FE-based Vehicle Analysis of Heavy Trucks
Part I: Methods and Models

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ABSTRACT: A method for complete vehicle analysis based on FE-technique will be presented. This method is used for analysis of complete vehicle features such as vehicle dynamics and durability. The large number of vehicle variants and increasing demands on optimisation and modularization for trucks call for effective tools to handle the complete vehicle analysis. In-house developed tools for pre- and postprocessing are used to achieve effectiveness. Necessary FE model complexity for the different types of complete vehicle analyses will be discussed. Finally full use of the method will be illustrated by an example.

NOTATION:

V_0	forward speed
$f_{z.ref}$	yaw rotation of vehicle
$x_{n.fe}$	local displacement in x-direction, node *n*
$y_{n.fe}$	local displacement in y-direction, node *n*
a	slip angle
s	relaxation length
$f_{z.ref}$	angular wheel rotation
$f_{z.ref}$	half contact length
y	local displacement in y-direction
$f_{z.wheel}$	yaw rotation of wheel
C_F	cornering stiffness (lateral force)
C_M	cornering stiffness (self aligning torque)
k'/V_0	angular damping constant
w	rotational speed of the wheel
I_w	moment of inertia of the wheel
r	tyre radius (loaded)

1 INTRODUCTION

Today there are increased demands on trucks not only on cost and weight but also on improved complete vehicle features.
This result in increased focus on optimisation and modularization which together with the large number of vehicle variants makes it necessary to use efficient analysis methods.

FE-based vehicle analysis has become an important part of the development process for a number of complete vehicle features such as:

- Ride comfort and vibrations
- Handling
- Durability and Reliability

The main objective of this paper is to describe how FE-based complete vehicle models are used for the typical dynamic analyses involved in the development of heavy trucks and what kind of demands this puts on the software. In addition, necessary complexity of the models for the different kind of analyses will be discussed as well as the importance of efficient pre- and postprocessing.

2 FE-BASED ANALYSIS

There are different methods and tools available for complete vehicle analysis. The different types of models are:

- Discrete linear models
- FE-models
- MBS-models

In the concept stage of a new project, discrete linear models are often used to find appropriate levels of stiffness and damping as well as optimal positions for the suspension elements. This is usually sufficient for modelling the vertical dynamics for road induced vibrations at early stages.
However for many of the necessary analyses in the feature optimisation a more detailed model description has to be used.

The different types of analyses involved for the different features are:

- Ride comfort and vibrations
  - Analysis of road induced vibrations
  - Analysis of wheel and brake induced vibrations
  - Analysis of driveline induced vibrations
- Handling
- Durability and Reliability
  - Analysis of static loadcases
  - Fatigue life prediction by simulating the proving ground or real life conditions

In the lateral direction it is difficult to represent the frame structure with rigid elements and discrete springs due to complex mode shapes and influence from local elasticity at the attachment points. This can be solved by using a FE-description of the frame and other components with distributed stiffness and mass. FE-based analysis is also to prefer for wheel, brake and drive-line induced vibrations as well as for fatigue life prediction where higher frequencies are important.
All these cases call for detailed information of the structure as well as a description of the global rigid body modes.
Therefore, besides the structural representation of the vehicle, more complex boundary conditions have to be used in most of the applications. The rolling tyres create cornering forces, self aligning torque, as well as gyroscopic effects that has to be included in the FE analysis.
In this way some of the advantages of a MBS model can be included in the full vehicle FE model to get a time-efficient analysis method.
The analyses are done both in the frequency domain and in the time domain. For analyses in the time domain, like fatigue life prediction, it is possible to include nonlinearities in the FE-models.

3 MODEL

The description of the model includes the following steps:

1. FE-model of the full vehicle
2. Reduction of problem size
3. Global motion and centrifugal forces
4. Lateral tyre model
5. Gyroscopic moments

Besides the traditional structural description of the vehicle the system matrices have to be modified with respect to the more complex boundary conditions. This is possible to do in a straightforward manner in Nastran. This results in non-symmetric system matrices.

3.1 FE-model of the full vehicle

A typical full vehicle FE-model for dynamic and static analysis of trucks is shown in Fig. 1. The application is a two-axle tractor with a trailer.
The model is assembled of the following main components, (subsystems):

a) **Frame.** FE-structure with shell elements using a mesh also acceptable for stress analysis.

b) **Superstructure.** Trailer connected to the frame by the fifth wheel. The stiffness and mass properties of the trailer are obtained by bar elements.

c) **Cab.**
   - Rigid body connected to the frame with springs, shock absorbers and anti roll bar.

   or
   - Full FE-model for structural stiffness-, damping- and mass properties, Fig. 2.

d) **Engine.**
   - Rigid body with engine mounts.

   or
   - Full FE-model for realistic engine excitation from the combustion.

e) **Axles and chassis suspensions.** Both front and rear axles are connected to the frame with leaf/air springs and anti-roll bars, modelled with bar elements. Shock absorbers are also included.

f) **Steering system.** Front wheel with angular freedom around kingpin. Caster and kingpin angles as well as steering geometry are modelled in laden condition. The steering gear is modelled with the gear ratio and angular stiffness and damping.

3.2 Reduction of problem size

It is necessary to reduce the problem size in order to allow for dynamic analyses and fast parameter studies. This is done in two steps.

A first reduction is achieved by use of superelements and component mode synthesis. The frame is always treated as a superelement, the cab and engine or other additional components are also treated as superelement if they are fully FE-modelled.

Further reduction of the full model, including the reduced frame, is achieved in a second step. By solving for real eigenmodes and using only modal data in the frequency range of interest for the final analysis of complex eigenvalues, frequency- or transient response.

This results in fast parameter studies when no structural modifications are done to the superelements.

3.3 Global motion and centrifugal force

Additional equations have to be added in order to take care of global rigid body motions and centrifugal forces, Fig. 3.

Global motion:

For node *n* in the FE-model

$$X_n = x_{n.fe} - V_0 t \qquad (1)$$

$$Y_n = y_{n.fe} - \int_0^t V_0 f_{z.ref}\, dt \qquad (2)$$

Centrifugal forces:

Addition of centrifugal forces to the centre of gravity for the main components such as superstructure, cab, engine, frame and axles.

$$F_c = m V_0 \overset{*}{f}_{z.ref} \qquad (3)$$

Eq. 3 is added to the damping matrix for the centre of gravity of these masses in order to cover the total mass distribution of the vehicle.

3.4 Lateral tyre model

The "straight tangent" lateral tyre model [1] is used to model the lateral boundary condition between the wheels and the road, Fig. 4. A point contact model is used for the boundary condition in vertical direction.

With the assumption that the directions for y and y' are equal and $a' = a$ the equation for the "straight tangent model" is

$$\frac{s}{V_0}\dot{a} + a = y - \frac{a}{V_0}\dot{y} - \frac{1}{V_0}\dot{y} \qquad (4)$$

In the FE-model the slip angle a is a control variable defined by a transfer function. However, due to the numerical solution algorithm, a mass term has to be included as well.

The transfer function to be implemented with a small $m\ddot{a}$:

$$m\ddot{a} + \frac{s}{V_0}\dot{a} + a = y - \frac{a}{V_0}\dot{y} - \frac{1}{V_0}\dot{y} \qquad (5)$$

where

$$y = -(f_{z.wheel} - f_{z.ref}) \qquad (6)$$

Forces and moments acting on the wheel in the road contact point:

$$F_y = C_a a \qquad (7)$$

$$M_z = C_M a \qquad (8)$$

$$M_z^{'} = \frac{k^{'}}{V_0}\dot{y} \qquad (9)$$

Eq. 7 and Eq. 8 are added to the stiffness matrix and Eq. 9 to the damping matrix.

3.5 Gyroscopic moments

Moments on the rotating wheel due to gyroscopic effects are included as well, Fig. 5.
General equation:

$$\overline{M} = I_w \overline{w} \times \dot{f} \qquad (10)$$

This gives moment in x and z directions on the wheels:

$$M_x = I_w w \dot{f}_z \qquad (11)$$

$$M_z = -I_w w \dot{f}_x \qquad (12)$$

where

$$w = -\frac{V_0}{r} \qquad (13)$$

Eq. 11and Eq. 12 is added to the damping matrix.

4 PRE- AND POSTPROCESSING

To be able to handle the large amount of truck variants a specialised pre- and postprocessing tool has been developed.

To accomplish increased quality, efficiency, flexibility and to lower the costs, the trucks are built from a global module concept. From a limited number of cabs, engines, gearboxes and axle installations etc., these components can be combined in many ways to fulfil different types of transport missions, Fig. 6. The architecture of the pre processing system reflects this reality, Fig. 7

From a database of FE models of cabs, engines, axle installations and frame components like cross members, the truck frame is automatically generated and the components assembled. Different type of fasteners like rivets are also automatically generated, Fig. 8 and Fig. 9.

The system can handle different types of analysis like static analysis or dynamic analysis in the frequency and time domain. Specialised postprocessing tools have also been developed for the different types of analysis, Fig. 10.

5 EXAMPLE

Wheel induced vibration has a large influence on the vibration environment for the driver. Even though the absolute levels are usually quite small, they can easily be felt when excited by a harmonic excitation. Wheel unbalance or wheel run-out can cause this excitation. Since the quality and conditions of tyres varies it is important to find a design where the vehicle sensitivity is minimised.

This example shows the vertical and lateral cab acceleration caused by a periodic excitation of the front axle due to a front wheel run-out.
Normal vehicle speeds (70-100 [km/h]) and typical tyre sizes results in an excitation frequency band of 6-9 [Hz] due to first order wheel-run-out.
A typical mode excited by first order wheel run-out at normal speed is shown in Fig. 11.
The vertical and lateral cab response as a function of the frequency can be seen in Fig. 12, where the interesting frequency band is from 6 to 9 Hz.

Further use of the method with focus on durability is exemplified in [2] where the full vehicle models are used to simulate the proving ground or real life conditions in the time domain.

6 CONCLUSIONS

In the automotive area there are different important analyses ranging from low to high complexity models with demands on accuracy from low to high frequency.

- FE-based complete vehicle analysis has the advantages to cover a broad range of analyses with only one type of model.

In the design process of trucks with the many different variants that has to be investigated and optimised it is important to have efficient analysis methods.

- FE-based complete vehicle analysis has the potential to be a time efficient method because:

    1. It is possible to adapt an efficient pre- and post processing to the models, specially an automatic model generator.

    2. By use of superelement technique there are almost no limitation of the complexity of the different subsystem of interest. The increased complexity is usually achieved without too high time consumption. Fast parameter studies and optimisation can still be done for all variants.

In order to maintain or even increase the efficiency of the method the following demands are put on the FE-software:

- Stable format for in- and output
- An easy handling of the substructuring
- Time efficient solver

REFERENCES

(1) Pacejka, H. B. (1980) “Tyre factors and front wheel vibrations”, *Int. J. of Vehicle Design*, Vol. 1, no. 2, 1980, pp. 97-119.

(2) Öijer, F., “FE-based Vehicle Analysis of Heavy Trucks, Part II: Prediction of Force Histories for Fatigue Life Analysis”, Proceedings of 2nd MSC Worldwide Automotive Conference, MSC, 2000.

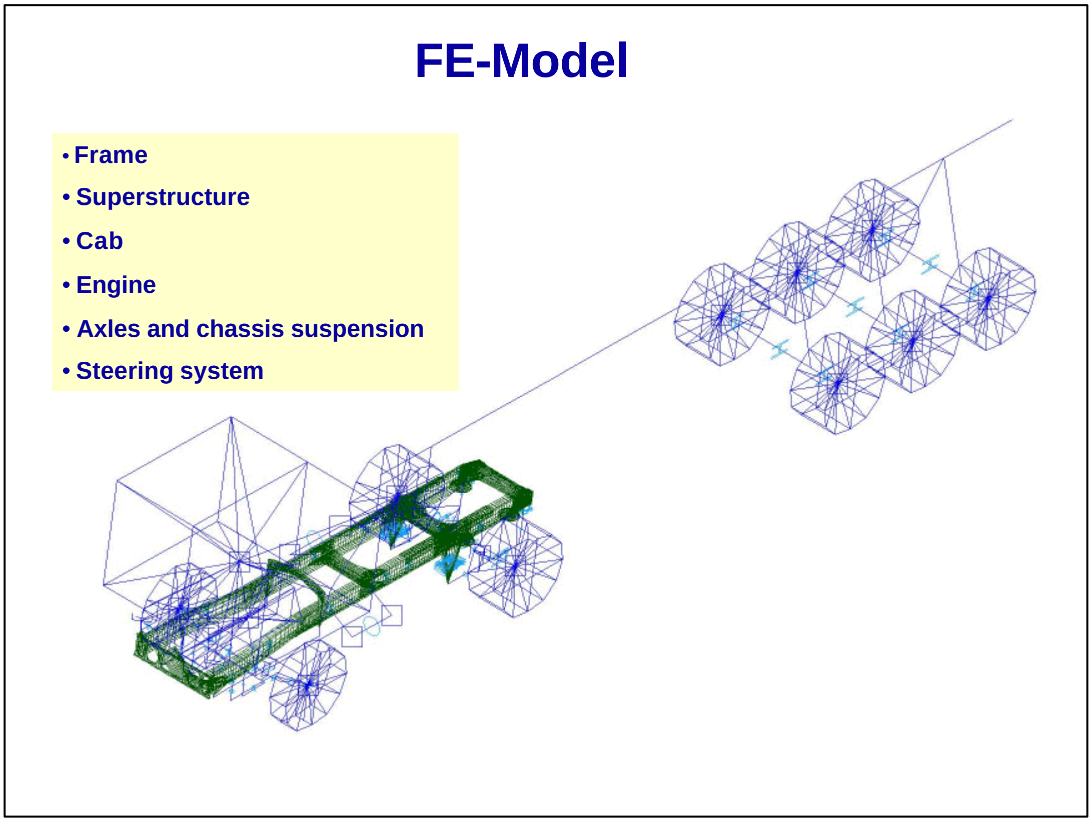


Figure 1.

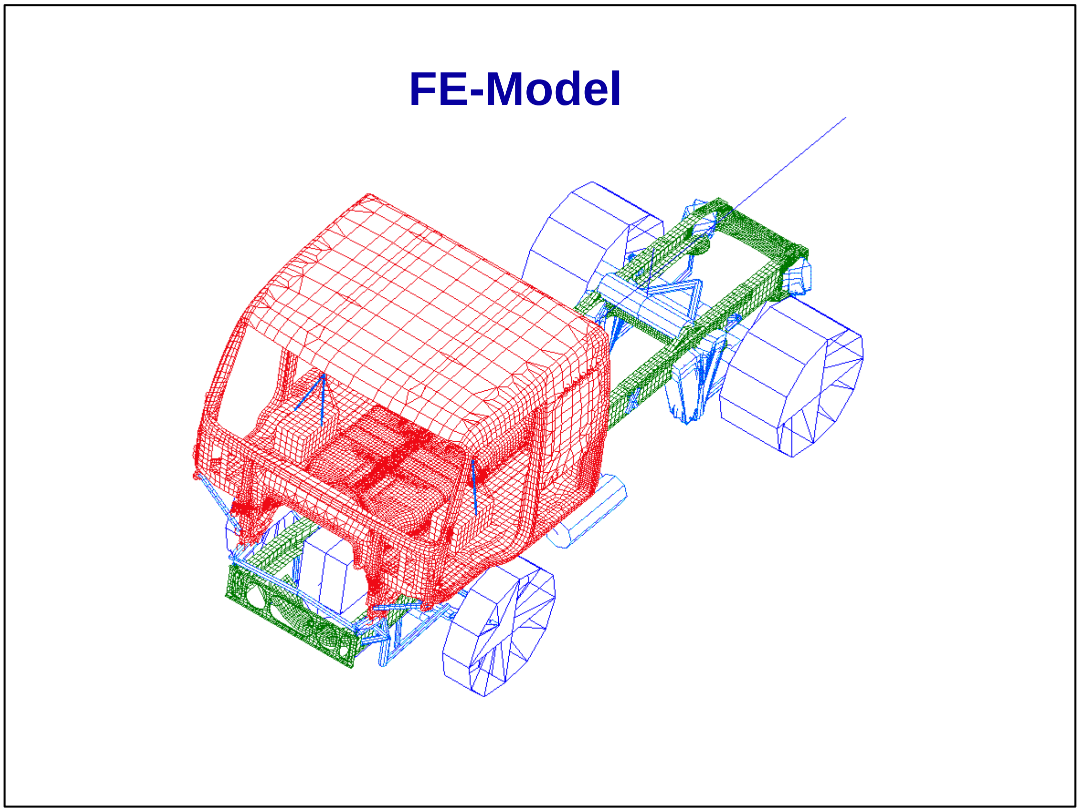


Figure 2.

Centrifugal forces

$$F_c = m V_0 \dot{\phi}_{z.ref}$$

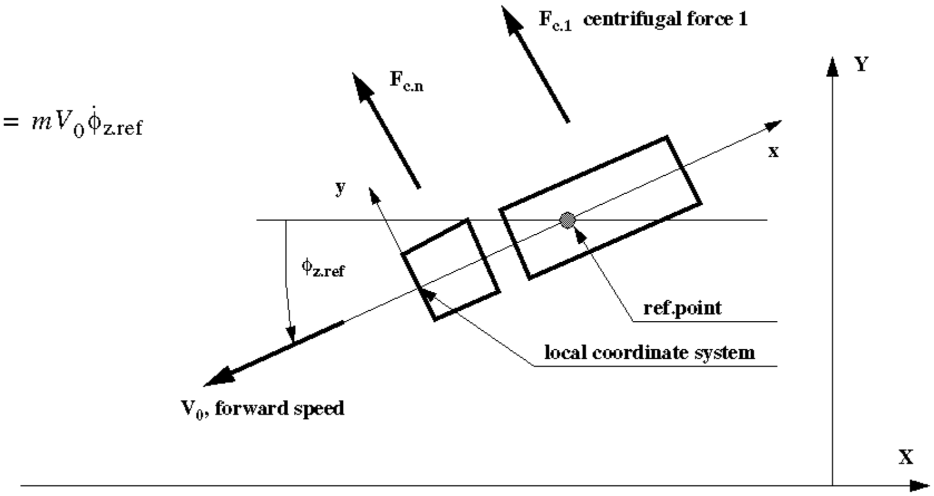
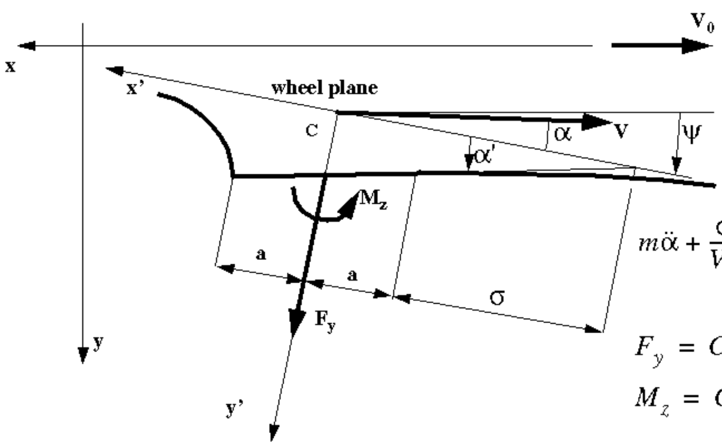


Figure 3.

Lateral tyre model

- slip angle
- steer angle
- relaxation
- small mass term



$$m\ddot{\alpha} + \frac{\sigma}{V_0}\dot{\alpha} + \alpha = \psi - \frac{a}{V_0}\dot{\psi} - \frac{1}{V_0}\dot{y}$$

$$F_y = C_F \alpha$$

$$M_z = C_M \alpha$$

$$M'_z = \frac{\kappa'}{V_0}\dot{\psi}$$

Figure 4.

Gyroscopic moments

$$M_x = I_\omega \omega \dot{\phi}_z$$

$$M_z = -I_\omega \omega \dot{\phi}_x$$

$$\omega = -\frac{V_0}{r}$$

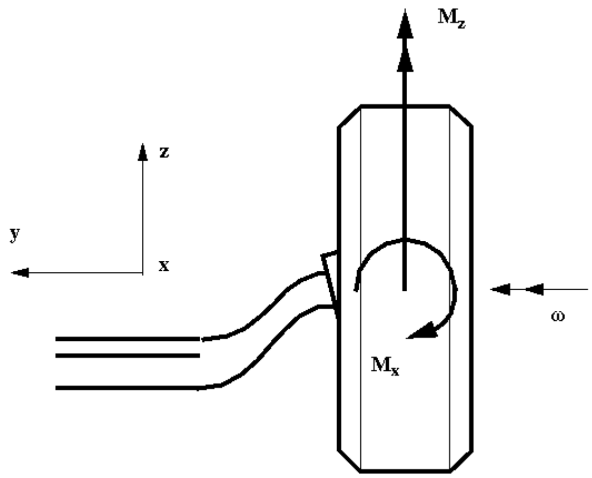
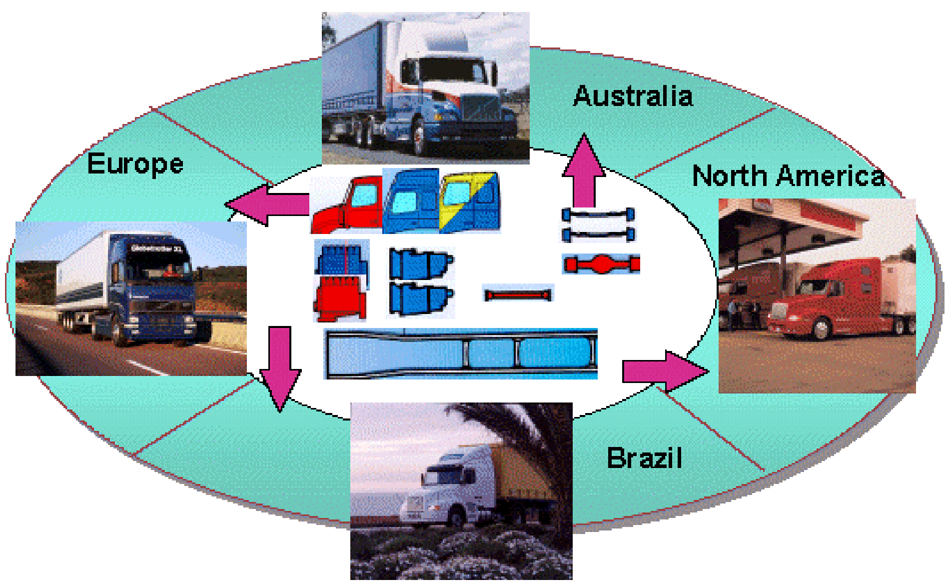


Figure 5.

Global module concept

*** Increased quality * Flexibiliy * Increased efficiency * Lower costs**



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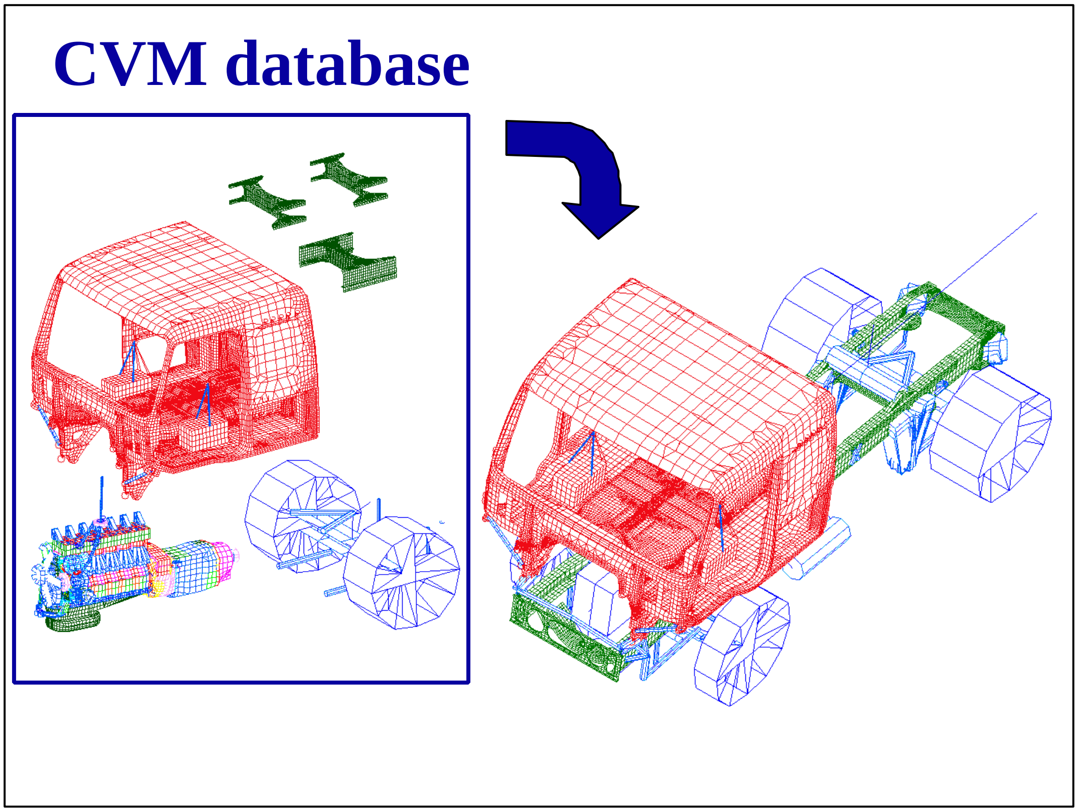

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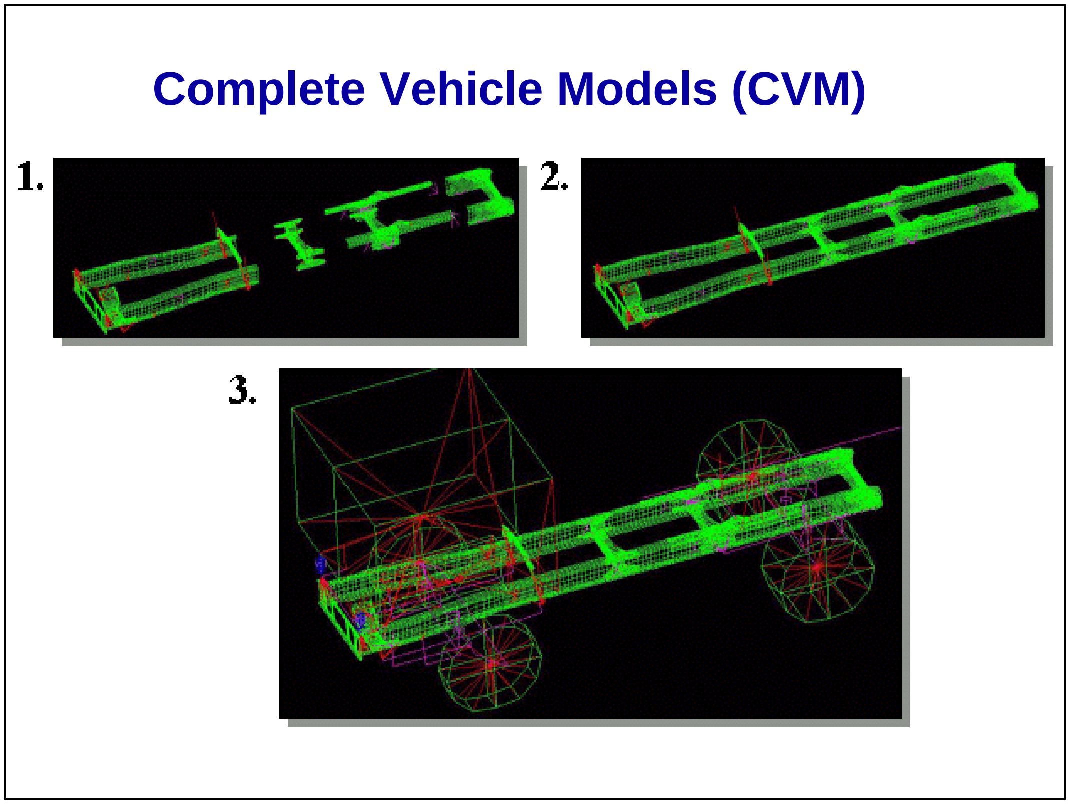

Figure 8.

Complete Vehicle Models (CVM)

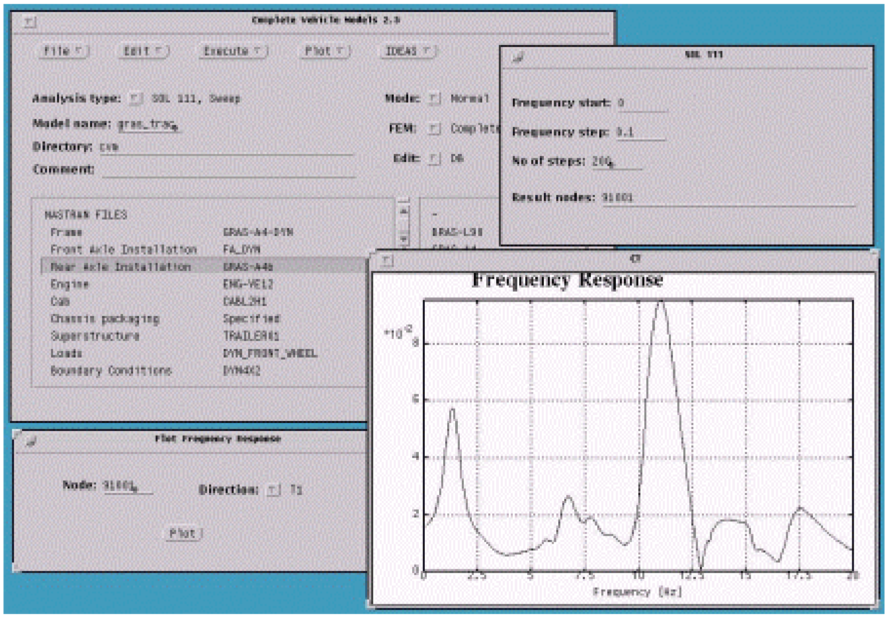

Figure 9.

Postprocessing

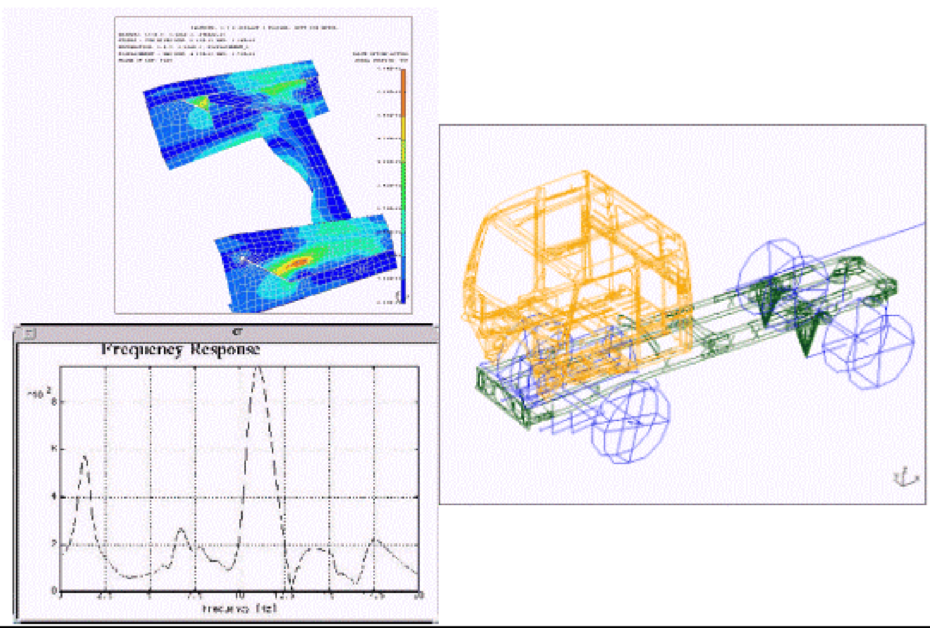

Figure 10.

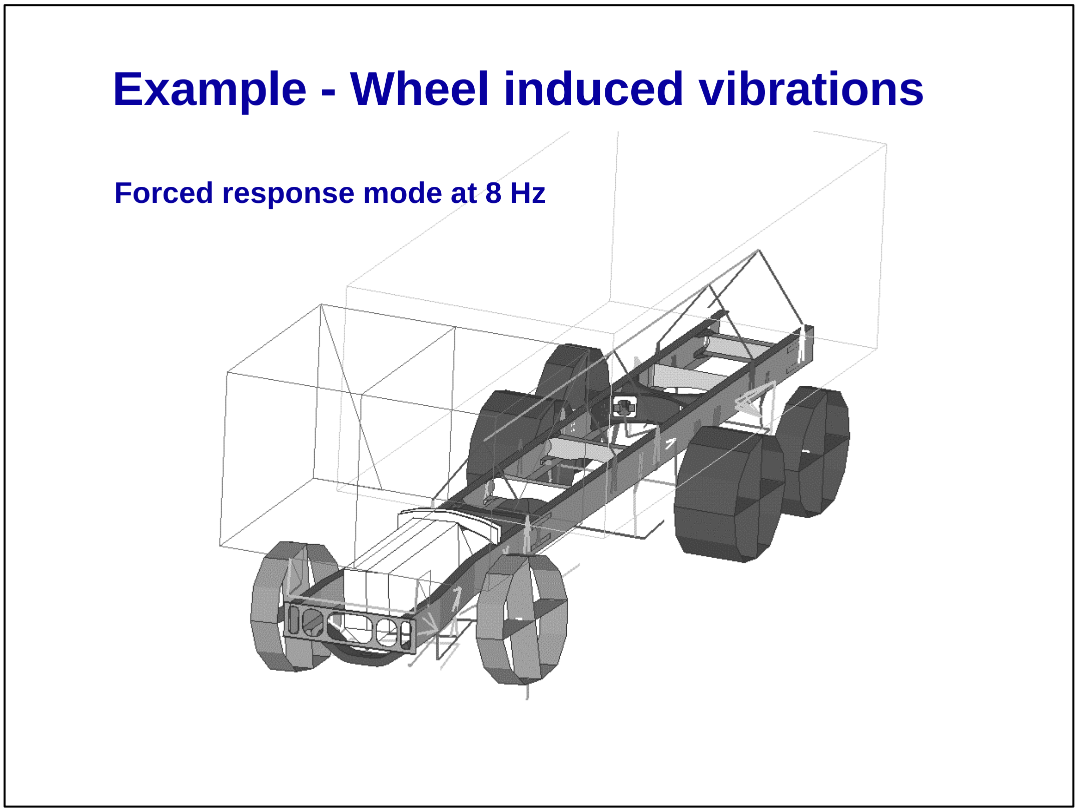

Figure 11.

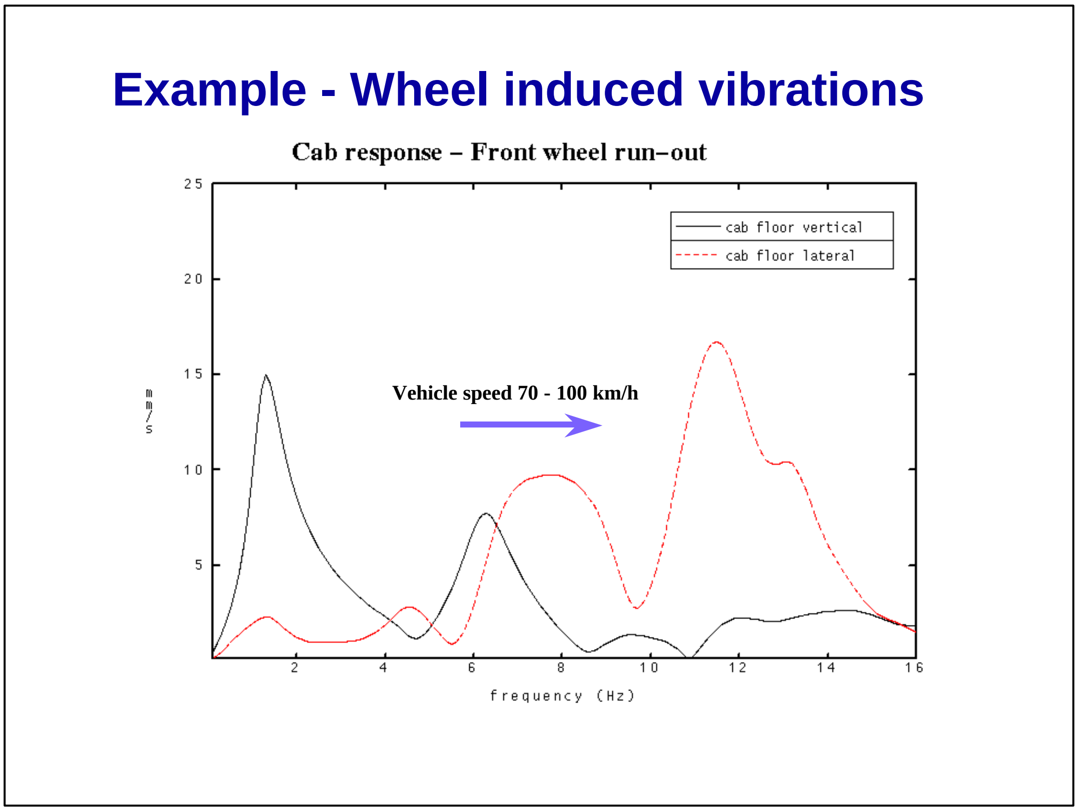

Figure 12.