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# Bubble Pump Cooling System

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**Indiana University-Purdue University Fort Wayne**

**Department of Engineering**

ME 487 – ME 488

Capstone Senior Design Project

*Report # 2*

Project Title: Bubble Pump Cooling System

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- Steve O'Shaughnessy Corporate Advisor
- Parker Hannifin Precision Cooling Business Unit Corporate Sponsor

## **Abstract/Summary**

The scope of our project was to design a bubble pump cooling system that could be used as a demonstration unit for potential customers of Parker Hannifin. The bubble pump system is a favorable design over conventional designs due to its reliability and low cost since there is no need for electricity to pump the fluid and no moving parts to fail. This system was required to remove 400W of heat from an Insulated Gate Bipolar Transistor (IGBT) without allowing its surface temperature to exceed 80°C.

Our design of the bubble pump was based on the theories of conduction and two phase flow in a pipe. As an electronic component increases in temperature this heat is conducted into a cold plate which in turn begins to boil. As the working fluid methanol begins to boil the vapor rises to the top of the cold plate and into the bubble pump tube. Similar to a piston cylinder action as the vapor rises in the tube the liquid is forced upward into an upper reservoir. This liquid and vapor is then condensed as the heat is removed and is returned to the cold plate as a saturated liquid. This cycle is what drives the system without the use of a pumping system. These components and flow rates required to remove the proper heat from the IGBT were determined by the use of an analytical model.

To verify that our system met the goals of the project thermocouples were used to measure the temperature of the lid to validate that it did not exceed 80°C. Our system also measured other parameters such as fluid temperatures and pressure at multiple state points in the system. This data was displayed and stored with the use of a data acquisition system and a personal computer with a program DasyLab.

The estimated cost of our project was \$1773.21 and the final cost was \$1,662.08. If this system or one similar was put into production the cost would be reduced due to mass manufacturing discounts.

Our recommendations would be to remove the fan from the condenser and use a heat sink. A heat sink would allow for free convection to remove the 400W of heat. Machine the upper reservoir so that all fluid would flow through the condenser. Increase the size of the upper reservoir to decrease the amount of fluid returning in the lift tube. Use multiple tubes to increase the flow rate and allow more heat to be removed from the system.

## **Section I. Detailed Description of the Selected Conceptual Design**

## Mechanical Design

The final design uses methanol as the working fluid to transfer the heat from the IGBT and maintain the surface temperature below 80°C. The IGBT mounts directly to a copper lid which is brazed to the open channel brass cold plate. As the methanol begins to boil it forces a slug flow through the bubble pump tube. The bubble pump tube is a polycarbonate tube which connects the cold plate to the liquid vapor separator/upper reservoir. As the liquid boils the methanol vapor drives the fluid into the upper reservoir. This action is what drives the system in lieu of a pump. The upper reservoir is made from brass and has a copper lid. Once the liquid is in the upper reservoir it is transferred through the evaporator. While in the evaporator the two phase fluid is condensed and exits into an accumulator. The accumulator allows for any expansion in the fluid to occur without allowing the system pressure to increase above the desired value. The methanol is then transferred back into the cold plate. The multiple components are connected by the use of flexible tube. To run the system there is some electrical requirements. There is a standard electrical box in which a rheostat is used to adjust the fan for the condenser and the wattage for the IGBT input. The initial design of the system can be seen in Figure 1.

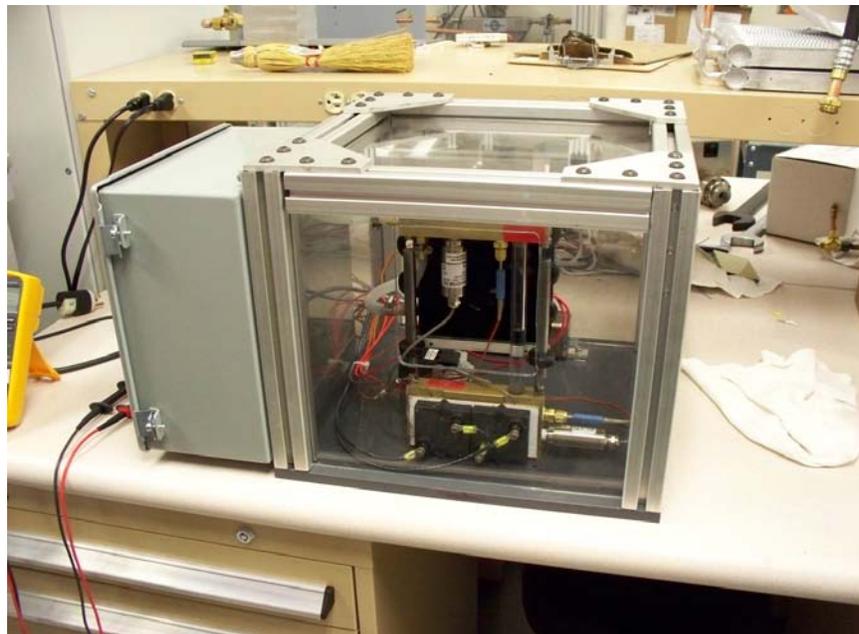


Figure 1: Initial design of bubble pump cooling system

## **Section II. Building Process**

## Mechanical Fabrication

The mechanical design as seen in *Figure 1* of the bubble pump cooling system consists of a cold plate evaporator, a bubble pump lift tube, an upper reservoir, a tube-fin condenser with an axial fan, a lower liquid reservoir and an enclosure. These items were machined, fabricated, and assembled to our required design specifications.

### Cold Plate:

The cold plate cavity as seen in *Figure 2* was machined from a brass alloy C360 bar stock. Brass was chosen for the cavity because of its thermal conductivity and because it can easily braze to copper. The brass cavity contains two tapped holes on the bottom surface for mounting to the base plate of the enclosure. There are two tapped holes on the top surface of the cavity in which standoffs for the upper reservoir are fastened. The stand-offs locate and secure the upper reservoir. Two 1/4" National Pipe Thread (NPT) holes were machined into the side surface of the cavity for a pressure transducer and a thermocouple. Another 1/4" National Pipe Thread (NPT) hole was added to the back surface of the cavity for a hose barb fitting for the fluid inlet. A 5/8" hole is machined into the top surface of the cavity for the mounting of the bubble pump lift tube.



Figure 2: Cold Plate Cavity

The cold plate lid in *Figure 3* was machined from copper alloy C110 bar stock and was furnace brazed to the front surface of the cavity using SILPHOS15 braze alloy. The lid separates the fluid from the resistive heating elements that mount to the cold plate. Copper was selected for the lid because it has a high thermal conductivity which allows the heat to be transferred from the

heaters to the fluid with little resistance. The braze alloy was purchased as a 0.004” thick sheet of foil and was cut to size. The braze foil was sandwiched between the cavity and lid and then sent down a furnace where it was heated and brazed. After the lid was brazed to the cavity two small holes were machined into the top surface of the lid for thermocouples and four threaded holes were added to the front face of the cold plate to mount the heaters.



Figure 3: Cold Plate Lid

The pressure transducer in *Figure 4* was installed into the cold plate followed by the invasive thermocouple which required the use of a 1/4 NPT to 1/8 A-lock adapter and then the hose barb fitting was installed. Teflon tape and loctite thread seal were used on all NPT joints to prevent leaks. A thermal interface material, thermal grease, was applied to the front copper surface of the cold plate to decrease thermal contact resistance between the cold plate and the heaters. The heaters were then mounted to the cold plate and two thermocouples were inserted into the small machined holes to measure surface temperatures.



Figure 4: Side View Assembly

**Bubble Pump Lift Tube:**

The bubble pump lift tube in *Figure 5* was fabricated from a polycarbonate, high strength Lexan, tubing. The standard size tubing was a purchased item with an outside diameter (OD) of 5/8" and an internal diameter (ID) of 3/8" and overall height of 8". The overall outside diameter was slightly reduce near the end of the lift tube were the o-ring grooves are located. The diameter reduction was done to ensure correct fit between the lift tube, cold plate, and upper reservoir. The lift tube connects the cold plate to the upper reservoir by slip fitting into counter bores in the cold plate and upper reservoir. The o-rings form a radial seal between the lift tube and the counter bores. The lift tube is then secured between the cold plate and upper reservoir by the stand-offs. Neoprene o-rings were lubricated using Dow Corning grease and installed onto the machined o-ring grooves. Neoprene and Dow Corning grease were chosen due to their compatibility with a large number of chemicals including methanol.



Figure 5: Bubble Pump Lift Tube

**Upper Reservoir/Liquid Vapor Separator:**

The upper reservoir in *Figure 6* consisted of a machined brass alloy C360 cavity and a machined copper alloy C110 lid. Like the cold plate the upper reservoir cavity and lid were furnace brazed together using a SILPHOS15 braze alloy. 1/4" National Pipe Thread (NPT) holes were machined into the bottom surface of the cavity for a pressure transducer, thermocouple, and a hose barb fitting for the fluid outlet. Another 1/4" NPT hole was machined on the side surface of the cavity so that the reservoir can be used as a liquid vapor separator. A 5/8" hole was machined into the bottom surface of the cavity for the mounting of the bubble pump lift tube. There are two threaded holes tapped on the bottom surface of the cavity for fastening the standoffs. The pressure transducer was installed into the cold plate followed by the invasive thermocouple which required

the use of a 1/4 NPT to 1/8 A-lok adapter and then the hose barb fittings were installed. Teflon tape and Loctite thread seal were used on all NPT joints to prevent leaks.



Figure 6: Liquid vapor Separator

### **Tube-Fin Condenser with Axial Fan:**

The tube fin condenser in *Figure 7* was purchased along with an axial fan. The fan mounted directly to the condenser and the condenser was installed behind the upper reservoir by fastening circular clamps around the stand-offs which were attached to the condenser. Clear silicon rubber tubing was used to connect the outlet of the upper reservoir to the inlet of the condenser and the outlet of the condenser to the lower liquid reservoir. The silicon rubber tubing was chosen due to its chemical compatibility, temperature rating, pressure rating and soft durometer which allow it to bend without kinking. Hose clamps were secured around the hose barb connections to further prevent leaks.



Figure 7: Tube Fin Condenser

**Lower Liquid Reservoir:**

The lower liquid reservoir in *Figure 8* was fabricated from stainless steel alloy SS304 and is used to accumulate liquid from the condenser and supply liquid to the cold plate without causing a drastic change in the height of the liquid. The reservoir was revised from the original design to reduce the size and to allow ease of fabrication. The reservoir is a welded assembly and is mounted directly to the base plate of the enclosure. 1/4" NPT holes were machined into the welded condenser assembly for an invasive thermocouple and hose barb fittings.



Figure 8: Reservoir

**Enclosure:**

The enclosure in *Figure 9* is a combination of extruded aluminum, and clear polycarbonate sheet. The top and side plates are made from clear polycarbonate sheet and are mounted to between uprights which are made from extruded aluminum. The base has clearance holes for the mounting of the cold plate evaporator and lower liquid reservoir. There are also holes put in the polycarbonate sheet which will allow for airflow across the condenser in line with the axial fan.



Figure 9: Bubble Pump Enclosure

## Electrical Design

The electrical design in *Figure 10* of this system consists of type t thermocouples to measure the temperature at set locations. The pressure is also measured in set locations throughout the system by the use of pressure transducers. A paddle wheel flow meter is used to measure the liquid volume flow rate into the cold plate. The outputs from all of the sensors are hooked into a P-DAQ-65 unit and are evaluated and displayed with the use of Dasy Lab software. Dasy Lab gives real time information that is used to watch the systems state variables. The power that is being put into the systems is calculated with the use of two multi-meters which track the voltage and current supplied to the system. The power to the heaters is controlled with the use of a rheostat. The axial fan speed is also controlled by a rheostat.

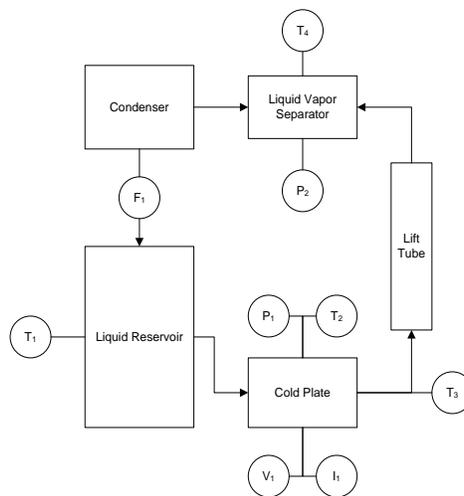


Figure 10-Measured Quantities of Bubble Pump Cooling System

## Mechanical Design Revised

The bubble pump tube which was cut to 8" was not sufficient to remove 400W. A new tube was cut to a length of 24" and 4 o-ring grooves were machined into the tube, two on each end. The overall height of the tube was increased from the original height of 8". This additional height was needed to correspond with a higher flow rate which was required to dissipate the 400W. The height increase was realized by the model that a taller tube will increase the head and the flow rate through the system. It was also determined through experimentation that the lower liquid reservoir created too large of restrictions of flow so it was removed and replaced by an accumulator. The accumulator in *Figure 11* was added to the system in place of the reservoir to create a driving head for the system. The accumulator also allowed for the liquid and vapor to

expand reducing the amount of pressure in the system. This reduction in pressure allowed for the system to flow properly and remove the required 400W of heat. The accumulator was placed in the system behind the liquid separator. There was a 3/8" NPT fittings added to the supply and return of the accumulator. There were 3/8" male hose barbs threaded to the inlet and outlets of the accumulator in which the flexible hose for the flow were mounted. The hose were affixed using hose clamps to prevent leakage of Methanol from the system. The final design of the system can be seen in Figure 12.

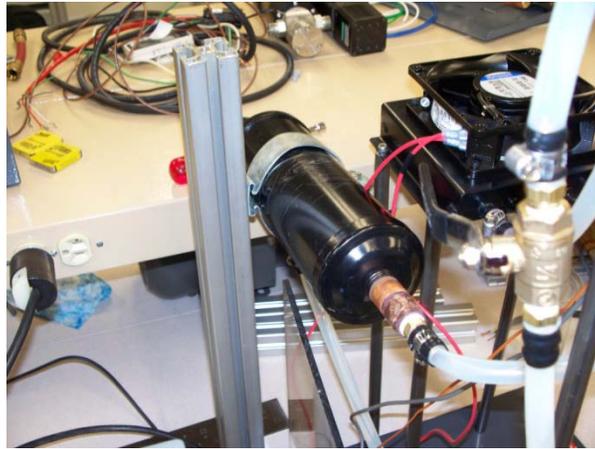


Figure 11: Liquid Accumulator

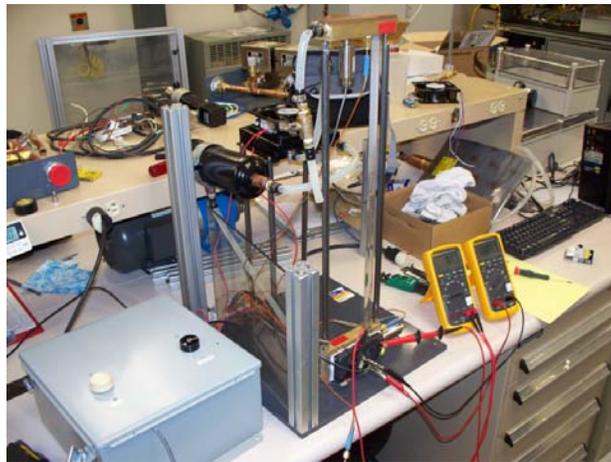


Figure 12: Final Design

## **Section III. Testing**

When a prototype is build it is essential that the new device be tested that the results of the test can be compared with the way the prototype was predicted to behave. The parameters that need to be measured must be set so that the correct data is obtained for analysis. In the bubble pump cooling system that parameters that justify whether it is working correctly are temperature, pressure, flow rate and heat applied. A detailed procedure of how that experiment was run must also be documented to ensure that no shortcuts were taken and that when the test is run more the once the results can be compared more accurately. Once all this is done several tests must be done and the results must be recorded. Once the tests are run the results and be analyzed and compared with the calculations made.

### Measured Parameters

In the building phase of the Senior Design Project there are several properties that must be measured to determine whether the bubble pump cooling system is operating according to the parameters that were given in the problem statement. One property that must be measured is the temperature at various locations which will be done by the use of a thermocouple. Pressure will also be measured at various locations by the use of a pressure transducer. Finally the flow rate will be measured with a flow meter. All these values will be measured and recorded to determine if the design meets the parameters that were given.

Figure shows the locations where the thermocouples, pressure transducers and the flow meter will be placed. Table 1 shows what the variables and subscripts refer to in the figure while Table 2 shows the quantities that must be calculated.

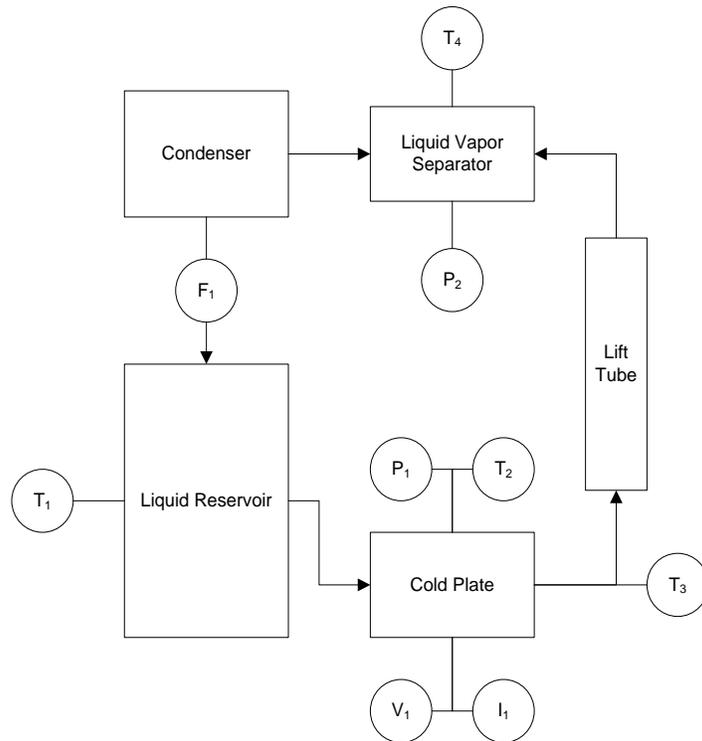


Figure 13-Measured Quantities of Bubble Pump Cooling System

**Table 1 -Variables and Subscripts of Measured Quantities**

Measure Parameters & Quantities

$T_1 =$	Temperature of fluid entering the cold plate
$T_2 =$	Cold plate surface temperature
$T_3 =$	Fluid temperature near the outlet of the cold plate
$T_4 =$	Temperature of fluid in the liquid/vapor separator
$P_1 =$	Fluid pressure inside of the cold plate
$P_2 =$	Fluid pressure inside of the liquid/vapor separator
$F_1 =$	Volumetric flow rate
$V_1 =$	Voltage across heating elements
$I_1 =$	Current supplied to heating elements

**Table 2- Calculated Quantities**

$Q =$	Rate of heat transferred to cold plate
$T_s =$	Average cold plate surface temperature
$R_{th} =$	Cold plate thermal resistance
$\Delta P =$	System pressure drop
$m_{dot} =$	Mass flow rate

### Calculated Quantities

The rate of heat transfer to the cold plate must be measured for testing purposes. During the testing the heat load will be varied to observe how the system behaves under different heat loads. The voltage and current across the heating elements will be measured and the heat load can then be calculated by the following equation.

$$Q = V_1 I_1$$

The pressure will be measured in the cold plate and also in the liquid vapor separator. These values will be measured so that a pressure drop can be found across the bubble pump. The pressure drop will be found by the following equation.

$$\Delta P = P_2 - P_1$$

Average cold plate temperature must be measured to ensure that the surface temperature of the heating element is kept below 80°C. The temperature will be measured at the cold plate surface and also at the outlet of the cold plate. With these two values the average surface temperature of the cold plate can then be calculated.

$$T_s = \frac{T_2 + T_3}{2}$$

The flow rate will be measured so that it can be compared with the analytical model created. The flow rate does not justify whether the system is working correctly but it must be measured in order to determine whether or not the analytical the model is correctly predicting the results that are obtained during the testing.

## Testing Procedure

Once the bubble pump cooling system was built the team began the testing phase of the project to determine if the design was capable of operating and meeting the specifications and parameters that were given and determined. In order to determine the capabilities of the design we placed thermocouples in four different locations. Two thermocouples, coated with thermal grease, were recessed into counter-bores that were machined in the copper lid near surface of the cold plate. These thermocouples measure the approximate surface temperatures of the cold plate in two locations where the heaters are mounted. Thermocouples were also placed at the inlet and outlet of the bubble pump lift tube to monitor the temperature of the fluid entering and leaving the lift tube. A pressure transducer was installed in the cold plate to monitor fluid pressure. Another pressure transducer was installed in the upper reservoir. The pressure transducers are used to monitor the system pressure and were used to determine the pressure drop across the lift tube. The pressure measurements were useful for ensuring we did not exceed a pressure of 10psig which is the maximum operating pressure of the silicon rubber hose used for plumbing the system. The team also needed to monitor the pressure because if it became too high, that would impede the flow of methanol or in some cases prevent it completely. The wattage applied to the cold plate would be known by the current and voltage that were measured. If the average cold plate surface temperature was under 80° C and the wattage applied is 400 Watts, then the team knows that the bubble cooling system is behaving according to the parameters.

A DasyLab program was set up to record the data from the P-Daq-65 unit. For the first tests the team recorded the data every minute during the testing onto to an Excel spreadsheet but later programmed DasyLab to write the data to a file. A testing procedure was then set up for running the bubble pump cooling system.

1. Make sure all the silicon rubber tubing is plumbed and hose clamps are secure as shown in Figure 14.

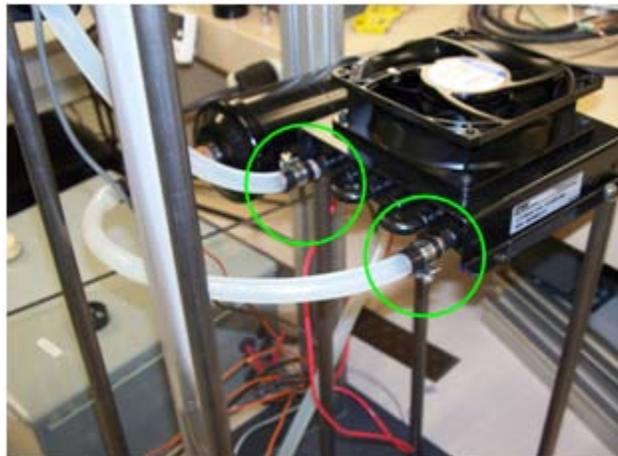
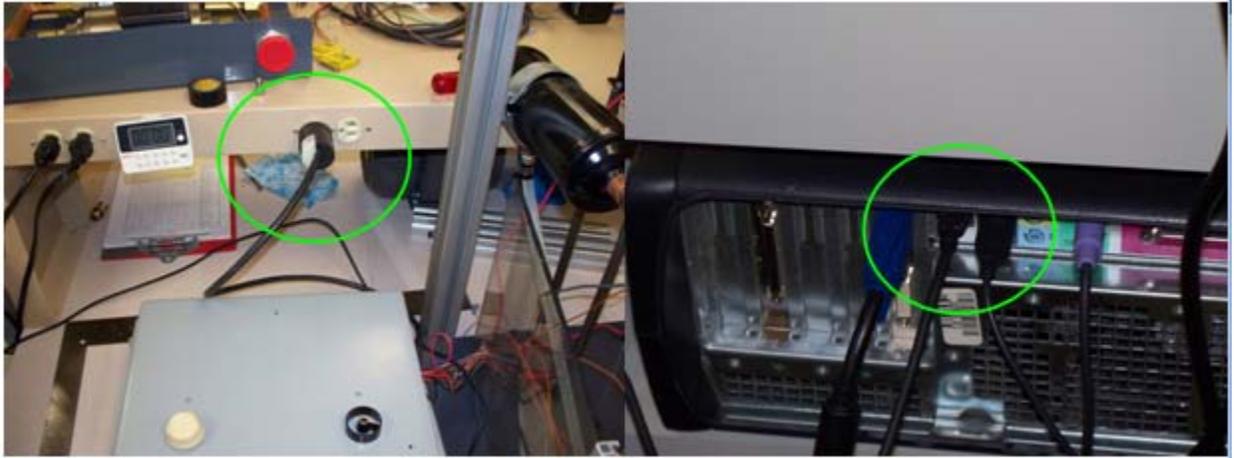


Figure 14-Silicon Rubber Hosing and Clamps

2. Plug in electrical enclosure into an 110 Volt outlet and USB cord to computer as shown in Figure 15.



**Figure 15-Electrical Plug and USB Cord**

3. Open bubblepump.dlb from desktop.

4. Double click on the Write00 icon and click the File button shown in Figure 16, enter in a new file name, click save, and then click ok.

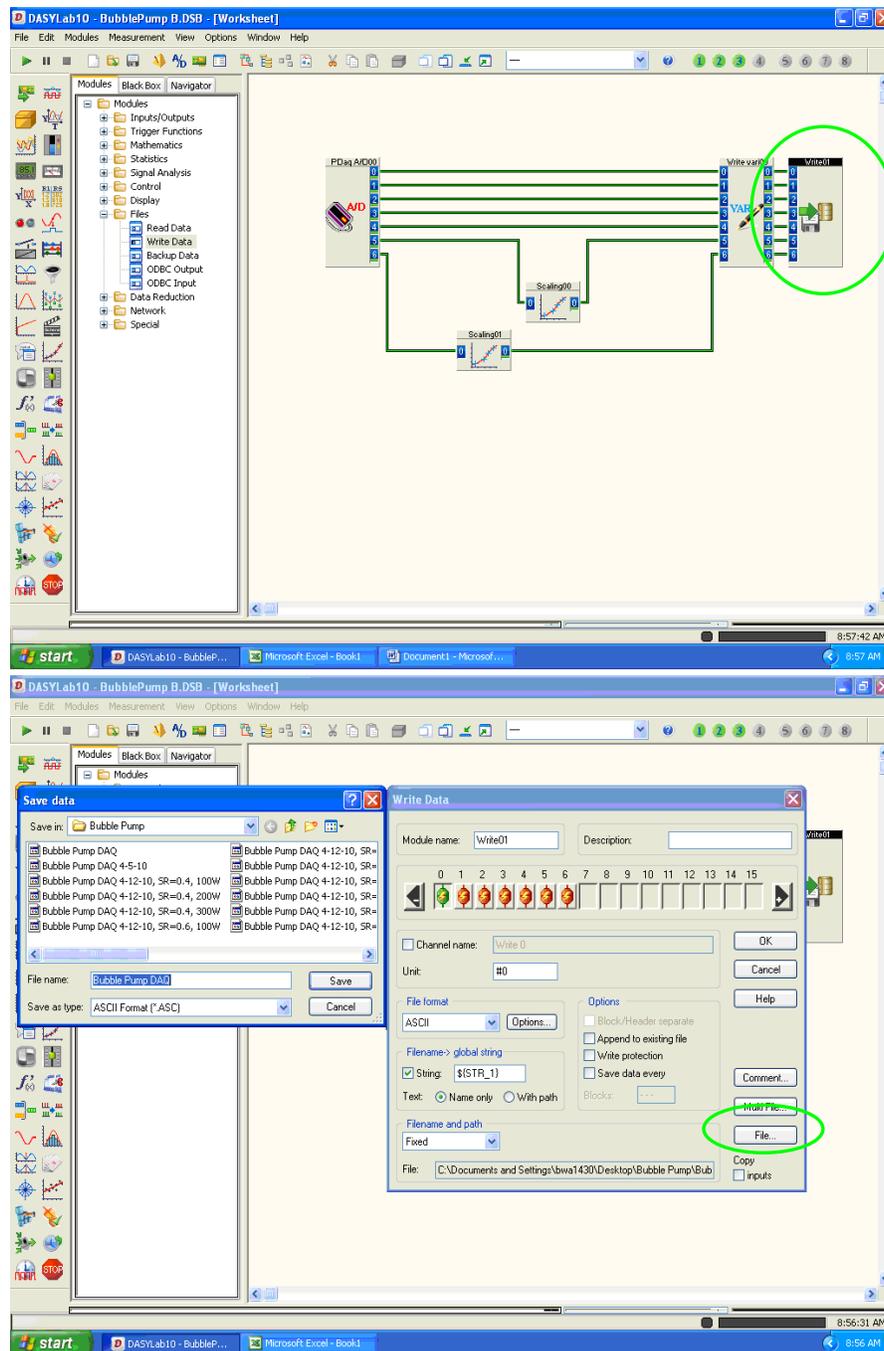
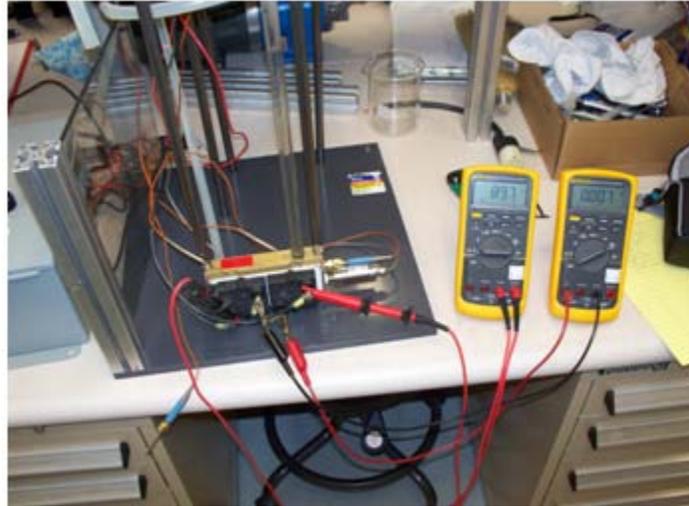


Figure 16-Writing Data to File

5. Click Window and select layout 2 and then click on the play icon to start recording data.

6. Clamp one multi-meter to the positive and negative leads on the heater as shown in Figure 5 and set it to measure the voltage.
7. Clamp the other multi-meter to the two remaining leads also shown in Figure 17 and set it to measure the current across them.



**Figure 17-Multi-Meter Connections**

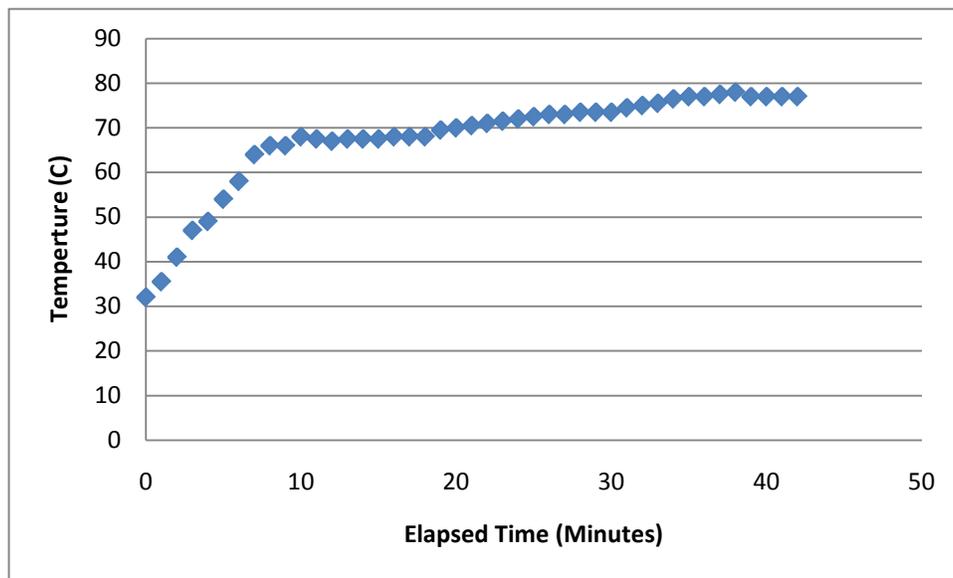
8. Turn on the fan via the labeled rheostat.
9. Turn on the heater via the other labeled rheostat and watch the multi-meter reading the voltage and turn rheostat until it reaches approximately 90 volts as shown in Figure 18.



**Figure 18-Multi-Meter Reading at Approximately 90 Volts**

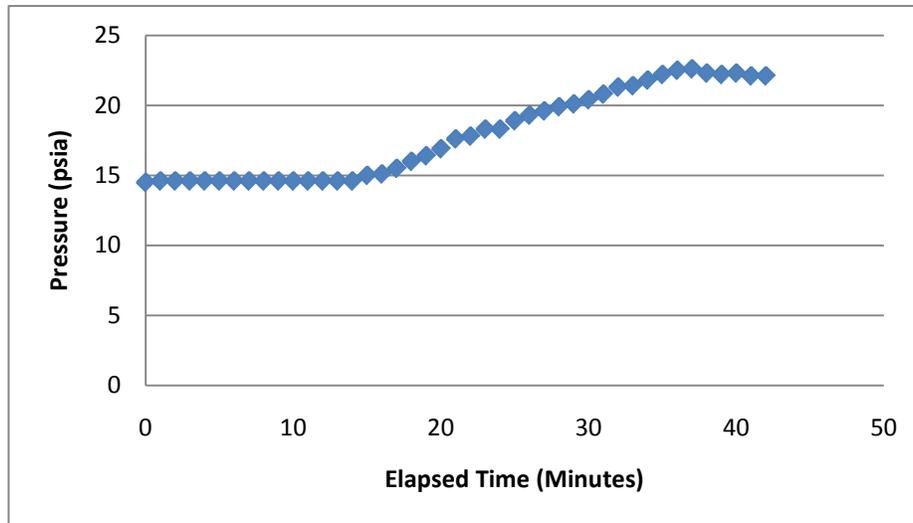
10. Record the temperatures and pressures at one minute intervals for reference.
11. Allow the unit to run as long as desired. It will take 10-15 minutes for the system to reach steady state. (If pressure exceeds 24 psia or the surface temperature exceeds 80°C, turn the heaters off immediately.)
12. Turn off heater and fan and click the stop button on the DasyLab program.
13. Unplug unit.

For the first test the team used their original design with the 8” bubble pump tube and ran the system with a heat input of 100 Watts. The bubble pump system was set up with a submergence ratio of 0.6 and the room air temperature was approximately 72<sup>0</sup> F. Data was recorded during the forty minute test duration. With these parameters the system reached steady state with an average cold plate surface temperature of 77<sup>0</sup> C. The results from the initial test are shown in Figure 19.



**Figure 19-Average Cold Plate Surface Temperature with 100 Watts Input**

As can be seen from the figure the cold plate surface temperature is close to 80<sup>0</sup> C at only 100 Watts input. The procedure was tested at 200 Watts but the cold plate surface temperature jumped above 80<sup>0</sup> C very quickly. One parameter the team noticed was that the pressure also increased as the temperature increased as can be seen from Figure 20.



**Figure 20-System Pressure**

As the pressure increases the temperature also increases and the flow is impeded. From the results of the first test the team decided that the system needed more volume to allow for the fluid to expand and keep the system at a lower pressure. The team removed the smaller reservoir and replaced it with a larger volume reservoir. The reservoir was mounted above the liquid level in the bubble pump so that the reservoir would not be flooded when the system was not running. The team also observed that there was not much flow through the condenser so to solve that problem the team had a 24" bubble pump tube machined and replaced the 8" tube with it. Having a longer bubble pump tube gave the fluid enough static head or momentum out of the upper reservoir to overcome the restrictions imposed by the condenser. The team also went back to the analytical model and from it higher flow rates were predicted with the use of a taller lift tube. The choice of 24" was made because an analysis was done to determine how much static head was needed to flow through the condenser and with the 24" inch tube there would be a sufficient amount of static head. The condenser was also mounted above the liquid level so that it too would not be flooded. Once these changes were made the system was ran again at heat loads of 100, 200, 300 and 400 Watts and the results can be seen in the figures. At 100 Watts the system was run with the fan off. The atmospheric conditions were the same as the first test.

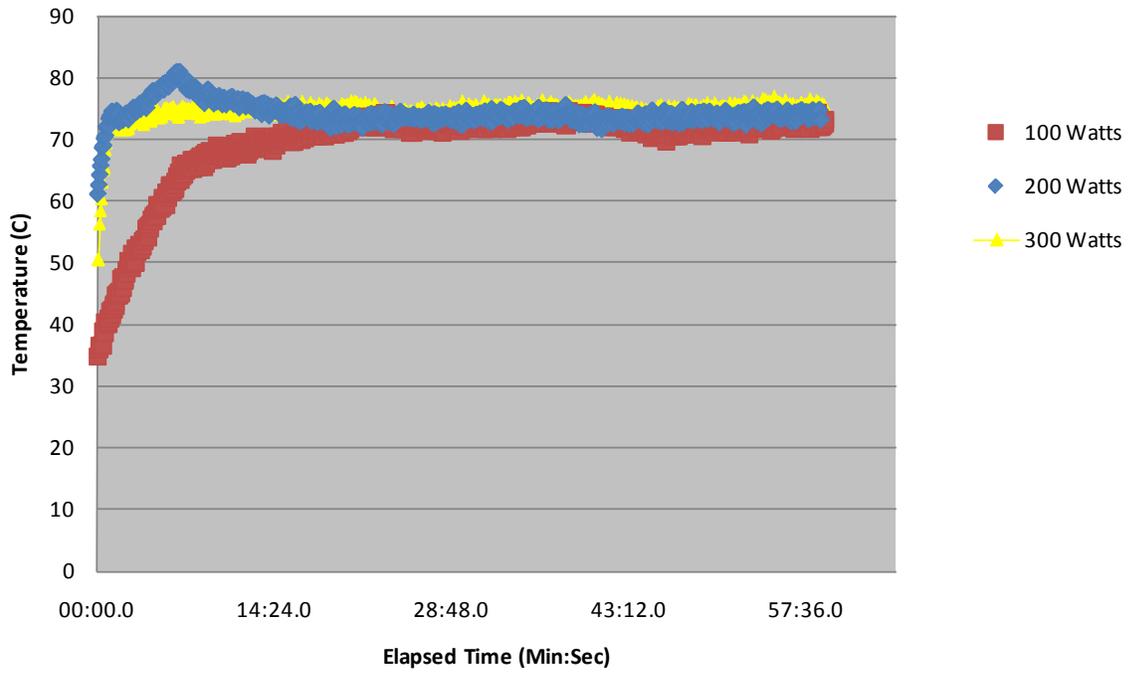


Figure 21-Average Cold Plate Surface Temperature

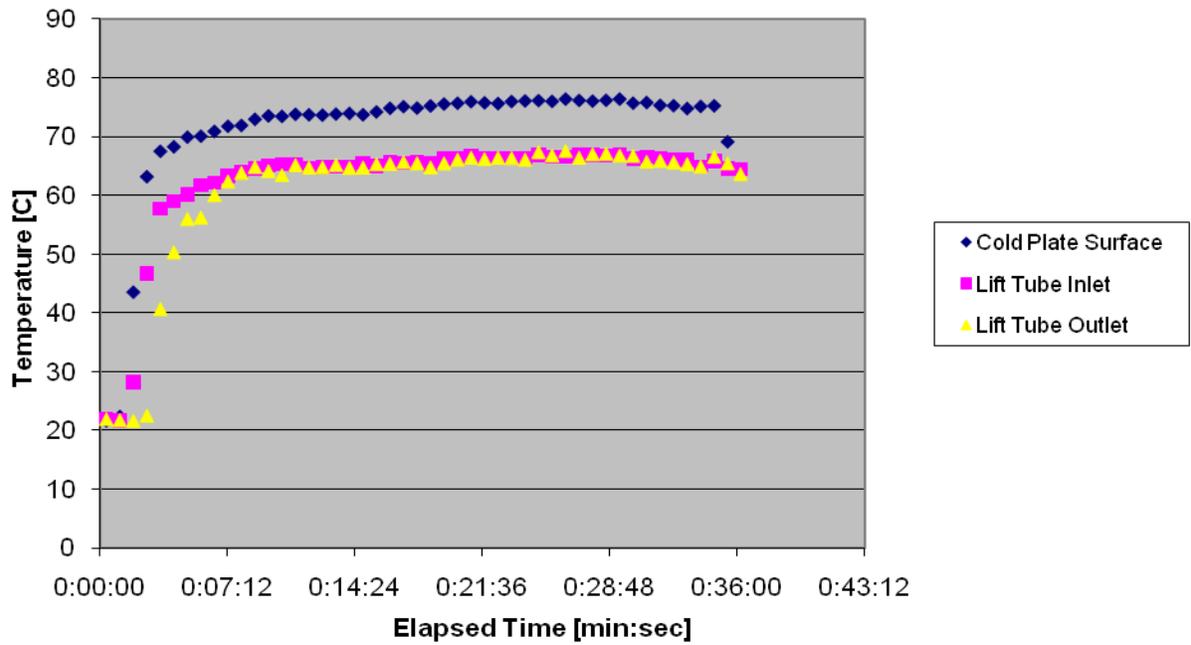


Figure 22=Cold Plate Surface Temperature at 400 Watts with Modified Design

As can be seen from Figure 21 and Figure 22 the cold plate surface temperature never exceeds 80° C. The spike in the 200 Watts test occurred because the team ran the test with no fan at startup. Once the cold plate surface temperature approached 80°C the fan was turned on and the temperature then dropped. The conclusion can be made that at 100 Watts the heat can be dissipated by free convection but at other heat loads forced convection must be used. Once the system reached steady state after about fifteen minutes the temperature remained close to 75°C at 400 Watts. The pressure was also much lower with the larger volume as can be seen from Figure 23. The pressure remained between 3 to 4 psig which is what the analytical model predicted.

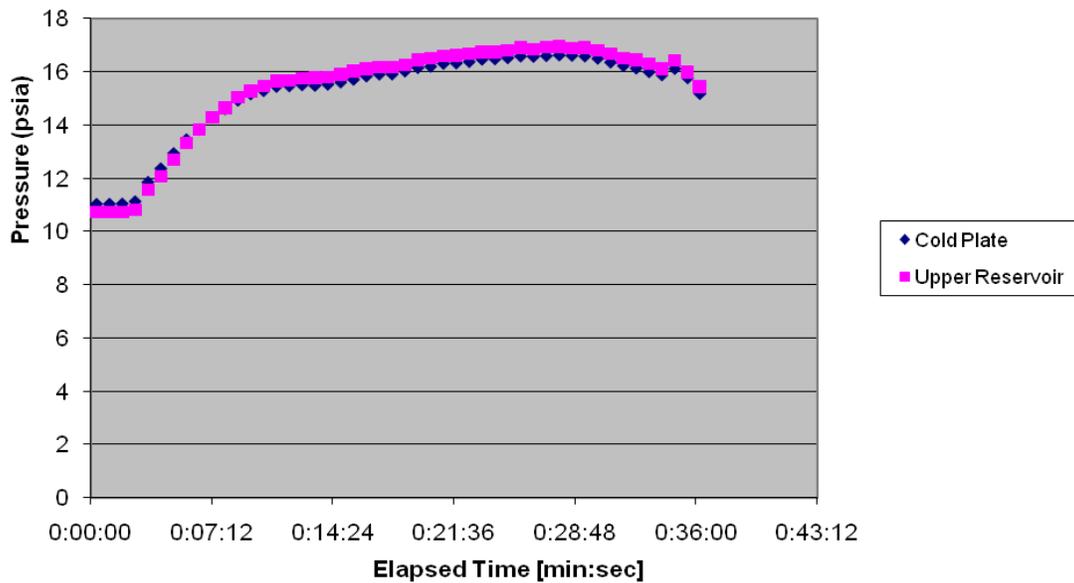


Figure 23=Absolute Pressure at 400 Watts with Modified Design

### Calculation/Analysis:

The cold plate utilizes a copper lid in which the heat load was applied. The heat was transferred through the lid and into the fluid. The surface area of the copper lid in contact with the fluid was designed to be proportional to the surface area in which the heat will be applied. The heat flux was then determined by dividing the rate of heat transfer by the heated surface area in contact with the fluid.

$$\begin{aligned}
Q &= 400W \\
L_0 &= 0.127m \\
h_0 &= 0.0508m \\
A_s &= L_0 h_0 = 0.006452m^2 \\
q_s &= \frac{Q}{A_s} = \bar{h} \theta_e = 62kW/m^2
\end{aligned}
\tag{1}$$

The excess temperature is defined as the temperature difference between the solid liquid interface and the saturation temperature of the fluid. The excess temperature for this analysis was estimated to be 10°C from data listed in various boiling curves where heat flux is plotted against excess temperature. Most fluids with an excess temperature of 10°C will be in the nucleated boiling regime which is the flow regime that slug flow occurs. From the temperature data listed in Figure 24 we can see that actual average excess temperature is around 10°C. With the heat flux determined with equation 1 and the excess temperature the heat transfer coefficient for the cold plate can be determined. In equation 2  $T_{sat}$  is assumed to be the fluid temperature near the inlet of the lift tube.

$$\begin{aligned}
T_{sat} &= 65^\circ C \\
\theta_e &= T_s - T_{sat} = 10^\circ C \\
T_s &= 75^\circ C
\end{aligned}
\tag{2}$$

$$\bar{h} = \frac{q_s}{\theta_e} = 6.2kW/m^2 \cdot ^\circ C
\tag{3}$$

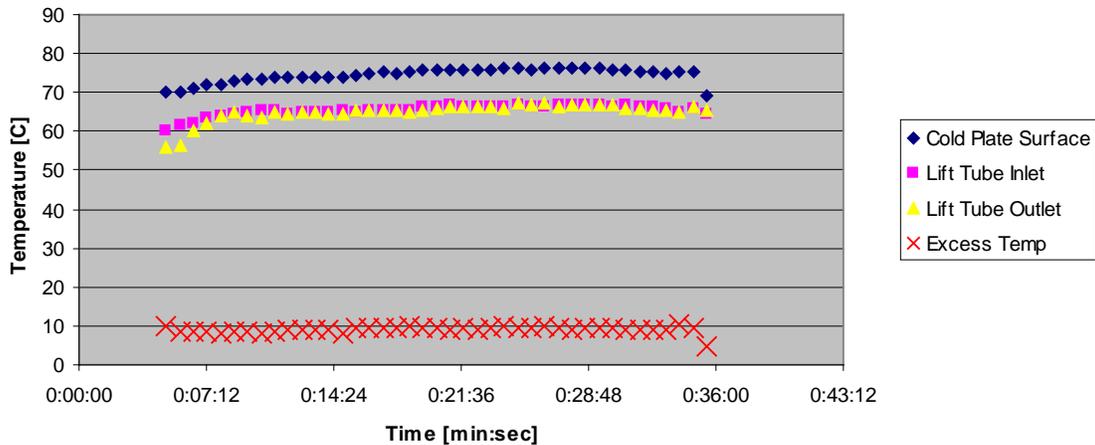


Figure 24: Revised design temperature measurements

The total thermal resistance estimated for the cold plate was determined by dividing the excess temperature by the rate of heat transfer.

$$R_t = \frac{\theta_e}{Q} = 0.023^\circ C/W \quad (4)$$

The mass flow rate at different heat loads determined from the analytical model is shown in Figure 25 along with the mass flow rate values measured at different heat loads. The mass flow rate was determined by measuring the time required to collect approximately 100mL of methanol pumped from the system. The mass of the liquid was then measured and the mass flow rate was determined by dividing the measured mass by the measured time. The analytical model predicted much higher flow rates than what was actually measured. There are possible many factors to account for the deviations but one of the main reasons we noticed was that some of the liquid being lifted in the tube fell back into the tube. This was most likely due to restrictions near the outlet of the lift tube in the upper reservoir. Due to system size constraints the upper reservoir was made small and liquid pulsing from the lift did not have enough room to expand and clear the outlet. Some of the pulsing liquid from the lift tube burst out and immediately collide with the lid of the upper reservoir and fell back into the lift tube decreasing the overall flow of the system.

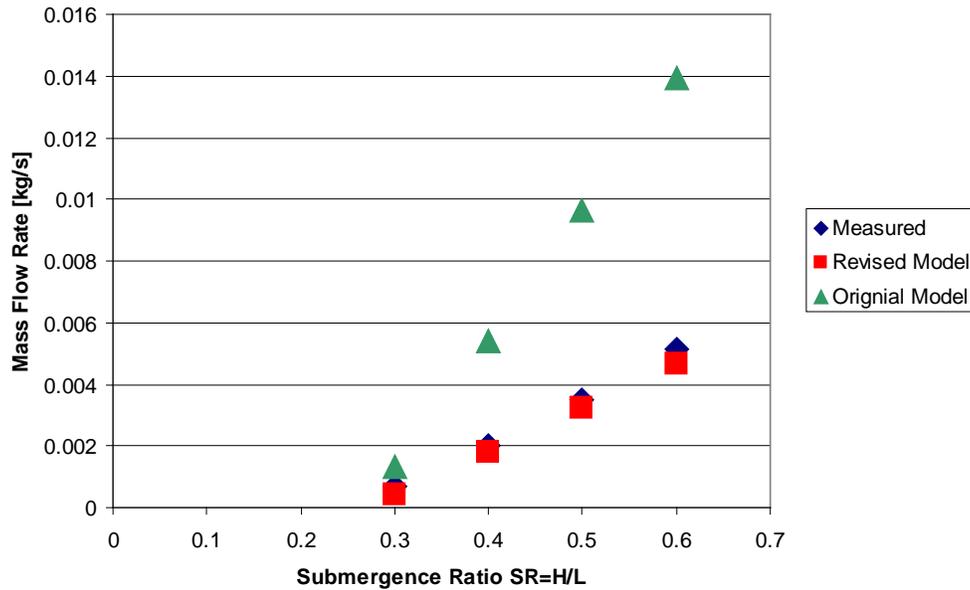


Figure 25: Flow Rate through the system

In the analytical model the friction coefficient  $K$  accounted for frictional losses in the lift tube, not the upper reservoir. The coefficient was determined from the friction factor which was determined by the Reynolds number. For the analysis we assumed liquid laminar flow for the Reynolds number, friction factor, and friction coefficient. These values were determined by the following equations.

$$Re = \frac{\rho_f V_1 D}{\mu_f} \quad (5)$$

$$f = \frac{64}{\text{Re}} \quad (6)$$

$$K = \frac{4fL}{D} \quad (7)$$

The analytical relationship for the lift tube used to determine the flow rate from last semester is listed in equation 8.

$$\frac{H}{L} - \frac{1}{\left(1 + \frac{\dot{V}_g}{s\dot{V}_f}\right)} = \frac{V_1^2}{2gL} \left[ (K+1) + (K+2) \frac{\dot{V}_g}{\dot{V}_f} \right] \quad (8)$$

The submergence ratio or the ratio of the height of liquid in the reservoir to the fixed length of the lift tube is defined by the following equation.

$$SR = \frac{H}{L} \quad (9)$$

The mass liquid flow rate was then determined by equation 10.

$$\dot{m}_f = \rho_f \dot{V}_f \quad (10)$$

From Figure 25 we can see the flow rate increases linearly with the heat load. The analytical results yield higher flow rates and change in flow rates compared to the measure results. To correct this correction factor was determined from the linear curve fit equations associated with the analytical and measured data sets. This correction factor was applied to equation 10 and was determined to be  $C=1/3$ .

$$\dot{m}_f = C\rho_f \dot{V}_f = \frac{\rho_f \dot{V}_f}{3} \quad (11)$$

Applying the correction factor shown in equation 11 the analytical results for the mass flow rate matched closely to the measure results.

## **Section IV. Evaluation and Recommendations**



## **Recommendations**

As there is with any design or prototype there are always aspects of the project that were later realized that could improve the design. The way the bubble pump cooling system is currently designed electricity is needed to run the fan. Without the fan the system would not run properly at the required heat load of 400 Watts. In order to eliminate the use of a fan a heat sink could be added to the system. With a heat sink the heat could be dissipated into the surroundings by free convection and not by forced convection.

Another good improvement that could be made is machined the upper reservoir differently. With the current design of the upper reservoir liquid is allowed to bounce off the lid and fall back into the lift tube. This aspect is can be observed by watching the system run. If the upper reservoir was machined so that a limited amount of fluid could fall back into the lift tube then system would have a greater flow rate and more heat could be removed.

If the problem statement required a greater heat load to be applied multiple tubes may need to be used. If multiple tubes were used then the system would have a greater flow rate and more heat could be applied and removed. The method of dissipating the heat would then have to be reevaluated because the current condenser may not be large enough to remove the higher heat loads.

## **Conclusion**

In conclusion, we felt that the final design is an acceptable demonstration unit for Parker Hannifin. Our final design was able to use the theories of two phase flow in a pipe and conduction to solve the problem set in front of us by our corporate sponsor. Our design was centered on the use of a bubble pump tube to drive the system and remove the 400W of heat without allowing the surface temperature of the IGBT to exceed the temperature of 80°C. Through experimentation and the use of the mathematical model it was determined that the original design would not be able to accomplish our goals. With the math model revised it was then possible to calculate the changes that would be required to achieve the parameters set forth at the beginning of this project. The math model determined that the height of the lift tube would need to be increased to 24" to achieve the proper flow rate to remove the required rate of heat while achieving the proper surface temperature. Through cost analysis it was found that the final cost of the project was \$112 cheaper than the estimated cost. The cost to make this system in a manufacturing setting would be reduced greatly through optimization of manufacturing practices. The final design met all the requirements set forth by Parker and will be a good start for their growing Precision Cooling Business Unit.

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## **Appendices**

## Copy of EES Program

"Bubble Pump Analysis"  
"ME488 Senior Design"

"Fluid Properties:"

F\$='Methanol'  
P\_sys=101.3 [kPa]  
T\_sat=T\_SAT(Methanol,P=P\_sys)  
rho\_f=DENSITY(F\$,P=P\_sys,x=0)  
rho\_g=DENSITY(F\$,P=P\_sys,x=1)  
v\_f=VOLUME(F\$,P=P\_sys,x=0)  
v\_g=VOLUME(F\$,P=P\_sys,x=1)  
h\_f=ENTHALPY(F\$,T=T\_sat,x=0)  
h\_g=ENTHALPY(F\$,T=T\_sat,x=1)  
mu\_f=VISCOSITY(F\$,T=T\_sat,x=0)  
mu\_g=0.000010925 [kg/m-s]  
sigma=SURFACETENSION(F\$,T=T\_sat)

"Working fluid"  
"Saturation pressure of fluid in system [kPa]"  
"Saturation temperature of fluid in system [C]"  
"Density of liquid [kg/m^3]"  
"Density of vapor [kg/m^3]"  
"Specific volume of liquid [m^3/kg]"  
"Specific volume of vapor [m^3/kg]"  
"Enthalpy of liquid [kJ/kg]"  
"Enthalpy of vapor [kJ/kg]"  
"Dynamic viscosity of liquid [kg/m-s]"  
"Dynamic viscosity of vapor [kg/m-s]"  
"Surface tension of fluid [N/m]"

"The maximum lift tube diameter in which slug flow occurs (Chisholm, 1983)"

$D_{max}=19*((\sigma*v_f)/(g*(1-v_f/v_g)))^{1/2}$

"Max lift tube diameter [m]"

"Geometry:"

SR=.6  
L=.2032 [m]  
H=SR\*L  
L\_0=0.127 [m]  
h\_0=0.0508 [m]  
w\_0=0.020 [m]  
A\_0=w\_0\*h\_0  
p=2\*(w\_0+h\_0)  
D\_0=4\*A\_0/p  
A\_s=L\_0\*h\_0  
D=0.009525 [m]  
A=pi/4\*D^2

"Submergence ratio"  
"Length of lift tube [m]"  
"Height of liquid in reservoir [m]"  
"Length of cold plate cavity [m]"  
"Height of cold plate cavity [m]"  
"Depth of cold plate cavity [m]"  
"Cross-sectional area of cold plate cavity [m^2]"  
"Perimeter of cold plate cavity [m]"  
"Hydraulic diameter of cold plate cavity [m]"  
"Heated solid/fluid interface surface area of cold plate [m^2]"  
"Diameter of lift tube [m]"  
"Cross-sectional area of lift tube [m^2]"

"Constants:"

g=9.81 [m/s^2]

"Acceleration due to gravity [m/s^2]"

"Known Parameters:"

Q=0.400 [kW]  
T\_s\_mas=80.0 [C]

"Rate of heat transferred to fluid [kW]"  
"Maximum cold plate surface temperature [C]"

"Cold Plate:"

T\_e=10.0 [C]  
T\_s=T\_sat+T\_e  
q\_s=Q/A\_s  
q\_s\_max=0.131\*h\_fg\*rho\_g\*((sigma\*g\*(rho\_f-rho\_g))/rho\_g^2)^{1/4}  
q\_s\_min=0.09\*rho\_g\*h\_fg\*((sigma\*(rho\_f-rho\_g))/(rho\_f+rho\_g)^2)^{1/4}  
h\_fg=h\_g-h\_f  
h\_s=q\_s/T\_e  
m\_dot\_g=Q/h\_fg  
V\_dot\_g=m\_dot\_g/rho\_g

"Estimated excess temperature of solid/fluid interface [C]"  
"Surface temperature of solid/fluid interface [C]"  
"Heat flux [kW/m^2]"  
"Latent heat of vaporization [kJ/kg]"  
"Heat transfer coefficient [kW/m^2-C]"  
"Mass flow rate of vapor [kg/s]"  
"Volume flow rate of vapor [m^3/s]"

"Bubble Pump: Governing Equations"

$H/L-1/(1+V_{dot_g}/(V_{dot_f*s}))=V_1^2/(2*L*g)+V_1*V_{dot_g}/(L*g*A)+K*V_1^2/(2*g*L)*(1+V_{dot_g}/V_{dot_f})$   
s=2.5

V\_1=V\_dot\_f/A  
m\_dot\_f=V\_dot\_f\*rho\_f  
K=(4\*f\*L)/D  
f=64/Re  
Re=(rho\_f\*V\_1\*D)/mu\_f  
P\_1=P\_sys+rho\_f\*g\*H-rho\_f\*V\_1^2/2  
x=m\_dot\_g/(m\_dot\_g+m\_dot\_f)  
v\_fg=v\_f+x\*(v\_g-v\_f)  
V\_2=v\_fg\*V\_1/v\_f  
P\_2=P\_1-rho\_f\*V\_1\*(V\_2-V\_1)  
W\_pump=(V\_dot\_f+V\_dot\_g)\*(P\_2-P\_sys)  
n\_eff=W\_pump/Q