ACOUSTIC RESONANCE PHENOMENA IN HIGH ENERGY VARIABLE SPEED CENTRIFUGAL PUMPS

by

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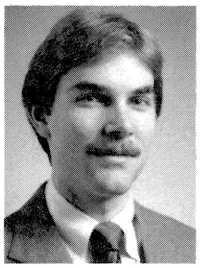

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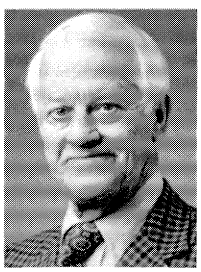

Richard M. Nelson, Chief Technical Consultant for the pump division of Bingham-Willamette Company, received his B.S. degree in Mechanical Engineering from Oregon State University in 1943. Mr. Nelson has spent more than 38 years on the design and analysis of centrifugal pumps. He is a member of ASME and has served on committees for the draft ASME Code for pumps and valves for nuclear power and ASME Section III working group on pumps. He is a company representative to the Hydraulic Institute, a registered professional engineer in the State of Oregon, and has been granted patents for adjustable guide vane hydraulic turbine and double case pump designs.

ABSTRACT

The design of reliable pumps is of the utmost importance in petrochemical and power installations. Downtime can be costly in terms of lost production and in the power industry loss of generating capability. In order to meet this challenge, it is necessary to use state-of-the-art engineering design and analysis capabilities. In recent years, it has been necessary to design pumps at relatively high energy levels capable of operating over a wide range of flow conditions and speeds. Due to impeller vane and volute lip interaction, hydraulic pulses are generated at vane passing frequencies. If the pulse frequency corresponds to an acoustic resonance, the pulse may be amplified by as much as 100 times. The resulting large pressure pulsations interact with the pump rotor to cause unacceptable vibration levels. Variable speeds pumps are particularly susceptible, since they can create pulsations over a wide range of speeds.

An analytical model for predicting acoustic resonance in the internal passages of high energy centrifugal pumps will be presented. This analytical model will establish these acoustic resonances and allow for appropriate design modifications prior to pump testing or in the field where they manifest themselves as unacceptably high rotor vibrations at vane passing frequencies.

Actual test data taken on several multistage pumps has correlated well with the analytical predictions.

INTRODUCTION

Acoustic resonance pulsations have been a problem for some time in pumps and piping systems, with resulting losses in reliability and efficiency. In the past, minimal consideration has been given to pulsation effects in the multistages of pumps or system design. Most approaches to controlling these pulsation effects involve use of physical restraints or modifying the machine, usually to the detriment of the performance and efficiency. Understanding the phenomena associated with the generation and transmission of these pulsations is the first step in controlling these effects [1].

Due to the interaction that occurs between the impeller vane and the pump volute lip, hydraulic pulses are always generated at vane passing frequencies or a harmonic of vane passing. If the pulse frequency corresponds to an acoustic resonance frequency, magnification can occur. As a result, actual pulsating wave amplitudes can be substantially greater than the induced level. Magnification factors at resonance may be as much as 30, and sometimes as high as 100.

In general, the acoustic resonance pulsation problem in centrifugal pumps is caused by an interaction of the pump dynamic characteristics with that of its internal passages. Thus, control of such problems lies in either altering the pump performance characteristics or the internal hydraulic design of the pump.

A simplistic analytical model which can predict acoustic resonant frequencies for the internal passages of a multistage centrifugal pump will be demonstrated in this paper. By predicting these resonant frequencies, appropriate design changes can be made prior to the testing to avoid exciting potential resonances in the operating speed range.

Theory

Analytical techniques exist, based on elementary physics, to predict acoustic resonance in piping systems. Resonances of an internal pump passage can be predicted in the same manner. The resonance of a passage is described by "organ pipe" resonance theory [1].

The relationship between acoustic velocity, wave length and frequency is:

$$a = \lambda f$$

where

a = acoustic velocity, fps
f = frequency, hz
λ = wave length, ft

This equation describes the pressure maxima and minima for either a standing or traveling wave. A standing wave is propagated and reflected when a pressure or velocity perturbation encounters a discontinuity in an acoustic system. Resonant conditions will result when the total propagation time up and down the system is such that the reflected wave reaches the excitation source in phase with a subsequent perturbation. Under these conditions the excitation frequency is said to be at one of the natural frequencies of the acoustic system.

Elementary physics of organ pipe resonances reveals that reflections from a closed end pipe will cause a pressure maximum (velocity minimum) at that end, while reflections from an open end pipe sustain a velocity maximum (pressure minimum) at the open end. Therefore, an open-open pipe will have the various resonant mode shapes as shown in Figure 1, while a closed-closed pipe will have the resonance mode, also shown in Figure 1 [2]. The lowest resonant frequency of these is called the fundamental frequency, wherein the length of the pipe is one-half wave length of the sound at that frequency [1].

Since resonance in a piping system occurs with a velocity maxima at open ends and pressure maximum at closed ends, it follows that the fundamental resonance of an open-closed pipe is a quarter-wave length resonance as shown in Figure 1.

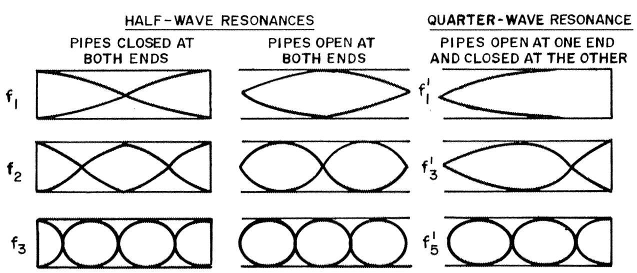

Figure 1. Standing Wave Patterns for Half and Quarter Wavelength Resonance Modes.

The resonant frequencies for the various modes of the three configurations shown in Figure 1 can be calculated from the equation defining the acoustic velocity [2].

Half-wave resonance:

$$f = N\frac{a}{2L}$$

where

N = 1, 2, 3 . . . harmonic

Quarter-wave resonance:

$$f = (2N - 1)\frac{a}{2L}$$

where

N = 1, 2, 3 . . . harmonic

An open end resonance can usually be sustained if the pipe size suddenly diverges to twice the flow area or, conversely, if the area decreases by a factor of two or more [1]. Sufficient reflection will occur to result in a resonance. This is particularly applicable to the crossover passage of a multistage pump. The crossover areas change to reduce the velocity of the fluid being transferred from one stage to another and to distribute the flow uniformly to the inlet of the next impeller [3]. Since changes in area are gradual, it is difficult to predict where a discontinuity exists.

Damping is quite small in this type of acoustic system, and low level pulsations can be amplified by a factor of 100, depending upon frequency, passage size, flow and termination impedance [2]. Since the amplification factor can be so large, a small change in the speed of a pump can result in a large change in pulsation amplitude as shown in Figure 2. Note that the frequency of pulsations generated by a centrifugal pump is a function of the speed. The magnitude of the pressure pulse in a non-resonant condition is related to the impeller diameter/cutwater clearance area, specific gravity of the fluid, developed head, the speed of sound in the fluid, and the horsepower per stage [4].

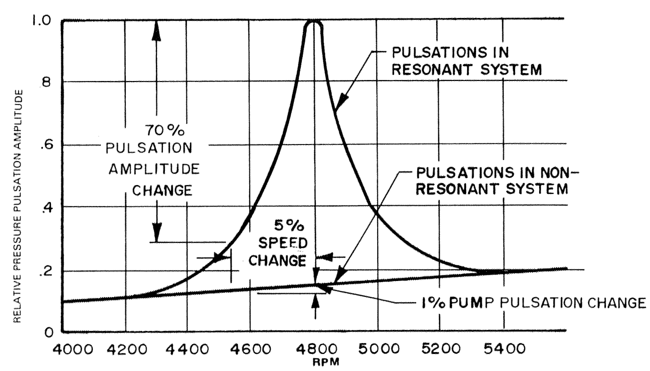

Figure 2. Relative Pressure Pulsation Levels Exhibited in a Resonant and Non-resonant System.

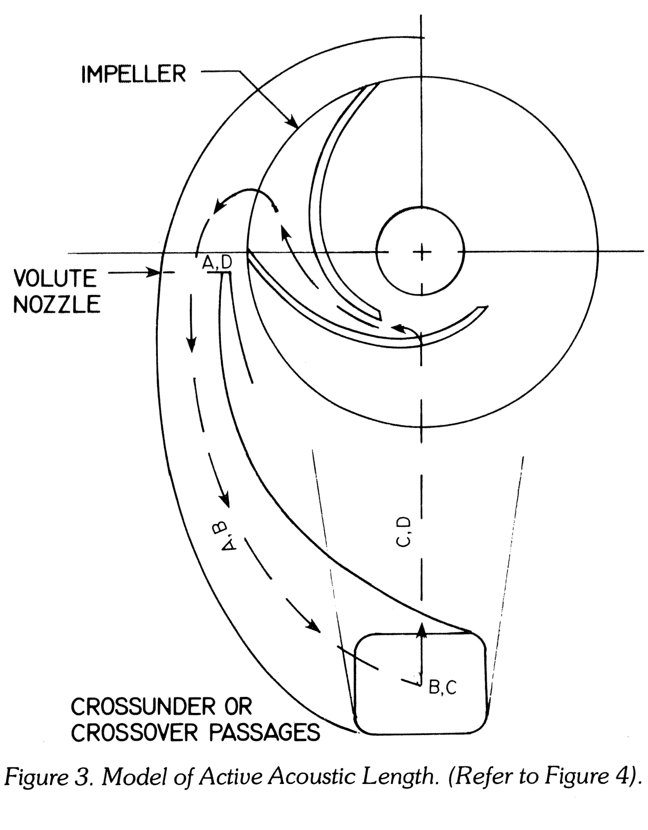

Figure 3. Model of Active Acoustic Length. (Refer to Figure 4).

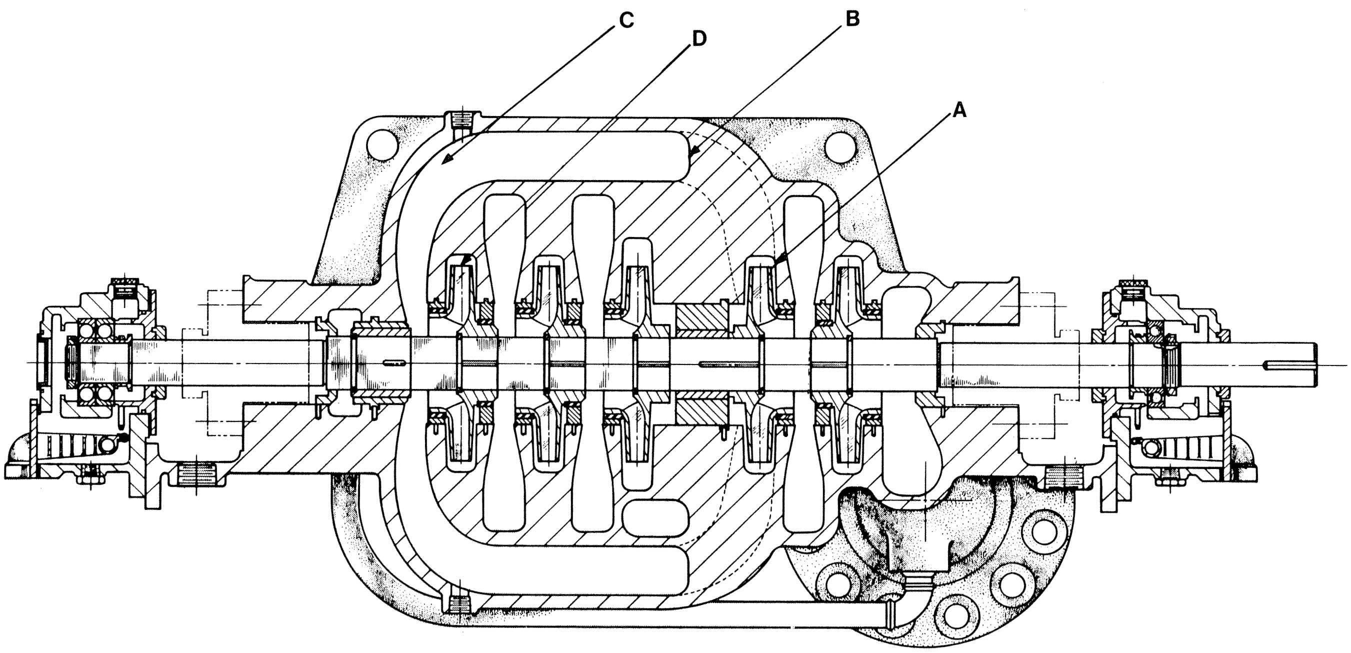

Figure 4. Cross Section of Multistage Centrifugal Pump Showing the Active Acoustic Length.

ANALYTICAL MODEL

Actual test stand data taken on several high energy multistage pumps operating over a wide range of speeds and flows indicated an acoustic resonance was present. From this data, a wavelength was calculated which helped to determine the passage in which the acoustic resonance was occurring. The active length, which is comprised of the multistage crossover and impeller, is illustrated in Figures 3 and 4. Thus, the impeller at each end of the crossover feeds the crossover passage with pressure pulsations at vane passing frequencies. It is interesting to note that the pulsations travel through the impeller at the far end of the passage without being affected.

Verification of the analytical model has been achieved through test data on two different pumps to-date. Pressure pulsations transducers were mounted in the crossover passage and the pressure pulsation amplitudes reviewed, using a real time analyzer. Pulsation amplitudes have been seen as large as 150 psi peak to peak at the resonant frequency. These pulsations have been accompanied by shaft vibrations in excess of 0.25 mils (0.4 in/sec) at vane passing (resonant) frequency. Further verification was obtained by changing the fluid temperature, thus affecting a change in the speed of sound which changed the acoustic resonance frequency, resulting in a change in the peak response speed.

CASE HISTORY NUMBER 1

A high energy water flood pump experienced high vane passing frequency rotor vibrations on test. The test was conducted on a high energy test facility which is capable of 20,000 hp and variable speed operation up to 8400 rpm. This pump had an eight inch suction and six inch discharge nozzle, five stages, five vane impellers and an eleven inch impeller diameter. Speeds on the test ranged from 4800 to 6600 rpm. A plot of vibration data taken on a frequency sweep from 4800 6600 cpm in shown in Figure 5. The pulsation amplitudes measured over the range of vane passing frequencies (400 through 550 Hz) peaked at 465 Hz at a value of .27 mils (.40 in/sec). Pressure pulsation transducers were mounted at the suction nozzle, discharge nozzle, single stage crossover and multistage crossover. The suction nozzle, discharge nozzle and short crossover had low pulsation levels (30 psi peak to peak) over the entire range of the frequency sweep. However, as seen in Figure 6, the multistage crossover had presure pulsation levels as high as 125 psi peak to peak at a vane passing frequency of 28,600 cpm, corresponding to 477 Hz.

It was concluded, based upon the pressure pulsation and rotor data, that an acoustic resonance existed. The active acoustic length matched the half wavelength of the frequency (477 Hz) corresponding to the peak response. Several alternatives were available to correct the problem:

- Avoid operation of the pump in the range of resonance.
- Design a new pump case with a different crossover length.
- Detune the crossover with a quarterwave stub.
- Change the number of vanes in the impellers,

The most practical option was to obtain new impeller

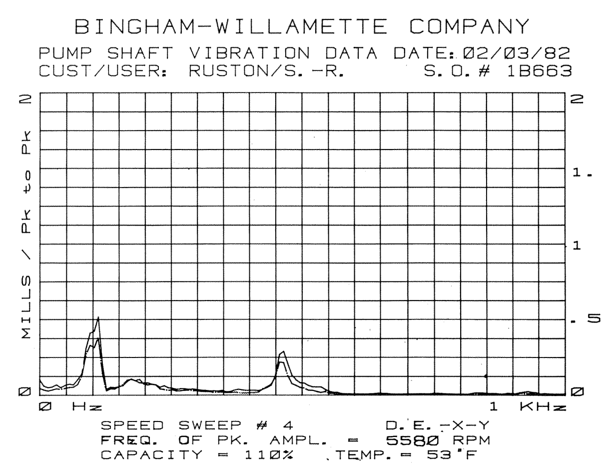

Figure 5. Frequency Sweep of Vibration with the Pump in Its Original Configuration.

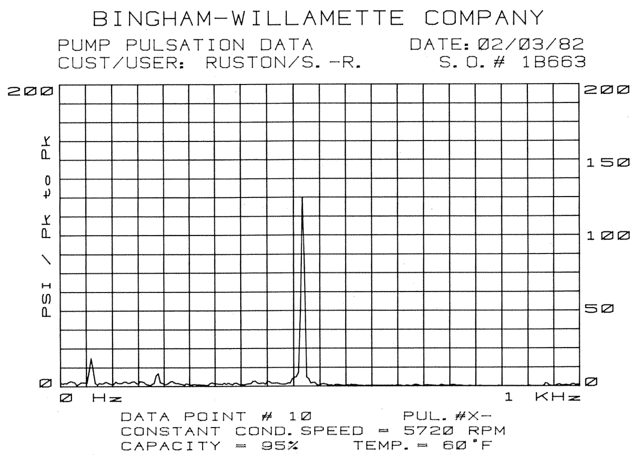

Figure 6. Frequency Sweep of Pulsation Levels in the Multistage Crossover Passage with the Pump in Its Original Configuration.

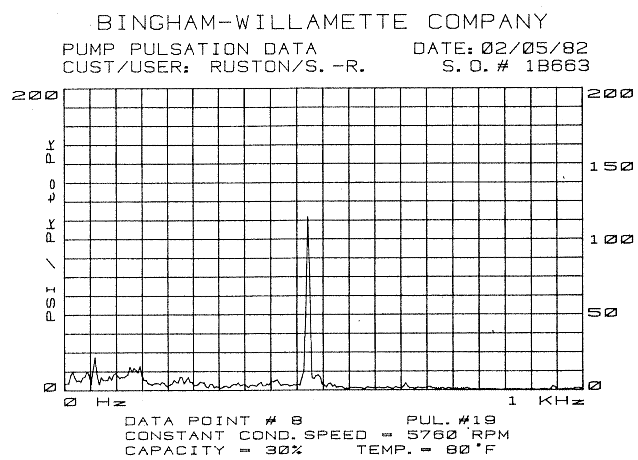

Figure 7. Frequency Sweep of Pulsation Levels in the Multistage Crossover Passage with One Five Vane Impeller Replaced with a Seven Vane Impeller.

castings with a different number of vanes. A two phase test program was established as follows:

1. Replace the five vane impeller immediately preceding the multistage crossover (second stage) with a seven vane impeller and obtain test data.

2. Replace the five vane impeller fed by the multistage crossover (third stage) with a seven vane impeller and obtain test data.

With the impeller in (1) above changed to seven vanes (second stage), the vibration amplitudes were reduced slightly, but the pressure pulsation levels were still high; 115 psi peak to peak (Figure 7). In order to verify the ability to excite the multistage crossover resonance with a seven vane impeller, the pump speed was lowered to 3960 rpm (well below the minimum required speed of the unit). The pressure pulsation plot at this speed is shown in Figure 8.

With both impellers changed to seven vanes on either side of the crossover, the vibrations and pressure pulsations were sharply reduced. The pressure pulsation frequency sweep is shown in Figure 9. It is interesting to note the presence of a large pulsation amplitude at 60 Hz, which did not appear in the other tests. While it is possible that this is an acoustic resonance, it seems unlikely, since the half wavelength would be very long (40 feet), and the resonance would involve the majority of the

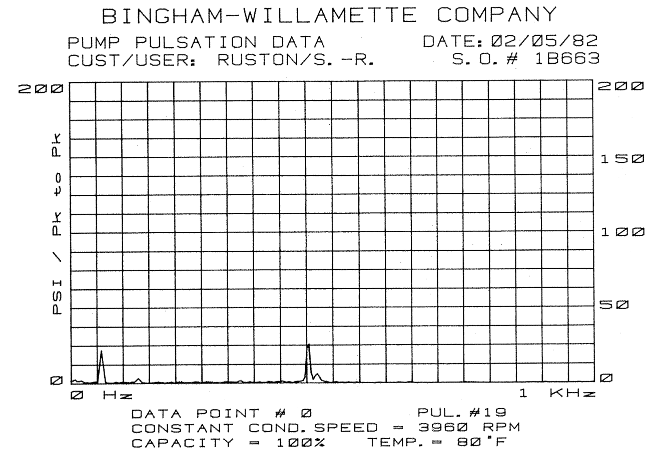

Figure 8. Frequency Sweep of Pulsation Levels in the Multistage Crossover Passage with One Seven Vane Impeller at 3960 RPM.

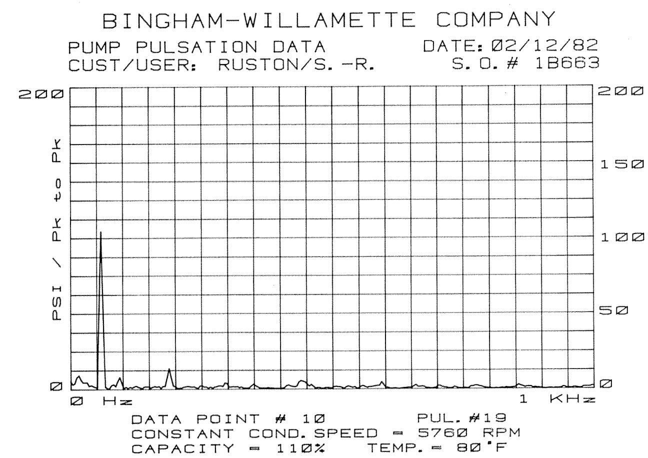

Figure 9. Frequency Sweep of Pulsation Levels in the Multistage Crossover Passage After the Five Vane Impellers Were Replaced with Seven Vane Impellers.

test piping. More likely there was some 60 Hz noise picked up in the instrument cabling.

CASE HISTORY NUMBER 2

Three horizontally split, double volute, multistaged pumps installed in a pipeline application, were experiencing high vane

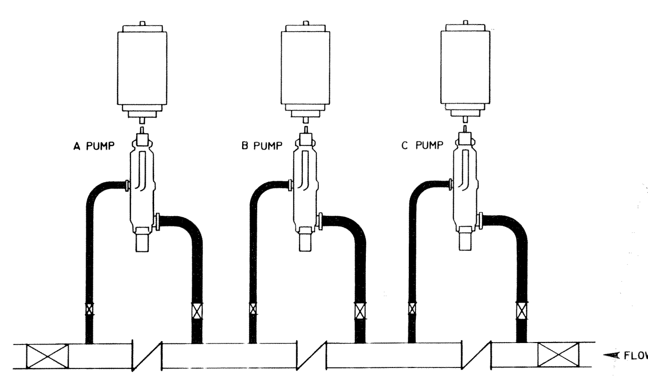

Figure 10. Schematic of the Field Installation with Three Pumps in Series.

passing rotor vibrations (five vane impellers × rpm). These pumps were manifolded such that two pumps could run in series (with the third as a spare) and pump liquified propane gas (LPG) (Figure 10). These pumps were four stage units, destaged to three and had an eight inch suction, six inch discharge, five vane impeller and an eleven and a half inch diameter impeller.

The A pump was started and ran very close to its best efficiency point at 1100 gpm and 1740 feet of head, at a speed of 3560 rpm. Suction pressure was 400 psi with a discharge presure of 800 psi. The required net positive suction head was less than 30 feet. A severe vane passing vibration on the outboard bearing housing was present. As a result of that vibration, the cooling line running to the outboard bearing had broken. Readings as high as 1.0 in/sec in the horizontal direction were seen at 18,000 cpm (300 Hz).

When this problem was initially reported, it was believed to be due to insufficient clearance between the impeller and volute lip. The pump was shipped back to the plant for inspection and testing. Inspection revealed that there was less than 60 mils (1 percent) clearance between the impeller outside diameter and the volute lips. The impeller was trimmed back from 11.80 inches to 11.56 inches, while the volute lips were cut back and the nozzles ground giving a 3 percent clearance. The pump was then retested with water at the plant, with no indication of a vane passing frequency problem. The pump was shipped back to the installation site where it still exhibited excessive vane passing frequency vibrations.

Since field reports did not indicate large structural vibrations on the pedestals or baseplate, an analysis on the acoustic wavelength internal to the unit was performed. Suspicions were further supported by the fact that the vibration amplitudes increased when two units were brought on line, possibly due to a change in the density, since LPG is compressible. The vibration spectrum on the bearing housing with one pump on-line is shown in Figure 11. The maximum vibration amplitude was 0.56 in/sec at 18,000 cpm. The vibration amplitude with two pumps on line is shown in Figure 12. The maximum vibration was in excess of 1.0 in/sec at 18,000 cpm, explaining why the seal piping was failing.

From previous experiences, the multistage crossover has been the passage involved in acoustic resonance problems. Halfwave resonance has been the most common type. Since the speed of sound in LPG was not readily available, it was calculated using a formula for fluid contained in a rigid-walled vessel [5].

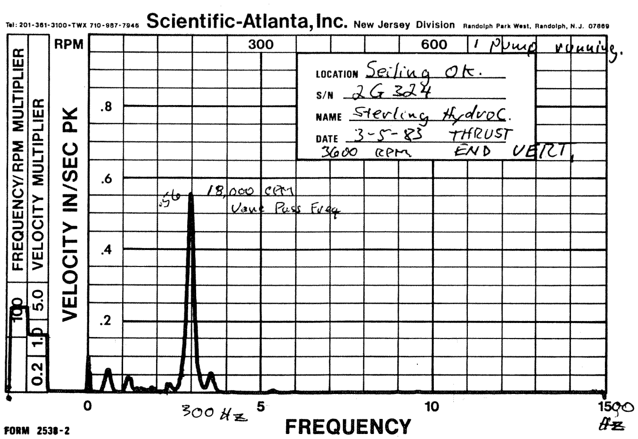

Figure 11. Vibration Spectrum of the Outboard Bearing Housing with One Pump Operating.

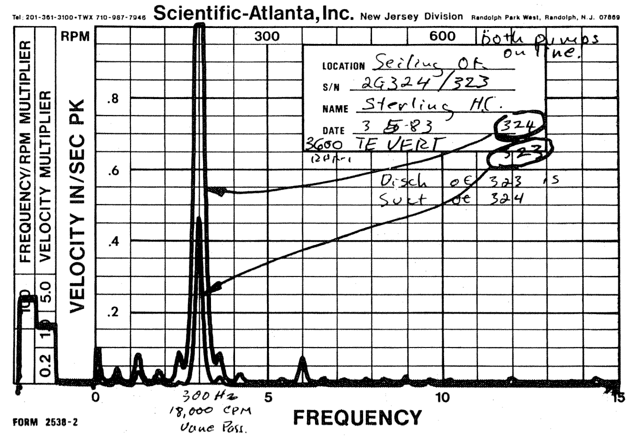

Figure 12. Vibration Spectrum of the Outboard Bearing Housing with Two Pumps Operating.

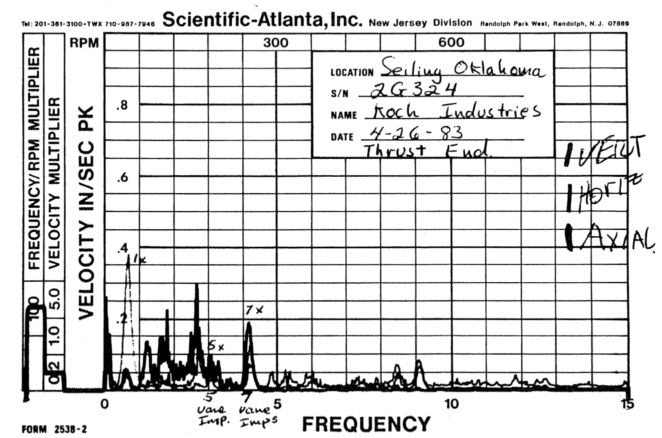

Figure 13. Vibration Spectrum of the Outboard Bearing Housing with Two Pumps Operating.

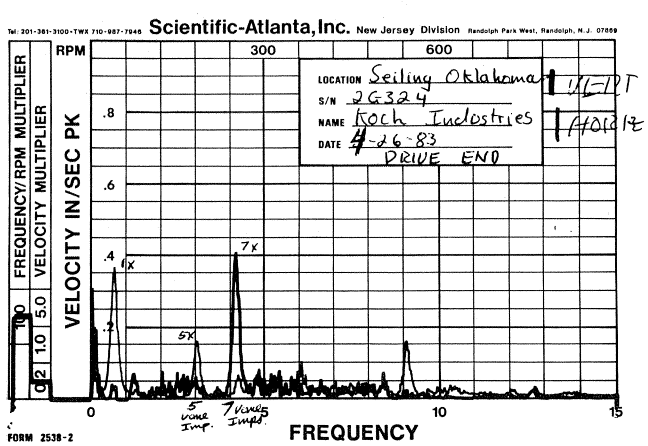

Figure 14. Vibration Spectrum of the Inboard Bearing Housing After the Five Vane Impellers Were Replaced with Seven Vane Impellers.

$$a = \left[g_c \left(\frac{\partial P}{\partial P} \right)_s \right]^{1/2}$$

where: $g_c = 32.17$ ft/sec^2

$\left(\frac{\partial P}{\partial \rho}\right)_s$ = partial derivative of the pressure with respect to density at constant entropy.

The partial derivative can be roughly approximated by using:

$\left(\frac{\Delta P}{\Delta \rho}\right)_s$ = Change in pressure with respect to the change in density at constant entropy.

Using the above approach, the speed of sound in LPG was calculated at several pressure levels. It was found that due to the compressibility of LPG, there was a 10 percent variation in the speed of sound between the suction and discharge pressures.

The calculation of the resonant wavelength was done utilizing field information to estimate the active acoustic length associated with the multistage crossover. The calculated half wavelength varied from 3.4 feet to 4.0 feet, because of the compressibility of the LPG, while the estimated passage length was 4.0 feet. The close match of length explained the behavior of the unit and also explained why the vibration increased when the second unit was brought on line. The additional compression of the LPG was apparently sufficient to change the speed of sound to better match the passage length, hence the higher amplitudes.

Based on the results of the analysis and prior experience, it was recommended that the impellers on each side of the multistage crossover be changed from 5 to 7 vanes. The five vane first stage impeller was left unchanged, since prior experience has shown that it will not excite the resonance significantly. The results of the recommended change of the two impellers are illustrated in Figures 13 and 14. The vibration readings were below 0.1 in/sec at all frequencies with one pump on line and did not increase when the second unit was brought on line.

CONCLUSION

This simple analytical model appears to correlate well with available test data. A developmental test plan is underway to further study acoustic resonance in crossovers of multistage centrifugal pumps. It is the intent of the program to further refine the analytical model by investigating the effect of passage geometry, and by using known input levels to determine actual amplification factors.

The ultimate outcome of this work will result in more reliable pump operation. Elimination of acoustic resonance problems in centrifugal pumps will reduce shaft vibrations which will result in better seal life. Also, high pressure pulsations have been known to cause fatigue failures of impeller shrouds and volute lips.

Improved pump reliability is the major goal of this development work, and to achieve this goal will require both analyses and tests. With the help of this analytical model, analysis can be performed prior to test to help assure a trouble free pump design and on line availability in the field.

REFERENCES

1. *Controlling the Effects of Pulsations and Fluid Transients in Piping Systems,* Southwest Research Institute, Report No. 160 (1981).
2. Sparks, C. R., and Wachel, J. D., "Pulsations in Liquid Pumps and Piping Systems," *Proceedings of the Fifth Turbomachinery Symposium,* Texas A&M University, College Station, Texas, pp. 55-61 (1976).
3. Stepanoff, A. J., *Centrifugal and Axial Flow Pumps,* New York: John Wiley and Sons, Inc., pp. 123-124 (1967).
4. Everett, W. S., "Causes and Cures of Pressure Pulsation," Hydrocarbon Processing, *43* (8), (1964).
5. Perry, R. H. and Chilton, C. H., *Chemical Engineering Handbook,* New York: McGraw-Hill Book Company, pp. 5-4 (1973).

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