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IDENTIFICATION OF THE VIBRATION EIGEN MODES FOR ELASTIC ENGINE MOUNTINGS

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Abstract: Engine mounting are an important vibrations path from the engine to the vehicle body. Reduction of automotive powertrain vibrations through proper design of mounts is of considerable interest. The goal of this paper is to present how to identify the eigen mode by dynamic run-up tests and impact hammer measurements. By comparing the results obtained by the two types of measurements can be evaluated the eigen frequencies in order to prevent resonance phenomena that can influence passenger comfort

1. INTRODUCTION

Mechanical vibrations and shocks occur in various mechanical structures, and are the outcome of the time variable load actions. These loads induced forced movements in a linear direction (strain) or angular (torque, torsion) or with a mixed movement on both directions. As a result, we can say that any vibration movement transmits forces and/or moments, with different displacements, and phases, resulting in negative effects for the entire structure

There are several substantial reasons for resilient mounting the engine and/or transmission. An increasingly highlighted reason is to decrease the engine structural noise and vibrations felt in the interior of the vehicle. Resilient mounting will help improve the life period of the frame, engine brackets and for the suspended parts as well as for the transmission, by diminishing the transient shock inputs and the torque loads. An engine is considered to be stable if the forces and moments which are transmitted to the mounting points have the size and direction of their vector constant. In case of the engine being unbalanced, the forces and moments transmitted to the engine mounts continuously change their vectors direct and size, and in these conditions, they produce vibrations in the vehicle body and other parts. These vibrations can become very dangerous if the oscillating frequencies overlap with the unbalanced forces and moments frequencies transmitted from the engine to the mounting points. The causes for the engine being unbalanced are the periodic vibrations of the inertia forces and their moments, as well as the lack of uniformity of the engine moment. In fact, the engine balancing is realized throw choosing the right number the disposition of the cylinders, cranks and choosing the suitable counterweights.

Until the 1960's a lot of research has been made on the noise and vibration caused by the engine and the transmission. Because the fact that the automotive business industry is a very competitive field, manufacturers must pay more attention to passenger comfort, therefore noise, vibration and harshness (NVH) have become increasingly important factors in the vehicle design. While the vibration issue has always been important for reliability and quality, the noise is of increasing importance to drivers and environmentalists. The harshness is referring to the sound quality. From the engine, loads are transmitted to the bushing attached to the body mount bracket. The bushing consists in an elastomer encased in housing. Its main role is that the bracket must react loads in all 3 translational directions. Also, the bushing must isolate from noise and vibration. In order for the mount to be effective, the stiffness of the bracket and the surrounding structure must be high enough. Each bracket engine mouning provides several functions like supporting the engine and transmission weight, providing to reaction to the drive-train torque and isolation from noise and vibration. Also it controls the power train motion.

The powertrain can be considered as a six degree of freedom system displacements:

- yaw, pitch and roll exhibiting:
- axial, lateral, vertical rotations.

It is forced into combinations of these motions by the forces, moments and torques caused by the crank train and the cyclic variations in gas pressures. A simple guide is that:

- vertical shaking forces encourage vertical displacements of the powertrain;
- horizontal shaking forces encourage lateral motion;
- moments in the vertical plane encourage pitching;
- moments in the horizontal plane encourage yawing;
- torques encourage roll.

The above guide is not truly accurate because the motions will be almost never pure but coupled (vertical motion coupled with pitching for example) and excited by a combination of forces, moments and torques.

There are some governing principles, however. The purpose of the engine mounts is threefold:

- To constrain the motion of the powertrain whilst the car is in motion in order to minimize the risk of damage caused by high cyclic stresses on the powertrain or on the chassis.
- To constrain the motion of the powertrain whilst the car is in motion in order to preserve driveability.
- Reduce structure-borne noise and vibration in the vehicle.

2. MODEL EQUATIONS

Most practical noise and vibration problems are related to resonance phenomena, where the operational forces excite one or more of the modes of vibration. Modes of vibration which lie within the frequency range of the operational dynamic forces, always represent potential problems.

An important property of modes is that any forced or free dynamic response of a structure can be reduced to a discrete set of modes.

The engine mount can be taken in consideration as the Kelvin-Voigt model.

m - mass of the vehicle frame

k,c - engine mount stiffness and damping coefficient

y_1 - engine excitation

According to the d’Alembert principle, we have the following motion ecuation:

$$m\ddot{y} + c(\dot{y} - \dot{y}_1) + k(y - y_1) = 0 \qquad (1)$$

And taking in consideration that the excitation is harmonic:

$$y_1(t) = Y_1 \cos \omega_b t \qquad (2)$$

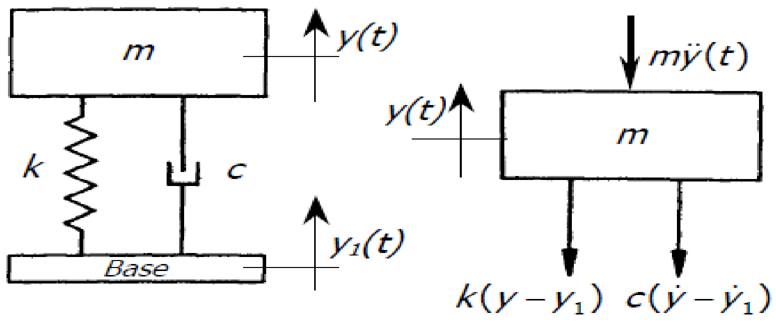

Figure 1. The Kelvin-Voigt model

Where:

Y_b - is the amplitude of the engine motion,

ω_b - is the frequency of the engine oscillation.

Thereby, from ecuations (1) and (2):

$$m\ddot{y} + c\dot{y} + ky = -c\omega_b Y_1 \sin\omega_b t + kY_1 \cos\omega_b t \quad (3)$$

Also, we can compute the transmissability (T) of the mounting structure:

$$T = \frac{Y}{Y_1} = \sqrt{\frac{1+4\zeta^2 u^2}{(1-u^2)^2 + 4\xi^2 u^2}} \quad (4)$$

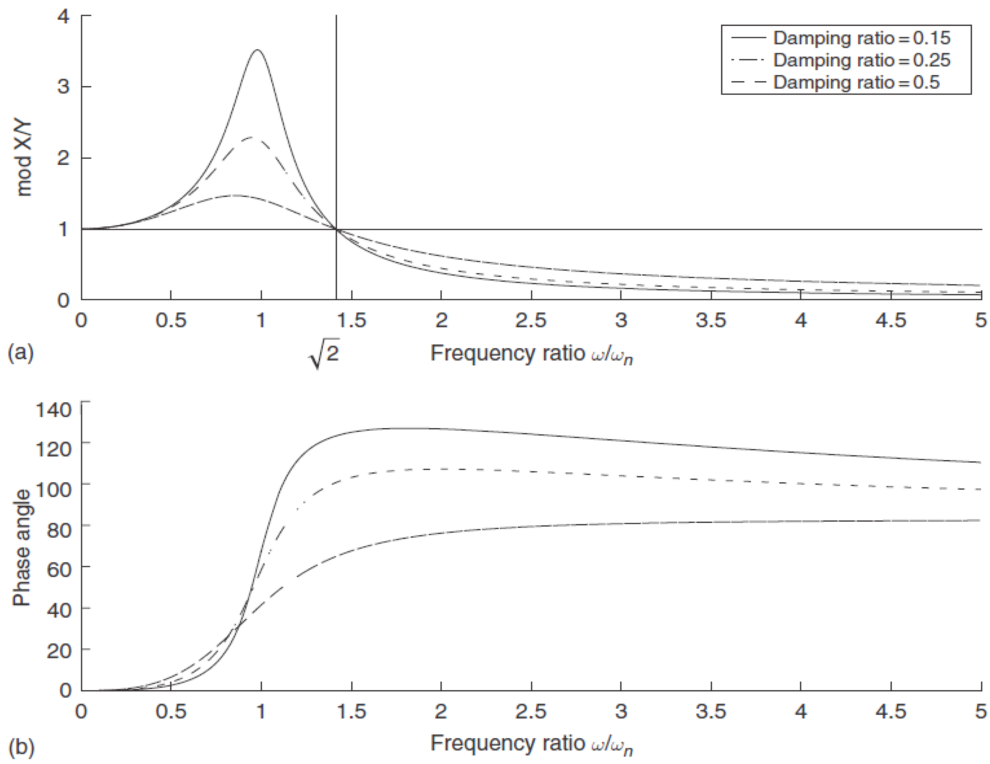

Figure 2. The response of the sdof when forced as its support

3. EXPERIMENT DESCRIPTION

The experiments took place using LMS equipment, acquisition and processing software – Testlab, and triaxial accelerometers B&K with a 10mV/g sensibility and weighting 20 grams each. Evaluation of the eigen values was made with an accelerometer fitted on the engine mount's bracket (SM02), and the assessment of the engine born excitations, with a sensor fitted on the engine cylinder head.

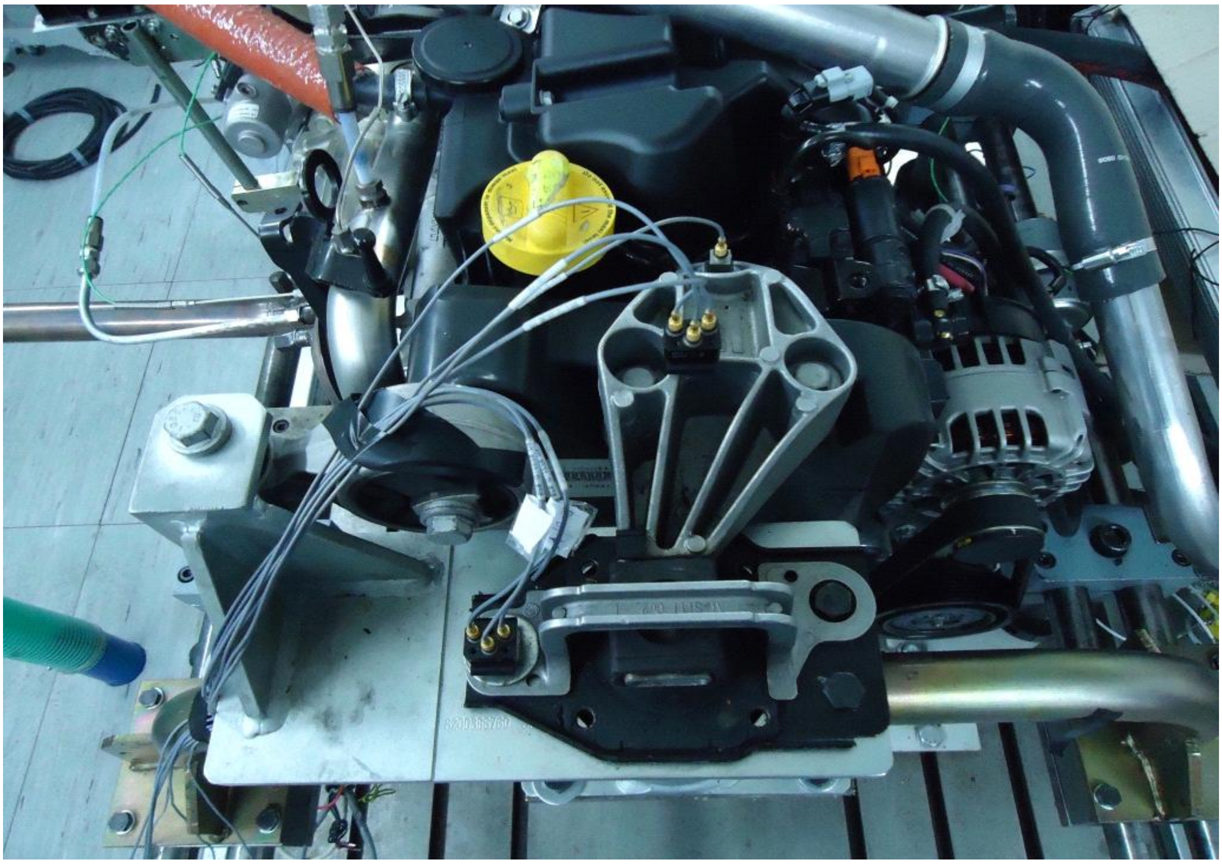

Figure 3. Sensors positioning

The acquisition procedure assumed the realization of measurements in an operational engine regime and impact testing, using the impact hammer. We performed an engine speed uphill regime from 950 up to 4500 rpm in order to identify the engine mounting's behavior under the entire range of engine speeds. Tests were run on the engine bench. For checking the repeatability of the measurements, we performed 3 sets of acquisitions and compared them.

The accelerometers were fitted with respect to the vehicle's coordinate system so that: x- longitudinal vehicle direction, y - transverse direction and z – vertical direction. The position of the sensors was not modified during the experiment in order to maintain the results accurate.

For the engine speed determination we used instead of a tachometer, TestLab Software, offline RPM Extraction mode. From the energy specter measured by sensors mounted on the engine head, the software determined the engine speed during the acquisitions

By introducing the tachometer signal we could evaluate the dynamic behavior of the engine mount under the engine born vibrations at different speed values. Therefore, by post processing the acquired signals, we can generate the Campbell Diagram, extract order, determine the overall level. Highlighting the eigen value from the dynamic measurements will be compared to the ones from the impact test.

At the impact test, the sensors maintained the same position as the dynamic test. The measurements were performed with the engine fixed in the test bench, not it was not in free-free conditions. Using a test hammer with 1.12 mV/N sensibility, we determined the accelerance/inertance in SM02 point (bracket) and the transfer function for RH02 (chassis frame).

4. RESULTS

A spectral map is a three-dimensional display of vibration or sound spectra as a function of time or speed. The spectra can be frequency or order spectra. A spectral map provides an overview of the frequency or order content of a signal related to time or speed. A spectral map can help you locate strong sound or vibration components, identify the components changing with the rotational speed, and identify the fixed components within a certain frequency range.

A colormap displays spectral map data in a customized intensity graph. The colormap uses different colors on the plot to represent the signal power distribution. When displaying a signal with a colormap, you can select any one of eight plot types to view different information related to time, speed, frequency, and order. The following front panel shows vibration results from a run-up test in RPM-Order displays.

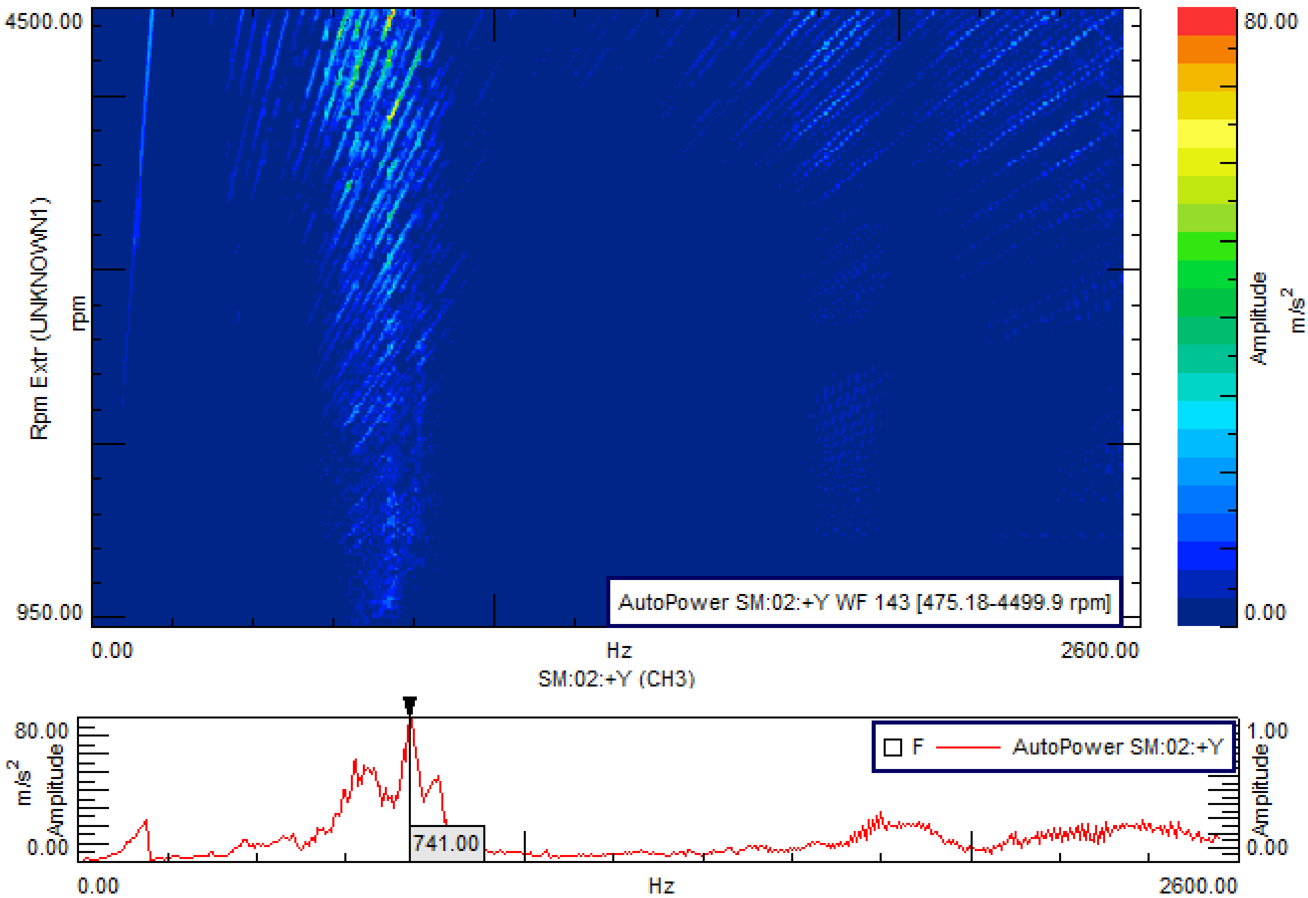

Figure 4. Colormap and peakhold

Before starting the post processing, a signal quality and measurement repeatability verification was performed. Post processing the acquired data was realized at a 2560 Hz bandwidth with a 25 rpm increment, for which the colormap and peak hold were extracted.

The colormap and peak hold are used for the medium frequency range, in order to analyze eigen mode, harmonics and shocks. The vertical line shows the first eigen mode of the engine mounting bracket at different values of the engine speed. We can see in Figure 4. that the first mode is at 741 Hz and the peak amplitude has a very high level (80m/s^2)

The most popular excitation technique used for modal analysis is impact, or hammer excitation. The waveform produced by an impact is a transient (short duration) energy transfer event.The most important measurement that is needed for experimental modal analysis is the frequency response function. Very simply stated, this is the ratio of the output response (SM:02) to the input excitation force (Hammer). This measurement is typically acquired using a dedicated instrument such as an FFT (Fast Fourier Transform) analyzer or a data acquisition system with software that performs the FFT (TestLab software).

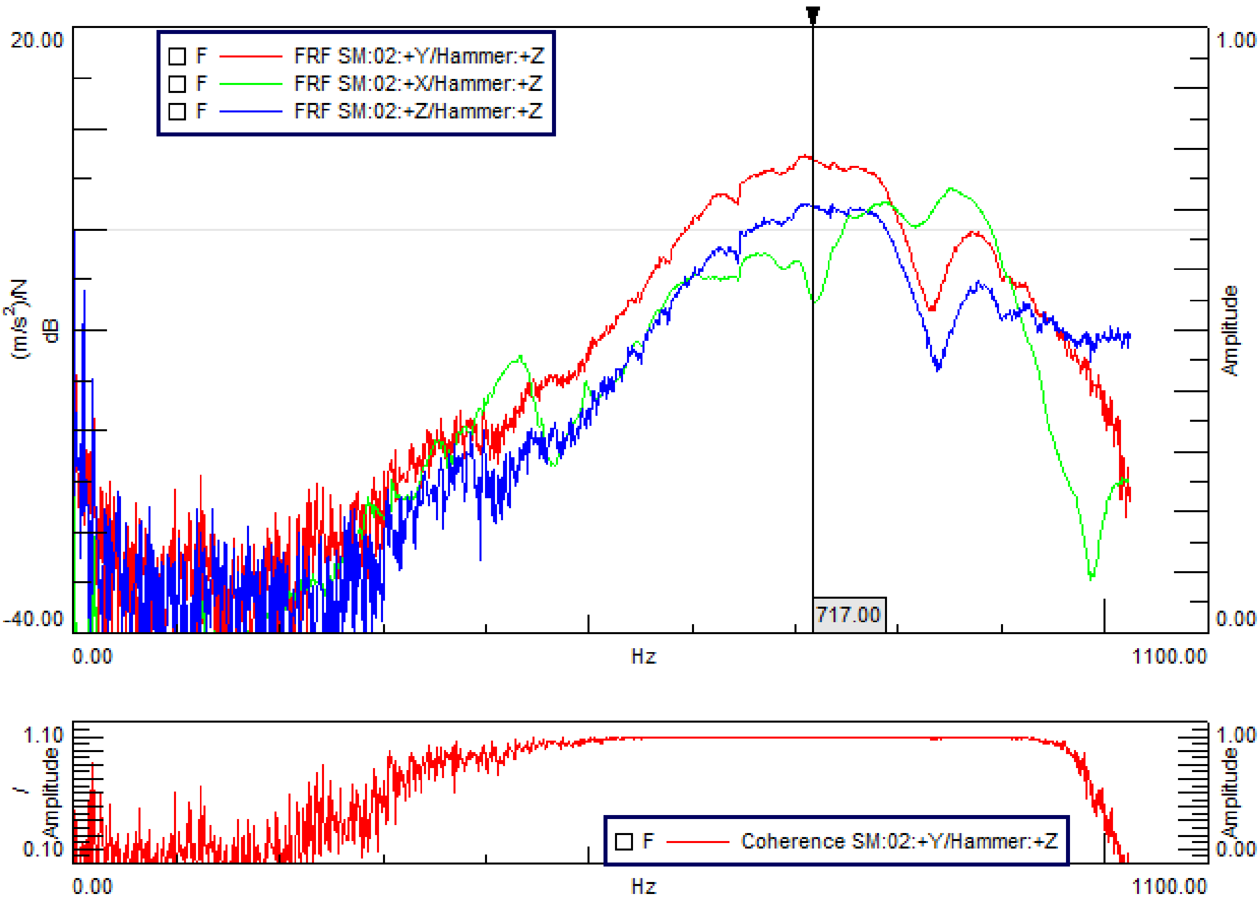

Figure 5. Frequency response function for impact testing

These functions are averaged and used to compute two important functions that are used for modal data acquisition – the frequency response function (FRF) and the coherence. The coherence function is used as a data quality assessment tool which identifies how much of the output signal is related to the measured input signal. The FRF contains information regarding the system frequency and damping and a collection of

FRFs contain information regarding the mode shape of the system at the measured locations

Measurements were performed by applying a total of 7 hits in each direction. Because the system response is measured with a triaxial accelerometer, was considered valid only measured signal corresponding to the impact direction. But, the fact that the frequency response graph shows the same eigen value for all measurement directions regardless of the direction of impact and coherence indicate a good quality of measurement, confirming the identification of the characteristic frequency at around 717 Hz.

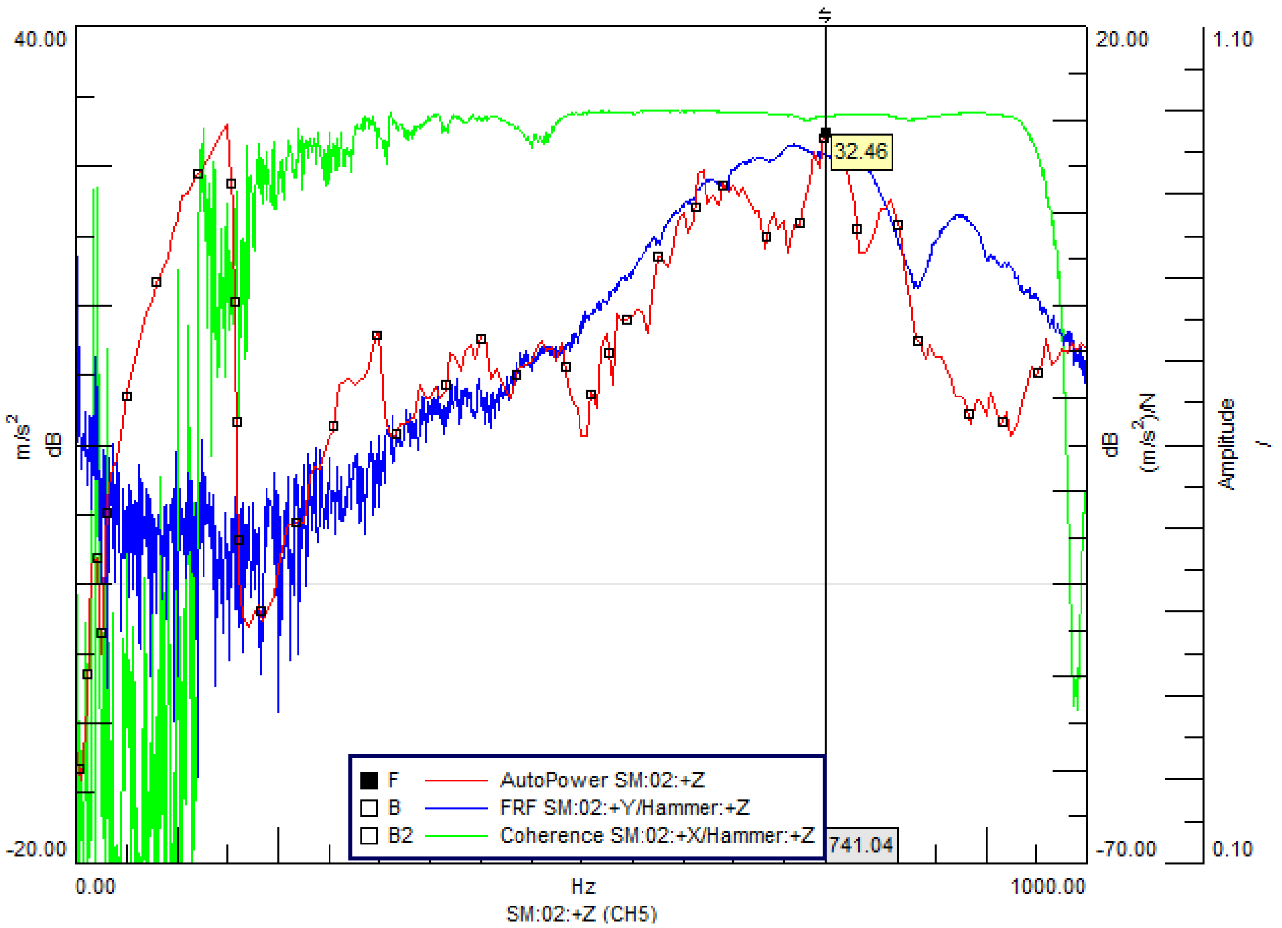

Figure 6. Peak Hold, FRF and Coherence comparison

By comparing the two types of measurements, the run-up and impact testing, we could identify the eigen mode of engine mounting at frequency range 717-740 Hz. Difference of 20 Hz can come from temperature differences of the engine mounting for every type of acquisition. For dynamic measurements run-up the bracket was heated and for hammer impact measurements were performed with cold engine bracket. Gap of 3.5% between the two measurements is acceptably and identify modes of vibration of the engine support around 740 Hz allows for further research and vibration measurements on the engine and its elastic elements.

5. CONCLUSIONS

The engine mounts have a very important role in damping the engine vibrations. The objective of this paper was to identify the eigen values of the engine mounts, based on the analysis of the impact hammer measurements and the engine dynamic run-up.

The vibratory characteristic was realized in the engine test bench on a K9K732 diesel engine. From the measurement results analysis under dynamic conditions we can observe an eigen value at the frequency of 741Hz. But on the analysis with the impact hammer we can see the eigen value is at 717 Hz. The gap between these 2 acquisitions represents a percentage of max. 3.5%. Even if very small difference, it is explained by different temperature conditions in which measurements were made. Thus, for measurements in dynamic engine run-up the engine mounting was preheated and heated during measurement but unlike the hammer impact measurements were made in cold engine support

These measurements help us in further research to assess the evaluate the vibration behavior of unconventional engine mounting compared to that on which measurements were made in this paper

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