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FATIGUE SIMULATION OF POWER TRAIN COMPONENTS DURING THE DESIGN PROCESS

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ABSTRACT−The lifetime of power train components can be improved dramatically by finding crack initiation points with suitable software tools and optimization of the critical areas. With increasing capacities of computers the prediction of the lifetime for components by numerical methods gets more and more important. This paper discusses some applications of the outstanding fatigue simulation program FEMFAT supporting the assessment of uniaxially and multiaxially loaded components (as well as welding seams and spot joints). The theory applied in FEMFAT differs in some aspects from classical approaches like the nominal stress concept or the local one and can be characterized by the term "influence parameter method". The specimen S/N-curve is locally modified by different influence parameters as stress-gradient to take into account notch effects, mean-stress influence which is quantified by means of a Haigh-diagram, surface roughness and treatments, temperature, technological size, etc. It is possible to consider plastic deformations resulting in mean-stress rearrangements. The dynamic loading of power train components is very often multiaxial, e.g. the stress state at each time is not proportional to one single stress state. Hence, the directions of the principal axes vary with time. We will present the way how such complex load situations can be handled with FEMFAT by the examples of a crank case and a gear box.

Key words : Fatigue, Simulation, Power train

1. INTRODUCTION

The reduction of vehicle weight is at least since the drastic raise of fuel prices in the seventies a basic requirement for the development of new cars, trucks and busses. Nevertheless there is the general tendency in the opposite direction. The reasons for this contradiction are on the one hand additional features for more comfort and safety as well as increased motor torque. On the other hand the potential for mass savings in the basic structure is not enormous. For example the benefit of a magnesium transmission housing for a passenger car compared to an equivalent made of aluminium is just about 5 kg.

To use materials with high strength is one possibility to realise lightweight structures. In this case one must not forget that the sensitivity to notches raises generally together with the ultimate stress limit if dynamical load is applied. Vehicles have to withstand dynamical loads and their components have lots of notches. Several technological treatments with positive or negative influence to the life time usually are applied. How can engineers handle notch factors and all other important influences to the lifetime during the development process?

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Of course, a very good method is to perform fatigue tests with components or total vehicles. Many influences can be considered without any mathematical model, nevertheless the results have absolute accuracy. Unfortunately, experimental tests give only poor information about low stressed areas so it is very expensive and time consuming to minimise a components mass mainly by experiments. Therefore the Technologie Zentrum Steyr, a profit center within the Steyr-Daimler-Puch AG developed a software for the fatigue simulation of dynamically loaded components on the basis of Finite Element examinations. The name of this program is FEMFAT® (Finite Element Method + FATigue). It will be described briefly in the following paragraph.

2. BASIC FATIGUE ANALYSIS CONCEPT OF FEMFAT (http://www.femfat.com)

At each user selected FE-node a component S/N-curve is calculated. The computation of this local S/N curve starts with the well known S/N-curve data of the unnotched specimen and is influenced by the local properties and linear calculated FE-stress tensors for the static and dynamic portions of the loading of the component (Figure 1). The calculation methods used for the assessment of

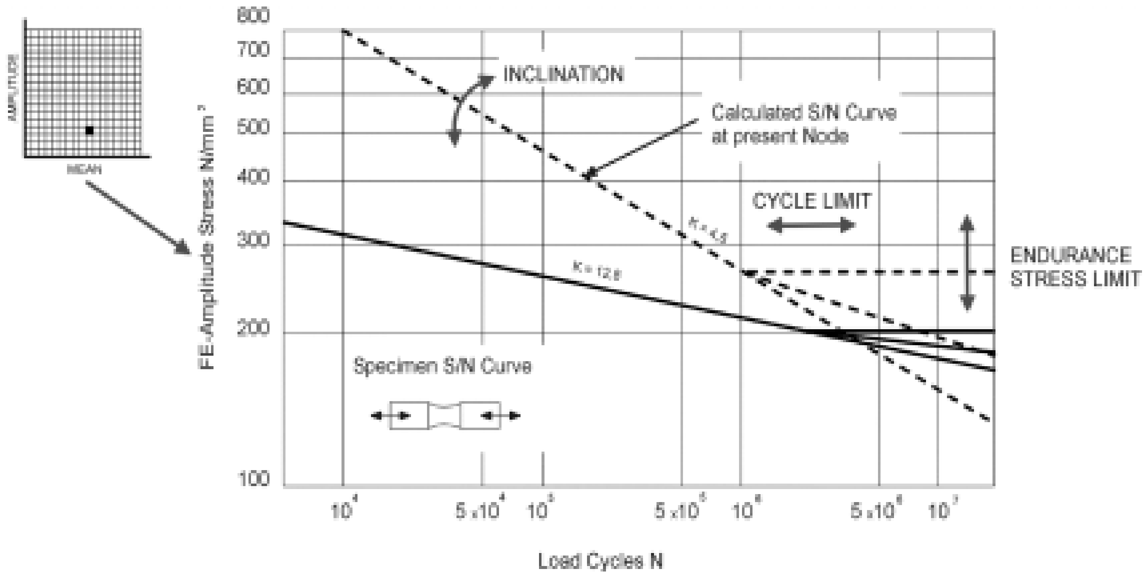


Figure 1. Modification of specimen to calculated component S/N curve.

FE-results in FEMFAT, can be allocated to the "B-Method". That means the used methods are based upon experimental and theoretical research results as well as on tests on special test structures and the experience gained in the operation. The cardinal procedure is based upon the Influence Parameter Concept described in (FKM-Richtlinie, 1994).

When applying this concept, a consequently distinction is made between amplitude-, mean-, and constant stress tensors. The definition of the S/N-curve-model in the double logarithmic coordinate system is that the linear damage accumulation is always done only with the FE-amplitude stress tensor of the current hysteresis. All other influences on the local operating strength of a component like stress distribution, notch effects, size, mean stress, mean stress rearrangement due to local plastification, residual stresses, temperature distribution, quenching and tempering conditions, surface influences like roughness, edge hardening by mechanical, chemical or thermal treatment, etc. are taken into account by a locally modified S/N-curve. Generally the endurance limit of the S/N curve is calculated as well as the slope and endurance load cycle limit of the straight line model. The concept of synthetically created S/N-curves was modified (e.g. Kt-free calculation of notch effects and more flexibility of the formula parameters) and extended by influence parameters described in references (FKM-Richtlinie, 1994).

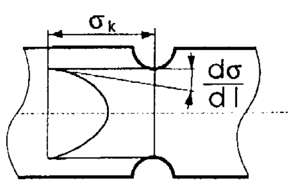
2.1. KT-Free Estimation of Component Notches

The classic procedure using notch factors defined over a stress ratio between linear notch stress and nominal stress is not possible due to usually non definable nominal cross sections. An estimation of the notch influence factors by the engineer (from component tests or based on his experience) would be possible, but is absurd with respect to the functions of FEMFAT in the process of component development.

Therefore a procedure is used which is based on the relative linear FE-stress gradient (see Figure 2), that represents the local geometrical conditions ("notch form") and the material behaviour. The method was verified by many specimen tests and has been used for many years in daily practice (Unyer, *et al.*, 1996). With this method it is possible to calculate the influence of any notch geometry in combination with any power strain.

2.2. Influence of Mean Stress

For the damage effect of hysteresis it is important to know the amplitude and the static position of the hysteresis. To consider the multiaxial interaction between amplitude- and mean stress tensors a critical plane method in combination with an uniaxial Haigh-diagram is used. Material specific differences between ductile and brittle materials are automatically considered by the use of the ratios between tension endurance limit/torsion



Stress Gradient:

$$\chi = \frac{\partial\sigma}{\partial l}$$

Relative Stress Gradient:

$$\chi' = \frac{1}{\sigma_k}\frac{\partial\sigma}{\partial l}$$

Figure 2. Definition of the stress gradient.

endurance limit and tension yield stress/torsion yield stress at equivalent stress calculation. If local stress peaks occur in the loading pattern of a component in the elastoplastic area of material, then mean stress tensors are repositioned by using the Neuber formula in the cyclic stabilised σ/ε-Diagram. This meets the arise of local residual stresses which may have a positive or negative effect to the life time of a component.

2.3. Multiaxial Loading

The module FEMFAT-MAX has been developed for the fatigue analysis of MultiAXially loaded structures. This tool helps development engineers to find the optimum dimension of a component according to extremely complex dynamical loads. Extremely complex means that the load history cannot be divided into one mean stress, one constant stress and one stress amplitude information.

An engine block, for example, is loaded in different directions at several locations. The relating forces effect the structure dynamically at the same time with different load histories. We call each force together with its time history a *channel*. This means the stress tensors of a channel corresponds to a certain FE-load case, for example the gas forces, while the history file shows the change of theses stress tensors in respect to time. FEMFAT-MAX can work with any number of different dynamic channels in one session, if there are enough resources in the hardware and operating system-software. As an option the load can be defined as a TRANSIENT computed sequence of stress tensors.

All load and history information is combined using an advanced method of the critical cutting plane approach. This method was derived from the procedure introduced in the concept for mean stress influence for the multiaxial combination of stress amplitude tensors and mean stress tensors. To compute the interaction between all channels, in a series of cutting planes the whole stress information is superposed, transformed to an equivalent stress and ranged into classes before the damage calculation starts. The results are damage values for all relevant cutting planes, which can be selected automatically by filters, while the highest damage value is selected to be the most critical in a certain plane.

The load history can be received by different ways, e.g. measurement or simulation with MBS (Multi Body System), such as ADAMS.

3. FATIGUE ANALYSIS OF A POWER DIVIDER

FEMFAT also provides a powerful tool for the computation of ultimate loads and safety factors against break respectively. For example, the housing of a new power divider developed in Steyr/Austria in 1997 was dimensioned against break at ultimate input torque since by experience this is the most critical load situation. The power divider is applicable for a great variety of heavy duty and all terrain vehicles up to 78 tons.

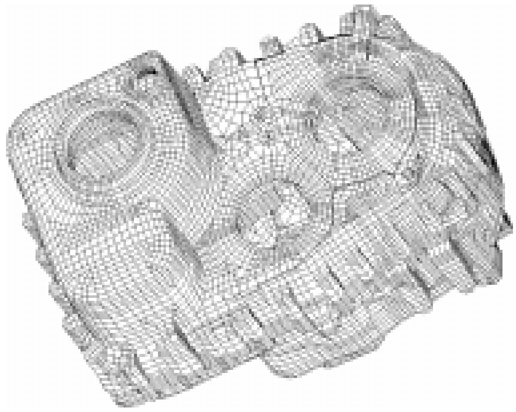


Figure 3. High-quality hexaeder mesh of a gearbox.

This example, however, should demonstrate the importance of stress analysis with common Finite Flement solvers which is the basis for the numerical fatigue analysis (Steiner, *et al.*, 1999).

3.1. Nonlinear Stress Analysis

Figure 3 shows a high-quality hexaeder-mesh of the gearbox. However, for the computation of realistic stress results the mesh quality is not the only point to be taken into account.

It was found that the stresses and deflections of critical regions are strongly influenced by the nonlinear behaviour of the tapered roller bearings. Thus, gap elements were used for modelling these parts. To highlight the effects of the nonlinear bearing model, Figure 4 shows a comparison of the stress results obtained from the nonlinear and from

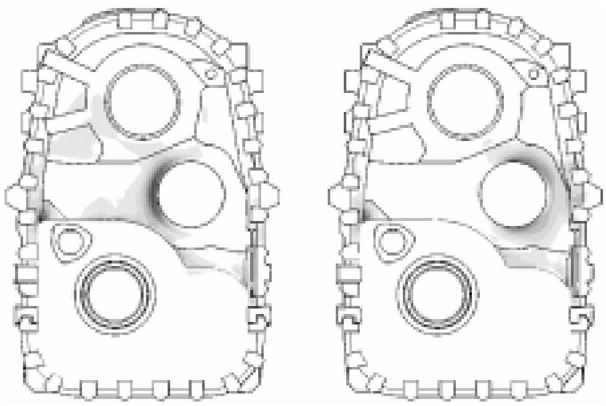


Figure 4. Stress results from FE-computations.

a linear computation.

In the linear case the bearings are able to transmit tensile loads which is not realistic. The comparision shows significant differences. The highest stresses occur in the vicinity of the bearings for the transmission shaft for a load case describing the conditions for backward motion off road.

The differences between the linear and the nonlinear results are greater than 100 percent in these domains. This observation shows clearly that nonlinear modelling of the tapered roller bearings is essential for stress assessment (and as a consequence for fatigue behaviour) of the gearbox.

3.2. Optimization Process with FEMFAT

The distribution of safety factors against break is shown in Figure 5. The plot provides most clear and compressed information for the fatigue assessment and indicates where improvements of the gearbox are required. In order to minimize weight it is reasonable to make the design stiffer only in those regions where it is necessary.

For the gearbox the safety against break could be improved significantly by executing just a single design loop. Figure 6 shows the improvement of the safety factors. The increase of the weight in the optimized structure is less than 3 percent.

The optimized gearbox was finally approved by computing the safety against endurance failure. For this task the input torque was applied as pulsating load. In this case the amplitude stresses are equal to the mean stresses. For each load case the input torque was chosen such that the minimum endurance safty factor is equal to 1. These are the maximum torques which the gearbox can endure. They are listed in the table below:

Description	maximum torque
off road, foreward motion	26 900 Nm
on the road, foreward motion	32 000 Nm
off road, backward motion	22 000 Nm
on the road,backward motion	30 500 Nm

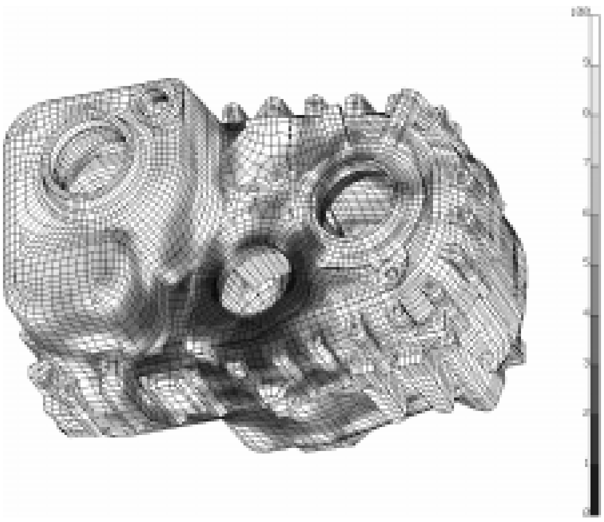


Figure 5. Safety factors against break.

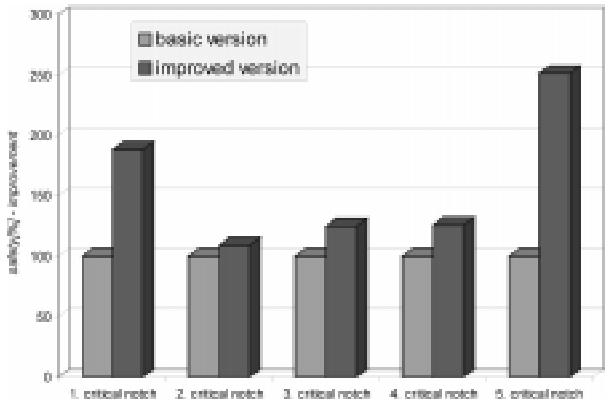


Figure 6. Improvement of safety factors.

It turns out that in foreward motion on the road higher input torques are possible than in backward motion off road. These features coincide with the demands on the gearbox in practice.

4. MULTIAXIAL FATIGUE ANALYSIS OF A CRANKCASE

The fatigue assessment of engine components such as crankcase (see Figure 7), crankshaft or conrod usually confronts the engineer with great problems. On the one hand, the loading and load distribution is often not known very well, on the other hand, there are lots of load states occuring during one engine cycle which have to be

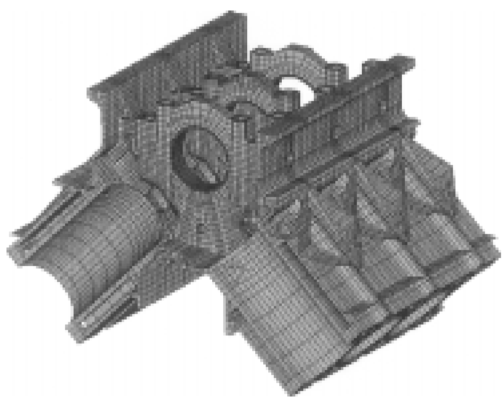


Figure 7. FE-model of a crankcase.

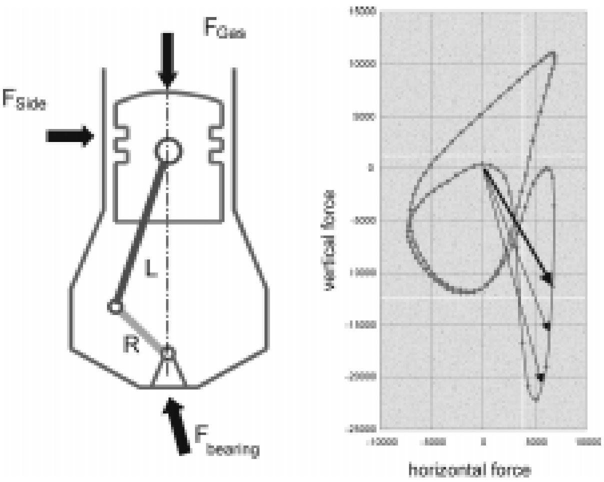


Figure 8. Loading of a crank case. The right picture shows a polar diagram of the main bearing resultant $F_{bearing}$.

analysed.

Clearly, each component of the stress tensor varies in a different way and the directions of the principal axes will not remain constant.

Strictly speaking, we have to deal with a transient fatigue process. In this section we demonstrate how to compute damage distributions for such complex load situations by means of FEMFAT-MAX.

4.1. Loading of the Crankcase

Dynamic loading of engine components results from gas- and mass-forces on the one hand, and from thermal cycles on the other hand. Here, we focus on the mechanical loading due to gas- and mass-forces which usually act millions of times during the life of an engine, whereas the engine will undergo just a few thousand thermal cycles.

Generally, the gas- and mass-forces yield the following loads in a crankcase (Figure 8):

· Crankshaft-bearing forces
· Balanceshaft-bearing forces
· Gas pressure on cylinder head acting on the crankcase via cylinder head bolts
· Side forces from the piston
· Inertia forces due to

For the computation of the main bearing forces several simulation methods are available. The most accurate results are provided by an elasto-hydrodynamic bearing analysis which takes into account the influence of the bearing deformation on the behaviour of the oil film.

Finally, the stress states for the loading for several consecutive positions of the crankshaft must be investigated. Usually a quasi-static stress-analysis is sufficient for a reliable fatigue analysis.

4.2. Elasto Hydro Dynamic Bearing Computation

Elastohydrodynamics deals with the problem of interacting elastic parts and hydrodynamic oil films. With the tool FEMFAT EHD it is possible to compute the loadings of plain journal bearings and generate a time history of these loadings.

The following influences are taken into consideration:

· Geometry of the bearing
· Arbitrary global loadings
· Oil data
· The flexible surrounding (FE-model)

The analysis is based on a quasistatic solution, vibration effects of the elastic structure are not taken into consideration. The geometry of the bearing can be arbitrary. There can be oil grooves, oil bores, unsymmetric bearing shells and boundary conditions like given oil pressure on an oil bore.

· The results of a FEMFAT EHD computation are:
· Dislocation orbits
· Oil film pressure dependent on crankangle
· Sommerfeld number
· Displacements of bearing shells
· Oil flow through bearings
· Time history of the bearing forces in every node of the FE-model

Among detailed information about the bearing properties, this data provides also the most advanced base for fatigue analysis of main bearings, crankshaft, connecting rod, etc., by the FEMFAT fatigue modules.

4.3. Fatigue Analysis using Femfat-Max

For the fatigue analysis of the crankcase the stress state is computed for rotation-intervals of 20 degrees. A preprocessing tool enables an automatic detection of the most critical state within each interval.

For a four-cycle engine we end up with 36 consecutive stress-states ($20 \times 36=720$) as the basis for the following fatigue computation.

Note that there exists a very simple and natural way

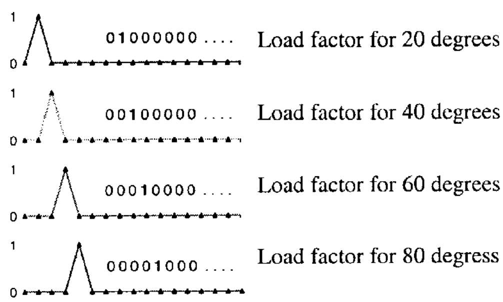


Figure 9. FEMFAT-MAX-channel definition.

how to analyse a complete transient process with the channel-conception of FEMFAT-MAX. One simply has to define a seperate channel for each stress state and a corresponding load factor equal to one if the state is currently active and equal to zero otherwise. In this sense the load factors are tent-functions (Figure 9) switching on and off the stress states in the correct chronological order. The load factor definition can be applied easily. However, in the current version of FEMFAT-MAX this method has been also implemented such that the loading factors for transient problems need not be defined any more.

Finally, the multi-axial fatigue analysis yields a damage distribution D_{cycle} for a 720-degree cycle. In order to obtain the fatigue assessment for a running engine the damage for one cycle has to be scaled. For example, for an engine that runs 400 hours at 7000 RPM we end up with

$$D_{400h} = D_{cycle} \cdot 400 \cdot 7000 \cdot 60/2$$

The factor ½ results from the four cycle process. Of course, the result of the fatigue analysis will depend on the step size of the time-discretization. For a 10-degree step size more accurate results will be obtained. However, the method should converge for an increasing number of stress states.

Figure 10 shows the computed damage at a few points of a three-cylinder crank case depending on the number of stress states. One can observe that the difference for 36 and 26 states is already zero for the first two points.

5. OPTIMIZATION OF A NEW MAGNESIUM GEAR BOX (AUDI MULTITRONIC CVT)

For the development of transmissions for high performance vehicles a lot of criteria have to be considered. This section discusses the step-by-step improvement of the stiffness and strength of the new AUDI MULTITRONIC (http://www.audi.com) with magnesium transmission case for passenger car engines of the upper power range.

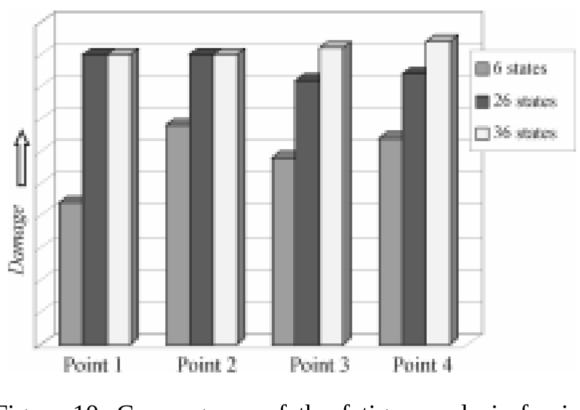


Figure 10. Convergence of the fatigue analysis for increasing number of investigated states.

The FEM & fatigue analyses and tests were realised in a co-operation of AUDI AG and the Technologie Zentrum Steyr TZS, a profit centre within Steyr Daimler Puch (Steinwender, *et al.*, 1998).

The strains in the flange area of the transmission/engine were measured in a vehicle during driving condition. Basing on the measurements a loading cycle was derived which was used for a laboratory test as well as for a fatigue simulation. The damage computation considers the interaction of the multiaxial loads.

5.1. Modelling Techniques

For the numerical investigations the gearbox including all cases, covers, bolts, bearings and shafts were modeled. The gear box is connected to an engine model. In the flange areas the model consists of three-dimensional hexaeder elements, however, for saving computer power, we used shell elements for thin-walled parts.

For a reliable assessment of the critical areas so-called sub-models with very fine meshes were generated. Deformations and boundary conditions of the global (coarse) model are projected onto the sub-models by means of the software FEDIS which is available as a stand-alone part of FEMFAT. The stresses computed for the sub-models are much more realistic than for the global model and thus, the fatigue analysis becomes more reliable.

The global model also includes all shafts and bearings. Stiffness and life time of the rolling bearing depend nonlinearily on the load magnitude and the load direction. The interaction between housing, shafts and bearings was simulated using an iterative procedure developed in co-operation with SKF Schweinfurt. For that purpose the FE-model was condensated to the bearing areas. The SKF-Software BEACON now computes stiffness-matrices of the rolling bearings defining the parameters of the bearing-elements in the FE-model. An updated stiffness-matrix of the housing is computed for the modified bearing elements and serves as input for the next loop. Just three loops were necessary for achieving convergence yielding much better stress-results and therefore an improved fatigue analysis.

5.2. Material Data and Simulations

Magnesium was used for gearboxes and crankcases already in the first VW-Käfer (VW-Beatle) in 1938. Because of the low density and the fine conformability during the casting process magnesium is now again used at AUDI and VW for new gears.

It turned out that the strength of magnesium decreases significantly under the influence of corrosion which has to be taken into account in the simulations during the developing process.

Additionally, the slope of the S/N-curve does not decrease for increasing load cycle numbers (Kriegel, *et al.*, 1989). For understanding this characteristic very time consuming experiments must be performed. Thus, a numerical simulation in combination with experimental tests can contribute to a time reduction of the whole developing process.

5.3. Multi-Axial Fatigue Analysis

For the experimental simulation of the loading under driving conditions the gear box was mounted on a fixed crankcase and dynamically tested by means of two forces. The experiment was performed on a servo-hydraulic test bench designed such that every phase-position between both cylinder-signals can be achieved. The combination of both cylinder-forces yields a rotating bending load. Phase position and force-magnitude are chosen such that the strains in the flange areas coincide with the measured strains during driving operation. These forces also serve as input for the numerical fatigue analysis of the gearbox. Since the two (time-dependent) forces are acting simultanously a multi-axial fatigue problem must be solved. Each channel is defined by a normalized stress distribution for two unit load cases and a time-dependent scale factor.

Figure 11 shows the result of the multi-axial fatigue analysis with FEMFAT-MAX for a sub-model of the flange area between gear and engine. One can observe different areas of maximum stress for each unit load case.

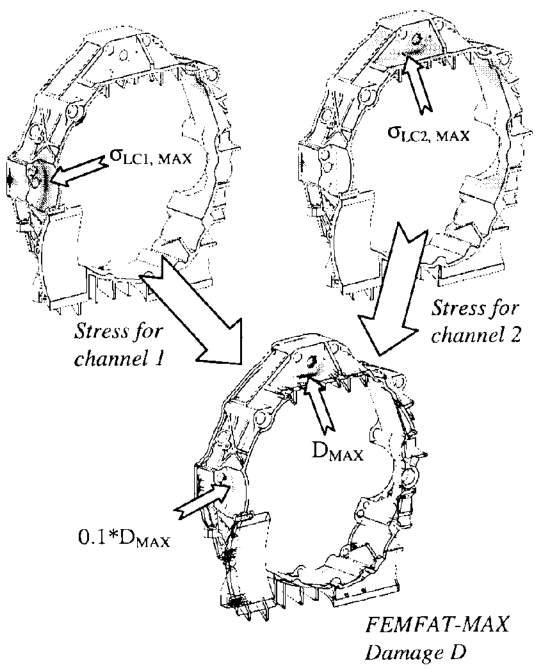


Figure 11. Result of the numerical fatigue analysis.

However, the fatigue analysis shows that the more critical zone with respect to the dynamic loading can be found in the upper part of the flange. The damage in this area is ten times higher than in the other crtical point.

The computed damage values are in very good agreement with the experimental results with respect to position of the critical spots and even with respect to magnitude. The absolute deviation between the numerical simulation and the experiment is equal to factor 2 to 4.

Based on the simulation results several modifications of the gear box could be recommended. The damage of critical area in Figure 11 could be improved by factor 9.

6. THERMOMECHANICAL FATIGUE

The objective of the new module FEMFAT HEAT is the low cycle fatigue analysis of thermal and mechanical highly stressed components. Especially engine parts like cylinder head, piston, exhaust manifold, etc. have to satisfy design criteria with regards to economy, efficiency and environment.

Together with BMW Motoren GmbH (Diesel Engine Development) a procedure has been developed that is based on experimental results. These are isothermal cyclic strain limits at different temperatures and strain limits for temperature loading. FEMFAT HEAT combines this material behaviour with component related properties like the geometry for each combination of thermal and mechanical loads.

The result is a damage distribution displayed with the users FEM-program at the component.

7. CONCLUSION

Today, the optimization of strength and stiffness by means

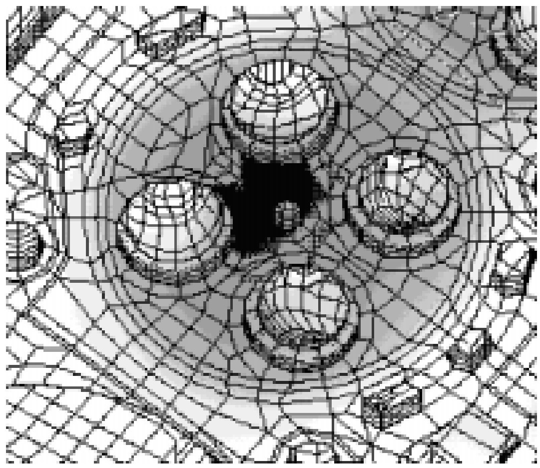


Figure 12. FEMFAT HEAT applied to a BMW V8 diesel engine cylinder head (http://www.bmw.de).

of numerical fatigue simulations plays an important role in the developing processes in automotive industry.

Despite the enormous capabilities of modern software products like FEMFAT experimental tests can not be omitted. However, the key benefit of numerical computations must be seen in the scope of comparisons of constructive modifications and of overall inspections of critical areas. Therefore, we might expect future decreases of developing times from a high-level numerical fatigue analysis.

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Online information about BMW V8 Diesel Engine: http://www.bmw.de/ (Die neue BMW Diesel Generation).

Online information about FEMFAT and the related fatigue fact forum: http://www.femfat.com.