# A PRELIMINARY STUDY OF NOISE REDUCTION IN A PASSENGER CAR THROUGH OPTIMIZATION OF THE ENGINE SUSPENSION

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## ABSTRACT

An investigative method based on dynamic analysis was used to optimize engine-body interaction, especially in terms of noise abatement. The body interacting with the engine and elastic supports was examined through modal analysis and experimental testing. The engine unit (engine plus gearing) behaves like two virtually stiff bodies joined by an elastic connection at the clutch. Natural frequencies of the body were found to be above the engine's rigid body modes. The dynamic deformations of the engine were reconstructed on the basis of the operating test results, making it possible to determine the optimal support points. Through analysis of the elastic support-body frequency response functions, variations in the vehicle structure could then be simulated that decreased vibration transmissibility and thus noise inside the vehicle.

## NOMENCLATURE

| | | |
|---|---|---|
| $a_i$ | = | acceleration measured at point i |
| DIR1 | = | direction of axis X |
| DIR2 | = | direction of axis Y |
| DIR3 | = | direction of axis Z |
| $F_i$ | = | force measured at point i |
| $H_{ij}$ | = | frequency response function between points i and j |
| $K_m$ | = | elastic support stiffness |
| $\ddot{X}_i$ | = | acceleration at point i |
| $X_i$ | = | displacement of point i |

## 1. INTRODUCTION

Noise, one of the major factors that influence the passengers' perception of automobile comfort, is the sum of several components, each of which traceable to a specific source. The main kinds of noise are:

1. Aerodynamic noise, which increases with speed.
2. Noise stemming from vehicle-road interaction, which is due to:
   - the deformation of the tires during their rolling movement
   - vibrations transmitted to the body via the suspension units.
3. Noise from the engine unit, which is either transmitted directly through the air to the cabin or generated in the cabin by the vibrations of the body surfaces excited by the engine when it is running.

This work describes an investigation undertaken to analyze engine-generated noise. An experimental dynamic analysis was conducted on an italian vehicle mounting a new-model 3000-cc V-6 engine with a four-valve-per-cylinder distribution system.

Any vibrating surface coupled to an elastic fluid induce in it sound pressure waves that are felt by the passengers of a road vehicle cabin like noise. Clearly, under resonance conditions of vobrating sirfaces, sound pressure level reaches its maximum. Consequently, the investigation was carried out in three stages:

1. Analysis of the engine's dynamic behavior during operation and of the forces transmitted to the body.
2. Modal analysis of the engine.
3. Analysis of the frame, especially the fore section where the driveline is located.

In all three stages our objective was to determine the points of limited movement, both of the engine in operation and of the body. If it were then possible to find coincident nonmoving points on the engine and the body, vibration transmission could be eliminated simply by connecting the elements trought theese points. This procedure would hold for any stiff, elastic flexible, or visco-elastic flexible connection, thereby abolishing the variable connected to the elastic support characteristics.

## 2. CALCULATION OF THE DYNAMIC DEFORMATIONS AND FORCE LEVELS

Acceleration measurements were made at operating conditions to 1) verify the stiff body behaviour of the engine and 2) to reconstruct the forces transmitted by the elastic supports. The latter information was especially important in determining the amount of engine excitation exerted on the body (Figure 1). The engine accelerations

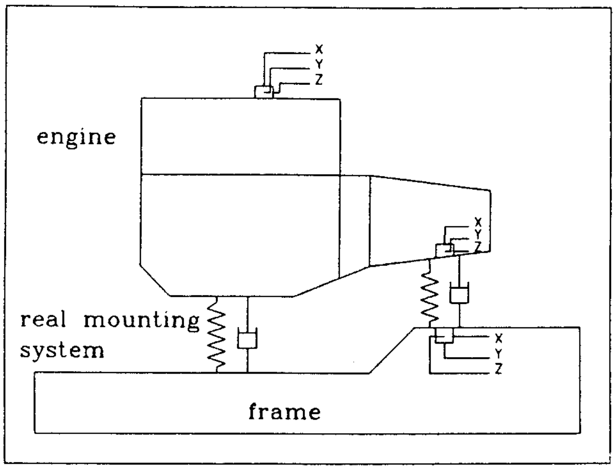



Fig 1: engine instrumentation layout during operational test

were measured on the engine unit of a complete car mounted on a roller test bench, using triaxial accelerometers placed on each data acquisition point during a reference maneuver (gradual increase in speed in third gear that would reproduce near-static conditions). The acquired accelerations were then plotted on waterfall diagrams (Figure 2). The deformed shapes were

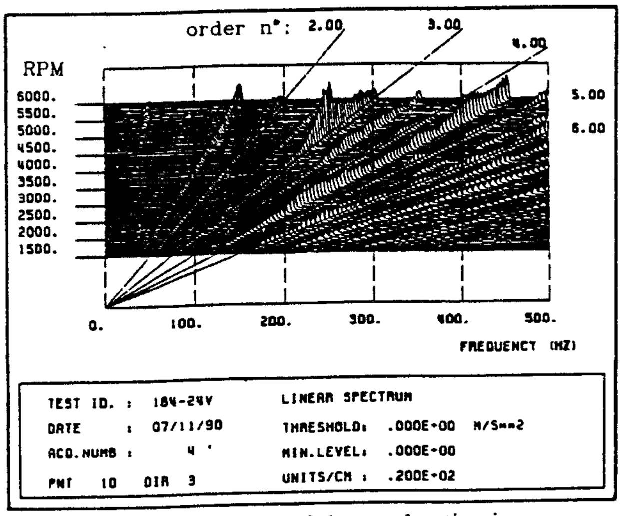



Fig 2: waterfall diagram of the acceleration in measurement point n° 10 in vertical direction

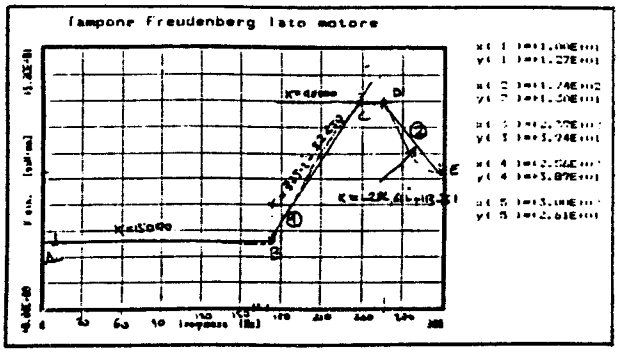
reconstructed using the speeds (1500, 3000, 4500, 5000, 5500, 5800, and 6000 rpm; Figure 3) corresponding to the

| point | order number | | | | | |
|---|---|---|---|---|---|---|
| n | 2.0 | 2.5 | 3.0 | 3.5 | 4.0 | 4.5 | 5.0 |
| 1 | / | / | 5000<br>5300 | 5800Z | / | 5800Z<br>5000 | 4500Y<br>5800Y<br>6000Z |
| 2 | 6000 | / | 5800<br>6000 | 5800Z | / | 5000<br>6000Y | 5800<br>6000 |
| 3 | 6000 | 6000Z | 5000<br>6000 | 5800Z | / | 6000Y | 5800<br>6000 |
| 4 | 6000 | / | 5000 | 5800 | / | 4800Z<br>6000X | 5800<br>6000 |
| 5 | 6000 | / | 5500 | / | / | 4500 | 5000 |
| 6 | 6000 | / | 5500 | / | / | 4500<br>5500 | 5700 |
| 7 | 6000 | / | / | 5800 | / | 4800<br>5500 | 4300<br>5500 |
| 8 | 5000 | / | 5400 | 5000 | / | 4500<br>5500 | 5800 |
| 9 | / | / | 5000<br>5800 | / | / | 4500<br>5500 | 5800 |
| 10 | 6000 | 5500 | 5000 | / | / | 3000<br>4500<br>5500 | / |
| 19 | 6000 | / | / | / | / | / | / |
| 20 | 6000 | / | 5800 | 4800<br>5800 | / | 4800 | 5800 |
| 21 | 6000 | / | 5000<br>5800 | 5000<br>5500 | / | 5500 | 5800 |
| 22 | 6000 | / | / | / | / | / | / |
| 23 | 6000 | / | 5000<br>5500<br>5800 | 4000 | / | 4000 | / |
| 24 | 6000 | / | 5500 | 5300<br>5800 | / | / | / |
| 25 | 6000 | / | 5700 | 5300 | / | 4000<br>5500 | 5500 |

Fig 3: table used to select the main RPM values

maximum accelerations on the diagrams.

Using the accelerometer measurements on the engine- and body-side elastic supports and using a piece wise linearization of their dynamic stiffness (Figures 4 and 5).




Preload: 890 daN Amplitude: +/− 0.1 mm

Sweep Time: 160"

Fig 4: piecewise linearization of the dynamic stiffness of the engine side elastic support

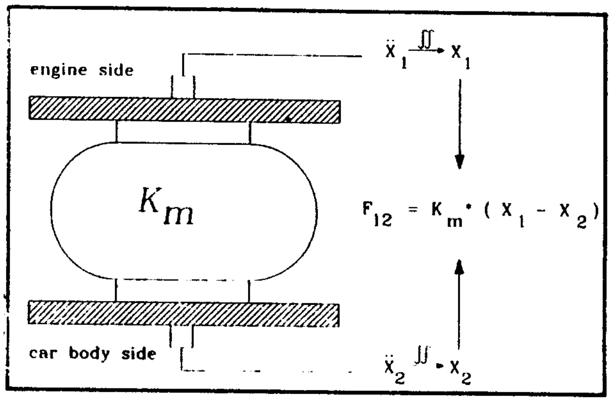



Fig 5: calculation of the force transmitted trought the elestic support

we found the main orders of excitation transmitted to the body to be 1.5, 2, and 3 (Figure 6), with the latter two increasing with speed. In thge following we will focuys our attention on the research of the optimal position of the engine elastic support, observing that the mean level of excitation in the fore elastic support is much less than in the others.

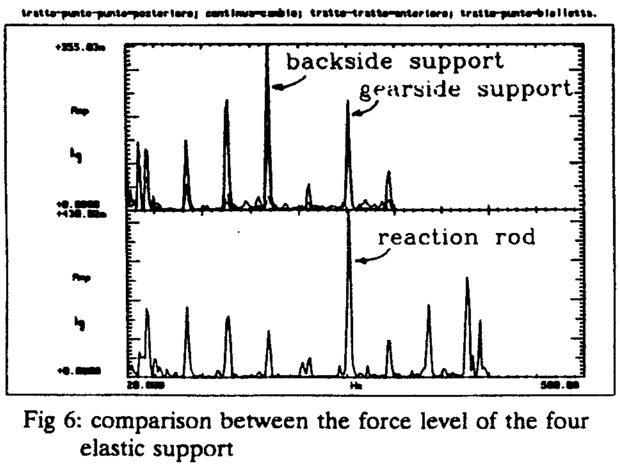



Fig 6: comparison between the force level of the four elastic support

## 3. MODAL ANALYSIS

*Engine*

In the second part of our investigation, a modal analysis of the engine was performed 1) to verify the results of the operational analysis, 2) to characterize the dynamic behavior of the accessories and the engine support brackets, and 3) to evaluate the frequencies of the rigid body modes.

To perform the modal analysis of the engine unit it has been mounted on a structure that has been designed and made as steef as possible to isolate any deformation of the frame from the engine's dynamic behavior. The excitation source was an electromagnetic shaker positioned on the end of the gearshift housing (Figure 7). The 125

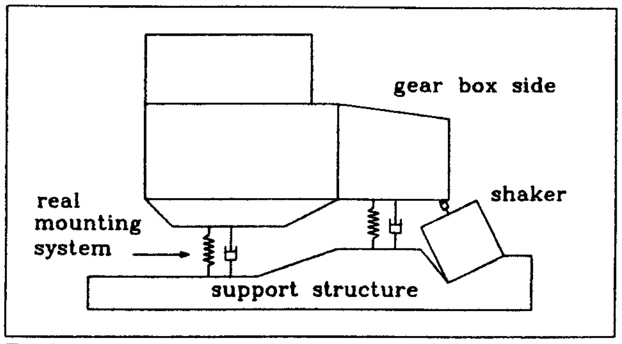



Fig 7: scheme of excitation system used for engine modal testing

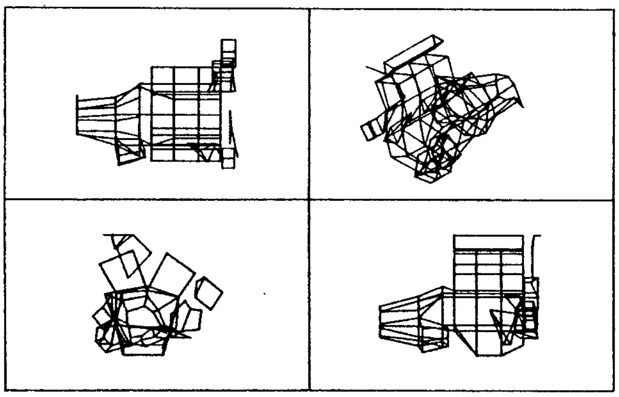


Fig 8: engine discretization

acquisition points were spread over the engine and accessories (Figure 8). The acquired data served to calculate:

- The engine's rigid-body mode shapes appeared between 0 and 50 Hz. The rigid-body frequencies were found to be well below the excitation frequencies of the engine during operation.
- The vibrating structure's natural frequencies (Figure 9). The mode shapes analyzed between 50 and 650 Hz, showing that 1) the engine deforms mostly according to bending movement and 2) revealing a yield zone in the clutch housing. As a result, the engine and gearshift could be considered as stiff bodies.

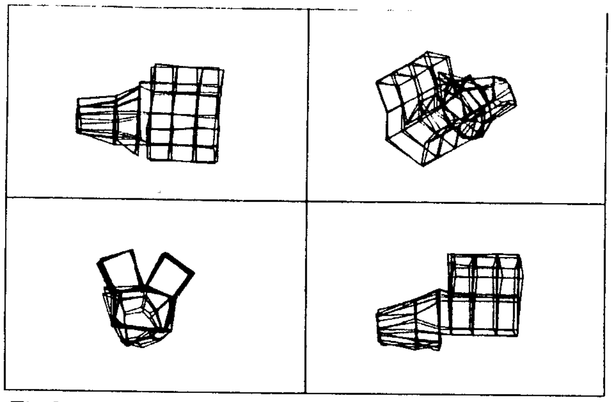


Fig 9: the main deformation mode of the engine

A local modal analysis carried out on the main accessories and the engine brackets between 50 and 650 Hz showed that:

- There are many resonance frequencies of the main accessories (Figure 10) but given the notable difference in mass between the accessories and the engine, these vibrations have no effect on cabin noise transmission.
- The rear and gearshift-side brackets showed a high number of local modes.

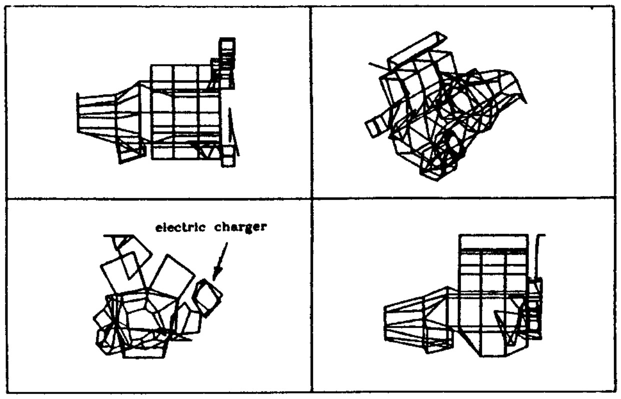



Fig 10: the electric charger mode

*Body*

The body was analyzed with all the irrelevant parts removed. The support system was designed to render the vehicle free in space, i.e., a constraint with a stiffness way below the system's, so that we could consider the natural frequencies of the system and its support structure decoupled. The system was implemented by four air springs connected via a simple pneumatic loop (Figure 11).

Testing was conducted by applying simple and multiple excitations. The modal analysis of the body was carried out between 0-100 Hz, since the high modal density made it impossible to go any higher. The 53 acceleration measuring points provided an optimal amount of data to ensure proper evaluation of the global modes. The acceleration-force FRFs were recorded during acquisition. The structural influence of the engine support frame on

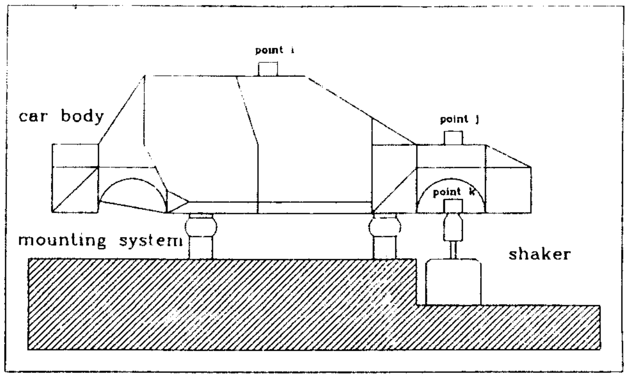



Fig 11: suspension and excitation system of the body used to perform body modal testing

the vehicle with and without it was examined. After extraction of the modal parameters, it could be seen that the structure's first natural frequency (torsional) (Figure 12) was not influenced by the presence of the frame,

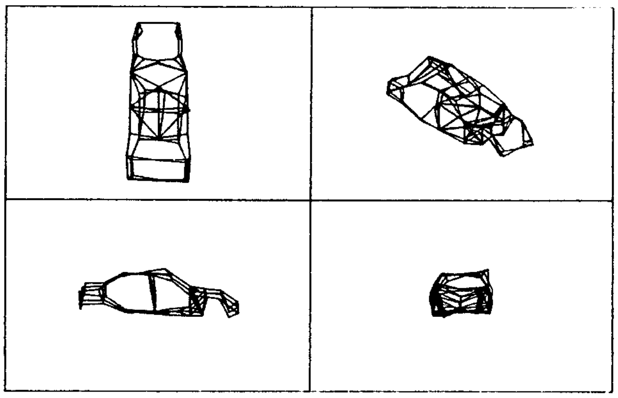


Fig 12: first modal shape of the body

whereas the second and successive ones (bending) were so greatly affected by the frame that some of the structure's natural frequencies found when there was no frame disappeared in the presence of the frame. (However, the frame could not be modified, since such modification would entail modification of the whole body.)

The fore section, including the support frame, was then analyzed in order to site the elastic supports at the nodes where vibration transmission was minimal. We evaluated the acceleration-force FRF between the excitation point and several points on the frame, which had been discretized into 47 points. Using extraction of the modal parameters, we again reconstructed the frame's mode shapes. Since, most of the nodes were found in proximity to the body-frame connection (with the exception of a few in the middle of the fore beam), the best place to connect the elastic supports to the frame turned out to be by the frame-body connection.

## 4. TRANSFER SYNTHESIS METHOD

Assuming that the system (body plus frame) is linear, we can see that an acceleration-force transfer function, $H_{ij}$, between two generic points of the system, can be calculated when the transfer functions between points k and i, $H_{ki}$, k and j, $H_{kj}$, and the point transfer function at point k, $H_{kk}$, are known. The relation linking these magnitudes can be deduced from the following simple considerations (Figure 11):

$$H_{ij} = a_j/F_i$$

$H_{ki} = a_i/F_k$ and reciprocally, $a_k/F_i = H_{ik}$, yielding

$$F_i = a_k/H_{ki}$$

$$a_j = H_{kj} * F_k$$

Substituting, and recalling that $H_{kk} = a_k/F_k$, we get

$$H_{ij} = H_{kj} * H_{ki}/H_{kk}$$

On this basis, we constructed the transfer functions between several points on the frame and several inside the cabin (quarter panels, roof, floor), exciting the structure at a single point on the frame (i.e., the same point used for the modal analysis).

The synthesis process, while theoretically viable, is actually rather difficult to implement. Since problems arise if the FRFs are processed in narrow band during the multiplication of records in the resonance and antiresonance frequencies, we used third of octave band. The experimental test results were in excellent agreement with the frame modal analysis results. Also, since displacement of the body-frame connection point of only a few centimeters produced excitations differing considerably from the excitation levels of the quarter panels, roof, and floor, this method resulted more accurate than modal analysis in siting the elastic supports.

## 5. CONCLUSIONS

The following emerged from our investigation:

1. The engine and gearshift behave as stiff bodies with localized bending by the clutch housing, and hence the most useful points for connection to the body are the middle ones.
2. Numerous natural frequencies appear for the accessory support brackets that, however, do not affect vibration transmission.
3. The engine support brackets, especially the rear and the gear ones, show several natural frequencies, and thus must undergo closer scrutiny.
4. Even slight movements of the elastic support-frame connection points reduce the excitation level inside the cabin.

## 6. ACKNOWLEDGMENTS

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