Simulation of a Comprehensive Heat Recovery Steam Generator

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ABSTRACT

With the increasing prevalence of renewables, it is important to optimize combined cycle power plants during transient conditions to better compliment the variable generation of renewable energy generators. There is a demand for dynamic heat recovery steam generators (HRSG) models that may enable improved control designs. Sufficiently accurate real-time simulations of HRSGs may serve as soft sensors and state estimators during transient conditions. Furthermore, real-time simulations of HRSGs can be used for operator training which does not incur the inherit risk of using plant hardware.

The focus of this research is on the development towards an integrated dynamic closed loop HRSG model, including a three-pressure level HRSG and a steam turbine, capable of real-time simulation. To that end, a simple dynamic steam turbine model was developed and integrated with the HRSG to accurately model the dynamic interaction of both systems. A dynamic control volume was used to model the mixing of IP steam turbine steam exhaust and LP superheater steam. Attemperators were modeled at superheater exits in the HP bypass flow path. Steam turbine and bypass flow paths were modeled, including their respective valves, enabling the simulation of transient conditions. During testing, simultaneous operation of both valves was used to simulate an HRSG and steam turbine running with hot start conditions. Simulation results for the integrated HRSG and steam turbine show good agreement with data provided by Siemens Energy. Once the closed loop simulations were demonstrated, a full HRSG model was created. This model was compiled and downloaded to Siemens PLC hardware and run without active controls at steady state conditions. The simulation results demonstrated numerical stability when run at real-time.
I dedicate this research to my grandfather Dr. Jose M. Castro. Thank you for all your help in raising me as a child and helping me become the person I am today. I hope that wherever you are, you are looking at me with pride and that you have found peace.
ACKNOWLEDGMENTS

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CHAPTER 1: INTRODUCTION

1.1 Research Motivation

As the world makes strides away from fossil fuels and towards renewables as cleaner sources of energy, it’s important to be aware of current technological limitations to better transition to green energy. A current issue with renewable energy sources is that they do not have a consistent output of power and fluctuate how much energy they can produce at any given time during the day. This is an advantage that traditional power plants that use fossil fuels or natural gas have since they are able to output consistent amounts of energy throughout the day. The goal of such power plants in areas where there is a lot of renewable energy being produced is to produce only as much power is needed to fill the gap. The dynamic power production that would be taking place requires improved control designs that can account for the fluctuations in power production. This can be achieved by creating dynamic models of power plants that respond to different controller designs that can show the effects that they will have on the plant without potentially damaging plant equipment.

1.2 Research Objectives

This research focuses on the development of a closed-loop model of a Heat Recovery Steam Generator (HRSG) along with the creation of a full multi-pressure level HRSG that can be run at real-time on PLC hardware. One of the goals of this research is to develop a simple dynamic steam turbine model that would then be integrated with an existing HRSG model within SPPA-T3000. This integrated model will capture the dynamic interaction of both systems while also demonstrating the start-up and transient thermodynamic properties and behaviours of a HRSG given input conditions similar to those of an actual HRSG system at hot start. A major challenge of this re-
search was creating a turbine model that could be run at real-time and be able to be integrated with the existing HRSG model without have to change or modify too much of the existing model. To accomplish this, both the bypass and steam turbine flow paths would have to be modeled, along with their valves, to run a simulation that included this interaction during transient conditions. The simulated closed loop HRSG would then be validated against trends and steady-state values provided by Siemens Energy Inc. as well as data results from previous validated models.

Along with the simulation of a closed loop HRSG in the T3000 plant control software, the creation of a multi-pressure level HRSG model that can be run on PLC hardware was another important part of this research. The challenge with this task was the recreation of the model that was already present within T3000 in Simulink and making sure that it produced results that closely matched those that were already seen in the existing model. Once a working recreation of the HRSG model is tested and validated, this research then aims to show that the model can be compiled to C++ code and then downloaded to actual PLC hardware so that it can be run in real-time.

1.3 Research Approach

A HRSG model, integrated with a steam turbine, will be created using Siemens T3000 plant control software and then run to check for numerical stability and accuracy. First, a simple dynamic steam turbine model must be constructed and tested so that it can then be integrated with the HRSG. Afterwards, there are some changes to the HRSG model that need to take place, specifically the addition of the steam turbine and bypass flow paths, the addition of attemperator blocks, and the new equation of state that will be used. Once closed-loop simulation is demonstrated, a multi-pressure level HRSG model will be created within Simulink. The model will then be compiled and downloaded to Siemens PLC hardware using Siemens TIA Portal. From TIA portal, the HRSG model will be run without any active controls at steady-state conditions and at real-time to see if
it responds as expected. An overview on what TIA Portal and SPPA-T3000 are will be discussed briefly in Chapter 2. The closed-loop HRSG model that is presented in this research is an adaptation of previous models presented in [1, 2] which will be discussed in Chapter 2. An overview of the structure of the steam turbine model, the integrated HRSG model, and the creation of the multi-pressure level HRSG in Simulink that will be run on a PLC will be discussed in Chapters 3, 4, and 5.
CHAPTER 2: BACKGROUND INFORMATION

2.1 Heat Recovery Steam Generators

A heat recovery steam generator (HRSG), shown in Figure 2.1 [3], is a waste heat recovery boiler that is added in series to a traditional power plant [4, 5]. It utilizes the exhaust gas that leaves a gas turbine of a combined cycle power plant (CCPP) and uses it to boil water into steam that will go through a steam turbine to produce power. The purpose of a HRSG is two-fold, it increases the efficiency of traditional power plants from about 40% to about 60% and it lowers the temperature of waste heat that would leave the plant and enter the atmosphere. All the heat that is being produced by the CCPP would ultimately enter the atmosphere regardless, but the HRSG makes more use of the heat before it leaves. A HRSG can have anywhere from 1 to 3 different pressure levels which are composed of three major sections which are Economizer, Evaporator(Boiler), and Superheater.

There is also an additional type of section within the HRSG that consists of the cold reheat (CRH) and the hot reheat(HRH). A HRH is just another name for a superheater that is placed after the CRH. The CRH mixer is a location where the exhaust steam from the HP steam turbine is heated back up to the temperature using the steam leaving the IP superheater. This steam then goes through an two HRH so that it can then be reintroduced into the IP steam turbine. The Siemens HRSG model that was created previously [1, 2] consisted of three pressure levels: High Pressure (HP), Intermediate Pressure (IP), and Low Pressure (LP). Each of these pressure levels consist of different combinations of economizers, evaporator, superheaters, CRH, and HRH with the ability to simulate start-up, transient, and steady-state behaviour.
2.2 Steam Turbines

A steam turbine, seen in Figure 2.2 [6], is a machine that extracts the thermal energy from pressurized steam and uses it to produce useful work that generates power. There has been research done to model dynamic steam turbines simulations, however some focus on the creation of models that are based on the internal geometry of the turbine blades [7] while others focus on creating models that closely produce results that match trend lines that were provided by outside sources [8]. A paper by Byregowda [7] focused on creating a steam turbine model in ANSYS that depends on the geometric properties of the turbine. This would not suit the purposes of this research since the intention is to create a simple dynamic turbine model that can be tuned to better fit different
operating conditions without depending on internal geometry. There was one research paper [9] that focused on using Stodola’s Law of the Ellipse [10, 11] to create equations that demonstrated the pressure and temperature changes that occur within the different pressure levels that are present in a multi-pressure level steam turbine. Since none of these existing steam turbine models matched the needs of the research, a different approach was taken. The purpose of this research is then to create a simple dynamic steam turbine model that can be easily integrated with the HRSG model that was already constructed.

The reference documents that were used to construct the steam turbine model was provided by Siemens Energy and contained three different pressure level systems. This reference document was the same one that was used in the creation of the HRSG model that will be used for integration.
The document also showed the presence of two separate mixing chambers, one of which that combined the same that was leaving the IP steam turbine and the steam from the LP superheater. The equations that were used to construct the steam turbine sections, as well as the mixing chamber, are discussed in Chapter 3.

2.3 Siemens T3000 Modeling Software

T3000 is a plant control software that was developed by Siemens Energy Inc. It is has the ability to conduct large scale transient simulations and has a library of blocks that can carry out a variety of functions, whether it be calculating thermodynamic properties given specific input conditions or an integral block. Additionally, it can also preform basic calculations through individual blocks, either numeric or logical, and a series of those blocks can be used to model equations.

Figure 2.3: Example of a Thermodynamic Calculation Block

Figure 2.3 shows an example of one of the blocks that could be used from the expansive library of blocks and figure 2.4 shows an example of how an equation would be constructed in T3000. One of the impressive qualities of T3000 is the fact that the simulation will continue running even
if there is an algebraic loop. It should be noted that while Simulink can run a simulation that involves an algebraic loop, it cannot run a simulation after a "nan", or a not a number, value is encountered. In T3000, the previous valid state is held, and the signal quality indicates a bad value but the simulation continues. All these different features mean that T3000 is a robust simulation software that can handle complex dynamic models. One downside to using T3000 over Simulink is that the time it takes to run a simulation within T3000 is much longer than it would be in Simulink. For example, the HRSG model that will be talked about in this research takes 5 hours of real-time simulation to run from start-up to steady-state in T3000 while it only takes about 30 minutes to run the same model within Simulink.

![Figure 2.4: Example of a Equation Construction in T3000](image)

2.4 Siemens TIA Portal

TIA Portal is a working environment for integrated engineering with the various SIMATIC systems being made available within a single environment. This program was originally created to serve as a software to automate tasks but it can also be used to download dynamic models to PLC hardware and monitor the results in real-time. An example of what that would look like in the TIA portal interface is shown in Figure 2.5.
The downloaded model is able to communicate to other systems through bus and network interfaces. Steady-state and dynamic I/O enables a dynamic model to mimic a physical system through simulation. Parameter values can also be adjusted in real-time and have an immediate effect on the model, which is then visualized within TIA portal.

2.5 Previous Works

Previous research has been done on the creation of dynamic multi-pressure level HRSG models that focused on creating accurate values and behaviour for start-up, transient, and steady-state simulation. There was a model created by Caesar [2] that focused on the creation of a complete LP HRSG that was able to run while producing accurate steady-state results as well as having transient behaviour that matched reference trends that were provided by Siemens Energy. A continuation of this work by McConnell [1, 12] saw the development of a multi-pressure level HRSG model that improved upon the evaporator component of the model. In particular it focused on developing a switching mechanism to account for the phase change that is present in the boiler subsystem that
Figure 2.6: Overview of McConnell HRSG Model

was robust to numerical fluctuations. This simulation showed good agreement with steady-state values and transient trends against values that were from an actual physical plant. Figure 2.6 shows an overview of the overall three pressure level HRSG that was developed by McConnell.

There has also been a significant amount of research done on the simulation of dynamic steam turbines models[13, 14]. The research conducted by Chaibakhsh [9] provided a modified equation that represented the relationship between mass flow and pressure drop that was first developed by Stodola [11] that accounted for the effect of the inlet temperature of steam. This equation will be discussed further in Chapter 3 since it was used in the construction of the steam turbine model that was developed for this research.
CHAPTER 3: STEAM TURBINE

The unknowns of the steam turbine are solved algebraically, the unknowns being the inlet turbine pressure, the mass flow exiting the turbine, and the outlet steam temperature. The following series of equations and blocks were chosen/created to create a model of a simple dynamic steam turbine that could be connected to an existing HRSG model that was created previously. Fig 3.1 shows a series of tables that display the inputs for each component of the steam turbine as well as the outputs for those same components. It should be stated that while the steam turbine model needs to calculate the inlet turbine pressure, it is used internally and is not an outlet of the turbine sections.

Figure 3.1: Process Variables for the Entire Steam Turbine
3.1 Stodola Turbine Equation

There was an equation that developed by Stodola [11] to define the relationship between mass flow and the pressure drop across the turbine known as the Ellipse Law. This is an empirical formulation that is used to describe an idealized turbine. This equation was later adjusted to account for the effect of inlet temperature on the steam as seen in the paper by Chaibakhsh[9] and is shown in Eq 3.1.

\[ \dot{m}_{n,in} = \frac{K_T}{T_{n,in}} \sqrt{\frac{P_{n,in}^2 - P_{n,out}^2}{T_{n,in}}} \]  

(3.1)

This equation was then manipulated to solve for \( K_T \), which is a turbine constant, that can be acquired using steady-state values that were provided in reference documents for the given turbine section. This value is assumed constant throughout the start up of the turbine into its steady-state operation.

\[ K_T = \frac{\dot{m}_{n,in}}{\sqrt{\frac{P_{n,in}^2 - P_{n,out}^2}{T_{n,in}}}} \]  

(3.2)

Solving explicitly for \( K_T \) using Eq. (3.2) for each of the turbine sections gives us the values shown in Table 3.1. These parameters will be used when solving for the inlet turbine pressure and mass flow for each of the turbine sections.

<table>
<thead>
<tr>
<th>Table 3.1: ( K_T ) Values for Different Pressure Level Systems</th>
</tr>
</thead>
<tbody>
<tr>
<td>HP Turbine</td>
</tr>
<tr>
<td>380.638</td>
</tr>
</tbody>
</table>

12
3.2 Valve Equation

The valve equation was also used when creating the turbine block that would calculate the pressure change and mass flow through the section since there are valves present before the steam enters the turbine that need to be accounted for. These valves control how much and at what point steam enters the turbine sections during transient conditions. The valve equation is given below

\[ \dot{m}_n = K_V \sqrt{\frac{\Delta P}{\nu}} \]  

(3.3)

which can be rearranged to solve for \( K_V \) as shown in the following equation.

\[ K_V = \frac{\dot{m}_n}{\sqrt{\frac{\Delta P}{\nu}}} \]  

(3.4)

It should be noted that the valve equation that is being used is the one used for incompressible fluids. We are assuming that the steam passing through the turbine sections are incompressible. This is because the velocity of the steam within the turbine is never fast enough for it to be considered a factor. The regime in which steam would be considered compressible is when it is approaching Mach 1, which is not something that occurs throughout the start-up, transient, or steady-state cases of the different turbine sections. If we were to consider steam to be compressible, then we would use a different form of the valve equation not only in the calculation of the \( K_V \) values for the different sections but also in the block that calculates pressure and mass flow. That equation is shown below

\[ \dot{m} = K_V P_{n,in} (1 - \frac{2\Delta P}{3P_{n,in}}) \sqrt{\frac{\Delta P}{P_{n,in} G_T^g T_{n,in}}} \]  

(3.5)

The values for the different \( K_V \)’s that will be used for each turbine section are shown in the table below.
Table 3.2: $K_V$ values for different pressure level systems

<table>
<thead>
<tr>
<th>HP Turbine</th>
<th>IP Turbine</th>
<th>LP Turbine</th>
</tr>
</thead>
<tbody>
<tr>
<td>35.670</td>
<td>181.981</td>
<td>65.148</td>
</tr>
</tbody>
</table>

3.3 HP & IP Turbine Sections

There HP and IP Turbine sections are both identical in construction with the only difference between them being what components they connect to. The HP Turbine connects its exhaust with the IP superheater into a mixing chamber that leads into the IP CRH. The IP Turbine connects to the mixing chamber that combines its exhaust with the superheated steam of the LP superheater which then enters the LP turbine. There are two main components to these turbine sections and they are the block that calculates the temperature drop across the turbine and the block that calculates both the inlet pressure of the turbine along with the mass flow throughout the turbine section.

There are two equations, the valve equation and the Stodola turbine equation, that both have the same two unknowns and because of this the unknowns can be solved for algebraically. The equation that is used to calculate the inlet pressure of the turbine is shown below with a full derivation of the equation present in Appendix A.

$$P_{\text{Turb,Inlet}} = -\lambda + \sqrt{\lambda^2 + 4(\lambda P_{\text{SH,Outlet}} + P_{\text{Turb,Outlet}}^2)}$$  \hspace{1cm} (3.6)

where

$$\lambda = \frac{K_V^2 T_{\text{Turb,Inlet}}}{K_T \nu}$$  \hspace{1cm} (3.7)

This equation is derived from combining Eq. 3.2 and Eq. 3.3 and solving for the inlet pressure of the turbine and then using that value and plugging it into the valve equation (Eq. 3.3) provides the mass flow through the section. The reason that both equations combined to solve for the inlet
turbine pressure is due to the presence of the valve before the turbine inlet. Because of this, both
equations are required to solve for both mass flow and inlet turbine pressure due to the fact that
there are two unknowns because of the two components and neither equation by itself can solve
for both. If the valve weren’t present then you could use the outlet superheater pressure in Eq. 3.2
while accounting for any loss in pressure due to pipe loss. A figure showing the construction of
this block within Simulink is shown in Figure 3.2.

Figure 3.2: Mass Flow and Inlet Pressure Calculation Block

There are two equations that calculate the outlet enthalpy of the turbine section which is then used,
along with the outlet turbine pressure, to calculate the outlet temperature of the turbine. The first
equation calculates the ideal outlet enthalpy of the turbine section and is shown below

\[
H_{Outlet, \text{Ideal}} = H_{Inlet} + RT \ln \frac{P_{Outlet}}{P_{Inlet}}
\] (3.8)

From there you use the \( H_{Outlet, \text{Ideal}} \) that is found in Eq. 3.8, along with the isentropic efficiency of
the turbine section, in Eq. 3.9 to get the actual outlet enthalpy for the turbine section.
\[ H_{\text{Outlet}} = H_{\text{Inlet}} - (H_{\text{Inlet}} - H_{\text{Outlet, Ideal}}) \eta_T \]  

As mentioned above, with this enthalpy value along with the outlet turbine pressure, one can calculate the outlet temperature of the turbine using thermodynamic property calculator. Figure 3.3 shows the construction of the outlet temperature calculation block. It should be mentioned that while SPPA-T3000 has blocks that calculate these thermodynamic properties easily, Simulink has no such blocks readily available. A fellow lab member created blocks that did these calculations in Simulink using XSteam, which is a matlab script implementation of the IF97 "fast formulation" for the thermodynamic properties for water. A benefit of using XSteam is that it can be compiled to C++ and used when downloading the HRSG model to PLC hardware. These blocks were used in both the construction of the Steam Turbine Sections and the three pressure level system HRSG conversion which will be discussed in the following chapter.

![Figure 3.3: Overview of the Temperature Calculation Blocks](image)

### 3.4 Mixing Chamber

The mixing chamber is a section of pipes where the steam leaving one component mixes with the steam leaving a different component. In this case it is the steam exhaust leaving the IP turbine and
the steam from the LP superheater. Originally, the mixing chamber only comprised of a simple mass-enthalpy balance that took in the mass flow and enthalpy from the IP steam turbine exhaust as well as the steam for the LP superheater. The issue with this approach was that the data in the transient case were unrealistic and prone to breaking the simulation. The mixing chamber was replaced by a control volume where temperature and mass are the two states of the system. The mixing chamber control volume uses the mass balance equation, the unsteady flow energy equation (UFEE), and the Redlich-Kwong equation of state to solve for the different outputs of the component. Figure 3.4 shows the construction of the mixing chamber which consists of five terms that, when combined, produce \( \dot{T}_{MC} \) and \( \dot{m}_{MC} \). Two integrators integrate \( \dot{T}_{MC} \) and \( \dot{m}_{MC} \) to update the states and convert them to \( T_{MC} \) and \( m_{MC} \). There is also a block that calculates the molar specific volume of the mixing chamber using \( m_{MC} \) and a Redlich-Kwong equation of state block that calculates the outlet pressure of the mixing chamber. Some of the construction of the mixing chamber mimics the construction of similar components from previous work in the full HRSG that was modified to work for this purpose, specifically the Redlich-Kwong block. As will be mentioned in Chapter 4, the Gibbons-Laughton equation of state was used to replace the Redlich-Kwong equation of state within the boiler and superheaters. The reason why the Gibbons-Laughton was not used within the mixing chamber in the turbine model is because the Redlich-Kwong produces results that are very close in accuracy to those that would be calculated using the Gibbons-Laughton for the range of temperature of steam entering the turbines.

Equation 3.10 calculates the total mass flow present in the control volume of the mixing chamber by taking the total amount of mass flow entering the system and subtracting the total mass flow leaving the system.

\[
\dot{m}_{MC} = \dot{m}_{LPSH,Outlet} + \dot{m}_{IPST,Outlet} - \dot{m}_{MC,Outlet}
\]  

(3.10)
Figure 3.4: Overview of Mixing Chamber Block

Due to the size of the equation that solves for $\dot{T}_{MC}$, the main components of the equation have been split into four terms that will then be used in the calculation of the $\dot{T}_{MC}$. Equations 3.11 - 3.15 show the different terms and the final equation.

\begin{align*}
x_1 &= \dot{m}_{LPSH,Outlet} h_{LPSH,Outlet} \quad (3.11) \\
x_2 &= \dot{m}_{IPST,Outlet} h_{IPST,Outlet} \quad (3.12) \\
x_3 &= (\dot{m}_{IPST,Outlet} + \dot{m}_{IPST,Outlet} - \dot{m}_{MC,Outlet})u \quad (3.13) \\
x_4 &= \dot{m}_{MC,Outlet} h_{MC,Outlet} \quad (3.14) \\
\dot{T}_{MC} &= \frac{x_1 + x_2 - x_3 - x_4}{m_{MC} C_V} \quad (3.15)
\end{align*}
3.5 LP Turbine Section

The LP steam turbine block follows the same basic construction as the previous two turbine sections with the exception in how it calculates the mass flow through the turbine. Instead of using Eq. 3.6 to calculate both the inlet pressure to the turbine and the mass flow, we instead use Eq. 3.1. There is no valve between the mixing chamber and the LP steam turbine and because of this both the inlet and outlet pressure of this section are known meaning there is no need to calculate the inlet turbine pressure. There is also the fact that the valve that controls steam flow from the LP superheater is placed before the mixing chamber. Because of this there is a separate valve block placed before the mixing chamber that calculates the amount of mass flow entering the mixing chamber from the LP superheater. Fig 3.5 shows an overview of the entire configuration of the steam turbine within Simulink. An identical version of this model was constructed within T3000 when integrating both the HRSG and the steam turbine together.

Figure 3.5: Overview of Entire Steam Turbine
CHAPTER 4: STEAM TURBINE AND HRSG INTEGRATION

This chapter will focus on the additions and changes that were done to the existing HRSG model to make it compatible for integration with the steam turbine model that was designed. These additions and changes include the addition of attemperator blocks, the new equation of state that was chosen for use in the boiler and superheater blocks, and the introduction of separate flow paths for the steam turbine and the bypass mass flows. Figure 4.1 shows a full schematic of the integrated HRSG model and the direction that the flow paths are going between components.

![Figure 4.1: Schematic of Integrated HRSG with Components](image)

Figure 4.1: Schematic of Integrated HRSG with Components
4.1 Attemperator

Attemperators are spray curtains that lower the temperature leaving one superheater before it enters the next component by spraying it with feed water from the LP economizer, otherwise known as the preheater, that is placed earlier within the HRSG. The equations that were used to calculate the mass flow and temperature are shown below. Equation 4.1 is the mass equation and equation 4.2 is the energy balance equation and they are both used within the attemperator block.

\[ \dot{m}_{Att} = \dot{m}_{FW} + \dot{m}_{n,SH} \]  
\[ h_{Att} = \frac{\dot{m}_{FW} h_{FW} + \dot{m}_{n,SH} h_{n,SH}}{\dot{m}_{Att}} \]

Once the enthalpy of the vapour leaving the attemperator is calculated, a thermodynamic block is used to calculate the temperature of the steam by using both the enthalpy and the current pressure of the preceding superheater/reheat. There are four attemperators present within the reference documents that was used in the development of the HRSG model. There is one attemperator
present after each superheater in the HP section of the HRSG and one present after each hot reheat component in the IP section of the HRSG. Figure 4.2 shows the construction of the attemperator block within Simulink.

The attemperators are not constantly on throughout operation. Instead they are only turned on when the temperature of the steam exceeds nominal levels and needs to be lowered. To model this, two switches were used to modulate between the temperature that was calculated by the attemperator block and the temperature calculated by the superheater component which is shown in Figure 4.3. This is an active switch that changes which block is being used depending on the mass flow entering from the feed water. For testing of this model, the operator can directly control the amount of feed water that is being entered into the system and at what temperature that feed water is entering at through a direct signal. In reality, these attemperators would not be controlled directly but would have valve actuation signals that would activate them when they are needed.
4.2 Gibbons and Laughton Equation of State

In the existing model for the HRSG developed by McConnell [15], the Redlich-Kwong equation of state has been used to describe the behaviour of water and vapor in the evaporator during boiling. While it has provided accurate results in the current model, it was discovered that it was not as effective as other equations of state, especially in low reduced pressures and temperatures that are during start-up and transient conditions. The Redlich-Kwong equation has poor agreement for strongly polar substances, such as water, due to the complex nature of polar interactions and the size of the molecules. It is because of this that equation of state that was used in the evaporator components were changed to the one developed by Gibbons and Laughton [15] in 1983. This is because the Gibbons-Laughton was designed to have strong agreement with polar substances. The equation that was developed is shown in Eq. 4.3 where \(a\) is the internal energy term at the critical point, \(\alpha\) is a function of temperature, \(\nu\) is the specific volume, \(R\) is the gas constant, \(P\) is the pressure, \(T\) is the temperature, the subscript \(c\) refers to the critical condition, \(T_R\) is equal to \(T/T_c\), and \(m\) is the mass.

\[
P \nu^3 - RT \nu^2 + (a\alpha - bRT - b^2P)\nu - aab = 0
\]  
(4.3)

where

\[
a = \frac{(RT_c)^2}{9kP_c}, \quad b = \frac{kRT_c}{3P_c}, \quad k = 2^{1/3} - 1
\]  
(4.4)

\[
\alpha = 1 + X(T_R - 1) + Y(T_R^{1/3} - 1), \quad X = m^2, \quad Y = -2m(m + 1)
\]  
(4.5)

It should be noted that both the Redlich-Kwong and the Gibbons-Laughton equations of state are cubic equations although the Gibbons-Laughton is an improvement on existing cubic equations of state. There are other equations like the Helmholtz and kinematic equations of state that could be
Table 4.1: Comparison of Percent Error of Calculated vs. Experimental Vapour Pressures

<table>
<thead>
<tr>
<th>Temp. (K)</th>
<th>Press. (bar)</th>
<th>Gibbons-Laughton error %</th>
<th>Redlich-Kwong-Soave error %</th>
</tr>
</thead>
<tbody>
<tr>
<td>273.16</td>
<td>0.006112</td>
<td>0.89</td>
<td>-33.64</td>
</tr>
<tr>
<td>293.15</td>
<td>0.023368</td>
<td>0.16</td>
<td>-27.02</td>
</tr>
<tr>
<td>323.15</td>
<td>0.123347</td>
<td>-0.33</td>
<td>-18.63</td>
</tr>
<tr>
<td>373.15</td>
<td>1.01325</td>
<td>-0.41</td>
<td>-8.54</td>
</tr>
<tr>
<td>423.15</td>
<td>4.7597</td>
<td>-0.13</td>
<td>-2.39</td>
</tr>
<tr>
<td>523.15</td>
<td>39.776</td>
<td>0.62</td>
<td>2.43</td>
</tr>
<tr>
<td>573.15</td>
<td>85.917</td>
<td>0.97</td>
<td>2.64</td>
</tr>
<tr>
<td>623.15</td>
<td>165.37</td>
<td>0.74</td>
<td>1.44</td>
</tr>
<tr>
<td>643.15</td>
<td>210.52</td>
<td>0.09</td>
<td>0.21</td>
</tr>
</tbody>
</table>

used to describe the behaviour of water. However cubic equations of state are easier to implement in a controls sense and they have a smaller computational cost.

Table 4.1 shows the percent error of values calculated using the Gibson-Laughton equation of state and the Redlich-Kwong-Soave equation of state compared to experimental vapour pressures for water vapour that were presented in the Gibbons and Laughton paper. It should be mentioned that the Redlich-Kwong-Soave EoS is in itself an improvement on the Redlich-Kwong EoS, by means of using compressibility factors. However, it did not specifically improve the results for polar molecules/compounds which Gibbons-Laughton did. The results show that the percent error using the Gibson-Laughton equation is consistently lower than the with Redlich-Kwong equation, especially at lower temperatures. This is important to note since that means that the implementation of this equation would improve the behaviour of the HRSG in the start-up and transient case.

### 4.3 Bypass and Steam Turbine Flow Switching

One of the things that needed to be addressed with integrating the steam turbine model with the existing HRSG model was the creation of two separate flow paths: one leading to the steam turbine
and one leading to a steam turbine bypass. In the previous version of the HRSG model that is being used, a single valve was used to approximate the downstream flow resistance that was being provided by both the bypass valve and the steam turbine. Additionally, due to the absence of a steam turbine model the CRH was using approximated HP steam turbine temperature and mass flow. In this research, the superheater to CRH flow path has been closed through the implementation of the steam turbine model and the bypass flow path.

During start-up conditions, there is no steam entering the steam turbine and any steam that leaves the superheaters or HRH is bypassed and introduced into the CRH before entering the IP level of the HRSG. Once certain pressures conditions are reached, the valves are opened to allow some of the steam to start entering the steam turbine sections. The steam isn’t introduced all at once after it starts leaving the superheater to prevent the sudden change in temperature between the steam and the metal from shocking the system. This is done to prevent any unnecessary damage to any of the expensive equipment. The valve opening present for the steam entering the turbine sections follows timings that match when the bypass valve is closing or openings. For testing purposes, an approximation of the valve controls were implemented for HRSG start-up simulation which is shown in Fig 4.4. The HP Turbine Post-MC block is a mixing chamber that mixes the steam leaving the HP steam turbine and the bypass valve. This was added because there is a point in time where both the steam turbine and bypass flow paths are active and mix with each other before entering the CRH. This mixing chamber is different than the one shown in Chapter 3 since it is a simple mass-enthalpy average block using Eqs 4.1 and 4.2 that were used within the attemperator block and it does not involve calculating pressure or integration.

When the HRSG system is starting up, the HP turbine valve starts closed until the pressure present in the second superheater of the HP section reaches 45 Bar, which happens after about 7 minutes, at which point the valve begins to open up until it is about 30% open. At that point some of the mass flow leaving the superheater enters the HP steam turbine while the rest goes straight to the
CRH through the bypass valve. This valve is different from the other two valves that lead into the IP and LP steam turbine present within this system because it doesn’t fully open once it reaches the desired pressure. Instead, it only opens partially to allow some steam into the turbine and once about 30 minutes have passed, then it fully opens. This is probably done since the temperature of the steam is much higher in the HP section than the temperature of the steam in the other two pressure levels of the HRSG. Limiting the amount of steam entering the HP steam turbine would prevent any potential damage to the equipment from the difference in temperatures. While the valve leading into the turbine is fully opening, the bypass valve that is allowing flow to enter the CRH is closing at the same rate and will remain closed in normal operation.
The IP turbine valve opens once the second HRH reaches 13 bar and fully opens in the course of 39 minutes. The steam flow goes through the IP turbine and enters the mixing chamber where it then enters the LP turbine before exiting into the condensor. The LP turbine valve is the last valve to open amongst the three valves. This valve stays close until the LP superheater reaches 0.9 bar. The steam then enters the mixing chamber where it mixes with the exhaust steam from the IP turbine before entering the LP turbine. Once the steam leaves the LP turbine, it then flows into the condenser.
CHAPTER 5: HRSG SIMULINK CONVERSION

5.1 Development of Full Model

As mentioned earlier, one of the goals of this research is to create a working HRSG model that can be compiled, downloaded into Siemens PLC hardware, and run in real-time. To that effect, a full model of a multi-pressure level HRSG was created in Simulink that matched the construction of the one that was previously created in SPPA-T3000 by McConnell [1]. A library of blocks was created to replicate the modules that were present within the T3000 model. The placement of the blocks within the individual clusters also matched the placement within the T3000 model as well as the configuration of components from reference documents provided by Siemens Energy. Figure 5.1 shows the run-time configuration of the entire HRSG model and the different parameters that were needed to properly test it within Simulink.

The main changes that were made in the creation of the Simulink model were the replacement of the Redlich-Kwong equation of state that was used within the boiler with the Gibbons-Laughton equation of state, that was mentioned in Chapter 4, to improve accuracy of the pressure calculations as well as the addition of attemperators after both superheaters and HRH present in the HP and IP systems. This model was not tested for start-up or transient accuracy, instead it was tested for numerical stability with time steps approaching real-time simulation. It was also tested on the accuracy relative to data that was gathered from the existing model within T3000 and reference data provided by Siemens Energy as it approaches steady-state conditions. Full start-up and transient testing of this model was not done since the main focus was on its creation and compilation. It would have also been difficult to recreate some of the blocks that were present within T3000 that would be necessary to run a start-up test case. One example of this are the block that calculates the flue gas properties. While a solution was found for use in Simulink, it is not as simple as T3000
and has room for improvement. There is a difference in the logic used within T3000 and Simulink that makes a seamless transition difficult.

5.2 Compilation of Model

Once the model was completely built and tested, it was later compiled to a C++ code using SIMATIC Target for Simulink, cluster by cluster. The reason the model was compiled by individual clusters, instead of a single HRSG model, was to avoid any overly cluttered databases within TIA portal. The open development kit (ODK) for SIMATIC S7 creates both functional blocks and object files from a compiled model that will then be imported to TIA portal and downloaded to the PLC. There were specific settings that the model needed to have in place in each individual cluster to be
properly compiled and used within the PLC hardware in series. First off, the stop time needed to be set to infinite so that it could run as long as the operator required it to instead of stopping after a preset amount of time. The ode4 (Runge-Kutta) solver was used for compilation and the time step was set to 1 millisecond. This particular step size was chosen because it was numerically stable and was a smaller step size than was used in previous simulations. Parameter access with STEP 7 was enabled because it allowed for the adjustment of the system parameters from within TIA portal without having to make the changes in Simulink and recompile.

Figure 5.2: View of the Settings for Compilation
5.3 Creation of Project in TIA Portal

After successful compilation of the individual clusters, the following steps are taken to create a project within Siemens TIA portal. This project will contain everything necessary to run the model on the PLC and monitor the results. To start, the correct PLC device needs to be selected within TIA Portal that corresponds to the PLC hardware that is being used. A virtual representation of what the PLC hardware would look like is shown in Fig 5.3. Once the right PLC is chosen, then the compiled cluster files are uploaded into the project to construct the model, cluster by cluster, in sequence using ladder logic by using the function blocks that were created during the compilation process. Fig 5.4 shows what one of those function blocks looks like within TIA portal. Along with the function blocks, other blocks are used to allow for the model to be run in the PLC hardware. Once the all the appropriate function and operation blocks are placed, the different databases for the individual clusters are created and populated with the different input and output parameters that are present within the cluster function block. With the databases populated, the different clusters are connected to each other using the parameters that were created within the databases. Once the project is properly configured, it is then downloaded to or onto the PLC hardware to run the simulation. The simulation is able to be run using steady-state conditions without controllers in real time and behaves as is expected according to the given inputs into the system.
Figure 5.3: Virtual Representation of a PLC in TIA Portal
Figure 5.4: View of a Downloaded Cluster Block
CHAPTER 6: VALIDATION AND SIMULATION

6.1 Isolated Testing of Turbine

The accuracy of the steam turbine model was first tested by inputting steady-state values provided by reference documents from Siemens Energy and comparing the expected outputs with those produced by the model. This testing showed an exact match with the reference data so further testing was conducted to further measure the accuracy of the model. The following test was conducted by running the model in an open-loop configuration using data that was provided by the HRSG model that was created by McConnell. The steam turbine is feed values from the HRSG data but is not feeding those values into anything, instead the HRSG is using its original approximated values. Since the data that is being used is from a full run of the HRSG, the final results of the steam turbine simulation show result that would be expected from the steam turbine in a integrated run between both models. The steam turbine model and the HRSG data were both tuned to a react to a particular test case from reference data that was provided by Siemens Energy. The results of the isolated testing of the turbine model using previous HRSG data is shown in Table 6.1.

The results show that the steam turbine was producing values that were within approximately 5% of reference data. The only exception to this was $\dot{m}_{LP,SH}$ which was over 200% higher than was expected. This was due to an issue with the data that was feed into the LP superheater valve. The LP superheater pressure from the simulation data was higher than expected pressure from the reference data. It is unknown why this pressure is higher than expected and since this is data from a previous model there is no way of being entirely certain of the reason behind the higher pressure. Because of this the $K_v$ was not tuned for the experienced pressure differential which led to higher than expected mass flow being calculated at the LP valve. This increase in expected mass flow would cause further error in parts of the system that follow the valve which explains the increase
Table 6.1: Full Run Simulation Results of Steam Turbine

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Ref. Data</th>
<th>Simulation Result</th>
<th>% Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{m}_{HP,ST}$</td>
<td>136.013 kg/s</td>
<td>139.852 kg/s</td>
<td>2.82</td>
</tr>
<tr>
<td>$\dot{m}_{IP,ST}$</td>
<td>162.183 kg/s</td>
<td>173.273 kg/s</td>
<td>6.84</td>
</tr>
<tr>
<td>$\dot{m}_{LP,SH}$</td>
<td>15.662 kg/s</td>
<td>47.792 kg/s</td>
<td>205.15</td>
</tr>
<tr>
<td>$\dot{m}_{LP,ST}$</td>
<td>183.560 kg/s</td>
<td>221.065 kg/s</td>
<td>20.43</td>
</tr>
<tr>
<td>$P_{Inlet,HPST}$</td>
<td>10.761 MPa</td>
<td>10.779 MPa</td>
<td>0.17</td>
</tr>
<tr>
<td>$P_{Inlet,IPST}$</td>
<td>2.641 MPa</td>
<td>2.784 MPa</td>
<td>5.41</td>
</tr>
<tr>
<td>$P_{Outlet,MC}$</td>
<td>0.427 MPa</td>
<td>0.511 MPa</td>
<td>19.67</td>
</tr>
<tr>
<td>$T_{Outlet,HPST}$</td>
<td>639.816 K</td>
<td>606.213 K</td>
<td>-5.25</td>
</tr>
<tr>
<td>$T_{Outlet,IPST}$</td>
<td>575.705 K</td>
<td>581.195 K</td>
<td>0.95</td>
</tr>
<tr>
<td>$T_{Outlet,MC}$</td>
<td>572.261 K</td>
<td>565.296 K</td>
<td>-1.22</td>
</tr>
<tr>
<td>$T_{Outlet,HPST}$</td>
<td>(N/A)excel</td>
<td>318.957 K</td>
<td>N/A</td>
</tr>
</tbody>
</table>

in $\dot{m}_{LP,ST}$ which was over 20% higher than expected, as well as $P_{MC}$. The increased mass flow from the LP SH entering the mixing chamber and would also have an impact the on the pressure calculation that occurs within the mixing chamber block. It should be mentioned that the reason why there is no comparison value for $T_{Outlet,HPST}$ is because that value was missing from the flow diagrams that were used as reference data however the temperature that is calculated is within realistic expectations.

6.2 Testing of HRSG Integrated with Steam Turbine

The following data is from a simulation run of the HRSG model integrated with the steam turbine model. It should be mentioned that due to the way T3000 is constructed, gathering data is a complicated task. Due to the logic inherit to T3000, there is no easy way to gather data when running a simulation. This leads to instances where there are gap in data that is collected. This is seen in Figures 6.3 and 6.4 where there are gaps in the data, however the simulation was closely monitored and producing accurate results during these gaps. One of the reasons why converting
the HRSG model from T3000 to Simulink is important is because there are fewer issues when it comes to data capture.

Figure 6.1 shows the start-up, transient, and steady-state behaviour of the mass flow through the HP steam turbine as well as the bypass and the HP-bypass mixing chamber mass flow. As is expected, there is no mass flow through the turbine section during the majority of the run until the valve starts to open and allow mass flow through. This plot also shows the switching between the bypass and steam turbine mass flows. Initially, all the mass flow is going through the bypass valve to be introduced to the CRH, then the bypass valve starts to close as the steam turbine valve opens. Previously in the simulated model of HRSG in isolation, these flow paths would have been approximated to simulate the downstream resistance and both would have combined into a single mass flow. Figure 6.2 shows the outlet temperature for the HP steam turbine, mixing chamber, and the CRH. The reason why the initial outlet temperature is higher than is expected in the case where
there is no mass flow through the turbine is due to the fact that there is no check condition within
the turbine model. Ideally there would be one that switches between constant values for a turbine
in hot start and those that are calculated when the steam turbine valve is open but one was not
implemented. As soon as the valve leading to the steam turbine is opened, the temperature drops
and approaches the values that are expected.

Figure 6.3 shows the mass flow through both the IP and LP steam turbines. As mentioned pre-
viously, there were some issues with gathering data using T3000, which has led to some gaps
where data is missing. There are mild oscillations present in the IP steam turbine mass flow but
they are centered around 150 kg/s, which is around the range of expected results. The LP steam
turbine mass flow is wildly oscillating which is caused by oscillations present in the IP-LP mixing
chamber pressure. These pressure calculations are caused due to an issue with the integral blocks
within T3000 since similar testing conducted in Simulink did not produce any oscillations with similar conditions. This could be caused by the fact that the time step used within T3000 is 6.4 milliseconds while the one used in Simulink is 1 millisecond. Using a smaller time-step within T3000 could resolve this oscillation at the risk of destabilizing the rest of the system. Further tuning of the mixing chamber component within T3000 or using the model that was constructed within Simulink would reduce the oscillations in the pressure calculation and in turn reduce them in the LP mass flow. Figure 6.4 shows the outlet temperature of the IP & LP steam turbines and the IP-LP mixing chamber. All three temperature lines start oscillating around the time that the pressure oscillations begin within the IP-LP mixing chamber. This is because these three components are connected to each other so the issues in the mixing-chamber pressure cascade and affect the other components. It should be noted that while oscillations are present, the peaks present at around 70 minutes stabilize around values that are expected for this particular test case. Those
being approximately 670 K, 420 K, and 720 K for the IP steam turbine, LP steam turbine, and mixing chamber respectively.

6.3 Validation of Full HRSG in Simulink

To test if the recreation of the full HRSG in Simulink is numerically stable and approaches appropriate steady-state values, steady-state input process values were input into the model and run to compare the results with reference data and see the relative error. As seen in Fig 6.5, most output values of the different components within the clusters are within 5% error with the exception being the flue gas temperatures and economizer temperatures within clusters 4a, 5, and 6b. The reason for the higher than expected flue gas temperature is because of the method of simplification of the economizers. Specifically, simplifying the economizer cluster into a single control volume
prevents the economizer fluid temperature leaving the control volume from being greater than the flue gas being the cluster. Higher economizer temperatures than flue gasses are present in clusters 5 and 6b and those same clusters show a significant amount of error. It should be said that the higher than expected flue gas temperature leaving cluster 5 could lead to higher pressures within the system, specifically the LP system.

Each of the following sections will show graphs that demonstrate the approach to steady-state for the different pressure levels for the HRSG as well as a comparison of the steady-state values to the expected values for temperature, pressure, and mass flow.
Figure 6.6: HRSG Flue Gas Temperature Comparison

6.3.1 Flue Gas Temperature

Figure 6.6 shows the behaviour of the flue gas as it approaches steady-state conditions. The flue gas temperature that were captured from the model are very close to those in the reference data apart from the flue gas leaving cluster 6B which was about 32% higher than expected, the highest error for the different flue gas temperatures. As stated previously, this is due to the fact that the economizers are not allowed to output feed water temperatures that are higher than the flue gas even though this is what is happening in reality. Since there is no block following cluster 6b to simulate ambient conditions, there is no way to mitigate this error and thus it is larger than those.
seen in other clusters with similar component structures.

At around 600 seconds, there is dip in the flue gas temperatures in clusters 1 and 2. This happens because the pressure within the HP and IP boilers exceeds the downstream pressure causing a drop in temperature. Other than this dip in temperature, the different flue gas temperatures approach steady-state values that are close to expected values and are numerically stable.

6.3.2 HP HRSG Results

Figures 6.7, 6.8, and 6.9 show the results of the temperature, pressure, and mass flow of the components that are present in the HP section of the HRSG.

![Figure 6.7: HP Temperature Comparison](image)

The outlet temperatures for all HP components, apart from the economizers in clusters 4a and 5,
follow the same trend as the pressures where the errors present are 5% or less. Both economizers are roughly 13% lower than is expected which is something that was mentioned briefly in the previously as an issue that is present in the construction of the economizer block. At 600 seconds, there is a drop in the temperature present in the superheater temperatures. This is due to the fact that boiler pressures at that time are exceeding the downstream pressure which is causing this drop in temperature. The reason for this is that valve present to prevent steam from leaving the boiler is a check valve and allows for mass flow out of the system if the pressure within the boiler exceeds the downstream pressure. In reality, the valve would stay closed until the drum pressure exceeded the pressure threshold at which point the valve would open and allow steam to leave. At around 1000 secs there is another drop in the superheater temperatures which is caused by the attemperators being activated and lowering the temperature of the outlet steam.

Figure 6.8: HP Pressure Comparison

The outlet pressures of the superheaters in clusters 1 and 2 and the drum in cluster 3 are within 5%
error or less and approach steady-state stably. This means that the HP HRSG section is behaving as expected and producing results that closely match the reference data that was provided as well as being numerically stable.

![HP Flow Comparison](image)

Figure 6.9: HP Flow Comparison

The initialization of mass within the vapour control volumes has not been optimized, because of this it has been seen that pressures are higher than expected given ambient conditions during the approach to steady-state. Because of this there are moments when the mass flow is different than expected which is seen within the first few seconds of the simulation with the large spike in mass flow. After a few seconds, the pressure stabilizes and leads to the mass flow decreasing to what is expected and stabilizing for a bit. The reason for the oscillations in mass flow from around 150 seconds to 600 seconds is that pressure is rising in the vapor control volumes, briefly exceeding the downstream, which causes for the valve to open to allow some mass flow to leave the boiler which drops down the pressure.
6.3.3 IP HRSG Results

Figures 6.10, 6.11, and 6.12 show the approach to steady-state of the temperature, pressure, and mass flow of the components that are present within the IP section of the HRSG. The percent error for most components within the IP HRSG are about 5% with the only exceptions being the IP superheater temperature in cluster 4a and the IP economizer temperature in cluster 5.

![IP Temperature Comparison](image)

**Figure 6.10: IP Temperature Comparison**

The issue with the economizer blocks has been discussed previously and the situation for the IP superheater temperature is a result of this. In cluster 4a, the components that are present are the second HP economizer and the IP SH pressure. The superheater is absorbing too much energy and is causing it to have a temperature that is about 9% higher than expected while the economizer is not absorbing enough energy due to how it constructed. Further tuning of the parameters was
attempted to lower the temperature in the superheater and increase it in the economizer but the improvements to the results became negligible after a certain point.

The drop in temperature in the superheater at 600 seconds is due to the pressure being higher than the downstream pressure leading to mass flow to enter the superheater which drops the temperature. The drop in temperature around the 1000 seconds in the superheaters is due to the attemperators being turned on and lowering the temperature of the steam leaving the component. The slight increase of temperature in the CRH at around 900 seconds is caused by a slight increase in mass flow leaving the superheater in cluster 4a.

![Figure 6.11: IP Pressure Comparison](image)

The approach to steady-state for the pressure is stable and the end values are very close to expected values from reference data and have low error.
The approach to steady-state for the mass flow present within the IP section of the HRSG are numerically stable and have low error values when compared to reference data. It should be stated that this pressure level has no oscillation or sharp spikes throughout the simulation, unlike what is seen within the HP and LP sections. There is a small rise in the cluster 6b economizer is caused by the rapid increase in pressure within the evaporator in cluster 4b. After the pressure stabilizes, the mass flow lowers and approaches the expected value.
6.3.4 LP HRSG Results

Figures 6.13, 6.14, and 6.15 show the approach to steady-state of the temperature, pressure, and mass flow of the components that are present within the LP section of the HRSG.

![LP Temperature Comparison](image)

**Figure 6.13: LP Temperature Comparison**

The temperature of most of the components of the LP system are less than 5% in error relative to comparison data with the exception of the temperature of the economizer/preheater in cluster 6b which is about 23% lower than expected. This issue with the economizer blocks has been discussed previously however the error for present in this cluster is larger than the others. The reason for this is that there is no cluster following it that mitigates this difference in temperature. It was discussed that adding an additional block to be a facsimile for the atmosphere conditions outside of the HRSG could decrease the temperature of the flue gas in cluster 6b, which should
increase the temperature of the economizer. However, that would be outside of the scope of this research and is not relevant for what we are trying to achieve. Another potential solution would be adjusting the metal control volume block to allow for the economizer to have a higher temperature than the flue gas which would solve both the lower than expected temperature of the economizer as well as the higher than expected flue gas temperature.

The approach to steady-state for the pressures in the LP superheater and boiler are numerically stable and produce values that are very close to those that are seen within the reference documents.

As mentioned when discussing the results for the HP system’s mass flows, the initialization of mass within the vapor control volumes has not been optimized. This leads to there being higher than expected pressures in the system in the being than is anticipated given ambient conditions which leads to the spike in mass flow in the beginning of the simulation. The spike isn’t as high as the one in Fig 6.9 but it takes longer for the mass flow in the economizer in cluster 6b to come
back to what is expected for the point in the simulation which is being at zero mass flow. There is a little bit of oscillation that happens around 900 seconds that is caused by the pressure within the component being higher than the downstream pressure which leads to small amount of mass flow to leave the component. However, it should be noted that this period of oscillation happens for a smaller amount of time in the simulation when compared to the HP mass flow approach to steady-state.
CHAPTER 7: CONCLUSION

7.1 Research Overview

A multi-pressure HRSG model was successfully built in Simulink with all the correct components in each individual cluster along with the addition of the attemperator blocks and a change of the equation of state for the vapour control volume. The model produced results that were numerically stable and agreed with steady-state reference data that was provided by Siemens along with the ability to run near real-time in Simulink. It also showed behaviour that had good agreement in the approach to steady-state with data provided by Siemens Energy for a reference HRSG as well as data that was provided from the HRSG model produced by McConnell in T3000. The model was also able to be successfully compiled to C++ code and used to create a project within Siemens TIA portal that was then be downloaded to a PLC hardware. This model was able to be run uncontrolled at real-time and reacted to changes that were being input into the system as they were happening.

A simple dynamic model of a steam turbine was able to be successfully constructed with the inclusion of three different turbines sections for each of the individual pressure levels present in the HRSG, as well as the inclusion of the mixing chamber control volume between the IP and LP turbine. This model was able to be integrated with the existing HRSG model in T3000 that was created by McConnell. Isolated testing of the turbine model showed good agreement with steady-state reference data that was provided by Siemens Energy given input data that was gathered from the HRSG model before integration. The testing of the integrated models provided results that agreed with steady-state values both from the reference data and from the original HRSG model that was used. The transient trends of the model were also behaving as expected.
7.2 Future Research

One area of research that could be conducted in the future would be the integration of a condenser model with the HRSG and the steam turbine to create a complete closed-loop model of a HRSG cycle. There was a model of a condenser created by Odeh [16] within T3000 that could be integrated with the existing models with some minor adjustments to make sure the three models are compatible with each other. There is also some additional work that could be done to improve the economizer blocks that were used within this research. Current construction of the metal control volume and the economizer blocks prevent the flue gas temperature from being lower than the water temperature leaving the economizers, which is observed in reference data. Other ways of characterizing the economizer, potential discretizing it into smaller blocks, could reduce the discrepancy in the temperature that is expected from the economizer and for the flue gas in clusters that contain an economizer block.
APPENDIX A: DERIVATION OF MASS FLOW/INLET TURBINE PRESSURE EQUATION
There are two unknowns to the system, mass flow and inlet turbine pressure, that must be solved by algebraically solving two different equations. These two equations are the valve equation and Stodola’s turbine equation. To derive the equation for the inlet pressure of the turbine, you start with turbine equation

\[ \dot{m} = K_T \sqrt{\frac{P_{Turb,Inlet}^2 - P_{Turb,Outlet}^2}{T_{Turb,Inlet}}} \] (A.1)

Squaring both sides and rearranging the equation we get

\[ \frac{\dot{m}^2}{K_T^2} T_{Turb,Inlet} = P_{Turb,Inlet}^2 - P_{Turb,Outlet}^2 \] (A.2)

Plugging in the valve equation

\[ \dot{m} = K_V \sqrt{\frac{P_{SH,Outlet} - P_{Turb,Inlet}}{\nu}} \] (A.3)

into Eq. A.2 we get

\[ \left( K_V \sqrt{\frac{P_{SH,Outlet} - P_{Turb,Inlet}}{\nu}} \right)^2 \frac{K_T^2}{\nu} T_{Turb,Inlet} = P_{Turb,Inlet}^2 - P_{Turb,Outlet}^2 \] (A.4)

Rearranging the equation and moving everything into the left hand side we get the following quadratic equation

\[ P_{Turb,Inlet}^2 + \lambda P_{Turb,Inlet} - (\lambda P_{SH,Outlet} + P_{Turb,Outlet}^2) = 0 \] (A.5)
where $\lambda$ was created to simplify the equation and is shown below.

$$\lambda = \frac{K^2 \nu T_{\text{Turb,Inlet}}}{K^2 \nu}$$  \hspace{1cm} (A.6)

Solving the quadratic equation and choosing the higher of the two solutions we get the the equation for the inlet turbine pressure.

$$P_{\text{Turb,Inlet}} = \frac{-\lambda + \sqrt{\lambda^2 + 4(\lambda P_{\text{SH,Outlet}} + P_{\text{Turb,Outlet}}^2)}}{2}$$  \hspace{1cm} (A.7)

The reason why the higher solution is used is because it is the only one that is physically possible. It would be incorrect to choose the result that produces a negative values because there is no such thing as a negative pressure. Once you solve for the value of inlet turbine pressure you plug in that value into Eq. A.3 and solve for the mass flow going through the turbine section.
APPENDIX B: DERIVATION OF TURBINE TEMPERATURE EQUATION
The equation for ideal outlet enthalpy of the turbine is derived from the 2nd law of thermodynamics

\[ \Delta S = \frac{\Delta Q}{T} \rightarrow ds = \frac{dQ}{T} \]  
(B.1)

along with the first law of thermodynamics in terms of enthalpy

\[ dQ = dH - VdP \]  
(B.2)

where

\[ V = \frac{RT}{P} \]  
(B.3)

From there you plug in Eq. B.2 into Eq. B.1 to get the following equation

\[ dS = \frac{dH}{T} - \frac{RTdP}{TP} \]  
(B.4)

You then get the integral of the above equation, which is shown below

\[ \int_{S_{\text{Inlet}}}^{S_{\text{Outlet}}} ds = \frac{1}{T} \int_{H_{\text{Inlet}}}^{H_{\text{Outlet}}} dH - R \int_{P_{\text{Inlet}}}^{P_{\text{Outlet}}} \frac{dP}{P} \]  
(B.5)

Once you solve the integrals you get

\[ S_{\text{Outlet}} - S_{\text{Inlet}} = \frac{H_{\text{Outlet}} - H_{\text{Inlet}}}{T} - R \ln \frac{P_{\text{Outlet}}}{P_{\text{Inlet}}} \]  
(B.6)

Since we are solving for the ideal outlet enthalpy, it means that there is no change in entropy. With that we get
\[
\frac{H_{\text{Outlet}} - H_{\text{Inlet}}}{T} - R \ln \frac{P_{\text{Outlet}}}{P_{\text{Inlet}}} = 0
\]  \hspace{1cm} (B.7)

which we can rearrange to get Eq. B.8 which is the ideal outlet enthalpy of the turbine section

\[
H_{\text{Outlet, Ideal}} = H_{\text{Inlet}} + RT \ln \frac{P_{\text{Outlet}}}{P_{\text{Inlet}}}
\]  \hspace{1cm} (B.8)

Then you use \(H_{\text{Outlet, Ideal}}\) in Eq.B.9 to get the actual outlet enthalpy of the turbine section. From there, you use a thermodynamic look up table along with outlet enthalpy and pressure to get the outlet temperature of the turbine.

\[
H_{\text{Outlet}} = H_{\text{Inlet}} - (H_{\text{Inlet}} - H_{\text{Outlet, Ideal}}) \eta_T
\]  \hspace{1cm} (B.9)
LIST OF REFERENCES


