Study concerning the influence of the engine mounting system on the vibration transmissibility to the truck cab

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Abstract:- The main objective of the study is to enhance driver and passenger comfort in the cab. The focus is on an analytical study concerning the optimization of the engine mounting system of a truck using a FEmodel. This model is useful for studies of the vibration transmissibility among different elements of the structure, in order to improve the comfort in the truck cab. The studies developed with this model are based on the Finite Element Method and the results are acquired through a transitory dynamic analysis. Experimental validations have been performed upon the analytically obtained results. The study on the engine mounting system optimization was realized by changing the values of elastic characteristics in the model. Also, the model is possible to be used to study the dynamic behavior of the truck structures under various work regimes.

Key-Words: - FEM, vibration transmissibility, comfort, engine mounting system, truck structure

1 Introduction
The researches concerning the comfort in the truck cabins implies several aspects like: decreasing the vibrations and noise due to the structure, reducing the vibrations transmissibility from both the road and the power unit to the cabin, and keeping up the climate parameters inside the cabin.

The engine mounting system of the vehicle is a vibration control system comprising several mounts that connect the engine to the vehicle structure. The major role of the mounting system is to reduce the noise, vibration and harshness perceived by occupants and to improve the ride comfort. The main vehicle NVH sources are low frequency road roughness and high frequency engine force[1,2].

The mounting system minimizes the transmitted forces from engine to the frame and prevents engine bounce caused by shock excitation. The system of interest in this study is the truck, whose engine is mounted to the chassis via three elastomeric mounts. Two of them are mounted in the front of engine, on the frame. The last is mounted on the traverse of the frame. The performances of the engine mounting system are correlated with the stiffness and damping values of the mounts. A high stiffness or damping value of mount can induce low vibration transmission at low frequency, but poor performance on high frequency. On the other side, low stiffness and low damping values can induce low noise level but high vibration level. To obtain balance between engine isolation and engine bounce a compromise is needed [3-5]. The vibration behavior of the mounting systems may be predicted, analyzed and improved by the analytical studies of the structure which uses modeling systems and structures with FE method.

2 The FE model of the structure
The Fig. 1 shows the analytical model of the truck and includes its main subassemblies and the mounting system that connects them. Some simplifications of the structure geometry were operated by choosing simpler forms for some structure elements. The finite elements mesh was generated by using “shell” elements for the surfaces modeling, and “spring” elements for the springs and dampers.

Fig. 1 The FE model of the structure
The masses of the subassemblies placed on the model structure were distributed over a number of points on the structure surface, so that the mass of the whole model becomes identical to the real truck’s one. The power unit was considered a concentrated mass placed in its gravity center. Because the weight of the analytical model is very important for the dynamic behavior of the truck, it was verified and corrected according to the real weight.

3. Model validation by experimental study
The presented model was validated by experimental studies. The studies were made on the same model of truck which was used to build the analytical model. The transmissibility was calculated as a ratio of the effective accelerations measured at M, R, C points on the truck structure. The accelerometers were placed on the truck structure in the same points as on the analytical model.

It was observed that the analytical vibration transmissibility curves (Fig. 2 and 3) are similar to the experimental ones, considering their shape and their transmissibility level for 1-250 Hz range [7].

4. Modal analysis and the vibration transmissibility study
The model presented in Fig. 1 was used for the study of the dynamic behavior of the main elements and of the whole system.

The analytical model offers the possibility to analyze the vibration at any point on the structure and allows determining the transmissibility between any two points of the model structure. To show the influence of the engine mounting system on the comfort in the cab the vibration transmissibility from the power-unit to the frame and to the cab floor were determined, Fig. 3-4, [8].

The Fourier analysis of the accelerations (Fig. 5) determined for these three points makes out besides the engine excitation frequencies. Excitation loads considered were: sinusoidal loads; one moment in the center of gravity of the engine, in the longitudinal direction, was introduced after the damping time generated by placing the cab on its mounting system. The frequency of the sinusoidal load was established according to the rotation speed of the engine. There were considered values from 550 min⁻¹ to 2500 min⁻¹, according with the real variation of the rotation speed of the real engine.
There were chosen three points: M, R and C situated on the engine (M), on the frame (R), respectively, on the truck cab structure, on the floor (C), where the values of vertical acceleration were determined. In Fig. 6 is shown the variation of the vertical acceleration for the M, R, C points for the sinusoidal loads.

Fig. 5 The Fourier analysis of the accelerations for the M, R and C points for the 1000 min⁻¹ sinusoidal loads.

Fig. 6 The variation of the vertical acceleration for M, R, C points for the sinusoidal loads

5. Results and discussion
The frequency of the sinusoidal load were considered from 550 min⁻¹ to 2500 min⁻¹, according with the real variation of the rotation speed of the real engine; this means a frequency variation from 25 Hz to 125 Hz.
For each of M, R and C points the time and the frequency variations of vertical accelerations were analyzed, Fig. 7.
In order to determine the vibration transmissibility, the parameters were calculated since that time when the torque load was introduced.
The transmissibility as ratio of the frequency variation of the vertical accelerations on the M, R and C points was calculated.
The elastic characteristics were easily changed in the model, in order to analyze the behavior of engine mounting system for different conditions [8, 10].
Were considered the optimum values of the stiffness and damping coefficients for the elastic cab system obtained by the analytical study.

Fig. 7 Time and frequency variations of vertical accelerations for M point
The graphs of transmissibility from power unit to the frame, respectively from the power unit to the cab, were determined, Fig. 8, 9.

Fig. 8 The frequency variation of transmissibility from the engine to the frame for different values of the engine mounts stiffness
The results highlight lower values of the C/M transmissibility than R/M vibration transmissibility. It can be observed that the variation of the transmissibility from the engine to the frame and from the engine to the cab too, present a maximum value around 50 Hz frequency. The explanation is that a twisting vibrational eigenmode of the truck system occurs at 49.3 Hz, Fig. 10.
Fig. 9 The frequency variation of transmissibility from the engine to the cab for different values of the engine mounts stiffness.

Fig. 10 Twisting vibrational eigenmode of the truck system at 49.3 Hz.

6. Conclusion
The analytical model was validated by experimental studies.

The presented model is a useful tool for the vibration behavior analyses of the automotive in order to improve the passengers’ comfort.

The model allows the optimization study of the component mounting systems.

The influence on the vibration transmissibility of the main subassembly components of the whole system can be studied and improved. In order to find an optimal solution, the elastic and damping characteristics can be easily changed in the model.

The model is also useful in the body vibration behavior analyses during various working regimes as: acceleration, braking, passing over obstacle and turning on different types of road.

References: