AN INTRODUCTION AND CASE HISTORY REVIEW OF ACTIVE MAGNETIC BEARINGS

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ABSTRACT

The concept of magnetic levitation is not a new one and can be easily traced back to the 1800s [1]. It is only recently, however, that the congruous technologies of electronic control systems, power electronics, and magnetic materials have begun to merge to make the magnetic suspension device a viable product.

A brief overview is presented of an active magnetic bearing technology [2]. The required systems engineering interface with the machine designer is discussed. Finally, case histories of various turbomachinery in North America presently operating on magnetic bearings are reviewed.

BACKGROUND

In theory, the principle is quite basic. An electromagnet will attract any piece of ferrous material. By using a stationary electromagnet (stator) and a rotating ferrous material (rotor) a shaft can be suspended in a magnetic field while maintaining accurate position under varying loads. This can be accomplished given a small space (air gap) between the stator and rotor and proper electronic control of the electromagnet. In the following case of the active magnetic bearing, this concept is utilized for both radial and axial configurations. It must be noted that the bearing system described here always operates in an attraction mode and never repulsion.

The radial and axial bearing rotors make use of a ferrous laminated sleeve and solid disc, respectively [3]. Applying ferrous rotor elements to the shaft allows the shaft material to be constructed from a nonmagnetic metal or composite material. While the radial bearing requires laminations due to the number of flux reversals during rotation, the axial rotor disc can be solid since the magnetic flux level is changing, but the polarity is not.

As with any type of electromagnet, a wound field stator is required to produce a force output. Both the radial and axial bearing stators incorporate laminations to minimize stray losses and improve the bearing response time. The radial bearing stator is wound to provide four independently controllable quadrants for maximum rotor stability. The axial bearing, attracting the rotor in only one plane, requires the use of two stators, one on either side of the rotor disc, to provide double acting control.

Inductive position sensors are used to detect the exact radial and axial location of the shaft. Similar to the bearings, these sensors utilize a ferrous rotor and a wound field stator. As the air gap at the sensors changes with shaft disturbances, the inductance bridge of the sensor also changes. It is this change in inductance with air gap variation, that provides the position feedback signal required for closed loop servo control.

An isometric view of both a radial and double acting axial bearing with their associated position sensors is shown in Figure 1.

Control electronics are required to process the position signal and power the appropriate bearing coils. The exact shaft location is detected by the position sensors, and a DC voltage is generated which is relating to rotor displacement. This DC voltage (where the shaft is) is compared to the position reference signal (where the shaft should be). Any difference between these two signals generates an error signal which is used to maintain control of the rotor. This signal is then amplified, filtered, and conditioned prior to commanding the specified power amplifier(s). Current is increased or decreased in the appropriate bearing coil(s) to maintain the rotor at equilibrium. A basic block diagram is shown in Figure 2 of the closed loop servo control.

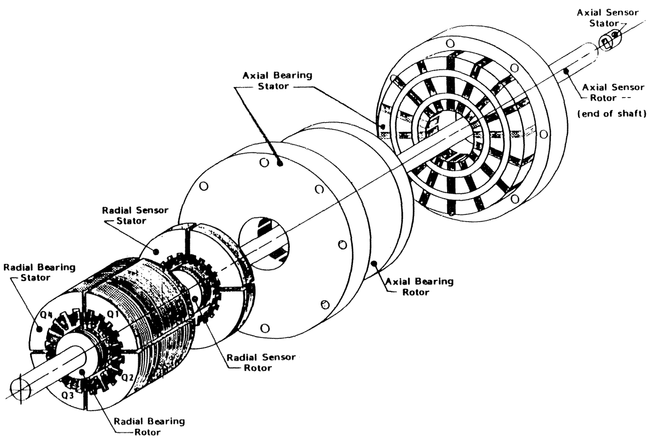

Figure 1. Radial and Axial Bearing Configuration.

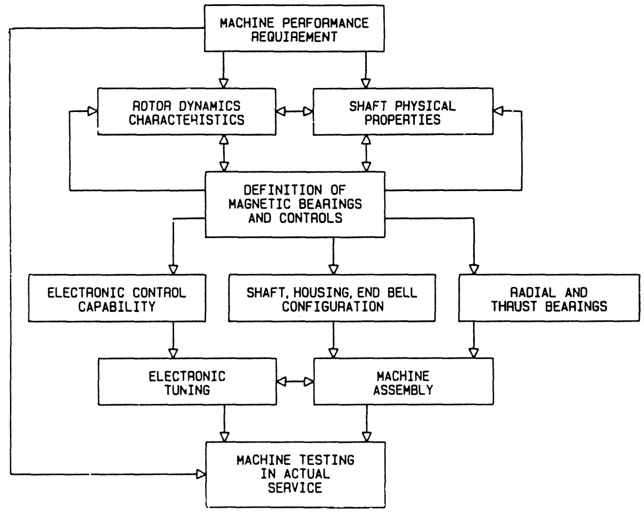

Figure 3. System Engineering Methodology.

SYSTEM ENGINEERING

A system engineering methodology has been developed to apply active magnetic bearings as described above to each specific turbomachine with a high degree of design success [4]. The process is flow charted in Figure 3, beginning with a machine performance specification.

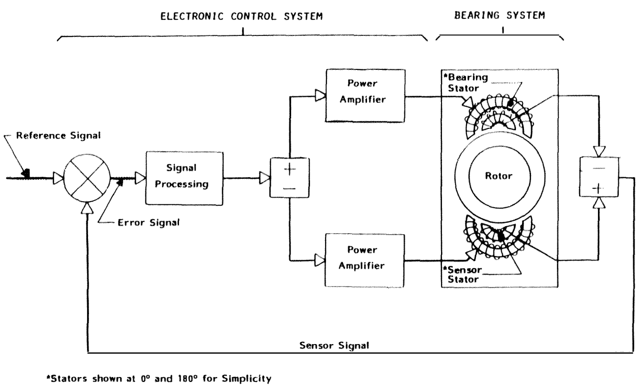

Figure 2. Basic Control Loop Diagram.

As shown by the steps in Figure 3, application of magnetic bearings from conceptual layout through equipment startup requires a "system" rather than a "component" engineering approach. Physical configuration and operating environment, static and dynamic load requirements, and rotordynamics must all be addressed to achieve successful implementation for a machine such as a centrifugal compressor.

Optimizing the physical configuration is an iterative process involving the compressor designer and the magnetic bearing designer. New options exist for the compressor designer such as bearings immersed in the process gas, larger shaft diameters at the bearing journals and more flexibility in locating bearings within the compressor case (i. e., midspan bearings, inboard or outboard thrust bearing). The bearing operating environment is usually established early in the design process. Of particular concern is the nature of the fluid in contact with the bearing components and the operating temperatures. This is of paramount importance in design of the bearing hardware, especially winding and encapsulant selection.

Detailed consideration must be given to the static and dynamic loads the magnetic bearing will control. Magnetic bearings are not as forgiving of gross overloads. It is often difficult to establish the exact load requirements, especially on a new machine. The magnetic bearing designer can manipulate several parameters (bus voltage, airgap, number of turns, control class) to optimize dynamic load capability, but ultimately bearing active area and the magnetic material saturation flux density will limit the bearing load capability. For this reason, it is desirable to select a bearing with reserve capacity. One approach is to design so that static levitation is achieved at about 1.0 Tesla flux density in the bearing airgap. This gives a margin of about two to one on static load capacity when using materials (silicon iron) with a 1.5 Tesla saturation flux density. This margin results from the magnetic force being proportional to the square of the magnetic flux density:

$$\left(\frac{1.5}{1.0}\right)^2 = 2.25 \tag{1}$$

Rotordynamics must be considered in parallel and often dictate the compressor physical configuration. The magnetic bearing controller governs the bearing dynamic characteristics of stiffness and damping which determines the rotor system dynamics. The compressor designer seeks to achieve an acceptable design based on critical speed margins, amplification factors and rotor stability, while the magnetic bearing designer must meet minimum gain and phase margins over a broad bandwidth for control loop (and therefore rotor) stability. The "system approach" found most useful is to first generate an undamped critical speed map and mode shapes for the rotor based on a desired physical configuration. This requires detailed model data from the machine designer. The mode shapes may indicate problems with nodes near the bearing or between the bearing and sensor. Machine modifications or bearing design changes such as sensor location or even multiple sensors may be necessary. If mode shapes are acceptable, reasonable bearing stiffness characteristics may be estimated from the critical speed map. The controller gain and phase characteristics may then be tailored (within limits) to the application. Controller characteristics are often based on previous working designs or modifications of previous designs. The magnetic bearing designer must ensure the proposed bearing characteristics are reasonable and achievable in practice from both control loop stability and bearing load capacity perspectives. A design, for example, that has a rotor running speed stiffness (gain) so high that the full dynamic load capacity of the bearing is commanded (by the controller) in a few microinches of rotor motion, is probably unreasonable for this class of machines. A listing of the proposed bearing characteristics is then supplied to the machine designer to verify acceptable rotor response using magnetic bearings.

The final step in the "system engineering" process is rotor levitation and control loop compensation with the actual compressor. The magnetic bearing designer must then confirm that the proposed bearing characteristics have been achieved while also satisfying more fundamental control loop stability requirements.

CASE HISTORIES

At the time of this writing, there are presently eleven industrial machines in North America operating on this type of magnetic bearing system. A listing of these units is provided in Table 1, and includes basic machinery data. Also shown is a listing of ten additional compressors scheduled for startup in 1989.

Table 1A. Magnetic Bearing Equipped Turbomachinery.

Machine	Rotor Type	Service	Duty	Comm	Rotor Weight Lb	Specified Thrust Load Lb	Speed rpm	Journal Diameter	Rating hp	Operating Hours (*)
CDP-230 Comp.	2 Stage Beam	Pipeline Methane	Seasonal	1985	3420	12000	5250	10.6"	14005	10466
CDP-416 Comp.	4 Stage Beam	Pipeline Methane	Seasonal	1986	300	2800	14500	3.7"	4150	13530
1B26 Comp.	1 Stage Beam	Pipeline Methane	Continuous	1956	790	4050	11000	6.5"	6448	11438
1B26 Comp.	1 Stage Beam	Pipeline Methane	Continuous	1987	790	4050	11000	6.5"	6448	8731
CBF-842 Comp.	8 Stage Beam	Refinery Wet Gas	Continuous	1987	1425	4600	10250	6.0"	4500	11245
3B37 Comp.	3 Stage Beam	Refinery Hydrogen	Continuous	1988	1050	4600	10230	6.0"	4500	6328
RFB30 Comp.	1 Stage Overhung	Pipeline Methane	Seasonal	1988	1730	15400	5200	10.5"	14256	1701
5P2 Comp.	2 Stage Beam	Pipeline Methane	Seasonal	1988	1500	10000	7140	6.0"	16600	1709
5P2 Comp.	2 Stage Beam	Pipeline Methane	Seasonal	1988	1500	10000	7140	6.0"	16600	657
5P2 Comp.	2 Stage Beam	Pipeline Methane	Seasonal	1988	1500	10000	7140	6.0"	16600	271
RF2BB30 Comp.	2 Stage Beam	Pipeline Methane	Seasonal	1988	1980	4000	5200	9.5"	33240	173
GT-51 Turbine	1 Stage Overhung	Pipeline	Seasonal	1988	3530	22000	5250	8.0" 12.0"	14005	532

*AMB Cabinet operating hours as of May 1989 — TOTAL HOURS: 66,781

Table 1B. Turbomachinery to be Commissioned With Magnetic Bearings.

Machine	Rotor Type	Service	Duty	Comm	Rotor Weight Lb	Specified Thrust Load Lb	Speed rpm	Journal Diameter	Rating hp	Operating Hours (*)
5P2 Comp.	2 Stage Beam	Pipeline	Seasonal	1989	1500	10000	7140	6.0"	16600	---
RFBB-36 Comp.	1 Stage Beam	Pipeline	Seasonal	1989	1980	8000	5250	9.5"	33240	---
RF2BB-30 Comp.	2 Stage Beam	Pipeline	Seasonal	1989	1980	8000	5250	9.5"	33240	---
RF2BB-30 Comp.	2 Stage Beam	Pipeline	Seasonal	1989	1980	8000	5250	9.5"	33240	---
RF2BB-32 Comp.	4 Stage Beam	Pipeline	Seasonal	1989	600	11000	10500	7.5"	14400	---
7.5P Comp.	1 Stage Beam	Pipeline	Seasonal	1989	3227	8000	5775	8.5"	29550	---
8P Comp.	1 Stage Beam	Pipeline	Seasonal	1989	5000	8000	5250	9.5"	20000	---
RF2BB30 Comp.	2 Stage Beam	Pipeline	Seasonal	1989	1980	8000	5250	9.5"	15405	---
C-30 Comp.	1 Stage Beam	Pipeline	Continuous	1989	225	2710	15775	3.75"	3830	---
PCS Comp.	2 Stage Overhung	Pipeline	Continuous	1989	1980	1760	10000	8.50"	8000	---

Reasons for utilizing magnetic bearings in rotating machinery vary with each particular application, although many common threads are evident. Heavy equipment users, typically employing oil lubricated tilting pad bearings, see many advantages, including efficiency and safety in eliminating the oil lubrication system [5]. Such a system which utilizes external lube oil pumps, piping, reservoirs and filters are also costly elements to install and maintain. In many cases, more heavy equipment downtime is attributable to failures in machinery subsystems than actual machinery failure itself. Other users of magnetic bearings cite higher speeds, harsh environment operation, and optimized rotordynamic characteristics as reasons for using magnetic bearings [6].

While researching the operating histories of the previously mentioned machinery it became apparent that discussing each unit in depth would become monotonous. Following the commissioning of each machine, multiple starts and stops have occurred and operating hours accrued with very little attention brought by the fact that it is a "magnetic bearing" machine. Therefore, it was decided to expand from Table 1 only two machines with varying dynamic conditions.

The first machine researched is an Ingersoll-Rand pipeline compressor model number CDP-230 [7]. The unit is part of the NOVA natural gas pipeline system in Alberta, Canada and was put into service at the Hussar compressor station in 1970. This train is ISO rated at 14,650 hp and consists of a General Electric LM-1500 gas generator exhausting into an Ingersoll-Rand GT-51 power turbine dry coupled to the compressor. The normal operating speed range is from 3000 to 5250 rpm.

As originally supplied, this compressor incorporated oil film seals and bearings. In 1982, the conventional oil seal system was replaced by a mechanical dry gas seal [8]. Three years later conversion of this unit to the world's first oil free compressor of its type in production service was completed with the retrofit of the oil film bearings to active magnetic bearings [9].

Following the installation of the magnetic bearings, extensive dynamic testing took place. Two bearing resonant frequencies were identified at 28 and 42 Hz, with the first three shaft modes occurring at 89, 142 and 190 Hz. It can be seen that the first bending mode at 89 Hz (5,340 rpm) is very near the maximum operating speed of 5,250. However, the bearing control system maintained shaft movements no less than 0.8 mils peak-to-peak with no noticeable excess current draw during operation. Bearing parameters were monitored under various load conditions and data collected as per Table 2.

Table 2. Test Values. Current, Bearing Loads and Power Consumption.

Bearing Location		Static Testing (0 rpm) Casing Pressurized	Normal Operation 3600 rpm $\Delta P = 112$ psi	Choke Condition 4500 rpm $\Delta P = 12$ psi	Surge Condition 4100 rpm $\Delta P = 152$ psi	In-Service 12-15-87 4280 rpm	
Outboard Radial Bearing	Upper (2) Quadrants (Average)	17.6	18.3	17.7	18.0	17.2	Current (amps)
		1370	1460	1370	1439	1339	Load (lbs)
	Lower (2) Quadrants (Average)	5.0	5.0	5.0	5.0	5.0	Current (amps)
		112	112	112	112	112	Load (lbs)
Inboard Radial Bearing	Upper (2) Quadrants (Average)	21.0	20.2	20.1	21.0	19.3	Current (amps)
		1933	1798	1776	1933	1705	Load (lbs)
	Lower (2) Quadrants (Average)	5.0	5.0	5.0	5.0	5.0	Current (amps)
		112	112	112	112	112	Load (lbs)
Thrust Bearing	Outboard	11.8	5.0	41.5	5.0	19.2	Current (amps)
		679	112	8610	112	1790	Load (lbs)
	Inboard	5.0	15.8	5.0	18.8	5.0	Current (amps)
		112	1259	112	1753	112	Load (lbs)
Total Bearing Power Consumption (Hp)		4.7	5.1	5.4	5.1	4.8	

Subsequent evaluations of the operating history of this machine provided additional economic and performance data [10]. While most of this data include improvement from the installation of both the gas seal and the magnetic bearings, it is representative of the benefits associated with a lubrication free machine.

By total elimination of the oil system parasitic and oil shear horsepower losses improved the units output power by approximately two percent. The magnetic bearing system on the compressor comsumes about five hp of energy. This compares to 302 hp lost in the conventional bearing and seal oil system.

Maintenance savings were also calculated and determined to be a rather substantial figure. With the total absence of contacting stationary and rotating components, no wear related maintenance was seen. Also maintenance to the lubrication and seal oil subsystems was eliminated. Overall machinery maintenance, call outs, and downtime have been reduced by 85 percent. With the total average scheduled maintenance cost for the compressor and associated equipment of $41,250 and $22,500 typically related to call outs and unscheduled maintenance, an annual maintenance savings of $54,187 was calculated.

Based on these maintenance savings and the additional savings associated with oil consumption and oil and pipeline contamination a payback period of 4.4 years is anticipated for this retrofit. Installation of magnetic bearings and dry gas seals in a new compressor, where the initial bearing and seal costs are offset by not purchasing a bearing and seal oil system can improve the payback period to less than one year.

A SPECIFIC APPLICATION

Another specific application is reviewed to highlight some details of the system engineering involved in an actual machine. The application was a magnetic bearing retrofit of a single stage natural gas pipeline compressor [4]. The machine had a 700 lb rotor operating at a maximum speed of 11,700 rpm. The radial bearings had a maximum load capacity of 890 lb per bearing quadrant. Static levitation was achieved at 0.8 Tesla flux density in a 0.020 in airgap. The compressor had a conventional physical configuration with all bearings outside of the process gas and the thrust bearing inboard of the radial bearings. Operating speed was under the first shaft bending mode (3rd critical) as shown in Figure 4. The first and third free-free (low bearing stiffness) mode shapes are shown in Figure 5 and Figure 6, respectively. The final controller gain and phase and resulting bearing stiffness and damping characteristics are shown in Figure 7. The bearing stiffness (k) and dynamic stiffness (K) are also plotted on the critical speed map (Figure 4). The dynamic stiffness K is the vector sum of the (real) stiffness and the (imaginary) damping stiffness and is noted as "KMAG" in Figure 7. Although a high gain control loop (i. e., stiffer bearings) were initially proposed, it was found during site tuning to cause the shaft second mode to encroach on the operating speed range. The data shown reflect the actual bearing characteristics as measured during startup at the site. A second machine, identical to the first unit, has since been commissioned. Controller and bearing characteristics of the two units are identical. At the time of this writing, two units have more than 20,000 hrs of operation.

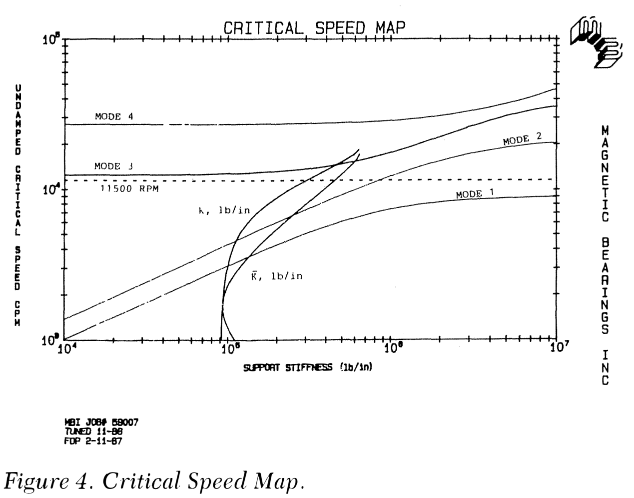

Figure 4. Critical Speed Map.

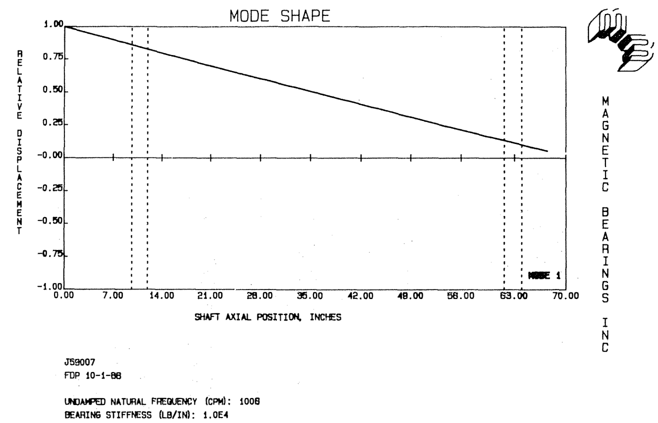

Figure 5. Mode Shape.

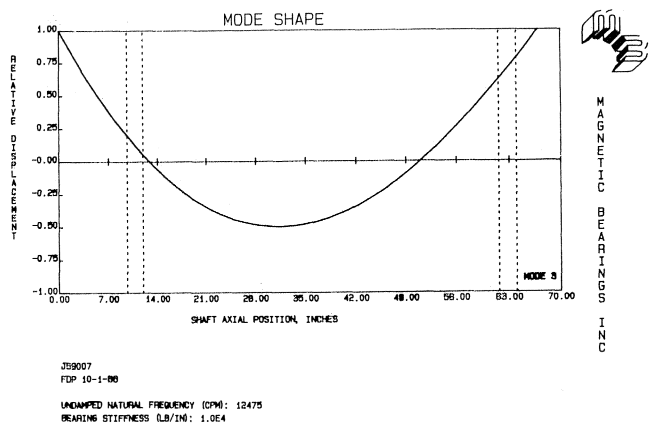

Figure 6. Mode Shape.

CONCLUSION

Magnetic bearings have proven to be a viable and attractive alternative to conventional bearings. Lubrication free equipment capable of harsh environment operation is definitely seen as the future of many types of rotating machinery. While turbomachinery users have benefitted greatly from applying magnetic bearings to existing designs, substantial advances can be made in rotating equipment by fully exploiting all of the operational advantages of magnetic bearings in future generation machinery.

REFERENCES

1. Earnshaw, S., "On the Nature of the Molecular Forces," Trans. Cambridge Phil. Soc. 7, pp. 97-112, (1842).
2. Weise, D. A., "Active Magnetic Bearings Provide Closed Loop Servo Control for Enhanced Dynamic Response," 27th IEEE Machine Tool Conference (October 1985).
3. Weise, D. A., "Present Industrial Applications of Active Magnetic Bearings," 22nd IECEC'87 Philadelphia, Pennsylvania (August 10-14, 1987).
4. Moses, H. J., Pinckney, F. D., and Weise, D. A., "Magnetic Bearing Turbomachinery Operating Experience," First International Symposium on Magnetic Bearings, Zurich, (1988).

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MAGNETIC BEARINGS INC
JOB #59007
BEARING CHARACTERISTICS
TUNED 11-86
SF=318265 K=345000LB/IN @ 150HZ
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FREQ	GAIN	PHASE	RPM	K,LB/IN	C,LB*SEC/IN	KMAG
32.750	0.298	7.85	1965.00	0.9408E+05	0.6300E+02	0.9497E+05
35.110	0.304	12.33	2106.60	0.9461E+05	0.9375E+02	0.9685E+05
37.650	0.312	16.67	2259.00	0.9525E+05	0.1206E+03	0.9943E+05
40.370	0.323	20.83	2422.20	0.9599E+05	0.1440E+03	0.1027E+06
43.290	0.335	24.79	2597.40	0.9685E+05	0.1645E+03	0.1067E+06
46.420	0.350	28.53	2785.20	0.9787E+05	0.1824E+03	0.1114E+06
49.770	0.367	32.02	2986.20	0.9903E+05	0.1980E+03	0.1168E+06
53.370	0.387	35.27	3202.20	0.1005E+06	0.2119E+03	0.1230E+06
57.220	0.409	38.28	3433.20	0.1021E+06	0.2240E+03	0.1300E+06
61.360	0.433	41.02	3681.60	0.1040E+06	0.2347E+03	0.1378E+06
65.790	0.461	43.52	3947.40	0.1063E+06	0.2441E+03	0.1466E+06
70.550	0.491	45.77	4233.00	0.1090E+06	0.2526E+03	0.1562E+06
75.650	0.524	47.76	4539.00	0.1122E+06	0.2600E+03	0.1669E+06
81.110	0.561	49.51	4866.60	0.1160E+06	0.2666E+03	0.1786E+06
86.970	0.602	51.01	5218.20	0.1205E+06	0.2725E+03	0.1916E+06
93.260	0.647	52.24	5595.60	0.1260E+06	0.2777E+03	0.2058E+06
100.000	0.696	53.23	6000.00	0.1325E+06	0.2822E+03	0.2214E+06
107.200	0.749	53.95	6432.00	0.1403E+06	0.2862E+03	0.2384E+06
115.000	0.808	54.40	6900.00	0.1497E+06	0.2893E+03	0.2571E+06
123.300	0.872	54.56	7398.00	0.1609E+06	0.2919E+03	0.2775E+06
132.200	0.942	54.42	7932.00	0.1744E+06	0.2935E+03	0.2998E+06
141.700	1.018	53.96	8502.00	0.1905E+06	0.2943E+03	0.3240E+06
152.000	1.101	53.20	9120.00	0.2099E+06	0.2938E+03	0.3504E+06
163.000	1.190	52.05	9780.00	0.2329E+06	0.2916E+03	0.3787E+06
174.800	1.286	50.51	10488.00	0.2603E+06	0.2876E+03	0.4093E+06
187.400	1.388	48.53	11244.00	0.2925E+06	0.2811E+03	0.4418E+06
200.900	1.497	46.06	12054.00	0.3306E+06	0.2718E+03	0.4764E+06
215.400	1.610	43.01	12924.00	0.3747E+06	0.2583E+03	0.5124E+06
231.000	1.726	39.27	13860.06	0.4253E+06	0.2396E+03	0.5493E+06
247.700	1.840	34.66	14862.00	0.4817E+06	0.2140E+03	0.5856E+06
265.600	1.944	28.91	15936.00	0.5416E+06	0.1792E+03	0.6187E+06

Figure 7. Tuned Bearing Characteristics.

5. Cataford, G. F., and Lancee, R. P. "Oil Free Compression on a Natural Gas Pipeline," ASME Paper No. 86-GT-293 (1986).
6. Hustak, J. F., Kirk, R. G., and Schoeneck, K. A., "Analysis and Test Results of Turbocompressors Using Active Magnetic Bearings," ASLE Preprint No. 86-AM-1A-1, (1986).
7. Weise, D. A., "Magnetic Bearing Turbomachinery Case Histories and Applications for Space Related Equipment," NASA, First Magnetic Suspension Technology Workshop, (February, 1988).
8. Hesje, R. C. and Peterson, R. A., "Mechanical Dry Seal Applied to Pipeline (Natural Gas) Centrifugal Compressors," ASME Paper No. 84-GT-3 (1984).
9. Foster, E. G., Kulle, V., and Peterson R. A., "The Application of Active Magnetic Bearings to a Natural Gas Pipeline Compressor," ASME Paper No. 86-GT-61 (1986).
10. Uptigrove, S. O., Harris, T. A., and Holzner, D. O., "Economic Justification of Magnetic Bearings and Mechanical Dry Seals for Centrifugal Compressors," ASME Paper No. 87-GT-174 (1987).