Significance of non-linearity and component-internal vibrations in an exhaust system

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Abstract: - To facilitate overall lay-out optimisation inexpensive dynamics simulation of automobile exhaust systems is desired. Identification of possible non-linearity as well as finding simplified component models is then important. A flexible joint is used between the manifold and the catalyst to allow for the motion of the engine and to reduce the transmission of vibrations to the rest of the exhaust system. This joint is significantly non-linear due to internal friction, which makes some kind of non-linear analysis necessary for the complete exhaust system. To investigate the significance of non-linearity and internal vibrations of other components a theoretical and experimental modal analysis of the part of a typical exhaust system that is downstream the flexible joint is performed. It is shown that non-linearity in this part is negligible. It is also shown that shell vibrations of the catalyst and mufflers as well as ovalling of the pipes are negligible in the frequency interval of interest. The results implies, for further dynamics studies, that the complete system could be idealised into a linear sub-system that is excited via the non-linear flexible joint, that the pipes could be modelled with beam elements and that the other components within the linear sub-system could also be modelled in a simplified way. Such simplified component models are suggested. The agreement between theoretical and experimental results is very good, which indicates the validity of the simplified modelling.

Key-Words: - Correlation; Dynamics; Exhaust system; Linear sub-system; Modal analysis; Non-linear joint

1 Introduction

There is a trend to use more computer simulations in the design of products. This is mainly due to demands on shortened time to market, higher product performance and greater product complexity. To be useful in the design process it is important that the simulation models are kept as simple as possible while still being accurate enough for the characteristics they are supposed to describe. To reveal weaknesses in the simulation models experimental investigation is often necessary. The simulation models can then be updated to better correlate with experimental results.

This study is a part of a co-operation project between the Department of Mechanical Engineering at the Blekinge Institute of Technology, Karlskrona, Sweden and Faurecia Exhaust Systems, Inc., Torsås, Sweden. The overall aim of the project is to find a procedure for effectively modelling and simulating the dynamics of customer-proposed exhaust system lay-outs at an early stage in the product development process, to support the dialogue with the costumer and for overall lay-out optimisation. Demands on, for example, higher combustion temperatures, reduced emissions, reduced weight, increased riding comfort and improved structural durability have made the design of exhaust systems more delicate over the years and more systematic methods have become necessary.

Examples of linear studies of exhaust systems are the works by Belingardi and Leonti [1] and Ling et al. [2], who focus on simulation models, and the work by Verboven et al. [3] who focus on experimental analysis. An introductory study of the present exhaust system is that of Myrén and Olsson [4].

Most modern cars have the engine mounted in the transverse direction. A flexible joint between the manifold and the rest of the exhaust system is therefore included to allow for the motion of the engine and to reduce the transmission of vibrations to the rest of the exhaust system. Recent suggestions of a stiffer attachment of the exhaust system to the chassis, as discussed by for example DeGaspari [5], with the purpose of reducing weight, makes this joint even more important. The commonly used type of joint is significantly non-linear due to internal friction, which makes some kind of non-linear dynamics analysis necessary for the complete system.

Thus a more comprehensive approach seems necessary. This paper represents an early step and focuses on the part of the exhaust system that is downstream the flexible joint. The purpose is to verify the assumption that this part is essentially linear so that, in the further studies, the complete system could be idealised into a linear sub-system that is excited via the non-linear flexible joint. The purpose is also to find a computationally effective and experimentally verified finite element (FE) model of this linear sub-system. This includes simplified modelling of the components.

2 Exhaust system design and excitation

The studied automobile exhaust system is shown in figure 1. The mass of the system is about 22 kg and it has a length of approximately 3.3 m.

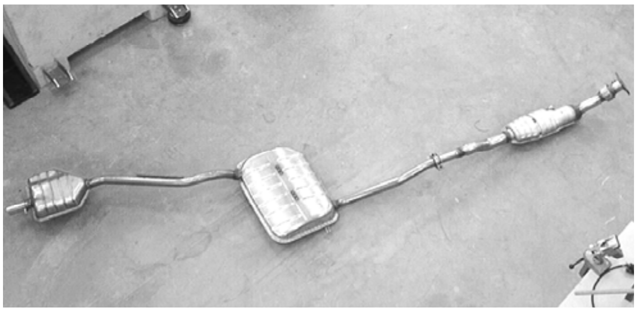

Fig. 1. The studied exhaust system.

The system consists of a front assembly and a rear assembly connected with a sleeve joint. Both are welded structures of stainless steel. The front part is attached to the manifold by a connection flange. The engine and manifold are not included in the study.

Between the manifold and the catalyst there is a flexible joint, consisting of a bellows expansion joint combined with an inside liner and an outside braid. This joint is significantly non-linear due to internal friction. More information on this type of joint is given by, for example, Cunningham et al. [6] and Broman et al. [7].

The front assembly, see figure 2, consists of this joint, the catalyst and pipes.

Fig. 2. Front assembly.

The catalyst includes a honeycomb ceramic and the outside shell structure is rather complicated. Thus, detailed modelling would be computationally expensive.

The rear assembly, see figure 3, consists of pipes, an intermediate muffler and a rear muffler.

Fig. 3. Rear assembly.

Perforated pipes pass through the mufflers. The mufflers are filled with sound silencing material. Their outside shell structure is also rather complicated.

Besides the connection to the manifold the exhaust system is attached to the chassis of the car by rubber hangers. Two hanger attachments are placed at the intermediate muffler and a third is placed just downstream the rear muffler.

The frequency interval of interest for the modal analysis is obtained by considering that a four-stroke engine with four cylinders gives its main excitation at a frequency of twice the rotational frequency. Usually the rotational speed is below 6000 rpm. Excitation at low frequencies may arise due to road irregularities, as discussed by, for example, Belangardi and Leonti [1] and Verboven et al. [3]. Thus, the interval is set to 0 – 200 Hz.

Free-free boundary conditions are generally desired to facilitate a comparison between the FE-results and the experimental results. This also makes it possible to easily exclude the influence of the non-linear joint in the present analysis. It is assured that the flexible joint does not have any internal deformations. Thus it will move as a rigid body in the present analysis.

3 Initial finite element model

An initial FE-model of the exhaust system is built in I-DEAS [8]. The outside shell structure of the mufflers and the catalyst are modelled with linear quadrilateral shell elements using the CAD-geometry. The mass of the internal material is distributed evenly to the shell elements. The pipes are modelled using parabolic beam elements. The flexible joint is modelled by stiff beam elements with a fictive density to reflect its mass and mass moment of inertia. Lumped mass elements are used to model the connection flange, attachments for the hangers, nipples and the heat shield. Connection between the beam elements representing the pipes and the shell elements representing the mufflers/catalyst is obtained by rigid elements.

By comparing different mesh densities it is found that approximately 140 beam elements and 1900 shell elements are sufficient. The total number of nodes are approximately 2200. This initial model is used as a basis for determining suitable transducer locations for the experimental modal analysis of the exhaust system.

The natural frequencies are solved for by the Lanczos method with free-free boundary conditions.

4 Experimental modal analysis

To sufficiently realise the free-free boundary conditions in the experimental modal analysis (EMA) the exhaust system is suspended, at the hanger attachments and at the connection flange, using soft adjustable rubber bands as shown in figure 4.

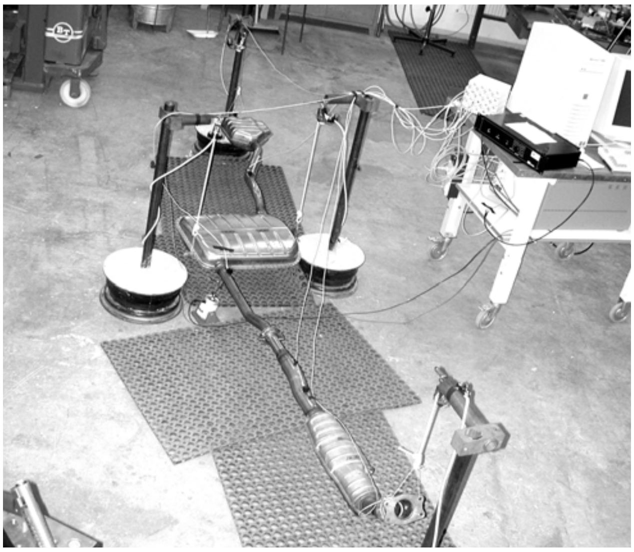

Fig. 4. The measurement set-up.

From the initial FE-analysis it is known that the motion is mainly in the plane (y-z) perpendicular to the length-direction (x) of the system. To be able to excite the system in both the y- and z- directions in one set-up the shaker is inclined. After consulting the FE-model several possible excitation points are tested. The final excitation point is taken just upstream the intermediate muffler, as seen in figure 4. The shaker is connected to the exhaust system via a stinger and a force transducer. A burst random signal is used to excite the exhaust system to avoid possible leakage problems. An HP VXI measuring system with 16 available channels is used. Five triaxial accelerometers could therefore be used in each measuring round. The accelerometers are attached on top of the exhaust system. To minimise the influence of the extra mass loading the accelerometers are evenly spread over the exhaust system in each measuring round.

Again considering the results from the initial FE-model it is concluded that 25 evenly distributed measuring points should be sufficient to represent the mode shapes in the frequency interval of interest. Using the AutoMAC, see figure 5, the chosen measurement points are checked to avoid spatial aliasing. The small off-diagonal terms in the AutoMAC indicate that the chosen measurement points sufficiently well describe the modes in the frequency interval of interest.

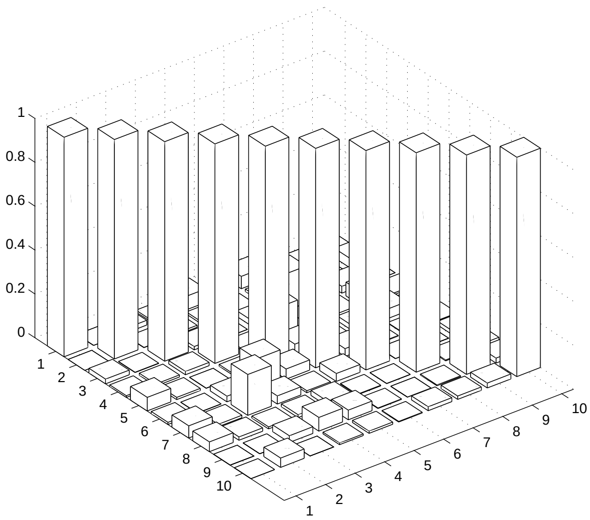

Fig. 5. The AutoMAC-matrix.

The quality of the experimental set-up is further assured by a linearity check, a reciprocity check and by investigating the driving point frequency response function (FRF). Also the coherence of some arbitrary FRFs is investigated. All the quality checks show satisfactory results.

Due to the long and slender geometry of the exhaust system concerns may arise that the static preload could have an undesired influence when the system hang horizontally. To ensure that this is not the case the exhaust system is also hanged vertically and some arbitrary FRFs are measured. The difference in natural frequencies is negligible between the two set-ups and it is therefore concluded that the initial set-up is satisfactory.

I-DEAS Test [9] is used to acquire the FRFs. The FRFs are exported to MATLAB [10] where they are analysed using the experimental structural dynamics toolbox developed by Saven Edutech AB [11]. The poles are extracted in the time domain using a global least square complex exponential method. The residues are found using a least squares frequency domain method. To improve the quality of the extracted modal parameters only data in the *y*- and *z*-directions are used. To get as good a fit as possible the curve fitting procedure is conducted in two steps; first in the interval 5-90 Hz and then in the interval 90-150 Hz. Above 150 Hz no significant modes are found, as seen in a typical FRF shown in figure 6.

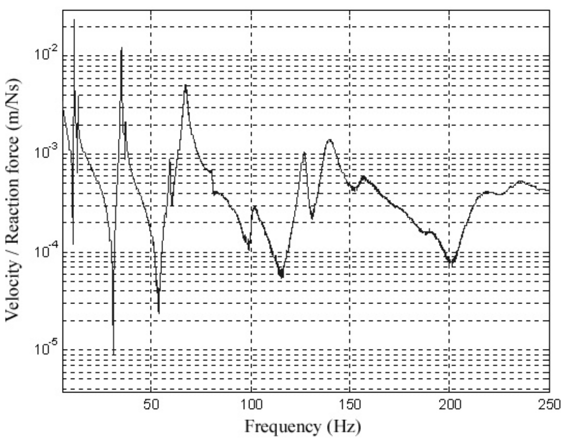

Fig. 6. Typical FRF.

5 Simplification and correlation

Determining the natural frequencies of the mufflers and the catalyst experimentally it is seen that no significant local modes are present in the frequency interval of interest. This was also found by Verboven et al. [3]. Therefore the modelling of the mufflers and the catalyst, which are responsible for most of the model size in the initial FE-model, can be significantly simplified. The mufflers and the catalyst are modelled by lumped mass and mass moment of inertia elements. The properties of these elements are obtained from the original FE-model. If more suitable in a general case these properties can also be obtained directly from the CAD-model or experimentally. The lumped mass and mass moment of inertia elements are connected to the beam elements representing the pipes by rigid elements.

The natural frequencies of the pipes are also investigated experimentally. No significant ovalling modes are found in the frequency interval of interest, which confirms the validity of modelling the pipes by beam elements.

To simulate the flexibility of the connections between the pipes and mufflers/catalyst, short beam elements with individual properties are used. These elements are located at the true connection locations, that is, with reference to the real system. Thus, they are placed between the rigid elements that are connected to the lumped mass and mass moment of inertia elements and the beam elements representing the pipes.

These individual beam properties are updated so that the difference between theoretical and experimental results is minimised. The updating procedure uses MATLAB [10] and ABAQUS [12] and is described in an accompanying paper (Englund et al. [13]).

The updated FE-model has approximately 200 nodes. Thus a reduction of over 90 % compared to the initial FE-model is obtained. Simplifications of this type are important if direct time integration becomes necessary for the non-linear dynamics analysis of the complete system. It is also important when a large number of simulations are necessary for overall exhaust system lay-out optimisation.

The FE modes are calculated without consideration of damping and are therefore real-valued. To be able to compare these modes with the modes obtained experimentally, which are complex due to damping, the experimental modes are converted into real-valued modes.

6 Results and correlation

To correlate the mode shapes from the updated FE-model and the experimental mode shapes a MAC-matrix is calculated, see figure 7.

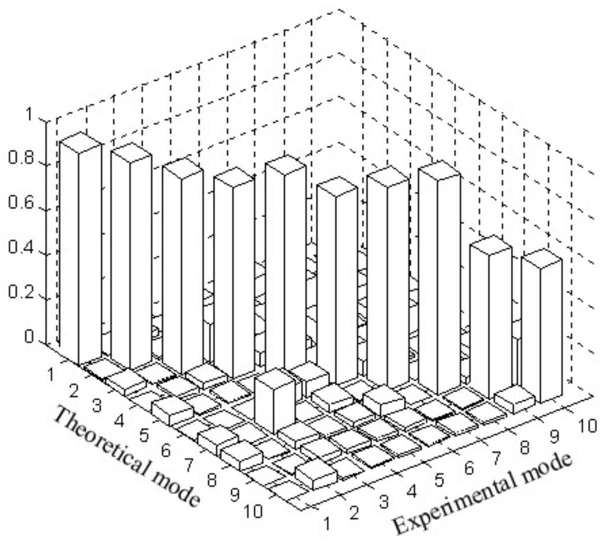

Fig. 7. The MAC-matrix.

Except for mode nine and ten the diagonal MAC-values are above 0.85, which indicates good correlation. All the off-diagonal values in the MAC-matrix are below 0.2. This indicates that the different mode shapes are non-correlated.

A comparison between theoretical and experimental natural frequencies is shown in figure 8. The 45-degree line represents perfect matching. The crosses indicate the frequency match for each correlated mode pair.

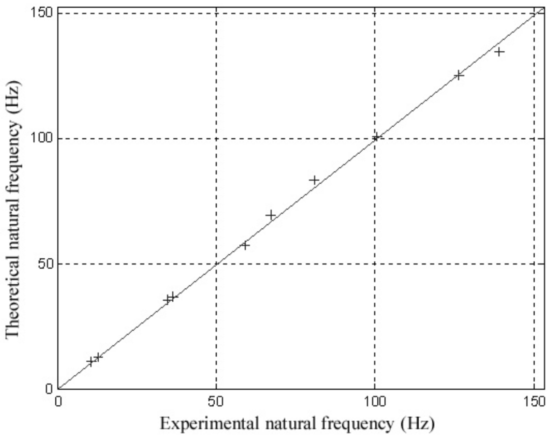

Fig. 8. Theoretical and experimental natural frequencies.

The maximum difference in corresponding natural frequencies is below four per cent. The small and randomly distributed scatter of the plotted points is normal for this type of modelling and measurement process [14].

The results are summarised in table 1.

Table 1. Results.

Mode	Experimental		Theoretical	Correlation[a] (%)	MAC
	Frequency (Hz)	Damping (%)	Frequency (Hz)		
1	10.9	0.32	10.9	0.24	0.95
2	12.9	0.52	12.8	-1.0	0.93
3	34.9	0.49	35.8	2.6	0.88
4	36.4	0.30	36.9	1.3	0.85
5	59.1	0.69	57.3	-3.0	0.93
6	67.1	1.5	69.7	3.9	0.85
7	80.8	0.79	83.7	3.6	0.91
8	101	1.6	101	0.30	0.96
9	127	0.91	126	-0.60	0.64
10	139	2.3	135	-2.9	0.60

[a] The correlations are calculated before rounding off.

The damping values are given as the fraction of critical damping and the correlation value is the relative difference between experimental and theoretical natural frequencies. Above 150 Hz no significant modes are found.

7 Conclusions

A dynamics study of an exhaust system that consists of a non-linear flexible joint and a main part including pipes, mufflers and a catalyst is presented. The good agreement between the theoretical and experimental modal analysis, as well as the satisfactory results of the linearity check, implies, for further dynamics studies, that the complete system could be idealised into a linear sub-system that is excited via the non-linear flexible joint.

It is also shown that shell vibrations of the catalyst and mufflers as well as ovalling of the pipes are negligible in the frequency interval of interest. This implies that the pipes could be modelled by beam elements and that the other components within the linear sub-system could be modelled by lumped mass and mass moment of inertia elements. The mass and inertia properties can be obtained either from a CAD-model or experimentally. Short beam elements with individual properties can be used successfully to model the flexibility of the connections between the mufflers/catalyst and the pipes. Automated updating of these individual properties is recommended since doing it manually is time consuming and difficult.

The agreement between results from the updated FE-model and the experimental investigations is very good. This implies that such simplified modelling is a valid approach and it may turn out important in coming non-linear analyses, since such analyses are often computationally expensive.

8 Acknowledgements

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