Faculty of

Engineering

An investigation into the performance of a Rankine-heat pump combined cycle

November 2012

School for Mechanical Engineering

Compiled by:

Stephanus Phillipus Oelofse

(13088785)

NORTH-WEST UNIVERSITY
YUNIBESITI YA BOKONE-BOPHIRIMA
NOORDWES-UNIVERSITEIT
POTCHEFSTROOM CAMPUS

An investigation into the performance of a Rankine-heat pump combined cycle

Stephanus Phillipus Oelofse

13088785

M.Eng (Mechanical) North West University Potchefstroom Campus

Dissertation submitted in partial fulfilment of the requirements for the degree of Masters in Engineering at the Potchefstroom Campus of the North-West University

Mentor: Professor C.P. Storm

Potchefstroom

November 2012

Abstract

The global growth in electricity consumption and the shortcomings of renewable electricity generation technologies are some of the reasons why it is still relevant to evaluate the performance of power conversion technologies that are used in fossil fuel power stations.

The power conversion technology that is widely used in fossil fuel power stations is the Rankine cycle. The goal of this study was to determine if the efficiency of a typical Rankine cycle can be improved by adding a heat pump as a bottoming cycle. Three simulation models were developed to perform this evaluation.

The first is a simulation model of a Rankine cycle. A quite detailed Rankine cycle configuration was evaluated. The simulation model was used to determine the heating requirements of the heat pump cycle as well as its operating temperature ranges. The efficiency of this Rankine cycle was calculated as 43.05 %.

A basic vapour compression cycle configuration was selected as the heat pump of the combined cycle. A simulation model of the vapour compression cycle and the interfaces with the Rankine cycle was developed as the second simulation model.

Working fluids that are typically used in vapour compression cycles cannot be used for this application, due to temperature limitations. The vapour compression cycle's simulation model was therefore also used to calculate the coefficient of performance (COP) for various working fluids in order to select a suitable working fluid. The best cycle COP (3.015 heating) was obtained with ethanol as working fluid.

These simulation models were combined to form the simulation model of the Rankine-heat pump combined cycle. This model was used to evaluate the performance of the combined cycle for two different compressor power sources.

This study showed that the concept of using steam turbine or electrical power to drive a compressor driven vapour compression cycle in the configuration proposed here does not improve the overall efficiency of the cycle.

The reasons for this were discovered and warrant future investigation.

Keywords: Rankine cycle, Vapour compression cycle, Power generation, Combined cycle, Thermal efficiency, Working fluids

Opsomming

Die groei in elektrisiteitsverbruik en die areas waar hernubare energie tegnologie nog te kort skiet is van die redes hoekom dit steeds van toepassing is om die uitset van krag omskakelingstegnologieë in fossiel brandstof kragstasies te bestudeer.

Die energie omskakelingstegnologie wat oor die algemeen gebruik word in hierdie kragstasies is die Rankine siklus. Die doel van hierdie studie is om te bepaal of die effektiwiteit van ‘n tipiese Rankine siklus kan verbeter kan word deur ‘n hitte pomp as die onderste siklus te gebruik. Drie simulasie modelle is ontwikkel om die evaluasie uit te voer.

Eerstens is die Rankine siklus gemodeleer. Die model was redelik omvattend. Die model is geevalueer om die verhittingsbehoeftes van die hittepomp te bepaal asook die temperatuur grense. Die effektiwiteit van die siklus is as 43.05% bereken.

‘n Basiese damp druk siklus konfigurasie is gekies om te dien as hittepomp vir die saamgestelde siklus. Die tweede model het die damp druk siklus en die raakvlakke met die Rankine siklus gesimuleer.

Dis nie moontlik om die tipiese werksvloeiers wat in damp druk siklusse gebruik word hier toe te pas nie as gevolg van die temperatuur beperkings. Die damp druk siklus se simulasie model is daarom ook gebruik om die koëffisiënt van werksverrigting (COP) vir verskeie werksvloeiers te bepaal. Sodoende is ‘n geskikte werksvloeier gekies. Die beste COP vir die siklus (3.015 verhitting) is gevind met etanol as werkvloeier.

Hierdie modelle is saamgevoeg om die Rankine damp druk saamgestelde siklus te vorm. Die model is gebruik om die uitset van die siklus te meet vir twee verskillende kompressor kragbronne.

Die studie wys dat die konsep om ‘n stoom turbine of elektriese kompressor te gebruik om ‘n damp druk siklus te dryf in die voorgestelde konfigurasie nie die effektiwiteit van die siklus as geheel verbeter nie.

Acknowledgements

I would like to give special to the following persons who supported me during this study:

- My wife for taking care of our children when I could not, for believing in me and for her motivation and patience throughout the duration of this study.
- My children, especially Linde, for understanding when I was not able to provide the attention they required.
- My parents for all their financial and moral support and for understanding when I was unable to visit.
- My in-laws for their help and support.
- Prof C.P. Storm for all the advice, insight and technical guidance he provided during this study.

Thank you for enabling me to complete my study.

Table of Contents

Table of Contents (Continued)

Table of Contents (Continued)

List of Tables

List of Figures

Abbreviations

BFP	Boiler Feed Pump
BFPTD	Boiler Feed Pump Turbine Drive
BS	Bled Steam
CTD	Compressor Turbine Drive
CEP	Condensate Extraction Pump
COP	Coefficient of Performance
CW	Cooling Water
DA	Deaerator
DCA	Drain Cooler Temperature Approach
DRUM	The boiler drum with natural circulation heaters
EES	Engineering Equation Solver
FW	Feedwater
FWH	Feedwater Heater
GSC	Gland Steam Condenser
HP	High Pressure
HTGR	High Temperature Gas-Cooled Reactor
HX	Heat Exchanger
IP	Intermediate Pressure
LP	Low Pressure
ORC	Organic Rankine Cycle
RE	Reheater
SH	Superheater
TDC	Turbine Drains Cooler
TTD	Terminal Temperature Difference
HPC	Heat Pump Condenser
RCC	Rankine Cycle Condenser

List of Variables

COP_H	The heating COP of a vapour compression cycle
COP_C	The cooling COP of a vapour compression cycle
c_p	The specific heat capacity of the fluid at constant pressure
c_v	The specific heat capacity of the fluid at constant volume
Eff_{HX}	HX effectiveness
h_e	Specific enthalpy at the outlet of an actual process or a control volume
h_{es}	Theoretical specific enthalpy at the outlet of an isentropic process
h_i	Specific enthalpy at the inlet of a process or a control volume
k	A polytropic expansion constant $[= \eta_{poly}(\gamma - 1)/\gamma]$
$\dot{m}_e$	Control volume outlet mass flow rate
$\dot{m}_i$	Control volume inlet mass flow rate
P_e	Total pressure at the outlet of the control volume
P_i	Total pressure at the inlet of the control volume
$\dot{Q}$	Total rate of heat transfer to the fluid
Q_h	Total cycle energy input
Q_H	Heat rejected in the condenser of a heat pump
Q_L	Heat absorbed in the evaporator of a heat pump
$T_{COND,out}$	The steam condenser outlet temperature
$T_{COND,sat}$	The saturation temperature of the steam condenser
$T_{COND,sub}$	The degree of subcooling present in the condenser
$T_{CW,in}$	The CW inlet temperature
$T_{CW,r}$	The CW temperature rise across the condenser
T_e	Total temperature at the outlet of the control volume
T_i	Total temperature at the inlet of the control volume
$T_{in,c}$	Temperature of “cold” fluid at the heat exchanger inlet
$T_{in,h}$	Temperature of the “hot” fluid at the heat exchanger inlet
T_L	Absolute temperature of the heat sink
T_H	Absolute temperature of the heat source
$T_{out,c}$	Temperature of the “cold” fluid at the outlet of the heat exchanger
$T_{out,h}$	Temperature of the “hot” fluid at the heat exchanger outlet

$T_{sat,h}$	Saturation temperature of condensing fluid
$\dot{W}$	Total rate of work done on the fluid
W_C	The cycle energy input in the form compressor work
W_{net}	Net Power Output of the Power Cycle

Greek Letters

ΔP	Lumped total pressure loss
η_c	Carnot efficiency of the cycle
η_{isen}	Isentropic efficiency
η_{poly}	The polytropic efficiency of the turbine
η_t	Overall thermal efficiency
γ	The ratio of the specific heat capacities

CHAPTER 1: INTRODUCTION

1.1 BACKGROUND

Electricity is currently mainly generated from coal, peat, liquid fuel, gas, nuclear and hydropower. In 2009, 81.9% of the world's electricity was generated by fossil fuel (coal, peat, liquid fuel and gas) and nuclear power plants (OECD, 2011:132).

Climate change triggered a global drive to reduce the greenhouse gas emissions and the energy sector was the "largest single source of global greenhouse gas emissions" in 2004 (EPA, 2012).

One way to control the greenhouse gas emissions of this sector is with the use of renewable energy sources. The U.S. Energy Information Administration (EIA, 2011:4) predicted that installed capacity of fossil fuel and nuclear power plants will increase, even though the focus of power generation is now shifting towards clean and sustainable electricity generation. Current technology shortcomings, an increase in electricity demand and financial feasibility are some of the main factors that prohibit the use of green electricity generation processes.

Other ways of reducing the greenhouse gas emissions of the electricity sector is the post combustion treatment of off-gasses and improving the energy conversion efficiencies of the fossil fuel and nuclear power plants.

A literature survey has shown that the Rankine cycle is widely used in fossil fuel and nuclear power plants. During 2006, more than 97% of South Africa's power was generated with the use of Rankine cycles (NERSA, 2010:31).

The Rankine cycle is typically used as a standalone cycle or in combination with other thermal cycles to improve thermodynamic efficiency of the power plant.

The focus of most of these combined cycles is:

- to overcome the high temperature/pressure limitations of the steam Rankine cycle, by adding a topping cycle or
- to combine power generation with a heating or cooling application.

Chapter 2 provides a more detailed background on the Rankine cycle and combined cycles containing Rankine cycles found in literature.

1.2 Problem Statement

A thorough literature survey revealed only one investigation (Agnew, et al., 2004:1509) into the concept of combining a Rankine cycle with a bottoming heat pump or refrigeration cycle to improve electricity generation efficiency.

It was also found that the concept of a Rankine-heat pump combined cycle, where the heat pump condenser (HPC) is used to heat the feed water (FW) and its evaporator is used to reduce the temperature of the cooling water before it enters the Rankine cycle condenser (RCC), have not yet been investigated.

1.3 Objective

The primary objective of this study is to determine if the thermal performance of a typical Rankine cycle can be improved by using a compressor driven heat pump cycle, instead of low pressure (LP) feedwater heaters (FWH), to heat the feedwater of the Rankine cycle in a Rankine-heat pump combined cycle.

In this cycle configuration, the evaporator of the heat pump is connected to the Rankine cycle's cooling water (CW) supply, indoor to reduce the operating temperature of the condensers as an added benefit.

1.4 METHOD OF INVESTIGATION

A simulation model of a rather detailed Rankine cycle was identified as the first requirement of this study. This simulation model had to serve as the Rankine part of the combined cycle and input to the heat pump simulation model.

Secondly, a simulation model of a heat pump, more specific, a compressor driven heat pump cycle was required. This model was used to select an appropriate working fluid.

These models were combined to form the simulation model of the combined cycle. The thermal performance of this combined cycle was compared with the performance of a stand-alone Rankine cycle or reference Rankine cycle as it is called in this document.

The Engineering Equation Solver (EES) was the software package selected to perform the simulations. EES has built-in fluid property calculation functions, with the ability to solve a vast number of coupled algebraic equations in an iterative process.

1.5 LIMITATIONS OF THE STUDY

This study will be limited to the theoretical investigation and will focus on the thermal performance of a Rankine-heat pump combined cycle. The design of physical equipment will therefore not be included, but the efficiencies of the equipment have been incorporated.

The Rankine-heat pump combined cycle analysis will be based on the first law of thermodynamics and will therefore exclude cycle optimising studies, exergy analyses and economic analyses.

Combining power generation with heating or cooling applications were also not included in the scope of the study.

1.6 DISSERTATION STRUCTURE

This document consists of a number of chapters. These chapters will briefly be discussed in this section.

CHAPTER 1

The first chapter serves as an introduction to this document. This chapter provides a brief background on the subject as well as the objective and limits of the study. A brief overview of the research procedure that was followed is also provided in this chapter.

CHAPTER 2

The aim of this chapter is to provide a detailed background on the subject of this study. The chapter discusses the Rankine cycle configuration changes that were implemented over the years. The typical limitations of the Rankine cycle are also provided in this chapter.

A detail investigation on combined cycle configurations which contains the Rankine cycle is also presented in this chapter. This investigation was performed to determine what had been done in the past, to prevent duplication of previous studies.

CHAPTER 3

A typical simulation model of a thermal cycle consists of conservation equations, fluid property equations and component characteristics equations. This chapter was used to present these equations and the simulation methodologies that were used in this study.

CHAPTER 4

This chapter was used to discuss the Rankine cycle configuration and the simulation inputs in detail. The results of the Rankine cycle simulation model are also presented at the end of this chapter.

CHAPTER 5

The heat pump technology selection and the simulation were discussed in this chapter. Covered in this chapter is:

- a cycle configuration discussion,
- the simulation inputs and
- the process that was followed to select the working fluid of the vapour compression cycle.

The simulation results are also provided at the end of the chapter.

CHAPTER 6

The process of combining the Rankine cycle with the vapour compression cycle is discussed in this section. Two combined cycle configurations are discussed in this section, along with the simulation models and results. The simulation results of the two simulation models are also compared with the results of the Rankine cycle in order to determine if the thermal efficiency Rankine cycle can be improve with the proposed combined cycle configurations.

CHAPTER 7

This is the final chapter of this document, which provides the conclusion of the study. This chapter also provides topics for possible future work on the subject of this study.

CHAPTER 2: LITERATURE SURVEY

2.1 INTRODUCTION

The literature survey conducted for study has been divided in two major topics, i.e. the reference Rankine cycle and combined cycles which contain a Rankine cycle.

The purpose of the Rankine cycle survey was to determine the typical cycle constraints, working fluids and cycle configurations that are used in power plants. This was done in order to ensure that relevant cycle is selected for the purpose of this study.

The thorough literature survey of combined cycles which contain a Rankine cycle was also conducted to determine what had been done in the past to ensure that the wheel is not reinvented by this study.

The literature survey of the Rankine cycle will be discussed first, followed by the literature survey that was on combined cycles which contain a Rankine cycle.

2.2 THE RANKINE CYCLE

Most of the world's thermal power plants incorporate the Rankine cycle to convert thermal energy to shaft power. The Rankine cycle is used, even though the Carnot cycle is known to be the most efficient thermal cycle operating between two constant temperature thermal reservoirs. The practical difficulties associated with controlling the condensing/compression transition point of the wet vapour Carnot cycle is one of the reasons for this. These issues are addressed in the basic Rankine cycle, where the condensing process is completed, before the working fluid is pressurised (Eastop & McConkey, 1993:235; Granet, 1980:320-324; Schroeder, 2000:125; Sonntag, et al., 2003:227-229,384).

2.2.1 Working Fluid

There are various working fluids used in Rankine or Rankine-type cycles. These include, but are not limited to, water, refrigerants, hydrocarbons and binary fluids.

Refrigerants and hydrocarbons have been investigated since the 1880s as an alternative working fluid for the Rankine cycle (Tchanche, et al., 2011:3964). Good thermodynamic performance have been achieved with these cycles, generally known as organic Rankine cycles (ORC), when used in low temperature applications (Jing, et al., 2010:11).

Alternative power cycles, using binary working fluids in Rankine-type or absorption-type cycles, have also been investigated. During 1953, Maloney and Robertson compared the thermodynamic performance of an absorption-type cycle and a Rankine cycle. They concluded that the absorption-type cycle had no advantage over the Rankine cycle (cited by Ibrahim & Klein, 1996:21). Kalina (1984:740) presented contradicting results with a novel absorption-type cycle. He presented evidence that an absorption-type cycle, with an ammonia-water working fluid, has some thermodynamic advantages in certain cases.

The working fluid most commonly used in Rankine cycles is water. The low working fluid cost and the thermal, dynamic and chemical properties of water (Tchanche, et al., 2011:3964) are the main reasons for this.

The low electricity generation costs associated with steam power stations (Haywood, 1987:15) have and will continue to promote the use of steam power cycles to generate electricity from large scale, high temperature energy sources.

2.2.2 Temperature Limitations

The efficiency of any thermal power cycle is limited to the efficiency of a Carnot cycle operating between constant maximum and minimum temperatures.

The efficiency of a Carnot cycle is defined as (Schroeder, 2000:125):

$$\eta_c = 1 - \frac{T_L}{T_H} \quad (1)$$

where:

η_c	–	Carnot efficiency of the cycle
T_L	–	Absolute temperature of the heat sink
T_H	–	Absolute temperature of the heat source

Equation 1 is used as a guide to the efficiency of thermal power cycles, even though the Carnot cycle is impractical. This equation clearly indicates that the cycle efficiency will increase by raising the temperature of the heat source and/or lowering the temperature of the heat sink (Lior, 1997:942).

The temperature of the heat source generally does not limit the maximum temperature of the thermal power cycle (Lior, 1997:943). The material properties are the main factor dictating the fluid temperature and pressure at the inlet of the high pressure turbine. Therefore the subcritical Rankine cycles were traditionally used in steam power plants. Material property improvements and the need for high cycle efficiency promoted the use of supercritical Rankine cycle in power plants (Beér, 2007:109).

With the use of Equation 1 it can be illustrated that lowering the heat sink temperature has a greater effect on cycle efficiency than increasing the heat source temperature with the same amount.

Lowering the condenser temperature of a thermal power cycle can improve its efficiency by up to 0.5%. This decrease in condenser temperature can be achieved by increasing heat transfer ability of the condensing unit or lowering the temperature of the coolant. An increase in the heat transfer ability of a heat exchanger (HX) normally increases the capital cost and pumping requirements. Options to lower the temperature of the cooling medium include (Lior, 1997:943):

- Cold air, water and/or ice of the polar regions
- Cold ocean water at depths below 500 m
- Using space as a heat sink

All of which is subjected to the location of the power plant, which is determined by the location and ease of transport of the fuel source, location of end users and transition losses (Lior, 1997:943). Other cooling options include air cooled condensers and secondary water recirculation systems with wet or dry cooling towers, all dependant on ambient temperatures.

2.2.3 Cycle Configurations

The low cost associated with steam power cycles promoted energy efficiency and optimising studies on the Rankine cycle. This resulted in complicated configurations for the Rankine cycle. The three main configuration changes to the simple Rankine cycle are superheating, reheating and regenerative feedwater heating (Granet, 1980:325-372; Sonntag, et al., 2003: 384-403). A further efficiency improvement has also been achieved by powering the feedwater pumps with steam turbines.

A typical cycle configuration used in power stations consists of (Storm, 2012):

- condensate extraction pumps (CEP)
- a gland steam condenser (GSC)
- a turbine drains cooler (TDC)
- three or four stages of LP FWH
- a deaerator (DA)
- boiler feed pumps (BFP)
- two or three stages of high pressure (HP) FWH's
- the boiler with associated HXs
- a HP turbine with an inlet throttling valve
- an intermediate pressure (IP) turbine with steam bleed-off points for rest of the HPFWH's and the DA
- two LP turbines with steam bleed-off points for the LPFWH's
- two RCC's, one for each LP turbine outlet
- and in some cases boiler feed pump turbine drives (BFPTD)

2.3 Cycle Combinations that Contain the Rankine Cycle

Different thermal cycle combinations have been investigated over the years. The aim of these cycle combinations are normally improved cycle efficiency or cogeneration. The cycle combinations that contain the Rankine cycle will be discussed in this section.

2.3.1 The Binary Vapour Cycle

Some of the disadvantages of using water as working fluid of the Rankine cycle are the poor heat transfer capabilities in superheated steam and the high vapour pressure of water (Granet, 1980:372).

The consequences of the high vapour pressure of water are greater pipe and HX wall thicknesses. This also reduces the heat transfer capabilities of the superheaters. The heat transfer capabilities of the superheaters increases the required heat transfer area and the working fluid/heat source temperature difference. The knock-on effect of the relatively large temperature difference is lower cycle efficiencies (Granet, 1980:372).

The effect of these poor qualities of water is reduced by combining a bottoming steam Rankine cycle with an additional topping Rankine cycle. The desired properties of the working fluid of the topping cycle, normally mercury, are a low vapour pressure and a high critical temperature (Granet, 1980:372; Cole, 1991:219). A simple schematic of a mercury-water binary cycle is presented in Figure 1.

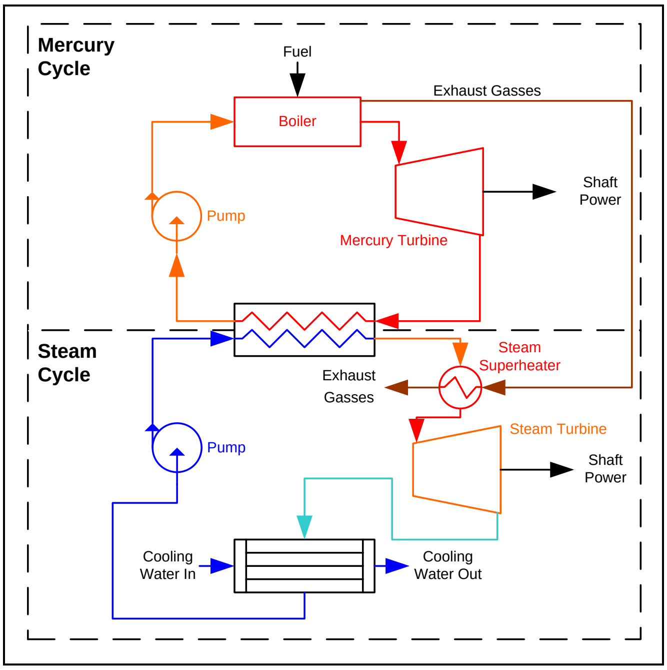

Figure 1: A simple mercury-water binary cycle.

One of the other advantages for this cycle configuration is the possibility of isothermal boiling processes in both the cycles (Sonntag, et al., 2003:446).

2.3.2 The Combined Cycle

The combined cycle (Figure 2) is a well-known cycle configuration, were the gas cycle is used as the topping cycle, with a Rankine or Rankine-type bottoming cycle.

Roughly 65% of the energy in a simple gas turbine cycle's fuel is lost through exhaust heat. In a combined cycle configuration, this heat is recovered by generating steam (vapour) for the Rankine bottoming cycle. This additional energy utilisation can increase the thermal efficiency of the power plant by 10% or more (Poullikkas, 2005:425).

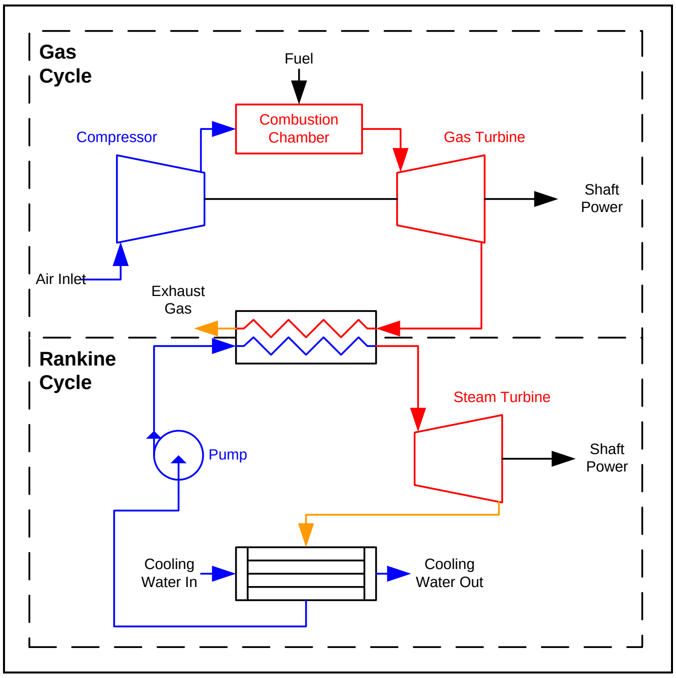

Figure 2: A simple combined cycle

Heyen & Kalitventzeff (1999:227) investigated two combined cycles, suitable for the upgrade of existing plants operating with a Rankine cycle with steam reheat and 3-level extraction. These cycles were:

- The Rankine cycle with a gas turbine topping cycle
- The Rankine cycle with a partial oxidation topping cycle

The gas turbine cycle they used was an open Brayton cycle. In this cycle air is compressed with a compressor, which is mixed with fuel in the combustion chamber. The combustion of the mixture adds heat to the air, which is fed into two turbines, placed in series. The function of the first turbine is producing shaft power for the compressor and the second compressor was used to generate electricity and the exhaust gasses was used as heat source for the Rankine cycle (Heyen & Kalitventzeff, 1999:230-231).

In the partial oxidation cycle, a smaller volume of air is compressed, which is mixed with hot gas before entering the first turbine. The gas mixture at the outlet of the first turbine is then mixed steam and more hot gas. It then enters a partial oxidation catalytic reactor. The resulting gas mixture is expanded in the second turbine and the flue gas is used as heat source for the

Rankine cycle. The benefit of this cycle is that less compressed air is required for the cycle and less compressor work is therefore required (Heyen & Kalitventzeff, 1999:230 231).

Both these cycles resulted in fairly low cycle efficiency improvement, mainly due to the constraints associated with upgrading existing power plants. Their study did show the greater efficiency improvements can be achieved with a partial oxidation topping cycle. Cycle efficiencies of above 60% can be expected with a proper design of the partial oxidation/Rankine combined cycle, according to Heyen & Kalitventzeff (1999:227).

The alternative Rankine-type bottoming cycles presented in literature is an absorption-type Rankine cycle (Kalina, 1984:737) and the ORC (Chacartegui, et al., 2009:2165).

The variable boiling temperature of the absorption-type cycle reduced the effect of thermal pinch. This boiling characteristic of the Kalina cycle resulted in a lower exhaust gas exit temperature, which increased the power output of the bottoming cycle and the overall plant efficiency (Kalina, 1984:739-741).

Chacartegui, et al. (2009:2165) calculated overall cycle efficiencies of just below 60 % for an ORC-gas turbine combined cycle.

2.3.3 The Rankine cycle with a Kalina Bottoming Cycle

One of the obstacles associated with parabolic trough solar fields is maintaining the temperature of the heat source/thermal store. This increases the insulation costs and reduces the plant availability.

Mittelman & Epstein (2010:1761) presented a Rankine-Kalina cycle for electricity generation at parabolic trough solar fields. They added the Kalina as a bottoming cycle to a Rankine as shown in Figure 3.

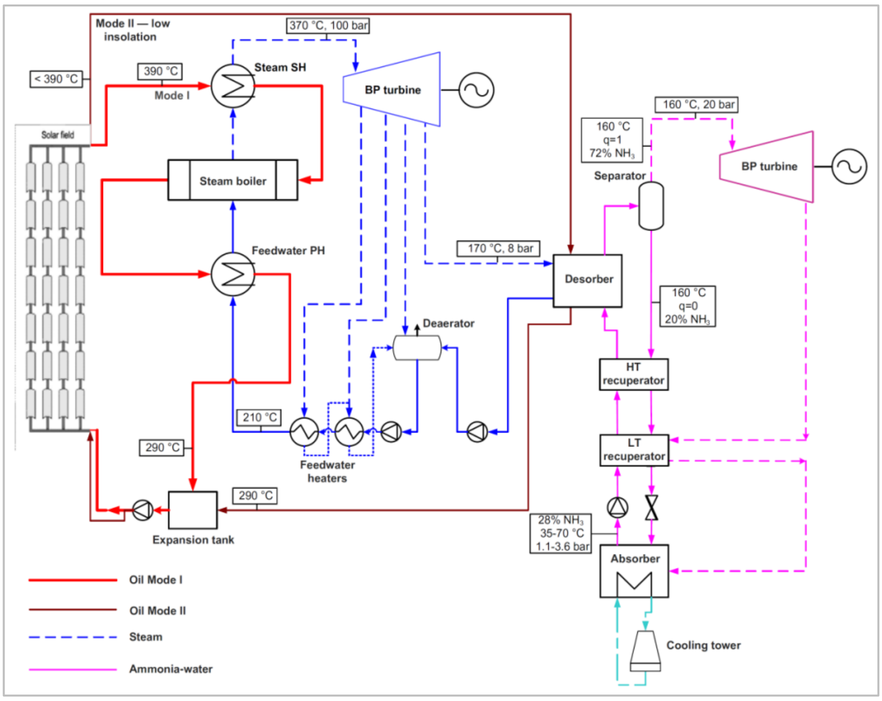

Figure 3: Combined Kalina-Rankine cycle driven by a parabolic trough field (Mittelman& Epstein, 2010:1765).

The configuration had the capability of bypassing the Rankine cycle when the temperature of the heat source/thermal store drops to low temperatures (<390 °C). The resulting increase in plant availability is significant enough to reduce the cost of electricity by 4-11 %, even though the efficiency of the combined cycle was ~ 5 % lower than that of the Rankine cycle (Mittelman & Epstein, 2010:1761).

2.3.4 The Rankine-Heat Pump Combined Cycles

Al-Sulaiman, et al. (2010:5106) presented cycle configuration with an ORC, which was connected to a bottoming absorption cycle. In their study, they investigated the cogeneration possibilities of the configuration and focussed on the cooling capabilities of the cycle.

Aphornratana & Sriveerakul (2010:2558) evaluated a Rankine-vapour-compression cycle, but the function of their cycle was refrigeration.

Only one Rankine-heat pump combined cycle with a function other than refrigeration or cogeneration was found in the literature survey. In this theoretical study, conducted by Agnew, et al. (2004:1509), a Rankine cycle with an absorption bottoming cycle was analysed. They aimed to improve the power generation performance of the Rankine cycle by reducing the operating temperature of the RCC, using the refrigerating properties of the absorption cycle. The absorption cycle was powered with heat extracted from the boiler flue gasses. Their study showed thermodynamic advantages, but they concluded that thermo-economic aspects may make the configuration unattractive.

2.3.5 Conclusion

A thorough literature survey revealed no reference to a Rankine-heat pump combined cycle, where the heat pump was used to perform FW heating.

CHAPTER 3: MODELLING METHODOLOGY

3.1 INTRODUCTION

The simulation models were developed:

- using high level conservation equations,
- component specific equations,
- fluid property equations,
- process inputs and
- a cycle efficiency equation.

This chapter will be used to provide and discuss the cycle efficiency equation, the conservation equations, the component characteristic equations and the fluid property equations that were used in this study.

The modelling methodologies which were used to model the attemperation process and the cooling water cycle will also be discussed in this chapter.

3.2 CYCLE EFFICIENCY

The thermal efficiency of a power cycle is simply defined by the relation:

$$\eta_t = \frac{W_{net}}{Q_h} \qquad (2)$$

where:

η_t – is the overall thermal efficiency,
W_{net} – is the net power output of the power cycle and
Q_h – is the total energy input.

However, a range of definitions is used in literature to define net power output and the total energy input of power cycles, depending on the nature of the study.

For this study, the total energy input was defined as the energy transferred from the heat source to the working fluid. All the combustion related boiler losses will be neglected.

The net power output was defined in terms of the net shaft power output of the cycle. It was therefore required to convert the power required by electrical equipment to the equivalent shaft power, before subtracting it from the power output of the turbines to calculate the net power output of the cycle.

3.3 Conservation Laws

Three conservation laws are normally used in thermal fluid system models. These laws are:

- the conservation of mass,
- the conservation of momentum (angular and linear) and
- the conservation of energy.

These laws are formulated into a set of equations, known as the conservation or governing equations. These conservation equations were derived for detailed transient thermal fluid models, where equipment details, like flow area, volume, heat capacity, etc., is available.

These equations were simplified by removing the time derivative, since only steady state simulation formed part of the study's scope. The simplified equations are presented below.

3.3.1 Conservation of Mass

The conservation of mass equation could then be reduced to:

$$\sum \dot{m}_e - \sum \dot{m}_i = 0 \qquad \textbf{3}$$

where:

$\dot{m}_i$ – is the control volume inlet mass flow rate and

$\dot{m}_e$ – is the control volume outlet mass flow rate.

3.3.2 Conservation of Momentum

Some of the terms in the conservation of momentum equation require quite some geometrical detail of the plant and its equipment. Since some of this detail is plant specific and not always generally available, it was decided to lump these terms into a single term, the lumped pressure loss.

These simplifications reduced the conservation of momentum equation to:

$$(P_e - P_i) + \Delta P = 0 \qquad \textbf{4}$$

where:

- P_i – is the total pressure at the inlet of the control volume
- P_e – is the total pressure at the outlet of the control volume and
- ΔP – is the lumped total pressure loss in the control volume.

3.3.3 Conservation of Energy

The change in energy due to a change in elevation also forms part of the conservation of energy equation. It was assumed that the effect of these elevation changes will balance out, since the simulated cycles are closed cycles.

The conservation of energy equation could then be reduced to:

$$\dot{Q} + \dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i \qquad \textbf{5}$$

where:

- $\dot{Q}$ – is the total rate of heat transfer to the fluid in the control volume,
- $\dot{W}$ – is the total rate of work done on the fluid in the control volume,
- h_i – is the specific enthalpy at the inlet of the control volume and
- h_e – is the actual specific enthalpy at the outlet of the control volume.

3.4 COMPONENT SPECIFIC EQUATIONS

It can be seen that two component characteristics are required to solve equations 3, 4 and 5. These characteristics are the total pressure drop and the total change in fluid energy across the component, i.e. the thermal behaviour of the component.

The main components of the Rankine cycle can be divided into the following three groups:

- Turbo machines
- HX's
- Valves

It was decided that the total pressure drop of each component will be provided as inputs to the simulation model and the thermal behaviour of the components will be characterised with the use of component performance equations.

The performance equations that were used for the components will be discussed in this section.

3.4.1 Turbo Machines

The performance of all turbo machines (pump, compressors and turbines) can be characterized using the definition of isentropic or polytropic efficiency. In both instances, the efficiency of the turbo machine is defined by a relation between the actual process and an ideal process.

The main difference between these definitions is that the definition of isentropic efficiency does not account for the diverting nature of the pressure lines of fluids, which is accounted for in the definition of polytropic efficiency (Saravanamuttoo, et al., 2001:16).

In this study, partially expanded steam was extracted from the IP turbine and LP turbines, thus two aspects had to be considered, i.e.:

1. how will the process conditions at the turbine outlet be calculated and
2. how will the process conditions at the steam extraction points be calculated.

The definition of isentropic efficiency was selected to characterize the overall efficiencies the turbo machines, since the effect of turbo machine pressure ratios will not be evaluated in this study.

The ratio between the pressures at the turbine inlet and the point of steam extraction differs significantly from the overall pressure ratios of the turbines. The isentropic efficiency approach could therefore not accurately be used to characterize efficiency of the turbine stage leading to the steam extraction point. For this reason, the definition of polytropic efficiency was used to calculate the total change in fluid energy across this section of the turbines.

Both these definitions are discussed below.

Isentropic Efficiency

The isentropic efficiency of compression processes is defined as:

$$\eta_{isen} = \frac{h_i - h_{es}}{h_i - h_e} \qquad \textbf{6}$$

where:

η_{isen} – is the isentropic efficiency and

h_{es} – is the theoretical specific enthalpy at the outlet of an isentropic process.

The isentropic efficiency of expansion processes is defined as:

$$\eta_{isen} = \frac{h_i - h_e}{h_i - h_{es}} \qquad \textbf{7}$$

The isentropic efficiencies that were used in this study are listed as in the appropriate chapters.

Polytropic Efficiency

The definition of polytropic efficiency for an expansion process can be written as:

$$\frac{T_i}{T_e} = \left(\frac{P_i}{P_e}\right)^{\eta_{poly}(\gamma-1)/\gamma} \qquad \textbf{8}$$

where:

T_i	–	is the total temperature at the inlet of the control volume,
T_e	–	is the total temperature at the exit of the control volume,
γ	–	is the ratio of the specific heat capacities and
η_{poly}	–	is the polytropic efficiency of the turbine.

The ratio of the specific heat capacities is calculated with:

$$\gamma = \frac{c_p}{c_v} \qquad \textbf{9}$$

where:

c_p	–	is the specific heat capacity of the fluid at constant pressure and
c_v	–	is the specific heat capacity of the fluid at constant volume.

Equation 8 could now be used to determine the polytropic efficiency of the turbine from the inlet and outlet conditions, but the fluid at the outlet of the LP turbines is a two-phase mixture and the heat capacities of a two-phase mixture are infinite. It was therefore decided to rather use a constant specific heat capacity ratio during this study.

If gamma is constant, equation 8 can be rewritten as:

$$\frac{T_i}{T_e} = \left(\frac{P_i}{P_e}\right)^{k} \qquad \textbf{10}$$

Where $k\left[= \eta_{poly}(\gamma - 1)/\gamma\right]$ is a polytropic expansion constant.

During this study, equation 10 was used with the inlet and outlet conditions of the turbine to determine the polytropic expansion constant for the turbine. Now the temperature of the bled steam can be calculated with equation 10.

3.4.2 Heat Exchangers

A number of HX's were encountered in this study. All of these HX's do not work on the same principles and some of them also did not require detailed modelling. A single HX definition could therefore not be used to characterise the performance of all the HX's.

The HX's were rather divided into groups to enable the use of different performance definition models. These groups are:

- the boiler HX's,
- the DA,
- the FWH's,
- the condensers,
- and the heat pump evaporator.

The performance definitions used for each HX group will now be discussed.

Boiler HXs

The HXs located in the boiler is heated by means of radiation and convection. The typical HX located in a pulverised coal boiler is listed below:

- an economiser
- the boiler drum with natural circulation heaters (DRUM)
- two primary super heaters (SH) with attemperation
- a secondary SH with attemperation
- two primary reheaters (RH) with attemperation
- and a secondary RH.

The cycle efficiency definition (section 3.2) leaves room to neglect heat transfer efficiencies of all the boiler components.

No HX performance definitions were therefore required for these HX's.

The Deaerator

The DA as well as the FWH's, are used for regenerative feedwater heating. Regenerative feedwater heating is used to heat the feedwater (FW), before it enters the boiler. The heat source for this heating process is partially expanded steam, extracted from the turbines. Increased cycle efficiencies are achieved with this configuration, even though the work output of the turbines are reduced. This is because the amount of energy rejected in the condenser is significantly reduced by the process (Granet, 1980:370-372).

The main difference between the two heaters lies in their method of heating.

The DA is a contact heater. All the incoming fluids of the DA are mixed to increase the heat transfer efficiency to 100 %. The outlet conditions of a DA can therefore easily be calculated with the conservation of energy equation.

The FWH's

The FWH is a closed heater, e.g. the FW is never in contact with the heating medium (BS). These heaters are predominantly counter flow HX's, which can essentially be divided into two or three sections, depending on whether subcooling occurs.

These sections are:

- the section where the heating medium is still superheated steam,
- the section where the heating medium is in the two-phase region and
- the section where the heating medium is subcooled.

The assumption is made that the heat transfer rate throughout a FWH is constant and the BS outlet is saturated water (no sub-cooling). The temperature distribution through that FWH will then be similar to that presented in Figure 4.

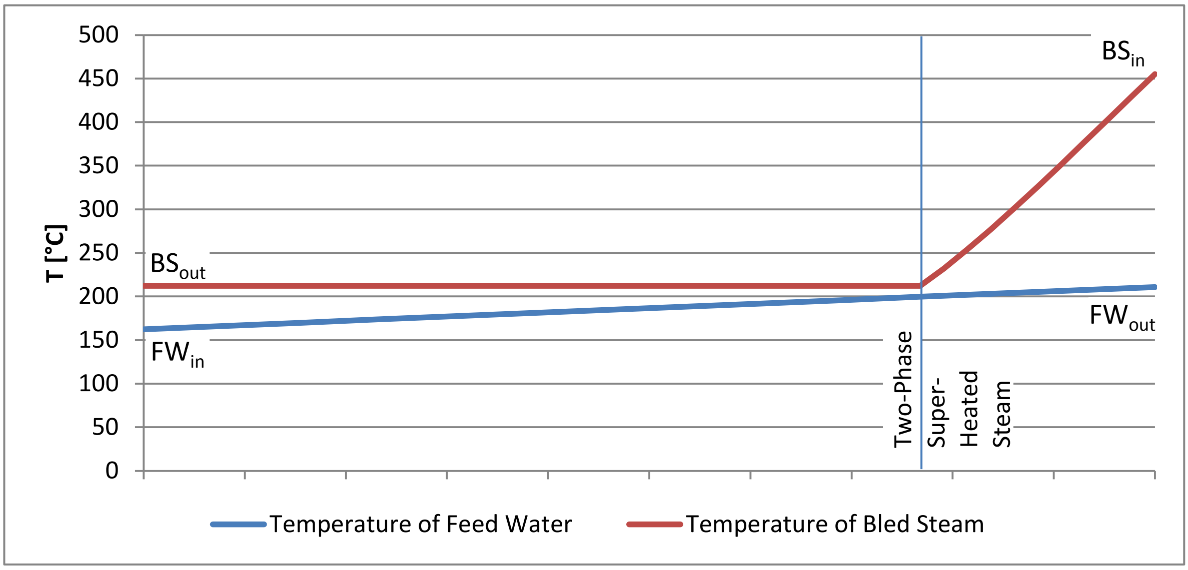

Figure 4: The temperature distribution through a feed water heater with a constant heat transfer rate.

It can be seen from Figure 4 that a typical FWH also has a thermal pinch point. A FWH can therefore not simply be analysed by using the definition of overall HX effectiveness. Generally researchers characterise the performance of the FWH's with the following definitions:

- the drain cooler temperature approach (DCA) and
- the terminal temperature difference (TTD).

The DCA of a FWH is essentially used to calculate the degree of subcooling that occurs within the heater, but for this project it is assumed that no subcooling occurs in the FWH. Only the definition of TTD was therefore used in this study.

The TTD of FWH is defined as:

$$\mathrm{TTD} = \mathrm{T_{sat,h}} - \mathrm{T_{out,c}} \qquad \mathbf{11}$$

where:

$T_{sat,h}$ – is the saturation temperature of condensing fluid and

$T_{out,c}$ – is the temperature of the FW at the HX outlet.

Condensers

Three major types of condensers are generally used in power stations. These condenser types are air cooled condensers, spray condensers and water cooled condensers.

All the condensers that form part of this study is water cooled. These condensers are the two RCC's and the condenser of the vapour compression cycle.

This temperature can be calculated as:

$$T_{COND,sat} = T_{CW,in} + T_{CW,r} + TTD \tag{12}$$

where:

$T_{COND,sat}$ – is the saturation temperature in the condenser,

$T_{CW,in}$ – is the CW inlet temperature and

$T_{CW,r}$ – is the CW temperature rise across the condenser.

The condenser pressure can now be calculated using the saturation temperature and the fluid property equations. It was assumed that subcooling can occur in the condensers.

The outlet temperature at the condensers is therefore calculated as:

$$T_{COND,out} = T_{COND,sat} - T_{COND,sub} \tag{13}$$

where:

$T_{COND,out}$ – is the steam condenser outlet temperature and

$T_{COND,sub}$ – is the degree of subcooling present in the condenser.

The heat pump evaporator

The definition, HX effectiveness, was used to characterise the evaporator of the heat pump. The HX effectiveness of evaporator was defined as:

$$\mathrm{Eff}_{\mathrm{HX}} = \frac{\mathrm{T}_{\mathrm{out,c}} - \mathrm{T}_{\mathrm{in,c}}}{\mathrm{T}_{\mathrm{in,h}} - \mathrm{T}_{\mathrm{in,c}}} \qquad \textbf{14}$$

where:

$\mathrm{Eff}_{\mathrm{HX}}$ – is the HX effectiveness,

$\mathrm{T}_{\mathrm{in,h}}$ – is the inlet temperature of the “hot” fluid,

3.4.3 Expansion Valves

A throttling process is described by Sonntag, et al. (2003:174) as a pressure drop at an almost constant enthalpy. They therefore assumed that the enthalpy stays constant across a throttling process for calculation purposes. The same methodology was followed during this study.

3.5 Fluid Property Equations

The additional required fluid properties were calculated with the use of the fluid property equations available in EES.

3.6 Modelling of the Attemperation Process

The steam turbines of a Rankine cycle have the ability to respond to the changing power requirements of the power grid, but the boiler’s responds is slow because of its thermal inertia. This is overcome with the attemperation process.

The attemperation is used to control the outlet temperature of the superheaters by spraying FW, extracted from the BFP, into the steam. This enables a fast, responsive control of the temperature at the turbine inlet. The FW used for attemperation is added to the steam at the inlet of the each SH stage, to protect the heaters against overheating.

In this study, FW were added at the outlet of the SHs, although in practice, this is not the case. The contradicting modelling configuration was selected, since the process information w.r.t. the attemperation process was not available.

The FW mass flow rates will be calculated based on a fixed required steam temperature change across the attemperation process.

3.7 COOLING WATER CYCLE MODELLING

The main components of a typical cooling water cycle are the RCC's, a cooling tower and a circulation pump. The key results of the cooling water cycle that is required when modelling a power cycle are:

- the temperature of the CW returning from the cooling tower and
- the water flow rate.

The temperature of the CW returning from the cooling tower is strongly dependent on the performance of the cooling tower. From the literature survey, it was found that Kröger is one of the leading researchers in the field of cooling towers. Kröger (2004) presented detailed cooling tower performance calculation methods for both wet and dry cooling towers.

After careful consideration it was concluded that the Rankine-heat pump combined cycle will not significantly affect the temperature of the cooling water returning from the cooling tower. It was therefore decided that the detail calculations presented by Kröger (2004) will not be performed for this study. For this study, the temperature of the CW entering the first RCC was instead fixed at a specific temperature.

The flow rate of the CW is mainly determined by the RCC design requirements, i.e. the temperature rise requirements and the amount of heat that needs to be extracted. These two RCC requirements were also used to calculate the required RCC flow rate during this study.

CHAPTER 4: THE REFERENCE RANKINE CYCLE

4.1 INTRODUCTION

A simulation model of the Rankine cycle was required:

- to determine the thermal efficiency of the reference cycle,
- to determine the operating envelope of the vapour compression cycle and
- to use as the Rankine part of the proposed cycle combination.

The focus of this chapter is to present the reference Rankine cycle configuration and the cycle input parameters.

4.2 CYCLE DESCRIPTION

A simplified configuration of the Rankine cycle was used in this study, since optimization of the Rankine does not form part of this study. The selected Rankine cycle configuration, presented in Figure 5, will be described in this section, starting at point 1.

It was assumed that FW at point 1 is a subcooled liquid, as stated in section 3.4.2. The FW at point 1 is then pressurised by the electrical driven CEP. The pressure at the outlet of the CEP (point 2) should be that of the DA plus the pressure losses between point 2 and the DA. This pumping process raises the temperature of the working fluid, which can be calculated using the isentropic efficiency equation presented in section 3.4.2.

A LP FWH is placed between points 2 and 3. The function of the LP FWH is to raise the temperature of the FW with the use of bled steam, extracted from the LP turbines (point 20). The maximum achievable temperature at point 3 is strongly dependant on the saturation temperature of the bled steam and the efficiency of the FWH.

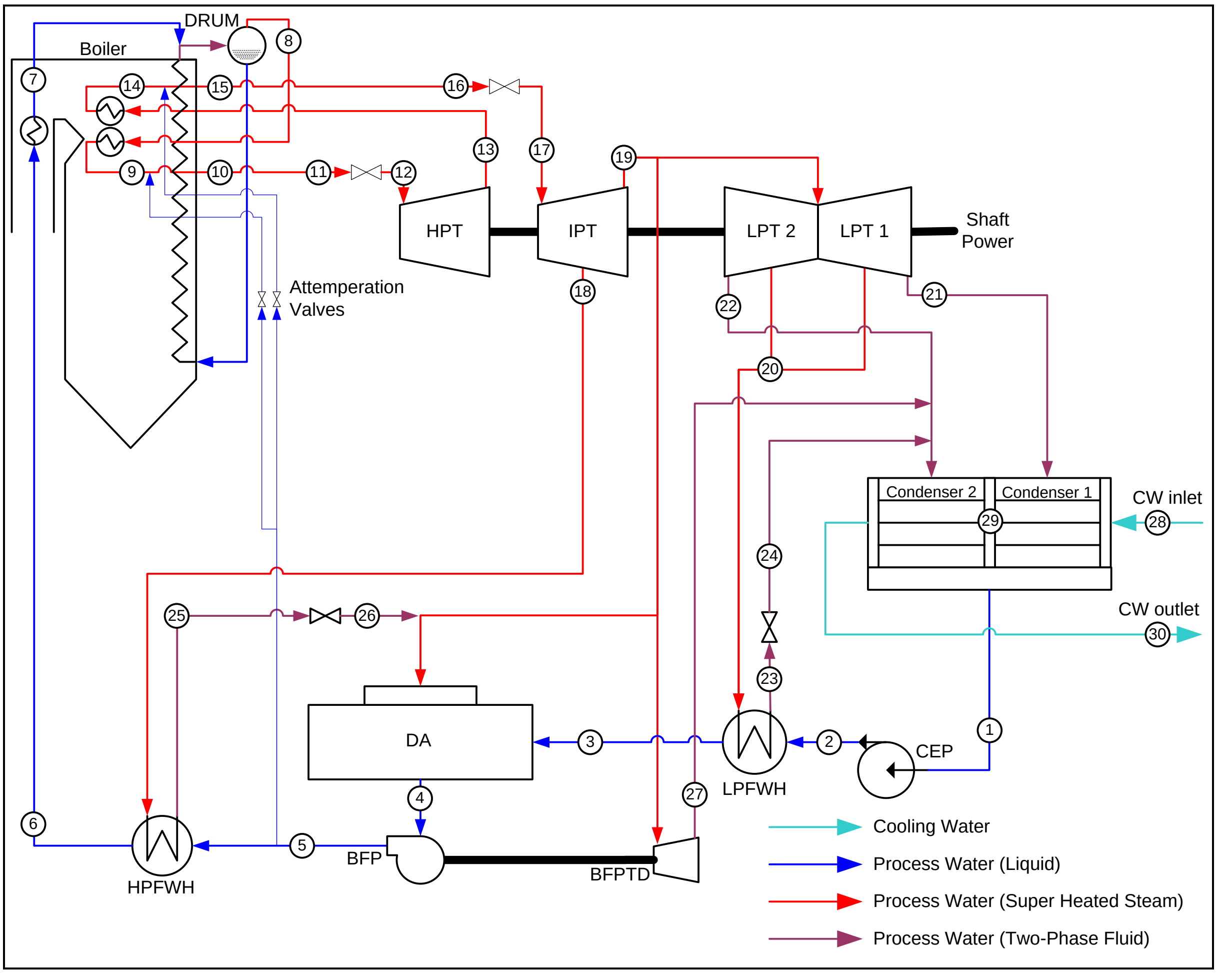

Figure 5: The Rankine cycle process configuration which has been used as the reference cycle of this study.

It was assumed that the bled steam condenses completely in the LP FWH, but is not subcooled. The distillate (point 23) was expanded through a throttling valve. The pressure at the outlet of the throttling valve (point 24) was set equal to the outlet pressure of one of the LP turbines. The two-phase fluid (point 24) was fed into the RCC.

Point 3 is the FW inlet of the DA. The DA heats the FW using bled steam, extracted at the outlet of the IP TURBINE, and the distillate coming from the HP FWH. The heating in a DA is done by physically mixing all the incoming fluids. The DA has a single outlet (point 4). It was assumed that the pressure is constant throughout in the DA and the FW is a saturated liquid at the outlet of the DA. The mass flow of the bled steam and temperature at the outlet were calculated with the conservation laws, described in section 3.3.

The steam turbine driven BFP pressurises the FW, extracted from the DA. The pressure at the BFP outlet (point 5) is the required HP turbine throttling valve inlet pressure (point 11) plus all the pressure losses between point 5 and point 11. The conditions at the BFP outlet were calculated with the same methods used for CEP.

A HP FWH was used to add more heat to the FW before entering the boiler. The HP FWH and the LP FWH is similar in function and description. In the HP FWH the FW is heated between points 5 and 6, with the use of bled steam extracted from the IP TURBINE (point 18). At the HP FWH, the distillate is fed into the DA (point 26), after expanding the throttling valve.

The feedwater (point 6) is fed into the boiler. In the boiler FW is initially heated in the economizer. It was assumed that the fluid exiting the economizer (point 7) is saturated liquid. The saturated liquid is fed into the drum, which serves as a separator.

The drum is connected to a number of natural circulation HX's, i.e. the boiler water walls. These natural circulation HX's heat the saturated water, extracted from the bottom of the drum and returns a two-phase mixture. This mixture is fed back into the top part of the drum, where it passes through a number of separators located in the drum. The saturated steam, accumulating in the top section of the drum is fed into the super heaters (point 8).

The attemperation FW spray is added to the steam between the outlet of the SH's (point 9) and point 10. The section between points 10 and 11 represents the section of pipe between the boiler and the turbine inlet valve, where heat is lost to the atmosphere. The heat loss in this section is calculated from the calculated conditions at points 10 and 11.

The steam at point 11 is expanded through the HP turbine inlet valve. The valve is used in practice to control the HP turbine inlet pressure at a fixed value. The combined effect of this valve's control capability and the steam attemperation process enables the control of the enthalpy at the turbine inlet.

Between points 12 and 13, the steam is expanded in the HP turbine to produce work. The steam at point 13 re-enters the boiler to be reheated. The process sequence between the HP turbine outlet (point 13) and the IP turbine inlet (point 17) is a repeat of the process sequence between the drum outlet (point 8) and the HP turbine inlet (point 12).

The steam at point 17 is expanded in the IP turbine. A small portion of the steam is extracted at point 18, after partially expanding. This steam is routed to the HP FWH. The rest of the steam is expanded further towards the outlet of the IP turbine.

The portion of the steam at the outlet of the IP turbine (point 19) is used as the heating medium in the DA, another portion is routed to the BFPTD and the rest of the steam is expanded in two LP turbines. The steam routed to the BFPTD is expanded in the turbine. The resulting shaft power is used to drive the BFP.

Partially expanded steam is also extracted from both the LP turbines. This steam is used as heating medium in the LP FWH. The remaining steam is expanded further to points 21 and 22.

The steam at the outlet of each LP turbine is routed to separate RCC's. These RCC's are cooled with cooling water, entering the first condenser at point 28. After the first condenser, the CW enters a water box (point 29), from where it is distributed into the second condenser. The CW at point 30 is routed back to the cooling towers. The condensers are therefore in series, when referring to the CW cycle. With this configuration, the first condenser operates at a lower saturation temperature. The pressure in this condenser is therefore lower, which means an increased power output of the connected LP turbine is achieved.

This condenser pressure difference causes the different liquid levels in the combined hot well of the condenser. The pressure at point 1 was therefore set equal to the pressure of the condenser with the highest operating pressure (the 'hot' condenser).

The fluid returning from the LP FWH and the BFPTD is added to the fluid entering the 'hot' condenser.

4.3 Modelling Inputs

The performance of any thermal cycle is dictated by the process conditions and the properties of each component of the cycle. This information, which served as inputs to reference Rankine cycle simulation model, will now be discussed.

4.3.1 Process Inputs

The fluid quality, temperatures and pressures at various points throughout the cycle were specified. Typical process conditions found in literature were used as inputs to the model, where possible. Unfortunately values for all the required process conditions were not found in literature, since the cycle presented in Figure 5 is a simplified representation of actual power station. Estimated values were used in these cases. These inputs are described below, with the specific values presented in Table 1.

The two major process inputs of any thermal cycle are the limiting temperatures, i.e. the temperatures of the heat source and heat sink or the maximum and minimum working fluid temperatures. In this study these temperatures are specified as the 'maximum' fluid temperature and the temperature of the heat sink.

The maximum fluid temperature of the reference Rankine cycle was specified at points 10 and 17. This is the bulk temperature of the fluid before entering the down-comers leading to the HP turbine inlet value. These points were selected since the temperature there is the maximum fluid temperature of a typical Rankine cycle configuration used in power stations. Kindly note, that it is not the actual maximum fluid temperature of the simulation model, due to the attemperation modelling configuration used in this study (discussed in section 3.6).The heat sink temperature input was specified as the inlet temperature CW.

Temperatures were also specified at the inlets of the HP turbine and the IP turbine, along with the pressures at these points. The other process properties that were specified are the pressure at each steam extraction point and the two-phase quality of the fluid at various points.

Table 1: Rankine cycle process inputs

Position	Point nr.	Value	Origin of Value
Temperatures [°C]			
HP down-comer inlet	10	540	Estimated
HP turbine inlet	12	538	Beér, 2007:109
IP down-comer inlet	15	540	Estimated
IP turbine inlet	17	538	Beér, 2007:109
Condenser CW inlet	28	20	Estimated
Pressure [MPa]			
HP turbine inlet	12	16.8	Beér, 2007:109
IP turbine inlet	17	3.4	Estimated
HP FWH steam inlet	18	2	Estimated
DA steam inlet	19	0.6	Estimated
HP FWH steam inlet	20	0.15	Estimated
Two-Phase Quality [0 – Saturated Liquid & 1 – Saturated Vapour]			
DA outlet	4	0	Assumption
Economiser outlet	7	0	Assumption
SH inlet	8	1	Assumption
LP FWH hot outlet	23	0	Assumption
HP FWH hot outlet	25	0	Assumption

4.3.2 Turbo Machine Efficiencies

The isentropic efficiencies used in the Rankine cycle simulation model are listed in Table 2.

Table 2: Isentropic efficiencies of the Rankine cycle's turbo machines

Turbo Machine Identifier	Between Points	Isentropic Efficiency [%]
CEP	1 & 2	85
BFP	4 & 5	85
HP turbine	12 & 13	89
IP turbine	17 & 19	89
LP turbines	19 & 21/22	85
BFPTD	19 & 27	85

It was already mentioned that the CEP will be running from electrical power. Thus, the pumping power of the CEP had to be converted to the equivalent shaft power. This was done by specifying efficiencies for each of the power conversion components between the FW pump and the main turbine shaft.

These components and their efficiencies are listed below:

- the generator of the plant (97%),
- a transformer (99.5%),
- the electric motor of the pump (97%)
- and the fluid coupling gearbox (75%)

These efficiencies were combined and modelled as an energy conversion efficiency (70.2%).

4.3.3 Component Pressure Losses

There are a number of factors contributing to pressure losses in a thermal cycle, but as mentioned in section 3.3.2, these pressure losses were lumped during this study. The pressure losses used, is listed in Table 3.

Table 3: Component pressure losses.

Component Identifier	Between Points	Pressure Loss [MPa]
LP FWH (FW)	2 & 3	0.2
HP FWH (FW)	5 & 6	0.2
Economizer	6 & 7	3.7
SH	8 & 9	1.2
HP turbine Inlet Valve	11 & 12	0.1
Reheaters	13 & 14	1.2
IP turbine Inlet Valve	16 & 17	0.1

4.3.4 Heat Exchanger Performance

Typical TTD values that are used by researchers for FWH's range from -3 °C and 3 °C (Hajabdollahi, et al., 2012:3651; Mittelman & Epstein, 2010:1768; Xiong, et al., 2012:490).

In this study, a TTD of 0 °C was selected for both FWH's and it was assumed that the distillate is not subcooled in the FWH's.

4.4 RANKINE CYCLE SIMULATION RESULTS

The simulation model of the cycle described in this chapter, with its results can be found in the appendix, section 8.1.

The Rankine cycle was simulated with total circulation flow rate of 1 kg/s, which is the flow rate through the BFP and the first section of the IP turbine.

The power output of the turbines was calculated as 330.5 kW for the HP turbine, 438.3 kW for the IP turbine and 508.3 for the LP turbines. The work required by the CEP was calculated as 0.7957 kW, which result an equivalent shaft power of 1.133 kW. This resulted in a net turbine shaft power output of 1 276 kW. The BFP power requirement of 27.56 kW is not subtracted from the total power produced by the turbine, since it is driven by a bled steam turbine.

The heat absorbed in the boiler was calculated as 2 964 kW and from these values the efficiency of the Rankine cycle was calculated as 43.05 %. The efficiency of a typical Rankine cycle operated power plant is 35 %, but this includes boiler losses and auxiliary power requirements. The calculated Rankine cycle efficiency therefore correlates well with the efficiency of a practical Rankine cycle.

The T-s diagram of the Rankine cycle is presented in Figure 6.

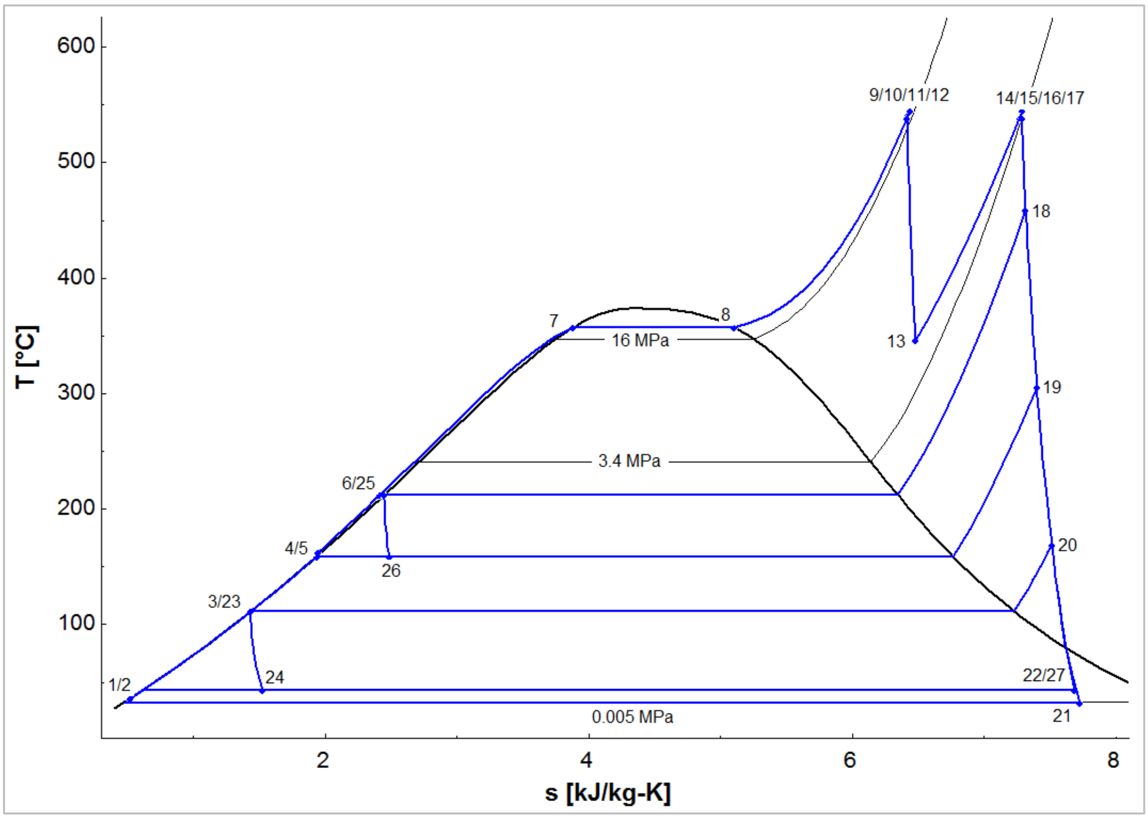

Figure 6: The T-s diagram of the Rankine cycle simulation model.

The process conditions of this Rankine cycle are presented in Table 4.

Table 4: Process conditions results for the reference Rankine simulation model.

Nr.	Pressure [MPa]	T [°C]	h [kJ/kg]	s [kJ/kg-K]	rho [kg/m3]
1	0.008651	35.59	149.1	0.5131	993.8
2	0.8	35.64	150	0.5136	994.1
3	0.6	111.3	467.5	1.433	950.1
4	0.6	158.8	670.4	1.931	908.6
5	22	162.3	697.9	1.94	917.6
6	21.8	212.4	915.8	2.414	865.3
7	18.1	357.5	1737	3.879	541.2
8	18.1	357.5	2506	5.099	134.8
9	16.9	545	3416	6.432	50.27
10	16.9	540	3402	6.414	50.75
11	16.9	538.4	3397	6.409	50.91
12	16.8	538	3397	6.411	50.61
13	4.7	345.8	3066	6.472	18.14

Nr.	Pressure [MPa]	T [°C]	h [kJ/kg]	s [kJ/kg-K]	rho [kg/m3]
14	3.5	545	3554	7.288	9.472
15	3.5	540	3542	7.274	9.536
16	3.5	538.4	3539	7.269	9.556
17	3.4	538	3539	7.282	9.282
18	2	458.5	3377	7.312	6.038
19	0.6	305.8	3074	7.395	2.278
20	0.15	169	2811	7.509	0.7428
21	0.00482	32.22	2349	7.718	0.03753
22	0.008651	43	2414	7.679	0.06386
23	0.15	111.3	467.1	1.434	949.9
24	0.008651	43	467.1	1.52	0.4968
25	2	212.4	908.5	2.447	849.8
26	0.6	158.8	908.5	2.482	27.03
27	0.008651	43	2414	7.679	0.06386
28	0.1	20	84.01	0.2965	998.2
29	0.1	29.22	122.6	0.426	995.9
30	0.1	40	167.6	0.5724	992.2

The various process water flow rates were all calculated as a fraction of the total circulating flow, which is the flow rate through the BFP and the first section of the IP turbine. The flow rates of the various bled steam streams, the attemperation process, BFPTD and the last section of each LP turbine are presented in Table 5. The required CW flow rate was calculated as 20.07 kg/s at a cycle circulation flow rate of 1 kg/s through the BFP.

Table 5: Various Rankine cycle flow rate results.

Process Stream	Flow Rate [kg/s]
HP FWH Bled Steam	0.08745
DA Bled Steam	0.06305
LP FWH Bled Steam	0.115
SH's attemperation	0.005175
RH's attemperation	0.003973
BFPTD steam	0.04397
LP turbine 1	0.3476
LP turbine 2	0.3429
CW	20.07

One of the main outcomes of this Rankine simulation model was to determine the operating ranges of the vapour compression cycle. These results will be discussed in the relevant sections of the next chapter.

CHAPTER 5: THE HEAT PUMP CYCLE

5.1 INTRODUCTION

Heat pumps are used to transfer heat from one fluid to another against the temperature gradient. The major heat pump configurations that are normally used, are the absorption cycle, the vapour compression cycle and jet refrigeration system.

These cycles are used for cooling or heating applications. Typical applications are:

- Cooling
  - Refrigeration
  - Space cooling
- Heating
  - Domestic water heating (pool or geyser)
  - Space heating

The performance of heat pumps is expressed in terms of the coefficient of performance (COP) and the vapour compression cycle generally has the highest COP. It was therefore selected as the heat pump of the Rankine-heat pump combined cycle.

5.2 CYCLE REQUIREMENTS

The function of the vapour compression cycle within the combined cycle had to be established, in order to select a cycle configuration, working fluid and operating parameters for the vapour compression cycle.

The goal of this study is to evaluate the concept of using a compressor driven vapour compression cycle, instead of LP FWH's, to heat the feedwater of a Rankine cycle, as mentioned in section 1.3.

The results of the Rankine cycle simulation model shows that the FW temperature at the outlet of the LP FWH (point 3) is 111.3 °C.

It was also decided to connect the evaporator to the CW supply of the RCC. This configuration was selected to reduce the operating temperatures and pressures of the RCC's by reducing the temperature of the CW.

The requirements of the vapour compression cycle can therefore be defined as:

1. *Heat the FW of the Rankine cycle to a Temperature of 111.3 °C.*
2. *Reduce the temperature of the CW.*

5.3 Cycle COP

As mentioned, the performance of a heat pump is generally expressed in terms of the COP of the cycle. Normally the COP is either expressed in terms of heat performance or cooling performance, depending on the objective of the cycle.

The COP of a heating vapour compression cycle ($\mathrm{COP_H}$) is expressed as:

$$\mathrm{COP_H} = \frac{\mathrm{Q_H}}{\mathrm{W_c}} \quad \textbf{15}$$

where:

$\mathrm{W_C}$ – is the cycle energy input in the form compressor work and

Q_H – is the heat rejected in the condenser of the heat pump.

The COP of a cooling vapour compression cycle ($\mathrm{COP_C}$) is expressed as:

$$\mathrm{COP_C} = \frac{\mathrm{Q_L}}{\mathrm{W_c}} \quad \textbf{16}$$

where:

Q_L – is the heat absorbed in the evaporator of a heat pump.

5.4 Cycle Configuration

A vapour compression cycle mainly consists of four different components, i.e.:

- evaporator(s),
- compressor(s),
- condenser(s) and
- expansion valve(s)

Various cycle configurations have been analysed by researchers over the years. Typical cycle configurations found in literature are:

- Configurations with multiple evaporators (Winkler, et al., 2008:931)
- Configurations with gas cooling between multiple compressor stages (Bertsch & Groll, 2008:1285)
- Configuration with cascading cycles (Bertsch & Groll, 2008:1285; Ratts & Brown, 2000:355)

The optimum configuration is normally application specific and the selection of such a configuration can be a quite an involved study. After careful consideration it was concluded that this will not form part of the scope of this study. A basic cycle configuration was therefore selected for this study.

The cycle configuration of the vapour compression cycle, selected for this study, is presented in Figure 7.

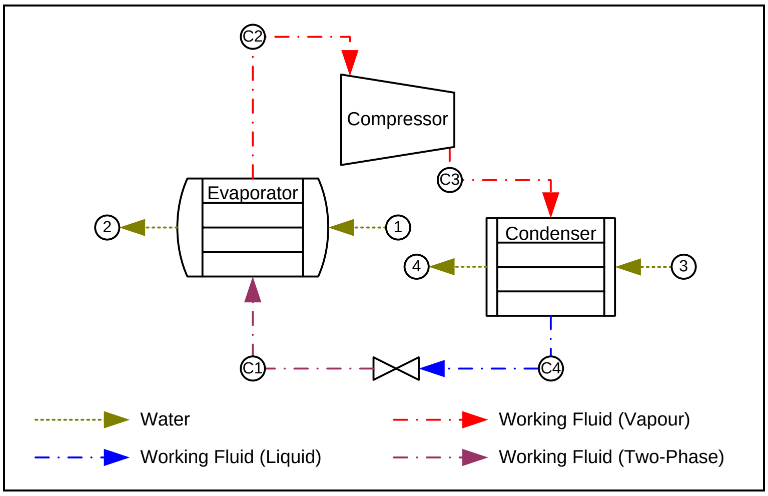

Figure 7: The basic vapour compression cycle configuration.

In this cycle the working fluid enters the evaporator at point C1. At this point the working fluid is a two-phase mixture. The working fluid is fully evaporated in the evaporator by absorbing heat from water, which reduces the temperature of the water.

The fluid at point C2 is then compressed in the compressor. This process increases the temperature and saturation temperature of the fluid. Heat can therefore now be rejected at a higher temperature.

In this cycle, heat is rejected in a single condenser, where the working fluid is fully condensed and subcooled. This heat is rejected to water (Rankine cycle FW), which is heated in the process.

An expansion valve is placed on the outlet of the condenser to lower the fluid pressure and close the cycle.

5.5 Modelling Inputs

5.5.1 Process Inputs

It has already been established in section 5.2 that the vapour compression has to heat the water exiting the CEP and reduce the temperature of the CW.

The process conditions at points 1, 2 and 3 could therefore be obtained from the Rankine cycle results, presented in Table 4. The flow rates of the FW and CW were obtained from Table 5.

The only other process condition that was required for the simulation model is the process conditions at point C2. For this project it was decided that no superheating will occur within the evaporator.

The values of the process condition inputs discussed above are presented in Table 6.

Table 6: Vapour compression cycle process inputs

Position	Point nr.	Value
Enthalpy [kJ/kg]		
Evaporator inlet (CW)	1	84.01
Condenser inlet (FW)	3	150.0
Condenser outlet (FW)	4	467.5
Pressure [MPa]		
Evaporator inlet (CW)	1	0.1
Condenser inlet (FW)	3	0.8
Condenser outlet (FW)	4	0.6
Two-Phase Quality [0-1]		
Evaporator outlet	C2	0
Mass Flows [kg/s]		
CW	n/a	20.07
FW	n/a	0.8495

5.5.2 Turbo Machine Efficiencies

The compressor is the only turbo machine to form part of the vapour compression cycle. An efficiency of 85 % was used for the compressor.

5.5.3 HX Performance

It is assumed that no subcooling will occur in the condenser, since it will replace the LP FWH and should therefore be modelled using the same assumptions.

The other HX that forms part of the cycle is the evaporator, which was modelled with an effectiveness of 85 %.

5.5.4 Component Pressure Losses

The only component that was modelled with a pressure drop was the condenser. The FW side of the condenser was also modelled with the same pressure loss that was used for the LP FWH.

5.6 WORKING FLUIDS

A list of countries has undertaken an agreement, the Kyoto Protocol, to reduce the emission of greenhouse gases. Some of the vapour compression cycle working fluids, such as R22, were classified as greenhouse gases. Another fact that has to be noted is that applications where vapour compression cycles are used operate at lower temperatures than required by the combined cycle.

A working fluid could therefore not be selected from previous studies. Another working fluid selection process was required instead.

5.6.1 The Working Fluid Selection Process

The fluid data base of EES was used as the starting point. The first steps of the selection process were aimed at eliminating fluids which could not be used for this application. The elimination steps that were followed, are discussed below:

1. *Eliminate fluids associated with green house and ozone depleting gases:*

   Motivation:

   The use of these types of gases in any application is currently being phased out throughout the world, as mentioned earlier in this section. It was therefore also not considered for this study.

2. *Eliminate fluids with critical temperatures below 111.3 °C.*

   Motivation:

   The fluid was eliminated because the cycle configuration that was selected for this study requires that the working fluid should condense in the condenser, with a final distillate temperature of 111.3 °C.

3. *Eliminate fluids if the slope of its saturate vapour line is positive on a T-s diagram.*

   Motivation:

   The fluid at the outlet of a compressor, in which the saturated vapour of such a fluid is compressed, is a wet vapour, as shown in Figure 8.

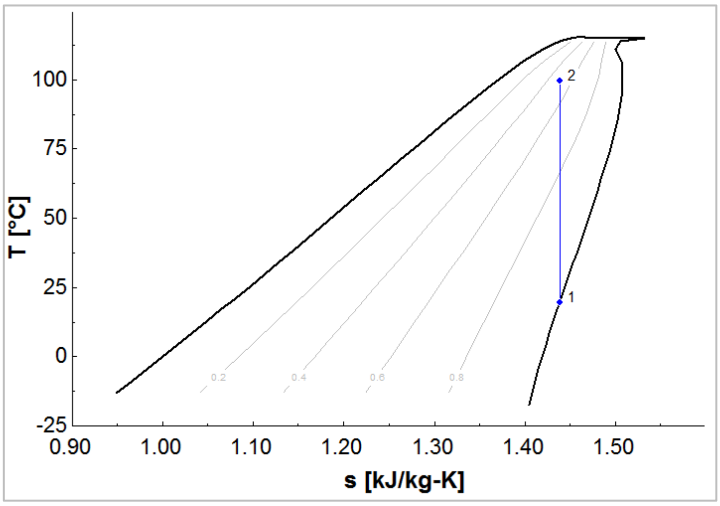

Figure 8: A T-s diagram, showing the compression of the saturated vapour of Perfluorocyclobutane (RC-318).

The simulation model of the vapour compression cycle described in this chapter was developed. This simulation model was used to calculate the cycle COP for each of the remaining fluids.

5.6.2 The Working Fluid Selection Process Results

The three working fluids which yielded the best cycle COP's are Ethanol, Methanol and Isopropanol. All three of these working fluids are unfortunately highly flammable. The non-flammable/non-toxic fluid which yielded the highest COP was water. These COP's are listed in Table 7.

Table 7: The cycle COP of the vapour compression cycle for the best performing working fluids.

Working Fluid	Cooling COP	Heating COP
Ethanol	2.015	3.015
Methanol	2.005	3.005
Isopropanol	1.927	2.927
Water	1.736	2.736

Evidence has been found that flammable fluids have been considered as working fluids of refrigeration systems in the past (McQuay International, 2002:44). The COP of the vapour compression cycle with water as working fluid is low when compared to the other working fluids. Water was therefore not considered as working fluid for this study.

It should also be noted that fluid properties, such as heat transfer coefficients and viscosity were neglected in the selection of the working fluid. These properties can have a significant impact on the cycle pressure losses and HX effectiveness. Both of which impact the cycle performance directly.

In the end the working fluid was selected purely on cycle COP results. Ethanol was therefore selected as the working fluid of the vapour compression cycle in the combined cycle.

5.7 Vapour Compression Cycle Simulation Results

The main results of the vapour compression cycle, as described in this chapter and with ethanol as working fluid, are presented in this section. The simulation model of the cycle and all its results can be found in the appendix, section 8.2.

The process condition results of the vapour compression cycle are presented in the table below.

Table 8: Process condition results for the reference vapour compression cycle simulation model.

Nr.	Pressure [MPa]	T [°C]	h [kJ/kg]	s [kJ/kg-K]	rho [kg/m3]
1	0.005029	17.48	368.3	1.464	0.3184
2	0.005029	17.48	1020	3.707	0.02507
3	0.328	202.9	1343	3.811	3.911
4	0.328	111.4	368.3	1.328	700

It can be seen from the results presented in the table above, that the pressures of the condenser and evaporator were calculated as 328.0 kPa and 5.029 kPa, respectively. This pressure is lower than the operating pressure of the RCC, where air and water in seepage are experienced at power stations. It should therefore be noted that the low operating pressure of the evaporator may have practical implications.

The T-s diagram of this vapour compression cycle is presented in Figure 9.

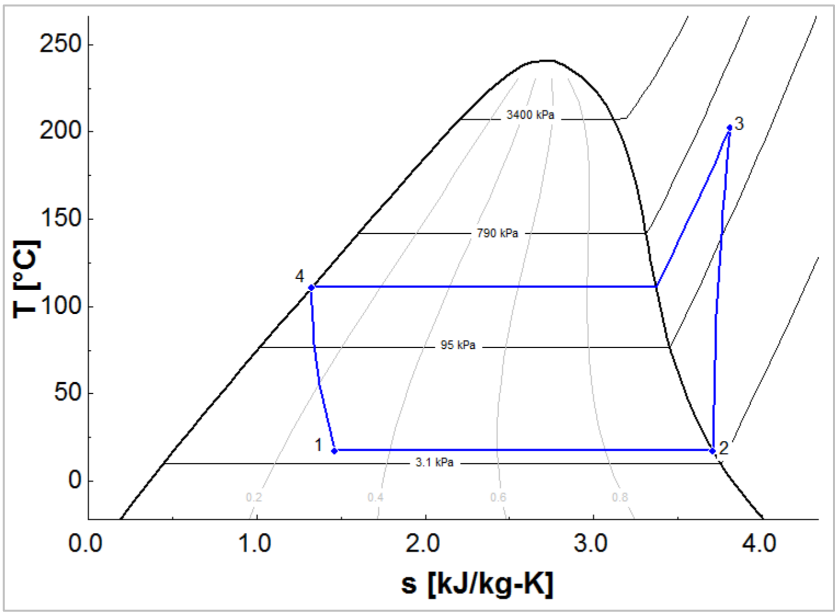

Figure 9: The T-s diagram of the selected vapour compression cycle.

It was calculated that an ethanol mass flow rate of 0.277 kg/s is required in order to heat the FW of the Rankine cycle to a temperature of 111.3 °C. The resulting condenser heat transfer rate was calculated as 269.7 kW, which corresponds with the heat transfer rate within the LP FWH of the Rankine cycle.

The energy required by the compressor was calculated as 89.46 kW and resulting heating COP of the cycle is calculated as 3.015.

The heat absorbed from the CW of the RCC was calculated as 180.3 kW. This process reduces the temperature of the CW by 2.15 °C. This reduction in CW temperature will in essence reduce the operating temperature of the two RCC's.

CHAPTER 6: COMBINED CYCLE

6.1 INTRODUCTION

The Rankine cycle simulation model and the vapour compression cycle have now been developed separately. All that remains is the integration of these cycles. This chapter will cover the simulation model of the combined Rankine-heat pump cycle.

6.2 CYCLE INTEGRATION

The functions and connection interfaces of the condenser and evaporator have already been discussed in the previous chapter. The only interface that still requires discussion is the power source of the compressor.

Two possible sources of power were evaluated for this study. These power sources are:

- an electric motor and
- a steam turbine drive.

The Stirling engine, using bled steam as heat source, was considered as a possible source of shaft power. The Stirling engine efficiencies that were found in literature indicate that efficiencies of ~30 % can be expected for temperature differences of about 120 °C (Kongtragool & Wongwises, 2003:148; Kongtragool & Wongwises, 2008:499). A power conversion efficiency of 30 % and the vapour compression cycle COP of 3.015 will essentially cancel out. There will therefore be no gain with this option.

The cycle configuration of the Rankine-heat pump combined cycle compressor steam turbine drive (CTD) is presented in Figure 10.

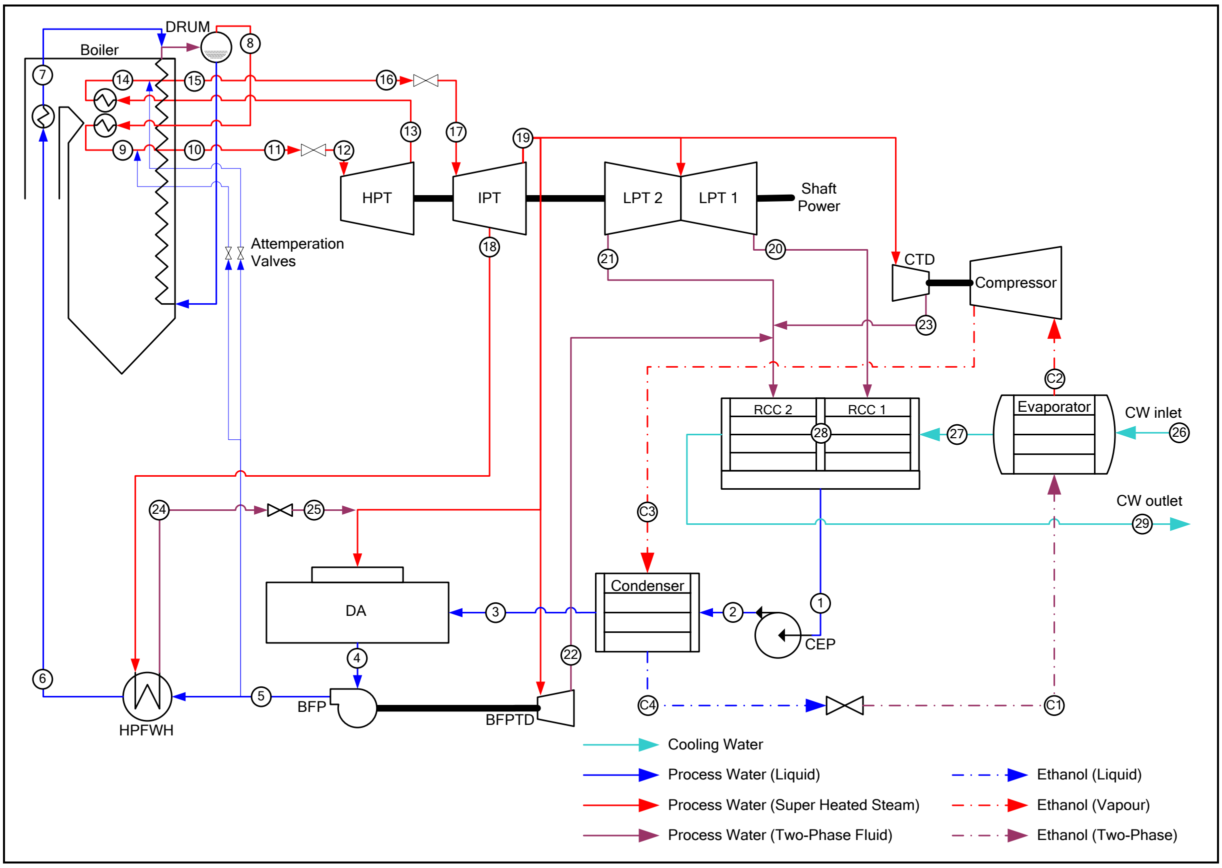

Figure 10: The Rankine-heat pump combined cycle with a turbine driven compressor.

The modelling assumptions that were used for these two compressor drive options will be discussed in the rest of this section.

6.2.1 The Electric Motor Driven Compressor

In this configuration, the power required by the compressor was converted back to an equivalent shaft power, similar to the modelling process that was used for the CEP. The efficiency values, as described for the CEP in section 4.3.2, were also used to power conversion for the electric motor driven compressor.

6.2.2 The Steam Turbine Driven Compressor

In this instance, the modelling approach and efficiency value that was used for the BFPTD was also used to model the compressor steam turbine driven compressor. The steam for the turbine was also extracted at the outlet of the IP turbine.

6.3 Rankine-Heat Pump Combined Cycle Simulation Results

The simulation model of the cycle described in this chapter, with all the results can be found in the appendix, section 0.

The section of the Rankine cycle, from the DA FW inlet to the LP turbines steam inlet, did not change as can be seen when comparing Figure 5 and Figure 10. The process input also did not change. This can also be seen from the results of the Rankine-heat pump combined cycle.

The power output of the HP turbine and IP turbine for the two turbine drive options was calculated as 330.5 kW for the HP turbine and 438.4 kW for the IP turbine.

The power output of the LP turbines of the Rankine-heat pump combined cycle, for the configuration with an electric motor drive, was calculated as 538.1 kW. The work required by the compressor was calculated as 89.85 kW, which resulted in an equivalent shaft power of 128 kW. The equivalent shaft power of the CEP was calculated as 1.133 kW. This resulted in a net turbine shaft power output of 1 177 kW.

The power output of the LP turbines of the Rankine-heat pump combined cycle, for the configuration with the CTD, was calculated as 442.6 kW. The net turbine shaft output of this configuration was calculated as 1 210 kW.

The process condition results of these two combined cycle configurations were the same, except for the conditions at point 23. This point does not form part of the configuration with an electric motor drive. These results are presented in Table 9.

Table 9: The process condition results of the combined cycle simulation model.

Nr.	Pressure [MPa]	T [°C]	h [kJ/kg]	s [kJ/kg-K]	rho [kg/m3]
1	0.008651	35.23	147.6	0.5083	993.9
2	0.8	35.29	148.6	0.5088	994.2
3	0.6	111.4	467.5	1.433	950.1
4	0.6	158.8	670.4	1.931	908.6
5	22	162.3	697.9	1.94	917.6
6	21.8	212.4	915.8	2.414	865.3
7	18.1	357.5	1737	3.879	541.2
8	18.1	357.5	2506	5.099	134.8
9	16.9	545	3416	6.432	50.27
10	16.9	540	3402	6.414	50.75
11	16.9	538.4	3397	6.409	50.91
12	16.8	538	3397	6.411	50.61
13	4.7	345.8	3066	6.472	18.14
14	3.5	545	3554	7.288	9.472
15	3.5	540	3542	7.274	9.536
16	3.5	538.4	3539	7.269	9.556
17	3.4	538	3539	7.282	9.282
18	2	458.5	3377	7.312	6.038
19	0.6	305.8	3074	7.395	2.278
20	0.004423	30.71	2365	7.807	0.03431
21	0.008651	43	2437	7.751	0.06322
22	0.008651	43	2437	7.751	0.06322
23*	0.008651	43	2437	7.751	0.06322
24	2	212.4	908.5	2.447	849.8
25	0.6	158.8	908.5	2.482	27.03
26	0.1	20	84.01	0.2965	998.2
27	0.1	18.11	76.11	0.2694	998.6
28	0.1	27.71	116.2	0.405	996.3
29	0.1	40	167.6	0.5724	992.2
C1	0.005124	17.78	368.3	1.463	0.3248
C2	0.005124	17.78	1020	3.705	0.09698
C3	0.328	202.5	1342	3.809	3.915
C4	0.328	111.4	368.3	1.328	700

* This is the compressor turbine drive outlet, which only forms part of the cycle configuration with the steam turbine drive.

The various fluid flow rates were all calculated as a fraction of the total circulating flow of the Rankine cycle. The flow rates of the various bled steam streams, the attemperation process, BFPTD and the last section of each LP turbine are presented in Table 10 for both the cycle configurations. This table also contains the required CW flow rate and ethanol flow rate of both the cycle configurations.

Table 10: Combined cycle flow rate results.

Process Stream	Flow Rate (CTD) [kg/s]	Flow Rate (Electric) [kg/s]
HP FWH Bled Steam	0.08752	0.08752
DA Bled Steam	0.06317	0.06317
SH's attemperation	0.005175	0.005175
RH's attemperation	0.003973	0.003973
BFPTD steam	0.04567	0.04567
CTD	0.1479	0
LP turbine 1	0.3294	0.4034
LP turbine 2	0.3282	0.4021
CW	22.98	22.97
Ethanol	0.2783	0.2791

The heat transfer rate between the FW and the ethanol, within the condenser of the vapour compression cycle was calculated as:

- The configuration with an electric motor drive - 271.8 kW
- The configuration with the steam turbine drive - 271.0 kW

The amount of energy that is transferred to the FW of the Rankine cycle is therefore significantly higher than the power required by the compressor.

In both cases the temperature of the cooling water is also reduced by ~1.9 °C. According to Carnot's law, the efficiency of a thermal cycle increases if the difference between the temperatures of the heat source and heat sink increases.

At first glance it seems that the efficiency of the combined cycle should improve, based on the facts mentioned above. This is in fact not the case. The thermal efficiencies of the two combined cycle configurations were calculated as:

- The configuration with an electric motor drive - 39.72 %
- The configuration with the steam turbine drive - 40.82 %

This is significantly lower than the efficiency of the reference Rankine cycle, which was calculated as 43.05 %.

After a detailed analysis of the simulation results it was found that the effect of latent heat was neglected/under-estimated. The change in the specific enthalpy of the bled steam in the LP FWH is 2 344 kJ/kg. The average change in the specific enthalpy of the remaining steam, between the LP FWH steam extraction point and the outlet of the LP turbines, is 430 kJ/kg. This is 5.5 times lower than the change in the specific enthalpy of the bled steam.

The COP of the heat pump cycle, driven with power generated from steam turbines, should therefore be above 5.5 in order to improve the thermal efficiency of a Rankine cycle with the proposed configuration.

CHAPTER 7: CONCLUSION AND RECOMMENDATIONS

7.1 CONCLUSION

The simulation models of a Rankine cycle, a vapour compression cycle and two Rankine-heat pump combined cycles were developed during this study.

The Rankine cycle was modelled to adequate detail. The efficiency of the reference Rankine cycle was calculated as 43.05 %.

The best COP's for a basic vapour compression cycle was obtained with ethanol as working fluid. The best cycle COP's with a non-toxic/non-flammable working fluid were obtained water, but it was significantly lower than the COP of the ethanol cycle. Ethanol was therefore used as the working fluid, yielding a heating COP of 3.015.

The only difference between the two combined cycle configurations is the compressor's power source. The two compressor power source options that were analysed, were a steam turbine and an electric motor. The Stirling engine was also considered as a possible compressor power source. The typical efficiencies that can be expected from a Stirling engine operating at the temperature ranges available in the combined cycle configuration were too low to make this a viable option.

The thermal efficiencies of both the two combined cycle configurations are significantly lower than the efficiency reference Rankine cycle. It was found that a heat pump COP of above 5.5 will be required in order to improve the thermal efficiency of the Rankine cycle with the combined cycle configurations tested in this study.

7.2 RECOMMENDATIONS

This study showed that the concept of using a steam turbine or electrical power to drive a compressor driven vapour compression cycle in the configuration proposed here does not improve the overall efficiency of the cycle. An alternative option that warrants investigation is the use of thermal driven pumps such as an absorption cycle, although the COP of these cycles are typically lower.

The argument that warrants the investigation as such is the fact that the latent heat of the bled steam can be fully utilised and not rejected to the heat sink via the condenser, which is the case with turbine work bled steam.

CHAPTER 8: APPENDIX

8.1 Rankine Cycle

8.1.1 Simulation Model

--*Component Functions*--

RCC

MODULE **Condenser** (T_R, TTD_{COND}, T_{SUB}, m_6, m_7, m_5, m_4, $P_{SAT,COND1}$, $P_{SAT,COND2}$, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, $h_{COND1,in}$, $h_{COND2,in}$, $h_{LP,FD,HTR,out}$, q_{loss}, $T_{CW,in}$, $P_{CW,in}$, $h_{CW,in}$, $s_{CW,in}$, $\rho_{CW,in}$, $T_{CW,int}$, $h_{CW,int}$, $s_{CW,int}$, $\rho_{CW,int}$, $T_{CW,out}$, $h_{CW,out}$, $s_{CW,out}$, $\rho_{CW,out}$)

Condenser Performance

$$T_{CW,out} = T_{CW,in} + T_R$$

$$T_{SAT,COND1} = T_{CW,int} + TTD_{COND}$$

$$T_{SAT,COND2} = T_{CW,out} + TTD_{COND}$$

Condenser Pressures

$$P_{SAT,COND1} = \mathbf{P}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{SAT,COND1}, x = 0 \right)$$

$$P_{SAT,COND2} = \mathbf{P}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{SAT,COND2}, x = 0 \right)$$

Outlet Conditions per Condenser

$$T_{COND1,out} = T_{SAT,COND1} - T_{SUB}$$

$$h_{COND1,out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{SAT,COND1}, T = T_{COND1,out} \right)$$

$$T_{COND2,out} = T_{SAT,COND2} - T_{SUB}$$

$$h_{COND2,out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{SAT,COND2}, T = T_{COND2,out} \right)$$

Outlet Mass Flow Rates per Condenser

$$\dot{m}_{COND1,out} = \dot{m}_6$$

$$\dot{m}_{COND2,out} = \dot{m}_7 + \dot{m}_4 + \dot{m}_5$$

Combined Outlet Conditions

$$P_{out} = P_{SAT,COND2}$$

$$T_{out} = \mathbf{T}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out} \right)$$

$$h_{out} \cdot \left(\dot{m}_{COND1,out} + \dot{m}_{COND2,out} \right) = \dot{m}_{COND1,out} \cdot h_{COND1,out} + \dot{m}_{COND2,out} \cdot h_{COND2,out}$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out})$$

Cooling Water Enthalpy

$$h_{CW,in} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,in}, P = P_{CW,in})$$

$$h_{CW,int} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,int}, P = P_{CW,in})$$

$$h_{CW,out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,out}, P = P_{CW,in})$$

Energy Balances

$$\dot{m}_{CW} \cdot (h_{CW,int} - h_{CW,in}) = \dot{m}_6 \cdot (h_{COND1,in} - h_{COND1,out})$$

$$\dot{m}_{CW} \cdot (h_{CW,out} - h_{CW,int}) = (\dot{m}_5 + \dot{m}_7) \cdot (h_{COND2,in} - h_{COND2,out}) + \dot{m}_4 \cdot (h_{LP,FD,HTR,out} - h_{COND2,out})$$

$$q_{loss} = \dot{m}_6 \cdot (h_{COND1,in} - h_{COND1,out}) + (\dot{m}_5 + \dot{m}_7) \cdot (h_{COND2,in} - h_{COND2,out}) + \dot{m}_4 \cdot (h_{LP,FD,HTR,out} - h_{COND2,out})$$

Other CW Properties

$$s_{CW,in} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,in}, P = P_{CW,in})$$

$$s_{CW,int} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,int}, P = P_{CW,in})$$

$$s_{CW,out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,out}, P = P_{CW,in})$$

$$\rho_{CW,in} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,in}, P = P_{CW,in})$$

$$\rho_{CW,int} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,int}, P = P_{CW,in})$$

$$\rho_{CW,out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,out}, P = P_{CW,in})$$

END **Condenser**

CEP

MODULE **ExtractionPump** ($P_{DA,BLEED}$, $\Delta P_{LP,HTR}$, η_{PUMP}, η_{LOSSES}, P_{in}, P_{out}, T_{in}, T_{out}, h_{in}, h_{out}, s_{in}, s_{out}, ρ_{in}, ρ_{out}, m_{tot}, m_2, m_3, W_{PUMP}, $W_{PUMP,CYCLE}$)

Outlet Conditions

$$P_{out} = P_{DA,BLEED} + \Delta P_{LP,HTR}$$

$$h_{isen,out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, s = s_{isen,out})$$

$$s_{isen,out} = s_{in}$$

$$T_{out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out}) \qquad s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, T = T_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\dot{m} = \dot{m}_{tot} - \dot{m}_2 - \dot{m}_3$$

Isentropic Efficiency

$$\eta_{PUMP} = \frac{h_{isen,out} - h_{in}}{h_{out} - h_{in}}$$

Pump Work

$$w_{PUMP} = (h_{out} - h_{in}) \cdot \dot{m}$$

Converting Pump Work to Shaft Power

$$w_{PUMP,CYCLE} = \frac{w_{PUMP}}{\eta_{LOSSES}}$$

END **ExtractionPump**

LP FWH

MODULE **LPFeedHeaters** (h_{in}, P_{DA}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out}, $P_{H,in}$, $h_{H,in}$, $P_{H,C}$, $T_{H,C}$, $h_{H,C}$, $s_{H,C}$, $\rho_{H,C}$, m_{tot}, m_2, m_3, m_4, P_{COND2}, $P_{H,out}$, $T_{H,out}$, $h_{H,out}$, $s_{H,out}$, $\rho_{H,out}$)

FW Outlet Conditions

$$P_{out} = P_{DA}$$

$$T_{out} = T_{H,C}$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Condensate Conditions

$$P_{H,C} = P_{H,in}$$

$$T_{H,C} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,C}, x = 0)$$

$$h_{H,C} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,C}, x = 0)$$

$$s_{H,C} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,C}, x = 0)$$

$$\rho_{H,C} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,C}, x = 0)$$

Energy Balance

$$(h_{out} - h_{in}) \cdot (\dot{m}_{tot} - \dot{m}_2 - \dot{m}_3) = (h_{H,in} - h_{H,C}) \cdot \dot{m}_4$$

Hot Side Outlet Conditions – After Expanding

$$P_{H,out} = P_{COND2}$$

$$T_{H,out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,out}, h = h_{H,out})$$

$h_{H,out} = h_{H,C}$

$s_{H,out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,out}, h = h_{H,out})$

$\rho_{H,out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,out}, h = h_{H,out})$

END **LPFeedHeaters**

HP FWH

MODULE **HPFeedHeaters** (P_{DRUM}, ΔP_{ECON}, m_1, m_2, T_{in}, P_{in}, h_{in}, s_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out}, $T_{H,in}$, $P_{H,in}$, $h_{H,in}$, $s_{H,in}$, $T_{H,C}$, $P_{H,C}$, $h_{H,C}$, $s_{H,C}$, $\rho_{H,C}$, P_{DA}, $T_{H,out}$, $P_{H,out}$, $h_{H,out}$, $s_{H,out}$, $\rho_{H,out}$)

FW Outlet Conditions

$P_{out} = P_{DRUM} + \Delta P_{ECON}$

$T_{out} = T_{H,C}$

$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$

$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$

$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$

Condensate conditions

$P_{H,C} = P_{H,in}$

$T_{H,C} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, x = 0, P = P_{H,C})$

$h_{H,C} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, x = 0, P = P_{H,C})$

$s_{H,C} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, x = 0, P = P_{H,C})$

$\rho_{H,C} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, x = 0, P = P_{H,C})$

Energy Balance

$\dot{m}_1 \cdot (h_{out} - h_{in}) = \dot{m}_2 \cdot (h_{H,in} - h_{H,C})$

Hot Side Outlet Conditions – After Expanding

$P_{H,out} = P_{DA}$

$T_{H,out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,out}, h = h_{H,out})$

$h_{H,out} = h_{H,C}$

$s_{H,out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,out}, h = h_{H,out})$

$\rho_{H,out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,out}, h = h_{H,out})$

END **HPFeedHeaters**

Dearator

MODULE **Dearator** ($P_{DA,BLEED}$, T_{out}, P_{DA}, h_{out}, s_{out}, ρ_{out}, h_{in}, $h_{H,in}$, $h_{C,FD,HTR,out}$, m_{tot}, m_2, m_3)

Outlet Conditions

$$P_{DA} = P_{DA,BLEED}$$

$$T_{out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DA}, x = 0)$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DA}, x = 0)$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DA}, x = 0)$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DA}, x = 0)$$

Energy Balance

$$(\dot{m}_{tot} - \dot{m}_2 - \dot{m}_3) \cdot (h_{out} - h_{in}) = \dot{m}_2 \cdot (h_{C,FD,HTR,out} - h_{out}) + \dot{m}_3 \cdot (h_{H,in} - h_{out})$$

END **Dearator**

BFP

MODULE **BoilerFeedPump** (η_{PUMP}, η_{MECH}, η_{TUR}, m_{tot}, m_5, P_{DRUM}, $\Delta P_{HP,HTR}$, ΔP_{ECON}, h_{in}, s_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out}, P_{COND2}, $h_{TUR,in}$, $s_{TUR,in}$, $P_{TUR,out}$, $T_{TUR,out}$, $h_{TUR,out}$, $s_{TUR,out}$, $\rho_{TUR,out}$)

Pump Outlet Conditions

$$P_{out} = P_{DRUM} + \Delta P_{HP,HTR} + \Delta P_{ECON}$$

$$h_{isen,out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, s = s_{isen,out})$$

$$s_{isen,out} = s_{in}$$

$$T_{out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Turbine Outlet Conditions

$$P_{TUR,out} = P_{COND2}$$

$$h_{TUR,isen,out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{TUR,out}, s = s_{TUR,isen,out})$$

$$s_{TUR,isen,out} = s_{TUR,in}$$

$$T_{TUR,out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{TUR,out}, h = h_{TUR,out})$$

$$s_{TUR,out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{TUR,out}, h = h_{TUR,out})$$

$$\rho_{TUR,out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{TUR,out}, h = h_{TUR,out})$$

Isentropic Efficiency

$$\eta_{PUMP} = \frac{h_{isen,out} - h_{in}}{h_{out} - h_{in}}$$

$$\eta_{TUR} = \frac{h_{TUR,out} - h_{TUR,in}}{h_{TUR,isen,out} - h_{TUR,in}}$$

Pump/Turbine Work

$$w_{PUMP} = (h_{out} - h_{in}) \cdot \dot{m}_{tot}$$

$$w_{TUR,out} = \dot{m}_5 \cdot (h_{TUR,in} - h_{TUR,out})$$

$$w_{PUMP} = w_{TUR,out} \cdot \eta_{MECH}$$

END **BoilerFeedPump**

Economiser

MODULE **Economiser** (P_{DRUM}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, h_{in}, q_{in}, m_1)

Outlet Conditions

$$P_{out} = P_{DRUM}$$

$$T_{out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, x = 0)$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, x = 0)$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, x = 0)$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, x = 0)$$

Energy Balance

$$q_{in} = \dot{m}_1 \cdot (h_{out} - h_{in})$$

END **Economiser**

Drum

MODULE **Drum** (P_{DRUM}, T_{out}, h_{out}, s_{out}, ρ_{out}, h_{in}, q_{in}, m_1)

Outlet Conditions

$$T_{out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DRUM}, x = 1)$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DRUM}, x = 1)$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DRUM}, x = 1) \qquad \rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DRUM}, x = 1)$$

Energy Balance

$$q_{in} = \dot{m}_1 \cdot (h_{out} - h_{in})$$

END **Drum**

Super Heaters

MODULE **HPSuperHeaters** (ΔP, T_{out}, P_{in}, P_{out}, h_{out}, s_{out}, ρ_{out}, h_{in}, q_{in}, m_1)

Outlet Conditions

$$P_{out} = P_{in} - \Delta P$$

$$h_{out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out} \right)$$

$$s_{out} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out} \right)$$

$$\rho_{out} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out} \right)$$

Energy Balance

$$q_{in} = \dot{m}_1 \cdot \left(h_{out} - h_{in} \right)$$

END **HPSuperHeaters**

Super Heater Attemperation

MODULE **HPAttemperation** (T_H, P_{in}, T_{in}, h_{in}, s_{in}, P_{att}, T_{att}, h_{att}, s_{att}, m_{tot}, m_1, m_8, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out})

Mass Balance

$$\dot{m}_1 = \dot{m}_{out} - \dot{m}_8$$

Super Heater Outlet Conditions If Attemperation Would Have Been Applied At Inlets

$$P_{out} = P_{in}$$

$$T_{out} = T_H$$

$$h_{out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out} \right)$$

$$s_{out} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out} \right) \qquad \rho_{out} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out} \right)$$

Energy Balance

$$\dot{m}_{out} \cdot h_{out} = \dot{m}_1 \cdot h_{in} + \dot{m}_8 \cdot h_{att}$$

END **HPAttemperation**

HP turbine Inlet Valve and Piping

MODULE **HPTurbineInletValve** (m_1, m_8, ΔP, P_{in}, T_{in}, h_{in}, s_{in}, ρ_{in}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, q_{loss}, $P_{ATT,out}$, $h_{ATT,out}$)

Turbine Inlet Valve Outlet Conditions

$$P_{out} = P_{in} - \Delta P$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Turbine Inlet Valve Inlet Conditions

$$P_{in} = P_{ATT,out}$$

$$T_{in} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{in}, h = h_{in})$$

$$h_{in} = h_{out}$$

$$s_{in} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{in}, P = P_{in})$$

$$\rho_{in} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{in}, P = P_{in})$$

Heat Loss In The Piping Between The Super Heaters And The Turbine Inlet Valve

$$\dot{q}_{loss} = (\dot{m}_1 + \dot{m}_8) \cdot (h_{ATT,out} - h_{in})$$

END **HPTurbineInletValve**

HP turbine

MODULE **HPTurbine** (m_1, m_8, η_{TUR}, h_{in}, s_{in}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, W_{out})

Outlet Conditions

$$h_{isen,out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, s = s_{isen,out})$$

$$s_{isen,out} = s_{in}$$

$$T_{out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Isentropic Efficiency

$$\eta_{isen} = \frac{h_{out} - h_{in}}{h_{isen,out} - h_{in}}$$

Turbine Work Output

$$w_{out} = (\dot{m}_1 + \dot{m}_8) \cdot (h_{in} - h_{out})$$

END **HPTurbine**

Reheaters

MODULE **Reheaters** (ΔP, T_{out}, P_{in}, P_{out}, h_{out}, s_{out}, ρ_{out}, h_{in}, q_{in}, m_1, m_8)

Outlet Conditions

$$P_{out} = P_{in} - \Delta P$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Heat Loss In The Piping Between The Reheaters And The Turbine Inlet Valve

$$q_{in} = (\dot{m}_1 + \dot{m}_8) \cdot (h_{out} - h_{in})$$

END **Reheaters**

Reheater Attemperation

MODULE **RHAttemperation** (T_H, P_{in}, T_{in}, h_{in}, s_{in}, P_{att}, T_{att}, h_{att}, s_{att}, m_{tot}, m_1, m_8, m_9, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out})

Mass Balance

$$\dot{m}_1 + \dot{m}_8 = \dot{m}_{tot} - \dot{m}_9$$

Super Heater Outlet Conditions If Attemperation Would Have Been Applied At Inlets

$$P_{out} = P_{in}$$

$$T_{out} = T_H$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Energy Balance

$$\dot{m}_{tot} \cdot h_{out} = (\dot{m}_1 + \dot{m}_8) \cdot h_{in} + \dot{m}_9 \cdot h_{att}$$

END **RHAttemperation**

IP Turbine Inlet Valve and Piping

MODULE **IPTurbineInletValve** (m, ΔP, P_{in}, T_{in}, h_{in}, s_{in}, ρ_{in}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, q_{loss}, $P_{SH,out}$, $h_{SH,out}$)

Turbine Inlet Valve Outlet Conditions

$$P_{out} = P_{in} - \Delta P$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Turbine Inlet Valve Inlet Conditions

$P_{in} = P_{SH,out}$

$T_{in} = \mathbf{T}(\text{'Steam}_{IAPWS}\text{'}, P = P_{in}, h = h_{in})$

$h_{in} = h_{out}$

$s_{in} = \mathbf{s}(\text{'Steam}_{IAPWS}\text{'}, T = T_{in}, P = P_{in})$

$\rho_{in} = \rho(\text{'Steam}_{IAPWS}\text{'}, T = T_{in}, P = P_{in})$

Energy Balance

$\dot{q}_{loss} = \dot{m} \cdot (h_{SH,out} - h_{in})$

END **IPTurbineInletValve**

IP Turbine

MODULE **IPTurbine** (m_{in}, m_{bs}, η_{isen}, k_{poly}, P_{FWH}, P_{outlet}, P_{in}, T_{in}, h_{in}, s_{in}, ρ_{in}, P_{bs}, T_{bs}, h_{bs}, s_{bs}, ρ_{bs}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, w_{out})

Mass Balance

$\dot{m}_{in} = \dot{m}_{out} + \dot{m}_{bs}$

Turbine Outlet Conditions – Isentropic Efficiency

$P_{out} = P_{outlet}$

$h_{isen,out} = \mathbf{h}(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, s = s_{in})$

$T_{out} = \mathbf{T}(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out})$

$s_{out} = \mathbf{s}(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$

$\rho_{out} = \rho(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$

Isentropic Efficiency

$$\eta_{isen} = \frac{h_{out} - h_{in}}{h_{isen,out} - h_{in}}$$

Converting temperature to Kelvin for polytropic efficiency calculations

$T_{in,K} = T_{in} + 273.15$ [K]

$T_{bs,K} = T_{bs} + 273.15$ [K]

$T_{out,K} = T_{out} + 273.15$ [K]

Bled Steam Conditions – Polytropic Efficiency

$P_{bs} = P_{FWH}$

$h_{bs} = \mathbf{h}(\text{'Steam}_{IAPWS}\text{'}, T = T_{bs}, P = P_{bs})$

$$s_{bs} = \mathbf{s}\,(\,'Steam_{IAPWS}'\,,T=T_{bs}\,,P=P_{bs}\,)$$

$$\rho_{bs} = \rho\,(\,'Steam_{IAPWS}'\,,T=T_{bs}\,,P=P_{bs}\,)$$

Polytropic Efficiency

$$\frac{T_{in,K}}{T_{out,K}} = \left[\frac{P_{in}}{P_{out}}\right]^{k_{poly}}$$

$$\frac{T_{in,K}}{T_{bs,K}} = \left[\frac{P_{in}}{P_{bs}}\right]^{k_{poly}}$$

Turbine Work Output

$$w_{out} = \dot{m}_{in} \cdot (h_{in} - h_{bs}) + \dot{m}_{out} \cdot (h_{bs} - h_{out})$$

END **IPTurbine**

LP turbine

MODULE **LPTurbines** (η_{TUR}, k_{poly}, P_{in}, T_{in}, h_{in}, s_{in}, P_{BLEED}, T_{BLEED}, h_{BLEED}, s_{BLEED}, ρ_{BLEED}, $P_{SAT,COND1}$, $T_{TUR1,S2,out}$, $h_{TUR1,S2,out}$, $s_{TUR1,S2,out}$, $\rho_{TUR1,S2,out}$, $P_{SAT,COND2}$, $T_{TUR2,S2,out}$, $h_{TUR2,S2,out}$, $s_{TUR2,S2,out}$, $\rho_{TUR2,S2,out}$, m_{tot}, m_2, m_3, m_4, m_5, m_6, m_7, W_{out})

Mass Balance

$$\dot{m}_{TUR1,S1} + \dot{m}_{TUR2,S1} = \dot{m}_{tot} - \dot{m}_2 - \dot{m}_3 - \dot{m}_5$$

$$\dot{m}_{TUR1,S1} + \dot{m}_{TUR2,S1} = \dot{m}_{TUR1,S2} + \dot{m}_{TUR2,S2} + \dot{m}_4$$

$$\dot{m}_{TUR1,S1} = \dot{m}_{TUR2,S1}$$

$$\frac{P_{BLEED} - P_{SAT,COND1}}{P_{BLEED} - P_{SAT,COND2}} = \frac{\dot{m}_{TUR1,S2}^{\;2}}{\dot{m}_{TUR2,S2}^{\;2}}$$

$$\dot{m}_{TUR1,S2} = \dot{m}_6$$

$$\dot{m}_{TUR2,S2} = \dot{m}_7$$

Turbine 1 Outlet Conditions – Isentropic Efficiency

$$P_{TUR1,S2,out} = P_{SAT,COND1}$$

$$h_{isen,TUR1,S2,out} = \mathbf{h}\,(\,'Steam_{IAPWS}'\,,P=P_{TUR1,S2,out}\,,s=s_{isen,TUR1,S2,out}\,)$$

$$s_{isen,TUR1,S2,out} = s_{in}$$

$$T_{TUR1,S2,out} = \mathbf{T}\,(\,'Steam_{IAPWS}'\,,P=P_{TUR1,S2,out}\,,h=h_{TUR1,S2,out}\,)$$

$$s_{TUR1,S2,out} = \mathbf{s}\,(\,'Steam_{IAPWS}'\,,h=h_{TUR1,S2,out}\,,P=P_{TUR1,S2,out}\,)$$

$$\rho_{TUR1,S2,out} = \rho\,(\,'Steam_{IAPWS}'\,,h=h_{TUR1,S2,out}\,,P=P_{TUR1,S2,out}\,)$$

Turbine 2 Outlet Conditions – Isentropic Efficiency

$P_{TUR2,S2,out} = P_{SAT,COND2}$

$h_{isen,TUR2,S2,out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{TUR2,S2,out}, s = s_{isen,TUR2,S2,out} \right)$

$s_{isen,TUR2,S2,out} = s_{in}$

$T_{TUR2,S2,out} = \mathbf{T}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{TUR2,S2,out}, h = h_{TUR2,S2,out} \right)$

$s_{TUR2,S2,out} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, h = h_{TUR2,S2,out}, P = P_{TUR2,S2,out} \right)$

$\rho_{TUR2,S2,out} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, h = h_{TUR2,S2,out}, P = P_{TUR2,S2,out} \right)$

Isentropic Efficiency

$$\eta_{TUR} = \frac{h_{TUR1,S2,out} - h_{in}}{h_{isen,TUR1,S2,out} - h_{in}}$$

$$\eta_{TUR} = \frac{h_{TUR2,S2,out} - h_{in}}{h_{isen,TUR2,S2,out} - h_{in}}$$

Converting temperature to Kelvin for polytropic efficiency calculations

$T_{in,K} = T_{in} + 273.15$ [K]

$T_{BLEED,K} = T_{BLEED} + 273.15$ [K]

Bled Steam Conditions – Polytropic Efficiency

$h_{BLEED} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{BLEED}, T = T_{BLEED} \right)$

$s_{BLEED} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{BLEED}, P = P_{BLEED} \right)$

$\rho_{BLEED} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{BLEED}, P = P_{BLEED} \right)$

Polytropic Efficiency

$$\frac{T_{in,K}}{T_{BLEED,K}} = \left[\frac{P_{in}}{P_{BLEED}} \right]^{k_{poly}}$$

Turbine Work Output

$$w_{out} = \left(\dot{m}_6 + \dot{m}_7 + \dot{m}_4 \right) \cdot \left(h_{in} - h_{BLEED} \right) + \dot{m}_6 \cdot \left(h_{BLEED} - h_{TUR1,S2,out} \right) + \dot{m}_7 \cdot \left(h_{BLEED} - h_{TUR2,S2,out} \right)$$

END **LPTurbines**

--*Main Program*--

Input Parameters

Carnot Parameters

$T_H = 540$ [C]

T_L = 20 [C]

RCC Parameters

T_R = 20 [C]

TTD_{COND} = 3 [C]

$T_{COND,SUB}$ = 3 [C]

CEP Parameters

η_{CEP} = 0.85 [-]

$\eta_{CEP,LOSSES}$ = 0.702 [-]

LP FWH Parameters

$\Delta P_{LP,HTR}$ = 0.2 [MPa]

$P_{LP,FD,HTR,BLEED}$ = 0.15 [MPa]

DA Parameters

$P_{DA,BLEED}$ = 0.6 [MPa]

BFP Parameters

$\eta_{BF,PUMP}$ = 0.85 [-]

$\eta_{BF,PUMP,TUR}$ = 0.88 [-]

$\eta_{BF,PUMP,MECH}$ = 0.95 [-]

HP FWH Parameters

$\Delta P_{HP,HTR}$ = 0.2 [MPa]

$P_{HP,FD,HTR,BLEED}$ = 2 [MPa]

Economiser Parameters

ΔP_{ECON} = 3.7 [MPa]

SH Parameters

$\Delta P_{HP,SH}$ = 1.2 [MPa]

$T_{HP,SH,out}$ = 545 [C]

HP turbine Inlet Valve Parameters

$\Delta P_{HPT,IV}$ = 0.1 [MPa]

CEP Parameters

$T_{HP,TUR,in}$ = 538 [C]

$P_{HP,TUR,in}$ = 16.8 [MPa]

$\eta_{HP,TUR}$ = 0.9 [-]

Reheaters Parameters

ΔP_{RH} = 1.2 [MPa]

$T_{RH,out}$ = 545 [C]

IP Turbine Inlet Valve Parameters

$\Delta P_{IPT,IV}$ = 0.1 [MPa]

IP Turbine Parameters

$\dot{m}_{tot}$ = 1 [kg/s]

$T_{IP,TUR,in}$ = 538 [C]

$P_{IP,TUR,in}$ = 3.4 [MPa]

$\eta_{IP,TUR}$ = 0.88 [-]

IP Turbine Parameters

$\eta_{LP,TUR}$ = 0.88 [-]

RCC

$T_{CW,in}$ = T_L

$P_{CW,in}$ = 0.1 [MPa]

Call **Condenser**[T_R, TTD_{COND}, $T_{COND,SUB}$, m_6, m_7, m_5, m_4, $P_{SAT,COND1}$, $P_{SAT,COND2}$, $P_{COND,out}$, $T_{COND,out}$, $h_{COND,out}$, $s_{COND,out}$, $\rho_{COND,out}$, $h_{TUR1,S2,out}$, $h_{TUR2,S2,out}$, $h_{LP,FD,HTR,COND2,in}$, q_{COND}, $T_{CW,in}$, $P_{CW,in}$, $h_{CW,in}$, $s_{CW,in}$, $\rho_{CW,in}$, $T_{CW,int}$, $h_{CW,int}$, $s_{CW,int}$, $\rho_{CW,int}$, $T_{CW,out}$, $h_{CW,out}$, $s_{CW,out}$, $\rho_{CW,out}$]

Extraction Pump

Call **ExtractionPump**[$P_{DA,BLEED}$, $\Delta P_{LP,HTR}$, η_{CEP}, $\eta_{CEP,LOSSES}$, $P_{COND,out}$, $P_{CEP,out}$, $T_{COND,out}$, $T_{CEP,out}$, $h_{COND,out}$, $h_{CEP,out}$, $s_{COND,out}$, $s_{CEP,out}$, $\rho_{COND,out}$, $\rho_{CEP,out}$, m_{tot}, m_2, m_3, w_{CEP}, $w_{CEP,CYCLE}$]

LP Feed Heaters

Call **LPFeedHeaters**[$h_{CEP,out}$, $P_{DA,BLEED}$, $T_{LP,HTR,out}$, $P_{LP,HTR,out}$, $h_{LP,HTR,out}$, $s_{LP,HTR,out}$, $\rho_{LP,HTR,out}$, $P_{LP,FD,HTR,BLEED}$, $h_{LP,FD,HTR,BLEED}$, $P_{LP,FD,HTR,C}$, $T_{LP,FD,HTR,C}$, $h_{LP,FD,HTR,C}$, $s_{LP,FD,HTR,C}$, $\rho_{LP,FD,HTR,C}$, m_{tot}, m_2, m_3, m_4, $P_{SAT,COND2}$, $P_{LP,FD,HTR,COND2,in}$, $T_{LP,FD,HTR,COND2,in}$, $h_{LP,FD,HTR,COND2,in}$, $s_{LP,FD,HTR,COND2,in}$, $\rho_{LP,FD,HTR,COND2,in}$]

Dearator

Call **Dearator**[$P_{DA,BLEED}$, $T_{DA,out}$, P_{DA}, $h_{DA,out}$, $s_{DA,out}$, $\rho_{DA,out}$, $h_{LP,HTR,out}$, $h_{IP,TUR,out}$, $h_{HP,FD,HTR,DA,in}$, m_{tot}, m_2, m_3]

Boiler Feed Pump

Call **BoilerFeedPump**[$\eta_{BF,PUMP}$, $\eta_{BF,PUMP,MECH}$, $\eta_{BF,PUMP,TUR}$, m_{tot}, m_5, P_{DRUM}, $\Delta P_{HP,HTR}$, ΔP_{ECON}, $h_{DA,out}$, $s_{DA,out}$, $T_{HP,PUMP,out}$, $P_{HP,PUMP,out}$, $h_{HP,PUMP,out}$, $s_{HP,PUMP,out}$, $\rho_{HP,PUMP,out}$, $P_{SAT,COND2}$, $h_{IP,TUR,out}$, $s_{IP,TUR,out}$, $P_{BFPTD,out}$, $T_{BFPTD,out}$, $h_{BFPTD,out}$, $s_{BFPTD,out}$, $\rho_{BFPTD,out}$]

HP Feed Heaters

Call **HPFeedHeaters**[P_{DRUM}, ΔP_{ECON}, m_1, m_2, $T_{HP,PUMP,out}$, $P_{HP,PUMP,out}$, $h_{HP,PUMP,out}$, $s_{HP,PUMP,out}$, $T_{HP,FD,HTR,out}$, $P_{HP,FD,HTR,out}$, $h_{HP,FD,HTR,out}$, $s_{HP,FD,HTR,out}$, $\rho_{HP,FD,HTR,out}$, $T_{HP,FD,HTR,H,in}$, $P_{HP,FD,HTR,H,in}$, $h_{HP,FD,HTR,H,in}$, $s_{HP,FD,HTR,C}$, $T_{HP,FD,HTR,C}$,

$P_{HP,FD,HTR,C}$, $h_{HP,FD,HTR,C}$, $s_{HP,FD,HTR,C}$, $\rho_{HP,FD,HTR,C}$, $P_{DA,BLEED}$, $T_{HP,FD,HTR,DA,in}$, $P_{HP,FD,HTR,DA,in}$, $h_{HP,FD,HTR,DA,in}$, $s_{HP,FD,HTR,DA,in}$, $\rho_{HP,FD,HTR,DA,in}$]

Economiser

Call **Economiser**[P_{DRUM}, $P_{ECON,out}$, $T_{ECON,out}$, $h_{ECON,out}$, $s_{ECON,out}$, $\rho_{ECON,out}$, $h_{HP,FD,HTR,out}$, q_{ECON}, m_1]

Drum

Call **Drum**[P_{DRUM}, $T_{DRUM,out}$, $h_{DRUM,out}$, $s_{DRUM,out}$, $\rho_{DRUM,out}$, $h_{ECON,out}$, q_{DRUM}, m_1]

HP Super Heaters

Call **HPSuperHeaters**[$\Delta P_{HP,SH}$, $T_{HP,SH,out}$, P_{DRUM}, $P_{HP,SH,out}$, $h_{HP,SH,out}$, $s_{HP,SH,out}$, $\rho_{HP,SH,out}$, $h_{DRUM,out}$, q_{SH}, m_1]

HP Super Heaters Attemperation

Call **HPAttemperation**[T_H, $P_{HP,SH,out}$, $T_{HP,SH,out}$, $h_{HP,SH,out}$, $s_{HP,SH,out}$, $P_{HP,PUMP,out}$, $T_{HP,PUMP,out}$, $h_{HP,PUMP,out}$, $s_{HP,PUMP,out}$, m_{tot}, m_1, m_8, $P_{HPT,IP,in}$, $T_{HPT,IP,in}$, $h_{HPT,IP,in}$, $s_{HPT,IP,in}$, $\rho_{HPT,IP,in}$]

HP Turbine Inlet Valve with Upstream Piping

Call **HPTurbineInletValve**[m_1, m_8, $\Delta P_{HPT,IV}$, $P_{HPT,IV,in}$, $T_{HPT,IV,in}$, $h_{HPT,IV,in}$, $s_{HPT,IV,in}$, $\rho_{HPT,IV,in}$, $P_{HP,TUR,in}$, $T_{HP,TUR,in}$, $h_{HP,TUR,in}$, $s_{HP,TUR,in}$, $\rho_{HP,TUR,in}$, $q_{HPT,IP,loss}$, $P_{HPT,IP,in}$, $h_{HPT,IP,in}$]

HP Turbine

Call **HPTurbine**[m_1, m_8, $\eta_{HP,TUR}$, $h_{HP,TUR,in}$, $s_{HP,TUR,in}$, $P_{HP,TUR,out}$, $T_{HP,TUR,out}$, $h_{HP,TUR,out}$, $s_{HP,TUR,out}$, $\rho_{HP,TUR,out}$, W_{HPT}]

Reheaters

Call **Reheaters**[ΔP_{RH}, $T_{RH,out}$, $P_{HP,TUR,out}$, $P_{RH,out}$, $h_{RH,out}$, $s_{RH,out}$, $\rho_{RH,out}$, $h_{HP,TUR,out}$, q_{RH}, m_1, m_8]

Reheaters Attemperation

Call **RHAttemperation**[T_H, $P_{RH,out}$, $T_{RH,out}$, $h_{RH,out}$, $s_{RH,out}$, $P_{HP,PUMP,out}$, $T_{HP,PUMP,out}$, $h_{HP,PUMP,out}$, $s_{HP,PUMP,out}$, m_{tot}, m_1, m_8, m_9, $P_{IPT,IP,in}$, $T_{IPT,IP,in}$, $h_{IPT,IP,in}$, $s_{IPT,IP,in}$, $\rho_{IPT,IP,in}$]

IP Turbine Inlet Valve with Upstream Piping

Call **IPTurbineInletValve**[m_{tot}, $\Delta P_{IPT,IV}$, $P_{IPT,IV,in}$, $T_{IPT,IV,in}$, $h_{IPT,IV,in}$, $s_{IPT,IV,in}$, $\rho_{IPT,IV,in}$, $P_{IP,TUR,in}$, $T_{IP,TUR,in}$, $h_{IP,TUR,in}$, $s_{IP,TUR,in}$, $\rho_{IP,TUR,in}$, $q_{IPT,IP,loss}$, $P_{IPT,IP,in}$, $h_{IPT,IP,in}$]

IP Turbine with HP Feedwater Bleed

Call **IPTurbine**[m_{tot}, m_2, $\eta_{IP,TUR}$, k_{poly}, $P_{HP,FD,HTR,BLEED}$, $P_{DA,BLEED}$, $P_{IP,TUR,in}$, $T_{IP,TUR,in}$, $h_{IP,TUR,in}$, $s_{IP,TUR,in}$, $\rho_{IP,TUR,in}$, $P_{HP,FD,HTR,H,in}$, $T_{HP,FD,HTR,H,in}$, $h_{HP,FD,HTR,H,in}$, $s_{HP,FD,HTR,H,in}$, $\rho_{HP,FD,HTR,H,in}$, $P_{IP,TUR,out}$, $T_{IP,TUR,out}$, $h_{IP,TUR,out}$, $s_{IP,TUR,out}$, $\rho_{IP,TUR,out}$, W_{IPT}]

LP Turbines

Call **LPTurbines**[$\eta_{LP,TUR}$, k_{poly}, $P_{IP,TUR,out}$, $T_{IP,TUR,out}$, $h_{IP,TUR,out}$, $s_{IP,TUR,out}$, $P_{LP,FD,HTR,BLEED}$, $T_{LP,FD,HTR,BLEED}$, $h_{LP,FD,HTR,BLEED}$, $s_{LP,FD,HTR,BLEED}$, $\rho_{LP,FD,HTR,BLEED}$, $P_{SAT,COND1}$, $T_{TUR1,S2,out}$, $h_{TUR1,S2,out}$, $s_{TUR1,S2,out}$, $\rho_{TUR1,S2,out}$, $P_{SAT,COND2}$, $T_{TUR2,S2,out}$, $h_{TUR2,S2,out}$, $s_{TUR2,S2,out}$, $\rho_{TUR2,S2,out}$, m_{tot}, m_2, m_3, m_4, m_5, m_6, m_7, W_{LPT}]

Efficiency Calculations

$$q_{loss} = \dot{q}_{HPT,IP,loss} + \dot{q}_{IPT,IP,loss} + q_{COND}$$

$$q_{in} = q_{ECON} + q_{DRUM} + q_{SH} + q_{RH}$$

$w_{in} = w_{CEP,CYCLE}$

$w_{out} = w_{HPT} + w_{IPT} + w_{LPT}$

$w_{net} = w_{out} - w_{in}$

$$\eta_{tot} = \frac{w_{net}}{q_{in}}$$

--Graph Nodes--

Node 1 - CEP Inlet

$P_1 = P_{COND,out}$

$h_1 = h_{COND,out}$

$s_1 = s_{COND,out}$

$\rho_1 = \rho_{COND,out}$

Node 2 - CEP Outlet

$T_2 = T_{CEP,out}$

$P_2 = P_{CEP,out}$

$h_2 = h_{CEP,out}$

$s_2 = s_{CEP,out}$

$\rho_2 = \rho_{CEP,out}$

Node 3 - LP FWH - FW Outlet

$T_3 = T_{LP,HTR,out}$

$P_3 = P_{LP,HTR,out}$

$h_3 = h_{LP,HTR,out}$

$s_3 = s_{LP,HTR,out}$

$\rho_3 = \rho_{LP,HTR,out}$

Node 4 - Dearator Outlet

$T_4 = T_{DA,out}$

$P_4 = P_{DA}$

$h_4 = h_{DA,out}$

$s_4 = s_{DA,out}$

$\rho_4 = \rho_{DA,out}$

Node 5 - BFP Outlet

$T_5 = T_{HP,PUMP,out}$

$P_5 = P_{HP,PUMP,out}$

$h_5 = h_{HP,PUMP,out}$

$s_5 = s_{HP,PUMP,out}$

$\rho_5 = \rho_{HP,PUMP,out}$

Node 6 - HP FWH - FW Outlet

$T_6 = T_{HP,FD,HTR,out}$

$P_6 = P_{HP,FD,HTR,out}$

$h_6 = h_{HP,FD,HTR,out}$

$s_6 = s_{HP,FD,HTR,out}$

$\rho_6 = \rho_{HP,FD,HTR,out}$

Node 7 - Economizer Outlet

$T_7 = T_{ECON,out}$

$P_7 = P_{ECON,out}$

$h_7 = h_{ECON,out}$

$s_7 = s_{ECON,out}$

$\rho_7 = \rho_{ECON,out}$

Node 8 - Drum Outlet

$T_8 = T_{DRUM,out}$

$P_8 = P_{DRUM}$

$h_8 = h_{DRUM,out}$

$s_8 = s_{DRUM,out}$

$\rho_8 = \rho_{DRUM,out}$

Node 9 - HP Super Heaters Outlet (Before accounting for attemperation)

$T_9 = T_{HP,SH,out}$

$P_9 = P_{HP,SH,out}$

$h_9 = h_{HP,SH,out}$

$s_9 = s_{HP,SH,out}$

$\rho_9 = \rho_{HP,SH,out}$

Node 10 - HP Super Heaters Outlet (After accounting for attemperation)

$T_{10} = T_{HPT,IP,in}$

$P_{10} = P_{HPT,IP,in}$

$h_{10} = h_{HPT,IP,in}$

$s_{10} = s_{HPT,IP,in}$

$\rho_{10} = \rho_{HPT,IP,in}$

Node 11 – HP turbine Inlet Valve Inlet

$T_{11} = T_{HPT,IV,in}$

$P_{11} = P_{HPT,IV,in}$

$h_{11} = h_{HPT,IV,in}$

$s_{11} = s_{HPT,IV,in}$

$\rho_{11} = \rho_{HPT,IV,in}$

Node 12 - HP turbine Inlet

$T_{12} = T_{HP,TUR,in}$

$P_{12} = P_{HP,TUR,in}$

$h_{12} = h_{HP,TUR,in}$

$s_{12} = s_{HP,TUR,in}$

$\rho_{12} = \rho_{HP,TUR,in}$

Node 13 - Reheaters Inlet

$T_{13} = T_{HP,TUR,out}$

$P_{13} = P_{HP,TUR,out}$

$h_{13} = h_{HP,TUR,out}$

$s_{13} = s_{HP,TUR,out}$

$\rho_{13} = \rho_{HP,TUR,out}$

Node 14 - Reheaters Outlet (Before accounting for attemperation)

$T_{14} = T_{RH,out}$

$P_{14} = P_{RH,out}$

$h_{14} = h_{RH,out}$

$s_{14} = s_{RH,out}$

$\rho_{14} = \rho_{RH,out}$

Node 15 - Reheaters Outlet (After accounting for attemperation)

$T_{15} = T_{IPT,IP,in}$

P_{15} = $P_{IPT,IP,in}$

h_{15} = $h_{IPT,IP,in}$

s_{15} = $s_{IPT,IP,in}$

ρ_{15} = $\rho_{IPT,IP,in}$

Node 16 – IP Turbine Inlet Valve Inlet

T_{16} = $T_{IPT,IV,in}$

P_{16} = $P_{IPT,IV,in}$

h_{16} = $h_{IPT,IV,in}$

s_{16} = $s_{IPT,IV,in}$

ρ_{16} = $\rho_{IPT,IV,in}$

Node 17 – IP Turbine Inlet

T_{17} = $T_{IP,TUR,in}$

P_{17} = $P_{IP,TUR,in}$

h_{17} = $h_{IP,TUR,in}$

s_{17} = $s_{IP,TUR,in}$

ρ_{17} = $\rho_{IP,TUR,in}$

Node 18 – IP Turbine Bleed Point

T_{18} = $T_{HP,FD,HTR,H,in}$

P_{18} = $P_{HP,FD,HTR,H,in}$

h_{18} = $h_{HP,FD,HTR,H,in}$

s_{18} = $s_{HP,FD,HTR,H,in}$

ρ_{18} = $\rho_{HP,FD,HTR,H,in}$

Node 19 – IP Turbine Outlet

T_{19} = $T_{IP,TUR,out}$

P_{19} = $P_{IP,TUR,out}$

h_{19} = $h_{IP,TUR,out}$

s_{19} = $s_{IP,TUR,out}$

ρ_{19} = $\rho_{IP,TUR,out}$

Node 20 - LP Turbine Bleed Points

T_{20} = $T_{LP,FD,HTR,BLEED}$

P_{20} = $P_{LP,FD,HTR,BLEED}$

$h_{20} = h_{LP,FD,HTR,BLEED}$

$s_{20} = s_{LP,FD,HTR,BLEED}$

$\rho_{20} = \rho_{LP,FD,HTR,BLEED}$

Node 21 - LP Turbine 1 Outlet

$T_{21} = T_{TUR1,S2,out}$

$P_{21} = P_{SAT,COND1}$

$h_{21} = h_{TUR1,S2,out}$

$s_{21} = s_{TUR1,S2,out}$

$\rho_{21} = \rho_{TUR1,S2,out}$

Node 22 - LP Turbine 2 Outlet

$T_{22} = T_{TUR2,S2,out}$

$P_{22} = P_{SAT,COND2}$

$h_{22} = h_{TUR2,S2,out}$

$s_{22} = s_{TUR2,S2,out}$

$\rho_{22} = \rho_{TUR2,S2,out}$

Node 23 - LP Feed Heater Condensate

$T_{23} = T_{LP,FD,HTR,C}$

$P_{23} = P_{LP,FD,HTR,C}$

$h_{23} = h_{LP,FD,HTR,C}$

$s_{23} = s_{LP,FD,HTR,C}$

$\rho_{23} = \rho_{LP,FD,HTR,C}$

Node 24 - LP Feed Heater Bleed Steam Condenser Inlet

$T_{24} = T_{LP,FD,HTR,COND2,in}$

$P_{24} = P_{LP,FD,HTR,COND2,in}$

$h_{24} = h_{LP,FD,HTR,COND2,in}$

$s_{24} = s_{LP,FD,HTR,COND2,in}$

$\rho_{24} = \rho_{LP,FD,HTR,COND2,in}$

Node 25 - HP Feed Heater Condensate

$T_{25} = T_{HP,FD,HTR,C}$

$P_{25} = P_{HP,FD,HTR,C}$

$h_{25} = h_{HP,FD,HTR,C}$

$s_{25} = s_{HP,FD,HTR,C}$

$\rho_{25} = \rho_{HP,FD,HTR,C}$

Node 26 - LP Feed Heater Bleed Steam Dearator Inlet

$T_{26} = T_{HP,FD,HTR,DA,in}$

$P_{26} = P_{HP,FD,HTR,DA,in}$

$h_{26} = h_{HP,FD,HTR,DA,in}$

$s_{26} = s_{HP,FD,HTR,DA,in}$

$\rho_{26} = \rho_{HP,FD,HTR,DA,in}$

Node 27 - BFPTD outlet

$T_{27} = T_{BFPTD,out}$

$P_{27} = P_{BFPTD,out}$

$h_{27} = h_{BFPTD,out}$

$s_{27} = s_{BFPTD,out}$

$\rho_{27} = \rho_{BFPTD,out}$

Node 28 - RCC CW Inlet

$T_{28} = T_{CW,in}$

$P_{28} = P_{CW,in}$

$h_{28} = h_{CW,in}$

$s_{28} = s_{CW,in}$

$\rho_{28} = \rho_{CW,in}$

Node 29 - RCC CW Intermediate

$T_{29} = T_{CW,int}$

$P_{29} = P_{CW,in}$

$h_{29} = h_{CW,int}$

$s_{29} = s_{CW,int}$

$\rho_{29} = \rho_{CW,int}$

Node 30 - CW Outlet

$T_{30} = T_{CW,out}$

$P_{30} = P_{CW,in}$

$h_{30} = h_{CW,out}$

$s_{30} = s_{CW,out}$

$\rho_{30} = \rho_{CW,out}$

8.1.2 Results

SOLUTION

Unit Settings: [kJ]/[C]/[MPa]/[kg]/[degrees]

$\Delta P_{ECON} = 3.7$ [MPa]
$\Delta P_{HPT,IV} = 0.1$ [MPa]
$\Delta P_{HP,HTR} = 0.2$ [MPa]
$\Delta P_{HP,SH} = 1.2$ [MPa]
$\Delta P_{IPT,IV} = 0.1$ [MPa]
$\Delta P_{LP,HTR} = 0.2$ [MPa]
$\Delta P_{RH} = 1.2$ [MPa]
$\eta_{BF,PUMP} = 0.85$ [-]
$\eta_{BF,PUMP,MECH} = 0.95$ [-]
$\eta_{BF,PUMP,TUR} = 0.88$ [-]
$\eta_{CEP} = 0.85$ [-]
$\eta_{CEP,LOSSES} = 0.702$ [-]
$\eta_{HP,TUR} = 0.9$ [-]
$\eta_{IP,TUR} = 0.88$ [-]
$\eta_{LP,TUR} = 0.88$ [-]
$\eta_{tot} = 0.4305$ [-]
$h_{BFPTD,out} = 2414$ [kJ/kg]
$h_{CEP,out} = 150$ [kJ/kg]
$h_{COND,out} = 149.1$ [kJ/kg]
$h_{CW,in} = 84.01$ [kJ/kg]
$h_{CW,int} = 122.6$ [kJ/kg]
$h_{CW,out} = 167.6$ [kJ/kg]
$h_{DA,out} = 670.4$ [kJ/kg]
$h_{DRUM,out} = 2506$ [kJ/kg]
$h_{ECON,out} = 1737$ [kJ/kg]
$h_{HPT,IP,in} = 3402$ [kJ/kg]
$h_{HPT,IV,in} = 3397$ [kJ/kg]
$h_{HP,FD,HTR,C} = 908.5$ [kJ/kg]
$h_{HP,FD,HTR,DA,in} = 908.5$ [kJ/kg]
$h_{HP,FD,HTR,H,in} = 3377$ [kJ/kg]
$h_{HP,FD,HTR,out} = 915.8$ [kJ/kg]
$h_{HP,PUMP,out} = 697.9$ [kJ/kg]
$h_{HP,SH,out} = 3416$ [kJ/kg]
$h_{HP,TUR,in} = 3397$ [kJ/kg]
$h_{HP,TUR,out} = 3066$ [kJ/kg]
$h_{IPT,IP,in} = 3542$ [kJ/kg]
$h_{IPT,IV,in} = 3539$ [kJ/kg]
$h_{IP,TUR,in} = 3539$ [kJ/kg]
$h_{IP,TUR,out} = 3074$ [kJ/kg]
$h_{LP,FD,HTR,BLEED} = 2811$ [kJ/kg]
$h_{LP,FD,HTR,C} = 467.1$ [kJ/kg]
$h_{LP,FD,HTR,COND2,in} = 467.1$ [kJ/kg]
$h_{LP,HTR,out} = 467.5$ [kJ/kg]
$h_{RH,out} = 3554$ [kJ/kg]
$h_{TUR1,S2,out} = 2349$ [kJ/kg]
$h_{TUR2,S2,out} = 2414$ [kJ/kg]
$k_{poly} = 0.1944$
$m_1 = 0.9909$ [kg/s]
$m_2 = 0.08745$ [kg/s]
$m_3 = 0.06305$ [kg/s]
$m_4 = 0.115$ [kg/s]
$m_5 = 0.04397$ [kg/s]
$m_6 = 0.3476$ [kg/s]
$m_7 = 0.3429$ [kg/s]
$m_8 = 0.005175$ [kg/s]
$m_9 = 0.003973$ [kg/s]
$m_{tot} = 1$ [kg/s]
$P_{BFPTD,out} = 0.008651$ [MPa]
$P_{CEP,out} = 0.8$ [MPa]
$P_{COND,out} = 0.008651$ [MPa]
$P_{CW,in} = 0.1$ [MPa]
$P_{DA} = 0.6$ [MPa]
$P_{DA,BLEED} = 0.6$ [MPa]
$P_{DRUM} = 18.1$ [MPa]
$P_{ECON,out} = 18.1$ [MPa]
$P_{HPT,IP,in} = 16.9$ [MPa]
$P_{HPT,IV,in} = 16.9$ [MPa]
$P_{HP,FD,HTR,BLEED} = 2$ [MPa]
$P_{HP,FD,HTR,C} = 2$ [MPa]
$P_{HP,FD,HTR,DA,in} = 0.6$ [MPa]
$P_{HP,FD,HTR,H,in} = 2$ [MPa]
$P_{HP,FD,HTR,out} = 21.8$ [MPa]
$P_{HP,PUMP,out} = 22$ [MPa]
$P_{HP,SH,out} = 16.9$ [MPa]
$P_{HP,TUR,in} = 16.8$ [MPa]
$P_{HP,TUR,out} = 4.7$ [MPa]
$P_{IPT,IP,in} = 3.5$ [MPa]
$P_{IPT,IV,in} = 3.5$ [MPa]
$P_{IP,TUR,in} = 3.4$ [MPa]
$P_{IP,TUR,out} = 0.6$ [MPa]
$P_{LP,FD,HTR,BLEED} = 0.15$ [MPa]
$P_{LP,FD,HTR,C} = 0.15$ [MPa]
$P_{LP,FD,HTR,COND2,in} = 0.008651$ [MPa]
$P_{LP,HTR,out} = 0.6$ [MPa]
$P_{RH,out} = 3.5$ [MPa]
$P_{SAT,COND1} = 0.00482$ [MPa]
$P_{SAT,COND2} = 0.008651$ [MPa]
$q_{COND} = 1678$ [kW]

$q_{HPT,IP,loss}$ = 4.518 [kW]	$q_{IPT,IP,loss}$ = 3.562 [kW]	q_{DRUM} = 762.4 [kW]
q_{ECON} = 813.2 [kW]	q_{in} = 2964 [kW]	q_{loss} = 1686 [kW]
q_{RH} = 486.1 [kW]	q_{SH} = 901.8 [kW]	$\rho_{BFPTD,out}$ = 0.06386 [kg/m^3]
$\rho_{CEP,out}$ = 994.1 [kg/m^3]	$\rho_{COND,out}$ = 993.8 [kg/m^3]	$\rho_{CW,in}$ = 998.2 [kg/m^3]
$\rho_{CW,int}$ = 995.9 [kg/m^3]	$\rho_{CW,out}$ = 992.2 [kg/m^3]	$\rho_{DA,out}$ = 908.6 [kg/m^3]
$\rho_{DRUM,out}$ = 134.8 [kg/m^3]	$\rho_{ECON,out}$ = 541.2 [kg/m^3]	$\rho_{HPT,IP,in}$ = 50.75 [kg/m^3]
$\rho_{HPT,IV,in}$ = 50.91 [kg/m^3]	$\rho_{HP,FD,HTR,C}$ = 849.8 [kg/m^3]	$\rho_{HP,FD,HTR,DA,in}$ = 27.03 [kg/m^3]
$\rho_{HP,FD,HTR,H,in}$ = 6.038 [kg/m^3]	$\rho_{HP,FD,HTR,out}$ = 865.3 [kg/m^3]	$\rho_{HP,PUMP,out}$ = 917.6 [kg/m^3]
$\rho_{HP,SH,out}$ = 50.27 [kg/m^3]	$\rho_{HP,TUR,in}$ = 50.61 [kg/m^3]	$\rho_{HP,TUR,out}$ = 18.14 [kg/m^3]
$\rho_{IPT,IP,in}$ = 9.536 [kg/m^3]	$\rho_{IPT,IV,in}$ = 9.556 [kg/m^3]	$\rho_{IP,TUR,in}$ = 9.282 [kg/m^3]

Local variables in Module ExtractionPump\1 CALL ExtractionPump

$\Delta P_{LP,HTR}$ = 0.2 [MPa]	η_{LOSSES} = 0.702 [-]	η_{PUMP} = 0.85 [-]
h_{in} = 149.1 [kJ/kg]	$h_{isen,out}$ = 149.9 [kJ/kg]	h_{out} = 150 [kJ/kg]
m = 0.8495 [kg/s]	m_2 = 0.08745 [kg/s]	m_3 = 0.06305 [kg/s]
m_{tot} = 1 [kg/s]	$P_{DA,BLEED}$ = 0.6 [MPa]	P_{in} = 0.008651 [MPa]
P_{out} = 0.8 [MPa]	ρ_{in} = 993.8 [kg/m^3]	ρ_{out} = 994.1 [kg/m^3]
s_{in} = 0.5131 [kJ/kg-K]	$s_{isen,out}$ = 0.5131 [kJ/kg-K]	s_{out} = 0.5136 [kJ/kg-K]
T_{in} = 35.59 [C]	T_{out} = 35.64 [C]	w_{PUMP} = 0.7957 [kW]
$w_{PUMP,CYCLE}$ = 1.133 [kW]		

Local variables in Module LPFeedHeaters\1 CALL LPFeedHeaters

$h_{H,C}$ = 467.1 [kJ/kg]	$h_{H,in}$ = 2811 [kJ/kg]	$h_{H,out}$ = 467.1 [kJ/kg]
h_{in} = 150 [kJ/kg]	h_{out} = 467.5 [kJ/kg]	m_2 = 0.08745 [kg/s]
m_3 = 0.06305 [kg/s]	m_4 = 0.115 [kg/s]	m_{tot} = 1 [kg/s]
P_{COND2} = 0.008651 [MPa]	P_{DA} = 0.6 [MPa]	$P_{H,C}$ = 0.15 [MPa]
$P_{H,in}$ = 0.15 [MPa]	$P_{H,out}$ = 0.008651 [MPa]	P_{out} = 0.6 [MPa]
$\rho_{H,C}$ = 949.9 [kg/m^3]	$\rho_{H,out}$ = 0.4968 [kg/m^3]	ρ_{out} = 950.1 [kg/m^3]
$s_{H,C}$ = 1.434 [kJ/kg-K]	$s_{H,out}$ = 1.52 [kJ/kg-K]	s_{out} = 1.433 [kJ/kg-K]
$T_{H,C}$ = 111.3 [C]	$T_{H,out}$ = 43 [C]	T_{out} = 111.3 [C]

Local variables in Module Dearator\1 CALL Dearator

$h_{C,FD,HTR,out}$ = 908.5 [kJ/kg]	$h_{H,in}$ = 3074 [kJ/kg]	h_{in} = 467.5 [kJ/kg]
h_{out} = 670.4 [kJ/kg]	m_2 = 0.08745 [kg/s]	m_3 = 0.06305 [kg/s]
m_{tot} = 1 [kg/s]	P_{DA} = 0.6 [MPa]	$P_{DA,BLEED}$ = 0.6 [MPa]
ρ_{out} = 908.6 [kg/m^3]	s_{out} = 1.931 [kJ/kg-K]	T_{out} = 158.8 [C]

Local variables in Module BoilerFeedPump\1 CALL BoilerFeedPump

ΔP_{ECON} = 3.7 [MPa]	$\Delta P_{HP,HTR}$ = 0.2 [MPa]	η_{MECH} = 0.95 [-]

η_{PUMP} = 0.85 [-]
η_{TUR} = 0.88 [-]
h_{in} = 670.4 [kJ/kg]
$h_{isen,out}$ = 693.8 [kJ/kg]
h_{out} = 697.9 [kJ/kg]
$h_{TUR,in}$ = 3074 [kJ/kg]
$h_{TUR,isen,out}$ = 2324 [kJ/kg]
$h_{TUR,out}$ = 2414 [kJ/kg]
m_5 = 0.04397 [kg/s]
m_{tot} = 1 [kg/s]
P_{COND2} = 0.008651 [MPa]
P_{DRUM} = 18.1 [MPa]
P_{out} = 22 [MPa]
$P_{TUR,out}$ = 0.008651 [MPa]
ρ_{out} = 917.6 [kg/m3]
$\rho_{TUR,out}$ = 0.06386 [kg/m3]
s_{in} = 1.931 [kJ/kg-K]
$s_{isen,out}$ = 1.931 [kJ/kg-K]
s_{out} = 1.94 [kJ/kg-K]
$s_{TUR,in}$ = 7.395 [kJ/kg-K]
$s_{TUR,isen,out}$ = 7.395 [kJ/kg-K]

$s_{TUR,out}$ = 7.679 [kJ/kg-K]
T_{out} = 162.3 [C]
$T_{TUR,out}$ = 43 [C]
w_{PUMP} = 27.56 [kW]
$w_{TUR,out}$ = 29.01 [kW]

Local variables in Module HPFeedHeaters\1 CALL HPFeedHeaters

ΔP_{ECON} = 3.7 [MPa]
$h_{H,C}$ = 908.5 [kJ/kg]
$h_{H,in}$ = 3377 [kJ/kg]
$h_{H,out}$ = 908.5 [kJ/kg]
h_{in} = 697.9 [kJ/kg]
h_{out} = 915.8 [kJ/kg]
m_1 = 0.9909 [kg/s]
m_2 = 0.08745 [kg/s]
P_{DA} = 0.6 [MPa]
P_{DRUM} = 18.1 [MPa]
$P_{H,C}$ = 2 [MPa]
$P_{H,in}$ = 2 [MPa]
$P_{H,out}$ = 0.6 [MPa]
P_{in} = 22 [MPa]
P_{out} = 21.8 [MPa]
$\rho_{H,C}$ = 849.8 [kg/m3]
$\rho_{H,out}$ = 27.03 [kg/m3]
ρ_{out} = 865.3 [kg/m3]
$s_{H,C}$ = 2.447 [kJ/kg-K]
$s_{H,in}$ = 2.447 [kJ/kg-K]
$s_{H,out}$ = 2.482 [kJ/kg-K]
s_{in} = 1.94 [kJ/kg-K]
s_{out} = 2.414 [kJ/kg-K]
$T_{H,C}$ = 212.4 [C]
$T_{H,in}$ = 458.5 [C]
$T_{H,out}$ = 158.8 [C]
T_{in} = 162.3 [C]
T_{out} = 212.4 [C]

Local variables in Module Economiser\1 CALL Economiser

h_{in} = 915.8 [kJ/kg]
h_{out} = 1737 [kJ/kg]
m_1 = 0.9909 [kg/s]
P_{DRUM} = 18.1 [MPa]
P_{out} = 18.1 [MPa]
q_{in} = 813.2 [kW]
ρ_{out} = 541.2 [kg/m3]
s_{out} = 3.879 [kJ/kg-K]
T_{out} = 357.5 [C]

Local variables in Module Drum\1 CALL Drum

h_{in} = 1737 [kJ/kg]
h_{out} = 2506 [kJ/kg]
m_1 = 0.9909 [kg/s]
P_{DRUM} = 18.1 [MPa]
q_{in} = 762.4 [kW]
ρ_{out} = 134.8 [kg/m3]
s_{out} = 5.099 [kJ/kg-K]
T_{out} = 357.5 [C]

Local variables in Module HPSuperHeaters\1 CALL HPSuperHeaters

ΔP = 1.2 [MPa]
h_{in} = 2506 [kJ/kg]
h_{out} = 3416 [kJ/kg]
m_1 = 0.9909 [kg/s]
P_{in} = 18.1 [MPa]
P_{out} = 16.9 [MPa]
q_{in} = 901.8 [kW]
ρ_{out} = 50.27 [kg/m3]
s_{out} = 6.432 [kJ/kg-K]
T_{out} = 545 [C]

Local variables in Module HPTurbine\1 CALL HPTurbine

η_{TUR} = 0.9 [-]	h_{in} = 3397 [kJ/kg]	$h_{isen,out}$ = 3029 [kJ/kg]
h_{out} = 3066 [kJ/kg]	m_1 = 0.9909 [kg/s]	m_8 = 0.005175 [kg/s]
P_{out} = 4.7 [MPa]	ρ_{out} = 18.14 [kg/m3]	s_{in} = 6.411 [kJ/kg-K]
$s_{isen,out}$ = 6.411 [kJ/kg-K]	s_{out} = 6.472 [kJ/kg-K]	T_{out} = 345.8 [C]
W_{out} = 330.5 [kW]		

Local variables in Module HPAttemperation\1 CALL HPAttemperation

h_{att} = 697.9 [kJ/kg]	h_{in} = 3416 [kJ/kg]	h_{out} = 3402 [kJ/kg]
m_1 = 0.9909 [kg/s]	m_8 = 0.005175 [kg/s]	m_{out} = 0.996 [kg/s]
m_{tot} = 1 [kg/s]	P_{att} = 22 [MPa]	P_{in} = 16.9 [MPa]
P_{out} = 16.9 [MPa]	ρ_{out} = 50.75 [kg/m3]	s_{att} = 1.94 [kJ/kg-K]
s_{in} = 6.432 [kJ/kg-K]	s_{out} = 6.414 [kJ/kg-K]	T_{att} = 162.3 [C]
T_H = 540 [C]	T_{in} = 545 [C]	T_{out} = 540 [C]

Local variables in Module HPTurbineInletValve\1 CALL HPTurbineInletValve

ΔP = 0.1 [MPa]	$h_{ATT,out}$ = 3402 [kJ/kg]	h_{in} = 3397 [kJ/kg]
h_{out} = 3397 [kJ/kg]	m_1 = 0.9909 [kg/s]	m_8 = 0.005175 [kg/s]
$P_{ATT,out}$ = 16.9 [MPa]	P_{in} = 16.9 [MPa]	P_{out} = 16.8 [MPa]
q_{loss} = 4.518 [kW]	ρ_{in} = 50.91 [kg/m3]	ρ_{out} = 50.61 [kg/m3]
s_{in} = 6.409 [kJ/kg-K]	s_{out} = 6.411 [kJ/kg-K]	T_{in} = 538.4 [C]
T_{out} = 538 [C]		

Local variables in Module Reheaters\1 CALL Reheaters

ΔP = 1.2 [MPa]	h_{in} = 3066 [kJ/kg]	h_{out} = 3554 [kJ/kg]
m_1 = 0.9909 [kg/s]	m_8 = 0.005175 [kg/s]	P_{in} = 4.7 [MPa]
P_{out} = 3.5 [MPa]	q_{in} = 486.1 [kW]	ρ_{out} = 9.472 [kg/m3]
s_{out} = 7.288 [kJ/kg-K]	T_{out} = 545 [C]	

Local variables in Module RHAttemperation\1 CALL RHAttemperation

h_{att} = 697.9 [kJ/kg]	h_{in} = 3554 [kJ/kg]	h_{out} = 3542 [kJ/kg]
m_1 = 0.9909 [kg/s]	m_8 = 0.005175 [kg/s]	m_9 = 0.003973 [kg/s]
m_{tot} = 1 [kg/s]	P_{att} = 22 [MPa]	P_{in} = 3.5 [MPa]
P_{out} = 3.5 [MPa]	ρ_{out} = 9.536 [kg/m3]	s_{att} = 1.94 [kJ/kg-K]
s_{in} = 7.288 [kJ/kg-K]	s_{out} = 7.274 [kJ/kg-K]	T_{att} = 162.3 [C]
T_H = 540 [C]	T_{in} = 545 [C]	T_{out} = 540 [C]

Local variables in Module IPTurbineInletValve\1 CALL IPTurbineInletValve

ΔP = 0.1 [MPa]	h_{in} = 3539 [kJ/kg]	h_{out} = 3539 [kJ/kg]
$h_{SH,out}$ = 3542 [kJ/kg]	m = 1 [kg/s]	P_{in} = 3.5 [MPa]

P_{out} = 3.4 [MPa]
$P_{SH,out}$ = 3.5 [MPa]
q_{loss} = 3.562 [kW]
ρ_{in} = 9.556 [kg/m3]
ρ_{out} = 9.282 [kg/m3]
s_{in} = 7.269 [kJ/kg-K]
s_{out} = 7.282 [kJ/kg-K]
T_{in} = 538.4 [C]
T_{out} = 538 [C]

Local variables in Module Condenser\1 CALL Condenser

$h_{COND1,in}$ = 2349 [kJ/kg]
$h_{COND1,out}$ = 122.5 [kJ/kg]
$h_{COND2,in}$ = 2414 [kJ/kg]
$h_{COND2,out}$ = 167.5 [kJ/kg]
$h_{CW,in}$ = 84.01 [kJ/kg]
$h_{CW,int}$ = 122.6 [kJ/kg]
$h_{CW,out}$ = 167.6 [kJ/kg]
$h_{LP,FD,HTR,out}$ = 467.1 [kJ/kg]
h_{out} = 149.1 [kJ/kg]
m_4 = 0.115 [kg/s]
m_5 = 0.04397 [kg/s]
m_6 = 0.3476 [kg/s]
m_7 = 0.3429 [kg/s]
$m_{COND1,out}$ = 0.3476 [kg/s]
$m_{COND2,out}$ = 0.5019 [kg/s]
m_{CW} = 20.07 [kg/s]
$P_{CW,in}$ = 0.1 [MPa]
P_{out} = 0.008651 [MPa]
$P_{SAT,COND1}$ = 0.00482 [MPa]
$P_{SAT,COND2}$ = 0.008651 [MPa]
q_{loss} = 1678 [kW]
$\rho_{CW,in}$ = 998.2 [kg/m3]
$\rho_{CW,int}$ = 995.9 [kg/m3]
$\rho_{CW,out}$ = 992.2 [kg/m3]
ρ_{out} = 993.8 [kg/m3]
$s_{CW,in}$ = 0.2965 [kJ/kg-K]
$s_{CW,int}$ = 0.426 [kJ/kg-K]
$s_{CW,out}$ = 0.5724 [kJ/kg-K]
s_{out} = 0.5131 [kJ/kg-K]
TTD_{COND} = 3 [C]
$T_{COND1,out}$ = 29.22 [C]
$T_{COND2,out}$ = 40 [C]
$T_{CW,in}$ = 20 [C]
$T_{CW,int}$ = 29.22 [C]
$T_{CW,out}$ = 40 [C]
T_{out} = 35.59 [C]
T_R = 20 [C]
$T_{SAT,COND1}$ = 32.22 [C]
$T_{SAT,COND2}$ = 43 [C]
T_{SUB} = 3 [C]

Local variables in Module LPTurbines\1 CALL LPTurbines

η_{TUR} = 0.88 [-]
h_{BLEED} = 2811 [kJ/kg]
h_{in} = 3074 [kJ/kg]
$h_{isen,TUR1,S2,out}$ = 2251 [kJ/kg]
$h_{isen,TUR2,S2,out}$ = 2324 [kJ/kg]
$h_{TUR1,S2,out}$ = 2349 [kJ/kg]
$h_{TUR2,S2,out}$ = 2414 [kJ/kg]
k_{poly} = 0.1944 [-]
m_2 = 0.08745 [kg/s]
m_3 = 0.06305 [kg/s]
m_4 = 0.115 [kg/s]
m_5 = 0.04397 [kg/s]
m_6 = 0.3476 [kg/s]
m_7 = 0.3429 [kg/s]
m_{tot} = 1 [kg/s]
$m_{TUR1,S1}$ = 0.4028 [kg/s]
$m_{TUR1,S2}$ = 0.3476 [kg/s]
$m_{TUR2,S1}$ = 0.4028 [kg/s]
$m_{TUR2,S2}$ = 0.3429 [kg/s]
P_{BLEED} = 0.15 [MPa]
P_{in} = 0.6 [MPa]
$P_{SAT,COND1}$ = 0.00482 [MPa]
$P_{SAT,COND2}$ = 0.008651 [MPa]
$P_{TUR1,S2,out}$ = 0.00482 [MPa]
$P_{TUR2,S2,out}$ = 0.008651 [MPa]
ρ_{BLEED} = 0.7428 [kg/m3]
$\rho_{TUR1,S2,out}$ = 0.03753 [kg/m3] $\rho_{TUR2,S2,out}$ = 0.06386 [kg/m3]
s_{BLEED} = 7.509 [kJ/kg-K]
s_{in} = 7.395 [kJ/kg-K]
$s_{isen,TUR1,S2,out}$ = 7.395 [kJ/kg-K]
$s_{isen,TUR2,S2,out}$ = 7.395 [kJ/kg-K]
$s_{TUR1,S2,out}$ = 7.718 [kJ/kg-K]
$s_{TUR2,S2,out}$ = 7.679 [kJ/kg-K]
T_{BLEED} = 169 [C]
$T_{BLEED,K}$ = 442.1 [C]
T_{in} = 305.8 [C]
$T_{in,K}$ = 578.9 [C]
$T_{TUR1,S2,out}$ = 32.22 [C]
$T_{TUR2,S2,out}$ = 43 [C]

Local variables in Module IPTurbine\1 CALL IPTurbine

η_{isen} = 0.88 [-]
h_{bs} = 3377 [kJ/kg]
h_{in} = 3539 [kJ/kg]
$h_{isen,out}$ = 3011 [kJ/kg]
h_{out} = 3074 [kJ/kg]
k_{poly} = 0.1944 [-]
m_{bs} = 0.08745 [kg/s]
m_{in} = 1 [kg/s]
m_{out} = 0.9126 [kg/s]

P_{bs} = 2 [MPa]
P_{FWH} = 2 [MPa]
P_{in} = 3.4 [MPa]
P_{out} = 0.6 [MPa]
P_{outlet} = 0.6 [MPa]
ρ_{bs} = 6.038 [kg/m3]
ρ_{in} = 9.282 [kg/m3]
ρ_{out} = 2.278 [kg/m3]
s_{bs} = 7.312 [kJ/kg-K]
s_{in} = 7.282 [kJ/kg-K]
s_{out} = 7.395 [kJ/kg-K]
T_{bs} = 458.5 [C]
$T_{bs,K}$ = 731.6 [C]
T_{in} = 538 [C]
$T_{in,K}$ = 811.2 [C]
T_{out} = 305.8 [C]
$T_{out,K}$ = 578.9 [C]
W_{out} = 438.3 [kW]

ARRAYS TABLE

Nr.	Pressure [MPa]	T [°C]	h [kJ/kg]	s [kJ/kg-K]	rho [kg/m^3]
1	0.008651	35.59	149.1	0.5131	993.8
2	0.8	35.64	150	0.5136	994.1
3	0.6	111.3	467.5	1.433	950.1
4	0.6	158.8	670.4	1.931	908.6
5	22	162.3	697.9	1.94	917.6
6	21.8	212.4	915.8	2.414	865.3
7	18.1	357.5	1737	3.879	541.2
8	18.1	357.5	2506	5.099	134.8
9	16.9	545	3416	6.432	50.27
10	16.9	540	3402	6.414	50.75
11	16.9	538.4	3397	6.409	50.91
12	16.8	538	3397	6.411	50.61
13	4.7	345.8	3066	6.472	18.14
14	3.5	545	3554	7.288	9.472
15	3.5	540	3542	7.274	9.536
16	3.5	538.4	3539	7.269	9.556
17	3.4	538	3539	7.282	9.282
18	2	458.5	3377	7.312	6.038
19	0.6	305.8	3074	7.395	2.278
20	0.15	169	2811	7.509	0.7428
21	0.00482	32.22	2349	7.718	0.03753
22	0.008651	43	2414	7.679	0.06386
23	0.15	111.3	467.1	1.434	949.9
24	0.008651	43	467.1	1.52	0.4968
25	2	212.4	908.5	2.447	849.8
26	0.6	158.8	908.5	2.482	27.03
27	0.008651	43	2414	7.679	0.06386
28	0.1	20	84.01	0.2965	998.2
29	0.1	29.22	122.6	0.426	995.9
30	0.1	40	167.6	0.5724	992.2

8.2 VAPOUR COMPRESSION CYCLE SIMULATION MODEL

8.2.1 Simulation Model

--Component Functions--

Expansion Valve

MODULE **VCC**$_{Expansion,Valve}$ (WF\$, T_{in}, P_{in}, h_{in}, s_{in}, ρ_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out})

Outlet Conditions

$$P_{out} = \mathbf{P}\,(\,WF\$\,,T=T_{out}\,,h=h_{out}\,)$$

$$h_{out} = h_{in}$$

$$s_{out} = \mathbf{s}\,(\,WF\$\,,T=T_{out}\,,h=h_{out}\,)$$

$$\rho_{out} = \rho\,(\,WF\$\,,T=T_{out}\,,h=h_{out}\,)$$

$$x_{out} = \mathbf{x}\,(\,WF\$\,,T=T_{out}\,,h=h_{out}\,)$$

END **VCC**$_{Expansion,Valve}$

--

Evaporator

MODULE **VCC**$_{Evaporator}$ (WF\$, Q_{EVAP}, m_{CW}, m_{VCC}, $HX_{eff,EVAP}$, T_{in}, P_{in}, h_{in}, s_{in}, ρ_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out}, $T_{CW,in}$, $P_{CW,in}$, $h_{CW,in}$, $s_{CW,in}$, $\rho_{CW,in}$, $T_{CW,out}$, $P_{CW,out}$, $h_{CW,out}$, $s_{CW,out}$, $\rho_{CW,out}$)

Energy Balance

$$\dot{Q}_{EVAP} = \dot{m}_{CW} \cdot (\,h_{CW,in} - h_{CW,out}\,)$$

$$\dot{Q}_{EVAP} = \dot{m}_{VCC} \cdot (\,h_{out} - h_{in}\,)$$

Performance

$$HX_{eff,EVAP} = \frac{T_{CW,in} - T_{CW,out}}{T_{CW,in} - T_{in}}$$

Outlet Conditions (CW)

$$T_{CW,out} = \mathbf{T}\,(\,'Steam_{IAPWS}'\,,P=P_{CW,out}\,,h=h_{CW,out}\,)$$

$$P_{CW,out} = P_{CW,in}$$

$$s_{CW,out} = \mathbf{s}\,(\,'Steam_{IAPWS}'\,,P=P_{CW,out}\,,h=h_{CW,out}\,)$$

$$\rho_{CW,out} = \rho\,(\,'Steam_{IAPWS}'\,,P=P_{CW,out}\,,h=h_{CW,out}\,)$$

Outlet Conditions (WF)

$$T_{out} = \mathbf{T}\,(\,WF\$\,,P=P_{out}\,,x=x_{out}\,)$$

$$P_{out} = P_{in}$$

$h_{out} = \mathbf{h}\,(WF\$\,, P = P_{out}\,, x = x_{out})$

$s_{out} = \mathbf{s}\,(WF\$\,, P = P_{out}\,, x = x_{out})$

$\rho_{out} = \rho\,(WF\$\,, T = T_{out}\,, h = h_{out})$

$x_{out} = 1$

END **VCC**Evaporator

MODULE **VCC**Compressor (WF\$, η_{COMP}, $\dot{W}$, $\dot{m}$, T_{in}, P_{in}, h_{in}, s_{in}, ρ_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out})

Outlet Conditions

$h_{isen,out} = \mathbf{h}\,(WF\$\,, P = P_{out}\,, s = s_{in})$

$T_{out} = \mathbf{T}\,(WF\$\,, P = P_{out}\,, h = h_{out})$

$s_{out} = \mathbf{s}\,(WF\$\,, P = P_{out}\,, h = h_{out})$

$\rho_{out} = \rho\,(WF\$\,, T = T_{out}\,, h = h_{out})$

Isentropic Efficiency

$$\eta_{COMP} = \frac{h_{isen,out} - h_{in}}{h_{out} - h_{in}}$$

Calculating Required Work

$\dot{W} = \dot{m} \cdot (h_{out} - h_{in})$

END **VCC**Compressor

Condenser

MODULE **VCC**Compressor (WF\$, η_{COMP}, $\dot{W}$, $\dot{m}$, T_{in}, P_{in}, h_{in}, s_{in}, ρ_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out})

Energy Balance

$\dot{Q}_{COND} = \dot{m}_{FW} \cdot (h_{FW,out} - h_{FW,in})$

$\dot{Q}_{COND} = \dot{m}_{VCC} \cdot (h_{in} - h_{out})$

Performance

$T_{COND,sat} = T_{FW,out}$

$T_{out} = T_{COND,sat}$

Outlet Conditions (WF)

$P_{out} = \mathbf{P}\,(WF\$\,, T = T_{out}\,, x = x_{out})$

$h_{out} = \mathbf{h}\,(WF\$\,, T = T_{out}\,, x = x_{out})$

$s_{out} = \mathbf{s}\,(\,WF\$\,,\,T = T_{out}\,,\,x = x_{out}\,)$

$\rho_{out} = \rho\,(\,WF\$\,,\,T = T_{out}\,,\,x = x_{out}\,)$

$x_{out} = 0$

Pressure Relations

$P_{out} = P_{in}$

$P_{FW,out} = P_{FW,in} - \Delta P_{FW}$

END **VCC**Condenser

--*Main Program*--

Rankine Cycle Input Parameters

CW Information

$T_{CW,in} = \mathbf{T}\,(\,'Steam_{IAPWS}'\,,\,P = P_{CW,in}\,,\,h = h_{CW,in}\,)$

$P_{CW,in} = 0.1$ [MPa]

$h_{CW,in} = 84.01$ [kJ/kg]

$s_{CW,in} = \mathbf{s}\,(\,'Steam_{IAPWS}'\,,\,P = P_{CW,in}\,,\,h = h_{CW,in}\,)$

$\rho_{CW,in} = \rho\,(\,'Steam_{IAPWS}'\,,\,P = P_{CW,in}\,,\,h = h_{CW,in}\,)$

$\dot{m}_{CW} = 20.07$ [kg/s]

FW Inlet Information

$T_{FW,in} = \mathbf{T}\,(\,'Steam_{IAPWS}'\,,\,P = P_{FW,in}\,,\,h = h_{FW,in}\,)$

$P_{FW,in} = 0.8$ [MPa]

$h_{FW,in} = 150$ [kJ/kg]

$s_{FW,in} = \mathbf{s}\,(\,'Steam_{IAPWS}'\,,\,P = P_{FW,in}\,,\,h = h_{FW,in}\,)$

$\rho_{FW,in} = \rho\,(\,'Steam_{IAPWS}'\,,\,P = P_{FW,in}\,,\,h = h_{FW,in}\,)$

FW Inlet Information

$T_{FW,out} = \mathbf{T}\,(\,'Steam_{IAPWS}'\,,\,P = P_{FW,out}\,,\,h = h_{FW,out}\,)$

$h_{FW,out} = 467.5$ [kJ/kg]

$s_{FW,out} = \mathbf{s}\,(\,'Steam_{IAPWS}'\,,\,P = P_{FW,out}\,,\,h = h_{FW,out}\,)$

$\rho_{FW,out} = \rho\,(\,'Steam_{IAPWS}'\,,\,P = P_{FW,out}\,,\,h = h_{FW,out}\,)$

$\dot{m}_{FW} = 0.8495$ [kg/s]

VAPOUR COMPRESSION CYCLE

Additional Inputs

WF$ = 'Ethanol'

$\eta_{COMP} = 0.85$

$HX_{eff,EVAP} = 0.85$

$\Delta P_{COND,FW} = 0.2$

Calculations

The Vapour Compression Cycle Expansion Valve

Call $\mathbf{VCC}_{Expansion,Valve}$[WF$, T_4, P_4, h_4, s_4, ρ_4, T_1, P_1, h_1, s_1, ρ_1]

The Vapour Compression Cycle Evaporator

Call $\mathbf{VCC}_{Evaporator}$[WF$, Q_{EVAP}, m_{CW}, m_{VCC}, $HX_{eff,EVAP}$, T_1, P_1, h_1, s_1, ρ_1, T_2, P_2, h_2, s_2, ρ_2, $T_{CW,in}$, $P_{CW,in}$, $h_{CW,in}$, $s_{CW,in}$, $\rho_{CW,in}$, $T_{CW,out}$, $P_{CW,out}$, $h_{CW,out}$, $s_{CW,out}$, $\rho_{CW,out}$]

The Vapour Compression Cycle Compressor

Call $\mathbf{VCC}_{Compressor}$[WF$, η_{COMP}, W_{COMP}, m_{VCC}, T_2, P_2, h_2, s_2, ρ_2, T_3, P_3, h_3, s_3, ρ_3]

The Vapour Compression Cycle Condenser

Call $\mathbf{VCC}_{Condenser}$[WF$, Q_{COND}, m_{FW}, m_{VCC}, $\Delta P_{COND,FW}$, T_3, P_3, h_3, s_3, ρ_3, T_4, P_4, h_4, s_4, ρ_4, $T_{FW,in}$, $P_{FW,in}$, $h_{FW,in}$, $s_{FW,in}$, $\rho_{FW,in}$, $T_{FW,out}$, $P_{FW,out}$, $h_{FW,out}$, $s_{FW,out}$, $\rho_{FW,out}$]

8.2.2 Results

SOLUTION

Unit Settings: [kJ]/[C]/[MPa]/[kg]/[degrees]

$\Delta P_{COND,FW} = 0.2$ [MPa]	$\eta_{COMP} = 0.85$ [-]	$HX_{eff,EVAP} = 0.85$ [-]
$h_{CW,in} = 84.01$ [kJ/kg]	$h_{CW,out} = 75.03$ [kJ/kg]	$h_{FW,in} = 150$ [kJ/kg]
$h_{FW,out} = 467.5$ [kJ/kg]	$m_{CW} = 20.07$ [kg/s]	$m_{FW} = 0.8495$ [kg/s]
$m_{VCC} = 0.2767$ [kg/s]	$P_{CW,in} = 0.1$ [MPa]	$P_{CW,out} = 0.1$ [MPa]
$P_{FW,in} = 0.8$ [MPa]	$P_{FW,out} = 0.6$ [MPa]	$Q_{COND} = 269.7$ [kW]
$Q_{EVAP} = 180.3$ [kW]	$\rho_{CW,in} = 998.2$ [kg/m3]	$\rho_{CW,out} = 998.6$ [kg/m3]
$\rho_{FW,in} = 994.1$ [kg/m3]	$\rho_{FW,out} = 950.1$ [kg/m3]	$s_{CW,in} = 0.2965$ [kJ/kg-K]
$s_{CW,out} = 0.2657$ [kJ/kg-K]	$s_{FW,in} = 0.5135$ [kJ/kg-K]	$s_{FW,out} = 1.433$ [kJ/kg-K]
$T_{CW,in} = 20$ [C]	$T_{CW,out} = 17.85$ [C]	$T_{FW,in} = 35.63$ [C]
$T_{FW,out} = 111.4$ [C]	WF$ = 'Ethanol'	$W_{COMP} = 89.46$ [kW]

Local variables in Module VCC_Expansion_Valve\1 CALL VCC_Expansion_Valve

$h_{in} = 368.3$ [kJ/kg]	$h_{out} = 368.3$ [kJ/kg]	$P_{in} = 0.328$ [MPa]
$P_{out} = 0.005029$ [MPa]	$\rho_{in} = 700$ [kg/m3]	$\rho_{out} = 0.3184$ [kg/m3]
$s_{in} = 1.328$ [kJ/kg-K]	$s_{out} = 1.464$ [kJ/kg-K]	$T_{in} = 111.4$ [C]
$T_{out} = 17.48$ [C]	WF$ = 'Ethanol'	$x_{out} = 0.2992$ [-]

Local variables in Module VCC_Evaporator\1 CALL VCC_Evaporator

$HX_{eff,EVAP}$ = 0.85 [-] $h_{CW,in}$ = 84.01 [kJ/kg] $h_{CW,out}$ = 75.03 [kJ/kg]
h_{in} = 368.3 [kJ/kg] h_{out} = 1020 [kJ/kg] m_{CW} = 20.07 [kg/s]
m_{VCC} = 0.2767 [kg/s] $P_{CW,in}$ = 0.1 [MPa] $P_{CW,out}$ = 0.1 [MPa]
P_{in} = 0.005029 [MPa] P_{out} = 0.005029 [MPa] Q_{EVAP} = 180.3 [kW]
$\rho_{CW,in}$ = 998.2 [kg/m3] $\rho_{CW,out}$ = 998.6 [kg/m3] ρ_{in} = 0.3184 [kg/m3]
ρ_{out} = 0.02507 [kg/m3] $s_{CW,in}$ = 0.2965 [kJ/kg-K] $s_{CW,out}$ = 0.2657 [kJ/kg-K]
s_{in} = 1.464 [kJ/kg-K] s_{out} = 3.707 [kJ/kg-K] $T_{CW,in}$ = 20 [C]
$T_{CW,out}$ = 17.85 [C] T_{in} = 17.48 [C] T_{out} = 17.48 [C]
WF$ = 'Ethanol' x_{out} = 1 [-]

Local variables in Module VCC_Compressor\1 CALL VCC_Compressor

η_{COMP} = 0.85 [-] h_{in} = 1020 [kJ/kg] $h_{isen,out}$ = 1295 [kJ/kg]
h_{out} = 1343 [kJ/kg] m = 0.2767 [kg/s] P_{in} = 0.005029 [MPa]
P_{out} = 0.328 [MPa] ρ_{in} = 0.02507 [kg/m3] ρ_{out} = 3.911 [kg/m3]
s_{in} = 3.707 [kJ/kg-K] s_{out} = 3.811 [kJ/kg-K] T_{in} = 17.48 [C]
T_{out} = 202.9 [C] WF$ = 'Ethanol' W = 89.46 [kJ/s]

Local variables in Module VCC_Condenser\1 CALL VCC_Condenser

ΔP_{FW} = 0.2 [MPa] $h_{FW,in}$ = 150 [kJ/kg] $h_{FW,out}$ = 467.5 [kJ/kg]
h_{in} = 1343 [kJ/kg] h_{out} = 368.3 [kJ/kg] m_{FW} = 0.8495 [kg/s]
m_{VCC} = 0.2767 [kg/s] $P_{FW,in}$ = 0.8 [MPa] $P_{FW,out}$ = 0.6 [MPa]
P_{in} = 0.328 [MPa] P_{out} = 0.328 [MPa] Q_{COND} = 269.7 [kW]
$\rho_{FW,in}$ = 994.1 [kg/m3] $\rho_{FW,out}$ = 950.1 [kg/m3] ρ_{in} = 3.911 [kg/m3]
ρ_{out} = 700 [kg/m3] $s_{FW,in}$ = 0.5135 [kJ/kg-K] $s_{FW,out}$ = 1.433 [kJ/kg-K]
s_{in} = 3.811 [kJ/kg-K] s_{out} = 1.328 [kJ/kg-K] $T_{COND,sat}$ = 111.4 [C]
$T_{FW,in}$ = 35.63 [C] $T_{FW,out}$ = 111.4 [C] T_{in} = 202.9 [C]
T_{out} = 111.4 [C] WF$ = 'Ethanol' x_{out} = 0 [-]

ARRAYS TABLE

Nr.	Pressure [MPa]	T [°C]	h [kJ/kg]	s [kJ/kg-K]	rho [kg/m3]
1	0.005029	17.48	368.3	1.464	0.3184
2	0.005029	17.48	1020	3.707	0.02507
3	0.328	202.9	1343	3.811	3.911
4	0.328	111.4	368.3	1.328	700

8.3 RANKINE-HEAT PUMP COMBINED CYCLE SIMULATION MODEL

8.3.1 Simulation Model

--Component Functions--

RCC

MODULE **Condenser** (T_R, TTD_{COND}, T_{SUB}, m_6, m_7, m_5, m_4, $P_{SAT,COND1}$, $P_{SAT,COND2}$, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, $h_{COND1,in}$, $h_{COND2,in}$, $h_{LP,FD,HTR,out}$, q_{loss}, $T_{CW,in}$, $P_{CW,in}$, $h_{CW,in}$, $s_{CW,in}$, $\rho_{CW,in}$, $T_{CW,int}$, $h_{CW,int}$, $s_{CW,int}$, $\rho_{CW,int}$, $T_{CW,out}$, $h_{CW,out}$, $s_{CW,out}$, $\rho_{CW,out}$)

Condenser Performance

$$T_{CW,out} = T_{CW,in} + T_R$$

$$T_{SAT,COND1} = T_{CW,int} + TTD_{COND}$$

$$T_{SAT,COND2} = T_{CW,out} + TTD_{COND}$$

Condenser Pressures

$$P_{SAT,COND1} = \mathbf{P}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{SAT,COND1}, x = 0 \right)$$

$$P_{SAT,COND2} = \mathbf{P}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{SAT,COND2}, x = 0 \right)$$

Outlet Conditions per Condenser

$$T_{COND1,out} = T_{SAT,COND1} - T_{SUB}$$

$$h_{COND1,out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{SAT,COND1}, T = T_{COND1,out} \right)$$

$$T_{COND2,out} = T_{SAT,COND2} - T_{SUB}$$

$$h_{COND2,out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{SAT,COND2}, T = T_{COND2,out} \right)$$

Outlet Mass Flow Rates per Condenser

$$\dot{m}_{COND1,out} = \dot{m}_6$$

$$\dot{m}_{COND2,out} = \dot{m}_7 + \dot{m}_4 + \dot{m}_5$$

Combined Outlet Conditions

$$P_{out} = P_{SAT,COND2}$$

$$T_{out} = \mathbf{T}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out} \right)$$

$$h_{out} \cdot \left(\dot{m}_{COND1,out} + \dot{m}_{COND2,out} \right) = \dot{m}_{COND1,out} \cdot h_{COND1,out} + \dot{m}_{COND2,out} \cdot h_{COND2,out}$$

$$s_{out} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out} \right)$$

$$\rho_{out} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out} \right)$$

Cooling Water Enthalpy

$$h_{CW,in} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,in}, P = P_{CW,in} \right)$$

$$h_{CW,int} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,int}, P = P_{CW,in})$$

$$h_{CW,out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,out}, P = P_{CW,in})$$

Energy Balances

$$\dot{m}_{CW} \cdot (h_{CW,int} - h_{CW,in}) = \dot{m}_6 \cdot (h_{COND1,in} - h_{COND1,out})$$

$$\dot{m}_{CW} \cdot (h_{CW,out} - h_{CW,int}) = (\dot{m}_5 + \dot{m}_7 + \dot{m}_4) \cdot (h_{COND2,in} - h_{COND2,out})$$

$$q_{loss} = \dot{m}_6 \cdot (h_{COND1,in} - h_{COND1,out}) + (\dot{m}_5 + \dot{m}_7 + \dot{m}_4) \cdot (h_{COND2,in} - h_{COND2,out})$$

Other CW Properties

$$s_{CW,in} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,in}, P = P_{CW,in})$$

$$s_{CW,out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,out}, P = P_{CW,in})$$

$$\rho_{CW,in} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,in}, P = P_{CW,in})$$

$$\rho_{CW,out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{CW,out}, P = P_{CW,in})$$

END **Condenser**

CEP

MODULE **ExtractionPump** ($P_{DA,BLEED}$, $\Delta P_{LP,HTR}$, η_{PUMP}, η_{LOSSES}, P_{in}, P_{out}, T_{in}, T_{out}, h_{in}, h_{out}, s_{in}, s_{out}, ρ_{in}, ρ_{out}, m_{tot}, m_2, m_3, w_{PUMP}, $w_{PUMP,CYCLE}$)

Outlet Conditions

$$P_{out} = P_{DA,BLEED} + \Delta P_{LP,HTR}$$

$$h_{isen,out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, s = s_{isen,out})$$

$$s_{isen,out} = s_{in}$$

$$T_{out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out}) \qquad s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, T = T_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\dot{m} = \dot{m}_{tot} - \dot{m}_2 - \dot{m}_3$$

Isentropic Efficiency

$$\eta_{PUMP} = \frac{h_{isen,out} - h_{in}}{h_{out} - h_{in}}$$

Pump Work

$$w_{PUMP} = (h_{out} - h_{in}) \cdot \dot{m}$$

Converting Pump Work to Shaft Power

$$w_{PUMP,CYCLE} = \frac{w_{PUMP}}{\eta_{LOSSES}}$$

END **ExtractionPump**

HP FWH

MODULE **HPFeedHeaters** (P_{DRUM}, ΔP_{ECON}, m_1, m_2, T_{in}, P_{in}, h_{in}, s_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out}, $T_{H,in}$, $P_{H,in}$, $h_{H,in}$, $s_{H,in}$, $T_{H,C}$, $P_{H,C}$, $h_{H,C}$, $s_{H,C}$, $\rho_{H,C}$, P_{DA}, $T_{H,out}$, $P_{H,out}$, $h_{H,out}$, $s_{H,out}$, $\rho_{H,out}$)

FW Outlet Conditions

$$P_{out} = P_{DRUM} + \Delta P_{ECON}$$

$$T_{out} = T_{H,C}$$

$$h_{out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out}\right)$$

$$s_{out} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out}\right)$$

$$\rho_{out} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out}\right)$$

Condensate conditions

$$P_{H,C} = P_{H,in}$$

$$T_{H,C} = \mathbf{T}\left(\text{'Steam}_{IAPWS}\text{'}, x = 0, P = P_{H,C}\right)$$

$$h_{H,C} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, x = 0, P = P_{H,C}\right)$$

$$s_{H,C} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, x = 0, P = P_{H,C}\right)$$

$$\rho_{H,C} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, x = 0, P = P_{H,C}\right)$$

Energy Balance

$$\dot{m}_1 \cdot \left(h_{out} - h_{in}\right) = \dot{m}_2 \cdot \left(h_{H,in} - h_{H,C}\right)$$

Hot Side Outlet Conditions – After Expanding

$$P_{H,out} = P_{DA}$$

$$T_{H,out} = \mathbf{T}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,out}, h = h_{H,out}\right)$$

$$h_{H,out} = h_{H,C}$$

$$s_{H,out} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,out}, h = h_{H,out}\right)$$

$$\rho_{H,out} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{H,out}, h = h_{H,out}\right)$$

END **HPFeedHeaters**

Dearator

MODULE **Dearator** ($P_{DA,BLEED}$, T_{out}, P_{DA}, h_{out}, s_{out}, ρ_{out}, h_{in}, $h_{H,in}$, $h_{C,FD,HTR,out}$, m_{tot}, m_2, m_3)

Outlet Conditions

$$P_{DA} = P_{DA,BLEED}$$

$$T_{out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DA}, x = 0)$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DA}, x = 0)$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DA}, x = 0)$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{DA}, x = 0)$$

Energy Balance

$$(\dot{m}_{tot} - \dot{m}_2 - \dot{m}_3) \cdot (h_{out} - h_{in}) = \dot{m}_2 \cdot (h_{C,FD,HTR,out} - h_{out}) + \dot{m}_3 \cdot (h_{H,in} - h_{out})$$

END **Dearator**

BFP

MODULE **BoilerFeedPump** (η_{PUMP}, η_{MECH}, η_{TUR}, m_{tot}, m_5, P_{DRUM}, $\Delta P_{HP,HTR}$, ΔP_{ECON}, h_{in}, s_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out}, P_{COND2}, $h_{TUR,in}$, $s_{TUR,in}$, $P_{TUR,out}$, $T_{TUR,out}$, $h_{TUR,out}$, $s_{TUR,out}$, $\rho_{TUR,out}$)

Pump Outlet Conditions

$$P_{out} = P_{DRUM} + \Delta P_{HP,HTR} + \Delta P_{ECON}$$

$$h_{isen,out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, s = s_{isen,out})$$

$$s_{isen,out} = s_{in}$$

$$T_{out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Turbine Outlet Conditions

$$P_{TUR,out} = P_{COND2}$$

$$h_{TUR,isen,out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{TUR,out}, s = s_{TUR,isen,out})$$

$$s_{TUR,isen,out} = s_{TUR,in}$$

$$T_{TUR,out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{TUR,out}, h = h_{TUR,out})$$

$$s_{TUR,out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{TUR,out}, h = h_{TUR,out})$$

$$\rho_{TUR,out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{TUR,out}, h = h_{TUR,out})$$

Isentropic Efficiency

$$\eta_{PUMP} = \frac{h_{isen,out} - h_{in}}{h_{out} - h_{in}}$$

$$\eta_{TUR} = \frac{h_{TUR,out} - h_{TUR,in}}{h_{TUR,isen,out} - h_{TUR,in}}$$

Pump/Turbine Work

$$w_{PUMP} = (h_{out} - h_{in}) \cdot \dot{m}_{tot}$$

$$w_{TUR,out} = \dot{m}_5 \cdot (h_{TUR,in} - h_{TUR,out})$$

$$w_{PUMP} = w_{TUR,out} \cdot \eta_{MECH}$$

END **BoilerFeedPump**

Economiser

MODULE **Economiser** (P_{DRUM}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, h_{in}, q_{in}, m_1)

Outlet Conditions

$$P_{out} = P_{DRUM}$$

$$T_{out} = \mathbf{T}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, x = 0 \right)$$

$$h_{out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, x = 0 \right)$$

$$s_{out} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, x = 0 \right)$$

$$\rho_{out} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, x = 0 \right)$$

Energy Balance

$$q_{in} = \dot{m}_1 \cdot (h_{out} - h_{in})$$

END **Economiser**

Drum

MODULE **Drum** (P_{DRUM}, T_{out}, h_{out}, s_{out}, ρ_{out}, h_{in}, q_{in}, m_1)

Outlet Conditions

$$T_{out} = \mathbf{T}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{DRUM}, x = 1 \right)$$

$$h_{out} = \mathbf{h}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{DRUM}, x = 1 \right)$$

$$s_{out} = \mathbf{s}\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{DRUM}, x = 1 \right) \qquad \rho_{out} = \rho\left(\text{'Steam}_{IAPWS}\text{'}, P = P_{DRUM}, x = 1 \right)$$

Energy Balance

$$q_{in} = \dot{m}_1 \cdot (h_{out} - h_{in})$$

END **Drum**

Super Heaters

MODULE **HPSuperHeaters** (ΔP, T_{out}, P_{in}, P_{out}, h_{out}, s_{out}, ρ_{out}, h_{in}, q_{in}, m_1)

Outlet Conditions

$$P_{out} = P_{in} - \Delta P$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T=T_{out}, P=P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T=T_{out}, P=P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T=T_{out}, P=P_{out})$$

Energy Balance

$$q_{in} = \dot{m}_1 \cdot (h_{out} - h_{in})$$

END **HPSuperHeaters**

Super Heater Attemperation

MODULE **HPAttemperation** (T_H, P_{in}, T_{in}, h_{in}, s_{in}, P_{att}, T_{att}, h_{att}, s_{att}, m_{tot}, m_1, m_8, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out})

Mass Balance

$$\dot{m}_1 = \dot{m}_{out} - \dot{m}_8$$

Super Heater Outlet Conditions If Attemperation Would Have Been Applied At Inlets

$$P_{out} = P_{in}$$

$$T_{out} = T_H$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T=T_{out}, P=P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T=T_{out}, P=P_{out}) \qquad \rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T=T_{out}, P=P_{out})$$

Energy Balance

$$\dot{m}_{out} \cdot h_{out} = \dot{m}_1 \cdot h_{in} + \dot{m}_8 \cdot h_{att}$$

END **HPAttemperation**

HP turbine Inlet Valve and Piping

MODULE **HPTurbineInletValve** (m_1, m_8, ΔP, P_{in}, T_{in}, h_{in}, s_{in}, ρ_{in}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, q_{loss}, $P_{ATT,out}$, $h_{ATT,out}$)

Turbine Inlet Valve Outlet Conditions

$$P_{out} = P_{in} - \Delta P$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T=T_{out}, P=P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T=T_{out}, P=P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T=T_{out}, P=P_{out})$$

Turbine Inlet Valve Inlet Conditions

$P_{in} = P_{ATT,out}$

$T_{in} = \mathbf{T}(\text{'Steam}_{IAPWS}\text{'}, P = P_{in}, h = h_{in})$

$h_{in} = h_{out}$

$s_{in} = \mathbf{s}(\text{'Steam}_{IAPWS}\text{'}, T = T_{in}, P = P_{in})$

$\rho_{in} = \rho(\text{'Steam}_{IAPWS}\text{'}, T = T_{in}, P = P_{in})$

Heat Loss In The Piping Between The Super Heaters And The Turbine Inlet Valve

$\dot{q}_{loss} = (\dot{m}_1 + \dot{m}_8) \cdot (h_{ATT,out} - h_{in})$

END **HPTurbineInletValve**

HP turbine

MODULE **HPTurbine** (m_1, m_8, η_{TUR}, h_{in}, s_{in}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, W_{out})

Outlet Conditions

$h_{isen,out} = \mathbf{h}(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, s = s_{isen,out})$

$s_{isen,out} = s_{in}$

$T_{out} = \mathbf{T}(\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out})$

$s_{out} = \mathbf{s}(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$

$\rho_{out} = \rho(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$

Isentropic Efficiency

$$\eta_{isen} = \frac{h_{out} - h_{in}}{h_{isen,out} - h_{in}}$$

Turbine Work Output

$w_{out} = (\dot{m}_1 + \dot{m}_8) \cdot (h_{in} - h_{out})$

END **HPTurbine**

Reheaters

MODULE **Reheaters** (ΔP, T_{out}, P_{in}, P_{out}, h_{out}, s_{out}, ρ_{out}, h_{in}, q_{in}, m_1, m_8)

Outlet Conditions

$P_{out} = P_{in} - \Delta P$

$h_{out} = \mathbf{h}(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Heat Loss In The Piping Between The Reheaters And The Turbine Inlet Valve

$$q_{in} = (\dot{m}_1 + \dot{m}_8) \cdot (h_{out} - h_{in})$$

END **Reheaters**

Reheater Attemperation

MODULE **RHAttemperation** (T_H, P_{in}, T_{in}, h_{in}, s_{in}, P_{att}, T_{att}, h_{att}, s_{att}, m_{tot}, m_1, m_8, m_9, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out})

Mass Balance

$$\dot{m}_1 + \dot{m}_8 = \dot{m}_{tot} - \dot{m}_9$$

Super Heater Outlet Conditions If Attemperation Would Have Been Applied At Inlets

$$P_{out} = P_{in}$$

$$T_{out} = T_H$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Energy Balance

$$\dot{m}_{tot} \cdot h_{out} = (\dot{m}_1 + \dot{m}_8) \cdot h_{in} + \dot{m}_9 \cdot h_{att}$$

END **RHAttemperation**

IP Turbine Inlet Valve and Piping

MODULE **IPTurbineInletValve** (m, ΔP, P_{in}, T_{in}, h_{in}, s_{in}, ρ_{in}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, q_{loss}, $P_{SH,out}$, $h_{SH,out}$)

Turbine Inlet Valve Outlet Conditions

$$P_{out} = P_{in} - \Delta P$$

$$h_{out} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$s_{out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Turbine Inlet Valve Inlet Conditions

$$P_{in} = P_{SH,out}$$

$$T_{in} = \mathbf{T}\ (\text{'Steam}_{IAPWS}\text{'}, P = P_{in}, h = h_{in})$$

$$h_{in} = h_{out}$$

$$s_{in} = \mathbf{s}\ (\text{'Steam}_{IAPWS}\text{'}, T = T_{in}, P = P_{in})$$

$$\rho_{in} = \rho\ (\text{'Steam}_{IAPWS}\text{'}, T = T_{in}, P = P_{in})$$

Energy Balance

$$\dot{q}_{loss} = \dot{m} \cdot (h_{SH,out} - h_{in})$$

END **IPTurbineInletValve**

IP Turbine

MODULE **IPTurbine** (m_{in}, m_{bs}, η_{isen}, k_{poly}, P_{FWH}, P_{outlet}, P_{in}, T_{in}, h_{in}, s_{in}, ρ_{in}, P_{bs}, T_{bs}, h_{bs}, s_{bs}, ρ_{bs}, P_{out}, T_{out}, h_{out}, s_{out}, ρ_{out}, W_{out})

Mass Balance

$$\dot{m}_{in} = \dot{m}_{out} + \dot{m}_{bs}$$

Turbine Outlet Conditions – Isentropic Efficiency

$$P_{out} = P_{outlet}$$

$$h_{isen,out} = \mathbf{h}\ (\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, s = s_{in})$$

$$T_{out} = \mathbf{T}\ (\text{'Steam}_{IAPWS}\text{'}, P = P_{out}, h = h_{out})$$

$$s_{out} = \mathbf{s}\ (\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

$$\rho_{out} = \rho\ (\text{'Steam}_{IAPWS}\text{'}, T = T_{out}, P = P_{out})$$

Isentropic Efficiency

$$\eta_{isen} = \frac{h_{out} - h_{in}}{h_{isen,out} - h_{in}}$$

Converting temperature to Kelvin for polytropic efficiency calculations

$$T_{in,K} = T_{in} + 273.15 \quad [K]$$

$$T_{bs,K} = T_{bs} + 273.15 \quad [K]$$

$$T_{out,K} = T_{out} + 273.15 \quad [K]$$

Bled Steam Conditions – Polytropic Efficiency

$$P_{bs} = P_{FWH}$$

$$h_{bs} = \mathbf{h}\,('Steam_{IAPWS}', T = T_{bs}, P = P_{bs})$$

$$s_{bs} = \mathbf{s}\,('Steam_{IAPWS}', T = T_{bs}, P = P_{bs})$$

$$\rho_{bs} = \rho\,('Steam_{IAPWS}', T = T_{bs}, P = P_{bs})$$

Polytropic Efficiency

$$\frac{T_{in,K}}{T_{out,K}} = \left[\frac{P_{in}}{P_{out}}\right]^{k_{poly}}$$

$$\frac{T_{in,K}}{T_{bs,K}} = \left[\frac{P_{in}}{P_{bs}}\right]^{k_{poly}}$$

Turbine Work Output

$$w_{out} = \dot{m}_{in} \cdot (h_{in} - h_{bs}) + \dot{m}_{out} \cdot (h_{bs} - h_{out})$$

END **IPTurbine**

LP turbine

MODULE **LPTurbines** (η_{TUR}, k_{poly}, P_{in}, T_{in}, h_{in}, s_{in}, P_{BLEED}, T_{BLEED}, h_{BLEED}, s_{BLEED}, ρ_{BLEED}, $P_{SAT,COND1}$, $T_{TUR1,S2,out}$, $h_{TUR1,S2,out}$, $s_{TUR1,S2,out}$, $\rho_{TUR1,S2,out}$, $P_{SAT,COND2}$, $T_{TUR2,S2,out}$, $h_{TUR2,S2,out}$, $s_{TUR2,S2,out}$, $\rho_{TUR2,S2,out}$, m_{tot}, m_2, m_3, m_4, m_5, m_6, m_7, W_{out})

Mass Balance

$$\dot{m}_{TUR1} + \dot{m}_{TUR2} = \dot{m}_{tot} - \dot{m}_2 - \dot{m}_3 - \dot{m}_4 - \dot{m}_5$$

$$\frac{P_{in} - P_{SAT,COND1}}{P_{in} - P_{SAT,COND2}} = \frac{\dot{m}_{TUR1}^{\;2}}{\dot{m}_{TUR2}^{\;2}}$$

$$\dot{m}_{TUR1} = \dot{m}_6$$

$$\dot{m}_{TUR2} = \dot{m}_7$$

Turbine 1 Outlet Conditions – Isentropic Efficiency

$$P_{TUR1,out} = P_{SAT,COND1}$$

$$h_{isen,TUR1,out} = \mathbf{h}\,('Steam_{IAPWS}', P = P_{TUR1,out}, s = s_{isen,TUR1,out})$$

$$s_{isen,TUR1,out} = s_{in}$$

$$T_{TUR1,out} = \mathbf{T}\,('Steam_{IAPWS}', P = P_{TUR1,out}, h = h_{TUR1,out})$$

$$s_{TUR1,out} = \mathbf{s}\,('Steam_{IAPWS}', h = h_{TUR1,out}, P = P_{TUR1,out})$$

$$\rho_{TUR1,out} = \rho\,('Steam_{IAPWS}', h = h_{TUR1,out}, P = P_{TUR1,out})$$

Turbine 2 Outlet Conditions – Isentropic Efficiency

$$P_{TUR2,out} = P_{SAT,COND2}$$

$$s_{isen,TUR2,out} = s_{in}$$

$$T_{TUR2,out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{TUR2,out}, h = h_{TUR2,out})$$

$$s_{TUR2,out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, h = h_{TUR2,out}, P = P_{TUR2,out})$$

$$\rho_{TUR2,out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, h = h_{TUR2,out}, P = P_{TUR2,out})$$

Isentropic Efficiency

$$\eta_{TUR} = \frac{h_{TUR1,out} - h_{in}}{h_{isen,TUR1,out} - h_{in}}$$

$$\eta_{TUR} = \frac{h_{TUR2,out} - h_{in}}{h_{isen,TUR2,out} - h_{in}}$$

Turbine Work Output

$$w_{out} = \dot{m}_6 \cdot (h_{in} - h_{TUR1,out}) + \dot{m}_7 \cdot (h_{in} - h_{TUR2,out})$$

END **LPTurbines**

Vapour Compression Cycle Expansion Valve

MODULE $\mathbf{VCC}_{Expansion,Valve}$ (WF$, T_{in}, P_{in}, h_{in}, s_{in}, ρ_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out})

Outlet Conditions

$$P_{out} = \mathbf{P}\,(WF\$, T = T_{out}, h = h_{out})$$

$$h_{out} = h_{in}$$

$$s_{out} = \mathbf{s}\,(WF\$, T = T_{out}, h = h_{out})$$

$$\rho_{out} = \rho\,(WF\$, T = T_{out}, h = h_{out})$$

$$x_{out} = \mathbf{x}\,(WF\$, T = T_{out}, h = h_{out})$$

END $\mathbf{VCC}_{Expansion,Valve}$

Vapour Compression Cycle Evaporator

MODULE $\mathbf{VCC}_{Evaporator}$ (WF$, Q_{EVAP}, m_{CW}, m_{VCC}, $HX_{eff,EVAP}$, T_{in}, P_{in}, h_{in}, s_{in}, ρ_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out}, $T_{CW,in}$, $P_{CW,in}$, $h_{CW,in}$, $s_{CW,in}$, $\rho_{CW,in}$, $T_{CW,out}$, $P_{CW,out}$, $h_{CW,out}$, $s_{CW,out}$, $\rho_{CW,out}$)

Inlet Conditions (CW)

$$h_{CW,in} = \mathbf{h}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{CW,in}, T = T_{CW,in})$$

$$s_{CW,in} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{CW,in}, T = T_{CW,in})$$

$$\rho_{CW,in} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, P = P_{CW,in}, T = T_{CW,in})$$

Energy Balance

$$\dot{Q}_{EVAP} = \dot{m}_{CW} \cdot (h_{CW,in} - h_{CW,out})$$

$$\dot{Q}_{EVAP} = \dot{m}_{VCC} \cdot (h_{out} - h_{in})$$

Performance

$$HX_{eff,EVAP} = \frac{T_{CW,in} - T_{CW,out}}{T_{CW,in} - T_{in}}$$

Outlet Conditions (CW)

$$T_{CW,out} = \mathbf{T}\,('Steam_{IAPWS}', P = P_{CW,out}, h = h_{CW,out})$$

$$P_{CW,out} = P_{CW,in}$$

$$s_{CW,out} = \mathbf{s}\,('Steam_{IAPWS}', P = P_{CW,out}, h = h_{CW,out})$$

$$\rho_{CW,out} = \rho\,('Steam_{IAPWS}', P = P_{CW,out}, h = h_{CW,out})$$

Outlet Conditions (WF)

$$T_{out} = \mathbf{T}\,(WF\$, P = P_{out}, x = x_{out})$$

$$P_{out} = P_{in}$$

$$h_{out} = \mathbf{h}\,(WF\$, P = P_{out}, x = x_{out})$$

$$s_{out} = \mathbf{s}\,(WF\$, P = P_{out}, x = x_{out})$$

$$\rho_{out} = \rho\,(WF\$, T = T_{out}, h = h_{out})$$

$$x_{out} = 1$$

END **VCC**Evaporator

Vapour Compression Cycle Compressor

MODULE **VCC**Compressor (WF\$, η_{COMP}, W, m, T_{in}, P_{in}, h_{in}, s_{in}, ρ_{in}, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out})

Outlet Conditions

$$h_{isen,out} = \mathbf{h}\,(WF\$, P = P_{out}, s = s_{in})$$

$$T_{out} = \mathbf{T}\,(WF\$, P = P_{out}, h = h_{out})$$

$$s_{out} = \mathbf{s}\,(WF\$, P = P_{out}, h = h_{out})$$

$$\rho_{out} = \rho\,(WF\$, T = T_{out}, h = h_{out})$$

Isentropic Efficiency

$$\eta_{COMP} = \frac{h_{isen,out} - h_{in}}{h_{out} - h_{in}}$$

Calculating Required Work

$$\dot{W} = \dot{m} \cdot (h_{out} - h_{in})$$

END **VCC**$_{Compressor}$

Vapour Compression Cycle Compressor Turbine Drive

MODULE VCC$_{Compressor,TD}$ (_MECH, m_s, $h_{TUR,in}$, $h_{TUR,out}$, w_{COMP})

Steam Mass Flow

$$w_{TUR,out} = \dot{m}_s \cdot (h_{TUR,in} - h_{TUR,out})$$

Energy Balance with Losses

$$w_{COMP} = w_{TUR,out} \cdot \eta_{MECH}$$

END VCC$_{Compressor,TD}$

Vapour Compression Cycle Condenser

MODULE VCC$_{Condenser}$ (WF\$, Q_{COND}, m_{FW}, m_{VCC}, ΔP_{FW}, $h_{FW,VCC,C,out}$, T_{in}, P_{in}, h_{in}, s_{in}, _in, T_{out}, P_{out}, h_{out}, s_{out}, ρ_{out}, $T_{FW,in}$, $P_{FW,in}$, $h_{FW,in}$, $s_{FW,in}$, $\rho_{FW,in}$, $T_{FW,out}$, $P_{FW,out}$, $h_{FW,out}$, $s_{FW,out}$, $\rho_{FW,out}$)

Energy Balance

$$\dot{Q}_{COND} = \dot{m}_{FW} \cdot (h_{FW,out} - h_{FW,in})$$

$$\dot{Q}_{COND} = \dot{m}_{VCC} \cdot (h_{in} - h_{out})$$

Performance

$$T_{COND,sat} = T_{FW,out}$$

$$T_{out} = T_{COND,sat}$$

Outlet Conditions (WF)

$$P_{out} = \mathbf{P}\,(\text{WF\$}, T = T_{out}, x = x_{out})$$

$$h_{out} = \mathbf{h}\,(\text{WF\$}, T = T_{out}, x = x_{out})$$

$$s_{out} = \mathbf{s}\,(\text{WF\$}, T = T_{out}, x = x_{out})$$

$$\rho_{out} = \rho\,(\text{WF\$}, T = T_{out}, x = x_{out})$$

$$x_{out} = 0$$

Outlet Conditions (FW)

$$T_{FW,out} = \mathbf{T}\,(\text{'Steam}_{IAPWS}\text{'}, h = h_{FW,out}, P = P_{FW,out})$$

$$h_{FW,out} = h_{FW,VCC,C,out}$$

$s_{FW,out} = \mathbf{s}\,(\text{'Steam}_{IAPWS}\text{'}, h = h_{FW,out}, P = P_{FW,out})$

$\rho_{FW,out} = \rho\,(\text{'Steam}_{IAPWS}\text{'}, h = h_{FW,out}, P = P_{FW,out})$

Pressure Relations

$P_{out} = P_{in}$

$P_{FW,out} = P_{FW,in} - \Delta P_{FW}$

END **VCC**$_{\text{Condenser}}$

---*Main Program*--

Rankine Cycle Input Parameters

Carnot Parameters

$T_H = 540$ [C]

$T_L = 20$ [C]

RCC Parameters

$T_R = 20$ [C]

$TTD_{COND} = 3$ [C]

$T_{COND,SUB} = 3$ [C]

CEP Parameters

$\eta_{CEP} = 0.85$ [-]

$\eta_{CEP,LOSSES} = 0.702$ [-]

LP FWH Parameters

$\Delta P_{LP,HTR} = 0.2$ [MPa]

$P_{LP,FD,HTR,BLEED} = 0.15$ [MPa]

DA Parameters

$P_{DA,BLEED} = 0.6$ [MPa]

BFP Parameters

$\eta_{BF,PUMP} = 0.85$ [-]

$\eta_{BF,PUMP,TUR} = 0.88$ [-]

$\eta_{BF,PUMP,MECH} = 0.95$ [-]

HP FWH Parameters

$\Delta P_{HP,HTR} = 0.2$ [MPa]

$P_{HP,FD,HTR,BLEED}$ = 2 [MPa]

Economiser Parameters

ΔP_{ECON} = 3.7 [MPa]

SH Parameters

$\Delta P_{HP,SH}$ = 1.2 [MPa]

$T_{HP,SH,out}$ = 545 [C]

HP turbine Inlet Valve Parameters

$\Delta P_{HPT,IV}$ = 0.1 [MPa]

CEP Parameters

$T_{HP,TUR,in}$ = 538 [C]

$P_{HP,TUR,in}$ = 16.8 [MPa]

$\eta_{HP,TUR}$ = 0.9 [-]

Reheaters Parameters

ΔP_{RH} = 1.2 [MPa]

$T_{RH,out}$ = 545 [C]

IP Turbine Inlet Valve Parameters

$\Delta P_{IPT,IV}$ = 0.1 [MPa]

IP Turbine Parameters

$\dot{m}_{tot}$ = 1 [kg/s]

$T_{IP,TUR,in}$ = 538 [C]

$P_{IP,TUR,in}$ = 3.4 [MPa]

$\eta_{IP,TUR}$ = 0.88 [-]

IP Turbine Parameters

$\eta_{LP,TUR}$ = 0.88 [-]

Vapour Compression Cycle Input Parameters

Working Fluid

WF$ = 'Ethanol'

η_{COMP} = 0.85

$\eta_{MECH,VCC,COMP}$ = 0.95

$HX_{eff,EVAP}$ = 0.85

$\Delta P_{COND,FW}$ = 0.2 [MPa]

$h_{FW,VCC,C,out}$ = 467.5 [kJ/kg]

RCC

$T_{CW,in} = T_L$

Call **Condenser**[$T_{CW,in}$, $P_{CW,RCOND,in}$, T_R, TTD_{COND}, $T_{COND,SUB}$, m_6, m_7, m_5, m_4, $P_{SAT,COND1}$, $P_{SAT,COND2}$, $P_{COND,out}$, $T_{COND,out}$, $h_{COND,out}$, $s_{COND,out}$, $\rho_{COND,out}$, $h_{TUR1,out}$, $h_{TUR2,out}$, q_{COND}, m_{CW}, $T_{CW,int}$, $h_{CW,int}$, $s_{CW,int}$, $\rho_{CW,int}$, $T_{CW,out}$, $h_{CW,out}$, $s_{CW,out}$, $\rho_{CW,out}$]

Extraction Pump

Call **ExtractionPump**[$P_{DA,BLEED}$, $\Delta P_{LP,HTR}$, η_{CEP}, $\eta_{CEP,LOSSES}$, $P_{COND,out}$, $P_{CEP,out}$, $T_{COND,out}$, $T_{CEP,out}$, $h_{COND,out}$, $h_{CEP,out}$, $s_{COND,out}$, $s_{CEP,out}$, $\rho_{COND,out}$, $\rho_{CEP,out}$, m_{tot}, m_2, m_3, W_{CEP}, $W_{CEP,CYCLE}$]

Dearator

Call **Dearator**[$P_{DA,BLEED}$, $T_{DA,out}$, P_{DA}, $h_{DA,out}$, $s_{DA,out}$, $\rho_{DA,out}$, $h_{LP,HTR,out}$, $h_{IP,TUR,out}$, $h_{HP,FD,HTR,DA,in}$, m_{tot}, m_2, m_3]

Boiler Feed Pump

Call **BoilerFeedPump**[$\eta_{BF,PUMP}$, $\eta_{BF,PUMP,MECH}$, $\eta_{BF,PUMP,TUR}$, m_{tot}, m_5, P_{DRUM}, $\Delta P_{HP,HTR}$, ΔP_{ECON}, $h_{DA,out}$, $s_{DA,out}$, $T_{HP,PUMP,out}$, $P_{HP,PUMP,out}$, $h_{HP,PUMP,out}$, $s_{HP,PUMP,out}$, $\rho_{HP,PUMP,out}$, $P_{SAT,COND2}$, $h_{IP,TUR,out}$, $s_{IP,TUR,out}$, $P_{BFPTD,out}$, $T_{BFPTD,out}$, $h_{BFPTD,out}$, $s_{BFPTD,out}$, $\rho_{BFPTD,out}$]

HP Feed Heaters

Call **HPFeedHeaters**[P_{DRUM}, ΔP_{ECON}, m_1, m_2, $T_{HP,PUMP,out}$, $P_{HP,PUMP,out}$, $h_{HP,PUMP,out}$, $s_{HP,PUMP,out}$, $T_{HP,FD,HTR,out}$, $P_{HP,FD,HTR,out}$, $h_{HP,FD,HTR,out}$, $s_{HP,FD,HTR,out}$, $\rho_{HP,FD,HTR,out}$, $T_{HP,FD,HTR,H,in}$, $P_{HP,FD,HTR,H,in}$, $h_{HP,FD,HTR,H,in}$, $s_{HP,FD,HTR,C}$, $T_{HP,FD,HTR,C}$, $P_{HP,FD,HTR,C}$, $h_{HP,FD,HTR,C}$, $s_{HP,FD,HTR,C}$, $\rho_{HP,FD,HTR,C}$, $P_{DA,BLEED}$, $T_{HP,FD,HTR,DA,in}$, $P_{HP,FD,HTR,DA,in}$, $h_{HP,FD,HTR,DA,in}$, $s_{HP,FD,HTR,DA,in}$, $\rho_{HP,FD,HTR,DA,in}$]

Economiser

Call **Economiser**[P_{DRUM}, $P_{ECON,out}$, $T_{ECON,out}$, $h_{ECON,out}$, $s_{ECON,out}$, $\rho_{ECON,out}$, $h_{HP,FD,HTR,out}$, q_{ECON}, m_1]

Drum

Call **Drum**[P_{DRUM}, $T_{DRUM,out}$, $h_{DRUM,out}$, $s_{DRUM,out}$, $\rho_{DRUM,out}$, $h_{ECON,out}$, q_{DRUM}, m_1]

HP Super Heaters

Call **HPSuperHeaters**[$\Delta P_{HP,SH}$, $T_{HP,SH,out}$, P_{DRUM}, $P_{HP,SH,out}$, $h_{HP,SH,out}$, $s_{HP,SH,out}$, $\rho_{HP,SH,out}$, $h_{DRUM,out}$, q_{SH}, m_1]

HP Super Heaters Attemperation

Call **HPAttemperation**[T_H, $P_{HP,SH,out}$, $T_{HP,SH,out}$, $h_{HP,SH,out}$, $s_{HP,SH,out}$, $P_{HP,PUMP,out}$, $T_{HP,PUMP,out}$, $h_{HP,PUMP,out}$, $s_{HP,PUMP,out}$, m_{tot}, m_1, m_8, $P_{HPT,IP,in}$, $T_{HPT,IP,in}$, $h_{HPT,IP,in}$, $s_{HPT,IP,in}$, $\rho_{HPT,IP,in}$]

HP Turbine Inlet Valve with Upstream Piping

Call **HPTurbineInletValve**[m_1, m_8, $\Delta P_{HPT,IV}$, $P_{HPT,IV,in}$, $T_{HPT,IV,in}$, $h_{HPT,IV,in}$, $s_{HPT,IV,in}$, $\rho_{HPT,IV,in}$, $P_{HP,TUR,in}$, $T_{HP,TUR,in}$, $h_{HP,TUR,in}$, $s_{HP,TUR,in}$, $\rho_{HP,TUR,in}$, $q_{HPT,IP,loss}$, $P_{HPT,IP,in}$, $h_{HPT,IP,in}$]

HP Turbine

Call **HPTurbine**[m_1, m_8, $\eta_{HP,TUR}$, $h_{HP,TUR,in}$, $s_{HP,TUR,in}$, $P_{HP,TUR,out}$, $T_{HP,TUR,out}$, $h_{HP,TUR,out}$, $s_{HP,TUR,out}$, $\rho_{HP,TUR,out}$, W_{HPT}]

Reheaters

Call **Reheaters**[ΔP_{RH}, $T_{RH,out}$, $P_{HP,TUR,out}$, $P_{RH,out}$, $h_{RH,out}$, $s_{RH,out}$, $\rho_{RH,out}$, $h_{HP,TUR,out}$, q_{RH}, m_1, m_8]

Reheaters Attemperation

Call **RHAttemperation**[T_H, $P_{RH,out}$, $T_{RH,out}$, $h_{RH,out}$, $s_{RH,out}$, $P_{HP,PUMP,out}$, $T_{HP,PUMP,out}$, $h_{HP,PUMP,out}$, $s_{HP,PUMP,out}$, m_{tot}, m_1, m_8, m_9, $P_{IPT,IP,in}$, $T_{IPT,IP,in}$, $h_{IPT,IP,in}$, $s_{IPT,IP,in}$, $\rho_{IPT,IP,in}$]

IP Turbine Inlet Valve with Upstream Piping

Call **IPTurbineInletValve**[m_{tot}, $\Delta P_{IPT,IV}$, $P_{IPT,IV,in}$, $T_{IPT,IV,in}$, $h_{IPT,IV,in}$, $s_{IPT,IV,in}$, $\rho_{IPT,IV,in}$, $P_{IP,TUR,in}$, $T_{IP,TUR,in}$, $h_{IP,TUR,in}$, $s_{IP,TUR,in}$, $\rho_{IP,TUR,in}$, $q_{IPT,IP,loss}$, $P_{IPT,IP,in}$, $h_{IPT,IP,in}$]

IP Turbine with HP Feedwater Bleed

Call **IPTurbine**[m_{tot}, m_2, $\eta_{IP,TUR}$, k_{poly}, $P_{HP,FD,HTR,BLEED}$, $P_{DA,BLEED}$, $P_{IP,TUR,in}$, $T_{IP,TUR,in}$, $h_{IP,TUR,in}$, $s_{IP,TUR,in}$, $\rho_{IP,TUR,in}$, $P_{HP,FD,HTR,H,in}$, $T_{HP,FD,HTR,H,in}$, $h_{HP,FD,HTR,H,in}$, $s_{HP,FD,HTR,H,in}$, $\rho_{HP,FD,HTR,H,in}$, $P_{IP,TUR,out}$, $T_{IP,TUR,out}$, $h_{IP,TUR,out}$, $s_{IP,TUR,out}$, $\rho_{IP,TUR,out}$, w_{IPT}]

LP Turbines

Call **LPTurbines**[$\eta_{P,TUR}$, $P_{IP,TUR,out}$, $h_{IP,TUR,out}$, $s_{IP,TUR,out}$, $P_{SAT,COND1}$, $T_{TUR1,out}$, $h_{TUR1,out}$, $s_{TUR1,out}$, $\rho_{TUR1,out}$, $P_{SAT,COND2}$, $T_{TUR2,out}$, $h_{TUR2,out}$, $s_{TUR2,out}$, $\rho_{TUR2,out}$, m_{tot}, m_2, m_3, m_4, m_5, m_6, m_7, w_{LPT}]

The Vapour Compression Cycle Expansion Valve

Call $\mathbf{VCC}_{\text{Expansion,Valve}}$[WF\$, $T_{CC,4}$, $P_{CC,4}$, $h_{CC,4}$, $s_{CC,4}$, $\rho_{CC,4}$, $T_{CC,1}$, $P_{CC,1}$, $h_{CC,1}$, $s_{CC,1}$, $\rho_{CC,1}$]

The Vapour Compression Cycle Evaporator

Call $\mathbf{VCC}_{\text{Evaporator}}$[WF\$, Q_{EVAP}, m_{CW}, m_{CC}, $HX_{eff,EVAP}$, $T_{CC,1}$, $P_{CC,1}$, $h_{CC,1}$, $s_{CC,1}$, _CC,1, $T_{CC,2}$, $P_{CC,2}$, $h_{CC,2}$, $s_{CC,2}$, $\rho_{CC,2}$, $T_{CW,in}$, $P_{CW,in}$, $h_{CW,in}$, $s_{CW,in}$, $\rho_{CW,in}$, $T_{CW,RCOND,in}$, $P_{CW,RCOND,in}$, $h_{CW,RCOND,in}$, $s_{CW,RCOND,in}$, $\rho_{CW,RCOND,in}$]

The Vapour Compression Cycle Compressor

Call $\mathbf{VCC}_{\text{Compressor}}$[WF\$, η_{COMP}, w_{COMP}, m_{CC}, $T_{CC,2}$, $P_{CC,2}$, $h_{CC,2}$, $s_{CC,2}$, $\rho_{CC,2}$, $T_{CC,3}$, $P_{CC,3}$, $h_{CC,3}$, $s_{CC,3}$, $\rho_{CC,3}$]

Steam Driven Compressor

$$\text{Call } \mathbf{VCC}_{\text{Compressor,TD}}\left(\eta_{MECH,VCC,COMP},\ \dot{m}_4,\ h_{IP,TUR,out},\ h_{TUR2,out},\ w_{COMP}\right)$$

$$w_{COMP,CYCLE} = 0$$

OR

Electric Driven Compressor

$$w_{COMP,CYCLE} = \frac{w_{COMP}}{\eta_{EXTR,PUMP,LOSSES}}$$

$$\dot{m}_4 = 0$$

The Vapour Compression Cycle Condenser

Call $\mathbf{VCC}_{\text{Condenser}}$[WF\$, Q_{COND}, $m_{tot} - m_2 - m_3$, m_{CC}, $\Delta P_{COND,FW}$, $h_{FW,VCC,C,out}$, $T_{CC,3}$, $P_{CC,3}$, $h_{CC,3}$, $s_{CC,3}$, $\rho_{CC,3}$, $T_{CC,4}$, $P_{CC,4}$, $h_{CC,4}$, $s_{CC,4}$, $\rho_{CC,4}$, $T_{EXTR,PUMP,out}$, $P_{EXTR,PUMP,out}$, $h_{EXTR,PUMP,out}$, $s_{EXTR,PUMP,out}$, $\rho_{EXTR,PUMP,out}$, $T_{LP,HTR,out}$, $P_{LP,HTR,out}$, $h_{LP,HTR,out}$, $s_{LP,HTR,out}$, $\rho_{LP,HTR,out}$]

Efficiency Calculations

$$q_{loss} = \dot{q}_{HPT,IP,loss} + \dot{q}_{IPT,IP,loss} + q_{COND}$$

$$q_{in} = q_{ECON} + q_{DRUM} + q_{SH} + q_{RH}$$

$$w_{in} = w_{CEP,CYCLE}$$

$$w_{out} = w_{HPT} + w_{IPT} + w_{LPT}$$

$$w_{net} = w_{out} - w_{in}$$

$$\eta_{tot} = \frac{w_{net}}{q_{in}}$$

$$COP_{CC,h} = \frac{h_{CC,3} - h_{CC,4}}{h_{CC,3} - h_{CC,2}}$$

$$COP_{CC,c} = \frac{h_{CC,2} - h_{CC,1}}{h_{CC,3} - h_{CC,2}}$$

---*Graph Nodes*---

Node 1 - CEP Inlet

$T_{RC,1} = T_{COND,out}$

$P_{RC,1} = P_{COND,out}$

$h_{RC,1} = h_{COND,out}$

$s_{RC,1} = s_{COND,out}$

$\rho_{RC,1} = \rho_{COND,out}$

Node 2 - CEP Outlet

$T_{RC,2} = T_{EXTR,PUMP,out}$

$P_{RC,2} = P_{EXTR,PUMP,out}$

$h_{RC,2} = h_{EXTR,PUMP,out}$

$s_{RC,2} = s_{EXTR,PUMP,out}$

$\rho_{RC,2} = \rho_{EXTR,PUMP,out}$

Node 3 – Vapour Compression Cycle Condenser - FW Outlet

$T_{RC,3} = T_{LP,HTR,out}$

$P_{RC,3} = P_{LP,HTR,out}$

$h_{RC,3} = h_{LP,HTR,out}$

$s_{RC,3} = s_{LP,HTR,out}$

$\rho_{RC,3} = \rho_{LP,HTR,out}$

Node 4 - Dearator Outlet

$T_{RC,4} = T_{DA,out}$

$P_{RC,4} = P_{DA}$

$h_{RC,4} = h_{DA,out}$

$s_{RC,4} = s_{DA,out}$

$\rho_{RC,4} = \rho_{DA,out}$

Node 5 - BFP Outlet

$T_{RC,5} = T_{HP,PUMP,out}$

$P_{RC,5} = P_{HP,PUMP,out}$

$h_{RC,5} = h_{HP,PUMP,out}$

$s_{RC,5} = s_{HP,PUMP,out}$

$\rho_{RC,5} = \rho_{HP,PUMP,out}$

Node 6 - HP FWH - FW Outlet

$T_{RC,6} = T_{HP,FD,HTR,out}$

$P_{RC,6} = P_{HP,FD,HTR,out}$

$h_{RC,6} = h_{HP,FD,HTR,out}$

$s_{RC,6} = s_{HP,FD,HTR,out}$

$\rho_{RC,6} = \rho_{HP,FD,HTR,out}$

Node 7 - Economizer Outlet

$T_{RC,7} = T_{ECON,out}$

$P_{RC,7} = P_{ECON,out}$

$h_{RC,7} = h_{ECON,out}$

$s_{RC,7} = s_{ECON,out}$

$\rho_{RC,7} = \rho_{ECON,out}$

Node 8 - Drum Outlet

$T_{RC,8} = T_{DRUM,out}$

$P_{RC,8} = P_{DRUM}$

$h_{RC,8} = h_{DRUM,out}$

$s_{RC,8} = s_{DRUM,out}$

$\rho_{RC,8} = \rho_{DRUM,out}$

Node 9 - HP Super Heaters Outlet (Before accounting for attemperation)

$T_{RC,9} = T_{HP,SH,out}$

$P_{RC,9} = P_{HP,SH,out}$

$h_{RC,9} = h_{HP,SH,out}$

$s_{RC,9} = s_{HP,SH,out}$

$\rho_{RC,9} = \rho_{HP,SH,out}$

Node 10 - HP Super Heaters Outlet (After accounting for attemperation)

$T_{RC,10} = T_{HPT,IP,in}$

$P_{RC,10} = P_{HPT,IP,in}$

$h_{RC,10} = h_{HPT,IP,in}$

$s_{RC,10} = s_{HPT,IP,in}$

$\rho_{RC,10} = \rho_{HPT,IP,in}$

Node 11 – HP turbine Inlet Valve Inlet

$T_{RC,11} = T_{HPT,IV,in}$

$P_{RC,11} = P_{HPT,IV,in}$

$h_{RC,11} = h_{HPT,IV,in}$

$s_{RC,11} = s_{HPT,IV,in}$

$\rho_{RC,11} = \rho_{HPT,IV,in}$

Node 12 - HP turbine Inlet

$T_{RC,12} = T_{HP,TUR,in}$

$P_{RC,12} = P_{HP,TUR,in}$

$h_{RC,12} = h_{HP,TUR,in}$

$s_{RC,12} = s_{HP,TUR,in}$

$\rho_{RC,12} = \rho_{HP,TUR,in}$

Node 13 - Reheaters Inlet

$T_{RC,13} = T_{HP,TUR,out}$

$P_{RC,13} = P_{HP,TUR,out}$

$h_{RC,13} = h_{HP,TUR,out}$

$s_{RC,13} = s_{HP,TUR,out}$

$\rho_{RC,13} = \rho_{HP,TUR,out}$

Node 14 - Reheaters Outlet (Before accounting for attemperation)

$T_{RC,14} = T_{RH,out}$

$P_{RC,14} = P_{RH,out}$

$h_{RC,14} = h_{RH,out}$

$s_{RC,14} = s_{RH,out}$

$\rho_{RC,14} = \rho_{RH,out}$

Node 15 - Reheaters Outlet (After accounting for attemperation)

$T_{RC,15} = T_{IPT,IP,in}$

$P_{RC,15} = P_{IPT,IP,in}$

$h_{RC,15} = h_{IPT,IP,in}$

$s_{RC,15} = s_{IPT,IP,in}$

$\rho_{RC,15} = \rho_{IPT,IP,in}$

Node 16 – IP Turbine Inlet Valve Inlet

$T_{RC,16} = T_{IPT,IV,in}$

$P_{RC,16} = P_{IPT,IV,in}$

$h_{RC,16} = h_{IPT,IV,in}$

$s_{RC,16} = s_{IPT,IV,in}$

$\rho_{RC,16} = \rho_{IPT,IV,in}$

Node 17 – IP Turbine Inlet

$T_{RC,17} = T_{IP,TUR,in}$

$P_{RC,17} = P_{IP,TUR,in}$

$h_{RC,17} = h_{IP,TUR,in}$

$s_{RC,17} = s_{IP,TUR,in}$

$\rho_{RC,17} = \rho_{IP,TUR,in}$

Node 18 – IP Turbine Bleed Point

$T_{RC,18} = T_{HP,FD,HTR,H,in}$

$P_{RC,18} = P_{HP,FD,HTR,H,in}$

$h_{RC,18} = h_{HP,FD,HTR,H,in}$

$s_{RC,18} = s_{HP,FD,HTR,H,in}$

$\rho_{RC,18} = \rho_{HP,FD,HTR,H,in}$

Node 19 – IP Turbine Outlet

$T_{RC,19} = T_{IP,TUR,out}$

$P_{RC,19} = P_{IP,TUR,out}$

$h_{RC,19} = h_{IP,TUR,out}$

$s_{RC,19} = s_{IP,TUR,out}$

$\rho_{RC,19} = \rho_{IP,TUR,out}$

Node 20 - LP Turbine 1 Outlet

$T_{RC,20} = T_{TUR1,out}$

$P_{RC,20} = P_{SAT,COND1}$

$h_{RC,20} = h_{TUR1,out}$

$s_{RC,20} = s_{TUR1,out}$

$\rho_{RC,20} = \rho_{TUR1,out}$

Node 21 - LP Turbine 2 Outlet

$T_{RC,21} = T_{TUR2,out}$

$P_{RC,21} = P_{SAT,COND2}$

$h_{RC,21} = h_{TUR2,out}$

$s_{RC,21} = s_{TUR2,out}$

$\rho_{RC,21} = \rho_{TUR2,out}$

Node 22 - BFPTD Outlet

$T_{RC,22} = T_{TUR2,out}$

$P_{RC,22} = P_{SAT,COND2}$

$h_{RC,22} = h_{TUR2,out}$

$s_{RC,22} = s_{TUR2,out}$

$\rho_{RC,22} = \rho_{TUR2,out}$

Node 23 - CTD Outlet

$T_{RC,23} = T_{TUR2,out}$

$P_{RC,23} = P_{SAT,COND2}$

$h_{RC,23} = h_{TUR2,out}$

$s_{RC,23} = s_{TUR2,out}$

$\rho_{RC,23} = \rho_{TUR2,out}$

Node 24 - HP Feed Heater Condensate

$T_{RC,24} = T_{HP,FD,HTR,C}$

$P_{RC,24} = P_{HP,FD,HTR,C}$

$h_{RC,24} = h_{HP,FD,HTR,C}$

$s_{RC,24} = s_{HP,FD,HTR,C}$

$\rho_{RC,24} = \rho_{HP,FD,HTR,C}$

Node 25 - HP Feed Heater Bleed Steam Dearator Inlet

$T_{RC,25} = T_{HP,FD,HTR,DA,in}$

$P_{RC,25} = P_{HP,FD,HTR,DA,in}$

$h_{RC,25} = h_{HP,FD,HTR,DA,in}$

$s_{RC,25} = s_{HP,FD,HTR,DA,in}$

$\rho_{RC,25} = \rho_{HP,FD,HTR,DA,in}$

Node 26 - BFPTD Outlet

$T_{RC,26} = T_{CW,in}$

$P_{RC,26} = P_{CW,in}$

$h_{RC,26} = h_{CW,in}$

$s_{RC,26} = s_{CW,in}$

$\rho_{RC,26} = \rho_{CW,in}$

Node 27 - RCC CW Inlet

$T_{RC,27} = T_{CW,RCOND,in}$

$P_{RC,27} = P_{CW,RCOND,in}$

$h_{RC,27} = h_{CW,RCOND,in}$

$s_{RC,27} = s_{CW,RCOND,in}$

$\rho_{RC,27} = \rho_{CW,RCOND,in}$

Node 28 - RCC CW Intermediate

$T_{RC,28} = T_{CW,int}$

$h_{RC,28} = h_{CW,int}$

$s_{RC,28} = s_{CW,int}$

$\rho_{RC,28} = \rho_{CW,int}$

Node 29 - CW Outlet

$T_{RC,29} = T_{CW,out}$

$P_{RC,29} = P_{CW,in}$

$s_{RC,29} = s_{CW,out}$

$\rho_{RC,29} = \rho_{CW,out}$

Node 30 -

$T_{30} = T_{CW,out}$

$P_{30} = P_{CW,in}$

$h_{30} = h_{CW,out}$

$s_{30} = s_{CW,out}$

$\rho_{30} = \rho_{CW,out}$

8.3.2 Results of the Cycle with a Steam Turbine Driven Compressor

SOLUTION

Unit Settings: [kJ]/[C]/[MPa]/[kg]/[degrees]

$COP_{CC,c}$ = 2.023	$COP_{CC,h}$ = 3.023	$\Delta P_{COND,FW}$ = 0.2 [MPa]
ΔP_{ECON} = 3.7 [MPa]	$\Delta P_{HPT,IV}$ = 0.1 [MPa]	$\Delta P_{HP,HTR}$ = 0.2 [MPa]
$\Delta P_{HP,SH}$ = 1.2 [MPa]	$\Delta P_{IPT,IV}$ = 0.1 [MPa]	ΔP_{RH} = 1.2 [MPa]
$\eta_{BF,PUMP}$ = 0.85 [-]	$\eta_{BF,PUMP,MECH}$ = 0.95 [-]	$\eta_{BF,PUMP,TUR}$ = 0.88 [-]
η_{COMP} = 0.85	$\eta_{EXTR,PUMP}$ = 0.85 [-]	$\eta_{EXTR,PUMP,LOSSES}$ = 0.702 [-]
$\eta_{HP,TUR}$ = 0.9 [-]	$\eta_{IP,TUR}$ = 0.88 [-]	$\eta_{LP,TUR}$ = 0.85 [-]
$\eta_{MECH,VCC,COMP}$ = 0.95 [-]	η_{tot} = 0.4106 [-]	$HX_{eff,EVAP}$ = 0.85[-]
$h_{BFPTD,out}$ = 2403 [kJ/kg]	$h_{COND,out}$ = 139.5 [kJ/kg]	$h_{CW,in}$ = 84.01 [kJ/kg]
$h_{CW,int}$ = 108.1 [kJ/kg]	$h_{CW,out}$ = 159.5 [kJ/kg]	$h_{CW,RCOND,in}$ = 75.9 [kJ/kg]
$h_{DA,out}$ = 670.4 [kJ/kg]	$h_{DRUM,out}$ = 2506 [kJ/kg]	$h_{ECON,out}$ = 1737 [kJ/kg]
$h_{EXTR,PUMP,out}$ = 140.5 [kJ/kg]	$h_{FW,VCC,C,out}$ = 467.5 [kJ/kg]	$h_{HPT,IP,in}$ = 3402 [kJ/kg]
$h_{HPT,IV,in}$ = 3397 [kJ/kg]	$h_{HP,FD,HTR,C}$ = 908.5 [kJ/kg]	$h_{HP,FD,HTR,DA,in}$ = 908.5 [kJ/kg]
$h_{HP,FD,HTR,H,in}$ = 3377 [kJ/kg]	$h_{HP,FD,HTR,out}$ = 915.8 [kJ/kg]	$h_{HP,PUMP,out}$ = 697.9 [kJ/kg]
$h_{HP,SH,out}$ = 3416 [kJ/kg]	$h_{HP,TUR,in}$ = 3397 [kJ/kg]	$h_{HP,TUR,out}$ = 3066 [kJ/kg]
$h_{IPT,IP,in}$ = 3542 [kJ/kg]	$h_{IPT,IV,in}$ = 3539 [kJ/kg]	$h_{IP,TUR,in}$ = 3539 [kJ/kg]
$h_{IP,TUR,out}$ = 3074 [kJ/kg]	$h_{LP,HTR,out}$ = 467.5 [kJ/kg]	$h_{RH,out}$ = 3554 [kJ/kg]
$h_{TUR1,out}$ = 2354 [kJ/kg]	$h_{TUR2,out}$ = 2426 [kJ/kg]	m_1 = 0.9909 [kg/s]
m_2 = 0.08745 [kg/s]	m_3 = 0.06304 [kg/s]	m_4 = 0.1492 [kg/s]
m_5 = 0.04321 [kg/s]	m_6 = 0.3291 [kg/s]	m_7 = 0.328 [kg/s]
m_8 = 0.005175 [kg/s]	m_9 = 0.003973 [kg/s]	m_{CC} = 0.2853
m_{CW} = 22.94 [kg/s]	m_{tot} = 1 [kg/s]	$P_{BFPTD,out}$ = 0.007814 [MPa]
$P_{COND,out}$ = 0.007814 [MPa]	$P_{CW,in}$ = 0.1 [MPa]	$P_{CW,RCOND,in}$ = 0.1 [MPa]
P_{DA} = 0.6 [MPa]	$P_{DA,BLEED}$ = 0.6 [MPa]	P_{DRUM} = 18.1 [MPa]
$P_{ECON,out}$ = 18.1 [MPa]	$P_{EXTR,PUMP,out}$ = 0.8 [MPa]	$P_{HPT,IP,in}$ = 16.9 [MPa]
$P_{HPT,IV,in}$ = 16.9 [MPa]	$P_{HP,FD,HTR,BLEED}$ = 2 [MPa]	$P_{HP,FD,HTR,C}$ = 2 [MPa]
$P_{HP,FD,HTR,DA,in}$ = 0.6 [MPa]	$P_{HP,FD,HTR,H,in}$ = 2 [MPa]	$P_{HP,FD,HTR,out}$ = 21.8 [MPa]
$P_{HP,PUMP,out}$ = 22 [MPa]	$P_{HP,SH,out}$ = 16.9 [MPa]	$P_{HP,TUR,in}$ = 16.8 [MPa]
$P_{HP,TUR,out}$ = 4.7 [MPa]	$P_{IPT,IP,in}$ = 3.5 [MPa]	$P_{IPT,IV,in}$ = 3.5 [MPa]
$P_{IP,TUR,in}$ = 3.4 [MPa]	$P_{IP,TUR,out}$ = 0.6 [MPa]	$P_{LP,HTR,out}$ = 0.6 [MPa]
$P_{RH,out}$ = 3.5 [MPa]	$P_{SAT,COND1}$ = 0.003955 [MPa]	$P_{SAT,COND2}$ = 0.007814 [MPa]
q_{COND} = 1918 [kW]	Q_{COND} = 277.8	Q_{EVAP} = 185.9
$q_{HPT,IP,loss}$ = 4.518 [kW]	$q_{IPT,IP,loss}$ = 3.562 [kW]	q_{DRUM} = 762.4 [kW]
q_{ECON} = 813.2 [kW]	q_{in} = 2964 [kW]	q_{loss} = 1926 [kW]
q_{RH} = 486.1 [kW]	q_{SH} = 901.8 [kW]	$\rho_{BFPTD,out}$ = 0.05823 [kg/m3]
$\rho_{COND,out}$ = 994.6 [kg/m3]	$\rho_{CW,in}$ = 998.2 [kg/m3]	$\rho_{CW,int}$ = 996.8 [kg/m3]
$\rho_{CW,out}$ = 992.9 [kg/m3]	$\rho_{CW,RCOND,in}$ = 998.6 [kg/m3]	$\rho_{DA,out}$ = 908.6 [kg/m3]

$\rho_{DRUM,out}$ = 134.8 [kg/m3]
$\rho_{ECON,out}$ = 541.2 [kg/m3]
$\rho_{EXTR,PUMP,out}$ = 994.9 [kg/m3]
$\rho_{HPT,IP,in}$ = 50.75 [kg/m3]
$\rho_{HPT,IV,in}$ = 50.91 [kg/m3]
$\rho_{HP,FD,HTR,C}$ = 849.8 [kg/m3]
$\rho_{HP,FD,HTR,DA,in}$ = 27.03 [kg/m3]
$\rho_{HP,FD,HTR,H,in}$ = 6.038 [kg/m3]
$\rho_{HP,FD,HTR,out}$ = 865.3 [kg/m3]
$\rho_{HP,PUMP,out}$ = 917.6 [kg/m3]
$\rho_{HP,SH,out}$ = 50.27 [kg/m3]
$\rho_{HP,TUR,in}$ = 50.61 [kg/m3]
$\rho_{HP,TUR,out}$ = 18.14 [kg/m3]
$\rho_{IPT,IP,in}$ = 9.536 [kg/m3]
$\rho_{IPT,IV,in}$ = 9.556 [kg/m3]
$\rho_{IP,TUR,in}$ = 9.282 [kg/m3]
$\rho_{IP,TUR,out}$ = 2.278 [kg/m3]
$\rho_{LP,HTR,out}$ = 950.1 [kg/m3]
$\rho_{RH,out}$ = 9.472 [kg/m3]
$\rho_{TUR1,out}$ = 0.03098 [kg/m3]
$\rho_{TUR2,out}$ = 0.05763 [kg/m3]
$s_{BFPTD,out}$ = 7.686 [kJ/kg-K]
$s_{COND,out}$ = 0.482 [kJ/kg-K]
$s_{CW,in}$ = 0.2965 [kJ/kg-K]
$s_{CW,int}$ = 0.3779 [kJ/kg-K]
$s_{CW,out}$ = 0.5464 [kJ/kg-K]
$s_{CW,RCOND,in}$ = 0.2687 [kJ/kg-K]
$s_{DA,out}$ = 1.931 [kJ/kg-K]
$s_{DRUM,out}$ = 5.099 [kJ/kg-K]
$s_{ECON,out}$ = 3.879 [kJ/kg-K]
$s_{EXTR,PUMP,out}$ = 0.4824 [kJ/kg-K]
$s_{HPT,IP,in}$ = 6.414 [kJ/kg-K]
$s_{HPT,IV,in}$ = 6.409 [kJ/kg-K]
$s_{HP,FD,HTR,C}$ = 2.447 [kJ/kg-K]
$s_{HP,FD,HTR,DA,in}$ = 2.482 [kJ/kg-K]
$s_{HP,FD,HTR,H,in}$ = 7.312 [kJ/kg-K]
$s_{HP,FD,HTR,out}$ = 2.414 [kJ/kg-K]
$s_{HP,PUMP,out}$ = 1.94 [kJ/kg-K]
$s_{HP,SH,out}$ = 6.432 [kJ/kg-K]
$s_{HP,TUR,in}$ = 6.411 [kJ/kg-K]
$s_{HP,TUR,out}$ = 6.472 [kJ/kg-K]
$s_{IPT,IP,in}$ = 7.274 [kJ/kg-K]
$s_{IPT,IV,in}$ = 7.269 [kJ/kg-K]
$s_{IP,TUR,in}$ = 7.282 [kJ/kg-K]
$s_{IP,TUR,out}$ = 7.395 [kJ/kg-K]
$s_{LP,HTR,out}$ = 1.433 [kJ/kg-K]
$s_{RH,out}$ = 7.288 [kJ/kg-K]
$s_{TUR1,out}$ = 7.816 [kJ/kg-K]
$s_{TUR2,out}$ = 7.759 [kJ/kg-K]
TTD_{COND} = 3 [C]
$T_{BFPTD,out}$ = 41.06 [C]
$T_{COND,out}$ = 33.3 [C]
$T_{COND,SUB}$ = 3 [C]
$T_{CW,in}$ = 20 [C]
$T_{CW,int}$ = 25.76 [C]
$T_{CW,out}$ = 38.06 [C]
$T_{CW,RCOND,in}$ = 18.06 [C]
$T_{DA,out}$ = 158.8 [C]
$T_{DRUM,out}$ = 357.5 [C]
$T_{ECON,out}$ = 357.5 [C]
$T_{EXTR,PUMP,out}$ = 33.35 [C]
T_H = 540 [C]
$T_{HPT,IP,in}$ = 540 [C]
$T_{HPT,IV,in}$ = 538.4 [C]
$T_{HP,FD,HTR,C}$ = 212.4 [C]
$T_{HP,FD,HTR,DA,in}$ = 158.8 [C]
$T_{HP,FD,HTR,H,in}$ = 458.5 [C]
$T_{HP,FD,HTR,out}$ = 212.4 [C]
$T_{HP,PUMP,out}$ = 162.3 [C]
$T_{HP,SH,out}$ = 545 [C]
$T_{HP,TUR,in}$ = 538 [C]
$T_{HP,TUR,out}$ = 345.8 [C]
$T_{IPT,IP,in}$ = 540 [C]
$T_{IPT,IV,in}$ = 538.4 [C]
$T_{IP,TUR,in}$ = 538 [C]
$T_{IP,TUR,out}$ = 305.8 [C]
T_L = 20 [C]
$T_{LP,HTR,out}$ = 111.4 [C]
T_R = 20 [C]
$T_{RH,out}$ = 545 [C]
$T_{TUR1,out}$ = 28.76 [C]
$T_{TUR2,out}$ = 41.06 [C]
WF$ = 'Ethanol'
w_{COMP} = 91.89
$w_{COMP,CYCLE}$ = 0 [kW]
$w_{EXTR,PUMP}$ = 1.134 [kW]
$w_{EXTR,PUMP,CYCLE}$ = 1.615 [kW]
w_{HPT} = 330.5 [kW]
w_{in} = 1.615 [kW]
w_{IPT} = 438.3 [kW]
w_{LPT} = 449.8 [kW]
w_{net} = 1217 [kW]
w_{out} = 1219 [kW]
$x_{EXON,out}$ = 0

Local variables in Module ExtractionPump\1 CALL ExtractionPump

$\Delta P_{LP,HTR}$ = 0.2 [MPa]
η_{LOSSES} = 0.702 [-]
η_{PUMP} = 0.85 [-]
h_{in} = 139.5 [kJ/kg]
$h_{isen,out}$ = 140.3 [kJ/kg]
h_{out} = 140.5 [kJ/kg]
m = 0.8495 [kg/s]
m_2 = 0.08745 [kg/s]
m_3 = 0.06304 [kg/s]
m_{tot} = 1 [kg/s]
$P_{DA,BLEED}$ = 0.6 [MPa]
P_{in} = 0.007814 [MPa]
P_{out} = 0.8 [MPa]
ρ_{in} = 994.6 [kg/m3]
ρ_{out} = 994.9 [kg/m3]
s_{in} = 0.482 [kJ/kg-K]
$s_{isen,out}$ = 0.482 [kJ/kg-K]
s_{out} = 0.4824 [kJ/kg-K]

T_{in} = 33.3 [C]	T_{out} = 33.35 [C]	w_{PUMP} = 1.134 [kW]
$w_{PUMP,CYCLE}$ = 1.615 [kW]		

Local variables in Module LPFeedHeaters\1 CALL LPFeedHeaters

$h_{H,C}$ = 467.1 [kJ/kg]	$h_{H,in}$ = 2811 [kJ/kg]	$h_{H,out}$ = 467.1 [kJ/kg]
h_{in} = 150 [kJ/kg]	h_{out} = 467.5 [kJ/kg]	m_2 = 0.08745 [kg/s]
m_3 = 0.06305 [kg/s]	m_4 = 0.115 [kg/s]	m_{tot} = 1 [kg/s]
P_{COND2} = 0.008651 [MPa]	P_{DA} = 0.6 [MPa]	$P_{H,C}$ = 0.15 [MPa]
$P_{H,in}$ = 0.15 [MPa]	$P_{H,out}$ = 0.008651 [MPa]	P_{out} = 0.6 [MPa]
$\rho_{H,C}$ = 949.9 [kg/m^3]	$\rho_{H,out}$ = 0.4968 [kg/m^3]	ρ_{out} = 950.1 [kg/m^3]
$s_{H,C}$ = 1.434 [kJ/kg-K]	$s_{H,out}$ = 1.52 [kJ/kg-K]	s_{out} = 1.433 [kJ/kg-K]
$T_{H,C}$ = 111.3 [C]	$T_{H,out}$ = 43 [C]	T_{out} = 111.3 [C]

Local variables in Module Dearator\1 CALL Dearator

$h_{C,FD,HTR,out}$ = 908.5 [kJ/kg]	$h_{H,in}$ = 3074 [kJ/kg]	h_{in} = 467.5 [kJ/kg]
h_{out} = 670.4 [kJ/kg]	m_2 = 0.08745 [kg/s]	m_3 = 0.06305 [kg/s]
m_{tot} = 1 [kg/s]	P_{DA} = 0.6 [MPa]	$P_{DA,BLEED}$ = 0.6 [MPa]
ρ_{out} = 908.6 [kg/m^3]	s_{out} = 1.931 [kJ/kg-K]	T_{out} = 158.8 [C]

Local variables in Module BoilerFeedPump\1 CALL BoilerFeedPump

ΔP_{ECON} = 3.7 [MPa]	$\Delta P_{HP,HTR}$ = 0.2 [MPa]	η_{MECH} = 0.95 [-]
η_{PUMP} = 0.85 [-]	η_{TUR} = 0.88 [-]	h_{in} = 670.4 [kJ/kg]
$h_{isen,out}$ = 693.8 [kJ/kg]	h_{out} = 697.9 [kJ/kg]	$h_{TUR,in}$ = 3074 [kJ/kg]
$h_{TUR,isen,out}$ = 2324 [kJ/kg]	$h_{TUR,out}$ = 2414 [kJ/kg]	m_5 = 0.04397 [kg/s]
m_{tot} = 1 [kg/s]	P_{COND2} = 0.008651 [MPa]	P_{DRUM} = 18.1 [MPa]
P_{out} = 22 [MPa]	$P_{TUR,out}$ = 0.008651 [MPa]	ρ_{out} = 917.6 [kg/m^3]
$\rho_{TUR,out}$ = 0.06386 [kg/m^3]	s_{in} = 1.931 [kJ/kg-K]	$s_{isen,out}$ = 1.931 [kJ/kg-K]
s_{out} = 1.94 [kJ/kg-K]	$s_{TUR,in}$ = 7.395 [kJ/kg-K]	$s_{TUR,isen,out}$ = 7.395 [kJ/kg-K]
$s_{TUR,out}$ = 7.679 [kJ/kg-K]	T_{out} = 162.3 [C]	$T_{TUR,out}$ = 43 [C]
w_{PUMP} = 27.56 [kW]	$w_{TUR,out}$ = 29.01 [kW]	

Local variables in Module HPFeedHeaters\1 CALL HPFeedHeaters

ΔP_{ECON} = 3.7 [MPa]	$h_{H,C}$ = 908.5 [kJ/kg]	$h_{H,in}$ = 3377 [kJ/kg]
$h_{H,out}$ = 908.5 [kJ/kg]	h_{in} = 697.9 [kJ/kg]	h_{out} = 915.8 [kJ/kg]
m_1 = 0.9909 [kg/s]	m_2 = 0.08745 [kg/s]	P_{DA} = 0.6 [MPa]
P_{DRUM} = 18.1 [MPa]	$P_{H,C}$ = 2 [MPa]	$P_{H,in}$ = 2 [MPa]
$P_{H,out}$ = 0.6 [MPa]	P_{in} = 22 [MPa]	P_{out} = 21.8 [MPa]

$\rho_{H,C}$ = 849.8 [kg/m3]
$\rho_{H,out}$ = 27.03 [kg/m3]
ρ_{out} = 865.3 [kg/m3]
$s_{H,C}$ = 2.447 [kJ/kg-K]
$s_{H,in}$ = 2.447 [kJ/kg-K]
$s_{H,out}$ = 2.482 [kJ/kg-K]
s_{in} = 1.94 [kJ/kg-K]
s_{out} = 2.414 [kJ/kg-K]
$T_{H,C}$ = 212.4 [C]
$T_{H,in}$ = 458.5 [C]
$T_{H,out}$ = 158.8 [C]
T_{in} = 162.3 [C]
T_{out} = 212.4 [C]

Local variables in Module Economiser\1 CALL Economiser

h_{in} = 915.8 [kJ/kg]
h_{out} = 1737 [kJ/kg]
m_1 = 0.9909 [kg/s]
P_{DRUM} = 18.1 [MPa]
P_{out} = 18.1 [MPa]
q_{in} = 813.2 [kW]
ρ_{out} = 541.2 [kg/m3]
s_{out} = 3.879 [kJ/kg-K]
T_{out} = 357.5 [C]

Local variables in Module Drum\1 CALL Drum

h_{in} = 1737 [kJ/kg]
h_{out} = 2506 [kJ/kg]
m_1 = 0.9909 [kg/s]
P_{DRUM} = 18.1 [MPa]
q_{in} = 762.4 [kW]
ρ_{out} = 134.8 [kg/m3]
s_{out} = 5.099 [kJ/kg-K]
T_{out} = 357.5 [C]

Local variables in Module HPSuperHeaters\1 CALL HPSuperHeaters

ΔP = 1.2 [MPa]
h_{in} = 2506 [kJ/kg]
h_{out} = 3416 [kJ/kg]
m_1 = 0.9909 [kg/s]
P_{in} = 18.1 [MPa]
P_{out} = 16.9 [MPa]
q_{in} = 901.8 [kW]
ρ_{out} = 50.27 [kg/m3]
s_{out} = 6.432 [kJ/kg-K]
T_{out} = 545 [C]

Local variables in Module HPTurbine\1 CALL HPTurbine

η_{TUR} = 0.9 [-]
h_{in} = 3397 [kJ/kg]
$h_{isen,out}$ = 3029 [kJ/kg]
h_{out} = 3066 [kJ/kg]
m_1 = 0.9909 [kg/s]
m_8 = 0.005175 [kg/s]
P_{out} = 4.7 [MPa]
ρ_{out} = 18.14 [kg/m3]
s_{in} = 6.411 [kJ/kg-K]
$s_{isen,out}$ = 6.411 [kJ/kg-K]
s_{out} = 6.472 [kJ/kg-K]
T_{out} = 345.8 [C]
W_{out} = 330.5 [kW]

Local variables in Module HPAttemperation\1 CALL HPAttemperation

h_{att} = 697.9 [kJ/kg]
h_{in} = 3416 [kJ/kg]
h_{out} = 3402 [kJ/kg]
m_1 = 0.9909 [kg/s]
m_8 = 0.005175 [kg/s]
m_{out} = 0.996 [kg/s]
m_{tot} = 1 [kg/s]
P_{att} = 22 [MPa]
P_{in} = 16.9 [MPa]
P_{out} = 16.9 [MPa]
ρ_{out} = 50.75 [kg/m3]
s_{att} = 1.94 [kJ/kg-K]
s_{in} = 6.432 [kJ/kg-K]
s_{out} = 6.414 [kJ/kg-K]
T_{att} = 162.3 [C]
T_H = 540 [C]
T_{in} = 545 [C]
T_{out} = 540 [C]

Local variables in Module HPTurbineInletValve\1 CALL HPTurbineInletValve

ΔP = 0.1 [MPa]
$h_{ATT,out}$ = 3402 [kJ/kg]
h_{in} = 3397 [kJ/kg]
h_{out} = 3397 [kJ/kg]
m_1 = 0.9909 [kg/s]
m_8 = 0.005175 [kg/s]
$P_{ATT,out}$ = 16.9 [MPa]
P_{in} = 16.9 [MPa]
P_{out} = 16.8 [MPa]
q_{loss} = 4.518 [kW]
ρ_{in} = 50.91 [kg/m3]
ρ_{out} = 50.61 [kg/m3]
s_{in} = 6.409 [kJ/kg-K]
s_{out} = 6.411 [kJ/kg-K]
T_{in} = 538.4 [C]
T_{out} = 538 [C]

Local variables in Module Reheaters\1 CALL Reheaters

ΔP = 1.2 [MPa]
h_{in} = 3066 [kJ/kg]
h_{out} = 3554 [kJ/kg]
m_1 = 0.9909 [kg/s]
m_8 = 0.005175 [kg/s]
P_{in} = 4.7 [MPa]
P_{out} = 3.5 [MPa]
q_{in} = 486.1 [kW]
ρ_{out} = 9.472 [kg/m3]
s_{out} = 7.288 [kJ/kg-K]
T_{out} = 545 [C]

Local variables in Module RHAttemperation\1 CALL RHAttemperation

h_{att} = 697.9 [kJ/kg]
h_{in} = 3554 [kJ/kg]
h_{out} = 3542 [kJ/kg]
m_1 = 0.9909 [kg/s]
m_8 = 0.005175 [kg/s]
m_9 = 0.003973 [kg/s]
m_{tot} = 1 [kg/s]
P_{att} = 22 [MPa]
P_{in} = 3.5 [MPa]
P_{out} = 3.5 [MPa]
ρ_{out} = 9.536 [kg/m3]
s_{att} = 1.94 [kJ/kg-K]
s_{in} = 7.288 [kJ/kg-K]
s_{out} = 7.274 [kJ/kg-K]
T_{att} = 162.3 [C]
T_H = 540 [C]
T_{in} = 545 [C]
T_{out} = 540 [C]

Local variables in Module IPTurbineInletValve\1 CALL IPTurbineInletValve

ΔP = 0.1 [MPa]
h_{in} = 3539 [kJ/kg]
h_{out} = 3539 [kJ/kg]
$h_{SH,out}$ = 3542 [kJ/kg]
m = 1 [kg/s]
P_{in} = 3.5 [MPa]
P_{out} = 3.4 [MPa]
$P_{SH,out}$ = 3.5 [MPa]
q_{loss} = 3.562 [kW]
ρ_{in} = 9.556 [kg/m3]
ρ_{out} = 9.282 [kg/m3]
s_{in} = 7.269 [kJ/kg-K]
s_{out} = 7.282 [kJ/kg-K]
T_{in} = 538.4 [C]
T_{out} = 538 [C]

Local variables in Module Condenser\1 CALL Condenser

$h_{COND1,in}$ = 2354 [kJ/kg]
$h_{COND1,out}$ = 108 [kJ/kg]
$h_{COND2,in}$ = 2426 [kJ/kg]
$h_{COND2,out}$ = 159.4 [kJ/kg]
$h_{CW,in}$ = 75.9 [kJ/kg]
$h_{CW,int}$ = 108.1 [kJ/kg]
$h_{CW,out}$ = 159.5 [kJ/kg]
h_{out} = 139.5 [kJ/kg]
m_4 = 0.1492 [kg/s]
m_5 = 0.04321 [kg/s]
m_6 = 0.3291 [kg/s]
m_7 = 0.328 [kg/s]
$m_{COND1,out}$ = 0.3291 [kg/s]
$m_{COND2,out}$ = 0.5204 [kg/s]
m_{CW} = 22.94 [kg/s]
$P_{CW,in}$ = 0.1 [MPa]
P_{out} = 0.007814 [MPa]
$P_{SAT,COND1}$ = 0.003955 [MPa]
$P_{SAT,COND2}$ = 0.007814 [MPa]
q_{loss} = 1918 [kW]
$\rho_{CW,int}$ = 996.8 [kg/m3]

$\rho_{CW,out}$ = 992.9 [kg/m3] ρ_{out} = 994.6 [kg/m3] $s_{CW,int}$ = 0.3779 [kJ/kg-K]

$s_{CW,out}$ = 0.5464 [kJ/kg-K] s_{out} = 0.482 [kJ/kg-K] TTD_{COND} = 3 [C]

$T_{COND1,out}$ = 25.76 [C] $T_{COND2,out}$ = 38.06 [C] $T_{CW,in}$ = 18.06 [C]

$T_{CW,int}$ = 25.76 [C] $T_{CW,out}$ = 38.06 [C] T_{out} = 33.3 [C]

T_R = 20 [C] $T_{SAT,COND1}$ = 28.76 [C] $T_{SAT,COND2}$ = 41.06 [C]

T_{SUB} = 3 [C]

Local variables in Module LPTurbines\1 CALL LPTurbines

η_{TUR} = 0.85 [-] h_{in} = 3074 [kJ/kg] $h_{isen,TUR1,out}$ = 2226 [kJ/kg]
$h_{isen,TUR2,out}$ = 2311 [kJ/kg] $h_{TUR2,out}$ = 2426 [kJ/kg] m_2 = 0.08745 [kg/s]
m_3 = 0.06304 [kg/s] m_4 = 0.1492 [kg/s] m_5 = 0.04321 [kg/s]
m_6 = 0.3291 [kg/s] m_7 = 0.328 [kg/s] m_{tot} = 1 [kg/s]
m_{TUR1} = 0.3291 [kg/s] m_{TUR2} = 0.328 [kg/s] P_{in} = 0.6 [MPa]
$P_{SAT,COND1}$ = 0.003955 [MPa] $P_{SAT,COND2}$ = 0.007814 [MPa] $P_{TUR1,out}$ = 0.003955 [MPa]
$P_{TUR2,out}$ = 0.007814 [MPa] $\rho_{TUR1,out}$ = 0.03098 [kg/m3] $\rho_{TUR2,out}$ = 0.05763 [kg/m3]
s_{in} = 7.395 [kJ/kg-K] $s_{isen,TUR1,out}$ = 7.395 [kJ/kg-K] $s_{isen,TUR2,out}$ = 7.395 [kJ/kg-K]
$s_{TUR1,out}$ = 7.816 [kJ/kg-K] $s_{TUR2,out}$ = 7.759 [kJ/kg-K] $T_{TUR1,out}$ = 28.76 [C]
$T_{TUR2,out}$ = 41.06 [C] W_{out} = 449.8 [kW]

Local variables in Module IPTurbine\1 CALL IPTurbine

η_{isen} = 0.88 [-] h_{bs} = 3377 [kJ/kg] h_{in} = 3539 [kJ/kg]

$h_{isen,out}$ = 3011 [kJ/kg] h_{out} = 3074 [kJ/kg] k_{poly} = 0.1944 [-]

m_{bs} = 0.08745 [kg/s] m_{in} = 1 [kg/s] m_{out} = 0.9126 [kg/s]

P_{bs} = 2 [MPa] P_{FWH} = 2 [MPa] P_{in} = 3.4 [MPa]

P_{out} = 0.6 [MPa] P_{outlet} = 0.6 [MPa] ρ_{bs} = 6.038 [kg/m3]

ρ_{in} = 9.282 [kg/m3] ρ_{out} = 2.278 [kg/m3] s_{bs} = 7.312 [kJ/kg-K]

s_{in} = 7.282 [kJ/kg-K] s_{out} = 7.395 [kJ/kg-K] T_{bs} = 458.5 [C]

$T_{bs,K}$ = 731.6 [C] T_{in} = 538 [C] $T_{in,K}$ = 811.2 [C]

T_{out} = 305.8 [C] $T_{out,K}$ = 578.9 [C] W_{out} = 438.3 [kW]

ARRAYS TABLE

Nr.	Pressure [MPa]	T [°C]	h [kJ/kg]	s [kJ/kg-K]	rho [kg/m3]
1	0.008651	35.23	147.6	0.5083	993.9
2	0.8	35.29	148.6	0.5088	994.2
3	0.6	111.4	467.5	1.433	950.1
4	0.6	158.8	670.4	1.931	908.6
5	22	162.3	697.9	1.94	917.6
6	21.8	212.4	915.8	2.414	865.3
7	18.1	357.5	1737	3.879	541.2
8	18.1	357.5	2506	5.099	134.8
9	16.9	545	3416	6.432	50.27
10	16.9	540	3402	6.414	50.75
11	16.9	538.4	3397	6.409	50.91
12	16.8	538	3397	6.411	50.61
13	4.7	345.8	3066	6.472	18.14
14	3.5	545	3554	7.288	9.472

15	3.5	540	3542	7.274	9.536
16	3.5	538.4	3539	7.269	9.556
17	3.4	538	3539	7.282	9.282
18	2	458.5	3377	7.312	6.038
19	0.6	305.8	3074	7.395	2.278
20	0.004423	30.71	2365	7.807	0.03431
21	0.008651	43	2437	7.751	0.06322
22	0.008651	43	2437	7.751	0.06322
23*	0.008651	43	2437	7.751	0.06322
24	2	212.4	908.5	2.447	849.8
25	0.6	158.8	908.5	2.482	27.03
26	0.1	20	84.01	0.2965	998.2
27	0.1	18.11	76.11	0.2694	998.6
28	0.1	27.71	116.2	0.405	996.3
29	0.1	40	167.6	0.5724	992.2
C1	0.005124	17.78	368.3	1.463	0.3248
C2	0.005124	17.78	1020	3.705	0.09698
C3	0.328	202.5	1342	3.809	3.915
C4	0.328	111.4	368.3	1.328	700

CHAPTER 9: REFERENCES

Agnew, B., Alaktiwi, A., Anderson, A. & Potts, I. 2004. Simulation of a combined Rankine–absorption cycle. *Applied thermal engineering,* 24(10):1501-1511.

Al-Sulaiman, F.A., Dincer, I. & Hamdullahpur, F. 2010. Energy analysis of a trigeneration plant based on solid oxide fuel cell and organic Rankine cycle. *International journal of hydrogen energy,* 35(10):5104-5113.

Aphornratana, S. & Sriveerakul, T. 2010. Analysis of a combined Rankine–vapour–compression refrigeration cycle. *Energy conversion and management,* 51(12):2557-2564.

Beér, J.M. 2007. High efficiency electric power generation: The environmental role. *Progress in Energy and Combustion Science,* 33(2):107-134.

Bertsch, S.S. & Groll, E.A. 2008. Two-stage air-source heat pump for residential heating and cooling applications in northern U.S. climates. *International journal of refrigeration,* 31(7):1282-1292.

Chacartegui, R., Sánchez, D., Muñoz, J.M. & Sánchez, T. 2009. Alternative ORC bottoming cycles for combined cycle power plants. *Applied energy,* 86(10):2162-2170.

Cole, G.H.A. 1991. Thermal power cycles. London: Edward Arnold.

Eastop, T.D. & McConkey, A. 1993. Applied thermodynamics for engineering technologists. 5th ed. Essex: Pearson Education.

EIA (U.S. Energy Information Administration). 2011. International Energy Outlook 2011. http://www.eia.gov/forecasts/ieo/pdf/0484(2011).pdf Date of access: 10 Sept. 2012.

EPA (U.S. Environmental Protection Agency). 2012. Global Emissions.http://www.eia.gov/ forecasts/ieo/pdf/0484(2011).pdf Date of access: 10 Sept. 2012.

Granet, I. 1980. Thermodynamics and heat power. 2nd ed. Reston: Reston.

Hajabdollahi, F., Hajabdollahi, Z. & Hajabdollahi, H. 2012. Soft computing based multi-objective optimization of steam cycle power plant using NSGA-II and ANN. *Applied soft computing,* 12(11):3648-3655.

Haywood, R.W. 1987. Analysis of engineering cycles. 3rd ed. Oxford: Pergamon.

Heyen, G. & Kalitventzeff, B. 1999. A comparison of advanced thermal cycles suitable for upgrading existing power plant. *Applied thermal engineering,* 19(3):227-237.

Ibrahim, O.M. & Klein, S.A. 1996. Absorption power cycles. Energy, 21(1):21-27.

Jing, L., Gang, P. & Jie, J. 2010. Optimization of low temperature solar thermal electric generation with organic Rankine cycle in different areas. *Applied energy,* 87(11):3355-3365.

Kalina, A.I. 1984. Combined system with novel bottoming cycle. *ASME journal of engineering for gas turbines and power,* 106(4):737-742.

Kongtragool, B. & Wongwises, S. 2003. A review of solar-powered Stirling engines and low temperature differential Stirling engines. *Renewable and sustainable energy reviews,* 7(2):131-154.

Kongtragool, B. & Wongwises, S., 2008. A four power-piston low-temperature differential Stirling engine using simulated solar energy as a heat source. *Solar energy,* 82(6):493-500.

Kröger, D.G. 2004. Air-cooled heat exchangers and cooling towers: thermal-flow performance evaluation and design.2nd ed. Tulsa: PennWell.

Lior, N. 1997. Advanced energy conversion to power. *Energy conversion and management,* 38(10-13):941-955.

McQuay International. 2002. Refrigerants - application guide.http://fluid.itcmp.pwr.wroc.pl/~zajaczkowski/files/docs/AG31-007.pdf Date of access: 14 Jun. 2012.

Mittelman, G. & Epstein, M. 2010. A novel power block for CSP systems. *Solar energy,* 84(10):1761-1771.

NERSA (National Energy Regulator of South Africa). 2010. Electricity supply statistic 2006. http://www.nersa.org.za/Admin/Document/Editor/file/News%20and%20Publications/Publications/Current%20Issues/Electricity%20Supply%20Statistics/Electricity%20supply%20statistics%202006.pdf Date of access: 14 Sept. 2011.

OECD (The Organisation for Economic Co-operation and Development). 2011. OECD factbook 2011-2012: economic, environmental and social statistics. http://www.oecd-ilibrary.org/ economics/oecd-factbook_18147364 Date of access: 12 Sept. 2012.

Poullikkas, A. 2005. An overview of current and future sustainable gas turbine technologies. *Renewable and sustainable energy reviews,* 9(5):409-443.

Saravanamuttoo, H.I.H., Rogers, G.F.C. & Cohen, H. 2001. Gas turbine theory. 5th ed. Dorchester: Pearson Education.

Schroeder, D.V. 2000. An introduction to thermal physics. San Franscisco: Addison Wesley Longman.

Sonntag, R.E., Borgnakke, C. & Van Wylen, G.J. 2003. Fundamentals of thermodynamics. 5th ed. Hoboken: John Wiley.

Storm, C.P., 2012. Operating parameters of a typical Rankine cycle [personal interveiw]. 16 Mar., Potchefstroom.

Tchanche, B.F., Lambrinos, G., Frangoudakis, A. & Papadakis, G. 2011. Low-grade heat conversion into power using organic Rankine cycles – a review of various applications. *Renewable and sustainable energy reviews,* 15(8):3963–3979.

Ratts, E.B. & Brown, J.S. 2000. A generalized analysis for cascading single fluid vapor compression refrigeration cycles using an entropy generation minimization method. *International journal of refrigeration,* 23(5):353-365.

Winkler, J., Aute, V. & Radermacher, R. 2008. Comprehensive investigation of numerical methods in simulating a steady-state vapor compression system. *International journal of refrigeration,* 31(5):930-942.

Xiong, J., Zhao, H., Zhang, C., Zheng, C., Luh, P.B. 2012. Thermoeconomic operation optimization of a coal-fired power plant. *Energy,* 42(1):486-496.