OPTIMIZATION OF AN ENGINE MOUNTING SYSTEM USING MODAL ANALYSIS TECHNIQUES

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Abstract

The following work confronts the problem of noise reduction inside the driving compartment of an automobile. It refers to that part of the noise generated by the vibration of the driving compartment walls due to the forces that the engine transmits to the body whilst running, via its mountings and its suspension frame. Signal analysis recorded in working conditions and experimental modal analysis are applied. Furthermore, reference is made to a numeric technique for the synthesis of force-acceleration transfer.

NOMENCLATURE

H_{ij} = transfer function between point i and point j

a_i = acceleration measured at point i

F_i = force measured at point i

FRF = transfer function

DIR1 = direction of X axis

DIR2 = direction of Y axis

DIR3 = direction of Z axis

K_m = elastic support stiffness

X_i = point i acceleration

X_i = point i displacement

1. Introduction

One of the factors that influence the comfort characteristic of an automobile is the noise noted by its occupants. One can say that the composition of such noise is the sum of various elementary disturbances, each of which is traceable to a precise source. The most important of these can be subdivided in the following way:
1) aerodynamic noise: the importance of which increases with speed;
2) noise due to road-vehicle interaction:
  a) noise due to the rolling of the tyres on the road
  b) noise due to the excitation that the roughness of the terrain transmits to the body via the suspension;
3) noise from the power unit:
  a)noise transmitted aerially by the engine and the exhaust to the driving compartment
  b)noise generated within the driving compartment by the vibration of the body surfaces during the running of the engine.

During the research attention has been centered on the last of these cases.

To reduce the level of such noise it is necessary to know the vibration characteristics transmitted by the engine to the inside of the vehicle. For this reason it was decided to study the behaviour of the engine, the anterior part of the body, the engine support frame and the influence of the engine mounting elastic supports' position. The basic principal of the work was the following: to determine the points that present limited movements, either on the body or on the power unit whilst running. The choice of this approach derives from the fact that if certain "stationary" points existed on the engine and on the body, and if these points were made to coincide by connecting the two elements via such points, we would not have any vibration transmission. This condition would be true for any type of connection used, (rigid, flexible elastic, flexible viscous-elastic) eliminating in this way the variable associated with the silent block characteristic.

In accordance with this consideration, the study was divided into the following parts:
- behaviour analysis of the power unit in operating conditions and the forces transmitted to the body
- modal analysis of the power unit
- analysis of the body with particular attention to the anterir part where the power unit is mounted.

The research was conducted using experimental dynamic analysis techniques, on the Alfa Romeo 164 automobile, on which a new version of the 3000 cc V6 with a distribution system of 4 valves per cylinder will be mounted.

2. Strain and dynamic deformation

The aim of the first phase of the work is to characterize the dynamic behaviour of the power unit in operating conditions and to calculate the value of the forces that the engine transmits to the body. During a reference manoeuver (slow acceleration in third gear in such a way as to reproduce successive almost static conditions) significant in the process of measuring the forces transmitted to the body, accelerations were measured with tri-axial accelerometers mounted on each discreet point of the group (fig. 1).

The reconstruction of the dynamic deformation is per number of engine revolutions, selecting the more interesting from the waterfall diagrams obtained during the measurements. In particular the following were chosen: 1500, 3000,4500, 5000, 5500, 5800 and 6000

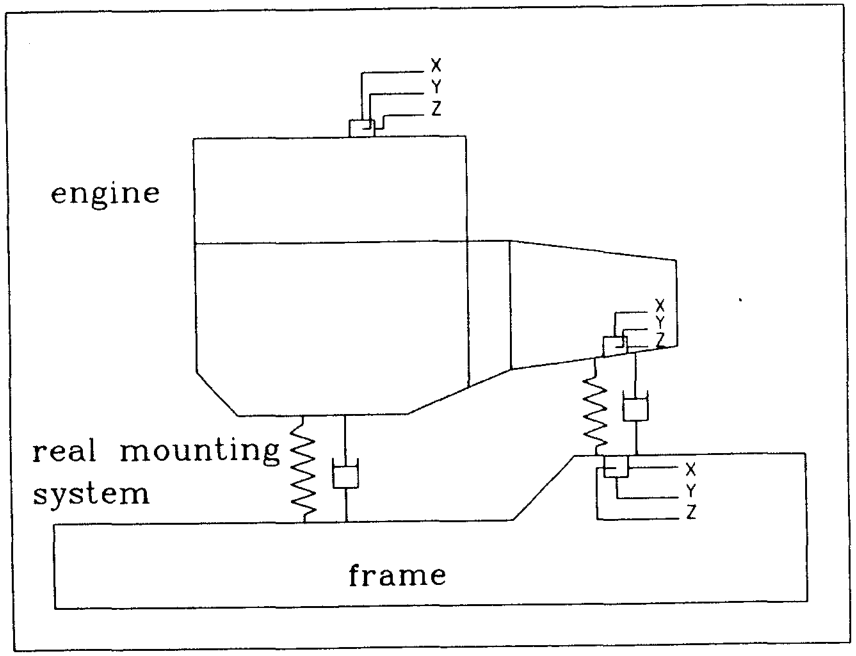

Fig. 1: engine instrumentation layout during operational test

RPM. It emerged that the unit can be considered as rigid up to a certain frequency value beyond which only the engine and the gearshift separately can be considered as such. Using the values of acceleration measured on the engine side and body side elastic supports, synthesizing the elastic support stiffness (fig. 2-3), via a double integration, it was established that the principal order of excitation that is transmitted to the body is that of the first and 1/2, the second and the third and that by increasing the number of revolutions, the last two become increasingly important (fig. 4-5). It was also observed that the average level of the force transmitted by the anterior elastic support was notably lower than that of the others. This result is very important as in the following phase it permits the concentration of attention on the engine mounting brackets.

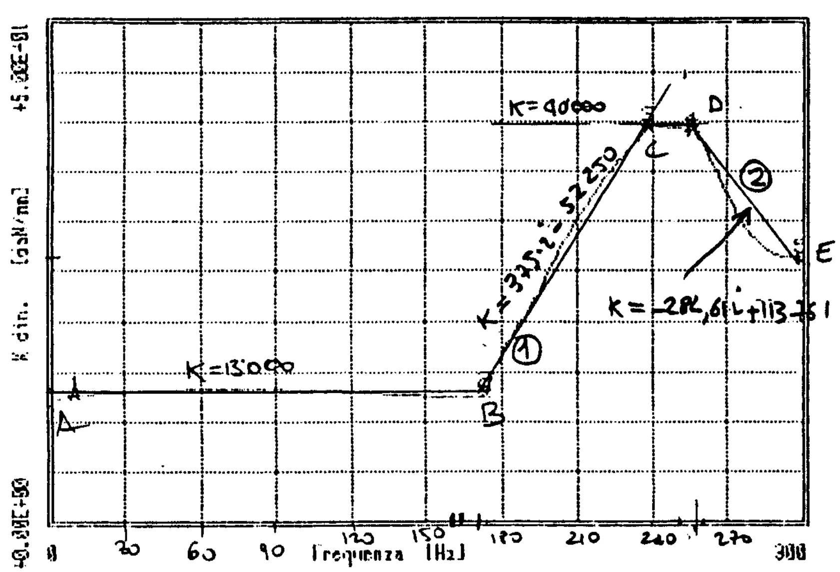

Fig. 2: piecewise linearization of the dynamic stiffness of the engine side support

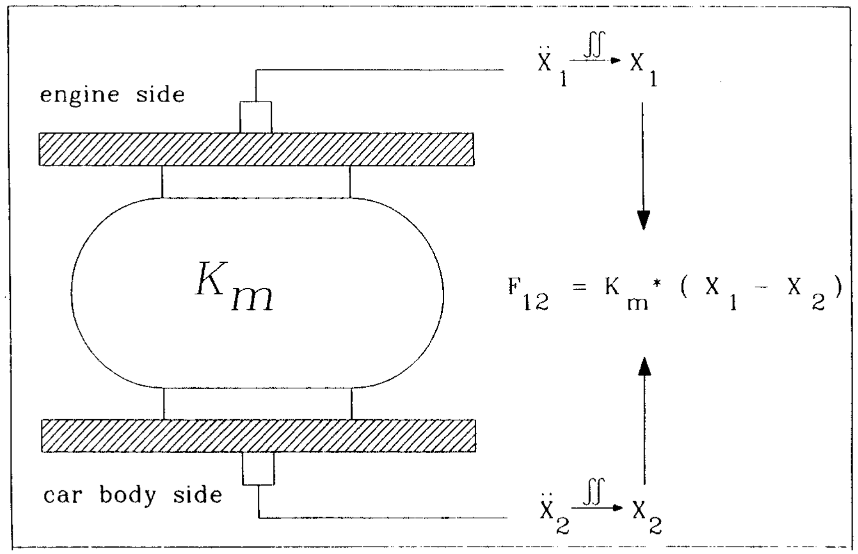

Fig. 3: calculation of the force transmitted throught the elastic support

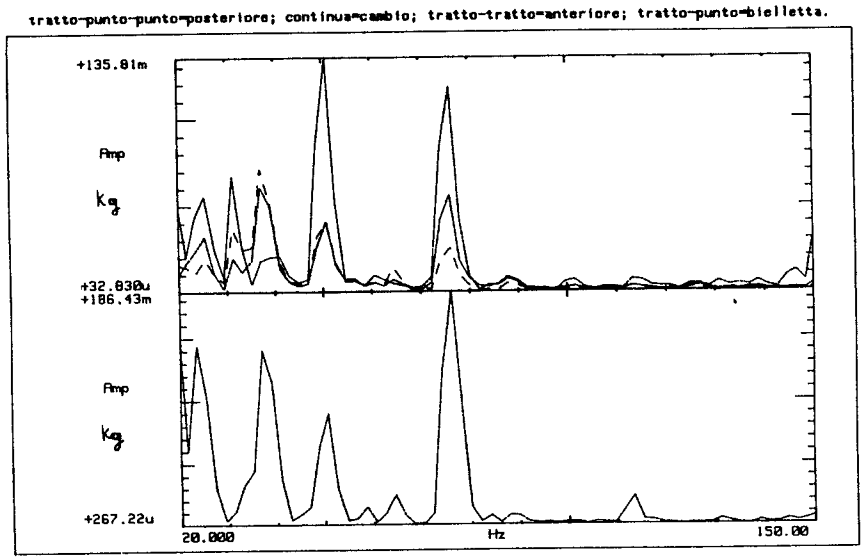

Fig. 4: force level comparison between the four elastic supports at 1500 RPM

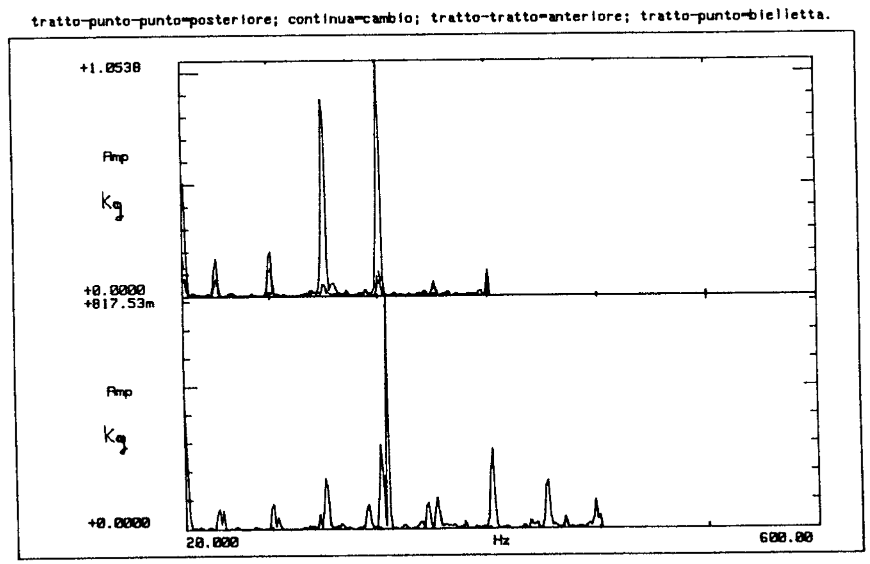

Fig. 5: force level comparison between the four elastic supports at 6000 RPM

3. Modal analysis of the power unit

The work's second part consists of the modal analysis of the power unit, with the aim of checking the results that emerged from the analysis in operating conditions and to characterize the dynamic behaviour of the accessories and the engine mounting brackets, notwithstanding the evaluation of rigid motion frequency. To execute the modal analysis, a power unit support plate was designed and made to which the unit and its frame were mounted. To conduct the engine modal analysis a simple excitation was used via an electromagnetic shaker, which after a series of preliminary tests was positioned on the terminal section of the transmission housing (fig. 6). 125 acquisition points were used covering the whole power unit group and accessories. Evaluating the data acquired, the following was calculated:

- the rigid motion of the power unit, verifying that the frequency was less than the excitation of the engine whilst running
- real vibration motion of the structure (fig. 7) (via a series of acquisitions between 50 and 650 Hz), that deform principally according to a bending motion denoting a yield zone of the clutch bell housing.

It was also established that the engine and gearshift can be considered as rigid bodies, thus confirming the analysis in operating conditions. Local modal analysis was then carried out on the engine mounting brackets and on the principal accessories, still in the 50 to 650 Hz frequency range, that made evident the following:

- there are various frequencies at which the accessories suffer deformations due to the engine mountings (fig. 8). Given the difference in mass between the accessories and the engine, such vibrations do not influence the transmission of noise to the driving compartment
- above all, the rear and transmission side mountings show a high level of real frequencies, confirming that which emerged from the tests in operating conditions.

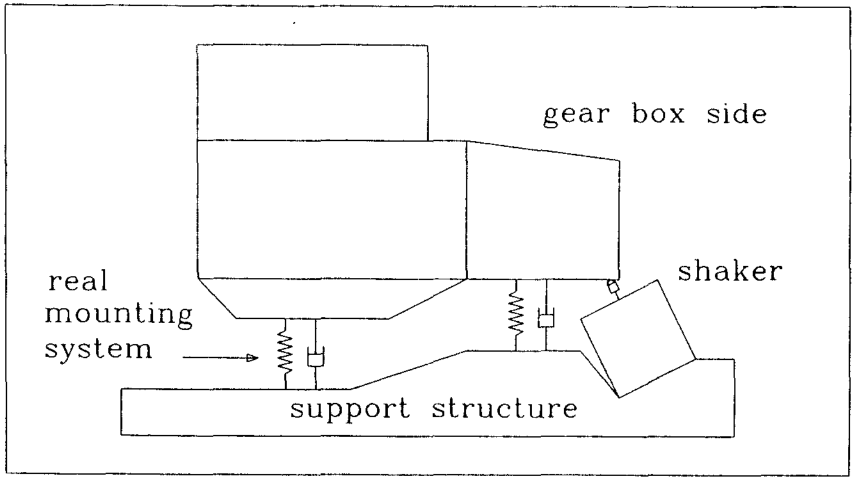

Fig. 6: scheme of excitation system used for engine modal testing

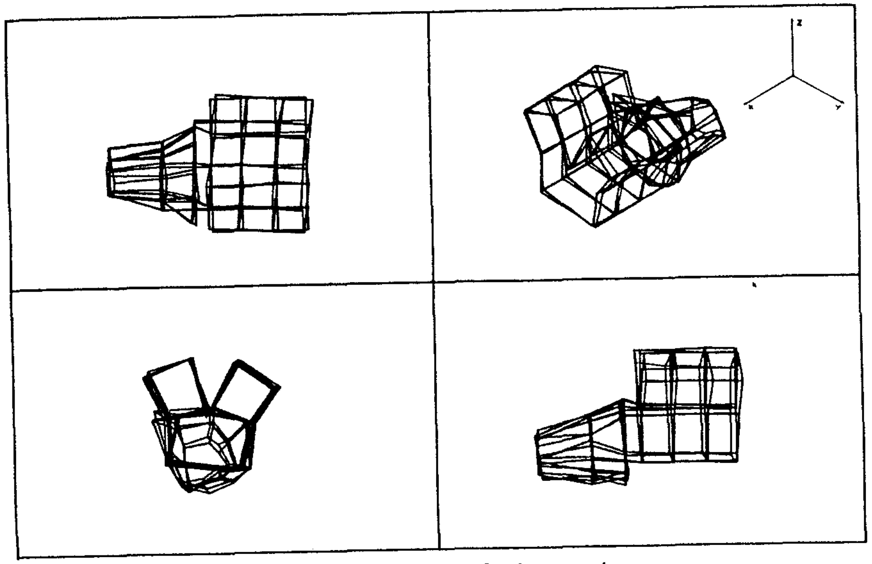

Fig. 7: the main deformation mode of the engine

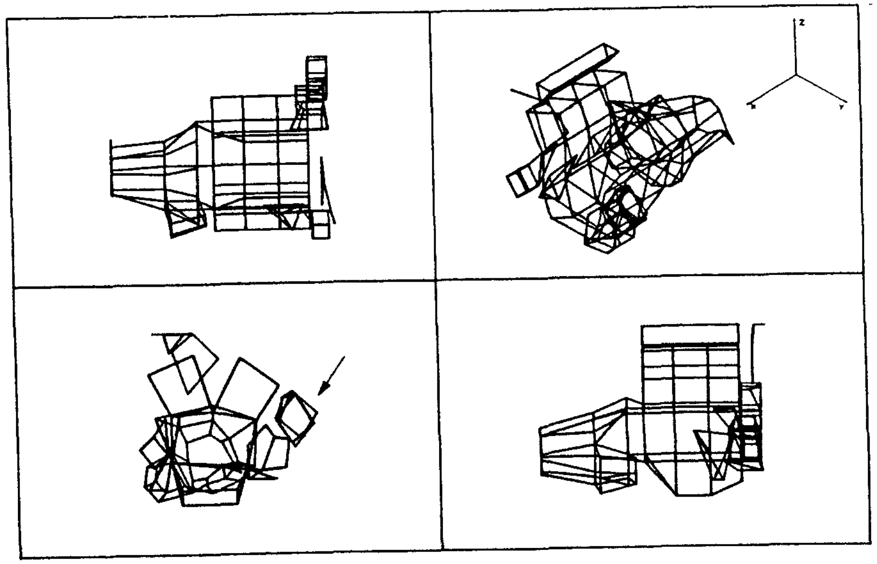

Fig. 8: the electric charger mode

4. Body modal analysis

The third part of the work consists of the modal analysis of the body to characterize the behaviour of the body and the anterior part of the structure. A method of support was manufactured that rendered the vehicle free in space, or precisely with a constraint having a rigidity much lower in respect to that of the system, such as to be able to consider the real frequencies of the system being studied as not linked to those of the support structure. A system consisting of four air dampers connected together in a simple pneumatic circuit (fig. 9) was designed.

Tests were conducted with simple and multiple excitation upon the vehicle. Modal analysis of the whole body was executed between 0 and 100 Hz. Accelerations in 53 positions in three orthogonal directions were taken. This choice allowed for the acquisition of not too many points but at the same time guaranteeing an optimum evaluation of global motions. The test was carried out on the vehicle equipped both with and without the power unit support frame, so as to establish the structural influence of the said element. By extracting the modal parameters one establishes that the first real motion of the structure is torsional whilst the second and also the following ones are bending (fig. 10). The analysis then moved to the anterior zone and in particular the support frame, with the aim of finding the most stationary points, or rather the knots that at the various frequencies could be used advantageously to position the silent blocks, such that excitation of the structure at said points would result in a reduced level of vibration transmission. The support was measured in 47 points. It was noted that for almost all the real frequencies found, the knots appear in the proximity of the attachments of the support frame to the body. As a result it is therefore convenient to connect the silent blocks to the support frame in areas near to the support frame-body anchors.

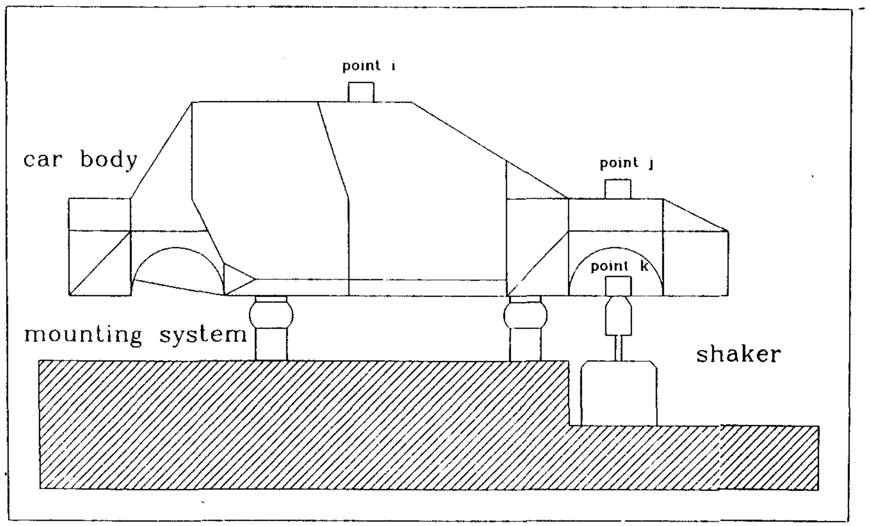

Fig. 9: suspension and excitation system of the body used to perform body modal testing

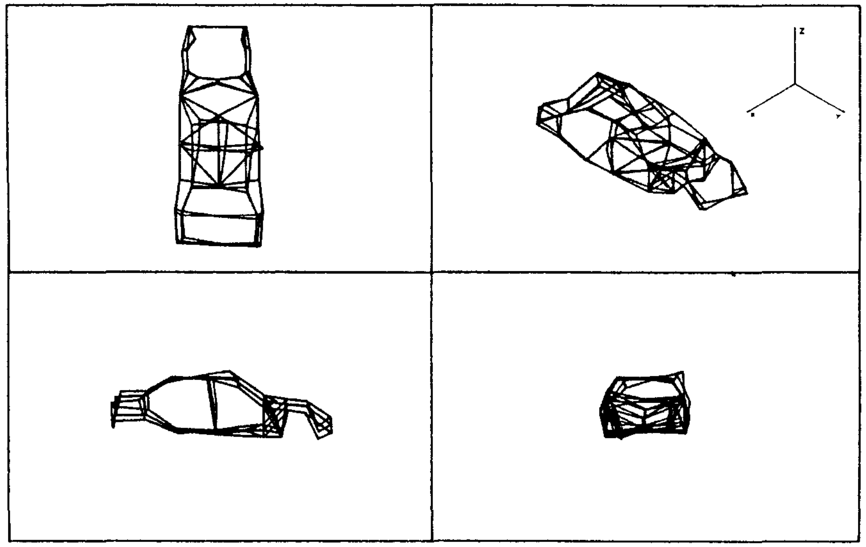

Fig.10: first modal shape of the body

5. Transfer synthesis

In the hypothesis that the system under examination is linear (body plus support frame), one can state that a transfer function H_{ij} (in terms of acceleration-force) between two general points of the system can be calculated by knowing the transfer between K and the respective points (H_{ki}, H_{kj}) and the punctual transfer at the point K (H_{kk}). The relationship that links such sizes is deductable from the following elementary considerations (fig. 9):

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

$$H_{ki} = a_i / F_k = \text{for the reciprocal} =$$

$$= a_k / F_i = H_{ik} \text{ from which } F_i = a_k / H_{ki}$$

$$H_{kj} = a_j / F_k \text{ from which } a_j = H_{kj} * F_k$$

substituting the values of aJ and Fi, as found, and remembering that:

$$H_{kk} = a_k / F_k$$

one has:

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

On the basis of this theory the transfer functions were constructed between various points on the support frame and some points on the inside of the driving compartment (fireproof partition, roof, floor) by exciting the structure at one point on the support frame (that used for the modal analysis). This test produced results in perfect accordance with the modal analysis of the support frame, supplying however further information: one can observe how by shifting the engine elastic support positions a few centimeters, a very different excitation of the various panels is provoked. Therefore this second method allows for higher precision in the identification of the points to position the silent blocks.

6. Conclusion

The principal results that have emerged from the research can be abbreviated as follows.

The engine and gearshift behave as rigid bodies showing a localized deflection in the area of the clutch bell housing, and therefore the most stationary and useful points for the connection to the body are to be found centrally.

The support mountings of the various accessories show numerous real frequencies, but their movements do not influence vibration transmission.

The engine support mountings, in particular those at the rear end of the gearshift, show numerous real frequencies and therefore one can foresee a specific study of such elements.

It was established that small changes in the position of the attachment points of the silent blocks to the support frame produce a reduction in the level of excitation in the inside walls of the driving compartment.

EXPERIMENTAL EVALUATION OF THE AXIAL CRUSHING BEHAVIOUR OF THIN WALLED COLUMNS WITH DIFFERENT CROSS-SECTIONS.*

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ABSTRACT 1.1.6

The paper deals with the experimental investigation of the post-buckling behaviour of thin walled columns with different cross-sections under axial quasi-static loading. The quasi-static collapse mechanism is almost the same of the dynamic collapse mechanism, but the lower deformation velocity allows a better insight.
Square and rectangular extruded sections and typical spot-welded hat-sections are considered. For this latter type of beams a parametric study is developed to point out the influence on the energy absorption of two design parameters: the width of the lateral flanges and the spot-welding pitch.
Experimental results are reported in terms of deformed shapes, of force/displacement diagrams and through some values characteristic of the crushing response. In the case of the hat-section beams these characteristic values are rather disperse; the applied statistical analysis allows interpretation.

1. INTRODUCTION.

The car-body structure plays some important roles in the vehicle design, one of these is connected with passengers safety during accident. In this case the kinetic energy of the vehicle should be dissipated very fast and in a controlled way.
This energy dissipation is generally obtained through crushing of car-body metallic structures. The resulting deceleration should not exceed certain values connected with the human body survival (this is the meaning of "controlled way"), moreover the passenger compartment should remain undeformed as much as possible.
It is well known that in the energy absorption of the car-body structures during impact the initial buckling of the frame beams is less important than their subsequent post-buckling behaviour.
This latter phenomenon is characterized by localized large plastic strains (that generate the so called "plastic hinges") in some positions along the beams. The plastic hinges allow large rotations (folding) of the beam wall.
The final result is a dramatic change of the shape of the structure.
During recent years there has been considerable research activities on the post-buckling (plastic) behaviour of thin walled beams that is the necessary preliminary step to understand the basic folding mechanism also involved in the collapse of more complex structures, of the type of car body structures.
Thus the behaviour of thin walled tubes with circular, square and rectangular cross-section, mainly subjected to axial loads has been widely investigated from the experimental [1,2,3] point of view. A theory based on the "moving plastic hinge" model has been developed [4,5,6,7] to explain the folding mechanism of the above

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