ACTIVE MAGNETIC BEARINGS FOR OPTIMUM TURBOMACHINERY DESIGN

Jerry Hustak, R. Gordon Kirk, and Kenneth A. Schoeneck
Ingersoll-Rand Company
Phillipsburg, New Jersey 08865

The design and shop test results are given for a high speed eight stage centrifugal compressor supported by active magnetic bearings. A brief summary of the rotor dynamics analysis is presented with specific attention given to design considerations for optimum rotor stability. The concerns for retrofit of magnetic bearings in existing machinery are discussed with supporting analysis of a four stage centrifugal compressor. Recommendations are given on design and analysis requirements for successful machinery operation of either retrofit or new design turbomachinery.

INTRODUCTION

A decade ago new technology in the form of dry gas seals was introduced into the field of industrial turbomachinery design to minimize the capital, operating, and maintenance costs associated with seal oil systems (1). Today this type of sealing system is gaining widespread recognition because it has and is continuing to demonstrate its superior mechanical, performance, and economic features in certain applications. Now once again new technology in the form of active magnetic bearings (AMB) is being introduced into the marketplace for use on individual turbomachinery. The features of this technology when applied to turbocompressor design result in several economic, performance, and versatility improvements unavailable to the industry at the present time. Active magnetic bearings used in conjunction with dry gas seals and dry couplings now enable both the manufacturer and user to think in terms of oil-free centrifugal compressors, certainly a dramatic change from only ten years ago (2).

Patent activity on passive, active, and combination magnetic bearing systems spans 150 years. The bulk of the initial investigations centered on permanent magnetic systems because they were easy to fabricate. It was later shown, however, that a passive magnetic suspension for three axes of displacement is unstable; a theory that is still valid today. In 1969 Societe Europeanne de Propulsion (SEP), a French research firm, began investigating the characteristics of both passive and active magnetic suspension systems for a satellite flywheel application. In 1970 they developed a totally active magnetic suspension system for the COMSAT communications satellite. In 1976 SEP formed a new company Societe de Mecanique Magnetique (S2M) to further develop and commercially market active magnetic bearing (AMB) systems internationally (3,4).

DEVELOPMENT CENTRIFUGAL COMPRESSOR WITH AMB

Figure 1 is a view of an eight stage, horizontally-split, back-to-back centrifugal compressor equipped with magnetic radial and thrust bearings and gas seals on test at the authors' company (1980). The eight stage rotor housed inside the compressor, originally designed to run at 10,000 RPM (167 Hz) on hydrodynamic bearings,

has since operated successfully at speeds up to 13,000 RPM (217 Hz) on magnetic bearings. The compressor is shown attached to two closed loops constructed for the purpose of operating the rotor in a pressurized environment over a wide range of pressures and flows from choke to surge.

One unique feature of this compressor was the installation of the thrust and journal bearings located on the free end of the rotor directly into the gas (nitrogen) pressurized environment thereby eliminating the need for one shaft seal. To illustrate the concept of a nonlubricated centrifugal compressor a gas seal was chosen as the main shaft seal on the coupling end of the rotor. Table 1 summarizes some of the important design features of the eight stage back-to-back rotor while Figure 2 illustrates the appearance of the fully assembled test rotor.

Before power is applied to the bearings the rotor is supported on two auxiliary cage, dry lubricated ball bearings located in close proximity to the AMB. The clearance between the rotor and the inner race of the ball bearing is selected to prevent rotor contact with the AMB pole pieces or the internal seals of the compressor while the rotor is at rest or during an emergency shutdown. Typical radial clearances in the AMB, internal seals, and auxiliary bearing area are 0.012 in. (.3 mm), 0.010 in. (.254 mm), and 0.006 in. (.15 mm), respectively. When power is applied to the electronic controls the electromagnets levitate the rotor in the magnetic field and rotation of the driving source such as a motor or turbine can be started. The sensors and control system regulate the strength and direction of the magnetic fields to maintain exact rotor position by continually adjusting to the changing forces on the rotor. Should both the main and redundant features of the AMB fail simultaneously, the auxiliary bearings and rotor system are designed to permit safe deceleration.

The undamped critical speed map shown in Figure 3 compares the standard fluid film bearing/oil seal design to the magnetic bearing/gas seal design for the 8 stage development compressor. The magnetic bearing design increased the first rigid bearing mode by reducing the bearing span but decreased the second, third, and fourth modes due to the additional weight of the ferromagnetic journal sleeves and the larger diameter thrust collar. Superimposed on this map are the as measured stiffness and damping properties of the electromagnetic bearings as a function of rotor speed. For a design speed of 10,000 RPM (167 Hz), Figure 3 indicates the rotor would have to pass through three critical speeds and operate approximately 20% above the third critical and 40% below the fourth critical. Furthermore, since both the first and second criticals are rigid body modes only a significant response at approximately 8000 RPM (133 Hz) would be expected as the rotor passed through its third (free-free mode) critical. Subsequent unbalance forced response calculations verified these expectations with acceptable operation of the compressor to 14,000 RPM (233 Hz) (see Figures 4 and 5).

The damping required for optimum stability may be arrived at by construction of a Lund stability map (5) from the calculated damped critical speeds (5,6,7). For the 1st mode stiffness of 130,000 lb/in (22.8 N/μm), Figure 6 shows the movement of the eigenvalues as the bearing damping varies. Increased damping levels cause the first and second modes to become critically damped. The third mode increases in stability up to a point of 501 lb-sec/in (87.7 N-sec/μm) but then decreases as damping is increased further. Since the first and second modes become critically damped, a second study was undertaken to determine if the third mode would go unstable at its corresponding stiffness of 229,000 lb/in (40.1 N/μm). The results of that analysis are presented in Figure 7 which indicate the third mode becomes critically damped as the damping is increased while the 1st mode damping would be at an optimum for 645

lb-sec/in (112.9 N-sec/mm). The behavior of the 1st and 3rd modes, resulting from the increase in bearing stiffness, is similar to results presented by Lund (5). These results indicate that a level of 400-500 lb-sec/in (70 - 87.5 N-sec/mm) would be ideal for all modes up to and including the fourth.

The actual measured stiffness and damping characteristics for the magnetic bearing are given in Figure 8. Active magnetic bearings have an important difference when compared to conventional fluid film bearings. Typical preloaded five shoe tilt pad bearings have characteristics generated predominantly by operating speed, with little influence from non-synchronous excitations (8). The active magnetic bearing characteristics, shown in Figure 8, are dependent on the frequency of excitation regardless of operating speed. For a given stiffness value selected to minimize unbalance forced response, the damping characteristics at subsynchronous excitation frequencies can be specified to assure optimum stability (3, 4).

The test program outlined for the compressor was directed toward confirming the analytical predictions for the dynamic behavior of the rotor and experimentally demonstrating the reliability of the complete system under typical operating conditions. The fully assembled compressor was installed on the test stand and operated at a maximum discharge pressure of 600 psig (4.1 MPa) with speeds up to 13,000 RPM (217 Hz). The results for a decel as recorded at the bearing probe locations are given in Figures 9 and 10. The results are in general agreement with the predicted unbalance forced response results. Since the amount and location of the actual unbalance distributions are never known, the amplitude of each damped response is difficult to predict. The predicted peak response speeds are considered to be in good agreement with the test results.

DESIGN EVALUATION OF A FIELD RETROFIT

The economic advantages of gas seals and/or magnetic bearings has prompted interest in retrofit of existing units. For either retrofit or new machinery, attention must be given to placement of critical speeds for both main and back-up bearings, response sensitivity, and overall stability considerations. The preliminary design study for a 4 stage high speed centrifugal compressor will illustrate in more detail the parameters that must be considered for total system dynamic analysis. The basic design parameters for this rotor are indicated in the second column of Table 1.

. The undamped critical speed map for the four stage compressor with dynamic stiffness values is shown in Figure 11. The magnetic bearing stiffness is positioned such that the compressor must pass through three critical speeds before reaching a maximum continuous operating speed of 14,500 RPM (241.7 Hz). Due to the rigid body nature of the second and third modes, the actual damped critical speeds will occur from the first and fourth modes as shown in Figures 13 and 14 at approximately 4300 (71.7 Hz) and 18,300 RPM (305 Hz) respectively. Figure 15 shows the Lund stability map using a constant first mode stiffness of 86,300 lb/in (15.1 N/μm) with variable damping values. The first forward mode typically goes unstable while the second and third modes become critically damped as the bearing damping is increased. For this compressor design, the first mode increases in stability as the damping increases up to 225 lb-sec/in (39.4 N-sec/mm) but then decreases as the damping is increased further. The damping value initially supplied, 140 lb-sec/in (24.5 N-sec/mm), should be increased by 61% based on the results of this analysis.

The optimum damping for stability was also calculated by an approximate method

using the modal mass, rigid bearing critical frequency, and actual bearing stiffness (see Table 1). The equation from Reference (9) can be written as follows (valid for high K ratios):

$$C_O = 1.356 \times 10^{-4} \; N_{cr} \left\{ M_m + \frac{70417.7 \; K_B}{N^2 cr} \right\}$$

Example for 4-stage 1st mode:

$$C_O = 1.356 \times 10^{-4} \times (5423) \times (146 + (70417.7) \times (86300)/(5423)^2)$$

$$= 259 \; lb\text{-}sec/in$$

This calculation gives an answer 15% higher than the actual optimum damping for this K ratio of 1.41.

Figure 16 shows a comparison of stability versus aerodynamic excitation between the conventional fluid film design and the magnetic bearing retrofit design. The increase in stability due to the magnetic bearings moves the log dec from near zero to a value of 1.41.

CONCLUSIONS AND RECOMMENDATIONS

The capability of an active magnetic bearing system to support a flexible turbocompressor rotor and simultaneously influence its vibrations has been successfully demonstrated. During the 350 hours of accumulated operating time for the development compressor supported by magnetic bearings the following observations have been made:

1. The rotor behaved in a stable manner at all times when accelerating/decelerating through its first three critical speeds.

2. The rotor behaved in a stable manner while undergoing surge cycles at maximum discharge pressure.

3. The rotor was able to satisfy commonly accepted vibration amplitude and critical speed amplification criteria at all operating speeds up to 13,000 RPM (217 Hz). Speeds beyond this point were limited by impeller stress considerations.

The analysis of the two compressors has clearly shown the advantages of adjustable bearing stiffness and damping to achieve minimum response sensitivity and optimum stability. For example, the increase in stability from a log dec of approximately zero to 1.41 for the 4-stage compressor could not be accomplished by conventional fluid-film bearings.

The following recommendations can be made for the design and analysis of magnetic bearing suspension turbomachinery:

1. Bearing stiffness should be selected by evaluation of shaft stiffness ratio with typical placement at the beginning of the 3rd mode ramp on the undamped critical speed map.

2. Bearing damping should be specified to give the optimum growth factors, with consideration given to all modes below maximum operating speed.

3. Consideration must be given to the next mode above operating speed (typically the 4th mode) to assure adequate separation margins.

4. All clearances in the bearings and seals should be selected to avoid rotor/stator contact (i.e., rubs) for normal expected operating conditions.

5. The machinery must be engineered to give a minimum of 10% separation margin on any continuous operating speed for operation on the auxiliary bearings.

REFERENCES

1. Schoeneck, K.A. and Hornschuch, H., "Design Concept of a High Speed-High Pressure Ratio Centrifugal Compressor," ASME Paper 75-PET-4, Presented at Petroleum Mech. Engrg. Conf., Tulsa, OK, Sept. 1975.

2. Schoeneck, K.A., "The Application of Gas Seals and Magnetic Bearings to Centrifugal Compressors," Pacific Coast Gas Assoc. Transmission Conference, Spokane, WA, April 18-19, 1985.

3. Haberman, H., "The Active Magnetic Bearing Enables Optimum Damping of Flexible Rotors," ASME Paper 84-GT-117.

4. Haberman, H. and Brunet, M., "The Active Magnetic Bearing Enables Optimum Control of Machine Vibrations," ASME Paper 85-GT-22, Presented at Gas Turbine Conference, Houston, TX, March 18-21, 1985.

5. Lund, J.W., "Stability and Damped Critical Speeds of a Flexible Rotor in Fluid-Film Bearings," J. of Eng. for Industry, Trans. ASME, Series B, 96, No. 2, May (1974).

6. Bansal, P. and Kirk, R.G., "Stability and Damped Critical Speeds of Rotor-Bearing Systems," J. of Eng. for Industry, Trans. ASME, 98 Series B, No. 1, February (1976).

7. Kirk, R.G., "Stability and Damped Critical Speeds - How to Calculate and Interpret the Results," CAGI Technical Digest, Vol. 12, No. 2, 1980.

8. Wilson, B.W. and Barnett, "The Effect of Eigenvalue-Dependent Tilt Pad Bearing Characteristics on the Stability of Rotor-Bearing Systems", University of Virginia, Report No. UVA643092/MAES5/321, January, 1985.

9. Barrett, L.E., Gunter, E.J., and Allaire, P.E., "Optimum Bearing and Support Damping for Unbalance Response and Stability of Rotating Machinery, Trans. ASME, J. of Engr. for Power, 1978.

TABLE 1

CENTRIFUGAL COMPRESSOR DESIGN PARAMETERS AND NOMENCLATURE

Parameter, Nomenclature, U.S. Units, (SI Units)	Eight Stage		Four Stage	
Operating Speed, N, RPM (Hz)	13000.	(217.7)	14500.	(241.7)
Total Weight, W, lb (N)	827.	(3678)	278.	(1237)
Bearing Span, , in. (mm)	49.97	(1269)	34.88	(886.0)
Shaft Length, , in. (mm)	74.90	(1902)	49.32	(1253)
Coupling End Overhang, , in. (mm)	12.00	(304.8)	7.28	(184.9)
Shaft Stiffness, K_s, lb/in (N/μm)	3.49E5	(61.1)	1.22E5	(21.4)
Bearing Stiffness @ MCOS, K_b, lb/in (N/μm)	3.65E5	(63.9)	1.60E5	(28.0)
Stiffness Ratio, K, Dim., (Dim.)	0.74	(0.74)	1.41	(1.41)
Mid-Span Diameter, , in. (mm)	4.88	(123.9)	3.00	(76.2)
Journal Diameter, , in. (mm)	7.38	(187.5)	3.66	(93.0)
First Rigid Bearing Critical, N_{cr}, RPM (Hz)	5551.	(92.52)	5423.	(90.38)
First Peak Response Speed, FPS1, RPM (Hz)	7700.	(129.5)	4300.	(71.67)
Second Peak Response Speed, FPS2, RPM (Hz)	16300.	(271.7)	18300.	(305.0)
First Mode Modal Mass, M_m, lb_m (N)	399.	(1775)	146.	(649)
Bearing Stiffness @ N_1, K_{b1}, lb/in (N/μm)	1.30E5	(22.8)	8.63E4	(15.1)
Optimum Damping (Ref. 9), C_o, lb-sec/in (N-sec/μm)	524.	(9.18E-2)	259.	(4.54E-2)

NOTE: Values are given in both SI and U.S. customary units. The measurements and calculations were made in U.S. customary units.

332

EIGHT STAGE DEVELOPMENT COMPRESSOR TEST SETUP
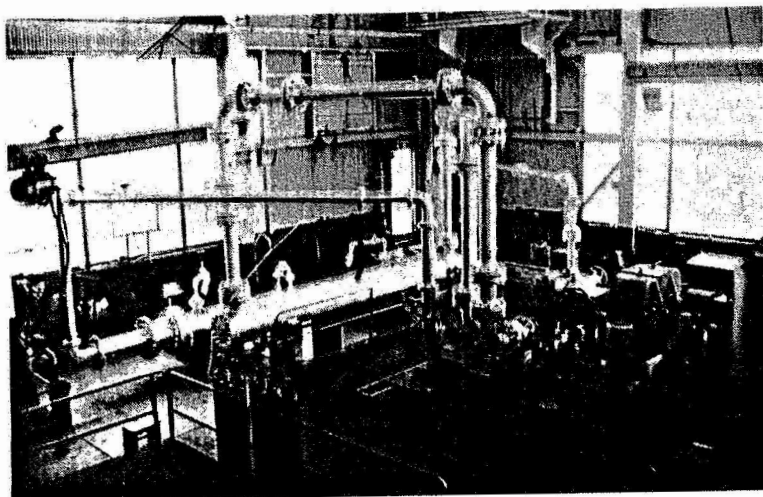
FIGURE 1

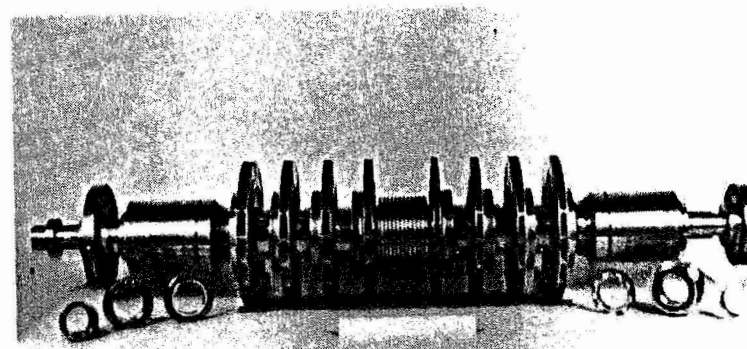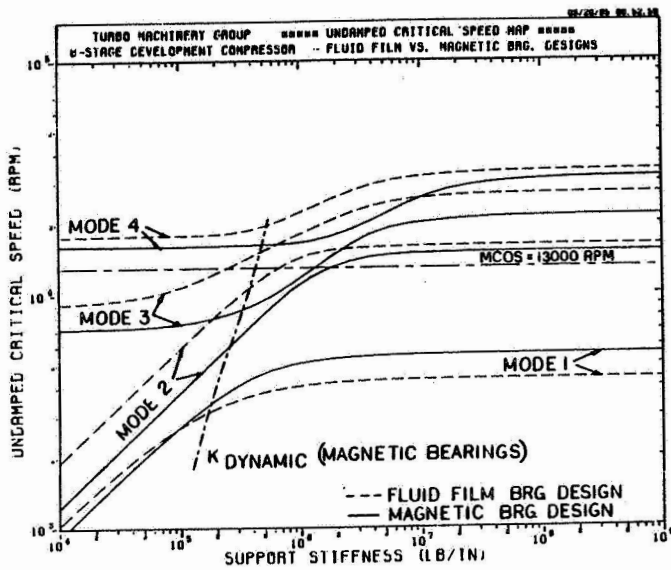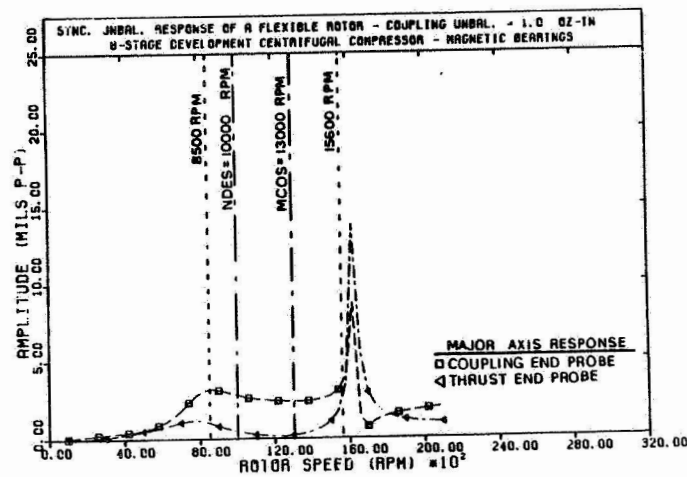EIGHT STAGE DEVELOPMENT COMPRESSOR ROTOR
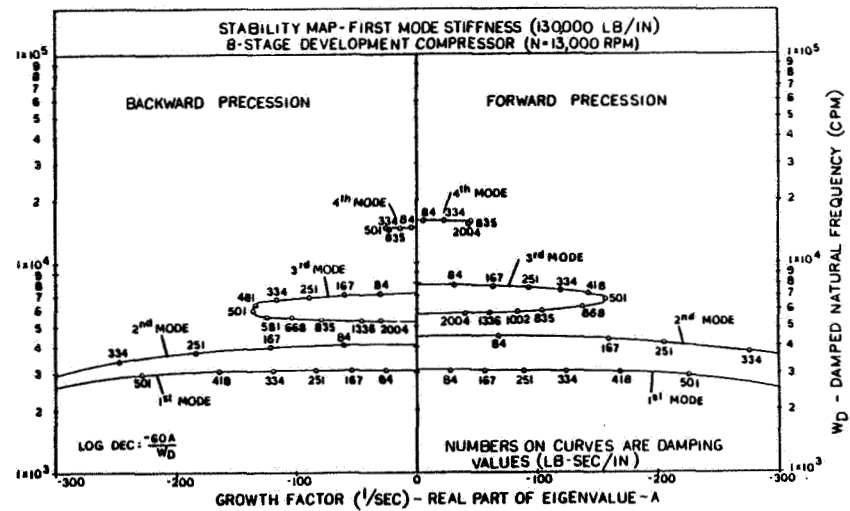

FIGURE 2

FIGURE 3

FIGURE 4

334
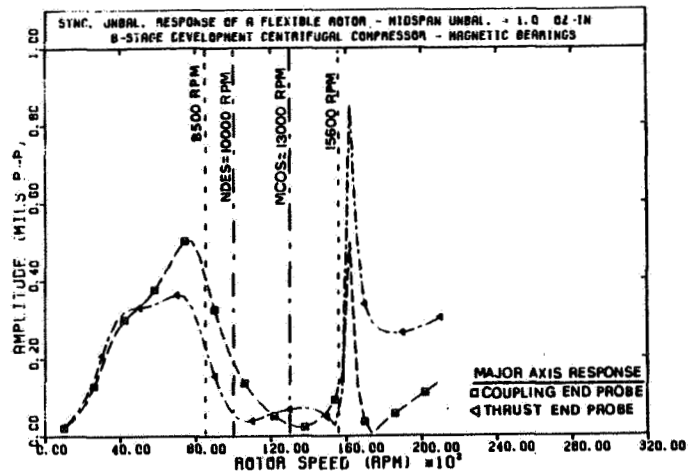
FIGURE 5

FIGURE 6

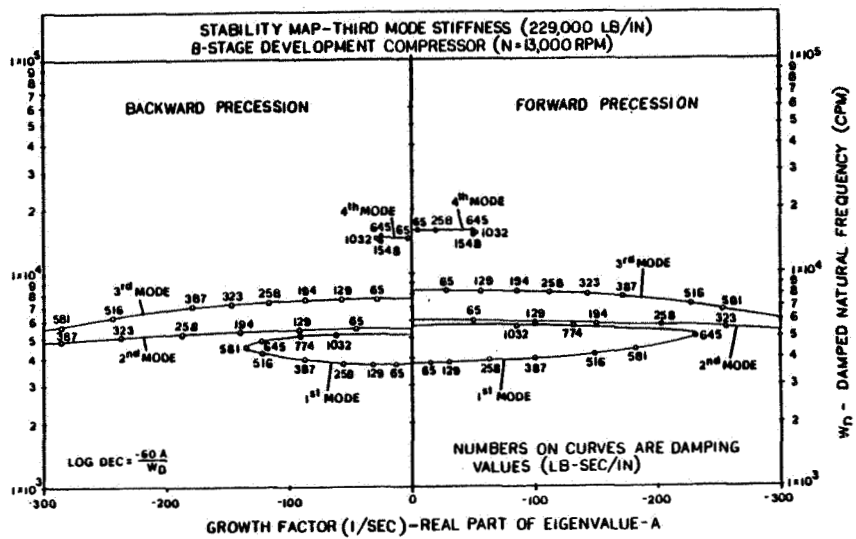**FIGURE 7**

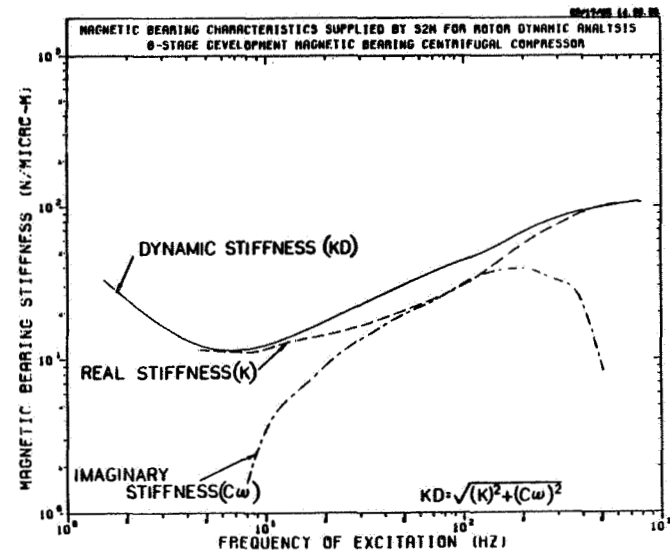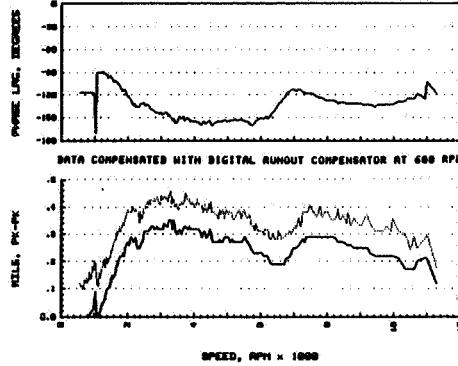**FIGURE 8**

INGERSOLL-RAND CO.
TURBO ENG. SCIENCES
PHILLIPSBURG, N.J.
PLANT ID: INGERSOLL-RAND
TRAIN ID: M.B. MACHINE
MACHINE ID: MGB 63
SOLID DATA: 1X Filt S2M M2-4   M2 - COUPLING VERT.
RUNDOWN   DASHED DATA:Overall S2M M2-4
RUN: 002   DATE: 5/17/85   TIME: 12:09

DATA COMPENSATED WITH DIGITAL RUNOUT COMPENSATOR AT 600 RPM

PHASE LAG, DEGREES

MILS, PK-PK

SPEED, RPM x 1000

FIGURE 9

INGERSOLL-RAND CO.
TURBO ENG. SCIENCES
PHILLIPSBURG, N.J.
PLANT ID: INGERSOLL-RAND
TRAIN ID: M.B. MACHINE
MACHINE ID: MGB 63
SOLID DATA: 1X Filt S2M V1-3   V1 - THRUST HORIZ.
RUNDOWN   DASHED DATA:Overall S2M V1-3
RUN: 002   DATE: 5/17/85   TIME: 12:09

DATA COMPENSATED WITH DIGITAL RUNOUT COMPENSATOR AT 600 RPM

PHASE LAG, DEGREES

MILS, PK-PK

SPEED, RPM x 1000

FIGURE 10

TURBO MACHINERY GROUP   ===== UNDAMPED CRITICAL SPEED MAP =====
4-STAGE RETROFIT COMPRESSOR - FLUID FILM VS. MAGNETIC BRG. DESIGNS

UNDAMPED CRITICAL SPEED (RPM)

MODE 4
MODE 3
MODE 2
MODE 1

MCOS=14500 RPM

$K_{DYNAMIC}$ (MAGNETIC BEARINGS)

- - - FLUID FILM BRG. DESIGN
——— MAGNETIC BRG. DESIGN

SUPPORT STIFFNESS (LB/IN)

FIGURE 11

MAGNETIC BEARING CHARACTERISTICS SUPPLIED BY S2M FOR ROTOR DYNAMIC ANALYSIS
4-STAGE MAGNETIC BEARING CENTRIFUGAL COMPRESSOR.

MAGNETIC BEARING STIFFNESS (N/MICRO-M)

DYNAMIC STIFFNESS (KD)

REAL STIFFNESS (K)

IMAGINARY STIFFNESS (Cω)

$KD = \sqrt{(K)^2 + (C\omega)^2}$

FREQUENCY OF EXCITATION (HZ)

FIGURE 12

FIGURE 13

FIGURE 14

FIGURE 15

FIGURE 16