Engine Induced Vibrations 2

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WFW-rapport 96.006
Stageverslag

Januari 1996

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Abstract

A full-vehicle FE-model of a FH12 truck has been used for studies of engine induced vibrations. The Super element technique in the FE-program CSA/NASTRAN has been used. Attention is given to the subject of dynamic analysis and the cab Super Element. Damping is also discussed. Besides this the modelling of certain parts of a truck are investigated: the fuel tank and the shock absorbers. The Finite Element model can be a good tool to investigate the influence of certain parameters on the engine induced vibrations.
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1. Introduction

A Finite Element model of a truck has been developed in the past in order to perform dynamic analyses. This FEM is a good tool to investigate the vibrations caused by the engine and the influence of different parameters on the response.

The truck modelled is a FH12 4x2 tractor, WB3800 with a L2H1 Cab and a D12 engine. In this studies the model is used to examine the engine induced vibrations in the frequency range 20-150 Hz.

Only half of the truck is modelled because in general a truck is more or less symmetric. Because anti-symmetric loadcases are used anti-symmetric boundary conditions are applied to the model. The model consists of approximately 24,000 elements and 23,000 nodes. Analyzing the structure will take a lot of calculation time if a reduction of degrees of freedom is not performed. Therefore the structure is divided into the following Superlements: frame, engine, cab and the residual. A modal reduction is performed on the Superlements of the frame, engine and cab (see [1] for a short description of the reduction method). This results in a reduction of degrees of freedom from about 140,000 to 2000. With this reduced model it is easier to make adjustments to the residual and analyse the new structure.

The model is used to perform various parameter studies. Input are different engine orders: 3.0\textsuperscript{rd} and 4.5\textsuperscript{th} order and the 3.0\textsuperscript{rd} order for the idle situation. The Output looked at will mainly be the vibrations of the cab suspensions.

First in chapter 2, the loadcases used as input will be shortly described. In chapter 3, the solution methods for dynamic analysis will be discussed. The necessary cab reduction in chapter 4. In chapter 5 attention will be given to the influence of damping on the response of the structure. The influence of the fuel tank is described in chapter 6 while the modelling of the shockabsorber is point of discussion in chapter 7. The conclusions and recommendations can be found in the chapters 8 and 9. Finally some general remarks are given in chapter 10.
2. Loadcases: engine excitation

Different engine orders are the input for the model. For every order dynamic loads are applied to the cylinders and bearings of the engine.

The following loadcases were used as excitation:

- 3.0^{rd} order : Full load
- 4.5^{th} order : Full load
- 3.0^{rd} order : Idle load

In this report mainly the third order idle loadcase is used.
3. Solution method dynamic analysis

Dynamic loads are applied on the cylinders and bearings of the engine. The response of the engine on these forces is the input for the rest of the truck. Dynamic response analysis is performed to obtain the engine induced vibrations of the truck. There are two possible ways to perform the dynamic response analysis: using a direct formulation or a modal formulation. A short description of both methods will be given next. After that the best method to be used in this situation will be point of discussion.

3.1 Direct formulation

In the direct method of dynamic problem formulation the displacements at the gridpoints are used as the degrees of freedom, \( \{u\} \). The linear equation of motion can be written as:

\[
[M]\{\ddot{u}\} + [B]\{\dot{u}\} + [K]\{u\} = P(t)
\]  

The frequency response solution of this equation is performed as follows:

\[
[-M\omega^2 + iB\omega + K]\{u(\omega)\} = \{P(\omega)\}
\]  

No eigenvalues and eigenmodes are calculated using this method. The direct method will usually be more efficient than the modal method for problems with dynamic coupling in which a large number of vibration modes are required for accurate results.

3.2 Modal formulation

In the modal method of dynamic problem formulation the vibration modes of the structure in a selected frequency range are used as degrees of freedom, also called generalized coordinates. This reduces the number of degrees of freedom while the accuracy is maintained in the selected frequency range. Suppose the number of modes used in the modal formulation equals \( m \) and \( [\phi] \) is the matrix of eigenvectors. Then \( \{u\} \) can be expressed in terms of the mode shapes as follows:

\[
\{u\} = \sum_{i=1}^{m} \{\phi_i\} \{\xi_i\}
\]

or

\[
\{u\} = [\phi]\{\xi\}
\]

Substituting equation (4) into equation (1) and premultiplying by \([\phi]^T\) yields

\[
\]
This can be written as

\[ [m] \dddot{x} + [b] \ddot{x} + [k] \dot{x} = p \]  

with \([m]\) the generalized mass, \([b]\) the generalized damping, \([k]\) the generalized stiffness and \(p\) the generalized force. This equation is solved in CSA/NASTRAN. The equations for the modal method become coupled or uncoupled depending on the specification of damping. In the used model damping due to viscous damping elements CDAMPi is present resulting in nondiagonal matrices and thus in coupled equations. In CSA/NASTRAN the coupled form of the modal formulation is therefore used which increases the calculation time compared to the uncoupled situation.

### 3.3 Direct versus Modal method

Simulations have been done to compare the both methods. The cab was reduced in the frequency range 0-150 Hz. The frame and the engine in the range 0-400 Hz. One solution has been obtained using the direct method. The modal method has been used for different frequency ranges of the residual:

- 10-550 Hz
- 0-400 Hz
- 0-550 Hz

A plot of the lateral velocities of the rear cab mount is given in appendix 3.3.1. In the direct method no eigenvalues and modes are calculated so the eigenfrequencies are implicitly taken into account. The modal method only uses the eigenmodes in the chosen frequency range. In the plot it can be seen that eigenmodes for high frequencies (over 500 Hz) are necessary to give the same results for the direct and modal method. This is a little strange for the engine supplies an input in the range 30-90 Hz. It could be that local eigenmodes are the reason for high frequency modes to be necessary because in other positions of the truck such high frequencies are not needed for the modal method to give comparable results with the direct method. The low frequency eigenmodes (0-10 Hz) are also necessary when using the modal method. Because a lot of eigenmodes are necessary for the modal method to give accurate results the computation time is quite long compared to that of the direct method. When using the direct method the CPU time needed is 206 seconds. The modal method, frequency range 0-550 Hz, needs 526 seconds.

In the model CDAMPi elements are used so there is dynamic coupling. Also a lot of eigenmodes are required to give accurate results. therefore the direct method will be more efficient than the modal method. The direct method will be used from now on in the various simulations.
4. Necessary cab reduction

The databases of the cab are the largest of the various databases. The sizes very much depend on the frequency range selected for the modal reduction. The size of the databases also effects the calculation time of the simulations. Database sizes as small as possible are preferred because of the required disc space to store them.

The cut-off frequency must be at least as large as the highest frequency present in the input signal, preferably even twice as large. The highest frequency present in the 1.5th order is 60 Hz while the highest frequency in the 3.0rd order (and also the third order idle loadcase) equals 90 Hz.

To validate whether a 0-150 Hz reduced cab can be used for the third order engine vibrations simulations with both a 0-150 Hz and a 0-400 Hz reduced cab have been performed. The results for the front cab mount are shown in appendix 4.1. For frequencies up to 125 Hz the response for both cabs is the same. This is also the case for other points in the model. So it is valid to use a 0-150 Hz reduced cab for the third order engine inputs (and of course the 1.5th order).

For the 4.5th engine order using a cut-off frequency twice as large as the input means that the cab should be modally reduced in a range of 0-300 Hz. This results in very large database sizes for the cab and a long calculation time for the simulations. Database sizes as small as possible are preferred. Therefore examined is if a cab reduction of 0-150 Hz is sufficient enough for the 4.5th engine order. Results for a cab reduction of 0-300 Hz are compared with results for a cab reduction of 0-150 Hz. They can be seen in appendix 4.2. Differences between the two cabs start to occur around 75 Hz (rule of thumb). Up to 100 Hz these differences are relatively small. At higher frequencies the difference becomes larger. For the 4.5th order therefore using a 0-150 Hz reduced cab is only valid for frequencies up to 75 Hz.

In this studies however we are not primarily interested in the exact values of the response but more in the influence of certain parameters on the response. Exact values are not really necessary in that case, only the relative change of the response when changing a parameter. Therefore the smaller cab can be used, but one has to be careful when looking at higher frequencies. The reason for not using the 0-300 Hz reduced cab is because of the database size. A 0-300 Hz reduced cab needs about 260 Mbytes and a 0-150 Hz reduced cab about 130 Mbytes. This is quite a large difference. Also the computation time will be less using the smaller cab. The elapsed time for a simulation with the large cab was 1367 seconds. For the small cab this was 282 seconds. But with the small cab an updated version of CSA/NASTRAN was used, which could also have reduced the computation time.
5. Damping

When no damping exists in the model then the equations of motion in the modal formulation can be written as:

\[ [m] \dddot{x} + [k] \ddot{x} = p \]  

(7)

Modal damping can be included in the model using the TABDAMP1 card. The equations of motion will then become:

\[ [m] \dddot{x} + [b] \ddot{x} + [k] \dot{x} = p \]  

(8)

The modal damping matrix is given by

\[ [b] = [\omega \gamma \omega] m \]  

(9)

For the direct formulation structural/material damping can be included in the model specifying the constant 'g', which is equal to twice the critical damping fraction, in either the PARAM bulk data card or in material property cards MATi. In this studies only the PARAM card is used.

The structural damping is added as the imaginary part of the stiffness matrix:

\[ [K_{dd}] = (1 + ig) [K^1_{dd}] + [K^2_{dd}] + i[K^4_{dd}] \]  

(10)

The superscript 1 indicates the matrix generated by the Element Matrix Assembler. The superscript 2 indicates the direct input matrices. The matrix \([K^4_{dd}]\) is a structural damping matrix obtained by multiplying the stiffness matrix of an individual structural element by the damping factor specified in the MATi card for that element.

The structural damping force is constant over the frequency range (for a constant displacement) as shown in figure 1.

\[ F^g = ig[K^1]u \]

Fig. 1. Structural damping for constant displacement
To find the influence of the structural damping several simulations have been done with different values for the damping constant $g$. The parameter $g$ is defined as twice the critical damping fraction, $g = 2 \frac{c}{c_0}$. The values used are: $g = 0.04$, $0.08$, $0.10$ and $0.20$. In appendix 5.1 the response of the front cab is shown. Increasing damping shows a reduction of the height of the peaks. With a damping factor of $0.20$ some peaks can hardly be seen. It is not obvious which damping factor is the most realistic one. A damping factor of $g = 0.10$ has been chosen for the rest of this studies. So the structural damping is $5\%$ of the critical damping value. With this damping the peaks can still be distinguished. When using a higher damping factor the peaks become very small and are very difficult to spot.
6. Fuel tank modelling

In the previous model no fuel tank was modelled. The mass of the fuel tank is quite large compared to that of the frame. A full fuel tank can weigh about 600 kg while the frame weighs about 300 kg. The tank is connected hanging sideways of the frame, therefore the fuel tank can have a significant influence on the response of the truck.

The fuel tank is modelled as shown in figure 2. A CBAR element is connected to the frame by 2 rigid bars. At the end of the CBAR concentrated masses were placed to model the mass of the fuel tank. The CBAR did not have a mass. Three suspension points were created to make it possible to place the fuel tank in three positions:

1.) In front of the crossmember
2.) Rear of the crossmember
3.) On both sides of the crossmember

In figure 2 the fuel tank is placed on both sides of the crossmember.

![Fuel tank installation diagram](image)

**Fig. 2. Fuel tank installation**

The stiffness, mass and position of the fuel tank have been examined. In the next paragraph attention will be given to the stiffness of the fuel tank.

6.1 Bending stiffness of the fuel tank

First the fuel tank is modelled with no torsion stiffness, only bending stiffness. Four situations have been examined. The values for the area cross-section of the bar and the area moments of
inertia are specified in Table 1. They were obtained by using values similar to that of round bars of 1, 50, 100 and 250 mm in diameter respectively \( \text{Area} = (\pi D^2)/4 \); \( I = (\pi D^4)/64 \).

Table 1. Values for fuel tank stiffness

<table>
<thead>
<tr>
<th>Situation</th>
<th>Area (( \text{mm}^2 ))</th>
<th>( I_{zz} ) (( \text{mm}^4 ))</th>
<th>( I_{yy} ) (( \text{mm}^4 ))</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.14</td>
<td>4.9E-2</td>
<td>4.9E-2</td>
</tr>
<tr>
<td>2</td>
<td>1963.5</td>
<td>3.07E5</td>
<td>3.07E5</td>
</tr>
<tr>
<td>3</td>
<td>7854.0</td>
<td>4.91E6</td>
<td>4.91E6</td>
</tr>
<tr>
<td>4</td>
<td>49100.0</td>
<td>1.92E8</td>
<td>1.92E8</td>
</tr>
</tbody>
</table>

The second situation is the one that has been used in the model so far. In all the parameter studies in the next chapters this is also the case, unless stated differently.

The input for the model was the third order and the mass of the fuel tank 600 kg. In Appendix 6.1.1 the results are plotted for the front cab mount with the fuel tank connected on both sides of the crossmember. Increasing the stiffness from situation 2 to situation 3 decreases the velocities. Further increasement of the stiffness however increases the velocities.

In Appendix 6.1.2 the results are plotted for the rear cab with the fuel tank connected in front of the crossmember. The results for situation 3 and 4 are almost similar. So when you start at a certain low stiffness and increase this stiffness then first the response will change but at a certain value increasing the stiffness has no effect.

It can be seen that the bending stiffness of the fuel tank can have a some influence on the dynamic behavior of the truck. Especially when the tank is mounted on both sides of the crossmember.

Real values of fuel tank stiffness are not known to me, so values to use in the simulations had to be chosen. In real life the fuel tank has not a very high bending stiffness, therefore situation 2 is chosen for the rest of the parameter studies.

6.2 Torsion stiffness of the fuel tank

Besides bending stiffness also torsion stiffness will now be given to the fuel tank. The torsion stiffness of the tank was given a value similar to that of a 250 mm round bar: \( J = (\pi D^4)/32 = 3.84\text{E8} \). Simulations were done for two engine inputs: the third order idle input and the 4.5th order. In both cases the mass of the fuel tank was 200 kg and placed in front of the crossmember.

Some results for the third order idle situation are plotted in Appendix 6.2.1. In general hardly any differences can be seen between the fuel tank with and without torsion stiffness for the third order idle situation. This is different with the 4.5th order as input. In Appendix 6.2.2 results can be seen for this situation. In this case it does make a difference if the fuel tank is modelled with a torsion stiffness. For the third order inputs the engine is rolling as a rigid body. The frame will then not greatly be ... on torsion and the fuel tank torsion stiffness will not be of large importance. For the 4.5th order (and the 1.5th order) however the engine is twisting causing torsion of the frame and a torsion stiffness of the fuel tank is of influence.

The lateral displacement of the cab suspension on the frame side is much larger than that at the
cab side (see appendix 6.2.3) so the torsion axis is located closer to the cab then to the frame. If
the cab mount is located on the torsion axis then the lateral displacement due to torsion will be
minimized.
In real life the torsion stiffness of the fuel tank is perhaps not as large as modelled because of
slip at the tank suspensions.

6.3 Mass of the fuel tank

The mass of the fuel tank has been varied from 0, 200, 400 to 600 kg. The third order idle load-
case was used with the fuel tank in front of the crossmember. For the front cab suspension and
the cab link bushing the results are plotted in appendices 6.3.1 and 6.3.2. The plots show a
large influence of the mass on the response. The way in which the response changes however
varies for different positions. At one position, in a certain frequency range the velocities can be
decreased with an increasing mass while at another position they can be increased.
So it is quite important to include the mass of the fuel tank in the model. It is not obvious what
mass should be used because in real life this changes when driving the truck.

6.4 Position of the fuel tank

As mentioned before in this chapter the fuel tank can be placed in three positions. To evaluate
the differences between the three positions simulations have been done with the third order idle
loadcase and a 200 kg fuel tank. In appendices 6.4.1 to 6.4.3 results can be seen for the cab link
bushing and the front cab mount. For the front and rear cab mounts the position does not really
make a great difference in the frequency range from 30-90 Hz (for all directions). The cab link
bushing displacements in longitudinal and lateral direction however show larger differences in
frequencies up to 50 Hz. Looking at lateral displacements the rear tank suspension seems the
best solution while that on both sides of the crossmembers should be avoided. However in lon-
gitudinal direction there is a large peak for the rear fuel tank at about 30 Hz which is not
wanted. So maybe the advantages and disadvantages level each other out and it does not really
matter which fuel tank position is used. In real life also the position of the fuel tank is very
truck dependent.
7. Shock absorber modelling

So far, the shock absorbers were modelled by linear viscous dampers. In this paragraph attention will be given of how to model the shock absorbers in a more realistic way. In reality two bushings at both ends of the damper are present to mount the shock absorber to the structure. The rubber bushings will act like two springs. This means that the shock absorber can be seen as a viscous damper in series with springs:

\[
\begin{align*}
\frac{k_1}{c} & \quad k_2 \\
\frac{k_f}{c} & = \frac{k_1 \cdot k_2}{k_1 + k_2}
\end{align*}
\]

For this system the transfer function \( H(f) = \frac{F}{x} \) equals:

\[
H(f) = \frac{2 \pi f k_j j}{2 \pi f + \frac{k_f}{c}}
\]  \hspace{1cm} (11)

The equivalent stiffness and damping can be defined by:

\[
K_{eq} = Re \{ H(f) \} \quad \text{(12)}
\]

\[
C_{eq} = \frac{Im \{ H(f) \}}{\omega} \quad \text{(13)}
\]

So the stiffness and damping of the system are frequency dependent. For \( c = 5.0 \text{ Ns/mm} \) and \( k_f = [100, 200, 400, 600, 800, 1000, 2000] \text{ N/mm} \), the equivalent stiffness and damping have been calculated with use of Matlab and are shown in the next figures.
For low frequencies the stiffness of the system is very low. This increases with increasing frequency to the value of the spring constant. The damping decreases with increasing frequency from the damper constant to a low value. So for high frequencies the system acts like a spring while for low frequencies the system acts like a viscous damper.

In reality however the mass of the shockabsorber is also involved. This has been modelled in CSA/NASTRAN. The mass $m$ was assumed to be $1.0 \text{ kg}$, $c = 5.0 \text{ Ns/mm}$ and $k_1 = k_2 = 2000 \text{ N/mm}$ so $k_t = 1000 \text{ N/mm}$. The model looks as in shown in figure 5.
Fig. 5. Model of shock absorber as modelled in CSA/NASTRAN

The equivalent stiffness and damping found for this system are given in the figures below.

Fig. 6. Equivalent stiffness of NASTRAN model of shock absorber

Fig. 7. Equivalent damping of NASTRAN model of shock absorber
Because of mass resonance for high frequencies a decrease in equivalent stiffness can be seen. Modelling the shockabsorber by a spring and damper in series is a more realistic way than just to use a viscous damper. The model as given above was implemented at the vertical front cab suspension and the rear cab suspension in lateral and vertical direction. As can be seen in the results (appendix 7.1 to 7.3) it makes quite a lot of difference if the shockabsorbers are modelled in this more realistic way. Therefore it can be good to model all the other shockabsorbers present in the system in this way.

The bushing is represented by a spring in this model. In reality however there is a rubber component. The stiffness of this rubber is also frequency dependent and perhaps there is also some damping. In the next version of CSA/NASTRAN (version 96) it is possible to define frequency dependent stiffness and damping for spring and damper elements. So this rubber properties can also be taken into account to give a better model for the shockabsorber.
8. Conclusions

In this report results are shown concerning the engine induced vibrations. A finite element model of a truck has been used to perform dynamic analyses. For most parameter studies the third order idle loadcase has been used.

In this case the direct method is the most efficient method to use. Storage of the database of the cab takes a lot of disc space, therefore a 0-150 Hz modally reduced cab has been used for the simulations.

Including a fuel tank in the model is quite important, the mass has the most influence. The modelling of certain parts of the truck can be very important, for example the front cab stabilizer and the shockabsorbers. Especially modelling the shockabsorber in a more realistic way has a large effect on the response.

The analysis with the truck model can be helpful to see the influence of certain factors on the engine induced vibrations. Changes can be applied in reality to see if they have a similar effect as found with the numerical model.
9. Recommendations

Further attention should be given to the way certain parts of the truck are modelled. Especially rubber components can be important. These components have frequency dependent characteristics (stiffness and damping). In the next CSA/NASTRAN version (version 96) it will be possible to model frequency dependent springs and dampers. These features should be applied to see their effect on the response. Also the modelling of the shockabsorbers needs a closer look because it is important to have a good description of the dynamical behavior of the shockabsorbers. In this report a very simple model of a shockabsorber was used. This model can be further developed, for example including masses and frequency dependent bushings.
10. General remarks

- **Database size**
  PHIAR contains the calculated eigenvectors. The storage of them take the most disc-space.
  So extracting less eigenmodes significantly decreases the needed disc space.
  A 0-400 Hz reduced cab needs 356 Mbyte.
  A 0-150 Hz reduced cab needs 130 Mbyte.

- **Blocks**
  In a previous report (see [1]) it was mentioned that using the option BLOCKS in the
  ASSIGN-card minimizes the database size. This feature has been used but there appeared to
  be very little difference in database size

- **Block shifted Lanczos**
  When using the 0-400 Hz reduced cab the block shifted Lanczos method of eigenvalue
  extraction will not be used. With smaller cabs it is possible to use the block shifted Lanczos
  method.

- Rigid body modes are not extracted if you use the following card:
  EIGR,4000,BLAN,0.0,...
  A frequency range from -0.1,... has to be specified.
11. References

1. Wijckmans, M; Engine induced vibrations; 1994


Direct vs Modal method
Third order
Bear cab. lateral

Veloc. (mm/s)

- csadir.03 420002 T2 MV
- csamod.01 420002 T2 MV
- csamod.02 420002 T2 MV
- csamod.04 420002 T2 MV
Cab 0-150 Hz versus Cab 0-300 Hz reduced

4.5 order

Bushing - vertical velocity

![Graph showing Cab 0-150 Hz and Cab 0-300 Hz reduced]
Influence of structural damping
third order
front cab., lateral velocities

![Graph showing the influence of structural damping with various curves representing different orders and velocities.](image-url)
Fuel tank stiffness
front cab, vertical velocity
fuel tank on both sides of crossmember

Velocity (mm/s)

Increasing stiffness

- fuel30.output01 420001 T3 MV
- fuel30.output02 420001 T3 MV
- fuel30.output03 420001 T3 MV
- fuel30.output04 420001 T3 MV
Fuel tank stiffness

Rear cab, vertical velocity

fuel tank in front of crossmember
Lat. displ. rear cab: frame & cab side
4.5 order

*10^-3

displ.
(mm)

10 6 5 4 3 2 1

40 60 80 100 120 140

torsion.f06 420002 T2 MD
 torsion.f06 4901 T2 MD

(6.2.3)
SHOCKABSORBERS REALISTIC MODELLING

RIDLE

acc. (mm/s²)

350
300
250
200
150
100
50

20 30 40 50 60 70 80 90 100

- - - -
frontcz.02 420001 T2 MA

- - - -
realshock.02 420001 T2 MA

- - - -
realshock.03 420001 T2 MA