SUBSYNCHRONOUS VIBRATION IN A LARGE WATER FLOOD PUMP

by

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ABSTRACT

In 1973 Aramco embarked on a large water flood projects for the Ghawar oil reservoir. After approximately 1 year in service three of the 16 × 16 × 20 2-stage DVMF pumps developed a severe shaft vibration of over 20 mils at a subsynchronous frequency. Testing of one unit with both vibration and pressure pulsation instrumentation indicated the problem was caused by a hydraulic excitation of a shaft critical speed. This paper describes the unique solution to the problem by modifying the suction and discharge piping from abrupt bell shaped reducing/expansion section to long tapers with a 3 to 1 length to diameter ratio. This piping change eliminated the vibration instability on all three machines without resorting to machinery redesigns. The modification was extended to the three additional units in the system and as a bonus, the life of impellers, which had suffered cavitation and vane cracking damage, was improved by a factor of three.

INTRODUCTION

In 1973, Aramco initiated a water flood program to maintain the pressure potential in the Ghawar reservoir. This reservoir, the largest known oil-field in the world, is approximately 200 Kms. long and roughly 35 Kms. wide. The initial water injection program, which has since been expanded, was designed to inject 7¼ million barrels per calendar day of water into the periphery of the field. Aquifer water obtained from supply wells is pumped up to the required injection pressure of 2,000 PSI through 8 injection plants in the Northern sector of the field. A typical injection train is shown on the schematic in Figure 1.

Key components in this system are seven 2-stage shipping pumps, Figure 2, located at four of the injection stations, each with a design capacity of 20,000 GPM. Initially, several of the units had vibration problems which are not unexpected during startup of very large, high speed, high head prototype pumps. These included normal causes, such as, misalignment, rotor unbalance and operation of the pump at off design conditions. After one of the units, SH WIP-II GT-II, had been in service for almost a year, it developed a new vibration characteristic which was extremely severe and which threatened to destroy the unit if not controlled. This was a sub-synchronous vibration which heretofore had occurred primarily in high speed compressors [1, 2]. Before the problem was solved, two more of the shipping pumps had developed the same sub-synchronous instability and operation of the water injection program was being jeopardized.

PUMP DESIGN

The 16 × 16 × 20 DVMF shipping pumps for the injection system is a large two stage, double suction volute pump and is directly driven by a 26,000 HP, iso rated, two shaft combustion gas turbine. Suction to the shipper is supplied by a 24 × 24 × 30 TVS booster pump which is driven off the outboard end of the shipper pump through a step-down gear. The shipper pump has a rated capacity of 20,000 GPM at 3,500 feet head and 4,900 RPM. The photo in Figure 3 shows the train layout along with the suction and discharge piping.

Bearings for the shipper are of a conventional pressure dam design to suppress oil whirl at the higher speeds. The bearing material was babbitt on steel and had a tendency to spall or crack when subjected to the high vibration imposed on the system. Wearrings for the shipper, which were later to be a factor affecting the instability, are of conventional design of a stellite overlay with, initially, API specified clearances of 21 to 24 mils.

The unit had throttle bushing seals with helical grooves to reduce the possibility of shaft seizure. On one unit, a face seal was tested, but a high failure rate due to high speeds and poor water quality forced us to terminate the experiment.

PIPING CONFIGURATION

Suction for the booster pumps is taken from a common header which feeds two injection trains in a given plant. The booster pump then supplies suction to the shipper through a pipe loop approximately 75 feet long, which can be seen in the aerial photograph in Figure 4. The suction and discharge piping is connected to the pump through bell shaped transition pieces. These transition pieces, shown in Figures 5 and 6, were later found to be a prime factor in the sub-synchronous vibration of the pumps.

VIBRATION PROBLEMS

During the initial year after startup of the units, various and sundry vibration problems were observed. These problems were rotor unbalance, produced quite often by the coupling, misalignment, and vane frequency excitation. The vane frequency excitation problem was particularly severe in the

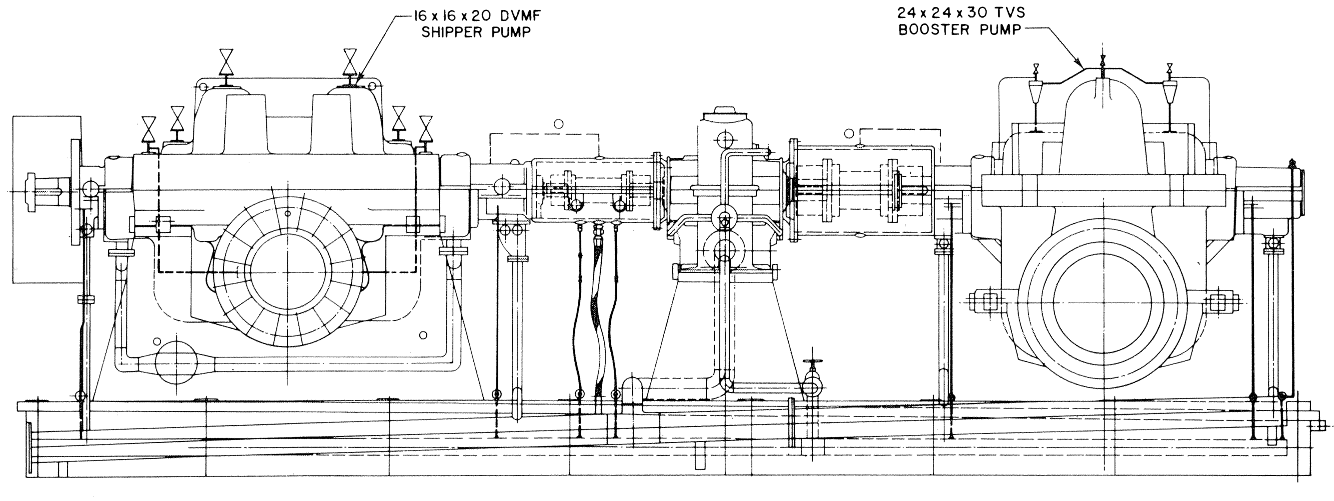

Figure 1. Schematic of a Typical Injection Train.

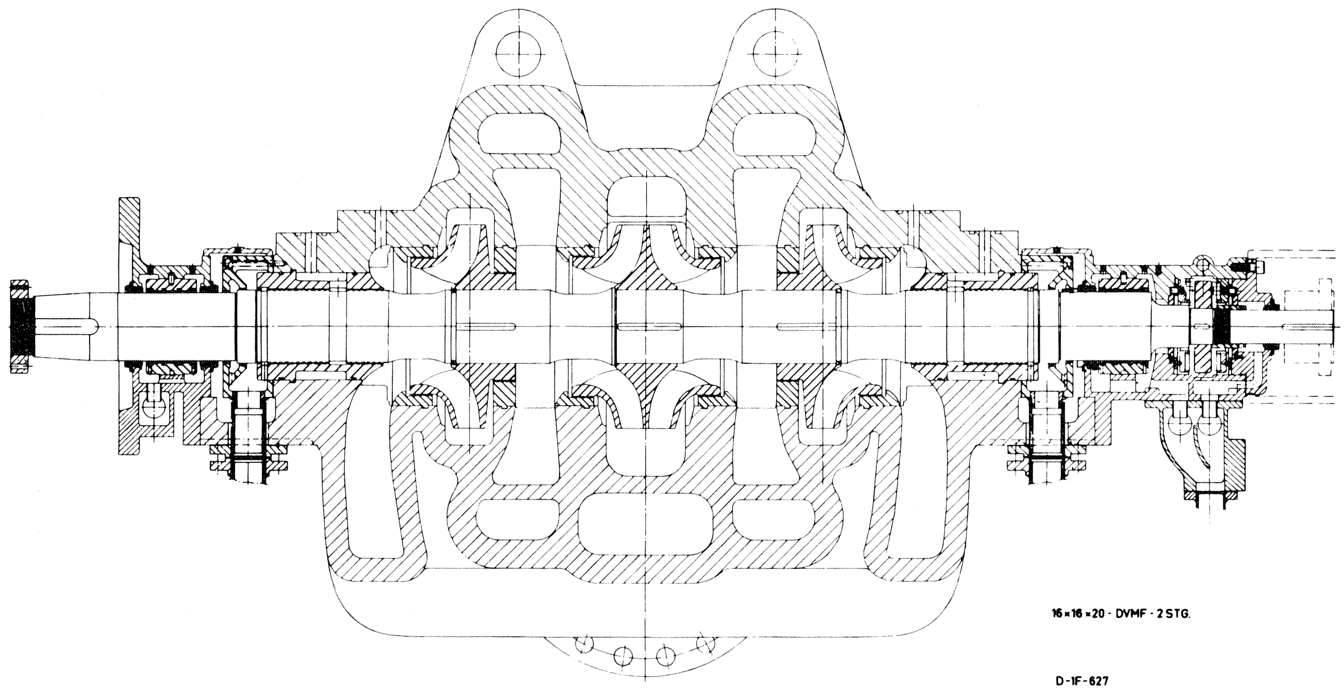

Figure 2. Outline Drawing of the 16×16×20 DVMF Shipping Pump.

initial stages of the plant startup. On startup of some units, the initial resistances of the line and injection wells were lower than design before the reservoir pressure could be brought up and the flowlines packed. This resulted in pump operation far out on its curve. In other units, injection well drilling was still in progress with the consequence that the pumps were operated well back on their curve. These problems, however, were those that could normally be expected in an installation with prototype equipment.

After a year's service, a vibration characteristic developed which could not be explained. At first, it was thought the vibration was at running speed of the units. At that time, we had access only to manually tunable vibration meters with relatively large band width filters. Consequently, when readings were taken using bearing mounted vibration pickups, it was thought that the vibration was occurring purely at running speed. Subsequently, more advanced instrumentation proved that this was not the case.

Because of the erroneous diagnosis of unbalance, the subject pump was brought into the shops on numerous occasions and balanced. When the unit would get to the field, the vibration amplitudes would be as bad as ever; in some cases, on the order of two inches per second on the bearings. After several of these exercises, Maintenance concluded that everything had been done to check out the unit mechanically. At this point, it was decided to instrument the shipper pump with proximity probes in order to obtain more complete data on the problem. The pump was brought into the shop and provisions were made for installing proximity probes. This installation proved somewhat difficult to do since the unit had not been originally designed for probes and there was a great deal of problem in finding sufficient shaft space to install them. Eventually, X and

Figure 3. Water Injection Pump Train.

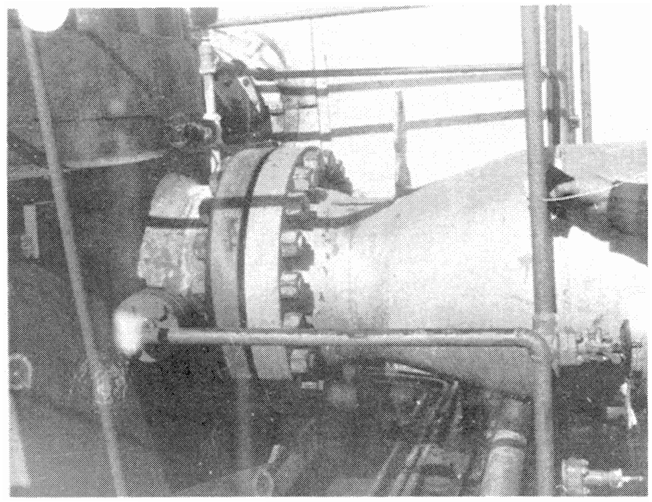

Figure 5. Original Shipper Pump Suction Transition Piping.

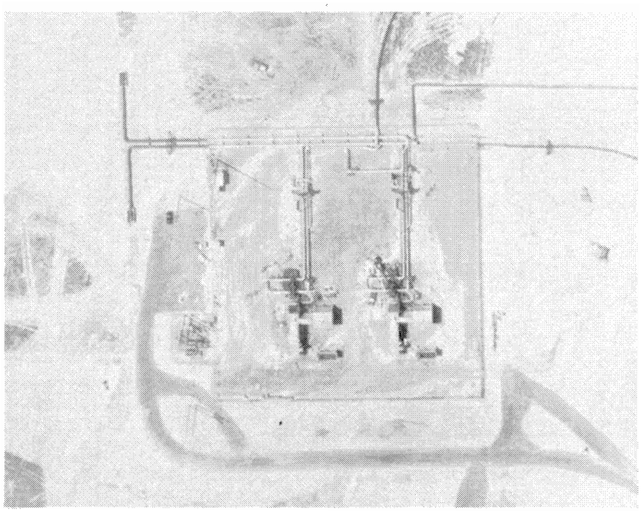

Figure 4. Aerial Photograph of Pump Station.

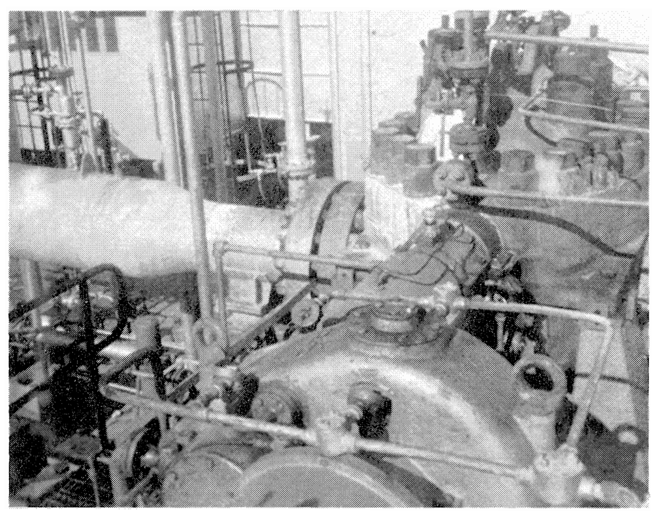

Figure 6. Original Shipper Pump Discharge Transition Piping.

Y probes were installed outboard of each bearing approximately three inches from the bearing and were oriented 15° from the vertical and horizontal planes. The unit was put back into service and testing commenced in early June 1974.

TESTING

The injection train was brought up to speed to a normal startup procedure of the gas turbine and all data appeared normal. The unit went through what appeared to be a well damped critical at 2,800 RPM as we brought the speed up in 400 RPM increments. During the test, vibration amplitudes and frequencies were recorded manually.

At a speed of approximately 3,300 RPM, a new characteristic became evident. This vibration pattern is shown graphically in Figure 7 as a beat frequency. Prior to this time, we had felt we were dealing exclusively with a one frequency problem. As the beat frequency pattern became evident, it was obvious we were dealing with two discreet frequencies which were close together. As we brought the machine further up in speed, the beat frequency decreased and the amplitude of the vibration increased. As the two frequencies became separated further, we were able to discriminate between the frequencies with our manual tunable filters. Taking the unit up towards its maximum speed, it was apparent that the sub-harmonic frequency remained relatively constant as the running speed increased. A plot of this vibration is seen in Figure 8 which shows vibration spectra as a function of a pump speed. As can be seen on the curves, the sub-harmonic vibration component increased in frequency slightly with running speed and the amplitude increased dramatically. The unbalanced running speed component of the unit increased as one would expect due to unbalance. At 4,600 RPM, which was the maximum speed we dared take the unit, the inboard bearing was showing a shaft amplitude of 20 mils. Needless to say, the data taken at this point was sparse. We brought the unit down in speed noticing the beat frequency until we dropped below 3,500 RPM.

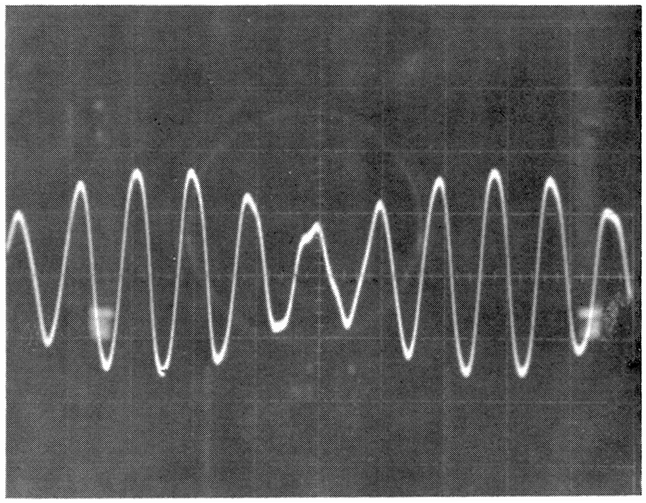

Figure 7. Beat Frequency Vibration Pattern.

SUBSEQUENT TESTS

Because we were dealing with two excitation frequencies, investigations were initiated to determine the source of the sub-synchronous component. We quickly eliminated any components in the drive train as being suspect and started concentrating on external factors of the system, such as, an acoustic or hydraulic excitation. As seen previously, the piping loop between the booster pump and the shipper pump was approximately 75 feet long. Initially, we thought we were dealing with an acoustic resonance similar to an organ pipe phenomena and this loop was of the approximate length to produce the excitation frequencies of 3,400 to 3,700 cycles per minute. The exact frequency could not be determined because of the unknown speed of sound in the water that was being pumped. The speed of sound in water is variable because of changes in water composition, dissolved gases and temperatures. Assuming a range of the speed of sound of the fluid between 4,200 and 4,800 feet per second, the 75 feet length of pipe became a prime candidate for a first mode organ pipe resonance. The subsequent tests which included probing the suction loop pipe with pressure transducers, failed to measure pressure fluctuations which could be driving the rotor. Although pressure measurements were made at several points in the loop, no frequencies corresponding to the sub-synchronous vibration were detected. However, on the discharge piping immediately downstream of the transition piece, we were able to pick up a very small excitation at the sub-synchronous frequency. This pressure pulsation was of the order of 5 PSI peak-to-peak as seen in Figure 9 and was a small portion of the pressure pulsation spectrum which was dominated by the vane frequency produced by the pump. Although it was suspected that the sub-synchronous component was significant to the problem, it was difficult to explain why a minor pressure pulsation could excite the rotor to the vibration amplitudes that were observed. An

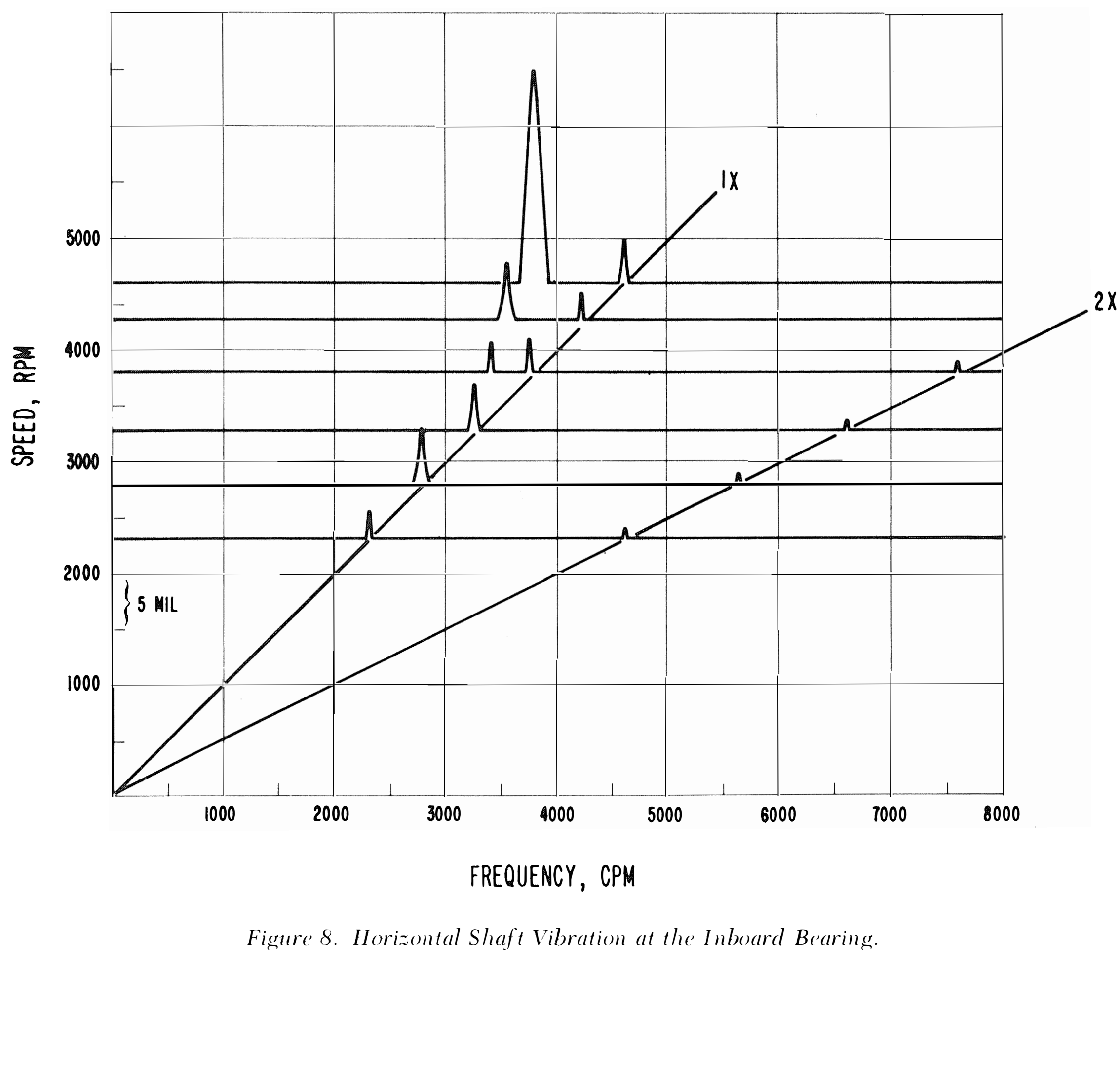

Figure 8. Horizontal Shaft Vibration at the Inboard Bearing.

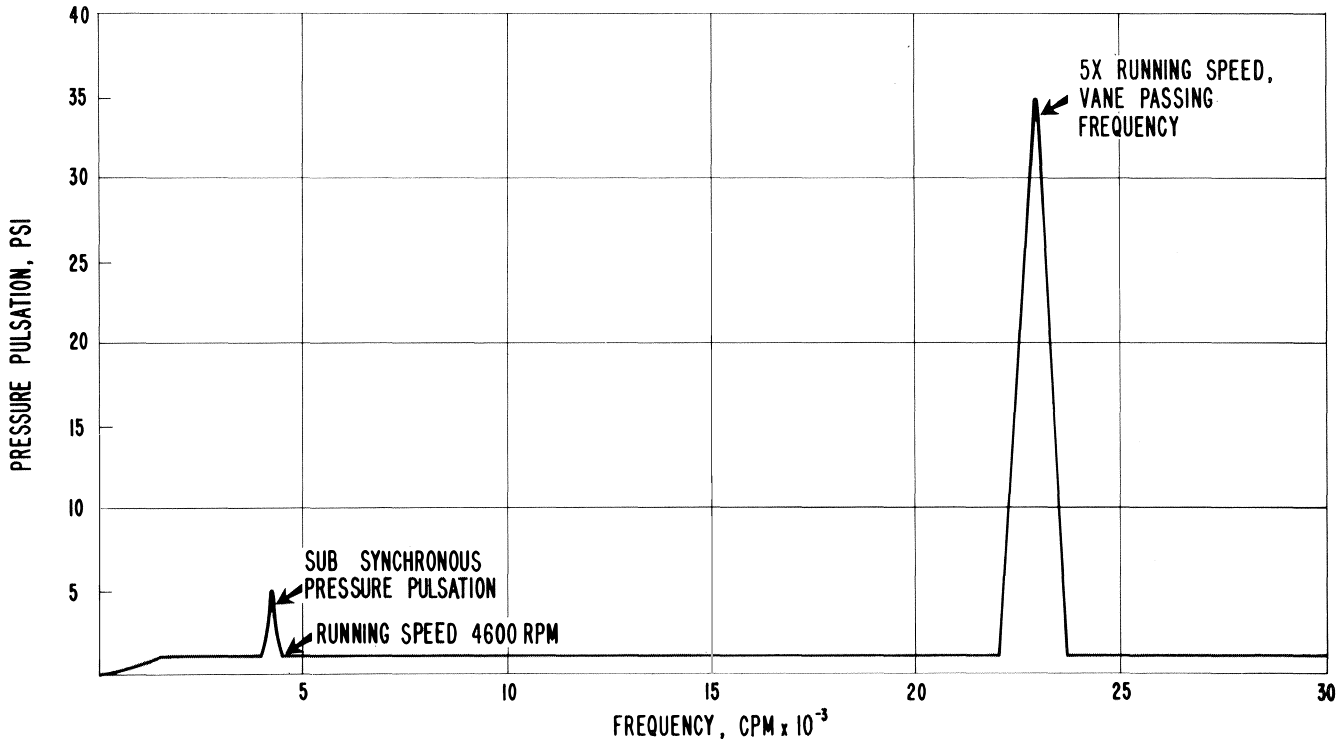

Figure 9. Pressure Pulsation Spectra as Measured in the Discharge Piping.

acoustic resonance was considered, but because of the long wavelengths involved for this low frequency, none of the piping components with the exception of the suction loop could have produced the required frequency.

A WORKABLE HYPOTHESIS

In the July 1973 issue of the "Scientific American," Arthur Benade wrote a paper titled "The Physics of Brasses," [3]. In this paper, Mr. Benade investigated the mechanism that allows a brass musical instrument to produce a musical tone. His paper describes the physics of how the acoustic energy in an instrument is fed back to the instrument through a reflection of the horn's bell. The paper investigates the feedback mechanism between a player's lips and the acoustic waves interacting with the instrument's bell. The mechanism described in the paper for the production of a musical tone is this; the player's lips vibrate producing pressure pulsations which go out to the end of the instrument and because of the shape of the bell, part of the acoustic energy is reflected back into the horn system, causing the player's lips to vibrate further. It is this coupled acoustic mechanical dynamic system which produces the musical tone of brass instruments.

Using this model of an acoustic mechanical interaction or coupled system, we hypothesized the following mechanism to account for the very large vibration amplitudes at the subsynchronous frequency. Some initial excursion of the shaft produced, say by unbalance, caused the shaft to deflect and produce a pressure pulsation which is transmitted through the pump volute to the discharge piping. At the abrupt transition piece of the piping, shown in Figure 6, part of the pressure energy is reflected back into the system similar to the way that sound is reflected by the bell of the horn. This reflected energy would then excite the shaft again at its resonance, setting up a coupled acoustic dynamic system. The shaft produces pressure waves which are reflected back causing further shaft movement.

A factor which supported this mechanism was the wearring and seal clearances on the particular unit which was causing problems. Because of lack of spare parts, Maintenance had been forced to install wearrings whose clearances greatly exceeded the required 21 to 24 mils. It was reported that the diametrical clearances were on the order of an eighth of an inch. Also, during the period of testing, two other machines in the system developed similar vibration symptoms and it was suspected that these machines also had wearring clearances which had been eroded due to sand in the water. We, therefore, had two conditions which interacted to produce an extremely severe vibration:

1. Clearances within the machine were such that shaft movement produced pressure or flow variations and light damping.
2. These pressure pulsations could be reflected back into the machine by the abrupt expansions of the transition pieces in the discharge piping.

ROTOR RESONANCE TEST

Because of the extremely high shaft amplitudes measured on the test, it was highly likely that the rotor was being excited at a resonance. The turbine vendor had predicted a pump critical speed at 2,600 RPM and we had measured a small peak slightly higher at 2,800 RPM. The once per rev component however did not peak in the same region of 3,300 to 3,700 RPM that the sub-synchronous vibration occurred.

To investigate this resonance question, a pump element was tested in the shop by supporting the shaft on V-blocks and measuring its response to impact with a real-time spectrum analysis. The response, shown in Figure 10, verified that the

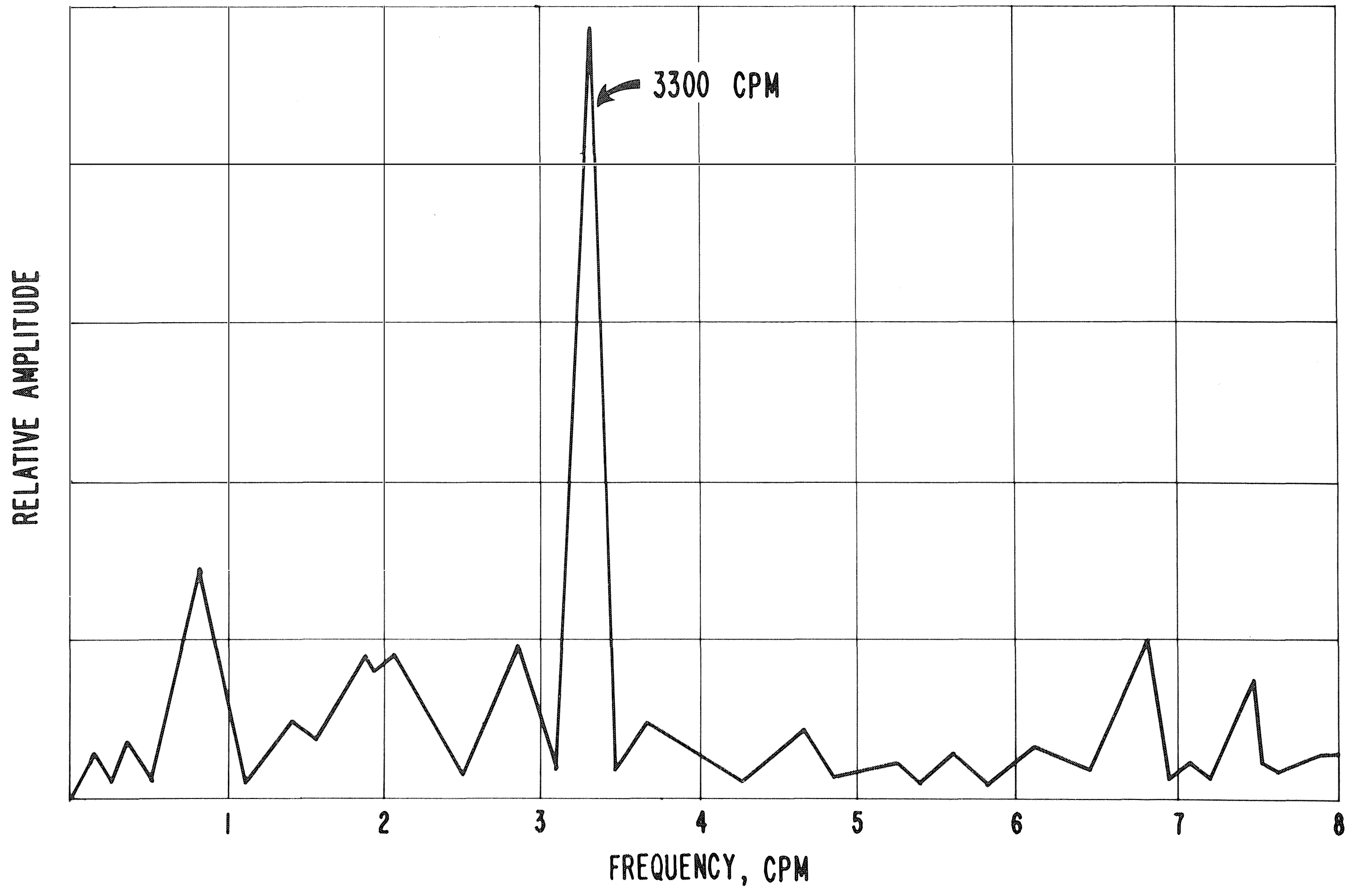

Figure 10. Impact Response of Pump Rotor.

fundamental frequency was 3,200 to 3,300 CPM or very near the speed where the sub-synchronous vibration started. It is still uncertain as to the reason the synchronous forces have never excited this mode.

THE SOLUTION

Using the hypothesis just described, it was proposed that the abrupt piping area changes be eliminated and replaced with transition pieces which had a much longer taper. We arbitrarily selected a L/D ratio of three to one for this change. Because of press of time, it was not possible to have the transition piece rolled in a normal manner. We, therefore, had the transition pieces fabricated by the orange peel method, that is, sections of the pipe were cut out and the discharge transition piece produced as shown in Figure 11. Since we still were not absolutely certain that interactions on the suction side of the pump were not causing problems, transition pieces with the long tapers were installed on the suction also, Figure 12. Unfortunately, due to the pressure to solve the problem as quickly as possible, we were forced to change more than one variable, the piping and also the wearrings. Therefore, for the initial test, we would not be certain as to which modification produced the solution; changing the piping or changing the wearrings.

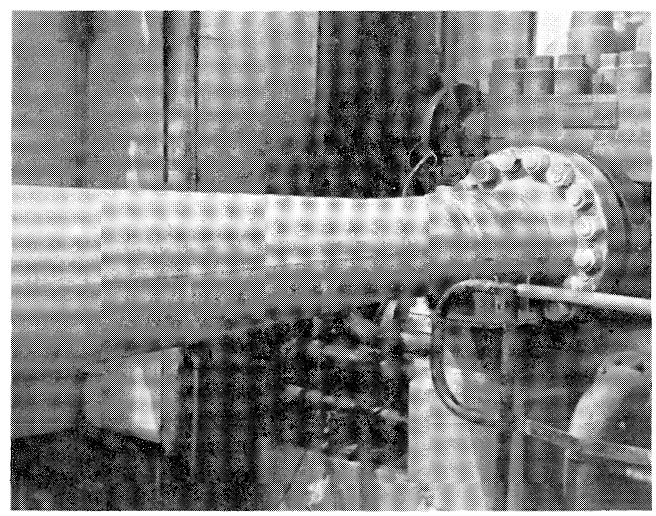

Figure 11. Modified Long Taper Transition for the Pump Discharge.

PROOF TESTING

After fabrication and installation of the long tapers, the earlier tests were repeated. The unit was brought up to speed in 400 RPM increments while measuring shaft motion, bearing motion and pressure pulsations on the discharge. During this test, the unit was brought to the design speed of 4,750 RPM with no evidence of the sub-harmonic excitation. The vibration patterns of the shaft, shown in Figure 13, indicated a

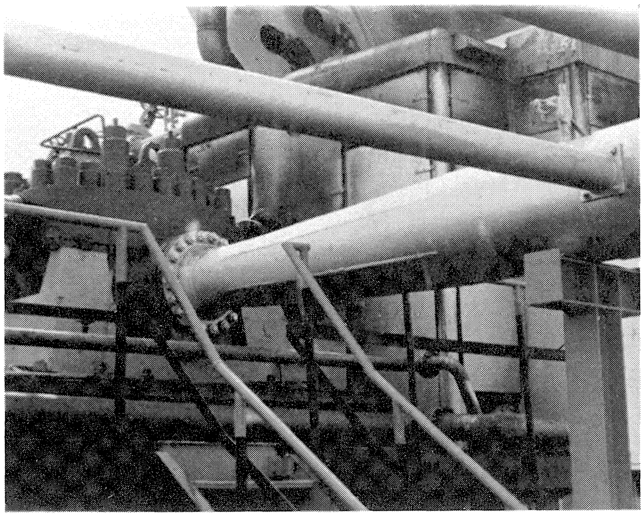

Figure 12. Modified Long Taper Transition for the Pump Suction.

maximum of 5 mils on the inward bearing at running speed. The outboard bearing vibration was on the order of 2 mils.

With the success of the long taper modification clouded only by the possibility that the solution was due to the wearring change, we then installed tapers on the other two similar units which were exhibiting sub-synchronous vibration. In both cases, the only modification made was the piping change from an abrupt expansion to a long taper. These changes proved highly successful in that the sub-synchronous vibration was completely eliminated on all machines. The long tapers proved so successful in eliminating the problem that a program was implemented to change the piping on all 16 injection and shipping pumps in the water flood system and has since become our standard discharge piping configuration on all higher energy shipping pumps.

A BONUS FOR IMPELLER FAILURES

On many of the injection system pumps there had been a problem with impeller vane cavitation and breakage. Before implementing the long taper program, we were sustaining impeller failures of the order of 2,700 hours. After implementing the long taper changes, the mean time to failure of the impellers increased to 7,500 hours as is seen on the hazard plots in Figure 14. It is suspected that this improvement in impeller life is due to the smoothing of the flow entering the pump through the suction transition.

CONCLUSIONS

The following conclusions have been drawn from the problem described in the paper:

1. The sub-synchronous vibration on the shipping pump was caused by hydraulic pulsations which were reflected from a sudden area change in the discharge piping. This energy excited the rotor's fundamental mode and produced a coupled hydraulic-mechanical self excited resonant system.

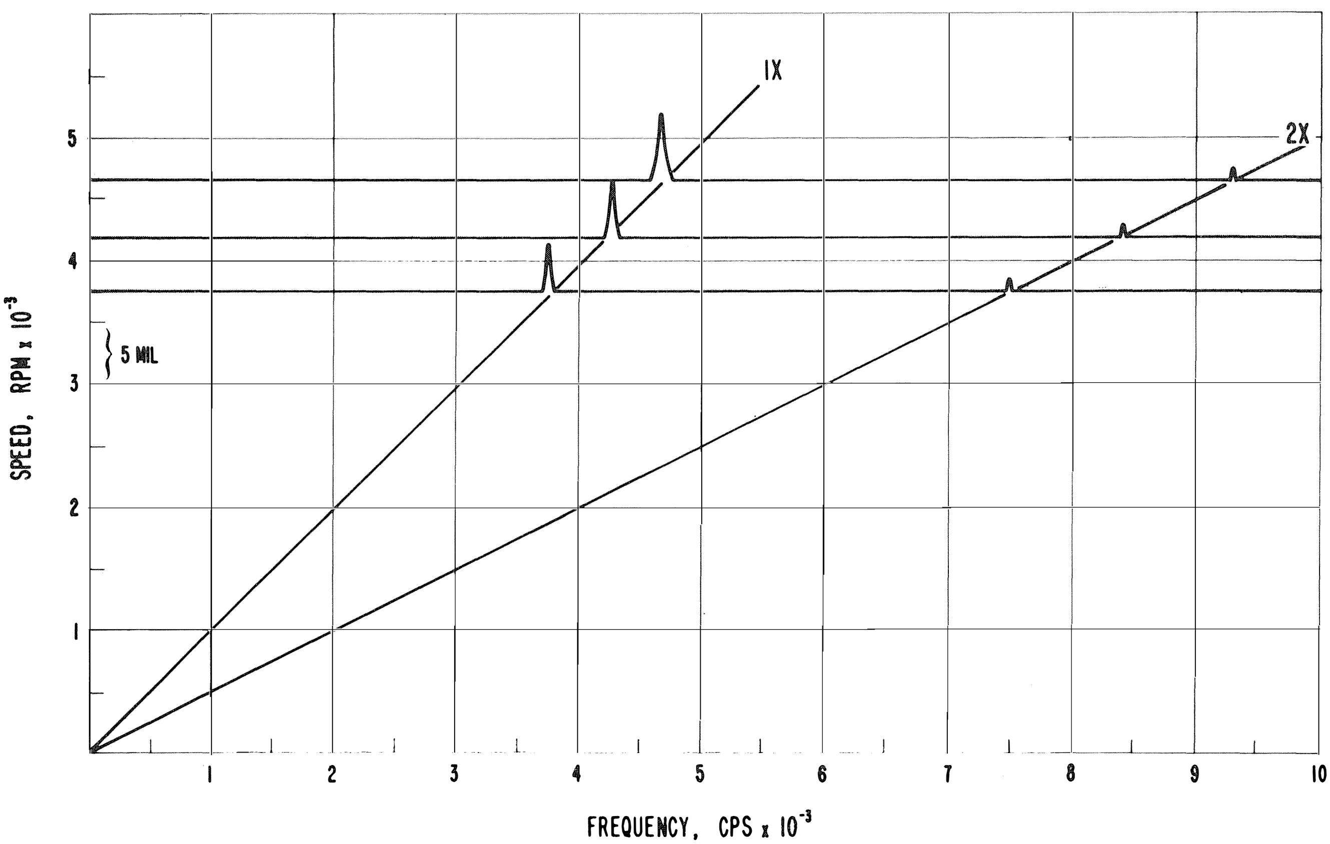

Figure 13. Vertical Shaft Vibration on the Inboard Bearing after Installation of Long Taper Transitions.

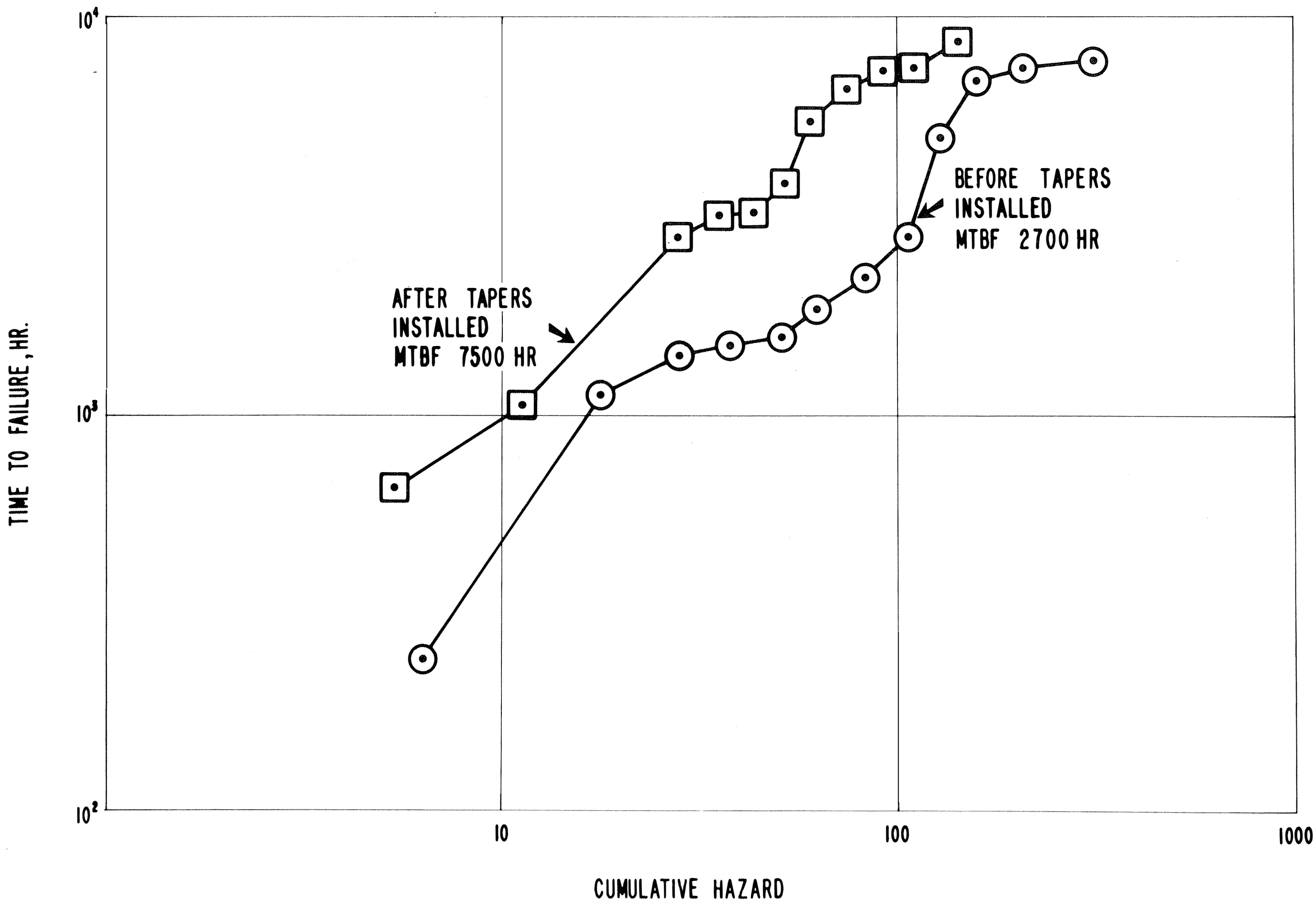

Figure 14. Improvement in Impeller Life after Installation of Long Taper Transitions.

2. Excessive clearances in the seals and wearrings of the pump contributed to the problem by reducing the damping and support stiffness of the rotor. Variation in the leakage flow across the wearrings during shaft excursions may also be the mechanism for producing the hydraulic pulsations.

RECOMMENDATIONS

This exercise has produced several lessons that should be applied to other rotating equipment problems.

1. When an unusual sub-synchronous vibration occurs, such as described in this paper, the engineer or diagnostician when approaching the problem should not put on blinders and concentrate entirely on the problem machine. The machine functions as a part of a system and it is this system, interacting with its components, which can produce a problem.
2. Any time a sub-synchronous instability arises, pressure pulsation data is required not only within the machine, but within the associated piping. All too often the problem machine is instrumented in such a manner that the machine cannot be seen for vibration pickups and probes attached to it and little if any thought is given to measuring the pressure pulsation produced by the acoustic or hydraulic excitation.
3. Area changes in piping should be as gradual as space permits. This can have two benefits:
   - The energy in the flow system will not be reflected back into the machine producing standing waves or interacting with the machine dynamics.
   - Flow is smooth and stabilized entering the machine, producing fewer problems with cavitation and impeller distress.

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