ASSESSMENT OF DISTRICT ENERGY INTEGRATION BETWEEN BUILDINGS

by

Oladayo Oladeji

A thesis submitted in partial fulfillment
of the requirements for the degree

of

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In

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GLOSSARY or NOMENCLATURE

MSU	Montana State University
HVAC	Heating Ventilation and Air Conditioning
EWT	Entering Water Temperature
LWT	Leaving Water Temperature
DOE	Department of Energy
COP	Coefficient of Performance

ABSTRACT

District heating or cooling is a system for distributing heat or chilled water in a centralized location through various systems which is intended for residential and commercial heating or cooling requirements. Montana State University (MSU) is looking at implementing a future energy district in form of a distributed heat pump model. Implementing such system will help in reducing carbon emissions in the atmosphere, provide energy savings and ensuring energy is being used efficiently.

In the summer season, there is a lack of substantial heat sinks in which heat could be utilized and in the winter season there is a lack of substantial heat sources available due to the extreme cold weather. This project identifies systems that serves as heat sinks and sources in buildings and provides substantial energy. This project also looks into the feasibility of connecting such systems together in a building to follow a recirculating heat pump model which operates in the temperature range of 60°F – 90°F. If this model provides much energy saving opportunities, it could be incorporated in buildings on campus here at MSU and connected to the future energy district.

The project scope was limited to Barnard Hall, in which heat sources opportunities identified include the building exhaust air system and the process cooling system while heat sink opportunities identified include the domestic hot water system and the outdoor air that needs to be pre heated majority of the time in Bozeman, Montana. Energy calculations were done for each system and imputed into TRNSYS, an extremely flexible graphically based software used to simulate the behavior of transient systems. The heat pump model was then designed and simulated for a time frame of 8760 hours (A year). The researched showed that this provided some energy savings opportunities and yields no profit in general.

CHAPTER ONE

INTRODUCTION

General Background

District heating or cooling is a system for distributing heat or chilled water in a centralized location through various systems which is intended for residential and commercial heating or cooling requirements. [1] District energy systems involves both district heating and cooling. District energy is the utilization of multiple energy sources or a single energy source to provide energy services to different users usually within a community. In a district energy system, thermal energy is being distributed via underground pipelines from the energy sources to various buildings that require such.

After energy is utilized by such buildings, the return pipes bring the energy back to the source to start another cycle of cooling or heating based on the requirements. Majority of the district energy system are connected to electricity generation facilities that provide power. [2] Integrating energy systems between buildings will help solve challenges that explore ways for various building energy systems to work together more efficiently thereby reducing carbon emissions in the atmosphere, providing energy savings and ensuring that energy is being used efficiently.

MSU's energy strategy is to reduce energy usage, reclaiming existing energy from sources or sinks and develop renewable energy sources. The MSU campus has over 5 million gross square feet of building and approximately 17,000 students. MSU is looking at implementing a future

energy district in form of a heat pump model in which various buildings will be connected to this model or heat pump loop designed to operate within a certain temperature range.[3]

Problem Statement

According to the US Energy Information Administration, commercial buildings account for nearly 20% of US energy consumption and 12% of the nation's greenhouse gas emissions. It concludes that much as 30% of that energy is wasted.[4] One of the major challenges faced in energy being wasted is in the aspect of heat energy. In the summer season, there is a lack of substantial heat sinks in which heat could be utilized, hence, there is a wastage of heat energy. Also, in the winter season, there is a lack of substantial heat sources available due to the extreme cold weather, thus, the heat demand cannot be met. This fueled the decision to analyze the energy potential of heat sources and heat sinks in buildings which will help guide decisions for an energy district and help in achieving MSU’s overall energy goal or strategy.



Figure 1. District Energy System [28]

Heat sources and sinks were identified in buildings and the energy potential of connecting such to a heat pump loop was identified.

Background Study: Joshua Talbert (2014) performed an energy assessment of buildings using photographic based model development alongside modeling calibration. He also proved the efficacy of modern energy modeling tools under development such as EnergyPlus. The modeling results generated support an overall reduction of campus building related greenhouse gas emissions and proved that emerging energy modeling tools can significantly reduce the time spent on accurate model development.[5]

Joshua Hays (2019) developed a method to assess a mini-district energy loop from historical utility data. A simple building model was developed using energy modeling from building construction specifications and inputs from the University Service Engineers at MSU. Calibration for the heat demand was done and the simulated data was validated with metered data for a high energy use laboratory building on campus. The model was also converted to be heat pump compatible which allowed for internal thermal energy sharing within the building to occur. The research also showed that high energy use laboratories have large potential internal energy savings and it was recommended that in order to maximize the energy sharing capabilities within the buildings, they should be converted to heat pump compatible buildings. [6]

Kouhia, et al (2019) performed an evaluation of a mid-sized district heating system design for different design objectives. A multi objective optimization model was formulated which solved for district heat primary energy factor, minimum exergy losses, minimum C02 emissions alongside other factors. It was concluded that a combination of profit maximization, CO2 emission minimization and district heat primary energy minimization factor is recognized as a feasible set

of design objectives which can result in heat supply systems with some environmental benefits and are relatively inexpensive. [7]

Kontu, et al (2019) examined the viability of large heat pump in existing district heating systems in which it was discovered based on simulations that the largest potential for heat pump is in a small district heating system, where fossil fuel usage was reduced. Also, it was discovered that in medium and large systems with economical combined heat and power production, the potential of the heat pump is smaller. [8]

Dongellini, et al (2017) performed a numerical analysis on the way the energy consumption of the heating, ventilation and air conditioning (HVAC) systems is influenced by a reversible electrical air-source heat pump modulating capability, the heat pump sizing and the climate. It was discovered that the heat pump sizing strongly affects seasonal and annual HVAC energy consumption, the ratio between the full thermal capacity of the heat pump and the building peak load strongly influences the energy performance of the system. [9]

The Heat Pump System

A Heat Pump can be defined as a system that makes use of a reversible refrigeration cycle to perform heating and cooling by reversing the function of both heat exchangers.

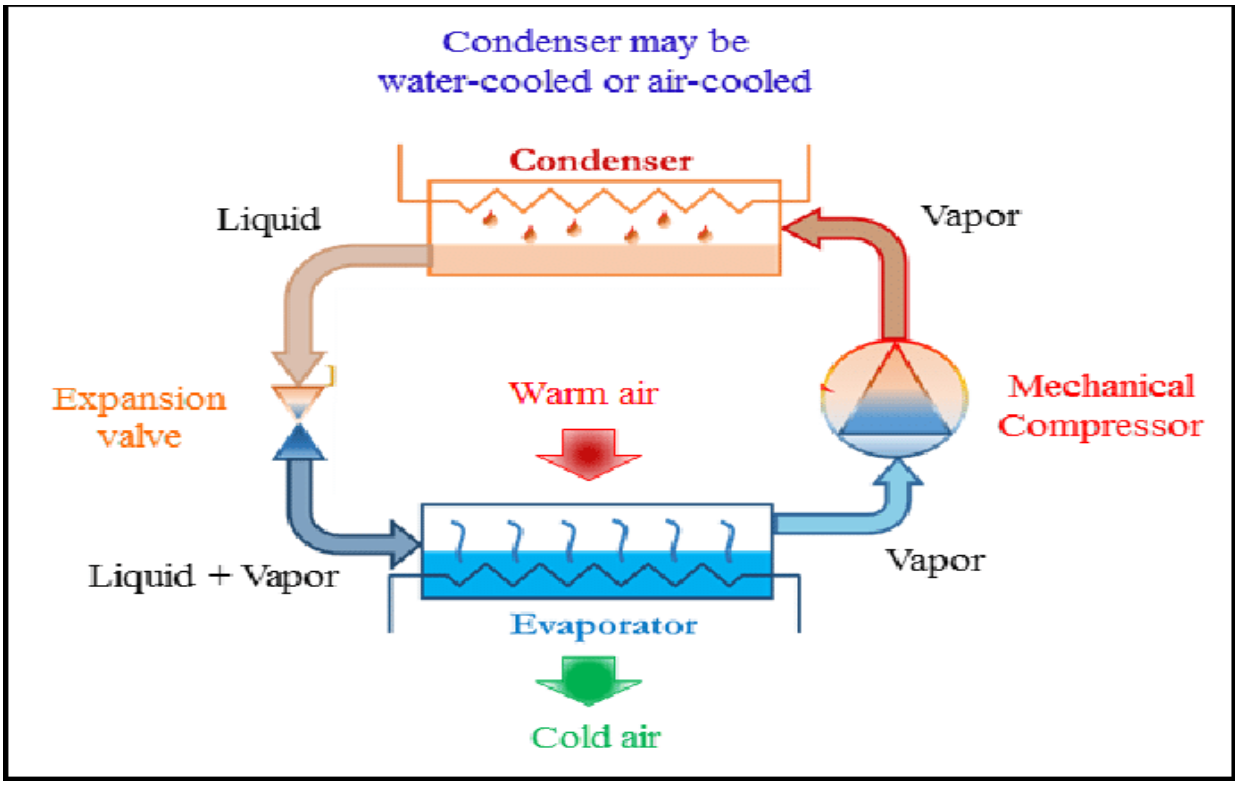


Figure 2. Vapor Compression Refrigeration Cycle [10]

The Vapor Compressor Refrigeration Cycle works by the following process;

(i) Refrigerant enters the evaporator in the form of a cool, low pressure mixture of liquid and vapor.

(ii) Heat is absorbed from the warm air or water that requires cooling changing the refrigerant to a low pressure gas.

(iii) The low pressure gas is then pumped from the evaporator to the compressor which causes an increase in pressure and temperature

(iv) The high pressure and temperature refrigerant leave the compressor and then enters the condenser where heat is rejected to the ambient air or water.

(v) This heat rejection in the condenser causes the refrigerant to change phase to the high pressure liquid state and flow from the condenser to the expansion valve.

(vi) The expansion valve creates a pressure drop that drops the pressure of the refrigerant to that of the evaporator.

(vii) The liquid vapor mixture of the refrigerant then travels to the evaporator to repeat this cycle. [11]

The major types of Heat Pumps include;

(i) Water Source Heat Pump; This type of heat pump transfers heat to or from a water source to the indoor air coil

(ii) Air Source Heat Pump; This type of heat pump transfers heat to or from the outdoor air to the indoor air coil

(iii) Water to Water Heat Pump; This type of Heat Pump transfers heat to or from one water loop to the other



Figure 3. Water Source Heat Pump used in the Norm Asbjornson Hall at MSU

CHAPTER TWO

ENERGY MODELING TOOL DISCUSSION

Energy Modeling

Energy modeling basically involves generating computer models of energy systems for the purpose of analyzing them. Results generated from modeling include system feasibility, economic analysis, system energy efficiency, greenhouse gas emissions and other technical and economic conditions involved. [12] There are many energy modeling software current available, hence it is of upmost importance to select the appropriate tool for a given application.

Factors involved in Tool Selection

The following factors were put into consideration when selecting the software to be used in designing the heat pump loop:

1. Systems Modeling: The software being selected should be able to incorporate various systems such as the HVAC systems, hydronic systems, thermal storage systems and any other systems necessary.
2. Configuration Flexibility: Different design configurations should be able to be modelled with the software selected, to determine which configuration serves as the most efficient and also to improve the energy efficiency.
3. Available Inputs: Inputs to the program should be able to provide an accurate representation of what is being required, i.e. for a pipe, the inner and outer diameter should be able to be inputed as well as the length of the pipe.

4. Available Outputs: Outputs generated should be able to show the energy usage e.g. power consumption as well as other necessary outputs.
5. Usability: A simple user interface is preferred which requires minimal expertise to operate the program.
6. Simulation Time: The program selected should be able to run simulations for a minimal period of a year(8760hours).
7. Learning Timeframe: The time frame required to get accustomed with the software should not be a lengthy period, to be able to meet up with the project deadline.
8. Interoperability: The software selected should be able to be used with other programs if necessary.

Tool Selection: After conversing with various thermal energy specialists, engineers, and professors, four major software packages were narrowed down and looked into for the energy modeling of this research. This include Trane TRACE 700, EnergyPlus, TRNSYS and the Integrated Environment Solution Virtual Environment Software (ApacheHVAC). The limitations observed after downloading and checking out these programs are stated below:

(i) Trane Trace 700; This program enables one to compare the energy and economic impact of building related selections such as architectural features, HVAC systems, HVAC equipment, building utilization or scheduling and financial options. The major disadvantage (with respect to this research) found after checking out the program is that there is no flexibility in adopting different design configuration which would help determine the most efficient design and also help improve energy savings. Also, there is a limitation to the available inputs in the program.

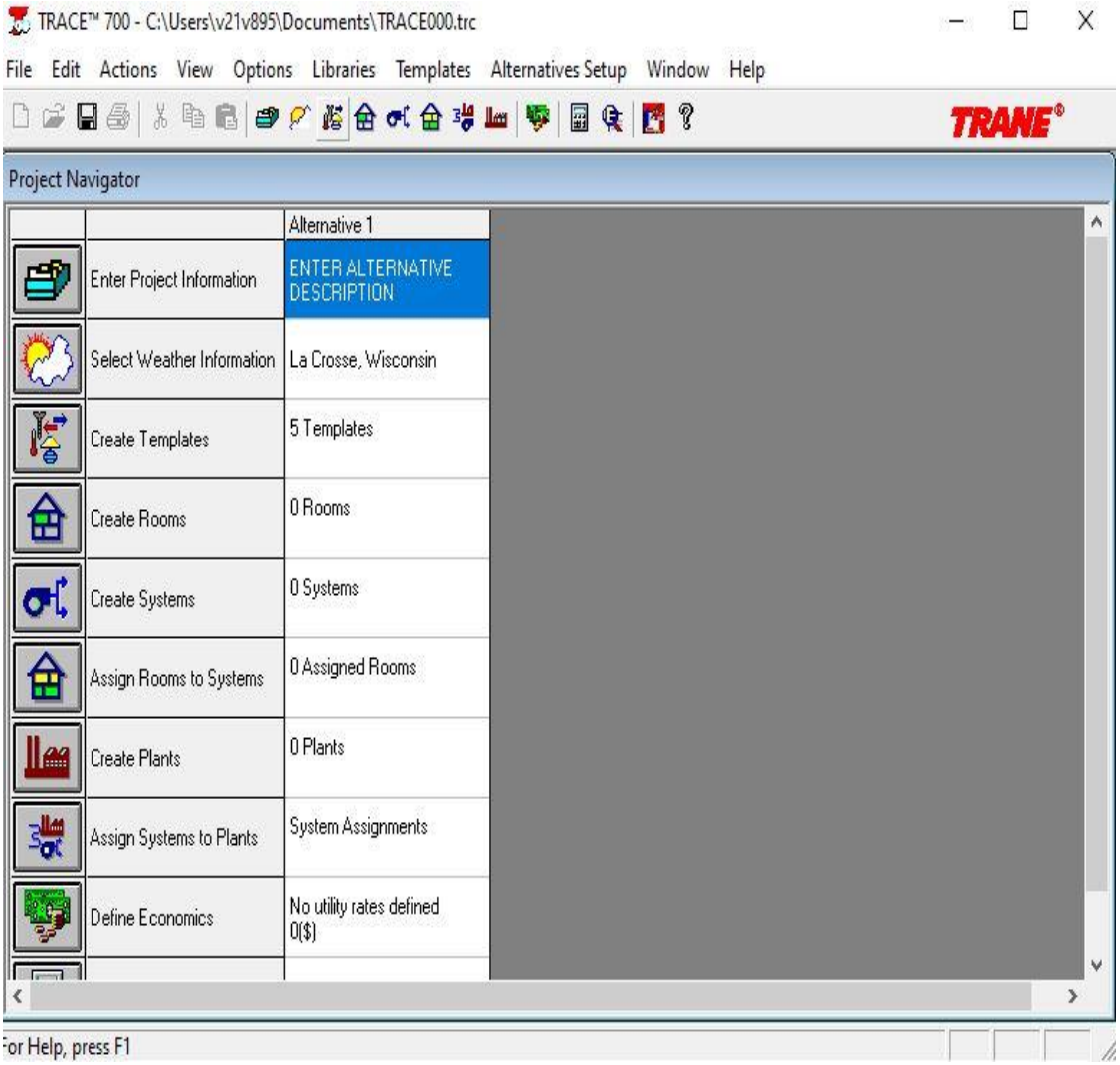


Figure 4. Trace 700 Interface

(ii) EnergyPlus: EnergyPlus is a building energy simulation tool that can be used to model the rate of energy consumption, process loads, lightning, water use, heating, ventilation and air conditioning in buildings.

[13] Although this software is free to purchase, the major disadvantage (with respect to this research) found after checking out the program is it takes much time to develop

and there are a lot of predefined standard loops. Hence, looking at the project time frame and also considering the design flexibility needed, the software was discarded.

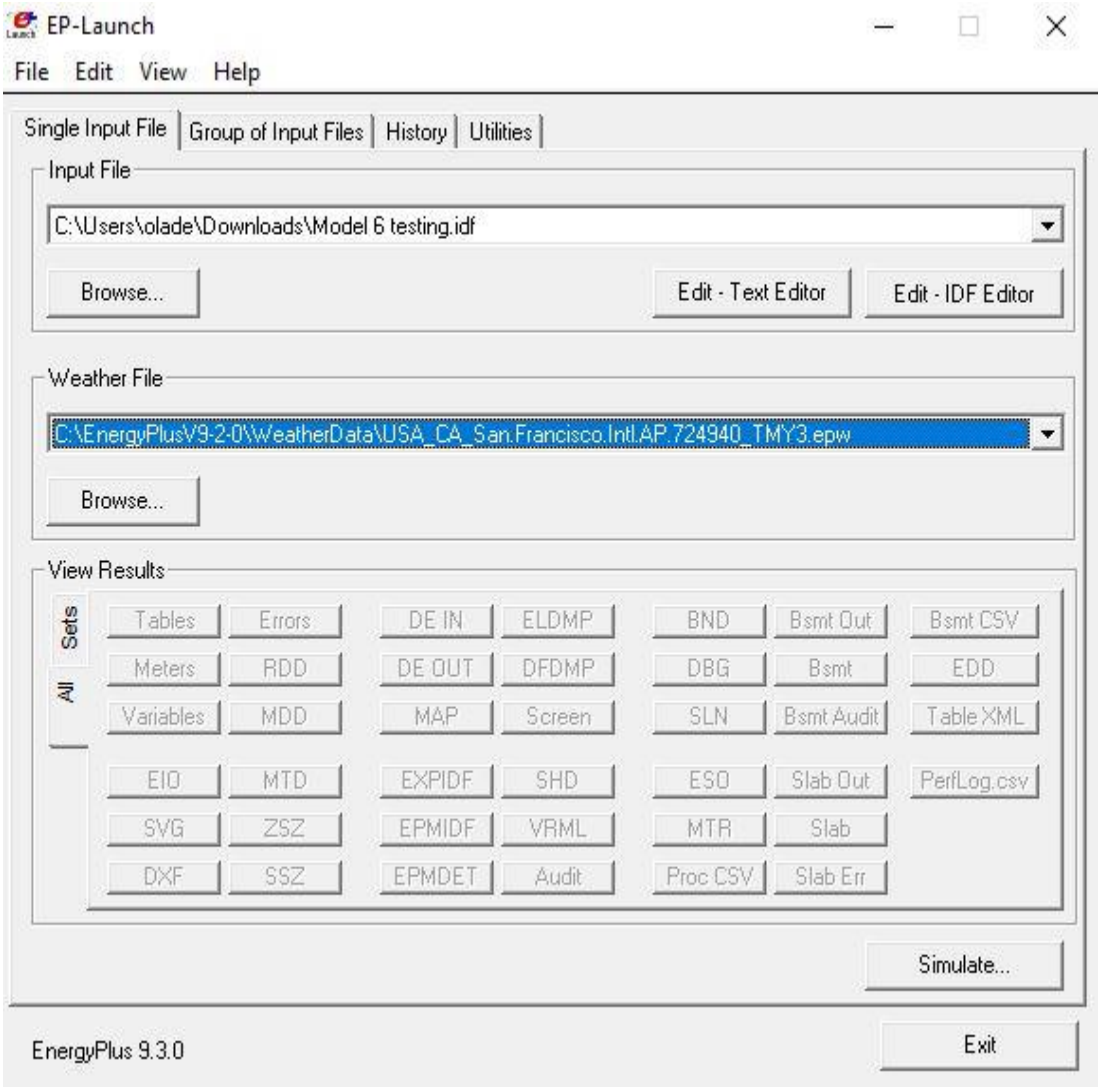


Figure 5. Energy Plus Interface

(iii) IES Virtual Environment Software: The ApacheHVAC software in this program can be used to model the HVAC system and various predefined loops as seen below. The limitations however from adopting such program stems for the fact that there are limitations on defining the arrangement for the systems for optimal performance, even

though this is more flexible compared to previous programs above. Also, the strength of the program lies in building analysis and connecting the building energy to HVAC systems, hence, the software was discarded.

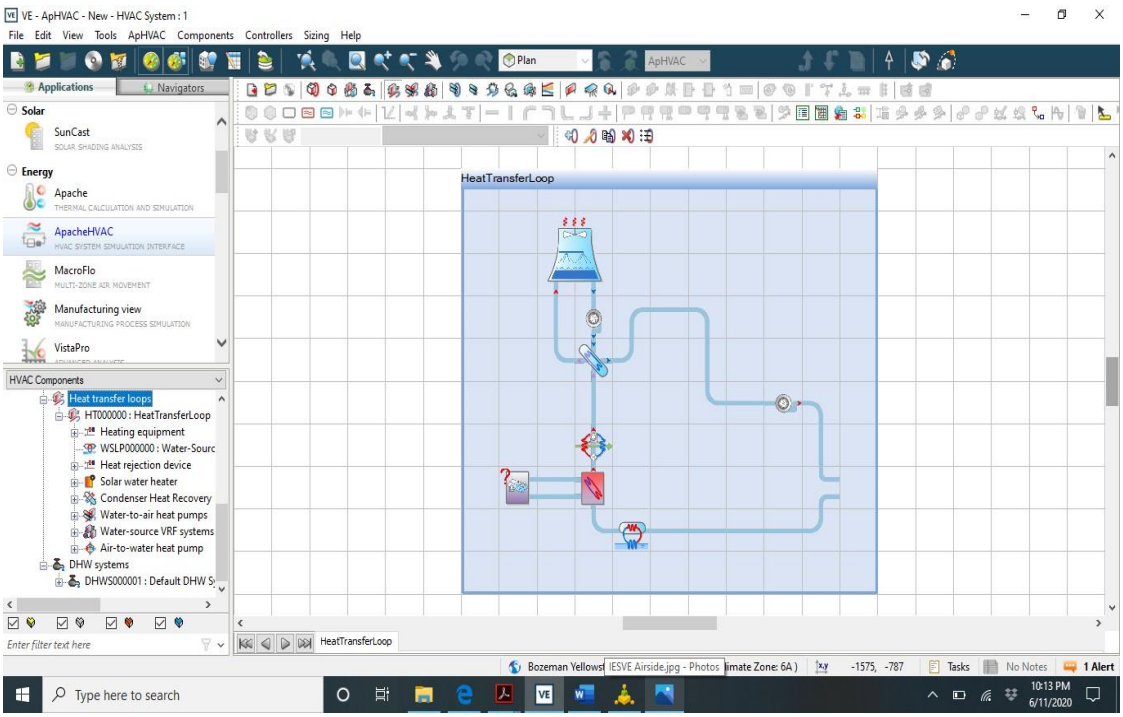


Figure 6. IESVES Interface

(iv) TRNSYS with the TESS libraries: TRNSYS is an extremely flexible graphically based software used to simulate the behavior of transient systems. The software is basically made up of two parts, the kernel that reads and processes the input file, solves the system, determines convergence and plot systems variables and the TESS library components that can model the performance of various transient systems. [14] The various TESS libraries available on the program includes;

- Application Library

- Cogeneration Library
- Controllers Library
- Electrical Library
- Geothermal Heat Pump Library
- Ground Coupling Library
- High Temperature Solar Library
- Heating, Ventilation and Air Conditioning Library
- Hydrogen Systems Library
- Hydronics Library
- Loads and Structures Library
- Obsolete Library
- Optimization Library
- Output Library
- Physical Phenomena Library
- Solar Library
- Solar Thermal Collectors Library
- Thermal Storage Library
- Utility Library
- Weather Data Reading and Processing

The various libraries above have individual components that can be used in the design when necessary. The program is user friendly and takes a reasonable amount of time to learn with respect to the project timeframe. It also offers flexible HVAC Loops. The only major disadvantage

was that the software was not free which was overlooked. Hence, the TRNSYS program was chosen for this research.

The image below shows a sample of the TRNSYS interface

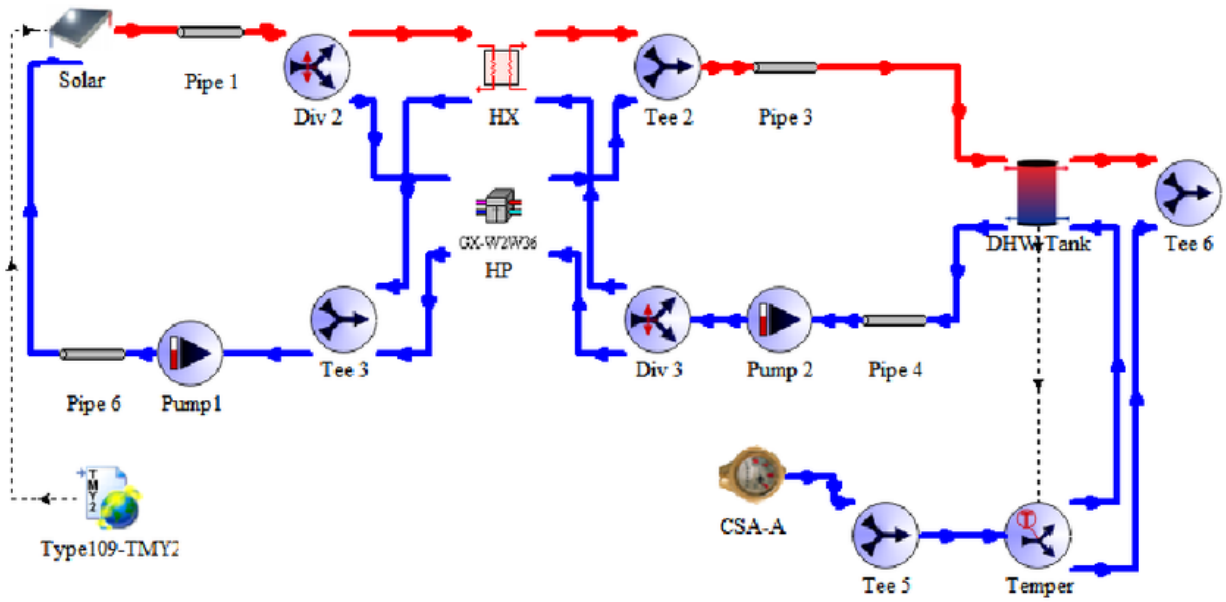


Figure 7. TRNSYS Interface [35]

CHAPTER THREE

MODEL DEVELOPMENT

Heat Sources and Sinks

Barnard Hall was chosen initially to see the feasibility of this model, if successful this model can be incorporated to other buildings on campus as well, with other buildings being connected to the overall heat pump loop. Heat sources and heat sinks were initially identified in Barnard Hall. Heat sources in general are system that generate heat energy while heat sinks are systems that harness such energy.



Figure 8. Barnard Hall

Heat sources that provided substantial opportunities identified in Barnard Hall include;

(A) The Process cooling system



Figure 9. Process cooling system in Barnard Hall

(B) Heat recovery from the exhaust part of the building



Figure 10. Exhaust air system of a building [15]

Heat sinks that provided substantial opportunities identified in Barnard Hall include;

(C) Outdoor Air that requires heating majority of the time



Figure 11. Outdoor air [16]

(D) Domestic hot water system that requires hot water during occupied hours

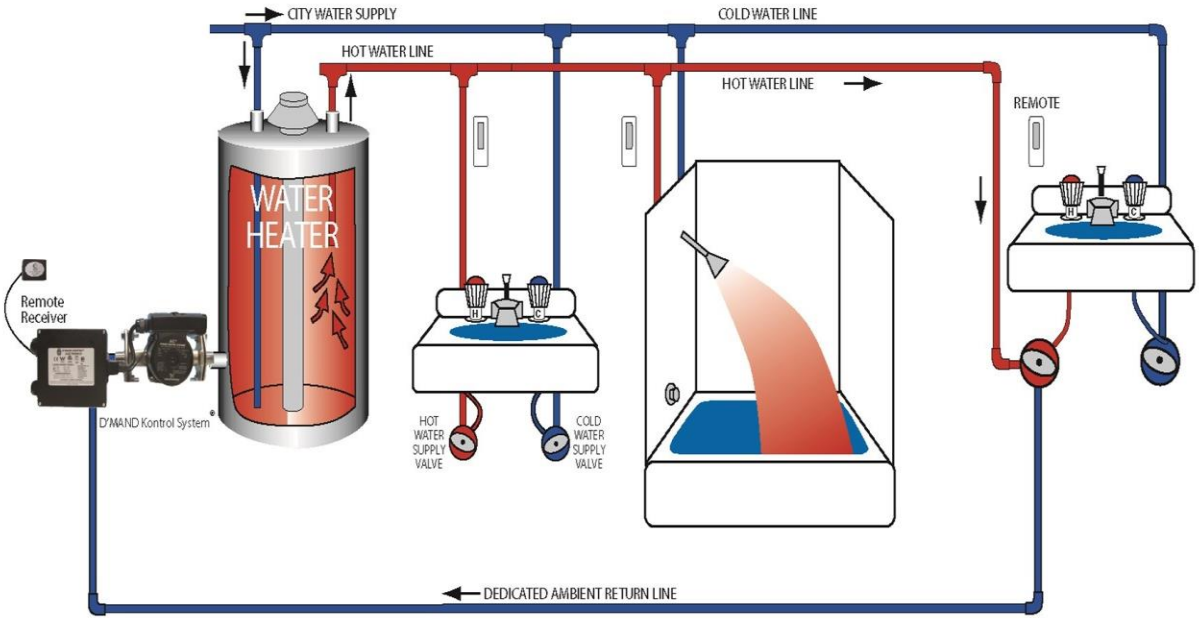


Figure 12. Domestic hot water system [17]

Each individual system was evaluated with the required loads and flow rate being calculated as shown in the next pages.

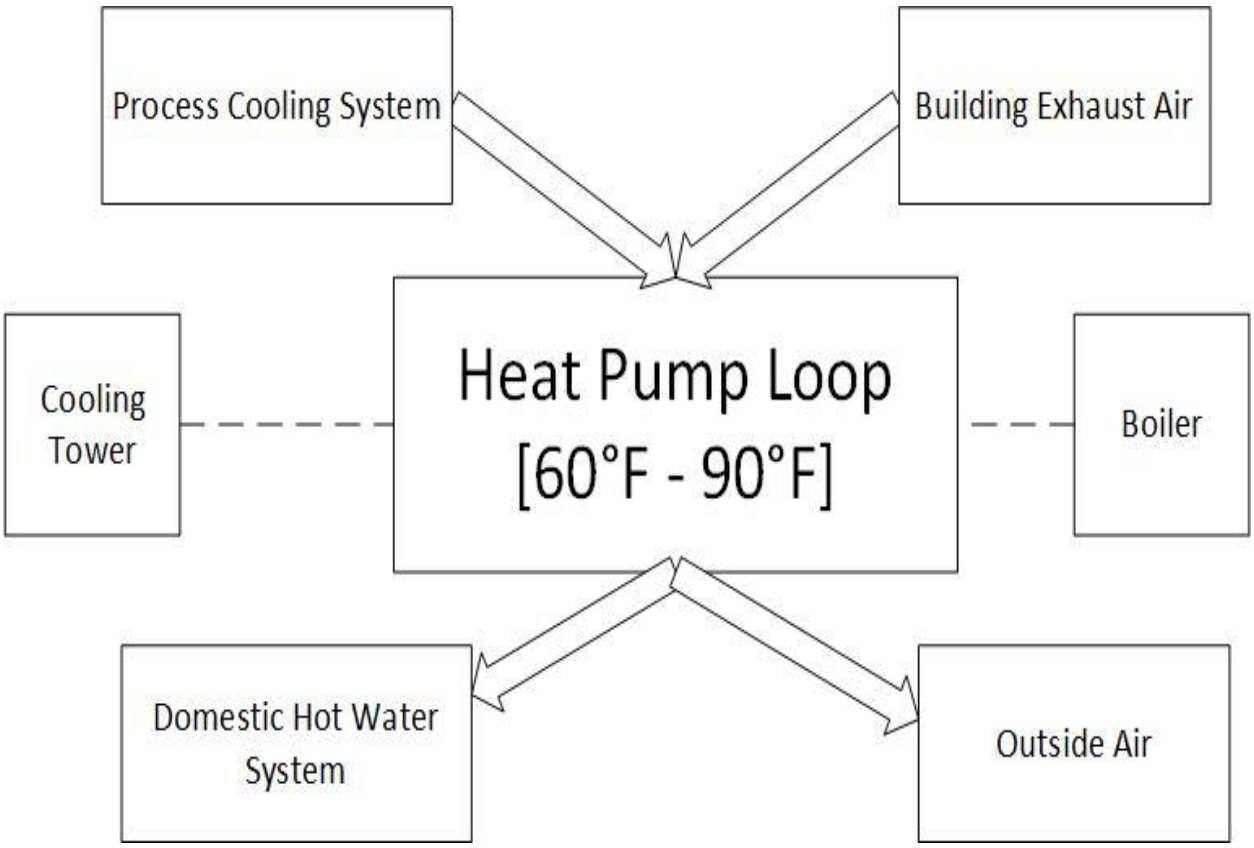


Figure 13. Heat Pump loop overview

(A) Process cooling system

The Process cooling system is a system designed to remove heat from processes, equipments or systems in a building. For the Process cooling system in Barnard Hall, the heat being removed is re-injected into the well with the help of a heat exchanger as shown in the image below and the cycle continues. Basically, we are looking at extracting the heat removed from equipments or processes with the aid of a heat pump before being re-injected to the well, this will be connected to the heat pump loop to serve as an heat source.

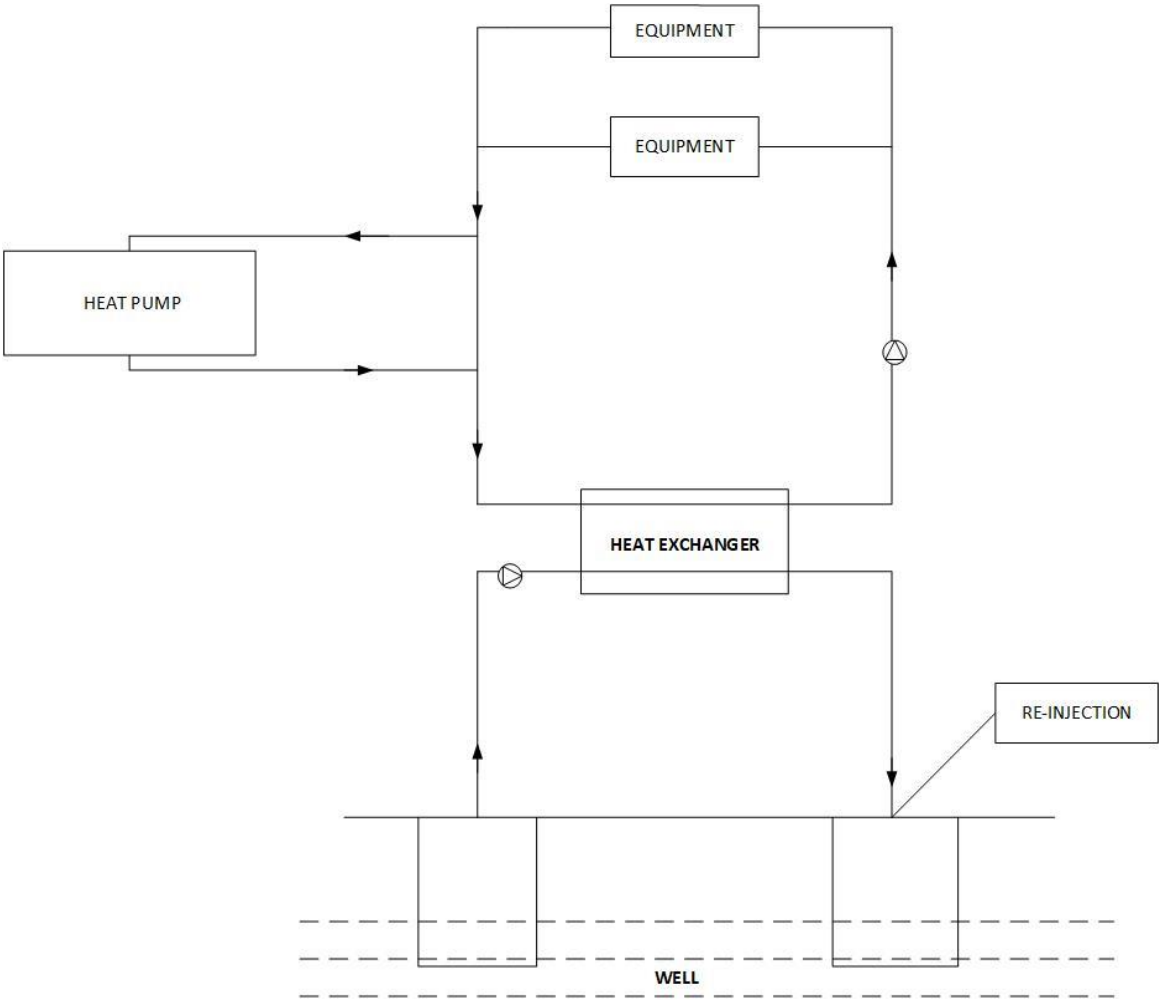


Figure 14. Process cooling system

Process Cooling Water Energy Assessment for Barnard Hall

$$Q(Btu/hr) = 500 \times \text{Flow rate (GPM)} \times \text{Temperature difference}(°F) \quad [18] \text{----(i)}$$

where Q(Btu/hr) = Energy content in Process cooling water,

GPM = Flow rate in Gallons per minute,

ΔT = Temperature difference(°F),

500 = Conversion factor (60min/hr x 8.33lbs/gallon x 1Btu/lb°F).

Temperature Difference = Chilled Water Return Temperature – Chilled Water Supply Temperature = 58°F - 55°F = 3°F. The respective temperature values were gotten from stable historical data trends from September 20th, 2019 to November 11th ,2019 provided by MSU facilities as shown in figure 15 below.

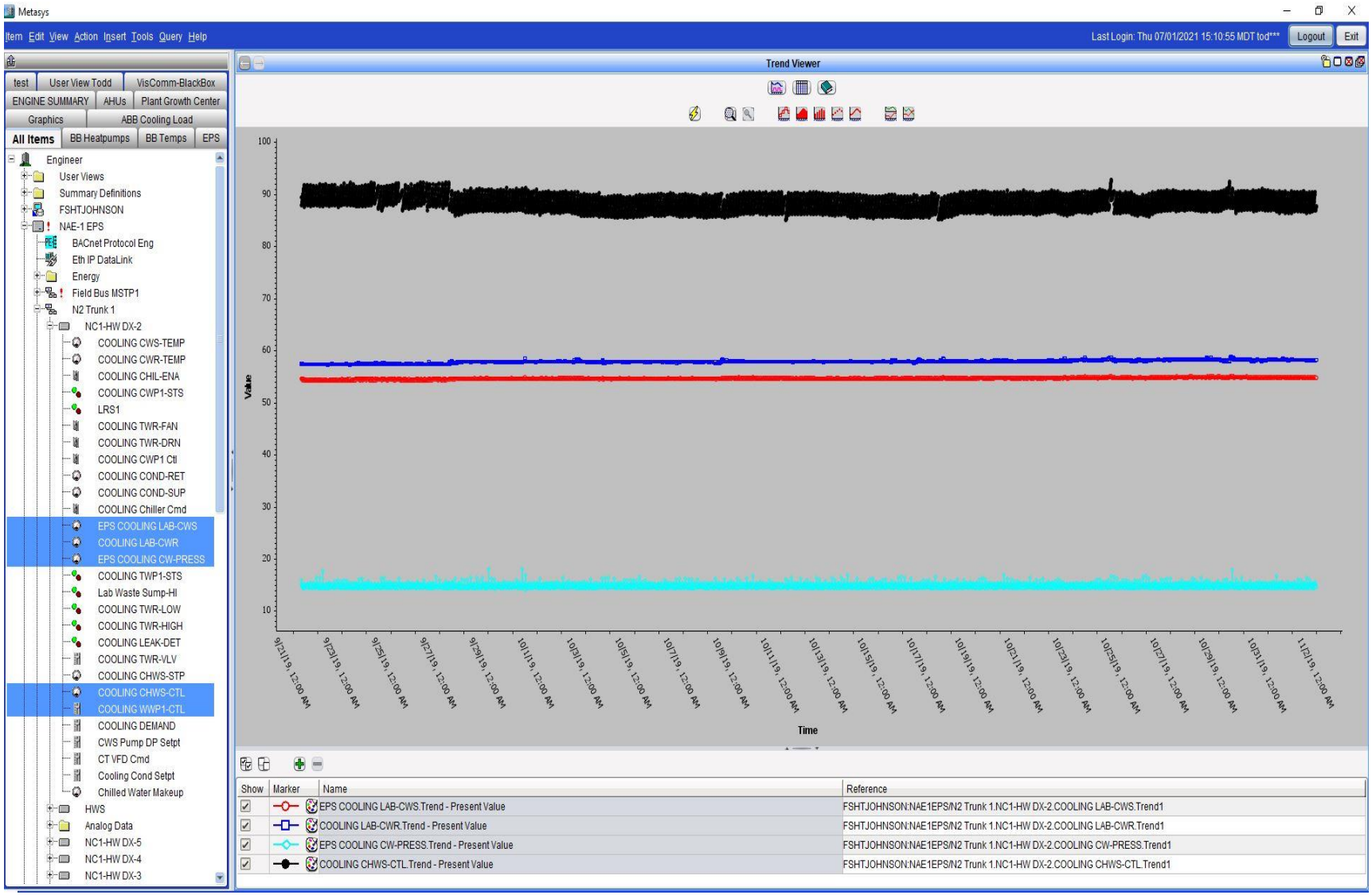


Figure 15. Process Cooling Temperature and Pressure Output

Also, Change in Pressure = Chilled Water Return Pressure – Chilled Water Supply Pressure. Thus, Change in Pressure = 74PSI – 36PSI = 38PSI (At a frequency of 60Hz).

This will be used in obtaining the flow rate from the pump curve. The respective pressure values were read from the pressure gauge by shutting the respective valve as shown below.



Figure 16. Chilled Water Supply Pressure

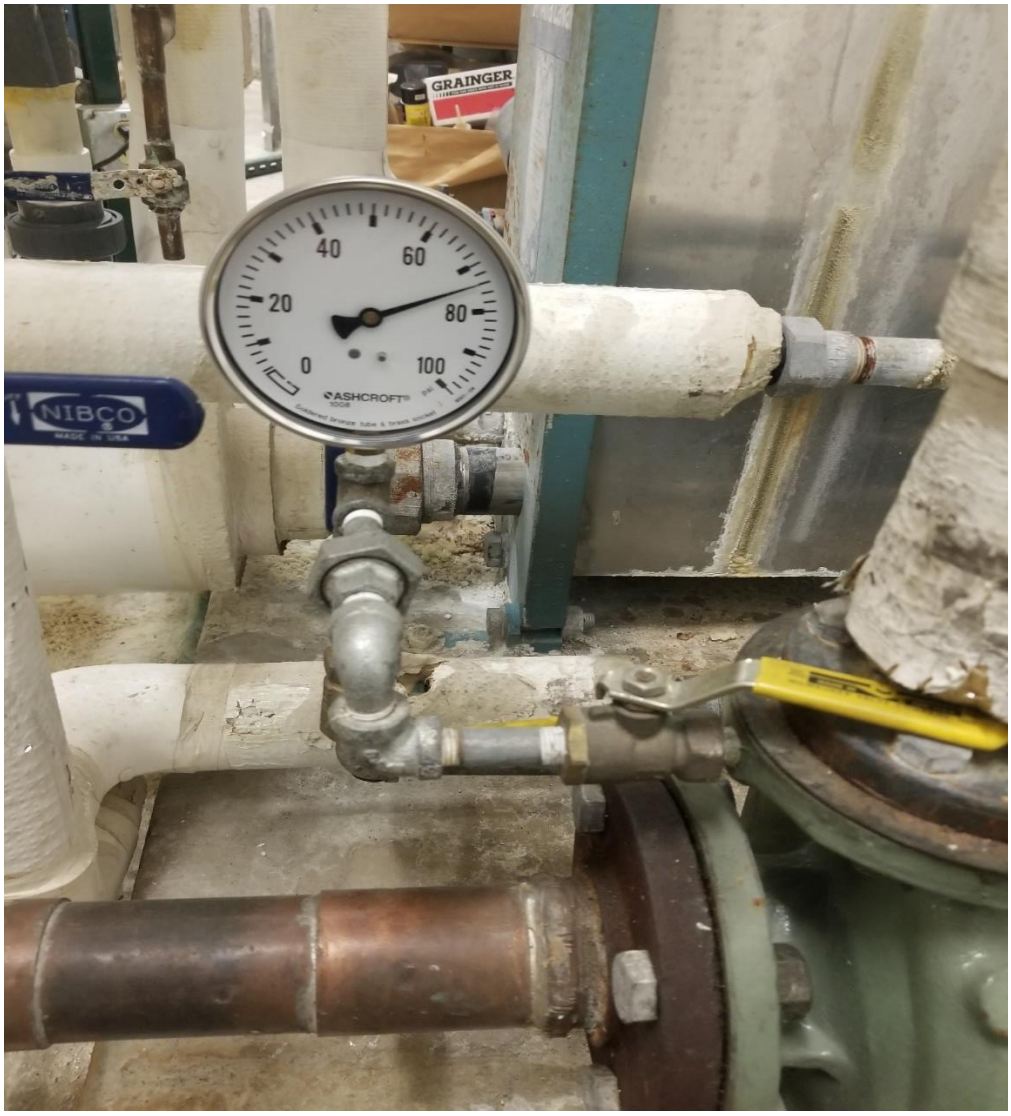


Figure 17. Chilled Water Return Pressure

Since 1PSI = 2.31 feet of head [19], hence 38 PSI change in pressure = 87.78 feet of head.

Pump Impeller Diameter Verification

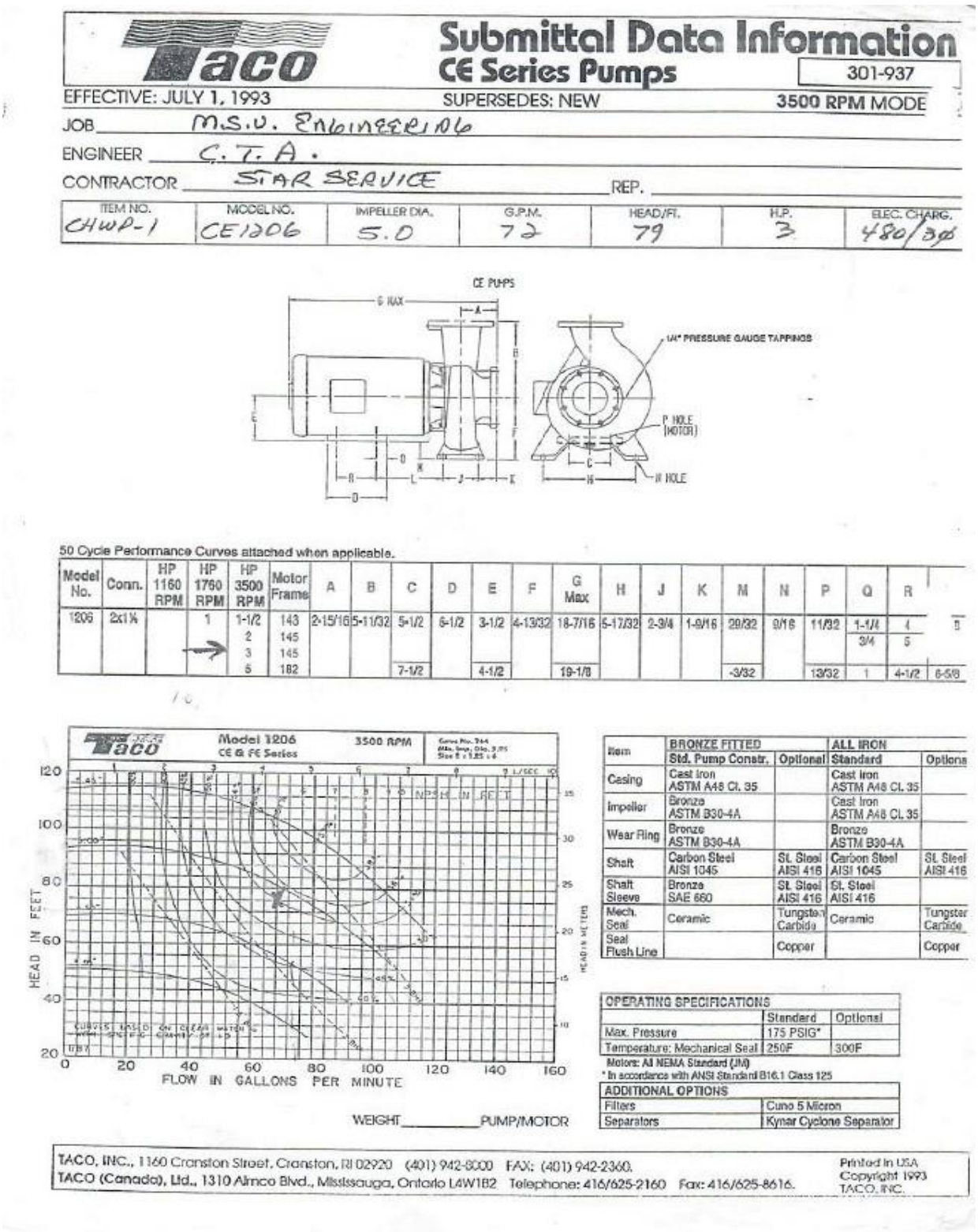
The dead-head pressure was also gotten to verify the impeller diameter of the pump, this was also read from the pressure gauge by closing the respective discharge valve. The dead-head condition was done for few minutes to prevent excessive heating which can damage expensive seals and reduce the life of the pump.

Outlet dead-head pressure = 95 PSI, Inlet dead-head pressure =43 PSI

Change in dead-head pressure(ΔP) = 95 PSI – 43 PSI =52 PSI

Since 1PSI = 2.31 feet of head, for 52 PSI change in pressure = 120.12 ≈ 120feet of head

The pump's model number was initially verified with the model number below to confirm if the data in figure 18 applies to the current pump being used, which matched, verifying the pump. Afterwards, 120.12 feet of head as obtained previously was checked in the pump curve as shown in figure 20, to verify the current impeller diameter, which was determined to be 5.45" and not 5" as stated in the data below. Hence, it was concluded that the pump impeller was changed.



Taco

Submittal Data Information

CE Series Pumps

301-937

EFFECTIVE: JULY 1, 1993 SUPERSEDES: NEW 3500 RPM MODE

JOB M.S.U. ENGINEERING

ENGINEER C.T.A.

CONTRACTOR STAR SERVICE REP.

ITEM NO.	MODEL NO.	IMPELLER DIA.	G.P.M.	HEAD/FT.	H.P.	ELEC. CHARG.
CHWP-1	CE1206	5.0	72	79	3	480/3ø

CE PUMPS

50 Cycle Performance Curves attached when applicable.

Model No.	Conn.	HP 1160 RPM	HP 1760 RPM	HP 3500 RPM	Motor Frame	A	B	C	D	E	F	G Max	H	J	K	M	N	P	Q	R	
1206	2x1¼		1	1-1/2	143	2-15/16	5-11/32	5-1/2	6-1/2	3-1/2	4-13/32	18-7/16	5-17/32	2-3/4	1-9/16	29/32	9/16	11/32	1-1/4	4	5
			→	2	145														3/4	5	
				3	145																
				5	182			7-1/2		4-1/2		19-1/8				-3/32		13/32	1	4-1/2	6-5/8

Item	BRONZE FITTED		ALL IRON	
	Std. Pump Constr.	Optional	Standard	Options
Casing	Cast Iron ASTM A48 Cl. 35		Cast Iron ASTM A48 CL 35	
Impeller	Bronze ASTM B30-4A		Cast Iron ASTM A48 CL 35	
Wear Ring	Bronze ASTM B30-4A		Bronze ASTM B30-4A	
Shaft	Carbon Steel AISI 1045	St. Steel AISI 416	Carbon Steel AISI 1045	St. Steel AISI 416
Shaft Sleeve	Bronze SAE 660	St. Steel AISI 416	St. Steel AISI 416	
Mech. Seal	Ceramic	Tungsten Carbide	Ceramic	Tungsten Carbide
Seal Flush Line		Copper		Copper

OPERATING SPECIFICATIONS		
	Standard	Optional
Max. Pressure	175 PSIG*	
Temperature: Mechanical Seal	250F	300F

Motors: All NEMA Standard (JM)

* In accordance with ANSI Standard B16.1 Class 125

ADDITIONAL OPTIONS	
Filters	Cuno 5 Micron
Separators	Kynar Cyclone Separator

WEIGHT_____PUMP/MOTOR

TACO, INC., 1160 Cranston Street, Cranston, RI 02920 (401) 942-8000 FAX: (401) 942-2360.
TACO (Canada), Ltd., 1310 Airnco Blvd., Mississauga, Ontario L4W1B2 Telephone: 416/625-2160 Fax: 416/625-8616.

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Figure 18. Taco Pump Data Information

Variable Speed Drive Application

The variable speed drive frequency was read from the VLT HVAC Drive as shown in figure19 below to be 54.1Hz (90.1%), this was also verified from figure 15 which showed an historical trend of 90%



Figure 19. Variable Speed Drive

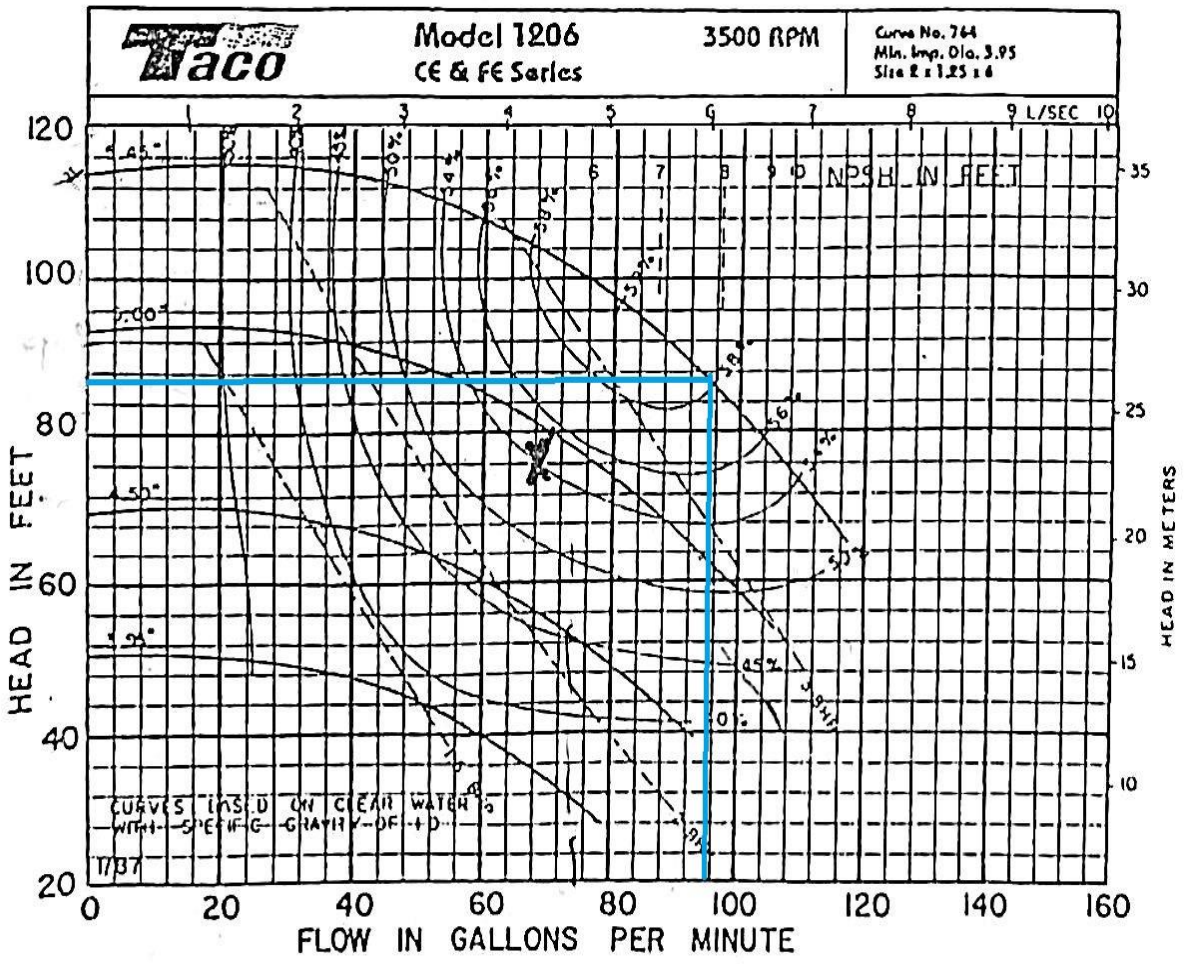


Figure 20. Taco Pump Curve

From the pump curve above, for 87.78 feet of head, we have 95GPM operating at 60Hz. In order to find the current flow rate at a frequency of 54.1 Hz (from the Variable Frequency Drive as seen in figure 19), we make use of the Affinity law. The Affinity law is stated as;

$$\frac{Q1}{Q2} = \frac{n1}{n2} \text{ and } \frac{n1}{n2} = \frac{F1}{F2} \text{ [3], thus } \frac{Q1}{Q2} = \frac{F1}{F2} \text{ ----equation(ii)}$$

Where Q = Flow rate (GPM), n = rotational speed and F= frequency.

Q1=95GPM, Q2 = ? , F1= 60HZ, F2 = 54.1Hz

Applying equation(ii) above to get Q2, we have Q2= 85.66GPM

From equation (i), Q(BTU/HR) = 85.66GPM x 500 x 3°F = 128,490BTU/HR ≈ 128,000 BTU/HR

Finding Qtotal for a year, since we have 8760hours in a year,

Qtotal = 128,490 x 8760 BTU = 1,125,572,400BTU

Value ≈ ($10/MMBTU) × (1,125,572,400BTU) = $11,255.72 ≈ $11,300

The next step with regards to this system is connecting this system to the heat pump loop which is performed in the next chapter of this thesis.

(B) Building Exhaust Air System

Basically, building exhaust air are designed to remove various pollutants, water vapor or heat and also to replace the air in buildings with fresh air. [20] The feasibility of using a heat exchanger to extract energy from the exhaust system of a building (Barnard Hall) was initially analyzed. The heat transfer calculator and the checal calculator were initially considered in sizing the heat exchanger to be used in recovering such energy for the building exhaust air stream. These calculators were proven not useful for this purpose due to limited inputs availability.

The Trane Official Product Selection Software (TOPSS) was used in selecting a coil for the exhaust stream due to many inputs being available. The input parameters used to generate the results are as follows:

(i) Airflow rate: 20,000 CFM

(ii) Number of Rows: 8

(iii) Area: 40.73 Square feet (This was obtained by selecting a Finned length of 115" and a Normal Coil height of 51")

(iv) Coil Utilization: Shipping Coil (Recommended by the Trane representative)

(v) Maximum face velocity: 800ft/min [21]

(vi) Normal fin spacing: 168 fins per foot (this value provided the maximum energy available)

(vii) Entering dry bulb air temperature: 70°F (Room Temperature)

(viii) Entering wet bulb temperature: 50°F [22]

The entering water flow rate was first kept constant at 55GPM with the entering water temperature being varied from 71°F to 90°F to see the correlation between the heat transfer as a function of the coil entering water temperature. Afterwards, the flow rates were varied, ranging from 35GPM to 200GPM as shown in the table below and then plotted using MATLAB. This was done to easily see the correlation between the entering water temperature and the flow rate with respect to the total capacity.

Table 1. Entering water temperature as a function of the flow rate and total capacity

Column	Flow rate =35gpm	Flow rate =45gpm	Flow rate = 55gpm	Flow rate = 65gpm	Flow rate = 75gpm	Flow rate = 85gpm	Flow rate = 95gpm	Flow rate = 100gpm
Entering Water Temp(°F)	Total Capacity (MBh)	Total Capacity (MBh)	Total Capacity (MBh)	Total Capacity (MBh)	Total Capacity (MBh)	Total Capacity (MBh)	Total Capacity (MBh)	Total Capacity (MBh)
71	13	14.9	16.18	17.05	17.68	18.15	18.51	18.67
72	26.04	29.83	32.38	34.12	35.37	36.31	37.04	37.35
73	39.11	44.79	48.60	51.20	53.08	54.49	55.59	56.04
74	52.20	59.78	64.84	68.30	70.81	72.69	74.14	74.75
75	65.33	74.78	81.10	85.42	88.55	90.90	92.72	93.47
76	78.49	89.82	97.38	102.56	106.31	109.13	111.30	112.21
77	91.67	104.88	113.68	119.72	124.09	127.37	129.90	130.96
78	104.88	119.96	130.00	136.89	141.88	145.63	148.52	149.72
79	118.12	135.06	146.34	154.07	159.69	163.90	167.14	168.50
80	131.38	150.18	162.70	171.28	177.51	182.18	185.78	187.28
81	144.67	165.33	179.06	188.50	195.34	200.48	204.44	206.08
82	157.98	180.49	195.45	205.73	213.20	218.79	223.10	224.90
83	171.31	195.68	211.85	222.98	231.06	237.12	241.78	243.72
84	184.67	210.88	228.27	240.25	248.94	255.46	260.47	262.56
85	198.05	226.11	244.73	257.53	266.84	273.81	279.17	281.40
86	211.46	241.35	261.17	274.82	284.74	292.18	297.89	300.26
87	224.89	256.61	277.63	292.13	302.67	310.55	316.61	319.13
88	238.33	271.88	294.11	309.45	320.60	328.95	335.35	338.02
89	251.79	287.18	310.60	326.80	338.55	347.35	354.10	356.91
90	265.28	302.49	327.10	344.14	356.51	365.76	372.86	375.81

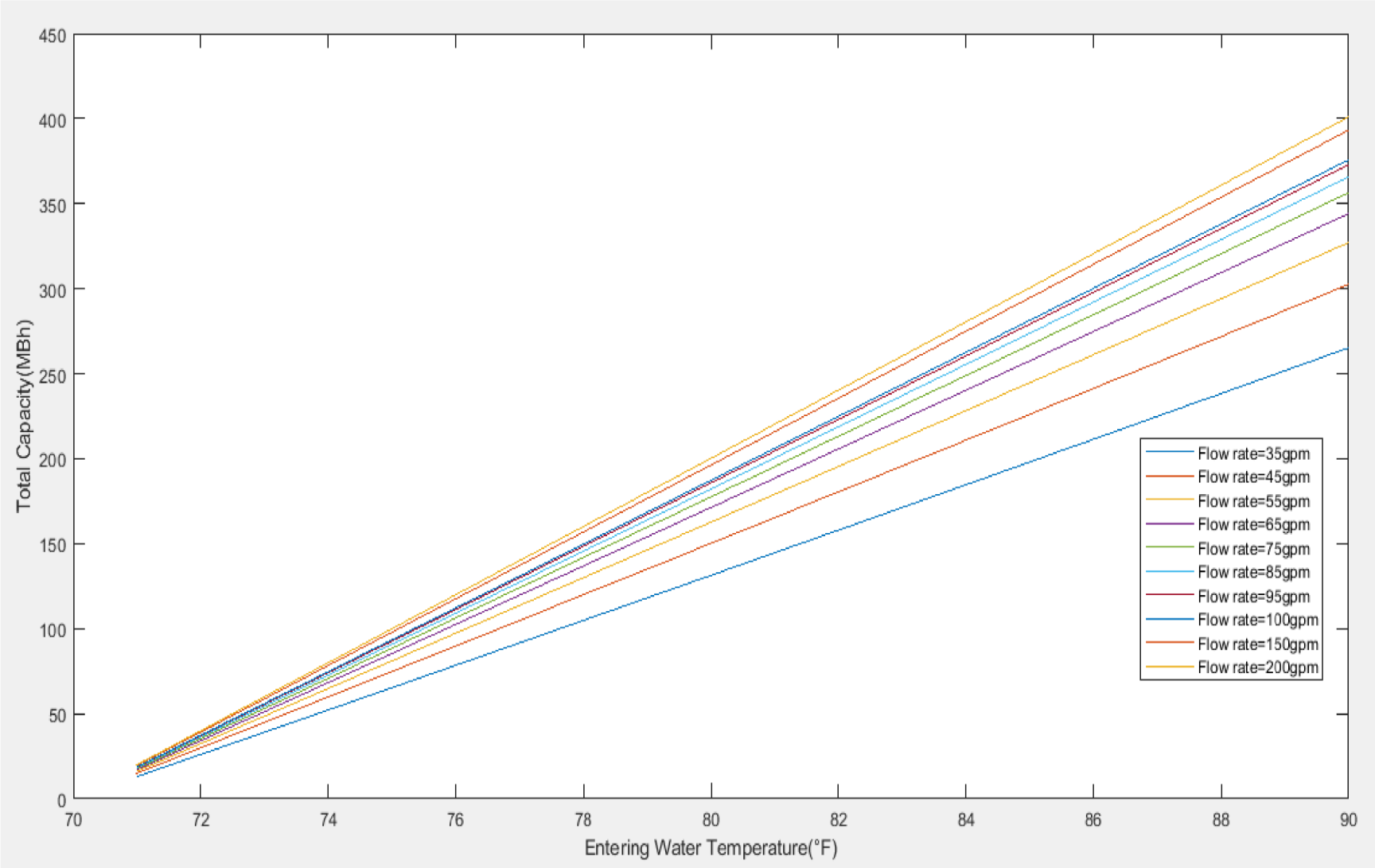


Figure 21. Entering water temperature as a function of the flow rate and total capacity

From the Graph, it was discovered that the entering water temperature varies linearly with the total capacity of the coil in which it increases as the entering water temperature increases. Also, the capacity also increases as the flow rate increases, and as the flow rate approaches 200GPM, less change in capacity is observed as seen in figure 21.

- A heat pump was then selected to be used as a replacement for the coil due to the low capacity being generated by the coil.

Hence, with respect to the heat pump, for the air flow side of the heat pump;

Qair = CFM x 0.92 x Temperature Difference

(This calculation does not depend on the coil selection above)

where CFM = Building Exhaust Air Flowrate = 20,000CFM (Design Air Flowrate)

Inlet Temperature = 70°F (Room Temperature)

Outlet Temperature = 55°F (Maximum Set point Cooling Temperature)

Hence, Qair = 20,000 x 0.92 x (70 – 55) = 20,000 x 0.92 x 15 = 276,000Btu/hr.

Heat Pump Data Analysis

Data from the McQuay Enfinity Water Source and Water to Water Heat Pumps were used in analyzing heat pump performance and trends in general. This will help guide the decisions in determining the parameters for the heat pump to be used. The various figures shown below were plotted using MATLAB.

(i) Water source heat pump - cooling mode(air)

For the heat pump in cooling mode (the water being heated, and the air being cooled), entering water temperatures from 60°F – 90°F were selected due to the heat pump loop being designed to operate within the temperature range of 60°F – 90°F. Also, the exhaust air temperature of 75°Fdb/63°Fwb was selected and kept constant due to the proximity to room temperature.

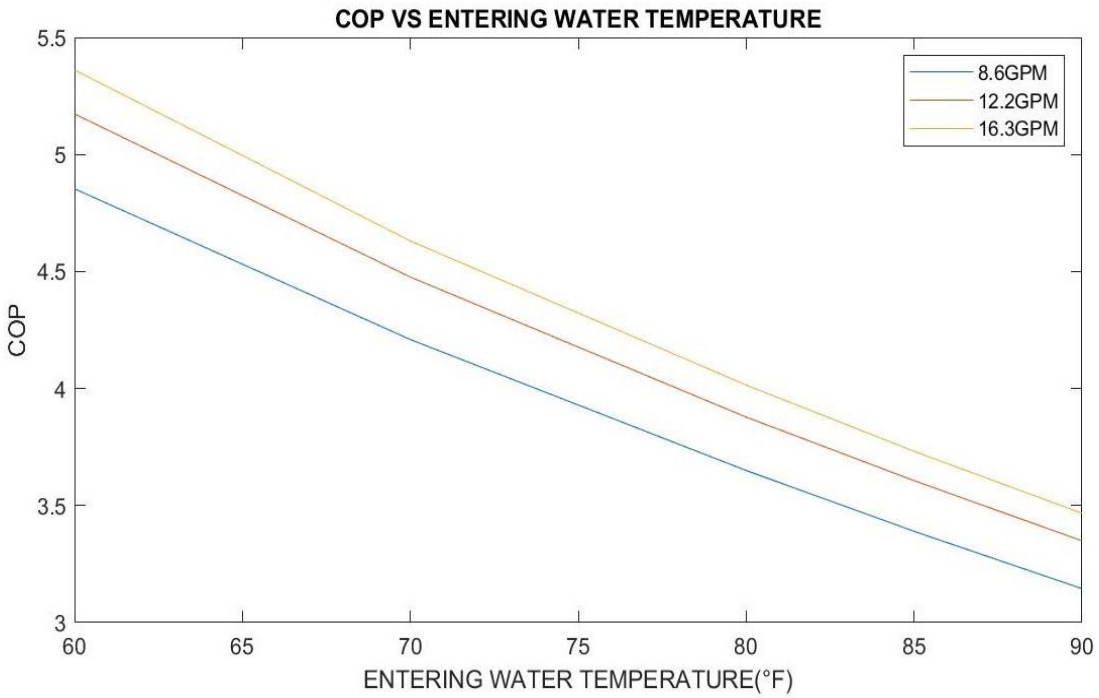


Figure 22. COP against Entering Water Temperature

As seen above for the water source heat pump in the cooling mode operation, the COP decreases as the entering water temperature increases when the heat pump is operating in the cooling mode. This occurs because of the heat pump being able to generate less cooling, requiring more energy to generate the quantity of cooling it would at a colder entering water temperature. Thus, more energy is required to drive the process in the cooling mode as the entering water temperature (EWT) is being increased. Hence, the heat pump performs more work which could be seen in figure 23 below, in which the cooling capacity of the heat pump decreases with the EWT.

This is typically explained by the decrease in COP as the refrigerant temperature/pressure difference (lift) increases between the hot and cold sides.

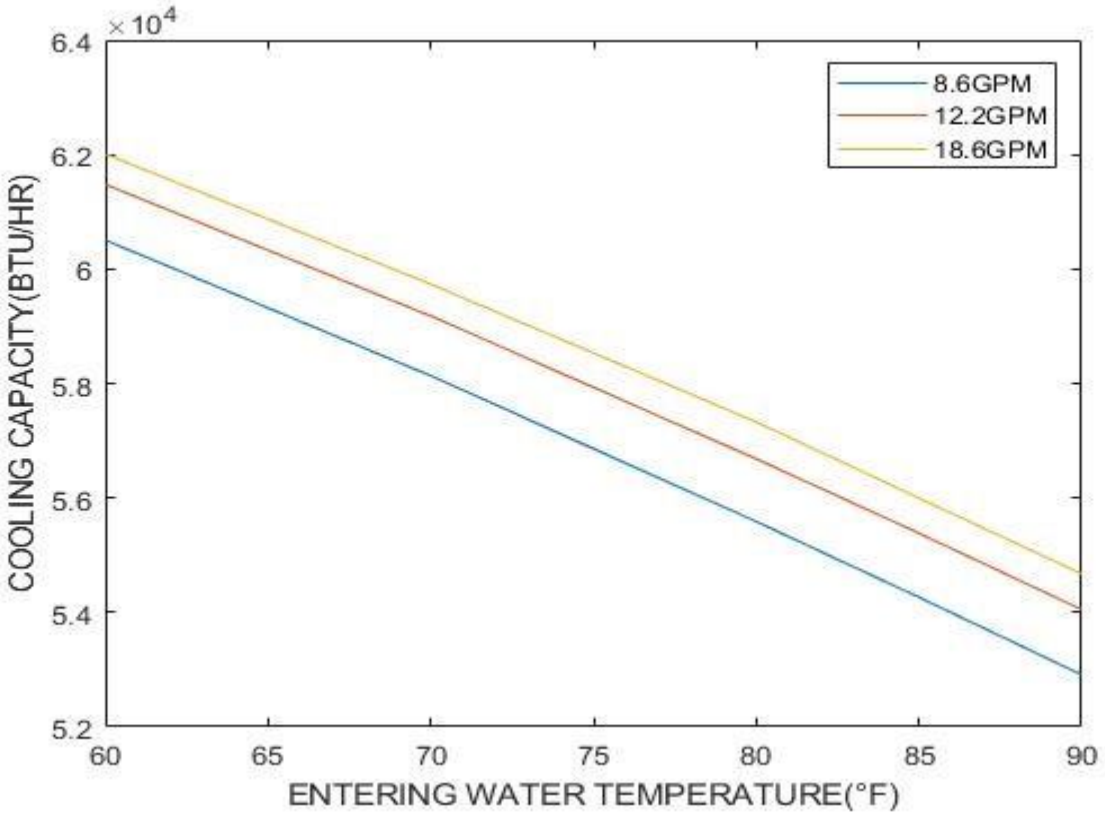


Figure 23. Cooling Capacity against Entering Water Temperature

Also, as we increase the flow rate (from 8.6GPM -16.3GPM) as shown in both figures above, we tend to improve the COP of the heat pump.

(ii) Water source heat pump – heating mode(air)

For the heat pump in heating mode (the water being cooled, and the air being heated), entering water temperature ranges from 60°F – 90°F were selected since the heat pump loop is designed to operate within the temperature range of 60°F – 90°F. Also, the exhaust air temperature of 70°F was selected due to the proximity of room temperature from the McQuay data available.

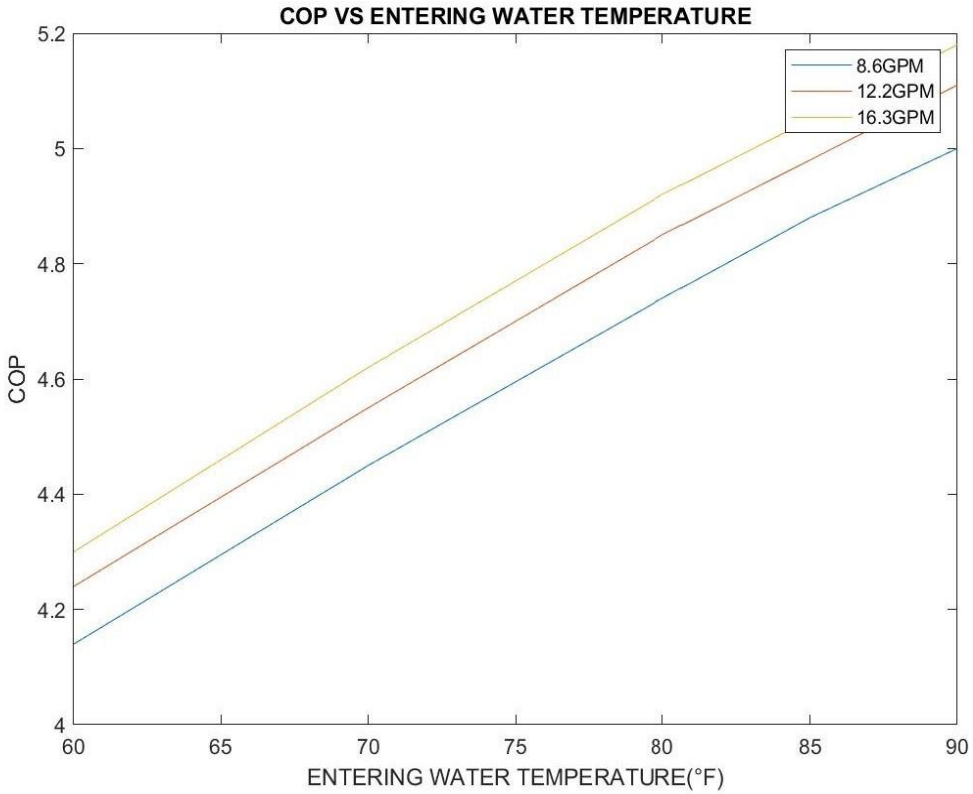


Figure 24. COP against Entering Water Temperature

As seen in figure 24, the COP increases as the EWT increases when the heat pump functions in the heating mode operation, this occurs as a result of the heat pump being able to generate more heat requiring less energy to generate the quantity of heat it would at a colder entering water temperature. Thus, less energy is required to drive the process in the heating mode, as the entering water temperature is being increased. Hence, the heat pump performs less work, this could be seen in the figure below in which the heating capacity of the heat pump increases with the EWT as shown below. This is typically explained by the increase in COP as the refrigerant temperature or pressure difference (lift) decreases between the hot and cold sides.

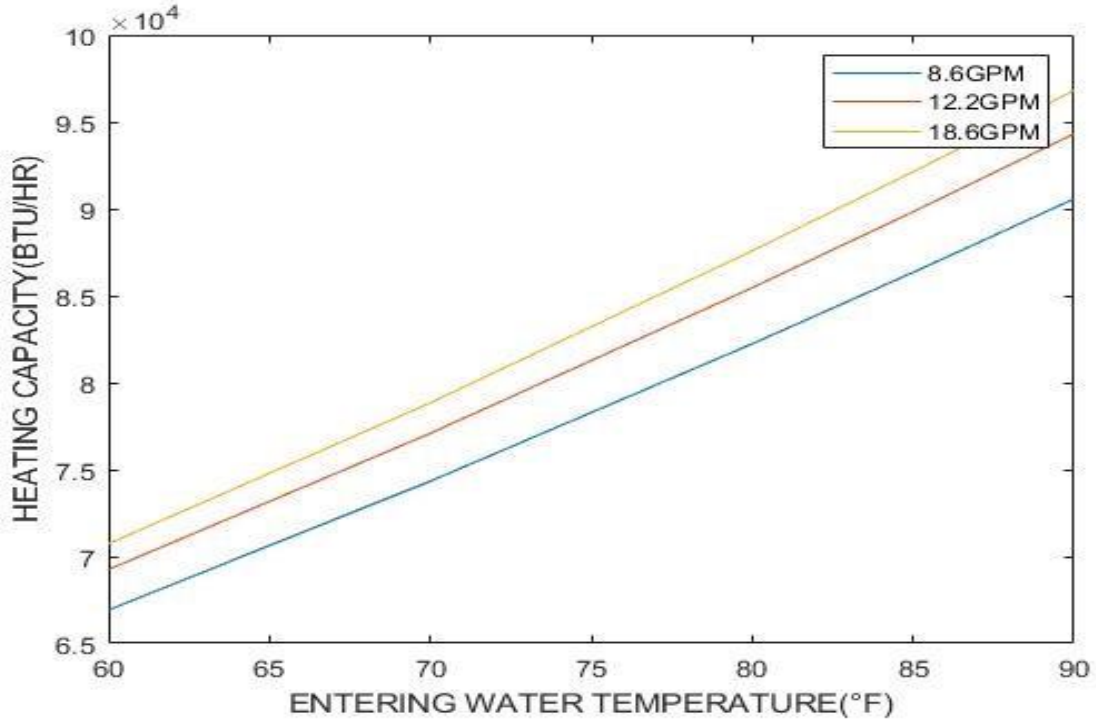


Figure 25. Heating Capacity against Entering Water Temperature

Also, as we increase the flow rate as seen in figure 25, the COP improves alongside.

(iii) Water to Water heat pump

For the water to water heat pump, we consider the correlation between the COP and the source entering water temperature as well as the load entering water temperature. The data was analyzed for the heating mode condition in which the water loop serving the terminal device or load is being heated. This correlation plays an important role in order to efficiently model the heat pump loop. The load is the water loop serving the air handler or terminal device while the source is the heat rejection water loop. The data was plotted using MATLAB for the entering source water temperature at 50°F, 70°F and 85°F, even though the heat pump loop will be designed to operate within the range of 60°F -90°F. This was done to get a better picture of the correlation. The COP and the source entering water temperature were compared at the operating ranges of 50°F, 70°F and 85°F as shown below from the McQuay Data made available. Also, the COP and the source entering water temperatures were compared for various flow rate conditions for the source gallons per minute (SGPM) and load gallon per minute (LGPM), i.e. either of them varied and the other made constant to see the correlation with the COP. e.g. at 11.2SGPM/11.2LGPM, 11.2SGPM/13.5LGPM, 13.5SGPM/ 11.2LGPM and 13.5SGPM/ 13.5LGPM.

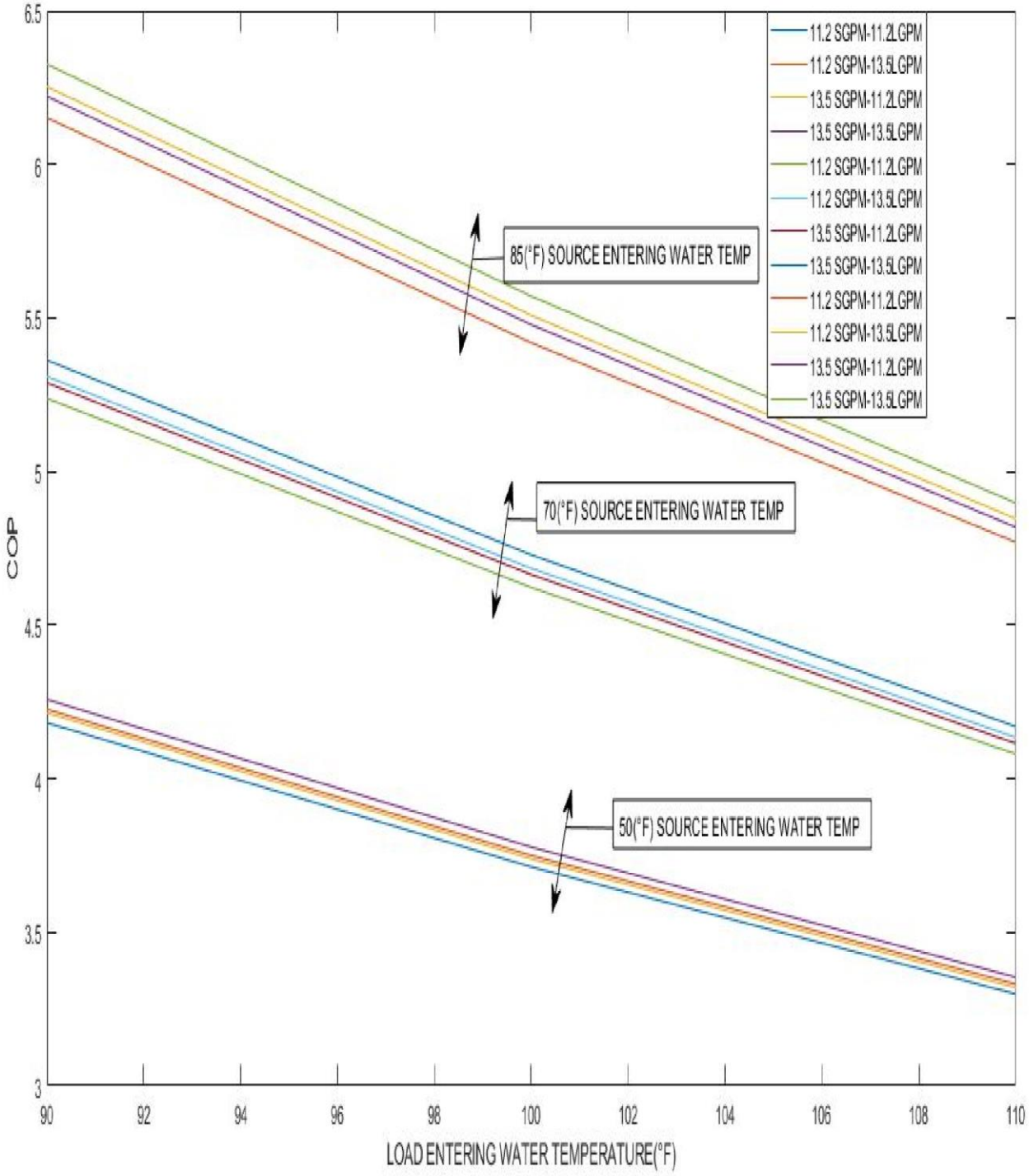


Figure 26. COP variation with the source and entering water temperature

From the figures above, the COP in general decreases as the load entering water temperature increases. Also, it is also seen as stated earlier that the COP increases as the source entering water temperature increases. Hence, to improve the COP for a water to water heat pump, a high source gallon per minute and a high load gallon per minute should be used as seen in figure 26 i.e. from the figure above, we tend to have the highest COP when the source gallon per minute and the load gallon per minute were both high (13.5GPM – 13.5GPM).

(C) The Outdoor Air

The outdoor air serving as a heat sink in the heat pump loop was first analyzed using the Bin method to get a broader picture of the temperature distribution and how the entering water temperature of the coil and the flow rate impacts the heat transfer. The Bin method refers to a process of discretizing hourly weather into various groups or bins of weather conditions.[23]

Previous bin/weather data for Bozeman was obtained from the MSU facility services as seen in Figure 30. Figure 30 incorporates an heat exchanger calculator for the UA (overall heat transfer coefficient, U, multiplied by the heat transfer area, A) of the heat exchanger as well as the heat capacity ratio based on the fixed UA value. The heat transfer, leaving air temperature and the leaving water temperature are then generated as an output for each bin based on the entering water temperature. Different entering water temperature can be imputed to see the impact being made on the total heat transfer.

The formulas used in the heat exchanger calculator are shown below;

$Q = UA\,\Delta t_m$ …….(iii)

where;

Q = Rate of heat transferred

U = Overall coefficient of heat transfer from fluid to fluid

A= Heat transfer area of the heat exchanger associated with U

Δt_m = Log mean temperature difference (LMTD)

For a heat exchanger with a constant U, the Δt_m is calculated as:

$$\Delta t_m = C_f \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln(T_1 - t_2)/(T_2 - t_1)}$$ ……..(iv)

where the correction factor was assumed to be 1 [24]

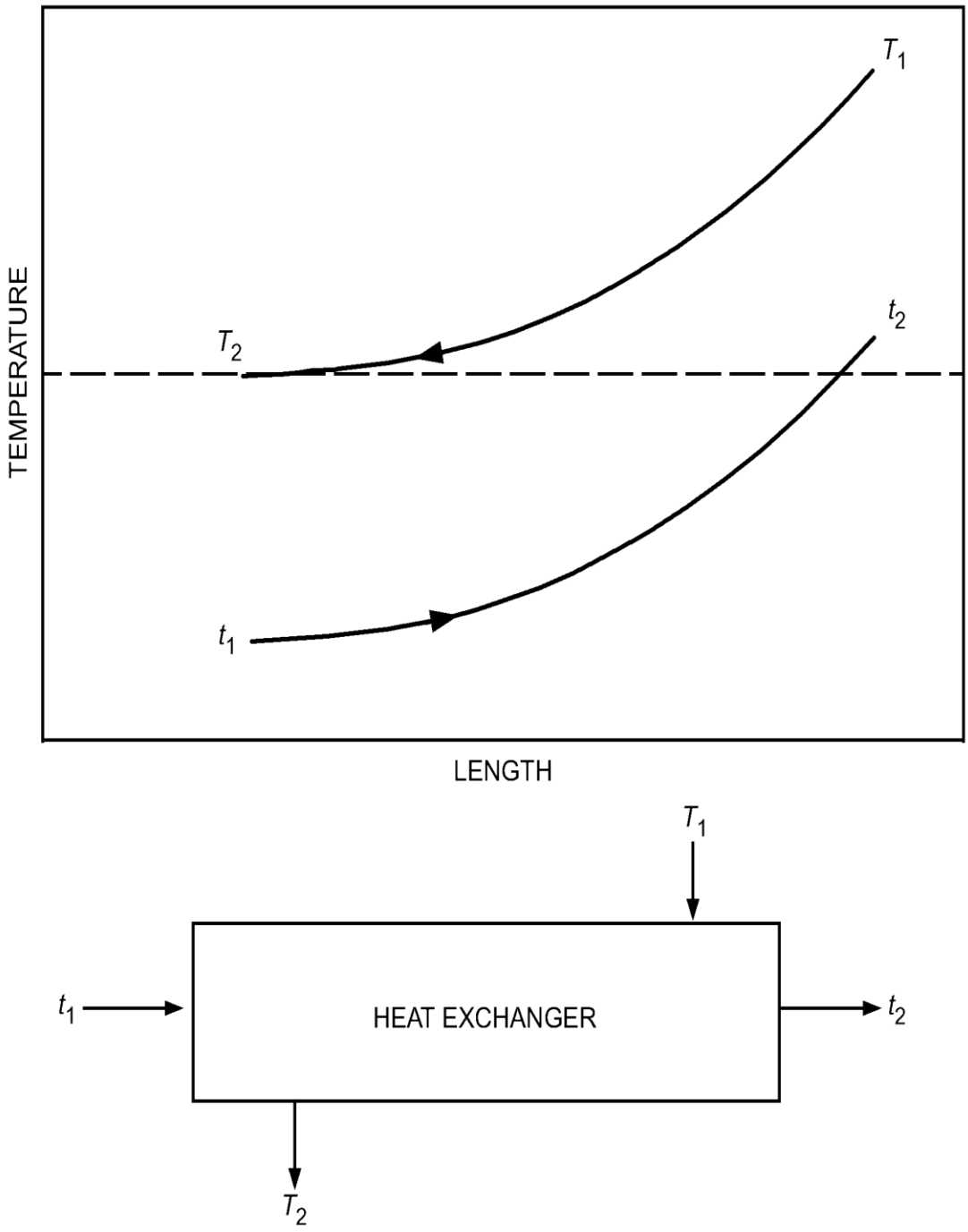


Figure 27. Temperature Distribution in Counterflow Heat Exchanger [24]

An adiabatic counterflow exchanger with inlet and outlet temperatures and capacitance rates of the hot and cold fluids is shown in figure 28 below. The overall heat transfer coefficient area product is *UA* with maximum possible temperature drop of the hot fluid occurring from T_{hi} to T_{ci}. The heat being transferred is thus given as;

$$Q_{\max} = (\dot{m}C_p)_h(T_{hi} - T_{ci}) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\text{(v)}$$

The maximum possible temperature rise of the cold fluid would be from T_{ci} to T_{hi}. The corresponding maximum heat transfer or exchange is given as;

$$Q_{\max} = (\dot{m}C_p)_c(T_{hi} - T_{ci}) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\text{(vi)}$$

The maximum heat transfer that could occur in the exchanger is thus deteremined by the minimum of the two capacitance rates, $(m\dot{C}_p)_{\min}$,

$$Q_{\max} = (\dot{m}C_p)_{min}(T_{hi} - T_{ci}) \ldots\ldots\ldots\ldots\ldots\ldots\text{(vii)}$$

The actual heat exchange Q is given by

$$Q_{\max} = (\dot{m}C_p)_c(T_{co} - T_{ci}) = (\dot{m}C_p)_h(T_{hi} - T_{ho}) \ldots\ldots\ldots\ldots\ldots\ldots\text{(viii)}$$

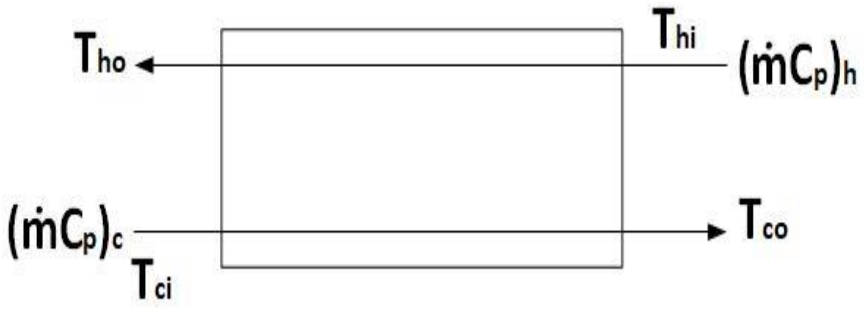


Figure 28. Schematic of an adiabatic counterflow heat exchanger showing temperatures and capacitance rates of the hot and cold fluids [25]

Effectiveness ε can be defined as the ratio of the actual heat transfer to the maximum heat transfer possible, the formula is stated below as;

$$\mathcal{E} = \frac{Q}{Q\max} = \frac{(\dot{m}C_p)_h(T_{hi} - T_{ho})}{(\dot{m}C_p)_{min}(T_{hi} - T_{ci})} = \frac{(\dot{m}C_p)_c(T_{co} - T_{ci})}{(\dot{m}C_p)_{min}(T_{hi} - T_{ci})} \quad \text{.....(ix)}$$

Since either the hot or cold fluid has the minimum capacitance rate, the effectiveness can always be expressed in terms of the temperatures only. The working equation for the heat exchanger is given as;

$Q = \varepsilon(\dot{m}C_p)_{min}(T_{hi} - T_{ci})$ ……..……(x)

For a counterflow exchanger, the effectiveness is given by

$$\mathcal{E} = \frac{1 - e^{-NTU(1-C^*)}}{1 - C^* e^{-NTU(1-C^*)}} \quad \text{if } C^* \neq 1 \quad \text{……...……(xi)}$$

$$\varepsilon = \frac{NTU}{1+NTU} \qquad \text{if } C^* = 1 \qquad \text{……..(xii)}$$

where NTU is the number of transfer units, defined as

$$\text{NTU} = \frac{UA}{(\dot{m}C_p)_{min}} \qquad \text{……(xiii)}$$

and the dimensionless capacitance rate is given by

$$C^* = \frac{(\dot{m}C_p)_{min}}{(\dot{m}C_p)_{max}} \quad [25] \qquad \text{…….….(xiv)}$$

For the Leaving Air Temperature and Leaving Water Temperature, this was gotten from the equation below by making the appropriate parameter the subject of the formula

Qair = 0.92 x CFM x (Entering Air Temperature – Leaving Air Temperature)

Qwater = 500 x GPM x (Leaving Water Temperature – Entering Water Temperature)

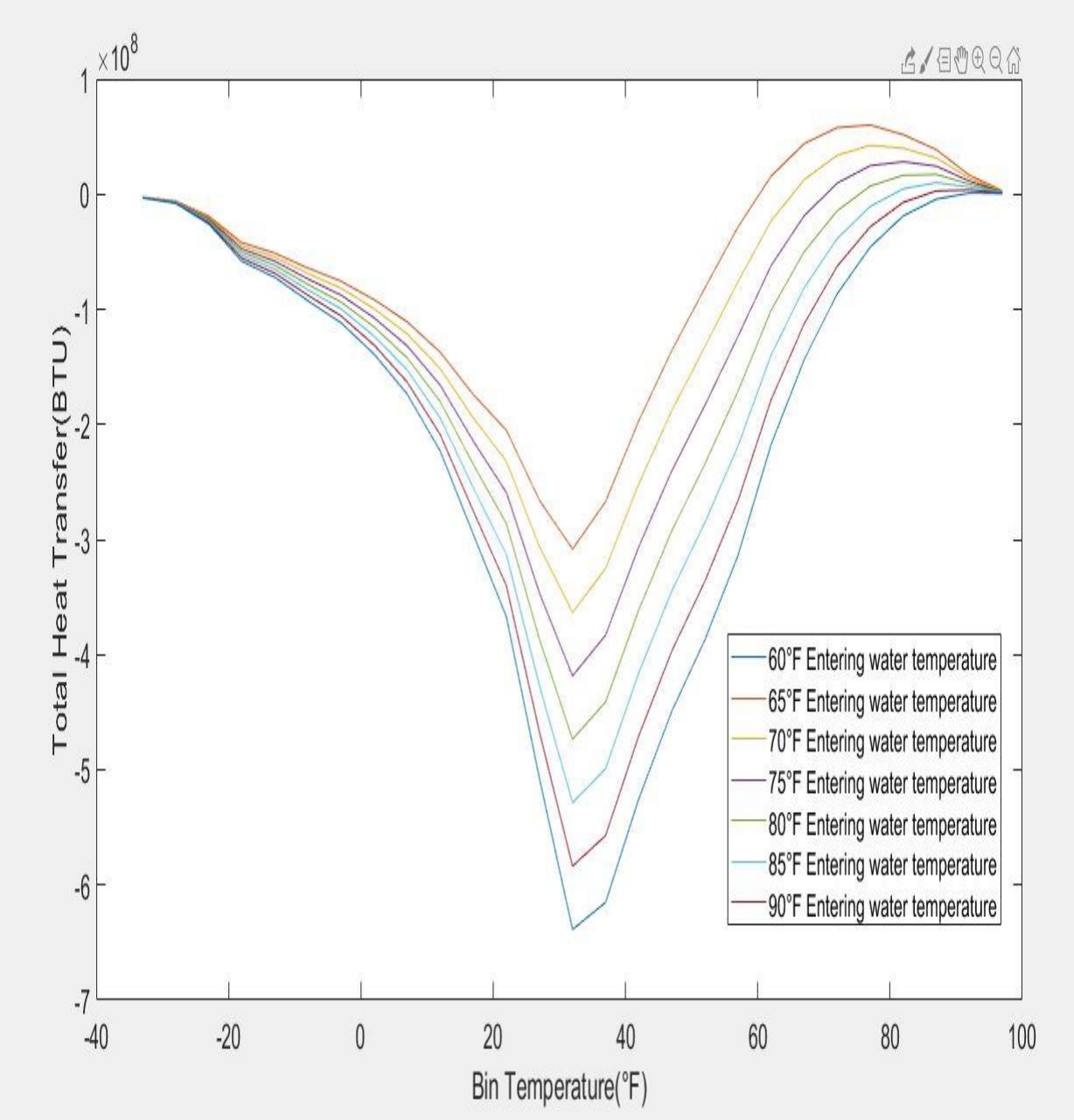


Figure 29. Graph showing how different entering water temperatures affect the heat transfer, with the flow rate kept constant (55GPM)

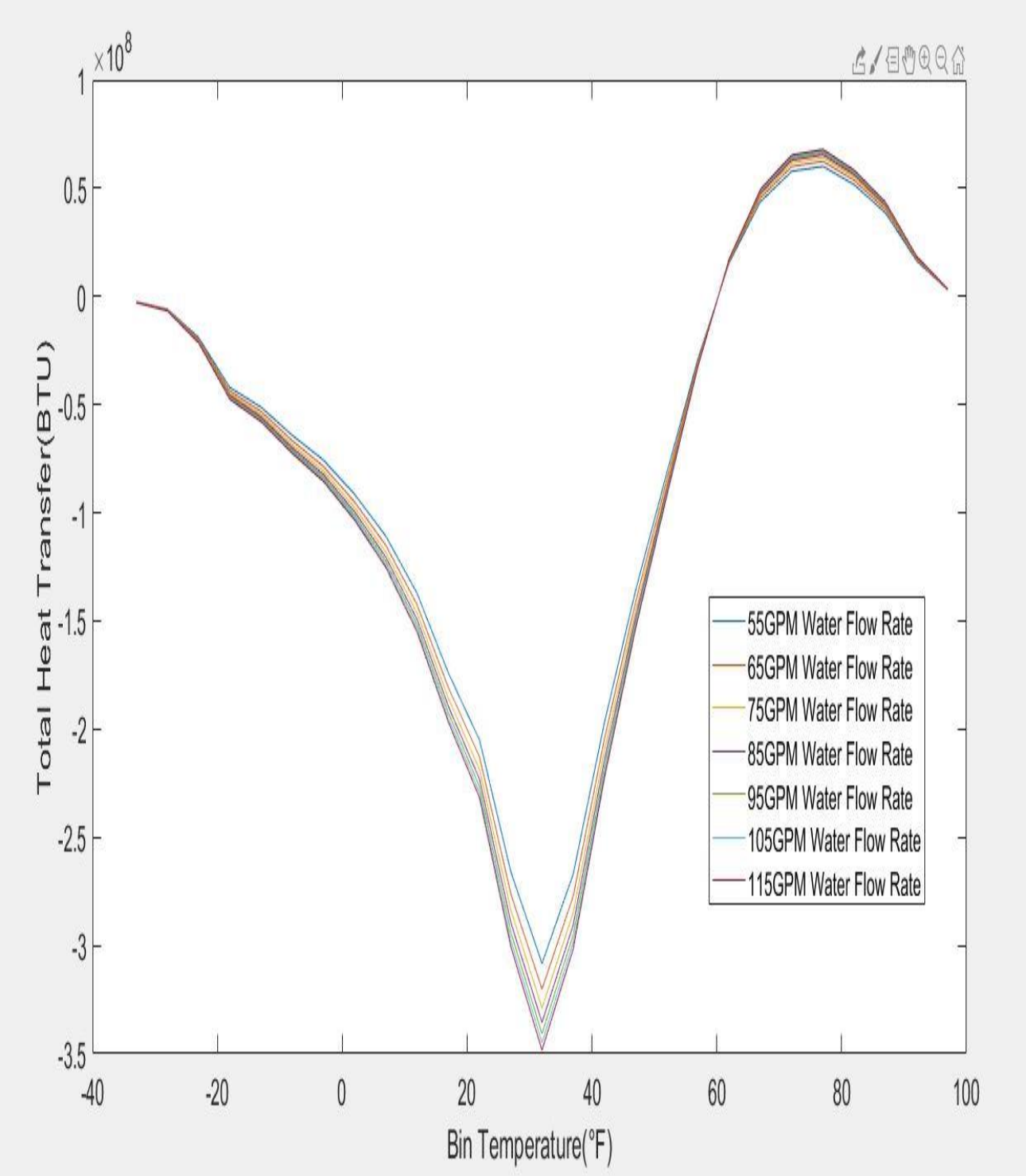


Figure 30. Graph showing how different water flow rates affect the heat transfer, with the entering water temperature kept constant (60°F)

From the analysis, the following observations were made:

- For the entering water temperature variation;
  - In the clustered column charts as seen in the appendix, the positive region of the graph represents the air being cooled while the negative region of the graph represents the air being heated at various bin temperatures (entering air temperatures).
  - The maximum heat transfer occurs at a bin/entering air temperature of 32°F because of the substantial annual occupied hours.
  - Also from the clustered column charts in the appendix, an increase in the entering water temperature leads to a decrease in the rate of heat transfer for the bin temperatures in which cooling takes place (i.e. at 65°F entering water temperature and bin temperature/entering air temperature of 92°F, we have an heat transfer rate of about 14 MMBtu, while at 90°F entering water temperature and bin temperature of 92°F**,** we have almost no heat being transferred).
  - Similarly, an increase in the entering water temperature leads to an increase in the rate of heat transfer for the bin temperatures in which heating takes place (e.g. at 65°F entering water temperature and bin temperature/entering air temperature of 32°F, we have an heat transfer rate of about 363 MMBtu, while at 90°F entering water temperature and bin temperature of 32°F, we have an heat transfer rate of about 639 MMBtu).

- For the flow rate variation, it is noticed from figure 30 that an increase in the flow rate leads to an increase in the heat transfer for both in the cooling and heating mode of operation

(D) Domestic hot water system

The domestic hot water system in Barnard Hall delivers hot water majorly to the various fixtures in the building. The system in Barnard Hall typically entails various hot water heated tanks installed at various part of the building to supply hot water when required.

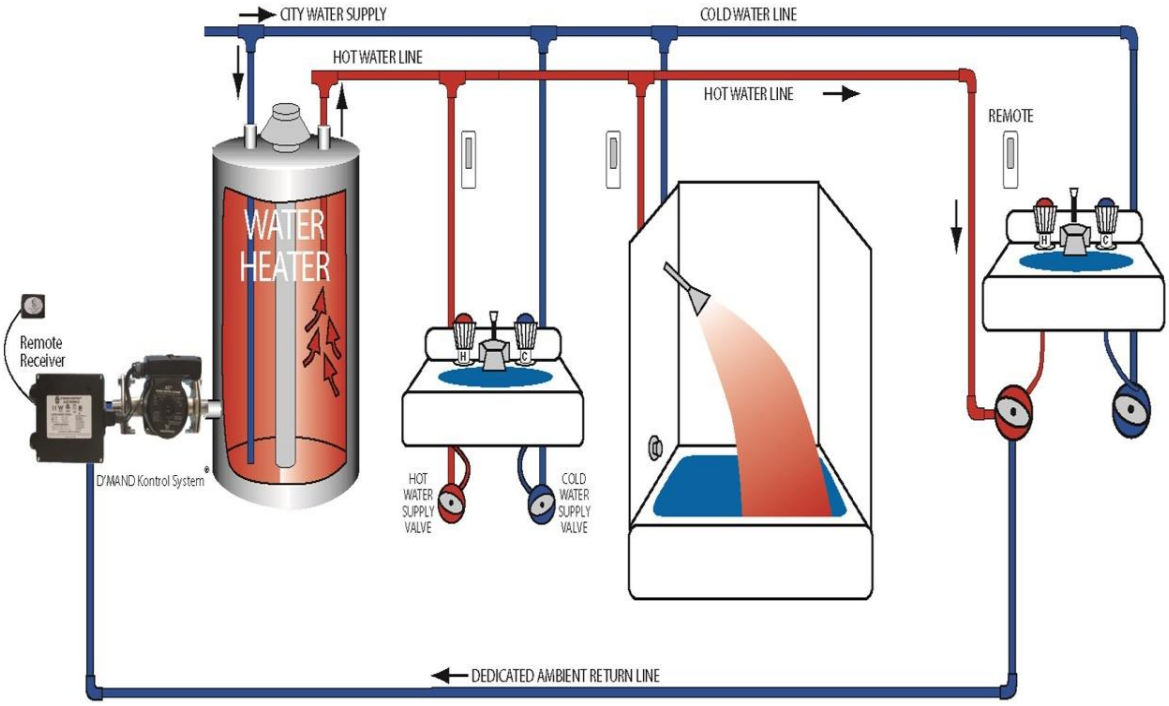


Figure 31. Domestic hot water system [36]

Hot Water Load Demand Calculation

Flow rate for basin public lavatory= 15 gallons of water per hour per fixture [26]

For 22 lavatories in Barnard Hall, total gallons of water per hour = 22 x 15 gallons of water per hour x demand factor, where demand factor = 0.4[26]

Therefore, total Gallons of water per hour = 132 gallons of water per hour or 2.2 gallons per minute

Hence, Flow rate = 2.2GPM

Inlet Cold Water Temperature = 40°F (Designed entering cold water temperature)

Outlet Hot Water Temperature= 140°F (Set Point Temperature)

Hence, Heating Capacity required = 500 x Flow rate x Temperature Difference

Adding 0.1 GPM to estimate the impact of pipe loss on flow rate,

New flow rate = 2.2GPM + 0.1GPM= 2.3GPM

Heating Capacity required = 500 x 2.3x (140 – 40) = 115,000Btu/hr

For Qpipingloss,

- $Q_{pipingloss} = UA_{flowing} L_{pipe} \Delta T_{lm}$ [26]----------(xv)

$$\Delta T_{lm} = \frac{[(T_{hot\ in} - T_{air}) - (T_{hot\ out} - T_{air})]}{\ln[(T_{hot\ in} - T_{air})/(T_{hot\ out} - T_{air})]} \quad [26]$$

$$T_{hot\ out} = T_{air} + (T_{hot\ in} - T_{air}) e^{-\frac{(UAflowing)(Lpipe)}{\dot{m} Cpwater}} \quad [26] \quad \text{----------(xvi)}$$

where;

ΔT_{lm} = log mean temperature difference (°F)

$Q_{pipingloss}$ = Piping heat loss rate (Btu/day)

$\dot{m}$ = water flow rate (lb_m/h)

c_p = specific heat of water ($Btu/lb_m°F$)

$T_{hot\ in}$ = water temperature entering the pipe (°F)

$T_{hot\ out}$ = water temperature leaving the pipe (°F)

$UA_{flowing}$= flowing heat loss factor per foot of pipe (Btu/h·ft·°F)

L_{pipe} = length of hot-water pipe (ft)

The water temperature leaving the pipe ($T_{hot\ out}$) was first determined from equation (iv) for each respective total length with respect to the domestic hot water nominal pipe size in Barnard Hall. The length of pipe for each domestic hot water piping was done by taking measurements as seen in the sample of the mechanical drawing drawn to scale below, thereafter scaling and converting from inches to feet. Temperature of Air (Tair) was estimated at 70°F and the hot water temperature entering the pipe was estimated at 140°F. The heat loss factor per foot of pipe was determined from referenced tables with respect to the respective pipe size as shown in table 2 below, the heat loss factor per foot of pipe for the pipe size that was not available in the referenced data was gotten by interpolation.

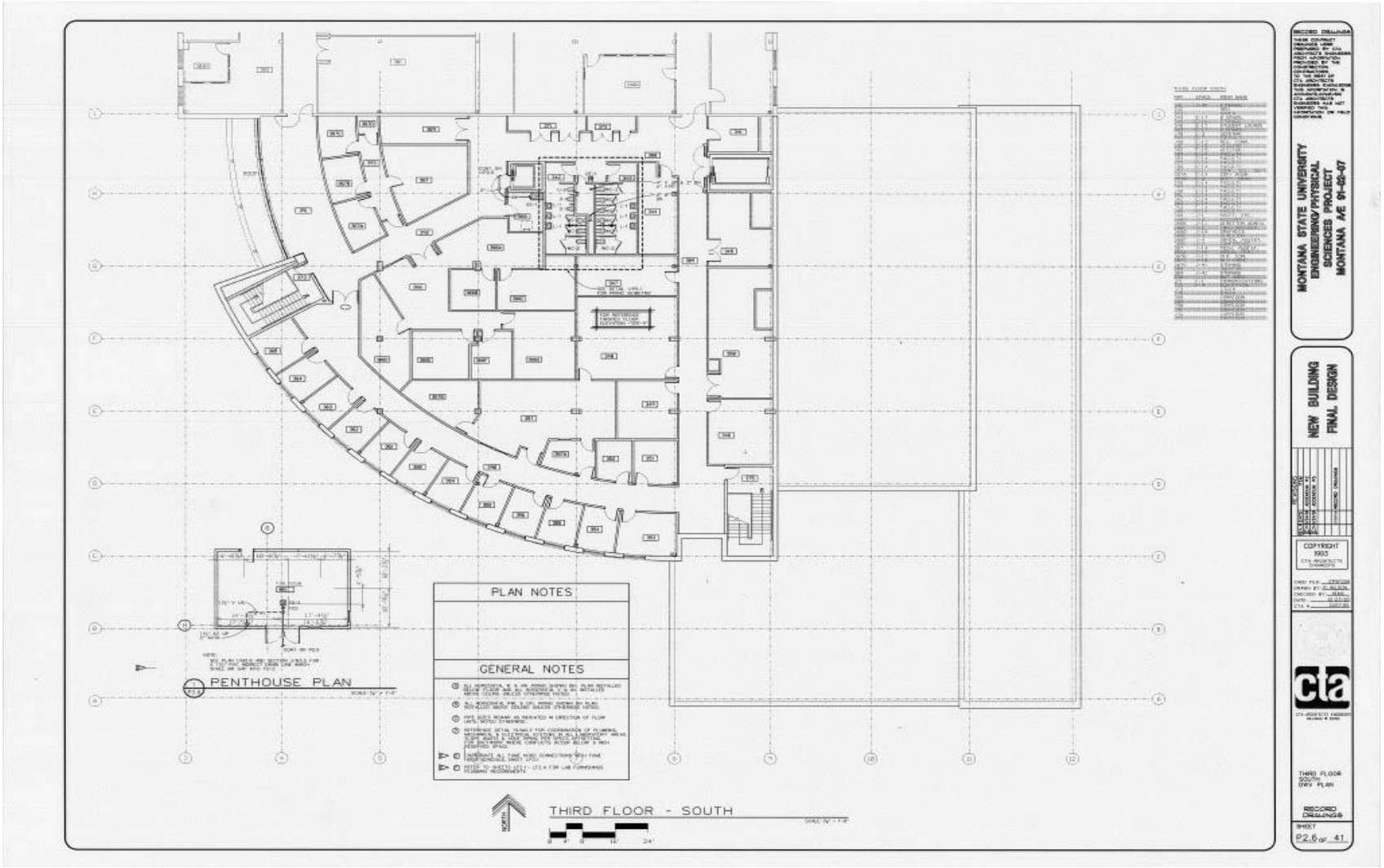


Figure 32. Barnard Hall Drawing Plan Sample provided by MSU Facility

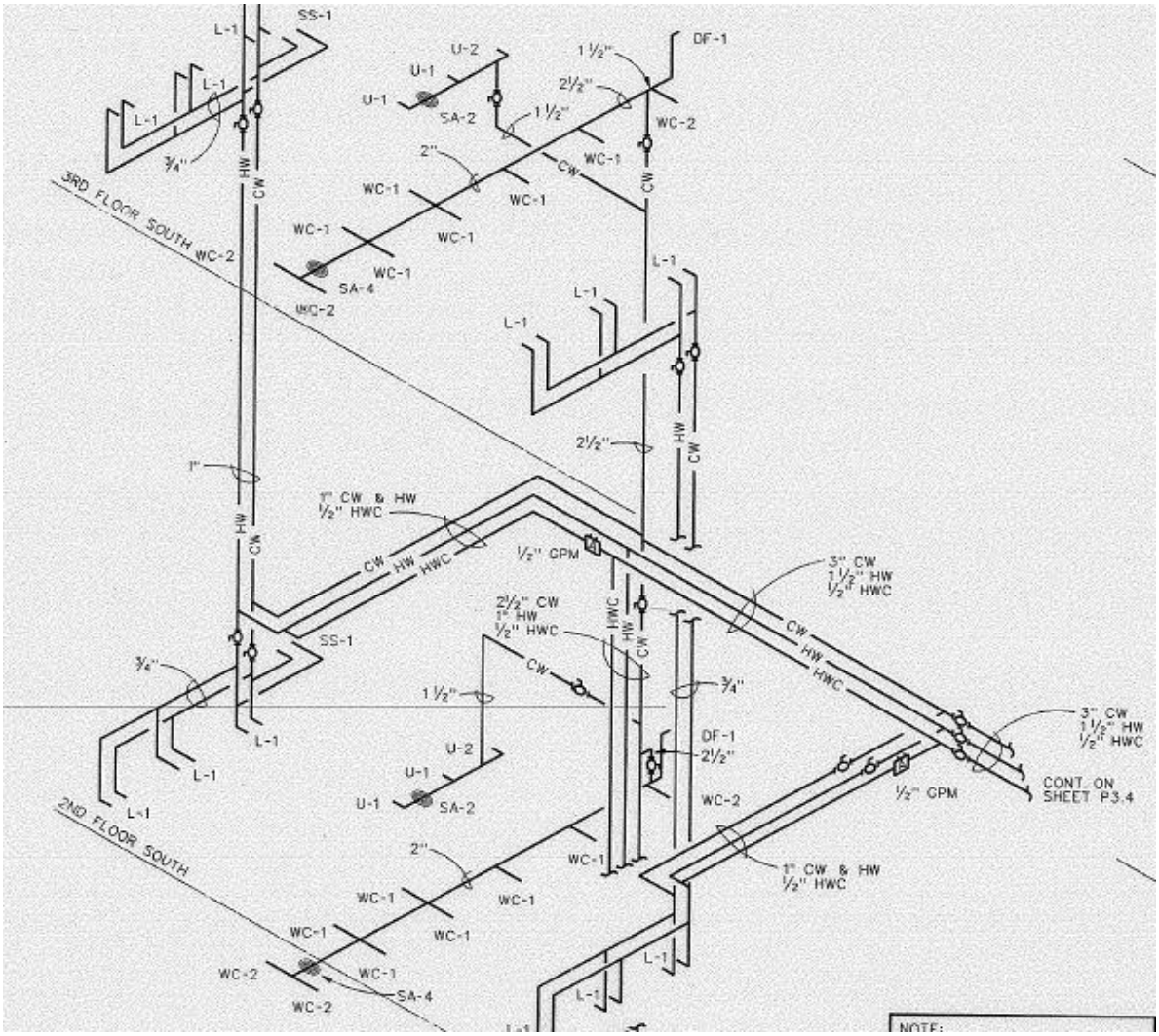


Figure 33. Barnard Hall Drawing Plan Sample provided by MSU Facility

Table 2. Heat Loss Factor Per Foot of Pipe

[27]

NOMINAL PIPE SIZE(INCHES)	**HEAT LOSS FACTOR PER FOOT OF PIPE(BTU/hr.ft°f)**
0.5"	0.1
0.75'	0.12
1"	0.13
1.5"	0.16
2.5"	0.2
3.0"	0.26

The specific heat of water (Cp) is given as 1 Btu/lb_m°F for the temperature range used in the heat pump loop while the mass flow rate is assumed to be 0.5gallons/minute as seen in some of the piping systems on the mechanical drawing. The various Hot water leaving temperatures for various pipe sizing is computed using an excel spreadsheet and equation (xvi) as seen in figure 36. The log mean temperature difference(ΔT_{lm}) was then computed with respect to the various pipe sizing using equation(iii). The piping loss was finally computed with respect to the various pipe sizing as seen in figure 36 and the total piping loss summed, where Qpipingloss = 124,380Btu/day = 5,182.5Btu/hr

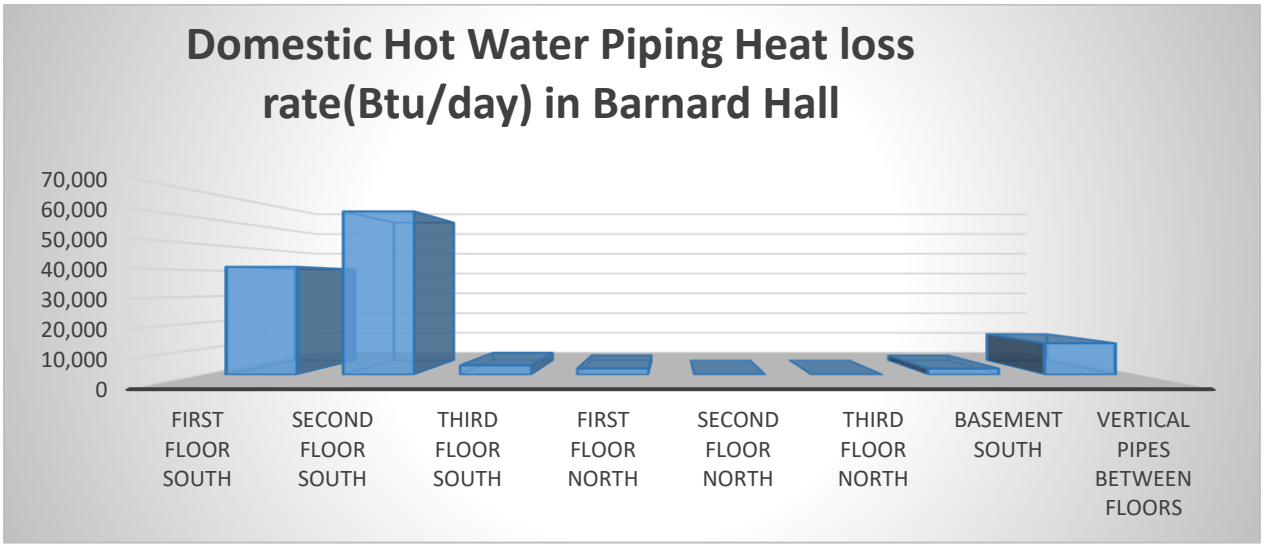


Figure 34. Domestic Hot Water Piping Heat Loss Rate in Barnard Hall

Thus, total Heat Input required is given as;

Qtotal = Qwater + Qpiping

=(500 x Flow rate x Temperature Difference) + Qpipingloss

= 115,000Btu/hr + 5,182.5Btu/hr = 120,182.5Btu/hr

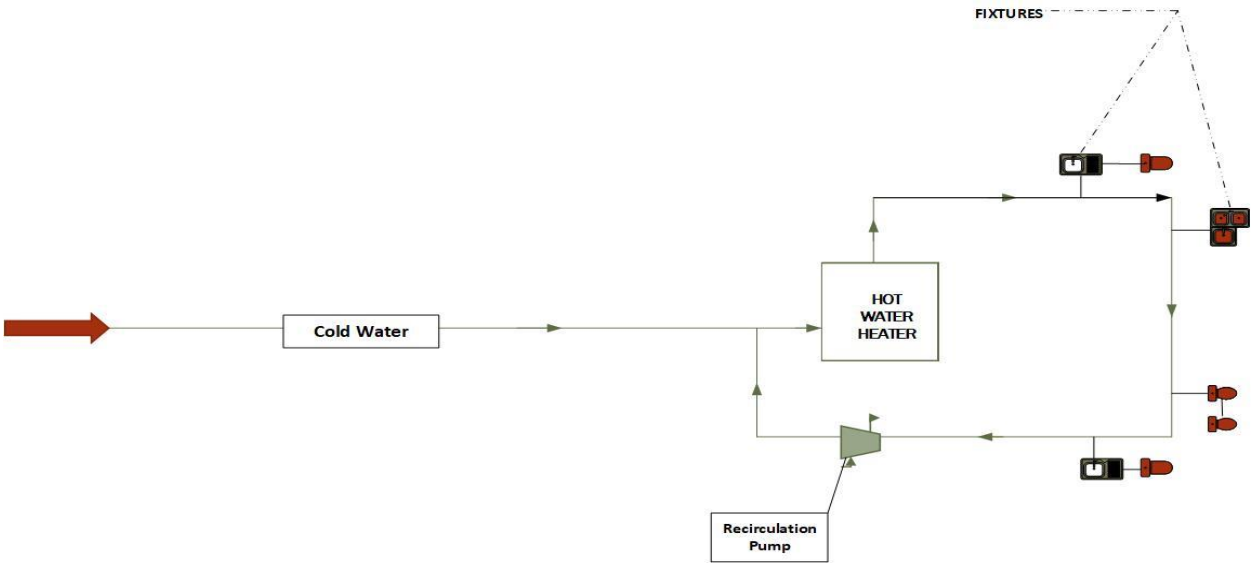


Figure 35. Domestic Water Heating System Sample in Barnard Hall

					DOMESTIC HOT WATER		
	NOMINAL PIPE SIZE(INCHES)	LENGTH OF PIPE(FEET)	LOG MEAN TEMP DIFFERENCE	Qpipingloss(BTU/hr)	Thot out(°F)	Qpipingloss(BTU/day)	
SECOND FLOOR SOUTH	HW	1.5	134	67	1438	134	34519
	HWC	0.5	134	68	913	136	21920
	HW	1	16	70	145	139	3480
	HWC	0.5	16	70	112	140	2679
FIRST FLOOR SOUTH	HW	0.75	71	69	586	138	14073
	HWC	0.5	53	69	367	139	8810
	HW	0.5	36	69	250	139	6005
	HW	0.75	34	69	283	139	6799
	HWC	0.5	34	70	236	139	5673
THIRD FLOOR SOUTH	HW	0.75	18	70	151	139	3613
FIRST FLOOR NORTH	HW	0.75	12	70	101	140	2412
SECOND FLOOR NORTH			N/A	-	-	-	-
THIRD FLOOR NORTH			N/A	-	-	-	-
BASEMENT SOUTH	HW	0.5	14	70	98	140	2345
VERTICAL PIPES BETWEEN FLOORS	HW(1st to 2nd Floor)	1	15	70	136	139	3263
	HWC(1st to 2nd Floor)	0.5	15	70	105	140	2512
	HW(2nd to 3rd Floor)	0.75	15	70	126	139	3013
	HW(To Penthouse)	1	15	70	136	139	3263
TOTAL					**5183**		**124380**

Figure 36. Spreadsheet Sample for calculating Hot Water Loss

Barnard Heat Pump Loop Load Calculations

1. (i) Domestic Hot Water Tank: The flow rate flowing through the domestic hot water pipe was previously calculated to be 2.3 GPM

Also, previously, Heat Input required = 500 x 2.3x (140 – 40) = 115,000btu/hr

Qpipingloss = 124,380btu/day = 5,182.5btu/hr (Previously Calculated)

Thus, total heat input required is given as;

Qtotal = Qwater + Qpiping

= (500 x Flow rate x Temperature Difference) + Qpipingloss

= 115,000btu/hr + 5,182.5btu/hr = 120,182.5btu/hr

(ii) Domestic Hot Water Heater Pump: From total domestic hot water load = 120,182.5 btu/hr, taking into account the heat pump effects, where the domestic hot water load demand serves as an heat sink, we reduce the heat taken from the water by 25% to simulate the compressor heat being added to the load. (Assumption made that the compressor heat accounts for 25% of the total heat based on data from actual heat pumps)

Hence, heat needed to be removed from the heat pump loop to meet the minimum load requirements = 75% x 120,182.5btu/hr = 90,136.88btu/hr

Calculating the Flowrate2 on the source side of the heat pump being connected to the heat pump loop, we have;

$$\text{FlowRate2} = \frac{\text{Qwater}}{500 \times \text{Temperature Difference}}$$

where Qwater = 90,136.88btu/hr, Temperature Difference = 10°F (Maximum Temperature Difference of 10°F assumed)

$$\text{Hence, Flow Rate2} = \frac{90{,}136.88}{500 \times 10} = 18.03\text{GPM}$$

2. Process cooling load: From previous calculations,

Process cooling load = 128,490 btu/hr

where Chilled water return temperature = 58°F

Chilled water supply temperature = 55°F

Flow rate = 85.66GPM (All from previous calculations and measurements)

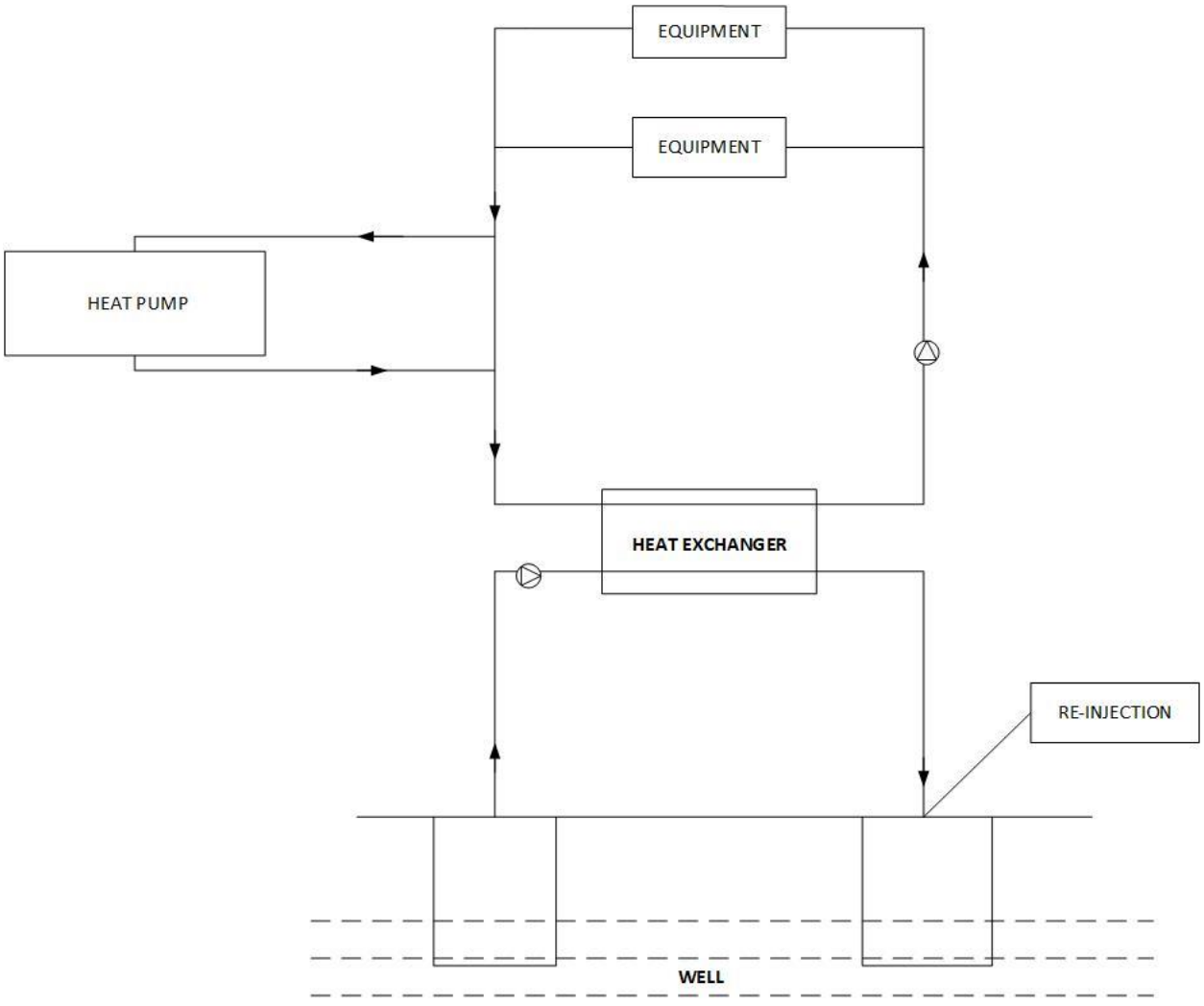


Figure 37. Process Cooling System Flow Diagram

Connecting a heat pump to the process cooling loop as shown in figure 51, where the process cooling loop serves as a heat source, we have;

Entering water temperature into the heat pump = Chilled water return temperature = 58°F,

For the leaving water temperature of the heat pump, we assume a maximum temperature rise of 10°F created by the heat pump

From Process cooling load = 128,490 btu/hr, we multiply the load by 4/3 to simulate the compressor heat rejection into the HP loop (working with the assumption made that the compressor heat accounts for 25% of the total heat based on data from actual heat pumps)

Hence, we have; 4/3 x 128,490 btu/hr = 171,320btu/hr

Thus, heat pump process cooling load = 500 x Flow rate2 x temperature difference(ΔT)

$$\text{FlowRate2} = \frac{\text{Heat Pump Process cooling load}}{500 \times \text{Temperature Difference}}$$

$$\text{FlowRate2} = \frac{171{,}320\text{btu/hr}}{500 \times 10} = 34.3\text{GPM}$$

where FlowRate2 represents the flow rate of the process cooling heat pump connected to the heat pump loop side.

3. Exhaust Air to Water Heat Pump:

For the air flow side of the heat pump, Qair = CFM x 0.92 x ΔT

where CFM = building exhaust air flowrate = 20,000CFM (design air flowrate)

Inlet Temperature = 70°F (room temperature)

Outlet Temperature = 55°F (maximum set point cooling temperature)

Hence, Qair = 20,000 x 0.92 x (70 – 55) = 20,000 x 0.92 x 15 = 276,000Btu/hr

Taking the building exhaust air as an heat source to the Heat Pump loop, we multiply the load by 4/3 to simulate the compressor heat rejection (working with the assumption made that the compressor heat accounts for 25% of the total heat based on data from actual heat pumps)

Thus we have, 4/3 x 276,000 = 368,000Btu/hr

For the water flow side of the heat pump, Qwater = GPM x 500 x temperature difference

$$GPM = \frac{Qwater}{500 \times \text{temperature difference}}$$

where Qwater = 368,000Btu/hr

Temperature difference = 10°F (we assume a maximum temperature rise of 10°F created by the heat pump)

Hence, $GPM = \frac{368{,}000}{500 \times 10} = 73.6GPM$

4. (i) Outdoor Air Heating Coil:

Qair = CFM x 0.92 x ΔT,

where CFM = Outdoor air flowrate = 20,000CFM

Tin = -24.32°F (Coldest air temperature chosen for the year to account for the maximum load, temperature data gotten from Energy Plus for Bozeman) [28]

Tout = Set point temperature = 55°F

Hence, Qair = 20,000 x 0.92 x (55 – (-24.32)) = 20,000 x 0.92 x (55 + 24.32)

Qair = 20,000 x 0.92 x 79.32 = 1,459,488 Btu/hr

For the water side of the heating coil, Qwater = GPM x 500 x temperature difference

Maximum temperature difference = $\frac{\text{Qwater}}{\text{GPM x 500}}$

where Qwater = Qair = 1,459,488Btu/hr

GPM = 55GPM (Although the total heat transfer increases as the flow rate increases as seen in figure 46, this flow rate was chosen as a result of the pump serving the coil consuming less power relative to other flow rates as seen later in the simulation results)

Maximum Water Temperature Difference(ΔT) = $\frac{1,459,488}{55 \text{ x } 500}$ = 53.07°F

(ii) Outdoor Air Cooling Coil

Qair = CFM x 0.92 x ΔT

where CFM = Outdoor air flow rate = 20,000CFM

Tin= 95°F (Hottest air temperature chosen for the year to account for the maximum load, temperature data gotten from Energy Plus for Bozeman) [28]

Tout= Set point temperature = 55°F

Hence, Qair = 20,000 x 0.92 x (95-55)) = 736,000 Btu/hr

5. Pump; Pump Power = $\frac{Q \times \Delta h}{3960 \times \eta_p}$ [29]

Q= water circulating rate (gallons per minute)

η_p = Pump efficiency

Δh = Pump head (Feet of water)

- For Domestic water heat pump;

Q = water circulating rate (gpm) = 18.8GPM

η_p = Pump efficiency = 0.6 (value assumed for typical pump efficiencies)

Δh = Pump head, feet of water = 120 feet of head (assumed from the pump head used in the process cooling system)

Hence, Pump Power = $\frac{18.8 \times 120}{3960 \times 0.6}$ = 0.9495hp

- For Process cooling heat pump;

Q = water circulating rate (GPM) = 34.3GPM

η_p = Pump efficiency = 0.6

Δh = Pump head, feet of water = 120

Pump Power = $\frac{34.3 \times 120}{3960 \times 0.6}$ = 1.732hp

- For Exhaust air heat pump;

Q = water circulating rate (GPM) = 74GPM

η_p = Pump efficiency = 0.6

Δh = Pump head, feet of water = 120

Pump Power = $\frac{74 \times 120}{3960 \times 0.6}$ = 3.737hp

- For Heating Coil;

Q = water circulating rate (GPM) = 55GPM

η_p = Pump efficiency = 0.6

Δh = Pump head, feet of water = 120

Pump Power = $\frac{55 \times 120}{3960 \times 0.6}$ = 2.778hp

6. Fan; Fan Power = $\frac{Q \times \Delta Pt}{6350 \times \eta_f}$ [29]

Q = Air circulation rate = 20,000cfm

ΔP_t = Total fan pressure, in. w.g. = 2.5 (Assumed)

η_f = Fan efficiency = 0.65 (Value assumed from typical fan efficiencies)

Hence, Fan Power = $\frac{20{,}000 \times 2.5}{6350 \times 0.65}$ = 12.11 hp

7. Boiler

Maximum Boiler Load = Total Heat Sink Load Demand – Total Heat Source Load Supply

Total Heat Sink Load Demand = Domestic Hot Water Load Demand + Maximum Outdoor Air Heating Load

Total Heat Sink Load Demand = 90,136.88Btu/hr +1,459,488 Btu/hr = 1,549,624 Btu/hr

Total Heat Sources Load = Process Cooling Heat Pump Load + Maximum Exhaust Air Heat Pump Load

Total Heat Source Loads = 171,320Btu/hr + 368,000Btu/hr = 539,320Btu/hr

Hence, Maximum Boiler Load = 1,549,624Btu/hr – 539,320Btu/hr = 1,010,304Btu/hr

Thus, 1,010,304Btu/hr maximum heat sink load demand will need to be met by the Boiler.

Also, Boiler Design Flow Rate = Exhaust Air Heat Pump Fluid Flow Rate + Heating Coil Fluid Flow Rate + Process Cooling Heat Pump Fluid Flow Rate + Domestic Water Heat Pump Fluid Flow Rate

Boiler Design Flow Rate = 73.66GPM + 55GPM + 34.3GPM + 18.03GPM = 180.99GPM

8. Cooling Tower:

For the Cooling Tower load calculation;

Total Heat Sink Load Demand = Domestic Hot Water Load Demand = 90,136.88Btu/hr

Total Heat Source Load Supply= Heat Pump Process Cooling Load + Maximum Exhaust Air Heat Pump load

Total Heat Source Load Supply = 171,320Btu/hr + 368,000Btu/hr = 539,320Btu/hr

Net Difference = Total Heat Sources Load Supply – Total Heat Sink Load Demand

Net Difference = 539,320Btu/hr - 90,136.88Btu/hr = 449,183.12Btu/hr

Also, Maximum Outdoor Air Cooling Load = 736,000 Btu/hr

Hence, load needed to be generated by the Cooling Tower to meet this Load = Maximum Outdoor Air Cooling Load + Net Difference = 736,000 Btu/hr + 449,183.12Btu/hr = 1,185,183.12Btu/hr

Similarly, Cooling Tower Flow Rate = Exhaust Air Heat Pump Fluid Flow Rate + Heating Coil Fluid Flow Rate + Process Cooling Heat Pump Fluid Flow Rate + Domestic Water Heat Pump Fluid Flow Rate [Since the Cooling Tower is to be connected in parallel with the Boiler]

Cooling Tower Flow Rate = 73.66GPM + 55GPM + 34.3GPM + 18.03GPM = 180.99GPM

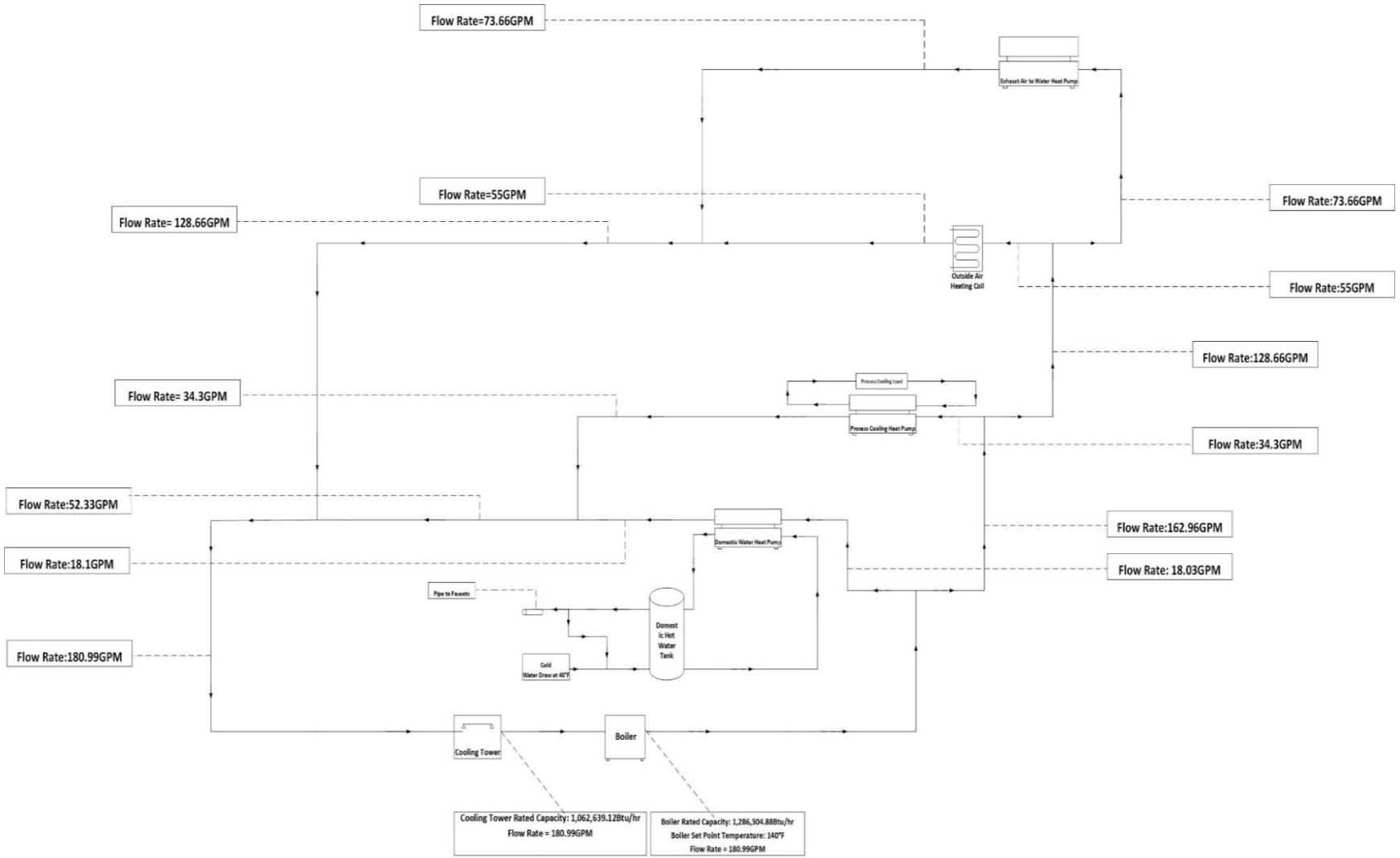


Figure 38. Ladder Diagram illustrating the Flow Rate

CHAPTER FOUR

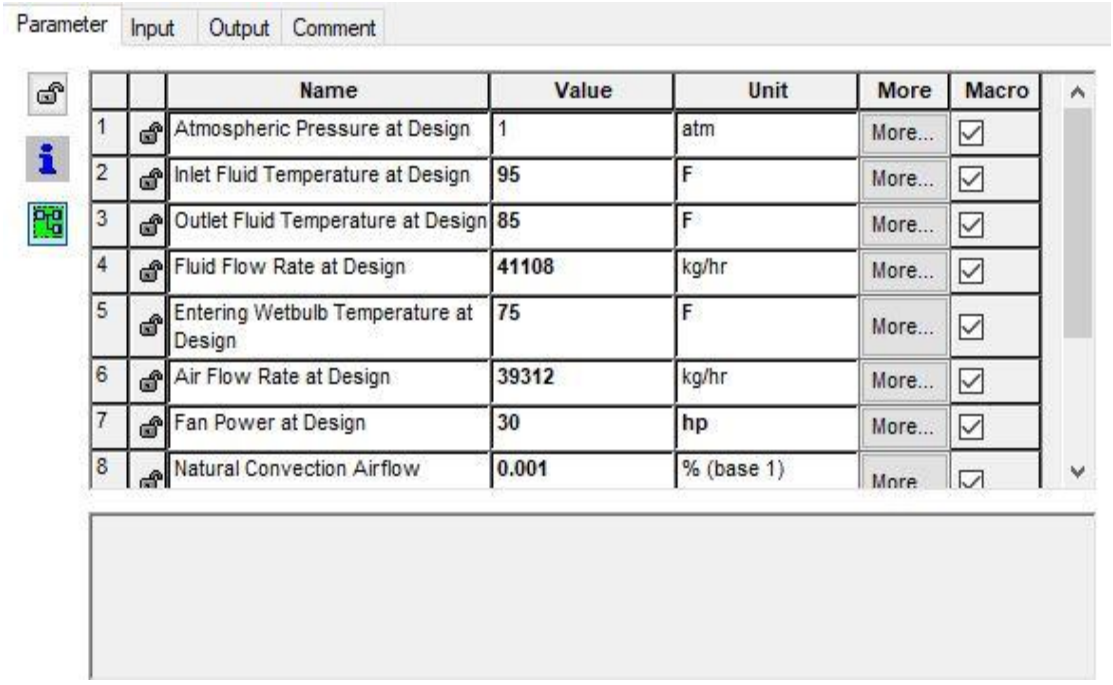
HEAT PUMP LOOP DESIGN

Parameters imputed into the TRNSYS program

The TRNSYS program by the default operates in the metric unit system, hence SI units previously calculated where converted to metric units before being imputed into the program. Detailed mathematical descriptions of the HVAC equations on which each component operates on can be found in the TRNSYS documentation library. The various components used in the design of the heat pump loop and details of the major imputed parameters are stated below;

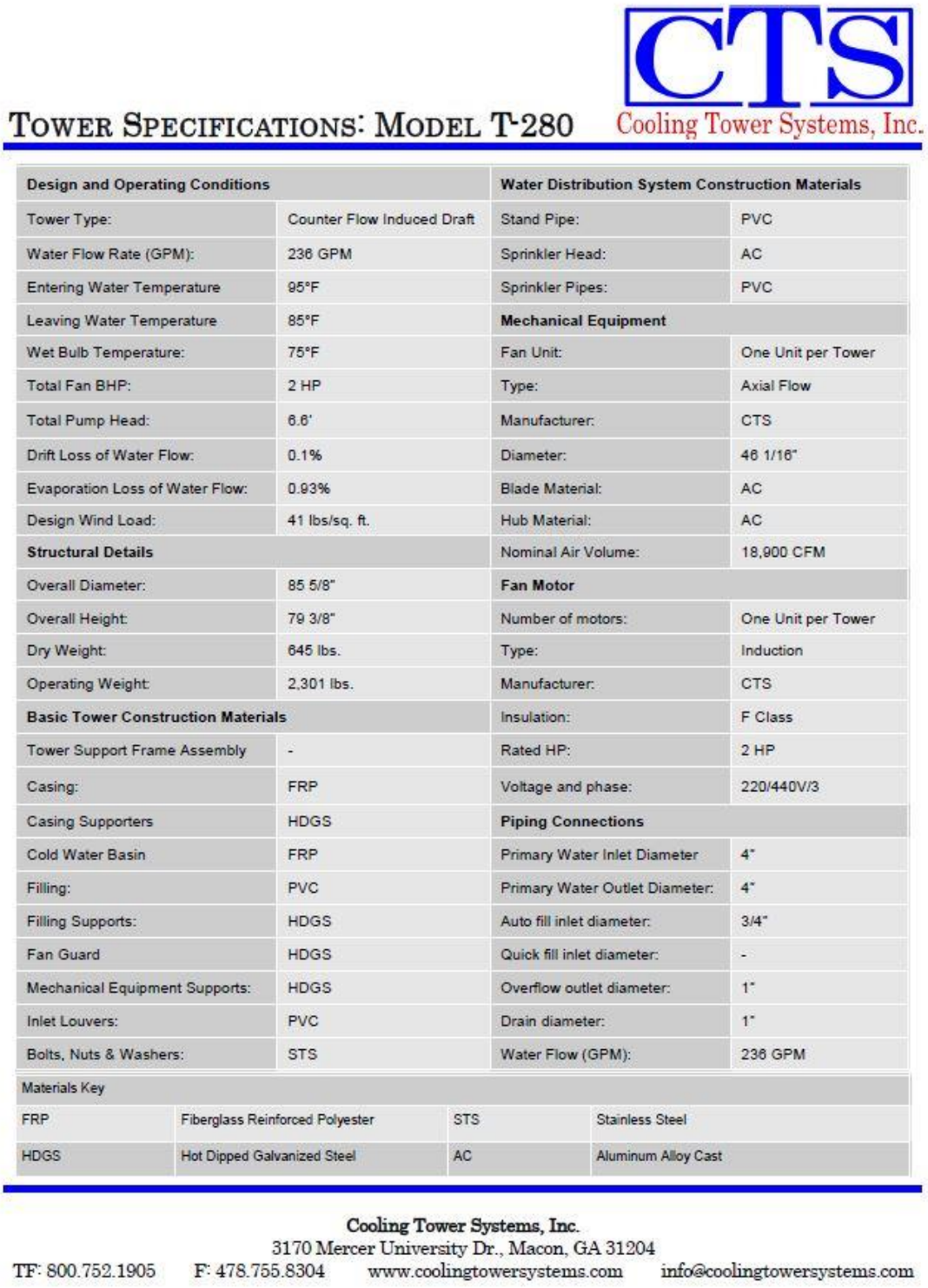
1. Cooling Tower: A single speed cooling tower was selected which uses the design inlet and outlet conditions to calculate an overall heat transfer coefficient(UA) for the tower and then uses that UA value to estimate performance at other inlet conditions.[30] Cooling tower design conditions was set under the parameter section with a flow rate design condition of the air through the cooling tower of 20,000 CFM, power fan design condition of 30HP and design fluid flow rate of 180.99GPM. Under the input section as shown below, the fluid entering temperature and the flow rate is gotten from previous flow conditions. The entering design fluid temperature, leaving temperature and wet bulb temperature was set to 95°F, 85°F and 75°F respectively. This was gotten from manufacturer's data as seen in figure 40 by selecting a cooling tower with a similar rated flow rate from previous calculations. Other parameters include the entering design air flow rate of 20,000CFM, entering design atmospheric pressure of 1atm, and the

cooling controller connected to the cooling tower. Weather data downloaded from the EnergyPlus website for Bozeman Yellowstone was connected to the cooling tower to provide the wet bulb temperature and atmospheric pressure for 8760hours, the time frame in which the simulation was computed.



		Name	Value	Unit	More	Macro
1		Atmospheric Pressure at Design	1	atm	More...	☑
2		Inlet Fluid Temperature at Design	95	F	More...	☑
3		Outlet Fluid Temperature at Design	85	F	More...	☑
4		Fluid Flow Rate at Design	41108	kg/hr	More...	☑
5		Entering Wetbulb Temperature at Design	75	F	More...	☑
6		Air Flow Rate at Design	39312	kg/hr	More...	☑
7		Fan Power at Design	30	hp	More...	☑
8		Natural Convection Airflow	0.001	% (base 1)	More	☑

Figure 39. TRNSYS impute details for the Cooling Tower



CTS
Cooling Tower Systems, Inc.

TOWER SPECIFICATIONS: MODEL T-280

Design and Operating Conditions		Water Distribution System Construction Materials	
Tower Type:	Counter Flow Induced Draft	Stand Pipe:	PVC
Water Flow Rate (GPM):	236 GPM	Sprinkler Head:	AC
Entering Water Temperature	95°F	Sprinkler Pipes:	PVC
Leaving Water Temperature	85°F	**Mechanical Equipment**	
Wet Bulb Temperature:	75°F	Fan Unit:	One Unit per Tower
Total Fan BHP:	2 HP	Type:	Axial Flow
Total Pump Head:	6.6'	Manufacturer:	CTS
Drift Loss of Water Flow:	0.1%	Diameter:	46 1/16"
Evaporation Loss of Water Flow:	0.93%	Blade Material:	AC
Design Wind Load:	41 lbs/sq. ft.	Hub Material:	AC
Structural Details		Nominal Air Volume:	18,900 CFM
Overall Diameter:	85 5/8"	**Fan Motor**	
Overall Height:	79 3/8"	Number of motors:	One Unit per Tower
Dry Weight:	645 lbs.	Type:	Induction
Operating Weight:	2,301 lbs.	Manufacturer:	CTS
Basic Tower Construction Materials		Insulation:	F Class
Tower Support Frame Assembly	-	Rated HP:	2 HP
Casing:	FRP	Voltage and phase:	220/440V/3
Casing Supporters	HDGS	**Piping Connections**	
Cold Water Basin	FRP	Primary Water Inlet Diameter	4"
Filling:	PVC	Primary Water Outlet Diameter:	4"
Filling Supports:	HDGS	Auto fill inlet diameter:	3/4"
Fan Guard	HDGS	Quick fill inlet diameter:	-
Mechanical Equipment Supports:	HDGS	Overflow outlet diameter:	1"
Inlet Louvers:	PVC	Drain diameter:	1"
Bolts, Nuts & Washers:	STS	Water Flow (GPM):	236 GPM

Materials Key			
FRP	Fiberglass Reinforced Polyester	STS	Stainless Steel
HDGS	Hot Dipped Galvanized Steel	AC	Aluminum Alloy Cast

Cooling Tower Systems, Inc.
3170 Mercer University Dr., Macon, GA 31204
TF: 800.752.1905 F: 478.755.8304 www.coolingtowersystems.com info@coolingtowersystems.com

Figure 40. Cooling Tower Specifications used



Figure 41. Evapco Cooling Tower sample used in Barnard Hall

2. Boiler: A Boiler was used with a rated capacity of 1,300,000 Btu/hr imputed under the parameter sections, this was gotten from previous calculations. The set point temperature of the boiler i.e. the desired temperature of the fluid exiting the device was set to 140°F. Boiler and consumption efficiency was assumed and set to be 75% and 75% respectively as shown below. Boiler rated flow rate was set to 180.99GPM from previous calculations. The other values were directly connected, such as the control signal being connected to the heating controller, the inlet fluid temperature and flow rate gotten from the incoming flow rate and temperature.



Figure 42. Heating Plant at Montana State University

Parameter | Input | Output | Comment

		Name	Value	Unit	More	Macro
1		Rated Capacity	1300000	BTU/hr	More...	☑
2		Fluid Specific Heat	4.190	kJ/kg.K	More...	☑
3		Minimum Turn-Down Ratio	0.2	-	More...	☑

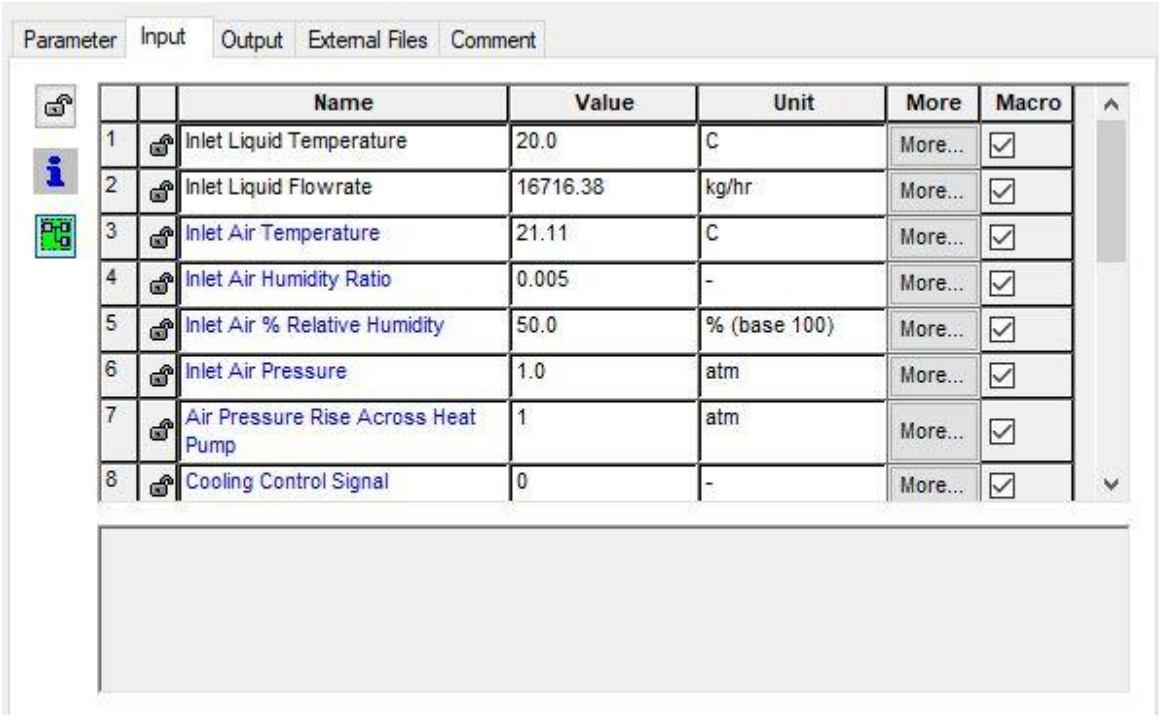
Parameter | Input | Output | Comment

		Name	Value	Unit	More	Macro
1		Inlet Fluid Temperature	10.0	C	More...	☑
2		Inlet Fluid Flowrate	41108	kg/hr	More...	☑
3		Input Control Signal	0.0	-	More...	☑
4		Set Point Temperature	140	F	More...	☑
5		Boiler Efficiency	0.75	Fraction	More...	☑
6		Combustion Efficiency	0.75	Fraction	More...	☑

(Values of components not highlighted in blue can be neglected as this takes its value from the incoming pipe)

Figure 43. TRNSYS impute parameters for the Boiler

3. Exhaust Air Heat Pump (Air to Water Heat Pump): This component models a single stage air source heat pump with a liquid stream on the load side[23]. The air side is the source and the heat pump can either heat or cool the liquid stream but was set to heat the liquid stream through control signals being sent by the heating controller. The heat pump is based off manufacturer's data from actual heat pumps. As seen in the figure below, the heat pump was set to maintain a water flowrate of 73.6 GPM whenever fluid flows through it, this was gotten from previous calculation. Other major inputs are the inlet air temperature: 70°F, inlet air humidity ratio: 0.005 (pounds moisture per pound dry air), inlet air relative humidity: 50%, inlet air pressure: 1atm and the inlet liquid temperature is the incoming liquid temperature from the previous pipe as seen in Figure 69 below. The rated capacity was determined from previous calculation.



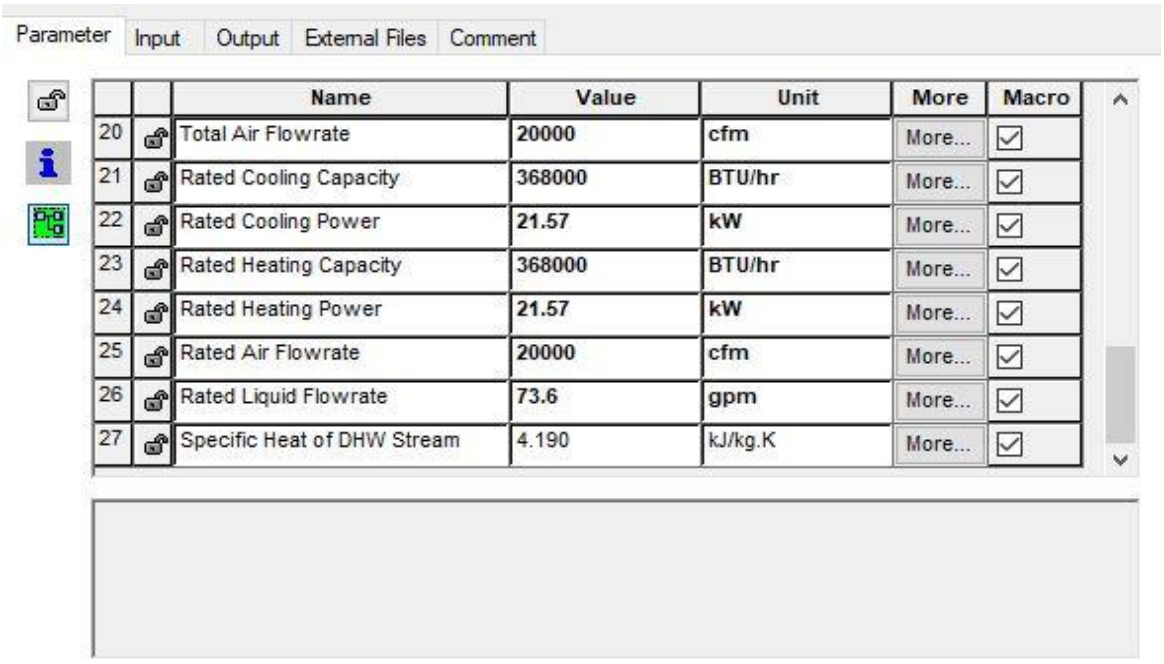
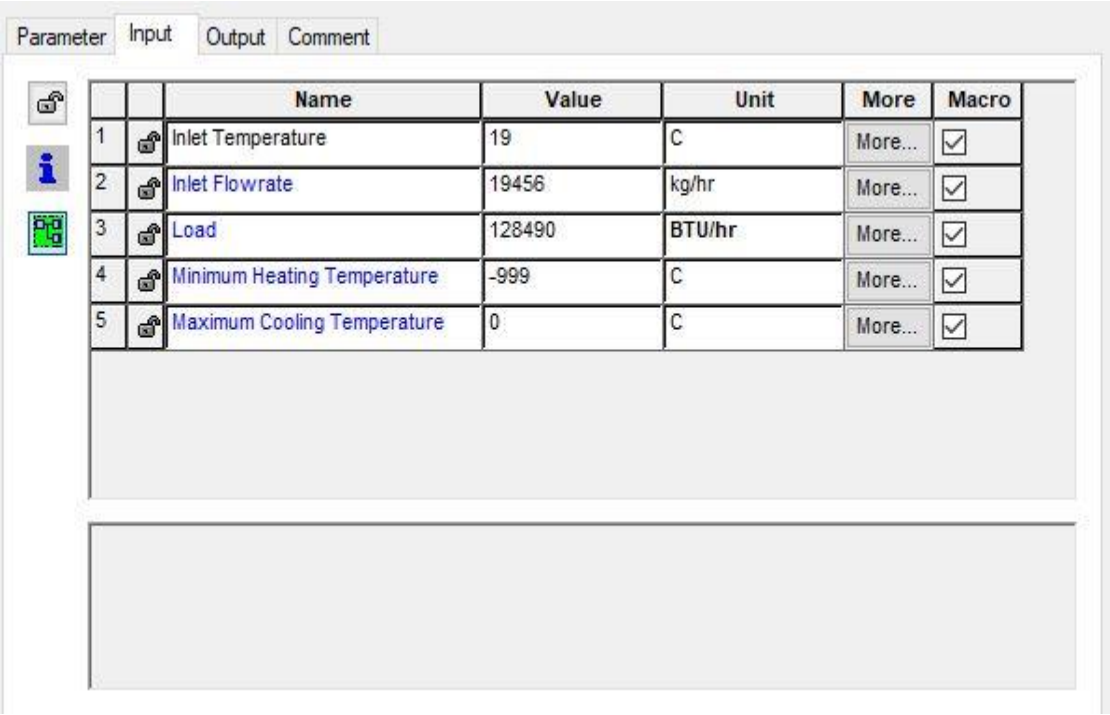


Figure 44. TRNSYS impute parameters for the Exhaust Air Heat Pump

4. Process Cooling Load: Pre calculated loads from the process loop was imputed into the program below which was 128,490 Btu/hr. The component was set to always supply a flow rate of 85.66GPM which was gotten from the measured data. The inlet temperature is gotten from the previous incoming temperature in the pipe.



(Values of components not highlighted in blue can be neglected as this takes its value from the incoming pipe)

Figure 45. TRNSYS impute parameters for the Process Cooling Load

5. Process Cooling Heat Pump (Water to Water Heat Pump): This component models a single-stage heat pump. The heat pump conditions a one liquid stream by rejecting energy to (cooling mode) or absorbing energy from (heating mode) a second. This model is a performance map meaning that is the results generated are based on supplied data files from actual Heat Pumps for the capacity and power draw as a function of entering source and load temperature [30]. The pre calculated heating capacity and flow rate were imputed as shown below.

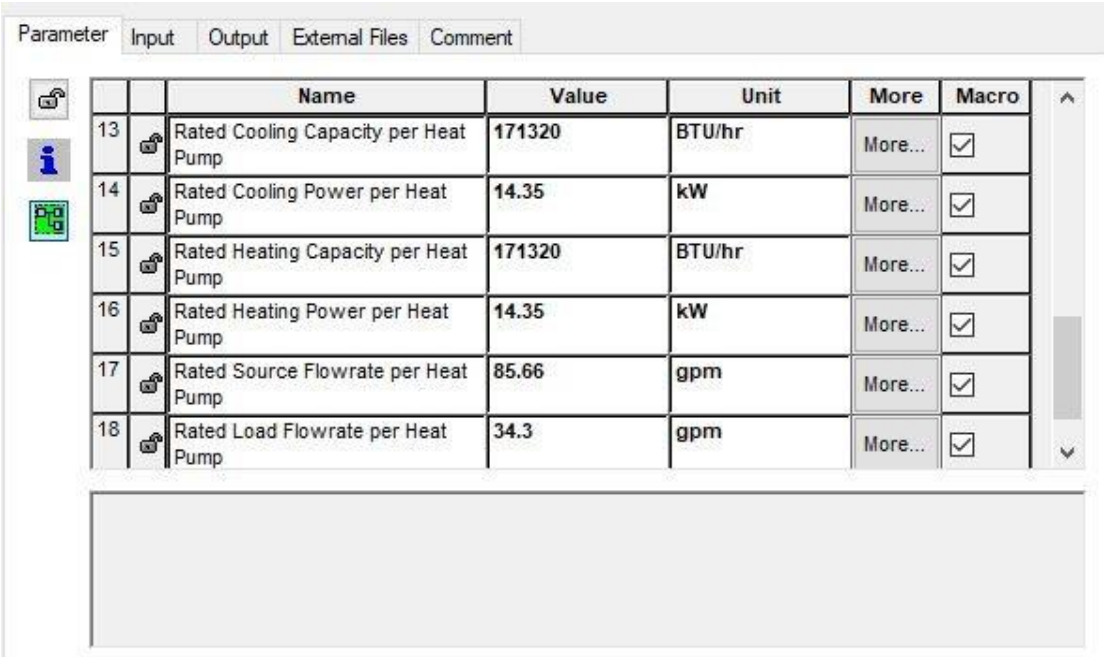


Figure 46. TRNSYS impute parameters for the Process Cooling Heat Pump

6. Domestic Hot Water Tank; A plug flow tank was used to model the domestic hot water tank. The domestic hot water tank provides auxiliary heating when necessary. Currently, no auxiliary heating was imputed or being provided since the total pre calculated heating capacity was imputed into the domestic hot water heat pump. A domestic hot water draw schedule was created and connected to the domestic hot water tank alongside the heat pump, the outlet of the tank is then connected to the faucets via pipes as seen figure 55.

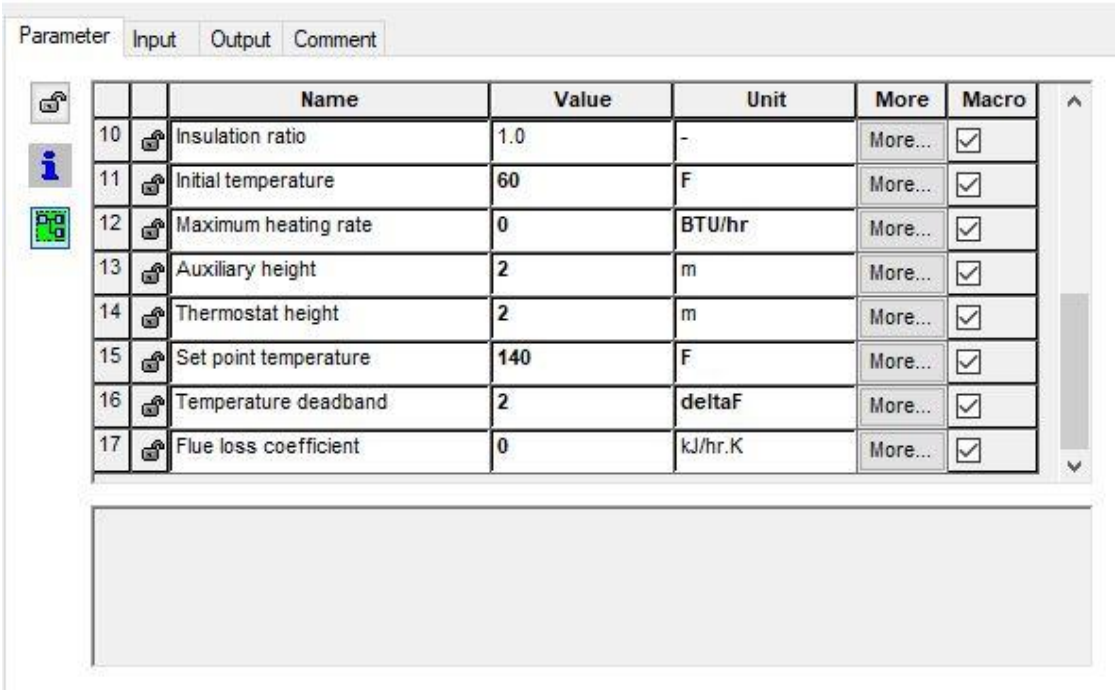


Figure 47. TRNSYS impute parameters for the Domestic Hot Water Tank

7. Water Draw Schedule; This component models a time dependent forcing function that has a behavior characterized by a repeated pattern. The water draw schedule for the domestic hot water was set. From previous calculations, the average flow rate for a faucet is 2.2GPM, 0.1GPM was added to estimate the impact of pipe loss on flow rate. A flow rate of 1.0GPM was selected for non-working hours. The water flow rate shown in the image below is in Kg/hr, the equivalent GPM conversion is shown below;

   12am – 7am: 227.12Kg/hr (1.0GPM)

   7am – 8am: 227.12Kg/hr (1.0GPM) – 522kg/hr (2.3GPM)

   8am – 6pm: 522Kg/hr (2.3GPM)

   6pm – 12am: 522Kg/hr(2.3GPM) – 227.12Kg/hr (1.0GPM)

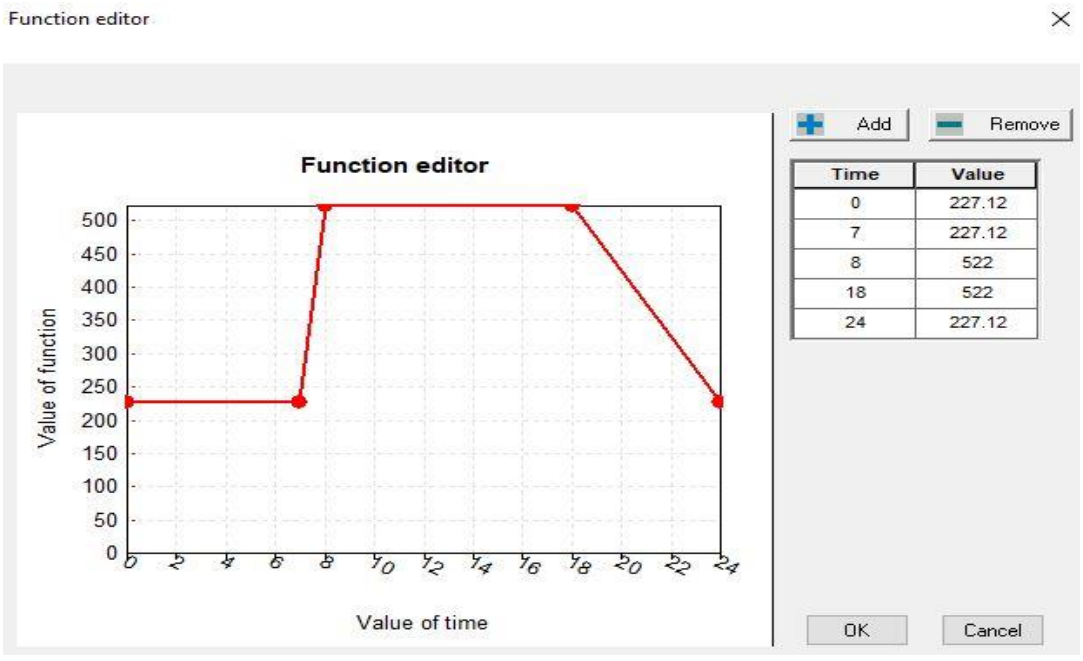


Figure 48. Water Draw Schedule

8. Domestic Water Heat Pump (Water to Water Heat Pump); This component models a single-stage heat pump. The heat pump conditions a one liquid stream by rejecting energy to (cooling mode) or absorbing energy from (heating mode) a second. This model is a performance map meaning that is the results generated are based on supplied data files from actual heat pumps for the capacity and power draw as a function of entering source and load temperature [30]. The pre calculated heating capacity and flow rate were imputed as shown below.

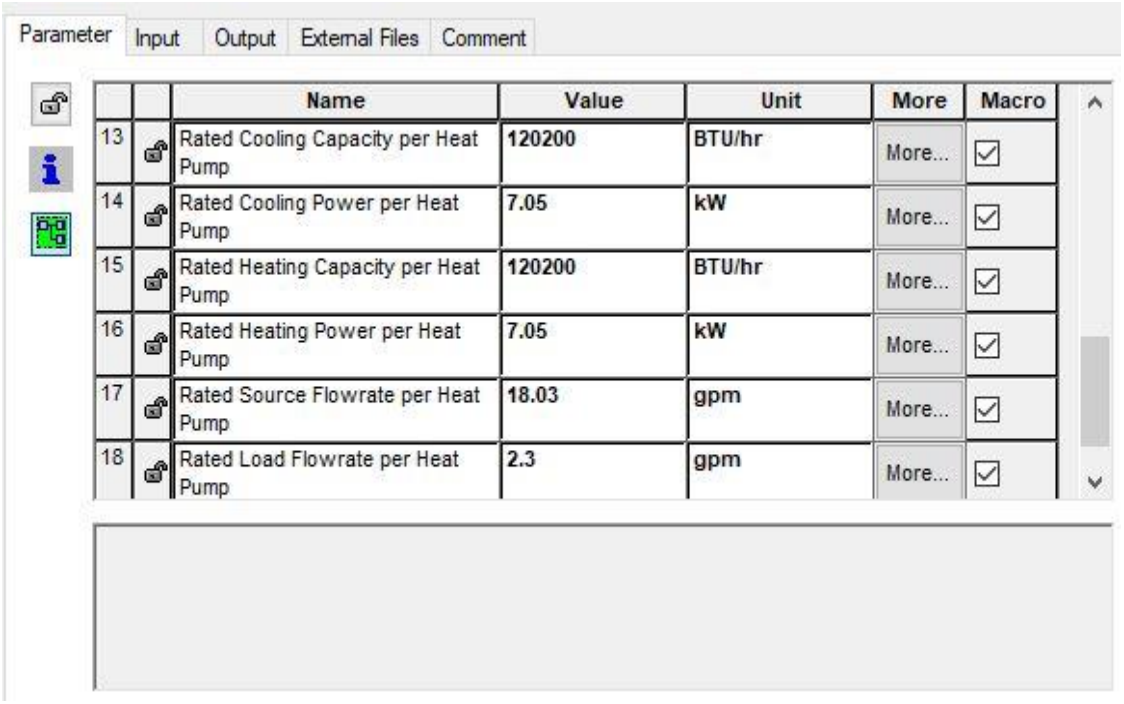


Figure 49. TRNSYS impute Parameters for the Domestic Water Heat Pump

9. Outdoor Air Heating Coil; A liquid source heating coil was used to heat the outdoor air since during most period of the year, Bozeman is typically cold. Weather Data downloaded from the EnergyPlus website for Bozeman Yellowstone was connected to the coil to provide the dry bulb temperature, humidity ratio, percentage relative humidity and atmospheric pressure for 8760hours, the time frame in which the simulation was computed. The outdoor air flow rate was set to 20,000CFM with the heating coil set to always maintain a specified flow rate whenever incoming water flows through the component. A water flow rate of 55GPM was imputed from previous calculations done.

The fluid outlet temperature of the coil is determined by;

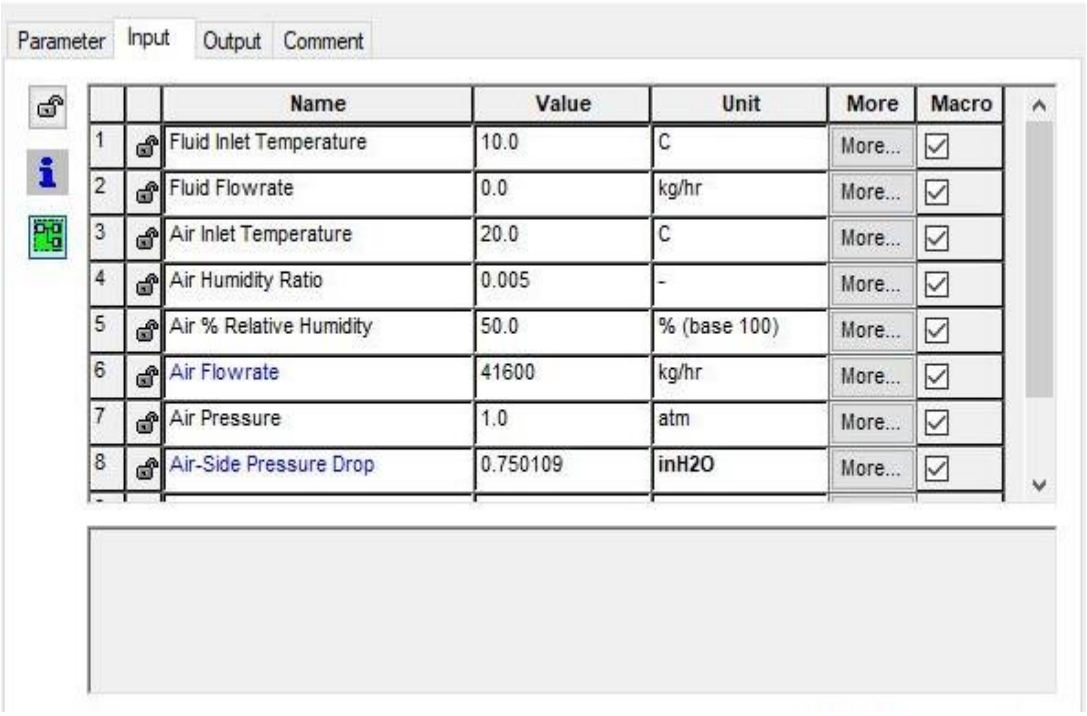
$$T_{fluidout} = T_{fluidin} + Q_{fluid}/\dot{m}_{fluid}Cp_{fluid}$$

where T_{in} = Temperature of fluid entering the coil

Q= Energy transferred from the air stream to the fluid stream

$\dot{m}_{fluid}$= Flow rate of fluid through the coil

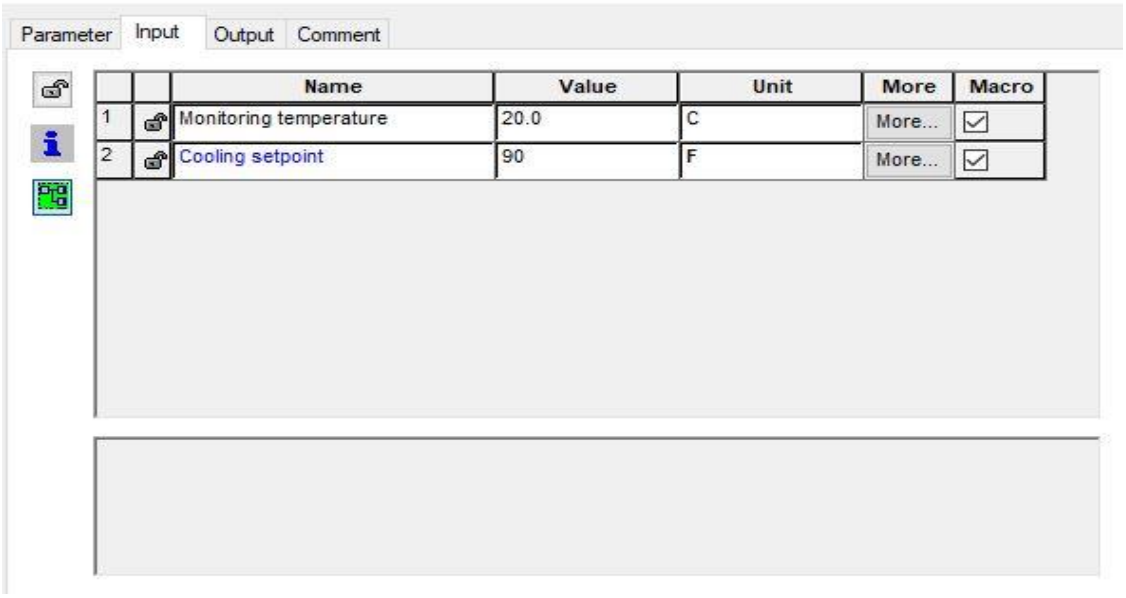
Cp_{fluid}= Specific heat of coil fluid



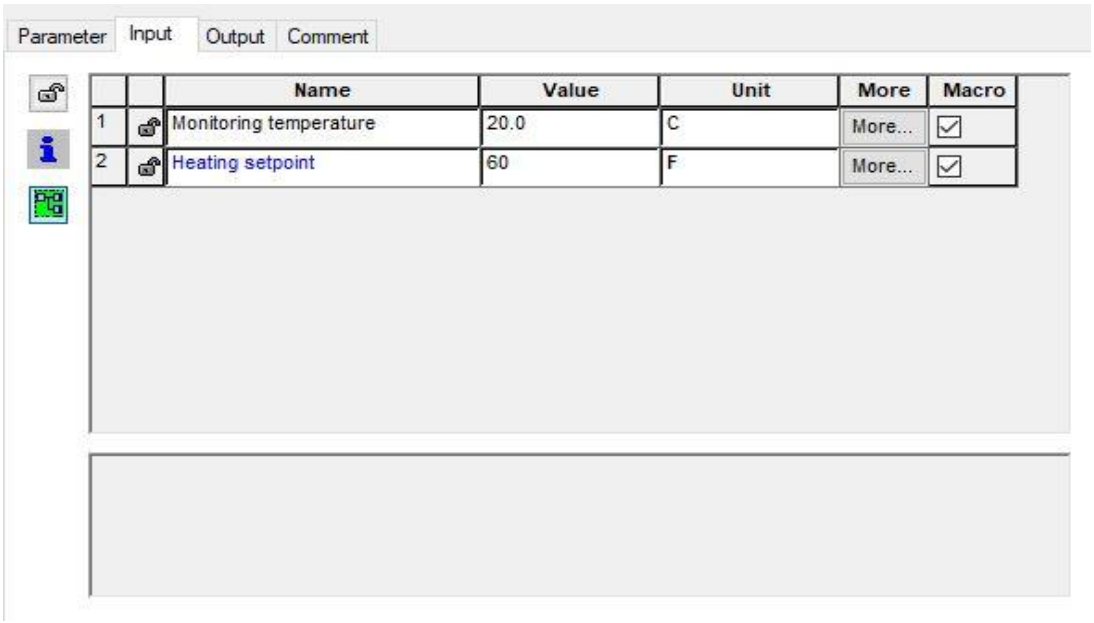
(Values of components not highlighted in blue can be neglected as this takes its value from the incoming pipe or weather data)

Figure 50. TRNSYS impute Parameters for the Outdoor Air Heating Coil

10. Cooling and Heating Controllers: For the Cooling and Heating controller, a setpoint input temperature of 90°F and 60°F respectively was inputed. For the cooling controller, the setpoint temperature 90°F is the temperature above which cooling is commanded from the Cooling Tower. Similarly, for the heating controller, the setpoint temperature 60°F is the temperature below which heating is commanded from the boiler.



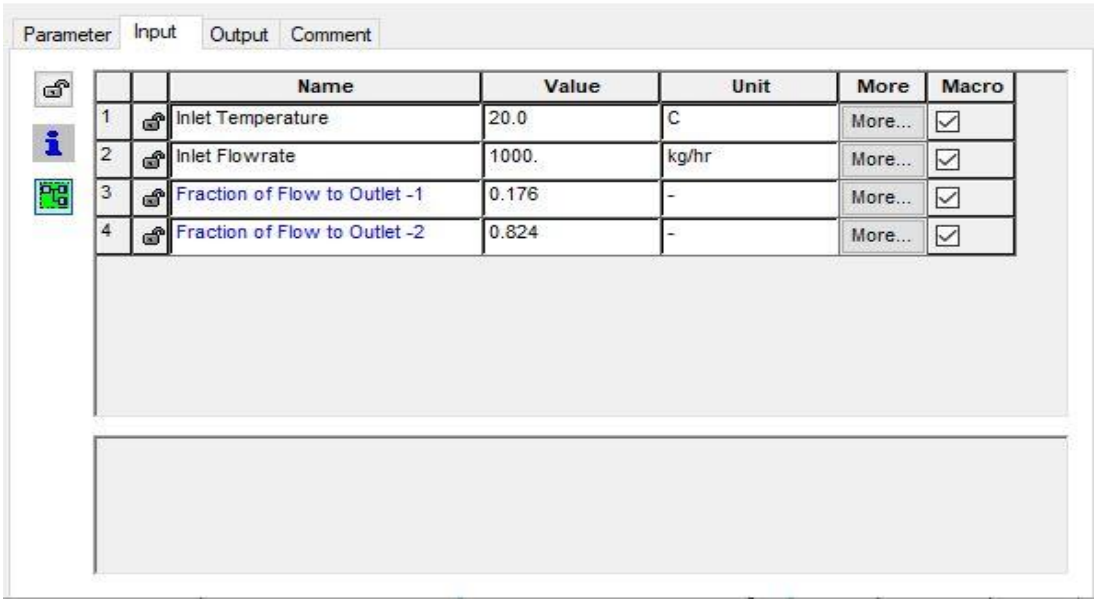
Cooling Controller



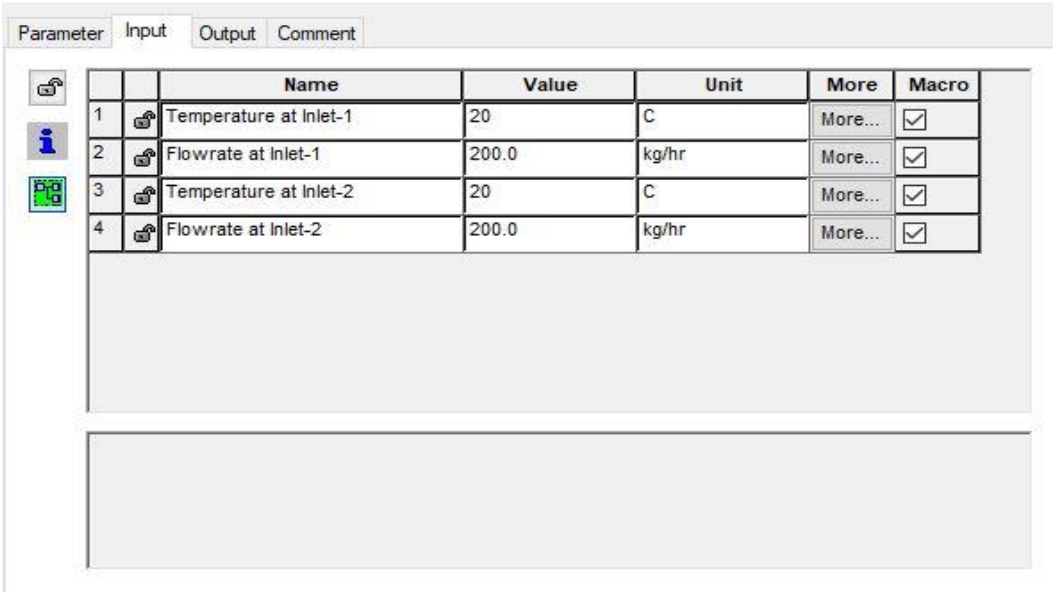
Heating Controller

Figure 51. TRNSYS impute Parameters for the Heating and Cooling Controller

11. Flow Diverter/ Flow Mixer: The flow diverter diverts the flow of water in the heat pump loop to the necessary systems based on the demand while the flow mixer converges the flow from different sources within the loop. The flow diverter works by imputing the fraction of flow needed to be diverted to the respective outlet. An example of the fraction of flow imputed into the flow divert component is shown below. For the various flow diverters used, the various fraction needed were calculated based on the pre calculated flow rate for each system. For the flow mixer, this consist of the previous incoming temperatures and flow rates converging together in the flow mixer where the flow rate is the net sum of both flow rates.



Flow Diverter



Flow Mixer

Figure 52. TRNSYS impute Parameters for the Flow Diverter and Mixer

12. Pump: A variable speed pump was used which can maintain any outlet mass flowrate between zero and a rated value. The four pumps used as seen in figure 55 have control signals connected to the systems they serve, i.e. the respective pump turns on when such system is on and turns off when such system is off.

    The pump rated power being imputed was calculated by;

$$\text{Pump rated power} = \frac{Q \text{ x } \Delta h}{3960 \text{ x } \eta_p} \quad [32]$$

Where; Q= Water circulating rate(GPM)

η_p = Pump efficiency

Δh = Pump head, feet of water

The Pump power draw is modeled using;

$P = Prated (a_0 + a_1Y + a_2Y^2 + a_3Y^3 + a_4Y^4 + \ldots.)$

Where Prated = Pump rated power,

a_0, a_1, a_2, a_3, a_4 = Polynomial coefficients in the pump power curve

Y = Pump control signal

The pump affinity law which states that the pump power is directly proportional to the cube root of the flow rate was then applied.

$$\frac{Q1}{Q2} = \frac{D1}{D2} \text{ and } \frac{D1\text{^}3}{D2\text{^}3} = \frac{BHP1}{BHP2} \text{ [33],}$$

$$\text{thus} \quad \frac{Q1\text{^}3}{Q2\text{^}3} = \frac{BHP1}{BHP2}$$

Where Q = Flow rate, D = Impeller Diameter and BHP= Brake horsepower.

Hence, a_0, a_1 and a_2 were set to 0 in the imput and a_3 set to 1 as shown in figure 53.

Hence $P = Prated (Y^3)$

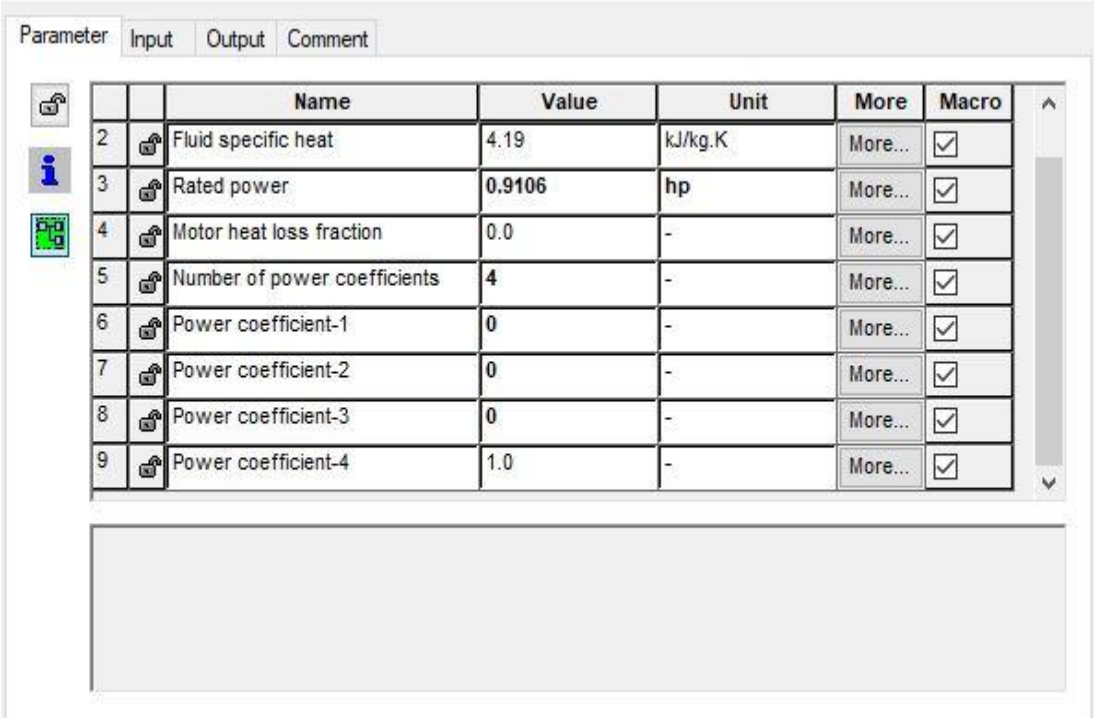


Figure 53. TRNSYS impute Parameters for the Pumps

13. Fan: A variable speed fan was used which is able to turn on at any speed between 0 (full stop) and its rated speed. The rated speed is the maximum volumetric flow rate. The two fans used as seen in figure 55 have control signals connected to the systems they serve, i.e. the respective fans turns on when such system is on and turns off when such system is off. The fan rated power being imputed was calculated by;

$$\text{Fan Power} = \frac{Q \times \Delta Pt}{6350 \times \eta_f}$$

Q = Air circulation rate(GPM),

ΔP_t = Total fan pressure (in. w.g.)

η_f = Fan efficiency

The Fan Power draw is modeled using;

$P = P_{rated} (a_0 + a_1Y + a_2Y^2 + a_3Y^3 + a_4Y^4 + \ldots)$

where Prated = Fan Rated Power,

a_0, a_1, a_2, a_3, a_4 = Polynomial coefficients in the pump power curve

Y = Pump control signal

The Fan affinity law which states that the fan power is directly proportional to the cube root of the flow rate was then applied

$$\frac{Q1}{Q2} = \frac{N1}{N2} \text{ and } \frac{N1\text{^}3}{N2\text{^}3} = \frac{BHP1}{BHP2} \text{ [34]},$$

$$\text{thus} \quad \frac{Q1\text{^}3}{Q2\text{^}3} = \frac{BHP1}{BHP2}$$

Where Q = Flow rate, N = Speed of Rotation and BHP= Brake horsepower.

Hence, a_0, a_1 and a_2 were set to 0 and a_3 set to 1 as shown in figure 54.

Hence $P = P_{rated} (Y^3)$

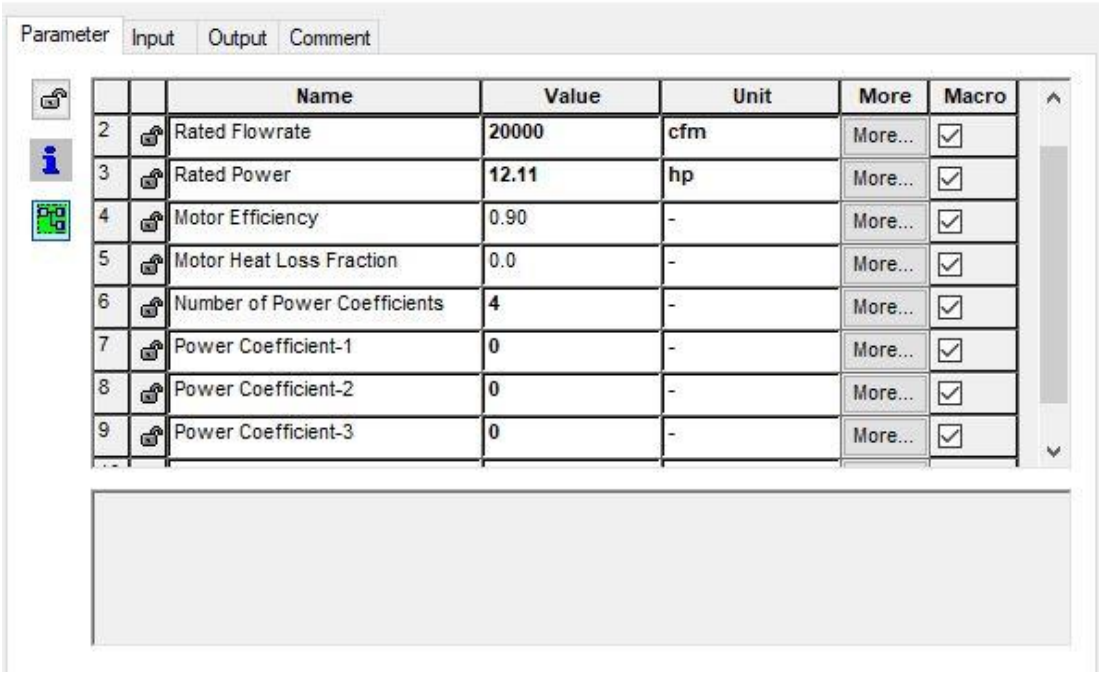


Figure 54. TRNSYS impute Parameters for the Fans

Heat Pump Loop Design with TRNSYS

The final arrangement as seen in figure 55 was designed and simulated with the TRNSYS program to evaluate the performance of the heat pump loop. Heat sources and heat sinks were connected together in the recirculating heat pump loop. This includes the process cooling load (serving as an heat source) connected to the loop through a water to water heat pump, the building exhaust air (serving as an heat source) in which heat is being extracted by the air to water heat pump, outdoor air (serving as an heat sink) connected to the heat pump loop through an heating coil (since Bozeman is majorly cold throughout the year) and the domestic hot water system (serving as an heat sink) connected to the heat pump loop through a water to water heat pump. The simulation was set to run for a full calendar year. Weather data was downloaded from EnergyPlus which is another building energy simulation program and connected to the heating coil as well as the cooling tower in the heat pump loop as shown in the figure below. This was done to provide values for parameters such as the dry bulb temperature, humidity ratio, relative humidity, atmospheric pressure, and the wet bulb temperature for the entire year.

The model was designed to operate between 60°F and 90°F, this was ensured by the heating and cooling controller, which are connected to the recirculating loop, to turn on and off the boilers and cooling towers respectively when necessary. For example, if the signal received by the controller indicates that the loop is operating within the temperature limits, the cooling tower and boiler remains off and the recirculation continues, however, if the temperature is below 60°F or above 90°F, the boiler turns on or the cooling tower turns on as required.

Also, signals were sent from a point in the recirculating loop to the exhaust air heat pump, process cooling heat pump and the outdoor air coil simultaneously. For the process cooling system, the process cooling heat pump controller receives signal from the point in the recirculating loop and sends signal to the heat pump and to pump 2 simultaneously as seen in figure 55, to turn on if the temperature of the loop goes below the heating setpoint(75°F). For the exhaust air system, the exhaust air heat pump controller receives signal from the loop and sends signal to the heat pump, pump 4 and fan 2 simultaneously to turn on if the temperature of the loop goes below the heating setpoint(70°F). Also, for the domestic water heat pump, the heat pump is always turned on due to the continuous load demand being imputed as the heat pump capacity.

Lastly, for the heating coil, the controller receives signal from the point in the loop and sends signal to fan1 and pump3 simultaneously to turn on if the temperature of the loop goes below the heating setpoint(55°F).

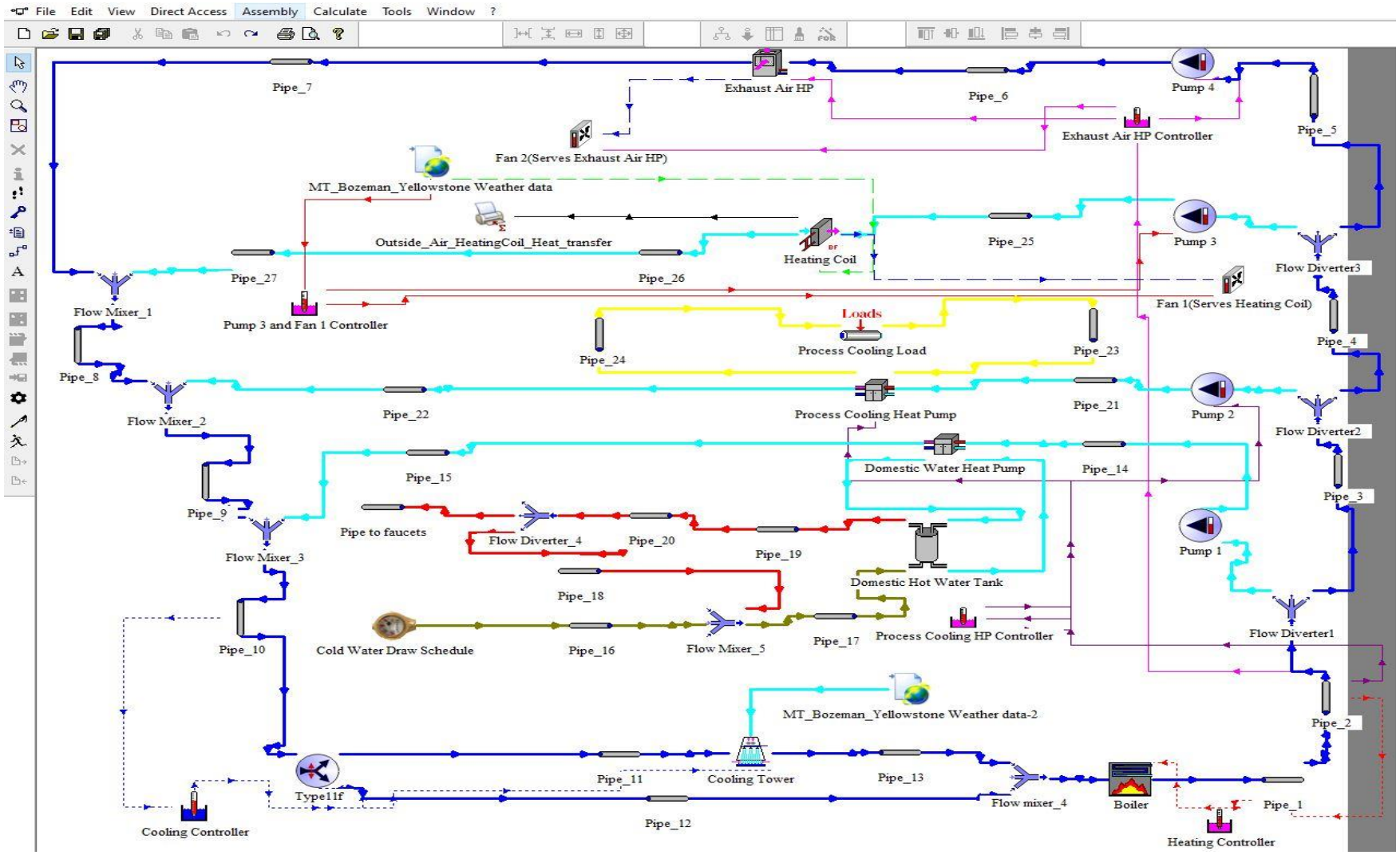


Figure 55. Heat Pump Loop Design

The setpoint temperatures of the controllers for the various heat pumps, fans, pumps and heating coil were determined by initially designing five different configurations of the various systems and then simulated to see which provided maximum efficiency/value among the designs.

Design 1: This has the exhaust air heat pump connected in series with the outdoor air heating coil. The process cooling heat pump as well as the domestic water heat pump are connected in parallel with the outdoor air heating coil as seen in figure 56.

Design 2: This has the exhaust air heat pump connected in series with the outdoor air heating coil. The process cooling heat pump is connected in series with the domestic water heat pump as seen in figure 57. The flow diverter to the process cooling heat pump occurs before the exhaust air heat pump

Design 3: This has the exhaust air heat pump connected in series with the outdoor air heating coil. The process cooling heat pump is connected in series with the domestic water heat pump as seen in figure 58. The flow diverter to the process cooling heat pump occurs after the exhaust air heat pump

Design 4: This has the outdoor air heating coil connected in series with the process cooling heat pump as seen in figure 59. The exhaust air heat pump is connected in series with the domestic water heat pump. The flow diverter to the process cooling heat pump occurs after the exhaust air heat pump

Design 5: This has the outdoor air heating coil connected in series with the process cooling heat pump as seen in figure 60. The exhaust air heat pump is connected in series with the domestic water heat pump. The flow diverter to the process cooling heat pump occurs before the exhaust air heat pump

Design 1

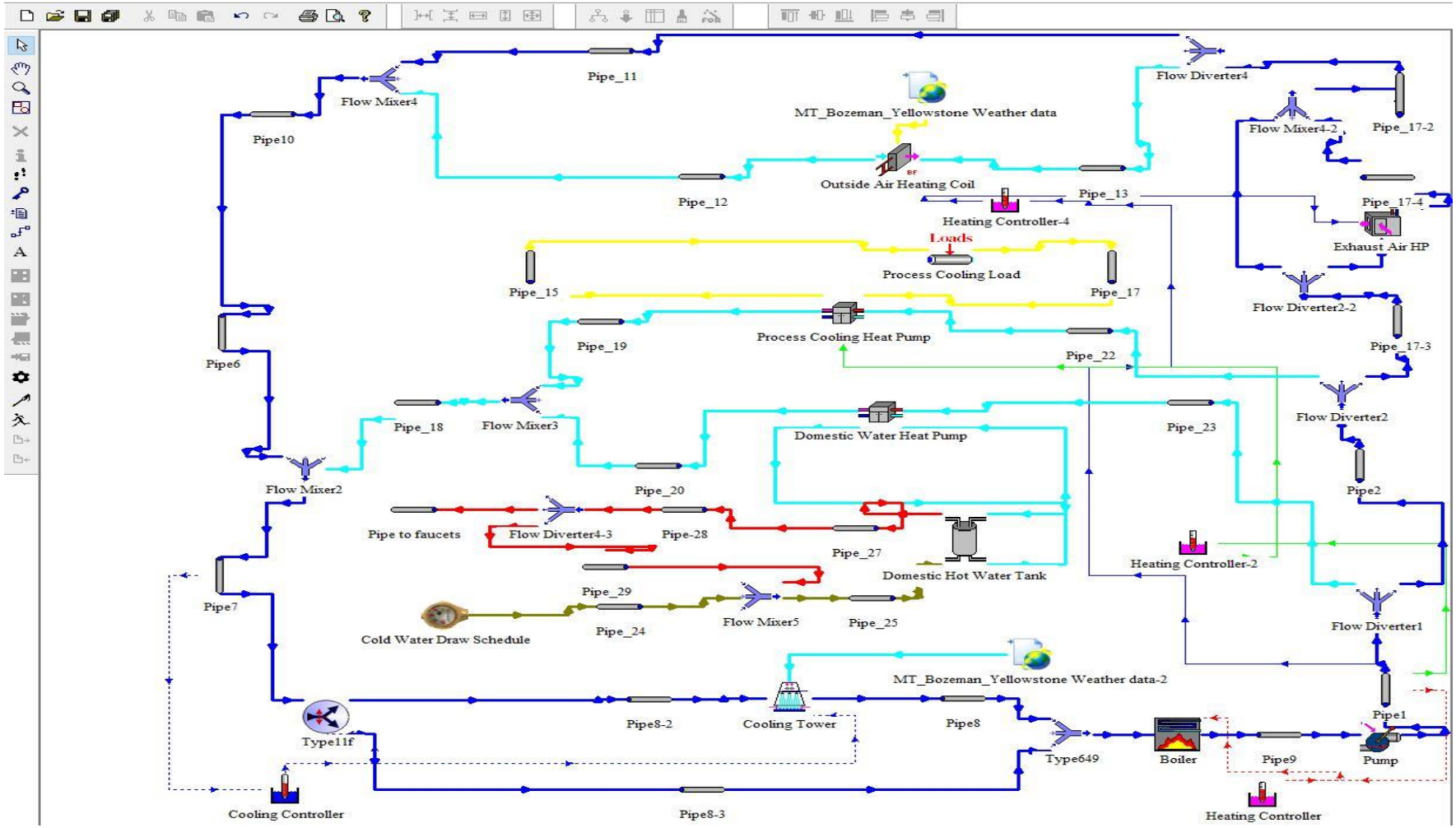


Figure 56. Heat Pump Loop Design 1

Design 2

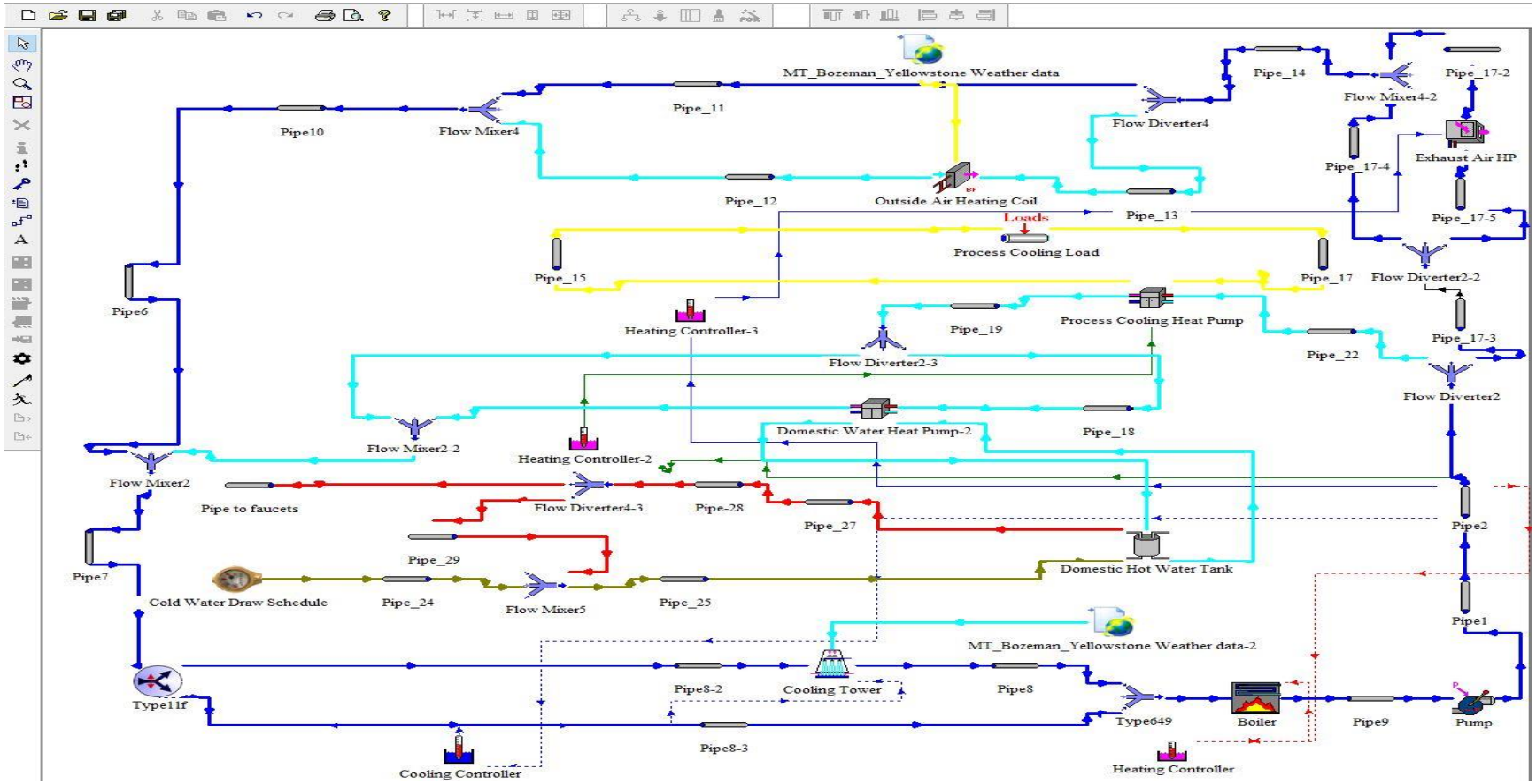


Figure 57. Heat Pump Loop Design 2

Design 3

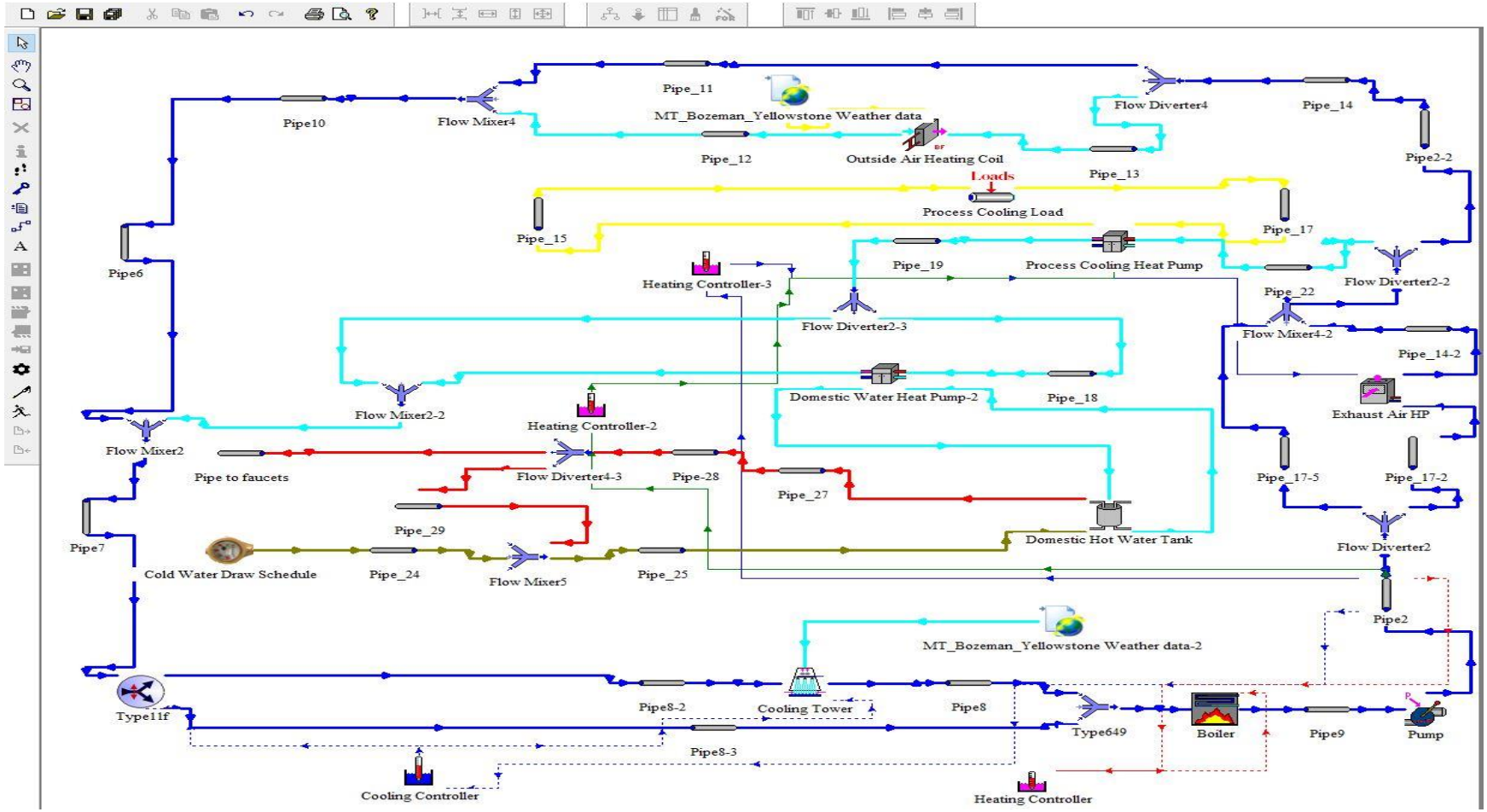


Figure 58. Heat Pump Loop Design 3

Design 4

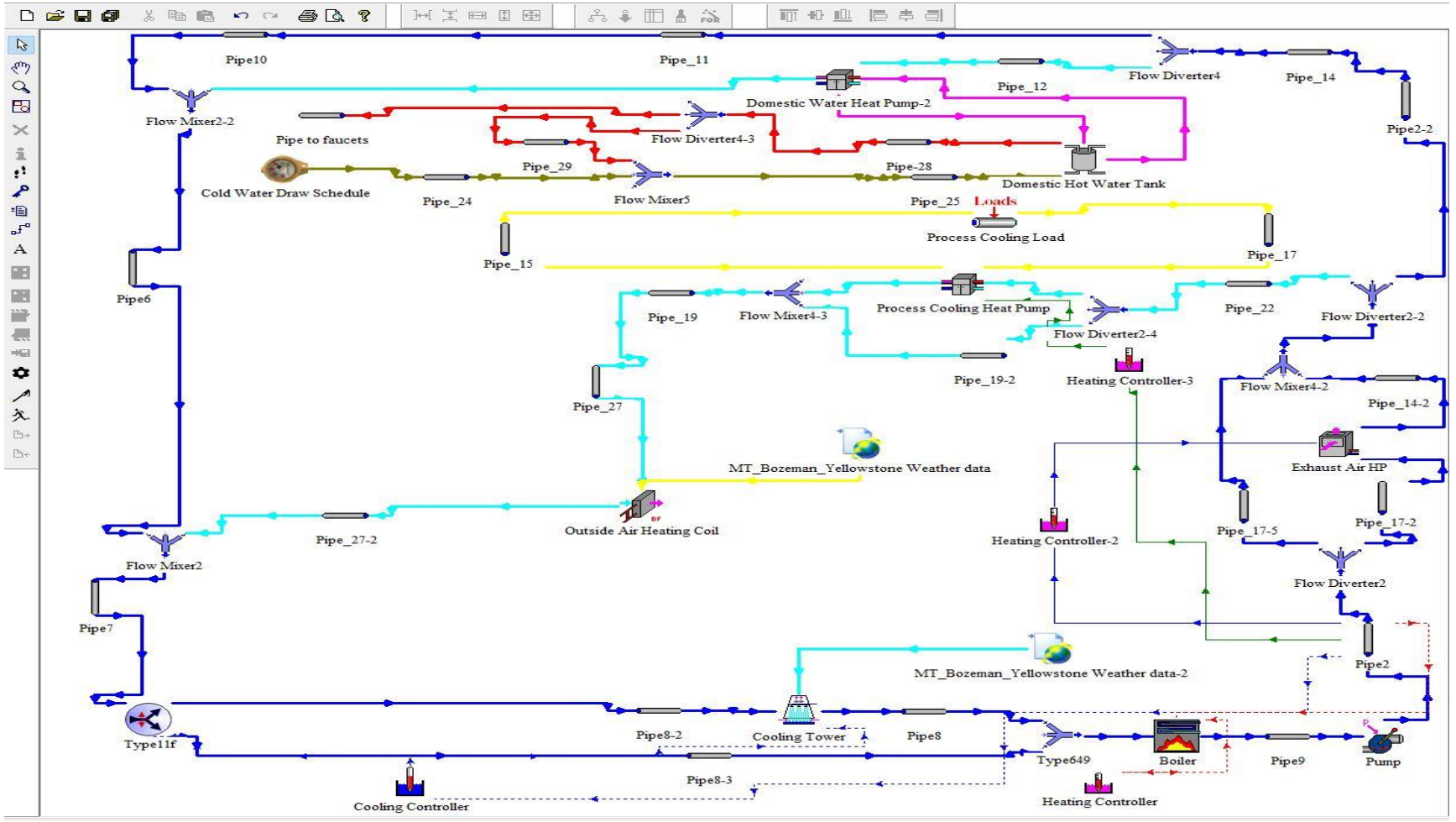


Figure 59. Heat Pump loop Design 4

Design 5

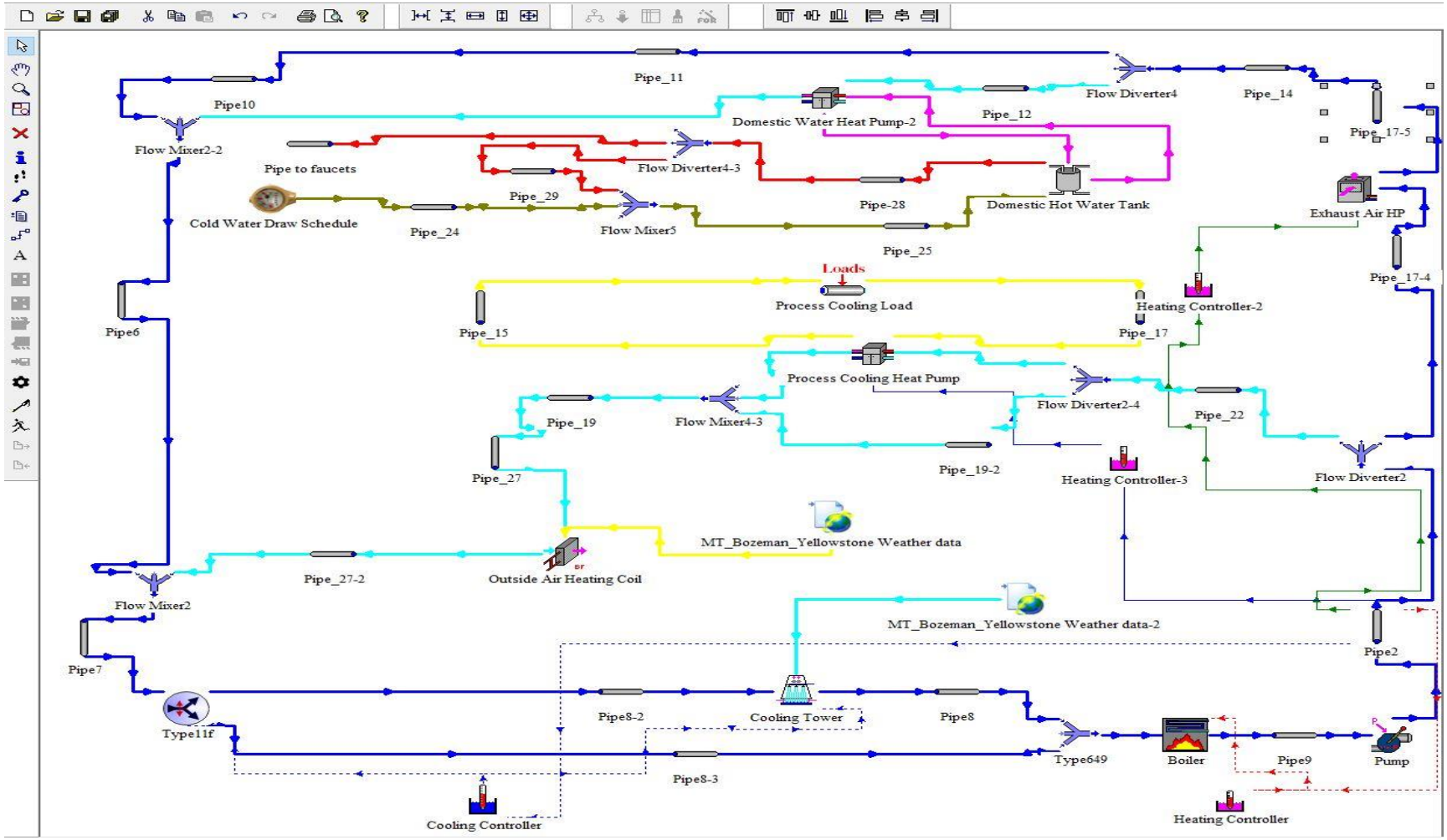


Figure 60. Heat Pump loop Design 5

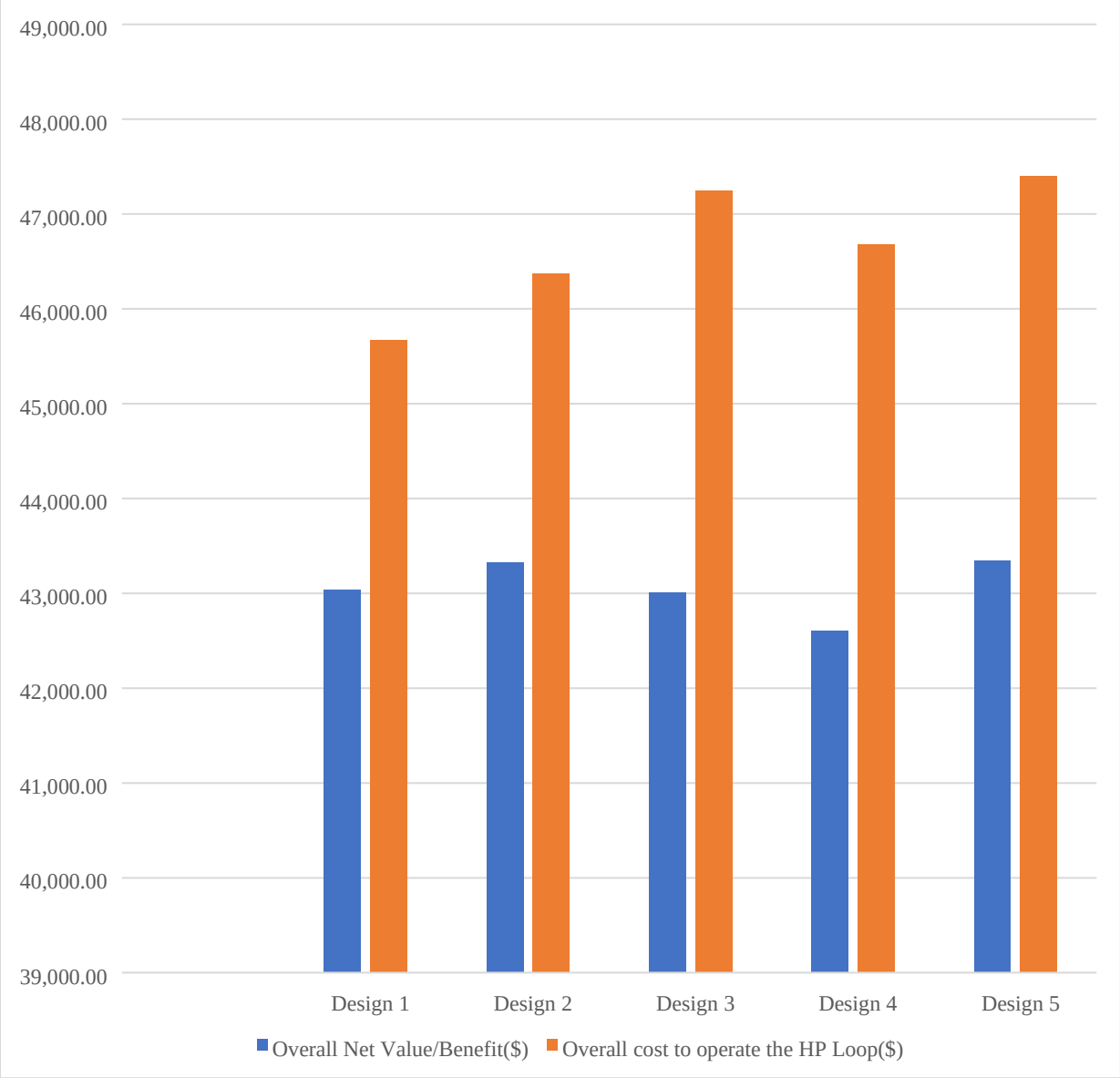


Figure 61. Bar Chart showing the net value and cost for the various designs

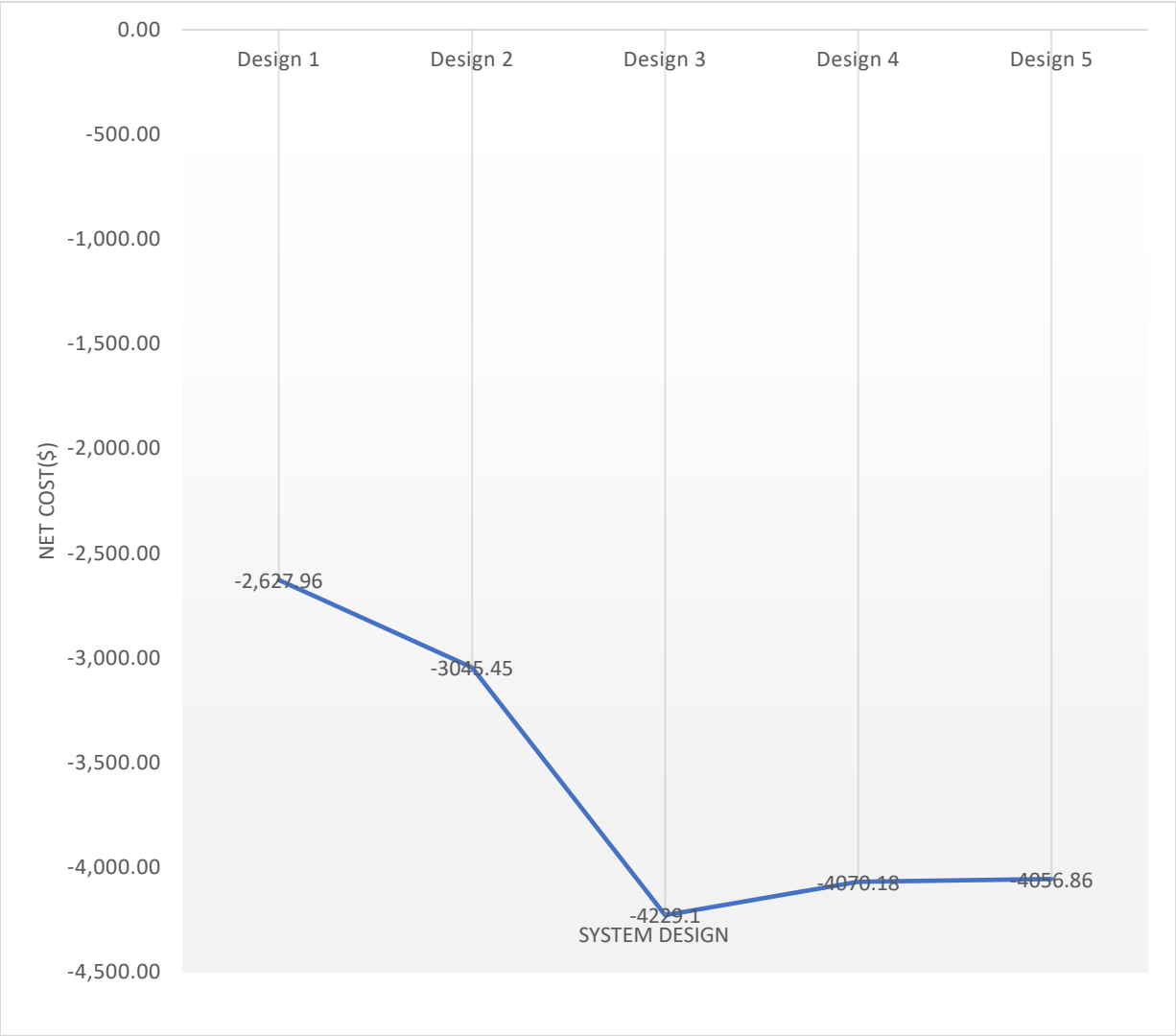


Figure 62. Net loss generated from the various designs

The graph shown above were preliminary results which showed a negative net value, after optimizing the system as discussed in the next page, a positive net value was gotten.

Calculation methodology used to arrive at the overall net value/benefits and cost to operate the heat pump loop were discussed in the next chapter under the economic analysis.

From the graphs above, design 1 was selected since it appeared the most efficient. The various controller set point temperatures were varied to see trends/patterns and to determine the setpoint temperatures that provided the most efficient.

The cooling and heating controller of the cooling tower and boiler was fixed at 90°F [the temperature above which cooling occurs] and 60°F [the temperature below which heating occurs] throughout the simulation. This was done to maintain the heat pump loop to be operating in the desired temperature range [60°F – 90°F]. Afterwards, the exhaust air heat pump controller temperature was fixed at 60°F, 65°F, 70°F, 75°F, 80°F, 85°F and 90°F with the process cooling heat pump temperature controller varied from 60°F – 90°F for each of the exhaust heat pump controller set point temperatures [temperatures below which the heat pump turns on and heating is commanded]. The overall value for each of this control points was calculated, and the graph plotted with MATLAB as seen in figure 63.

Note: The total pre-calculated heat capacity value was inputed into the domestic water heat pump rather than the total capacity being divided between the hot water tank which provides auxiliary heating and the heat pump as done in previous designs. This was done to ensure maximum efficiency from the domestic water heat pump and to provide ease when performing the economic analysis.

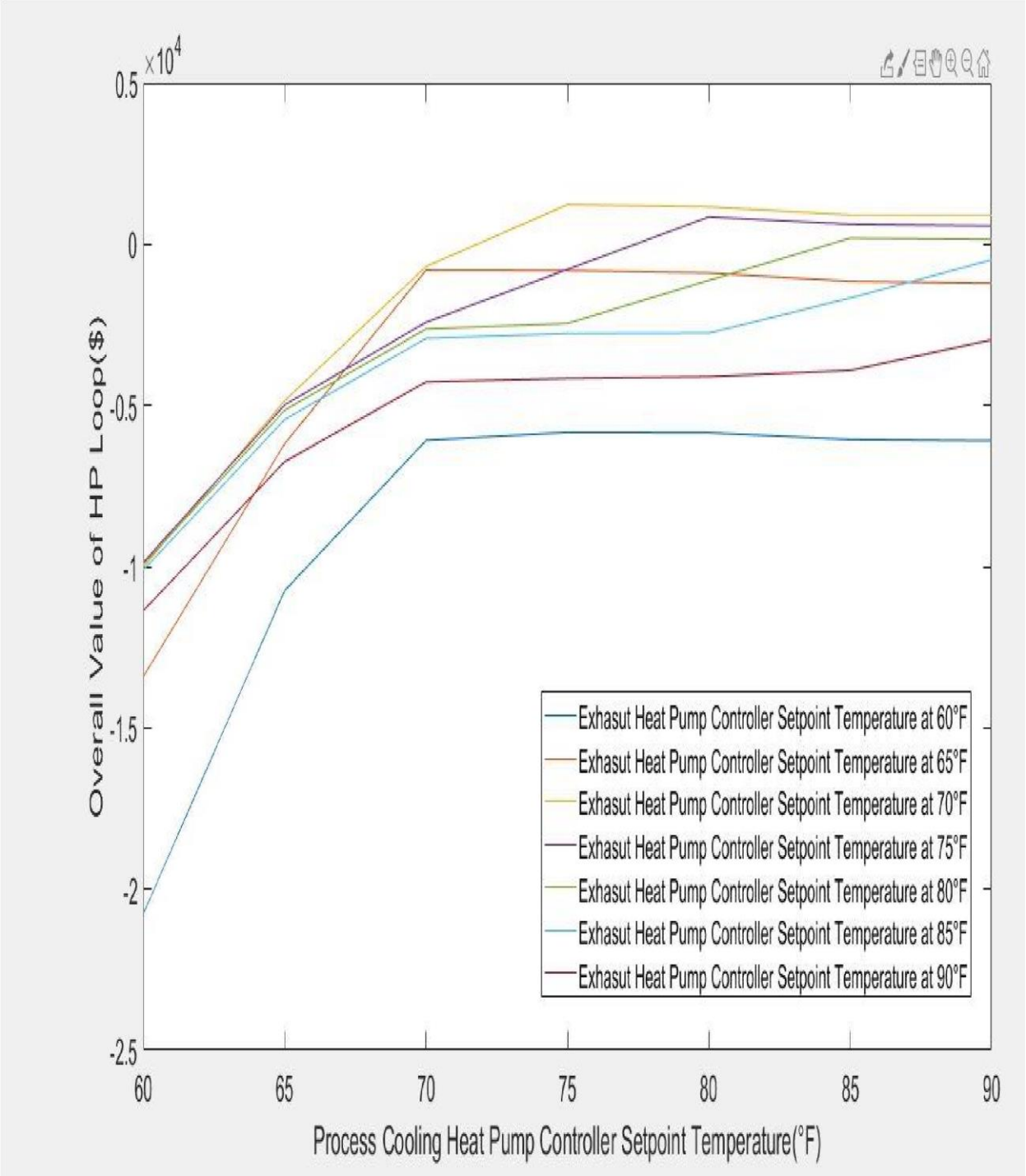


Figure 63. Process Cooling and Exhaust Air Heat Pump variations with the overall value of the Heat Pump loop

From the graph plotted, it is seen that the overall value/benefit of the heat pump loop increases as we increase the process cooling heat pump temperature and then stabilizes as we approach 90°F. Also, the most value obtained from the temperature variations occurs when the exhaust air heat pump controller is set at 70°F and the process cooling heat pump is set at 75°F with an overall value of $1225.77. Although an increase in value might still be obtained if the set point temperature of the controllers were increased a little bit, since the temperature variations currently done was in steps of 5°F.

As seen in the graph above, for the temperature variation in the exhaust air heat pump setpoint controller, the closer the controller set point temperature is to the building exhaust air temperature(room temperature;70°F), the more the increase in value of the heat pump loop. Thus, the exhaust air heat pump controller was set to 70°F while the process cooling heat pump controller was set to 75°F (the temperature below which heating is commanded).

Afterwards, design 1 was slightly modified to optimize it. The exhaust air heat pump was placed in parallel with other systems and various pumps were added to serve each system, this will enable the pumps to turn on and off according to the load demand, thus saving energy.

CHAPTER FIVE

SIMULATION RESULTS

The simulation results generated by the TRNSYS program for a time frame of 8760 hours are shown in the next few pages. The output generated includes the heat transfer, power consumption, leaving water temperatures, coefficient of performance alongside other parameters. This was done with the help of an online plotter attached to each component as seen in figure 65. Figure 64 shows the design of the heat pump loop used in generating the simulation result.

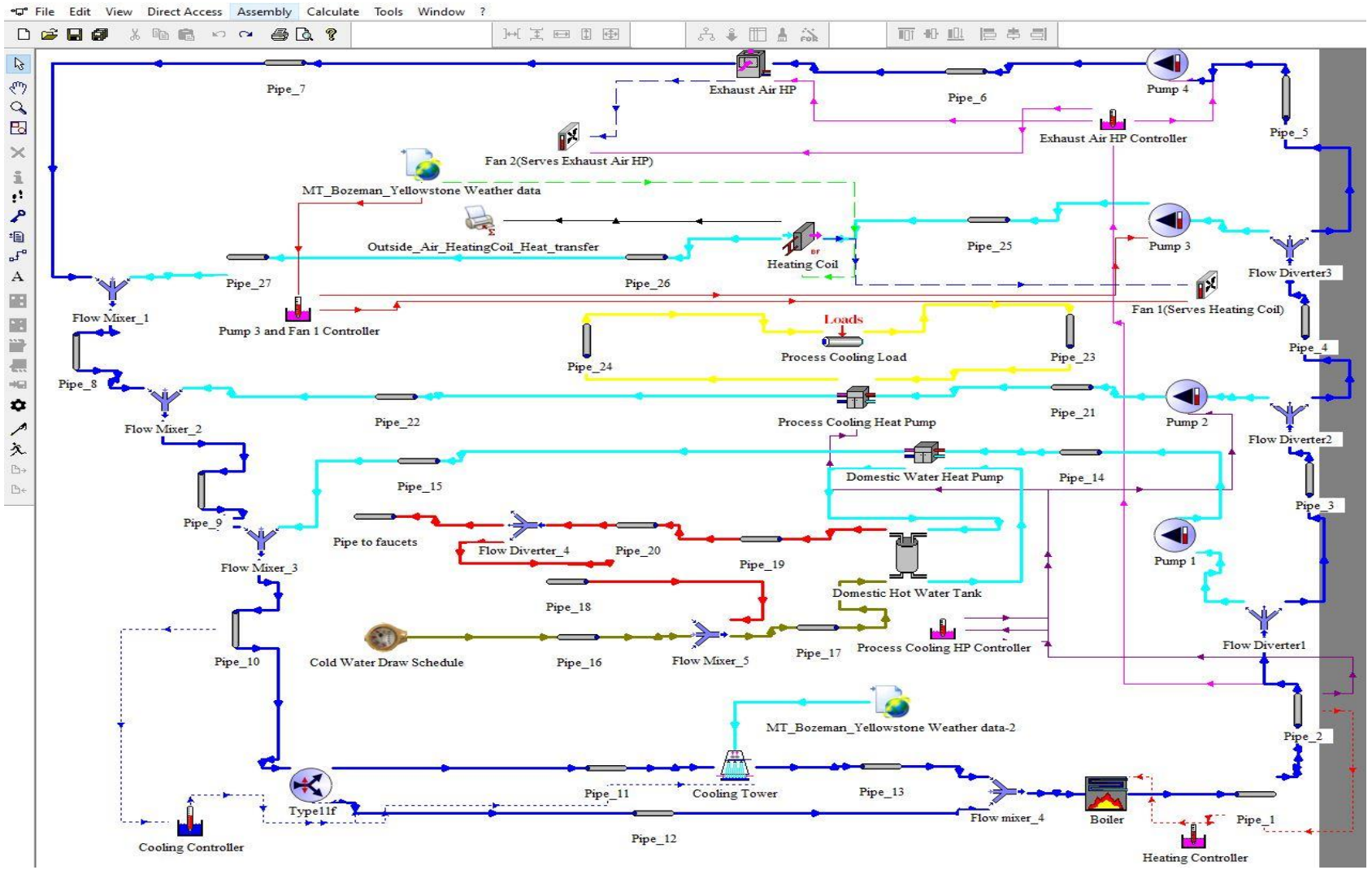


Figure 64. Final Heat Pump Loop Design

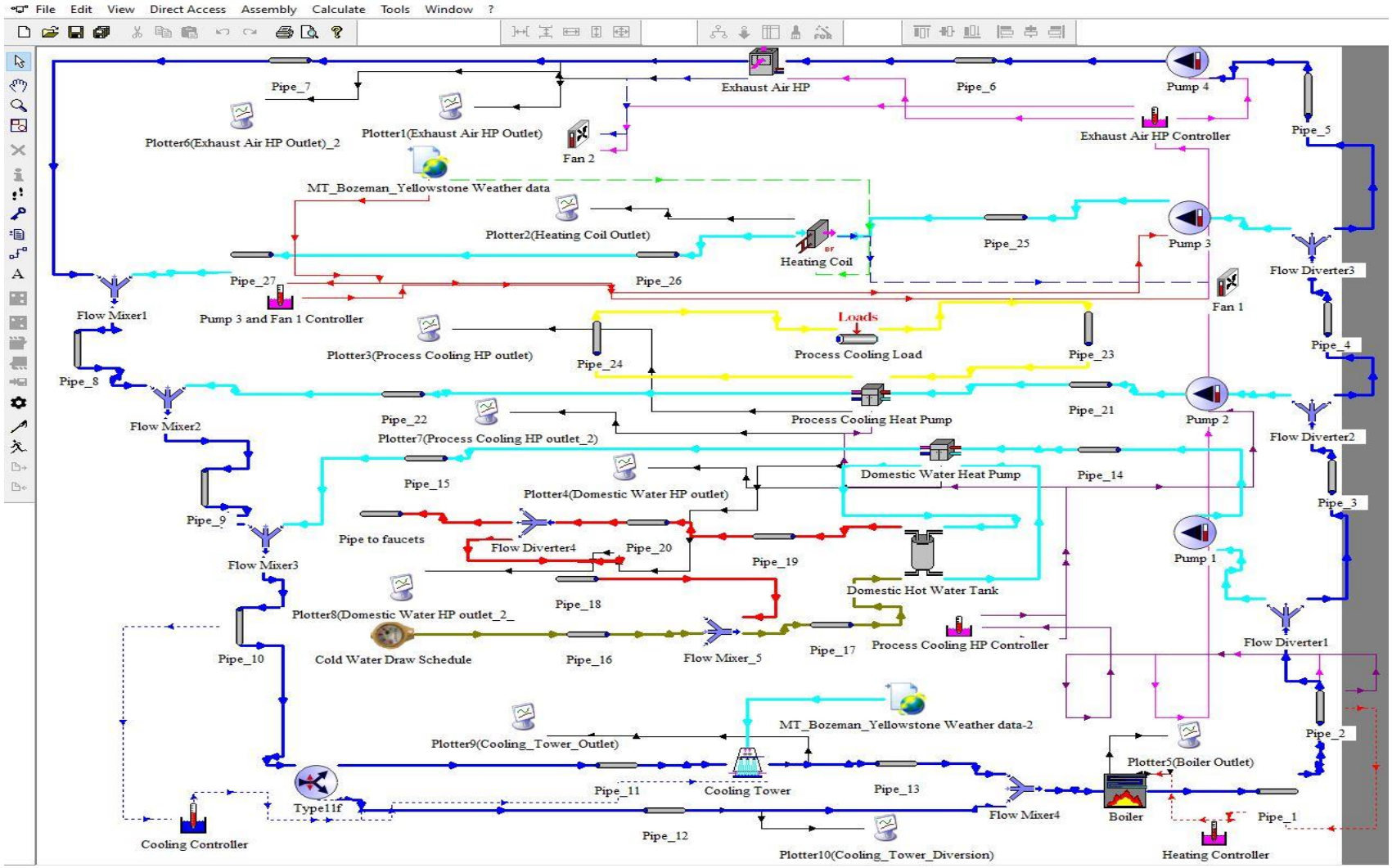


Figure 65. Final Heat Pump Loop Design with an Online Plotter

The simulation was computed for a period of 8760 hours and it was discovered initially that the cooling tower never rejected heat or came on throughout the simulation period as seen in figure 66, this is as a result of the heat pump loop temperature not exceeding the setpoint temperature [90°F] as seen in figure 67.

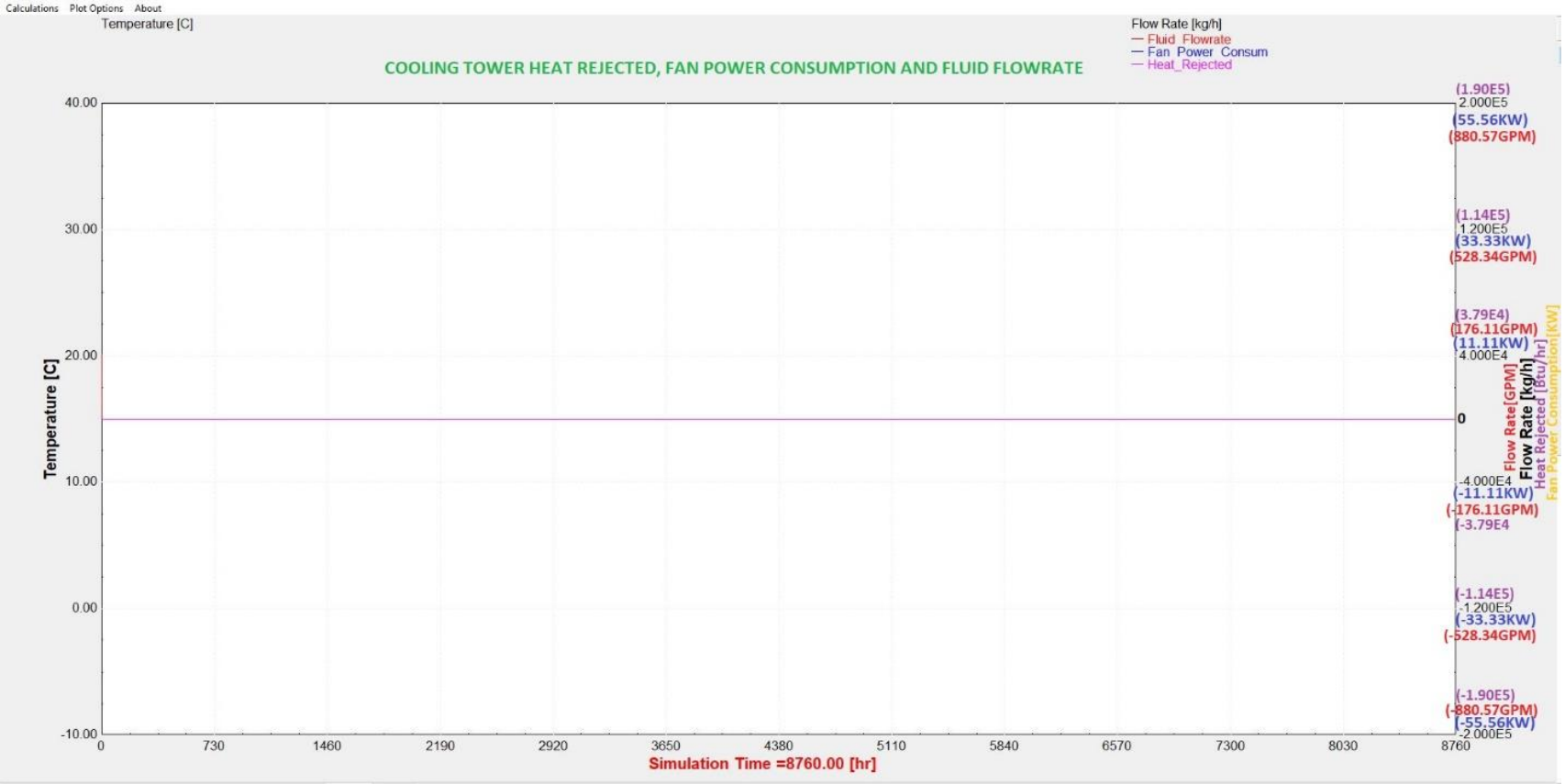
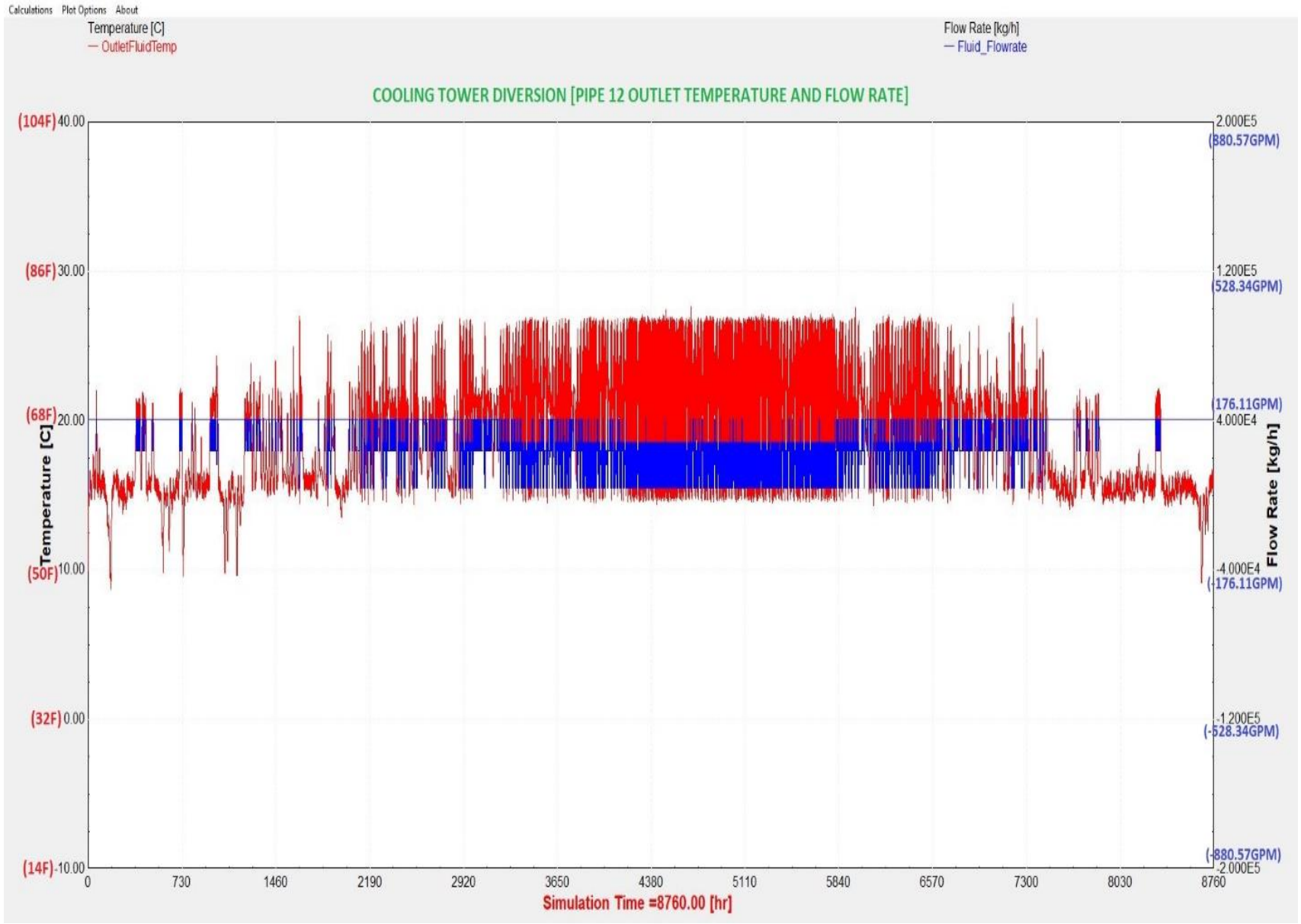


Figure 66. Cooling tower output simulation results


Calculations Plot Options About
Temperature [C]
OutletFluidTemp
Flow Rate [kg/h]
Fluid_Flowrate
COOLING TOWER DIVERSION [PIPE 12 OUTLET TEMPERATURE AND FLOW RATE]
(104F) 40.00
(86F) 30.00
(68F) 20.00
(50F) 10.00
(32F) 0.00
(14F) -10.00
Temperature [C]
2.000E5
(880.57GPM)
1.200E5
(528.34GPM)
(176.11GPM)
4.000E4
-4.000E4
(-176.11GPM)
-1.200E5
(-528.34GPM)
(-880.57GPM)
-2.000E5
Flow Rate [kg/h]
0
730
1460
2190
2920
3650
4380
5110
5840
6570
7300
8030
8760
Simulation Time =8760.00 [hr]

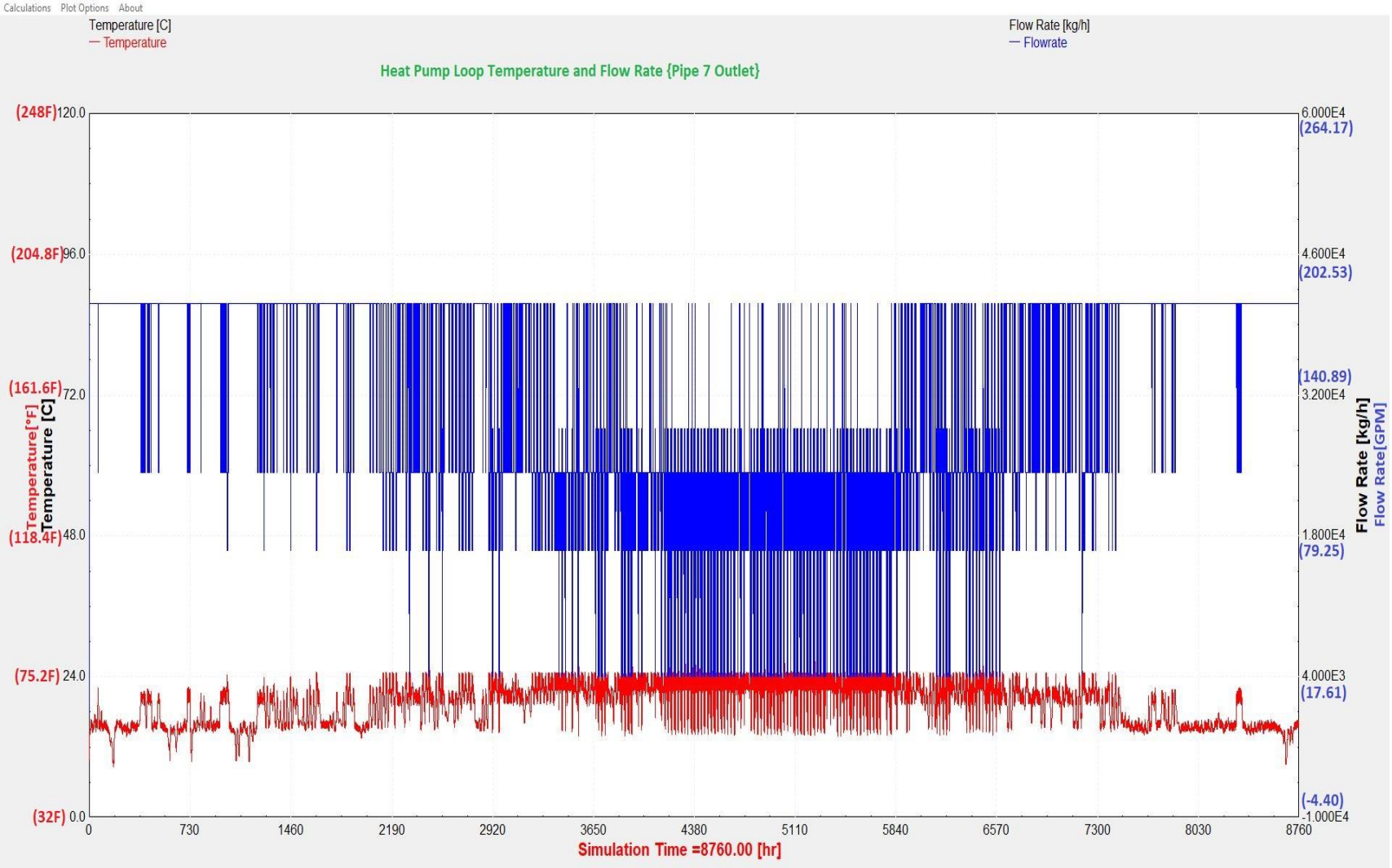


Figure 67. Heat Pump Loop Temperature and Flow Rate

Afterwards, the cooling tower was removed from the design since it never came on when the simulation was computed initially due to the set point temperature not exceeded. The new design without the cooling tower was simulated as seen in figure 68 with the simulation results shown in the next pages.

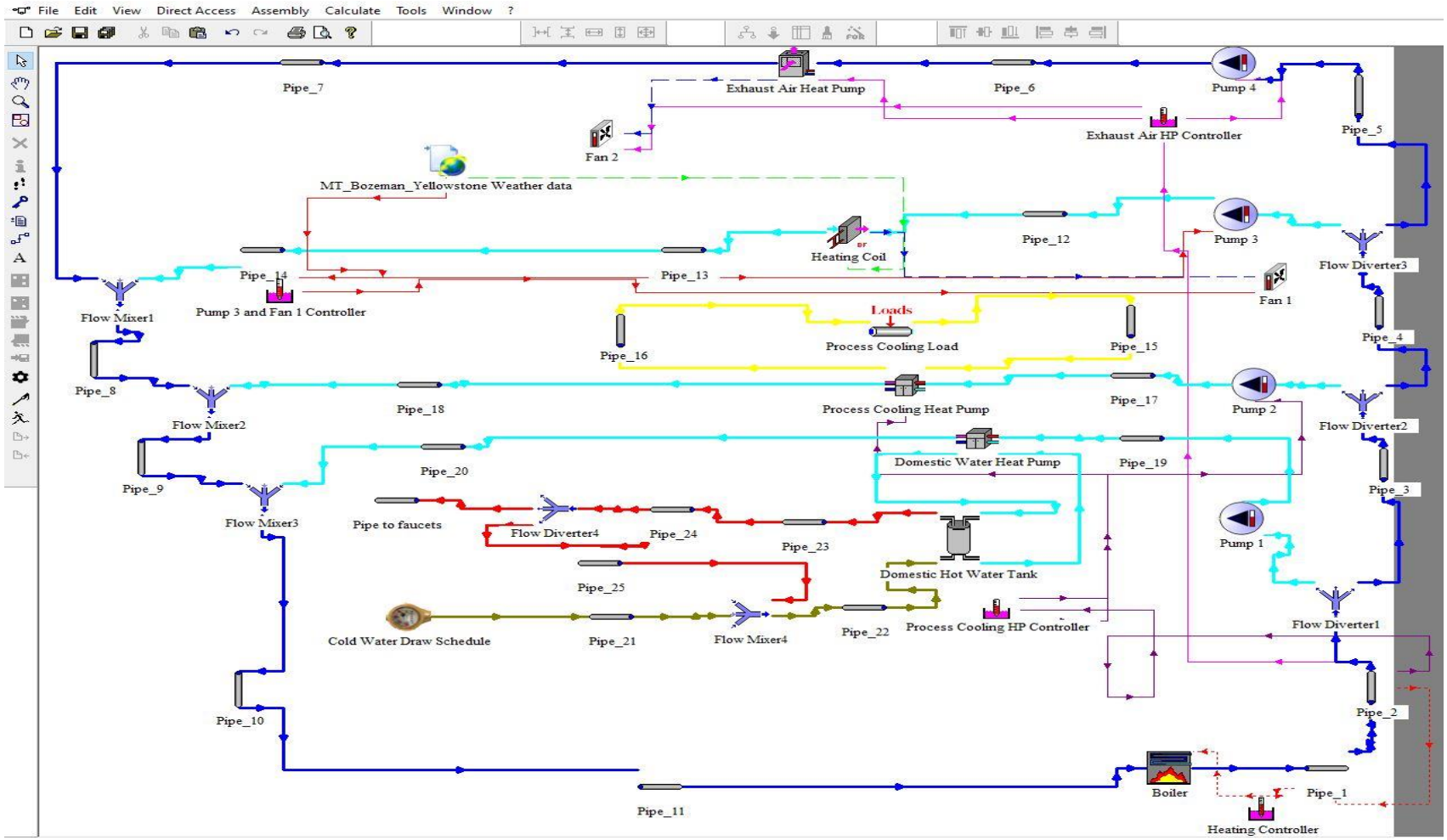
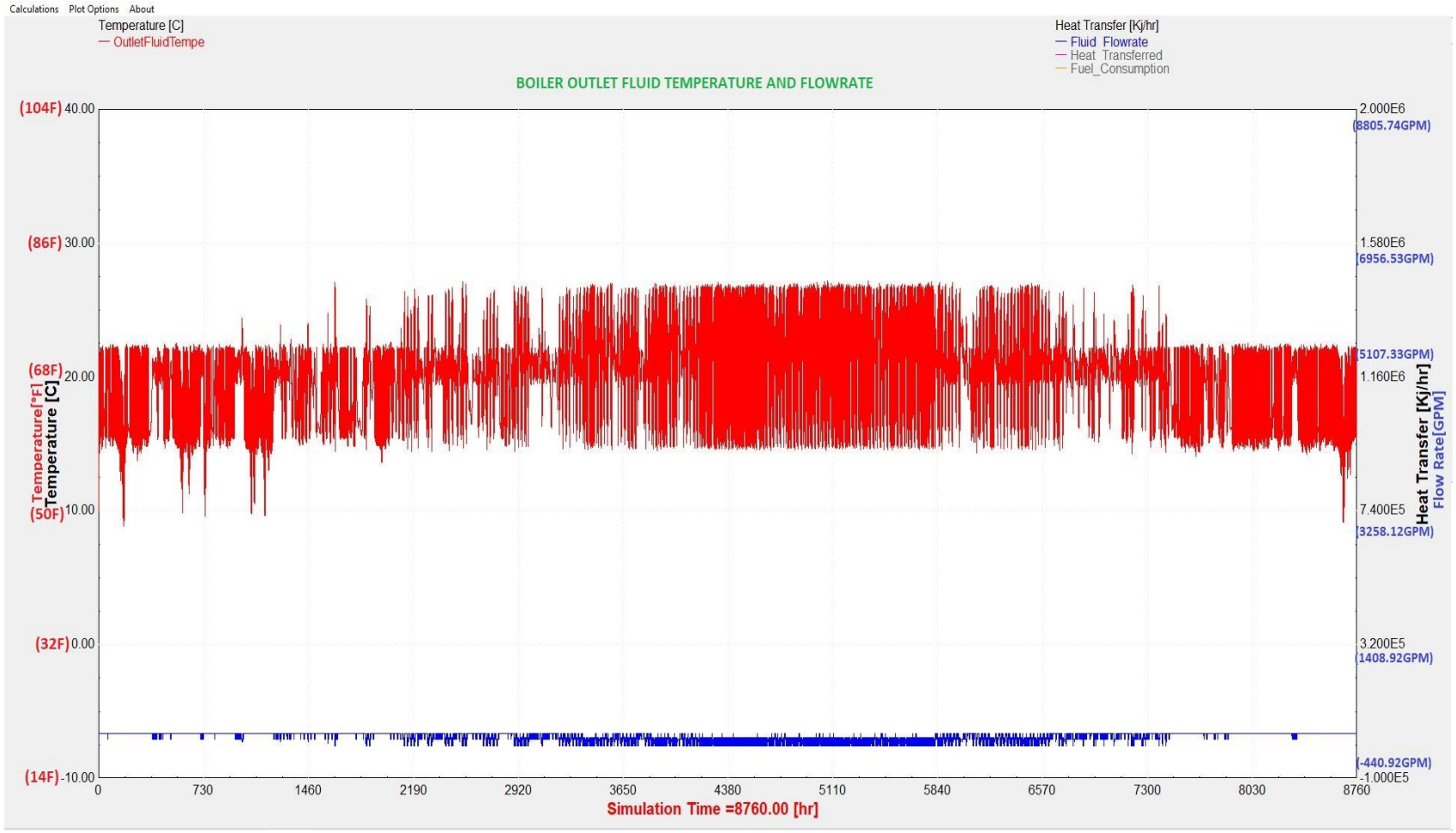
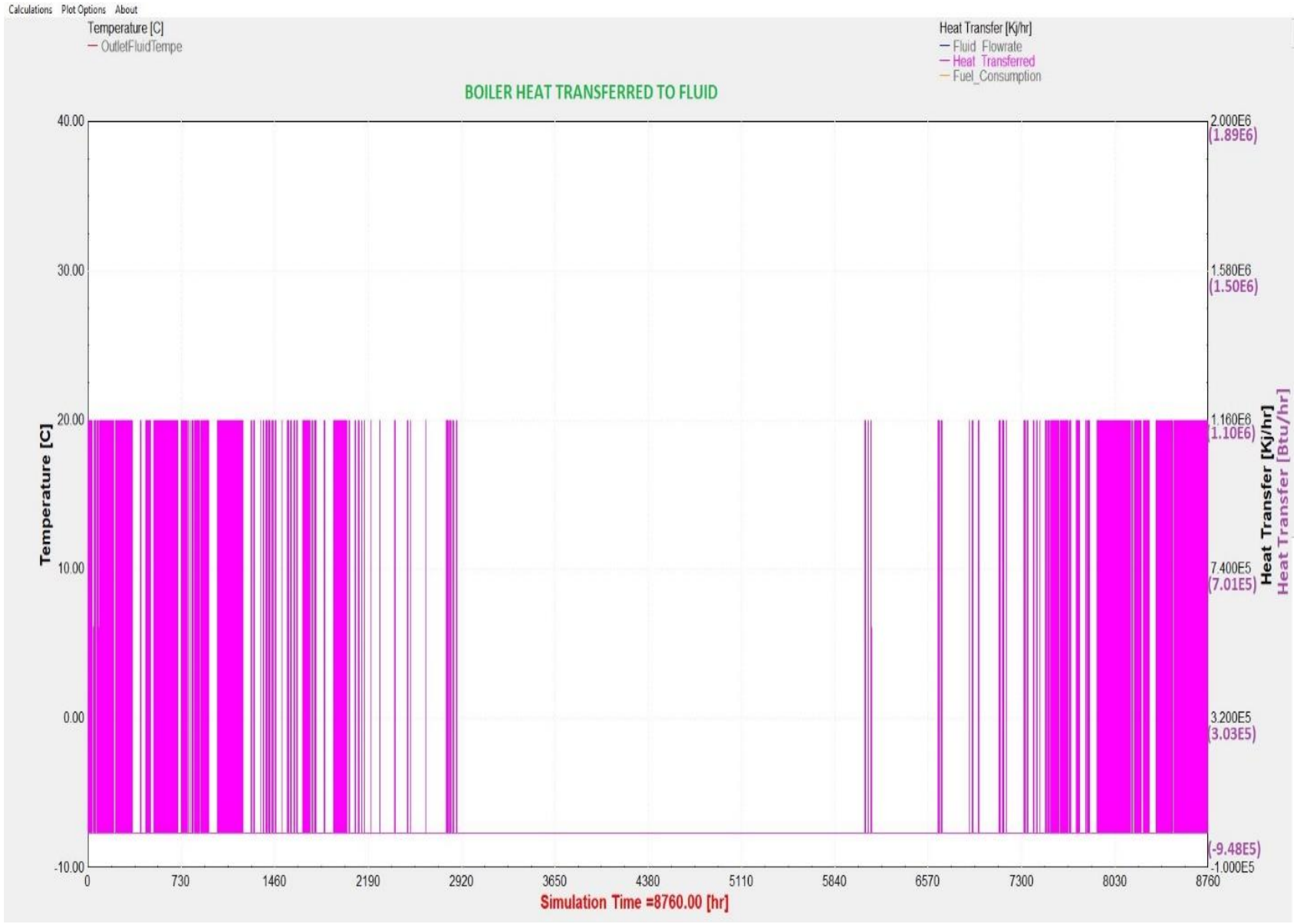


Figure 68. Heat pump loop design without the cooling tower

Detailed simulation results of each system are shown below;

(i) Boiler




Calculations Plot Options About
Temperature [C]
OutletFluidTempe
Heat Transfer [Kj/hr]
Fluid Flowrate
Heat Transferred
Fuel_Consumption
BOILER HEAT TRANSFERRED TO FLUID
40.00
30.00
20.00
10.00
0.00
-10.00
Temperature [C]
2.000E6
(1.89E6)
1.580E6
(1.50E6)
1.160E6
(1.10E6)
7.400E5
(7.01E5)
3.200E5
(3.03E5)
(-9.48E5)
-1.000E5
Heat Transfer [Kj/hr]
Heat Transfer [Btu/hr]
0
730
1460
2190
2920
3650
4380
5110
5840
6570
7300
8030
8760
Simulation Time =8760.00 [hr]

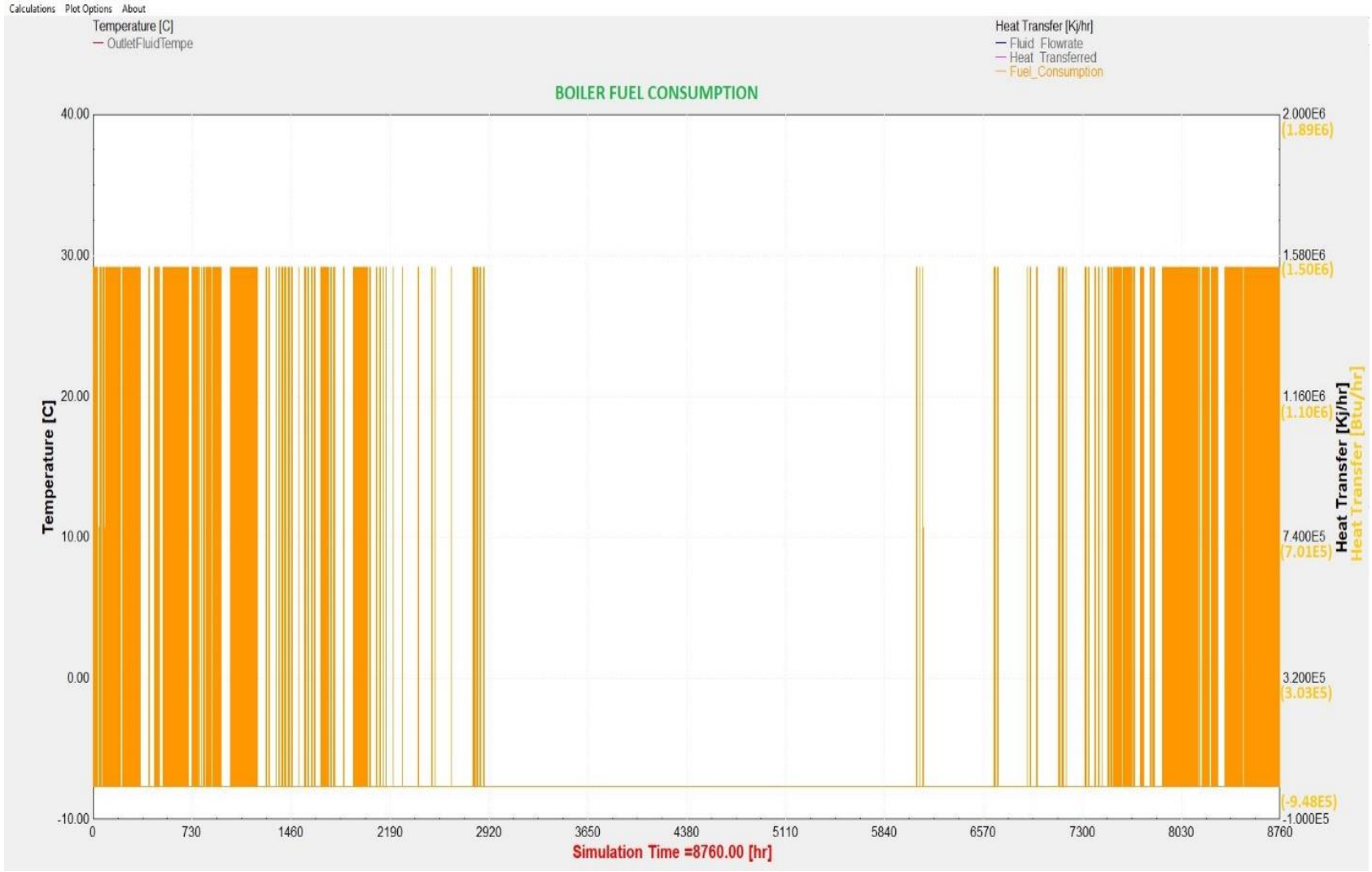
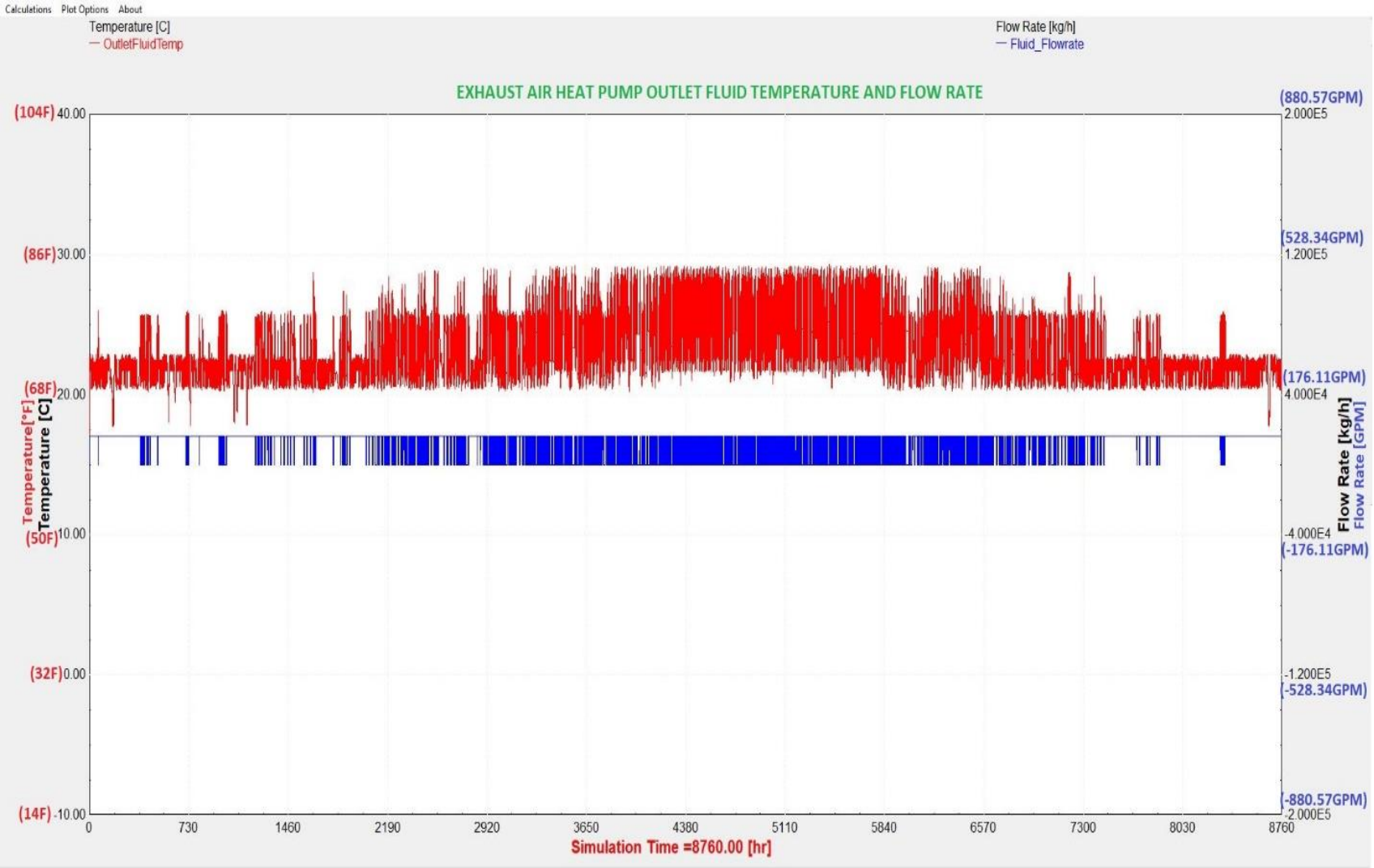
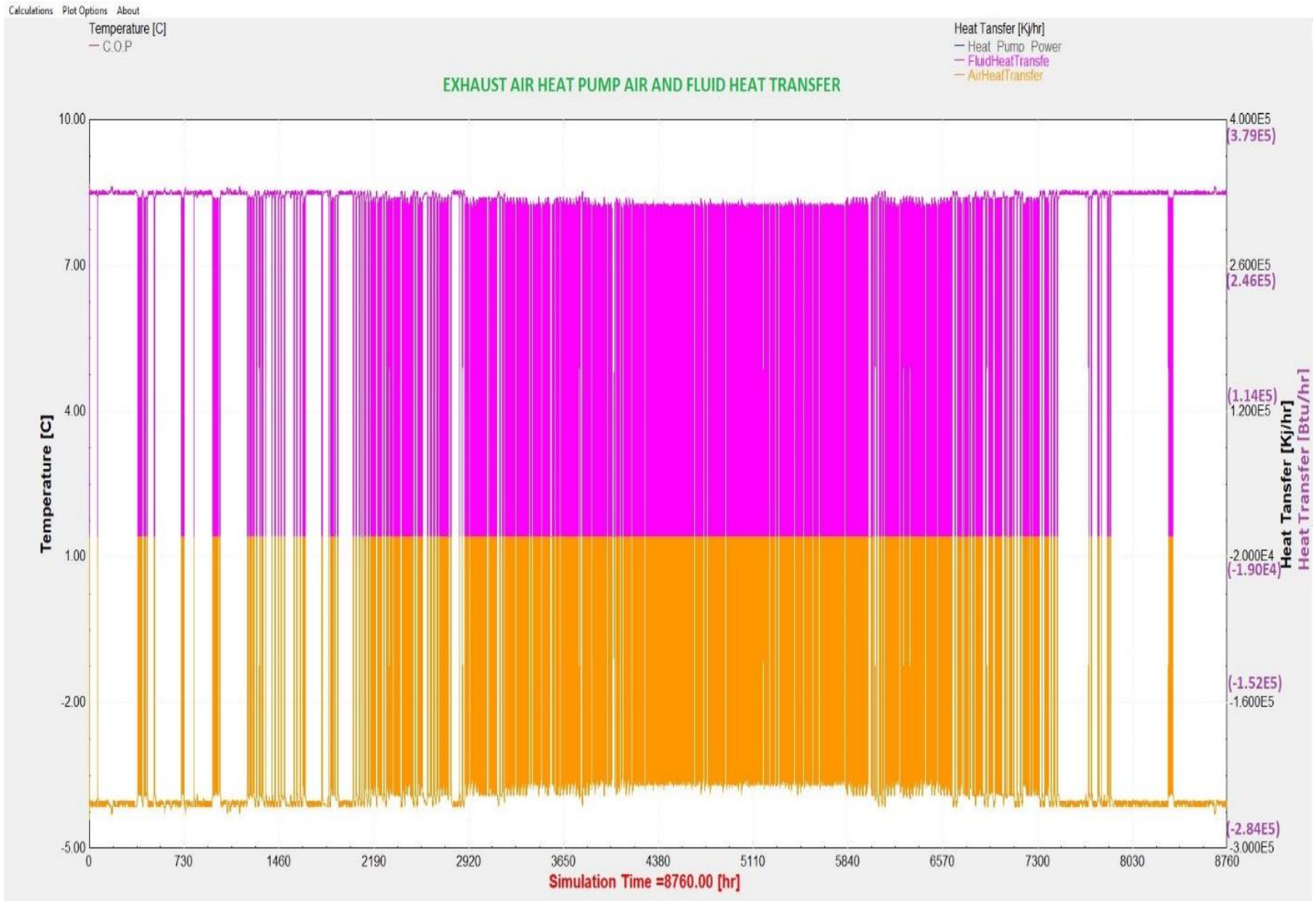


Figure 69. Boiler output simulation results

The boiler outlet fluid temperature in the heat pump loop can be seen to be majorly within the temperature range of 60°F- 90°F. The minimum outlet fluid temperature can be seen to be about 60°F since that was the controller setpoint temperature imputed. i.e. the temperature below which the boiler comes on. Also, from the boiler heat transferred and fuel consumption output, the boiler is seen to be off from May till August. This is expected due to moderate weather temperatures and less heating demand.

(ii) Exhaust Air Heat Pump




Calculations Plot Options About
Temperature [C]
C.O.P
Heat Tansfer [Kj/hr]
Heat Pump Power
FluidHeatTransfe
AirHeatTransfer
EXHAUST AIR HEAT PUMP AIR AND FLUID HEAT TRANSFER
10.00
7.00
4.00
1.00
-2.00
-5.00
Temperature [C]
4.000E5
(3.79E5)
2.600E5
(2.46E5)
(1.14E5)
1.200E5
-2.000E4
(-1.90E4)
(-1.52E5)
-1.600E5
(-2.84E5)
-3.000E5
Heat Tansfer [Kj/hr]
Heat Transfer [Btu/hr]
0
730
1460
2190
2920
3650
4380
5110
5840
6570
7300
8030
8760
Simulation Time =8760.00 [hr]

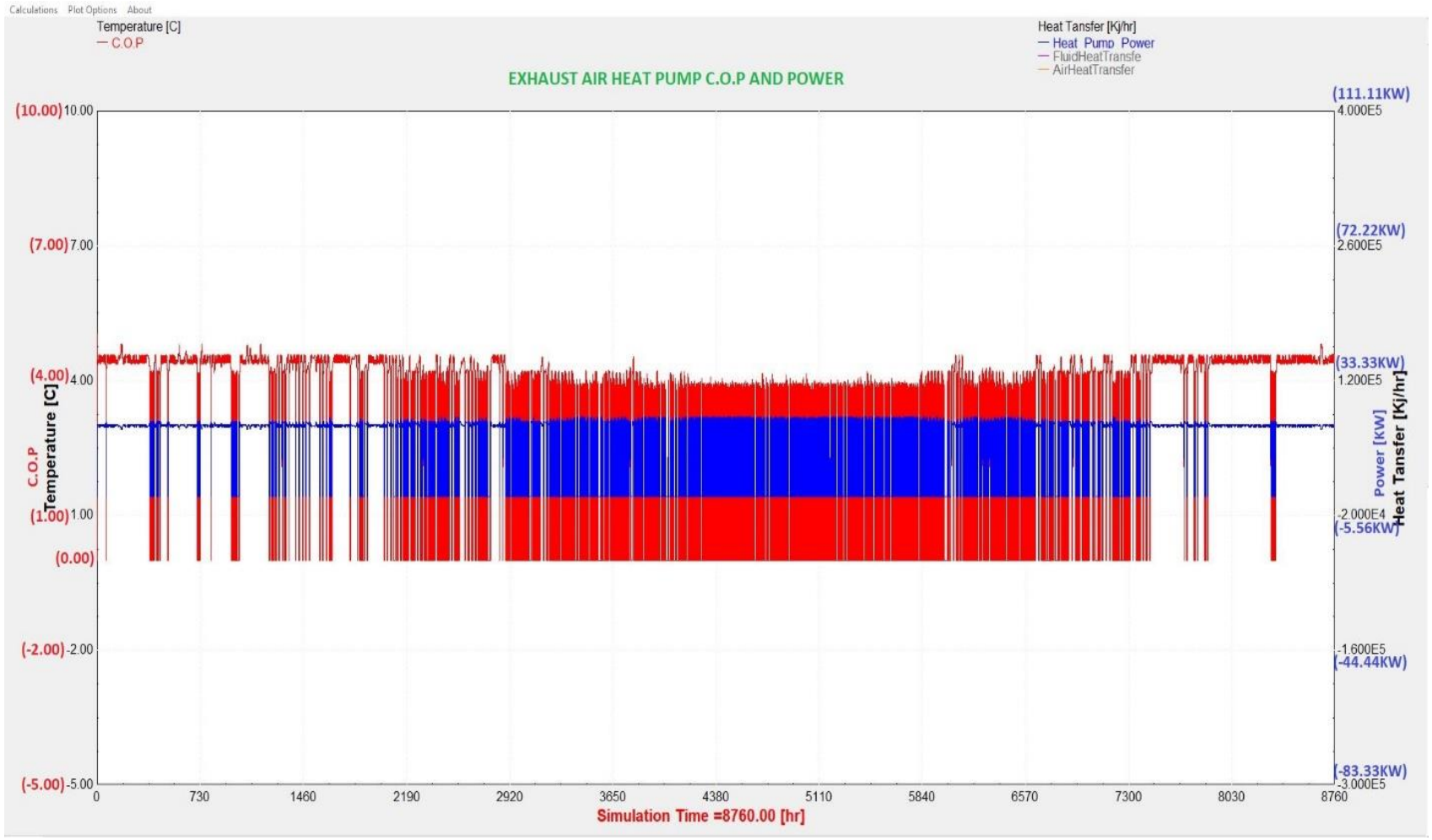
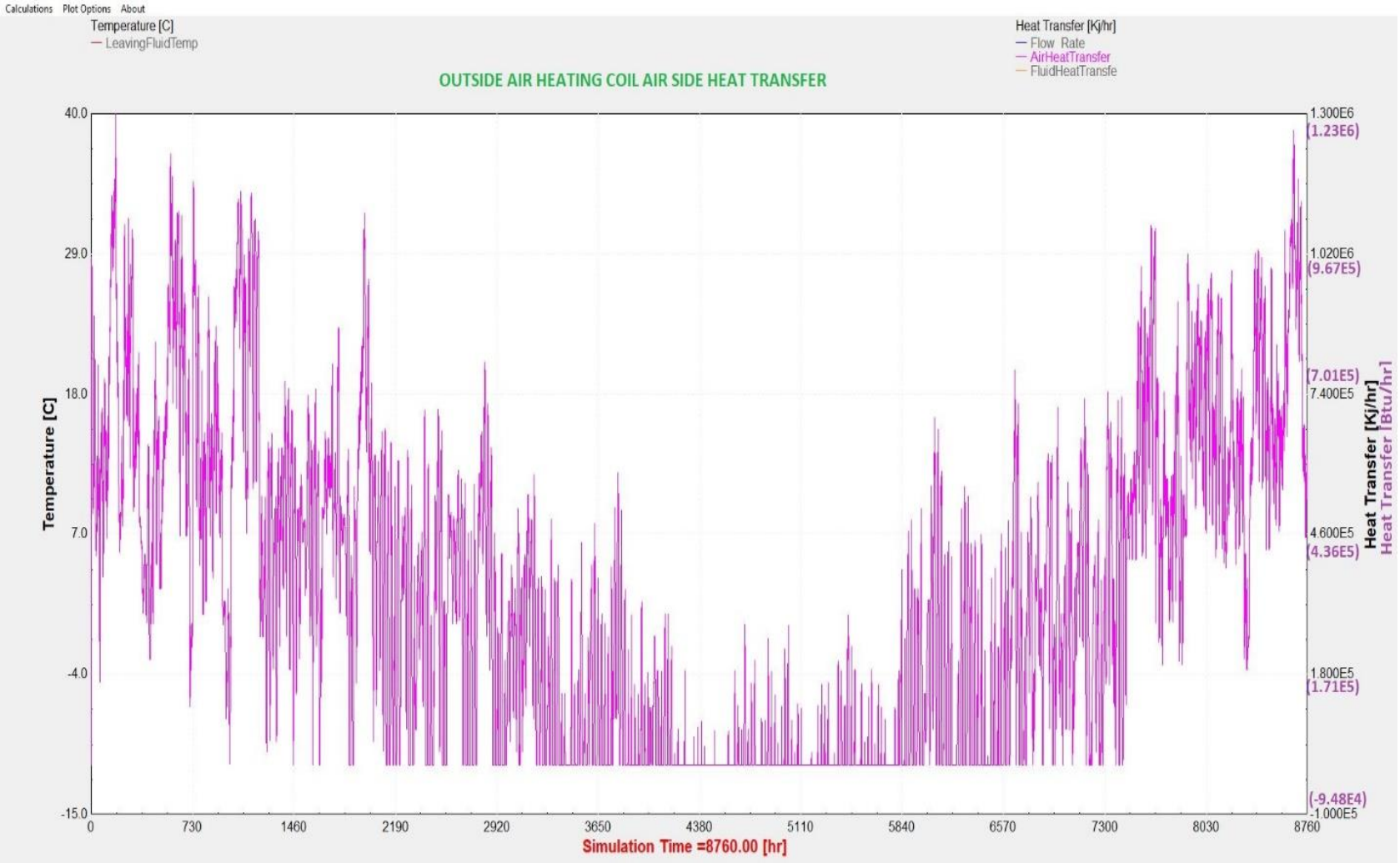
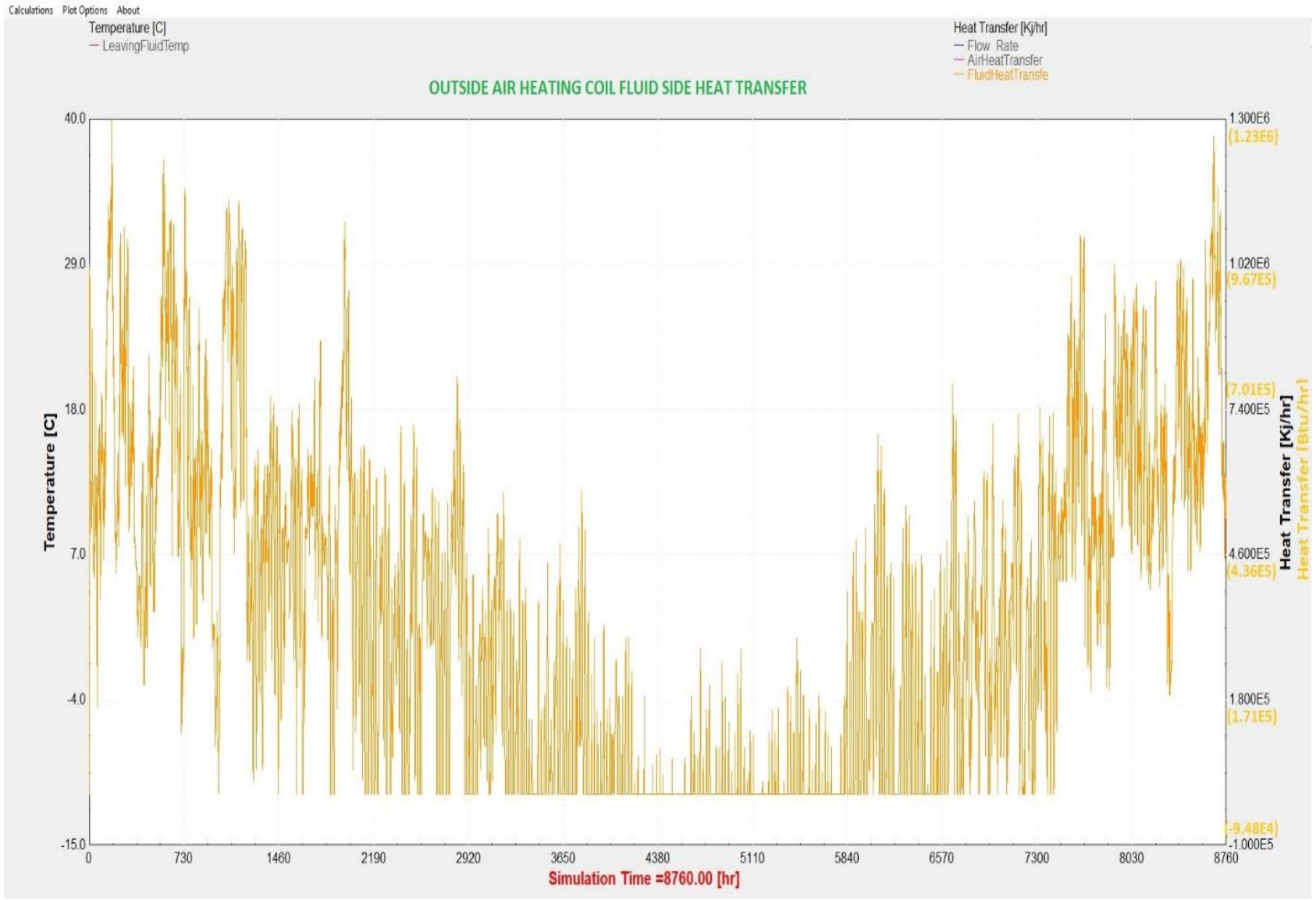


Figure 70. Exhaust Air Heat Pump simulation results

The exhaust air heat pump's outlet fluid temperature also is within the heat pump temperature range [60°F -90°F]. Heat energy can be seen to be transferred from the exhaust air to the fluid side of the heat pump. The heat pump maintains a COP majorly in the range of 4.0 - 4.5 as seen in the diagram above.

(iii) Outdoor Air Heating Coil




Calculations Plot Options About
Temperature [C]
LeavingFluidTemp
Heat Transfer [Kj/hr]
Flow Rate
AirHeatTransfer
FluidHeatTransfe
OUTSIDE AIR HEATING COIL FLUID SIDE HEAT TRANSFER
Temperature [C]
40.0
29.0
18.0
7.0
-4.0
-15.0
Heat Transfer [Kj/hr]
Heat Transfer [Btu/hr]
1.300E6
(1.23E6)
1.020E6
(9.67E5)
(7.01E5)
7.400E5
4.600E5
(4.36E5)
1.800E5
(1.71E5)
(-9.48E4)
-1.000E5
0
730
1460
2190
2920
3650
4380
5110
5840
6570
7300
8030
8760
Simulation Time =8760.00 [hr]

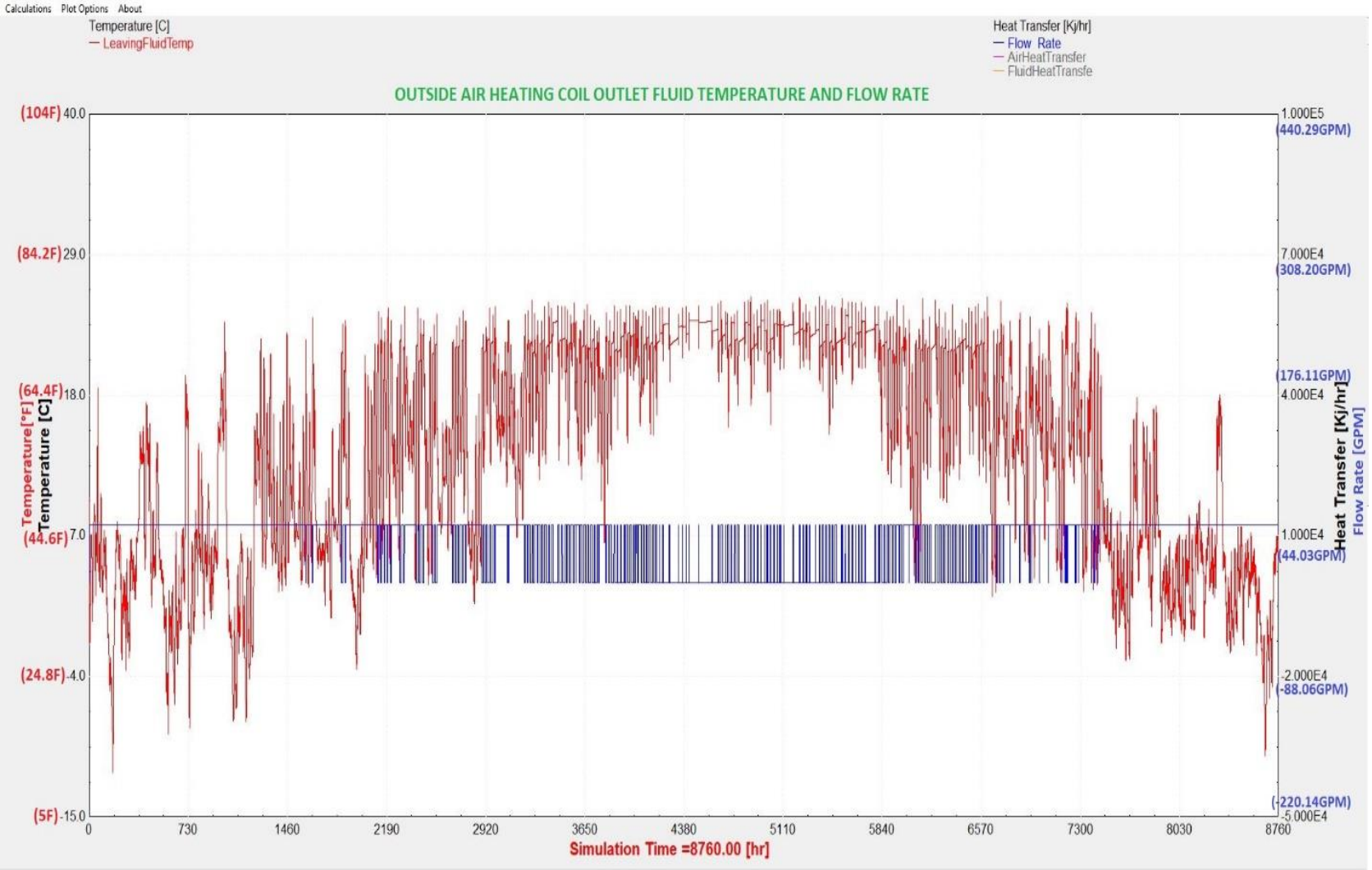
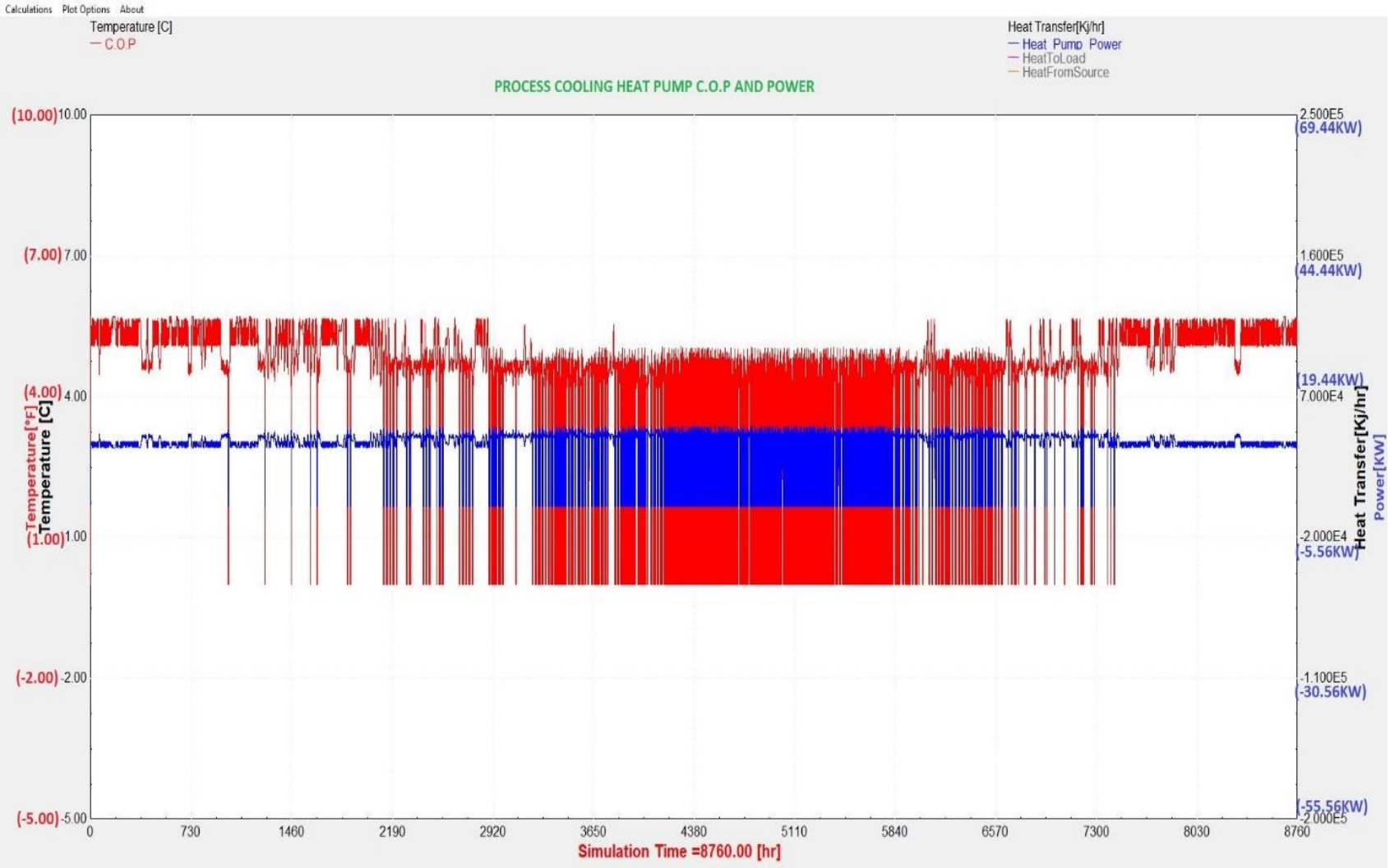
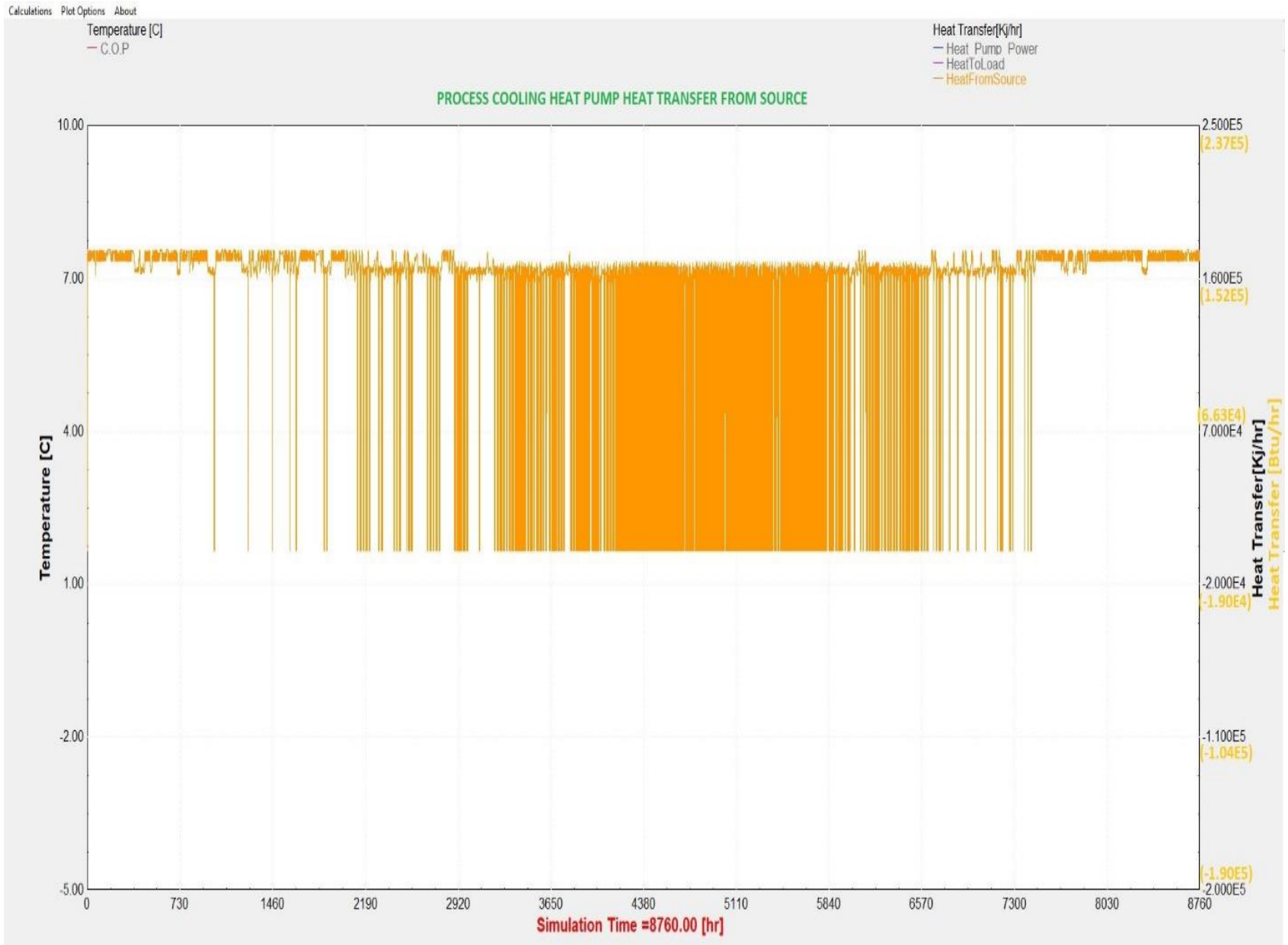


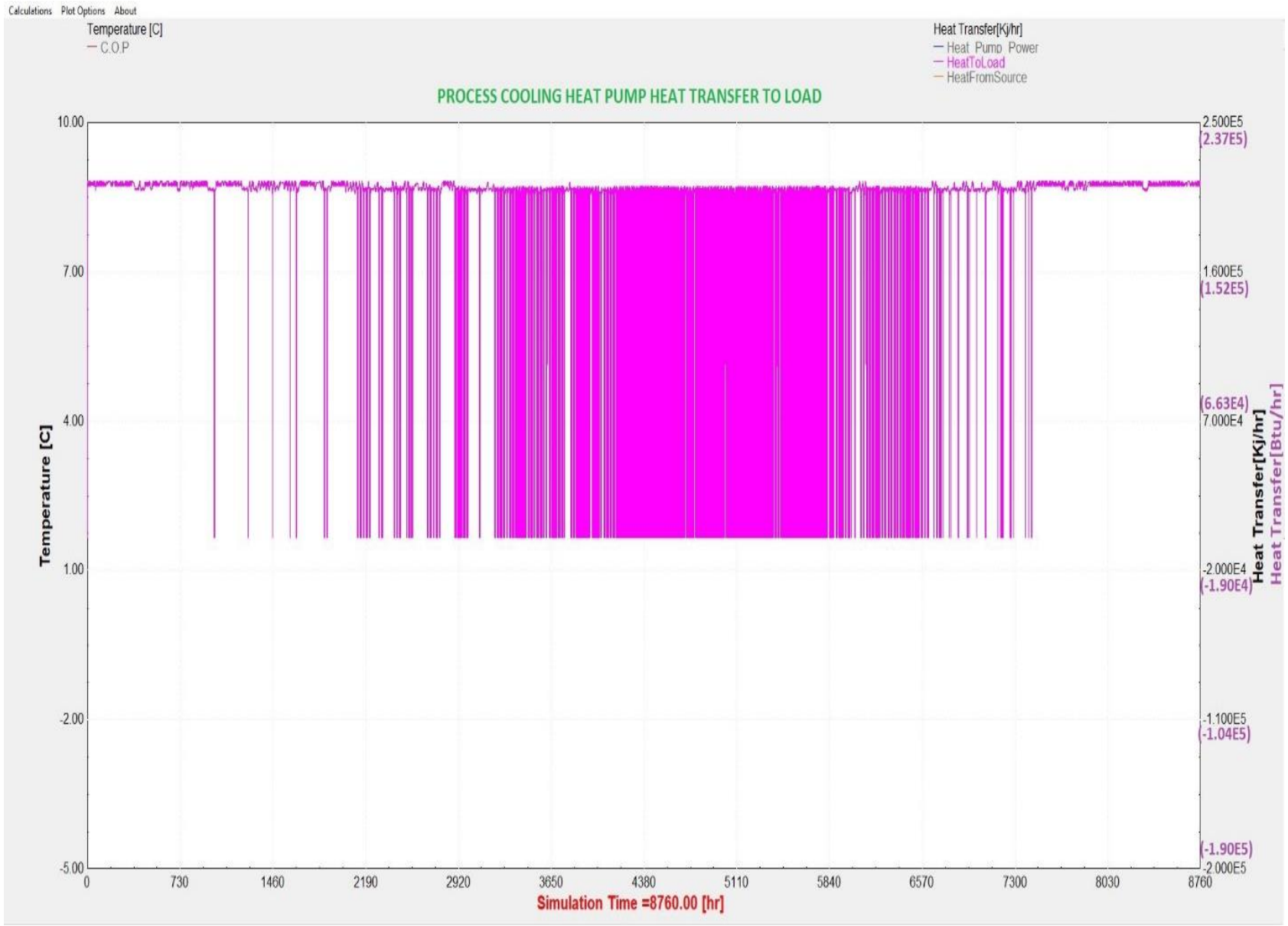
Figure 71. Outdoor Air Heating Coil simulation results

The heating coil's fluid leaving temperature can be seen to have low temperatures from November to January. This is due to the extreme cold weather temperatures that occur in Bozeman during that time of the year, hence, an anti-freeze additive will be required to be added in the fluids to prevent the pipes from freezing during the winter. Also, the heat transfer from the fluid can be seen equal in value to the heat transfer to the air, this is expected since it is a heating coil.

(iv) Process Cooling Heat Pump




Calculations Plot Options About
Temperature [C]
C.O.P
Heat Transfer[Kj/hr]
Heat Pump Power
HeatToLoad
HeatFromSource
PROCESS COOLING HEAT PUMP HEAT TRANSFER FROM SOURCE
10.00
7.00
4.00
1.00
-2.00
-5.00
Temperature [C]
2.500E5
(2.37E5)
1.600E5
(1.52E5)
(6.63E4)
7.000E4
-2.000E4
(-1.90E4)
-1.100E5
(-1.04E5)
(-1.90E5)
-2.000E5
Heat Transfer[Kj/hr]
Heat Transfer [Btu/hr]
0
730
1460
2190
2920
3650
4380
5110
5840
6570
7300
8030
8760
Simulation Time =8760.00 [hr]


Calculations Plot Options About
Temperature [C]
C.O.P
Heat Transfer[Kj/hr]
Heat Pump Power
HeatToLoad
HeatFromSource
PROCESS COOLING HEAT PUMP HEAT TRANSFER TO LOAD
10.00
7.00
4.00
1.00
-2.00
-5.00
Temperature [C]
2.500E5
(2.37E5)
1.600E5
(1.52E5)
(6.63E4)
7.000E4
-2.000E4
(-1.90E4)
-1.100E5
(-1.04E5)
(-1.90E5)
-2.000E5
Heat Transfer[Kj/hr]
Heat Transfer[Btu/hr]
0
730
1460
2190
2920
3650
4380
5110
5840
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Simulation Time =8760.00 [hr]

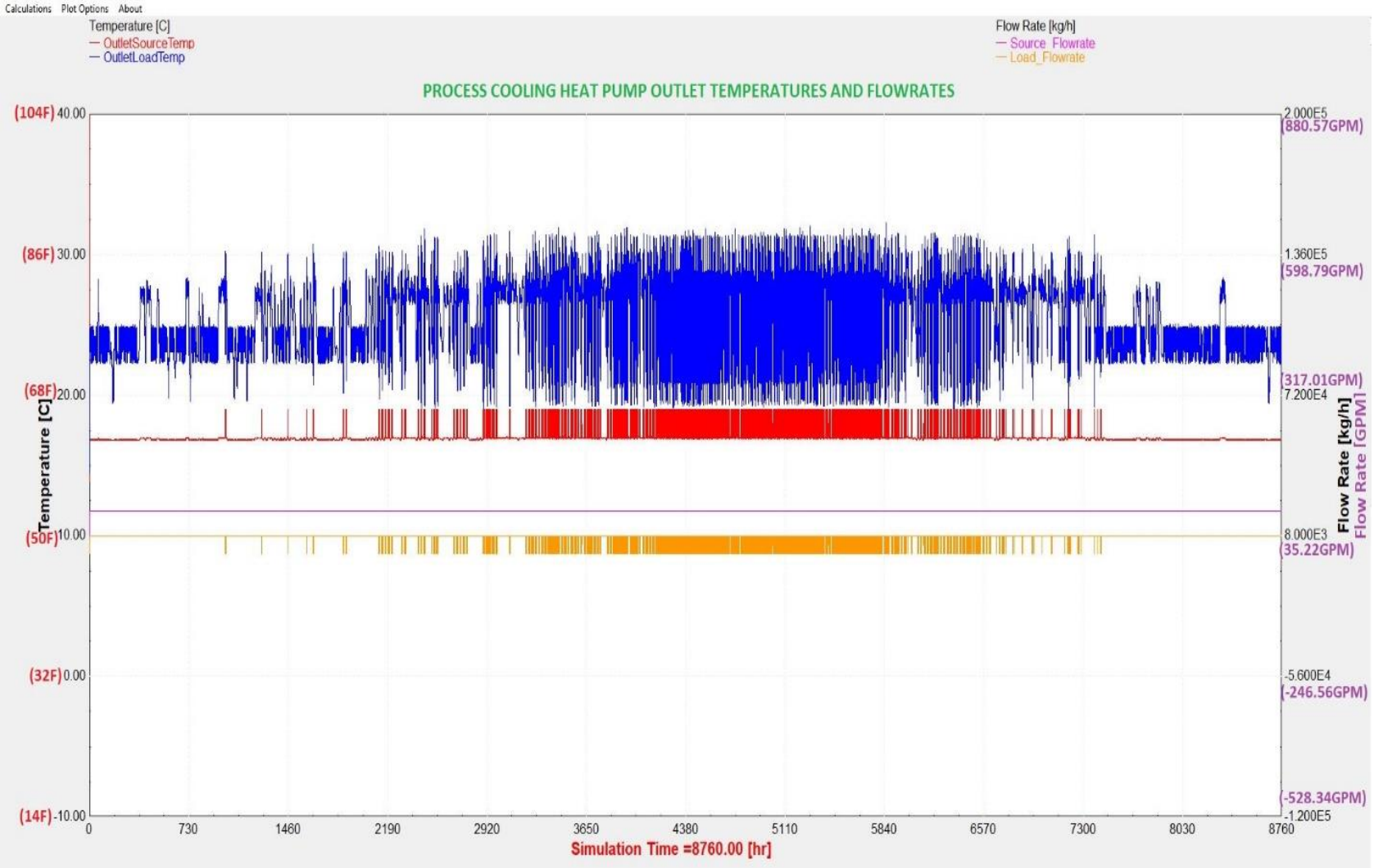
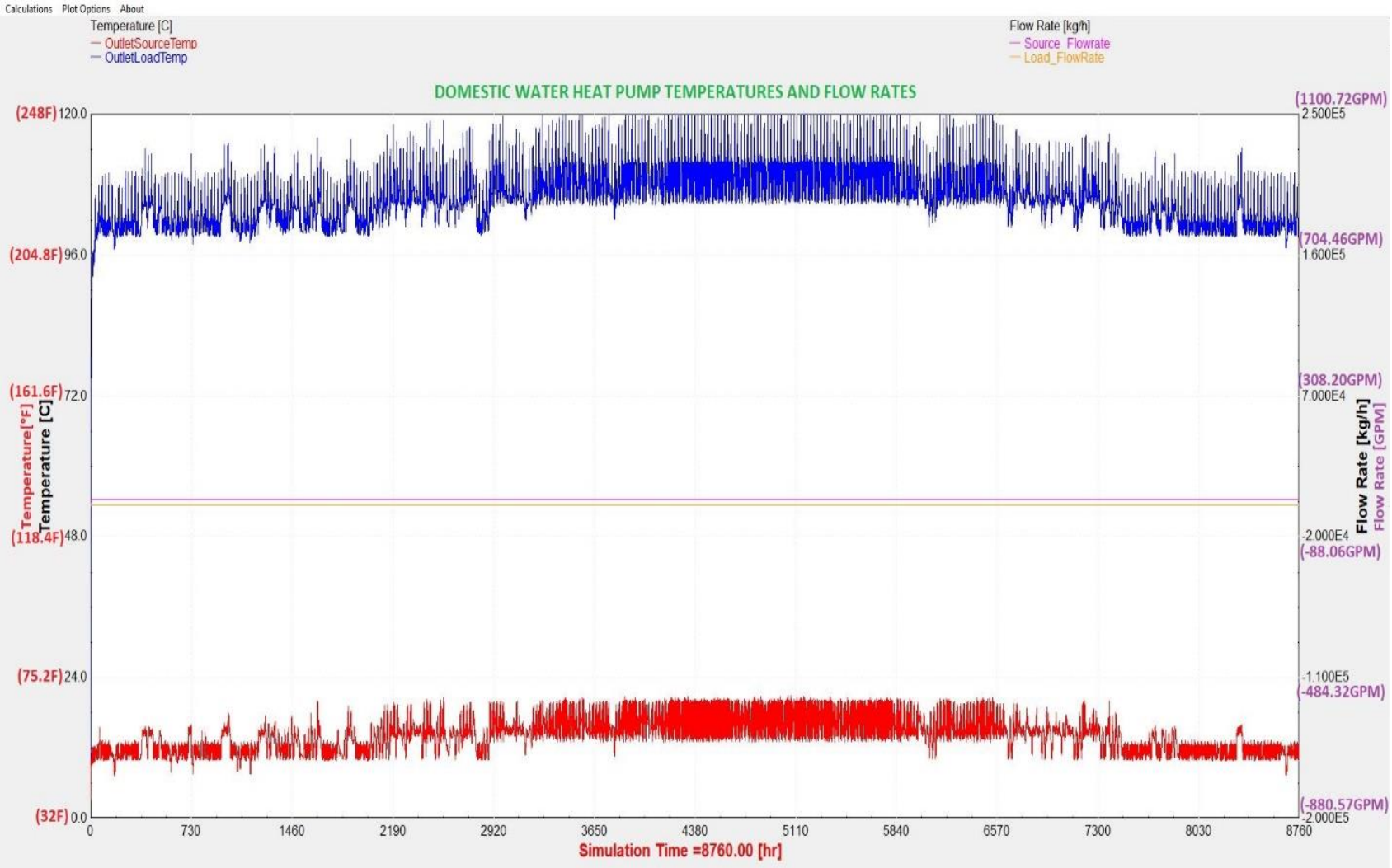


Figure 72. Process Cooling Heat Pump simulation results

The process cooling heat pump outlet leaving temperatures also is within the heat pump temperature range [60°F -90°F]. For the process cooling heat pump, the load side represents the heat pump loop while the source side represent the process cooling load or well. The heat pump maintains a COP majorly in the range of 4.5 - 5.5 as seen in the diagram above.

(v) Domestic Water Heat Pump



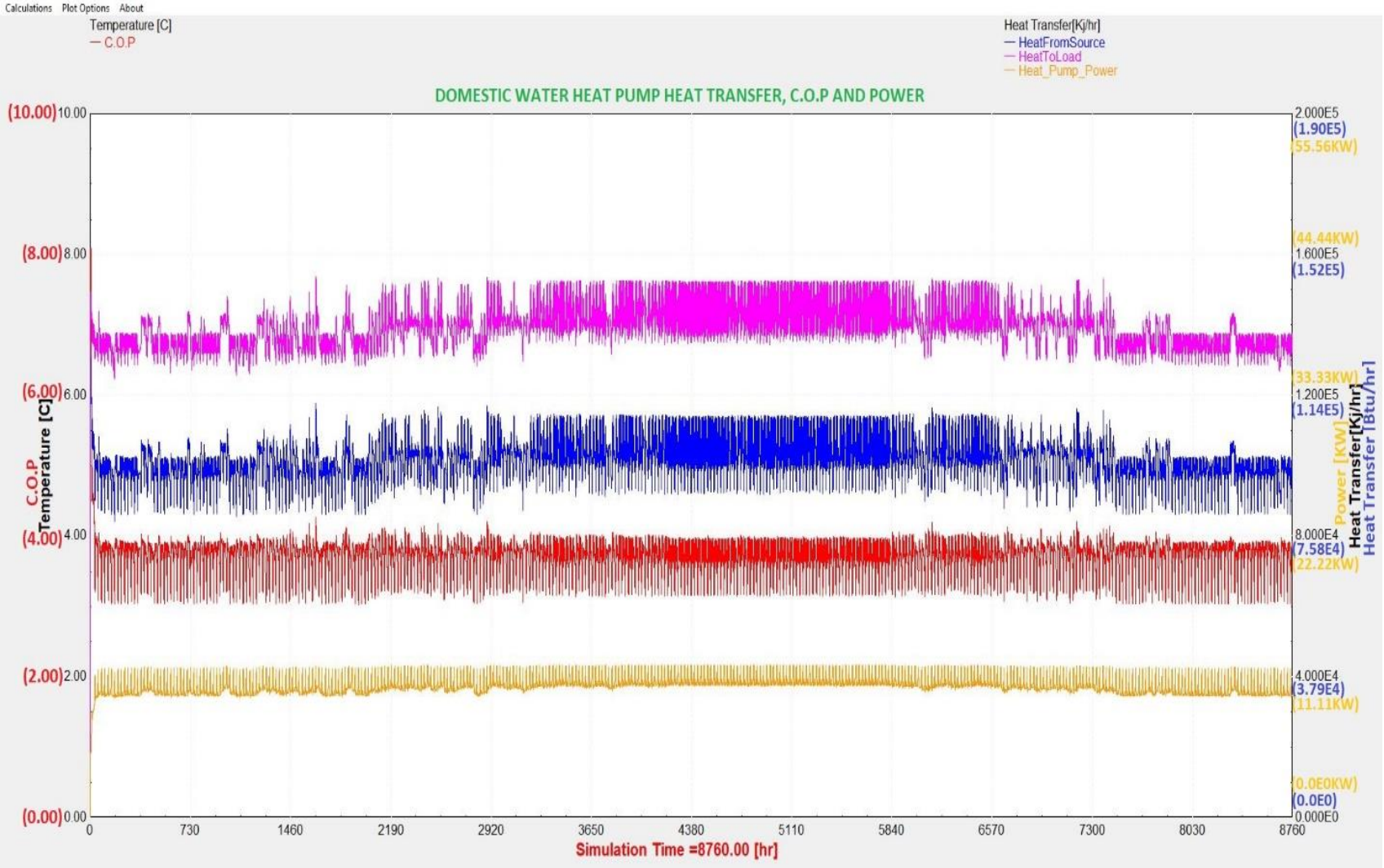


Figure 73. Domestic Water Heat Pump simulation results

For the domestic water heat pump, the COP can be seen to be majorly in the range 3.5 – 4.0. The source sides represent the heat pump loop while the load side represents the various load outlet that the hot water is being transferred to i.e. faucets. The domestic hot water tank is connected to the heat pump to provide auxiliary heating if needed but such was not imputed. For the outlet load temperature in the graph above, extremely high temperatures are seen in the graph above. This is as a result of the design in which there is a flow recirculation from the hot water tank to the heat pump and vice versa as seen in figure 68, alongside the hot water being distributed to various faucets. If the recirculation is disconnected as seen in figure 74, the outlet load temperature reduces as seen in figure 75. The temperature also reduces further if the pre-calculated rated capacity of the heat pump is also reduced and the amount reduced is imputed to the hot water tank to provide auxiliary heating in order to meet up with the pre-calculated capacity as seen in figure 76. However, the heat pump loop was left as seen in figure 68 to make it easier when estimating the economic cost and since this outlet load temperature would not have a direct impact on the heat pump loop load calculations, since a pre calculated heat capacity is always being extracted from the heat pump loop, with the outlet load temperature going directly to the loads or faucets.

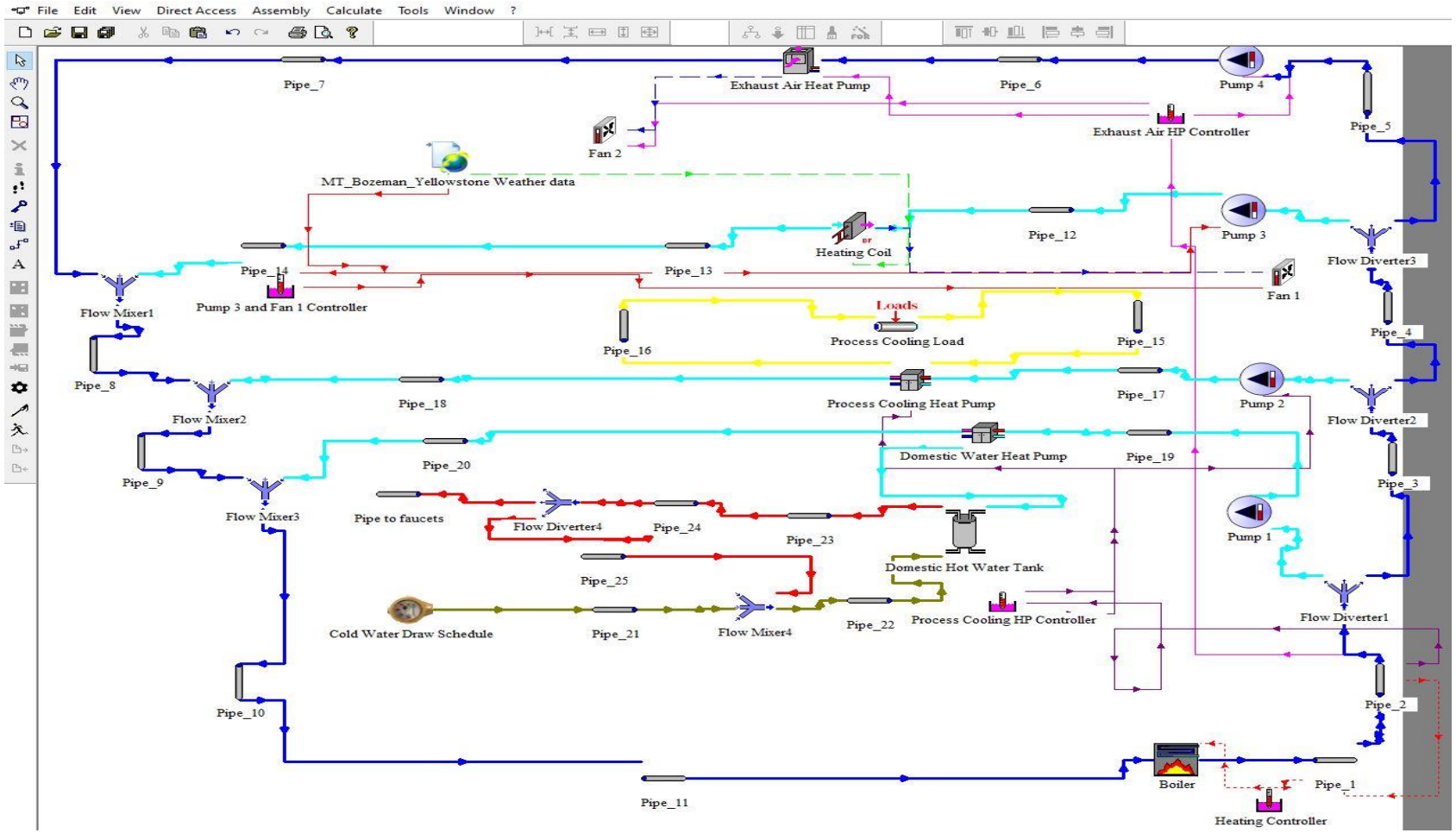


Figure 74. The Heat Pump Loop with the recirculation flow from the Domestic Hot Water Tank to the Heat Pump and vice versa disconnected

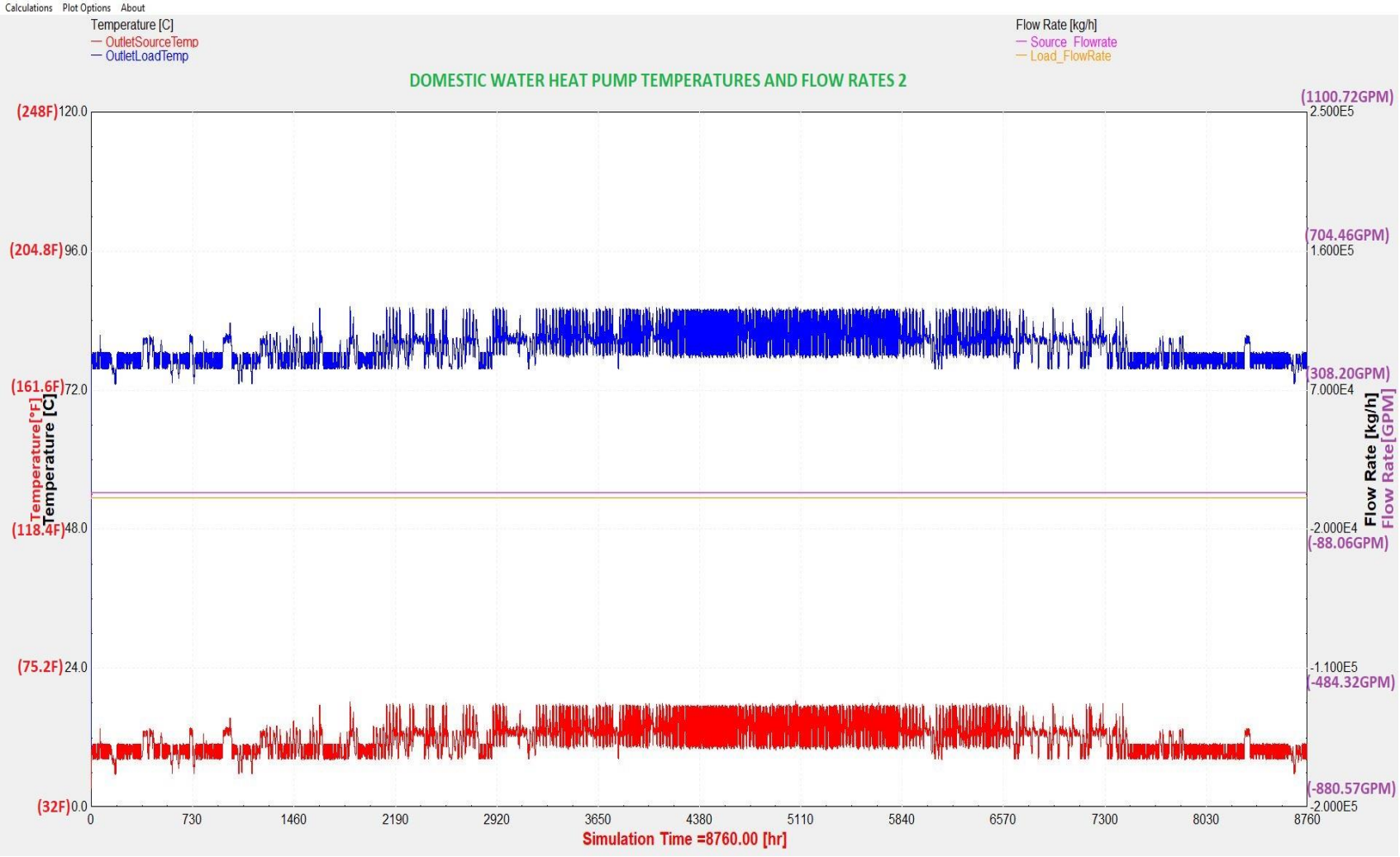


Figure 75. Domestic Water Heat Pump Outlet load temperature reduced as a result of the recirculation flow disconnected

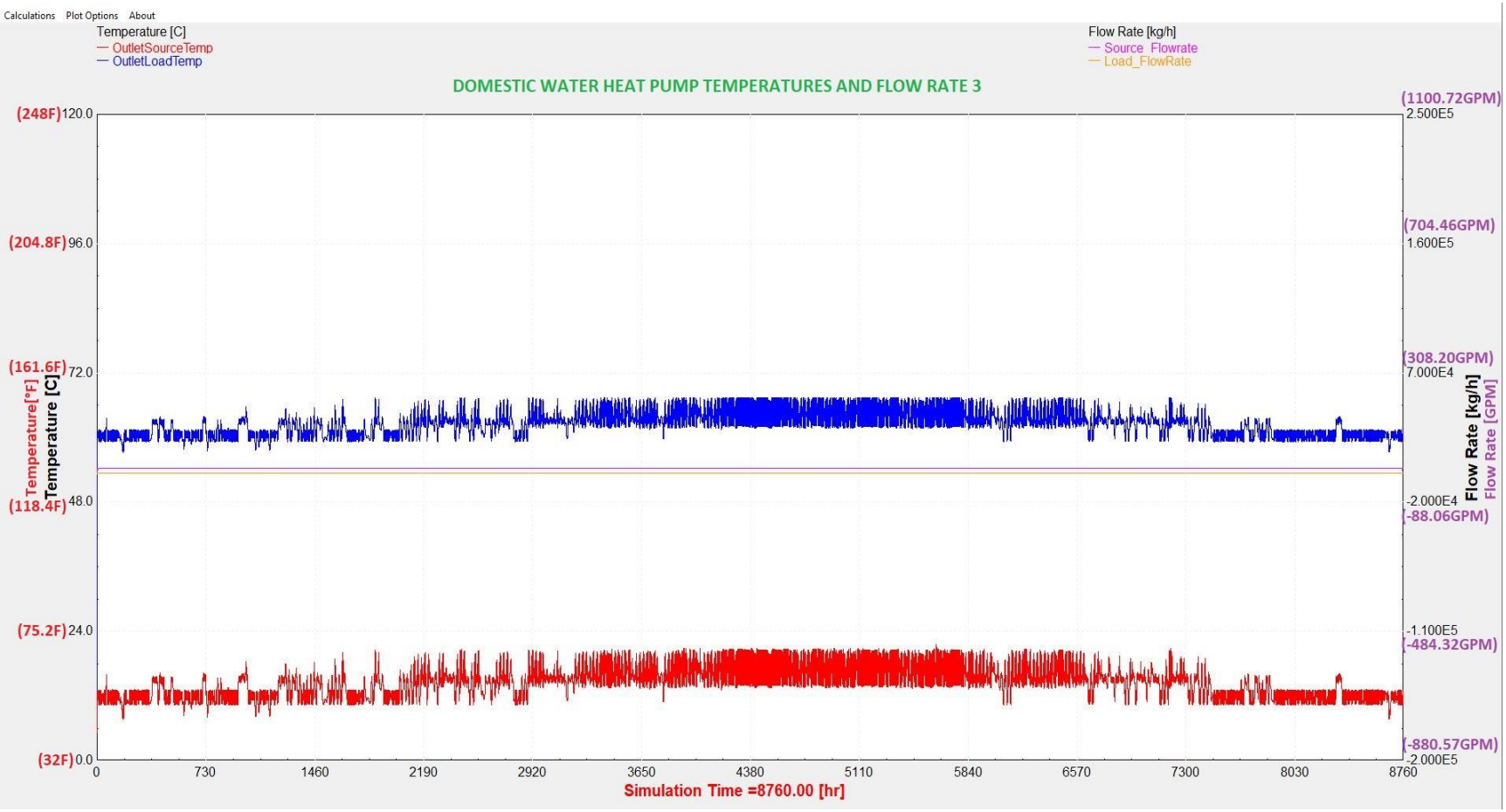


Figure 76. Domestic Water Heat Pump Outlet load temperature further reduced as a result of the recirculation flow disconnected and the Heat Pump Capacity reduced(From 120,200Btu/hr to 90,200Btu/hr). The net difference 30,000Btu/hr was imputed into the Domestic Hot Water Tank to provide auxiliary heating.

CHAPTER SIX

ECONOMIC ANALYSIS

Monthly Output

The printegrator component was used to generate values of each system on a monthly and hourly basis for the time frame of 8760hours of which the simulation was based on. The results on a monthly basis for each system can be seen below;

(i) Boiler

Table 3. Boiler Monthly Output

Month	Boiler Heat Energy Transfer (MJ/month)	Boiler Heat Energy Transfer (MMBtu/month)	Boiler Fuel Consumption (MJ/month)	Boiler Fuel Consumption (MMBtu/month)
January	172,634	164	230,178	218
February	120,698	114	160,931	152
March	54,256	51	72,342	69
April	9,430	9	12,573	12
May	580	1	774	1
June	-	-	-	-
July	-	-	-	-
August	-	-	-	-
September	1,306	1	1,741	2

October	7,544	7	10,058	10
November	93,570	89	124,760	118
December	183,804	174	245,072	232
Total	643,822	610	858,429	814

(ii) Process Cooling Heat Pump

Table 4. Process Cooling Heat Pump Monthly Output

Month	Process Cooling Heat Pump Heat Transfer to Load [To HP Loop] (MJ/month)	Process Cooling Heat Pump Heat Transfer to Load [To HP Loop] (MMBtu/month)	Process Cooling Heat Pump Heat Transfer from Source [From Process Cooling Load] (MJ/month)	Process Cooling Heat Pump Heat Transfer from Source [From Process Cooling Load] (MMBtu/month)
January	158,642	150	128,613	122
February	142,627	135	115,034	109
March	153,473	145	123,639	117
April	133,364	126	105,745	100
May	110,955	105	86,758	82
June	77,039	73	59,979	57
July	48,781	46	37,940	36
August	53,153	50	41,255	39
September	96,068	91	75,172	71
October	141,271	134	111,308	106
November	151,802	144	122,577	116
December	158,856	151	129,157	122
Total	1,426,031	1,350	1,137,177	1,077

Month	Process Cooling Heat Pump Power(MJ/month)	Process Cooling Heat Pump Power(MW)
January	30,029	8.34
February	27,593	7.66
March	29,834	8.29
April	27,619	7.67
May	24,197	6.72
June	17,060	4.74
July	10,842	3.01
August	11,898	3.31
September	20,896	5.80
October	29,963	8.32
November	29,226	8.12
December	29,699	8.25
Total	288,856	80.23

(iii) Domestic Water Heat Pump

Table 5. Domestic Water Heat Pump Monthly Output

Month	Domestic Water Heat Pump Heat Transfer to Load [To Hot Water Load] (MJ/month)	Domestic Water Heat Pump Heat Transfer to Load [To Hot Water Load] (MMBtu/month)	Domestic Water Heat Pump Heat Transfer from Source [From HP Loop] (MJ/month)	Domestic Water Heat Pump Heat Transfer from Source [From HP Loop] (MMBtu/month)
January	100,433	95	73,625	70
February	91,176	86	66,465	63
March	101,407	96	73,928	70
April	100,749	95	73,449	70
May	106,729	101	77,974	74
June	104,684	99	76,546	73
July	109,357	104	80,040	76
August	109,336	104	80,020	76
September	103,632	98	75,692	72
October	104,699	99	76,379	72
November	97,633	93	71,124	67
December	99,847	95	72,781	69
Total	1,229,682	1,165	898,023	852

Month	Domestic Water Heat Pump Power(MJ/month)	Domestic Water Heat Pump Power(MW)
January	26,808	7.45
February	24,711	6.86
March	27,480	7.63
April	27,301	7.58
May	28,755	7.99
June	28,138	7.82
July	29,318	8.14
August	29,316	8.14
September	27,940	7.76
October	28,320	7.87
November	26,509	7.36
December	27,066	7.52
Total	331,662	92.12

(iv) Exhaust Air to Water Heat Pump

Table 6. Exhaust Air to Water Heat Pump Monthly Output

Month	Total Heat Transfer from Air (MJ/month)	Total Heat Transfer from Air (MMBtu/month)	Total Heat Transfer to Liquid (MJ/month)	Total Heat Transfer to Liquid (MMBtu/month)
January	(182,002)	(173)	234,615	222
February	(152,224)	(144)	196,536	186
March	(160,556)	(152)	207,406	197
April	(109,652)	(104)	142,733	135
May	(52,429)	(50)	68,879	65
June	(25,588)	(24)	33,886	32
July	(15,865)	(15)	21,327	20
August	(15,060)	(14)	20,249	19
September	(48,782)	(46)	64,102	61
October	(93,576)	(89)	122,133	116
November	(166,797)	(158)	215,407	204
December	(186,644)	(177)	240,345	228
Total	(1,209,175)	(1,146)	1,567,618	1,485

Month	Total Power Consumption (MJ/month)	Total Power Consumption (MW)
January	52,613	14.61
February	44,312	12.31
March	46,850	13.01
April	33,082	9.19
May	16,450	4.57
June	8,297	2.30
July	5,462	1.52
August	5,189	1.44
September	15,320	4.26
October	28,557	7.93
November	48,610	13.50
December	53,701	14.92
Total	358,443	99.56

(v) Outdoor Air Heating Coil

Table 7. Outdoor Air Heating Coil Monthly Output

Month	Energy Transferred from Fluid to Air Stream (MJ/month)	Energy Transferred from Fluid to Air Stream (MMBtu/month)
January	506,801	480
February	405,428	384
March	355,024	336
April	224,788	213
May	115,708	110
June	50,110	47
July	9,808	9
August	12,604	12
September	100,513	95
October	206,055	195
November	403,917	383
December	525,629	498
Total	2,916,385	2,762

(vi) Pumps

Table 8. Pump Monthly Output

Month	Pump 1 Power Consumption (MJ/month) [Serves Domestic Hot Water System]	Pump 1 Power Consumption (MW) [Serves Domestic Hot Water System]	Pump 2 Power Consumption (MJ/month) [Serves Process Cooling System]	Pump 2 Power Consumption (MW) [Serves Process Cooling System]
January	1,819	0.51	3,459	0.96
February	1,643	0.46	3,118	0.87
March	1,819	0.51	3,356	0.93
April	1,760	0.49	2,938	0.82
May	1,819	0.51	2,460	0.68
June	1,760	0.49	1,711	0.48
July	1,819	0.51	1,084	0.30
August	1,819	0.51	1,182	0.33
September	1,760	0.49	2,129	0.59
October	1,819	0.51	3,121	0.87
November	1,760	0.49	3,316	0.92
December	1,819	0.51	3,459	0.96
Total	21,414	5.99	31,334	8.71

Month	Pump 3 Power Consumption (MJ/month) [Serves the Heating Coil]	Pump 3 Power Consumption (MW) [Serves the Heating Coil]	Pump 4 Power Consumption (MJ/month) [Serves the Exhaust Air System]	Pump 4 Power Consumption (MW) [Serves the Exhaust Air System]
January	5,549	1.54	7,130	1.98
February	5,012	1.39	5,980	1.66
March	5,339	1.48	6,314	1.75
April	4,527	1.26	4,372	1.21
May	3,564	0.99	2,126	0.59
June	2,146	0.60	1,053	0.29
July	846	0.23	672	0.19
August	1,032	0.29	638	0.18
September	2,888	0.80	1,979	0.55
October	4,893	1.36	3,748	1.04
November	5,303	1.47	6,556	1.82
December	5,549	1.54	7,298	2.03
Total	46,648	12.95	47,866	13.29

(vii) Fans

Table 9. Fan Monthly Output

Month	Fan 1 Power Consumption (MJ/month) [Serves Heating Coil]	Fan 1 Power Consumption (MW) [Serves Heating Coil]	Fan 2 Power Consumption (MJ/month) [Serves Exhaust Air System]	Fan 2 Power Consumption (MW) [Serves Exhaust Air System]
January	24,187	6.72	23,107	6.42
February	21,847	6.07	19,380	5.38
March	23,273	6.46	20,461	5.68
April	19,734	5.48	14,166	3.94
May	15,536	4.32	6,888	1.91
June	9,355	2.60	3,414	0.95
July	3,690	1.02	2,178	0.61
August	4,499	1.25	2,068	0.57
September	12,590	3.50	6,413	1.78
October	21,331	5.93	12,147	3.37
November	23,119	6.42	21,245	5.90
December	24,187	6.72	23,651	6.57
Total	203,348	56.49	155,118	43.08

Net Savings Calculation

Hence, Net Savings = Overall Net Benefits - Overall cost to operate the Heat Pump Loop.

Where Overall Net Value/Benefits = Process Cooling Heat Transfer to Load + Domestic Water Heat Pump Heat Transfer to Load + Outdoor Air Heating Coil Energy Transfer from Fluid to Air Stream = 1,350 MMBtu/year + 1,165 MMBtu/year + 2,762 MMBtu/year = 5277 MMBtu/year

Note: The Process cooling system heat transfer to load was considered a benefit because in most buildings in general, a chiller is used to extract and reject such heat energy from processes, and in this case we are not making use of such to extract the energy.

Converting to get the value in dollars using a rate of $10 per million Btu, Hence, for 5,277 MMBtu/year, we have a value of $52,770

Hence, Overall Net Value/Benefits = $52,770

Also, Overall cost to operate the Heat Pump Loop = Boiler Fuel Consumption + Process Cooling Heat Pump Power + Domestic Water Heat Pump Power + Exhaust Air to Water Heat Pump Power Consumption + Pump1 Power Consumption + Pump2 Power Consumption + Pump 3 Power Consumption + Pump 4 Power Consumption + Fan1 Power Consumption + Fan2 Power Consumption

Overall cost to operate the Heat Pump Loop = (814 MMBtu/year) + (80.23MW + 92.12MW + 99.56MW +5.99MW +8.71MW + 12.95MW + 13.29MW + 56.49MW + 43.08MW)

= (814 MMBtu/year) + (412.42MW)

Using a value of $10 per million Btu and $1 per 10KW,

Overall cost to operate the Heat Pump Loop = $8,140 + $41,242= $49,382

Hence, Net Savings = Overall Net Value/Benefits - Overall cost to operate the Heat Pump Loop

Net Savings = $52,770 - $49,382= $3,388

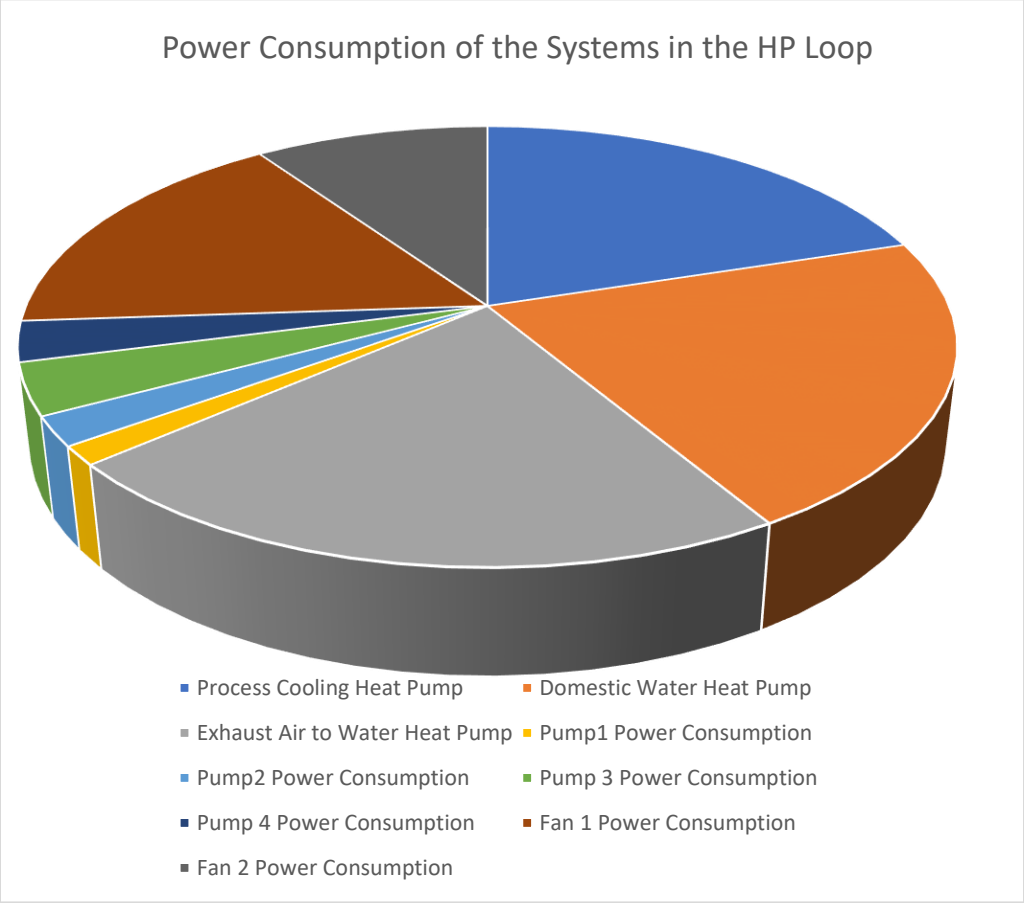


Figure 77. Power consumption of the systems in the heat pump loop

Economic analysis for the current systems in Barnard Hall

For some of the current systems in operation looked into in this project, the current cost of operation is looked into

(i) Process Cooling System: For the process cooling system, the major cost comes from the pump power consumption. The pump is always turned on throughout the year.

From the rated pump power consumption = 5HP,

For a year, 5HP x 8760 = 43,800HP = 32,661.65KW

Using a value of $1 per 10KW, for 32,661.65KW, we have $3,266.17

(ii) The Outdoor Air: From Qair = CFM x 0.92 x ΔT

Temperature data gotten from the EnergyPlus weather data was used and Qair was computed for temperatures below 55°F (set point temperature)

Hence, Qair = 1,415,918,640 Btu/year ≈ 1416 MMBtu

Using a value of $10 per million Btu, for 1416 MMBtu, we have $ 14,160

(iii) The Domestic Hot Water System: From the Precalculated load demand = 120,182.5 Btu/hr

For an assumed hours of usage of 8am – 6pm (10hours)

Qtotal in a day = 120,182.5Btu/hr x 10hours/day = 1,201,825Btu/day

Qtotal in a year = 1,201,825 Btu/day x 365days = 438,666,125Btu/year ≈ 439 MMBtu/year

Working with a Gas cost of $7.5/MMBtu, for 439 MMBtu/year, Gas cost = $3,292.50

(iv) For the Boiler, Barnard Hall accounts for 5% of the total campus steam

Total campus steam consumption = 323, 846 MMBTU (Year 2019)

Hence 5% of 323,846 MMBTU = 16192.30 MMBTU

This amount accounts for the whole building usage in Barnard Hall and cannot be accounted for the applied systems considered in this project due to limited data availability. Similar situation arises for the cooling tower alongside.

CONCLUSION

Various heat sources and sinks available have being identified and the energy savings from connecting such systems together in any building considered. This was done by using a building on campus (Barnard hall) as a case study in which if the results are promising, similar models can be adopted to other buildings and connected to an overall heat pump loop on campus. The project also succeeded in providing ways of assessing or measuring the energy value of various systems in any building and to determine the energy savings of connecting those systems together in a heat pump loop. This researched showed that this provided some energy savings opportunities and no profit in general.

Also, more savings can be generated by increasing the coefficient of performances of the heat pumps. The COP for the heat pumps used in this simulation were operating within the range of 3.5 – 5.5, hence more savings can be generated if this is increased. Selecting a fan and pump that is more efficient (unlike the fan and pump used in this design which had an efficiency of 65%) can also help increase the savings generated. Also, more savings can also be generated with the usage of true variable speed pumps, fans and heat pumps.

RECOMMENDATIONS

Future recommendations based on this project include looking into the energy potential and savings of incorporating the Geothermal system to majority of the buildings at MSU and connecting such to the overall heat pump loop. Also the feasibility of incorporating other renewable energy sources such as solar and wind energy, as well as the thermal energy storage system into the heat pump loop to store and save energy could be looked into. Another point worth mentioning is looking into the usage of true variable systems in the heat pump loop such as variable flow pumps, fans, and heat pumps. Lastly, the energy savings potential of connecting the district energy system to electricity generation facilities to provide power could be looked into.

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APPENDICES

APPENDIX A

TOPPS COIL OUTPUT SAMPLE

TRANE

Job01

Tag	HTCL-1	System type	Hot Water
Model Number	DW0B51		
Quantity	1		

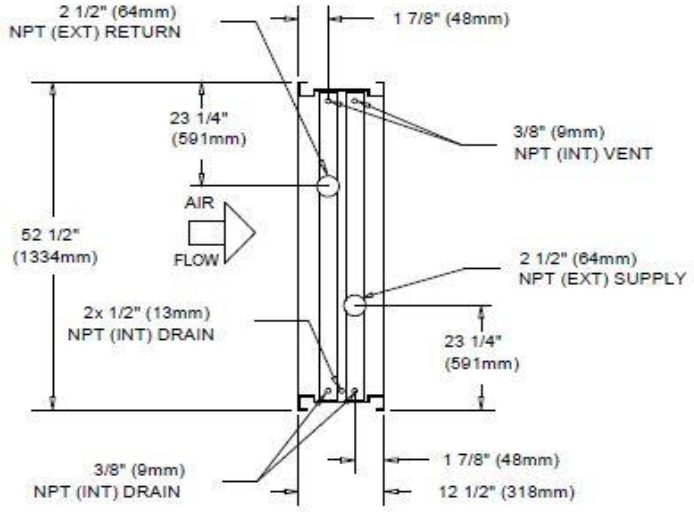
Coil Information

Coil type	W	Actual airflow	20000 cfm
Rows	6	Entering dry bulb	70.00 F
Finned length	115" (2921 mm)	Leaving dry bulb	73.42 F
Nominal coil height	51" (1295 mm)	Total capacity	74.14 MBh
Fin type	Prima-Flo E	Actual coil face area	40.73 sq ft
Fin material	Aluminum	APD	0.674 in H2O
Nominal fin spacing	168 fins per foot	Volume	33.02 gal
Tube matl/wall thickness	.020 (0.508 mm) copper	Elevation	50.00 ft
Corrosion resistant coating	No	Face velocity	491 ft/min
Coil tube drainability	Drainable	Rigging weight	1005.6 lb
Turbulators	No	Installed weight	1280.7 lb
		AHRI 410 classification	Outside scope of AHRI

Hot Water Information

Standard fluid flow rate	95.00 gpm	Fluid PD	7.50 ft H2O
Entering fluid temp	74.00 F	Fluid velocity	3.09 ft/sec
Leaving fluid temp	72.44 F	Fouling factor	0.00050 hr-sq ft-deg F/Btu
Fluid temp drop	1.56 F	Reynolds number	15827.47 Each
Fluid type	Water		

51" W 6 ROW
LEFT HAND



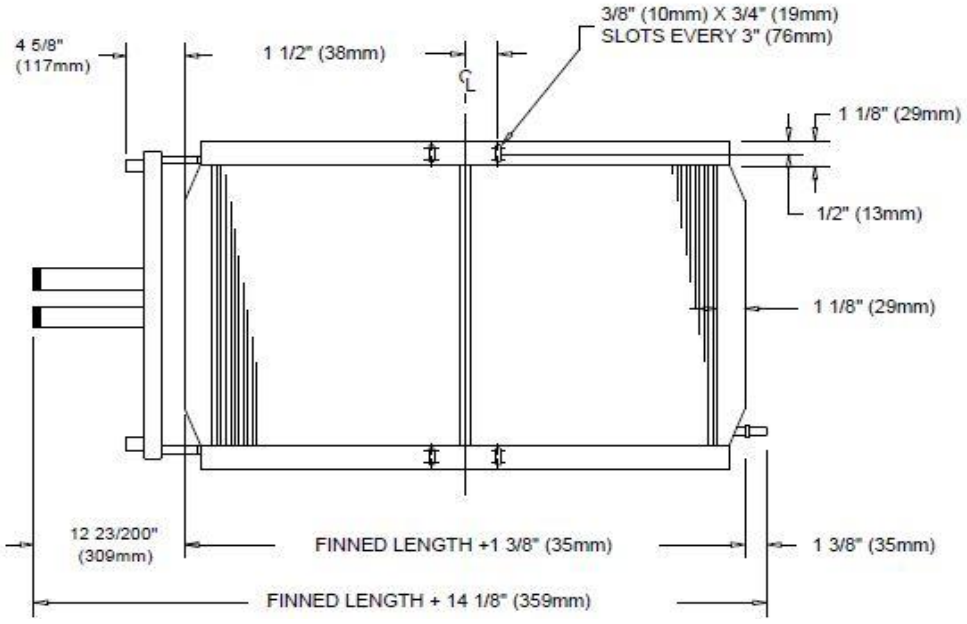


Figure 78. TOPPS Coil output sample

APPENDIX B

BIN CALCULATION AND HEAT EXCHANGER CALCULATOR SETUP FOR THE OUTSIDE AIR

Calculated UA for Heat Exchanger	43711.87
Entering Water Temperature(˚F)	60.00
Outside Air Flowrate(CFM)	20000.00
Heat Capacity (Air)	0.92
Air Conductance	18400.00
Water Flow Rate(GPM)	115.00
Heat Capacity (Water)	500.00
Water Conductance	57500.00
Heat Capacity Ratio, Z= CMIN/CMAX	0.320
Number of Transfer Units(NTU=UA/CMIN)	2.376
HX EFFECTIVENESS	
TYPE (COUNT (CF), CROSS (CS), PARA (PF), COND (CO))	CS
COUNTERFLOW HX	0.00
CROSSFLOW HW, Z= +\- 1, OR TD COLD +\- TD HOT	0.83
PARALLEL FLOW, Z<1	0.00
CONDENSING FLUID	0.00

CALCULATING UA	
Entering Water Temperature(˚F)	62.00
Leaving Water Temperature(˚F)	66.52
Entering Air Temperature(˚F)	70.00
Leaving Air Temperature(˚F)	64.29
Temperature Difference at Inlet of HX	2.29
Temperature Difference at outlet of HX	3.48
Calculated LMTD	2.84
Water Flow Rate(GPM)	55.00
Heat Capacity (Water)	500.00
Water Conductance	27500.00
Outside Air Flowrate(CFM)	20000.00
Heat Capacity (Air)	0.92
Air Conductance	18400.00
MEASURED HEAT TRANSFER (SET UP FOR THE WATER SIDE HX)	124300.00
CALCULATED UA = Q/LMTD (OR MTD) OF HEAT EXCHANGER	43711.87



Figure 79. Bin Calculation and Heat Exchanger Calculator Setup for the Outdoor Air

Table 10. Bin Calculation and Heat Exchanger Calculator Setup for the Outdoor Air

OCCUPIED PERIOD CALCULATIONS			
OCCUPIED MONTHS PER YEAR	12.00	MONTHS	1.00 FRACTION OF TOTAL
OCCUPIED DAYS PER WEEK	7.00	DAYS	1.00 FRACTION OF TOTAL
OCCUPIED FRACTION, 12 AM - 7:59 AM:	1.00	HOURS	0.13 FRACTION OF TOTAL
OCCUPIED FRACTION, 8 AM - 3:59 PM:	8.00	HOURS	1.00 FRACTION OF TOTAL
OCCUPIED FRACTION, 4 PM - 11:59 PM:	2.00	HOURS	0.25 FRACTION OF TOTAL

					ANNUAL TOTALS	
BIN CENTER TEMP(°F)	HOURS 1 THROUGH 8	HOURS 9 THROUGH 16	HOURS 17 THROUGH 24	BIN TOTALS	ANNUAL OCCUPIED HOURS	ANNUAL UNOCCUPIED HOURS
97.00	0.00	4.00	2.00	6.00	4.50	1.50
92.00	0.00	26.00	12.00	38.00	29.00	9.00
87.00	0.00	69.00	37.00	106.00	78.25	27.75
82.00	0.00	112.00	62.00	174.00	127.50	46.50
77.00	1.00	163.00	98.00	262.00	187.63	74.38
72.00	14.00	202.00	141.00	357.00	239.00	118.00
67.00	58.00	218.00	188.00	464.00	272.25	191.75
62.00	147.00	220.00	209.00	576.00	290.63	285.38
57.00	251.00	217.00	237.00	705.00	307.63	397.38
52.00	304.00	218.00	233.00	755.00	314.25	440.75
47.00	300.00	234.00	240.00	774.00	331.50	442.50
42.00	301.00	241.00	269.00	811.00	345.88	465.13
37.00	321.00	259.00	280.00	860.00	369.13	490.88
32.00	332.00	218.00	266.00	816.00	326.00	490.00
27.00	263.00	144.00	190.00	597.00	224.38	372.63
22.00	178.00	97.00	125.00	400.00	150.50	249.50
17.00	132.00	80.00	88.00	300.00	118.50	181.50
12.00	91.00	57.00	64.00	212.00	84.38	127.63
7.00	61.00	43.00	51.00	155.00	63.38	91.63
2.00	50.00	32.00	35.00	117.00	47.00	70.00
-3.00	36.00	25.00	28.00	89.00	36.50	52.50
-8.00	26.00	22.00	22.00	70.00	30.75	39.25
-13.00	21.00	14.00	17.00	52.00	20.88	31.13
-18.00	17.00	9.00	14.00	40.00	14.63	25.38
-23.00	11.00	2.00	4.00	17.00	4.38	12.63
-28.00	4.00	0.00	1.00	5.00	0.75	4.25
-33.00	1.00	0.00	1.00	2.00	0.38	1.63

Heat Transfer(Btu/hr)	Heat Transfer for Occupied Hours(Btu)	Heat Transfer for Unoccupied Hours(Btu)	Heat Transfer Total(Btu)	Leaving Air Temperature(°F)	Leaving Water Temperature(°F)
564112.61	2538506.74	846168.91	3384675.65	66.34	69.81
487881.17	14148554.07	4390930.57	18539484.64	65.48	68.48
411649.74	32211592.25	11423280.32	43634872.57	64.63	67.16
335418.31	42765834.22	15596951.30	58362785.53	63.77	65.83
259186.87	48629937.25	19277023.76	67906961.01	62.91	64.51
182955.44	43726350.29	21588741.98	65315092.27	62.06	63.18
106724.01	29055610.90	20464328.34	49519939.24	61.20	61.86
30492.57	8861904.15	8701818.14	17563722.29	60.34	60.53
-45738.86	-14070416.85	-18175479.54	-32245896.39	59.49	59.20
-121970.29	-38329164.79	-53758406.94	-92087571.73	58.63	57.88
-198201.73	-65703872.58	-87704264.30	-153408136.88	57.77	56.55
-274433.16	-94919569.49	-127645723.91	-222565293.40	56.91	55.23
-350664.59	-129439068.39	-172132482.75	-301571551.14	56.06	53.90
-426896.03	-139168105.09	-209179053.67	-348347158.76	55.20	52.58
-503127.46	-112889224.16	-187477870.32	-300367094.49	54.34	51.25
-579358.90	-87193513.70	-144550044.30	-231743558.00	53.49	49.92
-655590.33	-77687453.93	-118989644.63	-196677098.57	52.63	48.60
-731821.76	-61747461.18	-93398752.39	-155146213.57	51.77	47.27
-808053.20	-51210371.27	-74037874.05	-125248245.33	50.92	45.95
-884284.63	-41561377.57	-61899924.04	-103461301.62	50.06	44.62
-960516.06	-35058836.29	-50427093.30	-85485929.59	49.20	43.30
-1036747.50	-31879985.51	-40692339.23	-72572324.74	48.34	41.97
-1112978.93	-23233435.16	-34641469.19	-57874904.35	47.49	40.64
-1189210.36	-17392201.57	-30176212.97	-47568414.54	46.63	39.32
-1265441.80	-5536307.86	-15976202.69	-21512510.55	45.77	37.99
-1341673.23	-1006254.92	-5702111.23	-6708366.15	44.92	36.67
-1417904.66	-531714.25	-2304095.08	-2835809.33	44.06	35.34

APPENDIX C

ENTERING WATER TEMPERATURE VARIATION

Entering Water Temperature Variation

Clustered Column Chart

(i) 60°F Entering water temperature at fixed 55GPM

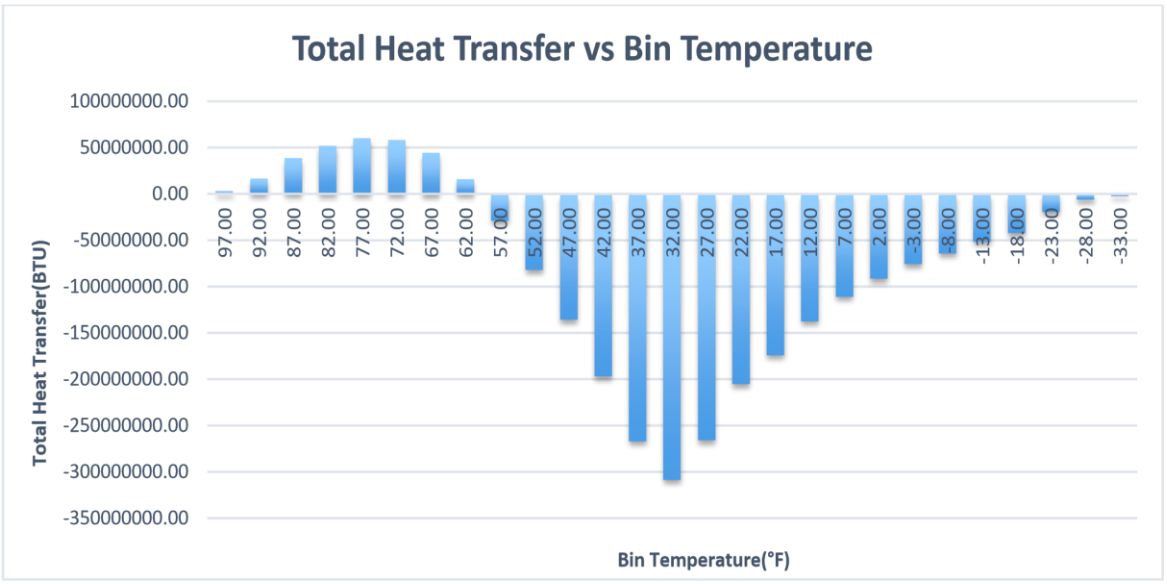


Figure 80. 60°F Entering water temperature at fixed 55GPM

(ii) 65°F Entering water temperature at fixed 55GPM

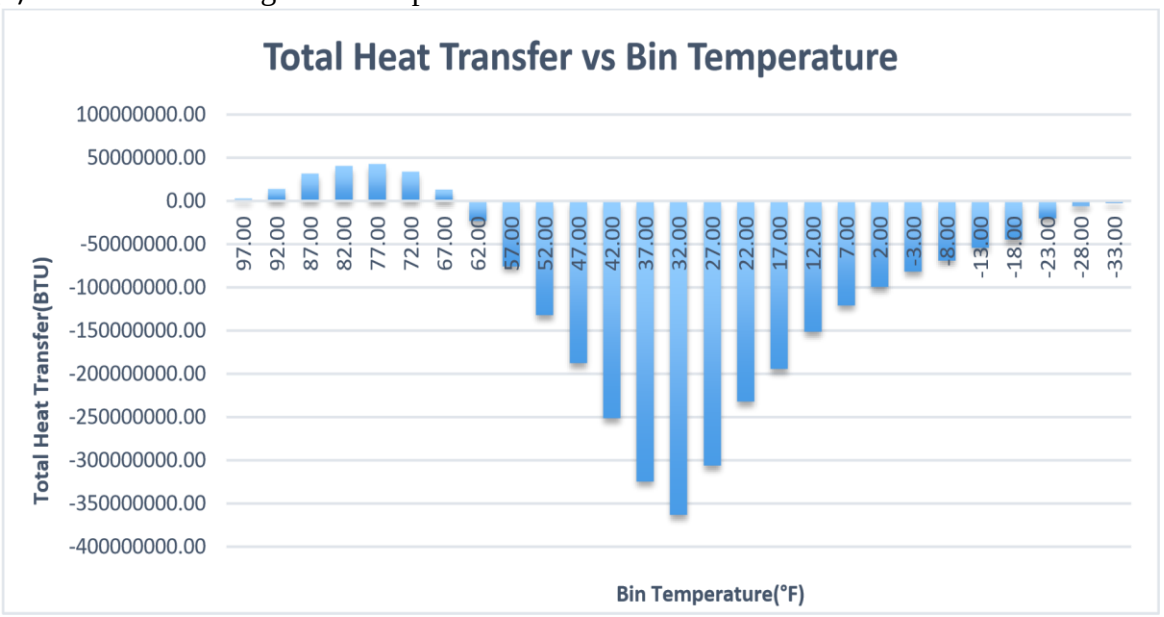


Figure 81. 65°F Entering water temperature at fixed 55GPM

(iii) 70°F Entering water temperature at fixed 55GPM

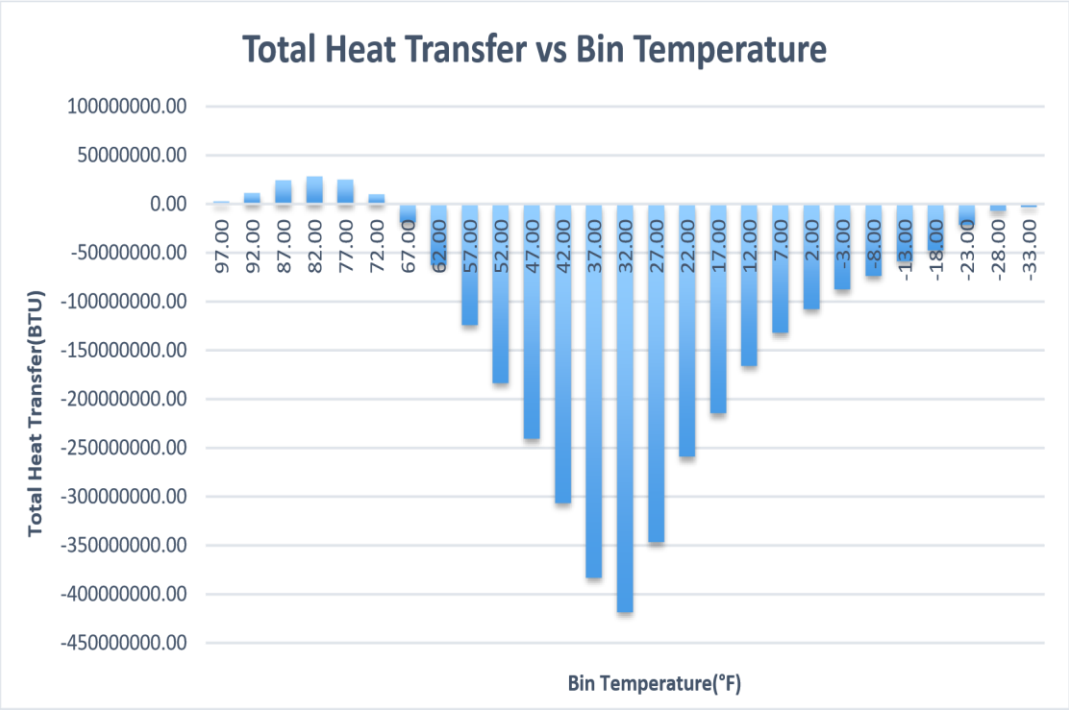


Figure 82. 70°F Entering water temperature at fixed 55GPM

(iv) 75°F Entering water temperature at fixed 55GPM

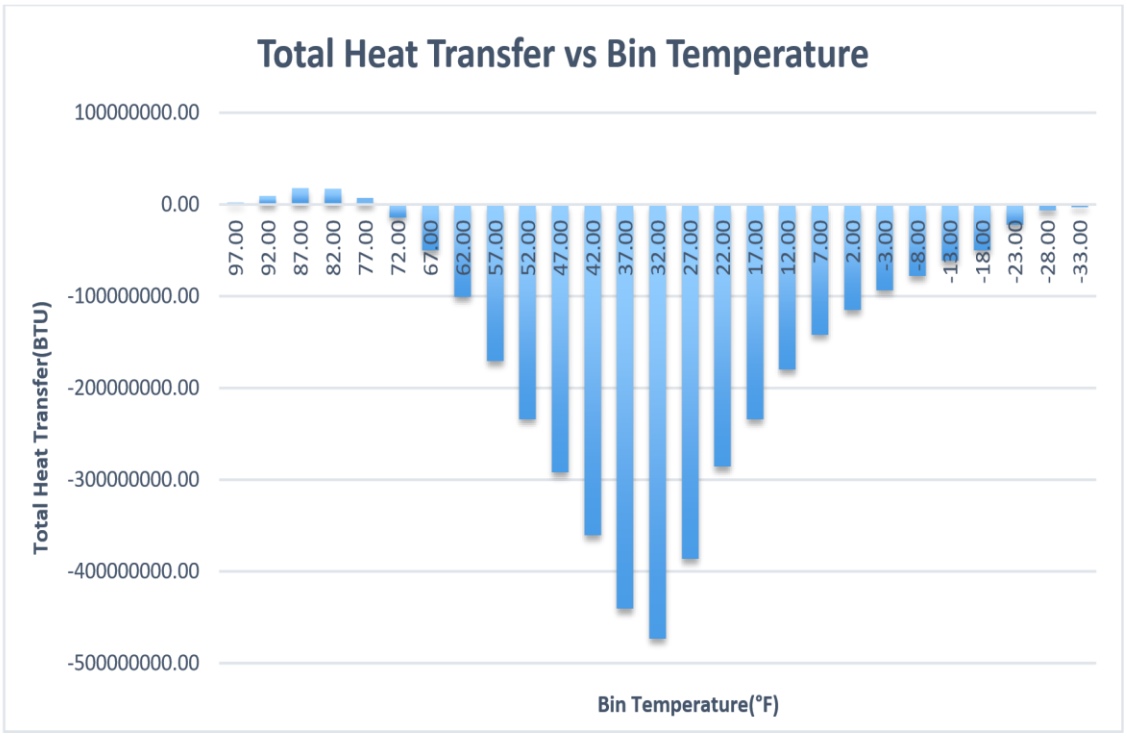


Figure 83. 75°F Entering water temperature at fixed 55GPM

(v) 80°F Entering water temperature at fixed 55GPM

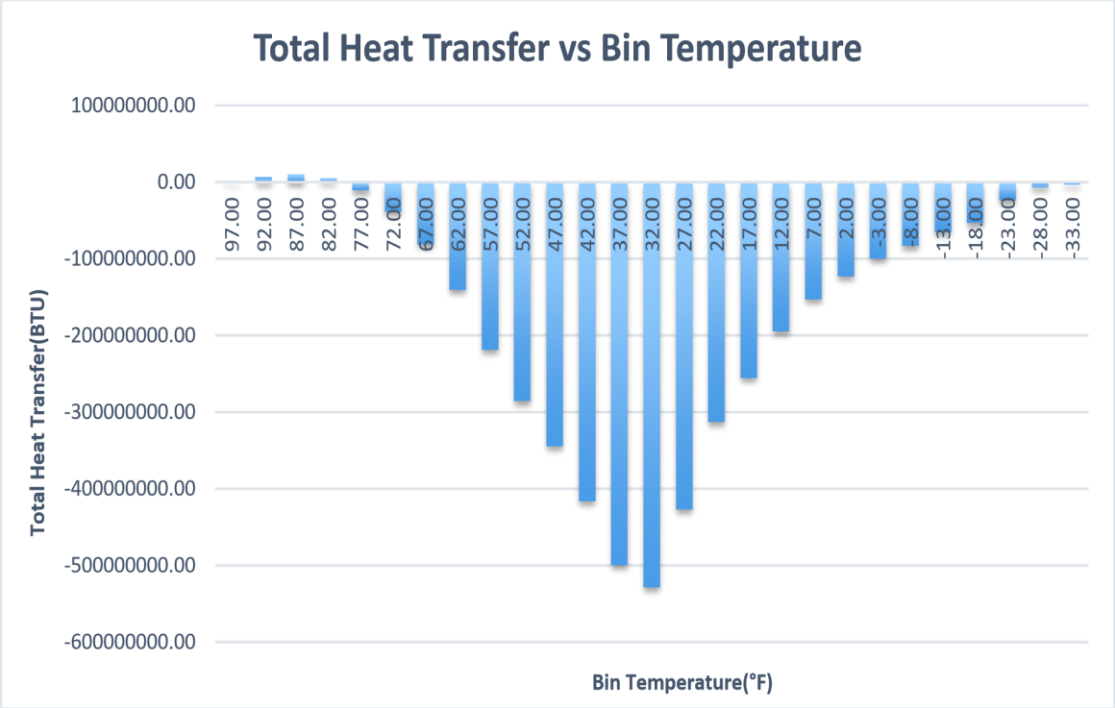


Figure 84. 80°F Entering water temperature at fixed 55GPM

(vi) 85°F Entering water temperature at fixed 55GPM

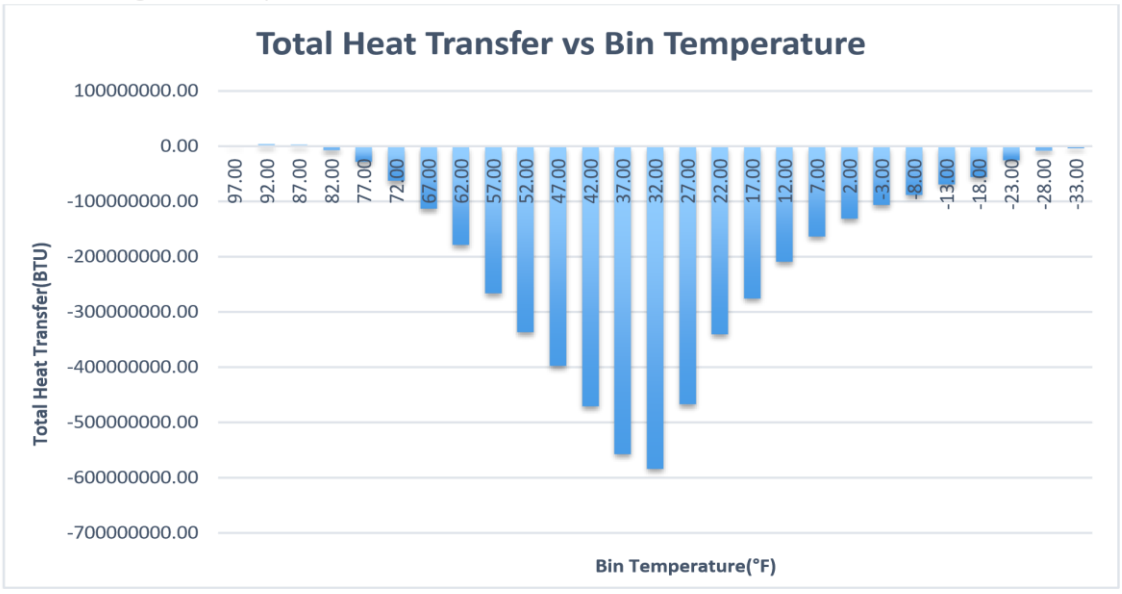


Figure 85. 85°F Entering water temperature at fixed 55GPM

(vii) 90°F Entering water temperature at fixed 55GPM

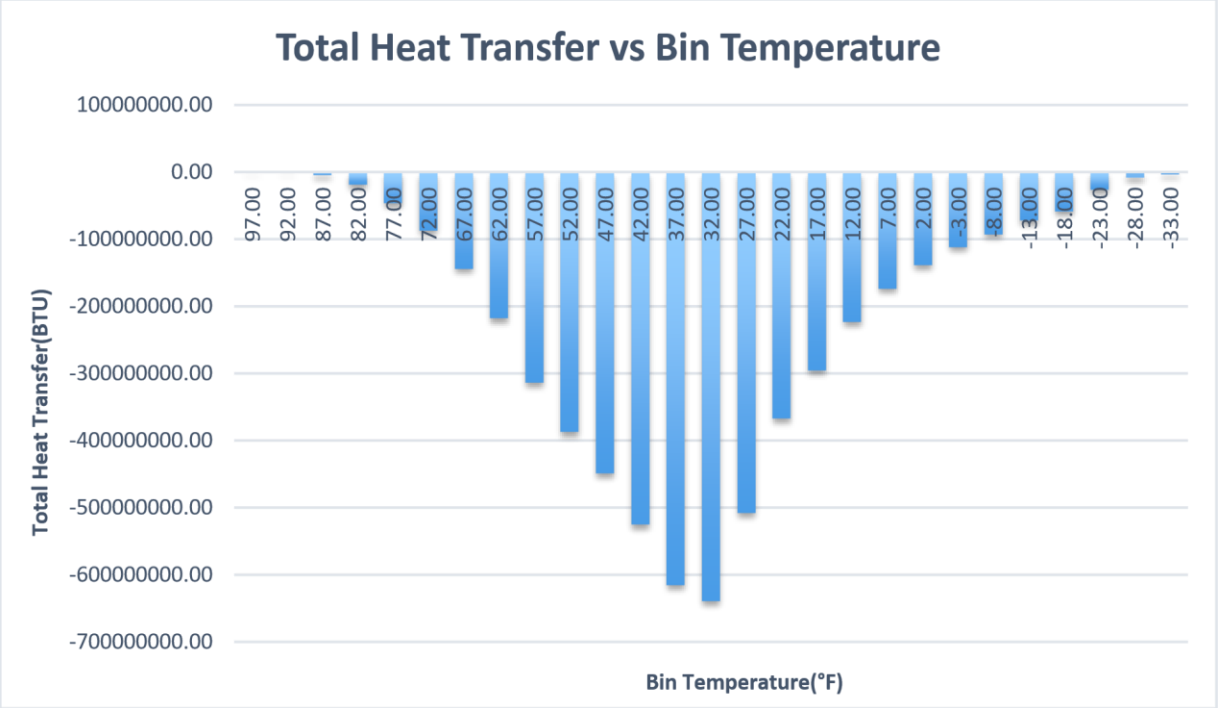


Figure 86. 90°F Entering water temperature at fixed 55GPM

APPENDIX D

WATER FLOW RATE VARIATION

Water Flow Rate Variation

Clustered Column Chart

(i) 55 GPM at 60°F Entering water temperature

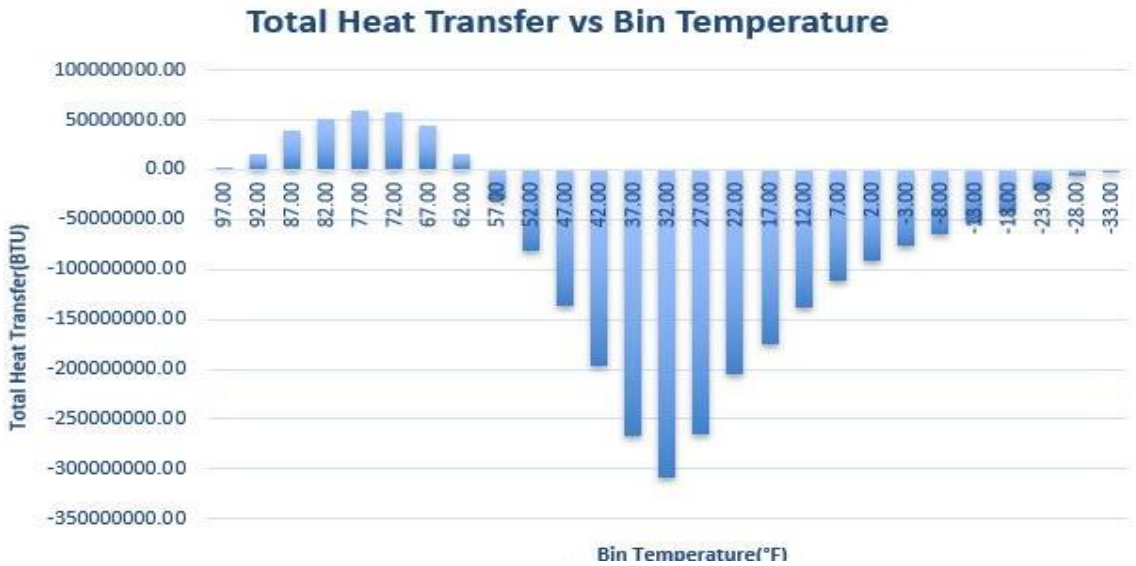


Figure 87. 55 GPM at 60°F Entering water temperature

(ii) 65 GPM at 60°F Entering water temperature

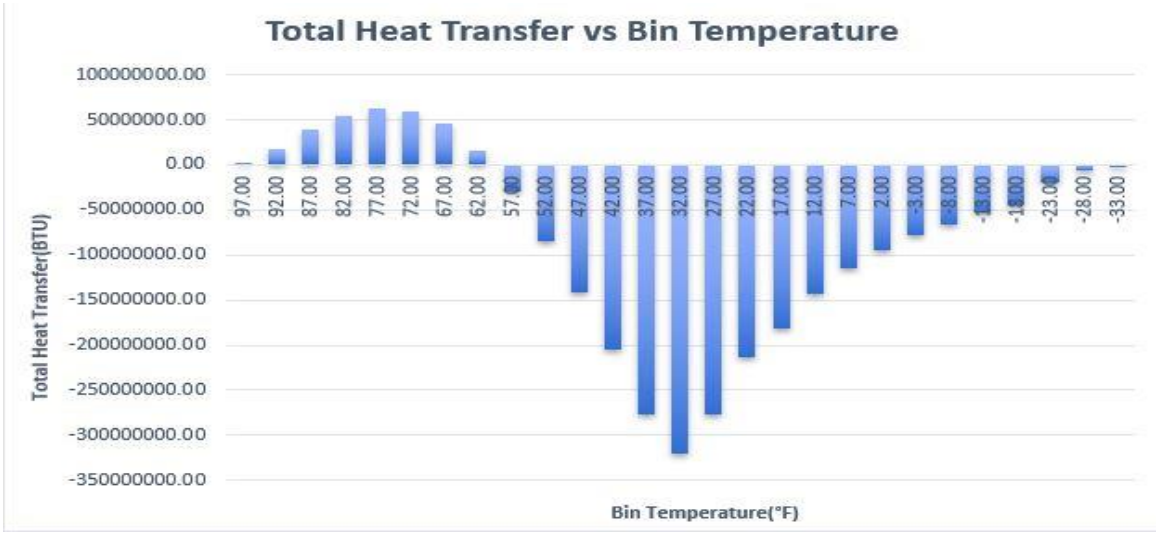


Figure 88. 65 GPM at 60°F Entering water temperature

(iii) 75 GPM at 60°F Entering water temperature

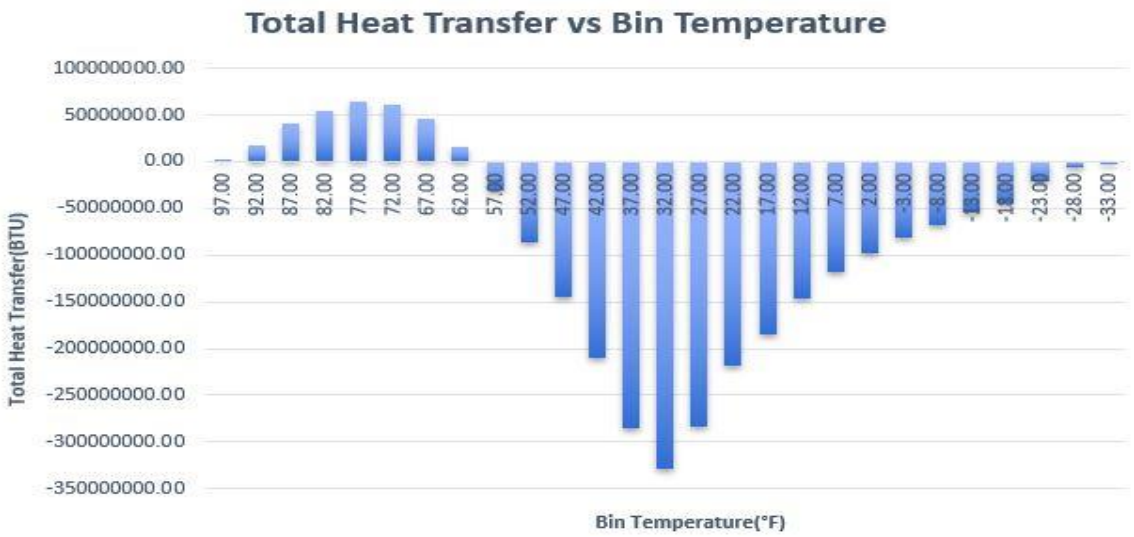


Figure 89. 75 GPM at 60°F Entering water temperature

(iv) 85 GPM at 60°F Entering water temperature

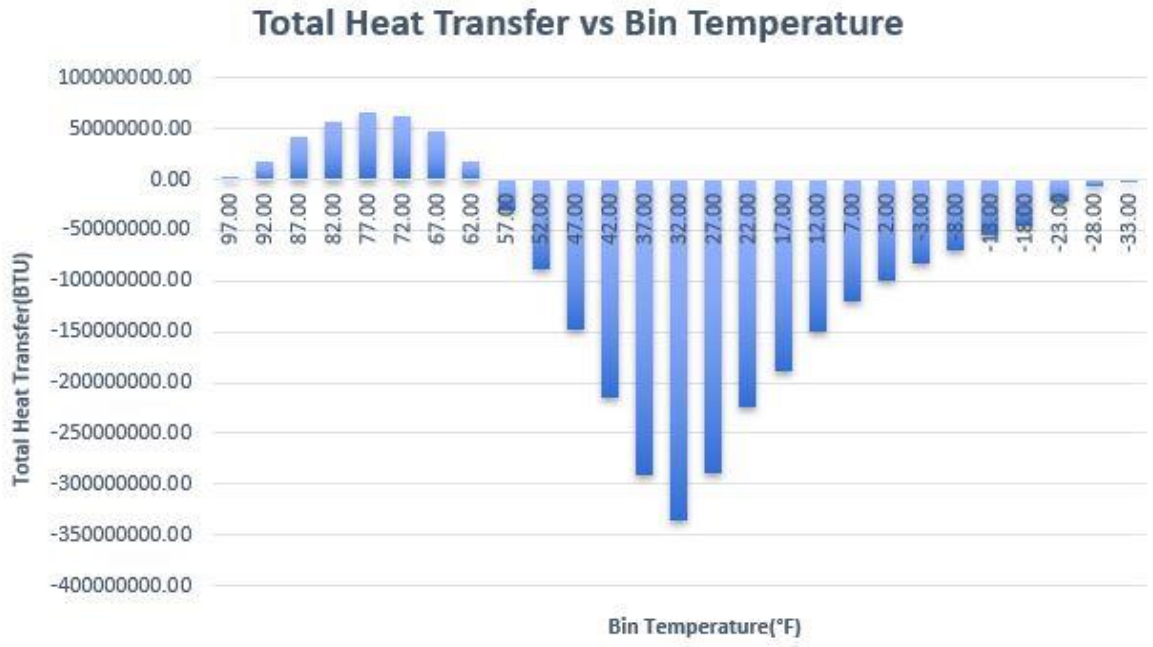


Figure 90. 85 GPM at 60°F Entering water temperature

(v) 95 GPM at 60°F Entering water temperature

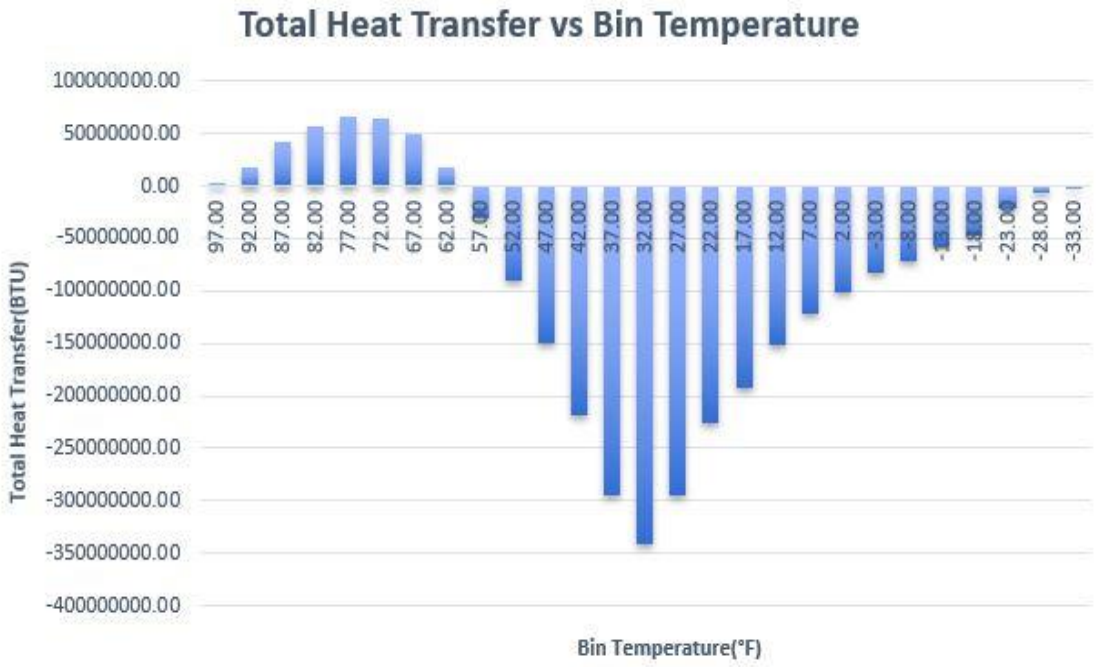


Figure 91. 95 GPM at 60°F Entering water temperature

(vi) 105 GPM at 60°F Entering water temperature

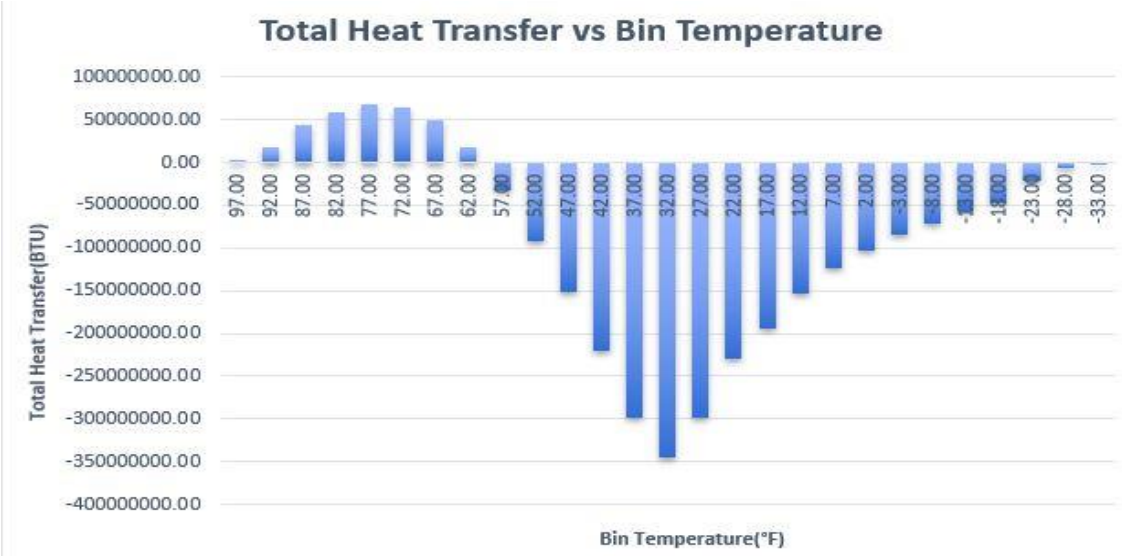


Figure 92. 105 GPM at 60°F Entering water temperature

(vii) 115GPM at 60°F Entering water temperature

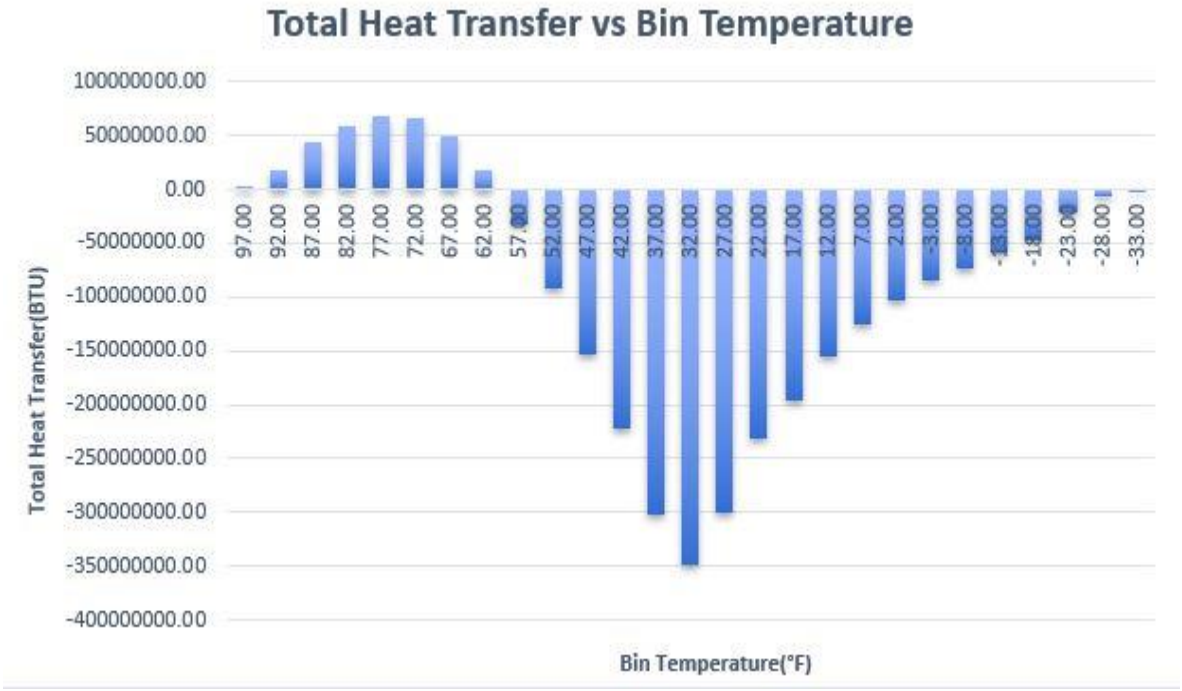


Figure 93. 115 GPM at 60°F Entering water temperature