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CRITICAL SPEEDS AND THE IMPORTANCE OF STIFFNESS— A CASE STUDY IN THE DESIGN AND TESTING OF A LARGE MECHANICAL DRIVE STEAM TURBINE TO API STANDARD 612

by

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ABSTRACT

Rotordynamics has long been one of the major considerations in the design of variable speed rotating equipment. Bearing support stiffness is a significant factor in the designer's ability to predict critical speeds and response to unbalance. This case study is used to describe the influence of support stiffness on the rotordynamics performance of a large variable speed mechanical drive steam turbine. This joint effort by the user and machinery vendor identifies several practical issues in meeting the intent of API Standard 612, Third Edition.

Modern analytical techniques can be used to predict bearing support stiffness in both the horizontal and vertical directions. When the stiffness in the two directions are dissimilar, multiple

critical speeds may occur, which complicate the interpretation of API 612 test results. Effects of critical speeds on response to unbalance are discussed. Also presented are some analytical and experimentally determined dynamic characteristics obtained during the design and test.

INTRODUCTION

The steam turbine discussed was one of three purchased to replace existing gas turbine compressor drivers used in a chemical plant. The turbines were required to meet the requirements of the API Standard 612, Third Edition [1].

API 612 Third Edition included significant changes from the previous edition regarding dynamics, both from an analytical perspective and from an actual test basis. These changes represented a belief by the industry as a whole that the state of the art of analytical rotordynamics modelling has progressed to the point that critical responses of rotors can be accurately determined analytically, and are verifiable by test. As a result of this confidence, API Standard 612, for the first time, related the required separation margins to the predicted critical response amplification factor. This confidence was predicated on the ability of the designer to accurately model the rotor and accurately determine required dynamic coefficients [2, 3].

Three identical units were built. One turbine drives two compressors in tandem, one drives a single compressor and one is a common spare. The response to unbalance verification test was performed on the spare unit. During response to unbalance testing, the second critical speed of the unit was observed to be within the speed range prohibited by the specification. Further testing identified flexibility of the low pressure bearing support as the cause of this unexpected result. Through analysis and testing, a suitable modification to the bearing support was designed and installed. Both the analysis and test results demonstrated that the sensitivity to unbalance at the bearing oil film was fairly independent of support stiffness and the exact locations of the critical speed.

ORIGINAL DESIGN, ANALYSIS, AND TEST

Design

The outline of the turbine and the support arrangement are shown in Figure 1. The turbine has 14 stages and generates 61000 hp at 4150 rpm. The rotor model, bearing span, and bearing reactions are shown in Figure 2. The rotor is supported by two tilting pad journal bearings having five shoes. The high pressure end journal bearing is solidly mounted on a bearing pedestal which is in turn supported by the base. The low pressure end bearing is supported by the vertical wall of the exhaust casing.

Analysis

As part of the normal design procedure, various types of rotordynamics calculations were performed to verify that the design met the requirements of API Standard 612. One of the required calculations was the response to unbalance. Prior design experience, along with impedance and rotordynamics testing, had established horizontal and vertical support stiffness as 8.0 million lb/in over the entire speed range. Response to both a midspan and an out-of-phase end unbalance were analyzed in separate calculations. With these analyses, the locations of critical speeds and sensitivity to unbalance were determined. For this rotor, the first rotor critical speed (bounce mode) was predicted at 1790 rpm, the second critical speed (rocking mode) at 4825 rpm and the third (bending) at 5990 rpm. The second critical was closest to the operating speed range. The rotor response at the bearing for the out-of-phase unbalances and the rotor mode shape are shown in Figures 3 and 4, respectively.

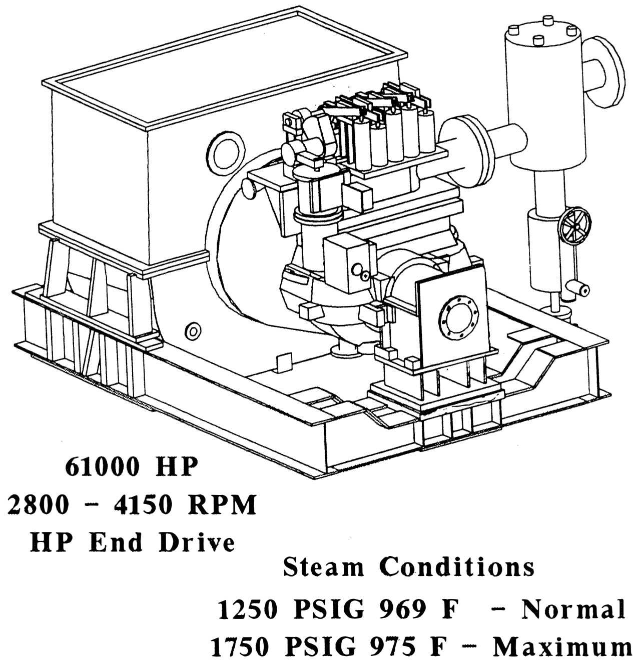

Figure 1. Turbine Outline.

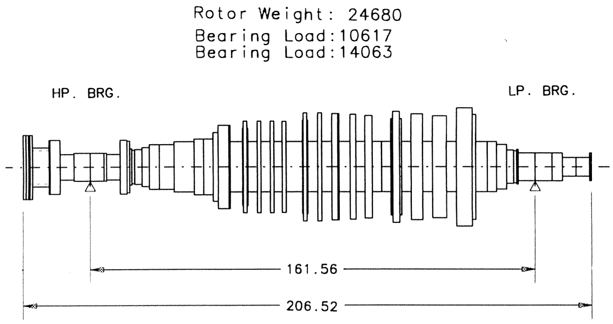

Figure 2. Turbine Rotor Model.

The bearing properties for the response to unbalance calculations are a function of speed. The calculated bearing properties at rated speed are given in Table 1.

Table 1. Dynamic Properties of Bearings.

	Stiffness		*Damping*	
	Vertical *(10^6lb/in)*	*Horizontal* *(10^6lb/in)*	*Vertical* *(10^3lb-sec/in)*	*Horizontal* *(10^3lb-sec/in)*
Drive End (HP)	4.1	2.3	4.5	4.0
Exhaust (LP)	5.2	2.9	4.6	4.0

Test

The third edition of API Standard 612 requires a response to unbalance test to be performed. The purpose of the unbalance test is to verify the accuracy of the analytical model by using an unbalance on the rotor on the test stand. The test response is compared with the predicted response. It was felt that this addition-

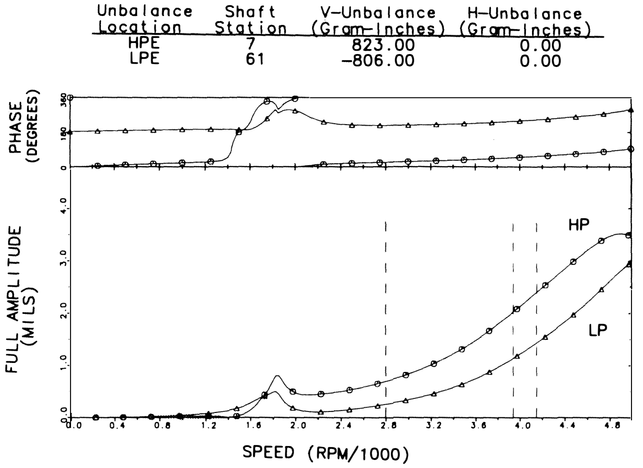

Figure 3. Response to Out-of-Phase Unbalance.

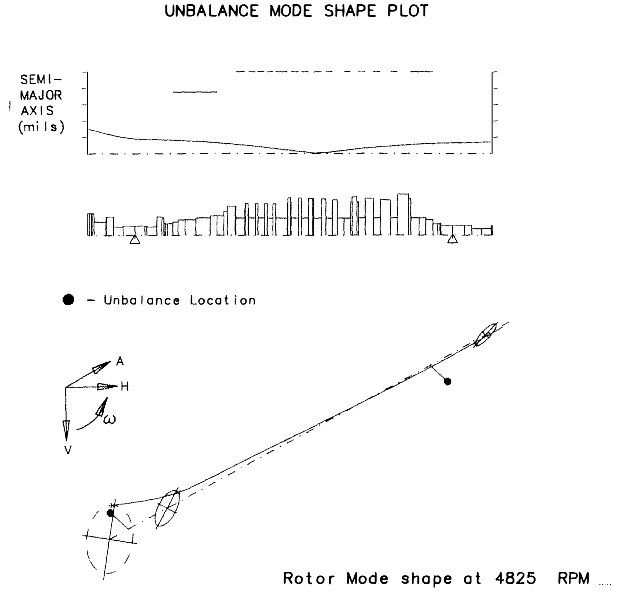

Figure 4. Rotor Deflection Shape at Second Critical.

al testing would provide feedback as to the accuracy of the original model, verifying general credibility of modelling techniques being used by the vendor. This would also reveal rotors which were marginally acceptable, but had their characteristics "masked" by having a well balanced rotor on test.

During the manufacturing cycle, the turbine rotors were assembled into the casings and run through the entire speed range by operating on steam. The first two turbines of this design were built and tested only in this manner. Both rotors were well balanced as indicated by the very low vibration levels shown in Figure 5. The response at the two proximetry probes on the low pressure bearing is shown in this figure as well as the following figures of test data.

The third turbine was successfully tested with unbalance weights added to the midspan of the rotor in order to excite the first critical speed. The test for the second critical speed had unbalance added to both ends of the rotor at opposite angular positions in order to stimulate the second critical speed. With these weights installed, unexpected response peaks in the range of 3800 rpm to 4500 rpm were present in the response curves (Figure 6).

The response was unexpected in several aspects; the second critical speed was expected to be higher than 4500 rpm but one peak was clearly below 4000 rpm. A second peak occurred at about 4500 rpm. Also, at the exhaust end, the LP 1 probe showed two distinct peaks, while the LP 2 probe showed only one at an intermediate speed. The two probes are located at 45 degrees on each side of the vertical centerline. This orientation is different from the horizontal and vertical axes normally used in the analysis, and makes it difficult to differentiate the horizontal and vertical criticals. Based on these data, the critical speeds are lower than predicted.

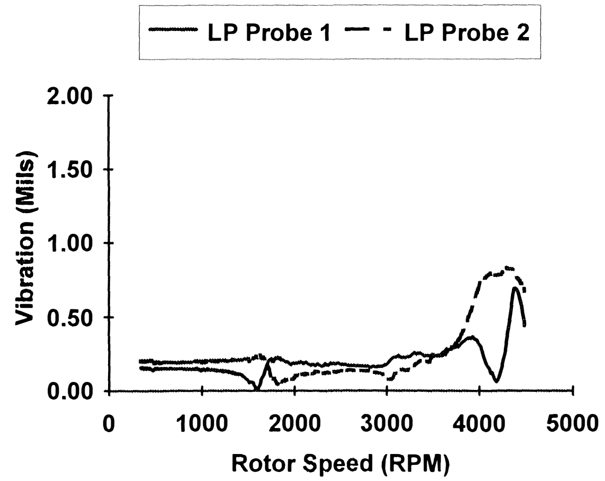

Figure 5. Balanced Rotor Shaft Vibration.

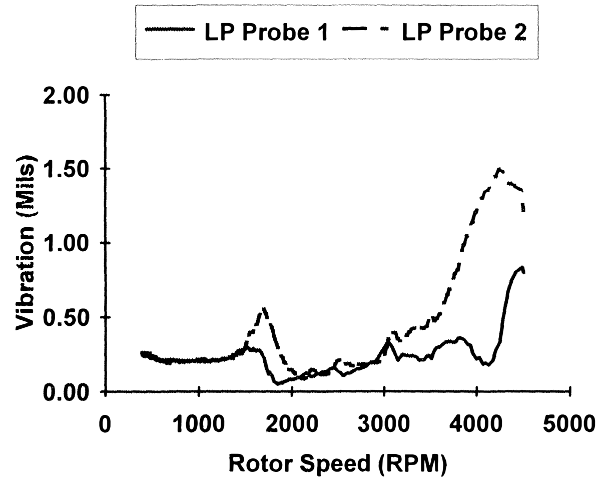

Figure 6. Response to Unbalance Test.

SPECIAL TESTS AND ANALYSES PERFORMED

Dynamic Testing of Assembled System

In order to characterize the natural frequencies of the turbine rotor, casing and any other components, dynamic testing was performed on the fully assembled unit. This testing was performed by artificially exciting the bearing caps with an electromagnetic shaker. The shaker applies a known force to the structure and allows the dynamic response to be measured. By varying the frequency of the force through the operating range, any natural frequencies or resonances of the system can be identified. The dynamic response was tested in both the horizontal and vertical directions. Testing in both directions was necessary because dissymmetry of the bearing and its supporting structure can cause significant differences in stiffness and natural frequencies in the two directions.

The natural frequencies identified by this process correspond roughly to the natural frequencies or critical speeds of the turbine. There is some difference between this nonoperational testing and actual turbine operation, mainly in the behavior of the bearings. The journal bearings used in this design are tilting pad bearings. They support the rotor on a thin film of oil generated by the rotation of the rotor. This oil film has specific properties of stiffness and damping that are included in the analysis of rotor critical speeds. When the rotor is not turning there is little oil in the bearing resulting in a stiffer interface from the rotor to the bearing. Because of this increased stiffness, nonoperational tests can be expected to show higher natural frequencies than operational testing would show. In spite of this discrepancy, nonoperational testing is a reasonable method to determine natural frequencies of this system.

The results of this testing (Figure 7) identified a dominant natural frequency at 74.75 Hz (4485 rpm) in the horizontal direction and 83.5 Hz (5010 rpm) in the vertical direction. As expected, these frequencies are higher than the critical speeds seen during operational testing. However, they are lower than would be predicted using the assumed bearing support stiffness of eight million lb/in (Table 2). If the combined stiffness of the bearing and support is eight million lb/in, the natural frequencies would be expected to be greater than 100 hz (6000 rpm).

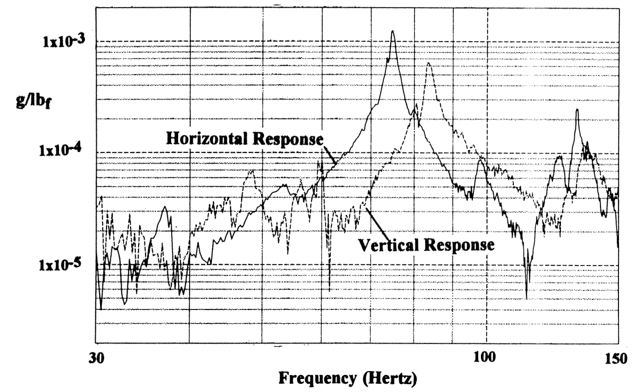

Figure 7. Transfer Functions—Original Measured Bearing Support Accelerance.

The actual stiffness of the low pressure bearing support structure was not determined directly from the test data. It was estimated from the calculations of the sensitivity of the second natural frequency to the support stiffness. The tested natural frequencies correspond to an exhaust support stiffness of approximately two million lb/in in the horizontal direction, and four million lb/in in the vertical direction. These values are much lower that the assumed stiffness of eight million lb/in. The cause of the low stiffness was attributed to the geometry of the end wall of the turbine exhaust, which supports the bearing. In most designs, there are a number of plates that both support the bearing and direct the steam flow. With this exhaust design, fewer plates were used. This made a less direct path from the bearing housing to the support feet.

Table 2. Stiffness of Bearing Supports.

Configuration	*High Pressure Bearing* *Vertical* *(10^6lb/in)*	 *Horizontal* *(10^6lb/in)*	*Low Pressure Bearing* *Vertical* *(10^6lb/in)*	 *Horizontal* *(10^6lb/in)*
Original Analysis	8.0	8.0	8.0	8.0
As-Built	15.0	15.0	4.0	2.0
As-Modified	15.0	15.0	4.5	4.0

The high pressure drive end support stiffness tested higher, approximately 15.0 million lb/in, than the original 8.0 million in both the vertical and horizontal directions. This high stiffness was the result of designing the high pressure end bearing support for a front end drive application.

FEA Analysis of LP Bearing Support

To confirm that the exhaust endwall was the cause of the low stiffness, a finite element analysis of the exhaust structure was performed. This analysis is not normally done for each application because of the similarity of the designs. In this case, however, it was clear that the analysis was necessary.

The finite element mesh shown in Figure 8 includes the end wall of the casing, the horizontal joint, the support feet and the bearing support structure. The primary support of the turbine is through the footplates extending on each side of the low pressure hood. The model was loaded independently in the horizontal and vertical directions, along the center of the bearing seat. The resulting displacement in both the horizontal and vertical direction was divided by the input load to determine the stiffness of the structure.

The results of the finite element analysis are a horizontal stiffness of approximately two million lb/in and a vertical stiffness of approximately four million lb/in. These stiffness results are consistent with the values derived from the response to unbalance test and the artificial excitation tests.

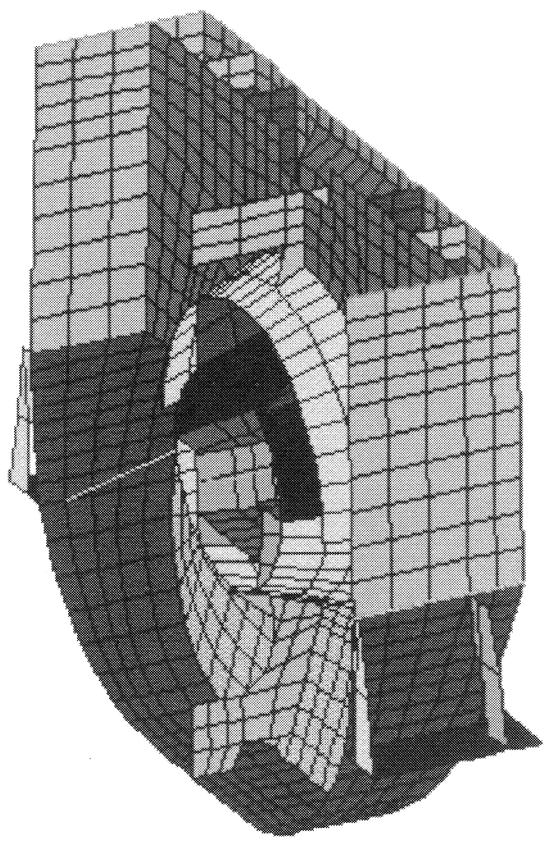

Figure 8. Exhaust Casing Finite Element Model.

MODIFIED DESIGN, ANALYSIS, AND TEST

Exhaust Bearing Support Modifications

Based on the testing and analyses made on the original design, the requirements of API Standard 612 on critical speed separation margin would be approached by significantly stiffening the exhaust end bearing support. Since the unit was completely assembled, it was desired to keep casing distortion due to structural modifications to a minimum.

After investigating several alternatives, the following modifications were made to increase the bearing support stiffness at the exhaust end. Four solid bar struts, each 2.50 in in diameter, were welded in the casing as shown in Figure 9. The function of the bars was to substantially increase the horizontal stiffness of the exhaust end bearing support, with a lesser increase in vertical stiffness. In addition to the struts, a triangular gusset was added on the vertical centerline of the exhaust to add additional support in the vertical direction. Two gussets were also added to stiffen the structure where the struts attach to the exhaust casing.

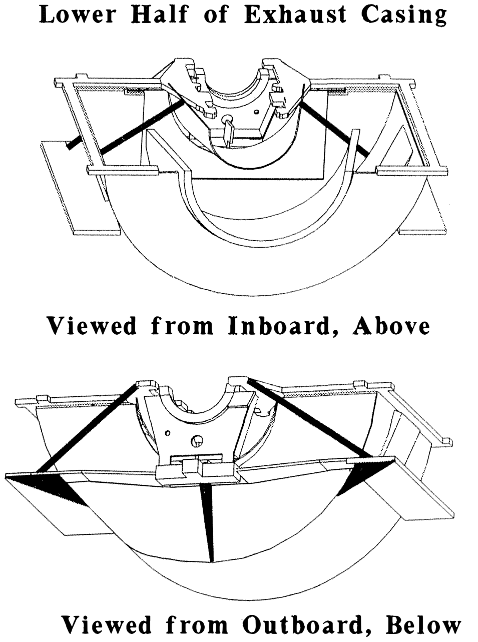
Lower Half of Exhaust Casing
Viewed from Inboard, Above
Viewed from Outboard, Below

Figure 9. Modifications to Bearing Support Structure.

Dynamic Testing of Assembled System

The artificial excitation tests were repeated on the low pressure bearing support after the structural modifications were installed. The natural frequencies in both the vertical and horizontal directions were improved as shown in Figure 10. The horizontal natural frequency increased from 4485 rpm to 5160 rpm. The vertical natural frequency increased from 4980 rpm to 5190 rpm. With these data, the support stiffness for the modified exhaust could be derived. This was done by increasing the assumed support stiffness values in the rotordynamics model until the calculated natural frequencies matched the test data. Based on this method, the exhaust end horizontal stiffness increased from 2.0 to 4.0 million and the vertical stiffness increased from 4.0 to 4.5 million lb/in.

Response to Unbalance with Modified Bearing Support

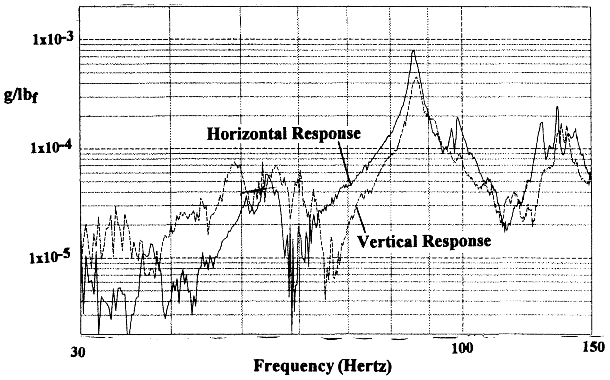

Figure 10. Transfer Functions —Modified System.

Following the installation of the structural modifications and artificial excitation testing, the unit was again run with the unbalance for the second critical speed. The results are shown in Figure 11. Although the critical speeds were increased, the relative vibration was not significantly reduced, and at some speeds was even increased. It had been speculated that by increasing the exhaust end support stiffness, the bearing would provide more damping to reduce rotor vibration in addition to increasing the critical speed. Indeed, when the measured vibratory motions of the bearing supports were combined with the shaft relative probe data, the absolute vibration of the shaft was substantially reduced.

To verify this result, response to unbalance calculations were made to predict the shaft motion relative to the bearing as measured by the vibration probes. The difference in calculated shaft absolute and shaft relative response can be seen by comparing Figures 12 and 13. The absolute response at the probe locations are shown in Figure 12 for the modified exhaust end bearing support. The peak response occurs at the exhaust end (LP) and is 3.5 mils. The shaft relative response at the same locations are shown in Figure 13. The relative vibration at the exhaust end is only forty percent of the absolute vibration. Because of the significantly higher drive end stiffness, the shaft relative motion is eighty percent of the absolute shaft motion.

Since the original design calculations were made on the basis of absolute vibration, the shaft relative motions for the original design support stiffness of 8.0 million were calculated in addition to the as-built and modified exhaust bearing supports. The calculated responses at the second critical for each configuration are given in Table 3.

The predicted shaft response relative to the bearings for the original 8.0 million lb/in support stiffnesses was highest at the

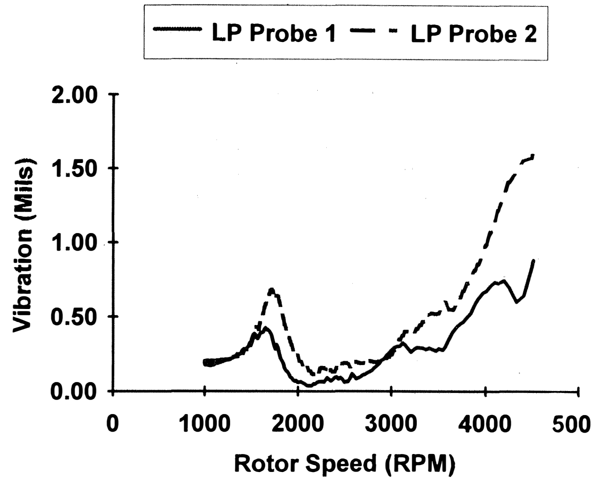

Figure 11. Response to Unbalance—Test of Unit as Modified.

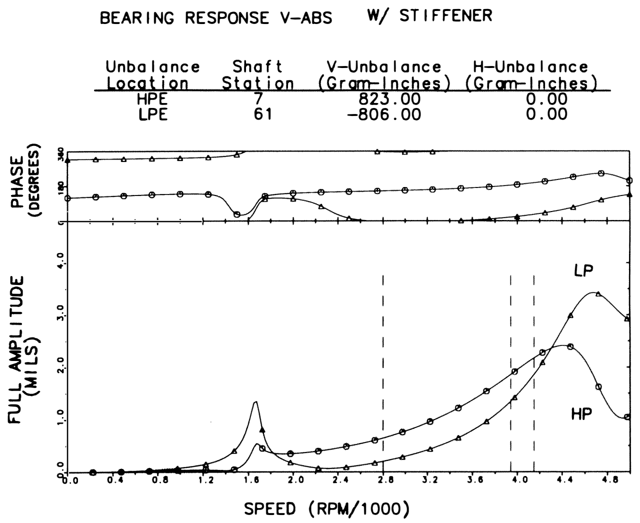

Figure 12. Response to Unbalance—Shaft Absolute Vibration with Modified Bearing Support.

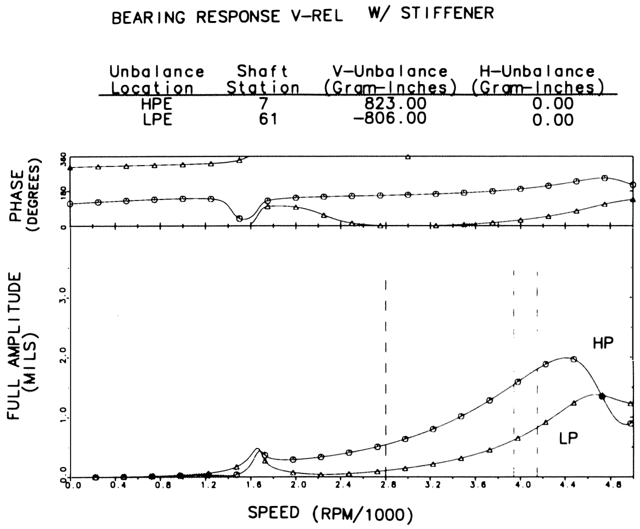

Figure 13. Response to Unbalance—Shaft Relative Vibration with Modified Bearing Support.

drive end (HP) with an amplitude of 2.5 mils and at a speed of 4760 rpm. The response at the exhaust end was 2.0 mils. The amplification factor at the critical was 4.0. Using the as-built support stiffness, the peak response at the drive end decreased to 1.9 mils and the exhaust end decreased to 1.5 mils. This closely matched the measured peak response shown in Figure 6. Even with the significantly lower critical speed, the calculated amplification factor increased by only 10 percent to 4.4. When the modified support stiffness values are used, the peak response changed by only 0.1 mil with a slight reduction in amplification factor. The calculated critical speed increased to 4675 rpm.

Table 3. Calculated Shaft Relative Vibration at Second Critical.

Configuration	*HP (mils)*	*LP (mils)*	*2nd Critical (rpm)*	*Amplification Factor*
Original Calculations	2.5	2.0	4825	4.0
As-Built	1.9	1.5	4300	4.4
As-Modified	2.0	1.4	4675	4.3

Based on the results of these calculations and the actual response to unbalance testing, the response at the vibration probes was insensitive to changes in stiffness in the low pressure bearing support. While the location of the second critical was influenced by a stiffer low pressure bearing support, the amplification factors for the different assumptions were not.

CONCLUSION

Although the second critical speed of this turbine was strongly influenced by the exhaust end bearing support stiffness, the response to unbalance as measured at the bearing was only slightly influenced. The comparisons of the original and modified calculations demonstrate that an acceptable design can be achieved even though the design requirements of API Standard 612 were not met.

For designs having a bearing support stiffness similar in magnitude to the bearing dynamic stiffness, comparisons between test stand data and calculated response must be done on a consistent basis. If test data are acquired on a shaft relative basis, the calculations should also be made on this basis. Shaft relative motion is a more accurate indication of the dynamic force transmitted through the bearing.

When test results do not match predictions, the cause of the discrepancy can be determined. Stationary excitation of the bearing supports with the rotor installed can be performed and the data interpreted to confirm the bearing support stiffness. This does require supplemental rotordynamic analyses to aid in this process.

DISCUSSION

Operational Experience of Two Other Units

The first two steam turbine drivers were successfully shipped early from the vendor and were already in operation prior to the verification testing of the third unit. Those two units are expected to have more separation margin than the third unit. One of the two units operates at a lower speed than used in the unbalance test, and the other has a lighter coupling than the tested unit. It was therefore determined that field modifications to the exhaust end on the installed units would represent more risk to equipment than would be gained.

The two operating units have performed flawlessly with very low vibration levels during operation. One unit operates at 4060 rpm and has 0.1 to 0.3 mils vibration. The other unit operates at 3220 rpm and has 0.1 to 0.4 mils vibration.

Teamwork and Cooperation of OEM and User

It is important to note that although a potential problem was uncovered during the shop verification test, close cooperation and openness between the buyer and vendor identified and resolved the discrepancies between the predicted and test responses. Analyses of the "as-built" units with a corrected analytical model were used to verify acceptable sensitivity to unbalance.

The stated purposes behind the addition of the verification test to API Standard 612 were certainly met on this unit, as it became a learning experience to all, and furnished the vendor with improved design data which will be used on future turbines utilizing this exhaust end. Although the design of the two operating units was not modified in any way, a benefit to the user was also experienced in having better definition of the operating envelope of these machines, which may in the future prove invaluable.

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