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MINIMISATION OF NOISE AND VIBRATION LEVEL FROM A STEEL PANEL

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Abstract

This work is part of a project to minimise the noise radiated from the casing of a constant speed gearbox by optimizing the positioning of stiffening ribs so as to create a "dead band" around the fixed gearmesh frequencies and their harmonics. Experimental studies were carried out on a baffled rectangular plate where a number of equally spaced stiffeners were attached, having been designed to minimize the radiation in a band around each of two harmonics of a simulated "gearmesh frequency". The band was made wide enough to allow for errors in the model updating process and for variations between different realizations of the "same" stiffened plate. The properties of epoxy joints and elastic supports were identified and were applied in the subsequent model updating process. A good agreement between the numerical and experimental modal models was achieved. Frequency response functions were synthesised from the updated FE model in terms of natural frequencies and mode shapes. A pseudo-inverse method was used to identify the two forcing functions acting on the plate (before modification) which in turn were used to stimulate the updated FE model. The resulting vibratory and acoustic responses from the experiment confirmed the validity of the optimisation proposed by numerical simulation. The optimal stiffener layout resulted in 'troughs' in the vibration and acoustic levels within the frequency bands of interest for the original plate.

1. INTRODUCTION

This work is part of a project to minimise the noise radiated from the casing of a constant speed gearbox by optimizing the positioning of stiffening ribs so as to create a "dead band" around the harmonics of the fixed gearmesh frequencies. An example plate is chosen to be similar to the main noise radiating surface of gearboxes of the type modelled (880 mm x 1000 mm x 16 mm thick), with primary gearmesh frequencies between 400 and 800 Hz. An earlier paper [1] has illustrated by numerical simulation the process of optimizing equally spaced stiffeners on such a steel plate with respect to the weighted vibration energy. Quad shell elements were used for the Finite Element (FE) modeling, with a mesh size approx. 35 mm. It was demonstrated that optimization could be made for virtually any gearmesh frequency (and its harmonics), in that case arbitrarily 600 and 800 Hz. In this paper, the results from experiments performed on the optimal stiffener configuration on the plate are presented and discussed (for 600 Hz simulated gearmesh frequency only).

2. PLATE UPDATE

The plate model used in the original optimisation study was considered to have very flexible elastic supports whose stiffness in the initial model was set to correspond to free-free boundary conditions. A test rig was set up for measuring the vibratory level and radiated sound power level for a number of panel configurations examined in the numerical model. A picture of a typical configuration is shown in Figure 1 below. Modal tests and multi-shaker experiments were first performed on a plain rectangular steel plate mounted in an aperture

Figure 1: Plate test rig

between two reverberation rooms. The plate was supported on rubber blocks at the bottom and ‘glued’ to the mounting support via Silastic sealant. In the finite element model, this elastic boundary condition was modelled using grounded spring elements. The properties of these spring elements were subsequently updated, based on the experimental results and held constant for the other plate configurations examined. Prior to executing the automatic Bayesian optimisation algorithm in *FEMtools* [2], a manual adjustment of the stiffness values of these spring elements was required to bring simulated rigid body modes to match those found in the test. Experience showed that if this manual adjustment were not performed beforehand, it would be almost impossible to obtain a convergence in the automatic updating. During the updating process, higher weightings were given to frequencies in the vicinities of the two target optimisation frequencies, namely, 600 and 1200 Hz (the two first harmonics of the simulated gearmesh frequency) as they are the primary frequency bands of interest. Table 1 presents the results of the two natural frequencies on either side of the optimisation frequencies of the plate before and after the boundary condition update. It clearly indicates that the elastic boundary conditions were close to but not exactly those of an ideal free-free boundary condition. At the same time, the damping values of the modes were updated to those determined from the experimental modal analysis.

2. OPTIMIZATION OF EQUALLY SPACED STIFFENERS

The numerical formulation of the optimization of the stiffened plate was detailed in an earlier paper [1] for two simulated gearmesh frequencies 600 Hz and 800 Hz. The experimental validation was limited to the 600 Hz case. The epoxy joints between stiffening ribs (cross section 50 x 20 mm) and the base plate were modelled using gap elements so that their properties could be investigated in the experiment and subsequently used to update the model for a more realistic representation of the actual connection. The optimization parameters used in the study are summarised in Table 2 below. The width of the "dead bands" around each "harmonic" were kept at ± 100 Hz as in the previous numerical study [1], to allow for errors in the modelling, variation from one unit to another, and sidebands in the gearmesh excitation spectrum.

Table 1: Updated resonance frequencies for the unstiffened plate

EMA – Experimental Modal Analysis FEA – Finite Element Analysis

Mode No.	Frequency (Hz)		
	EMA	FEA (before update)	FEA (after update)
14	496	472	490
15	539	529	535
16	603	595	598
17	636	625	640
18	664	663	668
29	1126	1119	1122
30	1151	1157	1150
31	1205	1206	1206
32	1240	1211	1223
33	1293	1300	1298

Table 2: Optimisation Parameters

<table>
<tr><th colspan="2">Optimisation Parameters</th><th>Example 2</th></tr>
<tr><td colspan="2" rowspan="2">Excitation Frequencies (Hz)</td><td>600</td></tr>
<tr><td>1200</td></tr>
<tr><td colspan="2" rowspan="2">Frequency Bands of Interest (Hz)</td><td>500 - 700</td></tr>
<tr><td>1100 - 1300</td></tr>
<tr><td rowspan="6">Constraints</td><td rowspan="2">Number of Ribs, X_{rbn}</td><td>X_{rbn}^{L}: 2</td></tr>
<tr><td>X_{rbn}^{U}: 9</td></tr>
<tr><td rowspan="2">Spacing, X_s (m)</td><td>X_s^{L}: 0.08</td></tr>
<tr><td>X_s^{U}: 0.94</td></tr>
<tr><td>TE_1</td><td>TE_1^{U}: 1.5e-4</td></tr>
<tr><td>TE_2</td><td>TE_1^{U}: 1.9e-5</td></tr>
</table>

TE = Total Energy (summed squared velocity)

The model parameters are listed in Table 3.

2.1 Identification and Application of the Forces

After the modal model of the plate was updated, two broadband excitation forces were applied to the model for the calculation of the temporal and spatial average vibration responses across the plate. The locations of the shakers are shown in Figure 1 earlier. The so-called pseudo-inverse method was used to identify the forces in the same way as proposed for the gearbox situation, where forces applied to the bearing housings cannot be measured.

Table 3: Model Parameters

Parameter	Value
Young's Modulus, E (GPa)	210
Density, ρ (kg/m^3)	7850
Panel length, L (m)	1
Panel width, w (m)	0.9
Panel thickness, t (m)	0.016
Rib length, l (m)	0.88
Rib thickness, tb (m)	0.02
Rib width, wb (m)	0.05

Since frequency response functions (FRFs) can be expressed in terms of resonance frequencies and scaled mode shapes as shown in expression (1) [3, 4], they can be synthesised from the updated FE model of the steel plate.

$$H_{ij}(\omega) = \sum_{r=1}^{n} \frac{R_{ijr}}{j\omega - p_r} + \frac{R_{ijr}^{*}}{j\omega - p_r^{*}} \tag{1}$$

where $H_{ij}(\omega)$ is the FRF between input DOF *j* and response DOF *i* at frequency ω, *r* is mode no., p_r is the *r*th pole and R_{ijr} is the residue.

Since $\{F^o\} = [H]^+\{X^o\}$ where $\{X^o\}$ is the operational deflection shape measured experimentally and $[H]^+$ is the pseudo-inverse of the transfer function matrix, which in this case is generated from the updated numerical model, the operational load $\{F^o\}$ can be calculated. A program was written in Matlab® to synthesise the FRFs from the updated modal parameters and to perform the pseudo-inverse calculation for finding the operational load. The results are shown in Figure 2.

They were then used as input forces in the updated FE model.

2.2 Responses for the Base Plate

The resulting vibration energy (velocity squared scaled to Joules by the mass of the plate) for the base plate (no ribs) is shown in Figure 3 below.

The dashed line represents the vibration energy level prior to the model update. The solid line represents the vibration energy level predicted by the FE model after taking the elastic boundary stiffness value (and updated damping) into account. This scaled squared velocity is related to the radiated sound by the radiation efficiency, which is close to unity for frequencies above 700 Hz with a plate of this thickness. A-weighting of the radiated sound pressure, in combination with lower radiation efficiency below the coincidence frequency mean that lower frequencies have little influence on the perceived sound, and thus the squared velocity was used as the reference for comparison with response levels obtained from the optimal stiffener configuration. The major reason why some resonance peaks are reduced is because of the revision of the damping values carried out during updating, but could partly be due to the fact that the identified force spectrum has lower peaks at some resonances. This has little effect on the values in the bands of interest in the final results as will be seen below.

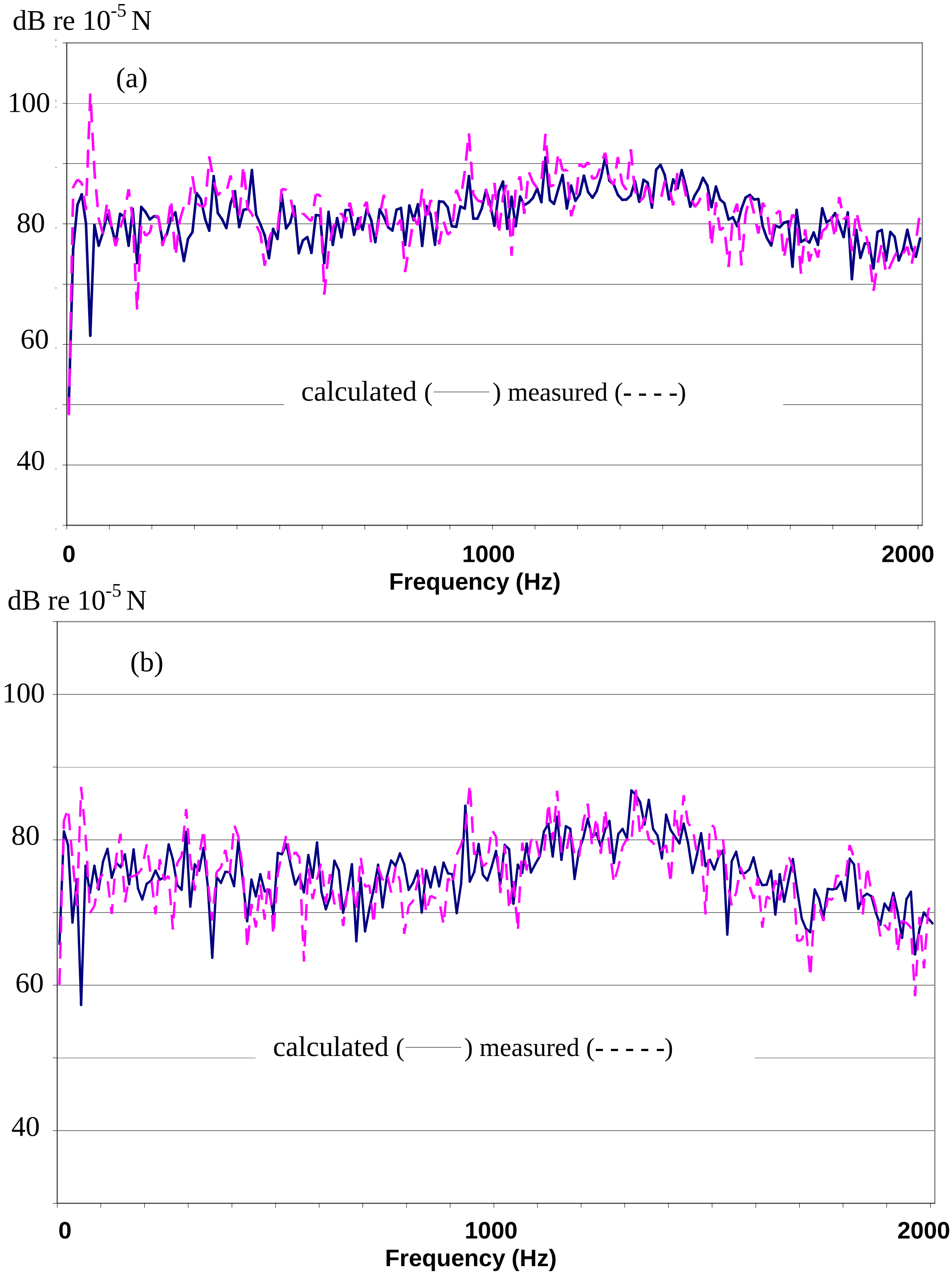

Figure 2. Identified force spectrum (a) Shaker 1 (b) Shaker 2

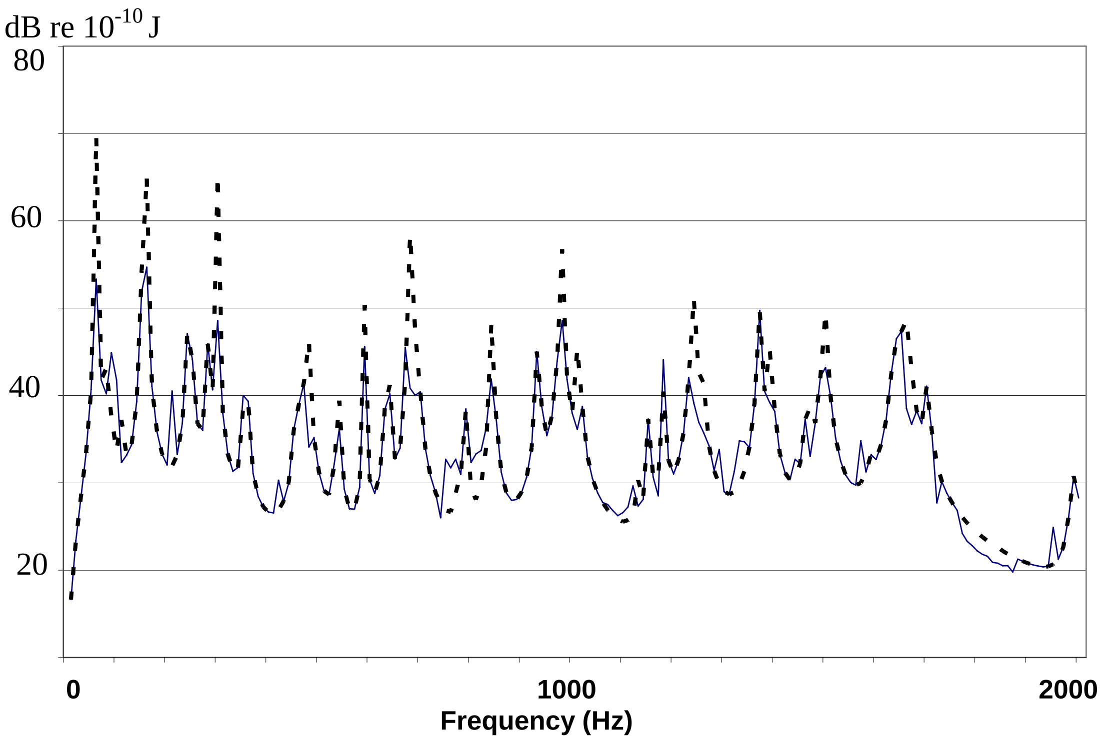

Figure 3: Base plate vibration energy: before update (- - - -) after update (——)

2.2 Responses for the optimal configuration

The commercial design software ANSYS which was used in the earlier paper [1] for identifying the optimal stiffener configuration on the plate was again used in this study on the updated plate model. For the frequencies of interest listed in Table 2, the resulting optimal configuration consisted of 5 stiffening ribs spaced at 200 mm apart. The resulting vibration energy predicted from ANSYS, based on the updated elastic boundary conditions and the forces estimated for the plate without ribs, is shown below in Figure 4.

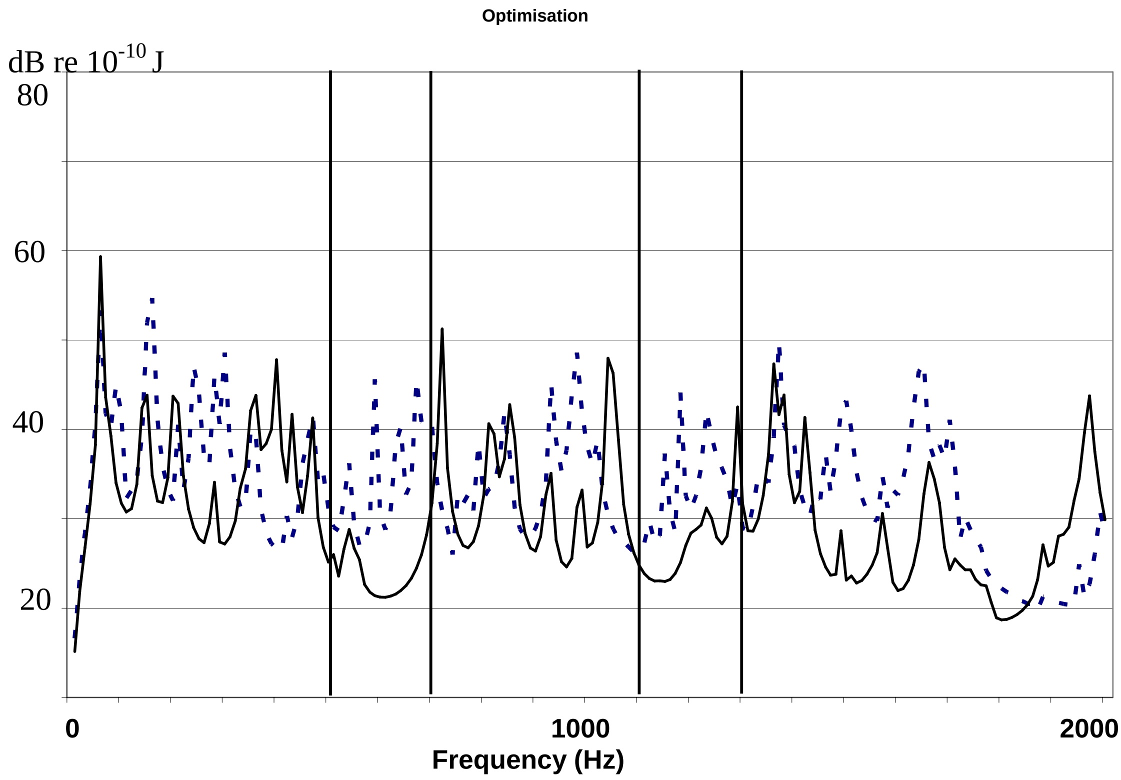

Figure 4: Vibration energy level for the optimal configuration: base plate (- - - -) 5-rib (———)

A comparison was also made between measured and predicted frequency distribution of vibration energy and the results are shown in Figure 5.

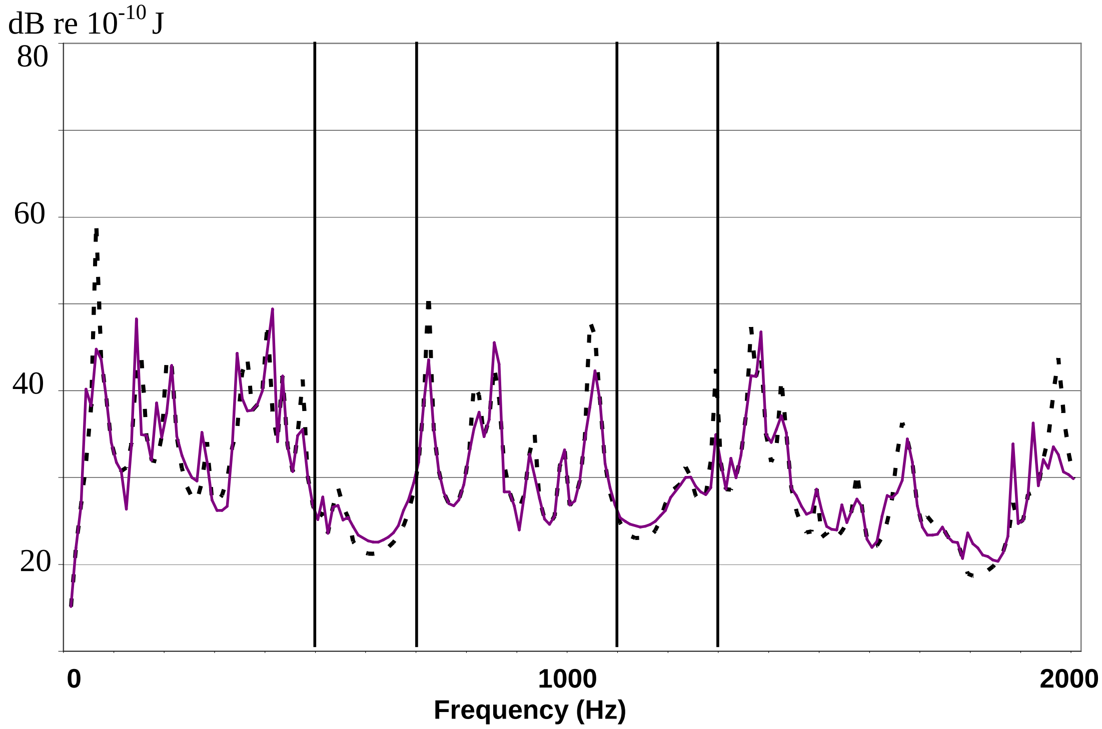

Figure 5: Vibration energy level for the optimal configuration: measured (- - - -) predicted (———)

The results clearly confirm the validity of the optimisation proposed by numerical simulation. There is a good correlation between the measured frequency response in terms of vibration energy and that predicted by numerical simulation. The frequency bands of interest clearly lie within the troughs of the response functions. A similar trend was also observed in the radiated sound power level (calculated using the Boundary Element software Sysnoise®), where above 700 Hz, the radiation efficiency became one and the radiated sound was entirely controlled by the vibratory response of the plate. Because the radiated sound spectra give basically the same information as the squared velocity, they are not reproduced here.

CONCLUSION

Optimisation was performed on a steel plate taking advantage of the formation of 'stop' bands resulting from the use of equally spaced stiffeners. Experimental studies were carried out on a baffled rectangular plate, with and without added ribs. The resulting vibratory and acoustic responses from the experiment confirmed the validity of the optimisation proposed by numerical simulation. The optimal stiffener layout resulted in 'troughs' in the frequency responses within the frequency bands of interest. Since the optimisation was pursued over specific frequency bands, the method does not guarantee the control of the vibratory responses outside those bands, but the sound from such constant speed gearboxes is normally dominated by the first two or three harmonics of gearmesh frequencies.

The results indicate that this should provide a viable solution to the problem of reducing the noise radiated from constant speed gearboxes, by optimising the placement of stiffening ribs to place stop bands around the harmonics of the fixed gearmesh frequencies.

ACKNOWLEDGEMENTS

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