

Vibro-Acoustic Modeling and Validation of Fuel Rails for GDI Wet Systems

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ABSTRACT

NVH simulations for an automotive component industry represent a convenient mean to compare different solutions and make decisions on design choices based on the predictions of the component vibro-acoustic behavior. This paper presents the vibro-acoustic characterization and comparison of two fuel rail assemblies (FRAs) by mean of simulations in Ansys Workbench & LMS Virtual.Lab.

These simulations required a preliminary finite element (FE) modal analysis on the FRAs. To verify the reliability of the FE models, an experimental modal analysis was performed on one of the two fuel rails in free-free condition. The correlation between FE and test models highlighted some differences: a sensitivity study proved that the differences depend on the modeling of some brazed joints.

The results of the following NVH simulations were checked by performing an acoustic impact test on the two FRAs in free-free condition inside an anechoic chamber. The comparison between the test and FE results proved that only a tuned FE model provides reliable results.

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INTRODUCTION

The growing need for reducing the car fuel consumption while increasing its performance contributed to the diffusion of high pressure gasoline direct injection systems, particularly in the European market [1-2]. Due to the new European Euro 6 particulate number restrictions, all the GDI engines have to meet legislative requirements significantly reducing exhaust emissions, especially with the implementation of Euro 6c, scheduled in 2017.

One of the strategies pursued by the vehicle producers, in order to reduce gasoline consumption and to fulfil the emission requirements, is to develop smaller engines (downsizing), which integrate turbocharge with direct injection systems and thus can work at higher loads and pressurization levels [3]. On

the other hand, the new GDI engines are a source of increased noise radiation and vibration transmission, which can be controlled only by a dedicated design of the fuel supply system layout [4-5].

In the last few years, the stronger customer demand for quieter vehicles has driven the engine R&D towards the investigation of noise and vibration sources in the fuel supply and delivery systems. Different investigations [6, 7, 8] showed that the high pressure pump and injectors are the main noise sources in GDI systems. The fuel rail assembly (FRA) acts as transmission path of vibration between the fuel pump and the injectors and its design strongly affects the acoustic radiation of the whole system (Figure 1).

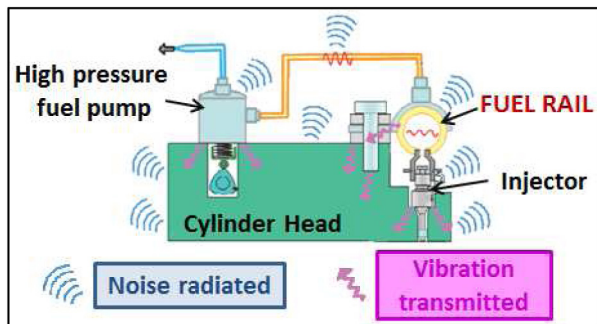


Figure 1. Scheme of a GDI engine with noise sources and vibration transmission.

In this paper a procedure for vibro-acoustic modeling of fuel rail systems is described and applied to compare two different design solutions. This activity has been carried in collaboration by the Modeling & Simulation Department of Continental Automotive Italy Spa, LMS Italiana Srl and the Department of Civil and Industrial Engineering of the University of Pisa.

FUEL RAILS ASSEMBLIES

The two fuel rail systems examined in this study are both designed for four in-line cylinders engines, equipped with the same type of injector. The FRA interfaces are the head of the engine, the fuel supply line and the injectors.

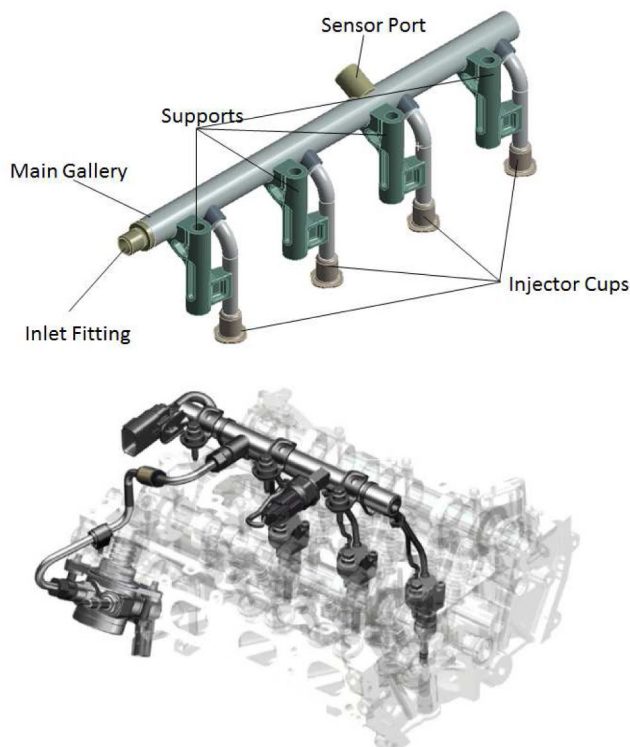


Figure 2. Top: Fuel rail assembly A. Bottom: View of the fuel rail assembly B and engine, from [3].

One assembly (FRA A) is characterised by four stiff supports at the connection with the head of the engine (Figure 2). On the other hand, the second assembly (FRA B) is connected to the

engine by means of ten thin brackets, separated from the cylinder head by damping components; additionally, it has longer and thinner fuel supply pipes. In their operating conditions, both FRAs are subjected to rapid loading cycles with forces transmitted by both the injectors and the cylinder head.

As far as modeling is concerned, since the whole system is very complicated to deal with, it was preferred to start from the characterization of the two FRAs under free-free conditions, i.e. without constraints, and then move to some analyses under simplified constrained configurations.

MODAL ANALYSIS

The first step in acoustic simulations is the determination of the modal behaviour of the structure. This means evaluating natural frequencies and mode shapes of the component in the frequency range of interest. In this section, the procedure for developing and validating the numerical model of FRAA is reported.

FEM Modal Analysis

The Finite Element models of both FRAs were developed in Ansys Workbench environment, starting from a simplification of their CAD geometry, consisting in the elimination of irrelevant details such as threads, trademarks, etc.. The material properties of an AISI 304 (200 GPa elastic modulus and 7.9 kg/dm³ density) were used.

An automatic mesh procedure with default 'Standard Mechanical' settings was chosen and the appropriate size of the elements was determined on the basis of a convergence analysis on the natural frequencies and mode shapes in the 0-10 kHz range. Such range was selected in the perspective of performing vibro-acoustic simulations in the field of higher sensitivity of the human ear (250-5000 Hz). The meshed model of FRAA is reported in Figure 3.

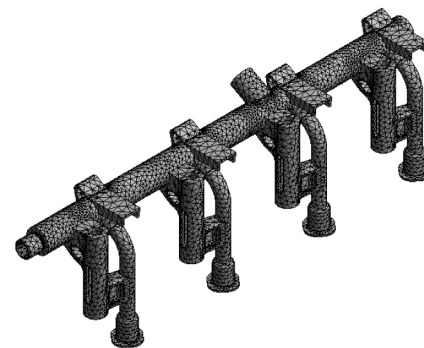


Figure 3. Mesh of FRAA for free-free modal analysis.

Experimental Modal Analysis

In order to validate the numerical FE models, experimental modal analyses (EMA) of the FRAs were performed in free-free conditions, after a preliminary check on the type of hanging, linearity and reciprocity of the system. The FRA was held by means of a rubber band as shown in Figure 4. A roving

hammer technique was applied, 24 impact points were selected on the basis of a geometrical criterion, and each one hit in three directions (Figure 4). A triaxial accelerometer was placed at the end of the main gallery.

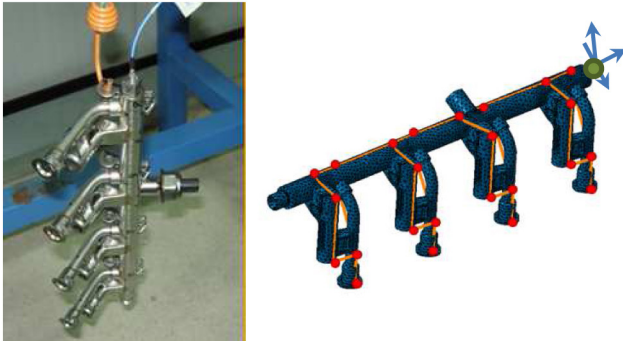


Figure 4. FRAA hanging for the EMA (left). Impact points in red and triaxial accelerometer in green on the right end (right).

The EMA was managed through the LMS Virtual Lab software. The acquired signals of the hammer and accelerometer were processed to estimate a total of 216 Frequency Response Functions (FRFs), from which the modal characteristics of the assemblies, such as the natural frequencies, mode shapes and experimental damping coefficients were obtained.

FEM-TEST Correlation

In order to assess the correlation between FEM and experimental modal analyses, the Modal Assurance Criterion (MAC) was applied [9]. This criterion estimates the correlation between a pair of modal eigenvectors, e.g. ϕ_i^{Test} and ϕ_j^{FEM} , by means of the following equation

$$MAC_{ij} = \frac{(\{\phi_i^{Test}\}^T \{\phi_j^{FEM}\})^2}{(\{\phi_i^{Test}\}^T \{\phi_i^{Test}\})(\{\phi_j^{FEM}\}^T \{\phi_j^{FEM}\})} \quad (1)$$

It can be noted that the MAC_{ij} value ranges from 0 to 1, indicating a bad or good correlation respectively. The comparison between test and FEM is commonly represented through a MAC matrix that for FRAA is shown in Figure 5. Rows and columns identify the modal shapes obtained by FEM analysis and EMA respectively and each element of the matrix gives the correlation between such modes, typically by means of a color, in a given color map. In this example, a perfect correlation between test and FEM modes would be described by a red diagonal over a green background.

Figure 5 shows that, for FRAA, there is a good agreement between FEM and experimental mode shapes, regardless of the value of the corresponding frequencies. The two shifts between modes 4-5 and 9-10 indicate an error in the calculation of natural frequencies.

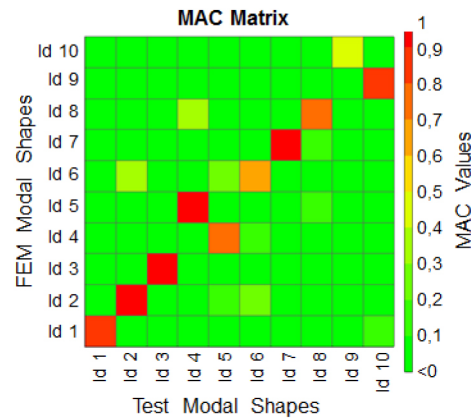


Figure 5. FRAA MAC matrix.

Additionally the Mode Pair Table reported in Table 1 shows the MAC values of the first 10 modes and the percentage differences of the natural frequencies.

Table 1. Mode Pair Table for Test-FEM modal correlation for FRAA.

Id Test	Id FEM	MAC Value	$f_{FEM} - f_{Test}$ (% of f_{FEM})
1	1	0.85	0.1
2	2	0.91	3.3
3	3	0.91	11.1
4	5	0.94	17.4
5	4	0.70	6.2
6	6	0.66	12.9
7	7	0.94	3.8
8	8	0.71	8
9	10	0.47	15.5
10	9	0.84	10.8

It can be observed from the MAC value in Table 1 that 8 out of 10 values can be considered acceptable, i.e. $MAC > 0.7$. Actually the correlation of mode shapes is usually quite critical above the first five modes. On the other side, a critical issue arises when considering the differences between the natural frequencies (last column in Table 1) that in some cases are higher than 10%. Such differences are confirmed by the comparison of the two FRFs shown in Figure 6, where the natural frequencies can be identified as the abscissae of the peaks of the two functions. It should be noted that, above the second mode, the peaks of the FRF for the FEM model are shifted to higher frequencies compared to those of the experimental FRF. This indicates that the FE model overestimates the ratio of stiffness and modal mass of the FRA.