Design And Performance Evaluation Of An Integrated Miniature Single Stage Centrifugal Compressor And Permanent Magnet Synchronous Motor

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University of Central Florida

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DESIGN AND PERFORMANCE EVALUATION OF AN INTEGRATED
MINIATURE SINGLE STAGE CENTRIFUGAL COMPRESSOR AND PERMANENT
MAGNET SYNCHRONOUS MOTOR

by

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A thesis submitted in partial fulfillment of the requirements
for the degree of Master of Science
in the Department of Mechanical, Materials and Aerospace Engineering
in the College of Engineering and Computer Science
at the University of Central Florida
Orlando, Florida

Summer Term
2006
ABSTRACT

An attempt has been made in this present work to design, fabricate and performance evaluate an integrated single stage centrifugal compressor and permanent magnet synchronous motor which is a key component of the reverse brayton cycle cryocooler. An off the shelf compressor – the driven and electric motor – the driver was not available commercially to suffice the requirements of the reverse brayton cryocooler. The integrated compressor-motor system was designed and tested with air as the working fluid at mass flow rate of 7.3 grams per sec, with a compression ratio of 1.58 and driven by a 2 KW permanent magnet synchronous motor at a design speed of 108,000 rpm.

A permanent magnet synchronous motor rotor was designed to operate to operate over 200,000 rpm at 77 Kelvin temperature. It involved iterative processes involving structural, thermal and rotordynamic analysis of the rotor. Selection of high speed ceramic ball bearings, their mounting, fit and pre-load played prominent role. Attempts were made to resolve misalignment issues for the compressor – motor system, which had severe impact on the rotordynamic performance of the system and therefore losses at high speeds [15], [16]. A custom designed flexible coupler was designed and fabricated to run the compressor – motor system.

An integrated compressor – motor system was an innovative design to resolve considerably several factors which hinder a high operational speed. Elimination of the coupler, reduction of number of bearings in the system and usage of fewer components on the rotor to increase the stiffness were distinct features of the integrated system.
Several custom designed test-rigs were built which involved precision translation stages and angle brackets. Motor control software, an emulator, a DSP and a custom designed motor controller was assembled to run the motor. A cooling system was specially designed to cool the stator – rotor system. A pre-loading structure was fabricated to adequately pre-load the bearings. Flow measurement instruments such as mass flow meter, pressure transducers and thermocouples were used at several locations on the test rig to monitor the flow. An adjustable inlet guide vane was designed to control the tip clearance of the impeller.
Dedicated to My loving Parents
ACKNOWLEDGMENTS

A voyage to explore and conquer the unknown is possible when accompanied by a goal oriented crew. Interdependency certainly yields productivity. The episode of my work would not be complete without being accompanied by a group of people on my project team. It is a pleasant aspect that I have now the opportunity to express my gratitude for all of them.

I would like to thank my advisor Dr. Jayanta Kapat and co-advisor Dr. Louis Chow. I have gained invaluable experience from their enthusiasm and integral view on research which I will carry as a trait and exhibit at my workplaces for many years to come. I express my sincere gratitude for their guidance, mental support and encouragement in this research work. I would like to thank Dr. Lei Zhou who really was my pseudo-supervisor, he who guided me and responded promptly to every problem in this project.

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CHAPTER 1. INTRODUCTION

1.1 Integrated Centrifugal Compressor and Permanent Magnet Synchronous Motor
– a key component of the reverse brayton cycle cryocooler.

An attempt has been made in this present work to design, fabricate and performance evaluate an integrated single stage centrifugal compressor and permanent magnet synchronous motor which is a key component of the reverse brayton cycle cryocooler. The centrifugal compressor, a dynamic component of the refrigeration cycle is governed by size restrictions, compression ratio and operational range of the cryocooling system. An off the shelf compressor – the driven and electric motor – the driver was not available commercially to suffice the requirements of the reverse brayton cryocooler.

The proposed reverse brayton cryocooler finds its application in advanced cryogenic cooling units such as next generation space based infrared sensors, superconducting electronics, and for cryogen management for future space applications. Some of distinguishable features of the cryocooler are high reliability, long-life, high efficiency, compact size, low cold-head vibration and minimal weight.

The reverse brayton cycle is a non-regenerative thermodynamic cycle. A gas compressor, an after-cooler (water cooled heat exchanger), a recuperative heat exchanger, an expansion device, and another heat exchanger to absorb the cooling load (this produced the refrigeration effect) comprise of the cryocooler [24].
Figure 1: A typical reverse brayton cycle components, [24].

Figure 2: A Temperature-Entropy Diagram of Reverse Brayton Cycle, [24].
On the Figure 1 and Figure 2,

Process $\text{(a-b)} \rightarrow$ represents an adiabatic compression of the gas from a low (ambient) pressure to a high pressure.

Process $\text{(b-c)} \rightarrow$ represents after cooling of the compressed gas back to the ambient temperature.

Process $\text{(d-e)} \rightarrow$ represents expansion using expander.

Process $\text{(e-f)} \rightarrow$ represents refrigeration effect, increase in enthalpy necessary to evaporate the fraction of working fluid that is liquefied to dry saturated condition.

The following Figure 3 and Figure 4 shows the comparison with other cryocoolers and a range over which the reverse turbo brayton cycle is operated.

Figure 3: System comparison related to input power (Walker, 1994)
Figure 4: System comparison related to operating temperature (Walker, 1994)
1.2 Single Stage Centrifugal Compressor

A centrifugal compressor is classified under the category of dynamic turbomachines. In contrast to positive displacement compressors, the dynamic compressor achieves its pressure rise by a dynamic transfer of energy to a continuously flowing fluid stream [1].

The centrifugal compressor consists of inlet guide vane, a rotating impeller followed by diffuser. The fluid is drawn through the inlet guide vane to the inlet of the impeller. The function of the impeller is to increase the energy level of the fluid by whirling it outwards, thereby increasing the angular momentum of fluid [4]. The pressure rise occurs in the diffuser where the fluid slows down to convert the kinetic energy into pressure energy.

The overall characteristic of a centrifugal compressor is generally presented in the following normalized data sets:

\[
\frac{P_{02}}{P_{01}}, \eta, \frac{P}{\rho_{01} N^3 D^3} = f \left\{ \frac{\dot{m} \sqrt{T_{01}}}{P_0 D^2 \sqrt{\gamma}}, \frac{ND}{\sqrt{\gamma RT_{01}}}, \frac{ND^2}{\gamma} \right\}
\]

where the required parameters from the experimental setup would be,

\( P_{02}, P_{01} \) – the stagnation pressures at outlet and inlet of compressor

\( \dot{m} \) – mass flow rate of the exit flow

\( T_{01} \) – Stagnation temperature at inlet of compressor

\( N \) – Rotational speed of the impeller.

The performance chart of a certain compressor is represented as follows
Figure 5: Overall Characteristics of a Centrifugal Compressor, [4].

Our goal in testing the compressor should be to create the same plot as Figure 5. In order to generate this plot we need to know the mass flow rate, the rotational speed, and the pressure ratio. The present work concentrates on a similitude scaled version single stage centrifugal compressor with air as the working fluid. The reverse brayton cryocooler was originally designed based on neon as the working fluid. But, the centrifugal compressor required to run such a system was limited by several factors like power requirement, manufacturability and especially the high speed of 313,000 rpm (see Appendix A). The specifications of the scaled single stage air compressor (B1’ – see Appendix A) are as follows as in Table 1.
In order to estimate a proper $P_{03}$, we must know the following at point 2: $P_2$, $T_2$ and $P_{02}$ or $C_2$. $C_2$ can be estimated using the mass flow rate measurement and the angle alpha, but it would be best to measure this value Figure 6. $C_2$ can be known if the total pressure at
point 2 is measured. Once this is known, we may estimate the performance through a proper diffuser.

The diffuser efficiency is defined as

\[ CP_{df} = \frac{P_{03} - P_2}{P_{02} - P_2} \]

From this we can back out the pressure ratio through the compressor, \( P_{03} / P_{01} \). From the equation

\[
\frac{P_{03}}{P_{01}} = \left[ 1 + \eta_c \sigma \frac{U_2^2}{C_p \cdot T_{01}} \right]^\frac{\gamma}{\gamma-1}
\]

we can find total compressor efficiency [8].
1.3 Components and Fabrication

The components of the single stage centrifugal compressor are as follows:-

1. Impeller – rotating part
2. Diffuser – stationary part
3. Inlet guide vanes – guiding the flow to the impeller
4. Top Plate – supports the inlet guide vanes
5. Collector – guides the pressurized flow towards a chamber or application
6. Driver or power source – electric motor, air turbine or any other power source.
7. Coupler – power transmission from driver side to driven side.
8. Bearings – the impeller shaft rotates on two bearings.
1. Impeller

The impeller of a centrifugal compressor is moving part of the compressor and the shaft energy gets transferred to the fluid energy. Typically, due to rotation of the impeller and impact of the vanes, the flow accelerates in between the blade passages leading to a drop in total pressure. The designed impeller is a backswept axial inlet and radial outlet impeller. The impeller design was based on cycle calculations and CFD. Preliminary research dedicated to design of the blades dictated two different types of impeller such the curved blade impeller and straight blade impeller. As the impeller was designed to run at 108,000 rpm the accuracy of machining and blade configuration were critical factors.

The curved blade impeller had an undercut so as to guide the fluid through the blade passages without escaping over the top of blade. But this approach gave rise to a complex manufacturing problem. Sand casting process with Aluminum 356 as the material was used to fabricate the impeller [12].

The straight blade impeller was a simplified version of the curved blade impeller. The deviation of height of the blades is important as it dictates the tip clearance and therefore the overall compression performance. The cost of manufacturing the impeller and accuracy obtained through the machining is high. A four axis CNC milling machine and Aluminum 6061-T6 was found to be adequate for machining the straight blade impeller [14],[22]. The Figure 7 and Figure 8 below show the rapid prototyped impeller, the aluminum cast curved blade impeller and four axis CNC milled straight blade impeller.
Balancing of rotating parts such as the impeller is done to remove the amount of imbalance in the rotor due non homogenous distribution of mass. Contract balancing was done from American Hofmann Corporation in Lynchburg, Virginia. Two cast impeller and one straight blade impellers were static balanced for quiet operation [6].
2. Diffuser

The design of turbomachinery is dominated by diffusion, which is conversion of velocity or “dynamic head” into static pressure [25]. The flow accelerated in the impeller needs to be slowed down and guided. The diffuser slows down the flow and pressure rise occurs [10]. An axial flow type diffuser is used in this system.

The following Figure 9 and Figure 10 show the solid model, rapid prototype and fabricated diffuser.

![Figure 9: Solid Model of Diffuser](image)

![Figure 10: Rapid Prototyped and Fabricated Solid Diffuser](image)
3. Inlet Guide Vanes

The inlet guide vanes were used at the inlet of impeller blades in order to reduce the incidence angle that the flow experiences and therefore enhance swirling Figure 11 and Figure 12.

Figure 11: Solid Model of Inlet Guide Vane

Figure 12: Rapid Prototyped and Fabricated Inlet Guide Vane
4. Top plate

The top plate guided the flow towards the center of the inlet guide vanes. The projected feature towards the center of the top cap imitated the inlet guide vane blade profile Figure 13.

![Top Plate and Side View](image1)

**Figure 13: Solid Model and Fabricated Inlet Guide Vane**

5. Collector

The collector guides the compress flow from the diffuser towards the mass flow meter. It was custom designed to accommodate the flanged connection on the mass flow meter Figure 14.

![Collector and Side View](image2)

**Figure 14: Solid Model and Fabricated Collector**
6. Electric Motor

A driver to drive the compressor at 108,000 rpm with 300 W of power was required. A custom designed electric motor from Koford engineering [9] was used Figure 15. The motor was designed to require 33 Volts and was anticipated to draw 28 Amperes of current in order to supply the 300 Watts running at 150,000 rpm. The motor tungsten carbide shaft had a diameter of 10 millimeters and used ceramic hybrid angular contact bearings. A sensorless motor controller (BLDC-004B) was supplied to drive the motor. The controller required a DC power supply with linear current limited to 100 Amperes.

![Figure 15: Koford Motor](image)

The power requirements of the Koford motor are as follows

<table>
<thead>
<tr>
<th>Shaft Power</th>
<th>Speed</th>
<th>Voltage</th>
<th>Current</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Watts)</td>
<td>(rpm)</td>
<td>(Volts)</td>
<td>(Amperes)</td>
</tr>
<tr>
<td>no load</td>
<td>100,000</td>
<td>22</td>
<td>5.2</td>
</tr>
<tr>
<td>no load</td>
<td>150,000</td>
<td>33</td>
<td>6.8</td>
</tr>
<tr>
<td>300</td>
<td>100,000</td>
<td>22</td>
<td>18.8</td>
</tr>
<tr>
<td>300</td>
<td>150,000</td>
<td>33</td>
<td>27.9</td>
</tr>
</tbody>
</table>

Table 2: Motor requirements from power supply (Koford, 2003)
7. Coupler

The means to transmit the power from driver to the driven side was by a coupler. A coupler was defined as a piece of hardware with the purpose of linking multiple shafts together so that the torque and angular velocity could be distributed throughout the system [11]. The coupler was bought from Rimitec Corporation and was rated at 28,000 rpm. The following Figure 17 shows the coupler used for the experiment.
1.4 Assembly and Experimental Set-up

The assembly of the experimental set-up can be discussed in two sections. The sequence of the above discussed components is as shown in the Figure 18 and Figure 19 below.

Figure 18: Compressor Assembly Exploded

Figure 19: Experimental Setup
Test Measurement Equipments

The inline vortex mass flow meter built by Sierra Instruments Inc. was used for measuring flow rate [21]. P-transducers varying in different ranges were used for pressure measurements. Temperature was measured with T-type thermocouples. The following Figure 20 shows the cross-section of the compressor and locations at which pressure and temperatures were measured.

1. Position 1 is the inlet conditions at atmospheric temperature and pressure
2. Position 2 is at exit of impeller and inlet of diffuser
3. Position 3 is at the exit of diffuser.
4. Position 4 is at the exit of collector

Figure 20: Pressure and Temperature measurement locations
1.5 Initial Results and Discussion

The compressor was expected to draw 300 W of power at 108,000 rpm with a pressure rise of 1.7 with a mass flow rate of 7.3 g/s. A series of tests were run with both straight blade impeller and curved blade impeller. The results of the tests are tabulated in Table 3: Curved Blade Impeller Test Results and Table 4.

<table>
<thead>
<tr>
<th>Power (Watts)</th>
<th>Speed (rpm)</th>
<th>Volumetric Flow Rate (cfs)</th>
<th>Density (kg/m³)</th>
<th>Mass Flow Rate (g/s)</th>
<th>Temperature Mass Flow Meter (ºC)</th>
<th>Diffuser (ºC)</th>
<th>Diffuser (psig)</th>
<th>Pressure Mass Flow Meter (psig)</th>
</tr>
</thead>
<tbody>
<tr>
<td>14.0</td>
<td>14493</td>
<td>0.019</td>
<td>1.207</td>
<td>0.649</td>
<td>23.10</td>
<td>24.20</td>
<td>0.135</td>
<td>0.180</td>
</tr>
<tr>
<td>28.5</td>
<td>23419</td>
<td>0.077</td>
<td>1.224</td>
<td>2.670</td>
<td>23.30</td>
<td>24.60</td>
<td>0.315</td>
<td>0.405</td>
</tr>
<tr>
<td>40.0</td>
<td>31348</td>
<td>0.056</td>
<td>1.241</td>
<td>1.969</td>
<td>23.60</td>
<td>24.80</td>
<td>0.540</td>
<td>0.630</td>
</tr>
<tr>
<td>60.0</td>
<td>38168</td>
<td>0.045</td>
<td>1.262</td>
<td>1.608</td>
<td>24.00</td>
<td>25.40</td>
<td>0.855</td>
<td>0.900</td>
</tr>
<tr>
<td>90.0</td>
<td>46512</td>
<td>0.108</td>
<td>1.290</td>
<td>3.946</td>
<td>24.90</td>
<td>26.00</td>
<td>1.215</td>
<td>1.305</td>
</tr>
<tr>
<td>119.0</td>
<td>54645</td>
<td>0.087</td>
<td>1.323</td>
<td>3.259</td>
<td>25.80</td>
<td>26.60</td>
<td>1.710</td>
<td>1.755</td>
</tr>
<tr>
<td>174.4</td>
<td>61920</td>
<td>0.073</td>
<td>1.354</td>
<td>2.799</td>
<td>27.70</td>
<td>29.10</td>
<td>2.205</td>
<td>2.250</td>
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<tr>
<td>241.2</td>
<td>89485</td>
<td>0.080</td>
<td>1.382</td>
<td>3.130</td>
<td>29.50</td>
<td>30.50</td>
<td>2.655</td>
<td>2.700</td>
</tr>
</tbody>
</table>

Table 3: Curved Blade Impeller Test Results

<table>
<thead>
<tr>
<th>Power (Watts)</th>
<th>Speed (rpm)</th>
<th>Volumetric Flow Rate (cfs)</th>
<th>Density (kg/m³)</th>
<th>Mass Flow Rate (g/s)</th>
<th>Temperature Mass Flow Meter (ºC)</th>
<th>Diffuser (ºC)</th>
<th>Diffuser (psig)</th>
<th>Pressure Mass Flow Meter (psig)</th>
</tr>
</thead>
<tbody>
<tr>
<td>63.2</td>
<td>28571</td>
<td>0.068</td>
<td>1.224</td>
<td>2.357</td>
<td>24.30</td>
<td>26.20</td>
<td>0.405</td>
<td>0.450</td>
</tr>
<tr>
<td>145.6</td>
<td>48309</td>
<td>0.106</td>
<td>1.282</td>
<td>3.849</td>
<td>25.10</td>
<td>26.80</td>
<td>1.170</td>
<td>1.215</td>
</tr>
<tr>
<td>212.8</td>
<td>59666</td>
<td>0.084</td>
<td>1.328</td>
<td>3.159</td>
<td>26.20</td>
<td>27.30</td>
<td>1.800</td>
<td>1.845</td>
</tr>
<tr>
<td>338.0</td>
<td>72464</td>
<td>0.086</td>
<td>1.392</td>
<td>3.389</td>
<td>28.10</td>
<td>27.70</td>
<td>2.745</td>
<td>2.745</td>
</tr>
<tr>
<td>540.0</td>
<td>82645</td>
<td>0.097</td>
<td>1.439</td>
<td>3.952</td>
<td>30.20</td>
<td>31.70</td>
<td>3.465</td>
<td>3.465</td>
</tr>
<tr>
<td>868.0</td>
<td>86655</td>
<td>na</td>
<td>na</td>
<td>na</td>
<td>na</td>
<td>37.00</td>
<td>3.735</td>
<td>3.780</td>
</tr>
</tbody>
</table>

Table 4: Straight Blade Impeller Test Results
It was observed that due to a cumulative effect of several structural issues the power drawn by the compressor soared way beyond 300 Watts. This fact can be visualized from the Figure 21. The misalignment of the compressor and motor shaft and incapability of the coupler to handle high speeds were the primary reasons for failure. Excessive current drawn to run the compressor also damaged the controller several times.

![Figure 21: Power drawn versus Speed Plot](image)

Due to bearing, motor controller and misalignment issues high operational speed could not be maintained and very few data points could be obtained to plot the compressor efficiency chart. The following Figure 21 shows a plot of the overall compressor characteristics diagram with scarce data points.
It was clear from these experiments that the issues to be focused upon to attain high speeds were

1. Resolve misalignment issues
2. Design of new coupler to handle high speeds
3. Design a new electric motor to handle the power required by compressor
CHAPTER 2. REDESIGN OF EXPERIMENTAL SETUP

2.1 Analysis and Design of Coupler

COUPLER

A coupler is defined as a piece of hardware with the purpose of linking multiple shafts together so that the torque and angular velocity could be distributed throughout the system [11].

2.1.1 Types of Coupler

The coupler is classified in two major types such as A) Rigid Coupler and B) Flexible Coupler.

A) Rigid Coupler

Rigid couplings are used to connect rotating members such as shafts and transmit torque and motion. Rigid couplings are generally made of aluminum, steel, or stainless steel. Solid, O-shaped couplings attach to shafts with set screws. Clamp couplings are solid, C-shaped devices with a split that is closed with tightening screws. Two-piece clamps are split axially into two pieces, each of which is closed onto the shaft with tightening screws. In multi-screw couplings, all of the screws are inserted from the same direction. Rigid couplings like shrink discs with two external rings that tighten together axially and exert inward force on a third ring via tapered surfaces. Rigid couplings with keyways include keys that can be used to stop the rotation of the shaft couplings.
B) Flexible Coupling

Flexible couplings couple or connect rotating members such as motors and drive shafts while allowing misalignment in either angular or parallel offset orientation [13]. They include sliding block, metal contoured diaphragm, roller or silent chain, steel grid, coil spring, wafer or flexible disc, Schmidt, beam or flexured (helical), gear or multi-jaw, metal bellows, elastomeric tire, magnetic, and flexible link. Sliding blocks are slotted or jawed flanges on shafts that transmit torque through a captured sliding disk or block. Contoured metallic diaphragms provide high horsepower and revolutions per minute (RPM). Chains are used with sprockets in Roller or silent chain couplings. They are useful for heavy-duty, high torque applications. Steel grids have slotted flanges on each shaft coupled by a flexible steel band that is laced through the slots providing good shock absorption. Coiled spring flexible couplings couple shafts with a coiled spring and are suitable for instrument and low horsepower applications. Torque through sheet-metal plates mounted to shafts and intermediate plates by two-point supports are used in Wafer or flexible disc couplings. They are suitable for light- to moderate-duty applications. Schmidt couplings transmit torque through a series of plates connected by links and allow large lateral misalignment. Beam, flexured or helical couplings prevent backlash and provide constant velocity with angular misalignment. With gear or multi-jaw couplings, gears with external teeth are mounted on shafts. A sleeve with internal teeth fits over the gears. These couplings provide high torque and allow axial misalignment, but usually needs lubrication and good shaft alignment. Cyclic speed variation is not provided in Metal bellows that operate at high speed. They are designed for light-duty applications. Elastomeric tire-shaped flexible
couplings are shaft hubs attached to either side of an elastomeric "tire." They accommodate angular and parallel misalignment and end-float. Magnetic couplings are permanent magnets mounted on shafts separated by nonmagnetic barrier. Torque ratings to 500 ft.-lb. can eliminate the need for seals. Flexible links are three cantilevered beams or links arranged in a triangle and are attached to hubs and torque sleeves. These flexible couplings can handle angular, parallel, axial misalignment.

Mechanical properties for both rigid and flexible couplings include rated torque, rated speed, torsional stiffness, and backlash. The rated torque is the maximum service torque for which the coupling is rated. The rated speed is the maximum rated rotational speed of the coupling. Stiffness is expressed in torque per unit angular deformation (e.g., required torque to deform the coupling one degree, etc.). Backlash is the rotational position loss due to a direction change. It is measured in angular units such as degrees. Important coupling dimensions to consider include bore diameter, coupling diameter, coupling length, and design units. The bore diameter is the internal diameter for mating to the motor or shaft-end. The coupling diameter is the outside diameter (OD) of the coupling and includes the housing, etc. The coupling length refers to the overall length of the flexible coupling.

Alignment and motion parameters to consider when specifying flexible couplings include angular misalignment tolerance, parallel misalignment tolerance, and axial motion allowed. The angular misalignment tolerance is the maximum angular misalignment between coupled shafts that flexible couplings can accommodate. The parallel misalignment tolerance is the maximum parallel offset between shafts that couplings can accommodate. The axial motion allowed refers to the relative axial motion
allowed by the coupler between shafts. Operating temperature is an important environmental parameter to consider when searching for flexible couplings.

2.1.2 Problems of the Rimitec Coupler

A miniature servo-insert type coupler manufactured by Rimitec Corporation [17] was used in the experiment Figure 23. The size of the coupler was restrained by the shaft size of 10 mm and was rated to 28,000 revolutions per minute. No other coupler was available on the market that could handle speeds over a few thousand revolutions per minute.

![Figure 23: Exploded view of Rimitec Coupler](image)

<table>
<thead>
<tr>
<th>Maximum Velocity</th>
<th>Lateral Misalignment</th>
<th>Axial Misalignment</th>
<th>Angular Misalignment</th>
<th>Mass</th>
</tr>
</thead>
<tbody>
<tr>
<td>30,000 rpm</td>
<td>±0.13mm</td>
<td>±0.8mm</td>
<td>±1°</td>
<td>20 grams</td>
</tr>
<tr>
<td></td>
<td>±0.005in</td>
<td>±0.032in</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 5: Specifications of the Rimitec Coupler

A servo-insert coupling allowed a restricted amount of misalignment. The orange elastic material in the middle began to plastically deform in the radial direction due to the
centrifugal stress imposed on it. In order to avoid this situation, a stainless steel retaining ring was slip fit on the elastic insert to prevent it to grow Figure 24. But, this made the coupler more rigid and prevented it from handling any misalignment. This resulted in bearing losses and even failure. Thus, it became even more difficult to attain high speeds.

![Steel Sleeve on coupler](image)

Figure 24: Steel Sleeve on coupler
A look at the coupler selection guide would give us an idea of the type of coupler we should use for our system. We required a coupler that operate at 108,000 rpm and could handle misalignments. WM Berg Inc. couplers are built in disc couplings style. Hub and the center are made of anodized aluminum alloy, rivets and washer of brass and discs of stainless steel. This coupler was definitely better than the Rimitech servo-insert coupler as seen from the manufacturer’s specifications as shown Figure 25.
This coupler allowed more angular, parallel and axial misalignment. The weight of the WM Berg coupler was less than that of the Rimitec coupler. The performance of this coupler was supposed to be better than the previous one.

The results after running the compressor-motor system with this coupler did not improve. The compressor – motor system could not go to high speeds. The system seemed to be excited by the coupler. At high speeds vibrations were observed and led excessive loss in the bearings including failure. It was understood that the coupler did not workout as expected. It rather brought down the rigid body modes and critical speed.

The requirement for a custom designed coupler specific for our system was clear.
2.1.4 Design of Helical Coupler

Helical Products Company Inc. was approached to custom design a helical coupler. The helical coupler was designed in collaboration with Helical Products Company Inc [5].

1) Why use a flexible shaft coupling?

The attachment of two “inline” shafts for the purpose of transmission of torque is a common, industrial activity. Rigid couplings would have been always used if it were possible to perfectly align these shafts and insure that operational variables such as temperature had no effect upon the relative shaft locations. However, the reality of most situations is that perfect alignment is not possible and operational parameters tend to further increase misalignments. Flexible Shaft Coupling is purposely created to transmit torque (and power) from one shaft to another while compensating for misalignments.

2) Helical Flexure

The Helical Flexure typically used the elastic aspects of specially machined, metal, bar stock for the transmission of torque and power from one shaft to another while compensating for parallel, angular and axial misalignments. The vast majority of the Flexures were of the Single Start variety with the number of coils selected to best compensate for the anticipated misalignments. Single Start Flexures could be statically balanced, but would always have been dynamically unbalanced (Helical Inc.)

3) Multiple Starts and Balance Considerations

Double Start, Triple Start, Four Start and Five Start Flexures are classified under Multiple Start Flexures. Multiple Start Flexures are inherently statically and dynamically
balanced within the production parameters of the process used to create the coils. On the basis of basic balance considerations a Multiple Start Flexure was selected to be the best for high speed applications.

A high speed coupling Flexure must be as perfectly balanced as possible. In order to balance a component material removal (or addition) from a section (or to a section) of the component is required. But it was not possible to remove material from the Flexure area or space available for addition of material to improve balance. The balancing was required to be as perfect it can be from inception.

ANSYS based analysis had shown that a Double Start Flexure with exactly one coil has the capability to transmit torque, compensate for small misalignments and be structurally capable of 150,000 rpm. The OD of the Flexure needs also to be small. OD = 0.425 inch appears to optimal. If more coils are enlisted and/or the OD is enlarged, the stress at speed exceeds the available material strength. If less than 1 coil is used, the misalignment capability becomes unusable (Helical Inc.)

4) Flexure Creation Process

Helical’s tried-and-true Generation Process had been used for decades for thousands of successful applications, where the issue of ultimate balance has not been needed. The Generation Process provides an rms product that when carefully applied, produced Flexures with elastic rates as low as a 0.1% tolerance. The drawback in this highly controller process was that it can result in somewhat uneven mass distribution and therefore prohibits its usage for high speed systems.
A Wire Electro Discharge Machining or EDM process was found to be an alternative process for Flexure manufacturing. EDM is a point process that allows the creation of Flexure slots two at a time. This information ensured that either a Double Start Flexure or possibly Four Start Flexure could be created. Since this process was untried for high speed systems, it was recommended that the proposed manufacturing be currently limited to a Double Start.

The EDM process is accomplished by machines with highly developed electronic controls. Specialized wire is used and the cutting process occurs under the surface of specially filtered water. EDM is a complex process and success, on occasion, can be redefined to be what has been done as a best effort, not by what was desired. Helical employs EDM houses from time to time, and the experience is almost always marked by compromise (Helical Inc.).

It was required that in order to accomplish this job the EDM house must have the newest machines (possibly Swiss ones) with the addition of a fourth axis rotational stage. Associated with wire EDM was the concern of recast material along with localized annealing of the material along the cut path. Recast material would not have the same elasticity as the parent material and hence, could be site for crack initiation. Localized annealing was a problem with the thin, 0.030 inch, coils. A very small amount of annealing (full or partial) was found to be beneficial to resist crack initiation.

5) Slot and Coil Width

The Slot Width was dictated by the wire size on the EDM machine. Research suggested that 0.008 inch diameter wire to be used. It was required the Coil width to be as narrow as possible to handle the applied torque (0.75 lbin) so that the maximum
misalignment capability was available. ANSYS analysis suggested that a coil width of 0.030 inch would be optimal (Helical Inc.). Minimum coil and slot width also helped to keep the length of the Flexure as short as possible which in turn exposed less coil to the high radial acceleration field experienced at high shaft speeds.

6) Material

Martensitic stainless steels make outstanding couplings and machined springs. One stainless steel, CC455 H900 per AMS 5617 was found to be particularly attractive. It exhibited very high strength, corrosion resistant and was a material of choice for EDM work related to the medical tool industry. It was important to add a solution anneal cycle (in addition to that accomplished by the mill) to any material selected. This process would insure that the Flexure was free of residual stresses that tend to distort a finished Flexure.

7) Attachments and Installation

The initial coupling sketch anticipated that motor and pump shafts will be inserted into the coupling. It is traditional to have the coupling provide the attachment mechanisms, but an alternative was to have the motor and pump ends provide the attachment mechanism. The traditional design forced a second turning/grinding operation on the opposite end of the coupling which may complicate the ability of one being able to make all surfaces concentric within very tight tolerances. If the coupling were simply a 0.425 inch diameter, thick wall tube with a Flexure in the middle, the ability to make all surfaces be concentric within very tight tolerances would be greatly enhanced.
Heating the coupling (or per the alternative, the receiving ends) to a temperature of 400 to 500 deg F and inserting the shafts was one way to insure concentric attachments. Cold (or warm) pressing was also considered to be an alternative. If one puts grooves in the coupling OD, special tools could be configured to insert or remove the coupling from the ends. The attachment of the coupling to the ends did not require to be particularly robust since only 0.75 lb-in of torque was being transmitted.

A special tool that clamps over the Flexure to make the coupling rigid for transportation, handling and installation was found to be prudent. A split-clamp coupling design was also considered to accomplish this task. This tool should be made from aluminum to prevent any marking of the coupling.

8) Misalignments

The misalignment capability of the coupling was required to be small to insure that the oscillating stresses caused by misalignment compensation are also small. Special attention towards this matter was given as typically the magnitude of the misalignments that wears-out or causes couplings to fail. In compressor –motor system, the driver and driven components was required to be aligned to well within the coupling’s parallel misalignment rating: 0.001 inch. It was assumed that any angular misalignment present will be very small. Angular misalignment could have been quantified, but was not expected to be a limiting issue.

Extension and compression was restricted within 0.005 inch. The proposed special tool for stabilizing the Flexure facilitated keeping these misalignments within limits. The Helical Flexure, particularly with an 0.008 in slot, will visually telegraph the
presents of misalignments. A simple look, would advise one if all was OK from a misalignment standpoint.

9) Flexure Hand and Torque

The Flexure was used in the windup direction. It meant that when torque was applied, the Flexure draws to a smaller dimension in the middle. The unwind direction is one where the Flexure tends to expand when the torque is applied. Unwind must never occur. Unwinding would be an issue during spin down or stopping of compressor – motor system. Stresses due to the high acceleration field (high speed operation) were relatively significant. When the torque was applied, the overall stress actually reduced. However, if the deceleration of the shaft mainly due to motor braking will be prominent then the opposite will be true and the Flexure may be endangered. A review of the anticipated duty cycles and torque reversals was required to further insure that a failure does not occur at shut down.

10) Stress Relief Holes

The coupling depicted on the first sketch (Sketch 1) exhibits large stress relief holes. They are about as large as possible given the ID size. This was done purposely. While maximum stresses on the Flexure move about on the coils depending upon the direction and nature of the forcing functions, one issue is always present. On a short coil the tendency for the stresses to be boundary condition dominant is significant. Hence, the larger the stress relief holes the better. From a manufacturing standpoint, it was intended to provide the coupling blank (no matter what the final attachments become) with all surfaces finished machined to the EDM processor. The EDM process was the last manufacturing process. In the middle of one of the stress relief holes a through hole was
provided for the wire to initially pass. The wire cutting procedure had the wire scroll out the first set of stress relief holes, then move tangentially into the slot, scroll the needed 1.5 turns of slot and finish with the final set of stress relief holes.

11) System Torsional Resistance

ANSYS modal computations strongly suggest that resonance within the Flexure will not occur at speeds up to 150,000 rpm (Helical Inc.). System torsional resonance analysis was done. The Flexure, a torsional spring with a torsional rate of 0.24 deg/lb-in, had a motor inertia on one side and compressor inertia on the other. The system was checked for torsional magnification for frequencies up to 2500 hz (150000 rpm).

The Table 8 listing stresses were created so that one can identify what stresses was associated to what Load Case. For instance, Load Case 2 is one where the 150,000 rpm acceleration field was applied to the Flexure. The resulting stresses are: Max Principal = 51.7 ksi and Max Von Mises = 52.8 ksi. Load Cases 1-8 apply to stresses resulting from a singular load case. Load Cases 5 and 6 were reserved for angular misalignments, in case we decide later to quantify those effects. Load Cases 10-13 refer to combined Load Cases. For instance, Load Case 10 combines effects from the 150,000 rpm acceleration field (LC=2) and wind up torque (LC=1). Of particular interest here is that windup torque tends to reduce the maximum torque levels (compare LC=2 and LC=10).

Three designs were proposed and one of them was modified to form the final design of the helical coupler Error! Reference source not found. show the designs.
### Table 8: ANSYS Simulated Analysis Load Cases (Courtesy: Gary Boehm, Helical Products Company Inc., 2004)

<table>
<thead>
<tr>
<th>Load Case (ANSYS Run Number)</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>13</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque, .75 lb/in windup, lb/in</td>
<td>*</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>*</td>
<td>*</td>
<td>*</td>
<td>*</td>
</tr>
<tr>
<td>150,000 rpm</td>
<td>*</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>*</td>
<td>*</td>
<td>*</td>
<td>*</td>
</tr>
<tr>
<td>Parallel Misalignment 0.001 in UX</td>
<td></td>
<td>*</td>
<td></td>
<td></td>
<td></td>
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<td>Intentionally left blank</td>
<td>*</td>
<td>*</td>
<td>*</td>
</tr>
<tr>
<td>Parallel Misalignment 0.001 in UY</td>
<td></td>
<td>*</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Intentionally left blank</td>
<td>*</td>
<td>*</td>
<td>*</td>
</tr>
<tr>
<td>Axial Extension 0.005 in UZ</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Axial Compression 0.005 in -UZ</td>
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<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Max Stress Principal Ksi</td>
<td>4.6</td>
<td>51.7</td>
<td>31.0</td>
<td>26.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Mod: Analysis For Model (2227.5 fs) MF = 1.135 at 150000 rpm (2900)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Max Stress Von Mises Ksi</td>
<td>7.5</td>
<td>52.8</td>
<td>31.0</td>
<td>46.7</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Material: CC455 H900 per ASM5617; YS = 245 ksi, US = 250 ksi
Figure 26: Helical Coupler Final Design
The Helical Coupler thus designed, developed and manufactured is as shown in the figure below.

![Figure 27: Helical Coupler Prototype](image)

<table>
<thead>
<tr>
<th>Properties</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque Capacity</td>
<td>.75 lb-in</td>
</tr>
<tr>
<td>Maximum Parallel Misalignment</td>
<td>0.001 in</td>
</tr>
<tr>
<td>Maximum Angular Misalignment</td>
<td>Near zero</td>
</tr>
<tr>
<td>Maximum Axial Misalignment</td>
<td>0.005 in</td>
</tr>
</tbody>
</table>

*Table 9: Properties of the Helical Coupler*
2.2 Alignment Issues

A coupling is generally used in power transmission equipments. Proper shaft alignment is required for dramatically increase the life of various components like bearings. Failure of bearings is inevitable due to vibration caused from misalignment of shafts. Misalignments increase the down-time, repair or replacement and lots of headaches.

The primary challenge of all methods used for shaft alignment on rotating machinery is obtaining valid measurement data [13]. The hardware setup and method used are the factors that dictate measurement accuracy. All measurement options including the laser alignment process include sources of error. The validity of measurement data is the most significant criterion of any shaft alignment process.

In the previous experimental set-up dial indicators were used. But, this process was a more of a trial and error method to determine the “high spot”. A ‘high spot’ between the shafts was noticed by rotating the coupler. As the shaft was rotated slowly about its axis, the coupler resisted the motion when the ‘high spot’ was approaching. Once the ‘high spot’ was passed, the shaft rotated freely until the same phase of rotation was approached again. Through the use of shims ranging in size from 0.0005 inches to 0.015 inches, the ‘high spot’ was minimized. This was a cumbersome process and it was very difficult to achieve the alignment without any notion of which direction to adjust. Therefore, a proper scientific method for alignment was necessary in order to quantify and direct this process.
2.2.1 Types of Alignment Procedures

The most common alignment methods used in industry are as follows:

1) Straight Edge

These rough alignment tools are still commonly used and have their place in the precision alignment process as a method of obtaining a rough alignment. The principles rely on the integrity of the coupling faces and their run out relative to the shaft axis. The methods are simple but not accurate enough for miniature shaft systems.

2) Laser Alignment Systems - Laser systems

Laser systems use laser beams and electronic detectors. One main advantage of laser beams is that there is no loss in measurement accuracy due to bar sag. All laser systems include a laser transmitter, a detector, and a computer that performs alignment calculations. This type of system uses auto collimation to measure offset and angle with a reflecting prism or a five-axis target. The target measures both vertical and horizontal components together with the angle simultaneously. This method is accurate in angular measurement over short distances. But, this process is susceptible to backlash when moving uncoupled machines. Re-measuring is required after each move as the reference is lost. The shaft size of 10 mm in our system restricted the use of Laser alignment. The equipments available in the market were meant for much bigger shaft diameters.
3) Dial indicator methods are

A) RIM FACE METHOD

Rim-Face method Figure 28 is recognized as the oldest method of shaft alignment. Many different variations of the rim-face method are used, including straight edge and feeler gauge methods, single dial rim-face, two dial rim-face, trial and error rim-face, etc. In this method, two dial indicators are used to determine the relative position of the movable shaft with respect to the stationary shaft.

![Figure 28: Rim Face Dial Indicator Method](image)

B) REVERSE RIM METHOD

The reverse rim method Figure 29 is widely acknowledged as the “preferred method” of shaft alignment. This method is used in laser alignment systems also. Essentially, the fundamental principle of the reverse rim dials method and laser systems is to determine shaft position based on two different offset measurements (rim readings) taken at two different points along the length of the shafts. Since no face readings are obtained with this method, measurement inaccuracies due to shaft endplay are essentially
eliminated. The rim-face method is used on machines whose shafts have axial end float greater than about 0.025 mm, and especially on machines with plain bearings, errors are introduced into the face reading. Since the reverse rim method requires no face readings, measurement errors that occur when using a rim-face setup due to shaft end float are eliminated.

As the face of the 10mm shaft was not easily accessible the reverse rim method was approved to be the best suitable method for our system. Laser alignment uses the reverse rim method but equipped with laser technology. The reverse rim method of alignment required a stationary body and a movable body. In our case, both the brackets were fixed. Thus, a requirement for a movable base arose.

2.2.2 Translation Stages

The requirement for a movable base to apply the reverse rim method urged us to redesign the experimental set-up. As the motor side of the experimental set-up was simple in design, it was chosen to be the movable side while the compressor assembly
remained fixed. Newport® 462-XYZ-M Translational Stages Figure 30 was used in combination to form X-Y-Z co-ordinate movable system.

![Figure 30: Translational Stages](image)

<table>
<thead>
<tr>
<th>Specifications</th>
<th>462 Series</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Stage Travel [in. (mm)]</td>
<td>1.0 (25)</td>
</tr>
<tr>
<td>Angular Deviation (µrad)</td>
<td>&lt;100</td>
</tr>
<tr>
<td>Load Capacity, Centered [lb (N)]</td>
<td>43 (191)</td>
</tr>
<tr>
<td>Load Capacity, Vertical [lb (N)]</td>
<td>15 (67)</td>
</tr>
</tbody>
</table>

Table 10: Specifications of the Translational Stage

A plate made of aluminum was machined to mount the motor assembly. Holes were drilled on this plate to match the holes on the translational stage. The motor assembly was mounted on the translational stage which could be moved in the X-Y-Z direction. The micrometers on the translational stage could adjust the motion precisely.
2.2.3 Reverse Rim Dial Indicator Method for Alignment

The alignment process can be affected by factors like “soft foot” and “sag”.

1) SOFT-FOOT

“Soft-Foot” is the term commonly applied to that condition which exists when all four the translational stage or base feet are not supporting the weight of the machine. Soft footing may cause the motor shaft to assume a different height dependent on the following factors:

1) amount of torque on the base fastenings
2) the mounting base is not true and level
3) existing shims are rusty, bent or broken
4) base was mounted on an unclean surface

The following should be done as an initial check for soft-foot:

1) Remove all dirt, rust, and burrs from the bottom of the base.
2) Set the base in place, but do NOT tighten the hold-down nuts.
3) Attempt to pass a thin shim underneath each point of the base fastened by bolts

Any point which is not solidly resting on the base is a “soft-foot.”

Final correction of soft-foot should be done as follows:

1) All bolts on the base to be aligned are tightened.
2) A dial indicator holder is secured in such a manner that the stem of the dial indicator is placed vertically above the foot which is to be checked for soft-foot.
3) The dial indicator value is set to 0.000. The bolts are completely loosened at that point only. The dial indicator value for foot movement is observed during the loosening. If the foot rises from the base when the bolt is loosened, an amount of
shims stock equal to the amount of deflection shown on the dial indicator is placed beneath it.

4) The bolt is retightened and repeated.

2) SAG

The dial indicator bar sag describes a bending of the hardware used to support a dial indicator or other part which spans the coupling. The bending action occurs as a result of gravity and cannot be totally eliminated in almost all cases of alignment. Numerous attempts have been made by fixture manufacturers to minimize the amount of sag that occurs; however, none have been successful in “eliminating” it for all alignment situations, only in minimizing it.

Factors that influence how much sag exists include:

1) Cantilever support of the dial indicator.

2) Span of the indicator bar and Stiffness of the fixture hardware materials

To determine sag:

1) The dial indicator plunger is set to zero.

2) The fixtures are rotated to read the amount of sag.
3) PROCEDURE FOR THE REVERSE RIM ALIGNMENT METHOD

![Diagram showing the procedure for the reverse rim alignment method.]

**Schematic Diagram of Compressor – Motor System for Reverse Rim Alignment Method**

- \( M \) is the offset in the plane of the movable indicator.
- \( S \) is the offset in the plane of the stationary indicator.
- \( A \) is the distance between the stationary and movable dial indicator plungers.
- \( B \) is the distance from the movable dial indicator plunger to the movable machine’s front feet bolt center.
- \( C \) is the distance between the movable machines’ front and rear feet bolt centers.

**Figure 31: Reverse Rim Alignment Compressor – Motor System**
**DIS** = DIAL INDICATOR STATIONARY SIDE

**DIM** = DIAL INDICATOR MOVABLE SIDE

**TIR** = TOTAL INDICATOR READING, is twice the amount of offset.

To obtain a complete set of readings, perform the steps below:

1. Rotate the dial indicators to 12:00.
2. Set both dials to the positive sag value.
3. Record the setting of both dials at 12:00.
4. Rotate the dial indicators to 3:00.
5. Determine and record the reading on both dials.
6. Rotate the dial indicators to 6:00.
7. Determine and record the reading on both dials.
8. Rotate the dial indicators to 9:00.
9. Determine and record the reading on both dials.
10. Rotate the dials to 12:00 and ensure that both dials return to their original setting.
i) Measuring Vertical Misalignment

1. Rotate the dial indicators to 12:00.
2. Set both dial indicators to the positive sag value.
3. Rotate both shafts (if possible) to 6:00.
4. Record the DIS and DIM dial indicator TIR values.

ii) Interpreting Vertical Misalignment Data

To determine vertical offset from 6:00 TIR’s, use the following rules:

- Stationary Side Offset = DIS TIR / 2
- Movable Side Offset = DIM TIR/2 & reverse the sign (+ to -) or (- to +)
- Coupling Centerline Offset = (Stationary Side Offset + Movable Side Offset) / 2

To determine vertical angularity from the two reverse rim offsets, use the following rule:

- Shaft Angularity = (Movable Side Offset - Stationary Side Offset) / A dimension
iii) Measuring Horizontal Misalignment

A major precaution for the measurement and interpretation of horizontal misalignment data is the establishment of the direction of view. For this training, all clock positions are referenced from the viewpoint shown below...standing behind the movable machine facing the stationary machine.

To measure horizontal misalignment, perform the following steps:

1. Rotate the dial indicators to 9:00.
2. Set both dial indicators to zero.
3. Rotate both shafts to 3:00.
4. Record the DIS and DIM dial indicator TIR values.

iv) Interpreting Horizontal Misalignment Data

To determine horizontal offset from 3:00 TIR’s, use the following rules:

- Stationary Side Offset = DIS TIR / 2 & reverse the sign (+ to -) or (- to +)
- Movable Side Offset = DIM TIR/2
- Coupling Centerline Offset = (Stationary Side Offset + Movable Side Offset) / 2.

To determine horizontal angularity from the two reverse rim offsets, use the following rule:

Shaft Angularity= (Movable Side Offset-Stationary Side Offset) ,(x100 = mm/100mm)

A dimension
2.3 Verification of components with FARO ARM

An attempt was made to find out whether the components of the single stage centrifugal compressor such as the impeller, diffuser, and inlet guide vanes were manufactured according to the modeled design and drawings. FARO ARM assisted with software CAM 2 was used to verify the manufactured components.

The Faro Arm Figure 33 is a multiple axis articulated coordinate measuring arm otherwise known as CMM with six degrees of freedom.

Figure 33: FARO ARM – UCF Rapid Prototyping Lab
Figure 34: Measurement of Impeller with FARO ARM

Figure 35: 3D Points for Blade and Splitter in CAM 2
Figure 34 shows the impeller measured with the FARO ARM probe. In order to measure any point on the manufactured part flat or round surfaces on the part was chosen for reference. The location of the tip probe is dictated by these reference surfaces. For the impeller, the two flat faces on the both ends of the shaft and the round surface of the shaft were used as reference. Figure 35 shows the real time visual representation of the 3D space in the CAM 2 software window. The two grids represent the two flat surfaces of the shaft and the pink circle defines the outer diameter of the impeller shaft. The probe was place on the tip of the impeller and splitter blades. The coordinates of the tip of the impeller blade and splitter blades can be seen as 3D green points and 3D yellow points in the Figure 35.

The alignment of bearings in the compressor assembly was also analyzed with the FARO ARM Figure 36. It was observed that the misalignment was insignificant.

Figure 36: Concentricity of Bearings Tested with FARO ARM
A faulty manufactured inlet guide vane was found. Figure 37 shows measurement of IGV. The profile of the incorrect IGV did not match with the solid model. A new IGV was manufactured Figure 38.
2.4 Experimental Results and Discussion

The experimental setup was run with the translational stages and custom designed helical coupler. A blank shaft test was conducted that would test all of the components except for the impeller. This test setup, consisted of utilizing the flexible coupler custom designed by Helical, to connect the motor shaft to a shaft similar in size and weight to that which will be utilized with the existence of the impeller. Reverse rim alignment method was used. The precision adjustment micrometers on the translational stages were used for fine alignment adjustments in the x-y-z plane for the Koford Motor shaft. The Figure 39 below shows the experimental setup.

![Experimental Setup with Translational Stages](image)

Figure 39: Experimental Setup with Translational Stages
Upon aligning the two shafts of the Koford motor and the blank shaft, an initial blank shaft test was performed with the Helical coupler to determine the power consumed to operate all of the components except for the impeller. Several blank shaft tests were conducted with Rimitec coupler previously, resulting in a four fold increase in losses due to the bearings and misalignment. The previously obtained results were to be compared to this current configuration blank shaft test (see Table 41 and Table 42), but various problems occurred during testing that prevented operating at elevated speeds. As the input to the motor controller was increased above 10V, a large spike in current occurred from the failure of one of the bearings. Upon replacing the bearings on the compressor housing, the currents detailed in figure 6 could not be duplicated. Upon configuring subsequent blank shaft tests, the best case current that resulted from the configuration was 5.7A, resulting in 40 Watts to operate at 33,000 rpm. This results from the second blank shaft test result in about a 35% increase in losses to operate at these same input conditions. Also, high levels of vibrations were also apparent on the motor side only, therefore to try and obtain better results the optical motor stage was replaced with the old test setup.

<table>
<thead>
<tr>
<th>Blank Shaft Test 1</th>
<th>8/17/2004</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage Volts</td>
<td>Current Amps</td>
</tr>
<tr>
<td>7</td>
<td>3.7</td>
</tr>
<tr>
<td>8</td>
<td>3.7</td>
</tr>
<tr>
<td>9</td>
<td>3.3</td>
</tr>
<tr>
<td>10</td>
<td>3.8</td>
</tr>
</tbody>
</table>

Table 11: Blank Shaft Test 1 – 8/17/2004
Due to the complications with induced vibration on the optical stage for the motor, the motor was mounted in the old mounting configuration. It was concluded that translational stages was not suitable for the experimental setup. A new experiment setup detailed in Figure 40 was intended to eliminate the vibration effects of translational stage.

<table>
<thead>
<tr>
<th>Voltage Volts</th>
<th>Current Amps</th>
<th>Watts</th>
<th>P=V*I</th>
<th>Speed rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>7</td>
<td>5.7</td>
<td>39.9</td>
<td></td>
<td>33,000</td>
</tr>
<tr>
<td>8</td>
<td>5.8</td>
<td>46.4</td>
<td></td>
<td>37,700</td>
</tr>
<tr>
<td>9</td>
<td>6.2</td>
<td>55.8</td>
<td></td>
<td>42,500</td>
</tr>
<tr>
<td>10</td>
<td>6.4</td>
<td>64</td>
<td></td>
<td>46,300</td>
</tr>
</tbody>
</table>

Table 12: Blank Shaft Test 2 – 8/17/2004

Figure 40: Helical Coupler used with fixed base
A blank shaft test was then attempted with this present configuration and the results reported in Table 13. Comparing the results obtained from this blank shaft test and the previous test performed with the old Teflon coupler, Figure 42 details the resulting consumed power and motor speed. For any motor speed, the current configuration with the Helical coupler resulted in a reduced consumed power. This trend remained consistent for the entire test. But, it could be observed that the use of helical coupler did not reduce the power consumption considerably. Experiments were conducted with curved blade impeller. Pressure, temperature and mass flow rate were measured at various valve positions. The overall compressor characteristics Figure 41 was plot.

Severe vibrations were observed beyond 87555 rpm and the power consumed was over 600 Watts. This phenomenon was succeeded with burning out of the motor controller and failure of the bearings. It was understood that although the coupler allowed misalignments in the axial, angular directions the critical speeds of rotor went down. The simulation in the coupler design had an assumption that the two shafts and the coupler were perfected aligned on a straight line. But, the rigid body modes or the critical speed of the three pieces, that is the coupler, the compressor shaft and motor shaft may not sum up to be the critical speeds for the whole system. Repeated oscillations and fatigue stress developed damaged the bearings and more current needed to drive the controller burnt it up. Also, the backlash provided in the helical coupler might have induced oscillations during spin up from one speed to another. It was therefore concluded that the use of the custom designed helical coupler, translational stage for alignment could not be used for high speed single stage centrifugal compressor testing.
Table 13: Blank Shaft Test with Helical Coupler – 8/26/2004

<table>
<thead>
<tr>
<th>Voltage (V)</th>
<th>Current (I)</th>
<th>Power (W)</th>
<th>Speed (RPM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>3.5</td>
<td>21</td>
<td>29600</td>
</tr>
<tr>
<td>7</td>
<td>3.7</td>
<td>26.32</td>
<td>33500</td>
</tr>
<tr>
<td>8</td>
<td>3.9</td>
<td>31.2</td>
<td>38300</td>
</tr>
<tr>
<td>9</td>
<td>4.5</td>
<td>40.5</td>
<td>43200</td>
</tr>
<tr>
<td>10</td>
<td>5.6</td>
<td>56</td>
<td>47800</td>
</tr>
<tr>
<td>11</td>
<td>5.7</td>
<td>62.7</td>
<td>52700</td>
</tr>
<tr>
<td>12</td>
<td>5.8</td>
<td>69.6</td>
<td>57700</td>
</tr>
<tr>
<td>13</td>
<td>5.2</td>
<td>67.6</td>
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<td>14</td>
<td>6</td>
<td>84</td>
<td>67600</td>
</tr>
<tr>
<td>15</td>
<td>7.5</td>
<td>112.5</td>
<td>72400</td>
</tr>
</tbody>
</table>

Note: The motor ran for five minutes at 30,000 rpm before collecting data points.

Figure 41: Blank Shaft Test Comparison
Figure 42: Compressor Performance Chart
Experimental Results with the new inlet guide vane was compared to the previous runs with the incorrect inlet guide vane. The experimental runs were done at same speeds for both cases. No significant changes were observed. The comparisons are shown in Table 14 and Table 15. The variations in the mass flow rate and power consumed with speed are shown in Figure 43 and Figure 44.

It was confusing that an impeller with a tip gap at the eye that is a few times larger than the blade height can give very similar performance as the one with correct tip gap. This was impossible unless the design is so far off from the appropriate tip gap that nothing matters. It was a known fact that tip gap may only affect efficiency, but it was not important on pressure rise performance.

Based on CFD simulations (Appendix B.) it was found that the diffuser blade angles were inappropriate and a new efficient diffuser was redesigned.
### Table 14: Test Results with Incorrect IGV

<table>
<thead>
<tr>
<th>V (volts)</th>
<th>I (amp)</th>
<th>Vol flow (ft³/sec)</th>
<th>Mdot (gms/sec)</th>
<th>RPM</th>
<th>P2</th>
<th>Valve Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>6.5</td>
<td>52</td>
<td>0.116</td>
<td>4.18992</td>
<td>37134</td>
<td>15.5</td>
</tr>
<tr>
<td>10</td>
<td>9</td>
<td>90</td>
<td>0.146</td>
<td>5.27352</td>
<td>46095</td>
<td>15.9</td>
</tr>
<tr>
<td>12</td>
<td>12.21</td>
<td>146.52</td>
<td>0.158</td>
<td>5.70696</td>
<td>58001</td>
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</tr>
<tr>
<td>13</td>
<td>13.8</td>
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<tr>
<td>14</td>
<td>15.8</td>
<td>221.2</td>
<td>0.174</td>
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<td>63224</td>
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</tr>
<tr>
<td>16</td>
<td>20.04</td>
<td>320.64</td>
<td>0.141</td>
<td>5.09392</td>
<td>71315</td>
<td>17.11</td>
</tr>
<tr>
<td>18</td>
<td>25</td>
<td>450</td>
<td>0.142</td>
<td>5.12904</td>
<td>78816</td>
<td>17.39</td>
</tr>
</tbody>
</table>

### Table 15: Test Results with Correct IGV

<table>
<thead>
<tr>
<th>V (volts)</th>
<th>I (amp)</th>
<th>Vol flow (ft³/sec)</th>
<th>Mdot (gms/sec)</th>
<th>RPM</th>
<th>P2</th>
<th>Valve Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>7</td>
<td>56</td>
<td>0.113</td>
<td>4.08156</td>
<td>37358</td>
<td>15.5</td>
</tr>
<tr>
<td>10</td>
<td>9.4</td>
<td>94</td>
<td>0.144</td>
<td>5.20128</td>
<td>46427</td>
<td>15.9</td>
</tr>
<tr>
<td>12</td>
<td>13.1</td>
<td>157.2</td>
<td>0.171</td>
<td>6.17652</td>
<td>55139</td>
<td>16.26</td>
</tr>
<tr>
<td>13</td>
<td>14.9</td>
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<td>0.171</td>
<td>6.17652</td>
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<td>14</td>
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<td>18</td>
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<td>450</td>
<td>0.103</td>
<td>3.72036</td>
<td>75148</td>
<td>17.27</td>
</tr>
</tbody>
</table>
Figure 43: RPM Comparison of IGVs

Figure 44: Mass Flow Rate Comparison of IGVs
CHAPTER 3. Permanent Magnet Synchronous Motor

3.1 Introduction to PMSM

The Koford motor used for the compressor test seemed to be unsuitable for driving the centrifugal compressor. The bearing losses, coupler losses raged at higher speeds. The motor was not capable of delivering the required output. Also, our application required the centrifugal compressor to be run at 77 K. The material of shaft in the Koford motor was not suitable for low cryogenic temperatures. The need for a custom designed motor was very clear.

A design for a permanent magnet synchronous motor\(^1\) was proposed with output power of 2000 W and running at 200,000 rpm at 77 K. A custom open loop digital DSP based controller\(^2\) was designed for the PMSM.

The mechanical design aspects of the PMSM motor is discussed in this present work.

---

\(^1\) The electrical design, development and testing of PMSM was done by Liping Zheng, Department of Electrical Engineering, University of Central Florida, Orlando.

\(^2\) The open loop controller was designed, developed and tested by Limei Zhao, Department of Electrical Engineering, University of Central Florida, Orlando.
Figure 45: Schematic Diagram of PMSM

<table>
<thead>
<tr>
<th>Specification</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output Power</td>
<td>2000 W</td>
</tr>
<tr>
<td>Operational Speed</td>
<td>200,000 rpm</td>
</tr>
<tr>
<td>Efficiency @ 77 K</td>
<td>89.9 %</td>
</tr>
<tr>
<td>Rotor</td>
<td>Titanium</td>
</tr>
<tr>
<td>Stator</td>
<td>Slot less Laminated</td>
</tr>
<tr>
<td>Wire</td>
<td>Multi-strand Litz Wire</td>
</tr>
<tr>
<td>Magnet</td>
<td>Samarium Cobalt</td>
</tr>
<tr>
<td>Winding</td>
<td>6 turns/phase/pole</td>
</tr>
<tr>
<td>Pitch</td>
<td>1 to 15</td>
</tr>
<tr>
<td>Stator Outer Diameter</td>
<td>30 mm</td>
</tr>
<tr>
<td>Stator Inner Diameter</td>
<td>23 mm</td>
</tr>
<tr>
<td>Rotor Diameter</td>
<td>16 mm</td>
</tr>
<tr>
<td>Permanent Magnet</td>
<td>19 mm</td>
</tr>
</tbody>
</table>

Table 16: Specifications and Features of the PMSM
3.2 Structural Design of Rotor

The structural design of the rotor involves material selection, stress analysis, thermal analysis, rotordynamic analysis, fabrication issues and assembly.

3.2.1 Material Selection of Rotor

The selection of the shaft material was based on certain major factors.

1) The shaft material should be non-magnetic and poses high yield strength to withstand the centrifugal stress developed in the shaft-magnet system while operating at 77 K and 200,000 rpm.

2) The assembly of the shaft and magnet should be done such that the magnet is at interference fit while operating at 77K and 200,000 rpm. An unequal expansion-contraction in shaft and magnet at 77 K as well as an expansion of the shaft due to centrifugal force while rotating at 200,000 rpm may cause the magnet to be loose fit and cause mechanical failure.

3) The shaft material should be machinable.

The materials taken in consideration were Titanium, Inconel and MP35N. All these materials have high yield Strength at 77 K. But density of Titanium is less [2]. The co-efficient of thermal expansion of Titanium is almost same as that of magnet material SmCo. Thermal conductivity of Titanium is more than other metals considered. The material used for shaft was commercially available Titanium 6 Al -4 V. This material poses a yield strength value of 1400 MPa at 77 K. It has similar co-efficient of thermal
expansion as that of magnet material (SmCo). This alloy has less density compared to other materials that were taken into consideration and is fairly machinable.

The following table shows the comparison of various alloys taken in consideration for the rotor of the motor.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Magnet SmCo</th>
<th>Titanium Ti-6Al-4V</th>
<th>INCONEL® 718SPF™ Nickel Superalloy, Annealed + Aged</th>
<th>MP35N</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young’s modulus N/m²</td>
<td>147e+9</td>
<td>122e+9</td>
<td>205e+9</td>
<td>234e+9</td>
</tr>
<tr>
<td>Poisson's Ratio</td>
<td>0.310</td>
<td>0.30</td>
<td>0.294</td>
<td></td>
</tr>
<tr>
<td>Bulk or Shear Modulus N/m²</td>
<td>0.299e+11</td>
<td>0.45e+11</td>
<td>0.80e+11</td>
<td>0.80e+11</td>
</tr>
<tr>
<td>Density kg/m³</td>
<td>8500</td>
<td>4420</td>
<td>8220</td>
<td>8430</td>
</tr>
<tr>
<td>Tensile Strength N/m²</td>
<td>39.2e+6</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Compressive Strength N/m²</td>
<td>833e+6</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Yield Strength N/m²</td>
<td>0.11e+9</td>
<td>0.14e+9</td>
<td>1.39e+9</td>
<td>1.62e+9</td>
</tr>
<tr>
<td>Coefficient of thermal expansion C 7e-6/°C</td>
<td>7e-6</td>
<td>9e-6</td>
<td>14e-6</td>
<td>12.8e-6</td>
</tr>
<tr>
<td>Thermal conductivity W/m°C</td>
<td>10.5</td>
<td>22</td>
<td>11.4</td>
<td>11.2</td>
</tr>
<tr>
<td>Specific heat C J/Kg°C</td>
<td>418</td>
<td>586</td>
<td>435</td>
<td></td>
</tr>
</tbody>
</table>

Table 17: Comparison of properties of material for rotor
3.2.2 Shape and Dimensions of the rotor

Figure 46: Schematic Diagram of Rotor

Figure 47: Dimensions of the Rotor
### 3.2.3 Stress Analysis

The rotor is the only moving part of the motor. It is subjected to various mode of failure. The shaft failure can be evaluated as follows,

1. **Shear Stress**

   It produces twisting in the shaft layers. Failure occurs when the maximum shear stress developed in the shaft material due to twisting exceeds the yield strength of the material in shear. According to the maximum shear stress theory [20], the inner diameter of the shaft is given by,

   \[
   d = \left[ \frac{D^4 - a \cdot F \cdot 32 \cdot T \cdot D}{\pi \cdot \tau_{\text{max}}} \right]^{\frac{1}{4}}
   \]

   The values considered for shear,

   - Power being transmitted = \( P = 2000 \) W
   - Speed of rotation of the shaft = \( N = 200000 \) rpm
   - Angular velocity of rotation = \( \omega = 2 \cdot \pi \cdot N/60 \)
   - Torque developed = \( T = P/ \omega = 0.095 \) N.m
   - Maximum allowable shear stress = \( \tau_{\text{max}} \)
   - Factor of Safety = \( F = 3 \)
   - \( a = \) ASME factor for shaft design for shear = \( \frac{3}{4} \)
   - Shaft cross-section – Hollow Shaft
   - No vertical shear stress
   - Rigidity modulus of the shaft m/l = \( \gamma = 117 \) GPa
   - Angle of shaft twist because of torsion = \( \alpha \) (maximum for the outer layer)
For a $\tau_{\text{max}}$ value of 20305 psi (for a Titanium), $d = 15.846$ mm

Thickness of the hollow shaft $= t = (D - d)/2 = 0.077$ mm << 0.5 mm

So, the shaft would not fail under pure shear.

Also, Angle of Twist $\alpha = 584 \times \text{Torque} \times \frac{1}{D^4 \times \gamma} \Rightarrow \alpha = 1.847 \times 10^{-4}$

2. Bending Stress

The values considered for the bending stress are as follows,

- Elastic modulus of shaft $= 116,000$ MPa
- Poisson’s ratio for shaft $= \nu = 0.34$
- Ultimate tensile strength of the shaft $= 220$ MPa
- Elastic modulus of permanent magnet $= 150,000$ MPa
- Poisson’s ratio for permanent magnet($\nu$) $= 0.3$
- Ultimate tensile strength of the permanent magnet $= 82.7$ MPa
- Bending moment due to impeller weight $= M = 3.74$ N-mm
- Shaft cross-section – Hollow Shaft
- Permanent magnet cross-section – Solid Shaft
- Density of the shaft m/l ($\rho_{\text{shaft}}$) $= 4500$ kg/m3
- Density of the permanent magnet ($\rho_{\text{magnet}}$) $= 7500$ kg/m3

For pure bending analysis, the shaft is considered as a composite material with the cross-section as shown below,
Maximum bending stress ($\sigma_{\text{max}}$) = $M/Z$, where $Z$ = section modulus.

$\sigma_{\text{max}}$, Titanium = 0.037 MPa << Ultimate tensile strength of shaft.

$\sigma_{\text{max}}$, Permanent Magnet = 0.048 MPa << Ultimate tensile strength of permanent magnet.

So the shaft would not fail under pure bending.

3. Fracture toughness

Fracture Toughness is an issue at cryogenic temperatures.

It is defined $k = cs(\sqrt{\pi*c})\alpha$, where

‘$k$’ is the ‘Critical Stress Intensity’,

$Cs$ – Critical Stress,

$c$ – crack length,

$\alpha$ – geometry factor (depends on the cross-section of the member)

‘$k$’ depends on the Bending Stress developed.
4. Centrifugal and Thermal Analysis

In order to design the shaft and assemble the magnet inside the shaft, it is absolutely necessary to determine the individual effect of centrifugal and thermal effects [18], [19] and [23]. The titanium shaft and Samarium cobalt magnet will radially expand due to centrifugal force experienced by it at 200,000 rpm. Unequal expansion (or contraction) will result a loose (or tight) magnet inside the shaft. Also, the grade of titanium and samarium cobalt magnet chosen should have yield stress within a safety margin.

The cross-section considered for the shaft and magnet is as shown in Figure 49.

![Figure 49: Section of the Titanium Rotor with SmCo magnet](image)

The magnet was intentionally made oval at the poles to ensure that it doesn’t slip inside the shaft. Also the roundness of the magnet at the poles would facilitate the assembly of shaft and magnet. The solid model of the rotor was created in Pro/engineer and Autodesk Inventor and was analyzed in IDEAS -9 and COSMOS.
1) Centrifugal stress

Figure 50: The centrifugal stress developed in the shaft was 728 MPa.

Figure 51: The centrifugal stress developed in the magnet at 200,000 rpm is 261.2 MPa.
Figure 52: The thermal stress developed in the shaft at 77 K was 329 MPa.

Figure 53: The thermal stress developed in the magnet at 77 K was 130 MPa
- Stress due to centrifugal force in shaft rotating at 200,000 rpm = 728 MPa
- Stress due to centrifugal force in magnet rotating at 200,000 rpm = 251.2 MPa
- Thermal stress developed in shaft due to operating at 77 K = 329 MPa

The Total Stress = 1308.2 MPa < Titanium Grade Yield Strength 1420 MPa

So the titanium shaft would not fail.

Also,

- Thermal Stress developed in magnet = 130 MPa
  < Compressive Strength = 833 MPa.

So, the magnet would not crack or crumble to powder.
3.2.4 Rotordynamic Analysis of Rotor

Rotordynamic analysis was done prior to fabrication. Two models were considered in these. The model is based on the first conceptual design Figure 47. The Figure 54 is a schematic of the FE rotordynamic model\(^3\) with 9 elements and 40 degrees of freedom. The upper half of the model shows the mass distribution while the lower half shows the stiffness distribution. The mass distribution shows the magnet counted for the mass, but not for the stiffness. The model shown has a certain bearing span. The bending critical speeds are quite sensitive to both bearing stiffness and bearing span. Ball bearing stiffness can vary widely depending on preload, operating internal clearance, etc. In this case the bearing stiffness can vary anywhere from 10000 lb/in to 200000 lb/in, resulting in fairly large variations in the 1st critical speed (\(\sim 155,000 \text{ rpm}\) to \(\sim 250,000 \text{ rpm}\)). Ball bearings have very little damping and hence operating near critical speeds results in very large transmitted forces.

\(^3\) Rotordynamic analysis was done with guidance of Dr. N. Arakere, University of Florida, Gainsville.
Figure 54: FE Model 1 for Rotordynamic Analysis

Figure 55: 1st Bending Critical Speed – 154933 rpm

Figure 55 represents the 1st Bending critical speed. Generally the 1st bending critical speed is kept at least 30% higher than the highest operating speeds. The 1st mode (bending mode) strain energy Figure 56 in the shaft represents nearly 94% of the total energy, with 6% of the strain energy due motion at the bearings. However, even this motion at the bearings will transmit large forces. The curves in the critical speed map represent the three bending modes. As the bearing stiffness increases the critical speeds go higher.
Mode No. = 1, Critical Speed = 154933 rpm = 2582.21 Hz
Potential Energy Distribution (s/w=1)
Overall: Shaft(S) = 93.55%, Bearing(Brg) = 6.45%

Figure 56: Strain Energy Distribution – 154933 rpm

Figure 57: Critical Speed Map – Model 1
The second model Figure 58 represents a modification of first model in which an attempt has been made to increase the first bending critical speed. The model 2 has a slightly smaller bearing span (by 14 mm) from model 1. The results changes considerably in the critical speed locations. The first mode increases considerably, to around 250k rpm.

Figure 58: FE Model 2 for Rotordynamic Analysis

Critical Speed Mode Shape, Mode No = 1  
SpinWhirl Ratio = 1, Stiffness: Kxx  
Critical Speed = 245290 rpm = 4088.17 Hz

Figure 59: 1st Bending Critical Speed – 245290 rpm
Mode No. = 1, Critical Speed = 245290 rpm = 4088.17 Hz
Potential Energy Distribution (s/w=1)
Overall: Shaft(S)= 85.57\%, Bearing(Brg)= 14.43\%

Figure 60: Strain Energy Distribution – 245290 rpm

Critical Speed Map

Figure 61: Critical Speed Map – Model 2
For the 1st mode (bending mode) strain energy in the shaft represents about 86% of the total energy, with 14% of the strain energy due to motion at the bearings. The first three critical speeds can vary a fair amount in the bearing stiffness range. We are only concerned about the 1st critical speed since it is the one near the max operating speed of 200,000 rpm. Bearing stiffness has to be around 100,000 lb/in to get the 1st mode above 200k rpm. The fabrication of the rotor of the Permanent Magnet Synchronous Motor was based on the second model shown above.
### 3.2.5 Assembly of the Rotor

Centrifugal Stress Analysis was done using FEM software at the operating speed. The shaft was observed to expand more than the magnet and would result in a clearance fit. A high risk of crumbling of the magnet while shuddering inside the shaft prevails. Thus, interference fit was required.

The Titanium Shaft has Coefficient of thermal expansion = 8.8 to $9e-6$ / °C

The Magnet(SmCo) has Coefficient of thermal expansion = 8.0 to $11e-6$ / °C

The tolerances on the magnet are specified as shown in Figure 62.

![Figure 62: Tolerances on SmCo Magnet](image)

The tolerance of the shaft was determined from this information. We required at least an interference fit of at least 0.01 mm between the shaft and magnet at room temperature running at 200,000 rpm. From FEM analysis it was seen that the centrifugal rotation would not expand the shaft walls more than 0.01 mm. Also, it was observed that some parts of the titanium shaft may shift inwards to compensate the stretching of the material.
on the heavier sides. And that shifting would still hold the magnet. Even if expansion due
to centrifugal action exceeds 0.01 mm, the magnet will still be held by two caps.

Figure 63: Results of Centrifugal Force on Titanium Shaft

Figure 64: Tolerances on Shaft
Table 18: Assembly Issues of PM and Shaft

<table>
<thead>
<tr>
<th></th>
<th>10 mm direction</th>
<th>13 mm direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>PM dimensions @ RT (mm)</td>
<td>9.995</td>
<td>10.005</td>
</tr>
<tr>
<td></td>
<td>12.995</td>
<td>13.005</td>
</tr>
<tr>
<td>Shaft inner dimensions @ RT (mm)</td>
<td>9.985</td>
<td>9.995</td>
</tr>
<tr>
<td>Interference @ RT (mm)</td>
<td>0.000</td>
<td>0.010</td>
</tr>
<tr>
<td>PM shrink @ 77K (mm)</td>
<td>0.022</td>
<td>0.020</td>
</tr>
<tr>
<td>Shaft expand (mm)</td>
<td>0.040</td>
<td>0.052</td>
</tr>
<tr>
<td>Clearance on each side while assembling (mm)</td>
<td>0.021</td>
<td>0.031</td>
</tr>
<tr>
<td>PM expansion @ 200,000 rpm (mm)</td>
<td>0.0018</td>
<td>0.0026</td>
</tr>
<tr>
<td>Shaft expansion @ 200,000 rpm (mm)</td>
<td>0.0034</td>
<td>0.0034</td>
</tr>
<tr>
<td>Interference @ 200,000 rpm (mm)</td>
<td>-0.001</td>
<td>0.009</td>
</tr>
<tr>
<td>CTE of PM (1/°C)</td>
<td>1.10E-05</td>
<td>0.00E-06</td>
</tr>
<tr>
<td>CTE of Shaft (1/°C)</td>
<td>1.00E-05</td>
<td></td>
</tr>
<tr>
<td>Shaft temperature while assembling (°C)</td>
<td>400.000</td>
<td></td>
</tr>
</tbody>
</table>

It was calculated that heating the shaft to 400°C and cooling the magnet to liquid nitrogen temperature would produce the necessary clearance to insert the magnet in the shaft. On cooling the interference fit would be established. Residual stresses arising due to this process of assembling was calculated. The magnet was tested for failure by compressive stress due to interference fit between the shaft and magnet. The centrifugal stress and residual stress would contribute to the total stress developed while the shaft is in operation. The maximum total stress observed was 852 MPa. This value was much less than the yield strength of Titanium grade that was used to build the shaft.
Figure 65: Von Mises Stress Analysis – 200,000 rpm
3.2.6 Fabrication of the Rotor

The fabrication of the motor includes the following parts.

1) Rotor 2) Stator and 3) Housing

The rotor of the motor was fabricated out of Titanium 6Al-4V. The shaft was machined by EDM process.

Figure 66: Drawing of Titanium Shaft

Figure 67: Hollow Shaft and Magnet
A metallic fixture Figure 68 was used to assemble the shaft and magnet. The shaft was heated 400°C and placed on it as shown. The magnet was cooled to 77 K with Liquid Nitrogen was dropped inside shaft. The position of the magnet inside the shaft was located by the projected part from the fixture base.

![Figure 68: Shaft – Magnet Assembly Structure](image)

The motor housing was machined with commercially available aluminum.

![Figure 69: Aluminum Casing with Stator inside](image)
The titanium plugs were partially machined by EDM and regular machining process.

Figure 70: Drawing of Titanium Plug

Figure 71: Drawing of Rotor Assembly
The titanium shaft and titanium plug was welded by EDM at the junction. Figure 72 shows the rotor held in a chuck inside the vacuum EDM chamber.

Figure 72: Electron Beam Welding of Rotor

Figure 73: Stator and Winding
Figure 74: Casing, caps and the assembled shaft
3.2.7 Motor Generator Test Set-up and Results

The permanent magnet synchronous motor built was tested successfully under no – load condition to speeds 200,000 rpm. The following Figure 75 shows the input motor power versus speed.

![Figure 75: No Load Test – PMSM](image-url)
In order to determine the efficiency of the motor it was required to run the motor under loaded conditions. A motor – generator set was built on a single shaft as no high speed coupler was available. Power was supplied to the motor and while the whole shaft rotated producing power on the generator side. This process gives the efficiency of the motor. The same principle applies to the integrated compressor – motor design which is discussed in the next chapter.

Figure 76: Motor – Generator Components

Figure 77: Motor – Generator Assembly
The following Figure 78 shows that the motor – generator set had 88.8 % efficiency at 109800 rpm which was the speed required for the integrated compressor – motor setup.

<table>
<thead>
<tr>
<th>Speed (krpm)</th>
<th>Back EMF (V)</th>
<th>Motor Input Current (A)</th>
<th>Motor Input Power (W)</th>
<th>Motor Input Voltage (V)</th>
<th>Motor Input Current (A)</th>
<th>Motor Input Power (W)</th>
<th>Torque (N.m)</th>
<th>M/G Efficiency (%)</th>
<th>Winding Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.1</td>
<td>0.35</td>
<td>6.0</td>
<td>6.8</td>
<td>0.51</td>
<td>2.8</td>
<td>4.3</td>
<td>0.008</td>
<td>63.9</td>
<td>23.6</td>
</tr>
<tr>
<td>10</td>
<td>0.57</td>
<td>8.9</td>
<td>16.5</td>
<td>0.85</td>
<td>4.6</td>
<td>11.8</td>
<td>0.013</td>
<td>71.5</td>
<td>25.1</td>
</tr>
<tr>
<td>20.1</td>
<td>1.15</td>
<td>16.7</td>
<td>61.6</td>
<td>1.70</td>
<td>9.3</td>
<td>47.1</td>
<td>0.026</td>
<td>76.4</td>
<td>27.5</td>
</tr>
<tr>
<td>30.5</td>
<td>1.74</td>
<td>24.5</td>
<td>137.2</td>
<td>2.53</td>
<td>14.2</td>
<td>107.6</td>
<td>0.038</td>
<td>78.5</td>
<td>33.9</td>
</tr>
<tr>
<td>36.6</td>
<td>2.09</td>
<td>28.7</td>
<td>192.7</td>
<td>3.03</td>
<td>16.8</td>
<td>152.7</td>
<td>0.045</td>
<td>79.2</td>
<td>38.7</td>
</tr>
<tr>
<td>42.7</td>
<td>2.43</td>
<td>32.0</td>
<td>250.8</td>
<td>3.51</td>
<td>19.2</td>
<td>202.0</td>
<td>0.050</td>
<td>80.5</td>
<td>44.5</td>
</tr>
<tr>
<td>48.8</td>
<td>2.78</td>
<td>34.8</td>
<td>310.8</td>
<td>3.96</td>
<td>22.0</td>
<td>261.1</td>
<td>0.056</td>
<td>84.0</td>
<td>50.8</td>
</tr>
<tr>
<td>54.9</td>
<td>3.13</td>
<td>37.6</td>
<td>376.7</td>
<td>4.39</td>
<td>24.5</td>
<td>322.4</td>
<td>0.061</td>
<td>85.6</td>
<td>54.7</td>
</tr>
<tr>
<td>61</td>
<td>3.48</td>
<td>30.5</td>
<td>333.1</td>
<td>5.36</td>
<td>18.0</td>
<td>289.4</td>
<td>0.049</td>
<td>86.9</td>
<td>42.7</td>
</tr>
<tr>
<td>73.2</td>
<td>4.17</td>
<td>35.5</td>
<td>465.8</td>
<td>6.39</td>
<td>21.1</td>
<td>404.9</td>
<td>0.057</td>
<td>86.9</td>
<td>50.3</td>
</tr>
<tr>
<td>97.6</td>
<td>5.56</td>
<td>46.2</td>
<td>809.5</td>
<td>8.36</td>
<td>27.7</td>
<td>695.2</td>
<td>0.073</td>
<td>85.9</td>
<td>71.0</td>
</tr>
<tr>
<td>109.8</td>
<td>6.26</td>
<td>37.0</td>
<td>717.5</td>
<td>10.00</td>
<td>21.2</td>
<td>637.2</td>
<td>0.059</td>
<td>88.8</td>
<td>58.3</td>
</tr>
</tbody>
</table>

Table 19: Motor – Generator Performance

Figure 78: Motor – Generator Efficiency Curve
CHAPTER 4. DESIGN OF INTEGRATED CENTRIFUGAL COMPRESSOR AND PERMANENT MAGNET SYNCHRONOUS MOTOR

4.1 Conceptual Design of Integrated Compressor Motor Structure

An attempt has been made in the present work to integrate the single stage centrifugal compressor and permanent magnet motor described in the previous chapters. It was very clear from the test results that the compressor – motor system coupled with helical coupler, aligned with translational stage didn’t fair well. It was understood that the use of a custom designed coupler and reverse rim alignment could not yield a high speed rotor system. Also, a major obstacle had to be confronted in finding a suitable driver for the single stage centrifugal compressor. The Koford electric motor was incapable of driving the compressor overcoming the losses with the present coupler and the alignment procedure. The following table makes it very clear that in considering cost and complexity, electric motor as a driver would be a good trade-off among the drivers.

<table>
<thead>
<tr>
<th></th>
<th>Electric Motor</th>
<th>Air- Turbine</th>
<th>Turbocharger</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Speeds</strong></td>
<td>158,000 rpm</td>
<td>250,000 rpm</td>
<td>200,000 rpm</td>
</tr>
<tr>
<td><strong>Accessories</strong></td>
<td>Controller</td>
<td>TC-Control System</td>
<td>Speed Controller</td>
</tr>
<tr>
<td><strong>Coupling</strong></td>
<td>Flexible Coupler</td>
<td>Spline Shaft</td>
<td>Spline Shaft</td>
</tr>
<tr>
<td><strong>Cost</strong></td>
<td>$2500</td>
<td>$30,000</td>
<td>$35,000</td>
</tr>
</tbody>
</table>

*Table 20: Comparison of various drivers*
The integration of the compressor impeller shaft and motor shaft would in fact,

1. eliminate the use of coupler
2. reduce the number of bearings and simplify to a two bearing system
3. increase the stiffness and also raise the critical speed of the system

High cost of manufacturing of new impeller shaft for integration was not recommended. The existing curved blade impeller shaft was machined and modified with attachments and integrated with a new motor shaft to form the integrated rotor. All the components of the rotor structure are described later.
4.2 Rotordynamic Analysis of the Integrated Rotor Structure

It was a prime requirement to have the rotordynamic analysis of the new rotor system with new bearing locations. Incorporation of five components to form the rotor structure may bring down the rigid body modes and critical speeds of the system. Several models varying in dimensions, material distribution, manufacturing methods and bearing locations were analyzed to form the rotor system. Final versions of 2D – rotordynamic analysis with software DyRoBeS are shown below.

The required dimensions of the rotor to model in the software are as shown in the Figure 79 below.

![Figure 79: Rotor System Diagram](image)
The bearing stiffness used in this system was 2000 N/mm. Figure 80 represents the 2D model of the integrated rotor system with bearings located at station 2 and station 8. Figure 81 represents the first rigid body mode at 31,056 rpm and Figure 82 strain energy distribution confirms the rigid body mode.

Critical Speed = 31056 rpm = 517.61 Hz

Figure 80: Model - Integrated Rotor System

Figure 81: First Rigid Body Mode – 31056 rpm
Figure 82: Strain Energy Distribution – 31056 rpm

Figure 83 represents the second rigid body mode at 64,848 rpm and Figure 84 shows the strain energy distribution at that speed.

Critical Speed = 64848 rpm = 1080.79 Hz

Figure 83: Second Rigid Body Mode – 64848 rpm
The first bending mode was observed at 244,852 rpm and is shown in Figure 85 and the corresponding strain energy distribution is shown in Figure 86.

Critical Speed = 244,852 rpm = 4030.87 Hz
The second bending mode was observed at 434,289 rpm and is shown in Figure 87 and the corresponding strain energy distribution is shown in Figure 88.

Critical Speed = 434429 rpm = 7240.48 Hz

Figure 86: Strain Energy Distribution – 244,852 rpm

Figure 87: Second Bending Mode – 434429 rpm
Figure 89 shows that with increasing stiffness, critical speeds increase. The topmost line represents the 1st bending mode, the bottom two are rigid body modes. It was clear from the rotordynamic analysis that the rotor structure has the first bending critical speed at 244, 452 rpm which was far beyond the operational speed of 108,000 rpm.
4.3 Fabrication of the Integrated Rotor Structure

The original impeller was manufactured with aluminum while the motor shaft was manufactured with titanium. Assembling the two different materials was a challenge. Welding of aluminum and titanium could not be done. The motor rotor consisted of two parts, the titanium plug and the titanium shaft housing the magnet. The titanium plug and titanium shaft were machined by EDM process and were welded together by Electron Beam Welding. An aluminum plug was used to connect the impeller to the titanium shaft. The aluminum plug was fabricated by EDM process and was welded to Aluminum Impeller. The two sets were assembled by press fit. The following Figure 90 shows the components of the integrated rotor.

![Components of Five Component Integrated Rotor](image-url)
The assembled shaft with the bearings mounted is shown in Figure 91.

Figure 91: Fabricated Five Component Integrated Rotor
4.4 Test Rig Structure

A new test rig structure was designed and fabricated. The previous translational stages were eliminated with Newport Angle Brackets. Aluminum plates were fabricated to mount the integrated compressor structure. The components of the structure are described in the following text.

Figure 92 represents the solid model of the integrated compressor – motor structure.

![Solid Model of Test Rig – Integrated Structure](image)

Figure 92: Solid Model of Test Rig – Integrated Structure
4.4.1 Components and Fabrication of the Integrated System

Figure 93 shows the section of the test rig with the label of each component of the integrated compressor – motor system.
List of Components (Labels from Figure 93)

1) Optical Table or Base
2) Newport angle bracket
3) Motor Plate
4) Bearing Holder
5) Spring Loader
6) Motor Cooling Jacket
7) Bearing Loader
8) Rotor Assembly
9) Stator
10) Guide Board
11) Top plate
12) Gas enclosure
13) Adjustable Inlet Guide Vane
14) Impeller
15) Diffuser
16) Diffuser Cover
17) Diffuser Plate
18) Compressor Plate
1) Optical Table or Base – is a portable base used to assemble the test-rig.

2) Newport Angle Brackets – supports the compressor – motor assembly.

3) Motor Plate – supports the bearing holder and motor assembly to the angle brackets.

4) Bearing Holder – houses the bearing on the motor side and holds the motor cooling jacket.

5) Spring Loader – it is used to pre-load the bearings on the motor side. Figure 94 shows the spring loader with the spring mounted on it.

6) Motor Cooling Jacket – cools the stator and windings. It has water cooling channels drilled through it. Figure 95 shows the motor cooling jacket with cooling inlet and outlet holes. The motor cooling jacket is threaded to mount the guide board assembly.
7) Bearing Loader – supports the spring and transmits the preload applied by the spring loader structure to the inner race of ceramic ball bearing on the motor side Figure 96. Figure 97 shows the edge of the bearing loader which supports the spring mounted by spring loader.
8) Rotor Assembly – consists of the integrated compressor – rotor structure as discussed in the previous section.

9) Stator – comprises of the iron core and windings. The motor cooling jacket houses and cools the stator.
10) Guide Board – is threaded on to the motor cooling jacket. It is a structure designed to hold the top plate, gas enclosure and inlet guide vane together and also facilitate axial movement with help of the threaded structure.

![Figure 99: Guide Board](image)

11) Gas Enclosure – is fabricated to house the inlet guide vane and also to incorporate various gases with help of 1/16” NPT connection as shown in Figure 100.

![Figure 100: Gas Enclosure](image)
12) Top plate – was fabricated in two parts and has the contour of the IGV blades as shown in Figure 105. The two parts of the top plate has the opening through which the integrated shaft passes.

![Top Plate](image)

**Figure 101: Top Plate**

13) Adjustable Inlet Guide Vane

This is an unique feature in this design. The inlet guide vanes structure was equipped with threads which facilitates axial movement. Tip clearance or the tip gap between the impeller blades and shroud can be controlled. Tip clearance has significant effect on the pressure rise of the compressor. Figure 102 shows the inlet guide vane threaded onto the enclosure part. Figure 103 shows the assembly procedure of the top plate, gas enclosure and inlet guide vanes.
14) Impeller – was discussed with rotor assembly in previous section.

15) Diffuser – Based on CFD analysis (Appendix C), the old diffuser was found to be inefficient and a new diffuser was designed and fabricated.
16) Diffuser Cover – encloses the diffuser to create an annular flow passage.

17) Diffuser Plate – supports the diffuser and diffuser cover. It is connected to the collector which allows the flow to the mass flow meter.

18) Compressor Plate – supports the compressor side assembly
4.5 Experimental Test Rig and Accessories

The experimental test rig consists of several components and accessories required to drive the motor and instruments used for flow measurements. The experimental setup can be classified as following:

1. Motor Control Software and Emulator
2. Motor Controller and Low Pass Filter

4.5.1 Code Composer and Emulator

![Figure 106: Code Composer]
Code composer is an integrated development environment (IDE) for TI DSP. It is used to develop and compile DSP source code. The code that we see and can be understood is call source code which is written in TI DSP assembly language or C. We have to compile this source code into executable code that can be run in DSP. That is, we can understand source code but we can not read executable code which is in binary format while DSP chip can understand and run binary code and doesn’t know anything about our source code. We use code composer to develop our source code. If our program has been well developed, we do not need code composer to run motor just like all commercial motor controllers. It sends out the compiled executable code to the emulator. The emulator executes the code and it sends out the pulse width modulation (PWM) signal to the controller. The PWM signals are used to control the on/off of the six MOSFETs in order to output sinusoidal waveforms to drive the motor.

4.5.2 Motor Controller and Low Pass Filter

An open loop controller was used to drive the motor. The major components of the controller are MOSFETs, gate drive ICs and DSP board. The DSP board is used to output PWM signal. Gate drive ICs are used to drive MOSFETs. Low pass filter are three inductors which are used to remove harmonics, so the current waveforms will be more like a sinusoidal waveform, thus the efficiency can be increased.
Figure 107: Motor Controller connected with DSP and Emulator

Figure 108: Low Pass Filter
Knowing input power to the compressor is important because that will determine the efficiency of the compressor input. In previous experiments, the power was measured using the line current and back emf. The formula has assumption of around 5% loss due to harmonic distortion. This calculation sometime doesn’t give accurate results when there is voltage and current measurement error present. It gets even worse if there are more harmonic presents in the line. Therefore, Shark 100 meter was bought which is shown in the following Figure 109.

![Power meter](image)

**Figure 109: Power meter**

This meter measures the three phase power either by directly connected to the voltage and current sources. The wiring diagram is a WYE, 4-Wire with No PTs, 3 CTs. Three CT in three lines were placed and connected at the CT connection lead. Similarly, three phase voltage were connected and the common wire in the voltage connection. Voltage Inputs are connected to the back of the unit via optional wire connectors. The
connectors accommodate up to AWG#12/2.5mm wire. The meter’s Ground Terminals were connected directly to the installation’s protective earth ground.

4.5.3 Flow Measurement Devices

Pressure transducers, manufactured by Setra, Inc. Figure 110, capable of measuring pressures in the range of 0-5 pounds per square inch gage as well as 0-15 pounds per square inch gage pressure were obtained. The transducers required an excitation voltage of 12 Volts. Each pressure transducer was calibrated by supplying a known constant pressure and measuring the output voltage.

Figure 110: Setra Wet – Wet Differential Pressure Transducers
The Sierra Innova Inline Vortex Mass Flow Meter is shown in the following Figure 111. The mass flow meter has inbuilt shredder bar and a sensor head. The shredder bar creates vortices in the flow stream. The velocity sensor head placed downstream of the shredder bar measures the frequency of the vortices and therefore measures the total fluid volume. The frequency of the flow wakes was measured in the mass flow meter device by incorporating a piezoelectric element that sensed the alternating lift forces produced by the vortices (Sierra, 1997).
4.5.4 Bearing Fit, Pre-load Calculations and Assembly Procedure

Bearing fit and mounting of the bearings is an important procedure in the assembly. The bearing carries the load from the rotor structure. The ball bearing stiffness depends on load, clearance, number of balls, contact angle. The shaft as well as the inner and outer races of the bearing expands due to joint effect of the centrifugal force due to rotation of the shaft and thermal expansion. The bearing fit must be precisely calculated and mounting procedures to be followed to elongate life time of the bearings. It was observed that for a 50°C temperature rise and 100,000 rpm speed, due to combined centrifugal and thermal effects, the shaft grows 3 microns radially. The inner race grows 3.85 microns. The bearing internal clearance specified by the manufacturer is 0.0002” – 0.0005”. The calculations are shown in Appendix E. The fit between the shaft and bearing was calculated to be a light press fit. The shaft was precision grinded.
Figure 112: Bearing Mounting Procedure (Courtesy: IBSCO),[7]
4.5.5 Alignment Procedure for the Integrated Compressor-Motor System

The alignment was done with two precision shafts. The bigger shaft had a tight fit with the two bearing holes on the compressor and the motor side. The shaft was inserted from the motor side and extended to the compressor side. The bearing hole on the motor side is not accessible, so the bearing hole on the compressor side was used to visually align the Newport precision angle bracket. The base bolts and metallic shims were used to align the system. The shaft when in place and aligned quite well could be rotated freely with hand. If the alignment of the two holes is not good then the shaft could not be freely rotated and manipulation on the base was required. A second smaller precision blank shaft imitating the actual rotor was used along with two bearings to further align the system. Two bearings were placed in the bearing holes. The smaller shaft having the same dimensions as that of the inner diameter of the bearing was inserted from the motor side extending to the compressor side. Dial indicators were used to measure the run-off of the shaft. A run-off precision up to third decimal places in inches could be achieved by this procedure.

![Figure 113: Alignment of bearing holes with precision shaft](image-url)
The assembled structure of the integrated centrifugal compressor and permanent magnet synchronous motor is shown in the following figure.

![Integrated Compressor – Motor Structure](image)

**Figure 114: Integrated Compressor – Motor Structure**

The assembled structure of the test–rig with the gas enclosure and adjustable IGV is shown in figure. The gas enclosure is threaded onto the motor cooling jacket which helps in axial movement. The adjustable IGV is threaded onto the gas enclosure assembly for axial movement. The adjustable IGV is the shroud of the impeller and controls the tip clearance between impeller blade and the shroud. The diffuser cover is not used in the test set-up during the free spin tests. The free spin tests are done to estimate the power required by the rotor while no aerodynamic work is done by the impeller.
Motor Jacket
Stator inside

Gas enclosure with adjustable IGV
to control tip clearance

Figure 115: Free Spin Test Integrated Compressor – Motor
CHAPTER 5. TEST RESULTS AND DISCUSSION

5.1 Test results - five piece integrated rotor

Free spin tests were conducted with five piece integrated rotor. The free spin test results were successful only to 42,000 rpm. The rotor was observed to be wobbling near the aluminum impeller plug by 0.008 inches. A hair crack Figure 116 was visible at the joint of the impeller plug and the impeller where the EBM welding was made.

![Hair Crack in Rotor](image)

Figure 116: Hair Crack in rotor

It was concluded that a minor misalignment or radial displacement of shaft during passing of rigid body mode around 31,056 rpm caused the failure at the weak junction. This five piece rotor assembly had a press fit between aluminum and titanium pieces. The rotor had a stress concentration at the welded junction where the failure occurred. It was clear that a single material shaft system is necessary.
5.2 Test results – Two piece rotor – Threaded Impeller

A two piece rotor assembly with the cast aluminum impeller threaded on to a stainless steel shaft was machined. Stainless steel was used instead of titanium to reduce manufacturability difficulties with titanium and to reduce machining cost. Titanium material was chosen for the rotor at initial stages of the design to use it for cryogenic temperatures around 77K. But, the present compressor test would be executed with air as the fluid at room temperatures. Stainless steel had higher density than titanium but trade-off could be made considering the machining cost and manufacturability. Stress and rotordynamic analysis confirmed the usage of the stainless steel as the material.

Stainless steel 303 was used to machine the plug and shaft of the rotor. The cast impeller was modified and internal threads were machined. A stainless steel ring was machined to support the impeller. The following Figure 117 shows the parts of the two piece rotor.

![Figure 117: Cast impeller with internal threads and shaft with external threads](image)

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The integrated compressor motor test was successful to 106,800 rpm. The power requirement was more than 1230 W. The bearings failed while the system was ramping up to 108,000 rpm. Pressure rise, temperature and mass flow rate data were measured up to 106,800 rpm. The following figures give the integrated compressor-motor data up to 106,800 rpm. An axial movement of the impeller was observed at speeds around 60,000 rpm to 70,000 rpm. The problem was investigated to deduce a structural design problem in the compressor. The pressurized flow from the diffuser was re-entering to the impeller side through the bearing passage as shown in the Figure 119. This caused some pressurized fluid to act on the disk surface of the impeller to create the axial load.
The cumulative effect of misalignment between the two bearing holes and the excessive frictional heat generated from the ceramic ball bearing due to negative operating radial clearance was understood to be the reason for the failure. The bearing stiffness calculations for ball bearings depend on load, clearance, number of balls, contact angle etc. It is difficult to exactly quantify the bearing stiffness and simulations are done with critical speeds as a function of bearing stiffness. The designer should choose an operating point where important variables like resonant frequencies do not change much. It is also difficult to estimate the exact preload applied to our system. A preload of 2 to 4 Newtons was applied to the outer race of the bearing in our system. The aerodynamic load from
the impeller and preload of the bearings may vary at different speeds finally leading to variable stiffness of the system. The rigid body modes are also the function of the stiffness. Lowering of the rigid body modes due to decrease in system stiffness will cause the set-up to experience excessive vibrations. It is quite clear from the rotordynamic analysis that the vibrations are experienced in the bearings during the rigid body modes. The amount of displacement occurring in the system may be enough to exhaust the operating internal clearance in the bearings and generate excessive heat due to friction. Misalignments present in the system makes this situation even worse. The damage may not be immediate while the system is being operated but the results for the cyclic fatigue is clearly experienced at higher speeds. This phenomenon is prominently visible with the high power consumption at higher speeds. Efficient cooling of the bearings also is necessary. The fit between the inner race of the bearing and shaft and the fit between the outer race and housing which contributes to the internal operating clearance also affects the system performance. Effective cooling of the inner race and outer race of the bearings may allow carrying away the heat generated and help the bearing to sustain few cycles of operation. The fabrication complexity in the present compressor – motor test rig restricted us to use effective means of bearing cooling.
Figure 120: Compressor Performance Chart
CHAPTER 6. CONCLUSION

The initial Compressor-Motor test setup developed and tested with innovations made in the alignment and coupler issues. Helical coupler was designed and tested. Alignment method was improved by Reverse Rim Method and Translational Stages. The compressor – motor testing was successful to 90,000 rpm due to excessive power consumption and damage of bearings.

A 2 KW Permanent Magnet Synchronous Motor designed and tested. Shaft material was selected; stress analysis was performed and optimized by rotordynamic analysis. A motor-generator set developed to obtain overall motor efficiency of 89.9%.

An Integrated Compressor – Motor Structure was designed and tested. Several versions of integrated rotor was designed and tested, bearing fit determined, pre-load structure designed, innovative procedure for alignment developed and adjustable IGV developed for control over tip clearance. Tests were successful to 106,800 rpm.
APPENDIX A: Scaling of Centrifugal Compressor
Several versions of Centrifugal Compressor were designed by scaling.

A₁ – Single Stage Centrifugal Compressor with Neon as working fluid; Diameter of impeller 1.5 cms
Operational Speed = 450,000 rpm.

B₁ – Scaled Single Stage Centrifugal Compressor with Neon as working fluid; Diameter of impeller 4.8 cms;
Operational Speed = 140,000 rpm.

B₁’ – Scaled Single Stage Centrifugal Compressor with Air as working fluid; Diameter of impeller 4.8 cms.; Operational
Speed = 107,000 rpm

C₁ – Scaled Single Stage Centrifugal Compressor with CO₂ as working fluid; Diameter of impeller 4.8 cms;
Operational Speed = 83,000 rpm

⇒ Design B₁’ was chosen because of its present feasibility and simplicity.
APPENDIX B: Sample Calculations for Reverse Rim Alignment Method
The position of the movable machine’s front feet and rear feet is determined using the following equation:

\[
\frac{(M - S) \cdot B}{A} + M \quad \text{-- front feet equation}
\]

\[
\frac{(M - S)}{A} \cdot (B + C) + M \quad \text{-- back feet equation}
\]

Positive value means that point is high and shims need to be removed.

Negative value means that point is low and shims need to be added.

Sample Data

Given the following vertical misalignment data:

- The stationary side offset (S) is say 0.002 inches.
- The movable side offset (M) is say 0.003 inches.
- \( A = 2 \) inches, \( B = 4 \) inches, \( C = 2 \) inches

**Front Foot Position Calculation**

\[
\frac{(M - S) \cdot B}{A} + M = \left( \frac{(0.003 - 0.002)}{2} \right) \cdot 4 + 0.003 = 0.005 \text{ inches}
\]

The front feet are 0.005 high; shims need to be removed.

**Rear Foot Position Calculation**
\[
\frac{(M - S)}{A} \cdot (B + C) + M = \frac{(0.003 - 0.002)}{2} \cdot (4 + 2) + 0.003 = 0.006 \text{ inches}
\]

The rear feet are 0.006 inches high; shims need to be removed.
APPENDIX C: Old Diffuser and New Diffuser
From CFD Calculations (Courtesy: Xiaoyi Li), it was found that the old diffuser was less efficient. A new diffuser was designed (Courtesy: Ray Zhou) and fabricated.

<table>
<thead>
<tr>
<th>Old Diffuser</th>
<th>New Diffuser</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1.png" alt="Old Diffuser Image" /></td>
<td><img src="image2.png" alt="New Diffuser Image" /></td>
</tr>
<tr>
<td><img src="image3.png" alt="Old Diffuser Flow" /></td>
<td><img src="image4.png" alt="New Diffuser Flow" /></td>
</tr>
<tr>
<td><img src="image5.png" alt="Old Diffuser Physical" /></td>
<td><img src="image6.png" alt="New Diffuser Physical" /></td>
</tr>
</tbody>
</table>
APPENDIX D: Bearing Fit and Pre-load Calculations
Defining properties of the bearing material

\[ \alpha_s := 9 \times 10^{-6} \quad \delta_s := 4500 \frac{\text{kg}}{\text{m}^3} \quad E_s := 117 \times 10^9 \text{Pa} \quad \nu := 0.3 \]

\[ R := 4.19 \text{mm} \quad R_0 := \frac{6.35}{2} \text{mm} \]

Expansion due to centrifugal force

\[ \Delta_{RCF} := \frac{\delta_s \pi (100000)^2}{1800 E_s} \left[ (1 - \nu) \left( \frac{R}{\sqrt{2}} \right)^3 + (3 + \nu) \left( \frac{R_0}{\sqrt{2}} \right)^2 \right] \]

Expansion due to thermal effects

\[ \Delta_{RS} := 50R \cdot K \cdot \alpha_s \]

\[ \Delta_{RS} = 1.886 \times 10^{-6} \text{ m} \]

\[ \Delta_{RCF} = 1.854 \times 10^{-7} \text{ m} \]
LIST OF REFERENCES


