# Virtual test rigs as a tool to reduce the time to market

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*Abstract: In order to reduce the time to market with maintained or improved quality, virtual test-rigs is emerging as an important technology in the commercial vehicle industry. Development testing consists of forcing a system B (a physical or virtual test rig) to behave like system A (a vehicle on the test track). This is achieved by choosing the "best" input signals to system B in a process generally referred to as "drive signal iteration". It is a classical inverse problem of a multivariable function where the key challenge is to get a good estimate of system transfer function. This paper describes how Abaqus may be used for this process. Following the drive signal iteration the Femfat fatigue post-processor is used to evaluate the severity of the experienced loads on a component level, i.e. the lifetime. The benefit of this exercise is that drive signal iteration and fatigue damage evaluation takes only a couple of days using a virtual test rig, while testing to failure of a component in a physical test rig may take several months. Virtual test rigs exploit the measured accelerations and strains from the test-track in a faster way than do physical test rigs, thus allowing for more construction loops and shorter time to market.*

*Keywords: Dynamics, Virtual testing, Fatigue*

## 1. Introduction

Physical testing has been a crucial part of the product development in the heavy vehicle industry for as long as trucks have been manufactured. In its simplest form it consists of driving a truck on public roads while observing the failures that possibly occur. A real break-trough in testing technology occurred in the 1960's when cheap servo hydraulic equipment became readily available. Due to this advancement it became interesting to move a large part of the testing away from public roads and test tracks and into the lab. The standard procedure today is to equip a vehicle with sensors (accelerometers and strain gauges), and drive it on a test track a few times while making sensor recordings. Thereafter the testing is continued in the lab by placing the vehicle or a subsystem thereof in a hydraulic test rig. The hydraulic actuators of the rig are moved in such manner that the sensor signals that were recorded on the test track are reproduced in the test rig. Then, any failures that occur in the test rig are recorded and actions of improvement may be undertaken.

Considering that the typical life-time of a truck is more than 1 million kilometers which corresponds to approximately 20000 hours of driving one realizes that testing on public roads inevitably takes a long time. Test track driving is less time consuming because test tracks are generally designed to treat the vehicle roughly, but still prohibitive. Thus, if testing is done on tracks the design engineers have to wait a very long time before they get direct feedback on their work and design-loops can not be numerous. With hydraulic test rigs this feedback-time is reduced considerably. The typical time to failure in a test rig is a few months compared to the more costly test track driving that takes more than a year.

The next evolutionary step in testing is virtual testing. The concept of virtual testing consists of using computer simulations with FEM or FEM/MBS simulating some relevant physical test. In theory, computer simulations have the potential of decreasing the test time significantly, particularly so when high cycle fatigue is studied.

In the heavy vehicle business today, the time to market is one of the major concerns for product development, obviously competing with the quality of the product. One of the limiting factors is the time spend in test rigs. Therefore it is of interest to use virtual testing instead of, or in parallel with physical testing.

Scania CV AB has developed a virtual test procedure described in this paper. It aims at reducing the time from the point when sensor signals have been measured on the test track to the point when failure is observed. The tool for doing so is a finite element model of a test rig with which dynamic simulations are performed. The driving displacements of the model are iterated in a way similar to that of a physical test rig in order to reproduce some sensor signals that were measured on the test track. Thereafter fatigue post-processing is performed on the stress histories obtained from the dynamic simulation. Instead of waiting for months for the physical component to fail in a test-rig an early response is obtained from the virtual test after about a week. Other advantages are better understanding of motion and force-paths and a fast and simple tool for evaluating design changes. There are several other automotive and test equipment manufacturers that do develop virtual test rigs and publish reports on this topic, see for example [1-3], although it is difficult to estimate the current importance of the method.

## 2. Methodology

This paper has implications on the product development process after the first prototype has been manufactured to the point where mass production is decided upon. The development process mentioned in the introduction is schematically visualized in Figure 1. In the here proposed methodology the “test rig” in Figure 1 is replaced by a virtual test rig (VTR) thereby reducing the time spend in testing considerably. It also introduces a natural way of evaluating minor design changes in new prototypes.

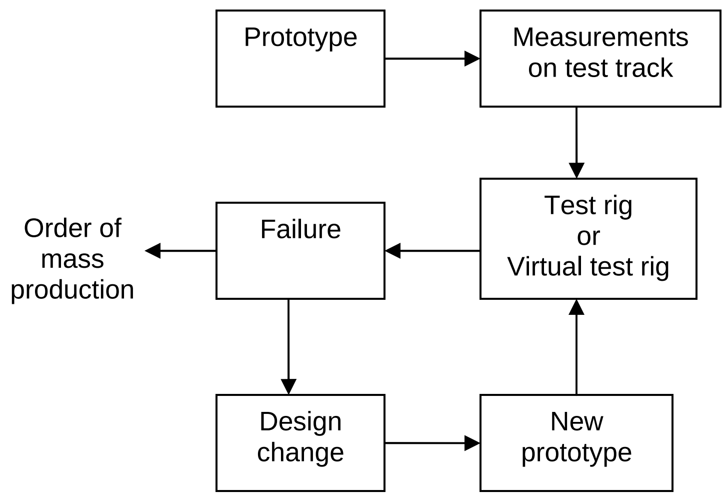


Figure 1. The product development process.

The VTR works very much the same as a physical test rig. The drive signals are iterated to produce response signals that correlate well with the signals measured on the test track. Inputs to the VTR are the measured test track signals and a finite element model. Output is a life time prediction. The basic functioning of the VTR is described below:

1. System identification in order to obtain a transfer function
2. Inversion of the transfer function
3. Initial guess of driving displacements
4. Dynamic simulation with driving displacements
5. Computation of the error on simulated signals, if the error is sufficiently small: jump to 8
6. Derivation of a new set of driving displacement by consideration of the error in simulated signals
7. Loop back to 4
8. Life time prediction based on the stress histories

The basic assumption of rig testing, physical or virtual, is that if the system in the test rig produces signals that are equal to or close to the signals measured on the test track, then the motion and thus the strain history in all points are close to or equal to the strain history on the test track. This assumption is not entirely correct since the systems in the test rig and on the test track in general are not identical, in particular so for virtual testing. However, it is out of the scope of this article to treat this complex question but it should be remembered that model errors may have a large impact on the usefulness of the method. In the following it is assumed that the model errors are small.

It is then of primary concern to produce output signals that correlate well with the measured signals in order to obtain correct strain histories. Which type of software used for this purpose is of secondary importance and there are several options available. Multi body simulations with flexible bodies created in FEM are frequently used for dynamic simulation of large vehicle systems. However, one may equally well envisage using FEM software also for the dynamic simulation; it is essentially a question of what is most practical. In this paper the dynamic simulations are performed in Abaqus [4] because it was the simplest way to profit from existing model libraries and engineering expertise.

Because of limitations in fatigue post processing it is currently unfeasible to handle very complicated stress histories for large models. (Example of an unfeasible task is $10^6$ DOF with $6 \cdot 10^4$ distinctly different stress states, corresponding to a modest sized model, 1 minute of simulated test track at 1 kHz sample rate). A feasible way to handle that same problem is to use modal stresses of 20 eigenmodes in combination with the time histories of the modal coordinates. This is the most common way to use VTRs although one is then limited to locally linear structures.

The two most critical steps in VTR are the first step (system identification) and the last step (fatigue post processing). In order to get a good correlation between measured and simulated signals it is critical to get a good estimation of the system's transfer function. This is generally much more difficult than to implement the dynamic simulation with acceptable accuracy. Furthermore the final step of fatigue post processing involves complex fatigue damage accumulation theory which at its current state introduces large uncertainties. These two challenging steps are discussed in some detail in the following.

### 2.1 System identification

A key challenge is to obtain a good estimate of the systems true transfer function *H*. Here the set of driving displacements are denoted *X* and the set of ouput signals denoted *Y*. With the discrete Fourier transform denoted $\Im(\bullet) \equiv \hat{\bullet}$ and assuming that the transfer function may be linearized one may write for each discrete frequency $\Omega_k$

$$\begin{bmatrix} H_1^1(\Omega_k) & \cdots & H_1^j(\Omega_k) \\ \vdots & \ddots & \\ H_i^1(\Omega_k) & & H_i^j(\Omega_k) \end{bmatrix} \begin{bmatrix} \hat{X}_1(\Omega_k) \\ \vdots \\ \hat{X}_j(\Omega_k) \end{bmatrix} = \begin{bmatrix} \hat{Y}_1(\Omega_k) \\ \vdots \\ \hat{Y}_i(\Omega_k) \end{bmatrix} \tag{1}$$

Eqn (1) is the fundamental relation that physical and virtual test rigs exploit, first to identify the transfer function *H* and then to compute the driving displacements *X*. It may be tempting to solve Eqn (1) directly for each discrete frequency in order to estimate *H*, but in practice the estimate thus obtained is generally poor. For physical testing the direct solution is often poor due to noise in measurements, limited signal treatment resources, and non-linear system behavior. In virtual testing some challenges for signal theorists remain even for linear systems, in particular when the simulated times are short and the signal is non-stationary. One way to overcome some of these difficulties is to update the system transfer function *H* between iterations.

However, for a linear system projected to modal coordinates the transfer function may be expressed analytically in terms of the modal properties. For example the receptance (displacement over force) is computed as

$$H_j^i(\Omega_k) = \sum_r \left[ \phi_r^i \frac{1}{m_r} \frac{1}{\omega_r^2 - \Omega_k^2 + i2\zeta\omega_r\Omega_k} \phi_r^j \right] \qquad (2)$$

where $\phi_r^i$ is displacement no. i in mode shape no. r, and $m_r$, $\zeta_r$ and $\omega_r$ are the modal mass, modal damping and angular frequency of resonance, respectively.

Hence, if a good estimate of the systems modal properties is available then the transfer function is known by Eqn (2). In practice one way to estimate *H* is to compute the modal properties in a frequency extraction in Abaqus and then compute Eqn (2) separately. Alternatively, it is possible to compute Eqn (2) with Abaqus using "*random response" or "*steady state dynamics".

Only for non-linear systems should one have to resort to system identification techniques used in experimental research. Suitable for (mildly) non-linear systems would be to assume almost linear behavior and identify the best-fit modal parameters with curve-fitting of polynomials in Ω. Note that Eqn(2) may be written in polynomial form

$$H_j^i(\Omega_k) = \frac{p(\Omega_k)}{q(\Omega_k)}, \qquad (3)$$

and it is possible to fit the polynomials *p* and *q* to the (noisy) transfer function *H* obtained by subjecting the structure (in the time domain) to a broad banded signal. However, it is quite possible that better results would be obtained by using the "*steady state dynamics, direct" option in Abaqus (this is unknown to the authors).

## 2.2 Life time prediction

Life time prediction is determining the number of repeated loadings that a structure will withstand before failing. As such it is assumed that the stress or strain history is known from previous analyses. For high cycle fatigue the load is most often expressed in stress and a linear damage accumulation is assumed. In the simplest form the damage $D_k$ of an individual load cycle *k* is given by

$$D_k = A\left(\sigma_e^k\right)^m, \qquad (4)$$

where $\sigma_e^k$ is the amplitude of some stress measure of load cycle $k$, $A$ is a material constant and $m$ is the fatigue exponent (dependent of material and geometry). The damages from individual load cycles are summed up and when the sum is greater than 1 failure is assumed to occur.

$$D = \sum_k D_k = A \sum_k \left(\sigma_e^k\right)^m \tag{5}$$

There are several difficulties here, for example; how are load cycles computed, what stress measure should be used and what test data should the material parameters be determined from.

Fatigue is fundamentally a complex process with crack initiation and crack growth as governing mechanisms. The way in which cycles are counted is strongly linked to which effective stress measure that is used, but the most prominent method is rainflow counting. There are numerous effective stress measures available in the literature, some do and some do not use the influence of stress gradient. Different stress measures require a different set of experimental data for determination of the material model parameters. An excellent review of some fatigue relevant stress measures is available in [5] where performance issues also are treated. Finally, there is quite a lot of research that indicates the existence of sequence-effects, i.e. if the load history contains cycles of different amplitudes it matters in which order the cycles are applied. This research implies that linear damage accumulation is an (over) simplification. For an extensive treatise on the topic of fatigue the reader is referred to [6].

In conclusion fatigue theory is still rather undeveloped from an engineering perspective and the mean life-time, even in simple controlled experiments, is difficult to estimate within a factor 2.

Despite these difficulties there exist several commercial fatigue post-processors, for example Femfat or FE-fatigue, that do implement elaborate variants of Eqn (4) and cycle counting in order to compute life time estimates from stress and strain histories for large complex structures. Fatigue properties of the materials are defined by supplying fatigue test data of material specimen in a predefined test matrix. The post-processor utilizes a stress field obtained from FE calculation and computes an effective stress (or strain) measure $\sigma_e$ in every node (or element). In combination with cycle counting this allows for life-time prediction of all or some nodes or elements in a finite element model. Exactly what stress measure that is used and how the parameters are determined is kept as business secrets.

The working procedure is to supplement modal stress vectors, modal displacements and material test data to the fatigue post-processor and obtain a life time estimate. Output from the fatigue post-processor is obtained both as an estimated life time for the structure and a result-file (in Abaqus .odb-format if that is desired) with the distribution of accumulated damage.

## 3. Implementation

There is currently no commercially available program that implement a VTR with the eight steps previously discussed (at least not known to the authors). LMS virtual labs which runs well with

Adams multi body tools [7] does implement the first seven steps but then an Adams model of the test rig has to be provided. Here a separate suite of programs/scripts have been developed in order to automatically link the necessary eight steps. The backbone of the suite is a Matlab [8] program that as inputs accepts an Abaqus model of the test rig with driven dofs specified and measured sensor signals with their corresponding location in the model.

The program prepares the Abaqus model and starts a steady state dynamics analysis by which an estimate of the transfer function is obtained. Thereafter the transfer function is inverted and the iteration loop is started. In each iteration the program launches Abaqus with a dynamic simulation (currently only *modal dynamic) and extracts the results in the sensor nodes or elements. The new drive signals are computed with the formula

$$x_j^{iter+1} = x_j^{iter} + \Im^{-1}\left(H^{-1}\left[\hat{y}^{measured} - \hat{y}^{iter}\right]\right) \tag{6}$$

When satisfactory convergence is reached the loop ends and a final Abaqus run is launched where data necessary for fatigue post processing is computed. The final result file is sent to Femfat [9] for fatigue damage computation where the output is presented both as a scalar value of expected life time of the structure and as an Abaqus formatted result file with the distribution of accumulated damage.

The implementation is schematically visualized in Figure 2.

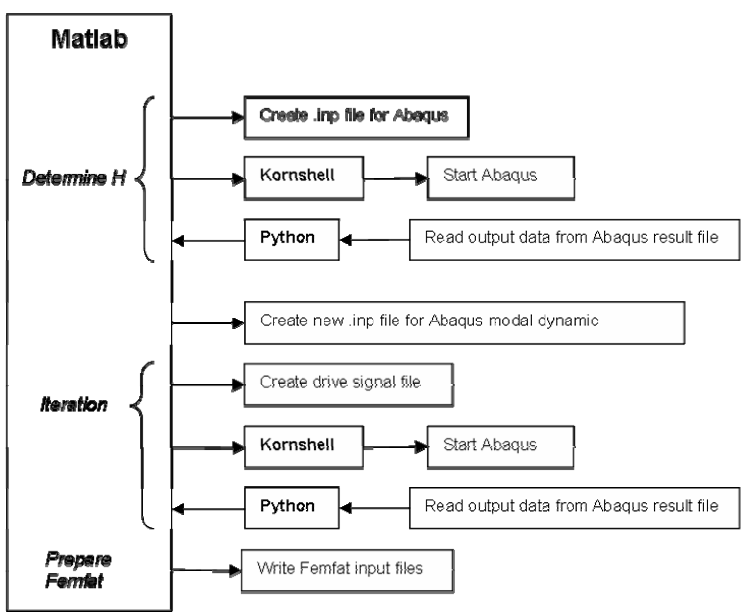


Figure 2. Schematic of the eight steps in the VTR.

### 3.1 Features

In physical testing as well as in virtual testing it is of interest to adjust the relative importance of different sensors. This is optionally done by providing scalar values acting as weights on the error of different channels.

Because fatigue post-processing of large models demands quite a lot of computer resources a selection feature has been developed which allows restricting the fatigue estimate to a subset of the model.

### 3.2 Limitations

The VTR is currently only implemented for linear models mainly due to the difficulty of treating linear super-elements in Abaqus and in Femfat. The desired mode of work would be to combine linear super-elements with a limited number of eigenmodes (so that fatigue damage may be estimated) with non-linear connections, much like the workings of multi body programs.

## 4. Example of application

In the development of a new mudguard bracket the initial design was prototyped. The prototype was equipped with strain gauges and mounted on a test vehicle that was driven on the test track. Sensor recordings were thus made available at the positions A and B indicated in Figure 3. A dual-channel virtual test rig was decided upon with one longitudinal and one vertical driven displacement, thus the system is quadratic with the same number controlled dofs as available sensor recordings. In general it is preferred to use over-determined systems but in this case both the longitudinal and the vertical forces were considered important and only two sensor recordings were available.

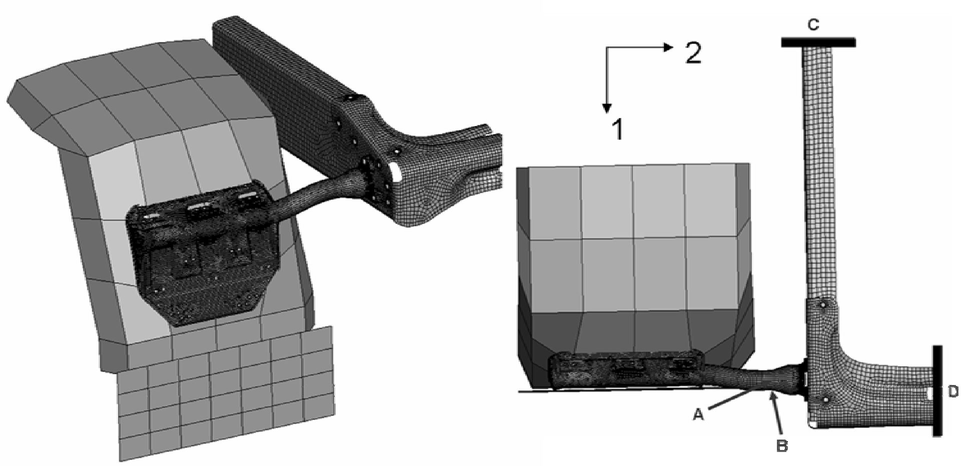


Figure 3. Virtual test rig for the mudguard bracket, strain gauges are positioned in A and B and displacement is forced at C and D in the 1- and 3-direction, respectively.

Because the system is quadratic it is possible to obtain a perfect match between measured and simulated sensor signals. For an over-determined system a good match on all sensors would indicate that the model agrees well with the physical structure. For quadratic systems this is not so, because it is always possible to find a set of drive signals that produces matching sensor signals. The quadric form makes the iteration procedure very rapid such that already after the first iteration a perfect match is obtained. The two most important natural modes are schematically depicted in Figure 5. The stress fields of all the retained modes (numbering 20 in total) together with the modal displacements are processed in Femfat. The result is an average life-time normalized to the design target of 450 per cent, and a fatigue damage plot showing the damage distribution, Figure 4. The calendar time from measurements on the test track to this result was approximately one week.

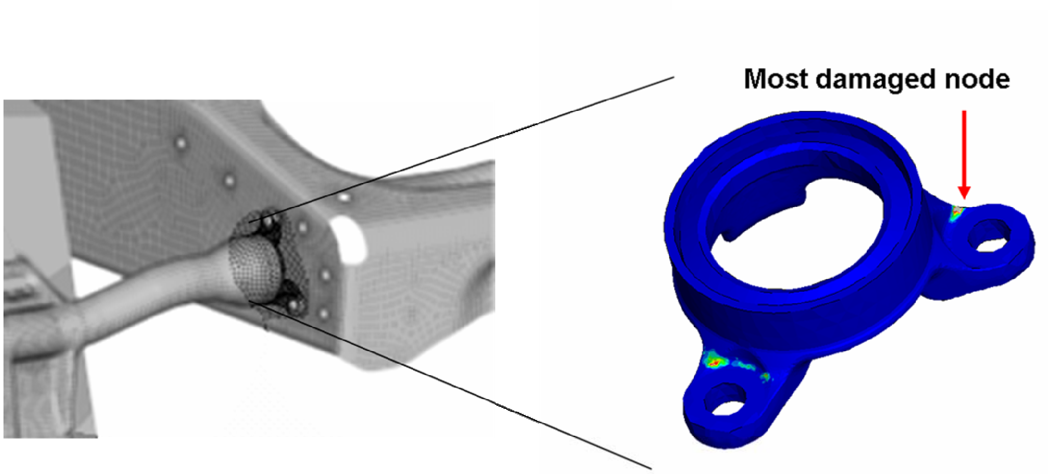


Figure 4. Damage distribution in the forged piece closest to the frame with the most damaged point indicated.

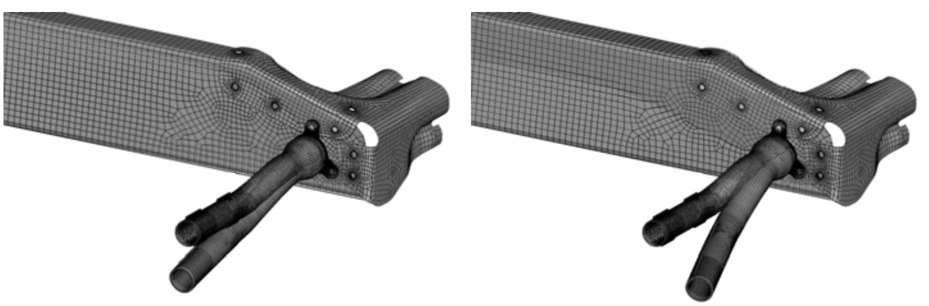

Figure 5. Illustration of the two modes that, when regarded individually, create the most damage.

Because of the important uncertainties associated with virtual testing and fatigue estimates it is reasonable to demand a larger safety factor than 4.5. The model uncertainties will naturally vary between components and so should the safety factor. Knowledge of appropriate safety factors for virtual testing is currently quite limited. In this particular case it was rather simple to improve the design by modifying the forged part of the bracket where maximum damage was estimated to occur.

To this extent a modified bracket was designed with reinforcements in the weak points as indicated in Figure 6. Assuming that the dynamic properties of the structure are unchanged it may be appropriate to use the same driving displacements in the virtual test rig as for the original bracket. Thus, no iterations were performed for the modified structure but the last set of driving displacements obtained with the original structure was used for excitation. When the modal stress vectors and displacements are post-processed in Femfat the average life time is estimated. According to this estimation the modified bracket will exceed the design target by a factor 50.

Figure 6. To the left: the original version of the forged piece, to the right: the modified reinforced version.

A separate test was performed in a physical test rig. The physical test rig incorporated much more of the truck including the entire frame, the cab, fuel tanks and other auxiliary equipment. It was a seven channel rig with 24 sensor points including those on the mudguard bracket. The outcome of the test for the original design was a life-time of 95 per cent normalized to the design target, Figure 7.

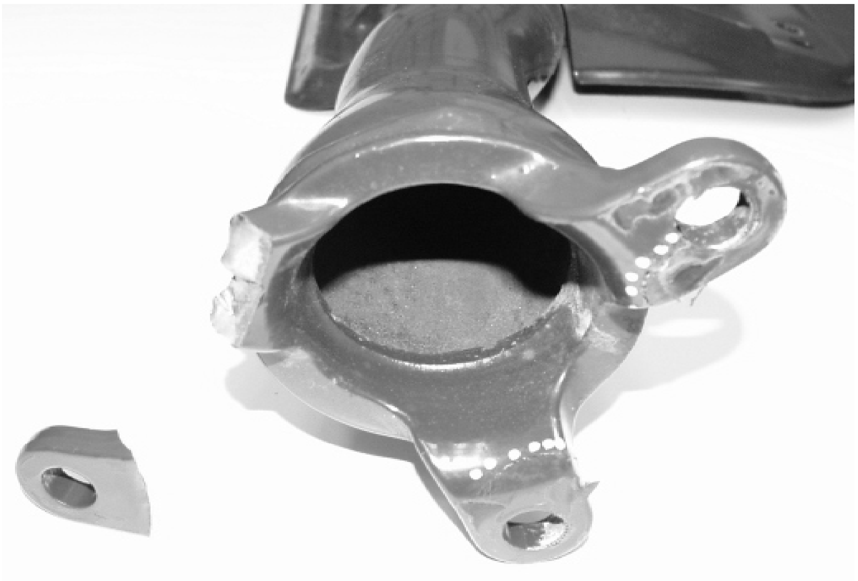

Figure 7. Photograph of the failure that occurred in physical testing after 95 per cent of design target completion.

Note that this result was obtained approximately 20 calendar weeks after the measurements on test track were available. Only one component was tested so nothing is known about the scatter certain to be present. However, the error of the virtual test rig in this case in the order of 5, gives insight for future safety factors. The modified design has at the time of writing been tested to 200 per cent of design target without failure.

Many reasons may be put forth to explain differences in result between this particular VTR and the physical test. A list of probable influence factors include

- Different rig designs means that different motions can be simulated
- The physical test rig is over-determined and does not show perfect correlation with test track measurements in the sensor points
- Poor fatigue damage theory
- Insufficient material data
- Discrepancy between drawing and physical component
- The failure occurs close to a bolt and the bolt model is grossly simplified with a beam spider
- Residual stresses from forging not accounted for

The fast feedback obtained from virtual testing allows for design changes early in the design process. In this way the work of introducing modifications of the design may be undertaken several months earlier than would be the case if the failure in the physical test rig was the only verification. Thus the time to market may be reduced with maintained quality without increasing the work load on design departments.

## 5. Conclusion

The methodology of virtual testing that has been outlined in this paper has already produced valuable results and product impact. Although the errors compared to physical testing appear significant they do not compromise the use of VTR for product development. Perhaps the error in the VTR can be reduced to a factor 2 with existing methods. However, because even very small design changes may have great impact on the fatigue life; methods with errors in the order of 10 are still useful in the early design process. The main advantage of virtual testing as it is implemented here is that it is faster and cheaper than physical testing. As such it provides an important tool for product development at Scania CV AB. In the foreseeable future it may also be envisaged to use virtual testing for determining which test vehicle that presents the most severe loads for a particular component.

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