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Application of Hybrid Frequency Domain Substructuring for Modelling an Automotive Engine Suspension

A practical application of hybrid FRF-coupling (Frequency Response Function) in the development of a passenger car is presented. First, a short review is given about FRF-coupling in general. Next, some problems are discussed which may be encountered when both analytical and experimental FRF-data is used in FRF-coupling. This is also known as hybrid modelling. The main part of this paper presents a successful application of hybrid FRF-coupling to analyze and solve an interior noise problem of a passenger car. Both analytical and experimental FRFs were used to create a hybrid dynamic model of a complete passenger car. The engine and its suspension system were modelled using finite elements, while the remainder of the car was modelled by experimentally derived FRFs. This hybrid model was then used to compute the response of the vehicle due to the engine excitation. Measured noise transfer function were used next to compute the interior sound pressure level using forced response results of the hybrid car model. Subsequently, the hybrid model was used to analyze the problem, and to predict the effects of an alternative design of the engine suspension on interior noise. Numerical results indicated that the alternative design would have a significant positive effect on noise. This was confirmed by verification measurements on a car.

1 Introduction

Numerical simulation techniques are commonly used today for the development of passenger cars. This is also the case at NedCar in Helmond, where such simulations are used extensively in vehicle design projects such as the Volvo 300 series, and in more recent projects such as the 400 series.

Typical for the approach used is that mainly theoretically derived numerical models are used, for example finite element models, and that experimental results mostly serve for the verification of numerical results. However, the general dynamic behavior of a passenger car is rather complex. In this case finite element models may be inadequate to simulate the dynamic behavior of the component, or only at the cost of excessive modelling efforts. In such a case one can alternatively derive the numerical model using both a theoretical and experimental approach. This is also known as “hybrid” modelling.

One of the techniques which can be used is known as FRF-coupling. A large part of the presented papers related to FRF-coupling, however, deal with theoretical aspects of this technique, and use relatively small sized problems as examples. A minority of papers describe practical applications to complex structures, e.g., Ewéns et al. (1980), Fingberg and Ahlersmeyer (1992), and Lim and Steyer (1992).

Because FRF-coupling is a relatively simple technique, with the potential of being able to solve complex dynamic problems, it was decided in the beginning of 1992 at the structural analysis department of NedCar that this technique should be made operational to simulate complex dynamic systems in general, and complete passenger cars in particular. Therefore, as a joint effort with the Eindhoven University of Technology, a modular computer program called COMIC (COMponent Impedance Coupling) was developed to make all necessary calculations. In addition, a number of modules were developed to support some pre- and postprocessing features.

In this paper some results achieved using this method will be presented. All of the results concern the development of an engine suspension system. We will start however with a short discussion of the theory behind FRF-coupling in general, and potential problems related to the use of experimentally derived FRF-data. For detailed information see Huizinga (1995).

2 Theory

In this section the governing equations of FRF-coupling are presented. First, an expression will be derived for the frequency response functions of a mechanical system. Next, it will be shown how this expression can be used for a load path analysis and to compute the sound pressure level in a cavity such as the interior of an automotive vehicle.

In this section the following assumptions are made:

- the mechanical system consists of two components $i$ and $j$
- both components are interconnected at a discrete number of locations
- both components behave linearly, time invariant. Reciprocity is assumed.

The displacements and applied loads of component $i$ can be partitioned as illustrated in Fig. 1 (the partitioning for component $j$ goes identical). In this figure the following notation is used:

$$x_i^{(i)}, \bar{x}_i^{(i)} = \text{coordinates belonging only to component } i \text{ or } j$$

$$(\text{subscript } i: \text{ "Internal"})$$

$$x_j^{(i)}, \bar{x}_j^{(i)} = \text{coordinates of component } i \text{ or } j \text{ which will be connected to the other component}$$

$$(\text{subscript } B: \text{ "Boundary"})$$

The frequency response functions of components $i$ and $j$ can be expressed as a matrix equation. Using the above partitioning we can write for component $i$:

$$
\begin{bmatrix}
    x_i^{(0)} \\
    x_i^{(1)}
\end{bmatrix} =
\begin{bmatrix}
    H_{bb}^{(0)} & H_{bj}^{(0)} \\
    H_{bi}^{(0)} & H_{jj}^{(0)}
\end{bmatrix}
\begin{bmatrix}
    f_b^{(0)} + f_j^{(0)} \\
    f_i^{(1)}
\end{bmatrix} = H^{(0)} f^{(1)}
$$

(1)

To write the FRF-matrix of component $j$, simply replace $i$ by $j$. Note that $H^{(0)}$ is symmetrical, and is defined for a discrete frequency.

When assembling both components, it is required that the boundary displacements of both components are equal, and that the loads are in equilibrium. Thus:

$$
\begin{bmatrix}
    x_B \\
    x_J
\end{bmatrix} =
\begin{bmatrix}
    H_{bb}^{(0)} & H_{bj}^{(0)} \\
    H_{bi}^{(0)} & H_{jj}^{(0)}
\end{bmatrix}
\begin{bmatrix}
    f_b^{(0)} + f_j^{(0)} \\
    f_i^{(1)} + f_j^{(1)}
\end{bmatrix} = f_B
$$

(3)

For this case the FRF-matrix of the assembled system $H$ can be written as (Urgeira, 1991):

$$
\begin{bmatrix}
    x_B \\
    x_J
\end{bmatrix} = H
\begin{bmatrix}
    f_b \\
    f_i^{(1)} + f_j^{(1)}
\end{bmatrix}
$$

(4)

Where:

$$
H =
\begin{bmatrix}
    H_{bb}^{(0)} & H_{bj}^{(0)} \\
    H_{bi}^{(0)} & H_{jj}^{(0)}
\end{bmatrix}
$$

(5)

This equation illustrates how to couple the FRF-matrices of two components. If the system consists of more than two components, the following two approaches can be used to compute the system FRF-matrix:

- Approach 1: Sequential coupling of components
  In this case FRF-coupling for two components is repeated a number of times. The procedure starts with the coupling of two components. The result is a sub-system FRF-matrix. The next FRF-coupling involves the FRF-matrix of this sub-system and the next component. The procedure is repeated until all components are processed.

- Approach 2: Simultaneous coupling of components
  The theory discussed before can easily be expanded to handle the coupling of more than two components simultaneously. This will result in a generalized coupling procedure. The derivation of the necessary equations is discussed in Jetmundsen et al. (1988).

Once the system FRF-matrix is computed, the response of the system due to a number of harmonic excitation loads can be computed by a straightforward matrix-vector multiplication:

$$
x = H f
$$

(6)

where:

- $x$ = response of all system coordinates
- $H$ = system FRF-matrix
- $f$ = vector with excitation loads on the system

- computation of boundary loads
  To compute the loads on the boundary of one of the components, e.g., component $i$, we use the first equation of Eq. (1), and substitute the solution for the boundary coordinates as computed by Eq. (5):

$$
x_B^{(1)} = H_{bb}^{(0)} f_b^{(0)} + H_{bj}^{(0)} f_j^{(0)}
$$

(8)

Let's define:

$$
f_B^{(1)} = f_b^{(0)} + f_j^{(0)}
$$

Then this results in:

$$
f_B^{(1)} = H_{bb}^{(0)} f_b^{(0)} + H_{bj}^{(0)} f_j^{(0)}
$$

(10)

An important feature of these component boundary loads is that they can be used to identify the most important connections for the response of the component. A procedure to do this is to compute the response of the component for each individual boundary load and plot all the resulting responses in the complex plane. Note that comparing the boundary loads themselves is not sufficient, because the sensitivity of the response to the loads is generally different. A vehicle engineering example will be presented in Section 4.

- computation of (structure-borne) sound pressure levels
  If an air cavity is included in one of the components (e.g., the interior of a vehicle), it is possible to compute structure-borne sound pressure levels inside the cavity due to an excitation of the system. The derivation of the equations involved is relatively simple:

For the response of a component we can write:

$$
x_i^{(0)} = H_{iB}^{(0)} (x_B^{(0)} - H_{BB}^{(0)} f_B^{(0)})
$$

(11)

$$
x_i^{(1)} = H_{iB}^{(0)} f_B^{(0)} + H_{iB}^{(1)} f_B^{(1)}
$$

(12)
where:

\[ f^{(i)} = \text{loads acting on component } i \]
\[ H^{(i)} = \text{FRF matrix of component } i \]
\[ x^{(i)} = \text{computed response of component } i \]

If the noise transfer functions are known from the coordinates \( x^{(i)} \) to pressure levels at one or more positions inside the cavity, then the following equation applies to the sound pressure levels at those positions:

\[ p^{(i)} = p^{(i)} f^{(i)} = p^{(i)} H^{(i)}^{-1} x^{(i)} \]  \hspace{1cm} (11)

where:

\[ p^{(i)} = \text{sound pressure levels inside the cavity of component } i \]
\[ P^{(i)} = \text{matrix with noise transfer functions} \]

Note that in the section above, the units of the noise transfer functions are \([\text{Pa/N}]\). A typical application is the computation of interior sound pressure levels of a vehicle, which will be demonstrated in Section 4.

The theory described above is a general one. This means that it can be applied to both analytically and experimentally derived FRF-data. However, typical problems may occur if experimentally derived FRF-data is being used, such as in hybrid modeling. A short discussion of these potential problems is presented in the next section.

3 Potential Problems of Hybrid Modelling and Suggested Solutions

For a hybrid FRF-coupling approach, where both analytically and experimentally derived FRFs are used, one has to be aware of the following potential problems:

- When FRFs are determined experimentally, the result is no more than an estimation of the true FRF. This means that the estimated FRF always contains an amount of uncertainty. Due to this, ill-conditioning can occur during the matrix inversion in Eq. (5). Other causes for ill-conditioning are (almost) dependent rows or columns in the component FRF-matrices (Urgearu, 1991). Ill-conditioning has to be avoided, because it can make the computed system FRF-data be noisy, or even incorrect. For this reason one should pay a lot of attention to the measurement of the FRFs, and try to avoid errors in the data as much as possible. If after all, the matrix to be inverted is still ill-conditioned, a least squares approximation of the true inverse, e.g., a pseudo-inverse, may be used to improve the accuracy of the system FRF-data (Leuridan et al., 1988; Otte et al., 1990).
- If components are connected to build a main system, the internal loads at the boundaries will generally consist of both forces and moments. This means that if FRF-coupling is used to compute the system FRF-matrix for such a system, it is required to include the rotational degrees of freedom (rdof's) at the boundaries too. The determination of rdof related FRFs is generally no problem if an analytical or numerical model is used. However, if the FRFs of a component have to be derived experimentally, as is the case in hybrid modelling, the determination of rdof related FRFs will generally be a problem. This would mean that unless the effect of boundary moments can be ignored (e.g., when both components are connected to each other by spherical hinges), we would not be able to compute a correct system FRF-matrix. If rdof related FRFs appear to be significant, and these FRFs cannot be measured directly, a number of methods is available to approximate them from measured translational data (Sattinger, 1980; O’Callahan et al., 1985).

4 FRF-Coupling in the Design of an Automotive Engine Suspension

Introduction. In this section a real life application of FRF-coupling in the development of an automotive engine suspension is presented. Because this development is part of a confidential vehicle project only "scaled" numerical results can be given, which still makes a valuable evaluation possible.

During the recent development of a vehicle a problem was identified with respect to interior noise levels. Various vehicles showed an unacceptable interior noise pressure level between 2200 and 2900 rpm due to the engine excitation. This is clearly visible from Fig. 2, which shows the measured interior noise pressure levels of two vehicles.

In order to solve the problem, it was decided to create a mathematical model to compute the interior noise pressure levels due to the excitation by the engine. This model would be used to analyze the problem and predict the effects of various structural modifications. Rather than trying to create an accurate finite element model of the vehicle including its air cavity, it was decided to use a hybrid FRF-coupling approach. In this way a good accuracy of the results could be obtained in significantly less time, because the time consuming generation of a finite element model (months) of the complete vehicle could be skipped. Instead, the vehicle would be characterized by measured results, which could be acquired within a time span of about one week.

The created hybrid model was comprised of two components:
Component 1: A (linear) finite element model of the engine and its suspension. This model would be used to compute the FRF-matrix (accelerance) of the engine/engine-suspension assembly.

Component 2: A measured FRF-matrix (accelerance) of the complete vehicle excluding the engine/engine-suspension. In addition the noise transfer function functions were measured from all the boundary coordinates to a location inside the cabin.

This model is depicted in Fig. 3.

Comments to Fig. 3.

Finite Element Model of the Engine With Suspension (Component 1).

- The engine (E) was modelled as a rigid body. Mass, moments of inertia and center of gravity were based on measurements
- The engine was suspended at four locations:
  - by two engine mounts situated left (LEM) respectively right (REM) of the engine. The other end of each engine mount is connected to the vehicle (points 19 and 20: front side members).
  - by two mounts called "roll stop mounts" situated at the front (FR) and rear (RR) of the engine. These are connected to the "center-member" (C). A basic model of the center member was created using one-dimensional beam elements.

Both the engine mounts and the roll stop mounts were part of the finite element model.

- Both ends of the center member are connected to the vehicle by rubber bushes (FB and RB, at points 21 and 22). These bushes are also part of the finite element model.
- All rubber parts (engine mounts, roller stoppers, bushes) were assumed to behave linearly.
- It was assumed that moments in the boundary dofs could be ignored, which limits the number of boundary degrees of freedom to twelve. Therefore, only scalar translational springs were used to model the engine mounts and roller stoppers.

Representation of Vehicle (Component 2). Only the twelve (translational) boundary degrees of freedom were used to determine the FRF-matrix. In addition mechano/acoustical noise transfer function functions were measured on the vehicle to enable the computation of noise pressure levels once the structural response of the complete vehicle was calculated. A check indicated that symmetry of the FRF-matrix could be assumed in order to save measurement time. The procedure used is described in Section 2.

Analysis of the Initial Design. FRF-coupling was used to analyze the initial design. The excitation used was the second order out of balance loads of the engine. Some results are presented in the following figures. Figure 4 shows the (scaled) second order computed acceleration level denoted "COMIC" at the rear of the center member. Figure 5 shows the computed interior noise pressure level. In addition to the computed responses the actual measured values (two vehicles) are displayed. From these figures we observe that:

- The computed acceleration levels of the rear of the center member correspond quite well to the measured levels. However, the results between 2700 and 4000 rpm seem to be shifted. In particular the peak at 3200 rpm is computed approximately 500 rpm too high.
- The computed interior noise pressure levels correspond quite well to the measured levels. Only between 1500 and 2000 rpm the computed levels are too high.

Possible reasons for these differences have been mentioned in Section 3. One of the reasons is probably that nonlinear
effects, which are likely to be present in the behavior of the rubber parts, are not represented by the linear model. Another reason was that the finite element representation of the center member was too coarse. At that time one-dimensional beam elements were used. A more detailed model, using 2-d shell elements, but still without including nonlinear effects, recently lead to slightly better results.

One of the potential problems of hybrid FRF-coupling mentioned in Section 3 is that experimentally derived FRFs always contain an amount of uncertainty. A closer look at the measured FRFs showed that the average random error of the FRFs was approximately a few percent (95 percent confidence interval). An error estimation technique was used to see what the effect of the random errors in the input data was. Figure 6 shows the relative (random) error of the coupled FRFs associated with both the (internal) degrees of freedom of the engine (1000-Z and 1000-RY), and the boundary degrees of freedom. The results are computed using a perturbation method with 500 perturbations on the input data, and are an approximation for the 95 percent confidence interval of the magnitude.

These results show that the relative error in the computed system FRFs is quite high (up to 100 percent) between 1000 and 5500 rpm. The large relative error can be explained by the error in the measured FRF data and the magnitude of the condition number, which was up to 100. Some authors (Leuridan et al., 1988; Otte et al., 1990) suggest using a pseudo-inverse rather than a direct inverse in Eq. (5) in case of ill-conditioning for better results. The question arose if this approach would improve the results. To investigate this, the coupling calculations were repeated using a pseudo-inverse. It was defined that the pseudo-inverse should be calculated such that the condition number would never exceed a value of 33.33. The ultimate results of this analysis, compared to the (direct) inverse results achieved earlier, are shown in Fig. 7. These results show that in this case it makes only a small difference to use a pseudo-inverse instead of a direct inverse. Furthermore, the results are not significantly improved compared to the results achieved with a direct inverse. For this reason no further use was made of the pseudo-inverse.

The next important step was to find a way to reduce the interior noise pressure levels by modifying the engine suspension. The contribution of each boundary force to the resultant noise pressure level was computed at 2700 rpm first. Note that for a given boundary force the (complex) contribution can be calculated by simply multiplying it with the corresponding mechanophysical noise transfer function. The resultant noise pressure level is the sum of all contributions.

In Fig. 8 the contribution of all boundary forces at 2700 rpm to the interior noise pressure level is plotted in the complex plane. In addition the resultant noise pressure level is plotted. Figure 8 shows the resultant ("TOTAL") which results from adding all the individual contributions of the boundary forces 19-X through 22-Z. The most significant boundary load to the interior noise pressure level is boundary force 22-Z. It has by far the largest magnitude, and if this force would be absent, the resultant noise pressure level would approximately be halved. These results were confirmed by a measurement on a vehicle where center members including the front and rear roller stoppers were removed. The results of this measurement is displayed in Fig. 9. It shows that the load input into the vehicle through the center member was a major contributor to the interior noise pressure level.

Analysis of the Alternative Design. The decision was made to alter the engine suspension design such that the Z-component of the boundary force at point 22 was reduced as far as possible, while the design of course still had to be feasible. After evaluating some alternative designs the most promising alternative seemed to be a design were the Z-stiffness of the rubber bushings was reduced, and some additional mass was added locally to the center member. The next figure shows the magnitude of boundary force 22-Z for both the initial and alternative (improved) design.

Fig. 8 All contributions to interior noise pressure levels at 2700 rpm: initial design
Figure 10 shows that the magnitude of boundary force 22-Z above 1700 rpm is always lower for the alternative design. Around 2500 rpm, which is the problem area, the magnitude is reduced by no less than 80 percent. This has a significant positive effect on the interior noise pressure level, as the next two figures illustrate.

Figure 11 shows the contributions of all boundary forces to the resultant noise pressure level, while Fig. 12 displays the interior noise pressure level for both the initial design and the alternative (improved) design.

Figure 11 shows that the contribution of force 22-Z is reduced to a level approximately equal in magnitude as the contributions of the other boundary forces. This reduces the magnitude of the resultant interior noise level. Figure 12 shows that the improved design is effective over a broad range. In particular around 2500 rpm the effect is significant (approximately 10 dB).

A side effect of a weaker engine suspension is that the engine motion will generally increase. Therefore it was also verified to what amount the motion of the engine was increased. Figure 13 shows the computed acceleration levels on the engine for both the initial and alternative design. Apparently the acceleration levels (and therefore also the motion) are changed. For the initial design the acceleration levels are increased by about 30
percent at 4500 rpm. In general however the differences were not dramatic, and no reason for concern.

The alternative design was evaluated on two vehicles. The measured interior noise pressure levels are shown in Fig. 14. This figure shows that the alternative design on both vehicles indeed resulted in a major improvement over the initial design. The improvement on vehicle #2 however was less than on vehicle #1.

5 Conclusions

The previous sections showed how the relatively simple technique of FRF-coupling was used very effectively in a vehicle development project. It proved to be a straightforward technique to create a hybrid dynamic model of an engine suspension of a passenger car. Nonlinear behavior of the rubber parts is probably one of the main reasons why some discrepancies existed between the predictions made with the hybrid model and measured reference data. Another reason was that the finite element model used was too coarse. Despite these differences the hybrid model was still able to identify a load path which caused an interior noise problem, and helped to solve it.

Because of the success gained with FRF-coupling in this project, the technique is now also used for problems within NedCar on noise problems related to wheel suspensions and road irregularities.

References


