Forty-Eight Case Histories of Intriguing Machinery Problems

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For 45 years, *Sound & Vibration* magazine has been published. Technological developments over that time period have allowed machine analysts to obtain ever more sophisticated vibration data on troubled equipment. These technological improvements have greatly aided troubleshooting specialists. However the art of diagnosing equipment problems relies as much on understanding the equipment being analyzed as it does on the technology of the instruments being used.

This series of case histories illustrates why a vibration specialist needs to comprehend how machines operate as much as they need to understand the complex instrumentation used to do the diagnosis. These cases cover a span of 35 years, which parallels a good portion of the span of the time that *S&V* has been in business. Each case in itself could be an article, but to show the diversity of problems that can be encountered, they have been condensed as much as possible. Forty-eight case histories are being represented, one for each year of publication of *Sound & Vibration* plus a bonus.

Case 1. 650-Megawatt Turbine

In the mid 1970s, a 650-megawatt turbine threw a piece of shroud off of its first-stage blades. To run the turbine until a new set of blades arrived, four blades under the damaged shroud area were removed. To keep the rotor in balance, four blades were removed on the opposite side of the blade group. The turbine came to speed without any issues, but as load was applied, the vibration grew worse as the load was increased. A vibration signature showed that all the vibration was at two times running speed.

To reduce throttling losses, large turbines sequence their valve openings so that one valve opens, then a second and so on. This is done instead of opening all the valves a small amount, thereby incurring throttling losses.

Analysis of the problem determined that the vibration was the result of the two gaps in the first-stage blade group going in and out of the active steam arc. The solution was to put the turbine into full arc operation, where all the valves open evenly and accept the throttling losses at low loads. When this was done, the 2× vibration disappeared, and it was possible to run the turbine until the new blades could be installed. This case illustrates why it is necessary to understand how a machine works to diagnose a problem and come up with a solution (see Figure 1).

Case 2. Power Plant Fan Motor

A large fossil generating station had two new units with two-speed 5500 HP motors that drove the ID Fans. While at low speed, the vibration at 120 Hz was found to be very high on one of the motors. The data were collected with a newly acquired swept-filter analyzer. Further testing showed that all the fans had the same problem. Levels of 1.0 in/sec were present on all the motors in the axial direction when they were operating at low speed. The manufacturer was contacted and admitted that there was a problem, and a motor of the same design had recently failed at another plant.

Diagnosis showed that the stator windings were resonant at 119.5 Hz. That combined with the winding sequencing of the low-speed coils caused the high vibration. Based on that knowledge, all four motors were sent back under warranty to the manufacturer on a planned and carefully orchestrated basis. This one case saved the utility enough money to fund its vibration monitoring program for many years.

Case 3. Pump Bearings

Three identical pumps sat side by side. One pump's bearings failed repeatedly. Pumps were swapped from the other two locations, but they always failed when put in the troubled location. Prior to taking any vibration readings, the vibration analyst noticed that the pump shaft slowly shuttled back and forth in the axial direction. Based upon the analyst's knowledge of pumps, this observation meant that the pump was most likely dead-headed. A measurement of the discharge pressure proved that it was indeed operating with almost no flow. This condition occurred when the tank to which it was pumping was full and the pump was put in a recirculation mode.

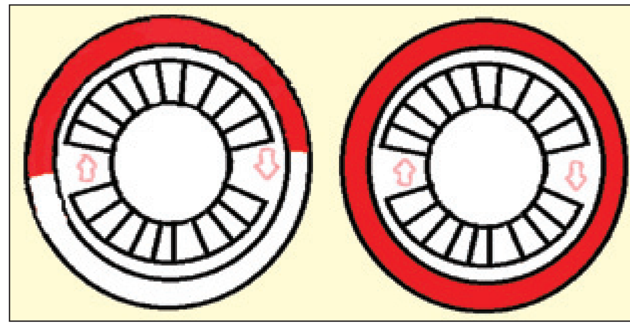

Figure 1. Steam admission to wheel; partial arc (left), gaps go in and out; full arc right, gaps are always in steam path.

Figure 2. Melted 500 megawatt generator phase lead.

Examining the recirculation line showed that there was supposed to be a 3-inch orifice in the line. The orifice tab said it was only 2 inches. When it was removed, the orifice actually only had a 1-inch hole. The pump bearings were failing due to the axial shuttling caused by the pump operating with almost no flow because of the improper bypass line flow orifice. This case illustrates why it would be impossible to solve the problem by looking at failing bearings with a spectrum analyzer. It was necessary in this case to understand what happens when a pump has insufficient flow.

Case 4. Pump Motor Vibration

Three pumps sat beside one another on a concrete foundation. Two of the motors on the pumps operated with low levels of vibration. The third motor had high levels of 120-Hz vibration. As with the previous case, when the good motors were put in the third location, the 120-Hz vibration was high.

A resonance test was performed, and it was discovered that, where the vibration levels were high, the foundation had a 120-Hz vertical resonance that was not present under the other two motors. When the foundation under the bad location was jack hammered out, it was discovered that the reinforcing bar had not been welded as designed. The last two case histories show that even though things may appear the same from the outside, there can be unseen problems that can affect either the process or the dynamic tuning.

Case 5. Cracked Hub of Fan

For several months the vibration on an exhaust fan had been changing. The fan could be balanced, but in a short time, the

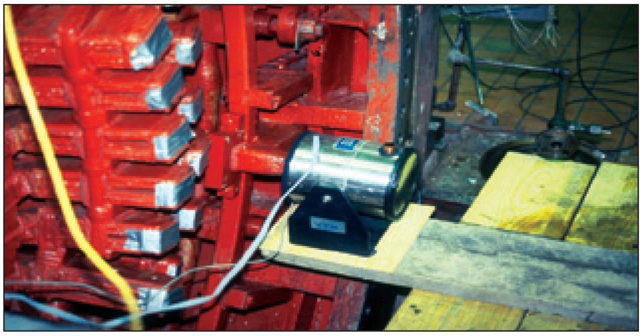

Figure 3. Electro-magnetic shaker testing phase leads.

high levels of vibration would return. The fan was examined for uneven dirt build-up on the blades but was always found to be clean. Amplitude and phase readings were monitored and found to vary with time. It was concluded that the fan was loose on its shaft, so the fan wheel was replaced, which solved the problem. The fan wheel was sent out to the scrap area, and several weeks later, following exposure to rain and weather, cracks were found in the hub by the presence of rust bleeding from the cracks.

This case provided a valuable lesson that if the vibration on a fan varies with time and there is no apparent cause, then it is advisable to perform a dye-penetrant check on the fan wheel and hub.

Case 6. 500-MW Generator Failure

During the early 1970s, a 500-MW generator failed catastrophically. It had an internal short that resulted in a phase-to-phase short. The generator stopped so suddenly that it sheared off 24, 2.5-inch bolts in the coupling between the generator and the turbine. The problem was traced to the phase leads' resonance shifting to 120 Hz as the generator aged. This problem resulted in the annual testing of the phase leads for the unit that had the failure and also on its sister machine.

One of the key lessons learned during many hours of testing these generators was that for electrical components, the stiffness of the insulating materials is affected significantly by temperature. For instance, if a phase lead tested as acceptable at room temperature, it might experience a 10-15 Hz drop when it was at operating temperature, thereby being unacceptable in the loaded state. The extent of this effect was discovered by accident when the windows to the turbine building were left open on a cold night. It was discovered that the natural frequency results on the phase leads were significantly different when the leads were cold versus at room temperature.

The lesson to learn here is to always be observant. The natural frequency of a vertical pump made of steel will not vary much whether the surrounding temperature is zero or 100 degrees, but electrical components that are covered with insulation can change (see Figures 2 and 3).

Case 7. Nuclear Plant Control-Rod Cooling Fan

While consulting for a nuclear power plant, a call was received that a control-rod cooling fan had vibration levels that had increased since its last outage. Examining the fan found no reason for the increase in the running-speed vibration. The fan wheel was clean, the bearings were in good condition, the coupling was OK, the hold-down bolts were tight, and there was no sign that any balance weights had been thrown. Despite no changes in the fan, the amplitude had more than tripled.

This was a case where it was advantageous to have knowledge in three areas, nuclear engineering, organic chemistry and dynamics. The first question that was asked was if the fan was on isolators, and the answer was "yes." The second question was whether the isolators were springs or rubber. They were rubber. Based on that knowledge, the problem was identified and quickly solved. The rubber isolators were exposed to gamma and neutron radiation from the reactor. This radiation destroyed the covalent bonds of the carbon-compound-based isolators, making them harder. This changed the tuning of the isolators, moving the isolated natural frequency (verified by an impact test) close to the running speed.

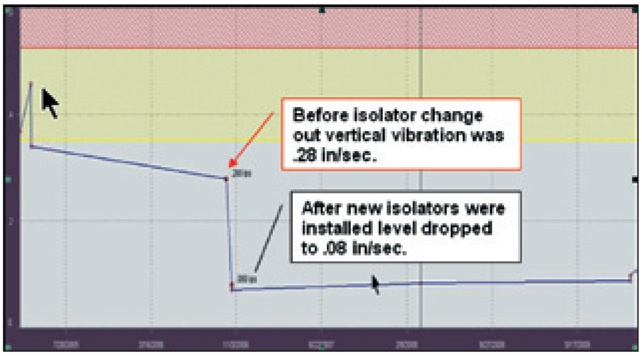

Figure 4 Vertical fan vibration levels before and after installing new isolators.

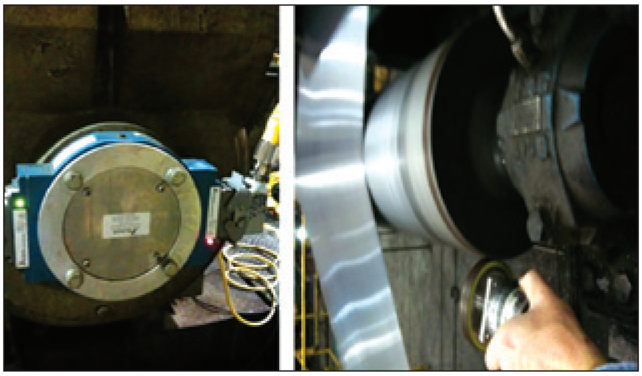

Figure 5. Torsional test setups: Shaft encoder on main rolls (left) and portable encoder for testing other rolls.

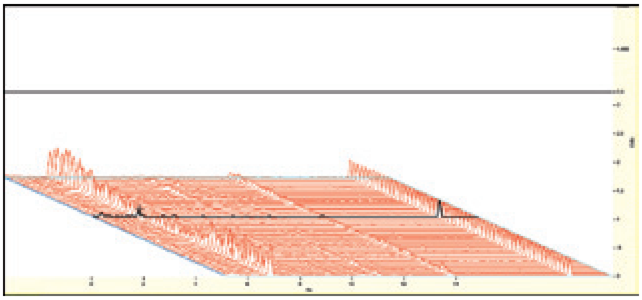

Figure 6. Map plot shows that when tension is reduced, vibration disappeared.

On the response curve, the system went from the isolation region to the amplification region. New isolators were installed, and the vibration amplitudes returned to their historical levels.

In this case, the instrumentation was used to confirm that the natural frequency of the isolated system had changed. However, the key to identifying the cause of the problem was to have an understanding of the nuclear environment and the chemistry of the isolators along with a knowledge of the nature of the dynamic response curve. Another important point to be learned from this case is that isolators are maintenance items. They do not last forever and any changes alter the dynamic response of the system (see Figure 4).

Case 8. Steel Plant Galvanization Line

A steel mill in the central part of the United States experienced a vibration problem on its galvanization line. The entire 300-foot-long line would vibrate at 7 Hz. Initial tests indicated that this frequency was nine times the bridle-roll speed, but further evaluation using higher resolution data and a tach pulse as a speed reference showed that the 7-Hz frequency was not an exact match to a multiple of the large bridle-roll speed. A consultant also wrote a lengthy report saying that the 7-Hz frequency matched one of the roll bearing's outer-race defect frequencies. Since it was doubtful that an outer-race defect could cause a 300-foot galvanization line to vibrate at high levels, a second opinion was requested. Torsional testing of the rolls verified that the entire system was oscillating at 7 Hz and that as the line speed varied; this frequency would change in direct correlation with the steel sheet speed.

In another test, when the tension on the plate was reduced, the vibration dropped significantly. The movement of the large structure was tied to the line speed and varied with the tension on the steel plate. This combination of factors pointed toward a control-related problem. The root cause was traced to bowed furnace rolls located near the tension control load cell. The bowed rolls would cause the tension to increase and decrease. This in turn resulted in the control system trying to compensate by increasing and decreasing the tension at the speed of the bowed rolls. Replacement of the rolls and desensitizing the control system solved the problem. (see Figures 5 and 6).

Case 9. Paper Mill Drive Motor

As noted in the previous case, control systems can cause vibration problems. For this particular case history, an unusual frequency (87.5 Hz) was present on the drive at a paper mill. The motor had been overheating and had failed. Using standard vibration analysis techniques, nothing mechanical was found that could explain the presence of the 87.5-Hz frequency. A current probe was then installed on the motor to measure any variations in current flow. The 87.5-Hz frequency seen in the vibration spectra also showed up clearly in the current spectrum. Based on that information, a controls person was brought in to adjust the drive controls. The 87.5-Hz component completely disappeared when the drive was tuned.

If you think about it, control systems are lot like vibrating systems. A control system wants to maintain a particular condition such as speed, tension or flow rate. In the case of speed, an encoder accurately measures speed and tells the system at what speed it is operating. The control system then makes a correction to move from the given speed to the desired speed. Depending on the amount of dead band and the response sensitivity of the system this can result in an oscillating response as the speed overshoots high then low. In this instance, the mechanical system consisted of a motor, gear box, long drive shaft and large roll. That complex mechanical system was then controlled by a speed-based control system. This resulted in the interaction of the electrical control system with the complex mechanical system, thereby creating the mystery frequency.

Case 10. 5000-HP Compressor Torsional Vibration

A 5000-HP compressor driven by a synchronous motor was experiencing high vibration during startup. On one startup, one of the shafts made contact with a seal and destroyed the shaft and the impeller. This case required both advanced instrumentation and knowledge of the equipment to solve. Since the vibration problem occurred during the brief 8-second startup period, it was necessary to install instruments that could record and post-process the data.

The data revealed that the vibration was high at the instant the motor was energized and then six seconds later, just prior to synchronization. The frequency of the vibration in both cases was the same (900 CPM). The second peak just prior to synchronization was the worst, with the vibration level using up all the bearing clearances.

The problem was determined to be that the first torsional mode was being excited initially by the motor's starting impulse and even more important in the second instance by the synchronous motor's magnetic torsional stimulus. Synchronous motors produce a torsional stimulus that starts out at 7200 CPM at zero speed and drops to 0 CPM when the motor synchronizes. At 6 seconds into the startup, when the 1200-RPM motor reached 1050 RPM, the torsional stimulus equaled the torsional natural frequency of the motor compressor combination, resulting in large oscillations of the bull gear, which translated into high levels of movement of the compressor stages being driven by the bull gear. The solution to the problem was to change out the coupling (see Figure 7).

Case 11. 750-RPM Power Plant Fan

A 5000-HP, 720-RPM fan at a power plant had above-normal levels of unbalance that resisted balance efforts. Several attempts were made to balance the fan, all of which were unsuccessful.

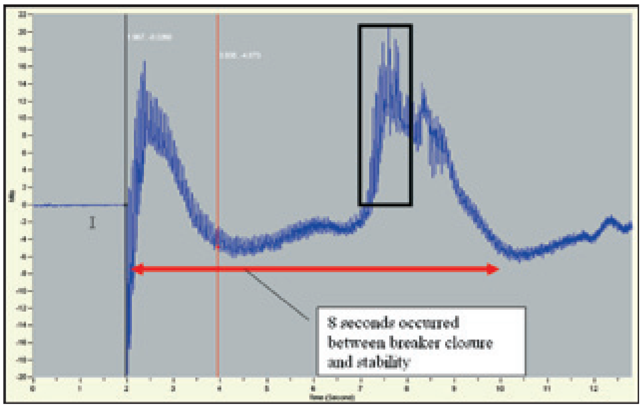

Figure 7. Torsional problem.

After attempting several balance shots in an induced-draft fan, the casing vibration levels were reduced to around 2-3 mils but would not go any lower. Since there was difficulty in balancing the fan, shaft stick measurements were taken to determine the absolute motion of the shaft.

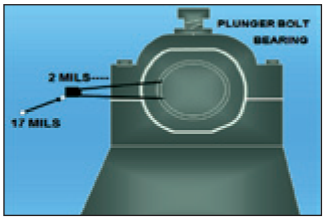

Figure 8. Motion of fan shaft.

The shaft movement was discovered to be more than 17 mils. Since the bearing clearance was only 8 mils, this was a strong indication that the bearing was moving in the housing. A large plunger bolt at the top of the bearing was tightened. After tightening, the casing vibration increased from 2 to 3 mils to 21.5 mils. The fan was then easily balanced to below the 1-mil level desired by the plant personnel. What had been occurring was that, with the bearing moving in the housing, a nonlinear system was present. This made balancing all but impossible (see Figure 8).

Case 12. Incorrect Selection of Isolators.

Several air handler units in New York City all had the same symptoms. The fans operated with acceptable levels of vibration; however, the motors for the belt-driven fans all had high levels of vibration in the vertical direction. When the spectra were examined, it was discovered that the primary component on the motor vibration signatures was at the fan's operating speed. An investigation resulted in finding that the isolators under the motor were sized improperly.

The isolators were too stiff. This resulted in the natural frequency of the isolated system matching the operating speed of the fans. The isolators were therefore acting as amplifiers of the fan vibration rather than isolators of the motor vibration. What appeared to have occurred is that the same size isolators were used for the motors as were used for the heavier fans.

Case 13. High Axial Vibration on Fan

A belt-driven exhaust fan operated at 1200 RPM. The fan's axial vibration was always high. The fan would be balanced and then a couple of weeks later, the axial vibration would go back up. Due to the sensitivity of the fan to unbalance, a resonance was suspected, so natural frequency tests were performed. There was no natural frequency match when the fan was struck in a lateral direction. However, when the fan was impacted axially, there was a match with running speed.

A mode shape found that the shaft was the node point and that the opposite sides of the fan were out of phase. This is commonly called a disk wobble natural frequency. Fans that have this problem exhibit sensitivity to unbalance, particularly in the axial direction. The solution in this case was to simply change the fan's speed. If the speed had not been an option, then stiffening of the back plate of the fan wheel would have been necessary.

Case 14. High Cross Effect from Fan to Motor

A motor had high levels of vibration. Its amplitudes were higher than on the fan it was driving, so the representative from the motor shop that had worked on the motor tried to balance the motor without success. Past experience on other large fans had shown that this might have been fan unbalance, instead of motor unbalance. When the phase angles were measured, the fan phase angles were leading the motor angles. Based on these data, the fan was balanced even though it had lower levels than the motor.

Balancing the fan brought the motor levels down to very acceptable amplitudes. The thing to remember is that the fan rotor will generally have a much higher level of polar-rotating inertia than a motor. It is quite common for a large fan to shake a motor, but much less common for a motor to shake a fan (see Figure 9).

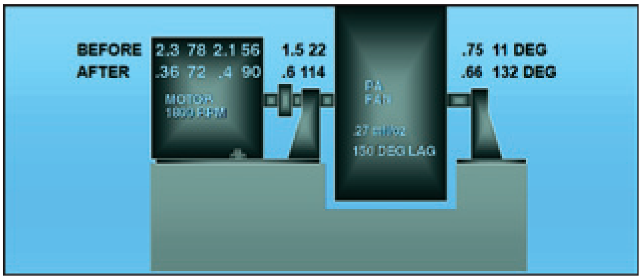

Figure 9. Cross effect from fan to motor.

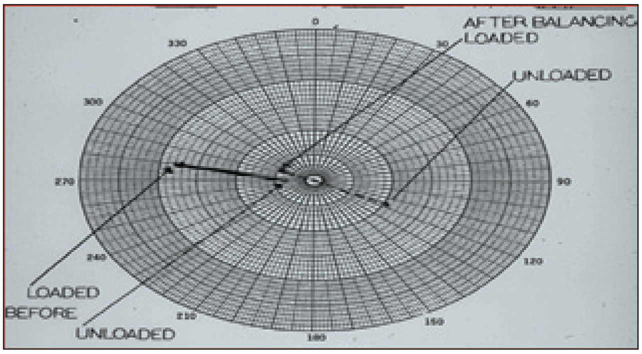

Figure 10. Balance shot.

Case 15. Unequal Air Gap on Motor.

An electric motor at a foundry was exhibiting high levels of vibration. In addition to having above normal levels of unbalance, it had almost 0.4 in/sec of 120-Hz vibration. When the motor was examined, it was discovered that there was a 0.035-inch variation in the air gap from the top to the bottom of the motor. Air gaps are measured by using long feeler gauges to measure the clearance between the rotor and the stator. This was a static air gap deviation, meaning that when the rotor was turned, the narrow gap remained in the same location. The magnetic interaction of the rotating field produced extra pull on the rotor at the narrow point. This extra pull on the rotor was translated to the bearings producing the 120-Hz vibration.

Case 16. Large 3600-RPM Motor with Thermal Vector

During a telephone conversation with an engineer at a refinery in Venezuela, a problem was described that sounded a bit unusual. The discussion went something like, “ever since we balanced our large motor, it doesn’t run well any more.” Further conversation helped clarify the situation. When the plant was brought down for an outage, the motor was run unloaded. During the unloaded operation, it was noted that the vibration on the proximity probes was more than 3 mils. Based on this amplitude, the plant personnel had elected to balance the motor. After balancing, the amplitude was approximately 1.5 mils.

Everything seemed to be going well until the plant was restarted and load was applied to the motor. As the motor was loaded, the amplitude went up to nearly 4 mils. To analyze the vibration, the plant engineer recorded the proximity probe signals along with a tach pulse signal. That data were sent back to the U.S., where it was analyzed on a Bently Nevada ADRE system. The analysis of the data showed that the motor had a large thermal vector. What had occurred was that the motor had been compromised-balanced in the past.

The compromise shot had been designed to let the motor have high levels of vibration in the low load condition, where it seldom operated. This decision was made so that when the thermal vector, which was 180 degrees out of phase from the low load vibration, took effect, the motor would operate with low levels at full load, where it spent most of its time. What had occurred was that when plant personnel had balanced the motor at low load, they had undone the compromise balance shot.

This story would have been interesting enough if it had ended at that point, however, the saga continued. Due to the presence of the large thermal vector, plant management elected to purchase another motor. Nothing was specified regarding thermal vectors in the new motor, and only a shaft vibration limit of 2 mils was mentioned in the purchase specifications. When the new motor was tested, it was found that it also had a thermal vector. In fact, the manufacturer stated that they routinely had to put in compromise balance shots on that model of their motors. They had performed approximately 128 such operations on that model of motor since it was introduced.

The moral of the story is that when dealing with 2-pole motors above 1000 HP, be aware of the fact that the 1× running speed vibration may vary significantly from low to high load. It is imperative that phase angles be recorded along with the amplitude to identify this type of situation. For instance, a motor might have 2 mils at 90° unloaded and 2 mils at 270° in the loaded condition. If only amplitude is recorded, there would appear to be no problem. However, if phase is taken into account, then it becomes apparent in this case that there has been a 4 mil change from low load to high load. This is therefore a situation where you do not want under any circumstances to balance out the 2 mils at low load, because if such an action is taken, then the vibration will increase at full load.

Case 17. Feed Pump Motor with Thermal Vector

A 4000-HP, 3600-RPM feed pump motor was sent for a normal inspection and cleaning. After returning from the motor shop, it was put into operation, and after 45 minutes, high vibration destroyed its bearings. It was returned to the motor shop where it was rebalanced. When it was returned to the plant, it again destroyed the bearings. It was then sent back to the manufacturer and put in a high-speed balance pit and balanced at speed. When it was reinstalled back at the plant, it destroyed the bearings for a third time.

Due to the nature of the problem, proximity probes were installed on the motor. When it was first started, the vibration was normal. However, as the motor was loaded, the vibration level increased to the point that the motion was nearly equal to the bearing clearance. It was determined that the motor had a thermal vector.

The solution was to balance the motor in the loaded condition. It operated with this thermal compromise shot installed for several years. It was discovered later that the motor shop had dropped the rotor on its first visit. This had damaged the laminations, causing a hot spot to develop. This hot spot on one side caused the rotor to bow as it heated up, thereby producing the sensitivity to load (see Figure 10).

Case 18. Cracked Rotor Bars

While an 1800-RPM, 250-HP water service motor was in operation, a noticeable variation in the sound pattern of this motor was evident. The current meter also showed oscillations in current draw. Based on these symptoms, the motor was connected to a dynamometer, and spectra of the current were obtained. The spectra were taken at various loads and showed the presence of side bands spaced at the number of poles times the slip frequency in both the current and vibration spectra.

Since this is a sign of broken rotor bars, the motor was disassembled and the rotor was re-barred. Following the repairs, the sidebands disappeared, and the sound and current oscillations also went away (see Figure 11)

Case 19. Cracked End Rings

A series of motors was tested at a processing facility; they all

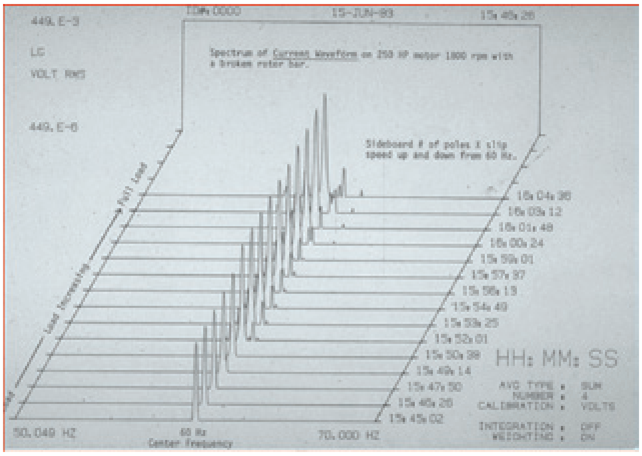

Figure 11. Cracked rotor bar analysis.

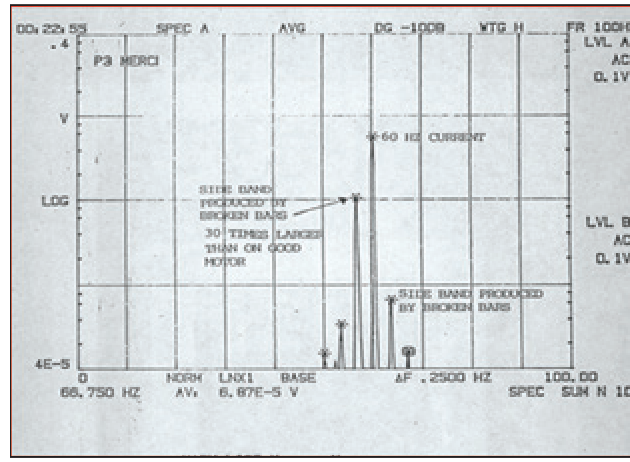

Figure 12. Cracked end ring analysis.

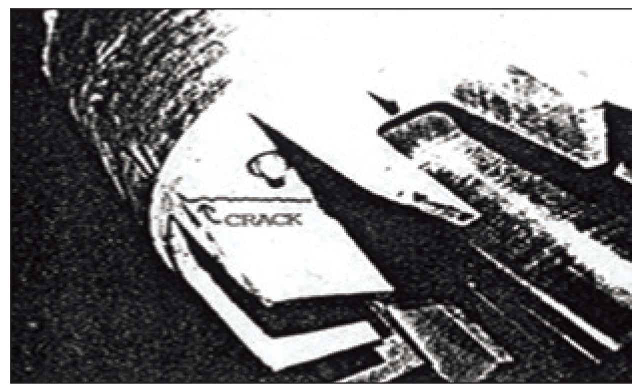

Figure 13. Cracked rotor bar.

powered large centrifuge units. The test consisted of putting the output from a current probe that was clamped around one of the phase leads into a spectrum analyzer that was connected to a computer. The computer had an expert system program that then checked the current spectrum for side bands that were related to problems with the rotor. The motor's amp meter in the electrical equipment room showed significant oscillation of the motor's current. In addition, the vibration on the motor and the sound coming from the motor varied in a periodic manner.

These observations were all consistent with a motor that has either broken rotor bars or cracked end rings. Despite all the symptoms that were observed, the expert system indicated that the motor was in good condition. Since there was so much evidence to the contrary, the current spectrum was examined. There was a very large lower side band 210 CPM (3.5 Hz) below 60 Hz. The side band was only 15 dB below a 60 Hz signal. All the evidence pointed toward a rotor problem. The expert system was over-ruled and the motor was sent in for an examination. Three cracks all the way across one end ring were discovered.

The reason that the expert system had missed the problem was that the motor was in such bad condition that the amount of slip had increased to a point that it pushed the side bands out of the expert system's normal side-band search range. A normal motor might have from 15 to 30 CPM slip, which on a four-pole motor would put the search range in the 60 to 120 CPM (1-2 Hz) above and below 60 Hz. This motor had 52.5 CPM of slip, which a placed the side bands at 210 CPM (3.5 Hz) above and below 60 Hz (see Figures 12 and 13).

Case 20. Rotor Eccentricity

Vibration analysis is a humbling profession. About the time you think you have something figured out, along comes a special case that proves there is still a lot to learn. This problem occurred on a 3600-RPM ash sluice pump motor. The motor had high levels of 1× running speed vibration that varied significantly in a periodic manner. The motor also sounded like it had a beat.

When a high-resolution zoom spectrum was taken, it was clear that the variation in level was not due to a beat, which results from two closely spaced frequencies, but was instead due to modulation. When there is a beat, the two signals' vectors add together. When modulation is present, one signal is multiplied by the other thereby producing side bands. The zoom spectrum showed that there were side bands in this case 70 CPM above and below the operating speed of the motor. The operating speed of the motor was 3565 RPM. This meant that the side bands were appearing at the number of poles two times the slip frequency (35) above and below the running speed.

This is a classic sign of either broken rotor bars or cracked end rings. To test that theory, a current probe was used to perform a current spectrum check. In the current spectrum, current modulation side bands are generated at the number of poles times the slip frequency around 60 Hz (50 Hz in countries with 50 cycle current). If the side bands in the current spectrum are larger than –54 dB as compared to the 60 Hz current, then there is the likelihood of a problem. When the current spectrum was examined, it was discovered that the side bands were –65 dB down from the 60 Hz component. This meant that it was very improbable that there was any problem related to the rotor bars.

Since the running speed vibration was also high, a coast-down curve using the peak hold feature of a spectrum analyzer was performed. The coast-down curve showed that the 1× running speed vibration dropped off rapidly as the rotor coasted down. Since that is an indication of a resonance, an impact test was performed on the rotor. To get the effect of the sleeve bearing's oil film, the rotor was rotated during impacting. The results of the impact test indicated that the rotor had a natural frequency of 3525 CPM. After having conversations with plant personal, it was discovered that the rotor had been changed on this motor. The original motor had an aluminum cast rotor. The rotor with the natural frequency problem was a laminated design with copper bars.

Based on the field test results, due to the proximity of the operating speed to the first natural frequency, the rotor was bowing when it was at operating speed. This bow resulted in a rotating air gap deviation. As the magnetic poles passed the narrow part of the rotating air gap deviation, there was extra magnetic pull that was translated into modulation of the running speed vibration. The rate at which the magnetic fields passed the rotating air gap was, as would be expected, the number of poles times the slip frequency.

To test this theory, it was decided to finely tune the balance level on the rotor. The thought was that even though there was a resonance, with the forcing function being reduced, the rotor would bow less and the side bands would therefore be reduced. The rotor was then balanced to as low a level as was practical. When it was tested following the balance work, the side bands had completely disappeared. Due to the reduced forcing function, the rotor was no longer operating in a bowed conditioned, thereby eliminating the rotating air gap and the resulting side bands. The owner was informed that because of the close proximity of the operating speed to the natural frequency that the problem would most likely reappear.

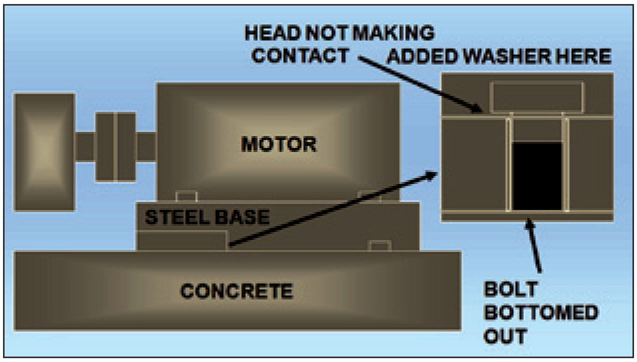

Figure 14. Motor vibration 8 mils; bolt tightened but not making contact; when washer added, vibration fell to beow 1 mil.

Case 21. Loose Base

When analyzing motors, it is easy to read more into a problem than actually exists. This case illustrates the need to always look for the simple solution first. While at a power plant working on a large fan, the vibration analyst requested that a large mill motor be examined. The motor had been sent out to motor shops twice. In each instance, the vibration remained high when the motor was put back into operation. Attempts were made to field balance the motor, but they were unsuccessful. The motor continued to have amplitudes of 8 mils of vibration, all of which were at its operating speed.

The first thing that was checked was the tightness of the motor to its base plate. All the bolts were tight. The next thing that was verified was the tightness of the base plate to the concrete pedestal. The bolts felt tight; however, motion between a bolt head and the base plate was apparent. What had happened was that the bolt had bottomed out. This meant that even though the bolt felt tight to the mechanic with a wrench, no force was applied between the bolt head and the plate. To solve the problem, a washer was installed. After the installation of the washer, the vibration dropped from 8 mils down to less than 1 mil. Over $30,000 had been spent on this large motor at the repair shops and the problem had been nothing more than the need to add a washer to a bolt (see Figure 14).

Case 22. DC Motor's Bad SCRs

On three phase motors with rectified current, six SCRs fire every 1/60th of a second. This results in a normal firing pattern of 360 Hz (300 Hz in 50 Hz countries). This 360-Hz signal will often appear on normal operating DC motors. In the case of this motor, a 120-Hz signal appeared. A 120-Hz signal is not that uncommon in an induction motor, because that is the rate at which magnetic poles pass a stationary element. When 120 Hz appears on a DC motor, however, it is an indication that there is an SCR or firing problem. The motor that was being tested had the 120-Hz signal as well as several harmonics of 120 Hz. In addition, it was noisy and overheating.

When a current probe was used to examine the waveform, it was discovered that the SCRs were not firing properly. Replacement of a bad SCR solved the vibration and heating problems and also quieted the motor (see Figure 15).

Case 23. DC Motor Process Related Vibration

An unidentified vibration was detected on a DC motor driving a couch roll at a paper mill. The frequency of the vibration did not match any known source. A current spectrum was taken on the motor, and it was discovered that the same unidentified frequency appeared in the current spectrum. What was occurring was that the motor was being loaded and unloaded at the unknown frequency.

The source of the current oscillations was traced to the fan pump blade-pass frequency. The fan pump, which was in the basement, was generating pressure pulsations that caused oscillations in the head box pressure. This in turn resulted in variations in the pulp thickness. When the thicker areas passed the vacuum rolls, the suction pulled harder against the fabric. This increased tension in the fabric caused the tangential force to increase on the couch roll, and that increased torque demand on the motor and varied the

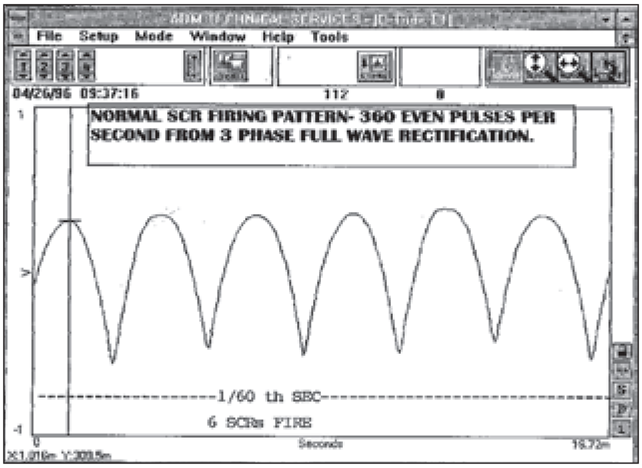

Figure 15. Waveform from pump drive following replacement of bad SCR.

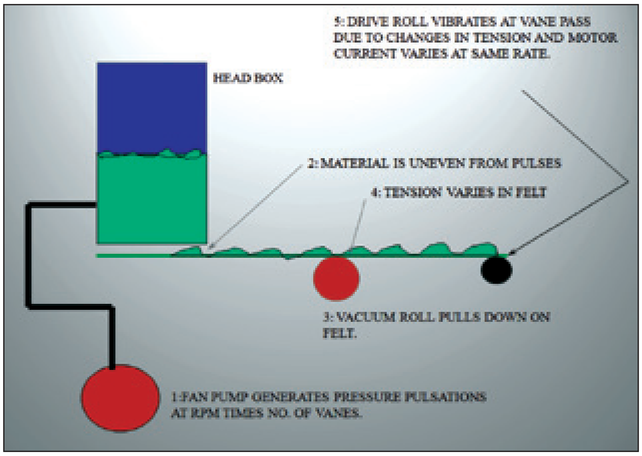

Figure 16. Paper machine problem.

amount of current draw. The fan pump was eventually replaced with one that generated much lower pressure pulsations. This improved the paper quality and eliminated the vibration on the couch roll drive (see Figure 16).

Case 24. Line Harmonics

After encountering a series of motor failures, a manufacturing facility requested assistance in finding the cause of the increase in failures. A consultant travelling around the area informed the plant management that the cause of the problem was that the local utility was supplying "bad power" that had several harmonics of 60 Hz present. Harmonics in the current supplied to motors or other electronic components cause excess heat and premature failure. Therefore, the utility requested that tests be performed to determine if the power they were supplying was the cause of the failed motors.

A dual-channel analyzer was used for the tests. A voltage input from a transformer was input into Channel A, and a current input from a current probe was input into Channel B. When the first measurement was made, an unusual amount of the fifth harmonic of the line frequency was noted. On the time waveform, it was also noted that instead of being smooth, the wave had a rough erratic appearance. Since the rough appearance could be from the rapid firing of SCR drives, the plant was asked if they had that type of drive in the plant. They responded that they had several such drives being used in their processes. They also stated that the drives had been in operation for a number of years, but that the failures had only started to increase the last few months. The plant engineer was then asked if any changes had been made to the electrical system. The only thing he could think of was that power factor correction capacitors had been added to cut their electric bill.

With this information, additional tests were performed. The current waveform and spectra were obtained with the power factor correction capacitors racked out and then with various amounts

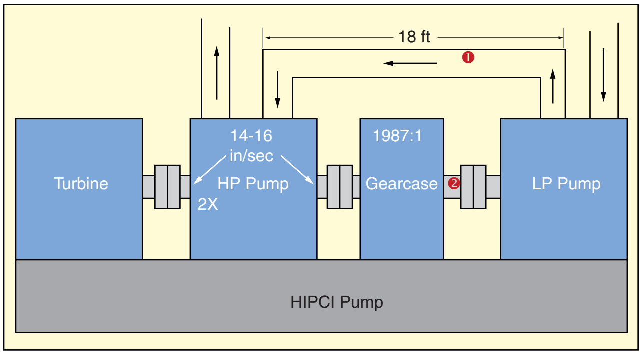

Figure 17. Pump layout.

of capacitance added back in. The results were dramatic. With the power factor correction capacitors all racked into the system, the fifth harmonic increased to a level that was 25% of the 60 Hz frequency. That meant that if 100 amps were flowing to a motor, there would be 25 amps of 300-Hz current also flowing through the motor. This would result in severe overheating in motors or transformers. As it turned out, the utility had nothing to do with this problem. The SCR drives were supplying the stimulus and the power factor correction capacitors in combination with the other electrical components formed a circuit that resonated at 300 Hz.

To solve this type of situation, it is necessary to have a power engineer design filters to lower the 300-Hz stimulus. Based on tests at other facilities, the seventh and eleventh harmonics can also cause problems.

Case 25. Nuclear Plant High-Pressure Core Injection Pump

Operators of nuclear power plants are required to comply with Section XI of the ASME code regarding in-service inspection. Prior to the time of this case, the code stated that baseline readings were to be taken in mils displacement and that if the baseline readings doubled, then action had to be taken. The code was then changed to measure vibration in velocity and this problem then appeared.

The machine train consisted of a turbine driver, a high-pressure pump, a gear case and a low-pressure booster pump. Vibration spectra were obtained for all the bearing locations on the pump train. The readings were at normal levels everywhere, except on the high-pressure pump in the horizontal direction. A level of 1.4 inches per second was present at what appeared to be twice the running speed of the high-speed components (see Figure 17). A cascade plot of vibration on the HP pump in the horizontal direction from the inboard end to the outboard end showed that the vibration was high at each end of the pump but was nearly zero in the center.

Due to the large difference between the vertical and horizontal readings and the presence of an apparent rigid body-pivoting mode, a horizontal resonance was suspected. An impact test was performed on the pump. A natural frequency at near twice running speed was found. This mode matched the response found while the pump was running (i.e., high on the ends and low in the middle). Since the vibration was predominately at what appeared to be twice running speed, it was suspected that misalignment might be the source that was exciting the horizontal structural resonance. The alignment was checked and found to be out of specifications. The alignment was corrected, and a test was run on the unit. There was no improvement in the level of vibration. In fact it was slightly higher on the test run than it had been previously.

This change in the level of vibration followed a pattern that showed that the vibration in the 2× filter bin would vary from 0.9 to 1.6 in/second from one test run to another. In an attempt to determine the phasing of the vibration, a once-per-revolution pulse was used as a reference trigger for the FFT analyzer, and a signature was taken using synchronous time averaging. As the number of averages increased, the vibration at what appeared to be twice running speed disappeared. This was one of the breakthroughs in the analysis of the problem. It meant that the vibration was not phase locked to the high-pressure pump shaft. Following the results of the synchronous time-average test, the pump train drawings were

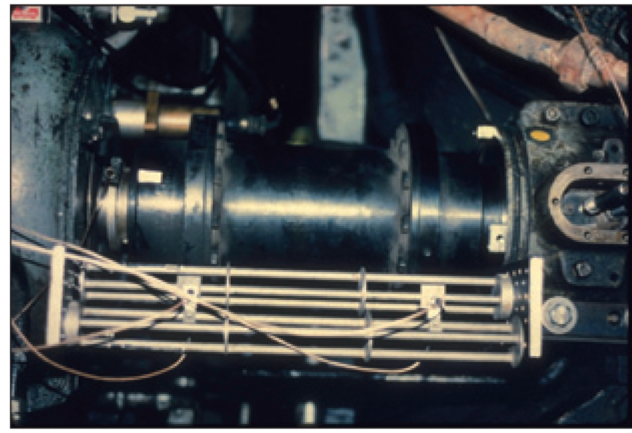

Figure 18. Dynalign bar setup.

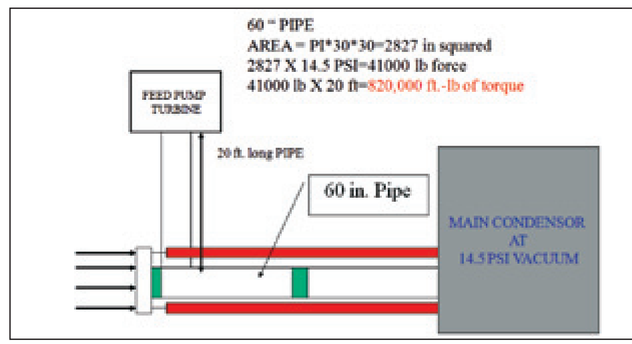

Figure 19. Force calculations.

re-examined. It was found that the gear case had a 1.987:1 reduction. In addition, it was discovered that the low-pressure pump had a four-vane impeller. The pieces were beginning to fall together.

The pump manufacturer was contacted with this information. The pump representative recalled that there had once been a case of an acoustical resonance with a similar pump. To determine if this could be contributing to the problem, the piping between the low- and high-pressure pump was measured. It was found that the length of pipe connecting the pumps was equal to one-half wave length of the low-pressure pump blade pass frequency. Since the low pressure pump had four vanes and the gear reducer had a 1.987:1 reduction, the vane pass frequency looked exactly like twice the running speed of the high-pressure pump. As a final test to confirm this theory, a tach pulse was put on the low-pressure shaft. The vibration pickup was placed on the high-pressure pump. The vibration on the HP pump was found to indeed be phase locked to the low-pressure pump.

The problem on the high-pressure pump was not at twice the HP pump running speed, as it had appeared. It was actually at the vane pass frequency of the low-pressure pump. It appeared to be at twice the HP pump running speed, because of the four-vane impeller in the LP pump and the almost exact 2:1 speed reduction between the two pumps. The problem was amplified by the acoustical resonance of the pipe connecting the discharge of the LP pump to the suction of the HP pump. The vibration was further amplified by the horizontal structural resonance of the HP pump casing.

The major clue to the solution of the problem was that the vibration on the HP pump was not phase locked to the shaft on that pump and was therefore coming from another source. The final clue was the past acoustical resonance problem encountered by the pump manufacturer. The recommenced solution was to change the four-vane impeller out to a five-vane design. The impeller change solved the problem.

Case 26. Alignment Change on Boiler Feed Pump

After being in operation only a few months, the boiler feed pump on a 500-megawatt turbine generator at a new generating station failed. The inboard seals were wiped out and the stage next to the coupling destroyed. Due to the type of failure, the plant initiated an alignment study.

To determine the amount of movement from the hot to cold condition, Dynalign bars were mounted between the pump and the turbine driver while the unit was in operation (see Figure 18). When the unit was brought off line, there were no significant alignment changes recorded. However, a few minutes later, the Dynalign system was found to be entirely out of range. When the operators were questioned as to what had happened during that period, they replied that the only thing they had done was to break vacuum on the main turbine. To determine if the vacuum had anything to do with the apparent alignment change, the bars were reset and long-range probes were installed to increase the measurement range of the Dodd bars. The vacuum was then reapplied to the system.

Everything appeared normal until the vacuum reached 11 inches of Hg. At that point, the gap voltages of the Dodd bar probes started to change. By the time full vacuum was achieved, the relative motion between the turbine and the pump was over 0.100 inch. The vacuum was then released and the readings moved 0.100 inch in the opposite direction. The test was repeated with identical results. An examination of the system was then performed. A 20-foot pipe descended downward from the feed pump turbine.

That pipe intersected with a horizontal pipe that was capped on one end and connected to the main turbine condenser on the other end. On the end of the pipe was an end cap. The horizontal pipe had three expansion joints. The purpose of the expansion joints was to isolate the boiler feed pump turbine from stresses induced by thermal growth of the horizontal condenser pipe. Thrust-canceling rods were installed between the end cap and the main condenser (see Figure 19). The thrust-canceling struts transmitted the atmospheric pressure load (14.7 psi) on the end cap to the condenser. The source of the problem was that threaded studs in the thrust-canceling struts were sliding into the struts. This was the result of failed welds on large nuts located on the backside of the strut end plates.

The net result of the failure was that atmospheric pressure being applied to the 6-foot-diameter end cap was pushing on the 20-foot vertical run of pipe attached to the bottom of the feed pump turbine. The large force applied to the 20-foot lever had the capability of generating 820,000 ft-lbs of torque to the turbine. Examining the concrete turbine base showed that the turbine foundation had several cracks from the bending torque. The thrust-canceling struts were repaired, and the alignment changes were rechecked as vacuum was reapplied. Following the repair, there were no significant changes in alignment as a result of variations in vacuum.

Case 27. Cavitation Destroys Impellers on Water Pumps

The impellers on large low-RPM, 156,000-gpm circulating water pumps on a cooling lake at a large power plant were failing. The failure mode appeared to be due to cavitation. The impellers looked like they had been attacked by metal eating termites. When a vibration spectrum was taken, the spectrum contained a large amount of broadband energy with no distinct peaks.

The key to the analysis, as is the case with a good percentage of pump problems, was to look at the flow head curve. The flow head curve indicated that at the design flow of 156,000 gallons per minute that the back pressure would be 30 ft. When the back pressure was measured, it was only 10 ft. What had happened was that during cold weather when the lake water was cool, operating personnel placed only one pump in operation to reduce power consumption.

The system was designed to operate against the back pressure produced by two pumps in parallel. When only one pump was on, the system back pressure dropped, and the one pump that was on line went into cavitation.

Case 28. Cracked Shafts in Vertical Pumps

Large vertical pumps were cracking shafts every few weeks. Due to the severity of the problem, underwater proximity probes were installed on one of the pumps. In addition, casing probes and torsional instrumentation was also installed. When the pump was put into operation, high levels of sub-synchronous vibration were observed. Natural frequency and mode shape measurements determined that the sub-synchronous vibration was centered around the shaft's first lateral natural frequency.

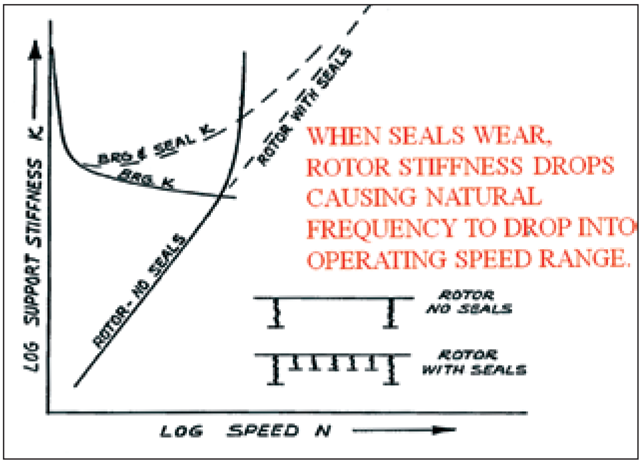

Figure 20. Lomakin effect.

The cause of the problem was traced to a maintenance superintendent purchasing impellers from a non OEM source. The design of the impellers varied significantly from the original design. This caused high levels of turbulence that excited the shaft's natural frequency. Since non-synchronous vibration causes stress reversals every revolution, this caused the shafts to fatigue.

Case 29. Lomakin Effect Alters Natural Frequency

Two boiler feed pumps would operate successfully for several months, then the running speed levels would start trending upward. The increase was due to higher levels of running speed vibration. When tests were finally run on the pumps, it was discovered that they were operating near a critical speed when fully loaded. This was determined when changes in speed resulted in large amplitude changes and shifts in the phase angles.

A newly overhauled pump did not show these same traits. It was determined that when the internal seals were wearing, this reduced the Lomakin stiffening effect allowing the shaft natural frequency to drop into the operating range. This example illustrates why Lomakin stiffening should not be depended upon to keep pump rotors from operating near a critical speed (see Figure 20).

Case 30. Containment Spray Pump Mass Addition

The containment spray pumps at a two-unit nuclear power plant had a history of high vibration at the vane pass frequency of the pumps. The pumps were vertical units that turned at 1785 RPM. The pump impellers had four vanes. This resulted in a vane pass frequency of 119 Hz. The highest levels of vane pass vibration (0.6 in/sec) occurred at the bottom of the motor in line with the discharge line. Since the levels were much higher in line with the discharge line as compared to 90 degrees out, a resonance was suspected.

An impact test was performed and there was a resonance identified at 111 Hz. Based on the natural frequency, the forcing frequency and the calculated damping, an amplification factor of 5.4 was computed by using the following equation.

$$\frac{X}{X_0} = \frac{1}{\sqrt{\left(\left(1-(\omega/\omega_n)^2\right)^2 + \left[2\xi(\omega/\omega_n)\right]^2\right)}}$$

where: forcing frequency = 119 Hz; natural frequency = 111 Hz; and damping = 0.05 (calculated by phase slope and half-power method).

In an attempt to understand why the level was highest at the bottom of the motor, impacts were made at several points up and down the motor and the pump. The imaginary parts of the transfer functions were then used to plot the mode shape of the 111 Hz resonance.

The mode shape agreed with the data in that the maximum response of the mode was at the base of the motor. Based on this information, it was elected to add mass at the base of the motor to lower the natural frequency and reduce the amplification factor. Calculations showed that adding 950 pounds of mass at the base of

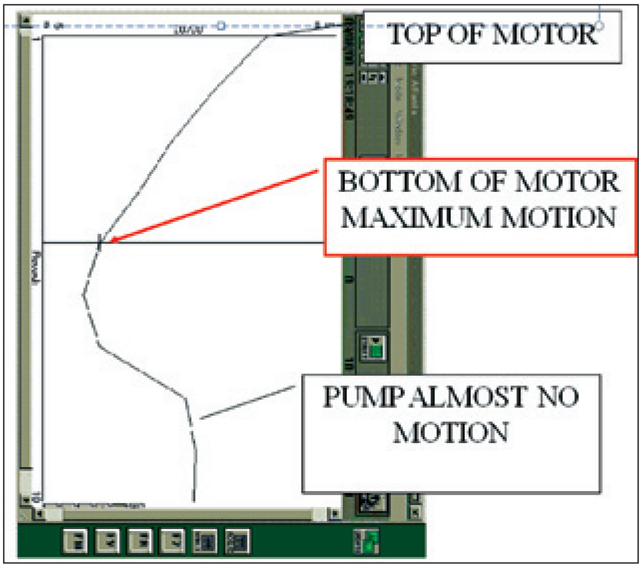

Figure 21. Mode shape.

the motor would drop the natural frequency to 101 Hz. According to the calculations, assuming the same damping, the amplification factor would be reduced to 2.5.

This amount of weight was added by installing channel iron around the base of the motor and then clamping lead weights between the two pieces of channel. Following the mass addition, the pump was tested. The vane pass levels dropped from 0.6 in/sec to 0.26 in/second. This same modification was made to all four pumps at this facility. The final solution was to change from a four-vane to a five-vane impeller (see Figure 21).

Case 31. Coupling Lock-Up of Nuclear Feed Pump

A feed pump turbine experienced high levels of twice-running-speed vibration. The figure-eight-shaped orbits indicated that the problem was misalignment. According to plant personnel, the unit had been aligned per the manufacturer's specifications. When the unit was brought down for an outage, the coupling was examined.

Its teeth were severely worn, the lubricant had failed, and it had evidently locked up. Based on this evidence, a study of the operating alignment was made. It was determined that the original specifications were wrong. Originally the pump had been set high relative to the turbine. The final setting required that the pump be set 0.020 inch lower than the turbine. The change was due to two main factors. The first was vacuum draw-down of the turbine. The second was that the assumed amount of growth of the pump was incorrect (see Figure 22).

Case 32. Oil Whip in 500-Megawatt Turbine

A large steam turbine had very peculiar behavior characteristics. It would operate with no problems for months at a time. If it then had to come off line for a few hours, it could not be started back up, due to high vibration from oil whip in the first LP rotor bearing. However, if the unit was off line for a day or so, it would start back up with no problem. It would also start up OK if it was restarted immediately after being brought off line. Such a situation has all the signs of a thermally related alignment problem. Since normal alignment equipment cannot be used on an operating turbine, a special system was developed to measure the elevation changes of the bearings.

This system showed that when the vacuum was drawn on the unit, the low-pressure rotor bearings dropped significantly. When the vacuum draw-down effect was combined with differential thermal shrinkage as the unit cooled (the LP hood cooled faster than the HP section), it resulted in the first LP bearing being unloaded enough to cause oil whirl. As the unit came up to speed, the oil whirl locked onto the rotor's first natural frequency and developed into oil whip. The solution was to install a tilt-pad bearing in the first LP position. After the installing the tilt-pad bearing, the hot startup problem was eliminated (see Figure 23).

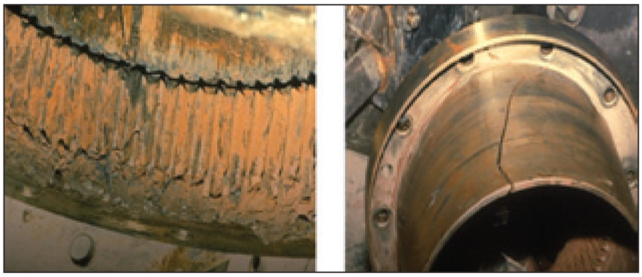

Figure 22. Couplings destroyed on steam generator feed pump at nuclear station.

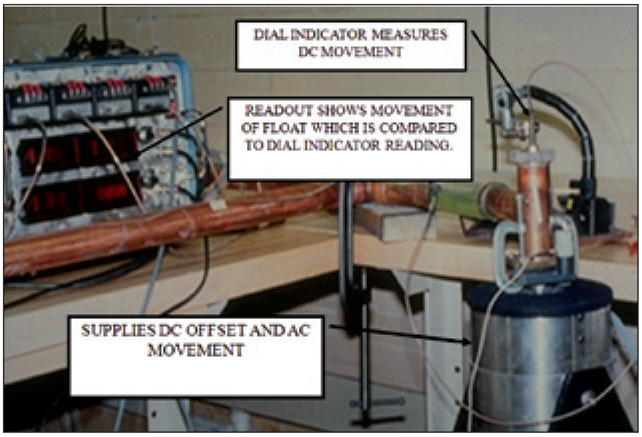

Figure 23. Alignment test system.

Case 33. Oil Whirl in Chiller Unit

The inboard bearing of a steam-turbine-driven chiller at a university campus heating facility experienced repeated failures. Examining the bearing showed that the top half had been fatigued to the point where babbitt was separating from the base metal. Vibration spectra contained high levels of sub-synchronous vibration. Since the bearing was located next to the coupling and oil whirl was present, misalignment was suspected.

A series of alignment measurements was made across the coupling, and it was found that there was significant movement during the first hour of operation. To determine which machine was moving, a laser was set up with a receiver mounted on an I-beam. It was determined that the chiller was moving. The root cause was determined to be the suction line on the chiller shrinking as the unit cooled. This shrinking caused the chiller to rock back. This in turn lifted the shaft, unloading the turbine bearing and causing it to go unstable (see Figures 24 and 25).

Case 34. Large Centrifugal Air Compressor Vibration

A compressed air facility had experienced severe difficulties bringing its main compressor unit into service. Its first-stage shaft had a natural frequency near operating speed of 16,200 RPM. The problem was so severe that the compressor had to be redesigned to accommodate a new shaft. After redesign, the compressor was brought into service. It was noticed that the level on the first stage was still higher than desired.

When a vibration signature was taken, it was discovered that there was another frequency present aside from the running speed. The frequency turned out to be four times the bull gear speed. Readings taken on the casing also showed the presence of the 4× bull gear frequency. A survey of the casing resulted in finding that an oil pump cantilevered to the case was vibrating at 3.0 in/sec at the 4× frequency. This turned out to be the lobe mesh frequency of the pump. When an impact test was performed, the cantilevered natural frequency of the pump matched the 4× frequency.

When a brace was installed on the pump, its level dropped significantly. In addition, the 4× bull gear vibration that was present in the signature of the first-stage proximity probes nearly disappeared. When the brace was removed, it reappeared. What had been occurring was that the oil pump had been shaking the compressor case. The first stage's newly modified first natural

Figure 24. Machine setup.

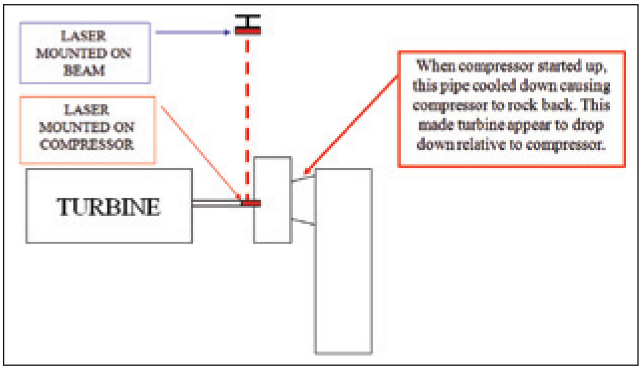

Figure 25. Laser set up for monitoring compressor alignment change during startup.

frequency was close enough to the 4× bull gear frequency that it amplified the response relative to the casing and was picked up on the proximity probes. The installation of a permanent brace on the oil pump eliminated the problem.

Case 35. Coupling Unbalance in High-Speed Compressors

A 6000-RPM turbine driving a chiller compressor was in alarm following an overhaul. The level was only high on the proximity probes on the inboard bearing next to the coupling. Since the vibration on the other end was low and the critical speed vibration was also low, it was decided to add weight to the coupling. It took only 3.8 grams/mil to balance out the problem. The level on the probes ended up below 0.5 mil.

A few months later, the unit was down for another inspection and the coupling was disassembled. It was discovered that on the previous overhaul the coupling had not been assembled with the match marks lined up. The match marks were put in line and the unit was restarted. The vibration was high like it had been after the previous overhaul. It was suspected that the reason that the balance weights had been installed in the first place was the match mark problem, so the balance weights were removed.

Following the removal of all the balance weights, the unit was restarted and it was again under 0.5 mil. This is a case that illustrates the importance of taking care to match mark couplings on high-speed units. Couplings represent overhung weight. This combined with the fact that the proximity probes are often located near the coupling can result in high-speed machines being very sensitive to any changes in coupling unbalance.

In another case involving a 10,000 RPM ammonia compressor, the owner had spent more than $100,000 trying to reduce the vibration to below the alarm setting. The unit had been disassembled three times and finally was sent back to the factory for a stack balance. When it came back, the problem recurred. The data showed that when the rotor went through its critical at 6000 RPM, there was very low response. However, from about 8000 to 10,000 RPM, the vibration on the coupling end probes increased to well above the alarm level. There was no phase shift during the 8000 to 1000

Figure 26. Morton effect caused shutdown of 23,000-HP compressor.

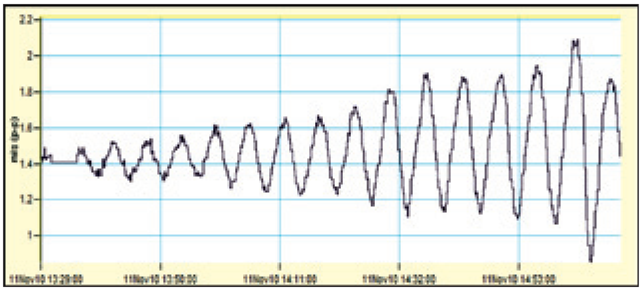

Figure 27. Overall amplitude showing Morton effect induced cycles that last 6-7 minutes.

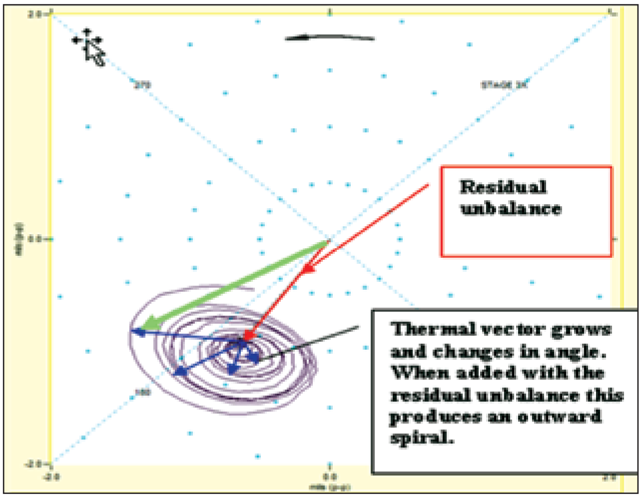

Figure 28. Polar plot that shows varying amplitude and phase spiraling out during Morton effect.

RPM range. It took one 11-gram washer on one of the coupling bolts to solve the problem. Several years later the customer called with a similar problem on the same unit. The compressor was balanced by adding weight to the coupling using the same sensitivity and lag angle that had been calculated for the previous coupling shot.

Case 36. Morton Effect in 23,000-HP Compressor

The vibration level on stage 3-4 X & Y probes of a 23,000 HP compressor would begin to oscillate. The time period of the oscillations varied from 6 to 7 minutes. In some cases, the level would reach a peak, then disappear after several cycles. In other instances, the amplitude would continue to build until the unit shut down on a high-vibration trip.

A National Instruments 20-channel analyzer was connected to the Bently Nevada outputs on the entire machine. The compressor had fortunately been set up with X-Y probes on all eight stages. Each stage also had a key phaser output, so it was possible to monitor the phase along with the amplitude when the transients occurred. The pattern was very repeatable. As the vibration started to change, the phase also changed along with the amplitude. The pattern looked exactly like what occurs with a rub, but in this instance, the problem had been present for several months. The air seals were made of soft aluminum, so any rub that would have occurred would have had to clear quickly. The vibration, when put into polar format, appeared like that of a snail shell with the amplitude and phase slowly changing and the amplitude getting larger on each cycle.

This problem was traced to what is called the Morton effect. When this occurs, a complex series of events takes place. It starts with the residual unbalance causing a high spot on the shaft as it goes by the tilt-pad bearing. The viscosity of the oil shearing causes a localized heating effect on the high spot of the shaft, which creates a small bow. Since these shafts operate above their first critical, the shaft throws out behind the resulting unbalance creating a new high spot and thus a new hot spot. This process continues, resulting in a slowly changing phase angle and ever-increasing amplitude. As a temporary fix, a new shaft was installed with a lower level of unbalance to reduce the possibility of the process being triggered. The long-term solution was to design a new bearing to reduce the localized heating of the shaft. Increasing of the oil temperature to reduce the viscosity also had a positive effect (see Figures 26-28)

Case 37. Surge in 5000-HP Centrifugal Compressor

Following an overhaul, a 5000-HP centrifugal air compressor at an air liquefaction facility experienced a vibration trip. Since the unit had recently been overhauled, the question arose as to whether the unit needed to be taken apart for inspection or could it be restarted. Following the overhaul, plant management had elected to use a system to monitor the vibration for a few days. This system was set up to continuously monitor all the proximity probes. Data storage was set up to store data every 60 seconds, on any change greater than 0.2 mil or if the speed changed. The data were buffered, so it was possible in the case of the amplitude based trigger to also store the data block that preceded the triggering event. That feature proved to be very helpful in diagnosing the cause of the vibration trip. Following the compressor vibration trip, the stored data were analyzed. The main questions were:

- Was the vibration real or instrument related?
- Did the machine experience any damage?
- Could the unit be restarted?

The data showed that a few seconds prior to the vibration trip, all the levels were normal. At the point of the trip, a frequency of 10,200 CPM with a 3.2-mil amplitude suddenly appeared in the spectrum. The rotor speed was 12,600 CPM, and the 1× component at 12,600 CPM remained constant through the transient. During the coast-down, it was noted that the peak in response was at the same 10,200 CPM that had been observed when the trip occurred. The conclusion was that the rotor's 10,200 CPM natural frequency had been excited and that is what caused the trip to occur.

Since a surge condition can excite a rotor's natural frequency, the plant was instructed to check the operating conditions to determine if the compressor was operating near the surge point. It was confirmed that the compressor had been operating very close to a surge condition. The time wave signals during the trip and coast-down were checked for any signs of clipping, which would have indicated a rub. In addition, the DC gap voltages were checked to determine if the shaft center line had changed; this would have been a sign of bearing damage. Since the wave shape was symmetrical, the DC gaps had not changed and the coast-down data were normal, it was concluded that no internal damage had occurred. The decision was therefore made to restart the compressor. When restarted, the levels returned to their previous values.

This case history illustrates how valuable it is to have a machine that has gone through an overhaul properly monitored for a few days until the process can be returned to normal. Having the vibration data from just before the trip, at the trip point and during the coast-down was very useful in determining whether the machine needed to be reopened for an inspection or simply restarted (see Figures 29 and 30).

Case 38. Vibration at Steel Strip Mill

An induction furnace in a steel mill was used to heat and diffuse galvanize steel strips. During the induction process, a loud

Figure 29. 5000-HP centrifugal compressor.

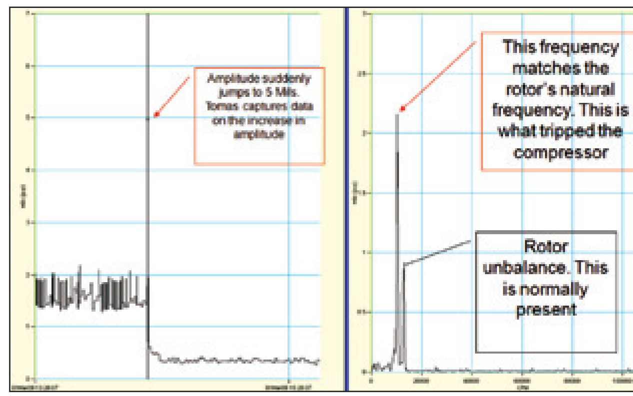

Figure 30. Trend showing sudden increase when compressor surged.

high-pitched frequency would radiate from the steel plate. When the sound would begin, vertical stripes would also appear on the plating. The stripes were causing the steel to be rejected by the customer of the steel mill.

An FFT analyzer was set up to determine the frequency of the sound along with the vibration on the induction furnace and the current being supplied to the induction coils. The frequency being detected in all three cases was at 7250 cycles/second. This frequency corresponded to the operating frequency of the induction furnace. To determine if a change in frequency would have an effect on the problem, the furnace frequency was increased to 9000 Hz. The stripes did not disappear at the higher frequency, but merely moved closer together.

It was determined that the induction furnace was exciting the natural frequencies of the plate, creating standing waves; this resulted in the stripes being formed as the galvanizing material flowed to the nodes. Since the thin plate had several natural frequencies within the normal operating range of the furnace, changing from one frequency to another did not help. Increasing the frequency made the stripes closer together, decreasing the frequency resulted in the stripes being further apart.

To solve the problem, the induction furnace's control circuit was designed to continuously vary its frequency several times a second. This rapid change in frequencies did not allow the plate to lock onto a particular frequency. When this modification was made, the striping problem was eliminated (see Figure 31).

Case 39. Vibration of Microscope in Microsurgery Room

Surgeons used a special microscope mounted to the ceiling of the operating room during microsurgery operations that involved replanting severed fingers and toes. The chief surgeon complained that the image was jittery and that it was very tiring to operate under those conditions, particularly when the scope was set for its maximum magnification.

The scope was set to its greatest magnification, and printed material was placed on the operating table. Vibration was clearly noticeable, just as the surgeon had indicated. Vibration spectra

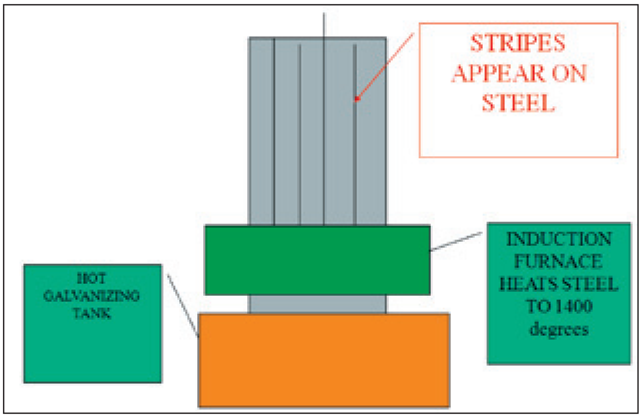

Figure 31. Trouble in steel mill.

Figure 32. FM transmitter.

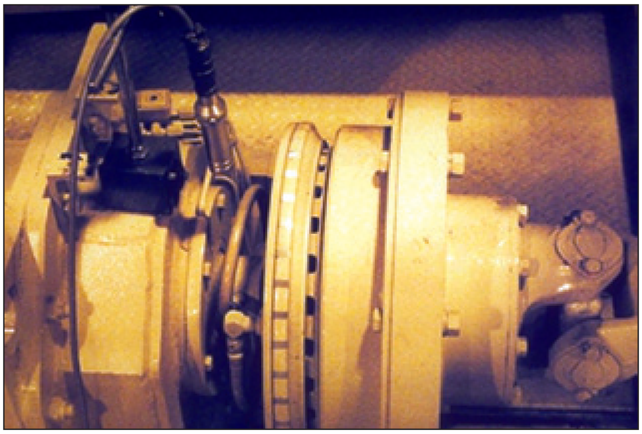

Figure 33. Multiple marks for torsional test.

were taken on both table and microscope. The levels on the table were very low across the spectrum. However, the levels on the microscope were significant. A view of the vibration spectra revealed that peaks were present at 225 CPM and 435 CPM. To trace the source of the vibration, levels were measured on the top of the microscope's vibration isolator and on the structural steel supporting the isolator . It was discovered that the levels on the isolator were seven times higher than on the steel support.

This meant that instead of isolating the microscope from structural vibration, the isolators were actually amplifying the vibration present on the I-beam. To determine the cause of the amplification, an impact test was performed on the microscope to determine its natural frequencies. It was found that the natural frequencies of the scope on its isolation system matched the vibration that was present on the I-beam.

Isolators perform their isolation function by creating a system with a natural frequency tuned much lower than the expected disturbing frequency. This in turn creates a mechanical low-pass filter, which will not pass the higher frequencies. However, a problem can occur if a low frequency is present near the low-tuned natural frequency of the isolated system. Instead of isolating the frequency, the isolators will then actually amplify the levels.

The solution in this case was to ground out the isolators. When the isolators were grounded out, the levels on the scope dropped to acceptable levels. The frequencies that had been present were due to isolators on fans above the room being tuned to the same frequencies as the microscope. Flow excitation in the fans excited the fans' isolated natural frequencies that were transmitted through the structural steel and amplified by the microscope isolators.

Grounding isolators should only be tried if nothing else works. When the isolators are grounded, higher frequencies, if any are present, will obviously pass through. In this particular case, grounding out the isolators didn't introduce any significant higher frequency vibrations.

Case 40. Torsional Vibration on Reciprocating Pump

Excessive torsional vibration at the pump speed of a 66 RPM reciprocating water pump driven by a gear box and belt reduction was being picked up at the gear case. Torsional testing generally is performed using either one of two methods. The first is the use of a strain gage to measure alternating torsional strain. The second method involves measuring the change in the passing frequency of equally spaced gear teeth or equally spaced reference marks. The change in passing frequency of equally spaced marks on a shaft is an indication of corresponding changes of angular velocity. This data can therefore be integrated to produce angular displacement.

For this test, both the strain gage and equally spaced reference mark techniques were used so that a comparison between the two methods could be made. A strain gage was mounted at a 45-degree angle on the drive shaft between the gear case and the belt driving the reciprocating pump. An FM transmitter and a battery were also mounted on the drive shaft to transmit the strain information.

This setup was calibrated by putting one end of the drive shaft in a vice and applying 100 ft-lb of torque to the other end. While the 100 ft-lb of torque was applied, the output of the demodulator was measured with a volt meter. The calibration constant from this test was then input into an FFT analyzer, resulting in readings directly in ft-lbs at each frequency present. The second test involved putting equally spaced photo reflective tapes at 20 locations around the output hub of the gear case. A photocell device was then mounted to pick up the pulse train of the reflective tapes. The output from the photocell was then input to a torsional-demodulator-integrator that produced an output of 200 mv/degree peak to peak.

With this combination, it was therefore possible to measure the torque being fed back to the gear case from the pump and the amount of angular displacement it produced. When the pump was operating against no appreciable back pressure, there was 1.5° of angular displacement present at the gear case hub at 66 cycles/minute. The torque from the strain gage at this condition was 27 ft-lb. As the back pressure on the pump was increased, the above values also increased. When the output pressure from the pump was 180 PSI, the torsional vibration had increased to 8.79° and the alternating torque to 123 ft-lb, both at the pump speed of 66 cycles/minute.

These data showed that the alternating torque peaks from the pump were too high. To correct this problem, a flywheel was added to level the torque peaks by absorbing energy during one half the cycle and return it to the system on the other half cycle. The alternating torque values were reduced by a factor of three by adding the flywheel.

This case shows how two different techniques led to the same conclusion. The test method used will depend on what the investigator needs to know, the test equipment available and the accessibility of the machinery to be tested. Another interesting note is that when coherence was measured between the two signals with a dual-channel analyzer, the level was 0.98. This indicates a direct correlation between the alternating torque on the drive shaft and the displacement on the gear case hub which would be expected (see Figures 32 and 33).

Case 41. Ghosts

In a small midwestern town, the residents complained that they felt movement of their houses, particularly at night. One of the residents said that her sister would no longer come and visit her because she thought the house was haunted with ghosts. She felt this way because lamp shades would move, pictures would rattle and rocking chairs would rock without anyone being in them.

In an attempt to identify the cause of this problem, vibration measurements were made at several of the houses in the community, along the sidewalks, and at a factory located near the houses. The testing was performed with an SD-380 analyzer using a 1000 mv/g low-frequency seismic accelerometer to convert the mechanical motion into an electronic signal.

The vibration signature taken at one of the houses where complaints had been registered showed a level of 1.22 mils at 300 CPM. It was observed that the amplitude of the vibration would oscillate, indicating that more than one frequency might be present. (Beat frequencies are generally produced by two or more closely spaced frequencies adding and subtracting as they go in and out of phase.) To determine if there was more than one frequency present, the zoom feature of the SD-380 analyzer was used. A 16:1 zoom plot of the vibration at one of the residences clearly showed the presence of several frequencies close to 300 CPM.

The next step in the investigation was to make a survey of the vibration present at the nearby factory. The factory was a foundry that contained several vibratory conveyors that moved parts from one area to another. Zoom plots were taken at each conveyor to determine their frequencies as closely as possible. Exact matches were found between the frequencies present at the houses and several of the vibratory conveyors in the factory. Additional testing confirmed the correlation between the operation of the conveyors and the vibration at the houses.

Rather than having ghosts, the residents of the small town were experiencing low-frequency vibrations from the vibratory conveyors in the nearby factory. The 5 Hz (300-CPM) frequency is easily perceived by the human body, particularly at night when other motion and noise is at a minimum. It also can excite low stiffness structures (i.e., lamp shades and rocking chairs). The factory eventually installed balancing devices to reduce the amount of force entering the ground from the conveyors.

Case 42. Vibration of Nuclear Magnetic Resonance Machine

Following the movement of a nuclear magnetic resonance (NMR) instrument used to test chemical samples from a second floor location to a third floor room, the instrument performed poorly. A test technician determined that everything was operating properly within the unit. The technician thought that vibration might be the cause of the problem, so tests were performed.

A signature taken at the probe of the NMR unit in the new location, showed a level of 8,190 micro g at a frequency of 26 Hz. The floor beside the NMR unit had a level at the same frequency of 1,550 micro g. The readings meant that the vibration on the NMR unit was 5.2 times higher than the level measured on the floor at the 26-Hz frequency of the vibration. To determine the cause of the amplification, a resonance check was performed. The floor was impacted and the response was measured on the detector of the NMR unit. The transfer function clearly showed a peak at 26 Hz, indicating that the unit was resonant at the frequency present on the floor.

It was concluded that fans in an HVAC room near the new third floor location were providing the 26 Hz forcing function. The resonant condition was significantly amplifying the vibration that was present. It was recommended that the NMR unit be installed on isolators with a 95% efficiency in attenuating the 26 Hz vibration. The 95% reduction resulted in levels that were lower than those the unit had been exposed to in its original location, where it had operated satisfactorily. Following installation of the isolators, the NMR unit performed well.

Case 43. Vibration Induced by Sound

After installing a rotary-casting conveyor in a foundry, vibration of the walls and particularly on the windows in the control room of the foundry experienced high levels of vibration. The locations with the highest levels of vibration were the windows in the control room. A plot of the vibration measured on the foundry windows showed a level of 39.2 mils near the center of one of the windows. A frequency of 885 cycles per minute was predominant in the spectrum. The 885 CPM vibration on the windows was also found to be present on the walls of all the offices in the foundry. This frequency matched the vibration frequency of a large rotary-casting conveyor. But vibration measurements next to the conveyor were low. The conveyor was mounted on springs and also was fitted with dynamic absorbers, which were, considering the low levels observed on the floor next to the conveyor, working as designed.

The next test involved taking measurements with a microphone. The output from the microphone was analyzed on an FFT analyzer and it was found that the sound level at 885 CPM (14.75 Hz) was more than 100 dB. Since this was below the normal hearing range for humans, the sound level did not seem bad; however, it could be felt, and a sheet of paper held in front of the conveyor would move noticeably. The final test involved performing a resonance check on the window. A plot of the response of one of the control room windows showed that the natural frequency of the window was very close to the frequency of the pressure waves being emitted by the rotary casting conveyor.

The rotary conveyor undoubtedly caused the vibration problem. However, the transmission path was through the air rather than through the structure. The windows being resonant near the operating frequency of the conveyor were further amplifying the problem. It was recommended that the windows be fitted with cross braces to move their natural frequencies away from the operating frequency of the conveyor and that a sound-absorbing enclosure with limp mass curtains also be built around the conveyor.

Case 44. High Levels of Vibration on End Cap of Large Pipe

A large pipe (37-inch diameter) at a refinery had very high levels of vibration on an end cap that was located after an expansion joint. The levels were over 4.0 inches/second. There had been failure of several of the retaining rods that spanned the expansion joint.

A vibration spectrum was taken on the end cap. The vibration was found to occur at 4500 cycles/minute with a level of 4.47 inches/second. The area surrounding the pipe was checked for rotating equipment operating at that frequency. No machinery was found that operated anywhere close to that speed. A visual exam of the pipe showed that there was a large butterfly valve upstream of the end cap. The end cap was on a dead end section of a tee.

Conversations were held with plant personnel to determine when the vibration had started and what, if anything, had been done to the piping prior to the high vibration and retaining rod failures. The first response was that work had been performed on the expansion joint but that "nothing had been changed." Further discussions and examination of the piping drawings did show that one thing had indeed been changed. A baffle had been removed just upstream of the expansion joint. The baffle was a thick flat plate with two small holes in it.

Plant personnel didn't think that it served any purpose. An overall review of the system showed that it was very important. What was occurring was that the butterfly valve was causing flow disturbances. The valve was found to be operating only 30% open. This resulted in pressure pulsations in the pipe. The baffle was acting like a low-pass filter and allowing static pressure to equalize but not letting the dynamic pressure pulsations pass. The pressure pulsations were probably small, but there were two design features that caused the vibration levels to be high. The first was the amount of area on the end cap. A 37-inch-diameter end cap has 1075 square inches of surface area. Therefore even a small pressure pulsation can generate a large force when acting on such a large area. The second reason was the length of the retaining rods. The retaining rods were 20 feet long, so the stiffness value was low.

The vibration problem was the result of pressure pulsations within the pipe acting on a large area with low stiffness restraints. Removal of the baffle had been a key element in the problem. The plant was advised to install short bolts across the expansion joint until the unit was brought down for its next outage. The purpose of this was to stiffen up the system and provide a backup to the long bolts that had been failing. This action was possible because

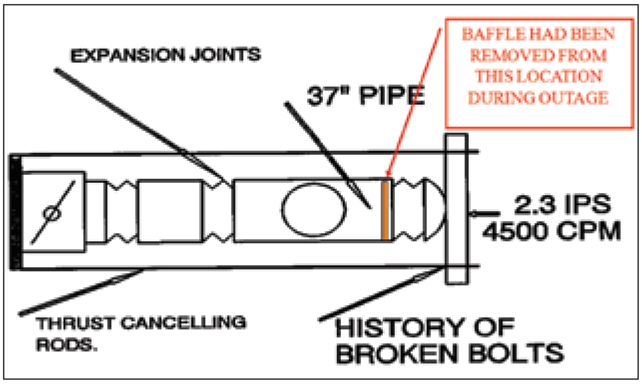

Figure 34. Missing component.

the piping was already at its maximum temperature, so the joint did not need to accommodate any further expansion. During the following outage, the baffle was reinstalled in the pipe, and the short bolts across the expansion joint were removed. The vibration problem was entirely eliminated (see Figure 34).

Case 45. Sound Annoys Dolphins

The zoo in Indianapolis, IN, had recently erected a new whale and dolphin building. It was a state of the art facility that allowed observation of the dolphins from a large area above the water and also from a restaurant below the facility. The dolphins arrived in town with much fanfare and their training began. A problem arose when the dolphins refused to follow instructions and seemed confused. The trainers noticed a sound in the auditorium area and requested that tests be performed to trace the source of the problem. A spectrum of the sound identified the noise as a pure tone at 1200 Hz. The level was 52 dB.

The sound was traced to a motor on one of the large vent fans built into each end of the building. The metal shell of the motor was acting like a speaker radiating the 1200 Hz frequency. The problem was simple to solve. The vent fans were on variable-speed drives. The speed at which this frequency occurred was programmed out of the drive. This stopped the problem and there were no further sound-related difficulties involved in training the dolphins.

Case 46. High Vibration on Plate Glass Window

An office building had an open atrium that went from ground level to the roof. In the center of the atrium was a large flowing fountain. Surrounding the fountain were meeting rooms. Each meeting room had a large plate glass window approximately eight feet tall and 12 feet wide, through which the individuals in the meeting room could view the fountain. The problem that was occurring was that these large plate glass windows had high levels of vibration. The frequency of the vibration was 240 cycles/minute.

The floors and walls were all tested, and no significant level of the 240 CPM vibration was observed in any location. The doors were also tested, and it was found that the doors that were not held tight against their latches also showed the presence of the 240 CPM vibration. It was concluded that the vibration was airborne and affected those items that had large surface areas and low stiffness values (plate glass windows and loose doors). The search began for the source of the 240 CPM stimulus.

The most obvious location to look for air borne transmission of pressure pulses was in the air handler area in the penthouse located at the top of the building. What made the search difficult was that the frequency was so low that it did not match any fan speed and certainly not any blade pass frequency. The breakthrough in solving the problem occurred while walking by an air handler and noticing that the pipe to the cooling coils in the discharge duct of the fan was vibrating. When a spectrum was taken on the pipe there was a match with the 240-CPM vibration on the window.

When that particular air handler was shut down, the vibration on the windows several stories below immediately stopped. When the fan was opened up, it was discovered that the screws in the braces that connected the heating coil to the duct work were missing. When the coil was impacted, its natural frequency was as an

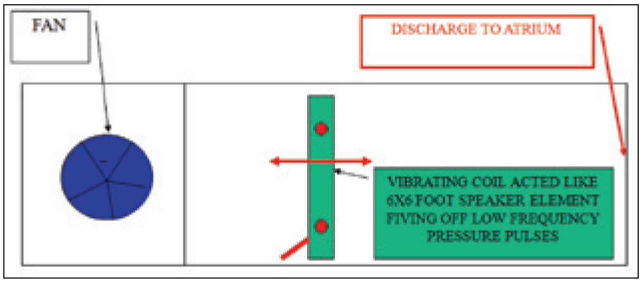

Figure 35. Loose coil.

expected 240 CPM. Without the supports, the coil was cantilevered off of the heating pipes resulting in a low natural frequency. When the fan was in operation, broad band flow energy excited this natural frequency causing the coil to vibrate in the duct at 240 cycles/minute.

Since the coil was approximately 6×6 feet, it generated a significant amount of air pulsation energy. The pulses were fed directly into the atrium area where they excited the windows. Since the windows had a very large cross sectional area and very little stiffness, they responded with high levels of vibration (see Figure 35).

Case 47. Drag Line Vibration

Only someone who has ever worked on a drag line can appreciate how difficult these machines are to work on. Large drag lines are used to remove the soil that is above a strip of coal. Sometimes the soil layer is 100 feet or more thick. The drag lines may weigh 5000 to 7000 tons, and their buckets can remove 100 or more cubic yards at a time. Typically there are several motor generator sets that generate DC current for the drag and hoist motors. This particular drag line had five of these very large motor generator sets.

Each motor generator set consisted of a large synchronous motor, two generators on one side of the motor and three generators on the other side. All the motors and generators were solidly coupled together and there was only one bearing per generator (the motor itself had two bearings). Since the motors are synchronous, that means that all the units operate at exactly the same speed, which in this case was 1200 RPM. To make things even worse, all the MG sets are located on the same metal deck. In this case it meant that there were 30 rotating elements (five synchronous motors and 25 generators) operating on one metal deck all rotating at the exact same 1200 RPM speed.

As would be expected, the vibration problem occurred at 1200 cycles per minute. This was a balance person's nightmare. The whole structure, even the boom, was shaking at 1200 CPM. However, with a little investigation, it was found that the vibration was higher on one of the MG sets. Efforts were then made to balance that unit. The highest reading was 11 mils in the vertical direction on the outboard generator of that unit. The first balance shot was installed in that generator.

The levels went down in the horizontal direction and up in the vertical direction. Therefore the influence coefficients indicated that it would not be possible to balance the unit in that rotor. In an attempt to simplify the problem, two of the generators were uncoupled from the motor. Even uncoupled, the generator vibration on the outboard generator was still 7.9 mils. Since the largest component in the MG train was the synchronous motor, it was elected to try a balance shot in that rotor.

When a trial shot was added to the motor, all the response vectors indicated that a solution could be obtained by adding the proper amount of weight at the proper angular location on the synchronous motor's rotor. Fifty ounces of weight had to be added at approximately a 30-inch radius on the motor. When this was done, the vibration level on the whole unit went back to normal. The lesson that was learned was that when in doubt, add the trial weight to the rotor with the largest polar moment of inertia.

Case 48. Sound and Vibration from Pulse Furnaces

A new building had been constructed at a university and it had objectionable levels of both sound and vibration. Both problems

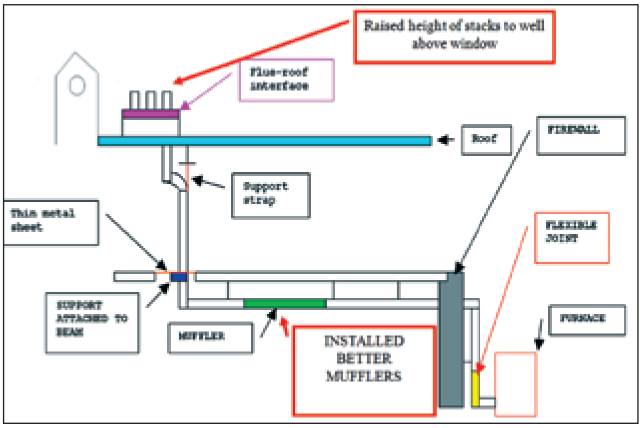

Figure 36. Pulse furnace building layout.

were related to the three pulse furnaces heating the building. The furnaces were located in the basement. Furnace flues ran under the floors then up a chase and out through short stacks in the roof. Vibration and sound spectra were taken in the offices areas where there were problems. The primary frequency in both the sound and vibration spectra were at approximately 37 Hz, which was the pulse firing frequency of the furnaces.

At the end of the building next to the offices with the highest sound and vibration levels, there was a hollow square tower that only contained stairs. The tower went up past the rooftop much like a church steeple. In the area of tower that stood above the roof, there was a 3-foot-diameter circular window. This window was directly across from the furnace exhaust stacks. It was discovered that this window was vibrating with extremely high levels of vibration at the furnace pulsing frequency.

When a resonance test was performed, it was discovered that the window's natural frequency was very close to the pulse firing frequency of the furnaces. This meant that near the top of the tower there was effectively a 3-foot speaker pumping energy into the tower. To make matters even worse. The height of the open tower was approximately 30 feet, which matches the wavelength of the 37-Hz frequency.

Two actions were taken to resolve the problem. The first was to put mufflers on the flue lines. The second was to raise the height of the flues to well above that of the window in the tower, so it would not be excited (see Figure 36). S&V

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