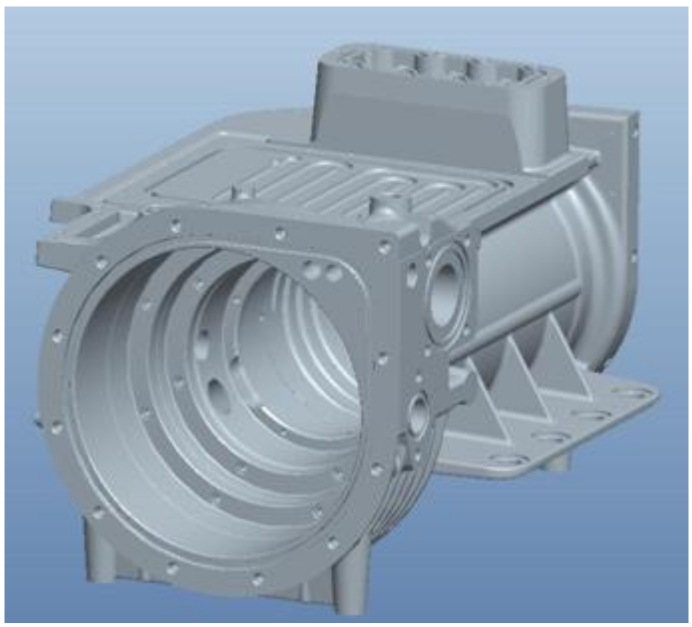
Final Design Report
Stator Insertion Machine Redesign



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Sponsored by:
Danfoss Turbocor

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Executive summary

Stator insertion process redesign is sponsored by Danfoss Turbocor. The main goal of this project is to come up with a way to heat up an aluminum compressor housing quickly, efficiently and to the desired specification. The desired application for this design is to be directly implemented on the production floor.

Our sponsor representative Robert Parsons from Danfoss Turbocor had specific needs for the project. Reduce the overall size of the design, lower the final temperature at which desired expansion is achieved, keep the compressor housing clean and dry throughout the process, and achieve needed expansion in less than 8 minutes.

To reach the goals specified a number of tasks were performed. To get the project started first an idea was postulated, that only the desired diameter was of the concern. Using linear expansion as the fundamental law Experiment 1 was performed and verified the predicted expansion. Next Matlab was used to simulate the requirements for the heat source under varying conditions, including different ambient temperatures, various convection coefficients, and numerous losses associated with the process. At the end using the data a conclusion was achieved describing that a 5600 Watt electric heater could achieve the goal of heating up the compressor housing using forced convection to the desired temperature of 85 degrees Celsius in just under 8 minutes. There are still many steps required to complete the project, but there is enough data to support the fabrication and testing of a proof of concept and then a prototype to be used on the production floor.

Introduction

One of the most valuable commodities in production is time, and the less time you spend doing any one task, the more efficient your overall process becomes

Danfoss Turbocor is a company that designs and manufactures large industrial refrigerant compressors. Their compressor is the first to utilize magnetic bearings, a clever way to improve efficiency and decrease noise. They have determined that one of the first stages in their production line is too slow relative to the others. If this production point can improve its turn over time to approximately eight minutes it would be able to keep up with the rest of the facility and create a smoother, more efficient flow in their manufacturing line.

So the project Danfoss Turbocor brought to us is to improve this slow stage of their production. This stage is responsible for heating the compressor housing and then inserting a stator into it. The housing needs to be heated to allow for the thermal expansion of the metal to enlarge the housing and hole that the stator fits into, this allows for easy installation of the stator. The assembly then begins cooling which shrinks the housing around the stator to lock it in place.

The housing is cast from A356 T6 aluminum. Currently a large convection oven is used to heat four housings at a time. The heating time is roughly 45 minutes at a temperature of 300 °F and before the housing can continue through production it must cool for approximately one hour.

Problem Definition

The fundamental task of this project is to redesign the current method for heating the compressor housings. The primary goal of this project is to design a quicker method of heating the housing while scaling down the overall size of the unit. Ideally the system used will be able to heat the housing in less than eight minutes and keep the overall part temperature down which reduces cooling time and allows it to be handled easily by the operator.

Concept Generation and Selection

For each of the following concepts the same six criteria are chosen to evaluate the designs based on their strong and weak characteristics. Performance includes efficiency, heating method, time required, and optimal operational temperature. Complexity deals with how simple the design would be to assemble and how easy it would be to replace expendable parts. Size is very important to the customer, that's why we included it as one of the main criteria. Durability considers what the expected life of the design is and how often the replaceable parts would need to be swapped out. Cost criteria was an obvious choice because like any business, saving money is always a plus, and finally user friendliness was chosen, since this is supposed to be installed on the production floor, so it should be easily operated by the necessary technicians.

Concept A – Convective Oven

Concept A uses forced convection to heat up the compressor housing. The insulated cabined would contain a powerful enough heater that would provide the necessary amount of heat input. Additionally a fan would re-circulate the air throughout the system. The air ducts would be incorporated into the cabinet to allow the air to be returned to the bottom and go through the cycle again. The compressor housing would simply slide horizontally on top of the cabined and be positioned in the desired location. Afterwards the insulated hood would be lowered on top to provide additional insulation and to help direct the air to the ducts that would re-circulate the air. Temperature probes could be inserted throughout the system to monitor the temperatures of the compressor housing.

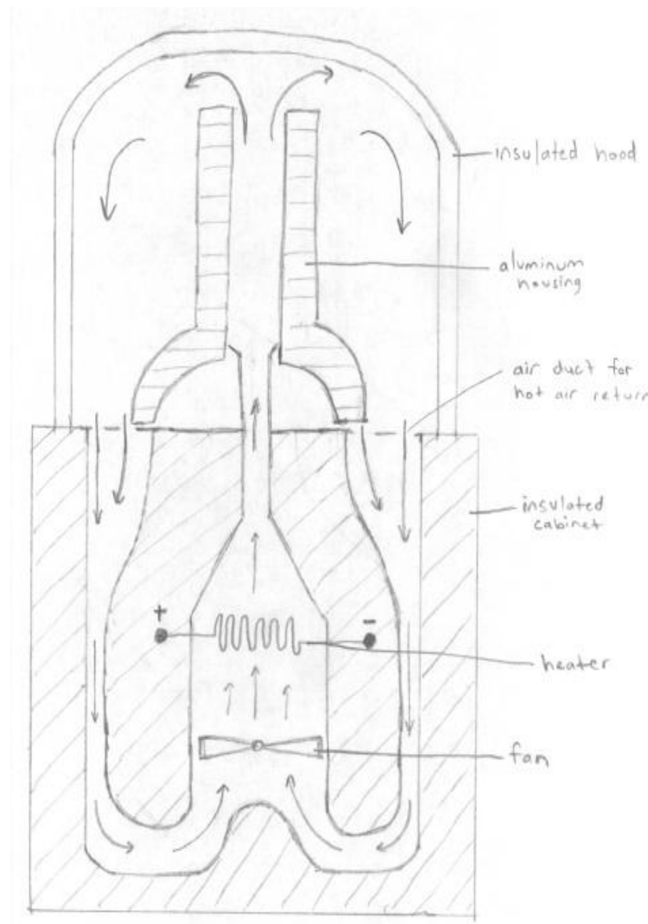


Figure 1: Concept

Analysis:

The positive characteristics of this design are high performance and the easy implementation into the production cycle. The heater cores could easily be found and substituted for the desired amount of heat input. The fan speed could also be regulated to control the forced convection. The fan would be appropriately chosen for the desired application, because of the high operating temperatures. The idea behind this concept is very simple because there are only two components to it, the heater and the fan. The compressor housing itself is used to channel the air and then the hood to return the air. The overall size of the design is comparably small because all the heater components are contained below in the insulated cabinet and because the stator insertion is a next process not incorporated in this design. The design is of average durability because the heater coils are subject to high operating temperatures, and will be the first thing to fail. Also the fan should be appropriately chosen not only to be able to provide the necessary air flow but also to be able to operate in the high temperatures required. The cost of the design is a little higher than some of the other concepts because of the requirements of the specific heater, fan and the required insulation. This is a very operator friendly concept because it only requires the housing to be moved into an appropriate position one time. Once the housing is located to the stator insertion mechanism no further manual input would be required.

Concept B – Oil Bath

The idea behind this concept is to use hot fluid to heat up a part. Since water cannot be heated above 212 degrees Fahrenheit it is not a good option for use as the heating fluid. Mineral oil on the other hand can be heated to about 500 degrees Fahrenheit without boiling, making it a good choice as the heating fluid. Now in order to heat the oil, an inductively heated steel tank is used, the steel tank would then conduct the heat into the fluid. As shown in Figure B the steel tank is designed for this specific housing to fit in it and around it by having a cylindrical extension in the middle to help heat the oil. The oil bath must be brought up to temperature before the housing is inserted because the induction heater can damage the aluminum housing when in operation. So the oil tank is inductively heated to about 500 degrees Fahrenheit and then the housing is inserted into the hot fluid where conduction heat transfer takes place. This method allows for a uniform heating of the part which eliminates residual stresses in the material.

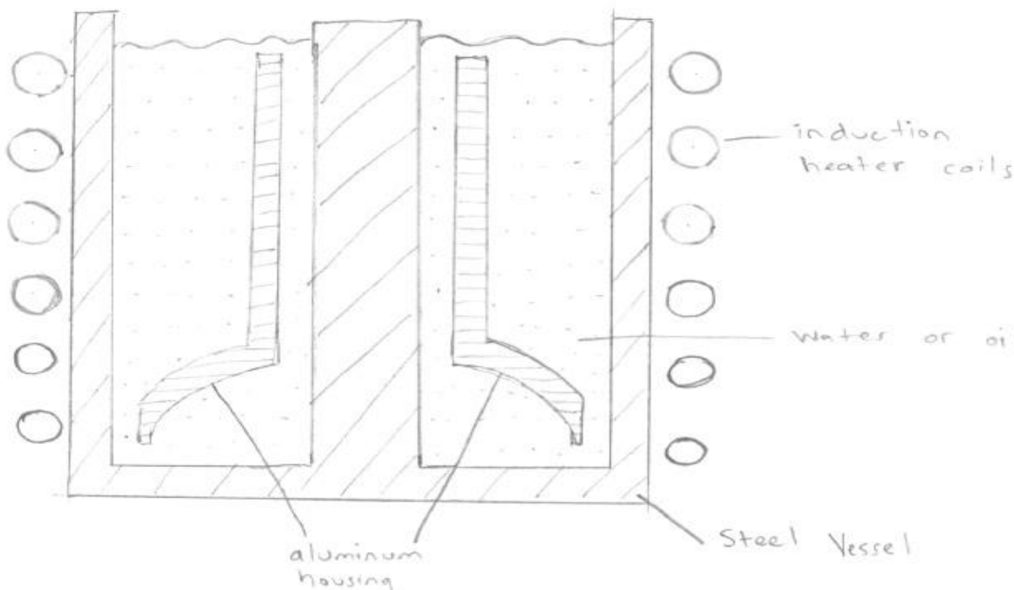


Figure 2: Concept B

Analysis:

This method of heating works well although the time needed to heat the part may become an issue. In this case time is an issue so the faster the part can be heated up the better. Adding a circulating pump to allow convection within the fluid would be an improvement to this design by increasing performance through time needed to heat up the part, but may slightly decrease durability while increasing complexity. This concept utilizes commonly available materials and equipment making it a rather simple design. The steel tank would be the most complex since it is specially designed for this part. This unit would be rather large since the part needs to fit inside it along with enough working fluid to effectively transfer enough heat. This is a problem for this facility due to limited floor space. With no moving parts except a possible circulating pump, this unit can basically last forever. If failure were to occur it would be within the induction heater itself. Another consideration is the replacement of oil as a small percentage of it will be lost throughout operation. Initial costs are relatively low since the parts and materials used are readily available and inexpensive. And maintenance costs will be low as well since the durability is so high but oil may need to be kept on hand for replacement and or topping off. This method is not very user friendly, it involves placing and removing hot parts from very hot oil, operator safety may be an issue.

Concept C – Internal Resistive Heating

This concept utilizes small size, relative low cost and ease of implementation into the existing production cycle. The concept consists of 2 pneumatic actuators. As the compressor housing comes into position on the production table, the lower actuator inserts and expands the heating element into the housing to provide conductive heating throughout the required area. After the required temperature has been reached, the lower actuator would lower and the same time the upper actuator would insert the preassembled stator into the housing. Afterwards the upper actuator would retract and grip another stator, while the assembled housing slides down the line for cooling and continuing assembling.

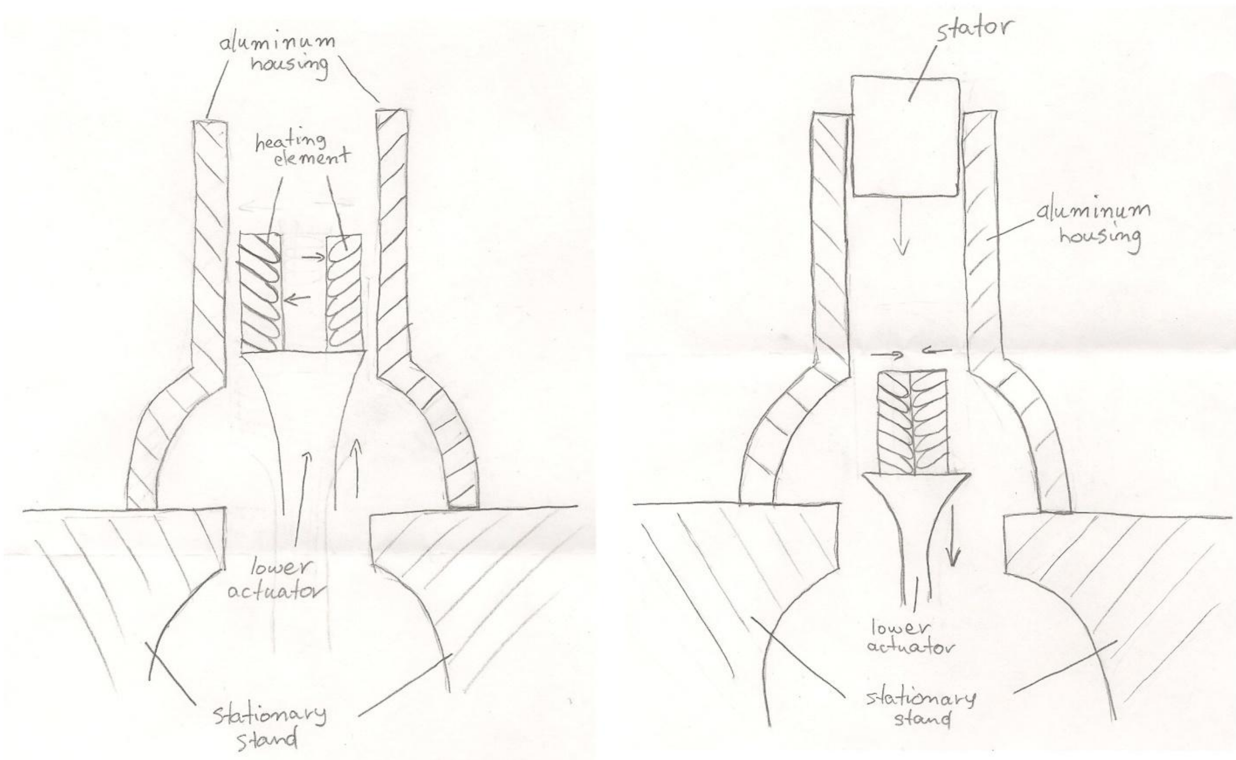


Figure 3: Concept C

Analysis:

Compared to the other concepts this design would have average performance, since the heating coils would heat up quickly but the transfer of this heat flux to the housing is compromised by the fact that the element does not touch the housing surface evenly throughout the required area. Also because only stator area of the housing is heated, there will stresses associated with uneven temperature distribution. This system would be relatively complex since there are a lot of moving parts, both of the actuators and the expansion of the heating element itself. Also because of the actuators precise positioning would be required. The overall size of the design would be a lot smaller than the other concepts because there is no need for any tanks or hoods and the heating coils are only as big as the compressor housing. Durability would suffer because again there are a lot of moving parts which will need to be properly maintained. Also there is hardly any way of predicting when the copper coils of the heating element will go bad, thus replacement ones will need to be on hand for substitution. The cost will go up because of low durability. Also constant maintenance will be required thus making it even more expensive to run. Replacement parts will need to be kept on hand which increases cost because even though the parts are paid for, they are not being used. After the initial installation setup, the operations would actually be very easy since the only actions required are the positioning of the housing, and initializing the start of the cycle.

Concept D – Internal Induction Heating

Another concept in heating the inside of the stator housing is to use internal inductive conduction. This just means that the heat will be dispersed by contact of a metal that will be heated up by an electric current that is caused to flow through the metal to the internal walls of the cylinder. The material of the housing, that is getting expanded, is aluminum, so it should take about 300 degrees F to change the dimensions of the housing. This copper will heat up by having current going up through it from a battery source. The process of heating the housing is, first to have the housing elevated over top of the copper cylinder. The copper cylinder will then be lifted up into the housing (lip side). Once fully in the housing, the copper cylinder will split open in four symmetrical quarter circles until each section is in contact with the inner wall of the housing. At that point, the current will go through the copper causing it to heat up to about 300 degrees F. This will cause the inner housing walls to expand to the desired dimension. Finally, the copper quarter circles will collapse back together and drop down out of the housing. Then the stator will be pressed into the heated and expanded housing and cooled to shrink fit stator in the housing.

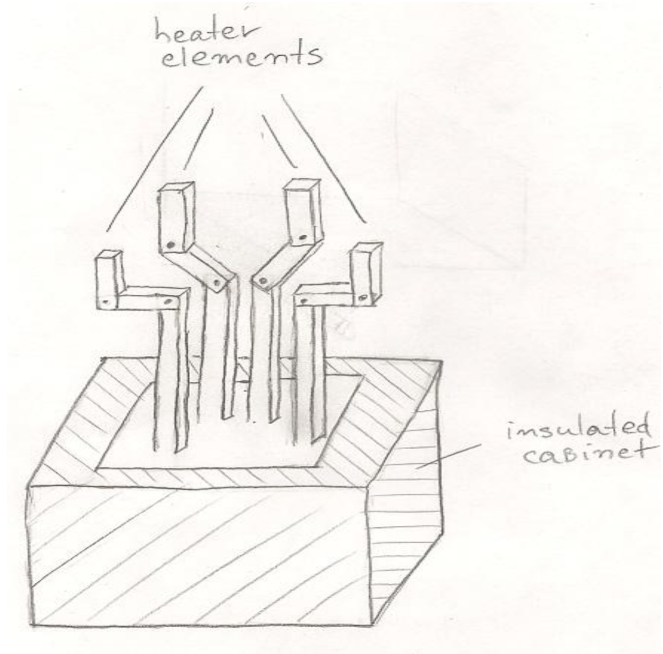


Figure 4: Concept D

Analysis:

The copper can reach a significantly high temperature with no issue. And will be able to disperse the heat throughout the inner wall of the housing evenly. This process should take no more than 15 minutes to heat and expand the inner walls, depending upon the amount of energy exuded. There are a lot of parts and mechanisms that are in this concept, causing it to be very complex. The most detailed area is the actual design of the shape of the soon-to-be heated copper, because the geometry of it has to be in a way that when it is expanded, it can be in contact with at least 90% of the inner wall of the housing. The linkages that are going to hold the copper and lift it will either be in copper or preferably steel so that the time for the copper to heat up will be shorter. The dimensions of the machine will not be big at all. It will be longer in height if anything because of the raises and lowering of the copper into the housing. It won't be as big as the current oven that is getting used to heat the housings now. So the customer's need of floor space can be acquired. With the mechanical arms being steel and the heat exchanger being copper, it makes the machine very durable. Both metals have high density, toughness, and fracture toughness. If any failure it will be in the joints of the mechanical arms. The economic value for this concept is not too expensive because steel, copper, joints, and a battery source are pretty easy to come by because they are heavily used in the industry. The most expensive part is the motor that will lift the mechanical arms and open them. The whole machine will be run by a controller that will direct it what to do; this makes it very safe and easy to use in any work place. The only part that may be done by human force is the removal of the completed shrink fitted stator in housing to the cooling area.

Concept Selection

The following decision matrix was constructed to aid in the making the decision. The weights are assigned to each criteria based on how important they are to the customer. Performance and size make up 50% of the overall design because those were the two most emphasized design characteristics. The rating is on a 1 to 5 scale, one being the lowest and five being the highest.

Table 1: Decision Matrix

Concept Selection									
		Convective Oven		**Oil bath**		**Internal Resistive**		**Internal Induction**	
Selection Criteria	Weight	Rating	Weight Score	Rating	Weight Score	Rating	Weight Score	Rating	Weight Score
Performance	30%	4	1.2	5	1.5	3	0.9	3	0.9
Complexity	20%	4	0.8	3	0.6	2	0.4	2	0.4
Size	20%	3	0.6	2	0.4	4	0.8	3	0.6
Durability	15%	3	0.45	4	0.6	1	0.15	2	0.3
Cost	10%	2	0.2	2	0.2	3	0.3	3	0.3
User Friendly	5%	3	0.15	1	0.05	3	0.15	3	0.15
Total		3.4		3.35		2.7		2.65	
Ranking		1		2		3		4	

Based on the decision matrix it was decided that Concept A – Convective Oven was chosen as our final design. Even though it has almost the same ranking as the Concept B, after presenting the choices to the customer, he eliminated the oil bath because the compressor housing must remain dry.

System Analysis

Linear Expansion

The stator utilizes an interference fit with the housing. The fit is at maximum material condition between the stator and the housing. The required clearance for this type of fit is based on the clearance fit section of the machinist handbook. The insertion process utilizes a basic shaft into hole free fit, the appropriate clearance for this is 60 microns. The temperature needed to reach the desired clearance was calculated using the thermal expansion relationship (Equation 1). The only dimension of interest was the internal diameter of the housing. The linear expansion equation is valid because we are not concerned with how the rest of the housing expands.

$$\Delta \boldsymbol{L} = \boldsymbol{L_o} * \boldsymbol{\alpha} * \Delta \boldsymbol{T} \qquad \textbf{(1)}$$

Based on the supplied drawings the smallest diameter of the housing is 0.1715m, the maximum stator dimension is 0.1716m. The coefficient for thermal expansion for aluminum A356 is 0.00023 m/m°C. Ambient temperature is considered 25°C. Using this information the clearance between the two parts was calculated based on the final temperature of the housing (FIG 5). From the graph the final temperature of the part should be approximately 85 °C (185°F).

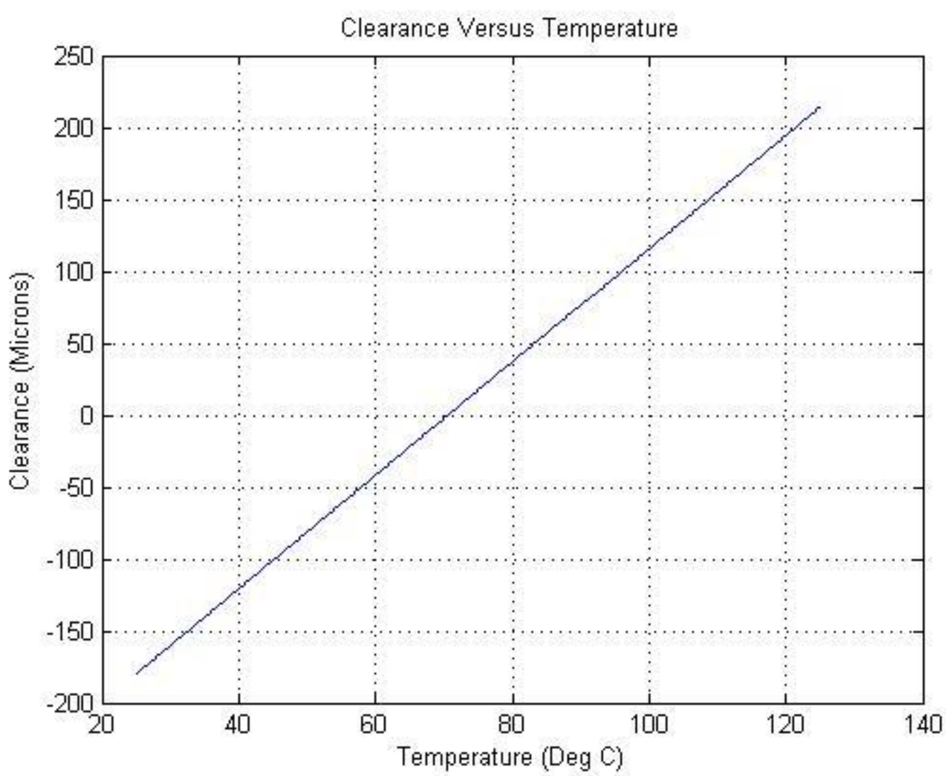


Figure 5: Linear Expansion versus Temperature

Internal Convection Coefficient

The internal heat convection coefficient is the primary mechanism of heat transfer to the housing. The nozzle attached to the end of the heater will promote turbulence and increase heat transfer. For our analysis the volumetric flow rate was varied until turbulent flow was achieved. Many of the parameters needed to calculate the convection coefficient are based on temperature of the medium. Our heating process is a transient problem in which the air temperature is constantly changing. In order to establish a temperature to find these properties at we needed to know the average air temperature. A MatLab program was created so that we could iterate until the average air temperature could be found. The initial properties were guessed. The program then finds the air temperature. The properties are then found for this new temperature. This process is completed until the temperature converges.

The first step was calculating the density of the air in the system. Air was treated as an ideal gas for our application because the reduced temperature was greater than 2, and the operating pressure was substantially lower than the critical pressure. The density was calculated using the ideal gas law (Equation 2). The next step was to calculate the Reynolds number so the proper Nusselt number correlation could be used (Equation 3).

$$\rho = \frac{P}{RT} \tag{2}$$

$$Re = \frac{\rho V D_h}{\mu} \tag{3}$$

For our system we simplified the housing geometry to a simple cylinder and treated the flow as flow within a pipe. The Reynolds number was 50,766. For turbulent flow in a tube the Dittus-Boelter equation was utilized to calculate the Nusselt number (Equation 4). Finally the convection coefficient was calculated for our system (Equation 5).

$$Nu = 0.023 Re^{0.8} Pr^{n} \tag{4}$$

$$h_i = \frac{Nuk}{D_h} \tag{5}$$

A detailed table of all the key property values can be found in Appendix II. The internal convection coefficient for our system based on a nozzle diameter of 10cm and a volumetric flow rate of 250 CFM was 15.9 W/m²K.

External Convection Coefficient

Natural convection is the mechanism of heat transfer from the exterior of our housing to the surrounding environment. Calculating the natural convection coefficient was a similar process to the forced convection coefficient. Many of the key properties needed are based on the film temperature, or the average between the surface and ambient temperatures. This again was an iterative process. Using the resistance network in the following section the properties were guessed. The total heat transfer was found and the surface temperature was solved for. This new temperature was then used to find the new film temperature. This process was repeated until the surface temperature converged. One of the key parameters in natural convection is the Graschof number (Equation 6). The Graschof number is a ratio of the buoyancy to viscous force acting on the fluid. The next step was calculated the Rayleigh number (Equation 7). Using the vertical plate correlation the Nusselt number was calculated (Equation 8). Finally the convection coefficient was calculated (Equation 9).

$$Gr = \frac{g\beta(T_s - T_\infty)L_c^3}{v^2} \tag{6}$$

$$Ra = GrPr \tag{7}$$

$$Nu = 0.59Ra^{\frac{1}{4}} \tag{8}$$

$$h_o = \frac{Nuk}{L_c} \tag{9}$$

A detailed table of all the key property values used for analysis can be found in Appendix II. Once the Nusselt number is calculated the natural convection coefficient was calculated. For our system the natural convection coefficient was 4.89 W/m²K.

Thermal Resistance Network

In order to calculate the heat loss from our system the thermal resistance approach was utilized. The total thermal resistance was the sum of the forced convection inside, the thermal conductivity of our hood material, and the external natural convection (FIG 6).

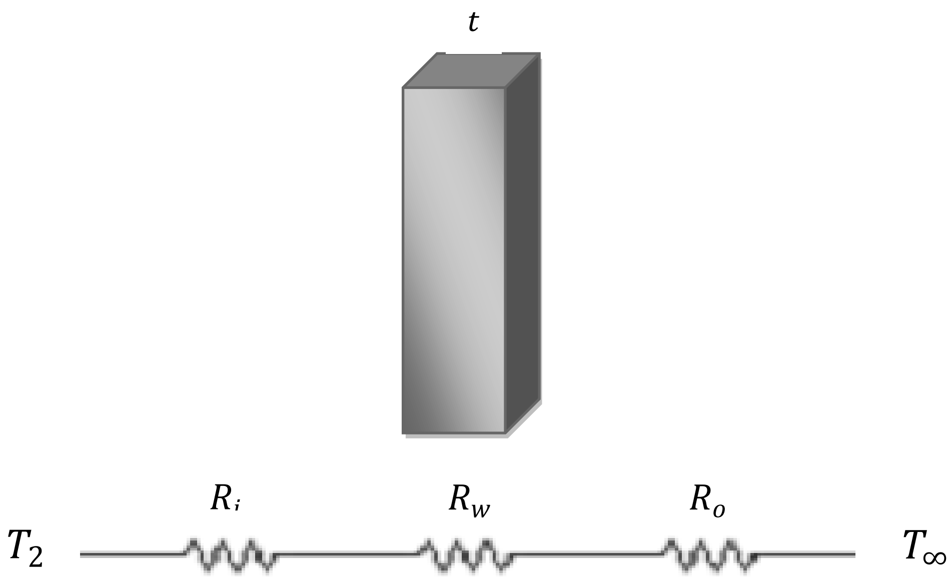


Figure 6: Thermal Resistance Network

The primary variables in the thermal resistance network are the hood material properties. For our hood material our first choice was polycarbonate. The thermal conductivity of Polycarbonate is 0.42 W/mK. For our system the Polycarbonate thickness was 10cm. The material and thickness are system variables that can be modified as needed during heat transfer analysis. In the figure the inside resistance is due to convection from the internal air to the hood surface (Equation 10). The wall resistance is the thermal resistance of the hood material (Equation 11). The outside resistance is due to natural convection from the hood to the ambient conditions (Equation 12).

$$\boldsymbol{R_i = \frac{1}{h_i A_{heated}}} \tag{10}$$

$$\boldsymbol{R_w = \frac{t}{k_{hood} A_{heated}}} \tag{11}$$

$$\boldsymbol{R_o = \frac{1}{h_o A_{heated}}} \tag{12}$$

All three resistances utilize the heated area. A detailed table of all the key property values can be found in Appendix II. For our system this is the hood surface area which is 3.29 m². The final calculation of our total thermal resistance was 0.1535 K/W.

Control Volume Analysis

The next phase in the analysis was to calculate the size of the heater we would need for our convection system. For analysis we considered the entire heating unit as a closed system. The main control volume will encompass the entire unit with a second control volume inside of the system (FIG 7).

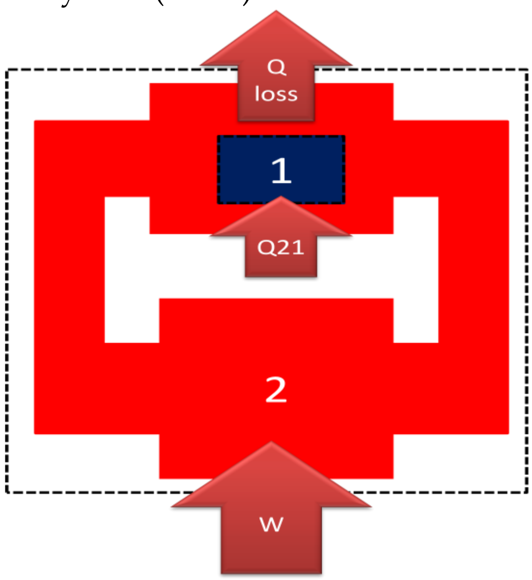


Figure 7: Control Volume

The only heat input into system one is the convective heat transfer from system two. System two loses heat to system one and through losses through the system. System two also receives power input from the heater. In order to complete the analysis the entire system was decomposed into two systems that depend on each other (FIG 8).

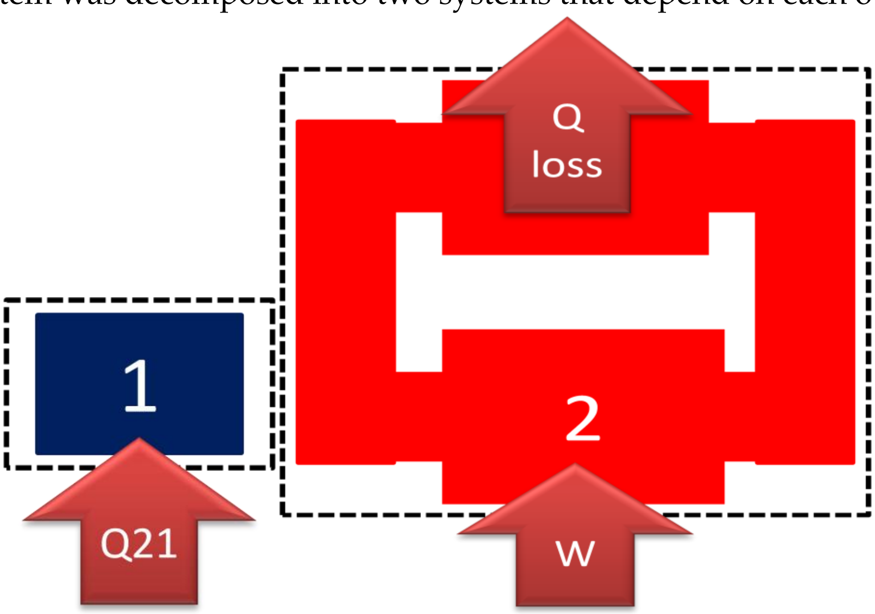


Figure 8: Control Volumes

Using the first law of Thermodynamics, equations were derived for the change in internal energy of the systems (Equation 13&14).

$$m_1 C_1 \frac{dT_1}{dt} = \dot{Q}_{21} \tag{13}$$

$$m_2 C_v \frac{dT_2}{dt} = \dot{W} - \dot{Q}_{12} - \dot{Q}_{loss} \tag{14}$$

The heat term from system two into system one is convective heat transfer from the air into the housing (Equation 15). The work term is the power rating of our heater. The heat loss term is based on a thermal resistance network from the inside of the system to the exterior (Equation 16).

$$\dot{Q}_{12} = h_i A_1 (T_2 - T_1) \tag{15}$$

$$\dot{Q}_{loss} = \frac{(T_2 - T_\infty)}{R_{tot}} \tag{16}$$

Simplifying the previous equations we end up with a coupled system of ordinary differential equations (Equation 17&18).

$$\frac{dT_1}{dt} = \frac{h_i A_1 (T_2 - T_1)}{m_1 C_1} \tag{17}$$

$$\frac{dT_2}{dt} = \frac{\dot{W}}{m_1 C_v} - \frac{h_i A_1 (T_2 - T_1)}{m_2 C_v} - \frac{(T_2 - T_\infty)}{R_{tot}} \tag{18}$$

Using MatLab a graph relating the temperature of both systems was generated (FIG 9). The MatLab code was created so that the heater value could be varied in order to find a sufficient heater in order to meet our heating time design requirements.

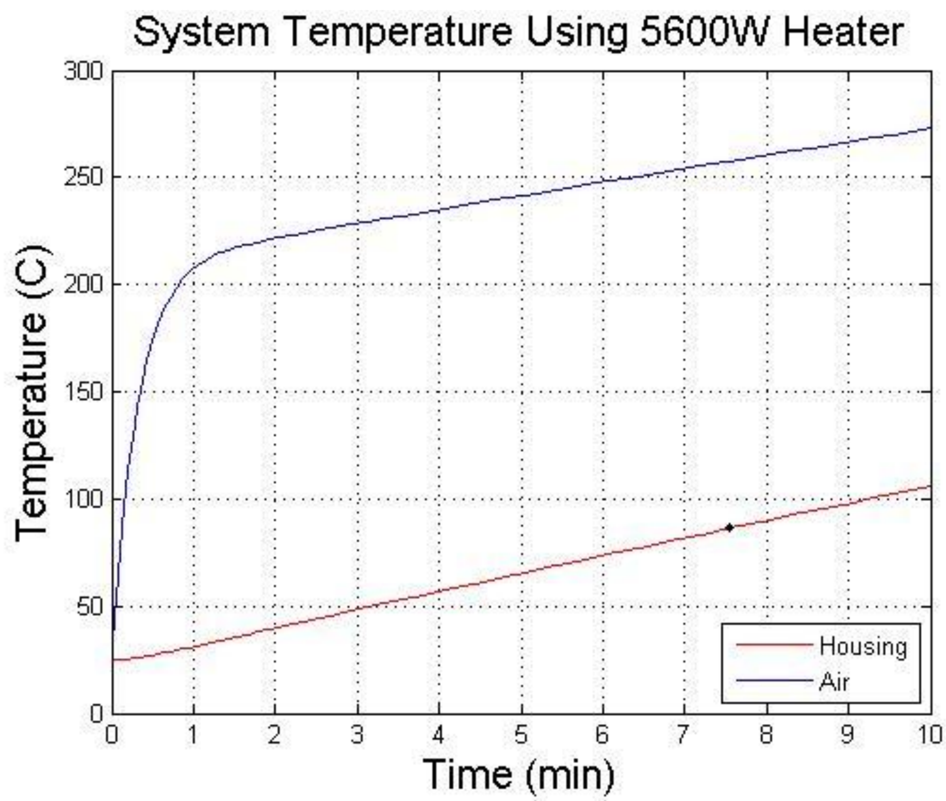


Figure 9: System Temperature versus Time

We should be able to reach the desired temperature of 85°C in less than 8 minutes. The maximum air temperature is also slightly under the 270°C melting temperature of Polycarbonate. If we have to change the system we may need to change the housing material and re-calculate all of the properties. Our MatLab simulation allows for this adjustment by changing the necessary values. A detailed table of all the key property values can be found in Appendix II.

Experimental Data

Experiment 1 was performed on site, at Turbocor to verify the linear expansion of the compressor housing. The housing was heated to more than 110 degrees Celsius and then the inner diameter and housing temperature measurement were taken with the bore gauge and thermocouples at specific time intervals as the housing was cooling down in ambient air. Once the experiment was completed we verified that the linear expansion relationship was valid (FIG 10). Ideally we would perform this experiment numerous times while testing many different housing locations. Unfortunately due to the nature of the manufacturing process at Turbocor, our oven time was limited. We feel our data does still verify that the critical final temperature of 85 °C will work for the purposes of this design. This final temperature is based on an initial temperature of 25 °C. Because the expansion formula is linear the actual change in temperature from needed is 60 °C.

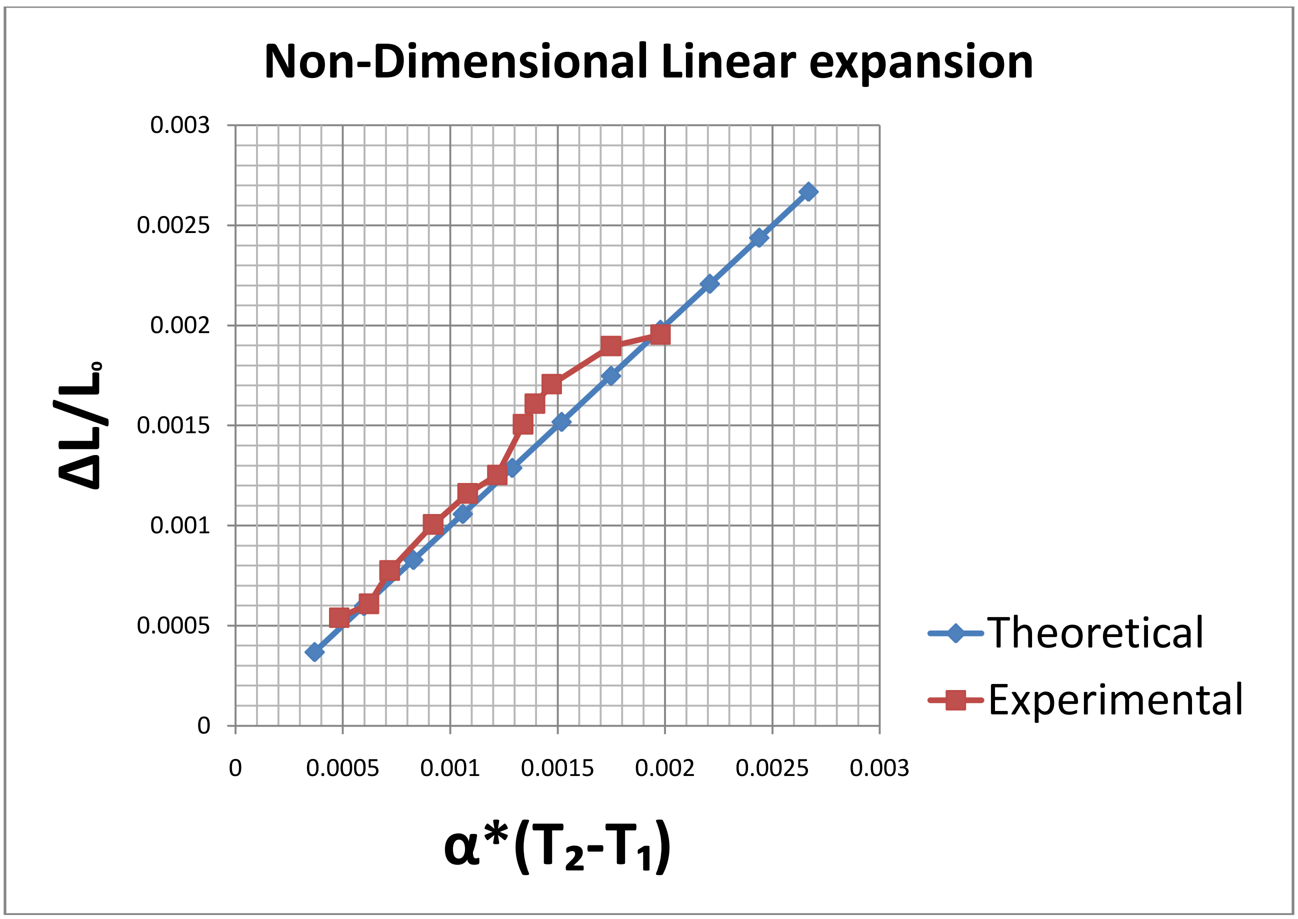


Figure 10: Theoretical & Experimental Data

System Optimization

The two critical parameters in our system are the internal convection coefficient and the heater wattage. In order to understand how these factors affect our system we used our Matlab program to plot the system response using various heater powers and convection coefficient (FIG 11).

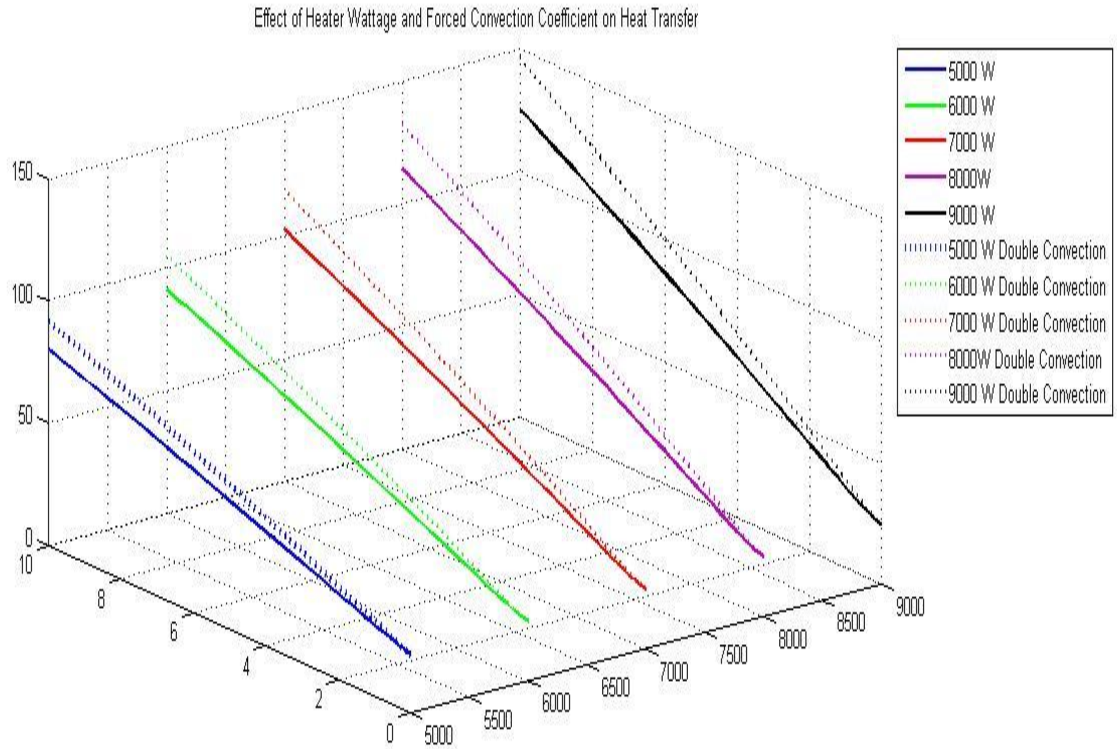


Figure 11: Heater Power & Convection Coefficients

An important trend in the data is that doubling the convection coefficient for a chosen heater size has nearly the same effect as raising the heater wattage by 1000W. Building a cost effective unit requires these properties to be analyzed. It could be possible to simply design a better nozzle or vary the flow rate to increase the convection coefficient. From our smoke tube analysis we have developed a nozzle that promotes turbulent flow within our system. The convection coefficient we used for our analysis was based on turbulent flow in a pipe. Our system obeys this initially but as the air passes through the top of the housing it then travels around the outside of the housing it is no longer flow in a pipe. There are no direct correlations for this type of flow. It is quite possible that the actual convection coefficient could be greater than what we calculated. When the flow changes direction it will become more turbulent and perhaps increase the convection coefficient. The only way to verify our model is to create an actual proof of concept and see how it reacts to the heater we have selected. If the initial nozzle doesn't work as expected we have several others to try. The blower we have also has the ability to vary the flow rate.

Increasing the heater size is another possible way to decrease the length of heating time. This is a very effective, but expensive way to increase the heat transfer into the housing. A off the shelf resistive heater bigger than the one we selected (5600W) will be difficult to come by. We found several companies online that can custom fabricate a specific heater but they are very expensive. This will be a last possible resort for us.

Material Selection and Cost Analysis

This is a preliminary cost analysis of the project to this date. The electric heater from MSC has already been purchased as well as the sheet metal and the flex hose for the purposes of the proof of concept requested by the sponsor. Full scale prototyping will begin during spring semester. This is just an estimated total for the project so far, detailed analysis of the parts and hardware required will be described in the final stages of the design.

Table 2: Cost Analysis

Table 2: Convection Heater Cost Analysis					
Part	**Description**	**Supplier**	**Unit Price ($)**	**QTY**	**Total Price ($)**
Electric Heater	5600W 100 CFM	MSC	138.09	1	138.09
Flange Mount Blower	250 CFM	MSC	112.59	2	225.18
Polycarbonate	96" x 48" x 3/8"	MSC	381.57	1	381.57
Expanded Sheet Metal	1/2" x 12" x 24" 18 Gauge	Lowes	9.37	2	18.74
Ultra Flex Hose	5' Length 4" ID	MSC	74.24	1	74.24
80/20 Extrusion	25 Series Mono Slot Bar 6m	TBD	48.00	4	192.00
80/20 Hardware	Misc. Hardware	TBD	100.00	1	100.00
				Total	**1129.82**

Prototype

The following images are of the preliminary design of the heater station assembly. The overall design consists of the table that is a part of the production line. Below the table top, there are 2 sealed off sections, the central one houses the heater and the shroud. This section has a hot air inlet on the bottom below the heater. After passing through the heater, hot air goes through the shroud and the nozzle and exits out through the table top, heating up the compressor housing. Air is then diverted by the hood back down and through the table slits that allow the air to enter the second sealed chamber, from which it is sucked out by the heater fan and redirected back to the heater in a continuous cycle. Above the table top rests the hood which has openings on two sides to allow the compressor housings to slide in and out during production, as well as an exhaust fan which is utilized during the cooling cycle.

1 Housing
2 Heater
3 Hood
4 Exhaust Fan
5 Table
6 Heater Fan
7 Shroud
8 Nozzle

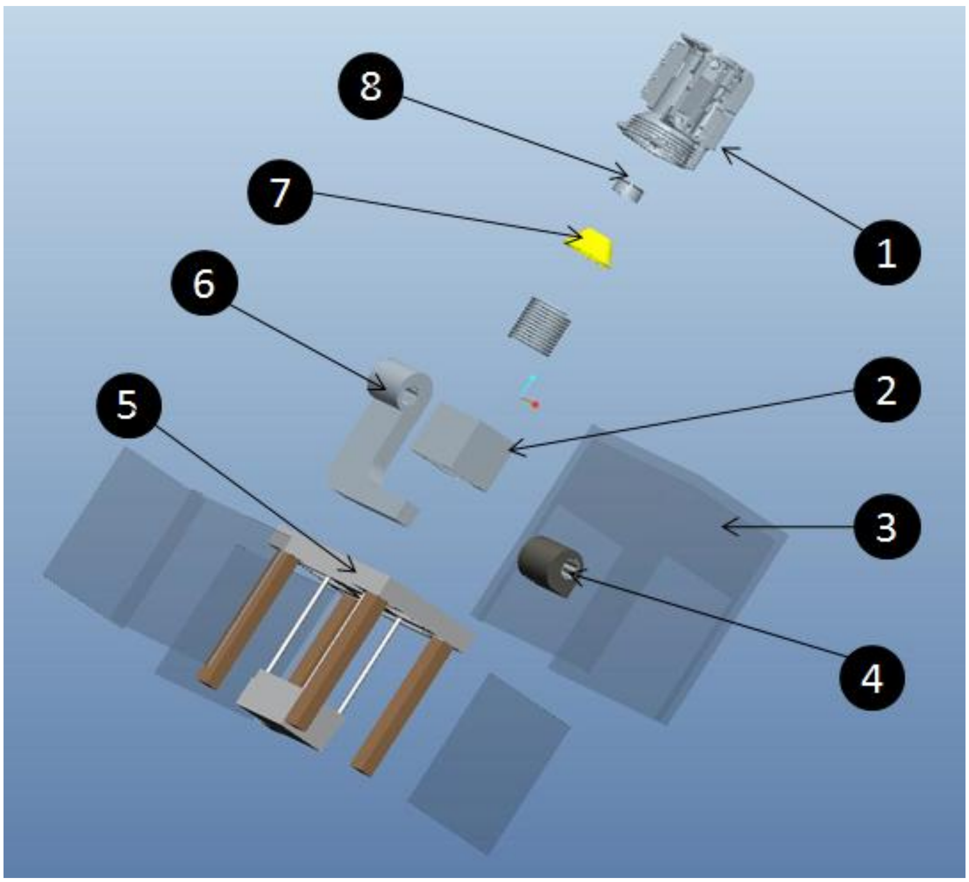


Figure 12: Heater Design Prototype, Exploded View

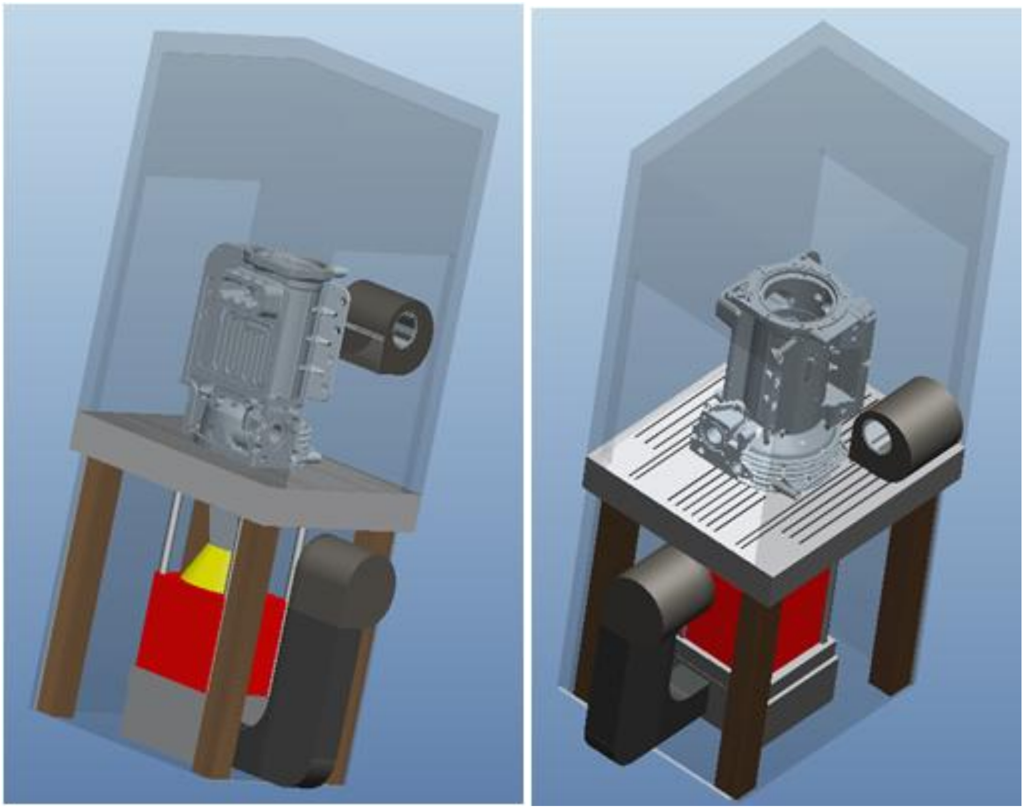


Figure 13: Heater prototype, Assembled view

Fabrication and Testing

As previously mentioned heater, flex hose and sheet metal have already been purchased for the purposes of proof of concept testing on site at Turbocor. This test will show can the purchased heater perform in the specified parameters. Full scale prototype build will start during spring semester right after the New Year's break.

References

- "Aluminum A356 T6 Properties." N.p., n.d. Web. 15 Nov 2010. <http://www.matweb.com/>.
- "Linear Expansion." N.p., n.d. Web. 15 Nov 2010. <http://hyperphysics.phy-astr.gsu.edu>.
- *Engineering Tool Box*. N.p., n.d. Web. 15 Nov 2010. <http://www.engineeringtoolbox.com/>.
- Cengel, Turner, Cimbala. *Thermal Fluid Sciences.* New York: McGraw Hill, 2008

Appendix I: Calculations

Thermal Expansion

$\Delta L = L_o * \alpha * \Delta T$

$\frac{\Delta L}{L_o} = \alpha * \Delta T$

Variable Definition

$\Delta L = Change\ in\ length$

$L_o = Initial\ length$

$\alpha = \ Thermal\ expansion\ coefficient$

$\Delta T = Change\ in\ temperature$

Thermal Expansion Sample Code

This code is specific to our particular housing.

```
%Calculating the clearance between the housing and the stator
%based on temperature. Linear thermal expansion of the radius is assumed
%Calculations assume thermal equilibrium within the material
%The calculation of heat needed to reach required
%temperature is based on second law
%Heat required is mC*(t2-t1)
%Specific Heat and Density designated 1 are for pure aluminum
```

```
%Specific Heat and Density designated 2 are for 2024 aluminum


clear all;
%Smallest possible part diameter based on drawing
Dinit = .171450;
%Maximum stator size
Stat = .171630;
%Linear expansion coeffient for aluminum (units m/m*degC)
alpha = 0.000023;
r2 = 0.230;
r1 = 0.171450;
h = 0.20065;
V = 0.012688855;
Tinit = 25;
Rho1 = 2670;
C1 = 963;
Q1 = zeros(1,201);
TfinalF = zeros(1,201);
TfinalC = zeros(1,201);
LDelta = zeros(1,201);
Linit = pi*Dinit;
DDelta = 0;
TDelta = 0;
Dnew = zeros(1,201);
Cler = zeros(1,201);
i = 0;
for t = 25:0.5:125
   i = i + 1;
   TDelta = t-Tinit;
   LDelta = alpha*Linit*TDelta;
   DDelta = LDelta/pi;
   Dnew = (Dinit + DDelta);
   Cler(i) = (Dnew-Stat)*10^6;
   TfinalF(i) = (9/5)*t+32;
   TfinalC(i) = t;
   Q1(i) = (Rho1*V*C1*(t-Tinit))/1000;

end
figure(1)

plot(TfinalF,Cler),title('Clearance Versus Temperature'),
xlabel('Temperature (Deg F)'),ylabel('Clearance (Microns)'),grid on

figure(2)
```

```
plot(TfinalC,Cler),title('Clearance Versus Temperature'),
xlabel('Temperature (Deg C)'),ylabel('Clearance (Microns)'),grid on

figure(3)

plot(TfinalC,Q1),title('Heat Versus Temperature (Aluminum A356 T6)'),
xlabel('Temperature (Deg C)'),ylabel('Heat (kJ)'),grid on

figure(4)

plot(TfinalF,Q1),title('Heat Versus Temperature (Aluminum A356 T6)'),
xlabel('Temperature (Deg F)'),ylabel('Heat (kJ)'),grid on
```

Heat Input

Heat Input Sample Code

This code allows the user to input a desired clearance in mcrons. The program returns the final temperature and the needed heat input.

```
%Calculating the temperature needed to reach desired clearance
%between the housing and the stator.
%Linear thermal expansion of the radius is assumed
%Calculations assume thermal equalibrium within the material
%Calculating the Heat needed to reach desired Temperature
%Treating the housing as a cylinder


clear all;
%Smallest possible part diameter based on drawing
Dinit = .171450;
%Maximum stator size
Stat = .171630;
%Linear expansion coeffient for aluminum (units m/m*degC)
alpha = 0.000023;
CL = input('Input desired clearance in microns >');
TI = input('Input initial temperature in degrees C >');
Rho = 2670;
C = 963;


DNew = (CL*10^-6)+Stat;
DeltaD = DNew - Dinit;
DeltaT = DeltaD/(Dinit*alpha);
TC = DeltaT + TI;
TF = (9/5)*TC+32;
```

```
r2 = 0.21910;
r1 = 0.171450;
V = 0.012688855;
Heat = (Rho*V*C*(TC-TI))/1000;



sprintf('Required Temperature is %.2f deg F',TF)
sprintf('Required Temperature is %.2f deg C',TC)
sprintf('Required Energy is %.2f kJ',Heat)
```

Internal Convection Coefficient

$\rho = \frac{P}{RT}$

$Re = \frac{\rho V D_h}{\mu}$

$Nu = 0.023 Re^{0.8} Pr^n$

$h_i = \frac{Nuk}{D_h}$

Variable Definitions

$P = Static\ Pressure$

$R = Gas\ Constant$

$T = Tempertaure$

$V = Velocity$

$D_h = Hydraulic\ Diameter$

$\mu = Dynamic\ Viscocity$

$Pr = Prandlt\ Number$

$n = Cooling\ Exponent$

$k = Thermal\ Conductivity$

Natural Convection Coefficient

$Gr = \frac{g\beta\,(T_s - T_\infty)L_c^3}{v^2}$

$Ra = GrPr$

$Nu = 0.59Ra^{\frac{1}{4}}$

$h_o = \frac{Nuk}{L_c}$

Variable Definitions

$g = Gravitational\ Constant$

$\beta = Volume\ Expansion\ Coefficient$

$T_s = Surface\ Temperature$

$T_\infty = Ambient\ Temperature$

$L_c = Characteristic\ Length$

$v = Kinematic\ Viscocity$

$Pr = Prandlt\ Number$

Thermal Resistance Network

$R_i = \frac{1}{h_i A_{heated}}$

$R_w = \frac{t}{k_{hood}\, A_{heated}}$

$R_o = \frac{1}{h_o A_{heated}}$

$R_{tot} = R_i + R_w + R_o$

$\dot{Q}_{loss} = \frac{(T_2 - T_\infty)}{R_{tot}}$

$T_2 = Hood\ Surface\ Temperature$

$T_\infty = Temperature\ of\ Surrounding\ Air$

$A_{heated} = Heated\ Housing\ Surface\ Area$

$t = Hood\ Material\ Thisckness$

$k_{hood} = Hood\ Material\ Thermal\ Conductivity$

$h_o = Outside\ Natural\ Convection\ Coefficient$

$h_i = Inside\ Convection\ Coefficient$

Control Volume Analysis

System 1 (Housing)

$$m_1 C_1 \frac{dT_1}{dt} = \dot{Q}_{21}$$

$$\dot{Q}_{12} = h_i A_1 (T_2 - T_1)$$

$$\frac{dT_1}{dt} = \frac{h_i A_1 (T_2 - T_1)}{m_1 C_1}$$

System 2(Air in system)

$$m_2 C_v \frac{dT_2}{dt} = \dot{W} - \dot{Q}_{12} - \dot{Q}_{loss}$$

$$\dot{W} = Heater\ Power$$

$$\frac{dT_2}{dt} = \frac{\dot{W}}{m_1 C_v} - \frac{h_i A_1 (T_2 - T_1)}{m_2 C_v} - \frac{(T_2 - T_\infty)}{R_{tot}}$$

Final Coupled ODE System

$$\frac{dT_1}{dt} = AT_1 - AT_2$$

$$\frac{dT_2}{dt} = BT_1 - CT_2 + D$$

$$A = \frac{h_i A_1}{m_1 C_1}$$

$$B = \frac{h_i A_1}{m_2 C_v}$$

$$C = \frac{h_i A_1}{m_2 C_v} + \frac{1}{m_2 C_v R_{tot}}$$

$$D = \frac{\dot{W}}{m_2 C_v} + \frac{T_\infty}{m_2 C_v R_{tot}}$$

Variable Definitions

$h_i = Inside\ Convection\ Coefficient$

$A_1 = Heated\ Housing\ Surface\ Area$

$m_1 = Mass\ of\ Housing$

$C_1 = Specific\ Heat\ of\ Aluminum\ Housing$

$m_2 = Mass\ of\ Air$

$C_v = Specific\ Heat\ of\ Air\ at\ Constant\ Volume$

$R_{tot} = Total\ Thermal\ Resistamce$

$\dot{W} = Heater\ Power$

$T_\infty = Temperature\ of\ Surrounding\ Air$

$A_{heated} = Heated\ Housing\ Surface\ Area$

$t = Hood\ Material\ Thisckness$

$k_{hood} = Hood\ Material\ Thermal\ Conductivity$

$h_o = Outside\ Natural\ Convection\ Coefficient$

Control Volume Analysis Sample Code

Initial System

This first code was used to evaluate all of the parameters for our system. All of the important variables are commented within the program.

```
function Tprime = HeatMaster(t,T)
%Estimating the time to heat the housing varying the heater wattage and
%the insulation of the hood.  This code was also used to determine
%film temperature to evaluate the necessary properties and determine
%convection coefficients



%System 1 (Housing) Properties
m1 = 32;                %mass
C1 = 900;               %Specific Heat
d2 = 0.2191;            %outer diameter
d1 = 0.17145;           %inner diameter
h = 0.5;                %height
A1 = pi*d2*h+pi*d1*h;   %heat transfer area

%System 2 (Air) Properties
%Control Volume
Lcv = 0.5;              %Length
Wcv = 0.5;              %Width
Hcv = 0.5;              %Height
Vcv = Lcv*Wcv*Hcv;      %Volume
%Hood Volume
L = 0.7;                %Length
W = 0.7;                %Width
H = 1.0;                %Height
Vhood = L*W*H;          %Hood Volume
%Duct Volume
Ld = 1;                 %Duct Length
Wd = 0.5;               %Duct Width
Vduct = Ld*(Wd^2);      %Duct Volume
%Cabinet Volume
Wc = 0.5;               %Cabinet Width
Hc = 1.0;               %Cabinet Height
Vcab = Hc*(Wc^2);       %Cabinet Volume
Vtotal = Vcv+Vhood+Vduct+Vcab;  %Total Volume of Air in system
%Air Properties
Pr = 101000;            %Static Pressure
Tc = 225;               %Air Temp           VARIABLE
Tk = Tc+273;            %Temp Kelvin
R = 287;                %Gas Constant
m2 = (Pr*Vtotal)/(R*Tk);    %Gas mass
Cp = 1013;              %Specific heat Cp          VARIABLE
```

```
Cv = Cp-R;               %Specific heat Cv

%Hood Properties
t = 0.1;                 %Thickness
Lh = L;                  %Length of 1 side
Wh = W;                  %width of 1 side
Hh = H;                  %Height of 1 side
Ah = 4*(Hh*Wh)+(Lh*Wh);  %Heat transfer surface area
kh = 0.42;               %Thermal Conductivity hood      VARIABLE

%Inside Convection Properties
Qus = 250;               %Volumetric flow (US)         VARIABLE
Qsi = Qus*0.0004719;     %Volumetric flow (SI)
rho = (Pr/(R*Tk));       %Density
Nd = 0.1;                %Nozzle Diameter              VARIABLE
Dh = 0.132;              %Hydraulic Diameter           VARIABLE
V = Qsi/((pi*Nd^2)/4);   %Velocity
mu = 2.76*10^-5;         %Dynamic Viscosity @250C      VARIABLE
Pr = 0.6946;             %Prandlt        @250C      VARIABLE
ka = 0.04104;            %Thermal Cond.    @250C       VARIABLE
Re = (rho*V*Dh)/mu;      %Reynolds #
n = 0.3;                 %Cooling Exponent
Nuf = 0.023*(Re^0.8)*Pr^n;  %Nusselt Dittus-Boeler corr.
hi = (Nuf*ka)/Dh;        %Convection Inside

%Outside Convection Properties
Tinf = 25;               %Temp far away
Tsurf = 121;             %Surface Temp                 VARIABLE
Tfilm = (Tinf+Tsurf)/2;  %Film Temp
TfilmK = Tfilm+273;      %Film temp K
g = 9.80;                %Gravity constant
beta = 1/TfilmK;         %Beta
Lc = Hh;                 %Characteristic Length
vn = 1.995*10^-5;        %Kinemetic visc.   @Tfilm      VARIABLE
Prn = 0.7177;            %Prandlt         @Tfilm      VARIABLE
kn = 0.02881;            %Thermal Cond.    @Tfilm      VARIABLE
Gr = (g*beta*(Tsurf-Tinf)*Lc^3)/(vn^2); %Graschoff
Ra = Gr*Prn;             %Raleigh Number
Nun = 0.1*Ra^(1/3);      %Nusselt natural
ho = (Nun*kn)/Lc;        %Convection inside



%Resistance Network
Ri = 1/(Ah*hi);          %Inside Resistance
Rw = t/(kh*Ah);          %Wall Resistance
```

```
Ro = 1/(Ah*ho);           %Outside Resistance
Rtot = Ri+Rw+Ro;           %Total Resistance

%Heater Properties
W = 5200;                 %Heater Power

%Simplifications
A = (hi*A1)/(m1*C1);
B = (hi*A1)/(m2*Cv);
C = ((hi*A1)/(m2*Cv))+(1/(m2*Cv*Rtot));
D = (W/(m2*Cv))+(Tinf/(m2*Cv*Rtot));

Tprime = [A*T(2)-A*T(1);B*T(1)-C*T(2)+D];
```

Solving the initial system

```
%This Program uses the coupled system of Ode
%to generate temperature profiles for the housing and air temperatures
clear all;

T0 = [25,25];
tspan = [0,600];
[t,T] = ode45(@HeatMaster,tspan,T0);
Tmin = t/60;
figure(1)
plot(Tmin,T(:,1),'r')
figure(2)
plot(Tmin,T(:,2),'b')
```

Control Volume Analysis Parameter System

This code allows variables to be passed to the ODE to see how the system reacts to changing variables.

```
function Tprime = HeatA(t,T,p)
%Estimating the time to heat the housing varying the properties
%listed as Variables
%VARIABLES
hi = p(1);       %Inside Convection           (37.2)
ho = p(2);       %Outside Convection           (4.89)
m2 = p(3);       %Air Mass                 (0.7879)
Cp = p(4);       %Specific Heat Air          (1013)
C1 = p(5);       %Specific Heat Aluminum       (900)
Ah = p(6);       %Hood Surface Area          (3.2900)
t = p(7);        %Hood Thickness              (0.1)
```

```
k = p(8);          %Hood Thermal Conductivity     (0.42)
W = p(9);          %Heater Wattage                (4000)

%CONSTANTS
A1 = 0.6135;       %Housing Area
R = 287;           %Gas Constant
m1 = 32;           %Housing Mass
Tinf = 25;         %Temperature Far Away

%CALCULATIONS
Cv = Cp-R;         %Specific Heat constant volume
Ri = 1/(hi*Ah);    %Inside Resistance
Ro = 1/(ho*Ah);    %Outside Resistance
Rw = t/(k*Ah);     %Wall Resistance
Rtot = Ri+Ro+Rw;   %Total Resistance



%Simplifications
A = (hi*A1)/(m1*C1);
B = (hi*A1)/(m2*Cv);
C = ((hi*A1)/(m2*Cv))+(1/(m2*Cv*Rtot));
D = (W/(m2*Cv))+(Tinf/(m2*Cv*Rtot));

Tprime = [A*T(2)-A*T(1);B*T(1)-C*T(2)+D];
```

Solving the variable system.

```
%This Program uses the coupled system of Ode
%to generate temperature profiles for the hoousing and air temperatures
%the p vector passes the variables to the ODE
clear all;

T0 = [25,25];
tspan = [0,980];                  %Time in seconds
p = [37.2 4.89 0.7879 1013 900 3.29 0.1 0.42 5200];
%Parameters
%[A B C D E F G H I]
%A = Inside convection coefficient        37.2
%B = Outside convection coefficient       4.89
%C = Air mass                       0.7879
%D = Specific heat of air              1013
%E = Specific heat of aluminum            900
%F = Hood surface area                 3.29
%G = Hood thickness                    0.1
%H = Hood thermal conductivity            0.42
%I = Heater Wattage                    4000
```

```
[t,T] = ode45(@HeatA,tspan,T0,[],p);
Tmin = t/60;
%PLOTS

figure(1)
plot(Tmin,T(:,1),'r'),
title('Housing Temperature'),xlabel('Time (min)'),
ylabel('Temperature (C)'),grid on

figure(2)
plot(Tmin,T(:,2),'b'),title('Air Temperature'),xlabel('Time (min)'),
ylabel('Temperature (C)'),grid on

figure(3)
plot(Tmin,T(:,1),'r',Tmin,T(:,2),'b'),title('System Temperature'),
xlabel('Time (min)'),ylabel('Temperature (C)'),grid on,
legend('Housing','Air','location','SouthOutside')
```

Appendix II: Property Tables

Table 3: Housing Properties

System 1 Properties (Housing)		
Property	**Unit**	**Value**
Mass	kg	32
Specific Heat	m	900
Outer Diameter	m	0.2191
Inner Diameter	m	0.17145
Height	m	0.5
Surface Area	m^2	0.6135

Table 4: Air Volume Properties

System 2 Air Volume Properties		
Property	**Unit**	**Value**
Heater		
Length	m	0.5
Width	m	0.5
Height	m	0.5
Heater Volume	m^3	0.125
Hood		
Length	m	0.7
Width	m	0.7
Height	m	0.7
Hood Volume	m^3	0.49
Duct		
Length	m	1
Width	m	0.5
Duct Volume	m^3	0.25
Cabinet		
Height	m	0.5
Width	m	1
Total Volume	m^3	0.25

Table 5: Air Properties at 250 deg C

System 2 Properties (Air) Evaluated at 250°C		
Property	Unit	Value
Static Pressure	kPa	101
Air Temp	°C	225
Air Temp	K	498
Gas Constant	J/kgK	287
Mass	kg	0.7879
Specific Heat (Constant Pressure)	J/kgK	1033
Specific Heat (Constant Volume)	J/kgK	746

Table 6: Convection Properties

Inside Convection Properties Evaluated at 250°C		
Flow Rate	CFM	250
Flow Rate	m^3/s	0.118
Density	kg/m^3	0.7067
Nozzle Diameter	m	0.1
Hydraulic Diameter	m	0.132
Velocity	m/s	15.02
Dynamic Viscosity	kg/ms	2.76E-05
Prandtl	N/A	0.6946
Thermal Conductivity	W/mK	0.041
Reynolds	N/A	1.76E+04
Cooling Exponent	N/A	0.3
Nusselt	N/A	51.28
Forced Convection Coefficient	W/m^2K	15.94

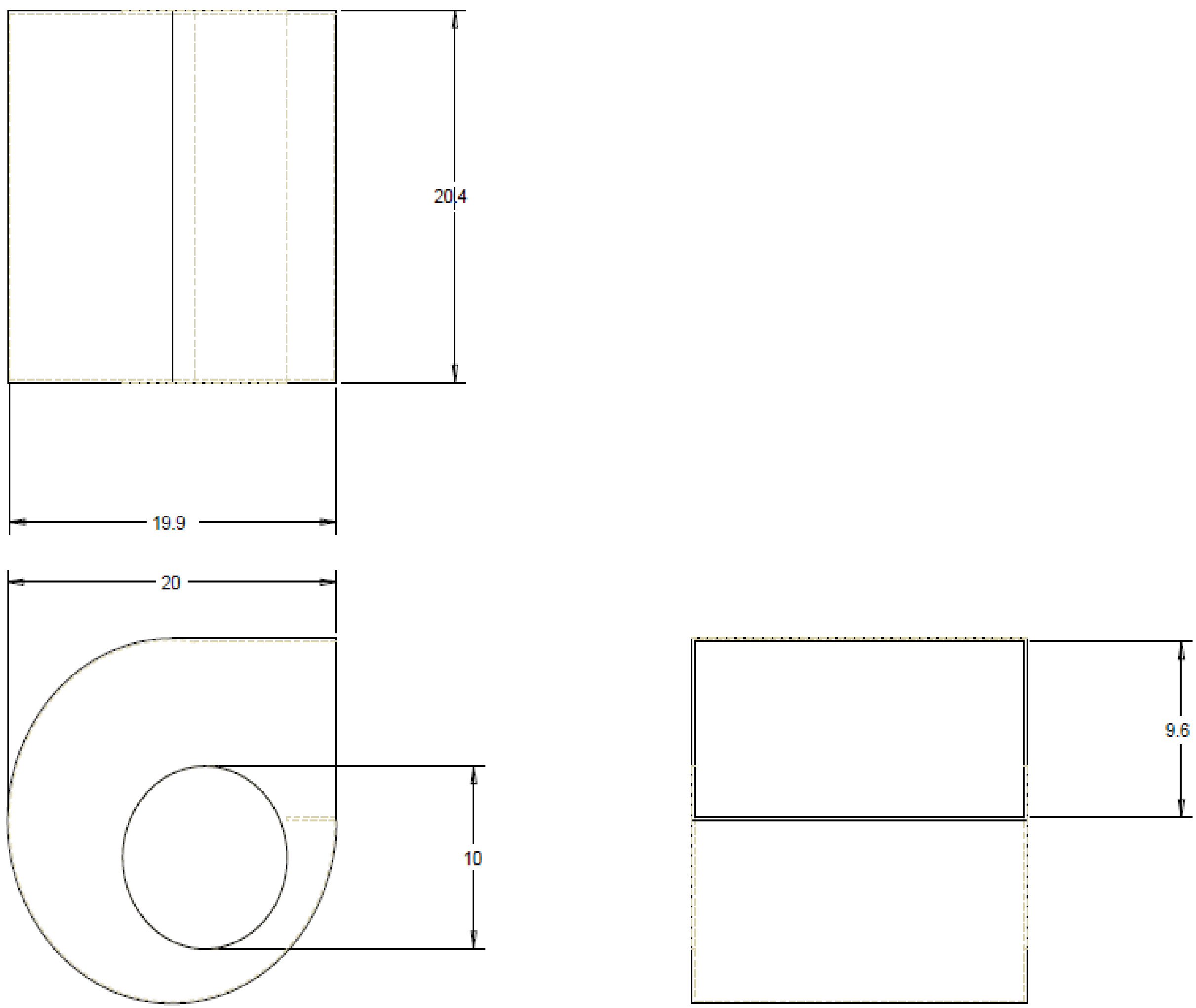
Table 7: Natural Convection Properties

Natural Convection Properties Evaluated at Film Temperature (73°C)		
Temperature Far Away	°C	25
Temperature Surface	°C	121
Film Temperature	°C	73
Film Temperature	K	346
Gravitational Constant	m/s^2	9.8
Beta	1/K	0.0029
Characteristic Length	m	1
Kinematic Voscocity	m^2/s	2.00E-05
Prandtl	N/A	0.7177
Thermal Conductivity	W/mK	0.02881
Graschoff	N/A	6.83E+09
Raleigh	N/A	4.90E+09
Nusselt	N/A	169.88
Natural Convection Coefficient	W/m^2K	4.49

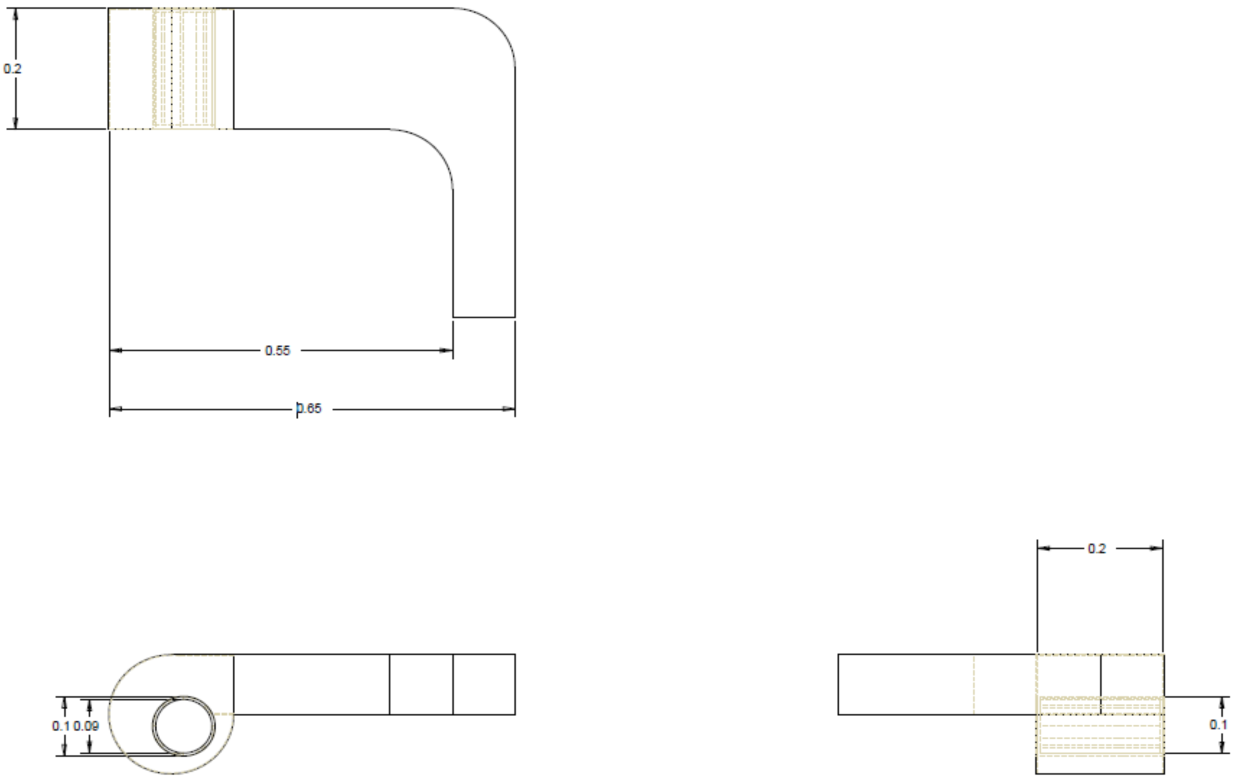
Table 8: Resistance Network Values

Resistance Network		
Hood Properties		
Thickness	m	0.1
Length	m	0.7
Width	m	0.7
Height	m	1
Hood Surface Area	m^2	3.29
Thermal Conductivity	W/mK	0.42
Inside Resistance	K/W	0.019
Wall Resistance	K/W	0.072
Outside Resistance	K/W	0.062
Total Resistance	K/W	0.154

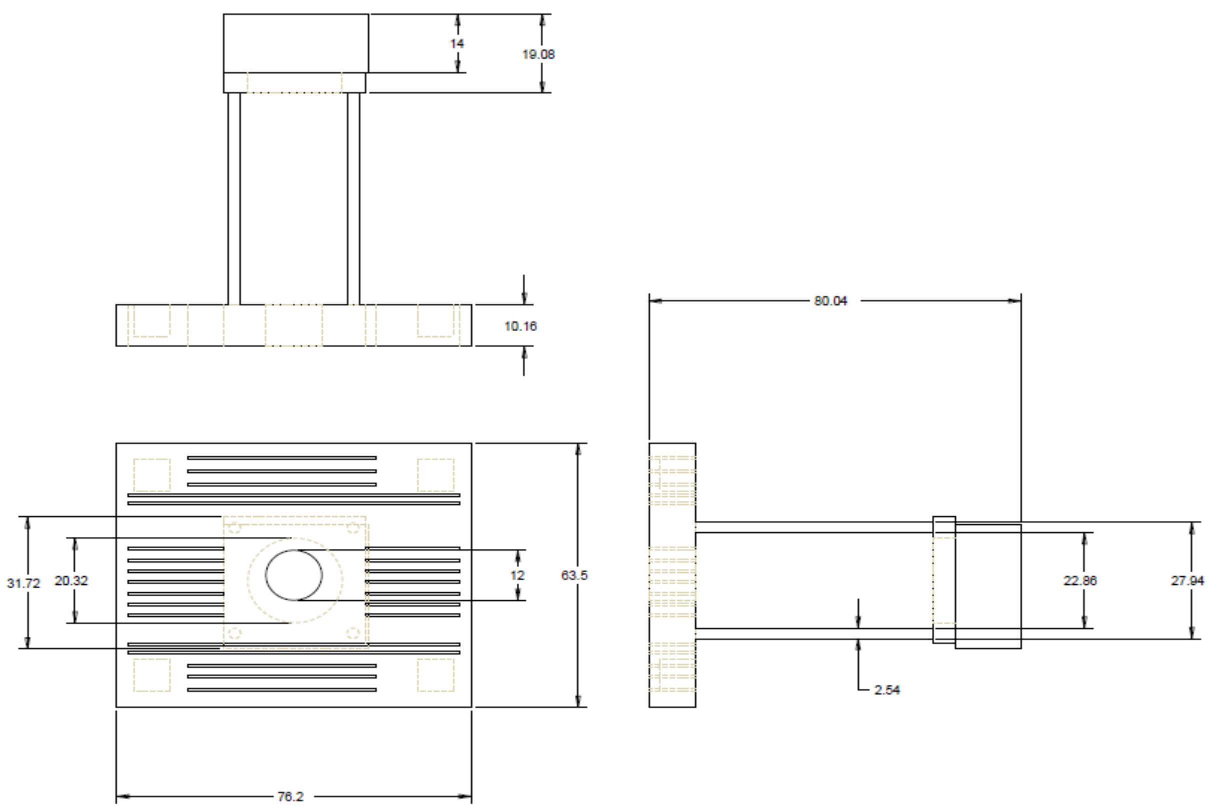
Appendix III: Pro Engineer Drawings



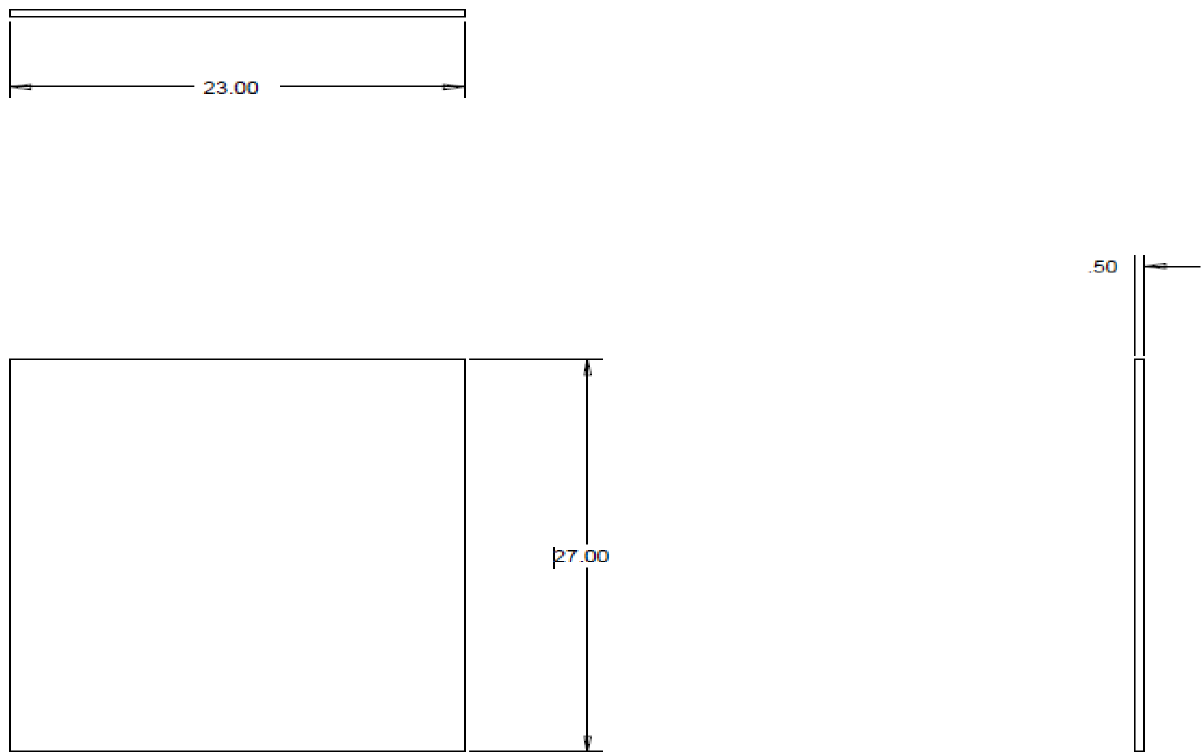
Drawing 1: Cooling Blower



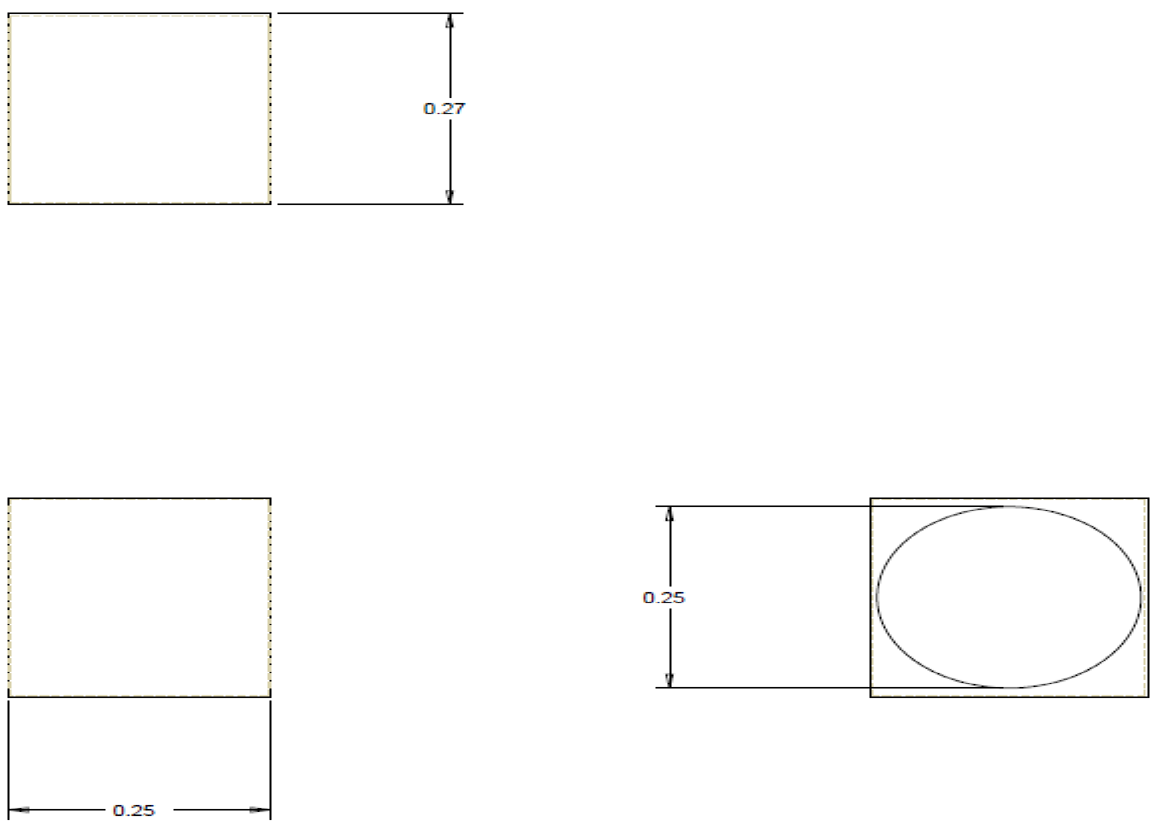
Drawing 2: Heater Fan & Duct



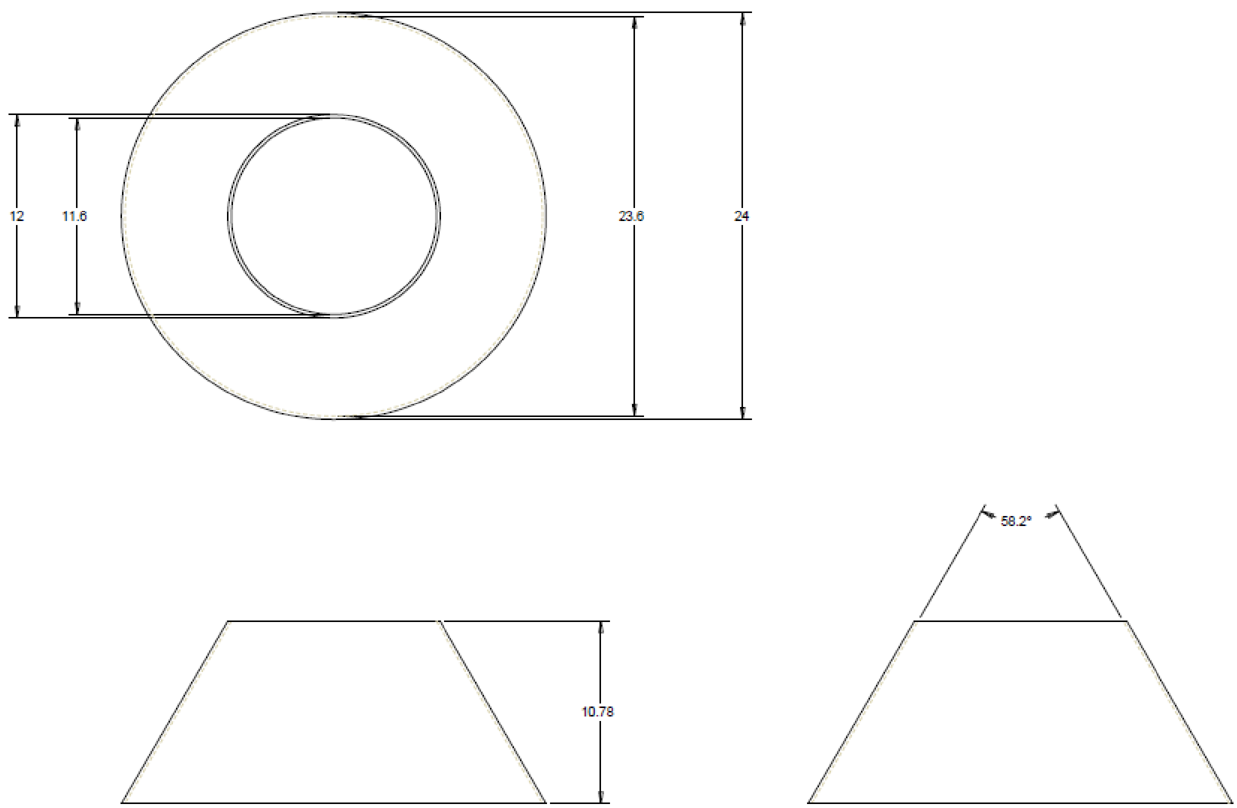
Drawing 3: Table



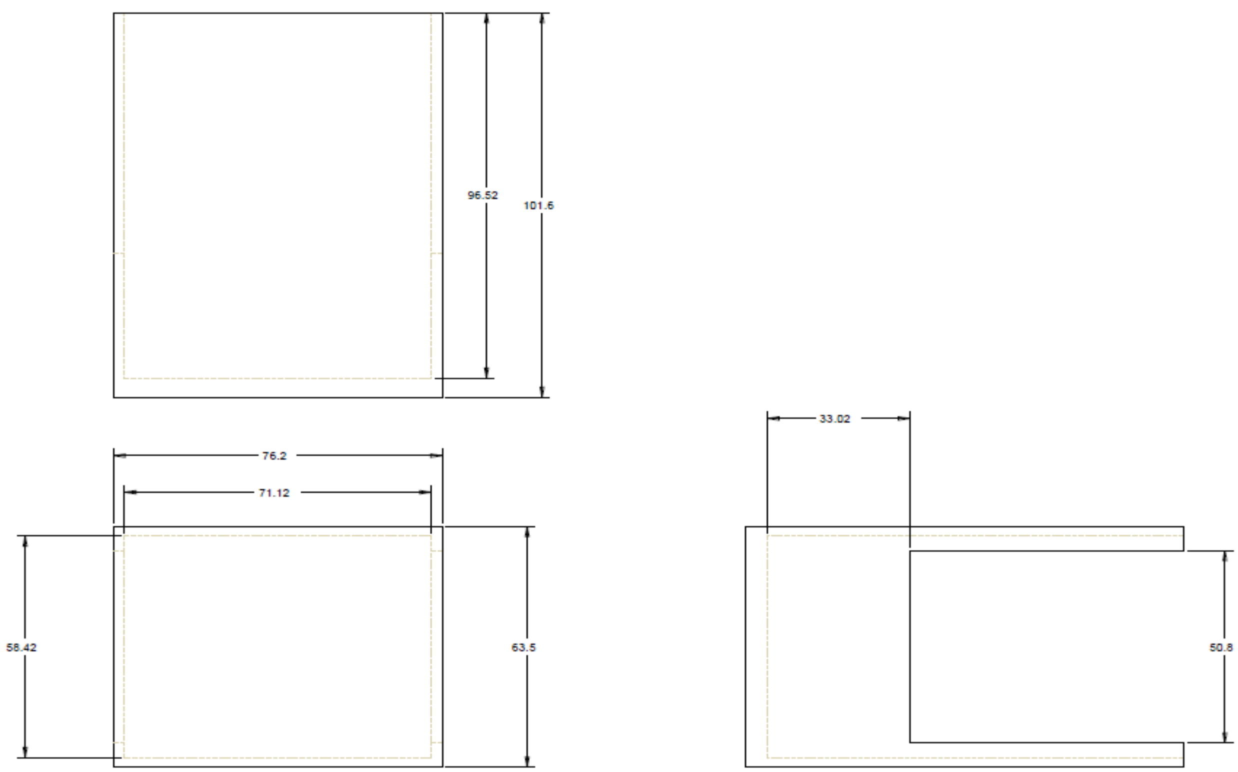
Drawing 4: Side Panel



Drawing 5: Heater, Outside Dimensions



Drawing 6: Heater to Nozzle Shroud



Drawing 7: Hood