EXHAUST MOUNT PLACEMENT OPTIMISATION WITH COMPARABLE METHODS

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Assessment of exhaust systems with virtual methods is very important to determine exhaust related NVH characteristic of road vehicles. In this study, exhaust system of a city bus is modeled virtually with finite element method including selected technical properties, like stiffness of flex-pipe, derived from specific tests. Firstly, a free-free normal mode analysis is conducted. The eigenvectors and eigenvalues are manipulated with a special method called ADDOFD for determination of number and location of exhaust hangers. Two different visualization procedures are used to show ADDOFD results and compare with each other. One is 1D model procedure which includes only selected points of exhaust system, the other is 3D graphical procedure which visualizes the results on whole model with a specifically written MATLAB code. After determination of hangers' specification, a constrained normal mode analysis is conducted to state modal status of the system and decouple exhaust modes with high effective mass from engine idle. With all these steps, connection details of an exhaust system have been defined mathematically and NVH status of the system has been determined.

1. Introduction

Development of technology affects automotive industry as new developed methods provide many opportunities to analyze factors that affect vehicle development and production phases. Important vehicle characteristics like vehicle dynamics, packaging, durability and NVH (noise, vibration & harshness) can be examined thoroughly before serial production and even prototype production with virtual analysis methods such as finite element method (FEM). Thus performance, working life and comfort level of vehicles have been improved by getting under control of essential problems and the affecting factors.

Exhaust system is exposed to vibration at high levels in vehicles with IC (internal combustion) engine because the system is connected directly to the engine. Because of the interaction with vehicle and engine, effects on both of them should be examined by extracting its NVH characteristic. This extraction is possible with virtual methods prior to prototype production.

Comprehensive studies are available about the subject in literature. Nefske et al. [1] and Englund et al. [2] verified an FE analysis of an exhaust system as 1D model. Fang et al. [3], Maruthi et al. [4] and Lupea [5] performed the correlation in 3D model. Furthermore Englund et al. [6] studied on effects of modelling flex-pipe on analysis of the system. Dwivedi et al. [7] and Gaonkar [8] carried on studies with different methods for locating exhaust hangers. Xu et al. [9], Zhien et al. [10], Maruthi et al. [11] and Wenzhu et al. [12] carried on similar studies by using ADDOFD (Average Driving Degree of Freedom Displacement) method and used 1D representation model to locate the hangers.

In this study, NVH characteristic of exhaust system of a city bus was determined and its effects to the vehicle were evaluated by examining its NVH analyses with FEM. After the definition of some important parameters of the system with tests and the completion of detailed system model, a free-free modal analysis was performed. Locations of exhaust hangers were determined by processing modal analysis' results analytically with ADDOFD method which calculates optimum number and location of exhaust hangers. ADDOFD results were visualized with two different methods. First process has already been used in industry: 1D representation model of the system was formed by selecting limited points on the model, the analyses' results were processed accordingly. As a second option, which is newly developed, whole model was analyzed and its results were processed by specifically written MATLAB code. Another modal analysis was performed by constraining the model at its hangers this time while test-specified stiffness' of flex-pipe and hanger isolators are used. By conducting all these steps, NVH characteristic of the system was defined and possibility of aligning system modes that have higher modal effective mass with important vehicle modes was examined.

2. ADDOFD Theory

ADDOFD method was used firstly to find locations of constraining points of a modal test model that minimally affect test results. The method locates the least deflecting points in a model. So it can be used to find optimum hanger placement locations of an exhaust system. In this way, exhaust system can be connected to body at locations where the least vibration occurs at a natural frequency.

Assuming the excitation is driven from a single point drive and considering modal analysis theory of multi-degree of freedom system, the response function between excitation point p and response point l is stated as Eq. (1) [13].

$$H_{lp}(\omega) = \sum_{r=l}^{N} \frac{\varphi_{lr}\varphi_{pr}}{M_r(\omega_r^2 - \omega^2) + j(2\zeta_r \omega \omega_r)} \tag{1}$$

In Eq. (1); φ_{lr} represents r order modal shape system of the l measuring point, M_r represents the modal mass, ω_r represents angular frequency of r order modal and ζ_r represents damping ratio of r order modal.

If the frequency of the excitation force is ω_r, then $H_{lp}(\omega)$ can approximate Eq. (2).

$$H_{lp}(\omega) \approx \sum_{r=l}^{N} \frac{\varphi_{lr}\varphi_{pr}}{j(2M_r \zeta_r \omega_r^2)} \tag{2}$$

For linear system, the amplitude of the displacement response is proportional to the amplitude of the frequency response function and can be defined as Eq. (3).

$$X(\omega_r) \propto H_{lp}(\omega) \approx \sum_{r=l}^{N} \frac{\varphi_{lr}\varphi_{pr}}{j(2M_r \zeta_r \omega_r^2)} \tag{3}$$

Equation (4) can be written, while the modal shape of the mass matrix is normalized and the mode damping is approximately equal.

$$X(\omega_r) \propto \frac{\varphi_{lr}\varphi_{pr}}{\omega_r^2} \tag{4}$$

To estimate the relative size of some degree of freedom's response displacement at a natural frequency, Eq. (5) gives ADDOFD value of the j degree of freedom [14]. In this equation, m represents number of modal order.

$$ADDOFD(j) = \sum_{r=1}^{m} \frac{\varphi_{jr}^2}{\omega_r^2} \tag{5}$$

Lower values of ADDOFD show the lower vibration response displacement. So locations of nodes with the lowest ADDOFD values are generally chosen as hanger placement locations.

3. Subsystem Tests

Exhaust system of a bus is usually placed inside the engine chamber where high temperature values are seen. So the system cannot be cooled with air that flows under the vehicle like the passenger cars. Operating temperature values, can rise up to 150 °C, are out of safe operating conditions for most of the rubber materials. Therefore, high temperature-resistant vibration isolators called wire-mesh are used, as seen in Fig. 1.

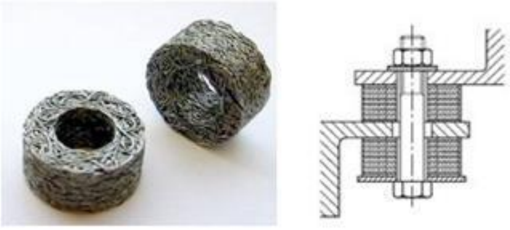

Figure 1: Wire mesh isolator.

Important stiffness values for modelling are determined with an MTS hydraulic test system [15]. At this system, tests have quasi-static characteristic and are performed at very low velocities, such as 0.5 mm/s for reducing dynamic interactions. Axial stiffness of the isolator increases while displacement increases, as expected and can be seen in Fig. 2. This situation can be interpreted that decrease of space between wires causes increase in stiffness progressively.

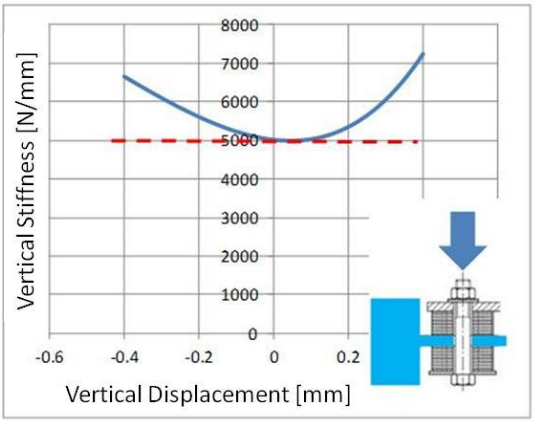

Figure 2: Vertical stiffness results of wire mesh isolator.

It can be seen at the axial stiffness results that 0.4 mm displacement results in 50% change in stiffness. Therefore, effect of assembly torque should be taken into account at the test of this isolator which has nonlinearity at high levels. The isolator should be tightened at assembly torque when it is fastened to the test system.

Unlike axial stiffness, increase in displacement results in decrease in stiffness seriously at radial stiffness results, as seen in Fig. 3, because of increasement in space between wires in radial direction.

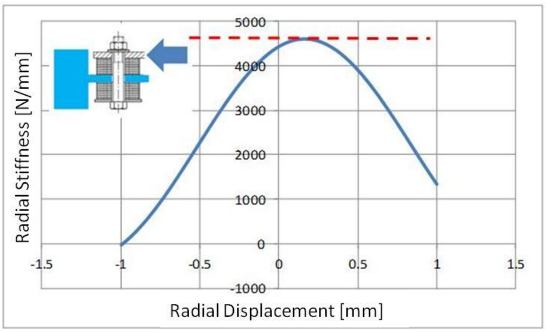

Figure 3: Radial stiffness results of wire mesh isolator.

Flex-pipe is used at exhaust systems to prevent the transmission of engine induced vibrations to the rest of the system and prevent possible fractures. Flex-pipe operates at both radial and axial directions. Radial and axial stiffness of the flex-pipe were obtained by connecting a known mass at end of the pipe at a chuck and then measuring the resulting displacement with a calliper, as seen in Fig. 4. The results are shown at Table 1.

Figure 4: Radial stiffness measurement for flex-pipe.

Table 1: Flex-pipe stiffness values.

Stiffness Measurement	Value [N/mm]
Axial	13.4
Radial	5.5

4. Modelling

Exhaust system was modelled using 1D, 2D and 3D elements at a commercial FE software, according to their usage in analyses. 2D (mostly quadrilateral) elements were used in pipe and plate components which compose most of the model. 3D (hexahedral) elements were preferred in welds used between pipes as their mass is significant in respect to the total system behaviour. 1D (rigid) elements were used in connecting profiles and brackets. Each wire mesh isolator was modelled as non-dimensional spring elements and a stiffness value for each translational axis was assigned to each of these elements. Mass of the engine was defined as a mass element and a couple of body-side connection profiles/brackets were also added to the model. The model consists of 75902 elements and is seen at Fig. 5.

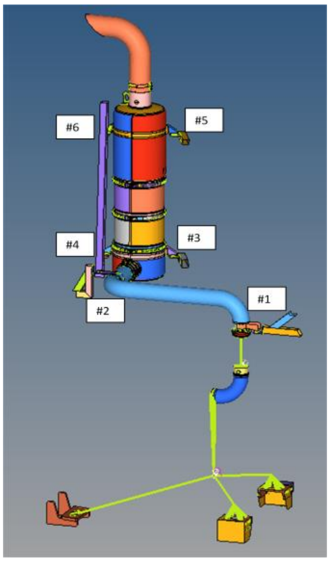

Figure 5: FE model of the exhaust system.

Flex-pipe is a special component which has high flexibility to reduce the transfer of engine induced vibrations through the exhaust line. Therefore, it is important to correctly model the flex-pipe. Analysis results of three different modelling methods, found in literature [6] or hypotized by authors, were compared with test results, given in Table 1. In first method, spring elements with different lengths and stiffness values were used throughout length of the flex-pipe. In second method, shell elements with three different material parameters were used. In last method, the flex-pipe was modelled as three spring elements, in which a stiffness value for each translational axis was assigned. All methods have advantages and disadvantages with regards to each other. Third method was preferred in this study because the main aim is only to extract NVH characteristic and modelling is much easier. However other methods are recommended for different analysis disciplines like static and fatigue.

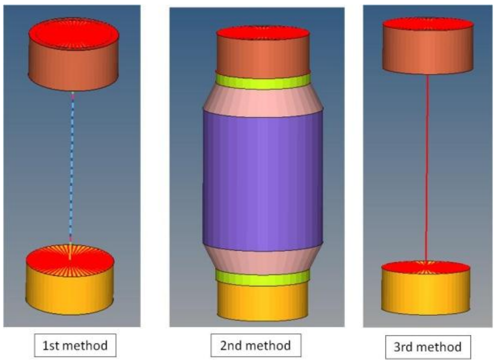

Figure 6: Flex-pipe modelling methods.

Specifically for 1D representation way of ADDOFD method, 1D plot elements were formed through the centre of the exhaust line, each has specific length and connected to the system with rigid elements. In Fig. 7, these plot elements can be seen as blue colour. In this way, modal results can be shown only via these elements and are processed with ADDOFD method easily.

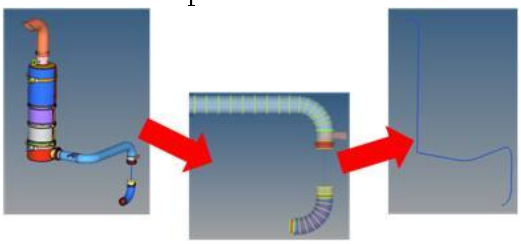

Figure 7: 1D plot elements for 1D representation of ADDOFD method.

5. FE Analyses and Results

5.1 Free-free modal analysis and ADDOFD method

A free-free modal analysis was performed on the model, from which flex-pipe, engine and engine mounts were extracted. Flex-pipe is assumed to have adequate flexibility for decoupling engine and the exhaust system. Modal shapes for using in ADDOFD calculation were evaluated by going through modal effective mass parameter which shows importance level of each mode.

As seen in Fig. 8, modal effective mass was calculated for only modes above firing frequency of the engine. Modes under this frequency are excited only for a short time during engine start-up and do not cause steady vibration problem. Modes which have high modal effective mass are marked in Fig. 8.

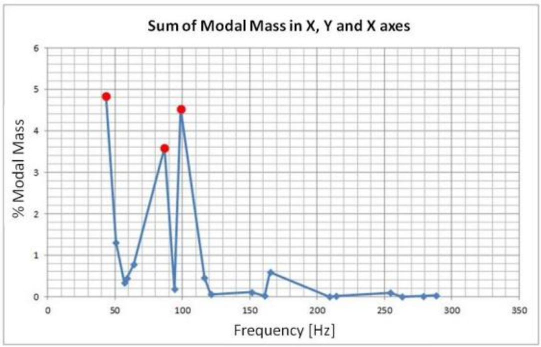

Figure 8: % Modal mass (X, Y and Z axes are summed).

5.1.1 1D representation of ADDOFD method

Using 1D plot elements shown in Fig. 7, ADDOFD calculation was processed for each node of plot elements. Because the number of nodes is low (50 in this study), it is easy to calculate ADDOFD value for each node. A very basic MATLAB algorithm is used for this calculation. Results are shown as a node number-ADDOFD value graph, as seen in Fig. 9. The anti-peak points in the graph can be defined as possible hanger locations.

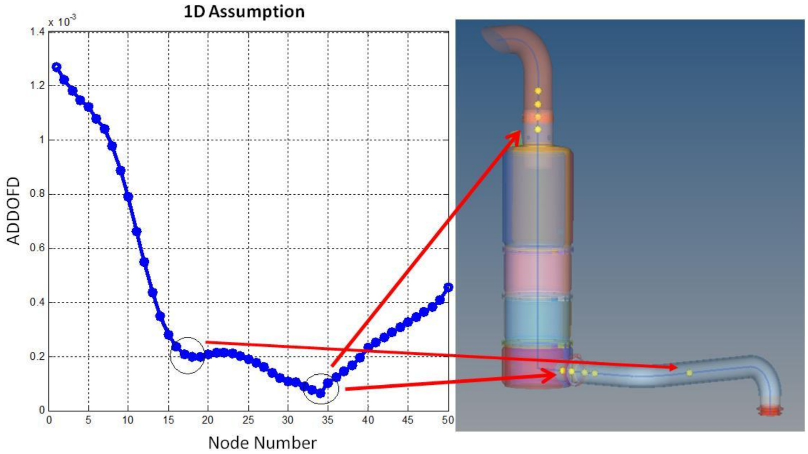

Figure 9: Node number-ADDOFD value graph and its representation on exhaust model.

5.1.2 3D representation of ADDOFD method

1D representation has generally been used because the computation and representation of whole data of the model is very expensive and complicated. 1D representation easies the computation by its lower numbers and the results can be shown easily by a simple graph, as shown in Fig. 9. But reducing 3D model into 1D results in misinterpreting results. Because reducing a whole cylindrical section of exhaust pipe into a point prevents the detection of local deflections at this section, as mentioned in the study of Nefske et al. [1]. 1D representation can show only an average value for whole section.

Because the number of nodes is very high (45376 in this study), a specific MATLAB algorithm needs to be written. In this algorithm, ADDOFD value for each node was calculated and the results were shown on the 3D model by matching node numbers with their own ADDOFD value, as shown in Fig. 10.

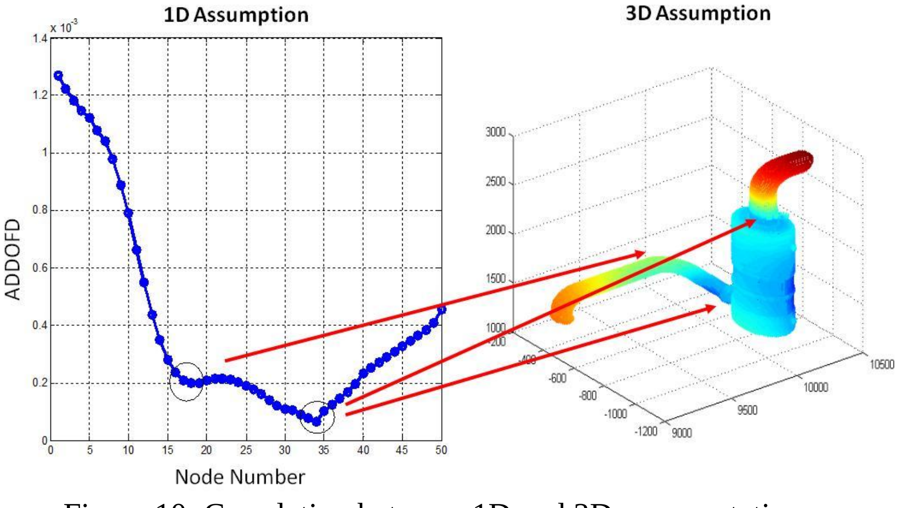

Figure 10: Correlation between 1D and 3D representations.

Comparison between 1D and 3D representations shows that

1. There is a good correlation in optimum mounting locations which is shown in Fig. 10.

2. Optimum hanger locations are also shown in circumferential directions in Fig. 10. For instance, there is a location just before the catalyst at both representations. 1D representation shows this as a point without showing the location on the circumference. However 3D representation shows clearly that hanger should be under the pipe. This is the main advantage of 3D representation.

Final hanger locations of exhaust system are determined with the help of optimisation results and BIW structure.

5.2 Constrained modal analysis

Another modal analysis by constraining full model of the exhaust system from the determined hanger locations was performed to obtain modal status of the system. Possible mode alignments and NVH problems were investigated by going through important modes and their mode shapes. The most noteworthy point in the evaluation of this analysis is to provide at least 10% separation of the most important frequency of the system from the firing frequency of the engine at idle operation (35 Hz). This frequency is 43.3 Hz at the system and it is above the threshold frequency, 39 Hz. It is seen in Fig. 12 that the modal shape is dominated by flex-pipe and flex-pipe has enough flexibility for decoupling the engine and exhaust system.

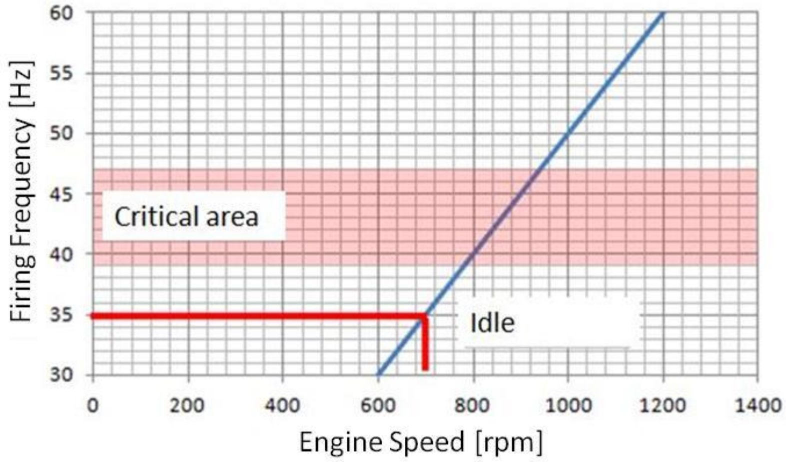

Figure 11: Modal alignment chart for exhaust system.

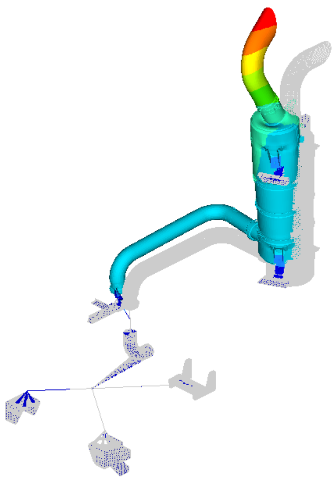

Figure 12: Modal shape of the system at 43.3 Hz.

6. Conclusion

This study summarizes virtual analyses of the exhaust system of a city bus to identify and prevent possible exhaust induced NVH problems by determining optimum exhaust hangers' locations. Firstly finite element model of the system was formed and some tests were conducted for generating the required data. To determine optimum hanger placement locations that transmit the lowest vibration between the body and the system, a free-free modal analysis was performed and the results were processed by ADDOFD method. Along with the traditional 1D representation of ADDOFD 3D representation, having an advantage to define hanger location with resolution in radial directions, was introduced. According to this study and suitability of BIW structure, hangers of the exhaust system were determined. A modal analysis by constraining the determined hanger locations was performed to obtain modal characteristic of the system.

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