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THE USE OF SURFACE POWER FOR CHARACTERISATION OF STRUCTURE-BORNE SOUND SOURCES OF LOW MODAL DENSITY

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1. INTRODUCTION

Different methods for quantifying the vibro-acoustic strength of machinery sources have been reviewed in refs. [1,2]. Most of these are based on simple velocity measurements of either a receiving plate to which the source is rigidly connected, or the feet of resiliently mounted machinery. However, novel measurement techniques based on power quantities are also devised [3 -10], and the most simple of these appears to be the socalled *equivalent source power* method [4,5,7], which is also called the *surface power* method.

This method is very suitable for evaluating the overall vibration characteristics of machines and for estimating the power produced by their internal source mechanisms. For rectangurlar box-shaped sources of moderate modal density (of .03 modes/Hz according to [11]), the method has been found very reliable and only little effected by the mounting conditions of the sources [5]. Whether this also applies to complex machinery sources of relatively low modal density is examined in this paper by determining the equivalent source power of a stiffened, shell-shaped model of a helicopter gearbox.The robustness of the method is tested for free and coupled mounting conditions, and its accuracy is evaluated by comparison with the power injected by the interior excitation mechanisms in the form of `instrumented´exciters.

The method assumes that the machinery vibration resulting from the (usually) unknown internal excitation can be adequately represented by the averaged input of complex power, J, from one or more *external* point forces. These equivalent forces and the associated input powers are determined by exciting the exterior parts of the machine structure. The magnitude spectrum of each force is then individually adjusted so it complies with the measured surface vibrations during machinery operation and, finally, an averaged value of the input powers is determined. It should be emphasised that the method does not presuppose that the vibratory field in the casing is reverberant.

A complex power description is necessary because both the real and reactive source powers are required in order to account for the non-resonant forced vibration, which is important at low and mid frequencies in cases of compact rigid sources like small pumps and compressors etc. However, for extended machinery such as gearboxes and multi-cylinder piston engines, it is sufficient to use the real part of the vibratory power as a source descriptor.

2. EQUIVALENT SOURCE POWER

Determination of the equivalent source power of a machine from measurements on its casing is based on three types of quantities [4,5]. These are:

(i) the point mobility $Y_{jj}(\omega)$ of a suitably selected external drive point, j, on the machinery casing

(ii) the spatially averaged, squared transfer mobility $\langle |Y_{pj}(\omega)|^2 \rangle$ relating the velocities at all response positions p on the casing to the external drive force F_j at position j

(iii) the correspondingly averaged mean-square velocity response $\langle v_p^2 \rangle$ of the casing during operation of the machine at a specified rpm and load condition.

These measurements can be performed with a suitable mounting arrangement of the machine on a factory test bench as schematically illustrated in Fig.1.

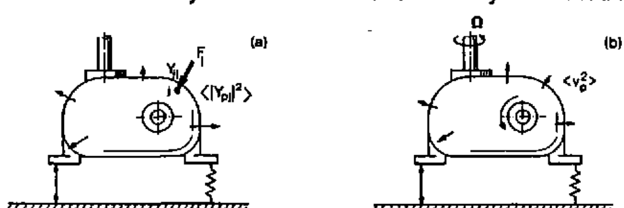

Fig. 1 Measurements of: (a) point and transfer mobilities of a machine casing when excited by an externally applied point force F_j at an arbitrary position j ; (b) casing velocity response of operating gearbox.

The power calibration of the machine follows from the ratio of (i) and (ii), which, when multiplied by (iii), yields the time-averaged and complex-valued equivalent source power $J_{eq,j} = P_{eq,j} + iQ_{eq,j}$. The variance of this estimate is of course improved by repeating the experiments using differently located drive points of various orientations. From the average of these uncorrelated estimates the equivalent souce power J_{eq} of the machine is obtained:

$$J_{eq} = \langle J_{eq,j} \rangle = \langle v_p^2 \rangle \left\langle \frac{Y_{jj}^{*}(\omega)}{\langle |Y_{pj}(\omega)|^2 \rangle} \right\rangle = \langle v_p^2 \rangle \langle \alpha_{pj}(\omega) \rangle \ , \qquad (1)$$

where pointed brackets imply spatial averages, and $\langle \alpha_{pj}(\omega) \rangle$ is the averaged proportionality function as defined by the last two terms of eq. (1); this represents the vibro-acoustic calibration of the machine type, whereas $\langle v_p^2 \rangle$

is specific to the individual machine at a given rpm and load. The active source power, P_{eq}, and reactive power, Q_{eq}, are obtained simply by taking the real and imaginary parts, respectively, of eq.(1). At the gear-meshing frequencies, P_{eq} is the most useful and robust quantity of the two.

Drive points. The optimum choice of drive point locations for external excitation is in rigid parts of the casing, that is, in stiffened shell or panel areas over ribs [5]. However, the specific selection of locations and directions is not critical as long as most of the structural modes are excited and provided that any dominating directions of major interior forces are included. Three or four drive points have been found to give consistent and accurate results.

Response points. The averaged transfer mobility and response of the casing are measured normal to the surface at a number of positions which adequately represents its kinetic energy. In cases where the surface mass varies the results should be averaged on a kinetic energy basis. It is preferable to include most parts of the main casing structure, but this is not a requirement.

Machinery mounting conditions. Experimental results with a moderately damped machine structure of rectangular box-form have shown [5] that the method provides reliable results, whether the source is resiliently mounted (ie freely suspended with ideal boundary conditions) or rigidly connected to a foundation or some form of supporting structure. (Whether this also applies to the low modal density source will be revealed in the following.)

3. EXPERIMENTAL RESULTS FOR HELICOPTER GEARBOX MODEL

Experiments were conducted with a 3/4-scale structural model of a BK117 helicopter gearbox (see Fig. 2a) in which the interior source mechanisms were simulated by two built-in vibration exciters that were fed by white noise signals. This dummy gearbox was instrumented so that the *interior* power input P_o from the exciters could be measured, and thus used as an absolute reference for evaluating the estimation accuracy of the equivalent source power method. Since the dominant gear-meshing components in practice occur in a frequency band from about 600 Hz to 4 kHz [12], this also constitutes the frequency range of main interest for this application. Two mounting conditions of the gearbox were considered: freely suspended, and installed in a 3/4-scale helicopter airframe model. This last condition serves the dual purpose of testing the robustness of the method towards changes in boundary conditions, and the sensitivity of the interior source mechanisms to realistic mounting conditions.

Power calibration and determination of averaged velocity response were carried out for each of the two mounting conditions. Fig. 2a shows positions of the four external drive points that were used (sequentially) in the power calibration of the gearbox. Points *A*, *B* and *C* are 'hard' whereas *D* is relatively 'soft' because it is in an unstiffened panel area. Averaged transfer mobilities for each of the drive points, and the average velocity response during operation, were all determined from 26 positions that were randomly distributed

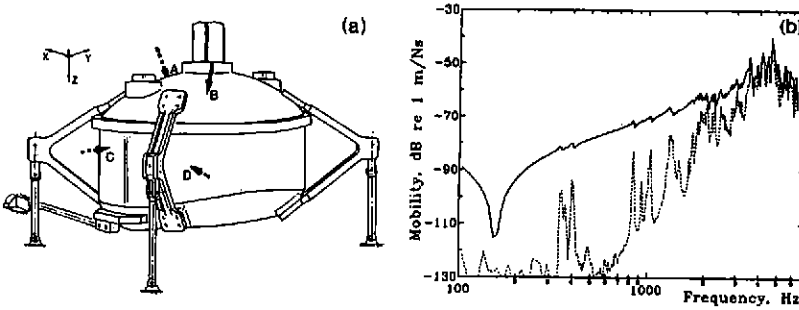

Fig. 2 (a) Positions of drive points A,B,C and D normal to gearbox surface. (b) Direct mobility of excitation point B: ——, modulus; ·······, real part.

over the casing surface (as little as 8 positions may be used). For a typical hard point, B, Fig. 2b shows that the input mobility is mostly very reactive, and that there are few modes below 3 kHz which corresponds to 0.7 times the ring frequency of the casing shell. This is expected, since no radial casing modes should ideally occur below that frequency [13]. However, a few distinct modes occur around 400, 1000 and 1300 Hz, corresponding to bending modes of the V-shaped mounting arms, shaft modes and flat panel modes,respectively.

The results of the two calibrations in Fig. 3 show that the proportionality function is virtually unaffected by coupling in the important mid and high frequency ranges, say, above 400 Hz. However, this is certainly not the case at low frequencies where especially the small real parts of the gearbox input mobilities are effected by the increased damping imposed by coupling to the airframe. Results that are very similar to these have recently been obtained from ground tests on a medium-size helicopter of the type BK 117 [12].

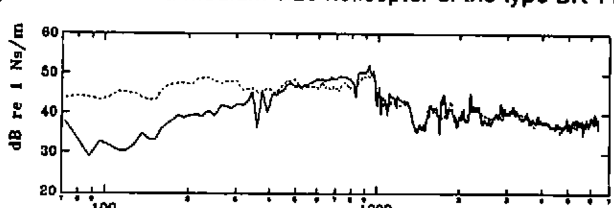

Fig. 3 Proportionality function Re($\alpha_{p}(\omega)$) of gearbox, averaged from estimates based on: excitation points A,B,C and D on casing. Gearbox mounting conditions: ——, freely suspended; - - - - -, installed in helicopter.

Also the strength of *interior* excitations is almost invariant to coupling in the frequency range of main interest. This is illustrated in Fig. 4 by the ratio of interior injected powers for the two conditions $P_{o,coupled} / P_{o,free}$, which shows that there is only a moderate increase in power of about 2 dB for frequencies above 600 Hz, whereas the increase at low frequencies is very significant with a broad peak of 15 dB at 250 Hz.

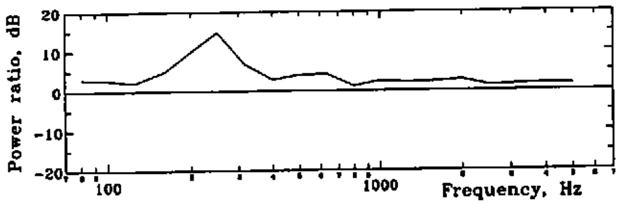

Fig. 4 Ratio of powers injected to gearbox in the coupled and free conditions, $P_{o,coupled} / P_{o,free}$, presented as the ratio of one-third octave band values.

The final results of the equivalent source power P_{eq} determined by surface measurements for the two mounting conditions are shown in Fig. 5 together with the power injected by the interior forces. The two conditions are compared in Fig. 6 where the equivalent source powers are normalised by the correspondingly injected interior power. Three major observations can be made from these results.

First, the robustness of the method towards different mounting conditions is clearly revealed by the close correspondence or merging of the two curves in Fig.6: in the frequency range from 250 Hz to 5 kHz the mutual deviations are found to be less than 2.5 dB (when the troublesome 400 Hz is ignored). Secondly it is seen from Figs.5 and 6 that the overall spectrum of the interior source power is accurately estimated by the equivalent source power method, say, to within 2 dB in the frequency range from 630 Hz to 3.16 kHz. In fact the interior power can be estimated with confidence even down to 250 Hz, which

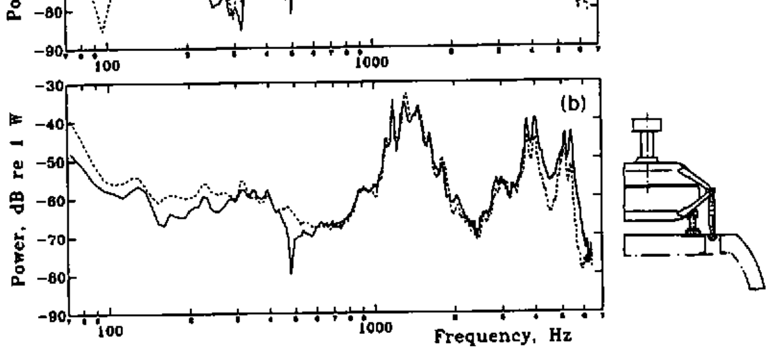

Fig. 5 Comparison of:——, equivalent source power derived from surface measurements, and ——, internally injected power. Gearbox mounting conditions: (a) freely suspended , (b) connected to airframe.

is a factor five below the 'fundamental' panel mode (1300 Hz) of the gearbox casing sides, located at pos. *D* in Fig. 2a. Thirdly it is demonstrated that the method predicts accurately the changes in the *interior* source power that is caused by coupling to a helicopter airframe.

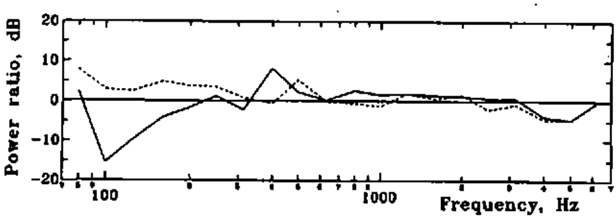

Fig. 6 Normalised equivalent source power P_{eq} / P_o for gearbox mounting conditions: ——, freely suspended; ------, connected to airframe.

4. CONCLUSION

Equivalent source power of an instrumented 3/4-scale structural model of a helicopter gearbox of low modal density has successfully been measured. The close correspondence of results for free and coupled mounting conditions has verified that this method is a very robust and accurate technique for quantifying the overall vibratory strength, even of machinery with a rather rigid casing. These surface measurements are also capable of accurately estimating the total power injected by the internal source mechanisms, and of disclosing that this power is not fully invariant to the gearbox mounting conditions. It is envisaged that the method may find use in the development stage of new machines, in comparison studies of different machines, as a vibro-acoustic specification and in factory quality control.

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