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Revision Page

Revision	Revised By	Date	Remarks
00	---	October 2003	Initial Issue
01	MJC	January 2004	Revised Table of Contents.
	PNB		Added revision page.
	JET		Revised Executive Summary and Results and Discussion Sections.
			Added Addendum.

**** DISCLAIMER ****

Abstract

Dresser-Rand completed the preliminary aerodynamic flowpath of the volute and inlet design for the compressor section. This has resulted in considerable progress being made on the development of the compressor section and ultimately towards the successful integration of the IEMDC System design. Significant effort was put forth in the design of aerodynamic components which resulted in a design that meets the limits of aerodynamically induced radial forces previously established. Substantial effort has begun on the mechanical design of the compressor pressure containing case and other internal components. These efforts show progression towards the successful integration of a centrifugal compressor and variable speed electric motor ventilated by the process gas. All efforts continue to confirm the feasibility of the IEMDC system design.

During the third quarter reporting period, the focus was to further refine the motor design and to ensure that the IEMDC rotor system supported on magnetic bearing is in compliance with the critical speed and vibration requirements of the API standards 617 and 541. Consequently specification to design magnetic bearings was developed and an RFQ to three magnetic bearing suppliers was issued.

Considerable work was also performed to complete preliminary reports on some of the deliverable tasks under phase1.0. These include specification for the VFD, RFQ for the magnetic bearings, and preliminary write-up for motor instrumentation and control schematic. In order to estimate motor efficiency at various operating points, plots of calculated motor losses, and motor cooling gas flow rates were also prepared. Preliminary evaluations of motor support concepts were performed via FEA to determine modal frequencies.

Presentation was made at DOE Morgantown on August 12,2003 to provide project status update. Preparations for the IEMDC motor-compressor presentation, at the GMRC conference in Salt Lake City to be held on October 5, 2003, were also started.

Detailed calculations of cooling gas flow requirements for the motor and magnetic bearings, per several new operating points designated by DR, confirmed that the required gas flow was within the compressor design guidelines. Previous thrust load calculations had confirmed that the magnetic thrust bearing design load capacity of 6,000 lb. was sufficient to handle the net thrust load produced by the motor and compressor pressure loading.

Thus the design data that has been generated, for the variable speed 10 MW 12,000 rpm motor, during the last three quarters, continue to confirm the feasibility of an efficient and robust motor design.

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Introduction

This report covers the third quarter (07/01/03 to 09/30/03) of the Phase 1 of the In-Line Electric Motor Driven Compressor (IEMDC) project.

The IEMDC project is the development of an in-line electric motor driven compressor to address the needs of the Natural Gas Industry. It represents a revolutionary advance in compressor system design and performance, in an improved package.

The objective of the first phase of the project is to develop the design of a direct coupled, seal-less, In-Line Electric Motor Driven Compressor to the point where detailed manufacturing drawings may be started. This gas compressor will be the world’s first that can be installed directly in the pipeline, utilizing a variable speed induction motor with magnetic bearings that is integrated with a centrifugal compressor. It will be an electrically powered, highly flexible, efficient, low maintenance compressor that can be quickly ramped up to meet peak demands. The unit design proposes to provide low cost, low maintenance gas compression for the Natural Gas Industry while minimizing the environmental, regulatory, and maintenance issues associated with gas turbine drives by the use of an electric motor as the prime mover.

Executive Summary

Unit Configuration and Integration
The issue of aerodynamically induced radial forces[1] generated in the aerodynamic flowpath gained significant progress during this quarter. Efforts during the design of the aerodynamic flowpath at the discharge focused on this problem. This issue was a design concern relative to the selection and design of the magnetic bearings. Preliminary evaluation of the primary flowpath components indicates that these forces can be kept within the limits established in the second quarter. This is a significant step in validating the feasibility of the design.

Compressor Design
Significant effort was put forth in the design and evaluation of the aerodynamic flowpath during this quarter. A preliminary design of the primary aerodynamic flowpath was completed for the IEMDC unit. The primary flowpath is the main passage that travels from the inlet nozzle and though the impeller and impeller diffuser system into the volute and out the discharge nozzle. This preliminary design consists of the basic geometry and areas defining the flowpath boundary and includes the appropriate levels of analysis to validate the design. This is a significant step toward the design of the compressor section of the IEMDC.

Several volute designs were analyzed using CFD during the development of the primary flowpath. An overhung volute style was chosen as it is expected to provide the best performance while maximizing the use of space within the IEMDC unit. The overhung volute style is more effective at using the available space than the other designs resulting in a more compact design. It also provides good static pressure recovery and low overall losses, which makes it a good choice for aerodynamic performance considerations. Preliminary analysis and evaluation of the chosen volute configuration indicates that aerodynamically induced radial forces can be kept within the limits established in the second quarter. Thus meeting the design load requirements for the radial magnetic bearings.

An aerodynamic inlet design concept was chosen for use in the IEMDC. Initially, two inlet concepts were evaluated for use in the IEMDC during the design process. The first inlet design was intended to simplify and accommodate standard fabrication methods and was previously evaluated in the second quarter. The second inlet design incorporates a scheduled area distribution to minimize turning losses. This second design was evaluated at the beginning of the third quarter. Both inlet designs were analyzed using computational fluid dynamics. A comparison of the two inlet designs was made in the third quarter to select the best option for the IEMDC. Results of the analyses indicated that there was no appreciable improvement in overall aerodynamic performance for the complexity of the geometry required for the second inlet design. Therefore first inlet design concept was chosen since it is expected to offer a better all around solution for both manufacturing and performance. However, some of the less complicated features of the second design may be incorporated into the chosen design concept.

Mechanical layout and design of the compressor pressure containing case was started after the completion of the primary aerodynamic flow passage design. Multiple design iterations have been performed on the case design to achieve a proper balance between mechanical and aerodynamic requirements. These design iterations have primarily been focused on the design of the nozzles that form the aerodynamic flowpath boundary. Preliminary calculations and analysis were performed on the case to check the structural design in accordance with industry standards. The design of the case is still in progress and is expected to undergo further optimization and analysis as validation of the design continues.

[1] See the first and second quarter technical reports for more detailed explanation of this issue.

Electric Motor

During this reporting period, the motor design made significant progress. An RFQ to procure magnetic bearings was written and sent to three magnetic bearing suppliers (see Appendix B). Preliminary design proposals for sizing the magnetic bearings from three independent magnetic bearing suppliers were reviewed, and have confirmed, that the IEMDC motor-compressor rotor design, supported on magnetic bearings, would meet the critical speed and vibration response requirements of the API standards 617, 7th edition and the 541. A summary of the analysis results is included in Table 1 of the section titled " Results and Discussion".

The technical specification for the Variable Frequency Drive (VFD) was completed and sent to Dresser-Rand for review. The VFD specification is included in Appendix C.

A preliminary schematic of the motor instrumentation and control was prepared and sent to Dresser-Rand for review. This is listed in Appendix D.

The motor thermal analysis was performed to determine motor cooling mass flow rate. Further refinements to the motor cooling flow requirements were carried out consistent with the additional operating points supplied by Dresser-Rand. Preliminary plots were generated showing the predicted motor cooling gas mass flow requirements as a function of compressor discharge temperature and inlet pressure at various operating speeds. Similar plots were also generated for the motor friction and windage losses. Additionally, graphs of motor electrical losses were generated as a function of compressor load and running speed. The resulting plots are discussed in the section titled," Results and Discussion" and were also sent to Dresser-Rand.

Preliminary Finite Element analysis of various concepts, to support the motor, was performed. The objective was to design a system that would provide sufficient separation of the base structure frequencies to preclude interactions with the excitation frequencies, thereby keeping the overall response at acceptable levels. The preliminary results of the FEA are discussed in the section titled, "Results and Discussion". Additional design and analysis work of the motor support system will be continued into the 4th quarter of 2003.

A technical proposal for extension of phase 1, to extend the design work beyond the 4th quarter of 2003, was prepared and submitted to Dresser-Rand for inclusion in their proposal to DOE.

A project progress presentation was made to the DOE at Morgantown, W VA on August 12, 2003. Preparations were started for the IEMDC motor-compressor technical presentation at the Gas Machinery Research Council (GMRC) conference in Salt Lake City to be held on October 5, 2003.

During the fourth quarter, further design work on the motor-compressor interface and motor support is planned. More work is also planned on the DOE deliverables for Phase 1.

In summary, during the third quarter, EMD has made significant progress in further improving the motor design and completing preliminary reports on some of the deliverables such as magnetic bearing specification and RFQ, technical specification for the VFD, motor instrumentation and control schematic. Further improvements to the motor cooling flow calculations, friction and windage losses, and motor electrical losses were also completed. Preliminary evaluations, via FEA, of motor support concepts were made. Thus, EMD has further confirmed the feasibility of the highly efficient and robust design for the 10 MW 12,000 rpm variable speed motor.

Experimental

No experimental work was performed during this reporting period.

Results and Discussion

Unit Configuration and Integration

Performance evaluation of the integrated system continues. Compressor aerodynamic operating predictions were made and motor cooling gas calculations were made. These were done in an effort to develop an operating performance prediction of the IEMDC unit.

Mechanical interface design concepts between the compressor and motor were explored in the third quarter. However, more focus on the interface is required as critical details are worked out on the compressor and electric motor design. Developing an interface design that provides the functionality of pressure separation and structural integrity while accommodating the required geometry and features of the motor and compressor is a requirement for the successful integration of the compressor-motor drive system. Design concepts are expected to be explored in detail in the fourth quarter.

Compressor Design

Considerable effort was put forth in the design and evaluation of the aerodynamic flowpath. Designing a flowpath for the inline configuration presents aerodynamic and mechanical challenges that result in multiple iteration during the design process. A balance must be attained between the aerodynamic design and the mechanical design of the unit. Constant evaluation of the design must be made to the aerodynamic flowpath design to facilitate the mechanical design of the unit. Modifications to the aerodynamic flowpath have been ongoing due to mechanical considerations of the inline configuration and will continue during the compressor case and internal component design. Typically changes are made in the form of flowpath cross-sectional area and shape as the geometry is developed. Evaluations have to be made to determine if the proper sectional areas and rate of change of can be maintained relative to the first past theoretical aerodynamic analysis. Appendix A contains figures of the aerodynamic flowpath (geometry) and solid models of mechanical components developed to establish the design of the flowpath. These figures show the geometry of the inlet designs and overall flowpath design developed, evaluated, and considered for the IEMDC unit

Several volute designs were evaluated to determine if established levels of aerodynamically induced radial forces could be met. Multiple volute designs were reviewed to determine the effect of geometry on aerodynamically induced radial loads and aerodynamic efficiency. The volutes were evaluated using results from CFD aerodynamic analyses for each configuration and comparing static pressure recovery and gradients within the volutes. A review of the various aerodynamic configurations determined that the overhung style of volute should be used for the inline configuration. Appendix A shows the various volute configurations and the relative aerodynamic radial forces generated in each configuration. The overhung style volute selected has the highest aerodynamic radial forces, but provides for the greatest aerodynamic efficiency. In addition, the aerodynamic forces are well within the established limits discussed in the second quarter report (See Appendix A).

Preliminary mechanical design calculations were performed on the compressor case design. These calculations consisted of simplified FEA and hand calculations with the goal of designing the case in accordance with API 617. Typically sections of ASME BPV Code Section VIII Division 1 and 2 are applied in the design process. The purpose of these calculations is to generate and evaluate dimensional information to ensure the mechanical integrity of the case design. Several iterations have been performed during the optimization process to improve the mechanical and aerodynamic geometry to facilitate the manufacturing process. A model of the case undergoing development is shown and discussed in Appendix A. A more detailed analysis of the case will be performed in the fourth quarter when the case design becomes more finalized.

Preliminary calculations were performed on the internal diffuser components to review levels of stress and deflection as dictated by the compressor configuration. Both simplified hand calculations and FEA was performed using conceptual geometry of the compressor end wall. The diffuser is an internal component that forms the internal pressure boundary at the motor compressor interface. A concept developed for the interface design is shown in Appendix A. The preliminary calculations have resulted in establishing a preliminary design method for the endwall that will help in establishing the interface design details.

Compressor mechanical design efforts will continue into the next quarter. These efforts will focus on the pressure containing case and internal component design.

Electric Motor Design

Rotordynamics: Prediction of an accurate rotordynamics behavior (critical speeds and vibration response) of the IEMDC motor-compressor supported on magnetic bearings is one of the critical design tasks. Furthermore, the selection of an optimized magnetic bearing design to provide desired radial and axial load capacity, and to control the rotor vibration response to unbalance loads is paramount to the trouble free operation of the IEMDC. Therefore, considerable thought was given to develop an appropriate design specification, to design and procure the magnetic and backup bearings. The design specification conforms to the rotordynamics requirements of the API standard 617, 7th edition for the centrifugal compressor and the API standard 541, 3rd edition, April 1995 for form-wound squirrel cage large induction motors. The specification was included in the Request For Quote (RFQ) that was sent to three reputed magnetic bearing suppliers. The magnetic bearing suppliers solicited were Kingsbury Magnetic Bearings (KMB), Revolve Magnetic Bearings, and Waukesha Magnetic Bearings (WMB). A copy of the RFQ is listed in Appendix B, for reference.

The written response to the RFQ from the three magnetic bearing suppliers was very encouraging. It confirmed that the proposed magnetic bearing designs and the incorporation of the specific design angular contact ball bearings, as backup bearings, for the IEMDC motor-compressor would indeed comply with the rotordynamic requirements of the applicable API standards. A summary of the predicted critical speeds of the IEMDC motor-compressor is presented in Table 1.

Table 1: Calculated preliminary rotor critical speeds excited by unbalance at 12,000 rpm

Mode #	Type	Critical Speed, rpm	Log Dec	AF	Comments
1	Rigid body, Translation	4,200	2.99	1.163	Critically damped mode
2	Rigid body, Tilt	8,250	1.62	2.0	Critically damped mode
3	First Bending Forward	15,840	0.314	10.0	32% margin over max speed

Note: All critical speeds excited by unbalance satisfy API 617 and API 541 requirements

VFD Technical Specification: One of the deliverables for Phase 1 is the preparation of a technical specification for the Variable Frequency Drive (VFD) to drive the 10 MW induction motor. For this purpose, preliminary contacts were made with three VFD suppliers (ABB, ALSTOM, and ASIRobicon) to solicit information on their product line. After reviewing the technical and budgetary pricing information that was received from the three suppliers, it was decided to work more closely with ASIRobicon to develop the required technical specification. Additional reasons to work with Robicon were the proximity of Robicon to EMD and also prior experience of using Robicon's VFD by Dresser-Rand and EMD. A copy of the resulting specification is listed in Appendix D. This specification will be used to procure the VFD during the prototype manufacturing and testing phase of the IEMDC motor-compressor. The key features of the WCII Harmony VFD for the IEMDC motor drive are listed in Table 2.

Table 2: Preliminary Characteristics of WCII Harmony VFD

Primary Voltage, kV	13.8
Motor Voltage, kV	6.9
Power Rating, HP/kW	13,400 / 10,494
Phases per phase required	6 @ 690V
Dimensions w/water to water HEX, in.	516L x 70D x 120H
Estimated weight, Lbs.	63,000
Losses, %/kW	4 / 400 (99% removed by water)

Motor Instrumentation and Control Schematic: A preliminary schematic of the motor instrumentation and control, as shown in Figure 1, was prepared and sent to Dresser-Rand for further enhancements to include the controls for the compressor. After completion, this will be one of the deliverables for Phase 1. A brief discussion of the proposed control schematic is given below but more detailed description is provided in Appendix D.

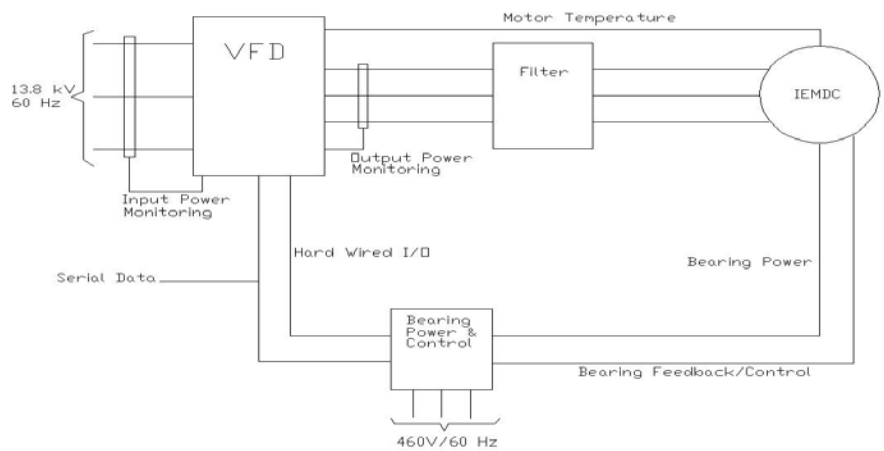

Figure 1: Proposed Motor Control and Instrumentation Schematic

In reference to Figure 1, the Master Control System of the VFD controls the compressor operation by monitoring data through a serial interface/communications board, hard-wired I/O, or a keypad located at the drive. VFD I/O is configurable per Table 3. The communications board is configurable to allow network communication via a variety of protocols.

Table 3– VFD I/O	
Parameter	**Maximum number**
Analog Inputs	24
Analog Outputs	16
Digital Inputs	96
Digital Outputs	64

Open Loop Vector Control is used to control compressor speed. In this method the VFD control estimates motor slip as a function of load torque (speed feedback is not required), so the motor speed is determined from motor stator frequency.

Motor stator temperature is monitored at the VFD via 6 Resistance Temperature Detectors (RTD) located in the motor stator slots. The Master Control System of the VFD, per Tables 4 and 5, monitors the input and output powers. Input side monitoring allows the drive to protect the secondary side of the input transformer. Output power measurements are used to implement rollback conditions to protect the drive as well as the motor.

Table 4- VFD Input Power Monitoring	
Phase Currents	Average Current THD
Phase Voltages	Efficiency
Frequency	kWh
Average Power	Reactive Power
Power Factor	

Table 5 - VFD Output Power Monitoring	
Motor Currents	Output Power
Motor Voltages	kWh
Motor Speed	Frequency

The Master Control software and hardware sense faults and alarms and store them within the fault logger. These faults can be the result of input line disturbances, motor/output disturbances, system related, system I/O related, external serial communications related, or user-defined faults. In the context of the motor/drive/magnetic bearing system a bearing fault can be used to fault the drive and initiate a rollback and/or shutdown sequence.

Motor Cooling Gas Flow Rate Requirements and Motor Losses:
Several design iterations were performed to estimate motor cooling flow requirements, motor friction and windage losses, and motor electrical losses per IEMDC compressor operating points that were provided by Dresser-Rand. Plots were then generated to investigate the trending of these parameters. It was decided to plot data at motor operating speeds of 8000 rpm, 10,000 rpm, and 12,000 rpm. These speed points are within the operating speed envelope of the IEMDC motor-compressor.

Due to the variable speed motor, it is difficult to generate a simple motor efficiency curve. It is, however, still possible to identify the primary factors that affect the motor losses and thereby the motor efficiency at a given operating point. The primary factors affecting the motor electrical losses are: type of motor rotor construction i.e., solid VS laminated, thickness of stator laminations, rotor-to-stator air gap, compressor load, and motor running speed. The overall motor friction and windage losses are affected by the running speed, motor cooling gas density, and magnetic bearing losses. The gas density is affected by the gas pressure and temperature.

Motor Cooling Gas Flow Rate: Figures 2-4 show calculated motor cooling flow rates at motor speeds of 8000 rpm, 10,000 rpm, and 12,000 rpm, respectively. At each of these three operating speeds, the cooling flow rate is plotted as a function of the compressor discharge temperature (60°F to 180°F). Superimposed on each figure is also the effect of changing the compressor inlet pressure on the motor cooling flow requirements. It is evident from the trends of these plots that the cooling flow increases by about 25% when the inlet pressure is increased from 500 psi to 800 psi and decreases by about 10% with the increased discharge temperature from 60F to 180 F. The effect of increased speed from 8,000 rpm to 12,000 rpm is relatively small. The required flow rate drops by about 4% to 5%. These trends can be utilized to assess the overall IEMDC performance and to design the flow control scheme for the IEMDC motor-compressor.

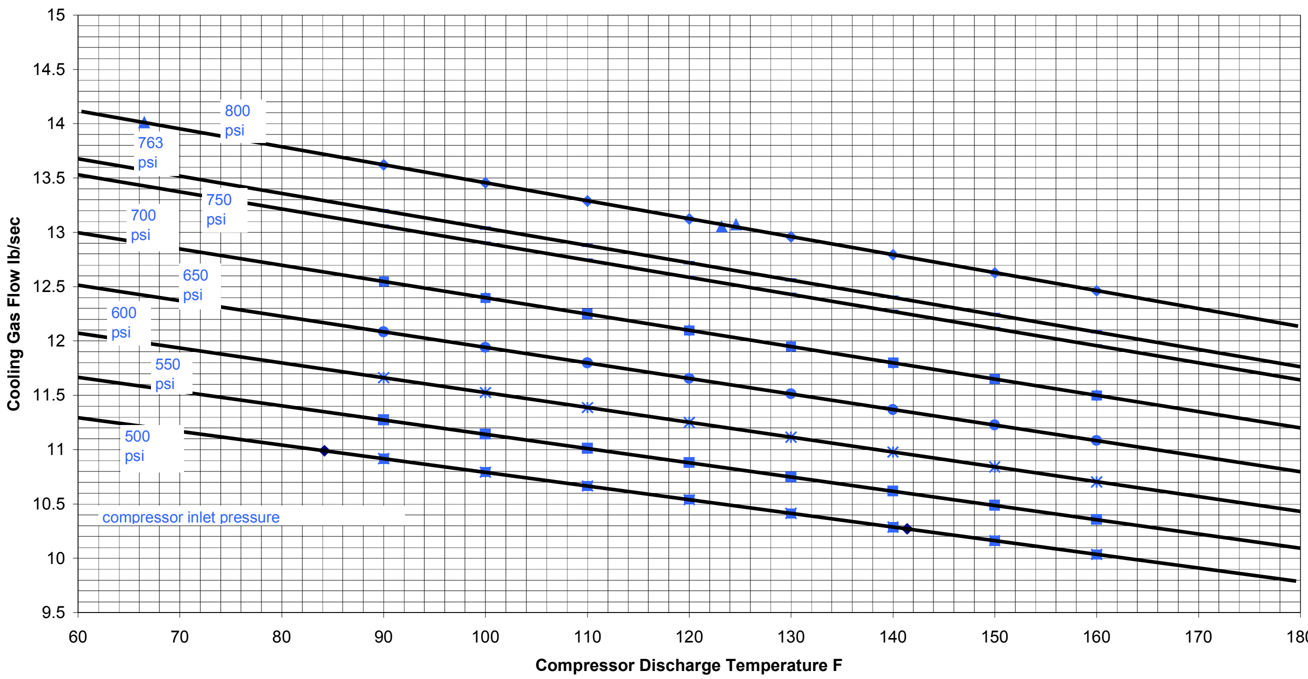

Figure 2 Calculated Motor Cooling Flow Rate at 8000 rpm

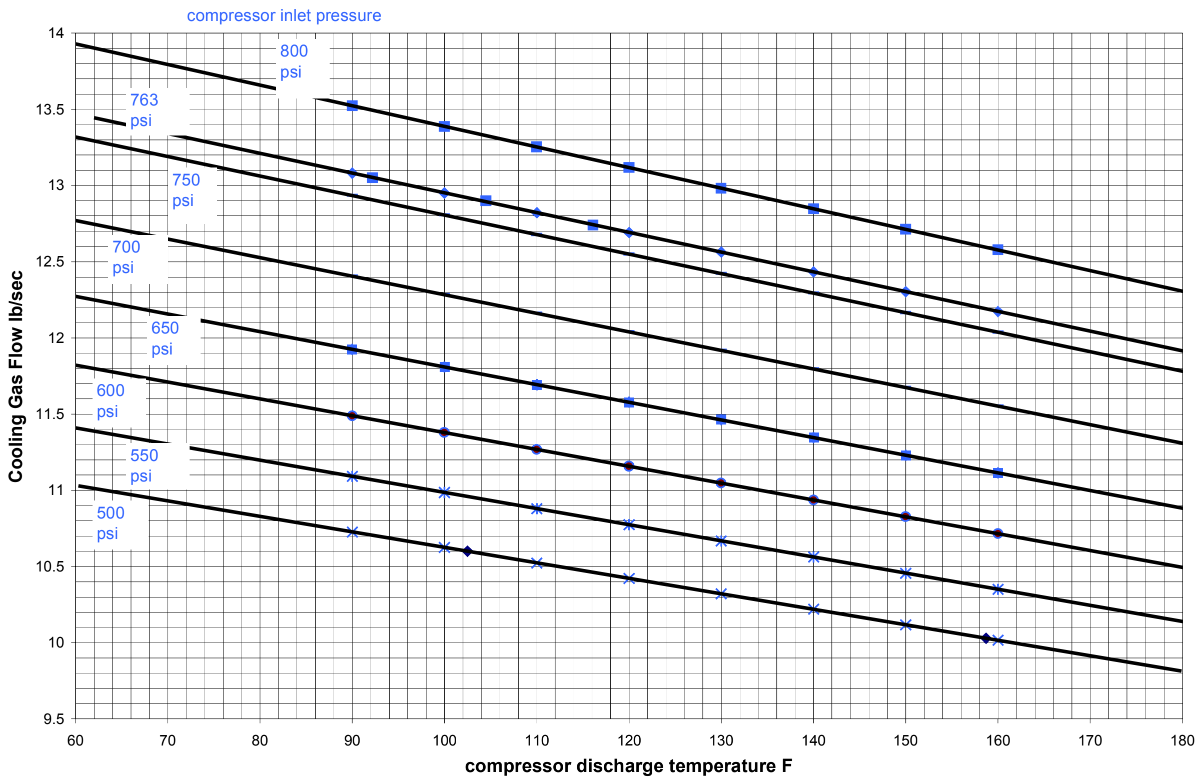

Figure 3 Calculated Motor Cooling Flow Rate at 10000 rpm

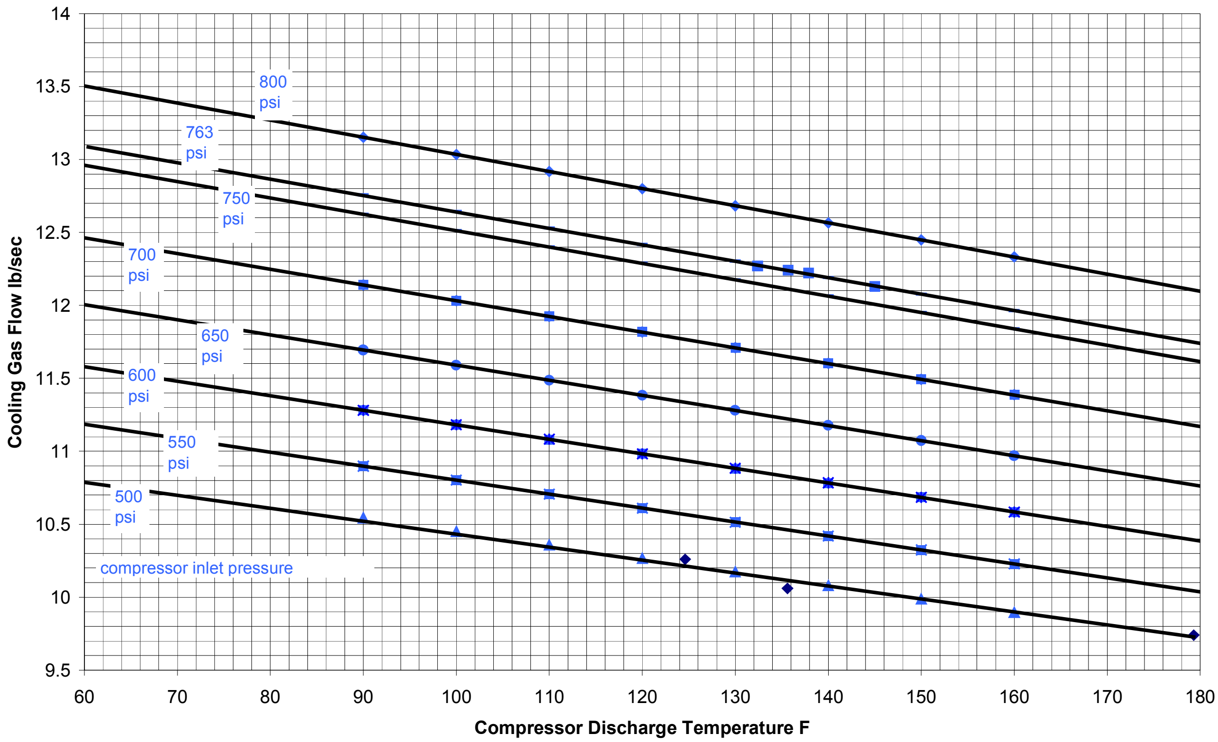

Figure 4 Calculated Motor Cooling Flow Rate at 12000 rpm

Motor Friction and Windage Losses: Figures 5-7 show similar trend plots for the motor friction and windage losses at 8,000 rpm, 10,000 rpm, and 12,000 rpm, respectively. The losses increase with increased rotor speed and the increased inlet pressure. They are significantly higher at lower compressor discharge temperatures as compared to the higher discharge temperatures. This is primarily due to the effect of gas temperature on the gas density. The losses are nearly proportional to the gas density.

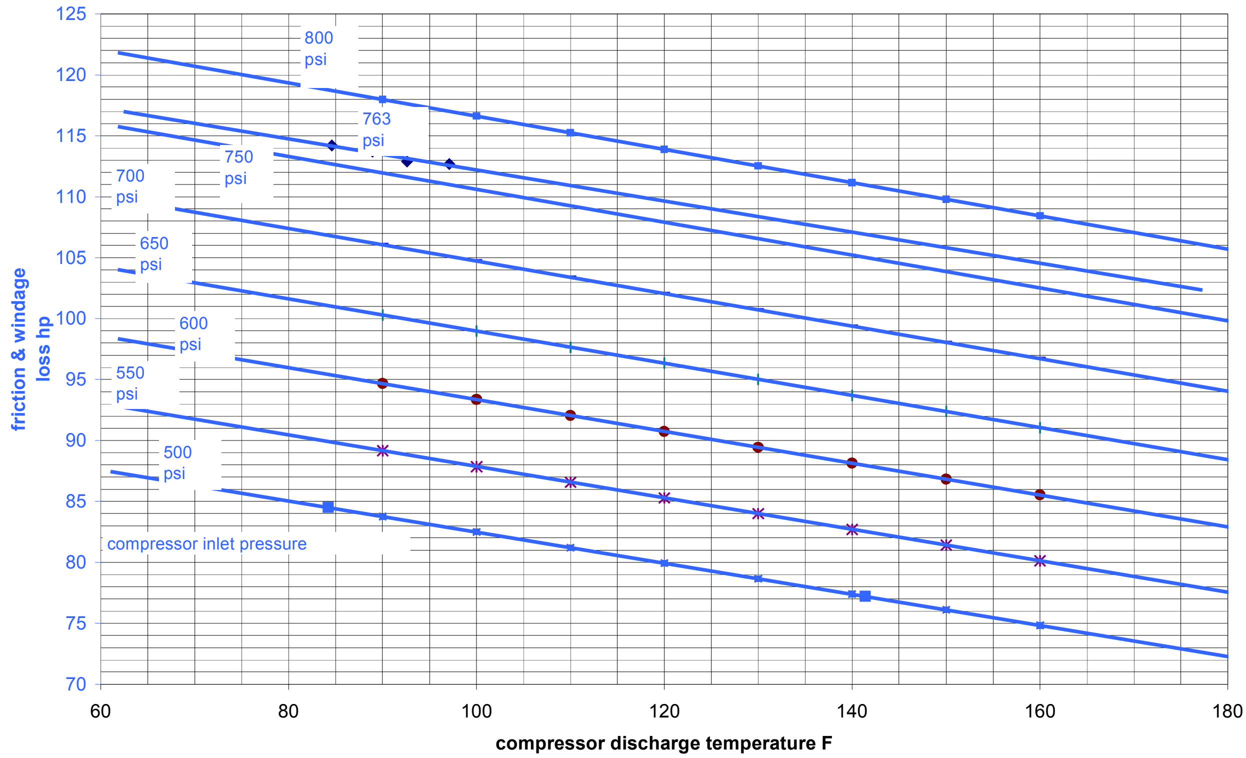

Figure 5 Calculated Motor Friction and Windage Loss at 8000 rpm

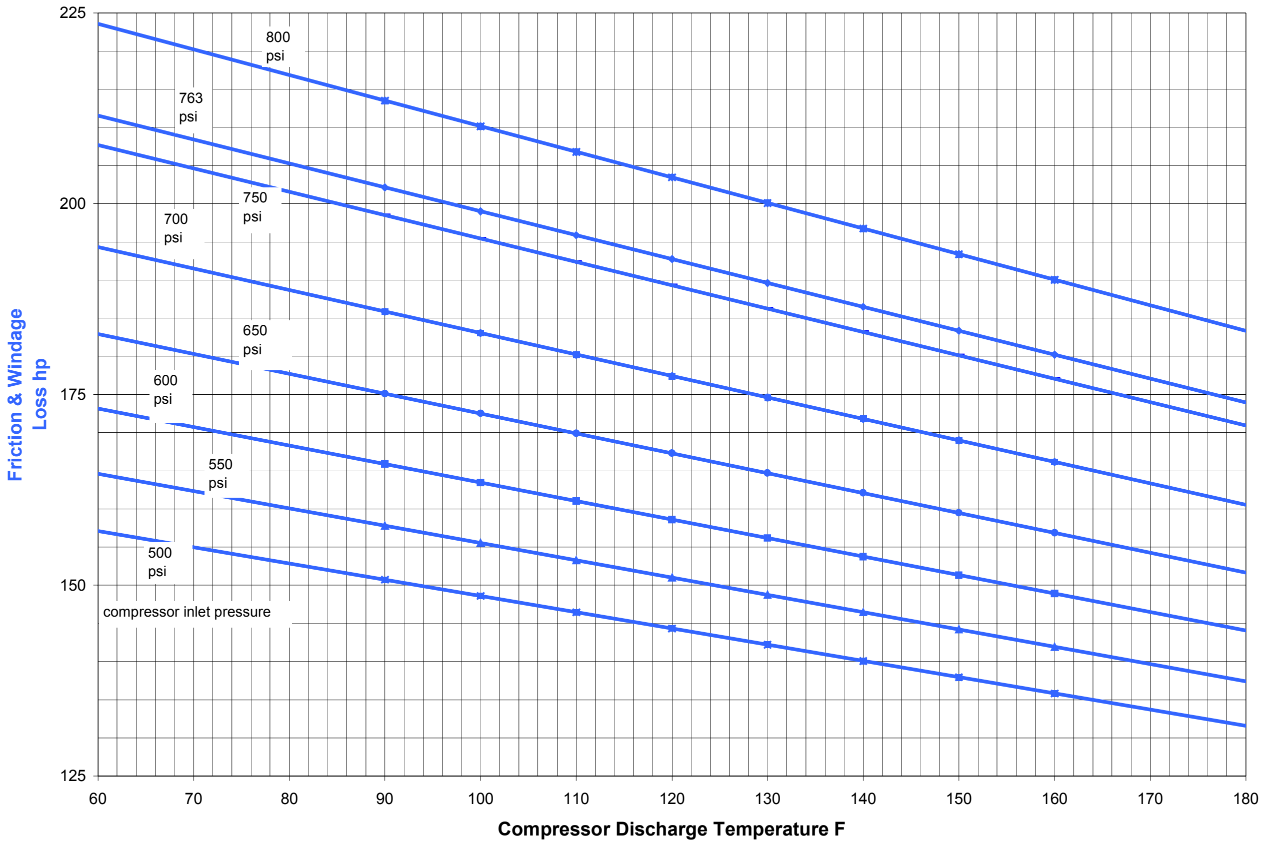

Figure 6 Calculated Motor Friction and Windage Loss at 10000 rpm

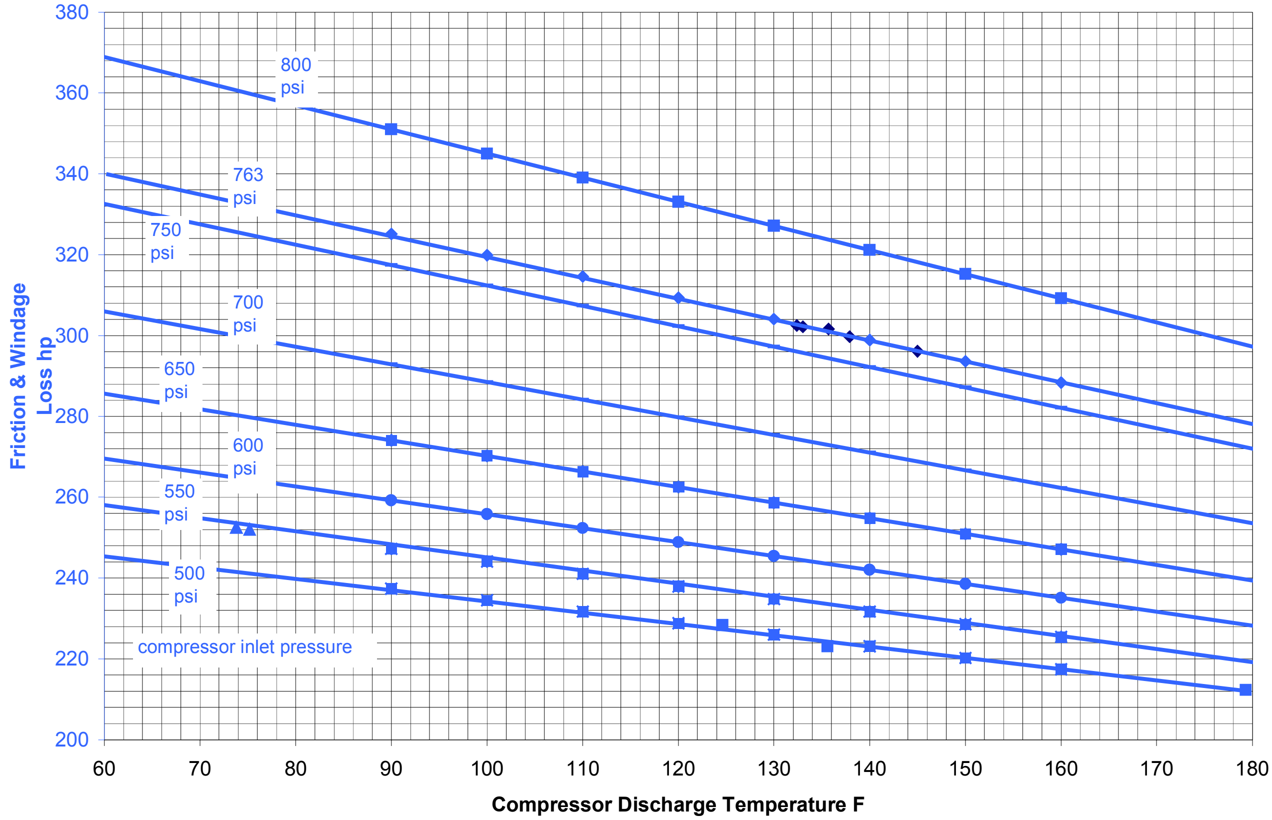

Figure 7 Calculated Motor Friction and Windage Loss at 12000 rpm

Motor Electrical Losses: Figure 8 shows trend plots for the motor electrical losses as a function of the shaft output power or compressor load at 8,000 rpm, 10,000 rpm, and 12,000 rpm, respectively. For a given motor operating condition, the electrical losses increase with motor speed and compressor load. Comparing Figures 5-7 with Figure 8, it is evident that as the rotor speed increases the friction and windage losses increase at faster rate than the electrical losses. During the motor design process, to minimize the friction and windage losses, it was decided to use an intermediate gas pressure rather than the compressor discharge pressure to cool the motor. Additional electrical design improvements such as roebeling and different top and bottom coil cross sections, were also implemented to improve the motor efficiency. Thus the 10 MW 12,000 rpm motor design has evolved into a robust and efficient design. The predicted motor efficiency at the design point is about 95%.

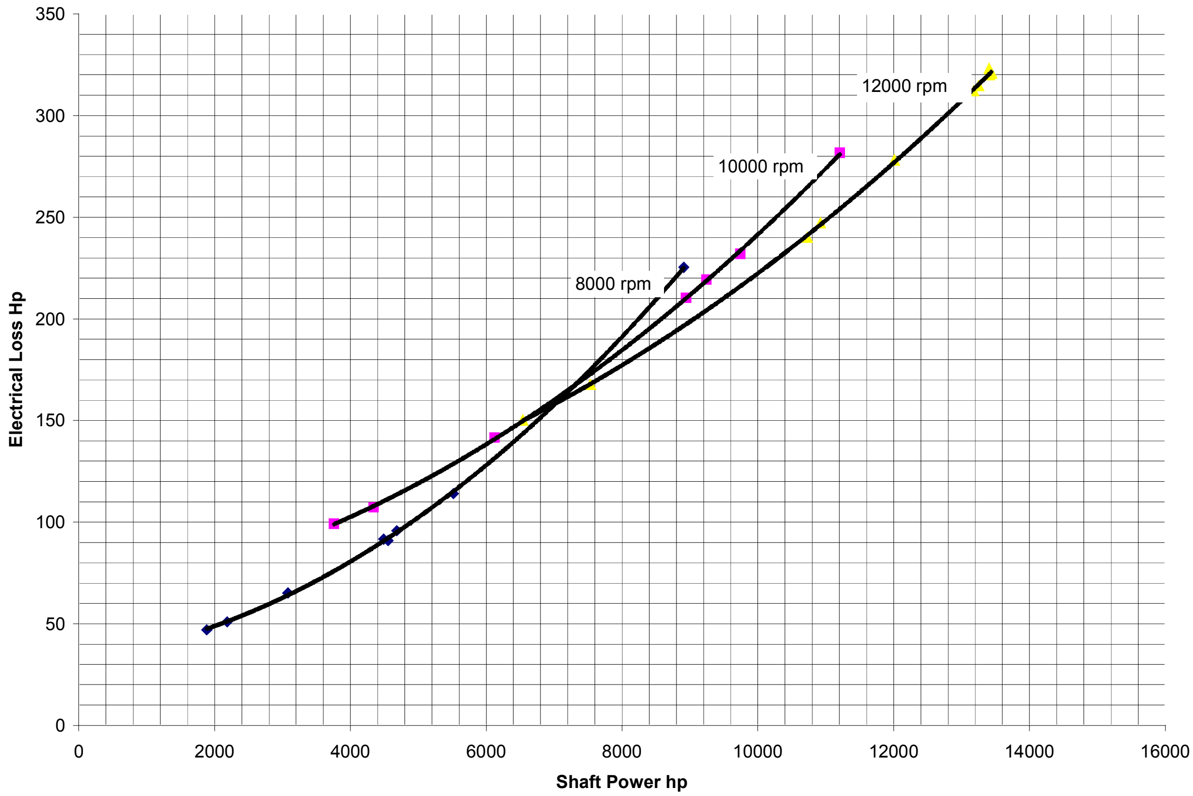

Figure 8 Motor Electrical Losses

Modal Frequency Analysis of Motor Foundation:
FEA models were developed for six motor-compressor foundation concepts as depicted in Figures 9 to 14. Modal frequencies were calculated for each model. The design objective for this analysis work was to determine the foundation stiffness that will enable the mount frequency to be above the motor-compressor maximum operating frequency of 200 Hz. Thereby precluding any force excitation by the motor foundation. The results of predicted frequencies, for each model that was analyzed, are listed in Table 6. The highest predicted frequency was 115.8 Hz, well below the goal of above 200 Hz. And the mount structure had already become unacceptably heavy. Therefore the approach to design out the mount frequency above the max operating speed is not feasible.

The alternate approach to achieve an acceptable design solution will be to analyze the vibration response of the casing structure due to rotor unbalance forces. Since the motor-compressor static structure is at least 15 times heavier than the rotating structure, we anticipate, based upon prior experience of analyzing similar structures the casing vibration response to be very small, less than 0.1 mil. This concept has been verified via a simplified preliminary rotor dynamics modeling of the rotor and the casing structures.

The plan, therefore, is to design an optimum motor mounting system by modifying the FEA models to investigate casing response with soft and rigid supports. Further analytical modeling work will be performed in the next quarters.

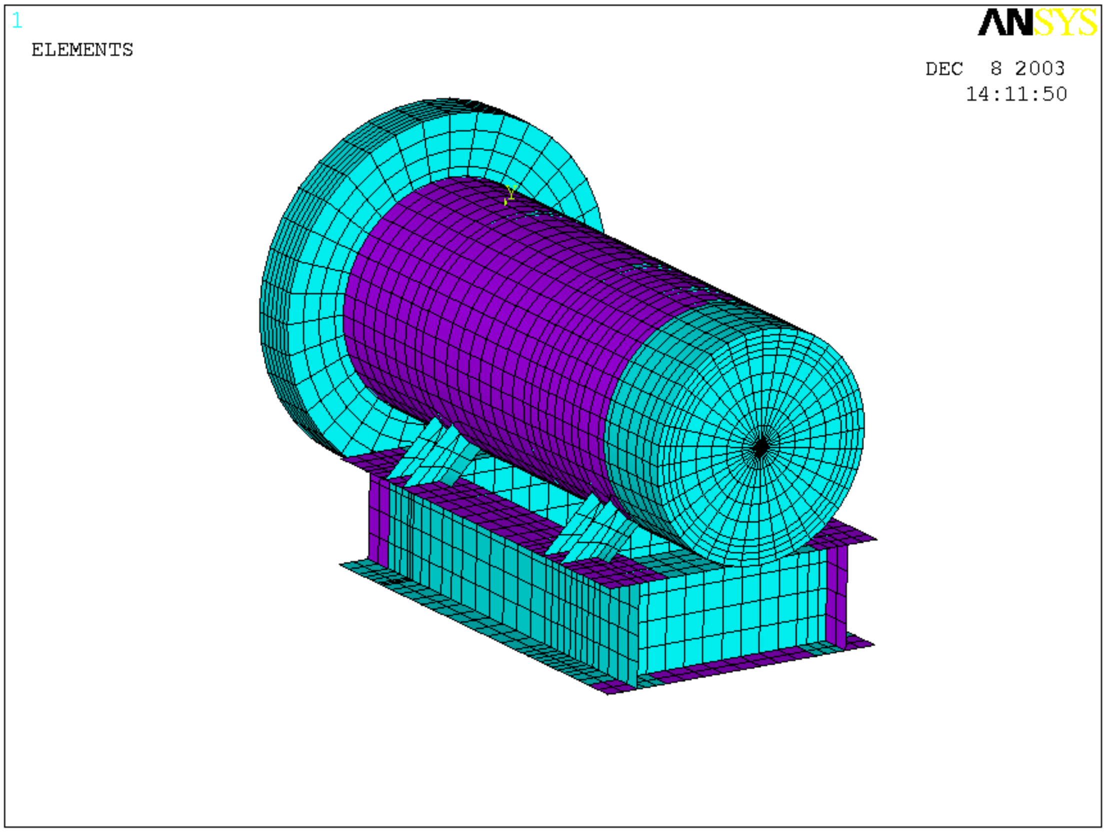

Figure 9: Motor Mount Concept 1

Concept 1 Description:

Welding I-beams in a box shape and closing the bottom with a plate form the foundation. The I-beam flanges are clipped to fit the webs together. This makes an easy to manufacture, cost effective weld joint. This configuration is a common one for motors of this size. The FEA showed a horizontal motion at the thrust end of the casing. This motion occurred at 98.3 Hz, which is below running frequency.

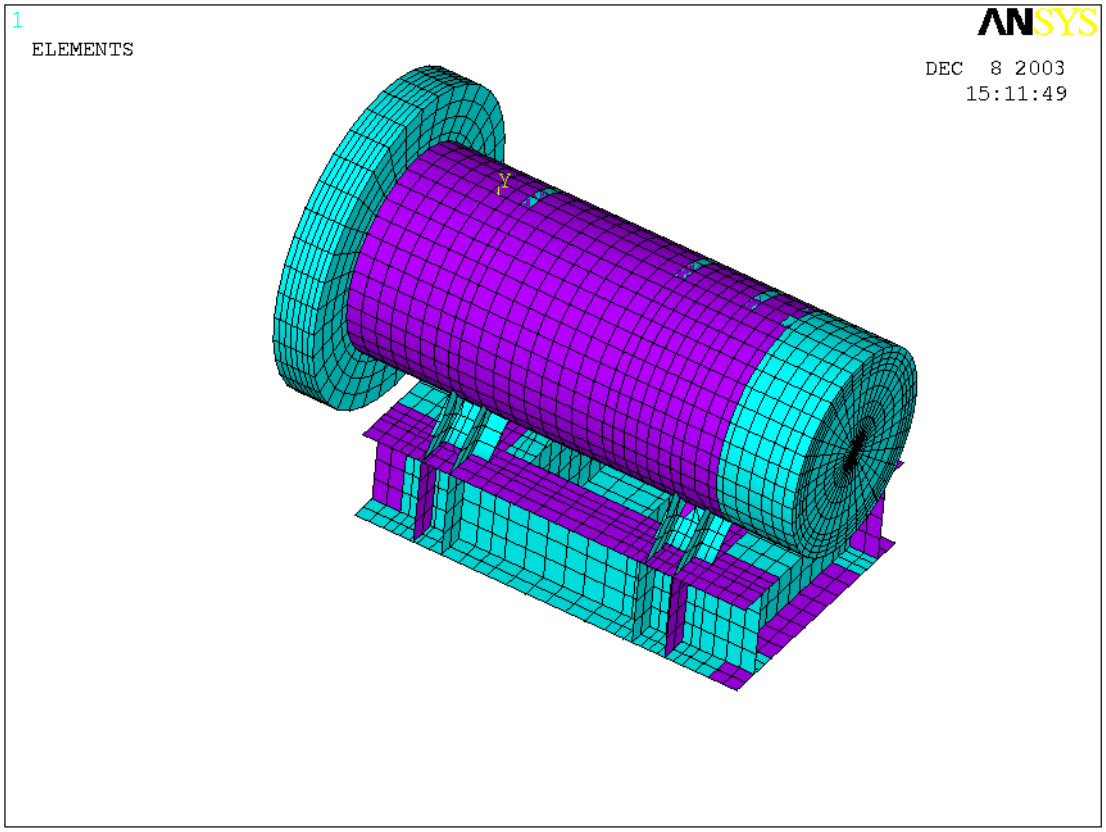

Figure 10: Motor Mount Concept 2

Concept 2 Description:

The FEA model of cocept1, as shown in Figure 9, produced casing horizontal motion with the I-beam flange twisting. To stiffen this up, we added vertical stiffeners under the mounting feet and additional plates between I-beams for concept2 modeling. This raised the frequency to 115.8 Hz from 98,3 Hz, which is below the minimum running frequency of 133 Hz.

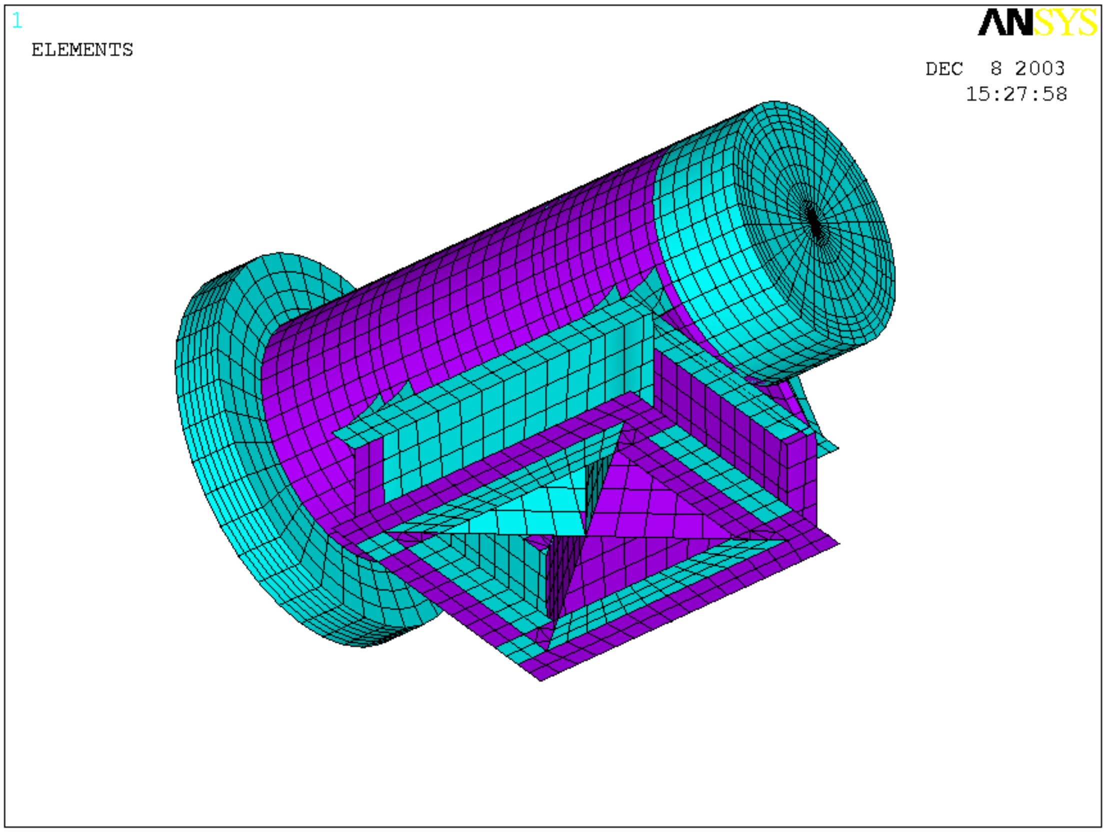

Figure 11: Motor Mount Concept 3

Concept 3 Description:

With concept 3, "x-bracing" was added from corner to corner of the I-beams in an attempt to stiffen the horizontal motion. This concept only raised the natural frequency to 99.7 Hz as compared to concept 1 model.

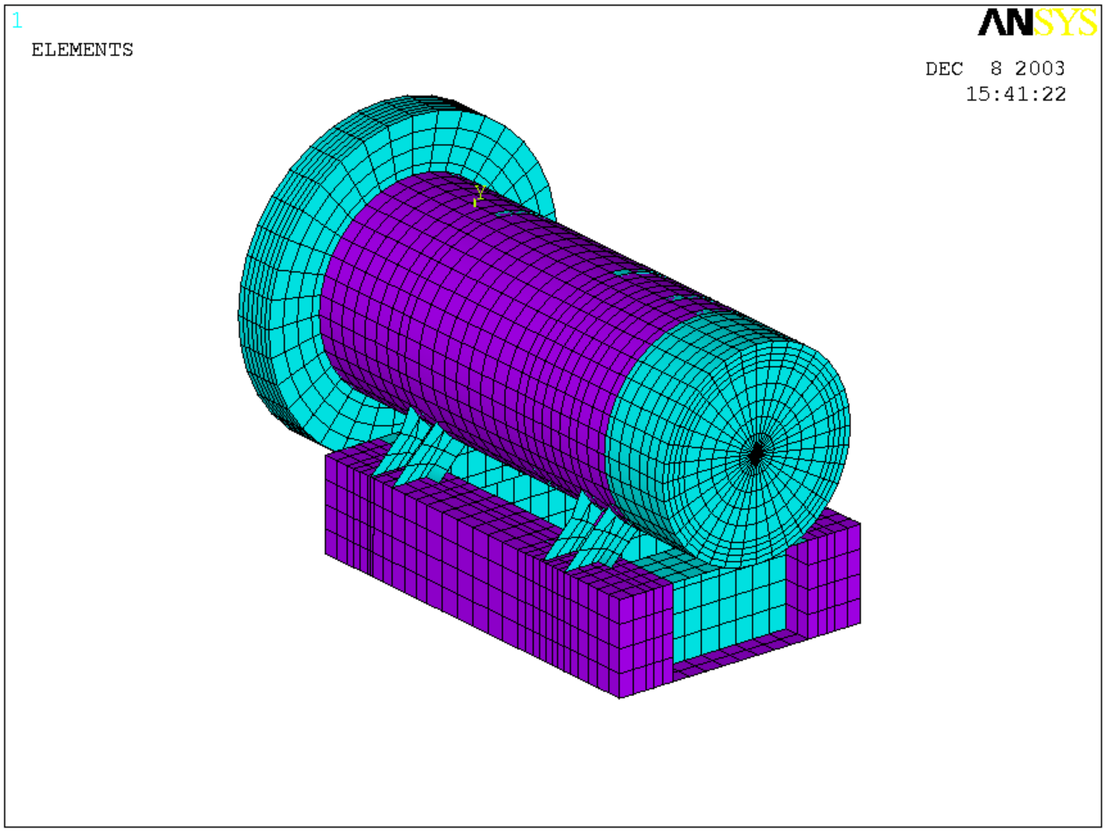

Figure 12: Motor Mount Concept 4

Concept 4 Description:

In this concept the I-beams were replaced with rectangular tubes to stiffen the horizontal motion. This raised the frequency to 114.5 Hz.

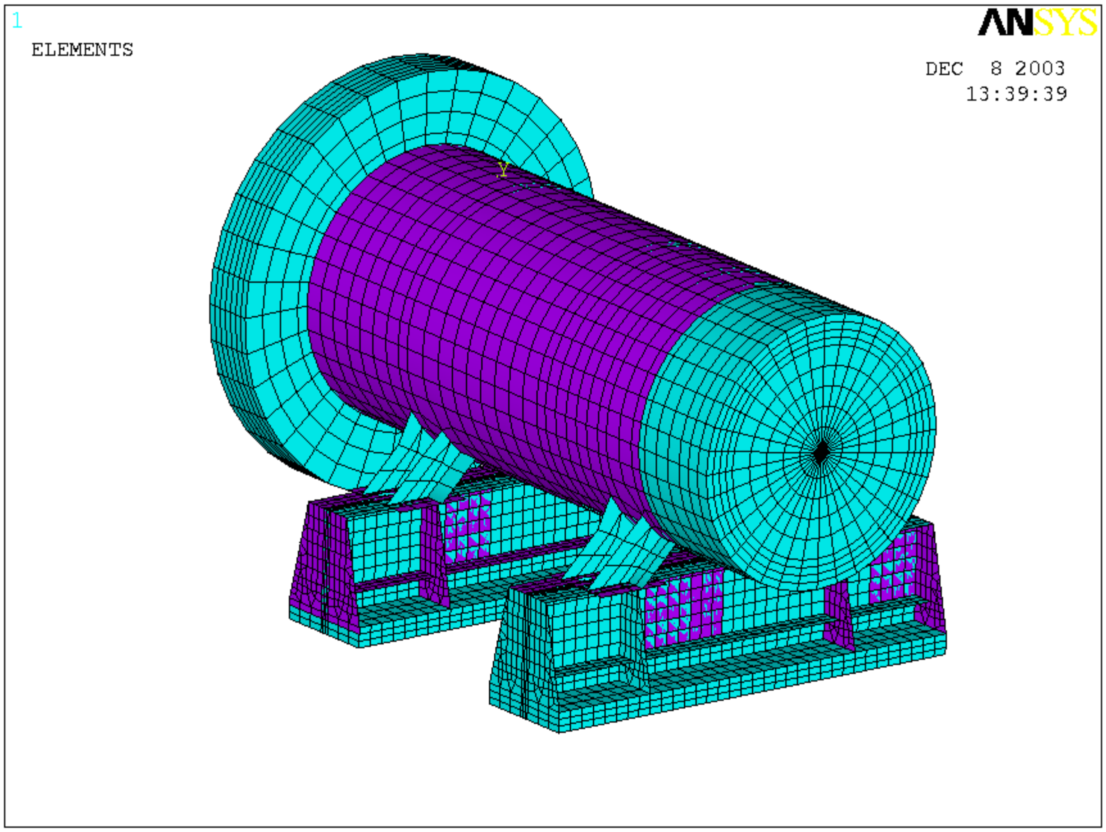

Figure 13: Motor Mount Concept 5

Concept 5 Description:

This concept attempted to stiffen concept 6 by making the foot and the plate between C-channels continuous across. Additionally, gussets were added to stiffen the motions parallel to the centerline of the motor. This raised the frequency to 114.8 Hz.

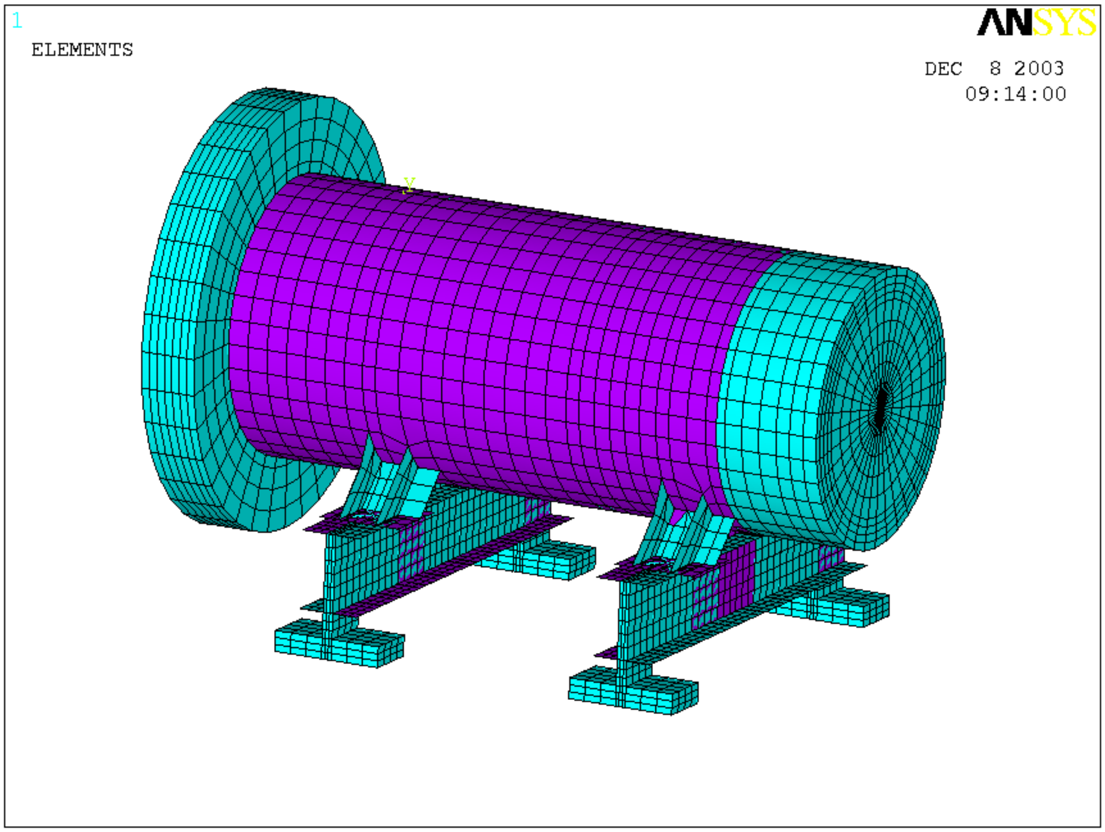

Figure 14: Motor Mount Concept 6

Concept 6 Description:

This concept attempted to reduce the horizontal motion by using back-to-back C-channels separated by a plate between motor supports. The frequency in the horizontal direction was raised to 109.6 Hz.

Table 6: Motor mount FEA predicted modal frequencies for various mount concepts

Motor Mount Design Concept	Frequency of Horizontal Cantilever Motion (Hz)
Concept 1	98.3
Concept 2	115.8
Concept 3	99.7
Concept 4	114.5
Concept 5	114.8
Concept 6	109.6

Conclusion

Work performed during the third quarter continued to focus on the design and development of the IEMDC unit. The design and development of the IEMDC is progressing into more detailed features of the design via mechanical design layout efforts of the compressor section. The compressor preliminary aerodynamic flowpath was completed and the design of the mechanical aspects of the aerodynamic flowpath such as the compressor containment case is in process. Design of the preliminary compressor discharge system confirmed that the aerodynamically pressure induced forces can be limited to within established limits. Substantial efforts such as those made during the design of the aerodynamic flowpath continue to demonstrate the feasibility of the revolutionary IEMDC configuration.

Continued progress over this reporting period shows that the technical advantages of the IEMDC over conventional technology continue to be maintained. These advantages include several key industry attributes such as operational flexibility, remote operation and automation, efficiency, reduced maintenance issues, and environmental benefits. Magnetic bearings require no lubrication; the variable speed electric motor produces no combustion by-products, and a sealed design does not allow process gas to leak into the atmosphere. It also maintains the capability to be installed in-line with the process piping and potentially in an underground vault.

The evolution of the integrated motor-compressor design continues to confirm the viability of the innovative IEMDC system. This is a very significant achievement. The successful development of this new advanced technology integrated motor-compressor system would provide to the Natural Gas Industry, a competitive low cost and low maintenance gas compression alternative.

References

API Standard 617, 7th Edition, July 2002, "Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical, and Gas Industry Services".

API 541, 3rd edition, April 1995,"Form-Wound Squirrel Cage Induction Motors – 250 Horsepower and Larger".

List of Acronyms and Abbreviations

AF	Amplification Factor
API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
BPV	Boiler and Pressure Vessel Code
CFD	Computational Fluid Dynamics
c.g.	Center-of -Gravity
DR	Dresser-Rand Company
EMD	Curtiss-Wright Electro-Mechanical Corporation (EMD)
F	Fahrenheit
FEA	Finite Element Analysis
GMRC	Gas Machinery Research Council
Hp	Horsepower
Hz	Hertz
IEMDC	In-Line Electric Motor Driven Compressor
I/O	Input / Output
kV	kilovolt
kW	kilowatt
kWh	Kilowatt hour
Lb	pound
Log Dec	Logarithmic Decrement
Mil	one one-thousands of an inch
MW	megawatt
PSI	Pounds per square inch
RFQ	Request for Quotation
RPM	Revolutions per minute
RTD	Resistance Temperature Detector
THD	Total Harmonic Distortion
V	Volts
VFD	Variable Frequency Drive