MAGNETIC BEARING RETROFIT OF A HIGH SPEED, EIGHT STAGE COKER COMPRESSOR

by

Pranabesh De Choudhury

Senior Consulting Engineer

Andrew P. Watson

Senior Product Engineer

Elliott Company

Jeannette, Pennsylvania

Donald R. Carter

Craft Supervisor

Marathon Oil Company

Robinson, Illinois

and

Chet Farabaugh

Senior Mechanical Engineer

Magnetic Bearings Inc.

Roanoke, Virginia

Pranabesh De Choudhury has worked for Elliott Co. in Jeannette, Pennsylvania for the past 22 years in the area of Rotor Bearing Systems Dynamics. In his current position as Senior Consulting Engineer, his responsibilities include rotor bearing dynamics, bearing design and analysis, torsional dynamics, blade vibration analysis, and troubleshooting field vibration problems. Dr. De Choudhury obtained a B.S.M.E. from Jadavpur University, Calcutta, India (1963), an M.S.M.E. from Bucknell University and his Ph.D in Mechanical Engineering from the University of Virginia (1971). He has written several technical papers, and has been awarded a patent. He is a registered Professional Engineer in the State of Pennsylvania, and is a member of ASME and STLE.

Andy Watson is Senior Design Engineer for axial and single-stage centrifugal compressors at Elliott Co., Jeannette, Pennsylvania. He is responsible for all technical aspects of these compressors, including aerodynamical performance, rotordynamics and mechanical design. Besides axial and single-stage compressors, he is assigned special projects involving multistage compressors. He assists Sales/Marketing in precontract compressor selection and technical data. He also works with Manufacturing and Test Engineering on inhouse orders and consults with Elliott Service organization on technical matters concerning units in the field. Mr. Watson has been with Elliott Co. since 1981 and prior to that had nine years experience in the design and development of turbomachinery. He has a B.S. in Mechanical Engineering from Pennsylvania State University (1973) and an M.S. in Mechanical Engineering from Rensselaer Polytechnic Institute (1981). He has written several technical papers, and is a member of ASME.

Don Carter is Craft Area Supervisor with Marathon Petroleum Company. He is responsible for electrical, instrument and machine shops. He has been with Marathon Petroleum Company for 19 years during which time he has been responsible for all types of equipment and related controls, predictive and preventive maintenance programs, and maintenance craft training. Prior to this, Mr. Carter worked as a performance analysis consultant with PMC Beta Corp. in the U.S.A. and South America, and as a field equipment specialist with Shell Oil Co. involved in maintenance of reciprocating and rotating compressors, electrical and instrument controls. He has written several technical papers and journal articles.

ABSTRACT

The design, integration, and implementation of a successful project to retrofit a high speed, eight stage centrifugal compressor with magnetic bearings and gas face seals is presented. The application is removal of light gases from the top of a fractionating tower in a refinery coker unit. The justification and reasons for the selection of the coker compressor retrofitted with magnetic bearings are discussed. The technical aspects of design, rotordynamics and operation of the old oil bearing and seal design to the retrofitted magnetic bearing and gas face seal designs are compared. The tuning of the magnetic bearings is also discussed along with startup and operation of the compressor running on magnetic bearings.

The significant challenge associated with retrofitting a compressor with magnetic bearings previously running on oil bearings is addressed from a design consideration and rotordynamics point of view. The development of assembly/disassembly techniques and manufacturing methods special to a magnetic bearing retrofit are described. Also presented is the logistics of meeting the short lead time and narrow turnaround window of 12 days at the user's facility.

INTRODUCTION

Retrofitting and upgrading turbomachinery to take advantage of new technology that the machine was not originally designed for can be both an opportunity and a challenge. The opportunity was to reduce operating and maintenance costs significantly on a 20 year old, high speed, eight stage intercooled coker compressor by installing magnetic bearings and gas face seals. The challenge was first to convince the refinery's upper management to take the risk. Then follow up with a plan and team made up of engineering personnel from the compressor, bearing and seal vendors, and refinery staff to make it happen.

Besides fixed space constraints associated with an existing machine, the Engineering team had to consider the resulting impact to machine rotordynamics, performance, and operations. Experience with dry running seals is almost common place nowadays, but the application of magnetic bearings to process compressors in refineries is relatively new.

Along with the normal demands associated with hardware retrofits, this project included the following additional constraints;

- The footprints of the units in this string would not change. The positions of the compressor casing, piping, and gearbox were fixed and new hardware must fit in the existing space.
- Accomplish compressor disassembly, hardware installation, magnetic bearing tuning, a test run on air, and ready to come back on process in 12 days.
- Make the magnetic bearing design similar to a design being developed for a new hydrogen recycle compressor to be installed at the same refinery.

This presentation first develops economic and technical justifications for this project. Second, a background of magnetic bearing operation and control is discussed. Design details are then presented followed by a comparison of rotordynamics between originally installed oil bearings and the new magnetic bearing design. Finally, a discussion of the preshutdown work and the results of the installation, startup, and operation.

JUSTIFICATION

Technical

The coker compressor was chosen as the best candidate for the new magnetic bearings for several reasons. The first reason being the adverse condition under which the machine operates. The coker operates on cycles of about 20 to 22 hr. The operating conditions, pressure, and flow of the machine vary widely within each cycle. When the coke drums are switched, the pressure is lowered and the flow falls off pushing the machine toward surge. As the drum switch is completed, the flow comes back while the pressure is still low, driving the machine into stone wall. This is also an intercooled machine, so the variations in the amount of condensate at off-design points effects second section performance and could push the machine further outside the normal operating envelope.

The second reason this machine was chosen was the age of the machine and instrumentation. The machine was due for an overhaul, and the 20+ year old instrumentation needed to be upgraded to meet today's standards. This machine had been picked as the next machine to be retrofitted with dry gas seals because of problems with the mechanical face seal. The addition of magnetic bearings would fit within the scope of work.

Additional technical benefits are as follows:

- Reduced gas and oil emissions, therefore environmentally a better machine
- Low mechanical losses. Less than eight horsepower total in seals and bearings compared to 104 hp loss for original oil bearings and seals, and oil pump power to supply seals and bearings
- Better rotordynamics such as being able to control the radial bearing dynamic stiffness and damping
- Diagnostic capabilities that are not available on a conventional oil bearing
  - Continuous bearing load readout
  - Ability to indicate surge
  - Ability to monitor balance piston performance
- Reduced maintenance cost
- Reduced operating cost
- Eliminating seal oil controls and associated hardware
- Eliminate the eductors to carbon cases on the steam turbine
- Upgrade the instrumentation to meet specifications

Return on Investment

The total cost of the retrofit as defined in the work scope was \$973,309. This price included remodelling the control room, which cost \$123,560. The cost for the turnaround on this machine without the seal and bearing upgrade is estimated at \$162,750. The instrument upgrade including new instrument panels and hardware installed was \$258,653. If these prices are taken out the total cost to upgrade to magnetic bearings and dry seals was \$428,346. The savings as calculated for magnetic and dry seals is \$941,426. The return on investment analysis, shown in Table 1, lists the payback period for this project as less than seven months. Other financial indices are also shown that indicate the definite cost benefit of this retrofit. A detailed list of the savings in operating costs by using magnetic bearings and gas face seals in place of oil bearings and seals is shown in Table 2.

SCOPE OF SUPPLY

Preshutdown Scope of Supply

- Send spare rotor to compressor vendor to unstack impellers and have them brought to standard, including inspection, heat treat, grinding bores, overspeed, and individual balance.

Table 1. Return on Investment Summary.

PRODUCT LIFE –	20 YEARS
PAYOUT –	0.55 YEARS
DISCOUNTED PAYOUT –	0.63 YEARS
	AFTER TAX
DCF–ROR –	185.87%
NPV AT 5% –	\$11,117,130
NPV AT 10% –	\$7,203,180
NPV AT 15% –	\$5,036,710
PV/INVEST RATIO AT 15% –	12.76
PERCENT CF FOR 15% ROI –	7.84%

Table 2. Operating Costs with Oil Bearings.

DESCRIPTION OF ITEMS	COST/YEAR
UNIT DOWN TIME TO REPLACE SEALS AND BEARINGS	$500,000
MAINTENANCE COST-MATERIALS AND LABOR	$16,200
ENERGY SAVINGS-TOTAL 106 hp- (3657 lb/hr STEAM)	$128,141
TREATMENT FOR OIL TO SEWER- AVERAGE 170 gal/day-@ $.70	$42,157
OIL USAGE-AVERAGE/DAY 170 gal	$142,094
PURCHASE AND WAREHOUSE OIL- 8 hr/week	$10,400
OPERATOR'S TIME TO HANDLE OIL- 2 hr/shift	$54,750
LAB. TIME TO PICK UP SAMPLE- 10 hr/week	$13,000
STEAM USAGE - CARBON SEALS AND EDUCTOR (420 lb/hr STEAM)	$14,716
OIL DRUM DISPOSAL	$19,968
TOTAL:	$941,426

- Make a new shaft and stack with new thrust disk, magnetic bearing laminations, auxiliary bearing landing sleeve, and existing balance piston, shaft sleeves and impellers.
- Balance rotor and ship to bearing vendor for preliminary tuning.
- Supply magnetic bearing control and UPS cabinets.
- Supply new inlet and discharge endwalls and thrust bearing housing.
- Furnish magnetic radial bearing housings, magnetic thrust bearing stator assemblies, auxiliary bearing assemblies and axial sensor assembly.
- Furnish abradable gas side seals and new coupling side shaft end seal.
- Furnish new tandem dry gas seals and filters.
- Supply new dry high performance disk couplings.
- Modify coupling guards to accept dry running couplings.
- New instrumentation to include local and remote panels capable of trending and storage and all necessary hardware such as smart transmitters, vibrator, level and flow transmitters, RTDs.
- New equipment manuals and installation tooling.

Shutdown Scope of Work

- The steam turbine driver would be overhauled and the carbons replaced with steam seals, thus eliminating the eductor piping and eductors.
- Gear box removed to shop for inspection and overhaul.
- Install new rotor, endwalls, magnetic radial bearing housings and magnetic thrust bearing housing assembly.
- Run unit on air with open inlet and discharge connections to dynamically tune bearings.
- Attach process piping and bring unit back online.

MAGNETIC BEARING BACKGROUND

Control

While much has been written regarding the basic operation of magnetic bearings [1], there is still little information about the control and tuning aspects associated with this technology. A basic control loop for one axis of control is shown in Figure 1. The control loop itself is shown by bold solid lines with diagnostics and system security features making up the remainder of the system. It is important to note that the actual rotor, or compressor shaft in this case, closes the loop. Control of the system is, therefore, based upon the position of the rotor, which is detected by position sensors mounted near the bearing. The sensors produce an error signal proportional to the rotor's deviation from a referenced center position. The signal is processed through a proportional integral derivative (PID) control loop card that develops a correction signal to recenter the rotor. Depending on the method of force actuation, the correction signal is either routed directly to the amplifier command card (Class A), or the signal is linearized and then routed to the amplifier command card (Class B). The correction signal is then conditioned and fed into the power amplifier that generates the current to produce the electromagnetic force necessary to move the rotor back to its centered position.

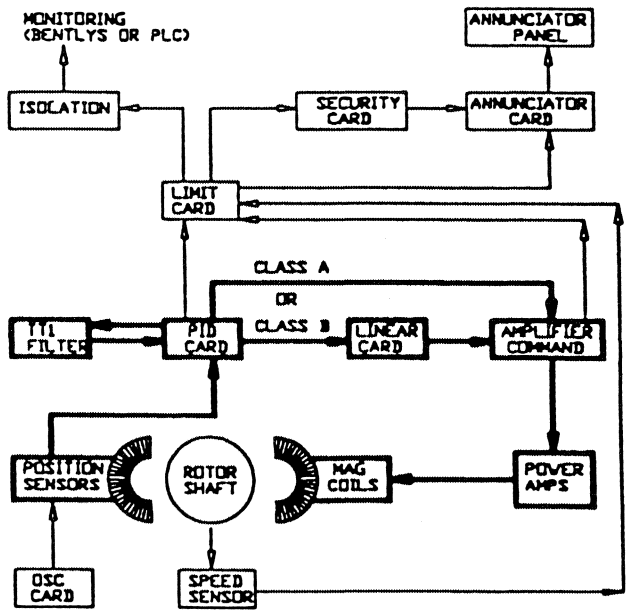

Figure 1. Control System Functional Block Diagram.

The two force actuation methods defined above (Class A *vs* Class B) offer tradeoffs regarding dynamic force capacity and losses associated with the magnetic bearings. A typical magnetic bearing arrangement is shown in Figure 2 consisting of two pairs of opposing electromagnets that provide control along two axes, "V" and "W." In Class B control only one electromagnet of the opposing pair is used at any given time. Therefore, the top two magnets support the static weight of the shaft and the bottom magnets are used only to control dynamic loads encountered during operation. In Class A control, opposing electromagnets are provided with a bias current so that they are always pulling against each other. Under static conditions, the upper magnets support the weight of the shaft with the bottom magnets providing opposing forces on the shaft. Dynamic loads encountered during operation are controlled by adding current to one electromagnet, while simultaneously, subtracting current from the opposing magnet. It can be shown that Class A results in a linear relationship between net bearing force and change in rotor position, for small motion, unlike Class B that requires a linearizer to develop the same relationship. Class A develops higher dynamic force capacity,

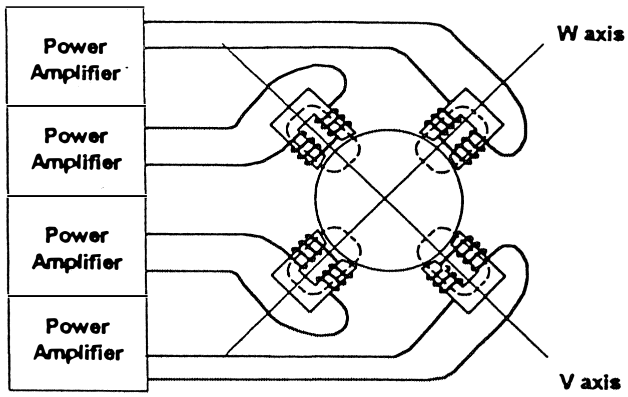

Figure 2. Radial Magnetic Bearing Configuration.

since there are two magnets and two power amplifiers acting along a given axis instead of one as in Class B.

Though there is no friction present, magnetic bearings have losses that are attributed to windage from the rotating components, I^2R losses in the windings, and hysteresis and eddy current losses in the rotor. These losses are typically an order of magnitude less than those of conventional oil bearings. In magnetic bearings the losses also depend on the type of force actuation. The higher dynamic force capacity offered by Class A results in higher power consumption (losses) *vs* Class B. Using two electromagnets at the same time, instead of one, increases the I^2R losses in the winding and also increases the number and magnitude of flux reversals in the rotor resulting in additional rotor losses for Class A. The magnetic bearing losses are dissipated by both conduction and by the addition of cooling air to the bearing housing.

Stiffness and damping characteristics of the bearing are determined by processing the position error signal in the PID control card. The gain and phase characteristics of the input error signal are modified to produce an output signal that defines the control loop transfer function and the bearing stiffness and damping properties [2]. Typically, an acceptable transfer function is defined in the design phase in conjunction with the rotordynamics analysis. Following installation, the transfer function is "tuned" to achieve compatibility with the natural frequencies of the shaft and stable operation of the machine.

Tuning

Since the shaft closes the control loop, final system integration and compensation is best established after the machine is assembled and the magnetic bearings are used to levitate the actual shaft. Usually control loop compensation, or tuning, is performed at the OEM plant to ensure stable operation throughout the speed range for mechanical and performance testing, and may be adjusted during field installation to account for any changes in bearing housing support stiffness, and effects caused by aerodynamic loads and coupled machinery.

Tuning consists of setting the control loop gain (stiffness) to allow adequate separation between the operating speeds and the undamped critical speeds. Then by using the magnetic bearings to "buzz" the shaft to verify the resonant frequencies, the transfer function is set up to ensure that phase lead (positive damping) is present at these frequencies. Magnetic bearing characteristics are dependent upon the excitation frequency and not the shaft rotational speed as in the oil bearings. The higher frequency shaft modes, that are beyond the operating speed range, but within the controller bandwidth, can become excited and adversely affect stability, and can lengthen the tuning phase.

In addition to modifying the control loop transfer function, the tuning phase includes various procedures associated with magnetic bearing system integration. These tuning procedures include:

- *Oscillator Setup.* Adjust components used to excite the radial and axial position sensors to minimize current draw.
- *Sensor Adjustment.* Compensate for length of lead wire between the control cabinet and the machine, and verify sensor sensitivity.
- *Flux Tuning.* Compensates for the negative stiffness inherent in a magnetic bearing.

DESIGN

This compressor was originally designed with pressure dam oil bearings and a mechanical face seal. The decision to modify the unit with magnetic bearings and gas face seals required radical change of the bearing and seal arrangement within the constraints mentioned earlier. The first step was to size the magnetic bearings based on the rotor weight and thrust load. The old and new bearing sizes, loads and capacities are shown in Table 3. The thrust bearing was oversized intentionally as a factor of safety. The thrust loads in off-design conditions, particularly surge, can be much higher than that calculated at design point. This is normally not a concern with oil bearings because oil bearings are designed to handle momentary overloads up to twice their operating capacity. The magnetic bearings on the other hand do not have this overload capability. When the capacity of the bearing is exceeded, the amplifiers saturate and magnetic force is overcome by the thrust force. The reduction in axial gap when this occurs will trip the unit. The radial bearing size does not require the same factor of safety, since the radial load is more predictable.

Table 3. Nominal Bearing Data.

	OIL BEARING SYSTEM	MAGNETIC BEARING SYSTEM
RADIAL BEARINGS		
BEARING DIAMETER (in) –	4.0	7.5
BEARING LENGTH (in) –	1.25	3.25
AVG. LOAD PER BRG. (lb) –	401.8	506.3
BEARING CAPACITY (lb) –	1250	1350
RADIAL CLEARANCE (in) –	0.0035	0.020
POWER LOSS (hp) –	18	1
THRUST BEARING		
THRUST DISK O.D. (in) –	6.74	14.0
THRUST BRG. AREA (in^2) –	15.6	125.7
RATED THRUST LOAD (lb) –	1959	1959
THRUST BEARING CAPACITY (lb) –	6240	8200
ACTIVE SIDE CLEARANCE (in) –	.001	.025
INACTIVE SIDE CLEARANCE (in) –	.012	.025
POWER LOSS (hp) –	26	2
AUXILIARY BEARING		
RADIAL CLEARANCE (in) –	N.A.	.0080
AXIAL CLEARANCE (in) –	N.A.	.0085

The redesign of the endwall to accommodate the tandem gas faced seal was straightforward, since retrofitting process compressors having mechanical face seals with gas face seals has become commonplace. The only change in the gas face seal was the addition of an outboard labyrinth seal to prevent the magnetic bearing cooling air from escaping through the secondary vent. The endwalls were then designed to incorporate the radial magnetic bearings. The size of the thrust bearing required a separate housing for it. The journal end magnetic bearing design required the shaft to be extended about 3.5 in. The original gap between the compressor and gear was 5.0 in, so the shaft would fit in but the gear would have to be removed to remove the coupling hub from the shaft.

The thrust end design is shown in Figure 3 and the journal end design is shown in Figure 4. As shown in both figures, the gas side laby and gas face seal are the same for each end of the compressor. The shutdown offered the opportunity to replace the gas side laby with an abradable seal to reduce buffer usage. The radial bearings are basically the same also. Descriptions of the radial and thrust bearings are given below.

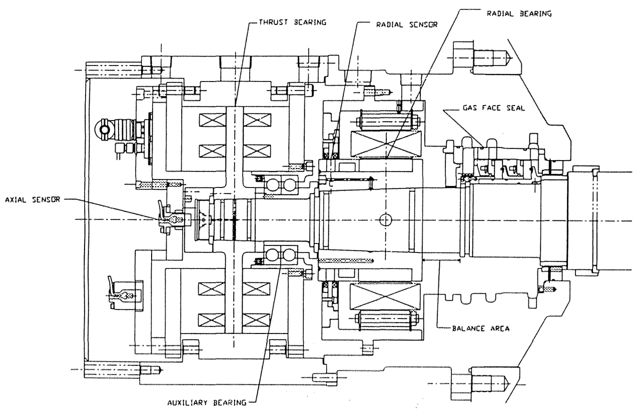

Figure 3. Thrust End Design.

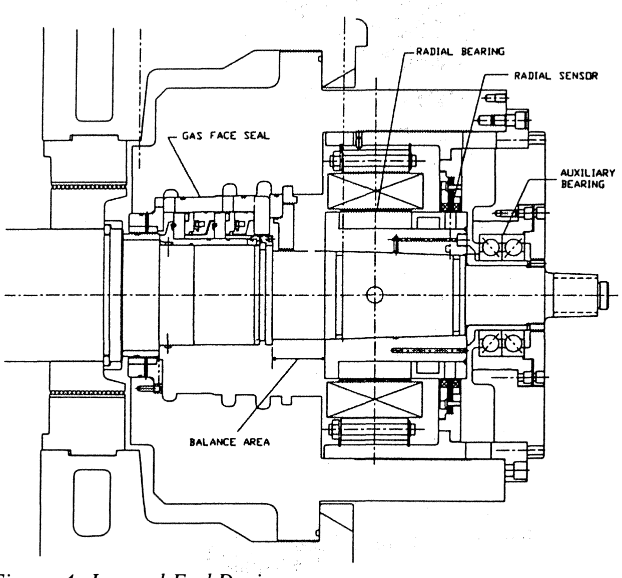

Figure 4. Journal End Design.

The space on each end the shaft, as shown in Figures 3 and 4, between the gas face seal sleeve and the magnetic radial bearing rotor assembly (from now on called the rotor hub), is required for at-speed balancing the rotor on oil bearings. The magnetic bearing laminations could be run on oil bearings, but the 7.5 in outer diameter bearings would require large pedestals in the at-speed balance that would not have sufficient sensitivity for a rotor of only 1000 lb. Outboard of this area is the radial bearing consisting of rotor laminations and stator assembly.

The rotor laminations, as shown in Figures 3 and 4, are hydraulically fit on a taper on the shaft. The rotor hub has a hydraulic fitting on the outboard face to supply radial hydraulic pressure. There are O-rings on the shaft under each end of the rotor hub. A hydraulic assembly/disassembly fixture was designed to supply the axial force to push the rotor hub on and as a catcher for removal. The rotor hub is positioned axially by an insert on the fixture that was custom ground to hit the shoulder on the shaft when the rotor hub was in position. Each rotor would have its own insert since the rotor hub position could vary slightly from rotor to rotor. The rotor hub is held axially by two shaft half keys that have a slight interference fit between a groove in the shaft and the rotor hub. The keys are ground to final thickness when the running position of the rotor hub is set. The keys are retained radially by the landing sleeve. The rotor hub is made of a solid metal base with stacked iron laminations on the inboard side for the magnetic bearing and a smaller stack on the outboard side for the radial sensor. These stacks are separated by a spacer. The rotor laminations are used to reduce the hysteresis and eddy current losses within the magnetic bearing.

The top quadrants, shown in Figure 2, of the magnetic radial bearings have RTDs built in to measure temperature rise due to I^2R and other losses. High temperatures will degrade the insulation of the windings. To prevent excessive heat build-up, cooling air is injected inboard of the bearing, shown in Figures 3 and 4, and flows through the clearance gap and out a vent hole on the outboard side. The radial bearing housings have a sliding fit into bores in the endwalls. The housings are then attached to the endwalls using an axial flange on the outboard end of the housing on the journal end and mid housing on the thrust end. The mid housing flange is needed to exit the radial bearing wires inboard of the thrust bearing housing.

The thrust bearing housing, shown in Figure 3, is cantilevered off the endwall. It has an inner thrust bearing stator assembly that also includes the auxiliary bearing assembly. Originally the auxiliary bearing was outboard of the thrust disk for easy access, but rotordynamic concerns required it to be moved inboard. The inner and outer stator assemblies are axially bolted to the thrust bearing housing and doweled after the auxiliary bearing is centered on the shaft. Shims are placed between the assemblies and the thrust bearing housing to obtain a nominal 25 mil clearance on each side of the thrust disk. The shims on the inboard thrust assembly also sets the 0.007 to 0.010 in axial clearance on the inboard side of the auxiliary bearing. The inner thrust assembly must be mounted when the thrust bearing housing is attached. The inner and outer thrust assemblies are made up of windings which are equally spaced circumferencially and apply an axial attractive force to the thrust disk when activated. Unlike the rotor hubs, the thrust disk does not contain laminations but is solid steel. The thrust disk is assembled and disassembled hydraulically. The radial hydraulic pressure is fed in through a drilled hole in the center of the shaft. O-rings on the shaft at each end of the disk contain the hydraulic pressure. The axial force to push the disk on and catch the disk when it is removed is provided by a fixture that attaches to the end of the shaft. The thrust disk position is initially set by a pull-up amount from a cold position without O-rings installed. Once this position is found, a landing shoulder on the inboard side of the disk is ground to thickness and is used to locate the disk on succeeding installations. The disk is held axially by shaft keys that are ground to give a light interference fit between the disk and groove in the shaft. The keys are retained radially by the rotor end cap that screws onto the end of the shaft. This cap also contains laminations in the center that provide a target for the axial position sensor contained in a bracket attached to the end of the thrust bearing housing.

The auxiliary bearings, shown in Figures 3 and 4, are face-to-face angular contact ball bearings. These bearings are preloaded to prevent rotation due to windage. The purpose of the auxiliary bearings is to support the rotor when the unit is shutdown or as emergency backup bearings. The radial clearance of the auxiliary bearings is about half that of the radial magnetic bearings to prevent contact between the magnetic bearing parts during shutdown or in the event the rotor drops due to the magnetic bearings losing or having insufficient power. The auxiliary bearings normally can withstand three to five full speed drops of the rotor before having to be replaced. The thrust end auxiliary bearing, shown in Figure 3, also takes up the axial thrust when the magnetic bearings lose power. To prevent the thrust assemblies from contacting the thrust disk, the axial clearance on the inboard side and outboard side is held to less than half the axial clearance of the magnetic thrust bearing. Underneath each auxiliary bearing is a hardened landing sleeve that contacts the auxiliary bearings if a drop occurs. The auxiliary bearings and landing sleeves also support the rotor weight when the unit is shutdown.

ROTORDYNAMIC ANALYSIS AND CORRELATION WITH TEST

A summary of the rotordynamic analytical results is shown in Table 4. Results have been presented for the existing rotor bearing system that had been operating for the past 20 years on pressure dam journal bearings at both ends, and with oil seals. Also presented are the results of the rotordynamic analyses on magnetic radial bearings, thrust bearings, and gas seals. Two sets of analytical results are presented for the system on magnetic bearings, the first one is with radial magnetic bearings as Class B, as designed configuration, and Class A, as tuned radial bearing configuration.

Table 4. Summary of Rotordynamic Analysis Speed Range: 10400 to 12500 RPM.

ITEM	ORIGINAL ROTOR DESIGN, PRESSURE DAM BEARINGS		MAGNETIC BEARING CONFIGURATION AS DESIGNED	MAGNETIC BEARING CONFIGURATION AS TUNED
BRG. SPAN (in)	63.563		68.04	68.04
SHAFT LENGTH (in)	76.613		89.868	89.868
STATIC BEARING REACTIONS (lb)				
THRUST END	390.64		528.55	528.55
JOURNAL END	412.94		484.11	484.11
RIGID SUPPORT CRITICAL SPEEDS (cpm)				
1	5452		4587	4587
2	20970		14422	14422
3	37509		21119	21119
4	–		35619	35619
UNDAMPED CRITICAL SPEEDS (cpm)				
	VERTICAL	HORIZONTAL		
	4797	3703	3028	3017
	16252	9969	5612	6722
	25050	18952	9080	9363
	–	–	16937	17249
UNBALANCE PEAK RESPONSE IN RPM (AMPLIFICATION FACTOR)				
THRUST END	3300	3300	3200	3200
	(2.50)	(3.02)	(2.09)	(2.69)
	4900	9500	7600	8600
	(7.37)	(1.04)	(1.96)	(2.86)
	14500	–	16400	16800
	(1.07)	–	(10.96)	(9.70)
JOURNAL END	5300	3700	3200	3200
	(4.63)	(3.34)	(2.15)	(2.48)
	9000	8500	7400	8400
	(1.84)	(1.56)	(2.08)	(2.48)
	19500	17500	16400	16800
	(2.60)	(2.68)	(10.08)	(9.36)
STABILITY – LOG DECREMENT @ 12500 rpm				
		.011	.261	.306

In Table 4, the bearing span has been presented of the rotor bearing system on pressure dam journal bearings, the total shaft length, the total rotor weight, and the static bearing reactions. These have been compared to the rotor bearing system on magnetic bearings. It shows that the bearing span with rotor on magnetic bearings was increased by 4.5 in, and the total rotor length increased by 11.0 in over the rotor on pressure dam journal bearings, and with oil seals. The total rotor assembly weight also increased by 209 lb. This is due to additional space and weight required by the magnetic radial, thrust bearings, and the gas seals.

In the rotordynamic summary are listed the rigid support critical speeds, the undamped critical speeds, the peak responses that include bearing damping, due to rotor unbalance and the amplification factors associated with each peak response. The results of the stability studies in the form of logarithmic decrement (log dec) associated with first forward even mode is presented at the maximum continuous speed of 12500 rpm.

A rotordynamic concern associated with magnetic bearings is the axial location of the sensors with respect to the bearings. Rotor mode shapes were analyzed to ensure that nodal points are not located at the sensor for modes below running speed. As shown in Figure 5, the fourth mode has a node at the radial sensor on the thrust end of the machine. Fortunately, this mode is above the operating speed range.

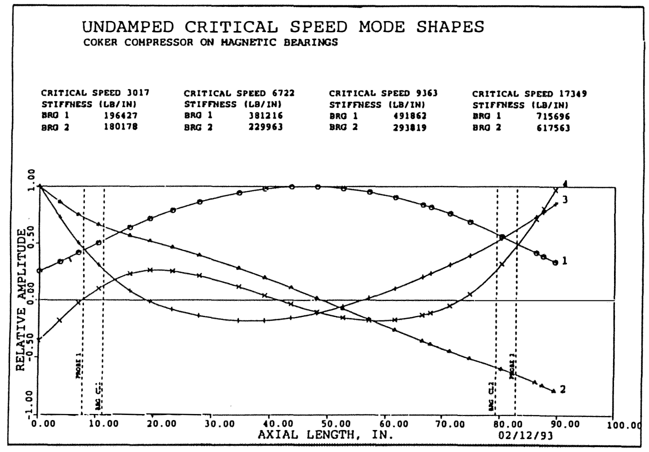

Figure 5. Undamped Mode Shapes.

As shown in Table 4, due to the increased bearing span, the first rigid support critical speed associated with magnetic bearing was lower than with the pressure dam bearings, also the higher critical speeds were lower due to the increased overhung and weight associated with the magnetic bearing thrust collar. The undamped critical speeds with magnetic bearings, as expected, are lower than those associated with pressure dam bearings due to lower bearing stiffness of magnetic bearings over the fluid film bearings. For both rotor bearing systems, first peak responses are seen well below the operating speed range of 10400 to 12500 rpm. However, a first peak response is predicted with pressure dam bearings at a speed higher than those associated with magnetic bearings at the journal end bearing. Well damped second peak responses are seen using both pressure dam and magnetic bearings below the minimum continuous speed. The as tuned magnetic bearing second peak response is slightly higher than the one associated with as designed magnetic bearing. The third peak responses are well above the maximum continuous speed for both pressure dam and magnetic bearings. While evaluating the rotordynamics with magnetic bearings in the design stage the requirements associated with API 617 5th Edition [3] were applied. The rotor bearing system with pressure dam bearings is predicted to be marginally stable. The predicted stability associated with magnetic bearing is much higher than with pressure dam bearings. The bearing stiffness and damping coefficients associated with radial magnetic bearings as designed and as tuned are shown in Figure 6. The correlation of the peak responses is shown in Figure 7 as predicted relative to the test data obtained with the as tuned magnetic bearings at the thrust end radial probe. The correlation between the observed and calculated

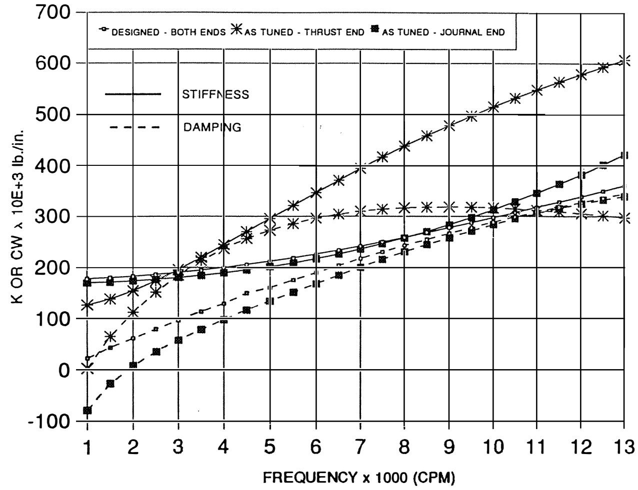

Figure 6. Radial Bearing Stiffness and Damping Coefficients.

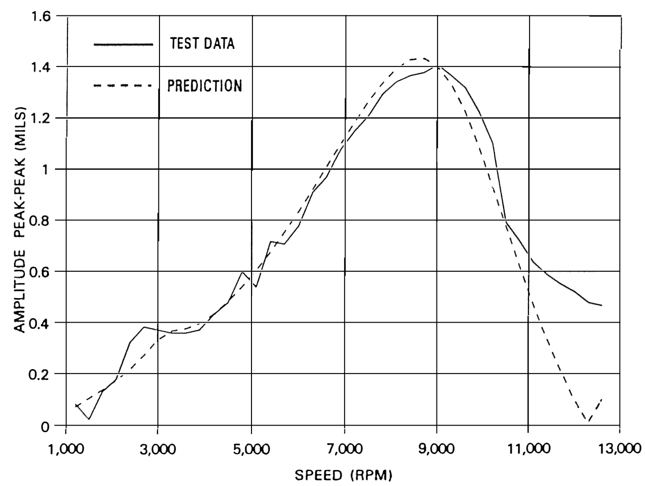

Figure 7. Correlation of Vibration Test Data with Prediction.

responses is good. A frequency plot is shown in Figure 8 at the thrust end at the maximum continuous speed of 12500 rpm. The rotor is stable as evidenced by the absence of subsynchronous frequency.

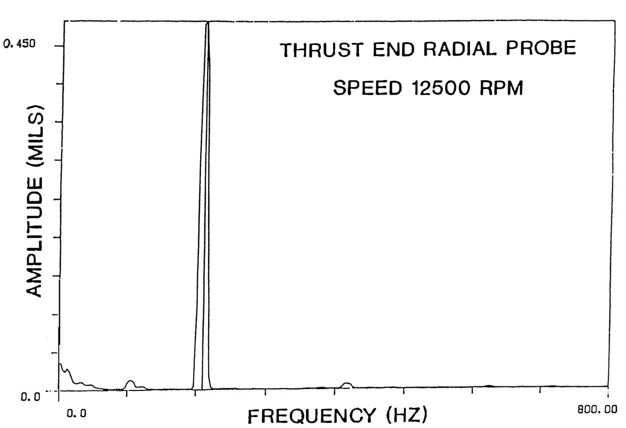

Figure 8. Frequency Plot at 12500 RPM—Thrust End.

INSTALLATION

Preliminary Tuning

Because of time constraints, it was decided that a certain amount of tuning could be performed in advance to minimize tuning time during the turnaround. The rotor was going to be available following at-speed balance and could be used to close the control loop. This was done using the test rig, shown in Figure 9, at the magnetic bearing manufacturer's facility, which was modified to levitate the compressor shaft using the bearings built for the job. Activities occurring during this preliminary tuning phase included:

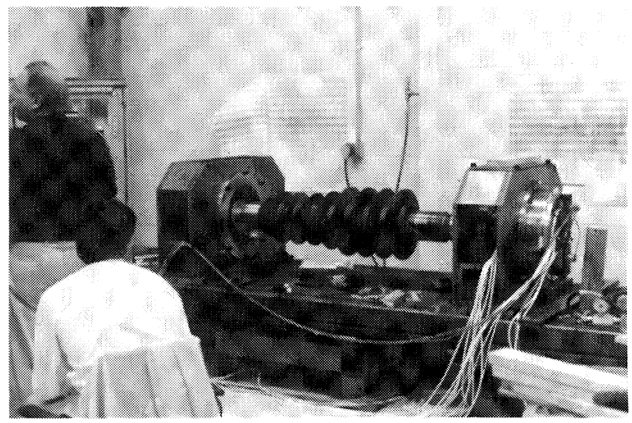

Figure 9. Preliminary Tuning at the Magnetic Bearing Manufacturer's Facility.

- Assembly of the shaft in pedestal mounted magnetic and auxiliary bearing assemblies.
- Integration of a magnetic bearing control system and power supply.
- Levitating the shaft and adjusting bearing stiffness to provide adequate separation between operating speeds and undamped critical speeds.
- Using the magnetic bearings to "buzz" shaft to identify all shaft resonant frequencies within the bandwidth of the controller.
- Modifying the controller transfer function to ensure stability.

The initial (as-designed) open loop transfer function for this application is shown in Figure 10 as Bodé plots displaying gain and phase *vs* frequency. Phase near the calculated first bending mode frequency of approximately 150 Hz (9000 rpm) is set at 36 degrees. In the tuning process, the control engineer is faced with a tradeoff between the need to limit the control loop bandwidth to eliminate adverse effects from electrical noise and higher frequency shaft modes *vs* maintaining phase lead (positive damping) at the resonant frequencies. Limiting bandwidth is done by decreasing the gain at higher frequencies, but this also drags the phase down resulting in lower damping at the operating frequencies and negative damping at the higher frequencies. To overcome this, various methods including notch filters are used at the higher frequencies to minimize gain at specific resonant frequencies without dramatically affecting the phase at the lower frequencies.

The preliminary tuning process identified various higher frequency shaft modes that, when excited, would cause the shaft to go unstable. The effects of wide *vs* narrow controller bandwidth with various combinations of notch filters were evaluated at this time. Due to the asymmetry of this shaft it was necessary to develop different transfer functions for each end of the machine. As shown

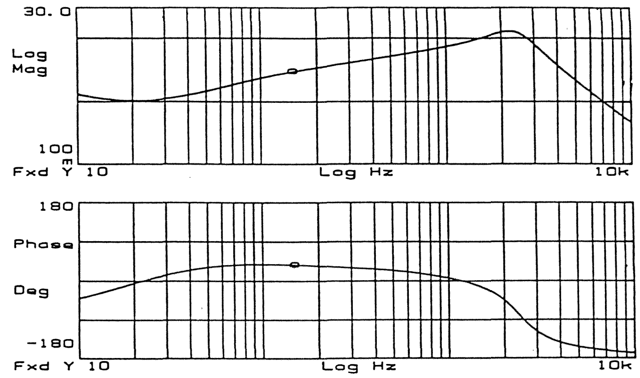

Figure 10. Proposed Controller Transfer Function.

in Figure 11, the resulting preliminary thrust end radial bearing transfer function has approximately 42 degrees phase at 158 Hz and notch filter at 1.3 Khz. The preliminary journal end transfer function with 42 degrees phase at 158 Hz and a notch filter at 800 Hz is shown in Figure 12.

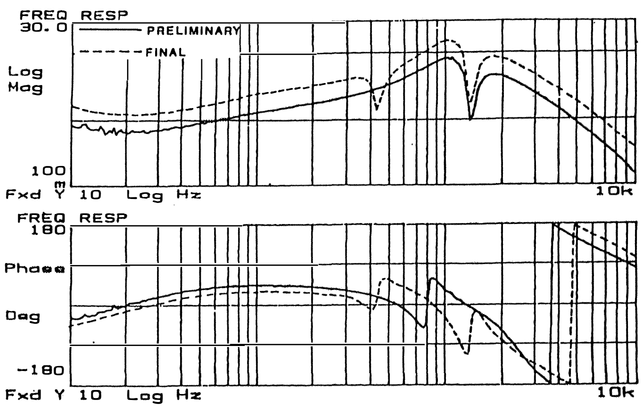

Figure 11. Thrust End Radial Bearing Transfer Function.

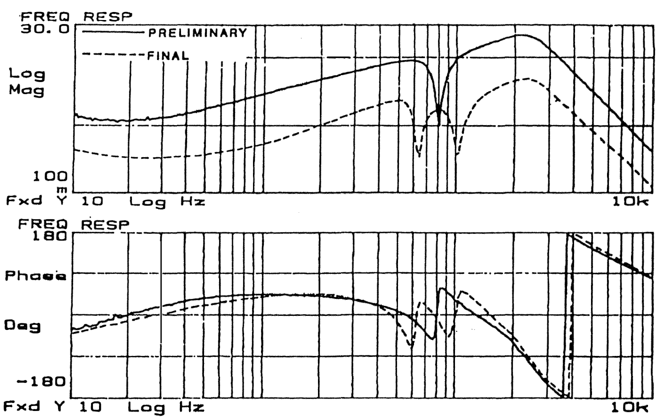

Figure 12. Journal End Radial Bearing Transfer Function .

It was also decided that additional tuning activities could be performed at the refinery during the initial teardown phase of the turnaround. While the compressor was being disassembled, an attempt was made to adjust and calibrate the sensors to account for the long lead length of wire between the unit and the control cabinet. In hindsight, this activity could have been skipped, since the results were questionable and the sensors had to be recalibrated after the bearings were installed in the machine.

Turnaround

To accomplish the work required for the upgrade of the regular coker compressor during the two-week outage was a huge task. There were several preshutdown jobs, besides those listed under *Scope of Supply*, done prior to shutdown. They were as follows:

- All instrumentation and the monitoring system, both local and remote, were mounted.
- The magnetic bearing panel and remote monitoring system panel were located in the control room approximately 500 ft from the compressor.
- All necessary utilities were hooked up—power and air.
- Conduit and wiring was run, including seven runs to the control room.
- The small bore pipe to the filters was prefabricated for the dry gas seal system.
- Old piping and instruments were marked for demolition once turnaround started.
- New operating procedures were written to include log sheets, startup, and shutdown.

The unit shutdown allowed 12 days for the retrofit to be accomplished. This 12-day span was broken into three basic stages. The mechanical completion was five days in duration. Four days were then needed to do the final bearing tuning, instrumentation loop check, and air run. The final three days were to ready the coker, including compressor piping, and to go back online. The preshutdown work had gone well; almost every phase had been completed before the unit shutdown.

The mechanical part of the job began on schedule. The machine was isolated, locked, and tagged out, following the correct procedures. The suction and discharge pipes were removed; the coupling bearings and seals were removed. While the compressor was being torn down, the steam turbine and gearbox were removed to the shop for overhaul. The steam turbine was retrofitted with a new set of steam seals and a high performance dry disk style coupling. The gearbox was inspected, overhauled, and fitted with the high performance disk couplings.

The overhaul of the compressor was proceeding, the case was split, the old rotor and diaphragms removed, cleaned, and inspected. After the cleaning and inspection had been completed, the diaphragms were reset. A mandrel had been machined prior to shutdown to be used with a dummy set of seals acting as centering rings to align the new end walls. With the lower endwalls installed, the rotor without magnetic bearing rotating parts was placed in the lower case and was set in running position. The rotor was supported by extensions to the shaft ends and the dummy seals were removed. The top end walls and casing were then replaced. The gas side seals and dry gas seals were installed next. The rotor hubs and shaft keys were installed next, aligning the match marks. The landing sleeves were installed by heating the sleeve in a rod oven. Next, the radial bearing stators were installed. After the radial bearing housings were installed, they were temporarily wired so the check out could start on the electrical end. The inboard thrust stator was positioned in the thrust bearing housing. The thrust stator was axially positioned by using shims. The thickness of the shim pack was determined from measurements taken after the rotor was in running position to obtain the required clearance on the inboard sides of the auxiliary and thrust bearings. Next, the thrust disk was installed with match marks properly aligned. The outboard thrust stator was installed with a shim pack to obtain proper clearance on the outboard thrust bearing. Finally, the sensor bracket containing the axial position and speed sensor and reference sensor was installed. With the thrust now wired up, the final wiring check could be made.

The rotor was ready for leviation; a term new to the refinery personnel. As the final static tuning checks were made, the gear box and turbine were set and the alignment done. The alignment was accomplished with the use of a laser alignment device.

Site Tuning and Air Run

The PID control cards modified during the preliminary tuning phase were loaded in the control cabinet installed at the refinery. The compressor shaft was again levitated and excited using the magnetic bearings to check that the transfer function was acceptable due to changes in support stiffness and the effects of coupling the unit.

The completion of mechanical, electrical and instrumentation phases, including static tuning moved the machine to the next step. The compressor would be operated on air, with open inlet and discharge nozzles, to perform the dynamic tuning. The air run was also used as a shakedown test to check shaft vibration, ensure clearances were properly set and the compressor and accessories were assembled and operating properly before bringing the coker back on line. Operating parameters were established, speed, temperature, etc, and the machine was ready to roll. The safety shutdown systems had been checked to insure proper operation. Dry clean instrument air had been temporarily hooked to the buffer gas filters, a positive pressure and flow would have to be maintained to ensure safe operation of the dry gas seals. The startup procedures were followed and the machine was brought up to slow roll, approximately 500 rpm turbine speed. Dynamic tuning could now proceed.

After starting the unit, it was realized that the radial dynamic loads were more than expected causing over-current in the journal end radial bearing that tripped the unit off-line at 172 Hz (10,300 rpm). With the unit operating in deep stonewall due to the open discharge on each section, it appeared that the dynamic loads were primarily aerodynamic. The inlets were orificed, increasing the pressure ratio of each section, to obtain more uniform aerodynamic loading. After the third unsuccessful attempt to bring the unit up to speed, it was decided to switch to Class A control to provide higher dynamic force capacity to control the dynamic loads present. The final tuning of the radial thrust and journal end bearings are shown in Figures 11 and 12, respectively.

In Class A control the compressor was successfully started and operated to 12,000 rpm. During this run there was 4.0 mils peak-to-peak (pk-pk) vibration. With the unit running, it was evident that unbalance also contributed to the excessive dynamic loads. Subsequent runs were made to field balance the rotor. This was accomplished by adding weights (setscrews) to the rotor end cap that also contains the targets for the axial position and speed sensors. Following field balance, rotor vibration at 12,000 rpm was recorded at 0.6 mils on the journal end and 0.3 mils on the thrust end.

As machine operation stabilized, magnetic bearing temperatures required that additional cooling air be supplied to the bearing housings. This was primarily due to the increased magnetic bearing losses associated with switching from Class B to Class A control. A summary of magnetic bearing losses and the conventional oil bearing losses is provided in Table 3.

The piping that was used for the air test was removed and the process piping reinstalled. Final alignment readings were taken and the compressor as well as the coker unit as a whole were readied for to return to process.

Operation with Process Gas

The unit shutdown was completed and returned to operations by the Maintenance Department. The unit operators had received training and had a written procedure for startup and operation of the compressor. Part of the contract with the compressor manufacturer and bearing manufacturer was to supply startup coverage for the machine until the coker had completed one complete cycle. This took approximately 24 hrs.

The machine was started and brought up to minimum governor using the startup procedure. The comparison of rotor position and voltage levels were very similar to those observed on the air run. The unit was brought on charge and the compressor moved to a normal operating range. Minor adjustments were made to seal buffer gas flows and the cooling air to the bearings. Normal operation and speed was achieved. The machine and all systems were operating flawlessly. The next test for the new system was the drum switch.

The drum switch occurred approximately 24 hr into operation; the machine handled the switch well. The new monitoring system had the capability to trend and plot the machine conditions and operating conditions, as the coke drum cycle occurred. The pressures lowered and raised as the flow changed. The ability to monitor the changes that occurred during this cycle was previously not possible. This machine now has health monitoring capabilities far and beyond any that currently operate at this refinery.

The completion of this run was the culmination of a successful project. The compressor and auxiliary systems operated as specified. The coker was back online doing the same job as before, but now at lower operating and maintenance costs.

CONCLUSIONS

- The successful conclusion of the project was achieved through meticulous planning, close cooperation and communication between the user, OEM, magnetic bearing, and seal manufacturer in all phases of the project.

- It was shown that if properly planned and executed the installation of the machine, tuning of the magnetic bearing and startup of the machinery could be done within 12 days, or less.

The initial high cost of retrofitting the compressor with magnetic bearings and dry gas seals was justified by reduced operating and maintenance costs, power savings, and ease of operation.

- Attention must be paid during the design stage to ensure acceptable design, ease of assembly, and attention to details. Several iterations are required to arrive at a final acceptable design.

- To optimize the size and design of magnetic bearings accurate evaluation of anticipated bearing loadings is essential.

- It is very essential that detail rotordynamics analysis be performed up front to avoid potential vibration problem, and to reduce magnetic bearing tuning time.

- The adherence to the rotordynamic requirements stated in API 617 5th edition proved adequate and acceptable in magnetic bearing application.

- Preliminary static tuning of the rotor assembly reduced total tuning time at site during the shutdown.

- The success of a project can sometimes be measured by how well the operators accept the changes. They lived with the old machine for more than twenty years. If it is not easier to operate and is less of a problem, you will have a hard time getting them to buy into it. Some comments from the operators were the ease of startup, how smooth it operates, less time required per shift, and no oil problems.

NOMENCLATURE

For Table 1

CF	Cash Flow
DCF	Discounted cash flow
NPV	Net present value
PV	Present value
ROR	Rate of return

For Figure 6

C	Damping coefficient, lb sec/in
K	Stiffness coefficient, lb/in
W	Frequency, rad/sec

REFERENCES

1. Weise, D. A. and Pinckney, F. D., "An Introduction and Case History Review of Active Magnetic Bearings," *Proceedings of the Eighteenth Turbomachinery Symposium*, The Turbomachinery Laboratory, The Texas A&M University System, College Station, Texas (1989).
2. Pinckney, F. D. and Keesee, J. M., "Magnetic Bearing Design and Control Optimization for a Four-Stage Centrifugal Compressor," STLE Preprint No. 91-TC-6A-1 (1991).
3. American Petroleum Institute, "Centrifugal Compressors for General Refinery Service," API Standard 617, 5th Edition (1988).

ACKNOWLEDGEMENTS

The authors are indebted to those people at Elliott Company, Marathon Oil Company, and Magnetic Bearings Inc., whose support and assistance made this paper possible.