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Advance Thermal Design

High Precision Picomotor Cooling Design

by

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1. Abstract

The goal of this project is to design an efficient cooling mechanism (heat transfer) for three picomotors which will be used on a miniature precision lathe that will be housed in a scanning electron microscope chamber. The reason efficient heat dissipation of the motors is important in the lathe is to avoid thermal expansion of any component of the lathe that may alter the resolution of the equipment. This equipment will be designed to have a resolution of 1 nm and in this nano-scale, thermal expansion can be a major issue. Each picomotor has a thermal dissipation of up to 32W. The proposed design to dissipate the heat out of the microscope chamber is to use a combined idea of heat pipes and heat sink. There will be heat pipes that will dissipate the heat out of the chamber and a common heat sink that will dissipate the heat from all heat pipes (from all 3 motors) through natural convection.

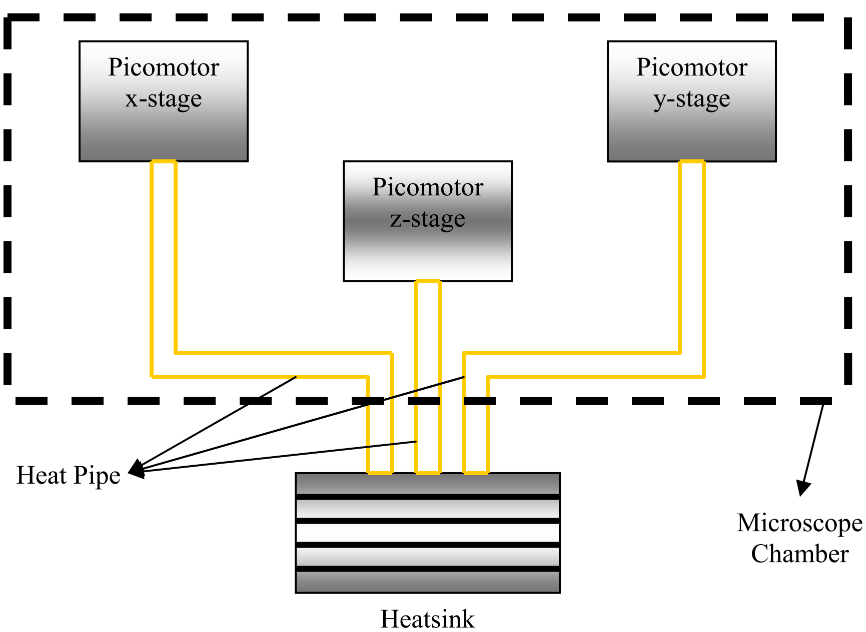

Figure 1: Schematic of the proposed thermal dissipation design for 3 picomotors on a precision lathe

2. Introduction

Precision engineering is currently one of industry's most promising fields. This is because almost any product used in today's world (i.e. computers, electronics, machinery, automotive etc) involve precision engineering. The technology window for precision engineering/machining continues to expand as human requirements/demands increase. The term precision engineering can be defined differently to each individual. In this project, we will define the term precision in the nanometer level.

In order to achieve nanometer level accuracy, it is important to make sure that the equipment being used has an accuracy about 10x more than that. It is also important to make sure that there are no external parameters that are playing a role in effecting this accuracy. External parameters could include noise, vibration, thermal expansion, heat generation, humidity and other environmental challenges. One of the most challenging problems faced in precision machines today is thermal expansion due to heat generation by the electronics/motors on the precision equipment.

Heat generation from the motors can result in thermal expansion of several parts of the precision lathe such as the spindle, tool holder, workpiece holder, cutting tool and many more. It only takes one of the above mentioned parts to expand in order to loose the machines actual resolution. For example, a change in the tool holder size due to thermal expansion of 2nm can result in a machining error of close to 200nm. This shows that the actual machining accuracy can be off by almost 100 times due to the thermal expansion of a single component on a precision lathe.

In order to solve this problem, some advanced thermal design techniques will have to be used. A combination of heat pipes and a heat sink will be design to dissipate heat generated by three picomotors (used to drive the x-y, and z-axis stage) on a precision lathe.

2.1 Heat Sinks

A heat sink is an environment or object that absorbs and dissipates heat from another object using thermal contact (either direct or radiant). Heat sinks are used in a wide range of applications wherever efficient heat dissipation is required; major examples include refrigeration, heat engines, cooling electronic devices and lasers. There are several essential characteristics that make a heat sink an efficient tool. These characteristics include factors such as:

- Heat sink surface - It's at the surface of the heat sink where the thermal transfer takes place. Therefore, heat sinks should be designed to have a large surface; this goal can be reached by using a large amount of fine fins, or by increasing the size of the heat sink itself.
- Aerodynamics - Heat sinks must be designed in a way that air can easily and quickly float through the cooler, and reach all cooling fins. Especially heat sinks having a very large amount of fine fins, with small distances between the fins may not allow good air flow. A compromise between high surface (many fins with small gaps between them) and good aerodynamics must be found. This also depends on the fan the heat sink is used with: A powerful fan can force air even through a heat sink with lots of fine fins with only small gaps for air flow - whereas on a passive heat sink, there should be fewer cooling fins with more

space between them. Therefore, simply adding a fan to a large heat sink designed for fanless usage doesn't necessarily result in a good cooler.

- Thermal design within heat sink - Large cooling fins are pointless if the heat can't reach them, so the heat sink must be designed to allow good thermal transfer from the heat source to the fins. Thicker fins have better thermal conductivity; so again, a compromise between high surface (many thin fins) and good thermal transfer (thicker fins) must be found. Of course, the material used has a major influence on thermal transfer within the heat sink. Sometimes, heat pipes are used to lead the heat from the heat source to the parts of the fins that are further away from the heat source.
- Flatness of contact area - The part of the heat sink that is in contact with the heat source must be perfectly flat. A flat contact area allows you to use a thinner layer of thermal compound, which will reduce the thermal resistance between heat sink and heat source.
- Mounting method - For good thermal transfer, the pressure between heat sink and heat source must be high. Heat sink clips must be designed to provide a strong pressure, while still being reasonably easy to install. Heat sink mountings with screws/springs are often better than regular clips. Thermoconductive glue or sticky tape should only be used in situations where mounting with clips or screws isn't possible.

Heat sinks function by efficiently transferring thermal energy ("heat") from an object at a relatively high temperature to a second object at a lower temperature with a much greater heat capacity. This rapid transfer of thermal energy quickly brings the first object into thermal equilibrium with the second, lowering the temperature of the first object, fulfilling the heat sink's role as a cooling device. Efficient function of a heat sink relies on rapid transfer of thermal energy from the first object to the heat sink, and the heat sink to the second object.

The most common design of a heat sink is a metal device with many fins. The high thermal conductivity of the metal combined with its large surface area due to the fins result in the rapid transfer of thermal energy to the surrounding, cooler, air. This cools the heat sink and whatever it is in direct thermal contact with. Use of fluids (for example coolants in refrigeration) and thermal interface material (in cooling electronic devices) ensures good transfer of thermal energy to the heat sink. Similarly a fan may improve the transfer of thermal energy from the heat sink to the air by moving cooler air between the fins.

Heat sink performance (including free convection, forced convection, liquid cooled, and any combination thereof) is a function of material, geometry, and overall surface heat transfer coefficient. Generally, forced convection heat sink thermal performance is improved by increasing the thermal conductivity of the heat sink materials, increasing the surface area (usually by adding extended surfaces, such as fins or foam metal) and by increasing the overall area heat transfer coefficient (usually by increase fluid velocity, such as adding fans, pumps, etc.).

2.2 Heat Pipes

A heat pipe is a device has an extremely high thermal conductivity, and is used to transport heat. In order to achieve this, heat pipes take advantage of simple physical effects: As a liquid evaporates, energy - in the form of heat - must be taken from the environment. Therefore, an evaporating liquid will cool the surrounding area. This is how a heat pipe effectively cools the heat source. However, this doesn't get rid of the heat; heat is just transported with the vapor. At the target side for heat transport, the heat pipe must be cooled, for example using a heatsink. Here, the inverse effect takes place: The liquid condenses, and therefore emits heat.

The heat pipe is, as the name suggests, a device of high thermal conductance; that is, it will transport thermal energy without an appreciable drop in temperature. The heat pipe may be regarded as a development of the thermal siphon. The thermal siphon consists of an evacuated sealed tube containing a small quantity of liquid. If this liquid at the bottom of the tube is heated, it will evaporate and the resulting vapor will flow to the top of the tube, where it will condense, returning to the evaporator region by gravitational forces. The net effect is for thermal energy to be transported from the evaporator to the condenser end of the tube. Since the energy exchange is by latent heat, the temperature difference between the two ends is small. Thus the effective thermal conductance of the tube is high.

Heat pipes employ evaporative cooling to transfer thermal energy from one point to another by the evaporation and condensation of a working fluid or coolant. Heat pipes rely on a temperature difference between the ends of the pipe, and cannot lower temperatures at either end beyond the ambient temperature (hence they tend to equalize the temperature within the pipe).When one end of the heat pipe is heated the working fluid inside the pipe at that end evaporates and increases the vapor pressure inside the cavity of the heat pipe. The latent heat of evaporation absorbed by the vaporization of the working fluid reduces the temperature at the hot end of the pipe.

The vapor pressure over the hot liquid working fluid at the hot end of the pipe is higher than the equilibrium vapor pressure over condensing working fluid at the cooler end of the pipe, and this pressure difference drives a rapid mass transfer to the condensing end where the excess vapor condenses, releases its latent heat, and warms the cool end of the pipe. Non-condensing gases (caused by contamination for instance) in the vapor impede the gas flow and reduce the effectiveness of the heat pipe, particularly at low temperatures, where vapor pressures are low. The velocity of molecules in a gas is approximately the speed of sound and in the absence of non condensing gases, this is the upper velocity with which they could travel in the heat pipe. In practice, the speed of the vapor through the heat pipe is dependent on the rate of condensation at the cold end.

The condensed working fluid then flows back to the hot end of the pipe. In the case of vertically-oriented heat pipes the fluid may be moved by the force of gravity. In the case of heat pipes containing wicks, the fluid is returned by capillary action. When making heat pipes, there is no need to create a vacuum in the pipe. One simply boils the working fluid in the heat pipe until the resulting vapor has purged the non condensing gases from the pipe and then seals the end. An interesting property of heat pipes is the temperature over which they are effective. Initially, it might be suspected that a water charged heat pipe would only work when the hot end reached the boiling point (100 °C) and steam was transferred to the cold end. However, the boiling point of water is dependent on absolute pressure inside the pipe. In an evacuated pipe, water will boil just slightly above its melting point (0 °C). The heat pipe will operate, therefore, when the hot end is just slightly warmer than the melting point of the working fluid. Similarly, a heat pipe with water as a working fluid can work well above the boiling point (100 °C), if the cold end is low enough in temperature to condense the fluid.

The main reason for the effectiveness of heat pipes is the evaporation and condensation of the working fluid. The heat of vaporization greatly exceeds the sensible heat capacity. Using water as an example, the energy needed to evaporate one gram of water is equivalent to the amount of energy needed to raise the temperature of that same gram of water by 540 °C (hypothetically, if the water was under extremely high pressure so it didn't vaporize or freeze over this temperature range). Almost all of that energy is rapidly transferred to the "cold" end when the fluid condenses there, making a very effective heat transfer system with no moving parts. The theory of the heat pipe and the physics behind is shown in Figure 2.

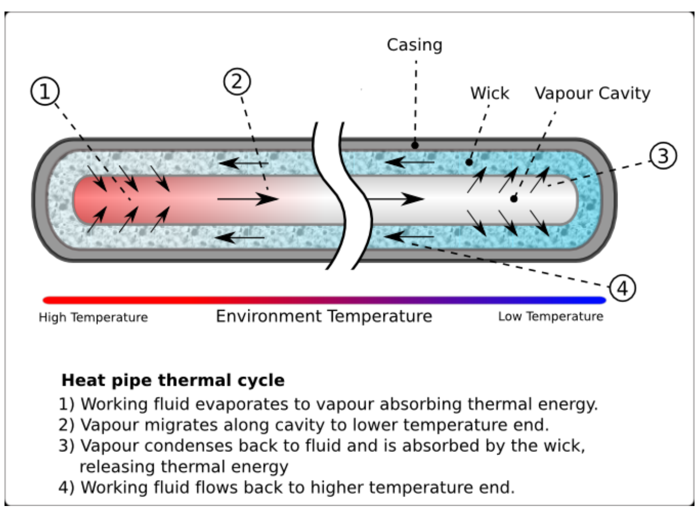

Figure 2: Heat pipe physics and theory

3. Design

3.1 Benchmarking

Benchmarking was done to have a baseline on the overall cooling system design. The specifications for a New Focus Picomotor (Model 8081) were used. A single picomotor generates approximately 32W of heat dissipation. A total of 3 motors will be used for this design. A picture of a picomotor is shown in Figure 3.

Figure 3: New Focus picomotor (model 8081)

Picomotors are ideal for precision lathes as they posses close to a nm resolution. However, it is not widely used in the industry for this purpose due to the heat dissipation problem that often reduces the accuracy of the machine.

In this project, a combination of heat pipes and heat sink will be designed to dissipate the heat generated by three picomotors. Various fluids and wick designs will be analyzed for the heat pipes. Different fin geometry, dimensions and optimum design will be investigated. The best combination that delivers the most efficient solution to dissipate 96W (32W from each picomotor) of heat generation will be simulated.

3.2 Constraints and Operating Conditions

The heat pipes will be mounted on the base of the picomotors that is 2cm x 2cm x 2cm (length x width x height) in size. Two of the pipes will bend in to the center and then three pipes will run parallel to the heat sink as shown in Figure 1. The entire enclosure where the heat pipes and will be placed in sealed and only the tip of each pipe will project outside the enclosure into the fins.

The benchmarked samples were used to set constraints on the heat pipe cooler to be designed. The heat pipe length was set at 15 cm with the evaporator and condenser sections set at 5 cm each. The outer diameter of the heat pipe was given a range between 5 to 8 mm. The constraints set on the cooling fins were a base length of 15 cm and profile height of 12 cm. the heat pipe must dissipate a total 96 W of heat generated. Other parameters of the heat pipe and cooling fins were to be optimized for maximum heat dissipation based on these constraints.

Each heat pipe must be able to dissipate 32 W of heat generated by the picomotor at maximum load. The operating temperature range is between 30 °C and 80 °C. The maximum operating temperature of 80 °C was chosen based on a factor of safety of 1.33 for the picomotor’s operating conditions. The heat pipe must also be able to withstand a bonding temperature of 160 °C in the manufacturing process.

3.3 Analytical Model (MathCAD)

MathCAD models were developed to optimize the heat pipe and cooling fin parameters for the required heat dissipation. Two separate modules were utilized for heat pipe design and cooling fin design respectively.

3.3.1 Heat Pipes

The first part of the heat pipe design module is the definition and interpolation routine of the working fluids for the heat pipe. Three different working fluids were selected to be evaluated eventually selecting the most suitable fluid for this application.

Acetone, ammonia and FLUTEC PP9 were the working fluids selected for evaluation. The interpolation routine used data for three different temperature points (0, 40 and 80 °C) to interpolate thermodynamic properties of the working fluids for any required temperature in that range.

The next section was the geometry definition section where the heat pipe length, operating temperatures, material properties and vapor area were defined. Vapor area was an arbitrarily selected number based on the outer radius constraint. An area of 0.25 cm^2 was selected as a starting point to which further optimizations could be made if necessary.

Based on the defined working fluids, geometry and operating temperatures heat pipe limits were calculated. The first limit evaluated was the sonic limit which is defined by the equation,

$$q_s := 0.474 h_{fg} \cdot A_v \cdot \sqrt{\rho_v \cdot P_v}$$

where q_s represents the sonic limit, A_v represents vapor area and h_{fg}, ρ_v and P_v represent latent heat, vapor density and vapor pressure of the working fluid at minimum operating temperature.

The next limit evaluated was the entrainment limit defined by the equation,

$$q_e := A_v \cdot h_{fg} \cdot \sqrt{\frac{\sigma \cdot \rho_v}{2 \cdot r_{hw}}}$$

where q_e represents the entrainment limit, A_v represents the vapor area and h_{fg}, σ, ρ_v and r_{hw} represent latent heat, surface tension, vapor density and hydraulic radius at the maximum operating temperature.

The next step was evaluating the boiling limit which is defined by the equation,

$$q_b := \frac{4 \cdot \pi \cdot L_e \cdot k_{eff} \cdot T_v \cdot \sigma}{h_{fg} \cdot \rho_v \cdot \ln\left(\frac{r_i}{r_v}\right)}$$

where q_b represents the boiling limit, L_e represents the evaporator length, k_{eff} represents the effective thermal conductivity, T_v represents the vapor temperature, r_i and r_v represent the inner and vapor radius and h_{fg}, σ, ρ_v represent latent heat, surface tension and vapor density at the maximum operating temperature.

After evaluating the sonic, entrainment and boiling limit, an evaluation of the required pipe wall thickness was done. This evaluation was done based on the maximum temperature the heat pipe would be exposed to which corresponds to the bonding temperature during manufacturing. The wall thickness evaluation is based on the equation,

$$t_{wall} := \frac{P_v \cdot r_i}{0.1 \cdot \sigma_u}$$

where, t_{wall} represents the wall thickness, P_v represents the vapor pressure at bonding temperature, r_i represents the inner radius of the heat pipe and σ_u represents the ultimate strength of the heat pipe material with a factor of safety of 10 taken into account.

The next evaluation carried out is the capillary limit. Two different wick structures which are the homogeneous screen wick and arterial wick were chosen to be evaluated. Capillary limits were calculated for both wick structures. For the homogenous screen wick the capillary limit is governed by the equation,

$$\Delta P_{cm} := \Delta P_{v} \cdot q_{c} + \Delta P_{l} \cdot q_{c} + \Delta P_{axial}$$

where q_c represents the capillary limit and ΔP_{cm}, ΔP_v, ΔP_l and ΔP_{axial} represent capillary, vapor, liquid and axial pressure drops respectively. The equation for the arterial wick is slightly more complicated defined by the next equation,

$$\Delta P_{cm} = \Delta P_{la} \cdot q_{c} + \Delta P_{lc} \cdot q_{c} + \frac{1}{\dfrac{1}{\Delta P_{v} \cdot q_{c}^{\frac{7}{4}}} + \dfrac{1}{\Delta P_{l} \cdot q_{c}}} + \Delta P_{axial}$$

where q represents the capillary limit and ΔP_{cm}, ΔP_{la}, ΔP_{lc}, ΔP_v, ΔP_l, ΔP_{axial} represent capillary, liquid along pipe length, liquid circumferential, vapor, liquid and axial pressure drops respectively.

With all the heat pipe limits evaluated, a performance map was generated to visualize the performance of the heat pipe. The total thermal resistance of the heat pipe was also calculated and used to determine the temperature drop from the evaporator to condenser. Calculations were also performed for heat pipe mass and heat sink temperature.

3.3.2 Heat sink

The next design module is the cooling fin design module. The section of the module defines the geometry of the cooling fins and the operating conditions. The base width, base length and profile height were all constrained parameters set at 5, 2 and 1.5 cm respectively as the fins had to fit in a given area in the notebook computer. A blower fan with an output of 20 cubic feet per minute (CFM) was selected based on common specifications. With fin dimensions and blower specifications set, properties such as flow velocity (U_{air}), Reynolds number (Re_L) and heat transfer coefficient (h_L) were determined. Based on these properties, optimum fin spacing (z_{opt}) and number of fins (N_f) were determined.

The next step was determining the characteristic length (L_c) which would be used in calculating the fin area (A_f), fin profile area (A_p) and fin cross section area (A_c). These values were then used to calculate fin efficiency which is given by the equation,

$$\eta_f := \frac{\tanh(mL_c)}{mL_c}$$

$$mL_c := \sqrt{\frac{2 \cdot h_L}{k_{al} \cdot A_p}} \cdot L_c^{\frac{3}{2}}$$

where η_f represents fin efficiency and mL_c is the quantity calculated using the values obtained previously. With fin efficiency now calculated, the next step was calculating overall efficiency which is given by the equation,

$$\eta_o := 1 - \frac{N_f \cdot A_f}{A_t} \cdot \left(1 - \eta_f\right)$$

where η_o represents overall efficiency and number of fins (N_f), fin area (A_f) and fin efficiency (η_f) all described previously.

The important step in this analysis is determining fin array heat dissipation which is described by the following equation,

$$q_f := h_L \cdot A_t \cdot \eta_o \cdot \theta_b$$

where q_f represents fin array heat dissipation and heat transfer coefficient (h_L), overall efficiency (η_o) and temperature difference (θ_b) all described previously.

Other calculations carried out were fin effectiveness, total thermal resistance, fin array mass and factor of safety. Plots of heat dissipation, overall and fin efficiency, fin effectiveness, thermal resistance and fin mass as a function of fin thickness were also made.

Details of equations used in the heat pipe and cooling fin analysis can be found in the Appendix section.

3.4 Simulation (CFD Model)

3.4.1 Heat Pipes

The CAD model was imported into GAMBIT using the ACIS format. Meshing the geometry was a challenge due to its complex nature. Many errors were encountered so a different file format was used. The new format, IGES, eliminated the errors and the meshing was able to be finished. The material was defined as aluminum for the pipe itself and acetone for the fluid inside. The meshed heat pipe model can be seen below.

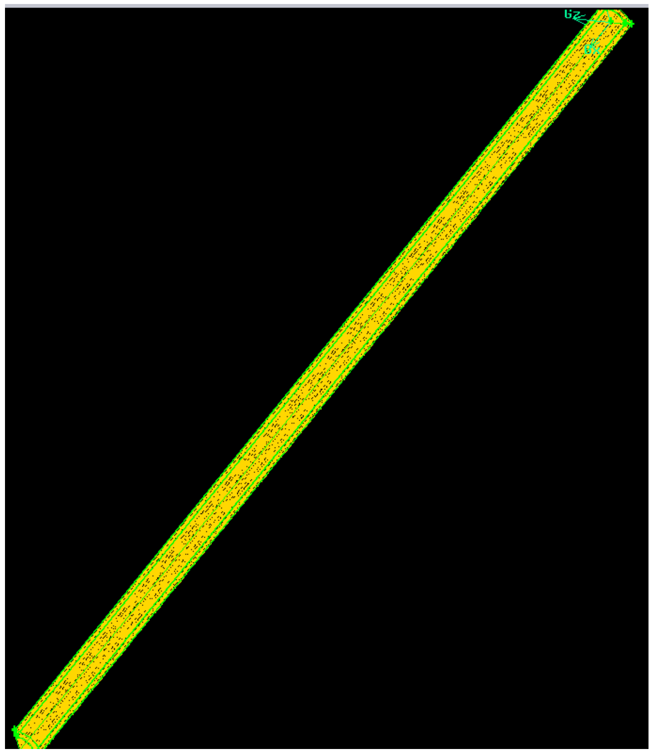

Figure 4: Image of meshed heat pipe from Gambit

The curves of the heat pipe were meshed first to define the end points of the heat pipe. The lengths were divided into 10 equal sections. The edges that define the length of the heat pipe were then separated into 150 sections. The Quad elements were used in this mesh for all of the faces. The Cooper Hex/Wedge elements were selected to mesh the volumes. The center section of the heat pipe was assigned as a counter flow to allow the acetone to flow freely throughout the pipe and best simulate the actual phenomenon. A schematic of this idea can be seen below. When the acetone is heated it vaporizes and flows to the condenser section. The vapor then returns to a liquid state and flows back to the evaporator to begin the process again. The flow is created by a pressure difference which is created by the temperature difference at the ends of the pipe.

The saved mesh file is then imported into Fluent. The scale is then set to mm because that is the unit the grid was created in. Turbulence models and standard solvers were then set to best model the flow. The turbulence model used for the heat pipe was the k-epsilon model. The operational temperature was then set to 300 K and gravity was set to 9.81 m/s in the negative Y direction. Finally, the boundary conditions were set and the material properties checked to ensure that everything was correct. The solver was then initialized and set to iterate 500 times.

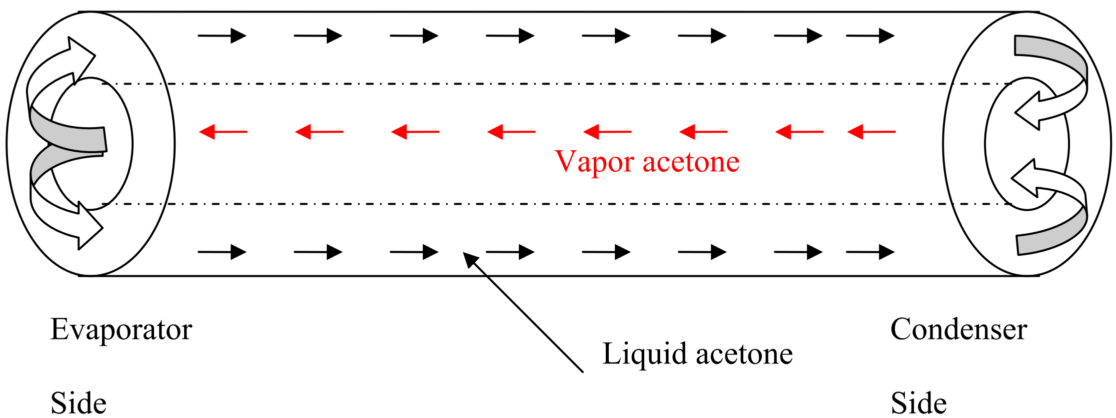

Figure 5: Schematic of heat pipe theory that was used for design

3.4.2 Heat Sink

The heat sink was modeled separately from the heat pipes themselves. The heat pipes were included to provide the necessary energy to the fin array but were not the focus. The fins were modeled and then saved in the IGES format to prevent the problems that were encountered previously from occurring. The fins were placed inside of a box that would provide a volume to simulate the air. The boundaries of this box were set to pressure outlets to prevent any interference between the sides of the box and the flow. All other faces were set to the wall boundary condition. The fins were set to solid and the volume representing the air was set to fluid. The heat pipes were meshed in a similar manner as the ones in the previous model. The fins and containment volume were meshed directly using the Cooper Hex/Wedge elements. The volume meshing was done using an interval of 2 instead of the specific number of interval method. The fin and containment mesh can be seen below. The same Fluent procedure was then followed with one notable exception. A turbulent model was not used because it was found that the flow around the fins would stay in the laminar regime. The properties of the materials that were used are shown in a table below.

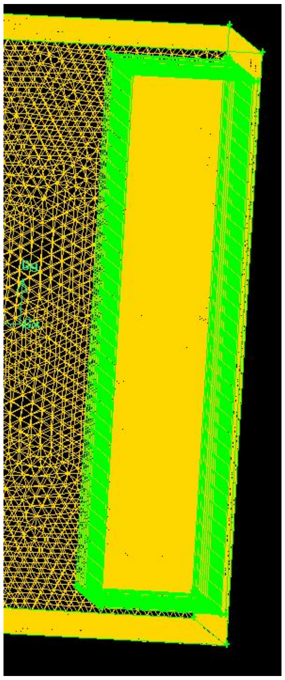

Figure 6: Meshed heat sink carried out on Gambit

Material	Type	Solid Density kg/m^3	Liquid Density kg/m^3	Vapor Density kg/m^3	Thermal Conductivity W/m K
Acetone	Liquid	-	782.3	-	0.1789
Acetone	Vapor	-	-	0.7835	0.1789
Aluminum	Solid	2702	-	-	237

Table 1: Material properties used for analytical and CFD sections

4. Results

4.1 Analytical Model (MathCAD)

4.1.1 Heat Pipes

After careful analysis and optimization in the MathCAD model, the specifications of the heat pipe were finalized. The initial constraints were maintained. These constraints are heat pipe length (L) which was set at 15 cm and evaporator and condenser length (L_e & L_c) which were set at 5 cm each. The minimum and maximum operating temperatures were set at 30 and 80 °C respectively. The selected heat pipe casing and wick material was aluminum and stainless steel respectively. The working fluid selected was acetone.

With all these parameters set, the maximum limits were determined. For the configuration above, the sonic, entrainment and boiling limits were 1025.56, 229.13 and 66.76 W respectively. Two different wick structures were examined to observe the differences in capillary limits of each. The first was a conventional homogeneous screen wick, which was 2 layers of #500 stainless steel mesh and the second was an arterial wick with an arterial depth (δ) of 0.5mm and an arterial width (ω) of 1.0 mm. The homogeneous screen wick possessed a capillary limit of 0.33 W, which was clearly not desirable in this application requiring 32 W of heat dissipation. The arterial wick on the other hand had a more desirable capillary limit of 65.34 W which is required for this application. The arterial wick was the selected wick for this application.

After analyzing the maximum operating temperature which occurred at bonding during the manufacturing process, the wall thickness of the heat pipe was set at 0.1 mm.

This resulted in the heat pipe outer diameter of 6.045 mm which falls within the constraint set between 5 to 8 mm. From these results it is shown that the capillary limit is the critical limit in this design. With a capillary limit of 65.34 W and a required heat dissipation of 40 W, this design provides a factor of safety of 1.634. Figure 7(a) and (b) below shows the performance map of this heat pipe design. There were a total of two final designs of heat pipes as one design was used for the x and y-stage picomotors and the other was used for the z-stage picomotor.

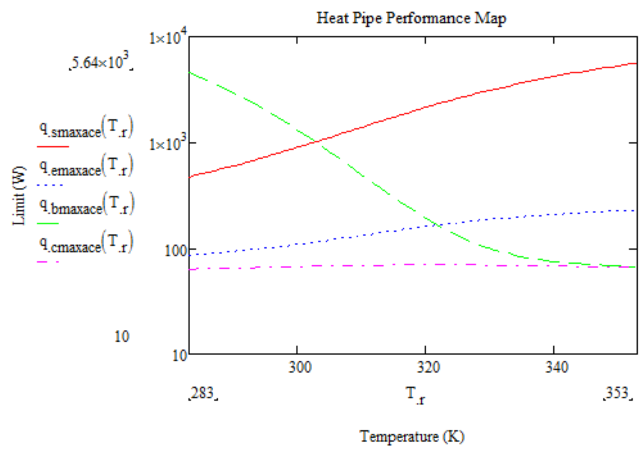

Figure 7(a): Heat pipe performance map for the x and y-stage picomotors

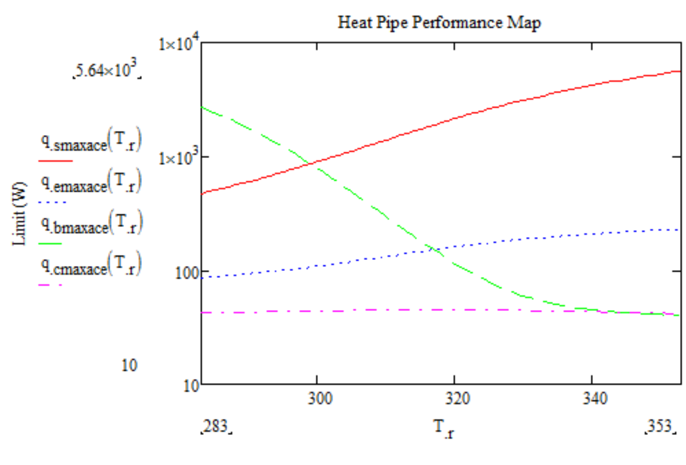

Figure 7(b): Heat pipe performance map for the x and y-stage picomotors

The overall thermal resistance of the heat pipes was determined to be 1.048 K/W which resulted in a temperature drop between the evaporator and condenser of 20.12 °C. The mass of the heat pipe with the arterial wick was determined to be 1.376 grams.

4.1.2 Heat Sink

For the design of the cooling fins, the initial constraints were. Keeping the constraint parameters in mind, the optimum fin spacing was determined to be 6.19 mm. The fin thickness was then set to 1 mm based on iterative solutions. With these parameters, the number of fins was calculated to be 20. This resulted in a fin and overall efficiency of 0.935 and 0.939 respectively. This fin array configuration was determined to dissipate 97.83 W of heat which satisfied the design requirement of 96 W with a factor of safety of 1.22. The fin effectiveness of the array was determined to be 113.176 with an overall thermal resistance of 0.613 K/W. Finally the mass of the fin array was determined to be 389 grams. Figures 8 through 12 show the variation of total heat transfer as a function of fin thickness.

Detailed analysis and results for the heat pipe and cooling fin design can be found in Appendix.

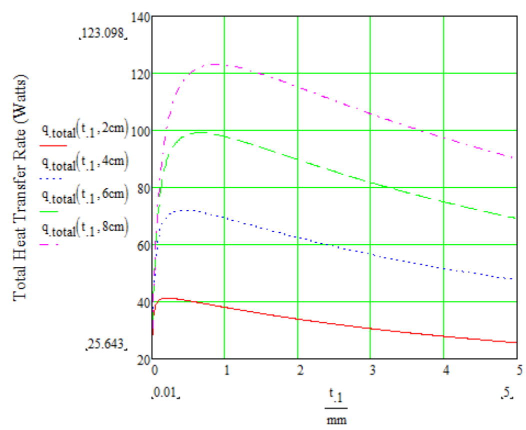

Figure 8: Heat Dissipation versus Fin Thickness

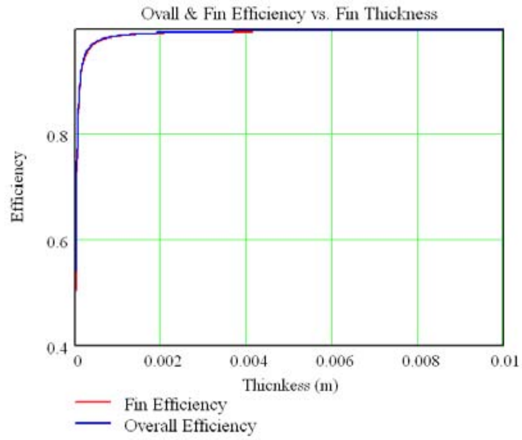

Figure 9 : Overall & Fin Efficiency vs. Fin Thickness

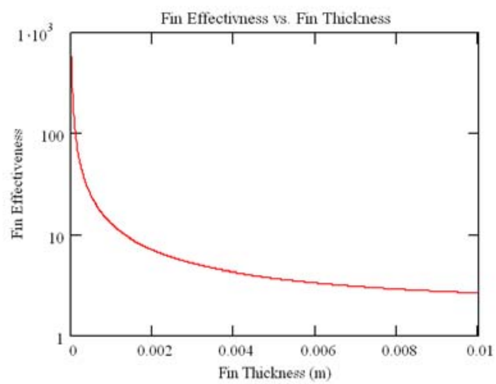

Figure 10: Fin Effectiveness vs. Fin Thickness

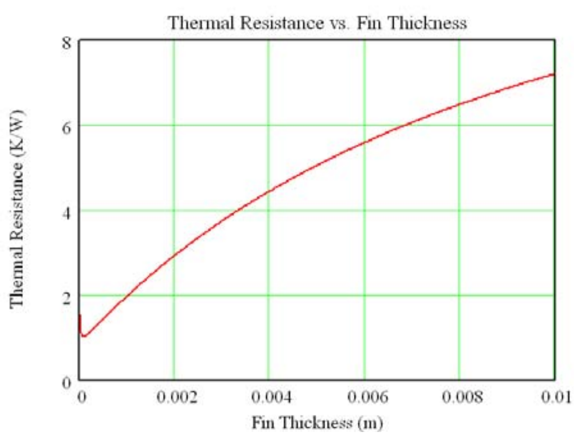

Figure 11: Thermal Resistance vs. Fin Thickness

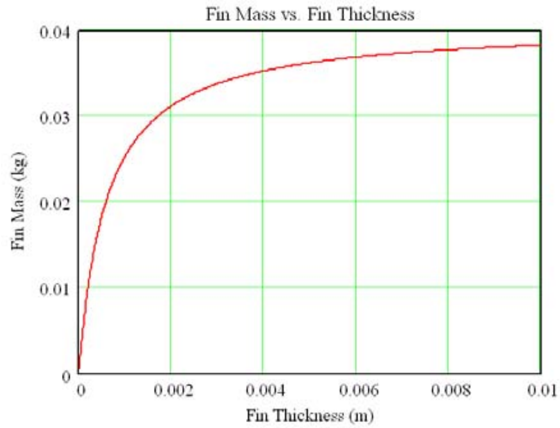

Figure 12: Fin Mass vs. Fin Thickness

PARAMETER	Max. optimum array
Fin thickness, **t** (mm)	1
Profile Length, **b**(cm)	6
Spacing, **z**(mm)	6.19
Heat Transfer Coefficient, **h**	5.677
Total heat Transfer rate, **qtotal**	97.83
Number of fins	20
Mass of fins, **m** (grams)	389
β = mb	1.421
Fin efficiency, **ηf**	0.935
Fin effectiveness, **εf**	113.176
Overall efficiency, **ηo**	0.939
Overall thermal Resistance	0.613

Table 2: Results obtained from the analytical model for the heat sink.

4.2 CAD Model

4.2.1 Heat Pipes

The heat pipe consists of three sections, evaporator, condenser, and adiabatic. The overall length of the pipe that was modeled is eight centimeters. The outer diameter is 6.045 mm with a thickness of 0.1 mm. The material selected for the heat pipe was aluminum with a density of 2702 kg/m^3. The fluid used in this design is acetone. It was chosen because of its ability to operate from 0 to 120 degrees Celsius. The heat pipe is shown below with the different sections labeled.

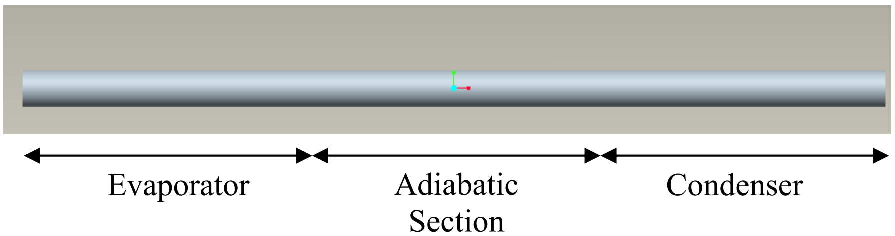

Figure 13: CAD model of the heat pipe with defined sections

4.2.2 Heat Sink/Heat Pipe

The heat sink was designed to dissipate the required energy. This resulted in an overall width of 15 cm, and a base height of 12 cm. Each fin has a length of 6 cm and a thickness of 1 mm. Optimum spacing for the fins is 6.19 mm which yields 20 fins for the given base dimensions. Material for the fins is the same aluminum which was selected for the heat pipe. The heat pipes and the heat sink were modeled together to carry out the computational fluid dynamics simulations.

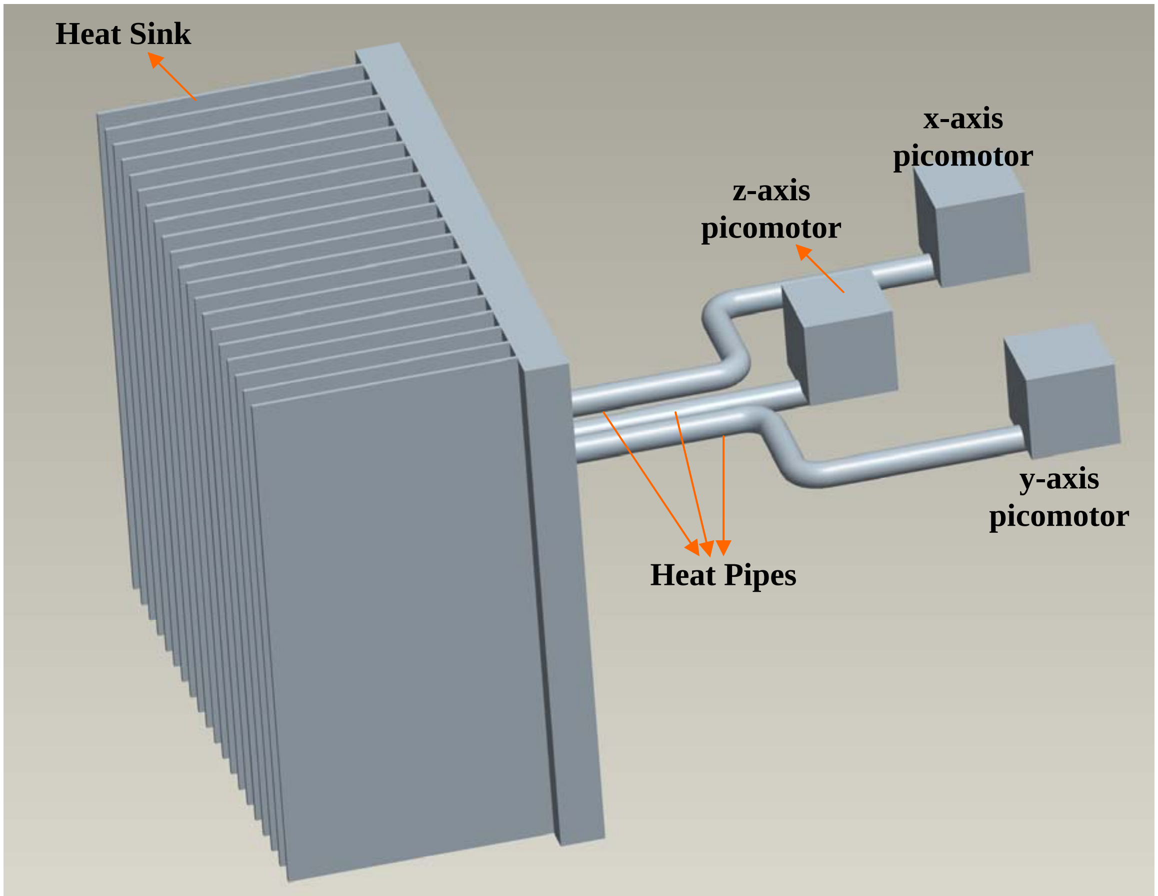

Figure 14: CAD drawing of the picomotors, heat pipes and heat sink carried out on SolidWORKS

4.3 Simulation Results (CFD)

4.3.1 Heat Pipes

Shown below are the simulation results obtained using the Fluent 6.3 software package.

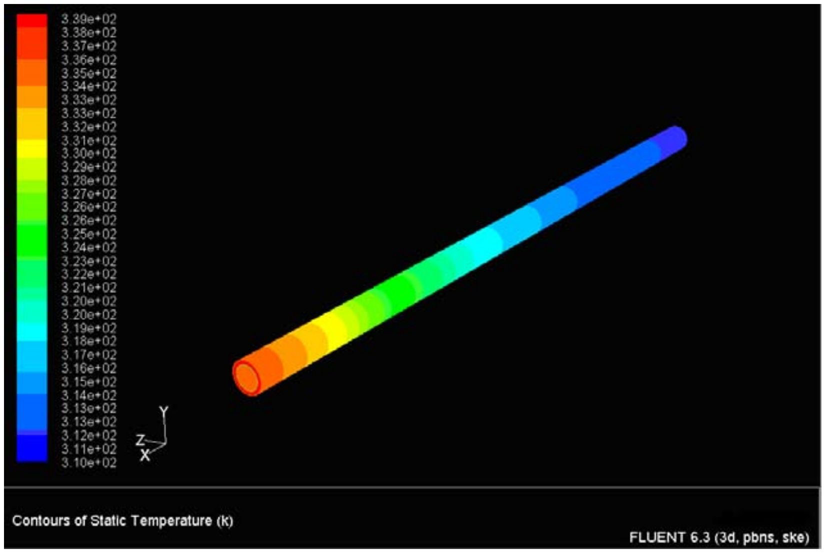

Figure 15: Static temperature contour for heat pipe

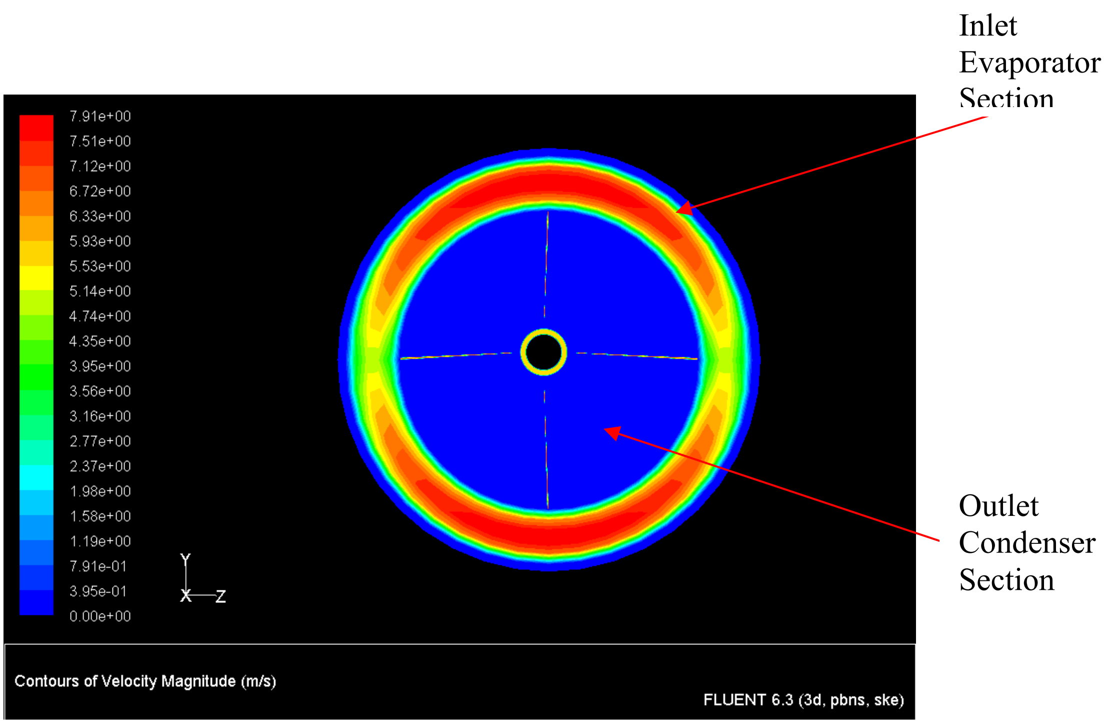

Figure 16: Cross section of heat pipe showing velocity magnitude contours

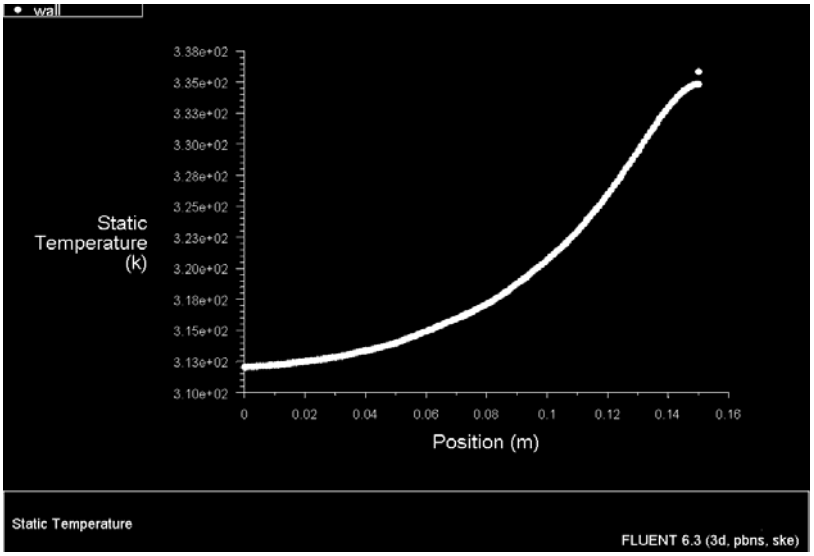

Figure 17: Static temperature distribution of heat pipes

4.3.2 Heat pipes/Heat sink

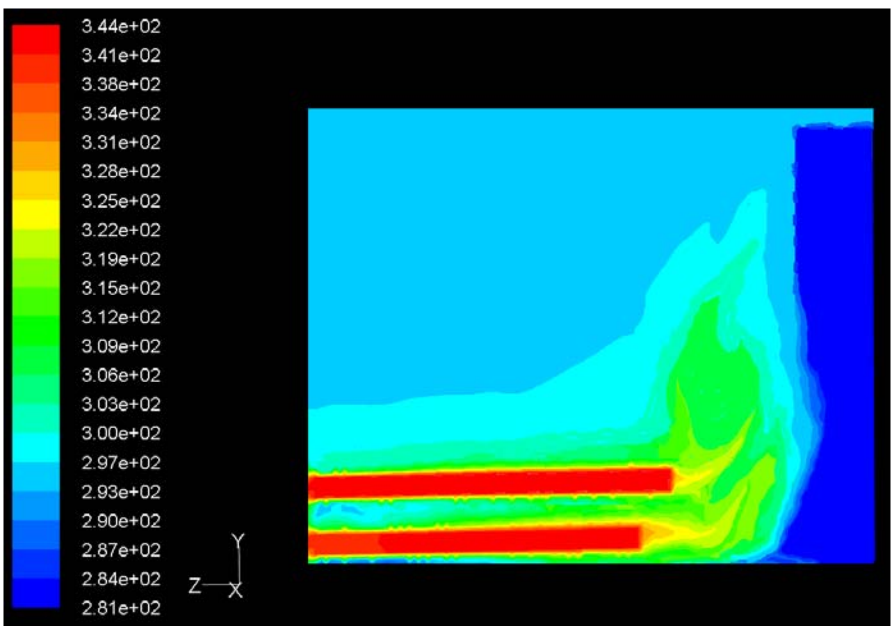

Figure 18: Static temperature dstribution of the heat pipes and heat sink.

From the temperature distribution seen in the above figure, the coldest part is the heat sink (as expected) and the hottest part are the heat pipes as the pipes are directly dispersing heat away from the picomotors. The temperature flow shows that the heat is rising and being dissipated by the fins/heat sink.

5. Summary

In summary, this project has provided a brief insight into the area of picomotors and precision lathes cooler design. Systematic design methods have been employed to successfully design picomotor coolers for the given heat dissipation requirements and constraints. A detailed MathCAD model has been developed to enable the optimization of different parameters in a precision lathe assembly to fit the specified requirements. Along with this a CAD model has also been developed to provide 3D visualization and a basis for CFD simulation. A systematic CFD study has also been performed to try and simulate the analytical MathCAD model. A robust design method is now in place for the analytical and numerical design approach for a picomotor cooler. In summary, this project has been successful in meeting its goals.

6. References

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8. Heat Pipe research at LANL

7. Appendix

7.1 Appendix A – MathCAD Model for Heat Pipes (x and y-stage picomotors)

7.2 Appendix B – MathCAD Model for Heat Pipes (z-stage picomotors)

7.3 Appendix C – MathCAD Model for Heat Sink