

Analytical Technique for Predicting Passenger Car Gearbox Structure Noise Using Vibration response analysis

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ABSTRACT

The gear pair assembly remains one of the major noise and vibration sources in power transmission systems typically used in automotive, aerospace and industrial applications. The gear vibration and noise signatures are often dominated by several high-level tonal peaks that occur at the fundamental gear mesh frequency and its harmonics. In this paper, development of the concept of a method for predicting the contributions to noise radiated by a vehicle gearbox structure using vibration response measured. An idealization of the gearbox surface as a set of flat plates was used to calculate the radiation efficiency from physical properties and edge constraints of each plate, and the vibration response of the gearbox structure surface was measured using accelerometers. These data were used in a simple acoustical power theory to determine 1/3-octave band sound pressure level under free field conditions for the whole and individual noise source. Moreover, the prediction of the individual noise sources in order to pinpoint the exact problem area in the gearbox structure so that treatment or redesign can be carried out in correct way.

Keywords: Idealization, Vibration Velocity, Noise, Radiation Efficiency; accelerometers, Air Density, Predicted Sound Pressure, 1/3-Octave band, Frequency

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NOMENCLATURE

a_i	area of i plate (panel) in the idealization	m^2
A	area of set of panels	m^2
ρ	density of air	$Kg.m^{-3}$
C	speed of sound in air	$m.s^{-1}$
I	sound intensity	$W.m^{-2}$
P	acoustic pressure	$N.m^{-2}$
S	area of a surface in a defined acoustic field = $2\pi r$ for free field conditions	m^2
r	distance	m
V	mean square time and space average velocity	$m.s^{-1}$
α	panel area ratio	
γ	wave number ratio	
$F_{m,n}$	fundamental natural frequency of panel	$Rad. S^{-1}$
E	Young's modulus	$N.m^{-2}$
a, b	panel dimensions (X,Y)	m
W_{rad}	radiated acoustic power	W
σ	radiation efficiency	
ν	Poisson's ratio	
$W_{vib.}$	vibration power	W
t	panel thickness	m
ρ_p	density of panel material	$Kg.m^{-3}$
m, n	Number of 1/2- sine wave along sides a, b (modal number)	

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1. INTRODUCTION

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A methodology for predicting noise and vibration of machines and their support structures has been introduced. Included is a heuristic energy-based criterion to assess the importance of dynamic coupling between a mechanism and its support structure based on a simplified analysis of a mechanism operating on a rigid base, which neglects the coupling. Also included was an analysis method that considers the coupling and to be used when the criterion reveals important coupling. The methodology was implemented using highly idealized closed form, more elaborate numerical descriptions and was checked against vibration and sound measurements of a plate subjected to periodic impacts by balls and a beam that rattles within a clearance bearing. The energy-based criterion was found to indicate situations in which mechanism-support coupling affects noise radiation. In some cases the coupling was observed to significantly affect vibration and noise radiation of the support structure, while having a relatively minor effect on mechanism response. Both the simple closed form and numerical descriptions are found to predict noise trends due to variations in machine speed and bearing clearance, and the numerical descriptions more accurately predict overall and band levels of noise radiation [1].

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The impulsive responses in geared systems with multiple clearances when the mean torque excitation and system load change abruptly, with application to a vehicle driveline with an automatic transmission has been studied. First, torsional lumped-mass models of the planetary and differential gear sets were formulated using matrix elements. The model was then reduced to address tractable nonlinear problems while successfully retaining the main modes of interest. Second, numerical simulations for the nonlinear model were performed for transient conditions and a typical driving situation that induces an impulsive behavior simulated. However, initial conditions and excitation and load profiles had to be carefully defined before the model can be numerically solved. It was shown that the impacts within the planetary or differential gears may occur under combinations of engine, braking and vehicle load transients. Their analysis shows that the shaping of the engine transient by the torque converter before reaching the clearance locations was more critical. Third, a free vibration experiment is developed for an analogous driveline with multiple clearances and three

experiments that excite different response regimes had been carried out. Good correlations validate the proposed methodology [2]

An investigation with the rattle noise, caused by the fluctuation of the engine torque (acyclic excitation) which, under special conditions, can cause multiple impacts inside the gearbox has been presented. The rattle phenomenon in a gearbox was described experimentally. First, a fully instrumented test rig consisting of a simplified gearbox was designed in order to recreate the rattle noise phenomenon for a multi harmonic excitation imposed to the input shaft of the gearbox. Second, different gearbox configurations were used to characterize the rattle threshold and the rattle noise evolution, in relation to excitation parameters and mechanical gearbox parameters. Third, a simplified model of the rattle phenomenon is drawn, aiming to determine the most significant parameters to the rattle noise [3].

The optimization of gearbox geometric design parameters to reduce rattle noise in an automotive transmission based on a torsional vibration model approach has been introduced. Rattle noise was calculated and simulated based on the design parameters of a 5-speed gearbox, and all pinion gears and wheel gears are helical. The effect of the design parameters on rattle noise was analyzed. The observed rattle noise profiles were obtained depending on the design parameters. During the optimization, a four-degree-of-freedom torsional vibration model of the pinion gear-wheel gear system was obtained and the minimum singular value of the transfer matrix was considered as the objective functions and design parameters were optimized under several constraints that include bending stress, contact stress and a constant distance between gear centers. Therefore, by optimizing the geometric parameters of the gearbox such as, the module, number of teeth, axial clearance, and backlash, it was possible to obtain a light-weight-gearbox structure and minimize the rattling noise. It was concluded that the optimized geometric design parameters lower the rattle noise by 10% compared to the calculated rattle noise for sample gearbox. All optimized geometric design parameters also satisfy all constraints [4].

Gear vibrations have been studied intensively for many years. They are mainly treated as forced damped vibrations caused by stiffness fluctuation of the gear teeth in mesh. Such approach gives good results in the sub-critical and resonant mesh frequency range. In the super-critical frequency range the vibration level increases slightly with the increase of the gear speed of rotation. The starting point of this work is the assumption that gear vibrations are free damped in which excitation is repeated by teeth impact at each entrance of teeth into the meshing-singular process. The new calculation model based on singular systems theory and on measurement results has been worked out. This model supports the measurement results in all three frequency ranges, the sub-critical, resonant and supercritical [5].

A study concentrated on modeling the complex vibro-acoustic behavior using a three-dimensional finite element (FE) approach for the structure combined with Rayleigh integral (RI) method for the acoustic radiation process has been introduced. A model analysis method is used to evaluate the elasto-acoustic modal characteristics for the coupled system. Effect of fluid inside the gearbox on the vibration response was discussed. The combined use of this FE/RI model enables evaluation of the acoustic response. A case study of a simplified gearbox internally excited with gear mesh stiffness fluctuation was presented. Vibratory analysis allows concluding that the vibro-acoustic coupling between elastic housing, air cavity and free acoustic field have not to be neglected [6].

Some insights has been provided on the vibration and radiated noise characteristics of a petrol engine structure, and possibly some recommendations which might be applicable to the solution of vibro-acoustic problem associated with the petrol engine structure. It is based on the concept of creation of the measurements, while the engine is in running conditions in the laboratory. Based on this concept a complete appraisal is made for the engine structure

and the reasons for its particular vibro-acoustic response were exposed. The potentially noisy surface areas forming the engine structure have been identified [7].

However, the objective of this paper In this paper, development of the concept of a method for predicting the contributions to noise radiated by a vehicle gearbox structure using vibration response measured.

2. RELATIONSHIP BETWEEN SURFACE VIBRATION AND RADIATED NOISE

2.1 Sound Power Theory

The theory of sound power radiated by a vibrating surface is dependent on the following parameters [8]:

- Velocity response of the vibrating surface
- The coupling coefficient between the vibrating surface and the surrounding space, and
- The surface area of the vibrating surface.

The equation relating these parameters is,

$$W_{rad} = W_{vib} \cdot \sigma = \rho \cdot C \cdot A \cdot \bar{V}^2 \cdot \sigma \quad (1)$$

States that the radiated acoustic power is dependent on the product of vibratory power in the surface and the efficiency with which power is transferred from the surface to its surrounding. For free fields conditions,

$$W_{rad} = I \cdot S \quad (2)$$

$$I = P^2 / \rho \cdot C \quad (3)$$

Substituting for W_{rad} in equation (1) and transposing, the space averaged sound pressure level (SPL) can be obtained as

$$P^2 = \rho^2 \cdot C^2 \cdot \bar{V}^2 \cdot \sigma \cdot A \cdot S^{-1} \quad (4)$$

Equation (4) determines the mean square space average SPL, that SPL which be expected at any point on a hypothetical surface in space at a given distance "r" from the source. The distance "r" must be at least equal to 0.6 meters to avoid the near field effects. This is assuming that the source is radiating sound uniformly fro its surface.

2.2 The Radiation Efficiency

The radiation efficiency (σ) which is presented in equation (4) can be established for the surface of a gearbox structure by idealizing the structure as an assembly of simple flat plates (panels) as shown in **Fig. 1**. This idealization has been chosen because there is evidence that parts of the gearbox do vibrate in a plate like manner and because there is available analytically derived data on radiation efficiency obtained by [9]. The details of estimated the individual plate in the idealization is presented in **APPENDIX**.

To calculate the radiation efficiency of a set of plates (panels). It is necessary now to account for the effect of each panel has on the sound radiated by the whole set. For this calculation

the panels are considered as independent panels with surface areas $a_1, a_2, a_3, \dots, a_n$ $A = \sum_{i=1}^{i=n} a_i$

is the total surface area of the set of panels. Therefore, the proportional effect on the radiation efficiency of the i^{th} panel on that of the whole set is given by $\sigma_i \cdot a_i / A$. The radiation efficiency of the set therefore being

$$\sigma_T = \sum_{i=1}^{i=n} \sigma_i \cdot \alpha_i \quad (5)$$

Where $\alpha_i = a_i / A$

Now, since (σ) is a quantity which varies with frequency as well as plate size, it is convenient to represent this in matrix form for calculation, thus.

$$\begin{bmatrix} \sigma_{11} & \dots & \sigma_{12} & \dots & \sigma_{1n} \\ \sigma_{21} & \dots & \sigma_{22} & \dots & \sigma_{2n} \\ \dots & \dots & \dots & \dots & \dots \\ \sigma_{m1} & \dots & \sigma_{m2} & \dots & \sigma_{mn} \end{bmatrix} \cdot \begin{bmatrix} \alpha_1 \\ \alpha_2 \\ \dots \\ \alpha_n \end{bmatrix} = \begin{bmatrix} \sigma_{T1} \\ \sigma_{T2} \\ \dots \\ \sigma_{Tm} \end{bmatrix} \quad (6)$$

$$\underline{\sigma} \cdot \underline{\alpha} = \underline{\sigma}_T \quad (7)$$

where $\underline{\sigma}$ is the matrix of m frequency rows and n panel (plate) columns, and $\underline{\alpha}$ is the vector of n panel area ratio (a_n/A).

2.3 The Vibration Response

The velocity response of the vibrating surface ($\bar{v}$) which appears in equation (4) cannot be expressed by a simple mathematical function. This means that this velocity needs to be measured. However, based on analysis, any structure has to be idealized as a set of independent plates (panels), for each of which an average vibration velocity can be determined. Therefore, the accuracy of the predicted sound pressure level (SPL) based on this theory will be limited by the number of structure panels used in idealization. Hence, the bigger the number of panels the more accurate will be the prediction of SPL resulting from the overall structure panels or part of it. The vibration velocity measurements must include narrow, 1/3 octave and RMS spectra form, when the RMS vibration velocity response $\langle \bar{V}_T \rangle$ of each area represented by a panel in the idealization has been measured in the octave spectrum form. The matrix solution can be applied to obtain the overall vibration velocity response spectrum for the structure represented by a set of panels.

To calculate the overall vibration velocity response of a set of panels, it is necessary to account for the effect of each panel has on the vibration velocity produced by the whole set. For this calculation, the proportional effect of the vibration velocity response of the i^{th} panel on that of the whole set is given by $\langle \bar{V}_T \rangle \cdot \alpha_i / A$. The vibration velocity of the set therefore being:

$$\langle \bar{V}_T \rangle = \sum_{i=1}^{i=n} \langle \bar{V}_i \rangle \cdot \alpha_i \quad (8)$$

Now, since $(\bar{V})$ is a quantity which varies with frequency as well as plate size, it is convenient to represent this in matrix form similar to equation (6).

$$\underline{\bar{V}} \cdot \underline{\alpha} = \langle \bar{V} \rangle_T \quad (9)$$

Having obtained values of σ_T and $\langle \bar{V} \rangle_T$ from structure idealization and vibration velocity response measurement, and a value for AS^{-1} from a knowledge of structure surface area and acoustic field idealization, equation (4) can be used to obtain 1/3 octave SPL either in dB or in dB(A)

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201 3. EXPERIMENTAL METHODOLOGY

202 3.1 Experimental Setup

203 The experimental setup is used for the vehicle gearbox manual transaxle system tested.
204 The vehicle gearbox manual transaxle system shift gears are tabulated in **Table 1**. The
205 entire system is settled in an oil basin in order to ensure proper lubrication. SAE 75W-90 oil
206 (ESSO Gear Oil GX) was used as a standard lubricant oil type. The full lubrication level is
207 100 mm and half lubrication level is 50 mm. The lubricant oil type specifications are listed in
208 **Table 2**. The gearbox is powered by the electric motor and consumes its power on a
209 hydraulic disc brake. A short shaft of 20 mm diameter is attached to the shaft of the motor
210 through a flexible coupling. This is to minimize effects of misalignment and transmission of
211 vibration from the motor. The shaft is supported at its ends through two ball bearings and
212 then the motion is transmitted directly to the gearbox. The system is illustrated schematically
213 in **Fig. 2**.

214 3.2 Test Procedure

215 Prior to testing, the gearbox had been separated from its vehicle and put on a heavy steel
216 stand made and mounted on it. The reason for doing this is to measure the gearbox structure
217 response away from other vehicle elements (tire/road interface, engine, etc) not only, but
218 also to allow the gearbox structure, to be tested with and without loads at different speed
219 conditions. The vibration velocity responses for the gearbox structure were measured by
220 accelerometers which had been mounted on the center of each panel and element surfaces.
221 The accelerometer signals were recorded were passed to the data analysis system analyzer
222 (PULSE labshop and PULSE analyzer). At the same time the gearbox noise in terms of
223 sound pressure level were by 1/2 in microphone placed at mainly four points around the
224 gearbox structure areas (right hand side, top, Left hand side and rear). The microphone
225 signals were passed to the data analysis system analyzer as in vibration velocity
226 measurements. The motor speed was measured by Photoelectric Tachometer Probe which
227 placed in front of the gearbox input shaft and its signals passed to the data analysis system
228 analyzer. Also, the torque load was computed by using the oil pressure measured on vehicle
229 wheel axle and transferred to the gearbox through the reduction ratio for the differential and
230 gearbox with the dimensions of the brake master cylinder and brake disc. The motor speed
231 was varied up to 1000 rpm, torque load is 12 Nm and the frequency range is 2000 Hz.

232 4. PREDICTION PROCEDURE DETAILS

233 The idealization of the vehicle gearbox structure panels and elements used in this thesis is
234 shown in **Fig. 1**, where four major panels are considered, i.e. right hand side (R.H.S), top, left
235 hand side (L.H.S) and rear. Each of which is divided into 4 elements areas. The panels and
236 elements dimensions, physical properties and end conditions are also included in the figure,
237 from which the fundamental modes (natural frequency) were computed based on equation
238 (A.1), **APPENDIX**. Consequently, the values of the normal modes radiation efficiency (σ) for
239 the idealized panels and elements were calculated according to equation (6). Furthermore,
240 the equivalent mean square vibration velocity of the rigid piston obtained by space and time
241 averaging the velocity of a surface ($\bar{V}$). It can not expressed by a simple mathematical
242 function. This means that this vibration velocity needs to be measured. It has to be noted that
243 the greater the number of elements (points) at which the vibration velocity measurements are
244 taken, the greater the accuracy of the estimation of ($\bar{V}$). The gearbox structure obviously,
245 has many modes of vibration, and it would be important critical to attempt to define the all to
246 find the position of maximum response for each mode. When the response ($\bar{V}$) of each area
247 represented by a panel in the idealization has been calculated in octave band spectrum form,
248 the matrix solution presented in equation (9) was applied to obtain the response spectrum for
249 an area of the whole gearbox structure represented by a set of panels. This can account the
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effect of each panel has on the vibration velocity response produced by whole set. Accordingly, when a surface in the gearbox structure at a discrete frequency and with a predictable mode shape, it is possible to calculate the sound pressure level (SPL) at a point above the surface. This calculation was achieved by using equation (4). Even so, a considerable amount of data is necessary to quantify the behavior of gearbox structure over a limited frequency band. The use of these analytical techniques on complex structure such as vehicle gearbox is completely precluded by the enormity of the calculation. However, by considering the total power radiated rather than pressure at discrete points, and by considering the use of one-octave bands rather than discrete frequencies, it is possible to obtain a considerable reduction in the calculation effort.

5. RESULTS AND DISCUSSION

Sample from the sound pressure level (SPL) in the form of narrow and 1/3-Octave bands for right hand side (R.H.S) panel are shown in **Figs. 3 and 4** respectively. These data present in the figures will be used later for the comparison between the predicted and measured SPL. **Figs. 5 and 6** show sample from the vibration velocity responses in the form of narrow and 1/3-Octave bands for element No.1 respectively.

5.1 Various Elements Measured Vibration Velocity

Figures 7 to 10 show the 1/3-octave bands of vibration velocity responses spectra (m/s) measured at the centers for all the elements individually considered in gearbox structure idealization. The gearbox structure has to be spitted into four major panels. Each major panel has four individual elements areas as shown in **Fig. 1**, which are right hand side (R.H.S), top, left hand side (L.H.S) and rear. The motor speed is being 1000 rpm, torque load and oil type of 75W90. Generally speaking, the spectra exhibit the same trend with different peaks and different locations. A common peak exists at 1/3-octave band of 50 Hz, while the level is different due to the element location in the idealization, its area and its end conditions. **Fig. 11** shows the overall RMS values for the individual element, where element No. 3 (R.H.S-panel) has the highest vibration velocity response followed by element No. 6 (top-panel), element No. 9 (L.H.S-panel) and element No. 15 (rear-panel).

5.2 Gearbox Structure Panels Radiation Efficiency

The radiation efficiencies for the individual the vehicle gearbox structure which are considered in the idealization along with the overall averages and are presented in **Table 3**. The gearbox structure has to be spitted into four major panels. Each major panel has four individual elements areas as shown in **Fig. 1**, which is right hand side (R.H.S), top, left hand side (L.H.S) and rear. The overall averages of the radiation efficiency for the individual elements are shown in **Fig. 12**, where the radiation efficiencies for the individual elements in each major panel have an equal value. This is true, where all the elements in any major panel are equal in area.

5.3 Comparison between Measured and Predicted SPL

Having calculating the whole average radiation efficiency and the whole RMS vibration velocity for the R.H.S panel based on equations (6) and (9) are plotted in **Figs. 13 and 14** respectively. The motor speed is being 1000 rpm with torque load of 12 Nm and oil type of 75W90.

Figure 6.15 shows the comparisons between 1/3-octave bands spectra sound pressure level (SPL) predicted in dB (A) based on equation (4) and those measured and shown in **Fig. 4**, which have a similar trend shapes. The predicted SPL spectrum is slightly higher than that for the measured spectra particularly over 1/3-octave of 250 Hz (**Fig. 15**). The fit of the predicted to the measured is not quite as close as expected but this is still acceptable. It observed that the SPL can be predicted to within 3 dB (A) of the measured level in each 1/3-Octave band. The SPL is expressed by dB (A) reference to $20 \times 10^{-5} \text{ N/m}^2$.

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307 **5.4 Various Elements Noise Source Contribution**

308 The element 1/3-octave bands of predicted at 0.6 m noise source in terms of SPL, dB(A)
309 contributions from the gearbox structure elements considered in this study were calculated
310 individually based on equation (4) and are shown in **Figs. 16 and 19**. The gearbox structure
311 has to be spitted into four major panels. Each major panel has four individual elements areas
312 as shown in **Fig. 1**, which is right hand side (R.H.S), top, left hand side (L.H.S) and rear. The
313 motor speed is being 1000 rpm without load and oil type of 75W90. Generally speaking, the
314 spectra exhibit the same trend with different peaks and different locations. A common peak
315 exists at 1/3-octave band of 50 Hz, while the level is different due to the element location in
316 the idealization, its area and its end conditions. **Fig. 20** shows the overall RMS values for the
317 individual element, where element No. 11 (L.H.S-panel) has the highest sound pressure level
318 response followed by element No. 5 (top-panel) and element No. 7 (top-panel).

319 **6. CONCLUSIONS**

320 1. The potentially noisy surface areas forming the vehicle gearbox structure have been
321 identified. The effect of vehicle gearbox speed and load on the performance of the vehicle
322 gearbox structure are included and critically examined with respect of improvements.

323 2. It has been shown that the noise in terms of sound pressure levels can be predicted to
324 within 6 dB (A) of those measured levels. Moreover, a reliable method of ranking the noise
325 sources based on the idealization technique and vibration measurements has been
326 developed which removes much of the uncertainty from noise reduction exercise.

327 3. The radiation efficiency of the vehicle gearbox surface can be calculated from an
328 idealization of the gearbox surface as a set of independent vibrating panels, without using
329 microphone measurements.

330 4. The calculation procedure lends itself to being fully computerized and providing that the
331 idealization and test is carefully designed, reliable frequency analysis can be achieved.
332 Furthermore, the measurements and analysis procedures presented in this thesis can be
333 applied on any structure, if the major noise sources need to be identified and treated.

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Table 1 Manual Transaxle System Specifications

S/No.	Model		M5EF2
1	Type		Forward 5 speed, reverse 1 speed
2	Engine displacement		1.1 MPI engine
3	Gear ratio	First	3.545
		Second	1.894
		Third	1.192
		Forth	0.853
		Fifth	0.719
		Reverse	3.636
4	Final gear ratio		4.437

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Table 2 Oil Type Lubricant Specifications

S/No.	SAE Grade	SAE 75W-90	Remarks
1	Viscosity Index	177	
2	Flash Point, COC: °C	170	
3	Pour Point °C	-42	
4	Kinematic viscosity@40°C	92.6	
5	Kinematic viscosity@100°C	15.4	
6	Density@ 15°C	875	

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Table 3 Predicted radiation efficiency (σ) of manual gearbox structure panels elements

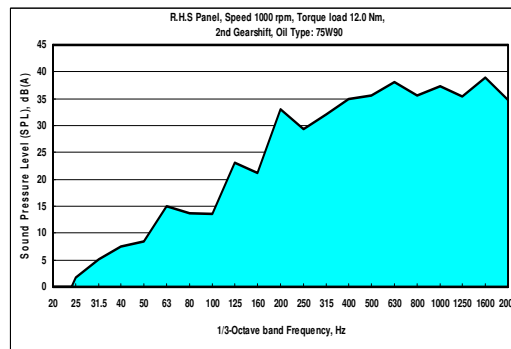
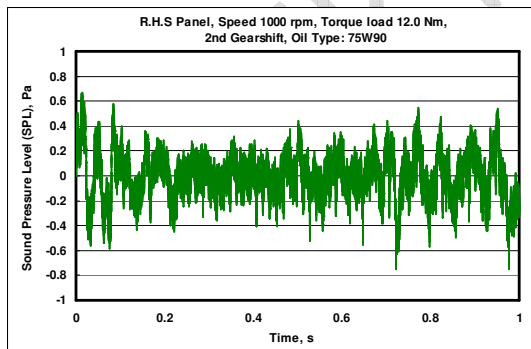
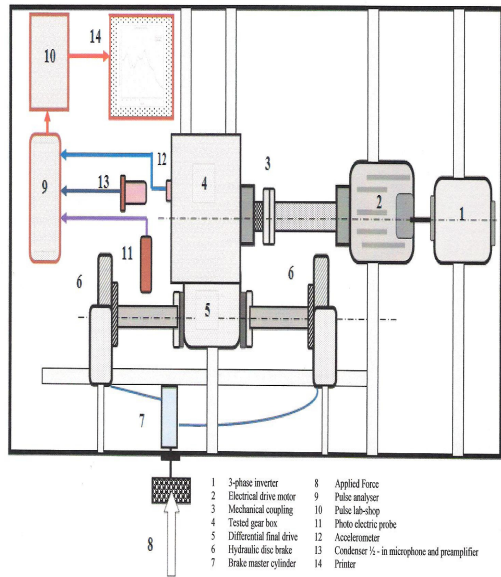
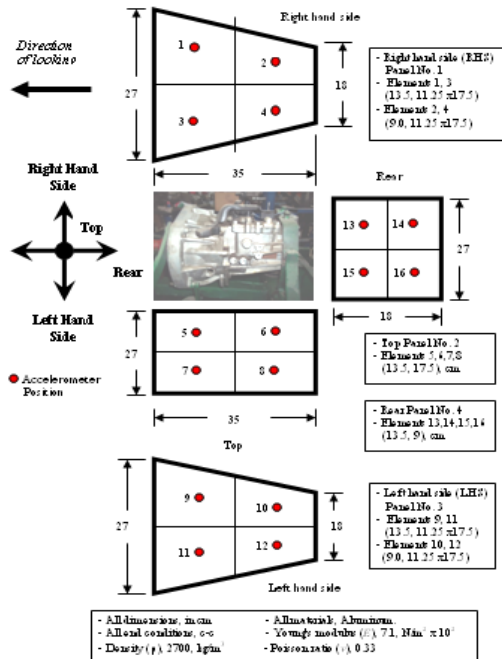
P. N	R.H.S				TOP				L.H.S				REAR			
El. N	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
F. N																
20	0.0006	0.0008	0.0008	0.0007	0.0007	0.001	0.001	0.001	0.0006	0.001	0.001	0.0008	0.0007	0.0007	0.0007	0.0007
25	0.0007	0.0009	0.0009	0.00075	0.0008	0.0008	0.001	0.00075	0.0007	0.001	0.001	0.0009	0.00075	0.00075	0.00075	0.00075
31.5	0.0008	0.002	0.002	0.0008	0.0008	0.0008	0.001	0.0008	0.0008	0.002	0.001	0.002	0.0008	0.0008	0.0008	0.0008
40	0.00095	0.004	0.004	0.0009	0.0009	0.0009	0.001	0.0009	0.00085	0.004	0.001	0.004	0.0009	0.0009	0.0009	0.0009
50	0.0025	0.005	0.005	0.001	0.0025	0.0025	0.003	0.0025	0.0025	0.005	0.003	0.005	0.001	0.001	0.001	0.001
63	0.003	0.0055	0.0055	0.0025	0.004	0.004	0.004	0.004	0.003	0.006	0.003	0.0055	0.0025	0.0025	0.0025	0.0025
80	0.005	0.006	0.006	0.0065	0.005	0.005	0.005	0.005	0.005	0.006	0.005	0.006	0.0065	0.0065	0.0065	0.0065
100	0.007	0.007	0.007	0.008	0.0075	0.0075	0.0075	0.0075	0.007	0.007	0.007	0.007	0.008	0.008	0.008	0.008
125	0.008	0.008	0.008	0.009	0.0085	0.0085	0.0085	0.0085	0.008	0.008	0.008	0.008	0.009	0.009	0.009	0.009
160	0.01	0.01	0.01	0.01	0.03	0.03	0.03	0.03	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
200	0.045	0.035	0.035	0.05	0.05	0.05	0.05	0.05	0.045	0.035	0.045	0.035	0.05	0.05	0.05	0.05
250	0.06	0.05	0.05	0.06	0.065	0.065	0.065	0.065	0.06	0.05	0.06	0.05	0.06	0.06	0.06	0.06
315	0.065	0.053	0.053	0.07	0.07	0.07	0.07	0.07	0.065	0.053	0.065	0.053	0.07	0.07	0.07	0.07
400	0.068	0.056	0.056	0.075	0.075	0.075	0.075	0.075	0.068	0.056	0.068	0.056	0.075	0.075	0.075	0.075
500	0.07	0.06	0.06	0.08	0.08	0.08	0.08	0.08	0.07	0.06	0.07	0.06	0.08	0.08	0.08	0.08
630	0.085	0.07	0.07	0.09	0.085	0.085	0.085	0.085	0.085	0.07	0.085	0.07	0.09	0.09	0.09	0.09
800	0.09	0.08	0.08	0.1	0.09	0.09	0.09	0.09	0.09	0.08	0.09	0.08	0.1	0.1	0.1	0.1
1000	0.095	0.09	0.09	0.13	0.095	0.095	0.095	0.095	0.095	0.09	0.095	0.09	0.13	0.13	0.13	0.13
1250	0.097	0.093	0.093	0.4	0.097	0.097	0.097	0.097	0.097	0.093	0.097	0.093	0.4	0.4	0.4	0.4
1600	0.098	0.095	0.095	0.5	0.098	0.098	0.098	0.098	0.098	0.095	0.098	0.095	0.5	0.5	0.5	0.5
0.1	0.1	0.1	0.1	0.6	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.6	0.6	0.6	0.6
Aver.	0.076	0.083	0.075	0.200	0.200	0.083	0.076	0.200	0.200	0.083	0.076	0.083	0.088	0.088	0.088	0.088

P. N.: Panel name

El. N.: Element Number

F. N.: Frequency 1/3-octave band

347
348



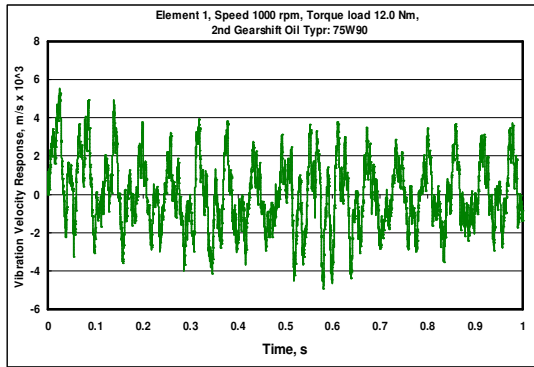


Fig. 5 Time history (domain) of measured vibration velocity for element 1

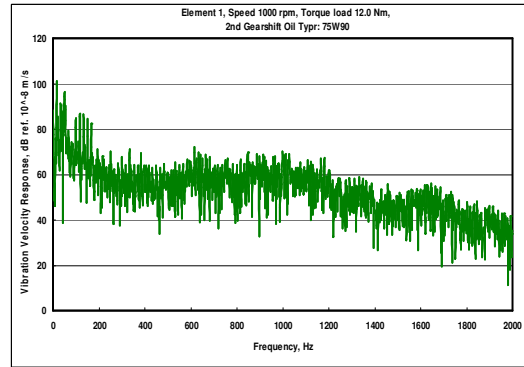


Fig. 6 Frequency domain of measured vibration velocity for element 1

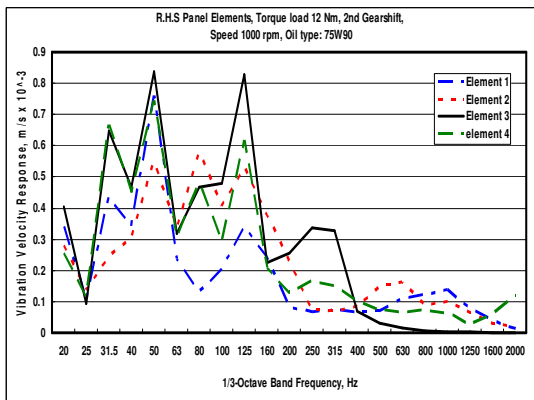


Fig. 7 1/3-Octave band vibration velocity measured at the center of R.H.S panel elements

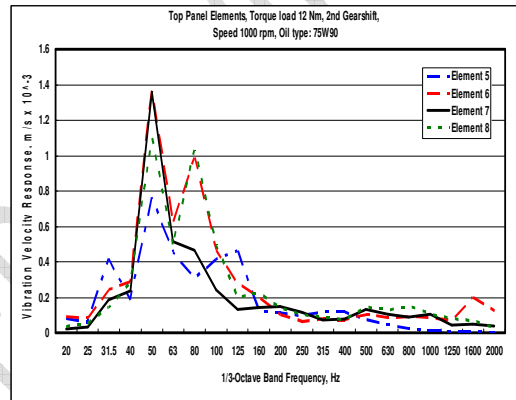


Fig. 8 1/3-Octave band vibration velocity measured at the center of top panel elements

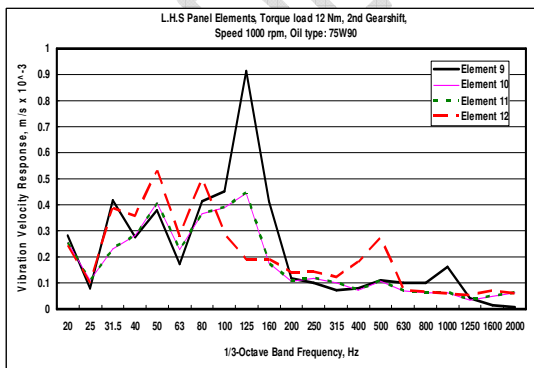


Fig. 9 1/3-Octave band vibration velocity measured at the center of L.H.S panel elements

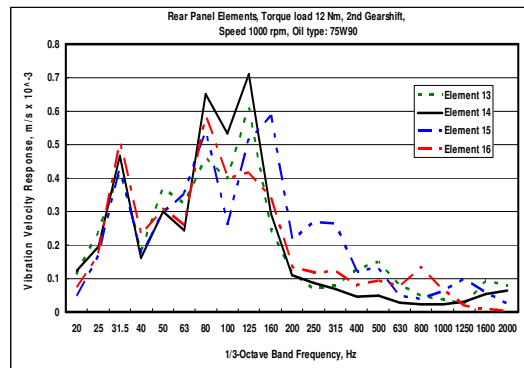


Fig. 10 1/3-Octave band vibration velocity measured at the center of rear panel elements

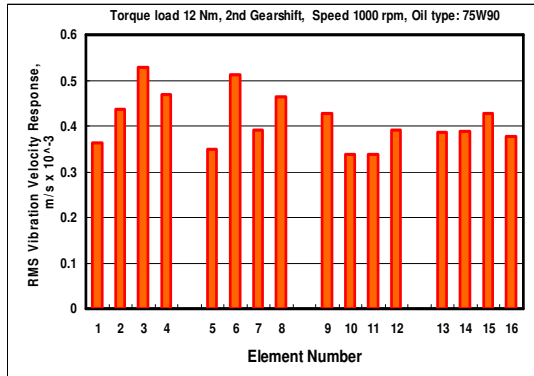


Fig. 11 Overall RMS vibration velocity calculated for all the gearbox structure elements

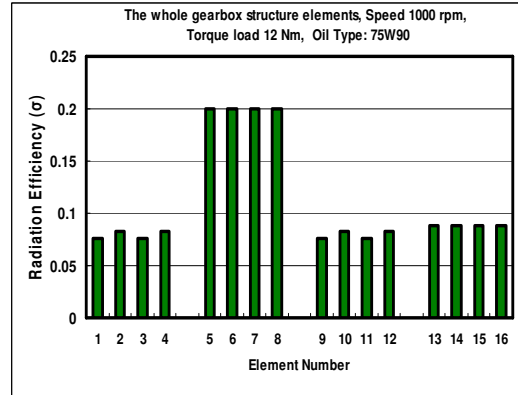


Fig. 12 Overall average of radiation efficiency calculated for all the gearbox structure elements

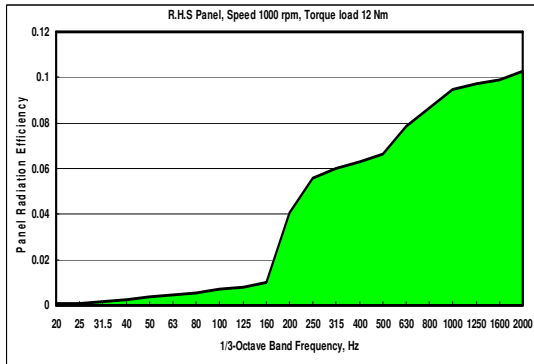


Fig. 13 The Whole average radiation efficiency calculated for the R.H.S panel

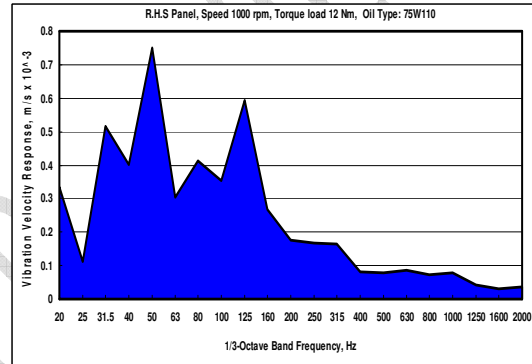


Fig. 14 The Whole RMS vibration velocity calculated for the R.H.S panel- Oil type 75W90

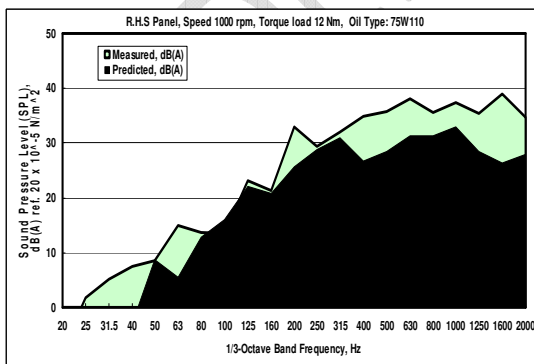


Fig. 15 Comparison between measured and predicted SPL calculated the R.H.S panel- Oil type 75W90

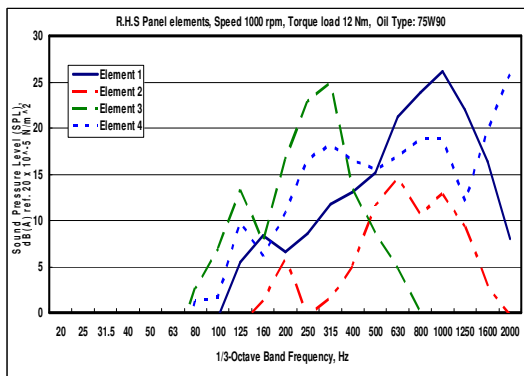


Fig. 16 1/3-Octave band sound pressure level (SPL) predicted in front of the center of panel R.H.S elements

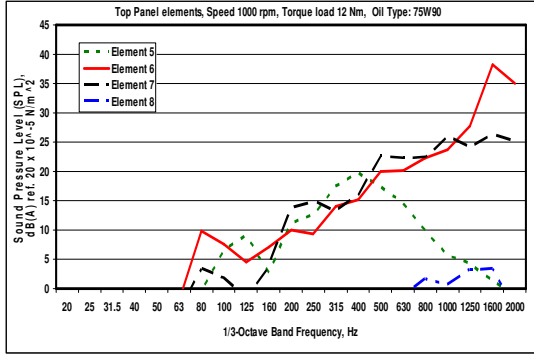


Fig. 17 1/3-Octave band sound pressure level (SPL) predicted in front of the center of top panel elements

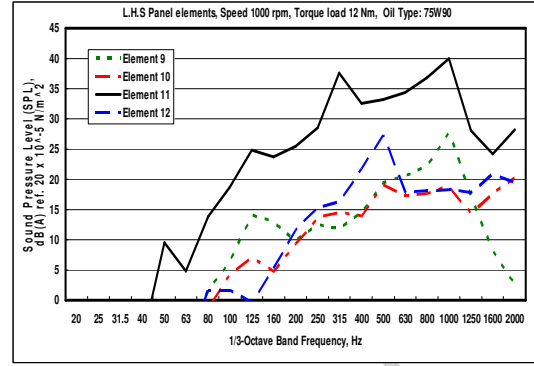


Fig. 18 1/3-Octave band sound pressure level (SPL) predicted in front of the center of L.H.S panel elements

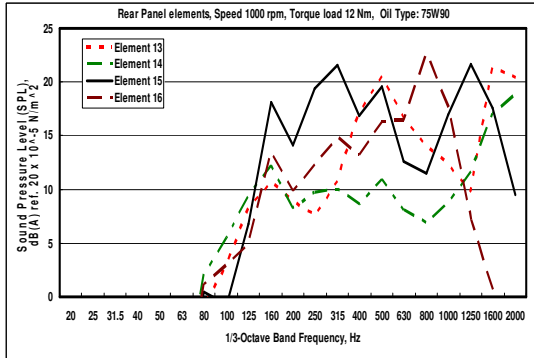


Fig. 19 1/3-Octave band sound pressure level (SPL) predicted in front of the center of rear panel elements

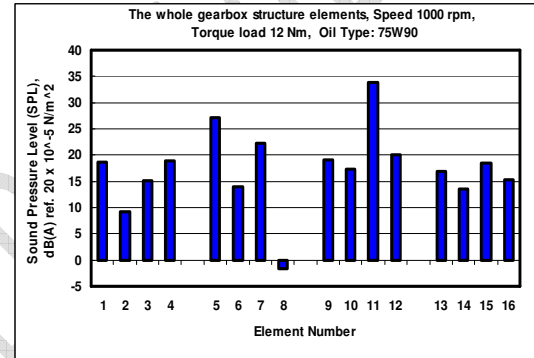


Fig. 20 Overall RMS sound pressure level (SPL) predicted for all the gearbox structure elements

APPENDIX

Radiation Efficiency Estimation

The fundamental modes can be calculated from an approach based on the Reyleigh Ritz classical analysis. This approach contains modal frequencies for simply support rectangular plate as shown in **Fig. A.1** using the following equation:

$$F_{m,n} = \sqrt{Et^3 / \rho_p (1 - \nu^2) 12 \{ (n\pi/a)^2 + (n\pi/b)^2 \}} \quad (\text{A.1})$$

In practical this is not always the case. Other boundary conditions (clamped or fixed) must be applied using the method described in [10], where values $F_{m,n}$ for clamped edge plates can be obtained as following:

- For a plate clamped on both edges in 'x' direction $\bar{b} \Rightarrow \frac{1.05n}{n+0.5} b$ (A.2)

- For a plate clamped on both edges in 'y' direction $\bar{a} \Rightarrow \frac{1.05m}{m+0.5}a$ (A.3)

The values of the radiation efficiency (σ) for various modes of vibration of a simply supported rectangular plate were derived by [9]. The simplification to Wallace's radiation efficiency curves could be made for third octave analysis. This work showed that calculated values were at least as accurate as measured results for radiation efficiency because the third octave filtering effectively merges the radiation efficiency of modes within third octave band. This simplification reduces the curves to one for the fundamental mode and one for the average of all other modes. Those curves are plotted against the non-dimensional wave number ratio (γ) as shown in **Fig. A.2**. The ratio (γ) is defined as follows:

$$(\gamma) = (\text{acoustic wave number})/(\text{structure wave number}) = K / K_p \quad (\text{A.4})$$

$$\text{Where } K = (F_{mn})/C \quad \text{and} \quad K_p = [(m\pi/a)^2 + (n\pi/b)^2]^{1/2} \quad (\text{A.5})$$

The relationship between wave number ratios at the fundamental mode frequency ($f_{1,1}$) and any other frequency for a simply supported rectangular plate is well documented in [10]. This shows that the wave number ratio for 1/3 octave band ($\gamma_{\Delta c}$) whose centre frequency is ($f_{\Delta c}$) is given by,

$$\gamma_{\Delta c} = (F_{\Delta c} / F_{1,1})^{1/2} \cdot \gamma_{1,1} \quad (\text{A.6})$$

Using equation (3.10) , and the curves of **Fig. A.2**, the predicted radiation efficiency of each plate considered can be determined

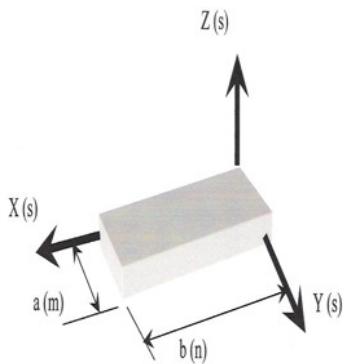


Fig. A.1 Simple support plate

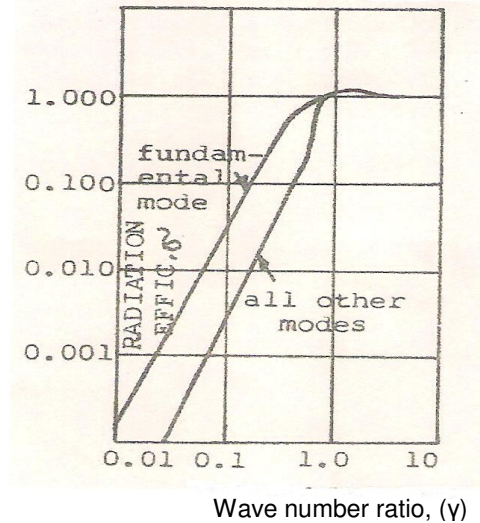


Fig. A.2 Radiation efficiency as a function