11 Problems – 11 Solutions

Case Histories of 11 Machinery Vibration Problems – Part 1

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This two-part article covers a series of eleven machinery vibration problems encountered over a three year period. While each case history is not necessarily outstanding in its own right, they do show the type of equipment problems encountered in today's industrial environment. Many problems were manifested by the lack of forethought on the part of the management team and in other cases management forethought eliminated additional problems. This article will cover cases in-depth. most required rotor dynamic modeling, structural modeling or both. Each case has a lesson to be learned. Part 1 covers case histories #1-6. Part 2 will cover the remaining 5 case histories and will be published in the May '07 issue of S&V.

Case #1 Unrepaired Turbine Alignment Issues

Problem – The concern was the difference in vibration amplitudes monitored in the control room from the low pressure bearing shaft rider of the turbine versus the proximity probe reading collected with the plant data collector. The turbine is instrumented with General Electric (OEM) shaft riders on both the high pressure bearing (inlet) and low pressure bearing (outlet). Plant personnel instrumented the low pressure turbine bearing with externally mounted proximity probes to look at shaft vibration.

Five years previously, a vibration analysis was performed on this equipment. The problem at that time was an alignment issue between the turbine and pump. During the analysis, it was discovered that straight bore bearings were installed in the turbine. This turbine manufacturer calls for a minimum of an elliptical bearing in both the low pressure and high pressure end.

Equipment Utilized for the Analysis – CSI 2115 data collector, four-channel TEAC analog tape recorder, proximity probes, accelerometers, multi-channel integrating signal conditioner.

Symptoms – There were no real symptoms other than the difference between the vibration from the proximity probes and the OEM shaft rider vibration amplitudes.

Test Data and Observations – The turbine data from the proximity probes and shaft riders were analyzed and concentrated on the difference between the control room data and the proximity probe data. The shaft rider data (Figure 1) contains 1× vibration along with a low frequency component.

Data from the horizontal proximity probe (Figure 2) contain no low frequency vibration; however, multiples of 1× are present. The data are essentially the same, the 1× amplitude from the shaft rider (Figure 1) is 2.6 mils$_{pk-pk}$ and the proximity probe data (Figure 2) is 2.1 mils$_{pk-pk}$. Data collected from a shaft rider is absolute data, meaning that it contains the shaft vibration plus the casing vibration.

The high pressure bearing (inlet) velocity data is very interesting. The data indicate an alignment issue (Figure 3); but, data from the inboard pump and inboard turbine casing do not show this issue. Ironically; these data are almost an exact copy of the data collected for the previous analysis of this turbine/pump combination (Figure 4).

It was documented that this turbine had the bearings replaced in May, 1997 and the pump overhauled in April, 2000. There is no record of the alignment being checked after the bearing replacement; however, the plant thought the alignment was checked after the pump overhaul.

The condition of the turbine was not an immediate concern, the 2× vibration had been present for a long time. The data are essentially the same as in a previous analysis and the unit has run reliably. The shaft vibration from the inboard turbine proximity probe indicates the shaft vibration is 2.45 mils$_{pk-pk}$ (Figure 5).

For equipment running above 3600 rpm, the shaft vibration should not exceed 30% of the bearing clearance. This turbine has a 6 in. journal; therefore, the bearing clearances should not exceed 9.0 mils. The acceptable shaft vibration would then be 2.7 mils$_{pk-pk}$. Presently, the vibration is 27% (2.45 mils/9.0 mils) of the bearing clearance. While this is not a serious problem, the vibration should be watched closely to see that it does not suddenly start to increase. There is indication of alignment issues on the turbine; however, these are not seen on the pump. This would mean the bearings on the turbine are not aligned or the turbine

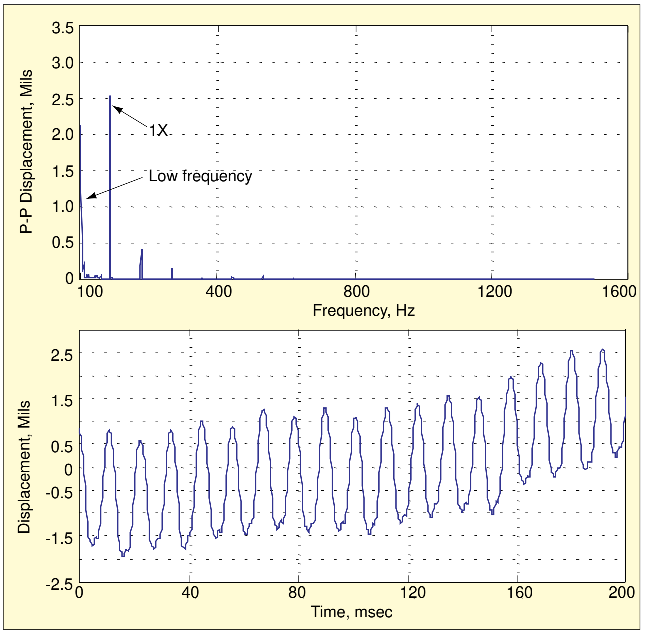

Figure 1. Shaft rider data.

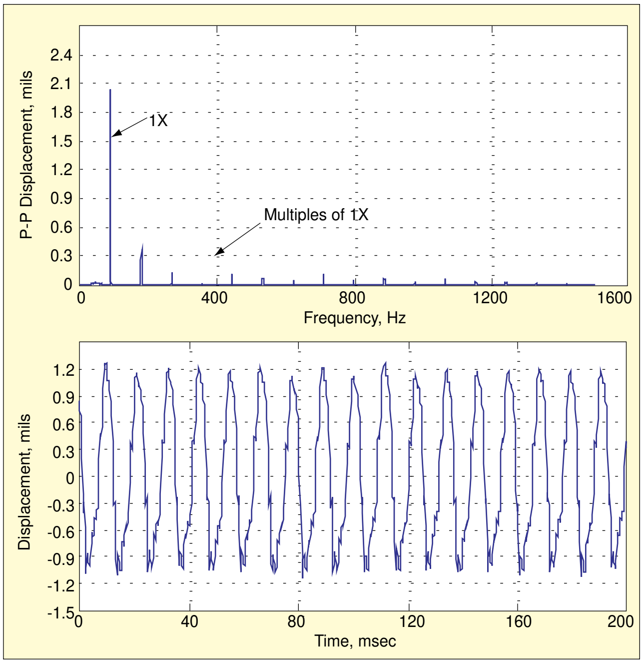

Figure 2. Low pressure bearing horizontal proximity probe data.

Based on a paper presented at the 60th Meeting of the MFPT Society, Virginia Beach, VA, April 2006.

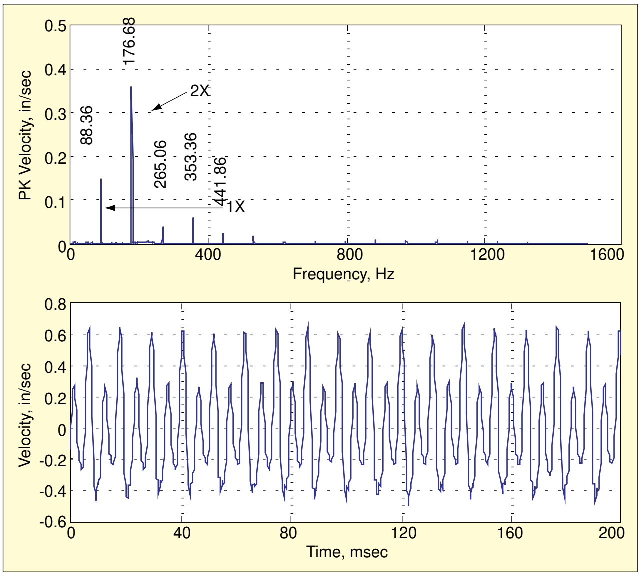

Figure 3. High pressure bearing shaft rider data.

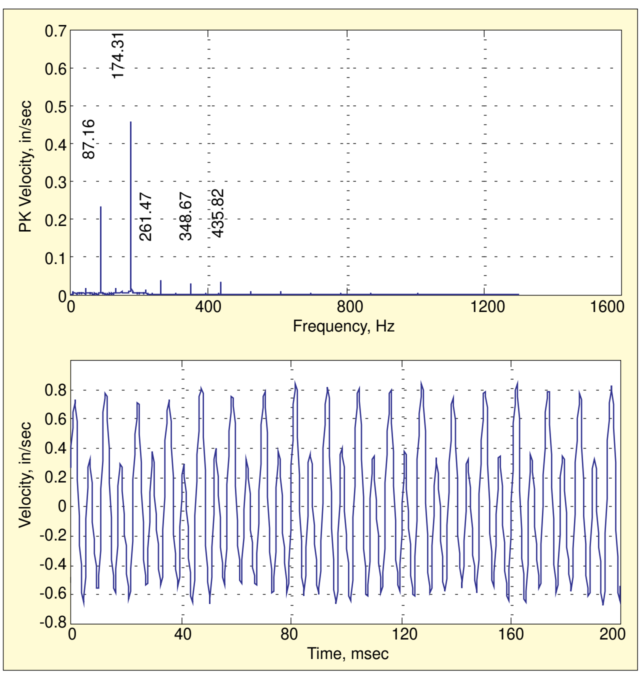

Figure 4. High pressure bearing seismic accelerometer data.

rotor is misaligned in the bearings.

The pump does have a strong blade passage frequency and multiples of this frequency component are present. While it is normal to see some blade pass components on a feed pump, the multiples are a concern. The peak amplitudes of the blade pass vibration are as high as 0.5 in/sec$_{0\text{-pk}}$ (Figure 6). The amplitudes of the blade pass vibration should be watched and the trends followed to see if they are remaining constant or if they are increasing.

Corrective Action – No corrective action was recommended. The unit, in spite of having alignment issues on the turbine, has run reliably. The plant, during the next overhaul, will check the bearing-to-bearing alignment on the turbine.

Results – The turbine has run reliably since the data were collected.

Conclusions – While there are alignment problems, the amplitudes are acceptable. The main concern was the difference between the amplitudes of the proximity probes and the shaft riders. There are several reasons for this difference. The shaft riders are an absolute probe collecting both shaft and casing data. The proximity probes are only looking at shaft data. Also, the frequency range being utilized for each reading is different. The OEM looks at a frequency range of 0-9000 cpm (0-150 Hz), while the plant was collecting data from the proximity probes for a frequency range of 0-2000 Hz.

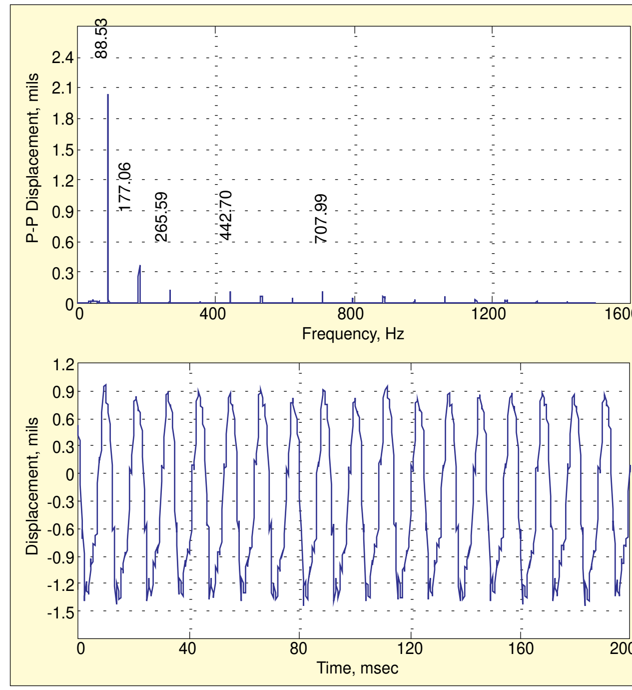

Figure 5. Low pressure bearing proximity probe data.

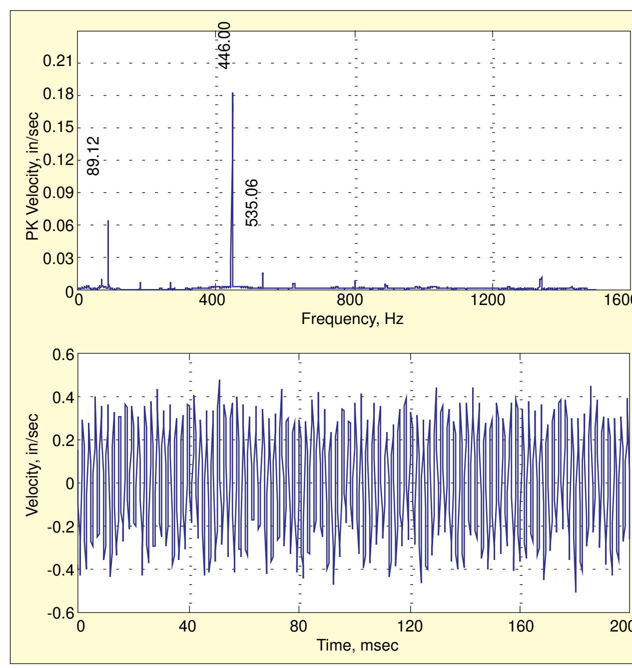

Figure 6. Inboard pump bearing seismic sensor data.

Case #2 Sub-harmonic Resonance

Problem (Continuation of Case History #1 – A boiler feed pump was put back in service after a complete overhaul of its turbine drive and high vibration was experienced on the turbine low pressure bearing (inboard bearing – exhaust end) within days of going back on line. The vibration was erratic and showed up instantaneously when the unit reached its maximum operating speed, 4800 rpm. The main turbine was de-rated to ensure the unit would not trip due to the loss of the feed pump. Turbine speed was to be held at 4725 rpm. Even with this de-rating, the unit could still experience vibration limit trips.

This is a General Electric turbine. The turbine can either be run on main steam or extraction steam with the normal turbine operating speed between 4300 to 5000 rpm.

Equipment Used for the Analysis – Bently Nevada System One

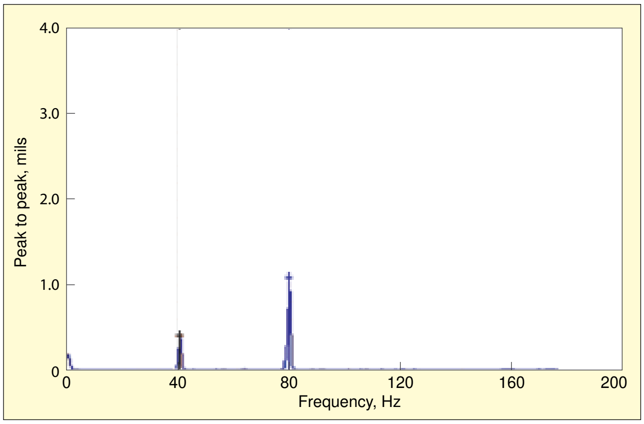

Figure 7. Low pressure bearing below vibration point – 4687 rpm.

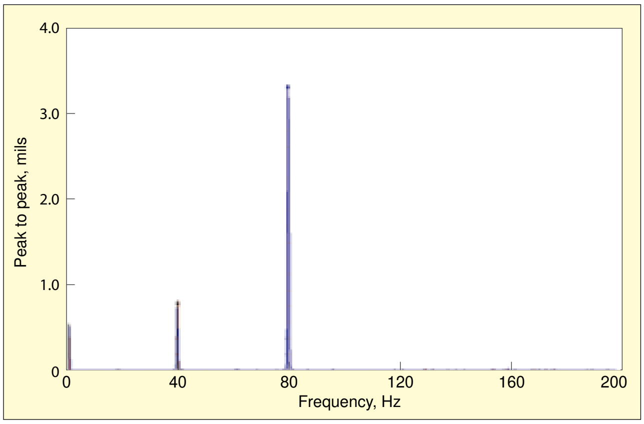

Figure 8. Low pressure bearing below vibration point – 4800 rpm.

software, proximity probes, IOtech ZonicBook 618E™ with eZ-Analyst™ software

Symptoms – This model turbine (DRV-631) has a power industry history of vibration problems related to subsynchronous vibration. These vibration problems are sudden high vibration amplitudes (equipment shutdowns for facilities utilizing vibration trips) in the 4700-4900 rpm range. Normally, the high vibration amplitudes are on the low pressure (exhaust end) bearing. Vibration alarms on the high pressure (inlet end) bearing do not occur unless the low pressure bearing vibration amplitudes are over 5.0-6.0 mils.

Test Data and Observations – Since the turbine just had a complete supervisory instrumentation retrofit, all data were collected from the newly installed proximity probes. Data were collected with the turbine operating just below the point where vibration would start to increase (Figure 7). Data indicated a subsynchronous vibration at 2400 rpm.

When the turbine speed reached 4800 rpm, the 2400 rpm vibration started to gain energy and become unstable (Figure 8). The average vibration at this point was still acceptable; however, it was beating up and down. This 1/2× vibration (2400 rpm) appears whenever the turbine speed is within ±10% of 4800 rpm. The amplitude and severity of the vibration depends on how much energy builds up in the system. The longer the turbine speed stays at 4800 rpm, the more energy builds up in the system.

This is a sub-harmonic resonance. This turbine design has a structural natural frequency at 4800 rpm. When the turbine operates at 4800 rpm, it excites the first critical speed of the turbine which is at 2400 rpm, one half (1/2×) running speed.

This turbine problem has been documented on many other turbines of this design. In fact a sister unit at this same plant had the same problem approximately fifteen years ago. It should be noted that in addition to the overhaul and new instrumentation installed during the outage, this turbine and pump had a major alignment problem.

Corrective Action – The data collected confirms the subharmonic resonance and the need for changing out the "elliptical" bearing to a "pressure dam" bearing. This type bearing has an oil dam opposite the load zone of the bearing. This dam increases the oil pressure in the load zone, stiffening the shaft and increasing the critical speed above 2400 rpm. The increased critical speed should be somewhere around 2650 rpm.

The design of the pressure dam bearing can vary from a round bearing with a dam at the top to an elliptical bearing with a dam at the top. The best solution to this problem is to use an elliptical bearing with 1.0 mil per in. of clearance per journal diameter for the vertical clearance and 1.5 mils per in. for the side to side clearance.

Pressure dam bearings are unidirectional, meaning they will only work in one direction. If the bearing is installed incorrectly, the dam will unload the bearing causing a severe unstable vibration.

Results – The pressure dam bearing was installed within four days of this analysis. The turbine has run for one year without return of the vibration problem.

Conclusions – This turbine was exhibiting what is called "sub-harmonic resonance." The cause of the problem is the interaction between the shaft and pedestal as the unit runs in the 4800 rpm range. Operating at 4800 rpm with the shaft and pedestal interaction excites the first critical at 2400 rpm; exactly one half running speed. This unit ran for years without this problem; however, it did have an alignment issue described in the previous case history.

At the end of the outage, an alignment check was done because of recommendations made on previous analysis procedures. When the alignment was corrected, which was a major change, the re-alignment lowered the load on the bearing. This unloading reduced the stiffness of the bearing, lowering the shaft critical speed back to around 2400 rpm.

It was discovered that many years ago, following previous vibration issues, this equipment was intentionally misaligned to increase the load on the bearing. This increased the stiffness and moved the critical speed higher.

Case #3 Perceived Balance Problem

Problem – A gas turbine driving a compressor had been in operation for three years without any problems. During a two month period, bearing vibration increased from less than 0.50 $\text{mils}_{\text{pk-pk}}$ to 1.3 $\text{mils}_{\text{pk-pk}}$ on the coupling end of the turbine. The gas turbine is direct coupled to the compressor. Operating speed is normally around 15,000 rpm (250 Hz). However, the speed could vary as low as 14,000 rpm (233.3 Hz) depending on system requirements. This compressor supplies a high volume of low pressure (40 PSI) air for fermentation in pharmaceutical manufacturing. The unit needed to run another nine months to a shut down before any bearing replacement could be performed.

Equipment Used for the Analysis – Bently Nevada 3500 monitoring panel, proximity probes, IOtech ZonicBook 618E with eZ-TOMAS™ Software

Symptoms – The only indication of a problem was the increase in shaft vibration from the proximity probe on the inboard turbine bearing. The vibration increase on the inboard turbine bearing was associated with an increase in temperature on both turbine bearings. The turbine bearings were operating at 195° F. The increase in temperature was in the 5 to 10° range. All four bearings on the turbine and compressor are monitored by a single proximity probe in the vertical direction and a keyphaser mounted on the compressor shaft.

Test Data and Observations – Due to production requirements, no changes in the operating speed of the turbo compressor could be made during data collection. Data were collected from each proximity probe and the keyphaser. This would allow for collection of time and spectrum plots along with phase angle data. The dominant frequency in all the data was operating speed (Figure 9) and the phase remained constant during the data collection (Figure 10).

The other concern was the temperature increase. During the testing, which took over five hours, at no time did the temperatures change. While 195° F is fairly high for certain types of babbitt bearings, this shaft is operating at 15,000 rpm. The drain oil temperatures did not change during data collection. One would expect the temperatures to be higher due to the higher speeds.

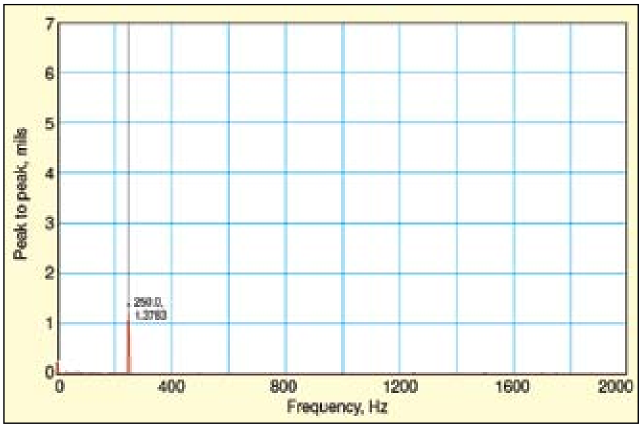

Figure 9 – Turbine inboard bearing proximity probe data.

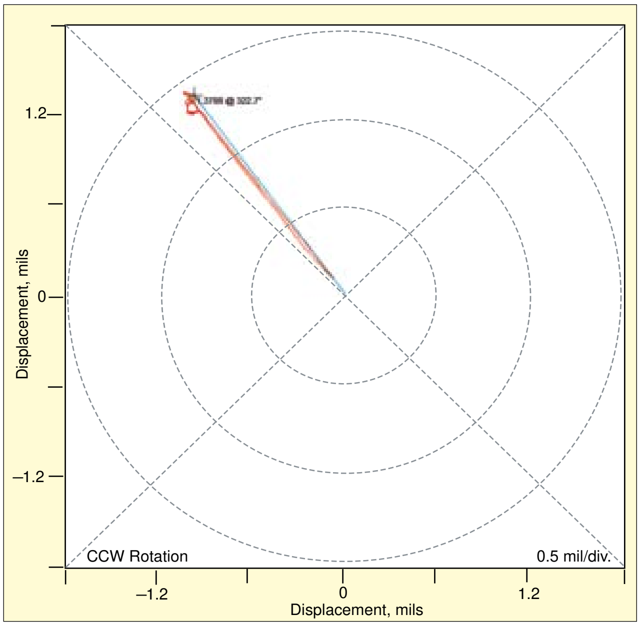

Figure 10. Turbine inboard bearing proximity probe polar plot.

Since there was no information on bearing clearance, it was estimated. The shaft diameters were 5.00 in. for the turbine and 6.50 in. for the compressor. There are two main rules for calculating bearing clearances. The first rule is there should be 1.5 mils (0.0015 in.) for each in. of journal diameter. The second rule states that you use 1.0 mil (0.001 in.) for each in. of journal diameter plus one mil. This means that for the first rule the clearance in the turbine should be 7.5 mils (0.0075 in.) and for the compressor the clearance should be 9.75 mils (0.00975 in.). The result for the second rule is 6.00 mils (0.006 in.) for the turbine and 7.50 mils (0.0075 in.) for the compressor.

It is understood that normal allowable shaft vibration is based on the ratio between shaft vibration and bearing clearance:

$$\text{Vibration Ratio} = \frac{\text{Shaft Vibration}}{\text{Bearing Clearance}} \tag{1}$$

The shaft speed is 15,000 rpm; therefore, if the ratio of shaft vibration to bearing clearance is less than 20%, the vibration is acceptable. The overall highest vibration was 1.60 $\text{mils}_{\text{pk-pk}}$ from the inboard turbine bearing (Figure 11, Table 1).

This plant used the first rule for calculating bearing clearance; 7.5

Table 1. Allowable shaft vibration/clearance ratio (Vibration Institute).

Maintenance	**Allowable R/C 3,600 rpm**	**10,000 rpm**
Normal	0.3	0.2
Surveillance	0.3-0.50	0.2-0.4
Shut down next convenient time	0.5	0.4
Shut down immediately	0.7	0.6

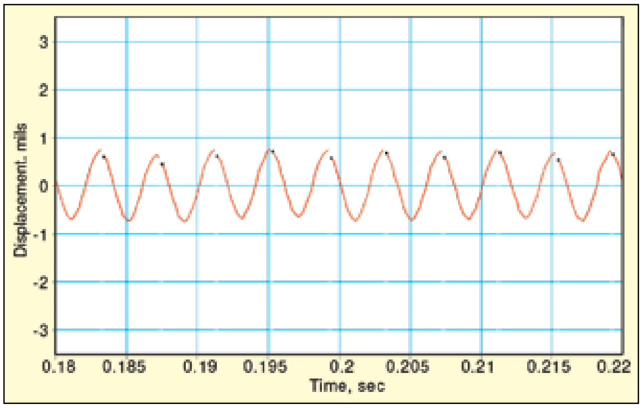

Figure 11. Turbine inboard bearing proximity probe data.

mils (0.0075 in.) for the turbine bearing. This means the vibration ratio is 0.21, right on the border between a normal condition and a surveillance condition. Surveillance means that if the vibration ratio is in this region the plant should cut the time between monitoring checks in half. Since this equipment has a permanent monitoring system installed, it has automatic tripping protection.

Corrective Actions – Since the vibration is stable and the bearing temperatures, while fairly warm, are also stable, it was recommended that the equipment be operated normally.

Results – This unit ran reliably for ten more months until the plant shut down. During the shutdown the turbine was serviced as scheduled. The bearings were inspected and found to be in good condition. There was no sign of damage due to the increased heat on the bearings.

It was later found that some of the piping hangers needed to be reset. This may have added some preload to the bearings; increasing bearing load and increasing the temperatures.

Conclusions – This plant has a fully functioning proactive maintenance program. The close watch on equipment condition pointed to a change in the equipment operating conditions even though no alarms were triggered. If this plant did not have a high quality monitoring program, this potential problem would not have been identified.

Case #4 – Vertical Pump Drive Problem

This was a new installation at a municipal water treatment plant. The vertical pumps are a two stage design driven by a 200 HP variable frequency drive (VFD) motor. The pumps were exhibiting vibration problems in the 1500-1700 rpm operating range. Previous data collected by the OEM consulting engineer indicated a possible natural frequency in this area. Maximum speed on the VFD motors is 1790 rpm. Operating data provided by the OEM showed a fairly narrow band of vibration excitation.

Equipment Used for the Analysis – IOtech ZonicBook 618E with eZ-Analyst and eZ-TOMAS Software, 100 mV/g accelerometers mounted with magnets, calibrated 3 lb modal impact hammer, multi-channel amplifying and integrating signal conditioner.

Symptoms – The pump was showing high vibration amplitudes in the upper 20% of the operating speed range. Amplitudes were recorded at over 1.0 in./$\text{sec}_{\text{0-pk}}$ (Figure 12).

Test Data and Observations – Data were collected for two operating conditions: normal operation with the pump below and above the vibration problem region; and transient mode when the speed was dropped from rated speed to 1200 rpm. Impact tests were performed on one of the new pumps in the vertical and horizontal directions (Figure 13). Accelerometers (100 mV/g) were installed in eight locations on the pump and motor. An optical tachometer triggering off reflective tape was used for a phase and speed reference. A calibrated 3 lb modal hammer (0.71 mV/lb) was used for force input. Excitation and response locations for data collection are shown in Figure 14. Impact tests were performed at Points #1 and #2 (Figure 13). Sensors were mounted opposite the impact points. Operational deflection shape data were collected in the horizontal and vertical directions (Figure 14) from the top of the motor down to the sole plate.

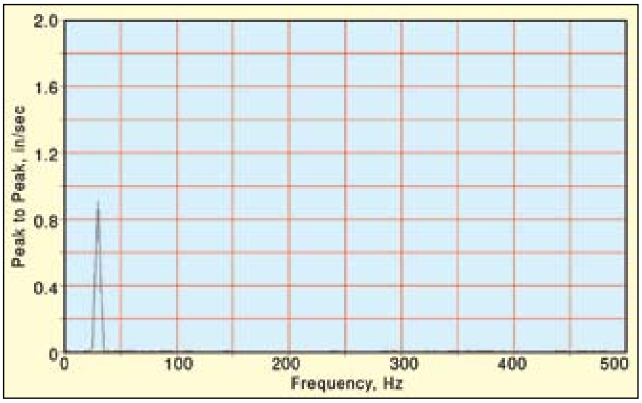

Figure 12. Outboard motor bearing – vertical.

Figure 13. Operating and impact test point locations.

Figure 14 – Operating deflection data points.

Transient data were analyzed first. There are two distinct natural frequencies present, 1597 rpm and 1690 rpm (Figure 15 and Figure 16). Impact tests confirmed the natural frequencies found on the coast down tests (Figures 17 and 18).

The pump and motor were operating in a resonant condition when they are in the 1550-1750 rpm range. This is a common problem with vertical pumps. It is compounded by the fact that the pump is driven by a VFD. If this were a fixed speed pump operating at 1780 rpm, one would never know that a natural frequency problem existed at 1690 rpm.

No one part of the system can be faulted. The problem is not related to the pump, motor or foundation individually; rather it is all three parts acting together. If there is one area for concern, it is the pad (foundation) poured on the floor that supports the pump (Figure 13 and Figure 14). The rule of design, based on "good engineering practice," dictates the foundation, at a minimum, must be five times heavier than the equipment it supports. This foundation is actually smaller in mass than the pump it is supporting.

Corrective Actions – It was recommended to utilize Fabreeka vibration isolation pads to break up the system dynamics by putting a 1/2 in. thick ring of Fabreeka pads between the motor and pump head. Additionally, the pads will add damping to the system – isolating and absorbing some of the vibration. It was also recommended that if, after the addition of the pads, the unbalance was still high, to balance the motor to reduce the running speed

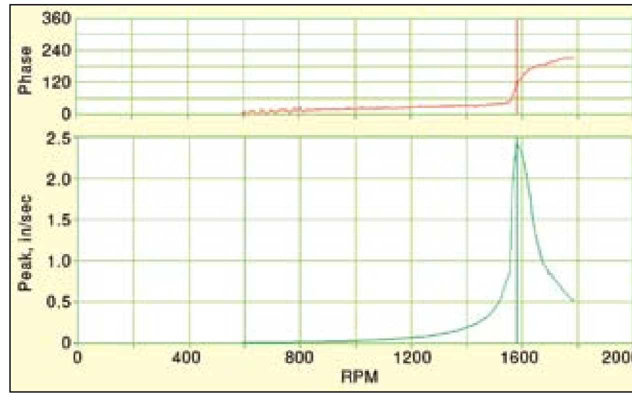

Figure 15 – Outboard motor – horizontal direction.

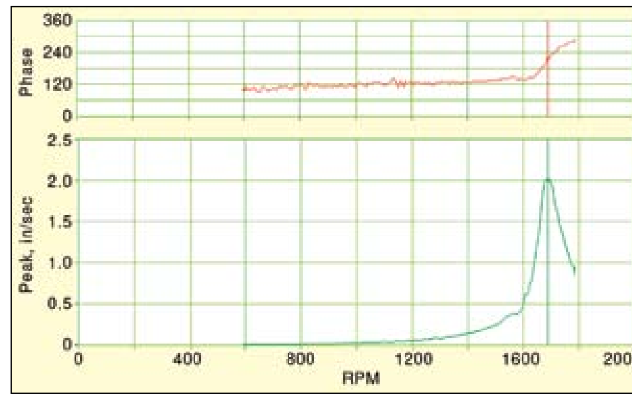

Figure 16 – Outboard motor – vertical direction.

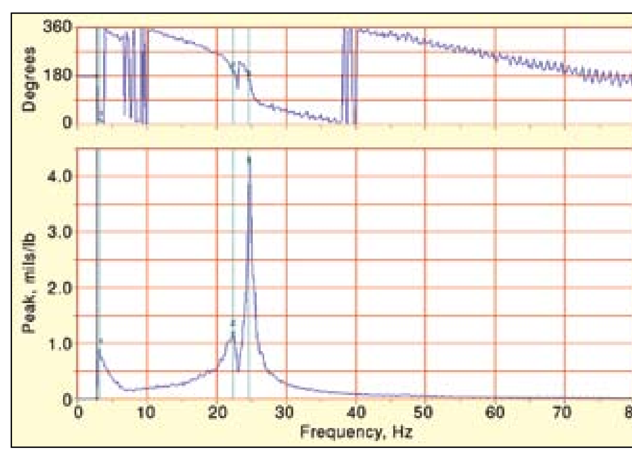

Figure 17 – Impact test – vertical direction.

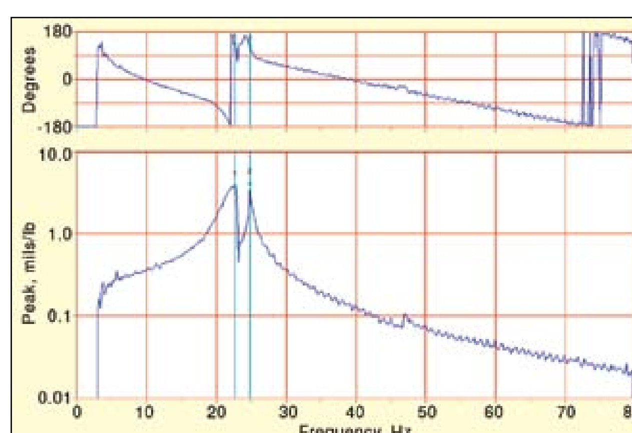

Figure 18 – Impact test – horizontal direction.

forcing function.

Results – Fabreeka vibration isolation pads reduced the vibration to the point that no additional balancing was required. The two pumps involved in this problem have both operated reliably since the repairs were made.

Conclusions – This pump suffered from a natural frequency problem that was a result of the design of the foundation. Total blame could not be placed on any one individual in this case. The original design of the building called for pumps that were smaller than those finally installed. When it was discovered that larger pumps would be needed, the foundations were already poured and ready for the original pumps. The municipality decided to use the foundations installed instead of tearing them out and putting in new foundations. It is likely that, as the pumps age, the resonance problem will be back as the piping and foundation systems weaken.

Case #5 – Pump with Excessive High Axial Vibration

Problem – A pump required overhaul due to rolling element failure of the pump bearings. The pump operates at 3575 rpm and is overhung. The plant vibration contractor identified the bearing problems. However, due to production requirements, it was decided to run the pump since there was no redundant backup. Vibration data prior to the bearing failures were dominant in the horizontal and vertical directions. There was some axial vibration at the beginning of the initial bearing failure data. The pump was repaired with a new shaft, bearings and a balanced impeller. Upon return to service the pump had high axial vibration.

Equipment Used for the Analysis – SKF single channel data collector with MachineView software, accelerometers mounted with magnets.

Symptoms – The pump had high axial vibration with velocity levels exceeding 3.0 in./sec$_{0\text{-pk}}$. The radial vibration amplitudes were below 0.50 in./sec$_{0\text{-pk}}$. Vibration levels could be changed by changing the flow settings of the system (Figure 19).

Test Data and Observations – The first overview of the pump and piping system indicated pump shaft thrusting and there was no recirculation line in the piping. The thrusting of the pump shaft is an indication of the pump running back on the flow curve. The symptom is called "low flow shuttling." This problem is compounded by the fact that the piping system does not have a recirculation line. When the pump flow requirements are cut back, the flow has nowhere to go, but stays in the pump causing the shaft to thrust back and forth due to hydraulic pressure building up on each side of the impeller. Data indicated vibration at 3× running speed. The impeller has three vanes; therefore, the 3× vibration is the vane passage frequency.

During the collection of data it was learned that the system requires pump flow of 2200 gpm. This pump was designed for 3200 gpm. Reviewing past vibration trend data; axial vibration was the dominant frequency at 3× running speed (vane pass). However, the vibration amplitudes were below 0.75 in./sec$_{0\text{-pk}}$. The data indicated that low flow shuttling had been an issue for a long time. In fact this pump had a long history of bearing failures and axial vibration.

The present concern was why is the vibration so high now? When vibration gets above 2.0 in./sec$_{0\text{-pk}}$, it becomes a safety issue and the recommendation was made to shut the pump down immediately. The plant decided to run the pump for production reasons.

Because the vibration was so excessive, it was decided to look into the impeller design. It was discovered that an oversized impeller was installed. The original impeller for this pump was 10.5 in. in diameter. Because the flow was too high, the impeller was cut down to 10.0 in. in diameter. The present impeller had a diameter of 12.0 in. This increased the specific speed of the pump and increased flow to over 4000 gpm when only 2200 was needed for production.

Corrective Actions – It was recommended that the pump be immediately shut down or a catastrophic failure of the shaft could occur. The plant ignored all warnings.

Results – The pump shaft failed within two weeks of this analysis. The plant reinstalled the same oversized impeller and had a second failure within a month.

Conclusions – The company originally installed an oversized pump because they felt that at some time they might increase production capacity. Until the pump is sized correctly and a recirculation line is installed, this pump will continue to have failures and reliability problems.

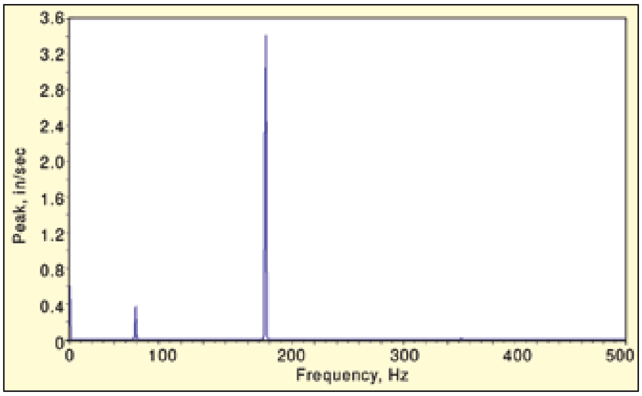

Figure 19. Inboard pump bearing axial vibration.

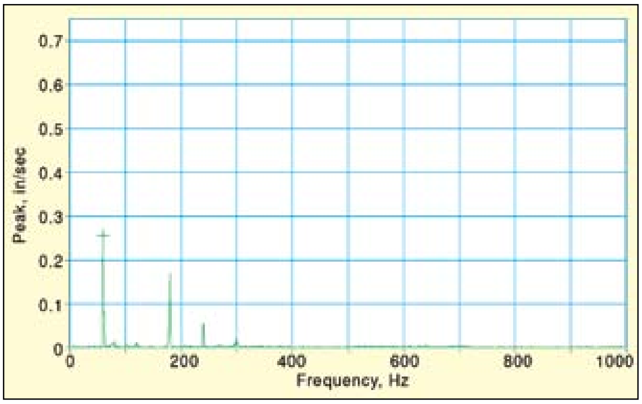

Figure 20. Outboard exhauster bearing vertical vibration.

Case #6 – Exhauster Vibration Vertical Direction

Problem – A vacuum exhauster was exhibiting vibration problems over the past few months. The present machine suffered a failure and was subsequently repaired. The repaired exhauster was placed back in service with vibration levels equal to levels experienced before the failure and repairs. The exhauster operates at 3575 rpm. The motor has sleeve bearings and the exhauster uses rolling element bearings.

Equipment Used for the Analysis – IOtech ZonicBook 618E with eZ-Analyst and eZ-TOMAS Software, 100 mV/g accelerometers mounted with magnets, calibrated 3.0 lb impact hammer, multi -channel amplifying and integrating signal conditioner.

Symptoms – The exhauster had acceptable vibration levels below 0.30 in./sec$_{0\text{-pk}}$ when started. However, as the exhauster runs the vibration levels increase to almost 1.0 in./sec$_{0\text{-pk}}$.

Test Data and Observations – The data collected were in velocity units for both the time history and spectrum. The data strongly point to mechanical looseness as the source of the vibration. The vibration is synchronous with multiples of 1× (Figure 20) present in the data from all exhauster points. In addition to the looseness there is a natural frequency at 3× running speed. The natural frequency could be excited by mechanical looseness.

One other item noted in the analysis is the presence of a natural frequency or critical speed around 2000 rpm on the coast down (Figure 21). The amplification factor on the coast down, depending on position, is around 6.0 which is acceptable for sleeve bearing machines. Sleeve bearings have damping which helps control the vibration. Rolling element bearings have no damping; therefore, no way to control vibration at critical speeds.

The repair company stated their test results indicated vibration levels below 0.10 in./sec rms. It was discovered that when the repair shop tested the exhauster at their shop, they ran the blower at 2200 rpm. The rule is to never operate a machine within ±20% of a natural frequency. Another issue learned during the testing was that the exhauster flow capacity was increased without any engineering study to access the effect on the rotor dynamics of the shaft and pedestal system.

Corrective Actions – The most likely source of the looseness is the fit between the blower wheel and shaft. The wheel is aluminum and with a steel shaft the expansion coefficients are different. If the shrink/interference fits are not correct the wheel will expand and quickly lose its fit with the shaft.

Results – The fits between the shaft (steel) and exhauster wheel (aluminum) were found to be correct. However; it was discovered that the grade of aluminum would creep (expand) under the added forces due to the increase in flow capacity.

Conclusions – If an engineering study were conducted before the upgrade in flow capacity it would have raised concerns about the ability of this aluminum wheel to take the added stress of the flow increase. The increased flow was causing more thermal growth than expected. This allowed the wheel to loosen on the shaft causing the operating problems. SV

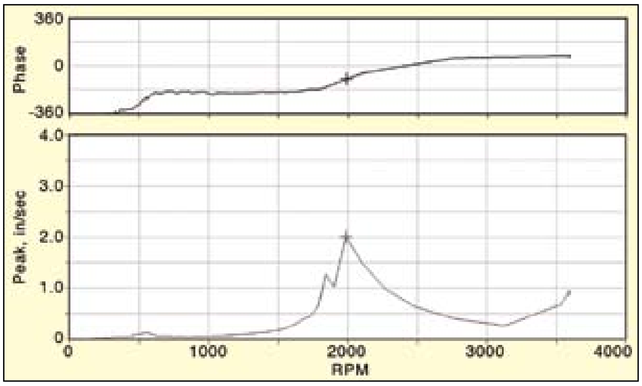

Figure 21. Coast down – outboard exhauster bearing – vertical.

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