VIBRO-ACOUSTIC TROUBLESHOOTING SOLVES 5MW BOILERFEED PUMP TESTRIG NOISE & VIBRATION PROBLEMS

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ABSTRACT This paper describes the global vibro-acoustic troubleshooting approach, used to identify and separate different sources of noise and vibrations on a boilerfeed pump testrig. The pump serves for rotor dynamic research of a EC-funded BRITE-Euram project. This approach resulted in the identification of local structural flexibilities in the connections between the machinery and the base plate. The relative importance of the modes during normal operation is revealed by comparison with operational deformation shapes. The use of sound intensity mapping allowed to calculate the total sound power and to rank the equipment according to its sound power contribution. High acoustic levels were found and related to the fluid drive and to the piping system. Modification of the piping section resulted in a reduction of noise and vibration levels along the test loop and smooth operation in a wide suction pressure range.

1. INTRODUCTION

An EC-BRITE-EURAM funded research programme to study and model fluid structure interactions in boilerfeed pumps (BFP) is run by three partners : BW/IP International BV (NL), LMS International NV (B) and the University of Kaiserslautern (D). [1] [2]
To produce sufficient data for the comparison of calculated and measured rotor dynamic coefficients, two testrigs are built. One testrig aims at component testing and is built at the University of Kaiserslautern laboratories. A larger testrig is built at BW/IP test facilities (Etten-Leur, The Netherlands) and is mainly designed to identify impeller force and moment coëfficients under actual power plant conditions. The equipment is equivalent to a 50% part load boilerfeed pump of a 300 MWatt powerplant. The pump is driven over a fluid coupling by an electric engine. The entire assembly is mounted on a steel honeycomb skid. To allow externally controlled dynamic excitation of the shaft, a full magnetic suspension is used. The bearing controller information is processed to obtain accurate force and displacement measurements. This rotor dynamic information is combined with accurate pump modelling to identify fluid related, motion dependent forces and displacements at inaccessible shaft sections. To verify the accuracy of the preliminary dry shaft model the rotor response to magnetic bearing excitation was compared to analytical FRF (see fig. 1). Apart from a shift in the rotor bending frequencies, additional resonance peaks are found in the measured FRF. To search for these resonances and fine tune the existing rotor model, the pump was disassembled and a modal analysis was performed on the pump rotor in free-free conditions.

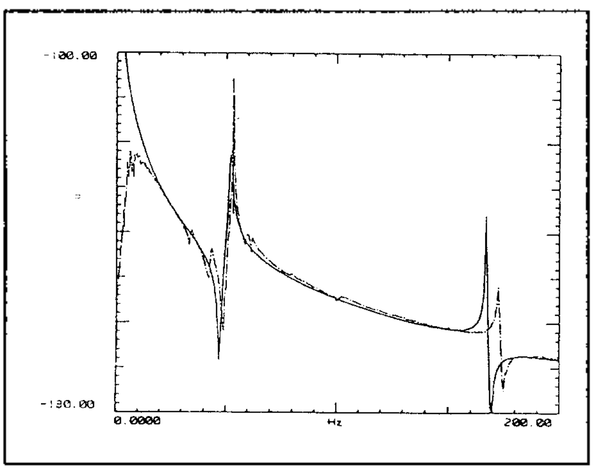


Fig. 1. Measured (dotted line) and predicted (full line) FRF in one bearing

As the proximity probes in the pump measure relative displacements between rotor and casing, dynamic phenomena of the stator part of the bearings or off the casing are included in the above

FRF. A limited set of acceleration pick-ups mounted on the casing support structure indicated high vibration values at several frequencies below 100 Hz when the shaft was excited by the magnetic bearings. To identify these phenomena, a full dynamic survey of the entire installation was set up.

2. DYNAMIC ANALYSIS OF THE PUMP ROTOR

Hammer impact excitation was used to identify rotor resonances in a effective bandwidth from 0 to 800 Hz. The results allowed to adjust some parameters in the analytical models used for the identification algorithms. This model is based on shaft geometry and material characteristics. Up to the 4th bending mode, the agreement between measured and modelled resonance frequencies is excellent. Higher order accuracy is limited by the model discretisation. Additional shaker excitation was used to search for local dynamic phenomena in the impellers and the thrust disc. In the frequency band of interest, no local resonances were found.

3. SIGNATURE ANALYSIS

Signature analysis is the appropriate technique to evaluate the changes of a measured quantity with varying rotational speed. Phenomena which coincide with a multiple of the rotating frequency are characterised by a peak frequency which varies with speed. Structural resonances appear at a fixed frequency. The acceleration levels show a rise and decline when the shaft speed or one of its harmonics crosses the resonance frequency.

The signature analysis was carried out between 500 and 3000 rpm over a frequency span of 0 to 250 Hz. The speed map included approximately 60 different speeds. Flow and head were continuously adjusted to maintain best efficiency point operation. During all measurements a total of 32 channels was measured simultaneously. Figure 2 shows a typical Campbell diagram. It represents vibration response in horizontal direction on one of the pump pedestals. Different phenomena are observed.

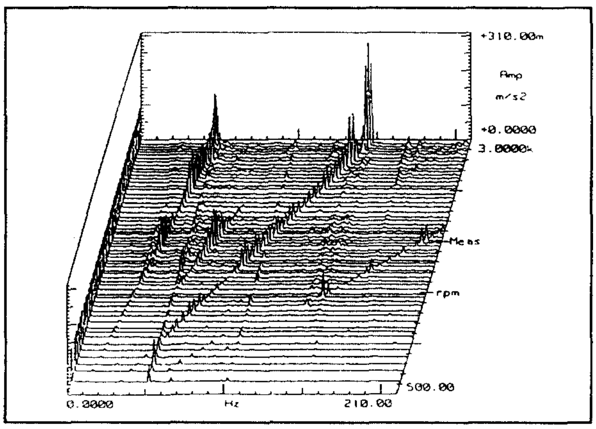


Fig. 2. Campbell diagram for a horizontal pedestal DOF

The first harmonic gets an amplified response around 2000 rpm (33 Hz), next around 2500 rpm (42 Hz) and climbs towards 3000 rpm drastically.

This last phenomena is explained by the first bending mode of the rotor, which descends from 54 Hz for a dry, non-rotating shaft to approximately 48 Hz for a fully operational pump at 3000 RPM.

Note that these values do not stem from a standard BW/IP pump design. The destaging of the test pump was done to create volontarily high dynamic response of the shaft.

The second order response is much smaller. It shows a distinct amplification around 1900 and 2000 rpm (65 Hz). The same 65 Hz is also slightly excited by the third harmonic. This harmonic is the vane frequency. It shows an increase around 1750 (87 Hz) and around 1800 and 3000 rpm (140 and 150 Hz). The latter corresponds again with a wet rotor resonance. As the pump is driven over a fluid coupling, the motor runs continuously at 3000 rpm, explaining the fixed 50 Hz component and harmonics.

Although the nature of excitation is completely different from the artificial excitation used in the actual test program, this survey of operational vibrations at different speeds confirms the presence of structure borne resonances in the frequency range of interest.

4. MODAL ANALYSIS

To identify the resonances indicated by the FRF measurements and the signature analysis, a full modal analysis from 0 to 200 Hz was performed on the testrig. In total about 300 points were measured, for 3 force inputs. 6 modes are identified which cause important motions of the pump casing (if excited) and can also be observed in the signature analysis. Table 2 lists the frequencies and the corresponding modeshape description.

Table 1. Important pump casing and skid resonances

Frequency Hz	Damping %	Description
34.4	3.4	1st bending skid XZ plane
63.0	0.9	rotation pump around X
79.6	2.0	rotation pump around Y
137.7	0.6	rotation pump around Z
146.1	0.7	skid 1st torsion
174.3	0.7	skid 2nd bending/torsion

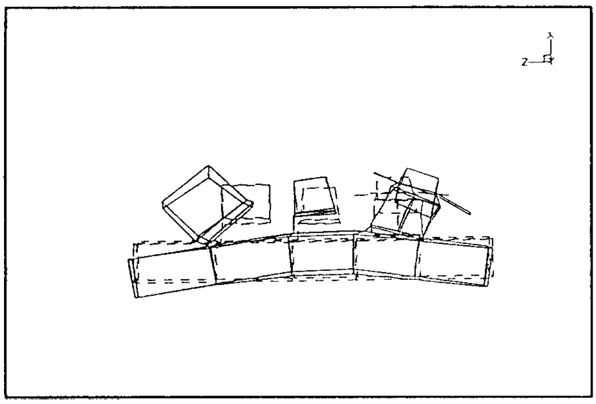
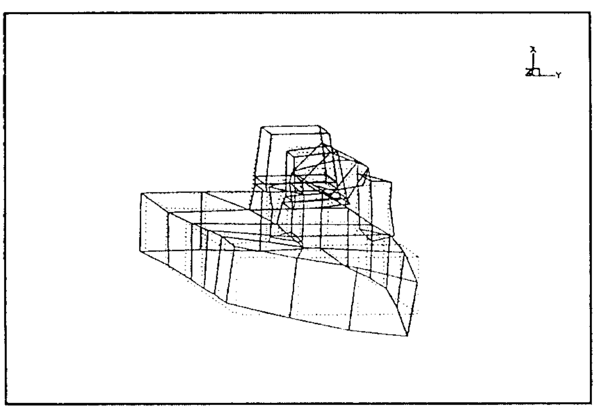


Figure 3 shows the first important modeshape. The skid has a first bending mode, but also the connection of motor and fluid coupling to the skid is a weak element which amplifies the vibration response. The pump pedestals show local flexibilities at the connection to the skid. The pedestal itself remains stiff. At 63 Hz the pump rotates around the vertical axis. The implies that the pedestals show an individual shear / torsion deformation.

Figure 3. 1st bending skid XZ plane

This mode is found to be excited very well by the magnetic bearing forces during FRF measurements. In summary the modal analysis reveals pronounced flexibilities in the interfaces motor - skid, fluid coupling, - skid, pedestal -skid and pump skid. The skid itself has several pronounced modes which interact with these local flexibilities. Elimination of the structural resonances by modification of the pedestals is impossible. From the nature of the modeshapes it is clear that some modifications could result in a limited shift of the eigenfrequencies. Structural modifications towards complete elimination or resonance shifts to frequencies above 200 Hz are not realistic.

5. RUNNING MODE ANALYSIS



The modal analysis results in a description of the dynamic behaviour of the testrig in terms of eigenfrequencies, damping values and modeshapes. The relative importance of every resonance is determined by the type of excitation present during stable operation at maximum speed. To identify which modes are excited by the normal pump operation, a running mode analysis is required.

Fig. 4. 148 Hz deformation pattern during operation at max. speed, 100% BEP

Running mode analysis allows a comparison of vibration amplitudes at different frequencies and an animated representation of the actual deformation pattern at every frequency, based on measurements in the same points as the modal analysis.
Apart from 1 x RPM and 3 x RPM peaks, peaks occur at frequencies corresponding to the eigenfrequencies as found in the modal analysis. Comparison of the deformation pattern with the modeshapes confirms the excitation of structural resonances during normal operation. For this particular mode of operation, the first torsional mode at 146 Hz is perfectly excited by the 3 x RPM forces (fig. 4). At lower frequencies, the rotations of the pump casing around the vertical axis at 63 Hz and its radial horizontal axis at 80 Hz are clearly excited. The deformation shapes under operation compared very well with the eigen modes, characterized by the modal analysis.

6. INTERPRETATION AND SOLUTION

The resonances of the assembly are found back in the rotor measurements on the pump because the measured displacement is the relative displacement between the rotor and the housing. A mechanical modification of the installation could be realistic, in a field application where resonances should not coincide with the normal operating conditions. In such cases, one could work on the different interfaces to shift frequencies around.

For this specific installation dynamic effects should be minimized drastically in the 0 to 100 Hz range and preferably even in the 0 to 200 Hz. The motor to skid connection was studied in detail, but the possible modifications were not expected to shift the first mode more than 50 Hz. The 63 Hz mode could also be tackled by increasing the torsion stiffnes of the pedestal by welding plates to it. All of the possible modifications are however only shifting problems.
It was decided to introduce correction measurements to compensate the relative shaft measurements for the absolute stator measurements. This makes the measurement instrumentation heavier, because all displacement sensors must be duplicated with an accelerometer. Note however that this correction of measurements is not fully correct. When doing so, one assumes that the magnetic bearing force acting on the rotor generates only rotor displacements and that the reaction force of the magnet on the bearing housing generates only stator vibrations. In reality, the fluid and seals couple rotor and stator in a dynamic way. This coupling is neglected in the correction and generates a residual error. Interpretation of the error is made possible by the modal model that is available now.

7. SOUND INTENSITY MEASUREMENTS

Apart from the unexpected dynamic behaviour that was found during artificial excitation, high noise levels were perceived during normal operation. To quantify and separate the contribution of the pump, the fluid drive and the motor to the total sound power, a sound intensity map of the installation is made. Therefore, 4 rectangular box-shaped measurement surfaces are chosen around motor, fluid drive, pump and valve. A 5th surface is created around a 1m long piping section downstream the discharge valve. To facilitate the scanning, a grid of strings is constructed around the equipment, defining 272 squares with a surface of 0,5m x 0,5m. The skid surface included in the measurement volume is highly reflective and has very little own contribution to the sound power. The skid side panels are not included in the measurement volume.
To measure the sound intensity, the sound intensity probe is scanned over each of the defined squares at a constant rate, yielding a time and spatial averaged value for each grid element. Every sweep takes about 1 minute. To improve the dynamic capability of the measurement chain, a phase error compensation is used. All measurements are performed using phase corrected signals and A-weighting. Good measurement quality is reached over the full frequency range.
Using the measured spectra and the geometry information from the LMS CADA-X database, following sound power values were obtained :

Table 2. Sound power dB(A) ref. 1.e-12 W for different components

Octave band Hz	Pump	Coupling	Motor	Valve	Tube 1 m
125	84.6dB+	72.9dB+	78.2dB+	71.7dB+	69.7dB-
250	93.5dB+	77.2dB-	92.8dB+	85.0dB+	84.0dB+
500	99.9dB+	93.0dB+	96.9dB+	94.5dB+	95.5dB+
1000	100.6dB+	105.7dB+	99.9dB+	100.1dB+	102.3dB+
2000	97.1dB+	102.5dB+	101.1dB+	95.2dB+	99.2dB+
TOTAL	104.6dB+	107.6dB+	104.7dB+	102.2dB+	104.6dB+

Following phenomena can be observed :

a) The fluid coupling has the highest contribution to the sound power. The energy is concentrated in the 1000 Hz octave band and stays high above this frequency.
b) At frequencies below 250 Hz most of the sound energy is generated by the pump. In between 250 Hz and 1000 Hz all components are
c) The sound power generated over a 1m length piping section close to and downstream the discharge valve is as high as the total sound power generated by the pump.

To investigate the high sound power contribution of the fluid drive, the sound intensity spectra at the surfaces in between the fluid drive and the motor are calculated, as well as the global averaged intensity value over the complete fluid drive surface. In fig. 5, the frequency distribution of the averaged sound intensity measured in between the coupling and the pump is shown. One can clearly observe the harmonic nature of the spectrum. Although the basic frequency does not show up as an important peak, very high levels are found at 3th and 4th harmonic of 273,5 Hz. All other harmonics are clearly present too.

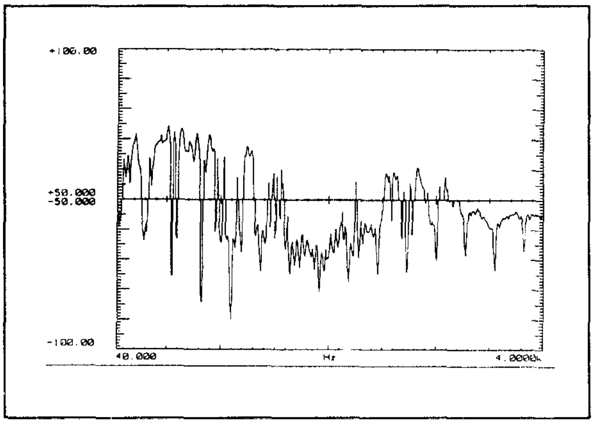


Fig.5. Averaged sound intensity from pump to fluid drive (top) and from fluid drive to pump (bottom).

The origin of the harmonic sound spectrum can be found in the fluid drive mechanics. At the operating conditions during the measurements, the tooth meshing frequency was 2740 Hz, which is exactly 10 times the "fundamental" frequency found in the sound spectra. Different gear wheel faults can generate low frequency sidebands. A close investigation of fig. 5 reveals increased levels at sidebands, spaced with the input shaft speed below and above the tooth meshing frequency and its harmonics, which confirms the gear deterioration hypotheses. The vibrational energy generated by the fluid drive mechanics is transmitted to the fluid drive housing. The thin plate sections of the housing transmit this energy by radiation of sound power. For the 821,9 Hz peak (3th harmonic) the highest intensity levels are found at the motor side, while for the 1093 Hz peak (4th harmonic), the pump side of the fluid drive generates most of the noise.

For both frequencies, the highest contribution to the total noise level is found at sections of the fluid drive which are free of auxiliary equipment. To reduce the sound power radiation from these surfaces, the mechanical impedance of the plates can be modified by covering them with commercially available damping layers. As only a limited part of the housing has to be treated, no cooling problems will occur. A close investigation of the fluid drive mechanics is advised to reduce the mechanical vibrations.

8. ADDITIONAL INVESTIGATIONS

A series of additional measurements are performed to investigate the dependency of the operational vibrations with suction pressure. Detailed pressure pulsation measurements combined with air injection testing resulted in the identification of acoustic resonances in the piping system. These resonances are excited by turbulent flow and cavitation introduced by the main valve and by the piping diameter change downstream this valve. An orifice was developed and built in the loop at the low pressure side of the main valve. This allowed to reduce the pressure drop over the valve and to smoothen the flow at the energy of the piping. Verification measurements showed excellent operation of the device, resulting in an important overall noise and vibration level reduction for a wide range of operating conditions.

9. CONCLUSION

The use of the experimental structural dynamic techniques complemented the insight in deviations between a rotor model of a boilerfeed pump and measurements on the pump. A standard troubleshooting approach based on signature analysis, modal and running mode analysis identified structural flexibilities in the connections between the machinery and the skid. Since the low frequencies of the modes, a modification would require a new development of the installation. The solution for the test programme is given by a compensation of the pump dynamics by additional acceleration measurements.
Sound intensity measurements allowed to seperate the fluid drive as the major sound source on the BRITE-EURAM testrig. A narrow band analysis from the noise measurements resulted in the identification of the source of mechanical vibrations. A close investigation of the fluid drive mechanics is advised. Sound intensity maps of the fluid drive showed high sound radiation at those housing sections where no additional equipment is mounted. The application of commercially available damping layers on these housing sections will reduce the overall noise level.
Additional sound intensity measurements of the main valve and a piping section indicated the presence of acoustic resonances which were confirmed by pressure pulsation measurements and air injection testrig. The piping section dowsnstream the discharge valve is modified, which resulted in a reduction of the noise level.

10. ACKNOWLEDGEMENT

BREU 3472-89 is a shared cost research programme partly funded by the European Commission BRITE-EURAM programme for which the authors want to express their gratitude.

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