VTT Rotor Back EMF Test Fixture

Mid Term Report

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Abstract

Turbocor requires a test fixture to be developed that can measure the back EMF generated when a third party manufactured rotor is rotated within a stator as a method of quality check. The test fixture must center and align the rotor within the stator, and provide means to overcome the 60-80 pound magnetic force that is exerted when the rotor is inserted into the stator. Several design decisions were made in order to optimize the efficiency of the design. The decision was made to move both the rotor and the stator to minimize the spatial footprint of the design. A live center will be utilized to keep to rotor centered within the stator. By using a weighted decision matrix, it was decided to implement a ball screw in order to overcome the magnetic force on the stator when it is moved over the magnetic portion of the rotor. The next step in the design process is to select a motor and a ball screw for the design, which will allow for other components to be selected. The Final Design Review will be held at Turbocor on November 20th, 2014.

1 Introduction

The Danfoss Group is a global leader with a wide range of products utilized in areas such as cooling food, air conditioning, heating buildings, and electric motors. Danfoss Turbocor is a wholly owned subsidiary of The Danfoss Group and is one of the pioneers of the oil-free centrifugal compressors. Turbocor blokiujhgfvcxvhjkuhgfcx[-p0dz[0[p;hvcxolhn egan as a R&D startup in Australia in 1993, and in 2004, Danfoss and Turbocor formed a 50/50 joint venture called Danfoss Turbocor.[1] They offer the world's first completely oil-free compressors designed for the Heating, Ventilation, and Air-Conditioning (HVAC) industry. The use of magnetic bearings sets Danfoss Turbocor apart from the other compressor manufacturers. This allows for oil-free operation while significantly reducing sound generation[2] Danfoss Turbocor has their headquarters and manufacturing facility in Tallahassee, FL, and does business around the world.

Danfoss Turbocor plans to launch a new compressor model before the end of 2014. Current production plans call for the use of a rotor that will be manufactured by a third party company. There needs to be a way to quality check these rotors to ensure they are up to Turbocor standards prior to installing them in the compressor. To test these rotors, Danfoss Turbocor must measure the back electromagnetic force delivered by the electric motor when the rotor is being rotated inside of the stator. Electromotive force, or EMF, typically refers to voltage generated when a motor is spun. Measuring this voltage can be used as a method to determine the rotational speed of the motor, which is called back EMF. The reason it is referred to as a back EMF force is because the voltage pushes against the current that induces it.[3] By measuring this back EMF force, Danfoss Turbocor will be able to verify the quality of the rotors being supplied by a third party manufacturer. Eventually, Turbocor plans to manufacture these rotors in-house, but until they switch over to manufacturing these themselves, they require this method of quality assurance.

To successfully and efficiently implement this testing procedure, a test fixture must be created that can be integrated into the manufacturing line. The equipment will be used to perform the back EMF measurements on each rotor prior to its assembly into the compressor. A previous test fixture has been developed by Turbocor for use on one of their smaller compressor models. The test fixture for this application will be similar; however, there are additional constraints that make the implementation more difficult. One of the biggest challenges is to determine a method of centering the rotor within the stator. This is essential because if the rotor is slightly off center, it cannot be tested properly. Additionally, there is a large magnetic force induced when the rotor is pushed into the stator. This is not of concern in the smaller compressor models as the small force can easily be overcome by a human; however, in the new larger compressor model this force is significant and it is not safe to manually load the rotor. Due to the magnetic nature of the components used in the assembly of the compressors, magnetic material should not be used in areas within the magnetic field of the rotor.

2 Project Definition

2.1 Background research

Turbocor has already created a test fixture for their smaller compressor, which will serve as guide for the new design to test a larger rotor. However, the current fixture cannot be modified to test the new rotor due to an increase in size, electromagnetic force and a need for a more reliable unit as discussed previously. The overall setup of this previously developed test fixture does give this senior design group an opportunity to view the essential features of the test fixture. A picture of the previously utilized back EMF test fixture can be seen below in Figure 1.

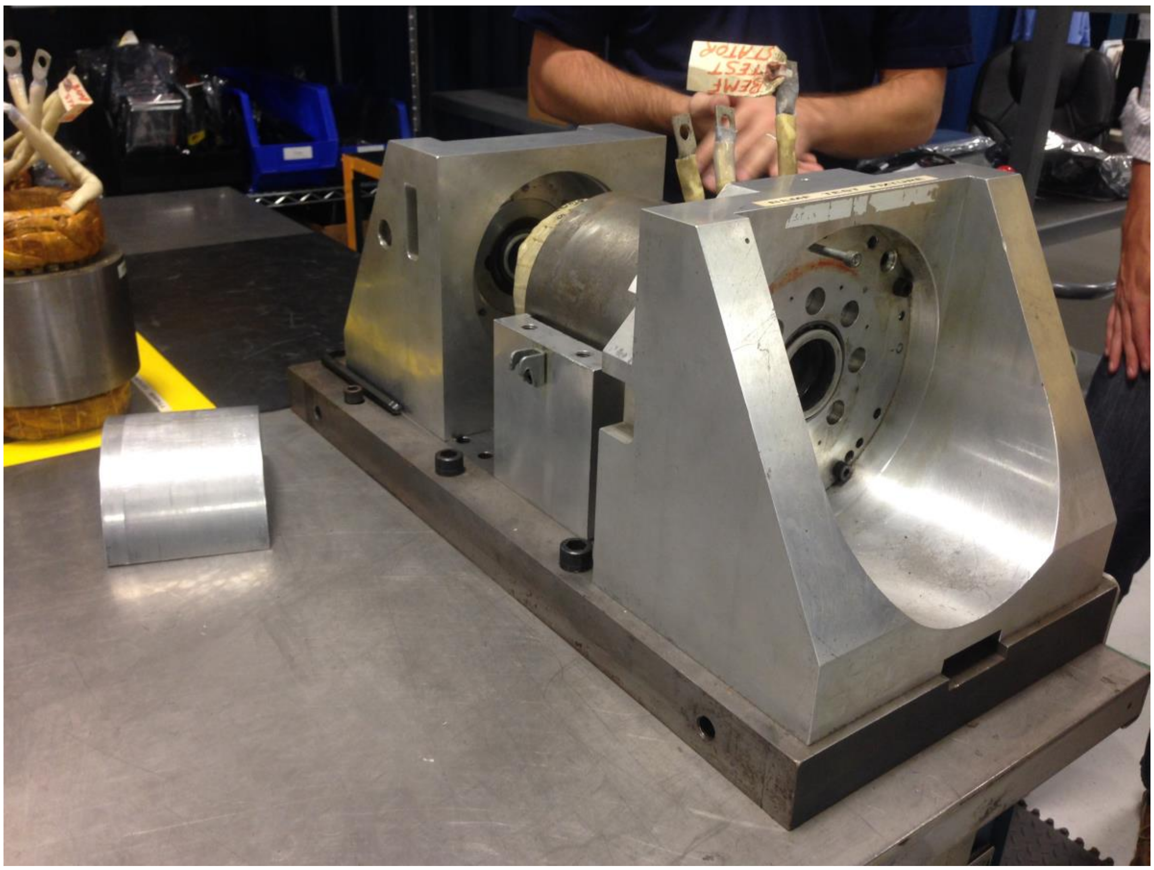

Figure 1: Previously designed Back EMF Test Fixture for smaller compressor model

In the test fixture for the smaller compressor model, there is a locking feature that locks the stator into place and can be unlocked, should the stator need to be replaced. This is an essential feature of the new design. The old design utilizes a bearing to ensure the centering of the rotor within the stator. This is an effective way to ensure that the rotor is centered; however, there is a high cost associated with the replacement of bearings over the life cycle of the test fixture, and thus an alternative method of centering should be developed. One key feature of the larger rotor is a key-like-hole centered on the end of the rotor. An isometric view of the rotor can be seen below in Figure 2, and this key like hole can be seen on the top left of this figure.

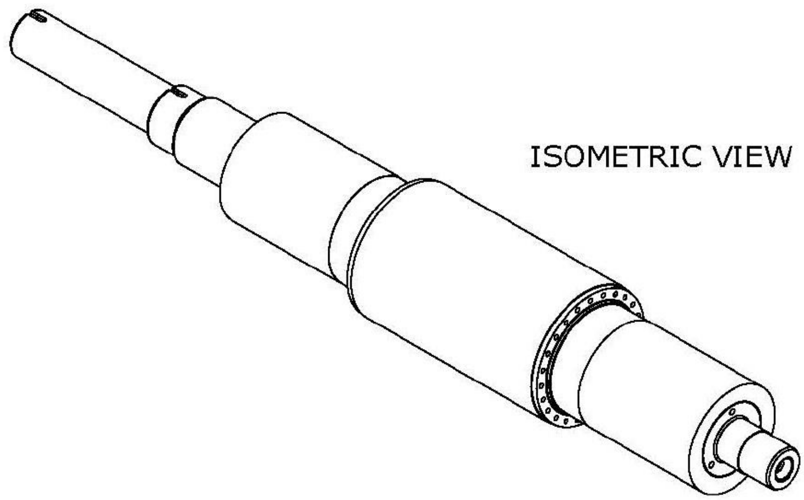

Figure 2: Isometric view of rotor showing key like hole for live center

The most effective way to center the rotor within the stator may be through the use of a live center. A live center, or lathe center is a tool that has a conical shape that is typically used in lathe work in order to provide a stable axis that can be easily replaced, while also providing an accurate method of centering. A live center typically consists of a sixty degree conical shape on one end that will align with an opening on the work piece that is shaped to accept the conical end at the given angle. The advantages of using a live center include the enabling high speed rotation while handling heavy loads, centering the work piece accurately from work area to work area, and feasibility of replacement.[4] The shape of the point will also have to be determined based on the work piece being used; the rotor that will be provided has a point angle that will accept a sixty degree conical shape.

Figure 3: Picture of an example live center

There are several design considerations that will need to be determined when designing this test fixture. The method for rotating the rotor will need to be decided upon. Several different methods will be evaluated in the design process to determine which method is the most suitable. As previously mentioned, due to the large size of the rotor, the magnetic force due to the magnetic field of the stator is significantly large. A design feature must be implemented that will assist the operator in manually loading the rotor into the stator. Because magnetic material cannot be used in the design, aluminum will be the most effective material for the test fixture housing as it is non-magnetic and low cost. Aluminum 8020 has been recommended to the senior design group as it is readily available in the Turbocor facility.

2.2 Need Statement

Danfoss Turbocor plans to launch a new compressor model before the end of 2014. Current production plans call for the use of a rotor that will be manufactured by a third party company. Danfoss Turbocor has a need for an ergonomic, efficient, and reliable device to test the back EMF of these third party manufactured rotors in a production facility. This is required because Turbocor uses a third party company to manufacture their rotors, and there is no other way to ensure the rotors are up to Turbocor's standards prior to implementation. The design of a back EMF test fixture will allow Danfoss Turbocor to properly evaluate the quality of the rotors they are receiving from the third party manufacturer and ensure their compressors will meet performance requirements when implemented.

"Turbocor does not have a method to verify the quality of third party manufactured rotors for their new compressor model"

2.3 Goal Statement & Objectives

"**The goal of this project is to have a fully designed, manufactured, and tested back EMF test fixture that meets Danfoss Turbocor's requirements prior to the conclusion of the senior design class in the spring**."

Objectives:

- This test fixture will be able to be implemented on Danfoss Turbocor's production line with a design life of seven years minimum.
- The submission package to Danfoss Turbocor will include a 3D prototype of the final design with a drawing package for each individual part sufficient for a re-creation of the test fixture should that be desired by Danfoss Turbocor.
- The final product will conform to all size and weight requirements outlined by Danfoss Turbocor (see constraints section)
- The back EMF test fixture will have a feature that centers the rotor within the stator to a specified tolerance.
- The back EMF test fixture will have a feature that aids in the manual insertion of the rotor into stator and provides a sufficient force to overcome the magnetic force generated.
- All other performance requirements outlined by Danfoss Turbocor will be met or exceeded.

2.4 Constraints

Although the previous existing version may serve as a template, there are several improvements that must be made in order for it to be effective in this application. The design must be strengthened and enlarged to support the weight and size of the new rotor and stator. Because of the magnetic nature of the stator and the rotor, magnetic materials may not be employed within 200mm surrounding the carbon fiber sleeve as they can affect the output given by the oscilloscope that will be used to measure the back EMF of the stator. Because this will be implemented in a manufacturing setting, it is important to optimize the spatial footprint of the test fixture so that easy movement between work cells will be possible. The size limitations imposed on this design are summarized in the design specifications section below.

There must be a method for measuring the voltage and waveform of the back EMF, and this is discussed in performance specifications section. Finally, but perhaps most critically, all Occupational Safety and Health Administration (OSHA) safety standards must be met. Specifically, OSHA 29 CFR 1920 must be met at all times and noise levels must be maintained less than 80 dB.[5] Dangerous areas of the test fixture must be clearly labeled with internationally recognized symbols, which may include pinch or shock points. Due to the magnetic nature of the compressors manufactured at Turbocor, stored energy may present an issue. For this application, the stored magnetic energy should not be significant; however, if the fixture is powered off, it may remain rotating at a high angular velocity for some finite amount of time prior to stopping. Because this may present a safety hazard, proper warning labels shall be attached to the test fixture.

2.4.1 Design Specifications

There are several design specifications that have been imposed on the back EMF test fixture and most of these are related to the size and weight of the design. The design must include a worktable, and spatial limitations of the design have been provided, which can be seen below in Figure 4. There should also be sufficient space for an oscilloscope on the workstation and a control panel that should be integrated into the design.

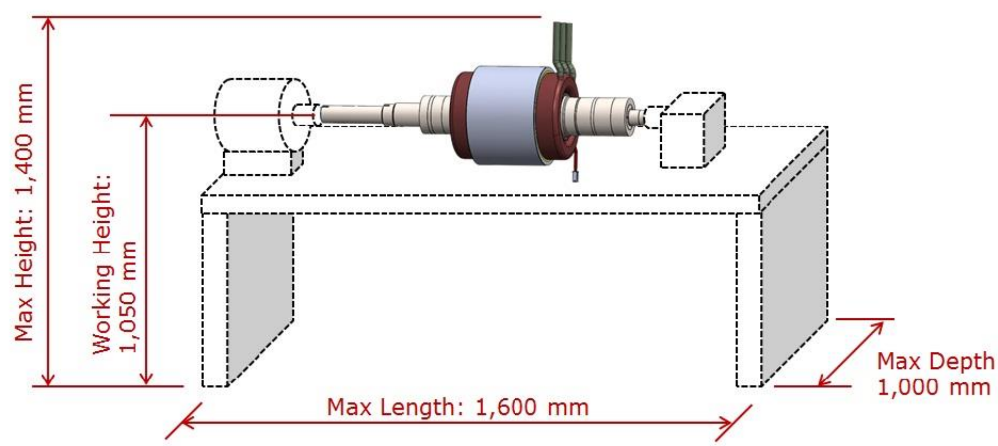

Figure 4: Spatial footprint requirements of test fixture

The spatial limitations are also summarized in Table 1, which includes the need for the center of the fixture to be oriented at a working height of 1,005 mm for ergonomic purposes.[5] There is no maximum weight of the test fixture outlined by Turbocor; however, it has been discussed that for both cost and ease of use purposes, the overall weight of the design shall be as low as reasonably possible.

Table 1: Summary of design constraints

Design Constraint	Requirement	Units
Max Bench Height	1,400	Length (mm)
Max Bench Length	1,600	Length (mm)
Max Bench Depth	1,000	Length (mm)
Bench Working Height	1,050	Length (mm)

2.4.2 Performance Specifications

The performance specifications have been briefly mentioned in previous sections, and will be addressed in detail here. The method of centering utilized in the design must ensure that while the motor is rotating the rotor, the rotor is centered within the stator within a tolerance of 0.1 mm. If the rotor is offset, two issues arise: the magnetic force exerted on the rotor by the stator will no longer be equal on all sides, and the back EMF voltage and wave form outputted by the stator will not be accurate. In addition to the centering mechanism, the test fixture must be capable of overcoming the magnetic force that is exerted by the magnetic field when the rotor is inserted into the stator. This is an estimated 60-80 pound force according to Turbocor. This must be addressed by using a design feature that will allow for the operator to manually insert the rotor into the stator in an efficient manner. The design life of the test fixture must exceed seven years.

Finally, there are performance requirements of the motor's performance. A 120V or 208V motor is preferred, and the motor must be capable of rotating the rotor within the stator at a constant angular velocity that exceeds 1,000 RPM.[5] It is important that the angular velocity is consistent between tests so each test is repeatable and can be compared to a standard value. The output given by the test fixture shall include the back EMF voltage, and the waveform for three different phases. This allows for verification of the angular velocity of the rotor, and this output provides the basis for rejection or acceptance of the rotors.

3 Design and Analysis:

3.1 Functional Analysis

There are several components, mostly mechanical, that will make up the test fixture. Some of these components are dependent on the final design chosen; however, there are some that are essential to the design regardless of the final configuration. These are listed below and will be addressed in this section:

- Motor
- Motor Drive
- Motor Connection/Vibration Reducer
- Method of Overcoming Magnetic Force
- Live Center Support (including Live Center)
- Live Center Track (including locking mechanism)
- Stator Housing/Linear Bearings
- Baseplate
- Table

The motor needs to be capable of rotating the rotor at a minimum angular velocity of 1,000 RPM. In order to angularly accelerate the rotor to this angular velocity, the motor needs to overcome the moment of inertia of the rotor and magnetic resistance due to the magnetic field of the stator. A minimum torque of 8.08 ft-lb and a power of 0.58 HP will be required in order to accelerate to and maintain this angular velocity. Calculations for this can be found in Appendix B. Additionally, because the rotor will be rotating at such a high RPM, analysis must be performed to determine the requirements for the internal bearings of the motor, which still needs to be performed. If it is determined that the loads experienced by the internal bearings of the motor are too large, an option to utilize roller bearings to support to weight of the rotor may be considered. It should be noted that a motor was initially selected; however, after further discussion with the sponsor and Senior Design professors, it was decided that more research would have to be performed before selecting a final motor. Due to the motor being the center of the test fixture, it is essential that the proper motor is selected.

It is important that when the motor is turned on, it is not instantly given full power as this will lead to an extremely high angular acceleration value. This will then lead to motor damage, and thus, a shorter lifetime for the motor. In order to offset this, a motor drive must be selected. This allows for controlled acceleration of the rotor, which will allow for a longer motor life. As the motor will be rotating at such a high angular velocity, vibration may be an issue. In order to mitigate this, a three pierce connector between the motor and the rotor shall be used. A vibration reducer, made of either nylon or hard rubber, will be connected to the shaft of the motor. On the rotor, there are keyways which can be utilized for connecting the motor. Between these two pieces there will be a connector. As with the motor shaft, stress analysis shall be performed to ensure that the motor connector pieces do not fail. The motor connection cannot be chosen until after the motor has been selected as the dimensions are dependent on the shaft of the motor. If it is deemed necessary, vibration analysis shall be performed on the motor to ensure that vibration will not be a significant issue during implementation of the test fixture.

As mentioned in the previous sections, when the rotor is inserted into the stator, there is a 60-80 pound force resisting it due to the opposing magnetic fields of the rotor and the stator. The design will need a method for overcoming this force, and allowing for the rotor to be properly centered within the stator. Additionally, whatever method is utilized may require an additional feature to prevent back drive so that the stator remains in place. Originally, it was thought that either the rotor would be inserted all the way into the stator, or that the stator would move to completely encompass the rotor while the other component remained stationary. This seemed to offer the simplest solution; however, it was determined that this would not be possible due to the spatial constraints that have been imposed on the test fixture. Due to these spatial constraints, both the rotor and the stator will need to move. All of the design concepts work on the same basic principle: the rotor shall be lowered into the test fixture, the non-magnetic portion will then slide into the stator, and finally, the rotor gets locked into place and the stator is slid over the rotor. This process will be discussed more in depth in later sections when the initial prototype is presented. It is important to note that the magnetic force will only need to be overcome when the stator is moving over the magnetic portion of the rotor, which is located in the center of the rotor.

Another key component of the design is the live center. This was introduced previously, and plays an essential role to allow for the rotor to spin at high angular velocities while maintaining linear and axial alignment within the stator. A live center was chosen because it will have very little internal friction and should have no problem supporting the weight of the rotor. The live center must be centered within a support system that will house it. The dimensions and tolerances of this support must be properly chosen so that the center of the live center axis lines up with the center of the stator and the axis of the motor shaft. The live center will be connected to the rotor after the rotor has been connected to the motor, but before the stator is moved over the magnetic portion of the rotor. This is significant because the live center support will not need to overcome the magnetic force as it is moved to connect with the rotor. The deviation of the live center from the central axis cannot be more than 0.5 mm so it is essential that all components that affect this alignment are machined to a specified tolerance to be determined when the design is finalized. The live center must run along a track that will keep it aligned. This also must include a locking feature to keep the live center in place once it is in the proper position.

The stator will be housed in a support system that will be referred to as the stator housing. The configuration of this design component shall depend on the method used to overcome the magnetic force. The role of the stator housing will be to ensure the stator stays in place during the testing of the rotors, and to be properly configured with the device used to overcome the magnetic force. Most likely, the stator housing will move along linear bearings. The purpose of the linear bearings shall be to reduce the friction that would otherwise be present due to the large weight of the stator. Additionally, depending on the method used to overcome the magnetic force during insertion, a moment may be imposed on the stator housing if this overcoming force is applied away from the center of the stator housing. The linear guides will act to negate this moment.

Once all components are selected, the base of the fixture must be made. This will most likely include the track that the live center runs along. The base will need to have connection points for the linear bearings, and the mounting for the motor. The base plate will serve as the reference point so the heights of the center of the motor shaft axis, the middle of the stator, and the axis of the live center are all equal. The last design feature will be the table the test fixture sits on. This shall be designed last and the height will be chosen so the center axis is at an ergonomic working height of 1,050 mm as outlined by Turbocor.

3.2 Design Concepts

There were several design choices that had to be made in order to determine the initial prototype. The three major decisions that had to be made were:

1. Rotor Centering: Bearing vs. Live Center
2. Rotor Connection (non-motor side): Hinged vs. Sliding
3. Overcoming Magnetic Force: Ball Screw vs. Rack and Pinion vs. Pneumatic Device

The first decision involved the main method for centering the rotor. One side of the rotor is constrained by the connection to the motor, and the other side of the rotor needs to be fixed by a different means to ensure proper alignment. The previous back EMF test fixture for the smaller rotor utilized ball bearings to achieve this. Using a ball bearing to center the rotor in this design was also considered, and the other option was to use a live center. The rotor is machined using a live center, which has been introduced in previous sections. Due to this, there is a feature on the end of the rotor that mates with a live center. These two ideas were both considered; however, it was not a difficult decision to use the live center as opposed to a ball bearing. Turbocor explicitly stated they had a preference against using a ball bearing to constrain the rotor. This was due to concerns with the bearings holding up over time. During testing the rotor will be spinning at over 1,000 RPM and with several tests performed daily ball bearings were not the preferred choice. Additionally, it is more difficult to center the rotor within a ball bearing, and because the rotor is pre-machined to mate with the live center this makes the most sense from an ease of application perspective. Turbocor is in full support of the choice of the live center for the rotor centering mechanism.

Upon the decision of using a live center for the centering of the rotor, the motion of the live center support needed to be determined. Two ideas were proposed for this, the first of which is essentially a hinged live center support. Prior to the rotor being lowered, the live center would lay back as seen below in Figure 5a. Once the rotor is attached to the motor, the live center could be hinged up and locked into position as seen in Figure 5b. It should be noted that these diagrams are shown just for proof of concept and the actual dimensions may vary.

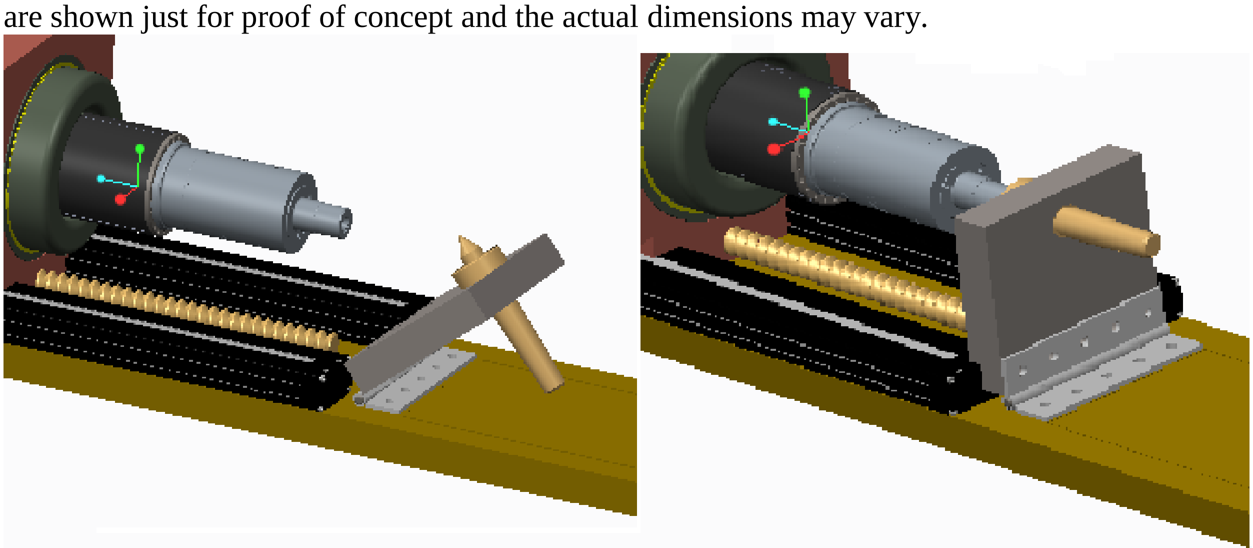

Figure 5: a. Hinged live center support open b. Hinged live center support closed

The second option involves sliding the live center along a track. This option would involve machining a track into either the base plate or a separate component that the live center support could slide along. This track will need to be precisely machining so that the position of the live center is constrained by the track. For both of these design options, a feature needs to be included that will allow for the live center support to be completely locked into place once it is connected to the rotor. This is an additional design consideration that must be taken into account. The two methods that have been discussed are utilizing bolts and wing nuts, or using a clamp.

The biggest decision that needed to be made for this design was the method for overcoming the magnetic force during the insertion process. Several ideas were proposed and presented to Turbocor, and three concepts were determined to be feasible options. These were a ball screw, a rack and pinion, and a linear pneumatic device. A ball screw is a device that will convert a rotary input into linear motion parallel to the axis of rotation. Variations that are similar to a ball screw are the power screw and the lead screw. A rack and pinion is similar to a ball screw in that it converts a rotary input into linear motion, except the linear motion is perpendicular to the axis of rotation. One issue that comes about with the rack and pinion is it easily back driven which is unacceptable for our application. Therefore, if a rack and pinion were to be utilized a separate feature would be needed to prevent back drive. The third option is a linear pneumatic actuator. Basically, this design would use pressurized air to exert a force on the stator housing to move the stator into the needed position. These three design options will now be presented and discussed in more detail. A schematic illustrating the linear pneumatic actuator can be seen below in Figure 6.

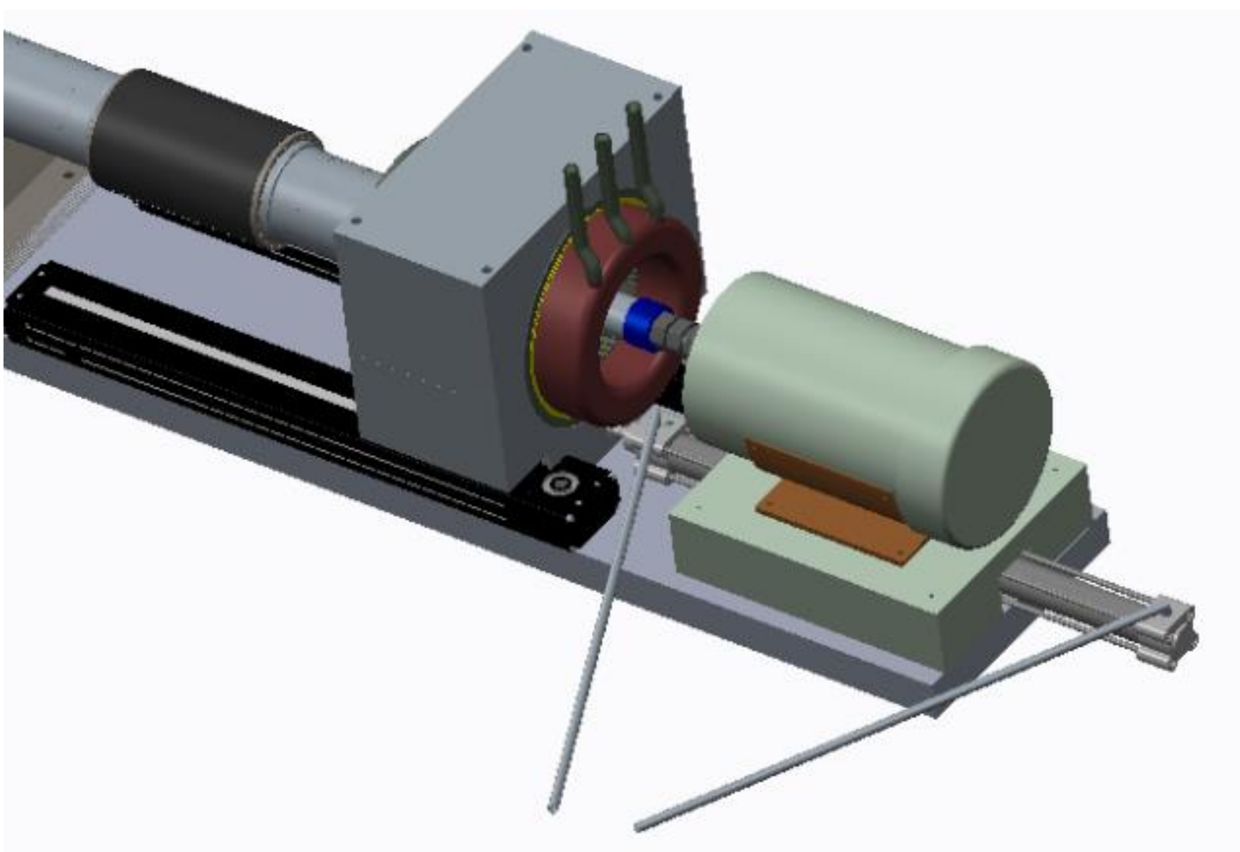

Figure 6: Conceptual prototype for pneumatic actuator

As mentioned regarding the live center hinge prototype presented previously, it is important to note that these designs being presented are prototypes only and do not represent any finalized designs. In Figure 6, two hoses can be seen connected to the linear pneumatic actuator. If this concept were to be implemented, these hoses would be connected to the shop air that is available at Turbocor. When connected, the pressure of the air could be regulated to provide the proper pressure (and thus force) needed to move the stator into position. Once in position, either a locking mechanism to keep the stator into place would be utilized, or the pressure would be kept constant to exert a constant force sufficient to prevent movement of the stator. More than likely, a locking mechanism would need to be utilized as the force exerted on the stator during the insertion process due to the magnetic field is not constant. There are two separate configuration options for the pneumatic application. The first of which is the one pictured in Figure 6, in which there are separate linear bearings for the stator housing to move along during the insertion process. The other option is for the linear bearings to be incorporated into the pneumatic device, which may actually be fore feasible. An example of one of these pneumatic actuators can be seen below in Figure 7.

Figure 7: Secondary option for pneumatic device

The option of utilizing a ball screw will now be discussed. An example prototype of the ball screw design is presented below in Figure 8. When evaluating the design choice of how the magnetic force will be overcome, the other decisions (live center vs. bearing, sliding vs. hinged live center support) are not taken into account as these mechanisms are independent from one another. For the prototypes of the methods being presented, all CAD designs are shown with the live center on a sliding live center support. For the ball screw, a crank would be connected to the ball screw which would be rotated by the operator. The ball screw would be connected to a fixed block on that set on the test fixture. Additionally, it would be connected to another moving block. This moving block would need to be fastened to the stator housing. Like with the pneumatic actuator, the stator housing needs to be connected to linear bearings. The purpose of the linear bearings is to reduce the friction that would be present without them. If the axis of the power screw is offset from the center of the stator housing, the force exerted on the stator housing will also generate a moment, and if two linear bearings are utilized they will work to offset this moment. It is possible for a ball screw to be back driven, so analysis will need to be performed to ensure that the ball screw cannot be back driven. If the ball screw selected is not sufficient to prevent back drive on its own, a separate mechanism such as a ratchet and pawl will need to be utilized to prevent back drive.

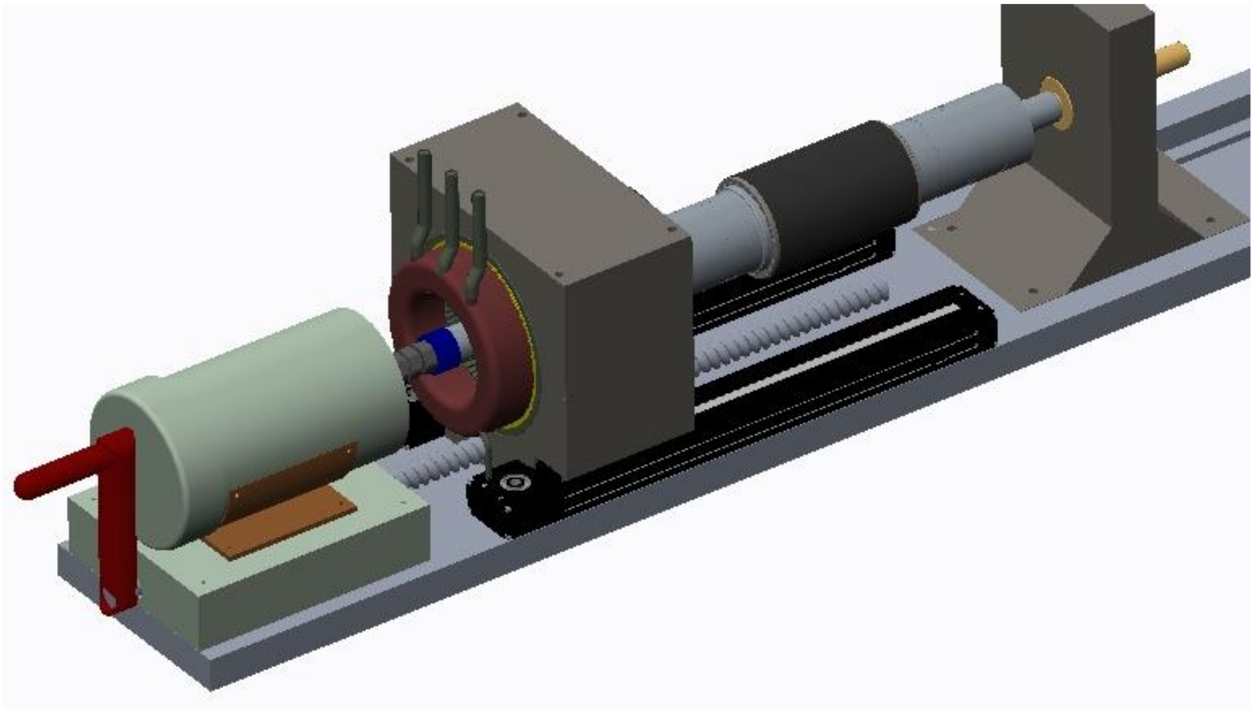

Figure 8: Conceptual prototype for power screw

The last option considered was the rack and pinion. As mentioned previously, the rack and pinion converts a rotary input into a linear output in a direction perpendicular to the axis of rotary input. Therefore, the crank input needs to be located on the side of the design. The CAD conceptual prototype for the rack and pinion option is seen below in Figure 9.

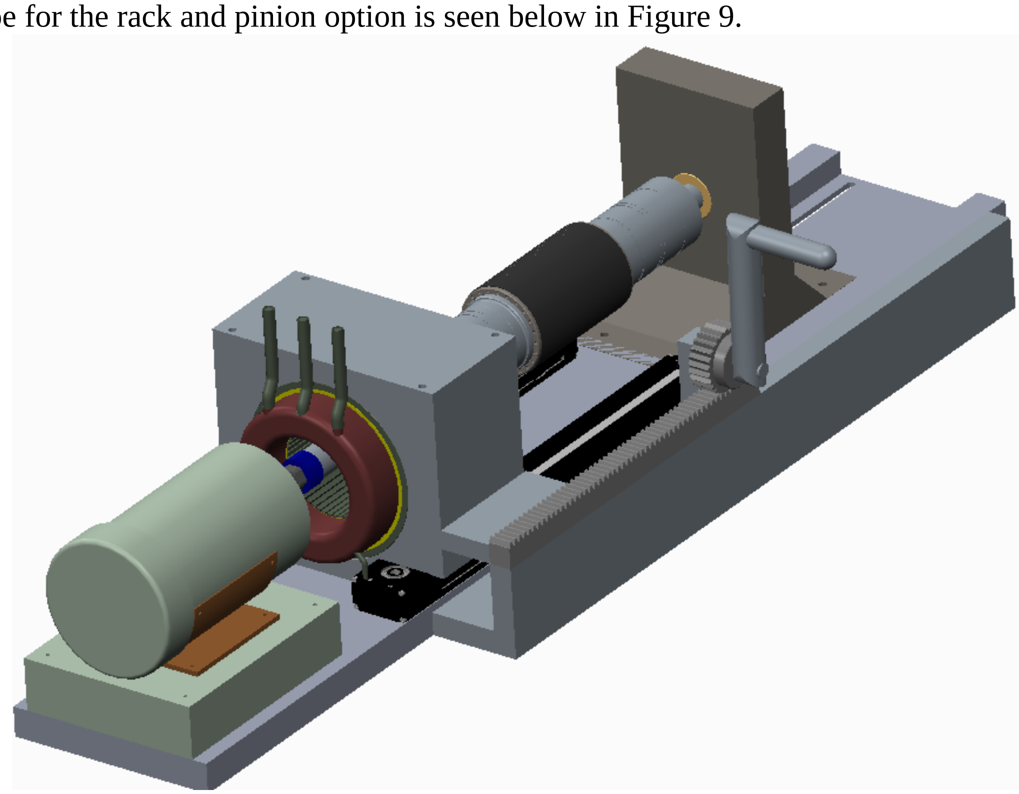

Figure 9: Conceptual prototype for rack and pinion

Back drive is a more significant issue with the rack and pinion. Back drive prevention is a necessary design component if the rack and pinion is to be utilized, but is not included in this conceptual CAD prototype. The two most viable options to prevent back drive is to utilize a mechanism that locks the crank into place, and a separate mechanism such as a ratchet and pawl that will prevent back drive on its own. With the rack and pinion, the rack would be connected to the stator housing, and the pinion would be fixed about an axis. When rotated, the pinion would move the rack linearly and thus the stator would be able to be moved into place.

3.3 Evaluation of Designs

As discussed previously, the decision to go with a live center as the method of centering over a roller bearing was due to the sponsor's preference to not utilize ball bearings in the design. The next design consideration introduced in the previous section was whether to implement the hinged live center support or a sliding live center support. A decision matrix was not needed to assist in this decision. The cost of using a hinge vs. sliding mechanism is insignificant relative to the costs of some of the other components in the design such as the motor or motor drive. The main advantage of using the hinged design would be to reduce the spatial footprint of the design. The maximum length of the test fixture is 1,600 mm. After running calculations to determine the length of track needed for a sliding live center support, it was determined that the test fixture could fit within the spatial constraints with the track feature. The smaller spatial footprint was the only advantage of the hinged assembly, and thus the track option is more viable. There is more positional play involved in the use of the hinged live center support, and it is essential that the live center is in the correct position, and that this position is consistent for each test. Because of this, it was decided to use the sliding track option as there was no advantage to the hinge. This has been discussed with Turbocor, and Turbocor is in agreement that the sliding live center support is the superior choice.

The last design choice that needed to be made was determining the method for overcoming the magnetic force during the insertion process. To determine which method was to be implemented, a weighted design decision matrix was used. This can be seen below in Table 2.

Table 2: Weighted decision matrix to determine method for overcoming magnetic force.

Design	Safety	Simplicity	Ease of Use	Cost	Durability	Weighted Sum
Weight	15	5	10	5	5	
Rack and Pinion	6	4	6	8	6	210
Pneumatic Device	2	2	4	2	8	130
Ball Screw	**8**	**6**	**8**	**8**	**6**	**300**

The three methods were evaluated based on safety, simplicity, ease of use, cost, and durability. It can be seen that the ball screw received the highest weighted sum of all design considered. It is also important to note that the ball screw would receive the highest score regardless of the weights chosen for the different categories because no other design has a higher score in any of the categories with the exception of the pneumatic device in the durability category.

3.3.1 Criteria, Method

The safety category was given a weight of 15, the ease of use category was both given a weight of 10, and the simplicity, cost, and durability categories were all given a weight of 5. This test fixture will be used by a human operator, and therefore safety is extremely important as the fixture will not be able to be implemented if it is unsafe. Ease of use was given a high weight because the test fixture is going to be implemented in a manufacturing setting. It is essential that the method is effective, and that the testing process can be completed in an efficient manner. Durability is important as the test fixture needs to have a design life of seven years. Simplicity was also taken into account because a complex method of overcoming the magnetic force may interfere with the other components of the design such as the motor or live center support.

The safety category was evaluated based on any design features that pose a risk to the operator, as well as any additional considerations involved with OSHA regulations. The pneumatic linear actuator received a 2/10 in this category. The pneumatic device would utilize air at high pressures, and that in itself would present a safety hazard. Additionally, in order to use the shop air, a safety enclosure would need to be added into the design to protect the operator. The other design ideas do not require this additional enclosure and present less of a hazard which is why the pneumatic device received such a low score. The rack and pinion device would involve the operator turning a crank to move the stator housing along a track. Rack and pinions are easily back drivable, and so a separate feature such as a ratchet and pawl may be used to prevent back drive. This may involve additional operator interaction with moving parts, and therefore additional safety hazards. However, no safety enclosure is required, so the rack and pinion still received a high score of a 6/10 in the safety category. The ball screw also involves the operator turning a crank to move the stator housing along a track. While the ball screw has not yet been selected, there are ball screws that are capable of preventing back drive, and so therefore a separate feature to prevent back drive would not be needed. This is the reason the ball screw received an 8/10 score in the safety category. If a balls screw is selected that is not able to prevent back drive and a separate feature if needed, this score would be adjusted accordingly to a 6/10.

The next category taken into account was simplicity of the design. As mentioned in the previous section, the pneumatic device would require a safety enclosure to be built around the test fixture in order to protect the operator. This would significantly complicate the design and thus the pneumatic device received a 2/10 in the simplicity category. The rack and pinion would essentially work by having the rack be connected to the stator housing; however, a separate design feature would be needed in order for the pinion to have a rigid connection. The ball screw would be able to be run through the block that the motor sits on because a ball screw works by outputting linear motion along the axis about which the crank rotates. Due to this difference, the ball screw offers a simpler solution that will not affect other components of the test fixture. Therefore, the ball screw received a 6/10 in the simplicity category and the rack and pinion received a 4/10.

Ease of use is extremely important due to the high volume of rotors to be tested using this test fixture. The pneumatic device received the lowest score in this category as well because the operator must navigate around the safety enclosure and connect and disconnect the air hoses each time the test fixture is used. The rack and pinion received a 6/10 because while it would be efficient to use, it would require the additional step of unlocking or locking in the device added to the design to prevent back drive. The ball screw received a score of an 8/10, which assumes that the ball screw selected will be capable of preventing back drive on its own. If the ball screw selected does not have this capability, the score would lower to a 6/10 to match that of the rack and pinion.

To determine the scores in the cost category, research was done to obtain quotes for the various design methods. It is important to note that this research was done for the purpose of determining a Rough Order of Magnitude (ROM) quote in order to estimate the price, and the products discussed in this analysis do not represent selected components. A rack and pinion was selected from Atlanta Drive Systems with a quality of 9 out of 10 meaning a backlash of less than 0.005 inches, and a max force feed per pinion contact of 225 pound force. The rack was 500 mm in length, had a module of 1.5, and a cost of $53. The pinion quoted had a 30 mm diameter, a module of 1.5, and a cost of $183.[6] Since back drive is an issue with the rack and pinion setup, a ratchet and pawl was also quoted from Quality Transmission Components. The ratchet and pawl found was 100 mm diameter, with a quality of 9, a 60 degree jaw angle, and manufactured from carbon steel.[7] The cost for the ratchet and pawl was $117, for a total cost of $353 for the rack and pinion setup. Next the pneumatic actuator was researched. The limiting constraint on the pneumatic actuator is the stroke needed, as the size of the actuator must be scaled up to support the long length of pneumatic piston in order to prevent buckling. A Parker product with a quote from a national retailer, Florida Motion Control, was obtained for a system with a 32 mm piston bore, two 12 mm support rods, a 500 mm stroke, and 80 psi pressure. This setup can exert a 100 pound force extension force and will meet our requirements. The total cost for the pneumatic actuator setup is $3261.[8] Finally the ball screw was researched. The ball screw selected is a lead screw from Thomson Product and is stainless steel with a 0.375 inch diameter screw with 5/8-18 threads and a 0.0625 inch/revolution lead. This lead screw is used in conjunction with a 0.375 inch bronze nut and a F37 flange that will mount to the stator housing. The lead screw is $137 dollars, the F37 flange is $35, and the bronze nut is $52 for a total cost of $224.[9] It is important to note that these are approximate costs used to obtain an estimate of the cost, and there are additional costs due to components needed to connect the methods of overcoming the magnetic force to the stator housing.

Durability was the last category considered. Durability is dependent on the final design chosen; however, it was possible to evaluate the design choices based on qualitative properties. The pneumatic device received the highest score of an 8/10 as it does not contain any components that will wear significantly with time. The pneumatic actuator rides on linear bearings which may wear slightly with time; however, the ball screw and rack and pinion also contain linear bearings. The ball screw and rack and pinion both received a score of a 6/10 in the durability category because the screw and rack both may wear over time. Overall, it can be seen that the ball screw was the best choice for the design, and it has been decided to move forward with the ball screw in this design.

3.3.2 Selection of Optimum Design

As discussed in the previous section, the prototype being moved forward with involves a sliding live center support and a ball screw to overcome the magnetic force. This prototype can be seen on page 16 in Figure 10, and will be discussed more in depth in this section. It is extremely important to note that at this point this simply represents an initial prototype of the design, and is in no way a finalized design. Drawings for the parts that make up this prototype can be found in Appendix C. It should also be noted that drawings were not made for the other prototypes seen in the previous section (pneumatic device, rack and pinion, hinged live center support) because these CAD prototypes were made purely to illustrate proof of concept.

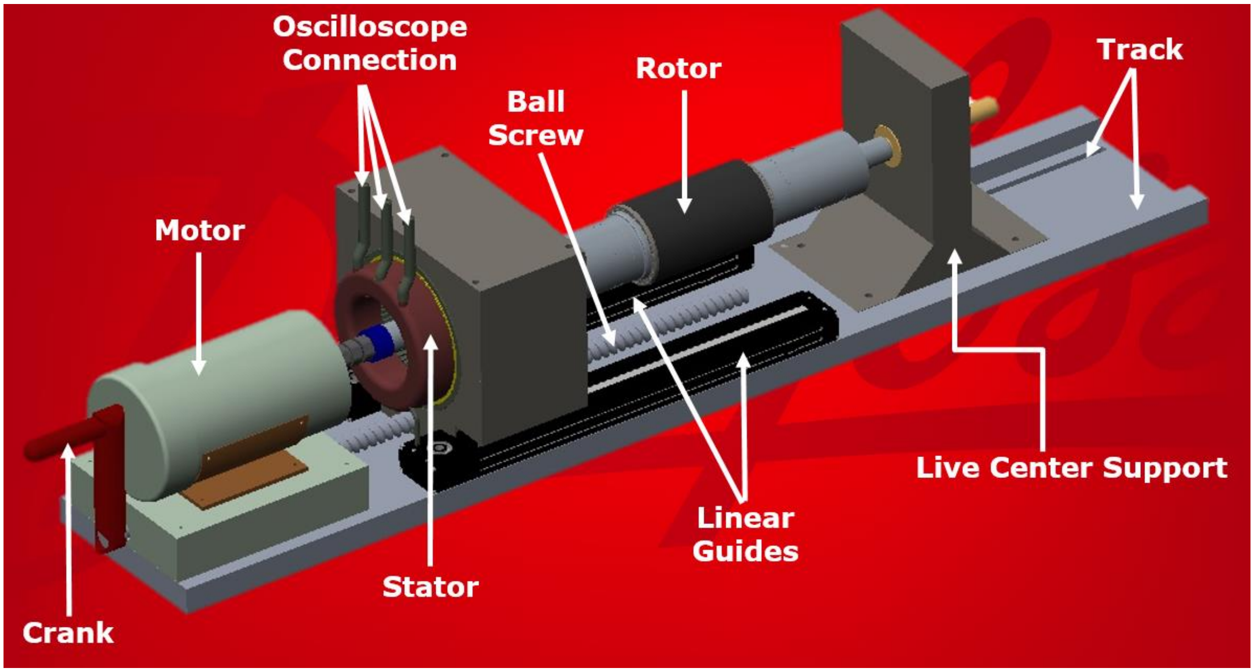

Figure 10: Initial Prototype Utilizing a Ball Screw

In this design, the rotor will be lowered down into the test fixture, and the live center support will initially be in the open position (shown closed and locked onto the rotor in Figure 10). The rotor will then be connected to the motor connection, and the live center support will be slid along the track and connected to the rotor. Once connected, the live center support will be locked into place by tightening down wing nuts (not pictured), which will be attached to bolts that run through the track. At this point, the rotor's position is fixed, and the stator will be moved along the linear bearings by turning the crank of the ball screw. Once the stator is in place, the test can be started. If the ball screw is not sufficient to prevent back drive, the stator must be locked into place. This will be done using either a ratchet and pawl system or a design feature that will lock the crank into its position. The motor can then be turned on and ramped up to the test speed using the motor drive. Once the test angular velocity is reached, readings will be taken by an oscilloscope, which is connected to the stator. From the output on the oscilloscope, it can be determined whether or not the rotor passes the quality check by looking at the back EMF. The final design will also include the table on which the test fixture sits.

4 Methodology

For this project, specific dates for the major deadlines of this project have already been identified, which will be discussed more in depth in the subsequent section. Weekly meetings have been organized with the representative from Turbocor, Brandon Pritchard, who is the liaison handling this project. This will ensure that the project is moving along at a pace that will allow for the completion of the project in a timely manner. Specific tasks have been assigned to the group members which are also discussed in the subsequent sections.

Once the preliminary design has been approved by Turbocor, all team members will contribute to the CAD design and analysis of the test fixture. The team leader will be in charge of all deliverables and delegating tasks to the group that come up over the course of the semester. A final design review has already been scheduled at Turbocor for November 20th, 2014. This date was chosen so that if any changes needed to be made to the design, there would be ample time prior to the winter break for these changes to take place. To ensure that no problems arise during the manufacturing stage of the spring semester, all parts and materials needed for the assembly of the test fixture shall be ordered prior to the end of the fall semester. One of the team members has been given the assignment of financial advisor and will be in charge of maintaining the budget of the project and insuring that funds are properly allocated.

Proper communication throughout the year will be an essential factor to the success of this project. Communication via telephone, text messaging, and email will ensure that all team members are aware of all meetings and deadlines. The weekly check-in meetings with the sponsor shall ensure that everyone involved in the project is on the same page and aware of any issues that may come up. The mentor for this project, Dr. Louis Cattafesta, shall be utilized as a technical advisor as needed. The senior design group has decided to aim for a project completion date of April 1st, 2015. This date was chosen so that if a delay in the design or manufacturing stage of the test fixture arises the project will be able to be completed within the timeframe of the class.

A block diagram has been created that shows a visual representation of how the various components of the design are related to each other. This shows the steps that need to be taken to complete the final design. This can be seen in Figure 11 below.

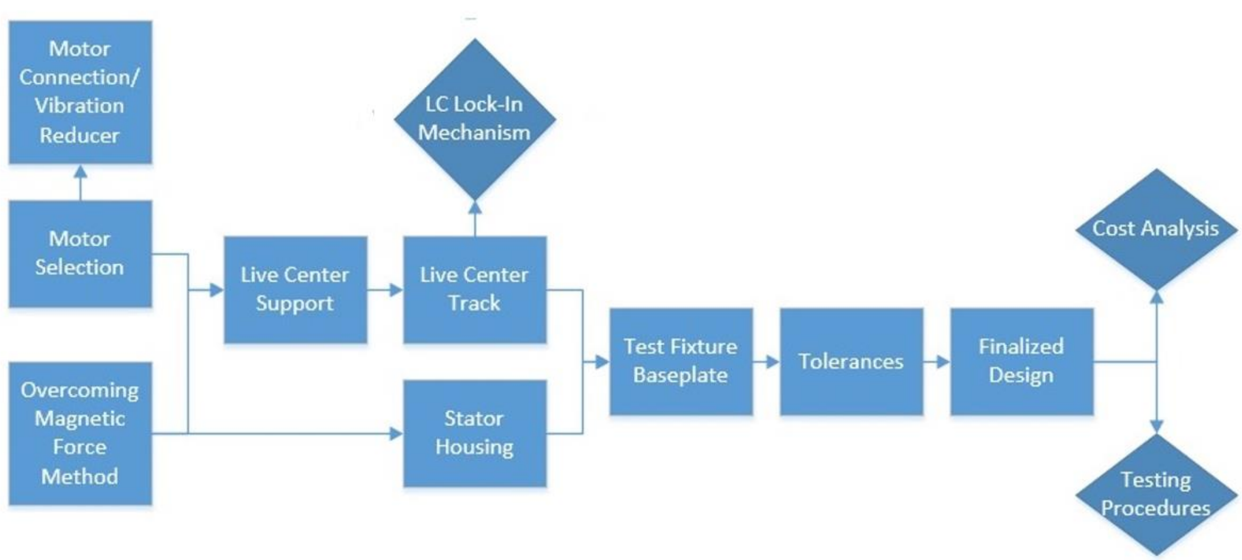

Figure 11: Work Breakdown Structure Block Diagram

4.1 Schedule

A Gantt chart is used to ensure that the project remains on schedule, and provides a view of the project progress. The Gantt Chart can be seen in Figure 12 in Appendix A. The first design stage is the Preliminary Design Stage, which encompasses research, the initial design conception, design development, and redesign. The initial design conception is brainstorming where each group member comes up with ideas to accomplish the project objective. Design development involves selecting the feasible ideas from the initial design conception stage and developing them further. Redesign is the last step of the Preliminary Design Stage in which the developed ideas are modified prior to being presented to the sponsor.

The second design stage is the Advanced Design Analysis. This stage involves taking the feedback from the sponsor, and adding it to the existing designs that were developed during the preliminary design stage, thus making the designs work more efficiently, save space, and perform better. The initial prototype was selected based on feedback from the sponsor and research performed by the group. The next step in the Advanced Design Analysis is performing further research to verify the initial prototype's feasibility. The last step in this stage is Final Prototype Selection which is where the team is currently. The last stage is the Final Design Stage, and the stages that make up this stage can best be illustrated in the Work Breakdown Structure (WBS) block diagram as seen in Figure 5 in the previous section. The final stage of the project for the fall semester is the Parts Ordering stage which will involve creating a bill of materials, getting Turbocor approval, ordering parts, and developing a testing procedure.

4.2 Resource Allocation

There are several tasks that need to be completed in order to successfully design and manufacture the back EMF test fixture and fulfill the requirements of the senior design class. The following roles have been assigned to each team member, and will be discussed more in depth in the subsequent paragraph:

- Team Leader – Russell Hamerski
- Webmaster – Andre Steimer
- Secretary – Thomas Razabdouski
- Financial Advisor – Tim Romano
- Lead Engineer – Andrew Panek

The team leader will be responsible for keeping the team on schedule, delegating responsibilities, and keeping all team members accountable for their responsibilities. The team leader is also responsible for ensuring all deliverables that need to be completed are of high quality, which includes reports, designs, CAD work, and presentations. The secretary acts as the assistant to the team leader, and is responsible for maintaining minutes of all meeting which include internal, external, and staff meetings. Additionally, the secretary is responsible for the proofreading and editing of all deliverables as a secondary check after the team leader. The financial advisor is responsible for maintaining the budget of the project and working with Turbocor to order all parts and materials required for the back EMF test fixture. The webmaster is required to build and maintain the project's website; he needs to ensure the website will exhibit sufficient information regarding the project's goal and progress.

There is significant engineering design and analysis required for this project. The lead engineer will be in charge of ensuring this design and analysis is completed in a timely manner and meets the constraints given to us by Danfoss Turbocor. All team members will be involved in the analysis of the design; however, major engineering decisions will be made by the team leader and lead engineer with input from the other team members. Per the October 30th meeting with Danfoss Turbocor, multiple tasks have been divided among team members as it is critical that all design work be completed before the final design review on November 20th. Russ will be taking the lead on the final motor selection for the test fixture. Thomas will be in charge of the final screw selection and making the decision on whether a ball or lead screw is used. Depending on the motor selected, Tim will be in charge of choosing the motor connection and vibration dampener. Tim will also be overlooking the financial data to ensure all Turbocor budget constraints are met. Andre and Andrew will be working on the CAD for the motor, motor housing, stator housing, and the selected screw. In addition to this, Andre will also be investigating different options for the live center and live center housings, while Andrew will also make the decisions on whether a linear bearing is used in the final design and choose said linear bearing if necessary. These tasks are in addition to the continued support roles that were delegated at the beginning of the semester.

5 Conclusion

Turbocor requires a test fixture to be developed that will measure the back EMF generated when the rotor is rotated within a stator. This is needed to verify the quality of the rotors as they are manufactured by a third party company. The key requirements of the design is that it must center and align the rotor within the stator to a tolerance of 0.5 mm, and it must contain a design feature that will overcome a 60-80 pound magnetic force that is exerted when the rotor is inserted into the stator. Additionally, the rotor must spin at a minimum of 1,000 RPM and the angular velocity must remain constant and repeatable so that tests may be compared to one another. Several important design considerations needed to be made in order to move forward with an initial prototype. Instead of moving only the rotor or only the stator, it was decided that both the rotor and stator would move in the final design in order to minimize the spatial footprint of the final design. The decision was made to utilize a live center to keep the rotor centered within the stator over a ball bearing, as the live center will have less durability issues. Additionally, Turbocor has indicated that a ball bearing is not preferred in the final design. Another key consideration was the live center support. The two proposed ideas were to use a hinged live center that would come up and connect to the end of the rotor, and a sliding live center that would move along a track. The main advantage of the hinged live center support was a reduction in space; however, it was determined that the sliding live center support could fit within the spatial constraints given. Therefore, it was decided to implement the sliding live center support. The main design decision that needed to be made was the method of overcoming the magnetic force exerted during the insertion process. A weighted decision matrix indicated that the most suitable choice for this was the use of a ball screw due to its safety, low cost, simplicity, and ease of use.

In order to stay on course, weekly meetings are held at Turbocor every week to ensure that there is a good line of communication between the team and the sponsor. Various team member roles were delegated to ensure all work related to the project is completed in an efficient manner. A Gantt chart was constructed based on the work breakdown structure to ensure all deadlines for the senior design class and Turbocor were met. Moving forward, the next step is to select a motor and ball screw for the design. Once these selections have been made, the linear bearings that the stator housing will move along can be selected and the dimensions for the stator housing can be finalized. The final design and all components will be completed by November 20th, 2014 when the Final Design Review will be held at Turbocor.

6 References

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[9] Thomson BSA Lead and Ball Screws (n.d.): n. pag. Web. <http://www.thomsonlinear.com/downloads/screws/Thomson_BSA_Lead_and_Ball_Screws_cten.pdf>

7 Appendix A - Schedule

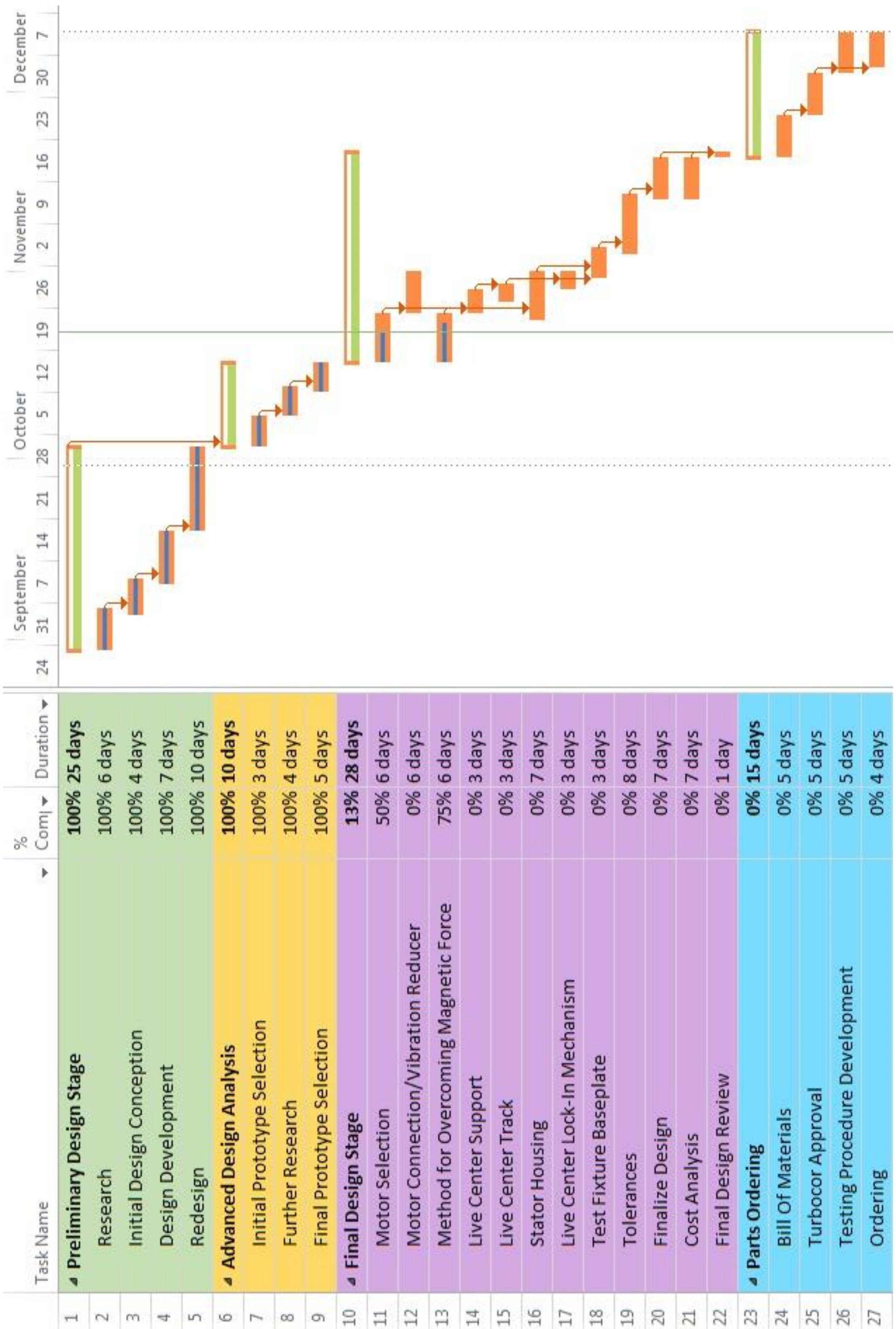

Figure 12: Fall Semester Gantt Chart

8 Appendix B – Calculations

$Inertia := 0.03027 kg \cdot m^2$ $time := 5 \cdot s$

$\omega := 1500 rpm$ $Torque_{resistance} := 10 N \cdot m$

$$alpha := \frac{\omega}{time} = 31.416 \cdot \frac{rad}{s^2}$$ (angular acceleration)

$$Torque_{inertia} := Inertia \cdot alpha = 0.951 \cdot N \cdot m$$

$$Torque_{total} := Torque_{inertia} + Torque_{resistance} = 10.951 \cdot N \cdot m$$

$$Torque_{total} = 8.077 \cdot ft \cdot lbf$$

Torque and the RPM are inversely proportionate and linearly realted. Therefore, the maximum power required is the product of half the max torque and half the max RPM

$$Power := \frac{(Torque_{total})}{2} \cdot \frac{\omega}{2} = 0.577 \cdot hp$$

9 Appendix C – Drawings

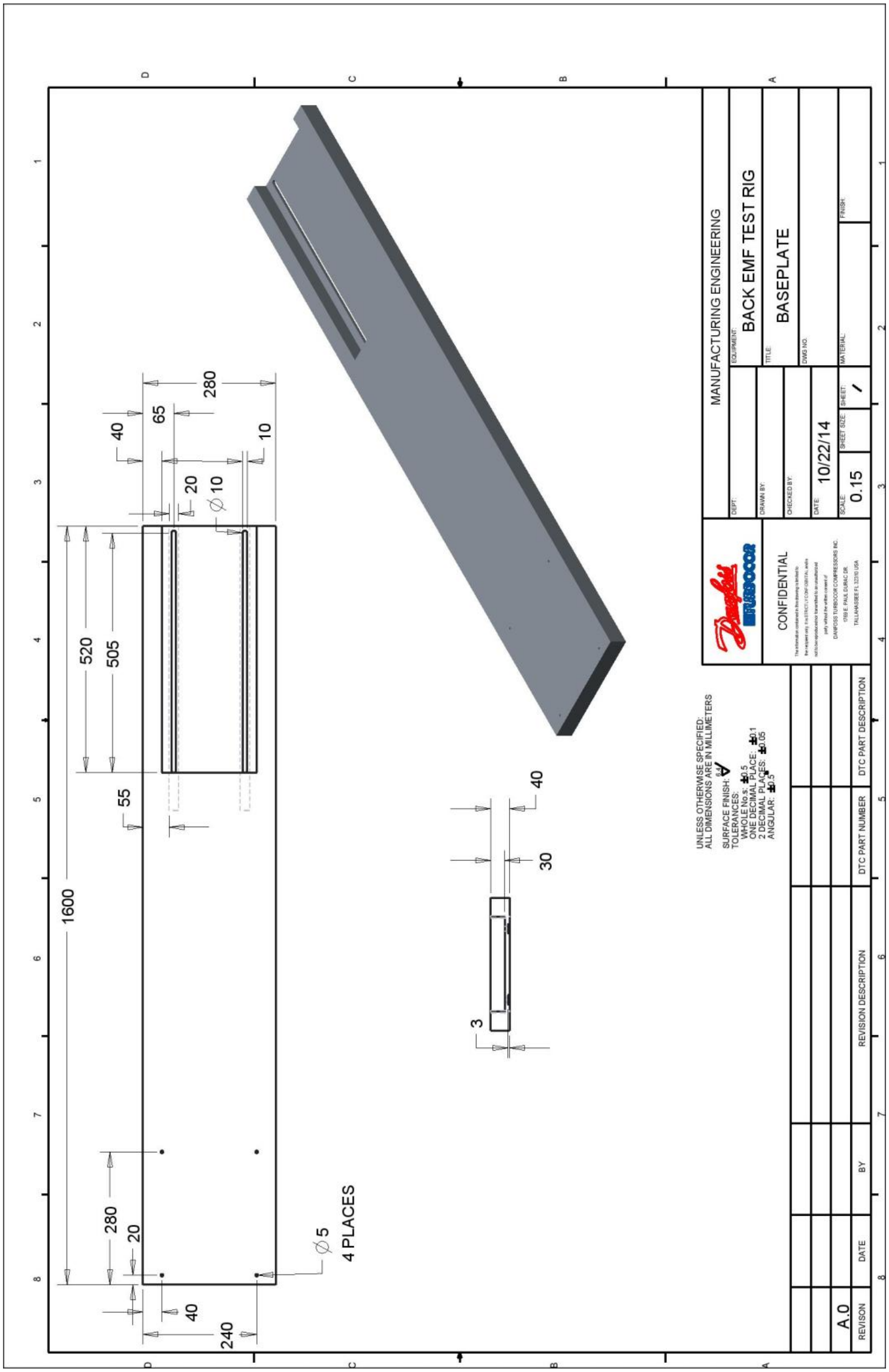

Figure 13: Baseplate Drawing

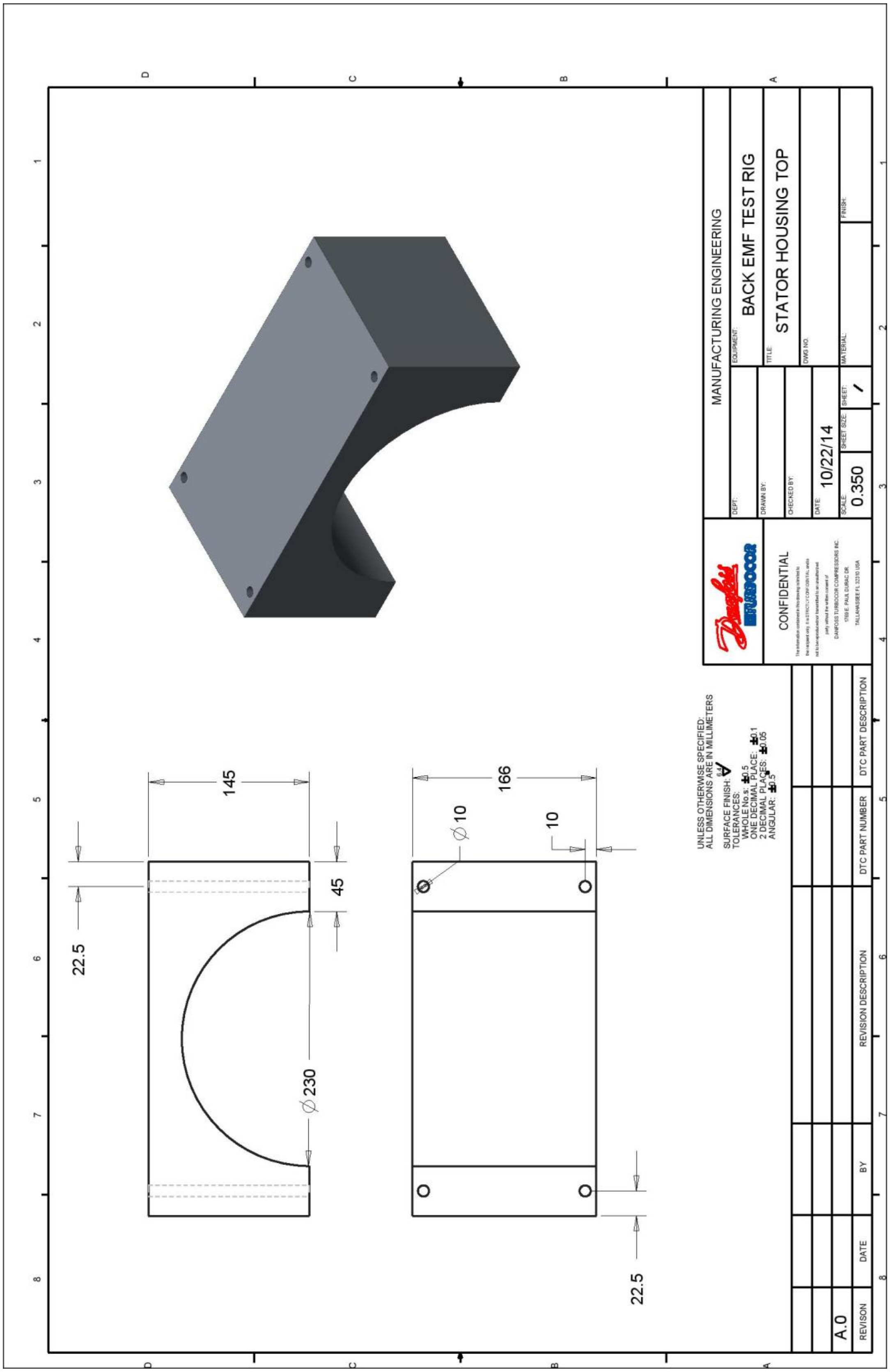

Figure 14: Top of Stator Housing Drawing

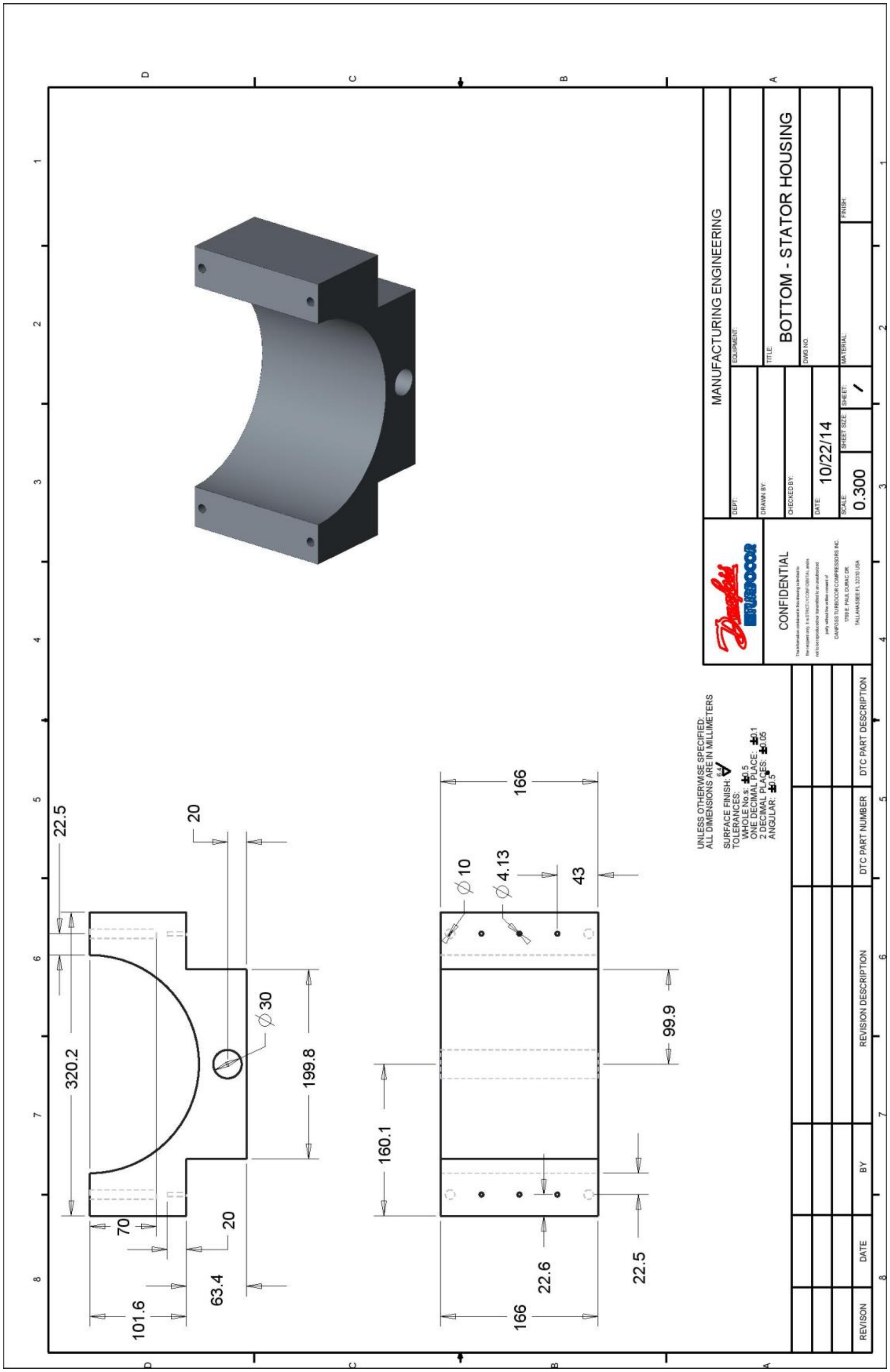

Figure 15: Bottom of Stator Housing Drawing

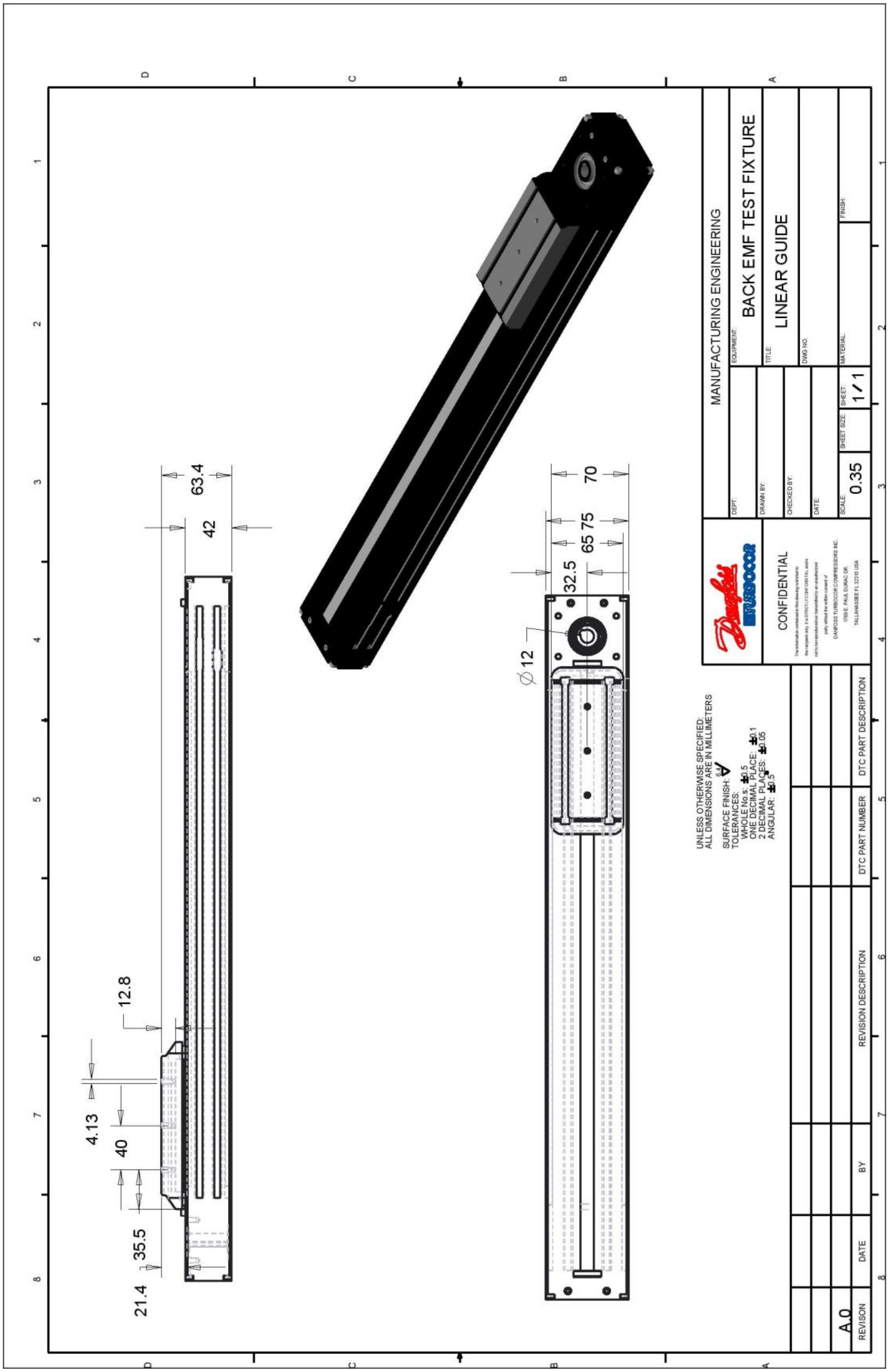

Figure 16: Linear Guide Drawing

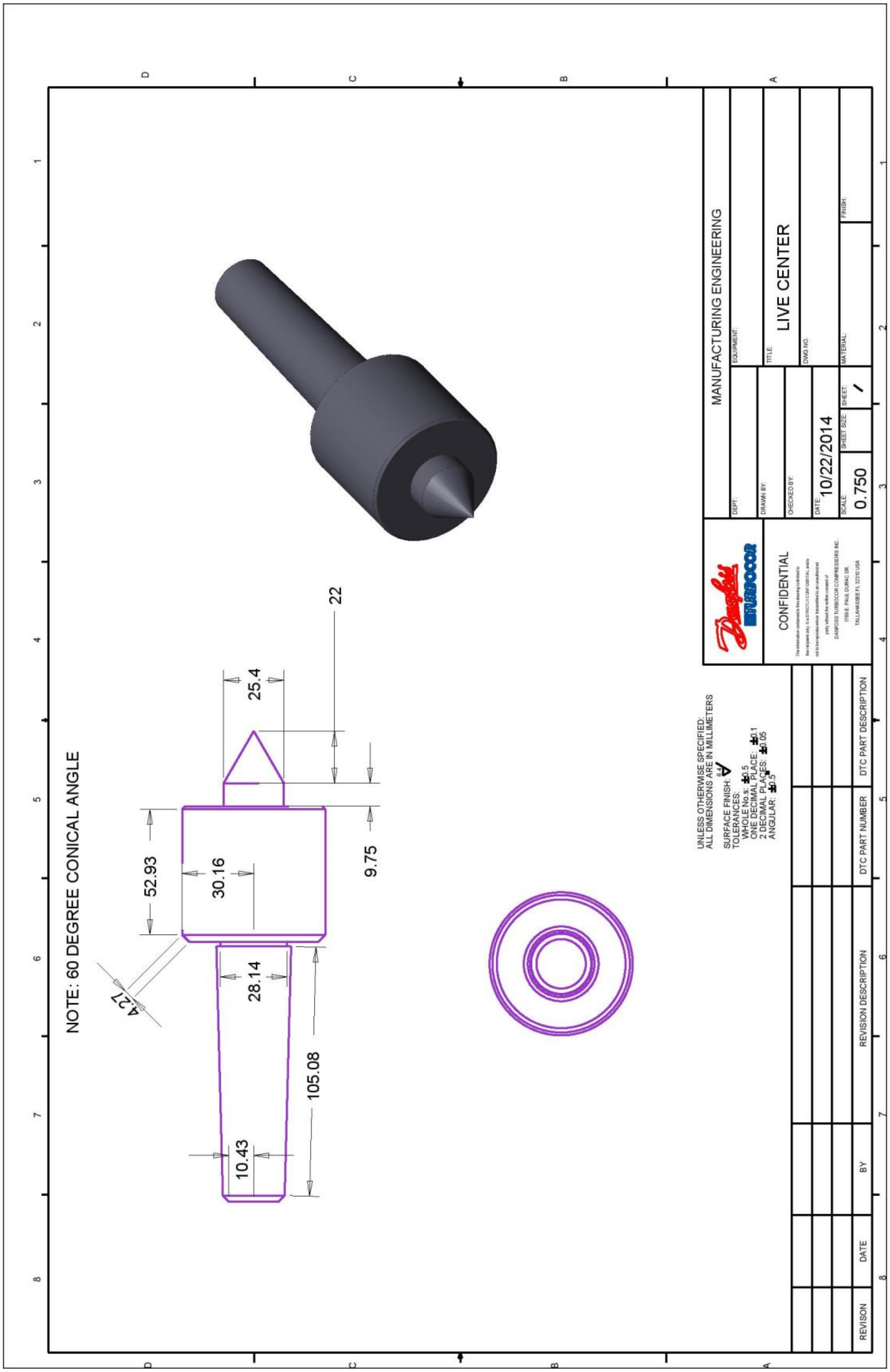

Figure 17: Live Center Drawing

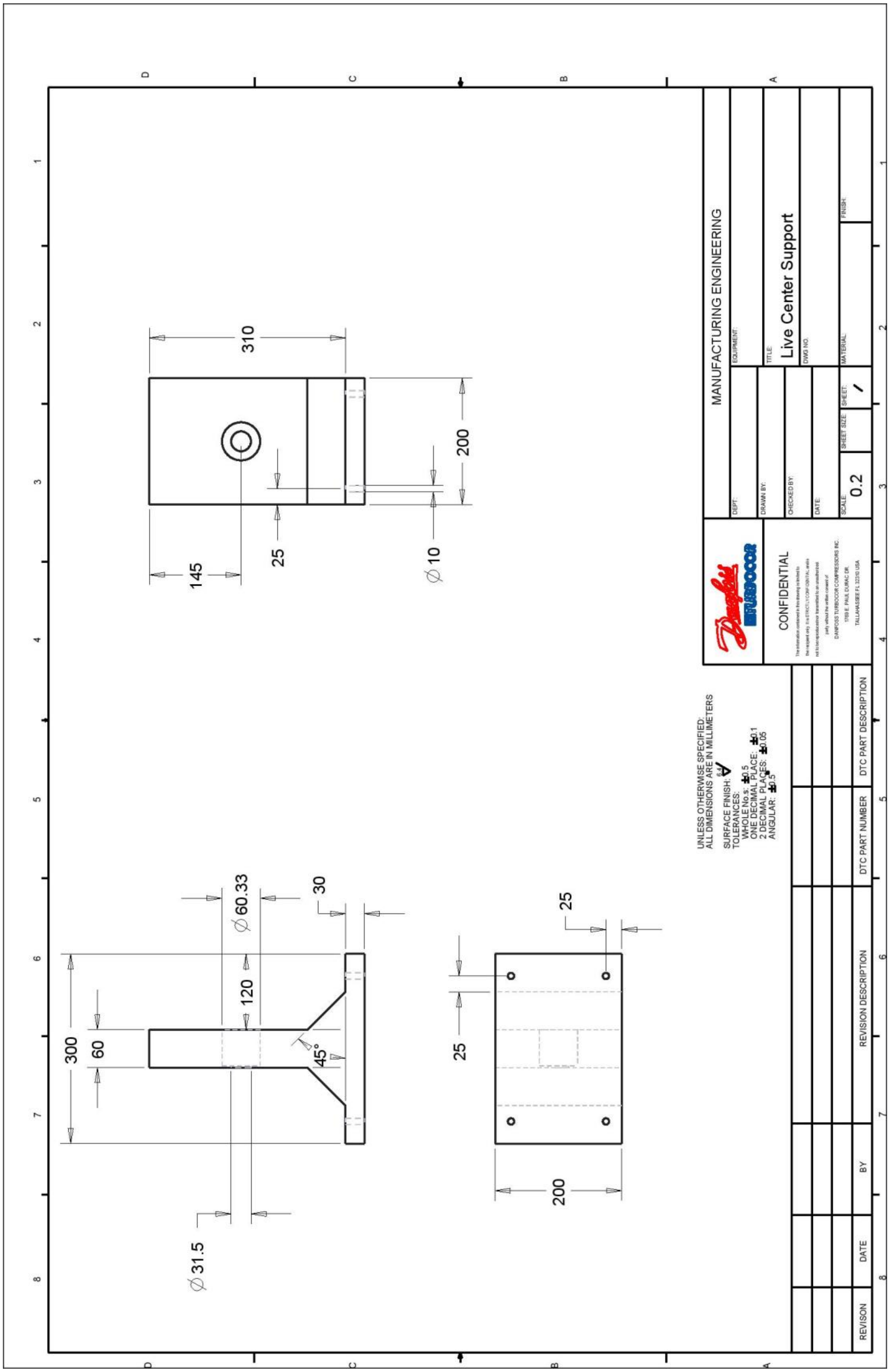

Figure 18: Live Center Support Drawing