Magnetic Bearing Turbomachinery Operating Experience

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Summary

The theory of active magnetic bearing technology utilizing electromagnetics and closed loop feedback control is presented. A system engineering process has been developed to apply active magnetic bearings to turbocompressors of various physical sizes, horsepower ratings and performance characteristics. This process is explained and a specific example reviewed. Characteristics of a number of North American machines designed using the system engineering methodology are tabulated. Future technology applications and enhancements are outlined.

Introduction

Volumes of research, development and analytical work can be identified relating to the history of magnetic bearings and suspension devices. The concept of magnetically suspending a rotor began to receive serious attention in the mid-nineteenth century. U.S. patents dating back to the early 1800's incorporated the use of "magnetic bearings" to support rotating shafts, reduce mechanical friction, and improve the efficiency of electrical meters. In 1842 Ernshaw analytically demonstrated the instability inherent in totally passive (permanent) magnetic bearing systems and presented the requirement of at least one active degree of freedom to achieve rotor stability.

In the years since then passive, active, and hybrid systems have been experimented with, each with its own advantages and disadvantages. While the basic theories involved have not significantly changed over the years, advancements in magnetic materials, servo circuitry, and power electronic components have provided many enhancements to the magnetic bearing technology. Today private corporations and institutes around the world are applying magnetic bearings in devices as diversified as centrifugal compressors, helium circulators, x-ray tubes, energy storage flywheels, high vacuum pumps, machine tool spindles, cryogenic compressors and space guidance. While the applications have been extremely varied, the technologies involved can be classified as passive, active, or hybrid systems. To date, the technology most successfully applied to large turbomachinery has been the fully active magnetic bearing system as described below.

Active Magnetic Bearing Operation

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 case of the active magnetic bearing, this concept is utilized for both radial and axial configurations. The bearing system described 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. 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 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.

Figure 1, Radial and Axial Bearing Configuration, shows an isometric view of both a radial and double acting axial bearing with their associated position sensors.

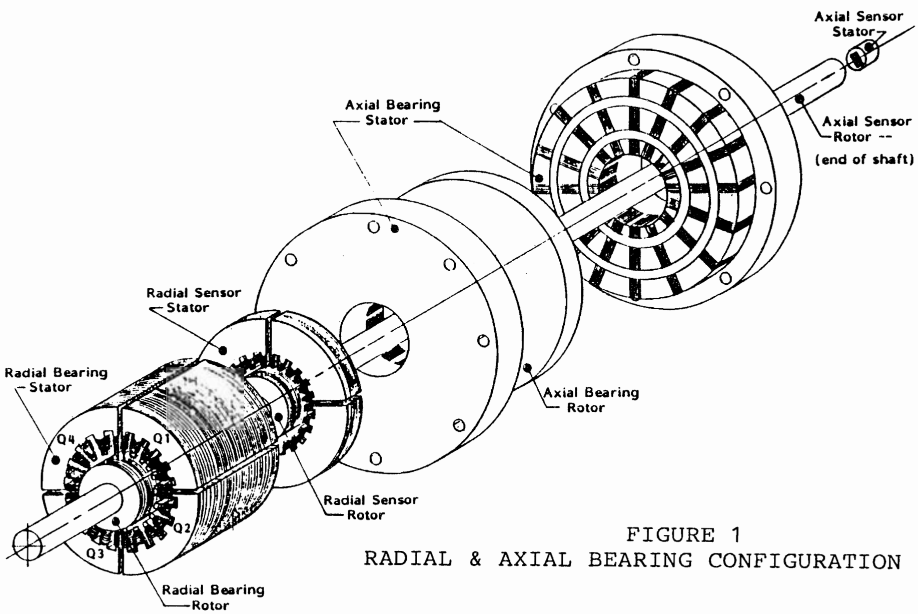
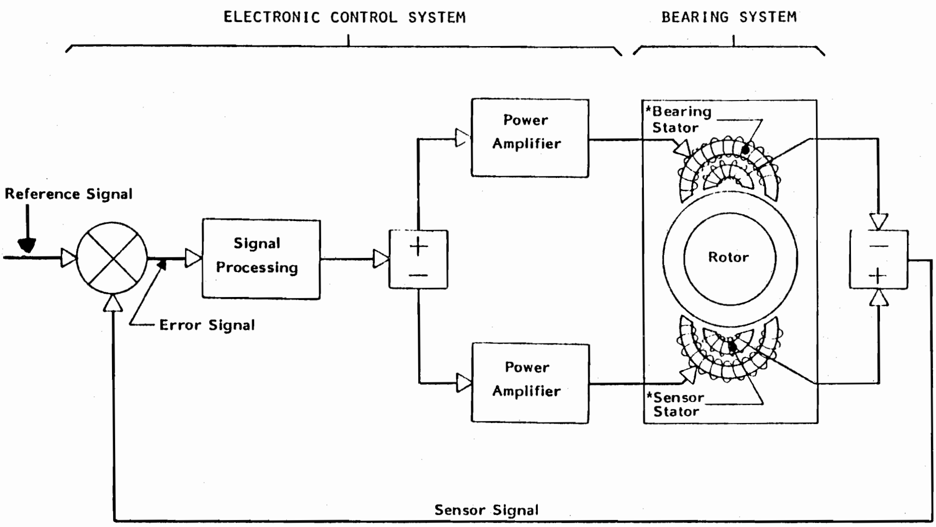


FIGURE 1
RADIAL & AXIAL BEARING CONFIGURATION

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 related 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. Figure 2, Basic Control Loop Diagram, shows a block diagram of the closed loop servo control.

BASIC CONTROL LOOP DIAGRAM



*Stators shown at 0° and 180° for Simplicity

FIGURE 2

System Engineering

A system engineering methodology has been developed to apply active magnetic bearings as described above to a specific turbomachine with a high degree of design success. Figure 3, System Engineering Methodology, flow charts the process beginning with a machine performance specification.

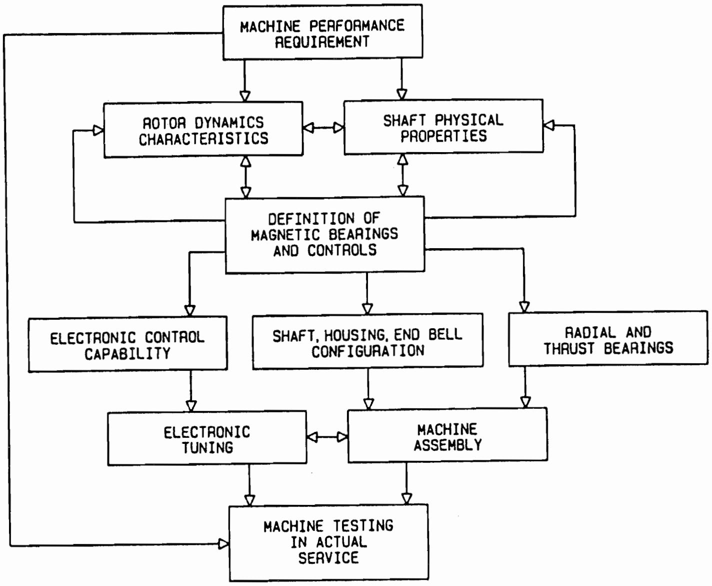


FIGURE 3
SYSTEM ENGINEERING METHODOLOGY

As shown by the steps in Figure 3, application of magnetic bearings, from conceptual layout through equipment start-up, requires a "system" rather than a "component" engineering approach. Physical configuration and operating environment, static and dynamic load requirements, and rotor dynamics 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 machine (compressor) designer and the magnetic bearing designer. New options exist for the machine (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 machine (compressor) case (i.e. midspan bearings, inboard or outboard thrust bearings). 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 forgiving of gross overloads and it is often difficult to establish the 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 often desirable to select a bearing with substantial 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:

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

Rotor dynamics must be considered in parallel, and often dictate, the machine (compressor) physical configuration. The magnetic bearing controller governs the bearing dynamic characteristics of stiffness and damping which determines the rotor system dynamics. The machine (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 insure 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 machine (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.

A Specific Application

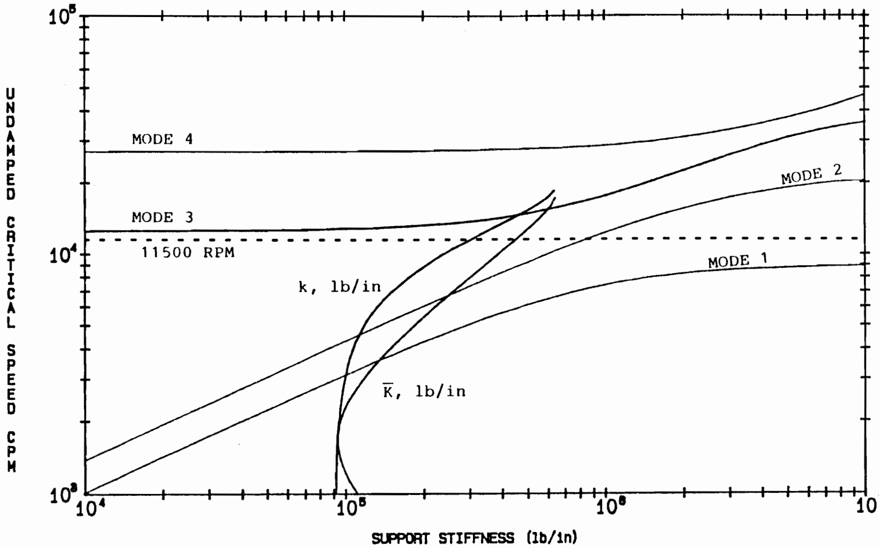
A specific application will now be reviewed to highlight some details of the system engineering involved in an actual machine. The application was a magnetic bearing retrofit of a single stage natural gas pipeline compressor. The machine had a 700 lb. rotor operating at a maximum speed of 11,700 rpm. 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 inch 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, Critical Speed Map. 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 higher 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 reflects the actual bearing characteristics as measured during start-up 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. The two units now have a total of more than 10,000 hours of operation.

Benefits of Active Magnetic Bearings

Application of active magnetic bearings depends not only on a successful implementation as described above but also on recognizing and quantifying the expected economic benefits.

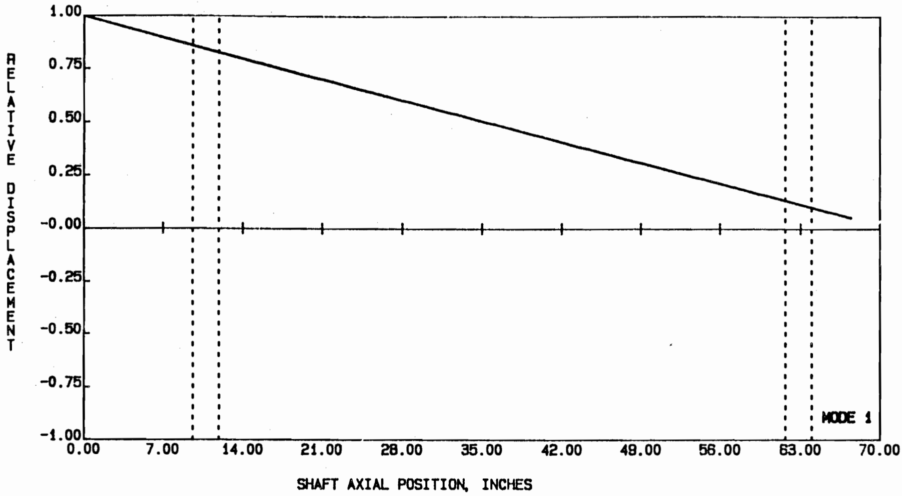
Reasons for utilizing magnetic bearings in rotating machinery vary with each particular application, although many common threads are evident. Heavy equipment users, typically employing oil lubricated tilting pad bearings, see many advantages including efficiency and safety in eliminating the oil lubrication system. Such a system which utilizes external lube oil pumps, piping, reservoirs, and filters also contains costly elements to install and maintain. In many cases, more heavy equipment down time 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 rotor dynamic characteristics as reasons for using magnetic bearings. A typical weighing of factors for a pipeline compressor is shown in Figure 8, Magnetic Bearing Advantages.



MBI JOB# 59007
TUNED 11-86
FDP 2-11-87

FIGURE 4
CRITICAL SPEED MAP



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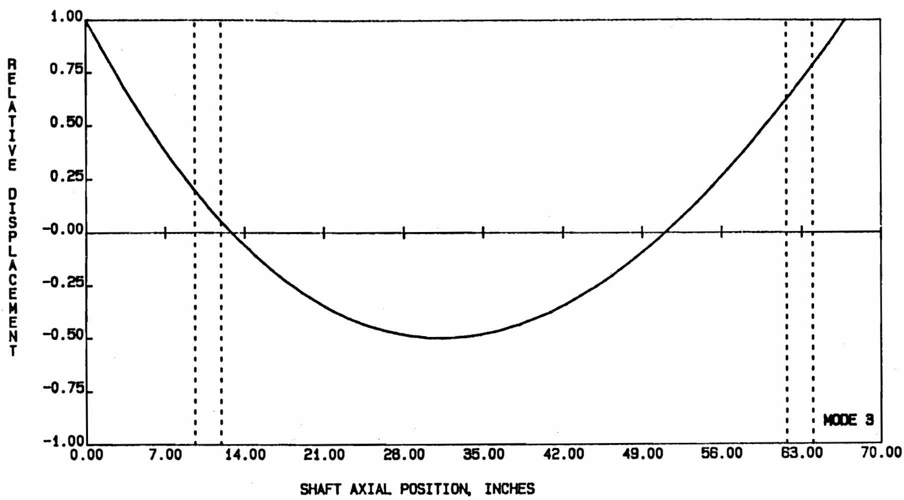
UNDAMPED NATURAL FREQUENCY (CPM): 1008
BEARING STIFFNESS (LB/IN): 1.0E4

FIGURE 5
MODE SHAPE



J59007
FDP 10-1-88

UNDAMPED NATURAL FREQUENCY (CPM): 12475
BEARING STIFFNESS (LB/IN): 1.0E4

FIGURE 6
MODE SHAPE

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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+05
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+05
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	13860.00	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

ADVANTAGES	WEIGHTING
-NO LUB SYSTEM REQUIRED	✓ ✓ ✓
-VERY LONG LIFE	✓ ✓ ✓
-MINIMAL MACHINERY VIBRATION	✓ ✓
-ACCURATE SHAFT POSITION	✓ ✓
-HIGH SPEED OPERATION	✓
-HARSH ENVIRONMENT OPERATION	✓ ✓ ✓
-NO PROCESS CONTAMINATION	✓ ✓ ✓
-REDUCED MAINTENANCE	✓ ✓
-HIGH RELIABILITY	✓ ✓
-ELECTRONIC SYSTEM INTERGRATION	✓ ✓

FIGURE 8: MAGNETIC BEARING ADVANTAGES

Current Experience

Having discussed the design process and reasons for incorporating active magnetic bearings, Figure 9, Turbomachinery Applications, shows the characteristics of a dozen magnetic bearing equipped machines which are either in service or planned for 1988 commissioning. Total operating time for this group is presently in excess of 30,000 hours. Based upon utilization estimates, it is anticipated that these machines will accumulate about 50,000 hours annually as the full set of units being constructed is placed in service. These turbomachines are owned by either Nova, an Alberta Corporation, Shell Canada Products Limited, or Trans Canada Pipelines and are located in Canada. The equipment was originally supplied by North American manufacturers either with magnetic bearings or retrofit modifications were performed to incorporate magnetic bearings into conventional units. Together, this grouping represents the largest concentration of turbomachinery operating on magnetic bearings anywhere in the world.

Experience to date continues to verify the system engineering process discussed above. Machine operation, once turned over to the operating organizations, has been trouble free. Reliable oil free operation with better efficiency, lower operating costs, and decreased vibration levels have all been achieved. Figure 9, Turbomachinery Applications, indicates a wide range of shaft rotor dynamics, machine sizes, loadings, and configurations. The design data base includes beam type and cantilever rotors, high thrust loadings, bearings operating directly in high temperature or corrosive process mediums, and repeat units of the same type. The latter are especially noteworthy in that they confirm the potential for lower cost standardized applications with predictable results.

MACHINE	ROTOR TYPE	SERVICE	DUTY	COMM	ROTOR WEIGHT LB	THRUST LOAD LB	SPEED RPM	JOURNAL DIAMETER	RATING HP	OPERATING HOURS (*)
CDP-230 COMP.	2 STAGE BEAM	PIPELINE METHANE	SEASONAL	1985	3200	12000	5250	10.6"	14650	9696
CDP-416 COMP.	4 STAGE BEAM	PIPELINE METHANE	SEASONAL	1986	280	3370	14500	3.7"	4150	9000
1B26 COMP.	1 STAGE BEAM	PIPELINE METHANE	CONTINUOUS	1986	780	4050	11000	6.5"	5540	8130
1B26 COMP.	1 STAGE BEAM	PIPELINE METHANE	CONTINUOUS	1987	780	4050	11000	6.5"	5540	2956
CBF-842 COMP.	8 STAGE BEAM	REFINERY WET GAS	CONTINUOUS	1987	1420	4590	10250	6.0"	4500	3900
3B37 COMP.	3 STAGE BEAM	REFINERY HYDROGEN	CONTINUOUS	1988	1050	4100	10230	6.0"	4500	---
RFB30 COMP.	1 STAGE OVERHUNG	PIPELINE METHANE	SEASONAL	1988	1730	15750	5200	10.0"	14250	---
5P2 COMP.	2 STAGE BEAM	PIPELINE METHANE	SEASONAL	1988	1500	10000	6800	6.0"	16600	---
5P2 COMP.	2 STAGE BEAM	PIPELINE METHANE	SEASONAL	1988	1500	10000	6800	6.0"	16600	---
5P2 COMP.	2 STATE BEAM	PIPELINE METHANE	SEASONAL	1988	1500	10000	6800	6.0"	16600	---
RF2BB30 COMP.	2 STATE BEAM	PIPELINE METHANE	SEASONAL	1988	2000	18000	5000	10.0"	29600	---
GT-51 TURBINE	1 STAGE OVERHUNG	PIPELINE	SEASONAL	1988	3530	24500	5250	8.3" 12.5"	14650	---

*AMB CABINET OPERATING HOURS AS OF APRIL 12, 1988

FIGURE 9
TURBOMACHINERY APPLICATIONS

Advanced Designs

Machines as detailed in Figure 9 have superior performance capabilities, but they do not fully exploit the possible improvements magnetic bearings can provide for turbomachinery. New machine designs are required to utilize the bearing and control capabilities in innovative ways. Turbomachinery performance depends heavily upon the dynamics of shaft and rotor design. Shaft suspension techniques have not changed markedly in decades, but magnetic bearings enable no contact operation at sustained high speeds with variable stiffness and damping coefficients and intelligent electronic control.

These characteristics can be applied to a number of turbomachinery types. Smaller compressors running at higher speeds with no oil system can save space on off shore platforms or in station modernization/upgrade situations. Higher efficiency steam turbines with no lubrication requirements are feasible. Gearboxes may be eliminated in some processes by directly matching driver and compressor speeds. Aircraft gas turbine thrust to weight improvements are possible through higher speeds and lighter weight structures. Canned pump life will be greatly enhanced by the ability of the magnetic bearing to perform in the process fluid. Fans and blowers will have bearings better able to survive harsh environments. Cascaded machinery strings for electrical generators or cogeneration can make use of the bearing capability as a transducer to damp out vibration.

Electronic control capabilities provide the features necessary to enable an intelligent bearing system. This intelligence can be applied in several ways. Machine performance can be altered adaptively depending on speeds or loads. Changes in operating conditions such as surge, unbalance, or seal wear can be recognized. Machine status, health maintenance trends, and internal bearing system self diagnosis data are all available within the electronic control system for local presentation or integration into control room functions.

Technology Trends

Technology advancements are proceeding in many areas applicable to further improving the performance and decreasing the cost of future magnetic bearing systems. Digital electronics utilizing custom chips and power semiconductors in hybrid packages will enable modular control systems to economically fulfill a wide variety of turbomachinery applications. Advances in magnetic materials and structures ultimately involving superconductivity will result in higher force density/smaller size bearings. Active magnetic bearing product enhancements will continue at a rapid pace because the basic technologies - computer simulation, control analysis, power electronics, electronic circuit packaging, and electro-magnetics are all strong areas of technology research.

Identification of Bearing and Rotor Parameters