STUDY OF LOW-GRADE WASTE HEAT RECOVERY AND ENERGY TRANSPORTATION SYSTEMS IN INDUSTRIAL APPLICATIONS

by

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A THESIS

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ABSTRACT

The following report has been compiled to provide a guideline for designing a waste heat recovery and transportation system in an industrial facility. The overwhelming availability of waste heat in the United States demonstrates the inefficiencies and the wasteful practices currently in the industrial sector. These inefficiencies are estimated to be anywhere between 20% and 50% of the total energy purchased and a large portion of this energy is contained in exhaust gas from combustion processes at temperatures below 450°F. This wasted energy does not only cost the manufacturer extra money to operate, but adds an additional environmental impact. This impact is not just from excess CO_2 production but from a steady source of high temperature exhaust gas flowing into the atmosphere.

To assist in recovering and determining the possible uses for this recovered energy, background information and recommended uses for waste heat recovery equipment are given. Additionally, thermal heating fluids and phase change materials which can be used to transport and store this recovered energy are discussed. A sample design of this energy transportation system, along with end uses is provided to demonstrate the method of determining the possible energy and cost savings impact which are estimated to be 14,700 MMBtu/year and $120,000/year respectively. This system design includes recovering energy from 485°F exhaust air flowing at 4,050 cfm and distributing this energy throughout the plant to perform several different tasks. These tasks include space heating, domestic hot water production, maintaining a minimum temperature in glue baths, powering an absorption chiller and preheating boiler feed-water. The remaining

energy is then stored in two thermal storage tanks, one for the sensible heat exchanger loop and one for the condensing economizer loop. The design shows a rough implementation cost based on the material and labor to be just over $323,000. These values can be used to demonstrate a typical payback period of 2.7 years. The information provided throughout the report is intended to inform the reader of the overall impacts of wasted heat energy while providing options and methods for recovering this wasted energy.

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Chapter 1

Introduction

Projected further increases in energy costs and global concerns of manmade climate change have made reduction in energy consumption a critical segment of all businesses, small and large. A large contribution to the overall cost and energy consumption at a given facility stems from inefficiencies in a process. This process inefficiency can be the result of a number of concerns ranging anywhere from a poor plant layout where extra time must be spent transporting goods from one station to the next to energy inefficient equipment and use. This demonstrates the consequences of energy inefficiency and the benefits that can result from improving it. For most cases, a typical system can be improved to decrease the amount of energy wasted during its operation. However once the equipment reaches a certain level, it will inherently require a certain amount of wasted energy to operate. A prime example is during a combustion process.

The most abundant form of this exhausted energy lies within a low temperature range which includes temperatures around 450°F. Although this lower temperature exhaust gas does not contain as much energy per pound of gas exhausted, the sheer volume of this wasted heat energy makes this temperature range very important in energy recovery. There are many different methods and equipment that can be used to recover, store, transport and consume this recovered energy and these different types of equipment can allow the facility to utilize this already purchased energy to offset costs elsewhere.

The following report has been compiled to provide a guideline for designing a waste heat recovery and transportation system in an industrial facility. The overwhelming availability of waste heat in the United States demonstrates the inefficiencies and the wasteful practices currently used in the industrial sector. These inefficiencies are estimated to be anywhere between 20% and 50% of the total energy purchased. This wasted energy does not only cost the manufacturer extra money to operate, it also adds an additional environmental impact. This impact is not just from excess CO_2 production but from a steady source of high temperature exhaust gas flowing into the atmosphere.

To assist in recovering and determining the possible uses for this recovered energy, background information and recommended uses for waste heat recovery equipment are given. Additionally, thermal heating fluids and phase change materials which can be used to transport and store this recovered energy are discussed. A sample design of this energy transport system, along with end uses is provided to demonstrate the method required to determine the possible energy and cost savings impact. This design system includes recovering the energy from 485°F exhaust air flowing at 4,050 cfm and distributing this energy throughout the plant to perform several different tasks. These tasks include heating of occupied spaces, domestic hot water production, maintaining a minimum temperature in 14 glue baths, powering an absorption chiller to provide year round cooling and preheating boiler feed water. The remaining energy is then stored in two thermal storage tanks, one for the sensible heat exchanger loop and one for the condensing economizer loop. The design shows a rough implementation cost based on the material and labor cost to demonstrate a typical payback period. The information provided throughout the report is intended to inform the reader of the overall impacts of wasted heat energy while providing options and methods for recovering this wasted energy.

Chapter 2

Economic and Social Impacts of Waste Heat

Waste Heat Potential

Throughout the past decade there has been an overwhelming push to reduce energy consumption in the United States and globally due to many different factors ranging from rising energy prices to lowering greenhouse gas emissions. For most people, to accomplish this means an entire overhaul of the current system of how tasks are accomplished and massive changes in technology. While this type of thinking is not wrong, a much easier and less expensive option is available through energy efficiency. This idea simply takes the same principles and ideas that are currently in place and improves on them, which in turn require less energy to operate.

A major source of energy that is thrown away into the atmosphere is waste heat from industrial processes. It is estimated that that industries in the United States consume 30% of the all energy produced. In 2009 this percentage translated into 28.2 Quadrillion (10^{15}) Btu's consumed to manufacture and assemble goods [2]. According to a report published by the Lawrence Livermore National Laboratory in 2010, from this energy usage 20% is rejected into the atmosphere as waste heat [38]. This amount is calculated based on the assumption that the end use efficiency of the entire industrial sector is 80% and in turn assuming that the other 20% of energy delivered is rejected as waste heat. Another study conducted by the Department of Energy on waste heat recovery in U.S. industry states that the exact quantities of waste heat have

been estimated to be as much as 20 to 50% of the total energy consumption [4]. Yet another study was conducted for the Department of Energy by Energetics, Incorporated and E3M, Incorporated. This study took place in 2004 and used a different method to determine the amount of wasted energy generated in industry [16]. This method took the approach that all energy supply, distribution systems and energy conversion process have inefficiencies and generate wasted heat. Average efficiencies for each process were then assumed and the amount of energy lost was calculated. This study estimated that about 32% of the energy input to plants is lost to its boundaries prior to its use for the intended purpose. This study did not include energy that is wasted after a process has been completed such as high temperature exhaust gas from combustion processes [16]. This study demonstrates that the amount of waste heat generated in industry is more likely on the upper side of the range found in the other studies.

While these previously described methods are able to quickly approximate the amount of energy wasted in the industrial sector, the gross assumption on efficiencies is only capable of giving an order of magnitude analysis of the actual amount. A more detailed analysis of the amount of energy wasted cannot be located and the true quantities are not monitored or known.

Quality of Waste Heat

Because every industrial process is somewhat different, waste heat is not rejected at a constant temperature. These temperatures will vary greatly depending on the type of equipment it is exhausted from, for what purpose the consumed heat is intended, the efficiency of the equipment along with several other contributions. To help evaluate a realistic recovery potential, the waste heat is characterized based on the temperature of the exhaust gas and from there the quantity of each category can be found. There can be any number of different cutoff points in categories based on temperature, however breaking them down based on technology and process limitations

allows for the actual potential recoverability for each to be seen. According to the DOE, these temperature groups are defined as shown below [4].

High Temperature	1,200 °F and above
Medium Temperature	450 °F to 1,200 °F
Low Temperature	450 °F and below

For the purposes of this paper, the recovery of the low temperature range and temperatures close to this threshold will be discussed. This is because this energy is the most difficult to recover which leads many facilities to simply ignore the potential energy which can be saved. This low temperature waste heat makes up about 60% of all waste heat rejected, making it a very large and available energy source. The following figure shows that although the low temperature range waste heat carries less thermal energy, it provides a much larger potential for energy recovery based on the volume exhausted.

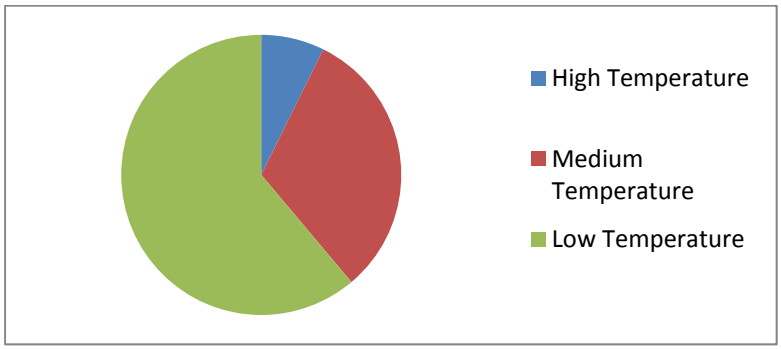

Figure 1. Unrecovered Waste Heat in Different Temperature Groups [4].

The above figure is based on results from a Department of Energy study on the opportunities for waste heat recovery. The figure indicates the potential energy content magnitude of the three different temperature categories of waste heat based on cooling the exhaust air stream to a temperature of 77°F [4].

As mentioned previously, low temperature waste heat recovery faces challenges due to the relatively small temperature difference between the higher energy exhaust gas and the lower temperature heat sink. This forces a heat exchanger with a large surface area to be implemented, which will not only require more space in the gas flow stream but have a higher initial cost to the customer. Other common limiting factors include a minimum temperature for the exhaust gas of about 250 to 300 °F. This is done to prevent water vapor and other corrosive gasses in the exhaust stream from condensing. While this prevents many corrosive substances from coming in contact with the heat exchanger, a great amount of energy is contained in the water vapor and can be recovered by condensing it. Another challenge in recovering the low temperature waste heat energy is finding a feasible use for the recovered energy [4]. Methods for overcoming the first two obstacles will be discussed in more detail in the following sections and a sample system will be developed in Chapter 6 that describes different end uses and methods of transportation of this energy.

Impacts of Waste Heat

The obvious impact of waste heat is the necessity to purchase more energy to make up for what is wasted while performing a task. This forces the operational cost to manufacture products to rise, hurting the company's profitability or causing these inefficiency costs to be passed along to the consumer. However a less obvious impact of wasted heat is the consequences on the environment caused by a steady flow of an estimated 55.5 Quadrillion Btus/hr of energy flowing into the atmosphere from the United States alone [38].

Due to the recent rise in concern of the effects of greenhouse gasses on the environment and global climate change, the impact of energy dissipated as heat into the atmosphere has also been explored as a possible source of manmade climate change. According to Mark Flanner, author

of *Integrating anthropogenic heat flux with global climate models*, all thermal energy released from non-renewable energy sources become an impacting variable for the climate. Only non-renewable sources are considered because they introduce an unexpected heat source onto the atmosphere that otherwise would not have been added. This released energy has long been considered a main cause to the "heat island" in highly industrialized cities, however is often overlooked as an important participant on a global scale. This is because the bulk of the wasted heat comes from a small percentage of the global land mass and causes a relatively small heat flux value when spread out globally. However, much less than 1% of this energy consumed is radiated from the earth without causing any increase in temperature somewhere in the atmosphere. The following figure shows the annual average density of waste heat, heat flux globally.

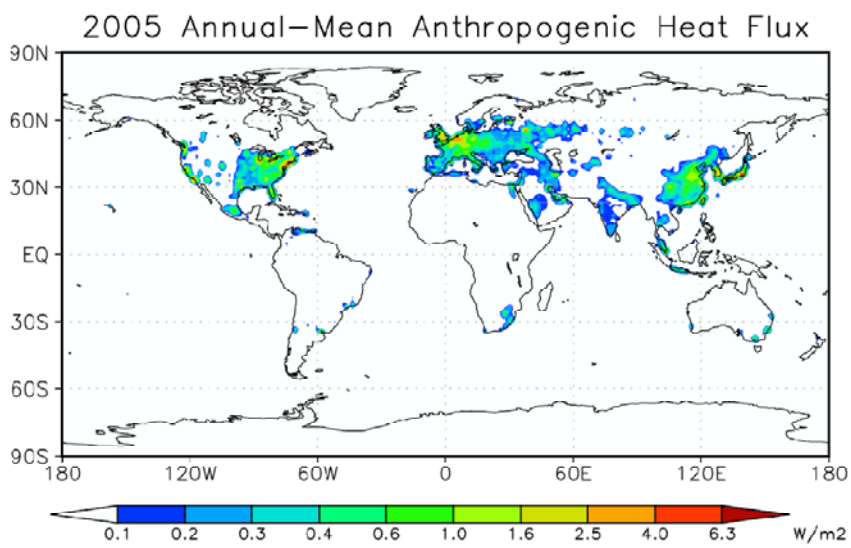

Figure 2. Density of Waste Heat Globally in 2005 [21].

By using this global average method previously mentioned, the heat released has less than 1% of the effect as greenhouse gasses are suspected to have. This has resulted in wasted heat energy

being neglected from the global climate models. This neglection has great potential to cause a deficiency with the current modeling system and to be most impactful on simulated surface temperatures within the model [21].

The results from Flanner's model improvements show that in the near future, there are minimal continental-scale surface temperature changes caused by waste heat. However the simulations show that the effects in the long term future, year 2100, could be an increase in annual mean warming by 0.72 to 1.62°F that occurs over large industrialized regions [21]. These results coincide with results found from a similar study conducted by Chaisson which says that waste heat from energy usage alone will cause an increase in global temperature by 5.4°F in by the year 2328 and by 18°F by year 2458 [10]. With this information, the value at year 2100 is found to be an increase of about 1.3°F which falls almost exactly in the middle of the range found by Flanner's study. This information was found by interpolating using a second order polynomial curve which matches exactly to the data points given by Chaisson.

The argument presented by Chassion with his study is to invest today in solar technology, renewable energy and energy efficiency so that the new technology will be in place when the effects of waste heat on the climate will become a major player. With this technology being implemented slowly throughout the next half century, even with continued growth in energy consumption, the potential threat that waste heat has on the climate will be reduced or at the very least slowed [62].

Both studies show that the current effects on climate change are minimal but have great potential to become very influential in the future and could cause a barrier on economic growth if left unappeased. According to Flanner, if the current growth rate in nonrenewable energy use

continues, the impact of waste heat on the environment will increase from its current effect of approximately 1% of that of greenhouse gasses to around a 50/50 split between the two in 200 years [69]. While these studies include all forms of waste heat and are not limited to industrial situations, it still shows a global climatic impact from waste heat is very possible in the future.

To further verify these studies, a simple calculation was performed to determine the effects that exhausting industrial waste heat into the atmosphere has on its overall temperature. According to research done by Dr. Adam Nieman, the volume of total air in the earth's atmosphere can be contained in a sphere with a diameter of 1,242 miles. This sphere contains a volume of almost 1.5×10^{20} cubic feet of air and can be seen in comparison with the volume of the earth in the following figure [45].

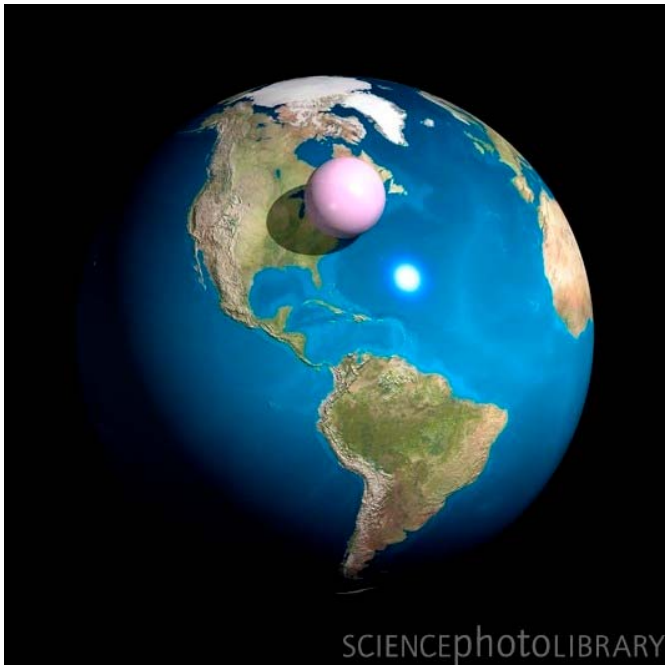

Figure 3. Global Air Volume [45].

The previous analysis conducted by Dr. Nieman also found the mass of air to be approximately 1.13×10^{19} pounds mass. This mass along with an average specific heat value for the air and the

amount of energy rejected as waste heat can be used to calculate a temperature increase to the atmosphere due to this flow of energy. The average specific heat can be assumed to be 0.24 Btu/lbm-°F based on a reference temperature of 68°F. The total waste heat energy emitted by industry can be found using the study conducted by Lawrence Livermore National Laboratory mentioned earlier to be 4.36 x 10^{15} Btus/yr . The following simple equation is used to find the increase in temperature which is simply a rearranged form of the simplified first law of thermodynamics.

$$\Delta T = \frac{Q_{heat}}{m_{air} \times C_{p,\,air}}$$

where

ΔT = increase in overall temperature of the air

Q_{heat} = amount of energy added to the atmosphere from waste heat

m_{air} = overall mass of air in the atmosphere

$C_{p,air}$ = average assumed specific heat of air based on 68°F air temperature

This simple analysis results in an increase in overall temperature of 0.0016°F per year but includes many large assumptions about each of the parameters involved. This simple calculation does however show an order of magnitude analysis on the effects of exhausting high temperature, high energy gas into the atmosphere and puts a separate check on the research presented earlier.

Chapter 3

Common Inefficiencies Resulting in Waste Heat Rejection

The following chapter briefly describes the major sources of inefficiencies when combustion processes occur. They are ordered with the most relevant first and the less significant source at the end.

Above Normal Flue Gas Exhaust Temperature

The main source of inefficiency that is discussed in this paper is the amount of wasted energy contained in exhaust gases from different processes. The paper focuses on how to recover this energy and what to do with it once it has been recovered. This is the greatest source of inefficiency for most combustion type heating processes and represents a relatively large amount of wasted energy. Average exhaust temperatures can range anywhere from about 75°F [63] if a highly efficient energy recovery system is implemented to over 3,000°F in certain processes such as a Nickel refining furnace where no energy recovery system is installed [4].

A process containing combustion as an indirect heating source with this type of inefficiency can be caused by many different reasons and due to the nature of the combustion process, the inefficiency cannot be completely eliminated. An exhaust gas temperature that is higher than normal could be due to an increase in demand for heat by the system. Because the heat transfer area of the system is fixed in the heat exchanger and the percentage of available energy which is able to be transferred to the working media does not follow the temperature differences exactly, some of the available energy is not able to transfer from the high temperature gas to the working

media. Because this extra energy in the exhausted flue gas cannot be transferred directly into the working media inside the equipment, instead of being exhausted into the atmosphere at high temperatures, the heat energy should be recovered by a method similar to the one described later in Chapter 5. As a general rule of thumb, for every 40°F reduction in exiting stack gas temperature, the overall efficiency of the process is increased by 1% [26]. Other sources of this inefficiency could be a faulty part in the equipment or fouling on the heat transfer surface. While it is still a good idea to recover the heat in the exhaust stack, this will not make up for the problems associated with the equipment and the source of the problem should be corrected to greatly increase the efficiency and productivity of the process.

Excess Air in the Combustion Process

During a combustion process, the maximum temperature possible occurs when there is just enough air to mix with the fuel and complete combustion is able to occur. This is called stoichiometric combustion and is not realistic for an actual combustion processes due to mixing limitations between the gasses. Although stoichiometric combustion is not realistic, it is desirable to only supply the minimum amount of excess air to the process to allow for complete combustion to occur. Complete combustion is desired so that unburned fuel particles are not sent through the stack. Incomplete combustion is not only a waste of the purchased fuel but also has the potential to create a safety hazard by creating the possibility for combustion to occur in undesired locations of the process. The reasoning behind using only the minimum amount of excess air is that for every molecule of O_2 found in air, assuming air is the source of oxygen for the process, there are 3.76 molecules of nitrogen that come along with it. This nitrogen is, for by far the most part, inert to the combustion process, however it absorbs energy by being heated from the initial temperature before combustion to the final temperature after combustion. This

heating of nitrogen robs the desired media from absorbing all the energy released during combustion and lowers the temperature of the combustion products which requires a larger energy input to create the desired temperature and results. With the use of modern controls, the O_2 in the exhaust stack can be maintained to about 1.5 to 3 percent for natural gas combustion and only slightly higher for number 2 and number 6 fuel oils as well as coal [26]. The use of older technology, which uses bars and linkages to create a positioning control system for regulating the incoming air, the amount of O_2 in the stack gas can still be kept well under 8%. The following figure shows the percent of the fuel's higher heating value (in this case natural gas) that is lost through the stack when the exhaust gas temperature is fixed at 450°F and 400°F and the outside conditions are set to 70°F. This type of analysis should only be considered accurate when there is less than 100 ppm of CO (combustible media) in the exhaust gas stream [26].

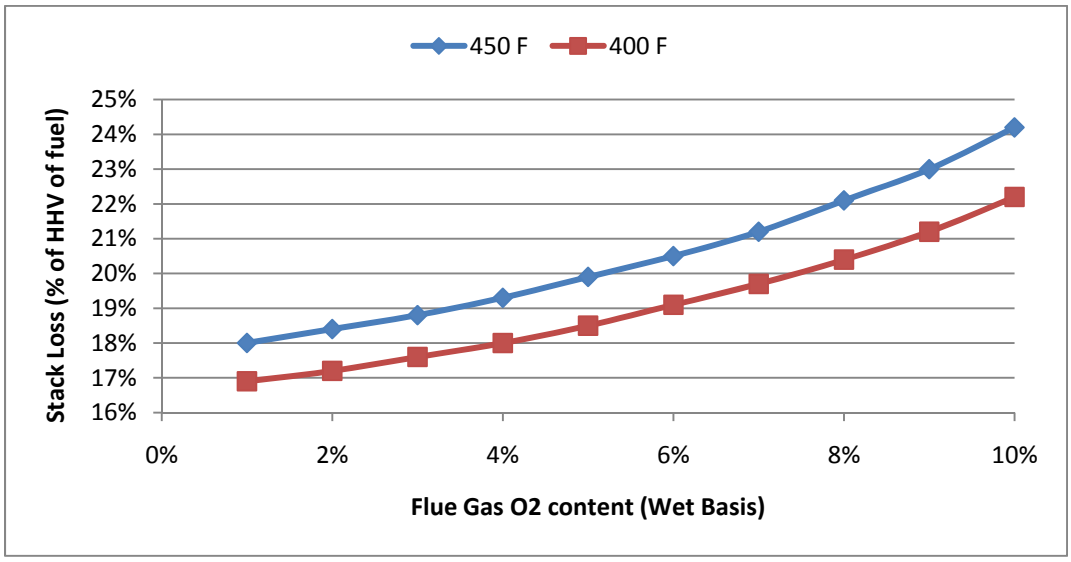

Figure 4. Stack Losses Due to Excess Air in Natural Gas Combustion [26].

The O_2 content in the flue gas can be easily measured using either a handheld combustion analyzer although some exhaust stacks may have permanent O_2 sensors mounted in which case

the value can be determined from there. However as a note of caution, most permanent O_2 sensors are made from zirconia and measure O_2 content on a wet basis, whereas most handheld combustion analyzers measure O_2 content on a dry basis and a different set of data should be used other than the type shown above.

It should also be noted that excess air losses are confounded with the previous section of inefficiencies, *Above Normal Flue Gas Exhaust Temperature.* As the temperature of the exhaust gas decreases, so will the losses due to excess O_2 content in the stack gas. This relationship is shown in the previous figure. Also when the amount of excess air in the system is decreased, the volumetric flow of the exhaust gas will also decrease, affecting the amount of energy that can be recovered by a heat exchanger.

Shell Losses

Another, but much less significant, major loss in a combustion heating system is the radiation and convective heat transfer losses from exposed hot surfaces on the equipment or distribution system, if they exist. This is generally only a major source of loss if the equipment has been shut down for an extended period of time and is turned on, requiring the equipment itself to be raised to the operating temperature from the ambient temperature. Overall this loss will account for less than 1% of the fuel energy input but can be easily reduced by the addition of insulation to the exposed hot surfaces [26].

Chapter 4

Low Temperature Waste Heat Recovery System Components

The following chapter describes different approaches to heat recovery and the equipment used to convert the heat energy into a useable form. The system components and the flow of the following section are shown in Figure 5 and are further broken down throughout the chapter.

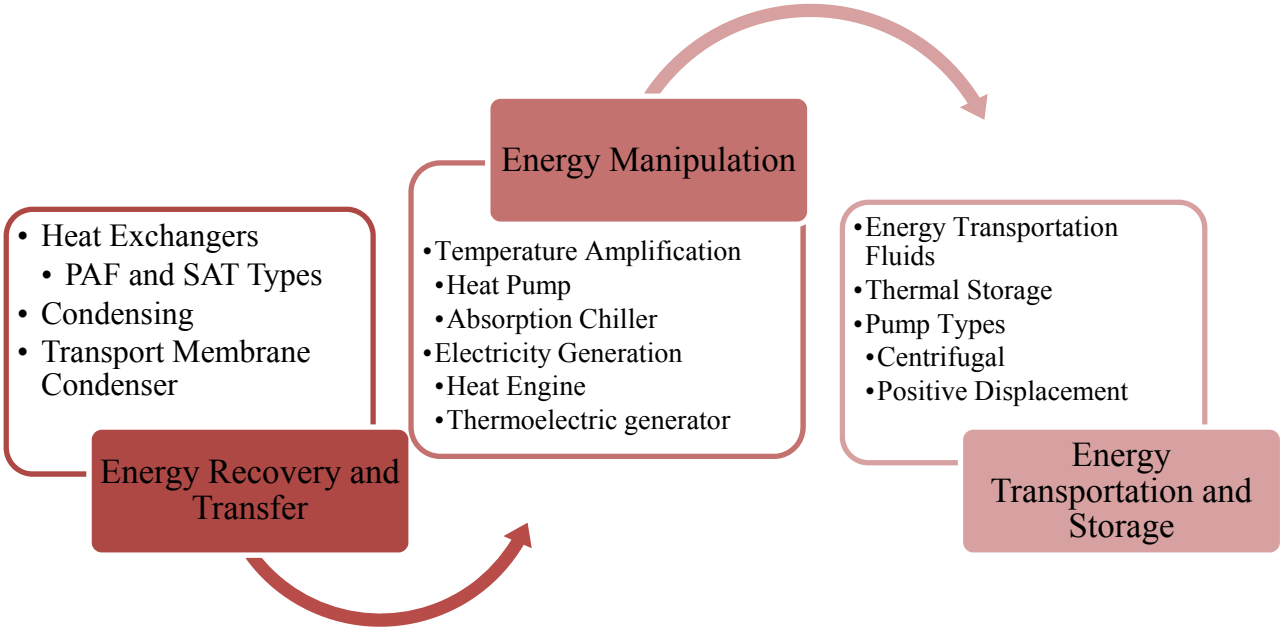

Figure 5. System Components Flow Chart.

The first portion of the chapter focuses on the equipment used to recover the energy and describes different types of heat exchangers and their preferred uses and advantages. There are four types of heat exchange equipment discussed, the first is the shell-and-tube heat exchanger followed by the plate-and-frame heat exchanger. These two heat exchangers are then compared for different applications and their uses along with when a certain type should be chosen over the other are discussed. The next two types are concerned with condensing water vapor from the exhaust and capturing its associated energy. This can be accomplished with either a condensing economizer or with a transport membrane condenser. The two types of technologies are

discussed along with the underlying goal behind condensing this water vapor from the exhaust stream.

The second section of this chapter focuses on methods of amplifying the waste heat temperature. This amplification takes in the waste heat and adds additional energy to the system which, in turn, increases the overall temperature of the waste heat stream. This can be accomplished by both a mechanical heat pump and an absorption heat pump. The absorption heat pump can be used for not only this increase in temperature of the waste heat but also to provide a cooling effect which can be used in space conditioning applications. The specific applications for which each of these types are recommended are listed and general performance efficiencies are shown.

The third portion of the chapter discusses different methods of converting the waste heat energy directly into electricity. This method is used when there is no or little direct use for a hot liquid nearby. The major methods discussed include power generation using an Organic Rankine Cycle, thermoelectric generators, thermophotovoltaics and a modified thermalphotovoltaic. The Organic Rankine Cycle uses the classic Rankine cycle and substitutes water with an organic fluid as the working fluid. This process uses waste heat to vaporize the working fluid and delivers it through a turbine which is connected to a generator to produce electricity. The thermoelectric generator uses the same concept as a thermocouple and creates a current flow by the temperature difference caused by waste heat flowing over a junction of dissimilar metals. The thermophotovoltaic operates similar to a solar PV cell however the energy source is switched from the sun to waste heat. This system design limits the heat transfer necessary to create an electrical current to that of radiation which sets the overall maximum efficiency of the system to that of a blackbody. More recently, modified thermophotovoltaic have been developed which eliminates efficiency limit problem by the addition of a converter which allows Coulomb

interaction within the system to occur. These units have been seen to achieve efficiencies much greater than that of a blackbody, however are currently only available on the millivolt scale.

The next portion of the chapter focuses on energy transportation and storage methods in the system.. The most important properties of the fluids discussed include the maximum operating temperatures and the volumetric heat capacitance. The fluids analyzed and compared with one another include water, triethylne and tetraethylene glycols both as a standalone fluid and mixed with water, DOWTHERM G, UCON HTF 500 and Therminol XP. Figure 26 and Figure 28 located in the *Energy Transportation Fluid Comparison* section shows these types of comparisons. Another type of transportation method that is discussed is the addition of microencapsulated phase change material to another fluid to increase its overall heat transportation capabilities. This method is not currently applicable for waste heat recovery systems but the concept shows promise for future applications once certain problems have been solved. Although microencapsulation of these phase change materials is not available, phase change materials can currently be used in a stationary thermal storage tank to capture energy in the latent heat of the material.

The final segment of this section is about pumping systems used in transporting the previously discussed fluids. These pumps include centrifugal and positive displacement types. The centrifugal type pumps are more common and recommended for most applications; however, the positive displacement pumps are capable of moving higher viscosity fluids. For either of the pump types the electric motors used to operate the pump are generally not fully loaded continuously throughout the year. To reduce energy consumption, variable speed drives are discussed as a method of conserving energy when idling the motor back. These systems operate by utilizing pulse width modulation which sends pulses of current to the motor to operate the

motor at a lower rotational speed. With this option implemented, the energy savings possibilities of 30% to 50% can be seen [20].

Heat Exchangers

Heat exchangers are a critical component in a waste heat recovery system and are the way that the energy is recovered from the stack gas and transferred to a useable media. The second law of thermodynamics states that heat can flow from one body to another when there is a temperature difference between the two and the energy will travel from higher temperature to lower. This shows that it is possible to recover the energy from the hot exhaust gas, however physical limitations to the heat exchanger put a limit onto how much can be recovered. The two main parameters that affect the amount of heat that can be recovered are the temperature difference and the surface area which connect the two fluids. Because the temperature differences between the two fluids are relatively fixed, to achieve more heat transfer the surface area of heat transfer must be increased. There are many commercially available types of heat exchangers that achieve this, however not all are practical or effective for waste heat recovery from exhaust gasses. The two main types that are used in this application are a shell-and-tube heat exchanger and a plate-and-frame heat exchanger.

Shell-and-tube

The shell-and-tube heat exchanger (SAT) is the most common type of heat exchanger used in industrial applications mainly because of its relatively low upfront cost, easy maintainability and availability. They are also capable of operating at a very wide range of pressures that can be accommodated because of well-established manufacturing techniques with precision metal tubes.

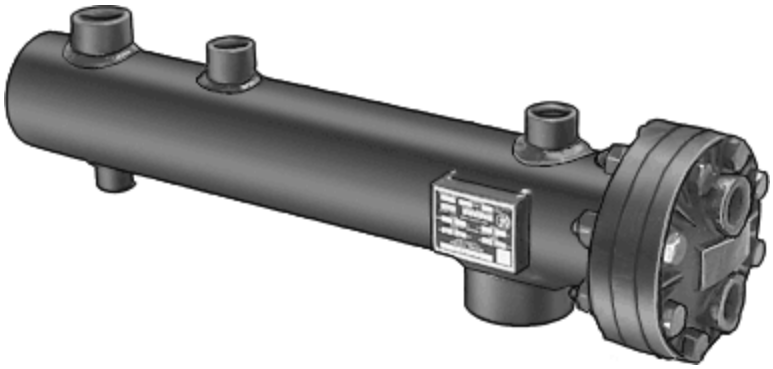

Figure 6. Shell-and-Tube Heat Exchanger [42].

This type of heat exchanger can be used for both air-to-liquid and liquid-to-liquid and sometimes air-to-air heat exchange. Air-to-liquid types are configured with the fluid being pumped through the tubes with very high values of convective heat transfer coefficients. The air is then forced over the outside of the tubes in the shell with much lower heat transfer coefficients. To make up for the small heat transfer rates, fins can be added to increase the available heat transfer surface area, although these are rarely installed. The tubes can be bundled in a variety of ways depending on the application and generally contain baffles to direct the flow of the fluid within the shell. These baffles are simply a disk in the flow path with a segment removed. These baffles can also be placed in a number of configurations but are generally placed in a segmental form with the cut edges alternating from top to bottom to force the fluid to move perpendicularly back and forth across the tubes [71]. Not only do these baffles channel the flow of the fluid in the shell, they also help support the tubes and restrain tube vibrations from fluid impingement [11]. This baffle setup can be seen in each of the following schematics of shell-and-tube heat exchangers.

The first configuration of tubes in the heat exchanger to be considered is called the *fixed tube sheet design* which has both tube sheets attached directly to the shell. This design is simple and most economical for low pressure and small temperature difference designs but should not be

considered when frequent shell side cleaning is required [11]. This design allows the heat exchanger to have a high degree of safety on mixing of the two fluids. This characteristic can be extremely beneficial if cross contamination of the two flows are not acceptable. The configuration also provides the opportunity for mechanical tube side cleaning where the heat exchanger can be disassembled and physically cleaned to increase the heat transfer rates. However the design does contain certain limitations including only having the capability for a small temperature difference between the working fluids without the addition of expansion joints. Other limitations are that only low pressure applications can be used, and the tubes are fixed inside the shell meaning that shell side cleaning is not possible without the use of chemicals [27]. The following figure demonstrates the configuration of this type of shell-in-tube heat exchanger.

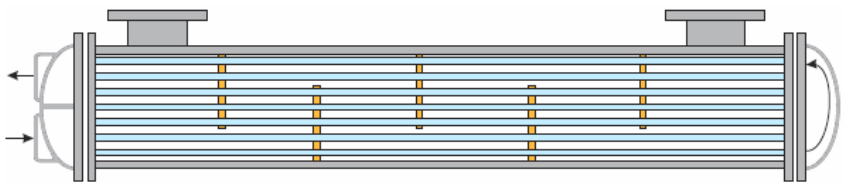

Figure 7. Fixed Tube sheet, 2-Pass Shell-and-tube Heat Exchanger [72].

The second configuration considered is a *U-tube design* which contains a single set of tubes in a "U" shape located inside the shell. This configuration is the simplest design and, because the fluid enters and exits from the same side, requires only one seal and joint compared to two in other designs. As a result of style, there are inherently less leaking points in the shell side and less concern with thermal expansion. This configuration is a very useful when high tube side pressure is required and also allows for easy access to maintenance on the shell side. However in the case of a failure of an individual tube, the entire bundle must be replaced due to access problems with tubes not located on the outer perimeter [11]. This access problem also leads to

only chemical cleaning to be possible on the tube side [27]. It should be noted that the bended surface area of the tubes are much less effective at transferring heat to the secondary fluid and are ignored when determining an overall effectiveness for the heat exchanger [11]. The following figure shows a shell-and-tube heat exchanger with a U-tube design.

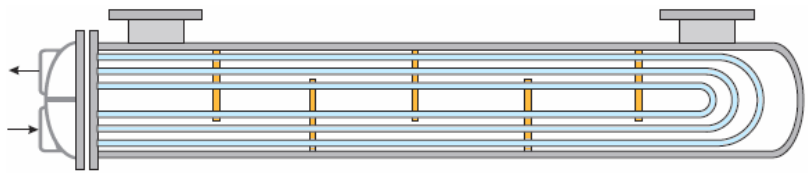

Figure 8. Diagram of U-tube Shell-and-tube Heat Exchanger [72].

A third type of shell-and-tube heat exchanger design is called a *pull-through floating head design* which consists of a fixed tube sheet at the end of the flow channel and a floating tube sheet at the opposite end. This setup allows for the tube bundle, along with individual tubes to be easily removed from the shell by only disassembling one side allowing for easy physical cleaning of both the shell and the tube. The floating head in this heat exchanger acts as a differential expansion, allowing for a greater temperature difference capacity. This floating head does sacrifice the ability to have as many tubes for a given shell size when compared to other designs because the floating head flange and bolt design require a large amount of area to work properly [11]. Another issue with this design is that, in addition to having less heat transfer surface area, the design contains fewer tubes. This results in a larger annular space between the tubes and the shell which leads to less flow of the shell side fluid coming into contact with the tubes, resulting in less overall heat transfer. To help combat this problem, sealing strips or dummy tubes can be added in the shell to prevent fluid bypass and in turn increase the heat transfer [72]. The design also has the potential for continuous, undetected leaks because the seal is located inside and not

visible from the outside [27]. The following figure shows the basic setup for this type of shell-and-tube heat exchanger and describes all its components.

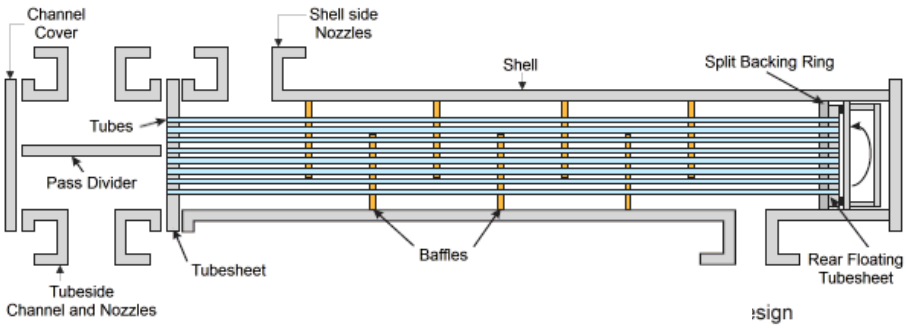

Figure 9. Diagram of Pull-Through Floating Head Shell-and-tube Heat Exchanger [72].

Yet another type of configuration of the shell-and-tube heat exchanger is known as a *split-ring floating head design.* This configuration consists of a fixed tube sheet at the channel end and a floating tube which is located between a split ring and a separate cover. This type is very similar to the *pull-through floating head design* discussed previously however this design allows for additional tubes to be added back into the effective usefulness of the heat exchanger [5]. This addition in effectiveness does add some complexity to the setup of the heat exchanger itself and to remove the tube bundle from the shell, both the front and rear covers along with the floating tube sheet covers must be removed. This addition in complexity increases the time required for maintenance on the heat exchanger, although cleaning is possible, making this design much more desirable if bundle repair is to be done in the field [11]. A form of this configuration is shown in the figure below.

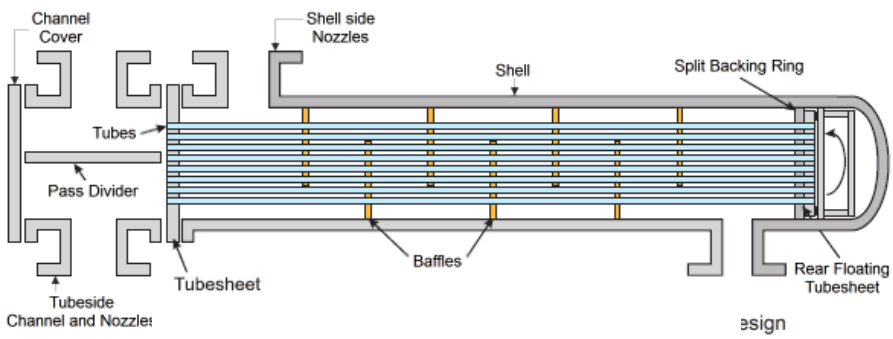

Figure 10. Diagram of Split-Ring Floating Head Shell-and-tube Heat Exchanger [72].

Finally the last configuration of shell-and-tube heat exchangers considered is the *outside packed floating head design.* This design itself has two variations, one being a lantern-ring type and the other being a stuffing-box type. The lantern-ring type has a floating head which slides against a lantern-ring packing that is compressed between the shell's flange and cover [11]. This floating head acts as an expansion joint and eliminates the need for an additional expansion joint to be added. Also because the individual tubes and tube bundles can be removed, both the tube side and shell side can be mechanically cleaned. The design limitations are a maximum temperature possible of 375°F and a maximum pressure of 300 psi setting a narrow band on its range of usefulness [27]. The second setup, a stuffing-box type, differs from the lantern-ring type in that the seal is attached against an extension of the floating tube and the tube sheet cover is attached to the tube sheet extension by a split-ring [11]. This design is able to operate at a high pressure on the tube size but limited to about 150 psi on the shell side and a maximum temperature of about 300°F. Like the lantern-ring type, both individual tubes and the whole bundle are able to be removed, making cleaning of both the shell and tube side possible. However, both of these configurations are susceptible to leaking to the atmosphere compared to other designs and therefore should never be used for hydrocarbons or toxic fluids [27]. The following figure shows the setup and flow patterns of both designs.

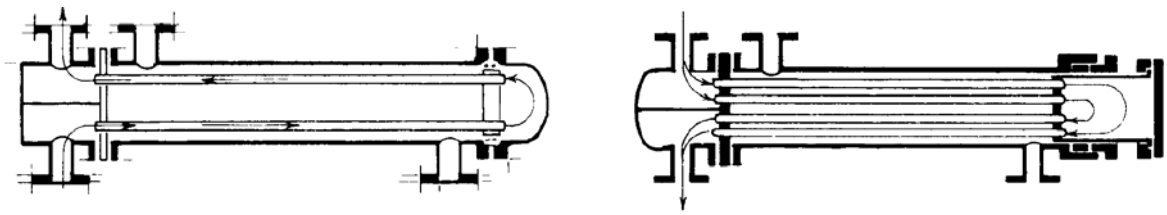

Figure 11. Outside Packed Lantern Ring Floating Head Exchanger (Left) and Outside Packed Stuffing Box Floating Head Exchanger (Right) [5].

The previous discussion was centered around the tube configurations in different types of shell-and-tube heat exchangers but the shell side of the heat exchanger can also have different configurations. These types of configurations include single pass, two pass and divided flow. The single pass shell is the most common type and occurs when the shell side inlet and outlet are physically located on opposite sides of the heat exchanger. This setup has the fluid flow enter on one side, pass through a series of baffles and across the tubes until the fluid makes its way to the opposite side, where it then exits the heat exchanger. The two pass shell has both the inlet and exit on the same side and requires a longitudinal baffle running down the center of the shell. Because of pressure drop and thermal stress and heat leakage in the baffle, this setup should only be used when a shell side pressure drop is a maximum of 10 psi and when the shell side temperature is less than 380°F. This system operates by the inlet fluid entering the heat exchanger and flowing through a series of baffles, across tubes on the top half until it reaches the opposite side. The flow is then directed down and back the opposite direction across another set of baffles and across the tubes on the lower half until it is exhausted below the inlet. The third type of shell design is the divided flow shell. This configuration has a central inlet nozzle and an outlet nozzle on each end, or vice versa. This type of shell configuration is typically used in condensing heat exchangers because it requires a smaller pressure drop [11].

The design of a shell-and-tube heat exchanger leads to the concern of the tubing as well as the shell fouling and in turn reducing the heat transfer between the two flows. The term fouling includes any type of deposit that appears on the heat transfer surface of the heat exchanger. The most significant types of fouling are sedimentation fouling, which occurs when suspended solids in the fluid settle out and stick to the heat transfer surface. The second type is inverse solubility fouling which happens when certain salts are less soluble in warm water than in cool water which causes the salts to leave the flow and crystallize on the heat transfer surface. The next type is chemical reaction fouling and is the result of a chemical reaction occurring in the fluid stream due to an increase in temperature and results in a solid forming on the surface. This fouling can be very difficult to remove and in extreme cases can require the deposits to be burnt off to restore the original heat transfer capabilities. Corrosion product fouling is yet another common type which results from the corrosion of heat exchanger parts that are installed. The last type of fouling common to shell-and-tube heat exchangers is biological fouling which is when an organism attaches itself to a heat transfer surface, such as algae, and causes a barrier between the fluid and the heat transfer surface which results in much lower heat transfer effectiveness [5].

These types of fouling can be slowed by the use of different materials when constructing the heat exchanger as well as intermittently adding certain chemicals to the process. Both of these measures are not permanent and eventually the heat exchanger will need to be physically cleaned. As mentioned earlier in this section, methods and frequency of cleaning shell-and-tube heat exchangers will vary greatly depending on the configuration of both the shell and the tube sides. The major techniques of mechanical cleaning include using a scraping or rotary brush to physically clean the inside and outside of the tubes as well as the shell or by using high velocity

water jets to pressure wash the heat exchanger. When these methods are not possible chemical cleaning must be used to remove the fouling particles and increase the overall heat transfer [5].

Plate-and-frame

The plate type heat exchanger is less commonly used in industrial applications however, because of certain characteristics it possesses, this type can be a much better choice in certain applications. These applications are generally limited to liquid-to-liquid heat transfer and not air-to-liquid which provides a substantial weakness in initial waste heat recovery from the exhaust stream.

The plate-and-frame (PAF) heat exchanger is made up of several pressed plates that are aligned on a frame and secured together. Gaskets are used on the perimeter of each plate and are set in a groove which seals the plates by a compressive load when pressed against the adjacent plate preventing leaking of the fluid. These gaskets can be made from several different materials depending on temperature and application requirements [37]. The hot and cold fluid flows in the pattern shown below within the cutout grooves in each of the plates with each plate alternating from the hot to the cold fluid. This flow pattern sandwiches a cold flow between two hot fluids creating a large heat transfer area while being more compact than the shell-and-tube type heat exchanger. This flow pattern is achieved by selectively removing a piece of the perimeter gasket that separates the plates, allowing for the fluid

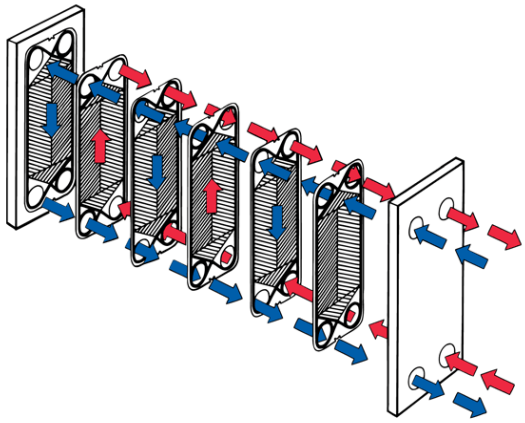

Figure 12. Flow Configuration in a Plate-and-frame Heat Exchanger [33].

to flow through the appropriate plate. Without a separate safety feature, this design has the potential to cross contaminate the streams in the event of a gasket failure in any of the plates. To prevent this from occurring, leakage grooves are installed into the gasket near the headers giving a backup system which allows a leak to flow to the atmosphere making cross contaminating virtually impossible [71]. After the flow is introduced into the plates, it follows the paths laid out by the flow passages which are built into the plate design itself. The flow paths contain a corrugated surface pattern which presses together with the next plate and cause narrow interrupted and convoluted flow paths. These corrugations not only create the fluid flow paths but also provide structural support to the plates and increase heat transfer rates and lower fouling resistance by creating turbulent flow in the fluid. This turbulence results in heat transfer coefficients as high as two to three times that of a comparable shell-and-tube heat exchanger under similar applications [36].

This system setup allows for a very effective and compact heat exchanger providing an approach temperature of as little as 2°F. The approach temperature is simply the difference in the outgoing hot fluid temperature and the incoming cold fluid temperature when a counter-flow

configuration is used. The setup of the plate-and-frame heat exchanger allows for additional plates to be added or to be removed at any time in response to changes in the process or temperature profile. Along the same lines, this characteristic provides easy maintainability of the plates because the plates can be easily added or removed, they can be cleaned in place and with a relatively small effort. Also the maintenance required for the plates are reduced because they are normally made from alloys or exotic metals which naturally have a high corrosion resistance [11].

As with all heat exchangers and mechanical equipment, maintenance issues become a major concern for PAF heat exchangers. The major source of maintenance stemming from the plate-and-frame heat exchanger is from fouling or scaling of unwanted material on the plates themselves. Fouling can occur due to sedimentation, crystallization, organic or biological growths, corrosion products or a combination of these. This fouling not only causes less heat transfer to be possible, it also increases the fluid flow resistance and if the heat flux is relatively high in a concentrated area, can lead to local hot spots and ultimately mechanical failure [36]. This problem can be dealt with by a number of methods but the most commonly used includes an upfront conservative design. This method supplies more heat transfer surface area than necessary so the heat exchanger can still effectively transfer the heat after a certain amount of fouling. However this method frequently causes a decrease in fluid velocity in the flow regime of the plate which creates stagnation zones. This, along with deposit formation, creates a random fouling growth rate. While a conservative design can be done before the heat exchanger is installed, it is not a permanent solution because over time the plates will still begin to foul and the effectiveness will diminish beneath the required level. To overcome this either chemical or mechanical cleaning of the individual plates must still be performed [75].

To help determine the frequency that the heat exchanger will have to undergo maintenance, several fouling growth models have been established. These models include linear, power-law, falling-rate and asymptotic fouling growth rate models and all attempt to predict the time-dependent behavior of most fouling phenomena. The linear fouling model is associated with predicting the impact from crystallization of well-formed deposits consisting of mostly pure salt. The power law fouling model investigates the deposition of $CaCO_3$ under relatively high temperature operations along with a representation of corrosion fouling data. The falling-rate fouling model normally occurs when the deposition rate is always greater than the removal rate and consists of modeling particulate fouling, but also includes some forms of crystallization fouling. Finally the asymptotic fouling model is generally observed in cooling water heat exchangers because of the formation of a scale layer of a weak, less coherent structure. These structures are associated with the simultaneous crystallization of salts of different shapes or with the presence of suspended particles embedded in the crystalline structure [75]. These models are still generic and depend on a considerable amount of assumptions and gross estimations of a random, time dependent process. These methods along with in-depth experimental data has shown that the fouling resistances for plate-and-frame heat exchanger are about 15 to 30% of that for a shell-and-tube heat exchanger when examined under similar conditions [76].

Although it is difficult to predict the exact timing that the heat exchanger must be cleaned, Zubair and Shah have developed a reliability-based and a cost-based cleaning strategy [76]. Both methods undergo the same cleaning scheme of one or more of the following maintenance items:

- The heat exchanger unit is hydro-tested to identify the damaged or leaking plates
- Open the unit to clean and replace damaged plate(s)

- All gaskets are thoroughly inspected for signs of aging, cracking or chemical attach
- Any plates that have cracks or show signs of corrosion, deformation from swollen gaskets or over-tightening are replaced
- Damaged plates are removed in pairs to maintain the critical plate pattern in the unit
- After washing, the plates may be sent to a plate-and-frame heat exchanger service center for reconditioning (including chemical cleaning, if needed) and dye penetrant inspection, and a failure analysis on damaged plates or gaskets.

The reliability-based strategy is based on the fouling models discussed earlier and an additional risk level which represents the probability of plates being fouled to a critical level. The authors present equations that give a good idea of how often the heat exchangers should be cleaned based on different assumptions of the quality of cleaning or breaks in operational time where the maintenance is possible. These equations use known statistics of heat exchangers and use a probability function to determine this frequency. The first case assumes that the cleaning is perfect and the heat exchanger is returned to its original performance level while the second case assumes a lowering of the performance based on the original due to imperfect maintenance. The third case assumes that the heat exchanger will be under a scheduled maintenance shutdown periodically and increases the risk level accordingly to account for the window of possible cleaning.

The cost-based cleaning strategy uses the same fouling models as before to predict a median time to reach a critical level when the heat exchanger will quit functioning properly and what that level of fouling will be. This model includes the cost per hour of the off-line cleaning, the cost of chemical cleaning and the cost of additional fuel consumption based on the loss of heat

exchanger effectiveness. All of these methods are discussed further and the equations are given by Zubair and Shah [76].

Heat Exchanger Comparison

While both types of heat exchangers are accepted in heat recovery applications, each has their advantages and disadvantages. The shell-and-tube heat exchanger is a very widely manufactured, used and a well-known heat exchanger. This leads to a certain confidence in the equipment. The shell-in-tube heat exchanger introduces very little pressure drop into the system and can be used at higher pressures and temperatures, depending on the configuration selected. This proven technology is, however, outperformed by the plate-and-frame heat exchanger in many different areas. Table 1 was created by Cheremisinoff & Cheremisinoff [11] and shows a comparison of a plate-and-frame heat exchanger and a shell-and-tube heat exchanger under the same design circumstances.

Table 1 demonstrates a relative comparison between a shell-and-tube heat exchanger and that under similar operating conditions a plate-and-frame heat exchanger has a "U" value or overall heat transfer coefficient of three to five times that of a shell-and-tube. This can be attributed to true countercurrent flow and the creation of turbulent flow which causes the heat transfer to greatly enhance while also lowering plate fouling. The plate-and-frame heat exchanger is also much more compact compared to a SAT heat exchanger and leads to less overall weight, hold up volume and less installation requirements. This comes from less total volume required to achieve the same amount of heat transfer where a reduction in surface area is made up for by the increased "U" value as mentioned before. This more compact size also provides the PAF heat exchanger with a lower initial cost when made from stainless steel or higher grade materials due

Table 1. Comparison of Plate-and-frame And Shell-and-tube Heat Exchangers [11]

Item	Plate & Frame	Shell-and-tube
Efficiency	High - "U" Value three to five times greater	Low
Space Required	10% to 50%	Twice as much to pull tube bundle
Ease of disassembly	Easy - Loosen bolts	Complex - tube bundle must be pulled
Costs	Less when stainless steel or higher grade material is required	Higher, except in all carbon steel construction
Fouling	Low due to corrugations an inherent turbulence	High due to circular cross-sectioning and channeling
Heat transfer surface	Plates easily added or subtracted	Fixed surface only
Weight and installation	Low- No concrete pad required	High - concrete pads normally required
Intermix between fluids	Impossible due to gasket design	Can mix, both at welds and at tube sheet
Inspection	Disassemble and inspect	Difficult - must normally pull tube bundle
Chemical Cleaning	Excellent due to corrugations/channels	Satisfactory but must be cautious of dead spots
Maximum viscosity	30,000 cps nominal	10,000 cps
Pressure Drop	Low to medium	Low
Heat Loss	Practically none - no insulation required	Great amount - insulation required
Temperature approach	Can be designed for a 2 degree F approach with more than 90% heat recovery attainable	5 degree to 10 degree minimum approach required
Design sizing	Computer custom-designed per application	Must always oversize to be safe
Hold-up volume	Low	Very high
Operations	Multiple duties possible with connecting plates	One unit required for each duty

to the decrease in total metal required, even when the SAT heat exchanger is more widely and cheaply manufactured. The simple design of the PAF heat exchanger and ease of disassembly easily allows for additional plates to be added or extra plates to be removed to respond to future process changes whereas the SAT heat exchanger is fixed at the manufactured size and cannot be easily expanded and requires substantial effort to disassemble.

The fouling rate is a big concern for the longevity of the product and decreased maintenance costs. The PAF heat exchanger is able to avoid the majority of this fouling due to its plate design and induced turbulence in the flow where the SAT heat exchanger has a much higher fouling rate because of its circular cross section and channeling. A study was performed by Heat Transfer Research Inc. which carried out experiments about comparative fouling rates and found that PAF heat exchangers to foul at a rate of one-quarter that of the TEMA recommended rates as well as the experimentally found cleaning rates for in SAT heat exchangers. This decrease in fouling greatly reduces the associated maintenance responsibilities that are incurred in SAT types [71].

The effectiveness of chemical cleaning is also greater in the PAF heat exchanger, leading to even less required shut-down cleaning. Other maintenance issues that arise from the SAT heat exchanger are that to inspect the working parts, the entire assembly must be removed. This increases the possibility of a complete failure occurring before the replacement of failed components. The SAT heat exchanger also has a much larger amount of heat loss to the atmosphere due to convection and radiation than does the PAF type, to reduce this heat loss the addition of insulation is required [11].

Other advantages of the plate-and-frame heat exchanger include less clearance space around the exchanger is required for maintenance considerations because of how it assembles and

disassembles. Another way the PAF heat exchanger can decrease the overall energy consumption of a facility is that the increased effectiveness of this heat exchanger over the SAT increases the temperature rise in the cold fluid which in turn reduces energy consumed by pumps to transport the fluids [71]. Also a very high number of transfer units are obtainable in a single pass and because there is less piping, it consists of very minimal entry and exit losses as well as pipe-work to be dealt with during assembly or disassembly [37].

A major concern that comes along with a plate-and-frame heat exchanger is the gaskets that seal the plates together. These gaskets are normally elastomeric and have a maximum temperature of about 300°F to 350°F and a maximum pressure of 400 psi [71] although newer gaskets made from a graphite base have a maximum temperature range of up to about 480°F [19]. The gaskets also have the potential to provide leaking points in the system if not properly installed or re-used after disassembly. Because these gaskets are recommended to be replaced after each disassembly to prevent leaking, the material cost and maintenance hours are increased. Another major limitation for the PAF heat exchanger is that it is generally only used for liquid to liquid heat exchange and not typically for gas to liquid [19]. This limitation causes the shell-and-tube heat exchanger to become an integral part in the initial stack gas energy recovery system whereas the plate-and-frame heat exchanger can be used for the subsequent applications.

Condensing Heat Exchangers

A condensing heat exchanger is useful in heat recovery applications because it is not only able to reduce the exiting exhaust gas temperature and recover the sensible energy, it is also able to condense the water vapor from the exhaust gas. This provides the possibility of recovering the latent heat energy stored in the exhaust by turning the water vapor in the exhaust back into water. The following schematic shows the basic layout of the heat exchanger and functionality. It

works by introducing a heat exchanger into the stack gas exit with a recovery fluid on one side of the exchanger to absorb the energy of the stack gas before it is exhausted to the atmosphere. This economizer is generally installed as a second heat exchanger in the exhaust stack and is only required to operate in a lower temperature range, potentially reducing the manufacturing cost.

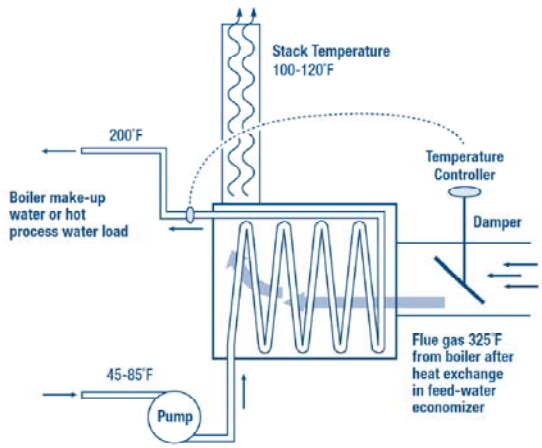

Figure 13. Condensing Heat Exchanger for Heating Boiler Feed Water [64].

The water vapor in the exhaust stream mentioned earlier is produced during the combustion process when hydrocarbons combine with oxygen. The most notable example of this is during the combustion of natural gas (CH_4). The following chemical equation is a very simplified representation of what occurs during natural gas combustion [63].

$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow 2H_2O + CO_2 + 7.52N_2$$

As this equation shows, for every one molecule of natural gas burned, two molecules of H_2O are produced and because this is a combustion process, temperatures are well above the boiling point of water. This produced water is immediately converted into steam and this change in phase of the liquid water to vapor consumes some of the combustion energy, about 970 Btu/lbm assuming

atmospheric pressure [63]. This is known as the latent heat of vaporization for water. From a simple chemistry analysis of this process, it can be seen that for every one pound of methane combusted, 2.25 pounds of water vapor are produced which works out to be about 12% of the total exhaust content by weight.

To express this in terms of energy lost simply by converting the produced H_2O into steam, based on the higher heating value of natural gas it requires the combustion of about 42 pounds mass of CH_4 to produce one million Btu's of energy. So for every one MMBtu of natural gas combusted 94.3 pounds of water is produced. This value, combined with the latent heat of vaporization of water, implies that for every one MMBtu of natural gas combusted, 91,495 Btus are consumed to convert the water into steam. This represents almost a tenth of the available energy of the natural gas [63].

As the previous simple analysis has shown, the ability to condense this vapor back into water and recover its energy can greatly increase the efficiency of the heating process and reduce the amount of energy wasted in the process. The effects of this production of water vapor are seen when other types of fuels are used that contain hydrogen, however most are not as significant as natural gas combustion. For example, when combusting one MMBtu of number 2 fuel oil only about 56 pounds mass of water is produced.

The system shown in the following figure demonstrates the amount of energy that can be recovered in a system containing a condensing heat exchanger and an initial heat exchanger to recover sensible heat. The system recovery efficiency is also calculated when only a condensing heat exchanger is installed.

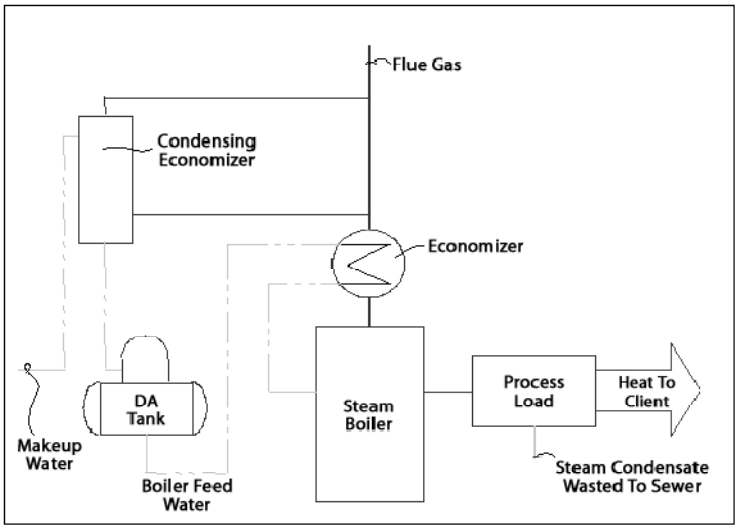

Figure 14. Heat Recovery Potential with a Condensing economizer System [9].

The example system consists of a boiler which is providing steam to be used as process heat for the client, this boiler produces 6,000 pounds of 100 psig saturated steam per hour. The exhaust gas temperature directly from the boiler is 530°F and leaves the initial economizer at a temperature of 320°F. The flow is then directed to either exhaust into the atmosphere or pass through the condensing economizer. This flow is split with 51% going to the condensing economizer and the remaining 49% exiting the system to the atmosphere [9].

With this described system, about 1.74 MMBtu/hr of energy is exhausted from the boiler in the form of waste heat. This heat is then passed through the sensible heat exchanger and about 0.43 MMBtu/hr can be recovered, or about 25% of the waste heat energy. The flow is then split between the condensing economizer and exhausting to the atmosphere. The condensing economizer is capable of recovering an additional 0.39 MMBtu/hr which means that with both heat exchangers installed, 0.82 MMbtu/hr can be recovered which is 47% of the total wasted energy. If the sensible heat exchanger is removed from the system, the condensing economizer by itself is only able to recover 0.44 MMBtu/hr which is just slightly more than the sensible heat

exchanger recovers when both are present [9]. This is because the condensing economizer works by lowering the exhaust gas below its dew point temperature and with such a high entering temperature of 530°F and a fixed surface area, the heat exchanger is not capable of condensing the steam in the exhaust, turning the condensing economizer into nothing more than a regular heat exchanger.

While the condensing economizer is capable of recovering large quantities of low temperature energy, there is a reason why this is not a widely used method of energy recovery. By cooling the exhaust gas stream below the dew point temperature, the water will condense releasing its latent heat but also corrosive materials contained in the water that can foul the heat exchanger used. Depending on the type of fuel which is being used, corrosive materials such as CO_2, NO_x, SO_x, unoxidized organics, minerals and the water itself are present. This causes heat exchangers constructed from ordinary materials to quickly foul and become ineffective at transferring heat energy. To prevent this occurrence and to be able to recover the latent heat in the exhaust gas stream, condensing heat exchangers can be made from certain advanced alloys and composite materials. This, however, significantly increases the implementation cost of such heat exchangers and is not economically feasible if a use for the relatively low temperature energy cannot be found [4].

Condensing heat exchangers can be configured as a shell-and-tube or plate-and-frame heat exchanger but the vast majorities are shell-and-tube. This is because the plate-and-frame types are generally used for liquid to liquid heat exchange where condensing does not take place [19]. Along with creating the heat exchanger from advanced alloys and composite materials, typical system designs can include carbon steel tubes with a throwaway section at the cold end. This throwaway section is designed to take to brunt of the corrosive elements and when it becomes no

longer effective the section of the heat exchanger can be disposed of and cheaply replaced. Another typical construction is to have a standard heat exchanger with stainless steel tubes or plates that are able to withstand the corrosive elements. This heat exchanger, with carbon steel tubes for the majority of the heat exchanger and stainless steel tubes for the cold end, contains this stainless steel end would take the place of the throwaway section described earlier. Finally using nonmetallic tubes in a heat exchanger can be used. This material can be glass or Teflon but both have large limitations in feasibility [24]. The shell-and-tube condensing heat exchanger is typically configured to route the condensing, low temperature fluid, in the tubes with the hot gas flowing through the shell [71].

Transport Membrane Condenser

Because of the relatively low temperatures that are required to condense the water vapor from the exhaust gas, large surface areas are required for typical condensing heat exchangers to be effective. The transport membrane condenser has been developed as a more compact alternative. The transport membrane condenser operates by removing the water vapor from the exhaust gas by selectively extracting the low pressure water vapor into small pores in the nanoporous ceramic membrane, in liquid form, through pore capillary condensation and also through the cooling effect of cold water on the shell side of the condenser. Because the water is recovered by transport through a membrane, the water recovered is pure and does not contain many of the impurities that can be found in condensed water from a traditional condensing heat exchanger. This property makes this system a perfect candidate to be used when preheating feed water in a boiler system or to replace a hot water generator. The following figure shows this basic process when configured into a boiler system [40]. This system is not limited to only boiler systems and can be used for any type of combustion process where the latent heat from water vapor is desired

to be recovered. However because of how the system works and current technology limitations, the transport membrane is currently limited to clean exhaust streams such as that from natural gas combustion [4].

A typical system operates in series with a traditional heat exchanger which is able to capture the majority of the sensible heat before the exhaust gas reaches the TMC [3]. After the exhaust gas is passed through the heat exchanger it enters the transport membrane condenser as shown in Figure 14.

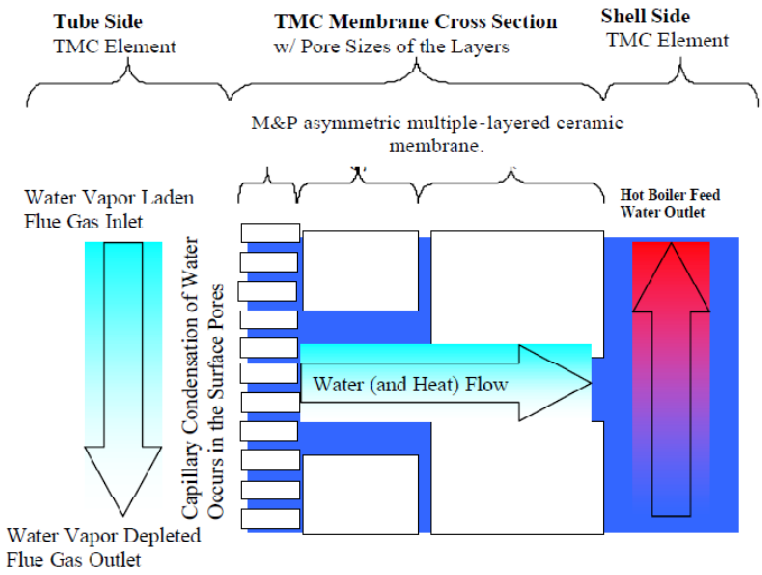

Figure 15. The Transport Membrane Condenser Concept [40].

The exhaust gas is passed through a series of tubes and the water vapor is condensed in the pores of the membrane surface. This condensation occurs from either a cooling effect from the shell side of the condenser or through capillary condensation. The latter of the two is very important because it allows for condensing to occur even when the pressure of the exhaust gas is above the condensing pressure, allowing for condensation to occur above the gas's dew point temperature, an effect which is impossible in normal condensing economizers. The reclaimed water is pulled

away from the tube surfaces by a slight negative pressure which comes from the shell side of the condenser. This water can be reclaimed into a boiler as warm feed water or if a system other than a boiler is used, this energy can be recovered through a separate liquid to gas or liquid to liquid heat exchanger to be used for various applications [3].

The transport membrane condenser also possesses enhanced heat transfer attributes. The first major advantage is the introduction of convection heat transfer which tends to be a much larger factor in overall heat transfer than conduction. The second major advantage of the transport membrane condenser is that the condensed water is sent directly into the unit where less heat energy is lost to the ambient. Finally the transport membrane condenser always exhausts the stack gasses at a temperature below their dew point, less than 125°F, with basically no water vapor present. This characteristic prevents the stack from corroding and reduces the risk of water vapor condensing in the stack [40].

This technology is a relatively new concept for waste heat recovery and is currently only manufactured by one company who has gained exclusive rights to the technology [8]. This lack of competition and new technology aspect will inevitably increase the implementation cost. Although analysis conducted by Liu [40] claims that based on their economic analysis a simple payback period of less than one year is achievable. This product is currently available for sale in the 200 to 400 boiler horsepower range and sizes are expected to be available for 400 to 800 boiler horsepower sometime within the year 2011 [7].

Heat Pumps

Heat pumps can be used in heat recovery applications to increase the temperature of the recovered fluid by adding additional work input. This increased temperature can be

accomplished in one of two methods. The first method takes advantage of the increased condensing temperature of the working fluid as the pressure is increased. The additional work input is used to compress this vapor, allowing it to be condensed at its higher temperature. This method utilizes mechanical power input and is therefore referred to as a mechanical type heat pump. The second method utilizes chemical reactions to create a temperature change while using the waste heat to separate the chemicals so the process can be repeated. This type of chemical process is known as absorption and operates in an absorption heat pump.

Mechanical Type

To recover the majority of the waste heat available, it must involve the recovery of energy from low temperature streams. While the exhaust stream contains a large amount of recoverable energy, the uses for such low temperature working fluids can sometimes be difficult to locate. In these situations a heat pump can be used to "upgrade" this energy so that is can be used for applications that require higher temperatures. A heat pump receives the waste heat from the process which is used in an evaporator to vaporize the heat pump working fluid. A compressor is then used to increase the pressure of this working fluid which also increases the condensing temperature. The working fluid is then sent to a condenser which then condenses it at this new, higher temperature. This heat gain is then used in the new heat recovery process stream [43]. This process then repeats itself, and is shown in the following figure.

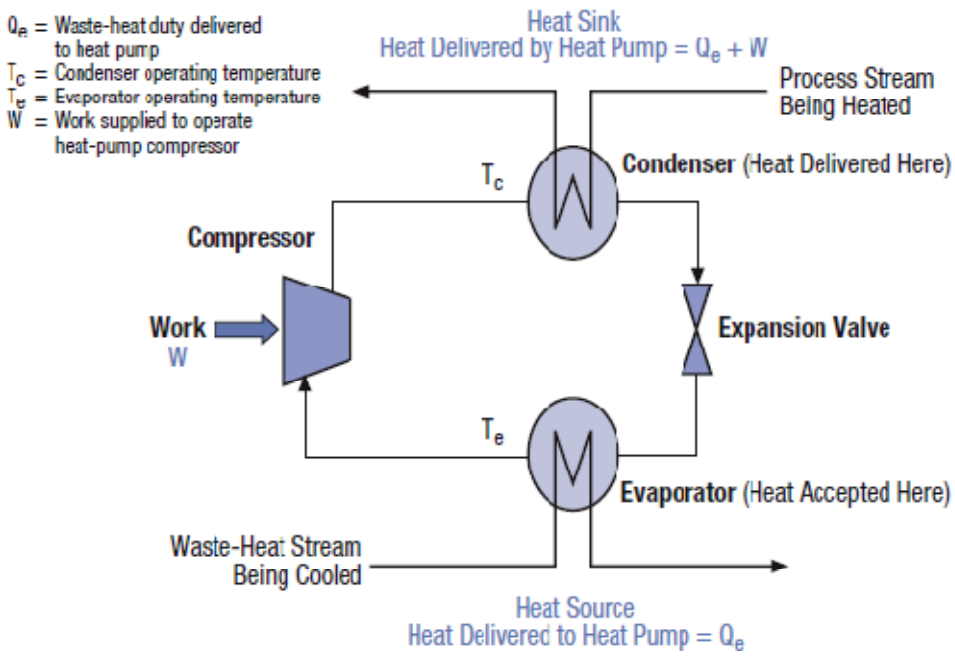

Figure 16. Simple Schematic of Mechanically Driven Heat Pump[43].

This type of heat pump brings in the waste heat on one side, and delivers a higher temperature heat source out through the condenser. Because this process uses a compression cycle, a coefficient of performance (COP) can be seen. The COP for mechanically driven heat pumps is defined as the ratio of heat output over the work input. This type of equipment typically has a COP value from three to six [24]. This means that for every one unit of electrical input to the compressor in the system, three to six units are created in the form of useful heat energy output. This process is not simply creating the extra energy and the remaining energy input to the system is made up for by the waste heat stream.

Absorption Chillers

Another type of heat pump that can be used is an absorption heat pump. This heat pump uses heat energy rather than mechanical power to cause an increase in temperature. The absorption chiller uses both a refrigerant and an absorbent to undergo a thermo-chemical process to produce a change in temperature due to the fluid's boiling and absorption properties. The refrigerant

vapor from the evaporator is mixed with the absorbent material and then pumped to a generator where the refrigerant is re-vaporized using the energy contained in the waste heat. The major advantage of this system is that while a typical mechanically driven heat pump is only capable of increasing the temperature by 50°F, an absorption type heat pump is able to increase the temperature by 200°F to 300°F creating many more potential uses. Another possible benefit from using an absorption type heat pump is the ability to simultaneously provide heating and cooling which has the potential to further increase the economics of installation. This also allows for the wasted heat energy to be used in cooling processes such as refrigeration and space conditioning [42]. Figure 16 shows a simple schematic of how an absorption heat pump operates.

There are two common types of absorption chiller chemical setups, the first type being a lithium bromide-water system. This type is used for larger tonnage applications and generally requires a water cooling system such as a cooling tower. The second type of system is an ammonia-water system which is more common for smaller applications and for applications requiring lower temperature. This system also requires less cooling and can be air-cooled [18]. The absorption chillers can also be configured into a single-stage or multiple-stage systems. The single-stage system was described earlier and rejects some heat to the environment due to incomplete mixing where as a multiple-stage system uses this rejected energy to generate more refrigerant vapor, increasing the efficiency of the process by about 40%. The multiple-stage chillers do require higher temperature heat input to be effective and may not be applicable for most low temperature waste heat recovery operations [52]. Both types of absorption heat pump setups have a lower coefficient of performance than that of mechanical chillers, however have the advantages of

converting waste heat energy into space conditioning during cooling months and require less electrical input to operate [65].

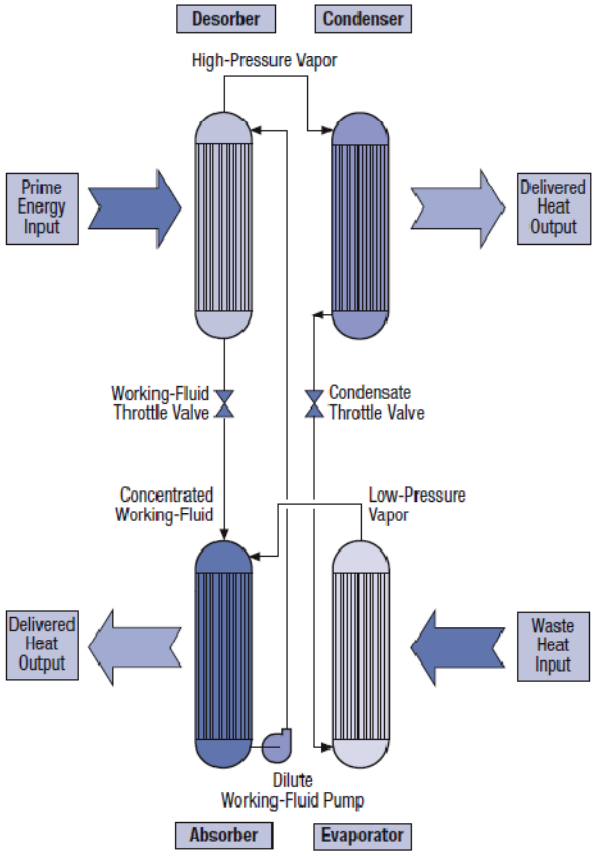

Figure 17. Simplified Schematic of an Absorption Heat Pump [43].

The absorption chiller is mainly available in larger sizes in the 100 to 1,500 ton range. There is one manufacturer currently available who produces smaller tonnage systems in a 10 to 30 ton range. The following list shows a current list of several manufactures and the minimum operating waste heat fluid supply temperature [25].

Table 2. Absorption Chiller Manufactured Capacities [25].

	Trane	York	Carrier	Yazaki	Sanyo
Capacity (Ton)	100-1,500	100-1,500	108-608	10,20,30	100-1,500
Hot Water Temperature Required (F)	200-270	210	250	167-212	94

Much like the mechanical type, the absorption chiller also contains a coefficient of performance, however because there is not a compressor it is defined differently. For this type of equipment, the COP is defined as the ratio of energy removed or added, depending on if the heat pump is used for heating or cooling, to the total energy input [43]. Typical COP values for an absorption chiller are 1.2 when heating is used and 0.7 when cooling is used [25].

Heat Pump Selection Criteria

The selection of which type of heat pump to use is a very important step in insuring the most beneficial system is implemented. There are a few characteristics about the desired system that should be known to help ensure the proper selection is made and provide a starting point for selection and economic considerations. These characteristics include the nature of the waste heat source, meaning whether it is a gas or liquid, the nature of the higher temperature heat sink and the desired increase in temperature. The following table is provided from *McMullan* [43] and shows what type of heat pump is recommended depending on the prior characteristics.

Because both heat pump cycles require either a compressor or pump to operate and therefore an extra electrical energy input, it should only be operated if there is a monetary benefit for the temperature increase in the wasted heat. To understand how much electrical energy is required to increase the waste heat temperature, the coefficient of performance (COP) must be considered. Typical paybacks have been estimated to be from 2 to 5 years [43].

Table 3. Guidelines for Selecting Heat-Pump Type [43]

Temperature Lift	Heat-Source Type	Heat-Sink Type	Suggested Heat-Pump Type
< 100 F	Sensible Cooling of liquid	Sensible heating of gas or liquid Boiling Liquid	1. Closed-cycle mechanical 2. Absorption (lithium bromide/water)
	Partial Condensation of liquid from vapor stream	Sensible heating of gas or liquid Boiling Liquid	1. Closed-cycle mechanical 2. Absorption (lithium bromide/water)
	Condensing Steam	Evaporation of water	1. Open cycle mechanical (single stage compressor) 2. Thermocompression
	Condensing vapor (steam or other)	Boiling Liquid Sensible heating of gas or liquid	1. Semi-open-cycle mechanical (single-stage compressor)
> 100 F	All heat sources (except steam)	All heat sinks (except steam)	1. Absorption (with high lift working fluid) 2. Multistage Mechanical compression
	LP steam	Higher-pressure steam header	1. Open cycle mechanical 2. Absorption (with high lift working fluid) 3. Multistage mechanical compression

Electrical Power Generation

Another method of reusing wasted heat energy is to convert the heat directly into electricity where it can be used for any number of applications. This process can be accomplished using three major methods however all are fairly inefficient in the conversion. The first method uses a Rankine cycle or heat engine to generate electricity. The second method uses a thermoelectric generator that directly converts heat energy into an electrical current and the third method uses a thermophotovoltaic device to accomplish the same effect.

Heat Engine

Because the scope of this paper is recovering low temperature waste heat, the Rankine cycle becomes the most efficient method of converting the heat energy into electrical energy. Also because of the low temperature heat energy, the organic Rankine cycle is recommended. The organic Rankine Cycle (ORC) is simply a conventional Rankine cycle however the working fluid is switched from water to some other type of fluid with a higher molecular mass and lower boiling point. The ORC consists of an evaporator where the energy is supplied to the fluid, a turbine which produces the electrical current, condenser where the fluid is converted back into a liquid state and a pump to transport the fluid in a configuration as shown below.

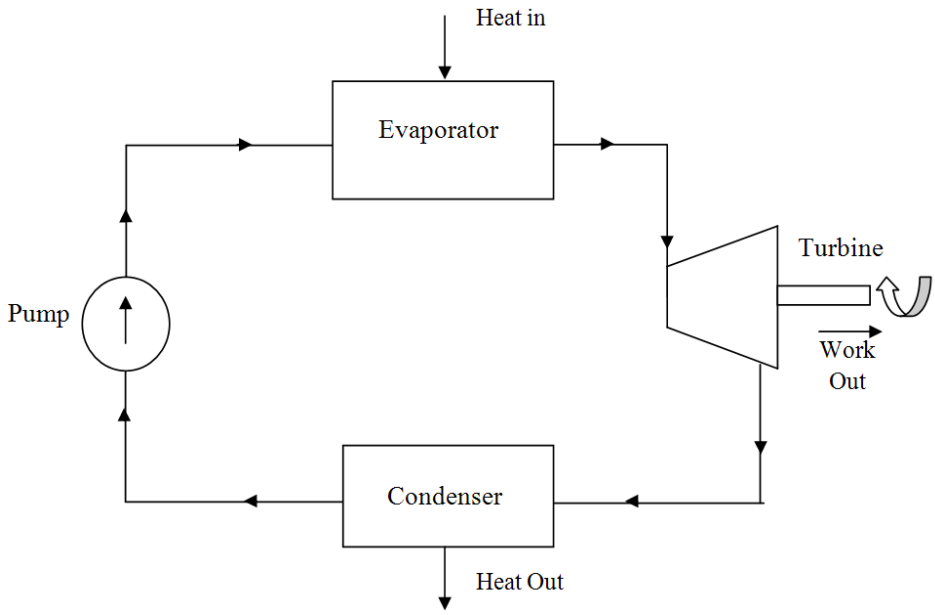

Figure 18. Simplified Rankine Cycle.

The cycle efficiency can vary greatly depending on the efficiency of the turbine and other components of the system. The overall cycle thermal efficiencies range from about 22% to 25% [15]. Typical values for isentropic efficiencies of turbines are shown in the following table for both steam systems and organic fluid systems.

Table 4. Comparison of Turbine Isentropic Efficiency vs. Size and Working Fluid Molecular Weight [54]

	Turbine Isentropic Efficiency %	
Power Level	**Steam**	**High Molecular Weight**
> 10 MW	70 - 80	75 - 80
1-5 MW	50 - 70	75 - 80
200-500 kW	30 - 50	78 - 80
10-100 kW	25 - 50	60 - 75

The next issue that arises when designing an ORC is which fluid should be used. Several studies have been conducted to determine the optimal organic fluid depending on several variables and some of the more effective ones include ammonia, R236EA, isobutane and butane. The first major parameter that should be considered when selecting a fluid is the temperature range over which it will effectively operate and compare that value with the incoming hot temperature reservoir. The second parameter to evaluate is to ensure the fluid does not solidify at temperatures around the ambient conditions. This is to ensure the liquid does not become a solid in the pipes while the unit is not in operation. After the fluid has been confirmed to be physically possible to work in the system, thermodynamic properties should be analyzed to ensure the best efficiency possible can be achieved. This efficiency is tied heavily into the fluids vaporization temperature and the best efficiencies are found when superheating of the substance are not done [15]. This requires knowledge of the high temperature source to design an efficient ORC and to help with this determination a comparative study was conducted by Dai, Wang and Gao that compares different working fluids and their net power output based on the incoming temperature to the turbine under the same operating conditions and each at their optimum operating pressures. Figure 18 shows the results taken from this study for ten different working fluids [12].

As Figure 17 shows, the net power output for both water and ammonia increase as the inlet temperature is raised. However, somewhat counter intuitively, the remainder of the fluids reduce

their net power output as the inlet temperatures increases. The figure also shows if the incoming inlet temperature is high, then ammonia provides the most efficient generation whereas if the incoming temperature is low, around 200°F, the use of R236EA is the most efficient. To optimize a system operating with water or ammonia as an operating fluid, it is seen that increasing the inlet temperature can increase output and

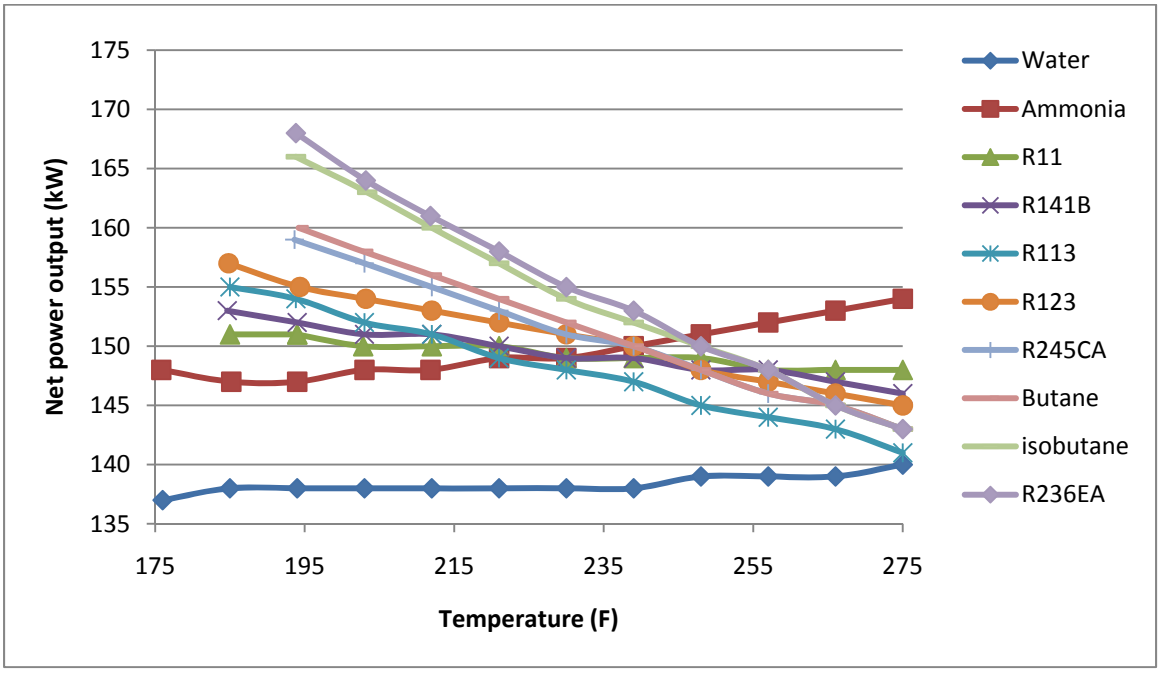

Figure 19.Variations of Net Power Output with Various Turbine Inlet Temperature under Optimal Pressure [12].

the highest efficiency point should be found using this logic. Using these "wet" fluids in the cycle could potentially cause the turbine equipment to be worn faster than with other options because if the fluid condenses, the impact of the condensation with the turbine blades causes rapid erosion and loss of effectiveness. To prevent this from happening, a superheater must be installed which requires additional input and increases the operational costs for the system. For the other fluids, optimizing the efficiency of the cycle includes raising the temperature of the medium just enough so that it is converted into a saturated vapor. The net power output will

begin to decrease as the vapor becomes superheated and will waste the energy input into the system [12].

Thermoelectric Generator

A thermoelectric generator (TEG) operates off the Seebeck effect which takes a temperature difference and converts it directly into electricity. This is the same principle on which an ordinary thermocouple operates. A junction of two different metals is created and when a temperature difference is introduced onto the junction, electrons are driven towards the lower temperature surface and a voltage is created which in turn produces an electrical current. The actual layout of the junction is set on a solid-state integrated circuit with two types of semi-conductor materials, forming the junction. This layout allows for the system to be very rugged and durable along with operating with no moving parts. The amount of voltage that can be created by a given temperature difference is proportional with the addition of the Seebeck coefficient. A large Seebeck coefficient is on the order of 167 microvolts per degree Fahrenheit temperature difference. This leads to the requirement of many TEGs to be added in series to make a useable form of electricity [55]. This conversion method is very inefficient though and generally only converts about 5 to 10% of the heat energy into electrical power and requires temperatures of almost 500°F to operate. Another limitation to the TEG is that current initial costs are about $13 per watt of capacity [32]. However, according to a case study conducted on thermoelectric recovery of waste heat, because of the long life, lack of maintenance requirements and a relatively free energy source, when operated over a seven year period the TEG can produce electricity at a similar cost per watt as purchased power from a major utility [53].

Thermophotovoltaics

A thermophotovoltaic (TPV) device is another method used to convert waste heat directly into electricity. This device uses a photovoltaic diode to create the electrical current based on a temperature difference exciting photons. The TPV functions in the same manner as a photovoltaic solar cell and replaces the sun's energy as the driving force with waste heat. This leads to a heavy dependence on thermal radiation to cause the generation of electricity and limits the efficiency to that of a blackbody. This also requires a relatively high temperature to achieve this radiation effect. This high temperature is well above the level of low temperature waste heat and a heat source of over 2,000 °F is recommended for higher efficiency. This requirement leads this technology to not be currently applicable for low level waste heat recover applications and is provided as information and with possible future technology developments could become applicable. This technology could also become relevant as the push for solar energy becomes important and as new techniques for solar photovoltaic cells become available, thermophotovoltaics will not be far behind.

Although the system is bounded by a blackbody efficiency limit, the system is not an actual black body and will incur losses. The major source of losses in the system is unconverted thermal radiation. This wasted thermal radiation is emitted at a wavelength or direction that the cell is not able to absorb and convert to electricity. To combat this, filters can be added to the system which helps direct and convert up to 75% of the radiation to a usable wavelength and direction. However current technologies still limit the conversion efficiency of heat to electricity to below 20%. The upper limit of this range is only possible with high temperature differences in the most efficient versions available and a more realistic conversion efficiency is around 12%. These systems tend to have an initial cost of around $1,000/kW and a simple payback of about 4

years can be expected when the average energy cost is assumed to be $0.10/kWh [22]. The following figure shows the basic setup of the TPV system and the flow of heat and electrical current.

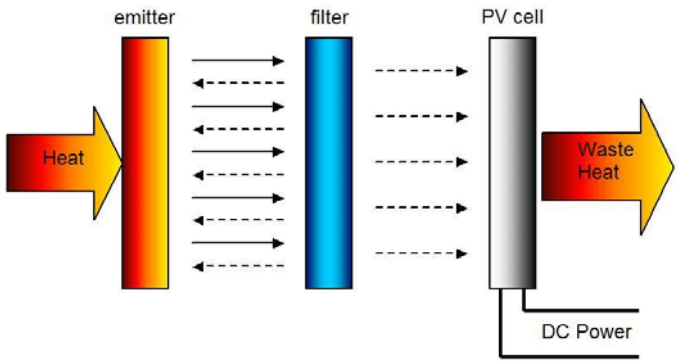

Figure 20. Essential Parts of a Thermophotovoltaic System [35]

Other Technologies

There have recently been breakthroughs in technology that have allowed the development of other systems to take place. One of these methods has recently developed by researchers at MIT in Cambridge, Massachusetts along with assistance from other sources [73]. This technology uses a quantum-coupled single-electron thermal to electric conversion scheme. This technology takes the same principles of thermophotovoltaics described in the previous section and adds to it, however this method can achieve higher power production per unit area than other methods by using thermal fluctuations in the near-surface electric fields. While the thermophotovoltaic alone is bounded in efficiency by the black body radiation concept, this technology has been observed to exceed this limit in lab experiments. To achieve this increase in efficiency, a converter is required to transport this thermal energy to the interior of the photo diode. This converter consists of a narrow, vacuum gap that separates the hot side from the cold side of the structure. It works by consisting of dipoles on each side of the gap which are coupled through Coulomb

interaction. This interaction allows the transfer of the energy from the hot to the cold side which creates the electrical current. The maximum efficiency of this lab test was found to be 43% with an 81°F temperature difference but the electricity produced was all on the millivolt scale [73].

Energy Transportation Fluid

After the energy is recovered from the waste heat, it must be transported to the end use. This can be accomplished by creating a closed loop system that transports the high energy fluid throughout the facility. This working fluid should be selected based on different physical and thermodynamic properties such as boiling point, density, specific heat and viscosity. For heat recovery systems, a high boiling point of the fluid is desirable to maintain the fluid as a liquid and not allow a phase change to gas to take place. While this transformation allows for more heat energy to be stored in the fluid's latent heat, it is not desirable to have a two phase flow in a system of such relatively low temperature. Maintaining a single, liquid phase will allow the system design to become much simpler and more importantly allow for much larger quantities of lower temperature energy to be recovered. By maintaining the fluid in a liquid phase also greatly reduces the initial implementation cost because there is no need to install vapor piping, special vent piping and pressure control devices [56].

The density of a fluid becomes important in its ability to transport energy because the denser a fluid, the more mass there is to absorb the available energy. The density is not constant for different temperatures and will become less and less as the temperature increases causing the fluid to expand. The specific heat of the fluid aids in the fluid's ability to gather and store the energy recovered and the higher the specific heat, the more energy that can be transported throughout the system per pound of fluid delivered. The density and specific heat can be combined together to gain a greater understanding of how much energy transportation potential a

specific fluid has. This combination is known as the fluid's volumetric heat capacity. A simplified version of the first law of thermodynamics states that the heat transferred is equal to the mass flow rate times the specific heat of the fluid times the temperature difference. The mass flow rate can then be expanded to include the multiplication of density, velocity and cross-sectional area. The velocity, area and change in temperature can be taken as constant no matter the fluid being analyzed which leaves only the density of the fluid multiplied by the fluid's specific heat as variables. This analysis can be a simple and quick check to determine which fluid will be able to transport the most heat energy. A graphical comparison of this property can be seen in Figure 28 in the *Energy Transportation Fluid Comparison* section. The final property discussed is the dynamic viscosity of a fluid which is also important during the selection process. This property does not directly affect the heat transfer capabilities but changes the flow characteristics and with a higher viscosity the required pumping energy can be increased. All of these fluid properties, along with initial cost and operating costs of the selected fluid should be considered. The following sections discuss different possible transportation fluids and the advantages and disadvantages of each.

Water

The most common type of energy transportation fluid because of its ease of availability and high heat capacity is water. Water is used for many heat transfer applications such as cooling water systems and also is used in process steam and hot water applications. This extensive usage as a working fluid has led to most pumping and heat recovery and transfer equipment to be developed with it in mind. However water does have certain limitations when it comes to temperature and when steam is not desired or expected to be created, water can become difficult or undesirable to be used. Along with these issues, water also has the potential to cause high corrosion problems.

The following figure shows the density and specific heats of saturated water at different temperatures in which it exists when under 60 psig pressure.

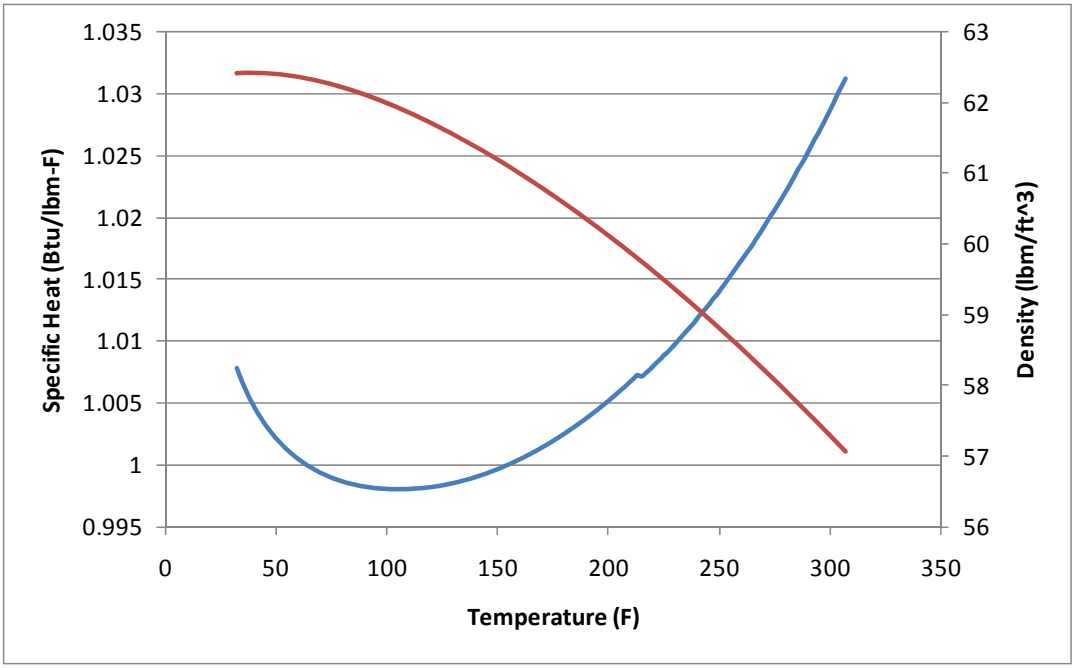

Figure 21. Specific Heat and Density of Saturated Water at Different Temperatures.

Because water has such a high specific heat relative to most other available fluids, additives have been developed to extend the boiling and freezing points so that the fluid mixture formed has a larger temperature range and can be used for more applications. This concept is not only employed in waste heat recovery operations but is also found in everyday use of an automobile's cooling system. The majority of these additives are glycol based fluids. The Dow Chemical Company produces both triethylene glycol and tetraethylene glycol which both serve for this purpose.

Triethylene Glycol

Triethylene glycol (TEG) is a colorless and odorless fluid composed of a large strand of hydrogen, oxygen and carbon which can be simplified to $C_6H_{14}O_4$. The fluid has moderate

viscosity, low volatility, high boiling point and is soluble in water and is also able to mix with other organic fluids. This property of the fluid can be used to enhance the heat transfer characteristics of certain fluids when needed. The boiling point of the fluid at atmospheric pressure is 550°F setting an upper limit on the usage of this fluid as an enhancement to an organic fluid. The fluid is considered chemically stable and will not decompose under normal circumstances. It is also non-corrosive to materials used in a typical distribution system. However, the boiling point of water is not greatly increased until the concentration becomes about 80% to 90% of the total weight and really doesn't increase substantially until all the water is removed. This becomes an issue because the specific heat of the triethylene glycol is much smaller than that of the water, taking away the advantage of using water as the heat transporter. The following figure describes this by showing different concentrations of triethylene glycol with water, based on weight, over its working temperature range [60]. This figure is based on an operating pressure of 1 atmosphere and does not take into account the increased temperature range of water seen by increasing the pressure to 60 psig.

As can be seen in Figure 21, the specific heat of the mixture continuously decreases as more and more triethylene glycol is added. The only possible advantage that the TEG contains is when no water is present and even though the specific heat of the fluid is still smaller, the temperature range is greatly increased to the boiling point of 550°F.

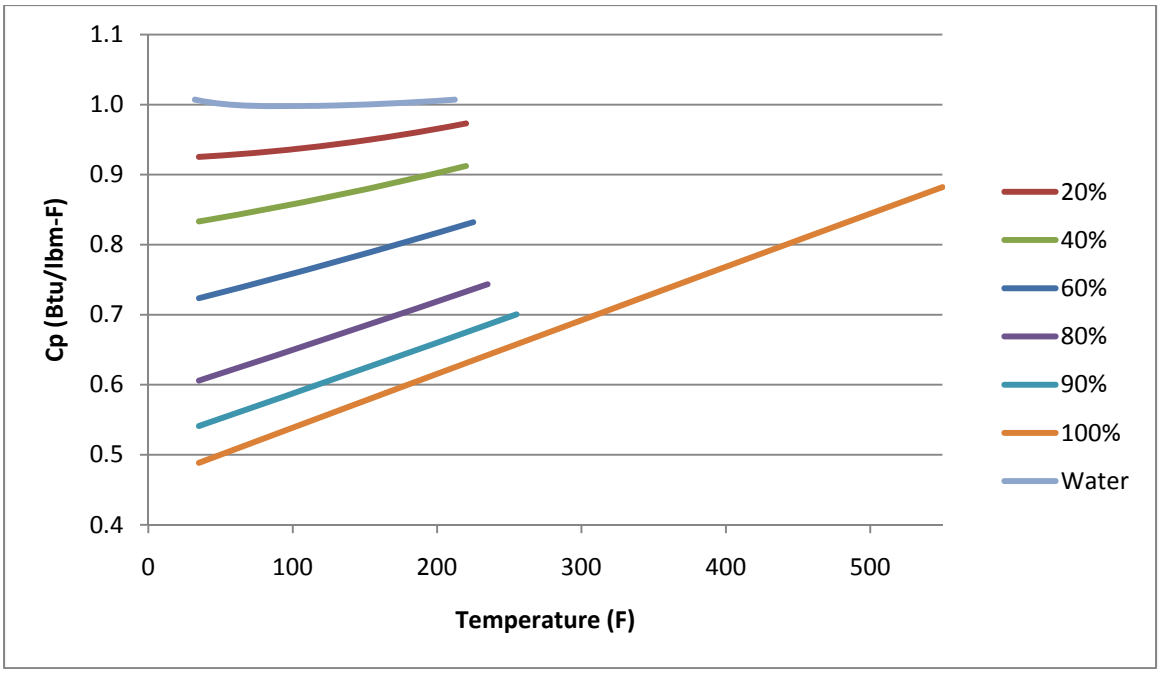

Figure 22. Specific Heat at Different Temperatures for Different Concentrations Based on Weight Over its Corresponding Temperature Range [60].

Tetraethylene Glycol

Tetraethylene glycol is also an odorless, transparent fluid made up of a large strand of hydrogen, oxygen and carbon which can be simplified to $C_8H_{18}O_5$. This fluid has very similar physical and thermal properties to the triethylene glycol but it has a higher boiling point and is denser. Although the boiling point is high, the maximum temperature range for the fluid is set by the decomposition temperature which is known to be at 464°F. The same problems arise as with TEG because of water's boiling temperature's slow increase and rapid decrease in overall specific heat as this concentration is raised. The following figure shows the specific heat at different concentrations when mixed with water throughout their operational temperature range [59]. This figure is based on an operating pressure of 1 atmosphere and does not take into account the increased temperature range of water seen by increasing the pressure to 60 psig.

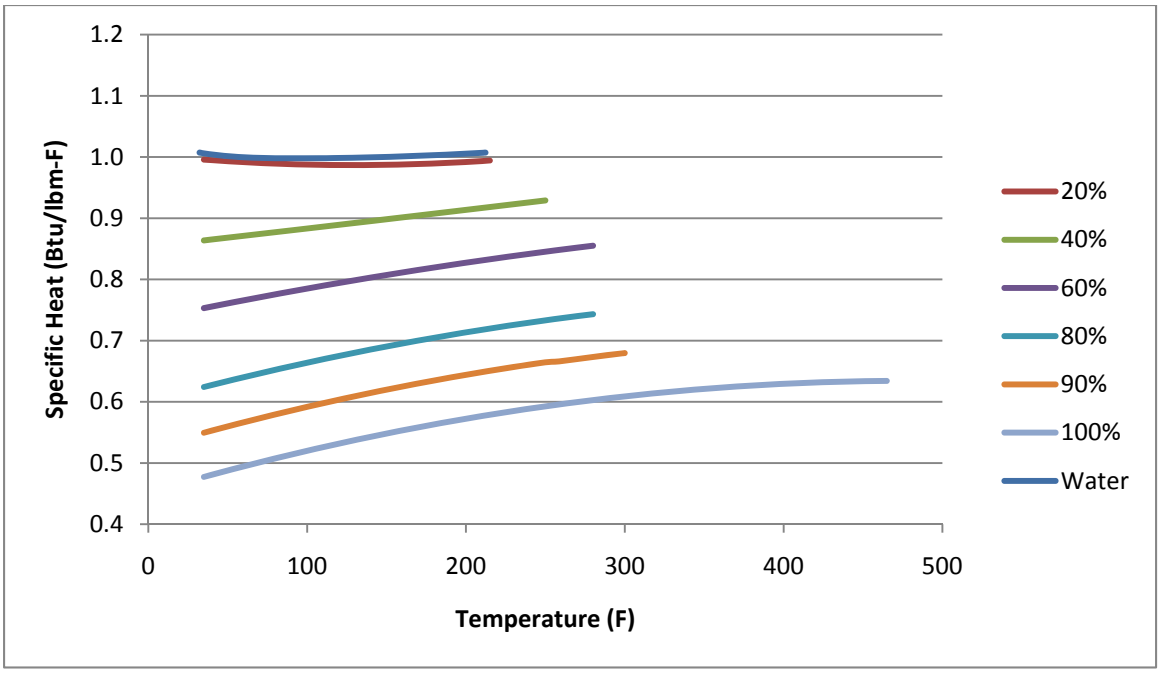

Figure 23. Specific Heat at Different Temperatures for Different Concentrations Based on Weight over its Corresponding Temperature Range [59].

DOWTHERM

DOWTHERM is an organic fluid developed by The Dow Chemical Company and is used as a heat transfer fluid in heat recovery applications. The fluid comes in different mixtures and can be used for temperatures of up to 700°F. DOWTHERM G is a blend of di- and tri-aryl compounds and offers the most attractive heat transfer and temperature range characteristics of the available mixtures of the DOWTHERM line of products. It is recommended for temperatures from 20°F to 675°F and remains in a liquid state a pressure of 48.8 psig at this temperature. However, DOWTHERM G is a flammable material and has an autoignition temperature of 810°F therefore the material should not be used if the process temperature is ever to reach above the recommended temperature range to avoid ignition of the fluid. The fluid is noncorrosive with common metals even when operated at high temperatures and most corrosion issues arise from chemicals infiltrating the system through cleaning or process leaks. The fluid

has a density of 56.5 lb/ft^3, a specific heat of 0.524 Btu/lbm-°F and a dynamic viscosity of 0.004 lbm/ft-s at 400°F. The following charts show these values at different temperatures in the fluid's operating range [14].

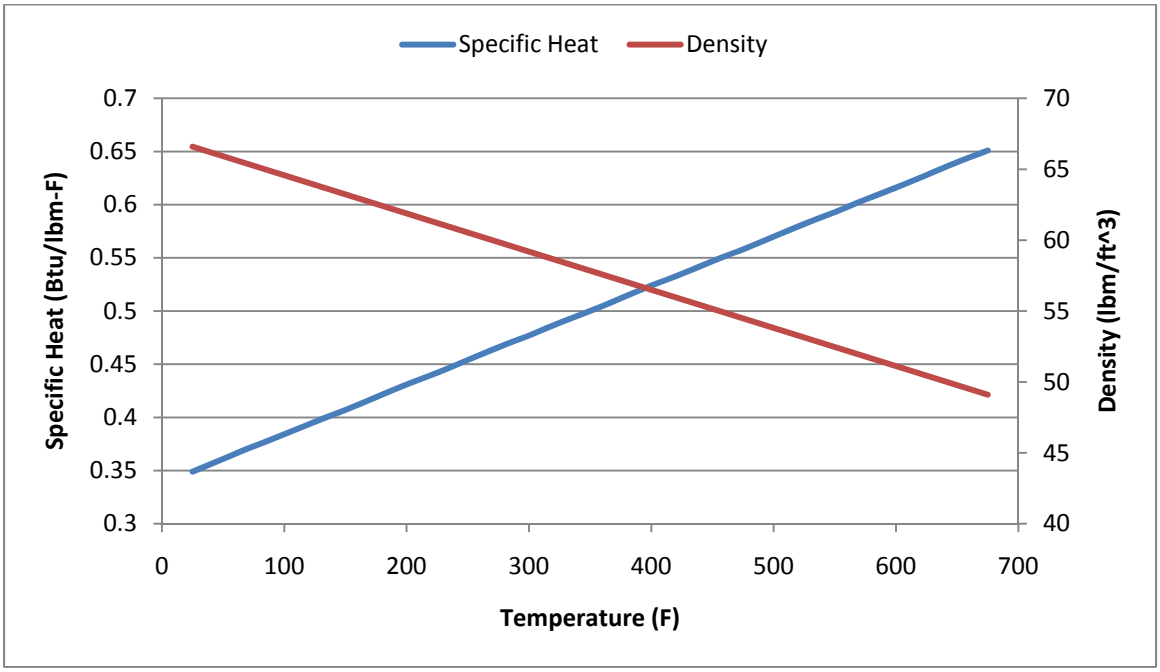

Figure 24. DOWTHERM G Specific Heats and Density at Different Temperatures [14].

UCON HTF 500

UCON HTF 500 is another heat transfer fluid manufactured by the Dow Chemical Company which is suitable for temperatures up to 500°F and is a polyalkylene glycol based fluid. This heat transfer fluid offers high initial heat transfer efficiency and does not produce a large amount of buildup in the pipes compared to petroleum based products which would prohibit heat transfer throughout the life of the system. The fluid is fully compatible and noncorrosive with most common metals that are used for piping and fittings such as steel and copper. It does, however, require the distribution system to contain a continuous vent to minimize the risk of system over pressurization and/or pump cavitation. The important physical properties of this fluid are quite

favorable up to the 400°F to 440°F range where it begins to slowly drop below the properties of other fluids [58]. The density and heat capacity of the fluid are shown at different temperatures in the figure below.

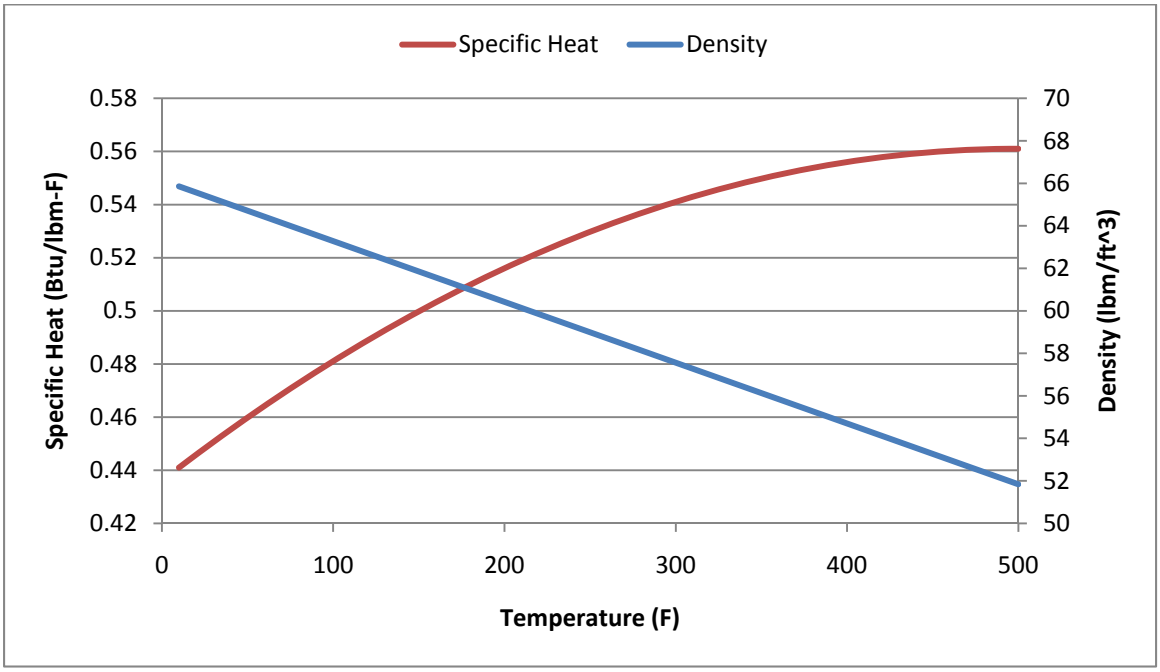

Figure 25. Properties of UCON HTF 500 Across Temperature Range [58].

Therminol XP

This heat transfer fluid is manufactured by Solutia and provides reliable heat transfer for temperatures up to 600°F and can operate all the way down to 0°F. However, this fluid is not classified as a fire-resistant heat transfer fluid and care should be taken that the fluid's ignition point of 655°F not be approached. The fluid is practically non-toxic and is specially formulated to minimize the fouling potential caused by oxidation and decomposition of the fluid throughout its life. The fluid is non-corrosive to commonly used metals in piping and distribution system and is composed of 100% petroleum based white mineral oil. The important physical and thermodynamic properties are almost identical to that of the DOWTHERM G fluid described

earlier, which can be seen later in the fluid comparison figure, however this fluid offers a different material composition and a different option depending on the particular situation encountered. The following figure shows the density and specific heat of Therminol XP as a function of temperature [56].

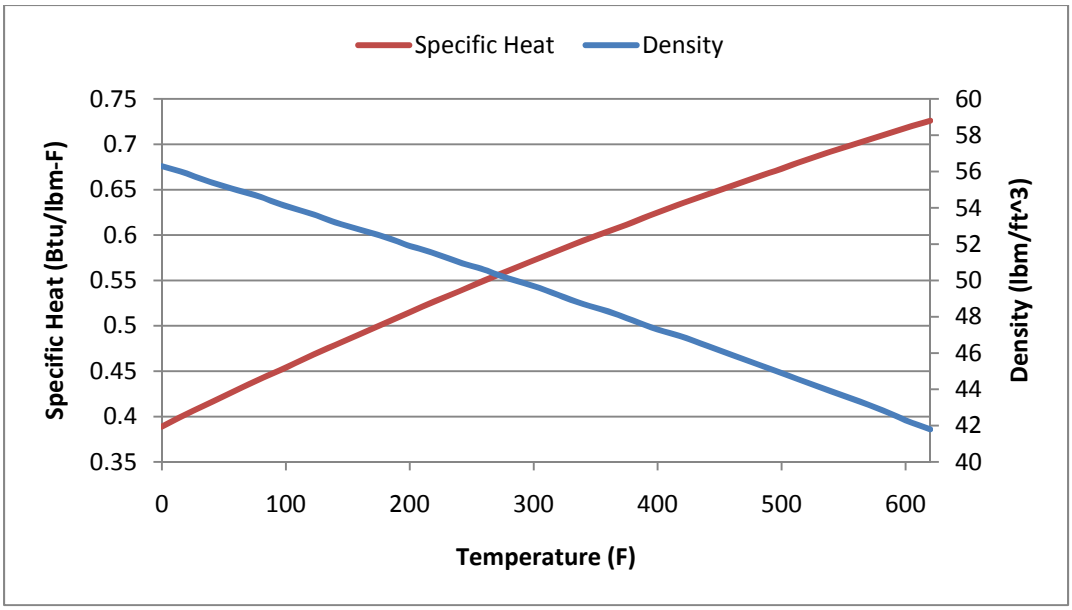

Figure 26. Properties of Therminol XP Across Temperature Range [56].

Microencapsulated Phase Change Material Slurries

A concept that can be implemented to increase the energy storage potential of one of the previously listed heat transfer fluids is adding microencapsulated phase change materials (MPCM) into the fluid, forming a slurry. The microencapsulated phase change material is a small material that consists of an outer, encapsulation layer shell which is on the order of micrometers in diameter. This shell is then filled with a core of phase change material. This setup leads to a large outer surface area relative to the volume which provides excellent heat transfer characteristics. When dispersed into a fluid it can be easily pumped without causing any mechanical damage to the system [28]. These MPCMs can be manufactured by a variety of

methods including spray-drying, fluidized bed and other chemical approaches. The addition of the MPCM increases the fluid's energy storage potential by adding the latent heat storage of the PCM. This is the energy that is required to turn the material from a solid into a liquid or liquid to a gas which will be released as the transformation is reversed [74]. This process happens with virtually no wasted energy and can provide an efficient means of thermal storage.

However, this concept is currently not available for temperatures in the 400°F range and only applicable for refrigeration applications. Limitations with the encapsulation materials' high temperature capabilities as well as the phase change material density variations in the mixture are a major concern and require further research and development before this application is feasible [47]. The current encapsulation materials have a maximum working temperature of much less than 400°F and are made from natural or synthesized polymers or plastics, and currently the only reasonable material to handle that type of temperatures are metals. This brings up another issue, because using metals as the encapsulation material takes away the ability for the slurry to be pumped without causing substantial damage to the system.

Energy Storage Tank with Phase Change Materials

To avoid the problems with MPCM slurries, a stationary phase change energy storage system can be used. This system allows for higher temperature applications to be used and takes away the current issues of microencapsulation materials and density variations. The process works by using a phase change material packed into a storage tank where the high temperature fluid is pumped through and energy is transferred into the materials, causing a phase change to occur. This energy can then be stored in the latent heat and later used in a different process when the demand is required. This storage is able to bridge the gap between the availability of the energy and the demand. There are commercially available PCMs that are capable of withstanding high

temperatures and include *PlusICE* PCM which is a high temperature salt based solution. This solution can be manipulated to have phase change temperatures at 230°F to up to 1,850°F [48]. Other types of high temperature PCMs include sugar alcohols, organic materials, molten single salts and metals [29].

For waste heat recovery systems there are two major configurations for thermal storage tanks. The first includes a thermal storage tank capable of storing all the energy recovered from the exhaust gas. This system type allows the user to capture the full energy and distribute it when a demand in the system is found. This supplies a buffer between when energy is supplied and when a demand for it is required. The major drawback of this system is the capacity of the storage tank and the larger upfront cost. The second type of system applies a similar principle as the previous type but locates the thermal storage tank at the end of the energy transportation loop. This storage tank is designed to capture the energy remaining in the transportation fluid before it enters back into the main heat exchanger which recovers energy from the stack gas. This system will not only recover the excess energy in the fluid but it will also lower the temperature of the fluid as it enters the heat recovery heat exchanger and allow for more energy to be captured in that process as well. The major advantage of this system is the capacity of the storage tank can be much smaller than the previous scenario. The system does contain drawbacks such as a requirement for an intermittent demand system to recover this stored energy along with a temperature controlled bypass loop to ensure the energy is not transferred from the storage tank back into the transportation fluid. These types are discussed further in Chapter 5.

Energy Transportation Fluid Comparison

Because each of the previously discussed energy transportation fluid has its advantages and disadvantages, a way to determine the best fluid for a specific application should be established.

The first major parameter that should be considered is the maximum operating temperature of the fluid and the minimum value should be based on the expected temperature of the waste heat which is being recovered. For the following comparisons, both triethylene glycol and tetraethylene glycol are taken to be at 100% concentration and not mixed with water. The following figure shows a plot of each fluid discussed maximum operating temperature.

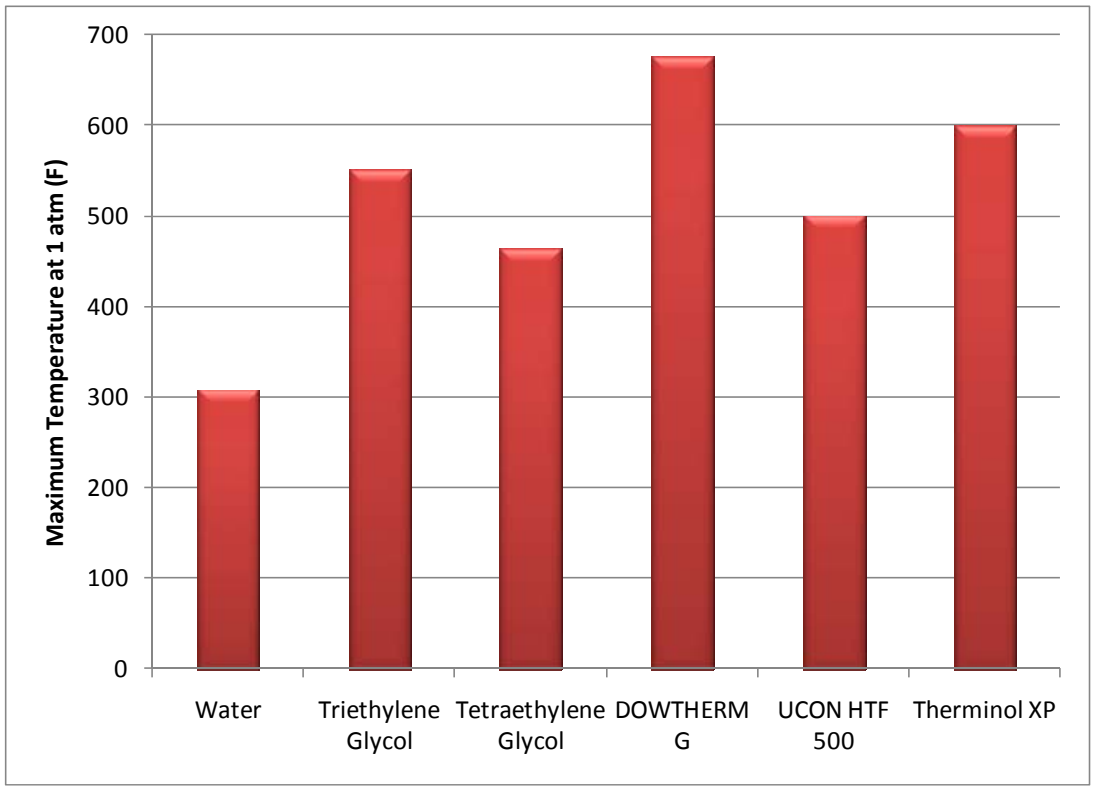

Figure 27. Maximum Operating Temperatures for Heat Transfer Fluids.

Another major parameter that becomes important in the design of the entire system is the fluid's viscosity. The viscosity of a fluid is basically how much resistance a fluid has to shear or tensile stress and directly relates to the ease of the fluid to be pumped through a network. So the higher the viscosity of the fluid, the more difficult it is for a pump to move the fluid throughout the system which leads to increased pumping horsepower and more energy usage. The following

chart shows a comparison of each of the fluid's viscosity at temperatures throughout its temperature range.

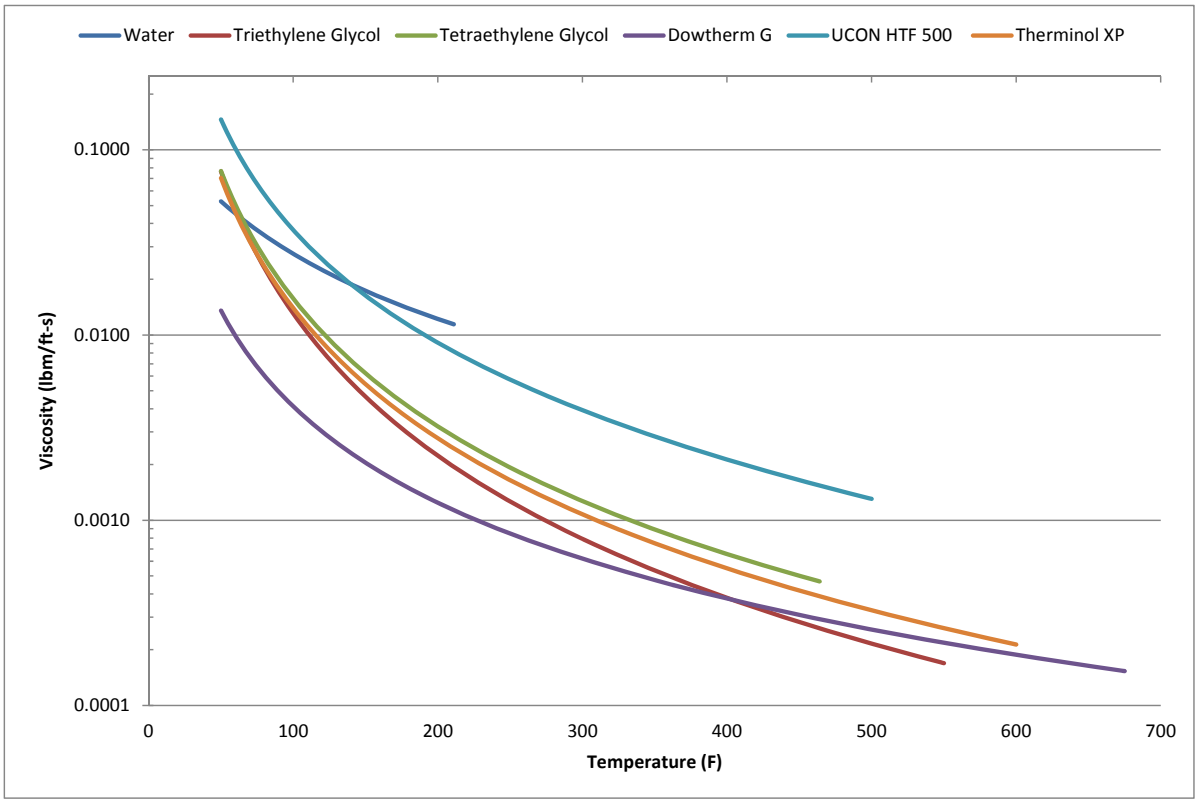

Figure 28. Viscosities vs. Temperature for Heat Transfer Fluids.

Two other important considerations when selecting a heat transfer fluid is the density and the specific heat of the fluid. The ideal heat transportation fluid would have the maximum value for this at the operational temperature because of reasons discussed earlier. Because these values are coupled into the first law of thermodynamics equation, they can be combined for the selection process. Figure 28 shows the product of density and specific heat for each of the heat transfer fluids for different temperature values in their operational range.

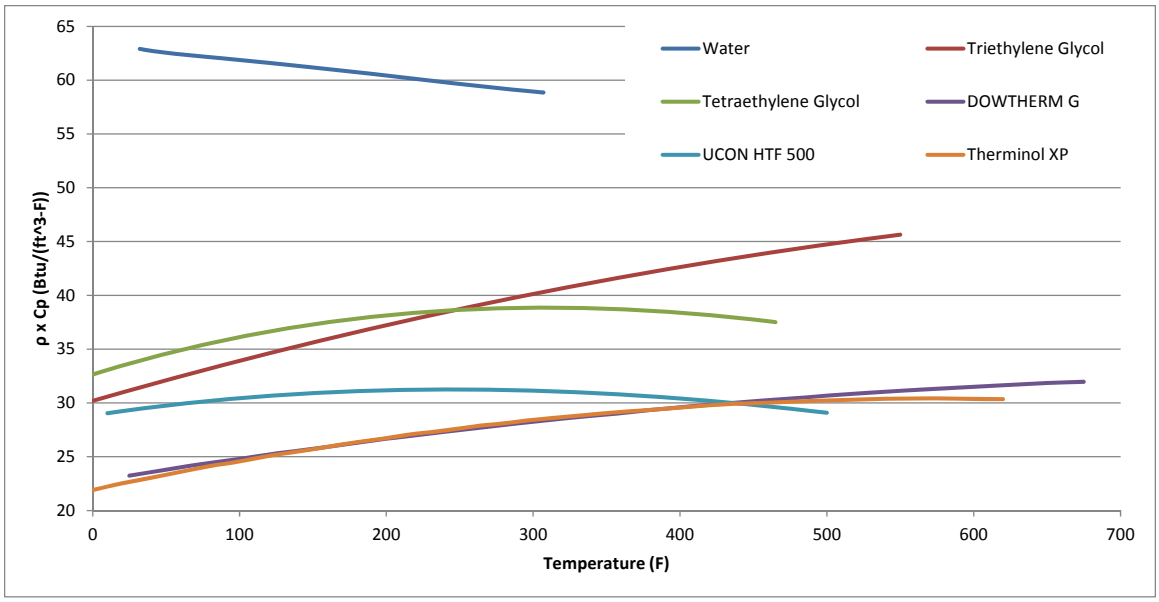

Figure 29. Volumetric Heat Capacity for Heat Transfer Fluids Across Temperature Range.

Figure 28 demonstrates that when temperatures less than 212°F are required, water is by far the most effective heat transfer fluid. However, this temperature range is rarely adequate to recover and transport waste heat energy. Triethylene glycol possesses the next best heat transportation characteristics and provides a high maximum temperature range while also providing a high combined density and specific heat. The tetraethylene glycol provides better low temperature characteristics when compared to the triethylene glycol, however this value declines as the temperature increases. This fluid also possesses a relatively low maximum operating temperature of 464°F, and becomes thermally unstable and begins to decompose at this temperature. When thermal heating oils are required, UCON HTF 500 demonstrates better heat transportation characteristics with temperatures less than 400°F but the slope of its density times specific heat curve becomes negative and decreases under both DOWTHERM G and Therminol XP at this temperature. The heat transportation characteristics of the DOWTHERM G and Therminol XP are relatively similar throughout the heating range with DOWTHERM G becoming slightly better at temperatures above 500°F. They both possess a high maximum

operating temperature with 675°F for DOWTHERM G and 600°F for Therminol XP and should be applicable for any type of recovery system when recovering low grade temperature.

Other considerations while selecting a heat transfer fluid includes the cost of the fluid as well as the overall cost of operation and installation. This can be made up by several components including piping materials required to be compatible and non-corrosive with the fluid selected. This can also include the pump sizing and operational costs for moving the fluid to overcome a more viscous fluid. The amount of heat energy that can be recovered from the system is another important variable when determining if the system will be cost effective to implement and the system should always benefit the facility when in operation.

Pump Types

Once the operating fluid has been selected, a pumping system then must be identified. There are several options that can be chosen regarding the pump type and the selection depends on different variables such as fluid viscosity, fluid flow requirements and the physical layout of the system. The major variations that will be discussed are what type of pump should be used for what application and whether or not a variable speed drive should be used to control the pump motor.

Centrifugal Pumps

A centrifugal pump creates flow and an increase in system pressure dynamically. This is accomplished by using an impeller, usually with vanes curved backwards in the direction of flow which is connected to a rotating shaft. The incoming fluid is forced in the inlet side of the pump housing by some other additional energy source which could be an upstream pump or gravity. The fluid is then moved towards the exit by the rotating impeller causing a reduced pressure at

the inlet, this lower pressure then draws in additional fluid and continues the fluid flow[70]. This type of a pumping system is referred to as a "rotodymanic" pump because the flow is created greatly through velocity changes as the fluid flows through the impeller and the passages throughout the pump [34]. The following figure shows the basic setup and flow patterns generated in a centrifugal pump.

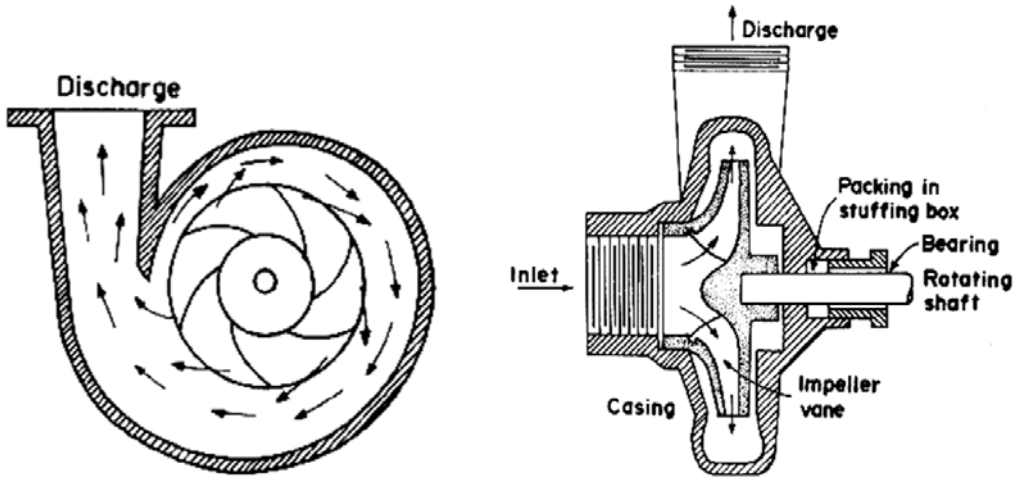

Figure 30. Centrifugal Pump Cross Sections [31]

As can be seen in the figure, after the fluid is directed from the inlet, across the impeller and towards the discharge, it is directed into an increasing cross-sectional area called a volute or diffuser. This area is designed with increasing cross-sectional area to reduce the linear velocity of the fluid, and due to the conservation of energy principle, the pressure of the system must be increased. To gain an increase in pressure or flow, additional stages can be added to the pump. This is accomplished by adding another impeller in series with the previous one and can be effective with up to thirty or more impellers operating in a single pump [23].

Positive Displacement Pumps

A positive displacement pump operates by taking the incoming fluid and capturing it in a moveable enclosure to transport it to the desired location. This moveable enclosure can be anything from a piston to a screw or roller. The positive displacement pump differs from a centrifugal pump by using a hydraulic force to add energy to the fluid directly. This results in a direct pressure increase in the fluid resulting in a head capacity curve when operating at a constant speed to become virtually a vertical straight line. This change in pressure does not significantly affect the flow characteristics of the pump. This is the result of its enclosure design as well as incompressible flow of the fluid, which causes the pump to move a specific volume of fluid no matter the pressure that is seen. This particular characteristic of the positive displacement pump also allows for the generation of any pressure at a given speed, leading to the need of a pressure relief valve to ensure the system pressure does not exceed a safe limit for the equipment it is feeding.

The positive displacement pump is capable of moving a wide variety of fluids which becomes an important variable in a waste heat recovery system. These are achievable because of the basic design principles of the pump itself, since energy is added to the fluid by force and it does not depend directly on acceleration. This causes pump to be able to handle liquids with extremely high viscosities and high temperatures [23].

With a reciprocating system, a piston and cylinder or some type of plunger acts as a displacer to move the fluid. This system is capable of almost any pressure range and flow capacity, however they tend to not be as efficient as and are more expensive than modern centrifugal pumps. Other problems with the system are the slow speed of operation and the pulsed nature of the flow. The reciprocating systems do still possess some inherent advantages over other systems because of its

design. These include easy controllability of volumetric flow by variable speed motors or simply stroke adjustments [23]. The following figure shows the components associated with a reciprocating pump and their layout.

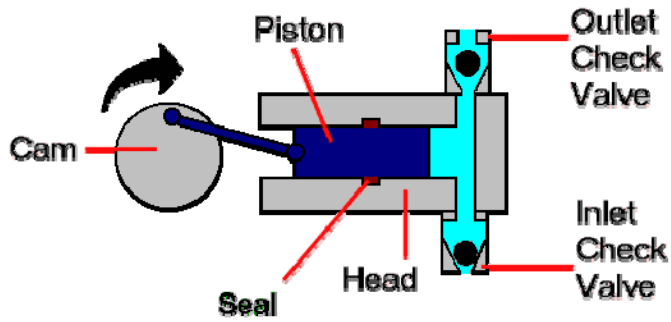

Figure 31. Layout and Components of Reciprocating Pump [39].

Another type of positive displacement pump is a rotary pump made up of multiple, intermeshing screws. This type of pump is typically applicable if the system requires high pressure, high capacities or a broad range of viscosities and temperature ranges are required. This pump is capable of moving highly viscous fluids such as greases, lube oils and fuel oil. The system can be designed in one of two ways, the first is with external bearings and the second consists of internal bearings. The external bearing design physically locates the bearings and gears in a separate chamber filled with oil for lubrication while the internal bearing design maintains the bearing and timing gears inside the pumping chamber and uses the pump's working fluid as lubrication. This obviously leads to the lubricating capabilities of the working fluid to play a major role in the design of the pump. The internal bearing location should be considered if the working fluid has a viscosity of 150 ssu and does not contain corrosive or hard compounds. However, as can be seen in the following table of operational design limitations for both design types, the internal bearing type has a maximum temperature of 325°F which is generally not high

enough for most heat recovery applications. This generally leads to the selection of the external housing [23].

Table 5. General Design Limitations of a Conventional Twin Screw Pump [23].

	Internal Design	External Design
Max. Capacity (gpm)	4,000	10,000
Max Delta P (psi)	2,000	2,500
Viscosity (ssu)	150 – 50,000	32 – $2x10^6$
Temperature (F)	325	850
Speed Range (rpm)	10 – 2,750	10 – 1,750
NPSHR (ft of fluid)	3 – 100	3 – 100

The following figure shows the components and layout of the internal bearing twin screw pump design. As stated before, this system only varies from the external bearing twin screw pump by the location of the bearing and gears.

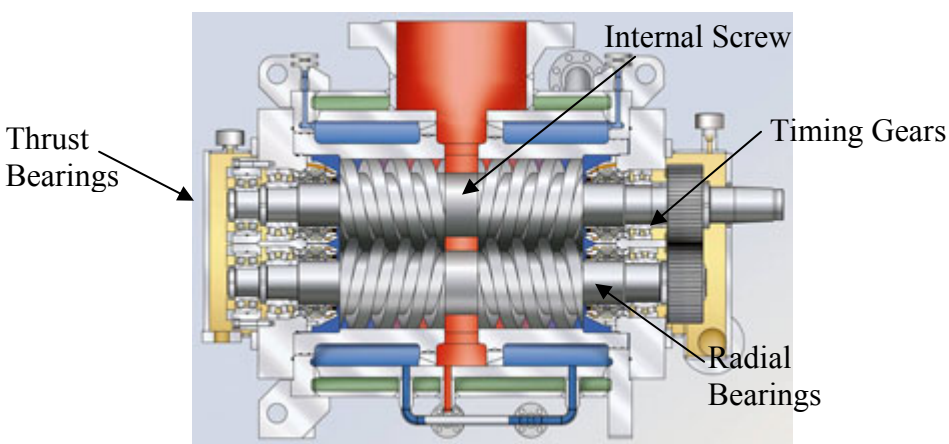

Figure 32. Internal Bearing Twin Screw Pump Design [1].

To overcome some of the major issues with the basic positive displacement pumps, a progressive cavity pump was developed. Garay describes it as consisting of single helical rotor turning inside a resilient stator with an opposite spiraling double helix cavity that is constantly opened and resealed with each revolution [23]. This design creates an almost pulseless delivery of the

fluid and the flow rate is proportional to the rotational speed of the pump. This pump can also be used for the movement of highly viscous fluids but cannot be operated at speeds from a direct drive. This requires a universal joint to be added to supply the appropriate amount of torque to the system and to maintain a long life of the pump relative motion between the rotor and the drive shaft must be considered. This design is adapted into many different types of simple positive displacement pumps including piston-cylinder pumps and screw types [23].

Variable Speed Drives

For each of the pumping types described, a variable speed drive (VSD) can be applied to the electric motor to control the speed and in turn the flow rate of the fluid. There are two types of system designs that are considered and dictate the control strategies for the pumping system. The first is a supply controlled system which removes incoming flow and are controlled to match the incoming flow rate received. This type of system can include flood protection schemes and discharging of process liquids. The second type is a demand controlled system which delivers fluid to a required process like a cooling water system. Both of these types of systems can be controlled by a constant speed pump when additional storage to the system is added, however a more effective and energy efficient method of controlling these pumps is to add a variable speed drive to reduce the pumping capacity with the demand to the system.

To demonstrate the magnitude of energy that can be saved by the addition of a variable speed drive, pump affinity laws must be introduced. Because a pump is a moving device, head is generated. For a centrifugal pump this happens as the impeller is rotated. For positive displacement pumps this occurs as the screw rotates or the piston pushes the fluid from the cylinder. For a centrifugal pump, the flow rate is directly proportional to the rotational speed, the head is proportional to the rotational speed squared and the power consumption is

proportional to the rotational speed cubed. For a positive displacement pump, the energy consumption is directly proportional to the volumetric flow rate. These simple proportionalities demonstrate the need for a way to control the rotational speed of the pump when different flow rates are required and shows that a reduction of rotation in a centrifugal pump will cause the power consumption to be reduced by a cubed factor [20].

A variable speed drive operates by using a concept known as pulse width modulation. This is a way of delivering energy through a succession of pulses rather than a continuous analog signal. The major advantage of this method is efficiency and works because the motor's own inductance acts like a filter and stores energy during the pulse cycle and releases it at the reference signal. Basically this along with the inertia of the system allows the motor to see a smooth, average current input rather than a choppy, on/off pulse. The following figure shows a sinusoidal wave being converted to a pulse-width modulated output with the addition of a sawtooth carrier [44]. This technology, along with an exciter control is able to vary the speed of a conventional electric motor.

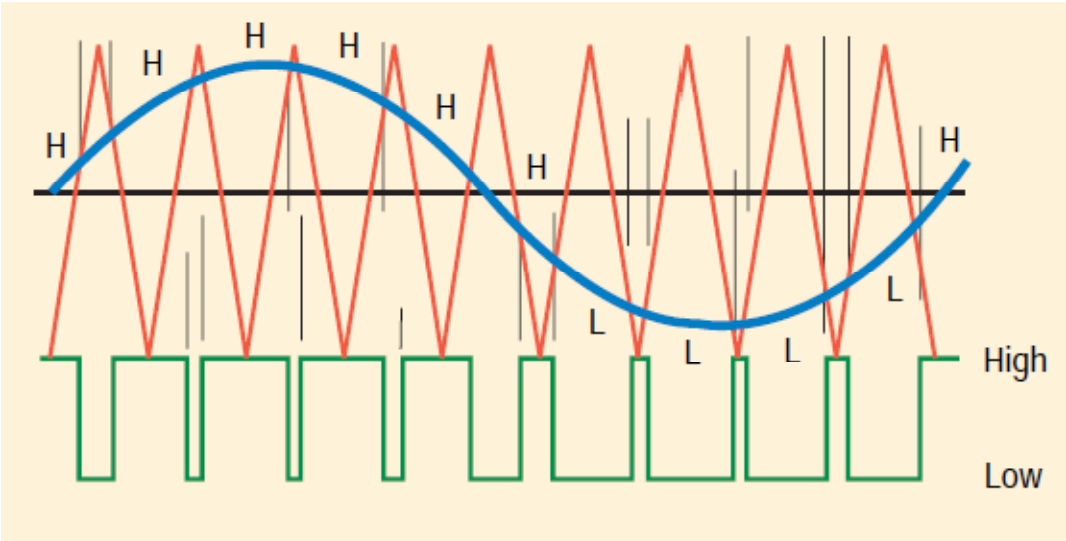

Figure 33. Example of Pulse-width Modulation [44].

As was mentioned earlier, the ability to control the rotational speed of the pump can save energy by a cubed factor with centrifugal pumps. Typical energy savings reductions can be seen to be anywhere from 30% to 50% [20]. Other advantages of implementing a variable speed drive to control the speed of the pump include improved process control, improved system reliability and soft starting of the pumps to decrease the overall demand of the facility. However there are also several disadvantages that can occur as a consequence of a VSD. These include such things as the requirement of having motors with reinforced "inverter duty" insulation in the United States because of the increased supply voltage. More disadvantages from installing a VSD are harmonic distortions, need for electrical screening, deterioration of older motor type insulation and the fact that long cable runs can cause a raised voltage at the motor input. Other possible issues include circulating currents in the drive shaft of larger motors which leads to increased bearing wear, losses due to frequency conversion, and wear of the equipment itself in less than ideal locations [20].

Pump Comparison

To adequately determine what pump is necessary for a certain application, a few parameters must be considered. The system head is a concept that relates the energy in the fluid to an equivalent column of water in terms of height. Three other system characteristics that become important are the volumetric flow rate requirements, operating pressure and the viscosity of the fluid. The head and flow rate produced by the pump are important parameters to ensure the pump is operating at its most efficient point on the pump curve while the viscosity of the fluid determines the ease of pumpability of the fluid in either pump [41].

Once these parameters in a system are known it is important to realize that different pumps operate more efficiently under certain design conditions. For example, when comparing flow

rate versus head, the centrifugal pump operates along a horizontal line that slopes negatively towards the end while a positive displacement pump operates along a vertical line at basically a fixed flow rate. The following figure shows this difference.

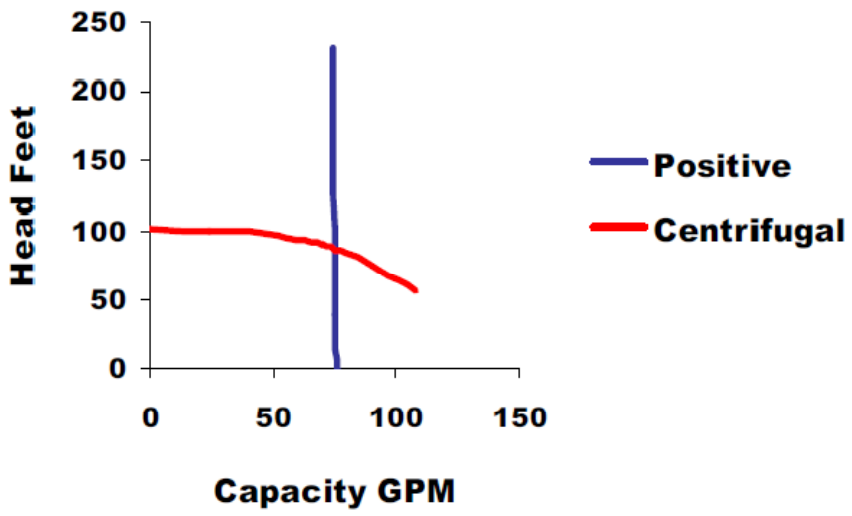

Figure 34. Flow Rate vs. Head for Positive Displacement and Centrifugal Pumps [49].

The viscosity of a fluid also has a drastically different impact on the pumping performance depending on the type of pump selected. The following figure shows that as a fluid becomes more and more viscous, the centrifugal pump rapidly loses its ability to move a fluid while a positive displacement pump will slightly increase the flow rate as the viscosity is increased. This can be explained by the more viscous fluid fill the "clearances" of the pump causing a higher volumetric efficiency [49]. This difference can be seen in the following figure. This property of a positive displacement pump quickly becomes one of its more important attributes. This allows for highly viscous fluids to be pumped when a centrifugal pump is not physically capable of moving the fluid [70].

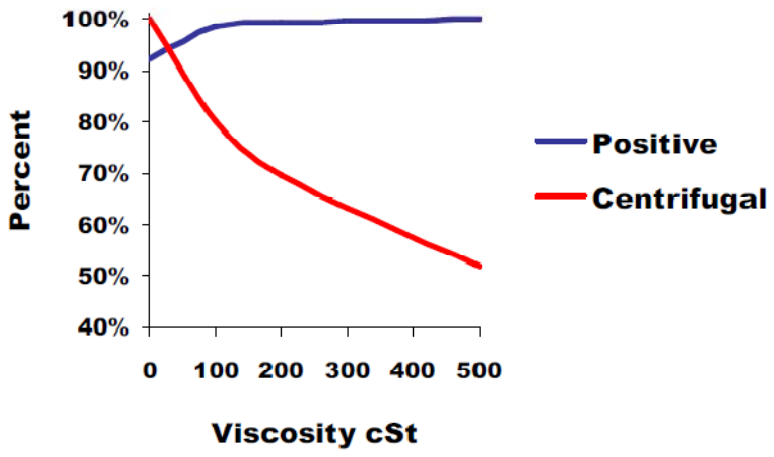

Figure 35. Flow Rate percentage vs. Viscosity of fluid [49].

Similarly, the efficiency of the pump behaves differently with an increase in pressure and as explained earlier, the positive displacement pump operates based on a given volumetric flow rate and is not readily affected by pressure fluctuations. Whereas a centrifugal pump has an optimum pressure point or head where the system is designed to operate but if the system pressure or head value is changed the efficiency of the pump will rapidly decrease. This is shown in Figure 35 where efficiency of the pump is compared to the amount of system head seen by the pump.

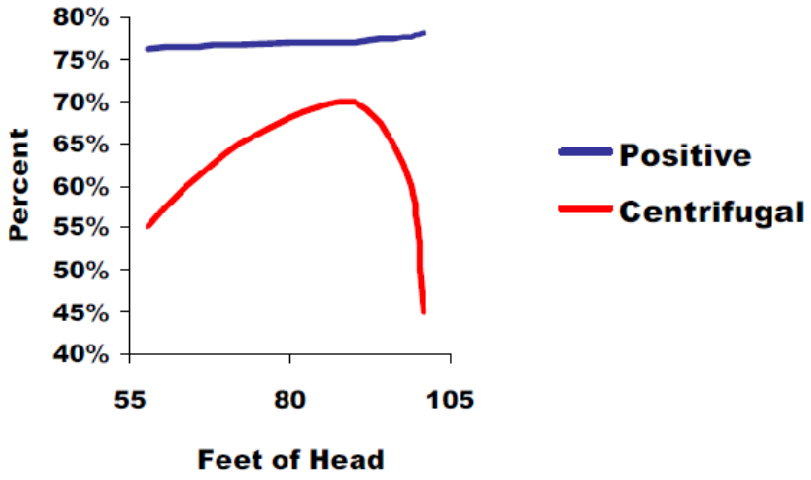

Figure 36. Efficiency vs. Head for PD and Centrifugal Pumps [47].

While the previous discussion leads to the belief that a positive displacement pump should be used for every application, it is only desirable under certain operating conditions. Centrifugal pumps are more widely used and generally incur lower maintenance costs because they possess fewer moving parts, do not require check valves, do not create large pressure pulsations, no rubbing contact with the pump rotor, and does not undergo fatigue loading due to reciprocation. The following list shows some certain applications and key criteria in which a positive displacement pump should be used over a centrifugal pump [70].

- High viscosity
- Self-priming
- High Pressure
- Low Flow
- Low shear
- Sealless pumping
- Constant flow/variable system pressure
- Two phase flow

Table 6 summarizes the previous points brought out and compares important operational functionalities between centrifugal pumps, reciprocating pumps and rotary (screw) type pumps [6].

Table 6. Comparison of Centrifugal and Positive Displacement Pumps [6].

Parameter	Centrifugal Pumps	Reciprocating Pumps	Rotary Pumps
Optimum Flow and Pressure Applications	Medium/High Capacity Low/Medium Pressure	Low Capacity High Pressure	Low/Medium Capacity Low/Medium Pressure
Maximum Flow Rate	100,000+ GPM	10,000+ GPM	10,000+ GPM
Low Flow Rate Capability	No	Yes	Yes
Maximum Pressure	6,000+ PSI	100,000+ PSI	4,000+ PSI
Requires Relief Valve	No	Yes	Yes
Smooth or Pulsating Flow	Smooth	Pulsating	Smooth
Variable or Constant Flow	Variable	Constant	Constant
Self-priming	No	Yes	Yes
Space Considerations	Requires Less Space	Requires More Space	Requires Less Space
Costs	Lower Initial Lower Maintenance Higher Power	Higher Initial Higher Maintenance Lower Power	Lower Initial Lower Maintenance Lower Power
Fluid Handling	Suitable for a wide range including clean, clear, non-abrasive fluids to fluids with abrasive, high-solid content. Not suitable for high viscosity fluids Lower tolerance for entrained gases	Suitable for clean, clear, non-abrasive fluids. Specially-fitted pumps suitable for abrasive-slurry service. Suitable for high viscosity fluids Higher tolerance for entrained gases	Requires clean, clear, non-abrasive fluid due to close tolerances Optimum performance with high viscosity fluids Higher tolerance for entrained gases

For the thermal heating fluids discussed, certain manufacturers recommend the use of a centrifugal type pump over a positive displacement. The Dow Chemical Company has published a guideline that describes the recommended hardware and assembly of the entire system. In this system guideline, it is recommended to use heavy duty, centrifugal process pumps made from cast steel [61]. The manufacturers of Therminol XP, Solutia, also have published a design guide for the heat transfer fluid. This document also recommends that a centrifugal pump made from cast steel be used [57].

Chapter 5

Potential Uses for Waste Heat

A major concern with implementing a waste heat recovery system is finding possible end uses for the recovered energy. The following sections discuss potential opportunities that can use this recovered energy instead of consuming purchased energy which will reduce the overall cost of operation. These recommendations are not the only possible opportunities and are only meant to offer starting points for design.

Waste Heat Boilers

A waste heat boiler is a separate system from the original combustion process which receives the hot exhaust gas and sends it through a high temperature heat exchanger to create low pressure steam. The waste heat boiler can also be supplemented with an additional burner in the instance that there is not enough energy available in the exhaust to create steam. This system is useful if there is a demand for low pressure steam that can be supplemented by this waste heat boiler, however in most cases should not be the only source of steam [4].

Preheating Combustion Air

For the combustion process to occur, the fuel and air mixture must be maintained at or above the ignition temperature, otherwise the combustion will extinguish. When the incoming air used in the combustion process is brought in at room or outdoor temperature, it must be heated from this temperature by the fuel combusting. This requires extra energy to be consumed unnecessarily to maintain the combustion process. To help minimize this waste, the incoming combustion air can

be preheated using a simple heat exchanger design which increases the incoming temperature closer to that of the exhaust gas temperature. This increase in temperature directly results in an energy savings because less energy from the combustion process is wasted on raising the initial air temperature. The following figure shows the energy savings potential in a 5.0 MMBtu/hr combustion process with an exhaust gas temperature of 400°F when the incoming air in increased from 40°F to 200°F.

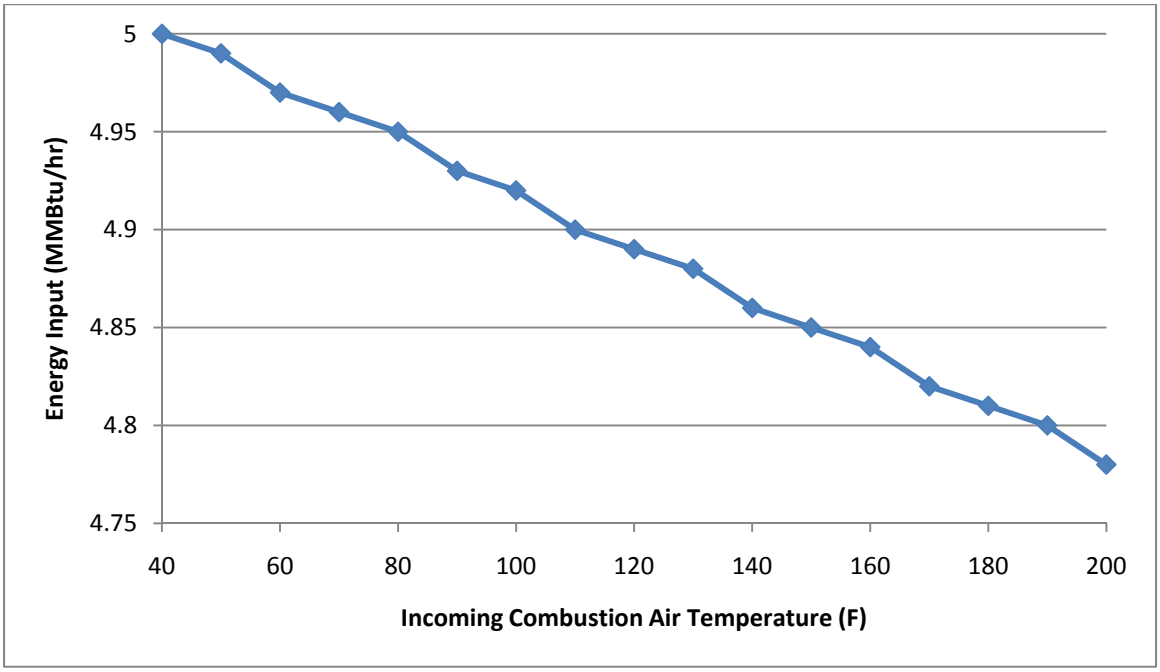

Figure 37. Energy Savings Potential by Preheating Combustion Air.

As can be seen in the previous figure, the energy input required to reach the same combustion results steadily decreases as the incoming air temperature is raised. This simple analysis shows that if the combustion process is operating all 8,760 hours a year, this increase in combustion air temperature can save the process almost 2,000 MMBtu/yr with no reduction in performance.

Preheating Feed Water for Boilers

One of the most common methods of recovering waste heat in a steam generation system is to install an economizer (heat exchanger) into the exhaust stack that will heat the incoming feed water to the system. This method takes some of the heat energy that would be released into the atmosphere and runs it through a gas to liquid heat exchanger that then heats the incoming feed-water before it enters the section of the boiler where it is converted into steam. This process can also be achieved directly by a transport membrane condenser. By implementing either of these systems, the amount of energy that the burner is required to input into the water to create the desired steam quality is reduced. The same basic concept of energy savings for preheating the combustion air pertains to preheating the boiler feed water. This is because the temperature of the water must be raised to a specified temperature to create the desired steam pressure and quality. This temperature increase of the water will require less energy when the incoming feed water is maintained at a higher temperature.

Load Preheating

When there is a nearby process that requires a component to be brought to a certain temperature before it can be used or if the heating a component will increase the efficiency of the process, load preheating can be used. This can be accomplished in a number of ways but mostly consists of simply blowing the exhaust gas directly on the component to be heated. This is not always possible if the materials are not capable of withstanding the components of the exhaust gas, in which case a heat exchanger can be installed to transfer the heat [4]. Common applications for this include removing the moisture from green wood used as furnace fuel and removing moisture from metals entering a melting furnace.

Space Conditioning

Another use for waste heat from industrial processes is for comfort space conditioning. Depending on the source of the waste heat, it can be used either directly or indirectly for heating a space. If the waste heat is from a clean source such as cooling air from a process that is simply blown over a heat exchanger and never comes into contact with a contamination source, it can be directly ducted into the area and blown into the space providing warm air to the surroundings. However if the waste heat is coming from a dirty source or a source with harmful contaminants present, such as the exhaust air from the products of a combustion reaction, the energy must first be recovered using a heat exchanger, ideally a condensing heat exchanger. The heated liquid can then be transported to a source which consists of a fan and coil to force clean air across a heat exchanger with the warm liquid inside and sends the clean, heated air to the space. By recovering this waste heat and using it for space conditioning, either energy can be saved by offsetting heat supplied to a space by another energy source or increase the comfort of employees.

Although it may sound counterintuitive, waste heat can also be used for space cooling. This can be accomplished by the use of an absorption chiller to convert the hot wasted heat energy into a cooling effect. The details of how an absorption chiller or heat pump operates is discussed in detail in the previous section, but basically the absorption chiller mixes two specific chemicals which when combined create a cooling effect. The waste heat energy is then used to separate the two chemicals from one another so the process can be repeated. Once the cooling effect occurs, the absorption chiller becomes just like an ordinary chiller and the cold fluid is passed through a heat exchanger where air can be passed through and ducted to be used as space cooling throughout the facility.

Hot Water

The vast majority of facilities have some use for domestic hot water whether it is for washing hands or used in a cafeteria. This system is generally of relatively low demand and can be approximated to be about one gallon of use per person per day. This leads to low total energy consumption when compared to a large combustion furnace. However, this energy consumption is not negligible and waste heat can be used to eliminate the need for a separate hot water heater and therefore eliminate the natural gas or electrical consumption associated with the system.

Another possible end use for generating hot water from waste heat can include process hot water. There are various processes that can require this type of warm to hot water including washing baths or dying equipment. These pieces of equipment can cause a reasonably high volume demand for hot water. This becomes a perfect candidate for producing the water from waste heat to reduce the purchased energy consumption required.

Chapter 6

Example System Design

The following chapter describes an example system created and the design steps required. The first step in designing a heat recovery system is to determine the amount of energy that is available in the exhaust stream which can be recovered. This can be accomplished using a simple first law of thermodynamic analysis on the flow of exhaust gas using reasonable temperature drops through the heat exchanger types used. Once the amount of energy available has been determined, the demand for the system should be calculated using different methods to find the maximum amount of energy required throughout the year. After the total energy recovered and the total system demands have been determined, the appropriate type of, or even need for, thermal storage in the system can be determined. The next step is to determine supply and return temperatures along with flow rates of the system. This information is then used to assist in the design of the next two systems, heat exchangers and pumps. The heat exchangers that must be designed initially include the sensible heat exchanger in the exhaust stream along with the condensing economizer. The pumps can be selected, one for each loop, using the information already known about the system flow rates and the head that the pump must overcome. These two system designs can be aided from software capable of designing and giving performance characteristics of each.

Once the system has been properly designed, the economic side of the project must be determined. To estimate the implementation cost of the entire system, cost estimating handbooks as well as online retailers and supplier quotes can be used to find the approximate purchasing and labor cost for each component. The amount of money that is saved from implementing the system can then be found by determining what source and the amount of energy which is being offset, in this case natural gas or electricity. From this information and an average energy cost for each of the sources, an estimated monetary savings can be found. This information together can be used to determine a simple payback of the system in years.

System Description

For this analysis, an actual industrial facility was chosen to simulate a realistic heat recovery situation. This facility contains two 400 hp natural gas fired boilers with one in operation continuously and one in standby as a backup. For the analysis, the system is assumed to operate with one boiler in fully loaded operation for all 8,760 hours a year. The boilers are used to create 120 psig steam air for manufacturing high precision lost foam casting molds. The exhaust gas conditions were measured and found to average 485°F with a flow rate of 4,050 cfm. This airflow rate was found from a simple combustion chemistry analysis of the system based on a measured O_2 exhaust gas content of 2.2% and the described exhaust temperature. Under these conditions the boilers are estimated to consume around 55,000 MMBtu a year in natural gas. The facility layout can be seen in the following drawing. The facility consists of a bead storage room which is maintained at a temperature of 70°F year round and, because of noxious gas being present, air is circulated regularly.

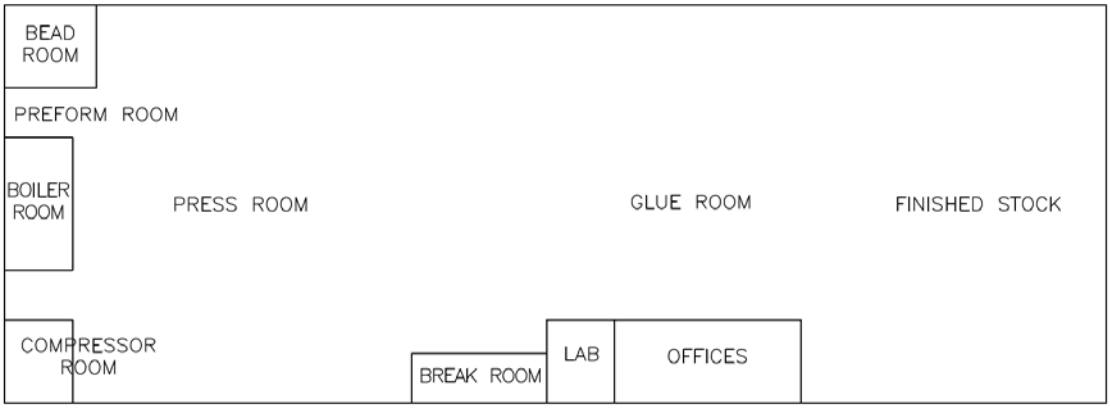

Figure 38. Example Facility Layout.

The beads are taken to the preform room where they are prepared to be sent into the press room. In the press room, specific machinery which uses once through steam, presses the beads together and fuses them creating a mold. Once the beads have been formed into the molds, they must immediately be cooled to prevent over expansion. This is accomplished by dropping the patterns into a bath of cooling water. For more complex molds, the manufactured molds must be glued together. This uses a type of glue similar to "hot glue" which must be maintained at a certain temperature or it will solidify. After the materials have been processed and randomly inspected in the lab, they are sent to the finished stock area.

The facility contains several opportunities to use heat energy, however, there is no current system in place and the logistics of the facility are very spread out. This facility contains opportunities in year round cooling by the use of an absorption chiller in the bead room, space conditioning in the offices, lab and break room areas, domestic hot water for the facility, along with a system to maintain the temperatures of the glue baths. Another major energy savings can be seen by installing a condensing economizer or transport membrane condenser to preheat boiler feed-water. This is because due to the process requirements, steam is simply blown across the molds and exhausted from the building which means that only about 10% of the total steam

mass flow is returned as condensate and the remaining 90% must be made up with water at a much lower temperature.

Design of a System

Energy Content Recoverable

The first step in designing the system is to determine the amount of energy available in the exhaust gas stream. This is helpful to determine a first order analysis of the potential magnitude that can be seen by recovering the waste heat and to determine how many of the proposed projects are feasible to be implemented at once. This can be easily calculated using the first law of thermodynamics and by making the assumption that the exhaust stream temperature is reduced from the current temperature of 485°F to a temperature slightly above the dew point temperature of 255°F. This is the amount of energy that can be recovered by a typical heat exchanger and will be the amount of energy supplied to the fluid in the waste heat recovery system.

$$Q_{available} = \dot{V}_{ex} \times \rho_{ex} \times C_{p,ex} \times (T_{hot} - T_{low})$$

$$= 4{,}050 \text{ cfm} \times 60 \text{ min}/1 \text{ hr} \times 0.045 \text{ lbm/ft}^3 \times 0.28 \text{ Btu/lbm-°F} \times (485 \text{ °F} - 260 \text{ °F}) \times 1 \text{ MMBtu}/10^6 \text{ Btu}$$

$$= 0.69 \text{ MMBtu/hr}$$

where

$Q_{available}$ = amount of energy in the exhaust stream for desired temperature range

$\dot{V}_{ex}$ = volumetric flow rate of exhaust gas stream

ρ_{ex} = density of exhaust gas stream at 485°F

$C_{p,ex}$ = average exhaust stream specific heat

T_{hot} = temperature of the initial exhaust gas stream

T_{low} = temperature the exhaust gas is to be cooled to after heat exchanger

The energy that can be recovered in by the condensing economizer can be calculated in a similar fashion but also considering the latent heat recovered from condensing the water vapor from the exhaust stream. This condensed water vapor will be used to supply hot feed-water into the boiler while the remainder of the energy recovered will be added to the a second fluid transport system. This process is shown below and assumes the condensing economizer will reduce the exhaust temperature from the previous heat exchanger exhaust temperature to 135°F and that all the moisture added during the combustion process will be condensed.

Q_{cond} = m_{ex} x $C_{p,ex,2}$ x ($T_{hot,2}$ - $T_{low,2}$) + LH x WV x FR

= [10,935 lbm/hr x 0.28 Btu/lbm-°F x (260 °F – 135 °F) + 970.3 Btu/lbm H_2O x 41.9 lbm H_2O/ 1 MMBtu NG x 12.55 MMBtu NG/hr] x 1 MMBtu/10^6 Btu

= 0.9 MMBtu/hr

where

Q_{cond} = amount of energy in the exhaust stream for desired temperature range

$\dot{m}_{ex}$ = because of conservation of mass, the same mass flow rate must enter the second heat exchanger as the first

$C_{p,ex,2}$ = average exhaust stream specific heat

T_{hot} = temperature of the inlet exhaust gas stream (from previous HX)

T_{low} = temperature the exhaust gas is to be cooled to after heat exchanger

LH = latent heat of vaporization of water

WV = amount of water vapor produced during the combustion of 1 MMBtu of natural gas

FR = firing rate of the boiler

Energy Demand for System

The next step is to calculate an estimate for the energy demand for each of the possible uses for this recovered energy. The design energy demand for each component should be calculated, however because previous equipment is already in place, the waste heat recovery system is not designed to replace the current infrastructure but rather to supplement the system. This allows for the current equipment to be turned off or idled back to save on energy costs. If a new system is being designed, care should be taken to ensure the waste heat source will be available throughout the operating hours of the system components and otherwise should be used as a supplemental heat source. For the system considered, the components should rarely be operated at the same time. However to avoid removing heat from the components, it is recommended to install thermostatically controlled valves in each system to ensure the appropriate temperature fluid required for each process is met by the incoming flow.

In this case the main demand in the system is space conditioning but also includes domestic hot water generation and process heating. To calculate the heating demand for the office, lab and break room space conditioning, the existing equipment was considered. To ensure this equipment was appropriately sized, a simple heat load calculation was performed using a spreadsheet that uses basic ASHRAE procedures [17]. The results of the model are a total capacity of 0.30 MMBtu/hr which matches up with the current installed capacity of 0.33 MMBtu/hr. This required demand for the system will be taken as the installed capacity. However if the current operation is not known or a more accurate answer is desired, a complex

building model can be built using one of several software suites to determine the building heating loads.

A similar type of analysis can be done to determine the amount of energy required to maintain the temperature of the bead room at a maximum temperature of 75°F year round. This can also be estimated by using the same spreadsheet cooling load model and results in a design cooling load of the room to be around 7 tons while the installed capacity is 8 tons. The spreadsheet calculation used several assumptions and the 8 ton value is taken to be correct. However, as mentioned earlier, the smallest capacity absorption chiller commercially available is 10 tons which will be the capacity installed.

To supply cold air to the room from the waste heat source, an absorption chiller must be used. Because of this, the coefficient of performance for the chiller must also be considered. To develop a first round estimate, a value for the COP is estimated based on the average values discussed earlier and product literature to be 0.7.

$$Q_{WH} = CC \times K_1 / COP \times K_2$$

$$= 10 \text{ tons cooling} \times 12{,}000 \text{ Btu/hr/ton} / 0.7 \times 1 \text{ MMBtu}/ 10^6 \text{ Btu}$$

$$= 0.17 \text{ MMBtu/hr}$$

where

Q_{WH} = required waste heat input to the chiller

CC = design system cooling capacity

K_1 = conversion from tons of cooling to Btu/hr

COP = coefficient of performance, assumed based on average data

K_2 = conversion from Btu to MMBtu

To determine the amount of energy required to supply the facility with domestic hot water, another thermodynamic analysis is considered. For this analysis, the amount of hot water consumption must be known. To determine this, as mentioned previously, it has been estimated that each employee will consume one gallon each day. For the facility considered, there are 35 employees that work for eight hours each day. This leads to an average consumption of about 5 gallons per hour. The incoming water temperature is also assumed to be an average of 55°F and heated to a temperature of 110°F.

$$Q_{water} = \dot{V}_{water} \times \rho_{water} \times C_{p,wat} \times (T_{hot} - T_{low})$$

$$= 5 \text{ gallons/hr} \times 0.134 \text{ ft}^3\text{/gallon} \times 62.2 \text{ lbm/ft}^3 \times 1.0 \text{ Btu/lbm-°F} \times (110°F - 55°F)$$

$$= 0.002 \text{ MMBtu/hr}$$

where

Q_{water} =amount of energy required to create demand of domestic hot water

$\dot{V}_{water}$ = volumetric flow rate of domestic hot water

ρ_{water} = average density of water

$C_{p,water}$ = average specific heat of water

T_{hot} = final temperature of domestic hot water

T_{low} = average inlet water temperature to the hot water

The complexity in this system comes in with the glue baths. For this analysis it will be assumed that the system can be heated by installing a tubular heat exchanger on the outside of the bath with an insulation backing. The energy input required to maintain the temperature above the melting point can be calculated and this amount will be taken as the required waste heat energy. The calculation is made by arguing that to maintain the current temperature within the glue

baths, energy must only be added to make up for that which is lost through convection and radiation. The glue is required to be maintained at a temperature of 285°F to prevent it from solidifying. With this as the internal temperature, the outside surface temperatures can be calculated. The outside surface temperature of three of the four outside surfaces is estimated to be the same temperature which can be found to be 83°F. The surface temperature for the remaining side which includes the heat exchanger operating at a temperature of 285°F located in the space between the tank and the insulation to supply the energy into the glue bath is found to be 95°. The outer surface temperature of the insulation was calculated using a computer software program published by the North American Insulation Manufacturers Association (NAIMA) called 3E Plus [46] which is also able to calculate the heat loss and the surface temperature of different types of surfaces and uses a database of many different types of insulation. This calculation takes into account both convection and radiation heat losses from the surface.

To more accurately predict the heat loss and in turn the amount of heat addition required, the tank was broken as previously described with three of the four outer sides having an interior surface temperature of 175°F. The one side containing the waste heat energy addition heat exchanger's surface temperature was fixed to the supply temperature of 285°F and the top is assumed to be open to the atmosphere. These operating temperatures were used to calculate the outer surface temperature and heat loss. The following calculation shows the total surface areas and the total amount of heat energy that must be added to the glue baths to maintain the temperature.

Q_{baths} $= N_{baths} \times (N_{SA,1} \times SA_1 \times Q_{loss,1} + SA_2 \times Q_{loss,2} + SA_3 \times Q_{loss,3} + SA_4 \times Q_{loss,4})$

$= 14 \times (2 \times 2.9\ ft^2 \times 15.27\ Btu/hr\text{-}ft^2 + 3.4\ ft^2 \times 15.27\ Btu/hr\text{-}ft^2 + 3.4\ ft^2 \times 39.66\ Btu/hr\text{-}ft^2 + 8.44\ ft^2 \times 513.9\ Btu/hr\text{-}ft^2) \times 1\ MMBtu / 10^6\ Btu$

$= 0.065\ MMBtu/hr$

where

N_{baths} = total number of glue baths in the facility

$N_{SA,1}$ = number of sides on each bath with this operating condition

SA_1 = surface area of one side of glue bath

$Q_{loss,1}$ = heat loss from this surface, calculated with 3E Plus

SA_2 = surface area of one side of glue bath

$Q_{loss,2}$ = heat loss from this surface, calculated with 3E Plus

SA_3 = surface area of one side of glue bath

$Q_{loss,3}$ = heat loss from this surface, calculated with 3E Plus

SA_4 = surface area of one side of glue bath

$Q_{loss,4}$ = heat loss from this surface, calculated with 3E Plus

Thermal Energy Storage Type Selection

From the previous calculations done, the appropriate type of thermal storage for the system can be selected. There are three main system design categories that are possible. These system designs are:

- No thermal storage
- Thermal energy recovered is sent directly to a storage tank where a separate system is then used to transport the thermal energy throughout the facility

- Thermal energy is recovered and distributed throughout the facility and thermal storage is added at the end of the loop to capture and store any remaining thermal energy

Each of these three design scenarios have their advantages and disadvantages and can be used in different scenarios depending on the particular system requirements. The first design is the cheapest and easiest to implement because a thermal storage system is not required to be purchased or installed. However if this system is not designed perfectly, if the operating conditions of the waste heat source or the end uses changes, then the system's energy transportation fluid will continue to be returned at a higher temperature and will not be able to absorb as much energy from the waste heat stream. This will lead to a decrease in effectiveness of the overall system and limit the amount of energy and cost savings that can be realized. The second energy transportation system design can become complex and requires a large thermal storage system which is adequately sized and capable of capturing and maintaining all the heat energy recovered until it is required elsewhere. By storing all of the heat energy directly as it comes from the waste heat source, the design allows for a non-constant end-use demand as well as a cycling waste heat source. However, this system requires a secondary loop to remove the energy from the storage system and transport it to the end uses and consequently will require more pumping horsepower and additional heat exchangers to operate. To effectively use the thermal storage tank, a separate system in the plant with an intermittent demand is needed. This system can then periodically remove energy and "drain" the tank while using the energy in a productive manner.

The third system design becomes a mixture of the pros and cons of the two previous systems. Similar to the second system design discussed, this design requires a separate secondary loop to transport the energy recovered by the storage tank. However the size requirements for this loop,

as well as the size of the storage tank will be much less. This is due to the energy magnitude that will be introduced into the storage system. Also, because of the secondary loop, this system will require additional pumping horsepower to operate. Although this system's horsepower requirements will be much less than the second design option because this secondary loop will possess a comparatively much smaller volumetric flow rate. The major drawback with this system, along with the second design, is the need for an intermittent end-use demand to effectively make use of the thermal storage as well as by placing the energy storage at the end of the loop, a thermostatically controlled bypass system will be needed. This bypass loop would allow the thermal heating fluid in the system to bypass the thermal storage tank if its temperature were too low and to prevent the fluid from undesirably recapturing the energy stored in the energy storage system.

The following chart summarizes some of the major advantages and disadvantages for each of the systems.

Table 7. Advantages and Disadvantages for Each System.

No Storage		**Initial Storage**		**End Storage**	
Advantages	**Disadvantages**	**Advantages**	**Disadvantages**	**Advantages**	**Disadvantages**
Lower Implementation cost	Loss of energy Recovery Effectiveness	Constant Waste Heat Supply not required	Large Storage system Required	Smaller Storage System Required	Need for an intermittent demand end use
Lower operational costs		Constant end-use demand not required	Secondary Loop Required	Constant Waste Heat Supply not required	Secondary Loop Required
		Can Recover and use all of the energy recovered when needed	Larger Pumping HP Required	Constant end-use demand not required	Bypass Loop around Storage Required
			Larger Heat Transfer Fluid Volume will be Required	Less Pumping HP than Initial Storage Design	
				Less Heat Transfer Fluid Volume Required	

The remaining capacity for each loop can be stored in an energy storage tank and used to supply intermittent, high demand systems. For the higher temperature energy recovered, the process that will receive the additional, stored capacity operates for two hours at a time, once at noon and again at midnight. This system requires a high demand of hot water and will be supplemented by the storage tank while the remaining demand will be met by an additional natural gas fired hot water heater. The excess energy stored from the lower temperature, condensing economizer storage tank will be used to offset costs of the cleaning process of equipment at the end of each shift where the use of hot water is required.

The previous analysis has shown that it is possible to operate the absorption chiller for the bead room on only the sensible energy recovered from the condensing economizer. This energy recovered will provide the minimum supply temperature of 266°F to the chiller and allow for sufficient energy to achieve nominal ratings. This fact leads the design to add an additional loop in the system for solely the condensing heat exchanger energy recovered. This separate system will lead to a lower implementation cost for each energy storage tank and less total flow rate through the other system. This type of design will also allow for a higher operating temperature to be achieved by the first heat exchanger loop because the two streams will not become thermally mixed. This will accommodate a wider range of possible end uses by supplying higher grade energy to systems that require this type of energy while still providing sufficient energy and temperature to other processes. On this separate loop, the sensible energy will be directed from the condensing economizer into the heat transfer fluid which will be pumped to the absorption chiller where the required energy will be used in this process. The remaining energy content in the heat transfer fluid will be sent through a plate-and-frame heat exchanger to preheat the incoming feed-water to the boiler and then through the heat storage tank to recover any

remaining energy. The plate-and-frame heat exchanger is chosen for this application due to its higher liquid-to-liquid heat transfer characteristics as discussed in Chapter 3 This feed-water preheating heat exchanger will be located prior to the location where the recovered condensed water is returned to the feed-water source to insure the maximum amount of energy can be recovered in the system.

Fluid Selection, Temperature and Flow Rates Design

For the example system developed, the second and third design options discussed in the previous section are selected and a waste heat source was chosen. The flow rate and temperature requirements for the system must then be determined. For the main loop, the maximum temperature required in the system is 285°F by the glue baths. To ensure that the fluid is delivered to the glue baths at a high enough temperature, taking into account the heat losses through the piping network, the temperature is set at 300°F. From this temperature setpoint, the mass flow rate of the main loop can then be calculated using the first law of thermodynamics. The energy available in the system was previously calculated by the energy recovered from the initial heat exchanger and the average densities and specific heats are known. For this loop, DOWTHERM G is used as the thermal fluid, and over the temperature range the average density is 61.2 lbm/ft^3 and the specific heat is 0.44 Btu/lbm-°F. The return temperature and flow rate can then be estimated to be 225°F and 30 gpm based on the energy consumption in the process. For the absorption chiller loop the flow rate can be calculated a similar way however the absorption chiller requires a minimum flow rate and temperature to operate effectively. The operating fluid for this loop is chosen to be UCON HTF 500 because of its better low temperature heat storage performance. By taking the thermodynamic analysis and the minimum requirements of the

chiller into account, the flow rate is found to be 35 gpm at a temperature of 225°F. The analysis was completed for each loop and the design flow rate and temperatures are shown below.

Table 8. Design Temperatures and Flow Rates for each Loop.

Loop	Flow rate (gpm)	Operating Supply Temperature (F)	Design Return Temperature (F)
Absorption Chiller	35	225	90
HX1 Sensible Heat Loop	130	400	225
Main Loop	30	300	225
Intermittent Loop	25	225	90

Piping Network Design

After the flow rates have been determined, the piping network and layout can be developed. This piping system is shown in Figure 38 and shows the general locations of the hot fluid supply piping (red) and fluid return piping (blue) with the minimum amount of thermostatically controlled valves for each section. These control valves are shown on the picture as a circle around the pipe. All piping shown is 2 inch standard schedule 40 seamless steel piping and should be welded together as recommended by the heat transfer fluid manufacturers [57]. The figure also shows the recommended design components based on the design parameters provided by the thermal heating fluid manufacturers including a thermal expansion tank bypass with a vent to minimize the risk of system pressurization and pump cavitation. This expansion tank will reduce difficulties in start-up and contribute to the system reliability. The system also consists of two thermal storage tanks consisting of phase change materials to store energy and transfer for current or later uses.

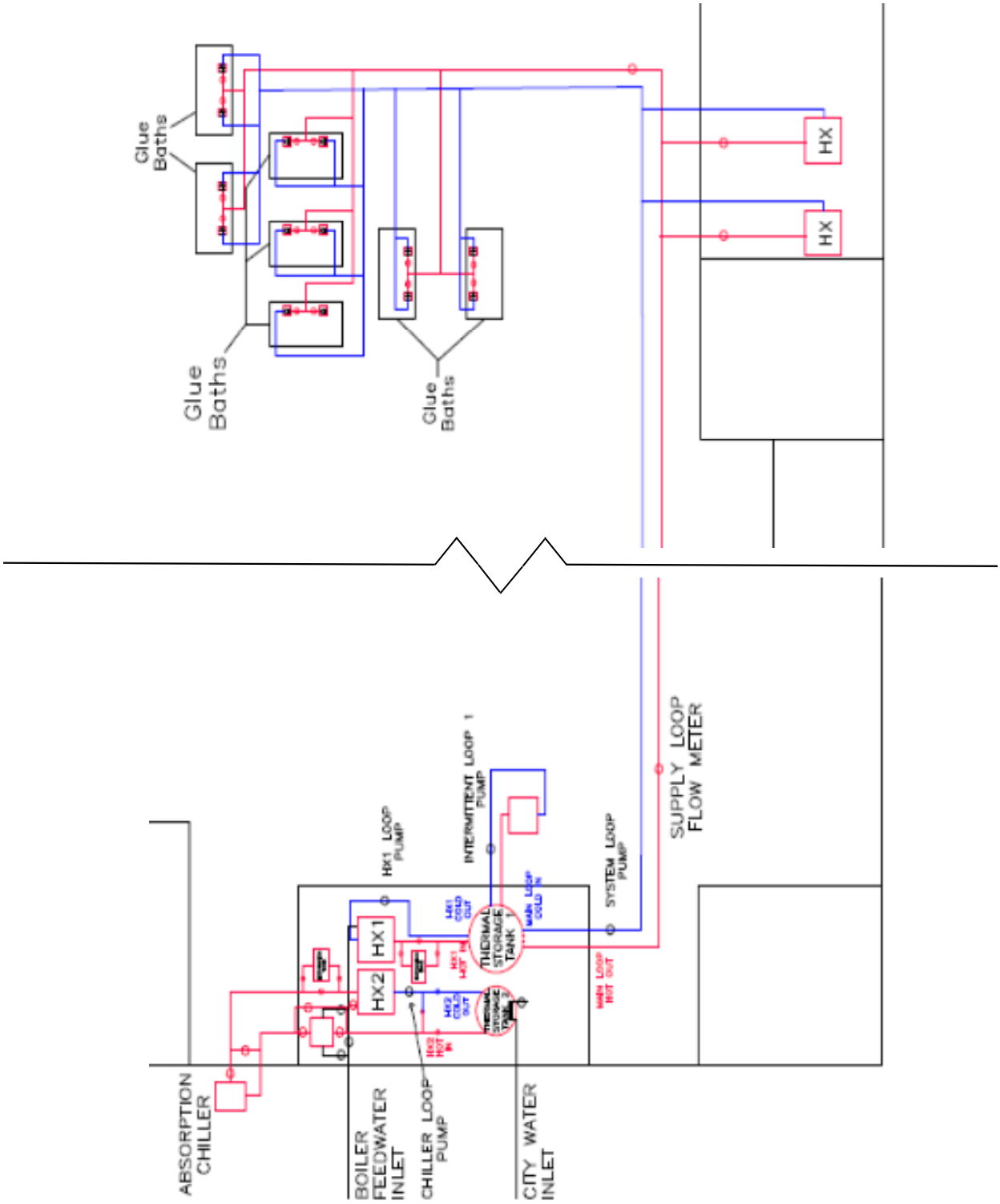

Figure 39. Piping Layout for Heat Recovery System.

Heat Exchanger Design

The next step in the process is to design the heat exchangers required to supply the appropriate amount of energy to the end processes. The selection criterion for which type of heat exchanger is to be chosen for each type of process is discussed in previous sections. To design an appropriate heat exchanger for each application, a computer software program developed by the Heat Transfer Research Institute titled *HTRI Xchanger Suite Educational 6.0* is used. The software allows the user to input detailed parameters about the entire system and fluids then, from this information, is able to develop a heat exchanger design [30].

The first heat exchanger designed is the initial economizer in the exhaust stack. This heat exchanger is required to recover sensible energy from the stack gas and lower its overall temperature from 485°F down to 260°F while increasing the thermal heating fluid from 225°F to 400°F. Because this application is for transferring energy from a gas to a fluid, a countercurrent, cross-flow, shell-in-tube heat exchanger is selected. From this basic design criterion, the following heat exchanger was designed and is shown in the following figure.

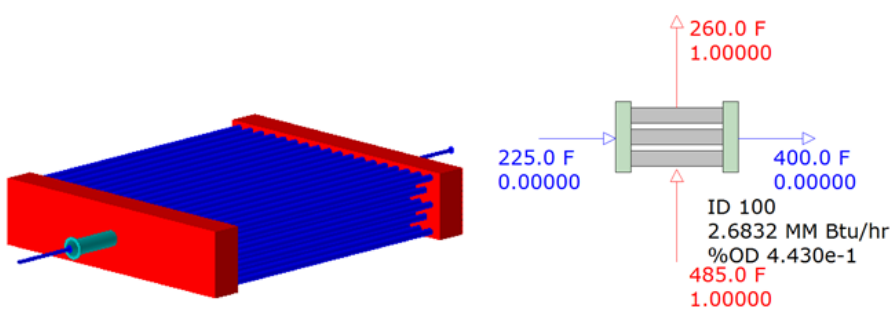

Figure 40. HX1 Design Model and Schematic.

The heat exchanger has an overall tube length of 3 feet which is able to fit inside the exhaust stack horizontally. The heat exchanger consists of 5 total tube rows which are 3 feet wide. The heat exchanger contains a total of 460 ft^2 heat transfer surface area with 0.875 inch diameter

tubes arranged in a staggered formation with a transverse pitch of 2 inches and a longitudinal pitch of 1.75 inches. The program is able to calculate an actual heat transfer coefficient for both inside and outside the tubes. This value for each is found to be 11.65 Btu/ft^2-hr-F and 10.06 Btu/ft^2-hr-F respectively under the design conditions. The heat exchanger is designed to recover the 0.69 MMBtu/hr that is calculated previously and is accomplished with the previous dimensions and style of heat exchanger with a percent over design of less than 1%.

The second heat exchanger that must be designed is the condensing economizer which is shown in the piping layout figure as HX2. This heat exchanger is used to reduce the temperature of the exhaust gas from the previous heat exchanger exit temperature of 260°F to 135°F, a temperature below the gas dew-point temperature, which in turn will condense the water vapor from the system. This economizer design schematics are shown in the following figure.

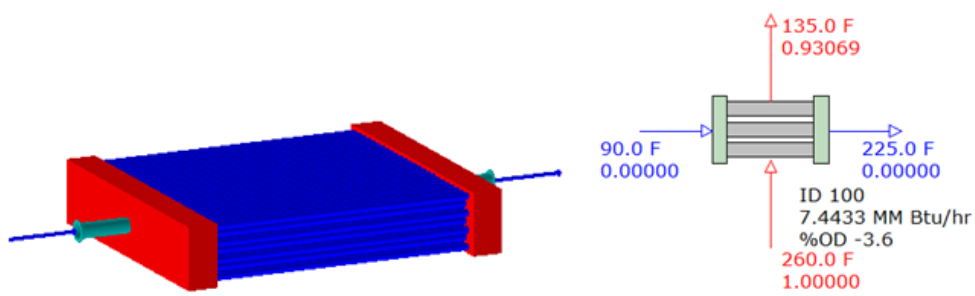

Figure 41. Condensing Economizer Design Model and Schematic.

The heat exchanger has an overall tube length of 3 feet just as the previous heat exchanger which ensures it will fit inside the exhaust stack horizontally. The condensing heat exchanger consists of 7 total tube rows which are spread evenly at a pitch of 1.4 inches in the transverse direction and 1.1 inches in the longitudinal direction. This formation is repeated for the full width of 3 feet. The heat exchanger contains a total of 1,500 ft^2 heat transfer surface area with 1 inch

diameter tubes arranged in a staggered formation. The program is able to calculate an actual heat transfer coefficient for both inside and outside the tubes. This value for each is found to be 6.3 Btu/ft^2-hr-F and 14.93 Btu/ft^2-hr-F respectively under the design conditions. The heat exchanger is designed to recover the 0.9 MMBtu/hr that is calculated previously and is this accomplished with the previous dimensions and style of heat exchanger with a percent under design of 3.6%. This means that 3.6% of the energy content that is to be recovered will not be absorbed into the heat transfer material or 0.032 MMBtu/hr are still exhausted to the atmosphere. The design performance information for this heat exchanger along with HX1 can be found in the Appendix.

After the energy is recovered from the stack it must be transported into the heat storage tanks. To accomplish this heat transfer, a series of piping loops are installed inside the tank that allows the hot fluid to flow through and transfer the energy into the phase change materials which are enclosed inside the tank. This type of heat exchange is chosen because the contents of the thermal storage tank are stationary and heat transfer through a traditional heat exchanger, a plate-and-frame or a shell-in-tube, is virtually impossible. To ensure that the appropriate energy is transferred into the system, a tubular heat transfer analysis can be conducted by knowing thermal properties of the piping materials and the phase change material. For the first heat exchanger loop, a total length of 105 feet must be used to transfer the appropriate amount of energy into the phase change materials. For the second heat exchanger loop, an additional 30 feet must be added.

Pumping Selection

Once the piping layout and the components of the system are known, the pumps can then be sized. To do this, a head loss analysis was conducted on the system and includes the associated losses from the heat exchanger, piping and valves and fittings. The system will require four

pumps to deliver flow throughout the system, one for each loop, however because the loops are relatively small, the required pumping power will also be small. Each loop's head loss and flow rate were calculated and shown below.

Table 9. Piping Loops Head Loss Results.

Loop	Flow rate (gpm)	Length (ft)	# Valves	# Fittings	# Heat Exchangers	Total Head (ft)
Absorption Chiller	35	100	5	18	3	20
HX1 Loop	130	50	5	9	2	120
Main Loop	30	840	19	38	17	80
Intermittent Loop	33	375	1	6	2	20

After the total head loss and the flow rates are known a pump can be selected for each loop. To assist in selecting a pump a computer software program was used which searches through a database of existing pump curves for a specified manufacturer [50]. For this analysis, Goulds Pumps were selected as the manufacturer however there are other manufacturers that produce pumps at the appropriate size.

Using the information from Table 5 along with certain properties of the UCON HTF 500, the absorption chiller loop pump was selected. The most efficient pump at the operating conditions calculated was chosen which is an end suction centrifugal pump operating at 65.6% efficiency with a 4.625 inch diameter impeller and requires a third of a horsepower motor to operate. The following figure shows the operating point of the system loop along the chosen pump's curve. The blue line in the figure represents the system curve and demonstrates the effect of changing the head or flow rate in the piping network.

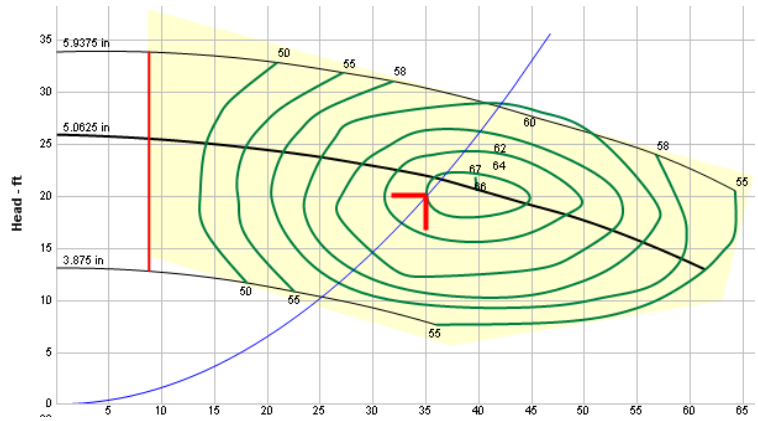

Figure 42. Absorption Chiller Loop Pump and System Curve [50].

The previous analysis was repeated for the HX1 loop, this is the loop that recovers the wasted energy in the initial heat exchanger and transports it into the thermal storage tank. This loop contains DOWTHERM G and the properties for this fluid must be entered into the program. Because the pumps have a lower allowable temperature than the operating temperature of 400°F, the pump is to be placed in the return line where the lower temperature of around 150°F can be seen. The most efficient pump for this loop was found to be a multistage centrifugal pump operating at 70.6% efficiency with a 5.75 horsepower motor power requirement to operate. The following figure shows the operating point of the loop along with its system curve imposed onto the pump curve.

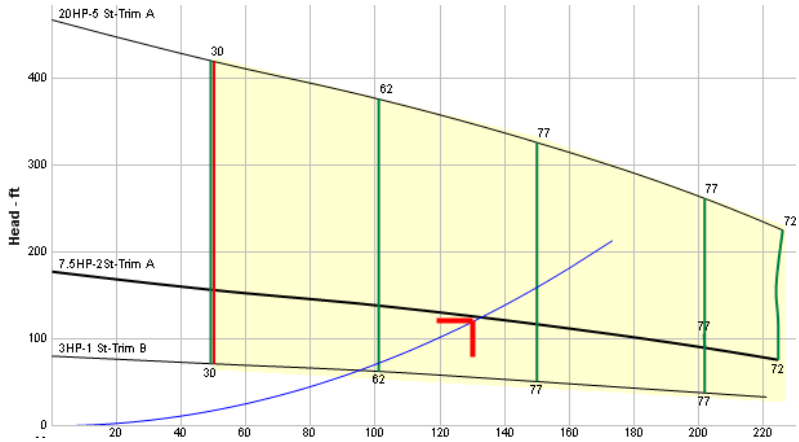

Figure 43. HX1 Loop Pump and System Curve [50].

The largest loop in the system is the main loop which transports the energy from the energy storage tank to create domestic hot water, produce comfort heating for the offices and to heat the glue baths. This loop contains DOWTHERM G heating fluid to transport the energy and again, because of the high temperature of the supply fluid, the pump is located in the return piping. The most efficient pump found is an end suction centrifugal pump operating at one and a quarter horsepower and a 4.875 inch diameter impeller. The pump has a best efficiency point of 64.5% but at the design conditions is operating at 59% efficient. Figure 43 represents the loop system curve in blue with the current operating point outlined by the red arrow. This is placed on top of the selected pump curve to show the operating efficiency of the pump.

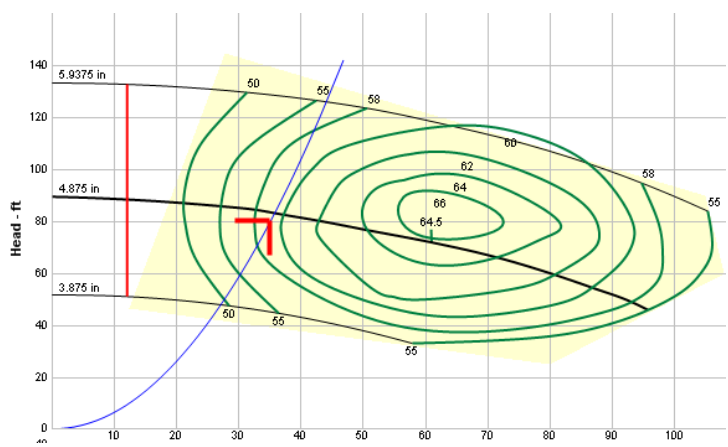

Figure 44. Main Loop Pump and System Curve [50].

The final pump that must be selected is for the larger intermittent loop. This pump will only be in operation for a limited time throughout the day which should be taken into account when deciding what magnitude of electrical efficiency is required. The low operating hours will more than likely not justify any extensive control system to be implemented on top of the basics. The pump that is chosen for the first heat exchanger loop is a centrifugal two stage end suction pump operating at an efficiency of 64.4% with a 5.0625 inch impeller. The motor power requirements

are just under a third of a horsepower to operate. The pump and system curves for this intermittent loop are shown together in the following figure.

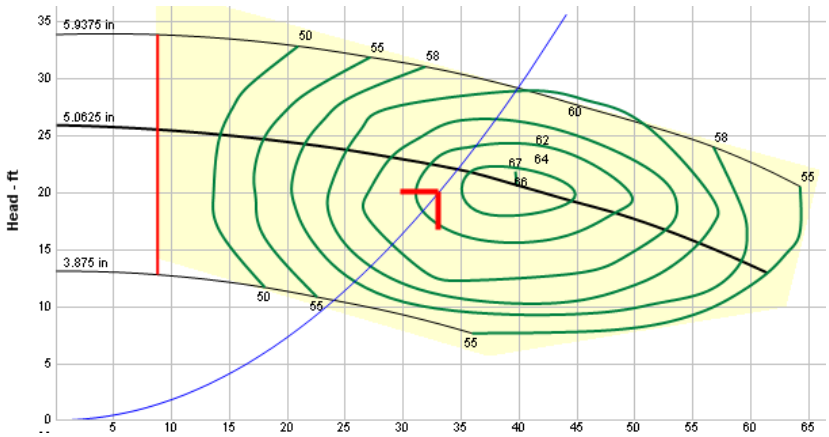

Figure 45. Intermittent Loop1 Pump and System Curve [50].

Thermal Storage Tank Design

The next step after the heat recovery potential, flow rates and temperature requirements have been determined is to design the thermal storage tanks. The larger of the two thermal storage tanks will be the one containing the initial stack heat exchanger loop. The main loop which this tank will be supplying must be maintained at an average temperature high enough to deliver heat energy to the system at 300°F. For this to occur, the tank should be filled with phase change materials that have a minimum phase change temperature of 300°F which can be found with commercially developed products. This is the minimum phase change temperature that can be used, however to store more energy, a phase change temperature closer to the supply temperature should be selected. A company called PCM Products Ltd. provides an organic phase change material that will change from a solid to a liquid at a temperature of 327°F with latent heat storage of 12,320 Btu/ft^3. This phase change material is the best energy storage option available for this process and although there are higher phase change temperature materials, this particular material is capable of storing more energy in the same volume as the other options. The size of

the thermal storage tank must also be considered and should be capable of storing enough energy to adequately supply the intermittent demand when required while still maintaining enough energy to supply the main, constant demand loop. For this system, the volume of the tank is estimated to be 750 gallons, giving a total storage capacity of 1.25 MMBtu which should be an adequate size to supply both required demands at a given time. The second storage tank will be located at the end of the absorption chiller loop which means it will receive the thermal heating fluid at a lower temperature. To adequately supply the intermittent hot water supply demand, the tank must be maintained at a temperature of about 130°F. Because the temperature of the fluid supplied to the tank is around this same temperature, a phase change material that changes phase at this temperature should be selected. Also because the tank is located at the end of the loop, a thermal bypass loop should be installed to ensure fluid is not delivered to the tank at a temperature lower than the current temperature of the contents of the tank. The same company as before also provides a hydrated salt type of phase change material which has a phase change temperature of 136°F and a latent heat storage of 6,476 Btu/ft^3. The volume of this storage tank is found slightly differently than before. Because the storage tank is only used to supply energy to produce hot water for a cleaning process at the end of each shift, it should only be designed to adequately store energy for this process. As mentioned previously, this design will reduce the implementation costs of the thermal storage tank but will also limit the amount of energy that can be recovered. The volume of the tank is estimated to be 325 gallons which has the capacity to store 0.3 Million Btus.

Temperature Control Strategy

After the required equipment has been selected for the system, the control strategy must be determined. Because of the system design, it is recommended to install a thermostatically

controlled bypass loop around each component where a specified low temperature must be met. This will prevent energy from being transferred from the process into the thermal fluid, the opposite of the desired effect, and also allow for adequate temperature and flow of hot air or water to be delivered. For the end-uses, the controls should be programmed to read the temperature of the incoming fluid and compare that with the required temperature for the process, which depends heavily on the process demand at the specified time. For example, the amount of energy required for heating the glue baths depend heavily on the ambient conditions. This control strategy should be developed to continuously measure the temperature of the glue and to ensure it is maintained at the appropriate temperature by adjusting the flow rate with an automated control valve. This control valve should be designed to open or close depending on the supplied hot fluid temperature to maintain the required heat flow. This system should also be able to turn off the heating fluid flow if appropriate temperatures are not available, in which case the backup system must be used to supply the required energy to the system.

Insulation of Hot Surfaces

To ensure the system maintains as much energy as possible, all exposed hot surfaces should be well insulated. This includes piping, valves, fittings, thermal storage tanks, and heat exchangers and processes when applicable. The insulation thickness should be determined to adequately maintain the energy for whatever purpose it is being used for in the process and for the system components. For most situations, normal, high temperature insulation can be used. This includes calcium silicate, mineral wool and cellular glass types to name a few. However special care should be taken to prevent leaks of the heating oil in the system, not only to conserve as much energy as possible and to reduce the need to purchase additional fluid, but fluid-saturated insulation can become a potential fire hazard when the fluid is operated at a high temperature.

Discussion of System Design

This particular type of system design was selected to separate the higher temperature energy recovered from the lower temperature energy. This design combines both the second and third concepts which were discussed earlier. This separation is accomplished by creating two entirely separate loops, one with the energy flowing directly into the thermal storage tank and the other delivering energy first into the process before returning through a thermal storage tank at the end. The first heat exchanger recovers energy from the stack gas and delivers the fluid at a temperature of 400 °F to the tank. This energy is transferred into the thermal storage tank and changes the phase of the enclosed materials at a temperature of 327°F while continuing to add additional energy into the system up to the fluid supply temperature. The second heat exchanger (condensing) sensible energy, used for the absorption chiller loop, is setup to supply energy to the process equipment first and, when the appropriate temperatures are available in the return line, the heating fluid is delivered through a second thermal storage tank. This type of system was selected for the condensing heat exchanger loop because there is a demand for the recovered energy fluid temperature and to separate the lower temperature fluid from the higher grade energy delivered from the initial heat exchanger. By separating the fluid streams into a high temperature loop and a low temperature loop, higher temperature end-use applications are possible and provide more overall opportunities for energy savings.

Energy and Monetary Savings

The energy savings that can be expected from implementing a system similar to this one described is found by determining the total amount of energy which can be offset by the recovered energy. This can be estimated by considering the approximate hours each year each

system will operate and to calculate the average demand for energy of each system throughout these hours. This information can be seen in the following table.

Table 10. Estimated Energy Demand.

System	Estimated Operating Hours (hrs/yr)	Average Energy Demand	
Space Heating	1,050	330,000	Btu/hr
Domestic Hot Water	8,760	2,000	Btu/hr
Glue Baths	8,760	65,000	Btu/hr
Intermittent Demand (Loop 1)	1,460	3,210,675	Btu/hr
Absorption Chiller	2,100	170,000	Btu/hr
Preheating Feedwater	8,760	750,000	Btu/hr
Intermittent Demand (Loop 2)	730	554,800	Btu/hr
HX1 Loop Pump	8,760	4.3	kW
System Loop Pump	8,760	0.9	kW
Chiller Loop Pump	8,760	0.2	kW
Intermittent Loop Pump	1,460	0.2	kW

The previous chart was compiled by using several methods of estimating the annual operating hours. For the space heating and the absorption chiller, local BIN weather data was used to determine how often the outdoor temperature would be above or below the temperature setpoint. From this information a usage factor was estimated to determine the percentage of the time the units would actually operate during these hours. The intermittent loads operating hours were determined from the estimated daily requirements and assumed to operate under these same conditions year round. The average energy input for these systems was found using the required flow rates along with the available energy remaining in the thermal energy fluid. The overall system was estimated to lose 7% of the energy recovered due to different considerations such as heat exchanger fouling and insulation losses.

Once the average input and operating hours are known, the annual energy reduction for each process can be determined. This information is shown in the following table and is broken up by the type of energy that the energy use reduction will offset, whether it be electricity or natural gas.

Table 11. Energy Purchased Reductions

System	Operating Hours (hrs/yr)	Average Energy Demand		Type of Energy Offset	Electrical Reduction (kWh/yr)	NG Reduct. (MMBtu/ yr)
Space Heating	1,050	330,000	Btu/hr	Electricity	101,553	-
Domestic Hot Water	8,760	2,000	Btu/hr	NG	-	18
Glue Baths	8,760	65,000	Btu/hr	Electricity	166,882	-
Intermittent Demand (L1)	1,460	3,210,675	Btu/hr	NG	-	4,688
Absorption Chiller	2,100	170,000	Btu/hr	Electricity	104,631	-
Preheating Feedwater	8,760	750,000	Btu/hr	NG	-	6,570
Intermittent Demand (L2)	730	554,800	Btu/hr	NG	-	405
HX1 Loop Pump	8,760	4.3	kW	Electricity	-37,576	-
System Loop Pump	8,760	0.9	kW	Electricity	-8,169	-
Chiller Loop Pump	8,760	0.2	kW	Electricity	-1,960	-
Intermittent Loop Pump	1,460	0.2	kW	Electricity	-327	-
Total					**325,034**	**11,680**

To translate this energy savings into a monetary savings, information about the electrical and natural gas bills must be known. The facility considered for this design is charged both an energy rate and demand rate for electricity. This cost is $0.0541/kWh for energy consumption and $13.7863/kW-month for the peak electrical demand throughout the month. Each of the purchased energy reductions that are tied to electrical consumption reduce the overall energy purchased for the facility by reducing the amount of energy consumed. However not all of the savings opportunities will necessarily reduce the system's peak demand. For this to occur, the specific piece of equipment would have to be operating at the moment of the peak energy demand, and then by being able to reduce or eliminate this electrical consumption, the peak

demand would therefore be reduced. Because all three of the electrical reductions are cyclical, and turn on and off based on ambient conditions, it cannot be guaranteed that the elimination of these process electrical consumption will reduce the facility's overall peak electrical demand. Therefore the electrical demand savings are not included in the cost savings for these recommendations. The natural gas cost is based only on the volume purchased and does not depend on when it is purchased or how much is being purchased at once. Because there is a standard quality of natural gas delivered, the volume of gas delivered can be expressed as energy content delivered which is determined based on the higher heating value of natural gas. This cost per unit of energy for this facility is $8.73/MMBtu. The cost savings which can be expected from implementing this system are summarized in the Table 12 and is estimated to save just under $120,000.

Table 12. Estimated Cost Savings of the System.

System	Type of Energy Offset	Electrical Consumption Reduction (kWh/yr)	Natural Gas Consumption Reduction (MMBtu/yr)	Estimated Cost Savings $/yr
Space Heating	Electricity	101,553	-	5,494
Domestic Hot Water	NG	-	18	153
Glue Baths	Electricity	166,882	-	9,028
Intermittent Demand (Loop 1)	NG	-	4,688	40,923
Absorption Chiller	Electricity	104,631	-	5,661
Preheating Feed water	NG	-	6,570	57,356
Intermittent Demand (Loop 2)	NG	-	405	3,536
HX1 Loop Pump	Electricity	-37,576	-	-2,033
System Loop Pump	Electricity	-8,169	-	-442
Chiller Loop Pump	Electricity	-1,960	-	-106
Intermittent Loop Pump	Electricity	-327	-	-18
Total		325,034	11,680	119,552

Environmental Impacts of Implementation

As mentioned in the *Impacts of Waste Heat* section of the report, the overall environmental impact that recovering waste has heat can become critically important in the future. The environmental impact of implementing the system can be calculated using information on the carbon dioxide emission levels depending on the type of fuel saved. This information can be found using information provided by the US Energy Information Administration and is broken down based on the geographical averages. The national averages consist of 1.341 pounds of CO_2 produced per kilowatt-hour produced [13] and 117 pounds of CO_2 produced per MMBtu of natural gas combusted [67]. Another environmental impact which should be considered is the effect of reducing the energy exhausted into the atmosphere. Because the recovered energy offsets the production of electricity at the point of generation and the combustion of natural gas in the facility, it can be shown as a reduction in heat energy added to the atmosphere. To determine the amount of energy reduced, the generation efficiency of electricity must be considered. The average overall efficiency of electrical power generation can be found from information provided by the U.S. Energy Information Administration to be about 37% [66]. To determine the reduction in energy content exhausted to the atmosphere by the combustion of natural gas, the total energy reduction is used. This information is shown in Table 13.

Table 13. Environmental Impact of System

System	Electrical Consumption Reduction (kWh/yr)	Natural Gas Consumption Reduction (MMBtu/yr)	CO2 Reduction (tons/yr)	Reduction in Energy Exhausted to Atmosphere (MMBtu/yr)
Space Heating	101,553	-	68	944
Domestic Hot Water	-	18	1	18
Glue Baths	166,882	-	112	1,551
Intermittent Demand (Loop 1)	-	4,688	274	4,688
Absorption Chiller	104,631	-	70	972
Preheating Feed water	-	6,570	384	6,570
Intermittent Demand (Loop 2)	-	405	24	405
HX1 Loop Pump	-37,576	-	-25	-349
System Loop Pump	-8,169	-	-5	-76
Chiller Loop Pump	-1,960	-	-1	-18
Intermittent Loop Pump	-327	-	0	-3
Total	**325,034**	**11,680**	**901**	**14,700**

Implementation Costs

A major consideration when deciding on whether to implement a system or not is the length of time that the system will require before the facility will begin to earn money and increase profitability. To determine this length of time to determine if the project is economically feasible, an approximated implementation cost must be determined based on all the system components used. The overall simple payback period for the system can then be calculated and each company will have different preferences about how long is too long for a project to pay back, however if a payback of under five years can be achieved, the project is recommended to be implemented. To determine the implementation cost for this described system each component is listed below and when possible, the RS Means Mechanical Cost Data 2010 Catalog [51] was used to determine the cost of the component as well as the labor to install the component. This implementation cost is summarized in the table on the following page.

This implementation cost is an estimate and should be taken only as that, not all specific recommended equipment was located and prices are based on similar equipment prices. The information does provide a rough estimate on the overall cost to implement a system similar to the one designed.

Table 14. Implementation Costs

Component	Quantity		Material Cost	Labor Cost	Total Cost	Source
Schedule 40, 2 inch diameter Steel Pipe with hangers	1,500	ft	12.60	11.70	36,450	[51]
2 inch Cast Steel Valves	35		6.30	10.40	585	[51]
2 inch 90° elbows	40		22.50	75	3,900	[51]
2 inch T Fittings	37		33.43	125	5,862	[42],[51]
Control System	15		675	38.50	10,703	[51]
2 inch Cellular Glass Insulation	1,500	ft	6.75	7.70	21,675	[51]
Valve Insulation	35		8.00	12.15	705	[51]
2 inch ID Insulation Jacketing (Aluminum)	1,500	ft	0.68	3.36	6,060	[51]
Flow Meter	2		112	104	432	[51]
Heat Exchanger 1	1		11,132	7,100	18,232	[51]
Heat Exchanger 2	1		16,758	10,011	26,769	[51]
Feed Water HX	1		24,000	10,650	34,650	[51]
Thermal Storage Tank 1	1		15,000	320	15,320	[51]
Thermal Storage Tank 2	1		8,675	139	8,814	[51]
Phase Change Material (Organic)	100	ft^3	603	-	60,462	[47]
Phase Change Material (Hydrated Salt)	43	ft^3	208	-	9,021	[47]
Domestic Hot Water System	1		1,200	390	1,590	[51]
Space Heating System	1		1,620	408	2,028	[51]
Glue Bath System	14		41.50	23.50	910	[51]
Absorption Chiller	1		30,000	1,500	31,500	[51]
Expansion Tank	2		3,750	93.50	7,687	[51]
DOWTHERM G	225	Gal	66.31	-	14,920	[68]
UCON 500 HTF	55	Gal	29.26	-	1,609	[68]
HX1 Loop Pump	1		600	325	925	[42],[51]
Absorption Chiller Loop Pump	1		450	325	775	[42],[51]
Main Loop Pump	1		450	325	775	[42],/[51]
Intermittent Loop 1 Pump	1		450	325	775	[42]/[51]
Total			247,460	75,673	323,134	

From the energy savings and implementation costs calculated, a simple payback period for the system design can be found. The simple payback calculates the number of years the investment will reduce energy costs by the same amount as the initial cost of implementation. This demonstrates the timeframe the project requires to begin "earning" the facility money [24]. The following calculation shows this simple payback period.

$$SP = IC / ES$$

$$= \$323{,}134 / \$119{,}552/yr$$

$$= 2.7 \text{ years}$$

where

SP = simple payback period

IC = total implementation cost for the entire system

ES = annual energy savings expected from the design

This system design is estimated to pay for itself in less than three years. This simple payback is calculated based on the initial costs along with the estimated energy consumption reduction and does not include any types of maintenance costs throughout the life of the system. However the system does not consist of many moving parts leading to the majority of the maintenance costs being associated with the pumps and cleaning of the heat exchangers. This maintenance is assumed to be minimal and should not impact the implementation of the system. However, certain maintenance interval estimations have been established and are discussed in previous sections which can be used to provide a more accurate representation of the payback period.

Different Design Scenarios

It is simply not true that all heat recovery applications will be able to use the previous design. Some facilities will have a variable demand while others will have a much larger source of waste heat and are able to effectively produce electricity. In the case that the system does not operate in the same manner as the previous design applies to, a few design tips for different operating points are discussed below.

The first variation that is considered is a furnace operation that has a changing energy input rate. This variable energy input rate will ultimately change the amount of recoverable energy in the exhaust stack by not only changing the temperature of the gas but also the volumetric flow rate. This change will not necessarily be directly proportional to the amount of energy input because furnaces tend to become less efficiently controlled as the firing rate is reduced, leading to more volume of flow but also a lower temperature. While the furnace will be exhausting gas at a lower temperature, the end uses of the system will still require a minimum temperature to operate. The previous analysis assumed a constant exhaust temperature and also a constant flow of heating fluid throughout the system. This simplifies the design of the pumps to only having one operating flow rate and a constant speed motor is all that is needed. In the situation where the exhaust temperature is not constant, a variable speed drive is recommended to be added to the pumps to vary the flow rate of the fluid to maintain the minimum temperature required. This type of system also increases the importance of the thermal storage tanks because the supply and demand will not be constant. By adding a thermal storage tank to the loop, this energy can still be recovered when it is available, stored in the latent heat of the phase change materials in the storage tank and then used by the processes as needed.

Another situation that is considered is where a large volume of high temperature flue gas is exhausted into the atmosphere and there is a relatively small direct use for the heat elsewhere in the plant. The simple recovery options should still be included such as preheating combustion air or feedwater where applicable however another option that should be considered is installing an Organic Rankine cycle to produce electricity which can be used for any electrical process in the facility. This option allows for the energy in the exhaust stream to be recovered for a useful application and can be directly seen as monetary savings as purchased electrical consumption is decreased. This conversion of waste heat into electricity could also be accomplished by other methods which were discussed in previous sections.

Chapter 7

Conclusion

Waste heat energy from industrial processes, in particular combustion equipment, has become a large, available source of untapped energy. The total amount of energy available is not monitored and is not known, however studies have been performed by and for the US Department of Energy that estimate this wasted energy can range anywhere from 20% to half of the total energy consumed by the industrial sector. This energy is lost through heat energy contained in hot exhaust gasses and depending on the process can be any temperature or quantity. The study conducted for this report focuses on the lower temperature exhaust, around 450°F, which makes up about 60% of all the wasted energy. This wasted energy temperature range is also the most difficult to recover which leads to the majority of facilities to simply ignore it.

Not only does this wasted heat lead to an increase in overall energy production and consumption, it also has two major environmental impacts. The first and most obvious is the carbon dioxide created for each unit of energy produced. These values have been researched by the US Department of Energy and for every kilowatt-hour production reduced, 1.341 pounds of CO_2 production can be eliminated. The same study also states that for every MMBtu of natural gas reduction in consumption, 117 pounds of CO_2 can be eliminated. The second type of environmental impact that occurs from waste heat is a flow of over 55 Quadrillion Btus of

energy steadily flowing into the atmosphere. This energy flow is beginning to gain consideration in global climate change models and is estimated to become an equal player along with CO_2 production, in the climate within the next two hundred years.

The recovery and reuse of waste heat requires many different types of equipment to achieve all possible requirements. The most important type of equipment in waste heat recovery is a heat exchanger which allows for the heat energy in the exhaust gas to be transferred to a more useable form. The shell-and-tube heat exchanger is the most commonly used type of heat exchanger for this application. However to recover additional energy from the stack gas by condensing the water vapor, a transport membrane condenser has been developed. This technology, along with a conventional shell-and-tube condensing economizer, is capable of recovering the latent heat stored by converting liquid water into a vapor form during the combustion process. This process allows for almost an additional 10% of energy to be recovered.

Other types of important equipment used in heat recovery applications include heat pumps, absorption chillers, heat engines and electrical conversion equipment, and pumps. Heat pumps can be used to increase the temperature of the fluid when the recovered energy temperature cannot be used in the facility. This provides additional end use possibilities by being able to increase the fluid temperature to meet certain equipment requirements. The absorption chiller can be used for the same application as the heat pump but can also be used to supply a cooling effect to the facility, whether it is space conditioning or refrigeration. Heat engines and electrical conversion equipment have the unique effect of allowing the user to create electricity from the waste heat when applications cannot be found for the heat energy directly. This allows an almost infinite number of possible applications for the energy, however these technologies have relatively low conversion efficiency and a large implementation cost leading to a tough payback

with current electricity prices. The final piece of equipment critical to the success of a waste heat recovery system is the pump. The pump allows the energy recovered from the hot exhaust gas which has been transferred into a fluid to be transported throughout the facility and to be consumed for useful applications. A centrifugal style pump is recommended in most applications and can provide adequate flow and efficiency for most types of heat recovery systems.

Another major contributor in the recovery of waste heat is the energy transportation fluid. The selection of this fluid will determine the amount of energy allowable that can not only be recovered but how much can be delivered to the end use. There are many different types of fluid available for this transportation however certain types have better characteristics under certain situations. The most common and abundant type of transportation fluid is water because it contains a high density and specific heat, but this fluid has certain limitations when a single phase flow is desired. Water contains a low boiling point and is limited in application by this. Additives have been developed to increase this boiling temperature but do not substantially increase it until large concentrations are seen. This fact decreases the benefit of using water as the heat transportation fluid because the additives contain much lower heat storage characteristics. Alternatives are available that contain much higher boiling points however the same energy transportation characteristics cannot be seen. These types of thermal heating fluids include DOWTHERM G, UCON HTF 500 and Therminol XP. DOWTHERM G contains the highest operating temperature of 675°F but UCON HTF 500 possesses greater lower temperature characteristics.

A waste heat recovery system was designed that takes into account the different types of considerations discussed. This system provides the opportunity to recover energy from a 400 hp

boiler continuously throughout the year which operates with an exhaust temperature of 485°F and a flow rate of 4,050 cfm. The heat is first recovered using a conventional shell-and-tube heat exchanger and this energy is sent to a thermal storage tank. This storage tank captures the energy contained in the thermal heating fluid and stores it using the latent heat from the enclosed phase change materials. This energy is then transferred from the phase change material into a separate loop of energy transportation fluid which is sent throughout the facility where it is used to create domestic hot water, add comfort heating to the offices, lab and break area, along with heating glue baths. The energy not captured in the first exhaust stack heat exchanger is then sent through a second heat exchanger where additional energy along with water vapor latent heat is recovered with the use of a condensing economizer. This recovered energy is used to pre-heat incoming feed-water along with supplying energy to operate an absorption chiller which supplies cooling to the bead room. The additional capacity recovered from the exhaust gas is stored in separate thermal storage tanks until an intermittent load is needed which "drains" the tanks of their energy and offsets additional energy requirements from other energy methods.

The system energy savings estimates were then calculated based on the amount of energy offset by the waste heat system along with the type of energy offset. The results of this analysis include a reduction in annual electrical consumption of about 325,000 kWh and a reduction of 11,680 MMBtu a year of natural gas consumption. This energy consumption reduction translates into a total energy cost savings of almost $120,000 a year. The associated energy reductions can also be used to calculate an estimated reduction in CO_2 production along with a total energy input decrease to the atmosphere. These values are found to be 901 tons a year and 14,700 MMBtu a year respectively for this facility.

The overall cost of implementing the system was also calculated. This installation cost includes both the price of each piece of equipment and the associated labor costs for each component. These values were estimated using an up to date cost estimating handbook along with additional resources when required. This estimation leads to a total implementation cost of just over $323,000. The implementation costs can then be compared to the annual estimated energy savings to predict a simple payback period. This payback period is found to be 2.7 years and is found by dividing the found implementation cost by the annual energy savings.

This system design shows that the potential for waste heat recovery savings can be seen no matter the size of the facility. The design also demonstrates that the system can achieve a relatively quick payback, even when sophisticated equipment such as an absorption chiller is used.

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APPENDIX

Heat Exchanger 1 Design Performance Data [30]

HTRI

Output Summary Page 1

Released to the following organization:
Microsoft
AIAC

Xace E Ver. 6.00 SP1 2/22/2011 17:48 SN: 1600211746 **US Units**

Design-Horizontal economizer countercurrent to crossflow

See Data Check Messages Report for Informative Messages.
See Runtime Message Report for Warning Messages.

Process Conditions		Outside		Tubeside	
Fluid name		Air		DOWTHERM G	
Fluid condition			Sens. Gas		Sens. Liquid
Total flow rate	(1000-lb/hr)		87.520 *		62.800
Weight fraction vapor, In/Out		1.000 *	1.000 *	0.000	0.000
Temperature, In/Out	(Deg F)	485.00	260.00	225.00	400.00
Skin temperature, Min/Max	(Deg F)	233.11	353.57	233.09	353.37
Pressure, Inlet/Outlet	(psia)	30.000	29.999	50.000	49.973
Pressure drop, Total/Allow	(psi) (psi)	9.074e-4	0.000	0.027	0.000
Midpoint velocity	(ft/sec)		6.79		0.12
- In/Out	(ft/sec)			0.13	0.12
Heat transfer safety factor	(--)		1		1
Fouling	(ft2-hr-F/Btu)		0.00000		0.00000

Exchanger Performance					
Outside film coef	(Btu/ft2-hr-F)	9.54	Actual U	(Btu/ft2-hr-F)	4.889
Tubeside film coef	(Btu/ft2-hr-F)	11.80	Required U	(Btu/ft2-hr-F)	4.867
Clean coef	(Btu/ft2-hr-F)	4.889	Area	(ft2)	460.287
Hot regime		Sens. Gas	Overdesign	(%)	0.44
Cold regime		Sens. Liquid			
EMTD	(Deg F)	308.0			
Duty	(MM Btu/hr)	2.683			

Unit Geometry				Tube Geometry		
Bays in parallel per unit			8	Tube type		Plain
Bundles parallel per bay			1	Tube OD	(inch)	0.8750
Extended area	(ft2)		460.287	Tube ID	(inch)	0.7450
Bare area	(ft2)		460.287	Length	(ft)	3.000
Bundle width	(ft)		3.000	Area ratio(out/in)	(--)	1.1745
Nozzle		**Inlet**	**Outlet**	Layout		Staggered
Number	(--)	1	1	Trans pitch	(inch)	2.0000
Diameter	(inch)	2.0670	2.0670	Long pitch	(inch)	1.7500
Velocity	(ft/sec)	1.53	1.42	Number of passes	(--)	1
R-V-SQ	(lb/ft-sec2)	143.05	132.85	Number of rows	(--)	5
Pressure drop	(psi)	0.015	7.169e-3	Tubecount	(--)	88
				Tubecount Odd/Even	(--)	18 / 17
				Tube material		Carbon steel

Fan Geometry			Fin Geometry		
No/bay	(--)	0	Type		None
Fan ring type			Fins/length	fin/inch	
Diameter	(ft)	0.000	Fin root	inch	
Ratio, Fan/bundle face area	(--)		Height	inch	
Driver power	(hp)	0.00	Base thickness	inch	
Tip clearance	(inch)	0.0000	Over fin	inch	
Efficiency	(%)	0	Efficiency	(%)	
			Area ratio (fin/bare)	(--)	
			Material		

Airside Velocities		Actual	Standard
Face	(ft/min)	254.04	
Maximum	(ft/sec)	7.91	
Flow	(1000 ft3/min)	18.291	
Velocity pressure	(inH2O)	0.000	
Bundle pressure drop	(inH2O)	0.025	
Bundle flow fraction	(--)	1.000	

Thermal Resistance, %	
Air	51.26
Tube	48.65
Fouling	0.00
Metal	0.10
Bond	0.00

Airside Pressure Drop, %					
Bundle	100.00			Louvers	0.00
Ground clearance	0.00	Fan guard	0.00	Hail screen	0.00
Fan ring	0.00	Fan area blockage	0.00	Steam coil	0.00

Condensing Economizer (HX2) Design Performance Data [30]

HTRI

Output Summary Page 1

Released to the following organization:
Microsoft
AIAC

Xace E Ver. 6.00 SP1 2/22/2011 20:16 SN: 1600211746 **US Units**

Design-Horizontal economizer countercurrent to crossflow

See Data Check Messages Report for Warning Messages.
See Runtime Message Report for Warning Messages.

Process Conditions		Outside		Tubeside	
Fluid name		Air		UCON HTF 500	
Fluid condition			Cond. Vapor		Sens. Liquid
Total flow rate	(1000-lb/hr)		142.220 *		17.400
Weight fraction vapor, In/Out		1.000 *	0.931 *	0.000	0.000
Temperature, In/Out	(Deg F)	260.00	135.00	90.00	225.00
Skin temperature, Min/Max	(Deg F)	113.85	193.64	113.82	193.56
Pressure, Inlet/Outlet	(psia)	30.000	29.995	25.000	24.999
Pressure drop, Total/Allow	(psi) (psi)	4.980e-3	0.000	8.265e-4	0.000
Midpoint velocity	(ft/sec)		11.39		7.986e-3
- In/Out	(ft/sec)			8.237e-3	7.986e-3
Heat transfer safety factor	(--)		1		1
Fouling	(ft2-hr-F/Btu)		0.00000		0.00000

Exchanger Performance					
Outside film coef	(Btu/ft2-hr-F)	14.93	Actual U	(Btu/ft2-hr-F)	4.008
Tubeside film coef	(Btu/ft2-hr-F)	6.30	Required U	(Btu/ft2-hr-F)	4.156
Clean coef	(Btu/ft2-hr-F)	4.008	Area	(ft2)	1524.43
Hot regime		Cond. Vapor	Overdesign	(%)	-3.58
Cold regime		Sens. Liquid			
EMTD	(Deg F)	142.0			
Duty	(MM Btu/hr)	7.443			

Unit Geometry				Tube Geometry		
Bays in parallel per unit			13	Tube type		Plain
Bundles parallel per bay			1	Tube OD	(inch)	1.0000
Extended area	(ft2)		1524.43	Tube ID	(inch)	0.8700
Bare area	(ft2)		1524.43	Length	(ft)	3.000
Bundle width	(ft)		3.000	Area ratio(out/in)	(--)	1.14943
Nozzle		**Inlet**	**Outlet**	Layout		Staggered
Number	(--)	1	1	Trans pitch	(inch)	1.4000
Diameter	(inch)	2.0670	2.0670	Long pitch	(inch)	1.1000
Velocity	(ft/sec)	0.25	0.24	Number of passes	(--)	1
R-V-SQ	(lb/ft-sec2)	4.00	3.88	Number of rows	(--)	7
Pressure drop	(psi)	4.321e-4	2.095e-4	Tubecount	(--)	172
				Tubecount Odd/Even	(--)	25 / 24
				Tube material		Carbon steel

Fan Geometry			Fin Geometry		
No/bay	(--)	0	Type		None
Fan ring type			Fins/length	fin/inch	
Diameter	(ft)	0.000	Fin root	inch	
Ratio, Fan/bundle face area	(--)		Height	inch	
Driver power	(hp)	0.00	Base thickness	inch	
Tip clearance	(inch)	0.0000	Over fin	inch	
Efficiency	(%)	0	Efficiency	(%)	
			Area ratio (fin/bare)	(--)	
			Material		

Airside Velocities		Actual	Standard
Face	(ft/min)	230.79	
Maximum	(ft/sec)	15.95	
Flow	(1000 ft3/min)	27.003	
Velocity pressure	(inH2O)	0.000	
Bundle pressure drop	(inH2O)	0.138	
Bundle flow fraction	(--)	1.000	

Thermal Resistance, %	
Air	26.84
Tube	73.09
Fouling	0.00
Metal	0.07
Bond	0.00

Airside Pressure Drop, %					
Bundle	100.00			Louvers	0.00
Ground clearance	0.00	Fan guard	0.00	Hail screen	0.00
Fan ring	0.00	Fan area blockage	0.00	Steam coil	0.00