Institutionen för systemteknik
Department of Electrical Engineering

Examensarbete

Modeling and Experimental Validation of a Rankine Cycle Based Exhaust WHR System for Heavy Duty Applications

Examensarbete utfört i Fordonssystem
vid Tekniska högskolan vid Linköpings universitet
av

Carin Carlsson

LiTH-ISY-EX--12/4595--SE

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Modellering och experimentell validering av ett Rankinecykelbaserat Waste Heat Recovery-system

Modeling and Experimental Validation of a Rankine Cycle Based Exhaust WHR System for Heavy Duty Applications

Författare
Author

Carin Carlsson

Sammanfattning
Abstract

To increase the efficiency of the engine is one of the biggest challenges for heavy vehicles. One possible method is the Rankine based Waste Heat Recovery. Crucial for Rankine based Waste Heat Recovery is to model the temperature and the state of the working fluid. If the state of the working fluid is not determined, not only the efficiency of the system could be decreased, the components of the system might be damaged.

A SIMULINK model based on the physical components in a system developed by SCANIA is proposed. The model for the complete system is validated against a reference model developed by SCANIA, and the component models are further validated against measurement data. The purpose of the model is to enable model based control, which is not possible with the reference model. The main focus on the thesis is to model the evaporation and condensation to determine state and temperature of the working fluid. The developed model is compared to a reference model with little differences for while stationary operating for both the components and the complete system. The developed model also follows the behavior from measurement data. The thesis shows that two phase modeling in SIMULINK is possible with models based on the physical components.

Nyckelord
Keywords

Component model, heat transfer, two phase, evaporation, condensation, heat exchanger, turbine, pump, expander, recuperator

Abstract

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A Simulink model based on the physical components in a system developed by Scania is proposed. The model for the complete system is validated against a reference model developed by Scania, and the component models are further validated against measurement data. The purpose of the model is to enable model based control, which is not possible with the reference model. The main focus on the thesis is to model the evaporation and condensation to determine state and temperature of the working fluid. The developed model is compared to a reference model with little differences for while stationary operating for both the components and the complete system. The developed model also follows the behavior from measurement data. The thesis shows that two phase modeling in Simulink is possible with models based on the physical components.

Sammanfattning

Att höja verkningsgraden för motorn är en av lastbilsbranschens största utmaningar. En metod för detta är ett Rankine-baserat Waste heat recovery. Avgörande för ett Rankine-baserat Waste heat recovery-system är temperatur- och fasmodellering av det använda arbetsmediet. Om inte fas på arbetsmediet kan fastställas riskeras inte bara verkningsgrad på systemet utan även skador på komponenter.

En Simulink-modell baserad på de fysiska komponenterna i ett av Scania utvecklat Waste heat recovery system är utvecklad. Modellen för det slutna systemet är validerad mot en av Scania utvecklad referensmodell, och delmodellerna är validerade mot det fysiska systemet. Syftet med modellen är att skapa bra modeller som möjliggör modellbaserad reglering. Huvudfokus har lagts på modellering av evaporation och kondensation för att säkerställa fas och temperatur på arbetsmediet. Den utvecklade modellen har jämförts mot en referensmodell med låga statiska skillnader både för varje komponent och för hela systemet. Modellen följer även beteendet från mätdata. Examensarbetet visar att tvåfasmodellering i Simulink är möjligt med modeller baserade på de fysiska komponenterna.

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I would like to thank my supervisor here at SCANIA, Kristian Lundh and my boss, Henrik Flemmer for letting me write my Thesis at NEPP at SCANIA. A big thanks for all the help and support during the process.

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Contents

Chapter 1

Introduction

This is the report for a Master's Thesis in Electrical Engineering. The thesis is examined at the division of Vehicular Systems at Linköping University and performed at SCANIA AB in Södertälje. The purpose of this report is to describe the work and results. The purpose, goals and background are described in this chapter.

1.1 Purpose and Goals

The need of lowering the fuel consumption of heavy duty vehicles has increased the last years due to a high oil price and new upcoming legislation regarding carbon dioxide [1].

The efficiency of a heavy duty vehicle is about 40 percent [2]. This means that 40 percent of the fuel becomes brake power and the remaining 60 percent is mainly turned into heat. The heat is called waste heat since it is not used for anything. One way to lower the fuel consumption is to recover energy from the waste heat. The energy may then be put back into the powertrain. That could be done with a process called Waste Heat Recovery (WHR). All the energy in the waste heat cannot be recovered, but it is still possible to lower the fuel consumption up to 5-6 percent [3]. This thesis focuses on Rankine Cycle based WHR. The Rankine Cycle is a thermodynamic cycle that turns heat into work by heating and cooling of a working fluid. The Rankine Cycle is further described in Section 1.2.3.

Waste Heat Recovery for heavy duty applications is a complicated process in several ways. First of all, adding a new system to a truck is not a simple thing to do. The system needs to be stable and to work in many conditions. A truck operates in both very cold and very hot climate which adds many demands to the system.

Secondly, the process of heating a liquid to gas is not simple to predict. It demands good models and good control systems to keep the system stable and able to operate. Above that, the system must be efficient enough to recover as much heat as possible. This adds demands to the components as well as to the working fluid but especially on the control system. Good control system will require good models and knowledge about the system.

The goal of this thesis has therefore been to develop models of the system in SIMULINK.

This thesis describes models of the components necessary for Rankine Cycle based waste heat recovery. It also describes a complete model and how to implement it. The focus on the modeling has been to model the heat transfer. A workinmg fluid is evaporated from liquid to gas and condensed back to liquid. Modeling evaporation and condensing, the two-phase flow, has been the main challenge and focus. This thesis describes the two phase flow, how to determine state of matter and how to model it. The models are mainly compared to a reference model and to measurement data from a test cell, both described in Section 1.4.

1.2 Background

The background to the thesis is presented in this section.

1.2.1 Reducing Fuel Consumption

There are different methods to lower the fuel consumption for a heavy duty vehicle. A lot of time and money are spent on research, and some examples of projects to save fuel beside waste heat recovery are look ahead control [4] and hybrid technology [5].

1.2.2 Waste Heat Recovery

There are different ways to recover energy from the waste heat within the combustion process. This thesis covers the thermodynamic process which is based on the Rankine cycle and can be seen in Figure 1.1. In the figure, the grey lines symbolize gas flow and the white lines liquid flow. The system schematically shown is a waste heat recovery system simplified to describe the Rankine cycle. The heat exchanger in the figure may be modeled as one, but to better describe the physical system it is modeled as three single units since this is the actual construction of the system studied in this thesis. The biggest challenge with the modeling is that the working fluid may be in both liquid and gas state while entering or leaving one of the heat exchangers. It is also very important that the working fluid is fully evaporated before it enters the turbine expander. Otherwise, not as much energy will be recovered and the turbine might even break.

A recuperator is placed after the turbine to extract energy from the working fluid if there is still extractable energy. The recuperator is a special purpose heat exchanger, that recovers waste heat and reuses it later [6]. In the recuperator it is possible to cool down the working fluid after the turbine and before the condenser.

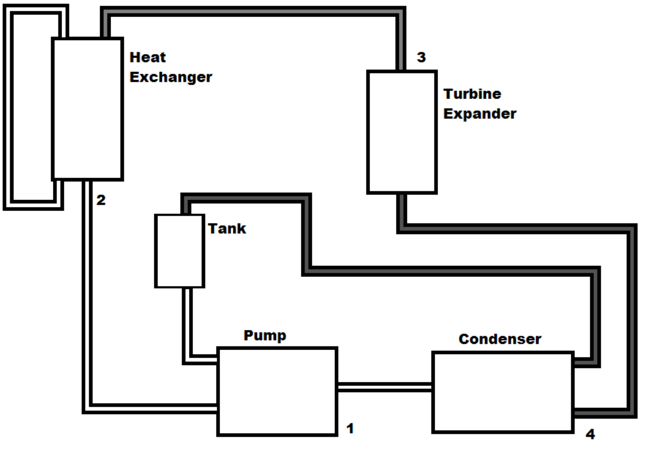


Figure 1.1. A waste heat recovery system based on the Rankine cycle. The working fluid is being pumped from low to high pressure by the pump. Then it vaporizes in the heat exchanger and enters the turbine expander as gas. The energy is extracted and then the working fluid condenses to liquid. The tank is used to store working fluid in. The control system regulates the system mass flow and uses the tank to store working fluid in. The control system may add or remove working fluid from the tank.

In the condenser, work is lost while cooling down the working fluid. Therefore, the less heat the working fluid contains while entering the condenser the better. The hot working fluid coming from the turbine is cooled down by heat transfer to the cooler working fluid coming from the pump.

The heat in the recuperator is saved and reused to heat up the working fluid before entering the heat exchanger.

The condenser seen in Figure 1.1 is modeled in two components to fit with the physical system. The working fluid enters first the condenser, where most of the working fluid is condensed to liquid. After the condenser, the working fluid enters a sub cooler where the rest of the fluid is condensed to liquid and cooled down to temperatures below boiling point.

1.2.3 The Rankine Cycle

The Rankine cycle consists of four general processes [7]. Figure 1.2 shows the Rankine cycle in an entropy versus temperature diagram. The figure also shows where the work and heat are put in and taken out. The process can be described as [8]

- (1) – (2): Reversible adiabatic compression of the working fluid, which is a liquid at this stage. The working fluid is being pumped from low to high pressure. This process is performed by the pump. The work done by the pump is called $\dot{W}_{in}$.

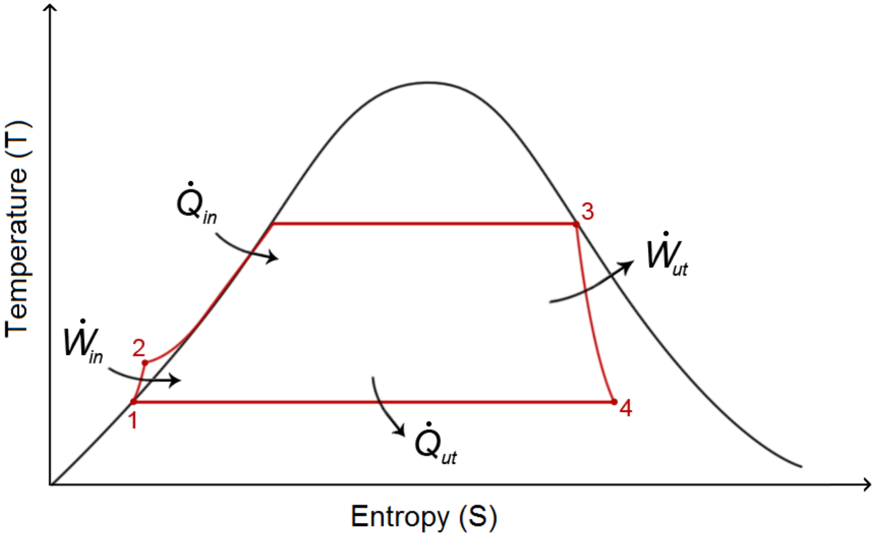


Figure 1.2. Entropy versus temperature diagram for the Rankine cycle. The curve shows the relationship between temperature and entropy. The Rankine cycle converts heat into work. The figure also shows where work and heat are put in and taken out. The picture is downloaded from Wikipeda 2012-05-06.

- $(2)-(3)$: The liquid is being heated at constant pressure to a dry saturated vapor by external heat sources. This process is performed by one or several heat exchangers. The heat transferred in the heat exchanger(s) is called $\dot{Q}_{in}$.
- $(3)-(4)$: The dry saturated vapor generates power while expanding through a turbine. The temperature and the pressure of the vapor decreases and some condensation may occur. This is an isentropic expansion of the working fluid vapor in an expander that generates mechanical power. The work extracted in the turbine expander is called $\dot{W}_{out}$.
- $(4)-(1)$: The wet vapor condenses in a condenser at a constant temperature to become a saturated liquid. This is an isobaric condensation of the working fluid. This process is performed by the condenser. The heat loss in the condenser is called $\dot{Q}_{out}$.

The ideal efficiency of the Rankine cycle is [9]

$$\eta_{ideal} = \frac{\dot{W}_{out} - \dot{W}_{in}}{\dot{Q}_{in}} = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}}. \tag{1.1}$$

1.2.4 Working Fluid

The purpose of the working fluid is to carry the heat. The heat is absorbed in the evaporator and transported to the turbine expander where it is turned into work. The more heat that can be turned into work, the higher efficiency of the complete system.

When choosing working fluid, several parameters needs to be considered. One of the parameters is the entropy, which has one of the most important roles since it affects how much work that is possible to extract. This can be seen in Figure 1.2. A low boiling point is also important to make overheating possible since this enables more extracting of energy [2]. Overheating refers to heat up a working fluid above its boiling point. Another parameter in the choice of working fluid is whether there are hardware components that are possible to use with the fluid, such as pump and heat exchanger. It is also important that the working fluid is possible to use in the environment of a heavy duty vehicle. The fluid is not supposed to freeze during the cooling for example [7].

The pressure and the temperature of a working fluid correlate with a lot of other variables. Heat capacity, c_p, defines the amount of energy that is required to achieve a certain raise in temperature of a certain mass in a specific material. This parameter affects the ability to increase the temperature in the working fluid. Viscosity, μ, refers to dynamic viscosity. Viscosity can be seen as a measurement of the flow resistance in fluids. This parameter affects the flow. Entropy, S, is how much of the energy in a system that cannot be transferred into work. This parameter affects the efficiency of the system. Enthalpy, h, is a measurement of the sum of the internal energy and the product of the pressure and the volume.

The change in enthalpy is the supplied heat. This parameter affects in which state of matter the fluid is in. Thermal conductivity, k, is a measurement of the ability for a specific material to conduct heat.

The behavior of the working fluid is included in the modeling of the components.

1.3 Related Research

The basics of the Rankine cycle and thermodynamic processes can be found in [9], [7] and in [8]. The dynamics of the Rankine cycle for power plants and boiling processes is described in [10] and [11].

Grebekov et al [12] describes how the change in temperature and pressure affects R245fa, the refrigerant used as working fluid for the system studied in this thesis. Density, pressure, thermal conductivity and temperature are investigated in vapor and liquid states.

Latz et al [13] presents a comparison of working fluids. The behavior of the working fluid is very important for the efficiency of the Rankine cycle and it needs more research.

Achkoudir and Hanna [14] describes the efficiency potential for water based waste heat recovery-systems for heavy duty vehicles. Their test is interesting as inspiration for modeling, but water is unfortunately not a good working fluid since heavy duty vehicles may work in very cold climates, where the temperature is below the freezing point for water. The boiling point of water is also too high to enable overheating of the fluid to exctract as much energy as possible.

Lemort et al [15] develops a model to investigate the potential of improvements of the waste heat recovery-system.

Park et al [3] includes test for different heat exchanger solutions. They evaluate the system for the following configurations:

1. Waste heat from EGR cooling for both evaporation and superheating.
2. Waste heat from EGR cooling for evaporation combined with waste heat from the exhaust gases for superheating.
3. Waste heat from the exhaust gases for both evaporation and superheating.

The different configurations give different test results for indicated turbine power and fuel consumption benefits. Different configurations are also studied in [16].

Teng et al [17] describes the differences between expander and turbine. Different working fluids for waste heat recovery based on the Rankine cycle are described in [2] and [18]. Taleshian et al [19] presents gas power and efficiency computations for gas turbines.

The research in the last years has been focused on efficiency in general and different working fluids. The efficiency has been tested to see how much fuel the waste heat recovery-process possibly can save and to whether it is worth further investigation. The working fluids could be one of the cornerstones in higher efficiency, but another cornerstone is the control systems. The processes need better models to increase control performance.

This thesis models the complete system. It also adds information of the heat transfer during different states of the working fluid. The thesis focuses on heat transfer and especially evaporation and condensing. This enables modeling in SIMULINK of the complete system.

1.4 Reference Model and Test Cell

The model developed in this thesis will be compared to a reference model and to measurement data from a test cell. The reference model and the test cell are proprietary but briefly described here.

The reference model is modeled in GT-Suite with the same physical component as in the developed model [20]. GT-Suite is a Computer-Aided Engineering tool used for simulation of vehicle system provided by Gamma Technologies Inc. This model is used today for testing and development of control systems to the WHR-system and includes all physical components. A control system is developed by SCANIA for the reference model, and during comparisons the control system may be used on the model developed in this thesis as well. But, GT-Suite does not enable model based control, since the models are not possible to extract from the system. Therefore a new model is needed.

The reference model is used for validation of the developed model together with data from the test cell. Measurement data from the test cell can be used to compare pressure and temperature, but the mass flow sensors are not reliable enought for comparisons. There are no enthalpy sensors, which means that mass flow and enthalpy are only possible to compare between the reference model and the developed model. Pressure and temperature will be compared to both the reference model and the temperature model.

The system in the test cell can be described schematically as in Figure 1.3. The three heat exchangers are schematically shown as one unit. The condenser and sub cooler are also shown as one unit. The pressure and temperature sensors used for validation are seen in the figure.

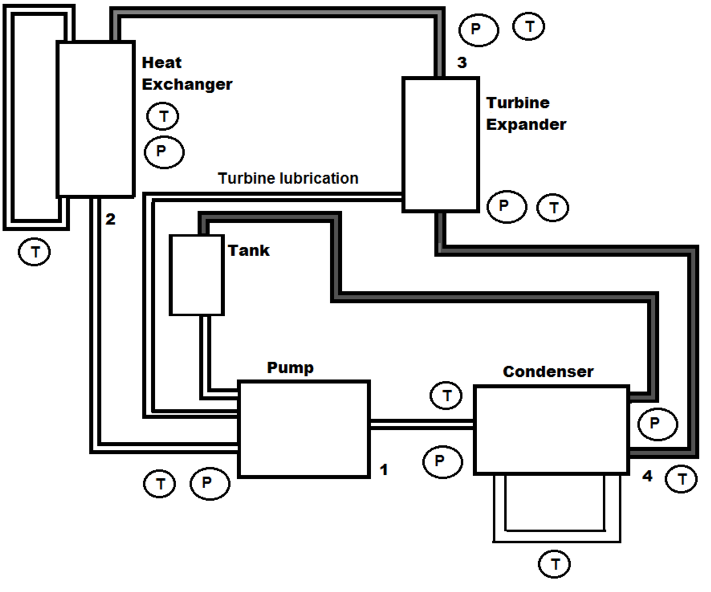


Figure 1.3. A schematic of the system installed in the engine test cell. The temperature sensors and pressure sensors used for validation are marked with T and P. The three heat exchangers are schematically shown as one and the same is for the condenser and subcooler.

1.5 Nomenclature and Abbreviation

Table 1.1. Nomenclature

Variable	Name	Unit
A	Area	m^2
c_p	Heat capacity during constant pressure	$\frac{J}{kgK}$
c_v	Heat capacity during constant volume	$\frac{J}{kgK}$
C_q	Flow coefficient	
d	Diameter	m
D_H	Hydraulic diameter	m
e	energy	J
h	Enthalpy	$\frac{J}{kg}$
h_t	Heat transfer coefficient	$\frac{W}{m^2K}$
k	Thermal conductivity	$\frac{W}{mK}$
$\dot{m}$	Mass flow	$\frac{kg}{s}$
n	Rotational speed	$\frac{Revolutions}{second}$
Nu	Nusselts number	
p	Pressure	Pa
P	Power	$\frac{J}{s}$
Pr	Prandtls number	
$\dot{q}$	Heat transfer	$\frac{J}{s}$
Q	Energy	J
Re	Reynolds number	
S	Entropy	$\frac{J}{kgK}$
T	Temperature	K
u	Internal energy	J
U	Speed	$\frac{m}{s}$
V	Volume	m^3
$\dot{V}$	Volume flow	$\frac{m^3}{s}$
$\dot{W}$	Power	$\frac{J}{s}$
x	Fraction of gas in mass flow	$-$
y	Thickness of plate	m
η	Efficiency	$-$
μ	Viscosity	Pas
ρ	Density	$\frac{kg}{m^3}$

Table 1.2. Abbreviations

Variable	Name
$cond$	Condensing
d	Displacement
$evap$	Evaporation
h	Hydraulic
l	Liquid
$metal$	Metal plate
r	Reference
rd	Reduced
v	Vapor
w	Wall
wf	Working fluid
$wf, cold$	Cold working fluid
wf, hot	Hot working fluid

Chapter 2

Component Modeling

In this chapter, the modeling of the components in the system is described. The components are schematically shown in Figure 1.3 and described in Section 1.2.2.

At first, the properties of the working fluid are modeled. Then the component models for the pump, the heat exchangers, the turbine expander and the condenser are provided and validated.

2.1 Working Fluid

The properties of the working fluid described in Section 1.2.4 are modeled with maps. Data to the maps are collected from [21]. The enthalpy h is modeled as

$$h = f_{map}(T, p) \tag{2.1}$$

where the parameters into the map are temperature and pressure. The specific heat capacity during constant pressure modeled as

$$c_p = f_{map}(T, p) \tag{2.2}$$

and the specific heat capacity during constant volume is modeled as

$$c_v = f_{map}(T, p). \tag{2.3}$$

The thermal conductivity is modeled as

$$k = f_{map}(T, p) \tag{2.4}$$

and the viscosity is modeled as

$$\mu = f_{map}(T, p). \tag{2.5}$$

The temperature and pressure levels are adjusted to likely levels in the WHR-system. All the maps consist of value points for the variable depending on pressure and temperature. The values between the points are calculated with linear extrapolation.

The boiling temperature depending on pressure and enthalpy is modeled with the inverse map of h as

$$T = f_{map,inv}(h, p) \tag{2.6}$$

and linear extrapolation is used to calculate values between the value points.

In due to hardware limitations are the maps nog as big as could be wished in this application. It is very important to receive the right value of boiling temperature for a certain pressure in order to get an accurate model. This since the amount of energy required to evaporate the working fluid is large compared to change the temperature with a few Kelvin.

2.2 Pump

A pump is used to pump the working fluid. The consumed pump work $\dot{W}_{in}$ is modeled as [22]

$$\dot{W}_{in} = \frac{(p_2 - p_1)\dot{m}}{\rho\eta_{pump}} \tag{2.7}$$

where p_2 and p_1 are the pressure in the working fluid after and before the pump, $\dot{m}$ the mass flow, ρ the density of the working fluid and η_{pump} the efficiency of the pump. The pump delivers a working fluid flow $\dot{V}$ to the heat exchanger. This might also cause a pressure change. The pressure change is caused by the volumetric flow between the pump and the turbine expander. There may be a pressure difference over the pump because of the expander. Since the pump delivers the working fluid, the pressure change needs to be included in the pump model.

The pump speed n_{pump} is used as a control signal for the complete WHR-system, since the volume flow affects both how much overheating that occurs and the pressure in the heat exchangers.

2.2.1 Modeling

Assumptions

- The heat remains constant during the compression in the pump.
- The density before and after the pump is constant. The variation in density for the working fluid in liquid state for the pressure range 1 bar to 32 bar is about 0.6% which does not give any large impact on the working fluid.

The pump produces a volumetric flow

$$\dot{V}_0 = \eta_v n_{pump} V_d \tag{2.8}$$

where the η_v is the volumetric efficiency, n the pump speed and V_d the volume displacement [14].

2.2.2 Modeling System Pressure

The pump is used to control the mass flow, which affects the pressure in the working fluid. The amount of working fluid that flows through the pipes is determined by the pump and decides the system pressure. Since the volumetric flow also is affected by the pressure change over the pump, the model needs to be extended as [23]

$$\dot{V} = \dot{V}_0 - a(\delta p)^b \tag{2.9}$$

where δp is the pressure difference over the pump, b is constant and a is given by

$$a = \frac{\dot{V}_r}{(\delta p_0)^b - (\delta p_r)^b} \tag{2.10}$$

where δp_0 is the pressure when the volumetric flow is zero. $\dot{V}_r$ is modeled as

$$\dot{V}_r = V_0(1 - \frac{\delta p_r}{\delta p_0})^b \tag{2.11}$$

In the further modeling, mass flow is used instead of volumetric flow as [7].

$$\dot{m} = \dot{V}\rho \tag{2.12}$$

The pressure difference depends on rotational speed, reference rotational speed and reference pressure difference [24], as

$$\delta p_n = \delta p_r(\frac{n_{pump}}{n_r})^2 \tag{2.13}$$

where δp_r is the pressure rise at the reference rotational speed n_r.

2.2.3 Turbine Lubrication Modeling

The turbine needs lubrication in order to run. The working fluid is used as lubrication. A small mass flow of working fluid is therefore led from the pump outlet to the turbine. This can be seen in Figure 1.3. The lubrication mass flow is modeled as [25]

$$\dot{m}_{expander} = c_q \pi d^2 * \sqrt{2\rho\delta p} \tag{2.14}$$

where c_q is the flow coefficient, d is the diameter of the expander, ρ is the density of the working fluid and δp is the pressure difference over the working fluid. The lubrication is about 2-5 percent of the complete mass flow. The small lubrication flow also leads to that the mass flow going through the heat exchanger and the turbine is thus slightly smaller than the flow leaving the pump. The heat exchanger flow is thus given as

$$\dot{m}_{he} = \dot{m}_{pump} - \dot{m}_{to-tank} - \dot{m}_{expander} \tag{2.15}$$

where $\dot{m}_{pump}$ is the mass flow after the pump and $\dot{m}_{to-tank}$ is the mass flow to the tank, given by the control system. The flow going in to the condenser, following the turbine, is the turbine flow added with the lubrication flow as

$$\dot{m}_{condensor} = \dot{m}_{pump} + \dot{m}_{from-tank} + \dot{m}_{expander} \tag{2.16}$$

where $\dot{m}_{from-tank}$ is the working fluid added from the tank given by the control system.

2.2.4 Validation

The model is validated with data from the reference model and with data from the physical system. Since the physical system only has reliable sensors for pressure and temperature it is only used to validate the pressure change caused by the volumetric flow. The reference model is used to compare pressure difference but also volumetric flow.

Validation against reference model

Figure 2.1 shows the mass flow in the model compared to the reference model. The comparison is made during start of the system and when the system is stationary operating after start. The large variations and peaks depend on the control system. Large variations in rotational speed gives large changes of the mass flow. Figure 2.1 shows that the mass flows have similar behavior except during large rotational speeds that gives large variations in mass flow.

Figure 2.2 shows the pressure difference over the pump in the model compared to the pressure difference in the reference model. The figure shows the mass flow during start of the system and when the system is stabilized after the start. The large variations and peaks depends on the control system for the reference model.

Validation against Cell Data

Figure 2.3 shows the pressure difference over the pump compared to pressure difference in the real system and the difference between them. The rotational speed is varying which causes changes in pressure differences. The behavior of the developed model agrees with the behavior seen in measurement data, but with a difference in values that decreases for higher pressure differences. Since the behavior agrees, the difference can depend on the reference speed. As seen in Equation (2.13), the differences caused by the reference speed may vary with the rotational speed of the tested system.

2.2.5 Simulink model

The pump with the extended pressure model is modeled in SIMULINK. Input to the model is the control signal n. The V_d, η, b, the pressure at zero flow, the pressure at reference speed and the reference speed are constants.

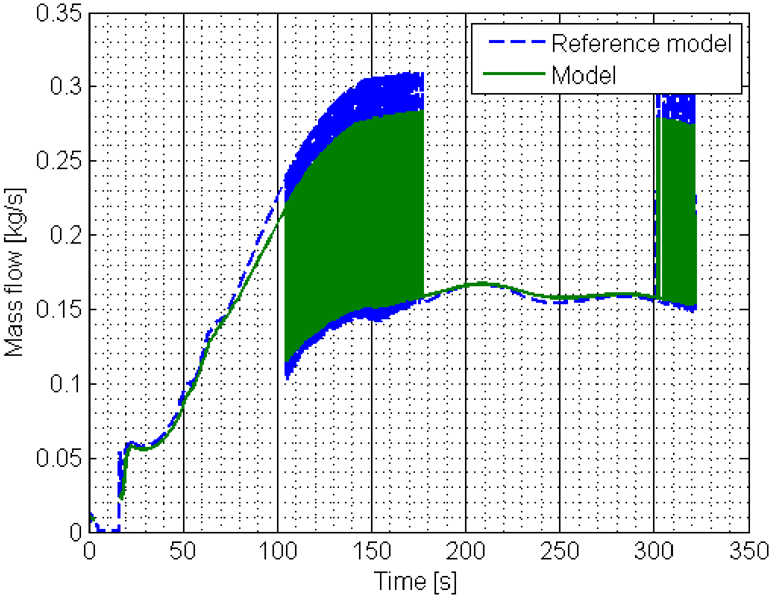


Figure 2.1. The mass flow in the model compared to the mass flow in the reference model. The figure shows the mass flow during start of the system and when the system is stationary operating after the start. The large variations and peaks between $t = 100$ s and $t = 180$ s and between $t = 300$ s and $t = 320$ s depend on the control system for the reference model. It can be seen that the behavior of the mass flows agrees with the exception of amplitud differences during large rotational speeds that gives large variations in mass flow.

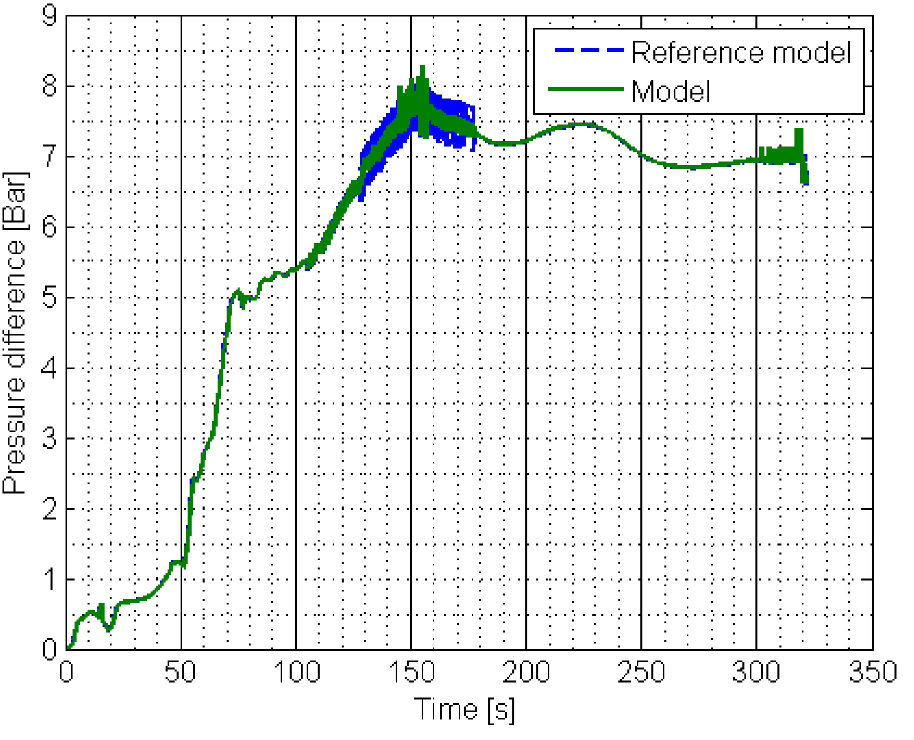


Figure 2.2. The pressure difference over the pump in the model compared to the pressure difference in the reference model. The figure shows how the pressure differs during the start and when the system is stationary operating after the start. The pressure differences agrees during the simulation except for the large variations in rotational speed which causes large pressure differences. The amplitude of the oscillations is greater for the reference model. During stationary operating are the differences very small and will not affect other parts of the system.

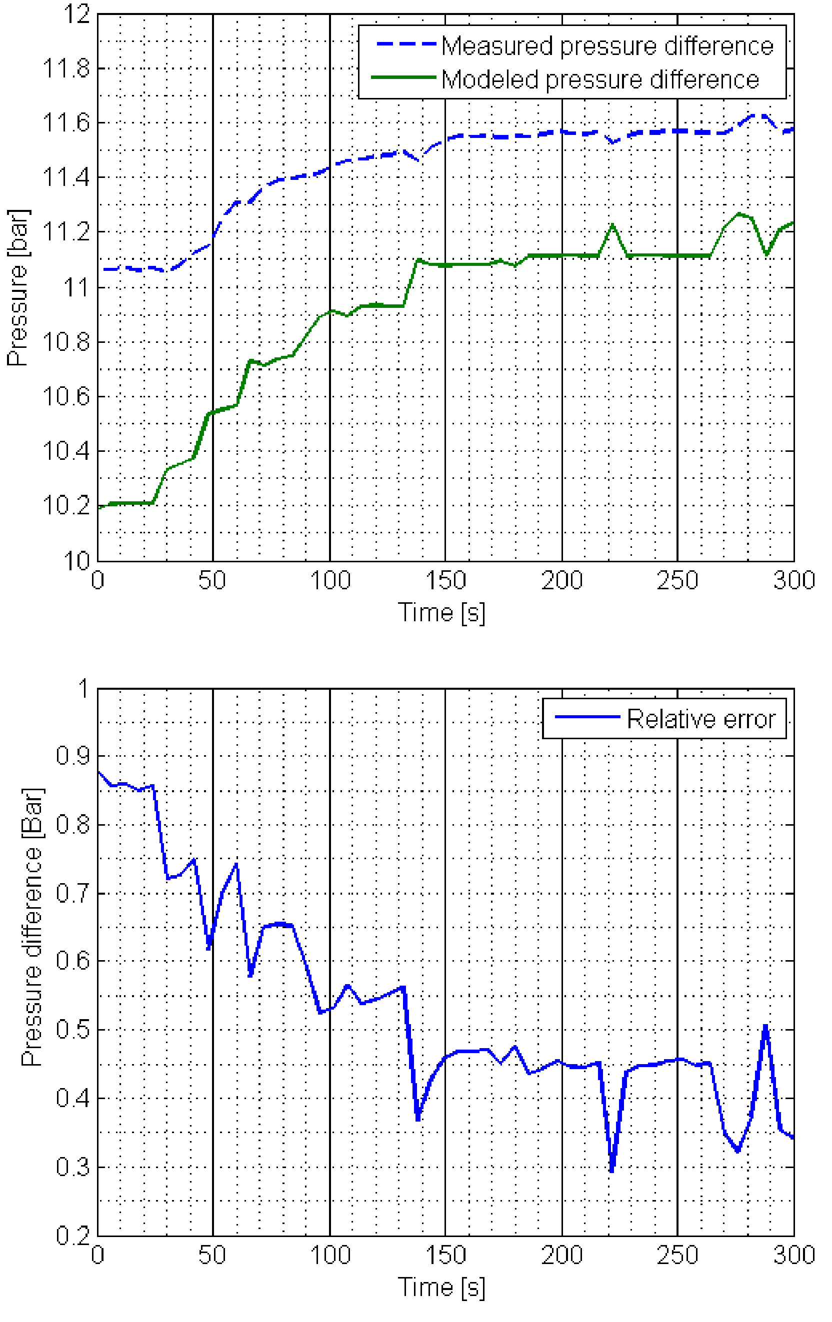


Figure 2.3. The pressure difference over the pump in the model compared to measurement data is shown in the upper figure. The behavior of the developed model agrees with the behavior seen in measurement data, but with a difference in values that decreases for higher pressure differences. Since the behavior agrees, the difference might depend on the reference rotational speed. The difference, that is seen in the lower figure, decreases for large pressure differences which indicates that the reference speed might not be accurate.

2.3 Heat Exchangers

The purpose of the heat exchangers is to transfer heat. There are three different kinds of heat transfer that will be described in the section. The first heat transfer is from a hot gas to a colder working fluid. This will be referred to as *heating*. The second heat transfer is from a hot working fluid to a colder refrigerant. This will be referred to as *cooling*. The last heat transfer is from a hot working fluid to a colder working fluid. This will be referred to as *recuperation*.

The principle of a heat exchanger can be seen in Figure 2.4. The figure shows a counter flow heat exchanger, where the two flows run in opposite direction [26]. The two flows are separated by a metal plate and one flow contains hot media, in this specific type, hot gas. The other flow consists of the working fluid. This kind of heat exchanger is used in the WHR-system studied in this thesis. In a counterflow heat exchanger, the hot gas is warmest just before the working fluid leaves the heat exchanger, which leads to a bigger heat transfer compared to a parallel flow heat exchanger [27].

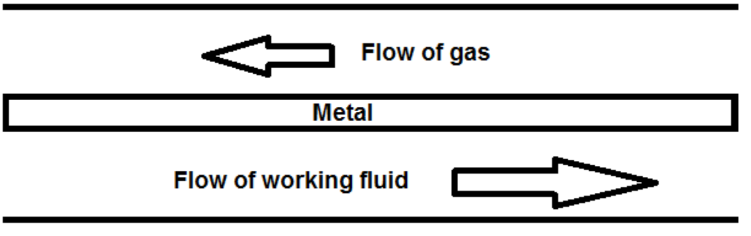


Figure 2.4. Principle of a heat exchanger. There are two flows, separated by a metal plate, where one flow contains hot media, in this specific type, hot gas. The other flow consists of the working fluid. This is a counterflow heat exchanger where the hot gas and the working fluid flow in the opposite direction of each other. This kind of the heat exchanger is used in the WHR-system studied in this thesis. In a counterflow heat exchanger, the hot gas is warmest just before the working fluid leaves the heat exchanger, which leads to a bigger heat transfer compared to a parallel flow heat exchanger.

2.3.1 Heat Transfer

Calculation of the heat transferred from one flow to another may be divided into three parts;

1. Forced convection from the hot gas to the metal.

2. Conduction in the metal plate.

3. Forced convection from the metal plate the working fluid.

The correlation between the temperature changes from the hot gas via the metal to the working fluid is modeled as [7]

$$h_{t,1}[T_{1,\infty}] = k\frac{\partial T(0,t)}{\partial x} \tag{2.17}$$

$$h_{t,2}[T_{2,\infty}] = k\frac{\partial T(L,t)}{\partial x} \tag{2.18}$$

where $h_{t,1}$ and $h_{t,2}$ are the heat transfer coefficient, $T_{1,\infty}$ and $T_{2,\infty}$ the temperatures in the flows and k the thermal conductivity in the metal plate. The heat transfer is schematically shown in Figure 2.5.

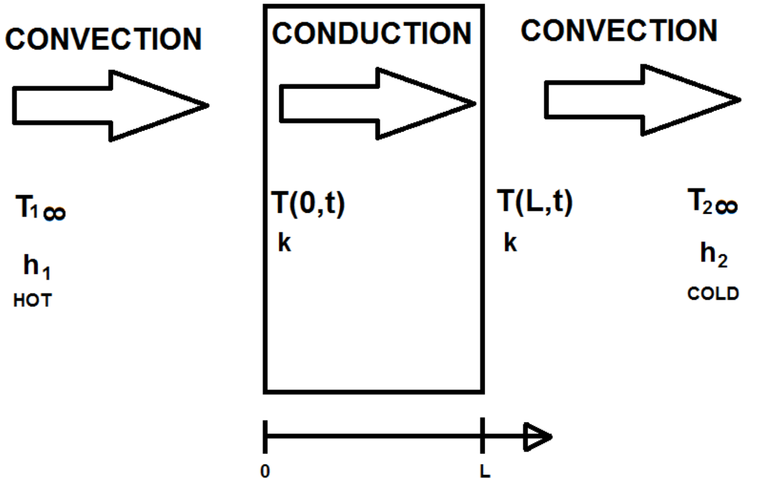


Figure 2.5. Heat transfer between a gas with high temperature and the working fluid through walls of the heat exchanger. This is the heat transfer that occurs in Figure 2.4. Forced convection from the hot gas to the metal plate heats up the metal plate. Conduction in the metal plate transfers heat from the hot side of the metal plate to its other side. Forced convection from the metal plate to the working fluid heats up the working fluid.

Convection from hot gas to the metal plate

The first part in the heat transfer is convection. The heat transfer in the heat exchanger goes from a fluid with a high temperature to the working fluid with lower temperature. The heat transfer can be seen in Figure 2.5. In Figure 2.5 the hot side is to the left. The hot gas flows through the heat exchanger. There will be a difference between the temperature of the hot gas, $T_{1,\infty}$ and the temperature of the heat exchangers wall, $\frac{\partial T(0,t)}{\partial x}$. That difference, together with the heat transfer coefficient $h_{t,1}$ determines how much heat per square meter the heat exchanger absorbs [7].

Conduction in the metal plate

Conduction describes how heat is transferred through the plate and can be described by Fourier's law of heat conduction as [9]

$$\dot{Q}_{cond} = -kA\frac{dT}{dy} \tag{2.19}$$

where A is the area of the plate, T is temperature and y is the thickness of the plate [7].

Convection from the metal plate to cold working fluid

The last part of the heat transfer is convection. The heat transfer from the hot metal to the colder working fluid is as in the forced convection between the hot gas and the metal plate [7].

2.3.2 Condensation and Evaporation

The most common states of matter on earth are solid, liquid and gas. The state the working fluid is in depends on enthalpy which is the sum of the internal energy and the product of pressure and volume. Enthalpy is described as [9]

$$h = u + pV \tag{2.20}$$

where u is internal energy, p pressure and V volume. In the heat exchanger, pressure and volume are assumed to be constant. Therefore, when heat is added the change in internal energy is equal to the change in enthalpy. While heating the fluid in a heat exchanger, energy is added (heat) which increases the enthalpy, and this may cause a change in state. The correlation between enthalpy and state can be seen in Figure 2.6. In a WHR-system the fluid is present as gas, liquid and in between those two states. The state when the fluid is evaporating or condensing is called two-phase.

The amount of the working fluid that is in gas state is modeled as x

$$x = \frac{m_{gas}}{m_{gas} + m_{liquid}} \tag{2.21}$$

where m_{gas} is the mass of the gas and m_{liquid} is the mass of the liquid in the two phase fluid. A heat exchanger containing working fluid is pictured in Figure 2.7. The working fluid enters the heat exchanger as liquid and leaves as gas. The line between $1 - x$ and x shows how much of the working fluid that is evaporated and how much that is still in a liquid phase. This line will be pushed closer or further away from the heat exchangers exit depending on the enthalpy level in the working fluid while entering the heat exchanger and how much heat that is transferred to the working fluid [9].

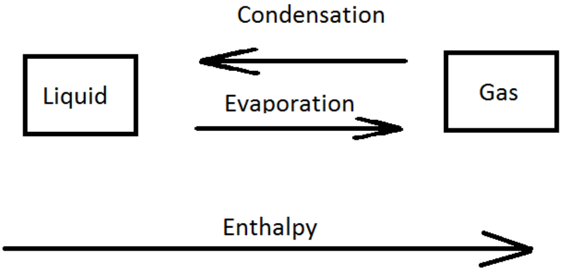


Figure 2.6. Liquid, gas and the two phase state that exist during evaporation and condensation are states the working fluid appears in during the heat exchange. The state of the working fluid depends on enthalpy. Increasing enthalpy leads to evaporation if the fluid is a liquid.

2.3.3 Heating

The purpose of heating is to evaporate a liquid to gas. Heating is considered for several different cases which are listed below.

- The working fluid enters the heat exchanger as liquid and leaves in the same phase.
- The working fluid enters the heat exchanger as liquid and leaves in two-phase.
- The working fluid enters the heat exchanger as liquid and leaves in gas phase.
- The working fluid enters the heat exchanger in two-phase and leaves in two-phase.
- The working fluid enters the heat exchanger in two-phase and leaves in gas phase.
- The working fluid enters the heat exchanger in gas phase and leaves in gas phase.

The process of evaporating a liquid to gas is schematically shown in Figure 2.8. The figure shows how a working fluid enters the heat exchanger as liquid and leaves as gas. The process is divided into three blocks where each block represents the working fluid in a certain state. The working fluid is heated up in the first block. Then it is vaporized to gas in the next block. In the last block, the working fluid is heated up to temperatures above its boiling point. The white lines symbolizes liquid and the grey lines gas.

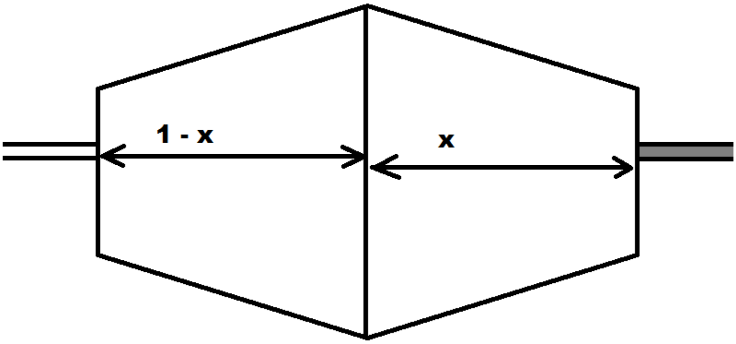


Figure 2.7. A heat exchanger containing working fluid. The working fluid enters the heat exchanger in liquid state and leaves as gas. The variable x indicates how much of the working fluid that has changed state from liquid to gas. One of the biggest challenges with the modeling is to handle the differences in temperature and pressure caused by different values of x.

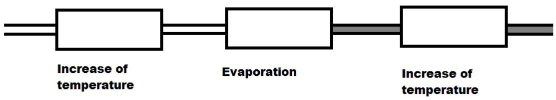


Figure 2.8. A working fluid enters the heat exchanger as liquid and leaves as gas. The process is divided into three blocks where each block represents heat in a certain state of matter. The working fluid is heated up in the first block. Then it is vaporized to gas in the next block. In the last block, the working fluid is heated up to temperatures above its boiling point. The white lines symbolizes liquid and the grey lines gas.

2.3.4 Modeling

The heat transfer in the heat exchanger is assumed to be isobaric. This means that the pressure after the heat transfer is the same as the pressure before. This assumption is valid for an ideal heat exchanger, and is probably not completely true. The change in pressure is though here very small, approximately 2-3%, and thus motivates this assumption. Further, the conduction in the metal is neglected and the temperature on one side of the metal plate is assumed to be the same as the temperature on the other side. The metal plate is very thin, which causes a very quick conduction process. In case of a thicker metal plate, the conduction may not be neglected.

Heat transfer in single phase

The heat transfer process of heating while in a state to the same state can be described by the following steps: Heat transfer between the hot gas and the metal is modeled as [26]

$$\dot{q}_1 = h_{t,1} A (T_{gas} - T_{metal}) \tag{2.22}$$

The temperature change in the hot gas due to heat transfer to the metal is modeled as [26]

$$\delta T = \frac{\dot{q}_1}{\dot{m}_{gas} c_{p,gas}} \tag{2.23}$$

where $\dot{m}_{gas}$ and $c_{p,gas}$ is the mass flow and heat capacity for the hot gas. The temperature change in the metal plate due to heat transfer to the metal is modeled as [26]

$$\delta T = \frac{\dot{q}_1}{m_{metal} c_{p,metal}} \tag{2.24}$$

where m_{metal} is the mass of the metal and and $c_{p,metal}$ its heat capacity. The heat transfer between the metal and the working fluid is modeled as [26]

$$\dot{q}_2 = h_{t,2} A (T_{metal} - T_{wf}) \tag{2.25}$$

The temperature change in the working fluid due to heat transfer from the metal is modeled as [26]

$$\delta T = \frac{\dot{q}_2}{\dot{m}_{wf} c_{p,wf}} \tag{2.26}$$

where $\dot{m}_{wf}$ and $c_{p,wf}$ is the mass flow and heat capacity for the working fluid. The temperature change in the metal plate due to heat transfer from the metal is modeled as [26]

$$\delta T = \frac{\dot{q}_1}{m_{metal} c_{p,metal}} \tag{2.27}$$

The parameters c_p depend on pressure and temperature. The area is constant and the mass flow may vary due to other factors in the system. The heat transfer coefficient $h_{t,x}$ depends on both whether the system is adding or removing heat and on the state. For heat transfer as adding heat and while in either liquid or gas phase, the heat transfer coefficient is modeled as the Dittus-Boelter equation [28]

$$h_t = 0.0243 Re^{0.8} Pr^{0.4} \frac{k}{D} \tag{2.28}$$

where Nu is the Nusselt number which is a ratio of convective to conductive heat transfer. Re is the Reynolds number which is a measure of the ratio of internal forces to viscous forces. Pr is the Prandtl number which is the ratio of kinematic viscosity to thermal diffusivity. Nu, Re and Pr are dimensionless [29]. Re is modeled as

$$Re = \frac{4\dot{m}}{\pi D_H \mu} = \frac{\rho U D_H}{\mu} = \frac{\dot{V} D_H \rho}{\mu A} \tag{2.29}$$

where D_H is the hydraulic diameter, μ is the viscosity and $\dot{V}$ is the volumetric flow. Nu is modeled as

$$Nu = \frac{h_t D_H}{k} \tag{2.30}$$

where k is the thermal conductivity. Pr is modeled as

$$Pr = \frac{\mu c_{p,w}}{k} \tag{2.31}$$

Heat transfer during two-phase-evaporation

The heat transfer between the hot gas and the metal is modeled as Equation 2.22-2.25. The enthalpy change in the working fluid is due to heat transfer from the metal. The temperature does not change during evaporation. The enthalpy change is modeled as [26]

$$\delta h_{wf} = \frac{\dot{q}}{\dot{m}_{wf}} \tag{2.32}$$

The temperature change in the metal plate is due to the heat transfer from the metal follows Equation 2.27.

The heat transfer coefficient depends on parameters for both liquid and gas phase. The equation is called the Klimenko correlation and is modeled as [28]

$$h_{t,evap} = 0.087 Re^{0.6} Pr_l^{1/6} (\frac{\rho_v}{\rho_l})^{0.2} (\frac{k_w}{k_l})^{0.09} \frac{k}{D_H} \tag{2.33}$$

where Pr_l, ρ_l and k_l are Prandtls number, density and thermal conductivity of the working fluid as liquid, just before the evaporation starts. ρ_v is density of the working fluid as gas, just after the evaporation. k_w is the thermal conductivity of the metal plate.

2.3.5 Recuperator

The purpose of the recuperator is to increase the efficiency of the system. The recuperator is used to extract heat from the working fluid after the turbine expander, and adding heat to the working fluid before the heating process starts in the heat exchangers. This will both increase the overheating, but also decrease the amount of necessary cooling in the condenser [26]. The recuperator may be described as follows; the heat transfer between the hot working fluid and the metal plate in the heat exchanger is modeled as [26]

$$\dot{q}_1 = h_{t,1} A (T_{wf,hot} - T_{metal}) \tag{2.34}$$

where the subscript wf, hot refers to working fluid that enters the recuperator after leaving the turbine expander. Temperature change in the hot due to the heat working fluid transfer to the metal is modeled as [26]

$$\delta T = \frac{\dot{q}_1}{\dot{m}_{wf,hot} c_{p,wf,hot}} \tag{2.35}$$

The temperature change in the metal plate due to the heat transfer to the metal is modeled as Equation 2.24. The heat transfer between the metal and the cold working fluid is modeled as [26]

$$\dot{q}_2 = h_{t,2} A (T_{metal} - T_{wf,cold}) \tag{2.36}$$

where the subscript $wf, cold$ refers to working fluid that enters the recuperator after leaving the pump. The temperature change in the cold working fluid due to the heat transfer from the metal is modeled as [26]

$$\delta T = \frac{\dot{q}_2}{\dot{m}_{wf,cold} c_{p,wf,cold}} \tag{2.37}$$

The temperature change in the metal plate is due to the heat transfer from the metal follows Equation 2.27.

2.3.6 Validation

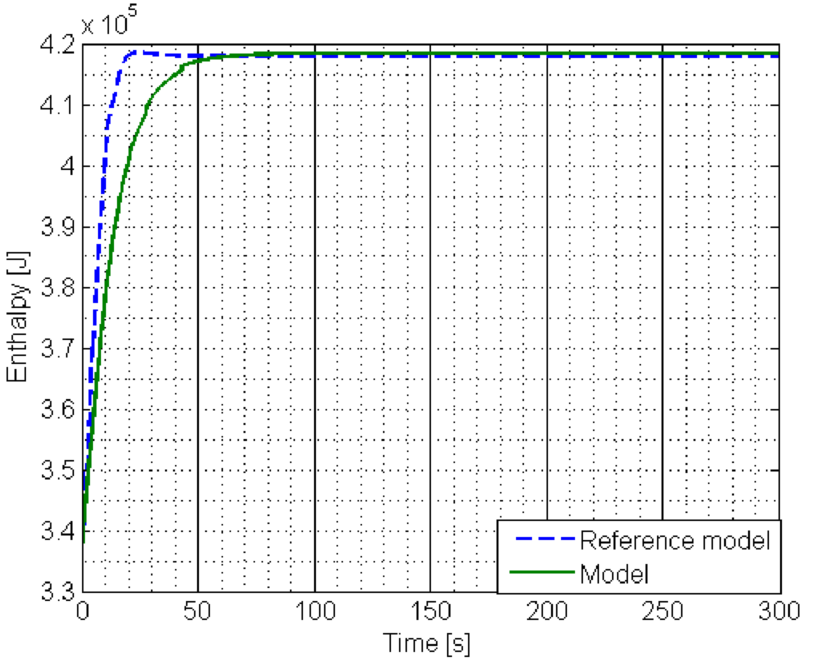
Validation against reference model

The heat exchangers are compared to the reference model with regard to enthalpy and temperature. Working fluid flows through two heat exchangers that are connected to each other. The working fluid enters the first heat exchanger in a liquid state and begins to evaporate. It enters the second heat exchanger as two-phase and leaves as gas. Each heat exchanger has a heating source in form of hot gas. The hot gas flows are not connected to each other. The first heat exchanger has a constant heating source and and the second heat exchanger has a step at time t = 150 seconds where its heating source's mass flow increases. This causes a larger amount of heat to be transferred to the working fluid.

Figure 2.9 shows how the enthalpy increases after the first heat exchanger. There is a peak in difference between the models in the beginning which depends on the time constants of the model. The developed model is slightly slower than the reference model, which in the first seconds causes a change of up to 10 percent. When a stationary operting point is reached, the difference decreases to about 0.2 percent.

Figure 2.10 shows how the temperature increases in the first heat exchanger. The temperature increases fast the first seconds and then reaches a fix level. The fix level is the boiling temperature, and here the evaporation begins. The enthalpy, seen in Figure 2.9, increases during the evaporation but the temperature is constant until the phase change is complete. There is a peak in difference during the first ten seconds, which depends on overshooting in the developed model. The difference is less than 0.01 percent at a stationary level. That difference depends probably of what boiling point the models have. During evaporation, the difference in enthalpy is more important.

Figure 2.11 shows the enthalpy level after the second heat exchanger. The second heat exchanger is dependent of good results from the previous heat exchanger. Otherwise, the starting points in enthalpy and difference would cause larger differences after this heat exchanger. Since the difference in enthalpy is only 0.2 percent, the difference is neglectable for the result after the heat exchanger. The enthalpy first increases to a stationary level. Then, there is a step in mass flow for the hot gas (at t = 150 s) which causes an increase in enthalpy. The difference has, just as in the first heat exchanger, a large peak of up to 10 percent of difference.



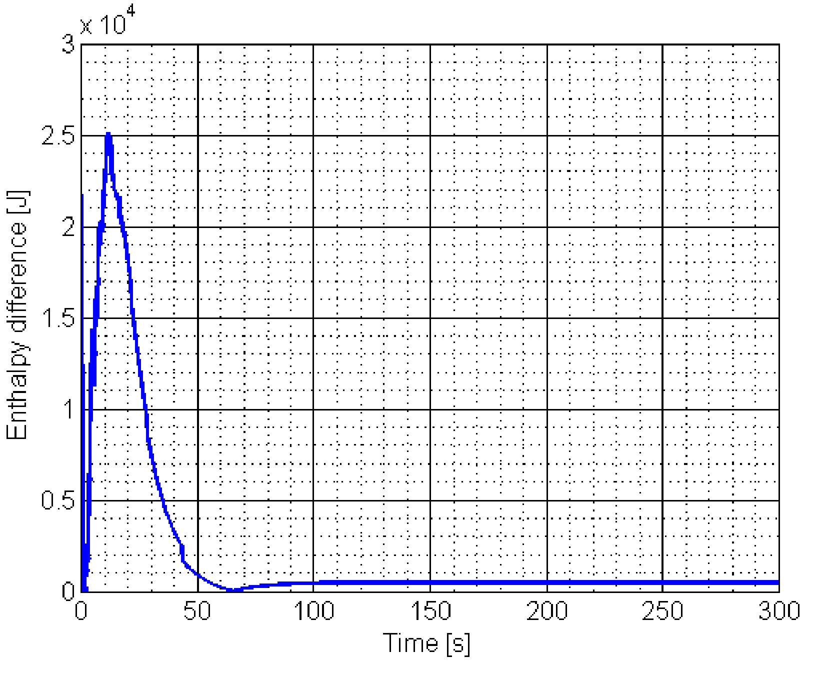
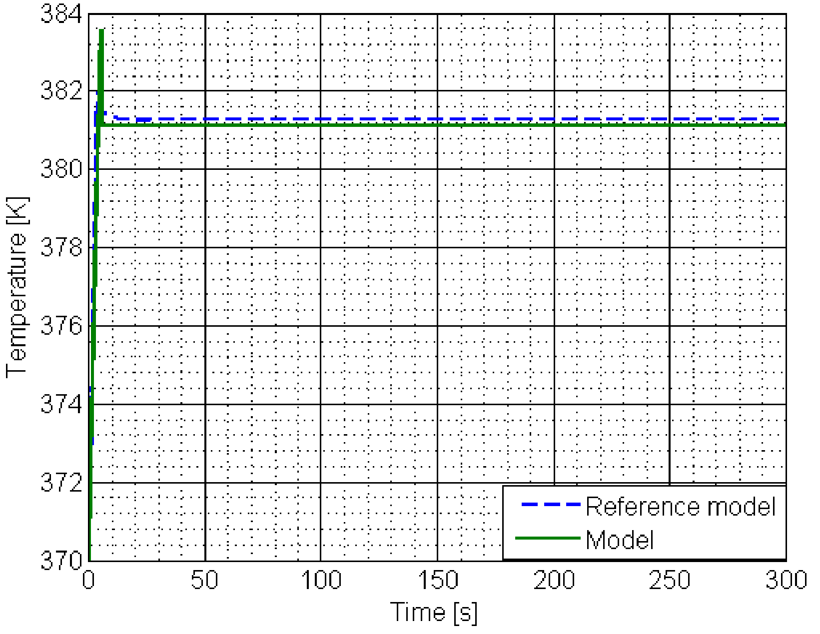


Figure 2.9. Enthalpy level of the working fluid after the first heat exchanger is shown in the upper figure. The heat source in form of hot gas is constant and the working fluid is starting to evaporate in the heat exchanger. The working fluid enters the heat exchanger as liquid and leaves in two-phase-state. The difference between the models are shown in the lower figure. There is a time constant difference in the models, which causes a faster increase of temperature for the reference model before the boiling point. Therefore, the difference is large in the first 50 seconds and then rapidly decreases when the system reaches a stationary level.



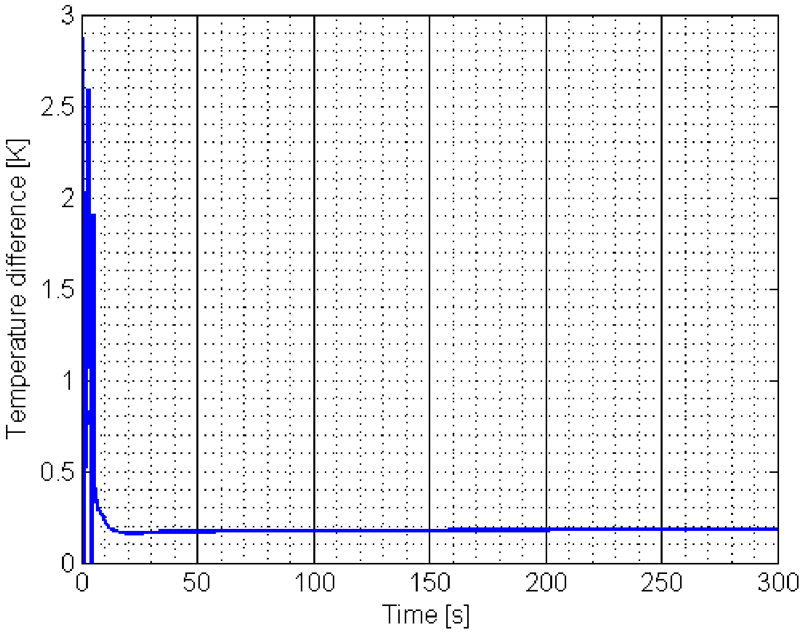


Figure 2.10. Temperature of the working fluid after the first heat exchanger is shown in the upper figure. The heat source in form of hot gas is constant and the working fluid is starting to evaporate in the heat exchanger. The working fluid enters the heat exchanger as liquid and leaves in two-phase-state. There is an overshoot in the developed model, but then the temperatures are constant at boiling temperature. There is a time constant difference in the models, which causes a faster increase of temperature for the reference model before the boiling point. Therefore, the difference, which is seen in the lower figure, in large in the first 10 seconds and then almost disappears during two-phase state.

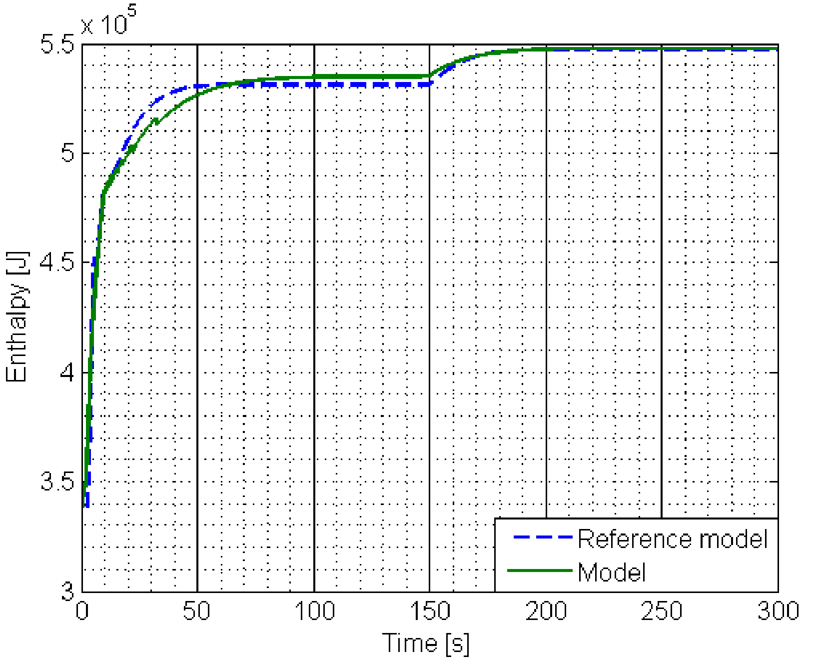
The difference at the stationary level before the step is about 0.7 percent. The difference at the stationary level after the increase in hot gas mass flow is less than 0.2 percent. This is a difference that will give a very small impact further results in the model.

Figure 2.12 shows the difference in temperature after the second heat exchanger. The difference in stationary level before the step is, as the difference in enthalpy, about 0.7 percent. The temperature difference at the stationary level after the step in hot gas mass flow is less than 0.2 percent.

To improve the model and decrease the temperature difference, the model describing the hot gas which is the heating source could be developed. The hot gas is modeled with constant heat transfer and c_p due to lack of data. The heat transfer and c_p are most likely not constant, and this might be one of the causes to the difference in temperature after the heat exchangers.

The data points used to model c_p, k, μ and ρ and the enthalpy-temperature correlation are collected from [21]. The maximum temperature is 500 K in [21]. The working fluid used in this WHR-system, R245fa, are not supposed to operate in temperature above 500 K for a long time [12], but it might happen for shorter time intervals. This means that when the working fluid is warmer than 500 K, the data is no longer valid. This was not the case for this comparison, but might affect other comparisons.

In the developed model, the metal between the flows are assumed to have the same temperature as the working fluid at the start of the simulation. The metal temperature of the reference model at start is not known, but if they are not equal, this can be one of the causes to the small delay in the developed model compared to the reference model.



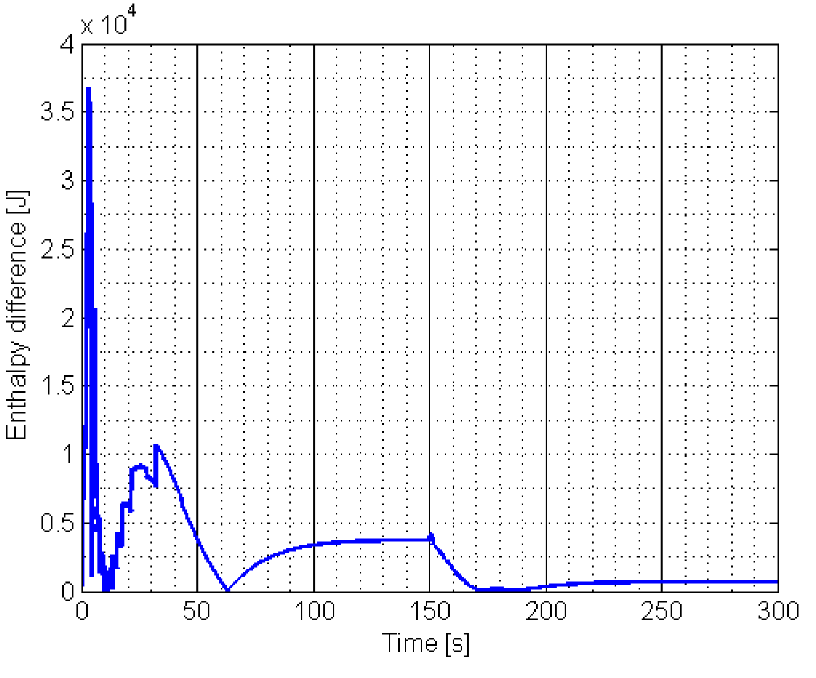


Figure 2.11. The enthalpy of the working fluid after the second heat exchanger is shown in the upper figure. The hot gas which is the heating source has a step with increased mass flow at time $t = 150$ s which causes a larger heat transfer to the working fluid. The working fluid enters the heat exchanger in two-phase state and leaves as gas. The difference, seen in the lower figure, has a peak since in the first 10 seconds because of different time constants in the models. After that, the model stabilizes at its stationary level before the step, and the difference is about 0.7 percent. After the increase of mass flow, the difference decreases to less than 0.2 percent.

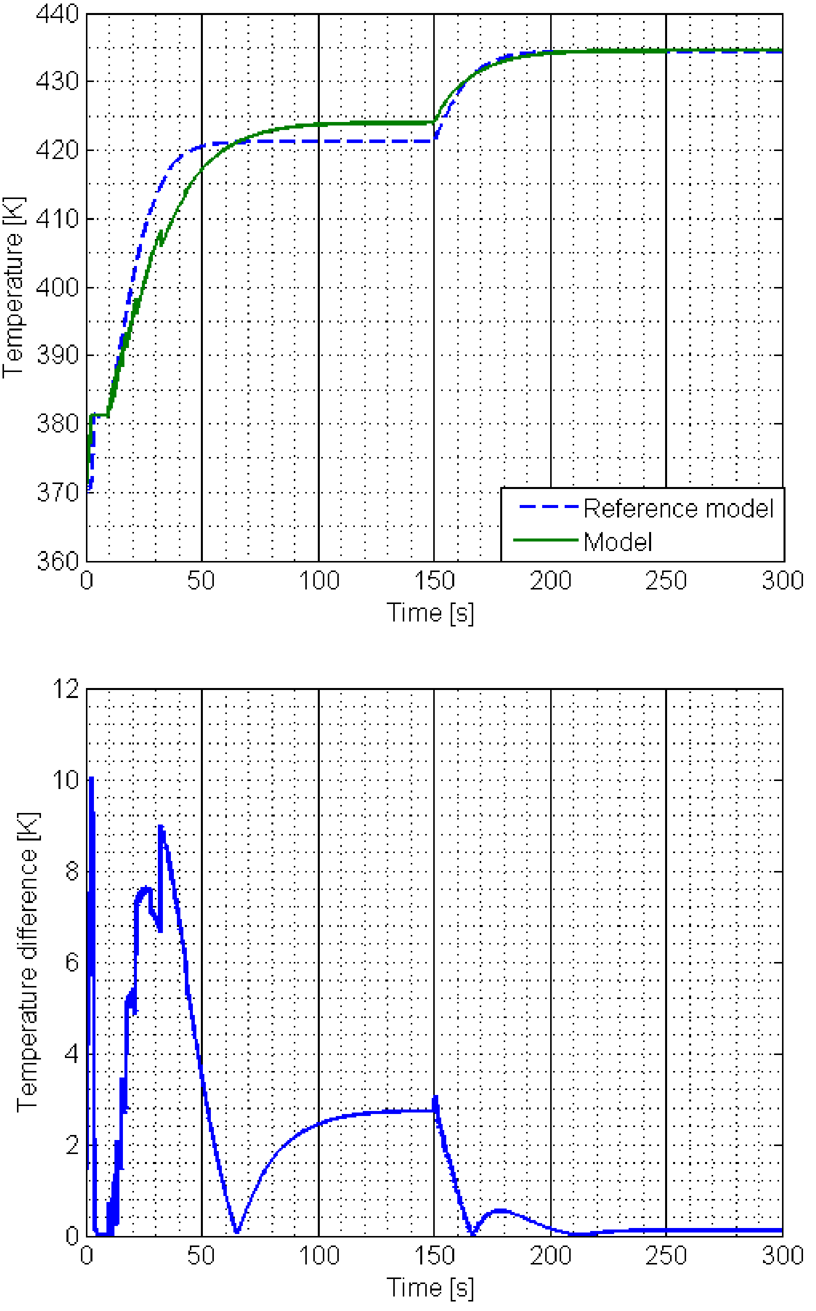


Figure 2.12. The temperature of the working fluid after the second heat exchanger is seen in the upper figure. The hot gas which is the heating source has a step with increased mass flow at time $t = 150$ s which causes a larger heat transfer to the working fluid. The working fluid enters the heat exchanger in two-phase state and leaves as gas. The difference, seen in the lower figure, has a peak since in the first 40 seconds because of different time constants in the models. After that, the model stabilizes at its neglectable before the step, and the difference is about 0.7 percent. After the increase of mass flow, the difference decreases to less than 0.2 percent.

Validation against cell data

A heat exchanger has been compared to measurement data. Measurement data does not include working fluid mass flow, which means the working fluid mass flow is modeled. The mass flow for the hot gas is measurement data. The working fluid enters the heat exchanger as liquid at first and as gas later in the simulation. The working fluid leaves the heat exchanger as gas. All input variables varies during the simulation and there is a big step in mass flow for the hot gas which causes a big increase in heat transfer. The resulting temperature after the heat exchanger can be seen in Figure 2.13. There is a large difference during the step in hot gas mass flow (which can be seen in Figure 2.14) which causes a delay in the modeled output. During the end of the simulation, the difference between measurement data and simulation model is about 3 percent.

The results seen in Figure 2.9 - 2.13 shows that it is possible to model heating and phase change from liquid to gas in SIMULINK.

2.3.7 Simulink model

The heat exchangers are modeled as a division into different sections. The sections are divided as seen in Figure 2.15. The sections are numbered from 1:m, where m is the total number of sections. The working fluid temperature and enthalpy level when entering the heat exchanger is the temperature and enthalpy of the working fluid in section 1. The temperature and enthalpy are increased, and the resulting values are the temperature and enthalpy that enters section 2. The temperature and enthalpy from section m are the resulting temperature and enthalpy. Different number of sections were tested, and while the number increased, the simulated pressure better matched the reference model pressure. First were 10 sections tested, but an increase to 20 gave a decreased difference between the reference model and the developed model. Higher numbers of sections than 20 did not give any differences in results compared to 20 sections.

The heat exchanger is modeled with STATEFLOW, which extends SIMULINK with state charts and flow graphs. An overview of the model is shown in Figure 2.16. Input variables to the model are temperature, mass flow and enthalpy of the working fluid and the temperature and mass flow of the hot gas. Output variables are the temperature, enthalpy and degrees of overheating of the working fluid and the temperature of the hot gas.

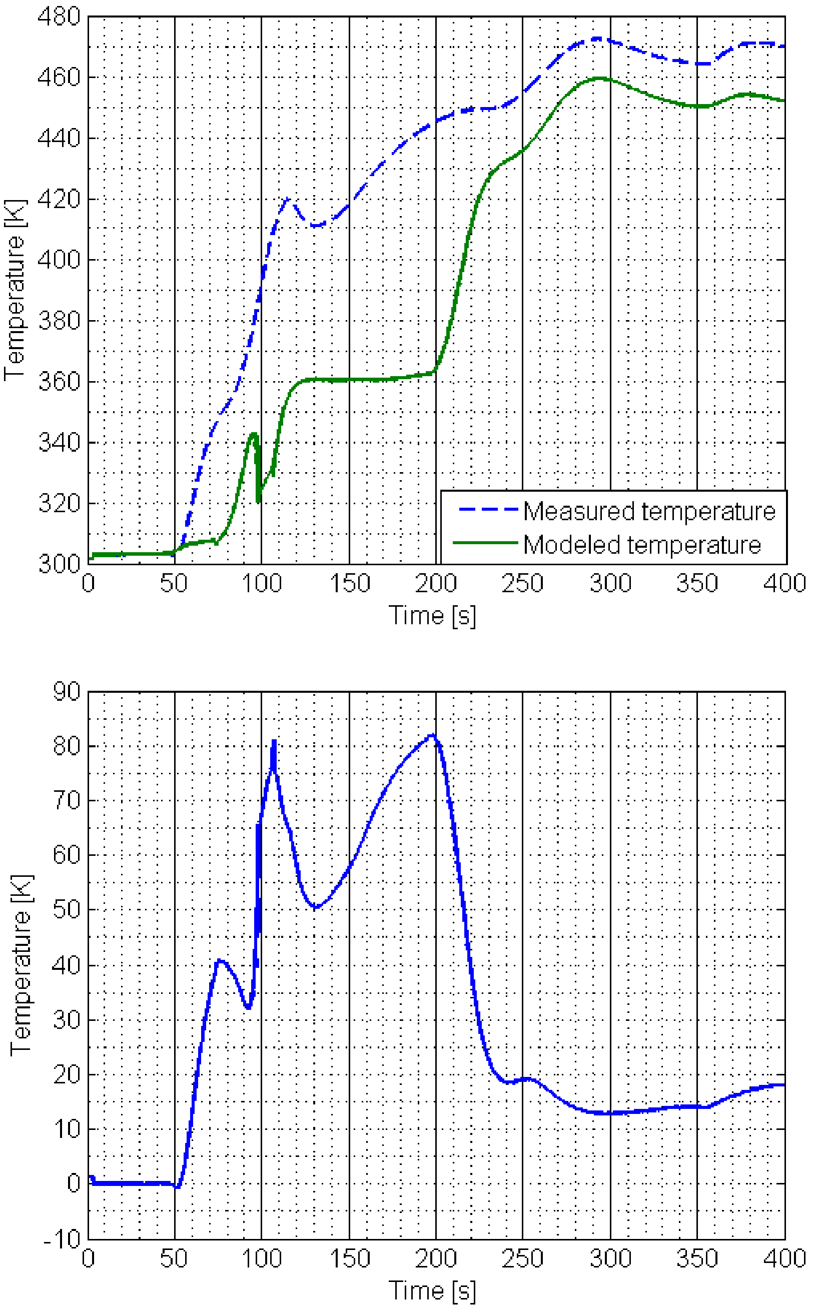


Figure 2.13. Output temperature from a heat exchanger is shown in the upper figure. The input working fluid is liquid during the first second and then gas. There is a step in mass flow between time $t = 75$ s and time $t = 100$ s. The developed model is compared to measurement data from cell. The difference can be seen in the lower figure.

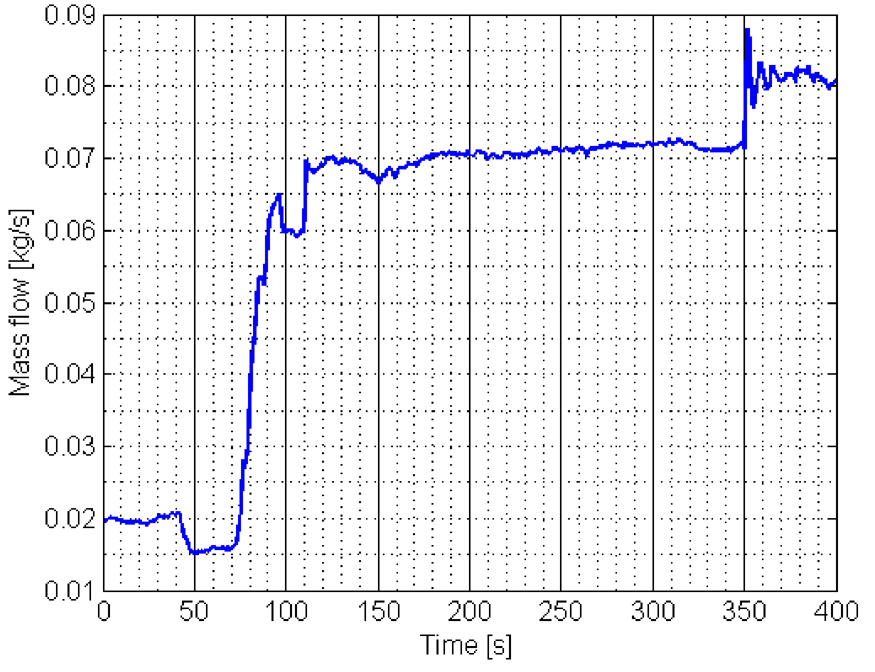


Figure 2.14. Measured input hot gas mass flow to the heat exchanger in Figure 2.13. There is a step in mass flow between time t = 75s and time t = 100 s.

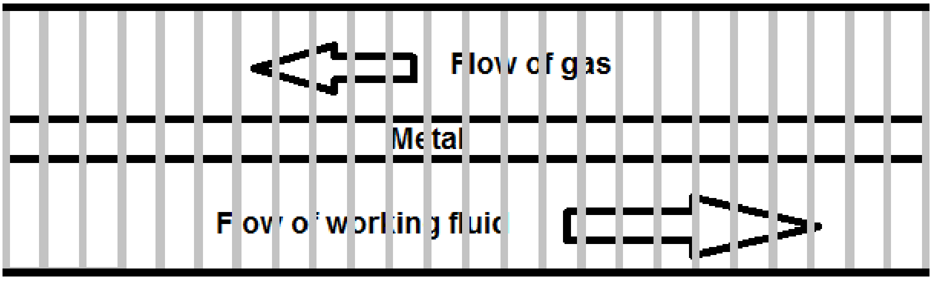


Figure 2.15. The figure shows how the heat exchangers are divided into several parts in the SIMULINK model.

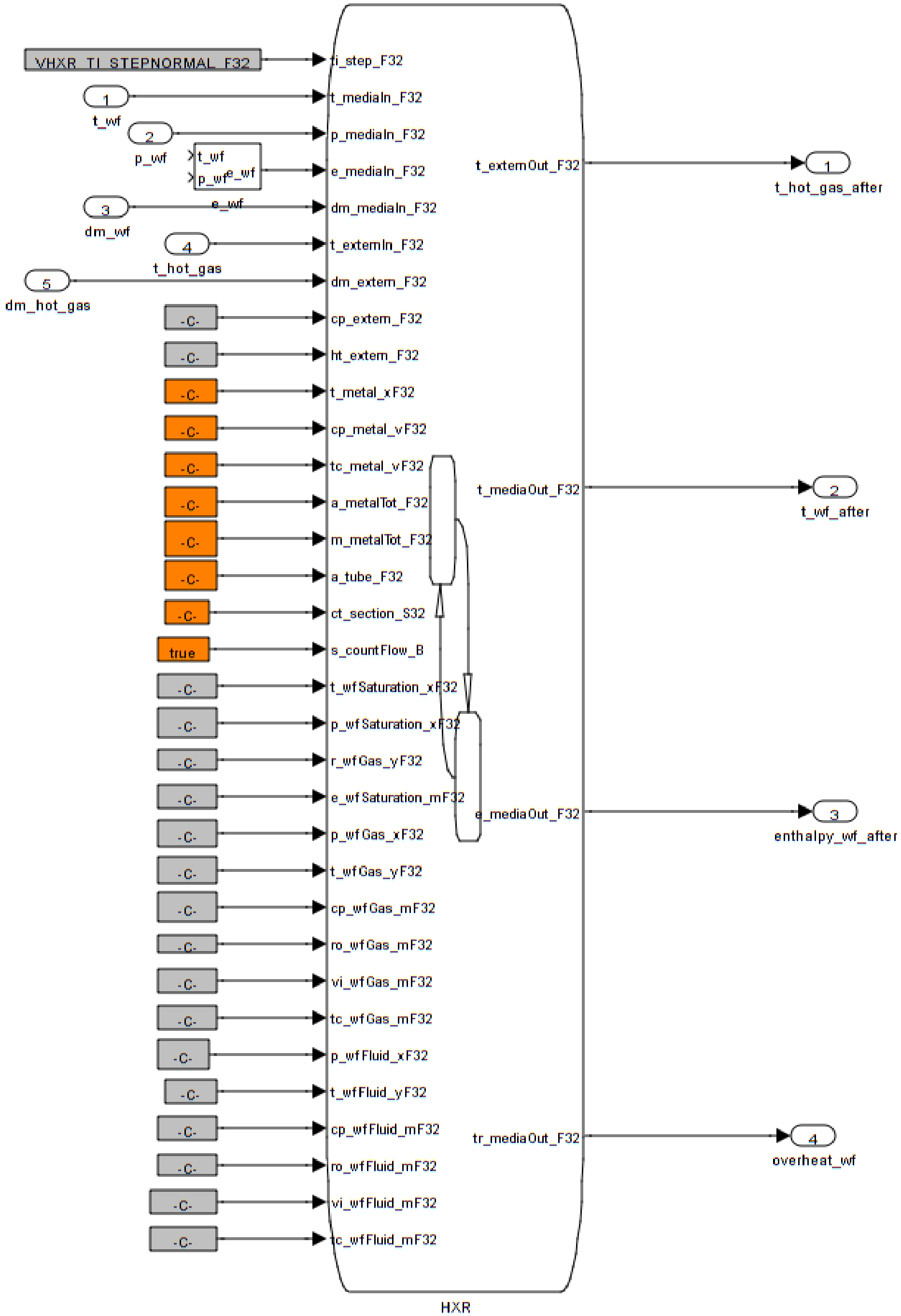


Figure 2.16. The heat exchanger model. Input variables are the temperature, pressure, mass flow and enthalpy of the working fluid and the temperature and mass flow of the hot gas. Output variables are the temperature, enthalpy and degrees of overheating of the working fluid and the temperature of the hot gas.

2.4 Turbine expander

The purpose of the turbine expander is to extract work from the hot working fluid.

2.4.1 Modeling

The mass flow before and after the turbine expander is constant but not the volumetric flow. This causes a pressure change that is modeled by the relation [30]

$$p_{out} = p_{in} r \tag{2.38}$$

where p_{in} is the pressure of the working fluid entering the turbine expander and r is the ratio of pressure. The ratio of pressure is modeled with a map that include the inputs [30]

$$r = r(p_{in}, T_{in}, n, \dot{m}) \tag{2.39}$$

where T_{in} and $\dot{m}$ are the temperature and mass flow of the working fluid entering the turbine expander and n is the rotational speed. Since the process is isentropic, the temperature after the turbine expander is modeled as [31]

$$T_{out} = T_{in} \frac{p_{out}}{p_{in}}^{\frac{k-1}{k}} \tag{2.40}$$

where k is [31]

$$\frac{c_p}{c_v} \tag{2.41}$$

where c_p is the specific heat capacity of the working fluid at constant pressure and c_v is the specific heat capacity of the working fluid at constant volume. The power output of the turbine expander is [31]

$$\dot{W} = \eta \dot{m}(h_{in} - h_{out}) = \eta c_p \dot{m}(T_{in} - T_{out}) \tag{2.42}$$

where h_{in} and h_{out} are the input and output enthalpy of the system and η is modeled with a map that includes the inputs [31]

$$\eta = \eta(p_{in}, T_{in}, n, \dot{m}). \tag{2.43}$$

2.4.2 Validation

The turbine expander is compared to the reference model. The turbine expander affects the pressure and temperature of the working fluid and those two are compared to the reference model.

Comparison to reference model

Pressure and temperature after the turbine have been compared between the developed model and the reference model. In Figure 2.17 the pressure after the turbine is seen. The simulation is performed during start of the system, and at first, there is a large difference in between them. After about 150 seconds, when

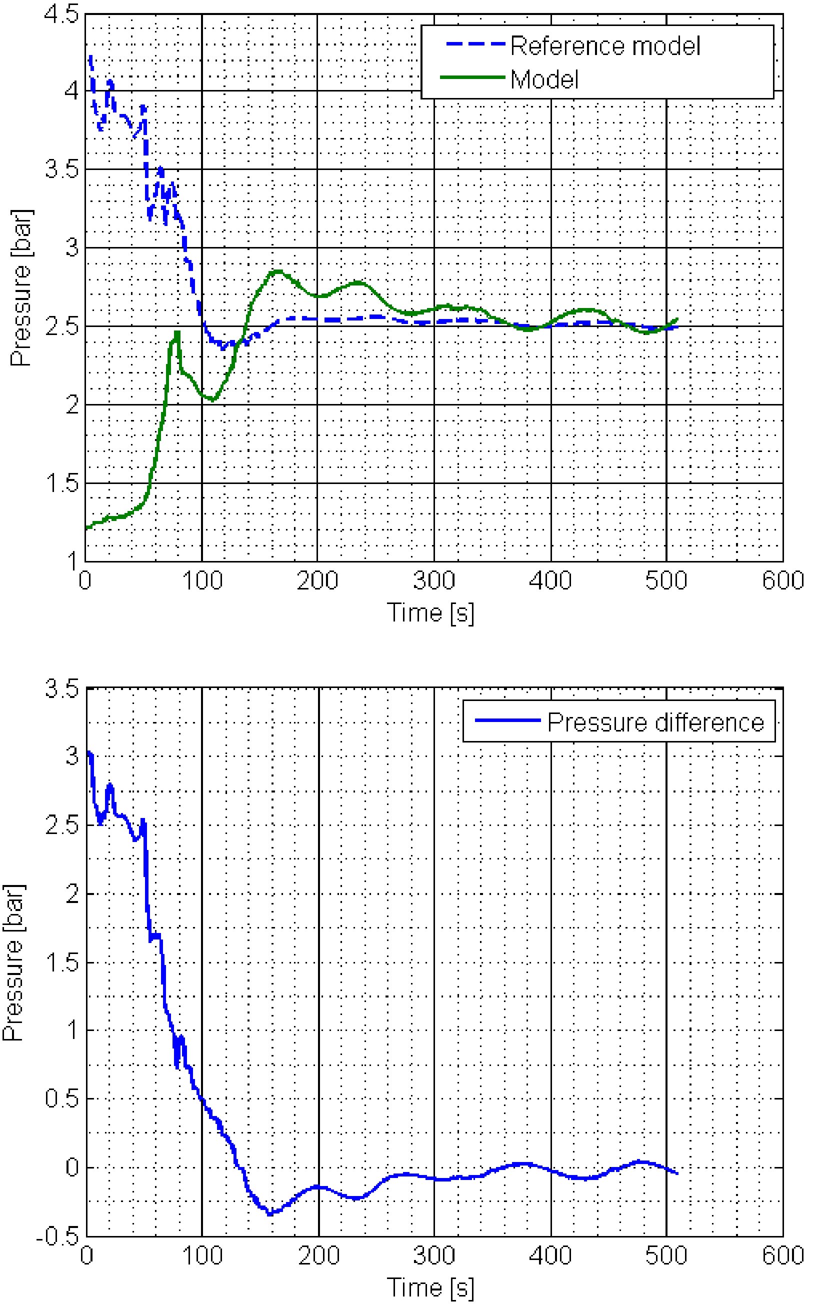


Figure 2.17. Pressure after the turbine is seen in the upper figure and the difference between the models in the lower figure. The working fluid entering both models has the same properties. In the turbine expander, the volumetric flow changes that causes a change in pressure. The comparison is made during start of the system and when the system is stationary operating after the start.

the system is stabilized, the difference is close to zero.

Figure 2.18 compares the temperature after the turbine between the developed model and the reference model after the turbine. The difference is small during the start of the simulation, but after about 200 seconds, the difference is about 2 percent. The difference depends probably on the assumption that the process is isentropic which means that there is no heat exchange with the environment.

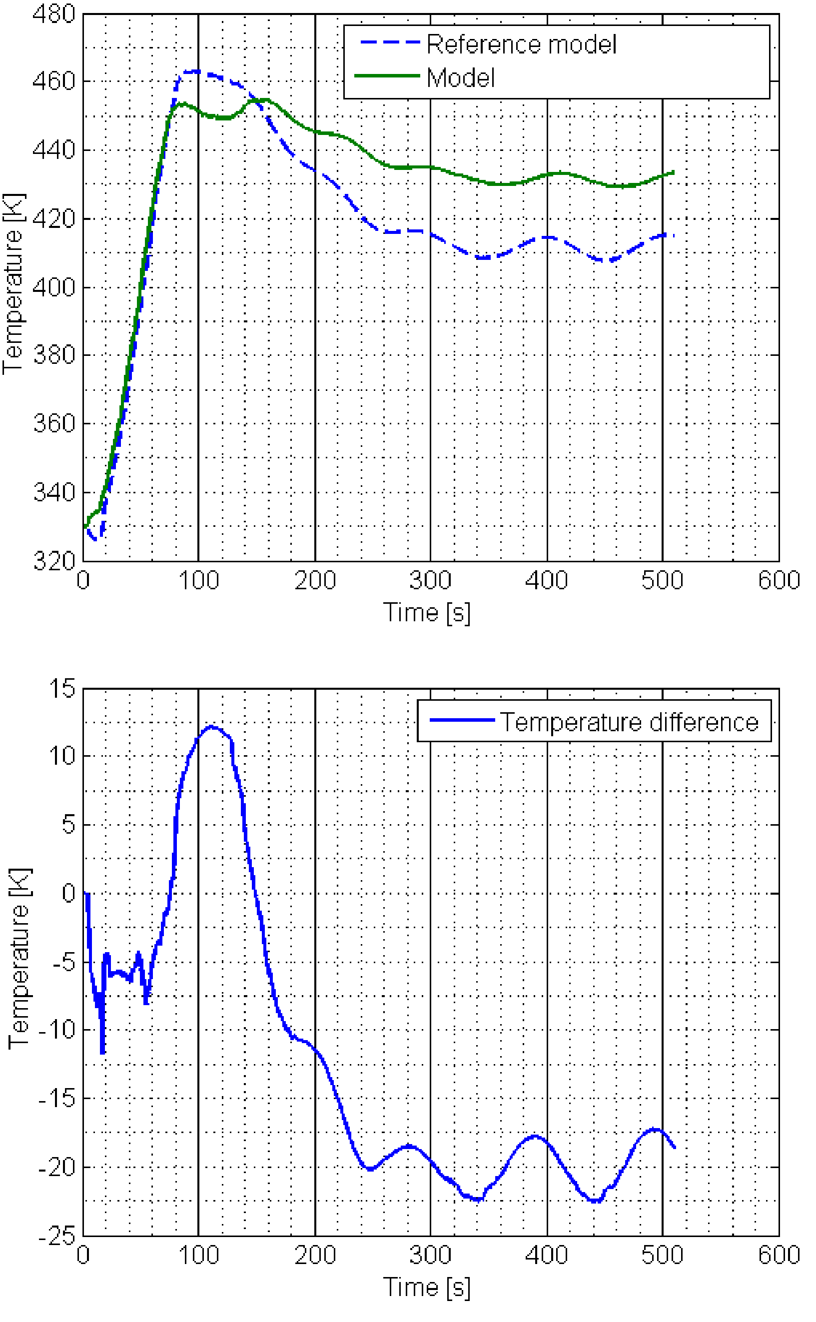


Figure 2.18. Temperature after the turbine is seen in the upper figure, and the difference between the models in the lower figure. The working fluid entering both models has the same properties. In the turbine expander, the volumetric flow changes that causes a change in pressure. The comparison is made during start of the system and when the system is stationary operating after the start.

Validation against measurement data

Pressure and temperature after the turbine have been compared between the developed model and measurement data. In Figure 2.19 the pressure and temperature after the turbine is seen. The behavior of the measurement data and the developed model are alike, but the values of both the pressure and the temperature differ. This might depend on the maps for c_v and c_p data, which needs to be extended to be more precise. The same is for the pressure ratio map.

2.4.3 Simulink model

The turbine expander is modeled in SIMULINK. Inputs to the model are temperature, mass flow, enthalpy and pressure of the working fluid and rotational speed of the turbine. Outputs are temperature, pressure and enthalpy of the working fluid. The mass flow remains constant.

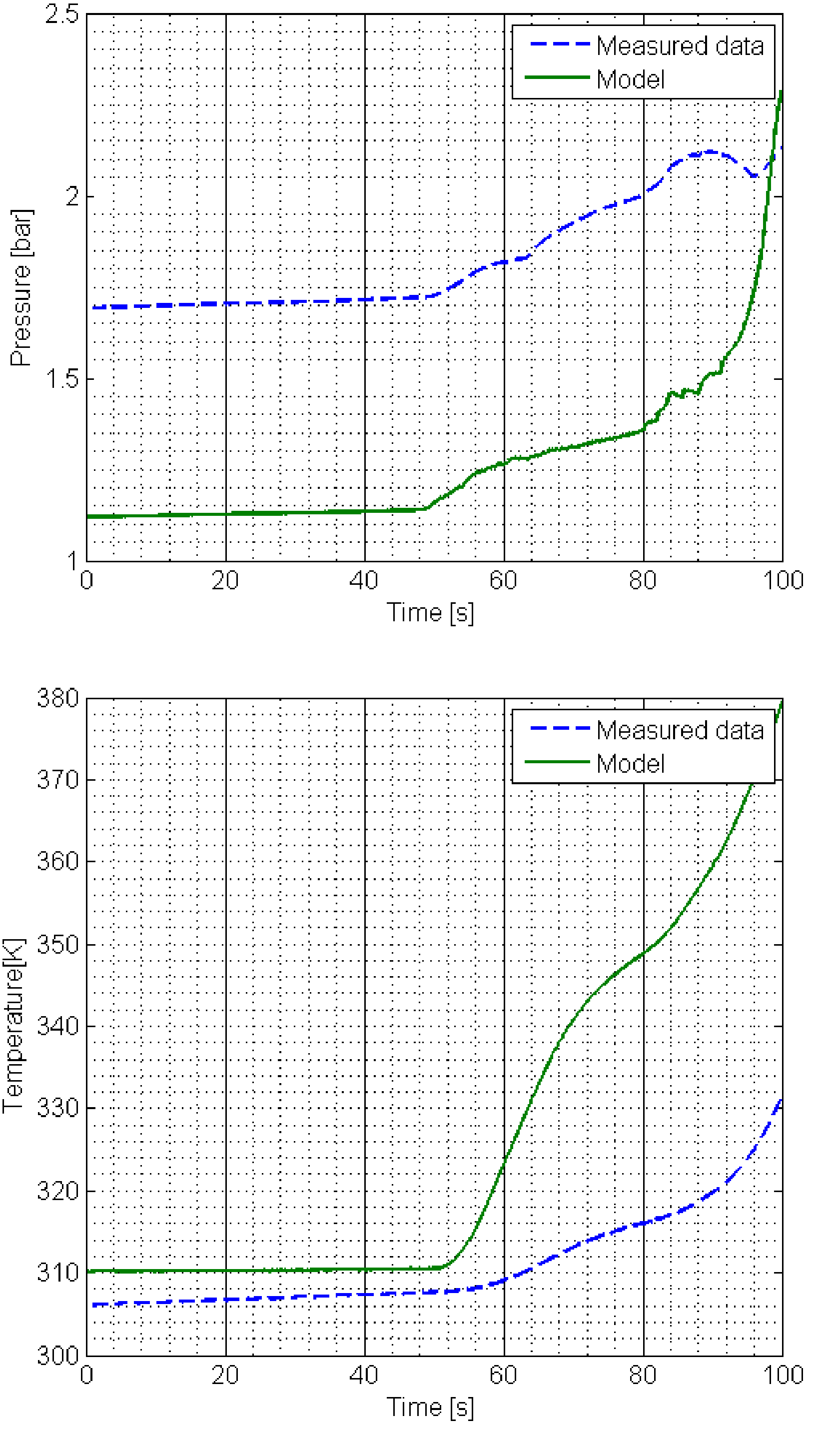


Figure 2.19. Pressure difference between measurement datal and the developed model after the turbine is seen in the upper figure and temperature difference in the lower. The working fluid entering both models has the same properties. In the turbine expander, the volumetric flow changes that causes a change in pressure. The comparison is made during start of the system and when the system is stationary operating after the start.

2.5 Condenser

A condenser is a heat exchanger where cooling instead of heating takes place. Cooling is heat transfer where heat is removed instead of added.

2.5.1 Modeling

Cooling in liquid and gas phase

The same steps are valid in heat transfer for cooling as for heating. When $\dot{q}_1$ is negative, heat will be added to the cooling fluid and removed from the working fluid. The heat transfer coefficient is slightly different for cooling compared to heating, but the Dittus-Boelter Equation 2.28 is still valid, with the exception of the exponent to Pr that is 0.3 instead of 0.4 [28].

$$h_t = 0.0243 Re^{0.8} Pr^{0.3} \frac{k}{D} \tag{2.44}$$

Cooling during condensation

Heat transfer during condensation occurs as heat transfer during evaporation, but enthalpy is removed instead of added. The heat transfer coefficient for condensation differs from both the Klimenko Correlation and the Dittus-Boelter Equation. The heat transfer coefficient for condensation is given by the Shah correlation [28]

$$h_{t,cond} = Nu \frac{(1-x)^{0.8} + 3.8x^{0.76}(1-x)^{0.4}}{p_{rd}^{0.38}} \frac{k}{D_H} \tag{2.45}$$

where x is the percent of gas in the working fluid and p_{rd} is the reduced pressure.

2.5.2 Validation

Validation against reference model

The condenser is compared to the reference model with regard to enthalpy and temperature. The working fluid mass flow and in temperature is constant, and the cooling source has a constant temperature but a step that increases the mass flow at time $t = 100$ s. This causes a larger heat transfer from the working fluid to the cooling refrigerant, which decreases the enthalpy and temperature of the working fluid. The working fluid enters the condenser as gas and leaves as liquid.

Figure 2.20 shows the enthalpy after the condenser. The developed model is faster than the reference model which causes a large difference the first 50 seconds. After the mass flow step the difference in enthalpy is 3 percent.

Figure 2.21 shows the temperature after the condenser. There is a rather large difference in temperature in the first 50 seconds, but then it reaches its stationary level and the difference in temperature is about 1 percent. The larger difference during the first 50 seconds depends probably both on the time constants of the

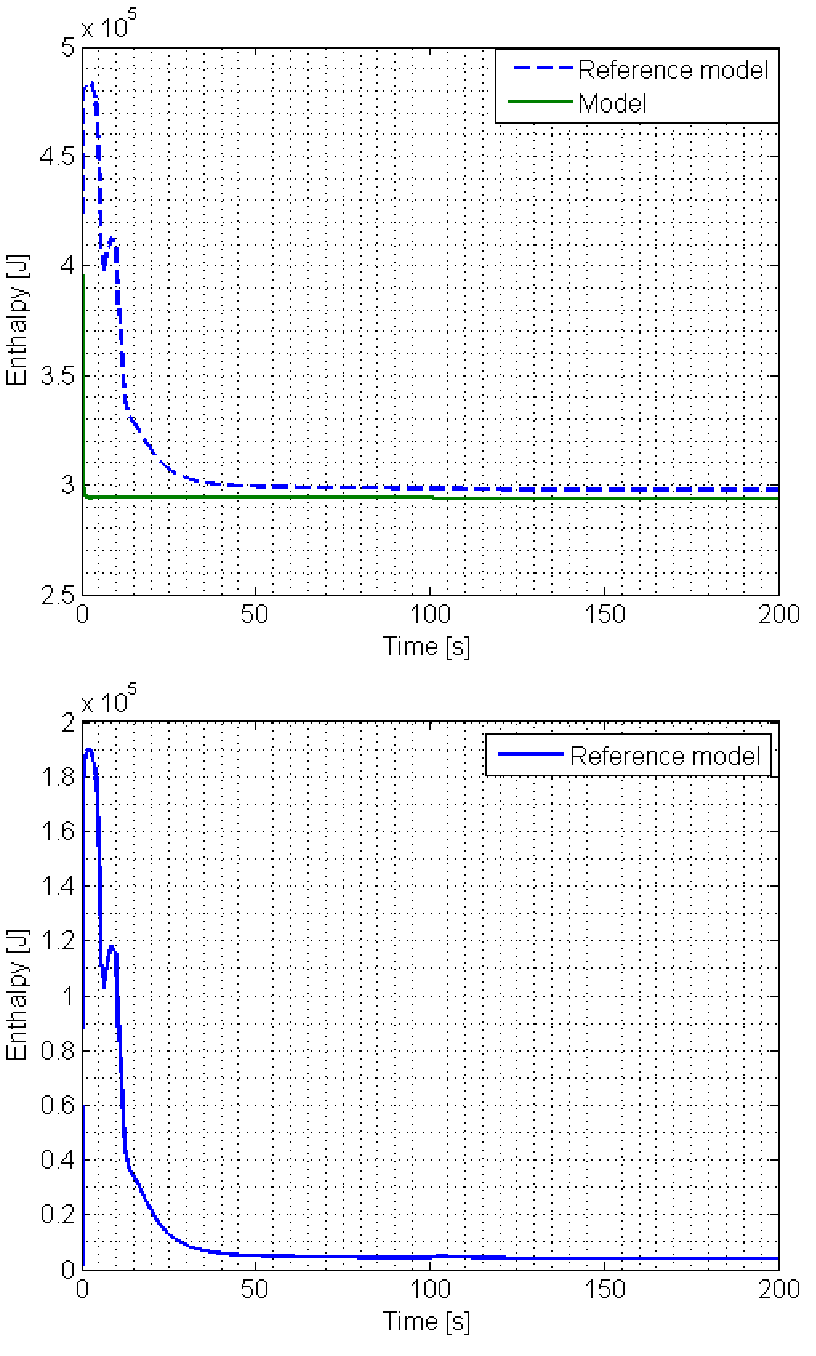


Figure 2.20. Enthalpy of the working fluid after the condenser is seen in the upper figure. The working fluid enters the condenser as gas and leaves as liquid. There is a step in mass flow for the cooling refrigerant at time t = 100s. The working fluid mass flow is constant. The difference, seen in the lower figure, is at its largest at the first 50 seconds, and then decreases to about 3 percent.

models, but also on the data used in the developed model. The data is more precise for higher temperatures due to limitations of hard ware. This causes a larger difference in temperature than for the heat exchangers.

Figure 2.20 - 2.21 shows that it is possible to model the condensation of the working fluid. It also shows how big the difference between the models is when the developed model completes its condensation before the reference model. There is a difference of more than 30 K when the reference model is in gas state and the developed model has condensed its working fluid. When the working fluid in both models are condensed, the temperature difference is about 3 K.

Validation against data from cell

Temperature after the condenser has been compared between the measurement data and the developed model. The input temperature to the model is the measured input temperature. There are no mass flow sensors for the working fluid which means that the mass flow is modeled. The mass flow into the condensor is modeled as the system mass flow. The input temperature is the temperature of the system mass flow, after the recuperator, which means that it could be a difference between the temperature from the tank and the temperature of the mass flow after the recuperator.

There is a difference between the models that increases during the simulation. This difference might depend on the assumption that the heat transfer coefficient and the specific heat capacity of the cooling refrigerant are constant. That would also explain why this difference is not seen in for example Figure 2.21 where the properties of the hot gas were constant.

2.5.3 Simulink Model

The condenser was divided into sections as the other heat exchangers. More sections were needed to get a good cooling result, but after $m = 40$, no change in result were seen. The larger number of sections compared to the heat exchangers could depend on the lack of data points for lower temperatures. The condenser is modeled in two steps that can be seen in Figure 2.23. The working fluid is first entering a condenser where the working fluid is condensed from gas to liquid. After the condenser the working fluid enters the sub-cooler. If the working fluid is not completely condensed, the sub-cooler condenses the two phase medium to liquid. Otherwise, the sub-cooler is used to cool the working fluid below boiling point.

Inputs to the model are temperature, pressure, enthalpy and mass flow of the working fluid and temperature and mass flow of the cooling refrigerant. Outputs are temperature and enthalpy of the working fluid and temperature of the cooling refrigerant.

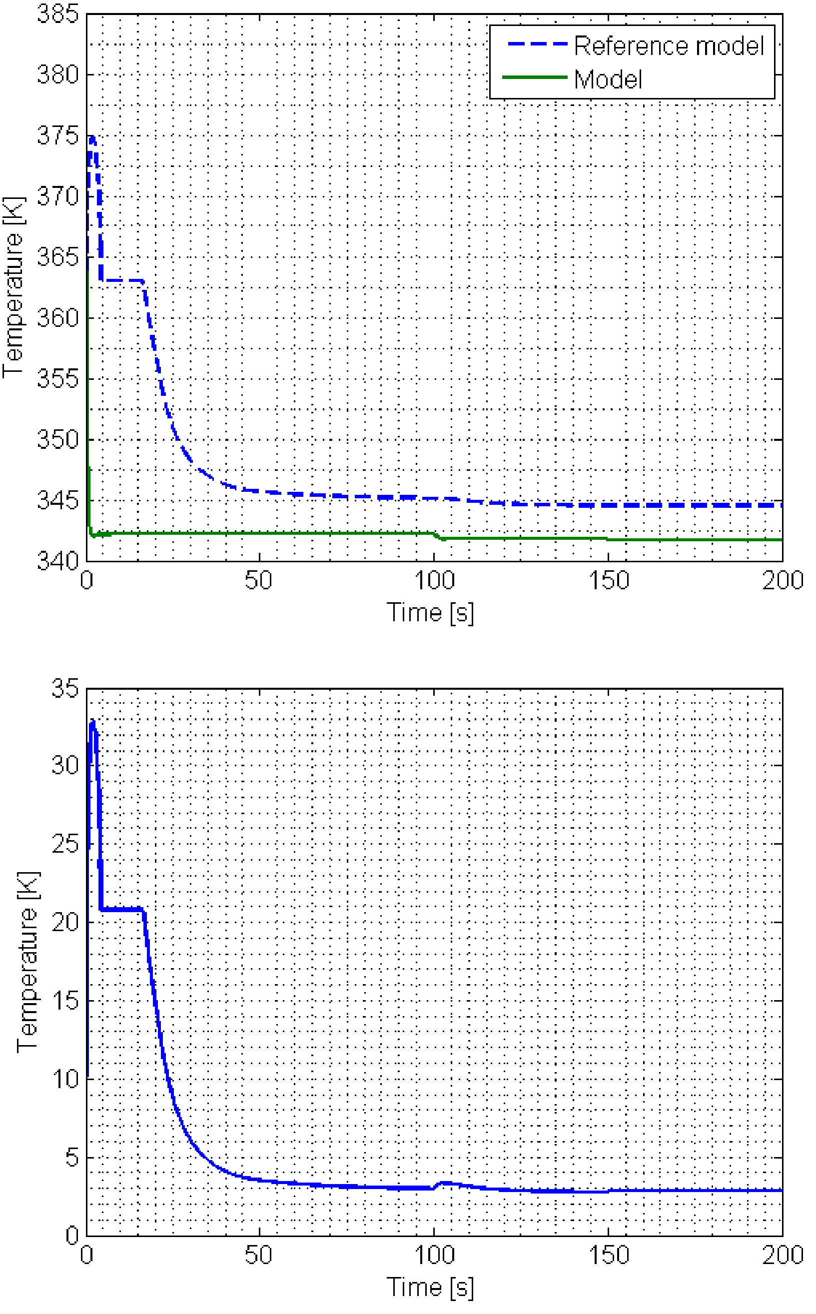
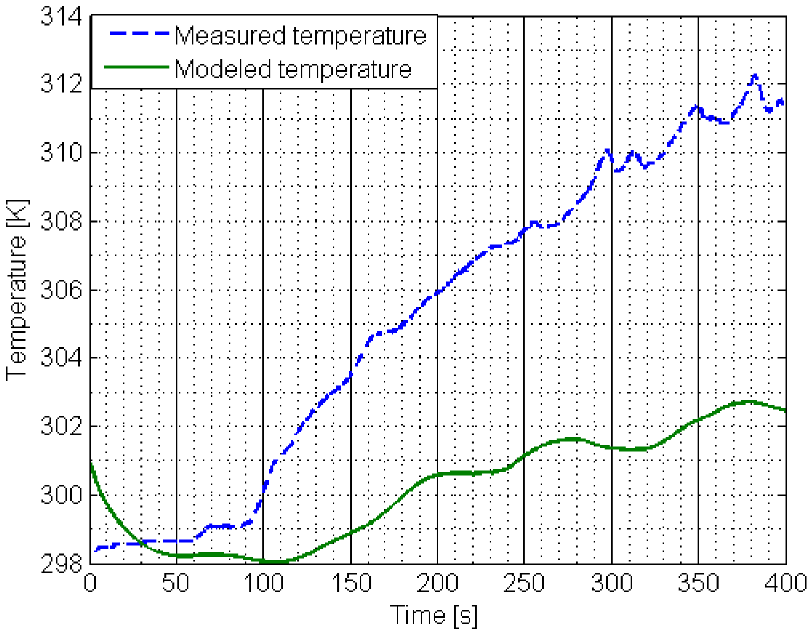


Figure 2.21. Temperature of the working fluid after the condenser is seen in the upper figure. The working fluid enters the condenser as gas and leaves as liquid. There is a step in mass flow for the cooling refrigerant at time t = 100s. The working fluid mass flow is constant. Difference in temperature of the working fluid is seen in the lower figure. The difference is at its largest at the first 50 seconds, and then decreases to about 1 percen



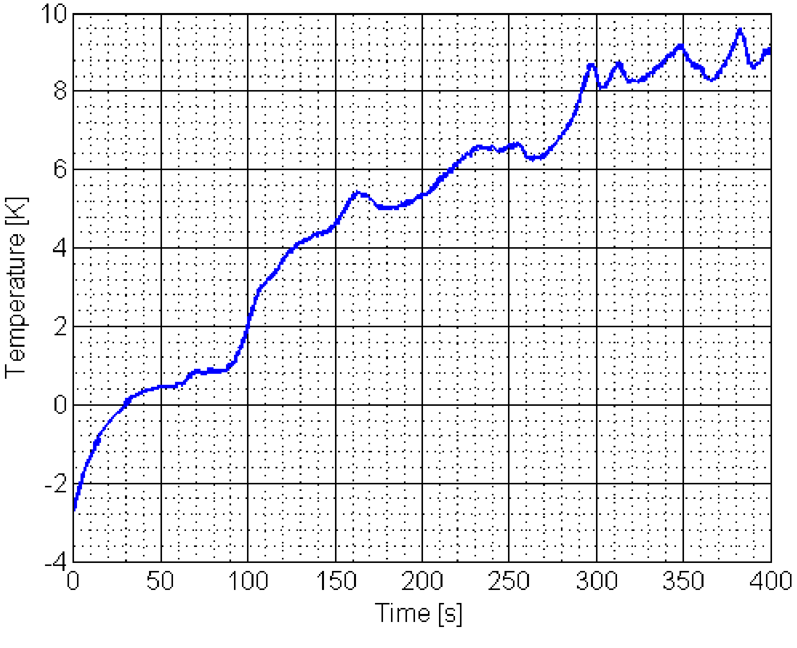


Figure 2.22. Temperature after the condenser is seen in the upper figure. Comparison is made between measurement data and the developed model. Input temperatures for both the working fluid and the cooling refrigerant varies as well as the pressure and the mass flow of the working fluid. The temperature difference increases during the simulation, but the fluctuations in temperature in the reference model can also be seen, but small, in the developed model. The difference, seen in the lower figure, increases during the simulation, which might depend on the assumption of assuming that the heat transfer coefficient and the specific heat capacity of the working fluid are constant.

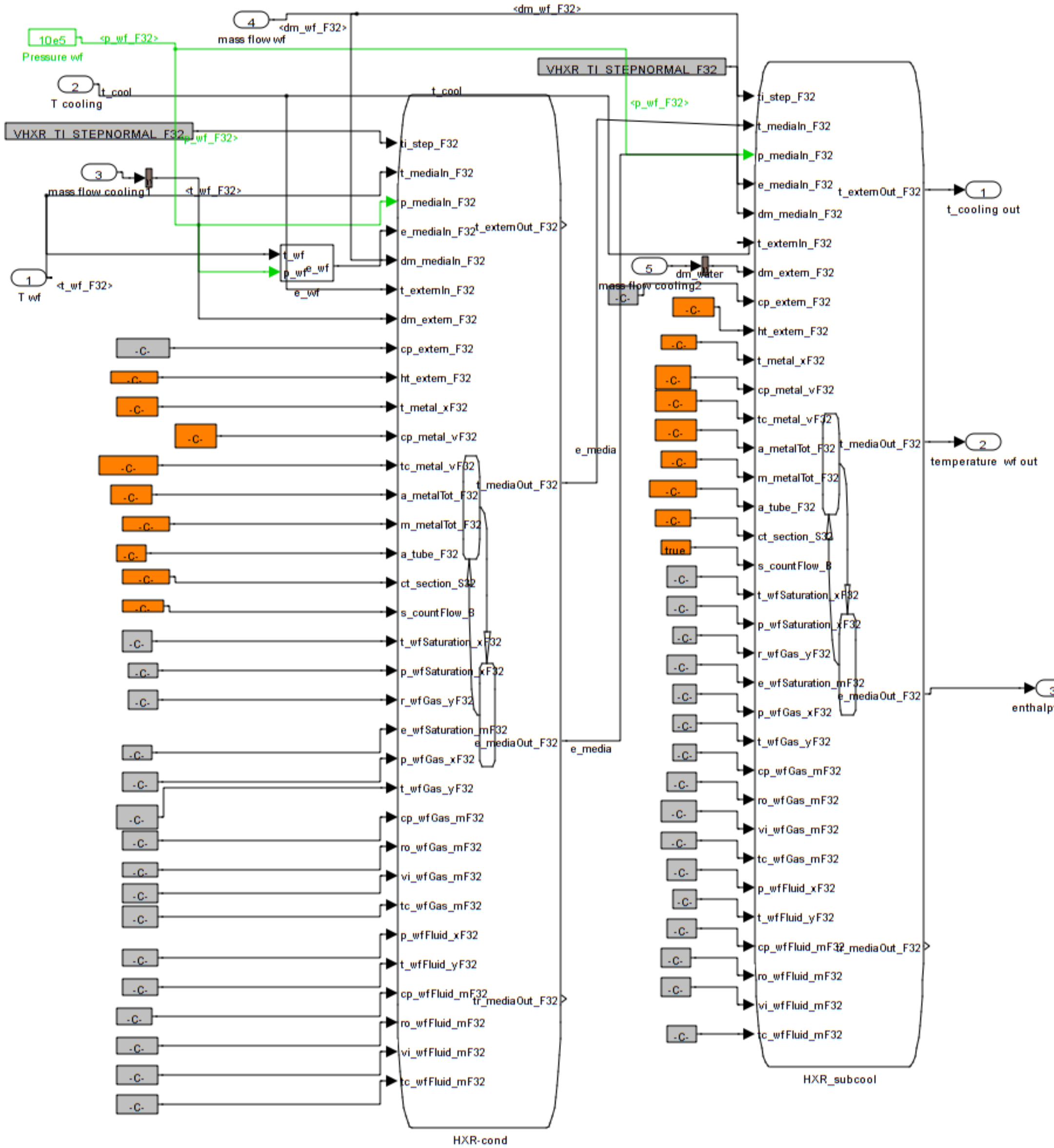


Figure 2.23. The condenser model. Inputs are temperature, pressure, enthalpy and mass flow of the working fluid and temperature and mass flow of the cooling source. Outputs are temperature and enthalpy of the working fluid and temperature of the cooling source.

2.6 Tank

The purpose of the tank is to store working fluid. The control system may then use the tank to add and remove working fluid to the system. The mass in the tank is described by [32]

$$m_{tank} = m_{tank,liquid} + m_{tank,two-phase} + m_{tank,gas} \tag{2.46}$$

where $m_{tank,liquid}$ is the mass of liquid in the tank, $m_{tank,two-phase}$ is the mass of two-phase working fluid in the tank and $m_{tank,gas}$ is the mass of gas in the tank. The mass difference in the tank is modeled with mass balance [32]

$$\delta m_{tank} = \dot{m}_{in} - \dot{m}_{out} \tag{2.47}$$

where $\dot{m}_{in}$ is the mass flow in to the tank and $\dot{m}_{out}$ is the mass flow out of the tank. The energy balance is described by [32]

$$\dot{e} = h_{in}\dot{m}_{in} - h_{out}\dot{m}_{out} \tag{2.48}$$

where h_{in} and h_{out} are the enthalpy of the working fluid added and removed from the tank.

Chapter 3

Complete System

In this section, the complete system is described. In order to run the complete system, a control system was needed. The control system is not a part of this thesis and will not be closer described. The system and inputs and outputs between the component models are described in this chapter with validations plots of the system. The physical components included in this model are a pump, three heat exchangers, a turbine expander and a condenser. The tank is a part of the control system and the recuperator is not included in this model. Not all of the working fluid enters the recuperator, how much is determined by the control system. To be able to run the simulation without receiving the mass flow from the reference model, the recuperator was not included.

There have been two sets of validations. In the first one, all parameters regarding the working fluid is modeled and the only inputs are temperature and mass flow of the hot gas/refrigerant. The other validation has the same mass flow as the reference model, but temperature, pressure and enthalpy of the working fluid is modeled.

Most attention has been paid to the temperatures of the working fluid after the last heat exchanger and after the condenser. The working fluid must be fully evaporated after the last heat exchanger, since the working fluid that enters the turbine must be gas. Otherwise, there is a risk of damage on the turbine. The working fluid that leaves the condenser is the working fluid entering the pump. The working fluid needs to be fully condensed by entering the pump, otherwise the efficiency of the pump will be very low. The pump is designed to pump fluid and not gas.

3.1 Validation against Reference Model

This comparison is made between the reference model and the developed model. The developed model is assigned starting values for pressure and temperature of the working fluid. The inputs to the developed model during the comparison are mass flow and temperature of the hot gas and the cooling refrigerant. The control system assigns the control variable n_{pump} and the mass flow to and from the tank. The temperature, enthalpy, pressure and mass flow of the working fluid is modeled. The comparison is done during the start of the system and when the system is stationary operating after the start.

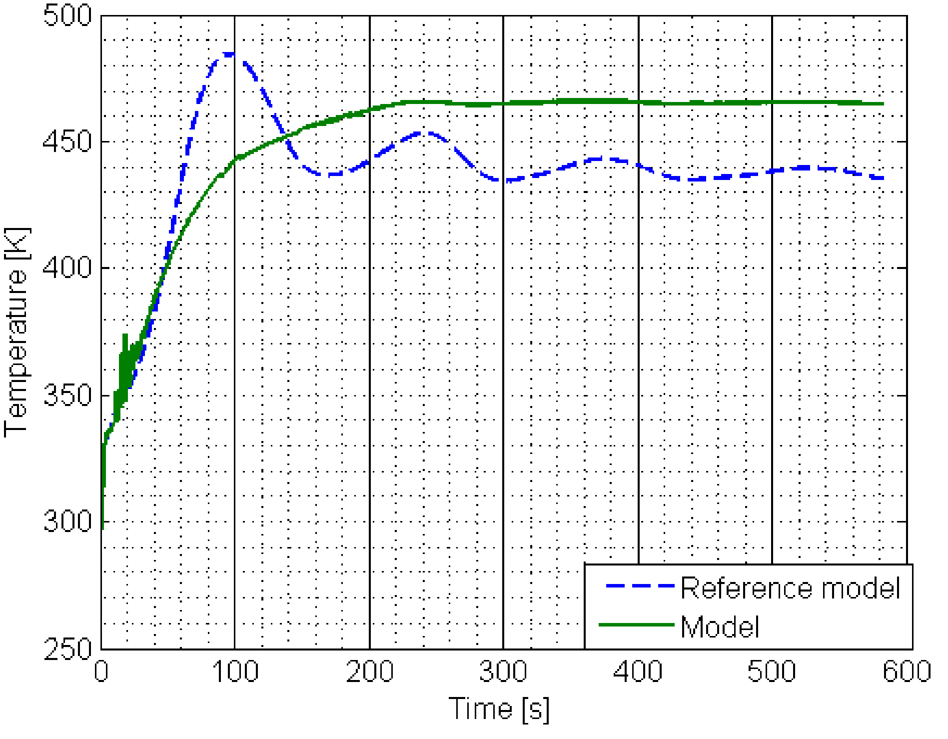


Figure 3.1. Temperature after the last heat exchanger. The simulation is made during start of the system and when the system is stabilized after the start. The working fluid is completely evaporated for both models. There is a difference in the model which might be explained by the mass flow modeling. A comparison with the same mass flow as the reference model can be seen in Figure 3.3.

Figure 3.1 shows the results of the temperature after the last heat exchanger. There is a difference of about 20 to 30 K, which is a rather large difference. There is also a large difference after the condenser, seen in Figure 3.2. Both Figure 3.1 and Figure 3.2 shows that the developed model does not follow the behavior of the reference model, even if the temperature differences are only about 5 K. The rather small differences but lack of agreement in behavior may be explained by Equation 2.36 and 2.37. The difference in temperature between the cooling refrigerant and the working fluid in the condenser is not as big as the difference

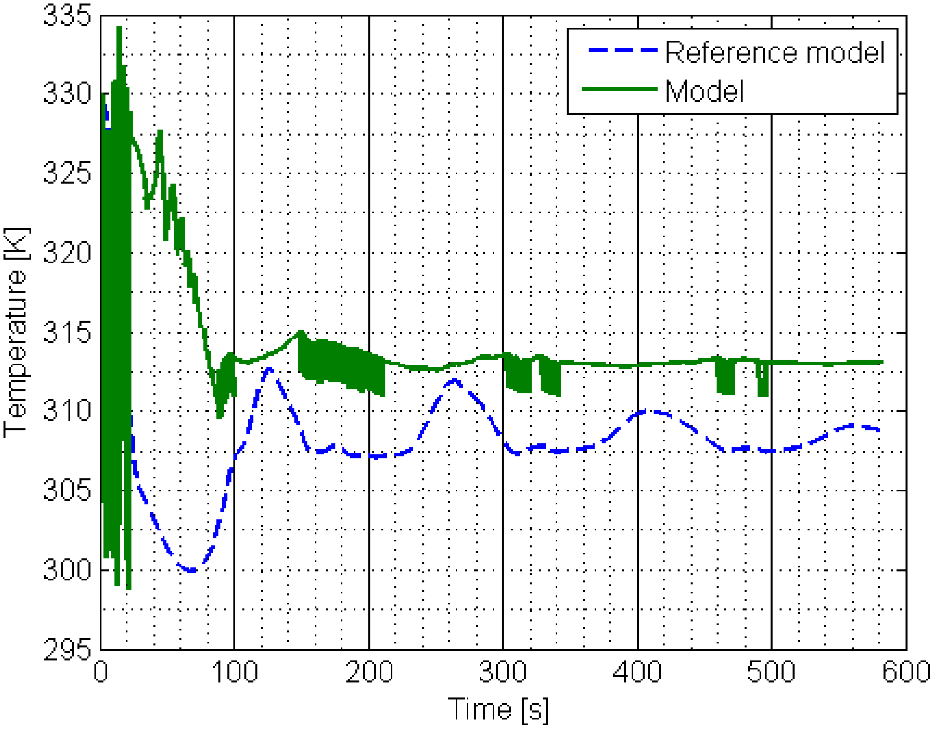


Figure 3.2. Temperature after condenser. The simulation is made during start of the system and when the system is stabilized after the start. The working fluid is completely condensed for both models. There is a difference in the model which might be explained by the mass flow modeling. A comparison with the same mass flow as the reference model can be seen in Figure 3.4

between the hot gas and the working fluid in the heat exchanger. Therefore, the difference in heat transfer is more dependent on input temperatures than on mass flow, compared to the heat exchangers. Therefore, the temperature levels agree with a small difference but the behavior does not.

The Figures in section 2.3.6 and 2.5.2 show that the component models compared to the reference model both have similar behavior and similar output temperatures for the same inputs. The input temperatures are of course of importance, but even more important is the mass flow into the heat exchangers/condenser. Equation 2.26 shows how the mass flow and temperature are correlated, and while not in two phase flow the mass flow is almost proportional to the temperature. Therefore, a validation with identical mass flow was performed.

3.2 Validation against Reference Model with Equal Mass Flows

This comparison is made between the reference model and the developed model. The developed model is assigned starting values of pressure and temperature of the working fluid. The temperature, pressure and enthalpy of the working fluid are modeled but the mass flow of working fluid is given as input from the reference model. Other inputs are temperature and mass flow of the hot gas and the rotational speed of the pump and turbine. The comparison is done during the start of the system and when the system is stationary operating after the start.

Figure 3.3 shows the temperature for the reference model compared to the developed model after the last heat exchanger for both models. During the start of the system, from $t = 0$ s to about $t = 200$ s, there are differences inbehavior between the models. After $t = 200$ s, the both models show a similar behavior except for a time lag. The developed model seems to react faster to changing inputs than the reference model.

Figure 3.4 shows the temperature after the condenser for both models. During the start of the system, there are large differences but after time $t = 200$ seconds, the differences are small. The behavior in the condenser is improved compared to Figure 3.2. With the same mass flow, the behavior of the reference model can be seen in the developed model, with smaller amplitudes. When the behavior of the developed model follows the oscillations of the reference model, the differences in amplitude might depend on the heat transfer coefficient and the specific heat capacity of the refrigerant. The heat transfer coefficient and the specific heat capacity of the cooling refrigerant and hot gas were assumed to be constant. This might affect the behavior and temperature of the working fluid in the developed model.

Figure 3.5 shows a comparison of pressure after the turbine. The amplitudes of the pressure difference in the developed model is smaller than in the reference

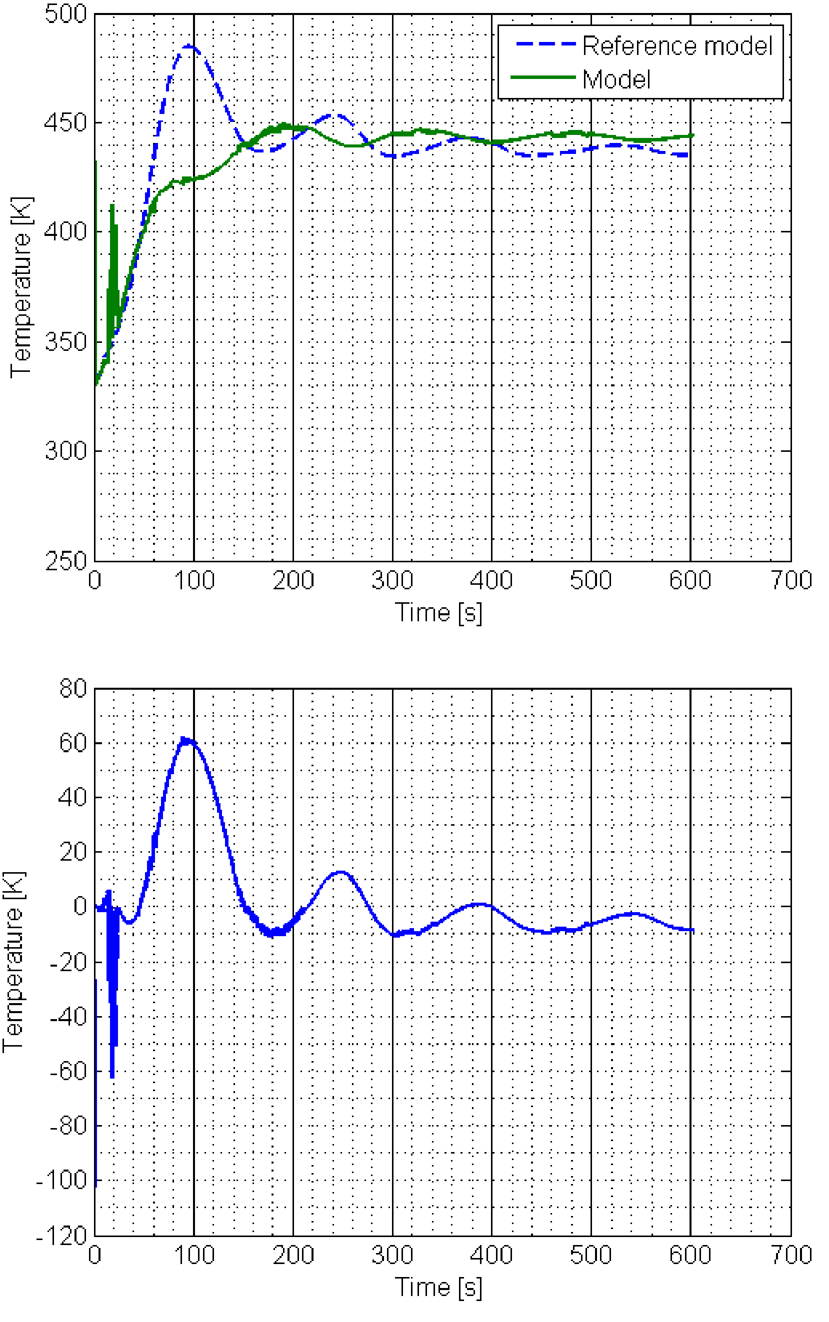
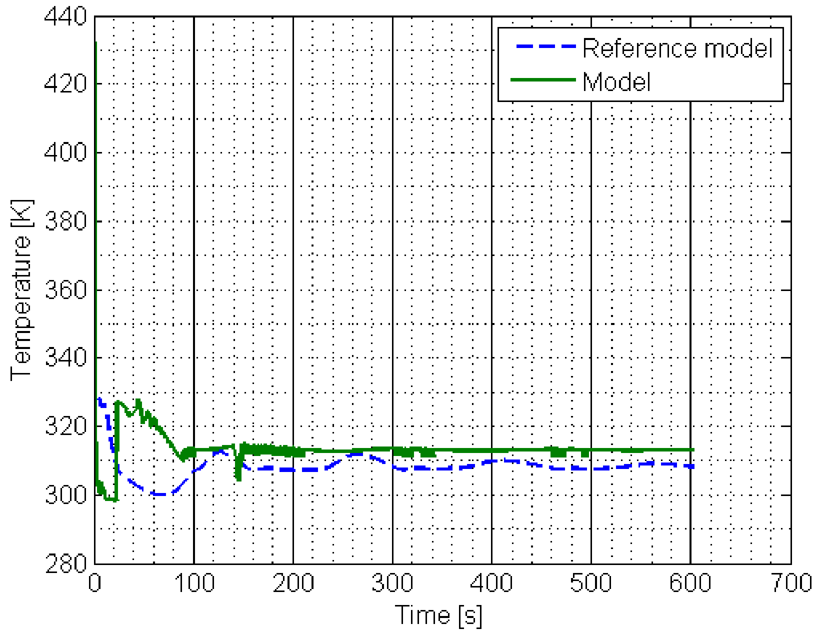


Figure 3.3. Temperature after last heat exchanger is seen in the upper figure and the difference between the in the lower figure. The working fluid in both models are completely evaporated. The simulation is made during start of the system and when the system is stationary operating after the start. The working fluid is completely condensed for both models. The behavior of the models agree but with a time difference and a difference in temperature of about 2.5 percent.



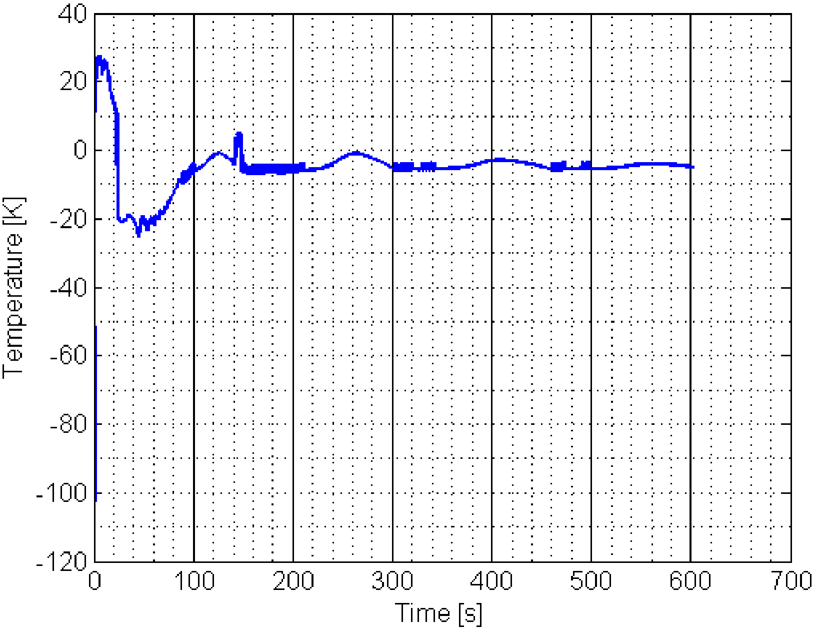


Figure 3.4. Temperature after condenser is seen in the upper figure and the difference between them in the lower figure. The simulation is made during start of the system and when the system is stationary operating after the start. The working fluid for both models are condensed. There is a little difference in temperature in temperature behavior, which might be explained with lack of data points for lower temperature levels. There is a peak during the start, which can be explained by how fast the different models begin the cooling. The condenser in the developed model begins to cool the system during the second sample which is faster the the reference model.

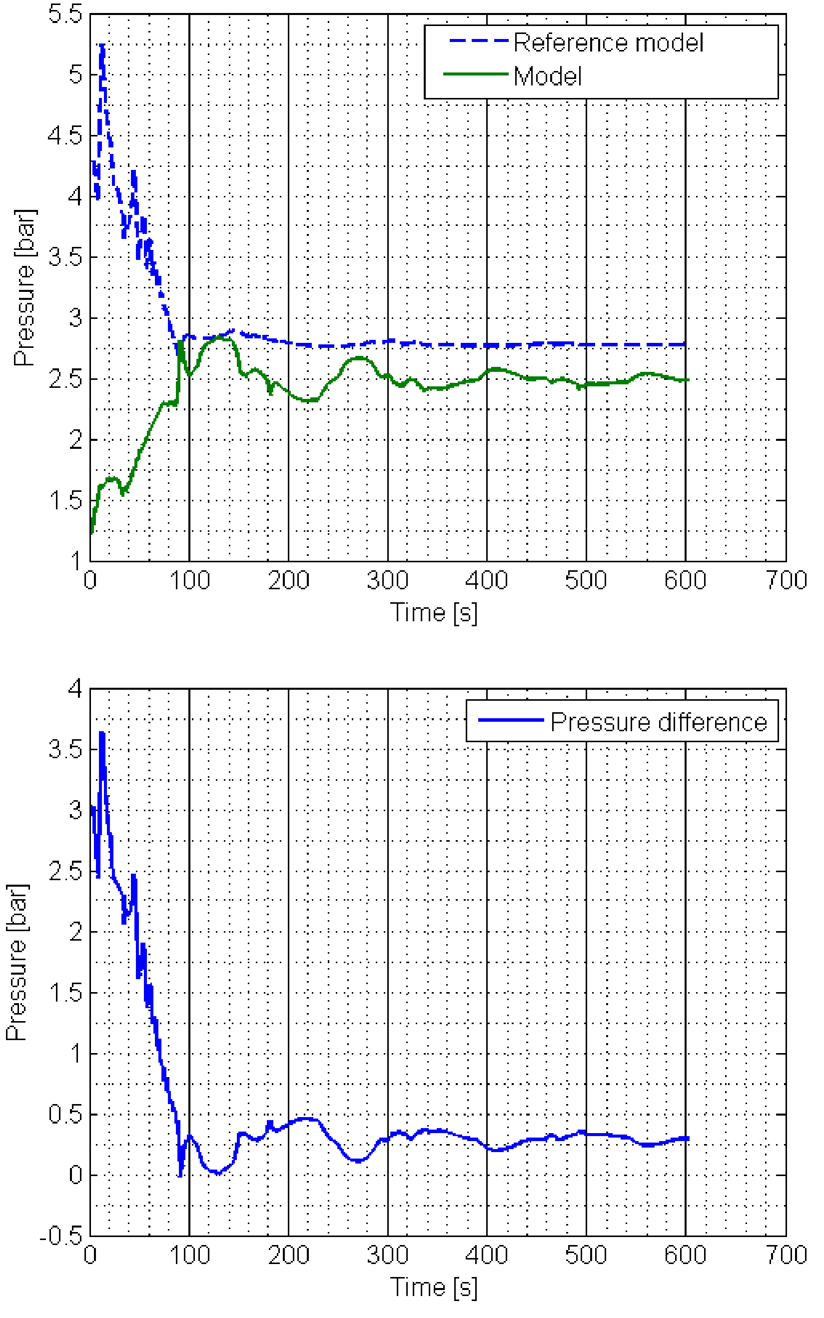


Figure 3.5. Pressure in working fluid after the turbine expander is seen in the upper figure and the difference between them in the lower figure. The simulation is made during start of the system and when the system is stationary operating after the start. The behavior of the working fluid in the reference model is seen in the developed model, but with a larger amplitude.

model. The behavior of the reference model is seen but very vauge in the developed model. This might be explained by the pressure ratio modeling, described in Equation 2.39. With a larger map describing the pressure ratio, the behavior might be improved.

The temperature after the heat exchanger shows agreeing behavior and values when the models have the same mass flow. The temperature after the condenser shows little differences in behavior but has small differences in levels. The pressure differences over the turbine in the developed model shows a vague behavior of the reference model, but might be improved with a larger map for the pressure ratio.

Chapter 4

Conclusions and Future Works

The goal of the thesis was to model a Rankine based waste heat recovery system in SIMULINK. The biggest challenge for the modeling was the two phase flow during evaporation and condensation, since a small difference in enthalpy between a model and a real system can lead to big temperature differences. Big temperature differences might lead to incorrect control system decisions which can lead to both poor system efficiency, but also damage the components.

4.1 Conclusions

The thesis shows that the developed models are capable of handling two phase flows. The component model for the heat exchanger shows a difference of less than 0.2 percent in enthalpy and temperature between the developed model and the reference model after evaporating the working fluid from liquid to gas. The component model for the condenser shows a difference of less than 1 percent compared to the reference model. This also shows that it is very important to know which state the working fluid is in. The difference in temperature when the working fluids in the reference and developed model are in different states is large. A prerequisite for the complete system is to know the enthalpy level of the working fluid. When the enthalpy level is known, the heat transfer can be modeled for all occurring states.

The modeling of the complete system shows that a small temperature deviation after the heat exchangers not is going to cause large temperature deviations after the condenser and vice versa. Temperature deviations do not seem to be transmitted. The modeling of the complete system also shows the importance of a correctly modeled mass flow since the mass flow is almost proportional to the temperature change while in gas or liquid state. Deviations in mass flow will causes deviations in temperature.

The complete system delivers temperatures of the working fluid after the heat exchangers and the condenser with small differences to the reference model. The maximum errors of models are approximately maximum 2.5 percent while stationary operating. This shows that the small errors of the component models still enable a complete system with small differences compared to the reference model.

4.2 Future Works

In order to improve the model, several adjustments can be done to the model. A few examples are presented here.

Focus in the modeling has been the components, and especially the heat exchangers. This means that there is still a lot of work concerning the pipes of the system. The pipes cause pressure changes in the working fluid depending on for example friction and different heights. The pipes may also affect the temperature in the working fluid by for example radiation and heat transfer.

As seen in section 3.1 the mass flow modeling needs to be improved in order to follow the behavior of the reference model, assigned by the control system. This means that for example the valves to and from the tank need models and that the division of the flow before the recuperator needs to be modeled. The mass flow from the pump is modeled, but if the mentioned models are added, a good model of the system mass flow would be possible.

The hot gas and cooling refrigerant need improved models for their heat transfer coefficient. As described in section 2.3.6, the heat transfer and c_p are modeled as constants in this thesis. It is a assumption that enables modeling of the complete heat exchanger, but in order to improve the heat exchangers, the heat transfer coefficient must be more accurate. In [33], heat transfer in the exhaust gas system is modeled and discussed. Those models can be used to extend the heat exchanger models to also include heat transfer coefficients for the hot gases.

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