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SOUND TRANSMISSION THROUGH GEARBOX: STUDY AND MODELLING OF THE INTERNAL ACOUSTIC SOURCES

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1. INTRODUCTION

The gearbox noise is mainly due to the acoustic radiation of the housing. This one vibrates under the action of two kinds of excitation: (1) dynamic efforts which arise in the region of tooth contact and which propagate through shafts and bearings, and (2) an internal acoustic field which results from vibrations of gear bodies and from an aerodynamic sound. This sound is generated by a fluid pumping action induced by the meshing process. The acoustic excitation has been not very studied unlike the structure-borne noise.

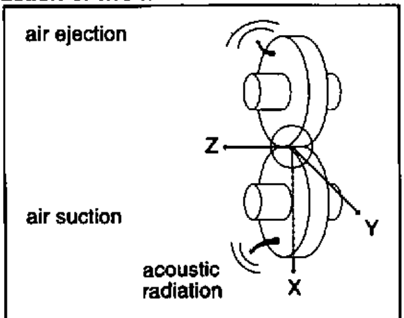

Fig. 1: Sources of internal noise.

The aim of this paper is to provide a set of theoretical and numerical results about the sources of internal noise.

2. MODELLING OF FLUID-PUMPING NOISE

The noise source associated with the fluid pumping action was modelled for a spur gear radiating in free space from previous experimental results obtained by Houjoh et al. [1]. The aerodynamic sound is emitted from the meshing region at the meshing frequency f_{mesh}, mainly in the XY-plane (see fig. 1) and towards the exit side of meshing. The radiated noise levels in this direction can be very high. They reach 130 dB 160 mm from a reference gear [1] at 3850-rpm rotation speed (160 mm ≅ one wheel radius).

Houjoh measured the effects of rotation speed on the noise levels, on the source directivity and on the phase difference between the source and a receiver. They are displayed on fig. 2 (fig. 2 was obtained from our modelling but is very closed to the experimental one).

Some remarks about these effects allowed us to conclude that the fluid-pumping noise is generated by two out-of-phase and directive elementary sources. One dominates above f_1 (see fig. 2) and radiates towards the exit side of meshing whereas the other is predominant for slow rotation speeds and emits towards the entrance side. According to Lighthill, the acoustic power radiated by an aerodynamic source is proportional to its typical speed. Thus we assumed that the intensity of elementary sources is proportional to $(\alpha f_{mesh} + b)^n$ and $(\alpha f_{mesh} + b)^m$, $n > m$. As the noise radiated by turbulent flows are often modelled by quadripoles, we supposed that each elementary source is composed of four monopoles located in the YZ-plane. We subsequently identified the characteristics of the eight monopoles associated with the reference gear by using measured directivity patterns (see table 1). The influence of rotation speed is expressed through the frequency dependence of the parameters of the monopoles.

The effects of the facewidth b and the module m can also be introduced in the model, by modifying the parameters found for the reference gear as follows: (1) the intensities are multiplied by 1/b and $(m^2/Z)/(m^2/Z)_{ref}$, and (2) the term f_{mesh} is replaced by $f_{mesh}.b/b_{ref}$ in all expressions. These modifications are deduced from Houjoh's measurements.

3. ACOUSTIC BEHAVIOUR OF GEAR BODIES

Vibrations

The vibrations and the acoustic radiation of the gear bodies constitute the second internal source of noise. The wheels are modelled by two thick annular disks. Their vibrations are governed by equations stemming from the Mindlin plate theory, in which the membrane stresses caused by rotation can be neglected. The boundary conditions are free-clamped edges. In fact, the coupling between the two wheels is not taken into account. The solicitation acting in the meshing zone is represented in a very simplified manner by a point force normal to the wheel plane. Whereas it does not move for an exterior observant, this force turns around the wheel because of the rotation of the gear.

Theoretical considerations and a few hypothesis lead to express the response of the wheel from the modal responses of a fixed disk having the same geometrical properties but being excited by a particular fixed solicitation.

$$\mathbf{w}[M, \omega, \omega_r, \omega_o] = \sum_{n} \mathbf{w}_{Dno}[M, \omega]_{\{F,0\}}.\delta[\omega-\omega_o]$$
$$+ \sum_{n,m\neq 0} \sum_{s=1,2} \mathbf{1/2.w}_{Dnm}[M, \omega+(-1)^s m\omega_r]_{\{(F_1,0)+(F_{2s},\pi/2m)\}}.\delta[\omega-\omega_o]$$

The excitation of non-axisymetric modes is composed of two forces F_1 and F_{2s} which have the same amplitude F but are out of phase ($F_{2s}=(-1)^s jF_1$). F_{2s} is shifted forward $\pi/2m$-angle (m is the number of nodal diameters).
Due to the motion of the real excitation in relation to the wheel, each natural mode responds at two particular frequencies, which depend on the excitation frequency, on the mode order and on the rotation speed. Nevertheless, as the excitation is motionless in the global coordinate system, the observed response to a ω_0-harmonic excitation is ω_0-harmonic. There is no Doppler effect.

Acoustic radiation

The acoustic properties of the wheel are studied in free space and in the absence of the shaft. The centre of the wheel is baffled. The acoustic variables associated with the wheel can easily be deduced from the disk ones. The behaviour of the disk associated with the reference gear was numerically investigated with two vibroacoustic softwares, ASTRYD and SYSNOISE, which use the Boundary Element Method respectively in time domain and frequency domain. The results are nearly identical.

The radiated noise is maximum when a natural mode of the disk is resonant. The radiated power reaches 90 dB when the disk is excited by a 100 N force (which is a rough estimate of the efforts acting on current gear wheels). The modal radiation efficiencies of the disk shown on fig. 3 rise quickly near critical frequencies, that verify the equality $k=k_{mn}$. These observations recall the well-known results about acoustic properties of baffled thin rectangular plates. The modes mainly radiate into a cone. If they are axisymetric, the cone angle is null. Otherwise it is given by the relation $\theta = \sin^{-1}(k_{mn}/k)$ (see fig. 4). Some secondary lobes appear when the frequency increases.

4. CONCLUSIONS

We found the main acoustic properties of the sources of internal noise for spur gears. The sources do not radiate in the same direction, but they both generate high sound levels. The modelling of these sources will be used to investigate the acoustic transmission through the housing, and to know the contribution of the air-borne noise to the total sound power radiated by a gearbox.

5. ACKNOWLEDGEMENTS

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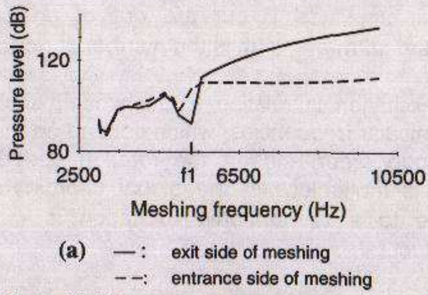
[1] H. Houjoh et al.: Proceedings of Internoise'88, pp619-622, 1988.



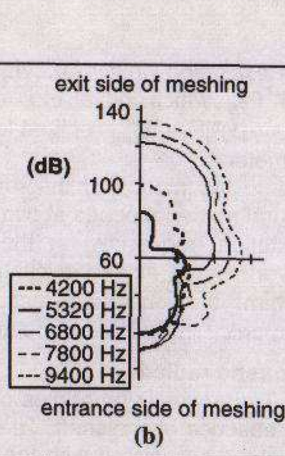
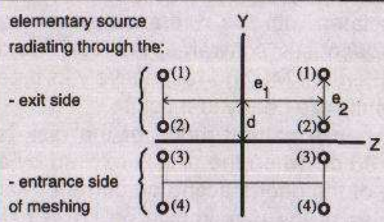


Fig. 2: Fluid-pumping noise source:
(a) frequency dependance of the noise level;
(b) directivity patterns in the YZ-plane.



(1) $A = A'.[\alpha.f_{mesh} - 1.4]^2$, $\varphi = 176°$

(2) $A = A'.[\alpha.f_{mesh} - 1.4]^2$, $\varphi = 176° + \Phi$

(3) $A = A'.[\alpha.f_{mesh} - 1.4]$, $\varphi = 0°$

(4) $A = A'.[\alpha.f_{mesh} - 1.4]$, $\varphi = \Phi$

	$f \leq f_1$	$f > f_1$
Φ'	179° (except for $f=f_1$: 160°)	57.3°
e_1 (mm)	201600/f	$-0.0027\times(f-f_1)+31$
e_2 (mm)	if $f \leq f_2$, $10^{[-0.00047\times(f-f_1)-0.53]}$ else $10^{[-0.0031\times(f-f_1)-2]}$	$-(f-f_1)/2600+10.6$
d (mm)	$0.014\times(f-f_1)$	0
A'	12.6	1

$f = f_{mesh}*b/b_{ref}$

$\alpha = \alpha_{ref}*b/b_{ref}$

$f_1 = 5320$ Hz

$f_2 = 4755$ Hz

$b_{ref} = 0.05$ mm

$\alpha_{ref} = 4.5.10^{-4}$

Table 1: Modelling of the fluid-pumping noise source.
The noise level is given by: $N(dB) = N(model) + 20.\log[m^2/(b.Z)] + 60$ dB

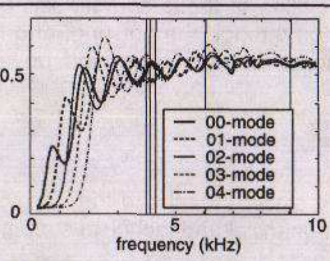


Fig. 3: Modal radiation efficiency of the disk.

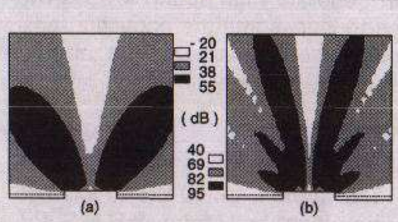


Fig. 4: Acoustic field radiated by the 02-mode of the disk, at 2000 Hz (a) and at its natural frequency (4320 Hz) (b); F=100N.

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ANALYSIS AND MODELLING OF THE VEHICLE GEARBOX NOISE

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INTRODUCTION

The gears in mechanical transmissions generate vibrations and noise during running which are then transfered through shafts and bearings onto the outer housing surfaces and adjoining areas which then emit the acoustic energy into a surrounding air space, in another word, acoustic subsystem.

When searching the noise level of mechanical transmission, it is necessary to follow the whole path - Fig. 1, e.g. from the cause of the excitation effects to the acoustic space.

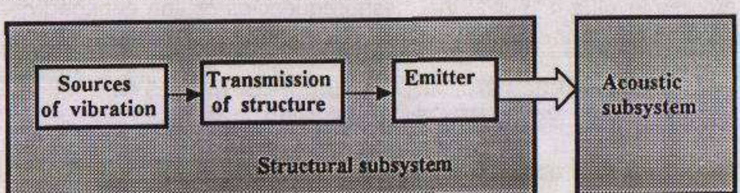

Fig.1.

Each of these blocks can expresively influence the total noise level as well as frequency spectrum of the noise emitted from the gearbox surface. Both structural subsystem and acoustic one form so caled structural-acoustic system.

OPERATIONAL NOISE OF THE GEARBOX

Higher operational vibrations of the transmission increase the level of noise. The total noise level and its distribution in each frequency band is changed according to the operational conditions, e.g. change in rotation, shifting of individual gears in the gearbox alone and also in other parts of the driving mechanism, etc.

Fig.2 shows Campbell diagram of the acoustic pressure levels of noise spectrum emitted from the back wall of the gearbox within 1 m from this wall. The noise was measured from the gearbox of the passenger car of type SKODA- FELICIA when the 4th gearbox ratio was engaged [3].

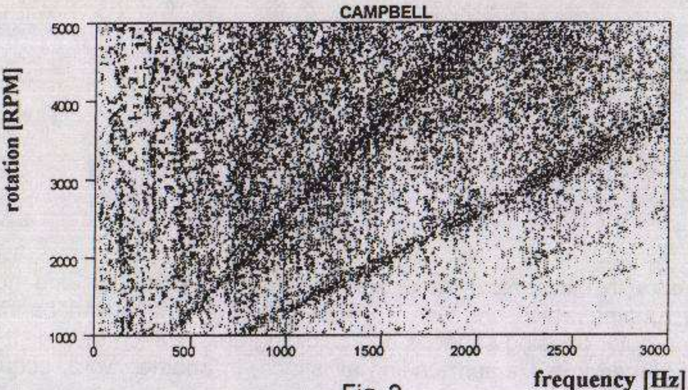

Fig. 2.

Fig 2 shows the evaluated noise spectrum in the frequency range up to 3000 Hz. The range of the acoustic pressure level is from 44 dB up 84 dB. The noise level is higher at the mesh frequencies of the engaged gears and the stable gear - see the inclined black lines.

SPECTRAL AND MODAL PROPERTIES OF THE GEARBOX HOUSING

The acoustic pressure field in the acoustic space is unambiguously determined by vibrations of the gearbox housing structure. So that the spectral and modal properties of the structure were being measured.

It is possible to monitor that each outer surfaces of the drive body housing vibrate with variable amplitudes according with corresponding mode shapes of vibration. It is therefore possible to detect at each mode shape when there is some apparent deformation going on the back wall of the gearbox housing and which of them have a strong influence as to the emition of noise into the monitored acoustic environment.

For example in Fig. 3., we see the measured mode shape, having natural frequency of f_0 = 465 Hz . Here it is evident that the back wall of the gearbox housing and the engine block vibrate very expressively.

Each mode shape of the drive housingl, according to the intensity of its excitation during running, contributes then by its noise in different measure to he total noise level which is emitted from the gearbox housing and engine block.

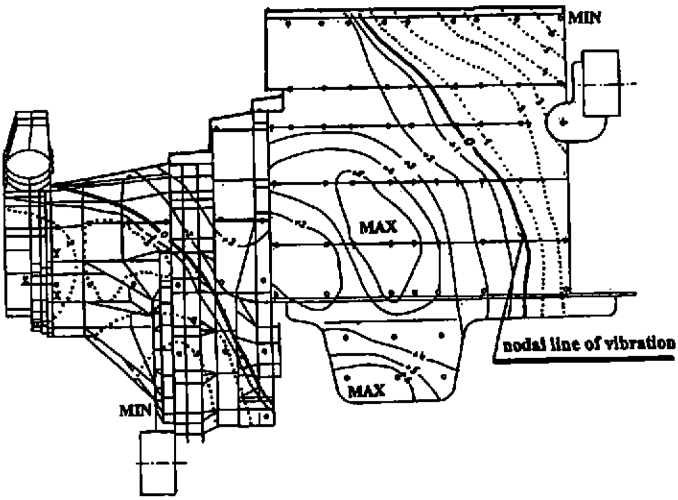

Fig. 3.

MODELLING OF THE STRUCTURAL-ACOUSTIC SYSTEM

In the view that only the noise emitted by the back wall of the gearbox housing and engine block is monitored, the acoustic area was modelled as a closed acoustic space in the shape of a cube. The acoustic space at the back wall of the gearbox was modelled as a closed space but with the properties of a free acoustic field.

Modelling of a coupled structural-acoustic system was done with a help of FEM programme system ANSYS 5.1.

The matrix equation of motion of the coupled structural-acoustic system is then [1], [4]

$$\begin{bmatrix} \mathbf{M}_s & 0 \\ \rho_0 \mathbf{A}^T & \mathbf{M}_a \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{u}} \\ \ddot{\mathbf{p}} \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & \mathbf{B}_a \end{bmatrix} \begin{bmatrix} \dot{\mathbf{u}} \\ \dot{\mathbf{p}} \end{bmatrix} + \begin{bmatrix} \mathbf{K}_s & -\mathbf{A} \\ 0 & \mathbf{K}_a \end{bmatrix} \begin{bmatrix} \mathbf{u} \\ \mathbf{p} \end{bmatrix} = \begin{bmatrix} \mathbf{f}_s \\ 0 \end{bmatrix} \tag{1}$$

in a short form

$$\mathbf{M}\,\ddot{\mathbf{x}} + \mathbf{B}\,\dot{\mathbf{x}} + \mathbf{K}\,\mathbf{x} = \mathbf{f} \tag{2}$$

where index : s - for structure (structural subsystem)
a - acoustic subsystem

u - column vector of the structure deformations in the node points

p - column vector of acoustic pressures in the nodal points
A - coupling matrix of both subsystems
M, **B**, **K** - mass matrix, damping matrix and stiffness matrix
f - column vector of exciting forces
and further column vector of total coordinates

$$\mathbf{x} = \begin{bmatrix} \mathbf{u} \\ \mathbf{p} \end{bmatrix} \tag{3}$$

RESPONSE OF THE SYSTEM ON THE HARMONIC EXCITATION

When the excitation is in the harmonic form

$$\mathbf{f} = \mathbf{F}\, e^{j\omega t} \tag{4}$$

the responses will also be harmonic [1].
The amplitude of the harmonic response of the coupled structural-acoustic system is then

$$\mathbf{X} = \sum_{i=1}^{n} \mathbf{v}_i\, \mathbf{Q}_i = \sum_{i=1}^{n} \frac{\mathbf{v}_i \mathbf{w}_i^T \mathbf{F}}{\mathbf{m}_i\left(\lambda_i - \omega^2\right)} \tag{5}$$

where $\mathbf{v}_i$, $\mathbf{w}_i$ are right and left sided eigenvectors of the coupled system, $\mathbf{m}_i$, λ_i - modal parameters of the system, ω - excitation circular frequency.

The relation (5) is possible to use for modeled acoustic subsystem as a diffusion field.

CONCLUSION

The introduced model of coupled structural-acoustic system allows to solve its interface interaction in both directions.

The presented algorithm allows to solve the coupled system with the help of modal analyses using the known method of mode superposition.

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