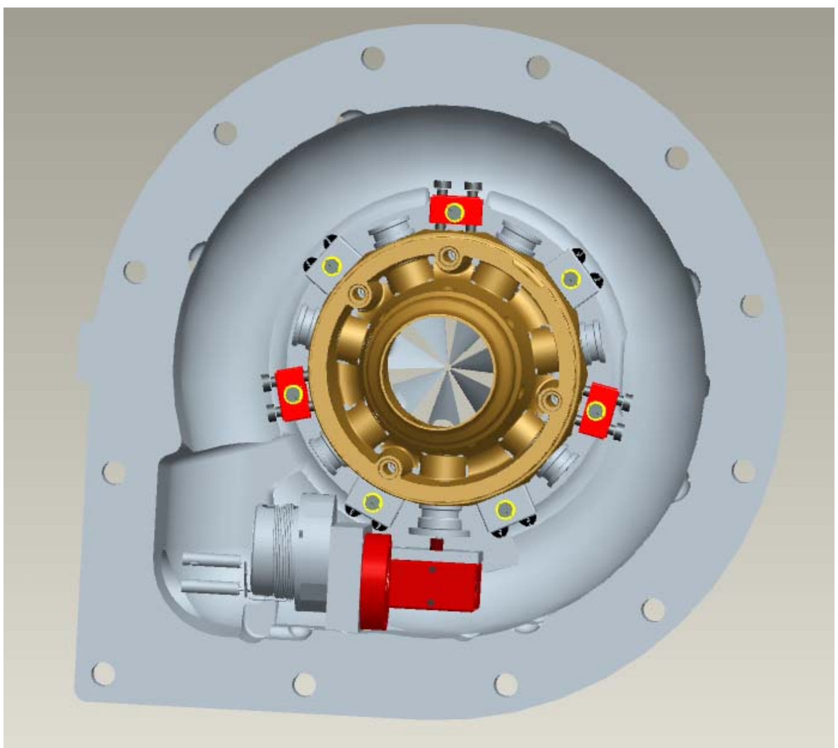
Inlet Guide Vane Control



Group 20:

Marc Alessandria
Bryan Ashley
Robert Hostetler
Mark Lawrence
Juan Pineda

TABLE OF CONTENTS

Executive Summary:

Compressors are a key part to many systems. One such system is the refrigeration system used by heating, ventilation and air conditioning. There are many ways to control the flow through compressors. One such way is with inlet guide vanes, IGV. IGV are triangular shaped pieces of material that are in the throat, inlet. The IGV need a system that will open and shut them in a very reliable fashion. In this project we will go through the process of designing a system to operate the IGV.

The process has many steps and needs. During the process we met frequently with Danfoss Turbocor, our sponsor, to learn their needs and desires. We designed with improved reliability, backwards compatible, manufacturing and cost in mind. These are the important items that any company would be concerned with.

The backward compatibility is the item that drove the design the most. The backward compatibility generated a problem with the turn down ratio that needed to be made. The ratio that was generated was 75000:110. The system that would allow for the gear ratio and reliability is a pulley system.

1.0 - Background:

Compressors require a means of controlling the capacity, when the speed drive is constant. The most common way to alter the capacity is to create a swirl in the refrigerant prior to entering an impeller. Inlet guide vanes, IGV, also referred to as pre-rotation vanes, produce the swirl.

To throttle the flow, IGV rotate from being parallel to the flow all the way to being perpendicular. The flow is throttled to reduce the work needed at the desired air outlet conditions. This decreases the input work and improves the specific power. The IGV can also increase the flow by pre-rotating the flow against the impellers direction of rotation.

Performance:

- The efficiency increase from having IGV on only the first stage decreases as the number of compression stages increase.
- The more precise the IGV are adjusted, the greater their impact on performance will be.
- IGV are only beneficial when the compressor is not fully loaded.
    - IGV benefit the power input when partially loaded.
    - IGV at over-throttle can increase the flow rate.
- IGV must be positioned as to not allow harmonics between the IGV and impeller.
- IGV don't increase turndown, they enhance efficiency during turndown.

IGV main benefit is the turndown control. IGV reduce the power input when the compressor is running at a flow rate lower than that of its design. There are other ways to control the flow such as;

- Butterfly Valves
- Variable Speed Drive Compressor
- Suction Throttling
- Impeller Throttling Sleeve
- Adjustable Diffuser Vanes

IGV are just more efficient at turndown control. This other methods however, can be combined with IGV to create an even higher efficiency, like the compressor that Danfoss Turbocor builds, maintains and designs.

Danfoss Turbocor's compressor is a combination of the variable speed drive and IGV. This leads to higher efficiency than the leading oil-flooded screw compressor.

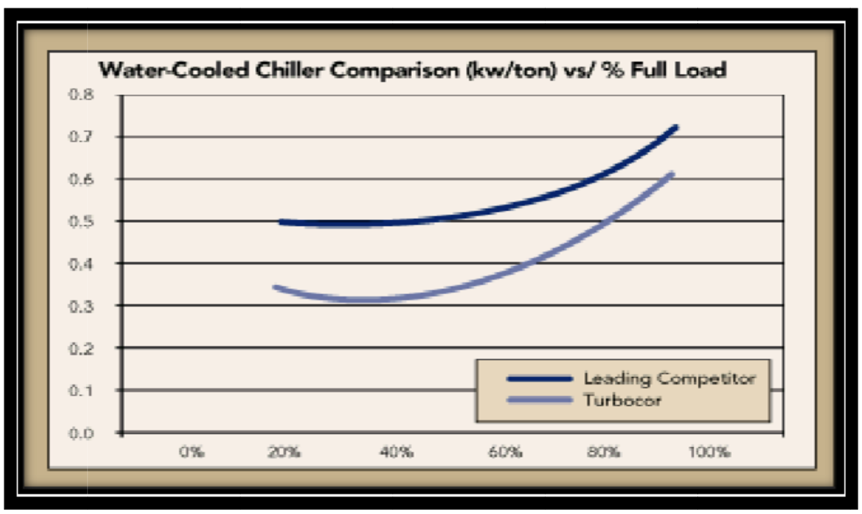


Figure 1.1 – 75 Ton Danfoss Turbocor Compressor vs. Leading Oil-Flooded Screw Compressor.

As can be seen in the above figure, Turbocor's compressor is more efficient at 100% of the load. However, at 20-60% the Turbocor compressor requires significantly less power, thus leading to a much higher efficiency and the need for IGV's.

2.0 – Project Scope and Needs Assessment:

Danfoss Turbocor produces state of the art HVAC and refrigeration compressors for commercial use. In order to keep a step ahead of the competition, they are continuously looking to improve the subsystems that make up their compressor. As a part of this process they have sponsored a Senior Design project for FAMU-FSU College of Engineering. The scope of this project is to design, build, and test a prototype for the Inlet Guide Vane control mechanism.

The redesign of the Inlet Guide Vanes (IGV) mechanical system implementation is one which will eliminate the use of a cam mechanism for the opening and closing of the fins. Our main goal is to emphasize attention to alternate means of mechanical or electrical type systems suitable for the substitution of the actual cam system.

The current IGV design is composed of:

- CAM Mechanism
- IGV Blades
- Drive Housing
- Worm Gear
- Stepper motor
- Compressor Intake/ Throat

All components work together as one unit. The motor drives the worm gear; the worm gear then rotates the drive housing. Inside the housing are slots in which the cam rollers follow. As the cam rollers follow the slots, the arm rotates and is directly attached to the IGV inside the throat.

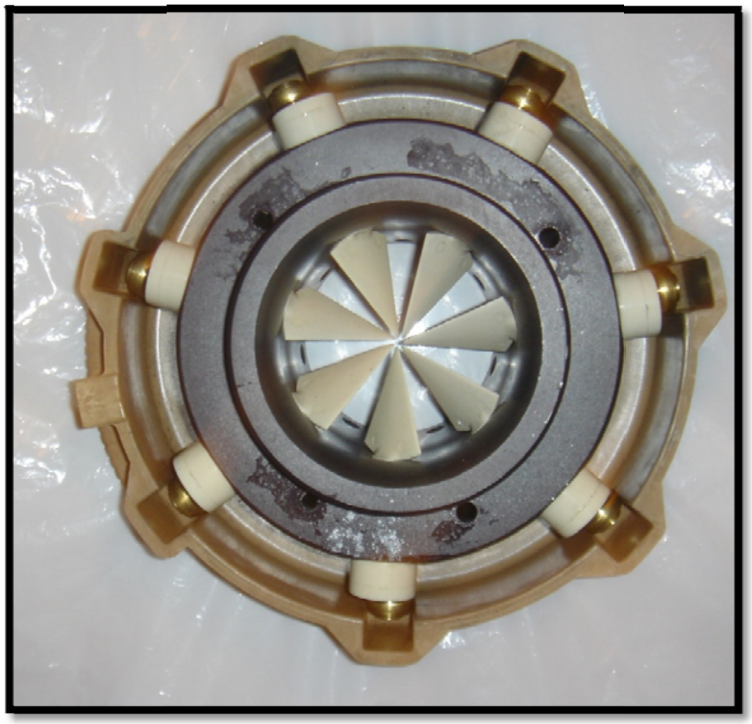


Fig 1.1 – The drive housing with the throat, IGV and cam arms attached. The black part is the throat, the cam rollers can be seen in the slots of the drive housing, and the IGV are the triangular shaped pieces in the middle.

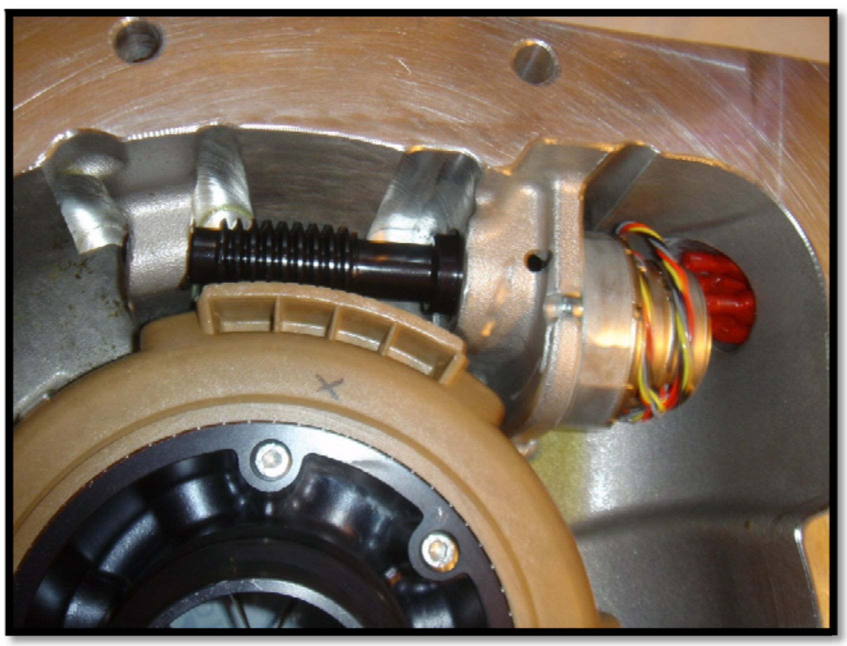


Figure 1.2 – The motor with worm gear attached and drive housing in the aluminum casing. The housing has small teeth on the outside, which collates with the worm gear.

As of now, the design works in a way in which the blades open from 0 degrees starting position to a 110° opened position. One complication emerges when the cam roller's component meet its natural toggle which at times causes the mechanism to jam.

The existing IGV assembly has a small failure rate, due to toggle position, but is extremely sensitive to manufacturing tolerance. The drive housing has a very small tolerance for the assembly to the aluminum casing. This is due to the motor and worm gear assembly which require very low tolerances for the worm gear to meet with the drive housing properly.

Needs to remember for the redesigning of the new mechanism are the following:

- Backwards Compatibility
- Reliability
- Increase in Tolerances
- Reduction of Costs

When redefining the new concepts behind this mechanism, backwards compatibility between the stepper motor and the rotational degrees of the fins must remain constant. The stepper motor must turn 75000° to the 110° of the fin; this translates to about 681.81:1 gear ratio from the stepper motor to the fin. In the current design, a worm gear is used which helps with the step down ratio needed.

Depending on what type of mechanism implemented either the use of a gearbox or gear step downs must be utilized if the worm gear is to be replaced. The location of the motor in the aluminum casing is not a fixed constrain in term of internal design or orientation. The inside of the aluminum casing can be redesign if needed for the relocation of the individual components.

The reliability factor of the redesigned mechanism is a very important need, due to the demands of Turbocor. The current design demanded that the whole aluminum casing be replaced when the system jams; this affected the consumer due to increase in service costs and it increased manufacturing and reliability costs. It is impossible to say that the system would eventually be a fail-safe, although prevention methods must be made. To do so, a look in the material selection is imperative, as well as mechanism safety and estimation on its duration and lifetime. The material selection is also important due to the use of R-22 refrigerant in the compressor. The materials must be compatible with R-22 or equivalent corrosiveness.

Ultimately the mission of our senior design group is to improve on the existing mechanism that operates the IGV for Danfoss Turbocor's compressors. The goal is to maintain constraints designated by our sponsor while allowing ideas to flow and expand. The goals are to maintain backwards compatibility, increase the reliability of the mechanism that operates the IGV, while lowering tolerances and reducing costs.

At the final closure of this project our group will have produced a design that meets both the needs of Danfoss Turbocor and their customers. As far as the current design of the IGV and the throat are not to be changed. The rest of the current design will either be altered or completely redesigned to produce the desired outcome of the specifications for this mechanical component. This new design will be more reliable, easy to manufacture and install. It will have retrofit capabilities, while maintaining cost effectiveness.

3.0 - Design Specifications:

Since there is a current design for the Inlet Guide Vanes control mechanism, the system specifications are well defined for the proposed redesign of the mechanism.

General

- The final design must be compatible to the current system. In order for this, the design must incorporate the current guide vanes and throat design and must fit in the bell housing.
    - Housing space is roughly the shape of a cylinder with a diameter of 294mm and a depth of 78mm.

Operation

- Mechanism must uniformly and smoothly (with no “jams” or “hang-up’s”)
- Angle of the IGV operation
    - 0° to 120°
        - At 0° the vanes are at a fully closed position perpendicular to the flow
        - At 90° the vanes are oriented parallel to the flow through the throat.
- In order to minimize turbulence due to vibration in the vanes, backlash must be minimized in the control mechanism. Backlash in the vanes must be held to the same or less than the current design.
    - Current backlash is approximately ± 5°

Motor

- The programming that operates the motor must remain the same and so must the motor.
    - The motor requires 10,000 steps at 7.5° per step to reach fully open at 110°.
    - To shut vanes the motor must run in reverse 11,000 steps.

Manufacturing and Assembly

- All materials used or defined in the final material list must be compatible with R22 refrigerant and other refrigerant and oils.
    - Aluminum Alloy of medium or high quality.
    - Chevron Phillips Ryton® BR42B Polyphenylene Sulfide Compound
        - Polyphenylene Sulfide, Thermoplastic, Polymer
        - Ryton® BR42B is a 40% fiberglass reinforced polyphenylene sulfide compound specially formulated to have a low coefficient of friction for use in low surface friction and wear applications. Has a wear rate of 0.001 mm/hr as per ASTM D3702.
        - Or higher grade material
- Design should open current tolerances.
    - 5-10% is the minimal opening of tolerances.
- Assembly Time
    - Maintain current assembly time or faster. The currently IGV assembly is assembled at 1 per hour. Less time is actually required.
- Manufacturing Cost
    - Cost must not exceed current cost of $371.06

4.1 - Design Concepts:

To solve our problem we considered several different ideas to rotate the IGV's. These concepts are explained below and shown in the attached drawings.

Concept 1: Gear Drive (Fig. 1)

This was the first concept considered. It involves removing the cams and replacing them with spur or bevel gears. These gears are then rotated in unison by a large ring gear. This idea does not further complicate the design because the gear would move in unison and would consist of about the same number of parts. This design could however present a problem with assembly because the gears and in turn the orientation of the fins would have to be specifically arranged so the mechanism could function. Another disadvantage could arise in the elevated cost due to the use of fine toothed gears to ensure minimal backlash. However, cost could be cut due the fact that less material would be needed to construct the IGV apparatus. Using gears also provides some freedom in the type and position of the motor. One of the greatest advantages to this potential design is the fact that it would give the IGV's a full range of motion from 0 to 360 degrees but implementing an effective gear ratio to achieve backwards compatibility would be a challenge. Overall, this was considered to be one of the soundest concepts.

Concepts 2: Cam Track Stops (Fig. 2)

This concept was the simplest of them all. Small stops would be inserted onto the cam tracks on the current disc drive. This would prevent the cams from moving into toggle by simply preventing them from ever reaching it. This design would be easy to implement, cheap to produce and would be completely backwards compatible with Turbocor's compressors. However, this design would limit the range of rotation of the fins and would use the existing cam design. These disadvantages would prevent us from satisfying our sponsor's requirements of extending rotation range and not using a cam design. The latter of which was made apparent after all the teams design ideas had been considered.

Concept 3: Gear Drive with Linkage (Fig. 3)

This design is very similar to the first concept the group had. The difference lies in the fact that instead of all the cams being replaced by gears, only one would be. This one gear would be in contact with a ring gear. When this gear would turn a linkage would be moved that was connected in series with other fins. This would cause the fins to rotate in unison. This design would increase the range of rotation, but this design might be difficult or near impossible to make backwards compatible with existing compressors. Because of the use of more parts, this design further complicates the existing mechanism. Also, if one linkage were to break due to vibration or fatigue, it would not only render the mechanism useless, but would cause detriment to the compressor. A higher number of parts would raise the cost. Due to an increase of complexity and decrease in reliability this design idea was thought to be ineffective.

Concept 4: Wire and Spring (Fig. 4)

This design equips the vane ends with wires. These wires are then in turn affixed to a spool connected to a motor, when this motor rotates, so do the fins. When the spool is turned to release the wires, the vanes will move back to position with the use of springs. Although assembly may not be very difficult, this design adds a considerable amount of complexity. A considerable number of parts would be needed for this design to function. More parts would reduce the reliability and promises a higher possibility of part failure. A gear train would have to be implemented such that the required turning ratio between the motor and spool would be adequate for backwards compatibility. Another disadvantage to this design is the fact that springs would be needed to establish tension in the opposite direction of the spool. If these opposing wires are not tight enough vibration could occur. A slight increase cost would be incurred due to the use of multiple cables. An alternate version of this design featured the use of 2 motors in opposite directions but was not considered feasible because of the higher complexity. While this design would provide a full 360 degrees of motion, its complexity made it somewhat ineffective.

Concept 5: Belt/Cable Drive - Pulley (Fig. 5)

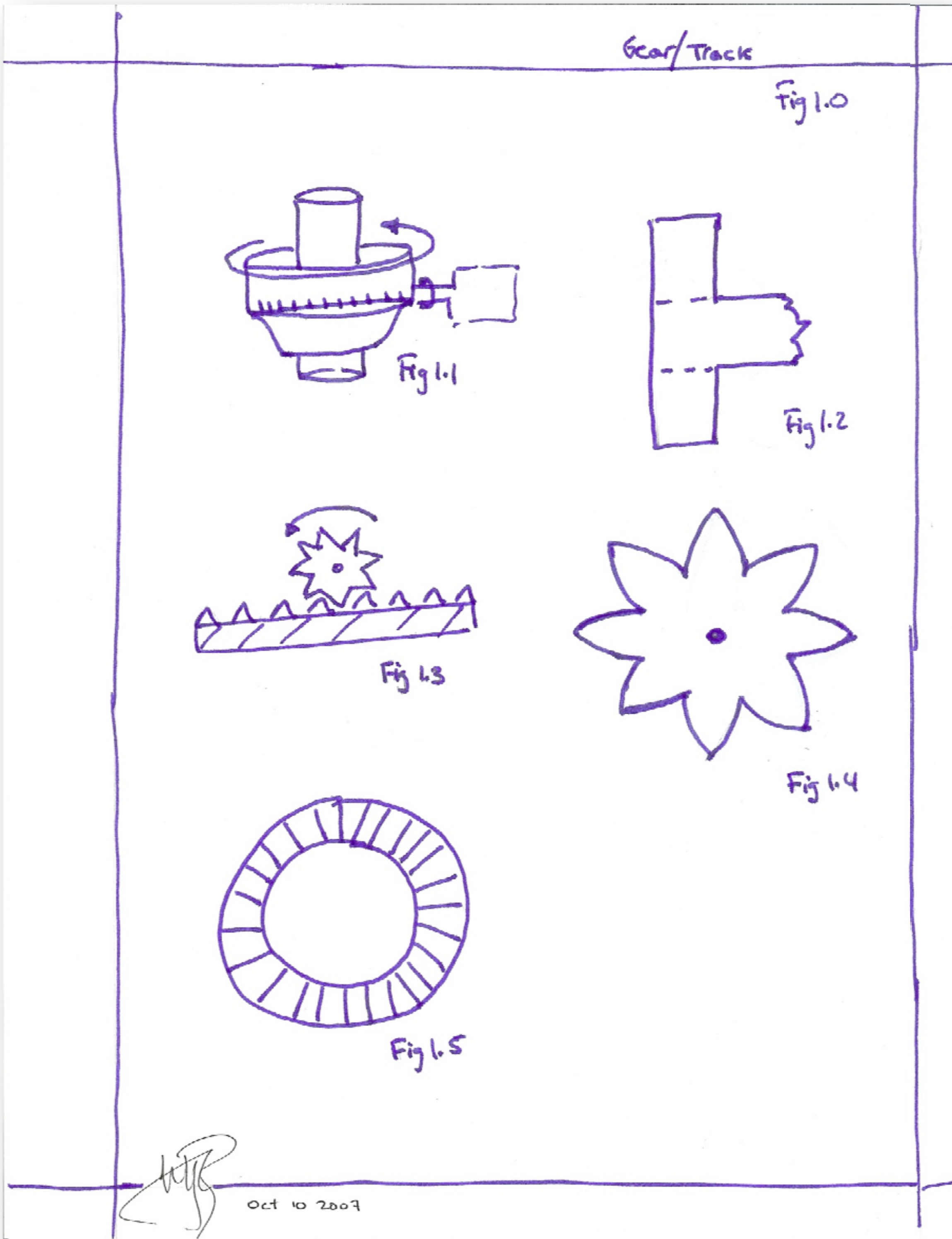
In this concept one belt is looped around the circular base of the IGV's. The belt/cable would consist of a material that would not corrode in the refrigerant used in the compressor. A set of pulleys would be added to the exterior of the throat such that the belt/cable could be wrapped around the vane base and then the pulley until completely encircling the throat. The belt/cable would then be tensioned with an adjustable pulley. While this design adds some complexity, the adjustable tension device would allow relaxed tolerances which in turn would make assembly easier. The belt/cable would cut down on any back lash, keeping vibration to a minimum. On site repair could be considerably easier since the belt/cable would be easy to replace. While a full 360 degree of motion would be achieved, the system would still need some modification to handle the high turning ratio specified by the projects backwards compatibility requirements. A disadvantage to this design is the fact that a rubber or plastic belt may break down considerably often causing a need for more service of the compressor. However, a metal cable may be used that will not break down as a polymer may. Overall, this was considered one of our strongest designs.

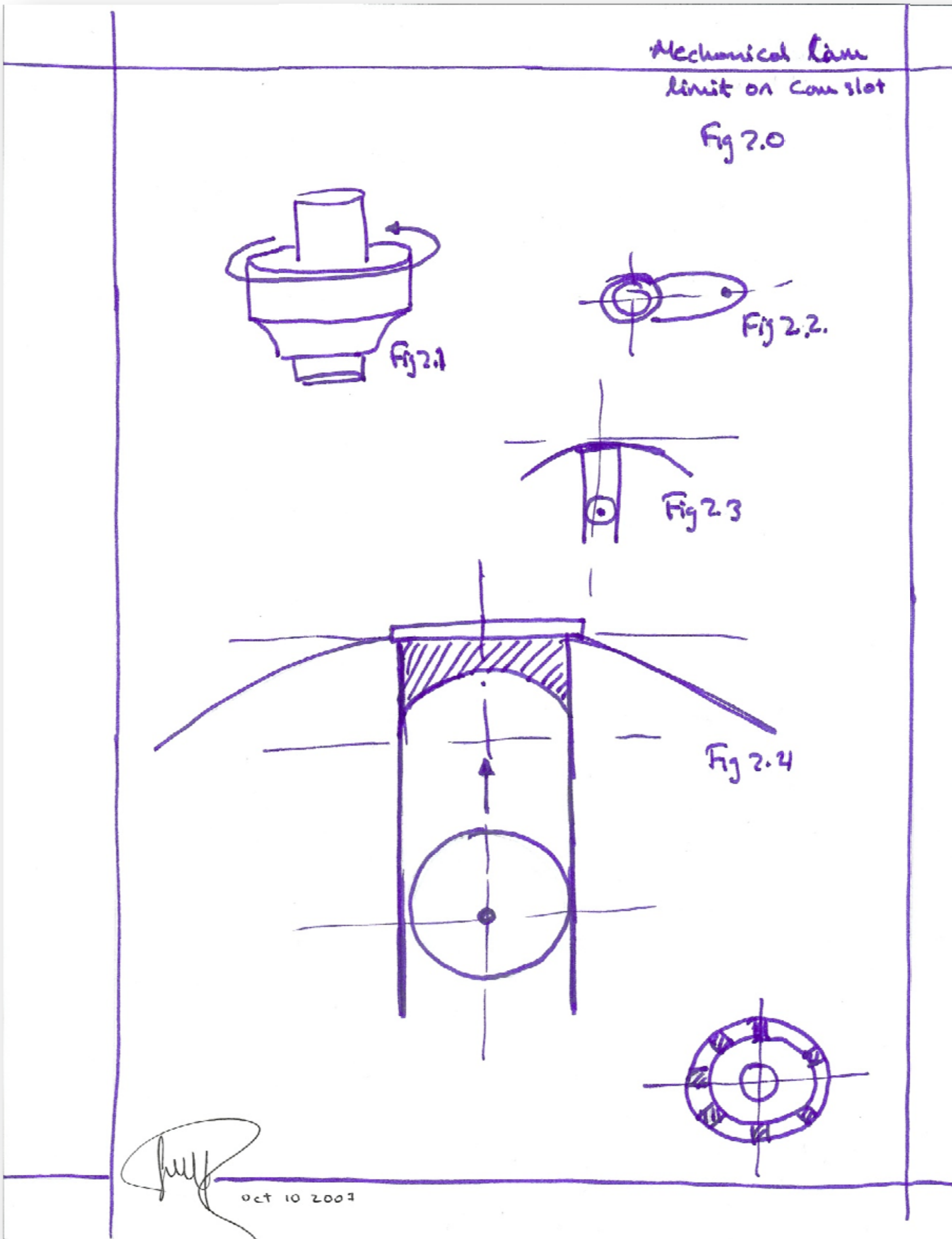
Concept 6: Cam Track Reorientation (Fig. 6)

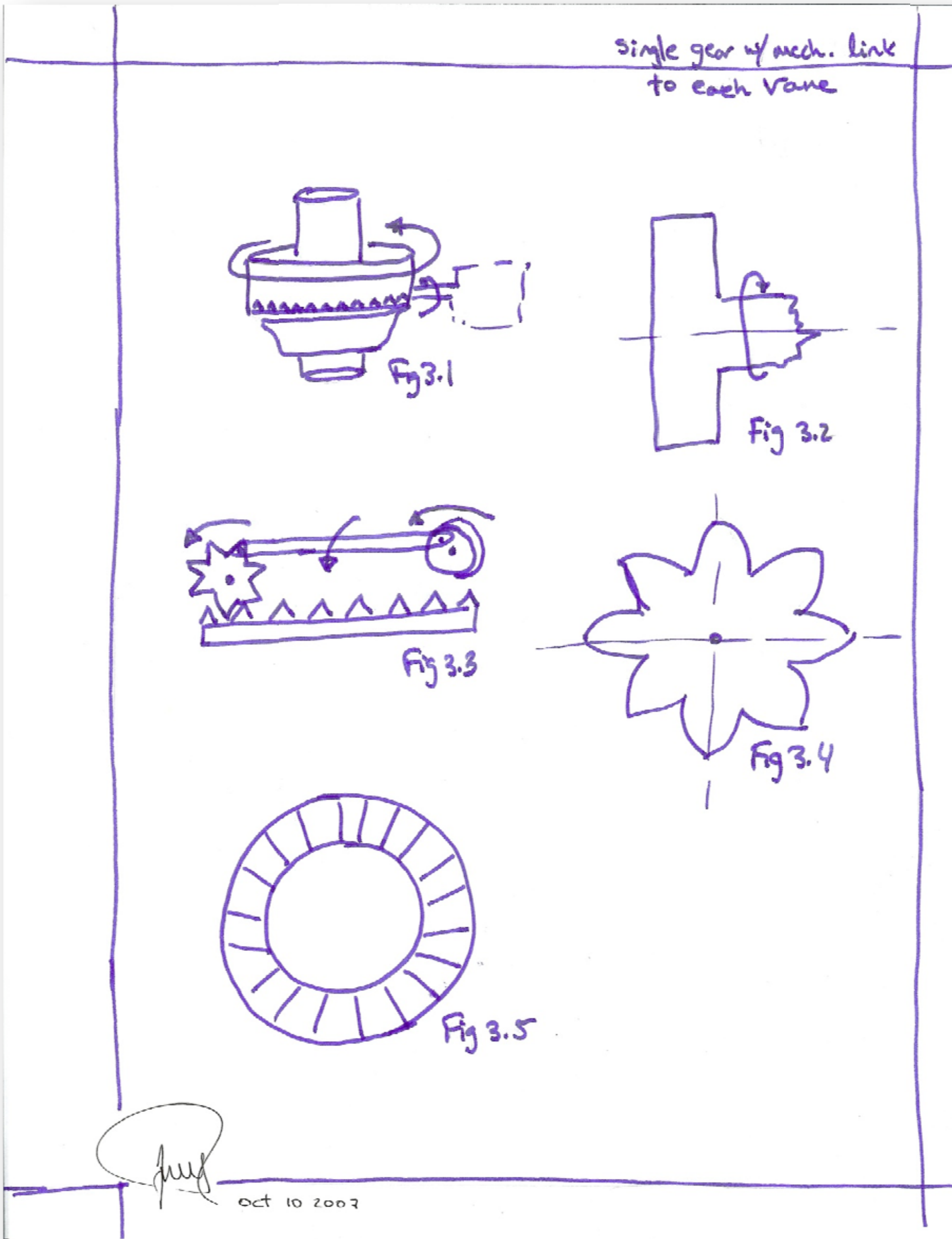
This concept provided a very simple solution. If the cam slots themselves would be tilted or curved, the position required for the cam to move into toggle would be change which could change the range of rotation of the attached vane. This idea would raise the cost higher than current design, due to more difficult molding and machining of the cam slots. This design would be as difficult to assemble as the current design and would require similarly strict tolerances. While the new cam slots could make the range of rotation greater and would be able to handle backwards compatibility requirements, it would still use cams, which was very discouraged by our sponsor and as such this design was seen as ineffective.

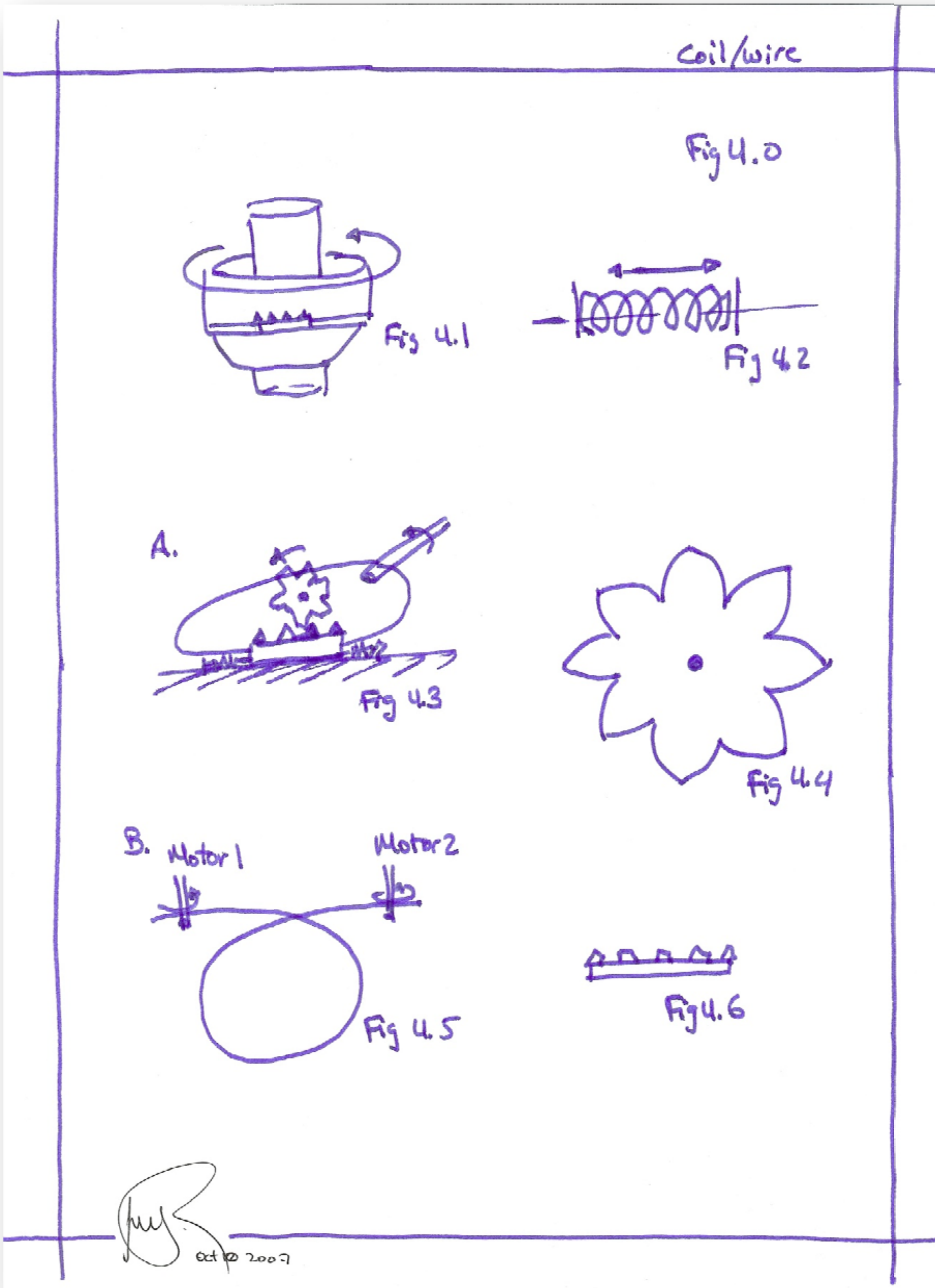
Concept 7: Individual Motors (Fig 7)

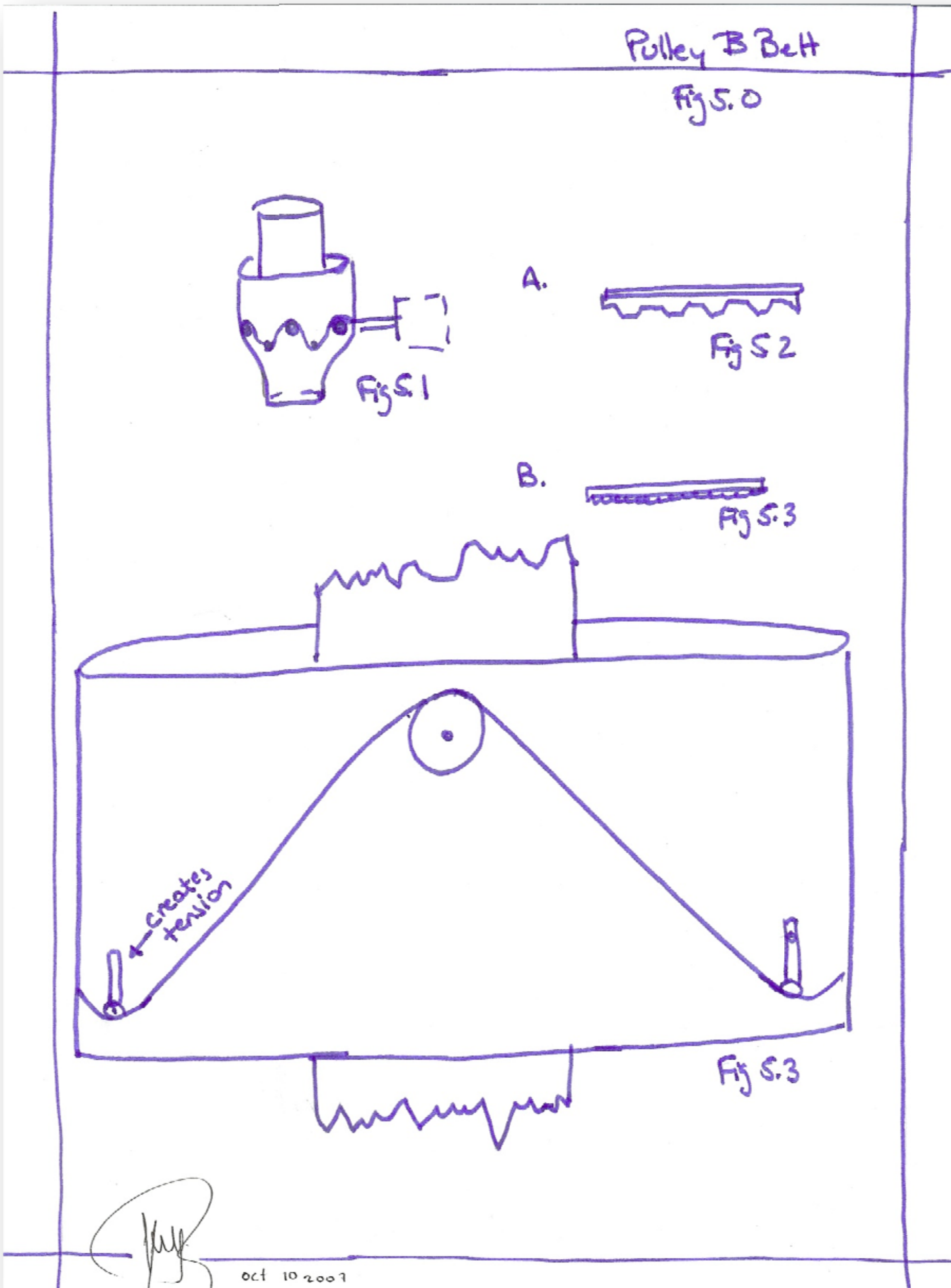
This design consists of affixing a motor to each vane and the motors move the vanes independently. While they can be moved in unison, if one motor were to fail, the compressor would be inoperative. Using additional motors only further complicates the design, by requiring additional controls to be written violating our backwards compatibility requirement completely. More motors also would also raise the cost. This design would be easy to assemble; however, it was considered a weak design because of the added complexity, cost and reduction in reliability. The other problem is the backward compatibility. The individual motors would not allow for the same controls and motor to be used.

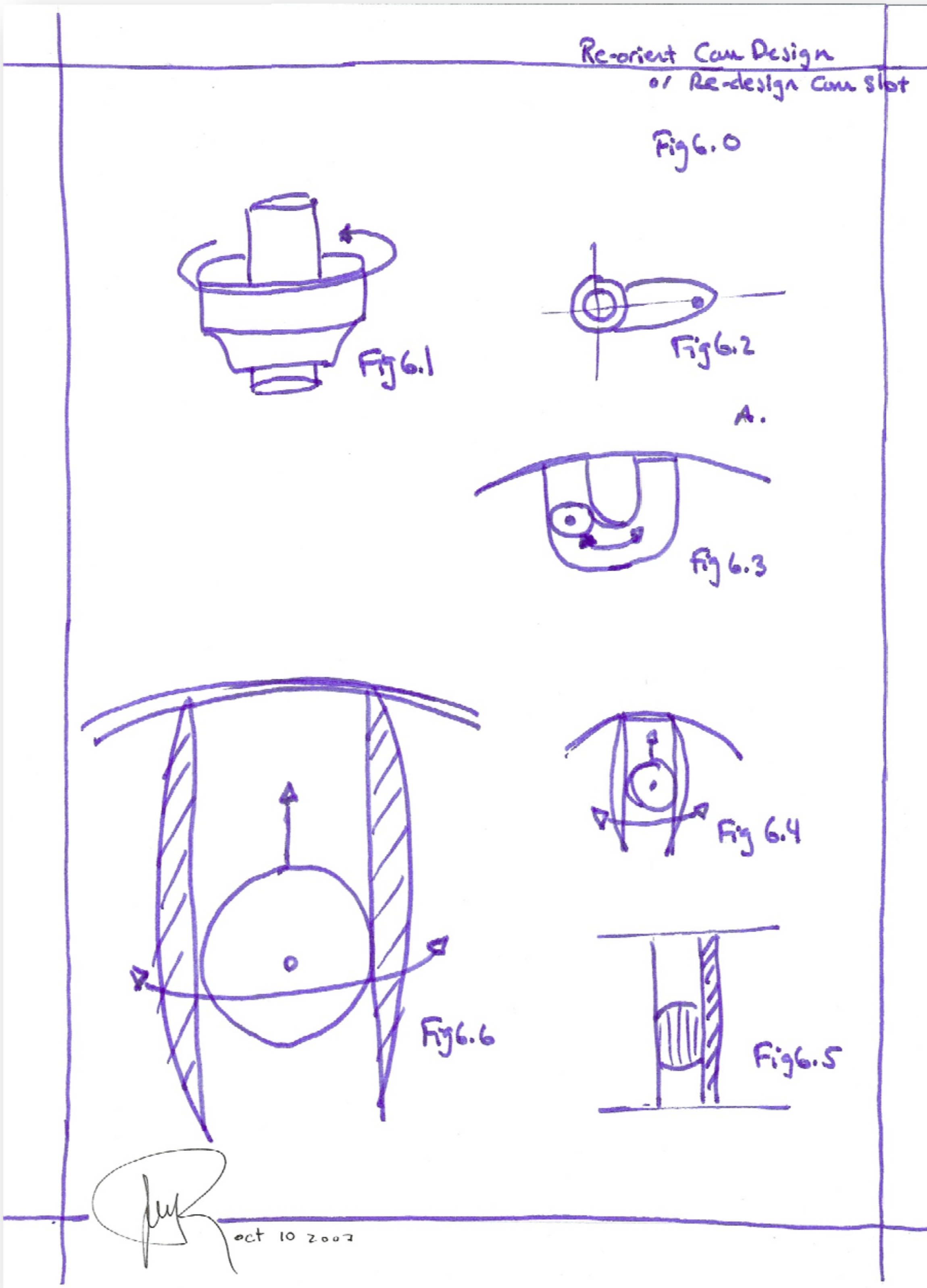

Gear/Track
Fig 1.0
Fig 1.1
Fig 1.2
Fig 1.3
Fig 1.4
Fig 1.5
Oct 10 2007


Mechanical Cam
limit on Cam slot
Fig 2.0
Fig 2.1
Fig 2.2.
Fig 2.3
Fig 2.4
Oct 10 2007

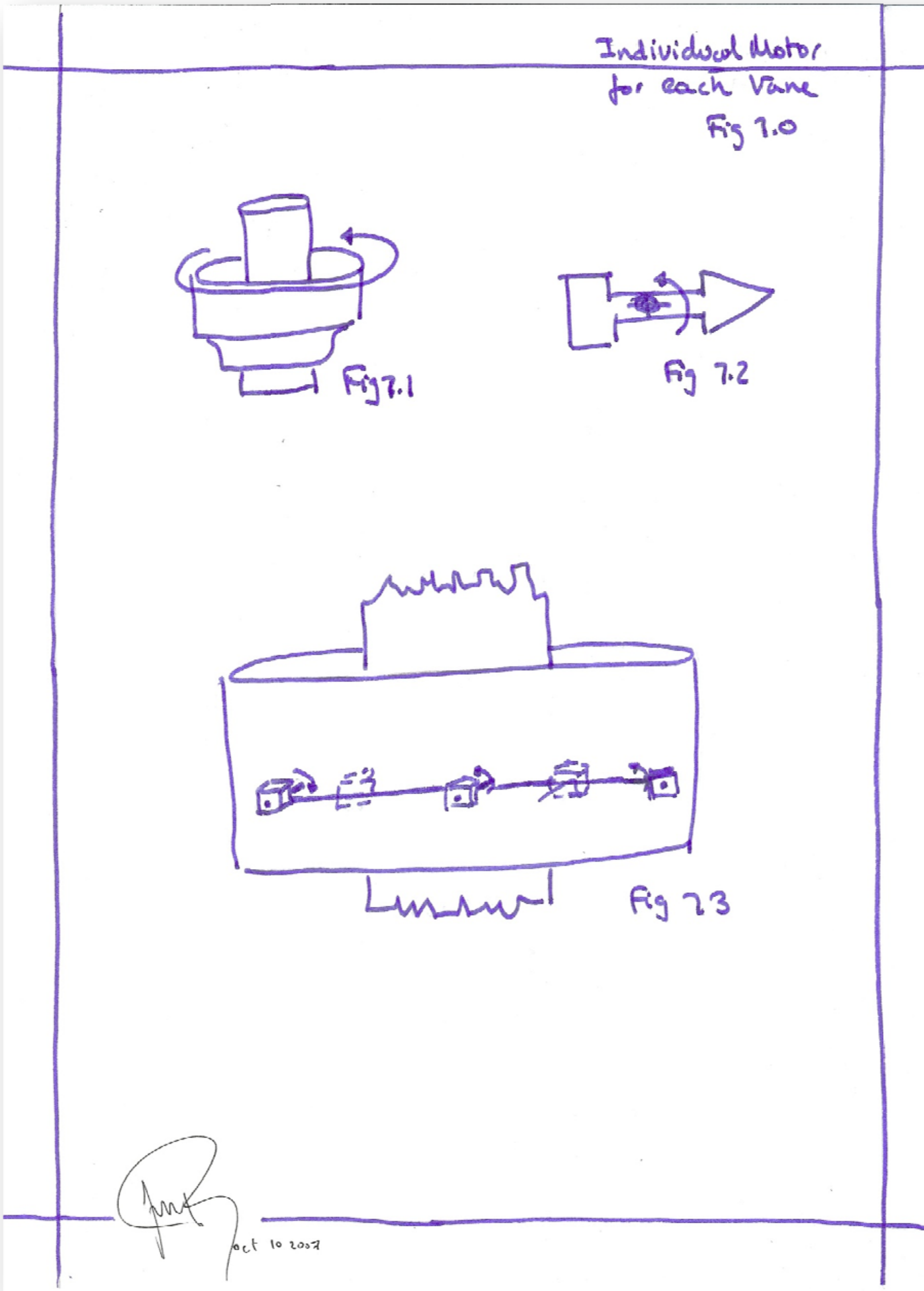

Single gear w/ mech. link
to each vane
Fig 3.1
Fig 3.2
Fig 3.3
Fig 3.4
Fig 3.5
Oct 10 2007


Coil/wire
Fig 4.0
Fig 4.1
Fig 4.2
A.
Fig 4.3
Fig 4.4
B.
Motor 1
Motor 2
Fig 4.5
Fig 4.6
oct 10 2007


Pulley B Belt
Fig 5.0
A.
Fig 5.2
Fig 5.1
B.
Fig 5.3
creates tension
Fig 5.3
oct 10 2007



Re-orient Cam Design
or Re-design Cam slot
Fig 6.0
Fig 6.1
Fig 6.2
A.
Fig 6.3
Fig 6.4
Fig 6.6
Fig 6.5
oct 10 2007


Individual Motor
for each Vane
Fig 7.0
Fig 7.1
Fig 7.2
Fig 7.3
oct 10 2007

4.2 - Design Selection Process:

Concept Selection consisted of two parts. The first part has a lot of involvement with sponsor influence and their desired expectation of the project. On October 9th, 2007, we had a meeting with our sponsor, Danfoss Turbocor and presented our first concept ideas. We had a round table discussion about each of the designs possible consideration and in doing so we gained further insight into the project as a whole. We learned that they are no longer having a locking problem with the current mechanism as it was resolved with the introduction of mechanical stops. These stops prevented the motion of the cam housing just before the toggle position. We also learned that they wanted us to pursue a completely new concept, in the attempt of increasing the range of motion for the IGV's.

With this new knowledge we were able to eliminate the idea of the cam stoppers immediately. The main reason for this decision was that Turbocor had already introduced mechanical stoppers into the current design. We both agreed that even though the stoppers in the current mechanism are different from our design, they both follow the same idea. The stoppers will not increase the range of motion for the IGV's; however, they do prevent the IGV's from locking. With this said, Turbocor expressed that they want to move away from the cam idea as a whole. This also eliminated the reorientation of cam slots concept. This would only increase the range of motion of the cams toggle position allowing the IGV's to reach the 110° without any problems.

The design of the individual motors was not particularly liked due to the fact that the motors are one of the most costly parts. And even though the assembly of the idea would be fairly easy, the computer programming and controls for such a design are very complex and integrate. This is because all the IGV's must open and close at the same time and be at the same corresponding angle throughout their operation. If this is not the case the IGV's would cause turbulence in the flow of the working fluid which could possibly damage the rest of the compressor system.

The gear drive with mechanical linkage was then discussed. The idea is very clean; however, the concept consisted of too many moving parts. The increase in parts would make assembly more difficult and could greatly decrease the reliability of the mechanism. With this in mind, the total cost of the design will be higher just due to labor and maintenance expenses. Tolerances would also be an issue with this idea because of the extra moving parts. Thus this idea was dismissed as well.

The wire and coil idea was thought of in depth. The problem with this concept is the fine control of the IGV's. Since the wire can only rotate the IGV's in one direction the assistance of springs would be needed to return IGV's to their original starting positions. The other problem would be the use of a tension coiler. This just added a lot issues concerning the working fluid and the reliability of the mechanism. Danfoss Turbocor thought this concept was just too complex for such a simple task.

The ideas that were thought of the most highly by Danfoss Turbocor are the gear drive with independent gears and the cable/ pulley drive. The gear drive is of high interest because of the simplicity of the concept and the full range of motion that would be generated. The down side of the concept was the assembly and the extra cost for the gears themselves. In order to get each IGV to operate simultaneously and be at the correct angle they must be positioned correctly at their starting point. This is where the assemble conflict arises. Also, with the new technology of computer aided CNC machining the cost to make gears has decreased when compared to conventional methods. However, in order to limit the amount of backlash for this design, the gears must have a high modulus with very fine teeth. This increases the cost of machining and the total cost of the system.

The cable/ pulley drive was also thought of in high interest. This design would help significantly in lowering the tolerance issue of the current mechanism. It also has the ability to extend the range of motion of the IGV's to any desired position up to 360° which could potentially increase the efficiency of the compressor. Turbocor did inform us that other companies had in fact already produced similar designs but failed to incorporate a method for tensioning the cable. They advised us to do some patent searches and if possible come up with a final design that would allow the cable to be adjusted either manually or automatically. The problem with this is that the cost of assembly and the materials that would be used is exceptionally high. However, Turbocor did like the concept of an already proven idea that can work that just needs to be improved upon. The term they used is that it would be the best "break through" of all the ideas that were proposed.

4.3 - Design Criteria:

As seen in the design matrix each of the proposed ideas were evaluated numerically using eight important aspects of each mechanism. These criteria were determined by both the sponsor and the design group and are as follows:

- **Reliability**

  This is a measure of how long a design is expected to operate while routinely producing the same desired results. The more the complex a design is, the higher chance its operating performance will change. Each design's reliability is judged by the ability to implement a mechanical stop such that the motor does not over shoot when returning the vanes to their closed position. Another part of a design's performance is affected by the tolerances allowed for the mechanism's parts. Too much tolerance and backlash can occur, giving slack in the system.

- **Manufacturability**

  The type of materials and the tolerances chosen for the individual parts must be within reason when considering the manufacturing cost. There designs must also have a minimal complexity to ensure that the machining process does not require any special methods and is not extremely difficult. Simply put, the Manufacturability of a design is a rating that describes the intricacy and in turn the cost associated with the fabrication of parts.

- **Ease of assembly**

The new designs are also ranked on their ability to be put together using a simple step by step process. This should be able to be completed by a single trained technician within a reasonable amount of time. The easier the design is to assemble, the more money will be saved in the total cost.

- **Life expectancy**

Each of the new designs were judged by how long they were expected to operate until wear on the mechanical parts would cause failure. Ideally the best design would have the longest operational life possible. This is done by reducing the internal forces that are produce through the moving parts and will eventually reduce the cost of maintenance on the IGV control system. The number of moving parts was also considered when evaluating the life expectancy of a design. The fewer parts a design has, the lower the probability of failure.

- **Backwards compatibility**

Turbocor has made it imperative that the new design uses the original computer software that controls the vanes' position. The sponsor has specified that the intermediate positions of the IGV's between the closed, 0°, and open, 110°, positions do not matter as long as they are at 110° when the motor completes 10,000 steps of rotation. How well a design can do this is describes how backwards compatible it will be.

- **Operational performance**

  This corresponds to how fast and accurate the new designs complete the desired task. Typically designs will need to have both of these attributes. However, since the speed of the IGV control system is not as important as its accuracy, operational performance ratings of the considered designs are based on how consistent they are in obtaining the 110° to 10,000 step ratios.

- **Simplicity**

  The ideal design should be able to be manufactured and assembled with ease. This corresponds with how many different parts the design requires, how complex each part is to machine and how long it takes to produce a finished product. This aspect of the design is rating of effectiveness of the design. The ideal design would contain the least amount of parts required to operate.

- **Total cost**

  One of the most important aspects of the designs was that of the total cost. This cost accounts for the price of the mechanism's parts as well as the labor involved in the assembly and up keep of the product. The total cost value for each design was weighed against these criteria.

These criteria reflect how the new designs will perform and how economically feasible they are when compared to each other and the requirements of the sponsor. Although some of the criteria are more important than others they were all given an equal value between zero and ten with ten being the highest possible score. This was done to simplify the selection process.

4.4 - Decision Matrix:

The next process of selection begins with the generation of a basic concept design matrix. This allowed us to compare the proposed ideas against each other in order to determine which one would be the most desirable selection for the intended operation of the IGV's. However, with the newly revised understanding of the term "backwards compatibility" along with the expectations of a completely new control design by Turbocor, we were able to immediately eliminate the cam stops, cam slot reorientation, and the individual motors designs. Although these designs were not eligible for further consideration they were still included in the design matrix in order to verify their elimination.

The design matrix can be seen in table 4.1, below and includes the most important engineering criteria that a new design should encompass as determined by the sponsor and the design group. The designs were then evaluated and allotted a rating for each of the criterion based on a group decision. After all the designs were analyzed, their criteria values were summed and the two ideas with the highest score were selected for the semi-final designs while the rest were denied but not forgotten completely. The two final selections were then transformed from hand drawings of ideas to basic 3-D representations with Pro-E models. Afterwards, we followed up with another meeting with Turbocor so that they could fully understand what each design was doing, how it was to be assembled, and to determine if there were any changes that needed to be made.

Table 4.1 – Concept Design Matrix									
	Reliability	**Manufacturability**	**Assembly**	**Cost**	**Life**	**Compatibility**	**Operational Performance**	**Simplicity**	**Total**
Gear Drive w/ Individual Gears	10	6	7	6	9	9	10	6	**63**
Cam Stops	8	8	9	9	7	10	1	8	**60**
Gear Drive w/ Mechanical Linkage	5	3	4	7	6	7	4	3	**39**
Wire and Springs concept	4	3	4	6	6	5	4	3	**35**
Cable/ Pulley Drive	9	6	8	8	8	9	10	7	**65**
Cam Slot Reorientation	6	5	6	5	8	4	5	6	**45**
Individual Motors	4	4	3	1	5	7	9	0	**33**

4.5 - Final Design Description:

The most important part of the final design is the inlet throat profile. It is crucial to the project that this does not change from the original mechanism as it would affect the performance of the main compressor. However, the outer housing of the throat profile has been altered to accommodate seven identical idler blocks that will assists in the operation of the IGV's. This can be seen in figure 9.2. from appendix V. Another important part that did not change from the original design is the stepper motor and attached gear reduction box. This will assist us in making the new mechanism completely backwards compatible using the original computer software as defined by Turbocor. As mentioned earlier in the report the motor moves through 10,000 steps of rotation to get 110° of motion through the IGV's. We achieve these same angles with the use of an additional gear reduction system that is mounted between the motors' gear box and the drive pulley of the control system.

The main IGV control mechanism consists of the following:

1. The throat housing
2. Seven IGV's with screw inserts
3. Seven IGV screws and accompanying loc washers
4. A drive IGV pulley with permanently attached reduction gear
5. Six standard IGV pulleys
6. Five stationary idler blocks which consists of:
    a. Idler block
    b. Idler pulley

c. 3/8” shoulder screw

d. 2 – 8/32 button cap screws

7. Two identical tension systems that consist of:

    a. Idler block

    b. Idler pulley

    c. 3/8” shoulder screw

    d. 2 – ¾” shoulder screws

    e. 2 – 6lb per inch springs ¾” long

8. 7 x 7 stainless steel non-coated cable 36” long

The standard and drive IGV pulleys replace the cams in the original mechanism but retain the original cams’ shaft. These shafts allow for the alignment of the pulleys with the IGV’s and can be seen in figure 9.4 from appendix V. The pulleys are what essentially drive or turn the IGV’s with the use of the cable. They have a 17mm diameter and are directly connected to the IGV’s via the IGV screws and loc washers figure 9.7 and figure 9.5 in appendix V. The idler blocks are located in between each IGV pulley along the outer housing of the throat profile. Mounted to each block with the use of the 3/8 shoulder screw is a 7.5mm diameter idler pulley as seen in figure 9.10 from appendix V. These pulleys are also mounted perpendicular to the IGV pulleys as to align the cable so that it is parallel with the outermost diameter of the IGV pulleys. This is done so that the cable may be wrapped completely around the IGV pulley in order to prevent slippage. Also, using this method the tensional forces from the cable pulling in each direction from the center of the IGV pulley will actually "cancel out". This allowed the IGV and the pulley shafts to have a minimal amount of wear and be made of the original self lubricating material with an illusion of an external force being applied with a magnitude of zero.

The two tensioning systems used are located directly across from the IGV drive pulley and are separated by one single stationary idler block. Using two tension systems allows the cable to be adjusted by a minimal amount verses just having one system that would require a lot of adjustment. As seen in figure 9.9 from appendix V, the tension blocks are the exact same as the idler block. This allows any of the idler blocks to be used in the tension system. The position of the tensioning system is also flexible as each mounting surface for the idler blocks are identical as well. The tensioning blocks are secured to the throat housing using two 3/4" long 3/16" diameter shoulder screws. There is also two ¾" long 6lb/in springs inserted behind the tension blocks to make them automatically adjust. This decreases or increases the overall diameter of the cable path and thus loosening or tightening the cable around the pulleys respectively.

The rest of the control system consists of the main outer housing, the drive motor with attached 10:1 reduction box and an external gear reduction system. The gear reduction system is made of a pre assembled gear box that utilizes a worm gear type arrangement to achieve a 12:1 reduction ratio. The gear box is then coupled with an external spur gear reduction pair that has a 5.6:1 reduction ratio giving a combined total of 672 revolutions of the motor shaft to 1 revolution of the IGV shaft. The external gears are made of Ryton® BR42B, the same material as the pulleys, to utilize its self lubricating properties. The complete reduction system allows the use of the original stepper motor and attached gear box which transmits 111.6° of motion to the IGV's. With the assistance of mechanical stops to prevent this slight overshooting of the desired angle the control system is completely backwards compatible.

Every part of the control system previously mentioned is attached to an aluminum alloy main housing using some sort of screw or bolt. This housing had to be modified from the original one to allow space for the new gear reduction system and larger throat housing assembly while still being able to be attached to the compressor. This was done by removing unnecessary material from the inside of the housing and adding some material to the outer surface in order to maintain the required thickness. The mount for the reduction system and the stepper motor was also added to the inside of the housing. With this complete it will now make changing a defective IGV control as simple as changing the entire control mechanism.

5.1- Gear Ratio Calculation:

To achieve backwards compatibility, the design must accommodate the compressor's current software that uses a specific motor. The motor is a series 42M048C stepper motor with gear train (type R). This motor must go through 10,000 steps to open the IGV to their fully open position of 110°. The motor's shaft rotates 7.5° per step of rotation. Thus the motor must turn 75,000° to the 110° of the IGV. The gear ratio is then calculated to be:

$$\frac{75000}{110} = \frac{7500}{11} = 681.818$$

To accomplish such a large gear ratio a worm gear box will be used. The gear box has a ratio of 120:1; this leaves a ratio of 375:66 to be made up.

The remaining ratio will be accomplished through a set of spur gears. First a module was assumed and the diameter of the shaft out of the worm gear box was known. With these values the number of teeth on the gear out of the worm gear box was determined; this is denoted by the subscript "shaft". With a whole number of teeth and the diameter a new module was generated. These calculations are shown below.

$$d_{shaft} = 5\text{mm} \qquad \text{module} = .3 = \frac{d_{shaft}}{N_{shaft}} \qquad N_{shaft} = 16.666$$

$$N_{Nshaft} := 17 \qquad \text{module}_N = .294117647$$

Then using the makeup gear ratio and the number of teeth on the shaft coming out off the gear box, number of teeth for the pulley gear was found. This is then rounded to the lowest whole number. This will prevent the IGV from not turning enough. Using the number of teeth on the pulley gear and the module found above, a diameter pulley gear can be found, as seen below.

$$\frac{N_{gear}}{N_{shaft}} = \frac{375}{66} \qquad N_{gear} = 96 \qquad d_{gear} = 28.235\,mm$$

With the number of teeth decided on each gear, the real gear ratio can be determined.

$$\frac{N_{gear}}{N_{shaft}} = \frac{96}{17} \qquad Ratio = \frac{N_{gear}}{N_{shaft}} \cdot Worm = \frac{96}{17} \cdot \frac{120}{1} = \frac{11520}{17}$$

For 10,000 steps or 75,000° of motor rotation, the vanes were calculated to rotate 110.677°. This value was a higher by little more than a half a degree than the required turning angle for the vane. However, IGV’s will be unable to rotate past 110° due to the implementation of a mechanical stop. This would prevent them from over turning and creating problems.

5.2 - Cable Tension Calculation:

The cable is what will allow all the vanes to turn in sync. This is achieved by use of the friction between the cable and pulley. The pulleys are made of Ryton® BR42B. This material is a polyphenylene sulfide compound with known mechanical properties. The properties can be seen in Appendix VII. Ryton has a static friction coefficient of 0.32 with steel.

When making the cable tension calculations, first the absolute maximum force on the blades was found. To do this the flow through the throat must be known. Using an industry standard the flow can be considered at Mach 0.2. Once the Mach number and the acoustic velocity of refrigerant were known, the velocity of the flow was solved for. The velocity in combination with the density of refrigerant at 0° C and the area of the throat then was used to yield a mass flow. This flow rate was then used for the calculation of the force. This force is then evaluated at the tip IGV furthest from the center axis. This creates a moment on the IGV and pulley. This moment can then be considered as a tangential force on its edge. These calculations are shown below.

$$F_{fluid} := c_i \cdot m_{flow} \qquad x := \left(.75 \cdot 11.4 + \frac{11.4}{2}\right) mm \qquad F_{fluid} = 0.04 \cdot kN$$

$$M_{oment} := F_{fluid} \cdot x \qquad M_{oment} = 0.572 \cdot N \cdot m$$

$$D := 17mm \qquad r := \frac{D}{2} \qquad F_{pulley} := \frac{M_{oment}}{r} \qquad F_{pulley} = 0.067 \cdot kN$$

With the torque know, the tension of the cable was calculated. The tension in the cable is equal to the force on the pulley divided by the friction coefficient. The process is shown below.

$$T_{ension} := \frac{F_{pulley}}{\mu} \qquad T_{ension} = 0.21 \cdot kN$$

The tension in the cable will be provided by idler pulleys. Three of these pulleys will be attached to a screw that will allow for the pulley to side in and out to increase and decrease the tension in the cable. (The full calculation can be seen in Appendix IV.)

5.3 – Cable Selection:

During the cable analysis, a big concern is the material selection, which will allow for the best control results of the IGV control mechanism. It's important to remember that this mechanism will work under an environment of low temperatures due to the fact that the working fluid of the compressor will either be a refrigerant, possibly R-22 or R-134a.

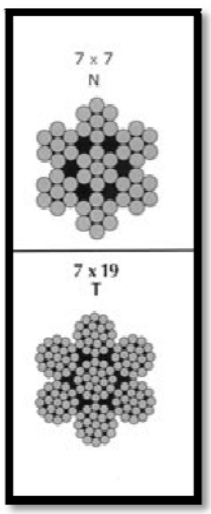
Material	Compatibility
Metals	
General Behavior : Slight risk of corrosion in presence of moisture.	
Aluminium	Satisfactory
Brass	Satisfactory
Copper	Satisfactory
Ferritic Steels (e.g. Carbon steels)	Satisfactory
Stainless Steel	Satisfactory
Plastics	
Polytetrafluoroethylene (PTFE)	Acceptable **but strong rate** of permeation.
Polychlorotrifluoroethylene (PCTFE)	Acceptable **but important swelling**.
Vinylidene polyfluoride (PVDF) (KYNAR™)	no data
Polyamide (PA) (NYLON™)	Satisfactory
Polypropylene (PP)	Acceptable **but strong rate** of permeation.
Elastomers	
Buthyl (isobutene - isoprene) rubber (IIR)	Acceptable **but important swelling**.
Nitrile rubber (NBR)	Non recommended, significant swelling.
Chloroprene (CR)	Acceptable **but important swelling**.
Chlorofluorocarbons (FKM) (VITON™)	Non recommended, significant swelling.
Silicon (Q)	Non recommended, significant swelling.
Ethylene - Propylene (EPDM)	Satisfactory
Lubricants	
Hydrocarbon based lubricant	Non recommended, significant loss of mass by extra chemical reaction.
Fluorocarbon based lubricant	Non recommended, significant loss of mass by extra chemical reaction.

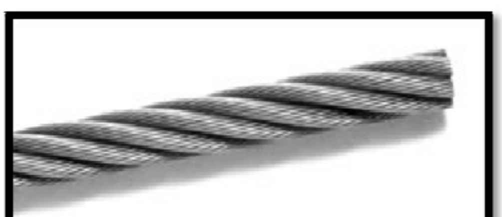
There are various materials available for the selection of the cable. The selected material is Stainless Steel because of its mechanical properties. The properties can be seen in the figure below.

Grade	Tensile Str (MPa) min	Yield Str 0.2% Proof (MPa) min	Elong (% in 50mm) min	Hardness Rockwell B (HR B) max	Hardness Brinell (HB) max
316	515	205	40	95	217
316L	485	170	40	95	217
316H	515	205	40	95	217

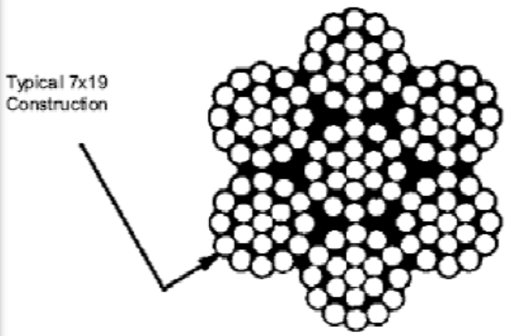
Grade	Density (kg/m³)	Elastic Modulus (GPa)	Mean Co-eff of Thermal Expansion (µm/m/°C) 0-100°C	Mean Co-eff of Thermal Expansion (µm/m/°C) 0-315°C	Mean Co-eff of Thermal Expansion (µm/m/°C) 0-538°C	Thermal Conductivity (W/m.K) At 100°C	Thermal Conductivity (W/m.K) At 500°C	Specific Heat 0-100°C (J/kg.K)	Elec Resistivity (nΩ.m)
316/L/H	8000	193	15.9	16.2	17.5	16.3	21.5	500	740

The type of wire to be used for the selection is a non-nylon coated cable as seen below. This cable is provided by Small Parts Inc. The interesting benefit about working with Small Parts Inc. is that if needed a costume coating may be made to fit any specifications or constrains. The data pertaining to the two types of breading and set up pattern of the wire, as seen to the left, tell much about its strength and durability. Also in the chart below, the information on the wires breaking strength and construction can be seen; which works as a function of the OD. For this specific system, the wire section would be MCX-024.





STYLE	MATERIALS	NOTE
PLAIN, UNJACKETED CABLE	302 STAINLESS STEEL	**COMPLIES WITH MILITARY SPECIFICATION**



STOCK NO.	CABLE CONST.	CABLE DIA.	CABLE LENGTH (FT.)	TENSILE STRENGTH (MIN. LBS.)
AB018-25FT	7X7	1/64	25	40
AB018-50FT	7X7	1/64	50	40
AB018-100FT	7X7	1/64	100	40
AB032-25FT	7X7	1/32	25	120
AB032-50FT	7X7	1/32	50	120
AB032-100FT	7X7	1/32	100	120
AB033-25FT	7X19	1/32	25	120
AB033-50FT	7X19	1/32	50	120
AB033-100FT	7X19	1/32	100	120
AB046-25FT	7X7	3/64	25	270
AB046-50FT	7X7	3/64	50	270
AB046-100FT	7X7	3/64	100	270
AB062-25FT	7X19	1/16	25	480
AB062-50FT	7X19	1/16	50	480
AB062-100FT	7X19	1/16	100	480
AB078-25FT	7X19	5/64	50	560
AB078-50FT	7X19	5/64	50	560
AB078-100FT	7X19	5/64	100	560
AB092-25FT	7X19	3/32	25	960
AB092-50FT	7X19	3/32	50	960
AB092-100FT	7X19	3/32	100	960
AB125-25FT	7X19	1/8	25	1760
AB125-50FT	7X19	1/8	50	1760
AB125-100FT	7X19	1/8	100	1760
AB156-25FT	7X19	5/32	25	2400
AB156-50FT	7X19	5/32	50	2400
AB156-100FT	7X19	5/32	100	2400

Stainless Steel Aircraft Cable is available in additional diameters and lengths.

5.4 - Cost Analysis:

When considering the cost of the new design, it would be an effective assessment to compare this cost to that of the original design. First, original cost must be considered. The original design of the IGV control apparatus consisted of many parts which cost $371.06 in parts. This value was calculated from the bill of materials given by the sponsor. This list containing pricing has been appended and is shown in appendix VI.

The new design consists of several different items from the original design. One item that was crucial to the new design was the worm gearbox. This item is manufactured by Berg Gear. Their product was considered to possess the necessary level of quality, that is, one with minimal backlash, to provide optimal performance of the new design. The Berg Gear worm gearbox cost $40.00. However, additional parts will be needed for this box to operate. These items are the input shaft and the output shaft with included gear; they will cost $1.55 and $2.25 respectively.

These shafts will be manufactured using a lathe and C&C machine.

Not all the parts from the original design needed to be replaced with complex parts like the gear box. It is estimated that the system's pulleys will cost about the same as the levers of the IGV's. Just like the levers, these pulleys will be injection molded. Also the brass cam rollers will not be needed. This could bring a slight reduction in overall cost. The connecting geared shaft was estimated to cost about the same as the output shaft from the worm gearbox as well.

The key items that will be removed from the original cost are the drive housing, ball bearings for disk drive housing, brass cam rollers, magnet, and the worm gear. These components cost a total of $69.81. Although this reduction in cost was considerable this decrease in cost was met with an increase by the cost of new parts.

The overall cost to assessment will ultimately include other costs associated with manufacturing, assembly time, and repair. As explained by the project's sponsors, the original design required low tolerances for the IGV mechanism to work otherwise the vanes would jam rendering the compressor inoperative. The new design also called for similar tolerances; however, an advantage arises when considering repair time and cost associated with this design. When the current mechanism failed, it was necessary to remove the throat and the housing which is quite costly. The new design would only need simpler repairs to make the IGV controller function. The new design also reduces repair costs because it is highly less likely to jam as the former cam mechanism did. This will save untold amounts over the life of the compressor. The bill of material can be seen in the table on the following page.

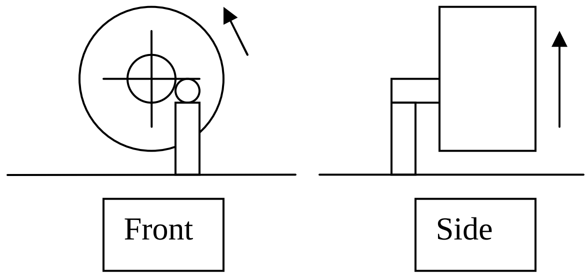
IGV Housing Assembly - TT-300 - Bill of Materials			
Component	Quantity	Cost per Unit	Cost per Assembly
Housing	1	$ 152.58	$ 152.58
Throat	1	$ 39.06	$ 39.06
Blade	7	$ 7.02	$ 49.14
IGV Pulley	7	$ 6.00	$ 42.00
Large Gear	1	$ 2.00	$ 2.00
Idler Pulley	7	$ 1.50	$ 10.50
Idler Block	7	$ 1.50	$ 10.50
3/8" Shoulder Screw	7	$ 0.30	$ 2.10
Tension Spring	4	$ 0.25	$ 1.00
Washer M3	11	$ 0.06	$ 0.66
Screw M3	20	$ 0.20	$ 4.00
Cable	1	$ 0.50	$ 0.50
Motor	1	$ 21.00	$ 21.00
Screw M5X	1	$ 0.02	$ 0.02
Pad-Anti Vibration	1	$ 0.18	$ 0.18
Hermetic Feed Thru	1	$ 17.22	$ 17.22
Retainer	1	$ 1.33	$ 1.33
Eye Bolt	1	$ 0.83	$ 0.83
Plug	1	$ 0.16	$ 0.16
Press/Temp Sensor	1	$ 35.36	$ 35.36
Screw M5X20	4	$ 0.34	$ 1.36
Small Gear	1	$ 1.25	$ 1.25
Small Gear Shaft	1	$ 0.30	$ 0.30
Gear Reduction Box	1	$ 15.00	$ 15.00
			$ 408.05

5.5 - Future Design Development:

Although the current prototype does operate and successfully ‘prove’ the design concept, there are a few details that should be worked out in future development. First, because of the oversight of the gear reduction assembly on the motor, the gear train must be reanalyzed. Also, it is necessary to provide a hard stop at the fully closed position of the IGV’s, and to provide some indication of the position of the vanes external to the compressor.

In order to address the gear ratio issue, there are a couple of viable options to consider. The simplest option would be to replace the 120:1 gearbox with a 4:1 gearbox that would fit in the same envelope. Although this would not require much of a design change, it could be costly to get a custom gearbox if an off the shelf option is not available. Another option would be to consider changing the gear reduction on the control motor along with changing the gearbox. The key aspect would be to define the gear reduction on the motor and the gear box in series to get the required 120:1 ratio. For example, if the motor could be ordered with a 10:1 ratio, then the current gearbox could be replaced with a 12.5:1 gearbox (off the shelf).

The following sketch shows the proposed addition of a mechanical stop at the fully closed position of the IGV’s. A stud would protrude from the outside face of one of the IGV pulleys and would meet a boss on the compressor housing when the vanes reach their closed position. This is a viable option because the desired motion of the vanes is only 110°.



A similar approach could also be used to provide an external indicator to show the position of the IGV's. Using a stud similar to the one shown above with a magnet protruding out near the wall of the housing, a ball bearing placed in a semi-circular slot would indicate the location of the vanes.

6.1 - Test Procedure:

Once the IGV control mechanism is assembled, the first step in verification is an initial functionality test. The purpose of this test is to ensure that the mechanism functions properly. The control motor must be powered with a DC power supply able to provide 12 volts to power the motor. When power is applied to the motor, all of the vanes should rotate about their respective axes. The vanes will also be driven to verify that their maximum turning angle is 110º.

After the initial functionality test, the performance of the mechanism must be analyzed. The performance of the mechanism will be measured by the accuracy of the vane angle given a certain input, and the mechanisms backlash which is translated to the vanes. To test the vane angle accuracy, the vane angle must be measured at multiple locations to ensure that with a given input the expected output is achieved based on the gear ratio. In order to meet the design specifications, the backlash in the mechanism must be minimized as well. The table 6.1 shows input steps to output angle of each vane. These values will be used to as a comparison to actual output angles.

Input Steps	Output Angle (degrees)
1000	11
2000	22
3000	33
4000	44
5000	55
6000	66
7000	77
8000	89
9000	100
10000	111

Table 6.1 - Input Steps vs. Output in Degrees

The final step is to test the reliability of the control mechanism. This test must duplicate the environment that the mechanism will see during actual operation in order to test its resistance to wear, material compatibility, and failure limit. Ideally, the mechanism should be installed into a compressor and operated through a ‘life’ of operation. The mechanism should not limit the life of the compressor. Because of the lack of oil or lubrication in the mechanism, it is important to verify that individual components of the mechanism do not fail due to pre-mature wear. In order to verify material compatibility, the mechanism should be operated in the compressor with R-22, or other refrigerant. After operation, it should be disassembled and inspected for swelling and corrosion. Inspection should not reveal any swelling or corrosion. It should also be verified that the failure limit of the mechanism is not less than the expected life of the compressor. To verify, the mechanism should be actuated from fully closed to fully open for the given compressor life.

This test will need to be performed at a later date. The prototype cannot be tested on an existing compressor; this is due to the pressure that is required for such a test. The prototype had material removed from the current pressure housing. This left a hole in the pressure housing, thus making pressure testing impossible.

6.2 - Proof of Concept:

After meeting with Danfoss Turbocor, it was made apparent that no extensive testing could be performed. This was due to the fact that Danfoss Turbocor follows specific procedures when it comes to testing new mechanisms. Before the mechanism can be tested over the life of a compressor, Danfoss Turbocor will have a panel convene and review the current design. The other problem preventing the full testing is a hole in the pressure housing. The system is pressurized; thus the hole would prevent any testing from occurring. However, the team was granted use of equipment to test if the prototype would function.

The devise was set to open the IGV is approximately 50 seconds. Instead, when the motor was driven, the IGV seemed stationary. It was first believed that the slippage was caused by slippage from the motor's shaft or possibly somewhere in the gear train. However, further observation proved that the gear train operated as expected. The IGV began to show signs of opening after close observation. It was discovered that the reason the vanes moved slower than anticipated was due to the fact that the motor used in the compressor was already geared down. The gear train was originally thought to be included in the motor specifications.

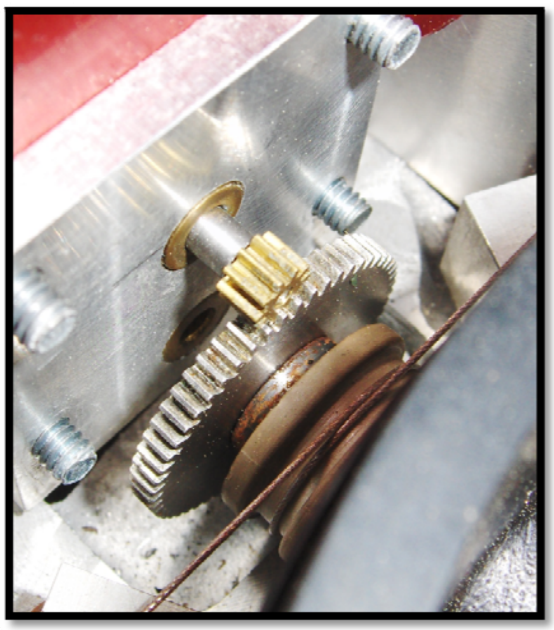


Figure 6.1 - Gear Train

The problem is very easy to remedy, by simply using a gear train that does not lower the turning ratio as much as the current one does.

Despite the unforeseeable obstacle of the over geared motor, the prototype operated smoothly. There was no slippage observed; this was mostly due to the fact that the self tensioning springs operated perfectly. The IGV opened fully and closed without any jamming or signs of problems. The system has no toggle position as the original design does; so the jamming should not exist. In short the applied concept was successful by virtue of its ability to open and close the IGV. The changing of the gearbox will provide the proper gear ratio and the design will be perfect.

Appendices

Appendix I - References:

- Small Parts, Inc. “CABLE – Stainless Steel

NYLON COATED CABLE - Stainless Steel Type 304” 2007. 28 Nov. 2007

http://www.smallparts.com/products/descriptions/mcx.cfm

- AIR LIQUIDE “Chlorodifluoromethane (R22)” 2007. 28 Nov. 2007

http://www.airliquide.com/en/home.html

- AIR LIQUIDE “Tetrafluoroethane-1,1,1,2 (R134A) “2007. 28 Nov. 2007

http://www.airliquide.com/en/home.html

- Continental Cable LLC. “Plastic Coated Cable” 2007. 28 Nov. 2007

http://www.gbgindustries.com/CoatedCable.htm

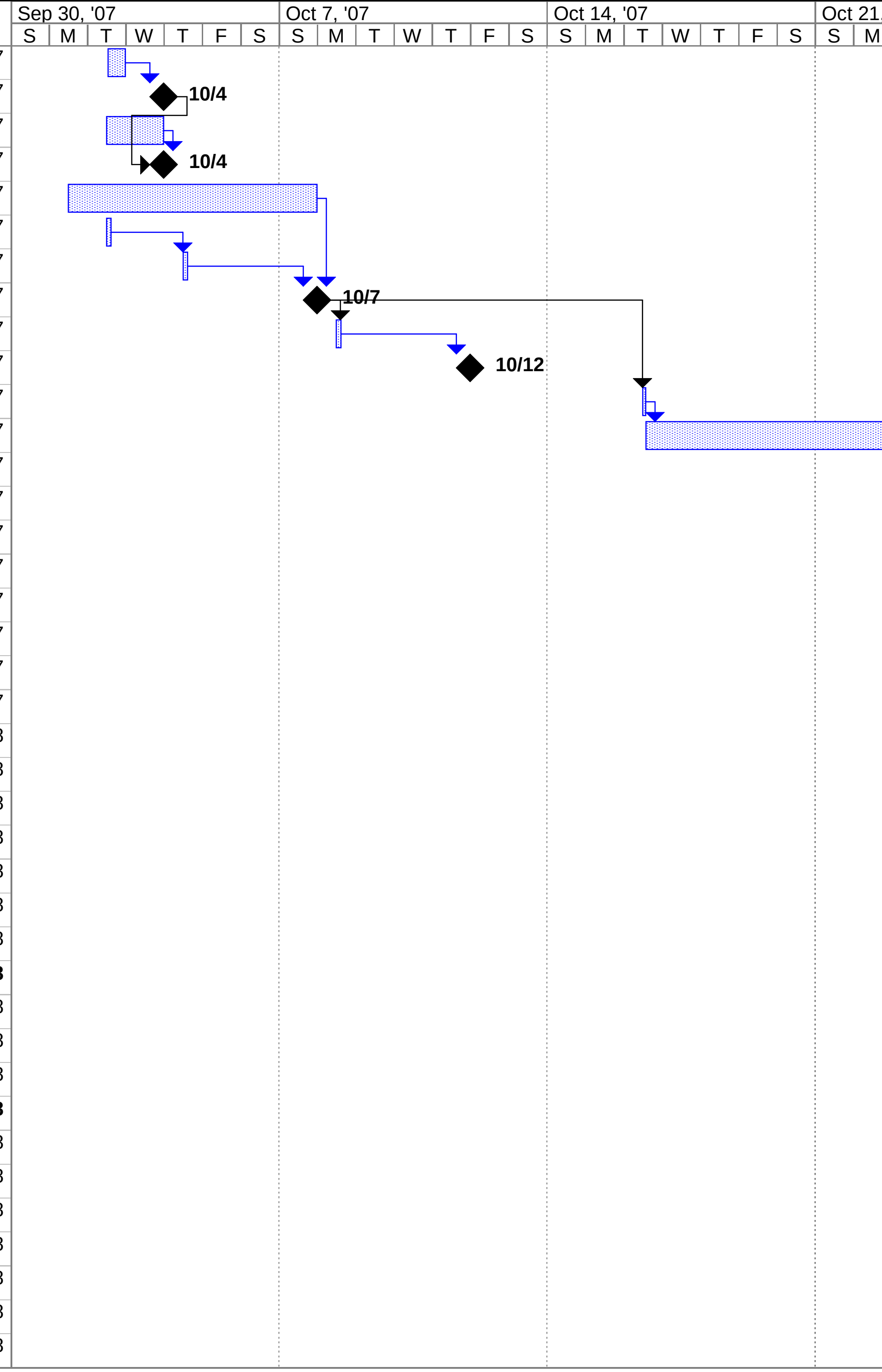
- Thermodynamic Properties of DuPont Freon 22 (R-22) Refrigerant. 2005.

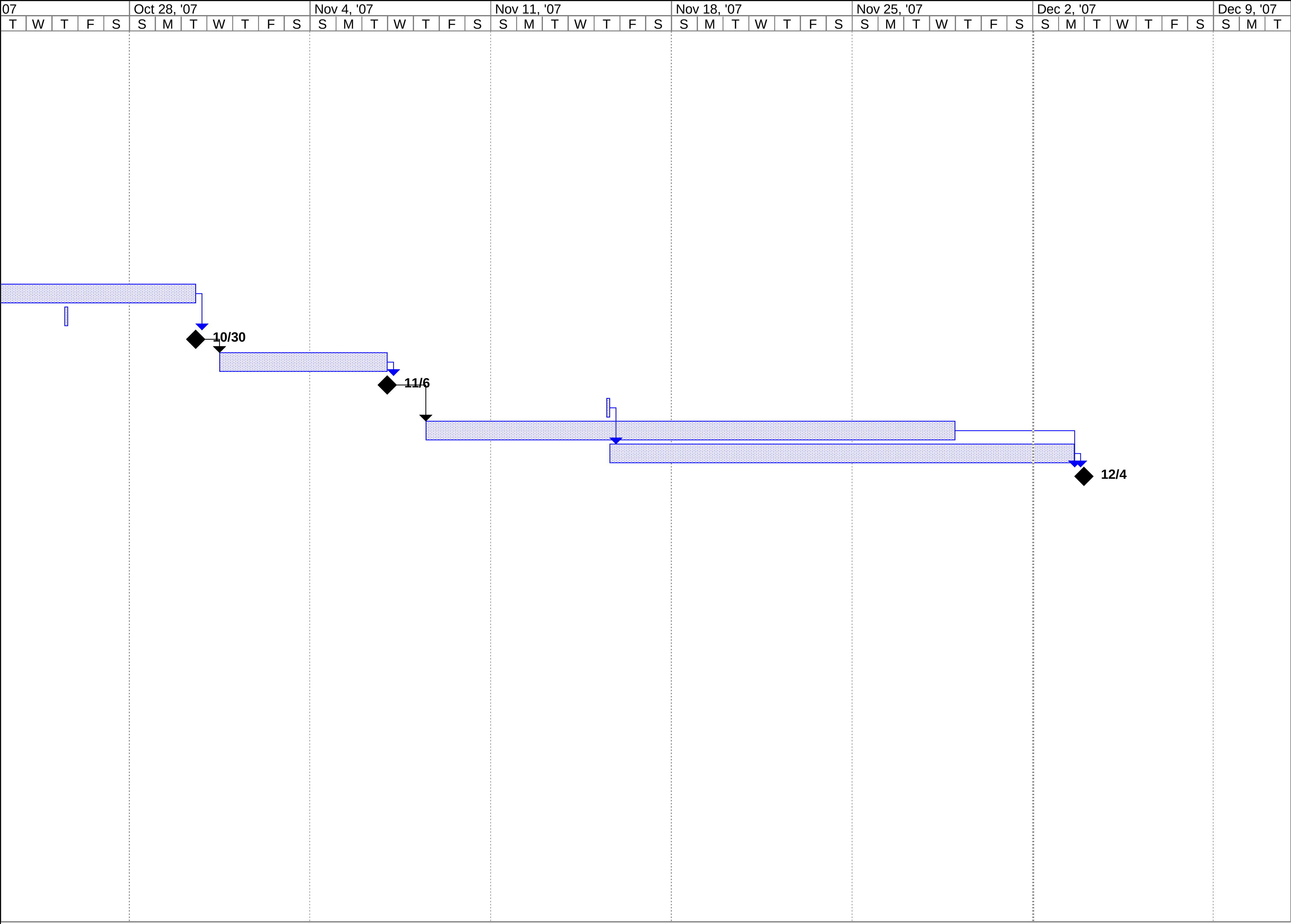
DuPont. http://refrigerants.dupont.com/Suva/en_US/pdf/k05736.pdf

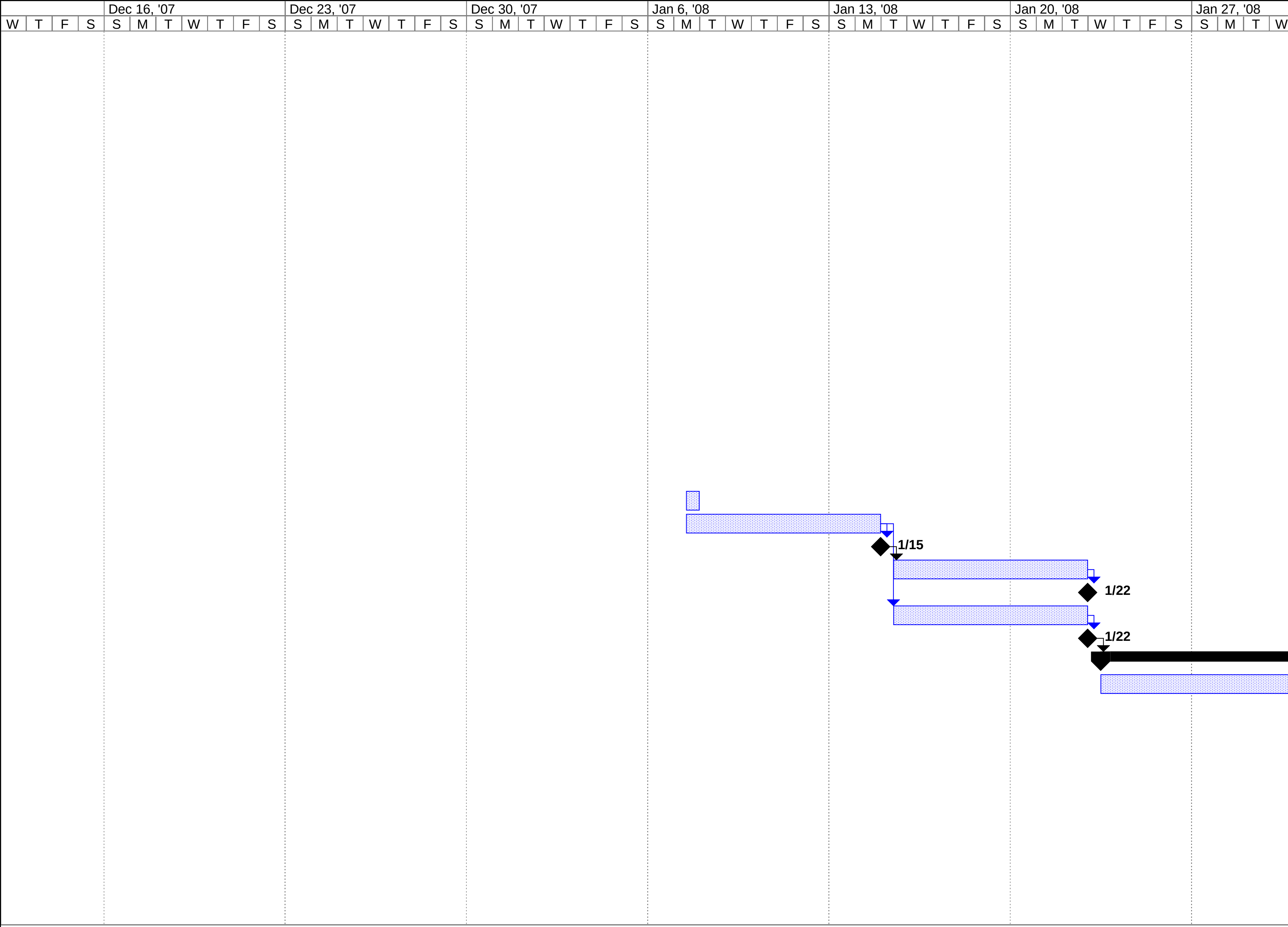
- W.M. Berg Gear, Gearbox and Cable.

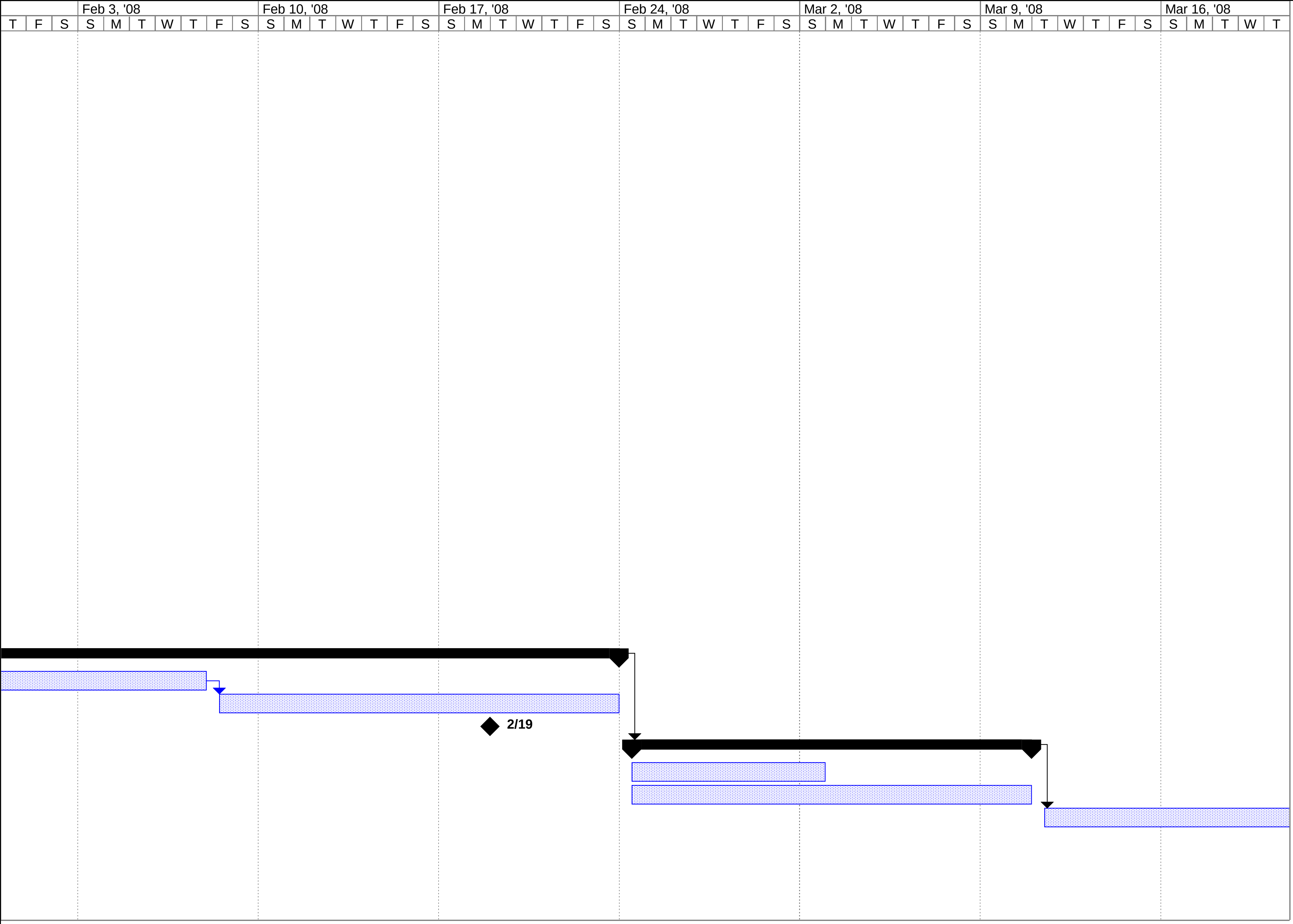
http://www.wmberg.com/

ID	Task Name	Duration	Start	Finish
2	Project Schedule	4 hrs	Tue 10/2/07	Tue 10/2/07
3	Schedule Due	0 days	Thu 10/4/07	Thu 10/4/07
4	Project Specifications	2 days	Tue 10/2/07	Wed 10/3/07
5	Specifications Due	0 days	Thu 10/4/07	Thu 10/4/07
6	Brain Storming	7 days	Mon 10/1/07	Sun 10/7/07
7	Group Meeting	3 hrs	Tue 10/2/07	Tue 10/2/07
8	Turbocor Meeting (Tentative)	3 hrs	Thu 10/4/07	Thu 10/4/07
9	Selection	0 days	Sun 10/7/07	Sun 10/7/07
10	Group Meeting	3 hrs	Mon 10/8/07	Mon 10/8/07
11	Conceptual Designs Due	0 days	Fri 10/12/07	Fri 10/12/07
12	Group Meeting (Divide Project)	2 hrs	Tue 10/16/07	Tue 10/16/07
13	Design of Selection	14 days	Tue 10/16/07	Tue 10/30/07
14	Group & Staff Meeting Progress Review	3 hrs	Thu 10/25/07	Thu 10/25/07
15	Pre-Interim Design	0 days	Tue 10/30/07	Tue 10/30/07
16	Work On Deliverable	7 days	Wed 10/31/07	Tue 11/6/07
17	Interim Design Due	0 days	Tue 11/6/07	Tue 11/6/07
18	Group Meeting	3 hrs?	Thu 11/15/07	Thu 11/15/07
19	Final Design	21 days	Thu 11/8/07	Wed 11/28/07
20	Final Design Pack	18 days	Thu 11/15/07	Mon 12/3/07
21	Final Design Pack & Spring Proposal	0 days	Tue 12/4/07	Tue 12/4/07
22	First Spring meeting to Regroup	1 day?	Mon 1/7/08	Mon 1/7/08
36	Finalize Bill of Material	1.14 wks	Mon 1/7/08	Mon 1/14/08
37	Scope Restatement and Project plan Due	0 days	Tue 1/15/08	Tue 1/15/08
38	Complete Purchase Order	1.14 wks	Tue 1/15/08	Tue 1/22/08
39	Deliver Purchase Order to FSU Purchasir	0 days	Tue 1/22/08	Tue 1/22/08
40	Organize Assembly Procedure Plan	1.14 wks	Tue 1/15/08	Tue 1/22/08
41	Progress Presentation	0 days	Tue 1/22/08	Tue 1/22/08
42	**Fabrication and Assembly**	**32 days**	**Wed 1/23/08**	**Sat 2/23/08**
43	Machining/Fabrication	2.29 wks	Wed 1/23/08	Thu 2/7/08
44	Assembly	2.29 wks	Fri 2/8/08	Sat 2/23/08
45	Mid-point Review Presentations	0 days	Tue 2/19/08	Tue 2/19/08
46	**Initial Test/Troubleshooting**	**16 days**	**Sun 2/24/08**	**Mon 3/10/08**
47	Initial Function Test	8 days	Sun 2/24/08	Sun 3/2/08
48	Troubleshooting	2.29 wks	Sun 2/24/08	Mon 3/10/08
49	In-plant testing @ Turbocor	2.29 wks	Tue 3/11/08	Wed 3/26/08
50	Final Project Review Presentation	0 days	Tue 3/25/08	Tue 3/25/08
51	Operations Manual	0 days	Tue 4/1/08	Tue 4/1/08
52	Webpage	0 days	Tue 4/1/08	Tue 4/1/08
53	Final Report	0 days	Tue 4/1/08	Tue 4/1/08




07
Oct 28, '07
Nov 4, '07
Nov 11, '07
Nov 18, '07
Nov 25, '07
Dec 2, '07
Dec 9, '07
T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T
10/30
11/6
12/4


Dec 16, '07
Dec 23, '07
Dec 30, '07
Jan 6, '08
Jan 13, '08
Jan 20, '08
Jan 27, '08
W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W
1/15
1/22
1/22


Feb 3, '08
Feb 10, '08
Feb 17, '08
Feb 24, '08
Mar 2, '08
Mar 9, '08
Mar 16, '08
T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T
2/19


Mar 23, '08
Mar 30, '08
Apr 6, '08
Apr 13, '08
Apr 20, '08
Apr 27, '08
May 4, '08
F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F S S M T W T F
3/25
4/1
4/1
4/1

Appendix III - Evolution of Scope:

As we have been working through the design process, the scope of what Turbocor desired has evolved. This evolution came as we, the design team, learned more and as we had more meetings with Turbocor. The meetings led to a deeper understanding of Turbocor's desires and needs. Most of the evolution was due to the understanding of what "backwards compatible" meant.

The scope started off simple and straight forward. The new design was to be backwards compatible, improve reliability, designed for manufacturing, and cost efficient. The understanding of backwards compatible was simple that the new design must fit in the existing housing and that the throat profile must remain the same. With this the design process started and the first scope was developed.

As we developed our concepts we had a meeting with Turbocor. We learned that one of Turbocor's desires is to completely get away from the cam design. The current design was a complex cam system. Want we understood that the current design had a problem with toggle if there was any overshoot from the motor. Thus we designed to fix the overshoot and to remove the cam design. This changed since the cam design was now obsolete. We had already developed our concepts, which now needed to evolve with our understanding.

We then selected a pair of designs that we felt best met the needs of Turbocor and fulfilled all the needs of the scope. With these designs we met with Turbocor again. This time we learned the biggest need of the scope. We were redefined as to what backwards compatible was.

The true meaning of what backwards compatible is for Turbocor involves the motor. The motor and programming that runs the motor must remain the same. This gave us our biggest challenge. The motor and programming remaining led to a gear ratio of 75000:110 with a minimal space in which to make the ratio up.

The evolution of our scope and project was now finished. We had already developed a scope, needs, specifications, and concepts prior to the finial evolution. The previous writings can be seen in the following.

Scope and Needs:

The designing of the Inlet Guide Vanes (IGV) mechanical system has many needs. The biggest is the reliability. The current system locks requires a complete replacement. The new system needs to be backward compatible and designed to reduce the tolerances. The cost also needs to be reduced or controlled. The new system also needs to prevent any backlash in the IGVs.

The idea behind the project is that in the actual IGV design which is composed of a cam system, IGV rotating blades, brown rotating housing, worm gear, 4 step motor and compressor intake, or throat, work together as one unit. The idea is that the motor drives the worm gear. The worm gear then rotates the brown housing. Inside the housing are slots in which the cam rollers follow. As the cam rollers follow the slots, the arm rotates and is directly attached to the IGV inside the throat.

Figure 1.1 – The Housing with the throat, IGV and cam arms attached. The Black part is the throat, the cam rollers can be seen in the slots of the housing, and the IGV are the triangular shaped pieces in the middle.

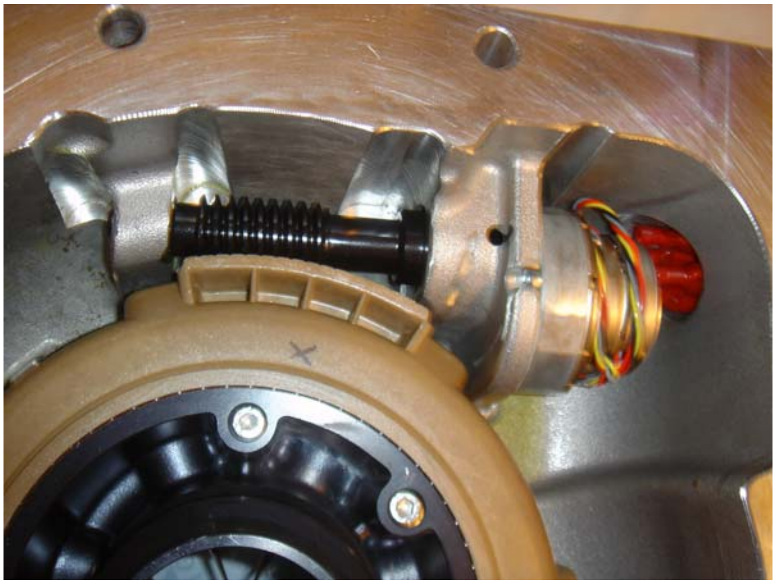


Figure 1.2 – The Motor with worm gear attached and housing in the compressor's body. The housing has small teeth on the outside, which collates with the worm gear.

As of now the problem with this design is that the blades open from 0 degrees starting position to a 110 degrees maximum opened position. The problem occurs that the natural toggle point for the system lies at the end on the maximum open position. This causes the cams to jam frequently. However the brown housing can not just be replaced. The tolerances are so small that the whole end has to be replaced.

The existing IGV assembly has a high failure rate and is extremely sensitive to manufacturing tolerance. The high failure rate is due to the sliding cams in the current mechanism jamming at their toggle positions, as seen in the figure 1.1 below. The motor and worm gear assembly is also awkward to assemble and provides very little tolerance.

The mission of our senior design group is to improve on the existing mechanism that operates the IGV for Danfoss Turbocor's compressors. The goal is to maintaining constraints designated by our sponsor while allowing ideas to flow and expand. The goal for the new design should be less costly to manufacture by implementing components that do not require very small tolerances to operate effectively. The most important goal is to increase the reliability of the mechanism that operates the IGV. Over all, our goal is to design a mechanism that is reliable, effective, and of lower cost to manufacture and assemble.

At the final closure of this project our group will have produced a design that both meet the needs of Danfoss Turbocor and their customers. As far as the current design of the IGV and the throat are not to be changed. The rest of the current housing design will either be altered or completely redesigned to produce the desired outcome of the Turbocor specifications for this particular model. This new design will be more reliable, easy to manufacture and install, have retrofit capabilities, while maintaining cost effectiveness.

Design Specifications:

Since there is a current design for the Inlet Guide Vanes control mechanism, the system specifications are well defined for the proposed redesign of the mechanism.

General

- The final design must be compatible to the current system. In order for this, the design must incorporate the current guide vanes and throat design and must fit in the bell housing.
    - Housing space is roughly the shape of a cylinder with a diameter of 294mm and a depth of 78mm.

Operation

- Mechanism must uniformly and smoothly (with no “jams” or “hang-up’s”)
- Angle of the IGV operation
    - 0° to 120°
        - At 0° the vanes are at a fully closed position perpendicular to the flow
        - At 90° the vanes are oriented parallel to the flow through the throat.
- In order to minimize turbulence due to vibration in the vanes, backlash must be minimized in the control mechanism. Backlash in the vanes must be held to the same or less than the current design.
    - Current backlash is approximately ± 5°

Motor

- The programming that operates the motor must remain the same and so must the motor.
  - The motor requires 10,000 steps at 7.5° per step to reach fully open at 110°.
  - To shut vanes the motor must run in reverse 11,000 steps.

Manufacturing and Assembly

- All materials used or defined in the final material list must be compatible with R22 refrigerant and other refrigerant and oils.
  - Aluminum Alloy of medium or high quality.
  - Chevron Phillips Ryton® BR42B Polyphenylene Sulfide Compound
    - Polyphenylene Sulfide, Thermoplastic, Polymer
    - Ryton® BR42B is a 40% fiberglass reinforced polyphenylene sulfide compound specially formulated to have a low coefficient of friction for use in low surface friction and wear applications. Has a wear rate of 0.001 mm/hr as per ASTM D3702.
    - Or higher grade material
- Design should open current tolerances.
  - 5-10% is the minimal opening of tolerances.
- Assembly Time – TBD
- Manufacturing Cost - TBD

Concept Generation:

To solve our problem we considered several different ideas to rotate the IGV. These concepts are explained below and shown in the attached drawings.

Concept 1: Gear Drive (Fig. 1 section 4.1)

This was the first concept considered. It involves removing the cams and replacing them with spur or bevel gears. These gears are then rotated in unison by a large ring gear. This idea does not further complicate the design because the gear would move in unison and would consist of about the same number of parts. This design could however present a problem with assembly because the gears and in turn the orientation of the fins would have to be specifically arranged so the mechanism could function. Another disadvantage could arise in the elevated cost due to the use of fine toothed gears to ensure minimal backlash. However, cost could be cut due the fact that less material would be needed to construct the IGV apparatus. Using gears also provides some freedom in the type and position of the motor. One of the greatest advantages to this potential design is the fact that it would give the IGV a full range of motion from 0 to 360 degrees with no toggle position or jamming. Overall, this was considered to be one of the soundest concepts.

Concepts 2: Cam Track Stops (Fig. 2 section 4.1)

This concept was the simplest of them all. Small stops would be inserted onto the cam tracks on the current disc drive. This would prevent the cams from moving into toggle by simply preventing them from ever reaching it. This design would be easy to implement and cheap to produce. However, this design would limit the range of rotation of the fins and would use the existing cam design. These disadvantages would prevent us from satisfying our sponsor’s requirements of extending rotation range and not using a cam design. The latter of which was made apparent after all the teams design ideas had been considered.

Concept 3: Gear Drive with Linkage (Fig. 3 section 4.1)

This design is very similar to the first concept the group had. The difference lies in the fact that instead of all the cams being replaced by gears, only one would be. This one gear would be in contact with a ring gear. When this gear would turn a linkage would be moved that was connected in series with other fins. This would cause the fins to rotate in unison. This design would increase the range of rotation. However, it would only give about 180 degrees of motion. Because of the use of more parts; this design further complicates the existing mechanism. If one linkage were to break due to vibration or fatigue, it would not only render the mechanism useless, but could be possibly detrimental to the compressor. A higher number of parts would raise the cost. Due to an increase of complexity and decrease in reliability this design idea was thought to be ineffective.

Concept 4: Wire and Spring (Fig. 4 section 4.1)

This design equips the vane ends with wires. These wires are then in turn affixed to a spool connected to a motor, when this motor rotates, so do the fins. When the spool is turned to release the wires, the vanes will move back to position with the use of springs. Although assembly may not be very difficult, this design adds a considerable amount of complexity. A considerable number of parts would be needed for this design to function. More parts would reduce the reliability and promises a higher possibility of part failure. Another disadvantage to this design is the fact that springs would be needed to establish tension in the opposite direction of the spool. If these opposing tensions aren't tight enough vibration could occur. A slight increase cost would be incurred due to the use of multiple cables. An alternate version of this design featured the use of 2 motors in opposite directions but wasn't considered feasible because of the higher complexity. While this design would provide a full 360 degrees of motion, its complexity made it somewhat ineffective.

Concept 5: Belt Drive (Fig. 5 section 4.1)

In this concept one belt is looped around the circular base of the IGVs. The belt would consist of a material that would not corrode in the refrigerant used in the compressor. A set of pulleys would be added to the exterior of the throat such that the belt could be wrapped around the vane base and then the pulley until completely encircling the throat. The belt would then be tensioned with an adjustable pulley. While this design adds some complexity, the adjustable tension device would allow relaxed tolerances which in turn would make assembly easier. The belt would cut down on any back lash, keeping vibration to a minimum. On site repair could be considerably easier since the belt would be easy to replace. Also a full 360 degree of motion would be achieved. A disadvantage to this design is the fact that a rubber or plastic belt may break down considerably often causing a need for more service of the compressor. There is however the possible use of a metal belt. Overall, this was considered one of our strongest designs.

Concept 6: Cam Track Reorientation (Fig. 6 section4.1)

This concept provided a very simple solution. If the cam slots themselves would be tilted or curved, the position required for the cam to move into toggle would be change which could change the range of rotation of the attached vane. This idea would raise the cost higher than current design, due to more difficult molding and machining of the cam slots. This design would be as difficult to assemble as the current design and would require similarly strict tolerances. While the new cam slots could make the range of rotation greater, it would still use cams, which was very discouraged by our sponsor. This design was seen as ineffective.

Concept 7: Individual Motors (Fig. 7 section 4.1)

This design consists of affixing a motor to each vane and the motors move the vanes independently. While they can be moved in unison, if one motor were to fail, the compressor would be inoperative. Using additional motors only further complicates the design, by requiring additional controls to be written. More motors also would also raise the cost. This design would be easy to assemble; however, it was considered a weak design because of the added complexity, cost and reduction in reliability.

Concept Selection:

Concept Selection consisted of two parts. The first part is sponsor influence. On October, 9th, 2007, we had a meeting with our sponsor, Danfoss Turbocor. In this meeting we presented our concepts. We then had a round table discussion about the all our concepts. During the process of our discussion, we gained further insight into the project as a whole. We learned that they are not having a locking problem at the current time. They want us to pursue a completely new concept, in the attempt of increasing the range of motion for the IGV.

With this new knowledge we were able to eliminate the idea of the stoppers. The stoppers will not increase the range of motion for the IGV. Danfoss Turbocor also expressed that they want to move away from the cam idea as a whole. This eliminated the reorientation of cam slots. The idea of the individual motors was not particularly liked. This is due to the fact that the motors are one of the most costly parts. Also the controls for such an idea are very complex and integrate.

The gear drive with mechanical linkage was then discussed. The idea is very clean; however, the concept consists of too many moving parts. The increase in parts decreases the reliability and increases the required tolerances. Thus this idea was dismissed. The wire and coil idea was thought of in depth. The problem with this concept is the fine control of the IGV. The other problem would be the use of a tension coiler. This would add a lot of complexity and issues concerning the working fluid. Danfoss Turbocor thought this concept was just too complex for such a simple task.

The ideas that were thought of the most highly by Danfoss Turbocor are the gear drive with independent gears and the belt drive. The gear drive is of high interest.

Danfoss Turbocor was interested, because of the simplicity of the concept and the full range of motion that would be generated. The down side of the concept was the assembly and cost. Gears are harder to assemble and are much more costly. The belt drive was also thought of in high interest. The belt drive would help significantly in lowering the tolerances. The problem would be the cost and materials that could be used.

The next process of selection was to generate a design matrix. The design matrix can be seen in table 8.1, below.

Table 8.1 - Concept Design Matrix

	Reliability	Manufacturability	Assembly	Cost	Life	Compatibility	Operational Performance	Simplicity	Total
Gear Drive w/ Individual Gears	10	6	7	6	9	7	9	7	**61**
Cam Stops	8	8	9	9	7	9	0	8	**58**
Gear Drive w/ Linkage	6	3	4	7	7	7	4	3	**41**
Wire and Springs	4	3	4	6	6	7	4	3	**37**
Belt Drive	8	7	8	8	7	7	7	8	**60**
Cam Reorientation	7	5	6	5	8	6	6	6	**49**
Individual Motors	4	4	5	1	5	6	8	1	**34**

With the use of the design matrix and sponsor influence, we have narrowed our concepts down to two. We will further expand on the concepts of the gear drive with independent gears and the belt drive.

Appendix IV – Cable Calculation

Using ASHRAE standards - Assuming R-22 Refrigerant

$D_{th} := 54.15\,mm$

$M = \frac{c_i}{a}$ $\quad a_{0deg} := 470\frac{ft}{s}$ $\quad M_i := .2$ $\quad c_i := a_{0deg} \cdot M_i$ $\quad c_i = 28.651\frac{m}{s}$

$r_{th} := \frac{D_{th}}{2}$ $\quad \rho_{0deg} := 21.23\frac{kg}{m^3}$ $\quad A_t := \pi \cdot r_{th}^2$ $\quad m_{flow} := \rho_{0deg} \cdot A_t \cdot c_i$

$\mu := .32$ $\quad F_{fluid} := c_i \cdot m_{flow}$ $\quad x := \left(.75 \cdot 11.4 + \frac{11.4}{2}\right) mm$ $\quad F_{fluid} = 0.04 \cdot kN$

$M_{oment} := F_{fluid} \cdot x$ $\quad M_{oment} = 0.572 \cdot N \cdot m$

$D := 17\,mm$ $\quad r := \frac{D}{2}$ $\quad F_{pulley} := \frac{M_{oment}}{r}$ $\quad F_{pulley} = 0.067 \cdot kN$

$T_{ension} := \frac{F_{pulley}}{\mu}$ $\quad \boxed{T_{ension} = 0.21 \cdot kN}$

Appendix V – Final Design Drawings:

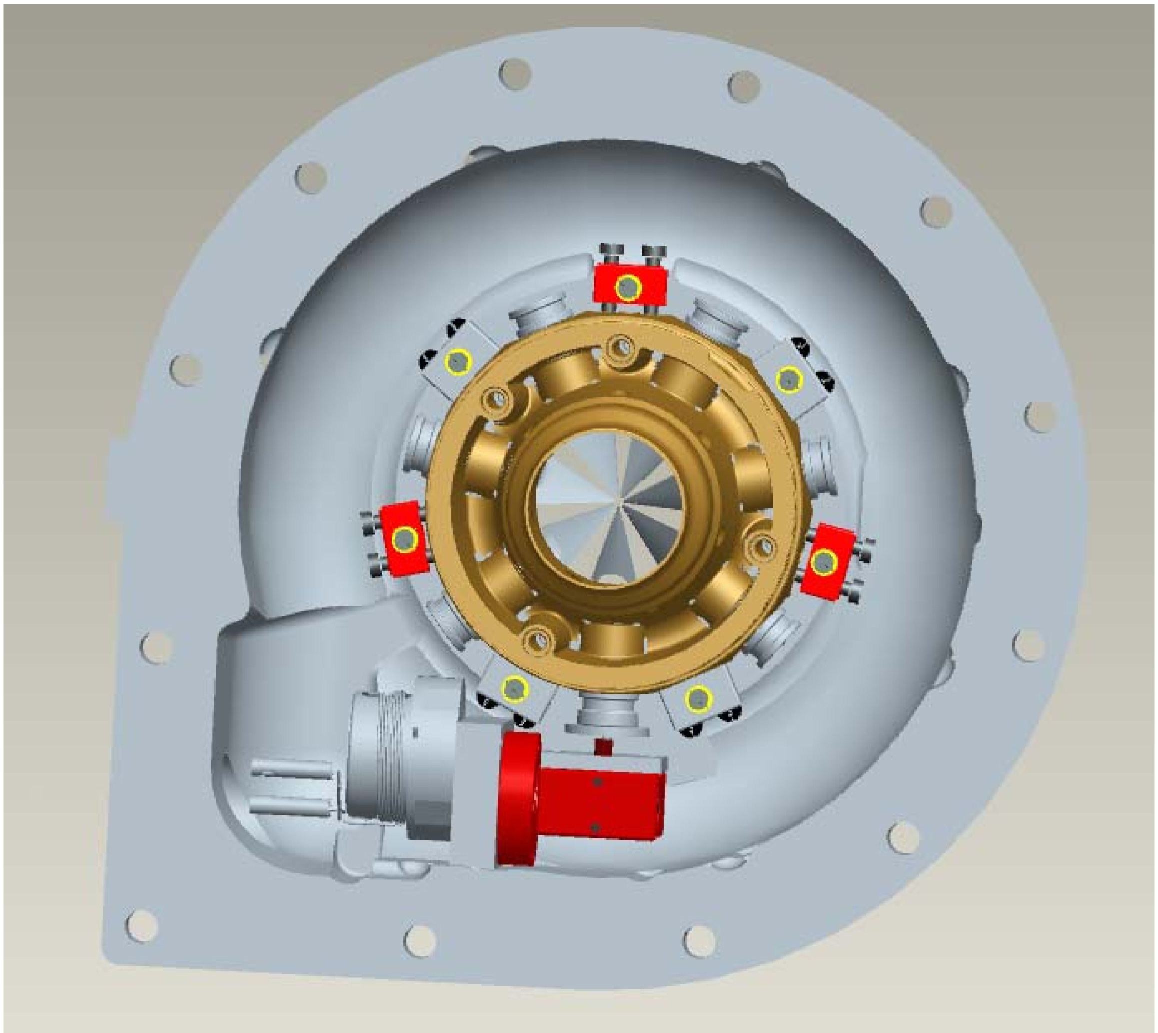


Figure 9.1------Completed Assembly

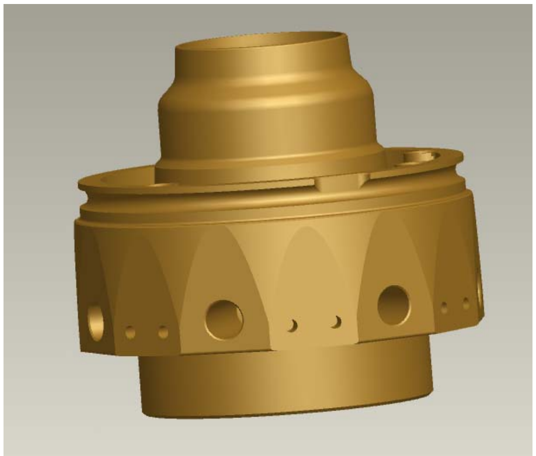


Figure 9.2------Throat Profile

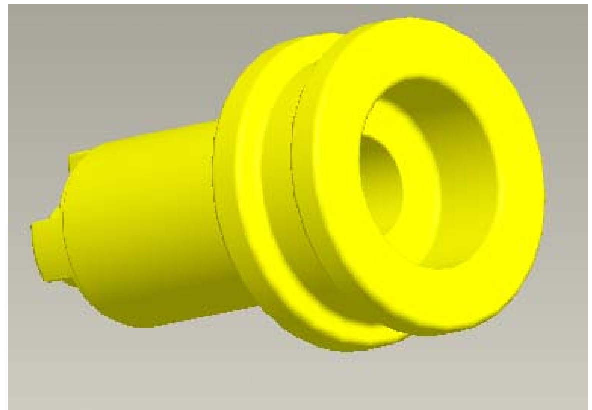


Figure 9.3------Standard IGV Pulley

Figure 9.4------IGV

Figure 9.5------IGV Lock Washer

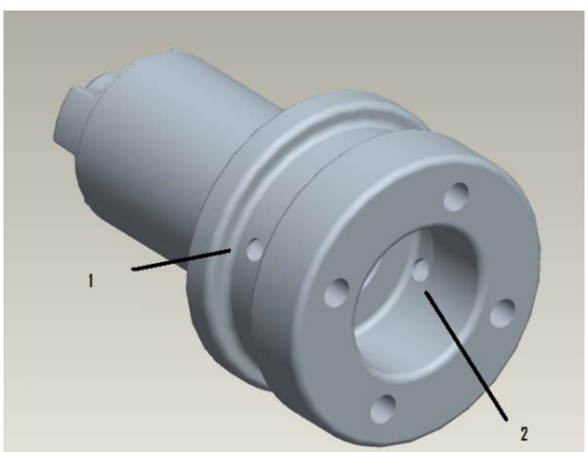


Figure 9.6------Input IGV Pulley

Figure 9.7------IGV Screw

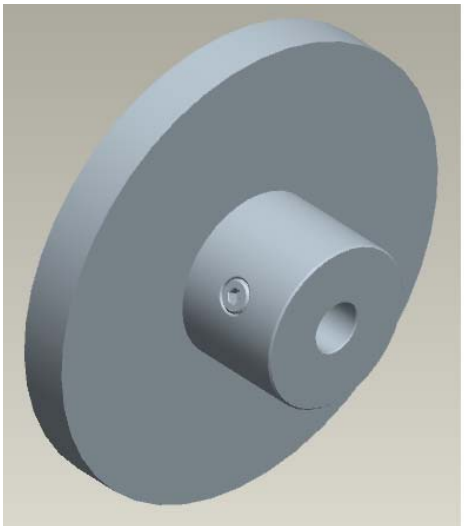


Figure 9.8------Large Gear

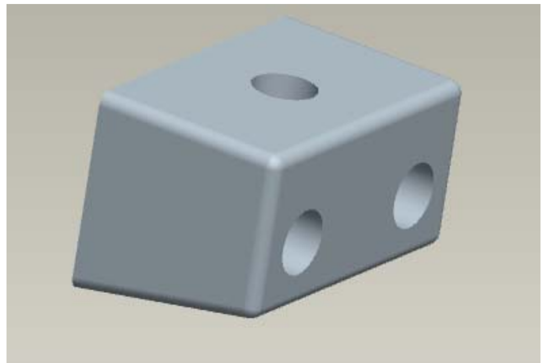


Figure 9.9----Idler Block/ Tension Block

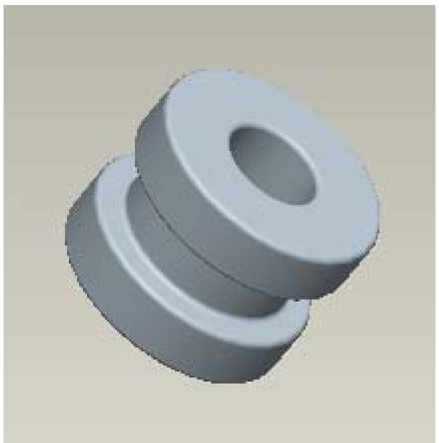


Figure 9.10------Idler Pulley

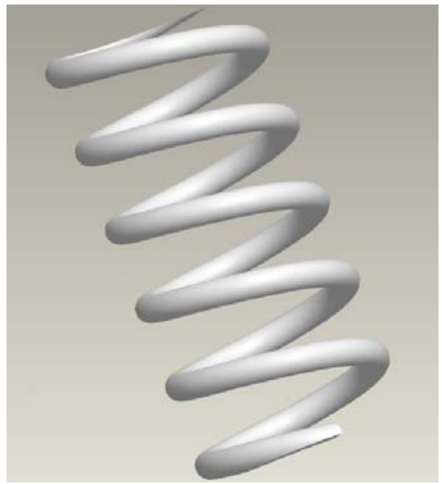


Figure 9.11-------Tensioning Spring

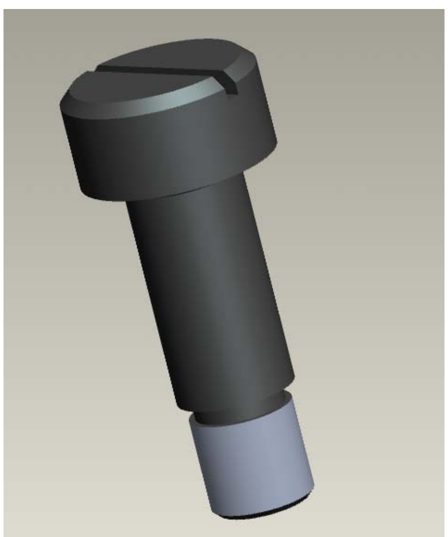


Figure 9.12------Idler Pulley Screw

Figure 9.13------Tension Screw

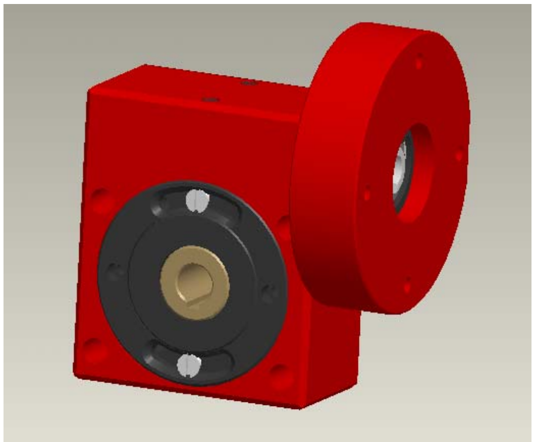


Figure 9.14------Gear Reduction Box

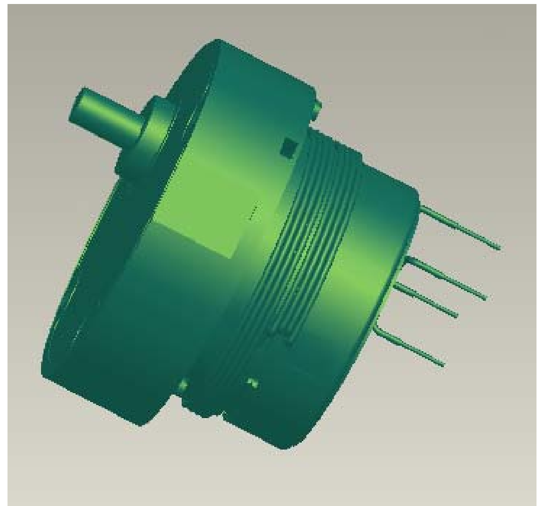


Figure 9.15------Stepper Motor

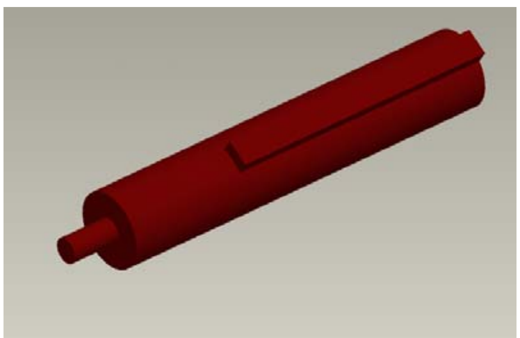


Figure 9.16------Keyed Gear Box Shaft

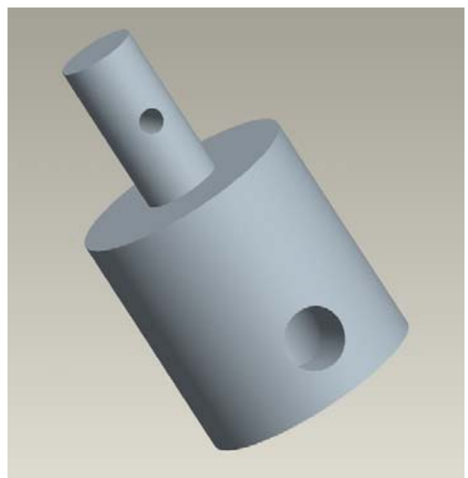


Figure 9.17------Motor Shaft

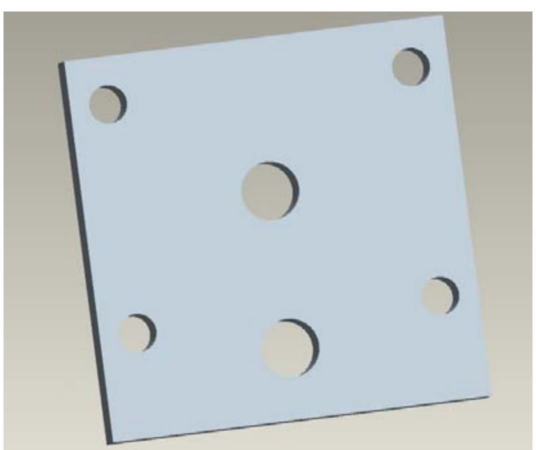


Figure 9.18------Bearing Block

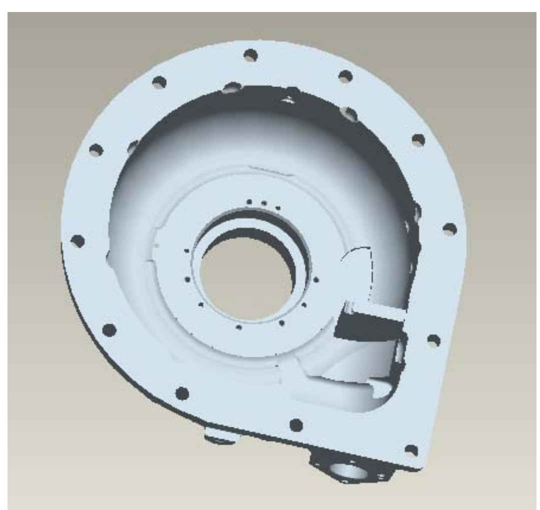
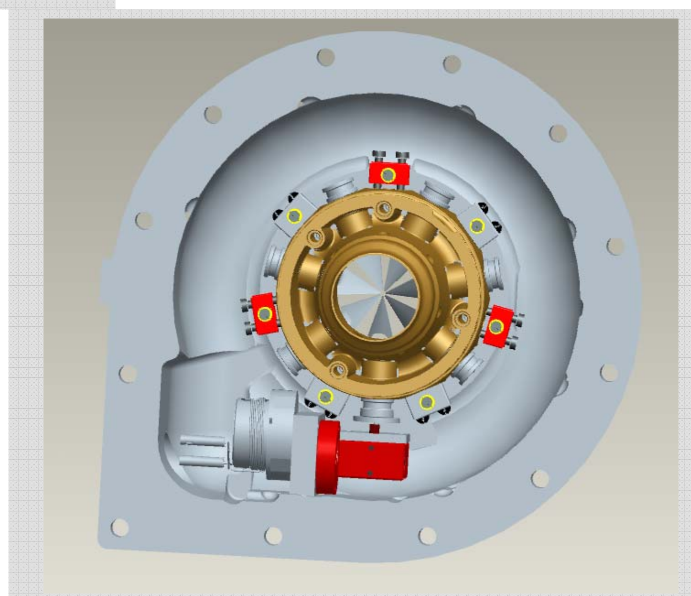


Figure 9.19------Main Housing

Assembly Manual for Cable Controlled Intel Guide Vanes

SRDP / 08-12

CC-IGV Series



CC-IGV SRDP Series

Contents

Information

The main benefit of the IGV's in the compressor inlet is their capability of turndown control. IGV's reduce the power input when the compressor is running at a flow rate lower than that of its design. However, there are other ways to control the flow such as:

- Butterfly Valves
- Variable Speed Drive Compressor
- Suction Throttling
- Impeller Throttling Sleeve
- Adjustable Diffuser Vanes

IGV's are chosen due to the fact that they are more efficient to the system and provide accurate and reliable operation. The other methods however, can be combined with an IGV system to achieve an even greater efficiency. These are much like the compressors that Danfoss Turbocor designs, builds and maintains.

Danfoss Turbocor's' compressor is a combination of the variable speed drive and the IGV flow control system. This leads to a higher efficiency than the leading oil-flooded screw compressors. As can be seen in the figure below, Turbocor's compressor is slightly more efficient at 100% of load capacity. However, at 20-60% the Turbocor compressor requires significantly less power, thus producing a much higher efficiency when compared to leading competition within this range.

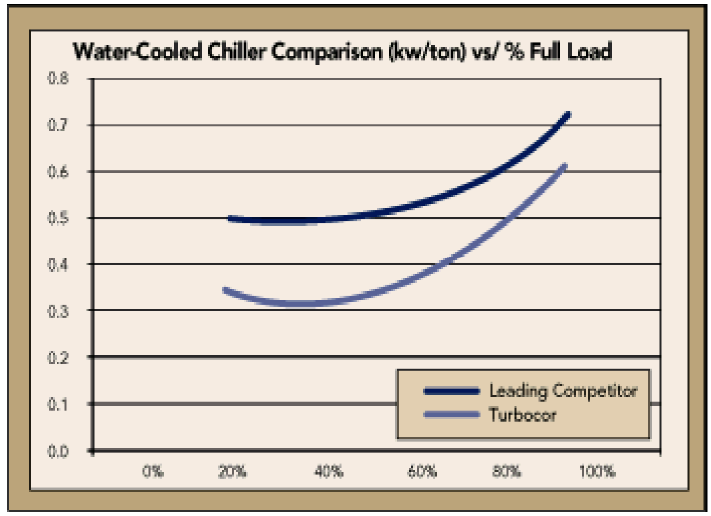


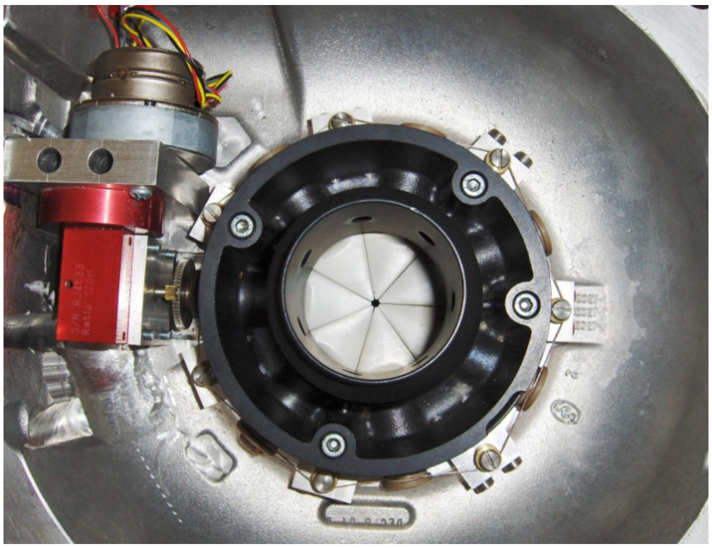
Figure 1 - 75 Ton Danfoss Turbocor Compressor vs. Leading Oil-Flooded Screw

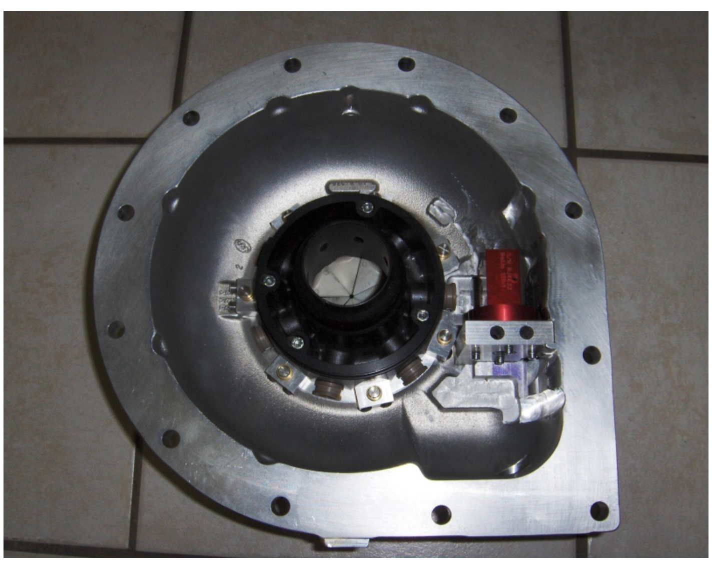
Performance

- The efficiency increase from having IGV's in only the first stage of compression decreases as the number of compression stages increase. This is to say that each stage of compression must have the IGV control system in order to fulfill maximum efficiency of the cycle.
- The more precise the IGV's are adjusted, the greater their impact on performance will be.
- IGV's are extremelyf beneficial when the compressor is not fully loaded. However at 100% load capacity, the IGV's are at 110° and actually increase the flow rate of the working fluid, thus increasing the compressors overall efficiency.
- The IGV's must be positioned as to not allow harmonics between the working fluid and impeller. If this happens the IGV's will

eventually be damaged due to vibration caused by turbulent flow in the system.

- IGV's do not directly increase turndown of the system; however, they do enhance the efficiency of the compressor during the turndown cycle.





Parts List # of parts

1. Throat housing -------------------1
2. IGV --------------------------------7
3. IGV Screw Insert ----------------7
4. IGV Input Pulley ----------------1
5. Standard IGV Pulley -----------6
6. IGV Lock Washer ---------------7
7. IGV Screw ------------------------7
8. Idler Pulley Post -----------------4
9. Idler Pulley -----------------------7
10. Idler Screw -----------------------7
11. Tension Slot ---------------------3
12. Tension Post --------------------3
13. Tension Screw ------------------3
14. Cable -----------------------------1
15. Gear Reduction Box -----------1
16. Keyed Gear Box Shaft --------1
17. Stepper motor ---------------------1
18. Main housing ---------------------1
19. Throat Profile Screws -----------4
20. Gear Box Screws -------------- 5

Assembly

A. IGV Control System

i. IGV Installation

1. Install the IGV screw Inserts in the IGV shafts by pressing them into the provided holes.
2. Insert the IGV's through inner throat housing using the holes that are provided.
3. Insert the IGV Pulleys into backside of the throat housing making sure that they are aligned with the IGV shafts.
4. Once in place, secure setup with the IGV screws and lock washers by inserting them into the holes located in the center of the IGV pulleys. This will hold the IGV's and IGV pulleys together.
5. Assemble all seven IGV's and the Pulleys together in the throat profile.

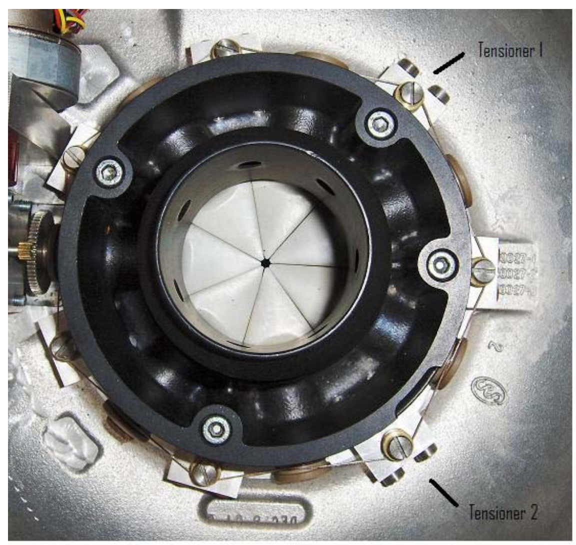
Disclaimer

The IGV's must be turned 90° from the base of the throat profile in order for all of them to be inserted without damage.

ii. Stationary Idler Block and Idler Pulley Attachment (Stationary Idler)

1. Attach Idler Pulley onto the seven idler blocks, using the 3/8" shoulder screws provided.
2. Select five of the seven blocks to assemble as stationary blocks and the other two will then become the tensioning idler blocks.
3. Using two 8/32-button cap screws, attach each of the five blocks around the outside of the throat profile at the desired location*.

Figure 2 – Tensioning Block Location



iii. Tensioner Idler Block Assembly

1. Follow through with the -*Stationary Idler*- in **step 1**.
2. The remaining two blocks will become the tensioning blocks.
3. Attach the two remaining blocks using the stripper bolts provided.
   - **a.** Begin by inserting the two ¾” long shoulder screws into the proper holes of the idler block.
   - **b.** Slide the springs over the shoulder screws and compress them against the outer throat profile. As you do so, screw in the two ¾” long shoulder screws.

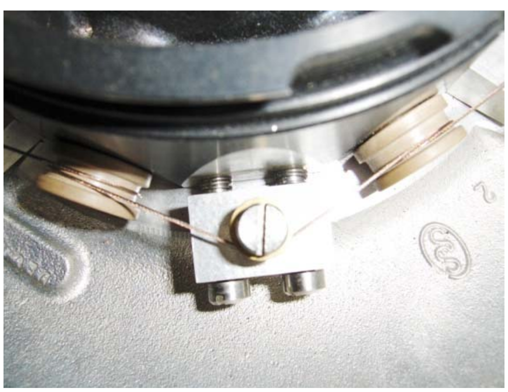


Figure 3 – Tensioning Block and Pulley

Disclaimer

- *A suggestion is to attach the stationary block as seen in **Fig. 2.**
- If need extra tensioning on the cable, an extra tensioner can be added in place of a stationary block.
- All blocks are interchangeable between the stationary blocks and tensioner blocks

iv. Cable Installation

1. Using the 7x7 stainless steel micro cable.
   - **a.** Begin with the drive pulley; insert one of the loose ends of the cable into one of the input pulleys cable holes (1). Wrap the cable around the pulley one time.
   - **b.** Now guide the cable around the outer perimeter of the throat housing to the first idler pulley and then to the next IGV pulley, as seen in **Fig. 4.** Continue around the throat profile repeating this process.

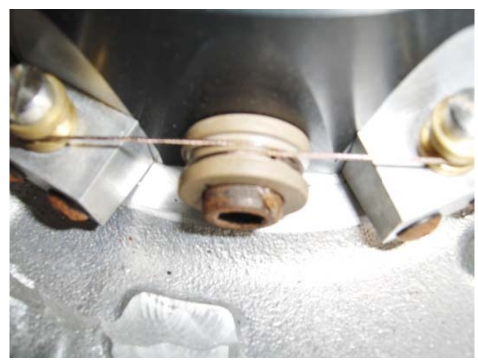


Figure 4 – Idler Block and IGV Pulley Set Up

   - **c.** Once the wire is completely wrapped around the throat, insert the second loose end of the cable into hole number 2 of the drive pulley.

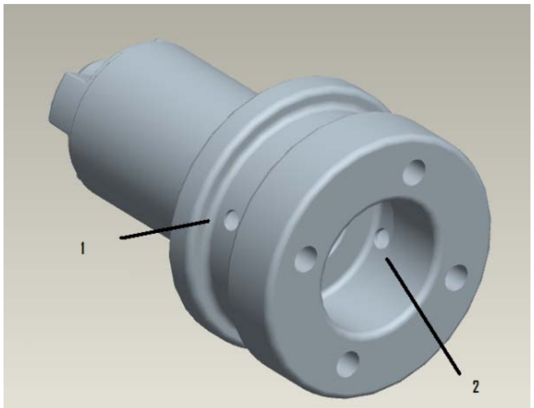


Figure 5 – Cable Tensioning Pulley

2. Use the *spring tool*** to compress springs while wrapping the wire around the system.
3. Using self locking pliers to hold the cable in place press the large gear bushing into the drive pulley; this will secure cable.
4. Release cable and spring tensioners. The cable will automatically tighten.
5. Insert the large gear shaft into the large gear and secure it with the set screw provided. Press large gear hub into the large gear bushing on the drive pulley.

B. Motion Control

i. Gear Reduction Box and Mounting Plate Assembly

1. Attach gear alignment plate to the external gear box using four 8/32 screws. This will keep the gears center to center distance constant and control deflection.
2. Press small gearbox shaft into gearbox.
3. Press small gear onto the small gear output shaft, **Fig. 6.**

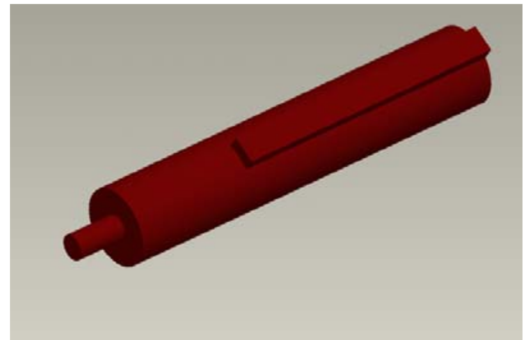


Figure 6 – Small Gear Shaft

4. Attach gear reduction box onto the mounting plate.

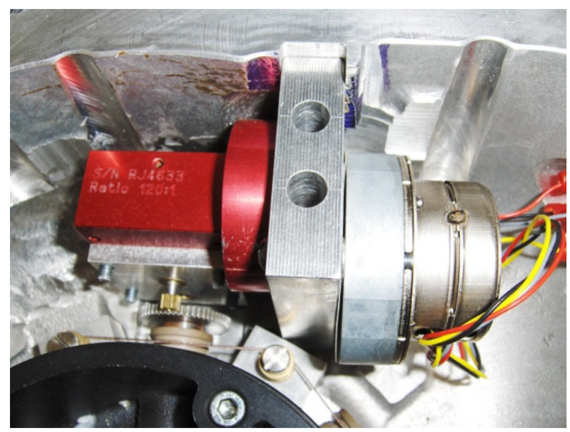


Figure 7 – Gear Set Up

C. Unification System Assembly

i. IGV Control System and Mounting Plate Installation

1. Slide the IGV control system **Part A** in unison with the mounting plate installation, **Part B** into housing.
2. Screw down throat and mounting plate.

3. Attach the motor shaft onto the motor output shaft using an 8/32 set screw.
4. Attach and secure motor onto the mounting plate using a single 8/32 screw.
5. Attach power input wiring to the motor connector.

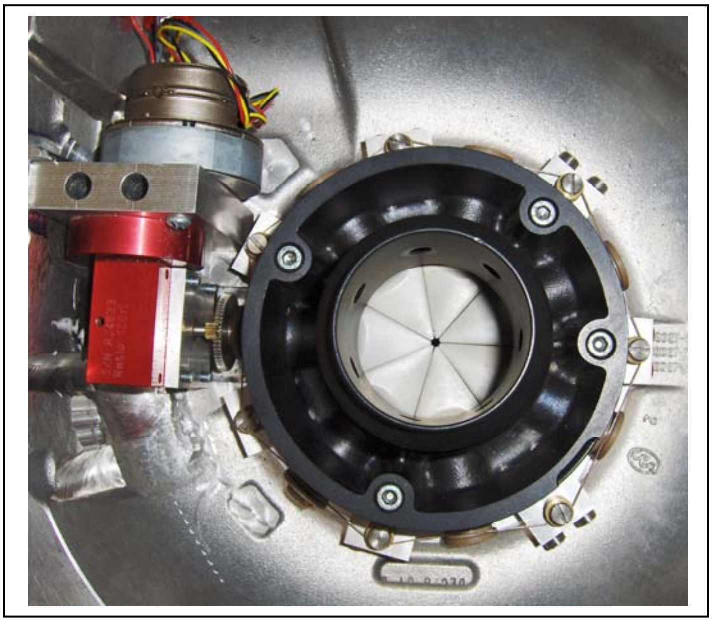


Figure 8 – Final Assembly

Danfoss Turbocor Inc.
High Efficiency Compressors
1769 Paul Dirac Drive
Tallahassee, FL 32310, USA
Phone: 850-504-4800
Fax: 850-575-2126
http://www.turbocor.com

Danfoss Turbocor Europe:
Seestrasse 199
CH-8820 Waedenswill Switzerland
Tel: ++ 41 43 477 9590
Fax: ++ 41 43 477 9591
pbolliger@turbocor.com

Appendix VII – Original Drawing and Cost

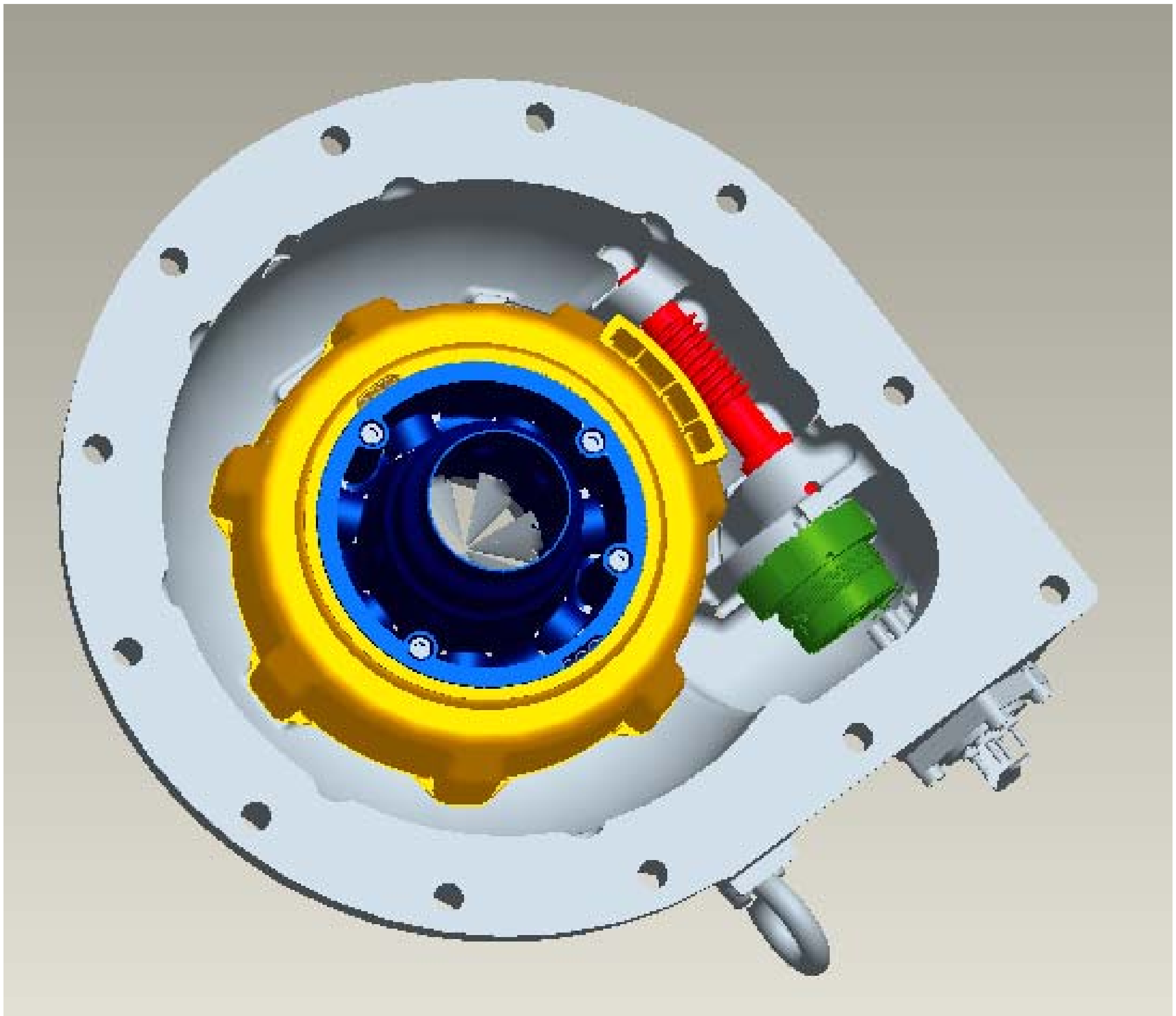


Figure 10.1------Original IGV Control Design

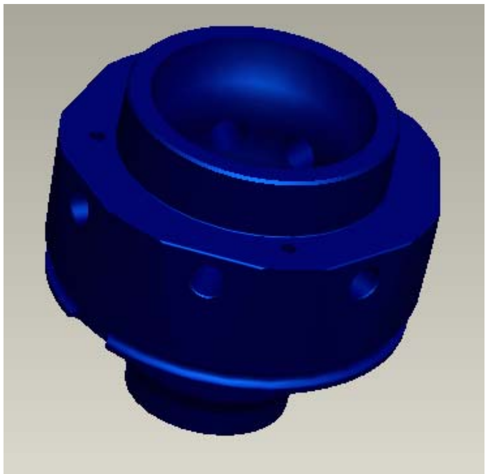


Figure 10.2------Original Throat Profile

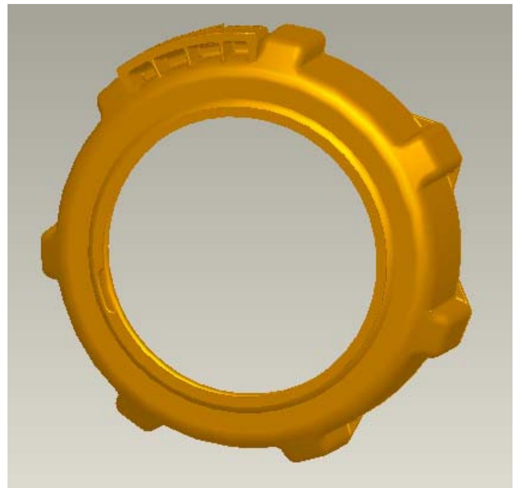


Figure 10.3------Cam Drive Housing

Figure 10.4------Cam Assembly

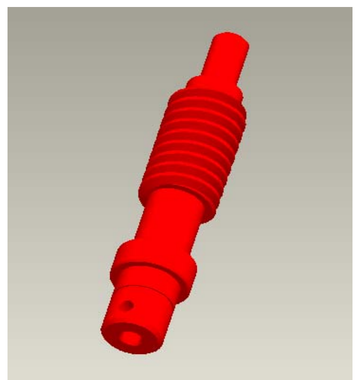


Figure 10.5------Worm Gear

Figure 10.6------Cam Roller

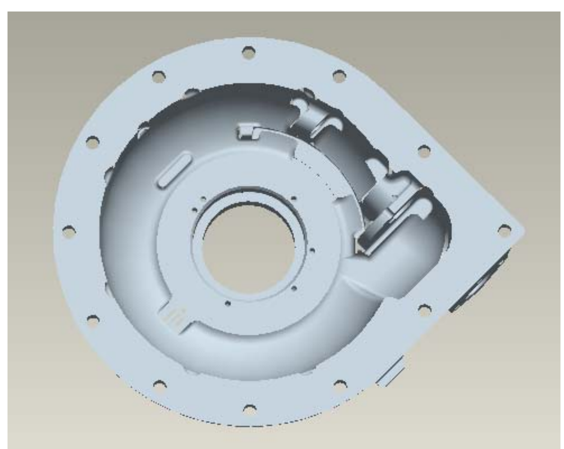


Figure 10.7------Original Main Housing

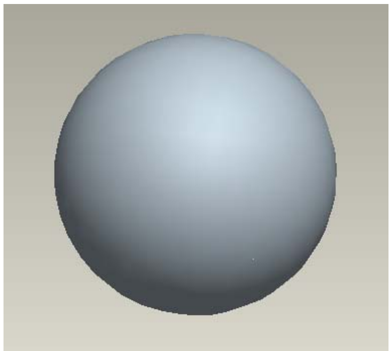


Figure 10.8------Ball Bearing

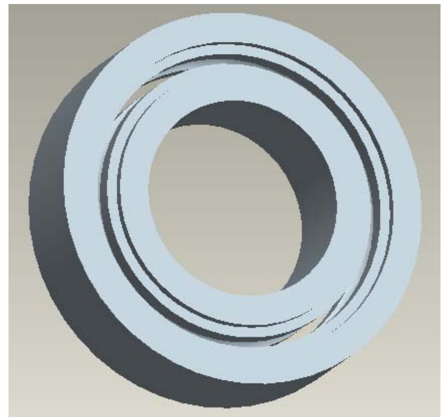


Figure 10.9------Worm Gear Bearing

Figure 10.10 ------IGV Position Indicator

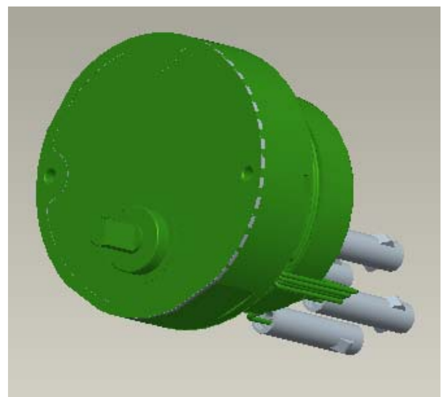


Figure 10.11------Original Motor Assembly

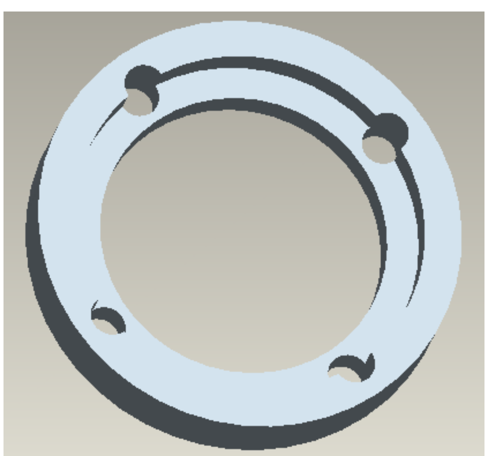


Figure 10.12 -Reverse Thread Retaining Nut

Figure 10.13------IGV

Cost Type: F - Frozen Standard

Parent Item	Description	M/B	Type	UM	Frozen Standard	Current Standard	Average/Actual
200144	IGV HOUSING ASSEMBLY - TT-300 EXTENDED RANGE	M	P	EACH	371.0606	350.7230	358.3928

Level	Component	Description	Type	Find	M B	Unit Meas	Quantity Per	Scrap	Vendor	Opn Seq	Effectivity Rev IN	Effectivity IN Date	Effectivity Rev OUT	Effectivity OUT Date	Total Cost
. 1	324007	IGV DRIVE ASSEMBLY - 75 TON	P		M		1.0000					01/08/07		12/31/99	96.4877
.. 2	430005	IGV LEVER ASSEMBLY-75 TON- DOC ONLY	P		M		7.0000				0001	01/25/05		12/31/99	33.9803
... 3	701515	RIVET - IGV BLADE ROLLER	S		B		1.0000		3492		0001	01/25/05		12/31/99	14.9310
... 3	701707	LEVER - IGV BLADE 75 TON	S		B		1.0000		3435		0001	01/25/05		12/31/99	7.0245
... 3	701708	RIVET - IGV BLADE ARM 75/90 TON	S		B		1.0000		3492		0002	05/20/05		12/31/99	12.0248
.. 2	700953	THROAT - IGV DRIVE TT-300 R134A	S		B		1.0000		3342		0001	01/25/05		12/31/99	39.0600
.. 2	701706	BLADE - IGV 75 TON	S		B		7.0000		3435		0001	01/25/05		12/31/99	7.0245
.. 2	741057	DISK - IGV DRIVE TOP	S		B		1.0000		3435		0001	01/25/05		12/31/99	11.1870
.. 2	900066	EPOXY RESIN - DEVCON FASTMETAL 10 (HVAC)	S		B		0.0000		3009		0001	01/25/05		12/31/99	0.0000
.. 2	900070	MAGNET - IGV 1/4" X 3/8"	S		B		1.0000		3473		0001	01/25/05		12/31/99	0.4179
.. 2	900071	BALL BEARING 6.0 DIA. MM ALLOY STEEL	S		B		62.0000		3009		0001	01/25/05		12/31/99	4.4640
.. 2	901021	WASHER M3 SPLIT TYPE ZINC PL LOCK	S		B		7.0000		4030		0001	01/25/05		12/31/99	0.0580
.. 2	901686	SCREW M3 x 20mm SHCS ZP	S		B		7.0000		4030		0001	01/25/05		12/31/99	0.2961
. 1	700083	PAD-ANTI VIBRATION - IGV	S		B		1.0000		3356			01/08/07		12/31/99	0.1800
. 1	710000	HERMETIC FEEDTHRU ASSY - 4 PIN - TAB	P		M		1.0000					01/08/07		12/31/99	17.2196
.. 2	710100	HERMETIC FEEDTHRU - 4 PIN MOUNTING PLATE - TAB	S		B		1.0000		1151		0001	07/13/05		12/31/99	15.5469

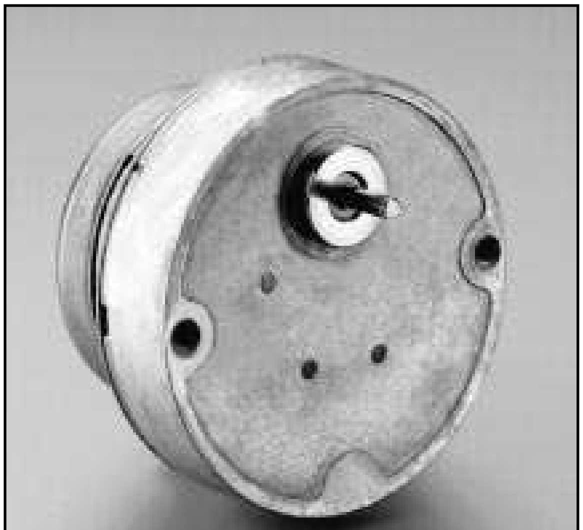
Level	Component	Description	Type	Find	M B	Unit Meas	Quantity Per	Scrap	Vendor	Opn Seq	Effectivity Rev IN	Effectivity IN Date	Effectivity Rev OUT	Effectivity OUT Date	Total Cost
.. 2	750378	HERMETIC FEEDTHRU 4 PIN - NEOPRENE OUTSIDE	S		B		1.0000		6941		0001	07/13/05		12/31/99	1.5120
.. 2	901352	'O' RING-PARKER#2-128-C1278-80	S		B		1.0000		6907			06/16/06		12/31/99	0.1607
. 1	710052	WORM IGV	S		B		1.0000		8584			01/08/07		12/31/99	28.0000
. 1	710081	COVER- IGV-INDICATOR	S		B		1.0000		4049			01/08/07		12/31/99	1.0530
. 1	710557	CLAMP, RETAINER, IGV, 4 AND 6 PIN PWM CLUSTER	S		B		1.0000		3488			01/08/07		12/31/99	1.3320
. 1	710606	COLAR - IGV WORM BEARING	S		B		1.0000		3001			01/08/07		12/31/99	4.4100
. 1	760027-2	HOUSING - IGV TT-300 EXT/TT-400	S		B		1.0000		3226			01/08/07		12/31/99	152.5821
. 1	900041	EYE BOLT M10X17LG ZP	S		B		1.0000		4030			01/08/07		12/31/99	0.8294
. 1	900043	PLUG 1/8"-27 NPT TAPER LVL S/HD ZP	S		B		1.0000		4030			01/08/07		12/31/99	0.1580
. 1	900071	BALL BEARING 6.0 DIA. MM ALLOY STEEL	S		B		1.0000		3009			01/08/07		12/31/99	0.0720
. 1	900095	SCREW M5X 8 S/SET FLT POINT Z/P	S		B		1.0000		4030			01/08/07		12/31/99	0.0252
. 1	900099	SCREW UNCX1/4"LG S/HD CAP #6-32 Z/P	S		B		1.0000		4030			01/08/07		12/31/99	0.2248
. 1	900549	LOCTITE 262	P		B		0.0000		3009			01/08/07		12/31/99	0.0000
. 1	900551	LOCTITE 55441 THREAD SEALANT	P		B		0.0000		3009			01/08/07		12/31/99	0.0000
. 1	900576	SENSOR-PRESS/TEMP-M12X1.25-150PSI	S		B		1.0000		3126			01/08/07		12/31/99	35.3571
. 1	900606	'O'RING SENSOR-PRESS/TEMP #76442-2	P		B		1.0000		3126			01/08/07		12/31/99	0.0000

Cost Type: F - Frozen Standard

Parent Item	Description	M/B	Type	UM	Frozen Standard	Current Standard	Average/Actual
200144	IGV HOUSING ASSEMBLY - TT-300 EXTENDED RANGE	M	P	EACH	371.0606	350.7230	358.3928

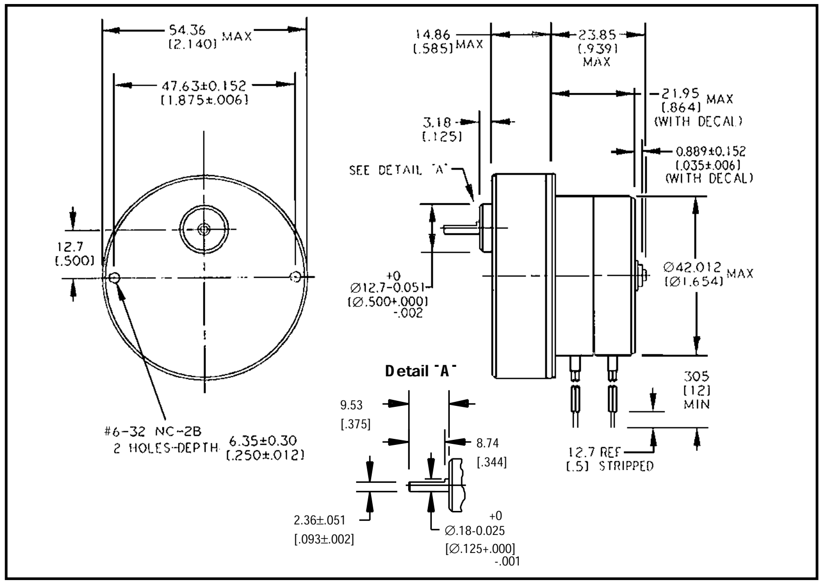
Level	Component	Type	Find	M B	Unit Meas	Quantity Per	Scrap	Vendor	Opn Seq	Effectivity Rev IN	Effectivity Date	Effectivity Rev OUT	Effectivity Date	Total Cost
. 1	900909 LOCTITE PRIMER N 7649	S		B		0.0000		3009			01/08/07		12/31/99	0.0000
. 1	900915 PIN SPRING �5x8x0.5WT STRAIGHT Z/P	S		B		1.0000		4030			01/08/07		12/31/99	0.0486
. 1	901471 SCREW M5X20 S/HD CAP BLACK DACROMET	S		B		4.0000		4030			01/08/07		12/31/99	0.3398
. 1	901609 LOCTITE 242	S		B		0.0000					01/08/07		12/31/99	0.0000
. 1	901671 SCREW M5x60mm SHCS ZP	S		B		4.0000		4030			01/08/07		12/31/99	0.8716
. 1	910098 MOTOR - STEPPER IGV 12VDC	S		B		1.0000		4233			01/08/07		12/31/99	21.0000
. 1	910243 CIRCLIP - �17 RETAINING RING EXTERNAL	S		B		1.0000		4030			01/08/07		12/31/99	0.0427
. 1	920241 BEARING, BALL - RADIAL �10 ID x �19 OD - NO	S		B		1.0000		3997			01/08/07		12/31/99	4.8420
. 1	920242 BEARING, BALL - RADIAL �17 ID x �30 OD - NO	S		B		1.0000		3997			01/08/07		12/31/99	5.9850

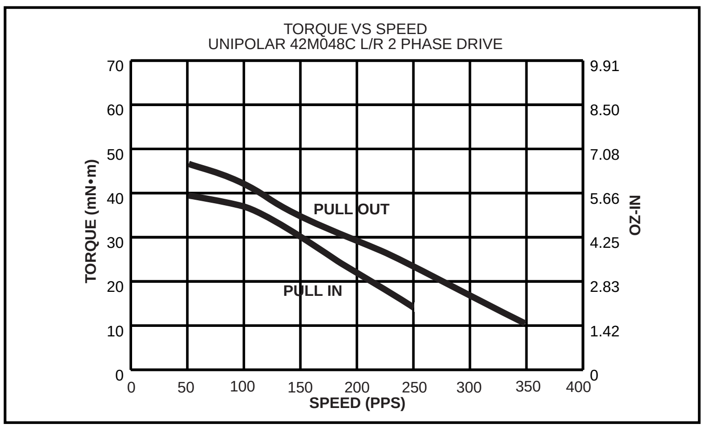
Series 42M048C Stepper Motors With Gear Trains (Type R)

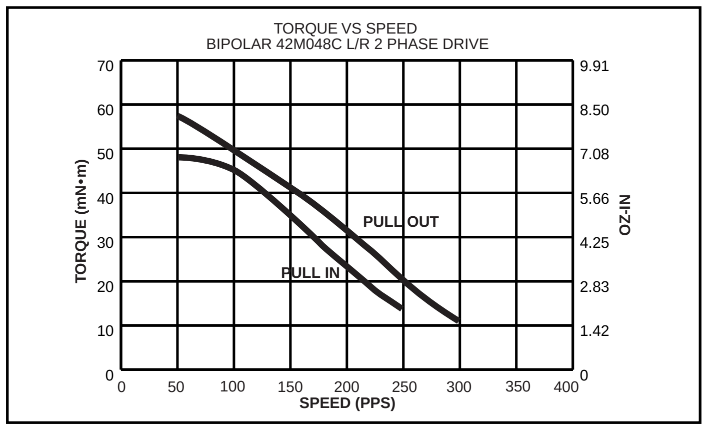


Gear Train Rating: 1.06 N•m/150 oz-in static
0.706 N•m/100 oz-in running

Dimensions: mm/inches







Specifications (Motor Only)

Part Number	Bipolar		Unipolar	
	42M048C1B	42M048C2B	42M048C1U	42M048C2U
DC Operating Voltage	5	12	5	12
Res. per Winding Ω	9.1	52.4	9.1	52.4
Ind. per Winding mH	14.3	77.9	7.5	46.8
Holding Torque mN•m/oz-in†	87.5/12.4		73.4/10.4	
Rotor Moment of Inertia g•m²	12.5 x 10^{-4}			
Step Angle	7.5°			
Step Angle Tolerance†	±0.5°			
Steps per Rev.	48			
Max Operating Temp.	100°C			
Ambient Temp Range				
Operating	-20°C to 70°C			
Storage	-40°C to 85°C			
Bearing Type	Bronze sleeve			
Insulation Res. at 500Vdc	100 megohms			
Weight g/oz	312/11.0			
Lead Wires	No. 26 AWG			

† Measured with 2 phases energized.

Available Gear Train Reductions

Part Suffix	Gear Ratio	Output Step Angle	Output Speed RPM @240 PPS	Running Torque @ 240 PPS N•m/oz-in
-R12	2.5:1	3.00°	50	0.078/11
-R16	5:1	1.50°	25	0.155/22
-R21	10:1	.75°	12.5	0.318/45
-R24	15:1	.50°	8.33	0.473/67
-R27	20:1	.375°	6.25	0.635/90
-R31	30:1	.25°	4.17	0.706/100 max
-R36	50:1	.15°	2.5	0.706/100 max
-R39	75:1	.10°	1.67	0.706/100 max

How to Order

1. List Series 42M048C.
2. Add suffix -R12 to -R39 for round diecast.

Example of Part Numbers

42M048C | 2U | -R24

-R24 — 15:1 (round)
42M048C — Basic Series
2U — Unipolar 12 Vdc

For information or to place an order in North America: 1 (203) 271-6444 Europe: (44) 1276-691622 Asia: (65) 7474-888

THOMSON AIRPAX MECHATRONICS

The specifications in this publication are believed to be accurate and reliable. However, it is the responsibility of the product user to determine the suitability of Thomson products for a specific application. While defective products will be replaced without charge if promptly returned, no liability is assumed beyond such replacement.

WORM & WORM WHEEL GEARS

SHAFT SIZE	TYPE
ØA= 1/8 ØB= 1/8	**BALL BEARING OR BRONZE BEARING**

BALL BEARINGS UNITS STOCK NO.	BRONZE BEARINGS UNITS STOCK NO.	PITCH	SHAFT SIZE ØA	SHAFT SIZE ØB	NOMINAL RATIO	C CENTER	D HEIGHT	MAXIMUM OUTPUT TORQUE (OZ.-IN.)
WX48B2-1	WX48P2-1	48 WORM & WHEEL	.1247	.1247	12.5:1	.6878	2-11/32	144
WX48B2-2	WX48P2-2				15:1	.7920	2-35/64	176
WX48B2-3	WX48P2-3				20:1	1.0003	2-31/32	240
WX48B2-4	WX48P2-4				22.5:1	1.1045	3-11/64	272
WX48B2-5	WX48P2-5				25:1	.6878	2-11/32	176
WX48B2-6	WX48P2-6				30:1	.7920	2-35/64	208
WX48B2-7	WX48P2-7				40:1	1.0003	2-31/32	288
WX48B2-8	WX48P2-8				45:1	1.1045	3-11/64	320
WX48B2-9	WX48P2-9				50:1	.6878	2-11/32	176
WX48B2-10	WX48P2-10				60:1	.7920	2-35/64	208
WX48B2-11	WX48P2-11				80:1	1.0003	2-31/32	288
WX48B2-12	WX48P2-12				90:1	1.1045	3-11/64	320

SHAFT SIZE	TYPE
ØA= 3/16 ØB= 3/16	**BALL BEARING OR BRONZE BEARING**

BALL BEARINGS UNITS STOCK NO.	BRONZE BEARINGS UNITS STOCK NO.	PITCH	SHAFT SIZE ØA	SHAFT SIZE ØB	NOMINAL RATIO	C CENTER	D HEIGHT	MAXIMUM OUTPUT TORQUE (OZ.-IN.)
WX48B3-1	WX48P3-1	48 WORM & WHEEL	.1872	.1872	12.5:1	.6878	2-11/32	144
WX48B3-2	WX48P3-2				15:1	.7920	2-35/64	176
WX48B3-3	WX48P3-3				20:1	1.0003	2-31/32	240
WX48B3-4	WX48P3-4				22.5:1	1.1045	3-11/64	272
WX48B3-5	WX48P3-5				25:1	.6878	2-11/32	176
WX48B3-6	WX48P3-6				30:1	.7920	2-35/64	208
WX48B3-7	WX48P3-7				40:1	1.0003	2-31/32	288
WX48B3-8	WX48P3-8				45:1	1.1045	3-11/64	320
WX48B3-9	WX48P3-9				50:1	.6878	2-11/32	176
WX48B3-10	WX48P3-10				60:1	.7920	2-35/64	208
WX48B3-11	WX48P3-11				80:1	1.0003	2-31/32	288
WX48B3-12	WX48P3-12				90:1	1.1045	3-11/64	320

SHAFT SIZE	TYPE
ØA= 3/16 ØB= 1/4	**BALL BEARING OR BRONZE BEARING**

BALL BEARINGS UNITS STOCK NO.	BRONZE BEARINGS UNITS STOCK NO.	PITCH	SHAFT SIZE ØA	SHAFT SIZE ØB	NOMINAL RATIO	C CENTER	D HEIGHT	MAXIMUM OUTPUT TORQUE (OZ.-IN.)
WX48B43-1	WX48P43-1	48 WORM & WHEEL	.1872	.2497	12.51	.6878	2-11/32	144
WX48B43-2	WX48P43-2				15:1	.7920	2-35/64	176
WX48B43-3	WX48P43-3				20:1	1.0003	2-31/32	240
WX48B43-4	WX48P43-4				22.5:1	1.1045	3-11/64	272
WX48B43-5	WX48P43-5				25:1	.6878	2-11/32	176
WX48B43-6	WX48P43-6				30:1	.7920	2-35/64	208
WX48B43-7	WX48P43-7				40:1	1.0003	2-31/32	288
WX48B43-8	WX48P43-8				45:1	1.1040	3-11/64	320
WX48B43-9	WX48P43-9				50:1	.6878	2-11/32	176
WX48B43-10	WX48P43-10				60:1	.7920	2-35/64	208
WX48B43-11	WX48P43-11				80:1	1.0003	2-31/32	288
WX48B43-12	WX48P43-12				90:1	1.1045	3-11/64	320

MatWeb Data Sheet

Chevron Phillips Ryton® BR42B Polyphenylene Sulfide Compound

Date: 9/14/2007 1:51:23 PM

KeyWords:
PPS
SubCat: Polyphenylene Sulfide, Thermoplastic, Polymer
Material Notes:
Natural Color Polyphenylene Sulfide Compound

Ryton® BR42B is a 40% fiberglass reinforced polyphenylene sulfide compound specially formulated to have a low coefficient of friction for use in low surface friction and wear applications. Has a wear rate of 0.001 mm/hr as per ASTM D3702.

Applications: Pump Vanes

Comments: Test specimen molding conditions: Stock Temperature, 315-345°C; Mold Temperature, 135°C; ASTM Values Converted to SI Units

Data provided by Chevron Phillips Chemical Company LP.

Properties	**Value**	**Min**	**Max**	**Comment**
Physical				
Density, g/cc	1.75	--	--	
Water Absorption, %	0.05	--	--	
Linear Mold Shrinkage, cm/cm	0.002	--	--	Measured on 102 mm X 102 mm X 3.2 mm Plaques, Edge Gated
Linear Mold Shrinkage, Transverse, cm/cm	0.005	--	--	Measured on 102 mm X 102 mm X 3.2 mm Plaques, Edge Gated
Mechanical				
Tensile Strength, Ultimate, MPa	170	--	--	
Elongation at Break, %	1.5	--	--	
Flexural Modulus, GPa	15	--	--	
Flexural Yield Strength, MPa	275	--	--	
Compressive Yield Strength, MPa	205	--	--	
Izod Impact, Notched, J/cm	0.85	--	--	
Izod Impact, Unnotched, J/cm	6.9	--	--	
Coefficient of Friction	0.32	--	--	Against 52100 steel; 100 hours; 22.7 kg load (1.7 MPa); 36 RPM; Dry; Ambient Temperature; PV=2500; ASTM D3702
Electrical				
Electrical Resistivity, ohm-cm	1.00E+13	--	--	
Dielectric Constant	4	--	--	1 MHz at 25°C; ASTM D150
Dielectric Constant, Low Frequency	4	--	--	1 kHz at 25°C; ASTM D150
Dielectric Strength, kV/mm	20	--	--	
Dissipation Factor	0.003	--	--	1 MHz at 25°C; ASTM D150
Dissipation Factor, Low Frequency	0.003	--	--	1 kHz at 25°C; ASTM D150
Comparative Tracking Index, V	150	--	--	UL 746A
Thermal				
CTE, linear 20°C, µm/m-°C	16	--	--	Axial; -50 to +50°C; ASTM E831

CTE, linear 20°C Transverse to Flow, µm/m-°C	43	--	--	
CTE, linear 100°C, µm/m-°C	98	--	--	Transverse; 100 to 200°C
CTE, linear 100°C, µm/m-°C	11	--	--	Axial; 100 to 200°C; ASTM E831
Thermal Conductivity, W/m-K	0.33	--	--	
Maximum Service Temperature, Air, °C	180	--	--	UL Temperature Index per UL746B
Deflection Temperature at 1.8 MPa (264 psi), °C	--	260	--	Annealed 2 hr @ 200°C; ASTM D654
Flammability, UL94	V-0	--	--	