

This document was prepared in conjunction with work accomplished under Contract No. DE-AC09-96SR18500 with the U.S. Department of Energy.

This work was prepared under an agreement with and funded by the U.S. Government. Neither the U. S. Government or its employees, nor any of its contractors, subcontractors or their employees, makes any express or implied: 1. warranty or assumes any legal liability for the accuracy, completeness, or for the use or results of such use of any information, product, or process disclosed; or 2. representation that such use or results of such use would not infringe privately owned rights; or 3. endorsement or recommendation of any specifically identified commercial product, process, or service. Any views and opinions of authors expressed in this work do not necessarily state or reflect those of the United States Government, or its contractors, or subcontractors.

Excitation of Structural Resonance Due to a Bearing Failure

Robert A. Leishear

Savannah River National Laboratory
Washington Savannah River Company
Aiken, South Carolina, 29803
803-725-2832, Robert.Leishear@SRS.gov

David B. Stefanko

Savannah River National Laboratory
Washington Savannah River Company
Aiken, South Carolina, 29803
803-725-7742, David.Stefanko@SRS.gov

Jerald D. Newton

Washington Savannah River Company
Aiken, South Carolina, 29803
803-557-8076, Jerald.Newton@SRS.gov

ABSTRACT

Vibration due to a bearing failure in a pump created significant vibrations in a fifteen foot by fifteen foot by eight feet tall mounting platform due to excitation of resonant frequencies. In this particular application, an 18,000 pound pump was mounted to a structural steel platform. When bearing damage commenced, the platform vibrated with sufficient magnitude that conversations could not be heard within forty feet of the pump. Vibration analysis determined that the frequency of the bearing was coincident to one of the natural frequencies of the pump, which was, in turn, coincident to one of the natural frequencies of the mounting platform. This coincidence of frequencies defines resonance. Resonance creates excessive vibrations when the natural frequency of a structure is coincident to an excitation frequency. In this well documented case, the excitation frequency was related to ball bearing failures.

The pump is a forty foot long vertical pump used to mix nuclear waste in 1,300,000 gallon tanks. A 300 horsepower drive motor is mounted to a structural steel platform on top of the tank. The pump hangs down into the tank from above to mix the waste and is inaccessible after installation. Initial awareness of the problem was due to increased noise from the pump. Initial vibration analysis indicated that the vibration levels of the bearing were within the expected range for this type of bearing, and the resonant condition was not obvious.

Further analysis consisted of disassembly of the motor to inspect the bearings and extensive vibration monitoring. Vibration data for the bearings was obtained from the manufacturer and compared to measured vibration plots for the pump and mounting platform.

Vibration data measured along the length of the pump was available from full scale testing, and vibrations were also measured at the installed pump.

One of the axial frequencies of the pump, the platform frequency in the vertical direction, and the ball spin frequency for the bearing were multiples of each other. This resonant condition was detected before other damage occurred, and further damage due to the resonant condition was prevented through vibration analysis.

KEYWORDS

Resonance, bearing failure, vibration analysis, centrifugal pump.

SYMBOLS

A	acceleration, G's, feet / second ²
D	displacement, inches
f	frequency, cycles / second
Hz	Hertz, cycle per second
G	gravitational constant = 32.174 feet / second ²
P	equivalent radial load, psi
psi	pounds per square inch
R	radial load, psi
SDOF	single degree of freedom
T	thrust load, psi
V	velocity, inches / second
τ	transmissibility
ω	forcing frequency, radians / second

ω_n natural frequency, radians / second
 ζ damping ratio

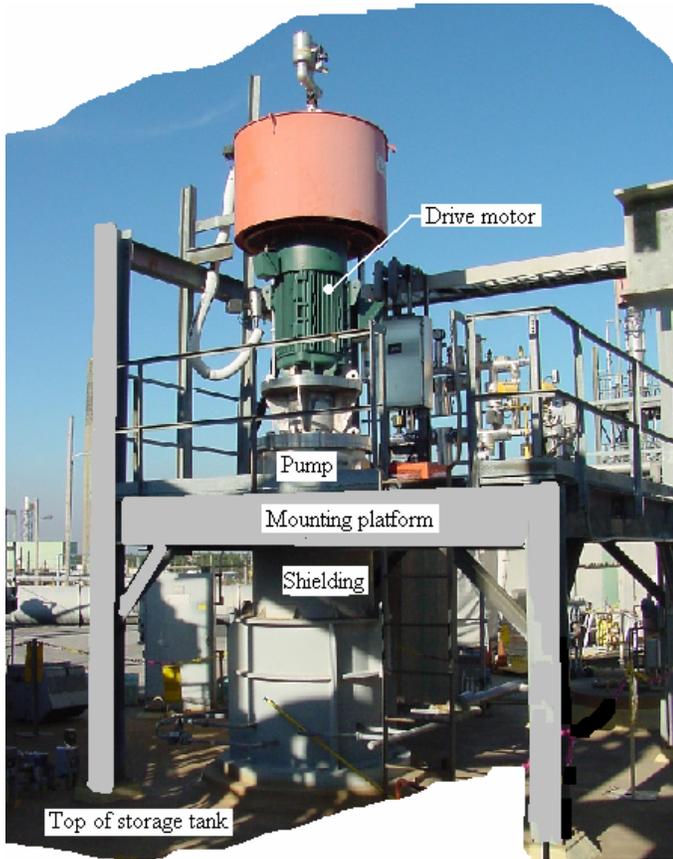


Figure 1: Installed Pump

INTRODUCTION

Vibration damage to rotating equipment due to resonance is a recognized phenomenon, but recognition of the resonant conditions leading to equipment failure is frequently difficult. In the case study considered here, data was collected and vibration analysis was performed to understand the vibration mechanics leading to a resonant condition, which had potential for significant equipment damage. Resolving the problem proceeded from problem recognition, to the vibration analysis, and then to disassembly and inspection of the pump motor to confirm bearing damage as the cause of vibration. To perform a vibration analysis, a description of the equipment, the measured vibration data, and a brief discussion to relate resonance to vibration is required.

EQUIPMENT DESCRIPTION

The equipment was one of several mixing pumps used on nuclear waste tanks at Savannah River Site, S. C. The top of the pump and its drive motor are located above a mounting platform, which is located on top of the tank as shown in Fig. 1. Most of the pump length hangs below the platform into the waste tank, and the pump assembly is shown in Fig. 2. The pump impeller is submerged in the waste tank and is driven by a 40 foot long shaft from the motor. The shaft is surrounded by a water filled column constructed of flanged piping.

Shown in Figs. 3 and 4, a cross section and plan view of the assembly provide further details, which are important to vibration analysis. In particular, the pump is connected to the platform by a turntable, which permits rotation of the pump. Shielding fabricated from pipe surrounds the pump above the tank top, and connects the platform to the tank top.

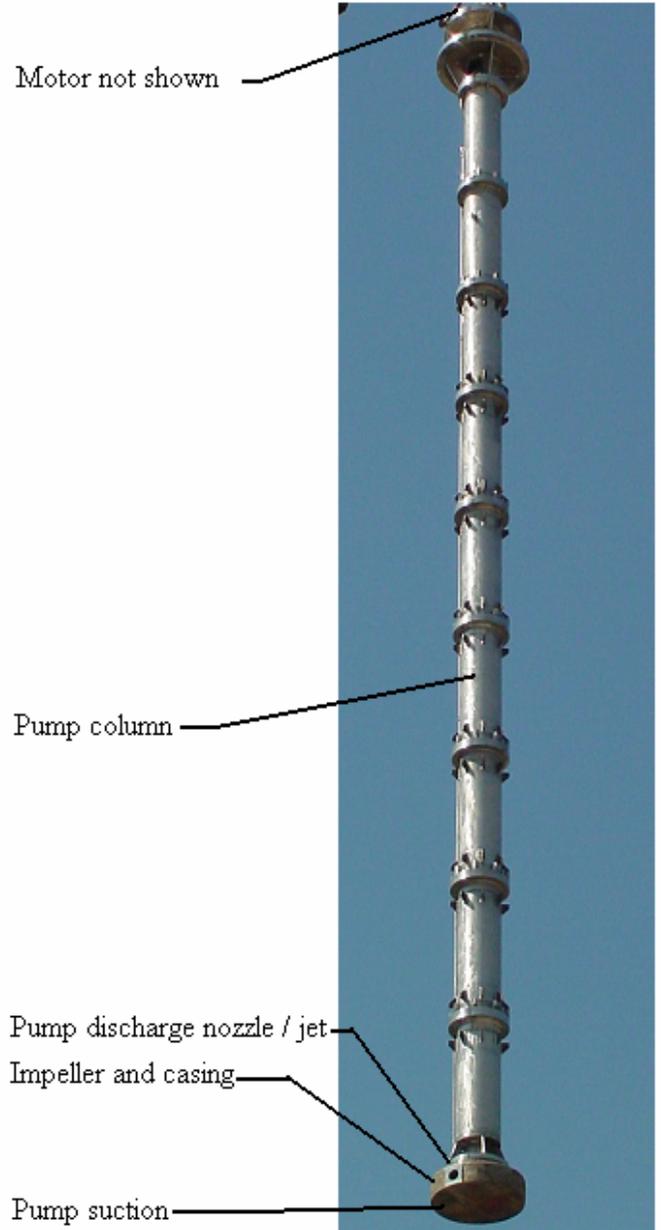


Figure 2: Pump Assembly

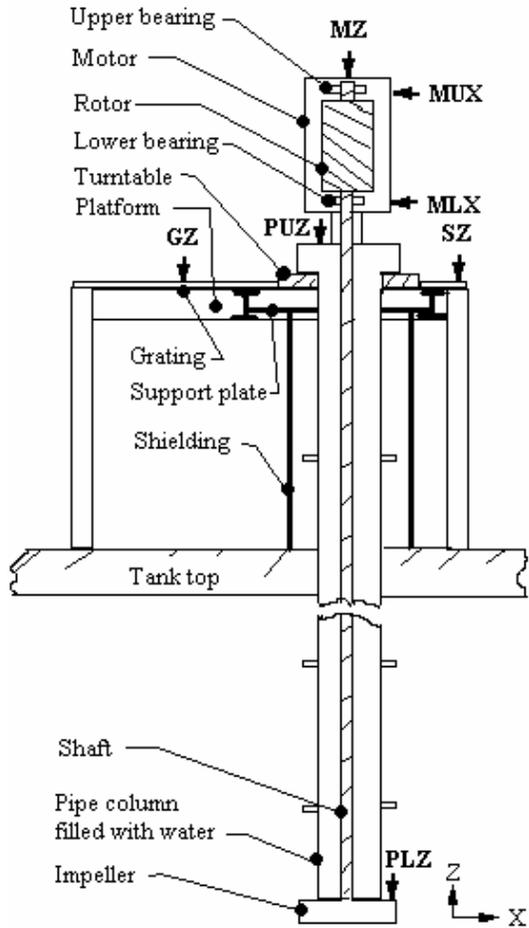


Figure 3: Elevation Cross Section of the Pump Assembly

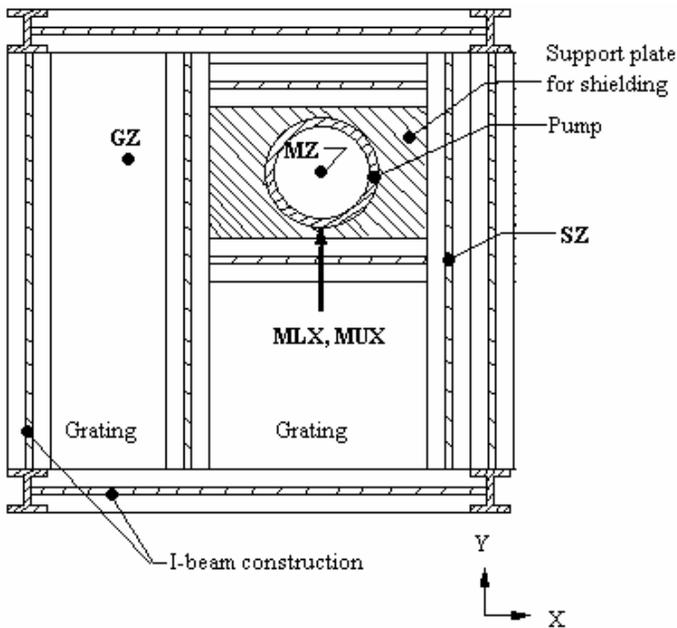


Figure 4: Plan View Cross Section of the Pump Assembly

PROBLEM DESCRIPTION

The initial problem was noted to be an increase in noise level by operations personnel. Normal conversations could only be heard at a distance of at least forty feet from the pump. The sound frequency was not measured, but was noticed to be considerably lower than the 60 Hertz frequency typically emanated from electrical transformers. Investigation noted that the entire platform on which the pump rested was vibrating. The vibration could be felt when standing on the platform. This vibration led to the increased noise level near the pump. Other than the increased noise level, apparent operating problems were unnoticed.

Motor Bearing Damage

The motor was disassembled and inspected for damage. To do so, the upper thrust bearing was cut from the shaft. The upper, double row, thrust bearing had a damaged cage as shown in Fig. 5, and the races and balls were damaged as shown in Figs. 6 and 7. Spalling, or pitting, of the ball surfaces and inner race was quite obvious. The race was loose with respect to the balls. Improper lubrication was suspected as the initial cause of bearing failure. The lower radial bearing shown in Fig. 8 was undamaged, and the race was still tightly fitted to the balls. To further evaluate the bearing damage, a short discussion of resonance provides a sense of direction for the remainder of this paper.



Figure 5: Damaged Outer Bearing Cage on the Upper Motor Thrust Bearing



Figure 6: Ball Damage on the Thrust Bearing



Figure 7: Races on the Thrust Bearing

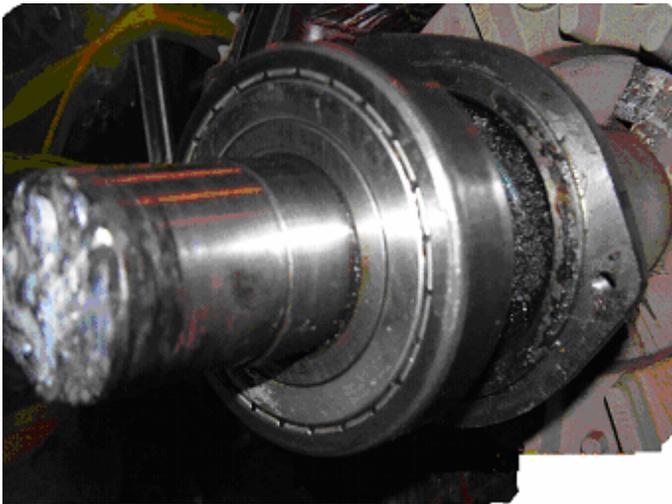


Figure 8: Lower Motor Bearing

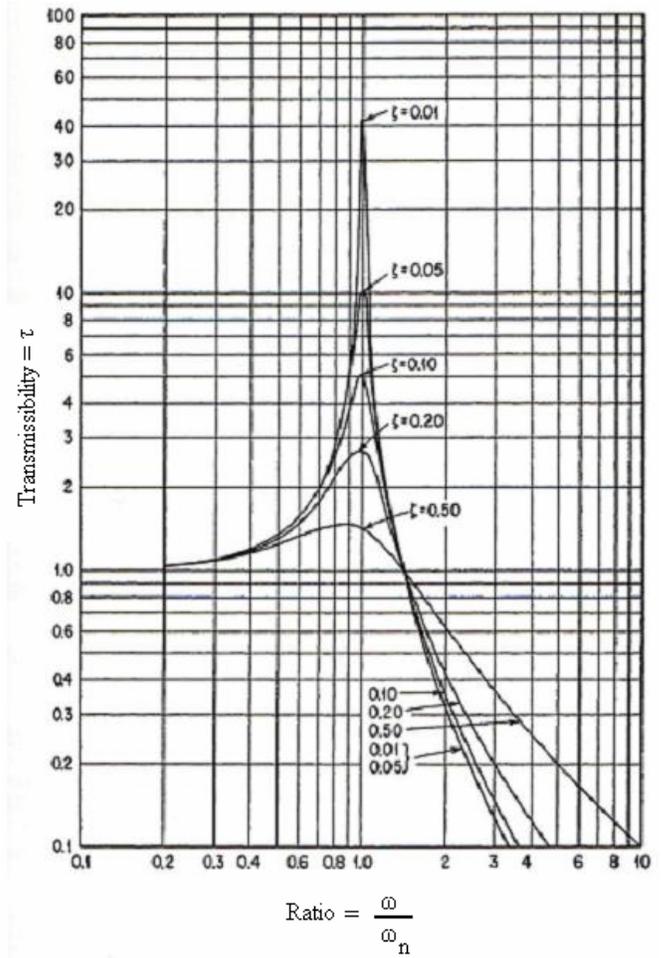


Figure 9: Vibration Response of a SDOF System (Harris [1])

RELATIONSHIP BETWEEN VIBRATIONS AND RESONANCE

Resonance occurs when a vibration frequency, or forcing frequency, ω , is coincident to the natural frequency, ω_n , of the excited structure (Thomson [2]). The response of a single degree of freedom (SDOF) system can be expressed mathematically, but a graphic presentation provides visual insight.

In Fig. 9, resonant responses are displayed in terms of dimensionless damping, frequency, and transmissibility terms for a single degree of freedom system. The damping ratio, ζ , quantifies the reduction of a vibration to an equilibrium state. The ratio term corresponds to the natural frequency divided by the excitation frequency (ω / ω_n). The transmissibility term, τ , describes the effective force transmitted to a structure during vibration. From the graph, note that if the frequency of the applied force is small with respect to the system's natural frequency, the transmitted force equals the applied force. In the case of a ball bearing, the vibration of the bearing is transmitted to a structure at the same frequency and magnitude. If the excitation frequency is high with respect to the structure's natural frequency, Fig. 9 shows that the response goes to zero. For a bearing, a high frequency fault will transmit little vibration to a structure. However, as the excitation frequency approaches the structure's natural

frequency, the magnitude of the vibration is magnified. Again considering the bearing, if the bearing vibrates at a frequency equal to the structure's frequency, vibrations from the bearing will be transmitted to the structure. These transmitted vibrations will have significantly higher magnitudes than the vibration at the bearing. For example, damping commonly occur in the range of 1 – 17 % (Blevins [3]). As seen in the figure, a damping ratio of 1 % at resonance yields a magnification of force equal to forty times the applied force. That is, the force imparted by a ball bearing vibration at resonance will be magnified forty times in the surrounding structure at 1 % damping. At the maximum damping observed in steel structures of

$$\zeta = 0.17 \quad (1)$$

the transmissibility is approximately

$$\tau = 3 \quad (2)$$

and the magnification of the bearing forces to the structure is tripled.

While the structure of concern is certainly a multiple degree of freedom system, simplifying assumptions with respect to specific vibrations permits the use of a SDOF model. That is, the structure is assumed to be a series of simple spring-mass-dampers as discussed below in the vibration analysis. Extensive vibration data was collected to understand the machinery dynamics through vibration analysis.

VIBRATION DATA

Vibration data was collected on the installed pump motor prior to removing the pump from service, and data was also available for points along the pump column from other testing. These two data sets were compared to vendor vibration data for the bearings. The comparison of measured vibration data and bearing frequencies was the basis for determining a resonant condition. In other words, motor vibrations, pump vibrations, platform vibrations, and ball bearing vibrations were available in order to be related to each other to evaluate resonance.

The relative locations of vibration measurements are shown in Figs. 3 and 4, where MUX and MLX are the upper and lower radial motor vibrations respectively, MZ is the axial motor vibration, SZ is a vertical structural vibration of an I-beam, GZ is the vertical vibration of one of the gratings, and PUZ and PLZ are upper and lower axial pump vibrations respectively.

Vibration Measurements on the Motor

When the pump is installed in the tank, only the motor and the top of the pump are accessible for vibration monitoring. Accordingly, the first step to resolve the problem was to measure the motor vibrations. Numerous measurements were taken on the motor, and the vibrations were comparable. Consequently, only partial results are reported here for radial and axial vibrations. The vibrations are presented here as a function of frequency vs. peak vibration velocity. Vibration velocity is typically expected to be a reasonable indicator for consideration of vibration severity in rotating equipment.

For example, all vibration measurements on the motor indicated reasonable vibration levels. In fact, the levels were within those expected for new equipment, where vibration levels of 0.1 inches / second are common. Even so, the thrust bearing was damaged due to vibration as mentioned.

Lower Motor Radial Bearing. Vibrations at the radial bearing were comparable to other vibrations on the motor, and the maximum vibration occurred at 271 Hz, as shown in Fig. 10. This vibration resulted from the axial vibrations of the pump, which in turn were a direct result of thrust bearing vibrations.

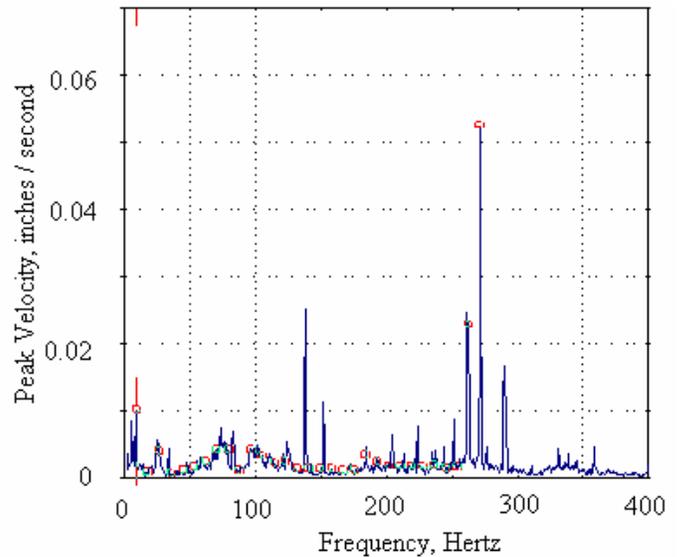


Figure 10: Radial Vibration Near the Lower Motor Radial Bearing, MLX

Axial and Thrust Bearing Vibrations. The vibrations at the top of the motor are displayed in Figs. 11 and 12, where the measurement locations are shown as MZ and MUX for the axial and radial vibrations respectively. Note that in Fig. 11 the radial vibrations are about a quarter of the axial vibrations shown in Fig. 12. The thrust bearing was significantly damaged, and the races of the double row bearing were effectively worn to a larger diameter as the predominant forces on the bearing became axially downward. The radial load would ordinarily be expected to be approximately 75% of the axial load (Mark [4]). Apparently, higher radial vibrations existed when the major damage to the races was in process. Figure 12 is of particular importance since it presents the axial vibrations related to the resonant condition and also presents the axial vibration at the thrust bearing. The thrust bearing was absorbing most of the load in the axial direction when the pump was removed from service. At that time, the vibrations were within acceptable limits for the thrust bearing as noted. Since the vibration levels are low, what caused the platform to vibrate? This question is related to the bearing, pump, and platform frequencies. The vibration of concern occurred at 271 Hz with a magnitude of approximately 0.034 inches / second.

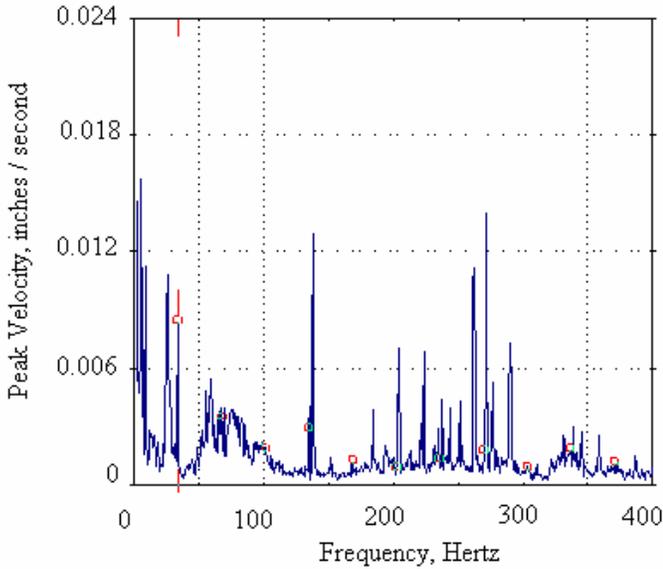


Figure 11: Radial Vibration Near the Top of the Motor at the Thrust Bearing, MUX

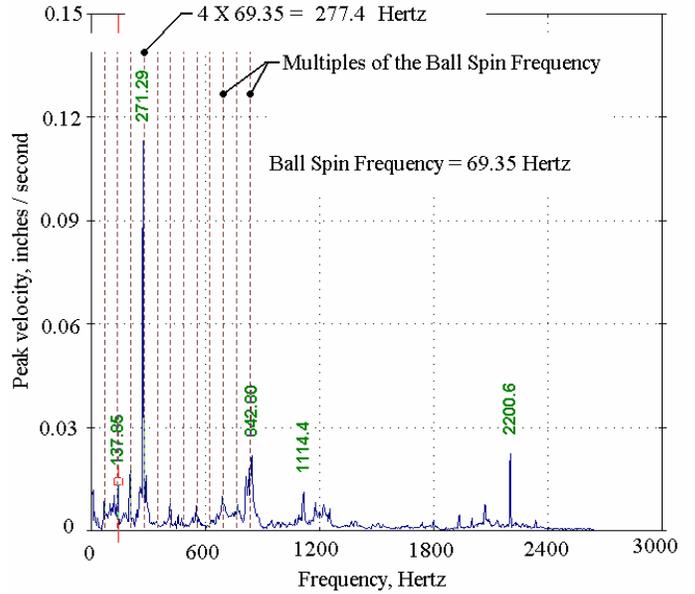


Figure 13: Ball Spin Frequency, Radial Motor Vibration, MLX

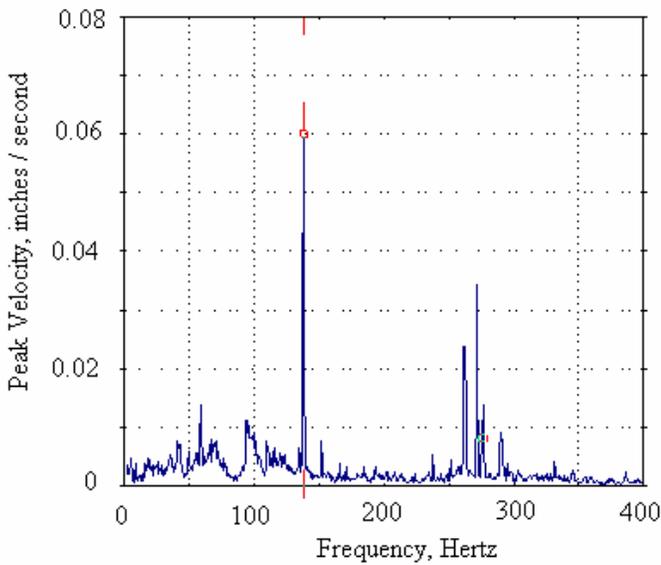


Figure 12: Axial Vibration Measured on the Motor, MZ

Bearing Frequencies

Each of the components within a bearing has a natural frequency at which it vibrates, i.e., the races, the cage, and the balls each vibrate at characteristic frequencies. The ball bearing frequency in operation is referred to as the ball spin frequency, which was obtained from the bearing manufacturer and is shown in Fig. 13. The ball spin frequency was noted in the figure to be coincident to most of the peak axial vibrations measured on the failing motor bearings. The vibrations shown in Fig. 13 were measured at the bottom of the motor, and the largest vibrations occurred at 271.29 Hertz, which are nearly coincident to a multiple of the ball spin frequency. The fourth mode of the bearing was excited by system vibration.

Axial Pump Frequency During Operation

As mentioned, vibrations during operations can only be measured at the top of the pump. Referring to Fig. 14, vibrations were measured at the location labeled PUZ to obtain the results shown in Fig. 14. Numerous vibrations are excited in the pump. Of interest is the vibration at 271 Hz, which has a small magnitude at PUZ of approximately 0.025 inches / second.

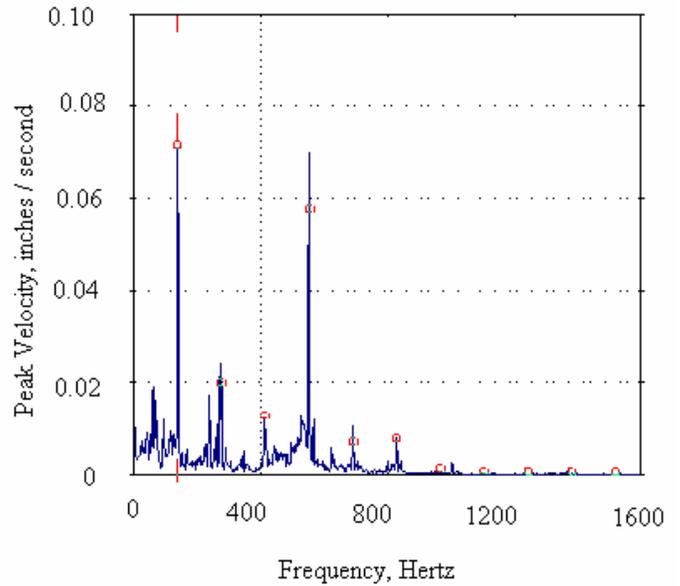


Figure 14: Axial Vibration Measured on the Pump During Operation, PUZ

Natural Frequencies in the Axial Direction for the Pump

Axial vibration data was available for an identical pump tested at Savannah River Site. In particular, resonance testing was performed at

numerous locations along the length of a pump mounted in a test platform (Leishear and Stefanko [5]). Known as a bump test, the test consisted of striking the pump and measuring the excited natural frequencies. Again, each of the frequencies corresponds to the vibration of a specific pump component. The results of interest were axial vibrations (PLZ), which were measured on the pump casing as shown in Figs. 2 and 3. The overall, or unfiltered, response to striking the casing is shown in Fig. 15. A filtered response due to the bump test was obtained from the vibration analyzer used to record the data and is shown in Fig. 16. The filtered response identifies vibration at specific frequencies. Note that a peak vibration occurs at approximately 271.3 Hz, which is nearly coincident to one of the ball spin frequency multiples noted in Fig. 13.

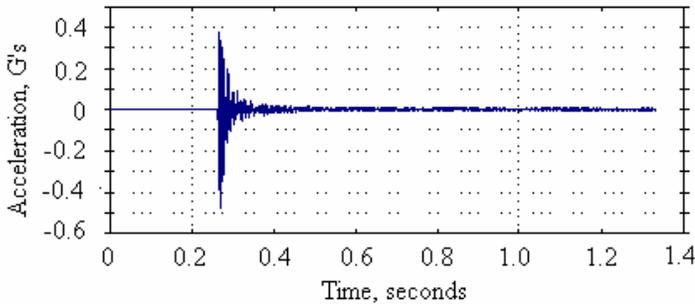


Figure 15: Unfiltered Axial Vibration Measured on the Pump Casing, PLZ

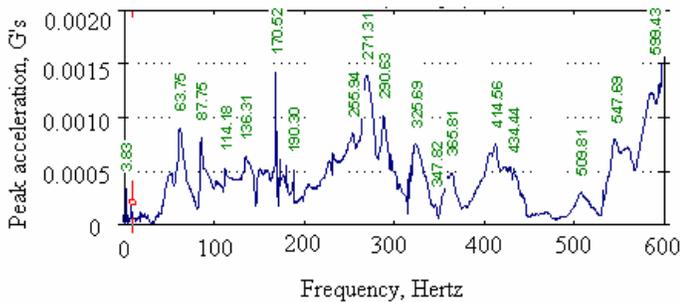


Figure 16: Filtered Axial Vibrations, or Natural Frequencies, Measured on the Pump Casing, PUZ

Mounting Platform Frequency

Platform frequencies were also identified by a bump test and are shown in Fig. 17. Numerous measurements were taken at different locations on the platform, and different natural frequencies occurred at different locations. Several locations had natural frequencies near the natural frequency of interest on the pump. Note that one of the platform natural frequencies at 265.87 Hz is nearly coincident to the pump natural frequency of 271.3 Hz. This frequency was measured at location SZ shown in Figs. 3 and 4.

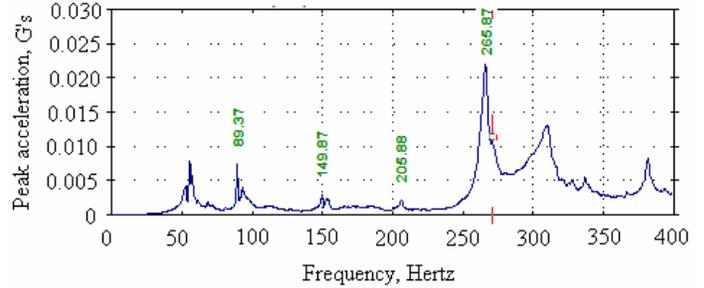


Figure 17: Structural Steel Platform Frequency, SZ

Grating Vibrations

Gratings are bolted to the platform to walk on, and the grating vibrations are typically quite random as shown in Fig. 14. The grating vibrated in response to structural vibrations of the platform. The loud, low frequency noise observed during operation resulted simply from the rattling of the gratings. The vibration response (GZ) of one of the gratings is shown in Fig. 14.

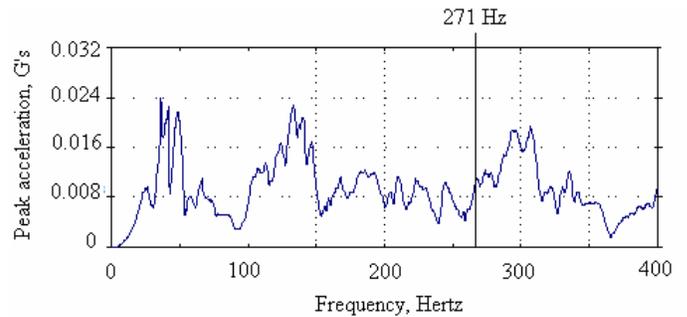


Figure 18: Grating Vibration, GZ

VIBRATION ANALYSIS

A vibration analysis combined a significant amount of data and theory provided thus far in this paper. That is, the numerous figures needed to be connected to the central theme that resonance is the cause of the high vibrations of the structure.

Structural Resonance

In the case considered here, the structural resonance condition was related to three distinct vibrations. Minor vibrations due to the damaged bearing transmitted vibration to the pump and the mounting platform. Each of these components had natural frequencies coincident to a bearing frequency. Vibration magnitudes were magnified as the bearing vibration was coupled to the structure and the pump, and the net result was that structural steel beams in the structure were vibrating up and down for a distance of at least $\pm 1/16$ of an inch. Vibrations in the pump shaft and impeller were expected to be similar, but could not be measured. Some discussion is required to support these statements. In particular, the vibration of the bearing needs to be related to the platform vibration.

Platform Vibration

Although a significant amount of data is available for this analysis, the vibration of the platform was not measured during pump operation prior to the motor replacement. Consequently, analysis is required.

The platform was certainly excited at 271 Hz, as shown in Fig. 17, and the forcing frequency was due to the bearing vibration. The question is, what magnitude of vibration was transmitted to the beam?

Figure 3 provides insight into the response of the platform. Assuming that the tank top, shielding, and vertical I-beams act as rigid structures, the vibrations are identical at each of the four corners of the platform and the shield which surrounds the pump. Then, all that is required is the vibration magnitude at any one of these five points to describe the vibration at the other four points. The vibration at the shield can be approximated from data already provided. Considering Figs. 12 and 14, the axial vibration magnitude of the pump and motor varies between 0.025 and 0.034 inches / second. Conservatively, assume that the vertical vibration of the shield is 0.034 inches / second. From this velocity, acceleration and displacement can be calculated as

$$A = \frac{f \cdot V}{123} \quad (3)$$

$$D = \frac{318 \cdot V}{f} \quad (4)$$

where A is the peak to peak acceleration, f is the frequency, D is the peak to peak displacement, and V is the vibration velocity. Since A is linearly related to V, the velocity is assumed to equal 0.034 inches / second at all four corners of the platform. In this case, the forcing function for each beam is essentially the movement of its ends with respect to the beam due to the vibration of the platform corners. Although a detailed analysis may be performed on the structure, the assumption made here is that the beam can be treated as a SDOF system subjected to a moving base. This assumption warrants the use of Fig. 9. Transmissibility can then be approximated.

From Eq. 4, the displacement at the bearing equals

$$D_{\text{bearing}} = \frac{318 \cdot 0.034}{271} = 0.039 \text{ inches_peak_to_peak} \quad (5)$$

For a complex structure like the one considered here, damping, ζ , is assumed to be large and equal to 0.17, where $\zeta = 3$ from Eq. 2. Structural damping will be due not only to the I-beam construction but from the concrete and steel construction of the tank top. In this case, the minimum displacement of the beam equals

$$D_{\text{beam}} = \tau \cdot D_{\text{bearing}} = 3 \cdot 0.039 = 0.120 \text{ inches_peak_to_peak} \quad (6)$$

At lower damping ratios, the calculated displacement will be even higher. In short, the beam is expected to deflect a minimum of approximately $\pm 1/32$ of an inch, and the gratings may vibrate nearly $\pm 1/8$ inch.

Pump Vibration

The pump is expected to vibrate similar to the platform except that damping is expected to be small. Referring to Fig. 3, structural damping between the pump shaft and bearing is not affected as much as the beam vibration. damping due to numerous structural elements is not present.

The total 1828 pound total weight of the shaft, rotor, and impeller impacts the bearing. Mechanical seals inside the pump or other

components could have been significantly damaged by the increasing vibration, but the bearing damage was identified and corrected before further damage occurred.

Problem Resolution

The motor was replaced and the vibrations were reduced as shown in Figs. 19 - 20. Using traditional vibration analysis acceptance criteria, the bearing vibration would have been considered acceptable. If the resonant condition was not identified, and the equipment was not removed from service, damage to other bearings and pump components was imminent. The vibrations inside the waste tank could not be measured, and the excitation of the massive pump would have caused impacting on other components.

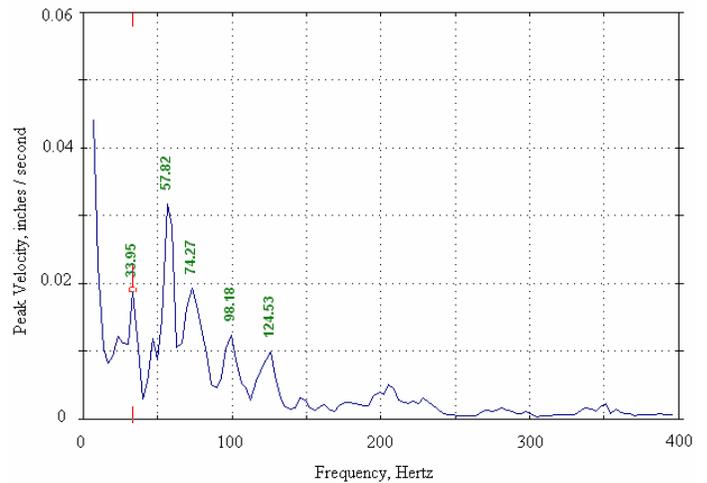


Figure 19: Radial Vibration Near the Lower Motor Bearing After Motor Replacement, MLX

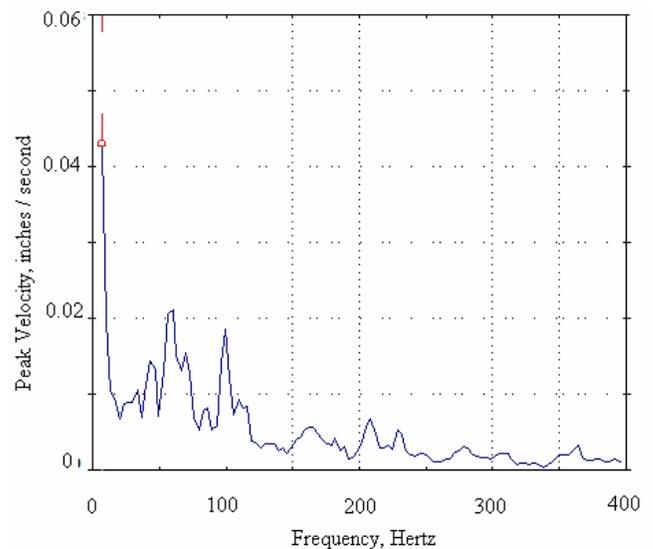


Figure 20: Axial Vibration Measured on the Motor After Motor Replacement, MZ

Grating Vibration

Grating vibrations resulting from the 271 Hz excitation can be explained using Fig. 18. Resonance does not occur since a natural frequency for the grating does not occur at 271 Hz. However, vibrations are transmitted from the vibrating beam to the grating in accordance with Fig. 9. For frequencies below 271 Hz, the transmissibility approaches zero and the effects of a high frequency excitation are negligible. For grating frequencies above 271 Hz, the transmissibility approaches one, and all of the higher frequencies will be affected. In other words, deflections of at least $\pm 1/16$ inch will occur at the grating at frequencies above 271 HZ. The low frequency vibrations heard from the grating were caused by the loose grating banging into the structure as the grating vibrated up and down at high frequencies. The lower frequency random vibrations seen in Fig. 18 created random noise at those frequencies, which interfered with speech frequencies. Consequently, conversations could not be heard near the pump.

ACKNOWLEDGEMENTS

Pump disassembly and photos of the damaged parts were provided by Brian Higby and Greg Beaver at Savannah River Site.

CONCLUSION

An understanding of structural resonance can prevent further equipment damage in operating facilities. In this case, increased noise levels indicated that a resonant condition was in process. Typical vibration acceptance criteria were met, but damage had already occurred, and the damage was accelerating. An understanding of structural resonance prevented further damage, and the intent of this paper is to simply provide a better understanding of resonance as it affects operating equipment. References such as Harris [1] and Thomson [2] provide sound theoretical discussions of resonance, and this paper further elucidates the topic by providing a well documented case of the relationship between vibration analysis and equipment resonance. Although examples of the resonant response of structures are documented in the literature, a specific example, such as this paper, to relate structural resonance to machinery vibrations was not found during a literature search.

REFERENCES

- [1] Harris, R. J., and Young, W. C., 1975, "Shock and Vibration Handbook", ASME, N. Y., p. 8 - 17.
- [2] Thomson, W. T., 1993, "Theory of Vibration with Applications", Prentice Hall, Inc., New Jersey, p. 54.
- [3] Blevins, R. D., 1990, "Flow Induced Vibration", Van Nostrand Reinhold, New York, pp. 326 - 327.
- [4] Avallone, E. A., Baumeister, T., "Mark's Standard Handbook for Mechanical Engineers", 1987, Mc Graw Hill, New York. p. 8-135
- [5] Leishear and Stefanko, 2005, "Relationship Between Vibrations and Mechanical Seal Failures in Centrifugal Pumps", International Mechanical Engineers Congress and Exposition, ASME, New York.