

SYNCHRONOUS MOTOR LATERAL VIBRATION DYNAMICS— DIAGNOSIS, RESOLUTION AND FIELD IMPLEMENTATION

by

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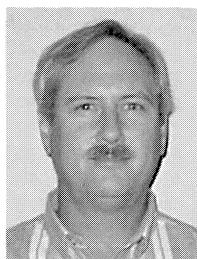
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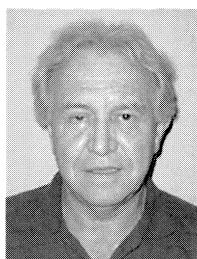
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ABSTRACT

A large, double ended drive synchronous motor experienced severe, limiting lateral vibration during commissioning of a major plant expansion. The motor driving a centrifugal compressor providing process air is decoupled during startup from a gearbox and steam turbine by a clutch. Steam generated by waste heat from the process drives the turbine under normal operation and the synchronizing clutch is engaged, reducing the motor load requirements. Excessive running speed ($1\times$) lateral vibrations came to light when the motor was run solo during field commissioning. Extreme sensitivity to small trim balance weight changes on the exciter end led to extensive field tests to identify the source of the problem. Rotordynamic analyses and coupled running tests confirmed an overhang resonance, and revealed the

only possible short term solution to be the addition of a third bearing on the motor shaft to control the overhang mode. Field installation of a 14 inch diameter tilting pad bearing, located on the clutch hub at the motor shaft end, was completed quickly to allow plant startup to proceed. *In situ* machining of the hub on the shaft, while the motor remained installed in the field, was required to assure proper journal shape and rapid project completion. This paper describes the process of problem identification, measurement/verification, rotordynamic analyses, resolution, and field implementation. Before and after field vibration measurements are presented to illustrate the success of the solution and demonstrate the additional instrumentation required.

INTRODUCTION

A lateral motor vibration problem was discovered during commissioning runs of two motors used for driving parallel centrifugal compressors. A complete second unit was being added to an existing facility. These compressors provide the main process flow, and delays in startup due to motor vibration problems would incur large financial consequences. Layouts of the two machinery trains, referred to as Train A and Train B, are shown schematically in Figure 1. Train A comprises a two stage integrally geared hot gas expander, a synchronous motor, and a four stage integrally geared compressor. These are connected by dry flexible couplings with long spacers. Train B has the same motor, compressor, and dry flexible coupling between them as Train A. However, for Train B, the hot gas expander is replaced by a steam turbine and a speed reducing gearbox. The gearbox and synchronous motor are connected by a synchronizing clutch, while the steam turbine is connected to the gearbox by another dry flexible coupling. The coupling and clutch hubs are keyed and shrunk onto the motor shaft ends. Both trains are elevated on a steel structure over three intercoolers, an oil reservoir, and for Train B, a steam condenser.

The compressors provide process air to the plant. During the process startup phase, the hot gas expander and steam turbine are not available to produce power, as the process must progress to provide the heat to operate these units. The hot gas expander on Train A is rotating, but cooling air is supplied until the process gas is available. Once expander operation is established, some of the remaining heat in the process gas stream is used to generate steam to drive the steam turbine. The expander and steam turbine are used to provide part of the compressor power requirements during normal plant operation, which reduces electric power consumption. During the startup of Train B, the steam turbine and gearbox are isolated from the motor by a synchronizing clutch. This allows the motor and compressor to rotate independent of the steam turbine and gearbox. Figures 2, 3, and 4 show the Train A compressor, Train A expander, and Train B motor, respectively.

Table 1 shows the motor rating and rotor information. Since the motor is a double ended drive, one shaft end must be longer than

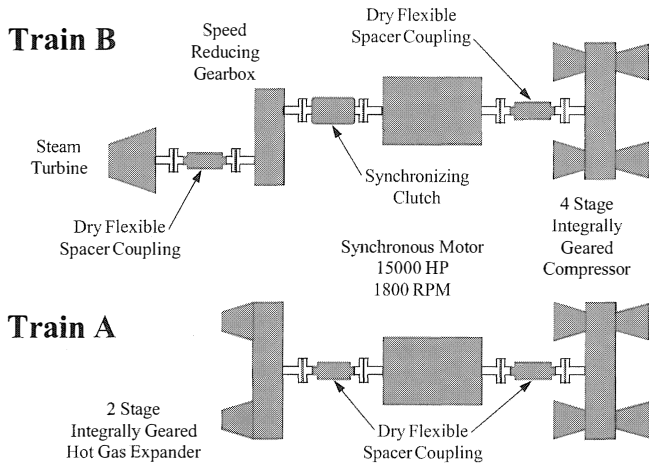


Figure 1. Equipment Layout Schematic.

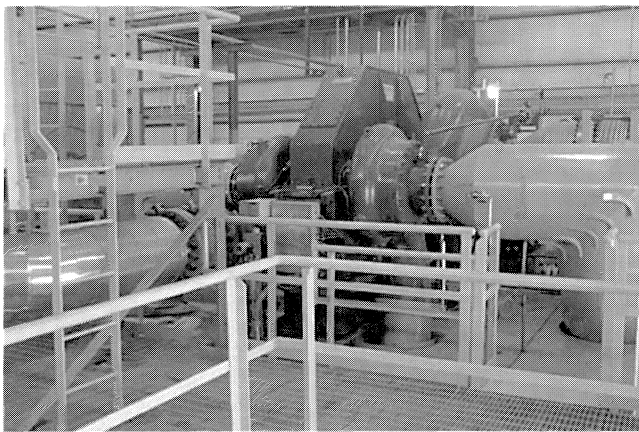


Figure 2. View of Train A Compressor Stage 2 Inlet.

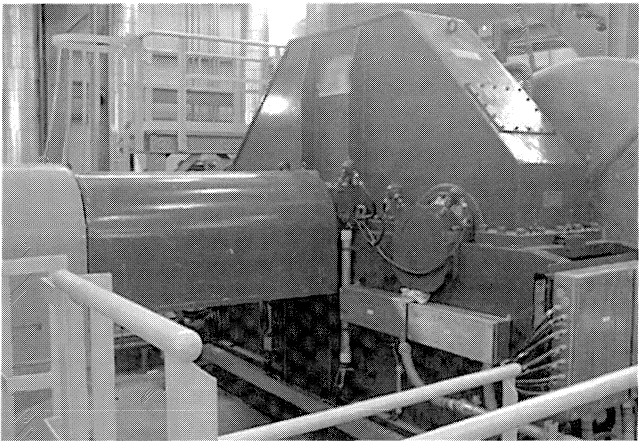


Figure 3. View of Train A Expander—Motor Coupling.

the other to accommodate not only the clutch hub, but also the starting resistors. The project motor specification called for a *stiff shaft* design and required any critical speed to be at or above 120 percent of the 1800 rpm operating speed. Figure 5 shows the motor manufacturer’s test stand vibration data prior to the motor being shipped. While there was an indication of a resonance below operating speed of 1800 rpm, it appeared that operation at synchronous speed would be in a *valley* between two critical speeds. This measurement generally agreed with the analytical predictions.

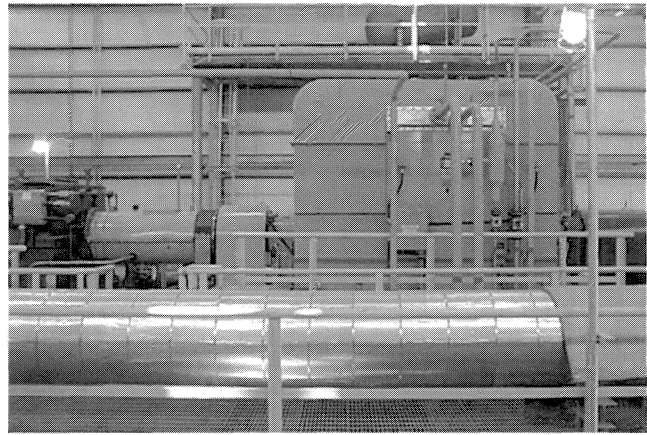


Figure 4. View of Train B Motor to Gearbox Clutch.

Table 1. Motor Information.

Power Rating	15,000 hp
Synchronous Speed	1800 rpm
Rotor Length	187.8 inches
Rotor Weight (without clutch or coupling)	16,000 pounds
Bearing Span	124.8 inches
Clutch Overhang (gear/turbine end)	42.5 inches
Coupling Overhang (compressor end)	20.5 inches

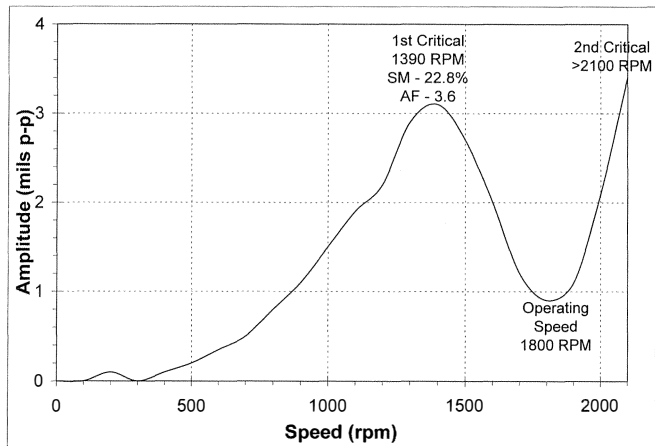


Figure 5. Motor Test Stand Vibration Data.

INITIAL SYMPTOMS

The problem became evident when excessive 1× vibrations were recorded during solo running and commissioning of the motors. Although the motors had been balanced and successfully tested prior to shipment from the factory, trim balance weights had to be added in the field. Review of the test stand data in Figure 5, in light of the actual field installation, raised the question of support stiffness contributing to the problem. Installation in the field required the motor to be located on a steel structure, while the test stand data were recorded with the motor supported in a different manner, most likely providing a more solid support. The fact that there were critical speeds on either side of operating speed, as shown in the test stand data, caused concerns from two

aspects. First, if the support stiffness in the field were softer than on the test stand, the resonant frequencies could shift downward and be causing the problem. Second, if the motors, as tested on the test stand, did not represent the as-installed conditions for weights and their center of gravity locations at the ends of the shaft, the resonant frequencies could also shift.

Balancing was difficult due to the extreme sensitivity of the rotors. Train B was determined to be more responsive than Train A, with a nominal sensitivity of 1.6 oz per mil for trim balance weights added in the vicinity of the exciter. Thus, for a 16,000 lb rotor, a 2 oz weight reduced synchronous vibration near the bearings from 4.25 mils, peak-to-peak, to 1.00 mil, peak-to-peak. The data measurement system employed was sufficient to accurately record the vibration and its phase relationship to a key phaser. The sensitivity and difficulty balancing could only be attributed to either loose parts moving relative to each other on the rotating element, or to the existence of a critical speed near operating speed. The fact that a 1 mil vibration amplitude could be achieved (after many balance runs), indicated the likelihood of loose parts was remote. These field trim balance run measurements were made with the motor uncoupled, but with the hubs for the coupling and clutch in place. The initial step taken was to thoroughly document and analyze the vibrations during motor startup, with a tracking data acquisition system and digital tape recorder monitoring the displacement vibration probes, permanently installed near the bearings.

DOCUMENT REVIEW

Review of the manufacturer’s critical speed analysis and comparison to test stand vibration data, when taken by itself, did not show a cause for alarm. Table 2 summarizes the predicted critical speeds. Operating speed appeared to be located between two damped critical speeds, and the predicted maximum amplitudes at those critical speeds were fairly low. Predicted running speed amplitude was less than 1 mil peak-to-peak. However, in reviewing the mass elastic data used in the manufacturer’s analysis, it was noted that the added weights and respective center of gravity locations representing the clutch and coupling at the ends of the motor shaft were assumed to be conventional values. Sufficient, as-built information on the mass elastic properties of the motor, couplings, and clutch was quickly gathered in case it became necessary to perform an independent critical speed analysis. Review of these data determined that significant discrepancies existed between the assumptions in the manufacturer’s analysis and the actual installed weights and center of gravity locations of the overhung weights. Table 3 shows a comparison of these values.

Table 2. Manufacturer’s Analysis Results.

Damped Critical Speeds	1325 rpm 2210 rpm
Max Amplitude	1.2 mil p-p at 1310 rpm 1.0 mil p-p at 2300 rpm
Predicted Running Speed Amplitude	0.75 mil p-p at 1800 rpm

Conventional values used for overhang weights

The actual, as-built values of the overhung weights were significantly higher than the values used in the analysis. For the compressor end of both A and B trains, the weight difference was 600 lb and the center of gravity location was approximately four and one-half inches further away from the bearing centerline. For the Train A expander end, the coupling weight was 400 lb heavier and almost eight inches further from the bearing center. For the Train B gear and turbine end where the clutch is located, the

Table 3. Overhang Weights and Center of Gravity Locations.

	Weight	Cg to Brg Ctr
Train A Actual		
Expander end	600 lb	43.29 inches
Compressor end	1100 lb	19.24 inches
Train B Actual		
Gear/Turbine end	945 lb	44.58 inches*
Compressor end	1100 lb	19.24 inches
Manufacturer’s Analysis (basis conventional values)		
Gear/Turbine end	500 lb	35.40 inches
Compressor end	500 lb	14.70 inches

*Center of gravity is beyond shaft end

additional weight was 445 lb and an additional 9.18 inches from the bearing centerline. This indicates the center of gravity of the overhung mass is actually beyond the end of the motor shaft. It should also be recalled that this end of the motor shaft has a longer overhang beyond the bearing than the other end. Critical speeds due to overhung masses on shaft ends are not a new phenomenon. Higher weight and longer overhang both contribute to lowering natural frequencies of vibration modes associated with overhung masses. The significant differences noted, between the analyzed mass elastic data and the actual as-built mass elastic data for the coupling and clutch, pointed to the fact that an independent rotordynamic analysis needed to be performed to determine if this could explain the vibration problem. Plant startup schedules dictated this had to be resolved rapidly. Several additional questions had to be addressed:

- Why was the Train A motor vibration better than the Train B motor?
- Why are similar motors in operation at other company plants without problems?
- Would there be problems with similar motors at other company plants that were under construction and nearing startup soon after this one?

While the purpose of this paper is to examine the technical problems and solutions, it is worthwhile to note that this project involved many manufacturers of the components of these compressor drive trains, which were located in several different countries, with different languages, separated by several time zones, and with highly disparate cultures. Given this situation, it is easy to see that insufficient communication could exist, resulting in some items *falling through the cracks*. It is important to point out that the technical problem result (vibration) is not directly attributable to a lack of technology on the part of any company, rather, it is attributable to cultural differences and the lack of timely, sufficient communication.

At this point, it was determined that if the plant startup was to occur in a timely fashion, all parties involved would have to work together independent of fault. There would have to be significant communication in all directions. Problem definition would most likely come from independent rotordynamic analyses that were undertaken concurrently by the authors and two of the vendors. The feasible technical solution would have to be determined quickly and implemented as fast as possible.

ROTOR DYNAMIC ANALYSIS

An independent rotordynamic analysis was performed on the motor rotating element, in various configurations both with and

without the overhung weights installed. The shaft model was developed from drawings supplied by the manufacturer and verified by field measurements where possible. Coupling and clutch models were verified with the vendors. Also, limits of possible modifications were investigated for the coupling and clutch along with replacing the clutch with a lighter coupling. This was not feasible from a process standpoint. Bearing information was developed based upon the motor manufacturer's information and verified by field measurements. The rotordynamic analysis consisted of calculating bearing/support stiffness and damping values, undamped critical speeds, and unbalance response. Many possible variations and combinations of the input parameters were investigated and compared with field data. Five of the cases that were analyzed during the rotordynamic analyses are shown in the following list.

- Case 1—Motor solo with hubs only
- Case 2—Motor with compressor coupled
- Case 3—Motor with compressor/turbine coupled
- Case 4—Modified system
- Case 5—Comparison to manufacturer's analysis

Case 1 was run to compare predictions to the motor solo vibration field data with the coupling hub and clutch hub installed. Case 2 was run to compare to actual field data with the compressor coupled, but with the clutch hub only on the other end. Case 3 was run as a prediction of what could happen with the compressor and gear both coupled to the motor (as-built, normal operation). Case 4 represents a modified system and will be discussed later, while Case 5 was run to compare to the manufacturer's initial analysis.

ROTORDYNAMIC RESULTS—CASES 1, 2, AND 3

Figure 6 shows a sketch of the mass elastic information for the motor as analyzed for Case 1, which includes only the coupling and clutch hubs on either end of the shaft. Modelling was begun at the clutch end of the shaft, which shows the longer overhang as compared with the compressor drive end of the shaft. The elliptical sleeve bearings have an L/D ratio of approximately 0.7.

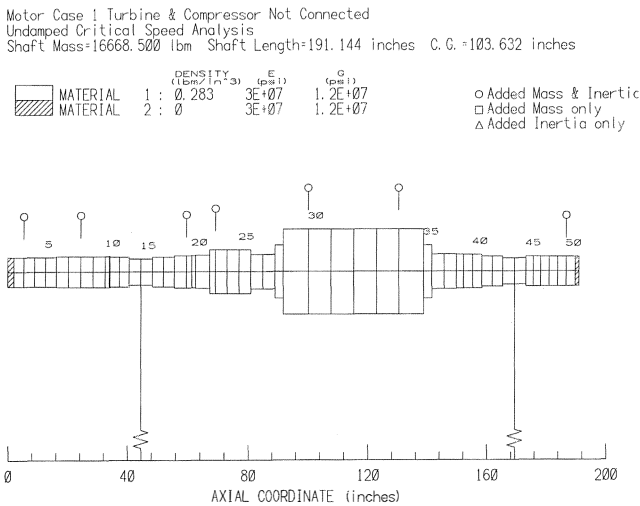


Figure 6. Motor Shaft Sketch—Case 1.

Figure 7 shows a comparison of the critical speed maps for Case 1 and Case 3. Bearing oil film stiffness lines are superimposed on the critical speed maps for orthogonal weak and strong directions. These stiffnesses are approximately constant at 900,000 lb/in, and 3.6 million lb/in for the speed range of concern. Case 1 shows a fairly typical critical speed map, but the comparison to Case 3 shows that the second mode *crosses over*

the first mode as support stiffness increases. Review of the mode shapes indicated the first mode at lower stiffness values is a classical bouncing mode of the shaft, while the second mode is an overhang mode of the clutch end of the shaft. Figures 8 and 9 show the lowest three mode shapes as calculated for the undamped critical speed maps for support stiffnesses of 900,000 lb/in, and 3.6 million lb/in, respectively. Comparing the second mode for the lower stiffness with the first mode for the higher stiffness indicates the modes have indeed *crossed over*, and that the first mode for Case 3 is the overhang mode at the higher stiffness values. Once the higher stiffness range is reached on the critical speed map, increasing support stiffness has very little impact on the frequency calculated for the first critical speed. The close proximity of intersections of operating speed (1800 rpm), bearing oil film stiffness curves, and modes 1 and 2, gave cause for concern for the as-built machine.

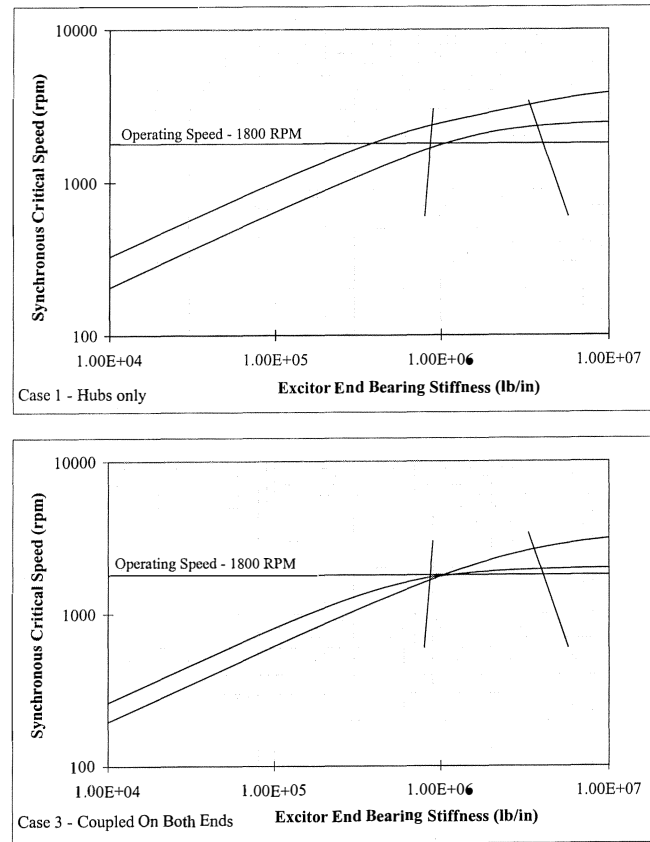


Figure 7. Undamped Critical Speed Maps, Cases 1 and 3.

Table 4 shows a summary of the calculated critical speeds for the two support stiffnesses referenced above. Also included are boxes tracing the various modes from case to case. The crossover between Case 2 and Case 3 is due to the increased weight included for Case 3, with the coupling and clutch installed. The undamped critical speed analysis with as-built mass elastic data (Case 3) showed there was a very good possibility that there was a critical speed near running speed. Case 3 had overhang modes at 1743 rpm for the weak axis stiffness direction and 1953 rpm for the strong axis stiffness direction. This is in addition to a first mode frequency of 1685 rpm for the weak stiffness direction. With this combination of three critical speeds predicted in the vicinity of 1800 rpm, it was determined that extensive efforts to refine the calculations were not necessary. Slight changes in support stiffness (bearing clearance) could move the prediction from one side of running speed to the other, but never far enough away to meet the required separation margin.

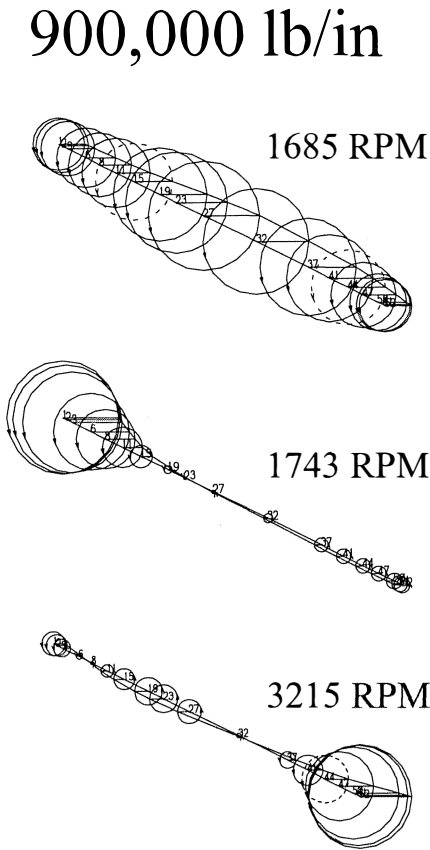


Figure 8. Undamped Mode Shapes—Case 3, Weak Axis Stiffness.

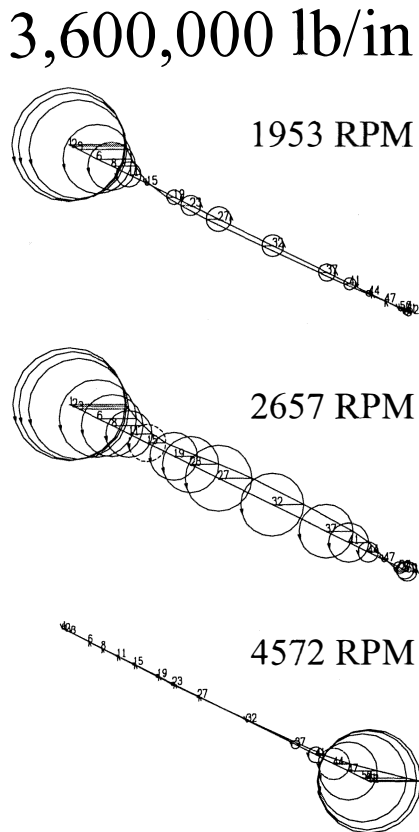


Figure 9. Undamped Mode Shapes—Case 3, Strong Axis Stiffness.

Table 4. Summary of Undamped Critical Speeds.

Frequency in RPM at a Support Stiffness = 900,000 lb/in					
	Case 1	Case 2	Case 3	Case 4	Case 5
1st Mode	1696	1686	1685	1687	1692
2nd Mode	2369	2336	1743 -3.17%	3148	2188
3rd Mode	4596	3321	3215	4438	4381
Frequency in RPM at a Support Stiffness = 3,600,000 lb/in					
	Case 1	Case 2	Case 3	Case 4	Case 5
1st Mode	2314	2311	1953 8.50%	2490	2273
2nd Mode	3255	3254	2657	4572	3029
3rd Mode	7370	4572	4572	8090	7073
				8779	
				9000+	

Unbalance response calculations were performed for center unbalance and overhang unbalance with a nominally acceptable unbalance weight. These indicated generally acceptable vibrations at the bearing locations. However, high vibrations were predicted at the end of the shaft where the clutch hub was located. Figure 10 shows the response calculated for the clutch end of the shaft for Case 3. For the *x* axis direction, peaks are seen at 1620 rpm and 1920 rpm. While the amplitudes appear acceptable and are under 2 mils peak-to-peak, the amplification factors and/or separation margins are unacceptable. The response in the *y* direction, shown in Figure 10, illustrates an extremely sharp resonant peak at 1920 cpm, with an amplification factor of 84. The separation margin for this calculation for a 2 in-oz unbalance was 10 mils peak-to-peak at the clutch end of the motor shaft. Mode shapes are shown in Figure 11, indicating the variation in the response at 1500 rpm and 1800 rpm. The mode shapes appear to be clutch end overhang modes, isolated in horizontal and vertical planes at these speeds. Again, support stiffness considerations could cause slight variations in the predicted frequencies, but not change the overall situation significantly. Support structure and foundation effects have been known to cause field installed machinery critical speeds to be located differently from previous calculations and/or test stand measurements. However, in this particular case, it was the significant differences in modelling overhung weights that was the culprit, not structural considerations.

Based upon which values were selected for various analysis parameters, critical speeds were predicted above and/or below the operating speed of the shaft, but always near. The analysis was *not optimized* to exactly correspond with field measured data due to time constraints, but sufficient knowledge was gained to provide significant insight into the dynamic characteristics of the system. The controlling mode of greatest interest is the overhang mode on the clutch end, which is a function of the overhang length, shaft diameter, and weight of the overhung mass. Very little impact is predicted for this mode, due to bearing oil film damping characteristics caused by *lower* vibrations at the bearings. No other source of damping was considered in this analysis. While the clutch does have a lubricant supply to it, and contains a lubricated journal bearing between its moving parts, modelling of this damping within the rotordynamic analysis was not considered, due to complexity and the aforementioned time limitations.

Since Train A has a smaller weight in this location (since it does not have a clutch, but a lighter weight coupling), it is understandable that it should have better vibrations and be less

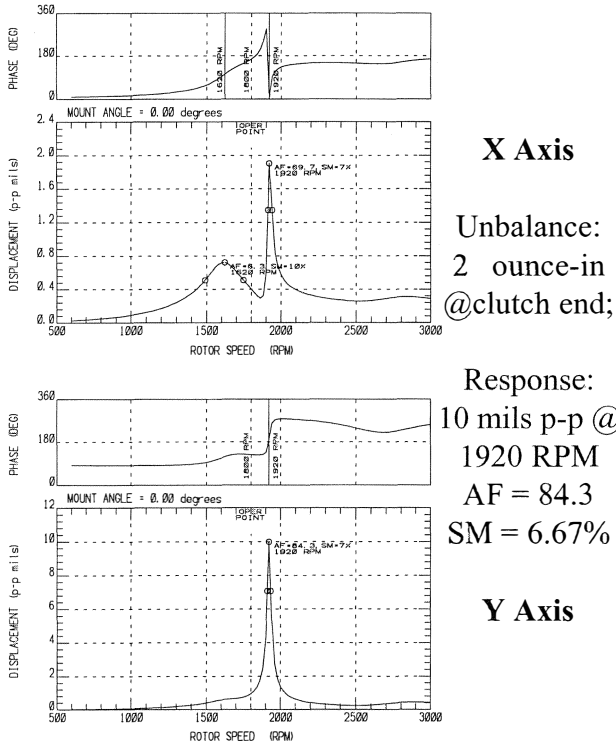


Figure 10. Unbalance Response—Case 3, Overhang Unbalance.

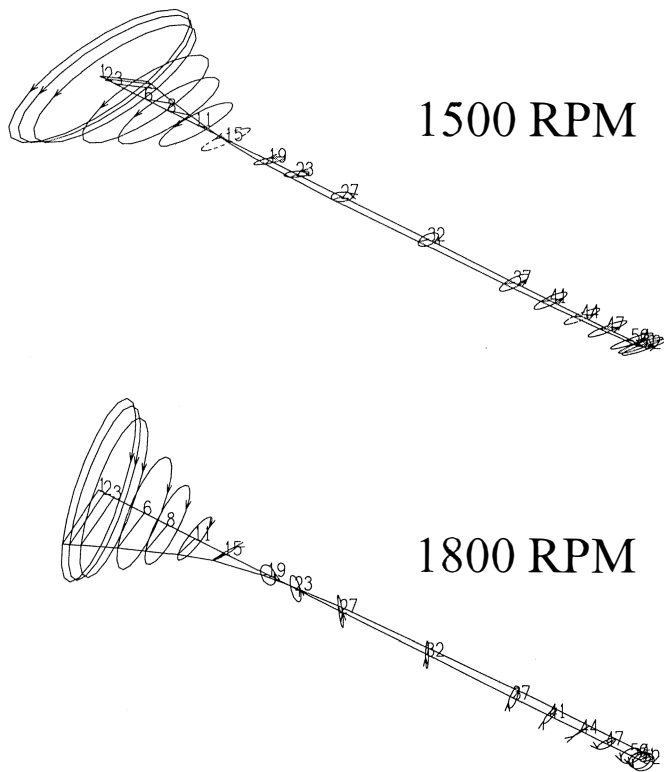


Figure 11. Unbalance Response Modes—Case 3, Overhang Unbalance.

responsive than Train B, as its resonant peak should be slightly higher in frequency. Also, similar motors operating at slower speeds in other plants (1500 versus 1800 rpm, due to electrical mains frequency differences) could account for the success of operation in those plants.

FIELD DATA—TRAIN B COUPLED

Excessive solo running vibrations of the motor were documented and the analysis showed a potential for a critical speed problem when coupled and run. Analysis showed no simple fix, such as an adjustment to existing bearings, could be implemented to change the predicted results. Analyses performed by both motor and compressor vendors in the same timeframe revealed similar results. A large number of companies were involved and many people within each company were reviewing various aspects of the available information. The inevitable question that had to be addressed was, “How difficult a problem is this to live with, since it appears that fixing it will incur significant costs and cause delays in startup?” This risk of experimental operation was assessed based on all available information, and the involved parties agreed that a better handle was needed on the actual vibrations that would be experienced when the unit was in operation. The decision was made to add some instrumentation to the motor, couple it to the compressor and gearbox, and observe vibrations as the motor was started.

In addition to the existing proximity probes in XY directions near the bearings, a proximity probe was added in a vertical direction at the shaft end to observe the clutch hub displacement, as shown in Figure 12. This had to be attached to the clutch cover, which was not an *inertial* reference, so an accelerometer was added on the housing of the clutch in the same direction. Thus, the absolute vibrations of the clutch hub in a vertical direction could be determined by vector addition of the two signals. Surface runout was evaluated and taken into consideration.

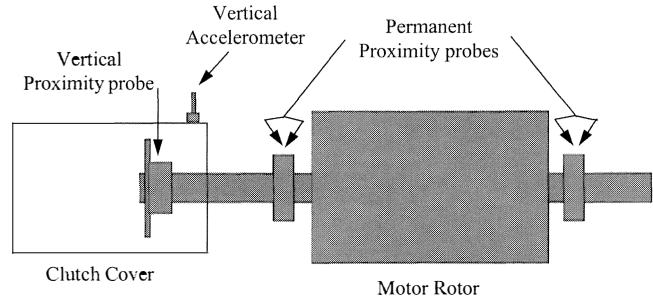


Figure 12. Additional Instrumentation at Clutch Hub.

The unit was operated with the new instrumentation installed, and a tape recorder was used along with a tracking data acquisition system to make sure the vibration data would be available for analysis at a later time. Figure 13 shows the proximity probe data near the bearings during the coupled startup. Maximum vibrations on the clutch end were 4.1 mils peak-to-peak at operating speed. Proximity probe data at the clutch indicated 21 mils peak-to-peak vibration as the unit achieved operating speed. Figure 14 shows this information. Displacement of the clutch housing was 2.6 mils at operating speed. Figure 15 shows the vector sum of these two signals, which gives the absolute motion of the clutch hub (assuming runout is negligible). The peak-to-peak vibration at running speed was 21.8 mils. The unit was operated for less than six minutes and shut down. Operation was allowed only for the time required to assure the necessary data had been permanently recorded.

The vibration data recorded during this coupled startup verified the existence and documented the severity of the problem. Measured vibration amplitudes at the shaft end exceeded the calculated values, which were based on nominal unbalance residuals. As mentioned previously, the analysis was not *optimized* to exactly match field data, but it was instrumental in providing insight into the nature of the problem and point toward a solution. The resonant amplitude measured did not indicate the sharpness of resonance as predicted in the analysis, possibly due to additional

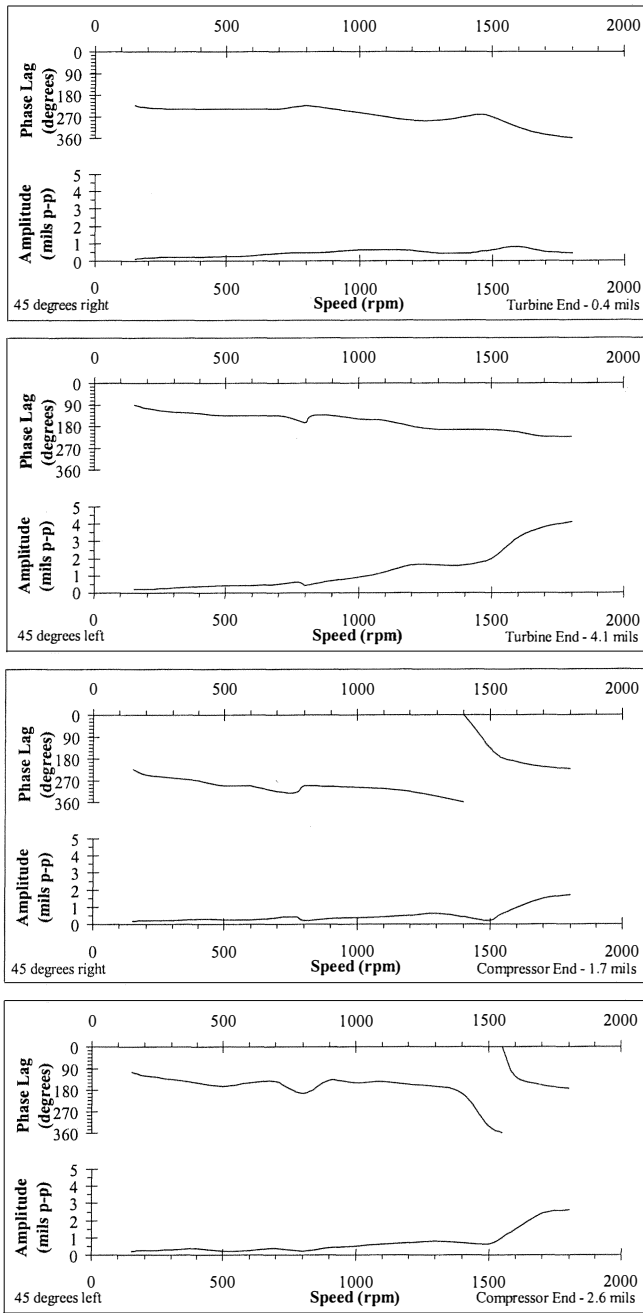


Figure 13. Proximity Probe Data at Bearings.

damping from the oil in the clutch or normally expected relative movement of parts within the clutch since it was not engaged.

The vendor required the clutch to be returned to a repair facility for disassembly and inspection due to the vibration experienced, otherwise the warranty would be voided. Fortunately, no damage was found.

These vibration data confirmed the suspicions that a critical speed was the root cause of the excessive $1\times$ vibration. There was a resonance at or very near running speed that was due to an overhang mode on the clutch end of the shaft. Having a resonance at or near running speed explained the sensitivity to small trim balance weights. The need to rebalance the unit in the field was explained by the fact that operation on the test stand was without the actual installed weights of either the clutch and/or the coupling. It is interesting to note that even though the mass elastic data for the compressor end of the motor shaft and for the expander drive

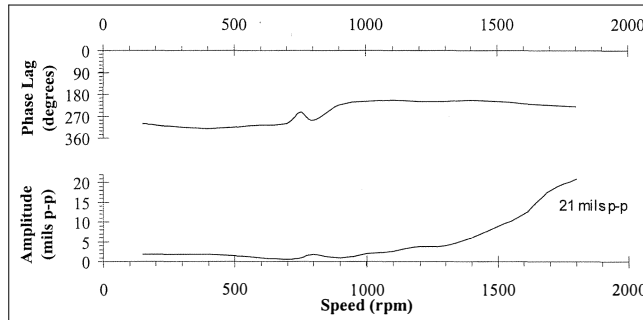


Figure 14. Proximity Probe Data at Clutch.

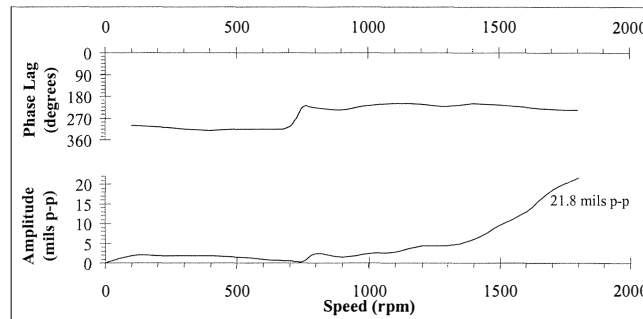


Figure 15. Absolute Displacement of Clutch Hub.

end of Train A were also incorrectly used in the manufacturer’s analysis, critical speed problems were not evident at those locations from field testing.

SOLUTION DEFINITION

It was determined that Train B would not operate reliably at these measured vibration levels on the clutch hub. It was also determined that Train A was *better* due to the smaller overhang weight with the dry coupling on the shaft connecting to the expander. Simple bearing modifications would not help the Train B situation and a shorter shaft overhang was impossible, due to the machinery already being in place. Investigations were made into changing the clutch weight by bore enlargement, but insufficient weight could be removed to make a significant difference.

All these conditions pointed to the need to install a third bearing at the clutch end of the shaft or remove the clutch and operate without the steam turbine. Evaluations were made of the economics of plant operation without the steam turbine. A conservative estimate was made based on local power costs and expected yearly operational use of the air compressor. Operation with the steam turbine providing some of the power to drive the compressor could provide a savings of \$2,300,000 per year. It was determined that installation of a third bearing was necessary.

ROTOR DYNAMIC ANALYSIS—CASE 4

It appeared, based upon the foregoing rotordynamic analysis and field vibration measurements, that the only feasible solution in the short term would be the addition of a third bearing at the clutch end of the motor shaft. Discussions were held with the clutch vendor, and it was found they normally recommend a third bearing be used at the clutch when driving through the exciter end of a synchronous motor. The vendor indicated this had been done on previous projects for other customers. A length of shaft for the journal can be added in the design phase, at the end of the shaft where the clutch hub fits. Had this information been available to all parties in the early stages of the design of the project, it might have been

possible to avoid this problem, but motor shaft design is not in the clutch vendor's area of responsibility.

A rotordynamic analysis commenced (Case 4) investigating the characteristics of the system with a third bearing located at the clutch end of the shaft. The only available axial length to locate this bearing was on the hub of the clutch, which was shrunk on and keyed to the motor shaft. Figure 16 shows a mass elastic sketch of the motor shaft with the third bearing on the clutch end of the shaft. Figure 17 shows a critical speed map for this arrangement. Case 4 included the mass of the clutch and coupling on the motor shaft ends similar to Case 3 and represents the system as it would exist during operation, fully coupled. Figure 17 indicates that the second mode no longer crosses over the first mode, but has increased in frequency all across the stiffness range of the critical speed map. Bearing support stiffness lines superimposed on Figure 17 are the same values as used in the prior analyses. A significantly smaller bearing stiffness was used at the clutch end of the shaft, since it was assumed that it would be difficult to install a very stiff bearing at that location due to support limitations. Mode shapes shown on Figures 18 and 19 are for 900,000 lb/in and 3.6 million lb/in support stiffness, respectively. These show that the overhang mode of the clutch end of the shaft has increased significantly in frequency and is no longer a critical speed concern for this motor, operating at 1,800 rpm.

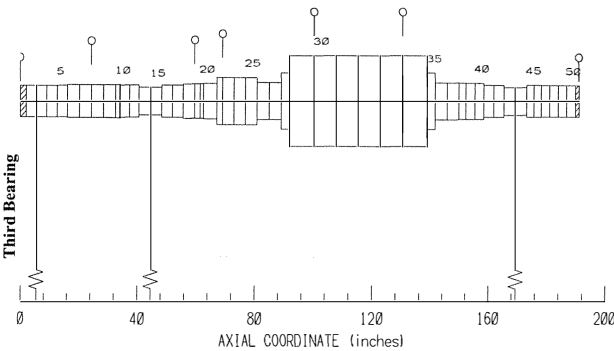
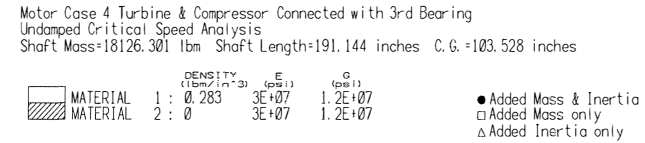


Figure 16. Motor Shaft Sketch—Case 4.

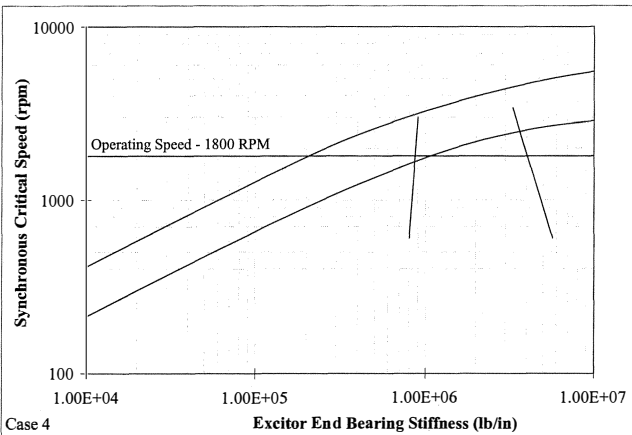


Figure 17. Undamped Critical Speed Map—Case 4.

900,000 lb/in

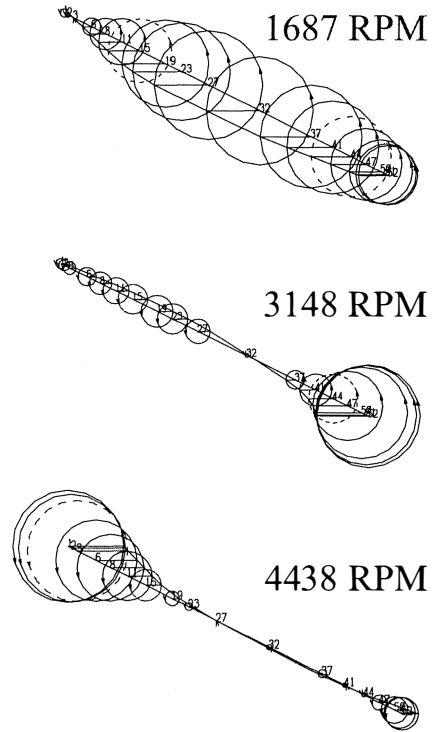


Figure 18. Undamped Mode Shapes—Case 4, Weak Axis Stiffness.

3,600,000 lb/in

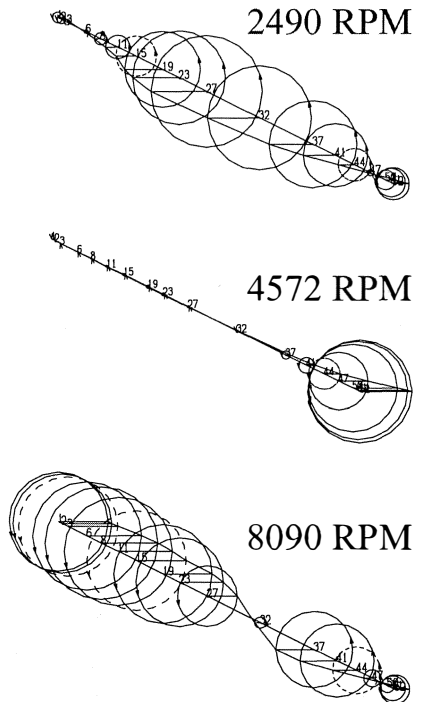
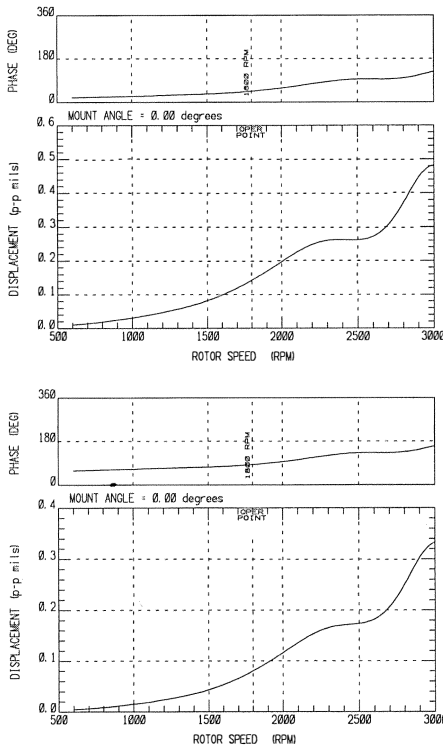


Figure 19. Undamped Mode Shapes—Case 4, Strong Axis Stiffness.

Returning to Table 4 for Case 4, the critical speed summary indicates that for the lower support stiffness of 900,000 lb/in, the second mode has increased from 1743 cpm to 4438 cpm, while at the higher support stiffness of 3.6 million lb/in, the problem mode has increased from 1953 cpm to somewhere above 9,000 cpm. Figure 18 shows the first critical speed to be 1687 rpm for the 900,000 lb/in support stiffness. While this is fairly close to operation at 1800 rpm, there is significant amplitude at the bearings to take advantage of available damping.

Figures 20 and 21 show the overhang unbalance response for Case 4. No critical speed is predicted within acceptable separation margins of running speed of 1800 rpm. Center unbalance response was investigated for Case 4. A very highly damped response was seen in the vicinity of 1500 rpm. The response due to weak bearing direction stiffnesses at 1687 rpm, as referenced above, was not evident in the unbalance response. The response mode shape shown in Figure 21 at 1800 rpm still indicates an overhang vibration is possible, but it is of a nonresonant nature and significantly reduced amplitudes are predicted, as shown in Figure 20.



X Axis
 Unbalance:
 2 ounce-in
 @clutch end;
 Response:
 < 0.3 mil p-p @
 ~2300 RPM
 AF < 2
 SM > 25 %

Y Axis

Figure 20. Unbalance Response—Case 4, Overhang Unbalance.

SYSTEM CONCERNS

While it is easy to analyze the system on paper with three bearings, there were significant implementation concerns that had to be addressed, as the decision point had been reached on whether or not to go ahead with a third bearing addition to the motor shaft. A three bearing system is statically indeterminate, thus the load on the bearing could not be accurately determined. Previous bad experiences with three bearing systems caused concern, since not only had they not solved problems in other company locations, but had actually become an additional problem. The light load on the third bearing was also a concern, since the bearing could be unstable if not properly loaded. Along with the problem of no axial space along the shaft to install the bearing (except on the hub), structural support for the bearing was nonexistent, due to the nature of the installation of the motor on an elevated steel frame. The overhang of the motor shaft was beyond the extent of the motor skid. Also, welding an extension on the existing skid and

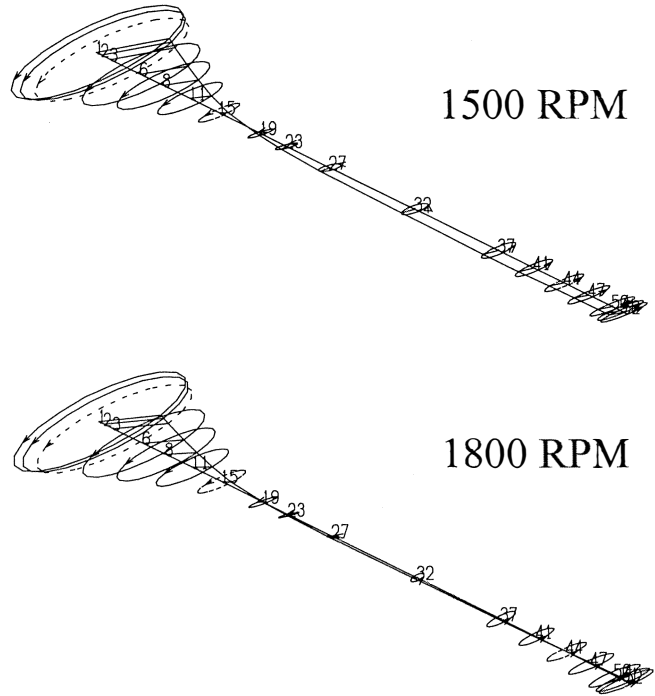


Figure 21. Unbalance Response Modes—Case 4, Overhang Unbalance.

baseplate could cause distortion and alignment problems. Adding another bearing to the system would cause lubrication supply demands to increase and it had to be determined if the existing system could accommodate them. The clutch cover would need modification to allow axial room for the bearing and still maintain a seal for the clutch oil.

The solution plan proposed was to add the third bearing at the clutch hub, which was 14 inches in diameter. A bit of good fortune came along to the project at this point (which was certainly needed) in that the clutch vendor had a spare 14 inch diameter tilting pad bearing in stock (without a housing), which could be supplied immediately. While the needed support at the shaft end did not specifically require the technology of a tilting pad bearing, this was the only available bearing without long manufacturing delays. Support for this bearing had to be developed and integrated with the clutch cover. Implementation of this solution was needed as soon as possible and, since the confidence level was very high that this would solve the problem, all parties involved energized the resources necessary to implement the proposed solution.

INSTALLATION

Two significant obstacles had to be overcome for the installation. First, massive structural modifications were necessary to provide a support for the bearing, and a pedestal had to be built. This was designed and built within the existing limits of a machine already installed and operational (Figures 22 and 23).

The second major obstacle was that a journal surface had to be created on the clutch hub. Neither the existing surface finish nor the hub runout was sufficient. Approximately one-eighth inch of diameter had to be machined off to obtain the proper dimension to fit the available bearing. Figure 24 shows the clutch hub schematically, along with diametrical measurements at three locations axially and three locations circumferentially that were taken prior to any machining. A 4 mil depression was discovered over the keyway that was due to a 14 mil shrink fit of the hub onto the shaft. Also, an axial taper was found to exist, since the hub is not symmetric from end to end. The existing surface where the journal was to be machined was neither circular nor constant

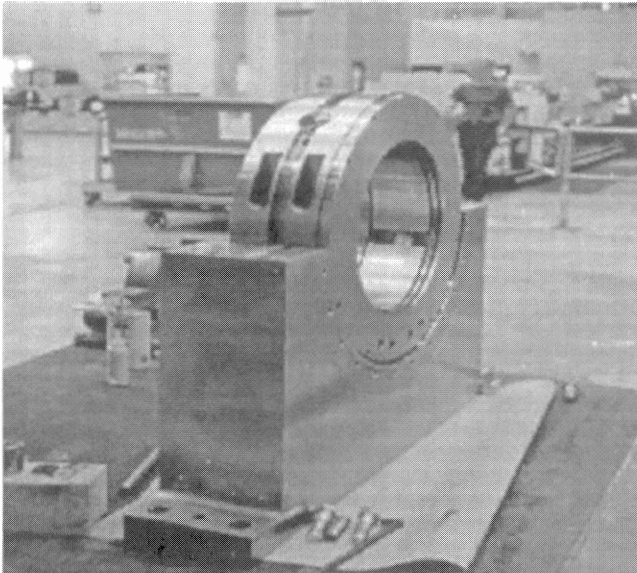


Figure 22. View of Third Bearing with Housing Under Construction.

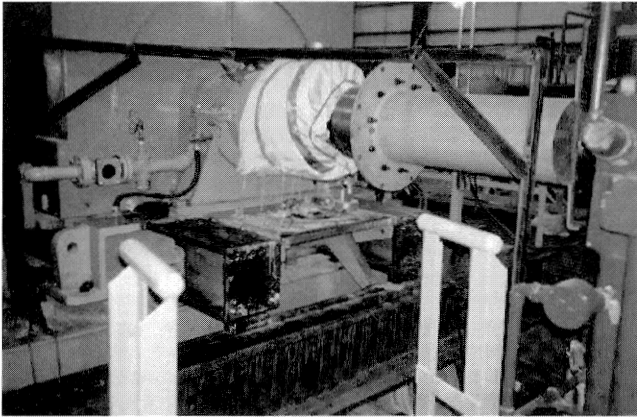


Figure 23. View of Structural Support Being Added.

diameter. If the hub was removed from the shaft for machining and then reinstalled, the actual shape of the journal would not remain circular or constant as needed to develop the proper oil film within the tilting pad bearing. In fact, it was unlikely that the clutch hub could be removed without galling the shaft due to the heavy shrink fit. The clutch hub would have to be machined in place, on the motor shaft, in the field, with the clutch mechanism removed, but without removal of the motor from its base. Extreme care for the machining operation was required. The cutting tool had to be adjusted as it was moved along the hub axially, since the static deflection slope of the shaft during machining was different than it would be in operation because of the weight of the clutch. The motor rotor was turned at a low speed, while the proximity probes near the existing bearings were continuously monitored to make sure that no damage was occurring to those bearings and journals during the machining (Figure 25).

SOLUTION PROVEN

After the above discussed modifications were made to the support structure and clutch hub, the bearing was installed and the system realigned (Figure 26). The bearing was installed slightly high, compared with a static deflection shape of the shaft, so that an approximate 2,000 lb load would be on the bearing. Similar instrumentation was used as before and the motor was started up

0 - 180 deg	14.178	14.172	14.1695
Keyway	14.174	14.163	14.165
45 - 225 deg	14.178	14.173	14.169

Note: 4 mil depression over keyway and axial taper

14 inch Outside Diameter
 9.8 inch Inside Diameter
 with 14 mil Shrink Fit on Shaft

Figure 24. Clutch Hub Diameter Runout Measurements.

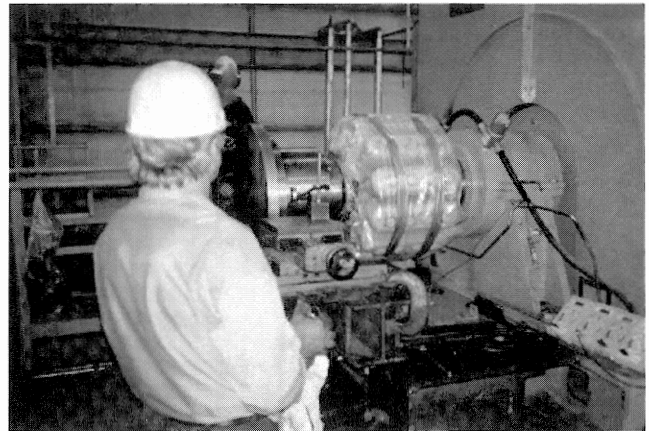


Figure 25. View of In-Place Machining Operation.

with both clutch and coupling installed. Figure 27 shows vibration data at the end of the shaft measured with the proximity probe relative to the clutch cover. Vibration amplitude at operating speed was 2 mils peak-to-peak. No resonant amplification was indicated by the vibration startup response. This showed the vibration had been reduced from 21 mils to 2 mils by the addition of the third bearing on the clutch hub. Plant startup proceeded and this third bearing has been in operation at normal capacity for over one year.

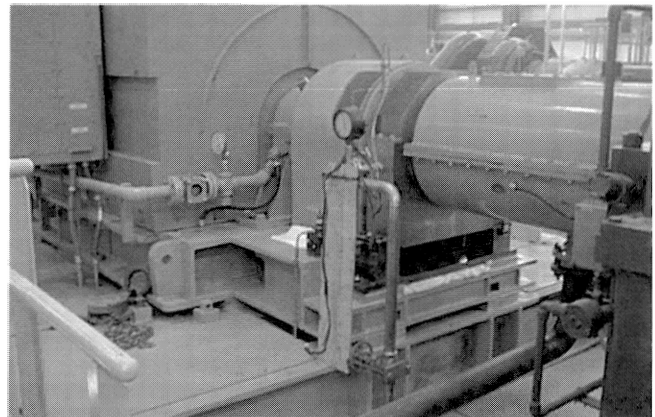


Figure 26. View of Third Bearing Installed and Operational.

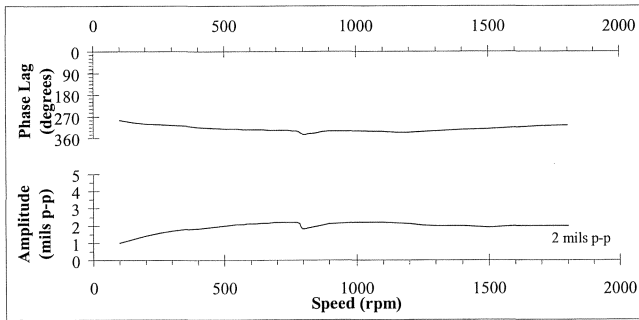


Figure 27. Proximity Probe Data at Clutch with Third Bearing Installed.

CONCLUSION

This paper has shown that a significant vibration problem was diagnosed, identified, and corrected during the commissioning phase of a plant expansion. This was an unanticipated problem that had to be dealt with, in addition to the normal course of business of building and commissioning a large plant. It was a significant accomplishment for all parties involved that this problem was resolved in a very short period of time. The following timeline indicates the elapsed time from discovery of the problem to successful operation of the motor.

- Excessive vibration during commissioning March 1997
- Analysis input data problem discovered 03/26/97
- Revised verified rotordynamic analyses completed 04/02/97
- Coupled vibration data confirmed problem 04/11/97
- Commitment to buy third bearing 04/15/97
- Successful operation with third bearing installed 05/15/97

A significant contribution to this timeline brevity was the fact that an existing bearing of the proper dimensions was quickly located and available. This provided a significant impetus to overcoming the other obstacles along the way to installing the successful solution prior to plant startup.

The technical achievements of resolving a critical speed problem are not new. Many sources of information and computer software are available to analyze problems of this nature. The same can be said for availability of instrumentation and software to analyze existing vibration problems. This paper has demonstrated that it is imperative in the design phase of a project to have the proper information communicated in a timely fashion to all parties involved. Even with the best technology available, we are still solving the same rotordynamic problems as occurred decades ago, rather than preventing them. With sufficient attention to system dynamics in the form of sufficient project *front end loading* problems, such as the one described above, can be avoided. Treatment of technology/information exchange issues before or at the time of order can provide sufficient lead time to prevent critical speed problems, but only with effective communication.

It must be reemphasized that this paper is not intended to blame or point out faults of any particular entity involved in this project, but to illustrate that the problems that have been solved many times before can still exist if the proper care is not taken.

ACKNOWLEDGEMENT

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