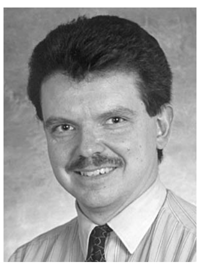
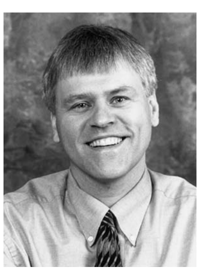
PRACTICAL USE OF ROTORDYNAMIC ANALYSIS AND TROUBLESHOOTING SKILLS TO CORRECT A LONG-TERM SYNCHRONOUS VIBRATION PROBLEM IN AN OVERHUNG TURBOBLOWER

by
Mark A. Corbo
President & Chief Engineer
No Bull Engineering
Delmar, New York
Thomas J. Kuli
Chief Engineer
and
H. Leslie (Les) Gutzwiller
Vice President of Technical Services
Robinson Industries, Inc.
Zelienople, Pennsylvania



Mark A. Corbo is President and Chief Engineer of No Bull Engineering, a high technology engineering/consulting firm, in Delmar, New York. He is responsible for rotating equipment consulting services in the forms of engineering design and analysis, troubleshooting, and third-party design audits. Before beginning his consulting career at MTI in 1995, he spent 12 years in the aerospace industry designing pumps, valves, and controls for gas turbine engines. His expertise includes rotordynamics, fluid-film journal and thrust bearings, dynamic simulations, hydraulic and pneumatic flow analysis, CFD, FEA, and mechanical design.

Mr. Corbo has B.S. and M.S. degrees (Mechanical Engineering) from Rensselaer Polytechnic Institute. He is a registered Professional Engineer in the State of New York, and is a member of ASME, STLE, and The Vibration Institute. He has authored more than a dozen technical publications, including one that won the "Best Case Study" award at Bently Nevada's ISCORMA conference in 2001.



Thomas J. Kuli is the Chief Engineer at Robinson Industries, Inc., a designer/manufacturer of heavy industrial fan and blower equipment, in Zelienople, Pennsylvania. He has been involved in the fan and blower industry for more than 19 years, which has included positions in the research and engineering departments. He has authored several articles associated with fan and blower design, applications, and troubleshooting.

Mr. Kuli has a BSME degree from the University of Pittsburgh. He is a registered Professional Engineer in the State of Pennsylvania, and is a member of ASME.

H. Leslie (Les) Gutzwiller is the Vice President of Technical Services at Robinson Industries, Inc., a designer/manufacturer of heavy industrial fan and blower equipment, in Zelienople, Pennsylvania. He has worked in the fan and blower industry for more than 26 years in various positions including research, engineering, manufacturing, as well as administrative duties (past President). He is a member of International Air Movement and Control Association (AMCA), where he has served on numerous committees and chaired the committee that published AMCA/ANSI Standard 204-96, "Balance Quality and Vibration Levels for Fans." He has several patents in the air movement industry and has authored numerous papers and articles associated with fan and blower design, applications, and troubleshooting.

Mr. Gutzwiller has a B.Ch.E. degree from Georgia Institute of Technology, and a Master's degree in Business Administration from Clemson/Ferman Universities.

ABSTRACT

The two subject blowers operate in parallel to circulate wet chlorine gas. Both units had large synchronous vibrations that led to multiple bearing failures. After simple rotordynamics studies failed to identify the problem, a comprehensive model that accounted for both the motor and blower was successful at identifying the problem as high sensitivity to unbalance loads due to an extremely lightly-loaded (less than one pound) condition at the blower's inboard bearing (refer to Gutzwiller and Corbo, 2001).

Based on the results of the rotordynamic analysis, two changes were made to both units. The couplings were changed from disk to gear couplings, and the blower's bearings were changed from plain cylindrical to tilting-pad designs. After implementing these changes, unit "A" ran smoothly for a period of six weeks, in accordance with the predictions of the rotordynamic analysis, and it appeared to all that the problem was completely solved.

However, when the "B" blower was then started up the two machines then commenced a three month period of operation in which each suffered from intermittent periods of high synchronous vibrations. During this time, the following behavior traits, which can only be described as bizarre, were observed:

1. After a smooth startup, vibration increased to a high level after about one week.

2. The transition from low vibrations to high vibrations was almost instantaneous. Additionally, this transition was always accompanied by an axial motion of the blower rotor away from the active thrust bearing.

3. The units appeared to be more likely to suffer high vibrations when both were running simultaneously.

4. Starting one machine might result in increased vibrations on the other, and shutting down a machine might lower vibrations on the other.

5. Someone observed that during a rainstorm the vibration levels decreased. (Note: the blowers are located outside.) Spraying water on the bearings sometimes (but not always) had a similar effect.

A task force of experts was then commissioned and an extensive troubleshooting effort commenced. Some of the potential root causes that were hypothesized included blower surging, starvation of the tilting-pad bearings, thermally-induced misalignment, insufficient blower thrust, acoustic resonance, axial vibration, seal rubbing, and Morton effects. After an extensive troubleshooting effort that included more rotordynamic analysis, bearing flow analysis, blower thrust analysis, and extensive studying of orbit plots, spectrum plots, and vibration and temperature histories, the task force concluded that the most likely cause was rubbing at the blower's carbon ring seal. Accordingly, the seal was disassembled and its locating pins were found to have come loose and generated a rub. Design changes were then implemented to provide better retention for the pins. The units were then restarted and it was verified that the modifications had finally eliminated the vibration problems.

This paper shows how the combination of rotordynamic analysis and troubleshooting skills was employed to identify and generate corrective actions for two independent causes of high synchronous vibrations.

INTRODUCTION

Overhung turbomachines tend to suffer more rotordynamics problems than do their straddle-mounted counterparts. Common problems experienced in such installations include failed bearings and wiped seals. Accordingly, a thorough lateral rotordynamics analysis should be included as an integral part of the design process for any critical process overhung machine. A comprehensive procedure for performing such an analysis is provided in Corbo and Malanoski (1998). Although this reference was directed toward pumps, the analytical procedure provided within it is general enough that it can be applied to any type of rotating machine.

One of the primary problems that plague overhung machines is that the prevalent use of plain cylindrical journal bearings tends to lead to rotordynamic instability problems. The instability is normally manifested as subsynchronous whirling and can lead to catastrophic events. The primary cause of this problem is that the majority of the machine's weight lies outboard of the bearing span. This often results in the bearing located at the opposite end of the shaft from the wheel being extremely lightly-loaded, making it unstable.

The fact that lightly-loaded plain cylindrical bearings tend to promote unstable shaft whirling is hardly an obscure phenomenon. Among the multitude of references that discuss this effect, Newkirk and Taylor (1924), Rentzipis and Sternlicht (1962), Pan and Sternlicht (1963), Lund and Saibel (1967), and Crandall (1982) are only a few. Accordingly, if the overhung blowers that are the subject of this paper had experienced subsynchronous whirling problems, that would hardly qualify as news. However, since the subject blowers have been troubled by synchronous, not subsynchronous, whirling, the number of precedents in the literature, if any, is greatly reduced.

The case to be discussed involves two 3000 hp motor-driven overhung turboblowers of titanium construction, manufactured by Robinson Industries, Inc., shown in Figure 1. The two subject blowers, units "A" and "B," are nominally identical and are located adjacent to one another. Operated in parallel, they are used to provide wet chlorine gas to a critical process at a large chemical plant in the southwestern United States. Both blowers run at a constant speed of 1780 rpm.

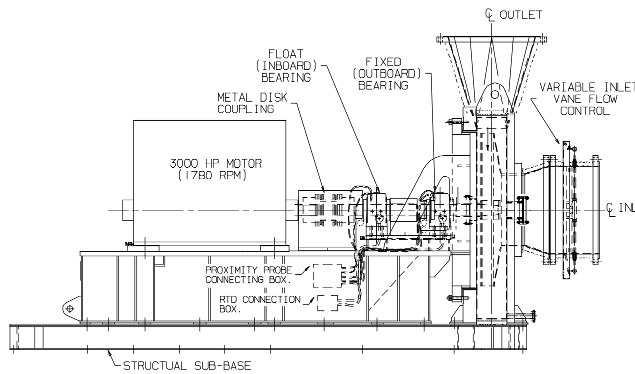


Figure 1. Blower Configuration.

After about one year of operation, both units began suffering from large synchronous vibrations that led to multiple bearing failures. After several conventional rotordynamic analyses failed to identify the cause of the problem, Robinson Industries hired No Bull Engineering to provide an independent evaluation. Their approach, which was less conventional than that of the previous analyses, utilized a comprehensive model that accounted for the motor, coupling, and blower in a single model. This analysis was successful at identifying the problem as high sensitivity to unbalance loads due to an extremely lightly-loaded (less than one pound) condition at the blower's inboard bearing (the one opposite the blower wheel). Since Gutzwiller and Corbo (2001) provide a detailed account of this problem and analysis, it is only described peripherally herein.

Based on the results of the rotordynamic analysis, two changes were made to both units. First, the couplings were changed from disk to gear couplings to eliminate the interdependency between the blower's and motor's rotordynamic behaviors. Second, the blower's bearings were changed from plain cylindrical to tilting-pad designs to reduce the inboard bearing's sensitivity to the lightly-loaded condition and to make both bearings more failure-resistant. After implementing these changes, unit "A" ran smoothly for a period of six weeks, in accordance with the predictions of the rotordynamic analysis, and it appeared to all that the problem was completely solved.

However, when the "B" blower was then started up, all illusions were shattered. The two machines then commenced a three month period of operation in which each suffered from intermittent periods of high synchronous vibrations. The behavior traits that these units exhibited can only be described as bizarre and are unlike any that have been previously published, to the best of the authors' knowledge.

A task force of experts was then commissioned and an extensive troubleshooting effort commenced. Some of the potential root causes that were hypothesized included blower surging, starvation of the tilting-pad bearings, thermally-induced misalignment, insufficient blower thrust, buildup of debris on the fan blades, acoustic resonance, locking up of the gear coupling, axial vibration, and seal rubbing. After an extensive troubleshooting effort that included more rotordynamic analysis, bearing flow analysis, blower thrust analysis, and extensive studying of orbit plots, spectrum plots, and vibration and temperature histories, the task force concluded that the most likely cause was rubbing at the blower's carbon ring seal. Accordingly, the seal was disassembled and its locating pins were found to have come loose and generated a rub. Design changes were then implemented to provide better retention for the pins. The units were then restarted and it was verified that the modifications had finally eliminated the vibration problems.

The history of this second problem and the troubleshooting effort that was employed to finally solve it are described in detail in this paper.

MACHINE DESCRIPTION

A major chemical manufacturer in the southeastern United States operates a process that involves the moving of wet chlorine gas. The system operates at a background pressure of up to 100 psig. A very heavy-duty process gas blower increases the pressure of the gas by approximately 6.5 psi while circulating a volume of 57,500 cubic feet per minute. This process must run for long periods of time with minimal shutdowns to maximize the profitability of the operation. Two nominally identical blowers (designated as units "A" and "B") operate in parallel and each provides the recirculation of the gas as described. A 3000 hp, 1780 rpm, constant speed AC motor drives each blower through a "flexible" coupling, which originally was a disk coupling. The blower, which is depicted in Figure 1, was originally configured with two plain cylindrical sleeve bearings, and an overhung blower rotor inside an extremely heavy-duty blower housing. The blower rotor is fabricated entirely of titanium Grade 7 and the housing is 6 inch thick mild steel with titanium "wallpaper" inside to provide effective corrosion resistance.

Figure 2 shows the loads that the blower shaft is normally subjected to. Since the weight of the wheel is substantially greater than that of the coupling and shaft, the blower's inboard bearing is normally subjected to a very light load, which, in the original design, acts upward. The thrust load, which is of aerodynamic origin, always loads the shaft toward the blower inlet, which is at the outboard end.

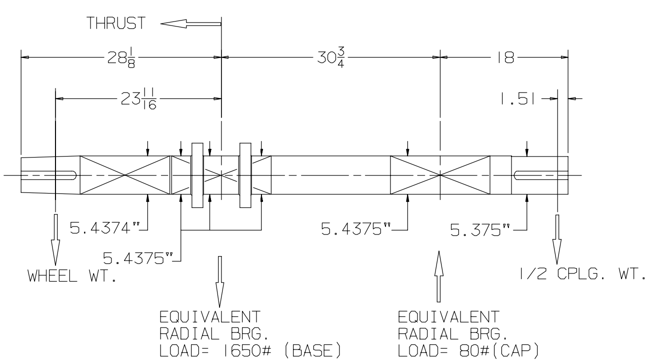


Figure 2. Loads at Bearings.

The entire blower and motor assembly rests on an integral subbase that is grouted to a heavy concrete foundation. Lubrication for the blower and motor bearings is supplied by a circulating oil system that provides filtered and constant temperature lubricant. The blower is monitored by means of the following:

- Blower outboard bearing
  - Two radial and one axial vibration proximity probes
  - One platinum resistance temperature detector (RTD)
- Blower inboard bearing
  - Two radial vibration proximity probes
  - One platinum RTD
- Motor inboard bearing
  - Two radial vibration proximity probes

All these measured parameters are continuously monitored using a monitoring system.

The blower's flow rate is controlled by means of a variable inlet vane damper of titanium Grade 7 construction that was supplied by the blower manufacturer. It is essential during operation of these blowers that no chlorine gas escapes to the atmosphere. A multiring carbon shaft seal with titanium Grade 7 housing construction is used to provide the shaft sealing. This includes a combined steam purge and nitrogen purge to ensure that the shaft surfaces remain wetted and that no steam and/or chlorine gas escapes to the atmosphere.

PROBLEM DESCRIPTION/HISTORY

After about one year of operation, both units were troubled by high synchronous vibrations that led to several failures of the blower's outboard bearing. A comprehensive static bearing loading and rotordynamics analysis of the entire drive train, consisting of motor, disk coupling, and blower, was then performed. A detailed description of this analysis is provided by Gutzwiller and Corbo (2001). The pertinent conclusions that were arrived at were as follows:

- The high angular rigidity of the disk coupling was causing the motor's and blower's bearing load distributions and rotordynamic behavior to be interrelated. Accordingly, traditional bearing loading and rotordynamics analyses, which assume the motor and blower to be independent of one another, were found to be insufficient for analyzing the behavior of this machine.
- As a result of this interrelated behavior, the static loading on the blower's inboard bearing was found to be extremely light, less than one pound for some cases of motor-to-blower radial misalignment.
- When this light-loading was taken into account, the plain cylindrical bearing at the blower's inboard location was found to generate virtually zero stiffness and damping. As a result, the rotordynamics analysis predicted large unbalance response amplitudes at this bearing, which were in agreement with the vibration measurements.

In order to rectify this situation, two major changes were made to the system. First, the disk coupling was replaced with a gear coupling (which is incapable of transmitting moments) to eliminate the interdependency of the motor's and blower's rotordynamic behaviors. Second, the blower's plain cylindrical bearings were replaced with tilting-pad bearings employing both geometric preload and offset pivots. Rotordynamic analysis verified that these two changes would eliminate the observed synchronous vibration problem while maintaining the unit's resistance to subsynchronous instability.

The "A" machine was then retrofitted with the new coupling and bearings and started up. For a period of about six weeks, this machine ran extremely smoothly with no vibration problems whatsoever. The synchronous vibrations at the blower's inboard bearing, which had consistently been 4 mil peak-to-peak or higher during the units' first year of operation, were reduced to well under 1 mil peak-to-peak, in accordance with the predictions of the rotordynamics analysis. This six week period represented, by far, the smoothest running period that either of these units had experienced in their entire history. This verified that the coupling and bearing changes had, indeed, solved the vibration and bearing failure problems that they had been designed for.

However, after the "A" machine had run trouble-free for this period, the "B" machine was started up. After a short period of time, the "A" machine started experiencing large synchronous vibrations at both of its bearings. At first, it was thought that this was a recurrence of the original synchronous vibration problem. However, this theory was quickly discarded since the new problem was observed to differ from the old one in the following ways:

- Whereas the old problem had generally exhibited high vibrations at the blower's inboard bearing only, the new problem had significant vibrations at both blower bearings.
- The new problem had a tendency to commence and disappear virtually instantaneously, similar to a rotordynamic instability problem. The unit would experience high vibrations for a certain period of time then dramatically drop down to the low vibrations that had been observed during the first six weeks of the "A" machine's running. The high vibrations could start just as suddenly. On the other hand, with the old system, the units exhibited high vibrations almost continuously.
- In the old system, all the vibrations had been synchronous and involved forward whirling, suggesting that they were triggered by unbalance. Conversely, although the new machine's vibrations were also observed to be synchronous, there was evidence of backward, as well as forward, whirling.
- Whereas the new problem was highly sensitive to both time and thermal conditions, the old system's vibrations were totally insensitive to these factors.

As a result of these major differences, all involved parties agreed that this was an entirely different problem than the original one. Over the next three month period, both units repeatedly experienced the vibration problems and both suffered several vibration-related shutdowns. During this time, the following behavior traits, which can only be described as bizarre, were observed:

- Whenever a unit was started up, its vibrations would invariably be low. It would take at least a week of continuous running before high vibrations could be observed.
- Even when a unit was shut down due to high vibrations and then restarted shortly thereafter, the vibrations usually returned to a low level.
- The transition from low vibrations to high vibrations was almost always instantaneous, suggesting that some sort of instability phenomenon was at work.
- The transition from low to high vibrations was always accompanied by a substantial (up to 8 mil) axial motion of the blower rotor away from the active thrust bearing (toward the coupling).
- The units appeared to be more likely to suffer high vibrations when both were running simultaneously.
- The simple act of starting up one machine could sometimes cause the other machine to transition from low to high vibrations.
- Conversely, shutting down one machine could greatly reduce the vibrations in the other.
- At one point when the vibrations were high, a sudden rainstorm caused them to instantaneously drop (note that these machines are exposed to the weather).
- Similarly, spraying cold water on the unit sometimes, but not always, was sufficient to eliminate high vibrations.
- Vibration levels *seemed* to increase when lube oil temperature increased.

Since these problems and their accompanying downtime were totally unacceptable to the user, a task force of experts, including the authors, was commissioned to perform an extensive troubleshooting effort and find the root cause of the problem. This effort is described in detail in the following sections.

TROUBLESHOOTING EFFORT

Of course, plant operating demands dictated that a quick solution be obtained. As is often the case in these situations, there were myriad observations, symptoms, hunches, etc., but very little hard data for analyzing the problem. Some of the potential root causes that were theorized included rotordynamics problems, starvation of the tilting-pad bearings, inadequate thrust load, blower axial vibrations, blower surging, loosening of the fan wheel, unloading of the blower inboard bearing, thermally-induced misalignment, rubbing on the carbon seals, and synchronous rotor instability. In the following sections, each potential root cause is briefly described along with reasons for/against the likelihood of it being the main culprit.

Rotordynamics Problems

Once the new problem was uncovered, the user cast a skeptical eye on the rotordynamics analysis that had been used to justify the changes in coupling and bearing design. This analysis, which is described in detail by Gutzwiller and Corbo (2001) had predicted low synchronous vibration levels (less than 1 mil peak-to-peak at both blower bearings) and had also predicted the rotor to be highly stable against subsynchronous vibrations (all damped natural frequencies above operating speed). Since the rotor model was identical to the one that had been used to identify the first synchronous vibration problem and which had predicted the observed synchronous vibration amplitudes with excellent accuracy, there was not much question about its validity. The main items that were in question were the tilting-pad bearing stiffness and damping coefficients that had been employed in the analysis of the new system.

Although the authors had a great amount of confidence in their tilting-pad bearing analysis code, in order to remove any lingering doubts, they hired another rotordynamics expert to independently calculate the bearing coefficients using one of the most widely-accepted tilting-pad bearing computer codes in the world. The coefficients that were obtained with the second code were then compared to the first set of coefficients and all were found to be in good agreement (less than 10 percent deviation). Most importantly, the rotordynamics analysis was rerun with the second set of coefficients and the predictions were found to be virtually identical to those obtained with the first set of coefficients. Accordingly, the rotordynamics analysis was concluded to be valid and it was, therefore, determined that the observed vibration problem was not of rotordynamic origin. This was not much of a surprise since the symptoms reported previously are hardly characteristic of a rotordynamics problem.

Starvation of the Tilting-Pad Bearings

The fact that the occurrence of the high vibrations seemed to be related to whether there were one or two blowers running led the troubleshooting team to search for systems that the two blowers had in common. One of the first ones identified was the bearing oil feed system—the tilting-pad bearings for both blowers (as well as the motor's sleeve bearings) are fed by a common oil system. It was, therefore, theorized that when only one blower was running, the oil supply system would feed the tilting-pad bearings all the oil that is needed to keep them operating flooded and the system would run smoothly. However, when both blowers were running simultaneously, the oil pump (a positive displacement design) could not supply sufficient oil to the tilting-pad bearings, causing them to run starved. In such a case, all the predictions of the rotodynamics analysis, which were based on the bearings operating flooded, would be invalidated. Corcoran, et al. (1997), present a case where the starvation of tilting-pad bearings led to a vibration problem in precisely this manner. This argument was strengthened further by the observation that the new tilting-pad bearings required larger supply flows than the previous plain cylindrical bearings.

In an effort to make a quick check of this theory, the user arbitrarily increased the oil supply pressure from 25 to 40 psig at a time when both blowers were running and both were suffering from high vibrations. The change was observed to have no impact on the high vibrations.

The authors checked into this situation further by performing a rigorous calculation of the oil flows required to maintain fully-flooded conditions in both of the blower's tilting-pad bearings. Such a calculation involves using standard orifice equations to set all the inflows to the bearing cavity equal to all the flows that will leak/drain out of the cavity and calculating the resulting average pressure level in the cavity. The hydrostatic pressure equation can then be used to convert this pressure into the height of oil present in the pressurized cavity. If this height is sufficient to keep all the tilting pads immersed in oil, then the bearing will operate flooded.

As with many tilting-pad bearing designs, the oil supply to each of these bearings was controlled via an inlet orifice upstream of the bearing. For this machine, two separate calculations were needed since the two bearings had different orifices and the supply system for the outboard bearing also had to provide flow to the unit's tapered land thrust bearing.

The inlet orifice originally used with the blower's inboard bearing was sized to give a supply flow of 0.8 gpm. For this orifice size, the calculations revealed that the cavity pressure was only 0.0071 psi, which was not nearly enough to keep the bearing flooded. Conversely, the 2.6 gpm that was being fed to the outboard bearing was found to be more than sufficient to keep the journal bearing flooded. However, since the thrust bearing, which is in series with the journal bearing, was calculated to need about 2.9 gpm (not including the flow to the inactive side), the thrust bearing was concluded to be operating in a starved condition.

During this exercise, one other item was noted with regard to the inboard bearing's oil supply system. That is, although the bearing has four pads, the original design only contained orifices supplying oil to the leading edges of two pads. Since it is the authors' experience that tilting-pad bearings perform best when all pads have their own supply of fresh, cool oil to their leading edges, even in flooded systems, this arrangement was considered to be less than optimum.

It should be noted here that it is the authors' experience that the above flow calculations tend to err on the conservative side. In actual practice, one can usually feed these types of bearings with somewhat lower flows than those calculated using the above procedure and still obtain satisfactory performance. How much lower is a matter of considerable debate.

Nevertheless, to ensure that bearing starvation was not the cause of the vibration problem, the orifice sizes were modified to increase the supply flows being provided to each bearing to the values that the calculations dictated. The supply flow to the inboard bearing was raised to 2.74 gpm and that to the outboard bearing was increased to 6.36 gpm. Additionally, at the authors' suggestion, two additional orifices were added to the inboard bearing to ensure that all four pads would be supplied with fresh oil at their leading edges.

When all these changes were implemented to the "B" blower, their impact was observed to be … absolutely nothing. The pump discharge pressure was measured and verified to remain at 25 psig, ensuring that the pump was supplying the orifices with the desired flows. However, these increased flows had no impact on the unit when it was experiencing high vibrations. Consequently, bearing starvation was ruled out as a potential cause of the problem.

Inadequate Thrust Load

The observation that high vibrations were always accompanied by an axial motion of the rotor from the active thrust bearing toward the inactive bearing led some members of the team to suspect that the cause of the problem was inadequate thrust loading on the blower wheel. This theory proposed that under certain conditions of operation, the thrust load on the wheel was reduced to a small enough level that it was not capable of holding the rotor against the active side of the thrust bearing. At these times, the rotor would drift away from the active thrust face and cause the high vibrations to commence.

This behavior was difficult to understand since the calculated thrust load acting on the blower wheel was large, approximately 3400 lb. Furthermore, detailed calculations by the blower manufacturer revealed that the most that this value could vary by was about ±10 percent over the expected system operating range. Thus, the theory that the thrust load was too light appeared to be ludicrous, at first.

However, later in the program the user measured the actual blower inlet density and static pressure rise during a period of high vibrations and they were found to be much lower than expected. Additionally, the volumetric flow rate was discovered to be much higher than expected. Using the measured parameters, the blower manufacturer recalculated the wheel's thrust load and found it to be only about 250 lb, about an order of magnitude lower than the calculated value for nominal conditions. This much lower thrust load made the axial drifting of the rotor from the active thrust surface much easier to understand than it previously had been.

In order to determine whether the low thrust load was the cause of the vibration problem, the user ran an experiment. The next time one of the blowers experienced high vibrations, the user closed the fan dampers to greatly reduce the volumetric flow and, thereby, greatly increase the aerodynamic thrust load. Although doing this seemed to momentarily reduce the high vibrations, it did not take long for the vibrations to build up to their previous high level.

In the authors' minds, this settled a "chicken-egg" type question that had received considerable debate amongst the task force. Namely, was the axial drifting causing the high radial vibrations or was it the other way around. After increasing the thrust was found to have no impact on the high vibrations, the authors concluded that the high radial vibrations caused the axial drift, not the other way around.

The authors never really believed otherwise since they could not come up with a credible reason why axial motion of the rotor away from the active thrust surface would lead to high radial vibrations. In an extensive search of the literature, the authors only found one reference, that of Besigk, et al. (1990), where such a phenomenon occurred. In the reference, they found that a radial vibration problem was greatly improved when the load on the thrust bearing was increased. The reason for this is that the thrust bearing was located at a vibration node and the thrust runner, therefore, was tilting back and forth as the rotor vibrated in the radial direction. This tilting generated damping in the fluid-film thrust bearing. Since the thrust bearing's damping increases substantially as the load on the thrust bearing increases, it was found that increasing the thrust load above a certain threshold level was sufficient to eliminate the radial vibration problem.

This blower resembles the machine in the reference since it also contains a fluid-film thrust bearing and the thrust bearing (which is located at the position of the outboard journal bearing) is located close to a node when the vibrations at the inboard bearing are high. However, since none of the authors or their colleagues have personally seen anything like this, such a scenario was considered to be a longshot, at best.

Blower Axial Vibrations

Since the equipment that was recording the rotor's axial position was only measuring the axial position of the rotor, not the axial vibration, it was theorized that the axial position changes of the rotor that always seemed to accompany the high radial vibrations might be due to axial vibration. This hypothesis assumed that there were certain conditions under which the rotor would reach axial resonance. Once the rotor started vibrating axially, it would free itself from the active thrust bearing and drift toward the inactive side. The axial vibrations were theorized to be converted into the observed radial vibrations by some unknown coupling mechanism.

In order to evaluate the possibility of this, the authors constructed a simple one degree of freedom mass-spring model. Since the gear coupling was assumed to separate the blower and motor shafts from one another, the blower rotor was taken to be the vibrating mass. The spring was simply the fluid-film stiffness of the tapered land thrust bearing. Using the normal blower wheel aerodynamic load of 3400 lb, the thrust bearing's stiffness was calculated and used to calculate the system's axial natural frequency. Since the natural frequency was calculated to be 10,100 cpm and the unit runs at 1780 rpm, this path, initially, appeared to be a dead end.

However, once it was discovered that the actual thrust load could be as small as 250 lb, everything changed. When informed of the reduced thrust load, the authors recalculated the thrust bearing's axial stiffness, which, naturally, was much lower than it had been under the larger thrust load. When the unit's axial natural frequency was recalculated, it turned out to be 1797 cpm, almost a dead-on resonance with the 1780 rpm operating speed. It was, therefore, theorized that when the blower's flow conditions were such that the aerodynamic thrust load was reduced to around 250 lb, the rotor could enter into resonance with its axial mode.

After this finding was made, the user changed their instrumentation to measure the rotor's axial vibration, as well as axial position changes. They then discovered that during periods of high radial vibrations, the rotor was also exhibiting axial synchronous vibrations of from 1 to 2 mil peak-to-peak. Although this finding made the axial vibration theory plausible, the authors remained skeptical that it was the cause of the problem. In the authors' experience, the percentage of turbomachinery vibration problems that are triggered by axial vibration is extremely small, due to the lack of excitations acting in the axial direction. Chen and Malanoski (1974) provide a description of one of the few turbomachinery axial vibration problems that the authors are personally acquainted with. In fact, the only types of rotating equipment that the authors are aware of where axial vibrations play a significant role are marine propulsion systems. As is described by Kane and McGoldrick (1949) and Ehrich (1992), the primary reason for this is the large axial excitations that are generated by the propeller.

Blower Surging

Blower surging was also considered a potential culprit since, if the blower were to enter a surge mode, the axial thrust on the wheel could easily drop to zero (or reverse itself) and the unstable condition could certainly generate high radial vibrations in the rotor. Since it was originally believed that the blower rotor was normally loaded toward the active thrust bearing with a sizable load (3400 lb), the troubleshooting team reasoned that it would take something drastic, like a surge condition, to force the rotor to move away from the active thrust surface. Additionally, since both blowers shared a common inlet duct, the likelihood of surge would depend on whether there were one or two blowers running.

In order to check into the likelihood of this, the authors reviewed the blower's aerodynamic performance curve. It was concluded that the blower's operation is highly stable at the normal operating point with no chance of significant pressure pulsations or surging. Furthermore, pressure readings taken at the blower inlet and discharge revealed none of the fluctuations that are characteristic of the surge condition.

Additionally, it is the authors' experience that when surge generates rotor vibrations, the vibrations are almost always nonsynchronous, not the synchronous vibrations that were being observed. Smith and Wachel (1984) agree, stating that flow instability phenomena, like surge, almost always generate subsynchronous vibrations. This is in agreement with a case reported by Ferrara (1977) where surging of a centrifugal compressor generated subsynchronous vibrations at approximately 10 percent of the running speed. Interestingly enough, this reference also observed axial motions of the rotor during surging. However, for all the above reasons, blower surging was eliminated as a possible cause of the problem.

In a related topic, acoustic resonances in the blower inlet or discharge lines were also briefly theorized to be a potential root cause. However, the absence of any pressure pulsations in these locations indicated that this was not the problem. Additionally, in the authors' experience, if an acoustic resonance were to occur, it would most likely generate vibrations at vane-passing frequency (12×), not 1×. This is consistent with the experiences of Schwartz and Nelson (1984).

Thermal Loosening of the Fan Wheel

The fact that the high vibration phenomena seemed to be time-related (i.e., high vibrations only seemed to occur after at least a week of running) led the authors to strongly suspect that the problem was, somehow, thermally related. Since the fan wheel is titanium and the shaft is steel, it was theorized that thermal growth effects could, somehow, cause the tapered interference fit that holds the wheel to the shaft to be lost after a certain period of time. This would allow the wheel to "flop" around on the shaft, causing the unbalance load to exceed its expected value, and resulting in a corresponding increase in synchronous vibration amplitudes.

Although this argument made good sense qualitatively, the numbers refuted it. A detailed study, using finite element analysis, performed by the authors revealed that the worst case combination of thermal and centrifugal effects could not come close to causing this interface to lose its original interference (15 mil diametral at room temperature). Thus, the authors could not imagine any scenario under which thermal effects could cause the wheel to run loose and this potential cause was, thereby, dismissed from consideration.

Unloading of the Blower Inboard Bearing

It is the authors' experience, and that of Palazzolo, et al. (1992), that the frictional loads generated within gear coupling teeth can be substantial. In fact, they have seen cases on overhung machines such as this one where these frictional loads were large enough to offset the gravitational load on the inboard bearing and cause that bearing to operate essentially unloaded. To test this possibility, the worst case in which the gear coupling's friction exactly cancelled out the gravity load on the inboard bearing was assumed. The stiffness and damping coefficients for this bearing were calculated for the zero load condition and the rotordynamics analysis was rerun with those coefficients. The unbalance response results turned out to be almost identical to those obtained with the nominal gravitational load of 282 lb. This extreme insensitivity to external loading is one of the many benefits reaped from the employment of tilting-pad bearings with geometric preload and offset pivots. The rotordynamics analysis, therefore, eliminated this potential cause from consideration.

Thermally-Induced Misalignment

The belief that this problem was somehow thermally-related led the task force to suspect thermal misalignment between the motor and blower as a potential cause. Detailed calculations and measurements verified that the motor and blower shaft centerlines moved several mils in both the horizontal and vertical directions with respect to each other under various thermal conditions. However, in the authors' experience, the manner in which these types of misalignments generate radial vibrations is by radically altering (i.e., reducing) the loads on the bearings, thereby dramatically changing (i.e., reducing) their stiffness and damping coefficients. An example in which this occurred in a steam turbine-driven compressor is reported by Martin, et al. (1978).

In order to check this theory, the rotordynamic analysis was run with both blower bearings in the unloaded condition. This would represent the most severe case that misalignment could impose. As was found in the previous section, the tilting-pad bearings were found to be highly insensitive to the load placed on them and the rotordynamic behavior was found to be virtually unchanged. The rotordynamic model was, therefore, again useful in dismissing another potential cause.

Rubbing on the Carbon Seals

The authors have worked with machines having symptoms similar to (but not as extreme as) this one where the culprit was an intermittent rub between the rotating shaft and a stationary structure, usually a seal. In those machines, certain thermal and/or misalignment conditions allowed the shaft to rub on the seal. This rubbing would generate a hot spot on the shaft, thereby causing the rotor to bow thermally, resulting in high synchronous vibrations in the forward direction. Nathoo and Crenwelge (1983) report of very similar behavior that occurred in a steam turbine.

The authors felt that the rubbing theory had a lot of potential for several reasons. First, in machines suffering from rub-induced vibrations, the high vibrations tend to appear and disappear in an unpredictable fashion. This blower is nothing if not unpredictable. Second, rub-induced vibrations do not tend to build up gradually—they tend to appear and disappear instantaneously, similar to this machine's vibrations (Figure 3). Third, the dependence of the high vibrations on time and weather conditions (i.e., rainstorms) would make sense since the rubs would most likely only occur under certain thermal conditions. Finally, rubs are probably the most well-known cause of reverse precessions in rotating equipment, and there were unconfirmed "reports" that reverse precessions had occasionally been observed on this machine.

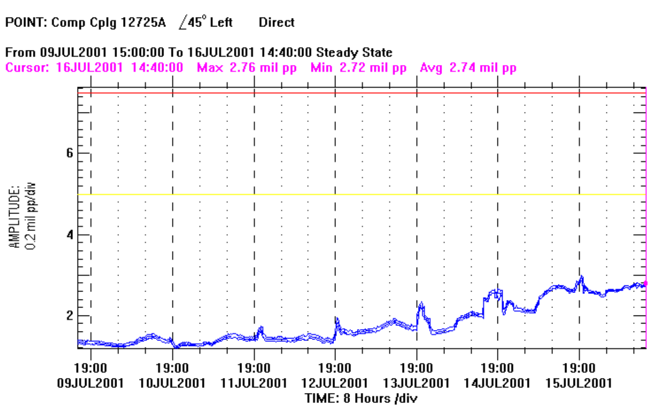


Figure 3. Vibration Amplitude Versus Time.

However, there were several members of the task force who opposed this theory based on an experiment that was performed early in the lives of these units. On one occasion when the original blower (with the disk coupling and plain cylindrical bearings) was suffering from high vibrations, a rub was suspected. The carbon seals were, therefore, removed from the unit and the unit was restarted. The vibrations were then observed to remain equally high and the rub was dismissed from consideration. The reason the authors did not consider this to be pertinent to the present question is that it involved the original unit, not the modified one. Since the original unit was known to suffer from high synchronous vibrations of rotordynamic origin, the fact that it was not rubbing at that time was not terribly surprising.

A detailed analysis of the carbon-ring shaft seal, which is shown in Figure 4, was performed by a third party. They recommended some slight modifications of the sealing faces of the carbon seal rings to reduce the side pressure loading. This was done to allow the carbon rings to more easily adjust to rotating shaft position changes. However, they concluded that, when properly installed, the carbon ring seal did not seem to have any inherent design characteristics that would lead to the intermittent vibration problem observed.

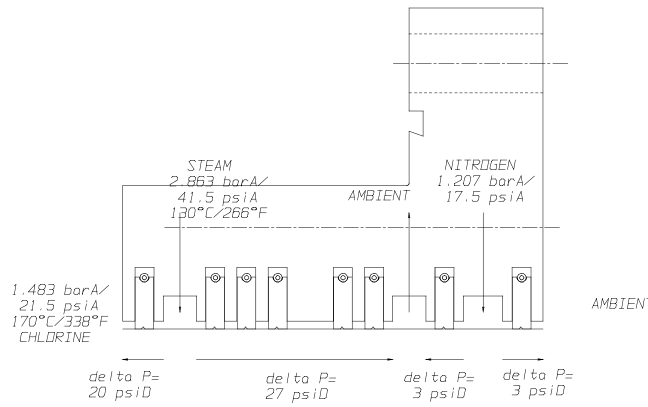


Figure 4. Eight-Stage Carbon-Ring Shaft Seal (Purged).

Shortly thereafter, shaft orbit plots during operation at "normal" and "high" vibration periods were finally collected (Figures 5, 6, 7, and 8). Inspection of these figures reveals clear evidence of both forward and reverse precession. Accordingly, in spite of the third party's findings, the authors continued to consider carbon seal rubbing as a primary candidate for the cause of this problem.

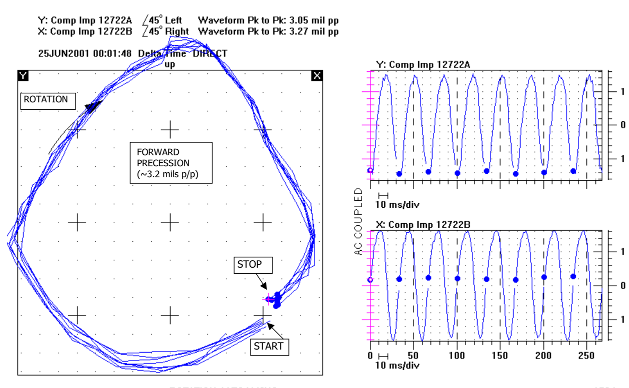


Figure 5. Shaft Orbit—Forward Precession.

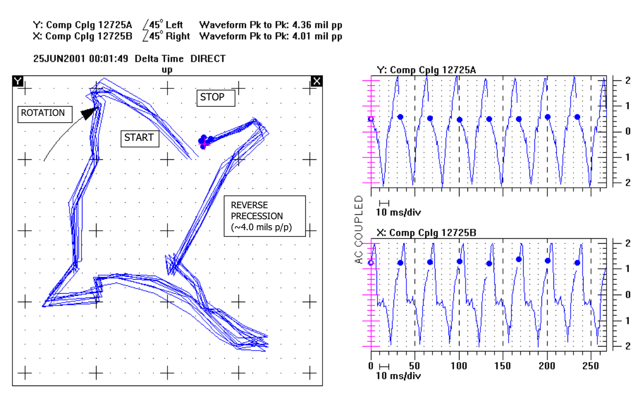


Figure 6. Shaft Orbit—Reverse Precession 1.

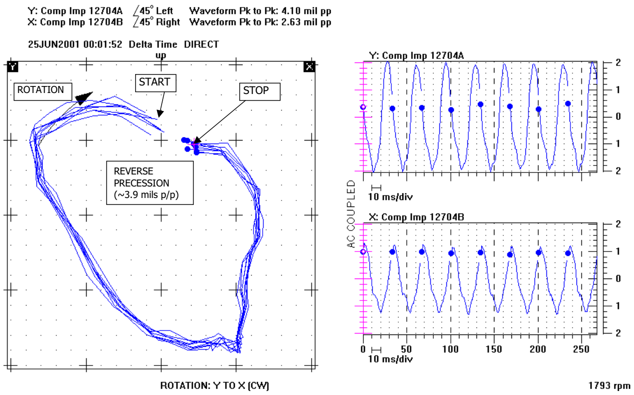


Figure 7. Shaft Orbit—Reverse Precession 2.

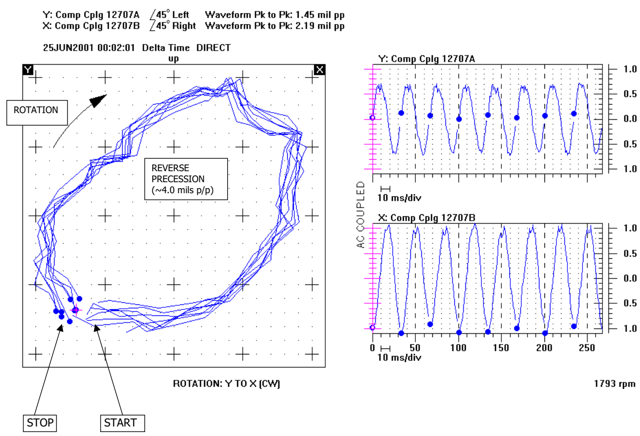


Figure 8. Shaft Orbit—Reverse Precession 3.

Synchronous Rotor Instability

This phenomenon, also referred to as the Morton Effect, is one which did not enter into the discussion until late in the troubleshooting effort. De Jongh and van der Hoeven (1998), Faulkner, et al. (1997), and Kirk and Balbahadur (2000) all give good descriptions of this phenomenon. In a nutshell, this effect is related to the fact that when a rotor is synchronously whirling within fluid-film journal bearings, one portion of the rotor remains at the bearing's minimum film thickness location at all times. Since the point of minimum film thickness is generally at a higher temperature than the remainder of the points around the bearing circumference, this can generate a temperature gradient across the shaft, which causes the shaft to bow, thereby increasing the synchronous whirling. Under certain thermal and mechanical unbalance conditions, this can generate an unstable situation under which increased bowing causes a greater thermal gradient that leads to more bowing, etc. This system was never checked for this effect since the problem was solved, as described in the following sections, before any checks could be performed.

FINAL DIAGNOSIS

After a complete review of all the data, observations, theories, "hunches," etc., the user requested that the authors provide them with their conclusions on the cause of the problems and to present an action plan. After summarizing the history of this very complex problem, they gave the following diagnosis:

> "Based on the information available, we believe that the most likely cause of the high radial vibrations is a thermally-induced rub at the carbon seal rings. While earlier inspections have shown no evidence of a metal-to-metal rub at the seal, it is certainly possible that forces exist that could result in binding of the free-floating carbon rings, which would result in a rub. We believe this for the following reasons:
>
> - The orbit plots show that both the "A" and "B" blowers are whirling in the backward direction during periods of high vibration. The only practical mechanism that can cause backward whirling is a rub.
> - In addition to the backward whirling, the blowers have also been observed to exhibit synchronous forward whirling during periods of high vibrations. If a rub were to occur, it would generate a local hot spot on the rotor, which would lead to a thermal bowing of the rotor. A bowed rotor would generate large synchronous vibrations in the forward direction.
> - The plant's monitoring expert has testified that he has seen evidence that the blowers were rubbing during periods of high vibration (this testimony was not uncovered until the time of this report). He also mentioned that when the rubbing was observed to stop, the radial vibrations were observed to instantaneously drop from 4 or 5 mil to around 1 mil peak-to-peak. The rubbing and high vibrations were, therefore, seen to go hand-in-hand with each other.
> - While the seal housings look okay, the carbon rings show extensive evidence that rubbing (and hot spots) has been occurring.
> - The occurrence of high vibration has been observed to be highly sensitive to time and thermal conditions (i.e., rainstorms). If rubbing were the cause of the vibrations, it might only occur when thermals caused sufficient radial misalignment to create contact between the rotating shaft and the carbon rings. Thus, rub-generated vibrations would be highly sensitive to both time and thermal conditions.
> - The observed behavior where the vibrations tend to greatly increase and decrease almost instantaneously suggests that some kind of instability is at play here. A rub would certainly qualify as an instability."

PROBLEM SOLUTION

After review of this report, the user shut down one of the blowers and the carbon ring shaft seal was carefully inspected. In three places, titanium antirotation detent plates (that prevent the carbon rings from rotating) were found to have come loose. These titanium parts were galled to the seal housing and rubbing on the rotating shaft sleeve, even to the point of causing a buildup of material on the shaft sleeve in two places (Figures 9 and 10).

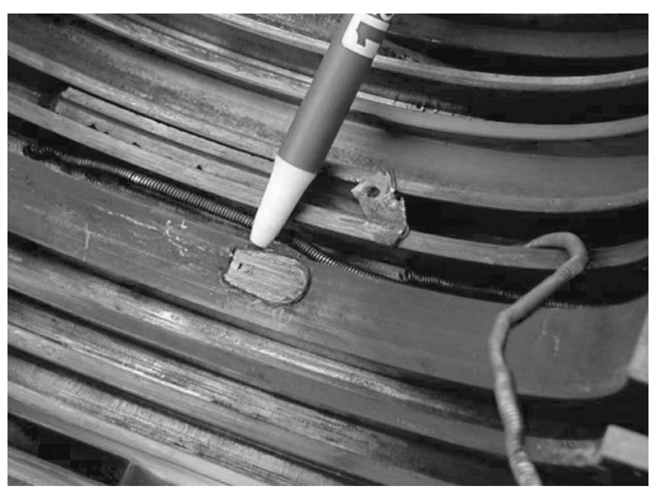


Figure 9. Antirotation Tab "Smeared" onto Seal Housing.

Figure 10. Seal with Slots for Antirotation Tabs.

This confirmed the authors' belief that the high vibrations were due to rubs between the shaft and the carbon ring seals. It was further concluded that the large radial vibrations were generating accompanying axial motions and forces and that the 250 lb aerodynamic thrust load was too low to prevent the rotor from moving away from the active thrust bearing once large radial vibrations commenced.

Once the cause of the problem had finally been uncovered, the only remaining task was to generate corrective action to prevent the antirotation plates from coming loose again. This problem was discussed with the seal manufacturer. After studying the photos and the vibration history of these blowers, they advised that the loosening of the antirotation plates was most likely associated with an improper assembly procedure. The correct procedure calls for the seal garter springs to pass through the holes in the antirotation detent plates. "If this was not done, it could have led to damage to the detent plates, the springs, and to the carbon rings." The manufacturer felt that it is more or less impossible for the detent plates to come loose from their mountings (when properly installed) unless the springs were broken. Although the seal manufacturer claimed that a broken garter spring is a rare occurrence, the plant had reported several broken springs.

In order to rectify the problem, the shaft seal antirotation detent plates were redesigned to eliminate the possibility of their becoming loose or falling out of position again during operation. A pin was also added to hold them in place. In addition, the clearance between the carbon rings' inner diameter and the rotating shaft sleeve was increased to 12 to 14 mil on a diameter. Finally, the carbon rings were grooved to decrease the side (axial) loading due to the purge gas pressure. This would allow the rings to "float" more easily in the radial direction to accommodate thermal expansion, etc.

Figure 11 shows grooved carbon rings, Figure 12 shows antirotation pins and detent plate, Figure 13 shows carbon rings and garter springs in place, and Figure 14 shows a completed seal assembly.

After some minor adjustments in the radial and axial clearances of the fan's tilting-pad bearings, the two fans were restarted. A report from the plant after about a year of operation stated, " . . . the two blowers have operated flawlessly, with one minor incident for the past year. The one issue: vibration rose on one of the blowers, but it was quickly traced to loose bolts on one of the bearing housings. Vibration disappeared after tightening the nuts."

Figure 11. Carbon Ring with Groove Added to Reduce Axial Loading.

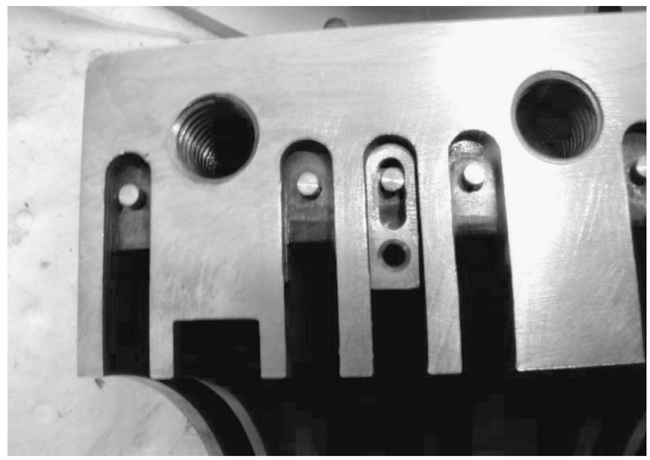


Figure 12. Shaft Seal Housing with Pins Added (for Antirotation Tabs).

Figure 13. Reassembled Shaft Seal Showing Carbon Rings, Springs, Etc.

CONCLUSION

Troubleshooting is very difficult on rotating machinery that is critical to continuous process operation. There are generally conflicting reports of symptoms. Theories as to the nature and cause of the problem abound, and there is commercial pressure to take corrective action immediately. The process of obtaining a solution is especially complicated by erratic machinery behavior, where problems seem to come and go unpredictably.

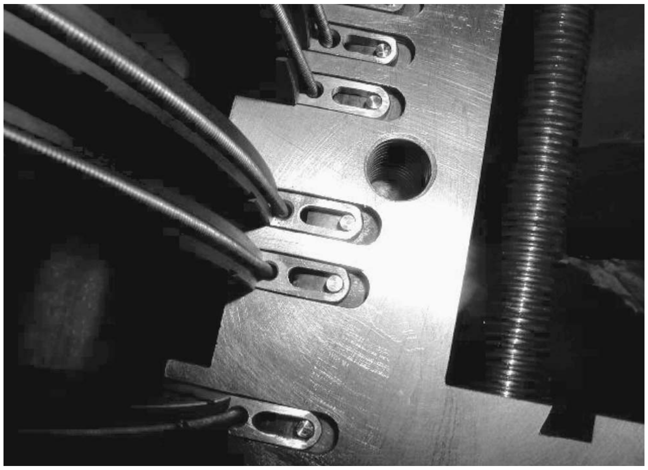


Figure 14. Revised Shaft Seal—Assembled.

For a team of problem solvers to be successful, they must:

• Look at the big picture. In the initial stages of this effort, very little progress was made because the team was focused on the rotor bearing system, to the exclusion of everything else. It was not until the team began working on a system level that the solution to the problem was uncovered.

• Take a systematic approach. Unfortunately, in the high pressure, "crisis" atmosphere that these types of problems tend to generate, teams often fall into the trap of taking a "scatter-gun" approach to solving the problem. However, that often simply leads to chaos. Instead, an organized, well-thought out, methodical approach is more likely to reap dividends.

• When dealing with rotating equipment vibrations, a good rotordynamic model is a must. Several of the proposed causes of this problem could not have been dismissed without the rotordynamic model.

• Listen carefully to *all* the observed symptoms . . . in this case observed slow vibration increases over relatively long periods, effect of weather changes on vibration levels, etc.

• Gather hard data . . . in this case shaft centerline plots showing forward *and* reverse precession, low oil flow rates versus the bearing requirements, etc.

• Review of case histories and first-hand experience.

In this high pressure environment, it is rare that the team ever has *all* the data it would like to have available. Therefore, the team must be able to, at some point, draw their very best conclusions from the information available, and take action. In this case, the conclusions drawn and recommendations made led to an inspection of the carbon shaft seal, redesign of the antirotation detent plates, and a solution that was successful.

REFERENCES

Besigk, C., Blank, W., Ohlmann, G., and Schulz, P., 1990, "Self-Excited Natural Vibrations of a Turbogenerator Set and their Elimination," Proceedings of the Third International Conference on Rotordynamics, IFToMM, Lyon, France, pp. 565-569.

Chen, H. M. and Malanoski, S. B., 1974, "Investigation of Compressor Axial Vibration for Machinery Train 18C1 at Koch Refining Company," MTI Report EA74-6, Mechanical Technology Incorporated, Latham, New York.

Corbo, M. A. and Malanoski, S. B., 1998, "Pump Rotordynamics Made Simple," *Proceedings of the Fifteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 167-204.

Corcoran, J. P., Rea, H., Cornejo, G. A., and Leonhard, M. L., 1997, "Discovering, the Hard Way, How a High Performance Coupling Influenced the Critical Speeds and Bearing Loading of an Overhung Radial Compressor—A Case History," *Proceedings of the Twenty-Sixth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 67-78.

Crandall, S. H., 1982, "A Heuristic Explanation of Journal Bearing Instability," Proceedings of the Workshop on Rotordynamic Instability Problems in High-Performance Turbomachinery, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 274-283.

de Jongh, F. M. and van der Hoeven, P., 1998, "Application of a Heat Barrier Sleeve to Prevent Synchronous Rotor Instability," *Proceedings of the Twenty-Seventh Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 17-26.

Ehrich, F. F., 1992, *Handbook of Rotordynamics*, New York, New York: McGraw-Hill.

Faulkner, H. B., Strong, W. F., and Kirk, R. G., 1997, "Thermally Induced Synchronous Instability of a Radial Inflow Overhung Turbine, Part I," Proceedings of the 1997 ASME Design Engineering Technical Conferences, Sacramento, California, Paper DETC97/VIB-4063.

Ferrara, P. L., 1977, "Vibrations in Very High Pressure Centrifugal Compressors," *Proceedings of the ASME Design Engineering Technical Conference*, ASME, Chicago, Illinois.

Gutzwiller, H. L. and Corbo, M. A., 2001, "Vibration and Stability of 3000 HP Titanium Construction Chemical Process Blower," *Proceedings of the First International Symposium on Stability Control of Rotating Machinery*, Bently Nevada Corporation, South Lake Tahoe, California.

Kane, J. R. and McGoldrick, R. T., 1949, "Longitudinal Vibrations of Marine Propulsion Systems," Transactions of the Society of Naval Architects and Marine Engineers, Jersey City, New Jersey, pp. 193-252.

Kirk, R. G., and Balbahadur, A. C., 2000, "Thermal Distortion Synchronous Rotor Instability," Proceedings of the Seventh International Conference on Vibrations in Rotating Machinery, University of Nottingham, United Kingdom.

Lund, J. W. and Saibel, E., 1967, "Oil Whip Whirl Orbits of a Rotor in Sleeve Bearings," ASME Journal of Engineering for Industry, pp. 813-823.

Martin, L. J., Malanoski, S. B., and Lewis, P., 1978, "Rotor-Bearing System Analysis of Turboblower," MTI Report 78TR86, Mechanical Technology Incorporated, Latham, New York.

Nathoo, N. S. and Crenwelge, O. E., 1983, "Case History of a Steam Turbine Rotordynamic Problem: Theoretical Versus Experimental Results," *Proceedings of the 1983 Annual Meeting of the Vibration Institute*, Houston, Texas, pp. 81-89.

Newkirk, B. L. and Taylor, H. D., 1924, "Shaft Whipping Due to Oil Action in Journal Bearings," General Electric Review, pp. 559-567.

Palazzolo, A. B., Locke, S. R., Calistrat, M. M., Clark, R. W., Jr., Ayoub, A., Calistrat, D., and Tang, P., 1992, "Gear Coupling Misalignment Induced Forces and Their Effects on Machinery Vibration," *Proceedings of the Twenty-First Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 83-96.

Pan, C. H. and Sternlicht, B., 1963, "Comparison Between Theories and Experiments for the Threshold of Instability of Rigid Rotor in Self-Acting, Plain Cylindrical Journal Bearings," ASME Paper 63-LubS-3.

Rentzipis, G. M. and Sternlicht, B., 1962, "On the Stability of Rotors in Cylindrical Journal Bearings," ASME Journal of Basic Engineering, pp. 521-532.

Schwartz, R. E. and Nelson, R. M., 1984, "Acoustic Resonance Phenomena in High Energy Variable Speed Centrifugal Pumps," *Proceedings of the First International Pump Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 23-28.

Smith, D. R. and Wachel, J. C., 1984, "Experience with Nonsynchronous Forced Vibration in Centrifugal Compressors," Proceedings of the Workshop on Rotordynamic Instability Problems in High-Performance Turbomachinery, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 37-52.

ACKNOWLEDGEMENTS

The authors would like to recognize the contributions of Stan Malanoski of No Bull Engineering and John Whalen and Clark Maloney of Turbo Components Engineering who provided valuable experience and insight during the troubleshooting effort. They also greatly appreciate the efforts of Audra Dodgson of Robinson Industries who assisted in the preparation of this manuscript. The authors would also like to thank Dr. Dara Childs and Robert Perez of the Turbomachinery Symposium Advisory Committee for their invaluable feedback and encouragement. Finally, they would like to thank Joanne Burnett of the Turbomachinery Laboratory, without whose patience and cooperation, this paper would never have made it to publication.