Final Report

Team 5

Motor Test Rig

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Acknowledgements

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Abstract

Danfoss Turbocor is the sponsor for team 5, and they are a lead manufacturer of compressors. They primarily deal with HVAC applications, and their company is unique because of their use of oil-free centrifugal compressors. They are able to utilize oil-free coolant because their compressors run with magnetic bearings that keep the shaft levitating while it runs, hence no contact on the shaft itself. Turbocor approached FSU last year to design a motor test rig, treating one compressor as a motor and the other as a generator. Their goal is to test and see what happens when external components are attached to their compressors as a research and development project. Last year's design team was able to run up to 700 rpm before the system shut down because of vibrations, and they were not able to test it on their adjustable frame. Utilizing last year's frame, team 5 was able to increase the rpm to above 700 easily, and plans to increase it and break the milestone of 1,000 rpm. They were unable to reach higher than 726 rpm because of a limited power supply, but their design held very well and did not struggle to maintain that speed at all.

1.0 Introduction

Danfoss Turbocor is a leader in compression technology, and is interested in running tests on their compressors to observe how they act when the speed increases. The company designs compressors for heat, vacuum, and air conditioner industry. Normally Turbocor runs their magnetic bearing compressors with internal shafts (inside of the compressor). Turbocor achieves a high efficiency due to a combination of magnetic bearings, which uses magnetic fields to create a contact free system between the shaft and bearings allowing high speeds (up to 40,000 RPM), and variable-speed centrifugal compression, which allows the use of the compressor with the rotation for the highest quality performance. The compressors have 9 sensors that re-align the compressors if they are pushed ever so slightly out of place. Therefore it is crucial to maintain alignment and reduce vibrations within the test system.

Danfoss Turbocor is still looking for a way to be able to test their compressor models at very high speeds along with maintaining the speed for longer periods of time. This year's team is currently building off of what last year's senior design team produced, which last year's team went off the image shown below that was Turbocor's original project scope in 2009.

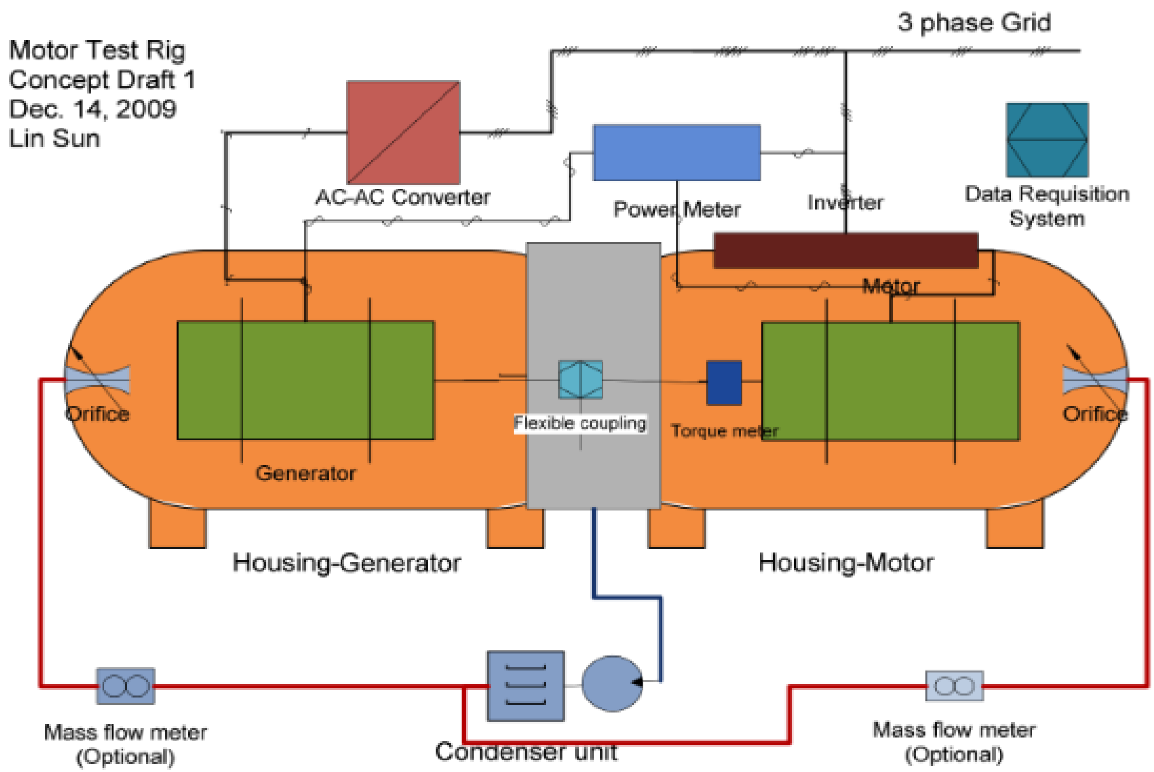


Figure 1 Initial Concept

The image shown below gives a gist of how a motor test rig works. The image shows that one compressor is the motor, while one acts as the generator. Therefore, one is the driver and one is driven. A torque meter will be placed in the middle to read the power efficiency of the compressors. Turbocor wants a similar design of a motor test rig that will be able to test any of the multiple compressors that they design and manufacture.

The senior design team that was tasked with this project last year was able to create a design that ran at low speeds. Last year's team designed an adjustable frame that was able to change the position of two compressors placed on it, align them, and then run compressors at low speeds. They designed a system with a one flexible coupler and two rigid couplers, which accounted for a certain amount of misalignment, but also caused problems later on in their project. The team was never able to implement a torque transducer to actually calculate the efficiency of Turbocor's compressors.

Team 4 ended up buying a dial alignment system to align the shafts with the compressors. Since a lot of human error occurs in a dial alignment system, this may have been a cause to some vibration issues that occurred when testing the design, therefore this year's team planned on swapping out the dial system for something more accurate. Although last year's team were able to stay levitated for a short amount of time, the vibration was bad enough that it caused the compressors to completely shut down. Last year's team improvised and added duct tape in order to dampen the vibration. This was done in order to be able to attempt to operate the machine. With the help of Julio Lopez (Turbocor's quality assurance expert), the test rig was able to run up to about 700 rpm before the vibrations caused the system to shut down. Julio thinks this may have stemmed from the compressor sensors fighting each other to realign, which caused the compressors to metaphorically be playing tug of war until the axial displacement was too large and the compressors shut down. This year's team aimed to create a system that would prevent the compressors from fighting each other. The figure below is the CAD drawing of the final design that last year's team ended up creating.

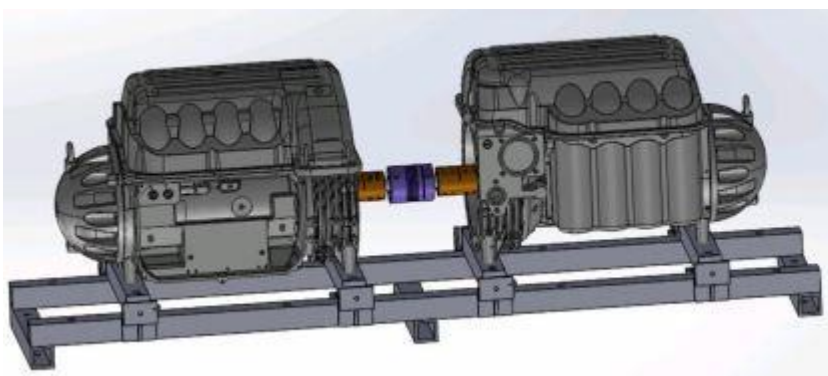


Figure 2 Last Year Design

The overall goal for the project this year is to be able to have the high speed motor test rig up and running up to about 10,000 rpm. Team 5's objectives are somewhat straightforward due to the achievements that last year's senior design team accomplished, as well as the failures they faced and recommendations they have passed on. Team 5 most important objectives for this year was having consistent levitation to allow the compressors to operate properly. Following this would be having accuracy in regards to the alignment and balancing in the test rig, then the desired durability and safety will be achieved as well. A huge thing that also wanted to be implemented was simplifying the operation process of the motor test rig.

The safety of team 5 and those who will be operating and maintaining the system is a priority. In order to achieve this safe system, team 5 will be doing different analyses when improving on or making changes to the last year's design. One way to do this is by producing an FMEA on parts that could potentially be bought and added to the test rig. An FMEA, Failure Mode and Effect Analysis, is a step by step approach for identifying all possible failures in a design, a manufacturing or assembly process, or a product or service . By doing an FMEA on every part that is considered for the design will show the likelihood of failures that can possibly occur. It also takes into account all the specifications in each part, therefore showing which products will work best with the design and project objectives. Team 5 members are aware of the dangers during the test rig assembling that could occur. This risks can occur from improper use of tools and also the weight of the compressors and other test rig components. When the test rig is in operation, safety shielding will be put in place to stop any potential projectiles that could be flung from the system. For any high-speed system, a loose part can cause serious injury or death. Therefore team 5 take this very seriously and will not only double check all of our connections, but have multiple supervisors from Turbocor present during testing.

Team 5 believes that by implementing a more flexible coupling that attaches to the shaft extruding out of the impeller, mounting a torque transducer in the middle of the compressors, and using more efficient equipment that the objectives of the project will be met.

2.0 Project Definition

Danfoss Turbocor is a leader in compression technology, but does not have a system to analyze the performance of their compressors when they run at high speeds. Turbocor achieves high efficiency in their compressors through a combination of magnetic bearings, which use magnetic fields to create a contact-free system between the shaft and bearings, allowing high speeds (up to 40,000 rpm), and variable-speed centrifugal compression, which allows the use of the compressor at the rotation required for the highest quality performance. Danfoss Turbocor requires a motor test rig that is capable of measuring efficiency and torque loads more accurately while utilizing little power consumption. The senior design team that was tasked with this project last year was able to create a design that ran at low speeds. Last year's team designed an adjustable frame that allowed the ability to change the position of the two compressors placed on it during the process of alignment using shims, screw sets, and a crowbar to lift and move the compressors.

The team then measured the alignment with a dial system, which contained a tremendous amount of human error. They used two rigid couplers that connected to the shafts extruding out of the impeller of the compressor then connected to a shaft that went into a heavy flexible coupler, which accounted for a certain amount of misalignment, which also may have caused oscillating vibrational problems later on in their project. Although Turbocor's compressors have rotation speeds that range from 13,000 rpm to 40,000 rpm, for safety reasons, this year's Team 5 has been tasked with designing a test rig that can withstand rotations of up to 10,000 rpm. Due to lead time and cost, instead of incorporating a torque transducer in the test rig, the team was asked to design a mock transducer to prove the validity of the proposed design theory. After analyzing last year's design three main problems arose: misalignment, complexity of design and set up, the compressors fighting each other. Improvements were expected to be made from the design that was presented last year; this includes adjusting the vibrational issues by using a different coupler, incorporating a fixed mock torque transducer that's mounted to the rig itself to prevent the compressors from fighting each other, and trying to fix the alignment issues that occurred last year through the use of a laser alignment tool. Last year's motor test rig operation process was also simplified by incorporating a car jack to help move the heavy compressors while aligning the system. Safety always remains top priority, therefore to prevent any casualties a safety shield was designed to fit over the critical components of the motor test rig.

3.0 Concept Generation

3.1 Design Requirements

This high speed motor test rig requires attention to four main aspects in order to successfully test the two compressors. These aspects are the magnetic bearings (a main feature of Danfoss Turbocor compressors), shaft misalignment, the alignment system of the test rig, and the torque monitoring process. It was decided by Danfoss Turbocor that the individual components are to be qualified to operate above 10,000 rpm and 64 Nm in torque. Danfoss Turbocor chose these requirements as compared to last year (50,000 rpm and 100 Nm of torque) in order to see if the method that the team chooses to run the motor test rig is successful. Upon achieving the goal, the project can then be moved to the next stage which was the original goal of team 4 last year. The benefit that Team 5 had going for them this year is the fact that the project is a continuation from last year. The team last year was able to make a base frame that the two compressors would sit on. The rotating assembly that connected to the two compressors together consisted of three couplers and two shafts. Two of the coupler were rigid couplers and a bellow flexible coupler that would be connected via two shafts.

3.1.1 Magnetic Bearings

The shaft in the compressors are levitated using magnetic bearings that are inside. There are nine sensors located in each compressor that will allow for adjustments to be made on the shaft. This is important to note because whenever there is a radial force of 200 Nm or more is exerted upon the compressor shaft, the sensors will make the system go into a lockdown mode. This mode will deactivate the motor and magnetic bearings, essentially shutting down the compressor. This will seize the shaft rotation as well if the compressors are running, if the components that are rotating are not properly balanced severe vibrations will be exerted on the shaft effectively activating the lockdown mode of the compressors and shutting them down. In order for this scenario to not occur, strict precision and accuracy for balancing must be considered for all that components that will be used in rotation.

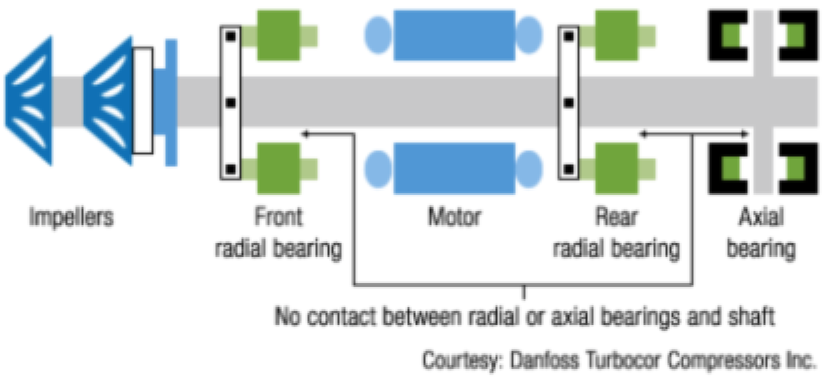


Figure 3 Schematic Magnetic Bearing System

3.2 Shaft Misalignment

A key factor to have the motor test rig up and running is dealing with the shaft misalignment. The figure below shows the possible orientations in which misalignment is to occur. Angular misalignment occurs when the axis of the shafts connect through intersection, but they are not parallel represented by the letters A and B. Parallel misalignment occurs when the axis of the shafts are parallel, but don't connect, the letter Y represents the misalignment. Axial misalignment occurs when the distance between the ends of the shafts changes during operation, represented by the letter X.

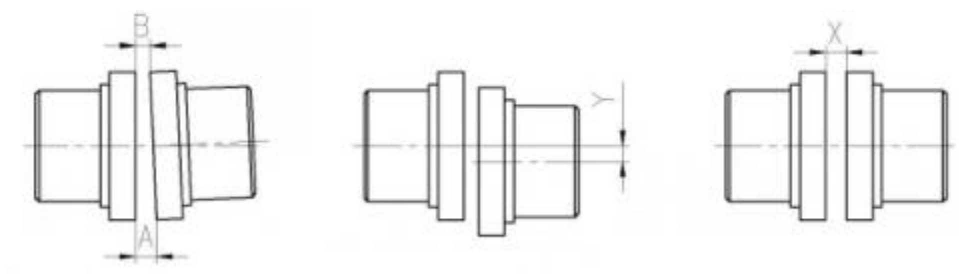


Figure 4 Shaft Misalignment Scenarios

Team 4 last year assumed that only assumed that angular and parallel misalignment ould played a role and axial would be minimal. They tackled the shaft misalignment by having a total of three couplers in the rotating portion of the motor test rig. They went with two rigid couplers and a bellow flexible coupler. An image of the assembled rotated assembly is seen in the figure below. Though team 4 didn't think axial misalignment played a role, it in fact had the biggest role that would impact the operation of the motor test rig. The sponsor mentioned to team 5 that since the impeller was removed, there had to be axial misalignment to be accounted for. Specifically 0.250 mm from each compressor for a total of 0.500 mm of axial misalignment before the motor test rig can be operated. The rigid coupler presented a problem with that because although it held the rotating assembly in terms of parallel and angular misalignment, there was little to no adjustment that it could have for axial misalignment. This may have been one of the reasons why team 4 last year dealt with vibrations when they started run the compressors.

Figure 5 Last Year's Rotating Assembly

Team 5 has decided to change the rotating assembly completely in order to tackle the axial misalignment issue. The amount of couplers was reduced from three to two, with both of the couplers being flexible couplers that attached to the compressors themselves. In the middle there will be a torque transducer that will act as a rigid component in the middle that will further reduce the misalignment in all three orientations.

3.2.1 Alignment System

While the flexible couplers can compensate for some of the misalignment, the overall alignment system of the motor test rig will need to accuracy and ensure that any misalignment that will occur is within the tolerance range that is determined by the two flexible couplers. To get more precise and accurate will be determined by the tool that is used to measure the angular and parallel misalignment of the shaft.

In addition to an alignment system that will be incorporated into the motor test rig, methods that are used to adjust the compressor in multiple directions will be implemented as well. In order to make a shaft angle change, it is required that a process which adjusts the elevation of the compressors in the four corners in the mounting locations. There will also be a need for a horizontal adjustment in order to move the compressors laterally.

3.2.2 Torque Monitoring

The motor test rig will be monitoring the efficiency during operation, it's required that a device be integrated in order to monitor the output that is being done by the motor driving the system. Team 5 has decided to go with a torque transducer to be implemented in order to measure the outside torque. The transducer should be able to operate up to the stated goal of 40,000 rpm and 100 Nm. Below is a list of the various compressors that Turbocor produces that would be used on the motor test rig.

Table 1 Rated Torque and RPM of TT Series Compressors Models

Compressor	Shaft Torque [Nm] (Max)	Speed [rpm]
TT300	22.8	37762
TT350	38.0	30598
TT400	37.2	25091
TT700 (Four Poles)	73	17000

3.3 Design Concept Selection

Based on the requirements that were presented from the sponsor, team 5 came up with 2 main design concepts for the test rig. The first one, which is considered the ideal design, incorporates all the requirements from the sponsor but unfortunately due to lead time constraints it had to be modified during the spring semester. The second design, the final design, is defined as the first phase of the ideal design and has for the most part all of the requirements except for the actual analysis that would occur while the compressors are operating. The final design will allow for further integration of all components that were originally required to fully operate any of the TT series compressors that Danfoss Turbocor produces.

3.4 Ideal Design

Team 4 from last year went with a dial alignment system in order to measure the misalignment between the motor shafts. The dial indicator system was cost effective and was purchased by the team for under $100 with the accuracy being up to 26 um. Dial alignment systems have a big learning curve. The operator must use his technical knowledge in using measurement tools in order to interpret the dial readings into compressor position adjustments. An image of one of the dial indicators is shown in the figure below.

Figure 6 Dial Indicator

Team 5 this year had decided to go with a laser alignment tool based on the recommendation left behind from last year's team. The laser alignment system that was selected by the team was the SKF TKSA 31 alignment tool. It has an accuracy of $\pm$5um. The total price for this comes in at $3847.01. An image of the system can be seen in the figure below.

Figure 7 Laser Alignment Tool

As stated previously, to monitor the performance of the motor that is driving the other compressor, it is necessary to incorporate a torque transducer. It must be able to handle speeds of up to 40,000 rpm and 100 Nm of torque. These values were chosen to over design the system to have a good factor of safety.

With further research into different types of torque transducers, a choice was made. Out of the ones that were researched, none really matched the required parameters. The transducers could match one of the parameters but not the other. Team 5 decided to go with a torque transducer that would have a high rpm limit. The transducer that was selected was the TMHS 310 that is produced by Magtrol. The specifications that it can reach was 32,000 rpm while having a nominal torque rating of 20 Nm. Though the rpm was lower than 40,000 rpm, the sponsor said that the transducer was fine choice. When the team contacted a sales representative from SKF, the team was assured that the TMHS 310 would be able to handle up to 100 Nm in range while it was in operation. An image of the transducer can be seen below.

Figure 8 Magtrol TMHS 310 Torque Transducer

The next step for the team was design a stand that would be attached to the torque transducer in order to act as a rigid point. Unfortunately, due the lead time of the transducer along with the cost of it, the sponsor advised the team to go into a different direction that would still validate the theory of having a rigid component in the middle of the rotating assembly.

In terms of the base frame that would be needed for the two compressor to stand on, the team decided to go ahead and keep the one that was created from team 4 last year. In terms of horizontal and vertical adjustments, the team went with two methods. Shims were used to adjust the compressor vertically. In the horizontal adjustment, lateral screw sets were used in order to move the driving compressor.

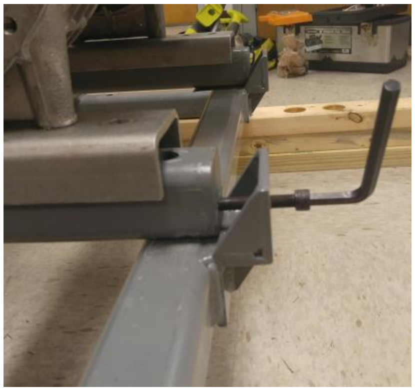


Figure 9 Horizontal Adjustment

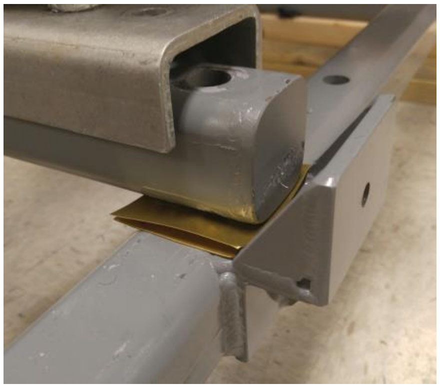


Figure 10 Vertical Adjustment

4.0 Final Design

4.1 Base Frame

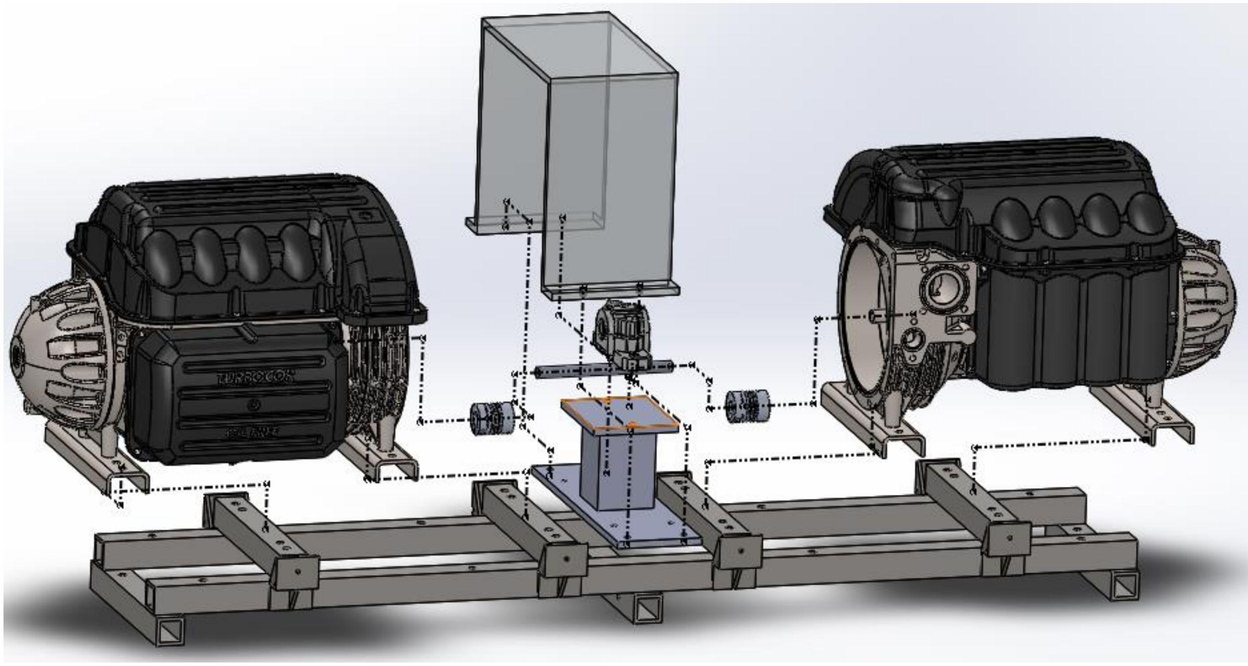


Figure 11 New Design with Base Frame

Last year's design of the base frame that the actual compressors will be sitting on worked, and therefore team 5 decided not to change it. Considering each compressors weighed about 300 pounds a strong material needed to be selected, therefore the frame was made out of steel. The FEA of the frame, located in the appendix, showed that there would be a maximum stress of 0.34 MPa and steel has a yield strength of 250 MPa, therefore there is no fear that the stand will fracture while the two compressors are sitting on top of it. The base frame is composed of two parallel long runners 2x2 inch square (boxed)steel (1/4 inches thick), four upper cross members also made of the same material, and three lower cross members for supporting purposes. The objective of the base frame is to support both compressors, the function is fulfilled basically with the upper cross members, and also allow the positioning of all other components needed to align the motors. The base frame dimensions can be visualized in the figure below.

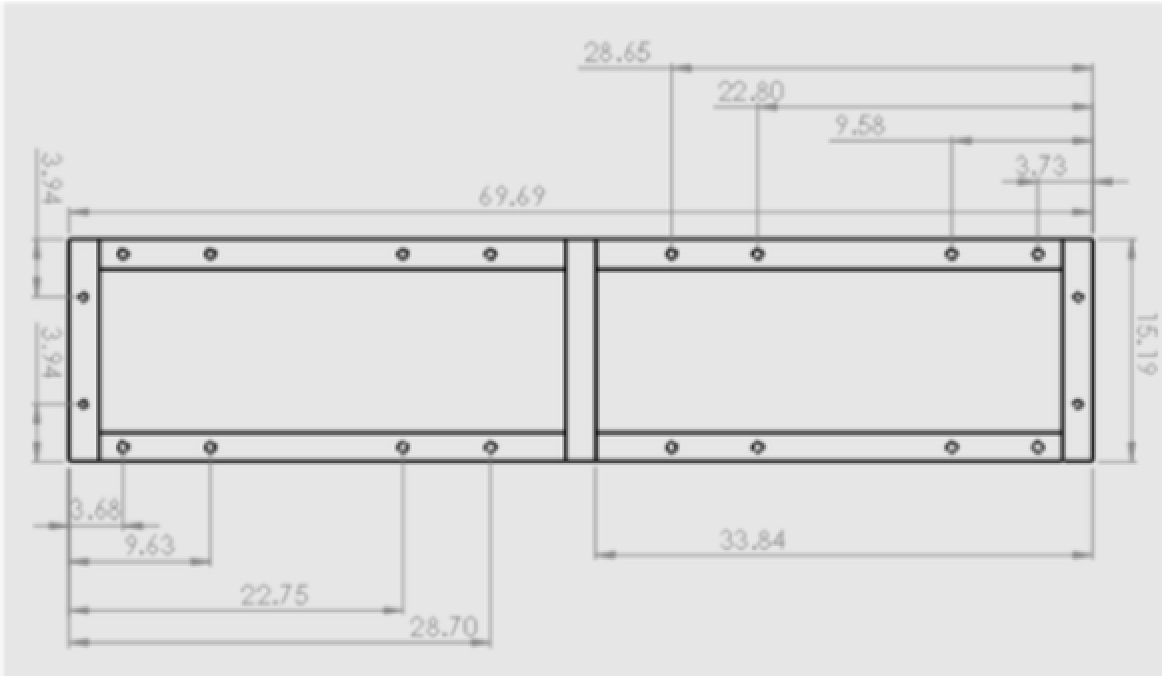


Figure 12 Base Frame Dimensions

Of the three lower cross members, the middle one is welded at mid length of the long runner, the other two lower cross members have two holes drilled in each and will be welded under the long runners at each end. The four upper cross members are going to be bolted to the frame with 1/2"-13 5.5" long cap screws. The supporting frame uses concrete fastening bolts to ensure their stability to the ground.

4.1.1 Double-flex disc coupling

After speaking with the sponsor multiple times and analyzing the FMEA of alternative couplers, team 5 came to the conclusion that the jaw couplings will not meet the requirements specified by the sponsor. The idea is to run the rig at 40,000 rpm eventually, and that needs to be kept in mind at all times. The jaw couplings might have been a solution for the lower speeds, specifically up to 3,500 rpm, but they do not operate well at the higher speeds. Thus team 5 further researched and decided on selecting the double-flex disc couplings that are made by Zero Max.

Figure 13 Zero Max Double Flex Disc Coupling

The various couplers from Zero Max offer many advantages, but the team decided to go the route of the double flex disc couplers that they produce. The couplings consist of two torsional stiff sections in combination with a unique patented disc that allows for a certain amount of misalignment and axial loads which is shown in figure 5. Out of the various options of double flex that are available from Zero Max, the team decided to go specifically with the Double Clamp A1C flex couplers. The specific model of the A1C coupler is the 6P18-A1C that can be seen in the figure below. Various factors were used in order to select this specific model. The main factor was the maximum speed that the coupler would be able to reach, which in this case is 15,000 rpm for the 6P18-A1C. The other factors that came into play was how much misalignment the coupler would allow while operating properly. The sponsor mentioned to the team that each coupler should be able to account for 0.250mm of misalignment in the axial direction for each compressor resulting in a total of 0.500mm of axial misalignment. The sponsor also mentioned that the coupler should be able to account for at least 1 degree of angular misalignment as well. Based on those factors, the 6P18-A1C was chosen because of the fact that it can account for 2 degrees of angular, 0.44mm of parallel, and 1.6mm of axial misalignment. The other models had a higher accountability for misalignment, but the one that was chosen had the highest maximum speed that be able to reach the goal that was set by the sponsor for the team this year.

Table 2 Double Flex Disc Coupler Models Information Part 1

Performance Information

Model	Continuous Torque	Peak Torque	Torsional Stiffness	Maximum Speed	Maximum Misalignments			Weight		Inertia	
					Angular	Parallel	Axial	Max Bore	Min Bore	Max Bore	Min Bore
	Nm	Nm	Nm/Rad	RPM	Degrees	mm	mm	kg	kg	10^{-3}kg-m^2	10^{-3}kg-m^2
6P18-A1C	20	40	5,500	15,000	2	0.44	1.6	0.25	0.30	0.30	0.11
6P22-A1C	30	60	8,482	13,500	2	0.58	1.8	0.39	0.47	0.22	0.24
6P26-A1C	53	106	9,712	11,500	2	0.55	2.2	0.54	0.65	0.41	0.43
6P30-A1C	90	180	20,923	9,500	2	0.85	2.6	0.97	1.14	1.00	1.10
6P37-A1C	181	362	32,700	7,900	2	1.00	3.6	2.03	2.43	3.17	3.31
6P45-A1C	282	564	60,324	6,700	2	1.24	4.6	3.7	4.6	8.50	9.00

- Consult factory for speeds higher than those listed and balancing requirements, if necessary.
- Consult factory for higher torque and higher torsional stiffness couplings.
- Available with or without keyway on clamp style hubs.

The couplers were custom bored out so that one end could attach onto a 22 mm shaft extruding out of the impeller of the compressor and a 20 mm shaft. This is within the range that the 6P18-A1C can have in terms of bore as seen in the figure below. The reason that the bore size of the coupler connecting to the shaft is 20mm is because the shaft that will connect the two couplers together is mimicking the diameter of the torque transducer that was originally selected by the team, but was ultimately rejected. The coupler was selected with an adjustable collar so that is has the ability to slide over the shafts and tighten to be securely fastened. Therefore this year, two 6P18-A1C were purchased at approximately $400 a piece from Zero Max to incorporate into the motor test rig.

Table 3 Double Flex Disc Coupler Models Information Part 2

Dimensional Information											
Model	A	B	C	D		E (bore)		F	G	H	L
				Bolt	Torque	Min	Max				
	mm	mm	mm	M	Nm	mm	mm	mm	mm	mm	mm
6P18-A1C	53	22.5	18	M6	13	8	26	20.1	7	18	63
6P22-A1C	62	26	23	M6	13	12	31	24.9	7	22	75
6P26-A1C	69.5	29.5	22	M8	32	14	35	25.4	9.14	24	81
6P30-A1C	82	32.5	34	M10	58	16	40	30.7	10	27.8	99
6P37-A1C	101	46	42	M12	100	18	51	38.4	12.7	36	134
6P45-A1C	123	60	48	M16	245	24	65	46	16.2	43.5	168

Another reason that the couplers were chosen from Zero Max was because of the capabilities further down the road. In talking with a sales representatives of Zero Max, they assured team 5 that they can make custom-made couplings that will be able to reach speeds of up to 40,000 rpm. They have made custom couplers before for similar projects and that the price per coupler would be roughly $1,500-2,500. They are expensive because they would be made out of carbon fiber and precisely balanced in order to meet the high requirements of rpm and torque rating. Team 5 has been designing while keeping in mind future teams down the road, so the 6P18-A1C will reach the goal of 10,000 rpm this year, but will pave the road for custom couplers to be added to the motor test rig.

4.2 Bearing Housing

Due to the fact that our torque transducer was not approved for purchase, the team was inclined to test their theory using a mock transducer. In order to mimic the transducer in the system, a rigid component was needed to hold the shaft. They selected the following bearing and housing kit shown below.

Figure 14 Bearing Housing SNL 505

The SNL 505 bearing housing is the corresponding size for the spherical roller bearing, shown below. It will be mounted onto a stand which is attached to the frame. This will prevent the compressors from fighting each other. The team selected a spherical roller bearing with an adapter sleeve to account for axial and radial loads that might be affecting the bearing to prevent failures at high speeds. It has a tapered bore which tightens onto the shaft and it has an adapter sleeve to fit the required 20mm shaft that connects the two compressors.

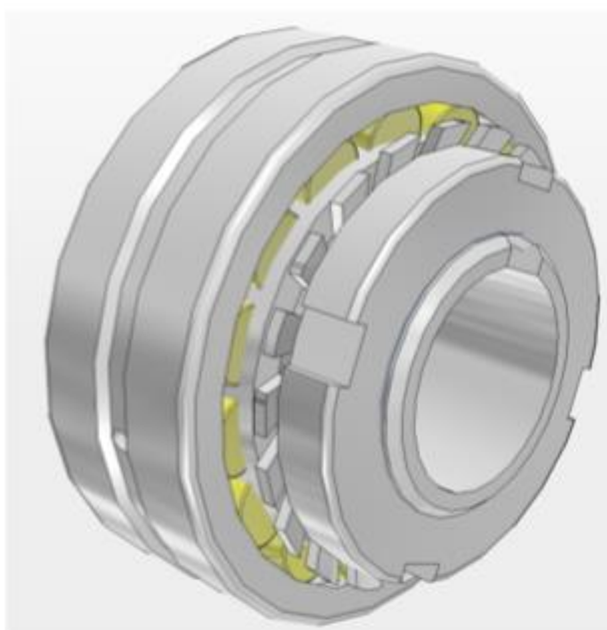


Figure 15 Spherical Roller Bearing

4.3 Safety Shield

A safety shield was used in order to help ensure the safety of everyone while the compressors were being run. It was a very simple box-type layout to prevent any projectiles from hitting anyone. The speeds that the system was planning to run at are considerably low, but as the speeds increase the safety shield will need to be upgraded and improved. A picture of our safety shield being implemented is shown in the design of experiment.

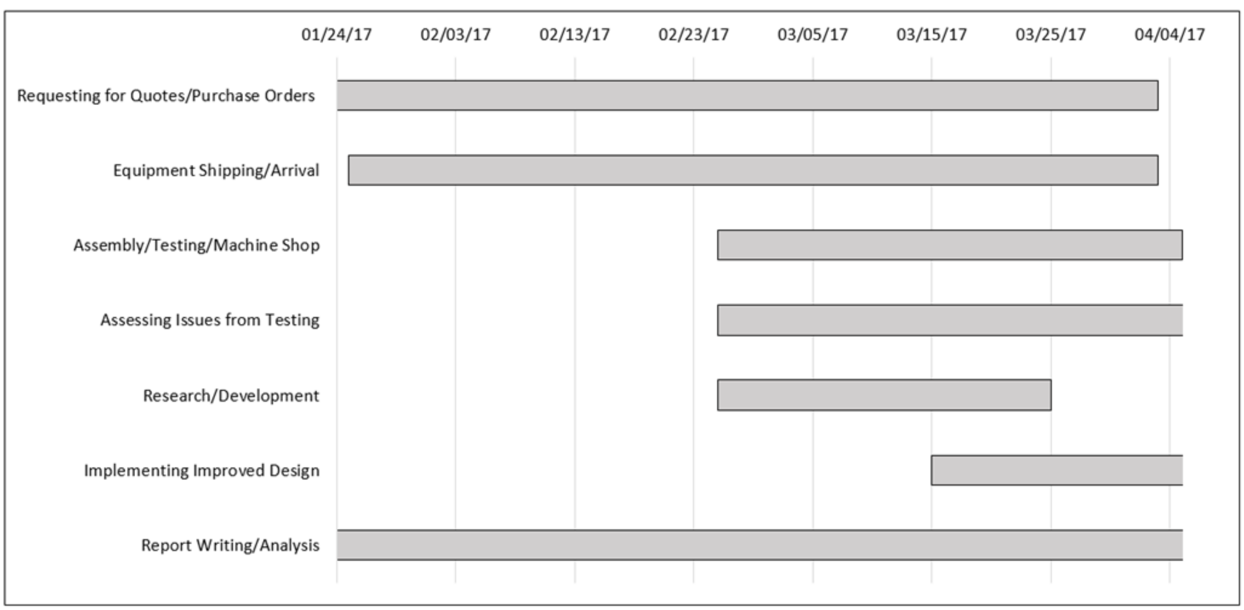
5.0 Project Management

Four basic factors greatly mattered to the realization of the goals of this project. These factors include scheduling, communications, risk assessment, and resource allocation and procurement. Each of these is discussed in details in the following sections.

5.1 Scheduling

The project had a lot of tasks involved in it and part of the requirements of the senior design course had to do with deliverables, which needed to be submitted at slated times during the period of the project. To therefore have a grip on the project, and not fall behind on time, the team adopted the use of a Gantt chart to schedule tasks. A Gantt chart is a chart that shows the start and finish dates of the terminal elements and summary elements of a project [1]. With the help of a Gantt chart, the team was cognizant of how much progress it was making, and revised it when needed, to accommodate unforeseen issues, including frequent changes in the course schedule and deliverable dates. The table below shows the typical layout of the Gantt chart we used throughout the period of this project.

Table 4 Gantt chart



5.2 Resource Allocation and Procurement

The team that worked on the design of the test rig was made up of four individuals, all mechanical engineering students, and there was no specific amount of money allocated to the team for this project. Although almost every task was eventually carried out together as a team, to ensure efficiency and productivity each team member was assigned to a specific responsibility, as follows:

- Alex Jurko was the team leader, responsible for communicating clearly and frequently with both sponsors and advisors, as well as keeping the team focused on the task at hand while staying on schedule.
- Jonathan De La Rosa was the lead design engineer, responsible for creating CAD or Pro E drawings of our design, as well as carrying out most calculation works with the assistance of the other members.
- Fehintoluwa Aponinuola was the lead web designer, responsible for keeping the senior design team's website up to date with deliverables, as well as being delegated to help any of the team members who needed assistance to execute an assigned task, if the website is up to date.
- Jack Pullo is the financial advisor, responsible managing the amount of money spent on the project, while researching alternative options and cost-effective solutions, as well as being delegated to help any of the team members who needed assistance to execute an assigned task, as the need rises.

A total of $4,916.25 was spent on this project (this is broken down on the table shown below). From the pie chart on the figure below, most of the budget, exactly 78.26%, was spent was on purchasing the laser alignment tool. The remaining 21.74% was spent on purchasing the materials needed for the mock-transducer stand, the shaft and the couplings.

Table 5 Economic Break Down

S/N	ITEM	QUANTITY	AMOUNT	TOTAL
1	Laser alignment tool	1	3,847.01	3,847.01
2	Double-flex disc coupling	2	417.85	835.7
3	Low-carbon steel bar 5/8" thick, 6" wide, 2' long	1	85.85	85.85
4	Low-carbon steel rectangular Bar 1/2" thick, 6" wide, 1' long	1	37.55	37.55
5	Low-carbon steel square tube 4" wide, 4" high, 0.250" wall thickness, 1' long	1	51.76	51.76
6	Low-carbon steel shaft 0.787" diameter	1	58.38	58.38
	Total			**4,916.25**

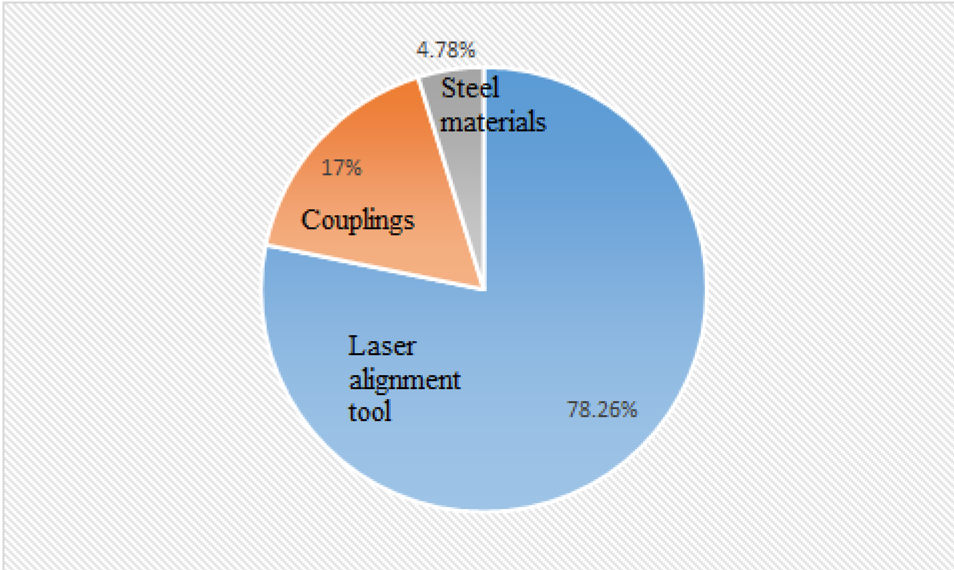


Figure 16 Economic Break Down - Pie Chart

5.3 Communications

At the beginning of the project, the team established means of keeping in contact with each other and the contact persons at Danfoss Turbocor. The team agreed to keep each other updated about project progress via GroupMe, a mobile group-messaging application, and email and phone calls when necessary. Communication with the liaison officer, Mr. Sun, and other contact persons at Danfoss was basically via email, and phone calls when necessary. Although the team achieved seamless communication among one another, communication stream between the team, suppliers and the contact person in charge of placing orders was not as smooth as would have been appreciated. Due to this miscommunication, some of the parts that were ordered were wrong and had to be returned, and some had to be remachined. This caused a lag in the project, causing the team to fall seriously behind schedule.

5.4 Risk Assessment

Failure mode effects analysis (FMEA) was carried out on the design the team came up with, which was approved by Turbocor; each of the components that made up the design was analyzed to observe how much harm they posed to the safety of the test rig operator. The result of the FMEA unveiled more possibilities of potential failures among the components between the two compressors, and an lexan polycarbonate safety shield was designed around this region to shield the test rig user from harm. The FMEA is located in the appendix along with an FEA that the team created.

6.0 Design of Experiment

6.1 Initial Calibrations

The TT500 is the model of compressor that the team had designed for the entire year. Every aspect of the dimensions was used in order to determine the centerline for the couplings and shaft, use of the base frame, and also the power and rpm that the compressor can reach. However, when the team arrived on site to test the new components, the only TT500 available was the one that was initially lent to them early on in the semester. This was a shock, but not a huge deal because the TT700 is the newer model. It is slightly larger and more efficient, but the centerline is the same as the TT500 and would connect to the rig the exact same as a TT500 would. For future design implementation the team recommends that the exact same compressor is used to eliminate possible anomalies.

Julio Lopez is an employee at Danfoss Turbocor and has helped tremendously in regards to setting up the rig and actually testing it. There was an issue of how the couplings were to be attached to the internal shaft of the compressor without affecting the layout of the shaft itself. He managed to come up with the plan to push out the compressor shaft more than normal and then lock it in place so that the majority of the torque load was on the front bearing when attaching the coupling instead of on the rear bearing which has a precise gap that is needed to maintain for levitation. When the rig is first being set up, it is important to note that there is always a change in how the compressor levitates when new components are added on to it. The first process is to calibrate the compressor with only a single coupling attached. After that is completed, the data shows what kind of a gain the system requires as compared to the compressor without the coupling attached. This is important to note because after each component is added, the transfer function must be adjusted to match the new additions as well as the frequency change between the front and rear bearings. The following figures and pictures show this process executed on the single TT500 used in testing.

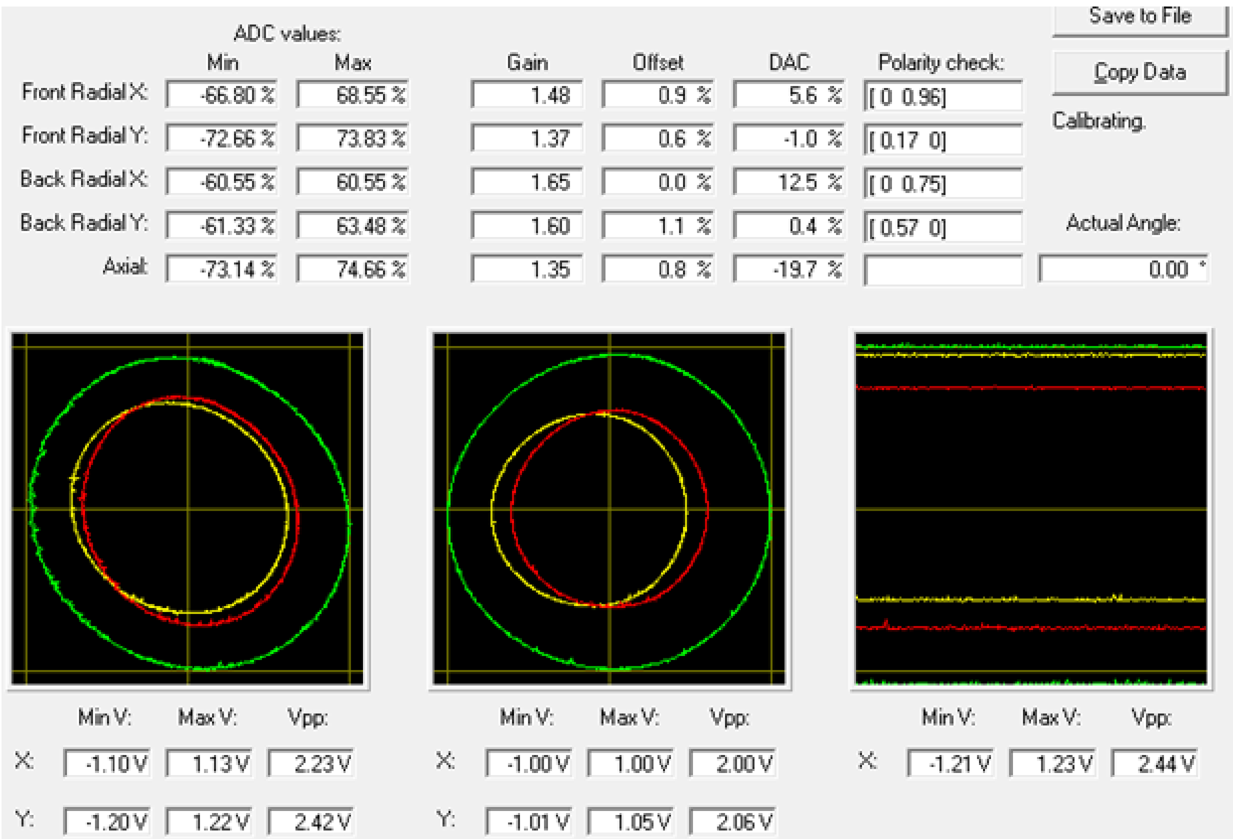


Figure 17 Initial test with no coupling

This was the base run of the compressor, showing the front and rear bearings with the circles to represent the concentricity between the front and rear bearings. As expected, without anything attached to the shaft it has no issue levitating and no noise that affects the natural frequency of the system.

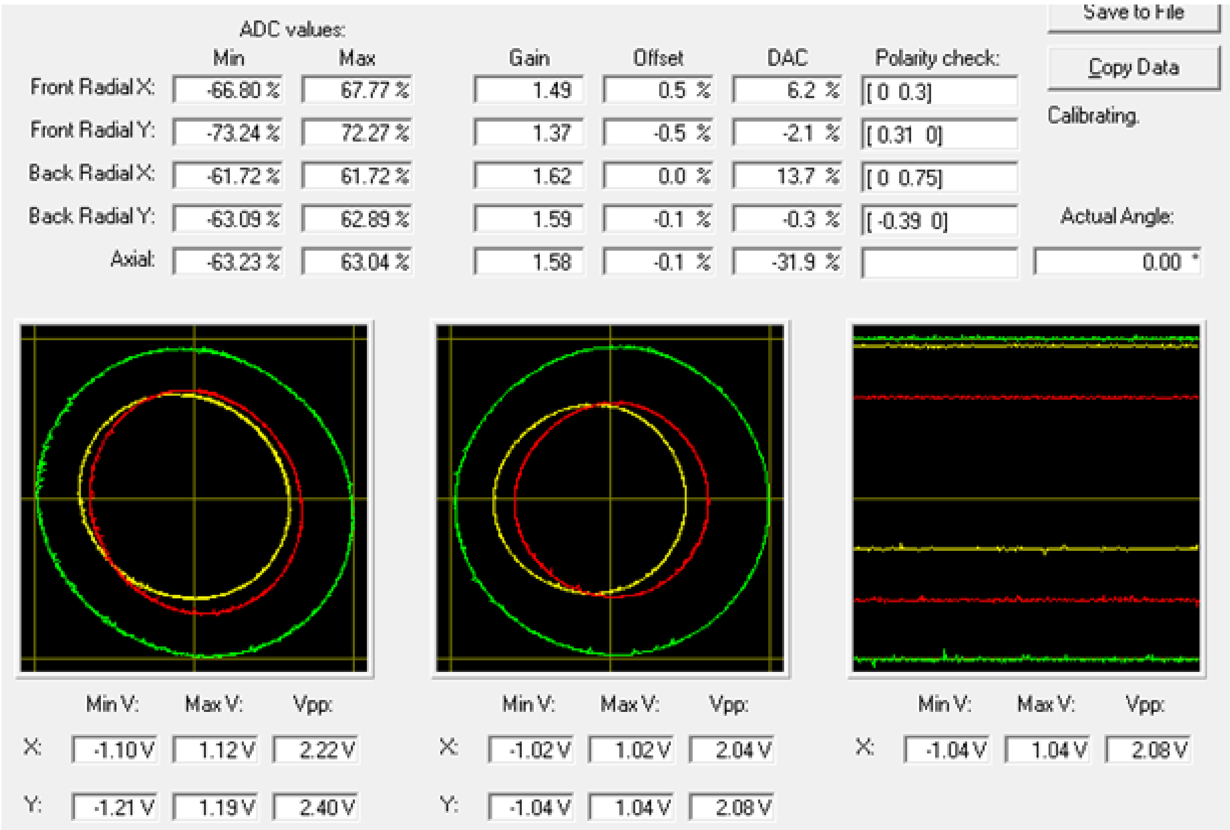


Figure 18 Initial test with single coupling

This is the same data sheet, but after the first coupling was attached to the system. Normally the coupling is attached using 13 Nm torque, but when the shaft was levitating and running, the air gap between the coupling and the internal shaft was too great and it contributed to noise in the system. To eliminate that, the team tightened the couplings to a higher torque but not more than achieved by a simple hand and allen wrench. This eliminated the noise and the compressor was able to run smoothly at multiple hundreds of rpm.

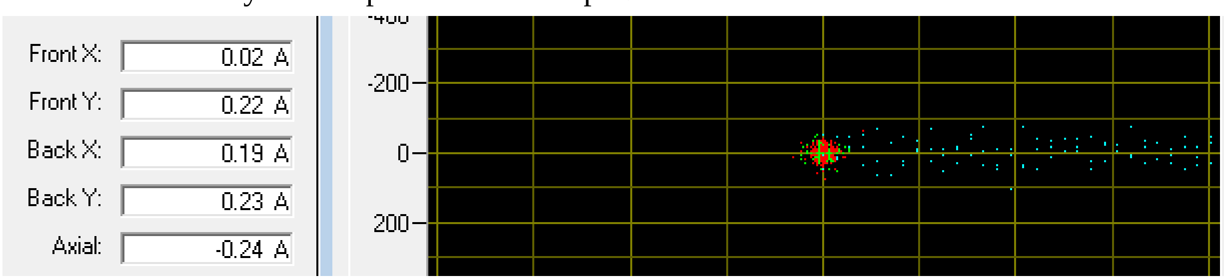


Figure 19 Initial test run with single coupling

This plot is the position of the shaft inside of the compressor. Since the compressor deals with electromagnetic bearings, it can determine the compensation required for weight distribution based on the amount of Amps flowing through the bearings at each sensor. At this point, the front x is extremely low because there is only a coupling attached that doesn't really affect the x direction, however the y direction causes it to be at 0.22A.

After this was tested and confirmed to not be an issue to the compressor's operation, it was time to add on the shaft. The shaft itself causes the flexible coupling to bend down slightly due to coupling's flexible nature as well as the force of gravity. Considering these aspects, when the compressor starts to run it will affect the shaft and cause it to rotate at an angle to the centerline because it is only supported on one end. A picture is shown below of the setup for a visual representation.

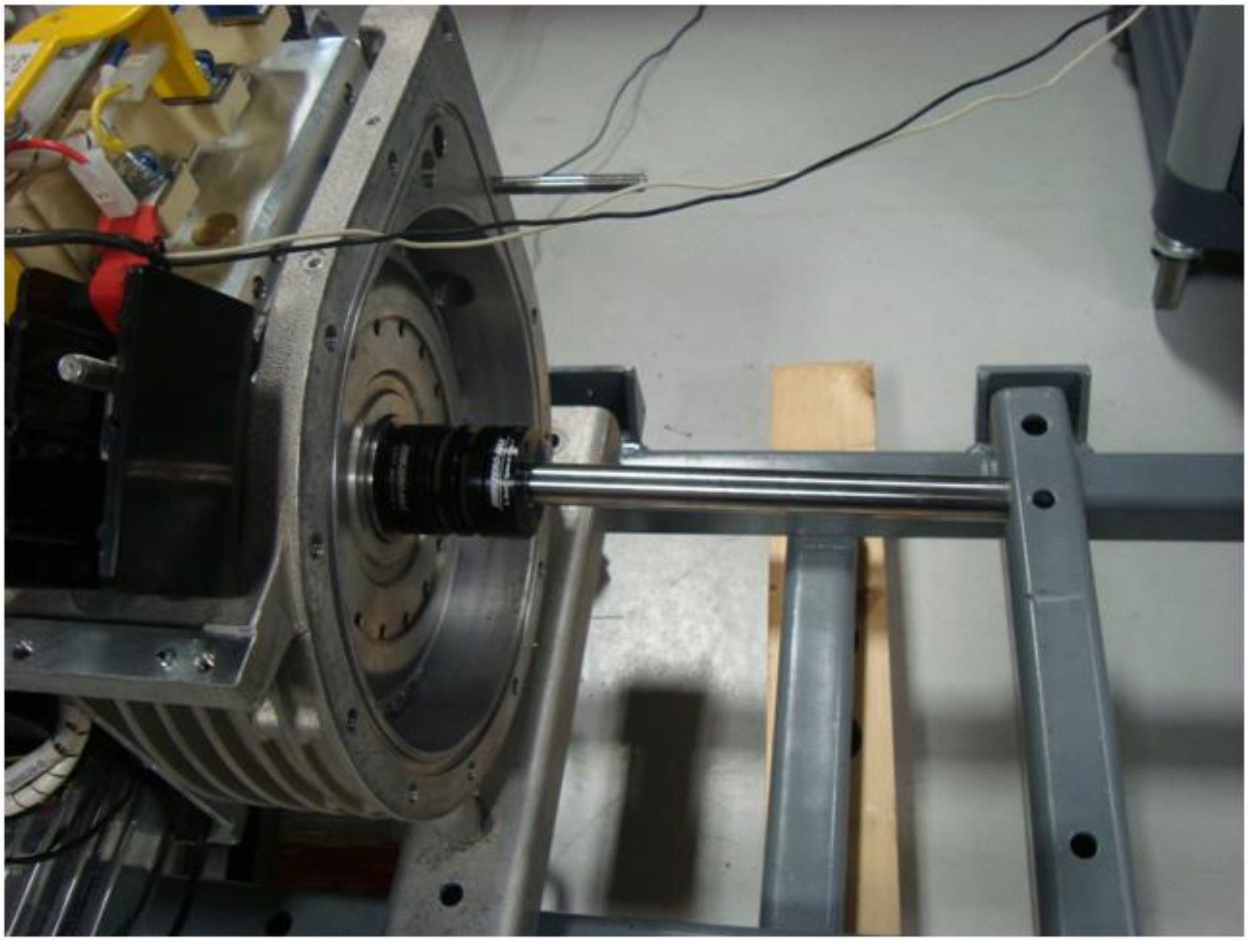


Figure 20 Shaft Added to System

The testing of the shaft was successful, there was a bit more compensation required for balancing as shown below. The front x is now at 0.12A and the front y is 0.30A as compared to 0.02A and 0.22A respectively for the front bearing in the previous test. Overall, it tested well and was able to run stable up to 300rpm. The team did not test faster than that because there could have been damage done to the coupling or shaft due to the lack of support on both sides.

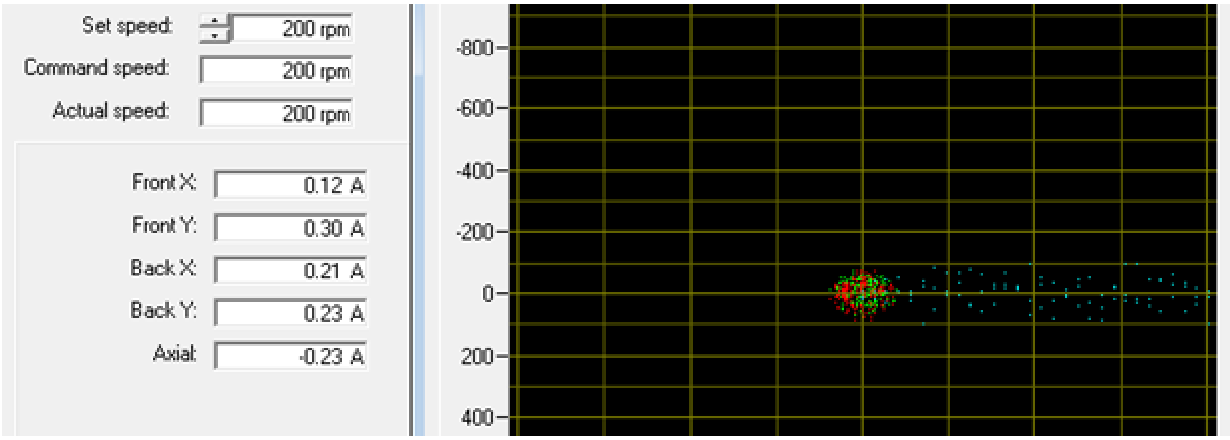


Figure 21 Testing with shaft and coupling

The last addition was to add the final coupling to the shaft and see how the compressor reacts with the entire connection series on a single compressor. The picture below shows how this was set up and what it looked like in the rig. It is very obvious to note the angle that the weight of the second coupling has on the shaft itself. The couplings are very lightweight but they are also very flexible, so the combination of the shaft and coupling weight on the right hand side causes the coupling attached directly to the compressor to flex downwards.

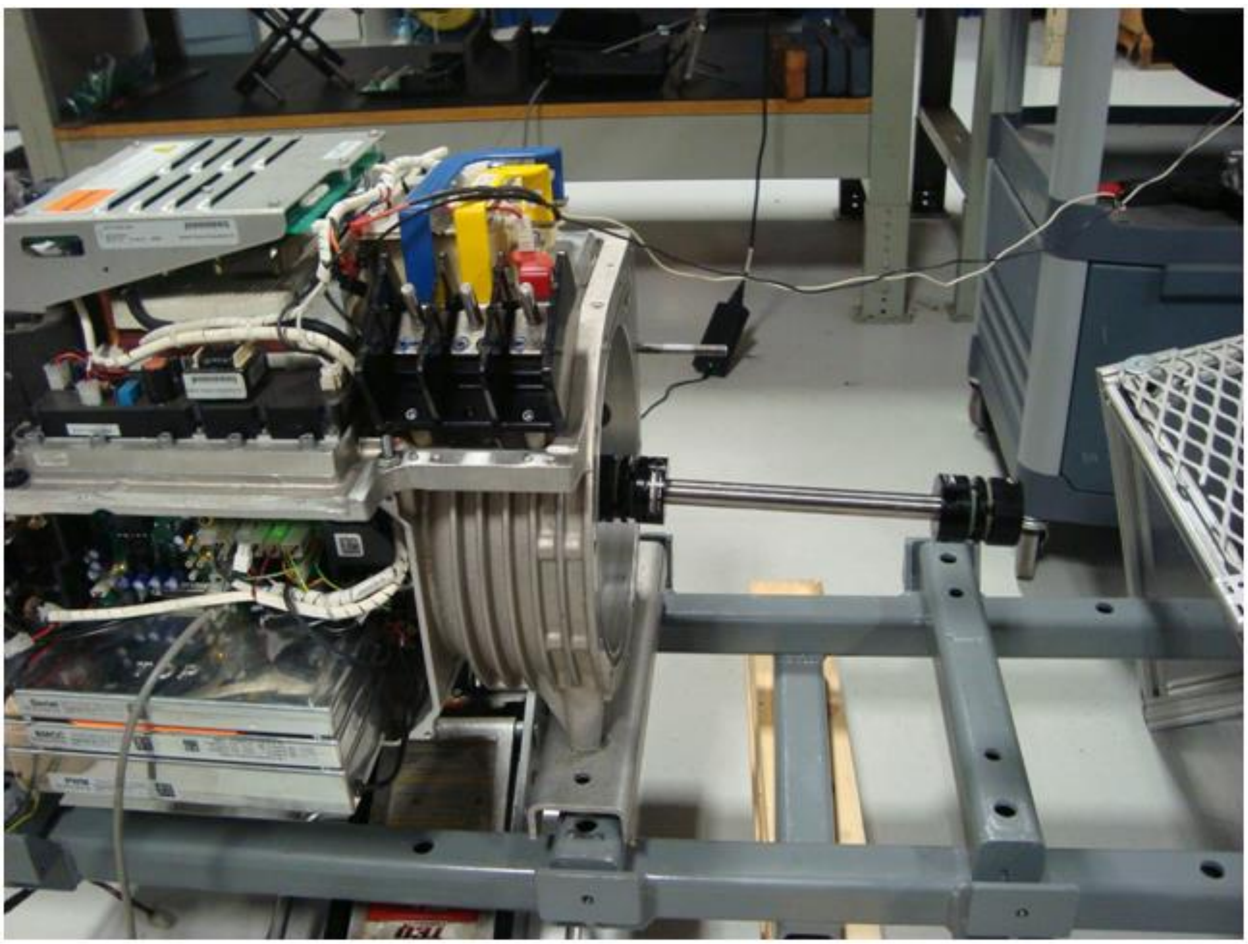


Figure 22 Shaft and second coupling attached

The same test was then executed with this setup and the results are shown below. There was a slight increase in the front bearing's y axis compensation, but other than that it is about the same. The levitation process was successful and there was no noise factor that could contribute to the compressors shutting down. Julio, the compressor expert helping the team, said that just based off of this initial calibration testing that the system would work far better than the previous design. Last year's team faced lots of issues just setting up the levitation and calibration portion because their setup involved a very clunky and heavy flexible coupling that actually caused more vibrations rather than dampening them. It also affected the natural frequency a lot more heavily than the setup from team 5. It was a good design idea to account for misalignment, but it ended up backfiring when they started reaching higher speeds and eventually shut down the compressors.

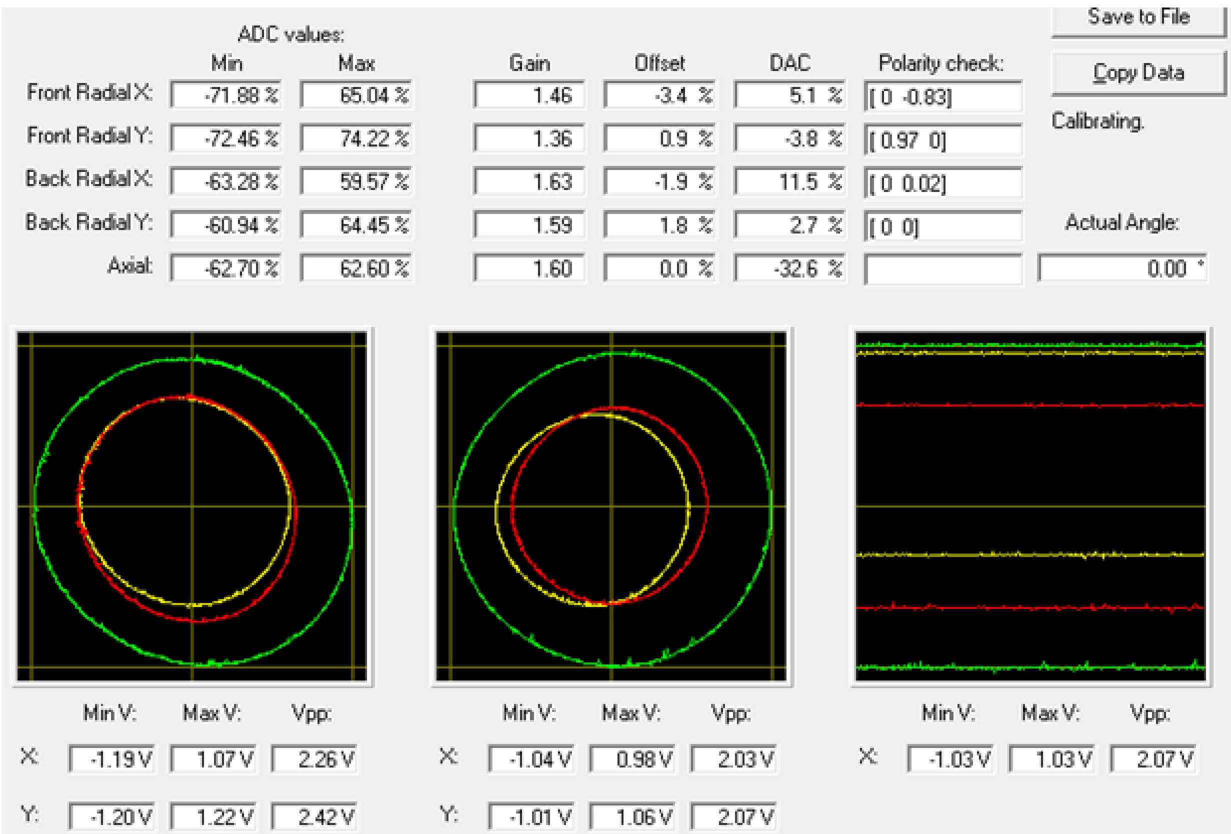


Figure 23 Test with shaft and second coupling

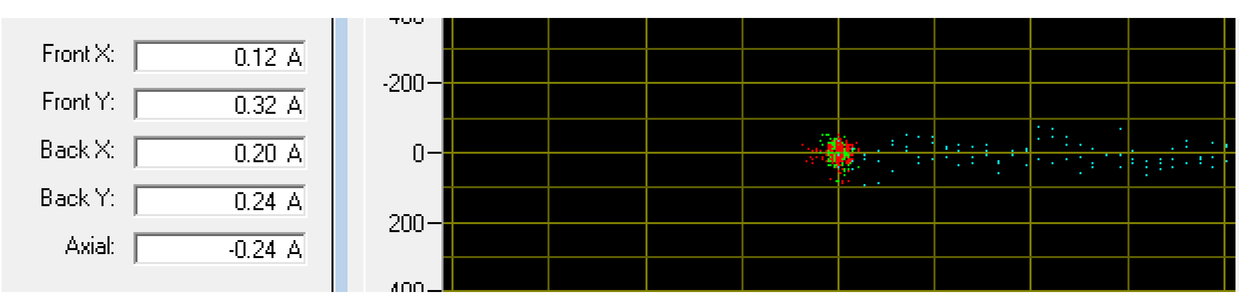


Figure 24 Test run with shaft and second coupling attached

6.2 Alignment

The alignment process is one of the most important aspects as discussed in this project. The lateral direction was aligned using the same technique as last year with lateral screw sets implemented in the frame itself. These worked great so there was no need to change that design. For the vertical direction however, to increase the simplicity, ease, and effectiveness of lifting the compressors the entire rig was elevated onto 3 wooden beams at the base legs of the frame. This lifted the frame up just enough so that a car jack with a low profile could roll underneath and easily lift the compressor at the front or rear legs for simple shimming and no danger of hurting the person lifting the compressor (they are very heavy). Below is a picture with the car jack underneath and the wooden beams clearly visible to show how it was elevated.

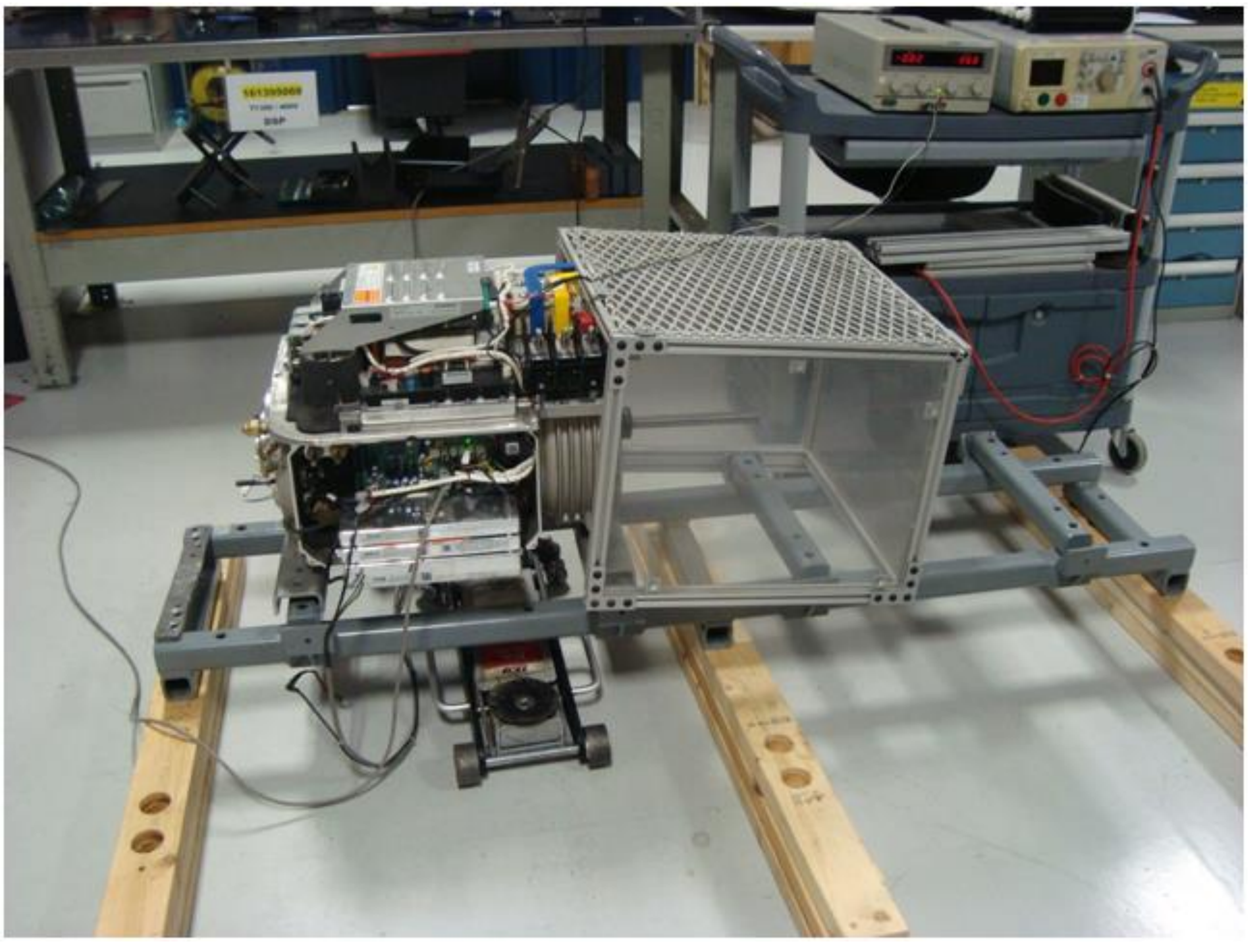


Figure 25 Picture of single compressor test with safety shield

Now that the actual lifting and moving of the compressor is accounted for, the tool to determine the alignment was the next most important thing. Last year they chose to use a hand dial alignment tool because it is relatively inexpensive and if used correctly can be very accurate. However, this has the issue of some human error based on the angle you are reading the dial from, as well as the fact that it is time consuming to adjust the compressor slightly and then take 3 readings again over and over until it’s close enough for safety. This year the team purchased a laser alignment tool, the TKSA 31. It takes 3 readings at 3 o’clock, 9 o’clock, and 12 o’clock relative to your centerline and creates a plane to determine the offset between the two tools as shown below. After it takes these readings it actually can show you in real time where you need to adjust more or less depending on which compressor you choose to be the "stationary reference" and the other one will be the one that's adjusted. It is a faster and more precise tool that eliminates majority of human error as long as it is set up according to the manual.

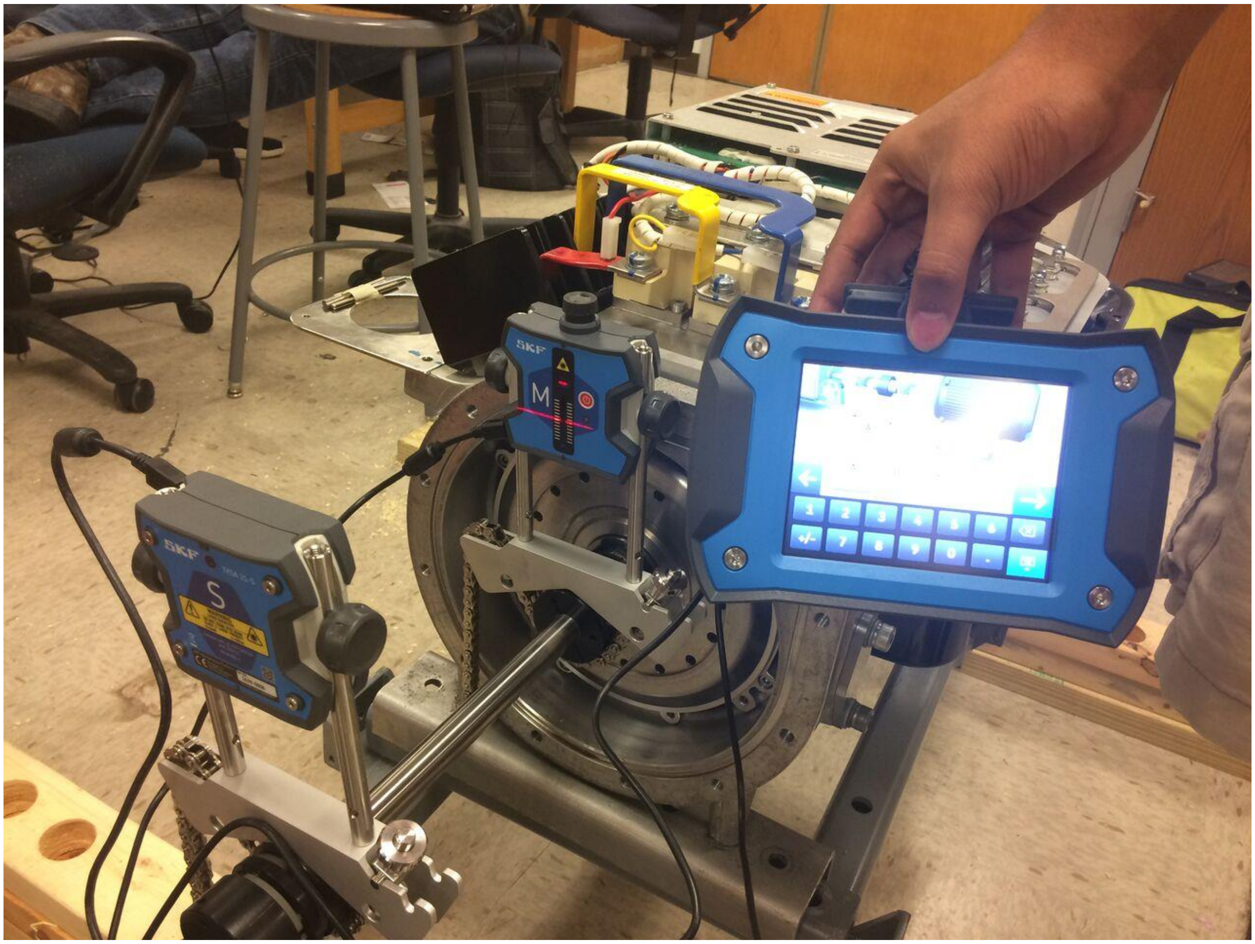


Figure 26 Alignment tool operation

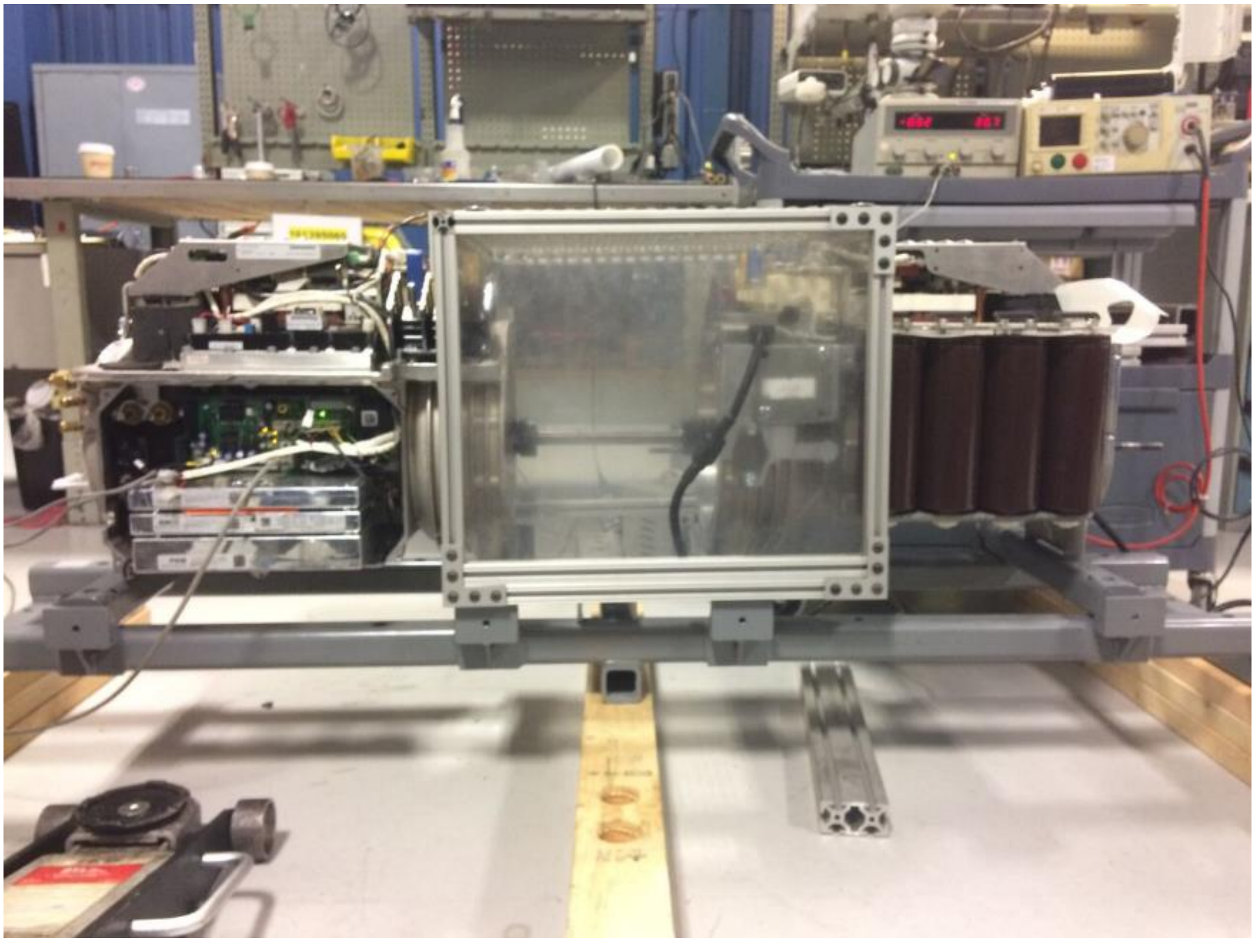


Figure 27 Both compressors set up with safety shield

7.0 Conclusion

7.1 Results

The team was successfully able to run the compressors with all the components attached. There were some issues running the TT700 with the power supply available. When the team switched the driving motor to the TT500 instead, they were able to reach 726 rpm recorded on video. The team will be following up in a week when a better power supply is available and plan to reach the milestone of 1,000 rpm. The sponsor voiced their approval and were impressed with the results given the setbacks that the team endured.

When initially running with the TT700 the torque load was too high for it to reach more than 200 rpm. This was the opinion of Julio, the compressor expert, and he came to the conclusion after the connecting components began to shutter or simply rock back and forth slowly when it was supposed to be running at 200-250 rpm according to the software. This was most likely due to the fact that the torques did not match up between the two compressors added to the fact that the power supply could only reach 6A. When the compressors were swapped, the components held up perfectly fine until around 300 rpm. Between 300-500 rpm the shaft began to "wobble". That word probably describes it best because it was not a back and forth oscillation, it just seemed that the concentricity between the two couplings was not ideal. After 500 rpm it started to level out and seemed to balance. This was a great sign for the future because it had absolutely no issue beating last year's team in terms of rpm, and the only reason the test had to stop was because the power supply could not support faster than the 6A required to reach that rpm.

7.2 Future Recommendations

Looking back, the team wishes that they had more communication between the distributors and the sponsor so no potential mix up will occur. The team encountered some issues with the components that were ordered, but luckily were able to be resolved in time to test the design. If the sponsor wishes to continue developing the High Speed motor test Rig, team 5 recommends having the sponsor allocate some of the resources that the future team can use from the get go. This would allow the next team to inherit the project to get to a running start. Also the team recommends that the next team try to get the torque transducer approved and purchased to incorporate into the system. Team 5 got the transducer initially approved, but had to go an alternate route due to the large lead time that it had. The next team should be ahead in researching and getting designs approved so there is enough time to make any adjustments that would be needed.

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3. http://www.agroengineers.com/bearings/types-of-couplings-2.shtml

4. http://www.rw-america.com/products/precision-couplings/metal-bellows-couplings/bk2.html

5. http://eng.fsu.edu/me/senior_design/2016/team04/finalreport.pdf

6. http://www.skf.com/binary/30-227821/MP5430EN.pdf

Appendix

Failure Modes Effects Analysis

Process or Product Name:	Design 4 (Selected Desgin With Transducer)	Prepared by:	Team 5	Page:	1 of 1
Process Owner:	Senior Design Team 05	FMEA Date (Orig):	11/10/2016	Rev.	

Key Process Step or Input	Potential Failure Mode	Potential Failure Effects	SEV	Potential Causes	OCC	Current Controls	DET	RPN	Actions Recommended	Resp.	Actions Taken	SEV	OCC	DET	RPN
What is the Process Step or Input?	In what ways can the Process Step or Input fail?	What is the impact on the Key Output Variables once it fails (customer or internal requirements)?	How **Severe** is the effect to the customer?	What causes the Key Input to go wrong?	How **often** does cause or FM **occur?**	What are the existing **controls** and procedures that prevent either the Cause or the Failure Mode?	How well can you **detect** the Cause or the Failure Mode?		What are the actions for reducing the occurrence of the cause, or improving detection?	Who is Responsible for the recommended action?	Note the actions taken. Include dates of completion.				
SKF TKSA 31	Not properly installed correctly	Having the motor test rig misaligned, resulting in higher chance of failure	8	Handling Error	7	Reading the instructions. Along with watching videos from the manufacturer on how to install the equipment properly	7	392							0
	Faulty equipment	Not being able to have the motor test rig properly aligned resulting in a very high chance of failure	8	Manufacturer	3	Standard Operating Procedures	9	216							0
	Lack of understanding with properly operating the equipment	Not having the full potential of the device. This results in not having a proper alignment	7	Human Error	5	Properly reading the instructions and watching tutorials on the device to ensure proper use of it	8	280							0
	Not properly fitting	Failure to align	10	Lack of Research on the range in which the laser aligment tool can attach to	4	Extensive Research into what range the laser alignment tool can attach	8	320							0
Curved Jaw Coupling	Couldn't handle desired rpm	The coupling getting damaged. Compressor shutting down due to possible vibrations that would occur. Also potentially damaging other components on the test rig. Wouldn't meet the end result.	10	Lack of coupler diversity within the prexisting market with such desired rpm speeds	7	Contact distributors to see if it's possible to contrsuct a custom curved jaw coupling	4	280							0
	Not properly installed	Damage can occur	8	Human Error	7	Reading the instruction on how to install the curved jaw coupling correctly	8	448							0

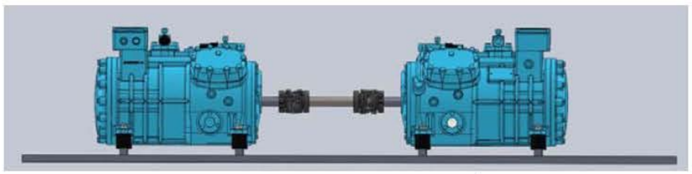
	Exceeded displacement range of the coupler	Causes the compressor to shut down. Potentially damaging the couper itself	10	Selected coupler from research done by the team	4	Further research on the jaw couplings. Discussing with professionals in the industry to select the right coupler	7	280							
	Faulty Equipment	Wouldn't function properly in order to test the design	10	Manufacturer	3	Standard Operating Procedures	9	270							0
2 Rigid Coupling	Couldn't handle desired rpm	The coupling getting damaged. Compressor shutting down due to possible vibrations that would occur. Also potentially damaging other components on the test rig. Wouldn't meet the end result.	10	Lack of coupler diversity within the prexisting market with such desired rpm speeds	7	Contact distributors to see if it's possible to contrsuct a custom curved jaw coupling	4	280							0
	Not properly installed	Damage can occur	9	Human Error	7	Reading the instruction on how to install the curved jaw coupling correctly	8	504							0
	Faulty Equipment	Wouldn't function properly in order to test the design	10	Manufacturer	3	Standard Operating Procedures	9	270							0
Shaft	Improper Length	Won't be able to properly connect the couplers to the compressors	10	The deisgn team	5	Adjustment of the compressors on the frame. Checking the legnth in between the compressors beforehand to properly purchase the correct shaft	7	350							0
	Frequency Issues	The software inside the compressor wouldn't operate properly	6	Complexity of the design due to the additional components	10	The Hammer (Or Dong) Test to see the frequency with the additional components. (Hard to test without purchasing)	10	600							0

Finite Element Analysis of Motor Test Rig Components

Jack Pullo
12/4/16

Overview

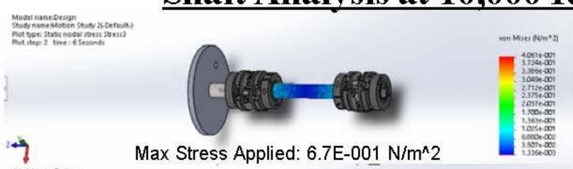
- Implementing a system to test performance of Danfoss Turbocor's TT500 Compressor
  - ❖ Before purchasing expensive components, specific parts were analyzed to determine if the design concept would fail
  - ❖ External conjoining shaft, base plate stand, selected couplings
- Used Creo Pro E and SolidWorks
  - ❖ Displacement, Von Mises Stress, and Factor of Safety diagrams were produced
  - ❖ Turbocor FEA Specialist verified results using ANSYS

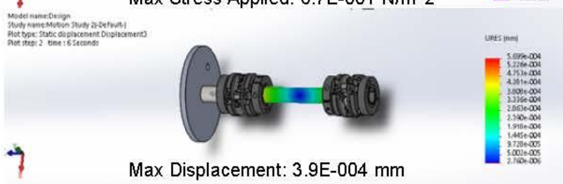


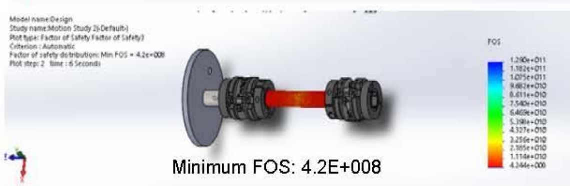
Procedure

- A motor was applied at 2000, 5000, 10000, and 30000 RPM on the internal shaft extruding out of the impeller.
  - ❖ The diagrams show results of external shaft connecting the two coupling to verify that the couplings would securely fasten the external shaft without causing alignment issues.
    - Due to space issues. FEA diagrams of the shaft are only shown at 10,000RPMs
    - Large displacement would effect alignment and result in the compressors shutting down.
    - If the Von Mises Stress surpassed the yield strength, the shaft would rupture
    - If a Factor of Safety under 1 occurred the design structure would fail
- A distributed load of 600 pounds was placed at the critical points on which the compressors located on the base stand
  - ❖ To assure the stand wouldn't collapse and drop the compressors during operation

Shaft Analysis at 10,000 RPM



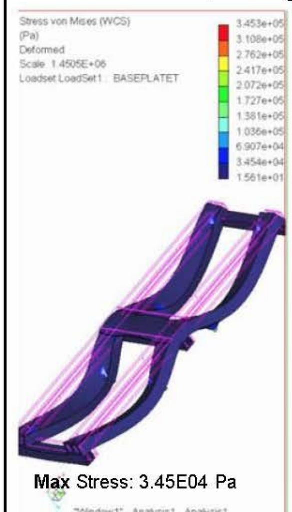


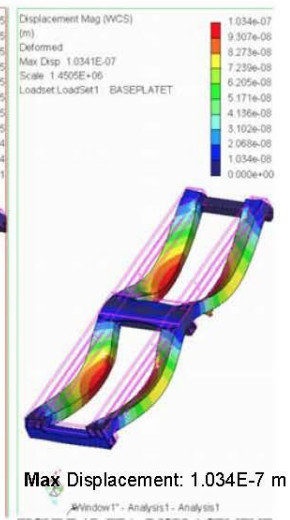


- The yield strength of the selected material is 220594000 N/m^2. Therefore the shaft would not fracture
- The torque applied from the motor doesn't't cause enough displacement to produce alignment concerns
- The factor of safety remains way above a value of 1
- At each speed the results verify that the couplings will work effectively

Confirms that the shaft will not fail during high speed operation

Compressor Base Stand Analysis





- The yield strength of the selected material is 513613000 Pa. Therefore the stand would not fracture
- The force applied from the load bearing of the two compressors doesn't produce enough displacement to cause failure in the design
- Can see the majority of stress and displacement occurs where compressor applies most of its weight

Confirms that the design of stand will not fail while compressors attached

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Q U O T A T I O N

PAGE 1 OF 1 **NUMBER** B959986

FROM
2402 SHORT STREET
ALBANY GA 31705

NUMBER B959986

TO
DANFOSS TURBOCOR
726 S. WESTOVER ROAD
ALBANY GA 31707

SHIP TO
DANFOSS TURBOCOR
1769 E. PAUL DIRAC DRIVE
TALLAHASSEE FL 32310

ATTN:

ACCOUNT NO.	SALESMAN	SHIP VIA	SHIP DATE	TAX	BRANCH NO.	PO / TAG ORDER NUMBER
CSHAGA	S99	UPS	17-JAN-17	NYYY	00323	

SEQ.	ITEM DESCRIPTION	ORDERED	BACK ORDERED	SHIPPED	U/M	PRICE	EXTENSION
	***** QUOTE **** QUOTE ** QUOTE ** QUOTE ** QUOTE **** QUOTE *****						
	***** QUOTE **** QUOTE ** QUOTE ** QUOTE ** QUOTE **** QUOTE *****						
	ZER 6P18A1C 25MMNOKWX10MMNOKW_ALUM SRV DBL FLEX						
1	0	2	0	0	EA	417.850	835.70
	THESE HAVE A LEAD TIME OF 2 WEEKS ARO THEN SHIP FROM MN.						

INITIAL: WLG N	BUYER: JACK PULLO	RESALE:	TAX	58.50
QUOTE EXPIRES: 16-FEB-17	CUSTOMER TELEPHONE: (954)235-1939	BUS:(229)432-0090 FAX:(229)432-0150	TOTAL	894.20

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Service Center Address: APPLIED IND TECH - DIXIE, INC. 1021 W JACKSON ST THOMASVILLE, GA 31792-6338 TEL: 229-228-0203 FAX: 229-228-5249 **Account Manager:** JAMES HOPKINS	**Quote Create Date (MM-DD-YYYY):** 01-27-2017	**Sales Rep:** MICHAEL ODOM **Phone:** 2292280203 **Email:** MODOM@APPLIED.COM	**Page:** 1/1
	Quotation Number: **504106812**	**PO #:** **Req #:** **Inquiry #:**	
Sold-To Address: **Account:** 1504643 MARPAN SUPPLY CO INC 356 MARPAN LN TALLAHASSEE FL 32305-0904	**Ship-To Address:** **Account:** 1250046 MARPAN RECYCLING 356 MARPAN LN TALLAHASSEE FL 32305-0904		
Payment Terms: NET 30 DAYS **Transport Mode and Means:** PARCEL **Terms:** FOB ORIGIN, FRGHT PREPAY & ADD	**Customer Contact:** **Customer Contact Phone:**		

Item #	Quantity	UOM	Manufacturer Part #	Customer Part #	Est Delivery Date	Unit Price (USD)	Extended Price (USD)
000010	1.000	EA	**SKF-TKSA 31** **Description:** SKF_TKSA 31			3,847.010	3,847.01
						SUBTOTAL (WITHOUT TAX)	3,847.01
						TAX	288.53
TOTAL IN WORDS: FOUR THOUSAND ONE HUNDRED THIRTY-FIVE DOLLARS AND FIFTY-FOUR CENTS						**TOTAL(USD)**	4,135.54

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