DYNAMIC IMPROVEMENT OF AN OVERHUNG SINGLE STAGE PUMP

by
Giacomo Marenco
Director
Alessandro Nicchio
and
Alberto Piva
Weir Gabbioneta Srl
Research and Development Department
Sesto San Giovanni, Italy

Giacomo Marenco is Director, Research and Development Department, Weir Gabbioneta Srl, in Sesto San Giovanni, Italy. He has more than 30 years of experience in the pump industry. He is a coauthor of several papers concerning rotordynamic analyses applied to pump design and/or pump troubleshooting.

Alessandro Nicchio is with the Research and Development Department, Weir Gabbioneta Srl, in Sesto San Giovanni, Italy. He has been with them since May 2006, studying all mechanical aspects of pumps with finite element methods and experimental modal identification techniques. In 2007, he began studying computational fluid dynamics and its application on pump engineering, being involved in the Gabbioneta fluid dynamics research projects.

Mr. Nicchio has a degree (Mechanical Engineering, 2005) from Politecnico di Milano, specializing in finite element methods and mechanical system dynamics. After receiving his degree, he worked as a researcher for the Politecnico di Milano continuing his thesis work on dynamics of high speed train pantographs.

Alberto Piva is with the Research and Development Department, Weir Gabbioneta Srl, in Sesto San Giovanni, Italy, involved in the development of new pump design. He began his career with them dealing with finite element analyses, computational fluid dynamics, and experimental activities.

Mr. Piva has a degree (Aeronautics Engineering, 1994) from Politecnico di Milano in, specializing in structural design. He then worked for a short time as a researcher for the Politecnico di Milano in the field of the mechanics of fracture.

ABSTRACT

Vibration control is an important aspect of all pump design. It helps assure quiet operation and high reliability. Therefore, the design of pump components should avoid any dangerous resonances with excitation forces, such as unbalance, misalignment, vane passing frequencies, etc.

A key pump component is the bearing housing, since this transmits forces from the shaft to the pump casing and therefore to foundations. Bearing housing designs having good dynamic behavior can reduce vibration levels and dramatically improve both the reliability and life of the whole pump assembly.

PROBLEM DESCRIPTION AND SOLUTION METHODS

This paper describes both the analysis and refinement of bearing housings used in a line of a horizontal overhung process pumps (API classification OH 2) (Figure 1). Nowadays, such product lines are modular and will be configured from highly standardized components. It was foreseen that certain component configurations (particularly when operating at higher speeds) might produce an unfavorable combination of variables, such as impeller size, shaft diameter, height of pump centerline, as well as whole pump mass. These are some of the factors affecting structural natural frequencies within the pump. Hence, they can also influence resonances with the rotational speed.

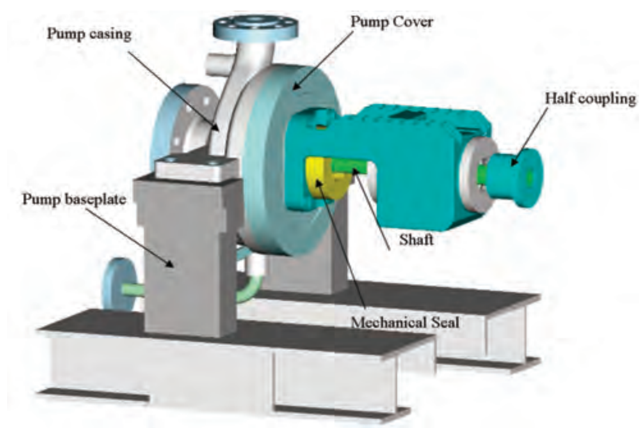


Figure 1. Typical View of the OH2 Type Pumps That Were Studied.

To avoid the above problem, it was required that the original bearing housing be further refined, but without compromising the functional characteristics of the pump. These functions include standardization of the housing, in spite of a great variety of mechanical seals and their auxiliary lines, good heat dissipation capability, mounting-space requirements, etc.

The whole pump line is comprehensive and spans an absorbed power range, from 10 kW to 750 kW. Such a wide range of modular possibilities is accomplished within only three sizes of bearing housing. Each housing can fit every type of auxiliary API plan without further customization. This helps keep production costs low. Moreover, changes in the execution are always possible with minimal expenses and without design intervention to the bearing housing. On the other hand, this makes the design refinement more difficult, because of the multiple constraints imposed by a modular concept.

The particular shape adopted for the bearing housings had previously been optimized to assure good heat dissipation (Cipolla, 2005). Thanks to this improved thermal behavior, water cooling is not required, even with pumped fluid temperature up to 752°F (400°C). This is mainly due to positioning of the support arms, which were relocated upwards and away from the center line of the bearing housing. This allows the best dissipation of heat transferred from the pump through the bearing housing support arms. Furthermore, the open slot in the upper part of the bearing housing helps further minimize direct heat transfer through the support arms. This aids reductions in the line bearing temperature. The technique applied, is summarized in Figure 2.

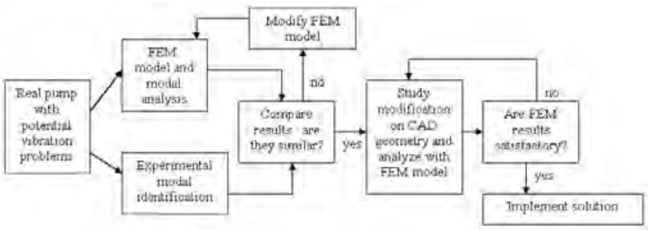


Figure 2. Adopted Technique Flowchart.

Initially, this technique was applied to the smaller bearing housing size. Experimental data were used to calibrate a finite element model of the pump. Then, using finite element modeling (FEM), the bearing housing design was optimized to improve its dynamic behavior. As any modification to reduce vibration levels can also change the heat dissipation capability, thermal analyses were implemented to assure the same thermal performance. This same methodology was then applied to other bearing housing sizes in order to extend the dynamic enhancements to the completely overhung pump line.

PUMP MODAL ANALYSIS

A simply modal analysis of the pump was carried out, measuring the acceleration response to a range of impacts. This simple technique allows quick identification of natural frequencies for the assembly. The frequency range considered for the analysis spanned 0 to 200 Hz. This range included the higher peaks that had been noted during prototype performance tests. In the authors' experience, most vibration problems can be related to mechanical exciting forces such as, unbalance (1× running speed), misalignment (2× running speed), and component resonances within this frequency range.

Excitation forces at higher frequencies (i.e., vane passing frequency) were excluded from the study because the first modes of the bearing housing have natural frequencies far below this. Furthermore, higher modes are typically more damped than lower ones, causing lower amplitude vibrations.

The data were recorded using a portable spectrum analyzer, acquiring data with a final frequency resolution of 0.625 Hz. The accelerations were measured using a 0 to 50 g range accelerometer, with the force impulses being measured with an "impact" hammer. A linear average of different acquisitions in the same test conditions was performed, in order to keep random "noise" in the measurements as low as possible.

The pump was analyzed in isolation, that is, without connections to piping and without the motor coupling. Although these conditions are different from reality, they do allow the investigation of the machine behavior without external influences. They also help assure good repeatability in the finite element model. Furthermore, these are the conditions mandated by API 610 (2004) for bearing housing resonance testing (paragraph 7.3.4.6).

Figure 3 and Figure 4 show, respectively, both the position and direction of force impulse for the modal analysis of the bearing housing. This drive end location, was chosen because the prototype pump's performance tests showed highest vibration here. Typical results (modulus and phase) are shown in Figure 5 for vertical direction and Figure 6 for horizontal direction).

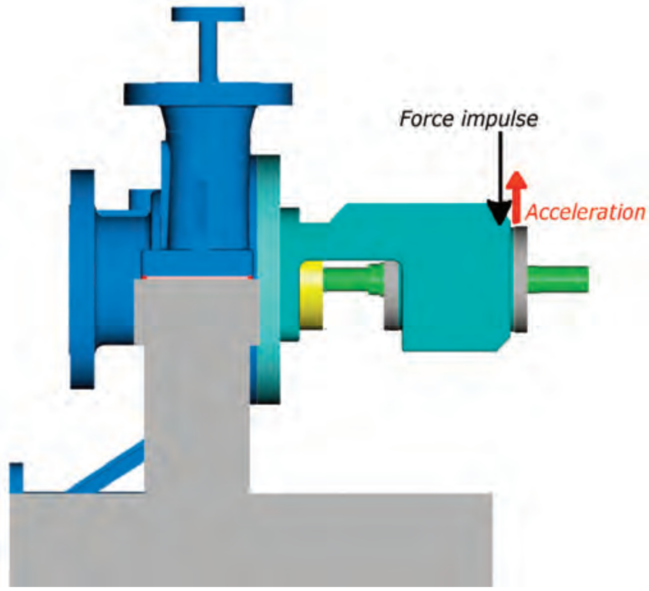


Figure 3. Drive End Impulse/Measurement Location in Vertical Direction.

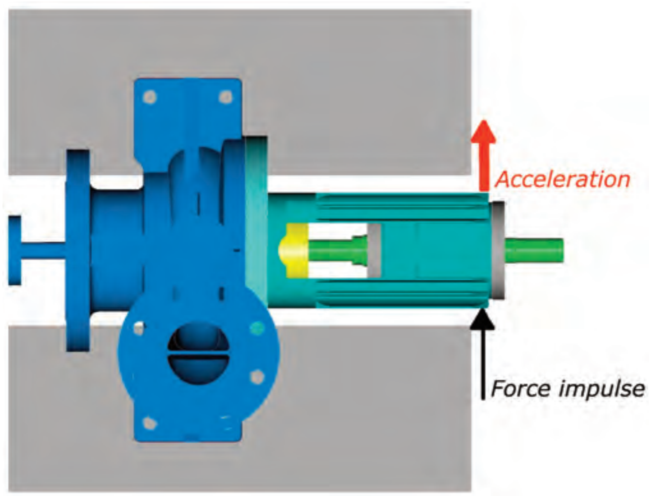


Figure 4. Drive End Impulse/Measurement Location in Horizontal Direction.

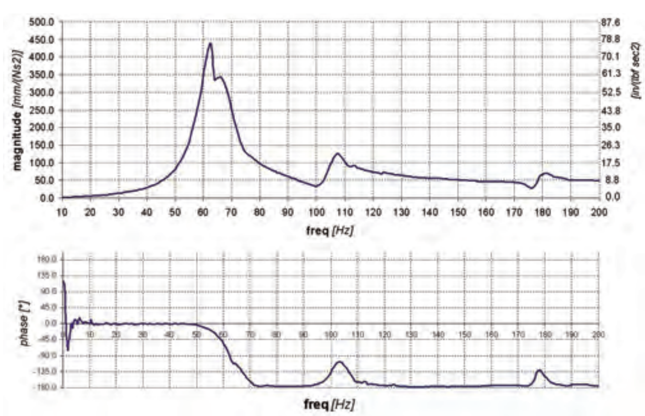


Figure 5. Drive End Frequency Response Magnitude and Phase, Vertical Direction.

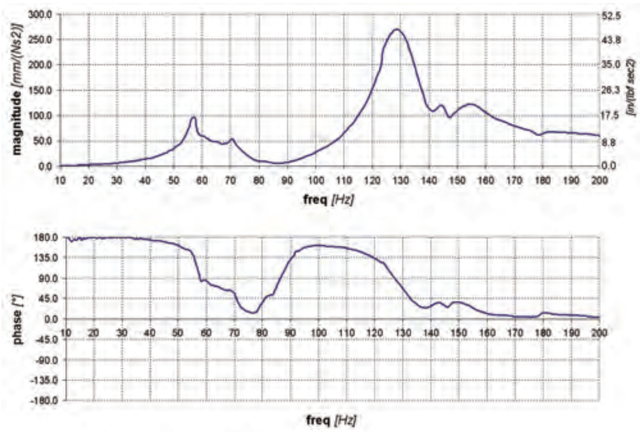


Figure 6. Drive End Frequency Response Magnitude and Phase, Horizontal Direction.

As previously mentioned, different modular configurations of the machine can shift the natural frequency of the spectra. This is illustrated in Figures 7 and 8. They show the drive end response functions for two different configurations (equipped with the same bearing housing) measured, respectively, in the vertical and horizontal direction. This variability in natural frequency complicates the design process, as one bearing housing design must assure good performance in many different modular configurations.

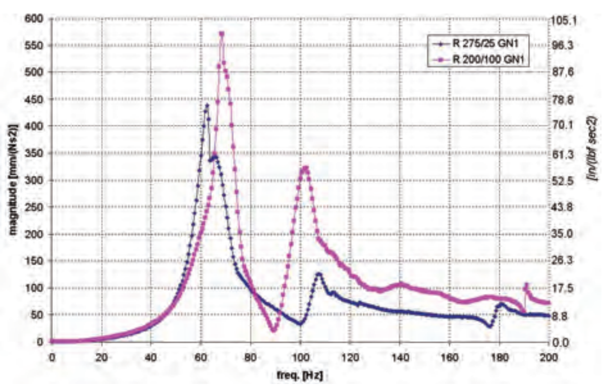


Figure 7. Comparison of Drive End Vertical Direction Frequency Response Magnitude for Two Different Pump Configurations Equipped with the Same Bearing Housing.

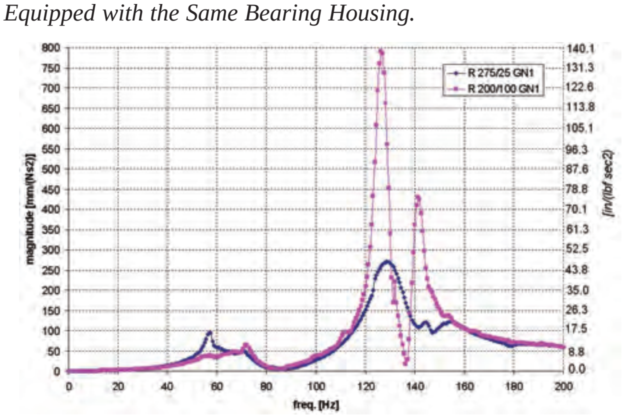


Figure 8. Comparison of Drive End Horizontal Direction Frequency Response Magnitude for Two Different Pump Configurations Equipped with the Same Bearing Housing.

The spectra for the vertical direction (Figures 5 and 7) show a margin of between 4 and 15 percent between the first natural frequency and a nominal running speed of 3600 rpm. Such proximity would indicate a potential for vibration problems.

FEM MODEL DESCRIPTION

The acquired data were used to calibrate a finite element model of the pump assembly, focusing on the bearing housing. This allowed a study of several possible design modifications. All were aimed at increasing the separation margin between the first vertical natural frequency of the machine and its rotational speed. The worst case found during prototype tests was considered.

Correct modeling of the connections between different parts of the assembly is critical to a good assessment of the model. The contact regions form a strong nonlinear constraint between the bodies so the greatest difficulty is their correct approximation in a linear modal analysis. In this study, two different techniques have been adopted concurrently, in order to solve this problem.

The first technique uses spring element of adequate stiffness to model the roller type bearings and to connect a node of the shaft to a set of nodes belonging to the bearing housing. The second concerns the contact between the bearing housing, the pump casing, and the baseplate. It involves creating a solid region with artificial material properties that make an approximation of the contact region. For simplicity, a unique set of parameters was selected to best fit all the vibration modes.

The optimization process requires several iterative computer runs, so simplification of the geometry adopted in the FEM model is useful. Only the influential features must be taken into account, in order to reduce the degrees of freedom of the model. Such simplification helps reduce the overall cycle time, but its justification relies on the knowledge of the studied phenomena (available experimental data, other similar past cases) and on previous experiences.

To further improve the efficiency of the FEM models, a high-order (P-element) solver was used to model the pump geometry. The P-elements used have shape function with a polynomial grade varying from 3 up to 9, according to the displacement gradient calculated during the solution. These kinds of elements have two main advantages. First, the calculation grid is definitely coarser than a classic linear or parabolic elements grid. This helps reduce the total number of degrees of freedom of the model. Second, this kind of element can map curved geometry thanks to the polynomial shape function.

A first modal analysis was performed to calculate both the natural frequencies and the modal shapes in the desired frequency range. Then a modal frequency response analysis was carried out and compared with the experimental one. Modal damping coefficients were introduced for each mode in order to achieve the best fitting of experimental data. Figure 9 and figure 10 compare the experimental response function with calculations for, respectively vertical and horizontal direction.

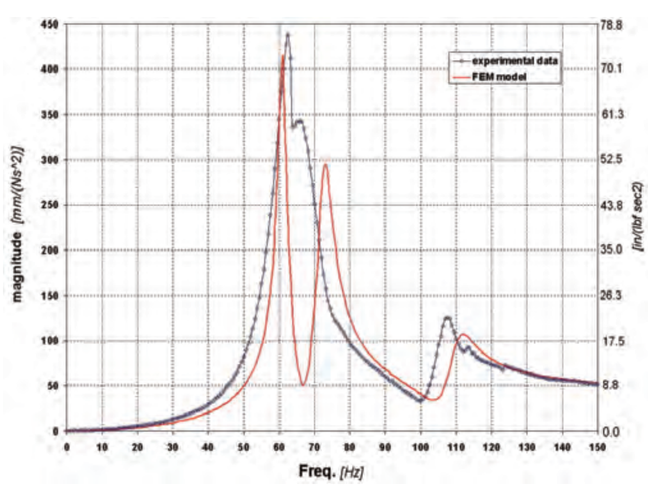


Figure 9. Comparison Between Calculated Frequency Response and Experimental Data for Drive and Vertical Direction.

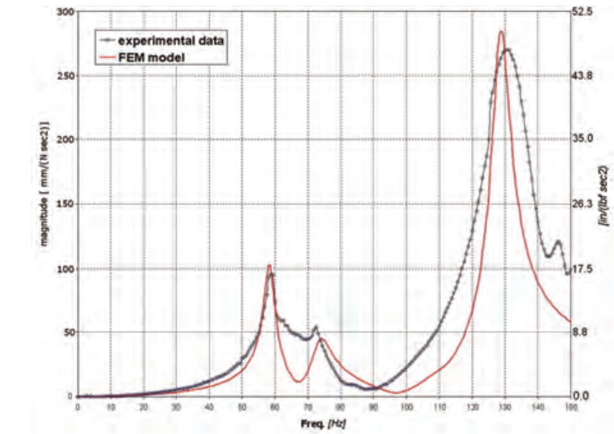


Figure 10. Comparison Between Calculated Frequency Response and Experimental Data for Drive and Horizontal Direction.

Table 1 summarizes both the natural frequencies and the discrepancies obtained during set up of the model. The first two modal shape forecasts are shown in Figures 11 and 12. Figure 13 shows the displacement calculated for the first vertical modal shape. It shows that the main deflection of the bearing housing occurs near the mounting flange. Displacements in the remaining part of the housing seem proportional to the distance from a point located at about 2.56 in (65 mm) from the pump casing.

Table 1. Comparison Between FEM Results and Experimental Data.

experimental frequency [Hz]	Calculated frequency [Hz]	Error [Hz]	error %
58.7	58.3	-0.4	-0.7
62.5	60.9	-1.6	-2.6
67	72.6	5.6	8.4
71	73.7	2.7	3.8
109.4	110.1	0.7	0.6
130.6	128.6	-2	-1.5

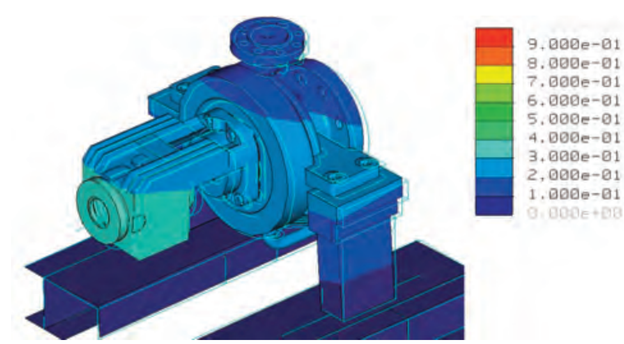


Figure 11. First Horizontal Modal Shape.

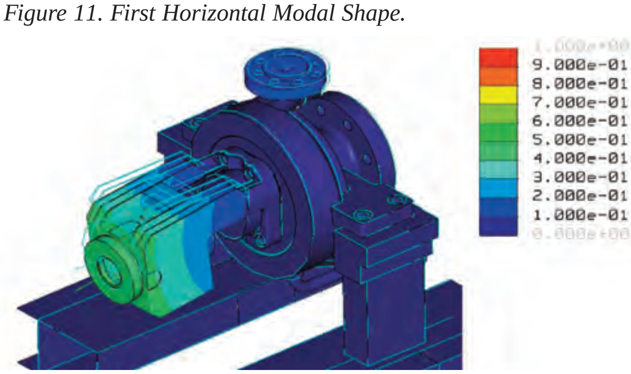


Figure 12. First Vertical Modal Shape.

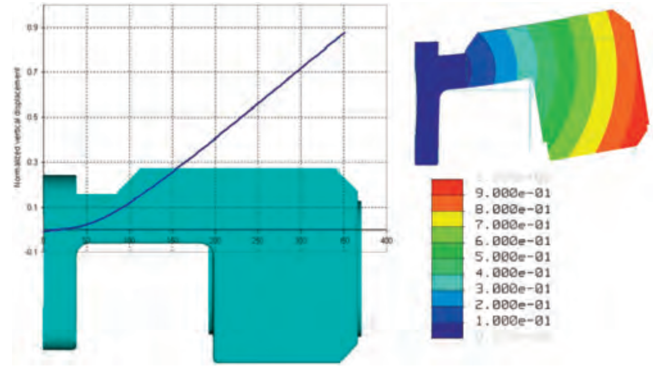


Figure 13. Bearing Housing Vertical First Modal Shape Displacements.

This analysis of the modal shape suggests:

- The area of greatest influence is the bearing housing support arms. Their cross section must be increased in order to help raise the natural frequency.
- The mass of the remaining part of the bearing housing must be reduced to further assure a higher natural frequency.

BEARING HOUSING REDESIGN AND TESTS

Starting from the FEM results, possible bearing housing changes can be implemented into the model to try to predict the real behavior of the new component. This allows accurate study of changed components without the use of expensive prototypes, thus saving money and time.

Figure 14 compares the initial shape of the bearing housing (on the left) with the final one (on the right). This shows an increased cross section of the support arms in the area of highest flexibility. It also shows the chamfer that has been introduced between the arms and the mounting flange.

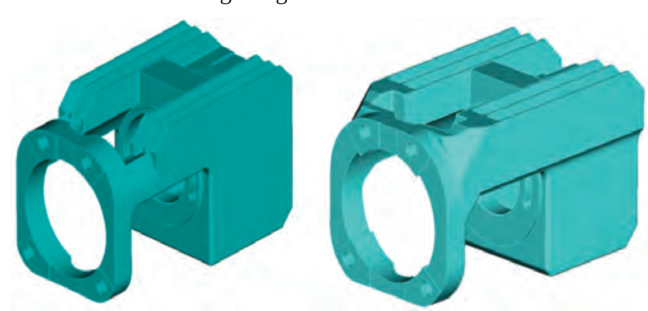


Figure 14. Comparison Between the Original (Left) and the Optimized Bearing Housing (Right).

Despite increasing the arm cross section, bearing housing mass has only increased by 2.5 percent (71.9 lb versus 70.1 lb, 32.6 kg versus 31.8 kg). This is because the remaining parts of the bearing housing were lightened. Consequently, the center of gravity moved closer to the bearing housing flange. Importantly, the improved bearing housing design retains interoperability with the old one, and no other components have been modified. In addition, the capability of fitting all auxiliary plans has been maintained.

Different sizes and configurations of pumps were analyzed with this new bearing housing, all adopting the same methodology described. In all cases, the new bearing housing has shown higher frequencies for the first vertical modal shape. There is a corresponding increase in the separation margin from the 60 Hz running speed.

Following the encouraging outcome of this theoretical study, a set of new bearing housings was produced and tested. Figure 15 compares both the FEM and experimental results. Although the finite element model will inevitably have some uncertainties, Figure 15 shows that the new design would have a significant increase in the vertical resonant frequency. A corresponding separation margin of about 20 percent is implied.

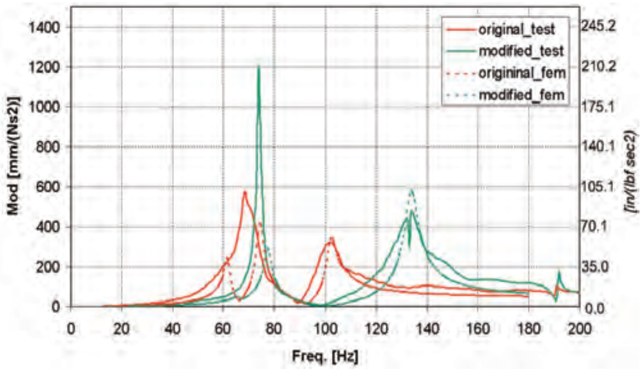


Figure 15. Drive End Frequency Response for Original and Modified Bearing Housing.

Uncertainties arise from generalizations and approximations in the modeling process, particularly regarding the component contact conditions. Nevertheless, such studies still yield useful results. They can help guide the designer in the direction of a more effective shape for the bearing housing.

Figure 16 shows the impact hammer test results obtained on other pumps that were equipped with the new bearing housing. It shows a first peak at 70 Hz in the vertical direction. As the dynamic improvement lead to an increase in support arm cross section, heat flux from pump casing increases as well. Consequently, it was necessary to find new solutions that reduce the heat flux. Hence, a new bearing housing support flange was designed, having reduced contact area with the pump cover. This was studied with a steady-state thermal analysis, and the results compared with those of Cipolla (2005) for the old bearing housing. The tests confirmed the same good heat dissipation.

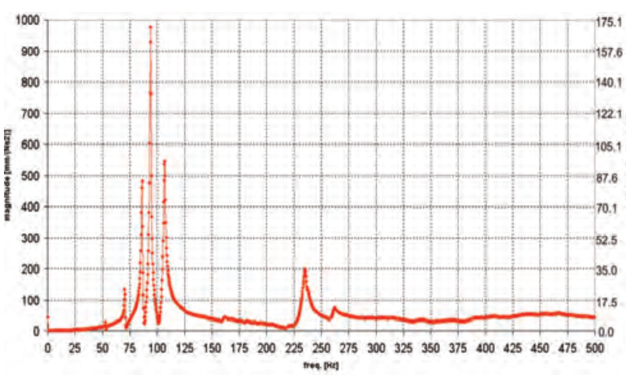


Figure 16. New Design Bearing Housing Frequency Response Magnitude.

LARGER BEARING HOUSING SIZES—REDESIGN TECHNIQUE EXTENSIONS

The same methodology was adopted for the larger bearing housing sizes. The objective here was to obtain similar improvements and thus achieve a better dynamic behavior across the complete pump line. Similar good results were obtained, as shown in Figure 17 (drive end vertical direction frequency response). The new design achieves a significant improvement, clearly evident in the comparison between Figures 18 and 19, reporting the velocity vibration values recorded on a pump running at about 3600 rpm, with the first design bearing housing (Figure 18) and with the optimized design (Figure 19).

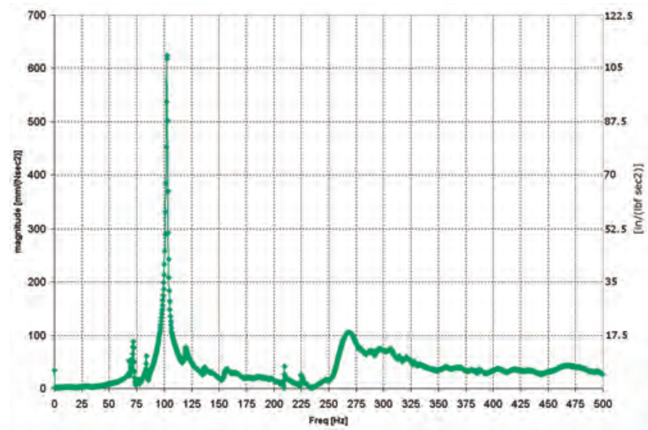
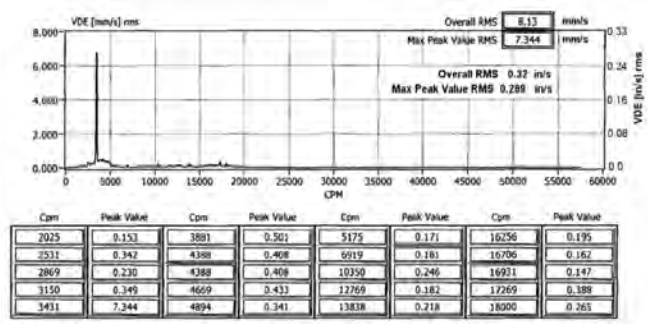
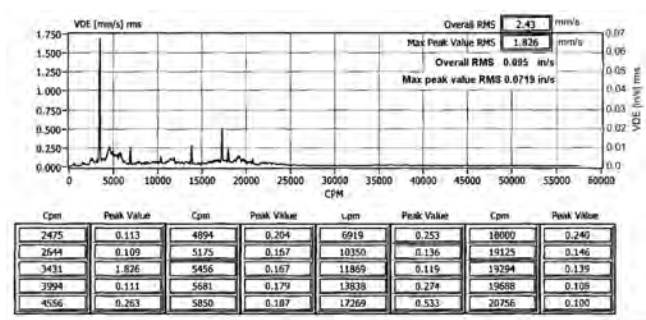


Figure 17. New Design Bearing Housing Response Function Magnitude.



Cpm	Peak Value	Cpm	Peak Value	Cpm	Peak Value	Cpm	Peak Value
2025	0.153	3881	0.501	5175	0.171	16256	0.195
2531	0.342	4388	0.408	6919	0.181	16706	0.162
2869	0.230	4388	0.408	10350	0.246	16931	0.147
3150	0.349	4669	0.433	12769	0.182	17269	0.188
3431	7.344	4894	0.341	13838	0.218	18000	0.265

Figure 18. Velocity Vibration Value (RMS) Measured at Drive End Position in Vertical Direction, Original Housing Design.



Cpm	Peak Value	Cpm	Peak Value	Cpm	Peak Value	Cpm	Peak Value
2475	0.113	4894	0.204	6919	0.253	18000	0.240
2644	0.109	5175	0.167	10350	0.136	19125	0.146
3431	1.826	5456	0.167	11869	0.119	19294	0.139
3994	0.111	5681	0.179	13838	0.274	19688	0.108
4556	0.263	5850	0.187	17269	0.533	20756	0.100

Figure 19. Velocity Vibration Value (RMS) Measured at Drive End Position in Vertical Direction, Improved Housing Design.

It is clear that the peak amplitude at 1× running speed, reduces from 0.289 in/s RMS (7.34 mm/s RMS), in the original design, to 0.072 in/s RMS (1.8 mm/s RMS) for the new optimized housing.

CONCLUSION

This paper describes the modal analyses carried out on the bearing housings of an overhung process pump line. It explains how a reduced separation margin between the running speed and the natural frequencies of the machine may cause high machine vibration during operation. To study this issue, a finite element model of the pump was set up and calibrated on experimental data. By studying this "virtual" model, it was possible to quickly explore several alternative solutions to the vibration behavior.

Changes to both the bearing housing support-arm (cross sectional area) and bearing housing total mass were made to "tune" the bearing housing. Furthermore, a new design of bearing housing mounting flange achieved the same good thermal behavior as before. This was accomplished despite increasing the cross sectional areas of the bearing housing support arms. The final bearing housing design was next shop tested in the real machine. It exhibited very good dynamic behavior and low vibration levels.

The same design methodology was subsequently extended to the larger bearing housings, so as to benefit the entire pump line. Overall, despite making assumptions made to simplify such a complex problem, the adopted technique appears to be quite a powerful tool in helping the designer to solve vibration problems.

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