FE-based Vehicle Analysis of Heavy Trucks; Part II: Prediction of Force Histories for Fatigue Life Analysis

F. Öijer*
Volvo Truck Corporation, S-405 08 Göteborg, Sweden

ABSTRACT

In the automotive industry, computer simulations of vehicle durability early on in the design process are becoming more and more important in order to decrease development cost and product time to market.

Accurate calculations of force histories are of utmost importance for reliable fatigue life estimates. The forces are often calculated by use of multi-body software and used as input for stress analysis in a FE package. A drawback is that the MBS calculations are very time consuming, especially if flexible bodies are included, and are thus, not well suited for fast parameter studies.

To overcome this a different method is proposed. In this method the force and/or stress histories are calculated directly in MSC.Nastran using complete vehicle models, where environment variables, such as road profile and curve radii, are used as input. This, in combination with modal superelement reduction, will result in fast design studies.

The calculated forces are compared with measurements for different road inputs. Correlation is good for frequencies up to 25 Hz. Studied transient obstacles also show good correlation to measurements.

* E-mail: fredrik_oijer@vtc.volvo.se, phone: +46 31 666710

1 Introduction

In heavy truck development a challenge is the large number of variants generated by different transport missions. From a simulation point of view, this means that a large number of vehicle configurations needs to be analysed, which calls for effective analysis tools if the whole population is to be optimised.

One type of analysis that is growing in importance is the simulation of forces for durability assessment. However, most (all) methods use multi-body simulation (MBS) codes to perform the calculation of the forces. Although very useful in many areas, MBS codes are quite slow for the integration of long time signals, such as, the ones needed for fatigue analysis, especially if flexible bodies are included. For truck simulations it is vital to have a flexible representation of the frame.

The objective of this study has been to find a more efficient method for the calculation of forces, which allows for fast analyses and easy implementation of flexible bodies.

2 Model

To calculate the forces due to road input models of both the vehicle itself and of the environment, which the vehicle is driven in, are needed.

2.1 Description of Environment

The environment, which the vehicle is driven in, is described by vertical road profile and curve radii. These environmental parameters have been calculated from extensive field measurements and are (almost) vehicle independent, i.e. they can be used as input for any vehicle model.

The road profile is applied to the tires of the vehicle. The curve radii (R) together with the forward velocity of the vehicle (V) give the yaw velocity as: $\dot{\varphi} = V / R$. A driver model is used to give the vehicle the desired yaw.

2.2 Vehicle Model

As described in [1] the vehicle structure is modeled by standard elements such as shells, bars, springs and dampers, see fig. 1. The cab and engine are modeled as either rigid or flexible.

To get more correct dynamic behavior some extra features have to be added, e.g. tire slip, centrifugal forces and gyroscopic moments.

When calculating forces due to road input in transient response analysis even more features are needed.

- More advanced tires

Depending on the road surface different tires are used. For low frequent road profiles an one-point contact tire is usually sufficient. For higher frequencies a more complex tire model is required and in this work a flexible ring tire is used. The tire is modeled mainly by bars, springs and dampers and its behavior agrees well with measured eigenfrequencies up to about 75 Hz. The tire can transfer longitudinal and vertical forces to the wheel spindles in addition to slip forces.

- Driver model

A simple driver model that steers the vehicle based on (mainly) yaw rate is included. In this model the steering wheel angles calculated from: $\dot{\delta} = f(\varphi, \dot{\varphi}, \ddot{\varphi})$, where δ = steering wheel angle, φ = vehicle yaw angle.

- Variable forward velocity

The coefficients for tire slip, centrifugal forces and gyroscopic moments are functions of (among other things) the forward velocity of the vehicle and therefore need to be updated with changing velocity. This is done quite easily in MSC.Nastran with NOLINx cards.

- Non-linear suspension characteristics

The most important non-linearities to include are damper characteristics, bump stops, leaf spring hysteresis and bushings.

3 Analysis

Superelement reduction, static or dynamic depending on the situation, is used to minimise computational time. The parts that are reduced are the frame, the cab and the engine, whereas the suspensions are kept in the residual.

Modal transient analysis (SOL 112) is used for the response studies. SOL 112 allows for very fast parameter studies together with easy implementation of non-linearities with EPOINTs, TFs and NOLINx cards. Since the non-linear forces at time t are applied at t+1, it is important to use a sufficiently small time step to ensure stability and accuracy.

4 Results

The method has been validated for various types of roads and truck variants.

The first check was to see if the forces measured on the vehicle used for the collection of the road parameters (see section 2.1) agreed with the calculated forces. The measured and calculated forces on a component in the front axle suspension can be seen in fig. 2. The input is a typically bad road with some curves and dips with fairly low frequency.

The agreement between simulation and measurement is pretty good. To condense the results further the `fictitious fatigue life' of the signals was calculated using a Wöhler curve with slope 5. This gave that the calculated signal had 1.7 times the life of the measured signal (calculated life/measured life = 1.7). Since the value is greater than one, the calculation is somewhat non-conservative, but still close.

Next, the road parameters was used as input for another vehicle and measured and calculated forces for a different component was compared. The forces in fig. 3 are for a component in a new rear axle suspension. Also here the agreement between measurement and calculation is good and the fictitious fatigue life ratio was, again, 1.7.

The method has also been used for roads with higher frequency. The first example is a road with a few curves and transient obstacles, such as curbstones and potholes. Plotted in fig. 4 is the force on a component in the cab suspension. Again the agreement is good if one looks at the time histories. The `fatigue life ratio' was 1.6, which is about the same as for the low frequent input.

The second example is a washboard with varying frequency. Fig. 5 shows an exceedance plot for measured and calculated longitudinal force in the front cab suspension. The fatigue life ratio was in this case 1.5.

Most studied cases have a `fictitious fatigue life ratio' between 1.5 and 2.

5 Discussion

The importance of including the truck frame flexibility in the analysis in order to get correct responses has been investigated before [2] and will not be discussed further here. Though it should be mentioned that the results in the previous section could not have been achieved with a rigid frame. However, a flexible part that usually is overlooked in this type of analysis is the truck cab. The calculation for fig. 5 was done with a flexible cab in the vehicle model. The reason is that when driving on roads with high frequency, an analysis with a rigid cab will sometimes overestimate the cab suspension forces with a factor two or more, resulting in a much too short fatigue life and a too heavy structure. This is clearly shown in fig. 6, where the longitudinal force in the front cab suspension has been calculated using both a flexible and a rigid cab. The vertical cab forces are not affected in the same way by the flexibility, but only the horizontal forces.

There is no doubt non-linearities in, for example, the axle suspensions in many cases are very important for good load estimations. But quite often other features are much more important for a correct response, for example cab flexibility. This is something that is sometimes forgotten today when analysts with ease can include every imaginable non-linearity in MBS softwares via a graphical interface without reflecting if they are actually needed. It should be noted that the analyses giving the results shown in fig. 2 and fig. 3 were completely linear!

For low frequencies (with TSTEP $\geq$ 0.01) most analyses run in real time or faster. The "worst" performance is 17 times real time. This is for the washboard mentioned above. This performance gives the opportunity to perform a wide range of parameter studies in the same time as one single MBS run without sacrificing accuracy.

To increase speed in fatigue life assessments further it would be desirable to have fatigue analysis included directly in MSC.Nastran's SOL 112. This would reduce the amount of data produced, since only one value per element/node needs to be stored, the accumulated damage, and not complete stress histories. Another obvious advantage is that no data needs to be transferred between softwares.

6 Conclusions

Using MSC.Nastran to calculate loads for fatigue analysis does have some benefits over the usual MBS alternative. It is very fast with good accuracy and flexible bodies are included pretty much by default.

This opens for the possibility to quickly analyse a large number of vehicle variants, something that is very important in truck development.

7 References

[1] Johansson I. and Gustavsson M., "FE-based Vehicle Analysis of Heavy Trucks, Part I: Methods & Models", Proceedings of 2nd MSC Worldwide Automotive Conference, MSC, 2000

[2] Koppenaal J. and Edlund S., "Commercial Vehicle Modeling and the Influence of Truck Frame Flexibility on Vehicle Response", Proceedings of 1997 International ADAMS User Conference, MDI, 1997

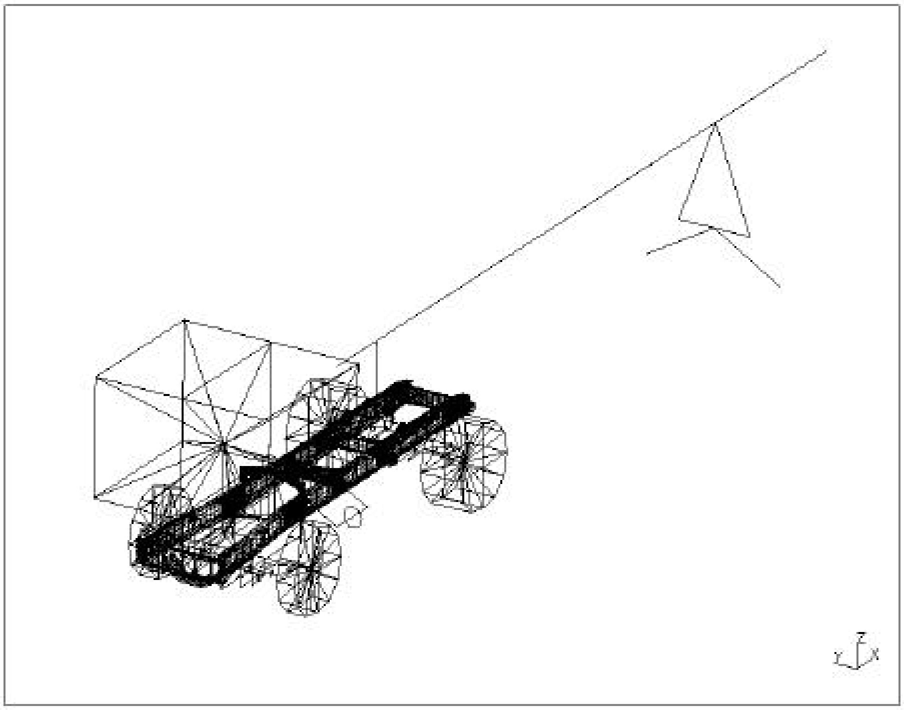

Figure 1. FE model of truck.

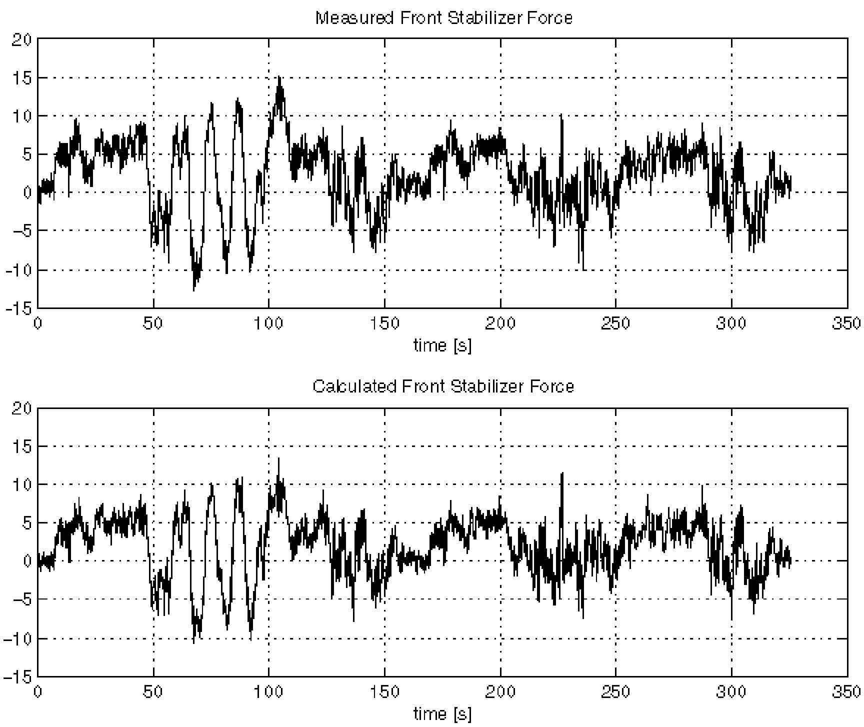

Figure 2. Comparison of measured and calculated force in a component in front axle installation.

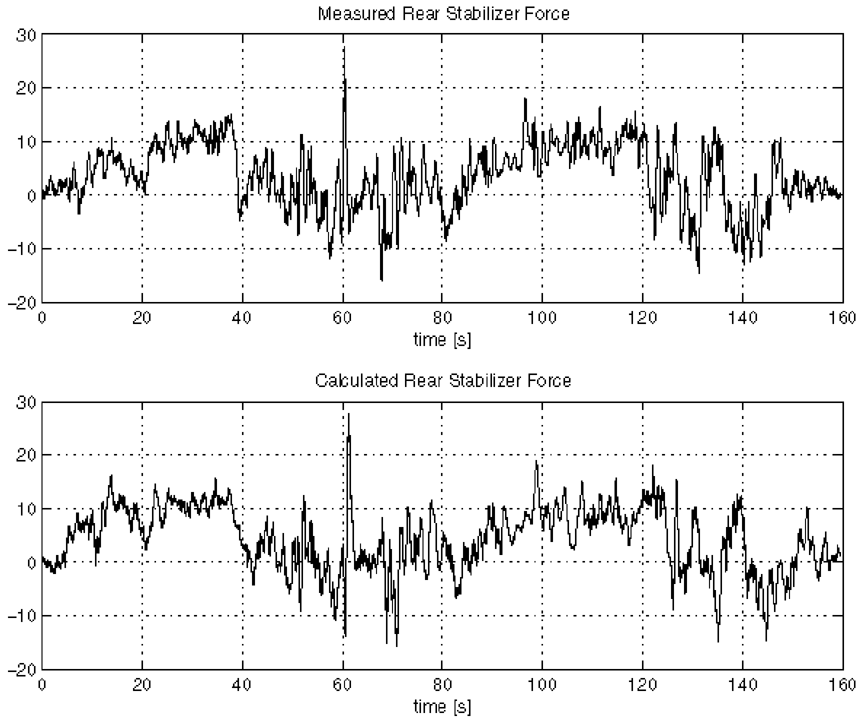

Figure 3. Force on component in rear axle installation

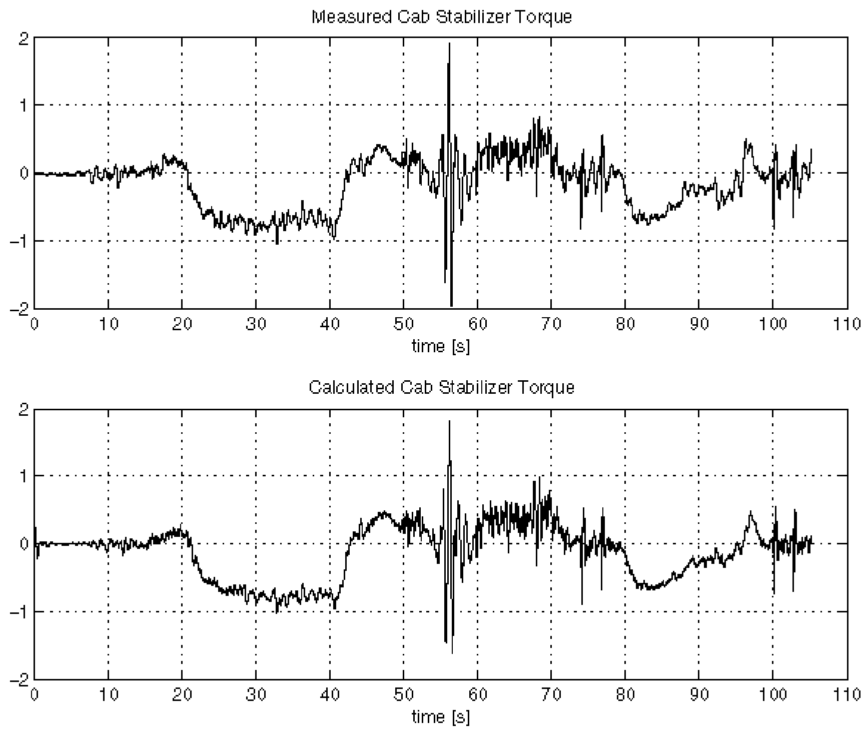

Figure 4. Cab stabilizer torque

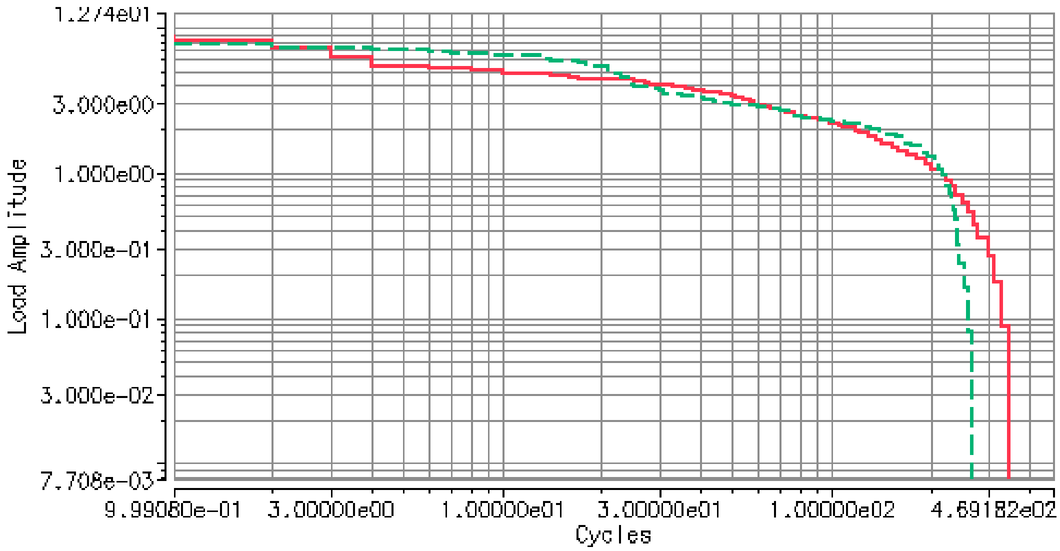

Figure 5. Longitudinal cab force. solid – measured, dashed – calculated.

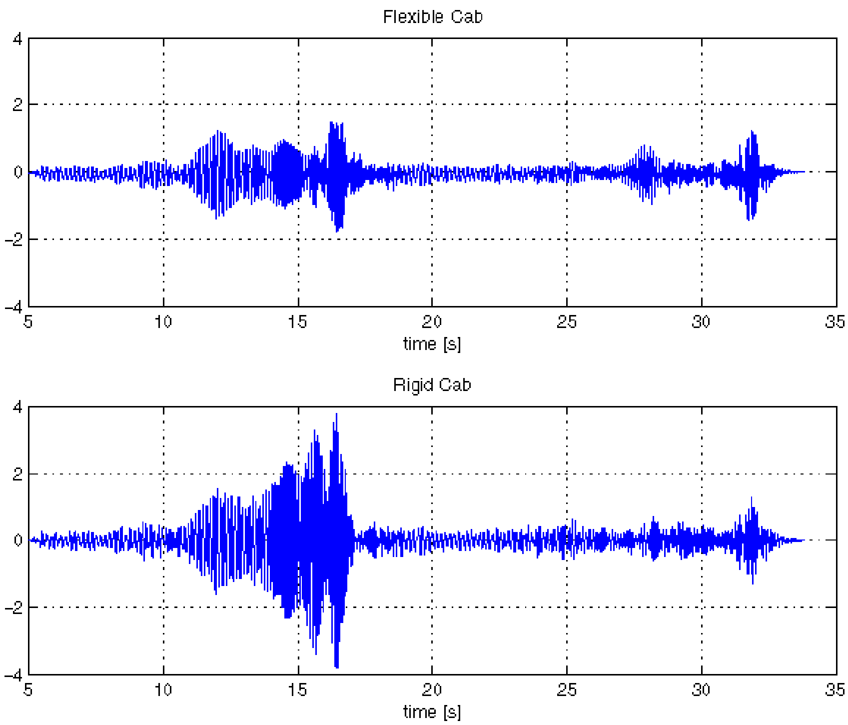

Figure 6. Comparison of forces in cab suspension with flexible and rigid cab models.