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MODAL CONTROL IN RELIABILITY TESTING OF TRUCK COMPONENTS

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Abstract. *Reliability testing of components is an important part of the development process for heavy trucks. Components such as cabins, fuel tanks and mudguards are tested as subsystems, e.g. without the complete vehicle, in test rigs. Reference signals of sensors are obtained through measurements on the complete vehicle in service conditions. The test rig is designed and controlled so as to reproduce the reference signals. It is generally held that a good correlation between the reference signals and the corresponding test signals indicate that the test reproduces the damaging process of the service conditions. Due to the differences between the test setup and the complete vehicle it is not possible to perfectly reproduce neither the service conditions nor the reference signals. It is proposed to use a modal filter to estimate the modal displacements, and to control the test rig so as to reproduce the estimated modal displacements. The modal filter has been applied to an industrial test of a truck cabin.*

1 INTRODUCTION

A truck is considered as an assembly of components on a frame. Examples of components are: the cabin, fuel tanks, mudguards, battery box, and similar. The mechanical integrity of these structures is a principal concern in the design process. For chassis components the most significant loads are road-induced vibrations, and the severity of the load on a component is influenced both by the speed, the road, and the vehicle on which the component is mounted. Although trucks come in many different configurations many components are the same for all vehicles. An example is the cabin which is used on many vehicles, from heavy mining trucks to postal services lorries driving on good city roads.

Validation of the strength is performed as one of the final stages in the design process. To this extent the component is assembled on a relevant vehicle. The component and the vehicle are then equipped with sensors and measurements are performed on a test track that represents some customer application. A section of the test track is shown in Figure 1a. The signals obtained in this way represent the service conditions. In order to save time, but also to spare the vehicle and the driver, the actual validation test is performed in a laboratory. The component and attached parts of the vehicle are put in a test rig where hydraulic cylinders are used to excite the structure. Such test rig is shown in Figure 1b.

Figure 1: a) Photograph of a vehicle on the test track in Södertälje. b) Photograph of a test rig at Scania AB that aims to reproduce the damage of the test track.

The aim of the test rig is to reproduce the damaging process of the test track. The smooth parts of the test track are removed from the signal but other than that, no acceleration of the test is performed.

It is generally held that if the test rig reproduces the measured signals from the test track it also reproduces the damaging process. The actual damaging process is difficult to characterize, it involves fatigue, fracture mechanics and sometimes plasticity.

In this paper it is investigated how modal filters can be designed and used to improve the damage reproduction in the test rig.

2 EXPERIMENTAL SETUP AND MEASUREMENTS

2.1 Physical measurements

Measurements on a complete vehicle were performed on the Scania test track in Södertälje, of which a piece is shown in Figure 1. The vehicle used for this measurement was a 2 axle heavy truck for haulage application and it was equipped with 16 accelerometers and also strain sensors on the anti-roll bar of the cabin. The strain in the anti-roll bar is an important measure of the loads on the cabin suspension, and the strain sensor is used as a reference.

2.2 Experimental setup – the test rig model

The strength validation of the cabin is completed with a physical test on a real cabin. Pre-testing is performed using a simulation model of the cabin in the test rig. The control of the pre-test and the validation test is almost identical; the same algorithm is used to control both tests, working with real sensors in the validation test and with simulated sensors in the pre-test. The pre-test provides an opportunity to investigate the test setup, and if necessary alter the design. The pre-test model is shown in Figure 2. In this paper the pre-test model is used to investigate different modal filters, and the correlation of the strain in the anti-roll bar is used as a measure of the test rig's performance.

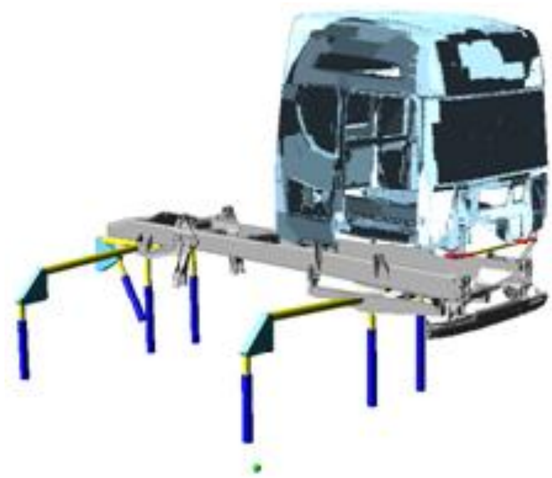

Figure 2: The simulation model used for pre-testing of the cabin.

3 TEST RIG CONTROL

The damage on the test track is not quantified, but it is believed to be closely correlated to the measured signals. In particular, it is believed that if the test reproduces the measured signals it also reproduces the damage. It is quite possible to check how well the test rig reproduces the reference strain in some points, and the reference measurement on the anti-roll bar is used for this purpose. However, because the test setup is different than the reference setup on the test track, it cannot reproduce the reference signals perfectly. The differences consist of both missing structure and missing excitation sources.

The test rig signals, $a_1(t)$ … $a_{n_o}(t)$, are transformed using the discrete Fourier transform (DFT) and gathered in the matrix $\mathbf{A} \in \mathbb{C}^{n_o \times N}$, where n_o is the number of sensors and N is the number of frequencies in the DFT. In the same manner the actuator forces are gathered in the matrix $\mathbf{F} \in \mathbb{C}^{n_i \times N}$, where n_i is the number of actuators, $n_i \leq n_o$. A linear frequency domain model of the structure is used so that a frequency response function (FRF), $\mathbf{H}(\omega_k) \in \mathbb{C}^{n_o \times n_i}, k = 1..N$, is defined such that

$$\mathbf{A}(:,\omega_k) = \mathbf{H}(\omega_k)\mathbf{F}(:,\omega_k). \quad (1)$$

The setup of the test begins with a system identification procedure where the FRF is identified. Then the actuator forces are computed in an iterative way. The difference between reference signals and test signals is computed each iteration. This difference is then passed to the pseudo-inverse of the FRF, $\mathbf{H}(\omega_k)^\dagger$, to compute the force update.

3.1 Synthetic sensors and weighting

It is possible to create "new" or synthetic sensors, v_r, as linear combinations of actual sensors. For some components it makes sense to study the acceleration of the component relative to the chassis. The design of synthetic sensors, *i.e.* which signals to combine, is a choice of the test engineer that is based on experience and understanding of the dynamics of the structure, the procedure is illustrated in Figure 3.

It is also possible to assign different weights to different sensors. In some applications the test engineer may realize that a specific sensor, synthetic or real, is closer correlated to the damage process than other sensors. That is a typical situation when that sensor is given higher weight than other sensors.

Between iterations the force update is computed as

$$\Delta\mathrm{F}(:,\omega_\mathrm{k}) = [\mathrm{W}\Psi\mathrm{H}(\omega_\mathrm{k})]^\dagger \mathrm{W}\Psi\left(\mathrm{A}^{\mathrm{ref}}(:,\omega_\mathrm{k}) - \mathrm{A}^{iteration}(:,\omega_\mathrm{k})\right), \quad (2)$$

where $\mathbf{\Psi} \in \mathbb{R}^{(n_o+p)\times n_o}$, p being the number of synthetic sensors and $\mathbf{\Psi}_{1..n_o\times 1..n_o} = \mathbf{I}$. The diagonal matrix $\mathbf{W} \in \mathbb{R}^{(n_o+p)\times(n_o+p)}$ is the weighting of the channels. Further reading on test rig control is provided in [1-2].

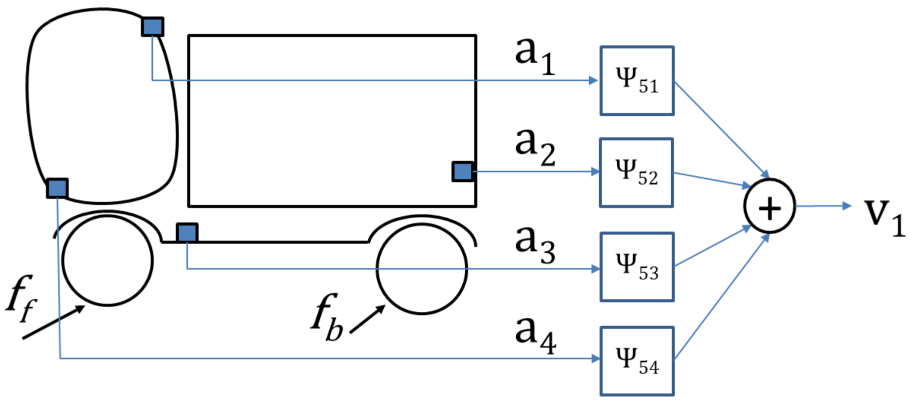

Figure 3: Schematic illustration of a synthetic channel, v_1, defined as a linear combination of four channels, $a_1 - a_4$, that are actually measured.

4 MODAL FILTER

Modal filtering was proposed by Meirovitch [3] as a method to control unstable eigenmodes. Ideally, a modal filter vector ψ_r is orthogonal to all but one of the mode shapes:

$$\psi_r^T \varphi_s = \begin{matrix} 0, & r \neq s \\ c_s, & r = s \end{matrix} \quad (3)$$

where φ_s is the mode shape of eigenmode no. s, and c_s is a scalar constant that depends on the scaling of the mode shape. In this discretized setup φ_s refers to the mode shape in the points where sensors are located. Thus, the length of both the mode shape vector φ_s and the modal filter vector ψ_r is n_o. Under the assumption of white noise excitation the modal filter vectors can be computed without knowledge of the excitation, using only the sensor signals.

For broad-banded excitation with a flat frequency spectrum there exist text-book methods for estimating the eigenfrequencies, ω_{ns}, and the modal damping, ζ_s, of the structure, [4]. When these are known a normalized modal transfer function, G_s for mode no. s can be defined,

$$G_s(\omega_k) = \frac{2\zeta_s\omega_k^2}{\omega_k^2 - \omega_{ns}^2 - i2\zeta_s\omega_k\omega_{ns}}. \quad (4)$$

The method outlined in [5] is based on the assumption that the excitation is broad-banded flat noise, that the structure is linear, and that the normalized modal transfer functions, Eq. (4), are known. Under these conditions the measured accelerations are

$$A(:,\omega_k) = \sum_{s=1}^{R} \varphi_s G_s(\omega_k) d_s, \tag{5}$$

where R is the number of modes in the modal sum and d_s is a constant that depends on the sensitivity of the eigenmode s to the excitation. Multiplication of Eq. (5) with a modal filter vector gives

$$A(:,\omega_k)^T \psi_s = G_s(\omega_k) c_s d_s. \tag{6}$$

Thus for $R \leq n_o$ the orthogonal modal filter vectors can be estimated from the measured data by solving the inverse problem Eq. (6) for some frequencies in the vicinity of the eigenfrequencies of the first modes.

$$\psi_s = (A^T)^{\dagger} G_s. \tag{7}$$

Eq. (7) describes a straightforward method to compute the modal filter vectors. These vectors are then orthogonalized using the Gram-Schmidt procedure.

5 RESULTS AND DISCUSSIONS

The performance of the test rig is evaluated using an error measure of the strain in the anti-roll bar,

$$e = \frac{\sum_k \left(\varepsilon(t_k) - \varepsilon^{ref}(t_k)\right)^2}{\sum_k \left(\varepsilon^{ref}(t_k)\right)^2}, \tag{8}$$

where ε^{ref} is the strain in the anti-roll bar from the test track, and ε is the corresponding quantity in the test rig. When no synthetic channels are used, p = 0, and all channels are given equal weight the error after converged iterations is e = 0.82. Time domain sequences of the strain signals are shown in Figure 4, the data displayed is centred on the peak strain recorded.

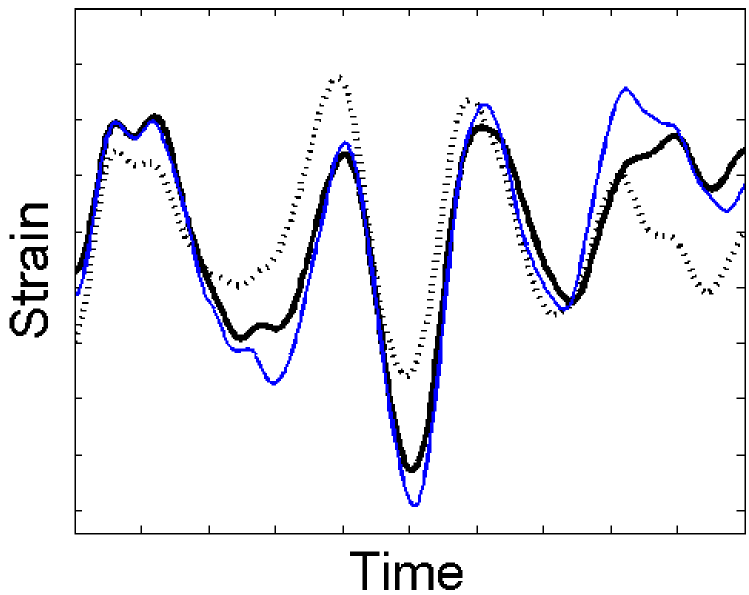

Figure 4: Strain in the anti-roll bar, the time window shown is about 1 s. The measured signal - solid black line, the test rig signal for p = 0 – dotted line, test rig signal for p = 7 – thin blue line.

The first seven eigenfrequencies are identified, ω_{n1} ... ω_{n9}. Together with damping estimation they are used to compute modal filter vectors using Eqs. (4, 7). The test rig was reiterated using p = 1…9 and with 4 times larger weight on the synthetic channels than on the original channels. The error of the strain in the anti-roll bar is reported in Table 1.

It can be observed that the first modal filter vector has a moderate impact on the test accuracy. As the number of modal filter vectors increase the performance of the test rig improves up to six modal vectors. For more than seven modal filter vectors the accuracy deteriorates.

The reduction in strain error due to the modal filter can be understood as follows; the cabin suspension is mainly stressed by low frequency motion of the cabin relative to the chassis. This motion is better captured by the modal filter based on low eigenfrequencies than by individual sensors. The higher order modes, filtered with v_7, v_8, and v_9, contribute less to the strain in the cabin suspension. It is noted that ω_{n6} corresponds to an eigenfrequency of about 3 Hz. It is also observed from the measured data that most of the energy in the strain signal lies in the frequency interval 0.5 – 4 Hz.

No. of modal filter channels p	Relative error e
0	0.82
1	0.78
2	0.66
3	0.68
4	0.60
5	0.49
6	0.31
7	0.31
8	0.40
9	0.41

Table 1: Strain error for different number of modal filter channels.

Traditionally a manual optimization of the channel weighting is performed by an experienced validation engineer. The manual optimization results in larger weighting coefficients for the sensors located on the cabin, in particular those directed in the vertical and lateral directions, and this reduces the error to e = 0.19. The manual optimization does not use the modal filter channels but relies on the experience of the engineer.

An important property of the modal filter method is that it relies on very little manual input and can thus be automated. It is possible that better results can be achieved with a more sophisticated estimation of the modal filter vectors, for example using an FE-model of the complete vehicle. However, creating an FE-model of a complete vehicle with sufficient accuracy requires many man-hours.

6 CONCLUSIONS

The results show that modal filters computed exclusively from output data can improve the accuracy in vehicle component testing significantly. The filtered problem remained stable and the test accuracy was on an order comparable with that of an experienced validation engineer.

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