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USE OF AIRBORNE NOISE CALCULATION TO DEVELOP LOW NOISE TRACTOR'S DRIVE SYSTEM HOUSING

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1. INTRODUCTION

The project described in this paper is aimed at acoustic radiation prediction using the Finite Element Method (FEM). As all legal regulations are based on measurements of air borne noise, it has to be a target to include the calculation of air borne noise in the optimization accompanying the design. The traditional sequential development approach that often resulted in costly retrofitting at the prototype testing stage late in the design process is being replaced with process based on computer simulation. Even if the Boundary Element Method (BEM) has established itself as the most important numerical procedure for exterior acoustic calculations [1], the ANSYS Finite Element software offers possibilities to represent the free acoustic field too (infinite space). In practice it is a field in which the effects of the boundaries are negligible over the region of interest. It can be for example in anechoic chamber. That way we are able to model free acoustic field [2].

2. FINITE ELEMENT MODEL

Generally, noise from the drive unit is emitted through the vibration on the drive unit surface [3],[4]. From this reason, our main attention is given only to the drive system housing a mainly to the housing of the gearbox contribution. For modeling of the structure, Fig. 1. the shell element SHELL63 was used. The drive system housing is installed through elastic springs (COMBIN14) for simulation of the investigation of structural and acoustic interaction. The acoustic space at the side of the gearbox wall was modeled as a closed space but with the properties of a free acoustic field. This was done using of the absorption coefficient, $\alpha = 1$ of the inner walls of the closed acoustic space. The acoustic element FLUID30 has been used for acoustic subsystem [5].

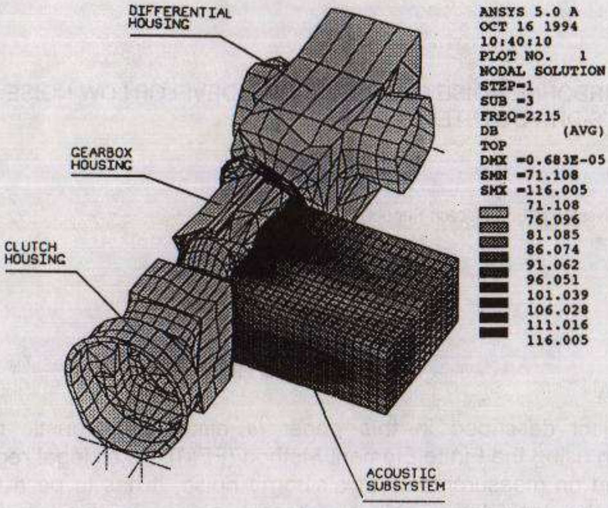

Fig.1 FEM model of the drive unit with acoustic subsystem

3. RESPONSE OF THE ACOUSTIC SUBSYSTEM ON THE HARMONIC EXCITATION

From a spectrum of acceleration measured [6] in a side-direction on the left side of the gearbox housing follows, that an expressive component f=2222Hz contributes the most to the total noise level. It seems to be a resonant frequency. For this reason the mode shape of the drive unit was measured by resonance method and modal analysis of the model of the gearbox also was done. Here, the natural frequency is f=2222Hz during measuring of the real body and f=2215 Hz during calculation of the model [6]. Harmonic response analysis gives us the ability to predict dynamic behavior of the structure and behavior of the surrounding acoustic space, Fig.1., because there is structure and acoustic interaction. Harmonically variable exciting force in side direction was applied in the bearing that is positioned in a diaphragm of the drive housing. The excitation frequency was f=2215 Hz in such a way to correspond with one of the natural frequency of the drive body housing. The nodal acoustic pressures were calculated and the sound pressure level relative to the reference pressure Re=2E-5 Pa was computed at the element's centroids. Fig.1. clearly shows that the highest level of an acoustic pressure of 116.0 dB is in the close distance near the gearbox wall. Its value decreases down to 71.1 dB at the remote wall of the acoustic space.

4. OPTIMIZATION

The goal to lower noise level of mechanical devices is to remove as many resonant regions of the device during running as possible. This reality can be reached by choosing adequate constructional parameters, or by changing them as needed. Fig. 2. shows the frequency range, that was investigated. It is narrow range in the surrounding of the natural frequency that is f=2215 Hz. Original thickness of the wall of the gearbox housing is 0.008 m. From Fig.2. it can be evaluated that noise level in the selected by chance points according to Fig.3. is expressively increasing in the surrounding of the natural frequency. Suitable tuning system can be achieved by changing of the thickness of the wall of the gearbox housing. By application 0.010 m thickness of the wall of the gearbox housing the noise level can be reduced. It is clearly detected from Fig.2. that corresponding levels are bellow the original noise level.

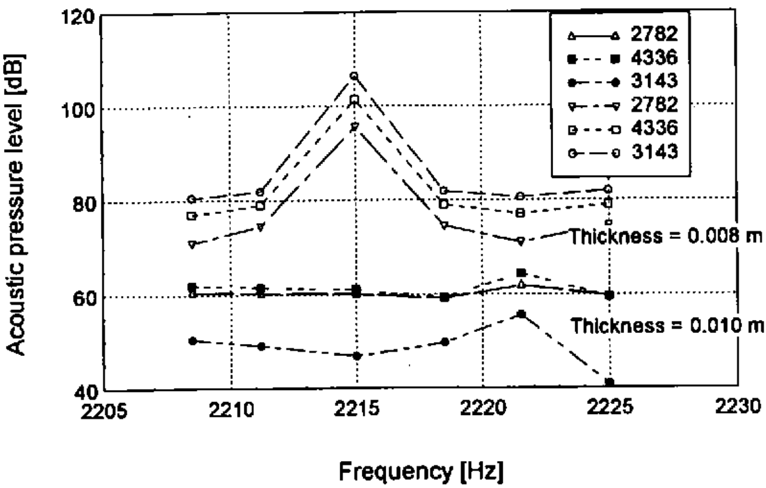

Fig.2 The resonant region 2205 Hz - 2230 Hz

With regard to the lack of knowledge as to the exciting in the real systems, it is clearly not possible to evaluate the absolute noise level near the gearbox model. It is possible to evaluate only the relative decrease or increase of the noise level by some constructional changes.

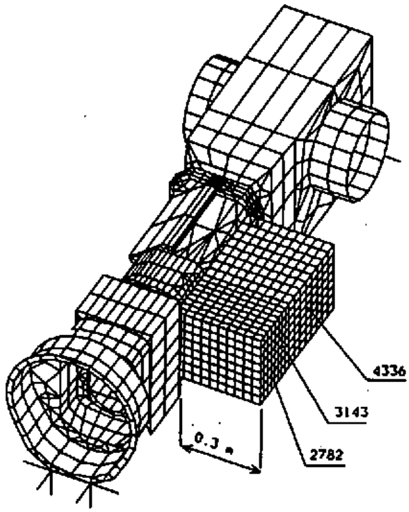

Fig.3 Selected by the chance points

5. CONCLUSION

The model allows to analyse resonance regions of the gearbox and air borne noise calculation. Farther it shows the influence of thickness of the wall of the gearbox with regard to the total level noise of the gearbox.

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