Advanced Heat Recovery Project

MEE 488

Mechanical Engineering Department

University of Maine

05/06/08

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Acknowledgements

Special thanks to James LaBrecque for the countless hours of HVAC mentoring and patience.

Also thanks to Stewart Harvey, Bill Lovejoy, and Norm Costello for their valued contributions and assistance in helping us complete this project.

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1. Problem Definition

1.1. Problem

The Engineering and Science Research Building houses laboratories which require high ventilation air rates. 54,000 cubic feet per minute (CFM) of outside air is required twenty-four hours per day, seven days per week, to meet OSHA regulations. Heating the makeup air to 65°F consumes roughly 70,000 gallons of #6 fuel oil annually, at a cost of over $120,000. The energy supplied to heat the air is lost as the building air is exhausted to the atmosphere. During the warmer summer months, a similar situation occurs, whereby outdoor air must be cooled, only to be exhausted to the atmosphere at a rate of 54,000 CFM.

A group of Mechanical Engineering students from 2006-2007, evaluated the advantages of installing a direct expansion heat recovery system in the Engineering and Science Research Building. They also analyzed the various elements within the system and conducted a rough environmental and economic analysis. The 2007-2008 group of Mechanical Engineering students was thus presented with the task of completing the project.

1.2. Objective

The following four objectives were used to engineer a heat recovery system:

1.2.1 Provide a cost analysis that evaluates the benefit of the project.

1.2.2 Provide a workable design to recover the maximum amount of energy at the lowest cost while reducing harmful environmental emissions.

1.2.3 Demonstrate that the project meets the criteria of the Green Loan Fund and request funding.

1.2.4 Produce a complete drawing and bill of materials package, which could be used by an HVAC contractor.

1.3. Solution

The goal of the Advanced Heat Recovery Project is to design and install a direct expansion heat pump system for the Engineering and Science Research Building (ESRB) at the University of Maine. The Advanced Heat Recovery Project seeks to produce an air recovery system to decrease the amount of energy required to maintain OSHA requirements at the ESRB.

The heat recovery system will take advantage of a direct expansion heat pump which has the capability to recover more energy with greater efficiency than a traditional water-glycol system. The recovered energy from the heat pump will preheat the incoming air and thus reduce the heating load. The heat pump will be adjusted automatically to maximize the performance over a large range of outdoor conditions. A state of the art compressor utilizing magnetic levitation bearings will power the direct expansion system, eliminating the need for oil lubrication within the compressor. This heat recovery system is expected to save the university roughly $80,000 per year in heating costs alone.

Reduction in energy consumption is at the forefront of the project's design. Annually, the heat recovery system will prevent 611 tons of carbon dioxide, a major green house gas, and 9.5 tons of sulfur dioxide, a major contributor of acid rain, from reaching the atmosphere. The Advanced Heat Recovery Project will lead the way for the University of Maine to reduce its energy consumption, and become a more environmentally conscious establishment.

2. Project Justification

2.1. Introduction to Heat Pump Analysis Spreadsheet

In order to evaluate the utility of installing this heat pump, it was necessary to conduct a thorough analysis of the impact that such a system would have on the cost of heating the Engineering and Science Research Building. It quickly became apparent, that this was formidable design challenge, and the data available from the 2006-2007 group could be improved upon. Building on the analysis which had been made, the group began to consider methods of accurately evaluating the heat pump's impact on the building's heat requirements.

The chosen method was a spreadsheet which divided the heating season into temperature groups, known as a 'BIN'. Each 'BIN' was represented by a row in the spreadsheet, and various calculations, described in the following pages, could be made for each condition. Summing the results for each temperature 'BIN' gave the totals for the heating season.

2.2. Climate Data

To evaluate the heating load, and the heat pump performance, it was necessary to obtain data on the number of hours at given temperatures throughout the heating season. The solution was BIN numbers, which give the number of hours during which the outdoor temperature falls within a given temperature range. The 2006-2007 project collected BIN numbers for Portland, ME; no BIN data was available for Bangor, ME, so the Portland, ME data was used. BIN numbers give the annual number of hours within a given temperature range, in increments of 5°F. This allowed us to divide the heating season into 'bins' of hours when the temperature can be approximated by the average of the given temperature range. For example, during the average year, there are 599 hours during which the temperature is between 27.5°F and 32.5°F. This is the 30°F 'bin'. The bin data, allowed us to evaluate, from thermodynamic principles, the amount of energy required to heat the air to 65°F.

2.3. Heating System Data

The next step in the analysis was to determine the amount of fuel, required to provide this quantity of heat to the ESRB. The heat is provided primarily by burning #6 fuel oil in the steam plant, to produce steam which is then piped to the ESRB, where it condenses, producing heat. The entire process must be analyzed to determine the economic and environmental benefit of reducing the heat load by a given amount.

From a review of the work done by last year's group, as well as the original design specifications of the ESRB, it was determined that the building continually required 54,000 CFM of makeup air. With this information and the Bin data, the group calculated that the building required 8,000 MBTU per year to heat the incoming air to 65°F.

The next step in the analysis was to determine the amount of fuel required to provide this heat, and the cost of the fuel. The group began by meeting with Steve Peary of UM Facilities Management. From this meeting, the following information was obtained:

- Non-fuel cost (capital & maintenance) of providing steam is $5.17 per million BTUs (MBTU) delivered.
- The boiler has an efficiency of roughly 0.8.
- Enthalpy losses from pressure reducing station are negligible.
- Piping losses are negligible.
- Facilities Management uses a system called Energy Watchdog to track energy cost and usage.

Data available from EnergyWatchdog.com, gave information on all University of Maine fuel purchases in the form of fuel 'tickets' or bills. Bills for No. 6 fuel oil from the 12 month period beginning August 2006 were used to calculate the average cost paid by the university for that time period. The resulting cost was $47.42/barrel or $1.13/gallon.

The autumn of 2007 brought a significant escalation in the general level of fuel prices, and $1.13/gallon was no longer representative of the University of Maine's fuel costs. In January 2008, faced with continuing volatility in oil markets, we concluded that a cursory review of recent fuel tickets would be the best way to obtain an accurate measure of current fuel costs. From this, we determined that a figure of $1.80/gallon was adequate for the purpose of our analysis. Currently, the most recent data available, for the month of March 2008, showed a price of $73.23/barrel or $1.86/gallon for No. 6 Fuel Oil delivered to the University of Maine.

2.4. Heat Pump Data

Once we had established the cost of heating the building, it was necessary to determine how the installation of a heat pump would affect the heating costs. Continuously changing outdoor conditions made this particularly challenging. This was further complicated by the fact that the heat pump was a closed loop, in which any change of condition, would cause changes in other parts of the system which would then lead to other changes in the loop. It was often necessary to use multiple iterations for each condition, to converge on a result.

After much consideration and analysis of the dynamics of the heat pump system, it was determined that the heat pump capacity was limited by the following three factors:

- Building Heat Requirements
- Compressor Load
- Energy Available to Evaporator

Our analysis had to account for these three factors.

2.5. Building Heat Requirements

Following the analysis discussed in sections 2.2 and 2.3, the amount of energy required to heat the makeup air to 65°F may be calculated. This limits the amount of energy provided by the heat pump; when outdoor temperatures are relatively warm (greater than 30F), it would be undesirable to heat the air beyond 65°F.

This part of the analysis was further complicated by the fact that the building heat load was calculated at the condenser, whereas the compressor load and the energy available to the evaporator were calculated at the evaporator. The condenser load is higher than the evaporator load because the compressor dissipates electrical heat into the evaporator. To find the evaporator load which would heat the air to 65°F, the electrical heat had to be subtracted from the building heat requirement. The electrical heat is a function of the compressor load and efficiency, which is a function of the operating conditions. As temperatures rose, the operating conditions became a function of the building heat requirement, and this led to an iterative solution. Compressor efficiencies had to be calculated (see 2.6) for a given evaporator load; the new efficiencies provided a new evaporator load requirement. This process was repeated iteratively until the results converged to a solution.

2.6. Compressor

Smardt, the vendor of the TurboCor compressor, provided us with a computer program called *TurboCor Selection Software [8]*, which provides the compressor performance data for various refrigerant states. From this data, a chart was created which provides the loads and efficiencies at various Saturated Suction Temperatures [SST] and Saturated Discharge Temperatures [SDT]. The group was then able to determine the optimum operating conditions of the compressor for a variety of outdoor air temperatures, and the corresponding maximum loads and efficiencies. The *TurboCor Selection Software* was also used to calculate efficiencies when the conditions were such that the compressor should operate below maximum capacity.

2.6.1. Energy Available to Evaporator

Another limitation on the capacity of the heat pump is the amount of energy which the evaporator can extract from the exhaust air at a given SST. At conditions where this became a limitation, the (compressor-optimized) SST was lowered to allow more heat transfer from the evaporator. Lowering the SST came at the expense of compressor efficiency.

2.7. Total Heat of Rejection (Condensor Heat Output)

The spreadsheet (see Appendix A) was setup so that the minimum of the three performance limitations, would dictate the performance of the compressor at a given condition. The minimum of the three performance limitations, taken at the evaporator, was referred to as the evaporator load at operating conditions. The electrical heat dissipated was added to the evaporator load to determine the Total Heat of Rejection (THR).

2.8. Electricity Costs

In order to compare the cost of heating the building with the heat pump, to the cost of heating with steam, it was necessary to determine the cost of the electricity required to run the compressor. The electricity load for each bin was calculated as the product of the evaporator load at operating conditions and the efficiency (kW-hr/ton) at these conditions. A figure of $0.10/kW-hr was used for the cost of electricity at the University of Maine.

2.9. Annual Savings

To calculate the total cost of heating the makeup air with the compressor, the cost of the steam required to provide the remaining heat was calculated. The total cost of heating the makeup air with the compressor was subtracted from the total cost without the compressor to provide the annual savings at each bin number. The bins were then summed to give the annual savings, resulting in an annual heating savings of $84,000. Because, this figure was more than enough to justify the capital cost, a conservative estimate of $20,000 was used for the cooling cost savings. James LaBrecque, an expert in heating and refrigeration, estimates the annual cooling savings to be $40,000 - $60,000, due to the reduced demand for the low COP glycol absorber.

2.10. Financial Analysis

From the annual fuel savings estimate, the installation costs of $195,000 and maintenance costs of $5,000 per year, the following financial parameters were calculated:

- Payback Period: 1.97 years
- Rate of Return: 78%
- Net Present Value: $1,030,447

See Appendix A for the in depth savings calculations.

2.11. Environmental Benefits

The environmental benefits of this project stem from the high COP of the compressor, and the relatively low emissions from electricity generation relative to those of the steam plant. Preheating the air to 65°F without the heat pump produces 890 tons of carbon dioxide emissions and 10.6 tons of sulfur dioxide emissions. With the heat pump, carbon dioxide emissions are reduced by 611.5 (a 69% reduction) tons and sulfur dioxide emissions are reduced by 9.45 tons (an 85% reduction).

2.12. Future Implementation with Similar Applications

One of the major advantages of the project's design is future applications in other buildings, with minimal additional engineering design required. An important design goal in this project was to design a compressor/receiver assembly with small dimensions such that it could be conveniently placed in virtually any building in need of a heat pump. A TurboCor compressor allows the system to be much smaller than alternatives; its placement on top of the receiver tank, along with the most complicated portion of the piping and valve system, limits the amount of planning and installation necessary for other buildings. The following buildings on the University of Maine campus would benefit from a heat pump: Aubert Hall, Hitchner Hall, Sawyer Environmental, Wells Commons, and the Advanced Manufacturing Center. All of these buildings have somewhat high to very high ventilation rates and a heat pump would provide substantial returns.

3. Concept & Design Description

3.1. Heat Pumps: How They Work

Heat pumps use the refrigeration cycle to transfer heat from one place to another. The refrigeration cycle is made up of four main components: the compressor, condenser, expansion valve, and evaporator. The cycle starts at the compressor, which increases the pressure and temperature of the refrigerant (now a hot gas) and forces it through the system. This gas flows through the condenser, which is a long coil with a large surface area so that heat can be released to the surrounding air. As the refrigerant loses heat, it changes state from a gas to a liquid. The refrigerant then flows through an expansion valve, which quickly drops the pressure and temperature. It leaves the expansion valve as a cold liquid with some gas. From there, it flows through the evaporator, which is another coil that allows the cold refrigerant to absorb heat from the surrounding air. Finally, the refrigerant returns to the compressor and the cycle repeats.

The key to the cycle's performance is in its use of the refrigerant's latent heat of vaporization to store energy. Since the refrigerant must boil to evaporate into a gas while in the evaporator, and its temperature stays at the boiling point while changing state, it is able to absorb a large amount of energy during the phase change. The energy associated with the phase change is known as the latent heat of vaporization, and is this same amount of energy released when the refrigerant changes back to a liquid in the condenser.

3.2. The Building's Current Heating System:

Currently the building does not have a heat pump and works as follows:

First outdoor air, also called makeup air, enters the building through intake louvers. The air enters two large air handlers, which contain steam pipes that heat the air to room temperature. The air then flows into the building's living space and is eventually sucked out through the exhaust ducts by the exhaust fans.

3.3. Advanced Heat Recovery: The Heat Pump

A heat pump uses exactly the same process as the refrigeration cycle to transfer heat from one place to another. Heat pumps can be used to cool or warm air, depending on which is desired and where the coils are placed. The heat pump in this project will mainly be used to warm incoming cold air. To accomplish this, the hot condenser coils are placed in the air handler to warm incoming air before it reaches the building's own steam pipes. This means that less heat is required from the steam pipes to warm the air to room temperature, which saves a great deal of energy.

On the other end of the cycle, the cold evaporator coils are placed in the exhaust ducts and absorb heat from the warm air exiting the building. The heat in the exhaust air, which would otherwise have been wasted, can be transferred through the system to the condenser coils. Since heat pumps are simply moving heat from one place to another, the only energy input needed is the energy needed to run the compressor. This makes it possible to get more energy back than is put in (since the vast majority of energy input is actually coming from otherwise wasted energy). Refer to Figure 1 for an in depth representation of how our system works. The Advanced Heat Recovery system will serve to heat and cool the Engineering and Science Research Building. Referring to Figure 1 below, the red arrows show the flow direction during heating and the blue arrows show the flow direction during cooling. The details of heating and cooling are described on the next page.

Heat Pump Schematic

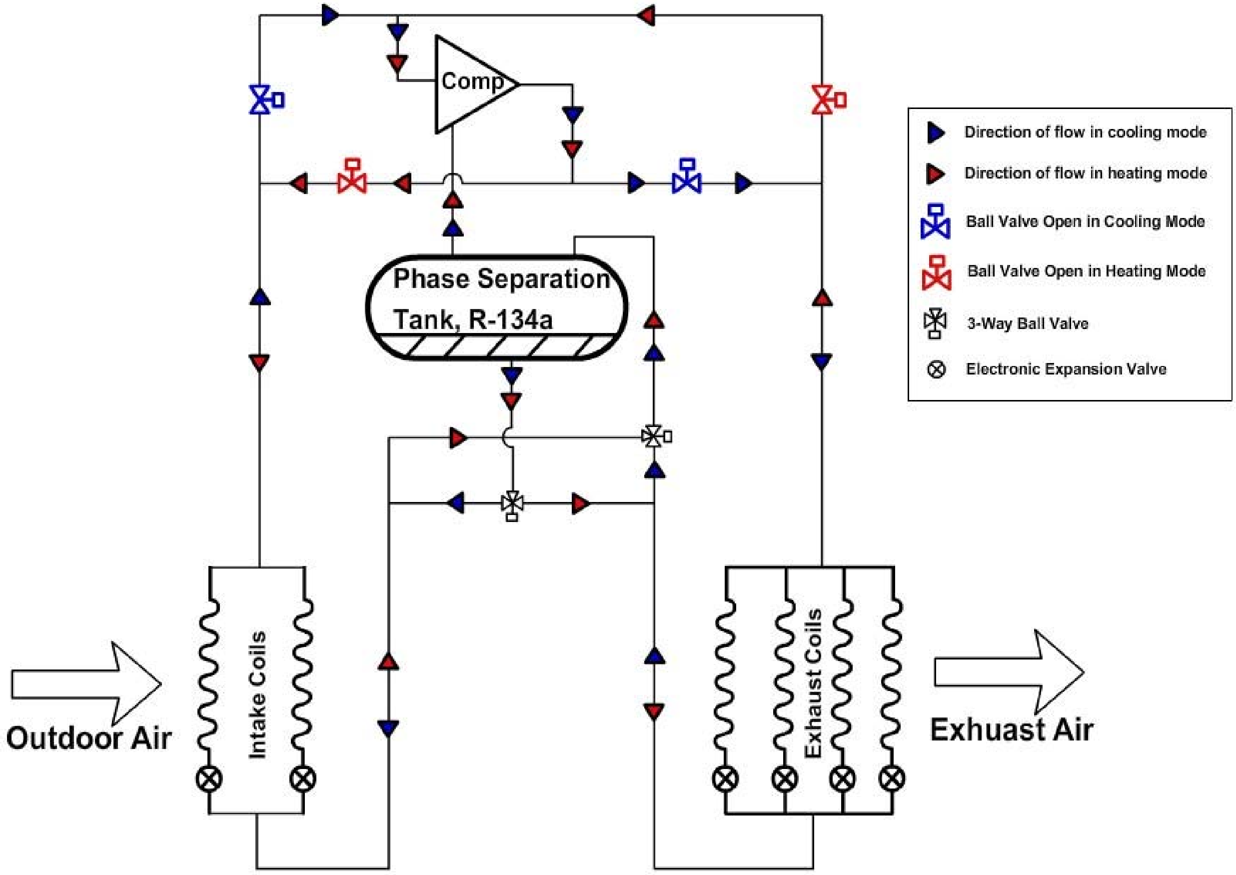


Figure 1: Schematic of Heat Recovery System

3.4. Heating

The compressed superheated refrigerant exits the compressor and travels through the two outdoor air coils where it enters as saturated vapor. After traveling through the coils, the refrigerant's temperature drops causing the refrigerant to change from a saturated phase to a liquid. The liquid refrigerant travels through the electronic expansion valves causing a drop in pressure.

Finally the saturated refrigerant returns to the receiver through the flash gas return.

Liquid refrigerant travels from the bottom of the receiver to the four exhaust air coils where it first encounters an electronic expansion valve. Exhaust heat is recovered through the exhaust air coils and the now superheated refrigerant returns to the compressor through the suction line.

3.5. Cooling

The process is the same as heating except the compressed refrigerant now travels through the four exhaust air coils instead of the outdoor air coils. The liquid line travels from the receiver to the outdoor air coils. As it passes through the coils, it absorbs heat from the incoming air which drops the temperature of the ventilation air, thereby cooling the building. Finally the refrigerant returns to the compressor through the suction line.

Ball valves on Figure 1 are used to reverse the direction of flow. This is necessary when the system switches from heating mode to cooling mode and vice versa.

3.6. Green Loan Fund

The Green Loan Fund was created through collaboration between the University of Maine Foundation and the University of Maine. Under the terms of the agreement, the Foundation will loan up to $300,000 to the University for projects designed to reduce energy consumption and improve campus sustainability [7].

The Green Loan is going to be the main contributor to the funding of the project. Much of our work this semester has been involved with several members of the Green Loan Fund Committee. They have been very supportive our project, as it meets the criteria for receiving funding and is beneficial to the state of Maine in preserving the environment. Our project could set precedence for future energy-saving projects at the University of Maine; we estimate that there are currently at least five buildings on campus that could benefit from a similar heat recovery system.

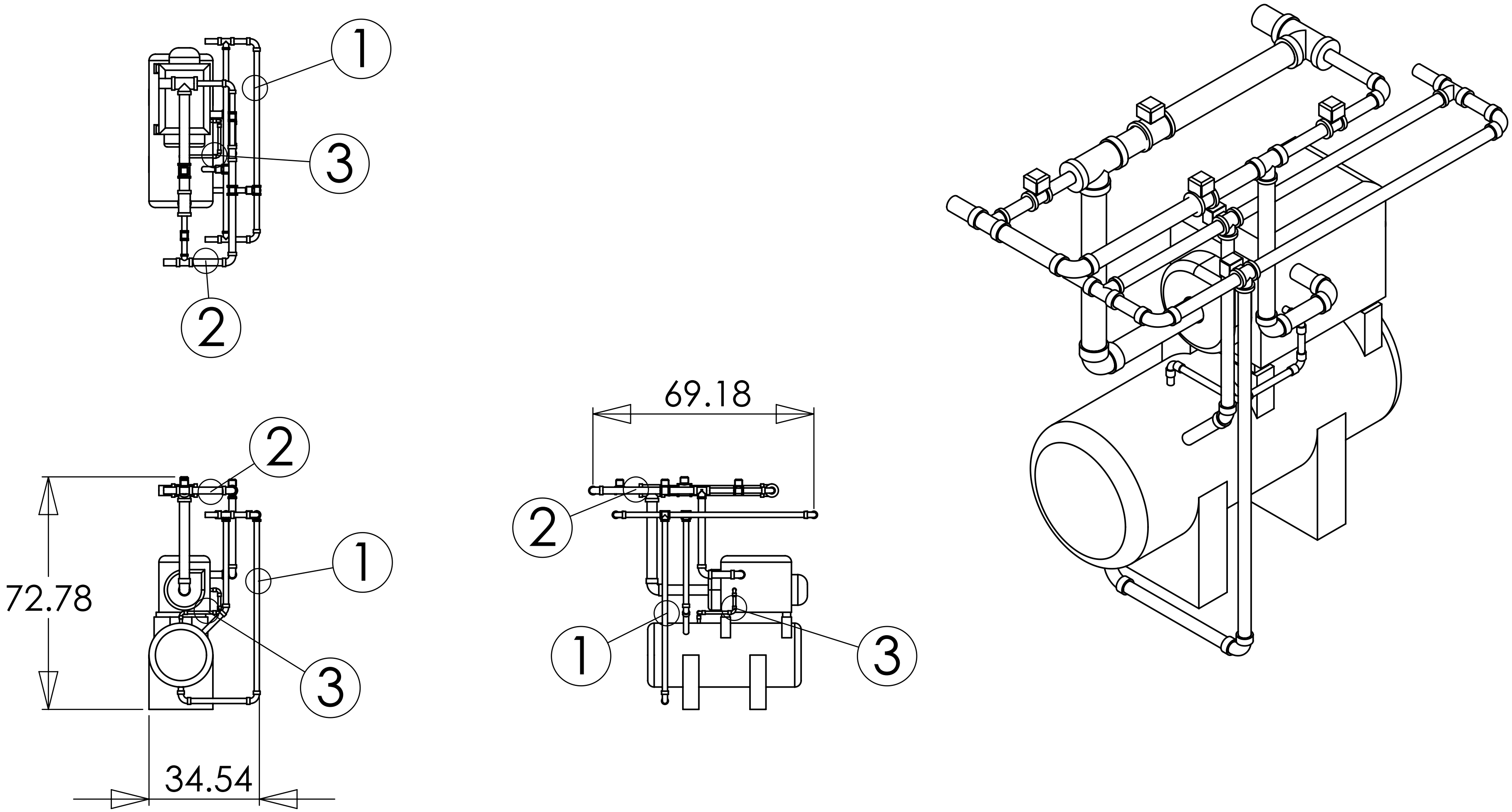
3.7. Drawings and Materials Package

3.7.1. Piping

Note: The following prices were confirmed with Stewart Harvey, Facilities Management Associate Director for Maintenance at the University of Maine.

		See Reference [1]				
		Copper	**Total**		**Steel Sch. 40**	**Total**
Compressor to Supply Air Coils:						
90 ft pipe (Size: 2 1/8")		$132 / 10ft	$1,188		$52.12 / 5ft	$938.16
6 90° long radius elbows		$19.75 ea.	$118.50		$18.05 ea.	$108.30
1 Tee		$17.87 ea.	$17.87		$27.79 ea.	$27.97
1 45° elbow		$9.38 ea.	$9.38		$11.00 ea.	$11.00
12 copper or 35 steel Couplings		$4.71 ea.	$56.52		$14.71 ea.	$514.85
		Subtotal:	**$1,390.27**			**$1,600.28**
Supply Air Coils to Receiver:						
90 ft pipe (Size: 1 5/8")		$78 / 10ft	$702		$38.49 / 5ft	$692.82
6 90° long radius elbows		$9.02 ea.	$54.12		$11.54 ea.	$69.24
1 Tee		$10.89 ea.	$10.89		$18.00 ea.	$18.00
1 45° elbow		$5.29 ea.	$5.29		$6.50 ea.	$6.50
12 copper or 35 steel Couplings		$2.91 ea.	$34.92		$7.50 ea.	$262.50
		Subtotal:	**$807.22**			**$1,049.60**
Receiver to Exhaust Air Handler Coils:						
100 ft pipe (Size: 1 5/8")		$78 / 10ft	$780		$38.49 / 5ft	$769.80
8 90° long radius elbows		$9.02 ea.	$72.16		$11.54 ea.	$92.32
3 Tees		$10.89 ea.	$32.67		$18.00 ea.	$54.00
1 45° elbow		$5.29 ea.	$5.29		$6.50 ea.	$6.50
14 copper or 50 steel Couplings		$2.91 ea.	$40.74		$7.50 ea.	$375.00
		Subtotal:	**$930.86**			**$1,297.62**
Exhaust Air Handler Coils to Compressor:						
100 ft pipe (2X Size: 3 1/8")		$240 / 10 ft	$4,800		$93.39 / 5ft	$3,735.40
16 90° long radius elbows*		$49.83 ea.	$797.28		$32.08 ea.	$513.28
6 Tees*		$52.35 ea.	$314.10		$38.87 ea.	$233.22
2 45° elbow*		$27.28 ea.	$54.56		$22.00 ea.	$44.00
28 copper or 100 steel Couplings*		$15.29 ea.	$428.12		$19.42 ea.	$1,942.00
		Subtotal:	**$6,394.06**			**$6,468.10**
		Total:	**$9,522.41**			**$10,415.60**

Section 3.7.2: Compressor and Receiver Package Drawings



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MFG APPR.		
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COMMENTS:
See Drawings 1 through 3 for details on each piping assembly

Advanced Heat Recovery Project

Compressor-Receiver Assembly

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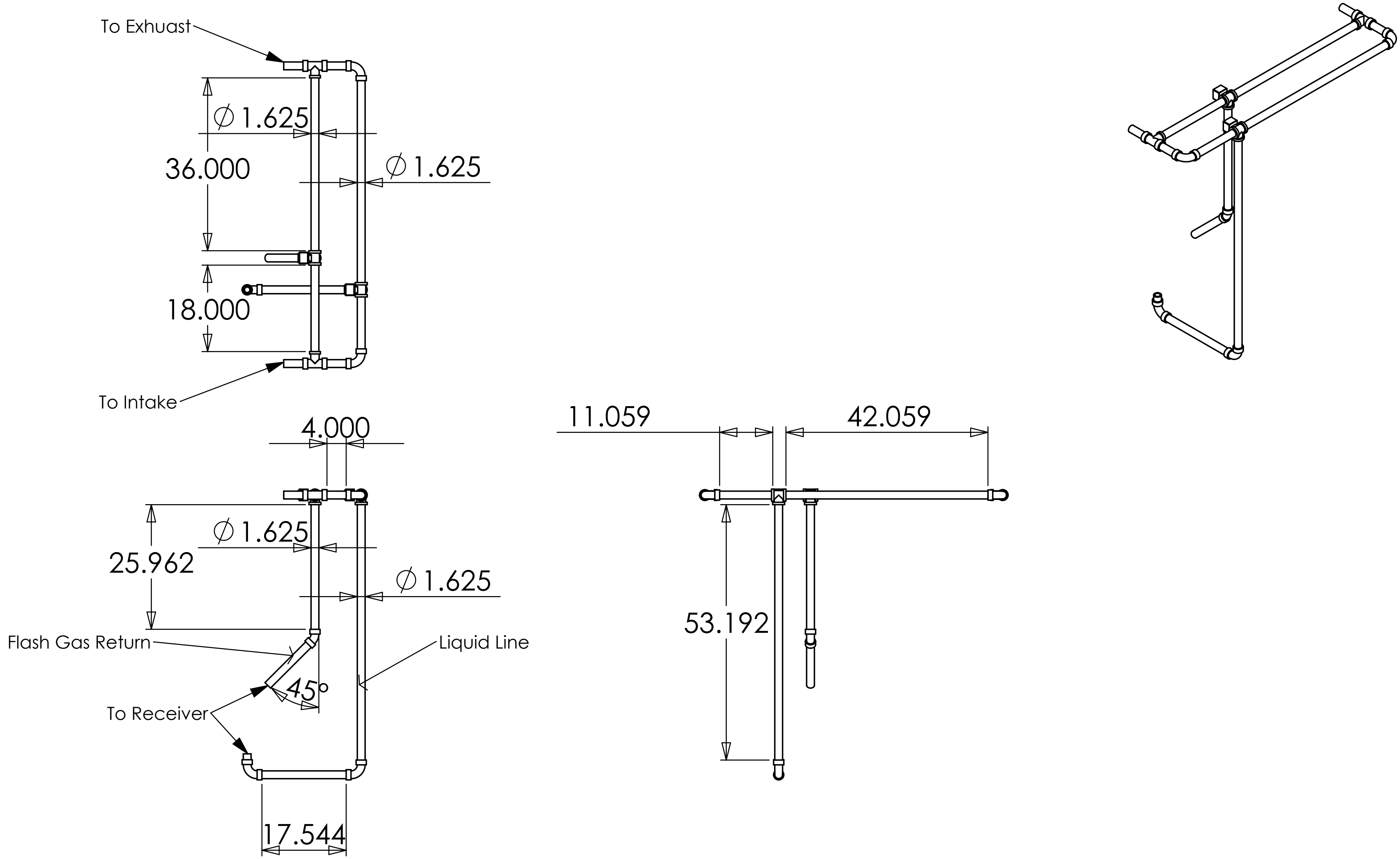

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Advanced Heat Recovery Project

Detail 1: Flash Gas Return Piping

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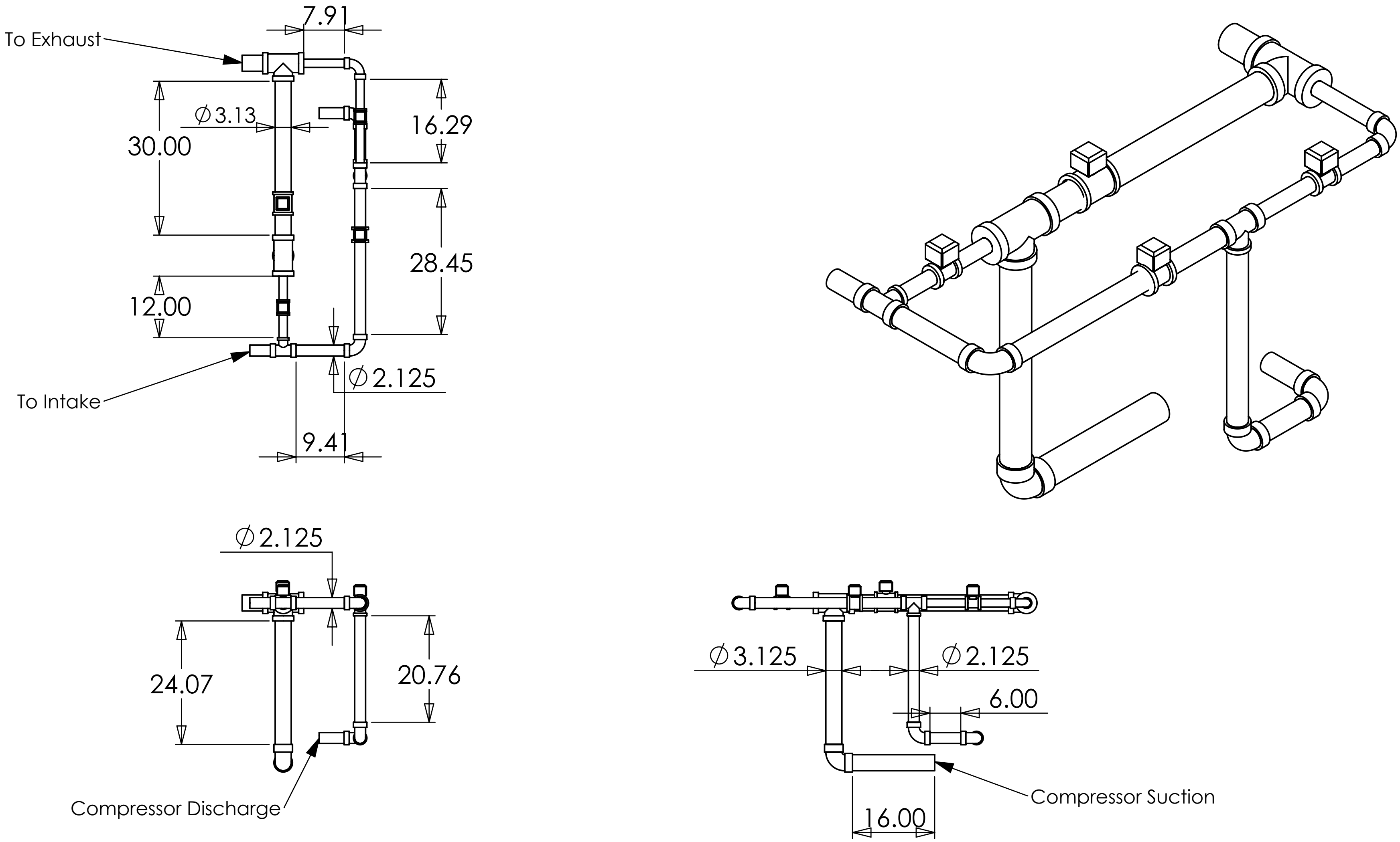

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COMMENTS:

Advanced Heat Recovery Project

Detail 2: Compressor Suction Piping

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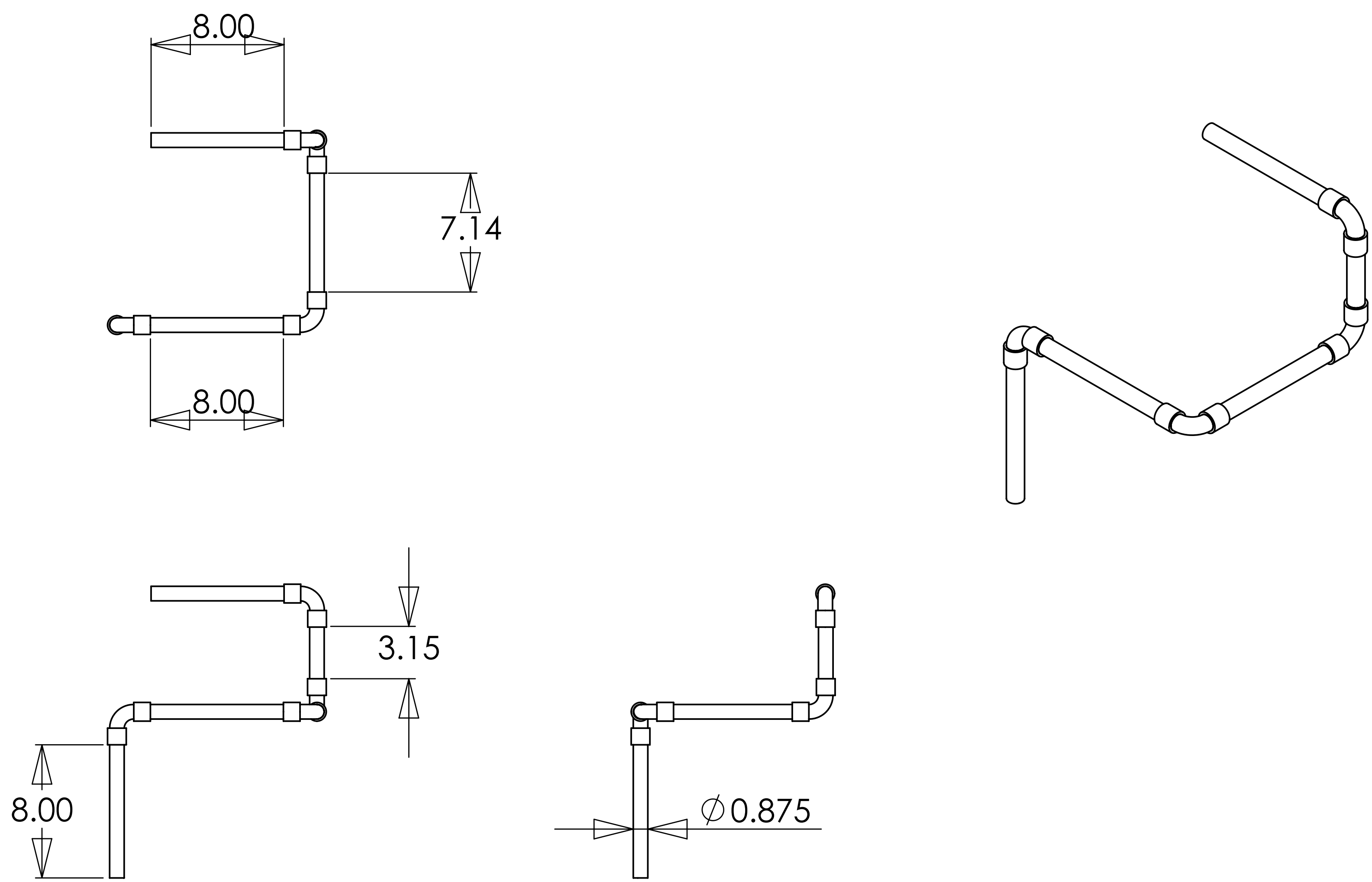

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MATERIAL --

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COMMENTS:

Advanced Heat Recovery Project

Detail 3: Economizer Piping

SIZE	DWG. NO.	REV.
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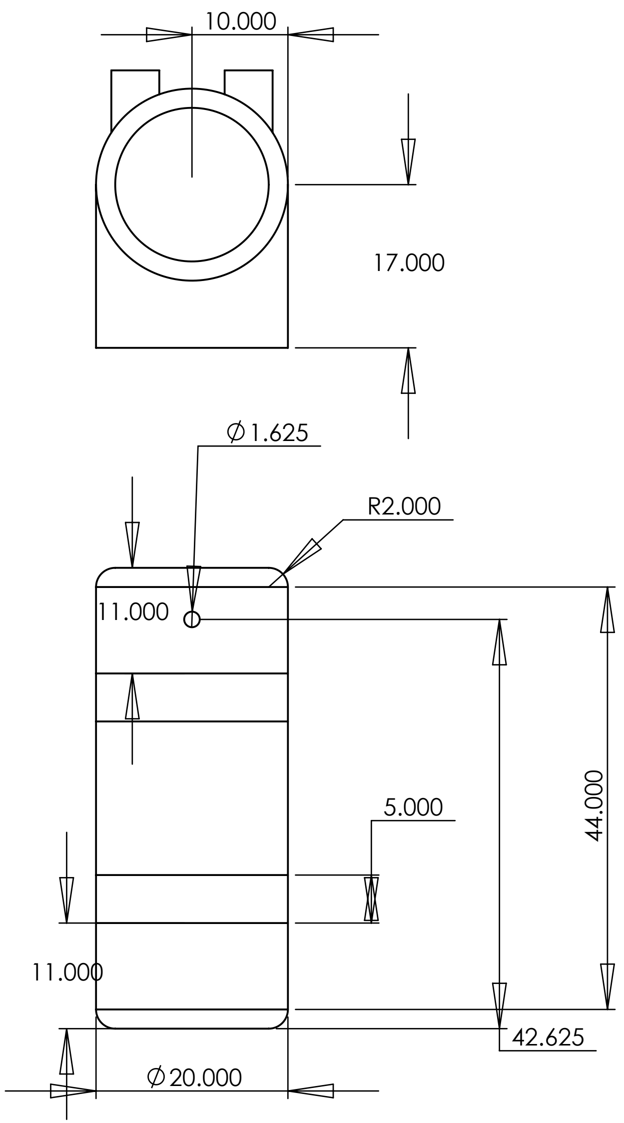

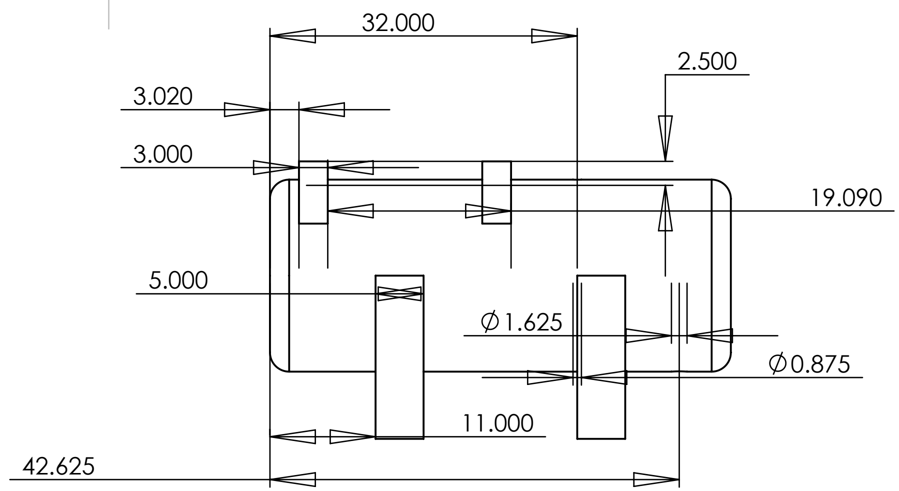

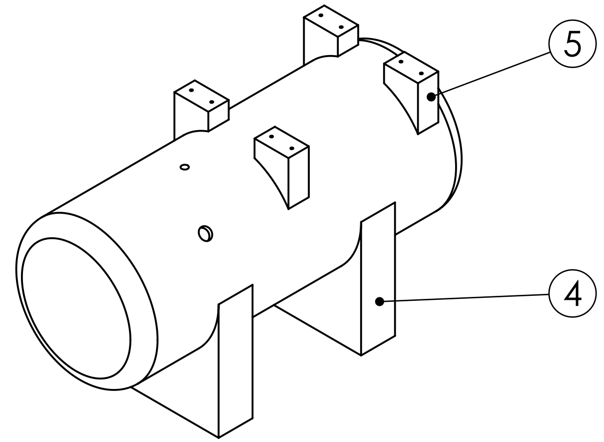

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DEBUR AND BREAK SHARP EDGES

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REVISION

Advanced Heat Recovery Project

	NAME	SIGNATURE	DATE
DRAWN			5-6-2008
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Receiver Tank

Comments:
See Details 4 and 5 for top and bottom mount specifications

DWG NO. 5

A4

WEIGHT:

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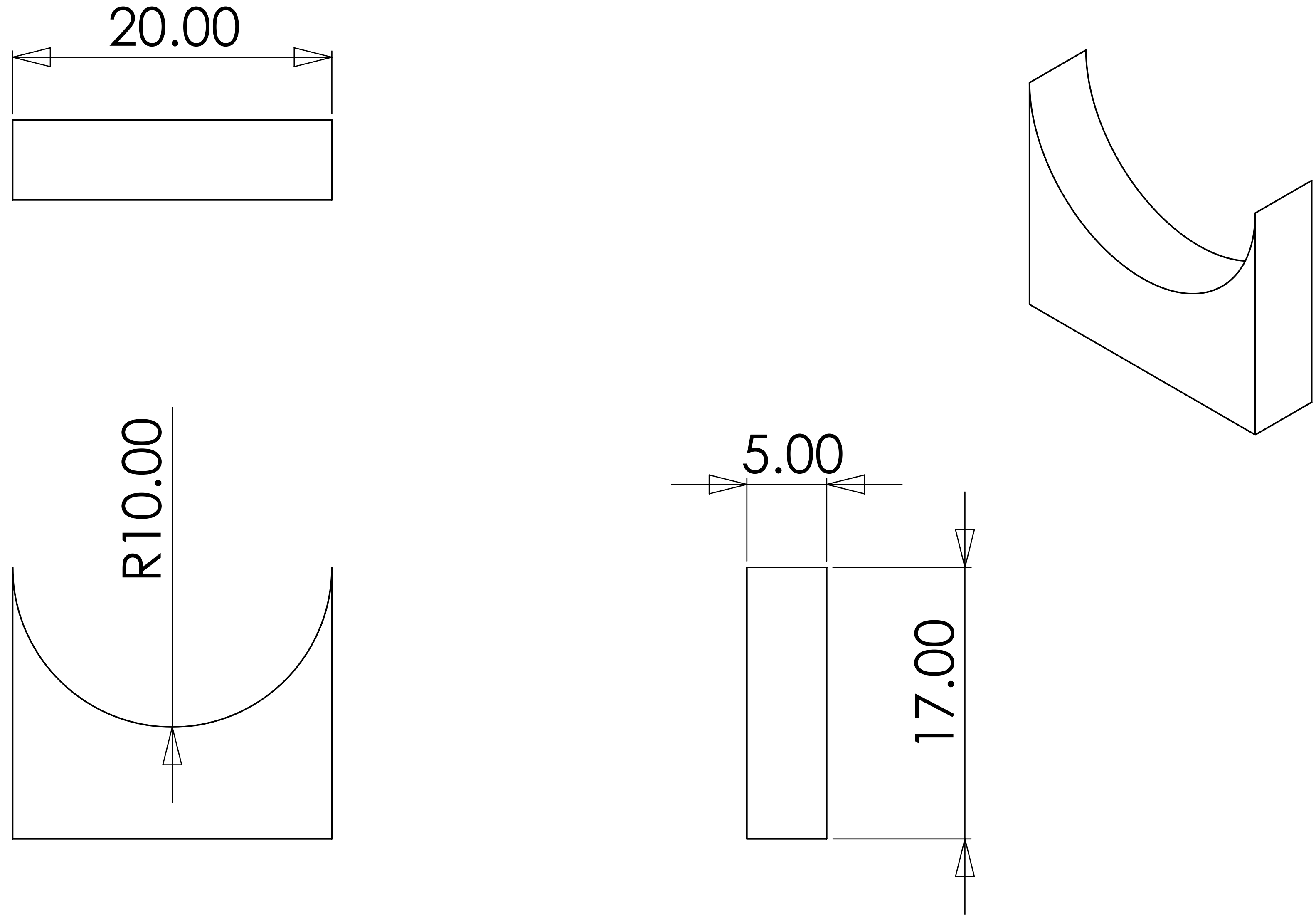
SHEET 1 OF 1



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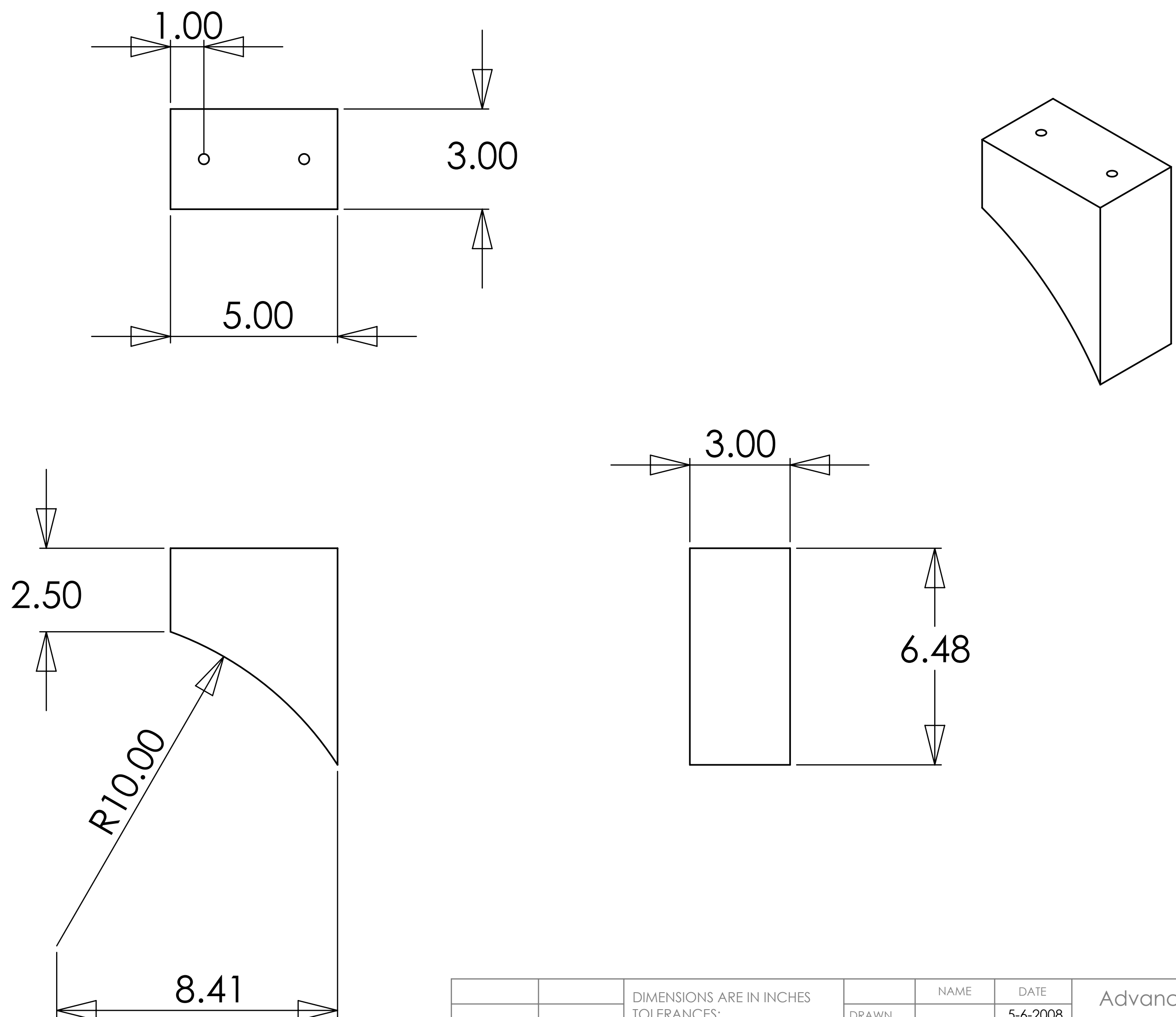
COMMENTS:

Advanced Heat Recovery Project

Detail 4: Receiver Base Mount

SIZE	DWG. NO.	REV.
A	6	

SCALE: 1:10 | WEIGHT: | SHEET 1 OF 1



PROPRIETARY AND CONFIDENTIAL

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NEXT ASSY	USED ON
APPLICATION	

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

MATERIAL --

FINISH --

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		5-6-2008
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

Advanced Heat Recovery Project

Detail 5: Receiver Tank Top Mount

SIZE	DWG. NO.	REV.
A	7	

SCALE:1:5 | WEIGHT: | SHEET 1 OF 1

3.8. Design Alternatives

3.8.1. Coils

The design of the coils for this project posed a particular challenge, because of their unusually large size. Direct expansion coils are primarily used in automotive radiators and are therefore much smaller than those required for this application. The coils were required to achieve the heat transfer required at design conditions (138 tons total for the evaporators and 156 tons for the condensers), while minimizing the pressure drop in the air and refrigerant. This had to be done in consideration of the cost of the coils, as well as the weight, which would be a factor in installation costs.

Initial Attempts

The first attempt at designing coils was to contact *Trane*, which was the manufacturer of the air handling units in which the coils were to be installed. Unfortunately *Trane* was unable to provide direct expansion coils of this size. After this, we contacted several large coil manufacturers including CanCoil and AlumaCool, but both were not responsive to our inquiries, presumably because they were unable to provide us with the required coils.

Micro-Channel Coils

The next attempt at designing coils was to use micro-channel coils. Micro-channels coils utilize very small passages in which refrigerant flows. This provides for a very large surface area for heat transfer and reduced weight.

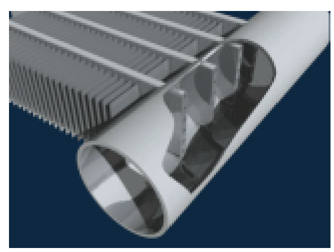


Figure _: Micro-Channel Coils [9]

These coils were designed using a program called *InstinctCode* provided by Norm Costello. This program's input requires properties of the compressor, the air and the refrigerant, and coil geometries, and provides the output of the system performance. This allows the user to design a coil which meets a given set of specifications. This program factored a tremendous of information into its calculations, which led to considerable challenges for designing the coils. On multiple occasions the group encountered errors, and had to have remote meetings with Norm Costello in order to overcome challenges. Once the coils were designed there was an additional issue of finding someone to manufacture them. The process of designing the micro-channel coils ourselves proved especially challenging, but it did give the group a great understanding of the dynamics of coils.

Conventional Coils

While proceeding with the design of micro-channel coils using *InstinctCode*, the group continued to investigate alternatives. In early April, with the end of the semester nearing, the group contacted four vendors in the hopes that at least one of them would be able to provide us with the required coils. We were able to contact three of the four vendors and sent our coil design specifications.

Chip Bond of *Briggs AC*, the local representative for *McQuay*, quickly responded to our request, but was unable to provide coils of the desired size for use with R-134a refrigerant. Seaver Woolfolk of Super Radiator Coils, responded several weeks later with a full quote for coils meeting the desired specifications. This gave the group a workable option for coils, with which to compare alternatives. Due to the reasonable cost and performance of this coil it was determined that this was the best option available.

3.8.2. Compressor

The design options for the compressor were very limited for this year's analysis. TurboCor is the only provider of large oil-free HVAC compressors, and an oil-free compressor was an essential component of the design. A conventional compressor would have introduced significant design complexity, due to the need for oil return to the compressor through the refrigerant pipes.

3.8.3. Air to Air Heat Exchanger

An air to air heat exchanger would not be desirable for this application because of the physical separation between the exhaust and intake air ducts, and the limited effectiveness of air to air heat exchangers. There would be great cost in running ductwork across the equipment room, and the system would not be able to provide nearly as much heat as a direct expansion heat pump.

3.8.4. Water-Glycol System

The original design of the building called for a water-glycol heat exchanger system, but this was never installed due to budget limitations. A water-glycol system simply transports heat using a water-glycol mixture which prevents freezing. The water-glycol mix is pumped between coils in the exhaust air and intake air. Because it does not use latent heat, more fluid is required to move the heat, and less heat can be added to the building. In mild temperatures when the temperature difference between the intake and exhaust air are lower, the efficiency is reduced because less heat is transferred, but the pump continues to run.

4. Conclusions and Recommendations

4.1. Strengths

The advantage of this system is the large amount of energy which can be recovered from the exhaust in the winter, and removed from the intake air in the summer. This will result in fuel savings estimated at $99,000 per year. The expected capital cost of this system is $195,000 providing a payback period of roughly 2 years. In addition to providing a very good return on investment for the University of Maine, the heat recovery system will significantly reduce emissions.

4.2. Weaknesses

The weakness of this system stems primarily from the relative novelty of the system. Because of this, there is a limited availability of qualified service personnel, as well as limited field experience and knowledge with direct expansion (DX) heat recovery systems. Finally, the entire system design relies on the use of an oil free compressor, which is only manufactured by Danfoss TurboCor.

4.3. Recommendations for Improvement:

The University of Maine could reduce energy costs and harmful environmental emissions by applying similar designs to other buildings on the University of Maine campus, which require high ventilation air rates. The following buildings on the University of Maine campus would benefit from the ESRB heat pump design: Aubert Hall, Hitchner Hall, Sawyer Environmental, Wells Commons, and the Advanced Manufacturing Center. Minimal engineering design would be required to modify the ESRB heat recovery design to meet the buildings' requirements. Piping and coil designs would require modification due to different energy requirements and geometries, but the compressor and receiver package would remain the same.

4.4. Reflections on the Design

During the fall semester, much of the group's time was spent gaining an understanding of HVAC systems. The conventions and terminology of HVAC professionals proved, especially challenging, and contrasted, with the standard conventions of science and engineering. As an example, in HVAC, pressure is measured in °F of saturation temperature and energy is measured in tons of ice.

The design of this system proved much more challenging than expected. The two main issues were the analysis of the system performance, and the sourcing of parts. The performance of the system was analyzed using calorimetry software [8] provided by TurboCor, and analyzing the resulting data using excel, as described in section 2. This analysis proved quite complicated, because of the various inter-related parameters of the system, which varied with temperature.

In addition, the selections of components were dependent on this analysis, and the analysis was dependent on the selection of components. A rough estimate of the time involved in this analysis is 300 engineering-hours. It should be kept in mind that this was with a team of four inexperienced engineers, working off and on. A smaller team, or even an individual working full time, and with slightly more experience, may have been able to accomplish this in 100 engineering-hours or less.

Another problem faced was the selection and sourcing of parts. The selection and sourcing of coils proved quite challenging, due to the absence of any similar applications. The attempt at designing micro-channel coils consumed a tremendous amount of time. This is estimated at 200 engineering-hours. In our attempts to select conventional coils, many vendors were not responsive to our inquiries. This may have been caused by the fact that vendors may have known that we were students, and though that they were not likely to make a sale, or by their inexperience with the design of coils for this type of application. The design of conventional coils and inquiries to vendors, consumed an estimated 80 hours, and led to only one received quote.

Creating three-dimensional drawings of the system proved useful in visualizing the installation, and any potential challenges. It should continue to be very useful when the system is installed. This consumed a large amount of engineering time, roughly 250 hours. Three-dimensional modeling allowed members of the group to gain experience in 3D drafting, but would prove expensive if performed by engineers. It may be possible to reduce the hours and cost, by outsourcing this activity, but that would also pose further logistical challenges.

A large amount of engineering time was spent designing the various components of this system and evaluating the system's performance. The fact that we could only devote a small part of our time made it particularly challenging to keep the project moving as productively as desired. Due to time constraints we frequently had to stop our work and return to it later. The group's lack of HVAC knowledge also proved a significant challenge in designing this system. The engineering-time in a future system would be significantly less because of the tools and knowledge that the group has acquired.

References

[1] Mechanical Engineering Senior Design. “Advanced Exhaust Air Energy Heat Pump.” [Online] 13 December 2007. <http://www.umaine.edu/MechEng/Peterson/Classes/Design/2006_7/Projects/HeatPump/Heat%20Pump%20Report.doc.>

[2] Meeting with Stewart Harvey and Steve Peary on September 25, 2007.

[3] Energy Information Administration “Voluntary Reporting of Greenhouse – Fuel and Energy Source Codes and Emission Coefficients” [Online] 13 December 2007 http://www.eia.doe.gov/oiaf/1605/coefficients.html

[4] Energy Information Administration “Electric Power Annual 2000 Volume II: Table A3” [Online] 13 December 2007 http://www.eia.doe.gov/cneaf/electricity/epav2/html_tables/epav2ta3p1.html

[5] One Billion Bulbs – An Energy TrekTM Initiative “Summary Statistics for Maine” http://www.onebillionbulbs.com/Stats/State/ME

[6] Maine Public Utilities Commission “Residential and Small Non Residential Standard Offer Service: Consumer Information about Your Electricity Supply” [Online] Accessed: 13 December 2007; Published: 10 November 2006 www.maine.gov/mpuc/industries/electricity/standard_offer/CMPres2006oct.pdf

[7] University of Maine Sustainability Alliance. “Green Loan Fund: Project Proposal Criteria and Process.” [Online] 13 December 2007.

<http://www.sustainability.umaine.edu/green_loan.html.>

[8] Danfoss Turbocor Compressors, Inc. Turbocor Selection Software Version 1.0.55.

[9] http://www.commercial.carrier.com/commercial/hvac/general/0,,CLI1_DIV12_ETI10722,00.html

Appendix A: Heat Pump Analysis Spreadsheet

Heating Plant Fuel Oil Tickets August 2006 - June 2007

Month	Ticket #	Date	Units	Cost	Cost/Barrel
	Barrels of #6 Fuel Oil				
Aug-06					
	6126934/6127265/6129172/6130329	8/19/2006	1172.15	$66,263.05	$56.53
	6130622/6131168/6131432	8/23/2006	581.26	$31,689.80	$54.52
	6132177/6133127/6133507/6134442	8/31/2006	975.91	$51,963.03	$53.25
	Bill Total		**2729.32**	**$149,915.88**	**$54.93**
Sep-06					
	6138665	9/14/2006	196.24	$8,977.97	$45.75
	6139269	9/15/2006	195.43	$8,856.88	$45.32
	6144499	9/29/2006	196.83	$8,465.65	$43.01
	6135358	9/5/2006	391.93	$20,055.05	$51.17
	6135793	9/6/2006	195.4	$9,764.13	$49.97
	Bill Total		**1175.83**	**$56,119.68**	**$47.73**
Nov-06					
	6170899/6171991/6177994/6172482	11/22/2006	1166.78	$71,177.78	$61.00
	6173519/6176405/6176405/6177993	11/30/2006	1959.18	$71,095.91	$36.29
	6163482/6164556/6165528/6166170	11/9/2006	1754	$83,606.43	$47.67
	6167706/6168324/6168962/6169375	11/16/2006	1174.07	$54,800.88	$46.68
	Bill Total		**6054.03**	**$280,681.00**	**$46.36**
Dec-06					
	6188739/6189569/6191581/6192794/61 93190	12/21/2006	1561.97	$71,018.38	$45.47
	6193929/6194228/6195049/6195728/61 96472	12/28/2006	1370.13	$61,054.07	$44.56
	6197358/6198365/6199157/6199806/62 00049	12/31/2006	1368.08	$58,738.04	$42.93
	6178704/6179827/6182137/6182136/61 82827	12/7/2006	1953.53	$91,525.67	$46.85
	6184226/6184720/6185463/6186658/61 87898	12/13/2006	1956.55	$89,523.53	$45.76
	Bill Total		**8210.26**	**371859.69**	**$45.29**
Jan-07					
	7049115/7051318/7052611/7052520	1/28/2007	1173.64	$48,841.19	$41.62
	7030400/7031574/7032122/7032993/70 34043	1/5/2007	2150.31	$90,967.47	$42.30
	7035282/7036526/7037165/7038467/70 39638	1/12/2007	1950.73	$80,597.81	$41.32
	7040624/7041965/7043548/7044422/70 45654	1/19/2007	2157.19	$88,276.80	$40.92
	7046041/7046140/7046785/7047453/70 48344	1/23/2007	1951.29	$80,100.39	$41.05
	Bill Total		**9383.16**	**$388,783.66**	**$41.43**
Feb-07					
	7081111/7082703	2/28/2007	978.26	$45,125.15	$46.13
	7057011/7059045/7060741/7061759/70 62247/7062881/7064512	2/8/2007	3128.98	$142,760.39	$45.63
	7066667/7067457	2/12/2007	1173.97	$56,585.28	$48.20
	7068673/7070021/7071179/7071312/70 72607/7073209	2/19/2007	3129.63	$148,257.09	$47.37
	7073881/7076000/7077413/7078288/70 80092	2/26/2007	2538.39	$117,045.46	$46.11
	Bill Total		**10949.23**	**$509,773.37**	**$46.56**

Mar-07					
	7101601/7103723/7105286	3/29/2007	1570.18	$77,292.27	$49.23
	7106434	3/30/2007	196.48	$9,951.71	$50.65
	7083578/7084750	3/2/2007	391.21	$17,776.77	$45.44
	7087055/7088291/7089134/709013;	3/9/2007	2952.04	$135,893.45	$46.03
	7092015/7092970/7094314/709532!	3/15/2007	1370.68	$63,252.00	$46.15
	7095822/7096596/7097256/7098258/70				
	99531/7100218/7102826	3/26/2007	2341.99	$109,529.99	$46.77
	Bill Total		**8822.58**	**$413,696.19**	**$46.89**
Apr-07					
	7123479/7124420	4/30/2007	583.83	$32,200.79	$55.15
	7107399/7108569/7109356/7110911/71				
	11978	4/9/2007	2155.16	$100,036.23	$46.42
	7112400/7114095	4/12/2007	1365.99	$67,057.30	$49.09
	7116368/7117077/7118023/711973(	4/20/2007	2155.94	$112,428.55	$52.15
	7120691/7121520/712226€	4/25/2007	785	$42,113.28	$53.65
	Bill Total		**7045.92**	**$353,836.15**	**$50.22**
May-07					
		5/30/2007	194.9	$11,017.10	$56.53
	7125240/7125983/712666€	5/3/2007	976.44	$44,368.90	$45.44
	7128014/7128759/7128760/7130159/71				
	30862	5/11/2007	1370.21	$75,199.40	$54.88
	7133281/7136039/7136612	5/22/2007	1747.92	$101,290.18	$57.95
	7137787/7139077	5/29/2007	779.19	$45,083.24	$57.86
	Bill Total		**5068.66**	**$276,958.82**	**$54.64**
Jun-07					
	7152210/7151769	6/26/2007	584.35	$33,647.01	$57.58
	7141787/7142405	6/5/2007	587.69	$26,734.26	$45.49
	7144160/7145567	6/11/2007	391.41	$20,308.95	$51.89
	7146265/7146713	6/13/2007	389.93	$22,729.58	$58.29
	7149279/7151110	6/22/2007	584.24	$34,105.04	$58.38
	Bill Total		**2537.62**	**$137,524.84**	**$54.19**
		#6 Fuel Oil Total:	**61976.61**	**$2,939,149.28**	**$47.42**
				Price per gallon	$1.13

Data for July 2007-March 2008 from pro.energywatchdog.com

Start	End	Barrels	Cost	Cost/Barrel	Cost/Gallon
6/30/2007	7/31/2007	195.26	$12,561.08	$64.33	$1.53
7/31/2007	8/31/2007	3,528.03	$216,458.25	$61.35	$1.46
8/31/2007	9/30/2007	0			
9/30/2007	10/31/2007	977.64	$73,085.05	$74.76	$1.78
10/31/2007	11/30/2007	1,378.14	$107,880.44	$78.28	$1.86
11/30/2007	12/31/2007	6,079.09	$459,795.27	$75.64	$1.80
12/31/2007	1/31/2008	4,893.11	$368,308.47	$75.27	$1.79
1/31/2008	2/29/2008	2,745.11	$204,304.53	$74.42	$1.77
2/29/2008	3/31/2008	1,569.34	$122,268.56	$77.91	$1.86
Total		21,365.72	$1,564,661.65	$73.23	$1.74

Source of Data:
pro.energywatchdog.com: University of Maine Heating Plant Fuel Ticket Reports

Analysis of Savings in Energy Costs and Carbon Emissions for a Heat Pump in the Engineering Science Research Building at the University of Maine in Orono
Adapted by the 07-08 HP Group from a file created by Andrew Kitchen in Spring 2007

Variables			
Fuel (#6 Residual Oil)	Cost/gallon	$1.80	
	BTUs/gallon	150000	
	lbs/gallon (+/-5%)	8.35	
	MBTU/Gallon	0.15	
	Boiler efficiency	0.8	
	Fuel Cost per MBTU Delivered	$15.00	
	Other Cost per 1000 lb of steam delivered	$6.10	*Source 1*
	Other Cost per MBTU delivered	$5.17	
	lbs CO2/MBTU burned	173.906	*Source 2*
	lbs CO2/Gallon Burned	26.0859	
	Percent Sulfur by Weight (S)	2.00%	
	Sulfur Multiplier ((.157*S)=lbs SO2/gal)	15.7	*Source 3*
	lbs SO2 Emissions per gal	0.314	
Heat Pump	Cost/kWh	$0.10	
	lbs CO2/kWh	1.43	*Source 4*
	lbs SO2/kWh	2.89E-03	*Source 5*

Building Air

Cp of air (BTU/lbm F)	0.24
CFM	54,000
density of air (lbm/ft^3)	0.073
Mdot (lbm/hr)	236520
Tsupply air (F)	**65**
q=Mdot*Cp*(Tsa-Toa)	

Conversion Factors

293.07	kWh/MBTU
0.848	MBTU/lb steam
1200 BTU/hr=1 ton ***	

Supply air temperature should be maximized but the problem is that if it's too high, then the temperature cannot be accurately controlled.

Sources

Source 1: Stewart Harvey and Steve Pearty Meeting of 9-25-07
Source 2: http://www.eia.doe.gov/oiaf/1605/coefficients.html
Source 3: http://www.eia.doe.gov/cneaf/electricity/epav2/html_tables/epav2ta3p1.html
Source 4: http://www.onebillionbulbs.com/Stats/State/ME
Please note: lbs of Carbon Dioxide is not equivalent to lbs of Carbon
Source 5: www.maine.gov/mpuc/industries/electricity/standard_offer/CMPres2006oct.pdf - 2006-10-11

University of Maine ESRB Heat Pump Project

TT400 TurboCor compressor Performance operating with R-134a, 460vAC supply

Spreadsheet created by Andrew Bennett on Feb-5-08 updated 2/12/08 AK

Calculated using TurboCor Software

$$1\,\mathrm{kW} = 3412.1\,\frac{\mathrm{BTU}}{\mathrm{hr}}$$

Assumptions:

Liquid Temperature at EXV = SDT

Suction Superheat = 10 deg F

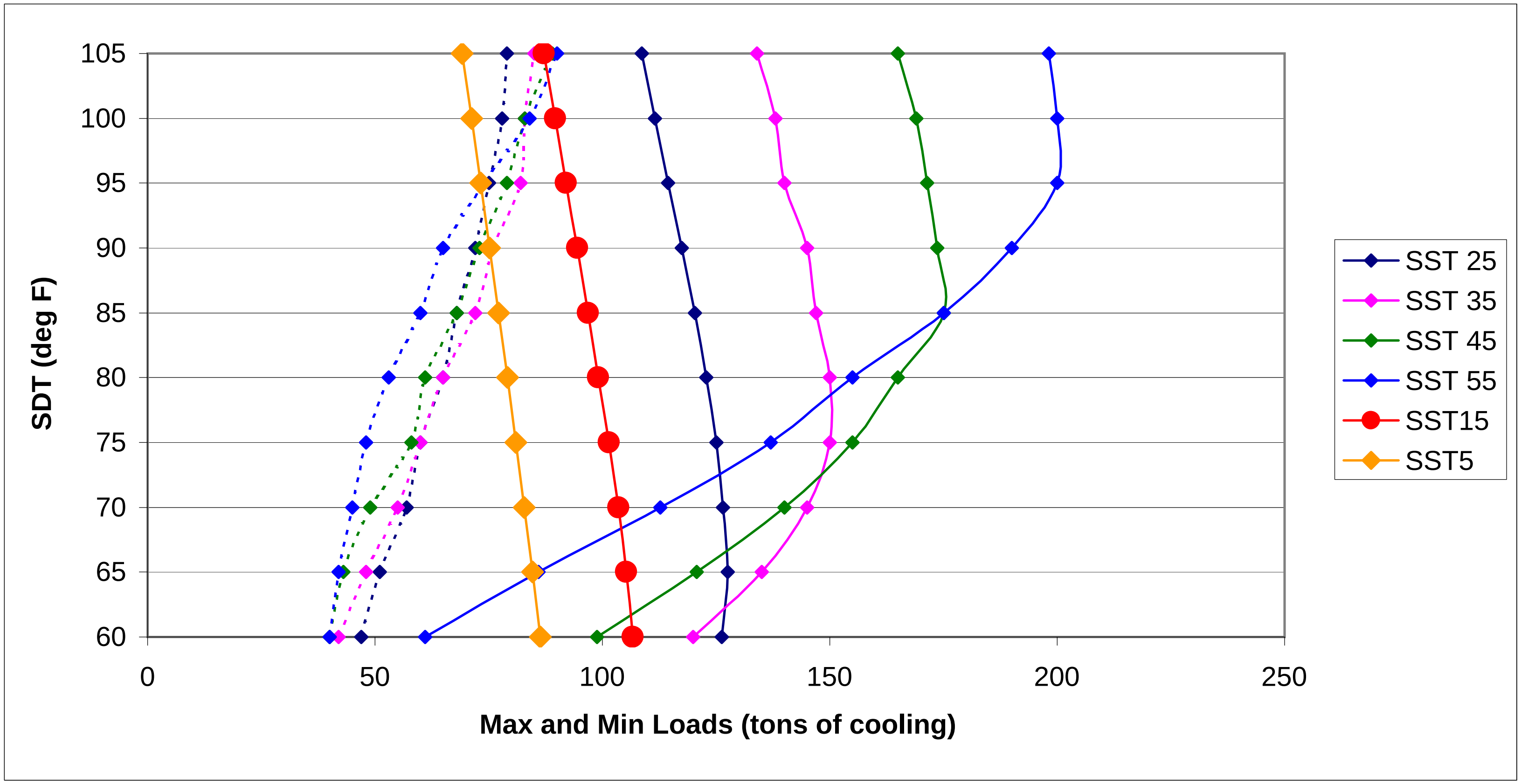
SST	SDT	Suctio Sucti on psig	Discharge Dischar ge psig	Comp Ratio	Max Load (tons)	Efficiency at Max (kW/ton)	COP at Max	Min Load	Efficiency at Min (kW/ton)	COP at Max
-5	25				73	0.41	9.58	25	0.35	11.048
-5	30			1.00	74	0.47	8.48	28	0.4	9.7922
-5	35	4.1	30.4	2.40	74	0.53	7.64	34	0.45	8.8153
-5	40	4.1	35	2.64	73.5	0.6	6.86	36	0.51	7.8959
-5	45	4.1	40.5	2.94	72.5	0.67	6.25	39	0.58	7.0636
-5	50	4.1	45.4	3.20	71.2	0.76	5.63	42	0.65	6.4106
-5	55	4.1	51.2	3.51	69.9	0.86	5.09	44	0.72	5.8846
-5	60	4.1	57.4	3.84	68.5	0.92	4.82	47	0.8	5.3961
-5	65	4.1	64.05	4.19	67	0.96	4.66	49	0.88	4.9965
-5	70	4.1	71.1	4.56	65.6	1	4.52	51	0.97	4.6257
5	40				89.7	0.47	8.48	35	0.39	10.018
5	45	9.05	40.5	2.32	89.8	0.53	7.64	41	0.44	8.9929
5	50	9.05	45.4	2.53	89	0.59	6.96	44	0.5	8.0338
5	55	9.05	51.2	2.77	87.8	0.66	6.33	46	0.56	7.2802
5	60	9.05	57.4	3.04	86.3	0.74	5.75	49	0.63	6.58
5	65	9.05	64.05	3.32	84.6	0.84	5.19	54	0.7	6.02
5	70	9.05	71.1	3.61	82.8	0.93	4.78	56	0.78	5.51
5	75	9.05	78.65	3.93	81	0.98	4.59	58	0.86	5.09
5	80	9.05	86.7	4.27	79.1	1.03	4.41	59	0.95	4.70
5	85	9.05	95.25	4.63	77.2	1.07	4.29	61	1.04	4.38
5	90	9.05	104.3	5.01	75.2	1.21	3.91	62	1.15	4.06
5	95	9.05	113.95	5.42	73.2	1.17	4.01	64	1.29	3.73
~~5~~	~~100~~	~~9.05~~	~~124.2~~	~~5.85~~	~~71.2~~	~~1.22~~	~~3.88~~	~~66~~	~~1.37~~	~~3.57~~
~~5~~	~~105~~	~~9.05~~	~~134.95~~	~~6.30~~	~~69.1~~	~~1.27~~	~~3.77~~	~~67~~	~~1.37~~	~~3.57~~
15	55				107.5	0.53	~~7.64~~	46	0.43	~~9.18~~

Not sure if these numbers are real because they start to go off the cart and the COP is really low

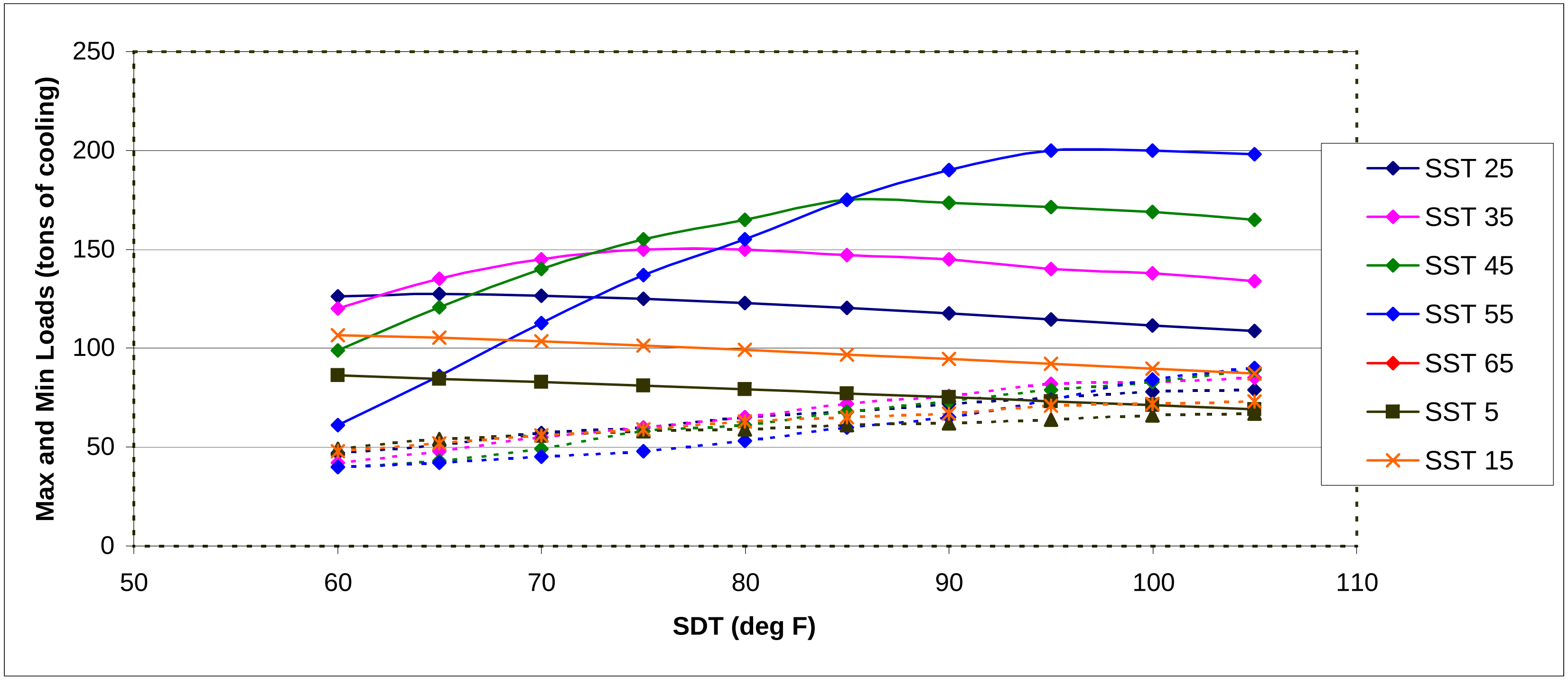
15	**60**	15.1	57.4	2.42	**106.7**	**0.59**	6.96	48	0.5	8.03
15	65	15.1	64.05	2.65	105.3	0.65	6.41	52	0.56	7.28
15	70	15.1	71.1	2.88	103.5	0.73	5.82	56	0.62	6.67
15	75	15.1	78.65	3.14	101.4	0.82	5.29	59	0.69	6.10
15	80	15.1	86.7	3.41	99.1	0.92	4.82	63	0.77	5.57
15	85	15.1	95.25	3.70	96.8	1	4.52	65	0.85	5.14
15	90	15.1	104.3	4.00	94.5	1.05	4.35	67	0.93	4.78
15	95	15.1	113.95	4.32	92	1.1	4.20	71	1.01	4.48
15	100	15.1	124.2	4.67	89.6	1.15	4.06	72	1.11	4.17
15	105	15.1	134.95	5.03	87.1	1.2	3.93	73	1.23	3.86
25	60	22	57.4	1.96	126.3	0.46	8.65	47	0.38	10.25
25	**65**	22	64.05	2.15	**127.5**	**0.52**	7.76	51	0.43	9.18
25	70	22	71.1	2.34	126.5	0.58	7.06	57	0.49	8.18
25	75	22	78.65	2.54	125	0.65	6.41	60	0.55	7.39
25	80	22	86.7	2.76	122.8	0.72	5.88	65	0.61	6.77
25	85	22	95.25	3.00	120.3	0.8	5.40	68	0.68	6.17
25	90	22	104.3	3.24	117.5	0.9	4.91	72	0.76	5.63
25	95	22	113.95	3.51	114.5	1.01	4.48	75	0.84	5.19
25	100	22	124.2	3.78	111.6	1.07	4.29	78	0.92	4.82
25	105	22	134.95	4.08	108.6	1.12	4.14	79	1.01	4.48
30	60	26.1	57.4	1.77	126.6	0.38	10.25			
30	65	26.1	64.05	1.93	136.5	0.46	8.65			
30	**70**	26.1	71.1	2.10	**138.3**	0.52	7.76			
30	**75**	26.1	78.65	2.29	137.5	0.58	7.06			
30	80	26.1	86.7	2.49	135.7	0.65	6.41			
30	85	26.1	95.25	2.69	133.3	0.72	5.88			
30	90	26.1	104.3	2.92	130.5	0.8	5.40			
30	95	26.1	113.95	3.15	127.5	0.89	4.95			
30	100	26.1	124.2	3.40	124.1	1	4.52			
30	105	26.1	134.95	3.67	120.8	1.08	4.26			
35	60	30.4	57.4	1.60	120	0.31	12.34	42	0.27	14.03
35	65	30.4	64.05	1.75	135	0.38	10.25	48	0.32	11.99
35	70	30.4	71.1	1.90	145	0.45	8.82	55	0.37	10.51

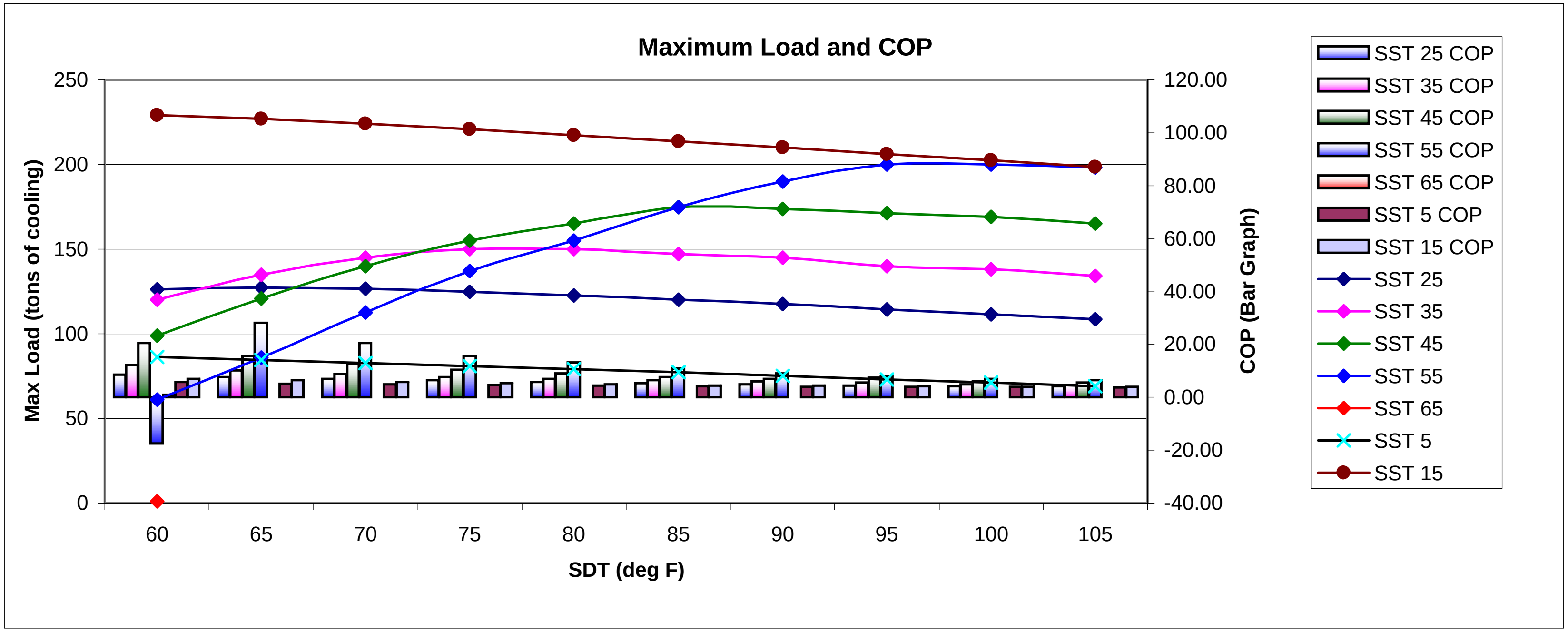
35	75	30.4	78.65	2.07	150	0.52	7.76	60	0.42	9.37	
35	**80**	30.4	86.7	2.25	**150**	**0.58**	7.06	65	0.48	8.33	
35	85	30.4	95.25	2.44	147	0.64	6.50	72	0.54	7.51	
35	90	30.4	104.3	2.64	145	0.71	5.95	76	0.6	6.86	
35	95	30.4	113.95	2.85	140	0.77	5.57	82	0.67	6.25	
35	100	30.4	124.2	3.08	138	0.88	5.00	83	0.74	5.75	
35	105	30.4	134.95	3.32	134	0.98	4.59	85	0.82	5.29	
40	60	35	57.4	1.45	112.9	0.24	15.65				
40	65	35	64.05	1.58	130.8	0.31	12.34				
40	70	35	71.1	1.73	145.5	0.38	10.25				
40	75	35	78.65	1.88	157.1	0.45	8.82				
40	**80**	35	86.7	2.04	**161.6**	0.52	7.76				
40	85	35	95.25	2.21	161	0.58	7.06				
40	90	35	104.3	2.39	158.9	0.64	6.50				
40	95	35	113.95	2.59	156.1	0.71	5.95				
40	100	35	124.2	2.79	152.7	0.79	5.45				
40	105	35	134.95	3.01	148.9	0.87	5.04				
45	60	40.1	57.4	**1.32**	98.8	0.18	20.54	40	0.15	24.45	
45	65	40.1	64.05	**1.44**	120.7	0.24	15.65	43	0.2	18.58	
45	70	40.1	71.1	1.57	140	0.3	12.72	49	0.26	14.53	
45	75	40.1	78.65	1.71	155	0.37	10.51	58	0.31	12.34	
45	80	40.1	86.7	1.85	165	0.44	8.99	61	0.37	10.51	
45	**85**	40.1	95.25	2.01	**175**	**0.52**	7.76	68	0.42	9.37	
45	90	40.1	104.3	2.17	173.6	0.58	7.06	73	0.48	8.33	
45	95	40.1	113.95	2.35	171.4	0.54	7.51	79	0.53	7.64	
45	100	40.1	124.2	2.54	169	0.71	5.95	83	0.6	6.86	
45	105	40.1	134.95	2.73	165	0.78	5.51	89	0.66	6.33	
55	60	51.2	57.4	**1.09**	61	-0.19	-17.51	40	0.09	40.0766	<-- Comp Ratio must be
55	65	51.2	64.05	**1.19**	86	0.13	28.05	42	0.18	20.54	G.T. 1.5
55	70	51.2	71.1	**1.30**	112.7	0.18	20.54	45	0.15	24.45	
55	75	51.2	78.65	**1.42**	137	0.24	15.65	48	0.2	18.58	
55	80	51.2	86.7	1.54	155	0.29	13.13	53	0.26	14.53	
55	85	51.2	95.25	1.67	175	0.36	10.77	60	0.32	11.99	

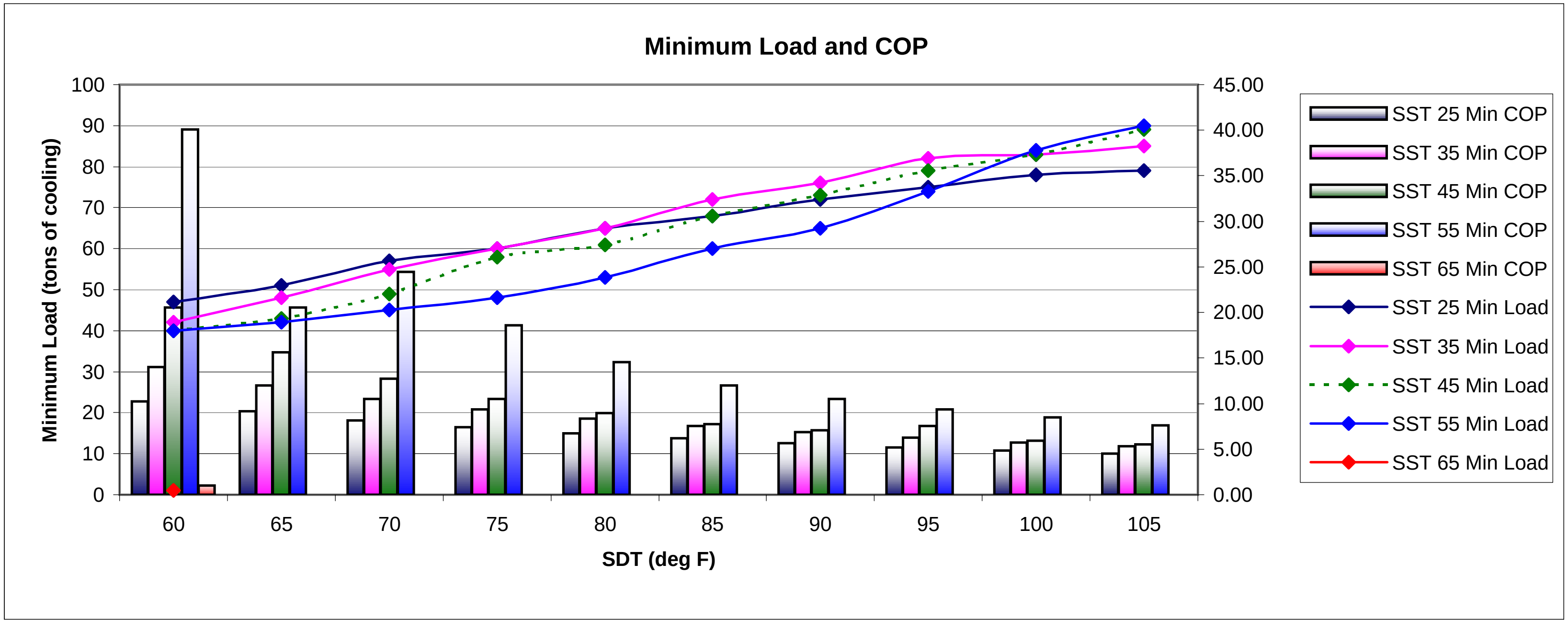
55	90	51.2	104.3	1.81	190	0.44	8.99	65	0.37	10.51
55	95	51.2	113.95	1.95	200	0.51	7.90	74	0.42	9.37
55	100	51.2	124.2	2.11	200	0.58	7.06	84	0.47	8.48
55	**105**	51.2	134.95	2.27	**198.2**	**0.64**	6.50	90	0.53	7.64
55	110									



Note: The direction of the axis was changed from last year because Loads are the dependent variable






Minimum Load and COP
Minimum Load (tons of cooling)
100
90
80
70
60
50
40
30
20
10
0
45.00
40.00
35.00
30.00
25.00
20.00
15.00
10.00
5.00
0.00
60
65
70
75
80
85
90
95
100
105
SDT (deg F)
SST 25 Min COP
SST 35 Min COP
SST 45 Min COP
SST 55 Min COP
SST 65 Min COP
SST 25 Min Load
SST 35 Min Load
SST 45 Min Load
SST 55 Min Load
SST 65 Min Load

Note: To Enable Circular References: Click 'Tools' --> Options --> Calculation --> Check Iteration

Rule of thumb approximatic

	Total Makeup Air Heating Requirements						Operating Temperatures		Maximum Compressor Load		Energy Available to Evaporator		Building Preheat Requirement		
T (F) of ODA	hrs per Year	q (BTU/hr)	q (Tons of Cooling)	q (BTU/yr)	Gal/Yr with no HP	Annual Cost with no HP	SST (F)	SDT (F)	Max Compressor Load at Evaporator (Tons)	Efficiency kW/ton at Max	(BTU/hr)	(Tons)	Building Heat Requirement (Tons)	Building Heat minus Electrical Heat	
-15 °F	1	4,541,184	378	4.541E+06	37.84	$68.12	30 °F	70 °F	138.3	0.52	1759708.8	146.6	378.4	360.6	
-10 °F	5	4,257,360	355	2.129E+07	177.39	$319.30	30 °F	70 °F	138.3	0.52	1759708.8	146.6	354.8	337.0	
-5 °F	15	3,973,536	331	5.960E+07	496.69	$894.05	30 °F	70 °F	138.3	0.52	1759708.8	146.6	331.1	313.3	
0 °F	29	3,689,712	307	1.070E+08	891.68	$1,605.02	30 °F	70 °F	138.3	0.52	1759708.8	146.6	307.5	289.7	
5 °F	60	3,405,888	284	2.044E+08	1702.94	$3,065.30	30 °F	70 °F	138.3	0.52	1759708.8	146.6	283.8	266.0	
10 °F	109	3,122,064	260	3.403E+08	2835.87	$5,104.57	30 °F	70 °F	138.3	0.52	1759708.8	146.6	260.2	242.4	
15 °F	190	2,838,240	237	5.393E+08	4493.88	$8,088.98	30 °F	70 °F	138.3	0.52	1759708.8	146.6	236.5	218.7	
20 °F	293	2,554,416	213	7.484E+08	6237.03	$11,226.66	30 °F	70 °F	138.3	0.52	1759708.8	146.6	212.9	195.1	
25 °F	408	2,270,592	189	9.264E+08	7720.01	$13,896.02	30 °F	70 °F	138.3	0.52	1759708.8	146.6	189.2	171.4	
30 °F	599	1,986,768	166	1.190E+09	9917.28	$17,851.11	30 °F	70 °F	138.3	0.52	1759708.8	146.6	165.6	147.7	
35 °F	820	1,702,944	142	1.396E+09	11636.78	$20,946.21	30 °F	70 °F	138.3	0.52	1759708.8	146.6	141.9	126.9	
40 °F	839	1,419,120	118	1.191E+09	9922.01	$17,859.63	35 °F	75 °F	150	0.52	1475884.8	123.0	118.3	106.9	
45 °F	722	1,135,296	95	8.197E+08	6830.70	$12,295.26	40 °F	75 °F	157.1	0.45	1192060.8	99.3	94.6	86.8	
50 °F	748	851,472	71	6.369E+08	5307.51	$9,553.52	45 °F	75 °F	155	0.37	908236.8	75.7	71.0	65.8	
Total				**8.185E+09**	**68207.64**	**$122,773.75**									

Annual Return
$84,061.57

8,184,916,512.00

$$\dot{q} = \dot{m} \cdot c_p \cdot (T_{\text{Supply}} - T_{\text{Outside Air}})$$

$$Q = \dot{q} \cdot \frac{hr}{yr}$$

$$\frac{Gal_{\text{fuel}}}{year} = Q \cdot \left(\frac{BTU}{Gal_{\text{fuel}}}\right) \cdot \eta_{\text{boiler}}$$

$$\frac{Cost_{\text{fuel}}}{year} = \left(\frac{Gal}{year}\right) \cdot \left(\frac{Cost}{Gallon}\right)$$

At High Temps, Lower the SST to provide more heat.

SST Below 30F will cause freezing.

Calculated from Turbocor Software

$$\dot{m} \cdot c_p \cdot (71°F - (SST + 10°F))$$

on: SST=Toa+10degF

Operating Conditions					THR				Heat Pump Energy Use		
Evaporator Load at Oper. Cond (Tons)	**Efficiency kW/ton at Oper. Cond.**	**COP at Operating Conditions**	**Temperature of Air Leaving Evaporator**	**Tons of Electrical Heat at Op Cond**	**THR to Condensor (Tons)**	***THR to Condensor (BTU/hr)***	***Total Heat Output to Condensor (BTU/yr)***	***Temperature of Air Leaving Condensor***	**Energy Use (kW)**	**kWh/yr for HP**	**$/yr of Electricity for HP**
138	0.52	7.8	41.8 °F	17.8	156.11	1,873,376	1,873,376	18.0 °F	71.92	72	$7.19
138	0.52	7.8	41.8 °F	17.8	156.11	1,873,376	9,366,881	23.0 °F	71.92	360	$35.96
138	0.52	7.8	41.8 °F	17.8	156.11	1,873,376	28,100,642	28.0 °F	71.92	1079	$107.87
138	0.52	7.8	41.8 °F	17.8	156.11	1,873,376	54,327,908	33.0 °F	71.92	2086	$208.56
138	0.52	7.8	41.8 °F	17.8	156.11	1,873,376	112,402,569	38.0 °F	71.92	4315	$431.50
138	0.52	7.8	41.8 °F	17.8	156.11	1,873,376	204,197,999	43.0 °F	71.92	7839	$783.88
138	0.52	7.8	41.8 °F	17.8	156.11	1,873,376	355,941,467	48.0 °F	71.92	13664	$1,366.40
138	0.52	7.8	41.8 °F	17.8	156.11	1,873,376	548,899,210	53.0 °F	71.92	21071	$2,107.14
138	0.52	7.8	41.8 °F	17.8	156.11	1,873,376	764,337,466	58.0 °F	71.92	29342	$2,934.17
138	*0.52*	7.8	41.8 °F	17.8	156.11	1,873,376	1,122,152,309	63.0 °F	71.92	43078	$4,307.77
127	*0.47*	8.5	44.2 °F	15.0	141.91	1,702,944	1,396,414,080	65.0 °F	59.66	48925	$4,892.53
107	*0.42*	9.4	48.4 °F	11.4	118.26	1,419,120	1,190,641,680	65.0 °F	44.88	37655	$3,765.53
87	*0.35*	11.0	52.7 °F	7.9	94.61	1,135,296	819,683,712	65.0 °F	30.36	21923	$2,192.31
66	*0.3*	12.7	57.1 °F	5.2	70.96	851,472	636,901,056	65.0 °F	19.74	14762	$1,476.22
											$24,617

$$= \text{Minimum}\left[\dot{Q}evap_{max},\ \dot{Q}comp_{max},\ \dot{Q}heat_{req} - \dot{Q}electric\right]$$

$$T_e = T_i + \frac{\Delta h}{c_p \cdot \dot{m}}$$

$$\frac{Q_{HP}}{yr} = \frac{Q_{HP}}{hr} \cdot \frac{hrs}{yr}$$

$$\boxed{\dot{W}_{comp} = Q_{evap} \cdot COP_{comp}}$$

Total Costs with Heat pump

q Steam with HP (BTU/yr)	Gal/Yr with HP	Fuel Cost ($/year)	Total Heating Cost with HP ($/year)
2,667,808	22.23173215	$40.02	$47.21
11,919,919	99.33266075	$178.80	$214.76
31,502,398	262.5199822	$472.54	$580.41
52,673,740	438.9478323	$790.11	$998.66
91,950,711	766.255929	$1,379.26	$1,810.76
136,106,977	1134.224804	$2,041.60	$2,825.49
183,324,133	1527.701108	$2,749.86	$4,116.27
199,544,678	1662.87232	$2,993.17	$5,100.31
162,064,070	1350.533917	$2,430.96	$5,365.13
67,921,723	566.0143576	$1,018.83	$5,326.59
0	0	$0.00	$4,892.53
0	0	$0.00	$3,765.53
0	0	$0.00	$2,192.31
0	0	$0.00	$1,476.22
		$14,095.14	**$38,712.18**

Total Savings: $84,061.57

	Annual Savings	Pounds of Carbon Dioxide Emissions			Pounds of Sulfur Dioxide Emissions		
		from fuel without HP	from HP	from fuel with HP	from fuel without HP	from HP	from fuel with HP
	$20.91	987	103	580	11.9	0.2	7.0
	$104.55	4,627	514	2,591	55.7	1.0	31.2
	$313.64	12,957	1,543	6,848	156.0	3.1	82.4
	$606.36	23,260	2,982	11,450	280.0	6.0	137.8
	$1,254.54	44,423	6,170	19,988	534.7	12.5	240.6
	$2,279.09	73,976	11,210	29,587	890.5	22.7	356.1
	$3,972.72	117,227	19,540	39,851	1411.1	39.5	479.7
	$6,126.35	162,699	30,132	43,378	1958.4	60.9	522.1
	$8,530.89	201,383	41,959	35,230	2424.1	84.8	424.1
	$12,524.52	258,701	61,601	14,765	3114.0	124.5	177.7
	$16,053.68	303,556	69,963	0	3654.0	141.4	0.0
	$14,094.10	258,825	53,847	0	3115.5	108.8	0.0
	$10,102.94	178,185	31,350	0	2144.8	63.4	0.0
	$8,077.29	138,451	21,110	0	1666.6	42.7	0.0
Total :		1,779,258	352,024	204,269	21,417	711	2,459
Total in Tons:		***890***	***176.01***	***102.13***	***10.67***	***0.36***	***1.23***
			Total with HP:	**278.15**		**Total with HP:**	**1.59**

Savings: **611.5** tons of CO2 per heating season — **9.09** tons of SO2 per heating season

Percent Reduction **68.73%** CO2 per heating season
85.15% SO2 per Heating Season

Return on Investment Results

Capital Cost:	$195,000		
Annual Heat Savings	$84,062		
Annual Cooling Savings	$20,000		
Maintenance	($5,000)	terest Rate: 4.5%	Inflation: 2%
Annual Return	$99,062	Lifespan: 15 years	
		IRR 78% / **NPV** $1,030,447	**Payback Period** **1.97** years

Andrew Bennett

From:	seaver.woolfolk@superradiatorcoils.com
Sent:	Monday, April 21, 2008 9:41 AM
To:	Andrew Bennett
Cc:	Jeff Spaeth
Subject:	Super Radiator Quote
Attachments:	Attach0.html; Spec Sheets.pdf; Q-06560.pdf

Here is your quote for the two coils. I have provided the specifications for the each of coils and you have two options for the intake air coil. Thanks again.

Seaver Woolfolk
Sales Account Specialist

Super Radiator Coils
(804)378-1367 Direct
(804)379-2118 Fax

Super Radiator Coils
Wet Evaporator Coil Design

Program Version 3.1.4.3, (c) 2007 Super Radiator Coils

Customer:	Super Radiator Coils	**Date:**	4/17/2008
Job Ref.:		**By:**	Jay Darji
Job Item:	License # SRC	**Record:**	JD04178AR0

Wet Evaporator Coil **Units (Eng.)**

AirSide Requirement:		
Air Flow	**SCFM**	13,500
Required Capacity	**BTUH**	414,000
Entering DB/WB Temp.	**°F**	70.0/55.0
Required DB/WB Temp.	**°F**	??/42.3
Airside Pressure	**PSIA**	14.696
Coil Hand		LH
Tubeside Requirement:		
Refrigerant Type		R-134a
Refrigerant Temp.	**°F**	30.0
Liquid Temp.	**°F**	100.0
Super Heat Temp.	**°F**	10.0

Exhaust Air Coil

Coil Selection:		
Model Number		40.5x47-8R-58/108
Part Number		01C04020470-1B08EW16001-C04A04G01IL
Die Surface		5/8 - 1-1/2 x 1.299 Stag.(Corr-#1)
Die Number & Location		1 (All Plants)
Face Area	**Ft^2**	13.22
Face Velocity	**Ft/Min.**	1021.3
Number of Circuits		27
Circuit Loading	**BTUH**	15,782
Configuration		Crossflow
Tube Material		CU
Tube Wall Thickness	**In.**	0.020
Fin Material		AL
Fin Thickness	**In.**	0.0075
Header Diam.	**In.**	3.125
Coil Wght. (Gal.Casing)	**Lb.**	328
Capacity:		
Total Capacity	**BTUH**	426,104
Sensible Capacity	**BTUH**	401,019
Leaving DB/WB Temp.	**°F**	42.8/41.9
Air Friction	**In. H2O**	2.70
Ref. Pres. Drop @ Rating	**Psi**	2.19
Ref. Pres. Drop @ Req'mt	**Psi**	2.08

SUPER RADIATOR COILS

Condenser Coil Calculations Summary

Program v3.1.4.3, (c) 2007 Super Radiator Coils

Design Parameters:

AirSide Requirements:	*Units (Eng.)*	*Input Data*
Air Flow	SCFM	27,000
Required Capacity	BTUH	936,000
Entering Air Temp	°F	30 0
Required Air Temp	°F	61.5
Airside Pressure	PSIA	14.696
Coil Hand		LH

Tubeside Requirements:		
Refrigerant Type		R-134a
Refrigerant Temp	°F	70 0
HG Temp	°F	150.0
Deg. Subcooling	°F	5.0

Intake Air Coil (Option 1)

Optimized Design Options:

Coil Option		#1	#2	#3	#4
Coils / Unit		1	1	1	1
Model - Face/Row		70x118-4R	70x118-4R	70x118-4R	70x118-6R
Model - Tube/Fin		38/108	38/132	38/144	38/108
Face Area Per Coil	Ft^2	57.36	57.36	57.36	57.36
Face Velocity	Ft/Min.	470.7	470.7	470.7	470 7
Die Surface		11X	11S	11C	11F
Circuits / Type		FU / CF	FU / CF	FU / CF	FU / CF
Circuit Loading	Btu/Hr.	14,035	13,429	13,662	13,499
Total Coil Capacity	Btu/Hr.	982,436	940,003	956,354	944,957
Leaving Air Temp.	°F	63.7	62.2	62.8	62.4
Air Friction	In.H2O/Coil	0.31	0 28	0.24	0 19
Ref. Pres. Drop @ Rating	Psi/Coil	6.81	6.34	6.52	9.59
Ref. Mass Flow	Lb/Hr./Coil	10191	9751	9921	9802
Header Diam. (HG/Liq.)	In.	3.625 / 1.375	3.625 / 1.375	3.625 / 1.375	3.625 / 1 375
Tube Material		Copper	Copper	Copper	Copper
* Tube Wall Thickness	In.	0.015 (Rifled)	0.015 (Rifled)	0.015 (Rifled)	0.015 (Rifled)
Fin Material		Aluminum	Aluminum	Aluminum	Aluminum
Fin Thickness	In.	0.0055	0.0055	0.0055	0.0055
Casing Matl.		16G Gal.Stl.	16G Gal.Stl.	16G Gal.Stl	16G Gal.Stl
Approx. Per Coil Wght.	Lbs.	398	429	444	571

SUPER RADIATOR COILS™

Condenser Coil Calculations Summary

Program v3.1.4.3, (c) 2007 Super Radiator Coils

Design Parameters:

AirSide Requirements:	Units (Eng.)	Input Data
Air Flow	SCFM	27,000
Required Capacity	BTUH	936,000
Entering Air Temp.	°F	30.0
Required Air Temp.	°F	61.5
Airside Pressure	PSIA	14.696
Coil Hand		LH
Tubeside Requirements:		
Refrigerant Type		R-134a
Refrigerant Temp.	°F	70.0
HG Temp.	°F	150.0
Deg. Subcooling	°F	5.0

Intake Air Coil (Option 2)

Optimized Design Options:

Coil Option		#1	#2	#3	#4
Coils / Unit		1	1	1	1
Model - Face/Row		70x118-4R	70x118-6R	70x118-6R	70x118-6R
Model - Tube/Fin		38/132	38/96	38/96	38/132
Face Area Per Coil	Ft^2	57.36	57.36	57.36	57.36
Face Velocity	Ft/Min.	470.7	470.7	470.7	470.7
Die Surface		11X	11S	11C	11F
Circuits / Type		FU / CF	FU / CF	FU / CF	FU / CF
Circuit Loading	Btu/Hr	13,778	13,678	13,435	13,811
Total Coil Capacity	Btu/Hr.	964,474	957,442	940,427	966,750
Leaving Air Temp.	°F	63.0	62.8	62.2	63.1
Air Friction	In.H2O/Coil	0.36	0.32	0.26	0.23
Ref. Pres. Drop @ Rating	Psi/Coil	4.48	6.65	6.44	6.77
Ref. Mass Flow	Lb/Hr./Coil	10005	9932	9755	10029
Header Diam. (HG/Liq.)	In.	3.625 / 1.375	3.625 / 1.375	3.625 / 1.375	3.625 / 1.375
Tube Material		Copper	Copper	Copper	Copper
* Tube Wall Thickness	In.	0.016	0.016	0.016	0.016
Fin Material		Aluminum	Aluminum	Aluminum	Aluminum
Fin Thickness	In.	0.0055	0.0055	0.0055	0.0055
Casing Matl.		16G Gal.Stl.	16G Gal.Stl.	16G Gal.Stl.	16G Gal.Stl.
Approx. Per Coil Wght.	Lbs.	440	565	565	635

SUPER RADIATOR COILS™

Quote #: 06560

Program Version 1.3.1

Customer:	**University of Maine**	**From:**	Jeff Spaeth
To:	Andrew Bennett	**Title:**	Lean Coils Product Manager
Rep #:	000	**Date:**	04/17/2008
Phone:		**Project:**	Heat Recovery
Fax:	804-794-7437	**Phone:**	804-378-1309
Email:	andrew.bennett@umit.main.edu	**Email:**	jeff.spaeth@superradiatorcoils.com

Coil Data / Net Price Each / Materials & Options

Part: Intake Air Coil (Option 1)
Type: Condenser Coil
Size: 70h x 118w (4R) 38/108

Qty: 2 $5,589.00 ea.

Tubes: 3/8" o.d. x .015" Rifled wall copper
Fins: .0075" thick aluminum
Headers: Type L copper
Connections: Copper
Casing: 16 gauge

Coil Data / Net Price Each / Materials & Options

Part: Intake Air Coil (Option 2)
Type: Condenser Coil
Size: 70h x 118w (4R) 38/132

Qty: 2 $5,785.00 ea.

Tubes: 3/8" o.d. x .016" wall copper
Fins: .0075" thick aluminum
Headers: Type L copper
Connections: Copper
Casing: 16 gauge galvanized steel

Coil Data / Net Price Each / Materials & Options

Part: Exhaust Air Coil
Type: Evaporator Coil
Size: 40.5h x 47w (8R) 58/108

Qty: 4 $3,916.00 ea.

Tubes: 3/8" o.d. x .020" wall copper
Fins: .0075" thick aluminum
Headers: Type L copper
Connections: Copper
Casing: 16 gauge galvanized steel
Distributor: Brass

Notes:
Prices are based on material costs on the date of quotation.

Conditions:
Coils will be shipped in approximately 4 weeks after receipt of order.
Freight is F.O.B. Richmond, PPD and credit terms are to be determined.

Disclaimer:
Prices are valid for 2 weeks from date on quotation and are subject to credit/surcharge due to material fluctuations. All coils must ship within 6 weeks of receipt of purchase order. If not shipped in full by the end of 6 weeks from the receipt of the order, the remaining coils on the purchase order will be subject to re-pricing at current material costs. Due to rising material costs, any delay caused by the customer could result in a price adjustment. Specifications are required with receipt of a purchase order. Drawing approvals are required within 48 hours after submittal.

Andrew Bennett

From: Amanda D. Kimball [Amanda_D._Kimball@umit.maine.edu]
Sent: Tuesday, May 06, 2008 3:12 PM
To: Timothy Hardy; Andrew Bennett
Subject: Fwd: Re: UMaine Receiving Tank
Attachments: Attach0.html; Receiver Assembly Feb 25 2008.pdf; Compressor Base.pdf

----- Original Message -----

Amanda,
Sorry that this took so long
S09533 ----- $2850.00 plus freight
20HR44 20"od x 44" Horz Rec. Per sketch 300psi
5 weeks
Mike Mo
eCommerce Manager
877-313-9100 ph.
(215)673-8324 fax

"Amanda D. Kimball" <Amanda_D._Kimball@umit.maine.edu>

02/26/2008 12:07 PM

To "Mike Motyczka" <mmotyczka@uri.com>

cc "Timothy Hardy" <Timothy_Hardy@umit.maine.edu>, "Andrew Bennett" <Andrew_Bennett@umit.maine.edu>, "Brennan Johnson" <Brennan_Johnson@umit.maine.edu>

Subject Re: UMaine Receiving Tank

Mike,

The working pressure will be below 250 psig (we are using R-134a).

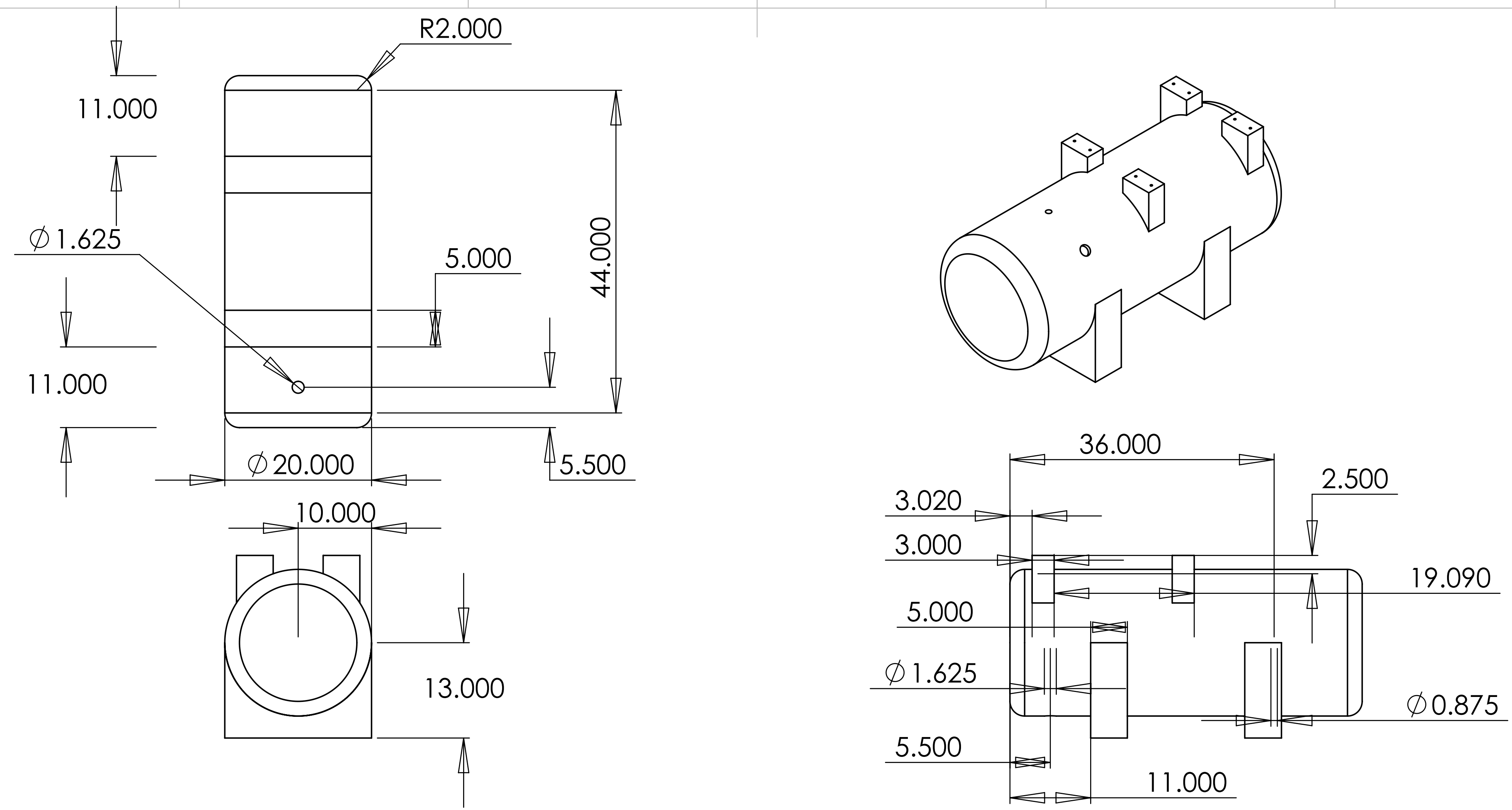
The dimensions can be the industry standards for 20 inches. We put a 2" dome in because we were unsure of the industrial standards and had to put a dimension on the drawing.

Attached are drawings of our newest receiver and compressor mounts. Let me know if you have further question.

Thanks,
Amanda

Mike Motyczka <mmotyczka@uri.com> on Monday, February 25, 2008 at
11:30 AM -0500 wrote:
>
>Amanda,
> What is the working pressure?
>What is the total overall lenth 20 inches? If so there can't be a
2" dome
>Mike Mo
>eCommerce Manager
>877-313-9100 ph.
>(215)673-8324 fax
>
>
>
>
>"Amanda D. Kimball" <Amanda_D._Kimball@umit.maine.edu>
>
>02/12/2008 03:27 PM
>
>
>To
>mmotyczka@uri.com
>
>cc
>"Timothy Hardy" <Timothy_Hardy@umit.maine.edu>, "Andrew Bennett"
<Andrew_Bennett@umit.maine.edu>, "Brennan Johnson"
<Brennan_Johnson@umit.maine.edu>
>
>Subject
>UMaine Receiving Tank
>
>
>
>
>
>
>
>
>
>We would like you to fabricate a receiving tank for refrigerant
134a.
>
>This is subject to change, but we would like a cost estimate and a
time estimate.
>

>Let me know if you need any more information.
>
>Thanks,
>Amanda
>
>



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING
			REVISION

	NAME	SIGNATURE	DATE
DRAWN	Timothy Hardy		
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:

WEIGHT:

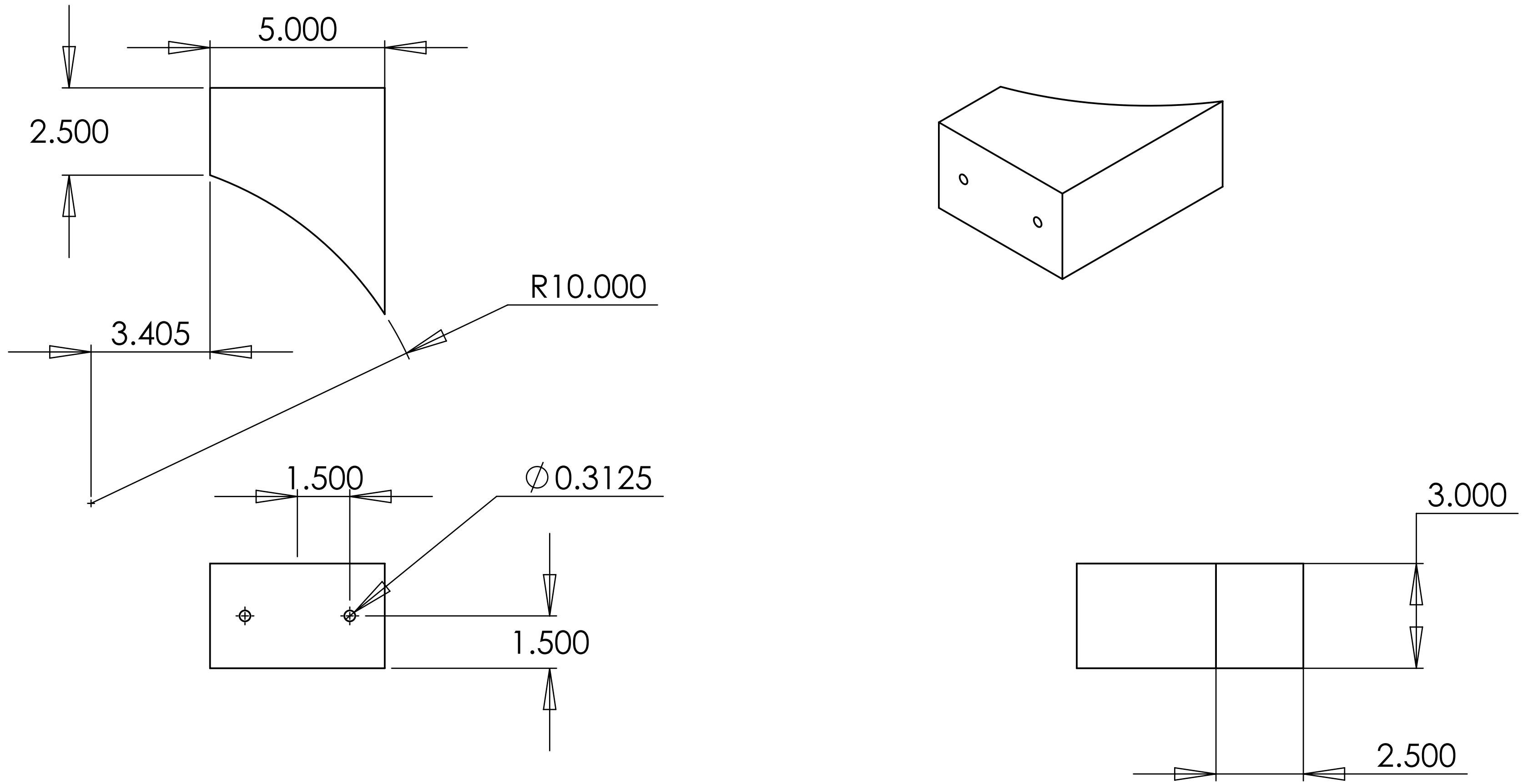
TITLE:

Receiver Assembly

DWG NO. A4

SCALE:1:20

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	Timothy Hardy		
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:

WEIGHT:

TITLE:

Compressor Base

DWG NO.

A4

SCALE:1:5

SHEET 1 OF 1