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NOISE REDUCTION FROM GEARBOXES

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INTRODUCTION

Noise is usually a problem in underground working, particularly because the area is hard walled confined space and it is difficult to apply acoustic absorption. Amongst the elements contributing to this noisy environment are gearboxes which are used extensively in the mining industry. Reducing noise levels from these gearboxes is needed. Although gearbox noise has received considerable attention over the past few years the main emphasis of the work has been the reduction of mesh excitation levels which was originally felt to be the most desirable approach, since maximum benefit was expected to result from efforts directed at the source of the problem [1,2]. Unfortunately, tooth profile accuracy requirements demanded by the excitation reduction criteria appeared to be beyond economically achievable tolerances. It is known that the non-uniformity of the torque due to transmission error will give rise to vibrations in the gearwheels in the shafts, and in the gearbox casing which supports the shaft housings, see figure 1.

The authors believe that attention to the gear case is more likely to give major improvements in reducing noise from gearboxes. Although the load carrying capacity of the modern gearboxes has improved enormously in the past few years because of the advances made in stress analysis, there has been no parallel advance made in noise reduction, due mainly to the lack of a suitable systematic technique that could be used at the drawing stage. As a result, a program has been put forward where such systematic technique can be used either at the drawing stage or to treat an already existing model. This technique relies on predicting the magnitude and direction of the forces transmitted to the different bearing housings of a gearbox case. These forces can subsequently be used to excite a finite element model of the case. The mean square volume (MSV) of air displaced by the deflection of the outer walls of the case is minimised with respect to the thickness of the case. Necessary steps can be taken to partially or fully modify the casing thickness where appropriate. It has been found that the MSV is a good criterion for judging whether a structural change will reduce or increase the noise. This method was successfully applied to engine structures [3]. This paper is concerned mainly with the prediction of the forces transmitted to the bearing housings. A general theory for predicting the magnitude and direction of the forces transmitted to bearing housings is described.

THEORY

Power transmitting units are generally complex in nature, depending on the type of gears used and their shaft lay-out, coupled lateral-torsional-axial vibration modes may exist. Figure 2 shows a typical multi-stage gearing system. To study the dynamic behaviour of a such multi-stage gearing system, consider one stage at a time. By assuming the system to behave linearly (though non-linearity can be included), the modelling consists of:

a) Shafts and gears: these are modelled by equivalent discrete and lumped masses at the appropriate nodes along the shaft length. These masses are inter-connected by springs which represent different shafts stiffnesses.
b) Gear teeth: meshing gear teeth are represented by equivalent springs.
c) Bearings: radial, lateral and axial stiffnesses of the bearings are represented by equivalent linear springs.

Figure 3 shows a mechanical model of a one stage involute helical gear unit, (this model was chosen because it involves most of modes of vibration that might exist). although different types of gearing system such as bevel and spur gears can be equally used. Depending on the type of the gearing system, the number of active degrees of freedom per node varies. In this particular case, five active degrees of freedom are considered. These are three displacements and two rotations. The displacements being equivalent to two bi-lateral movements (bending) and one axial movement (thrust); while the two rotations represent the rotational movement of the gear (torsion) and tipping respectively. By considering each degree of freedom in turn, it can be shown that the system of differential equations governing a multi-stage gearing system is of the form:

$$[M]_r \{\ddot{q}\}_r + [K]_r \{q\}_r = \{F\}_r \qquad (1)$$

where $[M]_r$ and $[K]_r$ are the mass and stiffness matrices respectively
$\{q\}_r$ = generalised coordinates which represent degrees of freedom of the same kind for the whole system. For instances, in the case of rotational degree of freedom $\{q\}_r$ will be given by:

$$\{q\}_{\theta_x} = \{\theta_{11}\ \theta_{12} - - \theta_{ij} - - \theta_{nm}\}^T$$

where,
θ is the rotational degree of freedom.
$i = 1,2, \ldots, n$ shaft number under consideration.
$j = 1,2, \ldots, m$ gear number mounted on shaft i.
$r = 1,2, \ldots, 5$ indices represents different degrees of freedom.
$\{F\}_r$ = internal and external excitations. External excitations being the forces due to torques applied to the gearbox, while the internal forces being the excitations generated at the meshing gear teeth.

By partitioning the internal excitations into two parts, one being due to the elastic deformation of the teeth in mesh and the other being due to tooth errors; equation (1) transforms into a set of differential equations representing a coupled multi-degrees of freedom system, with its right-hand side being expressed in terms of external forces and forces generated at meshing teeth and which are due to tooth errors. Possibly the most efficient way of solving this dynamic system is to use finite elements analysis. Bearing forces will be amongst quantities to be extracted from solving this system.

Application
To predict the magnitude and direction of the forces transmitted to the bearings, the theoretical model above was applied to a one hundred horsepower heavy duty mining gearbox, with an overall reduction of 22:1. This is achieved by three reduction stages, figure 2, one being bevel gear set to rotate the drive through 90°, while the remaining two stages are made of helical gear sets. The shafts are carried mainly by spherical and cylindrical roller bearings. The input shaft runs at 1500 rpm. Due to the robustness of the gear system, the following

assumptions were made : Bi-lateral movements; tipping and static transmission error were neglected. The dynamic system reduces to pure torsional problem. Furthermore, due to high torsional stiffness of the shafts and gear teeth; the problem reduces to a rigid body problem. Allowing for smooth transfer of power between gears, the forces transmitted to the bearing housings were computed in terms of the applied input torque to the gearbox. Figure 4 shows the angular variation (direction) of these forces which are transmitted to the different bearing housings. Note that these shown angular variations are in the plane of the bearings and that the axial direction is not shown here. From results shown, the transmitted static forces, in the plane of the bearings, tend to act in a confined sector, this differs from one bearing to another. Furthermore for each bearing, there exists an upper limit whereby it does not matter by how much the input torque is increased, the direction of the transmitted force will not change. To allow for non-uniform transfer of torque between gears, which is due to elastic deformation of gear teeth under load and to tooth profile effects which produce dynamic tooth forces at the gear mesh frequency and its harmonics, a fluctuation in input and output torques were added to the simplified model. In other words, the model is excited by static and dynamic loads. As a result of this, the total forces transmitted to the bearing housings will fluctuate in magnitude and direction about their static components. For the purpose of noise reduction from the gearbox, one can then assess the influence of these total transmitted forces on the gearbox case by using finite elements modelling in conjunction with optimisation techniques. For gearing systems where simplifications to the model can not be allowed, the full theoretical analysis has to be used, that is solving the coupled set of differential equations.

Vibration characteristics of gearbox

a) Empty case

Experiments were conducted on the case (without internal parts) of the gearbox described above, figure 2, to determine its natural frequencies and the corresponding mode shapes, figures 5 and 7. From finite elements modelling confirmed by experimental modal analysis, two types of modes of vibration were found to exist. In the first type, the gearbox case behaves as if it were an elastic solid, whereas in the second type, at higher frequencies, each cast panel of the case vibrates independently. Furthermore, these two types of modes of vibration tend to overlap. It is also found that the top and bottom panels of this gearbox case respond severely to the two types of modes of vibration. This is because they cover a large area of the gearbox case and most importantly they are much less stiffer than the other surface planes where the bearings house. It is also found that these panels are sensitive to the direction of the forces at the bearing housings, figure 5.

b) Running gearbox

Measurements of vibrational level and sound pressure levels (SPL) were recorded from a running gearbox, under typical working conditions, down a coal mine. The gearbox is identical to the one described above, figure 2. The acceleration spectrum which was measured at the centre of the top surface of the gearbox case, was integrated to obtain the corresponding velocity spectrum. The SPL was measured at 0.5m directly above the top surface of the case. Based on the notion that the noise radiated from a structure is proportional to the mean square velocity of its vibrating surface, the SPL and velocity spectrums, figure 6, have the same trend in frequency band of about 750Hz to 2.5KHz. This indicates that the main contributions to the noise radiated from this gearbox are its top and bottom panels. The disagreement of the two spectrums outside the frequency band 750Hz to 2.5KHz, suggests that the low frequency noise can be attributed to other sources of noise in the mine, such as the ventilation system, whereas the high frequency can be attributed to impact noise.

CONCLUSIONS

A general theory for predicting the magnitude and direction of the forces transmitted to the bearing housings of a gearbox is described. Theses forces and their directions were predicted for a mining gearbox using a simplified model. A study of the vibration characteristics of this gearbox show that the noise radiated from it is contributed mainly to its top and bottom panels and that these panels are sensitive to the forces at the bearings.

ACKNOWLEDGEMENTS

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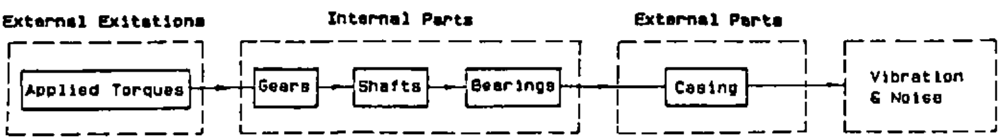

Figure 1

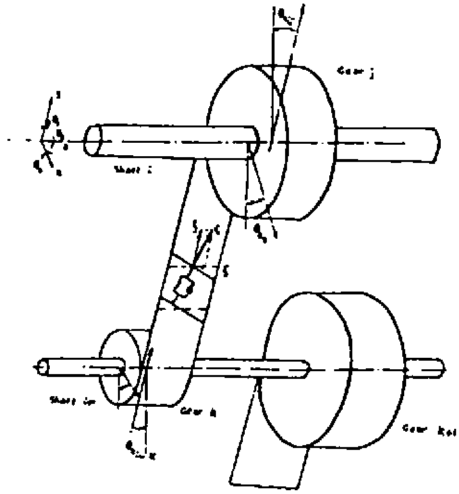

Figure 3: A mechanical model of a single stage involute helical gear unit

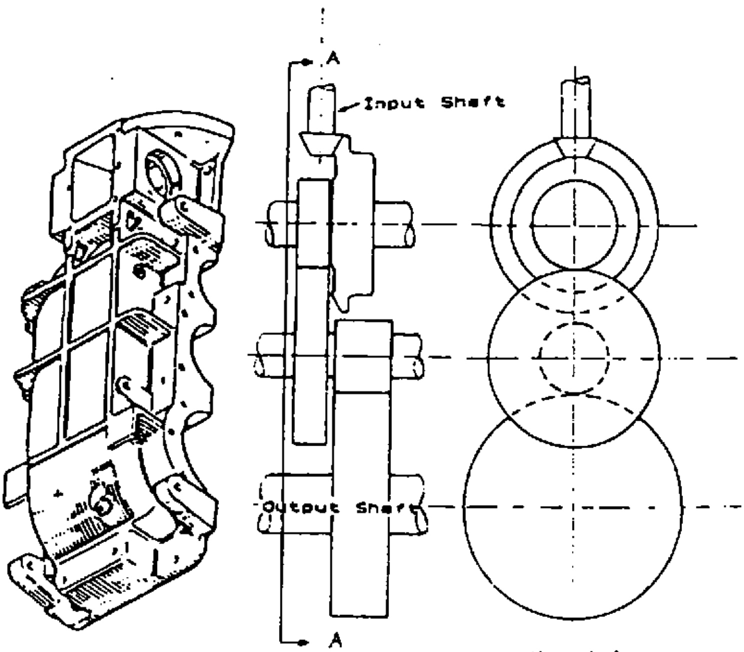

Figure 2: Heavy duty gearbox.

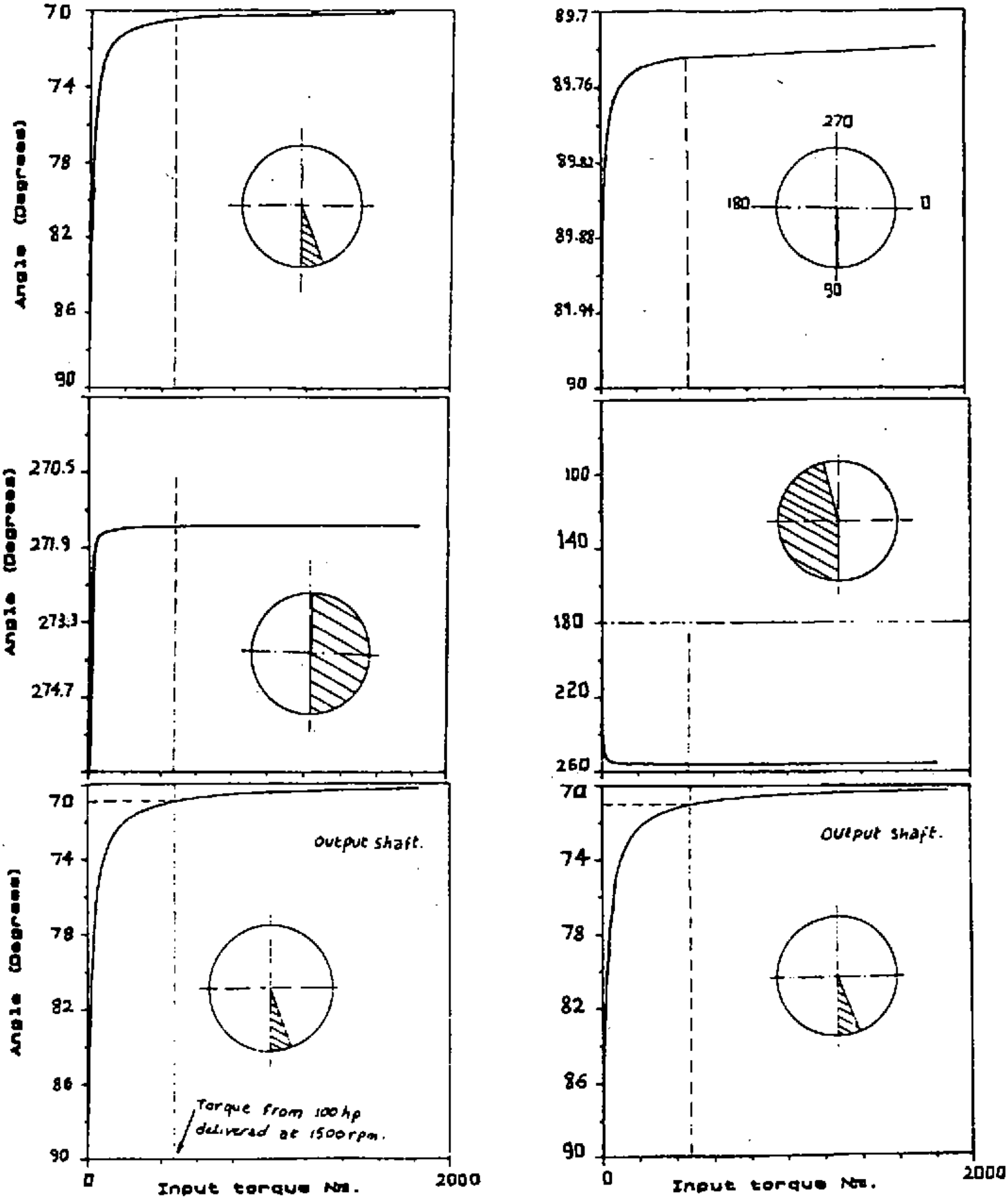

Figure 4: Predicted directions of transmitted forces at the bearings

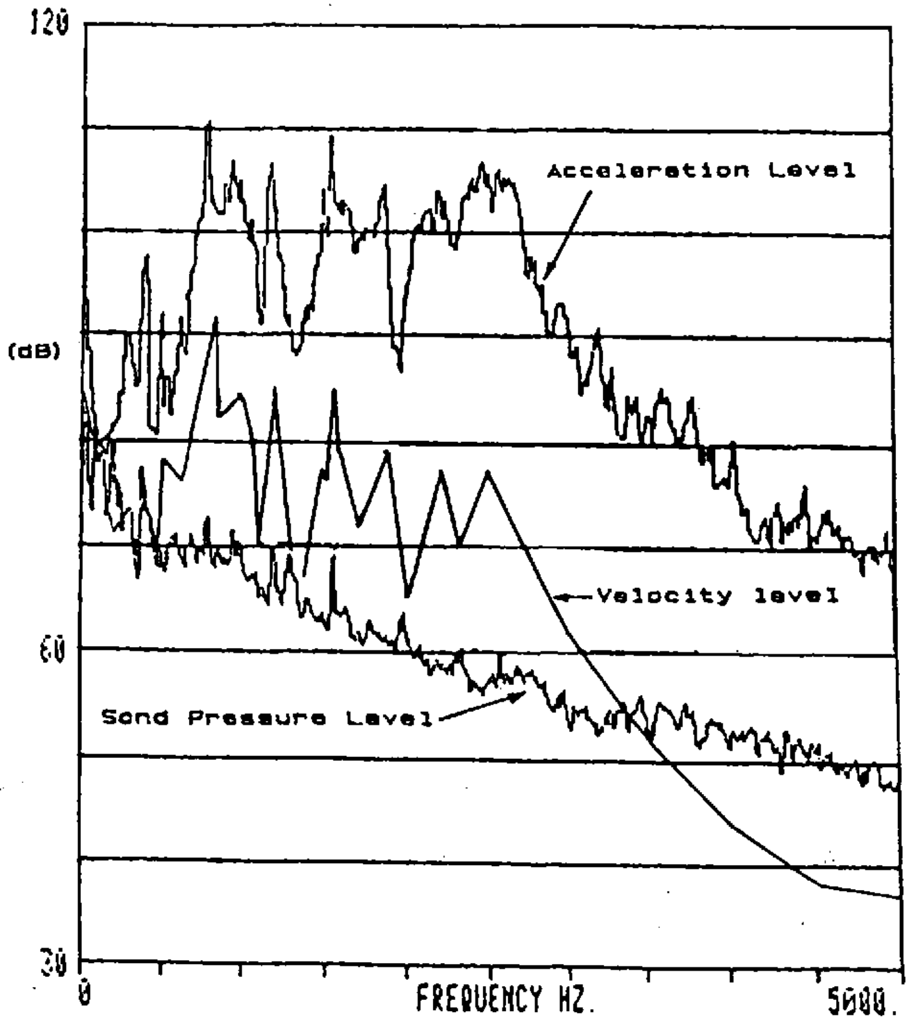

Figure 6: Vibration characteristics of The gearbox

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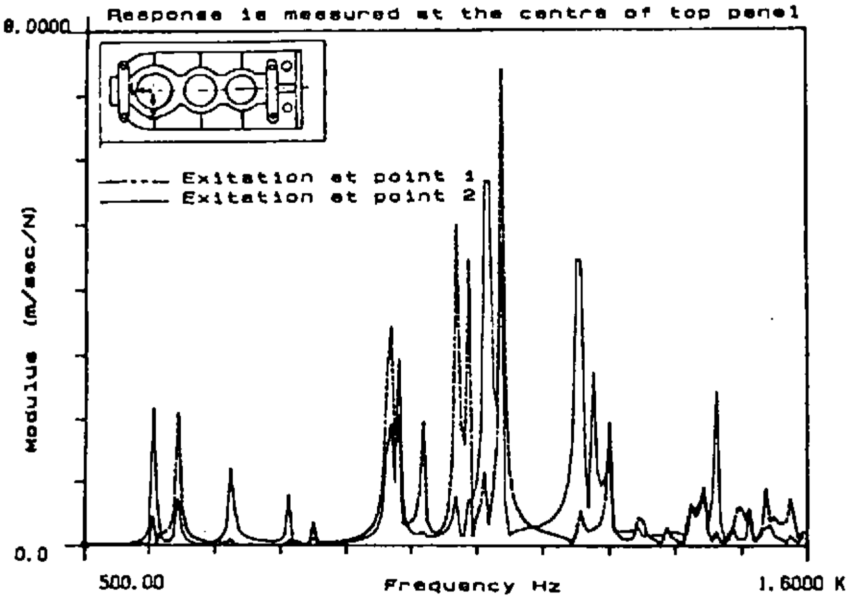

Figure 5: Sensitivity of top panel to forces at the bearings

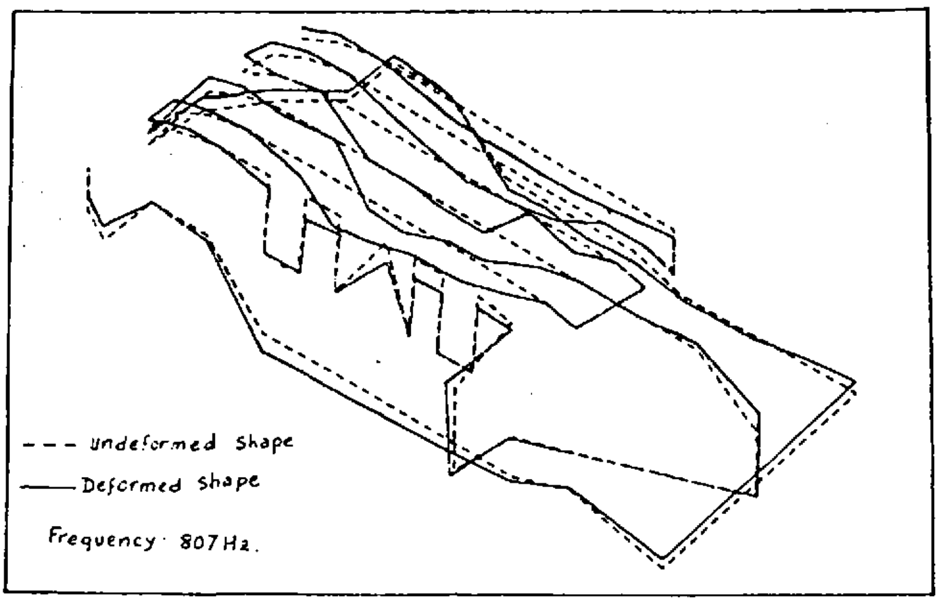

Figure 7: one of the gearbox mode shape