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A STUDY ON THE STRUCTUREBORNE NOISE OF HYDRAULIC GEAR PUMPS

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ABSTRACT

The main source of vibration and noise in hydraulic systems is the flow ripple at the pump output which interacts with the downstream components and generates pressure ripple in the current, structureborne noise in mechanical parts and airborne noise in the environment. The present research was suggested by the observation that some peaks in the airborne noise spectrum of a specific pump family were within a frequency range far from those typically related with the fluidborne or structureborne noise; unfortunately, in that range the human ear does not filter the noise efficiently.
In general, the investigation of the interaction between fluidborne noise and structurebornenoise can be carried through both "macro-analysis" techniques (sound emission analysis, pressure ripple measurement, modal analysis, etc.) and "micro-analysis" techniques (contact mechanics of gears, performance of the bearing system, etc.). The present research started from a "macro" approach inclusive of the modal analysis of a gear pump, the measurement of its pressure ripple and its acoustic mapping though the sound intensity technique. Currently, the focus is on the "micro" approach and specially on the investigation of the elasto-hydrodynamic bearing performance in view of deriving the influence of the bearing geometry on the overall structureborne noise.

KEY WORDS

Journal bearing, Modal analysis, Sound intensity analysis, Structureborne noise.

NOMENCLATURE

a: acceleration
b: adimensional ratio, $b = \frac{\Omega}{\omega_0}$
b_{ii}: damping coefficient
B_{ii}: dimensionless damping coefficient
c: radial clearance
$f(t)$: function of time
F_n : generic harmonic function
F_m : mean value of $f(t)$
F(Ω) : Fourier transform of $f(t)$
$G_{xx}(f)$: auto-spectrum of input signal
$G_{yy}(f)$: auto-spectrum of output signal
$G_{xy}(f)$: cross-spectrum of signal
h : damping adimensional ratio, $h = \frac{r}{r_c}$
H(Ω) : Harmonic Transfer Function
k_{ii}: stiffness coefficient
K_{ii}: dimensionless stiffness coefficient
m : deflected mass
m_a : shaft mass
M : pump mass
r_c : critical dumping
t : time
v: velocity
$\ddot{x}$: acceleration
$\ddot{X}$: acceleration of pump mass

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w_r : resultant load
ϑ_n : phase difference
ω_0 : natural frequency
ω_b : angular velocity
γ_{xy} : coherence function
Φ_l : angle between y axis and resultant load
Ω : angle velocity

INTRODUCTION

The external gear pumps seem to be the assembly of few and relatively simple mechanical parts. In spite of that, the design and analysis of such units is far from being simple, because more functional roles are associated with the individual parts, and consequently the overall behavior is the complex combination of multiple effects. Conversely, the piston units have more mechanical parts, but it's easier to associate a specific issue to a specific pump element, e.g. the pressure ripple to the kidney port or the flexural vibration to the swashplate.
The origin of the present research can be traced back to the analysis of the sound pressure spectrum (Figure 1) of a 12 teeth external gear pump (at different speed and pressure levels): in fact, high noise levels were found in a frequency range (1 – 2 kHz) which is rather distant from the dominant frequencies due to the pressure ripple (i.e. 200 - 600 Hz for a pump running at 1000 rpm). The idea of find the cause of this results brought to investigate the structurebornenoise of the pump and implements two different methods of study:
1) firstly, a number of techniques were applied at the "macro" level, i.e. sound analysis, pressure ripple measurement and modal analysis. The aim was to investigate whether the peaks were originated by a structureborne noise source due to the interaction between pressure ripple and pump structural members;
2) secondly, an analysis at the "micro" level was attempted, focused on a component not much studied by the scholars in this perspective: the journal bearing.

STRUCTUREBORNE NOISE MECHANICS

The pump structureborne noise is important for the same reason why the fluidborne noise is important: it causes other machine structural elements to vibrate and radiate airborne noise [1]. It is caused by all pump structural vibration modes, including those that don't produce sound, and is transmitted by bending or torsional vibrations or by compression or shear waves that travel through solids like sound waves travel through air.
Often the structureborne noise has 1000 times the energy of the pump airborne noise. By converting only a fraction of 1% of the pump structure-born noise into sound, a single responsive part can radiate more airborne noise than the whole pump if such a part is a good sound radiator.

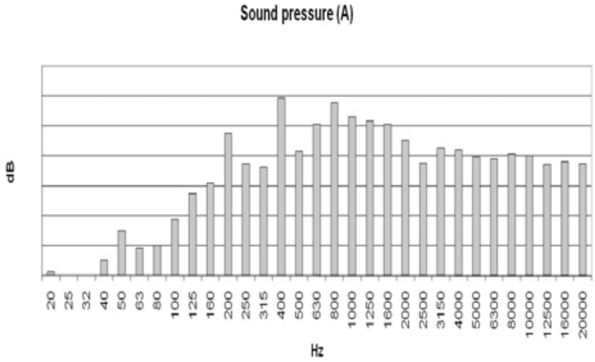

Figure 1 Spectrum of sound pressure level (A weighted) of a 12 teeth external gear pump (1000rpm; 180bar)

Most of the structureborne noise is due to the mechanism shown schematically in Figure 2. The small mass *m* is a part of the pump subject to the internal pumping forces which cause a movement relative to the rest of the pump mass *M* (e.g. the oscillation of the shaft in its bearing). The Newton's second law states that the center of the total pump mass must remain stationary unless affected by external forces, the mass *M* and the mass *m* must move in opposite directions to compensate and maintain the center of the total mass stationary. Since the center of gravity does not move, the larger mass *M* moves a fraction of the distance covered by the mass *m* relative to the center of gravity, as stated in Eq. (1)

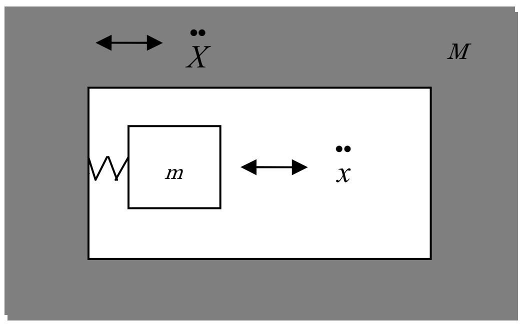

Figure 2 Basic structureborne noise mechanism

$$\ddot{X} = \frac{\ddot{x}\, m}{M} \tag{1}$$

The imbalance of rotating parts has a similar noise potential. Though most of these parts are fully machined, they can work as sources of structureborne noise because the pressure imbalance works on the pump gears.
The structureborne noise is not easy to detect, because

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the original vibration can be in one point of the fluid power plant (e.g. the pump) and then propagate through the circuit until it reaches a good emitter which converts the mechanical energy into acoustic energy, i.e. noise. In practice noise occurs when the energy of a vibrating structure is converted into pressure waves which propagate in air. So, vibration and noise are strictly related especially when resonance takes place, i.e. when the exiting forces work at the natural frequencies (or modal frequencies) of a structure.

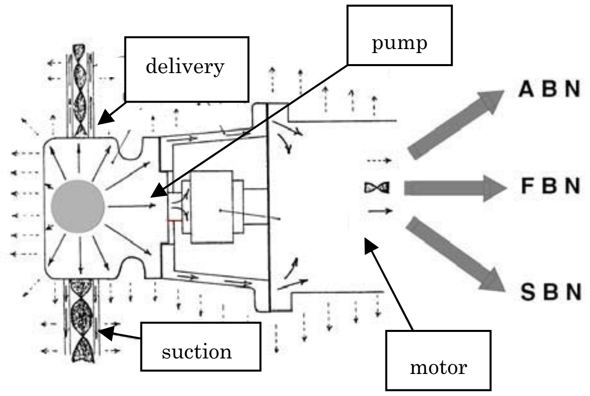

Figure 3 Noise sources in a fluid power pump: ABN (Airborne Noise); FBN (Fluidborne Noise); SBN (Structureborne Noise)

HARMONIC TRANSFER FUNCTION

By mean of the Fourier analysis [3] it is possible to express the cyclic forces as linear combinations of harmonic functions with frequencies that are integer multiples of a fundamental frequency Ω_0

$$f(t) = F_m + \sum_{1}^{\infty} F_{n0} \cos(n\Omega_0 t + \vartheta_n) \qquad (2)$$

Using the linear superimposition principle, the response of a linear system subject to a periodic force can be expressed as the combination of the responses to the harmonic components F_n (t) of the exiting force (the response to a harmonic force is usually defined *harmonic transfer function*). If the force *f(t)* is not periodic, its Fourier transform is to be used, as defined in Eq. (3)

$$F(\Omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} f(t) e^{-i\Omega t} d\Omega \qquad (3)$$

The behavior of a system subject to a generic force can be described by means of the typical "control" approach. From this point of view the force is the input signal and the vibration is the output signal: the vibrating system is a "black box" that receives a generic force *f(Ω)* as input and gives the displacement *x(Ω)* as output (Figure 4).

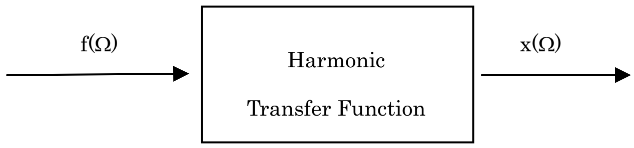

Figure 4 Harmonic Transfer Function definition

The "black box" behavior is fully described by the harmonic transfer function, as shown in Eq. (4)

$$X_p(\Omega) = H(\Omega) F(\Omega) \qquad (4)$$

The results provided by the harmonic transfer function depend on the numerator selected [3] [4]: displacement, velocity or acceleration. Three typical forms of the results provided are:

- Compliance $\frac{x}{F} = \frac{1}{(k - \omega^2 m) + i(\omega c)}$ (5)
- Mobility $\frac{v}{F} = \frac{i\omega}{(k - \omega^2 m) + i(\omega c)} = i\omega h(\Omega)$ (6)
- Accelerance $\frac{a}{F} = \frac{-\omega^2}{(k - \omega^2 m) + i(\omega c)} =$ (7)

$= -\omega^2 h(\Omega)$

MODAL ANALYSIS OF GEAR PUMPS

The modal analysis of the pump was performed by using an impact hammer as input excitation and the pump body reaction as output response: the complex ratio of the two is the harmonic transfer function. The accelerations were measured in different positions: on the suction, output and upper face, and on the back cover. The accelerometer was kept in a fixed position while the hammer hit the case: (a) firstly in the same position; (b) secondly, in all other positions. Once a series of measurements were completed, the accelerometer was moved and the measurements started again. In Figure 5, the spectrum, phase and accelerance of the pump body in one of the different configuration are shown.
The modal analysis proves the presence of a strong resonance frequency in the range of 5 – 7 kHz, clearly far from the peaks found in the range of 1 – 2 kHz, then, as first partial conclusion it can be state: that the noise produced in this range of frequencies probably is not caused by a pump's body resonance.

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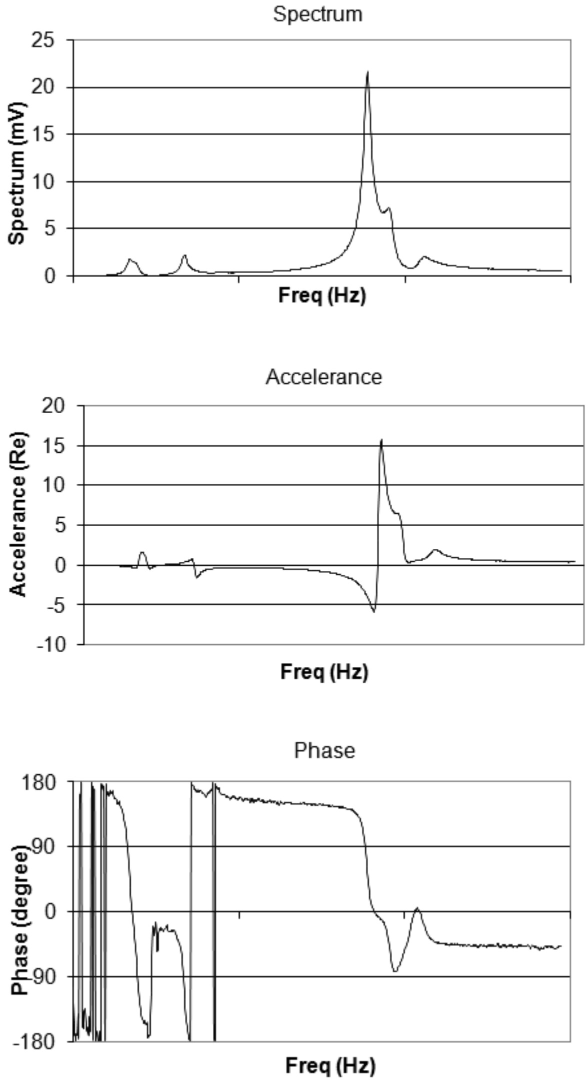

Figure 5 Harmonic transfer function:
Spectrum, accelerance and phase vs frequency

FLUID- AND AIRBORNE NOISE INTERACTION

By mean of the harmonic transfer function it is possible to investigate the interactions between the fluidborne and the structureborne noise, by using the pressure ripple signal as input and the acceleration of the pump body as output signal. The mathematical tool useful to check the reliability of such a relationship is the *coherence function* defined in Eq. (8). If this function is close to one, the output signal is substantially caused by the input signal; conversely, if the function is close to zero, the two signals are not related or linked by a cause/effect mechanism

$$\gamma^2{}_{xy}(f) = \frac{\left|G_{yx}(f)\right|^2}{G_{xx}(f)G_{yy}(f)} \Rightarrow 0 \le \gamma^2{}_{xy}(f) \le 1 \qquad (8)$$

From this point of view, a possible correlation between the airborne, fluidborne and structureborne noise was investigated. A series of measurements of pressure ripple and acceleration were performed at various pressure levels, at various running speed. The coherence function between acceleration and pressure ripple was computed to check the interaction between the fluidborne and the airborne noise. The *mathematical* coherence function proves the absolute deficiency of the *physical* coherence between the two signals in the range 1 – 2 kHz, because the pressure ripple is practically absent above 1kHz, as shown in Figure 6. So, as second partial conclusion, the noise frequency peaks is not due to the interaction between fluidborne and structureborne noise.

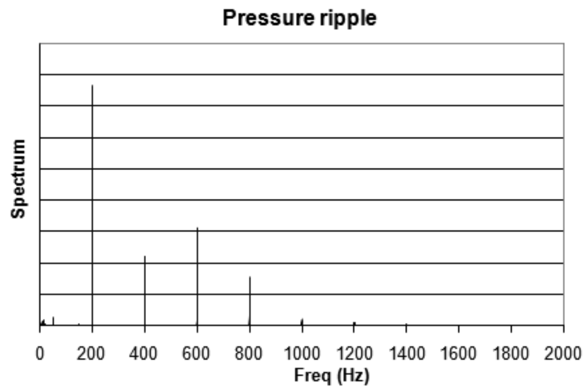

Figure 6 Pressure ripple (1000 rpm, 180 bar)

JOURNAL BEARING ANALYSIS

Finally, an acoustic analysis was carried out by means of the intensimetry measurement technique – a much more accurate method to describe the acoustic field if compared with the sound pressure measurements - and its results were compared with the acceleration measurements on the pump body.
The sound intensity analysis gave the opportunity to verify that the suction side of the pump body surface emits most of the sound power of the pump. For our purposes, however, the best pieces of information were provided by comparing the acceleration and sound power plots.
The spectrum of Figure 7 confirms the presence of two peaks centered at 315 Hz and 630 Hz, due to pressure ripple, but at the same time the sound power peak in the band centered at 2 kHz is of unknown origin.
In fact Figure 8 shows some peaks of the acceleration plot. A more accurate analysis of the acceleration data,

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in the range 1.6 – 2.0 kHz (Figure 9), discovers two lateral bands spaced from the principal one, centered at 1.8 kHz, by a regular interval of 25 Hz, just the same as the shaft revolution frequency.

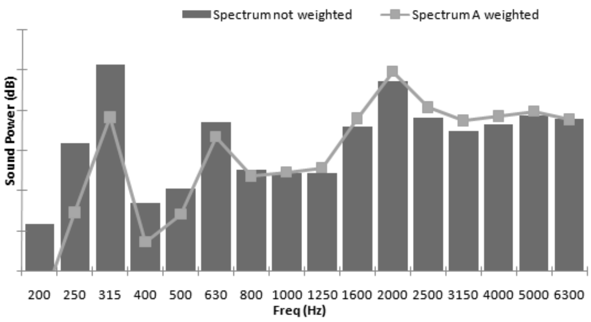

Figure 7 Sound power (1500 rpm, 180 bar)

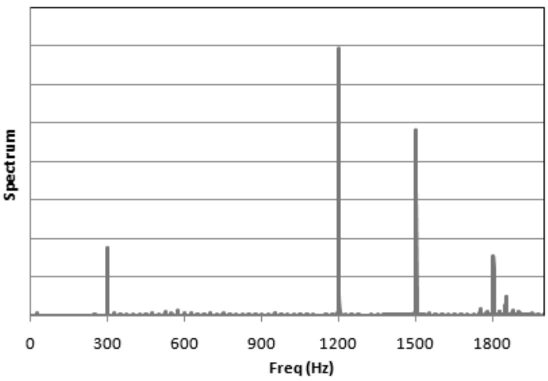

Figure 8 Spectrum of acceleration (1500 rpm, 180 bar)

On the basis of the journal bearing theory [2], it is possible to draw the conclusion that, being the band on the right higher than the other one, an eccentric operation of the shaft could be the origin of the airborne noise peaks.

The eccentricity is a normal operating condition of a shaft supported by a journal bearing (like an external gear pump), but it becomes critical from the structure-borne noise viewpoint if the bearing works in an unstable configuration.

The problem can be studied by applying the fluid film lubrication theory to a shaft of mass $2m_a$ supported by two identical well-aligned journal bearings.

Let the load be stationary, $w_r = w_{x,0}$ and $\Phi_l = 0$; then the linearized equation of the journal bearing is:

$$\begin{Bmatrix} m_a & 0 \\ 0 & m_a \end{Bmatrix}\begin{Bmatrix}\Delta\ddot{x}\\ \Delta\ddot{z}\end{Bmatrix}+\begin{Bmatrix}b_{xx} & b_{xz}\\ b_{zx} & b_{zz}\end{Bmatrix}\begin{Bmatrix}\Delta\dot{x}\\ \Delta\dot{z}\end{Bmatrix}+\begin{Bmatrix}k_{xx}&k_{xz}\\k_{zx}&k_{zz}\end{Bmatrix}\begin{Bmatrix}\Delta x\\ \Delta z\end{Bmatrix}=\begin{Bmatrix}0\\0\end{Bmatrix} \tag{12}$$

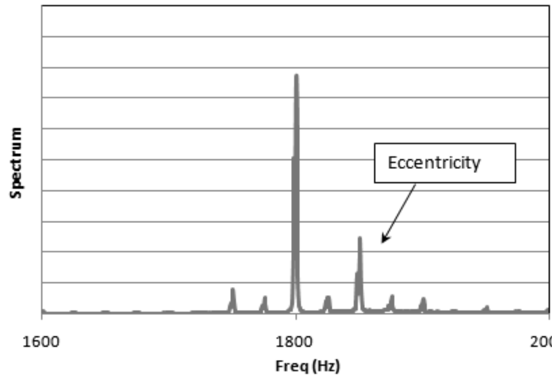

Figure 9 Spectrum of acceleration (1500 rpm, 180 bar)

The stiffness and damping coefficients can be made dimensionless through the following equations:

$$\begin{Bmatrix}\mathrm{K}_{xx} & \mathrm{K}_{xz}\\ \mathrm{K}_{zx} & \mathrm{K}_{zz}\end{Bmatrix}=\frac{c}{\omega_r}\begin{pmatrix}k_{xx} & k_{xz}\\ k_{zx} & k_{zz}\end{pmatrix} \tag{13}$$

$$\begin{Bmatrix}\mathrm{B}_{xx} & \mathrm{B}_{xz}\\ \mathrm{B}_{zx} & \mathrm{B}_{zz}\end{Bmatrix}=\frac{c\,\omega_b}{w_r}\begin{pmatrix}b_{xx} & b_{xz}\\ b_{zx} & b_{zz}\end{pmatrix} \tag{14}$$

The substitution the Eqs. (13) and (14) into Eq. (12) gives:

$$\begin{Bmatrix} m_a & 0 \\ 0 & m_a \end{Bmatrix}\begin{Bmatrix}\Delta\ddot{x}\\ \Delta\ddot{z}\end{Bmatrix}+\frac{w_r}{c\,\omega_b}\begin{Bmatrix}\mathrm{B}_{xx} & \mathrm{B}_{xz}\\ \mathrm{B}_{zx} & \mathrm{B}_{zz}\end{Bmatrix}\begin{Bmatrix}\Delta\dot{x}\\ \Delta\dot{z}\end{Bmatrix}+ \\ +\frac{\omega_r}{c}\begin{Bmatrix}\mathrm{K}_{xx} & \mathrm{K}_{xz}\\ \mathrm{K}_{zx} & \mathrm{K}_{zz}\end{Bmatrix}\begin{Bmatrix}\Delta x\\ \Delta z\end{Bmatrix}=\begin{Bmatrix}0\\0\end{Bmatrix} \tag{15}$$

The solution to Eq. (15) is of the form

$$\begin{Bmatrix}\Delta x\\ \Delta z\end{Bmatrix}=\begin{Bmatrix}x_h\\ z_h\end{Bmatrix}\exp\left(\bar{\Omega}\, t\,\omega_b\right) \tag{16}$$

The substituting Eq. (16) into Eq. (12) gives

$$\begin{Bmatrix}\mathrm{M}_a+\bar{\Omega}\mathrm{B}_{xx}+\mathrm{K}_{xx} & \bar{\Omega}\mathrm{B}_{xz}+\mathrm{K}_{xz}\\ \bar{\Omega}\mathrm{B}_{zx}+\mathrm{K}_{zx} & \mathrm{M}_a+\bar{\Omega}\mathrm{B}_{zz}+\mathrm{K}_{zz}\end{Bmatrix}\begin{Bmatrix}x_h\\ z_h\end{Bmatrix}\exp\left(\bar{\Omega}t\omega_b\right)=\begin{Bmatrix}0\\0\end{Bmatrix} \tag{17}$$

where

$$\mathrm{M}_a=\frac{c\,m_a\Omega^2}{w_r} \quad \text{and} \quad \bar{\Omega}=\frac{\Omega}{\omega_b} \tag{18}$$

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Thus the solution

$$\left(M_a + \bar{\Omega}\, B_{xx} + K_{xx} \right)\left(M_a + \Omega\, B_{zz} + K_{zz} \right) - \qquad (19)$$
$$- \left(\bar{\Omega}\, B_{zx} + K_{zx} \right)\left(\bar{\Omega}\, B_{xz} + K_{xz} \right) = 0$$

is an eigenvalue problem stating that, if the system should dislodge itself from the steady-state position, a transient vibration would result although the external load be constant.

When the load is very high (as in the case of a hydraulic pump) the lubrication's regime (viscous-elastic) could be influenced by two mayor physical effects: the elastic deformation of the walls and the increase in fluid viscosity with pressure. The lubricating regime could be studied by mean of tree dimensionless parameters :

$$\hat{H} = H_e \left(\frac{W}{U} \right)^2 \quad \textit{film thickness parameter} \qquad (20)$$

$$g_v = \frac{GW^3}{U^2} \quad \textit{viscosity parameter} \qquad (21)$$

$$g_E = \frac{W^{8/3}}{U^2} \quad \textit{elasticity parameter} \qquad (22)$$

These parameter could be used to compute the dimensionless minimum film thickness:

$$\hat{H}_{e,\min} = 3.63 U^{0.68} G^{0.49} W^{-0.073} \left(1 - e^{-0.68 k} \right) \qquad (23)$$

To reduce the instability, which is found particularly in the cylindrical journal bearing, a lot of efforts have been spent , but a universal solution is still (and far) to come. Since for some applications, new internal profiles have been proposed (Figure 10), some similar design options are under evaluation to check if the journal bearings are the true sources of the anomalous noise levels.

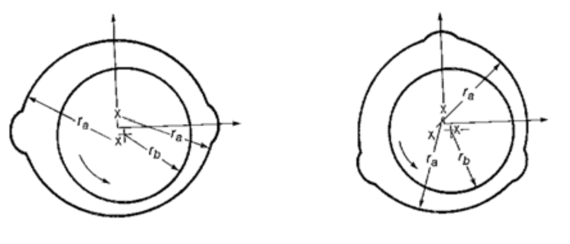

Figure 10 Journal bearings modified to reduce the instability (examples)

CONCLUSIONS

A study on structureborne noise in hydraulic external gear pump was carried out starting from the discovery of a high sound pressure level in a frequency range (1 – 2 kHz) which is far from those typically related with the fluidborne noise. A series of experiments were planned and evaluated:

- a modal analysis was used to derive the resonance frequencies of the pump body. Occurring in the range of 5 – 7 kHz, they are definitely far from the range of interest;
- the interaction between the fluidborne and structureborne noise was checked by applying the standard coherence function. The result was an absolute lack of coherence between the two signals in the range 1 – 2 kHz, because the pressure ripple is practically absent above 1kHz;
- finally the attention was focused on the journal bearings of the wheeled gears. Analyzing the acceleration spectra were find the presence of the two lateral bands spaced from the principal by a regular interval of 25 Hz, equal to the frequency of revolution. The journal bearing theory suggests that an eccentric operation of shaft could be responsible of the noise peaks in the range of 1 – 2 kHz. The same theory is useful to propose alternative bearing configurations to reduce the shaft unbalance.

The work under development now is focused on the experimental analysis – and hopefully validation - of a few newly designed journal bearings.

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