ANALYTICAL AND EXPERIMENTAL TECHNIQUES FOR SOLVING PUMP STRUCTURAL RESONANCE PROBLEMS

by

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ABSTRACT

Proper use of current analytical and experimental techniques is essential to finding and solving structural resonance problems in pumps. Available techniques vary from the relatively simple to the very complex. Accuracy of the analysis depends on the techniques used and how they are applied to the particular application. Various types of analytical and experimental techniques are discussed with guidelines on the proper use, and limitations, of these techniques. Three case studies of different pump resonance problems are presented as examples of how these techniques have been successfully applied.

INTRODUCTION

The analytical techniques for determining the existence of pump structural resonance vary from hand calculations to detailed finite-element modelling. For analyzing beams, plates, or simple piping, a hand calculation may be sufficient. These hand calculations used in industry are essentially the basic beam deflection equations modified by various dynamic and empirical coefficients [1]. This type of analysis can be easily performed by the equipment designer since minimal technical training is required. As the structure becomes more complex, a finite-element analysis may be used to accurately predict the vibratory behavior. These finite-element modelling techniques require a specialist trained in the latest computer programs. Since most pump/motor systems and pump/driver baseplates are complex structures, the analytical technique gaining the most acceptance is finite-element modelling. Analytical techniques for evaluating dynamic components, such as pump rotors, include rotor dynamic computer programs, written specifically for this purpose.

Choice of an experimental technique depends on the type of information needed to analyze the structure. The simplest and easiest experimental technique is measuring the natural or resonant frequencies of the part or assembly in question. This measurement is often performed initially to determine if the vibration problem is caused by a pump resonance or is related to unbalance or misalignment of the rotor. Modal analysis, a more involved experimental technique, shows how a given part is deforming at each of its resonant frequencies. Modal analysis is crucial in determining where modifications are needed to correct resonance problems.

The first case study involved high vibration in a vertical circulating-water pump installation. A frequency test and finite-element analysis were performed, and confirmed that the pump/motor system had a resonance near its operating speed. The finite-element analysis gave direction in showing how to modify the system, to move the natural frequency away from its running speed.

The second case study concerned the resonance of bedplate used with a multistage, horizontally split, volute pump on a crude-oil pipeline. The pump was driven by a diesel engine through a speed-increasing gear. Initial performance testing revealed high vibration levels at one of the pump's three operating speeds. A modal analysis using impulse techniques as described by Halvorsen and Brown [2] was performed on the bedplate and several resonances were found in the beams supporting the pump pedestals. The resonances created high relative motion at the pump mounting. The solution was to add braces to the bedplate beams under the pump.

The third case study involved excessive shaft vibration in multistage boiler-feed pump, during initial operation. Factory testing of the pump alone showed no vibration problems. However, a frequency test performed on the installed rotor demonstrated rotor resonances near the pump running speed. A rotordynamic computer analysis confirmed the frequency test, indicating significant rotor movement at the coupling. To solve the resonance problem, the rotor natural frequency was shifted by changing the coupling design.

All three case studies show the effectiveness of analytical and experimental techniques in solving pump resonance problems. The best results are usually obtained when both analytical and experimental techniques are used together to verify the problem and determine the best solution. Additional care must be taken in selecting the correct technique and applying it properly to obtain accurate and effective results.

An improperly produced finite-element model, or one with the wrong boundary conditions, will lead to the wrong conclusions. A modal test with the measured points in the wrong location, or testing in the wrong frequency range, will give misleading results. Performing a finite-element analysis to determine how the modal test should be set up and then taking modal test data to "fine tune" the finite-element model, affords the best change of solving pump resonance problems.

CASE I

Background

Several large, vertical, wet-pit (direct induction motor drive) circulating water pumps were installed in an overseas industrial plant. During the plant start-up, some of the units experienced vibration displacements that exceeded the contractual limits. A team of vibration specialists was dispatched to the plant site to determine the cause of the vibration. The on-site testing consisted of single plane balancing of the drive motor, impact testing of the unit at various water levels, and accurate vibration measurement on each unit. A typical cross-section of a vertical pump is shown in Figure 1.

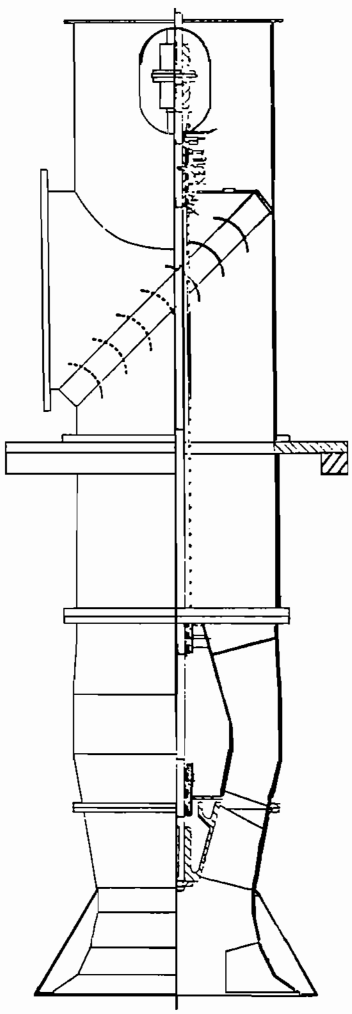

Figure 1. Vertical Pump Sectional Assembly.

The site testing showed that the vibration was a result of structural resonance modes occurring close to the unit rotating speed (6.6 Hz). A three-dimensional finite-element analysis was chosen as the best method of analyzing the machines. The analysis has the following objectives:

• Determine whether a finite-element model could be constructed that would accurately calculate resonant frequencies of the unit, as measured in the field.

• Determine what degree of model refinement would be required to obtain necessary accuracy.

• Determine whether a single finite-element model would properly describe measured field frequencies for 'dry" and "wet" conditions.

• Having validated a model, use it to confidently predict the effect that proposed structural modifications would have on unit resonance frequencies.

• The design goal was to have no wet resonances within 30 percent of 6.6 Hz, i.e., 4.63 to 8.59 Hz.

Analysis

A 3-D finite-element representation of the unit was required to obtain accurate modal frequency results for the following reasons:

• Difficulty in modelling one-dimensional spring equivalents for the mounting-plate trunnion effect.

• Difficulty in formulating a beam element that properly represents the asymmetrical nature of the discharge head/nozzle cut-out stiffness properties.

• Inability of a beam element model to simulate the true deflected shape of the segmented support head.

• A basic concern as to whether or not a beam model is an acceptable representation of a structure primarily made of thin-walled cylindrical components.

• Possible significance of "local" geometry effects. The difficulty of properly analyzing a complex dynamic system that involves a vertical pump, drive motor and foundation has been previously reported by Corley [3].

The ANSYS Rev. 4.1A general-purpose finite-element computer program was used throughout this analysis. Primary modelling techniques and assumptions are listed as follows:

• Plate and shell elements were used to obtain 3-D representations of the pump configuration.

• The drive motor was modelled as a lumped mass atop a beam element, having an equivalent stiffness—the combination of which gives the resonance frequency provided by the motor manufacturer.

• No special flexibility is included at bolted joints, because the specified preload forces of the bolts are considerably larger than any operating loads (flanges included as local stiffness).

• The mounting plate is assumed to be fixed to the foundation at the anchor bolt locations.

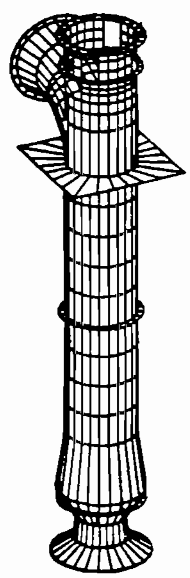

Figure 2. Finite Element Model Construction.

• The "wet" condition was simulated by the addition of mass, distrubuted throughout the wetted components. This was accomplished by increasing the density of pump components to include the mass of the component and the mass of the water.

• The flexible joint at the discharge is not included in the model, because the order of magnitude of stiffness of this joint is considered to be far less than that of the primary structural elements of the pump.

A view of the total finite-element pump model is shown in Figure 2. Modelling details of various pump components and the asymmetrical design of the discharge head and motor support are reflected in Figure 3. The pump rotor is not included in the model, since its contribution to the unit stiffness is quite small.

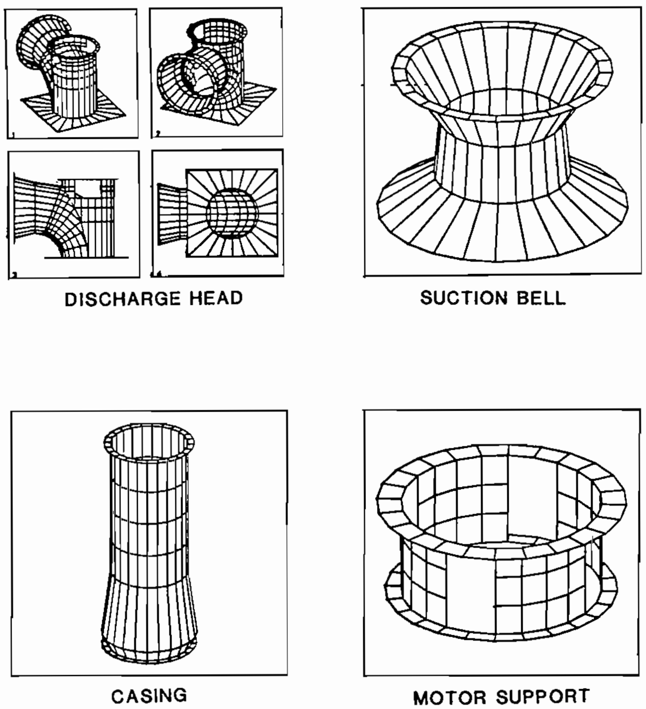

Figure 3. Modelling Details of Pump Components.

Results

The field shaker test data taken on one of the installed units was used as the benchmark for comparison with the ANSYS generated frequencies. This data was recorded with the discharge piping connected and the sump dry. The ANSYS analysis simulates these conditions. Once the dry correlation was made, a similar comparison was made with the sump in the watered condition, by adding a percentage of the mass of the entrained water. Comparative data are given in Table 1. The deflected mode shapes are shown in Figure 4 for the unmodified pump.

Table 1. Comparison of Field Measured Values with Computer Model Results.

RESONANCE		DIRECTION	"SHAKER" TEST DRY	"SHAKER" TEST WET	FINITE ELEMENT MODE DRY	FINITE ELEMENT MODE WET
f1	mode shape 1	//	4.58	3.50	4.69	3.54
f2		⊥	4.90	3.70	4.91	3.59
f3	mode shape 2	//	7.72	7.10	8.43	8.94
f4		⊥	8.75	7.85	8.73	7.46

ERROR ANALYSIS FREQUENCY	% ERROR
1 dry	+ .8
2 dry	- .2
3 dry	+12.6
4 dry	+ 2.5
1 wet	+ 1.1
2 wet	- 2.9
3 wet	- 2.3
4 wet	- 2.5

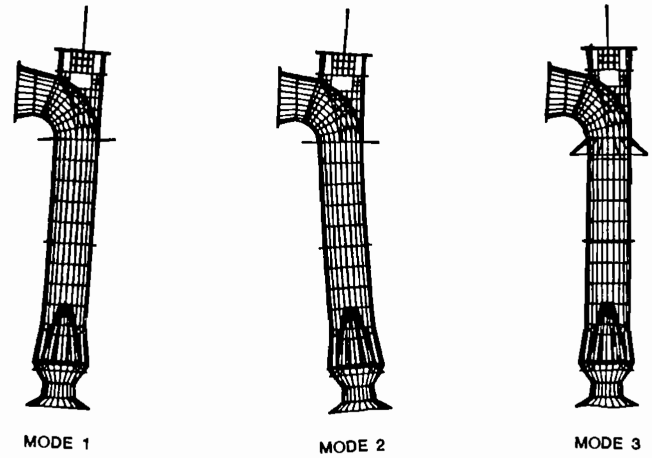

Figure 4. Deflected Mode Shapes.

After the model was developed, numerous design changes were analyzed for their ability to raise the third and fourth frequencies sufficiently above the operating speed and meet contract requirements without having an adverse effect on the first two frequencies. These design variations included approximately 20 iterations with changes to the lower pump column, pump mounting plate, pump discharge head, and motor support head. Since the pump column and discharge head were of cast construction, it was necessary to study the effect of various thicknesses of the castings.

With the aid of the analysis, new parts have been designed and will be used to retrofit the installed units. The following conclusions were established as a result of this analytical study:

• A full 3-D (plate/shell) model of the unit predicted the measured field frequencies, generally within -1 percent of the unmodified pump.

• A 2-D beam element model was not suitable, because of the difficulty in modelling the asymmetric properties of support and discharge head, and mounting-plate trunnion effects.

• The full 3-D model, when modified for the additional mass of entrained water, predicted measured "wet" field frequencies within $+1$ percent to -3 percent.

• The inclusion of the concrete foundation did not significantly alter the frequency results, and was not included in later runs.

• A series of possible structural modifications was analyzed, but no confirming field data are available until an actual modification has been made.

CASE II

Background

During initial testing of a horizontal multistage pump driven by a diesel engine through a sped increaser and mounted on an ungrouted baseplate, unacceptable vertical and horizontal vibrations were measured at one of the diesel engine operating speeds (1050 cpm).

The increased vibration levels at 1050 cpm suggested the possibility of a structural resonance inherent in the baseplate design. Consequently, a baseplate modal analysis was performed to substantiate the baseplate-resonance theory.

Analysis

The modal analysis was initiated by recording vibration levels and frequency spectrums at various locations on the unit while operating at 900 cpm, 1050 cpm, and 1200 cpm engine speeds. These data were used to verify previous information

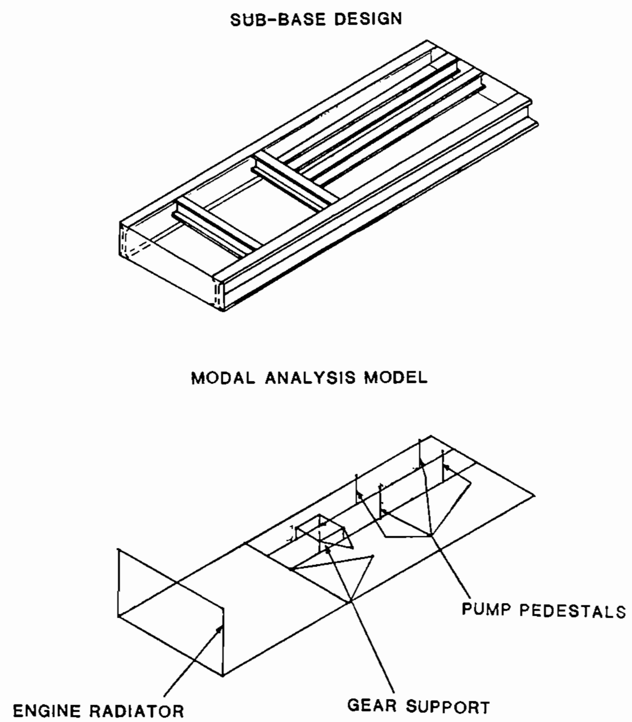

Figure 5. Baseplate Design and Analysis Model.

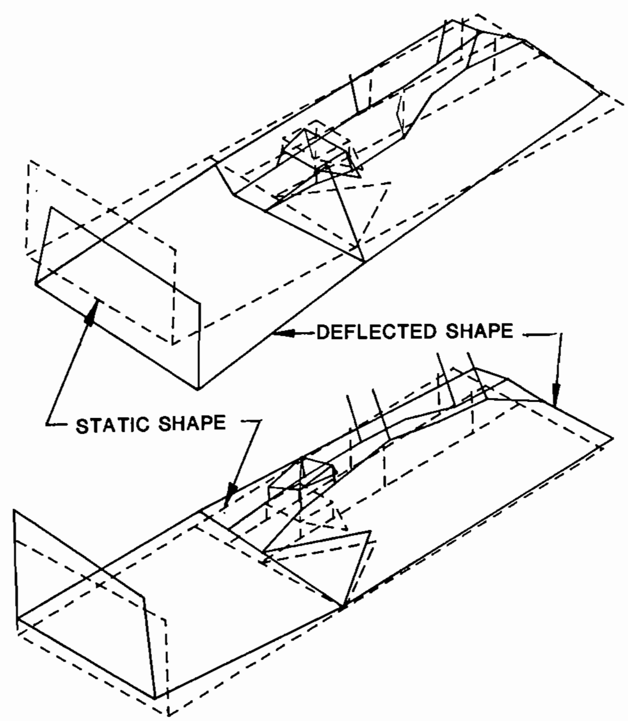

Figure 6. Typical Baseplate Deflected Mode Shapes.

and to determine locations of high vibration for inclusion in the modal model. The frequency spectra highlighted the main frequencies that were excited during operation. A modal model consisting of 34 points was constructed as shown in Figure 5.

The baseplate was then struck with an instrumented hammer, and an accelerometer was used to measure the response at each of the 34 modal points. A Hewlett-Packard 3582A structural dynamics analyzer and a Hewlett-Packard 9816 computer were used during the testing. Data were taken in the three principal directions at each of the modal points. Deflected and animated plots were then produced for each frequency of interest. A listing of the recorded frequencies and their relationship with either the low-speed diesel shaft or the high-speed pump shaft is reflected in Table 2. From these data, it can be observed that modes 5, 6, 7 correspond to the resonant speed range recorded during initial testing. Modes 1, 2, 3 and 11 are also within the operating range and may be contributing to the amplitude recorded at these frequencies. Several deflected mode shapes of the baseplate are shown in Figure 6. An examination of these shapes shows that all of the responses are related to the flexure of the outer beams of the bepplate or the two smaller beams under the pump and gear. Data taken during the performance testing of the pumps supported the high vibration displacements predicted by the modal analysis.

Results

To eliminate the flexure of the two longitudinal beams beneath gearbox and pump, it was evident that additional baseplate crossmemebers were needed at the gear and pump pedestals. Three rectangular box beams were welded into the baseplate under the two pump supports and the gearbox support, respectively. These modifications to the baseplate resulted in a substantial reduction of the vibration amplitudes as measured during the unit performance testing.

These design modifications to the baseplate were decided upon only after considering the ease with which they could be completed, since several of the units had already been shipped to the remote job site.

CASE III

Background

Commissioning tests on a large steam-turbine-driven boiler-feed pump indicated rotor vibration levels in excess of contractual requirements. The vibration indicated either rotor imbalance or operation near a rotor critical speed. Since full-speed shop testing did not exhibit the same high levels of vibration, the flexible coupling and the speed-increasing gear became areas of investigation.

Table 2. Measured Baseplate Modal Responses.

HZ	8.0	15.2	18.4	20.0	24.8	25.6	36.0	52.8	53.6	64.8	82.4	84.0
CPM	480	912	1104	1200	1488	1536	2160	3168	3216	3888	4944	5040
X	.5LS	1LS	1LS	1LS	—	—	2LS	1HS	1HS	4LS	5LS	5LS
LS CPM	960	912	1104	1200	—	—	1080	1046	1062	972	989	1008
MODE #	1	2	3	11	4	—	5	6	7	8	9	10

L.S. CPM = Low Speed Shaft CPM = Engine Speed
1LS = 1 × LS = 1 Times Low Speed Shaft
1HS = 1 × HS = 1 Times High Speed Shaft

Further study included the measurement of pump shaft vibration produced by a steady-state excitation applied to the pump coupling (more commonly referred to as a "shaker" test). The results plotted in Figure 7 reveal three system resonances within the frequency range investigated. Since the pump was not operating during this test, the stiffening effects of the impeller ring fits need not be considered, and the increased vibration amplitude at 56 Hz was recognized as the pump rotor first critical speed.

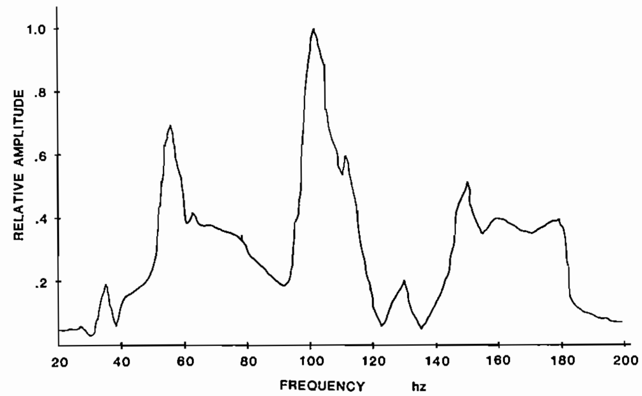

Figure 7. Relative Amplitude of Vibration Measured under Steady State Excitation.

The vibration peaks at 100 Hz and 151 Hz were not as readily identifiable. However, continued testing and calculation revealed the spacer coupling as the source of these system resonances. The system resonance at 100 Hz, combined with verified coupling imbalance, resulted in the increased operating vibration levels listed in Table 3. Vibration levels below 0.001 in peak-to-peak would be considered normal.

Table 3. Site Data From Initial Pump Testing.

	Gearbox Output Shaft	Pump Shaft at Driver End Bearing
Vertical		
Amplitude	0.0019 in P to P	0.00165 in P to P
Phase	286 Degrees	33 Degrees
Horizontal		
Amplitude	0.00235 in P to P	0.003 in P to P
Phase	292 Degrees	306 Degrees

Analysis

In an effort to analytically verify the pump first critical speed and to determine the effects of coupling weight on the rotor natural frequencies, an undamped critical speed program was used to compute rotordynamic data for various support stiffnesses as discussed by Bansal and Kirk [4]. Assuming the high-speed shaft was laterally isolated through the speed-increasing gear, the rotor model represented in Figure 8 was considered sufficient for this computer analysis.

The lateral speed computer program uses a transfer matrix solution technique which is a computerized version of the Myklestad-Prohl method. The rotor model is input by geometry data taken from machine layouts with the impeller weight and gyroscopic properties (calculated via a separate computer pro-

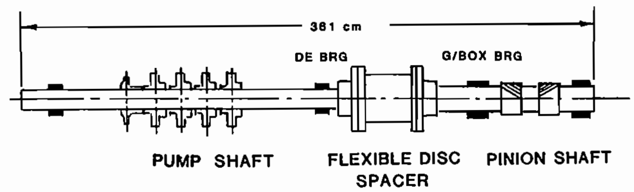

Figure 8. Rotor Model for Computer Analysis.

gram) input by specific external values at the appropriate axial location. The analysis considers shear deformation and gyroscopic moment effects for circular synchronous vibration. In addition to rotor critical speeds, the computer program provides rotor mode shapes coincident with the calculated support stiffness (tabulations in Figure 9). Critical speed data computed for the rotor model with the original coupling design agreed well with the shaker test results.

It became apparent that a solution to the vibration problem would be to decrease the coupling weight, thus increasing the critical speed of the overhung-coupling mode. As indicated in Figure 9, a 60 percent reduction in coupling weight resulted in a 20 percent increase in coupling critical speed. This provided sufficient margin for smooth pump operation and resulted in the ultimate solution to the vibration problem.

ROTOR CRITICAL SPEEDS

VIBRATION MODE	CPLG. 1 330# spacer	CPLG. 2 130# spacer	MODE SHAPE
1	49	50	
2	93	114	
3	147	181	

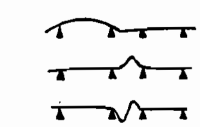

Figure 9. Comparison of Field Measured Values with the Computer Model Results.

To further investigate the effects of coupling imbalance, a forced-response program was utilized to compute damped vibration levels for the two rotor arrangements. The response of the rotor-bearing-seal system to imbalance was calculated by a digital computer program that treats the general case of elliptic response and thus permits the input of the eight stiffness and damping properties calculated by the bearing or seal programs. The features of the program are 1) geometry input, 2) accounting for shear deformation, 3) general linearized gyroscopics, 4) on-line calculation of bearing and seal properties, 5) automated line printer plots of dynamic mode shapes and response versus speed curves, 6) capability to monitor probe response at actual angular position of the probes. The computer program is a modified version of the Lund response program developed in

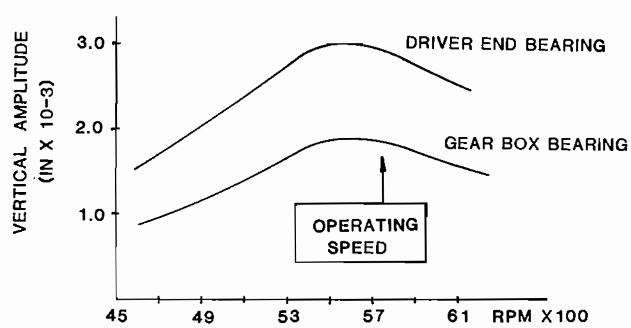

Figure 10. Forced Response Output Amplitude of Vibration at Various Pump Speeds.

the mid-60s [5]. With the heavier coupling and an imbalance of 7.9 oz-in, obtainable by offsetting the coupling spacer within the coupling hub aligning fits, the computer calculated vibration levels were approximately 1.5-2.0 mils.

The forced-response program results, illustrated in Figure 10, provide an indication of vibration levels for various operating speeds. Upon comparing the computed vibration levels with the field data as shown tabulated in Table 4, a fairly close agreement is realized. Also plotted in Table 4 are the forced-response results of the coupling configuration which is presently installed and operating successfully. The maximum imbalance included in the calculation is smaller, due partially to the reduced weight, and partially to a close-tolerance design.

Table 4. Comparison of Results.

Peak	Shaker Test Data	Computer Calculated Critical Speed Original Coupling	% Error
1	56 HZ	49 HZ	−14.3
2	100 HZ	93 HZ	−7.5
3	151 HZ	147 HZ	−2.7

Location	Field Data Vertical Amplitude	Forced Response Computer Original Coupling at Running Speed
Gear Box Output Shaft	0.0019 in	0.00175 in
Pump Driver End Bearing Shaft	0.00165 in	0.0028 in

Results

While the field data do not exactly agree with the computer calculations, a comparison of results included in Table 4 reveals fair agreement and can be considered good for non-laboratory investigation. This type of result demonstrates the ability of analytical techniques to predict trends and isolate problem areas. The disagreement between computer-calculated critical speeds and the "shaker" test results may be explained by the following, difficult-to-model, items:

•Internal rotor contact

•Stiffening effects of shrink fits (impeller, coupling hubs)

•Support stiffness at zero pump speed

The inability to compute the absolute magnitudeof vibration by forced-response programs may be explained as a direct result of the inability to model the actual magnitude of imbalance. In this case, the imbalance was caused by a random assembly technique in which the spacer was positioned by a loose-tolerance centering fit. The centering fit on the replacement coupling was eliminated by utilizing close-clearance bolts. However, by continued efforts to verify theoretical techniques, it is possible to predict, with reasonable accuracy, the rotordynamic performance of rotating equipment can be predicted with reasonable accuracy.

CONCLUSIONS

Three examples have been discussed, where the latest analytical techniques and testing procedures have been used to help solve operational pump problems. In all cases, the original designs were completed using the classical type of design calculations.

At the present time, these new analytical techniques are being used mainly to help solve problems, rather than being a part of normal machine design practice. This approach is a result of the time and work involved in performing the analytical work. The finite-element work involved in the vertical pump example is tremendously complex, consumes considerable computer time, and requires a skilled analyst to perform the work. More work is needed to sufficiently refine and simplify the degree of analysis which is required to make this type of work more cost effective. Management must be convinced that industry must continue to invest its resources in these new analytical and testing techniques.

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