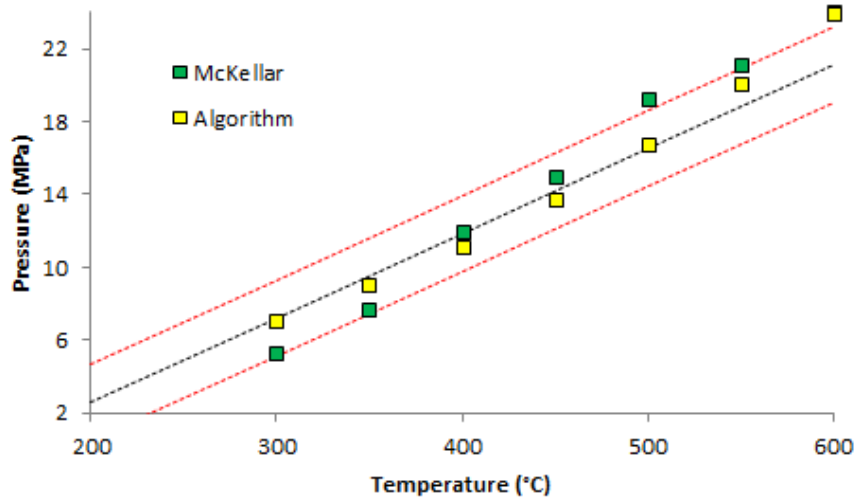


Figure 4.1: Comparison of data points from McKellar’s model and the algorithm with pressure vs. temperature operation

4.2 EXPANSION PRESSURE CONSIDERATIONS

A comparison of the data points in Tables 4.1 and 4.2 as well as Figure 4.1 show many important points. Firstly, for the most part, the results from the optimization algorithm give a higher thermal efficiency however slight it may be. This may be attributed to the increased expansion in the final turbine stage. Dr. McKellar’s models used a temperature difference of 15.51°C between the cooling water availability

temperature and the saturated turbine outlet temperature. This model has cited a better temperature difference as 12.25 °C which provides a better average of operating recommendations at various ambient temperatures. Although the lower temperature difference provides more accurate results, Table 4.3 shows the results when the final expansion is set to the same pressure used in Dr. McKellar's models. This was done by increasing the temperature difference between the cooling water and the turbine outlet to 15.51°C from 12.25°C.

Table 4.3: Data from my model expanded to the same final pressure used by Dr. McKellar

Steam Temperature (°C)	Steam Pressure (MPa)	Thermal Efficiency
300	7.119	35.34%
350	9.041	37.19%
400	11.25	38.99%
450	13.73	40.70%
500	16.58	42.35%
550	19.83	43.91%
600	24.00	45.57%

As a side note, the model does not converge to the same results each time due to the discrepancy in the adjust unit operations and its guess values. Therefore in some cases, the steam pressure increased, and in other cases it may have decreased compared to the values in Table 4.2. To further illustrate this point, the initial simulations for all temperatures was performed once more from the same starting point as Table 4.2. These results are presented in Table 4.4.

The steam pressure is another consideration in the comparison of the data between the two models. A higher operating pressure will give a higher thermal efficiency since the steam can expand more overall. This is highlighted by the differences in the thermal efficiency values between Tables 4.1 and 4.3 since these are expanded to the same condensing pressure. However, due to increased expansion in the results from Table 4.2, a higher operating pressure may not always be necessary to achieve a higher thermal efficiency.

Table 4.4: Another iteration of Data from the optimization algorithm with the same starting conditions used to generate the data in Table 4.2

Steam Temperature (°C)	Steam Pressure (MPa)	Thermal Efficiency
300	7.052	35.81%
350	8.954	37.62%
400	11.14	39.40%
450	13.73	41.15%
500	16.75	42.83%
550	20.03	44.39%
600	23.72	45.88%

4.3 USER CONSIDERATIONS

After the convergence of the system, further user manipulation is required to find the absolute optimal condition. The algorithm will produce an accurate estimate of where the Rankine cycle will perform based on the temperature set points. However, the HP turbine remains at the same pressure ratio throughout the pressure increase. This provides implications with the use of a reheater, since the model will perform counter intuitively at times. Because the steam temperature is raised to the limit in the reheater, the effect of the HP turbine expansion on the thermal efficiency may depend on other factors downstream in the system and not just on the pressure ratio.

Additionally, the model was optimized based on the feed water heater effects. Actual plant operation may require steam to be pulled at different pressures than those found by the feed water heater calculations. Therefore, the outlet pressure for each turbine stage can be adjusted accordingly to either meet plant specifications, turbine design specifications, or increase overall thermal efficiency of the system. It is recommended that the pressure rise and turbine outlet pressure user variable codes be disabled during manual manipulation, otherwise the system pressure and outlet pressures will be out of the control of the user.

Finally, the bottom adjust and the recycle may now be removed and the steam mass flow calculated by the spreadsheet for feed water heater one should be directly exported to the appropriate stream. This will allow for automatic convergence of the bottom of the system. Since the recycle and adjust are only required for the pressure change of the entire system, they can be removed. Manual manipulation will result in convergence of the system at the condensate pump inlet temperature desired by the user.

4.4 SENSITIVITY ANALYSIS ON OPERATING PRESSURES

To assist the user in further optimizing the model, a study was performed on each of the pressure specifications to show to what extent each affects the entire system. This analysis includes the steam pressure, the HP turbine pressure ratio, and each of the turbine stage outlet pressures excluding the final stage. This sensitivity analysis builds on the results generated by the optimization algorithm. Therefore, the pressure user variables were disabled and changes were made to the bottom of the cycle for consistent performance. The changes made have been previously discussed. The only exception is the inclusion of the constant feed water temperature change when varying the steam generator pressure. The data for the sensitivity analysis are from the model with a 500°C steam temperature. During the sensitivity analysis all other set points for pressures were reverted to the automated solution value for consistency and an independent analysis of each variable. The results for the steam pressure variance are presented in Figure 4.2, which plots the steam pressure vs. the thermal efficiency of the system. All other analyses contain plots of turbine pressure ratio vs. thermal efficiency. Even though, the outlet pressure was

the variable changed in most cases, the pressure ratio has more significance across all temperature and pressure operating conditions. Table 4.5 shows the pressure values for the steam and each turbine stage outlet as calculated by the optimization algorithm, as a reference for the sensitivity analysis.

Table 4.5: Pressure set points determined by the optimization algorithm at a steam temperature of 500°C

Pressure Stage	Pressure (MPa)	Pressure Ratio
Steam Pressure	16.25	-
HP Turbine	7.693	0.5000
IP Stage 1	4.682	0.6086
IP Stage 2	2.485	0.5308
LP Stage 1	1.046	0.4209
LP Stage 2	0.4914	0.4698
LP Stage 3	0.1689	0.3437
LP Stage 4	4.503e-2	0.2666
LP Stage 5	6.366e-3	0.1414

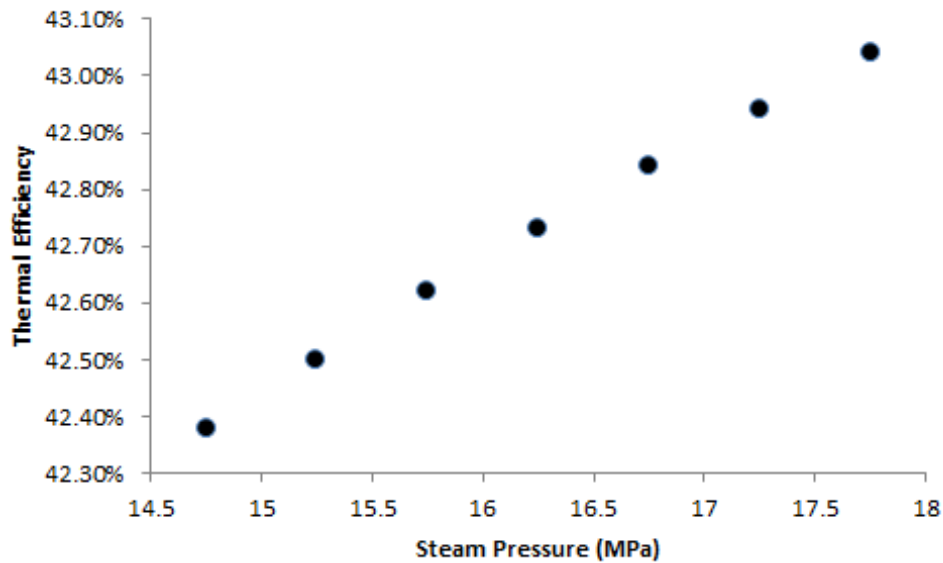


Figure 4.2: Variance of thermal efficiency with different steam pressures

As expected, the thermal efficiency increases as the steam generator pressure increases. However, the data in Figure 4.2 suggests that there is a theoretical maximum efficiency for this system since the data appear to be approaching a maximum. It is likely that this theoretical maximum occurs at an operating pressure which is considerably larger than what current materials are capable of handling and therefore is not of concern. The remaining data from the sensitivity analyses of each of the first seven turbine stages are shown in Figures 4.3 through 4.9.

Each of the sensitivity analyses attempted to center the data around the automatically generated result. It is interesting to note that for these seven turbine stages, there were four which have maximum

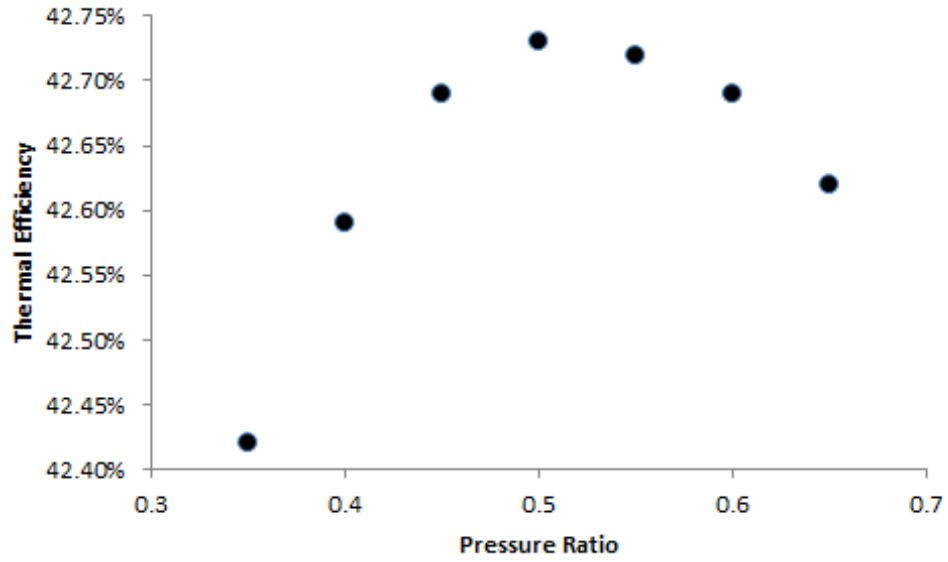


Figure 4.3: Variance in thermal efficiency vs. HP turbine pressure ratio

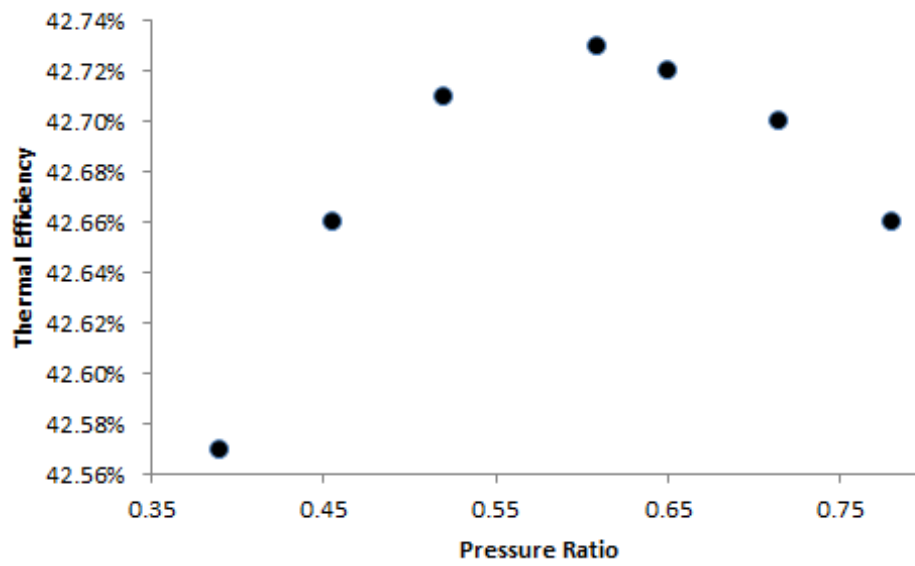


Figure 4.4: Variance in thermal efficiency vs. IP turbine stage 1 pressure ratio

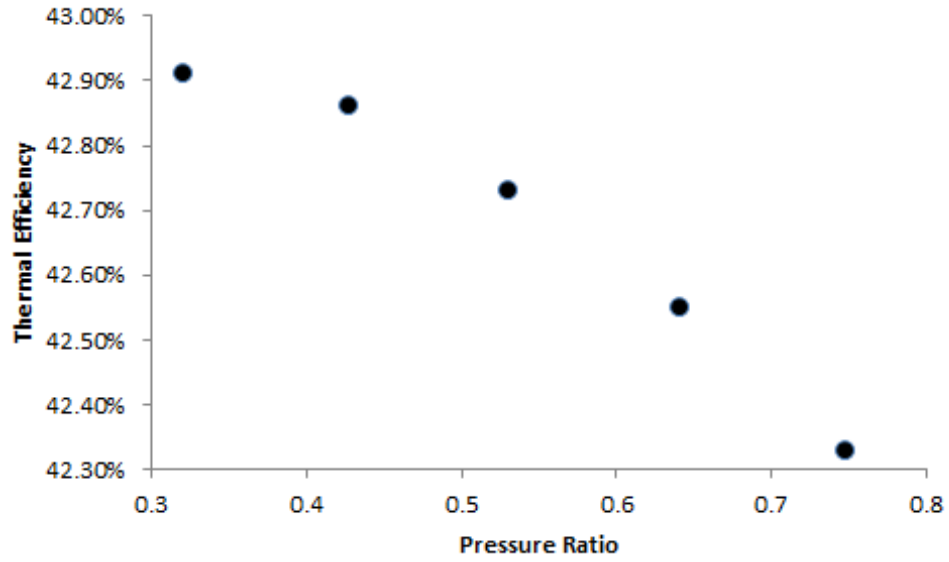


Figure 4.5: Variance in thermal efficiency vs. IP turbine stage 2 pressure ratio

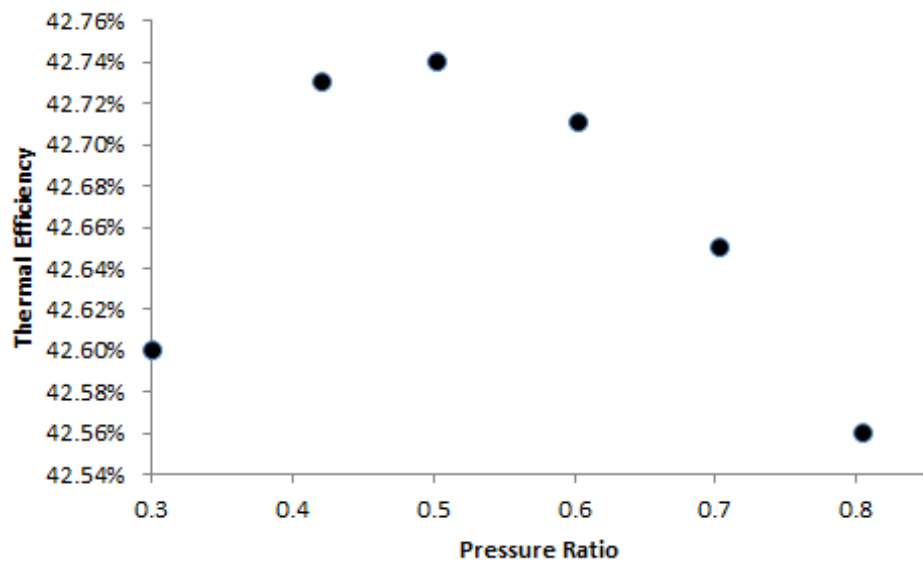


Figure 4.6: Variance in thermal efficiency vs. LP turbine stage 1 pressure ratio

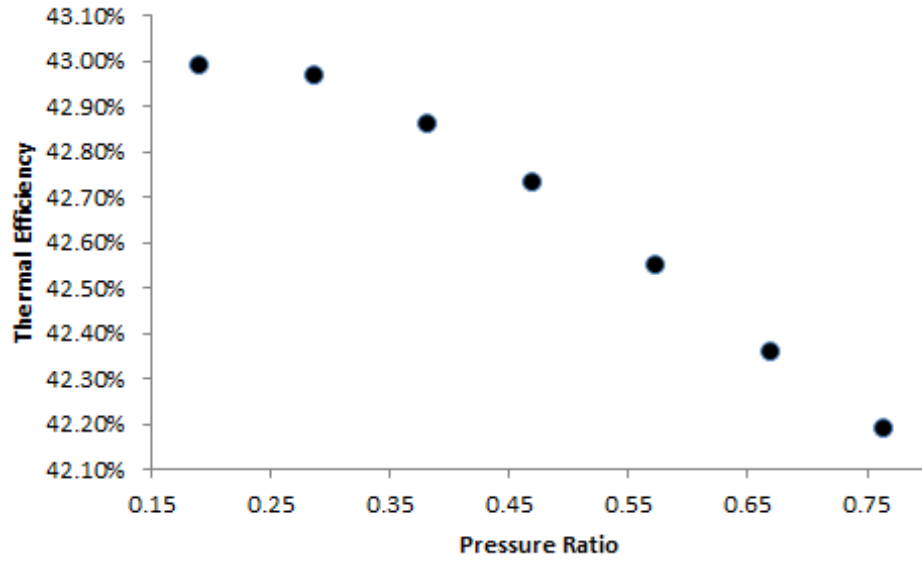


Figure 4.7: Variance in thermal efficiency vs. LP turbine stage 2 pressure ratio

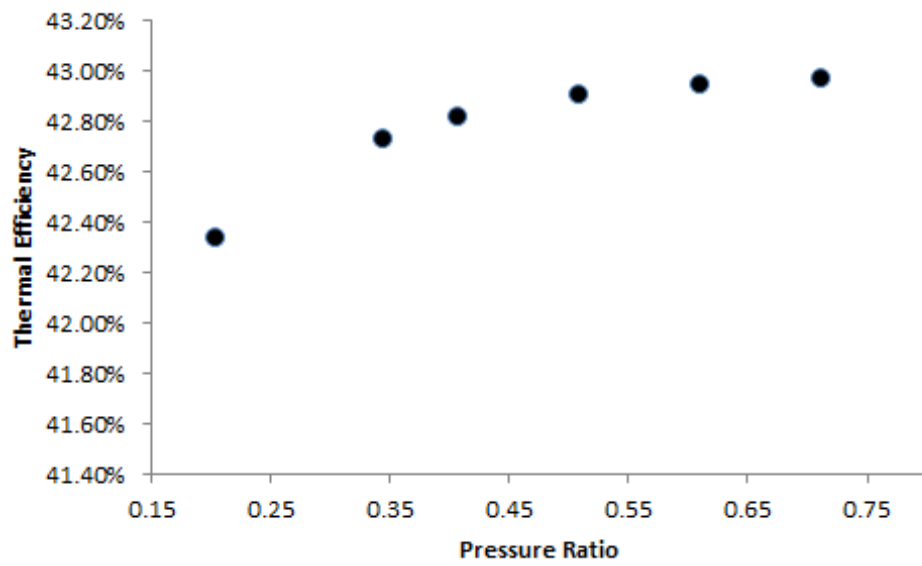


Figure 4.8: Variance in thermal efficiency vs. LP turbine stage 3 pressure ratio

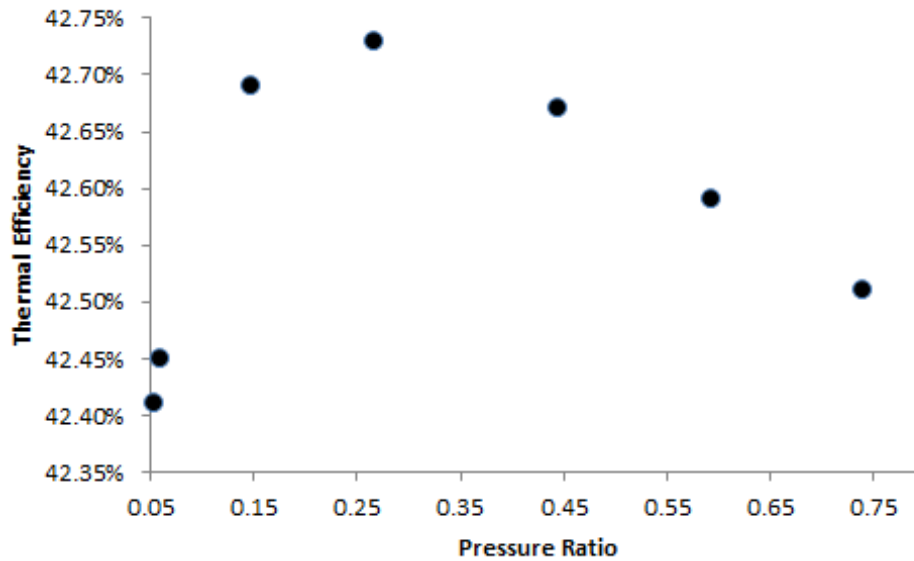


Figure 4.9: Variance in thermal efficiency vs. LP turbine stage 4 pressure ratio

thermal efficiency values around the pressure ratio which was generated by the optimization algorithm. The other three analyses do not have a maximum on the plot, but show that the maximum thermal efficiency may be reached as the outlet pressure reaches the outlet value of the surrounding turbine stages, indicated by a pressure ratio of 1 or a ratio that approaches zero. These stages which do not show a maximum value are in all cases not the first stage in each of the major turbines (HP, IP, or LP) but instead, are the second and/or third stages of the turbine. This result is not unexpected when considering the common configurations of turbines in steam power plants as discussed in Chapter 2. In this case all of the turbine stages are in series; however, from the diagram in Chapter 2, several configurations contain turbine stages in parallel. This turbine layout may explain why some plots show a higher thermal efficiency when the stage is eliminated.

Figures 4.3 through 4.9 show significant data that allows the user to further optimize the result of the algorithm with ease by understanding how each stage individually affects the overall thermal efficiency of the cycle. The optimized result for the 500°C Rankine cycle was achieved using the pressure ratios shown in Table 4.6 with an efficiency of 43.15%. Table 4.7 shows how this value compares with the result from Dr. McKellar’s model as well as values obtained from the different steam and condensing pressures which have been discussed up to this point. The models with the higher steam generator pressure have the highest thermal efficiency for reasons already discussed.

Table 4.7 shows the different high and low pressures that we are concerned with. The only result which does not produce a higher thermal efficiency than Dr. McKellar’s model has the lower high pressure and the higher low pressure. The high pressures were chosen because they represent the high pressure achieved

Table 4.6: Turbine pressure ratios for optimized Rankine cycle at 500°C with thermal efficiency of 43.15%

Turbine Stage	Pressure Ratio	Turbine Stage	Pressure Ratio
HP Turbine	0.5150	LP Stage 2	0.3000
IP Stage 1	0.5609	LP Stage 3	0.6667
IP Stage 2	0.4348	LP Stage 4	0.2250
LP Stage 1	0.5000	LP Stage 5	0.1415

Table 4.7: Comparisons of the thermal efficiency based on different high and low system pressures

High Pressure (MPa)	Low Pressure (MPa)	Thermal Efficiency
16.25	6.366e-3	43.15%
16.25	7.584e-3	42.69%
19.21	6.366e-3	43.74%
19.21	7.584e-3	43.28%
19.21 (McKellar)	7.584e-3	43.01%

by the optimization algorithm and the high pressure used in Dr. McKellar’s model. The low pressures represent the difference in condensing pressure from a temperature difference of 12.25°C, used by the optimization, and 15.51°C, used by Dr. McKellar.

4.5 COOLING TOWER CONSIDERATIONS

Although the cooling tower uses a relatively small amount of energy for operation, it is still important to see what kinds of effects the ambient conditions have on the thermal efficiency. For this analysis the results are from changes to the model operating at 400°C; there is no particular reason that this temperature set point was chosen. The base ambient conditions for each is 20°C moist air temperature at 50% relative humidity and cooling water source at 20°C with a 25°C cooling water availability temperature. The corresponding steam pressure and thermal efficiency are 11.00 MPa and 39.36%. In all cases, the user variable codes for the system pressure were disabled to keep the power system the same for the most part. The only change in the power system will come from the last LP turbine stage, which is dependent on the cooling water temperature.

4.5.1 RELATIVE HUMIDITY EFFECTS

The effect from changing the relative humidity comes from an increase in power for the fan used to drive the air flow through the tower. An increase in relative humidity decreases the amount of water vapor the air is capable of carrying in the cooling tower which decreases the overall effectiveness of the tower. Therefore, more air is required at a higher relative humidity and vice versa. This causes the power required to drive the fan to change according to total air flow required. This does not have any theoretical effect on the Rankine cycle operation, since the cooling water temperature remains unaltered. Table 4.8 shows the results from changing the relative humidity on both the required air flow and the

Table 4.8: Effects of relative humidity on the cooling tower operation and thermal efficiency

Relative Humidity	Air Mass Flow (kg/s)	Thermal Efficiency
10%	1211.14	39.44%
25%	1355.17	39.42%
40%	1538.60	39.39%
50%	1691.56	39.36%
60%	1878.61	39.33%
75%	2253.03	39.27%
90%	2815.27	39.17%

thermal efficiency of the system.

Overall the largest effect that relative humidity has on the system is the required air mass flow rate which becomes extremely relevant in the cooling tower design. The cooling tower must be capable of supplying enough air to successfully bring the cooling water to the set inlet temperature regardless of ambient conditions. The effect on thermal efficiency is not as severe; however, the air with higher humidity has a larger overall effect on thermal efficiency per change in relative humidity. This becomes a concern in areas with large fluctuations in humidity, which leads those areas to have more interest in natural draft cooling towers which do not require a fan to create the required air flow rate. In these cases, the losses are involved in the capital cost of the plant and not in the operation of the plant.

4.5.2 AMBIENT TEMPERATURE EFFECTS

Increasing the ambient temperature has a negative effect on the cooling capacity of the cooling tower. Although the higher temperature air is able to hold more water vapor and does a better job of evaporating the water that is sprayed into the cooling tower, the overall cooling is lowered since the air temperature will also increase the liquid water temperature as contact heat transfer occurs in the tower. Since more water is evaporating in the air, more make up water will be required. Also, the air flow rate will increase to balance the evaporation rate with the make up water rate while still maintaining the cooling water temperature set point. The base case make up water flow rate is 30.48 kg/s compared to the total cooling water flow rate of 2157 kg/s. Table 4.9 shows how increasing or decreasing the air temperature affects the air mass flow rate, evaporating water mass flow rate, and the make up water flow rate as well as the thermal efficiency of the overall system. Since the condensing side of the system parameters are held constant the overall cooling water mass flow rate remains the same.

As expected the thermal efficiency of the system decreases as the ambient temperature increases. This has a minor effect on the system when the ambient temperature is less than the cooling water source temperature. However, when the air becomes warmer than the water source temperature, the air requirement increases and thermal efficiency decreases rapidly. When the air temperature becomes higher than the cooling water availability temperature at 30°C, the effect becomes even larger. It does not make

Table 4.9: Effects of ambient temperature on the operation of the system

Air Temperature (°C)	Air Flow (kg/s)	Make-Up Water (kg/s)	Thermal Efficiency
5	982.212	23.12	39.49%
10	1120.16	24.82	39.46%
15	1330.03	27.08	39.43%
20	1691.56	30.48	39.36%
25	2465.92	37.02	39.23%
30	5253.44	58.86	38.74%

sense for the air temperature to be much different than the source temperature in most cases, since the source is usually a nearby lake, river, or the ocean. The air temperatures to pay the most attention to are 15°C, 20°C, and 25°C since these are the most reasonable with a source temperature at 20°C. The other points are shown to illustrate extreme conditions.

4.5.3 COOLING WATER SOURCE EFFECTS

The cooling water source has a direct impact on the temperature available for the condenser which directly affects the operation of the Rankine cycle. The final expansion pressure of the turbine is set based on the cooling water temperature and in turn determines the pump inlet and feed water train inlet temperatures. Therefore, as the cooling water temperature increases, the thermal efficiency will decrease because the turbine cannot expand to as low of a pressure resulting in less power generation. Table 4.10 shows the effects that changing the source temperature has on the system. The cooling water availability temperature will remain 5°C higher than the source temperature to allow for operation of the cooling tower. Since the low temperature of the system is changing the pressure codes must be run to find the new operating pressure of the system based on the cooling water temperature.

Table 4.10: Effects of changing cooling water availability temperature on the entire system

Water Source Temperature (°C)	Steam Pressure (MPa)	Thermal Efficiency
10	10.35	39.20%
15	10.46	39.59%
20	11.00	39.36%
25	11.32	38.85%
30	11.77	38.36%
35	12.85	38.06%

The results show that although the system pressure increases with an increase in cooling water temperature, the overall thermal efficiency of the system decreases since the lowest expansion pressure increases. This follows the limits on the Carnot efficiency as well. Since the high temperature remains the same while the low temperature increases, the absolute maximum efficiency decreases. This is reflected well in the results shown in Table 4.10, as it follows the limits placed by the Carnot efficiency. The 10°C data point is an out lier when it comes to this trend. Since the air temperature remained unchanged at 20°C,

it required a substantial amount of air to cool the water back to the set point of 15°C at that point. This is the cause of the decrease in efficiency. This illustrates the point that it is difficult to operate a cooling tower when the cooling water set point is significantly less than that of the air. This is a consideration to take into account when designing a cooling tower for operation.

CHAPTER 5: SUMMARY AND CONCLUSIONS

5.1 SUMMARY

Due to the economic factors driving the United States energy industry, any thermal power plant must be designed with an efficiency as high as possible to be competitive. Each design attempts to achieve the highest efficiency by adjusting power system designs and performing optimization on the chosen design. Higher operating temperatures or steam availability temperatures provide a higher thermal efficiency by default. This is one of the motivations behind the design of advanced nuclear reactors which are capable of delivering steam at a higher temperature [10].

The steam Rankine cycle is the most common way to produce large amounts of base load energy among various sources of heat [11]. The theory behind the complete Rankine cycle and accompanying unit operations was developed. This included the unit operations and systems that are required to design a basic process model. The cooling tower was also included in this explanation. Aspen HYSYS was introduced as the process modeling software used with information including the basics of operation and some advanced modeling capabilities.

An algorithm was generated for modeling the complete Rankine cycle. The model can be applied to any heat source; however, the intended source is a nuclear reactor, including advanced reactor technologies. The application in the Aspen HYSYS process modeling software is developed and used to test operating pressures and thermal efficiencies for steam temperatures in the range of 300°C to 600°C. The implementation of the optimization algorithm is explained with specifics regarding the implementation in HYSYS.

The results of the model at selected temperatures were compared to similar results from a model generated by Dr. Michael McKellar. The specific results at each temperature provide an automated result for the estimated optimal operating conditions. The automatic result does not arrive at the same solution every time due to some inaccuracies in the logical operations used in HYSYS for the model. The adjust unit operation, which uses guess values to check for a matching condition, is one of the sources of this inaccuracy. Additional user interaction with the model will help the model reach the optimal results. However, it is estimated that 95% of the optimization work is performed by the algorithm.

A sensitivity analysis of the model was performed with respect to the operating pressures, including all turbine outlet pressures and the steam pressure. The results of the analysis are intended to assist the user in determining which turbines to adjust to further increase thermal efficiency and in which direction to adjust the pressure under consideration. The effect of the ambient conditions on the cooling tower and the Rankine cycle were also investigated by adjusting the relative humidity the ambient air as well as the air and water source temperatures.

5.2 CONCLUSIONS

The results show that the algorithm automatically produces a Rankine cycle within a short amount of time which is comparable to other models with significant optimization efforts. Additionally, the model can be assigned to any user specified temperature, not only those previously investigated. The sensitivity analysis helps the user to determine how they can manually increase the thermal efficiency beyond the result produced by the algorithm. Because the algorithm operates within the bounds of physical operation for the system and results in a highly comparable thermal efficiency, the algorithm successfully optimizes the Rankine cycle.

Not only is the optimization algorithm capable of solving the system at any user specified temperatures within a highly reasonable amount of time, it can also easily be manipulated to work with any power cycle design. The simplicity of the embedded spreadsheets and user variables allow the entire optimization method to be implemented in any Rankine cycle configuration with minimal changes required. The basis for the optimization logic lies within the feed water heaters and their respective mass and energy balance. This includes how much steam must be extracted from each turbine stage as well as the pressure at which to extract the steam in order to ensure comparable temperature rise of the feed water across all heaters. This also allows for different turbine configurations to be explored as well as different amounts of feed water heaters.

5.3 FUTURE WORK

The results of this optimization algorithm can be used to further evaluate how other factors affect the thermal efficiency of a Rankine cycle. One example would be to compare the differences between a wet and a dry cooling tower. The model uses a forced draft wet cooling tower because the water is in direct contact with the air flowing through the cooling tower. If a dry cooling tower were to be used instead, the affects that ambient temperature has on the power system can be explored in a system with no external water source. In warmer climates, the air may not be capable of condensing the steam to the same degree if there is no cooling water available. This is because the warmer air would not be able to cool the cooling water as much as if there were a water source near by. By experimenting with the cooling tower design, a study may be performed to show to what degree the ambient conditions affect each system based on the ultimate heat sink design.

Another application of the algorithm is in complete coupled power system designs. This includes multi-output hybrid energy systems where the steam generated may be diverted to a nearby chemical process. This would affect the amount of steam flowing through the Rankine cycle. When performing parametric studies on such a system, varying the steam flow rate as well as the steam pressure would necessitate continual user manipulation in the Rankine cycle in the absence of the algorithm developed

in this thesis.

There is also the need for transient modeling of the system. This would not be an application of the algorithm discussed in this thesis, but instead a continuation of the work performed. Using the optimal system design as the starting point, a transient system may be designed within HYSYS to size each of the unit operations. This can be coupled to environmental data input into the cooling tower model, to evaluate daily and seasonal fluctuations in power output from the system. Since the model is capable of solving any number of different steam temperatures, this transient model can be implemented for different reactor types to help determine which reactor design and power system may be most suitable for a certain geographical area.

REFERENCES

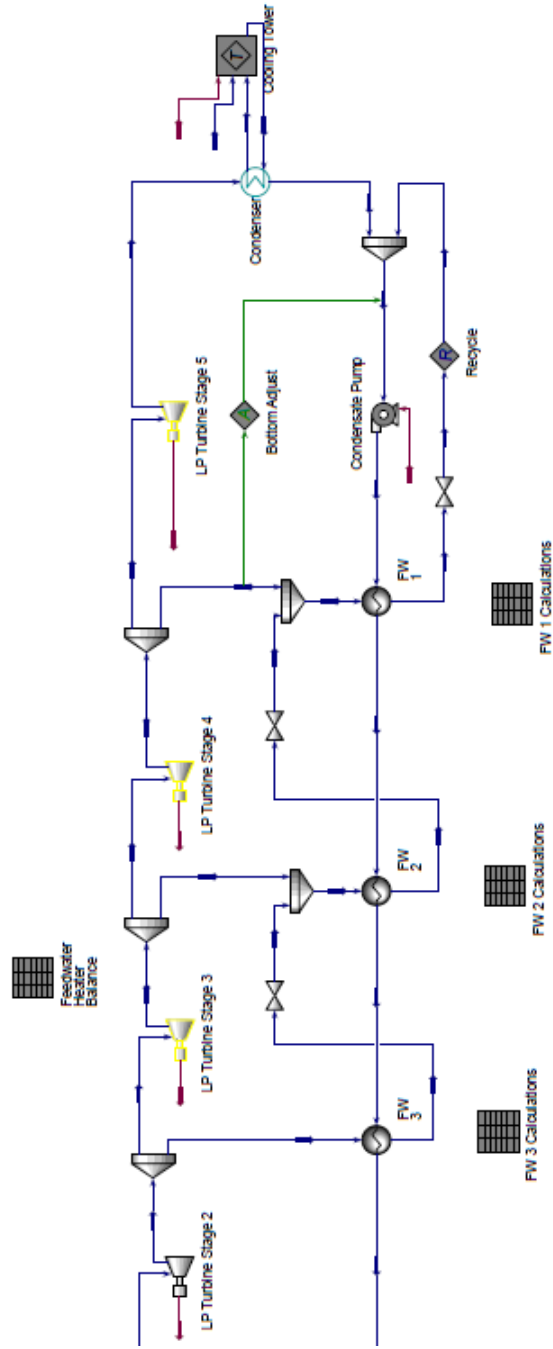
- [1] John J Conti. AEO2018 Reference case highlights. Technical report, 2018.
- [2] Christopher R Knittel, Konstantinos Metaxoglou, and Andre Trindade. NBER Working Paper Series - Natural Gas Prices and Coal Displacement: Evidence from Electricity Markets. Technical report, 2015.
- [3] Electric Power Annual 2016. Technical report, U.S Energy Information Administration, 2018.
- [4] Robbie Hayunga. When the Hurricanes Came, Nuclear Plants and Operators Were Ready, 2017.
- [5] A Lokhov. Load-following with nuclear power plants. Technical Report 29, 2011.
- [6] Geoffrey Haratyk. Early nuclear retirements in deregulated U.S. markets: Causes, implications and policy options. *Energy Policy*, 110:150–166, nov 2017.
- [7] Michael Buchdahl Roth and Paulina Jaramillo. Going nuclear for climate mitigation: An analysis of the cost effectiveness of preserving existing U.S. nuclear power plants as a carbon avoidance strategy. *Energy*, 131:67–77, jul 2017.
- [8] Thomas K. Bauer, Sebastian T. Braun, and Michael Kvasnicka. Nuclear power plant closures and local housing values: Evidence from Fukushima and the German housing market. *Journal of Urban Economics*, 99:94–106, may 2017.
- [9] NRC: Operating Reactors.
- [10] Neil E. Todreas and Mujid S. Kazimi. *Nuclear systems*. Taylor & Francis, 2012.
- [11] James H. Rust. *Nuclear power plant engineering*. Haralson Pub. Co, 1979.
- [12] GIF. Generation IV International Forum Annual Report 2007. Technical report, 2007.
- [13] NRC. Palo Verde Nuclear Generating Station, Unit 1, 2018.
- [14] Babcock & Wilcox Company. *Steam, its generation and use*. Babcock & Wilcox, 39th edition, 1978.
- [15] Pamela L Spath and Margaret K Mann. Life Cycle Assessment of a Natural Gas Combined-Cycle Power Generation System. Technical report, 2000.
- [16] Kam W. Li and A. Paul. Priddy. *Power plant system design*. 1985.
- [17] Bertrand F Tchanche, Gr Lambrinos, A Frangoudakis, and G Papadakis. Low-grade heat conversion into power using organic Rankine cycles â A review of various applications. *Renewable and Sustainable Energy Reviews*, 15:3963–3979, 2011.

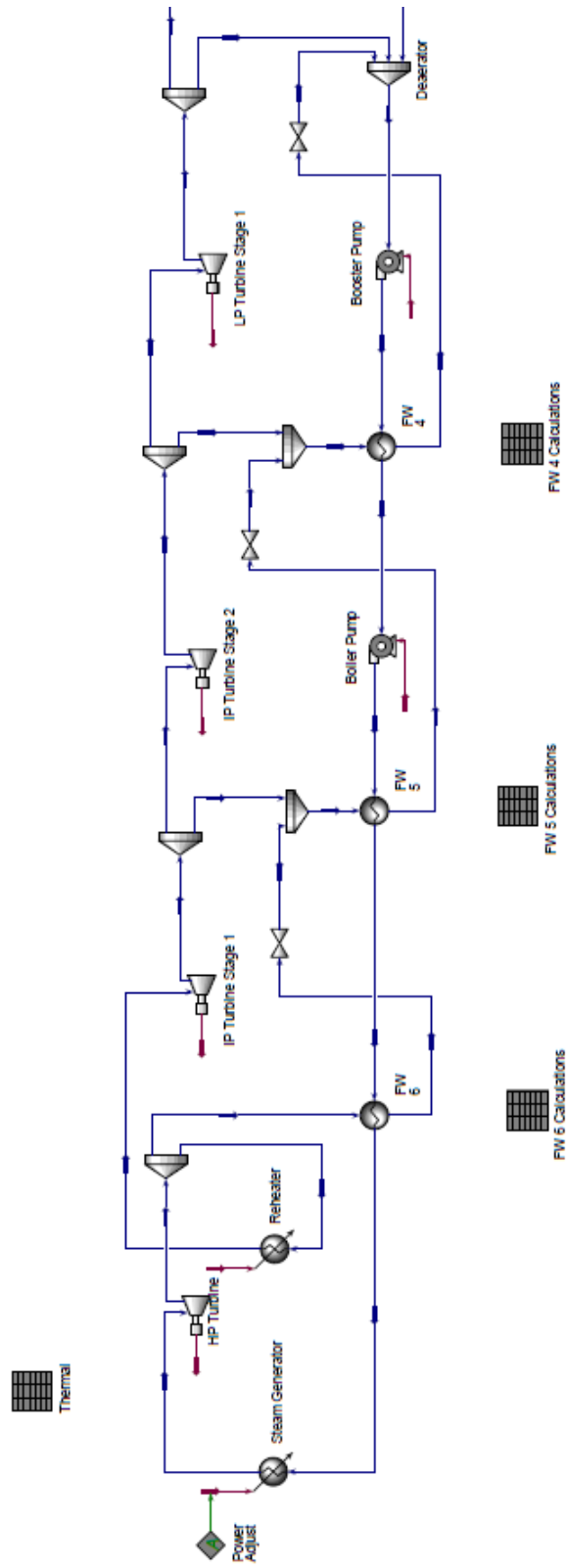
- [18] H D Madhawa Hettiarachchi, Mihajlo Golubovic, William M Worek, and Yasuyuki Ikegami. Optimum design criteria for an Organic Rankine cycle using low-temperature geothermal heat sources. *Energy*, 32:1698–1706, 2007.
- [19] John H. Lienhard IV and John H. Lienhard V. *A Heat Transfer Textbook, fourth edition*. 2012.
- [20] C H Fuller. Fort Saint Vrain Operation Experience.
- [21] Mario D Carelli, L E Conway, L Oriani, B Petrovi, C V Lombardi, M E Ricotti, A C O Barroso, J M Collado, L Cinotti, N E Todreas, D Grgi, M M Moraes, R D Boroughs, H Ninokata, D T Ingersoll, and F Oriolo. The design and safety features of the IRIS reactor. *Nuclear Engineering and Design*, 230:151–167, 2004.
- [22] Kellen R Mccarroll, John C Lee, Annalisa Manera, Matthew J Memmott, and Paolo Ferroni. Preliminary risk assessment of the Integral Inherently-Safe Light Water Reactor. 2017.
- [23] Muhammad Ilyas and Fatih Aydogan. Steam generator performance improvements for integral small modular reactors. *Nuclear Engineering and Technology*, 49(8):1669 – 1679, 2017.
- [24] BWR/4 Technology Manual (R-104B). Technical report, Nuclear Regulatory Commission.
- [25] RN Quade, PS Hunt, and WG Schuetzenduebel. The design of the fort st. vrain steam generators. *Nuclear Engineering and Design*, 26(1):118–134, 1974.
- [26] Zuoyi Zhang, Zongxin Wu, Dazhong Wang, Yuanhui Xu, Yuliang Sun, Fu Li, and Yujie Dong. Current status and technical description of Chinese 2 250 MW th HTR-PM demonstration plant. *Nuclear Engineering and Design*, 239:1212–1219, 2009.
- [27] Zuoyi Zhang, Yujie Dong, Fu Li, Zhengming Zhang, Haitao Wang, Xiaojin Huang, Hong Li, Bing Liu, Xinxin Wu, Hong Wang, Xingzhong Diao, Haiquan Zhang, and Jinhua Wang. The Shandong Shidao Bay 200 MW e High-Temperature Gas-Cooled Reactor Pebble-Bed Module (HTR-PM) Demonstration Power Plant: An Engineering and Technological Innovation. *Engineering*, 2:112–118, 2016.
- [28] John M. Cimbala Yunus A. Cengel. *Fluid Mechanics: Fundamentals and Applications*. 2nd edition, 2006.
- [29] Everett B. Woodruff, Herbert B. Lammers, and Thomas F. Lammers. *Steam Power Plant Operation Eighth Edition*. 2005.
- [30] D. S. Snell. The hydrogen-cooled turbine generator. *Electrical Engineering*, 59(1):35–50, Jan 1940.


- [31] S. Nagano, T. Kitajima, K. Yoshida, Y. Kazao, Y. Kabata, D. Murata, and K. Nagakura. Development of world's largest hydrogen-cooled turbine generator. In *IEEE Power Engineering Society Summer Meeting*, volume 2, pages 657–663 vol.2, July 2002.
- [32] Earl. Logan. *Handbook of turbomachinery*. M. Dekker, 1995.
- [33] G.F. Hewitt. *Heat Exchanger Design Handbook, 1998*. Number pts. 3-4 in Heat Exchanger Design Handbook, 1998. Begell House, 1998.
- [34] NRC: Diablo Canyon Power Plant, Unit 2.
- [35] Best Management Practice #10: Cooling Tower Management — Department of Energy.
- [36] AspenTech. Aspen HYSYS V8, 2012.
- [37] Operations Guide. Aspen HYSYS - Operations Guide. *AspenTech*, 2004.
- [38] AspenTech Documentation Team. Aspen HYSYS Customization Guide. 2006.
- [39] Lester. Haar, John S. (John Scott) Gallagher, George S. Kell, and National Standard Reference Data System (U.S.). *NBS/NRC steam tables : thermodynamic and transport properties and computer programs for vapor and liquid states of water in SI units*. Hemisphere Pub. Corp, 1984.
- [40] Juan Sebastian Lopez-Echeverry, Simn Reif-Acherman, and Eduard Araujo-Lopez. Peng-robinson equation of state: 40 years through cubics. 447, 05 2017.
- [41] WM Jacobi and YS Tang. Clinch river breeder reactor: a combined power and fuel source. Technical report, Westinghouse Electric Corp., Madison, Pa.(USA). Advanced Reactors Div., 1974.
- [42] Michael Mckellar and Shannon Bragg-sitton. Letter Report : Optimal Performance of Power Conversion Cycles and their Integration with Nuclear Reactors. 2014.
- [43] Michael G. McKellar. Power Cycles for the Generation of Electricity from a Next Generation Nuclear Plant. Technical Report 23843, Idaho National Laboratory, 2010.
- [44] Stephen Leeper. Wet Cooling Towers - Rule-Of-Thumb. pages 1–24, 1981.
- [45] J.M. Smith, H.C. Van Ness, and M.M. Abbott. *Introduction to chemical engineering thermodynamics*. McGraw-Hill chemical engineering series. McGraw-Hill, 2005.
- [46] J.G. Singer. *Combustion, fossil power systems: a reference book on fuel burning and steam generation*. Combustion Engineering, 1981.


APPENDIX A: HYSYS REPORT


This sections contains the process flow diagrams from the Aspen HYSYS model at 450°C. The Rankine cycle is shown first, followed by the report containing all stream information. Then the cooling tower and accompanying report is shown.







1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc			
2			Unit Set: Master's SI			
3			Date/Time: Tue Jul 31 17:36:15 2018			
4						
5						
6	Workbook: Case (Main)					
7						
8						
9	Heat Exchangers Fluid Pkg: All					
10						
11	Name	FW 6	FW 5	FW 4	FW 3	FW 2
12	Duty (MW)	7.597	6.588	6.808	4.983	4.924
13	Tube Side Feed Mass Flow (kg/s)	45.36	45.36	45.36	33.94	33.94
14	Shell Side Feed Mass Flow (kg/s)	3.895	6.596	9.353	4.112	4.098
15	Tube Inlet Temperature (C)	243.1	211.3	174.8	105.0	70.44
16	Tube Outlet Temperature (C)	277.6	243.1	208.6	139.5	105.0
17	Shell Inlet Temperature (C)	352.1	248.7	214.1	145.1	110.5
18	Shell Outlet Temperature (C)	248.7	216.8	191.3	144.3	76.00
19	Shell Side Pressure Drop (kPa)	134.6	77.69	41.37	8.321	2.916
20	Tube Side Pressure Drop (kPa)	280.2	285.9	82.31	17.79	18.16
21	Name	FW 1	Condenser			
22	Duty (MW)	4.900	58.65			
23	Tube Side Feed Mass Flow (kg/s)	33.94	2095			
24	Shell Side Feed Mass Flow (kg/s)	6.005	27.93			
25	Tube Inlet Temperature (C)	35.91	25.00 *			
26	Tube Outlet Temperature (C)	70.44	31.69			
27	Shell Inlet Temperature (C)	76.00	37.25			
28	Shell Outlet Temperature (C)	42.05	34.47			
29	Shell Side Pressure Drop (kPa)	0.8040	0.1273			
30	Tube Side Pressure Drop (kPa)	18.53	2.071			
31						
32	Heaters Fluid Pkg: All					
33	Name	Reheater	Steam Generator			
34	DUTY (MW)	10.96	89.04 *			
35	Feed Temperature (C)	352.1	277.6			
36	Product Temperature (C)	450.0	450.0 *			
37	Pressure Drop (kPa)	134.6	274.6			
38						
39	Dummy Streams Fluid Pkg: All					
40	Name	FW 6 Sat Steam	FW 5 Drain Set	FW 4 Sat Steam	FW 5 Sat Steam	FW 4 Drain Set
41	Temperature (C)	283.2	216.8	214.1	248.7	180.3
42	Pressure (MPa)	6.728	3.807	2.069	3.885	2.027
43	Name	FW 3 Drain Set	FW 2 Drain Set	FW Drain Set	FW 3 Sat Steam	FW 2 Sat Steam
44	Temperature (C)	110.5	76.00	41.47	145.1	110.5
45	Pressure (MPa)	0.4077	0.1429	3.940e-002	0.4160	0.1458
46	Name	FW 1 Sat Steam	FW 6 Drain Set	LP Stage 5 Saturated		
47	Temperature (C)	76.00	248.7	37.25		
48	Pressure (MPa)	4.020e-002	6.594	6.366e-003		
49						
50	Material Streams Fluid Pkg: All					
51	Name	Steam Generator Out	HP Turbine Out	Reheater In	FW 6 Steam In	Steam Generator In
52	Vapour Fraction	1.0000	1.0000	1.0000	1.0000	0.0000
53	Temperature (C)	450.0 *	352.1	352.1	352.1	277.6
54	Pressure (MPa)	13.46 *	6.728	6.728	6.728	13.73
55	Mass Flow (kg/s)	45.36	45.36	41.46	3.895	45.36
56	Name	FW 6 In	FW 6 Drain Out	Reheater Out	IP Stage 1 Out	FW 5 In
57	Vapour Fraction	0.0000	0.0000	1.0000	1.0000	0.0000
58	Temperature (C)	243.1	248.7	450.0	374.1	211.3
59	Pressure (MPa)	14.01	6.594	6.594	3.885 *	14.30
60	Mass Flow (kg/s)	45.36	3.895	41.46	41.46	45.36
61						
62						
63	Aspen Technology Inc.		Aspen HYSYS Version 9		Page 1 of 11	

1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc			
2			Unit Set: Master's SI			
3			Date/Time: Tue Jul 31 17:36:15 2018			
4						
5						
6	Workbook: Case (Main) (continued)					
7						
8						
9	Material Streams (continued)					
10					Fluid Pkg: All	
11	Name	Booster Pump In	IP Stage 2 In	FW 5 Steam In	FW 5 Drain Out	FW 5 Drain In
12	Vapour Fraction	0.0000	1.0000	1.0000	0.0000	0.0000
13	Temperature (C)	208.6	374.1	374.1	216.8	248.7
14	Pressure (MPa)	4.033	3.885	3.885	3.807	3.885
15	Mass Flow (kg/s)	45.36	38.76	2.700	6.596	3.895
16	Name	FW 5 Hot In	IP Stage 2 Out	LP Stage 1 In	FW 4 Steam In	FW 4 Drain In
17	Vapour Fraction	0.4932	1.0000	1.0000	1.0000	0.0068
18	Temperature (C)	248.7	293.5	293.5	293.5	214.1
19	Pressure (MPa)	3.885	2.069 *	2.069	2.069	2.069
20	Mass Flow (kg/s)	6.596	38.76	36.00	2.757	6.596
21	Name	FW 4 Hot In	FW 4 Drain Out	FW 4 In	LP Stage 1 Out	LP Stage 2 In
22	Vapour Fraction	0.3319	0.0000	0.0000	1.0000	1.0000
23	Temperature (C)	214.1	191.3	174.8	206.9	206.9
24	Pressure (MPa)	2.069	2.027	4.116	0.8718 *	0.8718
25	Mass Flow (kg/s)	9.353	9.353	45.36	36.00	33.94
26	Name	Deaerator Steam In	Deaerator Drain In	Deaerator Out	Deaerator In	LP Stage 2 Out
27	Vapour Fraction	1.0000	0.0376	0.0000 *	0.0000	0.9977
28	Temperature (C)	206.9	174.0	174.0	139.5	145.1
29	Pressure (MPa)	0.8718	0.8718	0.8718	0.8718	0.4160 *
30	Mass Flow (kg/s)	2.062	9.353	45.36	33.94	33.94
31	Name	LP Stage 3 In	FW 3 Steam In	FW 3 Drain Out	FW 3 In	LP Stage 3 Out
32	Vapour Fraction	0.9977	0.9977	0.4296	0.0000	0.9582
33	Temperature (C)	145.1	145.1	144.3	105.0	110.5
34	Pressure (MPa)	0.4160	0.4160	0.4077	0.8896	0.1458 *
35	Mass Flow (kg/s)	29.83	4.112	4.112	33.94	29.83
36	Name	LP Stage 4 In	FW 2 Steam In	FW 2 Hot In	FW 2 Drain In	FW 2 Drain Out
37	Vapour Fraction	0.9582	0.9582	0.4739	0.4756	0.0000
38	Temperature (C)	110.5	110.5	110.5	110.5	76.00
39	Pressure (MPa)	0.1458	0.1458	0.1458	0.1458	0.1429
40	Mass Flow (kg/s)	29.84	-1.414e-002	4.098	4.112	4.098
41	Name	FW 2 In	LP Stage 4 Out	FW 1 Drain In	LP Stage 5 In	FW 1 Steam In
42	Vapour Fraction	0.0000	0.9152	0.0000	0.9152	0.9152
43	Temperature (C)	70.44	76.00	76.00	76.00	76.00
44	Pressure (MPa)	0.9078	4.020e-002 *	4.020e-002	4.020e-002	4.020e-002
45	Mass Flow (kg/s)	33.94	29.84	4.098	27.93	1.907
46	Name	FW 1 Hot In	FW 1 Drain Out	LP Stage 5 Out	Condenser Out	Recycle In
47	Vapour Fraction	0.2907	0.0000	0.8654	0.0000	0.0090
48	Temperature (C)	76.00	42.05	37.25	34.47	36.88
49	Pressure (MPa)	4.020e-002	3.940e-002	6.366e-003	6.238e-003	6.238e-003
50	Mass Flow (kg/s)	6.005	6.005	27.93	27.93	6.005
51	Name	Condensate Pump In	FW 1 In	Recycle Out	Cooling Water In	Cooling Water Out
52	Vapour Fraction	0.0000	0.0000	0.0090	0.0000	0.0000
53	Temperature (C)	35.81 *	35.91	36.88	25.00 *	31.69
54	Pressure (MPa)	6.238e-003	0.9263	6.238e-003 *	0.1035	0.1015
55	Mass Flow (kg/s)	33.94	33.94	6.005	2095	2095
56	Name	Make Up Water				
57	Vapour Fraction	0.0000				
58	Temperature (C)	20.00				
59	Pressure (MPa)	0.1013				
60	Mass Flow (kg/s)	29.61				
61						
62						
63	Aspen Technology Inc.	Aspen HYSYS Version 9			Page 2 of 11	

1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc			
2			Unit Set: Master's SI			
3			Date/Time: Tue Jul 31 17:36:15 2018			
4						
5	Workbook: Case (Main) (continued)					
6	Energy Streams Fluid Pkg: All					
7	Name	HP Turbine Power	Reheater Heat	IP-1 Power	Boiler Pump Power	IP-2 Power
8	Heat Flow (MW)	7.027	10.96	5.789	0.7248	5.741
9	Name	LP-1 Power	Booster Pump Power	LP-2 Power	LP-3 Power	LP-4 Power
10	Heat Flow (MW)	5.574	0.2196	3.914	4.071	4.752
11	Name	LP-5 Power	Condensate Pump Po	Steam Generator Heat	Total Cooling Tower P	
12	Heat Flow (MW)	5.471	4.190e-002	89.04 *	0.2777	
13	Sets Fluid Pkg: All					
14	Name	FW 6 Tube Side Press	Steam Generator Pres	FW 6 Saturation Set P	FW 6 TTD	FW 6 Shell Side Press
15	Source Value (MPa)	14.01 *	13.73 *	6.728 *	283.2 *	6.728 *
16	Target Value (MPa)	13.73 *	13.46 *	6.728 *	277.6 *	6.594 *
17	Multiplier	0.9800 *	0.9800 *	1.000	1.000	0.9800 *
18	Increment (kPa)	0.0000	0.0000	0.0000	-5.556 *	0.0000
19	Name	FW 6 DCA	Reheater Pressure Dr	FW 5 Shell Side Press	FW 5 Tube Side Press	FW5 TTD
20	Source Value (MPa)	243.1 *	6.728 *	3.885 *	14.30 *	248.7 *
21	Target Value (MPa)	248.7 *	6.594 *	3.807 *	14.01 *	243.1 *
22	Multiplier	1.000	0.9800 *	0.9800 *	0.9800 *	1.000
23	Increment (kPa)	5.556 *	0.0000	0.0000	0.0000	-5.556 *
24	Name	FW 5 Pressure Set	FW 5 DCA	FW 5 Drain Pressure	FW 4 Pressure Set	FW 4 TTD
25	Source Value (MPa)	3.885 *	211.3 *	3.807 *	2.069 *	214.1 *
26	Target Value (MPa)	3.885 *	216.8 *	3.807 *	2.069 *	208.6 *
27	Multiplier	1.000	1.000	1.000	1.000	1.000
28	Increment (kPa)	0.0000	5.556 *	0.0000	0.0000	-5.556 *
29	Name	FW 4 Saturation Set P	FW 5 Saturation Set P	FW 4 Drain Pressure	FW 4 Shell Side Press	FW 4 Tube Side Press
30	Source Value (MPa)	2.069 *	3.885 *	2.027 *	2.069 *	4.116 *
31	Target Value (MPa)	2.069 *	3.885 *	2.027 *	2.027 *	4.033 *
32	Multiplier	1.000	1.000	1.000	0.9800 *	0.9800 *
33	Increment (kPa)	0.0000	0.0000	0.0000	0.0000	0.0000
34	Name	FW 4 DCA	FW 3 TTD	FW 3 Tube Side Press	FW 3 Shell Side Press	FW 3 Drain Pressure
35	Source Value (MPa)	174.8 *	145.1 *	0.8896 *	0.4160 *	0.4077 *
36	Target Value (MPa)	180.3 *	139.5 *	0.8718 *	0.4077 *	0.4077 *
37	Multiplier	1.000	1.000	0.9800 *	0.9800 *	1.000
38	Increment (kPa)	5.556 *	-5.556 *	0.0000	0.0000	0.0000
39	Name	FW 3 DCA	FW 2 Shell Side Press	FW 2 Tube Side Press	FW 2 DCA	FW 2 TTD
40	Source Value (MPa)	105.0 *	0.1458 *	0.9078 *	70.44 *	110.5 *
41	Target Value (MPa)	110.5 *	0.1429 *	0.8896 *	76.00 *	105.0 *
42	Multiplier	1.000	0.9800 *	0.9800 *	1.000	1.000
43	Increment (kPa)	5.556 *	0.0000	0.0000	5.556 *	-5.556 *
44	Name	FW 2 Pressure Set	FW 2 Drain Pressure	FW 1 Pressure Set	FW 1 Shell Side Press	FW 1 TTD
45	Source Value (MPa)	0.1458 *	0.1429 *	4.020e-002 *	4.020e-002 *	76.00 *
46	Target Value (MPa)	0.1458 *	0.1429 *	4.020e-002 *	3.940e-002 *	70.44 *
47	Multiplier	1.000	1.000	1.000	0.9800 *	1.000
48	Increment (kPa)	0.0000	0.0000	0.0000	0.0000	-5.556 *
49	Name	FW 1 Tube Side Press	FW 1 Drain Pressure	Condenser Tube Side	Condensing Pressure	FW 1 DCA
50	Source Value (MPa)	0.9263 *	3.940e-002 *	6.366e-003 *	6.238e-003 *	35.91 *
51	Target Value (MPa)	0.9078 *	3.940e-002 *	6.238e-003 *	6.238e-003 *	41.47 *
52	Multiplier	0.9800 *	1.000	0.9800 *	1.000	1.000
53	Increment (kPa)	0.0000	0.0000	0.0000	0.0000	5.556 *
54						
55						
56						
57						
58						
59						
60						
61						
62						
63	Aspen Technology Inc.		Aspen HYSYS Version 9		Page 3 of 11	


1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc			
2			Unit Set: Master's SI			
3			Date/Time: Tue Jul 31 17:36:15 2018			
4						
5						
6	Workbook: Case (Main) (continued)					
7						
8						
9	Sets (continued)					
10					Fluid Pkg: All	
11	Name	FW 3 Saturation Set P	FW 2 Saturation Set P	FW 1 Saturation Set P	High Temperature Set	FW 6 Drain Pressure
12	Source Value	(MPa) 0.4160 *	0.1458 *	4.020e-002 *	450.0 *	6.594 *
13	Target Value	(MPa) 0.4160 *	0.1458 *	4.020e-002 *	450.0 *	6.594 *
14	Multiplier	1.000	1.000	1.000	1.000	1.000
15	Increment	(kPa) 0.0000	0.0000	0.0000	0.0000	0.0000
16	Name	LP Stage 5 Pressure T	LP Stage 5 Pressure S			
17	Source Value	(MPa) 25.00 *	6.366e-003 *			
18	Target Value	(MPa) 37.25 *	6.366e-003 *			
19	Multiplier	1.000	1.000			
20	Increment	(kPa) 12.25 *	0.0000			
21						
22	Pumps					
23					Fluid Pkg: All	
24	Name	Boiler Pump	Booster Pump	Condensate Pump		
25	Power	(MW) 0.7248	0.2196	4.190e-002		
26	Feed Pressure	(MPa) 4.033	0.8718	6.238e-003		
27	Product Pressure	(MPa) 14.30	4.116	0.9263		
28	Product Temperature	(C) 211.3	174.8	35.91		
29	Adiabatic Efficiency	(%) 75.00	75.00 *	75.00		
30						
31	Expanders					
32					Fluid Pkg: All	
33	Name	HP Turbine	IP Turbine Stage 1	IP Turbine Stage 2	LP Turbine Stage 1	LP Turbine Stage 2
34	Power	(MW) 7.027	5.789	5.741	5.574	3.914
35	Feed Pressure	(MPa) 13.46 *	6.594	3.885	2.069	0.8718
36	Product Pressure	(MPa) 6.728	3.885 *	2.069 *	0.8718 *	0.4160 *
37	Product Temperature	(C) 352.1	374.1	293.5	206.9	145.1
38	Adiabatic Efficiency	85 *	90 *	90 *	80 *	80 *
39	Name	LP Turbine Stage 3	LP Turbine Stage 4	LP Turbine Stage 5		
40	Power	(MW) 4.071	4.752	5.471		
41	Feed Pressure	(MPa) 0.4160	0.1458	4.020e-002		
42	Product Pressure	(MPa) 0.1458 *	4.020e-002 *	6.366e-003		
43	Product Temperature	(C) 110.5	76.00	37.25		
44	Adiabatic Efficiency	75	80 *	80 *		
45						
46	Spreadsheet: FW 1 Calculations					
47					Units Set: Master's SI	
48						
49	CONNECTIONS					
50	Imported Variables					
51	Cell	Object	Variable Description		Value	
52	B3	Material Stream: FW 1 Steam In	Mass Enthalpy		-1.348e+004 kJ/kg	
53	B4	Material Stream: FW 1 Drain In	Mass Flow		4.098 kg/s	
54	B5	Material Stream: FW 1 Drain In	Mass Enthalpy		-1.561e+004 kJ/kg	
55	B6	Material Stream: FW 2 In	Mass Flow		33.94 kg/s	
56	B7	Material Stream: FW 1 In	Mass Enthalpy		-1.577e+004 kJ/kg	
57	B8	Material Stream: FW 2 In	Mass Enthalpy		-1.563e+004 kJ/kg	
58	B10	Material Stream: FW Drain Set	Mass Enthalpy		-1.575e+004 kJ/kg	
59	Exported Variables' Formula Results					
60	Cell	Object	Variable Description		Value	
61						
62						
63	Aspen Technology Inc.		Aspen HYSYS Version 9		Page 4 of 11	

1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc		
2			Unit Set: Master's SI		
3			Date/Time: Tue Jul 31 17:36:15 2018		
4					
5					
6	Spreadsheet: FW 1 Calculations (continued)				
7			Units Set:	Master's SI	
8					
9	PARAMETERS				
10	Exportable Variables				
11					
12					
13	Cell	Visible Name	Variable Description	Variable Type	Value
14	B1	B1:		---	<empty>
15	B2	B2:		---	<empty>
16	B9	B9:		Power	4.900 MW
17	B12	B12: Mass Flow	Mass Flow	Mass Flow	1.901 kg/s
18					
19	User Variables				
20	Name	Variable Type	Value	Units	Enabled
21	Try 2	User	---		No
22	Try 2	User	---		No
23					
24	FORMULAS				
25	Cell	Formula			Result
26	B9	=B6*(B8-B7)/1000			4.900 MW
27	B12	=(B9*1000+B4*(B10-B5))/(B3-B10)			1.901 kg/s
28					
29	Spreadsheet				
30		A	B	C	D
31	1		<empty> *		
32	2		<empty> *		<empty> *
33	3	Steam H *	-1.348e+004 kJ/kg *		<empty> *
34	4	Drain Flow *	4.098 kg/s *		<empty> *
35	5	Drain H *	-1.561e+004 kJ/kg *		<empty> *
36	6	FW Flow *	33.94 kg/s *		
37	7	FW H In *	-1.577e+004 kJ/kg *		
38	8	FW H Out *	-1.563e+004 kJ/kg *		
39	9	Duty *	4.900 MW *		
40	10	H Out *	-1.575e+004 kJ/kg *		
41	11		<empty> *		
42	12	Steam Flow *	1.901 kg/s *		
43					
44	Spreadsheet: FW 2 Calculations				
45			Units Set:	Master's SI	
46					
47	CONNECTIONS				
48	Imported Variables				
49					
50	Cell	Object	Variable Description	Value	
51	B3	Material Stream: FW 2 Steam In	Mass Enthalpy	-1.333e+004 kJ/kg	
52	B4	Material Stream: FW 2 Drain In	Mass Flow	4.112 kg/s	
53	B5	Material Stream: FW 2 Drain In	Mass Enthalpy	-1.440e+004 kJ/kg	
54	B6	Material Stream: FW 2 In	Mass Flow	33.94 kg/s	
55	B7	Material Stream: FW 2 In		-1.563e+004 kJ/kg	
56	B8	Material Stream: FW 3 In	Mass Enthalpy	-1.548e+004 kJ/kg	
57	B10	Material Stream: FW 2 Drain Set	Mass Enthalpy	-1.561e+004 kJ/kg	
58					
59	Exported Variables' Formula Results				
60	Cell	Object	Variable Description	Value	
61	B12	FW 2 Steam In	Mass Flow	-1.414e-002 kg/s	
62					
63	Aspen Technology Inc.		Aspen HYSYS Version 9	Page 5 of 11	


1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc		
2			Unit Set: Master's SI		
3			Date/Time: Tue Jul 31 17:36:15 2018		
4					
5					
6	Spreadsheet: FW 2 Calculations (continued)				
7			Units Set: Master's SI		
8					
9	PARAMETERS				
10	Exportable Variables				
11					
12					
13	Cell	Visible Name	Variable Description	Variable Type	Value
14	B1	B1:		---	<empty>
15	B2	B2:		---	<empty>
16	B9	B9:		Power	4.924 MW
17	B12	B12: Mass Flow	Mass Flow	Mass Flow	-1.414e-002 kg/s
18					
19	User Variables				
20	Name	Variable Type	Value	Units	Enabled
21	Try 2	User	---		No
22	Try 2	User	---		No
23					
24	FORMULAS				
25	Cell	Formula			Result
26	B9	=B6*(B8-B7)/1000			4.924 MW
27	B12	=(B9*1000+B4*(B10-B5))/(B3-B10)			-1.414e-002 kg/s
28					
29	Spreadsheet				
30		A	B	C	D
31	1		<empty> *		
32	2		<empty> *		
33	3	Steam H *	-1.333e+004 kJ/kg *		
34	4	Drain Flow *	4.112 kg/s *		
35	5	Drain H *	-1.440e+004 kJ/kg *		
36	6	FW Flow *	33.94 kg/s *		
37	7	FW H In *	-1.563e+004 kJ/kg *		
38	8	FW H Out *	-1.548e+004 kJ/kg *		
39	9	Duty *	4.924 MW *		
40	10	H Out *	-1.561e+004 kJ/kg *		
41	11				
42	12	Steam Flow *	-1.414e-002 kg/s *		
43					
44	Spreadsheet: FW 3 Calculations				
45			Units Set: Master's SI		
46					
47	CONNECTIONS				
48	Imported Variables				
49					
50	Cell	Object	Variable Description	Value	
51	B3	Material Stream: FW 3 Drain Set	Mass Enthalpy	-1.546e+004 kJ/kg	
52	B4	Material Stream: FW 3 Sat Steam	Mass Enthalpy	-1.425e+004 kJ/kg	
53	B5	Material Stream: FW 3 In	Mass Flow	33.94 kg/s	
54	B6	Material Stream: FW 3 In	Mass Enthalpy	-1.548e+004 kJ/kg	
55	B7	Material Stream: Deaerator In	Mass Enthalpy	-1.534e+004 kJ/kg	
56					
57	Exported Variables' Formula Results				
58	Cell	Object	Variable Description	Value	
59	B10	FW 3 Steam In	Mass Flow	4.112 kg/s	
60					
61					
62					
63	Aspen Technology Inc.		Aspen HYSYS Version 9	Page 6 of 11	


1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc		
2			Unit Set: Master's SI		
3			Date/Time: Tue Jul 31 17:36:15 2018		
4					
5					
6	Spreadsheet: FW 3 Calculations (continued)				
7			Units Set: Master's SI		
8					
9	PARAMETERS				
10	Exportable Variables				
11					
12					
13	Cell	Visible Name	Variable Description	Variable Type	Value
14	B1	B1:		---	<empty>
15	B2	B2:		---	<empty>
16	B8	B8:		Power	4.983 MW
17	B10	B10: Mass Flow	Mass Flow	Mass Flow	4.112 kg/s
18					
19	User Variables				
20	Name	Variable Type	Value	Units	Enabled
21	Try 2	User	---		No
22	Try 2	User	---		No
23					
24	FORMULAS				
25	Cell	Formula			Result
26	B8	=B5*(B7-B6)/1000			4.983 MW
27	B10	=(B8*1000)/(B4-B3)			4.112 kg/s
28					
29	Spreadsheet				
30		A	B	C	D
31	1		<empty> *		
32	2		<empty> *		
33	3	H out *	-1.546e+004 kJ/kg *		
34	4	H Steam *	-1.425e+004 kJ/kg *		
35	5	FW Flow *	33.94 kg/s *		
36	6	FW H IN *	-1.548e+004 kJ/kg *		
37	7	FW H Out *	-1.534e+004 kJ/kg *		
38	8	Duty *	4.983 MW *		
39	9				
40	10	Steam Flow *	4.112 kg/s *		
41					
42	Spreadsheet: FW 4 Calculations				
43			Units Set: Master's SI		
44					
45	CONNECTIONS				
46					
47	Imported Variables				
48	Cell	Object	Variable Description	Value	
49	B1	Material Stream: FW 4 Drain In	Mass Flow	6.596 kg/s	
50	B2	Material Stream: FW 4 Drain In	Mass Enthalpy	-1.499e+004 kJ/kg	
51	B3	Material Stream: FW 4 Drain Set	Mass Enthalpy	-1.516e+004 kJ/kg	
52	B4	Material Stream: FW 4 Steam In	Mass Enthalpy	-1.292e+004 kJ/kg	
53	B5	Material Stream: FW 4 In	Mass Flow	45.36 kg/s	
54	B6	Material Stream: FW 4 In	Mass Enthalpy	-1.518e+004 kJ/kg	
55	B7	Material Stream: Booster Pump In	Mass Enthalpy	-1.503e+004 kJ/kg	
56					
57	Exported Variables' Formula Results				
58	Cell	Object	Variable Description	Value	
59	B10	FW 4 Steam In	Mass Flow	2.757 kg/s	
60					
61					
62					
63	Aspen Technology Inc.		Aspen HYSYS Version 9	Page 7 of 11	


1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name:	450 c for hysys report.hsc	
2			Unit Set:	Master's SI	
3			Date/Time:	Tue Jul 31 17:36:15 2018	
4					
5					
6	Spreadsheet: FW 4 Calculations (continued)			Units Set:	Master's SI
7					
8	PARAMETERS				
9					
10	Exportable Variables				
11					
12					
13	Cell	Visible Name	Variable Description	Variable Type	Value
14	B8	B8:		Power	6.808 MW
15	B10	B10: Mass Flow	Mass Flow	Mass Flow	2.757 kg/s
16					
17	User Variables				
18	Name	Variable Type	Value	Units	Enabled
19	Try 2	User	---		No
20	Try 2	User	---		No
21					
22	FORMULAS				
23	Cell	Formula			Result
24	B8	=B5*(B7-B6)/1000			6.808 MW
25	B10	=(B8*1000+B1*(B3-B2))/(B4-B2)			2.757 kg/s
26					
27	Spreadsheet				
28		A	B	C	D
29	1	Drain Flow *	6.596 kg/s *		
30	2	Drain H *	-1.499e+004 kJ/kg *		
31	3	Hout *	-1.516e+004 kJ/kg *		
32	4	H Steam *	-1.292e+004 kJ/kg *		
33	5	FW Flow *	45.36 kg/s *		
34	6	FW H In *	-1.518e+004 kJ/kg *		
35	7	FW H Out *	-1.503e+004 kJ/kg *		
36	8	Duty *	6.808 MW *		
37	9				
38	10	Steam Flow *	2.757 kg/s *		
39					
40	Spreadsheet: FW 5 Calculations			Units Set:	Master's SI
41					
42	CONNECTIONS				
43					
44	Imported Variables				
45					
46	Cell	Object	Variable Description	Value	
47	B1	Material Stream: FW 5 Drain In	Mass Flow	3.895 kg/s	
48	B2	Material Stream: FW 5 Drain In	Mass Enthalpy	-1.485e+004 kJ/kg	
49	B3	Material Stream: FW 5 Drain Set	Mass Enthalpy	-1.499e+004 kJ/kg	
50	B4	Material Stream: FW 5 Steam In	Mass Enthalpy	-1.277e+004 kJ/kg	
51	B5	Material Stream: Booster Pump In	Mass Flow	45.36 kg/s	
52	B6	Material Stream: FW 5 In	Mass Enthalpy	-1.502e+004 kJ/kg	
53	B7	Material Stream: FW 6 In	Mass Enthalpy	-1.487e+004 kJ/kg	
54					
55	Exported Variables' Formula Results				
56	Cell	Object	Variable Description	Value	
57	D1	FW 5 Steam In	Mass Flow	2.700 kg/s	
58					
59					
60					
61					
62					
63	Aspen Technology Inc.		Aspen HYSYS Version 9	Page 8 of 11	


1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc		
2			Unit Set: Master's SI		
3			Date/Time: Tue Jul 31 17:36:15 2018		
4					
5					
6	Spreadsheet: FW 5 Calculations (continued)				
7			Units Set: Master's SI		
8					
9	PARAMETERS				
10	Exportable Variables				
11					
12					
13	Cell	Visible Name	Variable Description	Variable Type	Value
14	B8	B8: Mass Heat Capacity	Mass Heat Capacity	Power	6.588 MW
15	D1	D1: Mass Flow	Mass Flow	Mass Flow	2.700 kg/s
16					
17	User Variables				
18	Name	Variable Type	Value	Units	Enabled
19	Try 2	User	---		No
20	Try 2	User	---		No
21					
22	FORMULAS				
23	Cell	Formula			Result
24	B8	=B5*(B7-B6)/1000			6.588 MW
25	D1	=(B8*1000+B1*(B3-B2))/(B4-B3)			2.700 kg/s
26					
27	Spreadsheet				
28		A	B	C	D
29	1	Mass From 6 *	3.895 kg/s *	Steam Mass *	2.700 kg/s *
30	2	H 6 *	-1.485e+004 kJ/kg *		
31	3	H Out *	-1.499e+004 kJ/kg *		
32	4	H Steam *	-1.277e+004 kJ/kg *		
33	5	FW Mass Flow *	45.36 kg/s *		
34	6	FW H In *	-1.502e+004 kJ/kg *		
35	7	FW H out *	-1.487e+004 kJ/kg *		
36	8	Duty *	6.588 MW *		
37	9		<empty> *		
38	10		<empty> *		
39					
40	Spreadsheet: FW 6 Calculations				
41			Units Set: Master's SI		
42					
43	CONNECTIONS				
44					
45	Imported Variables				
46	Cell	Object	Variable Description	Value	
47	B1	Material Stream: FW 6 Drain Set	Mass Enthalpy	-1.485e+004 kJ/kg	
48	B2	Material Stream: FW 6 Steam In	Mass Enthalpy	-1.290e+004 kJ/kg	
49	B3	Material Stream: FW 6 In	Mass Flow	45.36 kg/s	
50	B4	Material Stream: FW 6 In	Mass Enthalpy	-1.487e+004 kJ/kg	
51	B5	Material Stream: Steam Generator In	Mass Enthalpy	-1.470e+004 kJ/kg	
52					
53	Exported Variables' Formula Results				
54	Cell	Object	Variable Description	Value	
55	B8	FW 6 Steam In	Mass Flow	3.895 kg/s	
56					
57	PARAMETERS				
58					
59	Exportable Variables				
60	Cell	Visible Name	Variable Description	Variable Type	Value
61	B6	B6:		Power	7.597 MW
62	B8	B8: Mass Flow	Mass Flow	Mass Flow	3.895 kg/s
63	Aspen Technology Inc.		Aspen HYSYS Version 9		Page 9 of 11


1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name:	450 c for hysys report.hsc	
2			Unit Set:	Master's SI	
3			Date/Time:	Tue Jul 31 17:36:15 2018	
4					
5					
6	Spreadsheet: FW 6 Calculations (continued)			Units Set:	Master's SI
7					
8					
9	User Variables				
10					
11	Name	Variable Type	Value	Units	Enabled
12	Try 2	User	---		No
13	Try 2	User	---		No
14					
15	FORMULAS				
16	Cell	Formula		Result	
17	B6	=B3*(B5-B4)/1000		7.597 MW	
18	B8	=B6*1000/(B2-B1)		3.895 kg/s	
19					
20	Spreadsheet				
21		A	B	C	D
22	1	H Out *	-1.485e+004 kJ/kg *		
23	2	H Steam *	-1.290e+004 kJ/kg *		
24	3	FW Mass Flow *	45,36 kg/s *		
25	4	FW H In *	-1.487e+004 kJ/kg *		
26	5	FW H Out *	-1.470e+004 kJ/kg *		
27	6	Duty *	7.597 MW *		
28	7				
29	8	Steam Mass *	3.895 kg/s *		
30	9				
31	10				
32					
33	Spreadsheet: Thermal			Units Set:	Master's SI
34					
35	CONNECTIONS				
36					
37	Imported Variables				
38					
39	Cell	Object	Variable Description	Value	
40	D6	Energy Stream: Steam Generator Heat	Power	89.04 MW	
41	D7	Energy Stream: Reheater Heat	Power	10.96 MW	
42	B2	Energy Stream: HP Turbine Power	Power	7.027 MW	
43	B3	Energy Stream: IP-1 Power	Power	5.789 MW	
44	B4	Energy Stream: IP-2 Power	Power	5.741 MW	
45	D2	Energy Stream: Boiler Pump Power	Power	0.7248 MW	
46	B5	Energy Stream: LP-1 Power	Power	5.574 MW	
47	B6	Energy Stream: LP-2 Power	Power	3.914 MW	
48	D3	Energy Stream: Booster Pump Power	Power	0.2196 MW	
49	B7	Energy Stream: LP-3 Power	Power	4.071 MW	
50	B8	Energy Stream: LP-4 Power	Power	4.752 MW	
51	B9	Energy Stream: LP-5 Power	Power	5.471 MW	
52	D4	Energy Stream: Condensate Pump Power	Power	4.190e-002 MW	
53	B10	Energy Stream: Total Cooling Tower Power	Power	0.2777 MW	
54					
55	Exported Variables' Formula Results				
56	Cell	Object	Variable Description	Value	
57					
58	PARAMETERS				
59					
60	Exportable Variables				
61	Cell	Visible Name	Variable Description	Variable Type	Value
62	A2	A2:		---	1.000
63	Aspen Technology Inc.		Aspen HYSYS Version 9		Page 10 of 11

1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc		
2			Unit Set: Master's SI		
3			Date/Time: Tue Jul 31 17:36:15 2018		
4					
5					
6	Spreadsheet: Thermal (continued)			Units Set: Master's SI	
7					
8					
9	PARAMETERS				
10					
11	Exportable Variables				
12					
13	Cell	Visible Name	Variable Description	Variable Type	Value
14	A3	A3:		---	2.000
15	A4	A4:		---	3.000
16	A5	A5:		---	4.000
17	A6	A6:		---	5.000
18	A7	A7:		---	6.000
19	A8	A8:		---	7.000
20	A9	A9:		---	8.000
21	C2	C2:		---	1.000
22	C3	C3:		---	2.000
23	C4	C4:		---	3.000
24	C6	C6:		---	1.000
25	C7	C7:		---	2.000
26	D9	D9:		Percent	41.08
27	D10	D10: Reactor Power	Reactor Power	Power	100.0 MW
28					
29	User Variables				
30	Name	Variable Type	Value	Units	Enabled
31	Try 2	User	---		No
32	Try 2	User	---		No
33					
34	FORMULAS				
35	Cell	Formula			Result
36	D9	=(B2+B3+B4+B5+B6+B7+B8+B9-D2-D3-D4-B10)/(D6+D7)*100			41.08
37	D10	=D6+D7			100.0 MW
38					
39	Spreadsheet				
40		A	B	C	D
41	1		Power Out *		Parasitic *
42	2	1.000 *	7.027 MW *	1.000 *	0.7248 MW *
43	3	2.000 *	5.789 MW *	2.000 *	0.2196 MW *
44	4	3.000 *	5.741 MW *	3.000 *	4.190e-002 MW *
45	5	4.000 *	5.574 MW *		Heat In *
46	6	5.000 *	3.914 MW *	1.000 *	89.04 MW *
47	7	6.000 *	4.071 MW *	2.000 *	10.96 MW *
48	8	7.000 *	4.752 MW *		Thermal Efficiency-ish *
49	9	8.000 *	5.471 MW *		41.08 *
50	10	Cooling Tower *	0.2777 MW *	Reactor Heat *	100.0 MW *
51					
52					
53					
54					
55					
56					
57					
58					
59					
60					
61					
62					
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1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc			
2			Unit Set: Master's SI			
3			Date/Time: Tue Jul 31 17:40:16 2018			
4						
5						
6	Workbook: Cooling Tower (TPL1)					
7						
8						
9	Material Streams Fluid Pkg: All					
10						
11	Name	Cooling Water Out	Dry Air	Water Vapor	Moist Air	Sat Press @ Moist Air
12	Vapour Fraction	0.0000	---	---	1.0000	1.0000 *
13	Temperature (C)	31.69	---	---	20.00 *	20.00
14	Pressure (MPa)	0.1015	0.1013	0.1013	0.1013 *	2.339e-003
15	Mass Flow (kg/s)	2095	1631	11.85	1643 *	---
16	Name	Air Draft	Mix High Pressure	Dew Point	Mix Ambient	Saturated Air
17	Vapour Fraction	1.0000	0.3364	1.0000 *	0.3364	1.0000
18	Temperature (C)	20.16	25.08	9.396	25.07	25.07
19	Pressure (MPa)	0.1015	0.1015	0.1013	0.1013	0.1013
20	Mass Flow (kg/s)	1643	3738	1643	3738	1664
21	Name	Liquid Water	Entrained Water	Blowdown	Water Return	Water Entrained Air
22	Vapour Fraction	0.0000	0.0000	0.0000	0.0000	0.9980
23	Temperature (C)	25.07	25.07	25.07	25.07	25.07
24	Pressure (MPa)	0.1013	0.1013	0.1013	0.1013	0.1013
25	Mass Flow (kg/s)	2074	2.095	6.566	2065	1666
26	Name	Make Up Water	Cooling Water Pump In	Cooling Water In a	Cooling Water In	
27	Vapour Fraction	0.0000	0.0000	0.0000	0.0000	
28	Temperature (C)	20.00 *	25.00	25.00	25.00 *	
29	Pressure (MPa)	0.1013	0.1013	0.1035	0.1035	
30	Mass Flow (kg/s)	29.61	2095	2095	2095	
31	Pumps Fluid Pkg: All					
32						
33	Name	Cooling Water Pump				
34	Power (MW)	6.153e-003				
35	Feed Pressure (MPa)	0.1013				
36	Product Pressure (MPa)	0.1035				
37	Product Temperature (C)	25.00				
38	Adiabatic Efficiency (%)	75.00 *				
39	Separators Fluid Pkg: All					
40						
41	Name	Air Water Separator				
42	Vessel Temperature (C)	25.07				
43	Vessel Pressure (MPa)	0.1013				
44	Vapour Mass Flow (kg/s)	1664				
45	Liquid Mass Flow (kg/s)	2074				
46	Compressors Fluid Pkg: All					
47						
48	Name	Air Fan				
49	Power (MW)	0.2715				
50	Feed Pressure (MPa)	0.1013 *				
51	Product Pressure (MPa)	0.1015				
52	Product Temperature (C)	20.16				
53	Adiabatic Efficiency	75				
54	Compositions Fluid Pkg: All					
55						
56	Name	Cooling Water Out	Dry Air	Water Vapor	Moist Air	Sat Press @ Moist Air
57	Master Comp Mole Frac (H2O)	1.0000	0.0000 *	1.0000 *	0.0115	1.0000 *
58	Master Comp Mole Frac (Nitrogen)	0.0000	0.7900 *	0.0000 *	0.7809	***
59	Master Comp Mole Frac (Oxygen)	0.0000	0.2100 *	0.0000 *	0.2076	***
60						
61						
62						
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1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc			
2			Unit Set: Master's SI			
3			Date/Time: Tue Jul 31 17:40:16 2018			
4						
5						
6	Workbook: Cooling Tower (TPL1) (continued)					
7						
8						
9	Compositions (continued) Fluid Pkg: All					
10						
11	Name	Air Draft	Mix High Pressure	Dew Point	Mix Ambient	Saturated Air
12	Master Comp Mole Frac (H2O)	0.0115	0.6741	0.0115	0.6741	0.0312
13	Master Comp Mole Frac (Nitrogen)	0.7809	0.2575	0.7809	0.2575	0.7653
14	Master Comp Mole Frac (Oxygen)	0.2076	0.0684	0.2076	0.0684	0.2034
15	Name	Liquid Water	Entrained Water	Blowdown	Water Return	Water Entrained Air
16	Master Comp Mole Frac (H2O)	1.0000	1.0000	1.0000	1.0000	0.0332
17	Master Comp Mole Frac (Nitrogen)	0.0000	0.0000	0.0000	0.0000	0.7638
18	Master Comp Mole Frac (Oxygen)	0.0000	0.0000	0.0000	0.0000	0.2030
19	Name	Make Up Water	Cooling Water Pump In	Cooling Water In a	Cooling Water In	
20	Master Comp Mole Frac (H2O)	1.0000 *	1.0000	1.0000	1.0000 *	
21	Master Comp Mole Frac (Nitrogen)	0.0000 *	0.0000	0.0000	0.0000 *	
22	Master Comp Mole Frac (Oxygen)	0.0000 *	0.0000	0.0000	0.0000 *	
23	Energy Streams Fluid Pkg: All					
24						
25	Name	Air Fan Power	Cooling Water Pump F	Total Cooling Tower P		
26	Heat Flow (MW)	0.2715	6.153e-003	0.2777		
27	Spreadsheet: Moist Air Conditions Units Set: NGNP1					
28						
29						
30	CONNECTIONS					
31						
32	Imported Variables					
33						
34	Cell	Object	Variable Description		Value	
35	D2	Material Stream: Dew Point	Temperature		9.396 C	
36	D1	Material Stream: Moist Air	Temperature		20.00 C	
37	B9	Material Stream: Moist Air	Mass Flow		1643 kg/s	
38	B7	Pure Component: Air	Molecular Weight		28.95	
39	B6	Pure Component: H2O	Molecular Weight		18.02	
40	B4	Material Stream: Moist Air	Pressure		0.1013 MPa	
41	B2	Material Stream: Sat Press @ Moist Air Ter	Pressure		2.339e-003 MPa	
42	Exported Variables' Formula Results					
43						
44	Cell	Object	Variable Description		Value	
45	B10	Dry Air	Mass Flow		1631 kg/s	
46	PARAMETERS					
47						
48	Exportable Variables					
49						
50	Cell	Visible Name	Variable Description	Variable Type	Value	
51	B1	B1: Relative Humidity	Relative Humidity	Percent	50.00	
52	B3	B3: Water Vapor Pressure	Water Vapor Pressure	Pressure	1.169e-003 MPa	
53	B5	B5: Air Pressure	Air Pressure	Pressure	0.1002 MPa	
54	B8	B8: Humidity Ratio	Humidity Ratio	---	7.266e-003	
55	B10	B10: Mass Flow	Mass Flow	Mass Flow	1631 kg/s	
56	D3	D3:		Temperature	<empty>	
57	User Variables					
58						
59	Name	Variable Type	Value	Units	Enabled	
60	Try 2	User	---		No	
61	Try 2	User	---		No	
62						
63	Aspen Technology Inc.		Aspen HYSYS Version 9		Page 2 of 4	

1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name: 450 c for hysys report.hsc	
2			Unit Set: Master's SI	
3			Date/Time: Tue Jul 31 17:40:16 2018	
4				
5				
6	Spreadsheet: Moist Air Conditions (continued)			
7				Units Set: NGNP1
8				
9	FORMULAS			
10				
11	Cell	Formula		Result
12	B3	=B1/100*B2		1.169e-003 MPa
13	B5	=B4-B3		0.1002 MPa
14	B8	=B6*B3/(B7*B5)		7.266e-003
15	B10	=B9/(1+B8)		1631 kg/s
16				
17	Spreadsheet			
18		A	B	C
19	1	Relative Humidity *	50.00 *	Dry Bulb Temperature *
20	2	Water Vapor Sat Pres @ Moist Air Temp *	2.339e-003 MPa *	Dew Point Temperature *
21	3	Water Vapor Pressure *	1.169e-003 MPa *	<empty> *
22	4	Moist Air Pressure *	0.1013 MPa *	
23	5	Air Pressure *	0.1002 MPa *	
24	6	Molecular Weight of Water *	18.02 *	
25	7	Molecular Weight of Air *	28.95 *	
26	8	Humidity Ratio *	7.266e-003 *	
27	9	Moist Air Mass Flow *	1643 kg/s *	
28	10	Dry Air Mass Flow *	1631 kg/s *	
29				
30	Spreadsheet: Water Cooling Calcs			
31				Units Set: NGNP1
32				
33	CONNECTIONS			
34				
35	Imported Variables			
36	Cell	Object	Variable Description	Value
37	D1	Energy Stream: Cooling Water Pump Pow	Power	6.153 kW
38	B7	Material Stream: Water Return	Mass Flow	2065 kg/s
39	B3	Material Stream: Saturated Air	Master Comp Mass Flow (H2O)	32.8302 kg/s
40	B5	Material Stream: Moist Air	Actual Volume Flow	1375 m3/s
41	B1	Material Stream: Cooling Water Out	Mass Flow	2095 kg/s
42				
43	Exported Variables' Formula Results			
44	Cell	Object	Variable Description	Value
45	D2	Total Cooling Tower Power	Power	277.7 kW
46	B8	Make Up Water	Mass Flow	29.61 kg/s
47	B4	Blowdown	Mass Flow	6.566 kg/s
48	B6	Air Fan Power	Power	271.5 kW
49	B2	Entrained Water	Mass Flow	2.095 kg/s
50				
51	PARAMETERS			
52				
53	Exportable Variables			
54	Cell	Visible Name	Variable Description	Variable Type
55	B2	B2: Mass Flow	Mass Flow	Mass Flow
56	B4	B4: Mass Flow	Mass Flow	Mass Flow
57	B6	B6: Power	Power	Power
58	B8	B8: Mass Flow	Mass Flow	Mass Flow
59	B9	B9:		---
60	D2	D2: Power	Power	Power
61	D7	D7:		---
62				
63	Aspen Technology Inc.		Aspen HYSYS Version 9	Page 3 of 4

1	 UNIVERSITY OF IDAHO Bedford, MA USA		Case Name:	450 c for hysys report.hsc	
2			Unit Set:	Master's SI	
3			Date/Time:	Tue Jul 31 17:40:16 2018	
4					
5	Spreadsheet: Water Cooling Calcs (continued)				
6				Units Set:	NGNP1
7	User Variables				
8					
9	FORMULAS				
10					
11	Name	Variable Type	Value	Units	Enabled
12	Try 2	User	---		No
13	Try 2	User	---		No
14	FORMULAS				
15					
16	Cell	Formula		Result	
17	B2	=.001*B1		2.095 kg/s	
18	B4	=.20*B3		6.566 kg/s	
19	B6	=B5/5.063		271.5 kW	
20	B8	=B1-B7		29.61 kg/s	
21	D2	=(D1+B6)		277.7 kW	
22	Spreadsheet				
23					
24		A	B	C	D
25	1	Cooling Water Mass Flow *	2095 kg/s *	Cooling Water Pump Power *	6.153 kW *
26	2	Entrained Water Mass Flow *	2.095 kg/s *	Total Cooling Tower Power *	277.7 kW *
27	3	Evaporated Water Mass Flow *	32.8302 kg/s *		
28	4	Blowdown Mass Flow *	6.566 kg/s *		
29	5	Moist Air Volume Flow *	1375 m3/s *		
30	6	Fan Power *	271.5 kW *		
31	7	Water Return Flow *	2065 kg/s *		<empty> *
32	8	Make Up Water Flow *	29.61 kg/s *		
33	9		<empty> *		
34	10		<empty> *		
35					
36					
37					
38					
39					
40					
41					
42					
43					
44					
45					
46					
47					
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63	Aspen Technology Inc.		Aspen HYSYS Version 9		Page 4 of 4

APPENDIX B: USER VARIABLE CODES

This appendix contains the user variable codes implemented in the HYSYS. The information includes not only the syntax, but the code type information as well as what unit operation it is implemented in.

Name: Adjust Min/Max
Tag: Adjust

Type: Code Only

Activation:
Enable In: Condenser Out

Code:

```

Sub PostExecute()
    Dim Source As HYSYS.ProcessStream 'Defines Source as a HYSYS process
stream
    Dim Adjust As HYSYS.AdjustOp      'Defines Adjust as a HYSYS adjust
operation
    Dim Cool As HYSYS.ProcessStream   'Defines Cool as a HYSYS process
stream
    Dim SatTemp As Double             'Defines SatTemp as a double
    Dim MaxTemp As Double             'Defines MaxTemp as a double
    Dim CoolTemp As Double            'Defines CoolTemp as a double
    Dim MinTemp As Double             'Defines Min Temp as a double
    Dim myFluid As Object              'Defines myFluid as an object

    On Error GoTo GetOut              'In the case of an error

    Set Source=ActiveObject
'Sets Source as the active object, in this case the stream
    Set Adjust=Source.Flowsheet.Operations.Item("Bottom Adjust")
'Sets Adjust as the "Bottom Adjust" operation
    Set myFluid=Source.DuplicateFluid
'Duplicates all fluid properties of Source
    Set Cool=Source.Flowsheet.MaterialStreams.Item("Cooling Water In")
'Sets Cool as the stream "Cooling Water In"
    'Pressure Vapor Falsh Operation
    FS=0                               'Sets a criteria for the
flash
    FS=myFluid.PVFlash(Source.PressureValue, 0) 'Performs a pressure vapor
fraction flash to the specified pressure and vapor fraction
    If FS=0 Then 'fsFlashOK            'Checks to see if the flash
was successful
        SatTemp=myFluid.Temperature.GetValue 'Gets the saturated
temperature of the flashed fluid
    End If

    MaxTemp=SatTemp-0.001              'Sets the MaxTemp slightly
less than saturated
    CoolTemp=Cool.Temperature.GetValue 'Sets the CoolTemp as the
actual temperature of the stream
    MinTemp=CoolTemp+8                 'Sets the MinTemp as 8
degrees higher than the cooling water in
    Adjust.AdjustedVarMax.SetValue(MaxTemp) 'Sets the value of the
Adjusted Variable Maximum value as MaxTemp
    Adjust.AdjustedVarMin.SetValue(MinTemp) 'Sets the value of the
Adjusted Variable Minimum value as MinTemp

    GetOut: 'Escape Route
    Exit Sub
End Sub

```

Name: High Pressure
Tag: Pmax

Type: Enumeration
Dimensions: Scalar

Enumeration Values:

Label	Value
Start	0
PressureAdjust	1
Done	2

Activation:

User Enabled in: Bottom Adjust

Code:

```
Const Start=0
```

```
Const PressureAdjust=1
```

```
Const Done=2
```

```
Sub PreExecute ()
    Dim Adjust As HYSYS.AdjustOp      'Adjust is defined as a HYSYS adjust
operation
    Set Adjust=ActiveObject           'Adjust is set to the active object
    On Error GoTo GetOut              'Escape path in the case of an error
    Adjust.IsIgnored=False            'Ensures that the adjust attempts to
solve everytime
    GetOut:                            'Escape path
    Exit Sub                          'Leaves
End Sub
```

```
Sub PostExecute ()
    Dim Adjust As HYSYS.AdjustOp      'Adjust is defined as a HYSYS
adjust operation
    Dim PumpIn As HYSYS.ProcessStream 'PumpIn is defined as a HYSYS
process stream
    Dim Steam As HYSYS.ProcessStream  'Steam is defined as a HYSYS
process stream
    Dim SGOut As HYSYS.ProcessStream  'SGOut is defined as a HYSYS
process stream
    Dim Sheet As HYSYS.SpreadsheetOp  'Sheet is defined as a HYSYS
spreadsheet operation
    Dim Mdot As HYSYS.SpreadsheetCell 'Mdot is defined as a HYSYS
spreadsheet cell
    Dim thisvar As HYSYS.RealVariable 'thisvar is defined as a HYSYS real
variable

    On Error GoTo GetOut              'Escape path in the case of an
error

    Set Adjust=ActiveObject           'Adjust is set to the active object
    Set PumpIn=Adjust.Flowsheet.MaterialStreams.Item("Condensate Pump In")
'PumpIn is set as stream "Pump In"
    Set Steam=Adjust.Flowsheet.MaterialStreams.Item("FW 1 Steam In")
'Steam is set as stream "To IP Mix"
```

```

    Set SGOut=Adjust.Flowsheet.MaterialStreams.Item("Steam Generator Out")
'SGOut is set as stream "SG Heat"
    Set Sheet=Adjust.Flowsheet.Operations.Item("FW 1 Calculations") 'Sheet
is set as spreadsheet "FW 1 Calculations"
    Set Mdot=Sheet.Cell("B12") 'Mdot
is set as cell B14 of "Sheet"
    Set thisvar=ActiveVariableWrapper.Variable
'thisvar is set as this variable
.....
''
    'Redirect Code Based On State
    If thisvar=Start Then GoTo Entrance
    If thisvar=PressureAdjust Then GoTo One
    If thisvar=Done Then GoTo Two
.....
''
    'Starting state advances the code
    Entrance:
    thisvar=PressureAdjust
    Exit Sub
.....
''
    'This adjusts the pressure of the system by changing the pump outlet
pressure
    One:
    Tmax=SGOut.Temperature.GetValue("C") 'Finds the hot steam
temperature
    Slope=0.0468 'This is the slope for the
linear fit of P vs T
    InterceptUpper=-6.889+1.925 'This is the y-intercept
plus two errors of the linear fit of P vs T
    Pmax=Tmax*Slope+Intercept 'This calculates the
maximum pressure on the steam generator outlet
    If PumpIn.TemperatureValue=Adjust.AdjustedVarMaxValue Then 'Tests
if the adjusted variable has hit the maximum value
        GoTo Maximum 'Tells
the code to go to Maximum
    ElseIf PumpIn.TemperatureValue=Adjust.AdjustedVarMinValue Then 'Tests
if the adjusted variable has hit the minimum value
        GoTo Minimum 'Tells
the code to go to Minimum
    ElseIf SGOut.Pressure.GetValue("MPa")>Pmax Then 'Tests
if the maximum pressure has been surpass
        GoTo MaxReached 'Tells
the code to go to MaxReached
    Else 'If not
        GoTo Continue 'Tells
the code to go to Continue
    End If

    Maximum: 'If the
maximum temperature was hit
    Adjust.IsIgnored=True 'Causes the
adjust to be ignored to avoid the dialog box
    PumpIn.Temperature.SetValue(PumpIn.TemperatureValue-1) 'Changes
the adjusted variable to avoid another maximum

```



```
.....  
''  
    GetOut:      'Error escape goat  
    Exit Sub     'Exit the sub so the code doesn't break  
End Sub
```

Name: Pressures
 Tag: Pressures

Type: Real
 Dimensions: Scalar

Activation:
 Enable In: Bottom Adjust

Code:

```

Sub PostExecute ()
.....
''
  'Define all of the required HYSYS objects and others
  Dim Adjust As HYSYS.AdjustOp
  Dim FW1In As HYSYS.ProcessStream
  Dim HPOut As HYSYS.ProcessStream
  Dim IP1Out As HYSYS.ProcessStream
  Dim IP2Out As HYSYS.ProcessStream
  Dim LP1Out As HYSYS.ProcessStream
  Dim LP2Out As HYSYS.ProcessStream
  Dim LP3Out As HYSYS.ProcessStream
  Dim LP4Out As HYSYS.ProcessStream
  Dim Ref As Object
  Dim Test As Object
.....
''
  'In the case of an error
  On Error GoTo GetOut
.....
''
  'Assign all of the defined variables to the corresponding HYSYS streams
and modules
  Set Adjust=ActiveObject
  Set Ref=Adjust.Flowsheet.MaterialStreams
  Set FW1In=Ref.Item("FW 1 In")
  Set HPOut=Ref.Item("HP Turbine Out")
  Set IP1Out=Ref.Item("IP Stage 1 Out")
  Set IP2Out=Ref.Item("IP Stage 2 Out")
  Set LP1Out=Ref.Item("LP Stage 1 Out")
  Set LP2Out=Ref.Item("LP Stage 2 Out")
  Set LP3Out=Ref.Item("LP Stage 3 Out")
  Set LP4Out=Ref.Item("LP Stage 4 Out")
.....
''
  'Set up the flash and find the HP turbine outlet saturated conditions
  Set Test=HPOut.DuplicateFluid
  FS=0
  FS=Test.PVFlash(HPOut.PressureValue, 1.0)
.....
''
  'Find the desired temperature difference accross each feedwater heater
  Tout=Test.Temperature.GetValue("C")-5.556
  Tin=FW1In.Temperature.GetValue("C")
  dT=(Tout-Tin)/7
.....
''
  'Set each feedwater outlet temperature

```

```

T1=Tin+dT
T2=T1+dT
T3=T2+dT
T4=T3+dT
T5=T4+dT
T6=T5+dT
.....
''
'Find the pressures that correspond with the needed saturation
temperatures through temperature vapor fraction flashes
FS=0
FS=Test.TVFlash(T1+5.556, 1.0)
P1=Test.PressureValue
FS=0
FS=Test.TVFlash(T2+5.556, 1.0)
P2=Test.PressureValue
FS=0
FS=Test.TVFlash(T3+5.556, 1.0)
P3=Test.PressureValue
FS=0
FS=Test.TVFlash(T4, 1.0)
P4=Test.PressureValue
FS=0
FS=Test.TVFlash(T5+5.556, 1.0)
P5=Test.PressureValue
FS=0
FS=Test.TVFlash(T6+5.556, 1.0)
P6=Test.PressureValue
.....
''
'Set pressure for turbine product streams
IP1Out.Pressure.SetValue(P6)
IP2Out.Pressure.SetValue(P5)
LP1Out.Pressure.SetValue(P4)
LP2Out.Pressure.SetValue(P3)
LP3Out.Pressure.SetValue(P2)
LP4Out.Pressure.SetValue(P1)
.....
''
'Error Escape
GetOut:
Exit Sub
End Sub

```
