Midterm 1 Report

Team 5

High Speed Motor Test Rig

Members:

Alexander Jurko adj13c

Jonathan De La Rosa jgd12

Fehintoluwa Aponinuola fehintoluwa1.aponinu

Jack Pullo jap12c

Faculty Advisor/s

Dr. Patrick Hollis

Sponsor

Danfoss Turbocor

Instructors

Dr. Gupta and Dr. Shih

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Table of Contents

Table of Figures

Table of Tables

ABSTRACT

Danfoss Turbocor is the only company that manufactures compressors with magnetic bearings. Magnetic bearings eliminate friction on the shaft inside the compressor, so it's extremely efficient and requires significantly less coolant to run at high speeds in comparison to other compressors. Turbocor's compressors use a process called "levitation" where the magnetized shaft literally levitates inside the compressor via electromagnets, and then it is ready to run. Turbocor approached the FSU-FAMU College of Engineering with the goal of receiving a high-speed motor test rig where they could test the power, efficiency, and heat management of their compressors. Instead of connecting a motor to the compressor, they will be connecting identical compressors while treating one as a motor and the other as the generator in the rig. Last year's senior design team managed to get the two compressors to run, however it was at low speeds due to multiple issues they came in contact with. Budget constraints restricted their availability for equipment, which is important when dealing with compressors that shut down when the sensors detect a displacement larger than their accounted threshold. Another one of their main struggles was dealing with the levitation of the shaft, this became an issue only when an external shaft was introduced (which is necessary when connecting a transducer and another compressor). The sensors on the compressor are dealing with the microns in terms of displacement, so any small issue with vibration or misalignment will cause the compressor to shut down or not even achieve levitation to start. With our increased budget this year, team 5 plans to incorporate ideas from last year's team with our own to align the compressors as best as possible with a laser alignment tool, eliminate the issue in axial displacement with 2 double-flex disc couplings system, as well as incorporate a transducer fixed to the frame to actually test the compressor as it runs. Fixing the transducer to the frame will hopefully prevent the compressors from fighting each other; as one compressor pulls one direction, it will only pull on the transducer and not on the other compressor which was an issue faced by last year's design team.

ACKNOWLEDGMENTS

Thank you to Danfoss Turbocor, for giving senior design team 5 the opportunity to tackle the task of the high speed motor test rig. Special thanks to William Sun, Kevin Lohman, and Julio Lopez for the time and effort they've spent teaching us about their compressors and how to use the software to run them. Thank you also to Dr. Gupta and Dr. Shih for the help and guidance as professors for senior design, as well as the staff advisor Dr. Hollis for answering specific questions about different mechanical aspects of the system.

1. Introduction

Danfoss Turbocor is a leader in compression technology, and is interested in running tests on their compressors to observe how they act when the speed increases. The company designs compressors for heat, vacuum, and air conditioner industry. Normally Turbocor runs their magnetic bearing compressors with internal shafts (inside of the compressor). Turbocor achieves a high efficiency due to a combination of magnetic bearings, which uses magnetic fields to create a contact free system between the shaft and bearings allowing high speeds (up to 40,000 RPM), and variable-speed centrifugal compression, which allows the use of the compressor with the rotation for the highest quality performance.

Danfoss Turbocor is still looking for a way to be able to test their compressor models at very high speeds along with maintaining the speed for longer periods of time. This year's team is currently building off of what last year's senior design team produced.

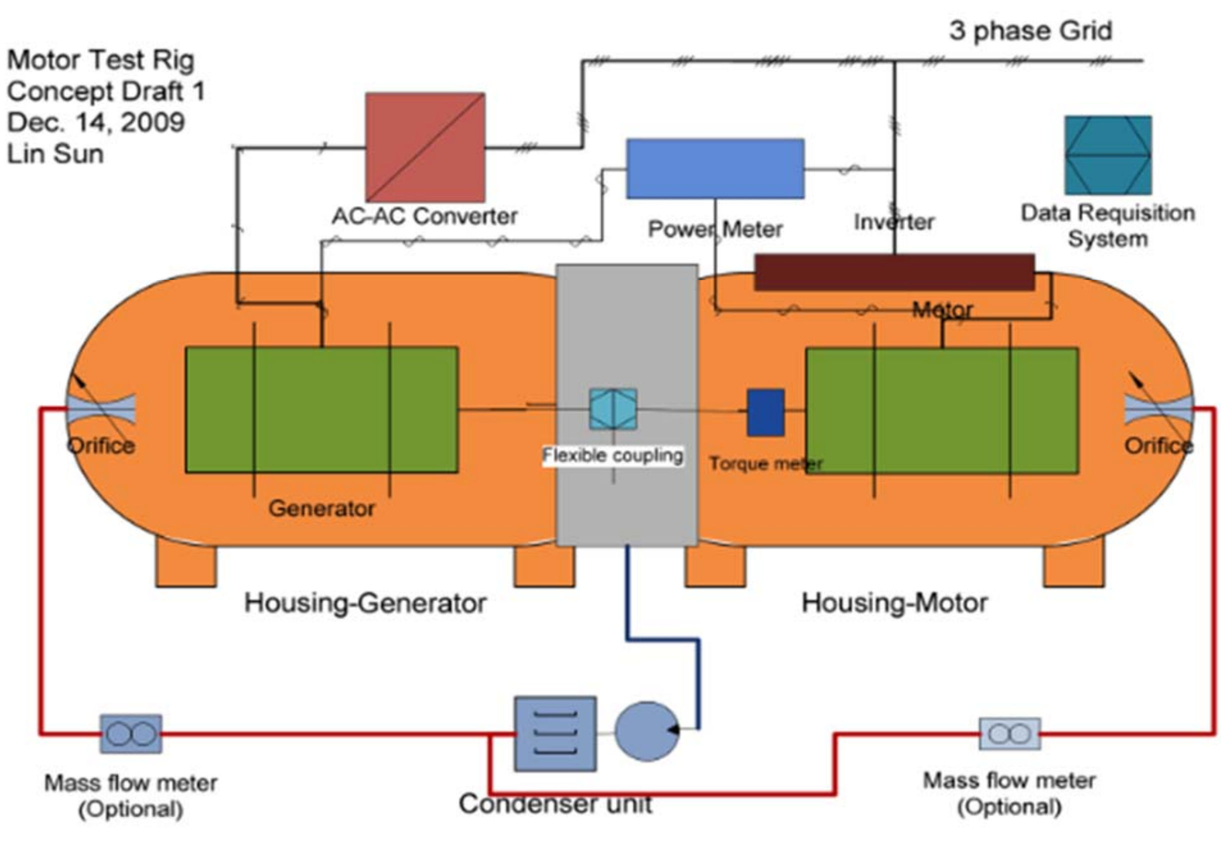


Figure 1 Motor Test Rig [1]

The senior design team that was tasked with this project last year was able to create a design that ran at low speeds. Last year's team designed an adjustable frame that was able to change the position of two compressors placed on it, align them, and run the compressors at low speeds. They designed a system with a flexible coupler, which accounted for a certain amount of misalignment, but that caused problems later on in their project.

Instead of having a laser alignment tool, team 4 ended up buying a dial alignment system to align the shafts with the compressors. This may have been a cause as to vibration issues occurred when testing the design. Although they were able to stay levitated for a short amount of time, the vibration was bad enough that it caused the compressors to completely shut down. Last year's team improvised and added duct tape in order to dampen the vibration. This was done in order to be able to attempt to operate the machine. With the help of Julio Lopez, the test rig was able to run up to about 700 rpm before the vibrations caused the system to shut down.

The figure below is the CAD drawing of the final design that last year's team ended up creating.

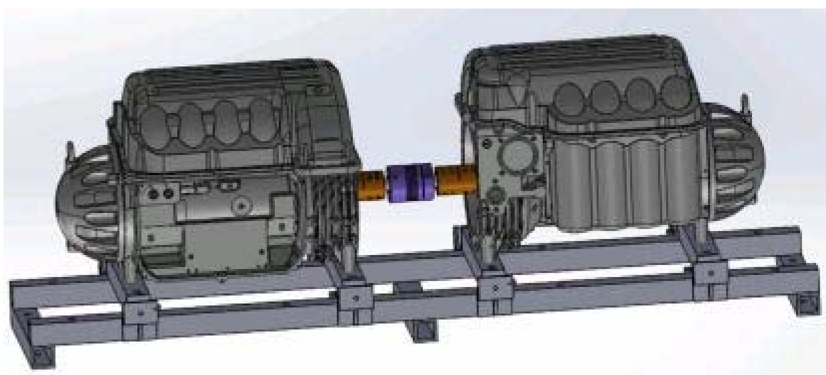


Figure 2 Last Year's Team's Final Design [1]

The overall goal for this project is to be able to have the high speed motor test rig up and running up to about 10,000 rpm without any issues by the end of spring 2017. In order to achieve this, all sub goals and objectives that are in place for Fall 2016 and Spring 2017 semesters will need to be met. Team 5's objectives are somewhat straightforward due to the achievements that last year's senior design team accomplished with the many constraints that were placed upon them. Team 5 has stated that having consistent levitation to allow the compressors to operate properly is the most important item. Following this would be having accuracy in regards the alignment and balancing in the test rig, then the desired durability and safety will be achieved as well.

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

2. Research/Analysis

2.1 Magnetic Bearings

Turbocor's compressors excel in four different areas: magnetic bearings, variable speed, digital control, and centrifugal compression. Magnetic properties are used to cushion the compressor's rotor shaft and impellers, providing contact-free levitation. Having one axial and two radial bearings, as well as nine sensors that provide real-time orbit feedback, the shafts are centered with very high precision. The sensors will shut down the compressor if a load of 200 lbf, roughly 270 Nm, is placed on the shaft that the magnetic bearing is holding The figure below represents the shaft beings supported by the front and rear radial bearings along with an axial bearing. The sensor rings are the devices that provide feedback that will control magnetic fields produced by each bearing.

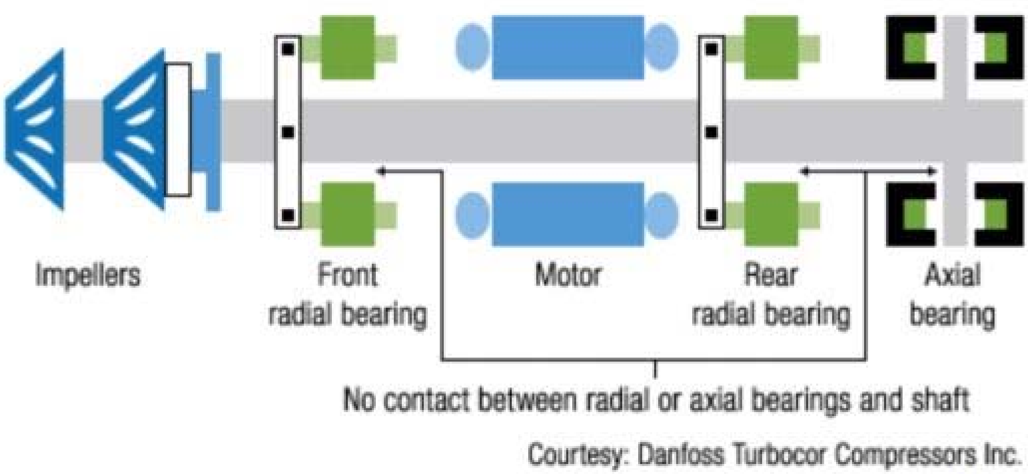


Figure 3 Schematic of Magnetic Bearings [1]

2.2 Laser Alignment Tool

The alignment between two of these magnetic bearing compressors is extremely important. Any misalignment will be amplified when running at high speeds, and this in turn can shut down the compressor. The previous design team used a dial alignment system to align the shafts, but that has a significant amount of human error. They researched and would have liked to use a laser alignment tool, but due to budget constraints that was impossible. This year, team 5 has used their insight to select a laser alignment tool appropriate for the project. The figure below shows the SKF TKSA 31 Laser Alignment Tool, which is around $3,250.00. This works by using three points, rotated around the shaft, to create a plane and determine the offset in each direction digitally (shown in figure 4).

Figure 4 SKF TKSA 31 Laser Alignment Tool [2]

2.3 Jaw Couplings

Initially, team 5 had designed our additions this year with two jaw couplings. Jaw couplings were a desired option because they are exceptional at dealing with axial misalignment and displacement. By using an elastomer spider located in the middle, shown in figure 5, the jaw coupling is able to rotate with a range of displacement that is much larger than other types of couplings. Not only does it account for misalignment, but it also has no metal on metal contact when it is in the threshold of the elastomer spider separating the two rigid portions of the coupling. One of the core issues that the previous team faced was the compressors fighting one another once the system started running at high speeds. By introducing the jaw couplings to the rig, it would replace the rigid couplings attached to the internal shafts as well as replace the flexible coupling located in the middle of last year’s design. By reducing the number of parts, this should increase the natural frequency to the required 904 Hz that the compressors are required to operate at.

Figure 5 Jaw Coupling [3]

2.4 Double-Flex Couplings

After speaking with the sponsor multiple times, team 5 came to the conclusion that the jaw couplings will not meet the requirements specified by the sponsor. The idea is to run the rig at 40,000 rpm eventually, and that needs to be kept in mind at all times. The jaw couplings might have been a solution for the lower speeds, but they do not operate well at the higher speeds. Thus team 5 has selected the double-flex disc couplings provided by Zero-Max.

Figure 6 Double-Flex Disc Coupling [4]

The double-flex disc couplings are available from stock, but limited in their rpm. The reason team 5 has chosen these couplings is because of the potential from Zero-Max. The couplings consist of two torsionally stiff sections in combination with a unique patented disc that allows for a certain amount of misalignment and axial loads which is shown in figure 5. They assured team

5 that they make custom-made couplings for similar projects frequently, and they estimated that for the 40,000 rpm the price per coupling would be roughly $1,500-2,500. They are expensive because they are most likely made out of carbon fiber and balanced in order to meet the high requirements of rpm and torque rating. Team 5 has been designing while keeping in mind future teams down the road, so the design will most likely reach the goal this year but pave the road ahead for simpler additions to the rig. When Turbocor produces the safety measures to hopefully increase the rpm capabilities with these external additions, the logical choice would be to look at the same company's custom couplings to crank up the speed. The double-flex disc coupling data sheet is located in Appendix A, showing the variations of sizes available as well as the rotational speeds and torque ratings.

2.5 Torque Transducer

Along with replacing the current couplings, a torque transducer will be added on this year. A torque transducer is used in order to gather data on the forces being applied to it from the motor and generator of the test rig. Team 5 has been thoroughly researching different types of torque transducers that could be used. The conditions that the torque transducer must be able to withstand are the power output of the compressor and also the high speed while the motor test rig is operational. The transducer should be able to withstand such conditions for a long duration of time. The team is in the process of speaking with a compressor specialist from Turbocor to understand the power output of the compressor.

The Magtrol 308 TMHS Series is the potential choice for the torque transducer. A schematic of this transducer is shown in figure 7. This torque transducer can withstand a speed of 50,000 rpm and 20Nm force. This choice, however, is subject to change, depending on the outcome of the meeting scheduled with the specialist in charge of Danfoss Turbocor's compressors.

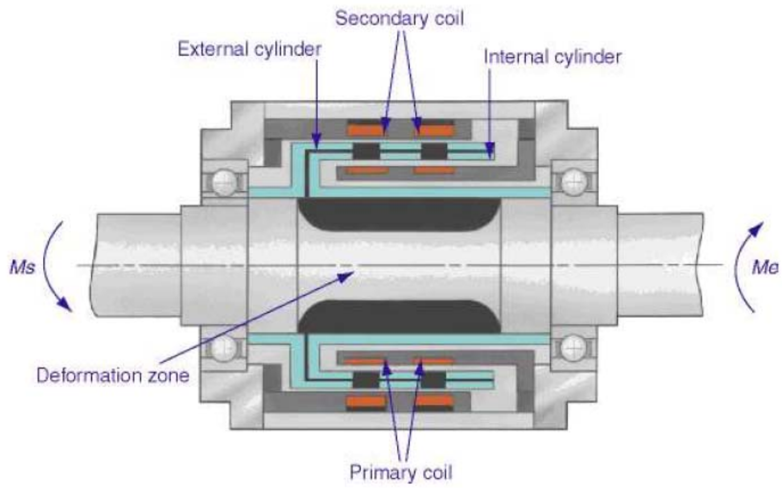


Figure 7 Schematic of Magtrol 308 TMHS Series Torque Transducer

There are two ways to connect a transducer to a system: the first is for it to be free-floating but strapped down, as shown in figure 8, and the other is for it to be completely fixed or grounded, as shown in figure 9. The benefits of a free-floating torque transducer is that it accounts for any misalignment, especially axial; however, fixing the transducer is much better for high-speed applications. The only issue with this method is that much more precision in the alignment of the system is required, compared to if it were free floating.

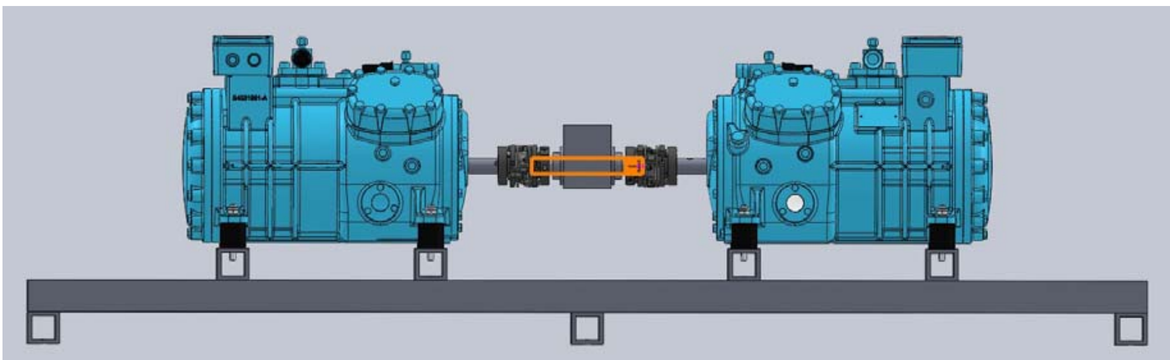


Figure 8 Free Floating Transducer (Strap Down)

As shown in figure 9, the torque transducer that will be incorporated in the test rig will be fixed on an adjustable stand, to allow for adjustments during alignment.

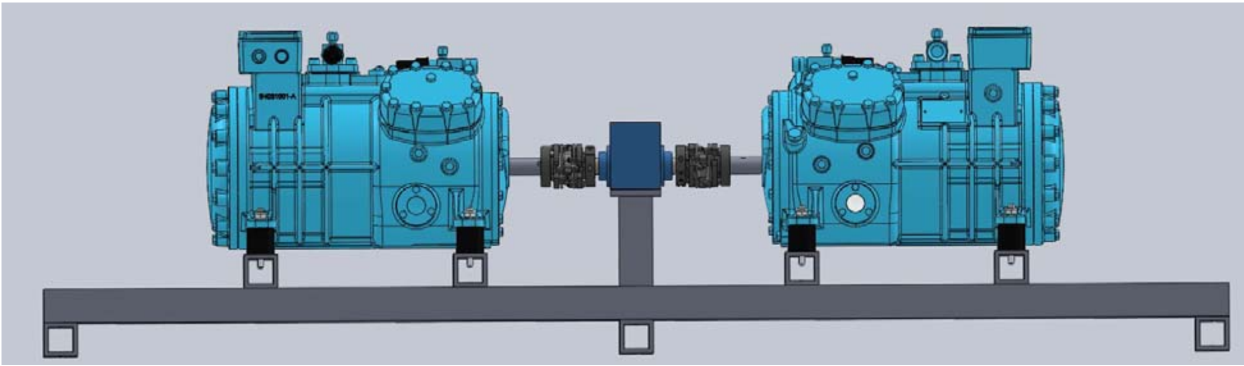


Figure 9 Fixed Transducer (Mounted)

Based on all the information that has been gathered, team 5 has come up with an initial design of what the motor test rig would look like. A torque transducer would sit on a stand in the middle of the frame. From there two curved jaw couplers would be attached on both sides of the transducer. A shaft is then attached on both ends of the couplers that are then attached to the rigid couplers from last year. The rigid couplers are then attached to the compressors as seen in figure 9 above.

2.6 FMEA

When choosing what components needed to be selected, an FMEA was conducted in order to see what components were the best to go with. The FMEA, shown in the following four figures, below shows the components that were selected for the design that the team is going forward with.

Table 1 FMEA Analysis of Laser Alignment Tool

Key Process Step or Input	Potential Failure Mode	Potential Failure Effects	SEV	Potential Causes	OCC	Current Controls	DET	RPN
SKF TKSA 31	Not properly installed correctly	Having the motor test rig misaligned, resulting in higher chance of failure	8	Handling Error	7	Reading the instructions. Along with watching videos from the manufacturer on how to install the equipment properly	7	392
	Faulty equipment	Not being able to have the motor test rig properly aligned resulting in a very high chance of failure	8	Manufacturer	3	Standard Operating Procedures	9	216
	Lack of understanding with properly operating the equipment	Not having the full potential of the device. This results in not having a proper alignment	7	Human Error	5	Properly reading the instructions and watching tutorials on the device to ensure proper use of it	8	280
	Not properly fitting	Failure to align	10	Lack of Research on the range in which the laser aligment tool can attach to	4	Extensive Research into what range the laser alignment tool can attach	8	320

In the table above, the SKF TKSA 31 laser alignment tool had an FMEA done in order to see whether or not it would be good to purchase. Some of the potential failures that the team saw that could occur include; not properly installing the system correctly, the component being faulty upon arrival, lack of understanding when it comes to using the laser alignment tool, and the tool itself not fitting on the coupler. From there some causes of said error were determined followed by what controls can be done in order for the failure to likely never happen. The scores were added up from each category to make the Risk Priority Number (RPN). The highest risk that the laser alignment tool has out of all the failures is not installing it correctly. The team decided that the laser alignment system was a go based upon the results last year's team had with a dial alignment system. The laser alignment system was included in each design's FMEA.

Table 2 FMEA Analysis of Double Flex Disc Couplings

Double Disk Coupler	Couldn't handle desired rpm	The coupling getting damaged. Compressor shutting down due to possible vibrations that would occur. Also potentially damaging other components on the test rig. Wouldn't meet the end result.	10	Lack of coupler diversity within the prexisting market with such desired rpm speeds	7	Contact distributors to see if it's possible to contrsuct a custom curved jaw coupling	4	280
	Not properly installed	Damage can occur	8	Human Error	7	Reading the instruction on how to install the curved jaw coupling correctly	8	448
	Exceeded displacement range of the coupler	Causes the compressor to shut down. Potentially damaging the couper itself	10	Selected coupler from research done by the team	4	Further research on the jaw couplings. Discussing with professionals in the industry to select the right coupler	7	280
	Faulty Equipment	Wouldn't function properly in order to test the design	10	Manufacturer	3	Standard Operating Procedures	9	270

The next component that the FMEA covered was the double flex disc couplings that were to replace the both the flexible bellow coupling and rigid couplings. Some of the failures that the team came up with that could happen would be not having the coupler properly installed, exceeding the displacement of the couplings, having the coupling come in faulty, and not properly installing the coupling correctly. Each one of those potential failure would be catastrophic when running the high speed motor test rig. Reasons for why such failures would occur were written in the figure above with the current controls that team could do to avoid those situations. When comparing the double flex disk couplings to the flexible bellow coupling, the rpn was higher on more points in the flexible bellow coupling and rigid couplings than the double flex dick couplings. So the team chose the disk couplings moving forward to replace the flexible bellow coupling and rigid couplings. This will make the design less complex by removing a coupler needed.

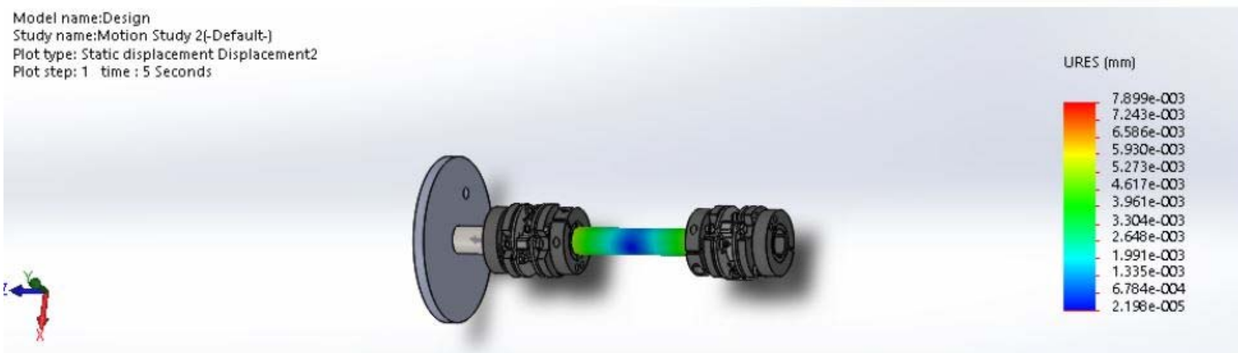
Table 3 FMEA Analysis of Shaft and Torque Transducer

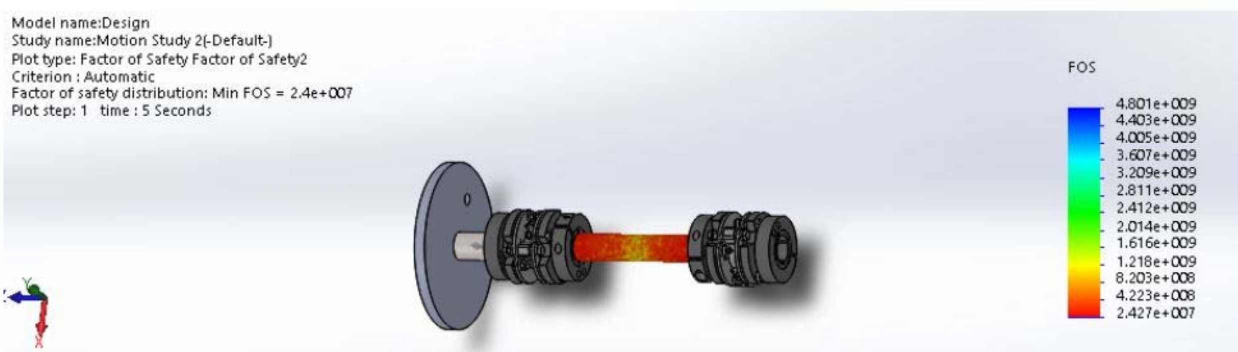
Shaft	Improper Length	Won't be able to properly connect the couplers to the compressors	10	The deisgn team	5	Adjustment of the compressors on the frame. Checking the legnth in between the compressors beforehand to properly purchase the correct shaft	7	350
	Frequency Issues	The software inside the compressor wouldn't operate properly	6	Complexity of the design due to the additional components	10	The Hammer (Or Dong) Test to see the frequency with the additional components. (Hard to test without purchasing)	10	600
Magtrol 308 Torque Transducer	Not Mounted Correctly	Can result in the compressor fighting each other, which will shut down the motor test rig	8	Human Error	6	Make sure to properly mount the transducer into the stand so that it is rigid	7	336
	Not Properly Installed	Damage can occur	9	Human Error	7	Read the instructions carefully in order to install properly	8	504

The last two components that are having an FMEA done are the shaft and the torque transducer. The two main failures that the team thought could likely happen are having the shaft at an improper length and frequency issues when connected to the compressor. If the shaft would have an improper length, the responsibility would fall on the design team for not accurately measuring the distance between the two couplings for the shaft to go through. Some solutions to the potential failures are listed as well. The shaft is incorporated in every design concept the team had, but it would only be a phase one for the current design as it would mimic the torque transducer while the team waits for the transducer to arrive when ordered. The two issues that popped up for the torque transducer is not properly installing the transducer to the couplings and not having it mounted. Not having it mounted properly poses a serious concern since the transducer would not be rigid and would allow the compressors to fight each other again when the high speed motor test rig would run. That would result in the compressors shutting down from the vibrations that would be occurring. Human error accounts for both failures occurring so the solution would be for the team to make sure it is properly installed and mounted with the help of workers from Turbocor.

2.7 FEA

To determine if a material or design will fail, it becomes essential to analyze specific properties before purchasing expensive components within the system such as the custom coupling or a torque transducer. For the conceptual design that was chosen, the von mises stress, displacement, and factor of safety were the most critical modules to produce and analyze. To calculate and verify results SolidWorks, Pro E, and ANSYS computer software's were used. It was important to use three finite element softwares to verify that the data was accurate. The overall objective of this design was to test the efficiency of the compressor. The critical parts to verify before purchasing included a carbon steel shaft, structural steel base plate, and the couplings. A motor was placed onto the shaft at 2000, 5000, 10000, and 30000 RPMs. For the shaft to fail, the highest von Mises stress applied 220594000 N/m^2, have a displacement of more than 1 mm, or a factor of safety less than 1. The highest stress applied was at 30,000 rpm and was approximately 1.5 N/m^2. The maximum displacement, also while the compressor was operated at 30,000 rpm, was roughly 4.7*10^-003 mm. The Figure of the data can be seen below.





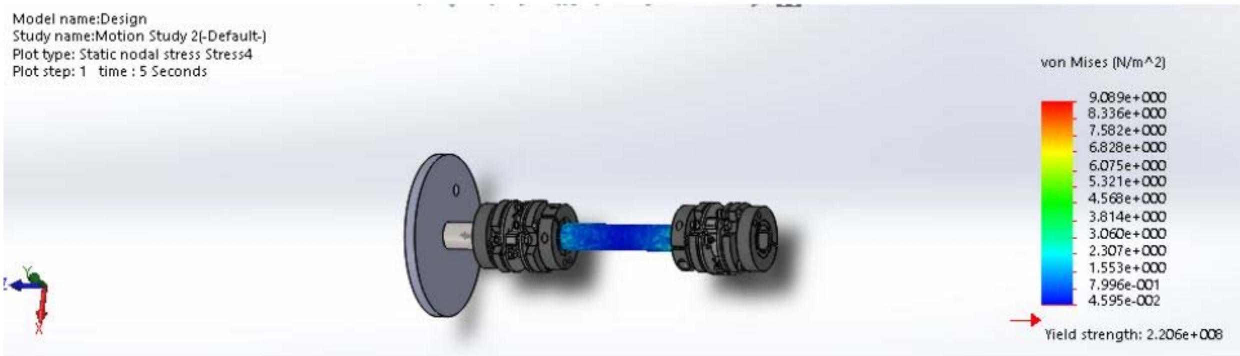


Figure 10 Displacement, Factor of Safety, and Von Mises Stress at 30,000 rpm

Although as the compressor elevated in speed, the factor of safety always stayed within excellent range. The lowest factor of safety received was 2.4E007. These results on the shaft prove that it would not fail and drop the compressors while running. A distributed load of 600 pounds was placed at the critical points on the stand. The maximum displacement was relatively low with a value of 1.034E-7 m. The highest von Mises stress received was roughly 3.45E04 Pa, which did not exceed the yield strength 513,613,000 Pa. Therefore, the results established that the concept of this high speed motor test rig would not fail during operation.

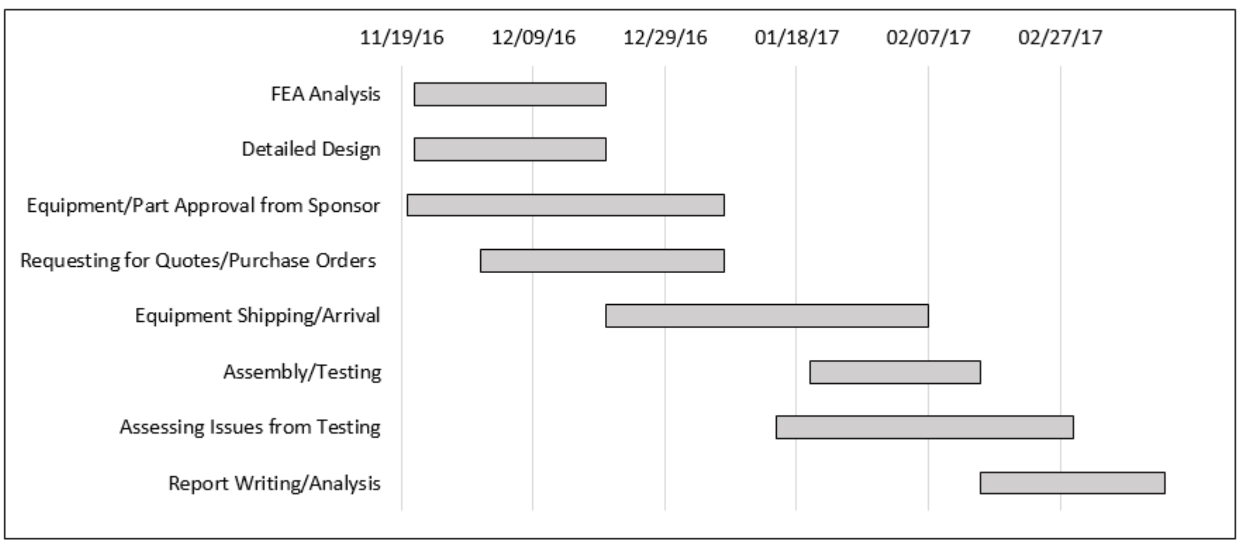
3. Project Management

Throughout the process of designing and building this motor test rig, the following team members have been delegated to specific tasks in order to maintain a consistent work environment:

- Alex Jurko is the team leader, assigned with communicating clearly and frequently with both sponsors and advisors as well as keeping the team focused on the task at hand while staying on schedule.
- Jonathon De La Rosa is the lead mechanical engineer, assigned with designing CAD or Pro E drawings of our design as well as doing most calculation work with assistance of the other members.
- Fehintoluwa Aponinuola is the lead web designer, assigned with keeping the senior design team's website up to date with deliverables as well as being delegated to whatever task needs assistance if the website is up to date.
- Jack Pullo is the financial advisor, assigned with managing the budget of the project while researching alternative options for cost-effective solutions as well as being delegated to whatever task needs assistance at the time.

The following Gantt chart, Table 2, has been created to show the design team's hopeful schedule for the remainder of the fall semester, winter break, and spring semester ahead.

Table 4 Gantt chart



4. Conclusion

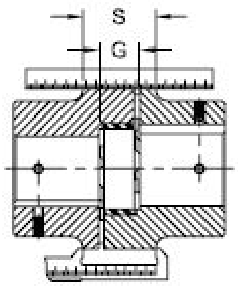
Senior design team 5 plans to have their rig up and running around the middle of February 2017. The tools that the team plans on using are the SKF TKSA 31 Laser Alignment Tool, which will improve upon the alignment process in comparison to last year's design team which used a dial alignment system. The laser alignment tool eliminates the problem of "eyeballing" the dial readings, and reduces human error in the alignment process. Team 5 also plans to use the Magtrol TM 308 torque transducer that was approved to analyze the speeds (up to 50,000rpm) and the torque (up to 100 N*m) for the rig in order to observe how the compressors react to an external shaft at high speeds. In order for team 5 to overcome the obstacle of levitation, the team plans to use double-flex disk couplings to connect the shafts replacing the rigid and bellow flexible couplings. Double-Flex disk couplings were deemed the best option for this project because they are resistant to vibration and are able to adjust in the axial direction. This axial direction displacement is believed to be the issue that the previous design team faced when testing their rig, and prevented the system to run properly. The shaft that connect two the couplings will only be in for phase one of the design. The difference between phase one and the following phase is replacing the shaft with the torque transducer. Therefore, the shaft will mimic the torque transducer dimensions when connected to the couplings while the team waits on the torque transducer lead time. As of the last meeting with the sponsor, all the parts have been approved and the team is currently looking for quotes on each component with the best lead time that will be purchased for the design.

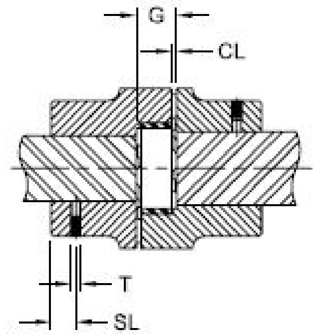
References

1. http://eng.fsu.edu/me/senior_design/2016/team04/Final%20Report%20-%20Team%204.pdf
2. http://www.skf.com/us/products/maintenance-products/alignment-tools/shaft-alignment-tools/shaft-alignment-tool-tksa31/index.html
3. http://www.lovejoy-inc.com/products/curved-jaw-couplings/gs-type.aspx
4. http://www.zero-max.com/documents/Zero-Max%20CD%20Series%20A1C%20A4.pdf
5. http://www.magtrol.com/torque/torquemeter_transducers.html
6. http://asq.org/learn-about-quality/process-analysis-tools/overview/fmea.html

Appendix A

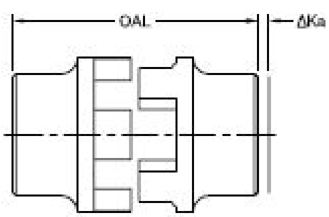
Table 5 Jaw Coupling Specification Sheet



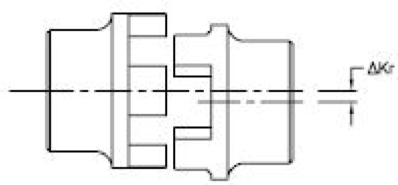


CJ Series Installation and Misalignment Capabilities

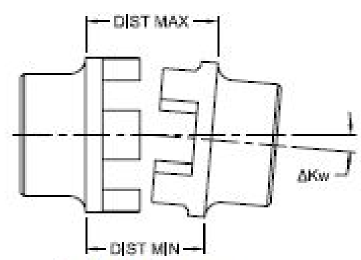
Size: / Dimensions	14	19	24	28	38	42	48	55	65	75	90	100	110	125	140	160	180
G	0.51	0.63	0.70	0.79	0.94	1.02	1.10	1.18	1.38	1.57	1.77	1.97	2.17	2.36	2.56	2.95	3.35
CL	0.06	0.08	0.08	0.1	0.12	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.35	0.41
H	0.39	0.71	1.06	1.18	1.50	1.81	2.01	2.36	2.68	3.15	3.94	4.45	5.00	5.79	6.50	7.48	8.66
S	—	1.02	1.18	1.34	1.57	1.81	1.97	2.20	2.48	2.83	3.27	3.62	4.06	4.57	5.00	5.71	6.42



Axial Displacement



Radial Displacement



Angular Displacement

CJ Series Displacement For Displacement / Misalignment (inches)

Size:	14	19	24	28	38	42	48	55	65	75	90	100	110	125	140	160	180
Max Axial Displacemnet (Ka)	0.04	0.047	0.055	0.06	0.07	0.079	0.082	0.87	0.102	0.12	0.133	0.15	0.165	0.18	0.19	0.22	0.25
Max Radial Displacement (Kr)	0.007	0.008	0.009	0.01	0.011	0.012	0.014	0.014	0.016	0.018	0.019	0.02	0.021	0.024	0.024	0.025	0.027
Kw Max angular displacement n=1500 [1/min] in deg (Kw)	1, 2	1,2	0, 9	0, 9	1, 0	1, 0	1, 1	1, 1	1, 2	1, 2	1, 2	1, 2	1, 3	1, 3	1, 2	1, 2	1, 2
Angular Displacement	0.03	0.03	0.04	0.05	0.07	0.07	0.08	0.09	0.11	0.13	0.17	0.19	0.22	0.25	0.26	0.3	0.35
Set Screw Information																	
Set Screw Size (T)	8-32	10-24	10-24	5/16-18	5/16-18	5/16-18	5/16-18	3/8-16	3/8-16	3/8-16	3/8-16	1/2-13	5/8-11	5/8-11	—	—	—
Set Screw Location (SL)	0.2	0.39	0.39	0.59	0.59	0.79	0.79	0.79	0.79	0.98	1.18	1.18	1.38	1.57	1.77	1.97	1.97

The intuitive and affordable laser shaft alignment system

SKF Shaft Alignment Tool TKSA 31

The TKSA 31 is SKF's most affordable solution for easy laser shaft alignment. The ergonomic display unit with touch screen makes the instrument very easy to use and the built-in machine library helps storing alignment reports for multiple machines. Large sized laser detectors in the measuring heads reduce the need for pre-alignments and the embedded soft foot tool helps establish the foundation for a successful alignment. Additional functions such as live view and automatic measurement support fast and effective alignment tasks and make the TKSA 31 an innovative laser shaft alignment tool that is affordable for almost every budget.

- Easy measurements can be performed by using the well-known three position measurement (9-12-3 o'clock) with additional positioning flexibility of 40° around each measurement position.
- High affordability is achieved by focussing on the standard shaft alignment process and essential functions to allow quick and effective shaft alignments.
- "Automatic measurement" enables hands-free measurements by detecting the position of the heads and only taking a measurement when the heads are in the right position.
- Automatic reports are generated after each alignment and can be customised with notes about the application. All reports can be exported as pdf files.
- Live view supports intuitive measurements and facilitates horizontal and vertical machine position corrections.
- The machine library gives an overview of all machines and alignment reports. It simplifies the machine identification and improves the alignment workflow.

Technical data

Designation	TKSA 31
Sensors and communication	29 mm *(1.1 in.)* CCD with line laser; electronic inclinometer ±0,5°; Wired communication with USB cables (included)
System Measuring distance[1]	0,07 to 2 m *(0.23 to 6.6 ft.)*
Measuring errors / displayed resolution	<0,5% ±5 µm / 10 µm (0.4 mils)
Display type	5.6" colour resistive touchscreen LCD
Software update	via USB stick
Shaft diameters	20 to 150 mm *(0.8 to 5.9 in.)* diameter Up to 300 mm *(11.8 in.)* with optional extension chains
Max. recommended coupling height[2]	105 mm *(4.2 in.)*
Mounting system	2 x V-brackets; 2 x chains 400 mm *(15.8 in.)*; 2 x threaded rods 150 mm *(5.9 in.)* per V-bracket
Alignment measurement	3 position measurement 9–12–3 3 position automatic measurement 9–12–3
Alignment correction	Live values for vertical and horizontal machine position correction; Laser soft foot tool
Alignment report	Automatic pdf report exportable via USB stick
System battery	up to 7 hours continuous use at 100% backlight (5 000 mAh rechargeable LiPo battery)
Carrying case dimensions	530 x 110 x 360 mm *(20.9 x 4.3 x 14.2 in.)*
Total weight (incl. case)	4,75 kg *(10.5 lb)*
Operating temperature	0 to 45 °C *(32 to 113 °F)*
IP rating	IP54
Calibration certificate	Supplied with 2 years validity

[1] With standard USB cables supplied, up to 4 m *(13.1 ft.)* possible
[2] Up to 195 mm *(7.7 in.)* with extension rods (optional)

Shaft alignment is recommended for almost every industry, as it helps to increase machine uptime, decrease production losses and reduce maintenance costs. The TKSA 31 is the ideal solution for entry-level laser shaft alignment. Its intuitive usage helps to achieve accurate alignments while the TKSA 31 remains affordable for almost every budget.

PUB MP/P8 15549 EN · February 2016

Features

- Integrated Torque and Speed Conditioning
- Torque Range: 0.1 N·m to 10,000 N·m (0.07 lb·ft to 7375 lb·ft)
- Accuracy: < 0.1%
- Overload Capacity: 200%
- Overload Limit: up to 400%
- High Speed Applications: up to 50,000 rpm
- Non-Contact (no slip rings)
- No Electronic Components in Rotation
- No Electrical Noise
- Single DC Power Supply: 20 to 32 VDC
- Immediate Speed Detection
- Adjustable Torque Signal Frequency Limitation
- Built-In Test Function
- Stainless Steel Shaft
- EMC Susceptibility Conforms to European Standards
- **Calibrated to METAS standards**

Specifications

Sales Drawings

For complete technical specifications and detailed dimension drawings, download the applicable data sheet (links at bottom of page).

Click here Conversion Calculators

Series Model Number	Nominal Rated Torque		TMB Series		TM Series		TMHS Series		Shaft Ends		View Drawing			
	N·m	lb·ft	Accuracy Class	Maximum Speed (rpm)*	Accuracy Class	Maximum Speed (rpm)*	Accuracy Class	Maximum Speed (rpm)*	Smooth	Splined	PDF TM	PDF TMB	STEP** TM	STEP** TMB
301	0.1	0.07	N/A		0.2%	20,000	N/A		X					
302	0.2	0.15	N/A		< 0.1%	20,000	N/A		X					
303	0.5	0.37	≤ 0.1%	6000	< 0.1%	20,000	< 0.1%	40,000	X					
304	1	0.7	≤ 0.1%	6000	< 0.1%	20,000	< 0.1%	50,000	X					
305	2	1.5	≤ 0.1%	6000	< 0.1%	20,000	< 0.1%	50,000	X					
306	5	3.7	≤ 0.1%	6000	< 0.1%	20,000	< 0.1%	50,000	X					
307	10	7.4	≤ 0.1%	6000	< 0.1%	20,000	< 0.1%	50,000	X					
308	20	15	≤ 0.1%	6000	< 0.1%	20,000	< 0.1%	50,000	X					
309	20	15	≤ 0.1%	4000	< 0.1%	10,000	< 0.1%	32,000	X					
310	50	37	≤ 0.1%	4000	< 0.1%	10,000	< 0.1%	32,000	X					
311	100	74	≤ 0.1%	4000	< 0.1%	10,000	< 0.1%	32,000	X					
312	200	148	≤ 0.1%	4000	< 0.1%	10,000	< 0.1%	24,000	X	X				

													3D	3D
313	500	369	≤ 0.1%	4000	< 0.1%	10,000	< 0.1%	24,000	X	X			3D	3D
314	1000	738	TMB Not Available		< 0.1%	7000	< 0.1%	16,000	X	X			3D	
315	2000	1475	TMB Not Available		< 0.1%	7000	< 0.1%	16,000	X	X			3D	
316	5000	3687	TMB Not Available		< 0.1%	5000	< 0.1%	12,000		X			3D	
317	10,000	7375	TMB Not Available		< 0.15%	5000	< 0.15%	12,000		X			3D	

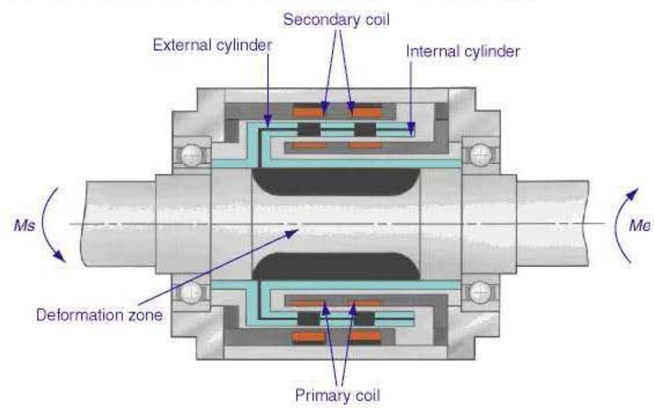
* For low speed testing (e.g. gear motors), consult Magtrol.
**Other CAD models available upon request.

NOTE: Dimensions are the same for every series (TM, TMHS, TMB)

Back to top

Operating Principles

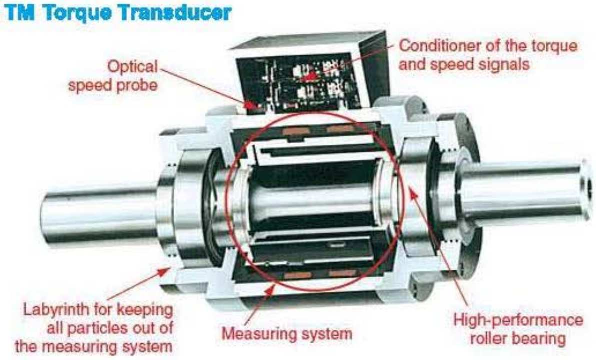
Simple and reliable, the torque meter measuring system is based on the principle of a variable, torque-proportional transformer coupling. The principle has been adapted by Magtrol for the measurement of torque. The measuring system consists of two concentric cylinders, shrunk on the shaft on each side of the shaft's deformation zone, and two concentric coils attached to the housing.



Both cylinders have a circularly disposed coinciding row of slots and rotate with the shaft inside the coils. A constant alternating current with the frequency of 20 kHz flows through the primary coil. When torque is applied, the slots on the two cylinders do not overlap. Instead, the deformation zone undergoes an angular deformation and the slots begin to overlap. Thus, a torque-proportional EMF is induced in the secondary coil.

The conditioning electronics convert the EMF into a voltage between +10 and -10 V, depending on the direction of the torque. Speed measurement is integrated by means of an inductive proximity transducer trained on a toothed path cut directly into the outer cylinder.

Inside View of TM Torque Transducer



Back to top

Applications

Magtrol torque meters provide dynamic torque and speed measurement of:

- **Propellers - aerospace, marine and helicopter**
- **Reduction gears and gearboxes**
- **Clutches**

- Windshield wipers, electrical windows, starters, generators and brakes in automobile industry
- Pumps - water and oil
- Motorized valves
- Drills, pneumatic tools and other machine tools
- Tachometers

Torque Limiter Testing on Drills

A Model TM 311 Torque Transducer is used in the design along with a disc brake, coupling, swivel and universal joint for accuracy and reliability.

BRP 125 Test Bench specially designed for testing torque limiter on a wide range of drills from 5 Nm to 200 Nm.

A large manufacturer of hand drills was in need of a test bench for testing the torque limiters of their drills. The aim of the test bench was to equip their repair shops with a versatile, easy and fast system capable of testing a wide range of repaired drills. The ideal test bench would be a stable setup with a maximum surface area of 800 × 800 mm, capable of measuring speed of rotation and maximum trip torque for a 5-200 Nm range of drills. The test cycle would include machine assembly , rotation at nominal speed and braking the machine until the torque limiter trips within a total test time of 3 minutes maximum per drill.

Magtrol SA was able to meet these expectations with the development of the BRP 125 Test Bench, using their TM 311 Torque Transducer along with a disc brake, coupling, swivel and universal joint. The drill is placed on the test bench and connected to the system with a quick male/female coupling, held by hand by the operator. A stop bar prevents the machine from turning on. The torque is transmitted by means of a Cardan (universal joint) and a swivel joint. The role of transmission by Cardan and swivel joint is to allow the test without having to align and fix the machine. Once the maximum speed is reached, the drill is braked by means of the disc brake controlled by an electropneumatic control.

The system is controlled by software that allows the user to save the test parameters and print out a test certificate for each drill tested. The certificate contains information on the type of drill, serial number , nominal speed, nominal torque, name of tester , date of the test and the test values, which are also output in a torque/speed diagram.

Since the TM 311 Torque Transducer is particularly suited to measurements of great accuracy in fairly large ranges, the BRP 125 Test Bench has been a success with great reliability , good mechanical stability and an excellent reproducibility (0.5%). The users are very satisfied with the performance and functionality of the bench.

Back to top

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Magtrol, Inc (USA): 716-668-5555 • magtrol@magtrol.com ~ Magtrol SA (Switzerland): 41 (0)26 407 3000 • magtrol@magtrol.ch


ZERO-MAX
CD® Couplings
SERIES A1C
6_26-A1C Tighten to 32 Nm
6_30-A1C torque clamp bolt to 58 Nm

ZERO-MAX CD® COUPLINGS SERIES A1C

- For today's most demanding servo motor and motion control applications. CD Couplings Series A1C are precise, robust, and available in sizes and models for every application.
- High torsional stiffness and high dynamic load capacity ensure reliable machine operation.
- Precise positioning under high speed reversing loads without fatigue for reliable 24/7 operation.
- Unique patented composite disc design provides misalignment capacity and long operational life.
- Clamp style hub design provides a superior method of shaft engagement.
- Eco-Friendly, adapted to RoHS Directive with no banned substances.

These next-generation CD Couplings Series A1C allow you to transmit high horsepower in a small envelope. They are ideal for cyclic applications where speed and repeatable accuracy are critical to keep 24/7 systems going.

CD Couplings Series A1C withstand the punishment and stress of a servo motor. In comparison, other couplings may have high torsional stiffness specifications; however, they can be too brittle to withstand the punishment of high speed reversing applications.

The working part of a CD Coupling Series A1C is made of high precision composite material. This patented design has high torsional stiffness, and yet allows for misalignment in high stress applications. CD Couplings Series A1C have excellent chemical and moisture resistance and operate without maintenance in hostile environments.

Standard and Custom CD Couplings Series A1C are available for every application. Do you need higher misalignment and greater torque capacity in your coupling? Need more flexibility and torsional stiffness? Need a very large bore diameter coupling? Or a long spacer coupling? Zero-Max CD Couplings Series A1C are available in a full range of styles, models and sizes to meet those needs. Zero-Max will design and build a custom CD Coupling Series A1C to handle your unique application.

CD® COUPLINGS SERIES A1C FOR THE MOST DIFFICULT MOTION APPLICATIONS

- Ideal for high precision applications including packaging machines, pick and place systems, printing machinery, machine tools and most systems using servo motors.
- Operating temperature range is -70° to +250° F (-57° to +121° C).
- Composite discs are resistant to many chemicals.
- Maintenance free.
- Hubs are machined to a high level of concentricity for smooth and quiet operation.
- RoHS compliant – manufactured of RoHS compliant materials and contains no banned substances.

CD® COUPLINGS **SINGLE FLEX** ALUMINUM

CD Coupling Series A1C has very low weight and inertia, making it an excellent choice for servo motor applications. The unique design delivers two features that are not often found in a precision coupling. High torsional stiffness and high durability!

The compact size, low inertia, and clamping system enable this coupling to fit into many applications.

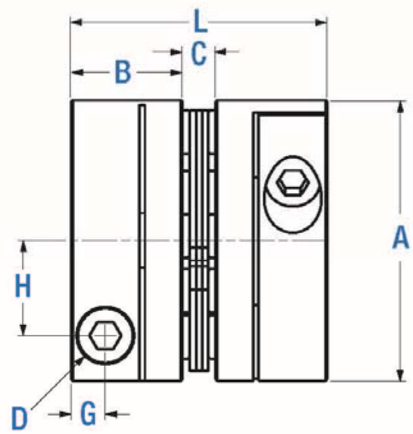
- Zero Backlash
- Torsionally Stiff
- Excellent for Reversing Loads
- Smooth Operation at High Speeds
- Compact

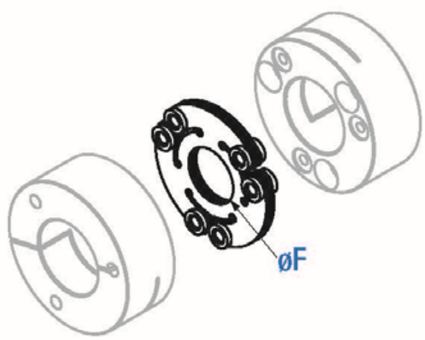
Performance Information

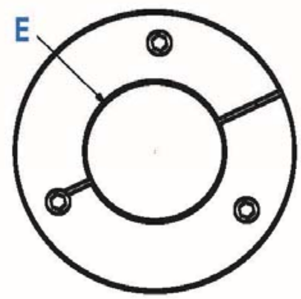
Model	Continuous Torque	Peak Torque	Torsional Stiffness	Maximum Speed	Maximum Misalignments			Weight		Inertia	
					Angular	Parallel	Axial	Max Bore	Min Bore	Max Bore	Min Bore
	Nm	Nm	Nm/Rad	RPM	Degrees	mm	mm	kg	kg	10^{-3}kg-m^2	10^{-3}kg-m^2
6A18-A1C	20	40	11,650	15,000	2	0.1	0.8	0.2	0.26	0.088	0.095
6A22-A1C	30	60	17,352	13,500	2	0.15	0.9	0.33	0.41	0.19	0.21
6A26-A1C	53	106	20,100	11,500	2	0.2	1.1	0.46	0.6	0.35	0.37
6A30-A1C	90	180	42,976	9,500	2	0.25	1.3	0.76	0.94	0.78	0.82
6A37-A1C	181	362	67,167	8,000	2	0.33	1.8	1.59	2.04	2.53	2.71
6A45-A1C	282	564	123,909	6,700	2	0.38	2.3	3	3.9	7.16	7.71

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

CD® COUPLINGS **SINGLE FLEX** ALUMINUM







Note: Typical keyway placement

Dimensional Information											
Model	A	B	C	D		E (bore)		F	G	H	L
				Bolt	Torque	Min	Max				
	mm	mm	mm	M	Nm	mm	mm	mm	mm	mm	mm
6A18-A1C	53	22.5	5.49	M6	13	9	27	20.1	7.25	18	50.5
6A22-A1C	62	26	5.74	M6	13	16	31	24.9	7.24	22	57.7
6A26-A1C	69.5	29.5	6.25	M8	32	14	36	25.4	9.14	24	65.2
6A30-A1C	82	32.5	9.65	M10	58	16	40	30.71	10	27.8	74.7
6A37-A1C	101	46	11.23	M12	100	18	52	38.4	12.7	36	103.2
6A45-A1C	123	60	12.75	M16	245	24	65	46	16.95	43.5	132.8

CD® COUPLINGS **DOUBLE FLEX** ALUMINUM

CD Coupling Series A1C has very low weight and inertia, making it an excellent choice for servo motor applications. The unique design delivers two features that are not often found in a precision coupling. High torsional stiffness and high durability!

The compact size, low inertia, and clamping system enable this coupling to fit into many applications.

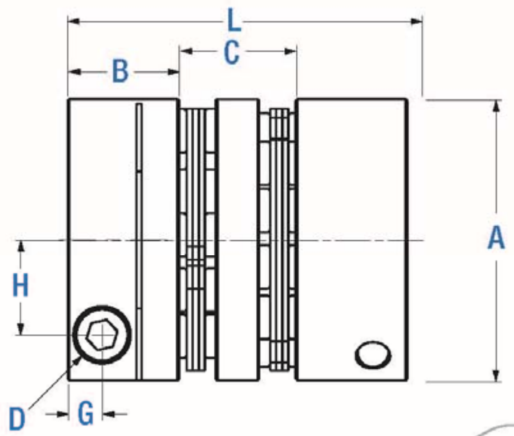
- Zero Backlash
- Torsionally Stiff
- Excellent for Reversing Loads
- Smooth Operation at High Speeds
- Compact

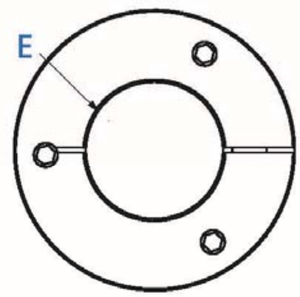
Performance Information

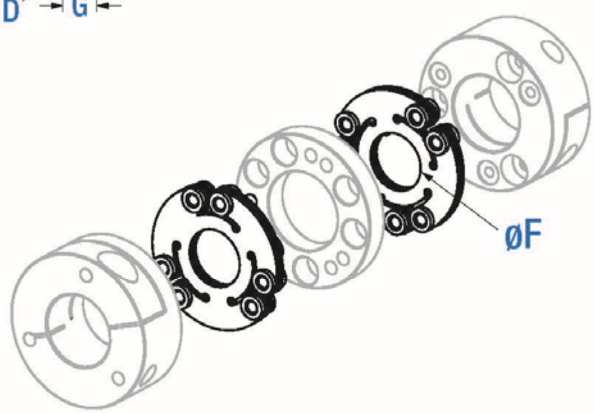
Model	Continuous Torque	Peak Torque	Torsional Stiffness	Maximum Speed	Maximum Misalignments			Weight		Inertia	
					Angular	Parallel	Axial	Max Bore	Min Bore	Max Bore	Min Bore
	Nm	Nm	Nm/Rad	RPM	Degrees	mm	mm	kg	kg	10^{-3}kg-m^2	10^{-3}kg-m^2
6P18-A1C	20	40	5,500	15,000	2	0.44	1.6	0.25	0.30	0.30	0.11
6P22-A1C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

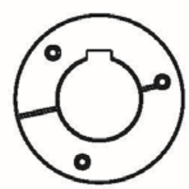
- Consult factory for speeds higher than those listed and balancing requirements, if necessary.
- Consult factory for higher torque and higher torsional stiffness couplings.
- Available with or without keyway on clamp style hubs.

CD® COUPLINGS **DOUBLE FLEX** ALUMINUM









Note: Typical keyway placement

Dimensional Information

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

SELECTING A *CD*® COUPLING

Feed Screw Systems

1. Oscillation phenomena of servomotors

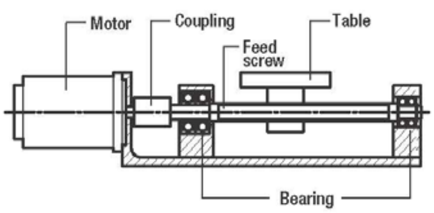
If the resonant frequency of the entire feed-screw system is under 400~500Hz, oscillation may occur depending on the gain adjustment of the servomotor. The problems can be avoided by raising the resonant frequency of the mechanical system or adjusting the tuning function (filter function) of the servomotor.

Contact us for unclear points concerning oscillation phenomena of servomotors.

How to evaluate the resonant frequency of feed-screw system

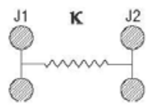
1. Select the coupling according to the normal operating torque and maximum torque of the servomotor/stepping motor.

2. In the following feed-screw system, evaluate the entire resonant frequency: Nf from the torsional spring constant: *K* of the coupling and feed screw, the moment of inertia: J1 of the driving side and the moment of inertia: J2 of the driven side.



$$Nf = \frac{1}{2\pi}\sqrt{K\left(\frac{1}{J1}+\frac{1}{J2}\right)}$$

Nf: Eigenfrequency of the entire feed-screw system [Hz]
K: Torsional spring constant of the coupling and feed screw [N • m/rad]
J1: Moment of inertia of the driving side
J2: Moment of inertia of the driven side



Selection Procedure

1. Calculate torque Ta applied to the coupling based on the motor output P and coupling operating rotation speed n.

$$Ta[N \cdot m] = 9550 \times \frac{P\ [kW]}{n\ [min^{-1}]}$$

2. Calculate corrected torque Td applied to the coupling after deciding the service factor K based on load conditions.
Td = Ta x K

In servomotor drive, multiply the service factor K=1.2~1.5 by the maximum torque of servomotor Ts.
Td = Ts × (1.2~1.5)

3. Select a coupling size with permissible torque Tn that becomes greater than the corrected torque Td.
$Tn \geq Td$

4. Depending on the bore diameters, the coupling permissible torque may be limited. Refer to the "Specification" and "Standard bore diameter".

5. Confirm if the required shaft diameter does not exceed the maximum bore diameter of the selected size.

Custom Designs Available Upon Request

If our standard line of couplings will not exactly fit your system needs, contact us for a custom design.

- Custom bores
- Ultra high speeds
- Special finishes
- Special Lengths
- Designed for operation in special environments

CD® HOW TO ORDER

Part Numbering Structure

TYPE | SIZE — A1C — Bore | TYPE — Bore | TYPE

TYPE

Code	Description
6A	Single Flex
6P	Double Flex

SIZE

Code	Description
18	Size 18
22	Size 22
26	Size 26
30	Size 30
37	Size 37
45	Size 45

Bore: See Chart

TYPE

Code	Description
B	without Keyway
K	with Keyway

Bore: See Chart

TYPE

Code	Description
B	without Keyway
K	with Keyway

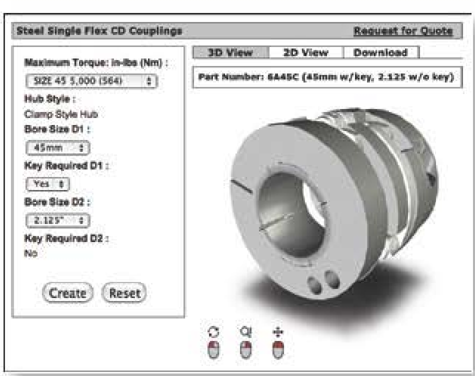
Note: The hub design of Series A1C CD Couplings will provide the necessary clamping force to hold the shaft in a dynamic application without the use of keyways.

Example:
6A30-A1C-20B-28B
- Single Flex
- Size 30
- 20mm bore without keyway x 28mm bore without keyway

Bore Size

Model	Bore (mm)	9	10	11	12	13	14	15	16	17	18	19	20	22	24	25	28	30	32	35	38	40	42	45	48	50	52	55	58	60	62	63	65
6A18-A1C 6P18-A1C		●	●	●	●	●	●	●	●	●	●	●	●	●	●	●																	
6A22-A1C 6P22-A1C									●	●	●	●	●	●	●	●	●	●															
6A26-A1C 6P26-A1C							●	●	●	●	●	●	●	●	●	●	●	●	●	●													
6A30-A1C 6P30-A1C									●	●	●	●	●	●	●	●	●	●	●	●	●	●											
6A37-A1C 6P37-A1C											●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●							
6A45-A1C 6P45-A1C															●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●

●: The coupling will transmit full peak torque on a shaft without a keyway. Please contact the factory for additional bores



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ZERO-MAX
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Phone: +45 8681 2288 Fax +45 8681 5388
www.zero-max.dk Email ext@zero-max.dk