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NEW HOUSING FOR REDUCTION OF NOISE FROM LARGE INDUSTRIAL GEARBOXES

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INTRODUCTION

Large industrial gearboxes are often very noisy and typically have service lengths of several decades. Newer gears can be made less noisy, but unfortunately they sometimes become even more noisy as new materials have been used in the gear rim, allowing more power to be transmitted per unit area.

One major source of noise in ore concentration plants are the gears, transmitting very high torques at mills that are grinding the mineral. In most cases the gears will not be replaced only because they are noisy. It is also generally considered far to expensive to regrind them for improved tolerances and thereby lower noise levels. We had to find a cheap, effective and durable method to reduce the noise from existing gears, with minimum interference in the construction and did not obstruct the maintenance. We decided to redesign the upper part of the gear housing to a new one with better absorption and higher reduction of the air-borne sound as well as the structure-borne sound.

EXISTING GEAR HOUSING

The gears at the ore concentration plants are of type Morgårdshammar ZE-1055. The gear housings have a lower part made of cast iron on which shafts and bearings are mounted, and an upper part welded in 12 mm steel. The upper part radiates most of the sound. The levels of vibration are more than 10 times greater on the upper steel part than on the lower more heavily damped cast-iron part.

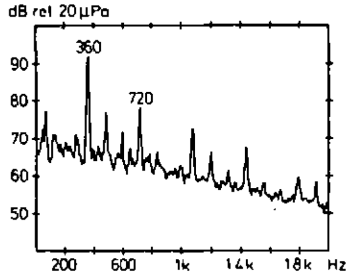

Figure 1

The sound pressure levels at a distance of 1 meter from the existing gears are about 94 dB(A), with a dominant pure tone at the gear mesh frequency of 360 Hz, shown in figure 1.

THE GEAR HOUSING

General design

The lower cast-iron part of the housing has been left in it's original form. The purpose of the existing upper steel part is to protect the gears and keep the lubrication oil in place. We have redesigned the upper part to

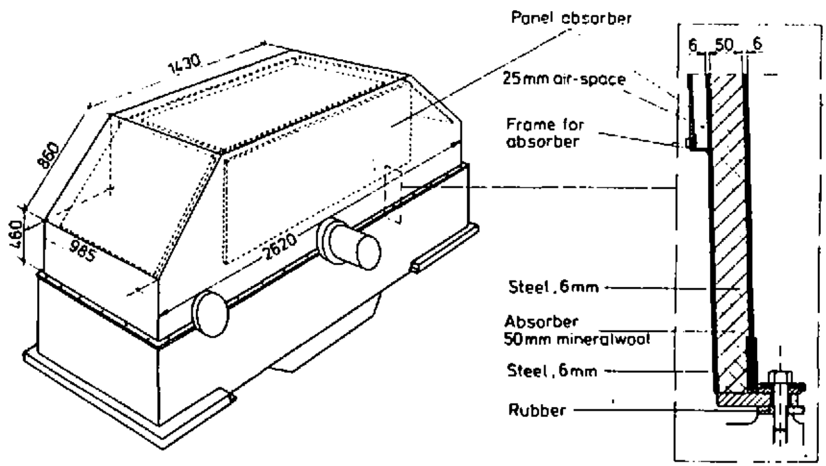

Figure 2 The new gear housing with double-wall and panel absorber.

also reduce the noise generated, see figure 2. The new upper housing is made of 2x6 mm steel with a 50 mm air-gap, filled with absorptive material for effective reduction of the air-borne sound. It will be mounted elastically, on rubber, to the lower part with the shafts, to reduce the structure-borne sound transmitted to it.

Panel absorber

Very large sound pressure levels are generated in the existing small and hard enclosure of the gear. Absorption is needed. It is impossible to employ the commonly used porous material in the hot and oily environment which exist inside a gear housing. We have therefore designed and conducted laboratory tests on a non-pourous panel absorber, made of 0,5 mm aluminium sheet and 20 mm air-space. A panel absorber can be regarded as a damped oscillator with one d.o.f. The air-space acts as the spring (with a spring impedance of $1/j\omega C_a$) and the aluminium sheet as the mass (with a mass impedance of $j\omega M_a$). R_a represents the loss of the system. The acoustic impedance (Z) of this system can be defined by the equation:

$$Z = R_a + j(\omega M_a - \frac{1}{\omega C_a}) \quad \text{where}$$

$$M_a = \frac{m}{S^2} \qquad C_a = \frac{Sd}{\rho c^2}$$

$\omega = 2\pi f$, f = frequency (Hz)
m = surface mass of absorber (kg/m^2)
S = surface area of absorber (m^2)
d = depth of enclosed air-space (m)
c = phase speed of sound (m/s)
ρ = density of air (kg/m^3)

Resonance of the system occurs when the reactance vanishes at the resonance frequency; that is when

$$f_o = \frac{c}{2\pi} \sqrt{\frac{\rho}{md}}$$

The energy loss of the system at resonance is large enough to provide absorption of the sound pressure field. The absorber has been designed to have its resonance frequency f_o at the gear mesh frequency of 360 Hz. This occurs for d = 0.025 m and m = 26 kg/m^2 (Al, 0,5 mm) for the resonator.

It is impossible to measure the absorption coefficient inside a small hard enclosure like a gear housing, since the measurement always will be impaired by a standing wave pattern. We therefore conducted laboratory tests, measuring outside a full scale model of the housing, in a reveberation room. The tests have shown a reduction of 6 dB at the resonance frequency, shown in figure 3. Since the gear mesh frequency dominates the gear noise spectra so heavily, a reduction of this frequency of 5 dB will result in almost the same reduction of the dB(A)-level.

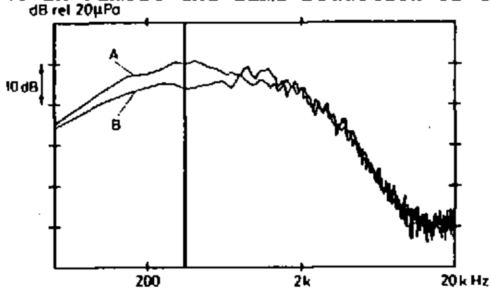

Figure 3 White noise in reveberation room
Upper line, A - without absorber
Lower line, B - with absorber

CONCLUDING REMARKS

The new elastically mounted upper housing with double-wall and panel absorber will reduce the noise level by 10-15 dB(A), compared with the existing housing. The construction of the new housing by Morgårdshammar commenced at the time of writing this paper. It will be tested on a gear at our reference plant, Boliden Laisvall. The background noise level around the gearboxes at the plant is high. However, we will try to resolve the problem of measurement of the reduction of gear noise by measuring coherent output power using our 2-channel FFT-analyzer, and present the results at the conference.