2019-24-0059 Published 09 Sep 2019

FEM Substructuring for the Vibrational Characterization of a Petrol Engine

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Citation: Citarella, R.G., Armentani, E., Giannella, V., Parente, A. et al., "FEM Substructuring for the Vibrational Characterization of a Petrol Engine," SAE Technical Paper 2019-24-0059, 2019, doi:10.4271/2019-24-0059.

Abstract

In this work the vibrational behavior of a 4-cylinder, 4-stroke, petrol engine has been simulated by leveraging on a reduced modelling strategy, based on the Component Mode Synthesis (CMS), adopted to reduce the size of the full FEM model of the engine.

The FEM model of the engine, comprising all of its sub-components, has been preliminary characterized from the vibrational standpoint; subsequently, the CMS has been adopted in order to reduce the FEM model size.

Frequency Response Function (FRF) analyses have been used to identify the resonant frequencies and mode shapes of the different FEM models, and the so-obtained results have been compared showing a very good agreement.

The reduced model has been able to reproduce with a high accuracy the vibration response at the engine mounts. The adopted reduced modelling strategy turned out to be effective in lowering the computational burden, keeping, at the same time, an accurate replication of the engine vibrational behavior. Runtimes have been significantly reduced from 24 hours for the full FEM model to less than 2 hours for the reduced model.

Introduction

Due to the ever increasing complexity of problems being solved and the limitations of power computing, several methods have been developed, since the early days of numerical computation, to improve the efficiency of numerical simulations. Many techniques have been investigated to reduce the computational effort required to solve large static and dynamic problems. Modal analysis techniques were developed to decouple the large set of ordinary differential equations and minimize the effort required to solve an iteration of the equations of motion [1, 2]. Substructuring methods were developed to approximate complex big structures as a collection of smaller single components, allowing the simulation process to be performed in a sequence of intermediate computationally lighter steps. These advances proved beneficial, especially in the simulation of dynamic systems, which require the solution of the equations of motion over large time intervals with many individual time steps.

Accurately assessing the dynamic behavior of these elements, when rather high vibration frequencies are involved, requires use of Finite Element (FE) models [3, 4, 5, 6] representing the geometry in considerable detail. Assembling the individual sub-components to build up a global FE model of the entire structure results in very large models, having a number of Degrees of Freedom (DoFs) which easily exceeds the limits of computer capacity, at least for reasonable run times. The question arises whether such FE models can be reduced in size, preserving at the same time the capability to represent the dynamic characteristics of the entire structure with sufficient accuracy. The substructuring method leverages on dividing a large model into subcomponents, these are separately analyzed and afterward re-assembled in a global model, through coupling of their mathematical description.

In the last decades, a variety of methods aimed at a model order reduction of dynamic problems have been developed within the area of structural mechanics, with the mode-based methods being the most frequently used. Fairly recently, methods originating from control theory, have been employed within structural mechanics. In contrast to mode-based methods which have an explicit physical interpretation, such modern reduction methods are developed from a purely mathematical point of view, see e.g. [7, 8, 9, 10].

In the present study, a substructuring technique is applied to a car petrol engine, modelled considering explicitly all its sub-components, i.e. gearbox, exhaust system, alternator, etc. In particular, the objective of the analyses carried out in the present investigation was to evaluate the performance of the Component Mode Synthesis (CMS) applied to the problem of a 4-cylinder, 4-stroke, petrol engine. A reference solution for the vibrational behavior of the engine was provided by a full

FEM model and used to validate the aforementioned reduced model.

A Frequency Response Function (FRF) analysis was used to identify the resonant frequencies and mode shapes of the different FEM models. The results were compared in terms of the vibrational response at the mounts, i.e. the gear mount and the engine mount. Also experimental results were available and used here for the FEM model validation.

The commercial FEM code MSC Nastran [11] was selected as FEM solver whereas the commercial code Siemens LMS Virtual Lab [12] was used for the dynamic analyses.

Full Engine FEM Modelling

The CAD model of the entire engine is shown in Fig. 1. Based on that model, a FEM model was created by means of the commercial code Altair Hypermesh [13]. Tetrahedral quadratic elements were used to model components such as engine crankcase, cylinder head, crankshaft, gearbox, etc., whereas quadrilateral quadratic surface elements were used to model thin elements such as the exhaust pipe or the oil pan. Bar elements were used to model the bolts that connects the various components. The average element size was set to 3 - 6 mm. The final full FEM model comprised nearly 5.7e6 elements and 4.7e6 nodes.

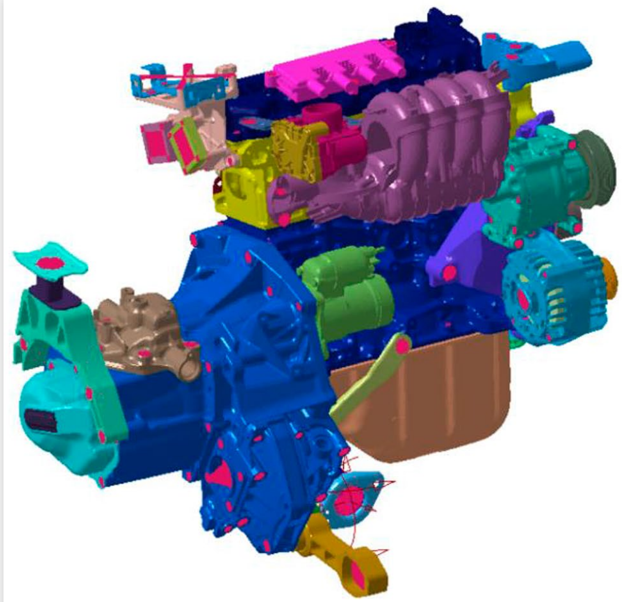
In order to validate such full FEM model, two Frequency Response Function (FRF) analyses were performed: one for the vertical bending and one for the lateral bending. A FRF analysis consists in the application of a unitary force in a certain point of the model and measurements of the accelerations in different points of the model. Such relevant points selected for the two FRF analyses of the engine are shown in Fig. 2.

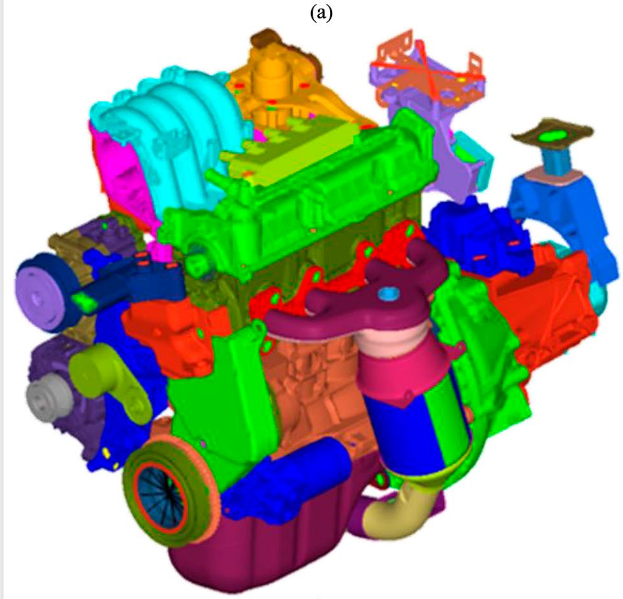
The FEM model was imported in the MSC Nastran environment to calculate the modal basis of the engine, up to a frequency of 1500 Hz (Lanczos method was used for such a purpose). A damping coefficient equal to 0.03 was considered for all the structural elements. Subsequently, the modal basis was imported in the Siemens LMS Virtual Lab environment, with input and output points defined for either the vertical (Fig. 2a) or lateral (Fig. 2b) bending. Finally, the models were solved allowing to calculate the accelerations. In particular, the impact of an additional engine bracket, shown in Fig. 3, on the acceleration levels at the previously mentioned relevant points was assessed.

Results of the Full Engine FEM Model

The accelerations were calculated at the engine and gear mounts for the two load cases of vertical and lateral bending. The ratio between acceleration and force are shown in Figs. 4-5. In particular, Fig. 4 shows results for the vertical bending at both mounts, with and without the engine bracket shown in Fig. 3. It is worth noting that the impact of the engine bracket is much more relevant for the vertical bending load

FIGURE 1 (a) Full CAD and (b) FEM model of the engine with all its sub-components.


(a)


(b)

case (Fig. 4), whereas it plays a minor role for the lateral bending load case (Fig. 5). This result was expectable due to the geometry of the bracket that allows to stiffen the engine-gearbox connection especially when undergoing vertical loads.

In addition, a comparison between numerical and experimental results is shown in Fig. 6. The FRF results show a satisfactory correlation with experimental outcomes in terms of peak frequencies, whereas the relevant differences in terms of magnitudes were judged as non critical for such kind of analyses. As a matter of fact, it is common practice in industry to consider the only frequency correlation as it would be too complex to arrange an experimental bench test aiming at a strict correlation between vibration amplitudes.

FIGURE 2 Full FEM model with details of the relevant points for the FRF analyses for load cases of (a) vertical bending and (b) lateral bending.

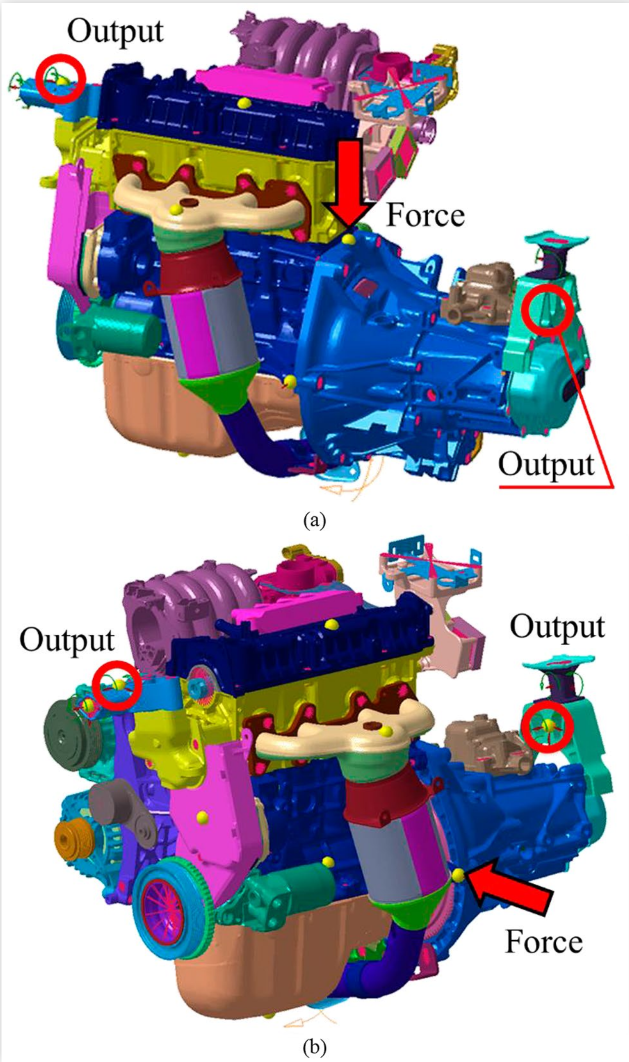


FIGURE 3 Close up of the engine bracket under analysis.

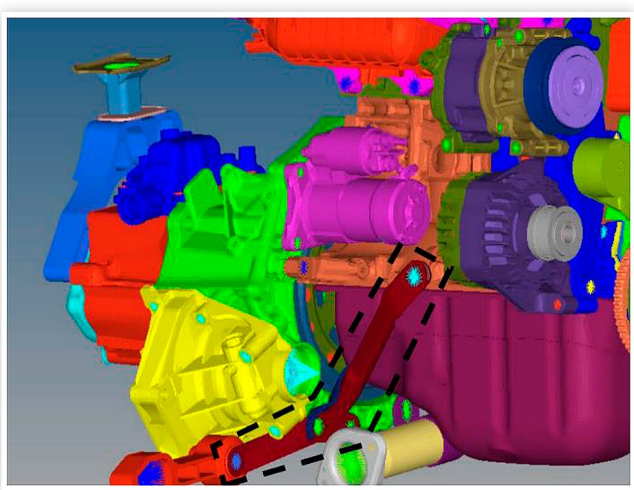


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FIGURE 4 Acceleration/Force ratio for the full FEM model for vertical bending load case with and without the engine bracket shown in Fig. 3 (a) at engine mount and (b) at gear mount.

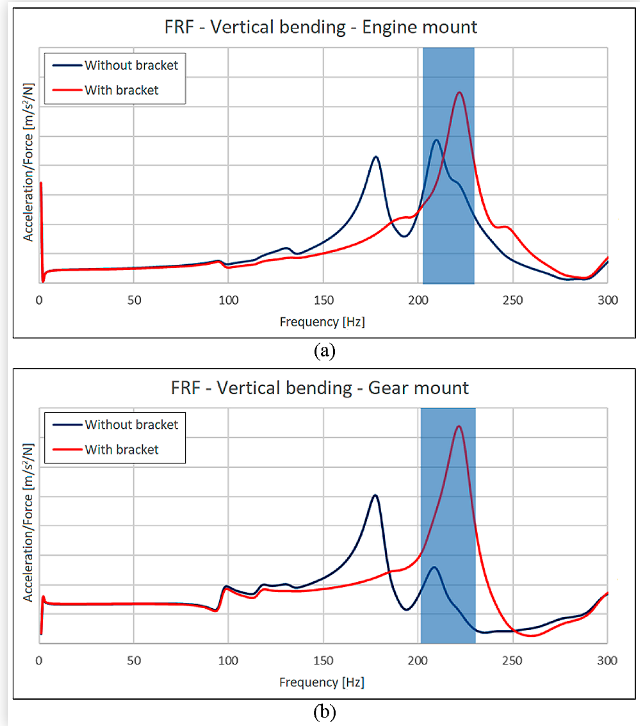


FIGURE 5 Acceleration/Force ratio for the full FEM model for lateral bending load case with and without the engine bracket shown in Fig. 3 (a) at engine mount and (b) at gear mount.

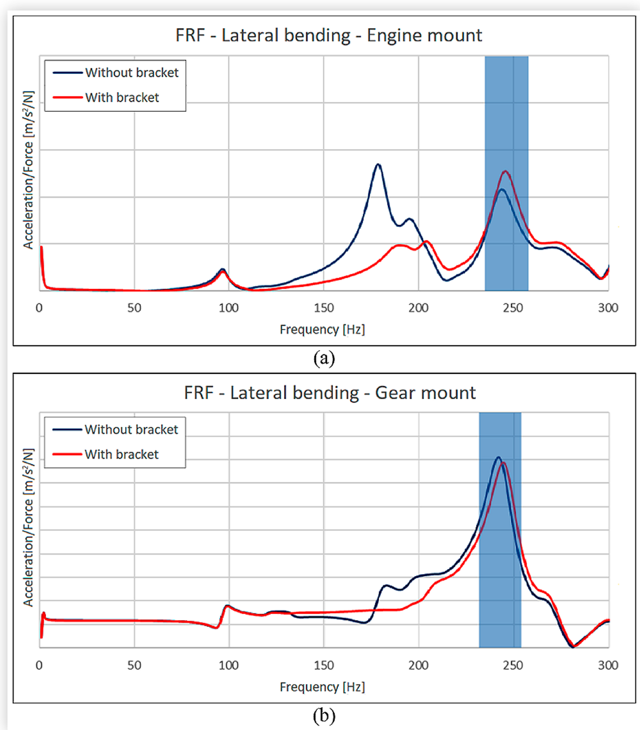
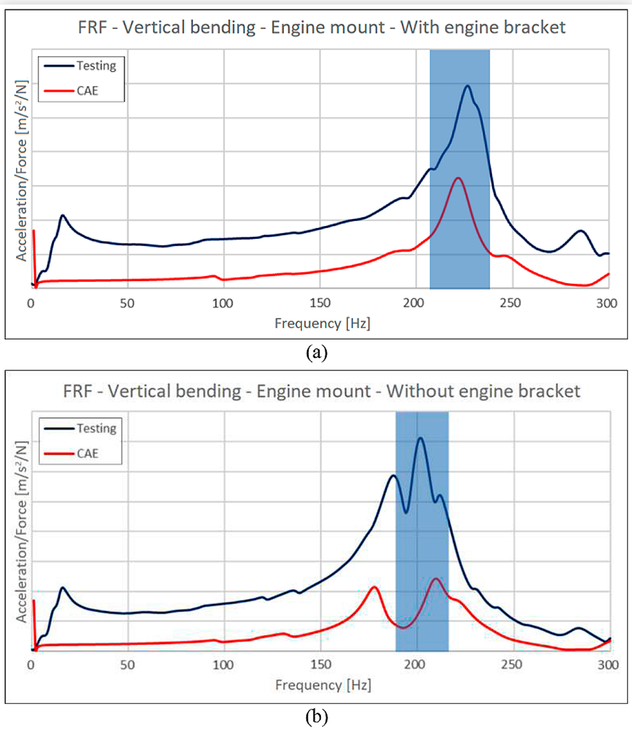


FIGURE 6 Numerical/experimental comparison on the Acceleration/Force ratio measured at the engine mount for the vertical bending load case: (a) with and (b) without engine bracket.



In particular, it is generally difficult to get a coincidence between the experimental and numerical excitation or monitoring points; in addition, the damping value set for the numerical simulation suffers from non-negligible approximations. Fig 6a shows a discrepancy between the peak frequencies of nearly 5 Hz for the model with the engine bracket, whereas Fig. 6b shows a discrepancy of nearly 10 Hz between the peak frequencies for the model without the engine bracket.

FEM Model Reduction

A reduced FEM model was created starting from the full FEM model. This was achieved by means of the ANSA MetaPost [14] code together with the pre-processor Altair Hypermesh and Nastran.

The reduction of the model was performed for the whole engine but for the sub-components, namely, the reduced model comprised a fully reduced engine model together with the engine original sub-components (still fully modelled by FEM). This modelling strategy was preferred since it allowed to efficiently deal with OEM's (original equipment manufacturers) that generally require the explicit modelling of their specific sub-components when assembled to the engine. From this standpoint, the model reduction of the engine allowed to significantly reduce the size of the FEM model to share, and also allowed the possibility to share the model without sharing proprietary information about the design of the engine.

Sub-components were removed from the full FEM model and PLOTEL elements [11] were used to model the engine. PLOTEL elements are Nastran dummy elements that are used only for display purposes without any significance from the dynamical standpoint. Such elements were adopted to represent the correct shape in the relevant positions, e.g. the load application points and bolt points, and also to keep the same engine shape and size.

The total mass and stiffness of the engine was associated to an RBE3 [11] element that was positioned in the cylinder head, whereas further RBE3 elements were also introduced to link the sub-components to the engine, in correspondence of the connecting bolts (these elements allowed to impose the displacements at the relevant positions directly connected to the sub-components.).

The final reduced model is shown in Fig. 8.

The model built up with PLOTEL elements is shown in Fig. 7a. It comprised 1885 elements and 1755 nodes (whereas the full FEM engine model required nearly 5.7e6 elements and 4.7e6 nodes). Such model was then imported in the ANSA MetaPost code and the reduced model was built up by considering the full FEM model modes as input modes (Fig. 7b).

FRF analyses were used in order to calculate the accelerations in both full FEM model and reduced model. Unitary forces were applied on the different sub-components and the accelerations were obtained in the same positions for both complete and reduced models. Fig. 9 shows the points considered on the alternator (Fig. 9a) and on the gear mount (Fig. 9b) in the FRF analyses.

Results of the Reduced Engine FEM Model

The FRF analyses were performed on the full and the reduced FEM model allowing to obtain the accelerations at the relevant points (e.g. those shown in Fig. 9). Fig. 10 shows the acceleration over force ratios calculated for two FRF evaluated at the alternator and the gear mount: a quite satisfactory matching is obtained.

Figure 11 shows an example of the modal shapes for the full FEM model and the reduced model: again a quite satisfactory matching is obtained.

The reduced FEM modelling allowed to calculate the accelerations at the sub-components with a reduced computational burden. In particular, the full FEM model required a solving time of nearly 24 hours, whereas the reduced FEM model required a runtime of nearly 2 hours. Such reduced model was then able to accurately replicate the engine vibrational behavior and it can be adopted in all the analyses that require the modelling of the engine with a very low contribution to the overall runtime. Moreover, since the sub-components are generally designed by external partners that require the analysis of the sub-component assembled to the engine, the adoption of the model reduction can overcome the key concern of not sharing proprietary design solutions.

FIGURE 7 Engine model built up with PLOTEL elements; in (b) also the RBE3 are shown.

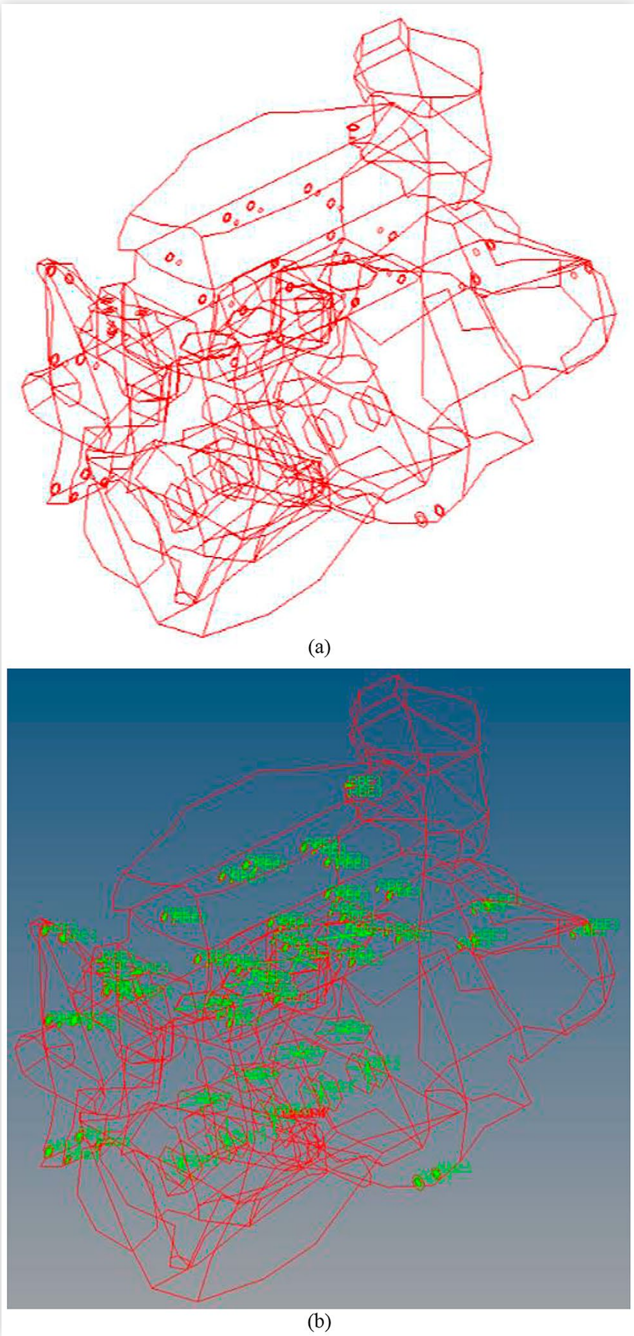


FIGURE 8 Final reduced model with all the FEM sub-components.

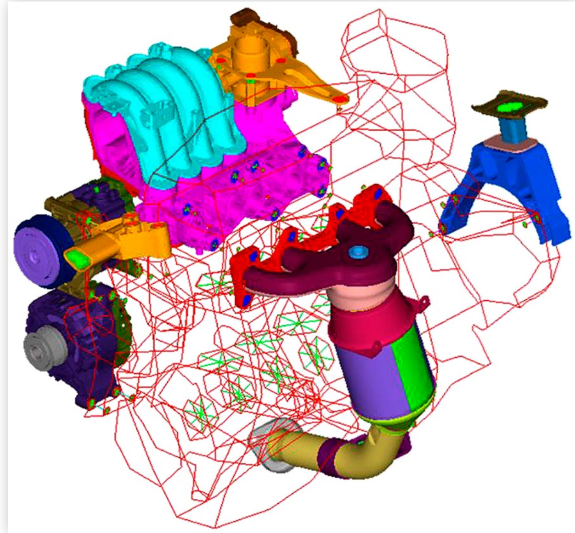


FIGURE 9 Points on the (a) alternator and (b) gear mount considered in the FRF analyses.

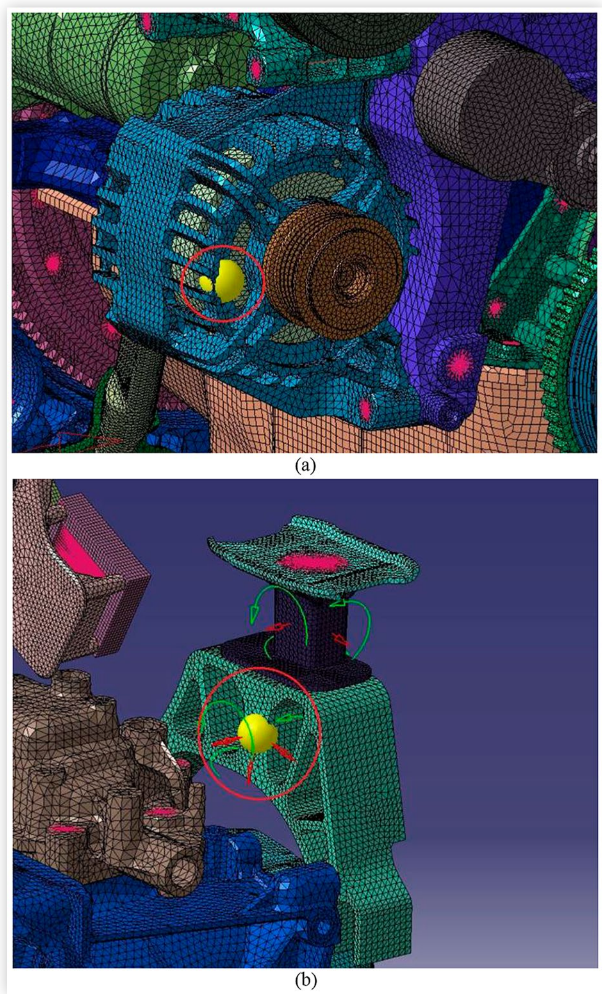
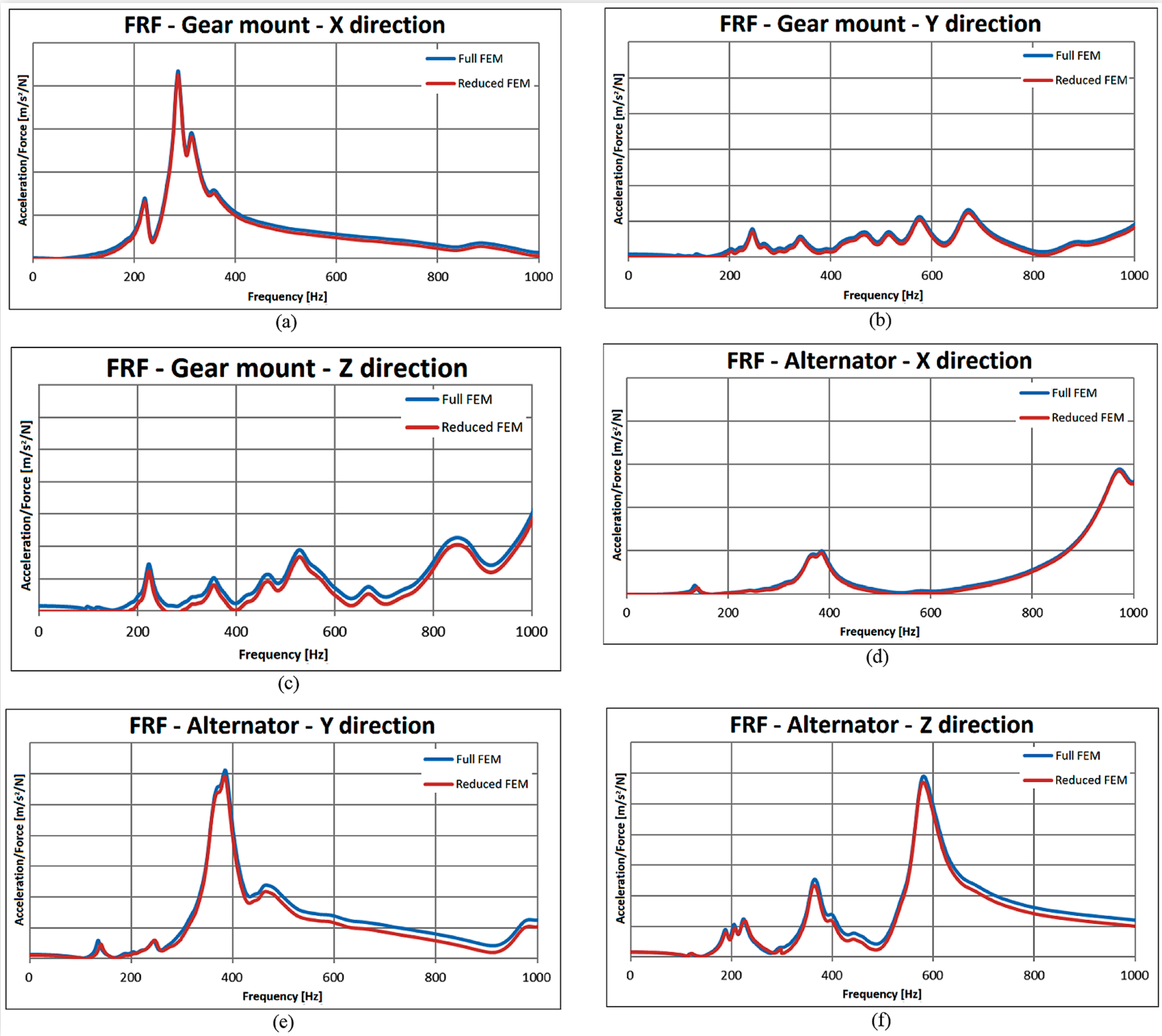


FIGURE 10 Comparisons of the acceleration/force ratio for the full and reduced FEM models (a-c) at the gear mount and (d-f) the alternator in the three directions.



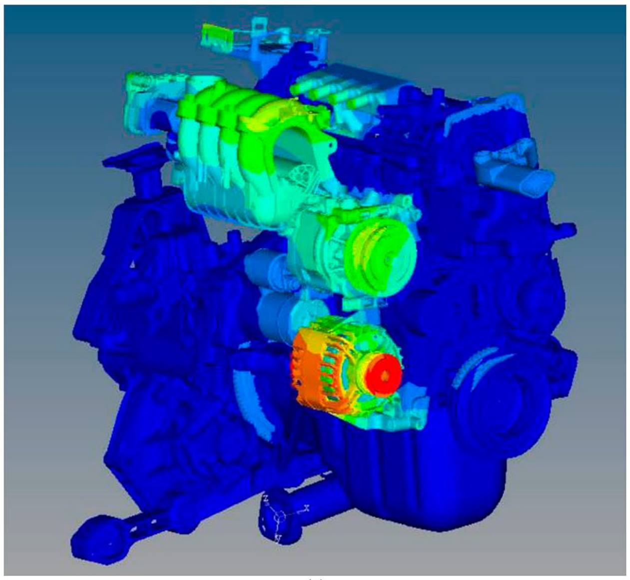
Summary/Conclusions

A 4-cylinder petrol engine has been simulated from the vibrational standpoint by leveraging on the Finite Element Method (FEM). In particular, a reduced modelling strategy based on the Component Mode Synthesis (CMS) has been adopted to reduce the size of the full FEM model of the engine.

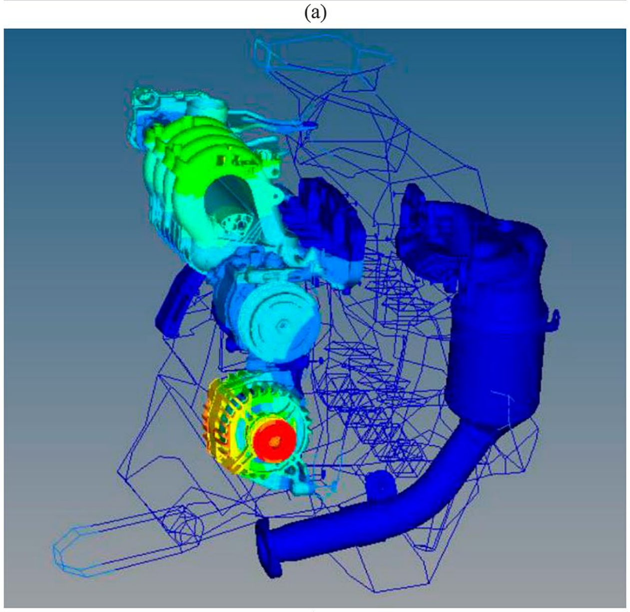
The full FEM model of the engine, comprising all its sub-components, has been preliminary characterized from the vibrational standpoint; subsequently, the CMS has been adopted in order to reduce the FEM model dimensions.

FRF analyses have been used to reproduce the vibration response of the engine and the corresponding acceleration levels have been compared between the models and test data showing a very good agreement. The adopted reduced modelling strategy turned out to be effective in lowering the computational burden from 24 hours to less than 2 hours, keeping at the same time an accurate replication of the engine vibrational behavior.

FIGURE 11 Comparisons of the modal shapes for (a) the full FEM and (b) the reduced FEM model.



(a)



(b)

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Definitions/Abbreviations

FEM - Finite Element Method
CMS - Component Mode Synthesis
FRF - Frequency Response Function
DoFs - Degrees of Freedom
CAD - Computer Aided Design

ISSN 0148-7191