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VEHICLE INTERIOR NOISE AND VIBRATION PREDICTION USING THE ADAMS SIMULATION TOOL

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ABSTRACT

The customer perception of a vehicle plays a key role in his purchasing decision. Amongst other factors the interior noise and vibration levels and the quality of the sound strongly influence the character of the vehicle.

The use of the multibody dynamic simulation package ADAMS for the optimisation of powertrain mounting systems is discussed. A method of combining the results of an ADAMS simulation with measured or predicted noise transfer functions to create a 'hybrid' model capable of predicting interior noise is shown. The application of this technique is demonstrated by a case history of the optimisation of the mounting system of a three cylinder powertrain. The interior noise and vibration predictions are shown validated against experimental measurements. The results of optimisation of the powertrain mounting system and engine balancing strategy are shown.

INTRODUCTION

The customer perception of a vehicle plays a key role in the purchasing decision. The interior noise and vibration levels and the quality of the sound strongly influence the character of the vehicle.

The interior noise and vibration levels of a vehicle are the sum of many sources but a major noise and vibration path for low frequency noise is the structural connection between the powertrain and the vehicle body using the engine mounts. This paper describes the use of the ADAMS multibody dynamic simulation package for the prediction of interior noise. A case history is given showing the application of these techniques to the optimisation of the mounting system of a three cylinder powertrain.

THE DESIGN OF POWERTRAIN SUSPENSION SYSTEMS USING ADAMS

There are conflicting requirements placed on the design of an automotive powertrain mounting system. In general, a stiff mounting system is required for good ride and handling performance while a soft mounting system is required for good isolation of the vehicle body from powertrain vibration. Further restrictions are imposed on the design of a mounting system by durability requirements, packaging constraints and interference of the powertrain with other parts of the vehicle.

The ability of ADAMS to carry out time domain solutions of non-linear systems makes it a very suitable tool for the analysis and optimisation of powertrains suspended on progressive mounts subject to time dependent forcing functions. In particular ADAMS may be used to carry out the following types of analyses:

1) The simulation of powertrain motion and mount deflection due to driveline loads and vehicle accelerations. This type of analysis is used to check for the interference of the powertrain with other parts of the vehicle and to ensure that mount deflections are within the limits governed by durability requirements.

2) The simulation of the powertrain rigid body modes using ADAMS/Linear. This type of analysis is used to ensure that the powertrain modes are separated from each other and known vehicle and powertrain forcing frequencies for good vehicle dynamics and noise and vibration isolation.

3) The simulation of the motion of the major engine components is used to predict the forced response of the powertrain, to low frequency engine excitation. This type of analysis can be used to predict the excitation of the vehicle body by the dynamic forces at the body side of the powertrain mounts. The vehicle interior noise can then be predicted by the application of noise transfer functions to the forces predicted at the powertrain mounts.

The remainder of this paper concentrates on the last of these analyses.

INTERIOR NOISE PREDICTION USING ADAMS

<u>General Approach</u>

In most cases the most important path for low frequency noise and vibration into the vehicle is structureborne via forces transmitted into the vehicle body at the mounting points and the amplification of these vibrations as they progress towards the passengers of the vehicle. The interior noise and vibration levels can thus be predicted by a two step process. Firstly, the forces at the mount positions are predicted using ADAMS. Secondly, noise transfer functions are applied to predict the vehicle interior noise. The following sections describe these two steps in more detail.

Mount Force Prediction

An ADAMS model of all the engine components which have a significant effect on low frequency forces at the engine mounts is generated. This would normally include representations of the rigid powertrain structure, pistons, connecting rods, crankshaft, flywheel and balancer shafts, as well as the non-linear stiffness and damping characteristic of each engine mount. An 'engine modeller' routine has been developed to speed up the process of generating the ADAMS model of a variety of engine configurations. The model is driven by measured or predicted cylinder pressure [1] or measured flywheel speed. The model is used to predict the dynamic forces at the vehicle body side of each mount as shown in Figure 1a.

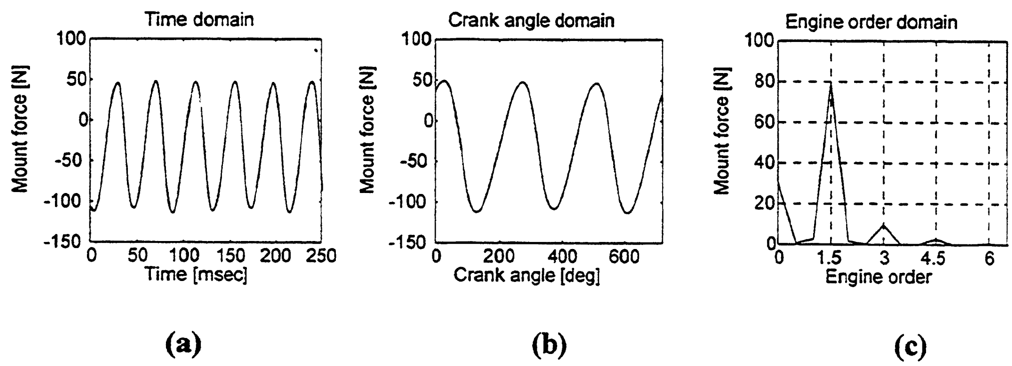

Figure 1: Left hand mount forces: a) original ADAMS output in time domain; b) Converted to Crank Angle Domain; c) Amplitudes in engine order domain

It is usual and more meaningful to plot noise and vibration data as engine order components (multiples of engine speed). Therefore, the forces at each mount, predicted by ADAMS, are exported to the mathematical analysis package 'MATLAB' using special routines and a Fourier transform is applied in order to plot the forces in order domain. Figure 1b shows the mount forces as a function of crank angle and Figure 1c shows the amplitude content of the vibration as a function of engine order. These data also contain the phase of each vibration order relative to the crankshaft rotation. It is therefore possible to re-calculate the force in the engine cycle domain for one order only.

MATLAB can also be used for advanced data presentation. For example, the main engine order mount forces in the fore/aft (X) and vertical (Z) directions can be plotted resulting in a two dimensional image of the force trace. A very clear presentation of all the mount forces can be achieved by creating a plot which shows the three orthogonal views of an engine bay, with scales representing the vehicle coordinates, and plotting the two dimensional force traces at the mount positions. An example of such a plot is given in Figure 8. This type of plot allows the effect of parameter changes on mount forces to be rapidly assessed.

Calculating Interior Noise and Vibration using Transfer Functions

Interior noise may be predicted by the application of noise transfer functions on the forces predicted by ADAMS at each mount position.

Transfer functions represent the relationship between the phase and amplitude of an input force signal to the phase and amplitude of an output signal across the frequency range. Transfer functions can be measured on an existing vehicle, or predicted using analysis packages such as NASTRAN. Figure 2a shows a typical noise transfer function.

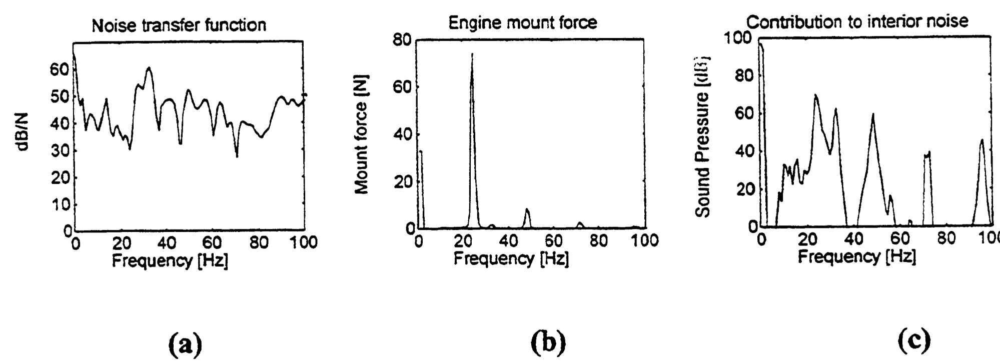

Figure 2: Calculation of the contribution to the interior noise level; a) Noise transfer function; b) Predicted force input; c) Resulting interior noise level

In order to apply a noise transfer function, the time domain ADAMS results are converted into frequency domain data with the same frequency range and resolution as the transfer function data. For example, the results of a Fourier transformation on a short time window of the forces shown in Figure 1a are shown in Figure 2b. The noise transfer function at each frequency is then multiplied by the predicted forces at the same frequency to provide an estimate of the contribution of each mount in each direction to the interior powertrain noise (Figure 2c). The interior noise level is equivalent to the sum of all these sources. As the multiplied and summed values are complex numbers, the phase and amplitude information is taken into account.

The predicted noise levels can be converted to time domain signals and played on loudspeakers for direct assessment. To give a realistic impression of the noise in a vehicle a sound recording of measured interior noise is edited. The noise generated by the first six engine orders is digitally removed and replaced by the predicted orders. This approach is valid if the changes assessed in the ADAMS model will primarily influence the lower frequencies and will have negligible effect on higher frequency engine harshness.

CASE HISTORY OF POWERTRAIN MOUNT OPTIMISATION FOR LOW FREQUENCY INTERIOR NOISE

The aim of this work was to assess the potential for modifications to the baseline mounting system to reduce the interior noise and vibration levels whilst at the same time maintaining the integrity of the mounting system for other design criteria.

Model Description

In order to predict the forces transmitted into the car body a functional ADAMS model of the engine was created using the Ricardo 'engine builder'. The model contains all the parts which contribute to the low frequency noise and vibration excitation, that is the combustion forces and the inertia forces of the reciprocating and rotating parts of the engine. Figure 3 shows an AVIEW picture of the model presented in this paper. The model represents a fully balanced 3 cylinder engine on a three point mounting system. It consists of pistons, conrods, crankshaft, balancer shaft, engine block, gearbox and the engine mounts. Two plain elastomeric mounts and one hydraulic mount were used in this installation. The hydraulic mount was modelled by converting its frequency dependent characteristic to a transfer function which can be implemented into ADAMS. To generate the unbalanced rotating couple, inherent to a three cylinder engine, the unbalanced

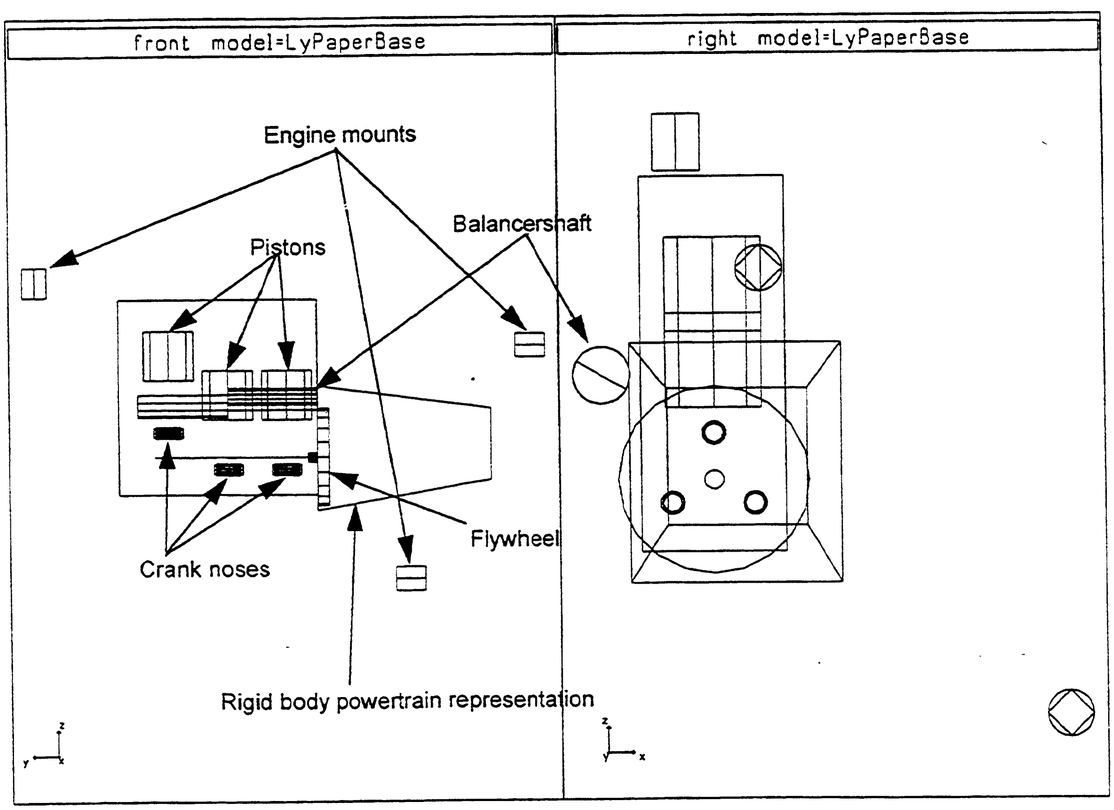

Figure 3 : ADAMS model of a 3 cylinder engine

crankshaft mass distribution was modelled in detail. To counteract this rotating couple, further masses were added to the crankshaft. The reciprocating couple is balanced 50% by additional masses on the crankshaft and 50% by a balancer shaft. Figure 4 shows the mass distribution on the crankshaft and the balancer shaft. By separately representing these masses, the balancing strategy can be changed and assessed.

The rotational speed of the engine was generated by applying a measured velocity history to the flywheel.

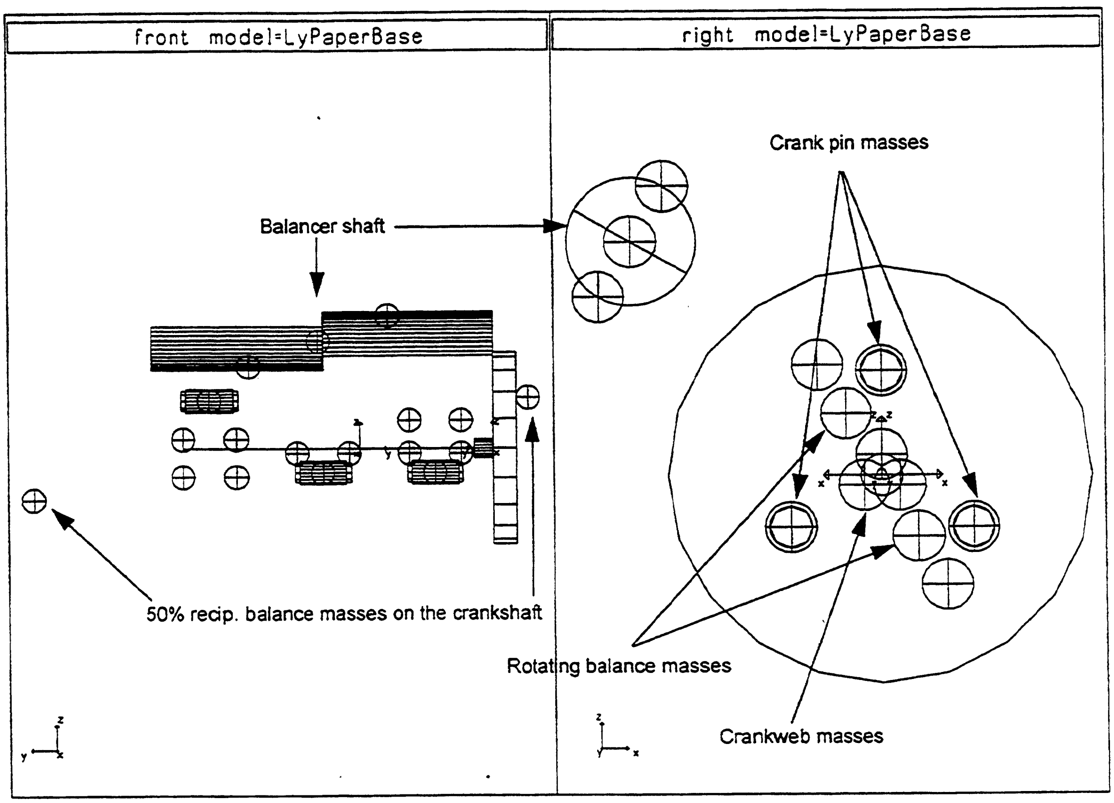

Figure 4: Mass distribution on the crankshaft

Transfer Function Measurement

The noise and the vibration transfer functions were measured on the car with the engine removed. The car body was excited at the mount positions by a hammer impact and the corresponding noise levels measured at the drivers left ear position. This test was carried out at each mount position in three orthogonal directions. Figure 2a shows the noise transfer function taken at the left hand mount in Z-direction. Similar vibration transferfunctions were measured with a triaxial accelerometer at the seat rail. These transfer functions were multiplied with the predicted forces from the ADAMS model.

Model validation

To validate the model and the approach, the simulation results were compared with measurements taken from a baseline vehicle fitted with the modelled engine. For simplicity the correlation was carried out at a fixed engine speed of 950 rev/min with the vehicle stationary.

To gain confidence in the ADAMS model, the predicted engine mount accelerations were verified. A direct correlation of the mount forces was not carried out, as force measurements are very complex and time consuming. However, the mount forces can be assumed proportional to the acceleration and the mount stiffness, as the motion of the powertrain at frequencies above its rigid body modes is inertia controlled. It can be seen in Figure 5 that a very good correlation was obtained for all engine orders.

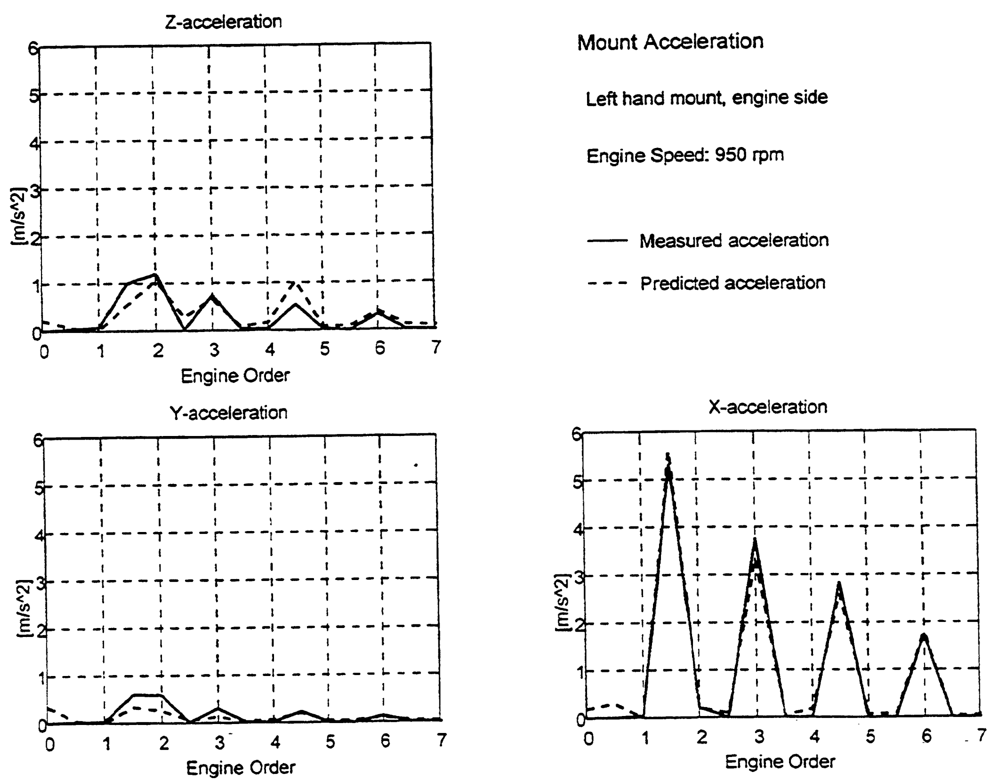

Figure 5: Comparison of the measured and predicted engine mount accelerations

The seat rail accelerations were calculated by multiplying the predicted mount forces with the measured vibration transfer functions. Figure 6 shows a comparison of these results with the measured acceleration levels at the seat rail. The vibration levels at the 1.5th, 3rd and 4.5th engine order have been predicted precisely. The 0.5th order vibrations occurring on the test vehicle are due to combustion irregularities which were not predicted by the model. The effect of a remaining engine unbalance due to production tolerances are evident in the high 1st order vibration levels of the real engine compared to the fully balanced engine model.

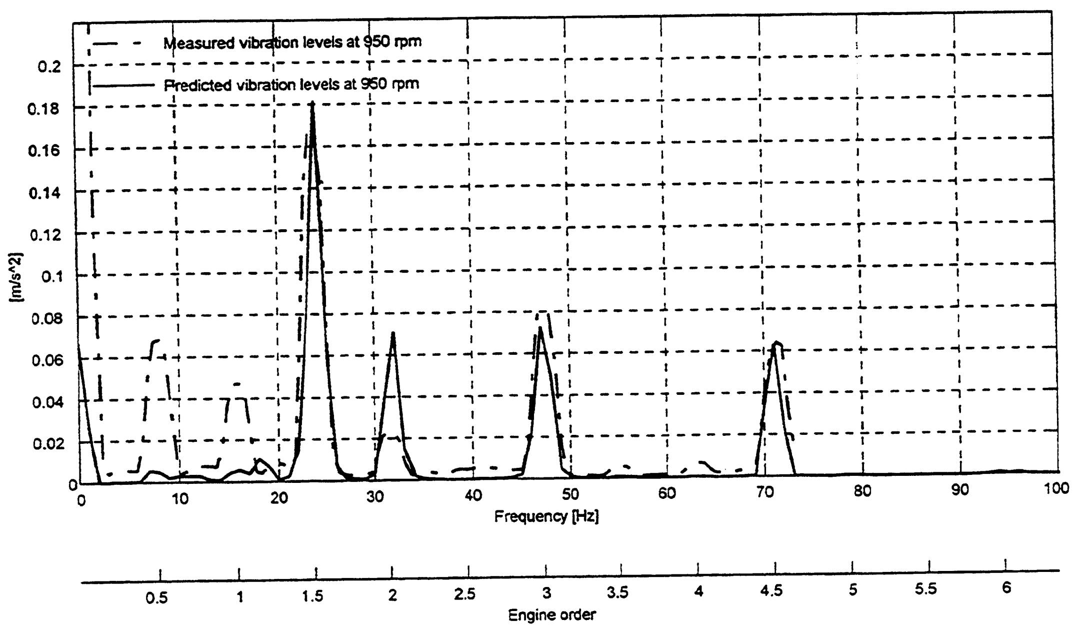

Figure 6: Comparison of measured and predicted seat rail vibration levels

The contribution of the engine mount forces to the interior noise level was calculated using the methods introduced above. A direct comparison of the predicted baseline noise levels with the measured noise levels can be found Figure 7. Unlike the seat rail vibrations the interior noise is also influenced by additional sources such as the intake system (contribution to 1.5th, 3rd and a 4th order resonance), exhaust system, alternator unbalance (2.5th order), vacuum pump and other sources. These introduce additional noise peaks which sum with the engine mount related noise. The noise levels predicted here would therefore be expected to be less than the measured noise levels. The correlation is generally good. The most significant differences between the measured and predicted sound pressure levels are at 1st order, which is underpredicted again due to a remaining engine unbalance, and at 2nd order, most likely due to a discrepancy between the modelled and the real vertical mount stiffness.

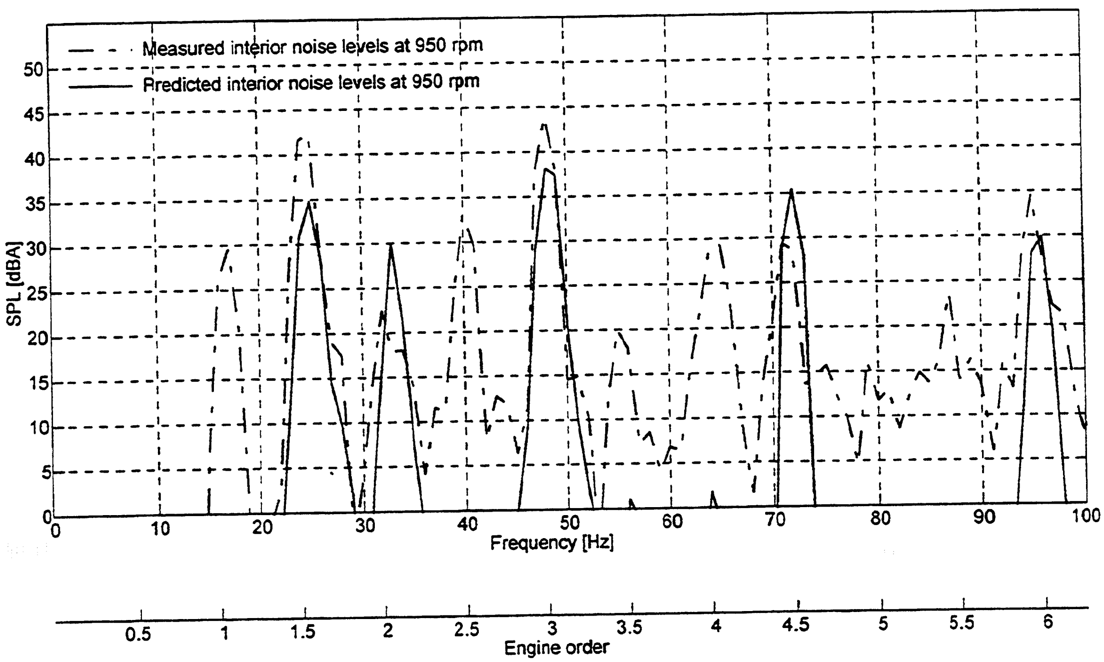

Figure 7: Comparison of the predicted and the measured interior noise levels

Improved Powertrain Suspension System

It was shown that an improved three point mounting system could be obtained by lowering the fore/aft (X) and the vertical (Z) stiffness rates of the right hand and the left hand mount. Figure 8 shows a 65% reduction in the (X) forces and a 57% reduction in the (Z) forces from the original mounting system (dashed lines) to the improved mounting system (solid lines). The new forces were multiplied by the measured force and noise transfer functions to predict the seat rail vibration and interior noise at idle. Figure 9 shows that the seat rail vibration could be reduced by 38% for 1.5th order, by 71% for 2nd order and by 28% for 3rd order. Figure 10 shows that the interior noise levels could be reduced by 6 dBA for the 1.5th order, 5 dBA for the 3rd order and 2 dBA for the 4.5th order.

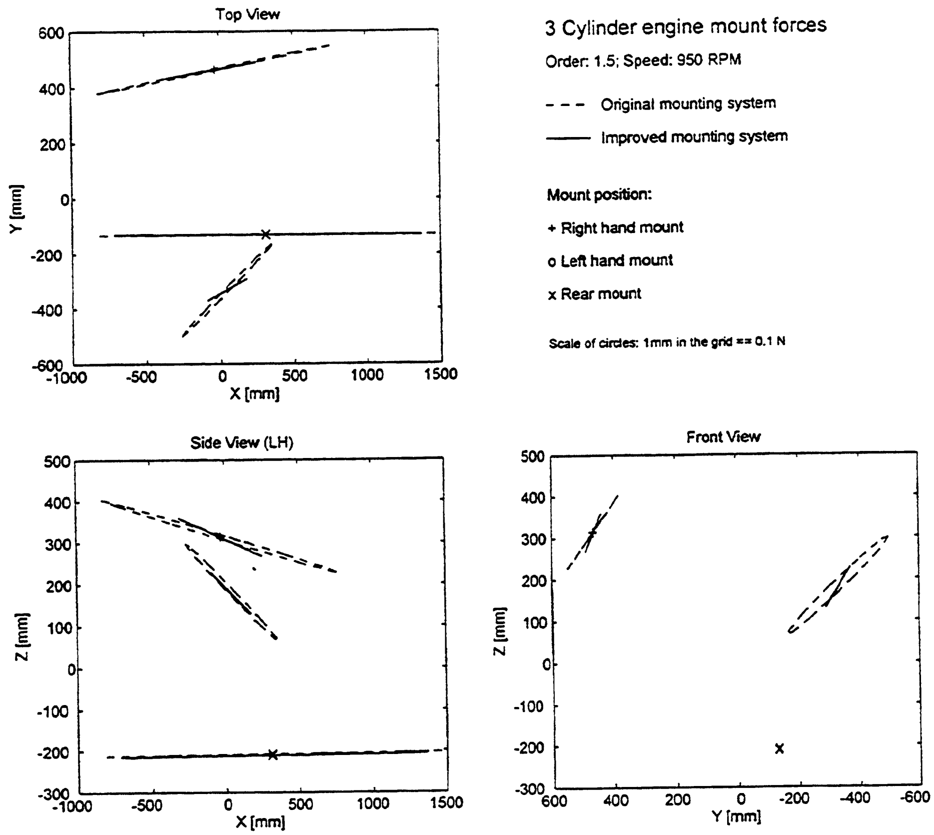

Figure 8: Mount force representation in the vehicle grid

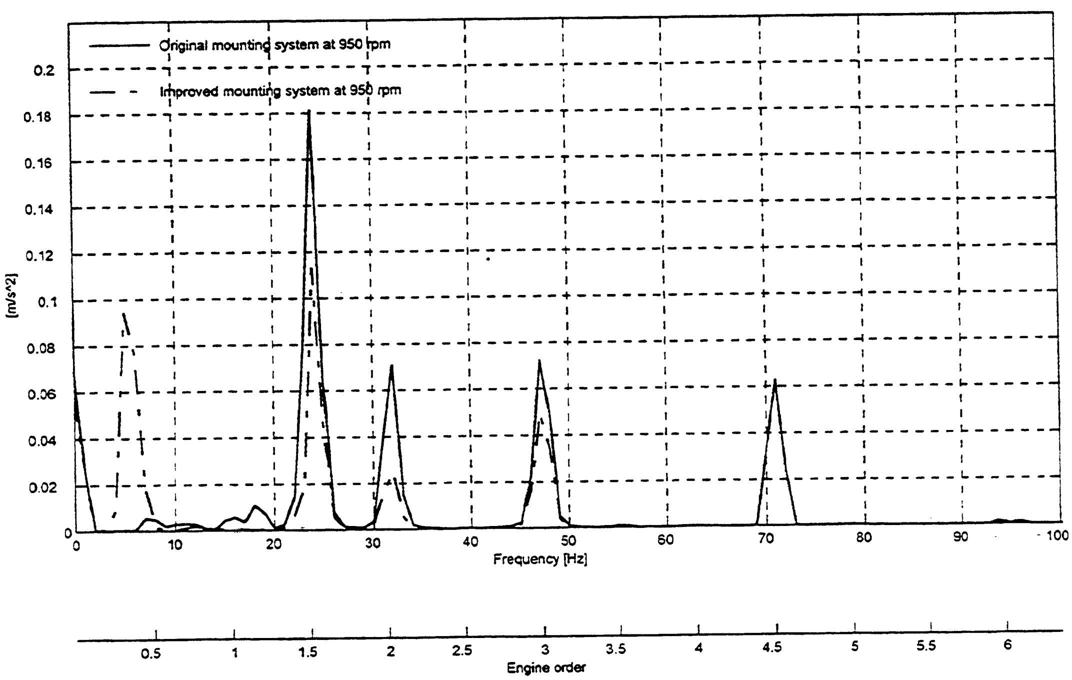

Figure 9: Predicted seat rail vibration reduction obtained by the modified mounting system.

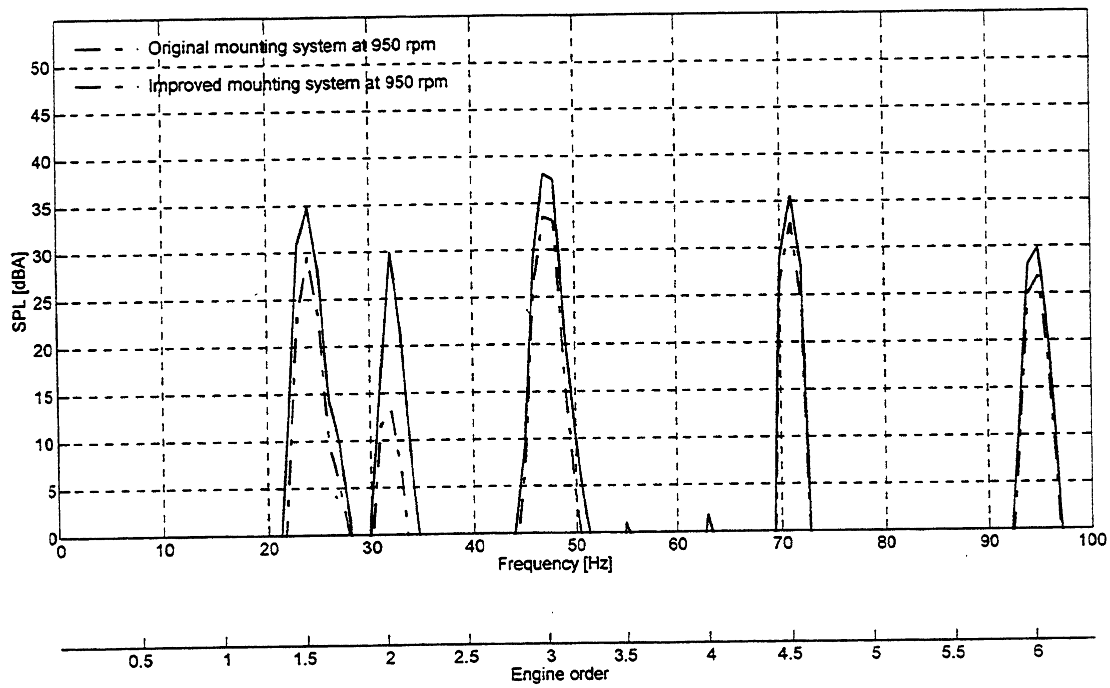

Figure 10: Predicted interior noise reduction obtained by the modified mounting system

CONCLUSIONS

Multibody dynamic simulation tools are an effective method for the optimisation of powertrain mounting systems. The application of transfer functions, to the results of a multibody dynamic simulation, is an valuable method for predicting vehicle interior noise and vibration.

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