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SOUND FIELD SIMULATION OF THE TRACTOR'S GEARBOX STRUCTURE USING STRUCTURAL AND ACOUSTIC INTERACTIONS

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The ANSYS 5.0 finite element software was used to develop a finite element for the determination of the sound field of the driving unit of a real tractor. Generally, noise from the gearbox structure is emitted through vibration on the product surface and it is necessary to predict correctly the sound field formed by this radiated noise. The sound field exterior environment was modelled with FLUID30 acoustic fluid elements. These elements are most useful for modelling the structural and acoustic interactions. The nodal acoustic pressures of the sound field are plotted as sound pressure contours. The Figures show the sound pressure contours near the vibrating surface and in the plane at the end of the investigated sound field. The paper deal with the modelling of the whole coupled structural-acoustic system of the gearbox in order to reduce the noise emitted from the vibrating surface.

1. Introduction

Gears are sources of structural excitation in mechanical systems. Two types of excitation forces are generated: the first type are forces of gear tooth impact, the second are inertial forces caused by changes in the gear tooth deformation during meshing. Tooth impact, changeable tooth deformations are periodic and are manifested in the form of dynamic forces.

The vibration energy is transmitted to the gearbox housing through the shafts and bearings, and the gearbox housing radiates structure borne noise. The very small pressure fluctuations in the surrounding air constitute a sound field. These pressure fluctuations are usually caused by solid vibrating surfaces. As the disturbance, which produces the sensation of sound, may propagate from the source through any elastic medium, the concept of a sound field will be extended to include structure-borne as well as air-borne noise.

The aim of this paper is to model the Fluid-Structure Interaction and describe those problems in which the structural and fluid responses are strongly coupled. In other words, it is aimed at the acoustic analysis of air-borne noise due to the real tractor's gearbox vibrations. The FEM programme system ANSYS 5.0 was used for solving the mentioned problem.

2. Modelling of the structural-acoustic system

2.1. The finite element model of the driving unit

The drive unit of a tractor is composed of many individual components. The dynamic characteristic of the fully assembled box-type structure system is a complex function of these individual components.

The individual components (Fig. 1) are box-type structures with varying thickness of the walls, flanges and diaphragms with cored holes for the bearings. The individual housings are connected by bolts. The drive unit is installed by elastic springs for simulation of the investigated structural and acoustic interactions.

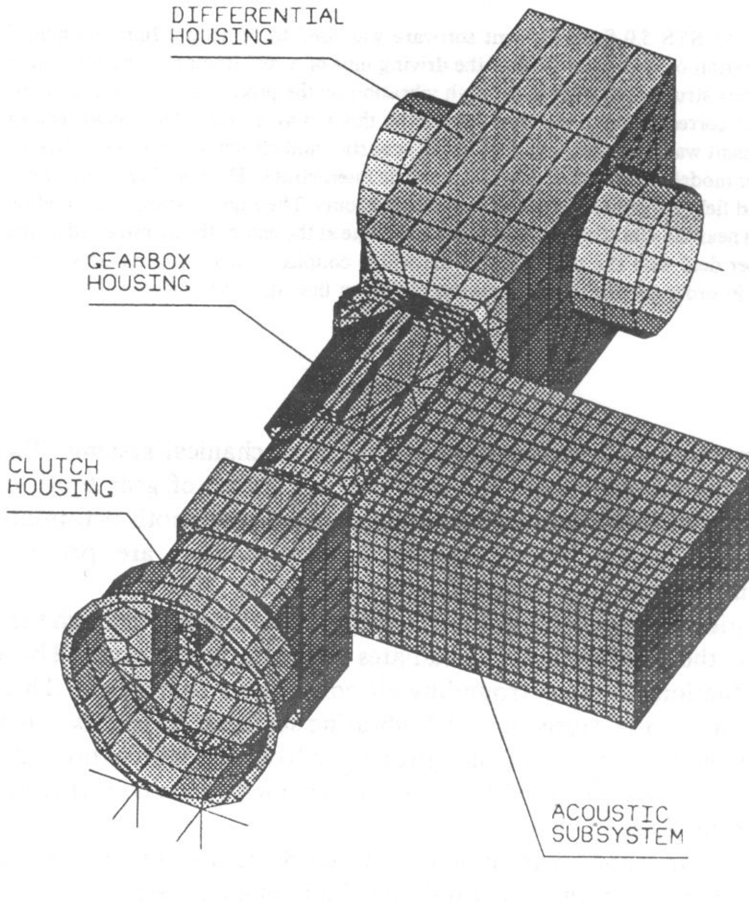

Fig.1 Body of drive unit with acoustic subsystem

The ANSYS finite element software was used to develop a finite element model for the determination of the box-type structures. For the modelling of the structures, the shell element SHELL 63 was used. The walls of the structures are relative thin and the bending moments of these walls are represented more exactly than by using volume elements. Elastic springs were modelled with COMBIN14 elements. More precise meshing was chosen on the gearbox's left side wall. The size of these elements was set at 0.03 m with regard to the acoustic space because this is the wall where the experimentally monitoring of the emitted acoustic energy was done.

2.2. Modellilng of the acoustic field

For the air-borne noise calculation of FE-model was used too. The sound field in the exterior environment was modeled with FLUID30 3-D acoustic fluid elements. They are most useful for modelling structural and acoustic interactions because they contain interfacing fluid elements that are in contact with the solid.

Interfacing fluid elements offer possibilities to represent the free acoustic field [10]. In practice it is field in which the effect of the boundaries are negligible in comparison with the region of interest. It can be, for example, in an anechoic chamber. This way we are able to model a free acoustic field. It is necessary to model the acoustic space as a close space in front of the vibrating wall. This closed space has to have properties of an anechoic chamber. The walls have to absorbe all the sound incident on them. For this reason it is necessary to choose the coefficient of absorption equal one. For a visual evaluation of the influence of the absorption coefficient of the walls in the modelled acoustic space, the 2D model has been used [6].

The size of the aocustic elements was chosen with regard to the wave length which is related to the wave motion frequency as follows:

$$\lambda=\frac{c}{f_{\max}}, \tag{2.1}$$

where $c=343\ \mathrm{ms}^{-1}$ — speed of sound in the air space, $f_{\max}=2500$ Hz — highest considered frequency of the sound wave.

The size of the acoustic element should be $L_{\max}=\lambda/6=0.03$ m.

2.3. The coupled structural-acoustic system

Both subsystems, the structure and the acoustic exterior environment are in mutual contact through the left side wall of the gearbox housing. The matrix equation of motion of the coupled structural-acoustic system is then [1]

$$\begin{bmatrix} M_s & 0 \\ \rho_0 A^T & M_a \end{bmatrix}\begin{bmatrix} \ddot{u} \\ \ddot{p} \end{bmatrix}+\begin{bmatrix} 0 & 0 \\ 0 & B_a \end{bmatrix}\begin{bmatrix} \dot{u} \\ \dot{p} \end{bmatrix}+\begin{bmatrix} K_s & -A \\ 0 & K_a \end{bmatrix}\begin{bmatrix} u \\ p \end{bmatrix}=\begin{bmatrix} f_s \\ 0 \end{bmatrix}, \tag{2.2}$$

where the index s for the structure and the index a for the acoustic subsystem or in a short form

$$\mathbf{M}\ddot{\mathbf{q}}+\mathbf{B}\dot{\mathbf{q}}+\mathbf{K}\mathbf{q}=\mathbf{f} \tag{2.3}$$

$\mathbf{u}$ — column vector of the structure deformations in the node points.
$\mathbf{p}$ — column vector of acoustic pressures in the nodal points.
$\mathbf{A}$ — coupling matrix of both subsystems.
$\mathbf{M}, \mathbf{B}, \mathbf{K}$ — mass matrix, damping matrix and stiffness matrix, respectively.
$\mathbf{f}$ — column vector of exciting forces.
$\mathbf{q}$ — column vector of total coordinates.
ρ_0 — density of air (1.2 kgm^{-3}).

3. Spectral and modal properties of the drive unit model

The basic dynamic characteristics of conservative mechanical systems are their spectral and modal properties.

The mentioned characteristics are derived from the homogeneous matrix equation

$$M_s\ddot{u}+K_s\dot{u}=0. \tag{3.1}$$

A modal analysis was performed using reduced modal extraction procedures with a total of 140 masters degrees of freedom. Table 1 shows some calculated natural frequencies of the drive unit housing. The first six frequencies correspond to a rigid body motion and the other ones correspond to the surface deformation on the surface vibration.

Table 1. Natural frequencies of the drive unit

Order	Frequen	Order	Frequen	Order	Frequen	Order	Frequen
1	2.29	7	164.3	31	892.6	62	2119.4
2	5.08	8	218.3	32	903.9	63	2177.8
3	6.02	9	331.5	33	913.6	64	2181.5
4	9.29	10	378.2	34	940.9	65	2199.2
5	10.56	11	436.3	35	963.9	66	2215.0
6	10.71	12	439.5	36	998.6	67	2241.7

4. Response of the acoustic subsystem to harmonic excitation

The exciting forces in the mechanical transmission originate inside the driving mechanism during running. The size of exciting forces are not known but their frequencies can be detected from a kinematic scheme of the whole driving mechanism [5]. From this point of view only their frequencies must be kept as they were.

The harmonically variable exciting force in the side direction was applied in the bearing which is positioned in a diaphragm in the gearbox. The exciting frequency

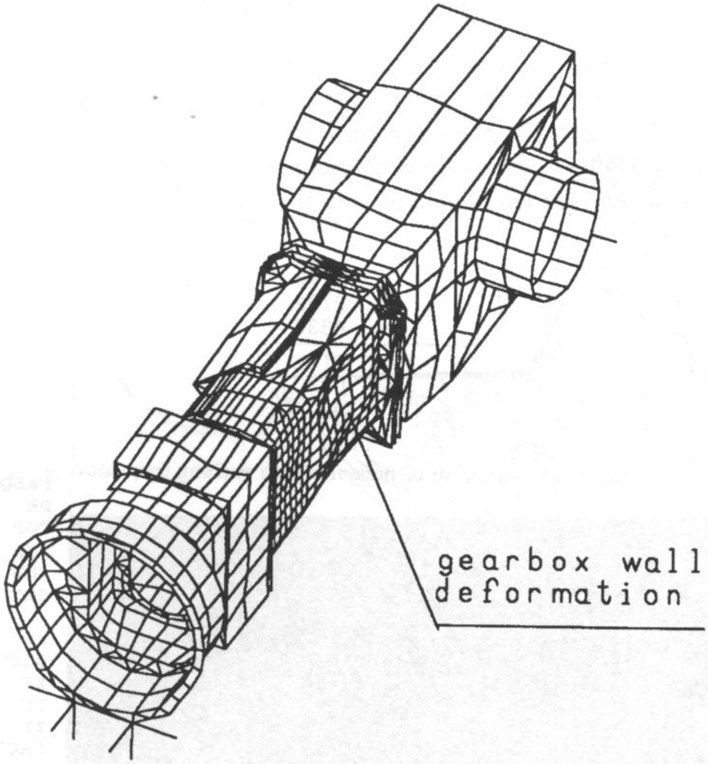

Fig. 2. Mode shape of the drive unit at $f=2215$ Hz.

was $f=2215$ Hz in order to correspond with one of the natural frequencies of the real drive unit housing 5 and the order 66 (see Table 1). If the structural motion is visualized Fig. 2, we can see that the side wall of the gearbox housing vibrates and the acoustic analysis can be performed.

4.1. Calculation of the air-borne noise

Figure 3 shows the entire frequency range that was investigated. It is a narow range in the surrounding of the natural frequency. The nodal acoustic pressures of the sound field were calculated and the sound pressure level relative to the reference pressure, Re=2.10E-5 Pa, is computed at the element's centroids. The contours near the vibrating surface and in the plane at the end of the investigated sound field i.e. at the distance 0.6 m from vibrating surface are plotted (see Figs. 4 and 5). The figures show clearly that the highest level of the acoustic pressure of 129 dB is at a close distance 0.03 m i.e near the gearbox wall. Its value decreases to 110 dB at the remote wall of the acoustic space. From Fig. 3 it can be evaluated that the noise level at points of the acoustic space selected by chance is expressively increasing in the surroundings of the natural frequencies.

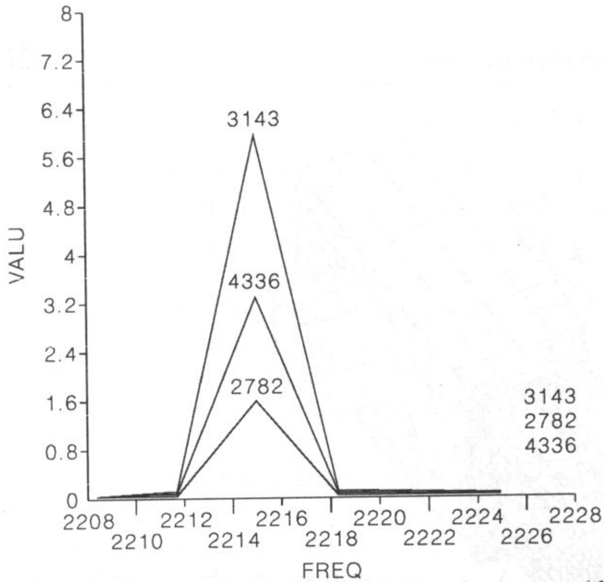

Fig. 3. The variation of pressures with exciting frequency.

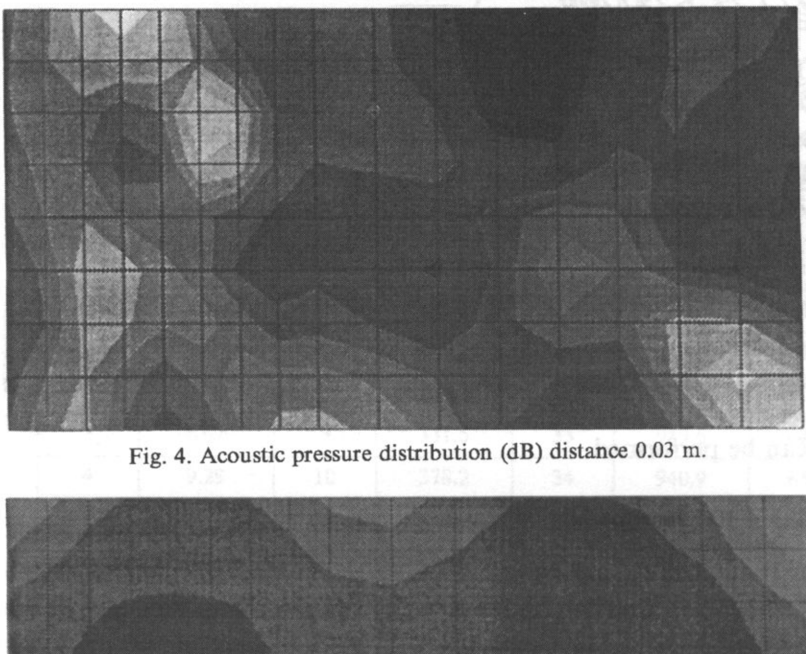

Fig. 4. Acoustic pressure distribution (dB) distance 0.03 m.

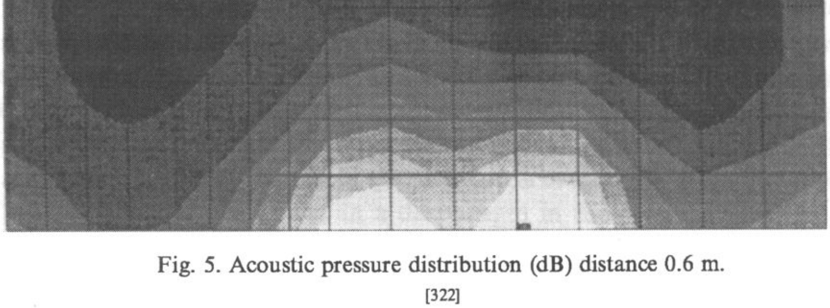

Fig. 5. Acoustic pressure distribution (dB) distance 0.6 m.

5. Model evaluation and measurements

For the comparison described in this paper, the mode shape of the real driving unit was measured by the resonance method. The coresponding mode shapes of the left side wall of the gearbox housing only are shown in Figs. 6 and 7. Here the natural frequency is $f=2222$ Hz during the measurements on the real body and $f=2215$ Hz during the calculation of the model. It can be concluded that we deal with the same mode shape of vibration.

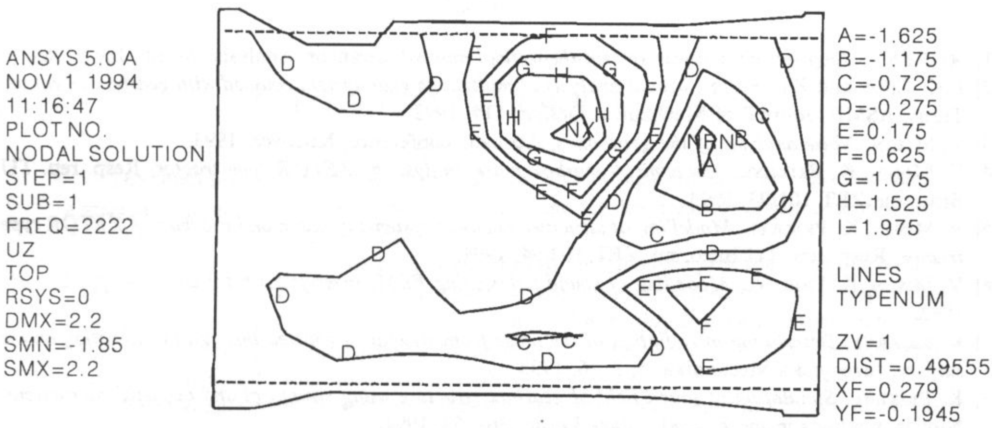

Fig. 6. Mode shape of the gearbox' side wall measurement.

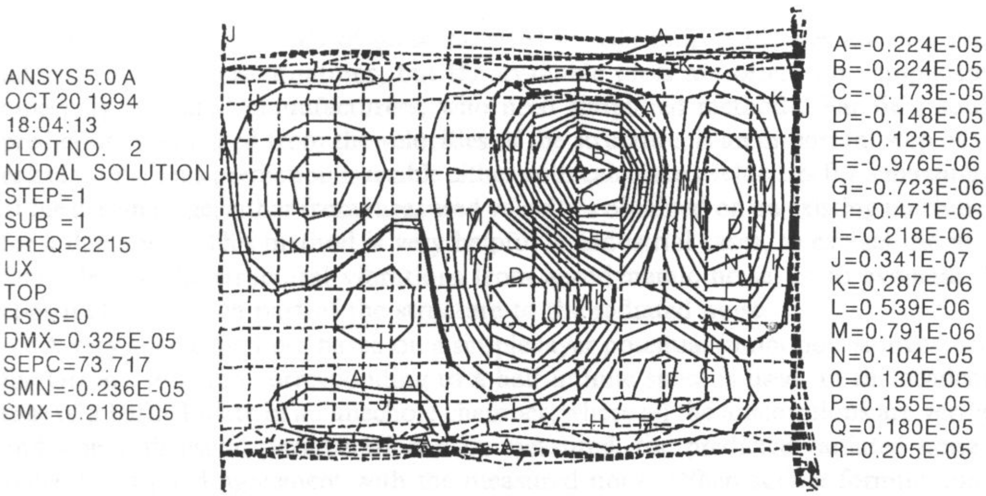

Fig. 7. Mode shape of the gearbox' side wall calculation.

6. Conclusion

Modelling of the coupled structural-acoustic system with FE-method by ANSYS 5.0 has been presented in this paper. Special attention was given to the air-borne noise. From the calculated results and the information on the surface vibration, effective changes in the individual constructional parameters for noise reduction can be found by the design optimization technique.

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