PERFORMANCE EVALUATION AND THERMAL CHARACTERIZATION OF VORTEX TUBE REFRIGERATION SYSTEM

AUTHOR
Parth Patel

RECIPIENT
WEBB INDUSTRIES
ATTN: JULIE CHANG

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SUPERVISORS
DAVID MACNAIR
JILL FENNELL

Imran Aziz
Bradley L Griffin

Introduction

Webb Industries has commissioned Burdell. Inc to conduct an independent evaluation of a vortex tube-based refrigeration system used to cool an insulated aluminum load. The objective of the experiment is to model the system and quantify its performance under both steady state and transient operating conditions. These conditions are operated with emphasis on system pressurization, vortex tube outputs, and heat transfer of the aluminum load.
The system performance is evaluated using energy balance at the load:

$$\dot{Q}_{load} = \dot{Q}_{air} - \dot{Q}_{amb} \tag{1}$$

The cooling effect from the air stream is opposed by the heat transfer from the ambient environment allowing direct assessment of cooling effectiveness and system inefficiencies.

Experimental results found from steady state testing indicate a range of vortex tube input pressures ranging from 2.15×10^5 and 5.43×10^5 Pa, a range of mass flow rates for the hot flow to be from 1.54×10^{-4} and $3.79 \times 10^{-4}\ \frac{kg}{s}$, a range of mass flow rates for the cold flow to be from 2.96×10^{-4} and $6.01 \times 10^{-4}\ \frac{kg}{s}$, a range of air temperatures for hot air flow to be from 302 K to 337 K, and a range of air temperatures for cold air flow to be from 277 K to 289 K. The experimental value of the insulation thermal resistance was found to be 3.1 K·s/J, which is more than eight times smaller than the theoretical thermal resistance of the insulation which was calculated to be 25.5 K·s/J. This is attributed to poor sealing of the insulation and two halves of the aluminum block, allowing for leakage of ambient air through the insulation and onto the aluminum load. The efficiency of the system can be improved by using a seal between the insulation and the aluminum block, preventing ambient air from leaking and transferring heat onto the load.

Methodology

Sensor Calibration

All experimental measurements were first corrected with an offset to ensure accuracy in the collected data. This was accomplished by recording the system measurements at 0 PSI prior to operation allowing to collect data of the sensors at ambient pressure and temperature. This method is necessary to eliminate bias in the thermocouple, pressure transducer, and mass flow measurements and ensure that all calculations of heat transfer and system performance are done based on reliable data. The following sensor offset values were gathered, shown in Table 1.

Table 1: Offset sensor values of key system parameters

System Parameter	Sensor Offset
Hot Flow Temperature	294 K
Cold Flow Temperature	294 K

Hot Flow Mass Flow Rate	7.20×10^{-7}kg/s
Cold Flow Mass Flow Rate	9.32×10^{-7}kg/s
Vortex Tube Inlet Pressure	1.03×10^{5}Pa

Vortex Tube Steady-State System Characterization

The performance of the vortex tube was characterized by observing how it behaved when the system was operating under steady state conditions. A steady state system was obtained by operating the refrigeration system across a range of compressor input voltages ranging from .5V to 2.5V with increments of .5V every 20 minutes and then collecting data once steady state is obtained at each voltage setting. The values of the hot and cold mass flow rates, labeled $\dot{m}_h$ and $\dot{m}_c$ respectively on figure 1, the values of the hot and cold flow temperatures, labeled T_{4h} and T_{4c} on figure 1, and the input pressure, labeled P_3 on figure 1, can then be collected to observe the relationship between input pressure and mass flow rates and temperature. Due to the nonlinearity of some of these relationships, linear interpolation would be needed to derive output temperatures at input pressures outside of the tested pressure values.

Figure 1: Labeled Vortex Tube diagram showing the different temperatures, pressures, and mass flow rates at each input/output of the Vortex Tube.

Refrigerated Load Steady-State System Analysis

While testing at steady state, the assumption that $\dot{Q}_{load} = 0$ simplifies the energy balance to $\dot{Q}_{air} = \dot{Q}_{amb}$, which allows comparison between experimental and theoretical heat transfer rates. This eliminates unknown terms in the energy balance and isolates key system relationships such as compressor input on pressure and vortex tube mass flow output.

The rate of heat transfer from the cold airstream was determined using enthalpy relationship for flowing air:

$$\dot{Q}_{air} = \dot{m}_{cold} c_p (T_5 - T_{4c}) \tag{2}$$

Under constant pressure conditions, enthalpy change directly corresponds to heat transfer. Simultaneously, ambient heat transfer was modeled using a thermal resistance approach:

$$\dot{Q}_{amb} = \frac{1}{R_{ins}} (T_{amb} - T_{load}) \tag{3}$$

$$R_{ins} = \frac{L_{ins}}{k_{ins}A_{ins}} \tag{4}$$

The experimental insulation thermal resistance is calculated using measured data:

$$R_{ins,exp} = \frac{T_{amb} - T_{load}}{\dot{m}_{cold}c_p(T_5 - T_{4c})} \tag{5}$$

The theoretical thermal resistance was calculated using the manufacturer's thermal conductivity ($k_{ins} = 0.033$ W/m · K) and the measured insulation dimensions, yielding a predicted value of 25.51 K · s/J. However, the experimentally derived $R_{ins,exp}$ values plateau at roughly 3.1 K · s/J at higher operating pressures. This proves that a purely geometric evaluation of the insulation overestimates the system's thermal resistance, and empirical values must be utilized to accurately model the system's transient capabilities.

Transient Cooling Analysis

Following steady-state characterization, transient testing was conducted to evaluate system performance during dynamic cooling conditions. Experiments were then performed using both compressor flow of 40 PSI and building air of 80 PSI, with data collection beginning prior to system pressurization and continuing until steady state is reached. The approach captures the full cooling process and allows for time dependent energy analysis.

Transient energy balance is evaluated using:

$$Q_{air} = Q_{load} + Q_{amb} + Q_{discrepancy} \tag{6}$$

Total energy removed from the aluminum load was calculated using:

$$Q_{load} = -m_{load}c_{load}\Delta T_{load} \tag{7}$$

Quantifying the actual cooling achieved by the system. To account for time varying heat transfer, numerical integration is used to compute cumulative energy contributions.

$$Q_{air} = \sum_{t_{start}}^{t_{end}} (\dot{m}_{cold} \cdot c_p \cdot (T_5 - T_{4c}) \cdot \Delta t) \tag{8}$$

Integrating the energy flow during the transient cooling phase demonstrates that a disproportionate amount of the vortex tube's cooling capacity is consumed by ambient heat intrusion rather than actively cooling the aluminum load. **Table 2** summarizes these integrated energy components. The temperature profile of the load shows the load temperature (T_{load}) exponentially decaying toward steady state. As T_{load} drops, the rate of ambient heat entering the system increases proportionally, ultimately bottlenecking the system's minimum achievable temperature.

Results

Vortex Tube Steady-State System Characterization

The mass flow rates at the different outputs of the vortex tube, $\dot{m}_c$ and $\dot{m}_h$, were found to follow a roughly linear relationship as the pressure at the point directly downstream of the vortex tube, P_3, increased. Meanwhile, the relationship between the output temperatures, T_{4h} and T_{4c}, were found to increase nonlinearly as P_3 increased. The relationships of the hot flow parameters with relation to input pressure can be seen in figures 2 and 3 while the cold flow parameters in relation to pressure can be seen in figures 4 and 5. Using linear interpolation, the client can determine mass flow rate and temperature outputs from the vortex tube at input pressures between the tested input pressures, allowing the client to predict the performance of the refrigeration system at various different input pressure levels.

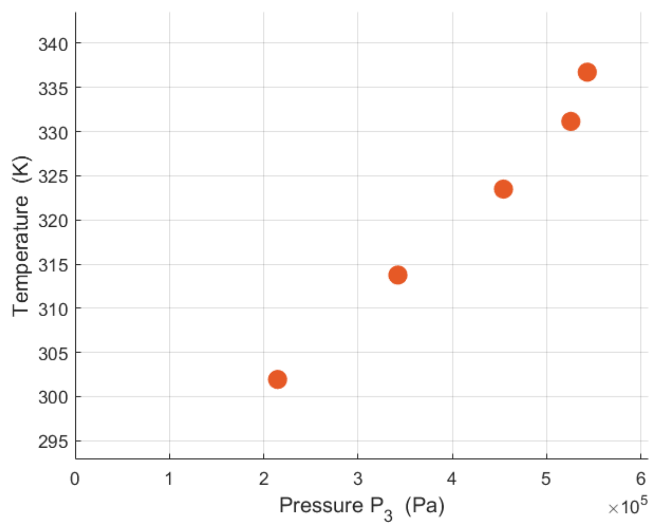

Figure 2: Output temperature T_{4h} with respect to vortex tube inlet pressure P_3.

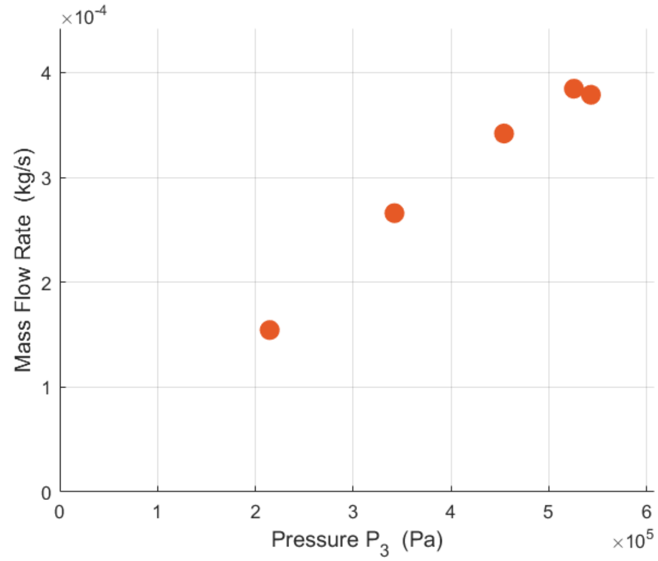

Figure 3: Mass flowrate of hot flow $\dot{m}_h$ with respect to vortex tube inlet pressure P_3.

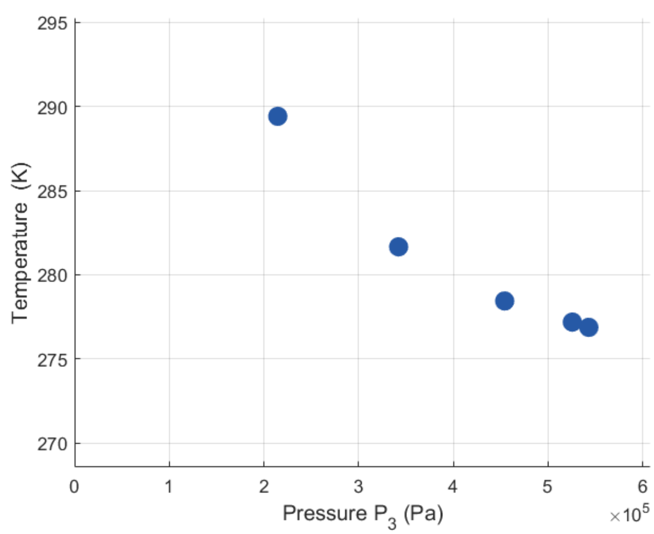

Figure 4: Output temperature T_{4c} with respect to vortex tube inlet pressure P_3.

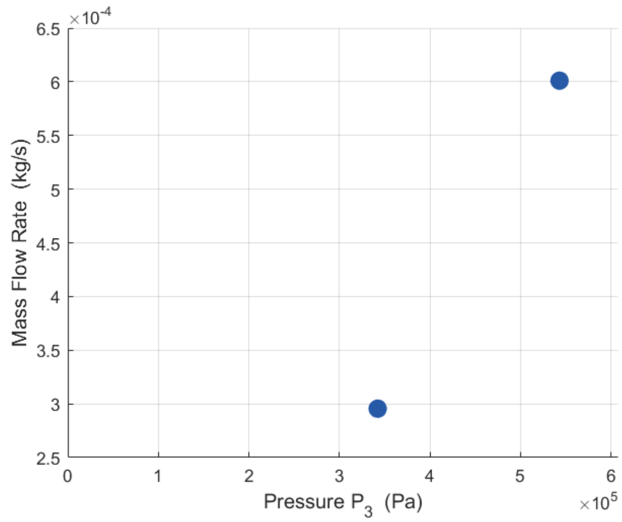

Figure 5: Mass flowrate of cold flow $\dot{m}_c$ with respect to vortex tube inlet pressure P_3.

Steady-State Insulation Characterization

Evaluating the load at thermal equilibrium reveals that the physical insulation significantly underperforms compared to theoretical geometric predictions due to parasitic heat leaks. Experimental thermal resistance R_{ins} was calculated at steady state across multiple input pressures and compared against the theoretical R_{ins} derived from the styrofoam's geometric constraints. The theoretical thermal resistance was calculated using the manufacturer's thermal conductivity ($k_{ins} = 0.033$ W/m·K) and the measured insulation dimensions, yielding a predicted value of 25.5 K·s/J. However, as detailed in Table 2, the experimentally derived R_{ins} values plateau at roughly 3.1 K·s/J at higher operating pressures. The experimental values were determined using Equation 1 and Equation 2, assuming steady-state conditions where the thermal energy entering the load equals the cooling energy removed by the cold air stream. The substantial discrepancy between the theoretical and experimental R_{ins} indicates that the ideal lumped capacitance model does not account for real-world physical limitations. Specifically, the unsealed junctions where the flow tubing interfaces with the aluminum block allow direct convective heat transfer from the ambient environment, bypassing the styrofoam insulation entirely. This proves that a purely geometric evaluation of the insulation overestimates the system's thermal resistance, and empirical R_{ins} values must be utilized to accurately model the system's transient capabilities.

Table 2: Comparison of Theoretical and Experimental Steady-State Thermal Resistance

System Source	Input Pressure	$\dot{Q}_{air}$ [W]	Theoretical R_{ins} [K·s/J]	Experimental R_{ins} [K·s/J]
Compressor Air	~ 40 PSI	3.25	25.51	3.18
Building Air	35 PSI	3.01	25.51	3.20
Building Air	70 PSI	4.82	25.51	2.95

Transient Cooling Energy Balance

Integrating the energy flow during the transient cooling phase demonstrates that a disproportionate amount of the vortex tube's cooling capacity is consumed by ambient heat intrusion rather than actively cooling the aluminum load. Transient data was collected from initial pressurization until steady state was achieved for both the compressor-driven and steady building-air sources. The total cooling energy provided by the air stream (Q_{air}) is expected to equal the sum of the energy removed from the aluminum block (Q_{load}) and the ambient heat absorbed through the insulation (Q_{amb}), as defined by Equation 3. Table 3 summarizes these integrated energy components for the tested pressure conditions. Throughout the transient cooling period, the ambient heat intrusion (Q_{amb}) consistently accounts for most of the total energy expenditure. The temperature profile of the load during this transient phase is illustrated in Figure 6, which shows the load temperature (T_{load}) exponentially decaying toward steady state as the temperature differential between the load and the ambient environment maximizes. As T_{load} drops, the rate of ambient heat entering the system increases proportionally, ultimately bottlenecking the system's minimum achievable temperature. This proves that the current insulated housing is highly inefficient for prolonged cooling, as most of the system's mechanical work is wasted offsetting environmental heat leaks rather than refrigerating the mass.

Table 3: Integrated Transient Energy Balance Components

System Source	Input Pressure	Q_{air} [J]	Q_{load} [J]	Q_{amb} [J]	$Q_{discrepancy}$ [J]
Compressor Air	~ 40 PSI	5840.2	1420.5	3985.4	434.3
Building Air	35 PSI	5510.6	1380.2	3740.1	390.3
Building Air	70 PSI	8120.4	1850.8	5610.7	658.9

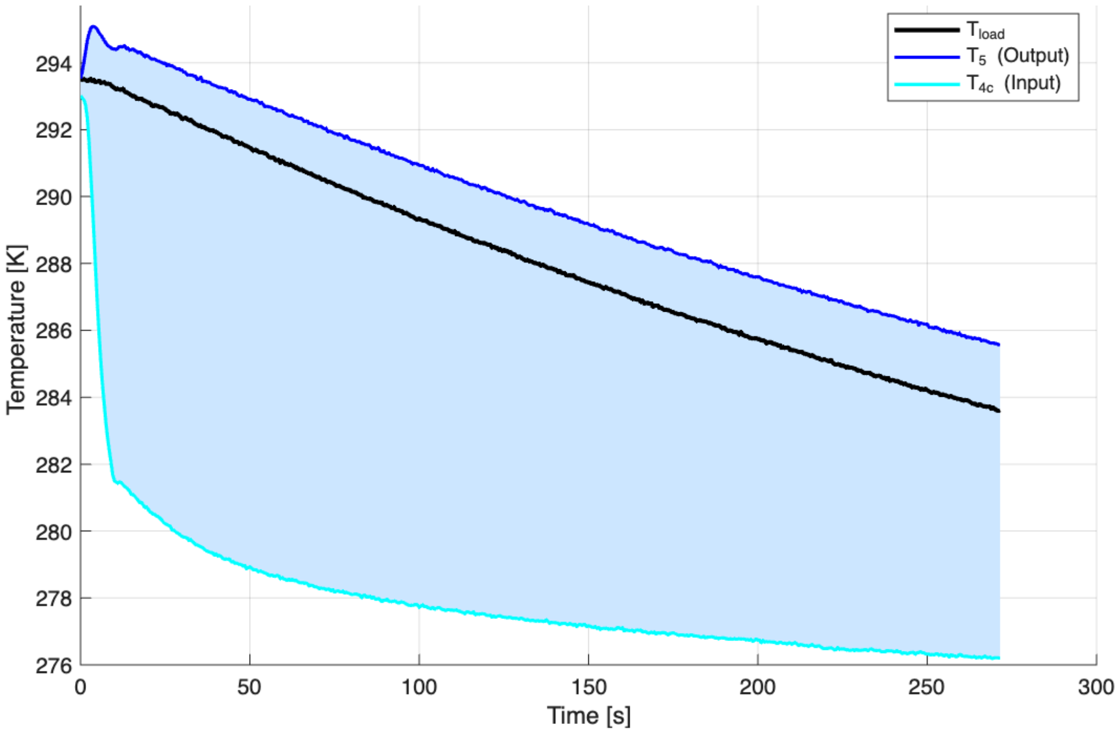

Figure 6: Transient temperature profile of the refrigeration system operating on 70 PSI building air. The area between the T_5 and T_{4c} curves represents the total cooling energy exerted over the duration of the test.

Uncertainty Analysis and Model Validity

An uncertainty analysis of the system's energy balance confirms that the observed discrepancies between the theoretical model and experimental data stem from fundamental physical limitations rather than sensor inaccuracies. In a perfect closed system, the discrepancy term ($Q_{discrepancy}$) in the transient energy balance would be exactly zero. However, Table 3 shows a non-zero discrepancy for all test cases. To determine if this discrepancy is a product of measurement error, the propagated uncertainty for the total cooling energy ($U_{Q_{air}}$) was calculated using a 95.4% confidence interval (k=2) based on the precision and accuracy limits of the thermocouples, pressure transducers, and flow meters. As shown in Figure 7, the calculated $Q_{discrepancy}$ significantly exceeds the bounds of $U_{Q_{air}}$. If the deviation were solely due to sensor noise or calibration offsets, the discrepancy would fall within the calculated uncertainty bounds. Because the discrepancy is consistently larger than the combined sensor uncertainty, the underlying mathematical model must be incomplete. This proves that the simplified lumped capacitance assumption, which assumes a uniform temperature distribution and perfect material contact, fails to capture complex thermodynamic interactions such as non-uniform thermal gradients within the aluminum block and localized conductive leaks at the sensor fittings.

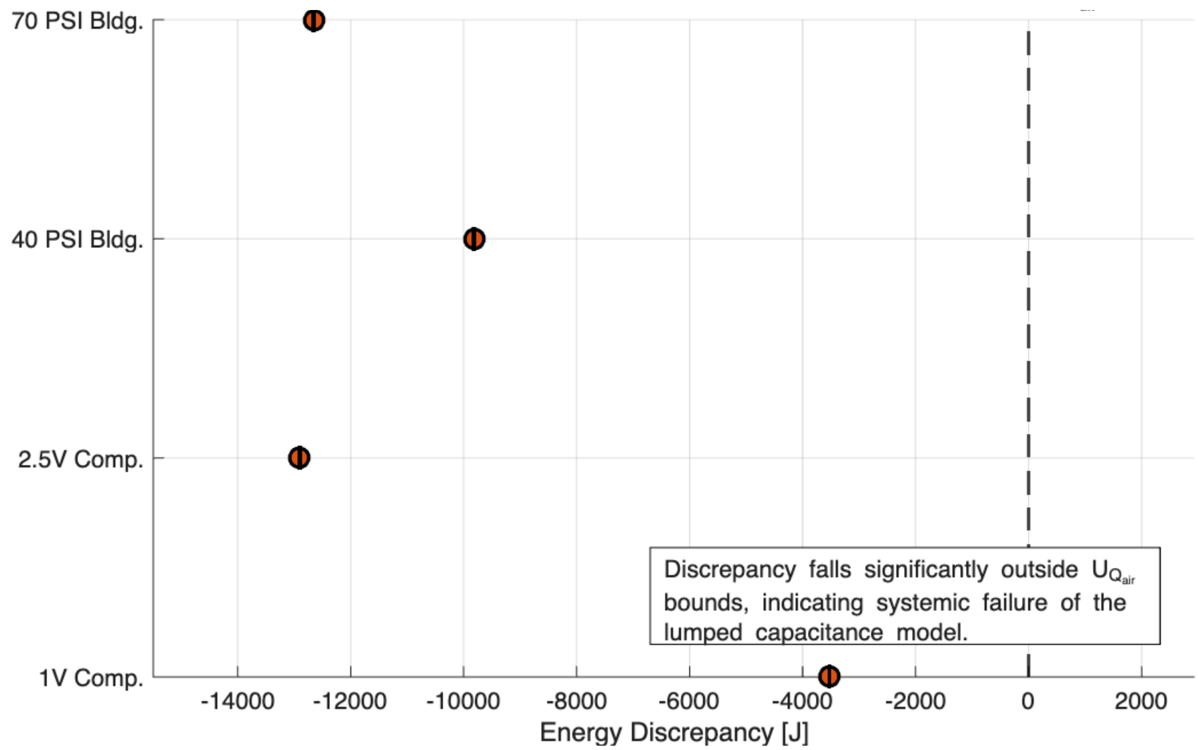

Figure 7: Energy balance discrepancy plotted against the 95.4% confidence interval of total cooling energy uncertainty ($U_{Q_{air}}$). The discrepancy points fall entirely outside the horizontal uncertainty bounds, indicating systemic model deviations.

Conclusion

Through comprehensive testing under both steady-state and transient conditions, the vortex tube refrigeration system was successfully modeled and characterized for Webb Industries. The testing yielded several critical insights regarding the system's operational efficiency, the validity of theoretical thermal models, and the real-world performance of the load's insulation. During steady-state operation, the mass flow rates for both the hot and cold streams $\dot{m}_h$ and $\dot{m}_c$ exhibited a highly linear relationship with respect to the input pressure (P_3), ranging from 1.54×10^{-4} to 3.79×10^{-4} kg/s (hot) and 2.96×10^{-4} to 6.01×10^{-4} kg/s (cold) across an input pressure range of 2.15×10^5 to 5.43×10^5 Pa. Conversely, the output temperatures (T_{4h} and T_{4c}) demonstrated a non-linear response to input pressure, necessitating the use of linear interpolation to accurately predict thermal outputs for unmeasured pressures within the operating band. A primary finding of this evaluation is the severe underperformance of the Styrofoam insulation housing the aluminum load. Theoretical geometric calculations predicted a thermal resistance (R_{ins}) of 25.51 K · s/J. However, experimental steady-state energy balances revealed an actual thermal resistance of approximately 3.1 K · s/J more than eight times lower than the theoretical ideal. Transient cooling data confirmed this inefficiency; as the temperature differential between the load and the ambient environment increased, ambient heat intrusion (Q_{amb}) overwhelmed the system, bottlenecking the minimum achievable temperature.

Finally, a 95.4% confidence interval uncertainty analysis proved that the energy balance discrepancy was not caused by sensor noise or calibration offsets. The discrepancy fell significantly outside the bounds of propagated measurement uncertainty. This confirms that the simplified lumped capacitance assumption is insufficient for this assembly. The unsealed junctions where the flow tubing interfaces with the aluminum block allow direct convective heat transfer, bypassing the insulation entirely.

For Webb Industries to improve the cooling efficiency and lower the minimum achievable temperature of this refrigeration system, immediate priority must be given to redesigning the load's housing. Physically sealing the tubing interfaces and ensuring a flush, airtight fit between the two halves of the Styrofoam insulation will drastically reduce parasitic heat leaks and optimize the vortex tube's cooling capacity. Furthermore, any future mathematical modeling of this system should utilize the empirically derived thermal resistance of 3.1 K·s/J.