INCREASING ISENTROPIC EFFICIENCY WITH HYDROSTATIC HEAD AND VENTURI EJECTION IN A RANKINE POWER CYCLE

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ABSTRACT

Increasing Isentropic Efficiency with Hydrostatic Head and Venturi Ejection in a Rankine Power Cycle

Nathan Daniel Ruiz

This thesis describes the modifications made to the Cal Poly Thermal Science Laboratory's steam turbine experiment. While the use of superheating or reheating is commonly used to increase efficiency in a Rankine cycle the methods prove unfeasible in a small scale project. For this reason, a mathematical model and proof of concept design using hydrostatic head generated by elevation and venturi ejection for use by the condenser is developed along with the equations needed to predict the changes to the system. These equations were used to create software to predict efficiency as well as lay down the foundation for future improvements of the system.

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First I would like to thank my family for helping and supporting me through my time at college. I would not be here if it wasn't for your continued love and care.

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CHAPTER 1

INTRODUCTION

The Rankine cycle is a thermodynamic cycle used to describe the process of converting heat to mechanical work through the use of a heat engine. Used in power generation plants the Rankine cycle is behind the steam turbine system which is responsible for most of the electricity generated in America. Due to its importance in the engineering world exposing engineering students to a complete working Rankine cycle can greatly benefit their practice of thermodynamics and turbo machinery. This thesis describes the changes made to the condenser of the steam turbine experiment used in the Cal Poly's thermal science laboratory in order to bring the system to a working condition using a new hydrostatic head vacuum based concept.

Traditionally, to increase the efficiency of a Rankine cycle the use of superheating, reheating, or regeneration have been used to modify the cycle (Wood 21). Although these are the improvement methods most frequently seen in industry they involve the addition of several new components to the system which can be problematic for an educational experiment where space and money are usually an issue. Additionally these methods generally deal with improving the heating process of the cycle, increasing the high pressure to the highest value possible, and rarely focus on improving the system by effecting the condenser. This paper will describe a new improvement method that is applied to the condenser which uses the elevation of the building the system occupies to create hydrostatic head along with the use of venturi ejection to lower the pressure in the condenser. In addition equations and software are developed to predict the changes in efficiency of the cycle which can be implemented easily into a laboratory manual for the

experiment. Also a new start up procedure that is specifically made for this experiment including all new modifications done is included. Since this modification is a proof of concept a permanent solution is also discussed.

CHAPTER 2

STEAM TURBINE SYSTEM

Figure 2.1 shows the basic components of a Rankine power cycle. These components include a boiler, turbine, condenser and pump. The working fluid, in this case water, is heated up in the boiler and leaves at high pressure and temperature [T_1, P_1]. The steam enters the turbine for an expansion process and the thermal energy in the steam is converted to mechanical work. Upon leaving the turbine the steam is a high quality liquid-vapor with low pressure [T_2, P_2]. Inside the condenser the steam condenses on the outside of the heat exchanger tubes which have the cooling water running through them. This process lowers the pressure inside the condenser. The condensate, now a saturated liquid, collects at the bottom of the condenser [T_3, P_3] and is then pumped back to the boiler [T_4, P_4]. This completes the Rankine cycle. For the ideal Rankine power cycle it is assumed the working fluid is flowing reversible and adiabatic in the turbine and pump and at constant pressure while through the boiler and condenser due to negligible frictional pressure drops.

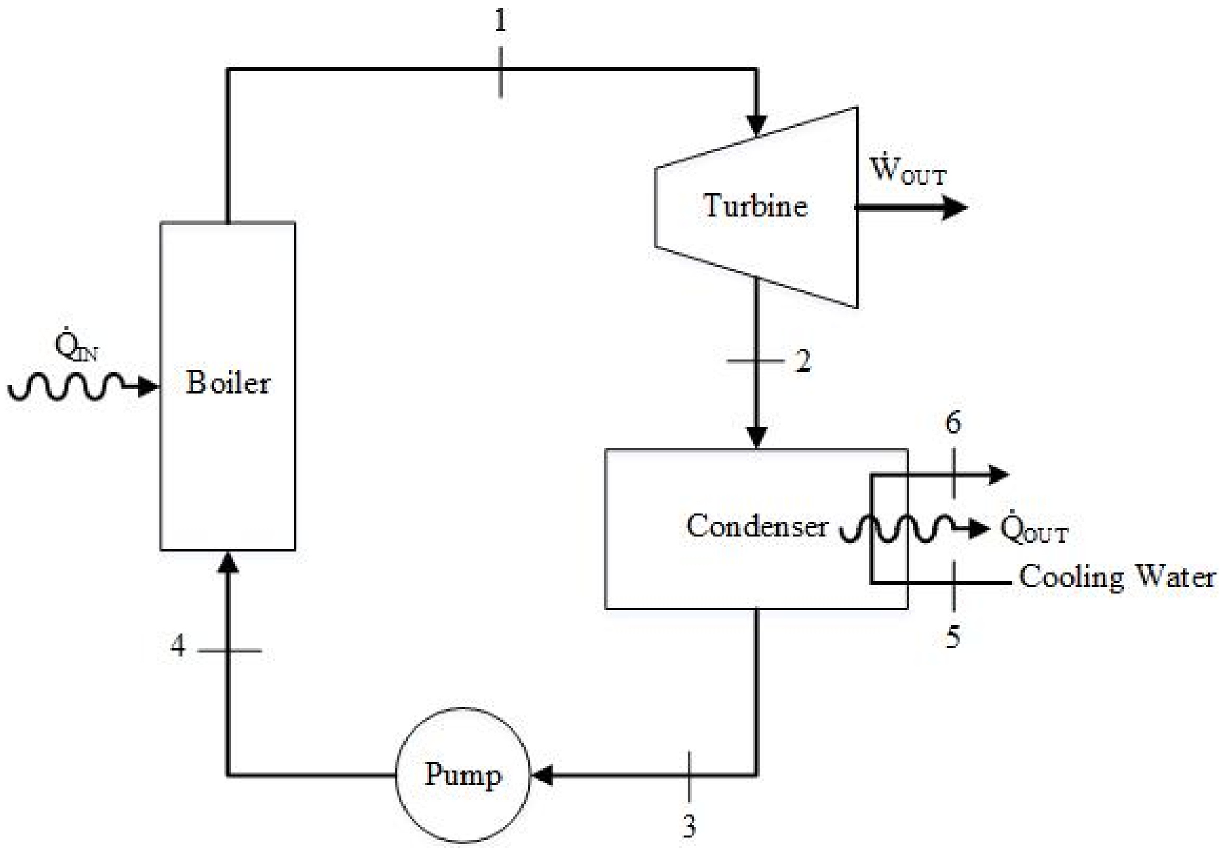

Figure 2.1: Basic Steam Turbine Rankine Cycle

The Cal Poly's steam turbine experiment contains each of these 4 major components along with a cooling tower and dynamometer to measure the work output of the turbine. Prior to 1998 the steam turbine experiment involved a General Electric single stage 50 horsepower steam turbine, Figure 2.2, powered by steam provided by a central boiler room on campus. In 1998 the central boiler room was taken down as the campus changed from a central steam system to a hot water system. The turbine experiment was replaced with a self-contained cycle that included a Tuthill Coppus 4.8 horsepower single stage turbine, a Fulton steam boiler, Wiegmann & Rose steam condenser, Evapco cooling tower, and Armstrong pump were installed to provide the necessary requirements to operate the Tuthill steam turbine. Figure 2.3 shows the Tuthill steam turbine system with

the condenser underneath. Figure 2.4 shows the Fulton steam boiler alongside the Evapco cooling tower. Appendix A has a complete schematic of this system which includes these components as well as all of the sensors and valves. Appendix B contains certain selections of the manuals for the turbine and boiler. Appendix C contains the pump curves for the Armstrong pump.

This current boiler is capable of generating enough steam to power the turbine under a no load condition. When any substantial load is applied by the dyno the system shuts down even though the turbine is rated at 4.8 horsepower. During preliminary tests the system runs at steady state up until 0.5 horsepower after which the boiler pressure starts to drastically fluctuate until finally the steam column inside the boiler fully collapses and the turbine loses all pressure steam. The core reasons behind the failure were discovered to be the atmospheric condenser pressure and boiler fluctuations. Each component was operating as designed so the operating problems did not arise from them being inherently broken. Mainly the boiler was not designed to produce the amount of steam needed to operate the turbine under a load. Simply replacing each device with an upgraded version could fix the problems, but proved to be too costly. So instead, the alterations discussed in the next chapter were implemented including the addition of the new hydrostatic head pressure system for lowering the condenser pressure for which this thesis is based upon.

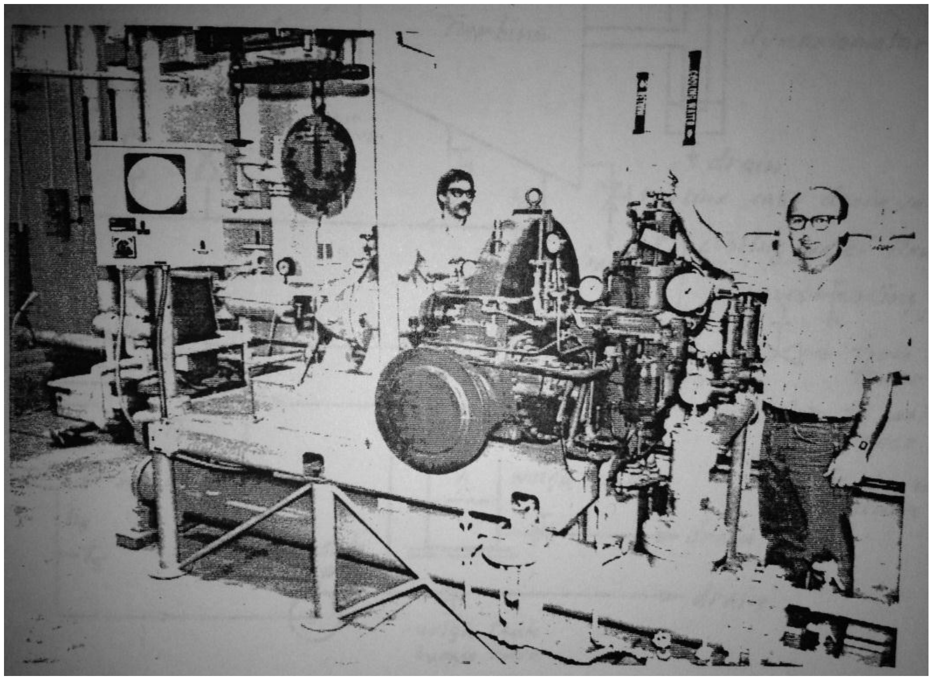

Figure 2.2: Original 50 HP Steam Turbine System

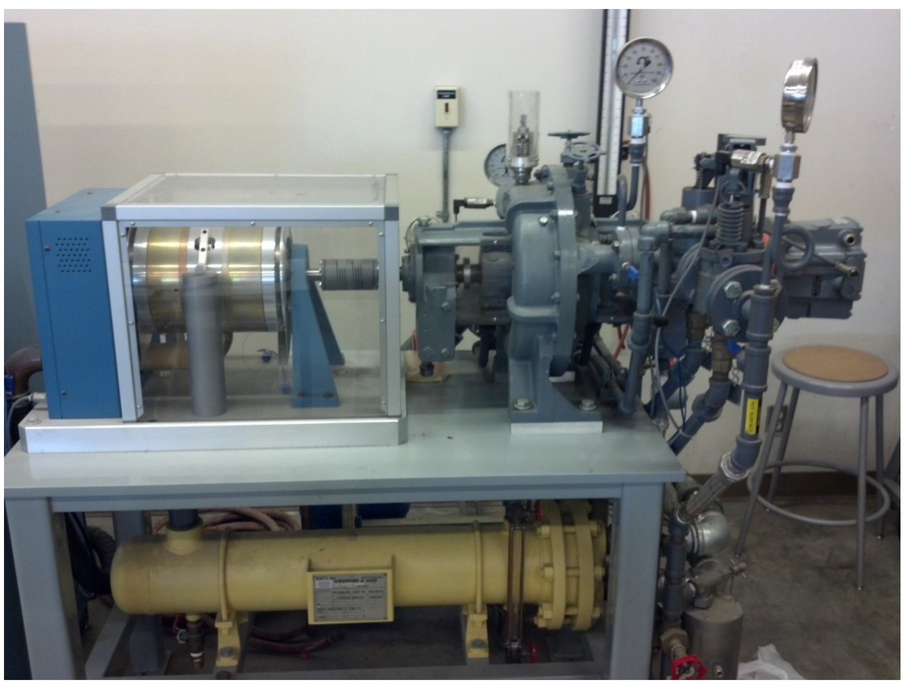

Figure 2.3: 4.8 HP Tuthill Steam Turbine System with W&R Condenser

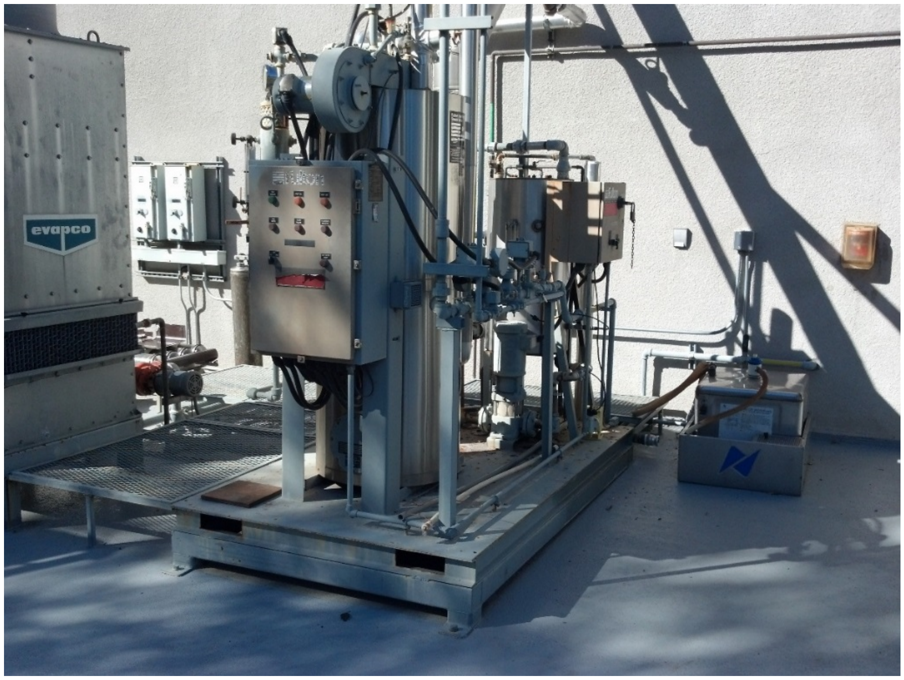

Figure 2.4: Fulton Steam Boiler and Evapco Cooling Tower

CHAPTER 3

SYSTEM MODIFICATIONS

Figure 3.1 shows a modified version of the basic steam turbine Rankine cycle, Figure 2.1, to reflect the actual steam turbine Rankine cycle. This mathematical model is for the actual system including all the new modifications. These modifications can be split into two categories: preliminary system changes and condenser vacuum changes.

Regarding the changes to the model, in order to get the model to correctly represent the actual system, the process path between the boiler and turbine is split in to two paths. This is to account for the pressure loss due to some of the steam splitting off and being used as sealing steam for the turbine shaft. For simplicity, the states are labeled with the terms defined by the turbine systems data logger. A full description of the exact values produced by the data logger is listed later in this report.

The preliminary system changes were done before the baseline test to allow for the system to run at no load for as long as possible. This allowed for the most accurate data, therefore distinguishing a clear difference in values between the old and new system. The first of these changes was performed on the boiler. Originally the high pressure steam lock out sensor permitted the boiler to cycle between 60 to 75 psig. This sensor was replaced to allow the boiler to cycle between 85 to 95 psig. Since the boiler has a maximum pressure rating of 100 psig the safety cut off and lock out on the boiler controls were adjusted to respond quicker to a pressure increase. Also as the boiler scheme stands some of the high pressure steam is bled off to act as a reheat for any water entering the

boiler. The optimized amount of steam to bleed off was found to allow the boiler to cycle while still producing the highest pressure steam possible.

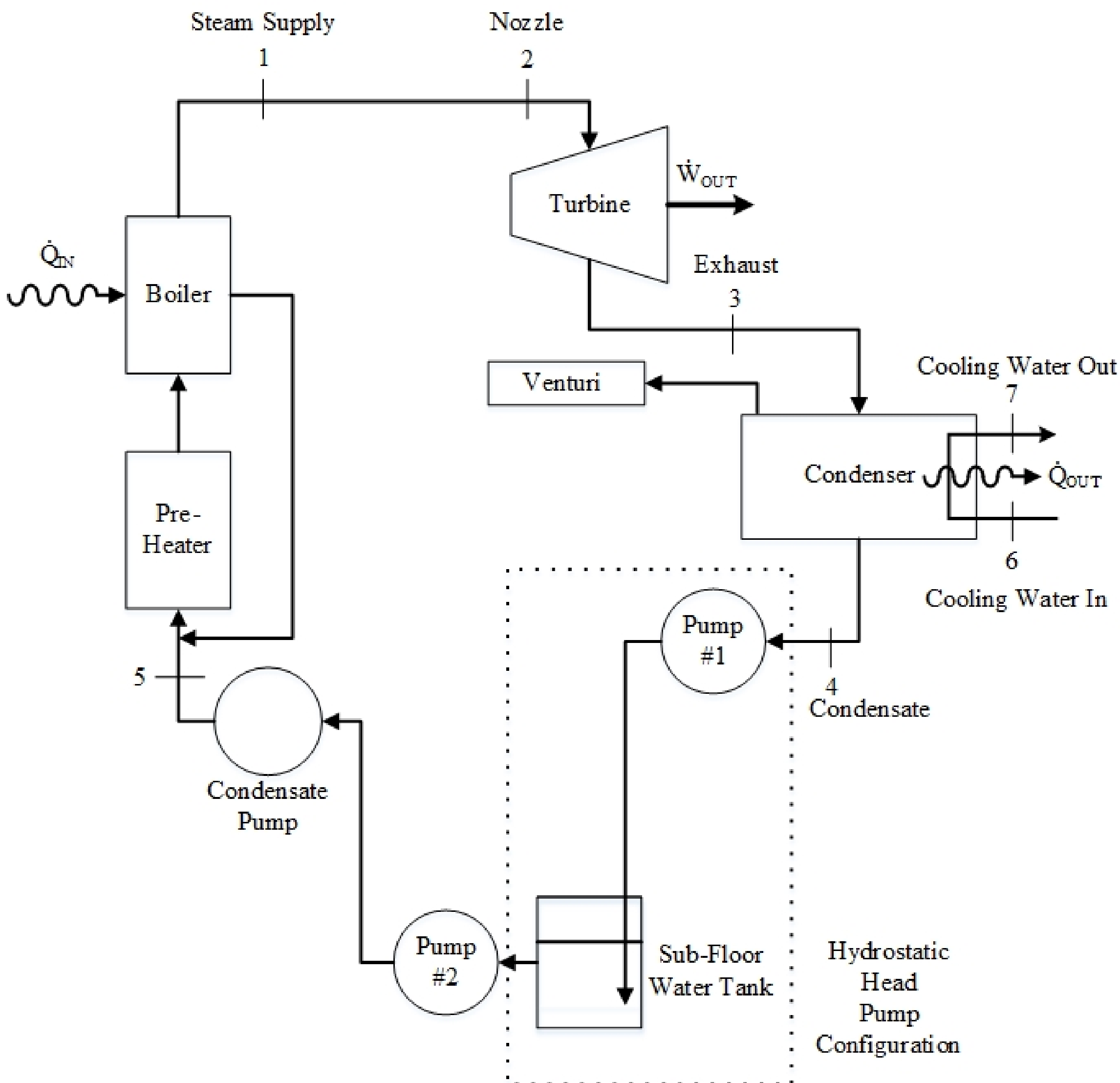

Figure 3.1: Modified Steam Turbine Rankine Cycle

Next the piping connecting the bottom of the condenser to the condensate pump was corrected to accurately connect both ends to the flow. Figure 3.2 shows the original layout of the condenser. Out of the four condenser valves only the two for exhaust and condensate flow were attached to the rest of the system with the other two closed off.

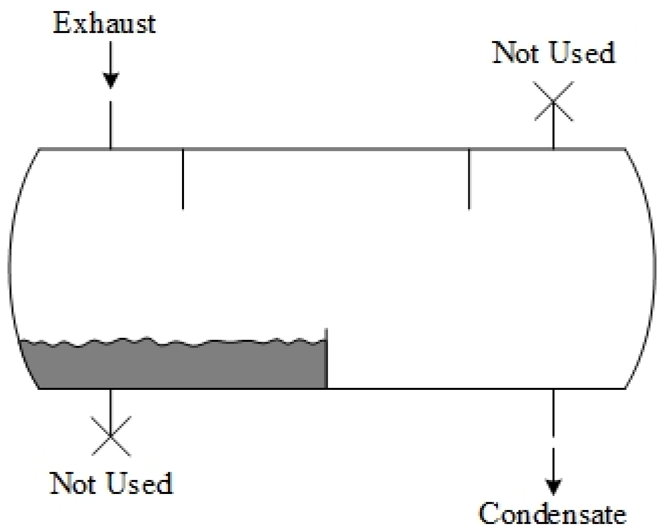

Figure 3.2: Cross Section Diagram of Original Condenser Layout

Looking at the diagram of the condenser revealed an inner lip inside the condenser where condensate could collect on the bottom left and remain in the condenser. This greatly limited the surface area where condensation occurred. Piping was installed to connect both bottom valves of the condenser together before running into the condensate pump. This kept all of the working fluid in the system reducing the load on the boiler and now the sight glass connected between the two rightmost valves accurately display the water level inside the condenser. Additionally the rightmost not used valve was used for the venturi discussed later. This can be seen in Figure 3.3.

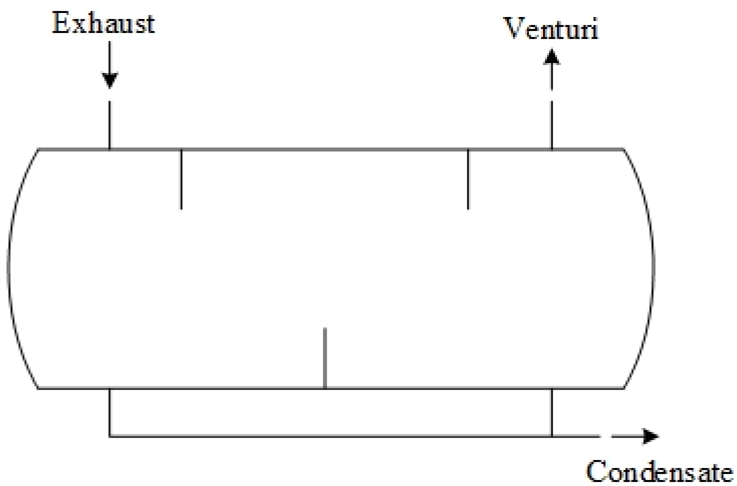

Figure 3.3: Cross Section Diagram of New Condenser Layout

Finally a way to quickly assess the data coming from the steam turbine was implemented. All of the pressure and temperature sensors along with the RPM and torque data from the dynamometer runs into a Hydra Logger Data Bucket which was used to only display one value at a time. LabVIEW is used to assess the data from a computer as well as control the Hydra Logger's settings. The control loop, front panel interface, and command list used to access the Hydra Logger and write the data to a text file can be found in Appendix F.

Once the preliminary changes were complete the subsequent series of changes were implemented to directly increase the condenser vacuum. The first was the installation of a venturi ejection to remove the incondensable gases, mostly air, found in the condenser. These gases interrupt the condensing steam process thus preventing a vacuum from developing. They exist inside the condenser shell during startup when the inside is at atmospheric conditions and can enter through the system's piping that is not sealed properly. However the main source of the incondensable gases during operation is

through the sealing steam injection used on the seals of the turbine shaft, and the reason behind the slight pressure difference between states 1 and 2 as mentioned earlier. This sealing steam is necessary to keep the atmospheric air conditions outside of the turbine shaft from entering the turbine, as seen in Figure 3.4

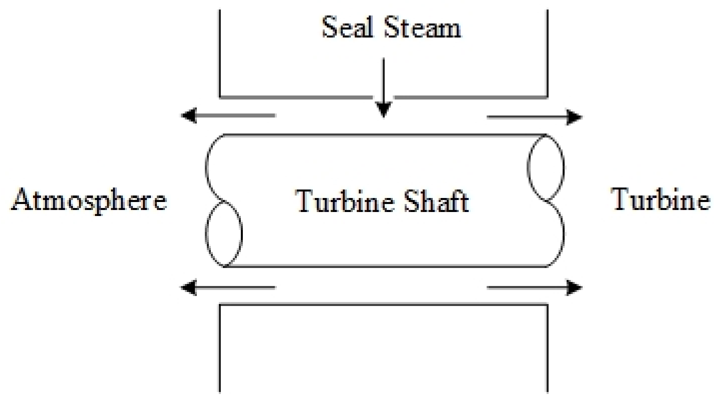

Figure 3.4: Cross Section of Seal Steam Injection

The venturi ejection uses the Bernoulli principal of increasing velocity in a nozzle to decrease pressure. The decrease in pressure provides a vacuum effect inside the chamber that pulls incompressible gasses from the vessel and ejects them to the atmosphere through a diffuser, as seen in Figure 3.5.

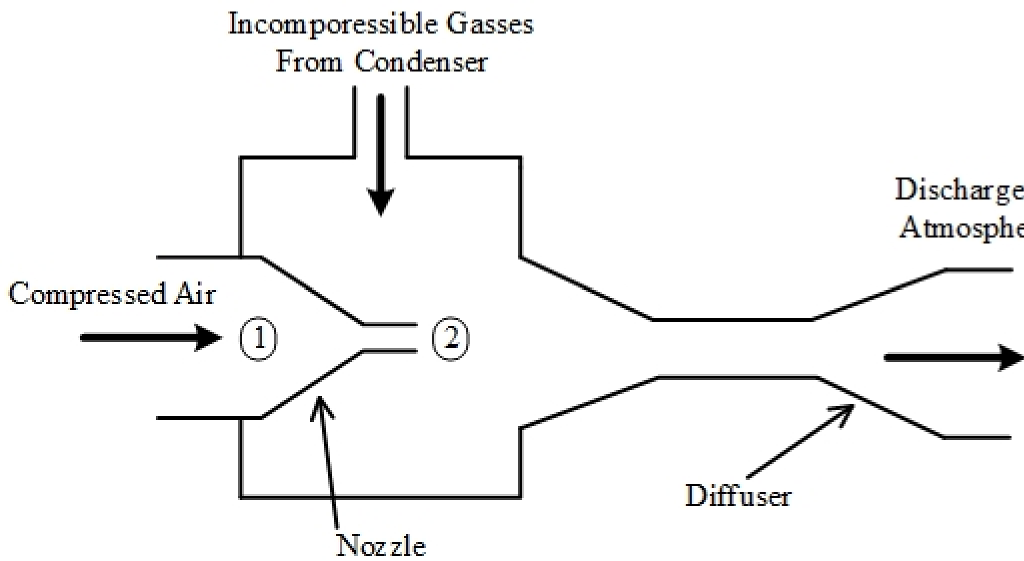

Figure 3.5: Cross Section of Venturi Ejection

The operation of this device is modeled using the Bernoulli Equation and Conservation of Mass with the assumption that flow remains horizontal ($Z_1 = Z_2$) and the fluid is incompressible ($\rho_1 = \rho_2$).

$$P_1+\frac{1}{2}\rho_1 V_1^2+Z_1 = P_2+\frac{1}{2}\rho_2 V_2^2+Z_2$$

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

$$V_2 = V_1 \frac{A_1}{A_2}$$

Although the venturi allows for a vacuum to occur it does not directly assist in the size of the vacuum developed. The venturi chosen is rated down to 1.9 psia which is far below any vacuum calculated for the condenser shell. Instead the vacuum will be directly caused by the hydrostatic head pump configuration which will rely on the vertical

distance of the water column between the condenser and the sub-floor water tank, the head loss caused by friction in the pipes, and the net positive suction head available from the pumps used. This configuration consisted of piping to connect the condenser exhaust flow to a tank located beneath the system as well as 2 pumps, one to assist the condensate flow to the ground floor and the other to pump the condensate back up to the original condensate pump. The exact pressure difference caused by this system is calculated in Chapter 5.

CHAPTER 4

THERMODYNAMIC EQUATIONS

The first part of analyzing this model is a system of thermodynamic equations to calculate the actual and ideal work done by the turbine and the heat flow inside the condenser. With this information the isentropic efficiency and heat transfer can be calculated. All of the equations developed including the pressure equations introduced in the following chapter are used to develop the software presented later.

In order for this thesis to be developed into a laboratory, the analysis will start with the most basic equation and move outward. Each device is modeled as an open system so each equation will be derived from the Reynolds transport theorem where N is the extensive property and η is the intensive property. For simplicity "i" will be denoted as each of the components inlet and "e" will be denoted as each of the components exit.

$$N=\eta*m$$

$$\left.\frac{\partial N}{\partial t}\right|_{System}=\frac{\partial}{\partial t}\int_{CV}\eta\rho d\forall+\int_{CSi}\eta\rho\vec{V}\cdot\vec{n}\,dA+\int_{CSe}\eta\rho\vec{V}\cdot\vec{n}\,dA$$

For the devices in question the Conservation of Mass and 1st law of Thermodynamics will be applied. The 2nd law of Thermodynamics is touched on lightly when discussing the turbine. Solving the Reynolds transport equation for Conservation of Mass develops the following:

$$\left.\frac{\partial m}{\partial t}\right|_{System}=0$$

$$\frac{\partial}{\partial t}\int_{CV}\rho d\forall + \int_{CSi}\rho\vec{V}\cdot\vec{n}\,dA+\int_{CSe}\rho\vec{V}\cdot\vec{n}\,dA=0$$

While this system is certainly not operating under ideal conditions certain assumptions will be made to make the analysis easier. First, each device is assumed to be quasi-static. This means the device is operating at thermodynamic equilibrium with no acceleration. Another effect is that there are no temperature gradients or pressure waves across the device. Although no real system is quasi-static, if the process is slow enough, this assumption can be valid. Moreover quasi-static must be assumed in order to use the thermodynamic equations and diagrams developed. Next each device is operating at steady state before data is recorded, so steady state is also a safe assumption. The pipes connecting each device are assumed to be inviscid (μ=0) meaning pressure losses due to friction can be neglected and that velocity is not a function of pipe area. Last, each device is assumed to have constant properties at the control surfaces. Applying these assumptions to the Conservation of Mass results in the following:

$$\cancel{\frac{\partial}{\partial t}\int_{CV}\rho d\forall} - \rho V\int_{CSi}dA + \rho V\int_{CSe}dA = 0$$

$$-(\rho VA)_i + (\rho VA)_e = 0$$

$$\dot{m}_i = \dot{m}_e = \dot{m}$$

Next, solving the Reynolds transport equation for the 1st law of Thermodynamics develops the following:

$$\left.\frac{\partial E}{\partial t}\right|_{System} = \dot{Q}_{IN} - \dot{W}_{OUT}$$

$$\frac{\partial}{\partial t}\int_{CV} e\rho d\forall + \int_{CSi}(e+pv)\rho\vec{V}\cdot\vec{n}\,dA+\int_{CSe}(e+pv)\rho\vec{V}\cdot\vec{n}\,dA = \dot{Q}_{IN}-\dot{W}_{OUT}$$

Where

$$e = \frac{V^2}{2}+gz+u$$

$$pv \equiv \text{Flow Work (Normal Stress)}$$

Therefore

$$\frac{\partial}{\partial t}\int_{CV}\left(\frac{V^2}{2}+gz+u\right)\rho d\forall + \int_{CSi}\left(\frac{V^2}{2}+gz+u+pv\right)\rho\vec{V}\cdot\vec{n}\,dA$$

$$+\int_{CSe}\left(\frac{V^2}{2}+gz+u+pv\right)\rho\vec{V}\cdot\vec{n}\,dA = \dot{Q}_{IN}-\dot{W}_{OUT}$$

All of the assumptions made for the Conservation of Mass (quasi-static, steady state, inviscid, constant properties at control surfaces) are also applied here. Furthermore the kinetic and potential energies are assumed negligible when compared to the internal energy.

Applying these assumptions to the 1st law results in the following:

$$\cancel{\frac{\partial}{\partial t}\int_{CV}\left(\frac{V^2}{2}+gz+u\right)\rho d\forall} + \int_{CSi}\left(\cancel{\frac{V^2}{2}}+\cancel{gz}+u+pv\right)\rho\vec{V}\cdot\vec{n}\,dA$$

$$+\int_{CSe}\left(\cancel{\frac{V^2}{2}}+\cancel{gz}+u+pv\right)\rho\vec{V}\cdot\vec{n}\,dA = \dot{Q}_{IN}-\dot{W}_{OUT}$$

Using the definition of enthalpy

$$h=u+pv$$

The previous equation becomes

$$-h\rho V\int_{CSi} dA + h\rho V\int_{CSe} dA = \dot{Q}_{IN} - \dot{W}_{OUT}$$

$$-h_i\dot{m}_i + h_e\dot{m}_e = \dot{Q}_{IN} - \dot{W}_{OUT}$$

Applying Conservation of Mass developed earlier this equations yields:

$$\dot{m}(h_e - h_i) = \dot{Q}_{IN} - \dot{W}_{OUT}$$

Last all of the previous assumptions and the Conservation of Mass can be applied to the 2nd Law of Thermodynamics.

$$\left.\frac{\partial S}{\partial t}\right|_{System} = \frac{\dot{Q}_{IN}}{T_b} + \dot{\sigma}$$

$$\frac{\partial}{\partial t}\int_{CV} s\rho d\forall + \int_{CSi} s\rho\vec{V}\cdot\vec{n}\, dA + \int_{CSe} s\rho\vec{V}\cdot\vec{n}\, dA = \frac{\dot{Q}_{IN}}{T_b} + \dot{\sigma}$$

$$-s\rho V\int_{CSi} dA + s\rho V\int_{CSe} dA = \frac{\dot{Q}_{IN}}{T_b} + \dot{\sigma}$$

$$-s_i\dot{m}_i + s_e\dot{m}_e = \frac{\dot{Q}_{IN}}{T_b} + \dot{\sigma}$$

$$\dot{m}(s_e - s_i) = \frac{\dot{Q}_{IN}}{T_b} + \dot{\sigma}$$

Referring back to Figure 3.1, the Conservation of Mass and 1st law of Thermodynamics is applied to the turbine, boiler, condenser, and condensate pump. Additionally the Tds equation will be applied to the condensate pump which can be seen below:

$$Tds = dh - vdp$$

Instead of Reynolds transport equations, Bernoulli's equation, along with pipe loss equations, are applied to the hydrostatic head pump configuration pumps to solve for the vacuum pressure at state 4.

Figure 4.1 shows the turbine along with its control volume

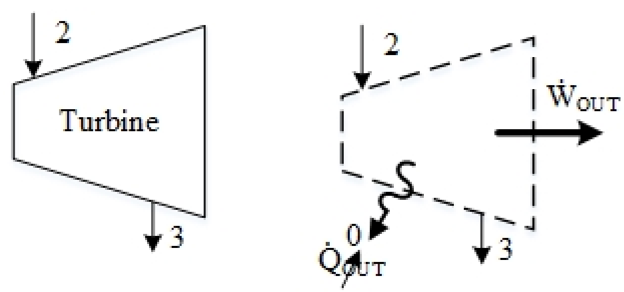

Figure 4.1: Rankine Cycle Turbine Control Volume

The Conservation of Mass shows

$$\dot{m}_2 = \dot{m}_3 = \dot{m}$$

Next applying the 1st of Thermodynamics

$$\dot{m}(h_3 - h_2) = \dot{Q}_{IN} - \dot{W}_{OUT}$$

Because the turbine is assumed adiabatic heat transfer becomes negligible.

$$\dot{m}(h_3 - h_2) = -\dot{W}_{OUT}$$

$$\dot{m}(h_2 - h_3) = \dot{W}_{OUT}$$

$$\frac{\dot{W}_{OUT}}{\dot{m}} = h_2 - h_3$$

Whatever heat actually lost is instead seen in the isentropic efficiency which is defined below:

$$\eta_{\text{Isentropic}} = \frac{\dot{W}_{\text{Actual}}}{\dot{W}_{\text{Ideal}}}$$

Since the 1st law applies to both the ideal and actual scenario:

$$\eta_{\text{Isentropic}} = \frac{\dot{m}\left(h_2 - h_{3,\text{actual}}\right)}{\dot{m}\left(h_2 - h_{3,\text{ideal}}\right)}$$

$$\eta_{\text{Isentropic}} = \frac{h_2 - h_{3a}}{h_2 - h_{3s}}$$

The data for the actual case is recorded by the system so h_2 and h_{3a} can be read off the table of data. To find h_{3s} the 2nd law of thermodynamics analysis is applied. In an ideal system the turbine is assumed to be adiabatic and reversible. Therefore the 2nd law for an ideal turbine becomes:

$$\dot{m}(s_3 - s_2) = \cancel{\frac{\dot{Q}_{IN}}{T_b}} + \cancel{\dot{\sigma}}$$

$$\dot{m}(s_3 - s_2) = 0$$

$$s_3 = s_2$$

This proves there is no change in entropy in an ideal turbine. Another important fact that is that since there is no super heater the steam coming into the turbine is a saturated vapor, the state lies on the vapor dome. Using this fact along with the temperature of state 3 being less then state 2 and that there is no change in entropy means state 3 is underneath the dome and therefore has a quality. Next all that is needed is to recognize

that the exit pressure is the same as the condenser pressure, P_3, and the quality of h_{3s} can be found. This is done by manipulating the equation for vapor quality below and solving for the s_f and s_g values at P_3.

$$s = s_f + x \cdot (s_g - s_f)$$

$$\frac{(s-s_f)}{(s_g-s_f)} = x$$

Once the quality is found P_3 and T_3 can be used to solve for h_{3s}

Figure 4.2 shows the boiler along with its control volume

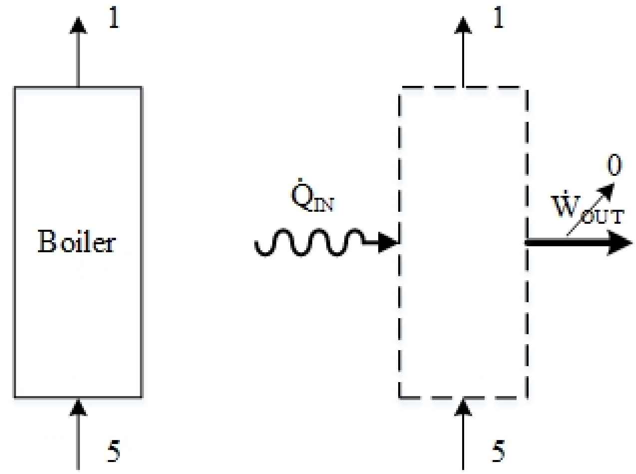

Figure 4.2: Rankine Cycle Boiler Control Volume

The Conservation of Mass shows

$$\dot{m}_5 = \dot{m}_1 = \dot{m}$$

Next applying the 1st of Thermodynamics

$$\dot{m}(h_1 - h_5) = \dot{Q}_{IN} - \dot{W}_{OUT}$$

The boiler has no shaft or moving boundary layer therefore no work done so:

$$\dot{m}(h_1 - h_5) = \dot{Q}_{IN}$$

$$\frac{\dot{Q}_{IN}}{\dot{m}} = h_1 - h_5$$

Figure 4.3 shows the condensate pump along with its control volume

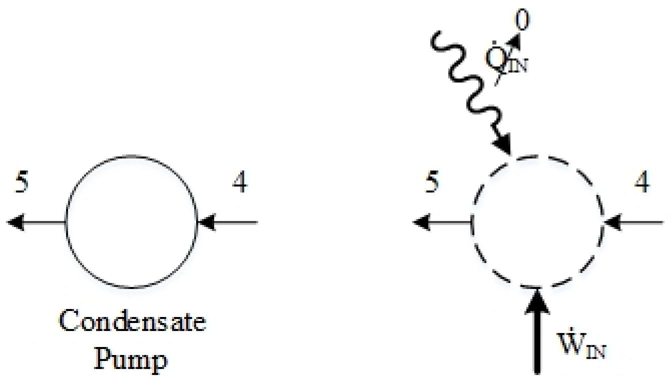

Figure 4.3: Rankine Cycle Condensate Pump Control Volume

The Conservation of Mass shows

$$\dot{m}_4 = \dot{m}_5 = \dot{m}$$

Next applying the 1st of Thermodynamics

$$\dot{m}(h_5 - h_4) = \dot{Q}_{IN} - \dot{W}_{OUT}$$

Because the turbine is assumed adiabatic heat transfer becomes negligible.

$$\dot{m}(h_5 - h_4) = -\dot{W}_{OUT}$$

$$\dot{m}(h_5 - h_4) = \dot{W}_{IN}$$

$$\frac{\dot{W}_{IN}}{\dot{m}} = h_5 - h_4$$

There is no sensors to record the data at state 5 so to get the value of work in the Tds equation is applied

$$Tds = dh - vdp$$

Just like with the turbine the condensate pump is also assumed reversible as well as adiabatic this means there is no change in entropy across the pump.

$$T\cancel{ds} = dh - vdp$$

$$dh = vdp$$

Now with specific volume and a pressure difference work of the pump can be found.

Figure 4.4 shows the condenser along with its control volume encompassing both flows

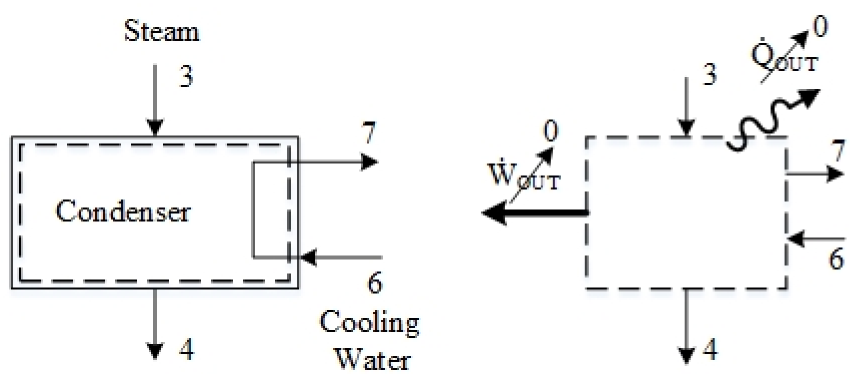

Figure 4.4: Rankine Cycle Condenser Control Volume for Both Flows

When analyzing the condenser the heat transfer by the system changes depending on how the control volume is defined. Work is always assumed zero but if the control volume

encompasses only one of the flows inside the condenser then there is heat transfer between the hot and cold streams while if the control volume encompasses both flows there is no heat transfer. First, a control volume encompassing both flows is used and therefore the system is assumed to be adiabatic. This will allow the ratio of flow rates, steam flow rate by cooling water flow rate, to be found only using the changes in enthalpy for each stream. This is done because the mass flow rate of the cooling water is read using a flow meter while the mass flow rate of steam is unknown. The legitimacy of the adiabatic assumption is proven in the heat transfer analysis of the condenser shell.

The Conservation of Mass and assuming no mixing of streams shows

$$-\dot{m}_3+\dot{m}_4-\dot{m}_6+\dot{m}_7 = 0$$

$$\dot{m}_3 = \dot{m}_4=\dot{m}_{Steam} \quad , \quad \dot{m}_6 = \dot{m}_7=\dot{m}_{Cooling}$$

Next applying the 1st of Thermodynamics

$$\dot{m}_{Steam}(h_4- h_3)+\dot{m}_{Cooling}(h_7- h_6) =\dot{Q}_{IN}-\dot{W}_{OUT}$$

$$\dot{m}_{Steam}(h_4- h_3)+\dot{m}_{Cooling}(h_7- h_6) =0$$

$$-\dot{m}_{Steam}(h_4- h_3)=\dot{m}_{Cooling}(h_7- h_6)$$

$$\dot{m}_{Steam}(h_3- h_4)=\dot{m}_{Cooling}(h_7- h_6)$$

$$\dot{m}_{Steam}=\dot{m}_{Cooling}\frac{(h_7- h_6)}{(h_3- h_4)}$$

In order to make the above assumption the heat transfer lost from the condenser to the environment is proven negligible. If heat loss is taken into account then following equation is used instead:

$$\dot{m}_{Steam}=\frac{\dot{m}_{Cooling}(h_7- h_6)-\dot{Q}_{LOSS}}{(h_3- h_4)}$$

The heat transfer loss is modeled as only free natural convection and radiation. Figure 4.5 shows the condenser modeled as a cylinder with dimensions given.

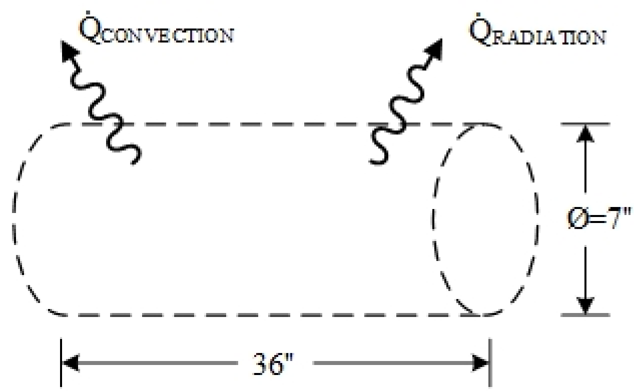

Figure 4.5: Condenser Modeled as a Cylinder for Heat Transfer Analysis

The heat loss is then calculated with the following (Incropera 554):

$$\dot{Q}_{Loss}=\dot{Q}_{Conv}+\dot{Q}_{Rad}$$

$$\dot{Q}_{LOSS}=\bar{h}A(T_{Condenser}-T_{Room})+ \sigma\varepsilon A({T_{Condenser}}^4-{T_{Room}}^4)$$

Where

$$\bar{h}=\frac{k}{D}\overline{Nu_D}$$

And the view factor, F, is assumed 1

Before solving for the Nusselt number the Rayleigh number is calculated:

$$\mathrm{Ra_D} = \frac{g\beta(T_S - T_\infty)}{\nu\alpha}$$

For a Rayleigh number range of

$$10^{-5} < \mathrm{Ra_D} < 10^{12}$$

The equation for Nusselt number is as follows:

$$\overline{Nu}_D = \left\{0.60 + \frac{0.387 * Ra_D{}^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}}\right\}^2$$

Second, a control volume only encompassing the steam flow is used which can be seen below in Figure 4.6

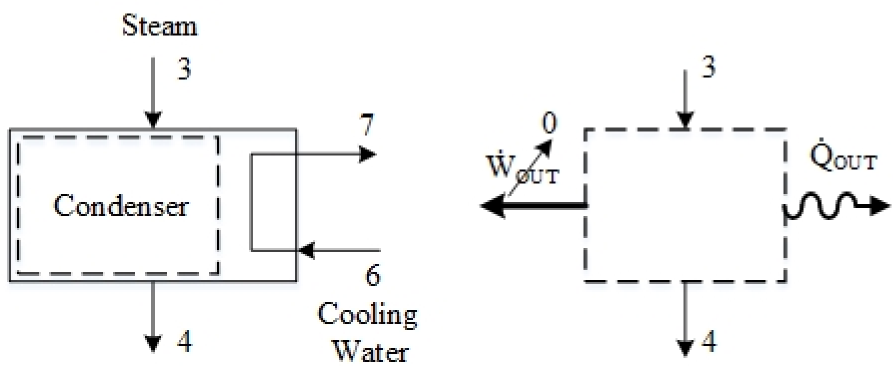

Figure 4.6: Rankine Cycle Condenser Control Volume for Steam Flow

The Conservation of Mass shows

$$\dot{m}_3 = \dot{m}_4 = \dot{m}$$

Next applying the 1st of Thermodynamics

$$\dot{m}(h_4 - h_3) = \dot{Q}_{IN} - \dot{W}_{OUT}$$

$$\dot{m}(h_4 - h_3) = \dot{Q}_{IN}$$

$$\dot{m}(h_4 - h_3) = -\dot{Q}_{OUT}$$

$$\frac{\dot{Q}_{OUT}}{\dot{m}} = h_3 - h_4$$

All of the equations derived in this chapter are solved with the software provided.

CHAPTER 5

PRESSURE EQUATIONS AND PIPING MODEL

An essential part of this model is to predict the pressure in the condenser caused by the hydrostatic head pressure configuration. Although the pressure produced in the condenser is recorded by the Hydra Data Logger a model of predicting this pressure allows future piping designs to be tested and optimized. Figure 5.1 shows the layout of the piping system.

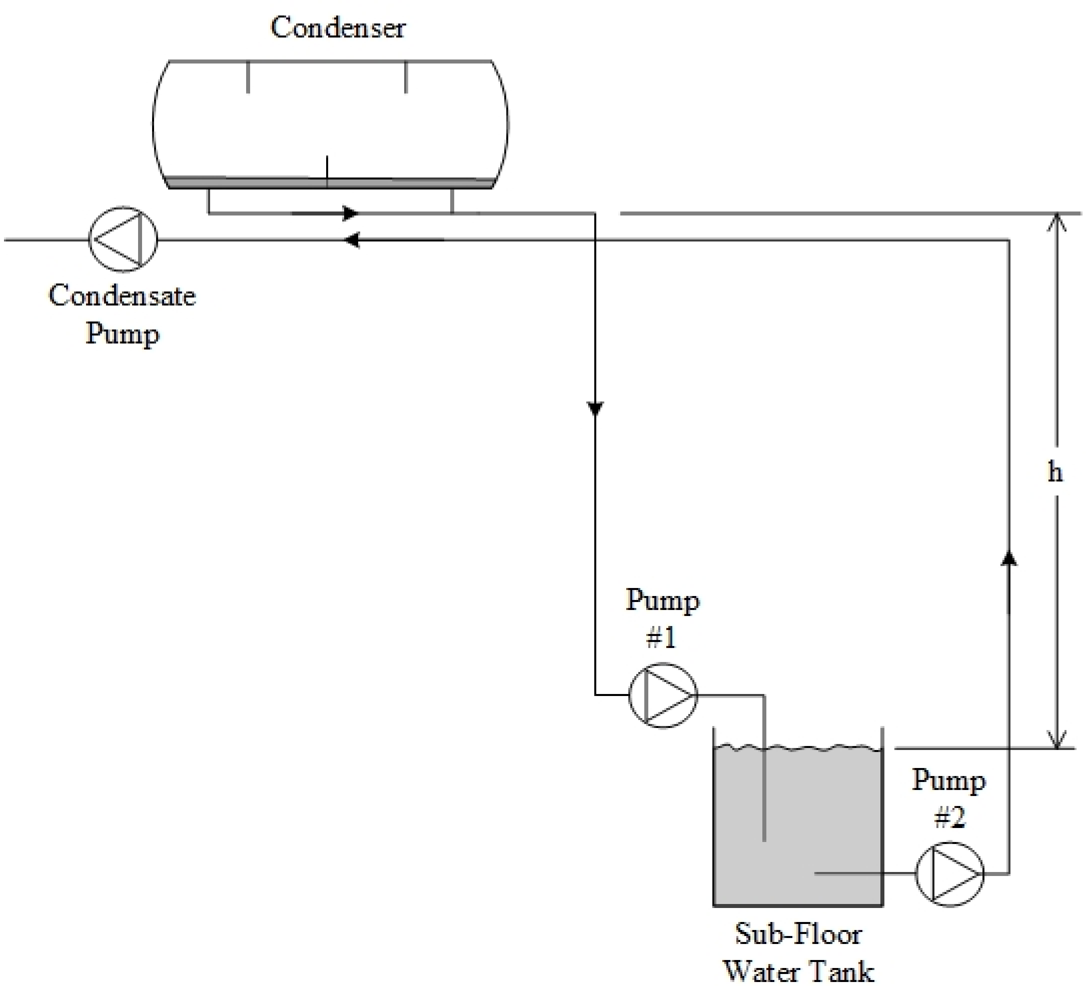

Figure 5.1: Hydrostatic Head Pump Configuration

To predict the vacuum pressure inside the condenser the original atmospheric condenser pressure, hydrostatic pressure, pressure loss due to friction, and pressure from the total pump head are taken into account. This is represented by the following equation:

$$P_{Vacuum}=P_{Condenser}-\Delta P_{Hydrostatic}-\Delta P_{Pump\ Head}+\Delta P_{Friction}$$

$P_{Hydrostatic}$ is negative due to the drop in elevation while $P_{Pump\ Head}$ is negative due to the pump's head assisting the flow. $P_{Friction}$ is always positive in the direction of the flow. $P_{Condenser}$ represents the initial pressure inside of the condenser which in this case is atmospheric therefore:

$$P_{Condenser}=14.7 \text{ psia}$$

$P_{Hydrostatic}$ is found by integrating the general differential equation which describes pressure and gravitational forces acting on a small element of fluid for constant density (Olson 42).

$$\frac{\partial p}{\partial z}=-\rho g$$

$$\int_{p_1}^{p_2} dp=-\rho g\int_{z_1}^{z_2} dz$$

$$p_2-p_1=-\rho g(z_2-z_1)$$

$$\Delta p=\gamma\Delta z$$

$$\Delta P_{Hydrostatic}=\gamma h$$

Where γ is specific weight of water and h is the height difference between the bottom of the condenser and the top of the water line in the sub-floor water tank.

$P_{Friction}$ is found using the Darcy-Weisback equation for fully developed incompressible flow in a circular pipe (Olson 313)

$$\frac{\Delta p}{L}=\frac{f}{D}\frac{\rho V^2}{2}$$

The length is relatively short so the entire piping system can be treated as one segment:

$$\Delta p=L\frac{f}{D}\frac{\rho V^2}{2}$$

Where L is length, D is diameter, f is friction factor, ρ is density, and V is velocity. Length and diameter come from the piping system lay out. Density of water is based on the temperature of the flow. Friction factor requires both the Reynolds number of the flow along with the pipe roughness. Reynolds number is based on velocity so start off the analysis flow rate is needed which was found before in the thermodynamic analysis.

$$\dot{m}=\rho VA$$

$$V=\frac{\dot{m}}{\rho A}$$

Next Reynolds number for a pipe can be found

$$Re_D=\frac{VD\rho}{\mu}$$

Substitute in kinematic viscosity, ν

$$Re_D=\frac{VD}{\nu}$$

The Reynolds number can then be calculated for each case. On average the flow of the system was around 0.094 lb_m/s which combined with the ¾ inch pipe used in the proof of concept designs meant a Reynolds number very close to around 10,000. This means assuming turbulent flow in the hydrostatic pipe became a safe assumption.

Since Reynolds will be between $4*10^3$ and 10^8 and the relative roughness of the pipes were never over 0.05 the following equation by Haaland (Olson 321) is used to find friction factor:

$$\frac{1}{\sqrt{f}}=-0.782 \ln\left[\frac{6.9}{Re_D}+\left(\frac{k}{3.7D}\right)^{1.11}\right]$$

Where k is the absolute roughness and D is the inner diameter of the pipe.

With this everything is known and therefore the change in pressure can be found using the equation stated earlier:

$$\Delta P_{Friction}=L\frac{f}{D}\frac{\rho V^2}{2}$$

Last $P_{Pump\ Head}$ is found by converting the pump's total dynamic head given by the manufacturer to pressure using the same equation used for finding $P_{Hydrostatic}$

$$\Delta P_{Pump\ Head}=\gamma h_{Total\ Dynamic\ Head}$$

Using these three pressure equations the vacuum pressure can be calculated. These equations are solved in the software provided.

CHAPTER 6

SOFTWARE

The Steam Turbine Rankine Cycle software takes the channel values read by the LabVIEW program and recorded by the Hydra Logger Data Bucket to calculate the turbine isentropic efficiency, total power produced by the turbine, heat loss in the condenser, and develop a temperature entropy (T-s) diagram to visualize the system.

From the Hydra Logger data the temperature and pressure of each of the fully known state points [defined before in Figure 3.1] are recorded along with cooling water flow rate, the speed and torque from the dynometer, and condenser shell temperature. Figure 6.1 shows the table in the software where this data gets inserted.

Table 6.1: Software Input Table

Steam Turbine Rankine Cycle
By: Nathan Ruiz
Steam Tables from Fundamentals of Engineering Thermodynamics 7th edition

Channel	Raw Value	Corrected Value	Units	Description
1	7.60E+01	76	°F	Ambient Air Temperature
2	2.98E+03	2984	RPM	RPM
3	8.53E-01	1	lb-ft	Torque
4	8.30E+01	83	°F	Dyno Temperature
5	7.50E+01	75	°F	Cooling Air In Temperature
6	8.50E+01	85	°F	Cooling Air Out Temperature
7	-3.00E-02			Cooling Air Flow Rate
8	2.11E+02	211	°F	Calorimeter Temperature
9	2.71E+02	271	°F	Steam Supply Temperature
10	2.71E+01	27	PSIG	Steam Supple Pressure
11	2.68E+02	268	°F	Nozzle Temperature
12	2.58E+01	26	PSIG	Nozzle Pressure
13	2.11E+02	211	°F	Exhaust Temperature
14	-5.00E-02	-0.050	PSIG	Exhaust Pressure
15	2.08E+02	208	°F	Condensate Temperature
16	-3.00E-03			Condensate Flow Rate
17	1.96E+02	196	°F	Condensor Shell Temperature
18	6.40E+01	64	°F	Cooling H_2O In Temperature
19	7.20E+01	72	°F	Cooling H_2O Out Temperature
20	4.36E+01	44	lb/s	Cooling H_2O Flow Rate

The software's steam property formula uses the temperature, pressure, and quality to find the steam properties (specific volume, internal energy, enthalpy, and entropy) of each point by interpolating their values from steam tables (Moran) that were transcribed into the software. The VBA software code can be seen in Appendix D. For this reason the tables are color coordinated to signify which points were from the raw data, read/interpolated off a table, solved for, or assumed. An extraction of the steam tables used can be found in Appendix E. In order to fill in the rest of the results the software relies on the equations developed in Chapter 4. The quality of ideal state 3 is found using the constant entropy conclusion and the assumption that both actual and ideal state 3 are at the condenser pressure. The condensate flow, and consequently due to the Conservation of Mass the flow throughout the entire system, is found using the various states enthalpy's as well as the recorded cooling water flow rate. Now that the both actual and ideal state 3 values are known, as well as the flow rate, the isentropic efficiency can be solved. Lastly, the software uses the temperature and entropy values to plot the state points and then using the pressure relationships presented in the steam tables plots the process paths between each point.

To justify the assumption that the condenser is adiabatic a free convection and radiation heat transfer analysis was performed modeling the condenser as a horizontal cylinder. The thermophysical properties table used in finding the variables (Incropera) is given in SI units so the parameters were changed to SI to finish the heat transfer investigation. After each test the condenser heat loss was found to be insignificant when compared to the change in enthalpy concluding that the adiabatic assumption was valid.

Last there is the ability to find the vacuum pressure inside the condenser by using the various equations derived in Chapter 5. The input parameters necessary are the total length of the pipe, the height difference between the two water levels, the pipe inner diameter, and the pipe's absolute roughness. Velocity is found from the flow rate solved in the first part of the software and the water state properties density, specific weight, and kinematic viscosity are found for water at the temperature exiting the condenser.

CHAPTER 7

PRESSURE AND EFFICIENCY RESULTS

First the system was left to run under no load until a steady state with the maximum pressure coming out of the boiler as possible. The Hydra Logger data values for this test can be seen before in Table 6.1. Table 7.1 has the results of the software and Figure 7.1 has the generated temperature entropy diagram. Next the system used the hydrostatic pressure head configuration using a rubber hose as the connective piping. The vacuum was only held for 10 seconds before the hose collapsed but the Hydra Logger data, Table 7.2, showed that the pressure decreased and that the isentropic efficiency increased. Table 7.3 has the results of this test and Figure 7.2 has the generated temperature entropy diagram. The heat transfer analysis of the system confirmed that the heat transfer loss of the condenser was negligible and had an insignificant effect on the mass flow rate of the system calculated. This can be seen in Table 7.4. As far as predicting the pressure drop inside the condenser Table 7.5 has the calculated pressure drop for the parameters of the hose test. Although the pressure recorded did not reach this value had the hose not collapsed the pressure could have approached this value. After this another test was performed using PVC pipe as the connective piping. Although the PVC pipe did not collapse when exposed to the vacuum pressure it could not maintain this pressure long enough to record accurate data. The reason behind this was due to there not being enough water in the pipes to maintain the vacuum at the location of the condenser. This problem can be solved by installing an additional tank directly underneath the condenser that would intercept the condensate before entering the hydrostatic head pump configuration.

Table 7.1: Software Results for No-Load Test

State	Temp.	Pres.	Quality	Specific Volume	Internal Energy	Enthalpy	Entropy
	T [°F]	P [lbf/in.^{2}]	x	v [ft^3/lb]	u [Btu/lb]	h [Btu/lb]	s [Btu/lb·°R]
1	271	41.8	1	9.9280	1094.03	1171.17	1.67199
2	268	40.5	1	10.4100	1093.44	1170.22	1.67581
3s	211	14.65	0.943	25.7540	1026.09	1094.74	1.67581
3a	211	14.65	1	27.3100	1077.31	1150.12	1.75835
4	208	14.65	0	0.0167	176.09	176.09	0.30608
5	233	41.8	0				0.30608
6	64	14.7	0	0.0160	32.09	32.09	0.06323
7	72	14.7	0	0.0161	40.09	40.09	0.07839

Legend
Solved
Look Up
Given
Assumed

Turbine Power [HP]	0.485	HP = Torque[lb-ft] × RPM
State 3 Ideal Quality	0.943	$\frac{(s-s_f)}{(s_g-s_f)} = x$
Condensate Flow Rate [lb/s]	0.358	$\dot{m}_{Steam} = \dot{m}_{Cooling} \frac{(h_7 - h_6)}{(h_3 - h_4)}$
Actual Isentropic Efficiency	0.048	$\eta_{Isentropic} = \frac{\dot{W}_{Actual}}{\dot{W}_{Ideal}}$

Condenser Heat Flow Loss Results		
Heat Loss	0.41	Taken from Heat Transfer Sheet
Condensate Flow Rate Assuming Adiabatic [lb/s]	0.358	$\dot{m}_{Steam} = \dot{m}_{Cooling} \frac{(h_7 - h_6)}{(h_3 - h_4)}$
Condensate Flow Rate Considering Heat Loss[lb/s]	0.358	$\dot{m}_{Steam} = \frac{\dot{m}_{Cooling}(h_7 - h_6) - \dot{Q}_{LOSS}}{(h_3 - h_4)}$

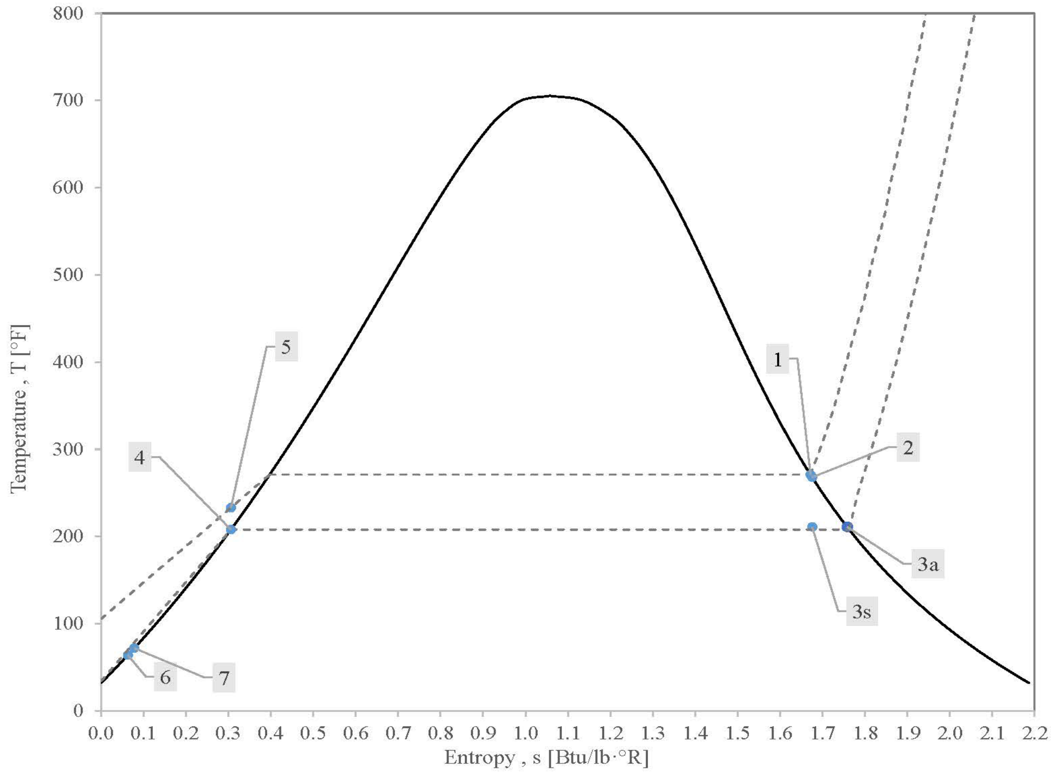

Figure 7.1: Temperature vs. Entropy Diagram for No-Load Test

Table 7.2: Software Input Table for Hose Test

Steam Turbine Rankine Cycle				
By: Nathan Ruiz				
Steam Tables from Fundamentals of Engineering Thermodynamics 7th edition				
Channel	Raw Value	Corrected Value	Units	Description
1	7.50E+01	75	°F	Ambient Air Temperature
2	2.79E+03	2790	RPM	RPM
3	6.34E+00	6	lb-ft	Torque
4	8.30E+01	83	°F	Dyno Temperature
5	7.50E+01	75	°F	Cooling Air In Temperature
6	8.50E+01	85	°F	Cooling Air Out Temperature
7	-3.00E-02			Cooling Air Flow Rate
8	2.11E+02	211	°F	Calorimeter Temperature
9	2.85E+02	285	°F	Steam Supply Temperature
10	3.52E+01	35	PSIG	Steam Supple Pressure
11	2.79E+02	279	°F	Nozzle Temperature
12	3.36E+01	34	PSIG	Nozzle Pressure
13	1.82E+02	182	°F	Exhaust Temperature
14	-5.36E+00	-5.360	PSIG	Exhaust Pressure
15	1.79E+02	179	°F	Condensate Temperature
16	-3.00E-03			Condensate Flow Rate
17	1.96E+02	196	°F	Condensor Shell Temperature
18	6.40E+01	64	°F	Cooling H_2O In Temperature
19	7.20E+01	72	°F	Cooling H_2O Out Temperature
20	4.36E+01	44	lb/s	Cooling H_2O Flow Rate

Table 7.3: Software Results for Hose Test

State	Temp.	Pres.	Quality	Specific Volume	Internal Energy	Enthalpy	Entropy
	T [°F]	P [lbf/in.^{2}]	x	v [ft^3/lb]	u [Btu/lb]	h [Btu/lb]	s [Btu/lb·°R]
1	285	49.9	1	8.0600	1096.78	1175.57	1.65419
2	279	48.3	1	8.7920	1095.60	1173.69	1.66182
3s	182	9.34	0.905	43.7743	981.73	1045.17	1.66182
3a	182	9.34	1	48.3600	1068.88	1138.86	1.80808
4	179	9.34	0	0.0165	146.97	146.99	0.26152
5	204	49.9	0				0.26152
6	64	14.7	0	0.0160	32.09	32.09	0.06323
7	72	14.7	0	0.0161	40.09	40.09	0.07839

Legend
Solved
Look Up
Given
Assumed

Turbine Power [HP]	3.368	HP = Torque[lb-ft] × RPM
State 3 Ideal Quality	0.905	$\frac{(s-s_f)}{(s_g-s_f)} = x$
Condensate Flow Rate [lb/s]	0.352	$\dot{m}_{Steam} = \dot{m}_{Cooling}\frac{(h_7-h_6)}{(h_3-h_4)}$
Actual Isentropic Efficiency	0.194	$\eta_{Isentropic} = \frac{\dot{W}_{Actual}}{\dot{W}_{Ideal}}$

Condenser Heat Flow Loss Results		
Heat Loss	0.42	Taken from Heat Transfer Sheet
Condensate Flow Rate Assuming Adiabatic [lb/s]	0.352	$\dot{m}_{Steam} = \dot{m}_{Cooling}\frac{(h_7-h_6)}{(h_3-h_4)}$
Condensate Flow Rate Considering Heat Loss[lb/s]	0.351	$\dot{m}_{Steam} = \frac{\dot{m}_{Cooling}(h_7-h_6) - \dot{Q}_{LOSS}}{(h_3-h_4)}$

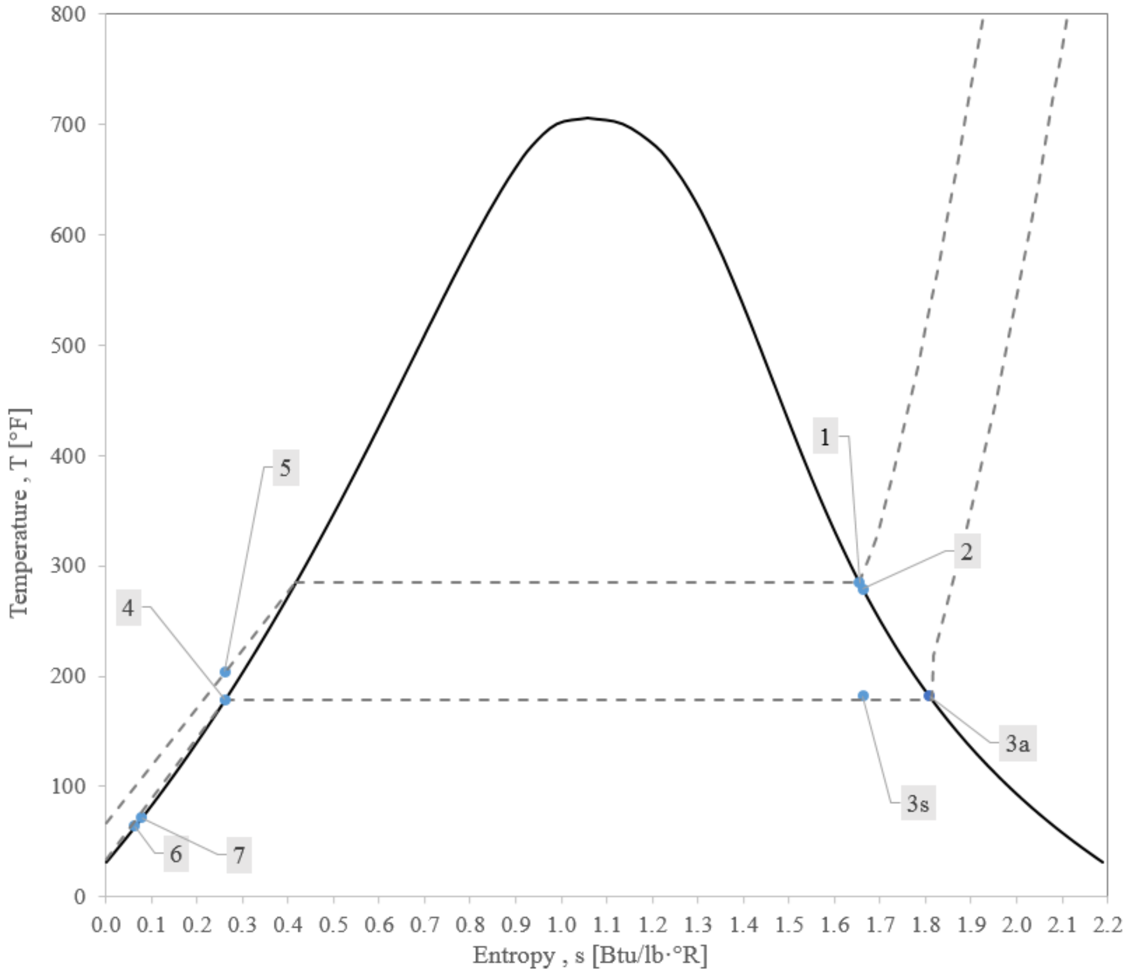

Figure 7.2: Temperature vs. Entropy Diagram for Hose Test

Table 7.4: Heat Transfer Analysis Results for Hose Test

Table A.4 Thermophysical Properties of Gases at Atmosperic Pressure (Excerpt)							
Taken from Fundamentals of Heat and Mass Transfer 3rd Ed.							
Air							
T[K]	ρ (kg/m$^{3)}$	c_p (kJ/kg·K)	$\mu\cdot10^7$ (N·s/m^2)	$\nu\cdot10^6$ (m^2/s)	$k\cdot10^3$ (W/m·K)	$\alpha\cdot10^6$ (m2/s)	Pr
100	3.5562	1.032	71.1	2.00	9.34	2.54	0.786
150	2.3364	1.012	103.4	4.43	13.80	5.84	0.758
200	1.7458	1.007	132.5	7.59	18.10	10.30	0.737
250	1.3947	1.006	159.6	11.44	22.30	15.90	0.720
300	1.1614	1.007	184.6	15.89	26.30	22.50	0.707
350	0.9950	1.009	208.2	20.92	30.00	29.90	0.700
400	0.8711	1.014	230.1	26.41	33.80	38.30	0.690
450	0.7740	1.021	250.1	32.39	37.30	47.20	0.686
change the range for interpolation manually							
330.9	1.0585	1.008	199.2	19.00	28.59	27.08	0.703
Multiplied by powers of ten to get actual values							
	1.0585	1.0082	1.99E-05	1.90E-05	0.029	2.71E-05	0.703

	°F	K
$T_{condenser}$	196	364.3
T_{room}	76	297.6

	K
T_f	330.9

	in.	m
Diameter	7	0.1778
Length	36	0.9144

	m^2
Surface Area	0.51

Emmisivity	0.85
Stefan-Boltzman Constant	5.67E-08

Rayliegh Number	2.16E+07	$Ra_D = \frac{g\beta(T_S - T_\infty)}{\nu\alpha} \quad 10^{-5} < Ra_D < 10^{12}$
Average Nusselt Number	35.5	$\overline{Nu}_D = \left\{0.60 + \frac{0.387 * Ra_D{}^{1/6}}{[1+(0.559/Pr)^{9/16}]^{8/27}}\right\}^2$
Heat Transfer Coefficient	5.70	$\bar{h} = \frac{k}{D}\overline{Nu}_D$
Heat Loss [W]	435.27	$\dot{Q}_{LOSS} = \bar{h}A(T_{Condenser} - T_{Room}) + \sigma\varepsilon A(T_{Condenser}{}^4 - T_{Room}{}^4)$
Heat Loss [Btu/s]	0.41	

Table 7.5: Hydrostatic Head Pressure Results for Hose Test

Variable	Description	Units	Value
L	Total Pipe Length	ft	114
D	Pipe Inner Diameter	in.	0.7
k	Pipe Absolute Roughness	mm	0.07
h	Water Level Height Differnece	ft	14
h_{Pump}	Total Dynamic Head	ft	7.5

Condensate Flow Rate [lb/s]	0.3517
Density [lb_m/ft^3]	59.848
Specific Weight [lb_f/ft^3]	62.43
Kinematic Viscosity [ft^2/s]	3.17E-06
Original Condenser Pressure [psia]	14.7

Velocity [ft/s]	2.199	$V=\frac{\dot{m}}{\rho A}$
Reynolds	40458	$Re_D=\frac{VD}{\nu}$
Friction Factor	0.031	$\frac{1}{\sqrt{f}}=-0.782\ln\left[\frac{6.9}{Re_D}+\left(\frac{k}{3.7D}\right)^{1.11}\right]$
Pressure Friction Loss [psi]	1.869	$\Delta P_{Friction}=L\frac{f}{D}\frac{\rho V^2}{2}$
Hydrostatic Pressure [psi]	6.070	$\Delta P_{Hydrostatic}=\gamma h$
Total Pump Head Pressure [psi]	3.252	$\Delta P_{Pump\ Head}=\gamma h_{Total\ Dynamic\ Head}$
Vacuum Pressure	7.248	$P_{Vacuum}=P_{Condenser}-\Delta P_{Hydrostatic}-\Delta P_{Pump\ Head}+\Delta P_{Friction}$

CHAPTER 8

CONCLUSIONS

A mathematical model of the effects of hydrostatic pressure head on the Rankine Power Cycle is complete. The software shows that a decrease in condenser pressure increases the work output of the turbine, decreases the load on the boiler, and increases the isentropic efficiency. Though the heat transfer analysis proved the condenser component adiabatic assumption the hydrostatic head pressure was not fully predicted due to complications in the piping. While the tests proved the proof of concept design can make the system run under a load further modifications can be made to better assist the system the greatest being to replace the condensate pump to add the much needed pressure head to the system directly and to install the plumbing for the hydrostatic pressure head configuration directly through the floor.

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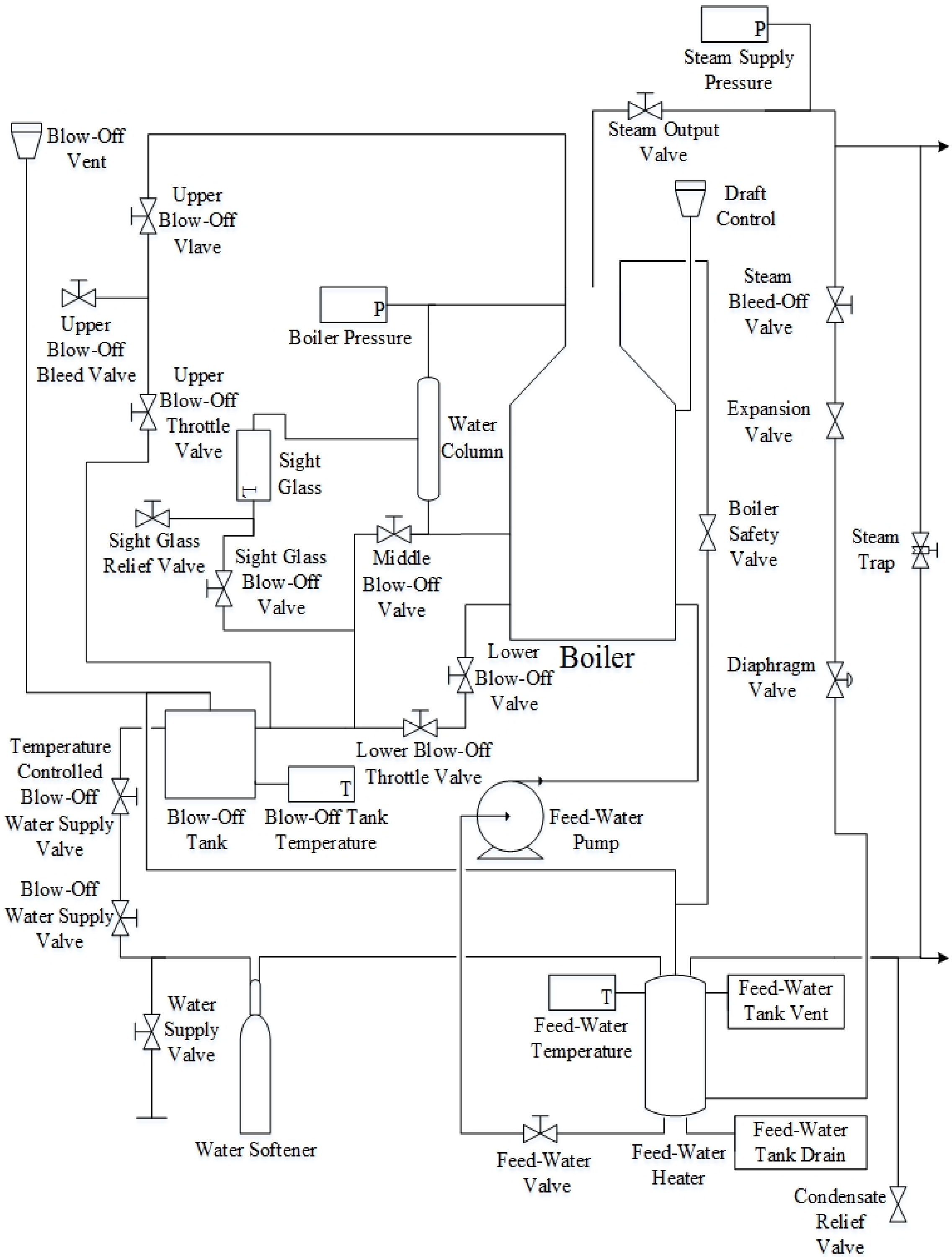
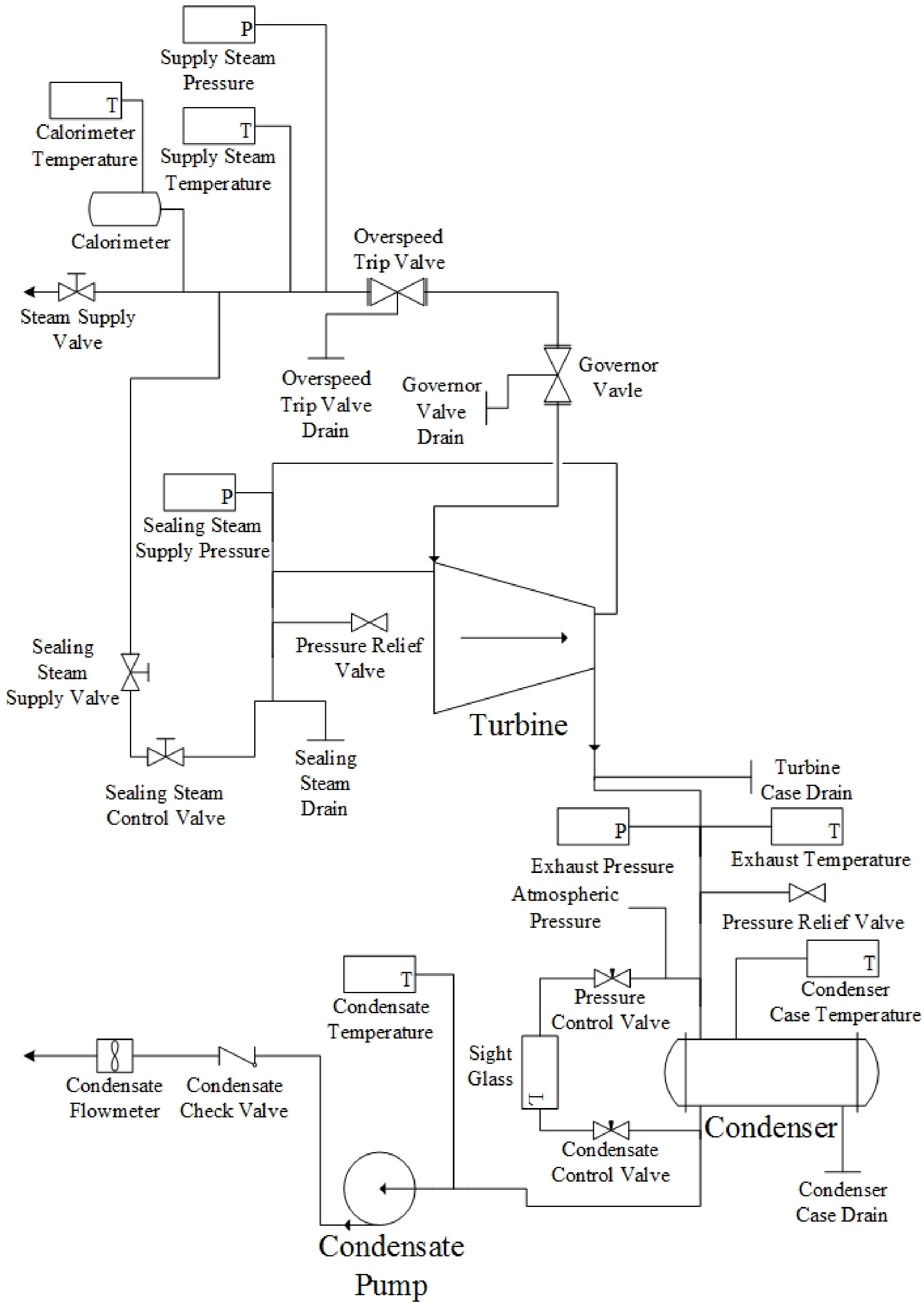
APPENDICES

APPENDIX A

SYSTEM SCHEMATIC




P
Supply Steam
Pressure
T
Calorimeter
Temperature
T
Supply Steam
Temperature
Calorimeter
Overspeed
Trip Valve
Steam Supply
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Trip Valve
Drain
Governor
Valve
Drain
Governor
Vavle
P
Sealing Steam
Supply Pressure
Pressure Relief
Valve
Sealing
Steam
Supply Valve
Turbine
Sealing Steam
Control Valve
Sealing
Steam
Drain
Turbine
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Atmospheric
Pressure
Pressure Relief Valve
T
Condenser
Case Temperature
T
Condensate
Temperature
Pressure
Control Valve
Sight
Glass
Condensate
Flowmeter
Condensate
Check Valve
Condenser
Condensate
Control Valve
Condenser
Case Drain
Condensate
Pump

APPENDIX B

SYSTEM COMPONENT MANUALS

TUTHILL CORPORATION / COPPUS TURBINE DIVISION
MILLBURY INDUSTRIAL PARK
P.O. BOX 8000
MILLBURY, MA 01527-8000

INSTRUCTION MANUAL NUMBER 98T4198

The Instruction manual that follows contains installation, operation, and maintenance instructions for the Coppus turbine identified below. It should be reviewed thoroughly before the turbine is installed and operated, and should be retained in a convenient location for a ready reference during operation and maintenance.

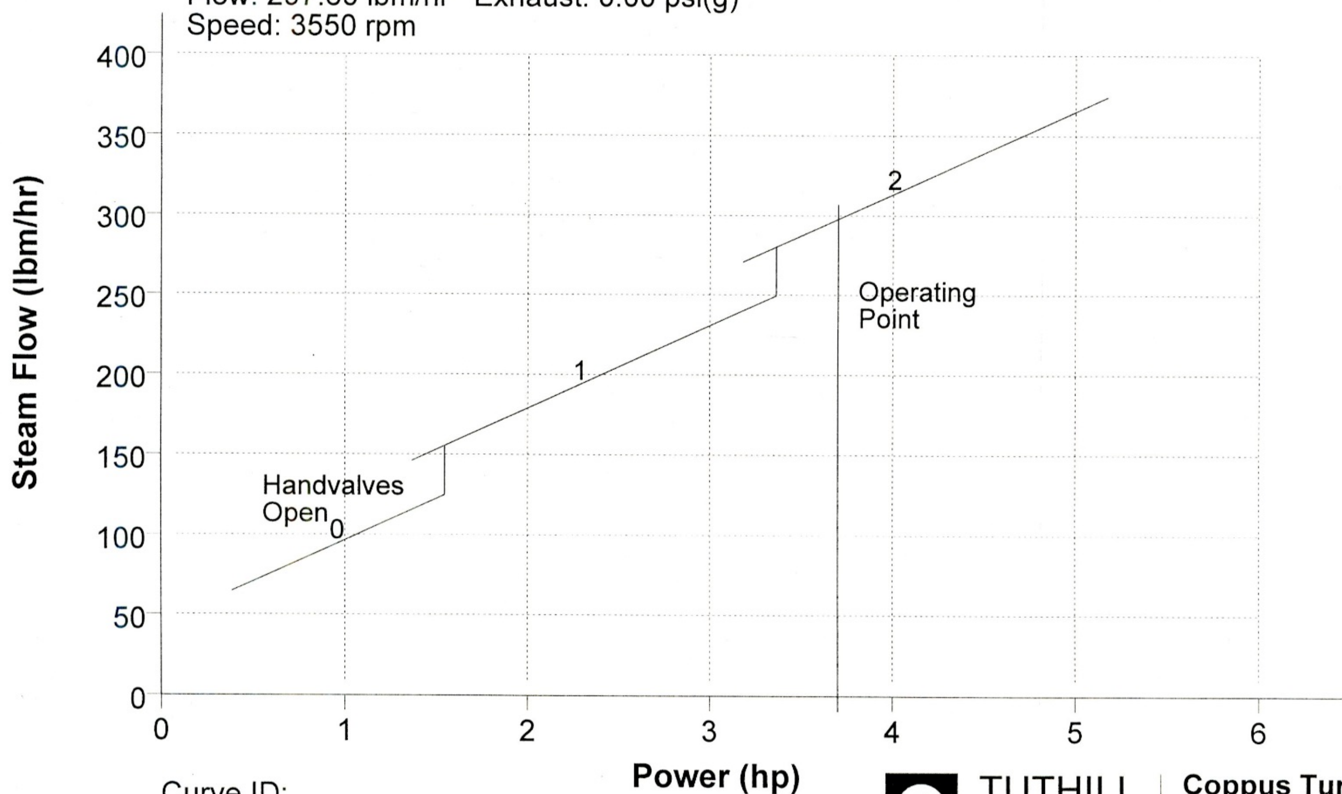
Turbine S/N	-	98T4198
Model & Frame Size	-	RLA 12M
HP and RPM	-	4.8 HP á 3550 RPM
Steam Conditions	-	INLET 100.0 PSI TEMPERATURE 338 Deg F EXHAUST PSI
Driven Equipment	-	
Purchaser	-	CALIFORNIA POLYTECHNIC STATE P807164
User Identification	-	CALIFORNIA POLYTECHNIC UNIV SAN LUIS OBISPO CA

9-418637-02 REV -

Turbine Performance Curve

Frame Size: RLA12M
No.: 98T4198
Power: 3.7 hp
Flow: 297.60 lbm/hr
Speed: 3550 rpm

Customer: California Polytechnic State U
Item / Tag No.:
Inlet: 100.0 psi(g) 338.0 °F
Exhaust: 0.00 psi(g)



Curve ID:
Date: 07-07-98
Engr: PJF
Version No. 1.00 03/31/98

TUTHILL CORPORATION® | Coppus Turbine Division

Millbury Industrial Park, P.O. Box 8000
Millbury, Massachusetts USA 01527-8000
Tel 508 756-8391 Fax 508 799-9531

TUTHILL/COPPUS NO LOAD TURBINE TEST REPORT 109481

1.0 Coppus Serial No. 98T4198 1.1 Turbine Size RLA 12M

2.0 Customer Order No. P807164 Test Time - Start 1230 Finish 1530

3.0 Test Conditions
- 3.1 Inlet Pressure 100 PSI — Gage # Used 1407
- 3.2 Exhaust Pressure 5 PSI — Gage # Used 1398
- 3.3 Inlet Temperature N/A °F or Saturated — Gage # Used ______ (if applicable)

4.0 Normal Speed 3550 RPM Tachometer Gage # Used 1977
4.1 No Load Speed Setting 3550 RPM

5.0 Overspeed Trip Setting Specified 4296 RPM
Tested Trip Speeds 5.1 4286 5.2 4293 5.3 4257

6.0 Governor Speed Control Tested/Accepted ok
6.1 Manual Trip Tested/Accepted ok

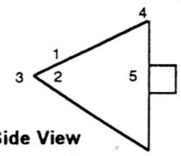
7.0/8.0 Vibration Level In/Sec., Velocity at Maximum Continuous Speed 3727 RPM
Vibration Meter/Gage # Used 1527

FILTERED

7.1 0.007 7.4 0.008
7.2 0.007 7.5 0.005
7.3 0.004

UNFILTERED

8.1 0.023 8.4 0.017
8.2 0.019 8.5 0.043
8.3 0.026



Side View

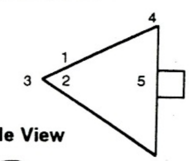
9.0/10.0 Vibration Level In/Sec., Velocity at approximately 100 RPM's Below Trip Speed 4196 RPM

FILTERED

9.1 0.021 9.4 0.023
9.2 0.038 9.5 0.032
9.3 0.006

UNFILTERED

10.1 0.033 10.4 0.031
10.2 0.043 10.5 0.050
10.3 0.021



Side View

11.0 Visual Inspection — Accept
11.1 Oil Leakage ok
11.2 Steam Leakage ok
11.3 Journal Bearings (RLHA only) N/A
11.4 Rotation (Facing Shaft Ext) (CW) (CCW)
11.5 Exhaust Location (Facing Shaft Ext) (Right) (Left) (Vertical)

12.0 Special Requirements
12.1 Sentinel Warning Valve Setting 5 PSI ok
12.2 Solenoid Trip - Test (3) Times N/A
12.3 Back Pressure Tested To 3 PSI ok Gage # Used 1398

12.4 Other ______

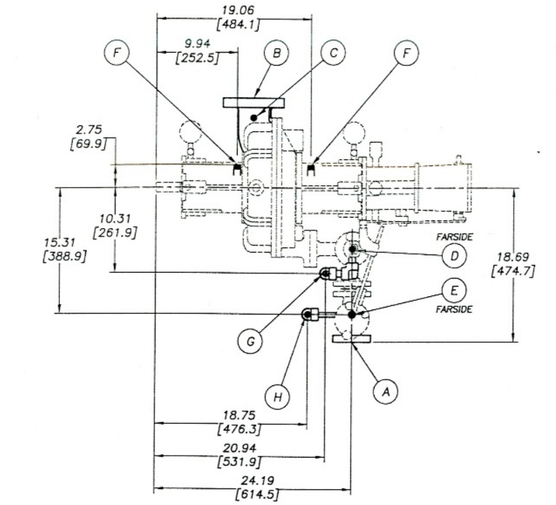
Inspected By Wayne Moore Date 10/21/98

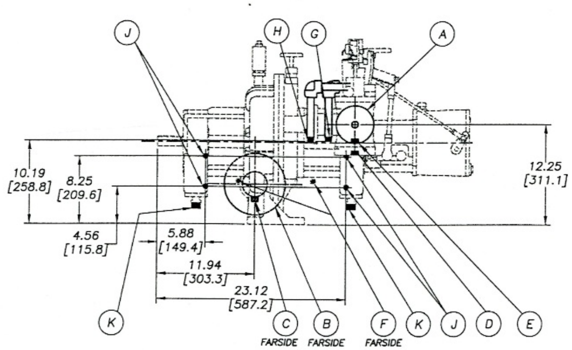
Quality Depart. Approval ______ Date ______

Witnessed By ______ Company ______ Date ______

COPIES: WHITE - QC, CANARY - ENGINEERING, GOLD - SALES

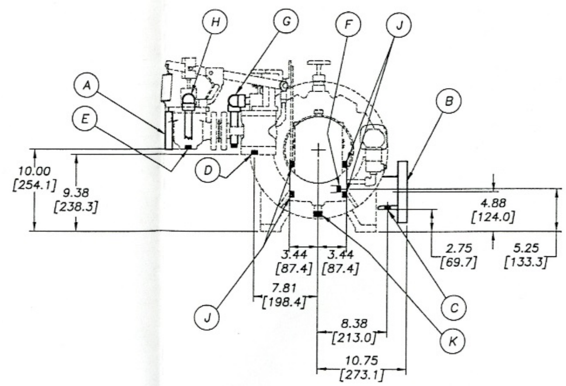
QC/SK2
REV. 5 10/94





PIPE CONNECTIONS			
CONN	QTY	SIZE	DESCRIPTION
A	1	0.75–300 FF	INLET, ANSI B16.5
B	1	3.00–125 FF	EXHAUST, ANSI B16.1
C	1	3/4 NPT F	DRAIN, EXHAUST CASING
D	1	3/4 NPT F	DRAIN, THROTTLE VALVE
E	1	3/4 NPT F	DRAIN, OVERSPEED TRIP VALVE
F	2	1/2 NPT M	LEAK–OFF, GLAND HOUSINGS
G	1	1/2 NPT M	LEAK–OFF, THROTTLE VALVE
H	1	1/2 NPT M	LEAK–OFF, OVERSPEED TRIP VALVE
J	8	3/8 NPT F	WATER COOLING CONN–SEE NOTE 3
K	2	1/2 NPT M	OIL DRAIN

NPT M = MALE THREAD NPT F = FEMALE THREAD



OCT 23 1998

REVISIONS			
REV	DESCRIPTION	DATE	BY

PURCHASER/USER INFORMATION

PURCHASER: CALIFORNIA POLYTECHNIC STATE UNIV

PURCHASE ORDER NUMBER: P807164

USER: CALIFORNIA POLYTECHNIC UNIV

LOCATION: SAN LUIS OBISPO, CA

USER EQUIPMENT IDENTIFICATION NUMBER: X

DRIVEN EQUIPMENT: X

CONSTRUCTION

INLET 200 CAST IRON

EXHAUST 200 CAST IRON

DIMENSIONS AND TOLERANCES

- DIMENSIONS ARE GIVEN IN INCHES AND [MM].
- DIMENSIONED FEATURES ARE TO SCALE, EXCEPT AS NOTED. UNDIMENSIONED FEATURES ARE FOR REFERENCE ONLY AND SHOULD NOT BE SCALED.
- OVERALL CENTER OF GRAVITY, AND CONNECTION LOCATION TOLERANCES ±0.50 [12.7]
- MOUNTING HOLE LOCATION AND SHAFT HEIGHT TOLERANCES ±0.06 [1.5]
- SHAFT AND KEY SEAT DIMENSIONS AND TOLERANCES ARE PER ANSI B17.1

NOTES

1. LEAK–OFF CONNECTIONS F, G, & H MUST BE PIPED TO OPEN AND UNRESTRICTED NON–MANIFOLDING DRAIN LINES WHICH DISCHARGE INTO A SAFE VISIBLE AREA.
2. DRAIN CONNECTIONS C, D, & E. EACH DRAIN MUST BE VALVED SEPARTELY AND PIPED TO NON–MANIFOLDING DRAIN LINES WHICH DISCHARGE INTO A SAFE VISIBLE AREA
3. WATER COOLING REQUIRED ABOVE 550°F [288°C] INLET TEMPERATURE. CONNECT INLET WATER TO A LOWER HOLE IN EACH BEARING HOUSING AND OUTLET WATER FROM THE UPPER HOLE IN THE OPPOSITE SIDE OF EACH BEARING HOUSING. THE UNUSED HOLES ARE TO REMAIN PLUGGED. SEE INSTRUCTION MANUAL FOR FURTHER INFORMATION.
4. ALL THREADED CONNECTIONS ARE PLUGGED PRIOR TO SHIPMENT.

TUTHILL ENERGY SYSTEMS | Coppus Turbine Millbury Massachusetts USA

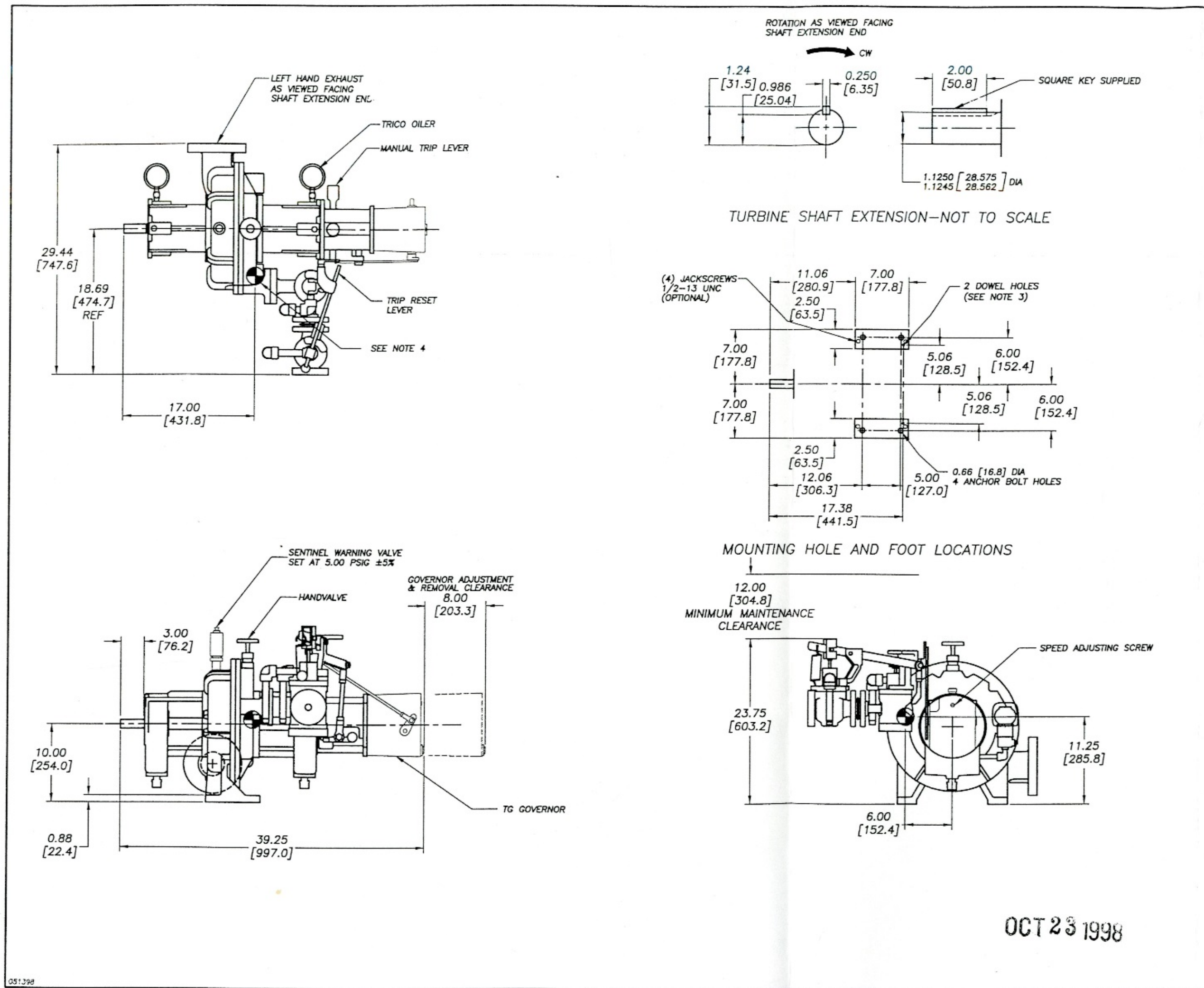
RLA12M INLET, EXHAUST, DRAIN, LEAK–OFF AND BEARING HOUSING CONNECTIONS

DRAWING NUMBER

SERIAL NUMBER 98T4198 | –02

DRAWN BY PJF DATE 07–07–98

CHECKED BY TEP DATE 07–08–98



REVISIONS

REV	DESCRIPTION	DATE	BY

PURCHASER/USER INFORMATION

PURCHASER	CALIFORNIA POLYTECHNIC STATE UNIV.
PURCHASE ORDER NUMBER	P807164
USER	CALIFORNIA POLYTECHNIC UNIV.
LOCATION	SAN LUIS OBISPO, CA
USER EQUIPMENT IDENTIFICATION NUMBER	X
DRIVEN EQUIPMENT	

CONSTRUCTION

INLET 200 CAST IRON
EXHAUST 200 CAST IRON

DIMENSIONS AND TOLERANCES

- DIMENSIONS ARE GIVEN IN INCHES AND [MM].
- DIMENSIONED FEATURES ARE TO SCALE, EXCEPT AS NOTED. UNDIMENSIONED FEATURES ARE FOR REFERENCE ONLY AND SHOULD NOT BE SCALED.
- OVERALL, CENTER OF GRAVITY, AND CONNECTION LOCATION TOLERANCES ±0.50 [12.7]
- MOUNTING HOLE LOCATION AND SHAFT HEIGHT TOLERANCES ±0.06 [1.5]
- SHAFT AND KEY SEAT DIMENSIONS AND TOLERANCES ARE PER ANSI B17.1

NOTES

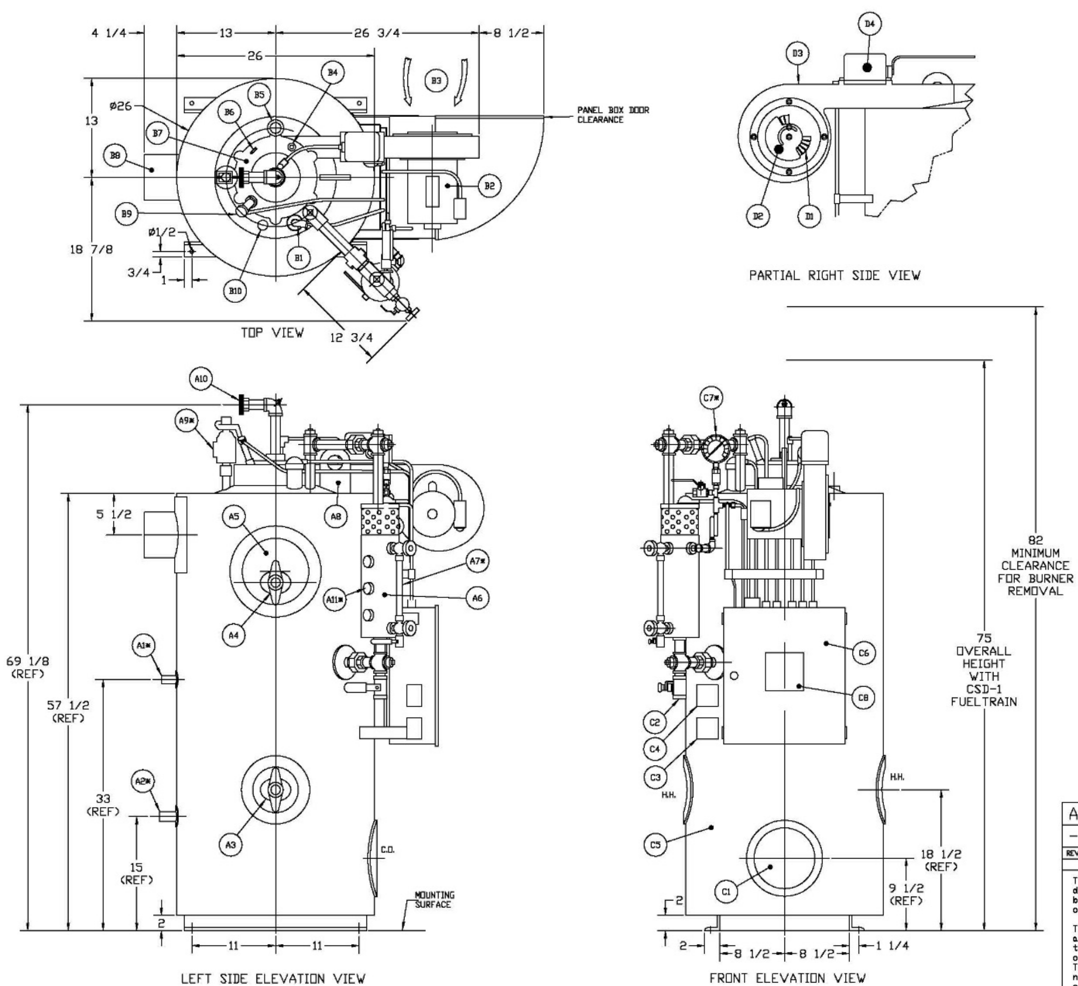
1. TURBINE INSTALLATION, OPERATION, REPAIR AND SERVICE MUST BE PERFORMED BY EXPERIENCED PERSONNEL ONLY. READ TURBINE INSTRUCTION MANUAL BEFORE INSTALLING, OPERATING, REPAIRING OR SERVICING.
2. THE TURBINE MUST BE MOUNTED ON A SOLID FOUNDATION OR BASEPLATE.
3. THE DOWEL HOLES ARE PILOT DRILLED TO .250 INCH [6.4MM] DIA AND ARE TO BE REAMED FOR #6 TAPER PIN AFTER HOT ALIGNMENT IS COMPLETE.
4. TARGET SYMBOL INDICATES THE TURBINE CENTER OF GRAVITY.

TUTHILL ENERGY SYSTEMS | Coppus Turbine Millbury Massachusetts USA

RLA12M TURBINE
BASIC DIMENSIONS

DRAWING NUMBER
SERIAL NUMBER 98T4198 —01

DRAWN BY PJF DATE 07-07-98
CHECKED BY TEP DATE 07-08-98



NOTES:
1. STEAM OUTPUT LB/HR : FROM 0 PSIG AND AT 212°F (0 KG/CM AT 100°C)
2. DUE TO POWER RATINGS, COMPONENT SIZES, TYPES, QUANTITIES, ETC., PANEL BOX SIZE MAY VARY FROM WHAT IS SHOWN. ALL PRECUT OPENINGS IN PANEL BOX THAT ARE UNUSED, WILL BE CAPPED.
3. WE RESERVE THE RIGHT TO CHANGE SPECIFICATIONS AND/OR DIMENSIONS.
4. OPERABLE FROM 75-125 PSIG.

A - LEFT SIDE ELEVATION VIEW
- A1 - 3/4" FEED WATER SUPPLY INLET (1EA CHECK & BALL VALVE SUPPLIED)
- A2 - 1" BLOWOFF OPENING ("Y" & BALL VALVE SUPPLIED)
- A3 - LOWER 3" x 4" HANDHOLE AND COVER ASSEMBLY (TYP 2 PL)
- A4 - UPPER 3" x 4" HANDHOLE AND COVER ASSEMBLY
- A5 - ASME CODE P.V. STAMPING AREA
- A6 - WATER COLUMN
- A7 - SIGHTGLASS ASSEMBLY WITH DRAIN
- A8 - LIFTING EYE
- A9 - SAFETY VALVE NOMINAL SIZE OF 3/4" INLET WITH 1" OUTLET
- A10- 1" GAS MANIFOLD ASSEMBLY CONNECTION (GASTRAIN NOT SHOWN)
- A11- TRICOCKS (OPTIONAL)

B - TOP VIEW
- B1 - 1/2" LOW WATER PROBE
- B2 - BURNER MOTOR
- B3 - AIR INTAKE (COMBUSTION)
- B4 - FLAME VIEW PORT
- B5 - 3/4" STEAM OUTLET. (1EA GATE VALVE SUPPLIED)
- B6 - AIR DAMPER HANDLE (COMBUSTION)
- B7 - TOP PLATE
- B8 - 6" DIA. MALE EXHAUST STACK OPNG STD
- B9 - UV SCANNER
- B10- 1" SURFACE BLOWDOWN OUTLET (1EA. BALL VALVE SUPPLIED)

C - FRONT ELEVATION VIEW
- C1 - 6" DIA. CLEANOUT OPENING WITH 10" DIA. COVER PLATE
- C2 - 1" WATER BOTTLE DRAIN (1EA. BALL VALVE SUPPLIED)
- C3 - TO OPERATING PRESSURE CONTROL
- C4 - TO HIGH LIMIT PRESSURE CONTROL
- C5 - STEEL JACKET
- C6 - CONTROL PANEL BOX (SEE NOTE 2)
- C7 - STEAM GAUGE ASSEMBLY WITH 1/4" BALL VALVE TEST/VENT PORT
- C8 - WINDOW KIT

D - PARTIAL RIGHT SIDE VIEW
- D1 - DAMPER PLATE
- D2 - AIR GATE PLATE
- D3 - BLOWER SCROLL
- D4 - IGNITION TRANSFORMER

DESCRIPTION
FUEL: GAS
ASME DESIGN PRESSURE: 150 PSIG (SEE NOTE 4)
BOILER INPUT: 257,000 BTU / HR
BOILER OUTPUT: 201,000 BTU / HR
STEAM OUTPUT & MIN. SRV CAP. : 207 LB/HR (SEE NOTE 1)
NET EFFECTIVE HEATING SURFACE : 18 SQ FT
WATER CAPACITY (OPERATING) : 16 GAL (U.S.)
APPROXIMATE SHIPPING WEIGHT (DRY): 1700 LBS
APPROXIMATE OPERATING WEIGHT : 1833 LBS
BURNER MOTOR HP: 1/3 H.P.

* COMPONENT(S) SHIPPED LOOSE WITH BOILER

REV.	DESCRIPTION	CHECKED:	APPROVAL:
A	CORRECTED MISC. L.D. 2011	L.D. 16 NOV 2011	L.D. 16 NOV 2011
—	INITIAL RELEASE	C.M. 11 DEC 2003	L.D. 11 DEC 2003

REVISIONS

The item shown in this drawing may be covered by one or more patents of The Fulton Companies.

This design and drawing are proprietary and are the exclusive property of The Fulton Companies. The corporation does not permit their use except with prior written consent.

DWN BY: B. SHERIDAN — DATE: 11 DEC 2003
SCALE: 1 / 6 — SHEET NO. 1 OF 1

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCE ON:
FRACTIONS ± 1/4
TWO PLACE DECIMALS ± .01
THREE PLACE DECIMALS ± .005
ANGLES ± 2 DEG.
SURFACE FINISH 250 MICROINCHES

Fulton Thermal Corp.
972 Centerville Road
Pulaski, New York, USA 13142

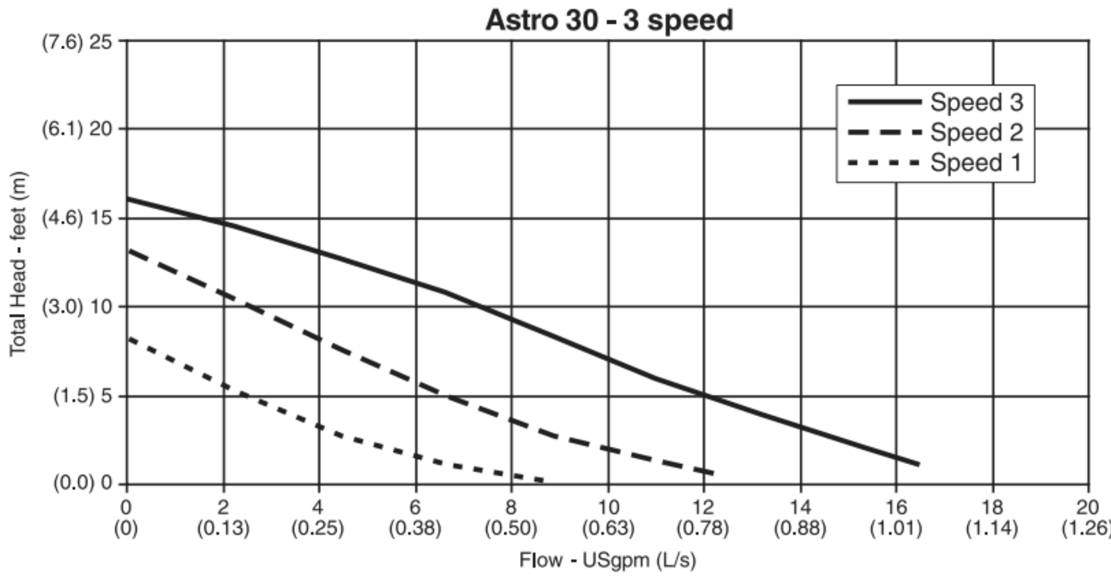
Fulton

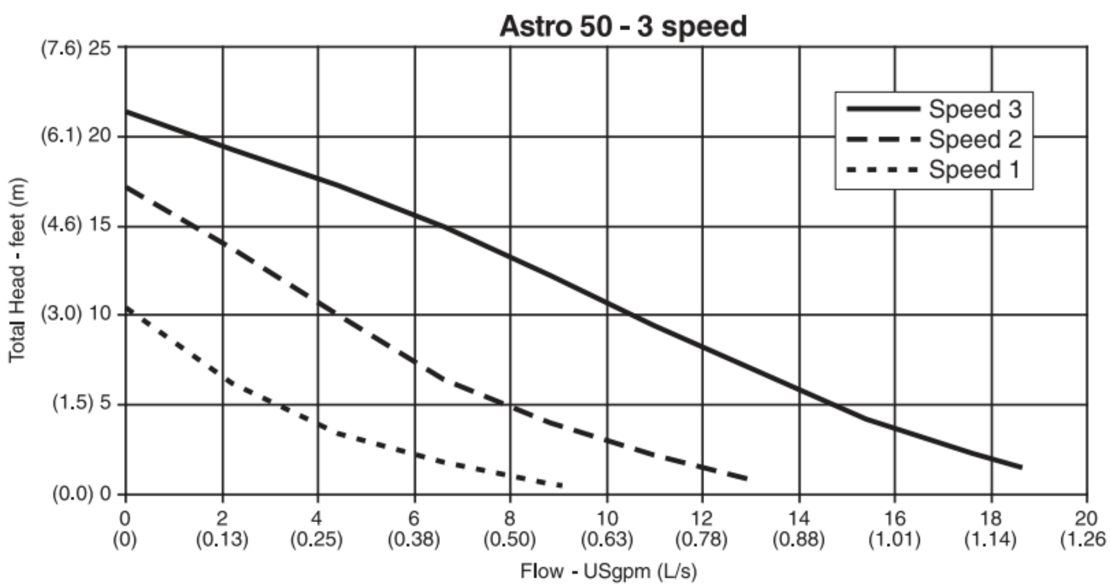
CLASSIC ICS 6 H.P.
GAS FIRED STEAM BOILER
(150 PSIG DESIGN)

SHEET SIZE: D — DWG NO. B91DR-1950 — REV. A

APPENDIX C

PUMP CURVES





APPENDIX D

VBA SOFTWARE CODE

```
'**Steam Turbine Rankine Cycle**'
'**Nathan Ruiz**'
'**Function to Identify State Properties off Steam Tables**'
Function vStateProp(Temp As Double, Pres As Double, Qual As Double)
'Function identifies specific volume'
'Notes: Some rows on Table A.5E were removed for coding simplicity'
    'while some rows were added to Table A.4E for coding simplicity;'
    'rows removed only effect accuracy of answers when interpolation
    'used but difference was deemed negligible'
'Notes: When dealing with a Liquid-Vapor the Temperature Table A.2E'
    'was used instead of the Pressure Table A.3E due to having more'
    'data points therefore increasing efficiency'
'Notes: Table A.6E, Properties of Saturated Water (Solid-Vapor):
    'Temperature Table, was not included. Integration with'
    'this code can be done easily if need arises'

'Define integer used to read specefic volume on the tables'
'This term will need to be changed for each property'
Dim a As Integer, af As Integer, ag As Integer
'From Table A.4E and A.5E'
a = 4
'From Table A.2E'
af = 3
ag = 4

'Assign all of the constants used during interpolation as doubles'
't = temperature, p = pressure, v = variable being measured'
'vf = prop. at saturated liquid, vg = prop. at dry saturated vapor'
Dim T1 As Double, T2 As Double, P1 As Double, P2 As Double
Dim v1 As Double, v2 As Double, vf As Double, vg As Double
Dim t11 As Double, t22 As Double, v11 As Double, v22 As Double
Dim vtemp1 As Double, vtemp2 As Double
Dim vf1 As Double, vf2 As Double, vg1 As Double, vg2 As Double

'Check to see if water is a superheated vapor'
If Qual = 100 Then
    'Checks if pressure is too low for the table'
    If Pres < 1 Then
    vStateProp = "Pres Low"
    Exit Function
    End If
    'Checks if pressure is too high for the table'
    If Pres > 4800 Then
```

```
vStateProp = "Pres High"
Exit Function
End If
'Index counter'
I = 6
For n = 1 To 36
   'Pressure index value'
   k = Sheets("Superheated Water Vapor").Cells(I, 1)
   'Finds lower bound pressure'
   If Pres > k Then
      I = I + 12
   'Solves for the case where pressure is an exact table value'
   ElseIf Pres = k Then
         'Checks if temperature is too low for the table'
         tmin = Sheets("Superheated Water Vapor").Cells(I, 3)
         If Temp < tmin Then
            vStateProp = "Temp Low"
            Exit Function
         End If
         'Checks if temperature is too high for the table'
         tmax = Sheets("Superheated Water Vapor").Cells(I + 11, 3)
         If Temp > tmax Then
            vStateProp = "Temp High"
            Exit Function
         End If
         For m = 1 To 12
         q = Sheets("Superheated Water Vapor").Cells(I, 3)
         'Finds lower bound temperature'
         If Temp > q Then
            I = I + 1
         'Solves for the case where temperature is an exact table value'
         ElseIf Temp = q Then
            vStateProp = Sheets("Superheated Water Vapor").Cells(I, a)
            Exit Function
         'Solves using interpolation'
         ElseIf Temp < q Then
            T1 = Sheets("Superheated Water Vapor").Cells(I - 1, 3)
            T2 = Sheets("Superheated Water Vapor").Cells(I, 3)
            v1 = Sheets("Superheated Water Vapor").Cells(I - 1, a)
            v2 = Sheets("Superheated Water Vapor").Cells(I, a)
            vStateProp = INTERPOLATE(T1, T2, Temp, v1, v2)
            Exit Function
         End If
         Next m
      Exit For
   'Solves using interpolation'
```

```
ElseIf Pres < k Then
    'Checks if temperature is too low for the table'
    tmin = Sheets("Superheated Water Vapor").Cells(I - 12, 3)
    If Temp < tmin Then
      vStateProp = "Temp Low"
      Exit Function
    End If
    'Checks if temperature is too high for the table'
    tmax = Sheets("Superheated Water Vapor").Cells(I + 11, 3)
    If Temp > tmax Then
      vStateProp = "Temp High"
      Exit Function
    End If
    'Checks if temperature is between saturation temperatures'
    tmin1 = Sheets("Superheated Water Vapor").Cells(I - 12, 3)
    tmin2 = Sheets("Superheated Water Vapor").Cells(I, 3)
    If Temp < tmin2 And Temp > tmin1 Then
      T1 = tmin1
      T2 = tmin2
      v1 = Sheets("Superheated Water Vapor").Cells(I - 12, a)
      v2 = Sheets("Superheated Water Vapor").Cells(I, a)
      vStateProp = INTERPOLATE(T1, T2, Temp, v1, v2)
      Exit Function
    End If
    OldPres = Sheets("Superheated Water Vapor").Cells(I - 12, 1)
    For m = 1 To 12
    q = Sheets("Superheated Water Vapor").Cells(I, 3)
    'Finds lower bound temperature'
    If Temp > q Then
      I = I + 1
    'Solves for the case where temperature is an exact table value'
    'Still uses interpolation between the pressures'
    ElseIf Temp = q Then
      P1 = OldPres
      P2 = k
      v1 = Sheets("Superheated Water Vapor").Cells(I - 12, a)
      v2 = Sheets("Superheated Water Vapor").Cells(I, a)
      vStateProp = INTERPOLATE(P1, P2, Pres, v1, v2)
      Exit Function
    'Solves for the case where temperature is also not on the table'
    'Uses double interpolation between the pressures and temperatures'
    ElseIf Temp < q Then
      P1 = OldPres
      P2 = k
      T1 = Sheets("Superheated Water Vapor").Cells(I - 1, 3)
      T2 = Sheets("Superheated Water Vapor").Cells(I, 3)
```

```
                v1 = Sheets("Superheated Water Vapor").Cells(I - 1, a)
                v2 = Sheets("Superheated Water Vapor").Cells(I, a)
                vtemp2 = INTERPOLATE(T1, T2, Temp, v1, v2)
                t11 = Sheets("Superheated Water Vapor").Cells(I - 13, 3)
                t22 = Sheets("Superheated Water Vapor").Cells(I - 12, 3)
                v11 = Sheets("Superheated Water Vapor").Cells(I - 13, a)
                v22 = Sheets("Superheated Water Vapor").Cells(I - 12, a)
                vtemp1 = INTERPOLATE(t11, t22, Temp, v11, v22)
                vStateProp = INTERPOLATE(P1, P2, Pres, vtemp1, vtemp2)
                Exit Function
            End If
            Next m
        Exit For
      End If
   Next n
End If

'Check to see if water is a liquid-vapor mixture'
If Qual < 100 And Qual > 0 Then
   'Checks if pressure is too low for the table'
   If Pres < 0.0886 Then
   vStateProp = "Pres Low"
   Exit Function
   End If
   'Checks if pressure is too high for the table'
   If Pres > 3204 Then
   vStateProp = "Pres High"
   Exit Function
   End If
   'Checks if temperature is too low for the table'
   If Temp < 32 Then
   vStateProp = "Temp Low"
   Exit Function
   End If
   'Checks if temperature is too high for the table'
   If Temp > 705.4 Then
   vStateProp = "Temp High"
   Exit Function
   End If
   'Index counter'
   I = 7
   For n = 1 To 81
      'Temperature index value'
      k = Sheets("Liquid Vapor Temperature").Cells(I, 1)
      'Finds lower bound temperature'
      If Temp > k Then
```

```
            I = I + 1
        'Solves for the case where temperature is an exact table value'
        ElseIf Temp = k Then
            'Checks that pressure is within 10% of saturation values'
            q = Sheets("Liquid Vapor Temperature").Cells(I, 2)
            q1 = q - (0.1 * q)
            q2 = q + (0.1 * q)
            If Pres < q1 Or Pres > q2 Then
            vStateProp = "Invalid Pres"
            Exit Function
            End If
            'Solves for value using two-phase system equation'
            vf = Sheets("Liquid Vapor Temperature").Cells(I, af)
            vg = Sheets("Liquid Vapor Temperature").Cells(I, ag)
            vStateProp = vf + (Qual / 100) * (vg - vf)
            Exit Function
        'Solves using interpolation'
        ElseIf Temp < k Then
            'Checks that pressure is within 5% of both bounds'
            r1 = Sheets("Liquid Vapor Temperature").Cells(I - 1, 2)
            r2 = Sheets("Liquid Vapor Temperature").Cells(I, 2)
            q1 = r1 - (0.05 * r1)
            q2 = r2 + (0.05 * r2)
            If Pres < q1 Or Pres > q2 Then
            vStateProp = "Invalid Pres"
            Exit Function
            End If
            'Solves for value using interpolation and two-phase system equation'
            T1 = Sheets("Liquid Vapor Temperature").Cells(I - 1, 1)
            T2 = k
            vf1 = Sheets("Liquid Vapor Temperature").Cells(I - 1, af)
            vg1 = Sheets("Liquid Vapor Temperature").Cells(I - 1, ag)
            vf2 = Sheets("Liquid Vapor Temperature").Cells(I, af)
            vg2 = Sheets("Liquid Vapor Temperature").Cells(I, ag)
            vtemp1 = vf1 + (Qual / 100) * (vg1 - vf1)
            vtemp2 = vf2 + (Qual / 100) * (vg2 - vf2)
            vStateProp = INTERPOLATE(T1, T2, Temp, vtemp1, vtemp2)
            Exit Function
        End If
    Next n
End If

'Check to see if water is a liquid or compressed liquid'
If Qual = 0 Then
    'Checks if pressure is too low for the table'
    If Pres < 0 Then
```

```
vStateProp = "Pres Low"
Exit Function
End If
'Checks if pressure is too high for the table'
If Pres > 4000 Then
vStateProp = "Pres High"
Exit Function
End If
'Checks if temperature is too low for the table'
If Temp < 32 Then
vStateProp = "Temp High"
Exit Function
End If
'Checks if temperature is too low for the table'
If Temp > 705.4 Then
vStateProp = "Temp High"
Exit Function
End If
'Check if water is a compressed liquid'
If Pres >= 500 And Temp < 467.1 Then
F = 6
For s = 1 To 8
   'Pressure index value'
   k = Sheets("Compressed Liquid Water").Cells(F, 1)
   'Finds lower bound pressure'
   If Pres > k Then
      F = F + 8
   'Solves for the case where pressure is an exact table value'
   ElseIf Pres = k Then
      For j = 1 To 8
         q = Sheets("Compressed Liquid Water").Cells(F, 3)
         'Finds lower bound temperature'
         If Temp > q Then
            F = F + 1
         'Solves for the case where temperature is an exact table value'
         ElseIf Temp = q Then
            vStateProp = Sheets("Compressed Liquid Water").Cells(F, a)
            Exit Function
         'Solves using interpolation'
         ElseIf Temp < q Then
            T1 = Sheets("Compressed Liquid Water").Cells(F - 1, 3)
            T2 = Sheets("Compressed Liquid Water").Cells(F, 3)
            v1 = Sheets("Compressed Liquid Water").Cells(F - 1, a)
            v2 = Sheets("Compressed Liquid Water").Cells(F, a)
            vStateProp = INTERPOLATE(T1, T2, Temp, v1, v2)
            Exit Function
```

```
        End If
      Next j
    'Solves by rounding to the nearest value'
    ElseIf Pres < k Then
        OldPres = Sheets("Compressed Liquid Water").Cells(F - 8, 1)
        Pavg = (OldPres + k) / 2
        'If pressure is closer to lower value use that'
        If Pres <= Pavg Then
          For j = 1 To 8
          q = Sheets("Compressed Liquid Water").Cells(F - 8, 3)
          'Finds lower bound temperature'
          If Temp > q Then
            F = F + 1
          'Solves for the case where temperature is an exact table value'
          ElseIf Temp = q Then
            vStateProp = Sheets("Compressed Liquid Water").Cells(F - 8, a)
            Exit Function
          'Solves using interpolation'
          ElseIf Temp < q Then
            T1 = Sheets("Compressed Liquid Water").Cells(F - 9, 3)
            T2 = Sheets("Compressed Liquid Water").Cells(F - 8, 3)
            v1 = Sheets("Compressed Liquid Water").Cells(F - 9, a)
            v2 = Sheets("Compressed Liquid Water").Cells(F - 8, a)
            vStateProp = INTERPOLATE(T1, T2, Temp, v1, v2)
            Exit Function
          End If
          Next j
        End If
        'If pressure is closer to higher value use that'
        If Pres > Pavg Then
          For j = 1 To 8
          q = Sheets("Compressed Liquid Water").Cells(I, 3)
          'Finds lower bound temperature'
            If Temp > q Then
              F = F + 1
            'Solves for the case where temperature is an exact table value'
            ElseIf Temp = q Then
              vStateProp = Sheets("Compressed Liquid Water").Cells(F, a)
              Exit Function
            'Solves using interpolation'
            ElseIf Temp < q Then
              T1 = Sheets("Compressed Liquid Water").Cells(F - 1, 3)
              T2 = Sheets("Compressed Liquid Water").Cells(F, 3)
              v1 = Sheets("Compressed Liquid Water").Cells(F - 1, a)
              v2 = Sheets("Compressed Liquid Water").Cells(F, a)
              vStateProp = INTERPOLATE(T1, T2, Temp, v1, v2)
```

```
                Exit Function
              End If
            Next j
          End If
      End If
    Next s
    'Otherwise assume water is not a compressed liquid'
    'Index counter'
    Else
      I = 7
      For n = 1 To 81
      'Temperature index value'
      k = Sheets("Liquid Vapor Temperature").Cells(I, 1)
      'Finds lower bound temperature'
      If Temp > k Then
        I = I + 1
      'Solves for the case where temperature is an exact table value'
      ElseIf Temp = k Then
        'Solves for value using only property at saturated liquid value'
        vf = Sheets("Liquid Vapor Temperature").Cells(I, af)
        vStateProp = vf
        Exit Function
      'Solves using interpolation'
      ElseIf Temp < k Then
        'Solves for value using interpolation and saturated liquid value'
        T1 = Sheets("Liquid Vapor Temperature").Cells(I - 1, 1)
        T2 = k
        vf1 = Sheets("Liquid Vapor Temperature").Cells(I - 1, af)
        vf2 = Sheets("Liquid Vapor Temperature").Cells(I, af)
        vStateProp = INTERPOLATE(T1, T2, Temp, vf1, vf2)
        Exit Function
      End If
      Next n
    End If
End If
End Function

'****Steam Turbine Rankine Cycle*****'
'****Nathan Ruiz****'
'****Function to Iterpolate Data***'
Function INTERPOLATE(xLow As Double, xHigh As Double, xMid As Double, yLow
As Double, yHigh As Double)
'Solves for the unknown value by interpolation, unknown value would be yMid'
INTERPOLATE = (((yHigh - yLow) * (xMid - xLow)) / (xHigh - xLow)) + yLow
End Function
```

APPENDIX E

STEAM TABLE SAMPLE

Table E.1: Properties of Saturated Water Liquid-Vapor Sample

Properties of Saturated Water (Liquid-Vapor): Temperature Table										
Table A.2E										
Temperature	Pressure	Specific Volume		Internal Energy		Enthalpy			Entropy	
		ft^3/lb		Btu/lb		Btu/lb			Btu/lb · °R	
		Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Sat. Vapor
°F	$lbf/in.^2$	v_f	v_g	u_f	u_g	h_f	h_{fg}	h_g	s_f	s_g
32	0.0886	0.01602	3305	-0.01	1021.2	-0.01	1075.4	1075.4	-0.00003	2.1870
35	0.0999	0.01602	2948	2.99	1022.2	3.00	1073.7	1076.7	0.00607	2.1764
40	0.1217	0.01602	2445	8.02	1023.9	8.02	1070.9	1078.9	0.01617	2.1592
45	0.1475	0.01602	2037	13.04	1025.5	13.04	1068.1	1081.1	0.02618	2.1423
50	0.1780	0.01602	1704	18.06	1027.2	18.06	1065.2	1083.3	0.03607	2.1259
52	0.1917	0.01603	1589	20.06	1027.8	20.07	1064.1	1084.2	0.04000	2.1195
54	0.2064	0.01603	1482	22.07	1028.5	22.07	1063.0	1085.1	0.04391	2.1131
56	0.2219	0.01603	1383	24.08	1029.1	24.08	1061.9	1085.9	0.04781	2.1068
58	0.2386	0.01603	1292	26.08	1029.8	26.08	1060.7	1086.8	0.05159	2.1005
60	0.2563	0.01604	1207	28.08	1030.4	28.08	1059.6	1087.7	0.05555	2.0943
62	0.2751	0.01604	1129	30.09	1031.1	30.09	1058.5	1088.6	0.05940	2.0882
64	0.2952	0.01604	1056	32.09	1031.8	32.09	1057.3	1089.4	0.06323	2.0821
66	0.3165	0.01604	988.4	34.09	1032.4	34.09	1056.2	1090.3	0.06704	2.0761
68	0.3391	0.01605	925.8	36.09	1033.1	36.09	1055.1	1091.2	0.07084	2.0701

APPENDIX F

LabVIEW

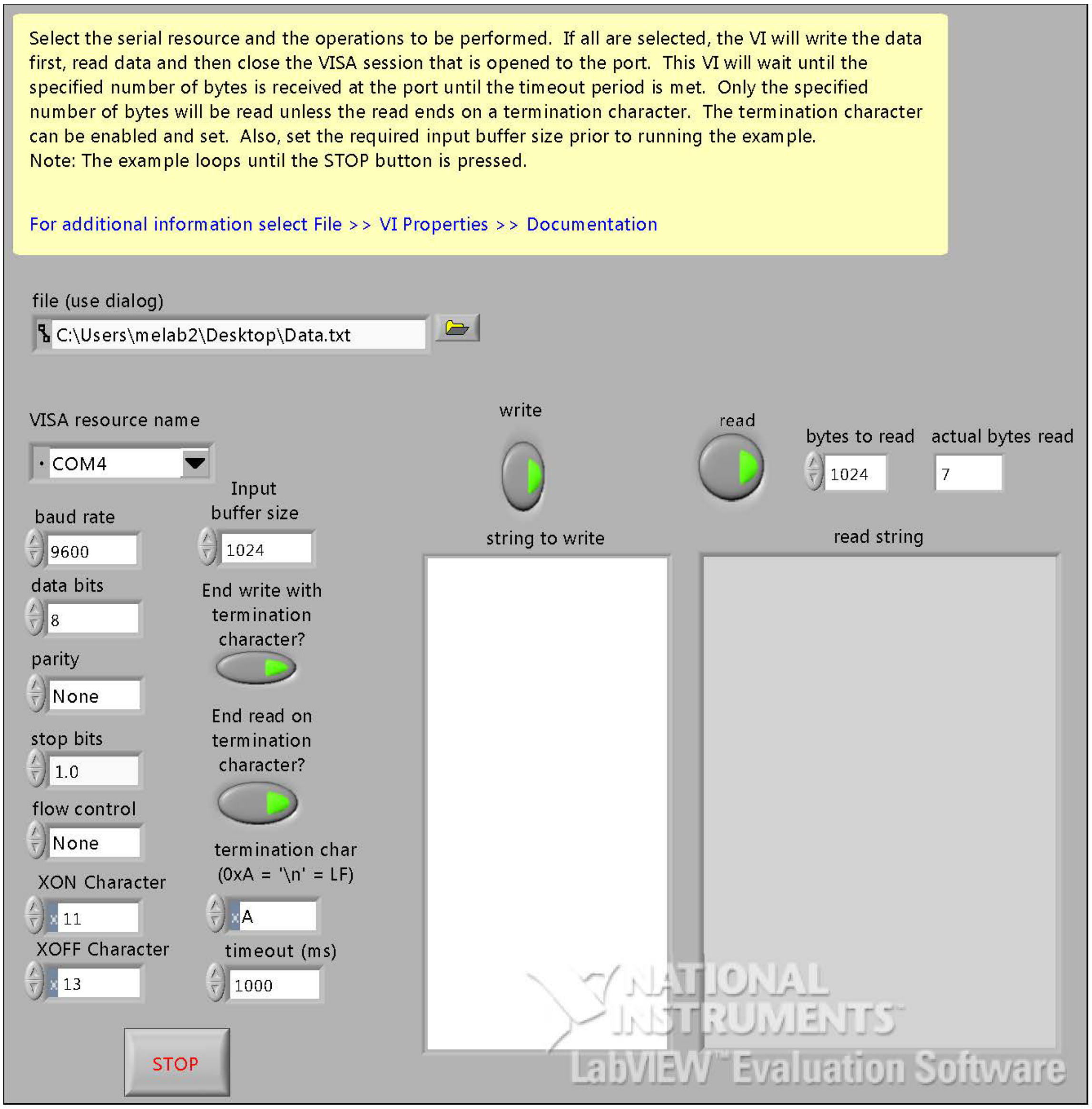


Figure F.1: LabVIEW Front Panel Interface

Table F.1: LabVIEW Command List and Descriptions

String to Write	Read String	Description
TIME <hours>,<minutes> DATE <month>,<date>,<year> TIME_DATE?	13,25,15,5,20,15	The time is 1:25:15 PM on May 20,2015 Set each time program is run so the time stamp on data points is accurate
INTVL 0,0,0 INTVL?	0,0,0	The interval between each scan is 0 hours, 0 minutes, and 0 seconds This shortens the time between each scan to as small as possible
PRINT 1 PRINT_TYPE 1,0	1 1,0	Data logging enabled Destination for logged data is internal memory and all scans are recorded
RATE 1 RATE?	1	Selects the fast measurement rate This records the data for all 20 channels in under a second
LOG_MODE 1 LOG_MODE?	1	Maintain the oldest scans and discard new
LOG_COUNT?	0	Returns how many scans are stored in internal memory Make sure LOG_COUNT returns 0 at the beginning of each trial and a higher number at the end of each scan
LOG_CLR	>!	Clear logged scans
SCAN 1	>!	Enable scanning
SCAN 0	>!	Disable scanning Will wait until current scan is finished then disable
LOG?		Returns the following information ~Date and time at the start of the logged scan ~Values for each channel ~Status of ALARM OUTPUTS, DIGITAL I/O, and totalize count

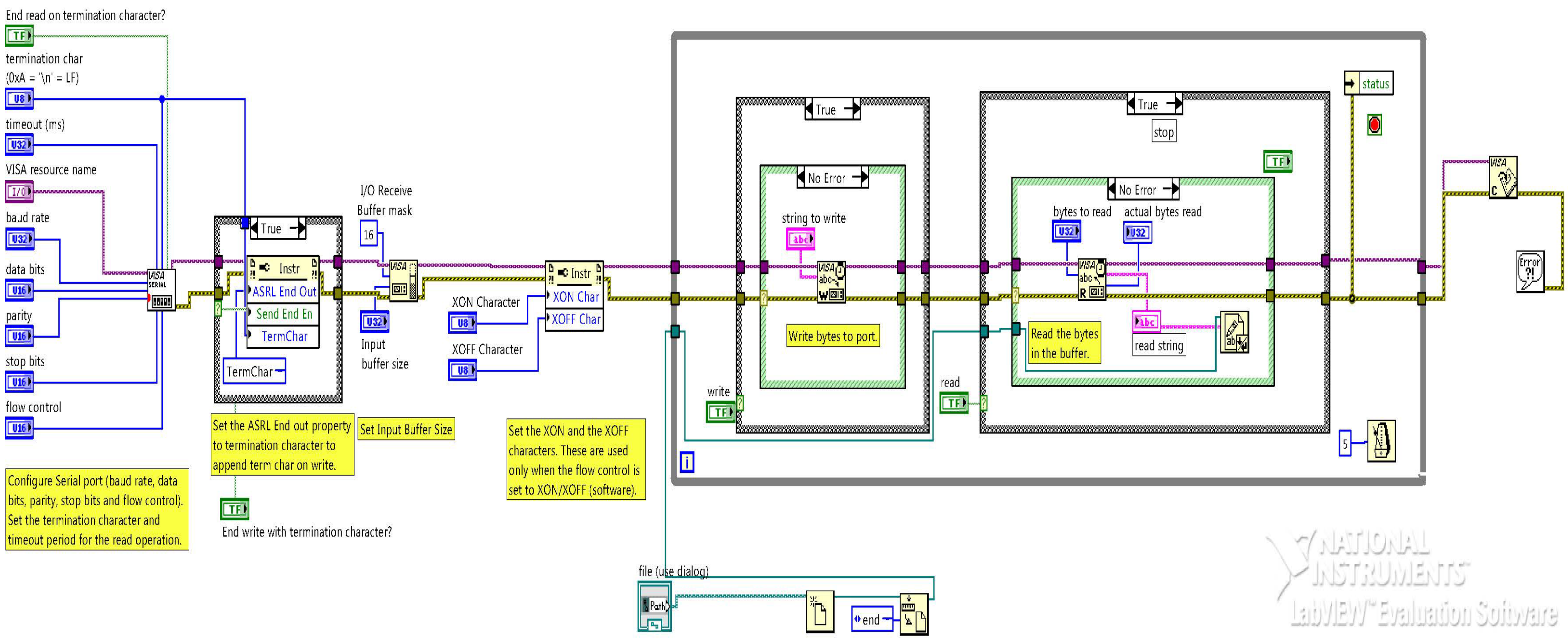


Figure F.2: LabVIEW Control Loop

APPENDIX G

START-UP PROCEDURE

Disclaimer: the following procedure is designed for use in the Cal Poly San Luis Obispo Thermal Science Lab. It is intended for use by trained personnel of the system and does not act as a standalone guide. All problems and malfunctions with the system should be reported to Cal Poly facilities.

Table G.1: Boiler, Pre-Heater, and Cooling Tower Valve Labeling and Descriptions

Valve Number	Description
01	Gas Supply
02	Main Gas Line
03	Main Gas Line
04	Gas Line Split
05	Gas Line Split
06	Cooling Tower Water Source
07	Cooling Tower Return
08	Cooling Tower Drain
09	Pre-Heater Tank Drain
10	Building Water Supply
11	Blow Down Water Source
12	Pre-Heater Pump Source
13	Suction Pump Feed Water Supply
14	Ball Lower Blow Off
15	Globe Lower Blow Off
16	Middle Blow Off
17	Upper Blow Off
18	Sight Glass
19	Boiler Pressure Gauge
20	Boiler Steam Supply
21	Boiler Steam Trap
22	Boiler Steam Return
23	Pre-Heater Steam Bleed Off

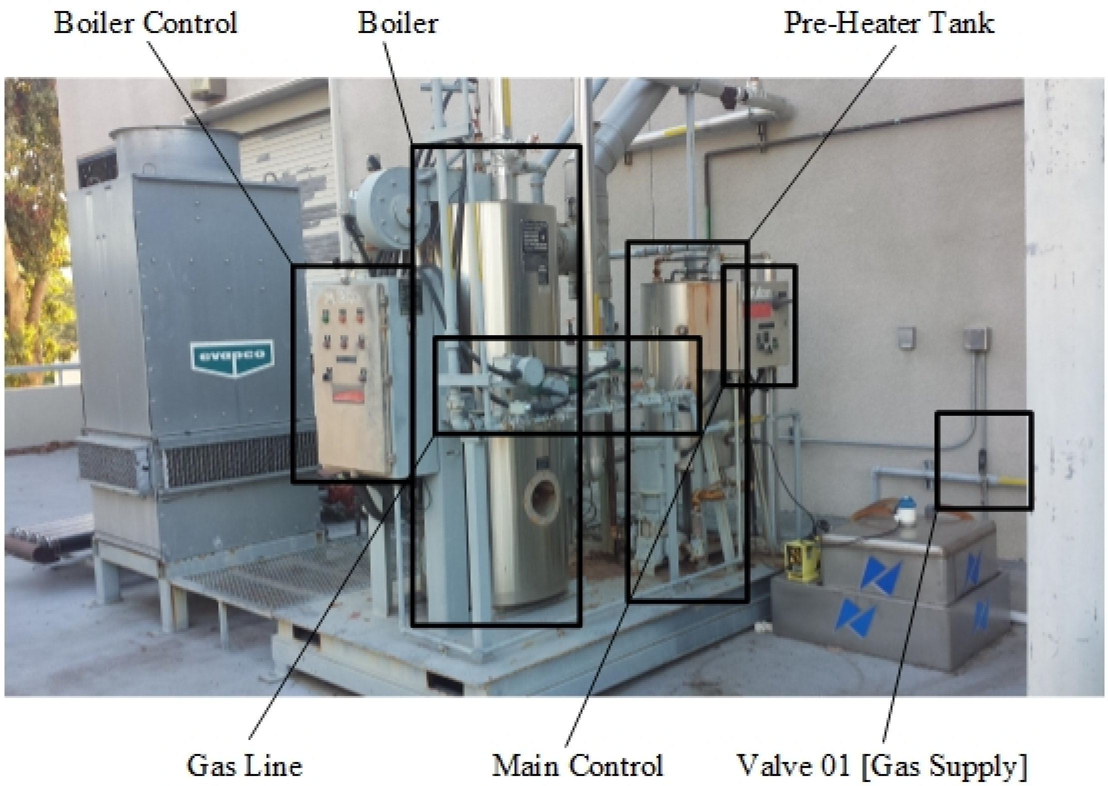


Figure G.1: Boiler and Cooling Tower System Front

Figure G.2: Boiler and Cooling Tower System Back

Boiler and Cooling Tower Preliminary Procedure

1 Gas Supply

- ☐ Open Valve 01 [Gas Supply] {Figure G.1}
- ☐ Verify Valve 02 and Valve 03 [Main Gas Line] Valve 04 and Valve 05 [Gas Line Split] opened {Figure G.3}
- ☐ "Low Gas Pressure Sensor" {Figure G.3} pressed down firmly
- ☐ "High Gas Pressure Sensor" {Figure G.3} pressed down until click heard (If higher pressure lock out switch won't budge the lock out is already reset)

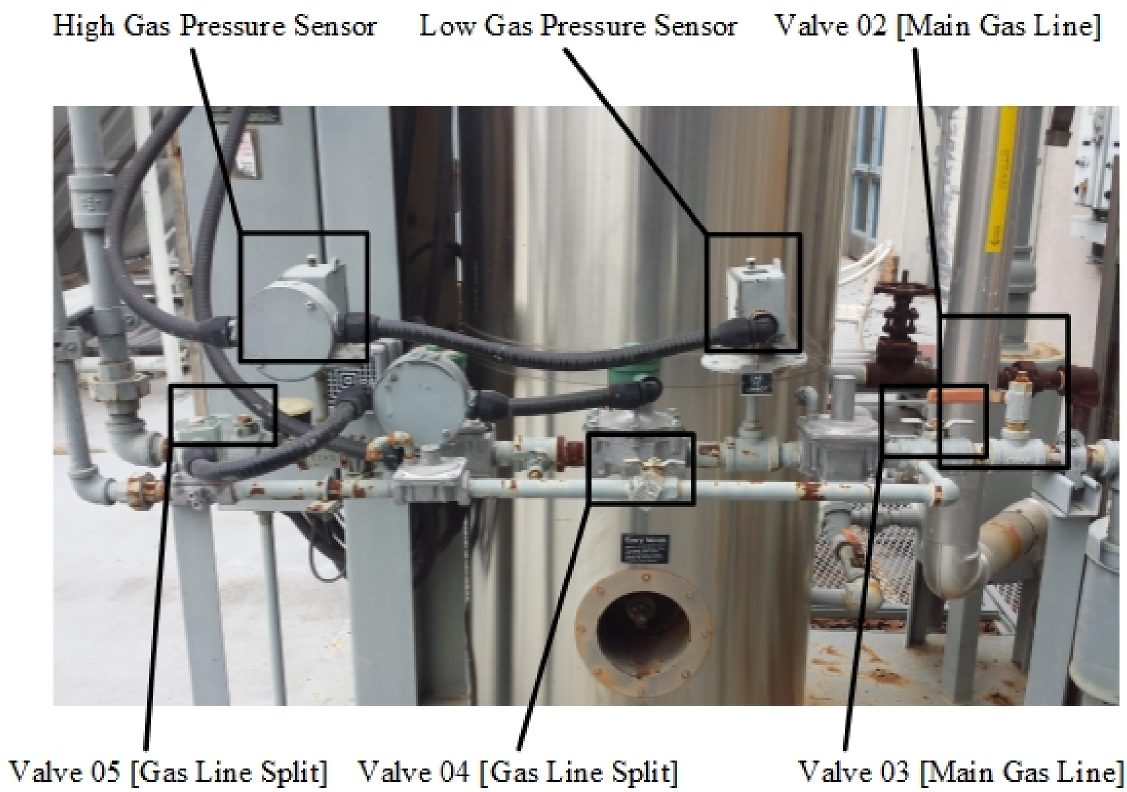


Figure G.3: Gas Line

2 Cooling Tower Water Supply

- ☐ Open Valve 06 [Cooling Tower Water Source] {Figure G.4}
- ☐ Verify Valve 07 [Cooling Tower Return] opened {Figure G.4}
- ☐ Close Valve 08 [Cooling Tower Drain] {Figure G.4}

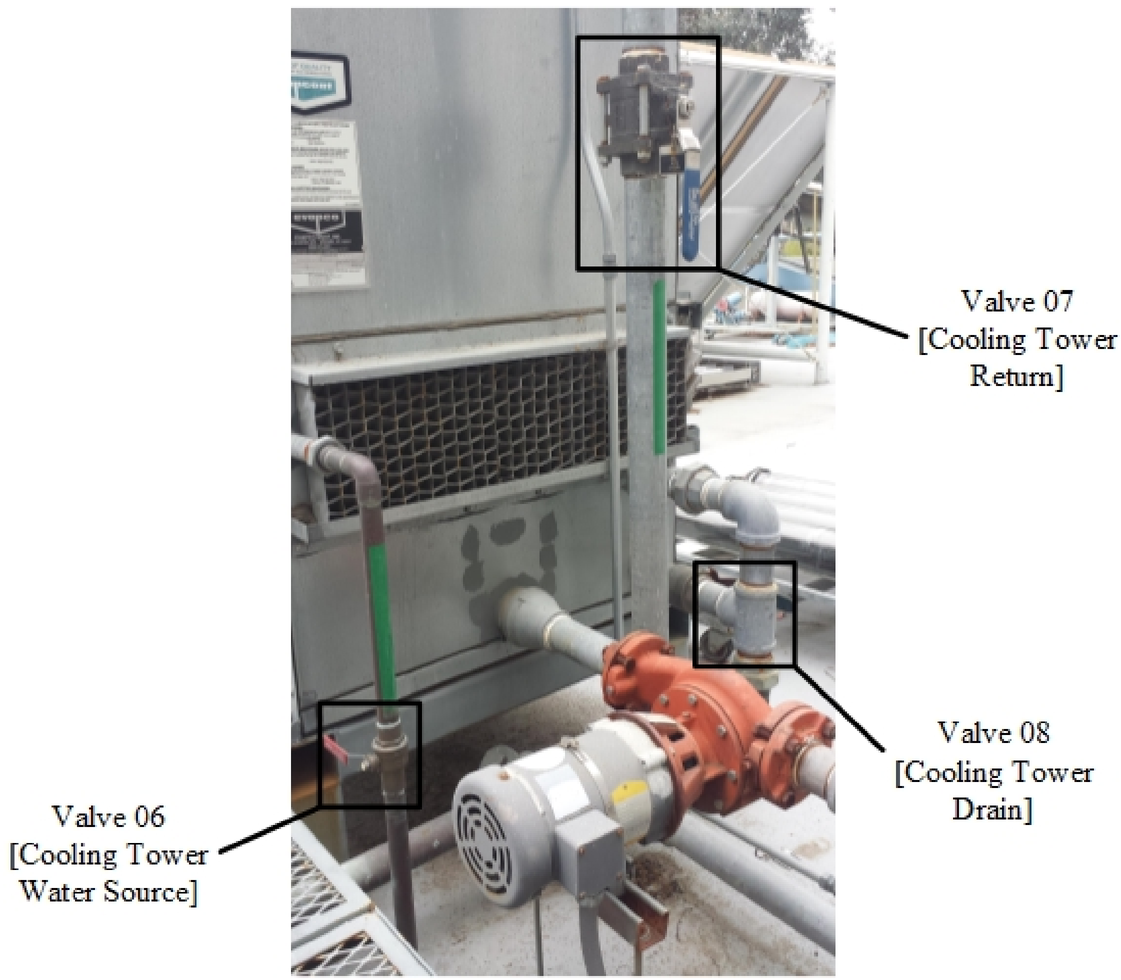


Figure G.4: Cooling Tower Valves

3 Pre-Heater Water Supply

- ☐ Close Valve 09 [Pre-Heater Tank Drain] underneath pre-heater tank {Figure G.1}
- ☐ Open Valve 10 [Building Water Supply] {Figure G.2}
    - Cooling tower and pre-heater tank will fill up and then stop flow themselves
    - Visually check water level with sight glass on side of pre-heater
    - Pre-heater will make squealing noise when tank close to being full
- ☐ Verify Valve 11 [Blow Down Water Source] opened {Figure G.5}
- ☐ Verify Valve 12 [Pre Heater Pump Source] opened {Figure G.5}
- ☐ Verify Valve 13 [Suction Pump Feed Water Supply] opened {Figure G.5}
    - Rising stem valve, turn handle so that stem is as high as possible then rotate quarter turn in the opposite direction while tension felt
- ☐ Drag drainage hose connected to the water drain to the drain by the lab doors

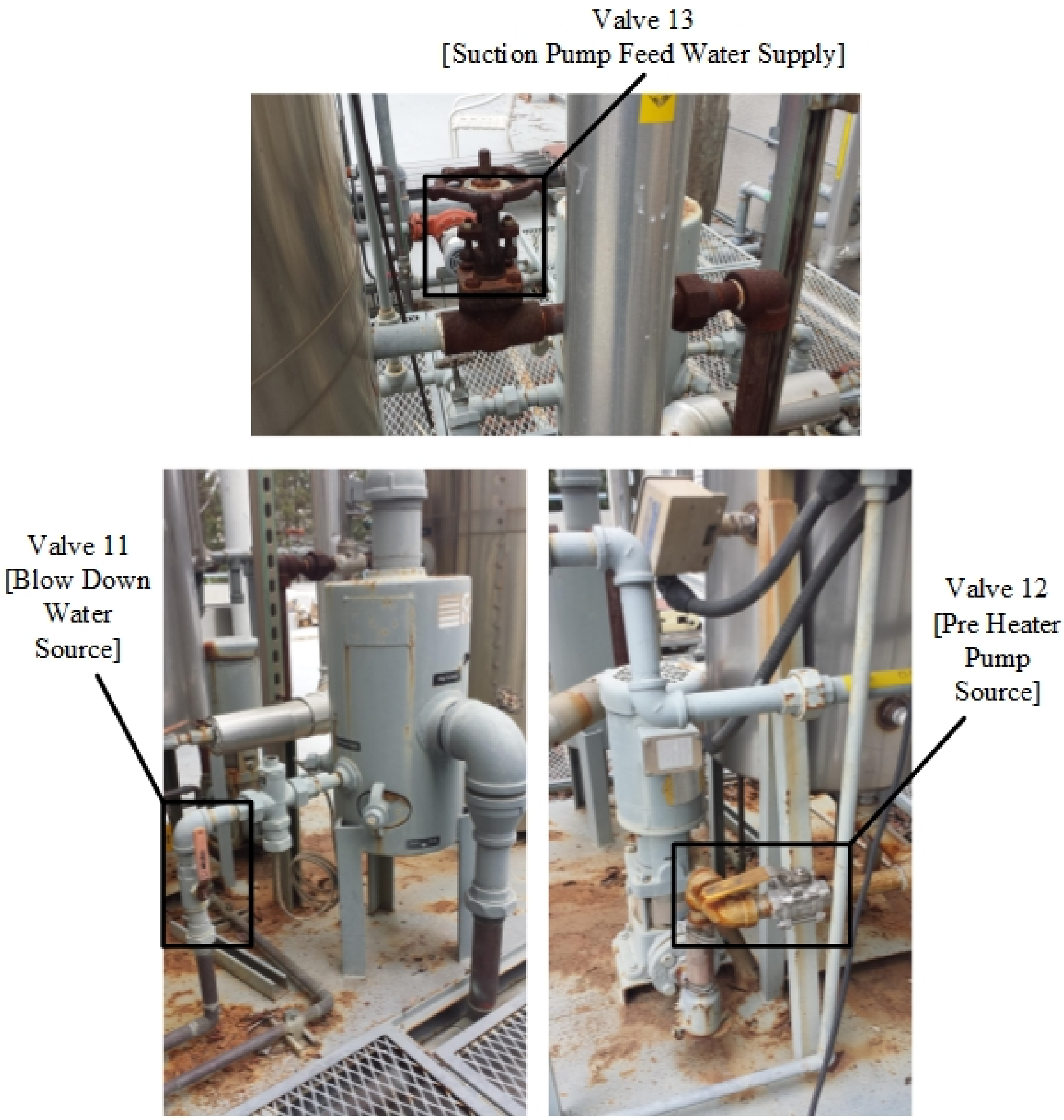
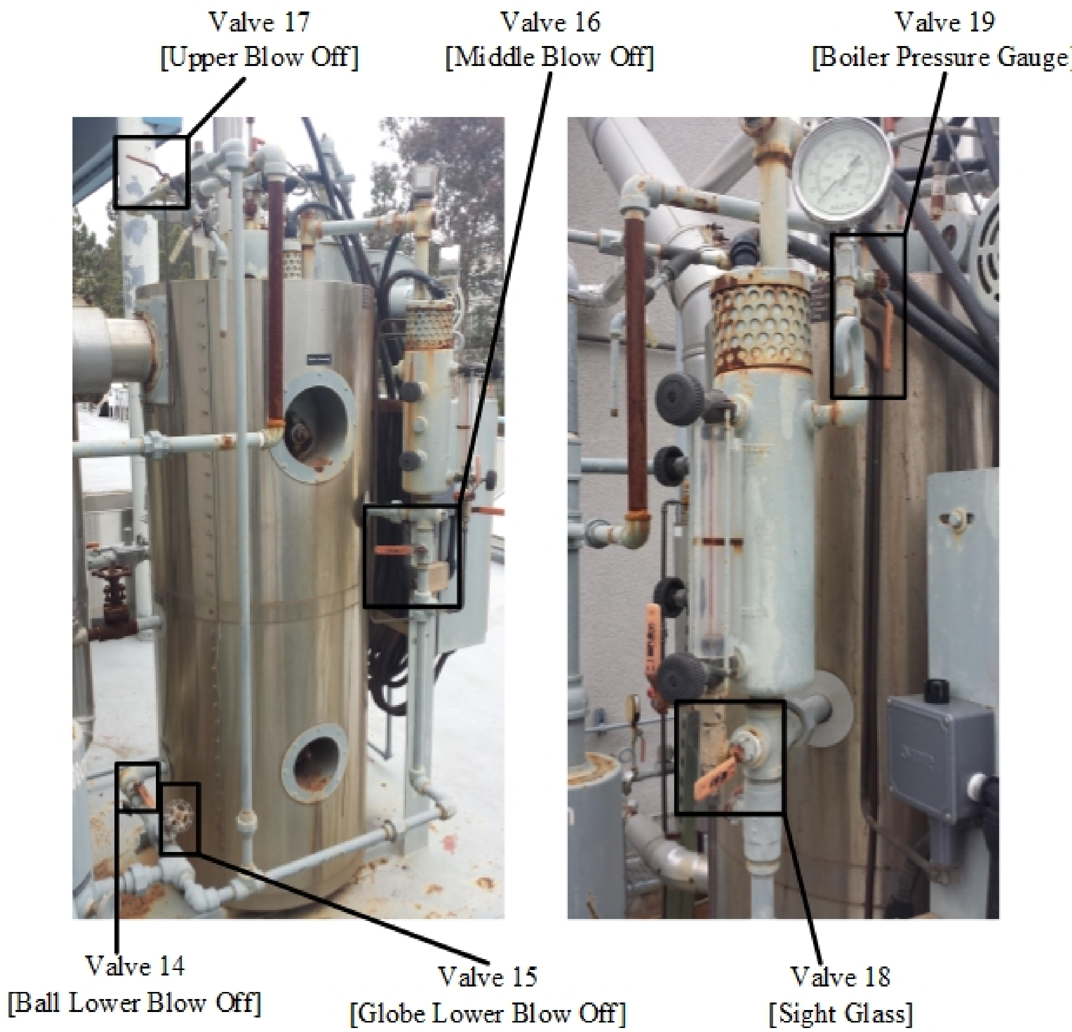


Figure G.5: Pre-Heater Valves

4 Boiler Water Supply

- ☐ Close Valve 14 [Ball Lower Blow Off] and Valve 15 [Globe Lower Blow Off] {Figure G.6}
- ☐ Close Valve 16 [Middle Blow Off] {Figure G.6}
- ☐ Verify Valve 17 [Upper Blow Off] closed {Figure G.6}
- ☐ Close Valve 18 [Sight Glass] {Figure G.6}
  - When valve is closed sight glass will read water level
- ☐ Close Valve 19 [Boiler Pressure Gauge] {Figure G.6}
  - When valve is closed pressure gauge will read pressure



5 Steam Line Settings

- ☐ Go inside of the thermal science lab
- ☐ Verify Valve “Turbine Steam Supply” closed {Figure G.7}

Figure G.7: Turbine System

- ☐ Go back outside
- ☐ Open Valve 20 [Boiler Steam Supply] {Figure G.8}
- ☐ Open Valve 21 [Pre-Heater Steam Bleed Off] {Figure G.8}

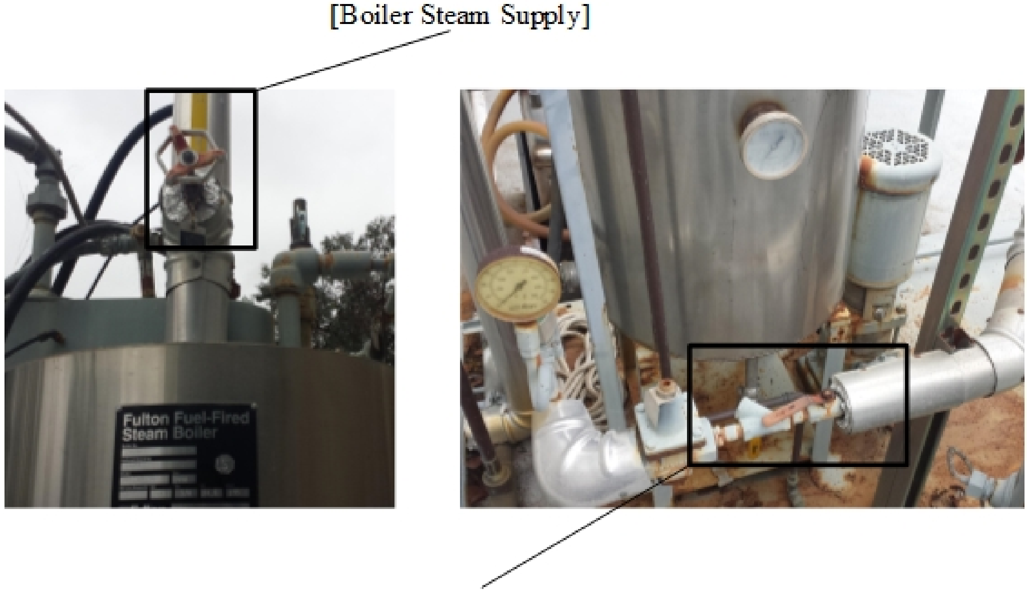


Figure G.8: Steam Boiler Valves

6 Steam Line Settings (Cont.)

- Both valves are located slightly underneath the floor grate

☐ Verify Valve 21 [Boiler Steam Trap] opened {Figure G.9}

☐ Close Valve 22 [Boiler Steam Return] {Figure G.9}

Figure G.9: Steam Boiler Drainage Valves

7 Electrical Settings

- ☐ Go over to the Boiler Control panel {Figure G.1}
- ☐ Verify "Boiler Control Switch" in off position
- ☐ Go over to Main Control {Figure G.1}
- ☐ Rotate "Main Power" switch to on
- ☐ Verify "Chemical Pump Switch" in off position
- ☐ "Feed Pump" HOA switch turned to Auto (IMPORTANT alarm will go off due to no water in boiler, alarm will remain on while boiler fills with water)
- ☐ Watch level of the sight glass on the Boiler, the Main Feed Water Pump will turn off once the level in the Boiler sight glass reaches around 5/8 of maximum value
- ☐ Alarm will remain on until "Low Water Reset" on boiler control panel is pressed

Figure G.10: Main Control

8 Safety Tests [To be done before each operation]

- ☐ Go over to the Boiler Control panel {Figure G.1}
  - The time delays for the Cut Off (top time dial) and Lock Out (lower time dial) are controlled here
- ☐ Open panel
- ☐ On left side there are two control boxes labeled “Lock Out” and “Cut Off” {Figure G.11}

Figure G.11: Inside Boiler Control Panel

- ☐ Cut Off/ Sight Glass Safety Test
  - Adjust time delay on Cut Off dial to zero seconds and Lock Out dial to 60 seconds
  - “Feed Pump” HOA switch turned to Off
  - Open Valve 14 [Ball Lower Blow Off] and Valve 15 [Globe Lower Blow Off] {Figure G.6}
  - Check sight glass, when water hits the bottom of the sight glass alarm should sound
    - If alarm goes off “Cut Off” works correctly
    - If alarm does not go off readjust time delay on Cut Off dial
    - If alarm still does not sound off again turn off system and call facilities
  - Close Valve 14 [Ball Lower Blow Off] and Valve 15 [Globe Lower Blow Off] {Figure G.6}

- "Feed Pump" HOA switch turned to Auto
- Alarm should turn off automatically

☐ Lock Out Safety Test

- Adjust time delay on Lock Out dial to zero seconds and Cut Off dial to 60 seconds
- "Feed Pump" HOA switch turned to Off
- Open Valve 14 [Ball Lower Blow Off] and Valve 15 [Globe Lower Blow Off] {Figure G.6}
- Check sight glass, because sight glass only displays top half of tank alarm should sound about 1-2 minutes after water passes the bottom of the sight glass
    - If alarm goes off 'Lock Out" works correctly
    - If alarm does not go off turn off system and call facilities
- Close Valve 14 [Ball Lower Blow Off] and Valve 15 [Globe Lower Blow Off] {Figure G.6}
- "Feed Pump" HOA switch turned to Auto
- Verify that alarm does not turn off automatically after water reaches max level in boiler
- Reset alarm with the "Low water Reset" switch

Boiler Start Up Procedure

9 Boiler Turn On

- ☐ Verify "Ball Lower Blow Off" and "Globe Lower Blow Off" valves are closed
- ☐ Verify "Boiler Pressure Gauge" valve closed
- ☐ Press "High Limit Pressure Switch" to reset steam lock out
  - Box on the left side of boiler control panel
- ☐ Set time delay on Cut Off to 10 seconds and Lock Out to 30 seconds
  - These values are subject to change depending on nature of boiler use
- ☐ Inside of boiler control panel find the Blue Honeywell Burner Control
  - Display should read "Stand By"
  - "Power" light should be flashing green
  - "Pilot, "Flame", and "Main" lights should all be off

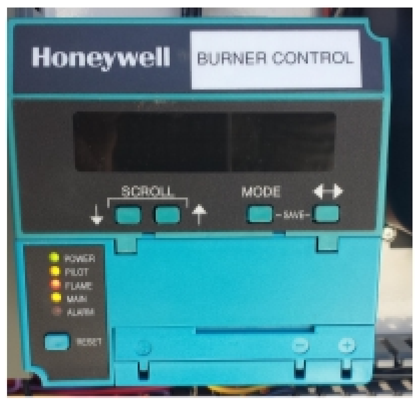


Figure G.12: Honeywell Burner Control

- ☐ Turn "Boiler Control" switch to LCL
  - IMPORTANT read the next part first to be prepared for startup verification process
  - Fan should immediately turn on
- ☐ Quickly verify that on the Blue Honeywell Burner Control all stages turn on correctly
  - "Power" light turned to solid green and Honeywell Control display reads "Purge"
  - Around 5 seconds later "Pilot" light turned to orange
  - During the next 10-15 seconds two clicks will be heard from gas line which verify both solenoids actuating
  - After both clicks are heard "Flame" light turned to red
  - "Main" light turned to orange and Honeywell Control display reads "Run"

10 Operation Procedures

- ☐ Wait 30-45 minutes for boiler to heat up
- ☐ When system is fully operational boiler will cycle
  - ▪ When boiler pressure hits 95 psig boiler will turn off and Honeywell Control display reads “Stand By”
  - ▪ When boiler pressure hits 85 psig boiler will turn back on automatically
- ☐ If water is shooting out of “Boiler Pressure Gauge” drain some of the water inside boiler by opening up “Ball Lower Blow Off” and “Globe Lower Blow Off” valves
  - ▪ IMPORTANT Open ball valve first and globe valve second
  - ▪ IMPORTANT Close globe valve first and ball valve second
- ☐ If water is shooting out of the sight glass open and close “Sight Glass” valve until leakage stops
- ☐ If water is shooting out of any of the pipe connections or joints turn off system and contact facilities
- ☐ Perform water column & sight glass blow downs to clean pipes
  - ▪ Open “Middle Blow Off” valve
  - ▪ After waiting a few seconds close “Middle Blow Off” valve
  - ▪ Open “Sight Glass” valve
  - ▪ After waiting a few seconds close “Sight Glass” valve
  - ▪ Repeat the previous steps 2-3 times
- ☐ Perform lower blow downs on system to clean boiler
  - ▪ Open “Ball Lower Blow Off” and “Globe Lower Blow Off” valves
    - IMPORTANT Open ball valve first and globe valve second
  - ▪ After a few seconds close “Ball Lower Blow Off” and “Globe Lower Blow Off” valves
    - IMPORTANT Close globe valve first and ball valve second
  - ▪ Repeat the last two steps

Turbine and Cooling Tower Start Up Procedure

11 Cooling Tower Turn On

- ☐ Go over to the Cooling Tower Controls {Figure G.2}
- ☐ Turn on power to both Cooling Tower Pump and Fan control boxes
- ☐ "Cooling Tower Pump" and "Cooling Tower Fan" switches {Figure G.13} to "Hand"

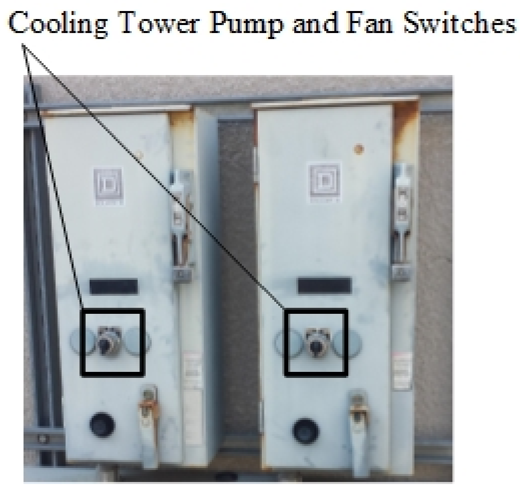


Figure G.13: Cooling Tower Controls

12 Turbine Warm Up

- ☐ Wait until boiler is cycling between 85 to 95 psig
- ☐ Turn on all Dynamometer controls and displays {Figure G.14}
  - 3 green buttons and one red switch
- ☐ Make sure "Governor Speed Adjust Screw" is turned counterclockwise to the low speed condition {Figure G.14}
  - High speed condition when screw fully turned clockwise, low speed condition is screw turned counterclockwise
  - A flat head screwdriver is required to rotate control

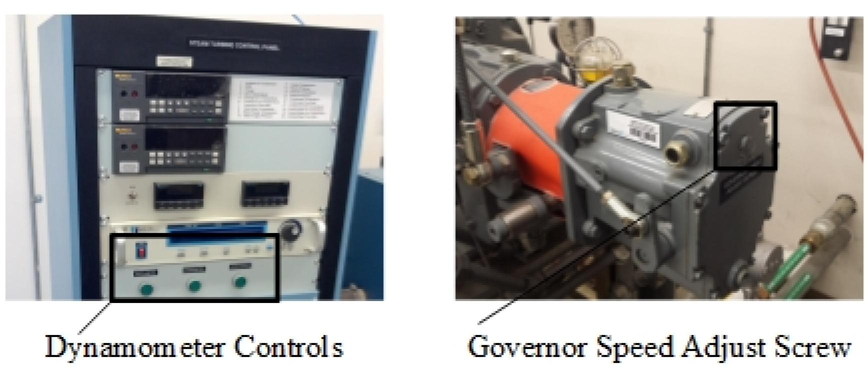


Figure G.14: Dynamometer and Governor Control

☐ Close “Turbine Sealing Steam Supply” and “Turbine Sealing Steam Control” {Figure G.15}

☐ Open “Turbine Drainage” valve {Figure G.15}

☐ Slowly open up “Turbine Steam Supply” valve {Figure G.15}

- After turning the valve slightly wait until each section of the turbine starts to warm up
- Once a section is warm close that “Turbine Drainage” and allow time for next section to warm up
- Once all the drain valves are closed turbine will start to spin

☐ Turn on Condensate pump by switch on wall

☐ Open “Turbine Steam Supply” even more until turbine is spinning at 900 RPM {Figure G.15}

☐ Open “Turbine Sealing Steam Drainage” valve {Figure G.15}

☐ Open “Turbine Sealing Steam Supply” valve {Figure G.15}

☐ Open “Turbine Sealing Steam Control” slowly {Figure G.15}

☐ Leave turbine spinning at 900 RPM and continuing to warm up for about half an hour

Figure G.15: Steam Turbine Valves

13 Turbine Operating Conditions

- ☐ After turbine is warmed up continue to open "Turbine Steam Supply" slowly until turbine is running around its operating speed of 3550 RPM.
- ☐ Plug in Venturi Vacuum Pump to air supply located on lab table near the front door
    - Open up venturi valve located near pressure gauge {Figure G.16}
    - Verify venturi works by checking for air flow out of venturi pump {Figure G.16}

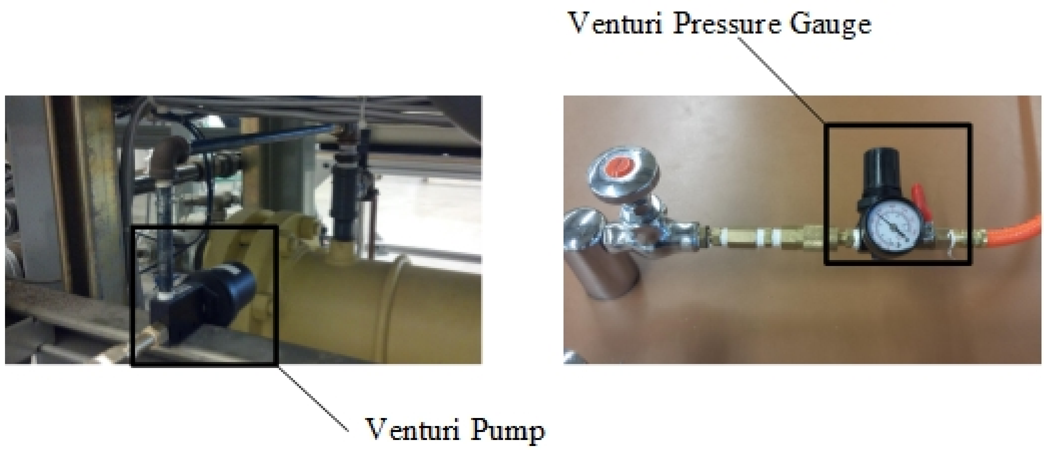


Figure G.16: Venturi System

- ☐ Let system return to a balanced state around 3550 RPM

14 Experiment Procedure

- ☐ Check pumps and reservoir to ensure enough water to hold vacuum
- ☐ Ensure that initial load of Dynamometer is zero by turning load dial all the way to the left
- ☐ Turn on brake
- ☐ Start to slowly turn load dial
    - Be aware of the pressure coming into the turbine, if the pressure starts to drastically drop stop increasing the load and wait for system to regain a balanced state
- ☐ Turn on hydrostatic head configuration pumps
- ☐ Run "SCAN 1" in LabVIEW to start recording values
- ☐ Continue to turn load dial until display reads 3.0 HP
- ☐ Once data is taken at steady state return load dial to zero and turn off brake
- ☐ Run "SCAN 0" in LabVIEW to stop recording value

System Shut Off Procedure

15 Turbine Shut Off
- ☐ Turn "Governor Speed Adjust Screw" to low speed condition
- ☐ Turn Dynamometer load dial until load is zero
  - If turbine speeds up out of control slightly close "Turbine Seal Supply"
- ☐ Open up all turbine drainage valves
- ☐ Close "Sealing Steam" valve {Figure G.15}
- ☐ Unplug Venturi Vacuum Pump
- ☐ Close "Turbine Steam Supply" valve {Figure G.15}

16 Cooling Tower Shut Off
- ☐ "Cooling Tower Pump and Fan Switches" to Off {Figure G.13}
- ☐ Turn off power to both Cooling Tower Pump and Fan control boxes

17 Boiler Cool down
- ☐ Turn "Boiler Control" switch to off position {Figure G.10}
- ☐ Open Valve 19 [Boiler Pressure Gauge] {Figure G.6}
  - This allows steam to quickly be released from inside boiler
- ☐ Open Valve 16 [Middle Blow Off], Valve 14 [Ball Lower Blow Off] and Valve 15 [Globe Lower Blow Off] {Figure G.6}
  - Leave valves open for a few minutes to drain out some of the hot water in the boiler and replace it with cooler water
- ☐ Close Valve 16 [Middle Blow Off], Valve 14 [Ball Lower Blow Off] and Valve 15 [Globe Lower Blow Off] {Figure G.6}
- ☐ Wait until boiler has cooled down

18 Boiler Shutdown
- ☐ "Feed Pump" HOA switch turned to Off
- ☐ Open Valve 16 [Middle Blow Off], Valve 14 [Ball Lower Blow Off] and Valve 15 [Globe Lower Blow Off] {Figure G.6}
- ☐ Rotate "Main Power" switch to Off {Figure G.10}
- ☐ Close Valve 10 [Building Water Supply] {Figure G.2}
- ☐ Open Valve 08 [Cooling Tower Drain] {Figure G.4}
- ☐ Open Valve 09 [Pre-Heater Tank Drain] underneath pre-heater tank {Figure G.1}
- ☐ Close Valve 01 [Gas Supply] {Figure G.1}