Engine induced vibrations

by

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ABSTRACT

Existing Finite Element models of chassis, cab, engine and gearbox have been merged into a complete full vehicle model by use of a superelement technique in CSA/Nastran version 93C. The superelement technique is used to reduce the degrees of freedom. Additional information about the internal nodes is achieved with component mode synthesis. To examine the engine induced vibrations and its transmission paths, a frequency response analyses is performed with loads for the different engine orders. The developed full vehicle model is a good tool to examine the transmission paths and to do fast parameter studies.
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1. Introduction

For front end dynamics ([1]), caused by excitation from wheel run-out and unbalance as well as from brake excitation and vertical road input, finite element models of frame and chassis have already been used to examine the dynamic behavior of the truck. For low frequencies, the cab and engine were modelled as rigid bodies. For higher frequencies, like engine induced vibrations, it was desirable to have finite element models of the cab and the engine in particular.

First the different finite element models used in this project will be described. The models of the cab, frame and engine were developed with respect to stress analyses. The models are therefore much too detailed for efficient analysis of vehicle dynamics. In these studies we are only interested in the mass- and stiffness-distribution within a certain frequency range. And not in the stress concentration in the different parts of the models. This goes in particular for the cab and the frame. For the engine some more details are requested, because the engine is excited by dynamic loads in the cylinders and the bearings. The response of the engine itself is used to excite the rest of the truck.

The superelement technique is used to reduce the number of degrees of freedom and thus to decrease the calculation time. Some background theory about reducing the degrees of freedom will be explained in chapter three. Next the model assumptions are discussed, while in chapter five the load case is introduced. These load case is used to examine the engine induced vibrations and the transmission paths.

2. Models of substructures

The full model consists of 24,157 elements and 23,039 nodes. The most common elements used in the several models are quadrilateral elements. Two types of these elements are used:

- **Quad4**: An isoparametric quadrilateral element with 4 nodes, which has membrane, bending and coupled membrane-bending stiffness. So there is stiffness in all 3 displacements and in the two out of plane rotations.
- **Tria3**: Same type as the Quad4, but with only 3 nodes per element.

The models also consist of isoparametric solid hexahedron elements with 8 up to 20 nodes. Because the truck is in general symmetric, only one half of the complete vehicle is modelled. The engine, with six cylinders in row causes anti-symmetric load cases, which are merely responsible for the high frequent vibrations in the cab. Therefore anti-symmetric boundary conditions are used in the plane of symmetry, which is the xz-plane. The direction of the global coordinate system can be seen in Figure 2.1.

2.1 Frame

The frame is built up by two siderails connected to each other by five cross-members and consists of the following elements and nodes:

- 3898 Quad4
- 38 Tria3
- 164 rigid bars
4464 interior nodes
15 exterior nodes

After a dynamic reduction, between 0.0 and 400.0 Hz, 32 generalized coordinates have been generated. When all exterior nodes are constrained (fixed eigenmodes), the frame has its lowest eigenfrequency at 80.1 Hz.

2.2 Engine

The engine is a 6 cylinder engine used in the new FH-series. It is a detailed model including the 6 cylinders, 7 bearings, gearbox, fan and even the dip-stick. However, the piston and piston rod have not been modelled. The model contains the following elements and nodes:

4431 Quad4
832 Tria3
334 Beams
48 Hexa
49 Conm2, to specify rigid mass.
412 rigid bars
5033 interior nodes
21 exterior nodes. 12 nodes for specifying the dynamic loads in the cylinders, 7 nodes for specifying the dynamic reaction forces in the bearings. 2 exterior points which connect the engine with the engine mounts. There are two engine mounts, each modelled by 3 linear translational springs and 3 linear translational viscous dampers. The front engine suspension is between exterior node 357201 and node 2500. The rear engine suspension is situated between exterior node 357101 and node 3901.

After a dynamic reduction 29 generalized coordinates have been generated.

To merge the engine into the full model, the engine had to be rotated with an angle of four degrees around the y-axis. After this was done, it occurred that there is an offset between the rear engine mount and the frame:

\[
\begin{align*}
x_{\text{offset}} &= 1.36 \text{ [mm]} \\
y_{\text{offset}} &= 85.15 \text{ [mm]} \\
z_{\text{offset}} &= 2.64 \text{ [mm]}
\end{align*}
\]

The offset in lateral direction is unacceptable large. This is corrected by defining a rigid bar between node 3901 and 357101, which is stiff connected to node 357101 in all 6 directions. This can be done because node 357101 itself is stiff connected to the engine by rigid bars.

For low frequencies the engine moves as a rigid body. Therefore it is important that the mechanical properties of the engine are comparable with the measured values. The mass and the moments of inertia have been calculated by means of Nastran. It turned out that the mechanical properties of the FE-model are comparable with the measured values.
2.3 Cab

A model of the L2H1 cab is used and contains the following elements:

- 12392 Quad4
- 878 Tria3
- 11 Beams
- 348 Hexa
- 122 Comm2
- 144 RBE2
- 13356 interior points
- 3 exterior points.

The bushing in front of the cab is modelled by three linear translational springs and three linear viscous dampers between exterior node 420.003 and 96.011. The bushing is mainly for lateral and longitudinal forces. The front cab suspension is only for vertical displacements. This is modelled by a linear spring and a linear viscous damper in vertical direction, acting between exterior node 420.001 and node 2501. The rear cab suspension is modelled by a vertical linear spring, a vertical linear viscous damper and a lateral linear viscous damper between exterior node 420.002 and node 4901.

The roll bar in front of the cab is modelled by beam elements between node 96.011, node 96003 (= node 2500, only rotational freedom in 5 direction) and node 96012.

A dynamic reduction generates 584 generalized coordinates.

To avoid modelling the plastics of the interior of the cab the specified density of the material is higher, than the density of steel, to take into account these additional masses of the plastics.

2.4 Residual

The residual contains the tyre models, steering gear, trailer, leaf suspensions between frame and axles and the suspensions mentioned above. This is done to make it easier to calculate different designs of the suspensions.

3. Reducing the degrees of freedom

Superelement analysis is used to reduce the number of degrees of freedom of a large structure. The structure is divided in substructures called superelements. Each superelement is connected to the residual in the exterior nodes. First the general form of this reduction will be described. The equations of motion for any superelement with \( n_G \) degrees of freedom is as follow:

\[
M_{GG} \ddot{U} + B_{GG} \dot{U} + K_{GG} U = F \ (i)
\]

In here is:
- \( U \): global degrees of freedom of a superelement, size \( n_G \times 1 \).
\[ M_{GG} \]: global mass matrix.
\[ K_{GG} \]: global stiffness matrix.
\[ B_{GG} \]: global dampings matrix.
\[ F(t) \]: exterior forces.

Reduction of the \( n_G \) degrees of freedom is possible by describing the global degrees of freedom \( \Upsilon \) as a linear combination of \( n_A \) generalized coordinates \( \eta \), whereas \( n_A < n_G \):

\[ \Upsilon = T \times \eta \Rightarrow T = \text{transformation matrix} (n_G \times n_A) \]  

(2)

Substituting (3) in (2) gives:

\[ M_{AA} \ddot{\eta} + B_{AA} \dot{\eta} + K_{AA} \eta = F_{AA}(t) \]

(3)

with:

\[ M_{AA} = T^T M_{GG} T : \text{reduced mass matrix.} \]
\[ K_{AA} = T^T K_{GG} T : \text{reduced stiffness matrix.} \]
\[ B_{AA} = T^T B_{GG} T : \text{reduced dampings matrix} \]
\[ F_{AA}(t) = T^T F(t) : \text{reduces forces} \]

The degrees of freedom of a superelement are now reduced to the analyses set, which contains a total number of degrees of freedom that is equal to the number of generalized coordinates. The transformation matrix can be developed in several ways.

In static analyses all superelements are reduced to the exterior nodes. The stiffness of a superelement can be described completely in those nodes. There is no loss of accuracy. But this is not accurate enough for dynamic analyses, because the number of boundary points is not enough to represent the eigenmodes. For dynamic analyses it is important that the reduced structure has a sufficient accurate dynamic behavior similar to the full model within a specified frequency range. A suitable method is the Component Mode Synthesis technique. CSA/Nastran uses the Craig-Bampton method. This method involves an eigenvalue extraction of each Superelement and assumes all exterior points to be fixed in all directions. For each extracted eigenvalue one generalized coordinate is added to the reduced set of degrees of freedoms. These generalized coordinates represent the eigenmodes of the superelement. In [2] it is explained how the transformation matrix \( T \) looks like for the Craig-Bampton method.

Because this method is an approximation, the problem is how many eigenmodes must be taken into account for Component Mode Synthesis in order to have a sufficient accurate dynamic behavior. As a rule of thumb, for each Superelement, all eigenvalues up to twice the highest frequency of interest for the complete structure, should be extracted. For the dynamic reduction all eigenmodes between 0 and 400 Hz have been taken into account. This made it possible to reduce the number of degrees of freedoms from 140,000 to nearly 2000.

The stiffness matrix of the residual is made by assembling all the reduced matrices of the superelements. The stiffness matrix of the residual is in general not sparse anymore, so CSA/Nastran will not use the sparse solver. For static problems the program uses the Cholesky decomposition which is much slower. Therefore it is important to have only a few grid points in the residual.
Remarks:

- Since the superelement technique is based on superposition, it is important that all the Superelements have linear conditions.
- In many cases it is enough to specify the exterior nodes only in the connection points of two substructures. Unless dynamic forces are acting on some nodes in a superelement. Then these nodes must be exterior nodes, so they are belonging to the residual. The reason for this is that when doing a reduction of a superelement, not only the mass and stiffness matrices are reduced, but also the loadvector. They must all be independent of the frequency, which is not the case for dynamic forces. Only static forces may act on the interior nodes of a superelement.
- A further reduction of the full model can be achieved by solving the real eigenmodes and using only modal data in the frequency range of interest for analyses of the frequency response.

4. Model assumptions

- The model is linear, thus only small deformations allowed
- As a truck is in general symmetric, only one half of the truck has been modelled.
- Only anti-symmetric loadcases, thus anti-symmetric boundary conditions are adapted on nodes in the plane of symmetry.
- For the dynamic reduction of every superelement all eigenmodes between 0.0 and 400.0 Hz have been taken into account. Therefore this model is only valid for frequencies up to 200.0 Hz.
- The geometry of the nodes applies to the loaded truck condition. The springs are in fact in a compressed state. If the mass of the truck is changed (unloaded), then in reality the geometry changes, but here the same geometry is considered.
- Only a vertical tyre model, modelled by a vertical spring for every wheel. No longitudinal and lateral tyre model, because these models are only important for low frequencies up to 10.0 Hz.
- No centrifugal and gyroscopic forces.
- Forces in the engine are calculated by using the pressure in the cylinder necessary to meet the euro2-legislation.

5. Loadcase

The dynamic response of the model is studied for different engine orders. The dynamic loads in the cylinders and the bearings have been calculated by the engine department. Because mainly
the torque in the engine is responsible for the excitation of the cab, only the anti-symmetric loadcase is used, which means that there are only forces acting in lateral direction. The dynamic loads are applied on one node in every bearing and on two nodes in every cylinder. In Figure 5.1 the magnitude of the forces for the third order can be seen. The magnitude and phase of the forces in each cylinder are the same.

6. Analyses

Since the superelement technique is an approximation, a real eigenvalue analyses has been used to compare the results of the model with and without a dynamic reduction. Only the first twenty eigenfrequencies have been calculated and they are exactly the same. The eigenmode at 4.2 Hz. however showed some differences:

- Exterior nodes: difference less than 0.01 %
- Interior nodes: frame: 0.01%
- engine max 0.4 %
- cab max 2.0 %

Off course the approximation will become worse when the frequency is close to 200.0 Herz. This model is however used to determine the direction in which the response alters due to different modifications of the structure.

During vibration measurements in the cab, the measurements showed an increasing acceleration level when the engine rotation is decreased towards 8.0 rotations per second. Due to cut out of the engine the peak has never been measured. This peak is expected to be in the range of 15.0 to 20.0 Hz. For this model the first eigenmodes however occurred around 11.5 Hz. The static stiffness of the engine mounts was not high enough. To get the first eigenmode, where the engine is rolling as a rigid body, in a realistic frequency range, the stiffness and damping values of the engine mounts have been increased to include dynamic stiffening effects found in rubber components.

The response of the nodes in the cab shows a lot of undamped eigenfrequencies, because there is no structural damping in the models. For nodes in the engine there is little distortion. But for the cab suspension the response is pretty distort due to a lot of undamped eigenfrequencies of the frame and the cab. The magnitude of this peaks obtained by a frequency response analyses are very sensitive to the size of each frequency increment. When parameter studies are done, some peaks could shift in the frequency. Peaks might disappear or new peaks might occur if the resolution in frequency is low.

One way to overcome this is to use very small frequency increments, which will result in a very large computation time. Another way to avoid this problem is by adding some material damping to the model. The peaks will decrease and become wider. Which means that the peaks are less sensitive to the size of the frequency increments. It also gives a clearer view about the most important eigenfrequencies.
In this model no well defined material damping mechanism exists. But with modal damping it is possible to include damping after the eigenvectors have been used to transform the equations of motion to the uncoupled single degree of freedom form. The uncoupled equations of motion for a single degree of freedom are:

\[ m_i \ddot{u}_i + b_i \dot{u}_i + k_i u_i = f_i \]  

(4)

Which can be written in non dimensional form as

\[ \ddot{u}_i + 2 \omega \zeta i \dot{u}_i + \omega^2 u_i = \frac{f_i}{m_i} \]  

(5)

where \( \zeta \) is the damping factor

\[ \zeta = \frac{c}{c_0 c_o} = 2 \sqrt{\frac{k}{m}} \]  

(6)

and \( c_0 \) is the critical damping value. With damping the resonant frequency becomes

\[ \beta = \omega \sqrt{1 - \zeta^2} \]  

(7)

When a damping value of 2.0\% of the critical value is used, thus the damping factor \( \zeta = 0.02 \), the presence of damping will not significantly alter the resonant frequency from the undamped value. But the response in the resonant frequency is entirely controlled by the damping. The rest of the response will not be affected by the damping. In Figure 6.1 it can be seen that adding modal damping gives a clearer view of the response.

This model is used as reference. For further analyses the following things have been done:

6.1 Exemplification of transmission paths for third engine order

Figure 6.2 shows the response of the cab in vertical direction. Out of the response of the cab some important eigenmodes can be extracted. That are 15.0, 32.0, 38.0, 43.5 and 58.0 Hz.

- Eigenmode at 15.0 Herz:

For this mode the engine is rolling as a rigid body. There is no deformation of the engine or the gearbox. There is also no deformation in the crossmember to which the front engine mount is attached. This crossmember follows the lower side of the engine with a certain phase delay. The crossmember itself excites the longitudinal beam of the frame, which leads to a torsion of this beam. The frame is in phase with the engine. From the front view the frame is swinging clockwise around some longitudinal axis (Figure 6.3). The frame is very soft.

It can be seen that most of the deformation of the cab appear in the front of the cab due to the bushing. The bushing is in phase with the engine and frame and moves also clockwise. Because there is no lateral spring in the rear cab suspension, most of the vibrations is transmitted through the bushing. Due to this the front cab is moving relative to the rear of the cab. This results in a twisting of the cab (Figure 6.4). For this frequency the vertical acceleration
(Figure 6.2) is very high for the front of the cab, even higher than the lateral acceleration. The magnitude of the front cab suspension and the bushing are the same. As the lateral position of the front cab suspension is twice the lateral position of the bushing, there is no roll of the cab. It can also be seen that there is a large deformation of the bottom plate of the cab (Figure 6.4).

7. Conclusions

- Once the superelements have been created it is fairly easy to do parameter studies and modifications of the residual. The extra effort put in the superelements pays itself back afterwards.

- To examine the transmission paths, some output files have to be read into Ideas which takes a long time. Sometimes it is difficult to see a difference between the modes, but in general it is possible to get a clear knowledge of the eigenmodes. The different engine modes found in the analysis did conform with experiences from measurements.

- The results of the parameter studies are rather trivial. This model can be used for parameter studies. The model should not be used to evaluate the absolute acceleration level in the cab. This model should mainly be used for determining the direction of the response change, due to some modification.

- The efficiency of the superelement technique is high and can be illustrated by the following examples:

  Sol3, only 20 eigenvalues, so number of Lanczos shifts = 2: 5000 cpu seconds.

  Dynamic reduction of cab, 400 Hz, 584 eigenvalues : 36,000 cpu sec.

  Calculations on residual a quite fast. Freq. resp. 200 Hz = 3300 cpu, mainly because of increments. 100 Hz = 184 eigenvalues, with modal damping: 3000 cpu.

  So it is much faster.

  But the database need a lot of diskspace. For the superelements a total of 300 Megabytes is needed.
8. References


2. Kraker A. de ‘Numeriek-experimentele analyse van dynamische systemen’.
Figure 5.1  
**Applied dynamic forces on engine**

- **load.fo6 302952 T2 MD: Top cylinder**
- **load.fo6 302832 T2 MD: lower node**
- **load.fo6 301001 T2 MD: bearing**
- **load.fo6 302001 T2 MD: bearing**
Figure 6.1  Modal damping

Lateral acc. of bushing

- No modal damping, 420003 T2 MA
- 2.0 %, 420003 T2 MA
Figure 6.2 Original model

- solid line: forig 420001 T3 MA: front
- dashed line: forig 420002 T3 MA: rear
- dash-dotted line: forig 420003 T3 MA: bushing
Figure 6.3

Third order
15.0 Hertz
10 %
Third order
15.0 Hertz
10 %