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Abstract

This report contains the final project summary and deliverables required by the award for the development of an In-line Electric Motor Driven Compressor (IEMDC). Extensive work was undertaken during the course of the project to develop the motor and the compressor section of the IEMDC unit. Multiple design iterations were performed to design an electric motor for operation in a natural gas environment and to successfully integrate the motor with a compressor. During the project execution, many challenges were successfully overcome in order to achieve the project goals and to maintain the system design integrity. Some of the challenges included limiting the magnitude of the compressor aerodynamic loading for appropriate sizing of the magnetic bearings, achieving a compact motor rotor size to meet the rotordynamic requirements of API standards, devising a motor cooling scheme using high pressure natural gas, minimizing the impact of cooling on system efficiency, and balancing the system thrust loads for the magnetic thrust bearing. Design methods that were used on the project included validated state-of-the-art techniques such as finite element analysis and computational fluid dynamics along with the combined expertise of both Curtiss-Wright Electro-Mechanical Corporation and Dresser-Rand Company.

One of the most significant areas of work undertaken on the project was the development of the unit configuration for the system. Determining the configuration of the unit was a significant step in achieving integration of the electric motor into a totally enclosed compression system. Product review of the IEMDC unit configuration was

performed during the course of the development process; this led to an alternate design configuration. The alternate configuration is a modular design with the electric motor and compressor section each being primarily contained in its own pressure containing case. This new concept resolved the previous conflict between the aerodynamic flow passage requirements and electric motor requirements for support and utilities by bounding the flowpath within the compressor section. However most importantly, the benefits delivered by the new design remained the same as those proposed by the goals of the project. In addition, this alternate configuration resulted in the achievement of a few additional advantages over the original concept such as easier maintenance, operation, and installation. Interaction and feedback solicited from target clients regarding the unit configuration supports the fact that the design addresses industry issues regarding accessibility, maintainability, preferred operating practice, and increased reliability.

The IEMDC developed during this project is a unique compression system that advances the state-of-the-art technology for natural gas pipeline service. This system will be driven by a 10 MW induction motor supported on magnetic bearings and powered by a 200 Hz variable frequency drive. A single centrifugal compressor stage is overhung from the shaft of an electric motor resulting in a completely integrated system. This electrically powered, highly flexible, efficient, and low total low life cycle cost system can be quickly brought online to supply gas during times of peak demands. In addition to the low total cost of ownership, the IEMDC design proposes to significantly mitigate critical gas industry concerns regarding environmental, regulatory, and maintenance requirements that are associated with natural gas powered compression equipment.

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Introduction

A recognized need exists within the gas transmission industry for a new generation compressor. The characteristics desired for this new compression system include: minimal maintenance requirements; capability of starting and stopping several times per day; easy installation; low total life cycle cost; and minimal environmental detriments. Several factors driving this need include: the increasing demand for natural gas; stringent environmental regulations; maintenance requirements of gas-fired equipment; and the age of the installed pipeline infrastructure.

Considering the current configurations of commercially available pipeline compressor systems, an alternative system designed for increased throughput to meet the growing demand on the aging pipeline infrastructure would provide an attractive solution to the challenges facing the gas industry. This system would need to be capable of readily replacing older compressors, re-powering compressor stations, and creating easily installed new compressor stations for the gas industry. Currently, there is an aging fleet of 20 to 50 year old gas driven compressors on pipelines. Maintaining this aging fleet of compression equipment can be a daunting challenge to gas pipeline operators due to: a) on site gas leakage; b) emissions that cause air pollution, d) availability and cost of spare parts, d) system monitoring capability, and e) noise. Many of these older installations use gas-fired compression equipment and lack operating flexibility, making it difficult to meet varying flow conditions. This is a concern given the projected growth in the gas market over the next decade. Therefore, the reliability, capacity, and efficiency of these older installations need to be upgraded through significant investment or replaced with more advanced and cost efficient designs.

The objective of developing a new integrated electric motor driven compression system was to achieve a reliable, low installed cost, and low maintenance design that provides high operational flexibility to meet the requirements of the gas industry. The idea was to develop a compressor system that could be installed directly in the pipeline, with the main objective of reducing overall pipeline costs. An overall pipeline cost savings would be achieved by decreasing the cost of the compressor station and having shorter distances between boosting stations thus increasing the average line pressure. In addition, if the unit had minimal building requirements (i.e., to house the compressor) and could be capable of underground installation, noise could be reduced, while also providing the potential of increased security for the compressor station.

The objectives of the project were to be accomplished by integrating a centrifugal compressor and a variable speed electric induction motor that utilizes magnetic bearings mounted inside the pressure-containing casing. Goals for the new compression system included low unit cost with a high degree of reliability to provide low maintenance gas compression that can be rapidly ramped up to meet peak demands. Some of the major benefits expected from the new compression system include highly reliable operation for the gas pipeline operator with no on-site emissions and reduced costs of system installation and maintenance.

Experimental

No true experimental work was performed during the performance period of the award. There was no testing vehicle developed or any empirical data gathered. However, there was experimental work done via computational analysis. That is, there were comparative studies performed on various configurations or component designs using various aerodynamic, rotor dynamic, or stress analysis codes to optimize or otherwise understand the new designs.

The analytical tools employed fall into two basic categories. The first are the empirical codes, which have been developed based on past design and test data. Such codes depend heavily on prior experience to assess the configurations being studied. The second are the more advanced finite element or finite volume codes, which use mathematical models to approximate or resolve parameters or features of interest. These latter codes do not depend on prior experience, relying instead on complex models of the mechanical or flow physics involved. In this latter category are the finite element analysis (FEA) codes for studying the response of materials to the forces imposed due to motion or external influences such as aerodynamic forces. Also in this category are the computational fluid dynamics (CFD) codes that can be used to analyze very complex flow passages. There will be no detailed discussion of the various codes used in this document as there is an abundance of material available in the open literature describing such codes.

The basic approach used for conducting analytical or computational experiments are identical to those used for vehicle or rig testing. The competing design options are analyzed using the various mechanical or aerodynamic codes. The parameters output by the codes for the different configurations are then weighed against one another and against the design requirements. Modifications are made to the component or system geometries, new analyses executed, and the new results again compared to the requirements and to prior iterations. This iterative process continues until an optimal solution is found.

There are clear economical advantages to conducting computational testing versus the costs associated with experimental test vehicles. Inadequate designs can be “filtered out” via the analytical assessment, leading to significant savings on any planned test program.

Further details on the mechanical and aerodynamic analytical approaches will be provided in the results and discussion sections that follow.

Results and Discussion

Summary of Design Objective Achievements

The objective of the project was to design a direct coupled, seal-less (hermetically sealed), In-Line Electric Motor Driven Gas Compressor (IEMDC). The required end result of the project was to develop the design of the IEMDC to the point where detailed manufacturing drawings may be started. This objective has been achieved and results of the project are summarized in the deliverables of the award.

The IEMDC system developed under this cooperative agreement with the DOE NETL is a natural gas compression system that is capable of being installed directly inline with the gas pipeline. This unique compression system design was achieved by integrating a centrifugal compressor and a variable speed, high power density, electric induction motor. The entire system including the electric motor is completely contained inside the pressure-containing casing boundary. This design eliminates the need for shaft seals that can leak gas to the atmosphere thereby eliminating any on-site emissions. Additionally, the system is designed for low cost unit installation and with a high degree of reliability for the purpose of providing low maintenance gas compression that can be rapidly ramped up to meet peak demands.

Several design goals, listed below, were established for the project in addition to the primary objective to develop the IEMDC system.

- Provide a cost effective design by developing a system that is simple to install and is low cost to operate,
- Provide a system that reduces environmental impact by eliminating shaft seal leakage and lubrication oil,
- Provide reduced or simplified maintenance concerns as compared to gas-fired equipment,
- Provide additional security by producing a design with the potential to be installed underground (compact design).

The IEMDC system was developed with the above goals as a criterion of the design. Furthermore, the final system design that has evolved after the completion of the project has successfully incorporated all the design goals that were established at the outset. Thus, success of the project can be measured not only by the achievement of the project objective and goals, but also by the benefits that the developed system will provide to the pipeline gas transmission industry. Market research performed during the course of the award and the positive feedback received from potential pipeline operators has confirmed the viability of this unique compression system.

One of the primary considerations of the IEMDC unit was to design a system that is environmentally friendly. This has been achieved by eliminating the shaft seals with the potential to leak to atmosphere, and by supporting the rotor on magnetic bearings. The magnetic bearings eliminate the need for oil lubrication. Designing a totally enclosed unit with no shaft seals that penetrate the pressure containment boundary eliminates shaft seal emissions. A totally enclosed design also reduces the requirements for blowing down the compressor station to atmosphere since the station compressor loop can remain pressurized during a shutdown.

Reliability of the system is expected to be high with an integrated electric motor with a minimum projected life of 20 years and 5,000 start-stop cycles. This will result in the reduction of operating costs over the entire life cycle of the unit. There are no major components that would require regular maintenance as usually occurs with gas-fired equipment. Additionally, there is easy accessibility to the compressor section since no process piping, instrumentation, or auxiliary connections would be required for removing the compressor bundle.

The IEMDC is a compact design relative to a traditional compressor train. Installation costs of the integrated and compact design of the IEMDC unit allows it to be direct mounted on a foundation that does not require a base plate. Eliminating the base plate results in reduced installation costs. This attribute in conjunction with the ability to install the unit inline with the process piping provides the unit with the potential to be installed or located underground. However, the control systems and power sources will most likely be located above ground until further advancements are made to reduce the size of these systems. The future advancement of fuel cells is one possibility for furthering the opportunity to install the IEMDC system underground.

In summary, the project has resulted in an integrated electric motor compression system design that meets the objective of the cooperative agreement and the goals of the project as determined by market research.

Summary of Project Activities

Scope

The scope of the work undertaken on the project included the following:

- Development of the compressor aerodynamic flowpath and pressure containment boundary.
- Development of the high-speed, gas-cooled motor.
- Development of the motor drive specification.
- Definition and engineering of the compressor/motor interfaces, including cable penetrations, gas cooling configuration, motor-mounting arrangement, system rotor dynamics, and system controls.
- Commercialization activities.

Integration

At the start of the project, it was understood that both the electric motor and the compressor have unique operating characteristics and requirements. Consequently, achieving successful integration of the motor and compressor as a common unit required that the interactions between the two be fully understood. The motor cooling gas circuit (configuration and method), rotor thrust balancing, and overall operating performance are a few of the significant areas affected by the interactions of the compressor and motor. Close teamwork and open communication was required between the compressor and motor design teams to understand the implications and impact of each others design changes and requirements. The teamwork and cooperative environment helped greatly to successfully accomplish the challenging design task of integrating the motor-compressor system.

The first consideration for the unit design was given to the thermodynamic operating performance of the system and the electric motor design. Motor cooling gas requirements have a significant impact on the system overall performance. The motor cooling circuit method establishes an arrangement for configuring the unit. Ultimately this arrangement and the geometry of the motor would result in establishing constraints around which the compressor aerodynamic flowpath would have to be designed. Many design iterations on the electric motor took place. During this process, the impact of the motor cooling circuit on the thermodynamic performance was continuously reviewed to determine if system losses could be maintained at an acceptable level. Evaluations performed on the system indicated that overall system losses would be similar to those of a conventional system comprised of a compressor, gearbox, and electric motor.

Many discussions ensued with regard to configuring the cooling circuit. These discussions included determining pressure levels of the motor cooling gas internal to the motor, the effect of cooling gas recycle on compressor power, and the impact of these design parameters on the overall unit operation. At this point, the critical interactions and elements for system integration were beginning to stand out. Therefore, the design team realized that for successful integration of the motor and the compressor, comprehensive evaluations of thermodynamic performance, motor and bearing cooling circuit configuration, and motor compressor thrust balancing must be performed simultaneously.

Thermodynamic performance and the impact of the electric motor cooling method was continuously reviewed between the compressor and electric motor design teams. The pros and cons of cooling the electric motor with process gas at inlet pressure or discharge pressure were being evaluated. Primary were the concerns of windage losses that would be generated in the motor and the magnetic bearings at high rotational speeds in the high-pressure gas environment, and the need for effective internal sealing. Using high-pressure natural gas as the motor cooling medium was a performance compromise since it increased the windage losses within the unit. The windage losses constitute a very significant portion of the overall motor losses and are the primary parasitic losses associated with running a high-speed electric motor in a high-pressure gas environment. However, these concerns were greatly minimized by throttling the pressure down to a level that is required to develop the required differential pressure across the motor.

Thrust balancing was another one of the technical design challenges identified at the beginning of the design process that received much attention. A method of balancing the thrust forces generated within the unit had to be developed. Various ideas were generated for pressure balancing the rotor. Each of the ideas were evaluated as a possible solution to the problem. The motor cooling circuit arrangement and unit configuration were a major consideration during the evaluation process since the pressure balancing method could not be developed independently of these items.

One method evaluated for balancing the thrust forces generated by the impeller was to use the thrust disc as a balance piston. The thrust disc was one of the only components on the rotor with enough area with which to create a differential pressure across and produce significant thrust forces. Access to the other major component on the shaft was more restricted by the configuration. This in turn resulted in a decision to use cooling gas at or near compressor discharge pressure. A high-pressure zone would be created on the inboard side of the thrust disc and a low-pressure zone on the outboard side. His was a concern regarding the accuracy of the predictability of pressures at the thrust collar. This concern was compounded by the need for developing a proper sealing system design at the OD of the thrust collar. Furthermore, concerns existed with regard to providing proper cooling of the magnetic thrust bearing without creating thermal distortion, and controlling the balancing system due to potential sensitivity of the system to pressure fluctuations. Additionally, the impact of the design on the mechanical configuration and assembly of the unit had to be considered.

In addition to the thrust issues and cooling configuration discussions, many design concepts were evaluated to arrive at solutions to the problems presented by the original configuration. These issues included the placement of internal gas passages, aerodynamic passage design, motor structural support, and assembly methods. Design considerations included in these evaluations were manufacturing tolerances, thermal expansion of components, sealing and leakage, anti-rotation devices, wire and cable passages, and motor structural support.

The original design concept of the IEMDC is shown in Figure 1. It can be seen that the compressor flowpath surrounds the motor. This design results in flow passages that traverse the entire axial length of the motor and supporting internal components. The flow passage geometry is constrained by the size of the electric motor, structural support components, and utility passage requirements. Long stream wise struts are necessary to support the motor and provide passages for instrumentation and power cables as well as cooling gas. The size and location of the passage geometry dictated the width of these struts. These constraints were expected to impede the compressor aerodynamic operation.

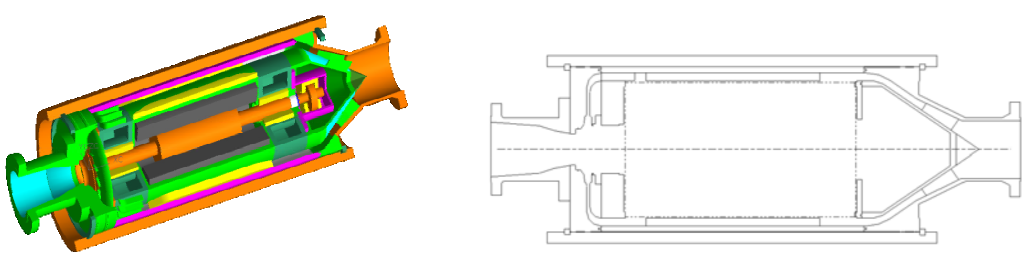


Figure 1. Original IEMDC Configuration

Many motor structural design support concepts were considered and discussed during the evaluation of the unit configuration. Various support concepts included multiple sleeve designs, spoked wheel shape supports, spring designs, etc. Unfortunately, many resulted in complex assemblies that could not meet the intent of the design considerations for the primary aerodynamic flow passage(s). Many of the concepts discussed presented difficult and challenging manufacturing and assembly configurations. At this point in the design process, there existed a major conflict between proper motor structural support and the aerodynamic design of the unit.

Subsequently, many design concepts for internal components were evaluated to address the balancing of the aerodynamic requirements of the flowpath passages with the requirements for supplying utilities and structural support for the motor. Many of these design concepts intended to address the issues of electrical cable penetrations, cooling duct passages, and motor structural support requirements and their impact on the aerodynamic design. Assembly and manufacturing issues were also considered to address the resulting increasing size and complexity presented by the various design concepts. The inability to balance the requirements for these items and the conflicts with the aerodynamic requirements led to a process check with regard to the design and commercialization of the original concept.

The goal of the process check was to review the design and determine if a more practical and simpler method could be developed to facilitate the integration of the centrifugal compressor and variable speed electric motor. Eventually the concept of an alternate configuration was envisioned. This alternate configuration was a modular design with the electric motor and compressor section each being primarily contained in its own pressure containing case. This proposed concept looked very promising to resolve the conflict between the aerodynamic flow passage requirements and electric motor requirements for support and utilities by bounding the flowpath within the compressor section. A meeting was therefore held to evaluate the concept. After considerable discussion and evaluation of the proposed concept, the team agreed to proceed with the alternate configuration concept.

The concept change, briefly described, was essentially a modular design that would still maintain the technological advantages of the IEMDC concept as well as the objective and goals of the project. This reconfiguration is designated as the "alternate configuration" as seen in Figure 2. Expectations for the configuration change at that time were that it would resolve or simplify the issues presented by the original IEMDC configuration concept shown in the award solicitation. This is due to the modular concept of the configuration that has both a compressor section and an electric motor section. The compressor section compartmentalizes the primary process gas flow passages minimizing conflicts with the electric motor structural requirements and utility interface requirements. In addition, compartmentalizing the electric motor section allowed easier access for utilities and fewer restrictions for structural support since there are no constraints from the aerodynamic flowpath.

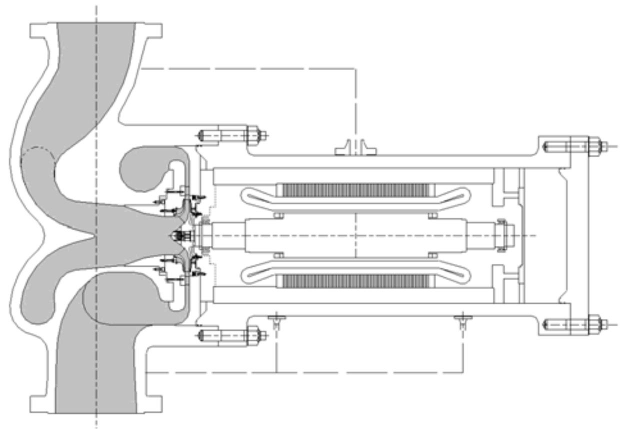


Figure 2. Alternate Configuration – Initial Concept

The final IEMDC modular configuration offers significant advantages over the original design concept. It is easier to maintain and is overall more commercially attractive from the design aspects of installation, operation, and maintenance. Interaction and feedback solicited from target clients regarding the design configuration supports the fact that the design addresses industry issues regarding accessibility, maintainability, preferred operating practice, and increased reliability. In addition to achieving benefits sought by the gas industry, the resulting system/unit configuration delivers the following advantages:

- Modular construction provides improved accessibility and maintainability of critical change out parts such as the impeller and bearings without the necessity to break critical system seals.
- Optimized aerodynamic flow path for the most efficient compression.
- Effective and efficient motor cooling design.

- Compact and easily maintained gas filtering arrangement.
- Minimized system weight.

A significant advantage of the alternate configuration was a more effective and efficient motor cooling scheme. A schematic representation of this cooling flow scheme is shown in Figure 3. This diagram shows the thermodynamic model used to develop the unit performance. As can be seen in the diagram, the motor cooling flow and internal leakage is recycled to the inlet of the compressor. The result is that the losses incurred within the motor appear as increased mass flow at the inlet of the compressor.

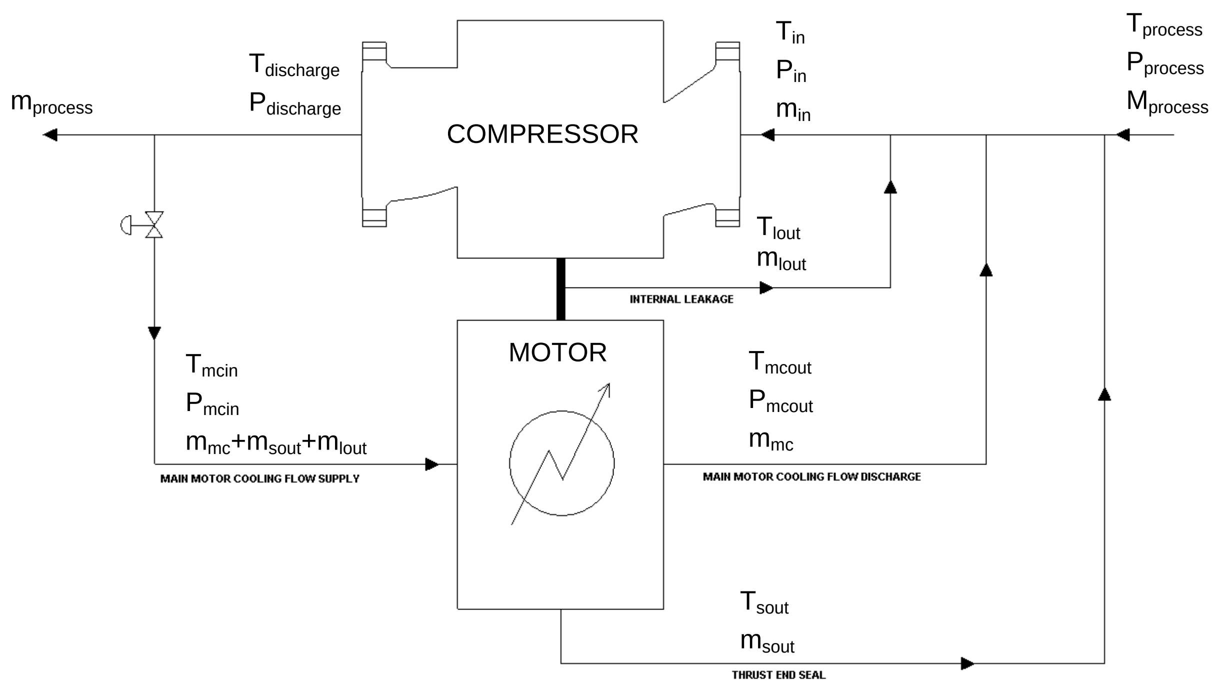


Figure 3. Diagram – Cooling Flows

It was recognized by the design team that in spite of many advantages the new modular concept also posed two significant new technical challenges. The first issue concerns the aerodynamic induced radial forces imposed upon the shaft from the discharge system, which were not an issue with the original concept. The second issue concerns the need for joining two separate pressure-containing cases. The aerodynamically induced radial loads reanalysis of the motor compressor rotor and redesign of the magnetic bearing at the impeller end to accommodate this additional load. Therefore, both compressor and motor design teams, to address the issue of aerodynamic radial loads resulting from the alternate configuration, expended significant design efforts. Initially, an empirical design prediction method was used to estimate the magnitude of these loads for subsequent evaluation by the motor design team. The electric motor team began spending considerable efforts to evaluate bearing design configurations and performing rotor dynamics evaluations to bring this issue to a resolution. The effort and evaluation of these design issues will be discussed later in the report.

Figure 4 shows a pictorial summary of the development cycle and evolution of the final IEMDC design configuration. Version 2, in Figure 4, shows changes to the end-style enclosures of the pressure containing case. End enclosures were changed from a shear ring style to a bolted shoulder fit. This change was made to increase the capability of the end enclosures to resist piping forces. The design of the bolted shoulder fit end enclosures was expected to better resist and support the pipe forces transferred to the compressor pressure containment housing and internal structure of the unit. Providing a solid structure upon which to attach the pipe to the unit significantly reduces the risk to the integrity of the sealing surfaces and internal components.

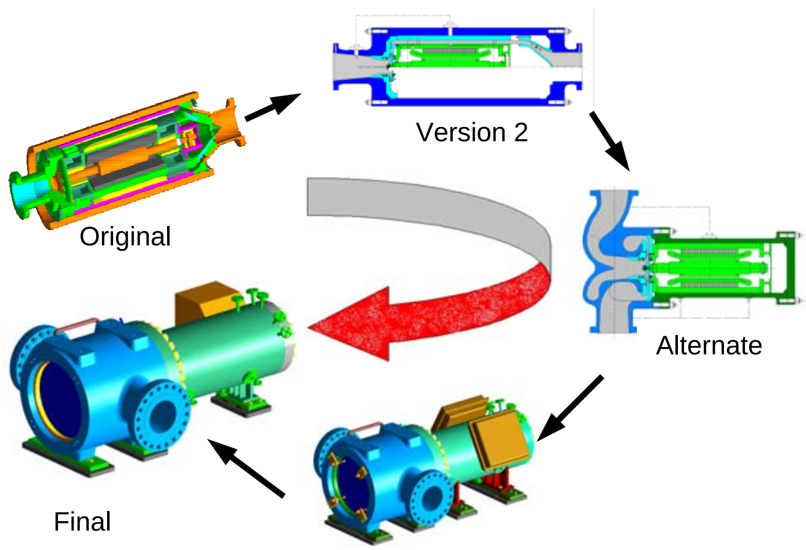


Figure 4. Configuration Development

Auxiliary Systems Development

Development of the unit configuration was primary to the development of the IEMDC system interface and control systems. The unit configuration had to be established before the motor cooling scheme, compressor and motor operational interaction, and utility penetrations could be determined. As a result of determining the unit configuration, the required customer interfaces required for operation of the IEMDC became apparent. Subsequently, the complete definition of the interfaces required to control the unit including the interaction with the supporting auxiliary equipment were defined for complete integration of the system.

An overall system instrumentation and control diagram is shown in Figure 5. System integration focused on the control interfaces and hardware required for system installation. There are several main auxiliary systems for control and control interfacing. These items consist of the variable frequency drive (VFD) system, magnetic bearing control panel, and main unit control panel. It was also identified that a cooling gas conditioning system is required to maximize the reliability of the system. However, note that no auxiliary system software was written or programmed during the period of this award.

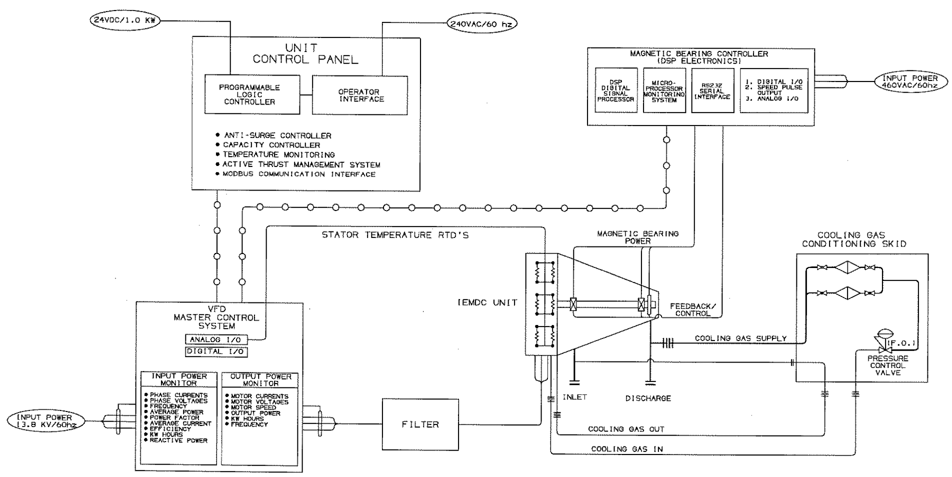


Figure 5. System Interface Diagram

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 1. The communications board is configurable to allow network communication via a variety of protocols.

Table 1 – 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 located in the motor stator slots. The Master Control System of the VFD, per Tables 2 and 3, 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 2 - VFD Input Power Monitoring	
Phase Currents	Average Current THD
Phase Voltages	Efficiency
Frequency	KW Hours
Average Power	Reactive Power
Power Factor	

Table 3 - VFD Output Power Monitoring	
Motor Currents	Output Power
Motor Voltages	KW Hours
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.

The gas conditioning system will regulate the supply pressure and filter the gas used to provide ventilation for the motor. Pressure regulation is required since the source of the gas supply is from the compressor discharge line. Using compressor discharge gas for cooling mitigates the risk of liquids entering the motor during operation since the heat of compression will warm the gas. A filtration system was selected that will remove both solid particles and trace liquids from the cooling gas before entering the electric motor. The filter system is designed with two parallel filters and transfer valves that permit changing of the filter elements while the unit is in operation. Transfer valves allow uninterrupted maintenance capability that was a design criterion established for the system. Temperature and pressure gauges will be provided with the system for the purpose of monitoring the cooling gas supply conditions. Check valves and isolation valves will also be supplied in the system as required by the compressor installation site.

Appendix A contains an assembly outline of the unit that identifies the required operator connections. These connections include those required for the process pipe, motor cooling, and interface control. A list of these connections is shown in Table 4.

Table 4. Unit Connections

CONNECTION LIST	
CONN POINT	DESCRIPTION
A,	INLET-IEMDC COMPRESSOR MAIN INLET
B,	DISCHARGE-IEMDC COMPRESSOR MAIN DISCHARGE
C,	DRAIN-IEMDC COMP'R. CASING DRAINS
D,	INLET-MOTOR COOLING GAS
E,	VENT-MOTOR SEAL VENT
F,	MOTOR HV POWER CONNECTION
G,	SUPPLY-MOTOR COOLING GAS FROM CUSTOMER'S DISCHARGE PIPING TO COOLING GAS CONSOLE
H,	DISCHARGE-MOTOR COOLING GAS DISCHARGE
J,	ELECTRICAL CONN.- COMPRESSOR END RADIAL BEARING TEMPERATURE ELEMENTS AND POSITION SENSORS (JB-1).
L,	ELECTRICAL CONN.- COMPRESSOR END RADIAL BEARING #1 POSITIONING POWER COILS (JB-2)
M,	ELECTRICAL CONN.- MOTOR FREE END BEARING TEMPERATURE ELEMENTS/ POSITION SENSORY SPEED SENSORS, AND FLUX SENSORS (JB-3).
N,	ELECTRICAL CONN.- MOTOR FREE END BEARING POSITIONING POWER COILS (JB-4).
P,	ELECTRICAL CONN.- MOTOR STATOR RTD'S. (JB-4).
Q,	MOTOR HV POWER CONNECTION
R,	MOTOR LOW POINT DRAINS
S,	COOLING GAS SUPPLY FROM CONSOLE TO MOTOR
T,	COOLING GAS REFERENCE FROM CUSTOMER'S SUCTION PIPING TO COOLING GAS CONSOLE
R,	INERTGAS PURGE CONNECTION

Operating Parameters

The nominal operating parameters of the IEMDC system that was developed under the award are shown in Table 5. These parameters can be optimized via speed variation or selection of the aerodynamic components. It is capable of a maximum nominal power output of 10MW at a maximum continuous operating speed of 12,000 RPM. A speed range of 70% to 105% was targeted for the design; however, the robust, sub-critical rotordynamics design permits operation down to around 50% speed depending upon actual operating conditions. Efficient performance can be achieved at low powers as the motor operates with an efficiency level in the range of 92% to 95% even at reduced power levels. This system was designed for operation in natural gas applications compressing pipeline quality natural gas.

Table 5
Nominal Operating Parameters

Parameter	Nominal Range
Pressure Rating (psig)	1500
Flow (ACFM)	4,000 to 10,500
Speed Range (RPM)	8,000 to 12,000
Power Output (MW)	5 to 10

Although the IEMDC is capable of operating over a wide range of parameters, the performance objective of the IEMDC unit was to select an operating set that would be applicable to a wide range of customer applications. Therefore, an operating design point was selected to establish design parameters for a subsequent prototype unit to be manufactured and tested. The design inlet pressure parameter for the unit design point falls outside the goal originally established for the project. Calculated design operating point selected for the IEMDC unit for the project is as follows. Table 6 lists the project design goal operating parameters as compared to the design point selected for the basis of the IEMDC design. Table 7 shows the electric motor operating parameters. Additionally, the operating map that was developed for the prototype unit is shown in Figure 6.

Table 6. Project Design Goal Parameters and Selected Design Point

Parameter	Project Goal	Selected Design point
Inlet volume flow rate (MMSCFD)	300 to 700	613
Inlet pressure range (psig)	500 to 700	763*
Discharge Pressure Range (psig)	700 to 1200	1100
Inlet Temperature Range (°F)	60 to 80	60
Operating Speed Range (RPM)	8,000 to 12,000	11268
Power Consumption (MW)	5 to 10	8.7

*Outside the anticipated range, but not outside the operating limits of the IEMDC.

Table 7. Motor Design Operating Parameters

Parameter	**Value**
Motor Type	Squirrel Cage Induction
Rated power (MW) [HP]	10 [13,405]
Rotational Speed (RPM)	12,000
Rated Torque (Ft-lb.)	5900
Maximum Torque (ft-lb)	6500
Efficiency at Rated Power (%)	95
Bearing Type	Magnetic
Phase Number	3
Winding Configuration	2 Parallel Wye
Maximum Frequency (Hz)	200
Pole Number	2
Number of Terminal Connections	6
Maximum Voltage (kV L-L)	6.9
Cooling System	Filtered Ventilation w/ Methane Gas
Phase Current (amps)	1118

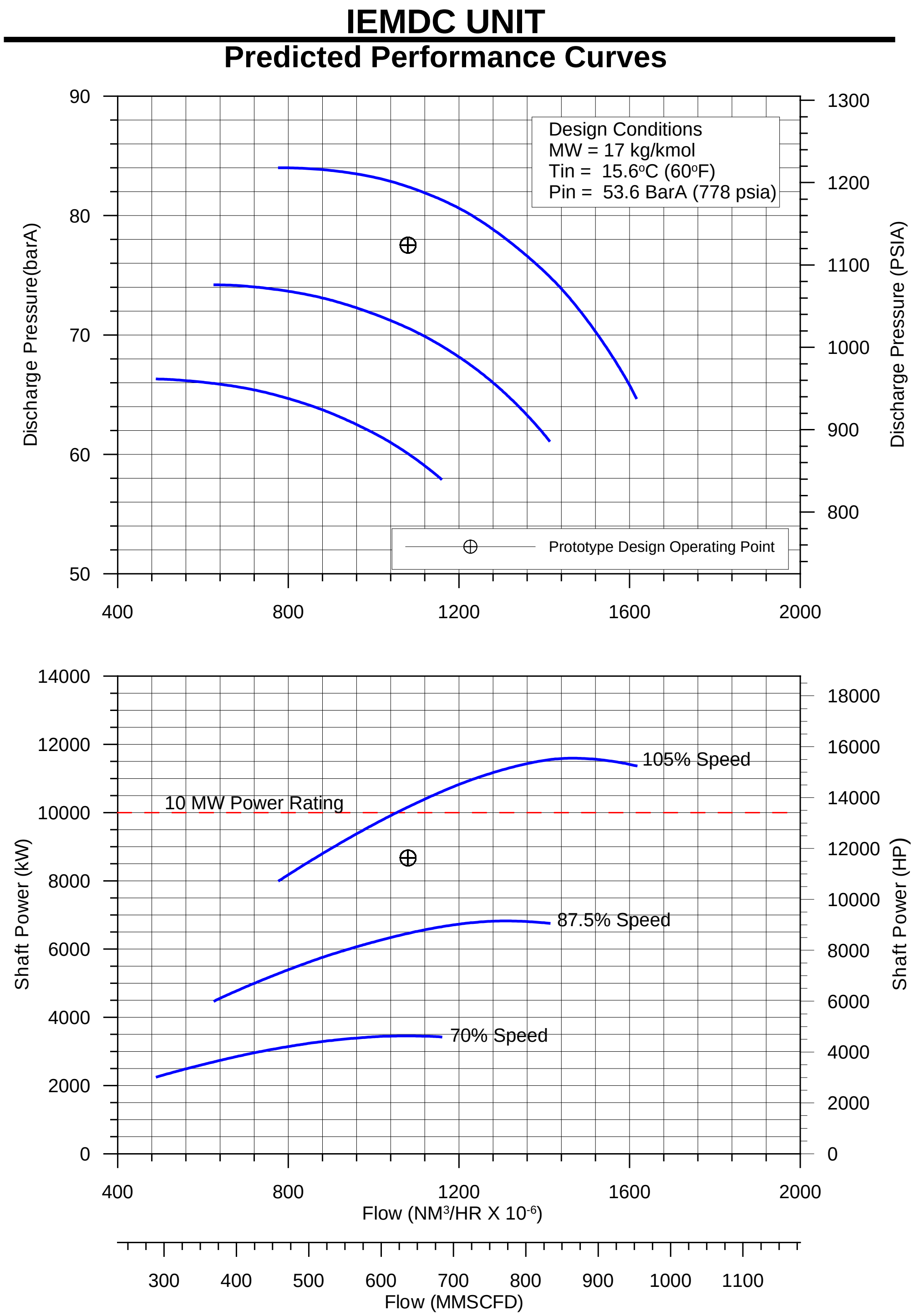


Figure 6. Prototype Design Operating Map

Compressor Section Design

Project efforts and tasks were undertaken to design the following components for the compressor section of the IEMDC:

- Aerodynamic flowpath design
- Pressure containing case
- Inlet
- Volute design
- Inlet guide
- Internal component and motor interface design
- Nozzle design
- Solid model development.

Features and operations that were considered and designed in conjunction with the motor efforts included:

- Thrust balancing arrangement
- Motor cooling flow scheme
- Unit performance
- Unit mounting arrangement.
- Off design operation.

Aerodynamic Flowpath design: The basic aerodynamic flowpath of the compressor is comprised of a radial flow inlet, an overhung impeller, a diffuser, and an overhung volute. All of these components are designed or selected to achieve the best possible performance for a single stage compressor. In addition, state-of-the-art techniques such as finite element analysis and computational fluid dynamics (CFD) were used throughout the compressor design process. Finite element analysis was used during the pressure containing case design process as well as for critical internal component design. Extensive computational fluid dynamics analyses were performed on the aerodynamic components of the compressor flowpath. These analytical efforts were undertaken to optimize and validate the design.

Significant effort was expended on developing the flowpath of the compressor section to design the inline configuration of the unit. The inline configuration results in a unique flowpath design that is optimized for maximum efficiency with a radial inlet design. Unlike a multistage compressor, where space is limited and compromises are made with regard to bearing span, there are no space restrictions on the single stage IEMDC. This allows the inlet to be designed such that the flow can be turned radially with low velocities. Subsequently, the configuration allows the gas to be accelerated along the axial path leading to the impeller eye. This approach has resulted in a very low loss radial inlet design.

This atypical flowpath arrangement resulting from the inline configuration also posed challenges with regard to determining the optimum volute geometry. This resulted in considerable effort being expended on designing a volute for the inline configuration both aerodynamically and mechanically. Many iterations were carried out to arrive at a flowpath that optimized the use of space and resulted in a design that was satisfactory for manufacturing. Figure 7 shows the compressor aerodynamic flowpath and resulting complex passage intertwined paths resulting from the inline configuration. The design shown in this figure was developed in the earlier part of the alternate configuration compressor development.

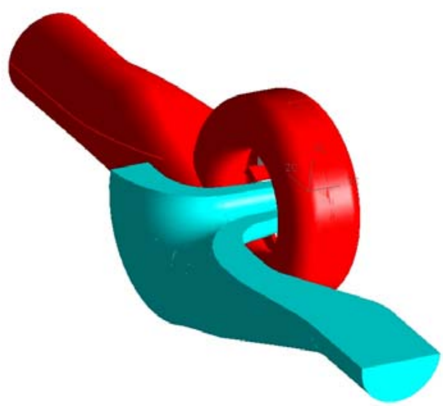


Figure 7. Aerodynamic flowpath surfaces.

Designing the flowpath for the inline configuration presented aerodynamic and mechanical challenges that resulted in multiple design iterations during the design process. A balance had to be maintained between the aerodynamic design and the mechanical design of the unit. Constant evaluation of the design was made to the aerodynamic flowpath design to facilitate the mechanical design of the unit. Modifications to the aerodynamic flowpath were ongoing throughout the project because of the mechanical considerations required of the inline configuration. Typically changes that were made during the design of the aerodynamic flowpath were in the form of flowpath cross-sectional area and shape as the geometry was developed. Constant evaluations were made to determine if the proper sectional areas and shape could be maintained relative to the results as determined by aerodynamic analysis.

Extensive computational fluid dynamics undertaken during the course of this award included both steady state and transient analyses. Analyses were performed both on individual aerodynamic component designs and on the entire flowpath geometry. These analyses used various modeling methods including pie-slice and 360-degree models. One of the complete computational fluid dynamics (CFD) analyses performed on a 360-degree model of the final compressor geometry included secondary flow passages such as the impeller seals. Analyses were also performed with and without diffuser vanes and the results were used to provide predictions of thermodynamic performance and aerodynamically induced radial loads. These predictions were then used to optimize the design of the aerodynamic flowpath components. Figure 8 shows a model of the final geometry of the IEMDC flow passage used in various analyses.

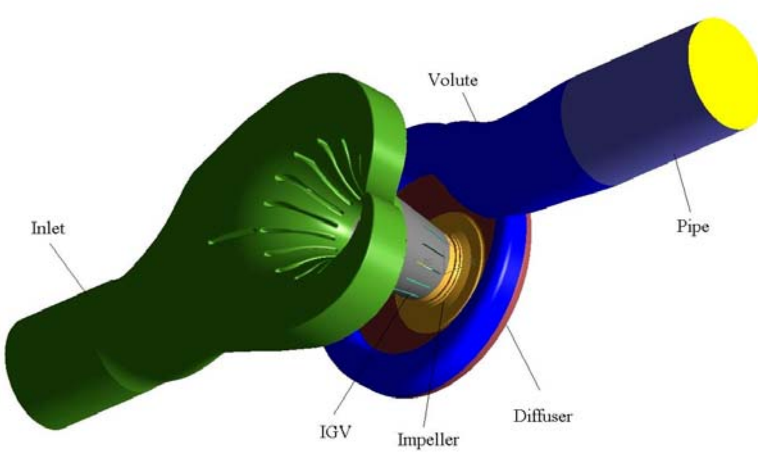


Figure 8. IEMDC Final Compressor Stage Geometry

CFD Analysis Models: Figure 9, 10 and 11 show meshes used for computational fluid dynamics analysis performed on flowpath components. Figure 9 shows a pie slice model of an impeller with an axial inlet extension used in the pie slice analyses. Analyses of the full 360 degree models include the inlet guide vane shown in Figure 10, and a full 360 degree impeller model with secondary passages as shown in Figure 11.

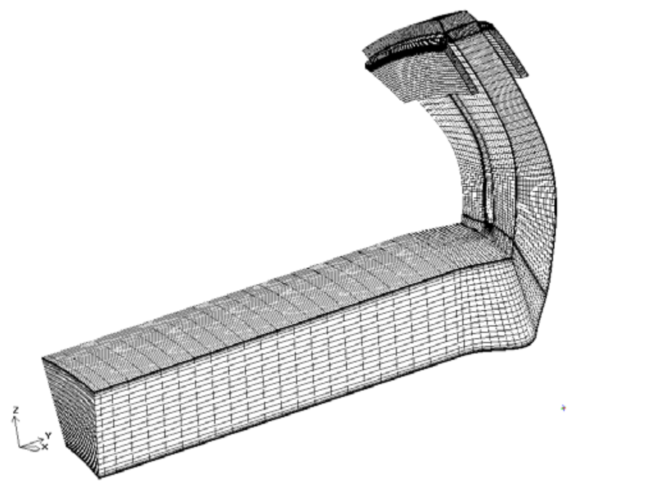


Figure 9. Pie slice impeller mesh with inlet extension.

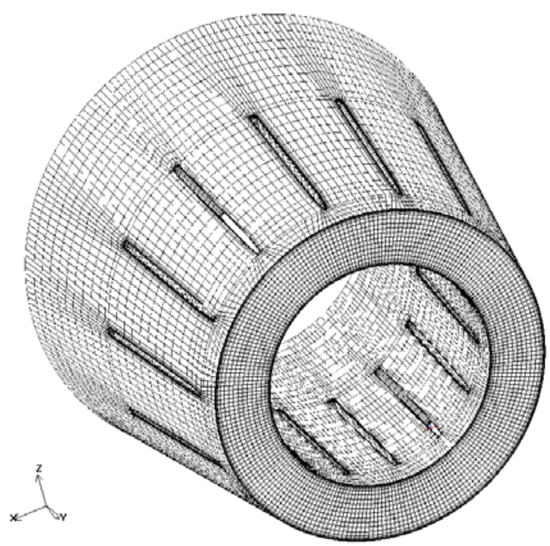


Figure 10. IGV computational mesh.

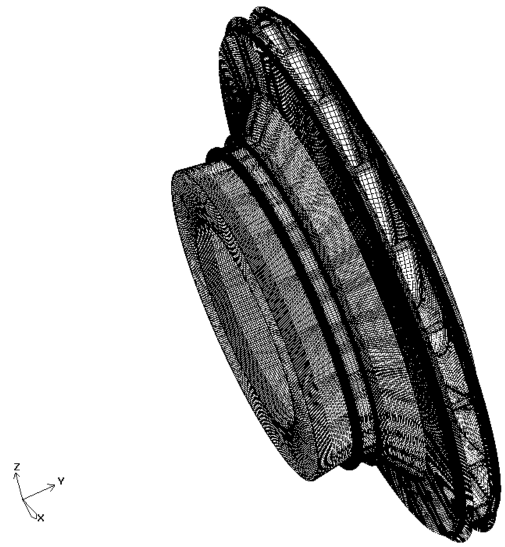


Figure 11. Full 360 impeller mesh with secondary passage

Inlet Design: The radial inlet of this compression system was designed to minimize the total pressure loss through the inlet passage as well as the flow distortion. Minimizing flow distortion is critical to obtaining a uniform flow field at the impeller eye. This was done to avoid undesirable flow characteristics that could negatively influence the performance of the downstream components. Further optimization of the inlet design was achieved by the use of turning and straightening vanes in the inlet flowpath. The vane setting angles were selected to minimize incidence angle variation over the entire operating flow range. This ensures a design that is insensitive to changes in the flow rate with regard to minimizing total pressure loss and flow distortion through the inlet.

Initially, two inlet concepts were evaluated for use in the IEMDC during the design process. These inlet designs are shown in Figures 12 and 13. 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 incorporated a scheduled area distribution to minimize turning losses. Both inlet designs were analyzed using computational fluid dynamics. A comparison of the two inlet designs was made to select the best option for the IEMDC. Results of the steady state computational fluid dynamics 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. Eventually some of the less complicated features of the second design were incorporated into the final inlet design of inlet number one for the purposes of optimizing the design. This inlet was designated inlet number 3.

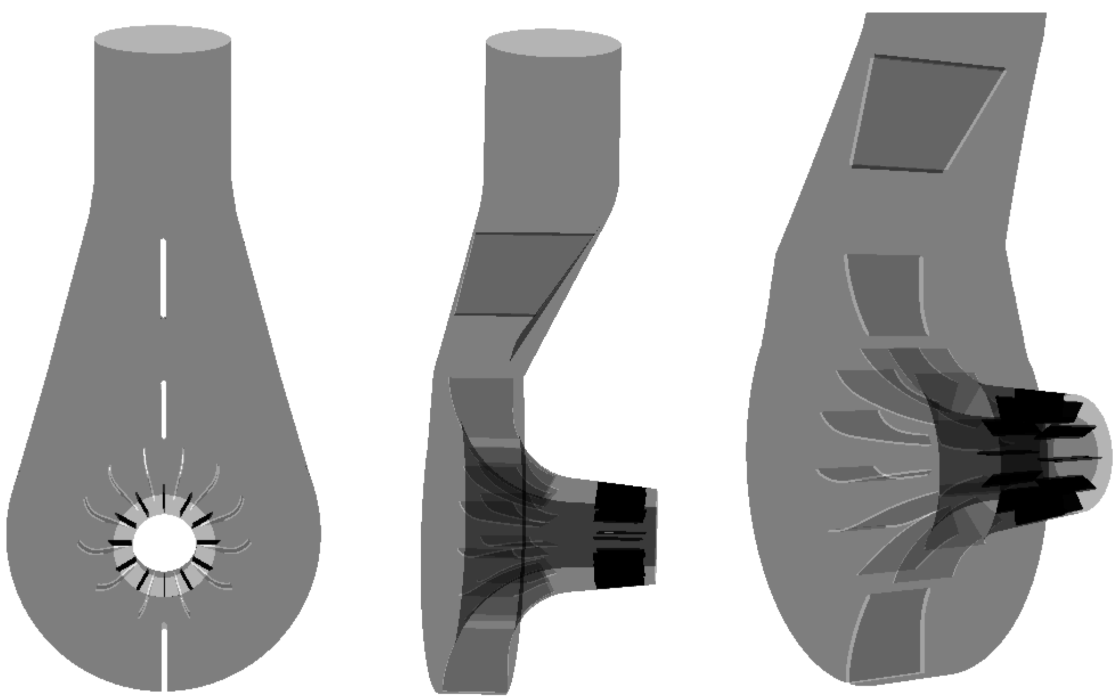


Figure 12. The geometry of radial inlet 1 (front, side, and rear view)

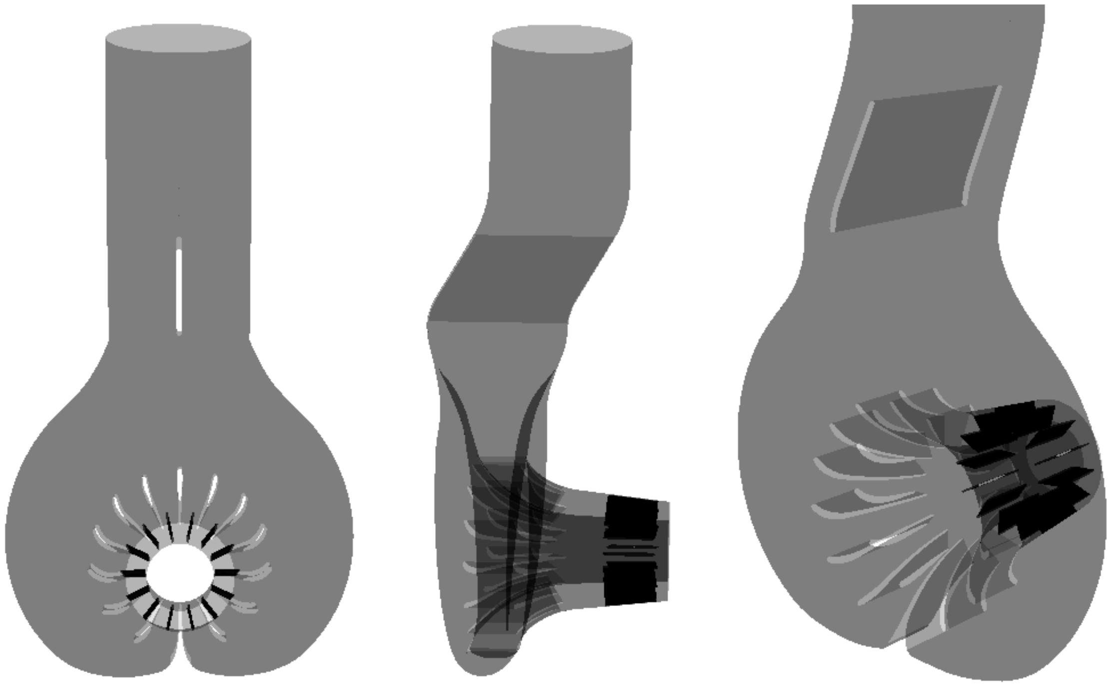


Figure 13. The geometry of radial inlet 2 (front, side, and rear view)

An aerodynamic analysis was completed on the final compressor inlet configuration (inlet number 3 shown in Figure 14). This inlet contains a combination of features that were analyzed in the first two inlet configurations. Inlet number three incorporates the inlet plenum region of the first inlet configuration and the vane designs of the second inlet configuration. As seen in Figure 14, the design of the geometry opposite the nozzle will provide improved area transition and better turning of the flow into the bottom inlet guide vanes. The new inlet design was evaluated using computational fluid dynamics at varying mass flow rates to determine the effect of changing operating flow conditions. Results of the analyses performed on the final inlet configuration indicated no adverse flow separation or incidence and that the inlet guide vanes direct the flow in a uniform pattern. Overall, the final inlet design was shown to be less sensitive over a wide range of mass flow conditions. A computational mesh of the final inlet is shown in Figure 15.

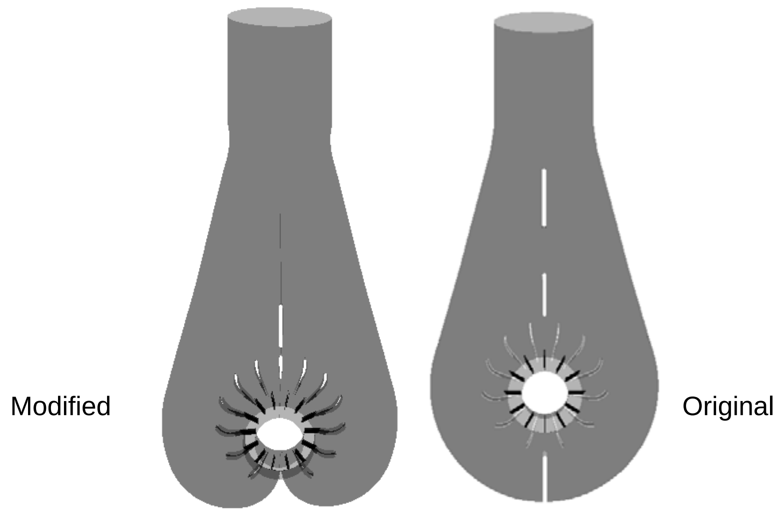


Figure 14. Inlet Configuration w/ Geometry Modifications (Inlet # 3).

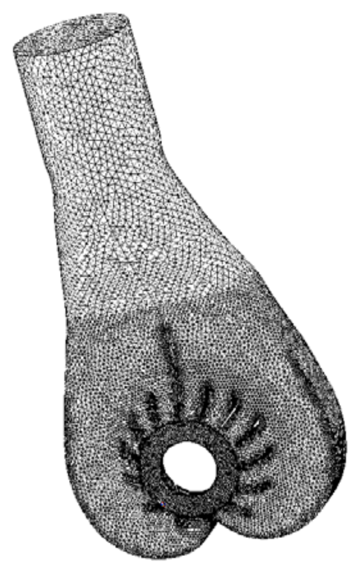


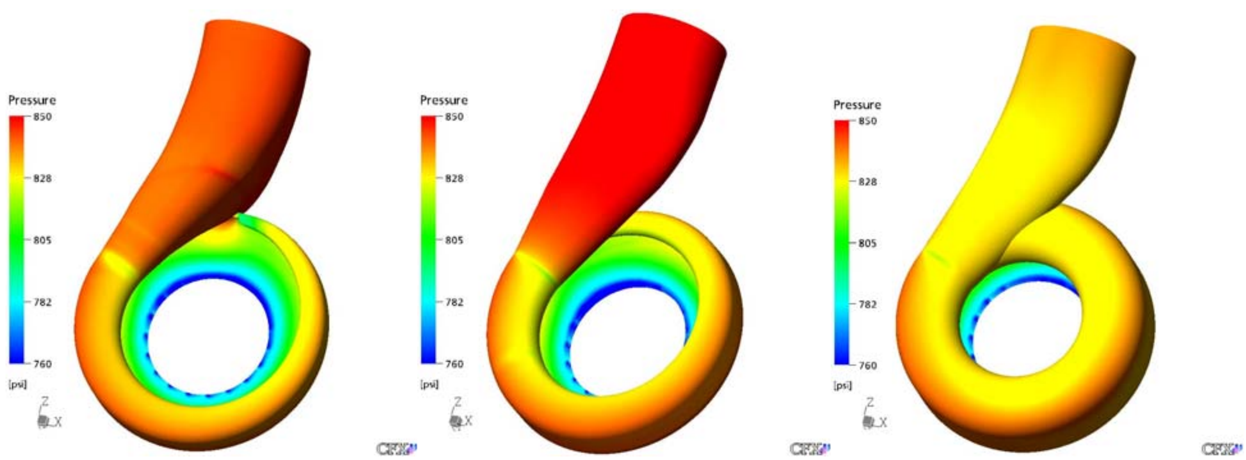
Figure 15. Inlet Computational Mesh for CFD Analysis.

Aerodynamic Radial Loads: Aerodynamically induced radial forces developed within the flowpath were an important parameter investigated during the design of the IEMDC. This design issue was brought about because of the change in the flowpath arrangement due to the change to the alternate configuration. The original concept proposed to use an axially inlet and volute-less discharge system. The new configuration required a radial flow inlet system with a volute or collector in the discharge system. This was an important design parameter considering that the unit has an overhung impeller and the rotor is supported on magnetic bearings. Aerodynamically induced forces are created by non-uniform pressure distributions within the flowpath. A resultant radial load is imparted onto the impeller because of this non-uniform pressure distribution. Consequently, this radial force must be accounted for when determining the reaction loads on the bearings. One of the design goals of the aerodynamic flowpath was to design a volute or collector that minimized the magnitude of the radial load force on the rotor, while achieving a high level of performance.

In order to achieve a high level of performance and minimize the aerodynamic radial loads, aerodynamic analysis including computational fluid dynamics work was performed throughout the development of the compressor section of the IEMDC. This effort was used to optimize the design of the aerodynamic flowpath and facilitate the mechanical design required for the inline configuration. Minimizing the aerodynamic radial forces required a significant amount of analysis and focus on the flowpath components downstream of the impeller, particularly the volute geometry and volute-diffuser interactions. The results of these efforts were evaluated to determine the effects of various volute tongue and diffuser geometry and their impact on thermodynamic performance and aerodynamically induced radial loads. The determination of the aerodynamically induced radial loads such that the subsequent impact on the magnetic bearing design could be ascertained was an important factor in the design. Figure 16 shows a 360 degree mesh of a volute used in performing a computational fluid dynamic analysis.

Figure 16. Volute Computational Mesh.

Volute Design: Several volute designs were analyzed using CFD during the development of the primary flowpath. These volute designs were evaluated to determine if established levels of aerodynamically induced radial forces could be met. Multiple volute designs were reviewed and compared 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 results on various aerodynamic configurations that were analyzed determined that on an overall basis the overhung style of volute was the preferred design configuration for the compressor. Figure 17 shows the various volute configurations and the relative aerodynamic radial forces generated in each configuration. Even though the overhung style volute selected has the highest aerodynamic radial forces, these forces are within the established design capacity of the impeller end magnetic bearing. Furthermore, this volute design has the highest predicted aerodynamic efficiency.



Partial Tongue Design

Full Tongue Design

Collector Design

Figure 17. Volute and Collector Configurations

The main difference between a collector and a volute as seen in Figure 17 is that collectors have circumferentially constant area, where as volutes have increasing area with the goal to maintain uniform circumferential velocity. Both volutes and collectors develop radial forces that are imparted upon the impeller. The radial force on the impeller and shaft is the result of a non-uniform static pressure developed along the volute circumference. This radial force is then transmitted to the bearings that must be designed to accommodate the additional load. Radial forces generated within the volute were an issue for the IEMDC because of the required use of magnetic bearings. Magnetic bearing technology has developed to the point where it is considered to be an alternative to traditional fluid film bearing technology, however, magnetic bearings are limited by the amount of radial load capacity that they can support. This necessitates that the radial loads are predicted with reasonable certainty and minimized to within acceptable levels. A maximum radial load limit of 1600l_{bf} was established as the allowable level for the impeller end magnetic bearing, based on the space envelope allocated for the magnetic bearing with regard to rotordynamic design considerations. Determining the magnitude of these loads, and whether or not these loads could be maintained within the established load level limit was a primary concern of the aerodynamic analysis.

Analysis of the three volute configurations discussed using steady state computational fluid dynamics resulted in the radial load vector development as shown in Figure 18. Magnitudes of the loads are scaled to the actual projected area of the impeller since the CFD analysis for the models under consideration did not include any secondary flow passages. The zero (0) degree circumferential position corresponds to the volute tongue location with the volute area increasing towards the direction of the shaft rotation. Notice that most of the plotted data points are located in the fourth quadrant where the volutes connect to the exit cone. It should also be noted that the collector shows minor variations in radial load with mass flow changes. In addition, the two volute designs with a tongue have higher loads at surge than the collector.

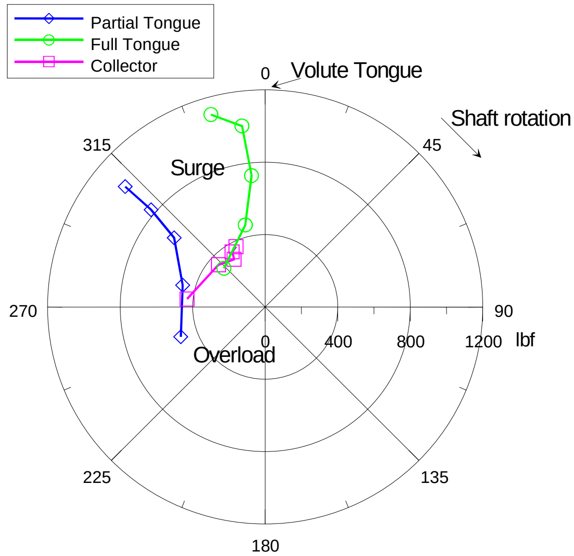


Figure 18. Radial load vector development in the volutes

Based on the results of the aerodynamic analysis performed on the three volute configurations as shown in figure 18, the following results were obtained,

- The collector has the lowest aerodynamic radial loads however, the performance is on the order of four percent lower in efficiency than the full tongue design and over two percent lower in efficiency than the partial tongue design.
- The full tongue volute design results in the lowest losses (best performance), but also generates the highest radial loads.
- Radial loads generated by each of the three volute designs are analytically shown to be within the 1600 lb_f limit. The radial loads of all the volutes are analytically shown to be less than 1200 lb_f.

Transient CFD Analysis: In addition to the steady state analysis methods used to analyze the flowpath for the developed aerodynamic radial forces, a 360-degree transient computational fluid dynamics analysis was performed. This analysis was performed on the complete compressor flowpath geometry. Figure 19 shows a particle traveling through the flowpath from the inlet to the outlet of the model. Symbols in this figure indicate the time between impeller revolutions as the particle travels along the flowpath.

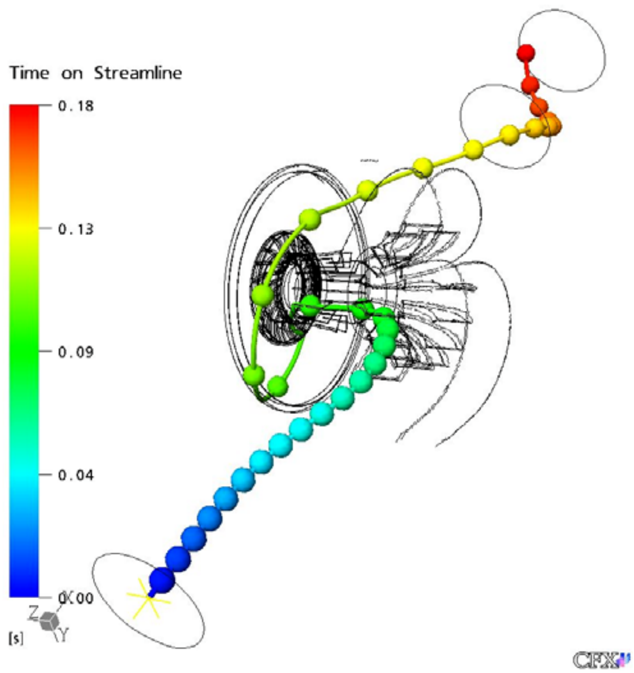


Figure 19. Particle time step through the flowpath.

Evaluation of the transient analysis, at the point at which the analysis was performed, indicates no substantial difference in the radial load calculation or thermodynamic behavior as determined by the steady state three hundred-sixty degree geometry analysis. Radial load results from the transient analysis are shown in Figure 20. Note that due to the extensive computation time required to perform a transient CFD analysis, only one flow point was evaluated and compared to the other results.

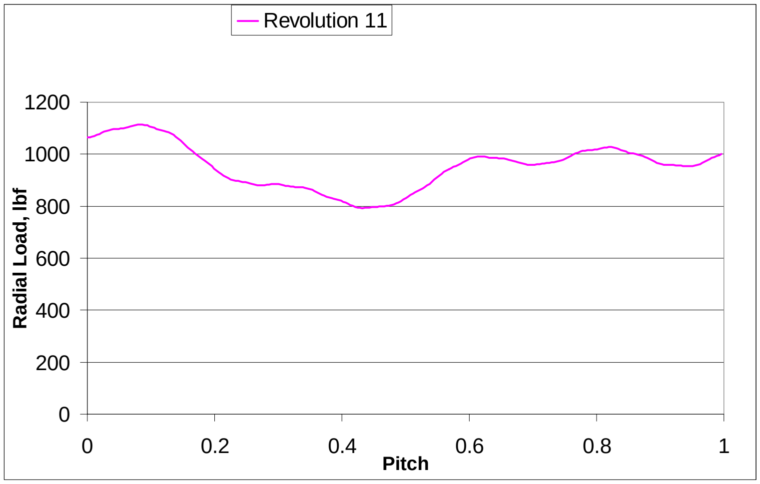


Figure 20. Radial load on impeller at the 11th revolution.

A transient CFD study was also performed at a compressor overload operating point using a transient sliding mesh for the entire 360-degree compressor flowpath. The transient runs modeled an operating conditions equivalent to 134% of design flow. The transient solution for this flow condition suggests that the efficiency will be higher and that the stage will have better range than predicted by 1-D analysis at the point analyzed. This analysis also shows that the distribution of the flow leaving the inlet guide vanes (igv) to be well-behaved and the flow field to be symmetric about its centerline. Further, the exit plane for this region shows that the flow is uniformly distributed and predominately in the axial direction. In the discharge system, there is incidence on the volute tongue that introduces unsteady losses as the flow leaves the machine. These unsteady losses are commonplace in a scroll-type volute in a stage operating in overload.

Pressure Casing: 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 were performed on the case design to achieve a proper balance between mechanical and aerodynamic requirements. Many of these design iterations focused on the design of the nozzles that form the aerodynamic flowpath boundary. Intermediate evaluations were performed using various calculation methods including numerical analysis techniques while designing the case to check the structural design integrity. The goal of the design was to achieve results that met or surpassed the requirements of relevant industry standards. Final calculations and analysis were performed on the case to check the structural design in accordance with industry standards. Table 7 shows some expected design operating parameters for the unit.

Table 8
Expected Design Operating Parameters

Service	Natural Gas	----
Maximum Allowable Working Pressure	1500	psig
Maximum Operating Pressure	1200	psig
Inlet Pressure Range (typical)	500 to 800	psig
Discharge Pressure Range (typical)	700 to 1200	psig
Inlet temperature		
Typical	60 to 80	°F
Range	35 to 95	°F
Speed		
MCOS	12000	RPM
100% Speed	11429	RPM
Minimum Speed	8000	RPM

Structural analysis was performed on mechanical compressor components such as the end wall, head, and necessary retaining devices (see Figure 21). Calculations and analyses were performed on the case to check the structural design. These analyses consist of hand calculations and numerical calculations such as finite element analysis. Results of the analysis were reviewed in conjunction with relevant industry standards and requirements. Design criteria are established by the application of industry codes and standards related to the design of pressure vessels and compressor. Relevant codes and standards referenced during the design of the IEMDC compressor section case include API 617 and ASME BPVC Section VIII, Division 1 and 2. The evaluation of the design using the results of the analysis indicates that the design is acceptable for the intended service. Figures 22, and 23 show detailed results from the analyses performed on the case and endwall. Figure 22 shows radial displacement of the case as a result of the design pressure loads. Figure 23 shows the stress on the endwall at the design pressure rating.

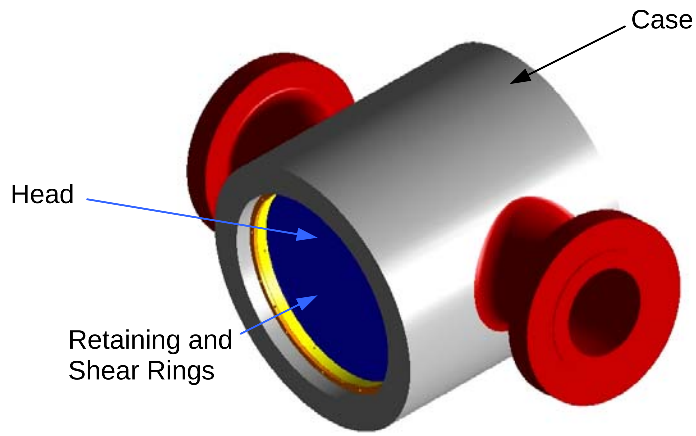


Figure 21. Case and Head Assembly

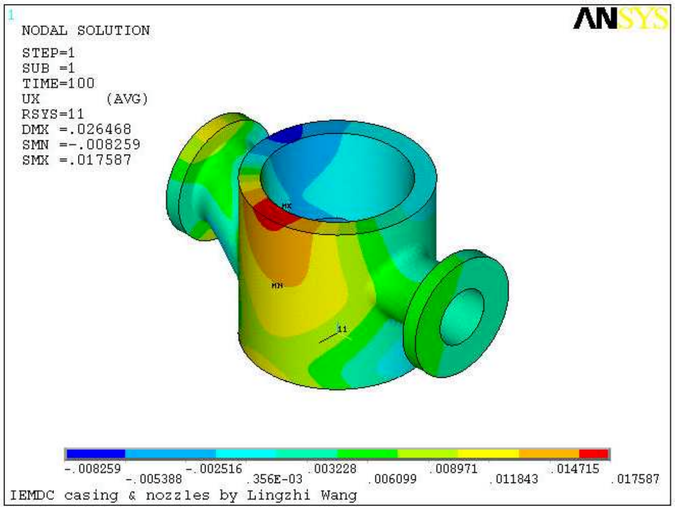


Figure 22. Case Displacement

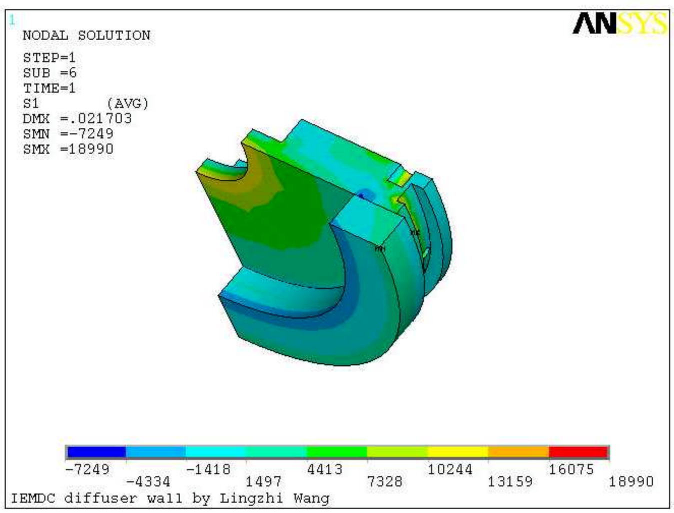


Figure 23. Endwall Stress

Finite element analysis was also used to design the pressure containing case interface geometry between the compressor and motor. This was an area designated as a technical challenge when the switch was made to the alternate configuration.

Determining and performing the appropriate level of analysis and evaluation was considered paramount to the integrity of the design. Design considerations made during the development of the interface geometry included items such as leakage, deflection (rigidity), and clamping force. The evaluation performed on the interface concluded that the design is suitably designed for the expected operating requirements. Figure 24 shows the mesh used in the finite element analysis of the interface and figure 25 shows some detailed results of the analysis and evaluation.

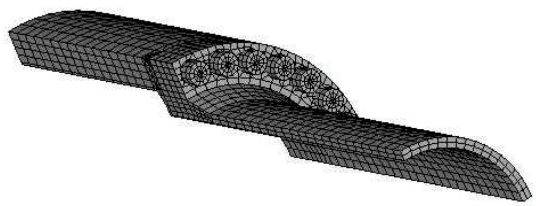


Figure 24. Mesh: Case Interface Geometry

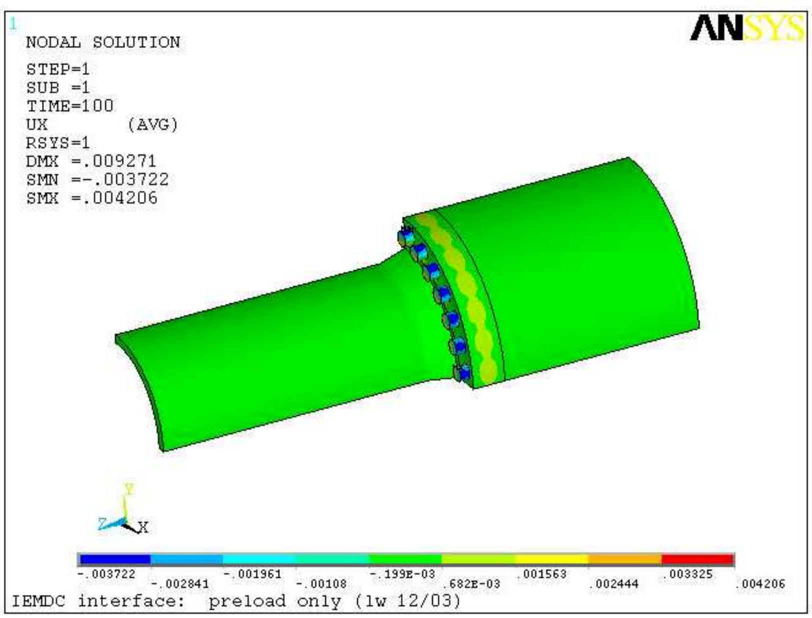


Figure 25. Interface Radial Displacement

Development of Solid Models: Solid models and solid model assemblies were developed during the mechanical layout of the unit. These models facilitate the design effort and are especially helpful as part of the design for manufacturing process. This process helps greatly in visualizing and reviewing unit assembly procedures during manufacture. This also ensures that appropriate manufacturing considerations are made during the design phase. Examples of benefits that can be derived from solid model assemblies are the tooling required for removing the bundle and internal parts from the unit. Solid models of components also assist in obtaining cost estimates from vendors. Parasolids can be transmitted to vendors that more accurately portray the component than two-dimensional drawings resulting in a more accurate cost estimate. Compressor case parasolids and volute parasolids were transmitted to vendors to obtain cost estimates of these components.

Figure 26 shows the final compressor configuration developed by the aforementioned efforts. It is the culmination of the work performed on the aerodynamic flowpath and the structural design of the compressor. In addition, the final compressor geometry reflects the results from the physical interface and interactions with the electric motor required for complete integration of the unit. This compressor section when coupled with the final motor design was used to create the outline drawing provided as a deliverable of the award.

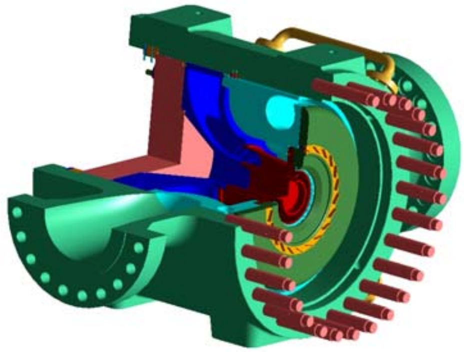


Figure 26. Compressor Section

Electric Motor Design

The design of the 10 MW 12,000 rpm variable speed induction motor, with an integrated overhung centrifugal compressor from the motor shaft, imposed several design challenges. The design process was iterative because of the need to develop an optimum, efficient, and robust design while resolving conflicting design requirements of competing factors. Listed below are some of the design challenges, which were resolved successfully during the execution of this project:

- 2-pole versus 4-pole design
- Impact of variable frequency drive on motor design
- Solid rotor versus laminated rotor design
- Design of connection rings and retaining rings
- Evaluation of motor and magnetic bearings losses at high speed
- Small foot print versus long foot print
- Sizing of magnetic radial bearings to accommodate system loading including compressor induced aerodynamic loads

- Reconciling rotor-bearing design with the API standards 617 and 541 rotordynamics design requirements
- Sizing of magnetic thrust bearing to support system thrust loads
- Stator frame vibration analysis
- Motor and magnetic bearing cooling with high pressure compressor discharge gas at different operating points of the compressor
- Impact of natural gas constituents on motor insulation life
- Design of pressure vessel and motor-compressor interface
- Design of explosion proof power junction box

The following sections provide a brief discussion of each of the items listed above.

Two Pole Design: Even though the 4-pole design is more efficient as compared to a 2-pole motor design, the motor design was selected as a 2-pole squirrel cage induction type because of commercial unavailability of a 400 Hz variable frequency power drive, which would have been required for a 4-pole 12,000-rpm motor. Consequently, a design specification for a 200 Hz VFD power drive for the motor, was developed in consultation with potential VFD suppliers. The 200 Hz VFD drive design specification was submitted to NETL as a final deliverable.

Solid Rotor: The solid rotor design was selected over a laminated rotor because it allowed the motor design to be more compact, made from a single piece forging, and designed with a smaller footprint. The design of the optimum rotor slots to insert the rotor bars was a specific challenge for this project. Therefore the rotor slot geometry was configured via FEA analysis to satisfy stress requirements, to fall within the growth constraints resulting from thermal effects, and to be capable of retaining the rotor bars under all loading conditions expected during the operation cycle. Additional factors affecting the decision to design a solid rotor include considerable EMD manufacturing experience with similar size solid rotors for US Navy and commercial products, and the rotor capability to withstand higher stresses at rotor surface speeds approaching 200 m/s. The overall rotor design was finalized via FEA analysis. Figure 27 shows the solid rotor configuration.

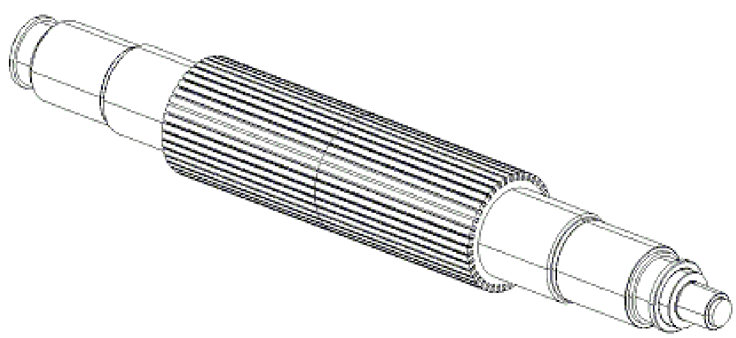


Figure 27: Motor Rotor

The design of the connection rings required selection of a copper alloy with high yield strength and good brazing characteristics. The stress requirements were established via rigorous FEA stress analysis under combined operating temperature and centrifugal loading at 12,000 rpm. The temperatures used in the analysis were obtained from the thermal analysis of the motor. The additional design constraint for the connection ring was to select a low cost alloy to keep the motor cost within budget. Therefore, a variety of material samples were procured, heat treated, and test evaluated under simulated brazing processes to identify the most appropriate material for this application. The material selected for the retaining rings to restrain the growth of connection rings was of very high strength. The retaining ring design was also evaluated via the FEA analysis. Further design constraints imposed included small overall size so as to keep electrical losses low, provide sufficient effective surface area for heat dissipation, and to allow efficient brazing of the connection rings to the rotor bars. The connection ring and the retaining ring combination were also analyzed for fatigue life to verify the motor start-stop life for 5,000 cycles. Both items are considered critical components of the motor.

Rotor Shaft Radii Evaluation: A detailed analysis was performed to evaluate the structural integrity of the rotor step radii at critical locations along the rotor shaft length such as the main rotor step, bearing collar, and bearing sleeve locations shown depicted on Figure 27. The primary importance of this analysis was to verify that the motor rotor meets the crack growth and fatigue life requirements of the API 541, which is a minimum of 5,000 start-stop cycles. The analysis considered loading conditions at zero speed at room temperature, 12,000 rpm at room temperature, 12,000 rpm at operating temperature, and zero rpm at operating temperature. The analysis results verified the robustness of the design by showing a range of safety margins that varied from 4.3 to 16.4 for the different conditions.

Characterization of Motor Losses: The accurate characterization of the motor electrical, and friction and windage losses at high operating speeds was another design challenge. Therefore an up-to-date literature search was performed to successfully identify the latest techniques for calculating friction and windage losses. These losses constitute a significant portion of the total motor losses that affect the motor cooling mass flow requirements, motor efficiency, and thermal characteristics, all of which impact motor insulation life. The resulting analyses performed were used to calculate the motor and magnetic bearing losses at various operating points of the IEMDC system. The motor-compressor operating points were developed by Dresser-Rand. The calculation of motor losses was a very significant effort because it included the evaluation of multiple data points for many operating conditions of the IEMDC system. Figure 28 shows plots of the calculated friction and windage losses at 12,000 rpm as a function of the compressor discharge gas temperature, at various inlet pressures. As shown on Figure 28, for a given operating speed of the motor, the friction and windage losses are dependent on the motor cooling gas pressure and temperature because of their effect on the gas density.

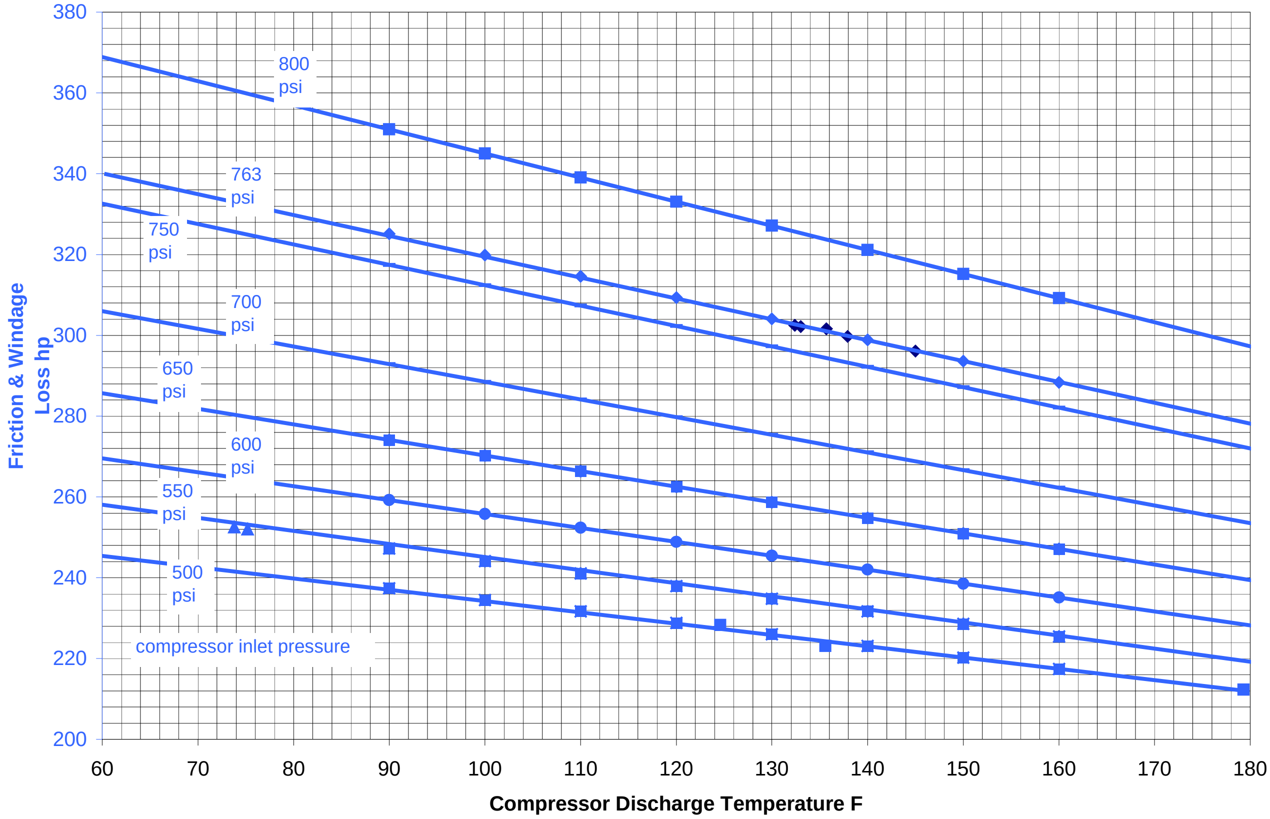


Figure 28. Calculated Motor Friction and Windage Losses at 12,000 rpm

Not a Typical Motor Design: Since this wasn't a typical stand alone motor design because of the overhung compressor stage from the motor shaft, another critical design challenge addressed was to reconcile the competing factors to keep the overall motor footprint as small as possible consistent with achieving a robust motor design. Specifically, this pertains to the rotordynamic analysis of the motor rotor that is levitated by the magnetic bearings and included an overhung compressor stage. The issues addressed include optimizing the size of the magnetic bearings capable of having acceptable design load margins to support the rotor weight, unbalance loads, and the compressor imposed aerodynamic loads. The compressor imposed aerodynamic loads were of significant importance for this design because of their large magnitude during the initial design stage of the compressor configuration. Therefore, it required iterative design trade offs among the design of the compressor, magnetic bearings, and the motor rotor. Hence a systems design approach was applied to develop the feasible and acceptable design solution by the three design teams of Curtiss-Wright, Dresser-Rand, and KMB/S2M, the magnetic bearing supplier. Eventually this resulted in a design solution with reduced aerodynamic loads via CFD design approach of the compressor volute and an increased magnetic bearing size at the impeller end to accommodate the predicted load envelope, and a slight reduction in shaft length to keep the bearing span length optimum. The final optimized rotor-bearing design configuration is a sub-critical design with two critically damped rigid body modes below 12,000 rpm and has a first bending critical speed of the rotor that is sufficiently higher than the maximum motor operating speed of 12,000 RPM, consistent with API 617 critical speed separation margin requirements. Furthermore, the predicted rotor response and bearing loads due to imposed rotor unbalance also satisfy the API 617 response requirements. The analyses were performed using validated rotordynamic techniques and simulation of the magnetic bearing control system, as discussed below.

Rotordynamic Analysis and Results: The design operating speed range of the IEMDC motor-compressor is 8,000 to 12,000 rpm. Rotordynamics analysis was performed and completed by S2M and the predictions independently verified by EMD. Both analyses have confirmed that the predicted critical speeds of the motor-compressor system are in conformance with the requirements of API standards 541 and 617. The motor-compressor operates above the two rigid body modes, translation and conical, and below the first bending mode of the rotor. The rigid body modes, as shown in Table 9, are critically damped and therefore require no separation margin. Also the first bending mode at 254 Hz is 27% above the maximum continuous operating speed of 12,000 rpm. This value of separation margin from the bending mode satisfies the rotordynamics requirements of API standards. Figures 29, 30, and 31 show the predicted rotor vibration due to 4xAPI unbalance with the synchronous AVR filter system turned on. The vibration level, shown in these plots, is below 0.001" peak-to-peak throughout the operating speed range. Thus, the design satisfies the unbalance response requirements as specified by API standards.

Table9: IEMDC Motor-Compressor Calculated Critical Speeds

Mode	Frequency (Hz)	Damping Ratio	Log Decrement
Translation	68	0.393	2.67
Conical	111	0.482	3.45
First Bending	254	0.051	0.321
Second Bending	447	0.011	0.07

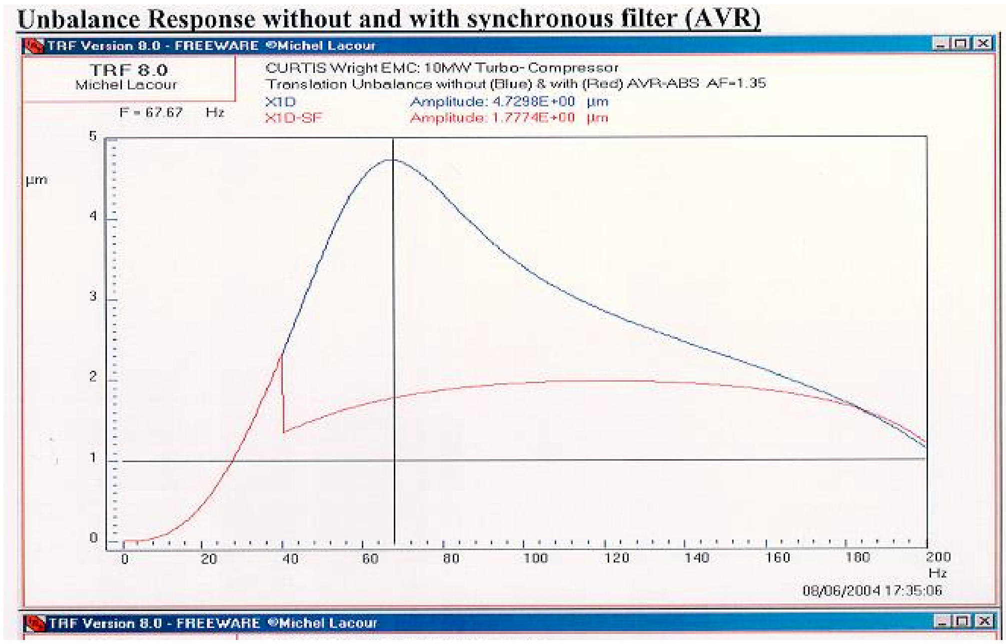


Figure 29

Predicted Unbalance Response at 111 Hz with and without AVR Filter

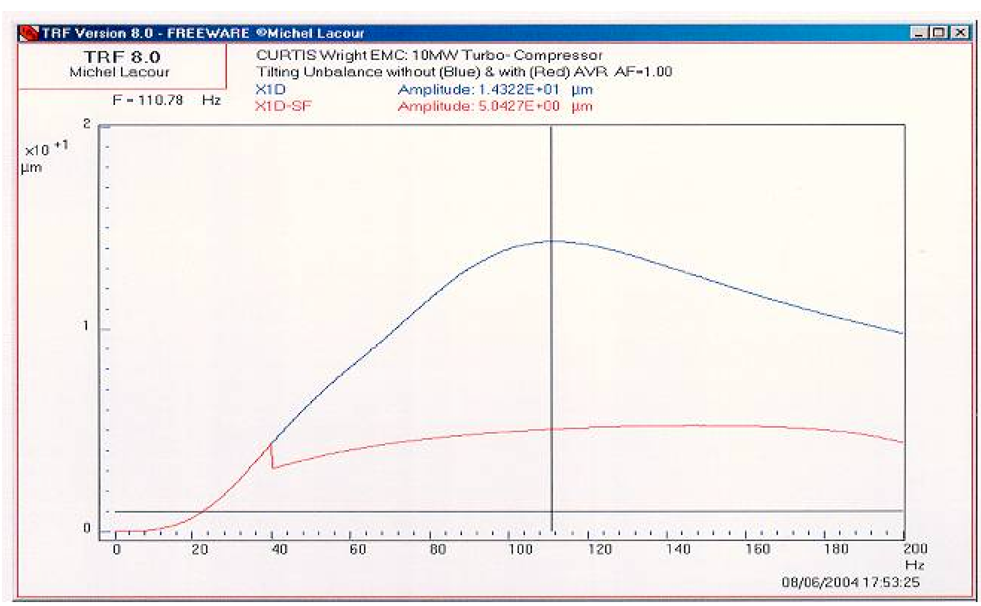


Figure 30

Predicted Unbalance Response at 254Hz with and without AVR Filter

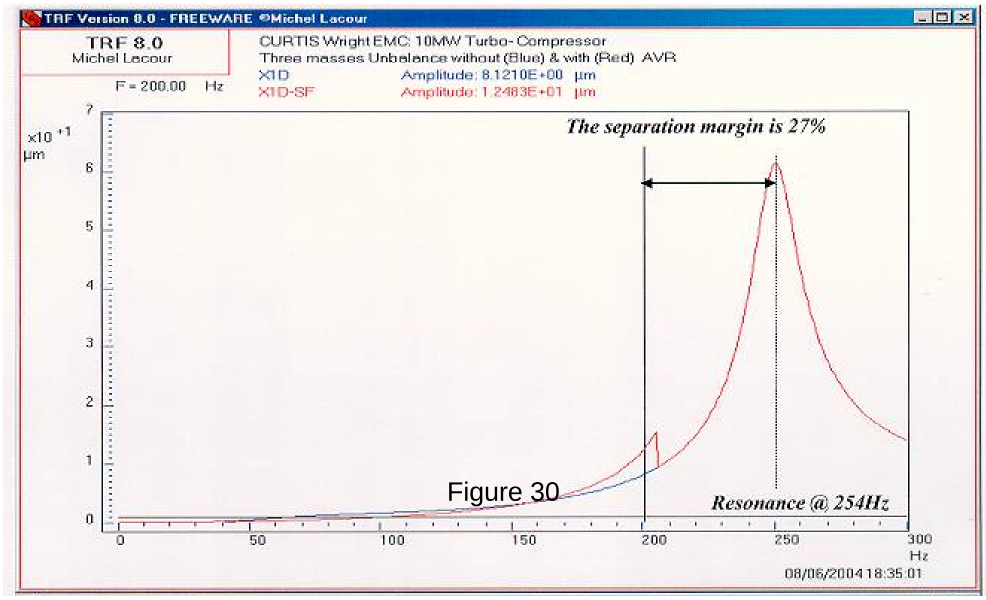


Figure 30

Figure 31

Rotor Thrust Load Balance: The rotor thrust load balancing consisted of loads generated due to pressure differentials in the motor housing by the motor cooling gas flow as well as the loads imposed by the pressure loading from the impeller flow. This required a system design approach to estimate the residual design loads under various operating conditions and then to size and optimize the thrust magnetic bearing with at least 100% design load margin to counteract this load. The selected magnetic thrust bearing has greater than 100% margin. Thrust balancing issues are also discussed in more detail in the compressor section of this report under IEMDC integration related development.

Stator Half-Coil design: The motor stator design also posed special design problems because of space limitations due to the compact design. Of specific concern was the design of full coil versus half coil. The full coil design was determined to be unacceptable. Therefore, the motor design incorporated a half coil design.

The design objective of this task was to verify that the insulated stator coil geometry would fit within the specified space envelope. The services of an experienced stator designer and coil winder were employed in the efforts to complete this task. This vendor has provided similar services to EMD on a variety of stator windings. Additionally, the stator coil insulation materials selected are based on a system that has been extensively tested and implemented on other EMD manufactured machines. Based on EMD's experience with the insulating tapes, the design incorporates the actual build dimensions rather than the vendor specified dimensions of the tapes. This allows the copper area to be maximized. However, in this particular application, the available space envelope (rotor length and diameter) for the electrical design was limited due to the placement of the rotor critical speeds to meet API standard design requirements. Therefore, it was necessary to optimize the allotted space for maximum conductor area in the coils.

Due to the small bore and the coil pitch, it was determined that a full coil design would not be feasible. Therefore, a half coil configuration utilizing different top and bottom half coil designs has been incorporated into the design in an effort to minimize the stator slot eddy loss in the stator coils. This design is shown in Figure 32. The top half coil stranding has been reduced in size at the expense of reduced copper area (greater number of strands, which have the same insulation thickness equals more area required for insulation). Because there is a fall off of the flux density deeper in the slot, the I^2R losses become more pronounced. Therefore, larger stranding is used for increased copper cross section in the bottom coil. Note that the eddy losses are proportional to the square of the strand height.

The IEMDC variable speed motor will be soft started unlike a line started motor with high inrush current. Therefore, the bracing of the end turns will experience reduced forces. The bracing scheme designed incorporates a support mechanism for the neutral ring. The phase coils will be made with extended start leads to facilitate connection to the terminal glands. The drafted layout supports the design envelope allowance for the stator end turns, phase connections, and neutral ring.

Verification of the layout with a stator mock-up is necessary, prior to production of the first unit. However, during the design phase of this task, it wasn’t feasible to construct the mockup due to lack of funding. Therefore, it is recommended that the mockup should be designed and built during the prototype-manufacturing phase of the program.

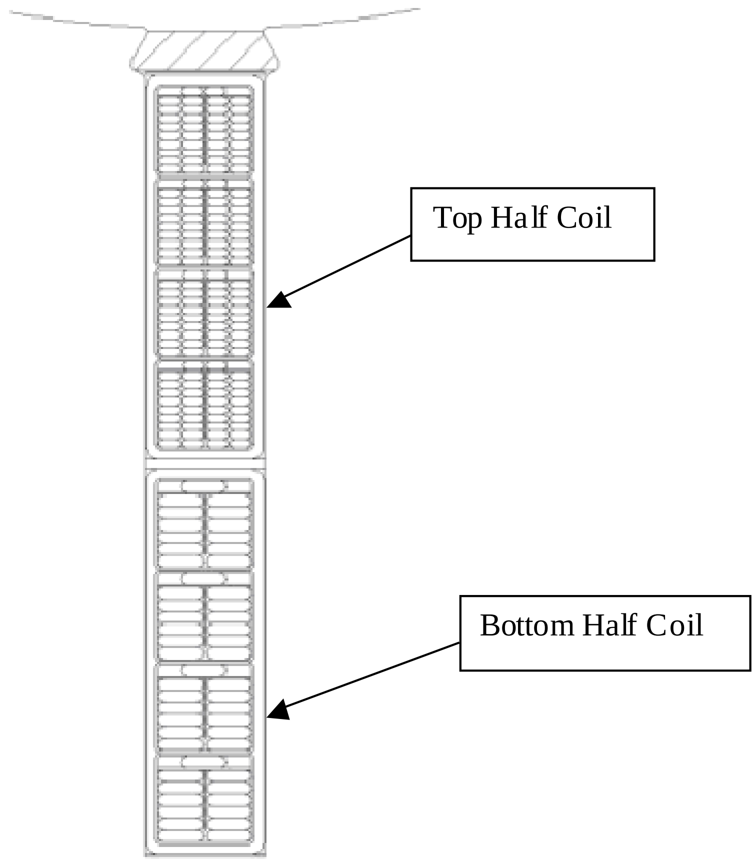


Figure 32: Stator Slot Cross-Section

Motor Frame Vibration Analysis

A modal frequency analysis model of the motor stator and frame was developed to investigate that the calculated frequencies satisfy the vibration design criterion of the API 541 standard. This evaluation considered all natural frequencies that could be excited by the motor's main magnetic field at 2E, and imbalance loads at multiple of rotor speed (1R, 2R, and 3R). The results show compliance for all significant modes. The stator frame vibration frequency analysis was carried out via FEA modeling and was consistent with the design requirements of API 541.

Motor and Magnetic Bearing Cooling Circuit: The motor and magnetic bearing cooling is accomplished by using the high-pressure compressor discharge gas. The cooling gas is introduced near the end turns at two locations and vented out at the middle and recycled back to the compressor inlet with appropriate controls. The challenges posed to cool the motor and magnetic bearings that were immersed in the high pressure gas flow, were in the design of an optimum cooling circuit and flow passages to achieve desired pressure drops throughout the machine for optimum amount of gas mass flow required under various compressor operating conditions. Some of the off-design points included extreme operating environment with respect to gas temperature and pressure. Ventilation system challenges included intricate design of vent plate and circulation of cooling flow around the motor and magnetic bearing windings and rotor surface so as to maintain acceptable motor and magnetic bearing maximum hot spot temperatures within the limits of the selected insulation class, and the evaluation of the impact of gas constituents on motor insulation life. Therefore, a special study was performed to investigate the effect of gas constituents. It was determined that under normal operating conditions and using gas filtered to 3-5 microns, to remove particulates, prior to entering the motor cavity, the motor insulation would last for 20 years. Further rotor and other critical component fatigue life calculations predicted that motor design is good for at least 5,000 start stop cycles. This meets the design objective expressed by several end users of the IEMDC system.

Figure 33 shows a cross-section of the stator and Figure 3 shows a schematic diagram of the cooling flows for the motor-compressor unit.

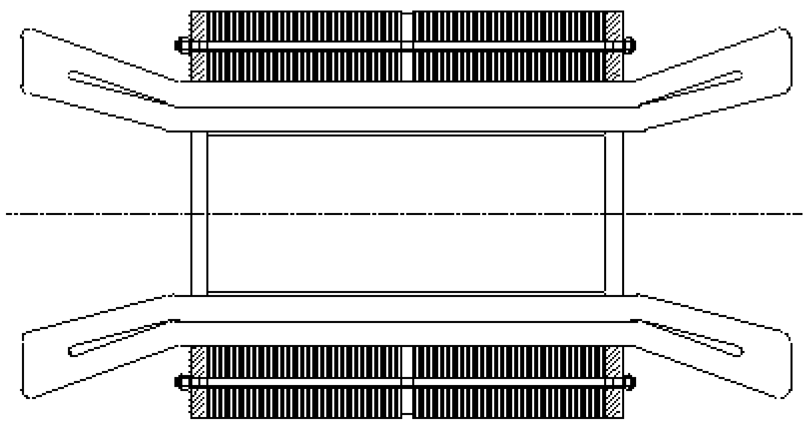


Figure 33: Motor Stator with Insulated Coil

Motor-Compressor Flange and Pressure Vessel Design: Figure 9 depicts a longitudinal section of the motor showing the location of the interface and the pressure housings. The design of the motor-compressor flange and pressure boundary was very critical to establish integrity of the system design. Therefore, very comprehensive and detailed FEA analyses were performed to validate the designs of the pressure vessel and motor-compressor interface flange. The conclusions of FEA analysis were that all applicable API 617 and ASME Section VIII limits were met for the assumed loading and maximum flange separation at o-rings was acceptable. For reference, the key design requirements are listed below:

Design requirements:

- Design governed by API 617, 7^{th} edition
- ASME Section VIII limits apply
- Design pressure of 1,500 psig at 350°F
- Hydro pressure of 2,250 psig at 70°F
- Maximum operating pressure of 900 psi
- Maximum operating temperature of 266°F
- Thermal stresses negligible
- 5,000 full range pressure cycles based on number of start-stop cycles specified in API 541, 3^{rd} edition
- 0.125 inch corrosion allowance
- Casing nozzle load reactions are negligible compared to blow off pressure force

The results of the FEA stress analysis are tabulated in Tables 10 to 13, showing the design margins of the calculated stresses as compared to the allowable stresses. As stated earlier, the components of the interface such as blind flange, reverse flange, shell, main flange, pipes, bolts, etc. all meet the loading requirements that were assumed for this analysis.

Table 10: Stress Summary for Non-bolting Components

Component	Minimum Wall Thickness (in)		Design Conditions (ksi)				Hydrotest Conditions (ksi)				Fatigue Usage	
			Membrane		Memb+Bend		Membrane		Memb+Bend			
	Req'd	Actual	Calc	Allow	Calc	Allow	Calc	Allow	Calc	Allow	Calc	Allow
Blind Flange	6.46	6.75	1.58	21.0	16.1	31.5	1.83	32.4	22.9	48.6	0.17	1.0
Reverse Flange	-	-	10.3	21.0	12.1	31.5	13.6	32.4	21.2	48.6	0.25	1.0
Shell	1.87	2.0	17.6	21.0	18.7	31.5	26.6	32.4	28.4	48.6	0.17	1.0
Main Flange	-	-	16.9	21.0	17.5	31.5	25.9	32.4	26.7	48.6	0.26	1.0
1.0 Dia. Sch. 80 304 SS Pipe	0.070	0.157	-	-	-	-	-	-	-	-	-	-
1.5 Dia. Sch. 80 304 SS Pipe	0.110	0.175	-	-	-	-	-	-	-	-	-	-

Table 11: Stress Summary for Bolts

Component	Design Cond. (ksi)		Hydrotest Conditions (ksi)				Allowable Number of Disassembly Cycles
	Average Stress		Membrane		Memb+Bend		
	Calc	Allow	Calc	Allow	Calc	Allow	
1.75 Dia. Studs	24.8	25.0	38.4	63.6	45.4	95.4	500
3.0 Dia. Studs	16.8	23.0	54.2	57.6	60.3	86.4	600

Table 12: Special Stress Limits

Component	Bearing (ksi)		Shear (ksi)	
	Calc.	Allow.	Calc.	Allow.
Reverse Flange	9.3	32.2	8.7	12.9
Main Flange	31.9	32.2	12.2	12.9

Table 13: Area Reinforcement Summary for Penetrations

Penetration	Required Area (in^2)	Area Provided (in^2)
3.0 dia. Gas Inlet	5.739	10.379
4.0 dia. Gas Outlet	7.363	14.094
1.5 dia.	none	-
1.0 dia. Drain Pipe	none	-

Motor Power Junction Box Design

The motor power junction box design requirements included definition of the number of junction boxes, development of the wiring diagram package, and development of the junction box mechanical design.

The IEMDC motor-compressor will be located in a Class 1, Division 2, Group D environment as defined in Article 500 of NFPA 70 NEC. The magnetic bearing power, instrumentation, and motor stator temperature detector connections are made with sealed connectors that mount directly on the motor casing. These connectors thread onto hermetically sealed receptacles that are threaded into the motor casing.

There is a single junction box where the motor power feed connections are located. Due to the motor operating at high voltage and current levels, this box has been designed as an explosion-proof enclosure. The design analysis was performed for an assumed hydrostatic pressure of 315 psi, which is three times the maximum expected internal pressure during the explosion tests. The analysis was performed to satisfy the intent of section VIII of the ASME Code, 2001 edition.

The power feed cables enter at the base of the box, as shown in Figure 34, where they are sealed with cable glands that are UL listed for use in Class 1, Division 1, Group D environment. The junction box is equipped with a methane gas monitoring system. The methane gas leak may be caused in the event of a hermetic seal failure of a power feed through gland.

A detailed design analysis of an explosion proof main power junction box was carried out. The junction box will be mounted on the IEMDC motor outer housing, as shown in Figure 34.

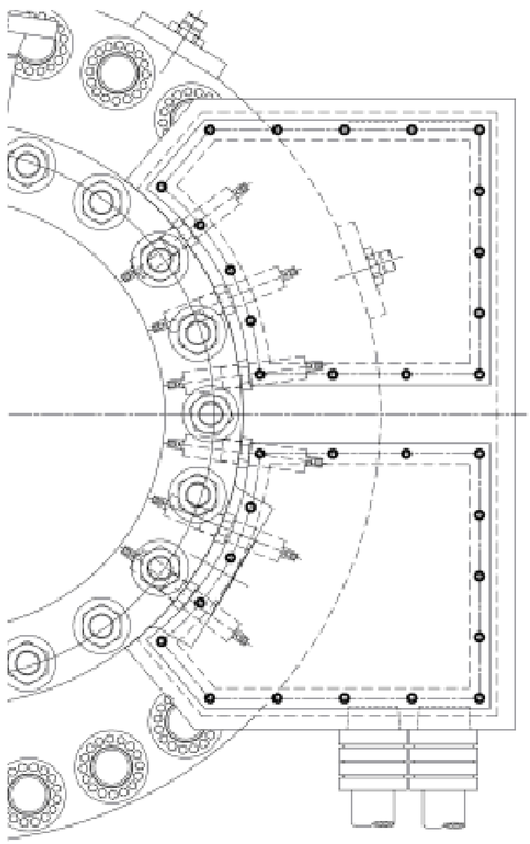


Figure 34: Power Junction Box Shown Mounted on the Motor Housing

The conclusions of the analysis are:

- The welds are acceptable for full penetration welds for the box, and 55% efficiency attachments to the motor shell.

- A fatigue evaluation of the attachment weld shows 15,000 full range pressure cycles for the motor shell, which exceeds the requirement of 5,000 cycles.
- The conduit penetrations satisfy the replacement requirements of the code.
- The addition of the conduit box has a negligible effect on the structural integrity of the motor shell.

Considerable effort was also spent in generating the motor layout drawings, which have been submitted to the DOE NETL. Figure 35 shows a longitudinal section of the final motor design configuration.

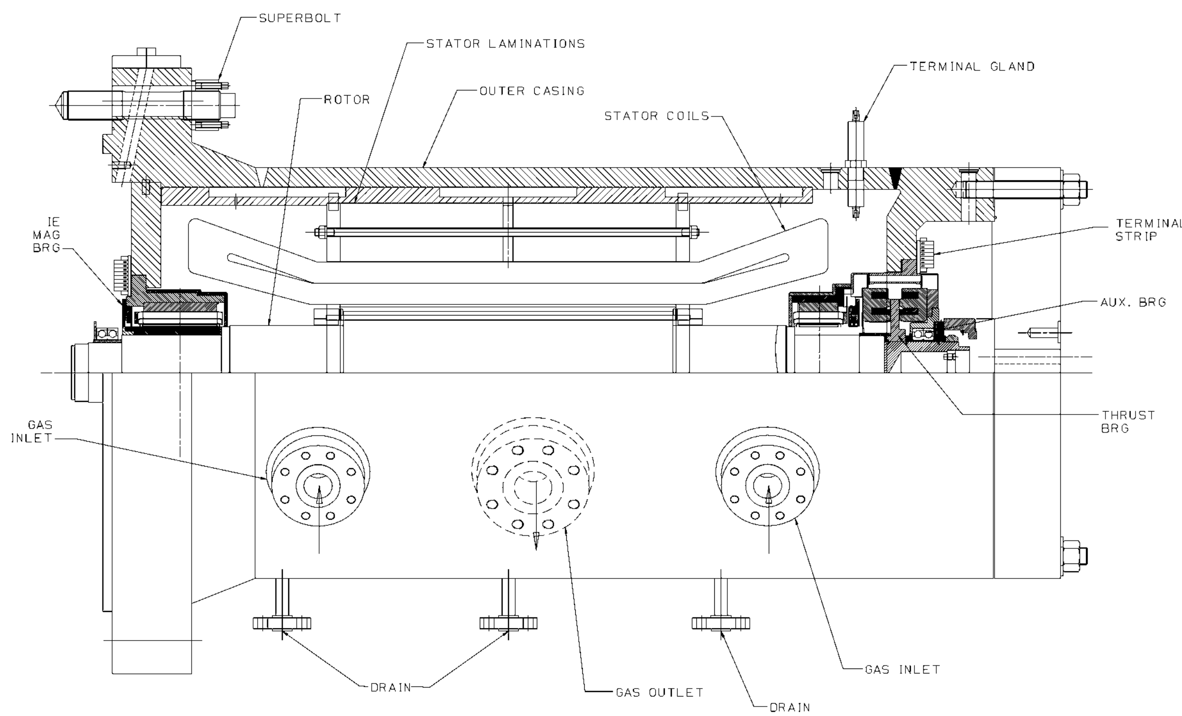


Figure 35: Motor Longitudinal Section

Commercialization Control

Activities were undertaken to ensure the successful commercialization of the IEMDC product. These activities included the review and consideration of the following,

- Product configuration,
- Component method of manufacture,
- Maintenance and operation requirements,
- Installation requirements,
- Gas cleanliness and composition,
- Operating capability,
- Market study.

Product configuration review: The development process included extensive design iterations of the original IEMDC configuration to evaluate the impact of design and performance trade-offs. Those evaluations led to the adoption of a modified configuration that achieved the proposed system benefits. The new design offers a configuration that is easier to maintain and is overall more commercially attractive from the design aspects of installation, operation, and maintenance. Interaction and feedback solicited from target clients regarding the modified configuration is both positive and supportive as it addresses industry issues regarding accessibility, maintainability, preferred operating practice, and increased reliability. In addition to maintaining the original benefits proposed by the project, the resulting system delivers the following advantages:

- Modular construction provides improved accessibility and maintainability of critical change out parts such as the impeller and bearings without the necessity to break critical system seals.
- Improved aerodynamic flow path for optimal and efficient compressor design.
- Enhanced system thrust balancing design.
- Improved motor cooling circuit design.
- Improved gas filtering arrangement over the original concept.
- Reduced system weight from the original configuration.

Gas cleanliness, dryness, and composition: A study was performed to review the range of composition and constituents expected for the intended service of the IEMDC. This study included reviewing prior experience of compressors in similar applications. In addition, various gas company tariffs, a gas sample from a customer was obtained, and various publications were reviewed. It was concluded that pipeline quality natural gas should have no deleterious effects on the design or operation of the motor as long as the gas is properly filtered before entering the motor.

Operating capabilities and requirements: Variable speed operation of the IEMDC permits maximum operating flexibility by allowing the unit to operate at varying flow conditions. Maximum operating flexibility has been identified as a key requirement of compressors in the intended application. Using a variable speed electric drive is a reliable and efficient means to achieve maximum operating flexibility.

Publicly, the IEMDC concept was presented at the 2000 International Pipeline Conference in Calgary, the 2000 Gas Machinery Conference (GMC) in Colorado Springs, the 2001 GMC in Austin, Texas, the 2002 GMC in Nashville, Tennessee, and the 2003 GMC in Salt Lake City. The IEMDC was also presented at several Gas-Electric Alliance Partnership Conference meetings, which are held annually in Houston. Each of these conferences was attended by a variety of potential clients and users. Considerable interest in the IEMDC was expressed from potential users at each of these conferences.

Technology Assessment

The gas transmission industry currently uses both gas fueled and electrically powered equipment as drivers for compression equipment. Although reciprocating engines and gas turbine drives comprise a majority of all installed gas compression and transmission equipment, in recent years the use of electric motor driven compression has increased significantly in gas transmission service. This has been facilitated, to a large extent, by the many technological advances in the variable frequency drive (VFD) converters as well as due to the superior economic advantages, operating flexibility and reliability of electric drives.

There is currently an aging fleet of 20 to 50 year old gas driven compressors on pipelines. This equipment typically has high associated maintenance and operating costs. Maintaining this aging fleet of compression equipment can be a challenge due to on site gas leakage, emissions that cause air pollution, availability of very costly spare parts, system monitoring capability, and noise. Older installations using gas fired compression equipment lack operating flexibility making it difficult to meet varying system flow conditions. This is a concern considering that in order to increase the capacity of the existing pipeline infrastructure to facilitate the projected growth in the gas market, the reliability, capacity, and efficiency of these installations needs to be maintained or enhanced.

Some of the issues with the currently installed compression base, and considerations that are required when selecting gas or electric powered compression equipment are as follows:

Emissions: Seal leakage that results in emissions is a concern for most currently installed compression equipment. A stand-alone centrifugal compressors in a conventional drive system requires shaft sealing. Shaft seals and associated seal systems increase expense, maintenance requirements, and introduce a reliability concern. Centrifugal compressors with shaft seals result in gas emissions as a result of seal leakage or when gas is vented and purged during start-up and shutdown. Gas leakage in reciprocating compressors is also a concern and a source of emissions. One of the most significant sources of emissions at natural gas compressor stations is the methane gas leaking from the rod packings of reciprocating compressors.

Reliability: Mechanical reliability is another important consideration when selecting a compressor driver. Reciprocating engines and gas turbines are mechanically more complex than electric motors. Frequent starts and stops can have a significant impact on gas fueled equipment integrity, reliability, and operational and maintenance costs resulting in decreased availability of the equipment. Electric motors have fewer moving parts resulting in an inherently more reliable drive system than a comparable mechanical drive system.

Moreover, variable speed electric motor drives, with proven reliability, typically offer significantly higher operating efficiency and flexibility.

Power Source: A distinct advantage of gas-fueled prime movers over electric drives is that they are better suited for gas pumping station sites that are located away from the electric power transmission grid. Gas fueled equipment receives its fuel directly from the pipeline so there is no need to acquire fuel from an alternate source. In contrast, electric motor driven units do require electric power and if it is not readily available there is a time and cost associated with extending transmission lines to the station site. In spite of this, the electric motor drives are often a better choice due to the inherently lower costs of owning and operating the units.

Operational: A major disadvantage of gas-fueled equipment is that the fuel supply is directly received from the pipeline thereby effectively reducing the capacity of the pipeline. Furthermore, the environmental impact of gas combustion can create emissions that result in increased installation costs as well as increased time for the construction and installation of the required abatement equipment. Permitting and air emissions requirements often result in equipment operating constraints. Additionally, new and existing installations are continually affected by changing emissions regulations and requirements. Capital investments in equipment are often necessary to achieve environmental compliance to changing regulations. Conversely, electric power has no site emissions and is therefore more environmentally friendly than gas fueled equipment. It is a far better solution, particularly for installations located in environmentally sensitive areas. This includes noise since electric power equipment typically has lower noise emission levels than gas fueled drives.

The IEMDC system proposes to provide significant benefits to the natural gas industry. It offers a unique configuration and features for a compressor with an integrated electric motor system of this size. The unit is comprised of a single stage overhung compressor with an impeller mounted directly on the shaft of an electric motor. The motor is cooled via an innovative cooling circuit utilizing the high-pressure discharge gas from the compressor resulting in a design efficiency of about 95%. A solid rotor construction is used with a very high magnetic shear rating that permits it to be a high power density machine without sacrificing motor efficiency. Additionally, the solid rotor can withstand higher centrifugal stresses as compared to a rotor with laminated construction. These features result in a robust motor design that allows the IEMDC to produce approximately 30% more power at a speed of 12,000 RPM than a laminated configuration, while maintaining a sub-critical rotor design. The motor design meets the design requirements of both API 617 and API 541 standards. This IEMDC system is an innovative package that can offer significant advantages when compared to various other currently known compressor systems. Some of the system offerings are as follows:

- It is a compact design offering reducing piping with a minimum of ancillary systems required for operation. This equates to easier installations as compared to some other systems.
- The direct drive electric motor is cooled via an innovative cooling circuit utilizing the high-pressure discharge gas from the compressor. This concept eliminates the need for motor cooling equipment such as heat exchangers and blowers.
- The rotor is a solid construction designed with a very high magnetic shear stress material that permits it to be a high power density machine. This design does not sacrifice motor efficiency and can withstand higher centrifugal stresses as compared to a rotor with a laminated construction.
- A modular bundle design provides the unit with the potential for fast maintenance turnaround times when changing aerodynamic compressor components. The modular bundle can be removed without disconnecting the process piping, instrumentation, or other auxiliary connections. This design lends itself to seasonal operating optimization of the unit by changing compressor internal flowpath components.
- There are many environmental benefits of the new IEMDC system. The unit has no emissions that are common with gas-fueled drivers. Methane emissions from the unit are also eliminated because it is hermetically sealed. In addition, the incorporation of magnetic bearings into the system eliminates issues associated with lubricating oil.

- Station compressor loop piping may be minimized as compared to some pipeline installations because of the single stage inline compressor configuration.
- Operational flexibility is provided in the form of variable speed for capacity control, and aerodynamic flexibility. The design readily permits changing aerodynamic configurations for design optimization that may be required for different seasonal operational conditions.
- The unit has the potential to be configured to operate at much reduced noise levels than other equipment resulting in very quiet operation. This is because the motor is encased by the pressure containing case, which should abate the noise and vibration of the system. In addition, noise attenuation devices can be installed within the compressor to significantly reduce compressor generated noise at the source.

Additional, some of the advantages that the IEMDC proposes to offer over current conventional technology are as follows:

- Capability to enhance the current infrastructure with greater reliability and operational flexibility,
- No emissions that are developed by the combustion of a gas fuel,
- Low impact in environmentally sensitive areas,
- Potential to create opportunities for using electric compression as replacement for older compressors, and for new site installations,
- Can be used as an enhancement of existing stations to increase operating flexibility.

The reader is referred to Appendix H for more information on the technology assessment of pipeline compression equipment.

Project Deliverables Summary

The following items are deliverables of the award. Each of the deliverables as noted below are provided in a separate appendix at the end of the report.

1. Outline drawing of the 10MW IEMDC unit Assembly

   See Appendix A of this report.

2. IEMDC system performance specification, including performance curves.

   See Appendix B of this report.

3. IEMDC system instrumentation and motor control schematic.

   See Appendix C of this report.

4. Layouts drawing of the 12,000 RPM, direct drive motor.

   See Appendix D of this report.

5. Rotordynamics analysis, including critical speed maps and rotor response analysis.

   See Appendix E of this report.

6. Performance specification for the required variable frequency drive.

   See Appendix F of this report.

7. Summary of the design objective achievement.

   This deliverable is outlined in the “Summary of Design Objective Achievements” section of this report.

8. Estimated cost and schedule to develop detailed manufacturing drawings, and manufacture and qualify the prototype 10 MW system.

   See Appendix G of this report.

9. Research Management Plan.

   Two research management plans were written as required during the course of the project. The first plan had outlined the initial performance for the first twelve-month period and a second plan was written and submitted to include an extension to the initial performance period. The award extension amended the scope of work and extended the performance period an additional six months. Both of these plans supported the project “Statement of Project Objectives” as outlined in the award.

10. Five page paper on the state-of the art technology for pipeline gas compression.

    A five page paper assessing the state of the pipeline gas compression equipment technology was written and submitted as a deliverable of the award. Contents of the paper included an in depth description of the state-of-the-art technology being developed for the IEMDC project as well as a detailed discussion of competing technologies, including the pros and cons of each topic.

11. A paper to the DOE NETL Annual Contractor’s Review Meeting.

    A comprehensive joint technical presentation about the IEMDC project development status was prepared and given by Dresser-Rand, Curtiss-Wright, and ASIRobicon at the 2003 GMC (Gas Machinery Conference).

Conclusion

The unique inline configuration of the IEMDC, as compared to existing compressor systems, makes this an innovative and attractive solution for the gas pipeline industry. It advances the concept of installing electrically powered, highly flexible, efficient, low maintenance compression by a combination of features that no other product currently offers as a complete package. It offers a compact design featuring a small footprint by the use of single stage compression utilizing an overhung impeller at the compressor end of the integrated electric motor shaft. Due to small footprint, the IEMDC offers competitive advantages as new compressor station equipment or for re-powering an existing station.

The IEMDC offers significant capabilities for a compressor with an integrated variable speed electric motor system of this size. The variable speed capability would provide operational flexibility to meet varying gas flow supply needs of the pipeline operators.

The motor is cooled via an innovative cooling circuit utilizing the high-pressure discharge gas from the compressor. This concept eliminates the need for auxiliary motor cooling equipment such as heat exchangers and blowers. The resulting motor design point efficiency is about 95%.

A solid rotor construction of the motor rotor with a very high magnetic shear stress design permits it to be a high power density machine without sacrificing motor efficiency. Additionally, the solid rotor and shafting can be manufactured from a single piece forging and can withstand higher centrifugal stresses as compared to a rotor with laminated construction. These features result in a very robust motor that allows the IEMDC to produce approximately 30% more power at a speed of 12,000 RPM than a comparable laminated configuration, while maintaining a sub-critical rotor design. The motor design meets the rotordynamic design requirements of both API 617 and API 541 standards.

An advanced compressor section design was achieved by fully utilizing Dresser-Rand's extensive design expertise and experience. The compressor utilizes a modular bundle design applied to a pipeline compressor. The modular bundle contains the head, inlet, volute and diffuser pieces of the compressor internals. This modular bundle assembly can be removed as one piece, which provides the advantage of quick turnaround in the field. In addition, the flowpath has been optimized via extensive CFD simulations and the aerodynamic efficiency is expected to be 86% or better at the design point.

One of the primary considerations of the IEMDC unit was to design an environmentally friendly system. This has been achieved by designing a hermetically sealed system levitated by magnetic bearings. A system that is hermetically sealed eliminates the emissions from shaft seals. Furthermore, emissions from combustion associated with gas-fuel driven compressors are eliminated. The result is a system that addresses the requirements for an installation that minimizes environmental and regulatory issues.

The incorporation of field-proven magnetic bearings and direct coupling of the compressor to a high-speed motor eliminates the need for lubricating oil. Since no lubrication oil is needed for this system, there is no potential for oil leaks and associated environmental issues such as clean up and disposal of costly lubricating oil. Additionally the design also eliminates the need for costly and cumbersome auxiliary equipment such as oil pumps, oil coolers, oil reservoir, oil reservoir heaters, oil filters, pressure switches and control valves, temperature switches and control valves, etc. Elimination of oil lubrication further enhances system reliability by eliminating the potential for shutdowns caused by the lubricating system and associated components.

Remote operation of compressor stations is also easier with a system that is totally electrically powered. This is another advantage of using magnetic bearings instead of oil film bearings. The IEMDC concept with the integrated motor and compressor set immersed in the process flow eliminates the needs for shaft end sealing. This coupled with the elimination of lubricating oil requirements and hazards, creates a system that is ideally suited to remote and/or unmanned installations.

The IEMDC is an inline single stage unit offering accessibility for ease of maintenance without the necessity of disconnecting the piping. Figure 36 shows the conceptual configuration of an installed IEMDC unit. If the unit were to require service, the compressor internals could quickly be accessed. Both the compressor and electric motor are contained in modular sections of the unit. Internal compressor components can be easily accessed through the compressor end head while the compressor case remains connected to the process piping. Easy internal compressor access permits rapid changing of aerodynamic components if required, since there is no main process piping or auxiliary connections to remove. This could be considered an important feature if compressor internal component changes are made to optimize seasonal operating conditions.

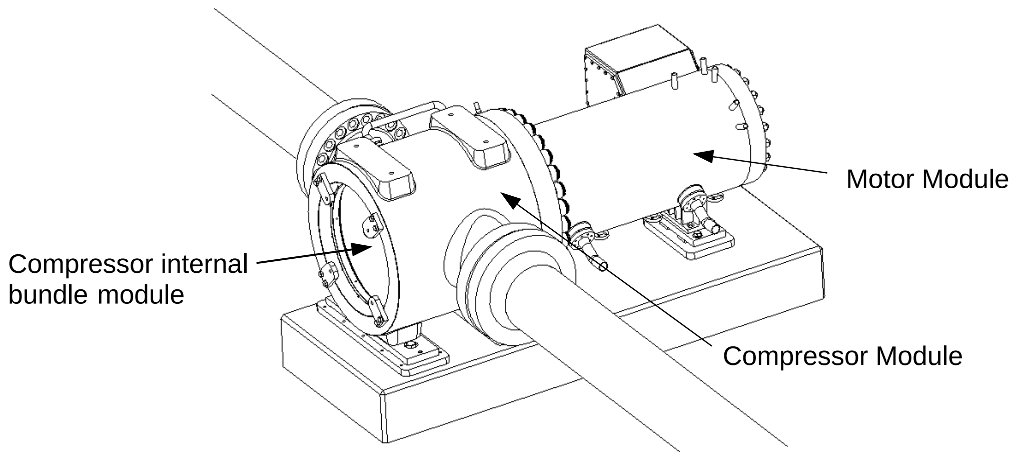


Figure 36. IEMDC Configuration – Maintenance Accessibility

A major installation benefit of the new system is that it is expected to result in less station piping which reduces the total system losses. When multiple elbows and piping loops are required by a system, there is a substantial increase in piping losses. Therefore, considerable benefit is obtained from minimizing the installation piping in terms of area, cost, and system losses. This is particularly important if this technology is going to be added to an existing installation or installed subsurface (i.e., underground). It is also important when considering a new installation to keep capital investment costs low.

Operational flexibility has also become an increasingly important issue in the gas transmission industry. This is from the perspective of economics and the response time required to meet changes in load demands. An integrated electric motor compression system can offer this needed flexibility. The full operating range of the IEMDC can be utilized, as there are no environmental implications when changing operating conditions. In addition, the IEMDC is efficient over a wide operating speed range, which allows significant capacity variation. This operating flexibility is, in part, a result of using a high-speed motor and variable frequency drive. It can quickly be started or undergo speed variations as required to respond to the load demand. Furthermore, the integrated IEMDC system can quickly be configured to meet seasonal aerodynamic component changes, if peak efficiency must be maintained under significantly varying seasonal load requirements.

Finally, in conclusion, as discussed in this report, the design efforts completed during the execution of the project have resulted in meeting the award objective as well as the key goals that were established at the outset. In addition, the project was completed on time and within budget. As required by the award objective, a basic design configuration of the IEMDC has been developed to the point where detailed manufacturing drawings could be started. Additionally, as outlined at the outset of the project, the IEMDC system will considerably reduce the cost of permitting to the pipeline gas transmission industry by minimizing on-site emissions. Furthermore, since the

IEMDC system is designed to be highly reliable with easy accessibility and low maintenance costs, it will result in overall reduced cost of installation and reduced total life cycle cost of operation. These potential benefits to the gas industry have been affirmed by the expressed interest of potential users of this new advanced compression system.

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

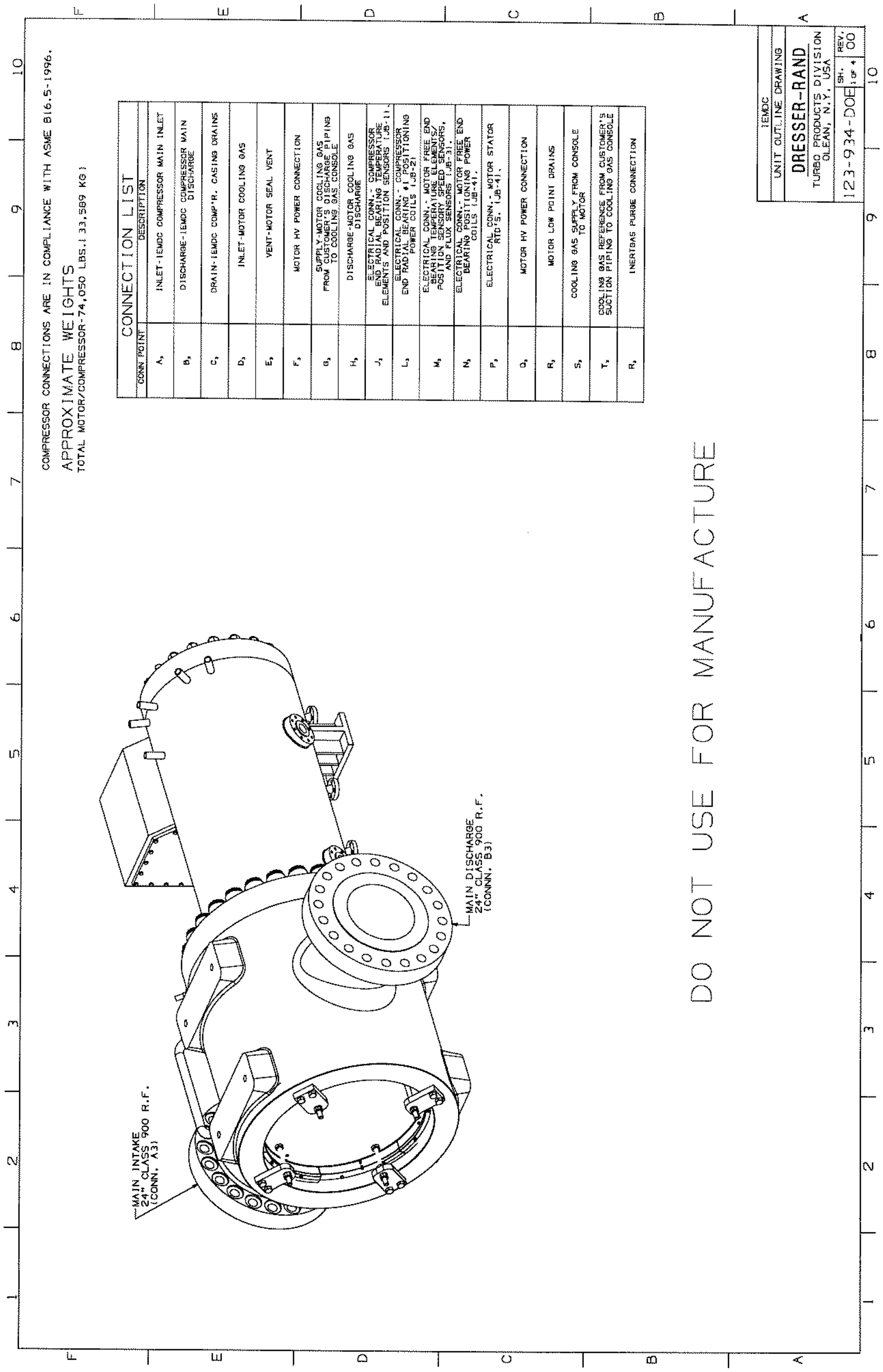
AC	Alternating Current
AF	Amplification Factor
API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
ASTM	American Society of Testing Materials
BPVC	Boiler and Pressure Vessel Code
BTU	British Thermal Unit
BWD	Backward
CCF	Hundred Cubic Feet
CFD	Computational Fluid Dynamics
c.g.	Center-of –Gravity
Cp	Specific Heat at Constant Pressure
CT	Current Transformer
DC	Direct Current
Deg.	Degree
DR	Dresser-Rand Company
DSP	Digital Signal Processing
EMD	Curtiss-Wright Electro-Mechanical Corporation (EMD)
ETA (η)	Efficiency
F	Fahrenheit
FAT	Factory Acceptance Test
FEA	Finite Element Analysis
FWD	Forward
HP	Horsepower
IE	Inlet end
IEEE	Institute of Electrical and Electronics Engineers
IEMDC	In-Line Electric Motor Driven Compressor
HVAC	Heating Ventilation and Air Conditioning
Hz	Hertz
in	Inches
IP	Polar Moment of Inertia
IT	Transverse Moment of Inertia
I/O	Input / Output
ISO	International Standards Organization
kV	Kilovolt

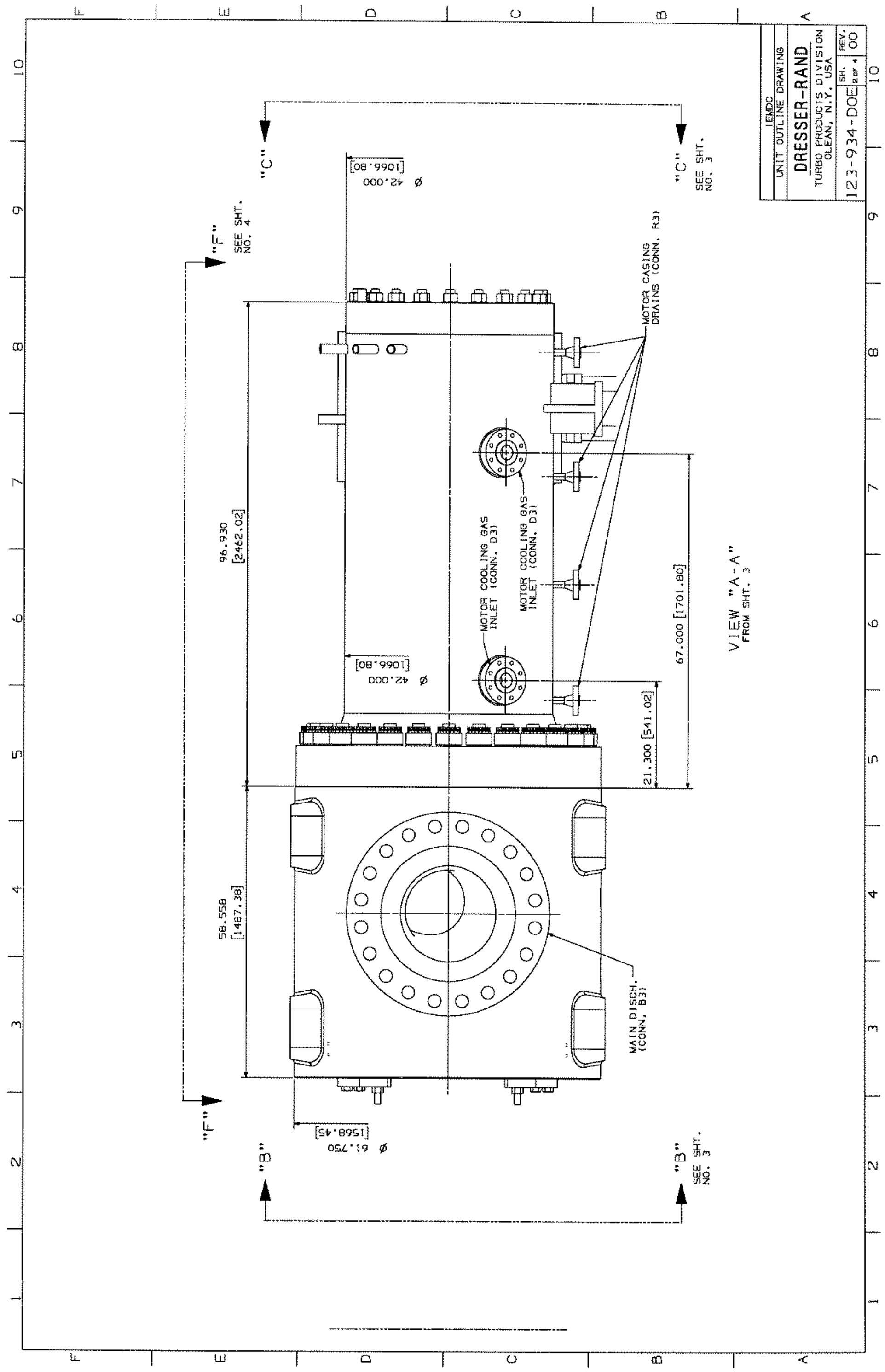
kW	Kilowatt
ksi	1000 pounds per square inch
Lb.	Pound
LEFM	Linear Elastic Fracture Mechanics
Log Dec	Logarithmic Decrement
m	Meter
MCOS	Maximum Continuous Operating Speed
mg	Milligram
Mil	one one-thousands of an inch
MIMO	Multi Input Multi Output
MMCF	Million Cubic Feet
MTBF	Mean Time Between Failures
MTBM	(Also MTBR) Mean time Between Maintenance/Repair
MW	Moleweight
N	Nomenclature for speed (usually RPM)
NEMA	National Electrical Manufacturers Association
Ni-Cr-Mo-V	Nickel-Chrome-Molybedenum-Vanadium (steel alloying elements)
Oz	Ounce
PCC	Point of Common Coupling
PLC	Programmable Logic Controller
psig	Pounds per square inch gauge
PT	Potential Transformer
RFQ	Request for Quotation
RPM	Revolutions per minute
RTD	Restistance Temperature Detector
SCADA	Supervisory Control and Data Acquisition
SISO	Single Input Single Output
SM	Separation Margin
TE	Thrust end
THD	Total Harmonic Distortion
UPS	Uninterruptible Power Supply
V	Volts
VFD	Variable Frequency Drive
σ_y	Yield Strength

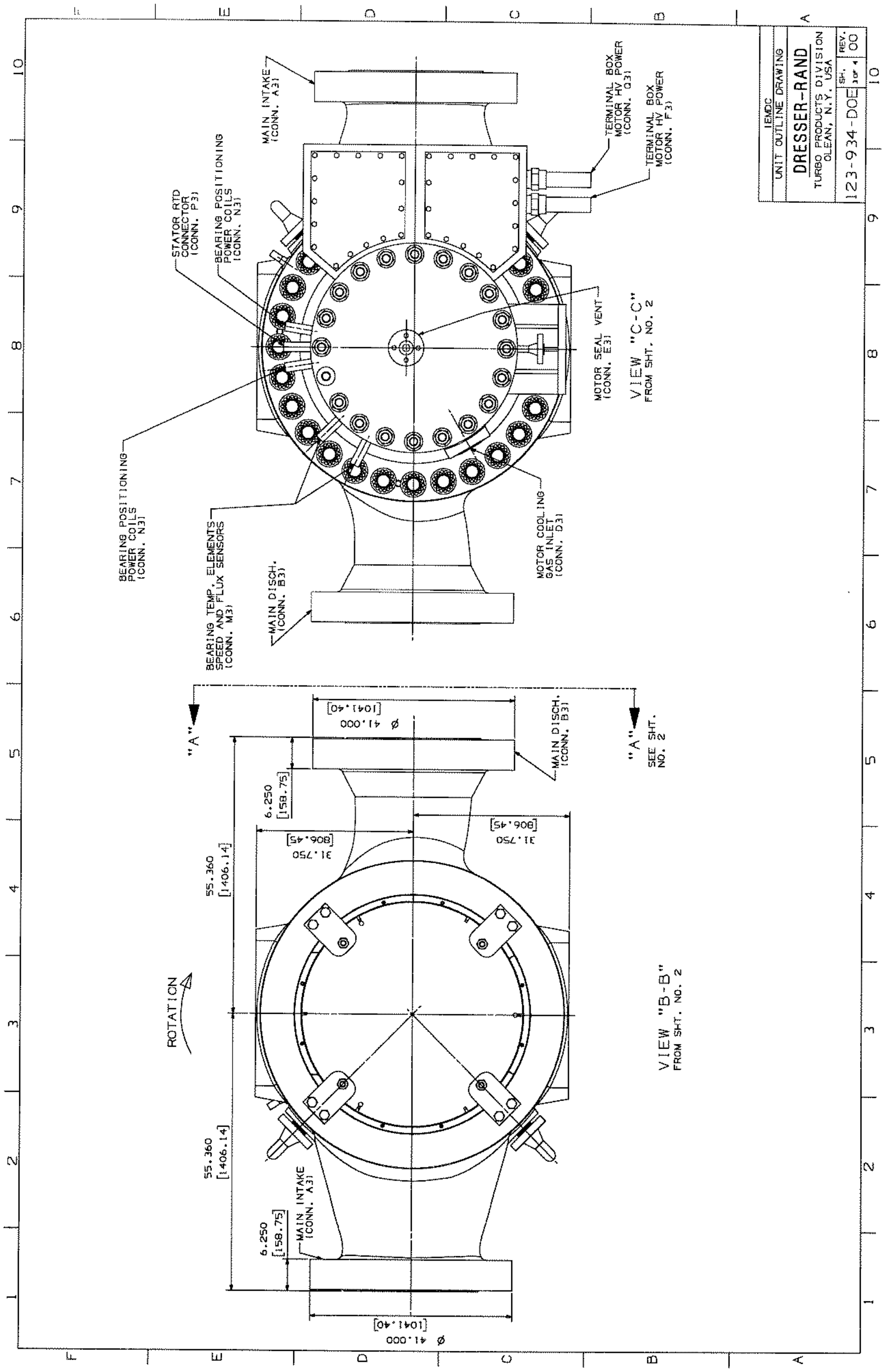
APPENDICES

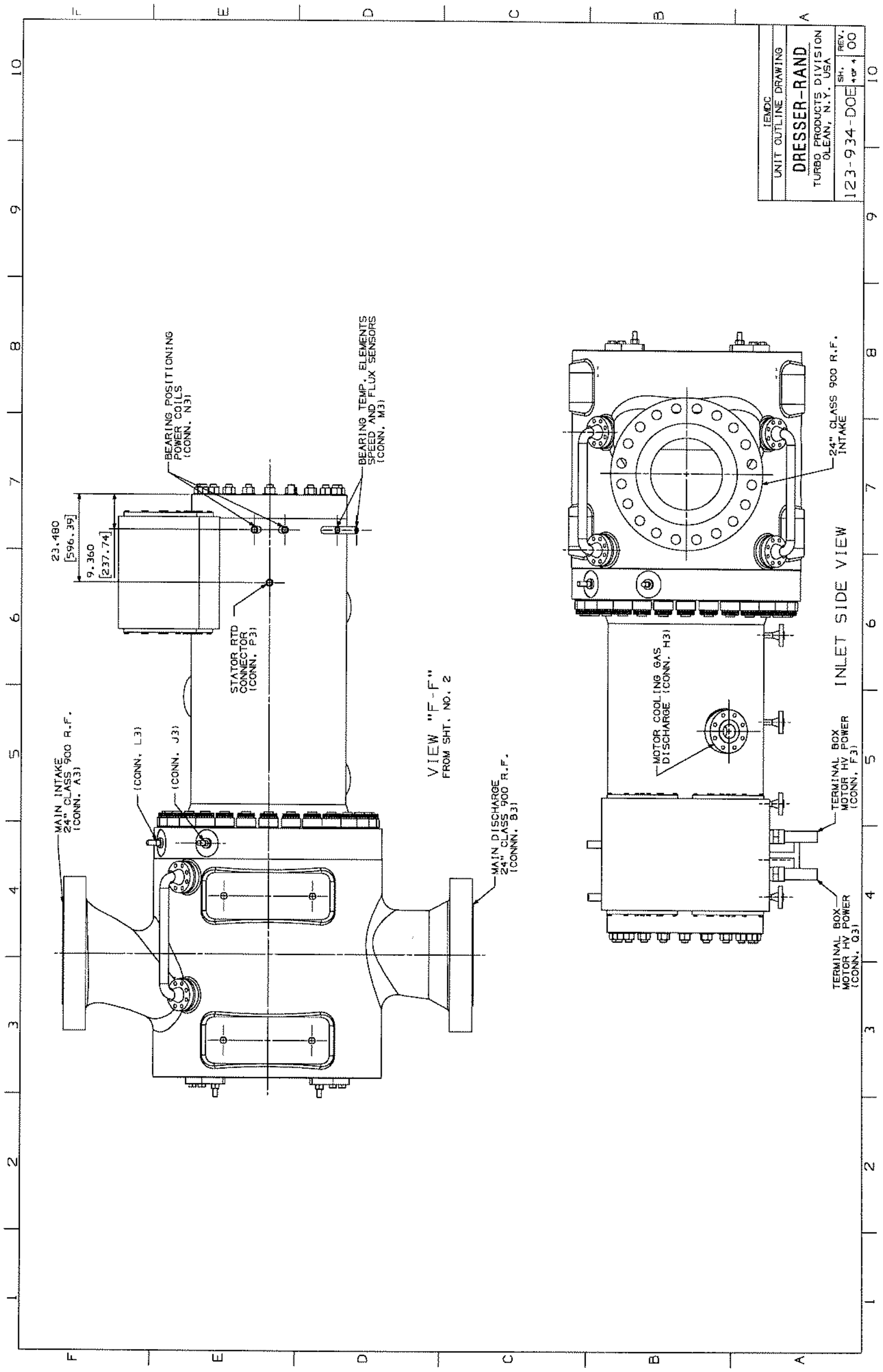
Appendix A

Outline Drawing of the IEMDC Unit Assembly


COMPRESSOR CONNECTIONS ARE IN COMPLIANCE WITH ASME B16.5-1996.
APPROXIMATE WEIGHTS
TOTAL MOTOR/COMPRESSOR-74,050 LBS.(33,589 KG)
CONNECTION LIST
CONN POINT | DESCRIPTION
A3 | INLET-IEMDC COMPRESSOR MAIN INLET
B3 | DISCHARGE-IEMDC COMPRESSOR MAIN DISCHARGE
C3 | DRAIN-IEMDC COMP'R. CASING DRAINS
D3 | INLET-MOTOR COOLING GAS
E3 | VENT-MOTOR SEAL VENT
F3 | MOTOR HV POWER CONNECTION
G3 | SUPPLY-MOTOR COOLING GAS FROM CUSTOMER'S DISCHARGE PIPING TO COOLING GAS CONSOLE
H3 | DISCHARGE-MOTOR COOLING GAS DISCHARGE
J3 | ELECTRICAL CONN.- COMPRESSOR END RADIAL BEARING TEMPERATURE ELEMENTS AND POSITION SENSORS (JB-1).
L3 | ELECTRICAL CONN.- COMPRESSOR END RADIAL BEARING #1 POSITIONING POWER COILS (JB-2)
M3 | ELECTRICAL CONN.- MOTOR FREE END BEARING TEMPERATURE ELEMENTS/ POSITION SENSORY SPEED SENSORS, AND FLUX SENSORS (JB-3).
N3 | ELECTRICAL CONN.- MOTOR FREE END BEARING POSITIONING POWER COILS (JB-4).
P3 | ELECTRICAL CONN.- MOTOR STATOR RTD'S. (JB-4).
Q3 | MOTOR HV POWER CONNECTION
R3 | MOTOR LOW POINT DRAINS
S3 | COOLING GAS SUPPLY FROM CONSOLE TO MOTOR
T3 | COOLING GAS REFERENCE FROM CUSTOMER'S SUCTION PIPING TO COOLING GAS CONSOLE
R3 | INERTGAS PURGE CONNECTION
MAIN INTAKE 24" CLASS 900 R.F. (CONN. A3)
MAIN DISCHARGE 24" CLASS 900 R.F. (CONNN. B3)
DO NOT USE FOR MANUFACTURE
IEMDC
UNIT OUTLINE DRAWING
DRESSER-RAND
TURBO PRODUCTS DIVISION
OLEAN, N.Y. USA
123-934-DOE
SH. 1 OF 4
REV. 00


IEMDC
UNIT OUTLINE DRAWING
DRESSER-RAND
TURBO PRODUCTS DIVISION
OLEAN, N.Y. USA
123-934-DOE
VIEW "A-A"
FROM SHT. 3
"C"
SEE SHT. NO. 3
"B"
SEE SHT. NO. 3
"F"
SEE SHT. NO. 4
MOTOR CASING DRAINS (CONN. R3)
MOTOR COOLING GAS INLET (CONN. D3)
MAIN DISCH. (CONN. B3)
Ø 42.000 [1066.80]
Ø 61.750 [1568.45]
96.930 [2462.02]
58.558 [1487.38]
67.000 [1701.80]
21.300 [541.02]


IEMDC
UNIT OUTLINE DRAWING
DRESSER-RAND
TURBO PRODUCTS DIVISION
OLEAN, N.Y. USA
123-934-DOE
VIEW "C-C"
FROM SHT. NO. 2
VIEW "B-B"
FROM SHT. NO. 2
MAIN INTAKE
(CONN. A3)
MAIN DISCH.
(CONN. B3)
MOTOR SEAL VENT
(CONN. E3)
MOTOR COOLING
GAS INLET
(CONN. D3)
ROTATION
"A"
SEE SHT. NO. 2


IEMDC
UNIT OUTLINE DRAWING
DRESSER-RAND
TURBO PRODUCTS DIVISION
OLEAN, N.Y. USA
123-934-DOE
SH. 1 OF 4
REV. 00
BEARING POSITIONING POWER COILS (CONN. N3)
BEARING TEMP. ELEMENTS SPEED AND FLUX SENSORS (CONN. M3)
23.480 [596.39]
9.360 [237.74]
STATOR RTD CONNECTOR (CONN. P3)
MAIN INTAKE 24" CLASS 900 R.F. (CONN. A3)
(CONN. L3)
(CONN. J3)
VIEW "F-F"
FROM SHT. NO. 2
MAIN DISCHARGE 24" CLASS 900 R.F. (CONN. B3)
24" CLASS 900 R.F. INTAKE
INLET SIDE VIEW
MOTOR COOLING GAS DISCHARGE (CONN. H3)
TERMINAL BOX MOTOR HV POWER (CONN. F3)
TERMINAL BOX MOTOR HV POWER (CONN. Q3)

Appendix B
IEMDC System Performance Specification

Summary: The performance objective of the IEMDC unit was to select an operating set that would be applicable to a wide range of customer applications. An operating point for the project was selected on this basis and used to set the design parameters for the project.

Table B1. Project Design Goal Parameters

Parameter	Project Goal	Selected Design point
Inlet volume flow rate (MMSCFD)	300 to 700	613
Inlet pressure range (psig)	500 to 700	763*
Discharge Pressure Range (psig)	700 to 1200	1100
Inlet Temperature Range (°F)	60 to 80	60
Operating Speed Range (RPM)	8,000 to 12,000	11268
Power Consumption (MW)	5 to 10	8.7

*Outside the anticipated range, but not outside the operating limits of the IEMDC.

A gas composition range was defined for use in establishing a design basis for the IEMDC unit. It is expected that the IEMDC will be compressing pipeline quality “sweet” natural gas. However, consideration was given to the effect of the listed constituents on the design as they may appear in the gas stream in varying concentrations.

Table B2. Natural Gas Composition

Constituent	Minimum Value (Mole %)	Maximum Value (Mole %)
Methane (C1)	80%	100
Ethane (C2)		14
Propane (C3)		5
Butanes (C4)		2
Pentanes and Heavier (typically C5 thru C7)		0.5
Total Diluents (Nitrogen, Carbon dioxide, Helium, Oxygen, and other inert gases)		18
Carbon Dioxide		5
Hydrogen		5
Total Unsaturated Hydrocarbons		0.5
Carbon Monoxide		0.1
Oxygen		1
Oxygen Trace Components		0.10 (10 ppm)
Hydrogen Sulfide		0.25 grains/CCF (5.7 mg/m3)
Mercaptan Sulfur		0.50 grains/CCF (11.5 mg/m3)
Total Sulfur		1.00 grains/CCF (22.9 mg/m3)
Water Vapor		7.0 pounds/MMCF (110 mg/m3)
Solids (undefined)		5 mg/m3
Liquids (Liquefiable Hydrocarbons)		0.2 gallon / MCF

Moleweight Range (approximately): 16 to 20

The unit design operating point selected for the IEMDC unit for the project is as follows.

Table. B3 Prototype Design Operating point

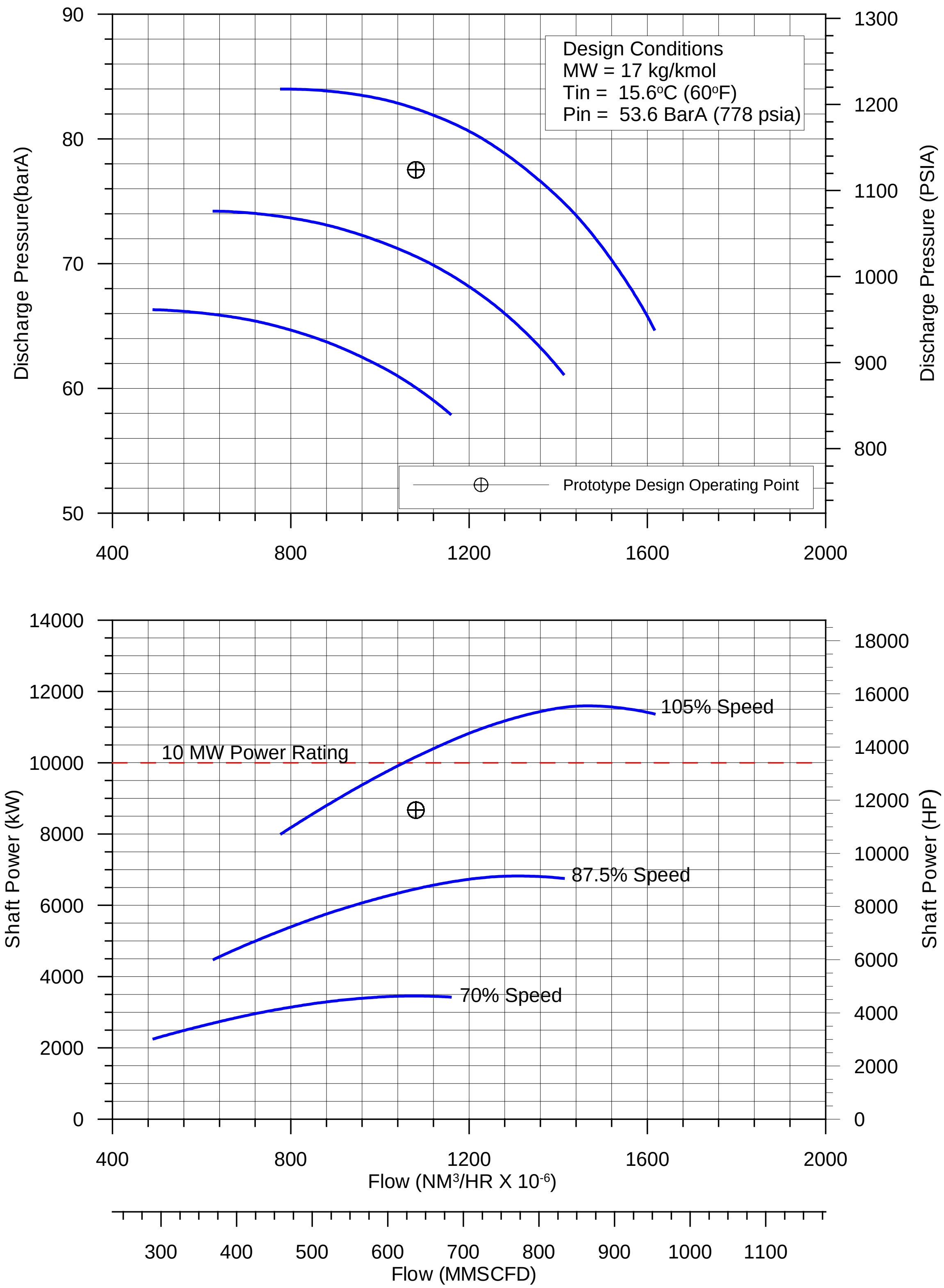
Parameter	IEMDC Prototype Unit Design Point
Gas Handled	Natural Gas
MMSCFD	613
Inlet Conditions	
Pressure (psig)	763
Temperature (F)	60
Discharge Conditions	
Pressure (psig)	1100
Speed (RPM)	11238

Table B4. Motor Design Operating Parameters

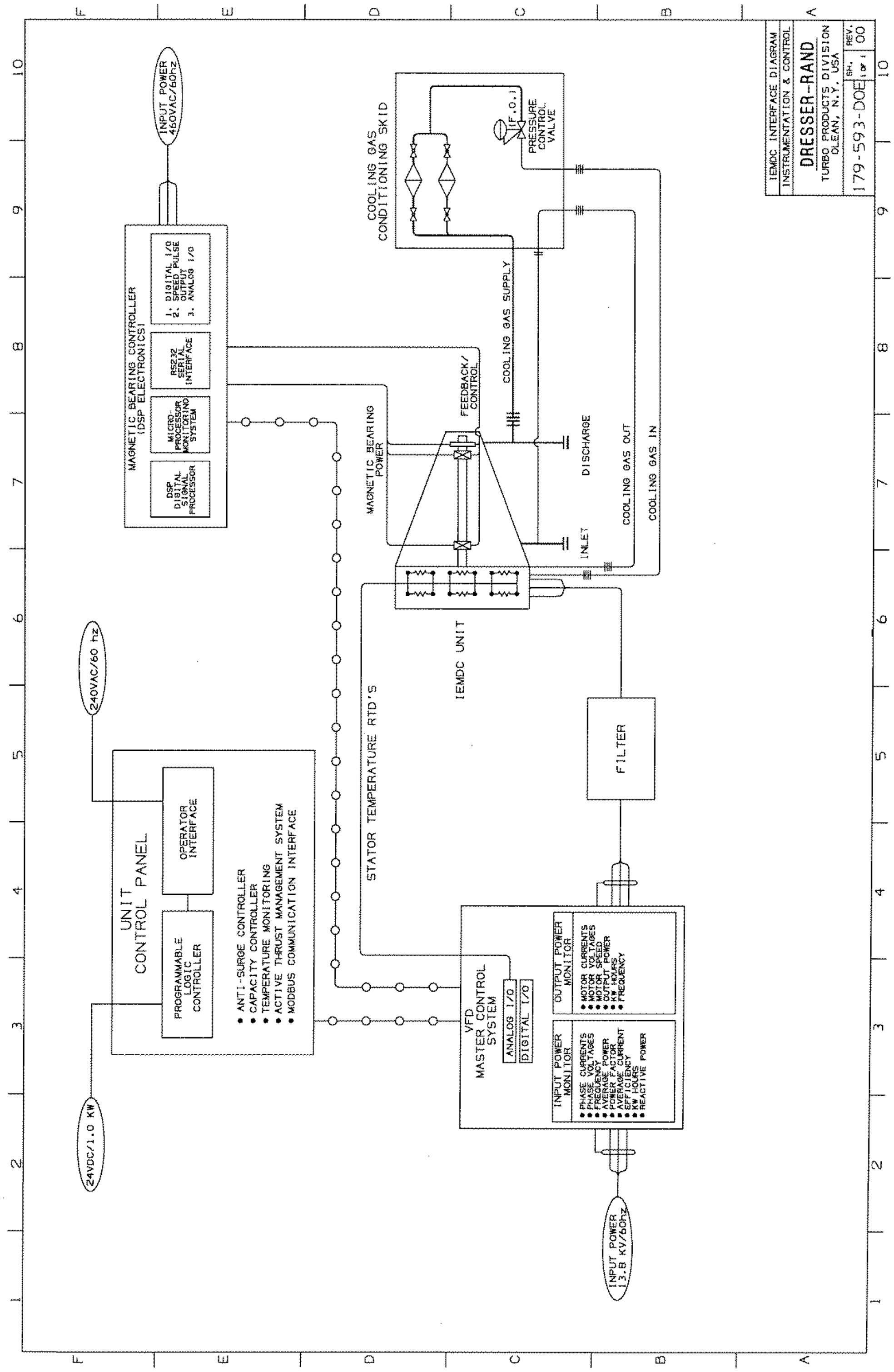
Parameter	**Value**
Motor Type	Squirrel Cage Induction
Rated power (MW) [HP]	10 [13,405]
Rotational Speed (RPM)	12,000
Rated Torque (Ft-lb.)	5900
Maximum Torque (ft-lb)	6500
Efficiency at Rated Power (%)	95
Bearing Type	Magnetic
Phase Number	3
Winding Configuration	2 Parallel Wye
Maximum Frequency (Hz)	200
Pole Number	2
Number of Terminal Connections	6
Maximum Voltage (kV L-L)	6.9
Cooling System	Filtered Ventilation w/ Methane Gas
Phase Current (amps)	1118

IEMDC UNIT

Predicted Performance Curves



Appendix C
IEMDC System Instrumentation and Motor Control Schematic


INPUT POWER 460VAC/60Hz
MAGNETIC BEARING CONTROLLER (DSP ELECTRONICS)
DSP DIGITAL SIGNAL PROCESSOR
MICRO-PROCESSOR MONITORING SYSTEM
RS232 SERIAL INTERFACE
1. DIGITAL I/O
2. SPEED PULSE OUTPUT
3. ANALOG I/O
COOLING GAS CONDITIONING SKID
(F.O.)
PRESSURE CONTROL VALVE
COOLING GAS SUPPLY
FEEDBACK/ CONTROL
MAGNETIC BEARING POWER
DISCHARGE
INLET
COOLING GAS OUT
COOLING GAS IN
IEMDC UNIT
STATOR TEMPERATURE RTD'S
FILTER
240VAC/60 hz
UNIT CONTROL PANEL
OPERATOR INTERFACE
PROGRAMMABLE LOGIC CONTROLLER
• ANTI-SURGE CONTROLLER
• CAPACITY CONTROLLER
• TEMPERATURE MONITORING
• ACTIVE THRUST MANAGEMENT SYSTEM
• MODBUS COMMUNICATION INTERFACE
24VDC/1.0 KW
VFD MASTER CONTROL SYSTEM
ANALOG I/O
DIGITAL I/O
INPUT POWER MONITOR
• PHASE CURRENTS
• PHASE VOLTAGES
• FREQUENCY
• AVERAGE POWER
• POWER FACTOR
• AVERAGE CURRENT
• EFFICIENCY
• KW HOURS
• REACTIVE POWER
OUTPUT POWER MONITOR
• MOTOR CURRENTS
• MOTOR VOLTAGES
• MOTOR SPEED
• OUTPUT POWER
• KW HOURS
• FREQUENCY
INPUT POWER 13.8 KV/60Hz
IEMDC INTERFACE DIAGRAM
INSTRUMENTATION & CONTROL
DRESSER-RAND
TURBO PRODUCTS DIVISION OLEAN, N.Y. USA
179-593-DOE
SH. 1 of 1
REV. 00

Appendix D

Layout Drawing of the 12,000 RPM Direct Drive Motor

REVISIONS

1	S.O. ENG04003
	AS OF 07/19/04

GRP REQD	PART NAME	SIZE AND REFERENCE INFORMATION	MATERIAL, PART NUMBER OR REF. DWG.	IDENT CLASS	NOTE	ITEM
.	END COVER	SA-266, SA-516	SA-105	.	..	30
.	INSUL WASHER	.	G-11	.	..	29
.	NUT	.	C'STEEL	.	..	28
.	INSUL THRU BOLT	.	NITRONIC	.	..	27
.	FINGER PLATE	.	304 SST	.	..	26
.	VENT PLT, CENTER	.	304 SST	.	..	25
						24
.	#3 GAS SEAL	.	.	.	..	23
.	THRUST BRG STAT	.	.	.	..	22
.	SPACER	.	.	.	..	21
.	THR COLL LCK KEY	.	.	.	..	20
.	THRUST COLLAR	.	.	.	..	19
.	TE CATCH BRG SLV	.	.	.	..	18
.	TE CATCHER BRG	.	.	.	..	17
.	TE MAG BRG PS	.	.	.	..	16
.	TE MAG BRG RTR	.	.	.	..	15
.	TE MAG BRG STAT	.	.	.	..	14
.	BRG SHIELD	.		.	..	13
.	IE CATCH BRG SLV	.	.	.	..	12
.	IE CATCHER BRG	.	.	.	..	11
.	IE MAG BRG P S	.	.	.	..	10
.	IE MAG BRG RTR	.	.	.	..	9
.	IE MAG BRG STAT	.	.	.	..	8
.	IMPELLER	.	.	.	..	7
.	CONNECT RING	.		.	..	6
.	ROTOR BARS	.		.	..	5
.	RETAINING RING	.	ALLOY STL	.	..	4
.	COILS	.	.	.	..	3
.	LAMINATIONS	.	.	.	..	2
.	ROTOR SHAFT	.	ALLOY STL	.	..	1

I.D. ⊲— IDENT CLASS
G.N. ⊲— GROUP NOTE

PARTS LIST

GRP REQD	PART NAME	SIZE AND REFERENCE INFORMATION	MATERIAL, PART NUMBER OR REF. DWG.	IDENT CLASS	NOTE	ITEM
						60
						59
						58
						57
						56
						55
						54
						53
.	RETAINING KEY	.	STEEL	.	..	52
						51
.	#1/#2 GAS SEAL	.	.	.	..	50
.	TERMINAL BOX	.	STEEL	.	..	49
.	TERMINAL GLAND	.	.	.	..	48
		.				47
		.				46
.	MAG BRG PWR	TERMINAL STRIP	.	.	..	45
.	MAG BRG CTRL	TERMINAL STRIP	.	.	..	44
.	COVER	.	AL	.	..	43
.	IE BRG HSG	.	C'STEEL	.	..	42
.	INNER FRAME	.	C'STEEL	.	..	41
.	MAG BRG CTRL CONN	FEED THROUGH	.	.	..	40
.	MAG BRG CONN	POWER FEED THROUGH	.	.	..	39
.	RTD CONN	FEED THROUGH	.	.	..	38
.	SUPERBOLT	(3.500-4)	.	.	..	37
.	STUD	(3.500-4)	ALLOY STL	.	..	36
.	WASHER	.	ALLOY STL	.	..	35
.	NUT	.	ALLOY STL	.	..	34
.	STUD	(2.750-8)	ALLOY STL	.	..	33
.	SHELL & FLANGE	SA-266, SA-516	SA-105	.	..	32
.	O-RING	.		.	..	31

I.D. ⊲— IDENT CLASS
G.N. ⊲— GROUP NOTE

PARTS LIST

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DFTG	BIERNACKI	3/23/04
CHKR		
DES		
MFG		
QA		
APP		
APP		
APP		
APP		
MGR/DESIGNEE		

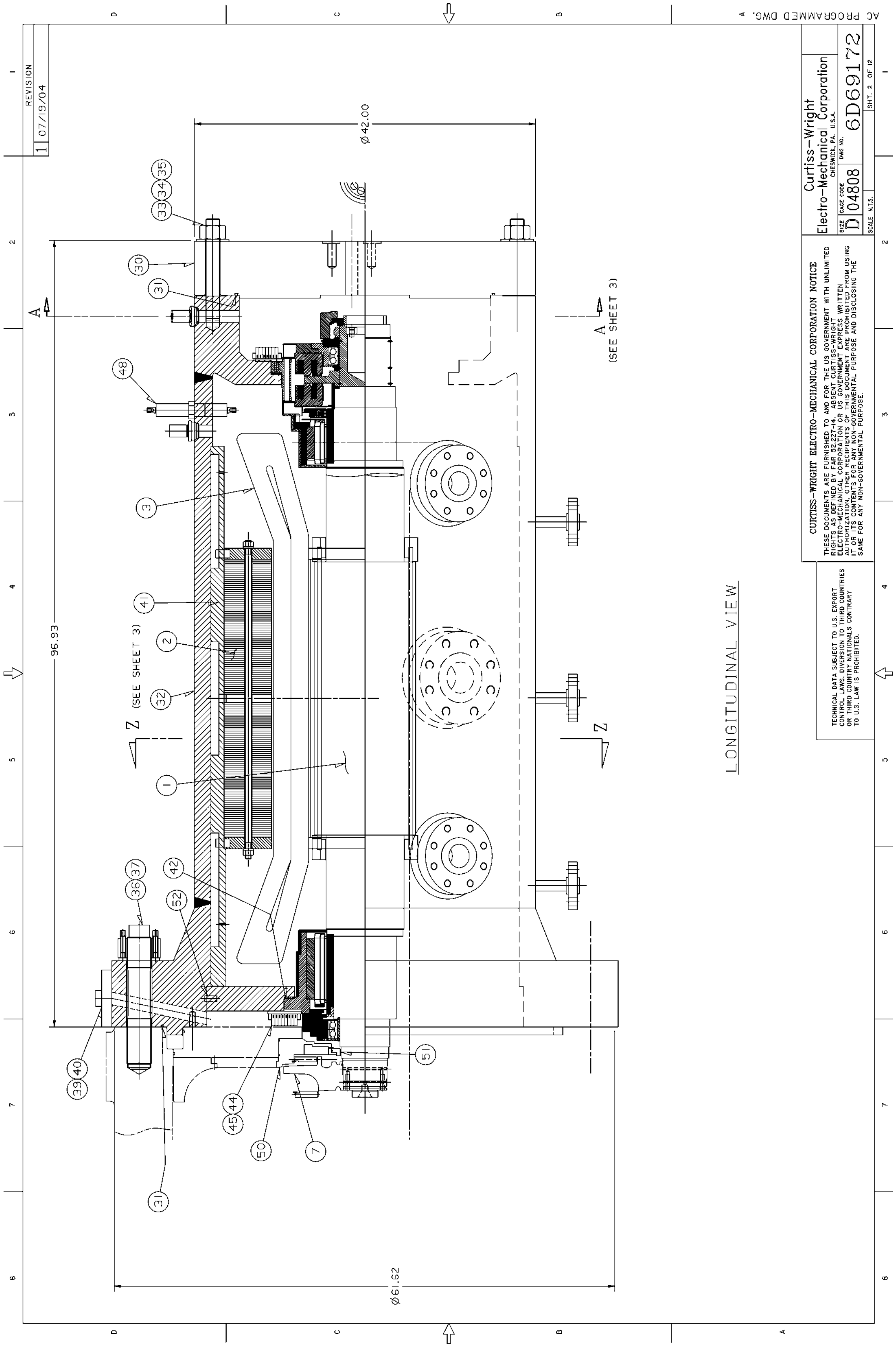
Curtiss-Wright
Electro-Mechanical Corporation
CHESWICK, PA. U.S.A.

IEMDC HIGH SPEED MOTOR
10 MEGAWATT
LAYOUT

SIZE	CAGE CODE	DWG NO.
D	04808	6D69172

SCALE N.T.S. WEIGHT 35,000 LBS. SHT. 1 OF 12

AC PROGRAMMED DWG.


LONGITUDINAL VIEW
(SEE SHEET 3)
Ø42.00
96.93
Ø61.62
REVISION
1 07/19/04
Curtiss-Wright
Electro-Mechanical Corporation
CHESWICK, PA. U.S.A.
D 04808
6D69172
SCALE N.T.S.
SHT. 2 OF 12
AC PROGRAMMED DWG.
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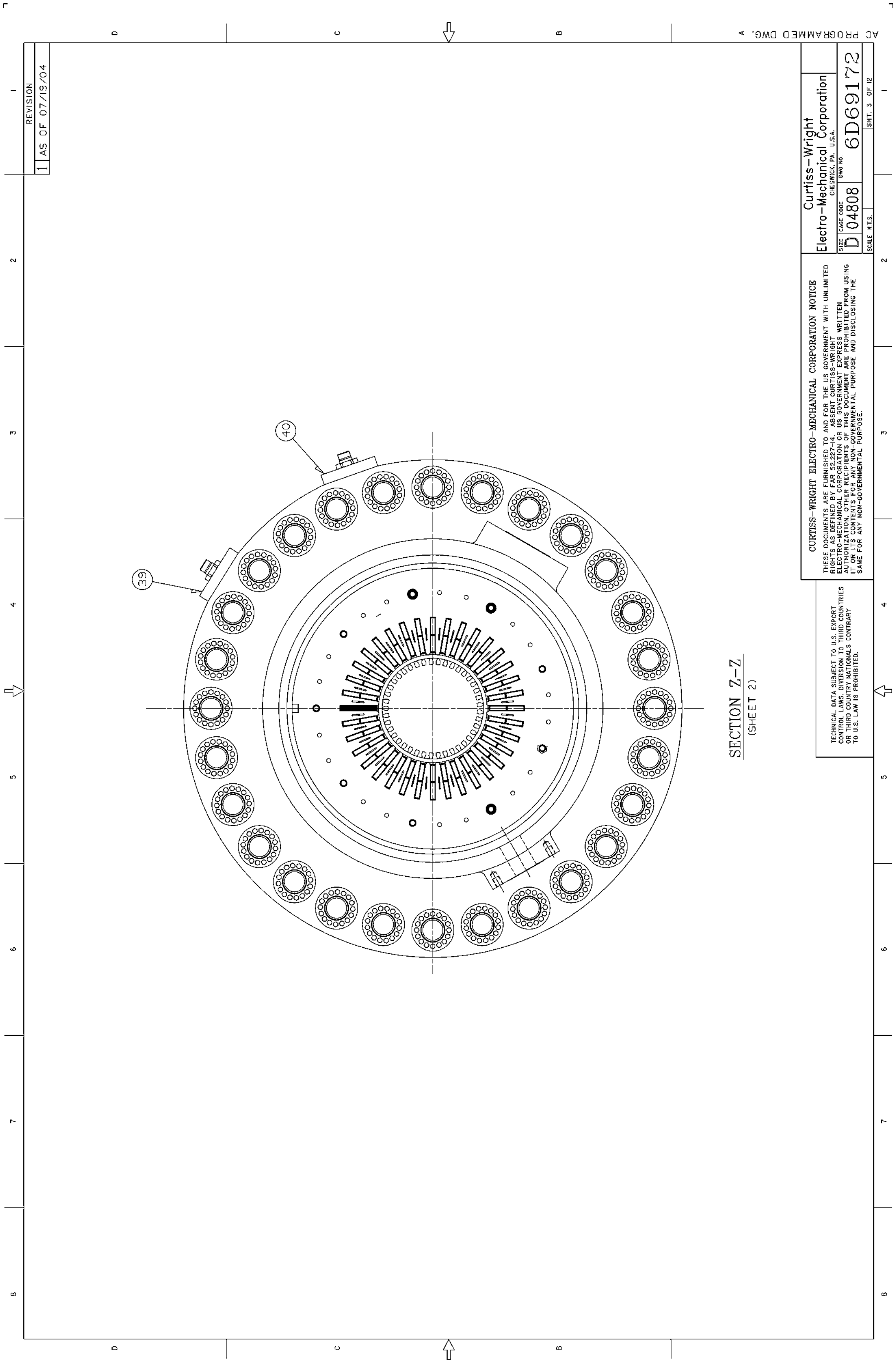
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Curtiss-Wright
Electro-Mechanical Corporation
D
04808
6D69172
REVISION
AS OF 07/19/04
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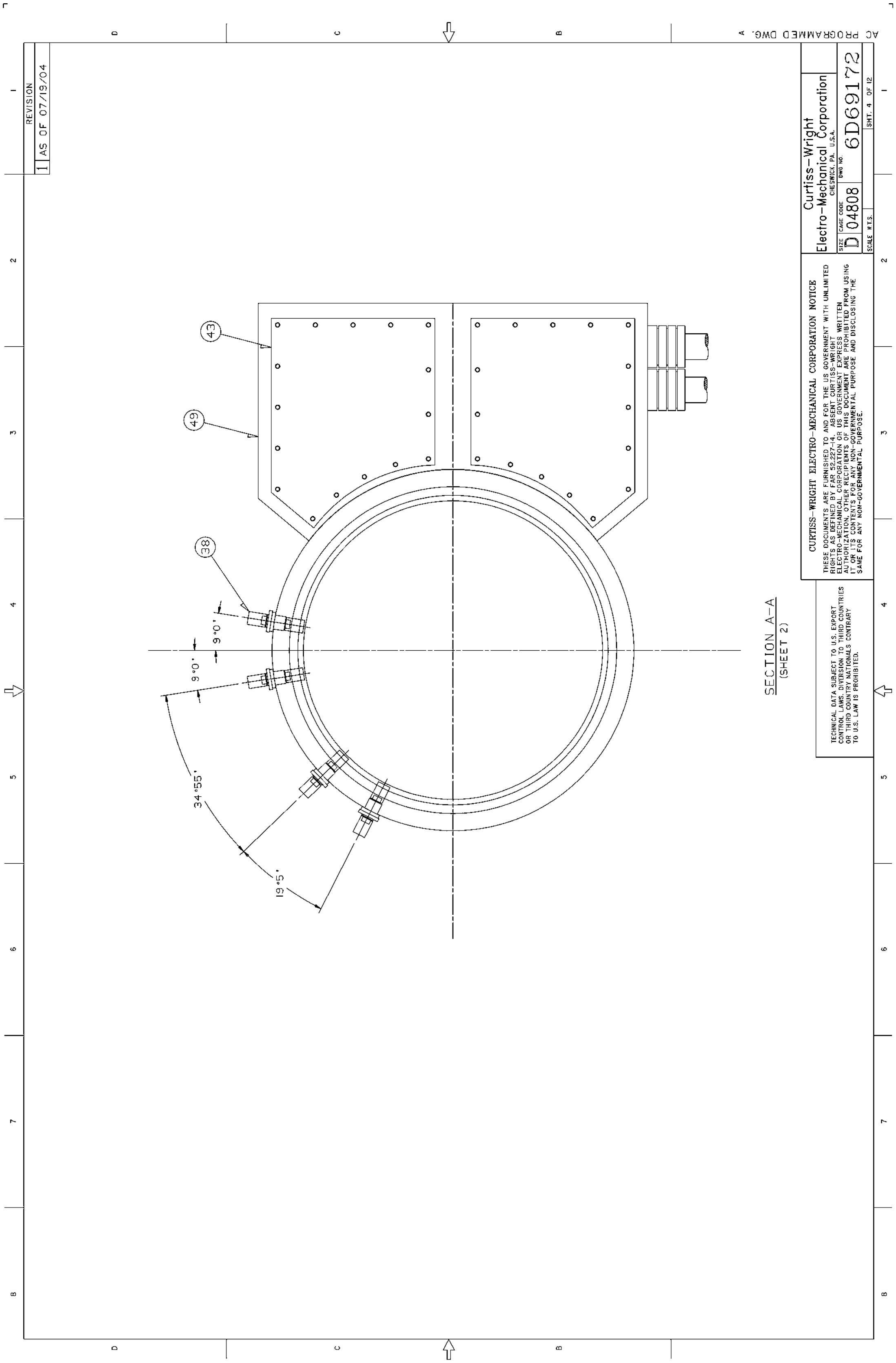
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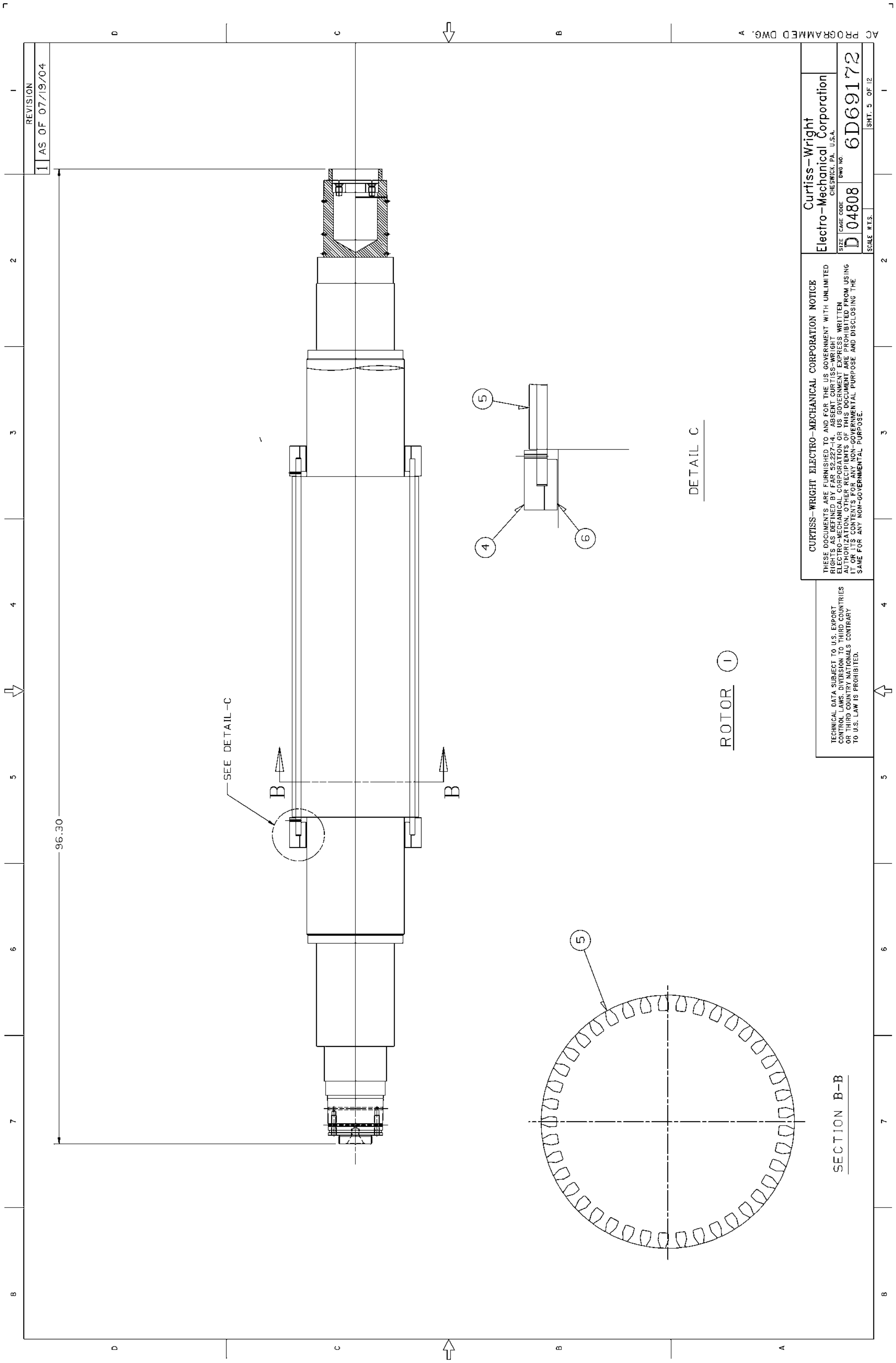
(SHEET 2)
Curtiss-Wright
Electro-Mechanical Corporation
D 04808
6D69172
REVISION
1 AS OF 07/19/04
9°0'
9°0'
34°55'
19°5'
43
49
38
AC PROGRAMMED DWG.
CURTISS-WRIGHT ELECTRO-MECHANICAL CORPORATION NOTICE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1 AS OF 07/19/04
AC PROGRAMMED DWG.
Curtiss-Wright
Electro-Mechanical Corporation
CHESWICK, PA. U.S.A.
D 04808
6D69172
SHT. 5 OF 12
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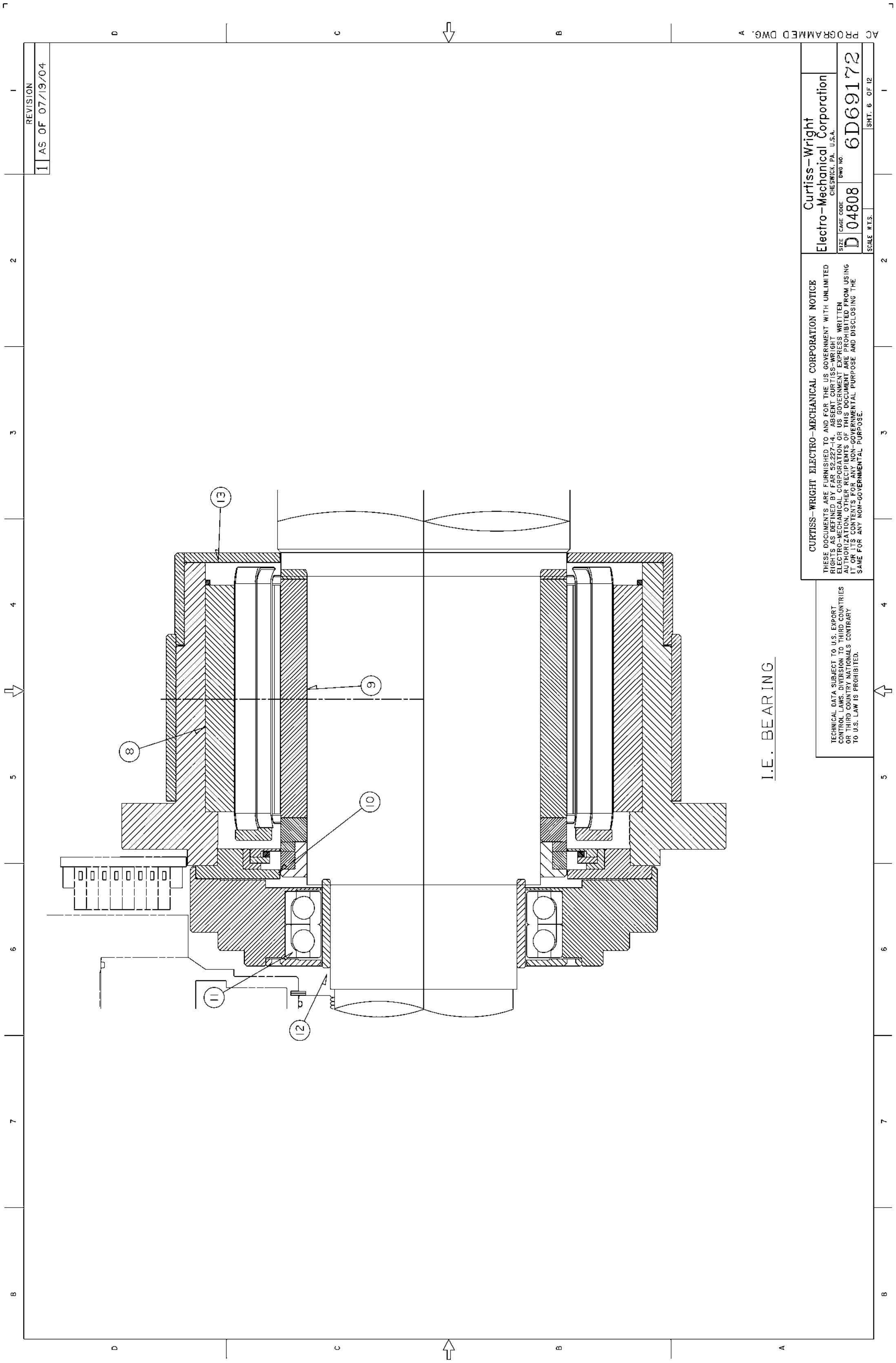
DETAIL C
ROTOR 1
SEE DETAIL-C
96.30
SECTION B-B


REVISION
AS OF 07/19/04
Curtiss-Wright
Electro-Mechanical Corporation
CHESWICK, PA. U.S.A.
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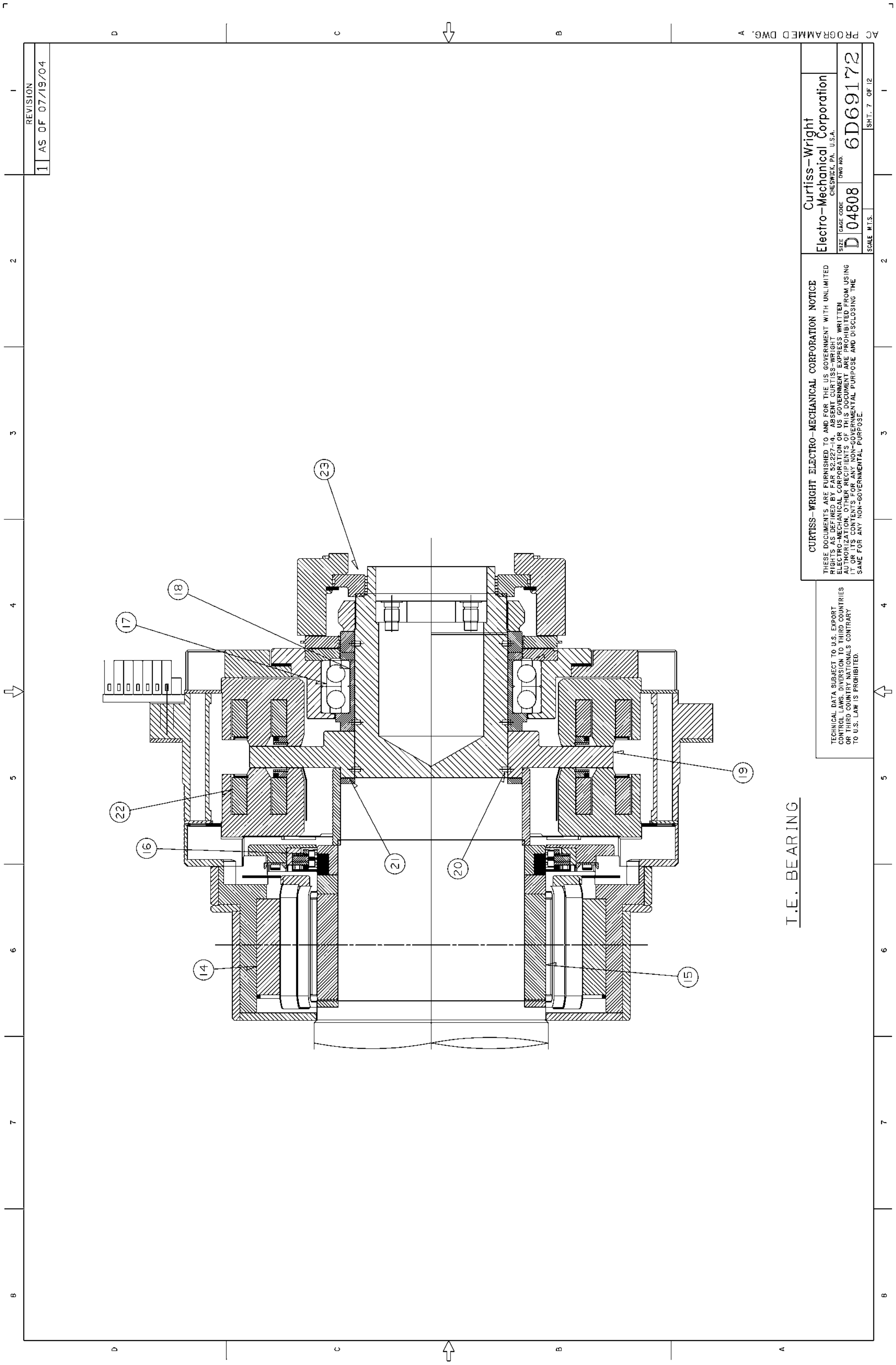
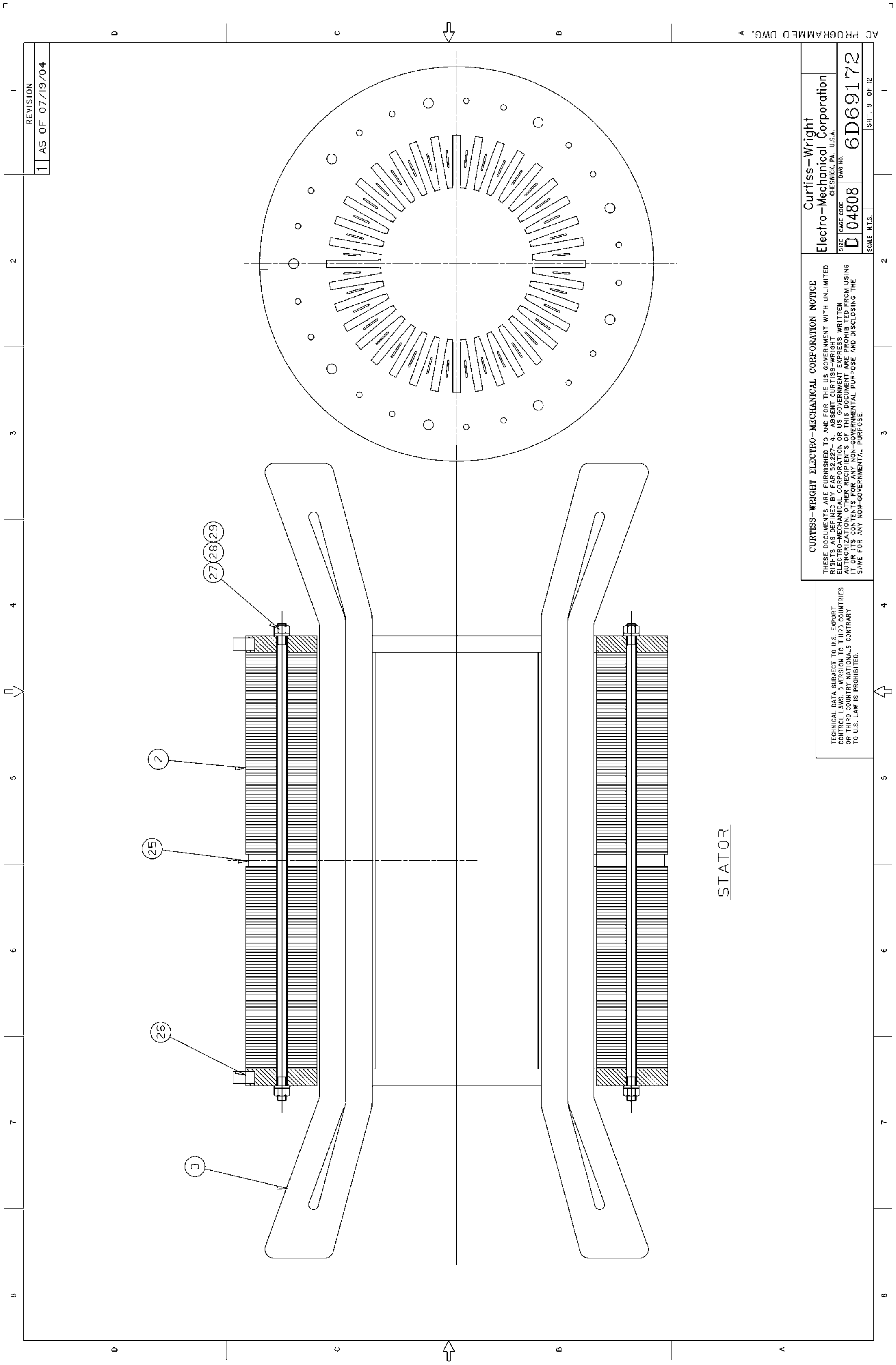
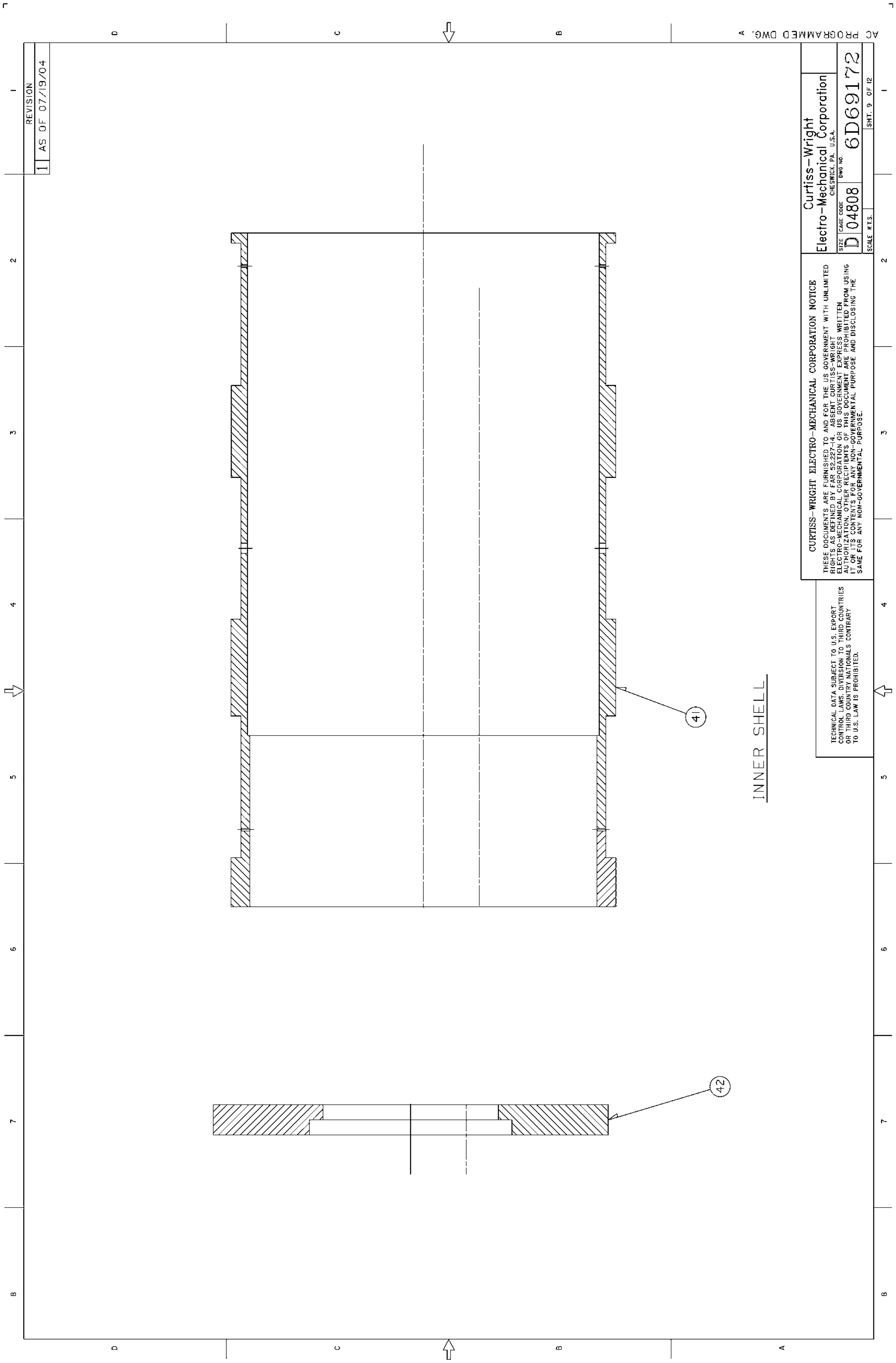
04808
6D69172
SHT. 6 OF 12
AC PROGRAMMED DWG.
CURTISS-WRIGHT ELECTRO-MECHANICAL CORPORATION NOTICE
I.E. BEARING


T.E. BEARING
Curtiss-Wright
Electro-Mechanical Corporation
CHESWICK, PA. U.S.A.
D 04808
6D69172
CURTISS-WRIGHT ELECTRO-MECHANICAL CORPORATION NOTICE
REVISION
1 AS OF 07/19/04
AC PROGRAMMED DWG.


STATOR
Curtiss-Wright
Electro-Mechanical Corporation
04808
6D69172


REVISION
1 AS OF 07/19/04
Curtiss-Wright
Electro-Mechanical Corporation
D 04808
6D69172
SHT. 9 OF 12
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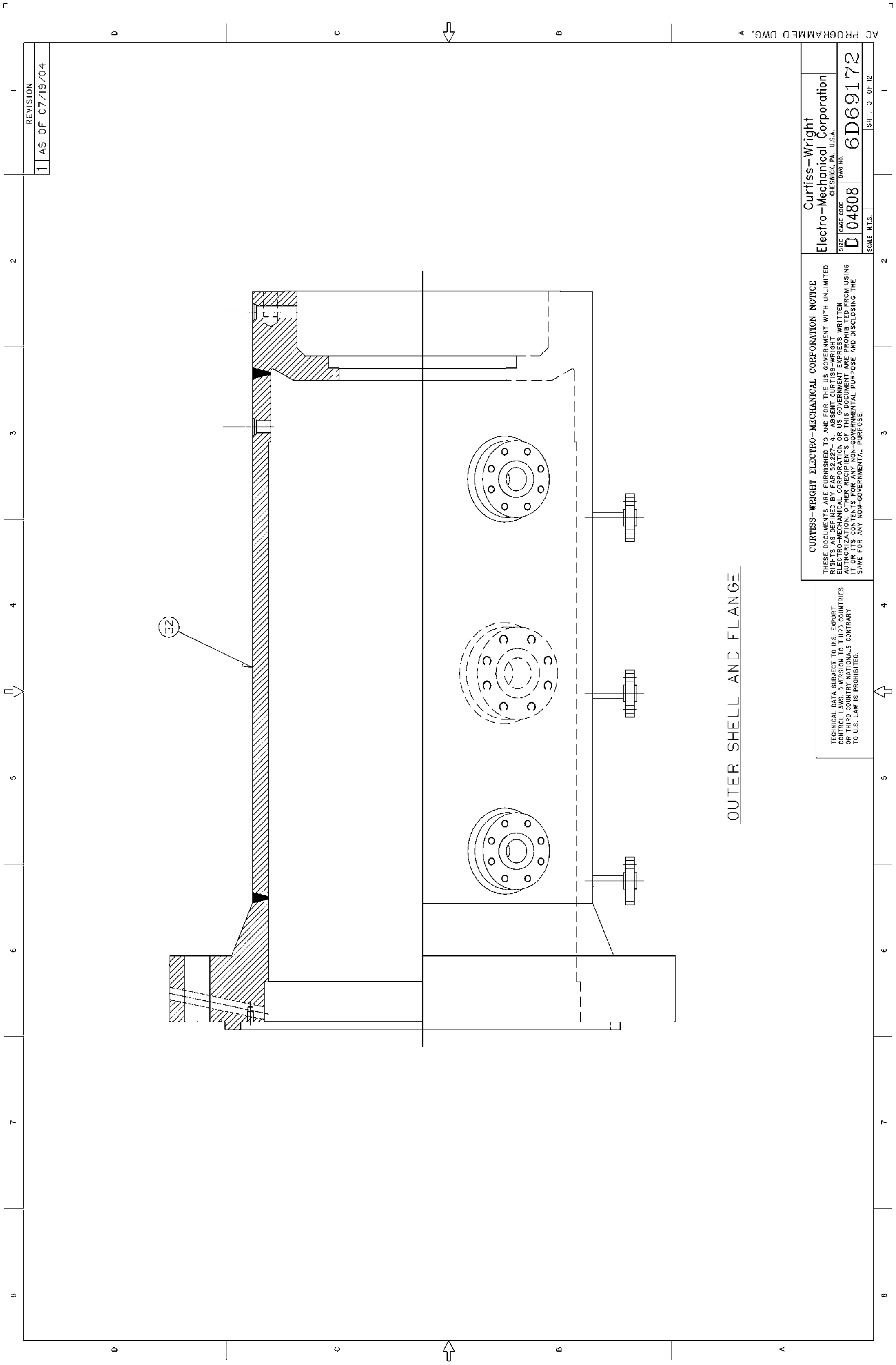
CURTISS-WRIGHT ELECTRO-MECHANICAL CORPORATION NOTICE
INNER SHELL
41
42


REVISION
1 AS OF 07/19/04
OUTER SHELL AND FLANGE
32
Curtiss-Wright
Electro-Mechanical Corporation
CHESWICK, PA. U.S.A.
D 04808
6D69172
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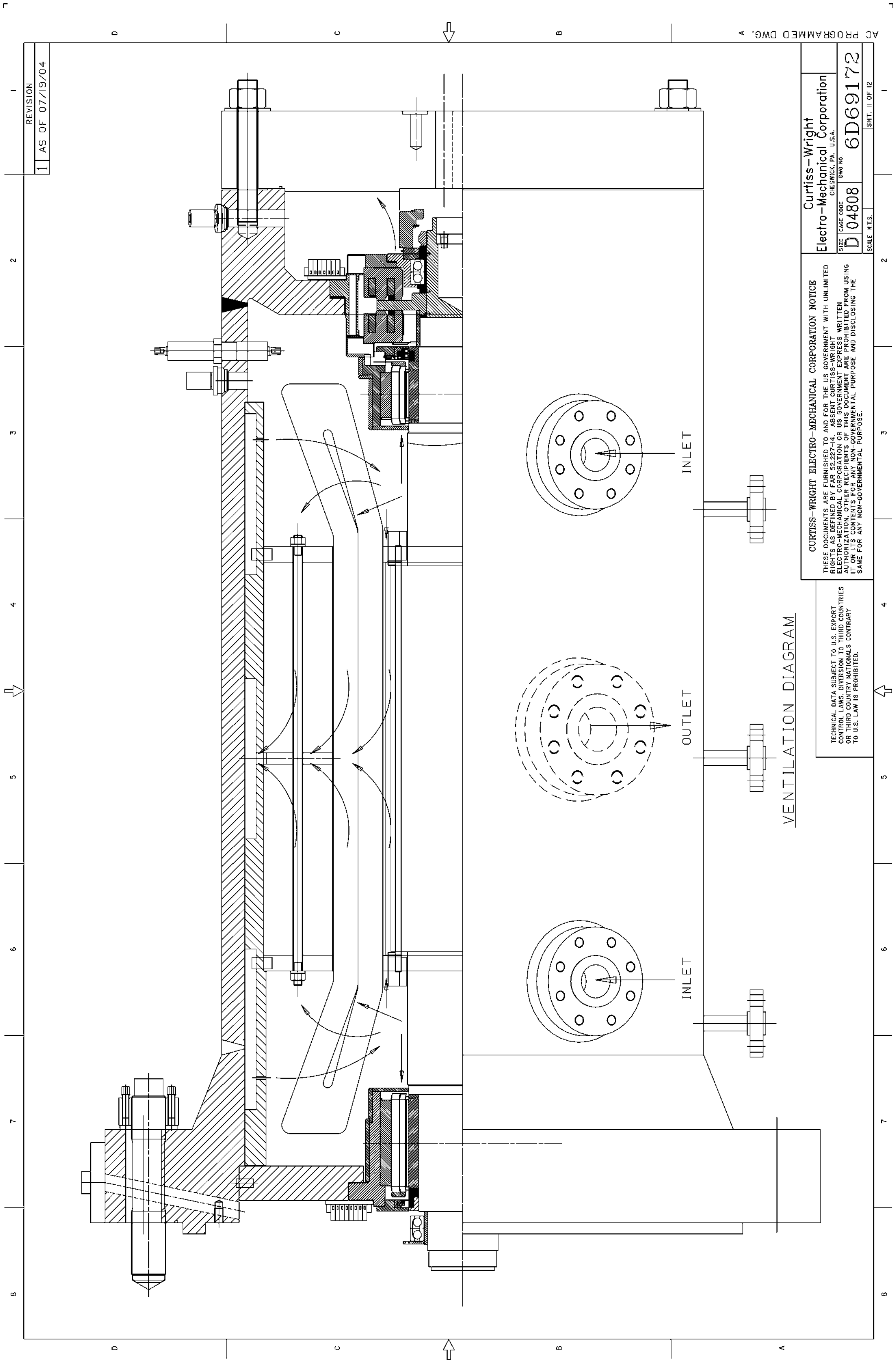
AC PROGRAMMED DWG.


VENTILATION DIAGRAM
INLET
OUTLET
INLET
Curtiss-Wright
Electro-Mechanical Corporation
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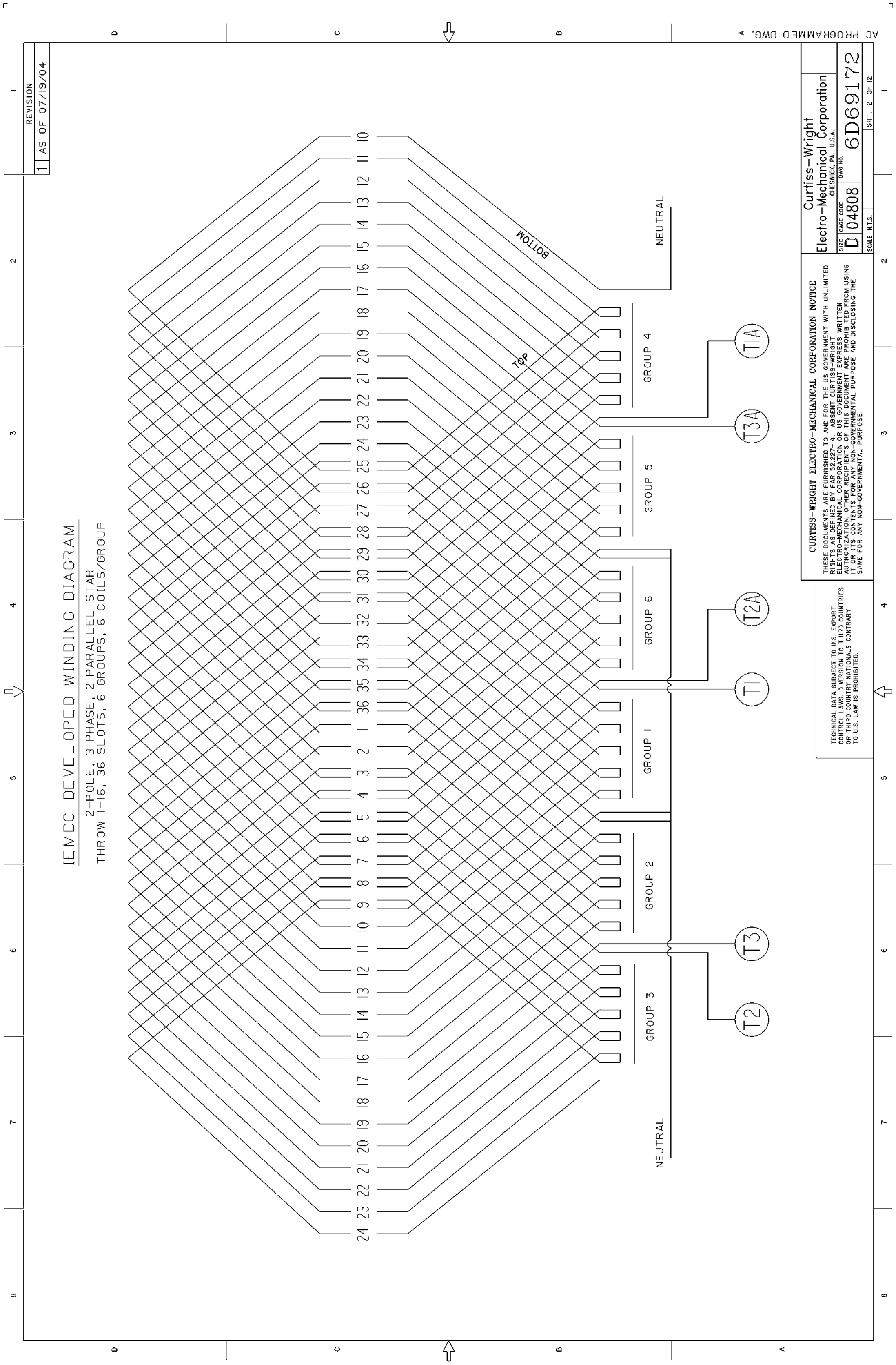
IEMDC DEVELOPED WINDING DIAGRAM
2-POLE, 3 PHASE, 2 PARALLEL STAR
THROW 1-16, 36 SLOTS, 6 GROUPS, 6 COILS/GROUP
REVISION
1 AS OF 07/19/04
BOTTOM
TOP
NEUTRAL
GROUP 1
GROUP 2
GROUP 3
GROUP 4
GROUP 5
GROUP 6
T1
T2
T3
T1A
T2A
T3A
Curtiss-Wright
Electro-Mechanical Corporation
CHESWICK, PA. U.S.A.
D 04808
6D69172
SHT. 12 OF 12
CURTISS-WRIGHT ELECTRO-MECHANICAL CORPORATION NOTICE
AC PROGRAMMED DWG.

Appendix E
Lateral Rotor Dynamics Analysis

IEMDC 10 MW MOTOR-COMPRESSOR
Lateral Critical Speed and Unbalance Response Analysis

Table of Contents

1.0 Introduction

The IEMDC motor-compressor was designed for 10-MW shaft output at 12,000 rpm with an operating speed range of 8,000 to 12,000 rpm. A rotordynamic analysis of the motor rotor was performed by S2M, with an overhung single stage centrifugal compressor and a thrust magnetic bearing disk at the opposite end of the impeller, and levitated by two radial magnetic bearings. The purpose of the analysis was to develop a robust rotor-bearing design that would conform to the rotor dynamic design requirements of the API standards 541 and 617, for critical speeds, unbalance vibration response and bearing loads

Appropriate FEA models were developed using the rotor geometry and the bearing transfer functions. The finite element program MADYN was used for the rotor dynamic analysis.

After the FEA model was completed, first off, free-free (with very soft bearing stiffness at both ends) critical speeds were calculated at 0-rpm and at trip speed of 12,000 rpm. This analysis confirmed that the selected bearing locations had adequate motion to effectively dampen out the vibrations for the bending mode as well as that the bearing span length would fit within the allocated space envelope of the motor design. The required system damping to reduce vibration response is provided by the magnetic bearings. Next, the analysis was performed to calculate and plot the critical speed map. This is a plot of the calculated undamped rotor frequencies versus bearing stiffness at 0-rpm. This plot helped to establish the required equivalent real stiffness of the impeller end and the thrust end bearings for rigid body and bending modes. After the separation of the undamped critical speeds with respect to the operating speed range was determined to be satisfactory, a damped critical speed and unbalance response analysis was performed to establish the magnitude of the rotor vibration at critical points along the rotor and the loads transmitted to the bearings due to rotor unbalance.

The rotordynamic model is discussed in Section 2, the results of critical speed analysis in Section 3, the unbalance response analysis results in Section 4, and the conclusion in Section 5.

2.0 Rotor Dynamics Model

Figure 1 shows the rotor configuration for the finite element model analysis. Also, shown on Figure 1 are the key elements such as motor rotor and shaft sections, impeller, thrust disk, connection rings, as well as the location of magnetic and backup bearing centerlines, the location of motion sensors that monitor the rotor displacement, and the unbalance masses that are applied to excite the synchronous rotor unbalance response of the two rigid body and the first bending rotor modes.

In addition to the bearing stations, the FEA model also included discrete critical stations along the rotor, where rotor displacements were to be calculated.

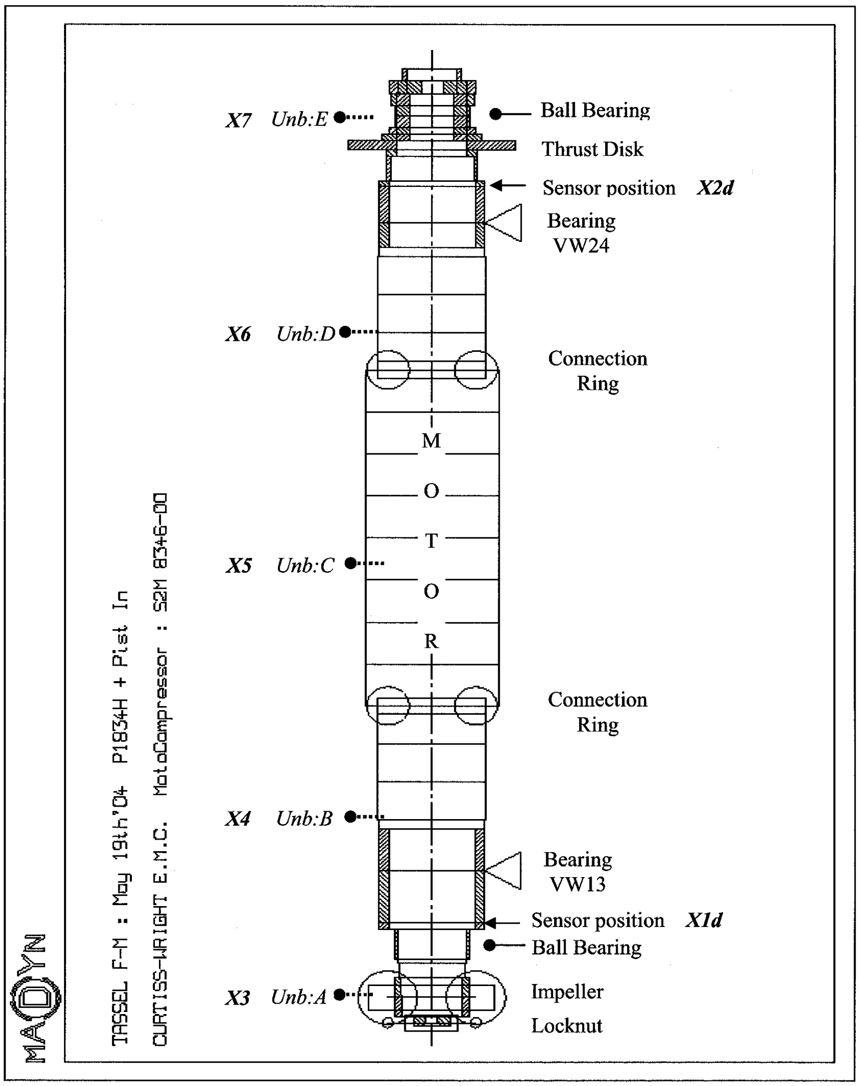

Figure 1. Rotor Configuration for FEA Model

3.0 Critical Speed Analysis

Figure 2 shows a plot of the first and second bending modes for the undamped free-free rotor configuration. The corresponding modal frequencies at 0-rpm rotor speed were predicted at about 220 Hz and 442 Hz, respectively.

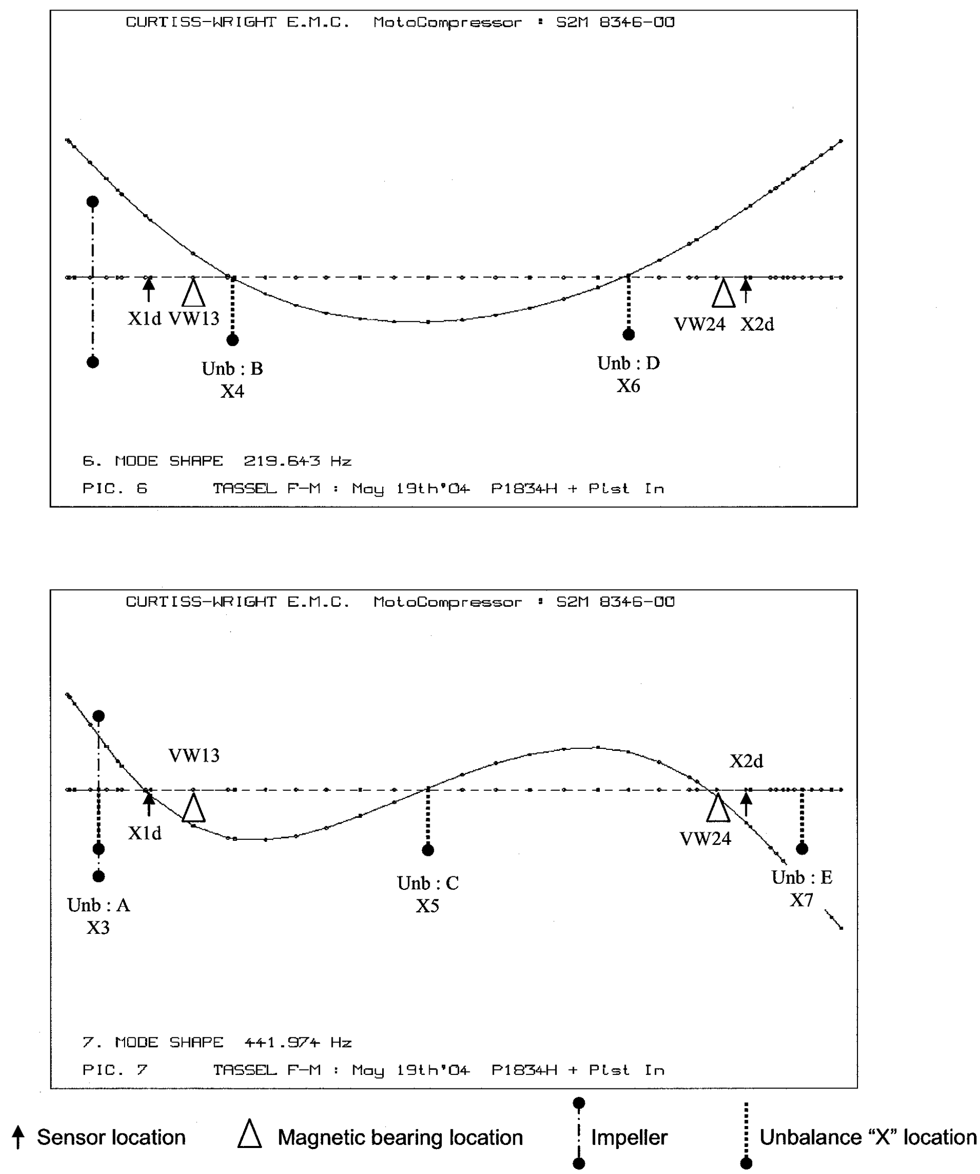

Figure 2 - Plots of Free-Free 1st and 2nd Bending Modes

E5

Figure 3 shows the gyroscopic effect of rotor speed at 12,000 rpm in splitting the bending modes into forward and backward whirl modes. As shown on Figure 2, the first forward mode was at 228.2 Hz and the backward mode at 210.8 Hz. Similarly, the second forward mode was at 462 Hz and 421 Hz.

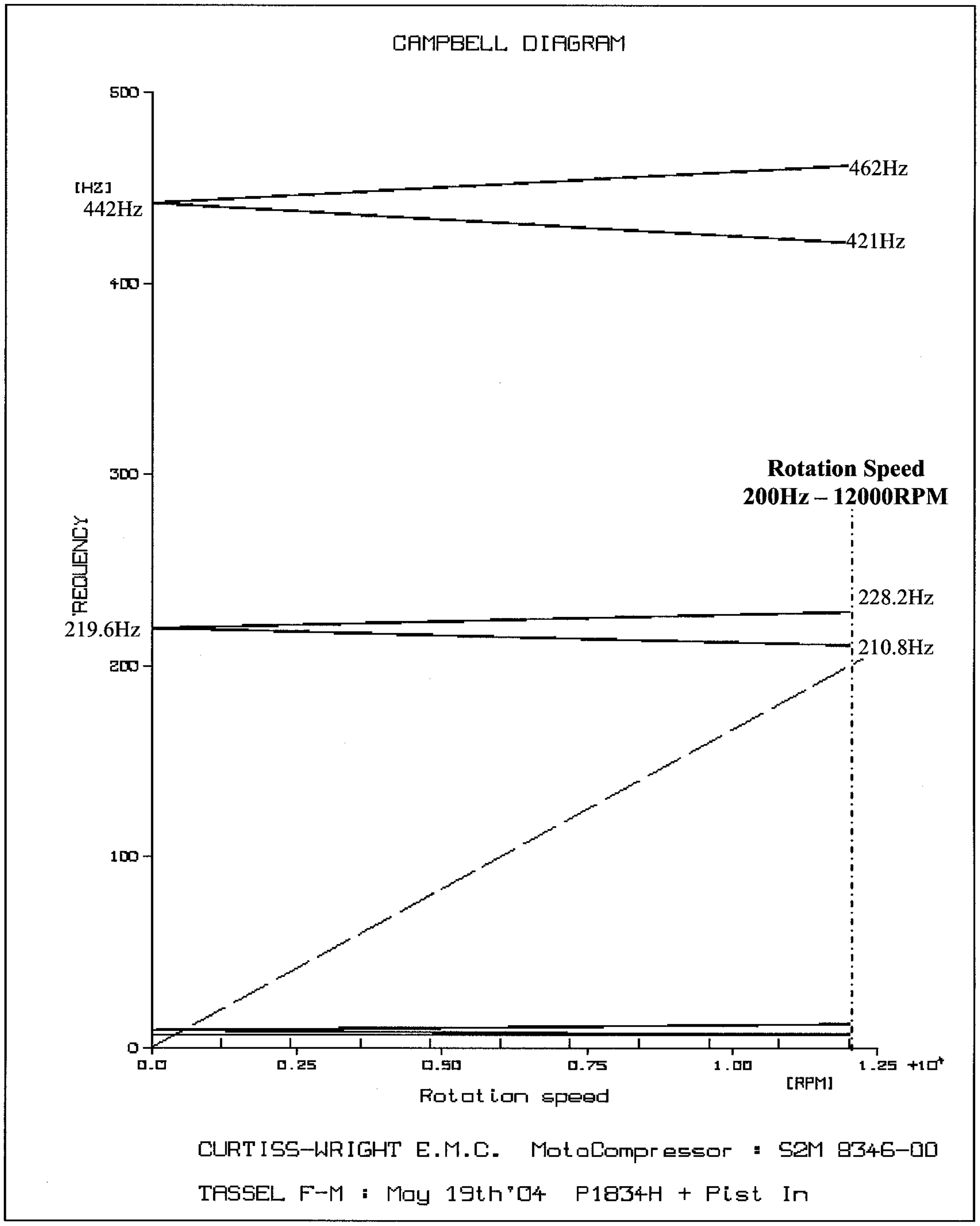

Figure 3 - Effect of Rotor Gyroscopic On Mode Split

Figure 4 shows the undamped critical speed map for the translation, conical, and first bending modes of the rotor. Plotted are rotor critical speeds as a function of the bearing stiffness. Superimposed on this plot are also the equivalent real stiffness of the impeller and thrust end magnetic bearings. The intersection of the bearing stiffness plots with the critical speed map identified the value of each of the rotor modes and the corresponding bearing stiffness. In this case, for example, the first forward rigid mode was predicted at 60 Hz with bearing stiffness of 462, 000 Lb/in. The second forward rigid body mode was at 90 Hz at a bearing stiffness of 571,000 Lb/in. And the first forward bending mode was at 246 Hz at a bearing stiffness of 1,220,000 Lb/in. This represented margin of 23% above the max operating speed of 200 Hz. However, as shown later this margin increased to 27% when the bearing damping was included exceeding the API requirement of 25% margin.

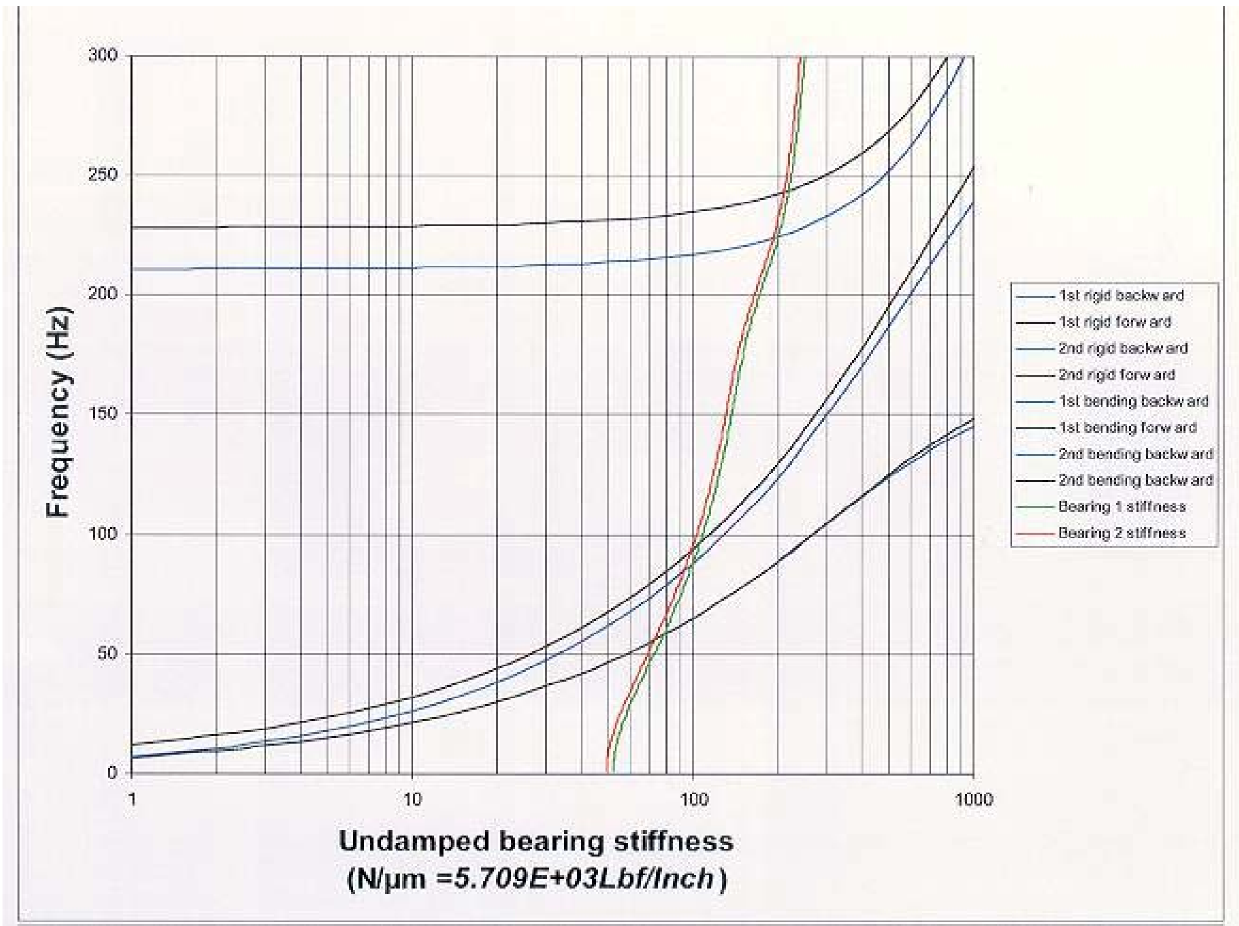

Figure 4 - Undamped Critical Speed Map

Figure 5 shows a plot of the calculated bearing stiffness at impeller, without the effect of the cross coupling, as a function of frequency. The minimum stiffness at the impeller at 27 Hz was predicted at 118,000 Lb/in. This is considered to be adequate to limit the compressor rotor excursions at low frequencies when the damping provided by the magnetic bearing system may not be sufficient to control rotor motion.

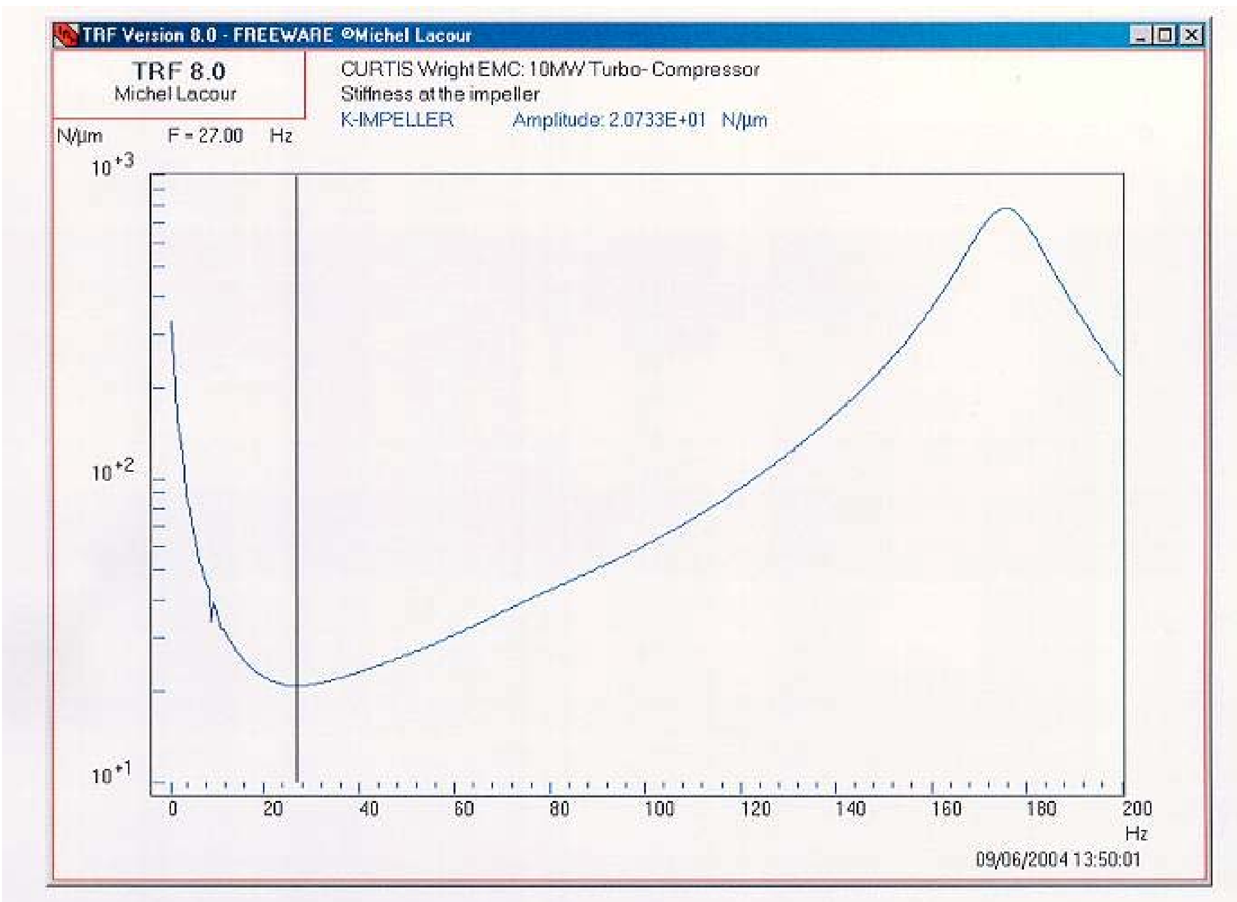

Figure 5 - Stiffness at Impeller VS Frequency Plot

The control type used to maintain rotor vibration within acceptable level is a Multi Input Multi Output (MIMO) type instead of the Single Input Single Output (SISO) type. The translation and conical rigid body modes were separated for control by two independent channels. The “parallel control” channel primarily controls the rotor first bending mode.

The rotor forward whirl rigid body modes, as shown in Table 1, are critically damped because the log decrement values are greater than 1, and therefore require no separation margins. Also the first bending mode at 254 Hz is 27% above the maximum continuous operating speed of 12,000 rpm, and satisfies the rotordynamic design requirements of the API standards.

Table 1

IEMDC Motor-Compressor Calculated Critical Speeds

E6

Mode	Frequency (Hz)	Damping Ratio	Log Decrement
Translation	68	0.393	2.67
Conical	111	0.482	3.45
First Bending	254	0.051	0.321
Second Bending	447	0.011	0.07

4.0 Unbalance Response

The rotor unbalance response was calculated for each of the three modes (two rigid body and first bending) according to the specified API unbalance levels. The 1X calculated API unbalance for this rotor configuration is 0.81 Oz-in, however, for rotor response calculations the unbalance applied was 4X API. The following three-unbalance cases, listed in Table 2, were analyzed to excite the translation, conical and bending modes, respectively.

Table 2

Magnitude and Location of Unbalance to Excite Three Rotor Modes

Unbalance Case	Unbalance Location, as marked on Figure 1	Magnitude (Oz-in)	Comments
1	At Plane B, IE of rotor shaft	1.63	Translation Mode
1	At plane D, TE of rotor shaft	1.63	Translation Mode
2	At plane A, Impeller	1.63	Conical Mode
2	At plane E, thrust disk	-1.63	Conical Mode
3	Plane A	0.81	1st bending
3	Plane C	-1.63	1st bending
3	Plane E	0.81	1st bending

Listed in Table 3 is the calculated synchronous rotor response for each of the three cases listed in Table 2. Also listed are the peak response speeds for each mode, which represents the damped resonance speeds for this rotor. The tabulated data also includes the effect of the AVR synchronous filter on rotor response and bearing loads.

Table 3
Predicted Rotor Response and bearing Loads at 200 Hz for 4X API Unbalance

Rotor Response Station	Translation 67 Hz		Conical 111 Hz		Bending 200 Hz		Comments
	Off	On	Off	On	Off	On	AVR Synchronous Filter Off or On
Sensor 1, XD1, mils	0.186	0.070	0.564	0.198	0.384	0.171	Impeller end sensor
Sensor 2, XD2, mils	0.131	0.068	-.551	-.210	-.359	-.280	TE sensor
Unbalance A, mils	0.187	0.072	0.693	0.215	0.421	0.102	Impeller at
Unbalance B, mils	0.187	0.069	0.408	0.162	0.311	0.205	At IE rotor shaft
Unbalance C, mils	0.186	0.070	0.020	0.015	0.024	0.062	Rotor mid-point
Unbalance D, mils	0.155	0.067	-.378	-.148	-.280	-.191	TE rotor shaft
Unbalance E, mils	0.124	0.068	-.614	-.228	-.378	-.300	Thrust disk
Bearing 1 Load, Lb.	121	7.2	393	21.8	379	23.2	IE Bearing
Bearing 2 Load, Lb.	93	6.7	364	24.1	355	39.3	Thrust End Bearing
Response Margins	2.68	7.18	0.89	2.52			Margins relative to 1 mil peak-to-peak for 4X API unbalance

4.1 Discussion of Vibration Response and Bearing Loads

Review of the vibration response and bearings loads tabulated in Table 4 clearly show the effectiveness of the AVR synchronous filter in reducing the rotor response and bearing loads. The predicted response has sufficient margin over the maximum 1 mil peak-to-peak response permitted by the API 541 and 617 standards at 12,000 rpm.

Figures 6, 7, and 8 show graphically the effect of the AVR synchronous filter on the predicted rotor vibration due to 4xAPI unbalance. The red color plot shows the response with the synchronous AVR filter system turned on and the blue with the filter turned off. The vibration level, shown in these plots, is below .001” peak-to-peak throughout the operating speed range. Thus, the design satisfies the unbalance response requirements as specified by API standards.

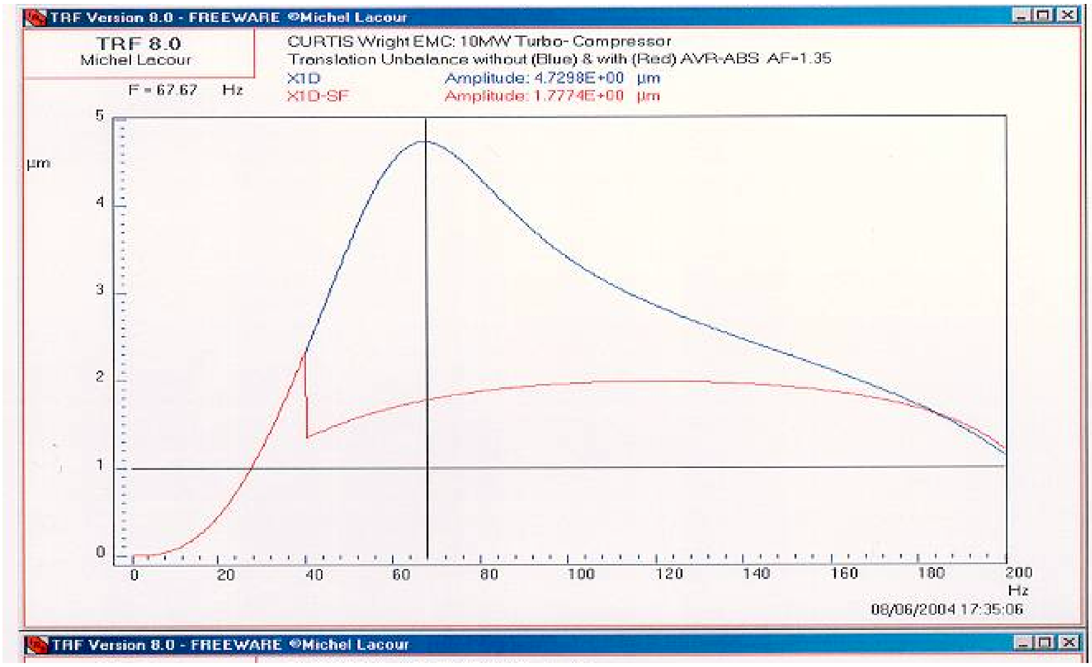

FIGURE 6 - Unbalance Response with and without Synchronous Filter (AVR)

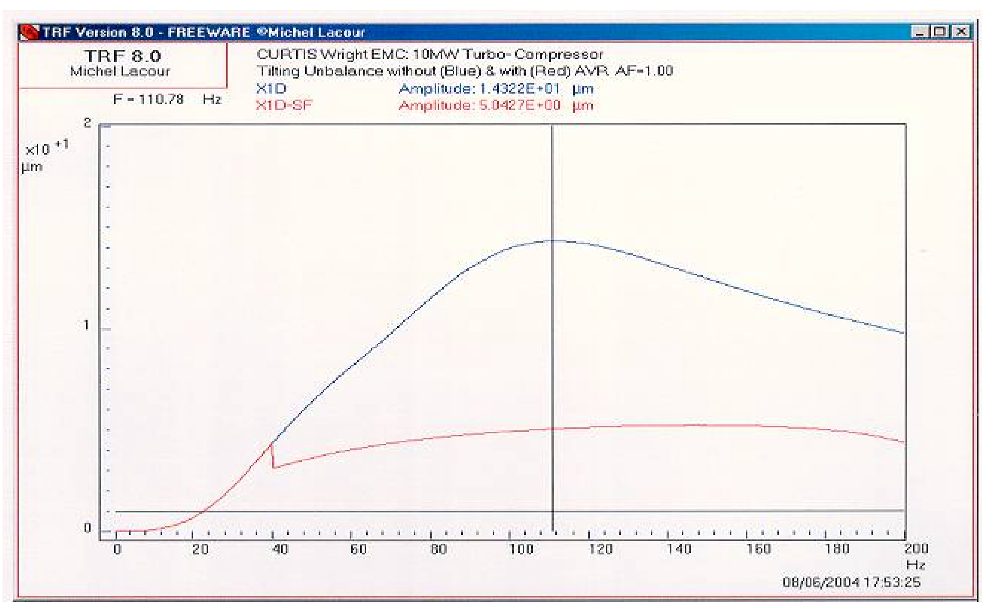

FIGURE 7 - Predicted Unbalance Response at 111 Hz with and without AVR Filter

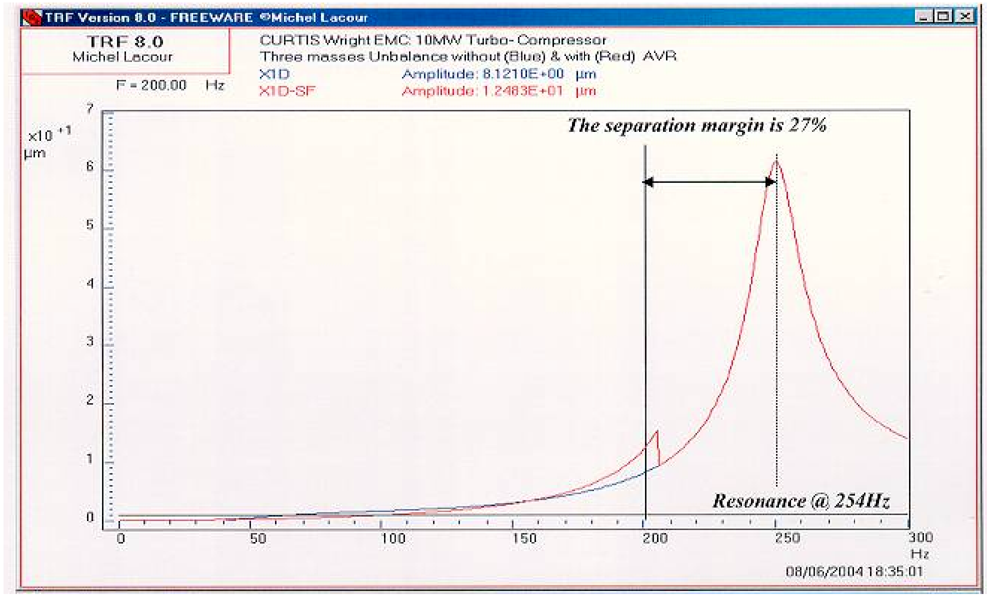

FIGURE 8 - Predicted Unbalance Response at 254Hz with and without AVR Filter

5.0 Conclusions

The rotor dynamics analysis results of the motor-compressor system that is levitated in the active magnetic bearings have clearly demonstrated that the design satisfies the rotor response and critical speed margins required by the API standards 541 and 617.

Appendix F
Motor Variable Frequency Drive specification

CURTISS-WRIGHT ELECTRO-MECHANICAL CORPORATION – EMD
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CHESWICK, PA 15024

DOE Deliverable Item #9 - Performance Specification for the Required Variable Frequency Drive

Prepared for Dresser-Rand Company
Under P.O. 51483
21-APR-03, Revised May 15, 2003

Prepared by:

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EMDC Project - Lead Engineer

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Generator/Motor Design & Technology

CURTISS-WRIGHT ELECTRO-MECHANICAL CORPORATION – EMD
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DOE ITEM #9 - TECHNICAL
SPECIFICATION FOR THE VARIABLE FREQUENCY DRIVE (VFD): INLINE ELECTRIC MOTOR DRIVEN COMPRESSOR (IEMDC)
EMD Document # IEMDC-CW-EMD-03-017

1.0 SCOPE

1.1 This specification covers the technical requirements for a medium voltage variable frequency drive (VFD) system for an induction motor application. The motor has been developed to operate as an integral driver for a compressor to transport natural gas through the pipeline.

1.2 The VFD system shall consist of all components required to meet the performance, protection, safety, testing, and certification criteria of this specification. These components may include incoming harmonic filter/power factor correction unit, input isolation transformer, VFD converter/DC-link/inverter, and output filter.

1.3 The VFD system must:

- Represent a fully integrated package.
- Include all material necessary to interconnect the VFD system elements, even if shipped separately.

1.4 The VFD system as defined (1.2, above) shall be completely factory pre-wired, assembled and then tested as a complete package by the VFD supplier, to assure a properly coordinated, fully integrated drive system.

2.0 APPLICABLE DOCUMENTS

2.1 IEEE STD 519-1992 or later – "IEEE Recommended Practices and Requirements for Harmonic Control in Power Systems"

2.2 ISO-9001 – "Quality Systems – Model for Quality Assurance in Design, Development, Production, Installation, and Service." 1987 or later

3.0 BASIS OF PURCHASE

3.1 Basic Specification Information: The following information is provided to the VFD supplier as design guidelines for the VFD. Refer to Table 3.1.7 for a summation of performance for the motor at rated conditions.

3.1.1 Input Voltage: The input voltage to the VFD will be 13.8 kV. The VFD shall be capable of continuous operation from +10%/–5% voltage at rated conditions and +10%/-30% at reduced power.

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3.1.2 Input Frequency: 60 Hz +5%

3.1.3 Output Frequency: The frequency range of continuous operation is from 133 to 200 Hz with rated frequency being 200 Hz.

3.1.4 Output Voltage Range: The motor is designed to operate at a line-to-line voltage of 6.9 kV. The VFD shall be capable of producing a variable ac voltage/frequency output to provide continuous operation over the normal system 65-100% speed range with 100% being 6.9 kV/200 Hz.

3.1.5 The VFD shall provide soft start protection in order to prevent overheating of the rotor.

3.1.6 The VFD shall provide some means of braking. Because the IEMDC rotor is supported on magnetic bearings, a means of braking is necessary in the event of bearing failure. The VFD supplier shall provide a brief description of the proposed means of braking and a method for quantifying its effectiveness. This information is to be provided at the time of the quote.

3.1.7 Expected motor performance values at rated load is given in table 3.1.7.

Table 3.1.7

Expected Motor Performance at Rated Load

Motor Terminal Voltage (V ac)	6900	Rated Current (amps)	1117
Frequency (Hz)	200	Power Factor (%)	78.8
Poles	2	Efficiency (%)	94.9
Rated Horsepower	13,400	Input Power (kW)	10,523

3.1.8 The VFD shall be capable of a momentary overload of 120% of rated motor current for one minute out of any ten minutes.

3.1.9 The VFD shall meet or exceed IEEE 519 Standard, section 10, for the suppression of harmonics, voltage distortion, and VFD transients transmitted into the power supply system. If the VFD cannot meet the requirements of IEEE 519, a filtering system shall be proposed by the supplier to meet the requirements. In this case, the supplier shall include a complete technical specification for the filter (e.g., electrical characteristics, dimensions, weight, heat dissipation, etc.). The supply of this filter along with proof of testing shall be included in the quote, if necessary.

3.1.9.1 Variable Frequency electrical system harmonics: Total Harmonic distortion (THD) up to 10.2 kHz (or higher if recommended by the supplier) for the line to line voltage and phase current at the output of the VFD shall not exceed 5%. This condition applies between 65% and 100% of rated speed regardless of load. If necessary, an output filter shall be provided to meet this requirement. A complete technical description for the filter (e.g., electrical characteristics, dimensions, weight, heat dissipation, etc.) shall be supplied.

3.1.10 The VFD supplier shall provide guaranteed efficiency and power factor information for the operating conditions indicated in table 3.1.10. This information is to be supplied prior to the award of order. Efficiency values shall include losses of input transformer, inverter, and output filter.

Table 3.1.10

Motor Operating points for Efficiency and Power Factor Data

Operating Points	Alt-1	Alt-13	Alt-19	Alt-20	Alt-21
Terminal Voltage (V ac)	4636.8	6934.5	6361.8	5785.6	5202.6
Frequency (Hz)	134.4	201	184.4	167.7	150.8
Shaft Horsepower	8910	13,441	12,247	11,209	8267
Line Current (amps)	1088	1115	1101	1105	900
P.F. (%)	79.1	78.8	79.0	78.9	79.5

Note: The VFD *system* efficiency is as follows:

$$\eta_{sys} = \eta_{VFD} \times \eta_{xfmr} \times \eta_{pfc} \times \eta_{harm} \times \eta_{filter}$$

Converter/Inverter (VFD)	η_{VFD}
Input Transformer	η_{xfmr}
Power Factor Correction	η_{pfc}
Input Harmonic Filter	η_{harm}
Output Filter	η_{filt}

3.1.11 Design Life: The system design life is 20 years or better.

3.2 Additional Requirements and Features

3.2.1 Operational Environment: The VFD system shall be capable of continuous operation in an average ambient temperature between 0°C and 40°C.

3.2.1.1 Enclosure Requirements: All VFD system components shall be mounted and wired by the VFD system manufacturer in a grounded enclosure. Enclosures for liquid-cooled units are to be NEMA 12 Non-Ventilated.

3.2.1.2 The VFD supplier shall provide information, with the proposal, on the heat generated from the VFD (and filters, if they exist) for rated conditions. (Refer to Table 3.1.7.)

3.2.2 VFD supplier shall provide all necessary interface information (power and access locations, cooling system details, etc.) to EMD for use in specifying the environmental housing. This shall include dimensioned drawings and weights for all components.

3.2.3 The VFD automatic shutdown features: The unit shall, at a minimum, protect against damage and incorporate automatic shutdown capability from the faults and overload conditions as displayed in Table 3.2.3. Automatic shutdown limits will be established during initial testing.

Table 3.2.3 VFD Automatic Shutdown

a) Line-to-Ground Faults[1]	g) Motor Locked Rotor
b) Line-to-Line Faults[1]	h) Low Speed Stall
c) Over Voltage[1]	i) Loss of Power[1]
d) Under Voltage (output of VFD)	j) Over Temperature of VFD
e) Loss of Phase[1]	j) Over Temperature of Motor
f) Over Current[1]	

(1)- These shutdown limits apply to both the input and output of the VFD

3.2.4 Shutdown Event Data Storage: As a minimum the VFD shall have the capability to record the parameters listed in Table 3.2.3 such that they can be retrieved after a shutdown event has occurred. This data shall be easily retrievable at the VFD.

3.2.5 Front Panel Display Monitor: As a minimum the VFD shall have the parameters displayed on the front panel as defined in table 3.2.5.

Table 3.2.5 VFD/Motor Front Panel Display	
a) Input Voltage	d) Output Frequency
b) Input Current	e) Output Voltage
c) Input Frequency	f) Output Current
d) Motor Speed	g) Total 3- Phase kW Output

3.2.6 At the time of the quote the VFD supplier shall provide details of the proposed programming interface.

3.2.7 Control System Integration: The VFD shall be capable of direct communication to an IBM compatible computer for serial link setup of parameters, fault diagnostics, trending and diagnostic log downloading. An RS-232 port shall be door-mounted for computer or printer interface. VFD parameters, fault log and diagnostic log shall be downloadable for hard copy printout via the RS-232 or Ethernet port and a standard serial printer. The VFD shall be provided with single port digital communication capability to allow direct control and status communication with a PLC, SCADA or other control system. The end user shall determine the communication protocol. Additional analog input and output signals and additional digital inputs and outputs shall be provided.

3.2.8 The VFD shall be equipped with an incoming load disconnect switch. This switch shall have the capability to isolate the VFD from the system power supply.

3.2.9 Current transformers (CT's) and potential transformers (PT's) shall be provided at the input and output of the VFD to monitor voltage and current waveforms. These CT's and PT's shall be provided with necessary hardware so that they may be terminated in the appropriate manner when not in use.

3.3 Test and Reliability Verification: EMD retains the option to witness the factory acceptance tests (FAT) of the VFD. The VFD supplier shall notify EMD 4 weeks prior to the start of the FAT so arrangements can be made to witness. The VFD supplier shall provide the FAT procedure 12 weeks after order placement to EMD for comment and approval.

3.3.1 On-Site Start-up and Testing: The test objective is to demonstrate the unit compliance with the IEEE 519 Standard, with the design specifications, and to check the unit functions to ensure proper operation.

3.3.1.1 Compliance shall be verified by the VFD supplier with field measurements of harmonic distortion differences at point of common coupling with and without VFD operating. The point of common coupling (PCC) for all harmonic calculations and field measurements for both voltage and current distortion shall be defined as the primary connection of the VFD input transformer.

3.3.1.2 The VFD supplier shall be responsible for the time to follow, tune and enhance the VFD.

3.3.1.3 The VFD supplier shall be responsible for taking current and voltage waveform harmonic distortion measurements up to 200 Hz. All equipment to take these measurements shall be provided by the VFD supplier. Single-phase measurements shall be made at the VFD input and output. Total harmonic distortion is not expected to vary phase to phase, however, limited measurement shall be taken on adjacent phases to compare waveforms to verify this expectation.

3.3.1.4 The VFD supplier shall provide a short report or summary documenting the harmonic spectrum and the VFD's level of compliance with the total harmonic distortion requirements given in section 3.1.9.

4.0 QUALITY ASSURANCE REQUIREMENTS

4.1 General Requirements

4.1.1 The supplier shall maintain a quality control system which meets the intent of ISO-9001. Should the supplier's quality system be qualified to another program, a copy shall be provided to EMD for approval.

4.1.2 The supplier shall provide access to supplier's facilities for witness of the VFD electrical tests by EMD personnel (refer to paragraph 3.3). In addition, the supplier shall provide access to the supplier's facilities for source inspection by EMD employees.

4.1.3 All documents generated and submitted to EMD shall be traceable to this specification and revision, purchase order, and must be legible and machine reproducible.

4.1.4 Deviations from the requirements of this specification shall be reported in writing to the EMD Purchasing contact.

4.2 Documentation Requirements

4.2.1 Document Submittal Requirements with the Proposal: The supplier shall provide the following drawings and information, in preliminary stages if necessary, with the proposal.

4.2.1.1 The supplier shall provide a 3 (or more) view outline drawing of the proposed equipment. The drawings shall specify the overall dimensions, weight and center of gravity of the assembly (or the constituent parts of the assembly in the event that it is supplied as modules) and the minimum space requirements for maintenance purpose (showing the dimensions at the front side, rear side, lateral side and over the VFD). The drawings will indicate the location of important interface features as listed in Table 4.2.1.1.

Table 4.2.1.1 VFD Outline Drawing Interface Locations
a) The location of connections for cooling water
b) The location of input and output power connections
c) The location of control wiring
d) The location of access panels for maintenance, including door swings etc.
e) The drawings should be sufficiently detailed so that HVAC, cooling water and structural support designs for the environmental housing can be prepared

4.2.1.2 Schematic drawings of the VFD shall be supplied showing rectifiers, inverters, inductors, capacitors, transformers, filters, etc., and specifications for all other main components.

4.2.1.3 The supplier shall provide a connection drawing(s) specifying the arrangement of input and output power, grounding and interconnections. Drawings specifying control, instrument and communications connections shall be supplied.

4.2.1.4 The supplier shall provide EMD with the Mean Time Between Failures (MTBF) for this voltage level and for the proposed type of VFD. In addition, the Mean Time For Repair (MTFR) shall also be provided. If inadequate information is available to calculate these parameters for this type and voltage level of VFD, reliability data for other similar VFD types may be utilized with EMD concurrence.

4.2.1.5 The supplier shall provide information related to the filtering system per 3.1.9, and 3.1.9.1, if applicable.

4.2.1.6 The supplier shall provide heat generation per 3.2.1.2.

4.2.1.7 The supplier shall provide information per 3.2.6.

4.2.2 Document Submittal Requirements for the Design/Construction: The VFD supplier shall provide the following drawings, interface specification and list of components and consumables, within 20 weeks of receipt of an order. In addition, the VFD manufacturer and EMD shall hold a technical interface meeting shortly after receipt of this information.

4.2.2.1 Outline drawings - The supplier shall provide a 3 (or more) view outline drawings of the supplied equipment. The drawing shall specify the overall dimensions, weight and center of gravity of the assembly (or the constituent parts of the assembly in the event that it is supplied as modules). The drawing will indicate the location of important interface features as listed in Table 4.2.1.1.

4.2.2.2 Schematic drawings of the VFD shall be supplied showing rectifiers, inverters, inductors, capacitors, transformers, filters, etc., and specifications for all other main components.

4.2.2.3 The supplier shall provide a connection drawing(s) specifying the arrangement of input and output power, grounding and interconnections. Drawings specifying control, instrument and communications connections shall be supplied.

4.2.2.4 The supplier shall provide EMD with the Mean Time Between Failures (MTBF) for this voltage level and type of VFD. In addition, the Mean Time For Repair (MTFR) shall also be provided. If inadequate information is available to calculate these parameters for this type and voltage level of VFD, reliability data for other similar VFD types may be utilized with EMD concurrence. EMD expects these values to be updated, if appropriate, from the proposal submittal. In addition, back-up information to support the MTBF and MTFR values shall be provided.

4.2.2.5 The supplier shall provide information related to the filtering system per 3.1.9 and 3.1.9.1, if applicable.

4.2.2.6 The supplier shall provide heat generation per 3.2.1.2.

4.2.2.7 The supplier shall provide information per 3.2.6.

4.2.2.8 The specified drawings must be provided in both paper and electronic format. Lists and operating procedure shall be supplied in Microsoft Word format. Drawings shall be supplied in one of the following formats; Autocad, IGES, Ideas or DFX format.

4.2.2.9 The drawings specified above shall not be supplied as proprietary and acceptance of the order conveys to EMD the ability to reproduce and include this material in communications to customers, technical publications, publicity and inclusion of the material in the overall system instruction books.

4.2.2.10 The supplier may maintain as proprietary other drawings and communications with EMD as they deem necessary for the proper specification and supply of this equipment. These materials should be appropriately marked at the time of submittal so that EMD can exercise the appropriate caution in storing and handling this material.

4.2.3 Document Submittal With Shipment

4.2.3.1 The VFD general specifications shall contain, as a minimum, a detailed explanation of the main characteristics, modulation type, control type, signaling for failures, grounding scheme for both the input and output sides, percentage limits of voltage, current, frequency and all other limited parameter variations. The supplier shall also include any behavior or operational differences that occur at different frequencies.

4.2.3.2 By the completion of the order the supplier shall provide an instruction manual detailing normal and emergency operation, control and interface requirements. The instruction manual shall include requirements for periodic maintenance and any replacement spare parts (fuses, cooling vent screens etc., required for 20 years of operation). The instruction manual should provide sufficient trouble shooting instructions to support normal breakdowns and repairs. The supplier shall provide as built drawings at the time of shipment.

Appendix G
Estimated Cost and Schedule – IEMDC Manufacture and Qualification

Scope

The estimate for the costs to manufacture and qualify the IEMDC system was developed during the course of the project. Manufacturing and qualifying a prototype unit was not a requirement of the project. The elements required for the manufacture and qualification of an IEMDC prototype unit developed under this award are shown below. It is expected that the manufacture and qualification of a prototype unit would be executed concurrently.

Manufacture

- development of detailed manufacturing drawings,
- manufacture of the compressor section components,
- manufacture of the motor and associated components,
- procure magnetic bearings and magnetic bearing control panel,
- procure VFD and associated components,
- procure gas conditioning system,
- procure system control panel,
- tooling,
- unit assembly (compressor section with motor).

Qualification Program

1. **Compressor module PTC type 2 inert gas test**

   This test is an independent validation test of the compressor section of the IEMDC. It requires designing and manufacturing special components to perform the testing. Special rig components that are required for this test include a shaft, bearings, seal(s), bearing housings, and a head.

   - Develop test rig components for a separate PTC type 2 inert gas test,
   - Manufacture test rig components,
   - Perform an inert PTC Type 2 gas test on the compressor test rig.

2. **PTC Type 2 Inert Gas Test**

   System performance evaluation of the IEMDC unit assembly (compressor with motor) prior to performing a hydrocarbon test.

3. **Hydrocarbon Test – PTC Type 1 Test**

   System validation test of the IEMDC using hydrocarbon gas as the test medium.

Costs

The estimated cost to manufacture and perform qualification is 5.61 million US dollars. Cost is dependent upon the actual initiation date of the project and may differ from the currently estimated cost.

Schedule

The estimated development schedule is shown in Figure G1.

Item #	Task Name	Duration
1	**Phase 2 - IEMDC Manufacture**	410 days
2	Detail Manufacturing Drawings	16 wks
3	Long Lead Materials	20 wks
4	Motor Rotor Machining	10 wks
5	Motor Rotor Assy and Balance	6 wks
6	Motor Stator Stacking and Winding	14 wks
7	Magnetic Bearing Procurement	48 wks
8	Motor Assy and Factory Test	8 wks
9	Procure Variable Frequency Drive	52 wks
10	Compressor Internals	58 wks
11	Compressor Casing	67 wks
12	Procure System Controls	40 wks
13		
14	**Phase 3 - IEMDC System Test**	347 days
15	IEMDC Test Planning	8 wks
16	IEMDC Test Loop Set-up	20 wks
17	IEMDC System Assembly	6 wks
18	IEMDC System Testing	14 wks

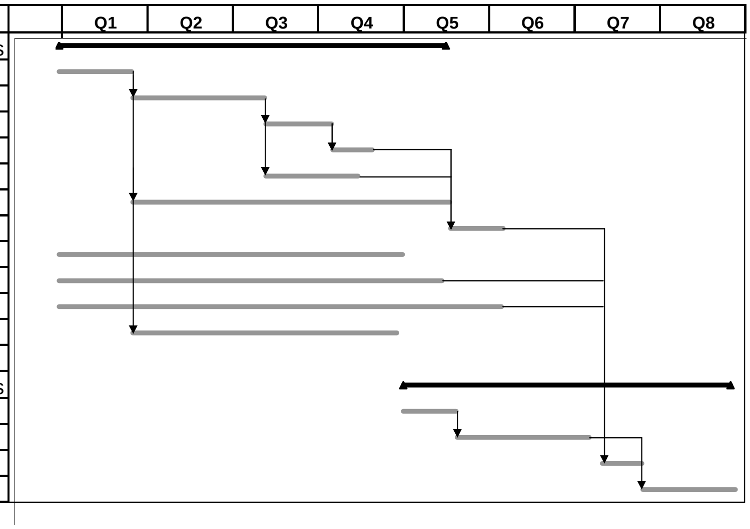

Figure G1. Estimated Development Schedule for Manufacture and Qualification.

Appendix H
Technology Assessment Paper

IEMDC: Inline Electric Motor Driven Compressor
Technology Assessment Paper

Award #: DE-FC26-02NT41643

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Abstract
This paper provides an assessment of the current compressor equipment technology available to the gas industry. Various compression system technologies for the compressor and prime mover are discussed including the benefits and limitations of these technologies. Additionally, the reader should keep in mind the possibility of combining the best available benefits of compressor and driver technologies and integrating them into a common unit. Such a driver-compressor system could combine the many advantages of the different technologies currently available. One such combined system is the integrated motor-compressor design presented by the Inline Electric Motor Driven Compressor (IEMDC) concept. The IEMDC concept is a single stage centrifugal compressor overhung from an electric motor shaft. It is an electrically powered, highly flexible, efficient, and low total cost compressor that can be quickly ramped up on the pipeline to supply gas during times of peak demands. In addition to the low cost, the proposed IEMDC design will also mitigate critical gas industry concerns regarding environmental, regulatory, and maintenance requirements that are associated with fossil fuel powered compression equipment.

Introduction
There are many factors to consider when selecting gas compression equipment. The choice of drive technology is as important as the selection of the compressor technology when sourcing new compression equipment. Proper consideration needs to be given to the matching of the drive technology to the compressor technology to achieve the full benefits of the gas compression equipment train. New technologies in the areas of control systems, emissions control, and variable frequency drives provide an array of available driver choices. These choices compel the user into performing a system evaluation of the equipment not only with regard to the proper matching of the equipment to site operating conditions, but to other factors such as environmental requirements, considerations for operational oversight, and overall total cost to install and operate the equipment.

Compressor technology for main gas transmission service currently focuses on reciprocating and centrifugal compressors. Reciprocating compressors comprise the majority of the installed compression horsepower in gas transmission service. This is primarily due to the fact that the pipeline infrastructure within the United States was designed around reciprocating compressors and engines at a time when there were fewer compressor and driver choices for the service. Integral engine compressors comprise the bulk of these reciprocating compressors, which have high horsepower and high volume flow capability. However, in the last several decades, centrifugal compressors have found considerable use in gas transmission applications. Centrifugal compressors are well suited for high horsepower applications in the gas transmission industry particularly with the current selection of available driver technology. Today there is a significant installed base of gas turbine driven centrifugal compressor packages in pipeline service.

Drivers powered by both gas fuel and electricity are currently being used to drive compression equipment in the gas transmission industry. [1-6] Reciprocating engines and gas turbine drives comprise the majority of all installed gas compression and transmission equipment. However, the use of electric motor driven compression in gas transmission service has significantly increased. This has been facilitated, to a large extent, by the many technological advances in variable frequency drive (VFD) converters as well as due to the economic advantages, operating flexibility and reliability of electric drives. Advances in areas of improved mechanical design, control system technology, monitoring capability, environmental compliance, and efficiency are continually being made because of the pipeline operator’s needs for different technologies to manage supply and demand requirements.

Assessment of Compressor Technology for Gas Compression
Integral engine compressors comprise the majority of all reciprocating compression equipment in service. The design combines reciprocating engine and compressor technology into a single unit. These units are low-speed, have high volume capability, and provide good operational flexibility. The U.S. pipeline infrastructure is made up of a significant number of these units that are still in operation. An advantage of these units is that pipeline stations can be placed far apart as a result of the high pressure ratio capability.

However these units are limited in production and are expensive to maintain due to failures and poor availability of spare parts. Other issues with integral engines include reduced efficiency, low power density, and more emissions than modern drivers because technological development essentially stopped several decades ago while other areas of driver technology advanced. Limited improvements in performance and life can be achieved by controlling the fuel air ratio and better monitoring, however, the long term trend is that these units will be replaced by more efficient modern drivers.

Separable gas compressors have essentially replaced integral engine compressors in the reciprocating compressor market. A separable compressor is a reciprocating compressor without the integral driver. These units have excellent operational flexibility and high efficiency. Modern units are high speed, which increases the power density allowing the unit to move high flow volumes with fewer machines than integral engine compressors. These units can be either gas engine driven or electric motor driven.

Centrifugal compressors provide a good match for gas transmission applications, which require high horsepower. A significant advantage of centrifugal compressors is the ability to provide greater volume capacity for a given footprint than a reciprocating compressor. There is also technology available to minimize methane emissions and noise from these compressors. Gas seal technology significantly reduces methane leakage and noise attenuation technology is available that can provide significant noise reductions as required by the site. Another advantage is that centrifugal compressors are free of the vibration problems that are typically experienced by reciprocating compressors due to inherent pulsations. Additionally, centrifugal compressors have a lower installed and maintenance cost, and have greater reliability than reciprocating compressors.

Assessment of Driver Technology for Gas Compression

Electric motors used in the gas industry are variable or fixed speed paired with either a reciprocating or a centrifugal compressor. These motors are typically stand-alone as opposed to being directly integrated with the compressor. Motors may be direct connected to a compressor using a coupling or connected with an intermediate gearbox to increase speed. Direct coupled electric motor arrangements are often used with small, low-speed reciprocating engines. These applications may use either constant speed or variable speed electric motor drives. Variable speed electric motors used in centrifugal compressor applications are low speed, usually less than 3,600 rpm and require a gearbox.

Reciprocating engines have historically been used to drive reciprocating compressors either as a separate unit or as an integral part of the reciprocating compressor. However, with recent advances in technology that improves efficiency and reduces emissions, reciprocating engines are currently being considered as drivers for centrifugal compressors. A recent publication (10) discusses the viability of driving a centrifugal compressor with a four-stroke lean burn reciprocating engine in conjunction with a Vorecon Mechanical Variable Speed Drive (MVSD) that utilizes a planetary gear design. These advanced high-speed (up to 1,000 rpm), lean burning engines are capable of delivering over 8,000 BHP. Additionally, these engines operate at a constant higher speed that results in an increased power density and allows operators to move larger gas volumes with fewer machines.

Mechanical reliability is an important consideration when selecting a prime mover for compression equipment. The total cost to operate the driver such as maintenance costs and staffing requirements factor into determining the most cost effective compression equipment solution. A duty cycle requiring frequent starts and stops can have a significant impact on gas fueled equipment integrity, reliability, and operational and maintenance costs resulting in decreased availability of the equipment. This is a significant consideration in driver selection knowing that reciprocating engines and gas turbines are mechanically complex. On the other hand, electric motors have fewer moving parts resulting in an inherently more reliable system than a comparable mechanical drive arrangement. Moreover, variable speed electric motor drives, with proven field reliability offer significantly better operating efficiency and flexibility.

A distinct advantage of gas-fueled prime movers over electric drives is that they are better suited for gas pumping station sites that are located away from the electric power transmission grid. Gas fueled equipment receives its fuel directly from the pipeline so there is no need to acquire fuel from an alternate source. Electric motor driven units require electric power and if it is not readily available there is a time and cost associated with extending transmission lines to the station site. Though from a reliability point of view, electric motor drives are often a better choice due to the inherently lower costs of owning and operating the units.

A major disadvantage of gas-fueled equipment is that the fuel used is directly received from the pipeline and thereby effectively reduces the capacity of the pipeline. It also has an adverse impact on the life of U.S. Gas Reserves. Furthermore, the environmental site impact of gas combustion can create emissions that result in increased installation costs and increased time for the construction and installation of the required abatement equipment. Permitting and air emissions requirements often result in equipment operating restrictions. New and existing installations are continually affected by changing emissions regulations and requirements. Capital investments in equipment are often necessary to achieve environmental compliance to changing regulations. Conversely, electric power has no site emissions and is therefore more environmentally friendly than gas fueled equipment. It is a far better solution, particularly for installations located in environmentally sensitive areas. This includes noise since electric power equipment typically has lower noise emission levels than gas fueled drives.

Application Considerations and Technology

While the capability of the compressor and driver is important when choosing equipment, there are additional factors that should be considered during the selection process. Factors such as emissions constraints, operating and maintenance costs, equipment footprint requirements, and construction costs contribute significantly to the overall cost of installing and operating a compressor station. Therefore consideration should be given to the technology that can be incorporated into the design and installation of the compression equipment to alleviate these concerns.

Environmental concerns are of critical importance when selecting a compressor. Seal leakage results in methane emissions and is a concern for most currently installed compression equipment. This is an issue with both reciprocating and centrifugal compressors. Shaft seals and the associated seal systems increase expense, increase maintenance requirements, and introduce a reliability concern. Centrifugal compressors with shaft seals can cause gas emissions when seals leak or when the gas is flared during venting and purging. However, the seal leakage in centrifugal compressors is significantly reduced with the use of modern gas seal technology. Gas leakage in reciprocating compressors is also a concern and a source of emissions. One of the most significant sources of emissions at natural gas compressor stations is the methane gas leaking from the rod packing of reciprocating compressors. Technology that can help to reduce or eliminate emissions from seal leakage will help to greatly reduce environmental concerns.

Gas fired engine driven compressors are becoming increasingly difficult to install. [7] Environmental restrictions have tightened, making permitting difficult. [7] The result is increased costs for safety and compliance with environmental regulatory requirements. [7] Additionally, Clean Air Act requirements for compression facilities make siting and new capacity additions difficult. [5] On the other hand, advancements are being made in the areas of emissions reductions in gas-fired equipment. Advancements in control technology have resulted in lean burning reciprocating gas engines with lower NOx emissions and higher efficiency. New power augmentation technologies such as air injection for gas turbine drivers can help reduce NOx emissions and increase power output from gas turbines. However, gas fired drivers require significant amounts of auxiliary equipment to reduce emissions which increases the overall installation and operating costs. [7]

Magnetic bearing technology is field proven reliable technology that can eliminate the issues and risks associated with using lubricating oil. Magnetic bearing technology eliminates the need for auxiliary equipment such as oil pumps, oil coolers, oil reservoir, oil reservoir heaters, oil filters, pressure switches and control valves, temperature switches, etc. Elimination of oil lubrication further enhances system reliability by eliminating the potential for shutdowns caused by the lubricating system itself. The lubrication oil requirement of conventional bearing technology creates the potential for oil leaks, fires, and increases the associated environmental risks of an oil spill. There are also handling issues and the associated costs of safe disposal. Incorporating technology such as field-proven magnetic bearings eliminates the issues and risks associated with using lubricating oil. Remote operation of compressor stations is also easier with magnetic bearings and without an auxiliary lube oil system. The overall size and footprint of the equipment has an effect on the handling, transportation, and installation cost as well as public nuisance issues and security of the facility. Compact designs create the opportunity to install boosting stations within existing rights-of-ways, the potential to install the unit in an underground bunker that eliminates booster stations from the public sight, and the potential to significantly reduce emitted noise. The elimination of public nuisance issues and reduced installation and operating costs enable more efficient and effective location of boosting system, and provides greater operational flexibility and throughput. Additionally, a compact equipment design has the potential to be installed underground or in a concrete vault offering increased security by eliminating the necessity of bringing the pipeline to the surface for boosting. The result is increased pipeline concealment, and reduced opportunities for unauthorized access.

A table of available compression equipment choices is given in table 1. Listed in the table are benefits and limitations of various compression equipment trains.

Summary

The installation of new compressor pipeline stations and the upgrading of existing stations rely heavily on the economics associated with such projects. The selection of the proper compression equipment that is compatible with the requirements of the pipeline operator can help greatly to lower the total life cycle costs. Furthermore, technology solutions that can address all of the requirements for an efficient, reliable, cost effective compression system will provide the maximum benefit to the gas transmission industry. Compression equipment and systems that can provide the key technological advantages listed below will provide the most economic and cost effective solution for pipeline operators and for the enhancement of the current pipeline infrastructure.

- Demonstrates high efficiency, reliability, ease of installation, and operational flexibility as compared to current systems,
- Reduces total life cycle costs and has the lowest per unit HP cost,
- Allows additional pipeline stations to be installed within the existing infrastructure ,
- Provide greater operating flexibility than integral engine compressor or separate electric drive systems,
- Provide greater reliability than mechanical drive systems,
- Minimizes environmental impact on sensitive station sites,
- Reduces emissions whether from combustion of fuel or methane leakage,
- Offers attractive attributes to replace older compressors and for new site installations.

Table 1. Overview: General Benefits and Limitations of Compression Equipment Trains

Compression Equipment (Compressor & Drive Technology)	Benefits	Limitations
Integral Engine Reciprocating Compressor (Power and compression cylinders are integrated on the same crankshaft.)	Good operational flexibility. Less sensitive to changes in gas composition and density. High volume. High horsepower capability. Simple installation as a result of single piece construction. High operating pressure ratios.	Fuel Efficiency (inefficient). Low Power Density. Emissions. Old Technology. Limited spare parts availability. Noise and vibration. High Maintenance costs. Reliability issues.
Gas Engine Driven / Separable Reciprocating Engine	Higher speed than integral engines. Increased power density over integral engines. Excellent operational flexibility. High fuel efficiency. Less sensitive to changes in gas composition and density. High operating pressure ratios.	Emissions. Noise and vibration as a result of pulsation. Gas engine driver power limitations. Maintenance costs. Large footprint of gas engine driver.
Gas Engine Driven Centrifugal Compressor	Alternative to motor or gas turbine. High fuel efficiency Large operational envelope. Small impact of low air density applications.	Requires a gearbox or mechanical variable speed drive (MVSD). Design is mechanically complex. Size and space requirements. Noise and vibration.
Gas Turbine Driven Centrifugal	High Power Density. High output powers. Low emissions with proper abatement equipment. High Availability. Continual technological advancements. Proven technology	Reduced performance at high altitude and high temperature. Limited range of efficient operation. Gas turbine is mechanically complex requiring frequent parts replacement expensive maintenance. Emissions. High initial capital cost.
Constant Speed Direct Drive Centrifugal Compressor.	High availability. Good for constant throughput applications. Low cost.	Limited operating flexibility can not react to changing demand. Limited applications.
Constant Speed Electric Motor Centrifugal Compressor w/ a Multi-Stage Variable Speed Drive (MVSD).	Good operational flexibility. High reliability. Low emissions. Same lube system for cooling and lubrication. High efficiency & High reliability.	MVSD efficiency drops off at part loads. Complicated design. Space and weight requirements.
Variable Low Speed Electric Motor Driven Centrifugal Compressor.	Good operational flexibility. Low emissions. Proven technology.	Requires high kVA power grid. Requires a gearbox. Lube oil system.
Variable High Speed Electric Motor Direct Driven Centrifugal Compressor	Good operational flexibility. High efficiency of drive system.	Limited availability of proven motor speed and power capability.
Integrated Electric Motor Driven Compressors	Good operational flexibility. Low operating pressure ratios. Eliminates on site emissions. High availability and reliability. Eliminates lube oil and oil systems. Compact unit design.	Station location limited to proximity to high voltage grid to minimize costs.

References

1. Pathways for Enhanced Integrity, Reliability and Deliverability, Natural Gas Infrastructure Reliability, DOE/NETL-2000/1130, September 2000.
2. DOE Solicitation No. DE-PS26-02NT41446, IEMDC-Totally Enclosed Electric Motor Driven Compressor.
3. Electric Powered Compression: A Viable Approach For Meeting New Pipelines, Wrenn, Frederick, K., Jr., Pipeline & Gas journal, October 2000.
4. Compressor Stations, Gas or electric compressor choices dictated by economics, David E. Reid, Pipe-line.com, July 1998 Vol. 81 No. 7.
5. Cost Efficient Electric High-Speed Drives for Gas Compression, Aherns, Marhus, Frei-Spreiter, Barbara, Weiser, Rudolf, 2000 International Pipeline Conference.
6. Totally Enclosed Inline Electric Motor Driven Gas Compressors, Johnson, Miller, Aherns, Flanery, Gas Research Machinery Council, October 2000
7. Advancements if Prime Movers Help Meet Growing Demands on Gas Compression, Hall, Kenneth C., Elijah, Jeffery, A., The American Oil & Gas Reporter, May 2001.
8. Engineering data Book Volume 1, Processors Gas Suppliers Association, Copyright 1998
9. High Power Four Stroke Lean Burn Gas engines with Centrifugal Compressors In Main Transmission Applications, Amundsen, Sautter, Goodenough, Gas Machinery Research Conference, October 2003.
10. Technical Report, Performance Improvement of Pipeline Combustion Turbines by Application of Air Injection Power Augmentation Technologies, Blades, Russel W., March 2003
11. Enhancing Operation of the Existing Natural Gas Compression Infrastructure, Harris, Bourn, Smalley, Gas Machinery Research Conference, October 2003.