High Speed Motor Test Rig

Final Report

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Team 4

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Abstract

Danfoss Turbocor has asked Senior Design Team 4 to develop a high-speed motor test rig to qualify their motor performance within the compressors. They have asked Team 4 for a motor-generator rig that uses one compressor motor to transmit power to another that can act as a variable load (generator). The original designs were rather expensive and Team 4 has offered a more affordable alternative to prove alignment within the system and to also eventually run the compressors at full speed output. After some redesign the new total cost of the proposed design is $ 1,046.76. Although the original design has been modified at Turbocor's discretion, this design makes the insertion of torque transducers possible. This will allow for the system to be to be ran at max speed or max torque in the future. The test rig construction was completed April 11th, 2016 and alignment testing was deemed a failure..

I. INTRODUCTION

A. Background

Turbocor started in Australia back in 1993, however, they did not install the TT300 (their first compressor) until the year of 2001 in California. Today they now stand amongst the leaders in the oil free centrifugal compressor product market. In order to combat the fast growing demand for their products, they moved to Tallahassee and opened up shop in 2006.

Turbocor wants a motor-generator system that can be utilized to qualify their compressor motor performances. They want to qualify the efficiency, power, and heat management of their high-speed compressors. Coupling a shaft that binds the motor to the generator creates the motor-generator system. This system is used in the situation in which power needs to be produced from the generator. Instead of using a motor and a generator, Turbocor wishes to use two compressors. The compressor that will serve as the generator will have the functionality of being able to be back driven by the other compressor that will serve as the motor.

For the purpose of this project, the impellers will be removed. With regards to alignment, special attention will be given since this motor generator system will be running at speeds much higher than the average motor generator system.

Turbocor has a high reputation of producing very reliable products, despite the increase of competitor respectability amongst consumers. In order to remain amongst the world's leaders in the oil free centrifugal product market, they have placed quality at the top of their lists. In response to their competitors, the company has invested in doing research to discover new and innovating ways to assure quality of their product. They have ventured onto FAMU-FSU College of Engineering Campus in hopes of gaining some help. Thus, they have granted Team 4 the opportunity of handling their top quality goal.

B. Objective

Team 4 has been tasked with designing and constructing a High Speed Motor Test Rig that can handle Turbocor's high-speed compressors (12,000 – 40,000 RPMs). Team 4 has placed heavy emphasis on the importance of choosing the correct coupler and alignment system that can efficiently work with the previously mentioned system. Since the compressors use magnetic bearings, they have varying load tolerances compared to standard ball bearings.

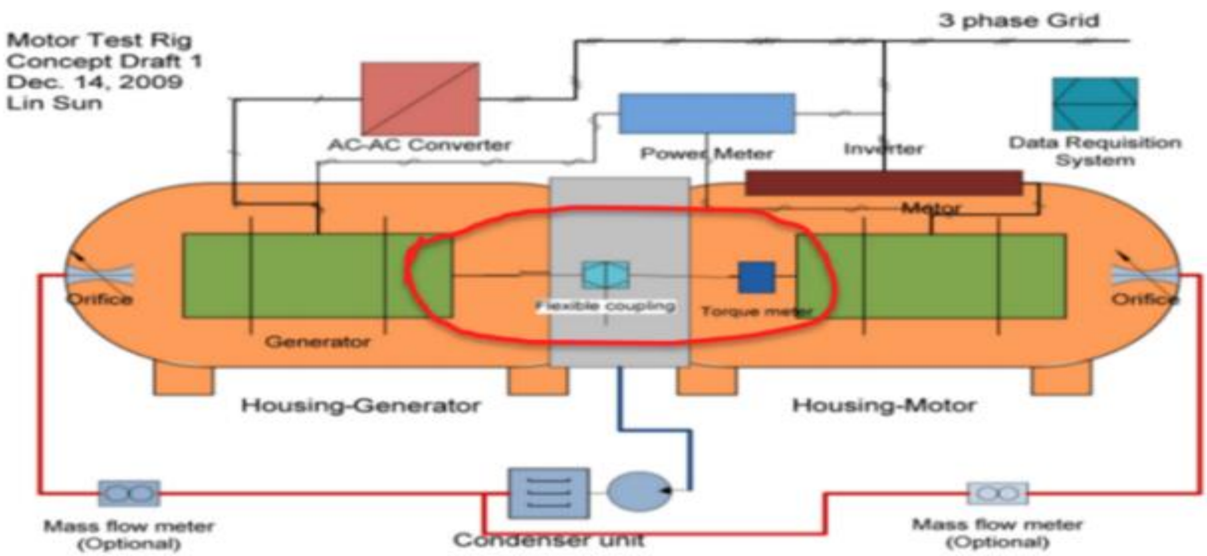


FIGURE 1: HIGH SPEED MOTOR TEST RIG DESIGN DRAFTED BY TURBOCOR

The red circle highlights the area of the motor-generator system that Team 4 has been focusing on. This section includes the flexible coupling (baby blue), the drive shaft, and the torque meter (blue). Since Turbocor is a local business, it was easy for Team 4 to communicate with them and be trained on how to run the compressors. Turbocor allowed Team 4 to keep a TT500 compressor on the FAMU-FSU College of Engineering's campus for experimental purposes.

C. Problem Statement

Danfoss Turbocor needs Team 4 to use innovative thinking to design a high-speed motor test rig to qualify their TT- Series compressor motor performance. Due to the high speeds during operation, special emphasis and attention will be placed on precision alignment requirements.

D. Design Requirements

Turbocor has required the High Speed Motor Test Rig to be compatible with various compressors that Turbocor produces. Due to the modification of Team 4's original design at Turbocor's discretion, the only thing that will be focused upon is the alignment process. Team 4 will be focusing on components that can be used across all compressor series. The max speed that the compressors can reach is 40,000 RPM and the max torque is 73 Nm. Team 4 has also placed a lot of focus on harmonic oscillations with regards to the test rig components. The original design was intended to utilize a torque transducer to give instant readings to record efficiency. The torque transducer was capable of operating for the non-normal capabilities of this motor- generator system.

II. CONCEPT GENERATION

A. Design requirements

This high-speed motor test rig requires attention to four main aspects to achieve successful compressor motor testing. 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 the client company that the individual components are to be qualified to operate up to 50,000 rpm and 100 Nm of torque. Danfoss Turbocor chose these requirements beyond the actual output of the motors to ensure an acceptable margin of safety in the mechanical system.

I. Magnetic bearings

When a radial force of 200 pounds or greater is exerted upon the compressor shaft, the system goes into a lockdown mode. This mode deactivates the motor and magnetic bearings, thus seizing the shafts rotation. If the components in rotation are not properly balanced, severe harmonic vibrations will exert forces in the radial direction. Because of this scenario, strict precision and accuracy for balancing must be considered for all components in rotation.

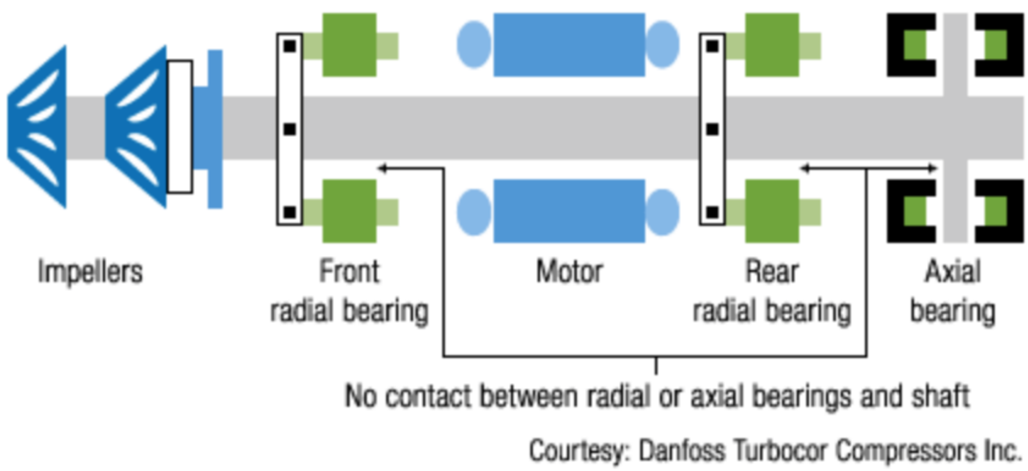


FIGURE 2. SCHEMATIC MAGNET BEARING SYSTEM

II. Shaft misalignment

One of the key components when dealing with shaft misalignment is the main coupler. To compensate for possible shaft misalignment, a flexible coupler will be used. Flexible couplers allow a limited amount of angular, parallel, and axial misalignment compensation. Figure 2 illustrates the possible misalignment orientations that can occur. Parallel misalignment is when the axis of the shafts are parallel, but do not connect as seen on Figure 2, the letter Y represents the misalignment. Angular misalignment occurs when the axes of the shafts connect through intersection, but are not parallel as seen on Figure 3 represented by letters A and B.

Axial misalignment is when the distance between the ends of the shafts changes during operation, shown on Figure 3 represented by letter X. While parallel and axial misalignment need to be considered in this project, axial alignment is not of huge concern since the compressor will be rigidly mounted to a rigid base with the couplings securely fastened.

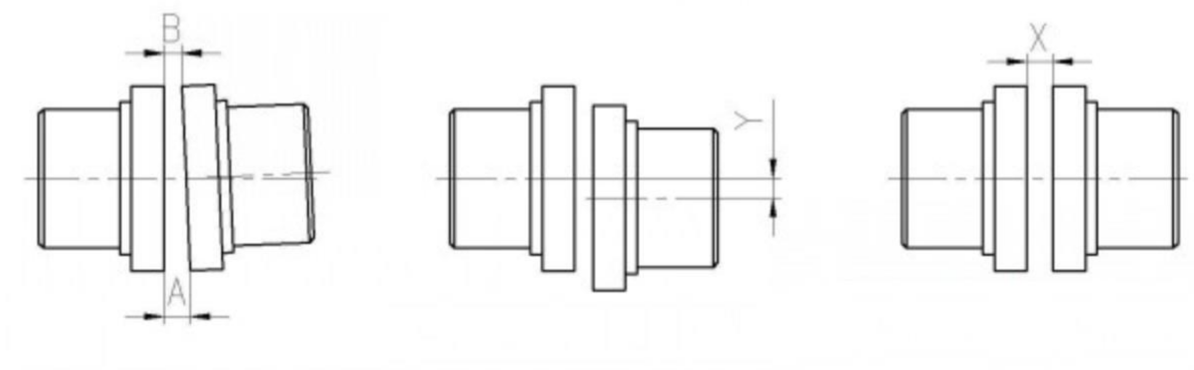


FIGURE 3: SHAFT MISALIGNMENT SCENARIOS

Another reason that a flexible coupler is desired is that it maintains a high Eigenvalue. If a rigid coupler were used in place of the flexible one, the dynamics of the characteristic equation for the system's rotation would be affected negatively. By using a rigid coupler, the two shafts would be coupled and considered to be one shaft in the differential equation to find the eigenvalue, which would increase the mass of rotating component. In turn, this mass increase would lower the eigenvalue of the shaft. The flexible coupler allows each shaft to be considered independent when analyzing the characteristic equation, which would lower the mass value, thus increasing the Eigenvalue.

III. Alignment system

While the flexible couplers can compensate for a certain level of misalignment, the alignment system needs to be accurate and ensure the misalignment is within the allowable range (determined by the flexible coupler). The precision and accuracy of the alignment will primarily stem from the tool used to measure parallel and angular alignment of the shafts.

In addition to an alignment tool, the alignment system is composed by methods to incrementally adjust the positioning of the compressors in multiple directions. To induce a shaft angle change, it is required that there be a process to adjust the elevation of the compressors in the four corners of its mounting locations. This elevation adjustment also assists in parallel alignment. The compressors also need to have lateral adjustment to horizontally position the shaft correctly.

IV. Torque monitoring

Since this test rig is monitoring the efficiency during operation, it is required that a device be integrated to monitor the output from the motor driving the system. It was requested by the client company that a torque transducer be implemented to detect the output torque. This transducer must be capable of operating up to the previously mentioned 50,000 RPM and 100 Nm. To ease efficiency calculations, this transducer must have the capability to connect to a computer that will give a user an easy ability to see the torque produced at the output shaft.

The Danfoss Turbocor compressors operate under a wide range of rotational speed. Table 1 shows the maximum torque and RPM specifications of various TT series compressors. The listed torque is the maximum output from the compressor, and the RPM is the speed at which that torque is made. This shows that the high standard of 50,000 RPM and 100 Nm will produce a good safety factor for each component, and therefore yield an acceptable overall factor of safely[3].

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

B. Design concept selection

Based on the requirements presented, Team 4 came up with 2 main design concepts for the test rig. The first one, considered as the ideal test rig, incorporates all the requirements demanded by the sponsor but due budget constraints it had to be modified during the spring semester. The second design, our final design, is defined as the first phase of the ideal design and remains with all the characteristics of the ideal concept but, as suggested by our liaison at Danfoss Turbocor, focuses on validating the alignment system and processes. The final design allows, in a probable future, the integration of all the components required to fully operate any of the TT series compressors.

I. Ideal Design

To measure the misalignment between both shafts it is highly recommended a laser alignment tool be used, since there is such an emphasis on precision. The laser aligner selected is in the median to lower end of laser alignment price and still allows a precision less than 20 µm. The SKF TSKA 31 alignment tool was selected and has an accuracy of ±5µm and comes in at a price of $3,500.

The TSKA 31 features innovative functions that expedite the alignment process. This will shorten system downtime while also accommodating a variety of users, regardless of the level of one's technical knowledge. First a user would fasten the laser reading devices to each shaft; next they would follow the manual to measure the shaft misalignment. Once this is completed, the tool will make calculations to determine how the compressor must be moved to align the shaft. Figures 4 & 5 shows an actual display from the TSKA 31, which is what the operator will see once they are proceeding with the alignment process.

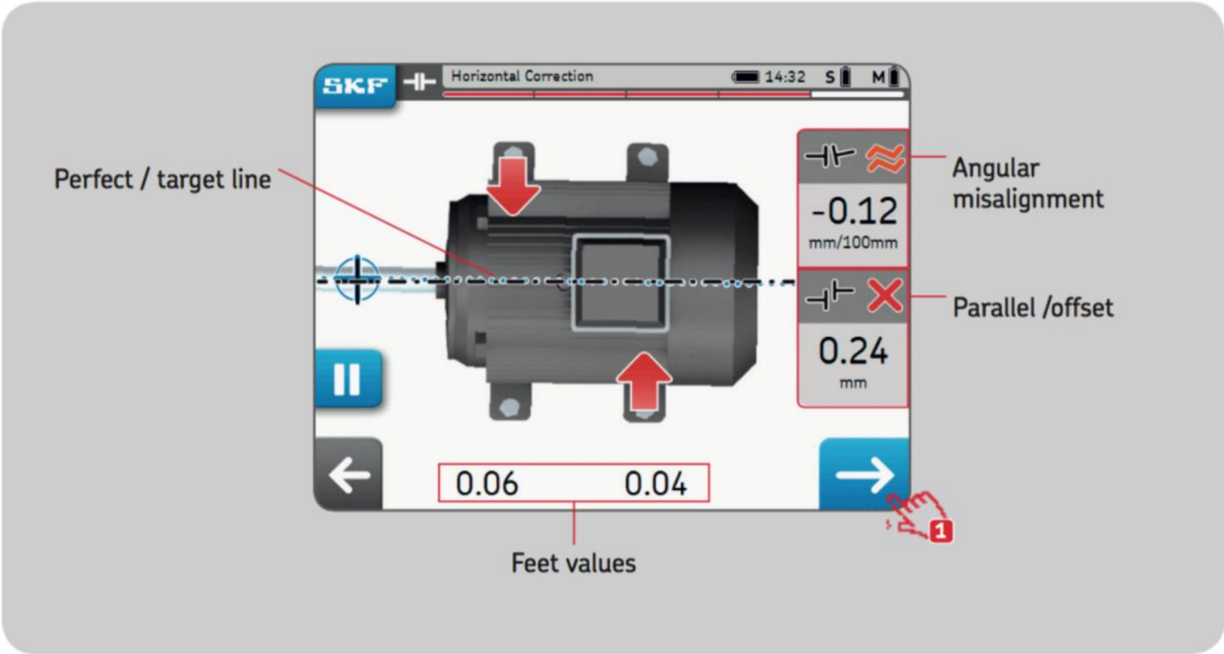


FIGURE 4: TSKA 31 HORIZONTAL CORRECTION GUIDE – TOP VIEW[3]

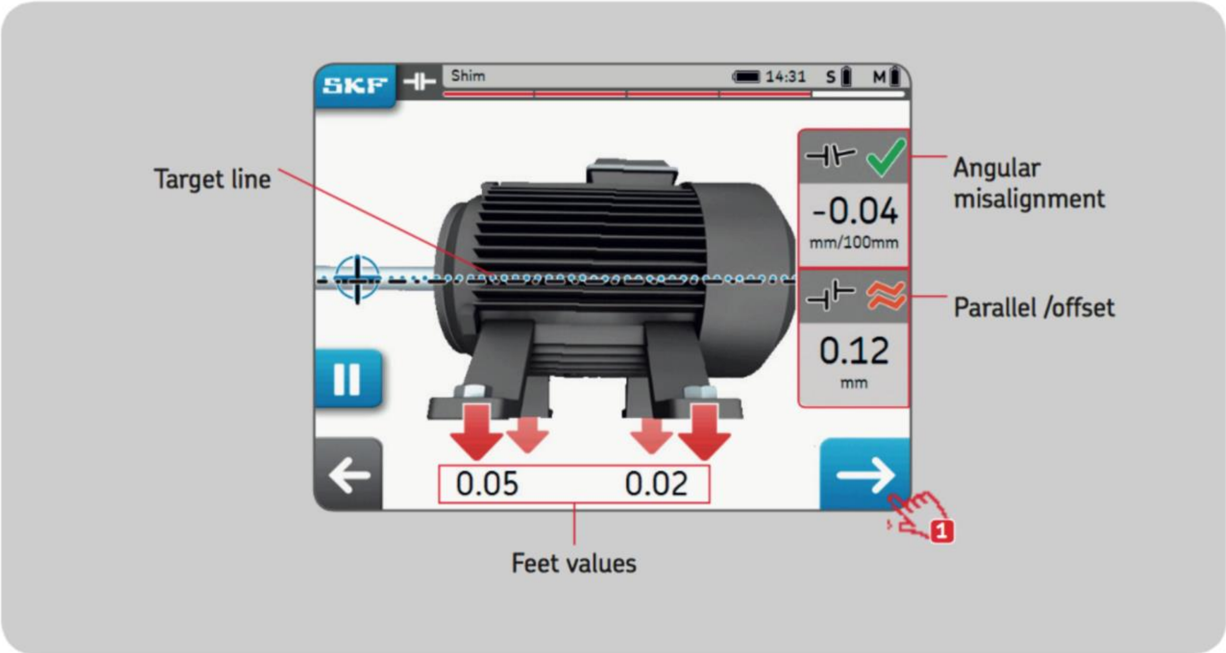


FIGURE 5: TSKA 31 VERTICAL CORRECTION GUIDE – SIDE VIEW[3]

As stated previously, to monitor motor performance it is necessary to integrate a torque transducer. The torque transducer must be able to handle the speeds up to 50,000 rpm while still being able to handle 100 Nm of torque. As previously stated, these values were chosen to over design the system and give a reasonable factor of safety.

Upon research of these components team 4 found that there is a caveat to these transducers. There isn't a transducer that can handle the given parameters and this is because when the angular speed is increased, the torque is decreased and vice versa. To accommodate this, the advisors at Danfoss Turbocor suggested that the team design a system that can handle multiple torque transducers for low torque and high RPM, and high torque and low RPM.

For the high RPM range, the Magtrol TMHS 308 (Figure 8) can handle 50,000-RPM speeds and a rated torque of 20 Nm, which was verified and deemed proper for this application by Turbocor. For the 100 Nm case, the Magtrol TMHS 311 (Figure 9) was selected for its maximum RPM range of 32,000 and a rated torque of 100 Nm.

FIGURE 6: MAGTROL TMHS 308/311[3]

Because the transducers have very different geometries, the team has designed spacers that will adjust the transducer elevation to align with the compressors' shaft axis. The TMHS 308 will need a spacer block with a height of 174.45mm. Because the mounting bosses on the 308 are under the transducer body, the elevation block will need to have clearance within the body to access the mounting bolts. The TMHS 311 will need an elevation block of 149.45mm in height. Both elevation pieces for the transducer will be tapped on their undersides and be bolted to a baseplate.

To carry the weight of two 300-pound compressors, the base plate must be designed with a material that has a good strength when in compression and that can be easily machined. Danfoss Turbocor's machine shop has a 2x2 inch square (boxed) steel (mild grade) tubing that has a 1/4-inch wall thickness and is perfect for the application of this project. The quarter inch thickness will allow for thread tapping and ensure that bolts can be used to fasten the majority of the baseplate.

The horizontal and vertical alignment systems are integrated into the base plate design. In the vertical direction, shims are used. While the thickness of the shim is critical, as it dictates the elevation increments, the other profile dimensions are also important. Figure 7 labels these dimensions that are dependent upon the fastener and mounting surface. C is the width of the opening that will slide around the mounting bolt. If A and B are too small, it will cause a large amount of force to be directed to a small surface area, which may fatigue the shim and cause it to lose width over time. Therefore B and A should be set to the size of the smaller surface that it is in contact with. In the test rig design, the shims will be between two pieces of 2”x2” square tubing, therefore it is unnecessary for dimensions A and B to be beyond 2”x2”

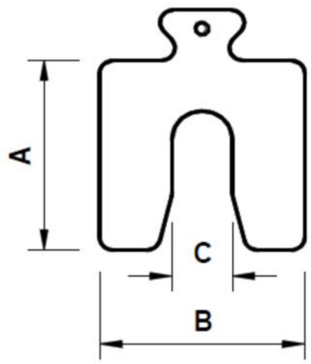


FIGURE 7: SHIM DIMENSIONS AND PROFILE[6]

The horizontal alignment mechanism is a setscrew system. This screw will push on the feet of the compressor in order to move it and for each compressor there will be a setscrew on each of the four feet. A solid piece of steel will be drilled and tapped for an accompanying bolt. The bolt will have a hex head that will allow the connection of a socket and wrench to turn it. A weld will mate the tapped steel piece to the base frame, and the bolt will be threaded through the hole. Each compressor will have four of these screw alignment apparatuses at the four corners where the compressor mounts to the frame. The screws will push into the square tubing that the compressor sits upon. To allow movement, yet still securely fasten to the frame; the hole for the mounting bolts will be drilled out to be larger than the bolts. This will allow a small amount of lateral movement, which will be sufficient for the alignment process.

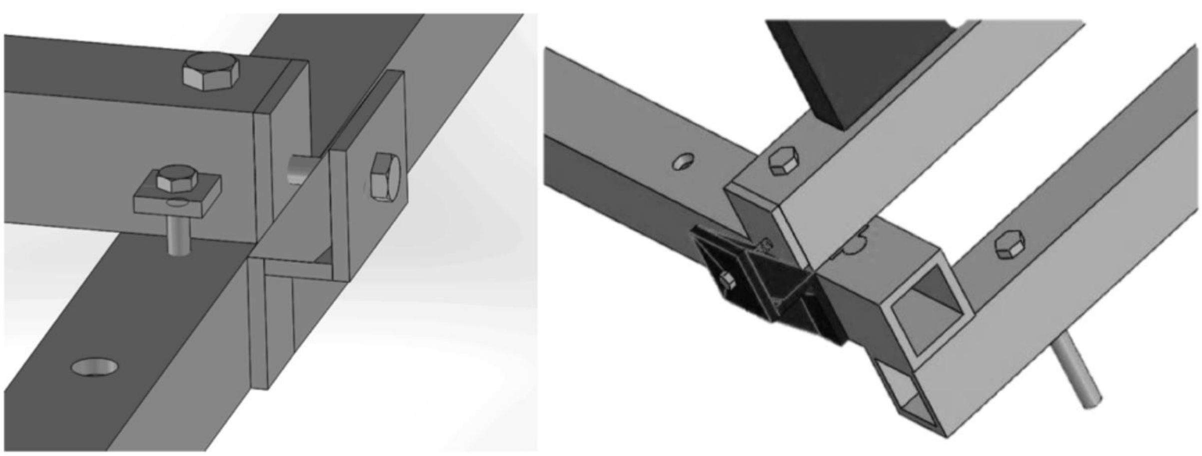
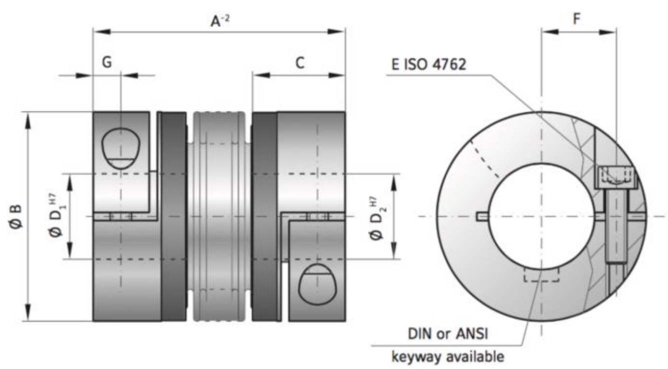


FIGURE 8: ALIGNMENT SYSTEM: SET SCREW FOR HORIZONTAL ADJUSTMENT[6]

Since the torque transducer will be the median between the two compressors the shaft of each compressor will need to be coupled to the shaft of the transducer. The coupler must be able to compensate for the misalignment that is going to be generated at such a high angular speed, up to 37,000 RPM. The R&W BK2 150 series coupler (Figure 14) was chosen because it can handle speeds up to 80,000 RPM. It also can tolerate up to 150 Nm of torque and be easily clamped to the shafts; since the bolting access is from the outer shell of the cylinder and not on the faces.



Overall length (mm)	A-2	95	107	144
Outside diameter (mm)	B	81		
Fit length (mm)	C	36		
Inside diameter possible from Ø to Ø H7 (mm)	D_1/ D_2	19-42		
Fastening screw ISO 4762	E	M10		
Tightening torque of the fastening screw (Nm)		70		
Distance between centerlines (mm)	F	27		
Distance (mm)	G	11		

FIGURE 9 : R & W BK2 COUPLER[6]

The rigid couplers selected are McMasterr-Carr Re-machinable Couplers and the shaft extenders are SUA 050 made of steel ASTM A108. All of these couplers were selected so both scenarios – with different torque transducers – become feasible with the base frame design. Figure 10 suggests how would it be the first and ideal design concept.

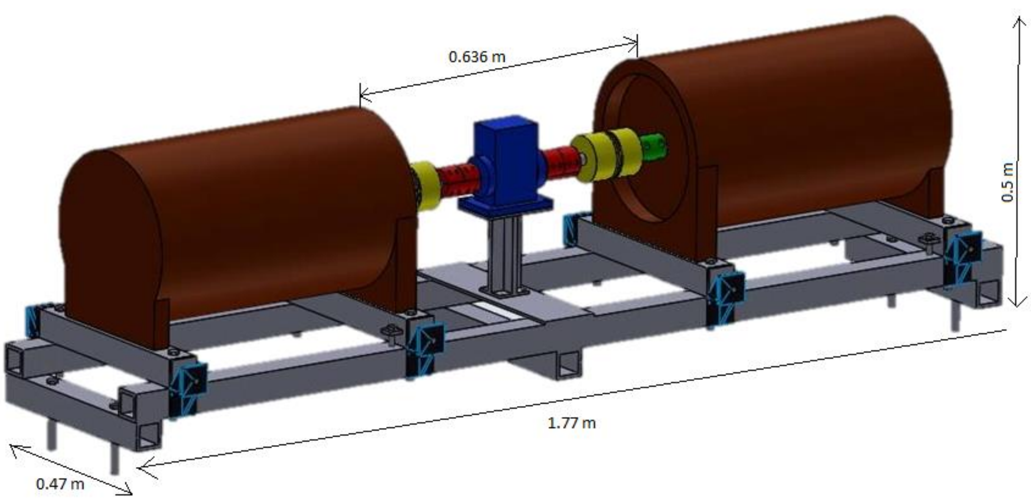


FIGURE 10: IDEAL DESIGN CONCEPT[6]

II. Alternative design

To measure the misalignment between the motors shafts a feasible alternative is necessary. The dial indicator system is definitely cost effective and can be found for about $30-$100 depending on the quality and is accurate to about 26 μm and that is definitely within the design specification parameters. Using dial indicators for shaft alignment has a learning curve; the operator must use his technical knowledge for using measurement tools to interpret dial readings into compressor position adjustments.

FIGURE 11: DIAL INDICATOR[5]

In order to qualify and validate the alignment system proposed, the team came up with an alternative concept with a dial indicator alignment method and without the torque transducers integrated to the system. This alternative design allows the alignment method to be tested and gives the sponsor the possibility to choose when it should acquire the torque transducers. It also dramatically reduces the overall cost of the project, further reducing the components to just 2 rigid couplers, 1 flexible coupler and 2 shafts.

III. Design analysis

It was recommended by the faculty advisor, Dr. Patrick Hollis, which analysis is to be done to understand the effects upon the shaft orientation that the different thickness of shims. The TT series compressors that the team is designing the system around, has its shaft positioned directly over the front base frame mounting location. Because of this, if one of the shims is on the front feet of the compressor, it will elevate the shaft by the same amount of the shim thickness. Not only will the shaft have a change in elevation, but also an angle will be induced. To predict the changes, trigonometry was used to analyze the changes in shaft orientation. Equation 1 shows the angle induced if a shim of width 485mm is used under the front mounting locations. The distance of 485mm is the displacement between the front and back mounting locations.

$$\tan^{-1}\left(\frac{a_1}{485mm}\right) = \theta_1 \quad (1)$$

Elevation on the rear of the compressor was also analyzed. This analysis found that the elevation at the end of the shaft changes with a fraction of the size of the shim used. Trigonometry was used and the angle induced from a shim of width a_2 is expressed in Equation 2.

$$\tan^{-1}\left(\frac{a_2}{485mm}\right) = \theta_2 \quad (2)$$

By using the angle induced from rear shims, the shaft elevation will change by the amount "*y*". The value *c* is the shaft height above the mounting surface.

$$y = c - cos(\theta_2) \cdot c \quad (3)$$

To finalize the adjustment analysis, Table 2 shows the values that will determine the orientation of the shaft. The chosen shim widths were the smallest shim sizes that the team found from McMaster Carr. By using the smallest shims for analysis, the team can see what the smallest possible amount that the shaft can be vertically adjusted. It is still undetermined by the team if larger shim sizes will be needed. However, the team has determined that these shims (10 μm, 25 μm, and 250 μm) produce very small increment amounts that are acceptable.

Table 2: Shim sizes and correlating shaft increment amounts

Front Shim Width (um)	Shaft Angled Induced (degrees)
10	0.0012
25	0.0023
250	0.0143
Rear Shim Width (um)	Shaft Angle Induced (degrees)
10	0.0012
25	0.0023
250	0.0143
Rear Shim Width (um)	Shaft Elevation Induced (mm)
10	0.0001188
25	0.000436425
250	0.016870138

II. FINITE ELEMENT ANALYSIS (FEA)

Once the final design of the baseplate was sketched in Pro Engineer, the model analysis commenced. Finite Element Analysis was conducted under the static load of the compressors just to validate that the base plate material and design selection was adequate. This is the only analysis that can be done at this time since the torque transducer, coupler, and compressors are qualified to run under the specified loads. The von Mises stresses (Figure 16) and displacement (Figure 17) models were created and it can be noted that the von Mises stresses are of no concern since the maximum displacement is going to be minimal due to the low magnitude of 1.034E-7 meters. This qualifies that the selected 2" steel tubing is an acceptable material and the design is safe.

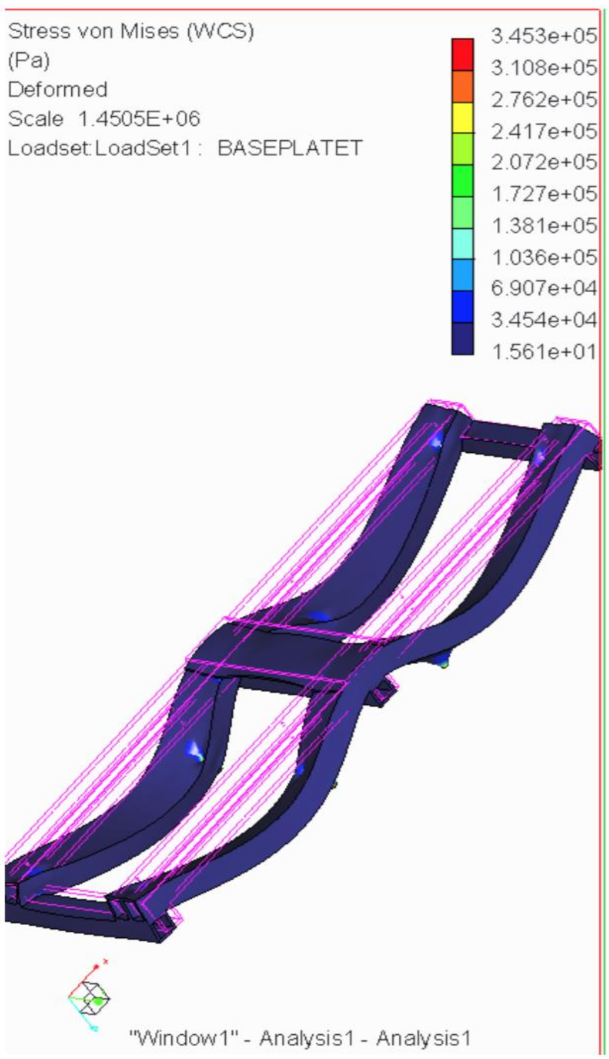


FIGURE 12: FEA, VON MISES STRESSES

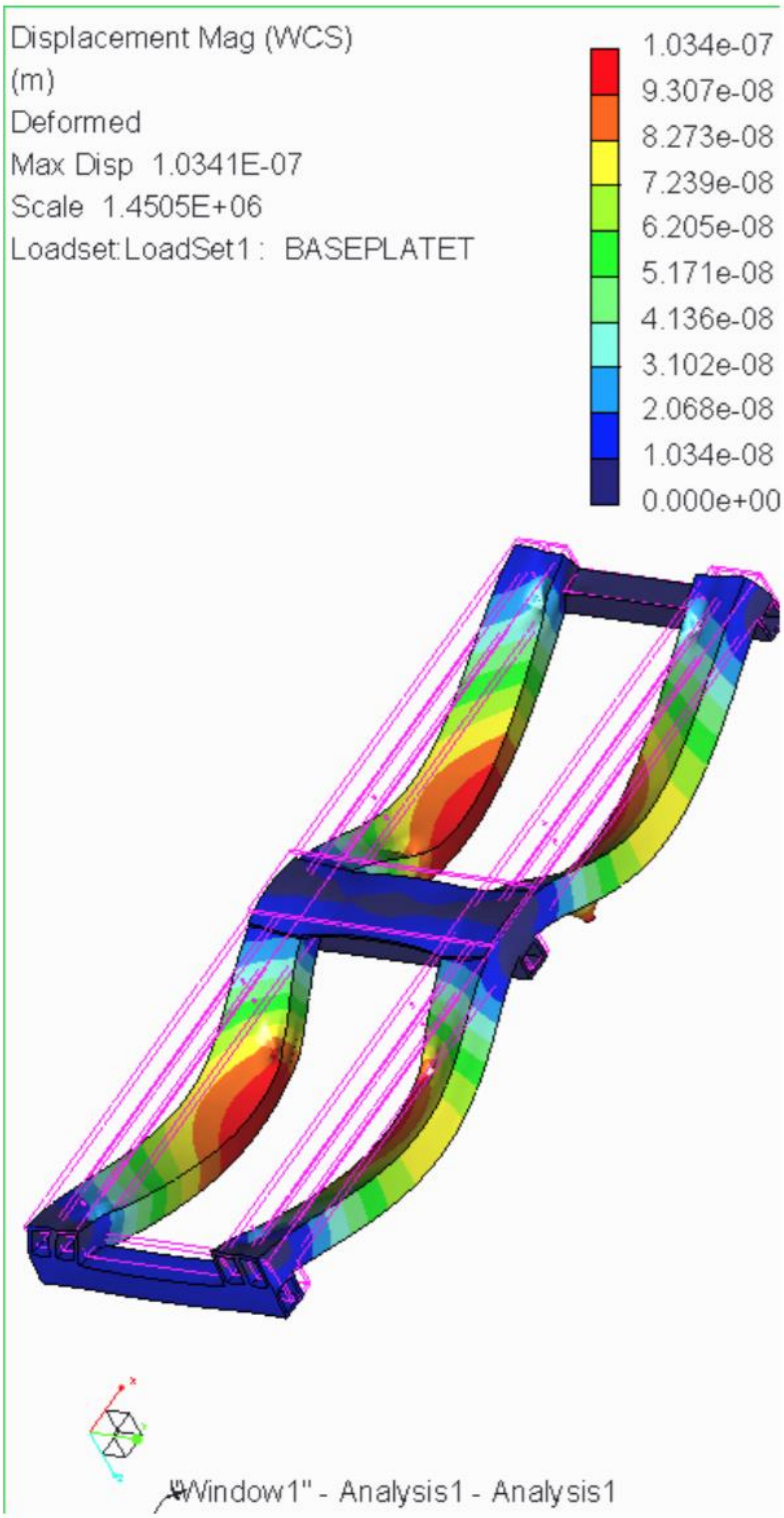


FIGURE 13: FEA, DISPLACEMENT

To further test the reliability of the results, the strain energy was plotted versus the number of passes the program used to compile the data. This was run to check against a 10% convergence and as one can see in Figure 18 the strain energy converges and levels off at the 8^{th} pass, which gives a reasonable accuracy of the data.

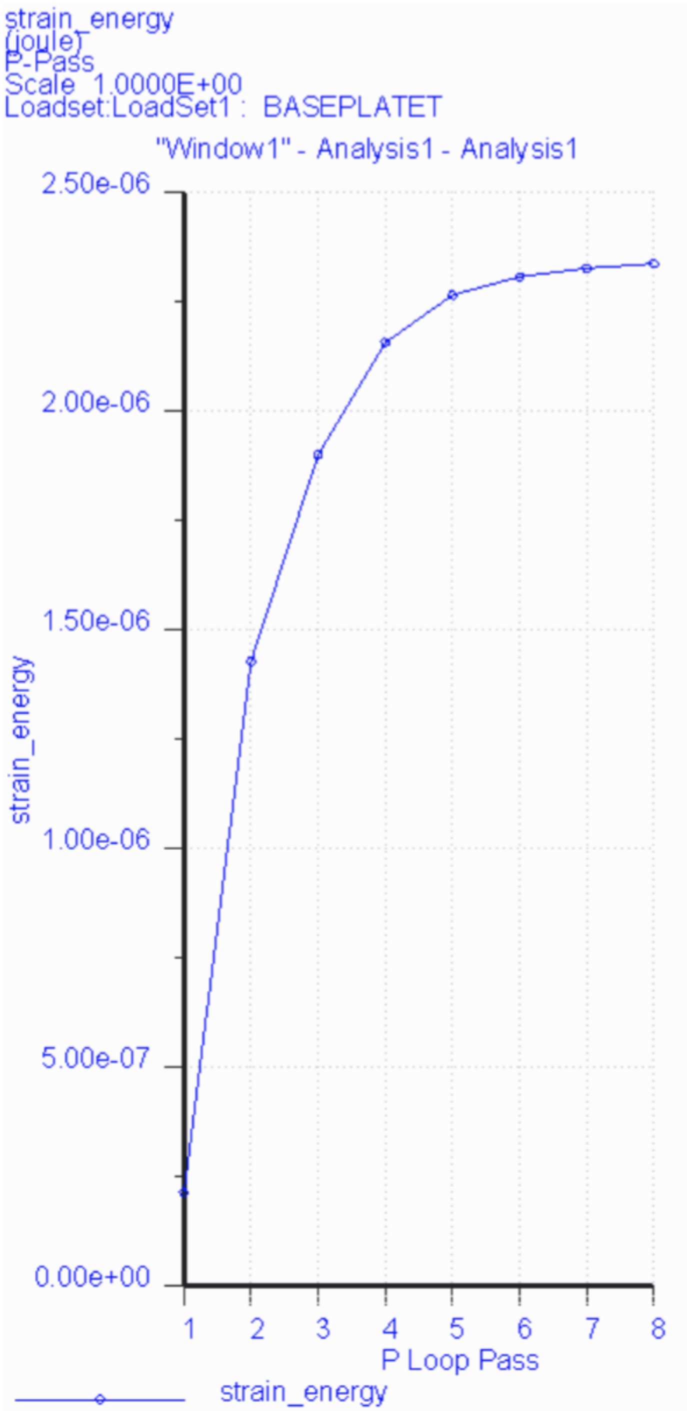


FIGURE 14: STRAIN ENERGY VALUES AT 10% CONVERGENCE

III. FINAL DESIGN

The final design and project scope were changed a few times over the course of the semester; this was due to unforeseen budget constraints and miscommunications. The Senior Design Team 4 faced some challenges during the design process and through the help of the faculty advisor and the project sponsor the team worked towards the best solution for each problem. The first major problem faced was due to the high cost of the project with torque transducers. Originally the project had the goal to include equipment that could measure the torque and the rotation speed between the compressors. The team came up with an idea of using a torque transducer between the compressors; the major problem with this equipment is that there is no torque transducer available in the market that can handle high-speed rotation and high torque. Instead of that there are two types of torque transducers, one that can handle high-speed rotation and low torque (Torque transducer 1 at 50,000RPM and 20Nm) and the other one that can handle low speed rotation and high torque (Torque transducer 2 at 20,000 RPM and 100Nm). Turbocor wants a method that can measure the compressor's motors in all possible scenarios so the team developed a test rig, which can fit both torque transducers. The other problem using two torque transducers is that torque transducer 1 (Magtrol 308) has a shaft with d1 = 10 mm and the torque transducer 2 (Magtrol 308) has a shaft with d2 = 20 mm. So for each torque transducer the system would require a unique pair of rigid couplers to connect the shaft from the torque transducer to the flexible coupler. This design can be seen in the Figure 15. Each torque transducer has an estimated price of $8,000.00, so this design was not practical because the transducers would cost $16,000.00 (for the pair). Instead the sponsor suggested to first prove the alignment system is worthy and if it matches the specification (0.2mm radial, 1 degree angular and 1mm axial) the team would have the budget to purchase the torque transducers in the future.

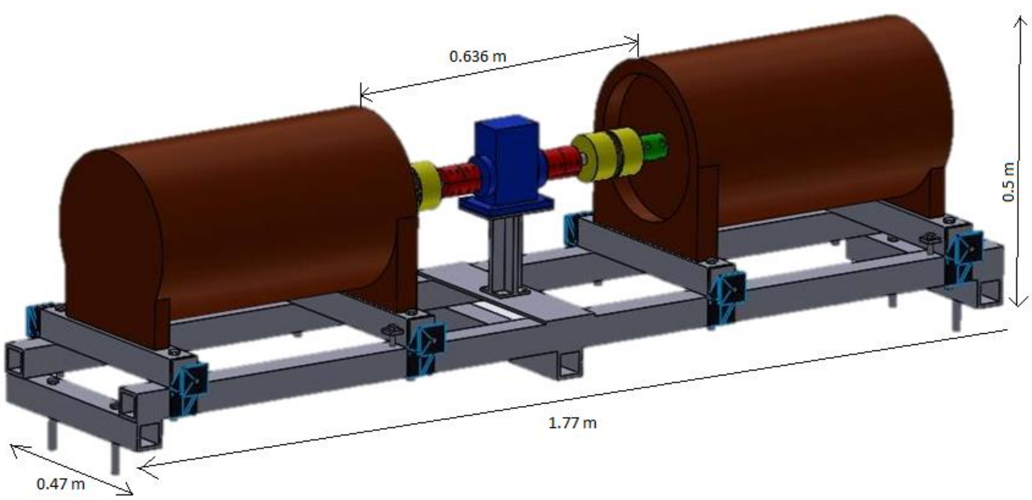


FIGURE 15: FIRST DESIGN[3]

To qualify the alignment system the team changed the project so the test rig would not have a torque transducer in the middle, a single flexible coupler is used instead. So the system would be simpler, the shaft of the compressor connects to the rigid coupler, which connect to another shaft and then the flexible coupler.

The second design was presented to Turbocor and the sponsor pointed out a problem. During the maximum rotation speed the rotating assembly will experience vibrations at 40,000 RPM (or 667Hz). For safety reasons, the natural frequency of the rotating system has to be higher than the operational frequency of the compressor shaft. If the system frequency is larger than the compressor natural frequency, the system it could hit resonance, causing catastrophic damage. The team reevaluated the design and noticed that the theoretical natural frequency was 530Hz, so this project was not safe to operate. To solve this problem, the team changed the design so the length of the rotating shaft is smaller and thicker; both changes increased the stiffness of the rotating assembly and consequently the natural frequency was heightened. With this changes the theoretical natural frequency was 840Hz. The Final Design can be seen in the Figure 16.

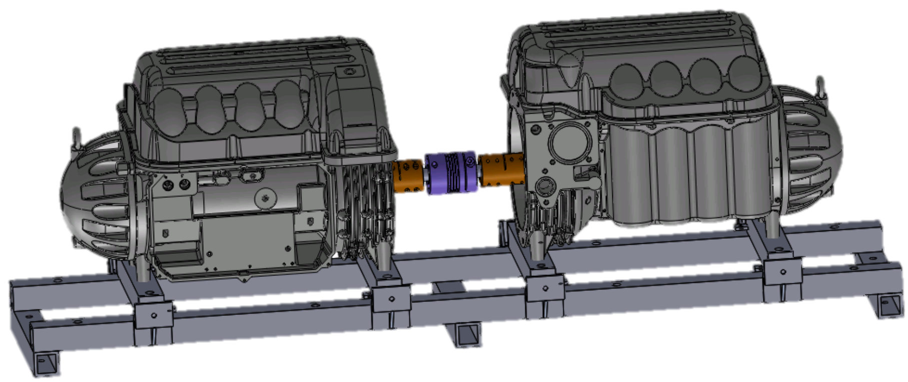


FIGURE 16: FINAL DESIGN[3]

To measure the actual natural frequency of the rotating assembly Turbocor[6] ran a test using a hammer with sensors. To perform this test the rotating assembly was connected to one shaft, which was suspended by rubber strings, simulating the effect of the magnetic bearings. The actual natural frequency observed was 940Hz, concluding that resonance will not be an issue. The method of the test among the results can be seen in Figure 17.

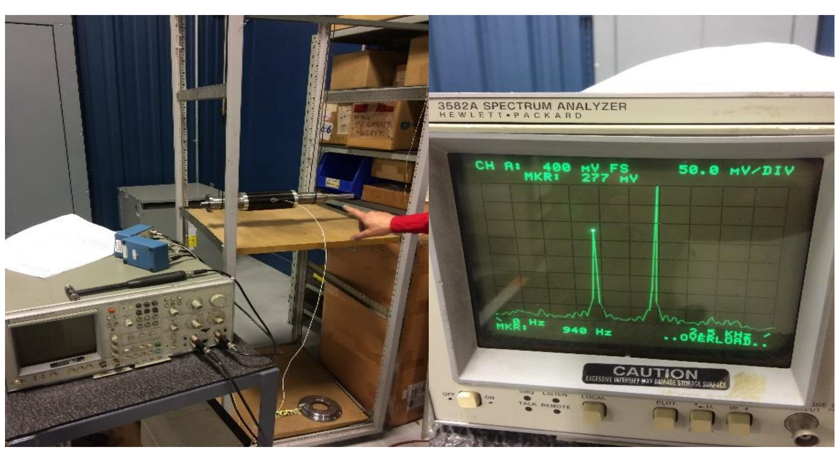


FIGURE 17: NATURAL FREQUENCY TEST AND RESULTS

A. Base Frame

The base frame is made of 2x2x0.25" steel and its main function is to settle the compressors in their proper position and rigidly fix the system to the ground. The raw materials were received at Turbocor's facilities and were machined there as well. The machining process mainly leveled the surface to minimize any inconsistencies on the base frame runners, drilled the holes to settle the compressors, drilled the holes to bolt to the ground, and welded the structure to guarantee the stiffness required. Turbocor has its desired distributors for raw materials that they use (McMaster-Carr) and they also stock this material in their facilities, the ordering process took less than one week and the machining process took longer than expected due to machine shop priority for the company.

B. Brackets and Screw Jacks

To move the compressor during the alignment, process the team designed brackets and screw jacks, which perform the horizontal and vertical movement respectively. Both parts are made of 1/4" steel and are welded to the structure. The brackets were difficult to manufacture because each bracket is made with smaller plates, which needed to be cut and welded to be one final part. For the actual completed base frame, the screw jacks were omitted due to time constraints and the proof of successful pry-bar use. The drawings include the screw jacks incase Danfoss desires to add them in the future. Figure 17 shows a view of the base frame with the brackets and screw jacks.

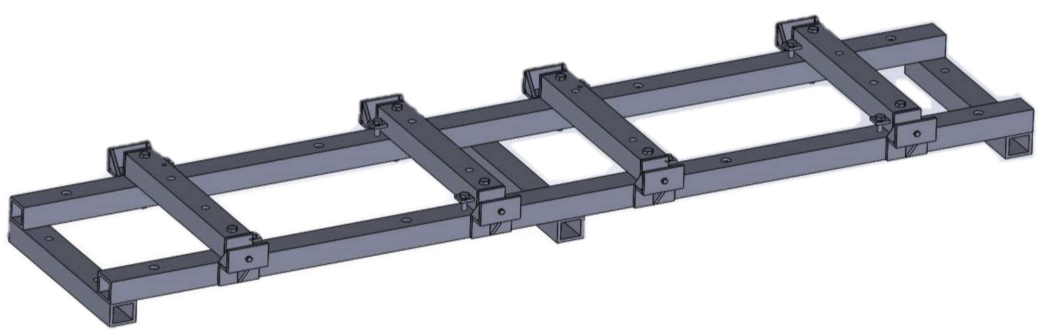


FIGURE 18: BASE FRAME WITH THE BRACKETS AND SCREWJACKS

C. Rotating Assembly

a. Shaft

The shaft to connect the rigid coupler to the flexible coupler is made of 1566 hardened steel. The system requires 2 shafts. Turbocor ordered the raw material and machined the shaft so the overall diameter is 25.4 with a 50µm precision.

b. Rigid Coupler

The rigid coupler selected by the team is the Re-Machinable Rigid Coupler supplied by McMaster-Carr. Its main function is to connect the compressor's shaft to a thicker shaft, which will connect to the flexible coupler. The stock inner bores were 22.25mm (which is the diameter of the shaft of the compressor). To increase the natural frequency of the system the team decided to increase the stiffness of the system by increasing the diameter of the shaft that connects the rigid coupler to the flexible coupler. To do so one of the inner bores was re-machined so it has 25.4mm diameter. The rigid couplers need to be balanced to withstand the high operational speed rotation. This system uses 2 rigid couplers.

c. Flexible Coupler

The flexible coupler selected by the team is the BK2 150 Bellows Coupler from R&W. This component can handle 150 Nm torque and comes balanced at 10,000-RPM. For higher speed operation the coupler will need to be re-balanced. A view of the final rotating assembly can be seen in Figure 19.

FIGURE 19: ROTATING ASSEMBLY

D. Assembly Procedure

The system is designed in order to allow for the fastest method to change the compressors that will be tested. This is possible because each compressor is mounted to the test rig by 4 bolts. To assemble the system, one compressor is rigidly fixed to the test rig to begin. During the alignment procedure this compressor will not move unless necessary. After fixing the first compressor, the second compressor will be mounted to the test rig as well. The next step is connecting the rotating assembly. The rotating assembly is composed of the two rigid couplers, the two shafts and the flexible coupler. The flexible coupler can be loose on one side, which allows an independent rotation between the compressor's shafts; this is a requirement to perform alignment with the dial indicator method.

E. Alignment Procedure

Alignment of compressor shaft requires the use of two dial indicators. The indicator should have a minimum accuracy of 0.001" and have an attachment that allows it to be rigidly mounted to a curved shaft surface. To properly function, the indicator attachment arms needs to have multiple pivot points, this allows the attaching base to mount to one side of the rotating assembly, and the dial to reach around to the other side.

The process for alignment begins with using the dial indicators. The indicators must be attached to one side of the rotating assembly and be able rotate 360 degrees with the shaft. It is predicted that the user preforming the alignment will need to mount the dial indicator base to a rigid coupler and rest the dial-reading arm upon the opposing side of the rotating assembly on the flexible coupler.

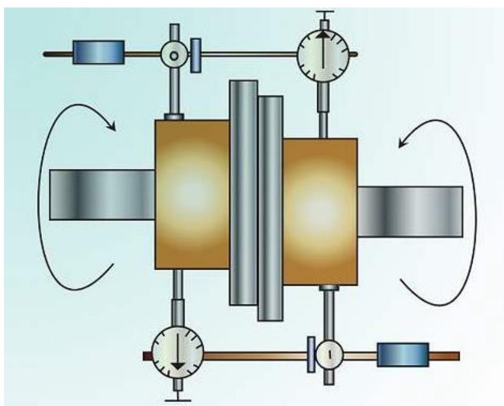


FIGURE 20: CROSS DIAL ALIGNMENT SETUP[4]

Dial reading arms must be 180 degrees from one another and perpendicular to the surface they are contact with. To achieve this, the team used their own developed method. Begin by clamping the flexible coupler with calipers, where the two arms come in contact is 180 degrees from each other. The users must do their best to accuratley mark on the flexible coupler these points. Next, remove calipers and set up the dials. See figures 20 and 21 of the set up. The tip of the dial arms must be on the 180 degree marks.

Negative dial readings are when the plunger arm extends, and the dial rotate counter clockwise. Positive dial readings are when the plunger is pushed in, and the dial rotates clockwise. This is important when getting the TIR values, which can be positive or negative.

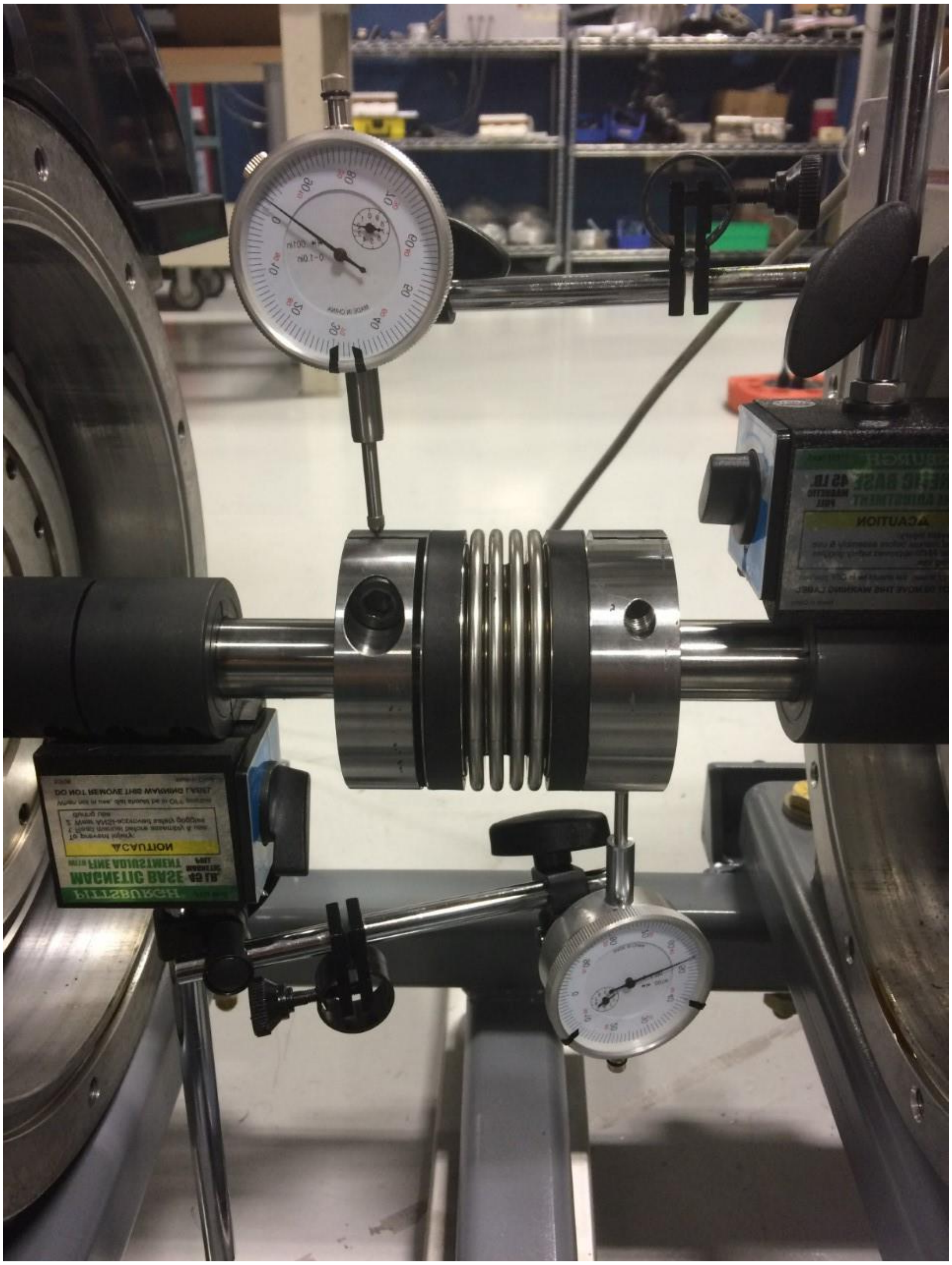


FIGURE 21: ACTUAL INDICATORS ON THE ROTATING ASSEMBLY AT DANFOSS TURBOCOR (NOTE: NOT ZEROED)

F. Vertical Adjustment

Vertical positioning adjustment must be done before horiztonal. This process relates to adjusting the vertical position of the compressor with shims. These shims will go under the upper cross members at the junctions they are mounted. After the dials have been mounted properly they must be zeroed. To zero, the dial face can be turned by hand until the "0" is in line with the tip of the dial arm. To fine tune, there is an adjustment screw on the horizontal indicator support that will adjust the zero. Zero each dial at the 12 o'clock position. Be careful to no disturb the position of the dial after it is mounted.

With the dials zeroed, begin the process to find the TIR, "True Indicated Reading". Rotate the assembly with the dials 180 degrees. The dial that was on the top is now on the bottom, note this reading and divide it by two. This the first TIR value. The value is divided because the displayed dial value is the amount of shaft displacement doubled. Repeat this for the other dial mounted to the opposing side of the rotating assembly. Zero at the 12 o'clock position and rotate until it is at 6 o'clock. Note this value and divide it by two. Now, both TIR values for both compressors has been obtained.

After the two vertical TIR values for are found, a formulated process is used to find the amount one compressor needs to be adjusted by. Technically, only one compressor needs to be adjusted. It the misalignment is extreme enough, the opposing compressor can be adjusted to compensate. The equations and process to obtain the adjustment values can be seen in figure 22, "Cross Dial Shaft Alignment Form". SM TIR is the TIR from the dial mounted on the left side compressor. MM TIR is for the dial mounted on the right side compressor. It is recommened to use the same clock orientation everytime alignment is done. Do this by having the statinoary machine on the left. The side that away from the user is the 3 o'clock position and the 9 o'clock position is facing the user.

The A value is the distance across from the tip of one dial reading arm to another. To find this value, the team developed their own method. Hold a marker to the tip of one of the dial tips, a second user should rotate the assembly as the other user maintains the same position of the marker. Once the assembly is rotated 180 degrees, measure the distance from the tip of other dial reading arm to the end of the marked line. Once all the euqations have been completed, the final values obtained will be the amount that the feet should adjusted. If the value is positive, shim the compressors up, if it is negative, remove shim material (it is recommeneded to insert shims before beginning alignment, this way the compressor position can be lowered to achieve the negative value).

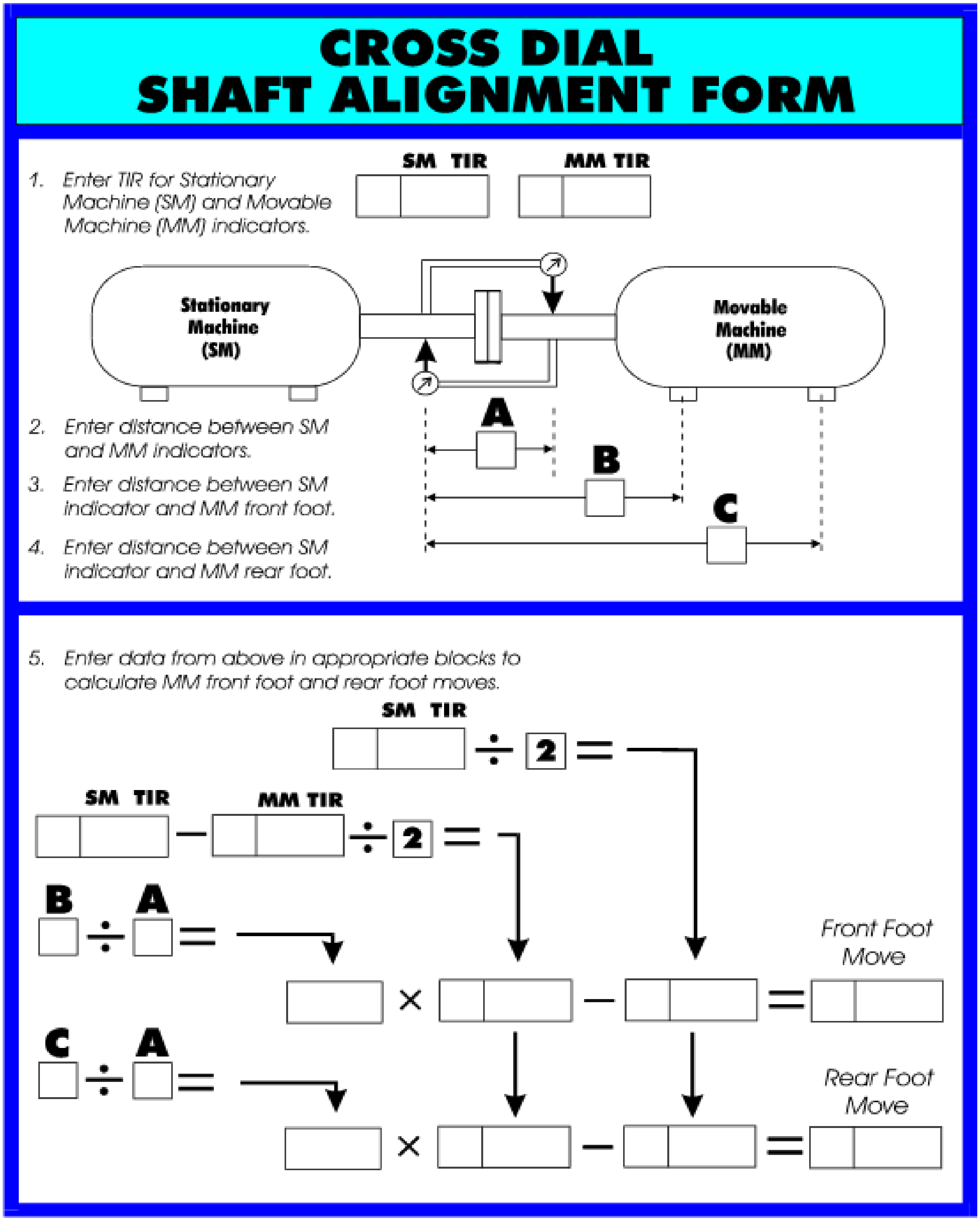


FIGURE 22: CROSS DIAL ALIGNMENT FORM[1]

G. Horizontal Adjustment

The horizontal alignment is similar to the vertical adjustment position. Zero the dials in the 3 o'clock position. Rotate the dials to the 9 o'clock position and record the values. Divide the values by two and these will be the TIR values. Use the "Cross Dial Shaft Alignment Form" again for the horizontal position adjustment. Negative values move the compressor towards 9 o'clock, and positive values are toward 3 o'clock.

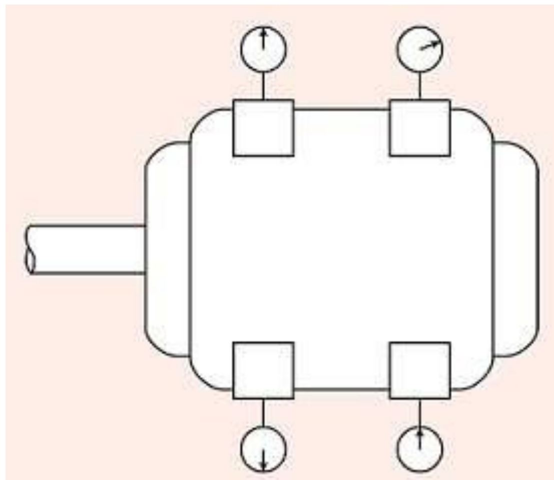


FIGURE 23: DOWNWAR VIEW OF COMPRESSOR WITH DIAL INDICATORS[2]

When the adjustment values have been obtained, this tells of how much the compressor will need to be laterally shifted. Rotate the setscrews in to push the compressors in the appropriate direction. Monitor this movement by attaching one of the rigid couplers to frame, and place the reading arm on the face of a cross member's end cover. As the set screw is rotated, watch the dial reading to achieve the calculated adjustment value. Figure 23 above shows the various locations that one may need to mount the dials to monitor the adjustment values.

H. Reliability

Being that most of the components of this test-rig are made of steel, the machining process was relatively simple. Low carbon steel was chose for it's high yield stress and for the purposes of the test rig there have been no forces of concern that will even come close to causing plastic deformation in the steel components. The max stress calculated from the FEA analysis shown in Appendix E gives a maximum stress value of 0.3 MPa in the base frame and 118 MPa in the setscrew alignment mounts. Looking at the fatigue limit of steel below in Figure 5, the carbon steel endurance limit is beyond the actual maximum load stresses on the steel components so it can be determined that there is no load life cycle concern with the test-rig.

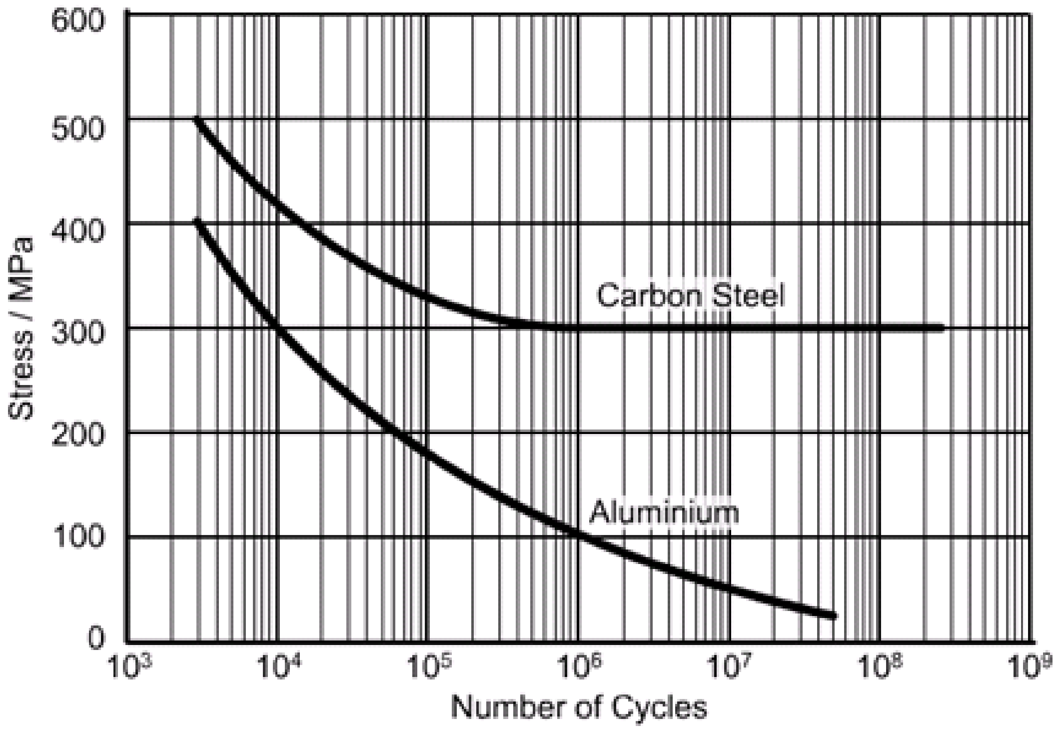


FIGURE 24: STRESS (MPa) PER NUMBER OF CYLCES[2]

Looking at the FMEA analysis performed by team 4, there are some areas of concern, which are unlikely but were deemed to have a 10 in severity. These potential issues are observed in the base frame, compressors, and shafts. In the base frame, the possible modes of failure are structural defects, rusting, and vibrations. Being that the base frame carbon steel runners are machined and welded, the processes could have adverse effects on the material's integrity. It is highlighted that fracture and bending could possibly occur due to mishandling and warping. This is not something that can be detected easily but the machinery and welders at Danfoss are of the highest caliber and the cause for a structural defect is highly unlikely (it received a 1 in occurrence). To go along with potential mishandling, rust can possibly plague this base-frame being that it isn't made of stainless steel. It is recommended that much care go into this frame; maintenance and an anti-rust paint should be applied to protect the rig.

Being that this designed to run at high speeds (50,000 rpm) and also at high torque (100 N-m), vibrational effects must be considered. The team is confident that the system can reach full operation for alignment purposes being that the natural frequency is 940 Hz, when the actual compressor natural frequency at around 667 Hz. Other considerations should be taken when Danfoss decides to run at full speed since the factor of safety calculated is only 1.13.

The compressor shaft and hardened shafts are a very key component to the rotating assembly of the rig. The hardened steel shafts could potentially fail, like the steel in the base frame, with a structural defect that could take place in the machining or balancing process. Since these shafts are suspended in air and attached to a flexible and rigid coupler, a structural defect that can cause fracture or bending can potentially launch a projectile. To combat this, the team has proposed a safety shielding made of mild grade steel to encase the rotating assembly and protect the operators in the event of a launched projectile. The compressor shafts are finely balanced and are attached with sensors that can stop the motors if a radial force of 200 lbf is experienced. The hard stop with all rotating components could cause damage to the compressor if the force is large enough. This is severe since these compressors are very expensive. As long as the machining process and assembly goes according to plan and all operators adhere to the alignment process specified in the operations manual, the test rig should be reliable and safe.

I. Financial Analysis

The total cost of the project thus far is $1,046.76. This price includes all hardware (nuts, bolts, washers, etc.), couplers, baseframe steel, and shaft material. As you can observe in Figure 6, the couplers and base frame make up the majority of the cost at $414.48 and $352.14. The shim stock was ordered in large amounts at multiple thicknesses so that the increments of height adjustment would range sufficiently. Fasteners and shafts were not a burden on the team for pricing.

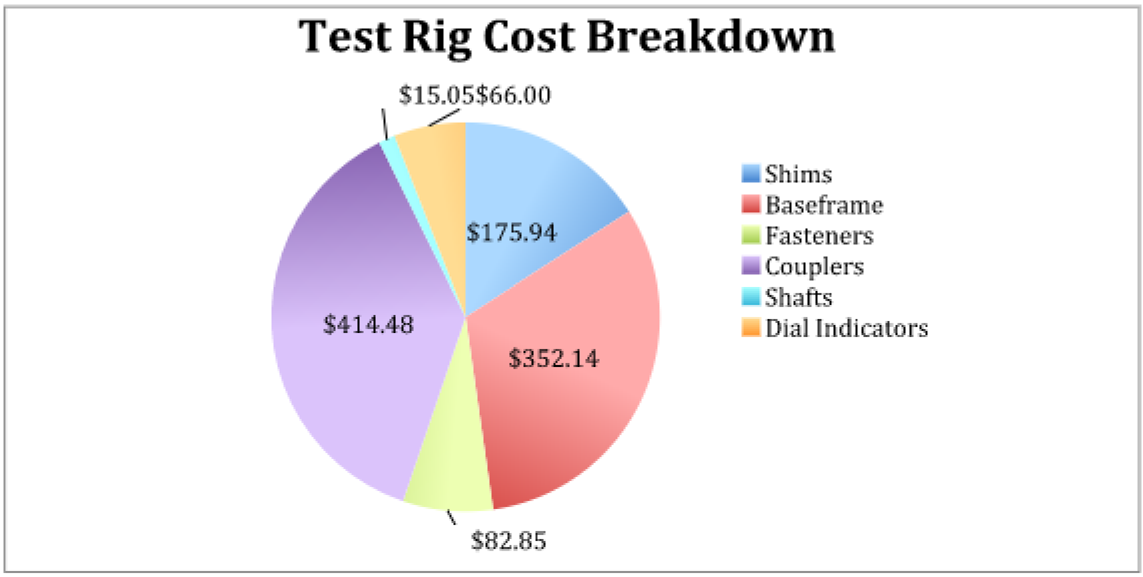


FIGURE 25: TEST RIG COST BREAKDOWN

Team 4 was never given a specified budget for the project and instead selected parts were purchased at Danfoss's discretion. As mentioned previously, the original design proposal called for two MAGTROL torque transducers. Each of these transducers was quoted at $8,000.00 and this would take the total for the design up to around $17,000.00, which was deemed too expensive at the moment by Danfoss. Another component that was omitted in the project due to price was the TSKA-31 laser alignment tool. It was valued at $3,500.00, but as previously stated the double dial indicator method is to be used in place of this. Overall the design has been cut by $19,500.00, but the base frame can still accommodate the original design if Danfoss decides to purchase the laser aligner and transducers.

There is not a current test rig that can handle the speeds that Danfoss requires and it was told to the team that the first phase in designing this test rig rung up a price near $70,000.00 - $80,000.00, as told by our liaison. This current design is simple, economical, and by design it is to be practical.

IV.DESIGN OF EXPERIMENT

A.Modal Testing

Three procedures were used to incrementally test the High Speed Motor Test Rig and its components. Since the focus of the final design is upon achieving high speed rotation and shaft alignment, the tests are focused upon the natural frequency of the rotating assembly and the alignment process of the shafts.

Once the rotating assembly components were obtained, they were connected to a TT 700 series shaft at Danfoss. This natural frequency of this assembly was then found through Modal testing. Modal tests are performed on objects by adapting an accelerometer to the object, and then striking with an impact hammer. From this, the modal (or natural) frequencies, modal masses, modal damping ratios, and modal shapes can be determined. These values are essential to successfully designing rotating components free from harmonic resonance.

This test works by striking the object with a hammer, where there is a known frequency content applied to the object. As the vibrations travel through the mass, some locations in the body will experience structural resonance, here there will be an amplified response, and this will be read in the response spectrum.

Danfoss setup this experiment by suspending the assembly from two cords, attaching the accelerometer, turning on the Modal testing system, and striking the assembly. It was a quick and essential test. Figures 26 and 27 show the set up and the results screen.

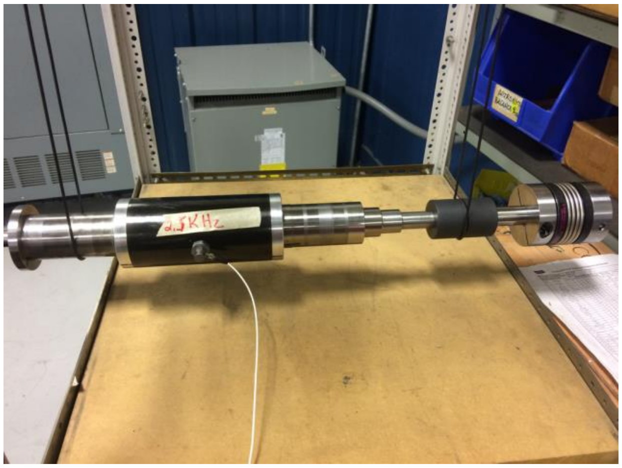


FIGURE 26: COMPRESSOR SHAFT WITH HALF OF THE ROTATING ASSEMBLY AND ACCERLEROMETER. TESTED AT 2.5 kHz OF APPLIED FREQUENCY.

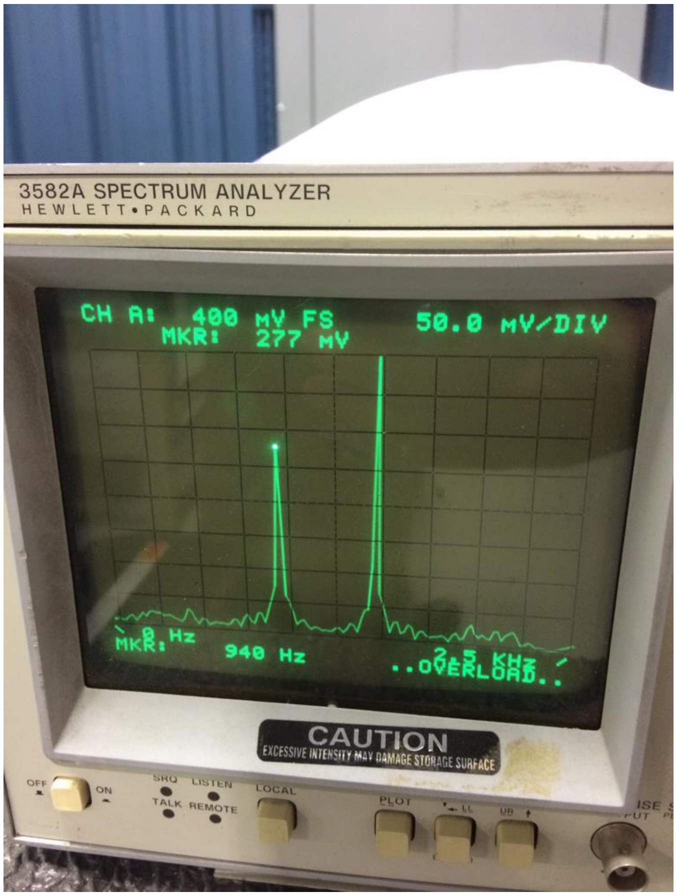


FIGURE 27: 940 Hz FOUND AT THE END OF THE ROTATING ASSEMBLY AT THE FLEXIBLE COUPLER.

B. Weighted Shaft Levitation

The second test performed was done at Danfoss to calibrate the shaft levitation with the weight of the couplers upon the shaft. When a compressor needs to be ran, the first steps involve calibrating the magnetic levitation of the shaft. Sensors within the system read the forces upon the shaft and determine how much magnetic force needs to be applied in a variety of directions to perfectly balancing it within the compressor. This ensures that the shaft is a even distance away from the surrounding walls that enclose upon it. This process was more challenging than anticipated. It was understood that this levitation was out of the ability of the students and therefore must be done by an experienced individual at Danfoss. The problem that occurred was was from the new forces that the coupling imposed upon the shaft. After trouble shooting, the levitation calibrating was finished. This took a Danfoss employee three days to complete. When the team returned to see the success, they were told that some parts of the software had to be bypassed, and that although it wasn't the best calibration it would be good enough for the alignment process to occur.

These unexpected issues were a surprise to the team and Danfoss. Because the calibration result was not ideal, but good enough, it is likely that if high speeds are attempted there will be issues with the centering of the shaft levitation within the compressor housing. Issues like these were far beyond the control of the team. It was never a concern to the team because it was understood that this task would be taken care of by Danfoss. Future acheivement of high speed rotation is highly depended upon more accruacte levitation calibrating with the rotating assembly.

C. Alignment Testing

After the shaft alignment process was done, a testing method needed to be done to evaluate the results. The only true way to test the alignment, was the repeat the alignment process to check for new misalignment. Comparing TIR values of the two dials for vertical and horizontal alignment tells how much the alignment changed by if the system was ran, or if there is an inconsistency in the alginment processes. The alignment inconsistencies that could occur are bar sag in the dial indicator or failure to follow the dial indicator set up process. To truly have the most percise alignment, the use of a laser aligner is highly recommended. The capabilities of the TSKA 31, previosly recommended by team 4, surpass most dial indicators for accuracy and eliminates many human errors that can occur with dial indicators. Team 4 recommends that no high speed testing be done without the alignment by a laser aligner and its accompanying software.

The expectations of this test were hopeful. Much research was put into understanding how to perform alignment, knowing the equipment needed, and performing the crucial steps along the way. Although the team could not purchase the laser equipment needed, they were able to focus on the weak points of the dial indicator solution and hopefully combat them to achieve the best results possible. Goals were set for the angular misalignment to be less that 0.8 degrees, and lateral misalignment to be 0.005”. These values were selected from what various industries often use for their alignment systems. Another way the team came to these values was by using the alignment adjustment equations, inputing them into an Excel file, and playing with possible results to see the outcome. This helped the team understand how much postion adjustment actually changes the shaft position by an induced angle.

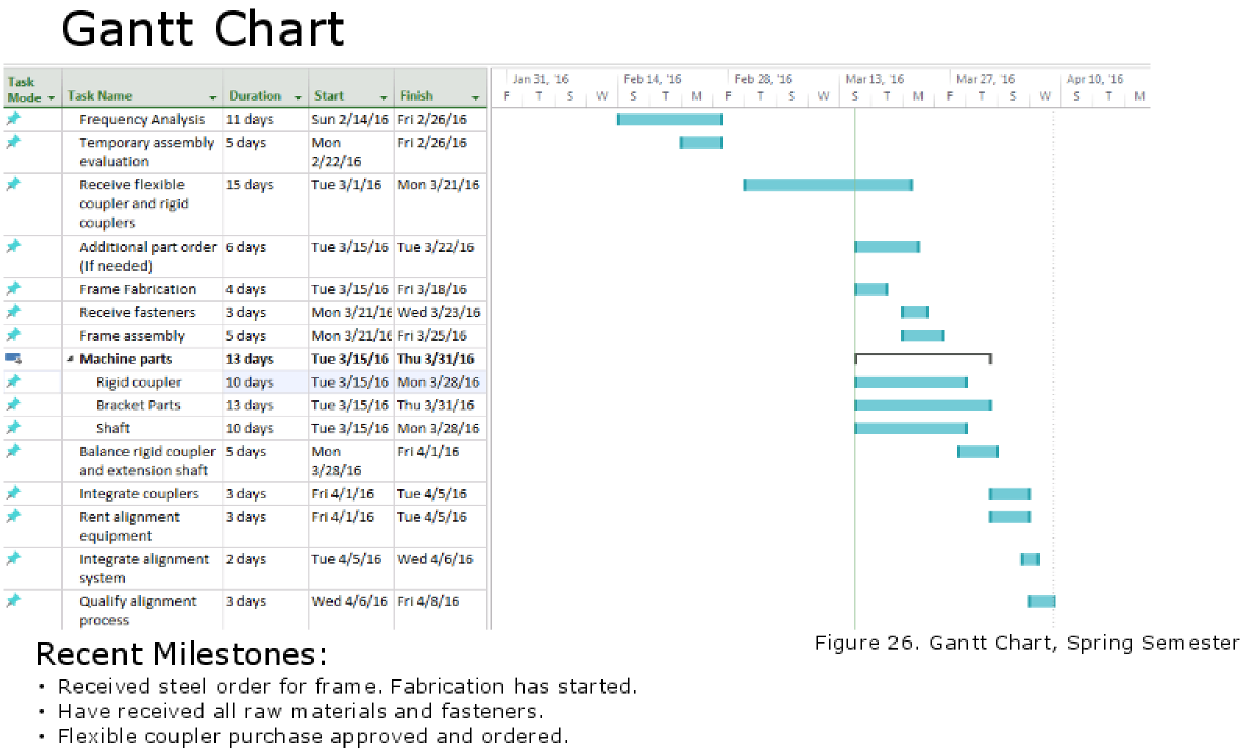
An iterative approach with taken by the team to achieve alignment. First they checked for vertical alignment. They chose a compressor that would be stationary and one that would be moved. The stationary one was shimmed up higher than the movable one, thus requiring the movable one to be shimmed up after the TIR values were obtained. After just one iteration, the lateral alignment was reduced from 0.040” to 0.001” (0.025mm), and an angularity of 0.003 degrees. The resolution of the dial was 0.001”, and the users saw that they could not improve anymore. Vertical alignment was sufficient. Next came the horizontal alignment. The users had more difficulty with this one, five iterations were preformed until the results converged. Final results showed that lateral misalignment was reduced from 0.004” to 0.0025” (0.064mm), and an angularity of 0.002 degrees. While watching the iteration results for the horizontal alignment, it was seen that the alignment with fluctuating beyond and below true alignment. This is because sign (positive and negative) on the correction adjustment values would change every other value or so. The team had come to a limit where no better horizontal values could be acquired. According to the specification sheet for the BK2 flexible coupler, the bellows coupling the team bought can handle up to 1 degree of angularity and 0.2mm of lateral misalignment. From the acquired results, it was proved that the two shaft had been aligned to a successful amount. This means that the alignment process the team developed for the system could be integrated into a high speed rotating system. Although the sponsor is recommended to check the balance of the rotating assembly before high speed rotation, the alignment system would achieve adequate results for lower speed rotations.

V. ENVIRONMENT, SAFETY, AND ETHICS

This project required the team to consider how the relating factors of this project could effect individuals and the enviroment in positive and negative ways. No harmful effects upon the enviroment were of concern to the team. This project was focused upon designing and fabricating a frame and achieving shaft alignment. There would be no impact from the Danfoss compressors or High Speed Motor Test Rig on the enviroment.

Because of the goal to achieve 40,000RPM, the team had to consider designing a safety shielding that would be compliant with the calculated values. The calcuated values showed that 3/16” mild grade steel would be sufficient to stop a projectile coming from the rotating assembly. Resources, such as professors and teaching assistants, were used to achieve the calculations for the required material thickness. It was understood by the team that these calcualtions must be precise, it is high priority that this system could ever harm someone, even in the worse possible failure mode.

The team members understand the academic responsibilty to be accountable for their work, to seek assistance, and strive for accuracy.   For an engineer, these are attributes that will positivley impact their professional work.



Task Name	Duration	Start	Finish
Frequency Analysis	11 days	Sun 2/14/16	Fri 2/26/16
Temporary assembly evaluation	5 days	Mon 2/22/16	Fri 2/26/16
Receive flexible coupler and rigid couplers	15 days	Tue 3/1/16	Mon 3/21/16
Additional part order (If needed)	6 days	Tue 3/15/16	Tue 3/22/16
Frame Fabrication	4 days	Tue 3/15/16	Fri 3/18/16
Receive fasteners	3 days	Mon 3/21/16	Wed 3/23/16
Frame assembly	5 days	Mon 3/21/16	Fri 3/25/16
Machine parts	**13 days**	**Tue 3/15/16**	**Thu 3/31/16**
Rigid coupler	10 days	Tue 3/15/16	Mon 3/28/16
Bracket Parts	13 days	Tue 3/15/16	Thu 3/31/16
Shaft	10 days	Tue 3/15/16	Mon 3/28/16
Balance rigid coupler and extension shaft	5 days	Mon 3/28/16	Fri 4/1/16
Integrate couplers	3 days	Fri 4/1/16	Tue 4/5/16
Rent alignment equipment	3 days	Fri 4/1/16	Tue 4/5/16
Integrate alignment system	2 days	Tue 4/5/16	Wed 4/6/16
Qualify alignment process	3 days	Wed 4/6/16	Fri 4/8/16

FIGURE 28: GANTT CHART [3]

A.Schedule

To help keep the team on track, the Gantt chart and Work Break Down Structure were used. The attempt to follow these was not entirely successful. The listed tasks were accomplished, but the timing and order at which they were completed was not as proejcted.

The nature of this project requries constant interaction with the sponsor. This project was not a high priority with the sponsor, and therefore the team expereinced highly delayed response times. To deal with this, the team looked to see what could be accomplished in the mean time. This is what led to the constant change of plans, and therefor the Gantt Chart and Work Break Down were not useful from a timing stand point. The possibilty to use a Critical Path Method Chart was not in question. The team expressed consistently that it was important to receive feedback in a timely fashion for continuous progression of the project. Once the sponsor was aware of this the team would notify their faculty advisor, Dr. Hollis, that way their academic expections could be realistic and adjusted for the limiting factor, that was their sponsor.

B.Resources

The COE Machine Shop and the Danfoss Machine Shop were available to the team for use. During the beginning of the project, the team discussed with the sponsor the possibilty of using the machine shop at Danfoss for fabrication. The sponsor ensured with no problems that the team could use this resource. Upon time for machining to occur, the team found out from one source that the machine shop could not help them, for the machine shop was too busy. Another source at Danfoss told the team that the machine shop could help them. This confusion greatly delayed the fabrication process and the team was in limbo between using the COE Maching Shop or the Danfoss machine shop. The teams faculty advisor and Senior Design Professor were notifited and updated when the situation occurred.

Danfoss decided that their welder would do the welding and cutting for the frame fabrication. At first the team did not plan on using the Danfoss machine shop, but the teams liason was able to find out weeks later that the Danfoss machine shop could produce some parts for the project. Eventually, the team used both the COE and Danfoss Machine Shops to split the load of the needed parts.

It came to the attention of the team early on in the project that the other Danfoss Senior Design Team was granted building access through badges. The badges granted the team the opportuninty to go the the building at their own leisure and work on the project and check in with their liaison as they pleased. Team 4 requested for the liaison to grant the team badge access through the HR department but each time this was brought up it was brushed off and never completed. The team repeatedly tried to get badge access and spoke with the front desk multiple times and nothing ever came of this. This proved to be a hinderance to the design process as the email communication was hardly ever timely and the team did not have the freedom to access the machine shop early on. This also was an issue towards the end of the project when assembly and testing took place. The team was relying on the schedule of the liaison and other helpful employees to be granted access to the warehouse to work at the teams desire. The team wanted to constantly be working on this project to finish sooner but as it was stated previously, it was extremely difficult without badge access. Much of this project depended on heavy equipment that required hoists and ample space to handle safely and the Senior Design Room A221 on campus was not suffienct space for this project, another reason why badge access was extremely necessary.

B.Procurement

No budget was ever given to this project. In the second meeting the team had with their sponsor in Fall 2015, they were told exactly “Do not worry about the budget”. The previous attempt the sponsor made at designing this project came to a price of $80,000. Team 4 was told to save money where possible and that the nature of the proejct was going to be expensive regardless. The team thought that this would help them achieve the best results, and it would have, except in Spring 2016 the sponsor decided to limit the purhasing power the team had. A crucial piece to shaft alignment was the TSKA 31 laser aligner for $3,500 purchase or rental for $2,100. This expense was denied, much to the surprise of team members and the faculty advisor. It is advised that for the next team to work on this project, that a budget be defined. By defining a budget, a team will have a consistent direction to work in, which is incredibly important to an engineering project and not abiding by a budget is ill-advised.

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D. Bill of Materials

Table 3: Bill of Materials

Component	Qty	Unit Price	Total Price
2"x2"x1/4" Low Carbon Steel tube, Length: 6ft	4	$69.82	$279.28
2"x1/4" Low Carbon Steel strip, Length: 6ft	1	$35.89	$35.89
2"x1/4" Low Carbon Steel strip, Length: 2ft	1	$16.15	$16.15
Brass Shim Stock, 6"x60" , Thickness: 0.001"	1	$11.53	$11.53
Brass Shim Stock, 6"x60" , Thickness: 0.003"	1	$11.42	$11.42
Brass Shim Stock, 6"x60", Thickness: 0.006"	1	$16.97	$16.97
Brass Shim Stock, 6"x60", Thickness: 0.009"	1	$22.65	$22.65
Brass Shim Stock, 6"x60", Thickness: 0.012"	1	$24.30	$24.30
Brass Shim Stock, 6"x60", Thickness: 0.02"	1	$35.60	$35.60
Brass Shim Stock, 6"x60", Thickness: 0.031"	1	$53.47	$53.47
Alloy Steel Socket Head Cap Screw, Thread size: 3/8"-24, Length: 2". Package Qty: 5	2	$8.87	$17.74
General Purpose Low Carbon Steel, 1/4" Thick, 2" Width, 3ft Length.	1	$20.82	$20.82
High Strength Steel Cap Screw, Zinc Yellow Chromate, 1/2"-13, Length: 5 1/2", Partially Threaded (Pack Qty: 5)	2	$12.58	$25.16
Extra-Wide Hex Nut, Zinc Yellow Chromate, 1/2"-13 (Pack Qty: 25)	1	$10.11	10.11
Over Sized Flat Washer, Zinc Yellow-Chromate Plated, 1/2" Screw Size, 0.531" ID, 1.062" OD. (Pack Qty: 25)	1	$11.44	$11.44
High Strength Steel Cap Screw, Zinc Yellow Chromate, M12 x 1.75, Length: 70mm, Partially Threaded. (Pack Qty: 5)	2	$9.20	$18.40
Bellows Coupler BK2 150 D1: 25.4mm D2: 25.4mm	1	$329.56	$329.56
Machine able-Bore One-Piece Clamp-On Rigid Shaft Coupling	2	$42.46	$84.92
Hardened Shaft, Steel, Diameter:1", Length: 10"	1	$15.05	$15.05
0.001 Inch Resolution Dial Indicator with Magnetic Base Stand	2	$33.00	$66.00
		Total Cost	$1,106.46

E. Communications

The team used texting and emails to communicate with one another, the sponsor, the faculty advisor, and the class professor. Immedialty into the project, in Fall 2015, the team expereinced terrible feedback from their liason. Many times the team would have to email multiple times throughout the week, hoping to hear back. At the beginging phase of the project, the team needed to know dimensions of the compressor and also needed access to CAD drawings and specifications for the compressor. It would be completley imposible to design the test rig without this imformation. This needed information was not given over easily. For example, the team asked what the torque output and RPM ranges were for the TT series compressor. Without this information it was impossible to select couplers and design the rotating assembly. This also prevented the team from designing the base framing because the over length was unknown. It took over a month until the sponsor would give the information. What caused this information to finally be given over was when the team went directly to Dr. Shih at the school, who then saught out an executive at Danfoss to help the team obtain the information.

Once the team realized that the liason was not going to be a helpful resource, they saught a solution. Upon a day meeting at Danfoss, the team leader walked around to find the machine shop area, and then looked to find someone with machining knowledge. The individual he found was acutally an engineer that gave some good feedback for the current phase of the project. The team then began using this individual (Kevin Lohman) for their primarry help on the project. Communication with Kevin was with email, and meetings that occurred once a week. Because Kevin was never intended to be involved with this project, he did not always have time during the week to meet for very long, and his email response was sometimes delayed to a week.

A meeting schedule was put into place with Dr. Hollis, the faculty advisor to this project. The team met with him, at a minimum, every Tuesday at 2:00. Whenever a break through or new challenge to the project occurred, Dr. Hollis was notified and his feedback was sought within day. The team also relied on these meetings with Dr. Hollis to inform one another about everyone's status with their involvement. This was the primary time for team members to seek assistance from one another, or seek involvement with another aspect of the project.

VII. CONCLUSION

The High Speed Motor Test Rig project was focused upon using the theory behind a motor-generator, and integrating two HVAC compressor from Danfoss Turbocor in place of the motor and generator. A motor-generator system couples the shaft of the motor to another motor. One motor drives the opposing motor, which acts as the generator, that can impose a varying load on the motor. In this project, one compressor motor is the driving motor and is coupled to another compressor motor (generator). Danfoss wishes to use this system in conjunction with a torque transducer to find efficiency values, by comparing actual torque to theoretical torque.. Because of the high speeds, 40,000RPM, high attention to shaft alignment process was required. This alignment eventually became the primary focus of team 4.

The technical constraints of the project set a goal rotation speed of 50,000 RPM and 100 Nm. These values are beyond the actual compressor performances and establishes a factor of safety. The team was able to design the system to withstand 150Nm, but the rotation speed did not meet the mark. Although of this short coming, the sponsor was still pleased, as the team was able to prove their alignment system setup and process.

A.Logistic Challenges

Team 4 dealt with a wide variety of challenges to accomplish this project. These challenges related to design constrains, component selection, communication with the sponsor, an undefined budget, and an inconsistent project scope. To overcome these, the team had to work patiently, yet adapt quickly to changing circumstances.

When the project was first assigned to the team, they immediately met with the sponsor to thuroughly understand all they could about the project. At this stage, the team was informed to not be concerend about the cost of the system. The staff at Danfoss staff was aware that the nature of this system causes it to be a costly design to execute. The team was told that years ago a High Speed Motor Test Rig for their systems was priced to be roughly $80,000. Therefore, the team immediately recognized a goal of the project was the development a cost effective solution, relative to the previously priced system. A non-limiting budget gave the team great control over the system design to ensure quality components could be ordered and integrated into the system.

Shortly into the Fall semester, the teams progress was completely stopped by the delayed and insufficient feedback. Email responses to the team were often times week(s) late, and at times the response did not contain the answers to the team's significant questions. The team was able to find ways around this. For example, they were able to get Dr. Shih to email an executive at Danfoss. Lin Sun, that then sought the individual who could answer the team. Another solution the team found was when they stumbled upon an engineer in the machine shop at Danfoss, where they got his contact information. This individual, Kevin Lohman, turned out to be the most help to the team, even though he was not formally assigned to help the team. The assigned liaison did not have near the knowledge or interest to help the team as Kevin did.

These challenges delayed the progress of the project. The first complete design that was thoroughly built in 3D CAD with accompanying drawing was not completed but till January 2016. It was the desire of the team to have this completed before December 2015. These challenges were beyond the control of the team. Lastly, the other factor beyond the team's control was the inability to have free access to the Danfoss facility. This luxury was given to the other team working with Danfoss, but our team was never granted that access. Team 4 has endlessly asked, and have never been able to get building access that allowed them to enter on their time.

B.Initial and Final Design

The unlimited budget allowed the team to develop a unique and quality design. The team established a completely thorough design that used torque transducers, laser alignment tooling, and a system that adjusted the position of the compressors, all of which rested upon a uniquely designed frame that passed through FEA analysis. The team presented this design (figure 15) to Danfoss and explained how assembly and alignment would be completed. It was understood by the team, and Danfoss, that there truly is no way to know what the minimum alignment requirement was, therefore the team design the best possible alignment that would be safe and accurate for the Turbocor compressor. By this understating, it was decided to use the TSKA 31 Laser Aligner, research shows that laser alignment processes are far more accurate than dial indicator methods. The chosen laser aligner to purchase eliminates common human errors and measures to 5 microns. This ideal system would meet all of the constraints proposed by Danfoss. Although, this design was not developed because of costs. The two torque transducers needed were priced at $8,000 a piece, and the laser aligner was $3,300. The laser aligner was not an immediate issue, but the transducers were not a reasonable price for the sponsor. It can at shock to the team when Danfoss decided that this was too much to spend on the project. None the less, the team adapted to the situation and design a new system.

To establish a consensus with the sponsor, the team communicated with the liaison to develop a new project scope. This new project scope still did not have a defined budget, but allowed the use of the laser aligner and the previously design frame and alignment system. It was then told to the team that they should focus upon proving the alignment system, and if possible, design a new rotating assembly that could be as close as possible to the 50,000 RPM and 100 Nm goal. The team then designed a system that used the previously developed frame, but a new rotating assembly that consisted of two rigid couplers, two shafts, and one flexible coupler. This design was presented to the sponsor, and again, was declined, but this time because of the price of the laser alignment tool ($3,300). This decision came a month before the senior design project was due, and the team scrambled for a solution.

Since the frame design was complete, they team sent off the designs to the fabricator at Danfoss. This fabrication process took weeks to accomplish. It became evident to the team that building the frame was of low priority to Danfoss, because the fabricator would work on it little by little. Correspondents at Danfoss gave the team multiple days that the frame would be completed by, but when the projected completion days came, they would inform the team that more time was needed. While this process took place, the team also had to find a cheaper solution to buying the laser aligner. Contact was made with a company that offered to rent the TSKA 31 laser alignment for one month to Danfoss for $2,100. This price was relayed to the sponsor, but not approved.

The only solution remaining for the alignment tooling problem was to use dial indicators. True shaft alignment dial indicator systems cost roughly $1,000, the team did not want to wait for a purchase approval for this, which historically took weeks. With a week left till the project was due, the team decided to buy their own dial indicators out of pocket from a local store, and develop their own alignment process.

C.Final Design Testing

Developing an alignment process was extremely challenging. Online operation manuals were found that outlined the alignment process. These extensive tests revealed that learning manual shaft alignment with a dial indicator is often times an extensive college level engineering elective that takes practice to accomplish. After team 4 thoroughly read into the process, they used equations that the text presented and developed an Excel file that used various parameters and dial readings to output the amount that the compressors needed to be shifted by in vertical and horizontal directions.

Once the frame was completed, there was three days left until the final due date of the senior design project. The team splits work tasks between assembly, testing, and academic deliverables to meet all expectations from the sponsor and college faculty. The system assembly went together very easily, there were not flaws with the fit of components with one another. Once this was complete, the alignment testing began. Users fitted the dial indicators to the rotating assembly and set them up using a mixture of the methods learned online, and the in-house methods developed by the users.

The first few attempts at vertical alignment were not successful, the users struggled to use the correct shims for compressor position adjustment. A new approach was then taken, the users decided to restart the vertical alignment process, but this time with one of the compressors (the stationary compressor) shimmed by 0.082" under each of its four feet. This caused the moveable compressor to be initially lower that the other compressor. The users then decided to iterate alignment to see how the misalignment trend either grew or shrunk. After just one iteration, the lateral alignment was reduced from 0.040" to 0.001" (0.025mm), and an angularity of 0.003 degrees. The resolution of the dial was 0.001", and the users saw that they could not improve anymore. Vertical alignment was sufficient. Next came the horizontal alignment. The users had more difficulty with this one, five iterations were preformed until the results converged. Final results showed that lateral misalignment was reduced from 0.004" to 0.0025" (0.064mm), and an angularity of 0.002 degrees. According to the specification sheet for the BK2 flexible coupler, the bellows coupling the team bought can handle up to 1 degree of angularity and 0.2mm of lateral misalignment. From the acquired results, it was proved that the two shaft had been aligned to a successful amount. This means that the alignment process the team developed for the system could be integrated into a high speed rotating system. Although the sponsor is recommended to check the balance of the rotating assembly before high speed rotation, the alignment system would achieve adequate results for lower speed rotations.

The team acquired these results two days before the final due date. They gave the sponsor the results they acquired and explained the process they used to get there. From that point on, all further operation of the test rig has been left up to the discretion of the sponsor. The team had accomplished their goal to prove the alignment process.

D.Future Recommendations

Looking back, the team wishes that they had developed multiple levels of backup plans in case one design or feature did not work or get approval. This mean not only just conceptually develop the idea, but acquire theoretical analysis results, CAD drawings, and outside research that would insure there were not holes in the backup plans.

If the sponsor wishes to continue developing the High Speed Motor Test Rig, team 4 recommends to increase the spending. To better the shaft alignment, a true shaft alignment system should be purchased. While the laser aligner is ideal, a dial indicator shaft aligner, with an accuracy better than 0.001", would be an improvement over the current system. If the sponsor wishes to next integrate the torque transducers, the team highly recommends using a laser aligner, which will help protect the expensive equipment at risk of being damage if misalignment is excessive. Currently, the shims being used were cut by hand from raw shim stock material. To better the user experience, it is recommended to label all shims according to their thickness and establish an organized box that sorts the shims.

One of the most important recommendations the team has to make is that the sponsor establishes a set budget for their next senior design team. By doing so, the team will have bounds to operate within, therefore the project scope is much less likely to change, and it happened to team 4. This budget needs to be set up with the school as well, so that the team can can quickly order parts when they need to. It was an incredible nuisance for the team to set up meetings just to explain the need to order basic parts. This process caused the team to order many tools and parts out of pocket. This would have been avoided if the team could order through an account at the school set up by the sponsor. Which is the common method for the majority of senior design teams at the FAMU-FSU College of Engineering.

Lastly, the team recommends that the next senior design team be granted badge access to the Danfoss building. It has already been established that another senior design team from this year was granted badge access to all of their team members, and individuals were able to come and go to achieve quick project progression. Team 4 found it un professional and somewhat insulting that the sponsor did not wish to help them in this way. It was never understood why the sponsor withheld this assistance to the team.

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APPENDIX A

High Speed Motor Test Rig

Operation Manual

4-1-2016

Team 4

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Abstract

Before an individual begins working with this High Speed Motor Test Rig, they should familiarize themselves with all the included components, processes, and risks. This project demands a high level of shaft alignment, to achieve this, the assembly process must be followed closely. Operators should thoroughly understand the alignment processes, and seek assistance if they need a better understanding. This system will require attention to component condition throughout its life. Operators must familiarize themselves with certain components that could become prone to structural failure. It is highly important that safety comes first, operators must follow instructions to protect themselves and others involved.

1. Functional Analysis

As delegated by Danfoss Turbocor, the High Speed Motor Test Rig designed by Senior Design Team 4 at FAMU/FSU College of Engineering was developed with the purpose of qualifying motor performance while using a precise alignment method. This is capable to work with all the TT-series of Turbocor compressors.

In order to qualify motor performance the test rig needs to be able to deal with high speeds (around 40,000 rpm) and be compatible with the specific features of Turbocor compressors. One of these main features is the use of magnetic bearings. Magnetic bearings are used to maintain these engines as oil free while at the same time supporting a limited value of radial load (around 200 lbs), which requires a certain level of attention when dealing with mechanical vibrations at high speeds.

To provide a feasible solution to all the requirements already mentioned and making future qualification possible in terms of power, efficiency and heat management of these motors, the high-speed motor test rig is a possible solution. Initially, the test rig was completely designed to allow the integration of a torque transducer and a laser alignment tool, but due to unforeseen budget constraints the overall design had to be changed to focus on validating the alignment process. This validation is to be done with dial indicators. The following is a breakdown of each subsystem, its components, and their respective functions.

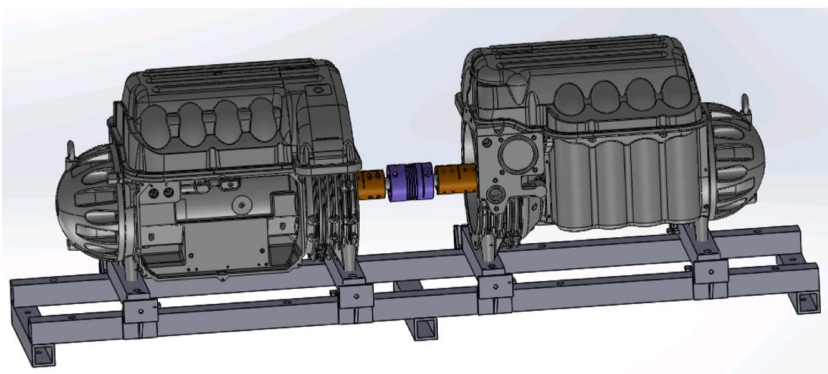


Figure 1: High-speed motor test rig assemble with two TT-500 compressors models.

1 2. Project Specifications

2 2.1 Base frame

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 upper cross members not only give support to the compressor but also allow the positioning of all other components needed for alignment. All the details about the alignment system and the other subsystems of the base frame can be found in the section Product Specifications.

The base frame dimensions can be visualized in Figure 6. Of the three lower cross members, the middle is welded at mid length of the long runner, the other two lower cross members have two holes drilled in each and are to be welded under the long runners at each end. The four upper cross members are going to be bolted to the frame with ½”-13 5.5” long cap screws. The supporting frame uses concrete fastening bolts to ensure their stability to the ground.

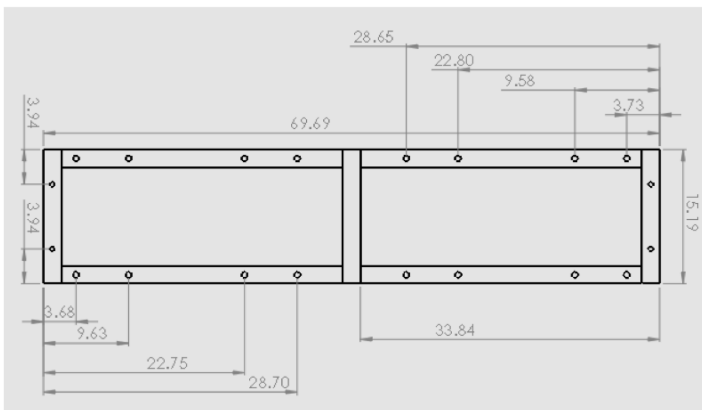


Figure 2: Upper view of supporting frame without upper cross members [units in mm]

3 2.2 Rotating assembly

Composed of two rigid coupler, one flexible coupler and two shafts, the rotating assembly is responsible for connecting both compressors to transmit power. Each component was selected/designed to fulfill the requirements demanded by the specific characteristics of Turbocor TT-series of compressors.

A flexible coupler is necessary in order to support axial, radial and angular misalignment. As the rotating assembly will be running at speeds around 40,000 rpm. A specific natural frequency analysis was done to ensure that this rotating assembly will not bring problems related to mechanical vibrations or any other misalignment issues.

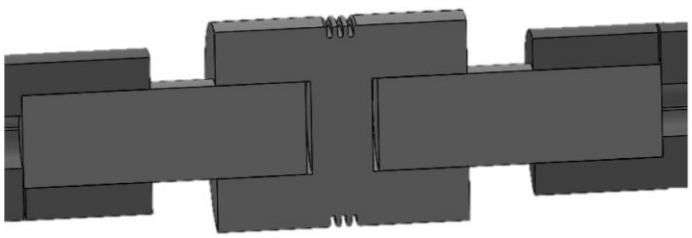
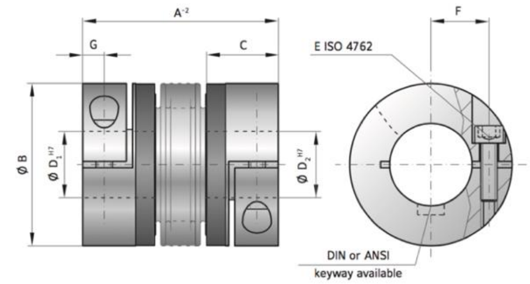


Figure 3: Section view of rotating assembly – 2 rigid couplers, 2 shafts and 1 flexible coupler

The rigid couplers selected are re-machined to the diameter of the compressor shaft. The flexible coupler is the R&W BK2 150. To conjoin the coupler, two shafts are used to fit into both couplers. The dimensions of the shaft can be founded on Figure 8, and the couplers specification on Figure 6 and 7.

Figure 4: Rigid coupler McMasterr-Carr Re-machinable.

The rigid coupler is mounted on the shaft of the compressor after the impellers are removed. The impellers are removed before any type of testing because the validation of the alignment system is the first concern of the test rig and at this point we are not interested yet in verifying the compression capacity of Turbocor compressors. This rigid coupler will be machined to reach inner diameters of 22 mm (compressor shaft) and 25.4 mm (connecting shaft). After being machined the coupler will be balanced at Turbocor's facilities in order to avoid any problems with unbalanced mass.



Overall length (mm)	A^{-2}	95	107	144
Outside diameter (mm)	B	81		
Fit length (mm)	C	36		
Inside diameter possible from Ø to Ø H7 (mm)	D_1/ D_2	19-42		
Fastening screw ISO 4762	E	M10		
Tightening torque of the fastening screw (Nm)	E	70		
Distance between centerlines (mm)	F	27		
Distance (mm)	G	11		

Figure 5: Bellow couple R&W BK2 150 dimensions.

The R&W BK2 150 is composed of aluminum hubs for low inertia and conical bushings for high clamping forces. This coupler is one of the key components because it can deal with the misalignments that the high speed can cause during a test for example.

The precision of the shaft is 50 µm. The shaft diameter and its length were determined mainly due the natural frequency analysis; the values are 25.4 mm and 107.0 mm. The shaft will connect the rigid coupler to the flexible coupler and is made of 1566 hardened steel.

4 2.3 Alignment system

The method to measure the misalignment is through the use of dial indicators. To measure the misalignment, it is suggested to use a laser alignment tool (SKF TSKA 31) in order to provide more accuracy, precision and also to avoid the possibility of human errors, although at this stage of the project the measure method will be with dial indicators. The decision to use dial indicators instead of a laser aligner was made by the sponsor, because of unforeseen budget limitations.

Other features of the alignment system include vertical and horizontal alignment adjustment. To adjust the compressors horizontally the test rig uses setscrew brackets, and to vertically adjust it utilizes shims of different thickness. The setscrews (eight in total) press into the upper cross members. Shims can be inserted in eight locations, under the upper cross member and between the long runners.

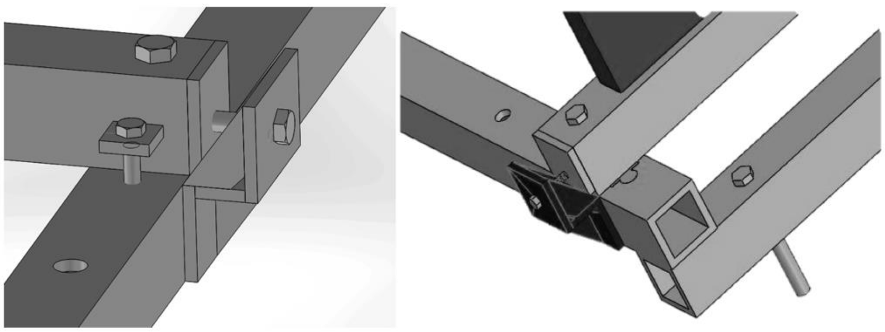


Figure 6: Alignment system: set screw brackets for horizontal adjustment and shims for vertical adjustment

The alignment system is basically the integration of the setscrew brackets with the supporting frame. To adjust the system horizontally the screw brackets are utilized and to adjust vertically the insertion of shims is used.

Brackets are composed of three triangular supports, two 2”x3” parts and one 1.25”x3”. The setscrew brackets are going to be welded on the long runner, as it will be explained on the Project Assembly section. A cap screw will allow the horizontal movement and a screw jack will allow the cross member elevation for shims insertion.

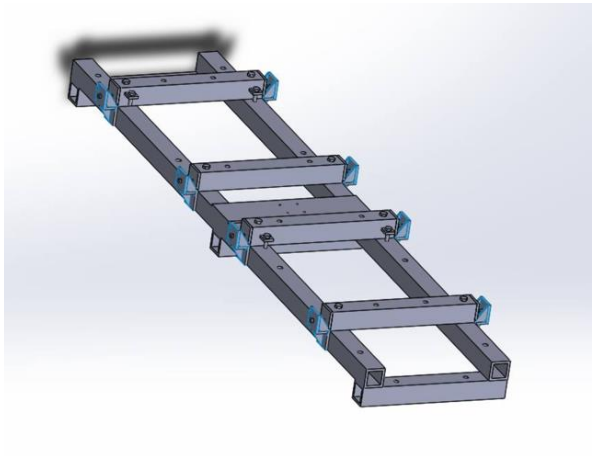


Figure 7: Horizontal and vertical alignment mechanisms integrated with the test rig base frame

The shims are made of brass and stainless steel and their thickness varies on a range from 0.001 inches to 0.031 inches. The dimensions of each shim are A=57 mm, B=51 mm and C=11 mm as on Figure 8.

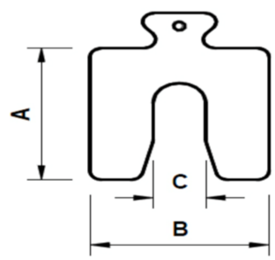


Figure 8: Shim upper view

5 2.4 Safety Shielding

The safety shielding was designed to prevent accidents due to possible failures and to provide better safety conditions to the operators of the test rig. It is made of steel with a thickness of 3/16". This shielding encompasses the rotating assembly and fits between the compressors.

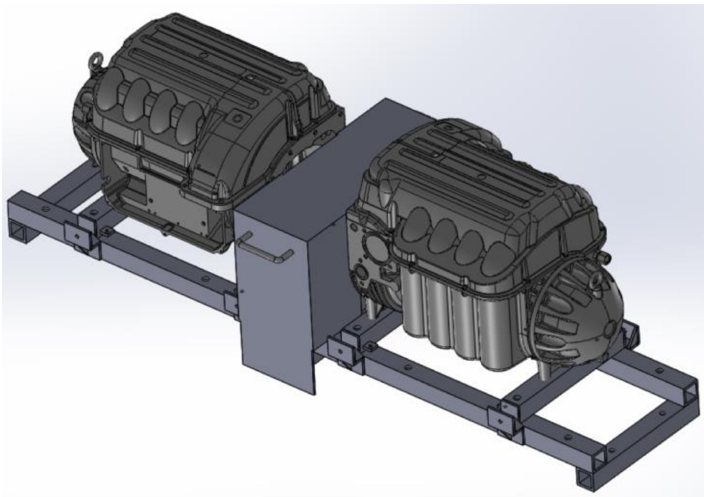


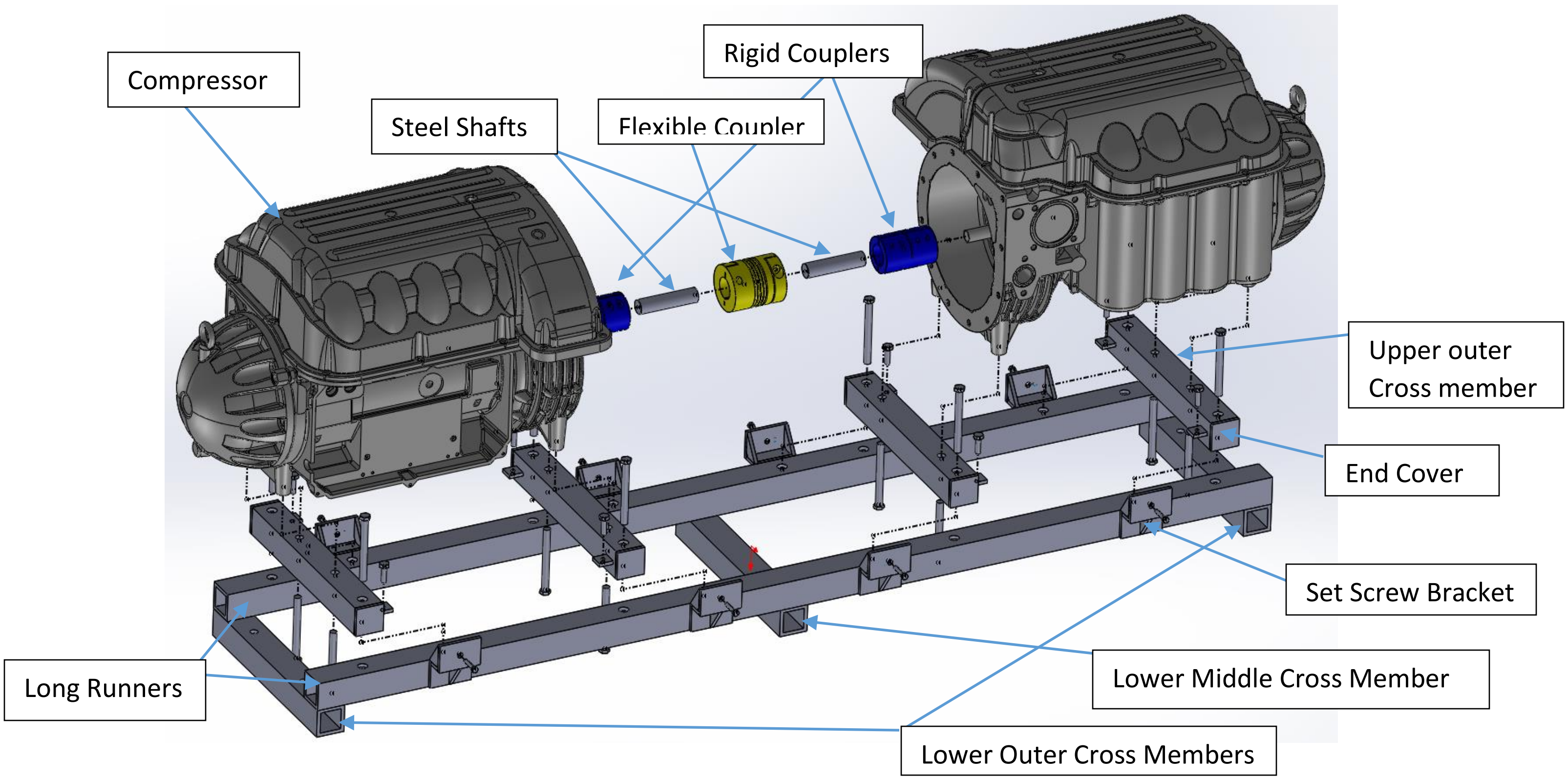
Figure 9: High speed motor test rig assembled with the safety shield

In summary, the high speed motor test rig needs to be able to operate in the following range of torque and speed:

The necessity to validate the alignment process required a proposal of testing and verifying all the alignment process. To do so the test rig was designed and each subsystem had their characteristics analyzed. Those characteristics can be verified as follows.

6 3. Project Assembly

This section will cover tools and equipment, and the assembly procedure that must be followed in the given order. By not using the necessary equipment and assembly process, error during alignment or structural failures may occur. Drawings of components and assemblies can be found in the appendix.



7 Figure 10. Exploded View.

8 3.1 Recommended Tooling and Equipment

Those preforming the assembly will require assistance from multiple items. These following items are given in the order that the user may need during system assembly.

Once the three lower cross members and two long runners are cut and their holes are drilled, they can then be welded together. The materials are of low carbon steel, the recommended welding equipment varies, and can be decided upon by welder. It is important that the welder is aware that steps be made to avoid warping the frame. Warping will cause the frame to lack straightness and the top surface will not be a level surface.

Multiple wrench fittings will be required to secure the fasteners within the test rig. Eight cap screws that will fasten through the four upper cross members and two long runners have a ½" socket head. Each of the eight setscrew brackets have cap screw that will require a 3/8" hex key. Eight compressor-mounting bolts will be fastened with a 12mm socket fitting. Two cap screws in the flexible coupler will require 10mm hex key. The two rigid couplers have four cap screws each and will require a 5mm hex key. Four anchor bolts will secure the frame to the concrete floor and will also use a 12mm socket fitting, these also require a hammer for their installation. The screw jack screw will require an 8mm socket fitting.

To hoist the compressors onto the frame, it is recommended that the individuals use the assistance of an overhead pulley or winching system. To avoid injury and damage to the system, it is not advised to lift the compressors without mechanical assistance.

Shaft alignment requires the use of either a SKF TKSA 31 laser aligner, or the use of a dial indicator. The dial indicator will be the assumed method and used in this operation manual. To properly fasten the dial indicator, the tool must have a strong magnetic base or clamp.

9 3.2 Frame

The supporting frame is composed of three lower cross members, two long runners, and four upper cross members. After these materials have been cut and their holes drilled, they can then be assembled. Two of the lower cross members have two holes drilled in each; these members will be welded on the opposing ends of the long runners (see figure 10). The lower middle cross member is to be welded mid length of the long runners. The four upper cross members are comprised of two different pieces, these are called “upper outer” and “upper inner” cross members. The “inner” pieces are bolted to the frame with ½”-13 5.5” long cap screws, and will be between, or “inside” the “outer pieces” (see figure 10). The four upper cross members will have end cover pieces, cut from ¼” steel, welded to the open ends (see figure 10) . The “outer” cross members will bolt to the long runners with the same cap screws, and be outside of the inner cross members. At each of these bolted junctions, there should be ½” washers, one washer under the head of the cap screw and one on the ½” nut that will be threaded onto the cap screw. At these junctions, the bolts should be torqued to 15Nm, but not until after the alignment process has been completed. Once the frame has been moved to its desired location, fasten it to the concrete using the concrete fastening bolts. Drill the concrete at the diameter of the M12 bolts. Use a hammer to punch the bolts through the lower outer cross members and into the concrete, be sure to leave bolt threads exposed above the surface of the cross member. Once they have tapped to the desired depth, thread the accompanying bolt onto the threads and tighten.

10 3.3 Set Screw Bracket

The setscrew brackets should be assembled by welding its seven sub parts together, following the prescribed assembly drawing (see appendix). Brackets consist of two 2”x3” parts, one 1.25”x3”, and three triangular supports. Once the eight brackets have been completed, they can be welded to the long runners of the assembled frame (see figure appendix for exact locations of set screw brackets). These brackets should be oriented so that the tapped hole is aligned with the center of the cross member end cover. Through each setscrew bracket, thread a 3/8”-24 2” cap screw.

11 3.4 Rotating Assembly

Components that comprise the rotating assembly are two steel shafts, two rigid couplers, and one flexible coupler. Starting first with the BK2 flexible coupler, loosen the two clamping screws. Slide each shaft 36mm into each end of the flexible coupler. Tighten the BK2 couplers screw to 70Nm (it may be easier to tighten these after the rotating assembly is connected to the compressors. Next, slide the rigid couplers 38mm onto the exposed ends of the shafts (see figure). The screws in the rigid couplers should be torqued to 70Nm (see appendix figure 19). Note: due to possible inaccuracies where the upper cross member supports are bolted, it may be necessary for the rotating assembly to extend further out or in to reach the compressor shafts.

12 3.5 Compressors

The High Speed Motor Test rig is designed around the use of the Danfoss Turbocor TT series compressors, and is not design to work with other series. For the system to function, the impellers must be removed from the compressor. To mount the compressors, it is recommended to use an overhead mechanical lifting system. Mount the first compressor onto one side of the frame, it does not matter which side is first. Be sure that the compressor shaft faces towards the center of the frame and not away. Align the mounting feet holes of the compressor with the holes in the upper cross members. Use the M12 bolts to fasten the compressor to the upper cross members (torque to 20Nm).

Once the first compressor is mounted, the next step is to attach the rotating assembly. Begin by taking the assembled rotating assembly and fixing one of the exposed rigid couplers to the exposed compressor shaft. These screws should be tightened to 70Nm.

The second compressor should be mounted to the system in the same manner as the first. Caution must be practiced when inserting the shaft into second exposed rigid coupler of the rotating assembly. The flexible couplers can allow for one degree of angular misalignment. Do not force the flexible coupler into a position that may exceed this, permanent damage may occur. Once the second compressor is fastened to the frame and rigid coupler, proceed in the assembly.

3.6 Safety Shielding

The safety shielding should be placed over the rotating assembly before the test rig is operated. With two people, grab the handles and place the shielding between the two compressors (figure 9). The feet of the shielding will sit on the ground, beyond the long runners. Make sure there is no contact with shielding and the rotating assembly.

13 4. Operating Instructions

Before the system is run, shaft alignment should always be performed. This ensures a safer and more efficient transfer of rotation from the driving motor to the generator motor, the greater the misalignment, the higher the chance of mechanical failure during rotation. The operation of the test rig after alignment requires control of the compressor motors and is beyond the realm of this user manual.

14 4.1 System Alignment

Alignment of compressor shaft requires the use of a dial indicator. The indicator should have an accuracy of 0.0001” and have an attachment that allows it to be rigidly mounted to a surface. To properly function, the indicator attachment arm needs to have multiple pivot points, this allows it to mount to one side of the rotating assembly, and reach around to the other.

The process for alignment begins with using the dial indicator. The indicator must be attached to one side of the rotating assembly and rotate 360 degrees. It is predicted that the user preforming the alignment will need to mount the dial indicator base to a rigid coupler and rest the dial-reading arm upon the other rigid couple.

15 4.1.2 Vertical Adjustment

With the dial base attached to a rigid coupler, reach the arm to a surface on the other side of the rotating assembly (must be somewhere past the opposing side of the flexible coupler). Start with the dial indicator in the 12 o'clock position and zero the reading. By hand, rotate the shafts to the 6 o'clock position and note the dial reading. During rotation, if the dial reading moves clockwise, the shaft being measured is lower than the shaft that the dial is mounted too. The value displayed at 6 o'clock is two times the vertical displacement between the shafts. To correct the vertical alignment, insert shims under the compressor that has the lower shaft position.

To select the appropriate shim(s), chose the largest shim width that is still less than the vertical offset value. Next, fine tune by using thinner shims to lessen the offset amount. The sum of the shim thicknesses used should be as close as possible to the measured offset value.

16 4.1.3 Horizontal Adjustment

Begin with the dial indicator rigidly fixed to one side of the rotating assembly (a rigid coupler is ideal). Reach the dial arm to the other side of the rotating assembly and set it upon a solid surface (the rigid coupler is ideal). Start with the indicator in the 3 o'clock position and zero the reading. Rotate the assembly to the 9 o'clock position and note the final reading. During shaft rotation if the dial rotates in the counter clockwise direction, the compressor opposing the side the dial indicator will need to be shifted towards the side of the 9olock indicator position. If the indicator dial rotated clockwise, the compressor opposite the side of the mounted indicator should be shifted to the 3 o'clock side (starting position of the dial indicator). Half of the dial value is the horizontal offset amount. To correct this offset, use the setscrews to shift the lateral position of the compressors. The set screw threads are 3/8"-24, this means that with one full rotation, the screw will thread in 0.0417 inches (1/24"). Use this value to estimate the required amount of rotations to shift the compressor. Be sure to use the same amount of rotation on the other setscrew that is on the same side of the frame.

17 5. Trouble Shooting

Following the order of assembly, this section will address possible issues that could arise. During the installation of the upper cross members and compressors, if the mounting holes do not align, it will be necessary to use a power drill to drill the bolt holes to a larger diameter. Only use small increments when increasing the hole size. While assembling the rotating assembly, if the fit between the couplings and shaft is too tight, turn down the corresponding shaft end diameter in small increments until the desired fit (h7 is ideal) is reached. While doing vertical alignment, if the dial indicator mounting position is disturbed after the dial readings have begun, the alignment process must start over. Maintaining the same position of the indicator is imperative to an accurate alignment. While preforming horizontal adjustment alignment, if the compressor does not want to shift, stop setscrew rotation. Excess load upon the setscrew may shear the threads. When a compressor cannot move laterally anymore, the bolts are being restricted by the frame hole diameter. If the opposing compressor cannot be adjusted instead, then the junctions at the long runners and cross members must be drilled to a larger diameter. After the system has been aligned and motor is run, if the shafts do not maintain alignment, check the break away torque on the frame and compressor. Due to the bolts relaxing from tensile load over time, the break away torque may be lower than the initial torque values. In this case, follow the alignment process and re-torque screw. If screw nuts become loose during system operation vibrations, apply Loctite 242 Blue Medium Strength Threadlocker to the threads. If any cracks are found within the frame or rotating assembly, replace the part immediately. If the safety shielding is found to have any significant impact damage, replacing before system operation.

18 6. Regular Maintenance

Before and after the system is run, the shaft alignment should be checked. During this time the torque on all fasteners should be checked to their intended values. If rust is found in frame, use an abrasive material to remove the rust and repaint the exposed metal. Inspect safety shielding before system operation for defects, replace if damage is found.

7. Conclusion

The team was tasked with designing a system that would test compressor efficiency by coupling two compressor shafts together. This system is referred to as a motor-generator system, and because of the high speeds Danfoss Turbocor compressors reach, accurate shaft alignment is critical. This alignment was the primary focus of the project.

The operations manual should be followed in the order the steps are presented. Failure to do so could jeopardize the success when preforming alignment and the safety during system operation

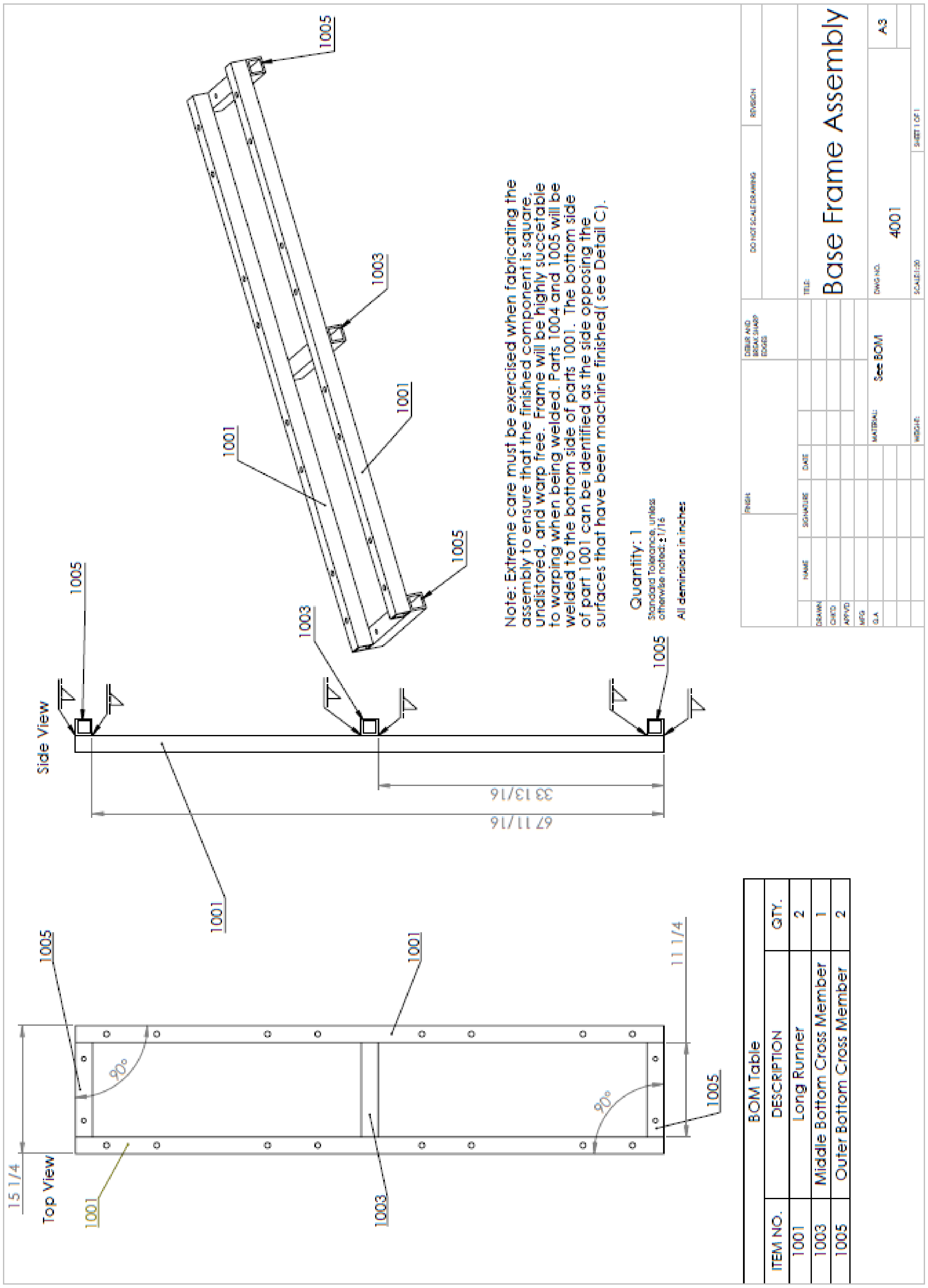


Figure 11. Base Frame Assembly

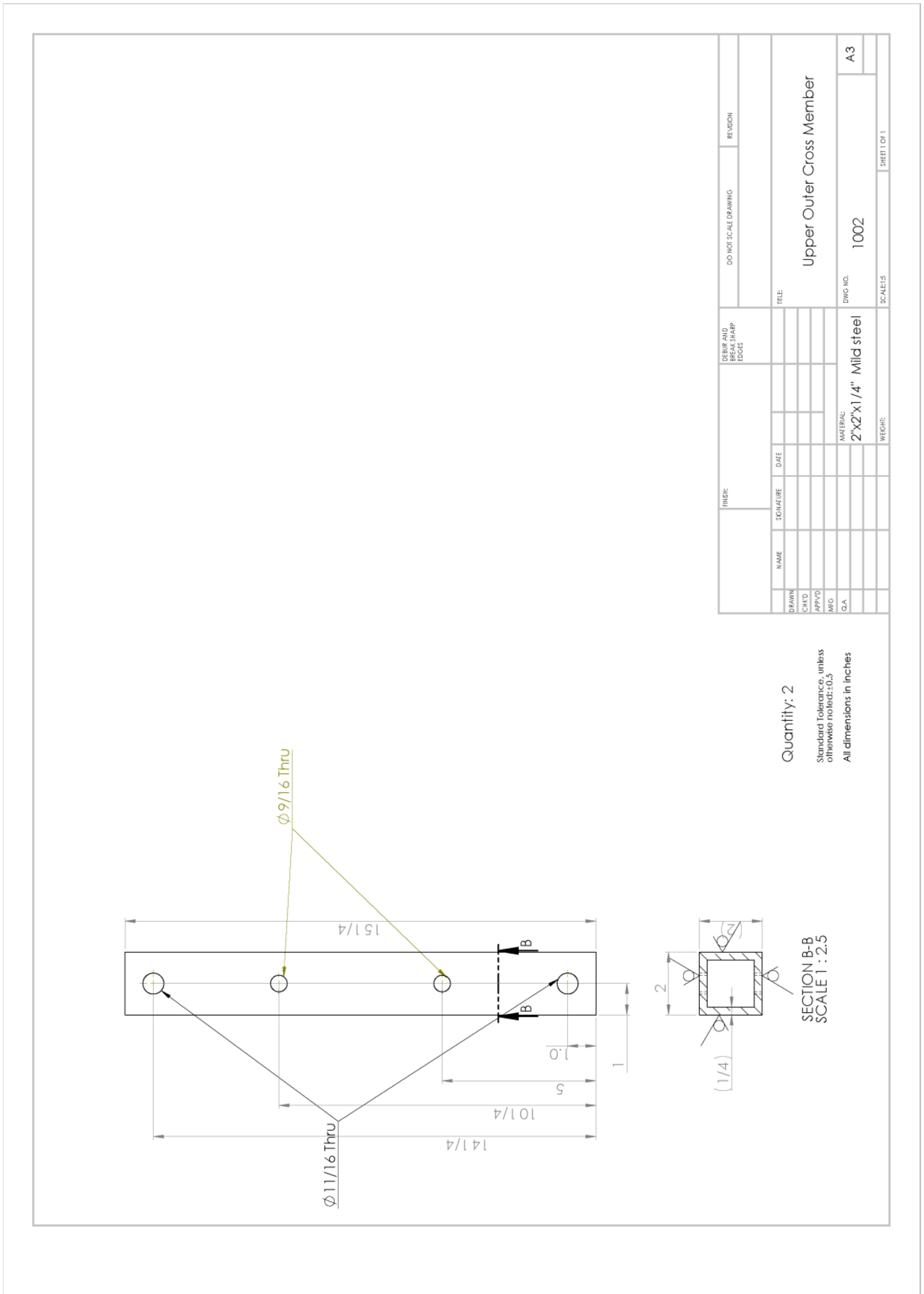


Figure 12. Upper Outer Cross Member

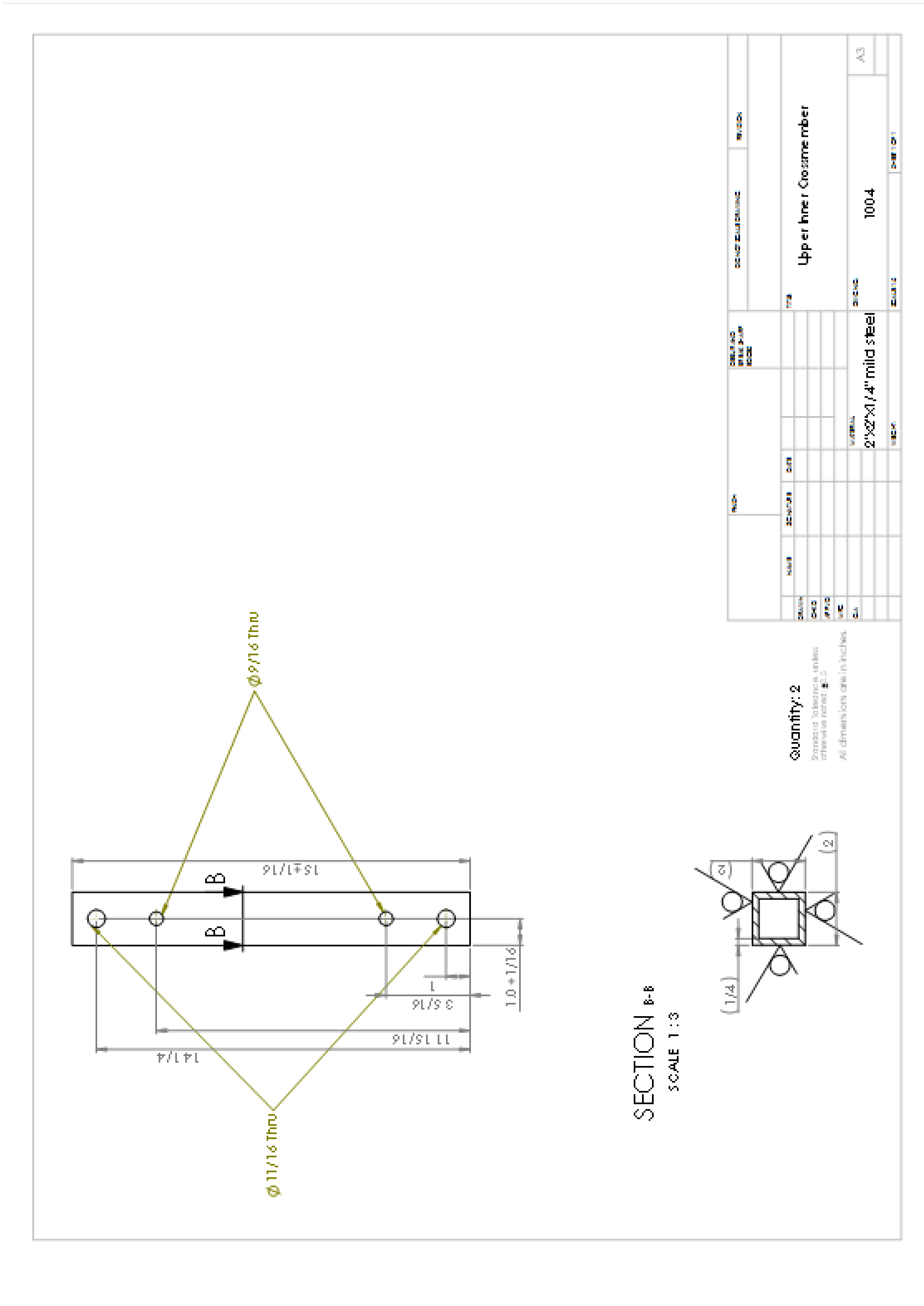


Figure 13. Upper Inner Cross Member

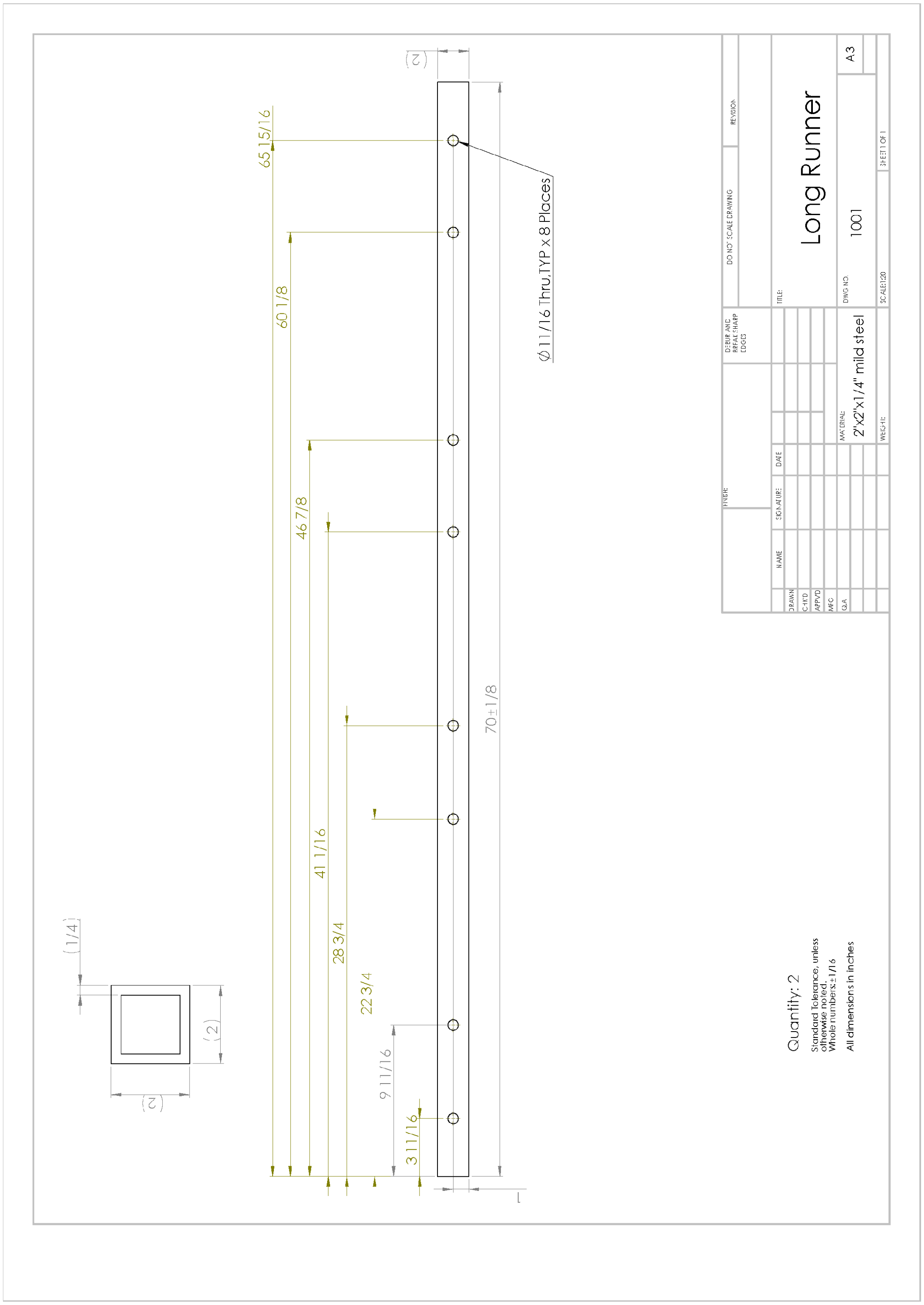


Figure 14. Long Runner

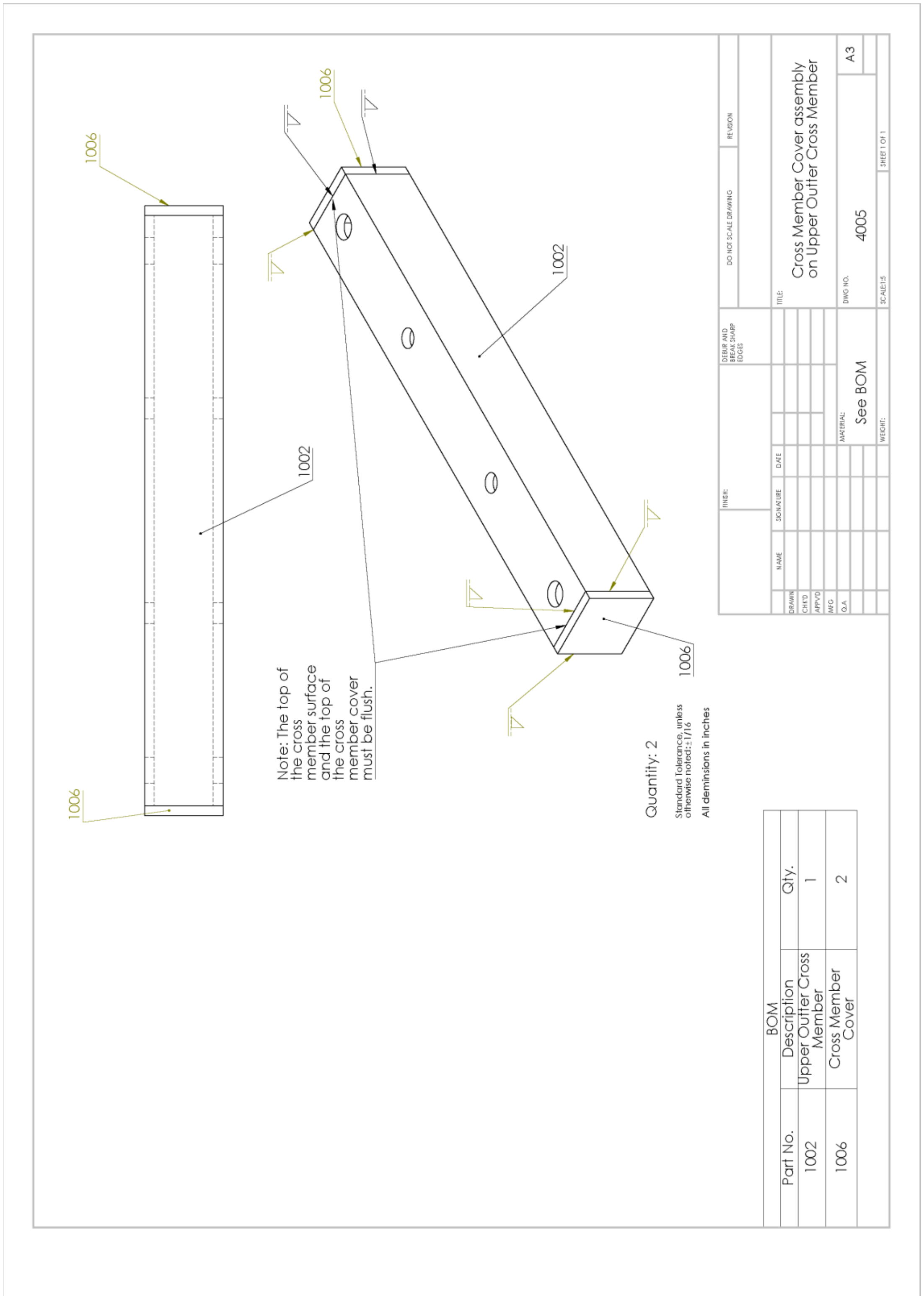


Figure 15. End Cover on Upper Outer Cross Member

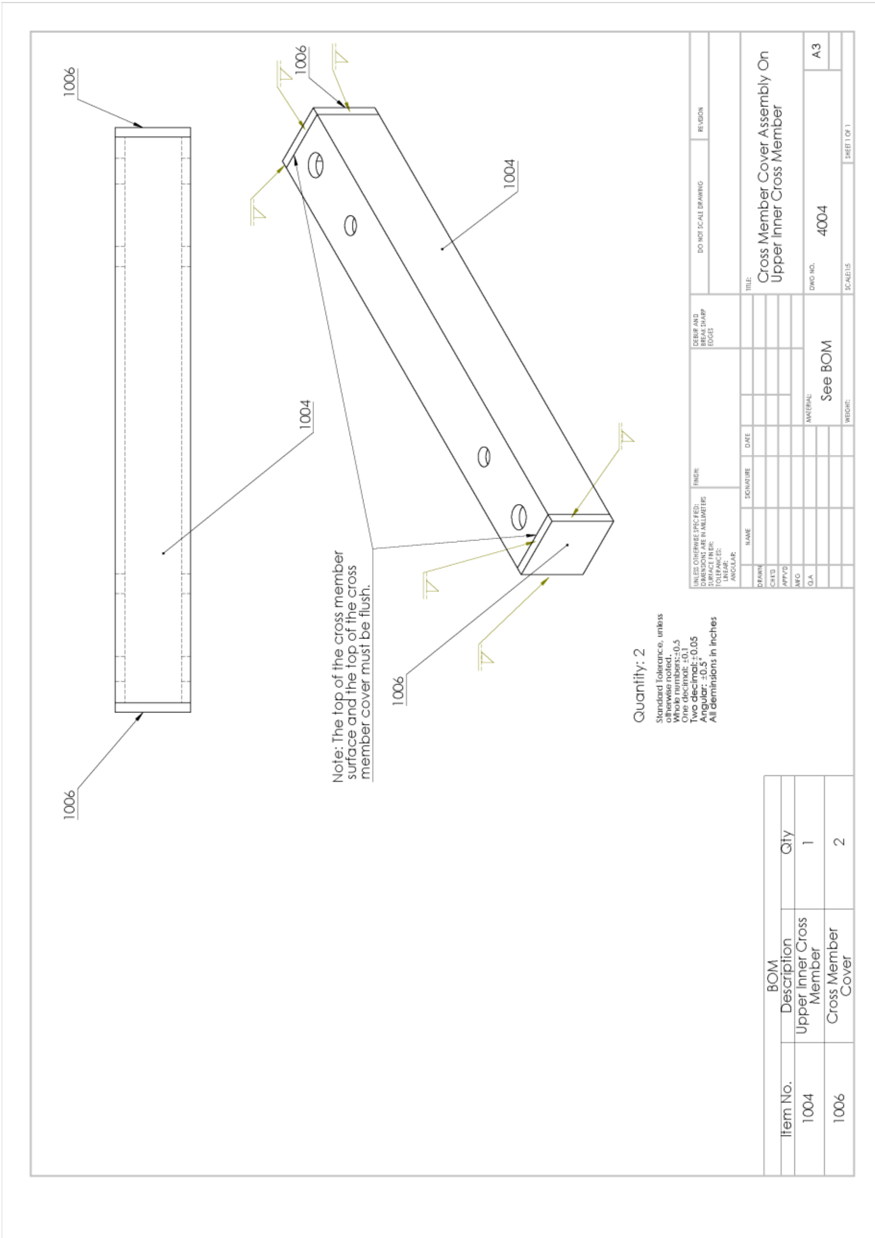


Figure 16. End Cover on Upper Inner Cross Member

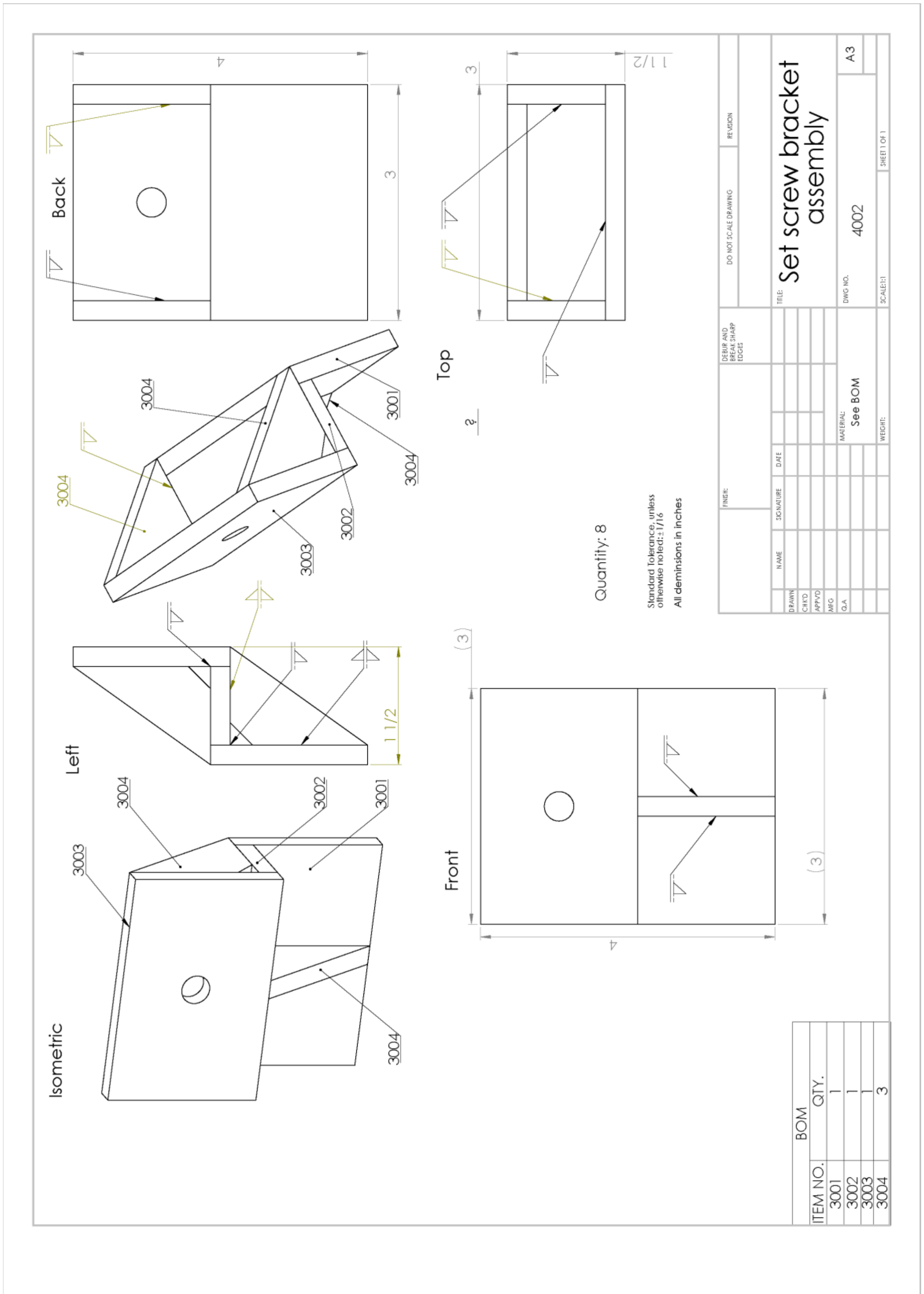


Figure 17. Set Screw Bracket Assembly

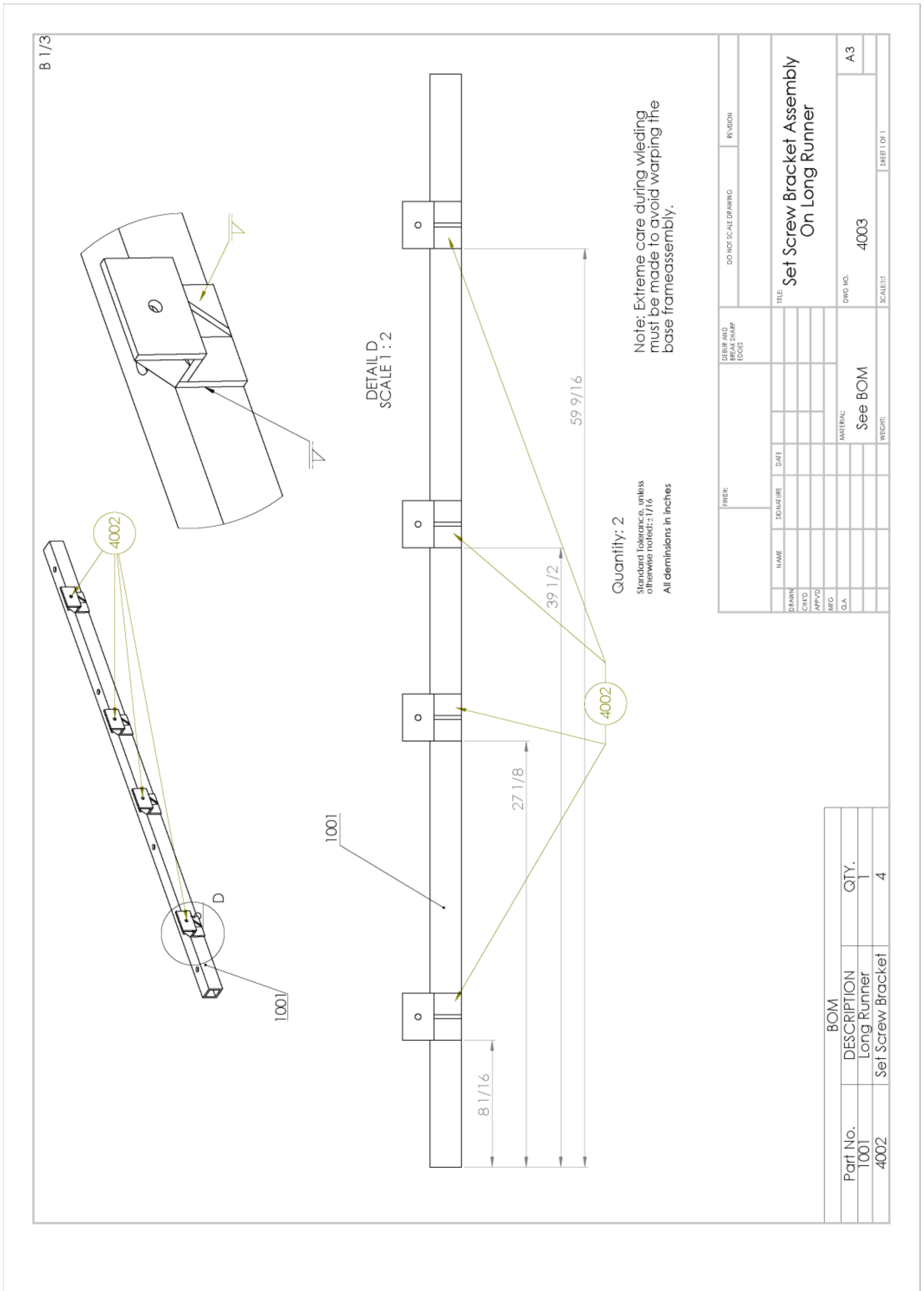


Figure 18. Set Screw Bracket Assembly on Long Runner

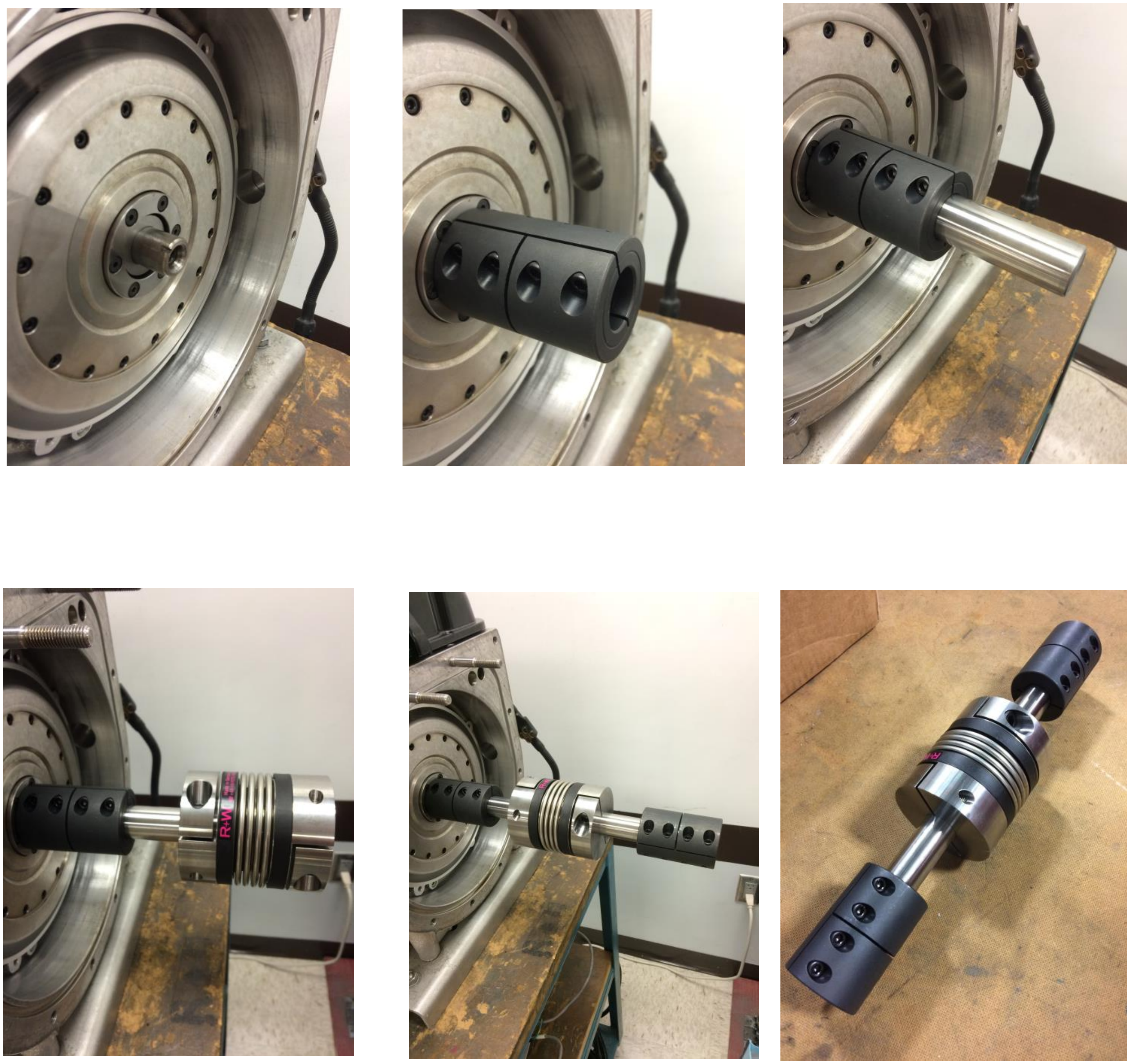


Figure 19. Rotating Assembly

APPENDIX B

High Speed Motor Test Rig

Design for Manufacturing, Reliability, and Economics

4-1-2016

Team 4

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Francisco Barreto (fjb11b)

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Sponsor: Danfoss Turbocor

Faculty Advisor: Dr. Patrick Hollis

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Abstract

Danfoss Turbocor has asked Senior Design Team 4 to develop a high-speed motor test rig to qualify their motor performance within the compressors. They have asked Team 4 for a motor-generator rig that uses one compressor motor to transmit power to another that can act as a variable load. The original designs were rather expensive and Team 4 has offered a more affordable alternative to prove alignment within the system and to also eventually run the compressors at full speed output. The total cost of the proposed design is $1,040.76 and this design is also suited for eventual input of torque transducers to run at max speed or max torque. The test rig is set to be complete the week of April 8th, 2016 and be ready to undergo testing.

Acknowledgements

Senior Design Team 4 would like to thank our instructors for the course, Dr. Nikhil Gupta and Dr. Chiang Shih, as well as the Teachers Assistants for their help. This project would not be possible without the support of our sponsor Danfoss Turbocor. Team 4 would also like to thank William Sum, Julio Lopez, and Kevin Lohman from Turbocor, for their guidance through the project.

1. Introduction

Danfoss Turbocor is the market leader in oil free compressors for different systems. The company designs compressors for heat, vacuum, and air conditioner industry. They achieve a high quality 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 best efficiency.

In order to test all the TT-Series compressors, Turbocor is looking for a system that can qualify their compressor's electric motor more accurately and more efficiently for power, efficiency and heat management. Danfoss's current test process is done in a chiller room, which is very expensive to operate. A motor-generator system could save test engineers time and money. Therefore, team 4 proposed to Turbocor a use of a new Motor-Generator Test Rig.

A Motor-Generator test rig is a system that couples two motors; one working as a motor and the other one as a motor load or generator, in this case they are using compressor motors. To connect the compressors to each other the use of a flexible coupler, which can handle misalignments, is required.

Our team designed a test rig that can handle the high-speed rotation with a considerable torque. To do so the team considered that the test rig should be well fixed to the ground, the compressors should be rigidly fixed to the base frame and it would require a precise method to move the compressor so the quality of the balancing would be guaranteed. The team designed a system that uses setscrews and shims to move the compressor horizontally and vertically.

For safety reasons it is required that the natural frequency should be at least higher than the operational frequency with a safety factor. The team assumed it too be at least 700 Hz, this because the theoretical natural frequency could be higher than the actual one.

Currently our design is complete and the team is awaiting the fabrication of the base frame. All other components have been acquired. As soon as the frame is completed, the final assembly will be made. As the test rig is made to test different compressors with simple installation, we are expecting that the assembly process should take only a couple hours.

2. Design for manufacturing

Before starting the assembly the team had to order the raw material and other components, such as the couplers. With the components in hand, some were machined and welded while others came prepared from the vendor. The ordering process took place through Danfoss Turbocor, who received a document with all price quotes of the components. The initial project considered a torque transducer between the two compressors, the project changed due the high cost this equipment ($8,000 each and the system requires two of them).

The base frame is made of 2x2x0.25” steel and its main function is to settle the compressors in their right position and rigidly fix the system to the ground. The raw materials were received at Turbocor’s facilities and were machined there as well. The machining process mainly leveled the surface to minimize any inconsistencies on the base frame runners, make the holes to settle the compressors and the ones to fix it to the ground, and weld the structure to guarantee the stiffness required. Figure 1 shows the structure as well as the welding lines, which are represented in red. Turbocor has some distributors of the raw material that they use (McMaster-Carr) and they also use this material in their facilities so the ordering process took less than one week and the machining process is still ongoing.

Figure 1: Base Frame with the weld lines indicated in red.

The rigid coupler selected by the team is the Re-Machinable Rigid Coupler supplied by McMaster-Carr. Its main function is to connect the compressor’s shaft to a thicker shaft, which will go inside the flexible coupler. The inner bores came originally with 22.25mm (which is the diameter of the shaft of the compressor). To increase the natural frequency of the system the team decided to increase the stiffness of the system by increasing the diameter of the shaft that connects the rigid coupler to the flexible coupler. To do so one of the inner bores was re-machined so it has 25.4mm diameter. After the bores were resized to the final diameter the rigid coupler needs to be balanced to deal with the high operational speed rotation. This system needs 2 rigid couplers and the ordering process ran through Turbocor and it took around 10 business days. Currently the rigid couplers are re-machined and the process took 5 business days, the balancing process should take 4 business days.

The flexible coupler selected by the team is the Bellow Coupler BK2 150, this component has been ordered through Turbocor and the senior design received this component. The coupler came with the desirable diameter so it does not require a re-machining process. This process should have taken 15 days but it has took longer than expected, we received this component only this week, it supposed to be delivered last week.

The shaft to connect the rigid coupler to the flexible coupler is made of 1566 hardened steel. The system requires 2 shafts. Turbocor has ordered the raw material and machined the shaft so they have an overall diameter is 25.4 with a 50µm precision. Figure 2 shows a section view of the rotating assembly, the yellow component represents the flexible coupler, the grey component represents the shaft and the blue component represents the rigid coupler.

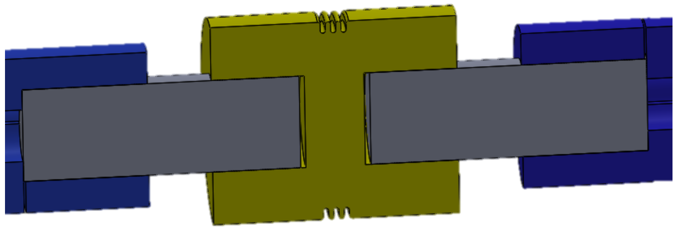


Figure 2: Section View of Rotating Assembly

To move the compressor during the alignment process the team designed brackets and tabs, which perform the horizontal and vertical movement respectively. Both parts are made of 1/4” steel and are welded to the structure. The tabs were simple to manufacture because they were designed to be a plate with a hole in the middle. The brackets were a little more complex to manufacture because each bracket is made with smaller plates, which needed to be cut and welded to be one final part. The manufacturing ran through the FSU/FAMU College of Engineering Machine Shop and took around 5 business days to cut all the pieces; the welding process will take a few days. A view with the brackets and the Tabs can be seen in the Figure 3, the red lines indicate the welding lines.

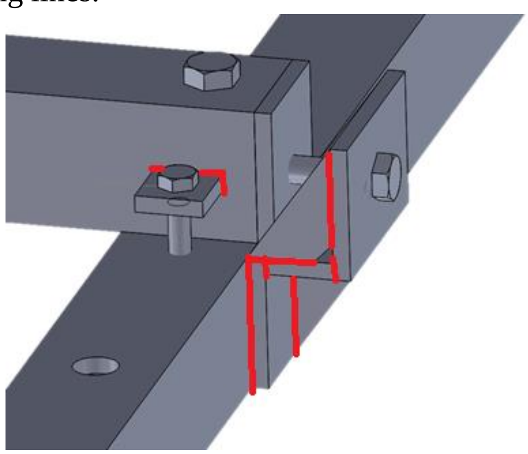


Figure 3: Brackets and tabs with the welding line indicated in red.

During the design process the team faced some problems. The first big problem was the budget constraint, initially the team designed the system to support one torque transducer between the two compressors. This was our ideal design but there is no torque transducer in the market that can handle different scenarios, high speed and low torque (40,000 RPM and 20 Nm) and low speed and high torque (10,000 RPM And 50Nm), so it would be necessary for two different torque transducers, one for each scenario. Another problem was that each transducer has a different shaft diameter, so again it would require different shafts and two more rigid couplers to connect the shafts to the torque transducer. Each torque transducer costs approximately $8,000. Our sponsor proposed to us to first prove the alignment system than buy the transducers.

To prove the alignment system we extended the shaft so it would fill the transducer space, reducing the costs and eliminating extra components, such as the second pair of rigid couplers. We presented this idea to our sponsor and then we faced another problem. If we replace the transducer by the shaft the system would not be rigid enough, which yields a small natural frequency (around 500 Hz). The maximum operational frequency is 40,000 RPM (667 Hz). For safety reasons the team assumed the minimum natural frequency for the system to be 700 Hz. To solve this problem we arranged the compressors to be closer to each other, to do so we designed the shaft to be shorter and thicker, which will increase the stiffness of the system and consequently the natural frequency. Our final design has a theoretical natural frequency of 708 Hz; the real value will be measured after the assembly in Turbocor's facilities.

After the design process the team came up with our final design, which is shown on the Figure 4.

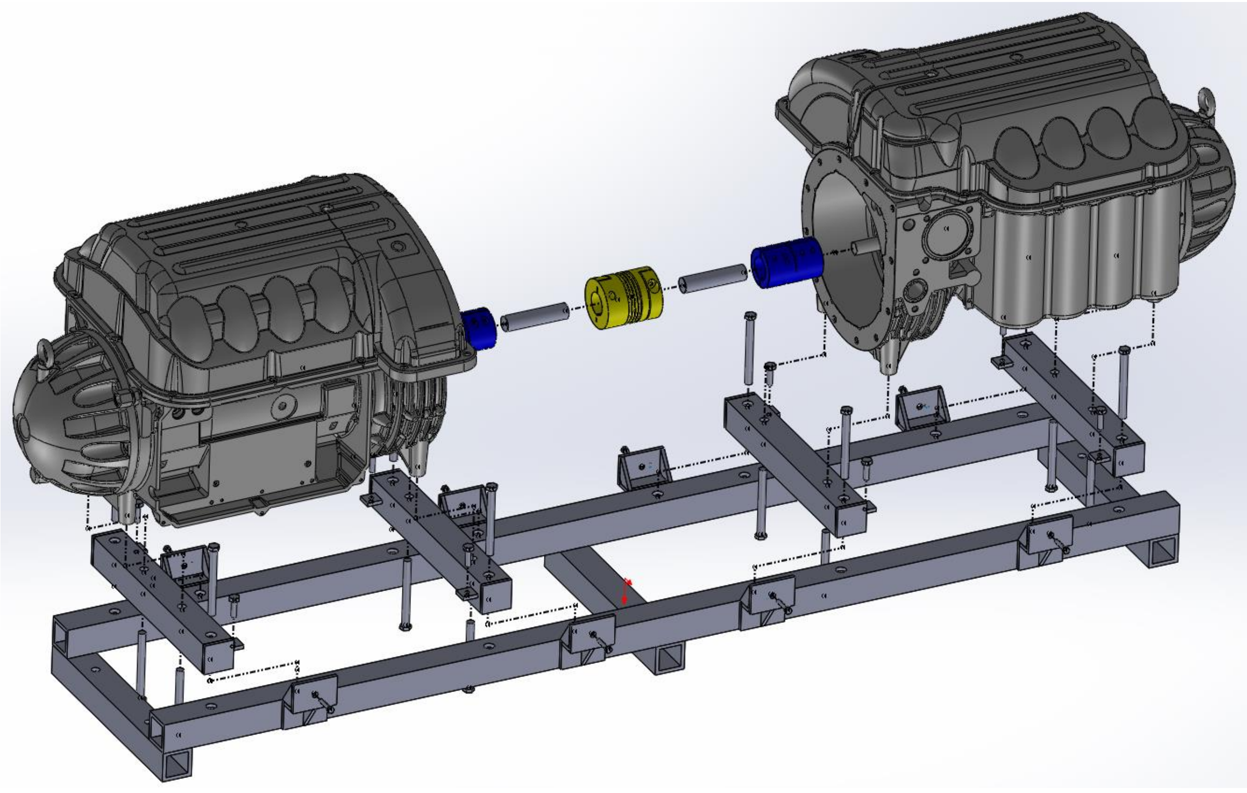


Figure 4 Exploded view of the final design.

With all the parts ready the assembly process can start, this implies that the base frame is machined, all the brackets and tabs are welded to the base frame, the rigid couplers are machined and the arrival of the flexible coupler is complete. The final assembly process should occur within one business day. The test rig will be used to test different compressors, so a fast assembly time is desirable, we estimate that the assembly time should take a couple hours, it would be interesting for a continuation of this project to develop a better way to assembly the system so it can be faster. The actual way to move the compressors during the alignment process does not indicate how much the compressor has been moved so another suggestion for a continuation would be to develop a system that indicates how much the compressor is moving.

3. Design for Reliability

Being that most of the components of this test-rig are made of steel, the machining process was relatively simple. Low carbon steel was chose for it's high yield stress and for the purposes of the test rig there have been no forces of concern that will even come close to causing plastic deformation in the steel components. The max stress calculated from the FEA analysis shown in Appendix E gives a maximum stress value of 0.3 MPa in the base frame and 118 MPa in the setscrew alignment mounts. Looking at the fatigue limit of steel below in Figure 5, the carbon steel endurance limit is beyond the actual maximum load stresses on the steel components so it can be determined that there is no load life cycle concern with the test-rig.

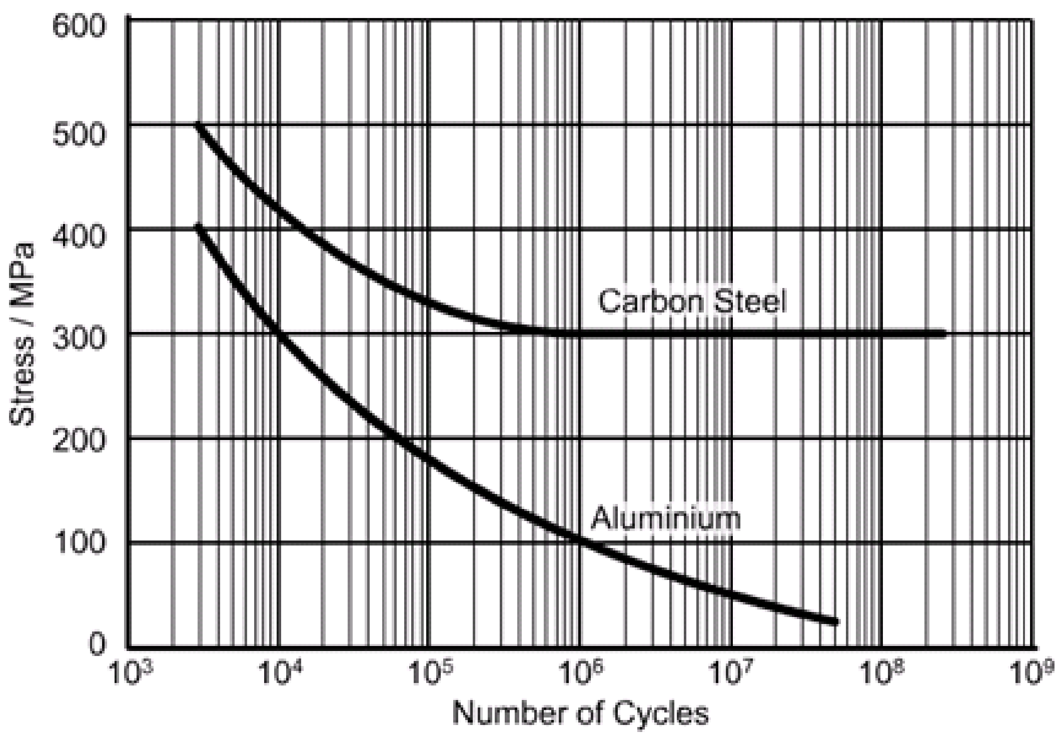


Figure 5: Stress (MPa) per number of cycles

Looking at the FMEA analysis performed by team 4, there are some areas of concern, which are unlikely but were deemed to have a 10 in severity. These potential issues are observed in the base frame, compressors, and shafts. In the base frame, the possible modes of failure are structural defects, rusting, and vibrations. Being that the base frame carbon steel runners are machined and welded, the processes could have adverse effects on the material's integrity. It is highlighted that fracture and bending could possibly occur due to mishandling and warping. This is not something that can be detected easily but the machinery and welders at Danfoss are of the highest caliber and the cause for a structural defect is highly unlikely (it received a 1 in occurrence). To go along with potential mishandling, rust can possibly plague this base-frame being that it isn't made of stainless steel. It is recommended that much care go into this frame; maintenance and an anti-rust paint should be applied to protect the rig.

Being that this designed to run at high speeds (50,000 rpm) and also at high torque (100 N-m), vibrational effects must be considered. The team is confident that the system can reach full operation for alignment purposes being that the natural frequency is 708 Hz, when the actual compressor natural frequency at around 667 Hz. Other considerations should be taken when Danfoss decides to run at full speed since the factor of safety calculated is only 1.13.

The compressor shaft and hardened shafts are a very key component to the rotating components of the rig. The hardened steel shafts could potentially fail like the steel in the base frame with a structural defect that could take place in the machining or balancing process. Since these shafts are suspended in air and attached to a flexible and rigid coupler, a structural defect that can cause fracture or bending can potentially launch a projectile. To combat this the team has proposed a safety shielding made of mild grade steel to encase the rotating assembly and protect the operators in the event. The compressor shafts are finely balanced and are attached with sensors that can stop the motors if a radial force of 200 lbf is experienced. The hard stop with all rotating components could cause damage to the compressor if the force is large enough. This is severe since these compressors are very expensive. As long as the machining process and assembly goes according to plan and all operators adhere to the alignment process specified in the operations manual, the test rig should be reliable and safe.

4. Design for Economics.

The total cost of the project thus far is $1,046.76. This price includes all hardware (nuts, bolts, washers, etc.), couplers, baseframe steel, and shaft material. As you can observe in Figure 6, the couplers and base frame make up the majority of the cost at $414.48 and $352.14. The shim stock was ordered in large amounts at multiple thicknesses so that the increments of height adjustment would range sufficiently. Fasteners and shafts were not a burden on the team for pricing.

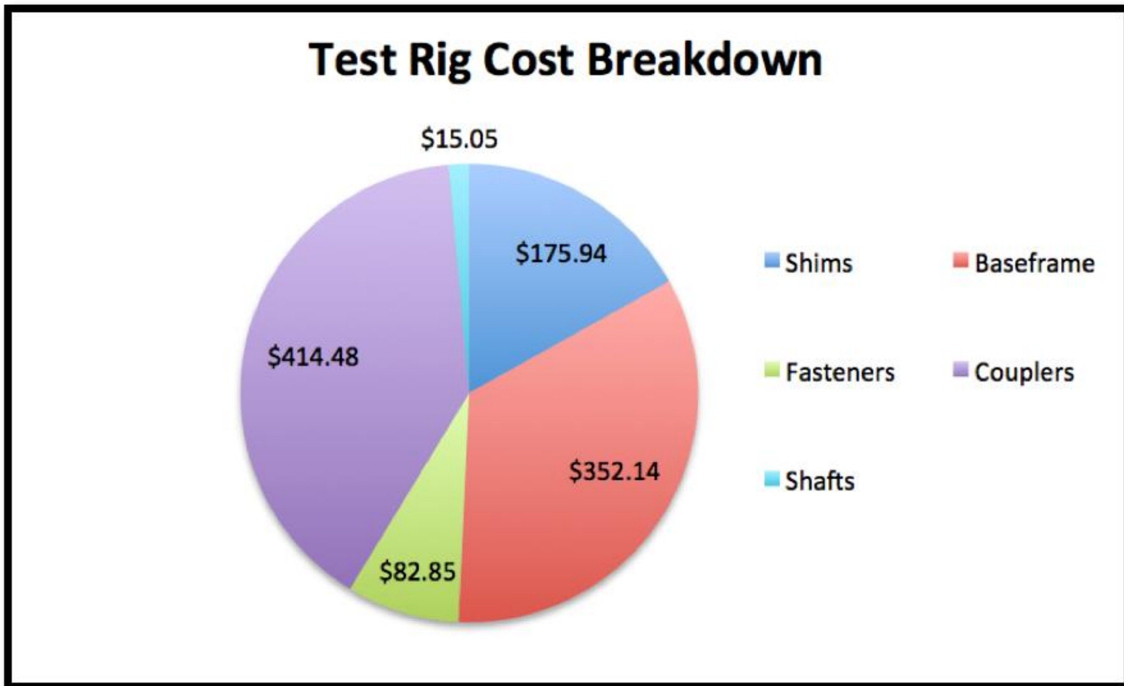


Figure 6: Test Rig Cost Breakdown chart.

Team 4 was never given a specified budget for the project and instead selected parts were purchased at Danfoss's discretion. As mentioned previously, the original design proposal called for two MAGTROL torque transducers. Each of these transducers was quoted at $8,000.00 and this would take the total for the design up to around $17,000.00, which was deemed too expensive at the moment by Danfoss. Another component that was omitted in the project due to price was the TSKA-31 laser alignment tool. It was valued at $3,500.00, but as previously stated the double dial indicator method is to be used in place of this. Overall the design has been cut by $19,500.00, but the base frame can still accommodate the original design if Danfoss decides to purchase the laser aligner and transducers.

There is not a current test rig that can handle the speeds that Danfoss requires and it was told to the team that the first phase in designing this test rig rung up a price near $70,000.00 - $80,000.00, as told by our liaison. This current design is simple, economical, and by design it is to be practical. All order forms are formally documented and can be viewed in Appendix B, Appendix C, and Appendix D.

5. Conclusion

Danfoss Turbocor is in need of a high-speed motor test rig that can withstand angular speeds up to 50,000 rpm and torques up to 100 N-m. There isn't a current system on the market that can perform under these conditions and Danfoss originally worked through a design running them up to $70,000.00-$80,000.00 in costs. Team 4 has offered an alternative design using a motor-generator set up with possibility for alignment validation and high-speed capability (if Danfoss desires). This design when ultimately completed with all transducers and laser aligners will save Danfoss close to $50,000.00 based on what the first phase was to cost. Fabrication of the prototype should be completed the week of April 8th, 2016 and undergo testing.

Reference

[1] http://eng.fsu.edu/me/senior_design/2015/team04/

Appendix A

Table 1: DFMEA Analysis

Item / Function	Requirement	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Classification	Potential Cause(s)/ Mechanism(s) of Failure	Current design: Controls Prevention	Occurance	Current design: Controls Detection	Detection	RPN	Recommended Action(s)	Responsibility	Action Results: Actions Taken	Sev	Occ	Det	R.P.N.
Frame	Support Compressors	Structural Defect (fracture and bending)	Misalignment during system operation	10		High heat from welding (warping), mishandling frame (fracture)	Careful handling and fabrication of components	1		2	20	Fracture: welding component. Bend: replace	Machinist, Welder Opera					
		Yielding in the steel components	Destruction of test rig	8		Overloading	Proper assembly of base frame and perform testing within specified	1		1	8	Shut down rig and replace damaged	Operator					
		Vibration in steel components	Damage to rig and compressors	10		Improper fastening	Be sure that bolts are torqued properly and rig is secured to floor	6		6	360	Shut down rig and be sure to fully fasten bolts	Operator					
		Rust	Could degrade material and cause serious structural issues	10		Lack of maintenance	Paint	7		7	490	Regularly inspect for rust, re-paint periodically	Operator, machinist					
Set screw brackets	Lateral Alignmnet	Bending	Unable to preform lateral alignment,	7		Overlaoding on set screw.	Ensure proper fastening and weld to frame properly	5		4	140	Shut down rig, re-thread hole and replace screw	Operator, machinist					
		Fracture	Unable to preform lateral alignment,	8		Overloading on set screw (compression)	Ensure proper fastening	2		4	64	Shut down rig, re-thread hole and replace screw	Operator					
		Hole shear	Unable to preform lateral alignment,	8		Overloading and vibrational effects	Ensure proper fastening	3		3	72	Shut down rig, re-thread hole and replace screw	Operator					
		Bolt shear	Unable to preform lateral alignment,	3		Vibrational effects and over torqued bolt	Ensure proper fastening	3		3	27	Shut down rig, re-thread hole and replace screw	Operator					
Shims	Vertical alignment	Warping	Uneven surface for compressor to sit, could compromise alignment	1		Compression from weight of compressor	cut out shims properly to not degrade material's integrity	1		7	7	Lift compressor and replace shim	Operator, Machinist					
Rigid Coupler	Couple compressor shaft to steel shaft	Vibrations	Could detatch from assembly or cause system to hit reasonance if	6		Improper fastening, improper balancing	Make sure to machine to proper balance specifications, fasten fully	4		4	96	Remove, balance, and re-fasten	Operator, Machinist					
Flexible coupler	Couple both steel shafts	Vibrations	Could detatch from assembly or cause system to hit reasonance if	4		Improper fastening, improper balancing	Make sure to machine to proper balance specifications, fasten fully	3		4	48	Remove, balance, and re-fasten	Operator, Machinist					
Steel shafts	Couple both compressors	Structural Defect (fracture and bending)	Could shear and turn into projectile and cause damage to compressor	10		mishandling (fracture), stall torque, poor machining	Careful fabrication and proper test rig operation	2		2	40	Replace shaft	Operator, Machinist					
Compressor	Motor and generator for test rig	Stall	Could damage inner components or onboard computers	10		improper assembly and coupling of shafts	Follow operations manual and have trained technician running	3		7	210	Remove compressor, recalibrate	Operator					

Appendix B

Table 2: Price order sheet 1

PURCHASE ORDER REQUISITION

Danfoss TURBOCOR

Vendor: McMaster Carr

DATE: **10-Mar-16**

DATE REQUIRED:

CAPITAL EXPENDITURE (please tick):

CURRENCY: **USD**

Contact: www.mcmaster.com

NOTE: THIS IS NOT A PURCHASE ORDER AND CANNOT BE ISSUED TO SUPPLIER

TURBOCOR P/N	DESCRIPTION	VENDOR P/N	QTY	UNIT PRICE	TOTAL PRICE	PROJECT NUMBER	ACCOUNT NUMBER
	Alloy Steel Socket Head Cap Screw, Thread size: 3/8"-24, Length: 2". Package Qty: 5	90044A158	2	$8.87	$ 17.74		
	Machinable-Bore One-Piece Clamp-On Rigid Shaft Coupling	3084K34	2	$42.46	$ 84.92		
	Hardened Shaft, Steel, Diameter:1", Length: 10"	6061K608	1	$15.05	$ 15.05		
	General Purpose Low Carbon Steel, 1/4" Thick, 2" Width, 3ft Length.	8910K557	1	$ 20.82	$ 20.82		
	High Strength Steel Cap Screw, Zinc Yelloww Chromate, 1/2"-13, Length: 5 1/2", Partially Threaded (Pack Qty: 5)	91257A734	2	$ 12.58	$ 25.16		
	Extra-Wide Hex Nut, Zinc Yellow Chromate, 1/2"-13 (Pack Qty: 25)	96460A370	1	$ 10.11	10.11		
	Over Sized Flat Washer, Zinc Yellow-Chromate Plated, 1/2" Screw Size, 0.531" ID, 1.062" OD. (Pack Qty: 25)	98025A133	1	$ 11.44	$ 11.44		
	High Stength Steel Cap Screw, Zinc Yellow Chromate, M12 x 1.75, Length: 70mm, Partially Threaded. (Pack Qty: 5)	95327A695	2	$ 9.20	$ 18.40		

FREIGHT: A) PREPAID (included)
B) PREPAID & CHARGE
C) COLLECT
D) FIXED AMOUNT amount

TOTAL $ 203.64

Special instructions:

Prepared by: (Print name)

Approved by: (Manager)

Approved by: (Director)

PUR-00007F01

Appendix C

Table 3: Price order sheet 2

PURCHASE ORDER REQUISITION

Vendor: McMaster-Carr

DATE: **22-Feb-16**

DATE REQUIRED: **ASAP**

CAPITAL EXPENDITURE (please tick):

CURRENCY: **USD**

Contact: www.mcmaster.com Phone: 404-346-7000

NOTE: THIS IS NOT A PURCHASE ORDER AND CANNOT BE ISSUED TO SUPPLIER

TURBOCOR P/N	DESCRIPTION	VENDOR P/N	QTY	UNIT PRICE	TOTAL PRICE	PROJECT NUMBER	ACCOUNT NUMBER
	2"x2"x1/4" Low Carbon Steel tube, Length: 6ft	6527K614	4	$69.82	$ 279.28		
	2"x1/4" Low Carbon Steel strip, Length: 6ft	8910K557	1	$35.89	$ 35.89		
	2"x1/4" Low Carbon Steel strip, Length: 2ft	8910K557	1	$16.15	$ 16.15		
	Brass Shim Stock, 6"x60" , Thickness: 0.001"	9504K41	1	$ 11.53	$ 11.53		
	Brass Shim Stock, 6"x60" , Thickness: 0.003"	9504K45	1	$ 11.42	$ 11.42		
	Brass Shim Stock, 6"x60", Thickness: 0.006"	9504K49	1	$ 16.97	$ 16.97		
	Brass Shim Stock, 6"x60", Thickness: 0.009"	9504K53	1	$ 22.65	$ 22.65		
	Brass Shim Stock, 6"x60", Thickness: 0.012"	9504K55	1	$ 24.30	$ 24.30		
	Brass Shim Stock, 6"x60", Thickness: 0.02"	9504K58	1	$ 35.60	$ 35.60		
	Brass Shim Stock, 6"x60", Thickness: 0.031"	9504K6	1	$ 53.47	$ 53.47		

FREIGHT: A) PREPAID (included)
B) PREPAID & CHARGE
C) COLLECT
D) FIXED AMOUNT — amount

TOTAL $ 507.26

Special instructions:

Prepared by: ______________________ (Print name)

Approved by: ______________________ (Manager)

Approved by: ______________________ (Director)

PUR-00007F01

Appendix D

R+W
COUPLING TECHNOLOGY

R+W America
1120 Tower Lane
Bensenville, IL 60106
Phone: 630-521-9911
Fax: 630-521-0366
Email: info@rw-america.com
Web: www.rw-america.com

Danfoss
Mr. Kevin Lohman
1769 E. Paul Dirac Drive
Tallahassee, FL 32310

Fax: kevin.lohman@danfoss.com

SALES QUOTE # 65010	
Date	03-10-2016 Page 1/1
Ref.# / Cust.#	65010 / 209644 (40)
R+W contact	Leon Voskov

Dear Kevin:

Thanks for the opportunity to quote this project. We are pleased to offer the following:

Line	Qty.	Description	Unit Price	Total
(1)	1	**Bellows Coupling** BK2 / 150 / 95 / 25.4 / 25.4 Bore D1: 25.4 H7 Bore D2: 25.4 H7	329.56	329.56 USD
			Total	329.56 USD

Payment Terms Net 30

Lead time: 2-3 weeks

Feel free to contact us with any questions or changes.

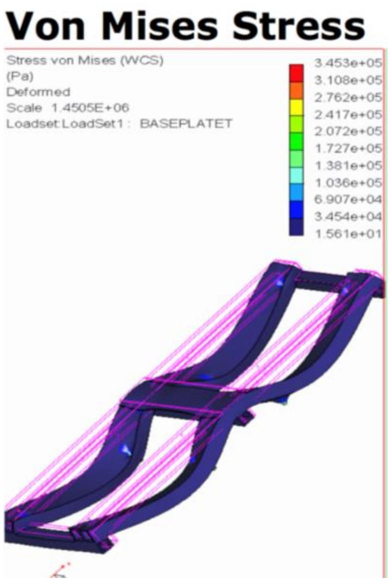
This quote is valid for 3 months and subject to our general terms and conditions.
Terms and conditions can be found at: info.rw-america.com/organization

Best regards,

R+W America
Leon Voskov

Figure 7: R&W Price quote

Appendix E



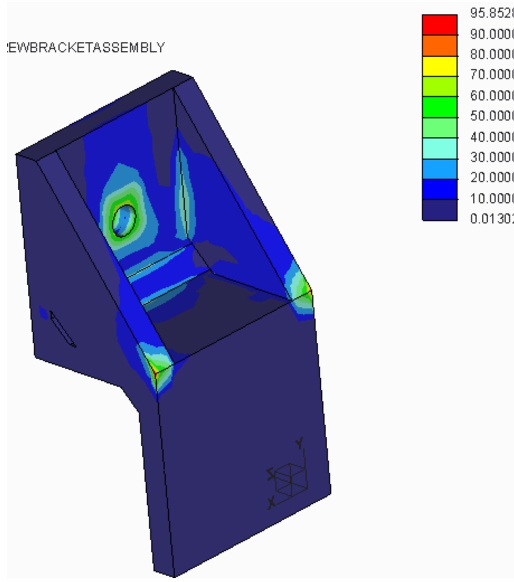


Figure 8: Base frame **Figure 9: Setscrew mount horizontal**

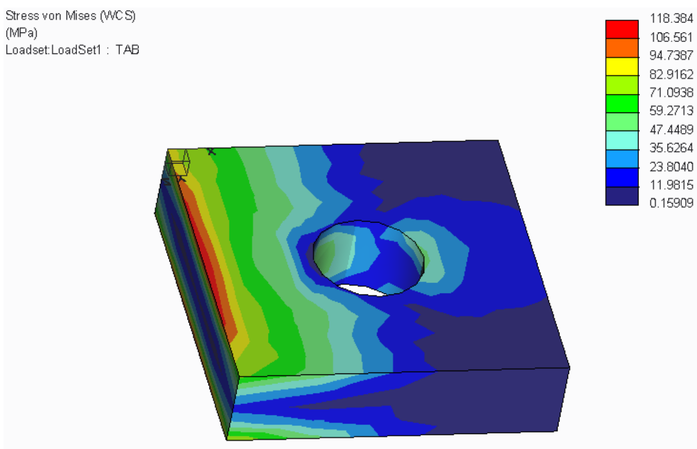


Figure 10: Setscrew mount vertical