Abstract

Nowadays, the highly competitive automotive business industry requires manufacturers to pay more attention to passenger comfort and riding quality. This has forced designers to direct their attention to the development of high quality engine mounting devices, with traditional physical prototyping and testing being gradually replaced by virtual prototyping and numerical simulations. Following this trend, this article presents the work that has been done in order to create a finite element model of an available automotive elastomeric engine mount so that a deeper understanding on the dynamics of these components can be obtained to form a basis for improved design and development of future mounts.

In this work, special attention is given to the accurate modelling of nonlinear effects on the static behaviour of the mount with a comparison being performed between experimental and numerical tests. Based on the finite element model created, the mount’s frequency response function curves are determined and multi-dimensional effects in the mount’s response are observed. Also, the mount’s time dependent response to step-like moderate amplitude loads is obtained. The overall results indicate that the modelled mount has an acceptable performance in both isolating engine induced vibration and suppressing high amplitude engine shake movement. Finally, some remarks are made about the use of parallel associated rubbers and nonlinear dynamic properties as a mean for improving the mount’s dynamic behaviour.

Keywords: vibration, isolation, engine mount, rubber, finite element method.

1. Introduction

An engine which is perfectly balanced for forces and moments will have no tendency to move or to transmit vibration to the frame or foundation to which it is attached. Unfortunately, there is no such thing as a perfectly balanced engine. As a consequence, flexible mounts are needed for supporting an automobile engine and its peripheral components (e.g. gearbox). Among the reasons for using flexible mounts one may mention: to prevent the fatigue failure (caused by small dynamic disturbances) of the engine and gearbox support points; to reduce the amplitude of engine vibration which is transmitted to the vehicle’s body structure; to reduce human discomfort and fatigue by partially isolating the engine vibrations from the body by means of an elastic media. Hence, the design of engine mounts must be performed with care in order to ensure that each and every one of the engine mount’s functions can be accomplished successfully within some performance parameters ranges.

To isolate the vibration caused by the engine unbalanced disturbances, i.e. in order to ensure low values of vibration transmissibility, low elastic stiffness and damping are needed as the forces transmitted to the structure are related to the stiffness and damping of the mounts. However, if the mount’s elastic stiffness and damping are too low, the transient response of the engine mount system can become problematic in the case of shock excitations caused, for example, by sudden vehicle and engine acceleration and deceleration, braking and riding on uneven roads. Hence, from this point of view, high elastic stiffness and damping are required to minimize the engine motion and absorb engine shake. From this discussion is easy to deduce that, in the development phase, there usually must be a compromise between the fulfilment of some performance requirements.

Since the first elastomeric mount proposed in 1930 [1] to isolate vehicle structure from engine vibration, many design proposals have been presented [2]. Traditionally, the development of elastomeric engine mounts has been tackled using physical prototyping and testing...
techniques (highly time-consuming and expensive processes). However, these are gradually being replaced by virtual prototyping and numerical simulations [3]. This article presents the work done in order to create a finite element (FE) model of a particular type of engine mount used in a current commercially available passenger vehicle. In Fig. 1.1 a picture of this mount is shown.

This mount, as many others, is build upon a metal structure which is attached to the body of the vehicle, an elastomeric nucleus that acts as a vibration isolator and a metal core that provides the attachment support of the engine through a bracket type component. Overload and rebound control is accomplished with two rubber blocks positioned in the lower and upper section of the mount, respectively. As the vertical load increases and, consequently, as the middle V’ shaped rubber block moves towards full distortion, contact between the deformable ‘V’ shaped block and one of these rubber stoppers will take place. If the load continues to increase, these rubber blocks are squeezed to each other increasing significantly the mount’s stiffness and, thus, controlling the overload and eventual rebound behaviour.

2. Methodology

2.1. Mount’s Static Behaviour

The standard approach to determine the behaviour of rubber-like materials is to perform a set of experimental tests in order to obtain the material’s stress-strain curves for a certain group of possible modes of deformation and then calibrate the material coefficients in the hyperelastic strain energy functions from this experimental information using a least-squares-fit procedure, which minimizes the relative error in stress [4]. Depending on the accuracy requested, one must perform one or more of these tests in order to fully describe the behaviour of the elastomeric materials when subjected to various loading conditions.

For the case of the work presented, it wasn’t possible do perform such tests due to the inexistence of samples made out of the same material as the one used in the mount. The alternative, and adopted, approach was to perform a pair of static tests of the engine mount in order to obtain its load-displacement curves in two orthogonal directions (Fig. 2.1). Afterwards, using the FE model of the mount, this experimental test was numerically reproduced and its resulting load-displacement curve was used to determine the material coefficients that best described the behaviour of the elastomeric material.

Fig. 2.1 – Experimental set-up to obtain the static load-displacement curves

2.1.1. Finite Element Model

To create a finite element model of the mount, a set of simplifications to the geometry had to be made to maintain its complexity at a reasonable level. To begin with, the metallic components in the mount were
modelled as rigid (by specifying a pair of surfaces with an imposed rigid behaviour) since their stiffness is several orders of magnitude higher than natural rubber’s stiffness. In addition, to improve mesh quality over the rubber isolator, this component was divided in three individual regions defining a ‘V’ shaped core component and two other ones placed in opposite sides of the core respectively as depicted in Fig. 2.2.

In terms of the mesh’s characteristics, where it was possible, hexahedral elements were used to build the mesh, while in certain regions the complexity of the geometry made inadequate the use of this element shape, being opted the use of tetrahedral shapes. Following the recommendations made by [4], in the case of the hexahedral elements, an 8-node linear interpolation with hybrid formulation element (C3D8H) was used, while in the case of the tetrahedral elements, a 4-node linear interpolated element with hybrid formulation (C3D4H) was used instead, with compatibility between different meshed regions being imposed using tie constraints. Either one of these element types have 3 translational degrees-of-freedom for each node.

In order to account for the presence of large displacements and deformations, contact interactions between surfaces and a hyperelastic material model, a nonlinear static analysis was performed.

In terms of dynamic properties of the mount’s rubber, the information available was scarce. It was known that the rubber could be approximately modelled with a hysteretic damping model, i.e. with constant dynamic properties, having a loss factor of approximately 0.1. Without knowing the exact rubber’s dynamic behaviour,

2.2. Mount’s Dynamic Behaviour

2.2.1. Frequency domain behaviour

Engine mounts for application in the automotive industry are subjected to various loading conditions depending on the engine working regime. Starting from engine disturbances loads with low amplitude-high frequency characteristics, elastomeric mounts must also support the engine’s weight while being able to cope with high amplitude-low frequency loads that tend to make the engine bounce in its compartment. For the engine disturbances loads, experimental tests performed by an automotive constructor indicate that, at an engine idling speed of 750 rpm, these small disturbances could be idealized as:

\[ F_{dist} = \sum_{i=1}^{2} A_i \sin(\omega_i t) \]  

with

\[ A_i = 1 \text{ N} \]
\[ \omega_1 = 25 \text{ Hz} \]
\[ \omega_2 = 50 \text{ Hz} \]  

If, for instance, the engine was subjected to an acceleration until 3000 rpm the results shown that the disturbances would not be the same as (1). Instead:

\[ F_{dist} = F_0 + \sum_{i=1}^{3} A_i \sin(\omega_i t) \]  

with

\[ F_0 = 1200 \text{ N} \]
\[ A_i = 1 \text{ N} \]
\[ \omega_1 = 100 \text{ Hz} \]
\[ \omega_2 = 200 \text{ Hz} \]
\[ \omega_2 = 250 \text{ Hz} \]  

i.e., besides the small engine disturbances loads, the mount was subjected to a static load cause by the changes in torque that would result from vehicle acceleration.

As mentioned, the experimental measures performed were made in two orthogonal directions to determine the static load-displacement curves of the mount. Hence, the boundary conditions imposed in the FE model tried to simulate with the best accuracy possible the experimental tests constraints. Since it was requested to determine the mount’s static load-displacement in the three mutually orthogonal directions, three sets of boundary conditions were applied: common to all of them, the external sleeve rigid surface was pinned, i.e. all the degrees of freedom (DOF) were restrained; additionally the displacements of all the nodes belonging to the internal sleeve rigid surface were restrained in all coordinate directions except the one corresponding to the applied load (three different load direction cases were analysed).
material properties available in [5] for several rubbers were tested to see which one resulted in a dynamic behaviour of the mount similar to the one observed when one uses the hysteretic damping model with a loss factor of approximately 0.1 in the FE model.

To determine the frequency dependent response of the mount, a direct-solution steady-state dynamic analysis was performed [4]. This kind of analysis provides the steady-state amplitude and phase angle of the response of the system due to harmonic excitation at a given frequency. Two base states were considered in this analysis: the one corresponding to the load cause by the engine’s weight applied in the mount (mass of 50 kg); and the other with an additional load of -1200 N in the vertical direction, which simulates the conditions of vehicle acceleration until 3000 rpm. The frequency of the applied force (of unitary amplitude) was swept over the range 1 – 300 Hz (approximately 0.16 - 1885 radians per second).

2.2.2. Time domain behaviour

In normal operating circumstances, the mount is subjected to various loading conditions, from harmonic loads with high frequencies and small amplitudes to quasi-instantaneous loads of large absolute value. To analyse the transient response caused by each possible combination of loading conditions is very difficult and time consuming whilst it may prove to be unnecessary. The loads more capable of inducing significant engine bounce are the one caused by sudden vehicle acceleration or by high amplitude road disturbances. A conservative way of analysing these loading conditions is to assume that the loads are applied instantaneously and remain constant with time.

Although the fact that most of the loads applied in this fashion to the mount have a magnitude high enough to result in temporary contact between the mount’s ‘V’ shaped rubber core and its overload and rebound rubber stoppers, this constraint could not be treated with the current FE model. It is of common knowledge that high speed contact between surfaces is very difficult or maybe impossible to treat in some finite element models. This was the case of the mount’s FE model. Earlier analyses showed that the inclusion of contact interactions was out of reach and could not be performed. However, for loads with a smaller magnitude which didn’t yield in contact between surfaces several conclusions could also be made. Hence, to analyse the mount’s transient response to instantaneously applied loads, a load of 250 N applied in the mount’s downward vertical direction was considered as shown in Fig. 2.3. Similar to what was performed in the frequency domain analyses, the mount was fixed in its outer rigid sleeve surface and a static nonlinear analysis was performed to determined the initial conditions caused by the presence of the engine’s weight.

Classical approaches for solving transient nonlinear dynamic problems using FEM analysis involve using a direct integration method [6]. Although initially it was considered the option of using a direct integration method for computing the transient response of the mount, this could not be done due to the nature of the material properties specification. ABAQUS software requires, in this type of analysis, dynamic material properties to be specified in the time domain, namely by specifying the material’s relaxation modulus. Although ABAQUS offers a tool for determine the relaxation modulus (by terms of a Prony series) based on the frequency dependent dynamic information it was verified that the algorithm that lay behind this tool could not adjust properly to the existing dynamic data. Hence, for being able to test the same materials mentioned in §2.2.1, an alternative type of analysis had to be considered – a transient modal decomposition analysis.

![Fig. 2.3 – Boundary conditions for the time domain analysis of the mount’s dynamic behaviour](image)

A validation step were results from both analysis types were compared showed that there is a general good agreement between results from both analyses indicating that is possible to analyse with a reasonable degree of accuracy the transient dynamic behaviour of this mount using a linear modal decomposition analysis.

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1 A pair of low damping rubbers: an unfilled natural rubber and a natural rubber filled with 50 parts by weight of HAF carbon black per 100 parts of rubber; and a high damping butyl rubber filled with 40 parts by weight of MPC carbon black per 100 parts of rubber.
3. Results

3.1. Hyperelastic Parameters Determination

To determine the material parameters that were able to describe the static behaviour of the mount, three material models (linear elastic, neo-Hookean and Mooney-Rivlin\(^3\) hyperelastic material models) were tested in the finite element model for the vertical direction (the most dominant one in terms of the actual loading conditions of the mount when the vehicle’s engine is turned on) and their best fitting parameters were determined.

Despite the relatively simplicity of the Mooney-Rivlin strain energy function, results indicate that, without any additional experimental data concerning the static behaviour of the rubber at hand, it can be used as a first choice in the modelling of the hyperelastic behaviour of this kind of natural rubber. For reference only, according to [7] the properties shown in Tab. 1 resemble the ones possessed by rubbers with a shore A hardness\(^4\) of 55.

In terms of the macroscopic behaviour of the mount, results showed that the vertical and the transversal loading directions have similar stiffnesses, namely 140 N/mm and 125 N/mm respectively, whilst the horizontal loading direction exhibit a much greater stiffness – approximately 470 N/mm for small displacement values. Since the longitudinal direction is expected to be the one which will be more affected by the engine’s running disturbances, the corresponding stiffness value determined will result in a system’s natural frequency (considering that the engine’s weight applied on each mount will be 50 kg) that falls somewhere between the range 6 – 12 Hz recommended by some authors [8]. This shows evidence that the design stage of the mount took into account not only the static performance of the mount but also its dynamic capabilities to absorb engine induced vibration.

![Graph showing load-displacement curves comparison for the longitudinal direction](image)

**Fig. 3.1 – Load-displacement curves comparison for the longitudinal direction**

<table>
<thead>
<tr>
<th>Strain energy function</th>
<th>(C_{10}) [MPa]</th>
<th>(C_{01}) [MPa]</th>
<th>(D_1) [MPa(^3)]</th>
</tr>
</thead>
<tbody>
<tr>
<td>(W = C_{10}(I_1^3 - 3) + C_{01}(I_2^3 - 3) + \frac{1}{D_1}(J - 1)^2)</td>
<td>0.382</td>
<td>0.096</td>
<td>0.01</td>
</tr>
</tbody>
</table>

\(^3\) The choice to use a Mooney-Rivlin hyperelastic material model was made because, while it is still a relatively simple model, it is the most widely used constitutive relationship for the nonlinear stress analysis of elastomers [10].

\(^4\) Rubber hardness is defined as its resistance to indentation. Hardness measurements in rubber are expressed in Shore A or Shore D units according to the ASTM D2240 test procedures.
3.2. Frequency Domain Dynamic Behaviour

The measured reactions (forces and moments) about the external rigid surface’s reference point are plotted in Fig. 3.2 and Fig. 3.3 (using dynamic properties corresponding to a hysteretic damping model).

![Graph showing reaction forces and moments](image)

**a) Reaction forces**

**b) Reaction moments**

*Fig. 3.2 – Frequency dependence (in idling state) of the reactions - constant $G'$ and $\eta$ material*

It can be seen that, opposing to common one-dimensional treatment of these problems, the mount cannot be considered as a SDOF system since, when one applies a load in the vertical direction, there are reaction components in all coordinate directions. Furthermore, despite the good agreement between the reaction in the direction of the applied load and the equivalent SDOF model for low frequencies, the fact is that for higher frequencies multidimensional effects are visible, corresponding to the influence of other vibration modes shapes. To be able to assess the effectiveness of the isolation it is necessary to determine the resulting reaction from all of these components. The resulting reaction (equivalent to the system’s transmissibility since the applied load had an amplitude of 1 N) is shown in **Fig. 3.4** and compared to the transmissibility of a SDOF system with equivalent properties.

![Graph showing reaction forces and moments](image)

**b) Reaction moments**

*Fig. 3.3 – Frequency dependence (in acceleration state) of the reactions - constant $G'$ and $\eta$ material*

The frequency domain analysis conducted on the FE model showed that, despite common one-dimensional treatment of these kinds of problems, the mount should be analysed as a three-dimensional structure since, even when dynamic loads are applied in just one direction, three-dimensional effects are evident in the analysis of the mount’s transmitted loads. However, the overall conclusions that resulted from the one-dimensional approach were proved to be valid, with the mount being able to isolate from the vehicle’s body the disturbances produced by the engine in either idling or acceleration states, with transmissibility values that are very reasonable.

Starting from a simple one-dimensional model results showed that the static properties of the mount were carefully chosen since they result in a low natural frequency (between the range 6 – 12 Hz recommended by [8]) and, hence, in the excitation frequency of interest, indicated that the mount should performed according to what is expected of it, i.e. it is capable of isolating the engine induced disturbances from the vehicle’s body. Furthermore, the comparison between the results obtained from several rubber types and the ones yielded by an hysteretic damping model showed that the rubber used in the mount should be some sort of filled natural rubber with dynamic properties similar to those of a natural rubber filled with 50 parts by weight of HAF carbon black per 100 parts of rubber.
a) One-dimensional model in idling conditions

b) Estimate load transmissibility in the idling state using the FE model

c) Estimate load transmissibility in the acceleration state using the FE model

Fig. 3.4 – Mount’s transmissibility curves for a load applied in the longitudinal direction

3.3. Time Domain Dynamic Behaviour

The time dependent relative displacements caused by an instantaneously applied of 250 N in the mount’s downward vertical direction are shown in Fig. 3.5. From the analysis of the dynamic results several conclusions can be taken. First it is possible to see a good agreement between the results obtained for the mount with a filled natural rubber and the ideal material with a constant storage modulus and loss factor, reinforcing the conclusions taken from the results in the frequency domain. Next, as expected, it is possible to verify that an unfilled natural rubber, which possessed the best vibration isolation capabilities to low amplitude-high frequency loads, now has the worst performance of the lot of rubbers analysed, mainly due to its low damping capabilities. In contrast a filled butyl rubber has the best performance with the engine bounce being suppressed in about 0.5 seconds. With intermediate performance is the filled natural rubber being able to most of the engine bouncing movement in approximately 1 second.

Joining these conclusions to the ones obtained in the frequency domain analyses it is possible to conclude that special care was taken in selecting the appropriate rubber for use in this mount, with its static and dynamic properties being able to offer a good compromise between vibration isolation of the engine’s low amplitude-high frequency disturbances and engine bounce control when the mount is subjected to high amplitude-low frequency loads arising from sudden vehicle acceleration and/or driving in rough terrain.

4. Suggestion for Improving the Dynamic Performance of Anti-Vibration Rubber Mounts

4.1. Rubbers Placed In parallel

The fact that high damping rubbers, which perform quite well in suppressing engine shake, have a storage modulus that increases very severely with frequency usually makes them unsuitable for application in anti-vibration mounts. Snowdon in [5] and [9] mentions an approach for overcoming these limitations of high damping rubbers as vibration isolating materials. He
suggests that pairs of high and low damping rubbers should be mounted in a parallel configuration of suitable cross-sectional areas in a way that the overall storage modulus increases relatively slowly with frequency while the loss factor still maintains reasonable values. A schematic diagram of this approach is depicted in Fig. 4.1.

Let us assume that $A_1$ and $A_2$ are the cross-sectional areas of the low and high damping rubbers respectively. If $G'_1(\omega)$ and $G'_2(\omega)$ are the storage moduli of the same rubbers, the resulting system’s storage modulus is given by:

$$G'_e(\omega) = \frac{A_1}{A_1 + A_2} G'_1(\omega) + \frac{A_2}{A_1 + A_2} G'_2(\omega) \quad (5)$$

Defining $a$ as ratio of the cross-sectional areas, i.e., $a = A_2/A_1$, it follows that:

$$G'_e(\omega) = \frac{G'_1(\omega) + a G'_2(\omega)}{1 + a} \quad (6)$$

For the loss factor, it can be showed that:

$$\eta'_e(\omega) = \frac{G'_1(\omega) \eta_1(\omega) + a G'_2(\omega) \eta_2(\omega)}{G'_1(\omega) + a G'_2(\omega)} \quad (7)$$

To take advantage of the parallel placement of rubbers, a rubber with high damping at low frequencies should be chosen. Hence, a commercially available unfilled synthetic rubber of brand name Thiokol RD\textsuperscript{®} developed by Thiokol Chemical Corporation whose dynamic properties are available in [5] was chosen to be placed in parallel with an unfilled natural rubber. This synthetic rubber is characterized for having a considerably high loss factor throughout the usable frequency range of common structural applications, namely due to the fact that its transition frequency, as it can be seen in Fig. 4.2b for large values of $a$, occurs at about 900 Hz. On the other hand, its storage modulus increases severely with frequency making it unusable for direct application in the vibration isolation application studied in this work.

Fig. 4.1 – SDOF system with parallel mounted rubbers

Let us assume that $A_1$ and $A_2$ are the cross-sectional areas of the low and high damping rubbers respectively. If $G'_1(\omega)$ and $G'_2(\omega)$ are the storage moduli of the same rubbers, the resulting system’s storage modulus is given by:

$$G'_e(\omega) = \frac{A_1}{A_1 + A_2} G'_1(\omega) + \frac{A_2}{A_1 + A_2} G'_2(\omega) \quad (5)$$

Defining $a$ as ratio of the cross-sectional areas, i.e., $a = A_2/A_1$, it follows that:

$$G'_e(\omega) = \frac{G'_1(\omega) + a G'_2(\omega)}{1 + a} \quad (6)$$

For the loss factor, it can be showed that:

$$\eta'_e(\omega) = \frac{G'_1(\omega) \eta_1(\omega) + a G'_2(\omega) \eta_2(\omega)}{G'_1(\omega) + a G'_2(\omega)} \quad (7)$$

The association of this high damping rubber with an unfilled natural rubber in parallel configuration results in the dynamic properties depicted in Fig. 4.2. From the analysis of Fig. 4.2, one may already see that even for low values of $a$ the equivalent loss factor takes relatively large values throughout the frequency range of interest in the isolation of engine induced disturbances (0 – 1000 Hz).

Fig. 4.3 shows the transmissibility curves for a SDOF system with static properties equal to those treated in §3.2 ($K_{static} = 140 [N/mm]$ and $M = 50 [kg]$) whose dynamic properties are those depicted in Fig. 4.2. An analysis of the results obtained reveal that there are no obvious advantages in using a parallel mount composed of an association of an unfilled natural rubber with an unfilled Thiokol RD\textsuperscript{®} rubber in parallel when compared with the results using a filled butyl rubber (presented in Fig. 3.4a). In fact, if one wants to improve the mount’s performance near the resonance frequency (hence
improving its response to instantaneously applied loads) by increasing the relative proportion of high damping rubber in the mount this will lead to poor performance in the isolation of small high frequency engine induced disturbances.

This conclusion may not, however, be generalized to all possible combinations of known rubbers. Each case should be analysed individually as a function of each rubber’s dynamic properties, since, as mentioned throughout this work, the static and dynamic properties of rubbers can be very distinct between any two sets of rubbers.

4.2. Nonlinear Dynamic Properties

All the results presented till now are based upon the linear modelling of the dynamic behaviour of the mount, i.e., for a given excitation frequency, the dynamic behaviour of the mount does not depend on the magnitude of the resulting displacement. However, the introduction of nonlinearities in the dynamic behaviour of the mount’s materials can prove to be very useful in improving the overall response to the various loads imposed upon the mount.

A schematic drawing of one of such nonlinear models is presented in Fig. 4.4. Consider a pair of low and high damping rubbers placed in parallel configuration with the latter being shorter that the former, i.e. leading to the existence of a gap.

To understand the advantages of this model, one need to remember the characteristics of the loads usually applied to the mount: first, it is required that the mount isolates low amplitude high frequency loads – this is accomplished by this model since, when the mount’s strains are relatively low, the dissipated energy is also low favouring low transmissibility values; second, the mount should prevent the engine to oscillate severely when high amplitude low frequency loads are applied – since the dissipated energy is very high for high strain values, the engine bounce is therefore suppressed quickly.

Nevertheless, despite the obvious advantages of this kind of approach, the fact is that the numerical modelling of these types of nonlinear models is difficult to perform. In the particular case of the finite element software ABAQUS, only linear viscoelastic models (whose behaviour is only frequency and not amplitude dependent) are numerically implemented. In order to use such nonlinear models, ABAQUS have the ability to allow the user to program a subroutine called UMAT which can be used as part of the material definition, and, hence, to define material models that are not implemented in ABAQUS’ native code. However, due to the complexity and difficulty of this kind of programming, the advantages of using nonlinear dynamic material models were not tested using the FE model developed in this work. These tests are then suggested to be performed in future works.

5. Conclusions

In this work, computational modelling and simulation of an available engine mount was performed to gain an understanding of the dynamics of automotive elastomeric engine mounts and to evaluate the effectiveness of current market available solutions. Special attention was given to the correct modelling of nonlinear effects on the static behaviour of the mount. The experimental static analysis of the mount revealed the presence of two distinct regions in the load-displacement curve: one (with displacements up to 10 and 6.5 mm in the downward and upward direction respectively) in which the mount had an almost constant stiffness and another where the existence of contact between surfaces caused an abrupt increase in the
mount’s stiffness. The numerical simulation of the experimental tests revealed that the mount’s rubber nonlinear static material behaviour could be modelled fairly accurately using a simple two parameter Mooney-Rivlin hyperelastic model.

Since the mount is mainly subjected to two kinds of dynamic loads, the mount’s dynamic response was simulated using the FE model created. First, the response to harmonic loads of small amplitude was determined. A linear steady-state dynamic analysis was performed using dynamic properties for three types of commercially available rubbers and the results compared to the ones obtained using a hysteretic damping material model suggested by the automotive constructor. The results indicated that the rubber used in the mount had dynamic properties similar to those of a particular type of filled natural rubber. Furthermore, the results showed that mount performs quite satisfactorily in both engine idling and accelerating operating conditions with transmissibility values lower that 15% even when the operating conditions led to an increase in the mount’s static stiffness and, hence, a shift in the transmissibility curve to higher frequencies. An important fact that was verified in this work was the multi-dimensional nature of the mount’s dynamic response (despite the common treatment of these problems as one-dimensional ones). Nevertheless the analyses showed that the mount is well developed since its task to isolate, from the vehicle’s body, engine induced vibrations is successfully performed.

Another type of dynamic loads applied to the mount are high amplitude low frequency ones. The analysis of the mount’s response to these kinds of loads required the computation of the mount’s time dependent transient response. Problems with the specification of the rubber’s dynamic properties in the time domain arose making impossible the use of a nonlinear direct integration time domain dynamic analysis. The alternative was to use a linear mode decomposition analysis which proved to yield acceptable results for moderate amplitude loads. The results indicated again a good agreement between the use of the dynamic properties of a filled natural rubber and those of a hysteretic damping model proposed by the automotive constructor. Furthermore, the time dependent response using this latter model proved to be acceptable, with the mount being able to suppress in about 1 second the engine shake caused by a step-like load of 250 N.

The analysis of the mount’s response to these two kinds of loads confirmed the existing knowledge that the rubber which yields better results in isolating engine vibration from the vehicle’s structure (a rubber with a low dynamic stiffness) has the worst performance in suppressing engine shake arising from high amplitude transient loads and vice-versa.

Finally, the approach of using rubbers placed in parallel as a mean to improve the mount’s performance was tested. The results indicate that this approach should be considered in a case by case analysis since the conclusions taken for a pair of rubbers cannot be extrapolated for every possible rubber combination.

6. Bibliography