ME-496
Senior Design Project

Design of A Waste Energy Recovery System
For the Pettit National Ice Center

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ABSTRACT

The Pettit National Ice Center has high-energy bills with approximately $23,000 spent on electricity each month and heating costs reaching $9,000 a month. In addition to the large heating bills, a substantial amount of waste heat from ice making equipment is rejected to the environment. The objective of this project is to select and design a waste energy recovery system for the Pettit National Ice Center. The existing system is examined to determine where heat is used and rejected. It is estimated that the evaporative condenser rejects 4.16 million Btu/h to the environment and a natural gas boiler provides roughly 1.75 million Btu/h to heat the facility. Different methods of heat recovery are examined to determine which are feasible with the PNIC system. A shell and tube heat exchanger capable of transferring 2 million Btu/h of heat from the refrigeration system to the heated water system and a small auxiliary boiler is designed.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

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1. INTRODUCTION

Energy has always been a significant component of industrial operations. Industrial heating and cooling processes generate large amounts of waste energy that are rejected to the environment. When energy is abundant and cheap, the initial cost of the equipment has a higher priority than the cost of operation. However, as sources of fossil fuels are depleted, the price of fuel increases, system-operating costs increase, and the use of waste energy becomes more important. (Van Arsdell, 2000)

One method of reducing operating costs is to harness and reuse wasted energy, resulting in lower utility bills. Energy recovery is the beneficial use of heating or cooling energy that would typically be rejected to the environment (U.S. Department of Energy, 2001). Technologies that recover heating and/or cooling energy reduce the cost and consumption of energy in commercial and institutional buildings. The recaptured energy may be useful for space heating, water heating, or other industrial processes.

The Pettit National Ice Center has high-energy bills with approximately $22,000 to $25,000 spent on electricity each month and with heating costs reaching $9,000 a month. In addition to the large heating bills, a substantial amount of waste heat from ice making equipment is rejected to the environment. It is desirable to lower the operational costs of the PNIC by lowering the amount of fuel needed to heat the facility. This can be accomplished by implementing a waste energy recovery system.

The objective of this project is to design and select a waste energy recovery system for the Pettit National Ice Center. Several tasks need to be completed in order to accomplish this objective. A schematic drawing of the cooling/heating system at the PNIC must be created to determine how much waste heat is available, and how much heat is needed. Several options of

energy recovery systems are examined and analyzed to determine if they are applicable to the PNIC. A full design for an energy recovery system is performed on the two most feasible options, and a life cycle cost analyses compares these options.

This report consists of a full analysis of the Pettit National Ice Center as well as the solution to its high-energy bills. Section 2 of the report gives a system description of the PNIC, which includes an introduction to the system as well as the components involved. This includes schematic diagrams of the energy flows and their respective characteristics, i.e., flow rates, pressures, and temperatures. Section 3 details the analysis of the solution to the high-energy bills and evaluates possible areas where the waste energy can be utilized in the PNIC. Section 4 highlights the results of the feasibility studies and life cycle cost analyses. This section shows the criterion used to select the best energy recovery system. Section 5 concludes the report and gives a final recommendation of the two best energy recovery systems, as well as the approach to integrate the energy recovery system with the current system.

2. SYSTEM DESCRIPTION

2.1 Introduction

The Pettit National Ice Center, located in West Allis, WI, was built in 1992 as an Olympic training facility for the 1994 Olympics. The PNIC is 127,000 ft^2, with 97,000 ft^2 of ice. Within the PNIC, there is a 450-meter running/walking track, a 400-meter speed-skating oval, two international-size hockey/figure skating rinks, offices, conference rooms, locker rooms, showers, and a pro-shop. An ammonia refrigeration system is used to maintain the ice and a heating, ventilation, and air-conditioning (HVAC) system conditions the air within the facility.

2.2 Ammonia Refrigeration System

Figure 2.1 is a schematic of the refrigeration system at the PNIC. The refrigeration system consists of two loops, a primary ammonia refrigeration loop and a secondary chilled glycol loop. The equipment in the ammonia loop consists of compressors, a condenser, a high-pressure receiver, and the oval/rink chiller. There are eight compressors, 2-150 horsepower and 6-125 horsepower compressors. Of the eight compressors, only two 150 horsepower compressors operate at a given time.

The ammonia enters the compressors as a low-pressure vapor and is compressed to a saturated high-pressure vapor. Part of the high-pressure ammonia passes through the glycol heater and part passes through the evaporative condenser, where heat is removed from the vapor. The saturated high-pressure vapor leaves the condenser as a saturated vapor and is collected in the high- pressure receiver before entering the oval/rink chiller. Heat is transferred from the glycol loop to the ammonia in the chiller. The chiller is a flooded chiller that receives the ammonia as a low-pressure liquid and discharges it as a low-pressure vapor. The equipment specifications are given in Tables A.1 to A.3 in Appendix A.

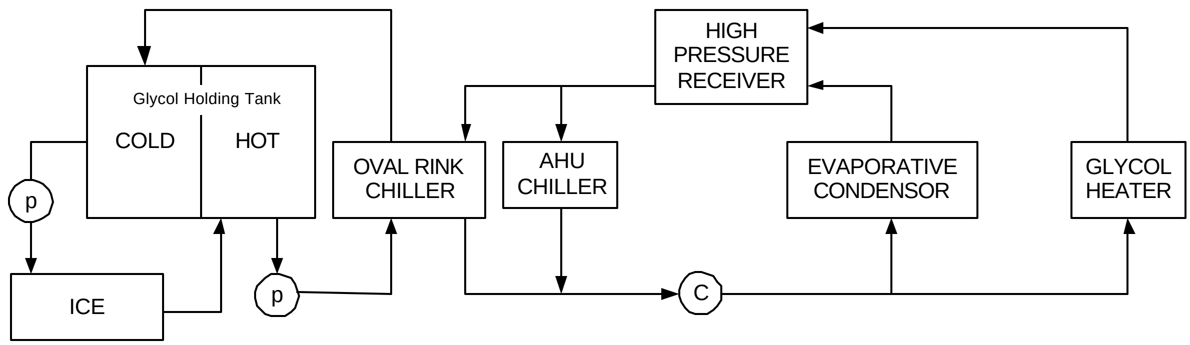

Figure 2.1: Ammonia Refrigeration Loop

2.3 Chilled Glycol Loop

The chilled glycol loop supplies cold glycol to the east rink, the west rink, and the oval. The equipment in the chilled glycol loop consists of several pumps, a glycol holding tank, and an oval/rink chiller. The chilled glycol loop is shown in Figure 2.2. There are eight pumps, 2-25 horsepower pumps, 3-60 horsepower pumps, and 3-125 horsepower pumps. The pump specifications are given in Table A.4 in Appendix A. The glycol holding tank is made of

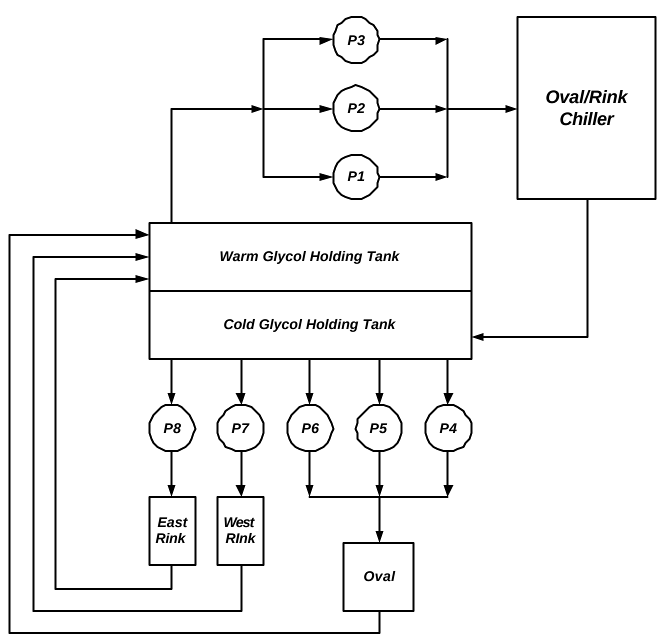

Figure 2.2: Chilled Glycol Loop

pre-cast concrete and is roughly 21’ Long x 18’ Wide x 9’ High. The tank has a divider that separates the warm glycol and cold glycol.

After the heat is transferred from the glycol to the ammonia in the oval/rink chiller, it enters the cold glycol holding tank where it is stored. The glycol is then pumped from the cold glycol holding tank to the east rink, the west rink, and the oval. Of the three pumps supplying the oval, only one is used at any given time. After the glycol is piped under the ice, it is discharged to the warm glycol holding tank where it is stored. Pumps 1, 2, and/or 3, depending on the demand, then pump the warm glycol to the oval/rink chiller.

2.4 Air-Handling Unit System

The air-handling units (AHUs) are used to condition the facility. The AHU system consists of six AHUs, two natural gas boilers as a source of heated water, and an AHU chiller as a source of chilled water. The AHU chiller specifications are given in Table A.5 of Appendix A and the AHU system is shown in Figure 2.3.

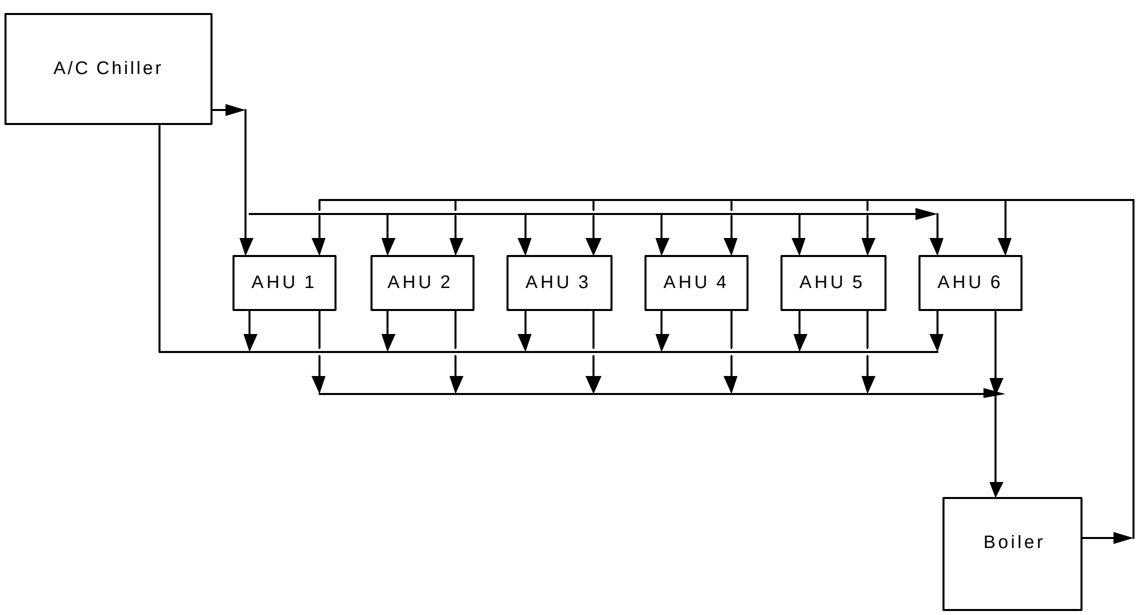

Figure 2.3: Air-Handling System

	AHU-1	**AHU-2**	**AHU-3**	**AHU-4**	**AHU-5**	**AHU-6**
Flow Rate (GPM)	49	49	49	49	64.5	9.1
Temp. Difference (F)	20	20	20	20	20	20
Heating Load (Btu/hr)	490,000	490,000	490,000	490,000	645,000	91,000

Table 2.1: AHU Heating Loads

There are six AHUs; AHU-1 to AHU-4 conditions the arena, AHU-5 conditions the showers and locker rooms, and AHU-6 conditions the office space, conference area and the pro-shop. Of the four AHUs supplying the arenas, only two are used at any given time. The designed heating load is 1.716 million Btu/hr. The specifications of the AHUs are given in Table 2.1. The heating loads for the six AHUs are given by (ASHRAE, 2001)

$$q = (Q)(\rho)(C_p)(\Delta T) \qquad (2.1)$$

where, q is the heating load, Q is the flow rate, ρ is the fluid density, C_p is the specific heat at constant pressure, and ΔT is the difference of temperatures between inlet and outlet.

The cooling coils in the AHUs are supplied with chilled water from the flooded ammonia AHU chiller (see Figure 2.1). The AHU chiller receives low-pressure liquid ammonia from the high-pressure receiver and discharges a low-pressure vapor to the compressors where it is then cycled through the ammonia refrigeration system.

The heating coils in the AHUs are supplied with hot water from one of the two natural gas boilers. Due to the size of the two natural gas boilers, only one is needed to supply the AHUs and the glycol/hot water exchanger. The glycol/hot water exchanger is used for rapidly melting the ice. The glycol/hot water exchanger is connected to the discharge flow from the cold glycol-holding tank. The schematic drawing is shown on Figure B.1 in Appendix B.(entire system schematic)

Several valves allow the glycol/hot water exchanger to heat the glycol from the cold glycol-holding tank discharge. The glycol can be heated and pumped under the east rink, the west rink, and the oval to rapidly melt the ice.

2.5 Heating Glycol Loop

Figure 2.4 is a schematic diagram of the heating glycol loop at the PNIC. The heating glycol loop is used to prevent perma-frost and to provide heat for the snow pit. The equipment in the heating glycol loop consists of pumps and a glycol heater. There are four pumps, 2-7.5 horsepower pumps and 2-5 horsepower pumps. The glycol heater specifications are given in Table A.6 and A.7 of Appendix A.

Once a year, the glycol heater supplies warm glycol to the east rink, the west rink, and the oval to prevent perma-frost. Warm glycol is supplied daily to the snow pit. This allows ice shavings to be melted.

The heating glycol loop utilizes waste heat from the primary ammonia refrigeration loop. A portion of the vapor ammonia is extracted from the refrigeration loop and used to heat the glycol. The glycol heater is a phase-change shell and tube heat exchanger and transfers the heat from the ammonia to the glycol. In the heat exchange process, the ammonia undergoes a

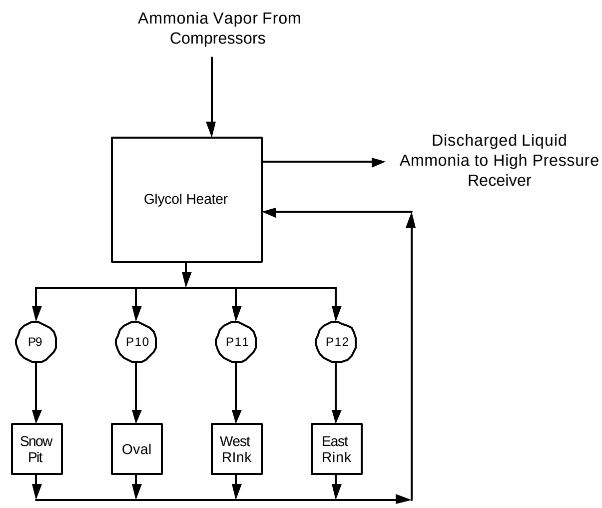

Figure 2.4: Heating Glycol Loop

phase-change of vapor to liquid. After the phase change, the liquid is then discharged to the high-pressure receiver.

From the analysis of the current system from section two, it was found that the PNIC is in need of an energy recovery system to lower utility bills. Section three will outline the available energy recovery options that may be implemented to make the PNIC more energy efficient. An initial assessment of each option's feasibility is given to determine which options are most favorable to the current system.

3. ENERGY RECOVERY OPTIONS

3.1 Introduction

Energy recovery systems typically incorporate heat exchange equipment to reduce energy costs. Energy recovery systems are designed to extract heat that would otherwise be wasted or expelled to the environment, and have become extremely popular within the last few years due to their economical and environmental benefits. Such benefits include energy cost savings, cost effectiveness compared to installing a new system, and reducing fuel consumption.

Waste energy recovery systems are implemented throughout the world. Some applications that generate wasted energy include industrial heating processes (i.e. releasing steam through stacks), refrigeration processes that expel heat directly to the evaporative condenser, and the rejection of heat from industrial components, such as compressors. In order for companies to stay competitive, several steps are chosen to reduce energy costs and other unnecessary overhead.

For this project, several options for heat recovery were considered. Such options include run-around coil systems, regenerative heat wheels, heat pipes, fixed-plate exchangers, single condenser systems, dual condenser systems, heat recovery heat pumps, cascade multi-stage systems, desuperheaters, and shell and tube heat exchangers. The following sections give a general description of each option considered, a short feasibility consideration, and a few benefits of each option.

3.2 Run-Around Coil Systems

A run-around coil system is a simple piping loop that has a pump, a finned-tube coil in the exhaust air, and a finned-tube coil in the supply air. The run-around coil system recovers energy by heating the system fluid in the exhaust air and using that heat to warm the supply air. For example, waste heat in the process cooling water from the equipment can be recovered. Water chiller waste

heat can provide domestic hot water and space heating for laboratories and offices with a run-around coil system. (Schicht, 1991) Run-around coil systems are primarily used with HVAC applications to recover sensible heat. A liquid-to-liquid heat exchange application is not generally used with run-around coil systems, and was therefore not considered as an energy recovery option.

3.3 Heat Pipes

Heat pipe exchangers offer a unique, compactly sized method of energy recovery that requires no external energy; however, they are not often used because of the restrictions placed on the location of supply and exhaust air streams. A heat pipe is a closed loop evaporation-condensation cycle. The heat pipe includes a heat transfer fluid that, when heated by a source, vaporizes and flows towards the cool end. The heat transfer fluid then condenses, releasing the heat of vaporization. After the heat transfer fluid condenses, it flows through designated channels back to the warm end for recycling. Heat pipes require that the heat transfer fluids be adjacent to each other, which limits the design options and makes heat pipes infeasible.

3.4 Regenerative Heat Wheels

A regenerative heat wheel is a revolving disc filled with an air-permeable medium. When the exhaust air passes through the medium, heat is transferred to the medium. As the medium rotates into the make-up air stream, the warmed medium transfers the heat to the cooler make-up air. Regenerative heat wheels require relatively high maintenance due to the rotating parts and there is a possibility of cross-contamination (Besant, 1995). Regenerative heat wheels are often used with air-to-air systems and not a feasible energy recovery option for the PNIC.

3.5 Plate Frame Heat Exchangers

Plate frame heat exchangers are often used for industrial, commercial, HVAC, heat recovery, refrigeration, and many other applications. Plate frame heat exchangers are typically

coated, fluid-to-fluid aluminum heat exchangers that have no moving parts. A gasket separates alternative layers of plates and the streams flow through alternate passages. Heat is transferred directly from the warm fluid through the separated plates into the cooler fluid. The design of a plate frame heat exchanger creates a turbulent flow and ultimately high heat transfer rates. Plate frame heat exchangers use counter-flow, parallel flow, and cross-flow patterns. Plate frame heat exchangers are often rather large (J.L. Herman and Associates Inc.). Due to the limited space in the Pettit National Ice Center, a plate frame heat exchanger is not a feasible energy recovery option.

3.6 Heat Recovery Heat Pumps

A heat pump extracts heat from a source and transfers it to a sink at a higher temperature. A heat pump operates like a refrigeration unit with the only difference being the desired effect. A heat pump adds heat and a refrigeration unit removes heat (Van Wylen). Heat pump advantages include simple controls, its ability to increase fluid quality, and its flexible installation in existing systems. (Dorgan, 1999) In order to implement a heat pump with the PNIC, several changes need to be made to the existing system. New AHU heating coils designed for a lower entering water temperature (130° F vs. 200° F) would need to be added to the system. The lower exiting water temperature would result in a larger heating coil (more rows of coil) and the addition of more fins to the outside of the coil to improve the heat transfer to the air. These changes would increase the pressure drop of both the air and the working fluid. The increased change in pressure would require new AHU fans and pumps for the heating system. Although still a feasible option, replacing the entire HVAC system would be costly and leave the PNIC inoperable for a large period of time.

3.7 Single Condenser System

The single condenser system sends the condenser water to the heating coils before reaching the cooling tower. When heat is to be recovered from a single-condenser chiller, the condenser water circuits are normally designed according to one of the following approaches: a closed-circuit cooler is used to reject excess heat, the heat recovery chiller is dedicated to the heating operation, and the open cooling tower is on a common circuit with heat recovery (Dorgan, 1999). Some advantages of a single condenser system are: simple control strategy, efficiency maintained during cooling, and lower first cost for chillers (Dorgan, 1999). The single condenser system was considered infeasible because it is incapable of economically providing water temperatures above 130°F and would require redesigning the HVAC system similar to the heat pump system.

3.8 Dual Condenser System

A dual condenser system is found most often in chiller heat recovery (Dorgan, 1999). It is composed of a single condenser shell with two independent water/refrigeration circuits. One of the water circuits is connected to the heating load equipment and the other is connected to a cooling tower to reject excess heat. Some advantages of a double bundle, or dual condenser system are: standard construction may be used which minimizes the cost of heat recovery, efficiency maintained during cooling only system, and the heat recovery loop is a closed system and is isolated from the cooling tower (Dorgan, 1999). However, there are a few limitations with the dual condenser system which include: inability to provide water temperatures above 140°F, cooling tower winterization and standby heating capability are required, they are typically more expensive and require additional piping and associated equipment, and the control system for the chiller is often more complex in order to handle proper staging of condensers (Dorgan, 1999). By implementing a

dual condenser system, the PNIC's refrigeration system would need to be altered by adding a new chiller. From an economical standpoint, the dual condenser system is not a feasible option.

3.9 Cascade Multi-Stage System

The cascade multi-stage system links chillers or heat pumps in a step arrangement to increase the temperature of recovered heat. In this system, one chiller condenser provides the heat source for the next chiller evaporator. Unfortunately, it is not possible to handle small cooling loads and large heating loads simultaneously but it can handle small heating loads with large cooling loads. It is imperative that high efficiency chillers be used. Some advantages of the cascade system include: discharge temps up to 180 degrees Fahrenheit are available, standard manufacturer chillers may be employed without need for special construction, and efficiency is maintained during cooling-only operation by taking energy out of the high temperature chiller (Dorgan, 1999). Similar to the dual condenser system, a cascade multi-stage system is feasible, but not economical.

3.10 Desuperheater

A desuperheater is a heat exchanger that removes the heat from the superheated gases before the refrigerant enters the condenser. It is a device in which chiller heat can be recovered at elevated temperatures in limited quantities and in larger quantities at a lower heating temperature. A common application of a desuperheater is in preheating of domestic hot water. A desuperheater water heater is installed in a vapor compression system between the compressor and the condenser. The desuperheater utilizes hot refrigerant vapor exiting the compressor to heat water (Iowa Energy Center). The desuperheater is not feasible with the PNIC system because only a limited amount of heat can be recovered.

3.11 Shell and Tube Heat Exchanger

Shell and tube heat exchangers are well suited for several different applications. Since there is no possibility of cross contamination, virtually any two fluids can be used for the exchange of heat. The shell and tube heat exchangers are extremely flexible in their choice of materials, temperatures, and pressure limitations (Ellis, 1999). The two basic designs of shell and tube heat exchangers are the multi-pass and single pass. The multi-pass design consists of straight length tubes bent into a u-shape. The tubes are placed in a shell to contain the fluid on the outside of the tube bundle. (Ellis, 1999) The single-pass design is best suited for high fouling fluids. The single pass design allows for easy maintenance and large temperature differences between the shell-side and tube-side. Given the current conditions at the PNIC, a shell and tube heat exchanger is the most favorable energy recovery option. The relatively small size of a shell and tube heat exchanger allows for an economical and practical installation at the PNIC.

There are many energy recovery options available. Traditional run-around coil systems and regenerative heat wheels are designed for air to air or air to water heat transfer and do not apply to the PNIC. A heat pipe system requires the heat source and sinks to be relatively close and do not apply to the PNIC. A heat pump, a single condenser system, and a dual condenser system supply hot water at only 130°F and would require redesigning the entire HVAC system, which would be cost prohibitive for the PNIC project. A shell and tube heat exchanger was chosen as the most feasible energy recovery option. Along with the shell and tube heat exchanger, a run-around system with a coil will make up the complete energy recovery system. The run-around system with the coil consists of the heat exchanger and the AHU coils. Based on the other energy recovery options researched, it was found that a shell and tube heat exchanger and the run-around system with the

coil will provide the most economical and feasible solution to the needs of the Pettit National Ice Center.

4. SYSTEM DESIGN

4.1 Introduction

Given that 4.16 million Btu/h are rejected to the environment and that two natural gas boilers are supplying 2 million Btu/h to the AHUs, it is desirable to replace the boilers with a shell-and-tube heat exchanger. This section examines the feasibility of using an existing shell-and-tube heat exchanger; describes the design of a new shell-and-tube heat exchanger, a new variable capacity boiler, and pumps to circulate the heated water; and estimates the cost of installing the equipment at the PNIC.

4.2 Existing Glycol Heater

Figure B.1 shows the existing glycol heater used to prevent perma-frost and heat the snow pit. The glycol heater is a shell-and-tube heat exchanger that removes heat from the high-pressure ammonia and transfers it to a glycol solution. If the capacity of the existing glycol heater were large enough to use for the snow pit and for heat recovery, it would save a substantial amount of money.

Figure 4.1 shows a picture of the glycol heater. Ammonia vapor at 205°F enters the heater and is discharged as 205°F liquid. The heater was designed for a mass flow rate of 2719 lb/h of ammonia. The amount of heat transferred to the ammonia is given by (Incropera & DeWitt, 1996)

$$q = \dot{m}(h_{fg}) \qquad (4.1)$$

where, $\dot{m}$ is the mass flow rate of the ammonia, q is the heat transfer, and h_{fg} is the latent heat of evaporation. For the existing system, this relates to 876,982 Btu/h. If the entering water temperature to the AHU heating coil is 200°F, and the exiting water temperature is 180°F, the

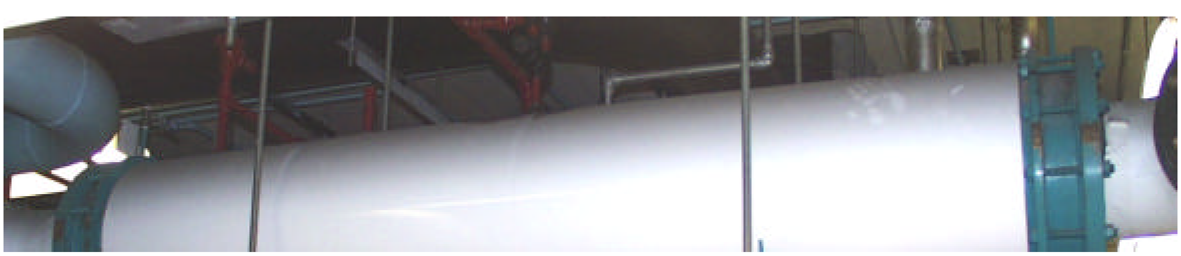

Figure 4.1: Glycol Heater

maximum volumetric flow rate of the heated water from the glycol heater can be estimated from (Incropera & DeWitt, 1996)

$$Q = \frac{q}{(\rho)(C_p)(K)(\Delta T)} \tag{4.2}$$

where Q is the volumetric flow rate in GPM, and K is a constant accounting for unit conversion. For the design conditions, the estimated flow rate is 100 GPM. The detailed calculations can be found in Appendix C. Since the design specifications for the AHUs require 175 GPM, the existing heat exchanger would be unable to supply this flow rate. As a result, a new heat exchanger is needed to meet the AHU specifications.

4.3 Heat Exchanger Design

The shell-and-tube heat exchanger to be implemented into the system must be able to provide an outlet water temperature of 200°F. Given that the water temperature will drop 20°F through the air-handling units, it is reasonable to assume an inlet water temperature of 180°F into the tube side. The vapor ammonia being discharged from the compressors will serve as the hot side fluid coming into the shell at approximately 205°F. Since the ammonia is in a liquid state, it will bypass the evaporative condenser and be piped directly to the high-pressure receiver. The schematic diagram of how the new heat exchanger is implemented into

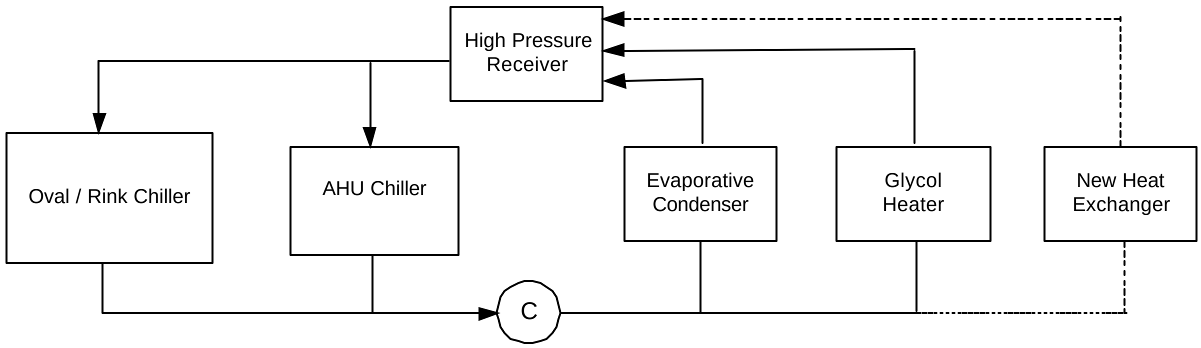

Figure 4.2: New Heat Exchanger in Existing System

the existing system is given in Figure 4.2. The heat exchanger will supply 200°F water to the air-handling units, thus eliminating the need for the boilers.

In order to properly size the heat exchanger, the mass flow and temperatures of each fluid must be known. The inlet and outlet temperatures of the water were given in the AHU design specifications as well as the required volumetric flow rate through the heating coils. Knowing that a phase change will occur in the heat exchanger, the temperature of the ammonia will remain constant at 205 degrees Fahrenheit. The necessary mass flow rate of the ammonia is the only unknown and is found to be 5253 lb/h. Detailed calculations can be found in Appendix C. The heat exchanger was designed according to these specifications.

Knowing that the heat exchanger will supply hot water to the AHUs, a pump and boiler system will be included as a safeguard in case the heat exchanger is not functioning properly. It is necessary to design and size a pump and boiler according to the heat exchanger design specifications. The boiler design is given in Section 4.3.

4.4 Boiler Design

As a safeguard in case the shell and tube heat exchanger is not supplying the necessary temperature to the AHUs, a secondary loop will be implemented containing a 1750 MBH boiler. The 1750 MBH heating load for the boiler was chosen given the extreme condition

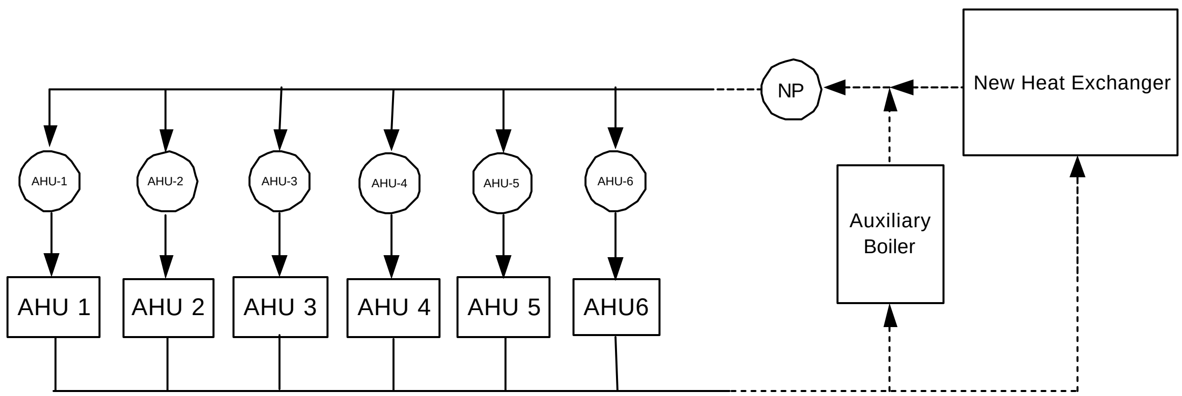

Figure 4.3: Energy Recovery System with AHUs, New HEX, New Boiler, and New Pump

that the heat exchanger installed would not be functioning at all. The AHUs need an overall volumetric flow rate of 171.6 gallons per minute with a temperature drop of 20°F. However, the small auxiliary boiler will be sized with a volumetric flow rate of 175 GPM in order to accommodate extreme conditions. The 1750 MBH was derived as follows:

$$Q = 500 \times GPM \times \Delta T \qquad (4.3)$$

$$Q = 500 \times 175 \times (200°F - 180°F) \qquad (4.4)$$

$$Q = 1{,}750{,}000 \text{ BTU/h} = 1750 \text{ MBH} \qquad (4.5)$$

A 3-way temperature valve will be added before and after the secondary loop to allow the system to bypass the boiler if the water leaving the heat exchanger is at the desired temperature. The addition of the energy recovery system will eliminate the need for the existing boilers, thus making the PNIC more energy efficient. The schematic diagram of how the AHUs, new heat exchanger, new boiler, and possible pump implemented into the existing system is given in Figure 4.3.

The piping and pumps that lead to the AHUs from the boilers will remain in place and be utilized for the new design so as to save on installation costs. The small auxiliary boiler will be installed where the current boilers are located in the PNIC. A pump was not designed for the

energy recovery system because most of the calculations have come primarily from equipment design specifications. Actual operating data was not obtained and therefore a pump could not be correctly sized or selected. The schematic diagram of the existing heating and cooling system is shown in Figure B.1 and the schematic diagram of the existing heating and cooling system along with the energy recovery system is given in Figure B.2.

4.5 Cost Analysis

The aim of installing a new energy recovery system at the Pettit National Ice Center was to be able to reduce fuel costs caused by heat being lost to the environment. Table E.1 in Appendix E charts the monthly fuel costs for the PNIC from May 2001-January 2002. The PNIC uses an average of 19,090 therms per month, which is equivalent to 1.9 billion Btu per month. Assuming an average boiler efficiency of 70 percent, the average price per Btu from May to January turned out to be $\$4.116 \times 10^{-6}$. Over those nine months the PNIC was paying on average $7857.24 per month on fuel costs. The detailed calculations can be found in Appendix G. By implementing the energy recovery system, the existing boilers will not be needed, thus eliminating most of the fuel usage each month.

The existing boilers will not be needed as a result of the implementation of the energy recovery system. It is estimated that the two boilers can be salvaged for $30,000. The new auxiliary boiler was sized to be 2 million Btu/h, with a cost of $22,000. The existing boilers are 4.4 million Btu/h and had a value of $40,000 per boiler when originally purchased. The $30,000 obtained by salvaging the existing boilers will be used to partially cover the initial costs of the energy recovery system.

An estimate for a phase-change shell-and-tube heat exchanger with a heat transfer load of 1.7 million Btu/h was given to be $5000.00. An estimate for the three-way temperature valve was given to be $1200.00. Compared to the price of the heat exchanger and the valve, wiring is negligible.

The installation process is estimated to be 2 months. This was determined by considering pipe installation, boiler removal, boiler installation, heat exchanger installation, and pump installation. Considering 2 months, eight hours a day, and $50.00/hr for labor and materials, this installation cost is $24,000. The estimated cost of engineering fees was figured to be $75,000 for 500 hours of consulting.

The total cost of the project is estimated to be $127,200. It should also be noted that $7,860 will be saved per month by eliminating the two boilers and $30,000 will be acquired for salvaging the existing boilers. The project, therefore, has a total estimated cost of $97,200 and has a payback period of roughly 13 months. All detailed calculations can be found in Appendix E.

5. Conclusions and Recommendations

The objective of this project was to select and design a waste energy recovery system for the PNIC. The existing system was examined and found to reject 4.16 million Btu/h of energy to the environment while a natural gas boiler supplied roughly 2 million Btu/h of heat. Different methods of heat recovery were examined to determine which are most feasible with the PNIC system. A shell and tube heat exchanger with a small auxiliary boiler was selected and designed to recover heat from the ammonia refrigeration system and supply heat to the boiler. The total cost of the system is estimated at roughly $97,200.

The main focus of this project was to reduce the cost of energy bills by reducing the amount of fuel used to heat the facility. By providing an energy recovery system, significant savings will be realized in approximately one year. The payback period for this project would be 13 months, after which the PNIC would experience savings of approximately $90, 000 yearly in fuel costs.

Several aspects of the design need further investigation before implementing the energy recovery system. Such investigations should include installing flow meters, temperature gauges, and pressure gauges to determine actual properties of the existing system. With the implementation of these instruments, the mass flow rates, temperatures, and pressures of the existing system can be determined. With the actual system properties known, it can be determined whether or not the four existing boiler pumps can be utilized in the energy recovery system.

To determine if the existing boiler pumps can be used, the total head of the existing system must be determined. The total head of the existing system and the energy recovery system at a specified mass flow rate must be determined as well. If the total head of the existing system and the energy recovery system is beyond the capacity of the existing boiler

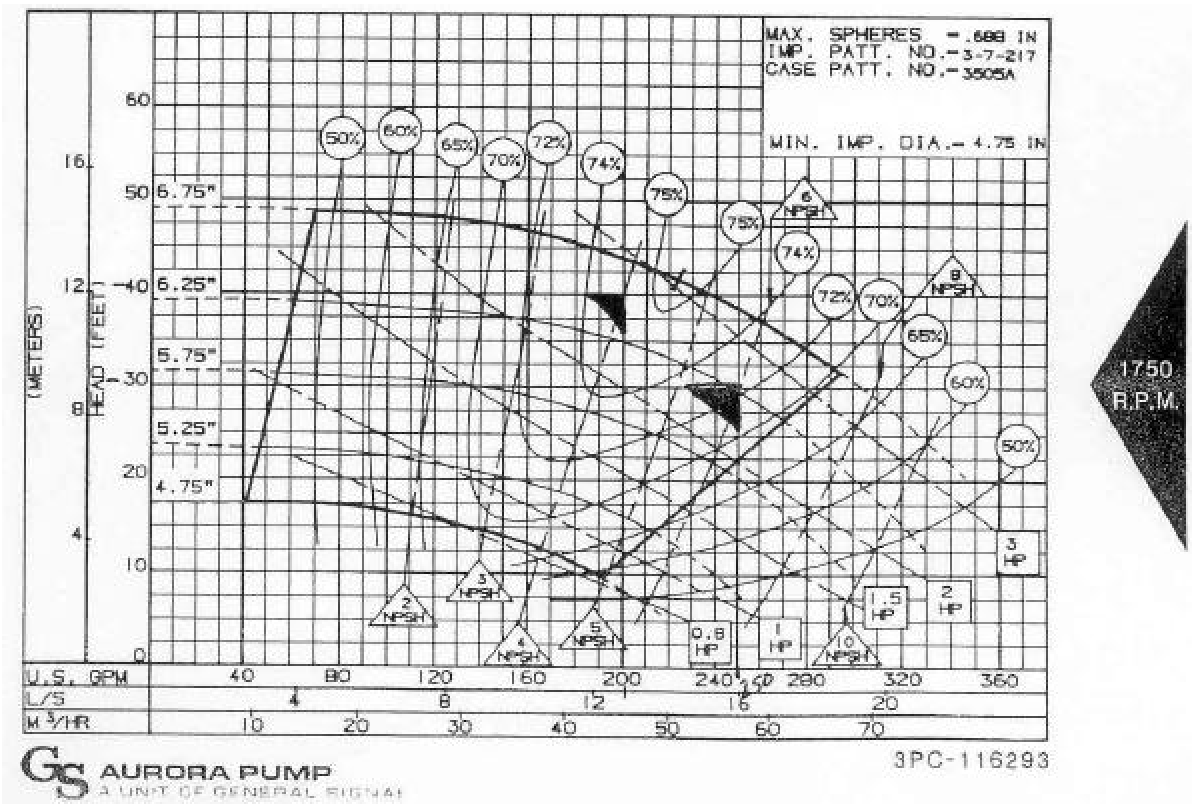

Figure 5.1: Pump Curve

pumps, it will not feasible to use the existing boiler pumps. The total head for a given flow rate can be determined using a pump curve. A typical pump curve is given in Figure 5.1.

There are four existing boiler pumps located in the boiler room (see boiler pumps in Figure B.1 in Appendix B). Of the four boiler pumps, two are 3-horsepower and create 20 ft. of head, while the other two are 15-horsepower and create 75 ft. of head. The two smaller pumps are the primary pumps and the two larger pumps are secondary pumps. If the existing pumps are large enough to handle the existing system including the new energy recovery system, the total cost of the project will be reduced.

Another point of interest is the temperature control of the three-way valve. The possibility of the water leaving the heat exchanger at a temperature above 200°F must be considered. A means to reduce the temperature leaving the heat exchanger is needed for the overall system design as well.

One way to decrease the temperature of the water is to install a throttling valve. By installing a throttling valve, a pressure drop will decrease the temperature of the water. However, the effect of the pressure drop on the entire system must be considered.

A major factor in the decision to implement a shell and tube heat exchanger as the primary source of energy recovery was the limited amount of space allotted in the mechanical room. It is desired to install the new heat exchanger in close proximity to the compressors, which will limit the amount of stainless steel pipe needed. Currently, there is approximately 30 square feet of available space next to the glycol heater. This is the most desirable location for the shell and tube heat exchanger. By eliminating the existing boilers and replacing them with a smaller auxiliary boiler, more space will be available for extra piping needed in the energy recovery system. This extra piping includes the three-way temperature valve, bypass system, and the possibility of extra pumps if the existing pumps are not feasible.

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APPENDIX A

		Shell Side	Tube Side
Fluid Name		Ammonia	40WT% Ethylene Glycol
Fluid Quantity, (in/out) [F]		10295	3,999,010
	Vapor (in/out)	0 / 10295	0/0
	Liquid	10295 / 0	3999010 / 3999010
Temperature (in/out) [F]		5 / 5	18.8/17
Dew Point [F]		5	--
Density (in/out) [lb/ft^3]		41.169 / 0.117	66.37 / 66.368
Viscosity (in/out) [cp]		0.191 / 0.009	8.809 / 9.214
Specific Heat (in/out) [Btu/(lb*ft)]		1.084 / 0.499	0.808
Thermal Conductivity [BTU/(ft*h*F)]		0.325	0.257
Latent Heat [Btu/lb]		565	--
Inlet Pressure [psia]		34.2	65
Velocity [ft/s]		--	7.8
Pressure Drop (Allow./Calc.) [psi]		0/0	20 / 1.139
Heat Exchanged [Btu/h]		5,821,302	

Table A.1: Oval/Rink Chiller Specifications

*Table provided by Cimco Refrigeration

Table A.2: Compressor Specifications

	Compressor Manufacturer	**Compressor Capacity**	**Motor Manufacturer**	**Motor Horsepower**	**Drive Arrangement**
Compressor 1	Vilter	79.5 TR @ 1130 rpm	Marathon	125 @ 1800 rpm	Belt
Compressor 2	Vilter	79.5 TR @ 1130 rpm	Marathon	125 @ 1800 rpm	Belt
Compressor 3	Vilter	79.5 TR @ 1130 rpm	Marathon	125 @ 1800 rpm	Belt
Compressor 4	Vilter	79.5 TR @ 1130 rpm	Marathon	125 @ 1800 rpm	Belt
Compressor 5	Vilter	79.5 TR @ 1130 rpm	Marathon	125 @ 1800 rpm	Belt
Compressor 6	Vilter	79.5 TR @ 1130 rpm	Marathon	125 @ 1800 rpm	Belt
Compressor 7	Vilter	135.6TR @ 1200 rpm	Marathon	150 @ 1800 rpm	Belt
Compressor 8	Vilter	135.6TR @ 1200 rpm	Marathon	150 @ 1800 rpm	Belt

*Table provided by Cimco Refrigeration

Table A.3: Evaporative Condenser Specifications

Unit Type	Counter Flow Blow-Through
Capacity	12,646 MBH
Fluid Type	Ammonia R717
Fans	(2) – 15 Hp
Fans	(2) – 7.5 Hp
Size	36'-2" L x 13' 2-3/4" H x 9' 11-3/4" W

*Table provided by Cimco Refrigeration

Table A.4: Pump Specifications

	Pump Function	Horsepower	Actual RPM	Voltage
			Shell Side	**Tube Side**
Pump 1	Chiller	60	1185	460
Pump 2	Chiller (Stand By)	60	1185	460
Pump 3	Chiller	60	1185	460
Pump 4	Oval Cooling	125	1760	460
Pump 5	Oval Cooling (Stand By)	125	1760	460
Pump 6	Oval Cooling	125	1760	460
Pump 7	West Arena	25	1760	460
Pump 8	East Arena	25	1760	460
Pump 9	Snow Pit	7.5	1770	460
Pump 10	Oval Heating	7.5	1770	460
Pump 11	West Arena Heating	5	1770	460
Pump 12	East Arena	5	1745	460
Pump 13	West Condenser Water	20	1773	460
Pump 14	East Condenser Heating	20	1773	460
Pump 15	Separator	5	1745	460
Pump 16	Hot Water	3	1800	460

*Table provided by Cimco Refrigeration

Table A.5: AHU Chiller Specifications

Fluid Name		Ammonia	20WT% Ethylene Glycol
Fluid Quantity, (in/out) [F]		6030	359,440
	Vapor (in/out)	0 / 6030	0 / 0
	Liquid	6030 / 0	359440 / 359440
Temperature (in/out) [F]		22 / 22	42/32
Dew Point [F]		22	--
Density (in/out) [lb/ft^3]		40.355 / 0.171	65.17 / 65.238
Viscosity (in/out) [cp]		0.164 / 0.009	2.539 / 2.918
Specific Heat (in/out) [Btu/(lb*ft)]		1.096 / 0.541	0.926
Thermal Conductivity [BTU/(ft*h*F)]		0.316	0.279
Latent Heat [Btu/lb]		551	--
Inlet Pressure [psia]		50.2	65
Velocity [ft/s]		--	5.9
Pressure Drop (Allow./Calc.) [psi]		0 / 0	20 / 9.821
Heat Exchanged [Btu/h]		3,324,000	

*Table provided by Cimco Refrigeration

Table A.6: Glycol Heater Specification (Case A)

*Table provided by Cimco Refrigeration

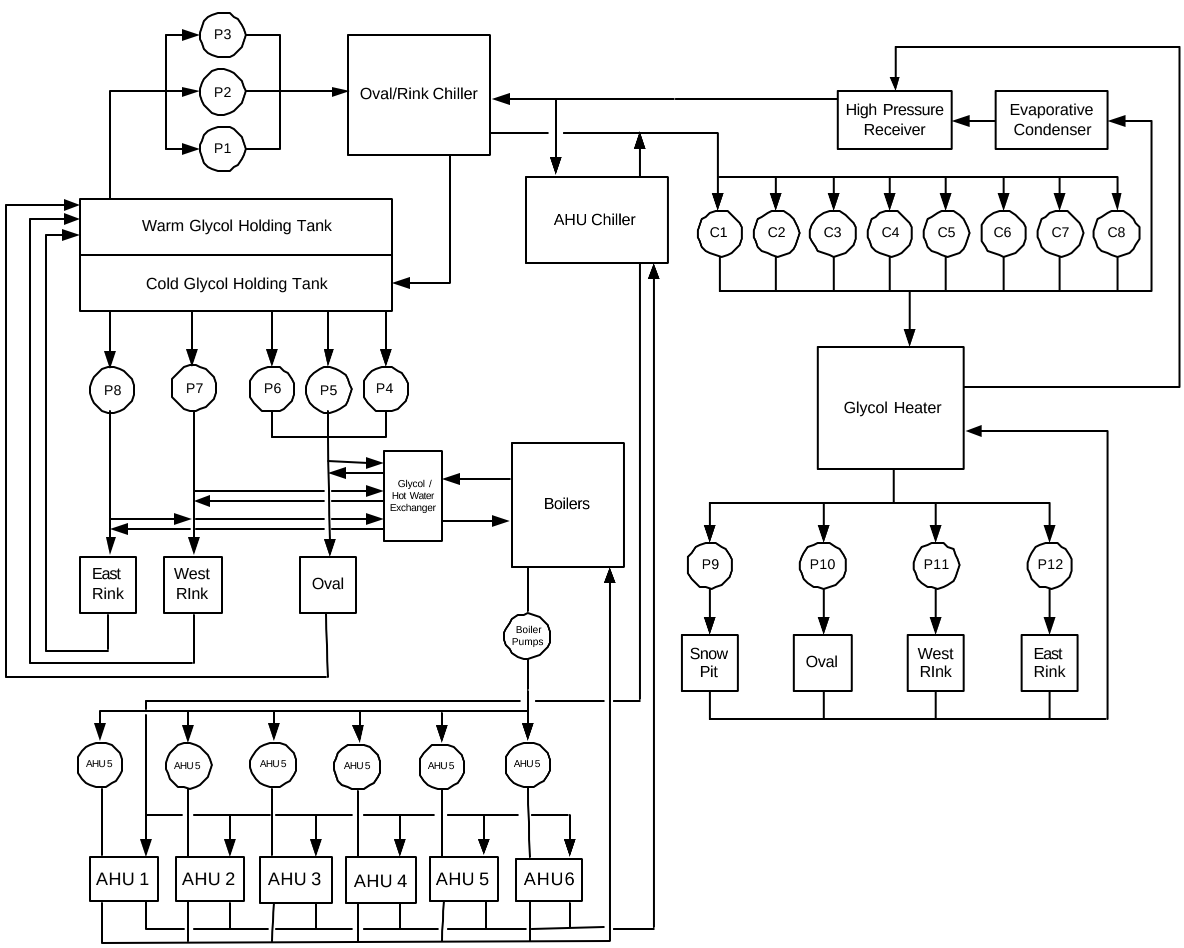
		Shell Side	Tube Side
Fluid Quantity, (in/out) [F]		2719	605696
Fluid Name		Ammonia	40WT% Ethylene Glycol
	Vapor (in/out)	2719 / 0	0 / 0
	Liquid	0 / 2719	605696 / 605696
Temperature (in/out) [F]		95 / 95	70.5 / 73.1
Dew Point [F]		95	--
Density (in/out) [lb/ft^3]		0.619 / 36.682	65.497 / 65.455
Viscosity (in/out) [cp]		0.011 / 0.098	2.658 / 2.535
Specific Heat (in/out) [Btu/(lb*ft)]		0.619 / 1.16	0.843
Thermal Conductivity [BTU/(ft*h*F)]		0.27	0.26
Latent Heat [Btu/lb]		484	--
Inlet Pressure [psia]		195.8	35
Velocity [ft/s]		2.1	6.1
Pressure Drop (Allow./Calc.) [psi]		0.5 / 0.28	15 / 1.599
Heat Exchanged [Btu/h]		1314918	

		Shell Side	Tube Side
Fluid Quantity, (in/out) [F]		446	211879
	Vapor (in/out)	446 / 0	0 / 0
	Liquid	0 / 446	211879 / 211879
Temperature (in/out) [F]		95 / 95	46.8/48
Dew Point [F]		95	--
Density (in/out) [lb/ft^3]		0.619 / 36.682	65.881 / 65.861
Viscosity (in/out) [cp]		0.011 / 0.103	4.414 / 4.286
Specific Heat (in/out) [Btu/(lb*ft)]		0.619 / 1.151	0.828
Thermal Conductivity [BTU/(ft*h*F)]		0.276	0.289
Latent Heat [Btu/lb]		484	--
Inlet Pressure [psia]		195.8	35
Velocity [ft/s]		0.3	2.1
Pressure Drop (Allow./Calc.) [psi]		0.5 / 0.009	15 / 0.311
Heat Exchanged [Btu/h]		215687	

Table A.7: Glycol Heater Specification (Case B)

*Table provided by Cimco Refrigeration

APPENDIX B



P3
P2
P1
Oval/Rink Chiller
High Pressure Receiver
Evaporative Condenser
Warm Glycol Holding Tank
Cold Glycol Holding Tank
AHU Chiller
C1
C2
C3
C4
C5
C6
C7
C8
P8
P7
P6
P5
P4
Glycol Heater
Glycol / Hot Water Exchanger
Boilers
East Rink
West RInk
Oval
P9
P10
P11
P12
Snow Pit
Oval
West RInk
East Rink
Boiler Pumps
AHU 5
AHU 5
AHU 5
AHU 5
AHU 5
AHU 5
AHU 1
AHU 2
AHU 3
AHU 4
AHU 5
AHU6

Figure B.1: Schematic of Existing PNIC System

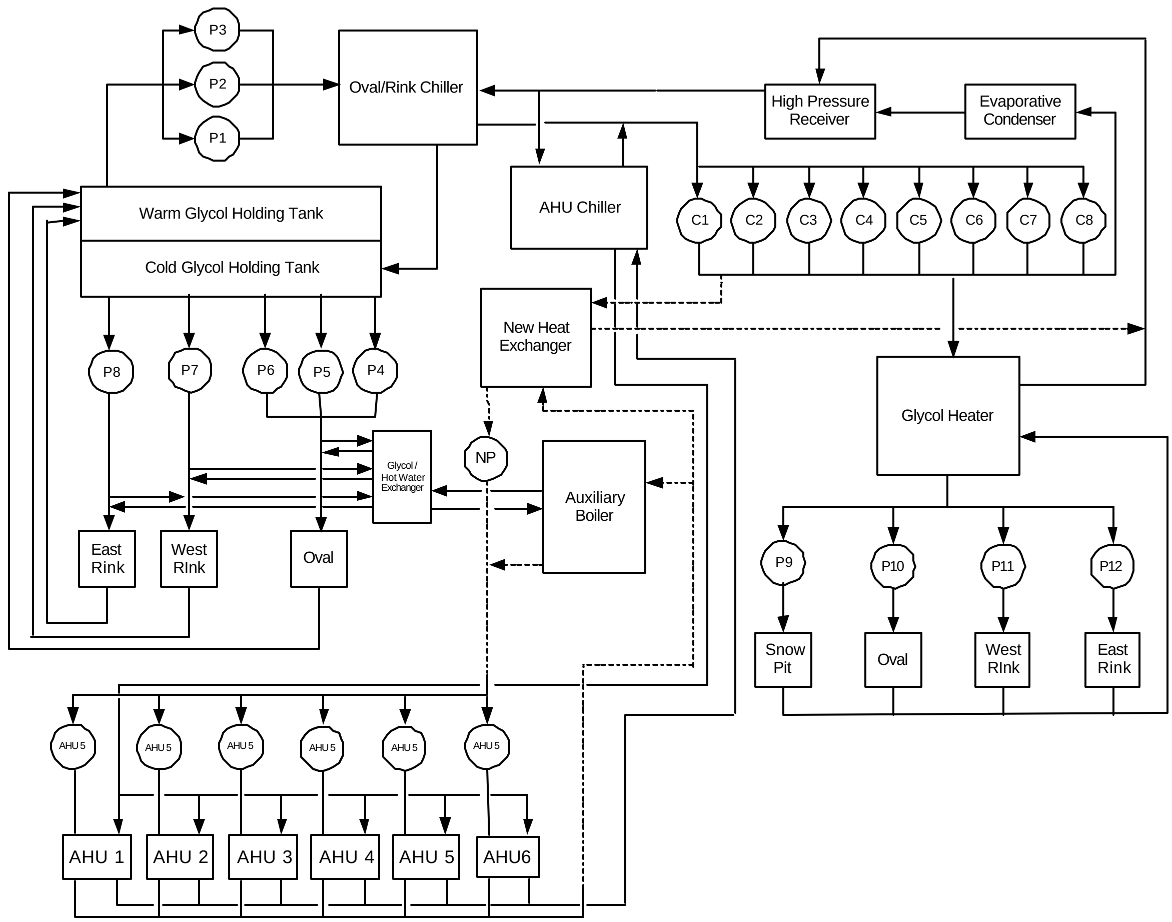


Figure B.2: Existing System With Energy Recovery System

APPENDIX C

In order to determine the maximum mass flow rate of the water to transfer a required amount of heat involves several steps. The following is the detailed process to determine the maximum amount of water.

Determine the heat transfer capabilities of the glycol heater. The enthalpy of evaporation, h_{fg}, is determined at a temperature of 205°F. Therefore, the heat transfer of the glycol heater is as follows:

$$q = (\dot{m}_g)(h_{fg}) = \text{(2719 lb/h)(322.5 Btu/lb) = -876,972 Btu/h} \qquad \text{(D.1)}$$

The glycol temperature entering the heat exchanger, T_i, is 180°F. The required glycol temperature leaving the heat exchanger, T_o, is 200°F. Now, solve for the mass flow rate.

Take T_o = 200°F and solve for the mass flow as follows:

$$\Delta T_i = 180 - 205 = -25°F \qquad \text{(D.2)}$$

$$\Delta T_o = 200 - 205 = -5°F \qquad \text{(D.3)}$$

The Log Mean Temperature Difference is found as follows:

$$LMTD = \frac{\Delta T_i - \Delta T_o}{\ln \frac{\Delta T_i}{\Delta T_o}} = \frac{-25+5}{\ln(\frac{25}{5})} = -12.43 \qquad \text{(D.4)}$$

Knowing that the LMTD is –12.93, the specific heat, C_p, can be determined. The specific heat value is equal to 0.843. The mass flow rate can then be determined as follows:

$$q = \dot{m} C_p (T_i - T_o) \qquad \text{(D.5)}$$

$$\dot{m} = \frac{q}{C_p (T_i - T_o)} = \frac{-876{,}972}{4.89(180°F - 200°F)} = 52{,}014.9 \frac{lb}{h} \qquad \text{(D.6)}$$

The mass flow can then be converted to gallon per minute for consistency as follows:

$$\dot{m} = \frac{52{,}014.9 lb}{1 hr}\left(\frac{1 ft^3}{65.497 lb}\right)\left(\frac{7.481 gal}{1 ft^3}\right)\left(\frac{1 hr}{60 \min utes}\right) \qquad \text{(D.7)}$$

$$\dot{m} = 99.01 \frac{gal}{\min} \approx 100 \frac{gal}{\min} \qquad \text{(D.8)}$$

When it was decided to implement a new heat exchanger, the required mass flow of ammonia needed to heat water from 180°F to 200°F needed to be determined. The following are the detailed calculations to properly size the new boiler.

First, the mass flow needed must be converted to IPS.

$$\dot{m}_w = 175\frac{gal}{\min}\left(\frac{60\min}{hour}\right)\left(\frac{1\,ft^3}{7.481gal}\right)\left(\frac{60.350lb}{1\,ft^3}\right) \tag{D.9}$$

$$\dot{m}_w = 84{,}704.6\frac{lb}{hr} \tag{D.10}$$

Knowing the required temperature difference of the water for the AHUs and the mass flow rate determined above, the heat transfer can be determined as follows:

$$q = \dot{m}_w C_p (T_i - T_o) \tag{D.11}$$

$$q = \left(84{,}704.6\frac{lb}{h}\right)\left(1.00\frac{Btu}{lb\cdot °F}\right)(180°F - 200°F) \tag{D.12}$$

$$q = -1{,}694{,}092\frac{Btu}{h} \tag{D.13}$$

Now, with the heat transfer known, one can find the required mass flow rate of the ammonia by the following equation:

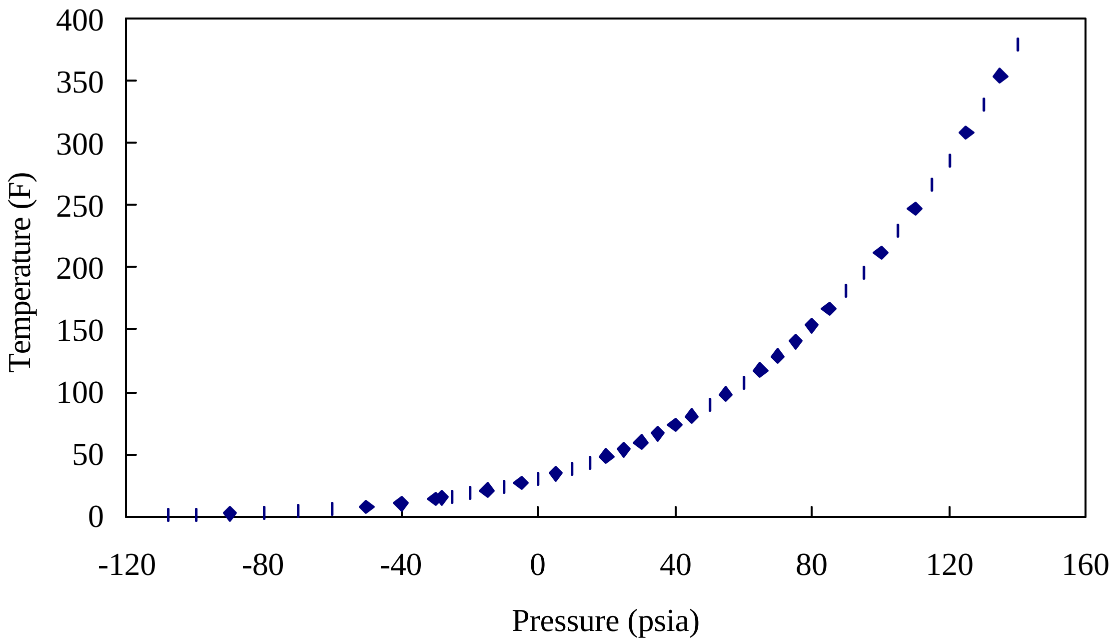
$$\dot{m}_a = \frac{q}{h_{fg}} \tag{D.14}$$

The enthalpy of evaporation, h_{fg}, for ammonia was found to be 322.5 Btu/lb at a temperature of 205 F. Therefore, the required mass flow rate of the ammonia is found as follows:

$$\dot{m}_a = \frac{1{,}694{,}092\frac{Btu}{h}}{322.5\frac{Btu}{lb}} = 5{,}253\frac{lb}{h} \tag{D.15}$$

APPENDIX D

Figure D.1: Pressure-Temperature Diagram for Refrigerant 717 (Ammonia)



APPENDIX E

Month	Therms	Btu	Cost of Fuel & Transportation [$/therm]	Price of Fuel [$/Btu]	# of Days in Billing Cycle	# of Hours in Billing Cycle	Total Heat per Hour	Cost of Fuel [$/Btu]
May	22685	2269000000	0.6234	0.000006234	30	720	3150694	0.000008906
July	14871	1487000000	0.4588	0.000004588	29	696	2136638	0.000006554
August	13667	1367000000	0.4032	0.000004032	28	672	2033780	0.000005760
September	16077	1608000000	0.4017	0.000004017	32	768	2093359	0.000005739
October	18412	1841000000	0.3145	0.000003145	27	648	2841358	0.000004493
November	20504	2050000000	0.268	0.00000268	33	792	2588889	0.000003829
December	21500	2150000000	--	--	--	--	--	--
January	25000	2500000000	--	--	--	--	--	--
Average	**19089.5**	**1909000000**	**0.4116**	**0.000004116**	**29.83**	**716**	**2474120**	**0.00000588**

Table E.1: Cost Analysis