VIBRATIONAL PROBLEMS OF LARGE VERTICAL PUMPS AND MOTORS

by
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ABSTRACT

A large vertical pump with its associated motor driver constitutes a very complex dynamic system. The apparent simplicity of the visible cantilever portion of the machine belies the complex interaction of the unit with its piping, foundation, soil and process fluid, all of which must be considered in order to achieve a successful design. This system approach is presented in this paper by an analytical and experimental study of a 3000 hp crude oil loading pump. The study shows the large number of possible vibrational modes which can exist near the operational speed of a typical system and the factors which should be considered in predicting resonant frequencies. Also included in the paper are several case histories which illustrate some of the dynamic problems which are common to vertical machines.

INTRODUCTION

Vertical canned pumps are usually applied in the oil and petrochemical industries to transport fluids when the available suction pressure is low or when space is at a premium. Over the years, the size of these units has increased until now the top of the motor driver may be fifteen or twenty feet above the base (Figure 1), and the barrel and drop column may extend even further below the ground. While this size and configuration may produce an efficient pumping system, it also introduces several dynamic problems which must be considered in order to obtain a reliable and trouble free installation.

The first problem is the long flexible cantilever structure above the ground, with the motor driver on top. The motor, with a small amount of unbalance, can act as a very efficient vibration excitation source. The second major problem is that the other half of the system, the barrel and drop column, is located below the ground and is not readily accessible, either visually or with normal instrumentation. These too can be considered as flexible cantilevers, as shown in Figure 2, but with the added inconvenience that one has very little opportunity to test, measure, or monitor what is happening below the surface.

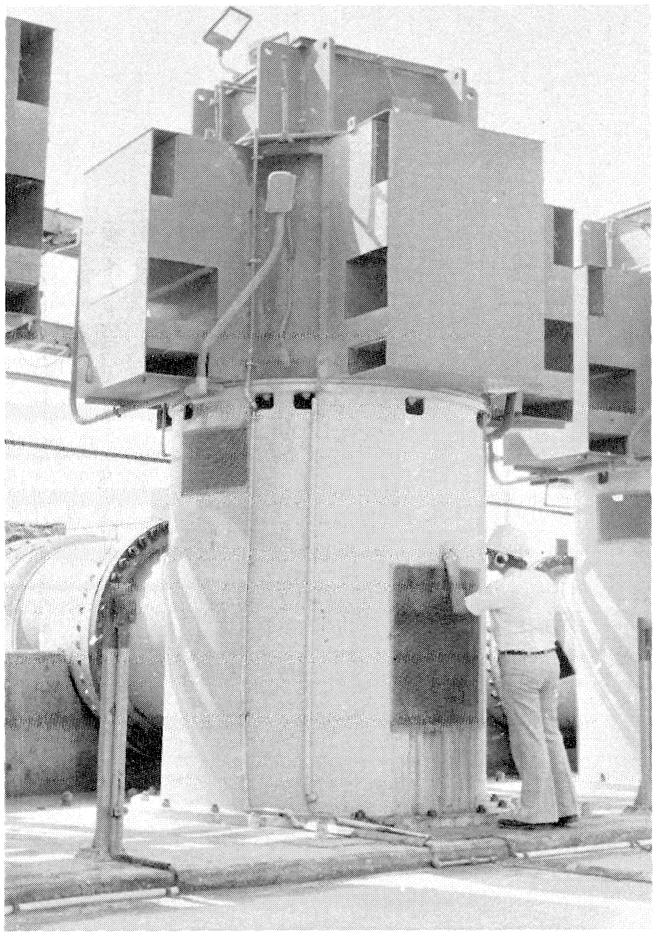

Figure 1. Typical Example of a Large Vertical Pump.

The third factor which makes vertical pumps a difficult dynamic problem is that the pump, the motor driver, the foundation (including the soil) and the attached suction and discharge piping are all elements of a total dynamic system. This system must be analyzed as a unit before a rational approach can be taken to designing and understanding vertical pumps. This system approach is further complicated by the fact that, in general, each major dynamic element of the system is designed independently of one another and the first time they

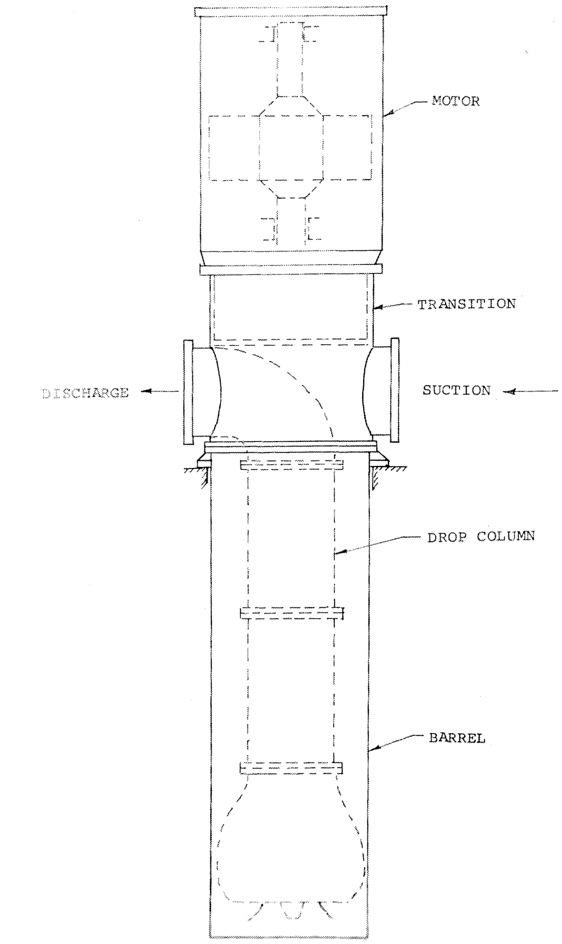

Figure 2. *Schematic of a Vertical Pump.*

merge into a total unit is during installation and commissioning of the plant. At that point it is frequently discovered that the unit is operating on or near a system resonance and the vibration amplitudes are excessive.

This paper presents an approach which can be taken to analyze vertical pump dynamics and investigates the effects of several factors which should be considered in order to accurately produce a mathematical model of the system. It also describes the instrumentation and testing techniques which have proven useful in diagnosing problems and points out areas where further work is needed. Finally, several case histories are presented to demonstrate the wide variety of problems which can arise from the apparently simple structure towering above the ground.

ANALYSIS OF A 3000 HP PUMP

In 1978, the Arabian American Oil Company added an increment of shipping capacity to its Ju'aymah oil terminal on the Arabian Gulf. A part of this shipping system consisted of two 3000 hp, 880 rpm vertical booster pumps. These units pumped crude oil from storage tanks to the shipper pumps used to load tankers and were of an identical design to four other units previously installed except that the two new pumps were mounted on a common pile foundation.

During the initial testing of the motors, prior to the introduction of crude to the system, it was found that the pumps were operating near or on a resonant frequency with the result that the vibration levels were excessive. Over 8 mils peak to peak displacement was measured at the top of the motor. Spectrum traces of the vibration and the peak amplitude response during a shutdown are shown in Figure 3.

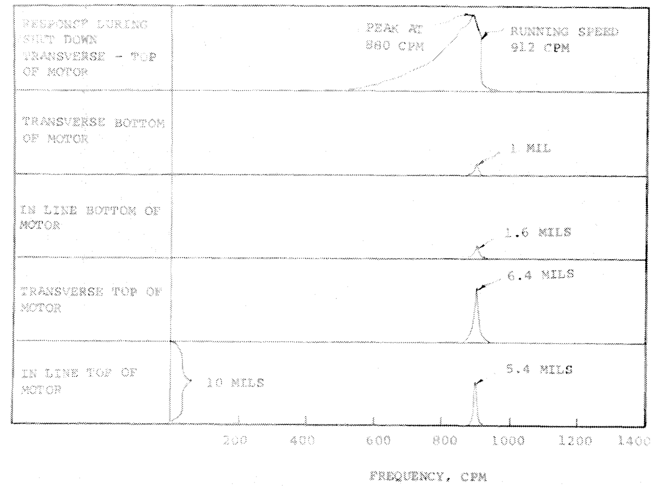

Figure 3. Vibration Spectrum of 3000 hp Pump and Response Curve During Coastdown.

Because the four sister machines had not experienced any resonant frequency problems, it was requested that the pump manufacturer analyze the dynamic system in detail to determine why these units were experiencing problems. The manufacturer agreed and put together a comprehensive finite element model of the pump system which included the piping.

MODELING THE SYSTEM

The initial model, which consisted only of a single pumping unit and infinitely rigid supports for the foundation, predicted the problem to be a lateral rocking mode at 1185 cpm. From the field data this seemed to be in error by about 32%. Although this model might be the approach a manufacturer would normally take in analyzing his machine, since in most cases he does not have access to the detail foundation design and soil properties at the time the pump is designed, it was shown in this case that the accuracy of the calculation was unacceptable. To improve the calculation, it was decided that a total system approach was required in which both pumps, the foundation, the soil and the piping should all be modeled. This model, shown in simplified form in Figure 4, was found to produce accurate results. In this model the process fluid, crude oil, was also included because it contributes significant mass to the system and because it also weakly couples the drop column modes to the modes of the barrel.

Although including the second pump into the mathematical model increased the accuracy of the analysis, it also essentially doubled the number of modes. The model showed that with this more realistic system, the pumps had both symmetrical and asymmetrical modes. For example, instead of a single cantilever type mode for a single pump, the true situation resulted in two possible modes: one in which the two pumps moved in the same direction and another in which they moved in the opposite direction to each other. Because of these factors, the model showed that up to a frequency of twice the running speed, the region of primary interest, the total

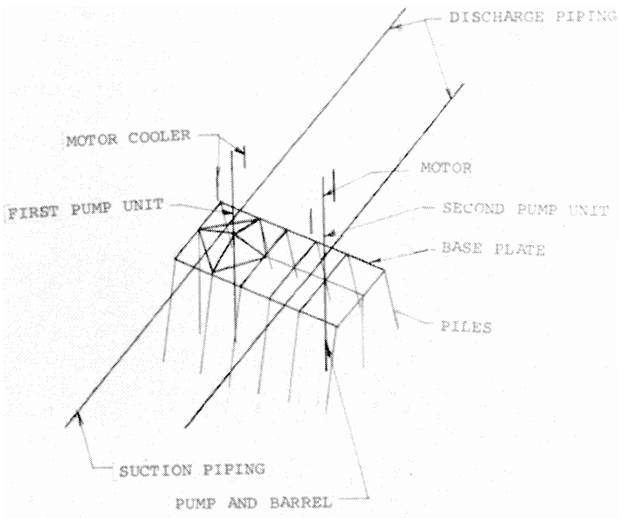

Figure 4. Finite Element Model of a Two Pump System.

dynamic system could have as many as twenty different resonant frequencies. Unfortunately, several of the more easily excited resonances, the inline and transverse first bending modes, seemed to cluster near the running speed.

Since these modes were within 10% of the running speed of the unit, several mathematical perturbations were made on the system to attempt to separate the modes further from the running speed excitation. These included adding mass to the foundation, changing the flexibility of the pump section below the motor, and introducing flexible elements at the pump feet. Several of these perturbations are shown in the frequency matrix in Figure 5.

A closer look at case S, the actual pump configuration filled with crude, is given in Table 1. This table shows that up

TABLE 1 - Computed Natural Frequencies for Case S, The Actual Pump, Filled with Crude

1 FREQUENCY	2 DIRECT	3 TYPE	4 2ND PUMP	5 TOP
74	T	-	Antis	YES
156	IL	Rock	Sym	YES
295	IL	-	Antis	YES
431	T	Pump	Sym	NO
432	T	Pump	Antis	NO
433	IL	Pump	Sym	NO
433	IL	Pump	Antis	NO
805	IL	Rock	Sym	YES
975	IL	Rock	Antis	YES
984	T	Rock	Antis	YES
994	T	Rock	Sym	YES
1043	TORS	-	Sym	NO
1044	TORS	-	Antis	NO

Column - 1: Significant frequencies (C.P.M) under 1200

Column - 2: In line (IL), Transverse (T), Torsion (TORS), Mixed (M)

Column - 3: Type of vibration: Inner part of pump (Pump) / General rocking (Rock)

Column - 4: Behaviour of second pump: Mirror symmetry (Sym) / Mirror antisymmetry (Antis)

Column - 5: Top of motor moving (YES or not (NO)

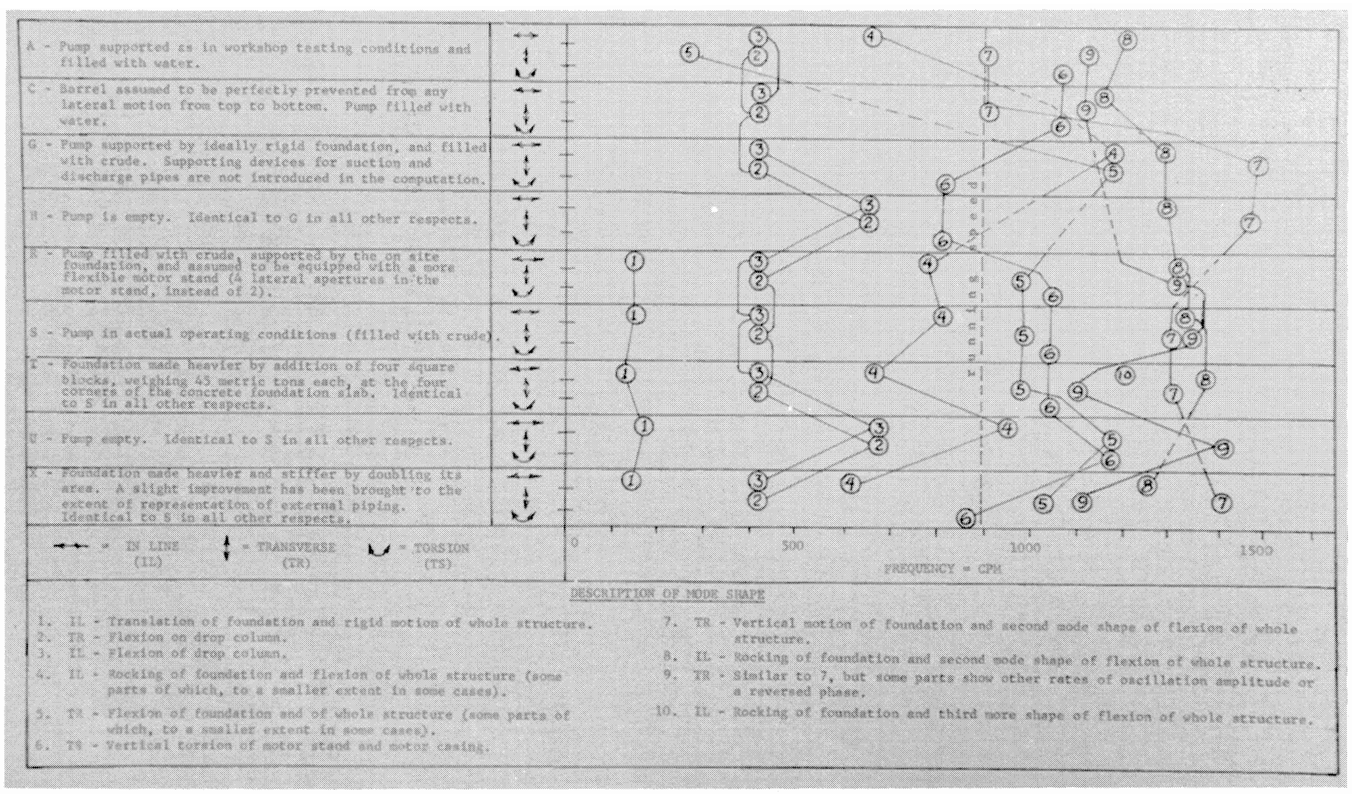

Figure 5. Frequency Matrix Showing System Responses for Several Cases Analyzed.

to 1200 cpm there are 13 significant natural frequencies. Several of these frequencies are very close together being symmetric and asymmetric modes of the two pumps. In column 5, it is indicated that in only seven of the thirteen modes is the top of the motor moving. These are generally rocking modes of the structure, both inline with the piping and transverse to the piping. For the other modes shown, the motor does not move significantly and thus malfunctions which excite these modes might be difficult to detect with normal instrumentation installed on the unit above ground. This could include oil whirl in the pump bearings or flow induced vibration.

SHOP TEST COMPARED WITH FIELD TEST

Comparing case A, a model of the workshop testing conditions, with case S, the pump in the actual operating condition, points out the futility of using a shop test to determine vibration characteristics expected in the field. The two modes of concern in the field installation, an inline rocking of the entire structure at 805 cpm, and a transverse flexure of the foundation and structure at 994 cpm, are seen to occur at 657 and 265 cpm, respectively, on the shop test.

For the shop test, case A in Figure 5, a vertical motion of the foundation combined with the second mode of flexure of the structure is seen to occur within 3% of the running speed or 910 cpm. In the actual field installation this mode has shifted to 1307 cpm and consequently is of no concern. This great lack of agreement between the field and the shop test, due primarily to differences in mounting, points out the problems of accepting equipment based solely on a shop test vibration criterion.

EFFECTS OF FLUID IN THE SYSTEM

It is of interest to consider the effects of having the pumped fluid in the system. As mentioned previously, the initial field tests were made running the motor uncoupled with no crude in the system. A comparison of case U, no crude, with case S, a full system, shows that the additional mass of fluid lowered a mode of primary concern, number 4 in Figure 5, from 960 cpm to 805 cpm, traversing the running speed of 880 cpm. Thus, it is seen that to accurately predict the frequencies of interest, the properties of the pumped fluid must also be considered.

PIPING EFFECTS

The effects of the piping on the various modes were also investigated. It was found that the piping primarily affected the torsional modes in this system and had little impact on the transverse modes near the running speed. However, this does not imply that the piping and its supports might not be significant for a different pump design.

Because many of the designs which were analyzed would require a major effort in time and expense to implement to the already existing system, it was decided to leave the system as designed and to operate it for a period of time. By trim balancing the motor and paying careful attention to the alignment of the motor to the pump, the vibration levels on the units were reduced to acceptable levels of 2 mils.

COMPARISON BETWEEN MODEL AND PUMP

As a mathematical model is being developed, it is highly desirable that correlation between the model and actual field data be made. In this way, the model can be checked to ascertain its accuracy, and once the model shows agreement with reality, one can proceed to perturbate the design with some assurance in the resulting answers. For the 3000 hp crude booster pump, a wide variety of vibration tests were conducted and the results fed back to the pump vendor as the model was being evolved.

RESPONSE TESTS

To establish the resonant frequencies which were of concern to us, several response tests were conducted. In this case three methods were used to excite the structure: impacting, pump rundown, and wind shaker. Although impact tests and rundown response are well known techniques and won't be discussed here, the wind shaker may need some elaboration. In this test, the random turbulant force of the wind is used to excite the structure. By mounting a velocity pickup on the top of the motor and amplifying the output with a gain of up to 20,000, the response of the pump could be measured. This technique was considerably easier than mounting an electromagnetic exciter as is sometimes used. Typical results of the wind shaker are shown in Figure 6 with the pickup mounted inline. The major resonance shown at 800 cpm compares within one percent of the 805 cpm inline mode shown in case S of Figure 5.

MODE SHAPE MEASUREMENTS

Mode shape measurements on the units were also made so that the measured response peaks could be identified with the

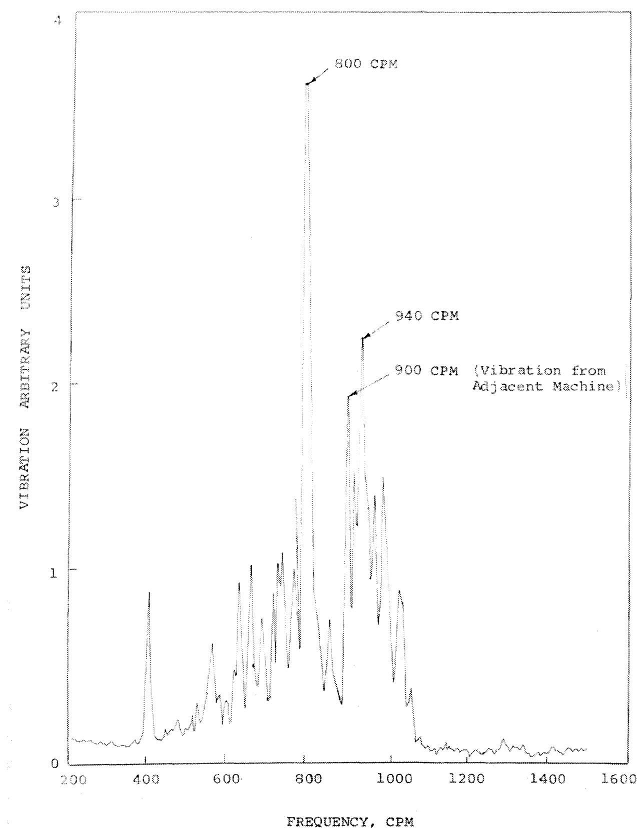

Figure 6. Response of 3000 hp Pump Using Wind as the Excitation.

calculated resonances. A comparison of the mode shape of the structure while running, and the calculated mode is shown in Figure 7. This plot also illustrates one of the primary benefits of having a model of the system: it gives an indication of what is happening to the pump below the surface. The plot shows that for this particular mode, there is significant motion of the barrel and drop column as well as motion of the motor. One can thus conclude that a measured high vibration on the motor implies that the vibration is high below the surface as well.

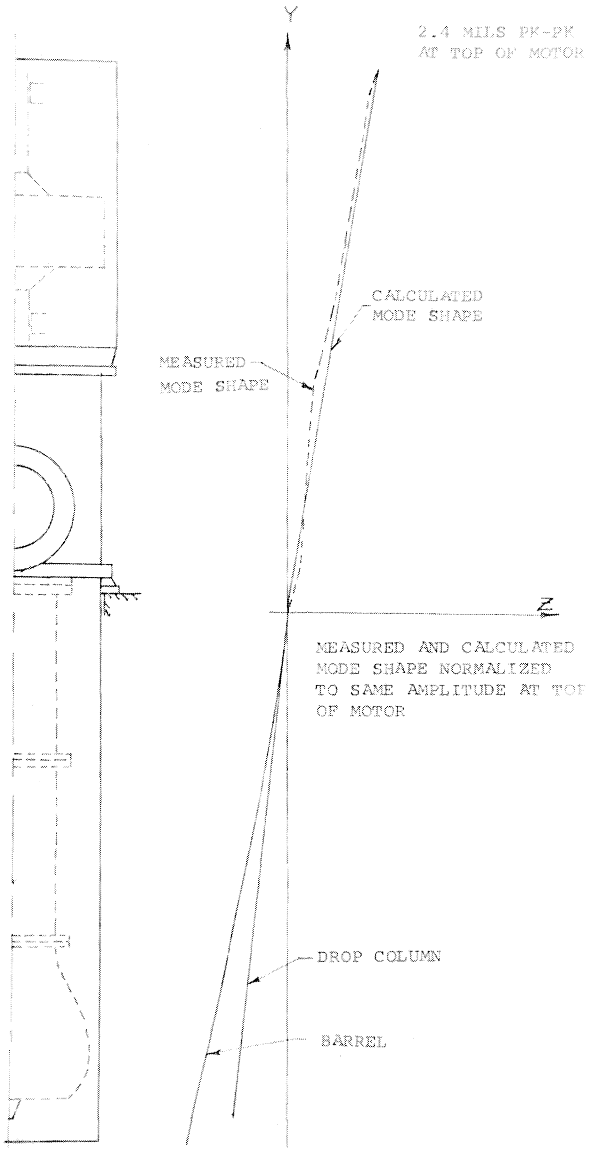

Figure 7. Comparison of Experimental and Analytical Mode Shape for a Transverse Mode Near Running Speed.

INSTRUMENTATION

Measuring the vibration of a vertical pump and motor can present problems which are somewhat different from those of horizontal machines.

Frequency Range

Many of the larger vertical pumps operate at a much slower speed than is typical for horizontal units. For example, a large lift pump for a power plant's cooling water intake might operate in the 400 to 600 rpm range. This speed is well below the linear operating range for most common velocity transducers and thus will introduce a significant measurement error if not considered.

Accelerometers have an advantage over velocity transducers in that their response is flat to very low frequencies. However, care must be taken in selecting the proper sensitivity. An accelerometer whose output is adequate for measuring gear mesh vibration at 2000 Hz may well lack the required sensitivity to measure the structural response of a pump at 5 Hz. A low signal to noise ratio can be a particular problem if the data is tape recorded for analysis later. Since in most cases of a vertical pump the displacement of the structure is of primary interest, the acceleration signal must be integrated twice. This integration can greatly accentuate the low frequency rumble common to many tape recorders and will introduce significant error to acceleration signals. For the low frequency measurement common to large vertical pumps, it is essential that either a high sensitivity accelerometer be used or the signal must be integrated prior to tape recording.

Transducer Location

On a vertical pump, locating a vibration transducer for monitoring purposes is governed by three factors:

1. Accessibility.
2. Sensitivity.
3. Economics.

Ideally, a vibration monitoring system for a large vertical pump would consist of the following:

1. Transducers mounted downhole on the drop column measuring the vibration of the shaft/impellers to detect bearing, wear-ring, or impeller failures.
2. Velocity or accelerometer mounted on the motor structure to detect motor failure or deterioration in balance.
3. Accelerometers or proximity probes, depending on the bearing type, mounted internally on the motor bearings to detect bearing failure.

However, in the real world many of these measurement points are not available. Because of factors such as fluid compatibility, safety, reliability and accessibility, it is generally not practical to install transducers downhole and thus, half of the unit is inaccessable for monitoring. We are therefore left with installing the transducer on the pump/motor column above the ground.

If sleeve bearings are used in the motor, proximity probes measuring the shaft motion relative to the structure can be used. These probes have the advantage of their excellent low frequency response which makes them ideal for the low pump speeds often encountered. The more common motor designs, however, use antifriction bearings for the rotor support and proximity probes for these systems are not suitable due to the lack of relative motion between the bearing and the shaft. For the antifriction bearings design, a velocity pickup mounted on the top of the motor has been found to be the most reliable and sensitive monitoring system. Generally, because of economics, only one pickup is used. Ideally two should be installed to measure both in-line and transverse motion.

There are two schools of thought as to whether to mount the pickup low, near the seal and coupling, or high, on top of the motor. Our standard monitoring system has a velocity pickup mounted on top of the motor flange. This location was chosen from an analysis of the mode shape of the structure which causes most of the vibration problems, a lateral, first bending or rocking mode. In this mode, the seal and coupling area is frequently near a node with little or no vibration. Thus a problem might well go undetected if measurements were made in the lower location.

On top of the motor, on the other hand, the displacements are generally higher and consequently the vibration problems easier to detect. Moreover, for several problem modes, the top of the pumping unit is frequently the mirror of the pump bottom and thus gives an indication of problems occurring downhole.

VIBRATION CRITERIA

At the present time, vibration limits for large vertical pumps are not well established in the petroleum industry. Historically, most of the vibration limits commonly used were derived from data on horizontal pumps operating at relatively high speeds. These criteria, such as the original Rathbone chart, are concerned primarily with velocity measurements made on the bearing caps. However, at low frequencies, velocity, which is related to energy, and acceleration, which is related to force, tend to become insignificant and displacement, relating to stress, becomes the parameter of interest.

Although the Hydraulics Institute's criteria for acceptable vibration of vertical pumps is based upon displacement, it is seen in Figure 8 that, at low frequencies, this criteria would allow amplitudes of up to 7 mils. Our experience indicates that these levels are excessive and allow little margin for deterioration in service. We have found that on machines with amplitudes of 10 to 15 mils, structural damage has occurred due to fatigue failures of the spider assembly supporting the drop column. Based on the premise that if vibration of the order of 10 mils measured on the top of the motor can cause damage, then, to be conservative, the maximum levels should be no more than half of this, or 5 mils. The reasoning led to our adoption of the following vibration criteria:

—For a pump speed greater than 600 rpm, the maximum amplitude, d_{max}, peak to peak mils is given by

$$d_{max} = \sqrt{\frac{15000}{RPM}} \text{ mils, peak to peak} \tag{1}$$

— For pump speeds less than or equal to 600 rpm, the maximum allowable amplitude is:

$$d_{max} = 5 \text{ mils, peak to peak} \tag{2}$$

This criteria is compared in Figure 8 with the Hydraulic Institute's criteria. As seen, the criteria presented here is somewhat more stringent than the Hydraulic Institute's. However, it has been found that with proper attention to alignment and with reasonable care in balancing the motor, these vibration levels can generally be met and will result in problem-free service.

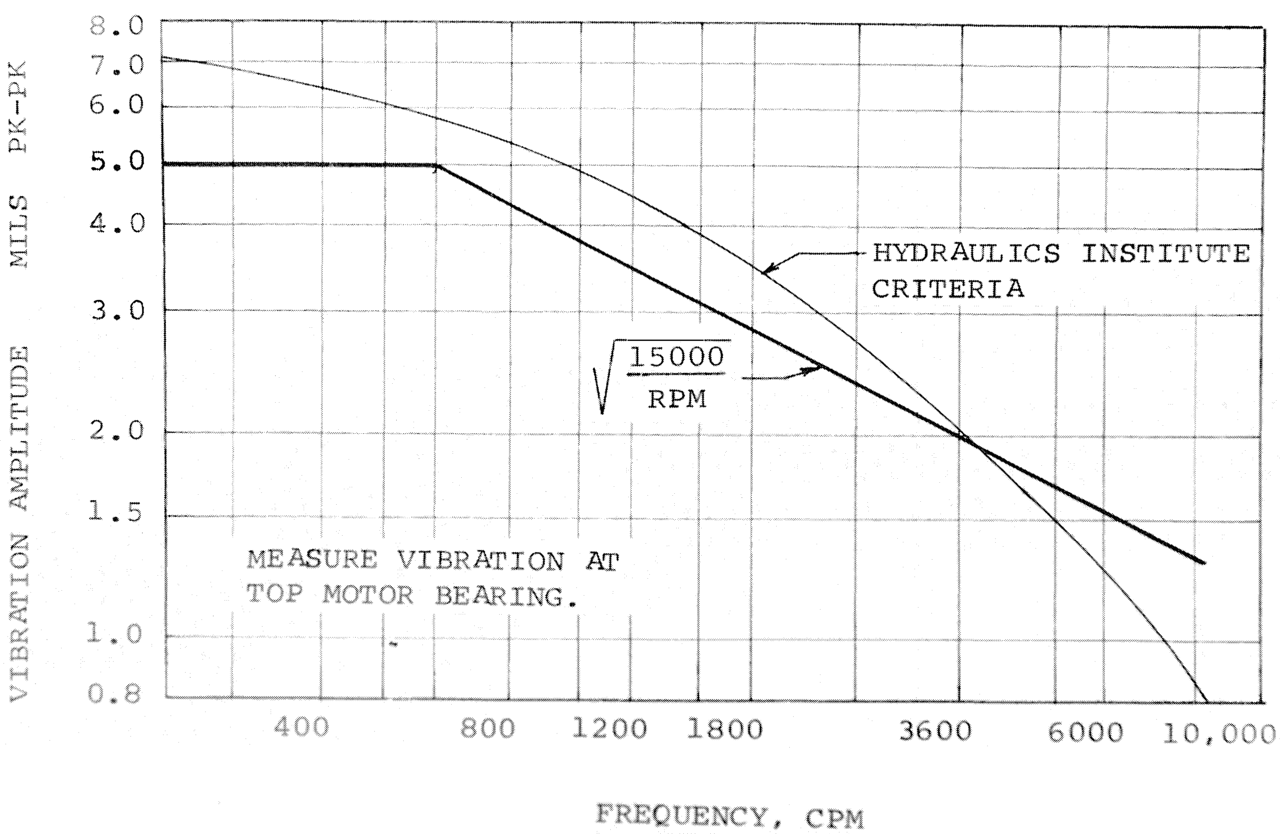

Figure 8. Comparison of Vibration Criteria for Vertical Pumps.

CASE HISTORIES

For examples where vertical pump installations are not trouble-free and had significant vibration problems, several case histories are presented. The cases selected are typical of some of the atypical problems one experiences when dealing with vertical pumps.

Case 1: The Galloping Propane Pump

This problem is a classical case of the resonance problem discussed earlier. A 900 hp, 1200 rpm vertical pump in propane loading service had experienced high vibration of up to 15 mils since installation. After a history of seal failures and fatigue failures on the spider, the problem was diagnosed as a resonance located near the running speed. Although efforts were made to detune the resonance by stiffening the structure by welding braces to the body of the pump, this was only marginally effective and resulted in shifting the resonance by only a few percent.

The problem was finally solved by eliminating the forcing functions rather than by shifting the resonance. By very careful attention to the alignment of the motor to the pump and by performing a single plane trim balance on the motor, the vibration levels were reduced to under 3 mils as measured on the top of the motor.

This case illustrates that the most successful tactic to use on intractable resonance problems, (i.e., those which would require a major pump redesign) is to reduce the common forcing functions of unbalance and alignment.

Case 2: Whose Fault Is It

In this case, a large 2250 hp, 393 rpm, 70,000 gpm seawater intake pump was being commissioned, and was experiencing vibration levels of 14 mils at the running speed. A difference of opinion arose between the pump vendor and the motor vendor as to where the vibration was originating. Although the vibration reduced significantly when the motor was run uncoupled, a load related electrical fault could not be ruled out.

The problem was isolated by measuring the vibration phase relative to the motor shaft and then rotating the pump shaft 180 degrees. When the unit was then run, the vibration was also found to have shifted 180 degrees. The problem was thus diagnosed as originating in the pump and the major effort of pulling the pump could be justified. Inspection of the pump shaft found a 15 mil bow in the shaft. After correction of the shaft the vibration levels were reduced to normal.

Case 3: Unstable At Any Speed

Bearing oil whirl is generally thought to be a high speed malady associated with compressors and turbines. However, during the commissioning of a 1800 rpm, 2250 hp heater drain pump in a nuclear power plant, the pump developed a high vibration of over 15 mils. A spectrum analysis of the vibration indicated that the predominant vibration was at 43% of the running speed or 760 rpm. When the unit was started cold, the low frequency vibration peak did not appear until several minutes after the pump reached speed. Further, by running the motor uncoupled from the pump, the vibration was found to be associated with the motor. This test also eliminated the possibility that flow induced turbulence exciting a resonance was responsible for the problem. An examination of the motor drawings indicated that the top bearing was a simple sleeve bearing with two axial grooves. Because a vertical rotor has very little, if any, side loading due to gravity, it was surmised that the bearing was operating at a very high Sommerfeld number and was thus unstable. The fault was diagnosed as oil whirl and a pressure dam bearing was recommended which would impose a side load on the shaft. Instead, the axial grooves were rounded so that there was a smooth transition between the groove and bearing. After this modification was made the vibration levels were reduced to acceptable limits.

Case 4: A Small Tornado In The Suction

On a 400 mW steam power plant, the condenser cooling water was pulled from a seawater sump by four 2000 hp, 396 rpm vertical pumps. During commissioning it was found that the pumps were vibrating at 9 mils, predominantly at the running speed. The maximum vibration seemed to occur when a surface vortex was formed which sucked air into the suction of the pump. The pump vendor initially thought the two phase flow was exciting a structural resonance of the unit and several attempts to detune or shift the resonances were unsuccessful. At the same time a hydraulic model study of the sump was initiated to eliminate the vortex.

A shaker test of the pump structure, shown in Figure 9, indicated that the resonances of the unit were sufficiently removed from the running speed frequency and indeed the structure seemed to be optimally tuned with an impedance maxima almost exactly at running speed. The results of the hydraulic model of the sump indicated that the vortex could be eliminated by

1. a baffle of the surface of the water;
2. extending the pump suction to within 2 feet of the bottom of the sump;

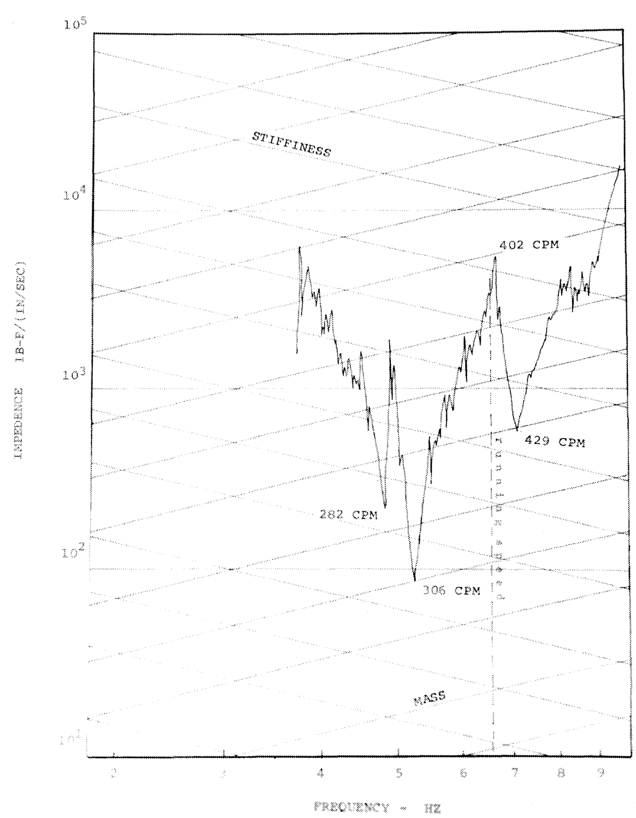

Figure 9. Impedence Plot of a Large Vertical Pump.

3. installing radius blocks between the side walls and bottom of the sump; and
4. installing a cone under the suction of the pump.

With these modifications to the system, the vortex was eliminated and the vibration levels were reduced to under 2 mils.

Case 5: The Long Slow Beat

In making routine vibration measurements on the four 2250 hp, 393 rpm lift pumps previously discussed in Case 2, a difference of opinion on the vibration level developed between the central engineering vibration group and the local maintenance engineers. One group measured over 6 mils on one of the machines while the other group reported the vibration to be under 2 mils. After the usual discussion as to whose instruments were properly calibrated, it was found that both measurements were correct. It was found that the vibration varied periodically from a minimum of 1.5 mils to a maximum of 6 mils over a period of 18 minutes. The two groups had simply taken a quick measurement at different parts of the cycle and the variation was too slow to be readily detected. It was determined that the vibration of two or more machines were interacting due to their being mounted on a common foundation. Because the units had a very slow speed to begin with and were operating at very close to the same load, the speed of the units were very close to each other but not exactly the same. With two units running, their speed was within six percent of each other, producing a beat period of 18 minutes.

This example points out the need to mount vertical pumps on separate foundations, if possible, to eliminate pump interactions. It also demonstrates, on very slow speed, multiple machine foundations, the need to measure the vibration for an extended period of time.

Case 6: Weaker Is Better

Although this case pertains to a smaller vertical machine than has generally been discussed in this paper, it illustrates the principle that the way to correct a vibration problem is not always to "beef" it up. This 40 hp motor driven mixer was found to be vibrating at excessive levels primarily in the transverse direction at a blade passing frequency of 1550 cpm. This transverse vibration was 14 mils compared to 8 mils in the in-line direction. Because of the disparity in vibration, it was concluded that the problem was not simply unbalance. An impact test on the unit, measuring the response in both the in-line and transverse direction, quickly showed that the vibration was caused by a transverse cantilever mode at 1530 cpm or within 3% of the blade frequency.

In this case, rather than try to stiffen the structure, as is frequently attempted to detune the resonance, it was decided to remove gusset braces from the sides of the motor support. This had the effect of decreasing the transverse stiffness and lowering the resonant frequency to 1322 cpm. With the mode thus detuned from the running speed excitation, the vibration dropped to 6 mils.

CONCLUDING REMARKS

From the analysis and examples presented in this paper, several general conclusions can be drawn:

1. A vertical pump and motor is a complex dynamic system which does not lend itself to simple modeling. To accurately predict resonant frequencies, many elements must be considered which include the pump, motor, piping, foundation and soil. It is felt that this total analysis should be the responsibility of the pump vendor.
2. Shop tests of a vertical pump bear little resemblence to the actual field installation and will give no guarantee of satisfactory operation in the field. A shop test will only offer a reasonable assurance that a unit will be balanced well enough to run long enough to solve any problems in the field.
3. Available vibration monitoring instrumentation is not entirely satisfactory. Further development is needed by the industry to solve the problems of measuring what is happening below the surface.
4. It is further felt that improvements are needed in industry acceptability criteria for vertical machines. This includes establishing amplitude limits which will assure trouble-free operation and a better definition as to where these measurements should be made.

ACKNOWLEDGEMENT

The author wishes to thank the ACEC Pump Company of Belgium for permission to use the analysis presented in this paper and for the assistance provided on several of the figures. He also wishes to thank the members of the Dynamic Analysis Group of Aramco for supplying much of the data and many of the solutions for the case histories described.