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VIBRATION ISOLATION AND NOVEL COOLING ON A RECIPROCATING COMPRESSOR AT THE ROCHESTER INSTITUTE OF TECHNOLOGY

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ABSTRACT

Senior Design Project P12452 had two separate aspects designed to improve the operation of the reciprocating compressor located in the Mechanical Engineering Machine Shop. The first component of the project was to create a vibration reduction system to reduce the vibrations that occur while the compressor is running. The second component was to modify the existing cooling system to allow for a passive thermosyphoning cooling system, acting as a test bed that could be analyzed to better understand how these systems work.

The compressor now vibrates 23% less under load while the MR dampers are in use. With the MR dampers installed (donated by LORD Corporation) the compressor can be used as a vibration laboratory for students, showing the effects of different damping ratios upon the dynamic motion and deflection of the system. The compressor can now also run on either the forced cooling system installed by P11452, or with the thermosyphoning system designed and installed during the course of this project. This system can now also be used as a fluid mechanics lab, investigating the principles of convective heat transfer and flow.

Figure 1: Dresser-Rand Compressor

PROJECT BACKGROUND

The overall goal of this project is to continue improving upon the setup of the Dresser-Rand (D-R) compressor that was installed in the Mechanical Engineering department machine shop by Senior Design Team P11452. This D-R compressor is a single-piston, dual-action compressor, meaning that the piston compresses during both the forward and return strokes, smoothing the flow of the compressed air from the short bursts, characteristic of a single-action compressor. Unfortunately, due to the fact that it is a single-piston configuration, the system is inherently unbalanced. In typical installations, a large concrete pad is poured at the mounting location and the compressor is then hard-mounted to the pad. This minimizes possible vibrations within the system. Unfortunately, in the current installation at RIT, there is a basement beneath the test cell where the compressor was installed, so hard-mounting the compressor was never a viable option.

Team P11452 designed their installation around elastomeric lattice mounts produced by LORD to minimize the vibrations the floor would see during operation (in the z-direction of an x-, y-, z-coordinate system). An unexpected side effect is that the compressor "floats" on these blocks, causing an unwanted amount of vibration in the horizontal plane (the x-direction, in line with the piston). The primary task of this project is to eliminate as much of this horizontal vibration as possible, with a baseline goal of fifty percent reduction.

The secondary task for this project involved implementing a novel cooling system to supplement the current pump-driven system. The cooling system to be used is referred to as a thermosyphoning system, and is used by D-R on many of their compressors. These thermosyphoning systems use the natural convective properties of their coolant liquid (in this case, water) to circulate through and cool the system.

Thermosyphoning systems are more desirable than a traditional pump-based system for use on these compressors due to the fact that they are closed-loop, self-driven systems that require no external power inputs to perform their cooling tasks. The primary component of these systems is a section (or multiple sections) of finned tubing, which acts as a heat exchanger with the environment around the system. The compressor causes the coolant liquid to heat up due to the higher temperatures created from gas compression, and the finned tubing cools the coolant liquid down towards the surrounding ambient air temperature. This temperature differential causes the liquid to circulate in an effort to balance the temperature distribution. An additional challenge for implementing a thermosyphoning system onto the D-R compressor is that the customers wanted to keep the original pump-based system intact and fully operational, requiring minimal changeover time and labor. The two primary goals for this portion of the project were to keep the systems easily interchangeable, as well as to design the thermosyphoning system to perform equally as well or better than the current cooling system.

An essential tool utilized throughout the project was the data acquisition system implemented by Team P12453. This system added temperature, flow rate, and vibration data acquisition sensors (amongst others) to allow for the full characterization the compressor's vital signs as it runs. Accelerometer data can be collected and analyzed to examine the vibration characteristics of the system, while thermocouple and flow meter data can be used to analyze and characterize the coolant system of the compressor.

PROCESS/METHODOLOGY

Vibration Reduction Concept Selection

The Pugh Concept Selection Method was used to select the vibration reduction approach we pursued. Shown in Figures 2 and 3 are the matrix of brainstormed concepts and the resulting evaluation.

	Current Installation	Car Strut	MR	Hanging	Inertial Shock	Another Recip	Weighted Motor	Magnets	New Lattice Mounts	Solid Attach	Pool of Pitch
Power Consumption	s	s	-	s	s	-	-	-	s	s	s
Adjustability	s	+	+	s	s	s	+	+	s	s	s
Price	+	+	+	+	+	-	+	-	s	+	-
Time Line	+	+	s	s	s	-	-	-	s	+	s
Installation	+	+	+	-	+	-	+	+	s	s	s
Ease to Remove	s	+	+	-	s	-	+	s	s	s	-
Safety	s	+	+	s	-	s	-	-	s	s	s
Floor Transmissibility	-	s	s	+	s	-	+	-	s	-	s
Area needed	s	-	-	-	-	-	-	-	s	s	-
Ease to Mount	s	+	+	s	+	s	+	+	s	s	s
Maintenance	s	+	+	s	+	+	+	+	s	s	S
Complexity	s	+	+	-	+	+	-	+	s	+	-
Speed to response	s	s	s	s	s	s	s	s	s	s	s
Number of +'s	3	9	8	2	5	2	7	5	0	3	0
Number of -'s	1	1	2	4	2	7	5	6	0	1	4
Number of "Same"s	9	3	3	7	6	4	1	2	13	9	9

Figure 2: Pugh Analysis For Vibration System

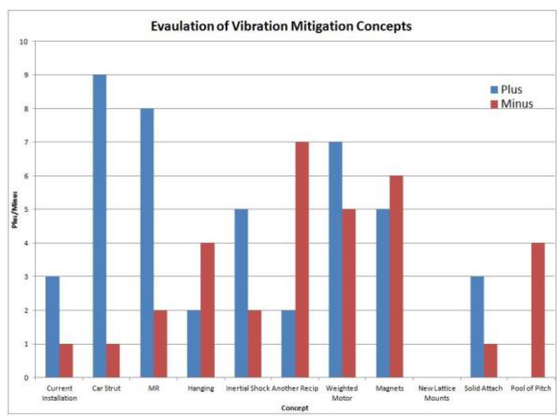
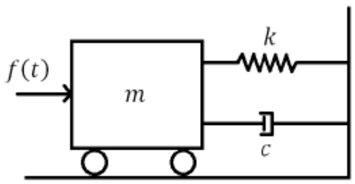


Figure 3: Pugh Evaluation

Based on the above data the Team decided to pursue both the car damper and the MR damper approaches. This imposed a unique set of design and manufacturing challenges which greatly increased the complexity and lead time of this portion of the project.

Vibration Damping Design

A critical design requirement for the vibration damping system was that it be extremely overdesigned, for safety reasons. As a result, excessive factors of safety were implemented in every aspect of the design. As such, the driving characteristics of the design were durability, strength, and low cost. Another requirement was that the system be interchangeable between MR dampers and automotive dampers, a requirement that was relatively easily met due to the similarity in form factor between the MR and automotive systems. In an effort to keep costs down, the mounting system was manufactured without outside assistance, and is not optimized to minimize material, but instead to minimize machine time. Furthermore, the raw material purchased was intentionally sized as close to the final dimensions as possible in order to reduce machine time. The size and capability of the machine shops on campus also limited designs; due to physical space available on the mills, the design was 1" shorter than designed. That said, a benefit of this simplistic design was that economical A-36 steel could be used, which helped drive costs down further.



$$m\ddot{x} + c\dot{x} + kx = f(t)$$

Figure 4: Spring-Mass-Damper Model of Compressor System

Extensive FEA analysis was performed to ensure that the system would meet the design requirements and remain safe for students to be around over the entire lifespan of the compressor. These FEA analyses were double-checked against hand calculations when possible in order to verify the results, with good overall good correlation. FEA also simplified the design process, lowering the turn-around time between a design and the verification of that design. Since the solid modeling and FEA programs were all part of the same software package (SolidWorks), prints for manufacturing were easily produced, but, some alterations made to the design after computational analysis was completed resulted in problems later on during testing of the system.

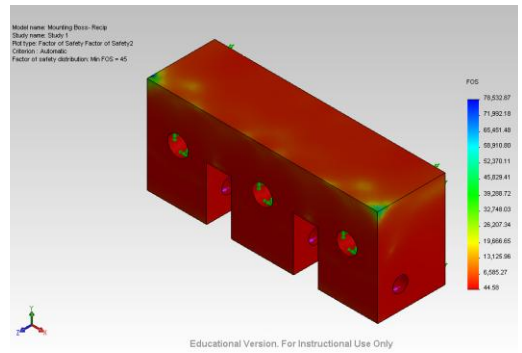
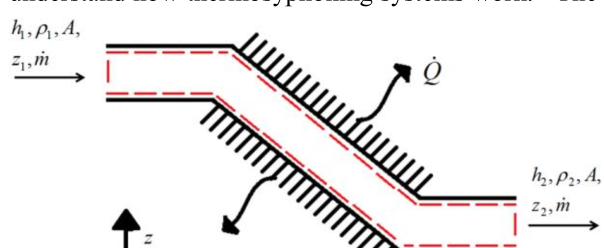


Figure 5: Compressor Mounts FEA

Thermosyphoning Concept Selection and Design

For the thermosyphoning system there was no traditional concept selection phase comparable to what occurred for the vibration isolation system. Due to the fact that this system is one that D-R already uses on some of their existing commercially-available compressors, the intent of this portion of the project was to create a rig to test and better understand how thermosyphoning systems work. The intent was to provide D-R with a test bench and a tool that they could use to further their knowledge on this topic.



$$\dot{Q} = \dot{m}\left[(h_2-h_1)+\left(\frac{\dot{m}^2}{2\rho_2^2A^2}-\frac{\dot{m}^2}{2\rho_1^2A^2}\right)-gz_1+32\frac{\bar{u}}{\bar{\rho}}\frac{L}{D^2}\frac{\dot{m}}{\bar{\rho}A}+\zeta_T\frac{\dot{m}^2}{2\bar{\rho}A^2}\right]$$

Figure 6: Control Volume for Thermosyphoning Analysis

First, the feasibility of installing a thermosyphoning cooling system onto the RIT compressor was examined. This was done by testing the compressor as is, and seeing how much heat the current system removes from the compressor. Using that data, a Microsoft Excel sheet was created that calculated the performance of both the proposed systems, and any possible future system of this type. This tool drove the design process to help find the basic dimensions and characteristics of the system so that the thermosyphoning system would, in theory, perform at least as well as the pump-driven system. Next, a virtual layout of the compressor and cooling system was created in SolidWorks, which served as a tools to assist with ordering material and fittings for the system, as well as determining pipe cut lengths.

The final design of the new system allowed for the compressor to be changed over from forced cooling to thermosyphoning by moving three valves. The old system remains entirely intact, and could replace the new system in the event of a major problem. Due to the analysis requirements for the thermosyphoning system, two thermocouples and a flow meter were installed in the thermosyphoning system. Due to the vibration of the system and for ease of manufacturing, it was decided that the coolant pipe would be welded rather than threaded. Exact lengths were not imperative, as they would have been had all connections been threaded.

INSTALLATION

Vibration Isolation

The installation of the compressor mounting blocks into the floor of the compressor room was accomplished utilizing Hilti concrete epoxy and anchors, which were chosen due to the extremely comprehensive data they make available to assist engineers. Using the Hilti system, holes were drilled in the reinforced concrete floor of the test cell to the appropriate depth, and threaded studs were then sunk into these holes and epoxied in place. The analyses performed on the mounting systems all showed an extremely high factor of safety, (in excess of 20) which was necessary to meet customer requirements. The mounting plates slide onto the floor studs, and nuts and washers were then tightened down to a Hilti-provided specification of 65 ft-lbs. of torque. Once these plates were mounted, the mounting blocks were then bolted into place. Two mounting locations were created at the front and rear of each plate, for the MR and automotive systems respectively. The dampers are held in place with M12 bolts at each end. For the automotive dampers, this is the final step. The MR dampers require a DC power supply capable of providing at least 4A, (1A per damper) which controls the damping coefficient of each damper.

Thermosyphoning

Fabrication was done with the system attached to the compressor. Rob Kraynik, one of the shop supervisors for the Mechanical Engineering department at RIT, tacked the piping while it was connected to its final position to make sure that the flange orientation was consistent and the pipe did not rotate or displace to an incorrect alignment. This saved time during the final installation, since the new plumbing fit comfortably with little adjustment required to fit the system into place.

The finned tubing section was also fabricated and installed while on the machine for the same reasons as above. The finned tubing sections were cut iteratively to ensure that the fit was correct to match both the bottom and top flanges. Steel sleeves were created to slip inside the copper finned tubing, and welded to the steel flanges. Due to this interface being between copper and steel, a two-part epoxy was used to make sure that it was a strong, permanent, water-tight connection, since welding was not an option. The finned tubing section is easily removable by simply unbolting the flanges at the top and bottom of the section, which allows for configurations to be tested easily.

DATA COLLECTION

Vibration System Data Collection Methodology

A point on the edge of the skid, located on the far end of the compressor skid equidistant between the dampers, was chosen as a data collection point. This point was chosen due to its proximity to the dampers, as well as its central location – ideal for the collection of baseline and comparative results. A tri-axis accelerometer was placed at this point to collect acceleration data in all three primary axes. The compressor was the turned on, and run until it reached steady state. At this point, acceleration data was taken. This methodology provided for a simple, consistent, and easily reproducible method for measuring the vibration of the compressor under different conditions and at different times.

Thermosyphoning

For the thermosyphoning system testing, the compressor was run in the original configuration using the pump-driven coolant system, and coolant temperature in, temperature out and the flow rate of the cooling system was recorded. From this, the heat rate removed from the compressor by the forced-cooling system was calculated. After the thermosyphoning system was manufactured and installed, a similar test was run to ensure the new system did not have any unforeseen flow restrictions or leaks. Next, the compressor was run solely using the thermosyphoning system, while collecting the same data as above, as well as the temperature into the finned-tubing, out of the finned-tubing, and the flow rate of the thermosyphoning system.

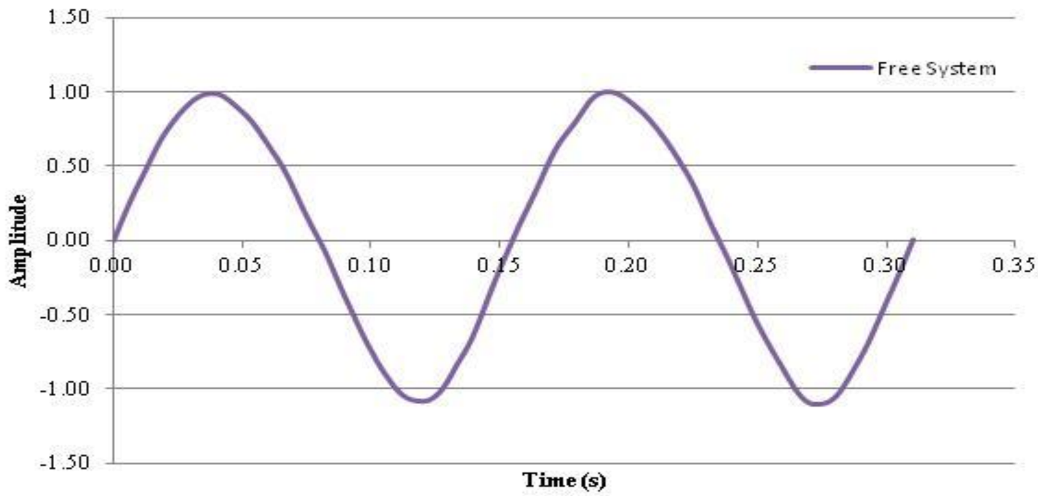


Figure 7. Normalized acceleration amplitude versus time for the initial, undamped system.

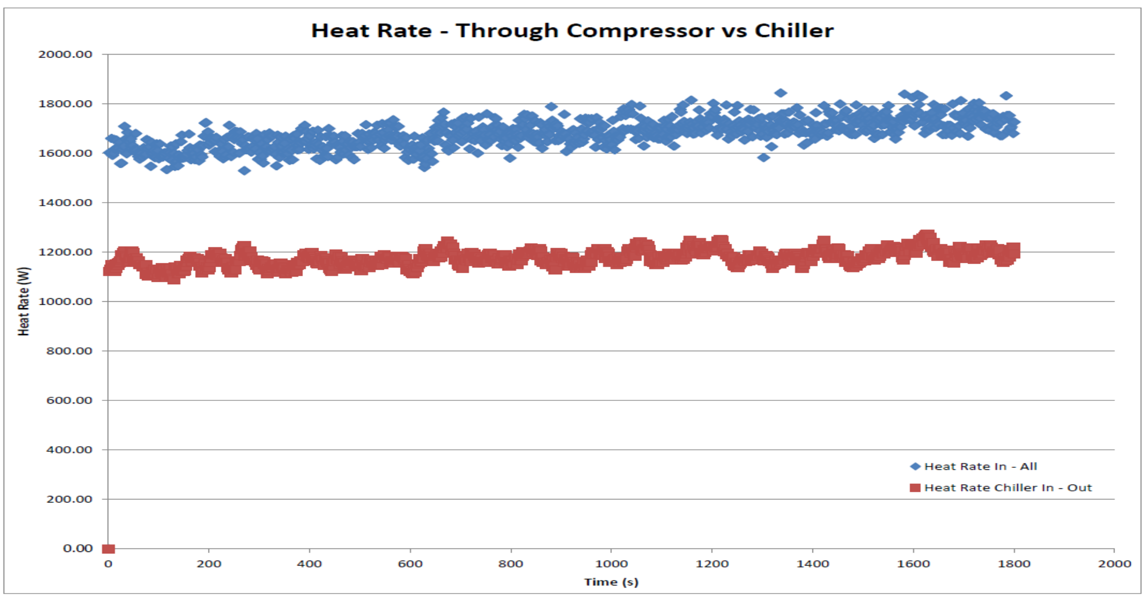


Figure 8: Initial Heat Rate Data Through Compressor

Pre-Installation Data

Vibration System

Using the planned data collection methodology, baseline data of the compressor's vibration with no added damping was taken for reference and comparison. This data was normalized so the peak amplitude of the vibration in the x-direction was equal to 1. This normalization would allow for easy comparison – 50% reduction would result in an amplitude of 0.5, or 50% amplification would result in an amplitude of 1.5. Figure 7 is a plot of the collected data for the acceleration of the system.

Thermosyphoning

To obtain initial data for the thermosyphoning system, the compressor was run at full load for approximately two hours using the forced cooling system. Temperature and flow data was collected and the heat rate through the current system calculated. This data can be seen in Figure 8. At steady state, the pump-driven cooling system has a flow of approximately 8.4 gallons per minute (gpm).

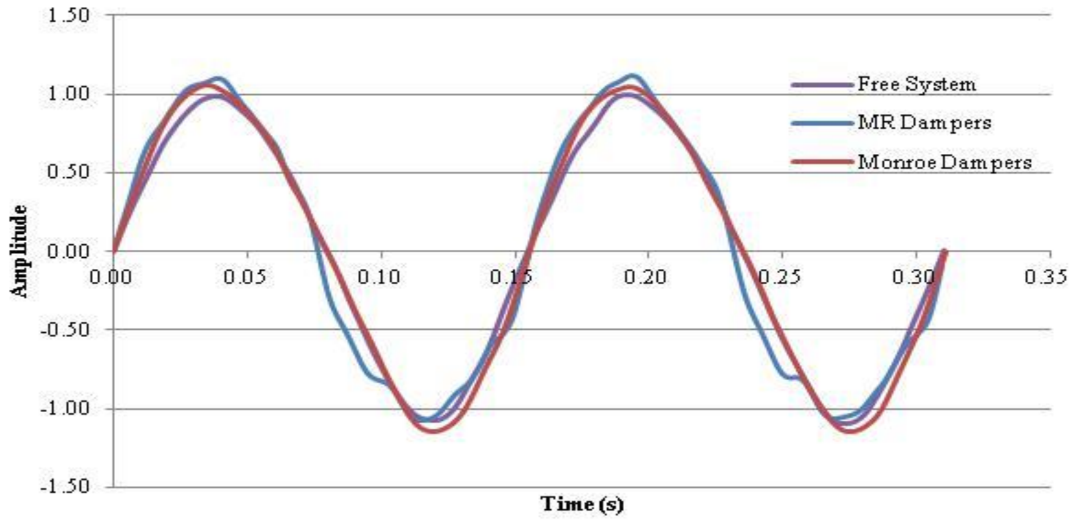


Figure 9: Normalized Acceleration Comparison of the Undamped System, MR Dampers, and Automotive Dampers

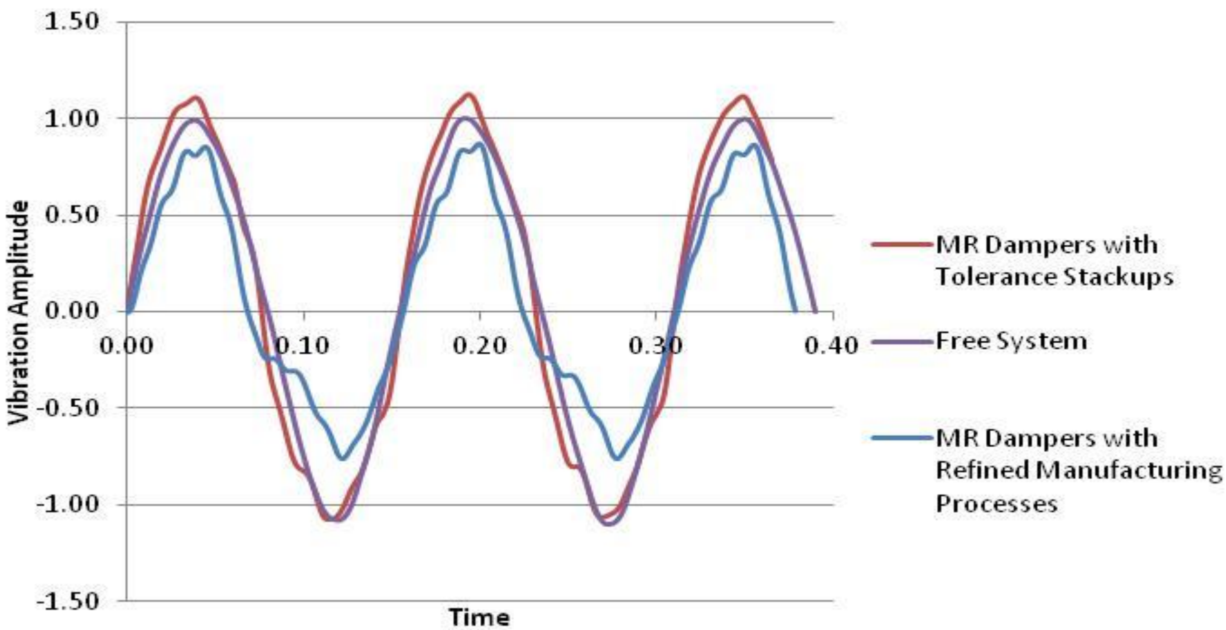


Figure 10: Normalized Acceleration Comparison of Undamped System, MR Dampers with tolerance stackups, and revised MR Damping System

Post-Installation Data

Vibration System

Data was collected using the same methodology as above for both implemented systems, the MR dampers and automotive dampers, the ensure that it could be compared against itself. This data was normalized using the same constant as the data from the system with no added damping. Figure 9, above, shows the direct comparison of the acceleration in the x-direction for each system, normalized to the free system.

Unfortunately, due to manufacturing tolerance stack-ups, and an unforeseen design flaw, there was an excessive amount of slop in the MR system as it was originally installed. As result, the MR dampers actually amplify the vibration in the X direction while installed with the faulty components. This led to a rapid redesign and remanufacturing phase of the project to address the issues that were discovered as a result of this test, which in turn led to the results above, Figure 10.

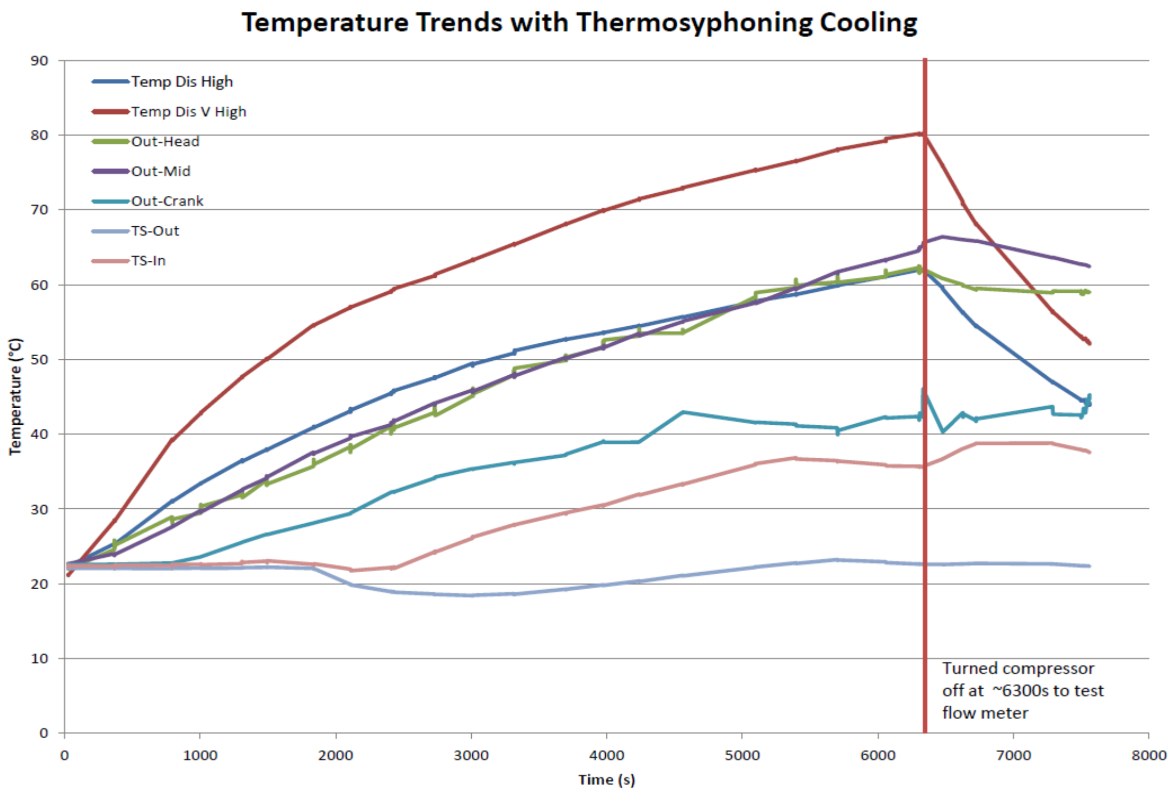


Figure 11: Temperature Trend Data for Compressor with Thermosyphoning Cooling System

As the plot above shows, once the MR dampers were properly installed, the damping improved considerably, resulting in a net damping of the vibration by about 23%. The causation of this problem and the details of the redesign are discussed below in the "Discussion" section.

Thermosyphoning

Once the thermosyphoning system was installed and operational, the system was run for the same amount of time as in the pre-installation data collection. The temperature data from all of the thermocouples was collected, which can be seen in Figure 11. The maximum temperature of the system reached over the course of testing was approximately 80 degrees Celsius.

A flow measurement was unable to be obtained while the thermosyphoning system was in use. The flow meter sources and installed for this system is able to read flows down to 0.13 gpm, so this was confusing given that initial design calculations predicted a flow of approximately 0.3 gpm. Initially, it was thought that the vibration of the system was causing impairment of the flow meter, so at approximately 6,300 seconds, the compressor was turned off to see if the flow meter would start measuring a flow without vibration present. This should have produced a flow if vibration was the problem, since the system was still hot and would continue to circulate and cool until ambient temperatures were reached. Unfortunately, this yielded no measured flow rate, leaning to the conclusion that the flow may be lower than the measurable 0.13 gpm limit of the flow meter. An additional test to check the calibration of the flow meter was conducted to ensure the meter's effectiveness by running a set amount of fluid in a set amount of time through the flow meter, which compared within 1.25% of the expected value. This secondary check ensured the flow meter's proper operation, and indicated that the lack of a flow reading from the thermosyphoning system was a problem elsewhere in the system or initial model.

DISCUSSION

Vibration System

As the vibration system was originally designed, there were two damping methods- automotive dampers, and the MR dampers supplied by Lord Corp. Once the first revision of the system was complete, the automotive dampers were tested, and found to have virtually no impact on the motion of the system, as the reader can see in Figure 9. When the system was switched to the MR dampers from the automotive dampers, a number of problems became apparent. First, the pocket size necessary to fit the automotive dampers were much, much larger than the width of the MR dampers, resulting in a large bending moment on the mounting bolts (see Figure 12). Furthermore, due to a tolerance stackup, and a difference between the print for the MR dampers and the dampers we received from Lord, there was excessive play in the mounting bolts (Figure 12). This resulted in excessive and unacceptable motion. As a result, the authors executed a rapid redesign that tightened up the tolerance stackups, and narrowed the pockets where the MR dampers mount to reduce the bending moment on the mounting bolts. The revised data can be seen in Figure 10.

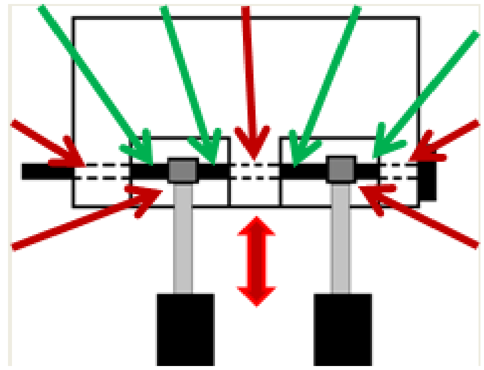


Figure 12: Original MR Damping System: Green Arrows Indicate Bending, Red Arrows Indicate Tolerance Stackup Locations

Unfortunately, this reduction in the X direction resulted in a slight increase in the pitching moment of the system, which was not foreseen by the authors' analytical model. This could be reduced by mirroring the installation onto the other side of the compressor skid, but this solution is not within the time or budgetary constraints of our system. This is a good example of possible future work.

Thermosyphoning System

After running the thermosyphoning cooling system for about one hour and forty five minutes the flow meter did not register any flow. However, from looking at the discharge air temperature inside of the discharge tank, it is evident that there is cooling occurring. Examining the above plot, the thermosyphoning system is on par with the forced cooling system. Currently, the authors believe that that the initial model was too generous with the assumptions, and the actual flow is below the lower limit of the flow meter, which explains the lack of flow across the flow meter.

The flow issue is something that can be looked into for future projects. The main reason why this flow meter was selected is for its price. A flow meter can be purchased that has the capability to measure the hypothesized lower flow. This will allow for a better analysis of the thermosyphoning system to be performed.

CONCLUDING REMARKS

Working on this project was very rewarding for the team's future engineering endeavors. There are multiple areas that future teams can work on, including the vibration system, the cooling system, and many others. In regards to the vibration system, future groups can work on the vibration transmission to the floor. The thermosyphoning system has the potential for teams to look into different finned-tubing arrangements and investigate the flow properties of the system. The overall compressor test cell also provides endless areas of graduate work, such as current thesis projects focusing on valve failure and bearing.

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