

Simulation of an Engine Speed-Up Run: Integration of MBS – FE – EHD – Fatigue

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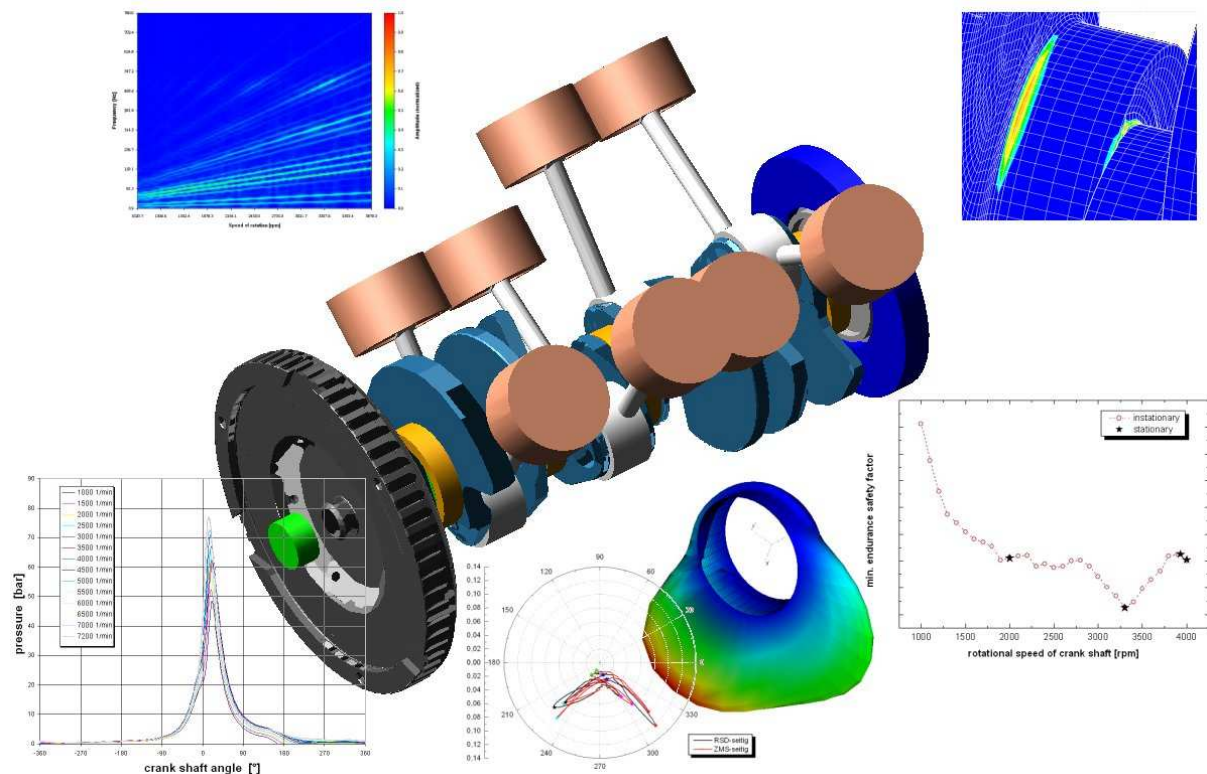


Figure 1: Integrated simulation of an engine speed-up run

1 Summary

This paper deals with the integration of an elasto-hydrodynamic oil film model (EHD) into ADAMS by means of an User Written Subroutine (co-simulation) in order to perform an accurate fatigue lifetime prediction of a crank shaft, based on an engine speed-up run. For this purpose it is necessary to consider the bending line of the crank shaft as accurate as possible. Attempts in the past [1] of using substitute systems (Kelvin-Voigt-elements,...) are only the first step in this direction. Consequently, using an EHD model, which is able to completely describe the dynamic properties of the oil film in the lubrication gap of the main bearings of the crank shaft, will yield realistic boundary conditions for an accurate representation of the bending line of the crank shaft in the simulation process. In this paper we will present a modally based fatigue analysis of a crank shaft, using an integrated simulation process of FE, MBS and EHD.

2 Introduction

In order to perform an accurate fatigue life time prediction of a crank shaft it is necessary to take a couple of different effects into account. The crank shaft is a linear reacting structure which undergoes large nonlinear displacements. Additionally, an

accurate fatigue computation of such an engine component, e.g. in the notch area of the crank shaft main bearing, requires an elasto-hydrodynamic (EHD) oil film model. An efficient computation of all these effects requires different algorithms (e.g. FE for linear reacting structure, MBS for large nonlinear displacements and EHD software which is optimized to compute the oil film dynamics).

A modally based fatigue analysis of the crank shaft of a V6-engine is presented. An elasto-hydrodynamic oil film model is regarded. The FE, MBS, EHD and fatigue software contribute to the integrated simulation process with their particular advantages. The modal representation of the crank shaft is imported into ADAMS. The reaction force and torque of the elasto-hydrodynamic oil film is computed in a co-simulation process (see section 3) utilizing a GFORCE - User Written Subroutine (GFOSUB). The result of the MBS simulation are the modal coordinates of the crank shaft. This modal coordinates together with the modal stresses are the input for the final durability analysis using FEMFAT.

The fatigue result regards the dynamic effects of all engine components and the dynamic properties of the oil film in the lubrication gap.

3 Elasto-Hydrodynamic Oil Film Model

As mentioned above, modeling and simulation of the plain journal bearings of the crank shaft is an essential part of an accurate fatigue life time prediction. Attempts in the past to use substitute systems like a series of Kelvin-Voigt-elements along the circumference and width of the bearing do not result in a realistic bending line of the crank shaft and will yield poor results. Therefore, also more detailed models have been introduced [1] but they are not able to completely describe the dynamics of the oil film in the lubrication gap, which is essentially important when considering the movement of the crank shaft in its main bearings (e.g.: ignition).

In order to accurately compute the deformations of the crank shaft, a very sophisticated oil film model has been developed.

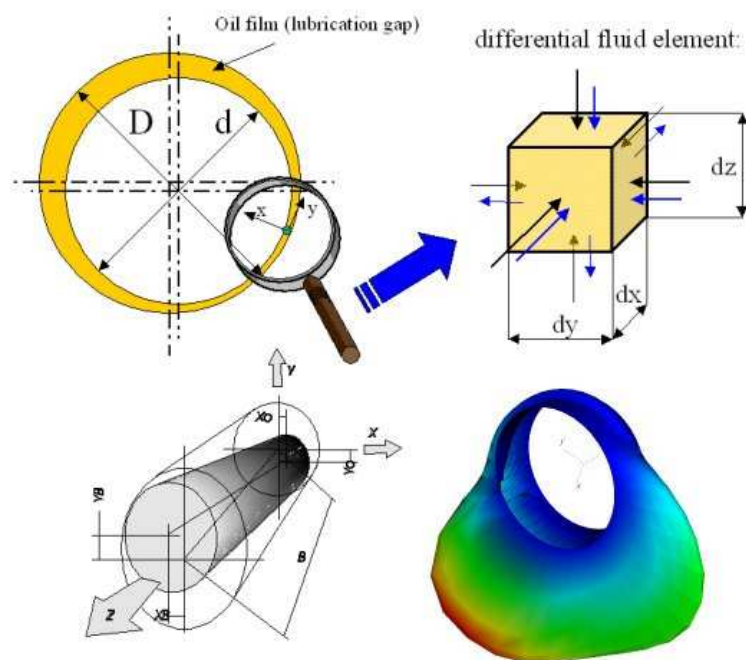


Figure 2: Principles of the Elasto Hydro Dynamic oil film model

Theoretical Background: [7,8]

Assuming an incompressible fluid (Newton's fluid) with a constant viscosity in a laminar lubricating flow, an equation describing the force equilibrium in a differential fluid element can be established. This equation is also known as the Navier-Stokes equation and can be written as:

$$\rho^* \mathbf{w}(\nabla \mathbf{w}) = -\nabla p + \eta^* \nabla^2 \mathbf{w} + \rho^* \mathbf{g} \quad (\text{Eq. 1})$$

with $\rho^* \mathbf{w}(\nabla \mathbf{w})$ denoting specific inertia forces, ∇p specific pressure forces, $\eta^* \nabla^2 \mathbf{w}$ specific viscosity forces, $\rho^* \mathbf{g}$ specific mass forces and \mathbf{w} the velocity vector of the fluid element.

Making several reasonable assumptions and applying appropriate transformations and boundary conditions, equation 1 can be rewritten to the form of:

$$\frac{\partial^2 \Pi}{\partial \varphi^2} + \left(\frac{D}{B}\right)^2 * \frac{\partial^2 \Pi}{\partial Z^2} + a(\varphi, Z) * \Pi = b(\varphi, Z) \quad (\text{Eq. 2})$$

which is also known as Reynold's differential equation. Π denotes a transformed pressure distribution, φ and Z are the coordinates in circumference and width direction, D/B is a form factor, a represents a position- and b a velocity-dependent coefficient. Equation 2, which is a partial elliptic differential equation, can not be solved analytically. However, equation 2 can be solved numerically, using an iterative algorithm, as it is done in the EHD subroutine.

Modeling:

In order to achieve precise and reliable fatigue results, not only the oil film itself but also properties of the bearing cage must be considered as well.

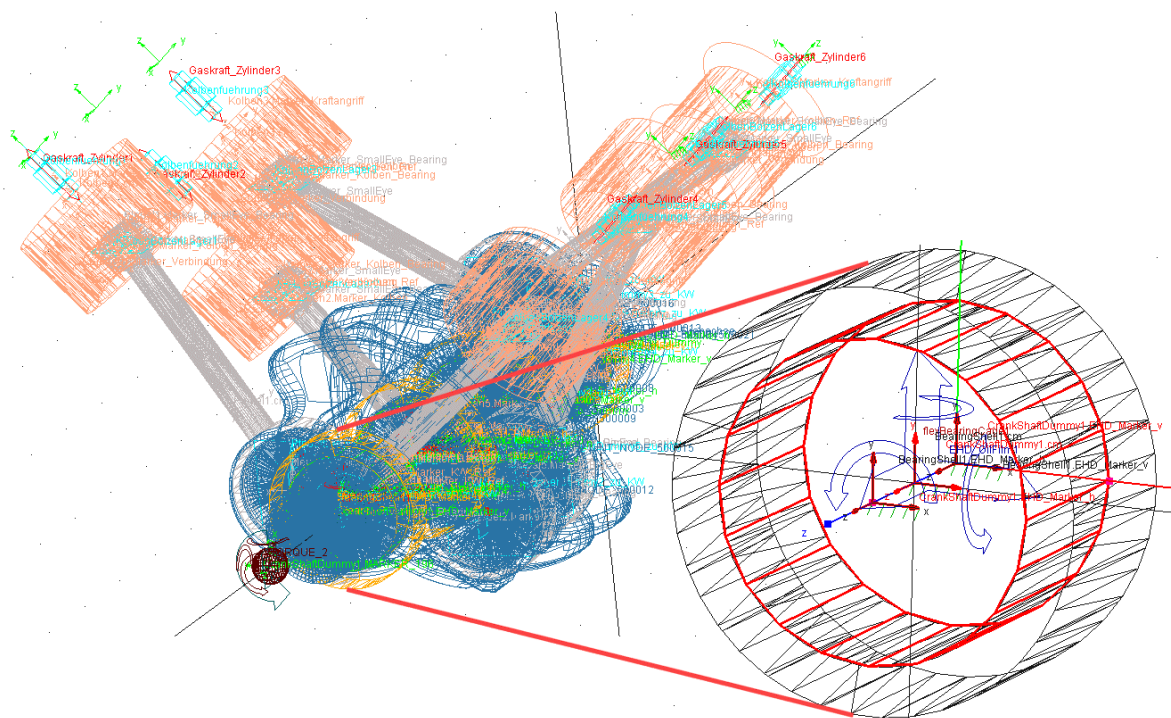


Figure 3: Example of an ADAMS model with oil film model

For this purpose, linearized stiffness values have been determined for each crank shaft main bearing housing using static FE calculations of the engine block. To implement these elasticities into the MBS calculation each bearing shell is suspended against the engine block using a GFORCE with the determined linearized stiffness coefficients in each direction. This elasticities of the main bearing housings will work towards a realistic global bending line of the crank shaft. However, one could also implement a flexible engine block (modal). But since this is again a linear reacting structure only the computational load would be increased. Using the above described EHD oil film model, which is based on equation 2, the pressure distribution inside the lubrication gap (figure 2, 4 and 5) is numerically iterated and integrated to determine the resulting force and torque which is acting on the crank shaft related to the bearing center.

To simplify the modeling process, an ADAMS macro was written, which automatically sets up such a crank shaft main bearing with all parameters including the linearized stiffness of the bearing housing.

Co-Simulation:

The implementation of our EHD oil film model into ADAMS is done using a GFORCE User-Written-Subroutine (GFOSUB). The parameters that are needed by the subroutine are: rotational speed of the crank shaft, position and velocity (vector) of journal relative to the bearing shell at each bearing side to consider an inclined position of the crank shaft inside the bearing shell. To keep track of the actual bearing state two initially coincident markers (shell and journal) are placed at each bearing side, as illustrated in figure 3.

In the parameter list of the user-written subroutine ADMAS passes the Adams_IDs of these four markers and the bearing number (user defined), which can be used to reference different bearing boundary conditions (diameter, width, oil viscosity, lubricating groove, oil supply bore, ...). Optionally, a third marker pair can be defined to consider a locally bending of the crank shaft within the bearing area. During the simulation process our subroutine is updated with the actual bearing state and recomputes the oil pressure distribution by solving the Reynolds equation (Eq. 2) and finally returns the resulting reaction force and torque back to ADAMS. Figure 4 illustrates this co-simulation process. Since the subroutine is called by ADAMS several times during a single time step to build up its Jacobean matrix the algorithm to solve equation 2 has to be highly optimized for speed. Therefore, a sophisticated "solver" has been developed.

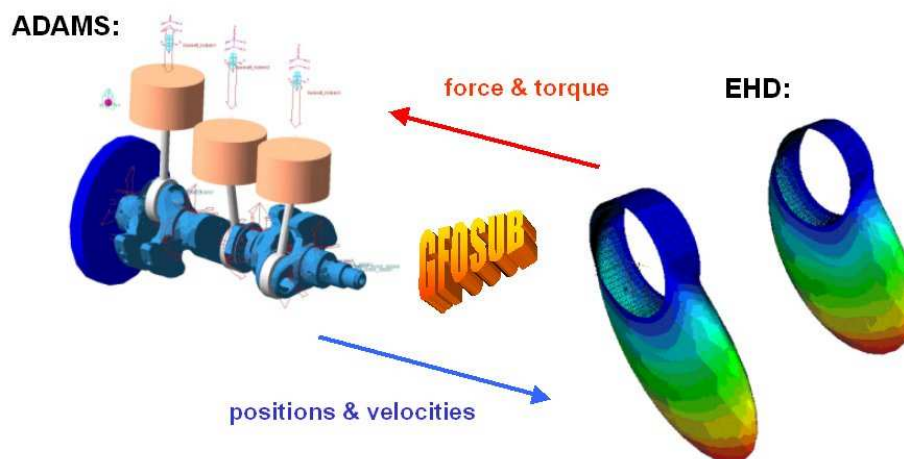


Figure 4: Integrated simulation process of ADAMS and EHD subroutine

The system integration time depends on the selected resolution of the discretization grid that is used to solve equation 2 of our oil film model. For the future it is planned to use the MFORCE statement of ADAMS to directly scale the element pressures at the bearing surface [10].

Since resonances are likely to occur, each mode form is selectively damped by a certain value using a DMPSUB user-written subroutine. The normal modes are assigned a low material damping value whereas the constraint modes are assigned a higher damping value (static corrections).

Outputs:

During an integrated simulation run our subroutine can output user-selectable results like dislocation orbits, pressure distributions, lubrication gap geometries and bearing forces for further use in postprocessing programs like FEMFAT-EHD. They can also be monitored online to trace the results of the integrated simulation process.

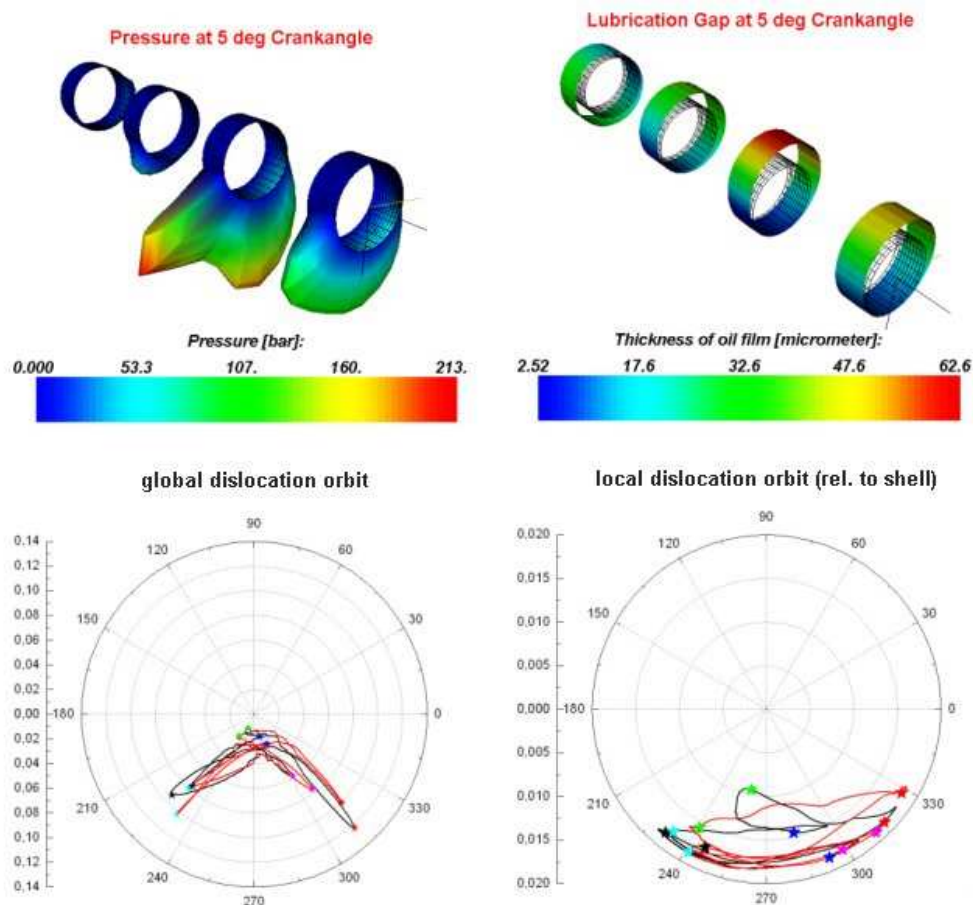


Figure 5: Example outputs of the integrated simulation process

The time history of the selected outputs as well as exported data from the ADAMS postprocessor can be used to generate Campbell diagrams to determine the critical rotational speed regions of interest, where resonance excitation occurs. Figure 6 is an example of a Campbell diagram showing resonances of a torsional crank shaft mode (modal coordinate). At about 3300 rpm resonance is excited by the 6. and 9. engine order. When looking at the rpm sensitive fatigue results given in figure 9, this resonance is responsible of lowering the minimum endurance safety factor.

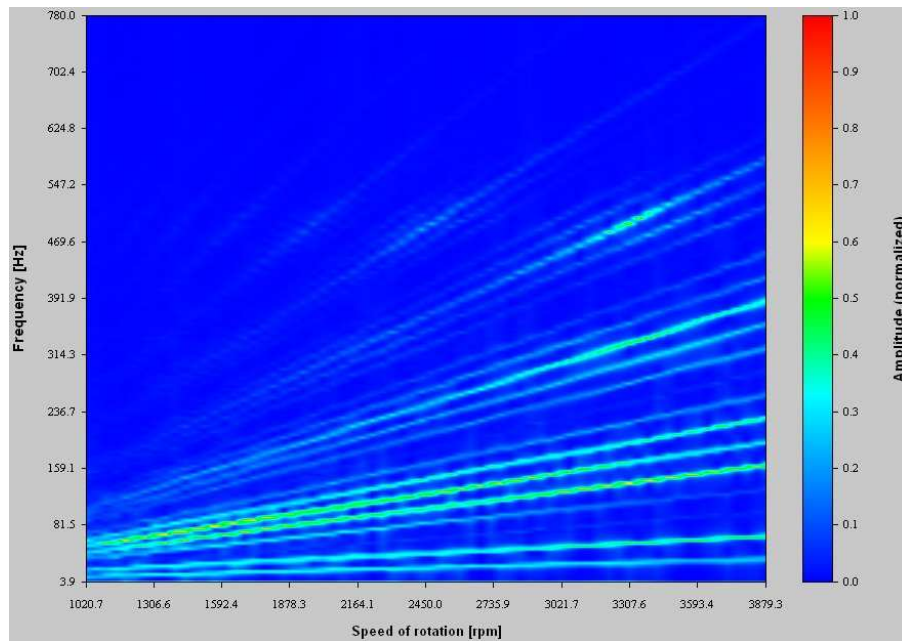


Figure 6: Example of a Campbell diagram to identify critical resonance regions

Verification:

In order to check the results of the integration of our oil film model into ADAMS several case studies (unbalanced shaft, shaft with static and dynamic loads) were performed. The results were compared to the corresponding output of FEMFAT-EHD, which is a stand-alone program, being developed at our company, for computation of journal dislocation orbits, oil pressures, deformations of bearing shells and much more in journal bearings.

The results obtained using ADAMS outfitted with our subroutine are in best agreement to those of FEMFAT-EHD.

4 Modally Based Durability Analysis [2]

General Remarks:

Within the MBS the position of the elastic body is computed by superposing its rigid body motion and elastic deformation. The elastic deformation \mathbf{u} of all degrees of freedom is approximated by a linear combination of suitable modes as outlined in figure 7. The so-called 'Component Modes Synthesis' turned out to be a reliable technique in order to determine such a set of modes [3, 4, 5].

The orthogonalized component modes Φ are used to import the elastic properties of a FE model into the MBS software ADAMS. This set of modes contains both, the FE structures static and dynamic properties and is a result of a FE analysis. The result of the total structure's MBS is a mode share \mathbf{q} of the total deformation \mathbf{u} at each time step, as expressed by:

$$\mathbf{u}(t) = \Phi * \mathbf{q}(t) \quad (\text{Eq. 3})$$

This share is called 'modal coordinate' and is given in figure 7 as q_i . Each deformation of a FE structure is related to a clearly defined stress distribution.

Consequently, each orthogonalized component mode (deformation) corresponds to a clearly assigned stress distribution (modal stress). The meaning of the modal stresses is analog to the meaning of the orthogonalized component modes. The resulting stress state of a FE structure is computed by a linear combination of modal stresses as given in figure 7. The single stress shape's modal coordinate is the same as the modal coordinate of the corresponding component mode and a result of the MBS.

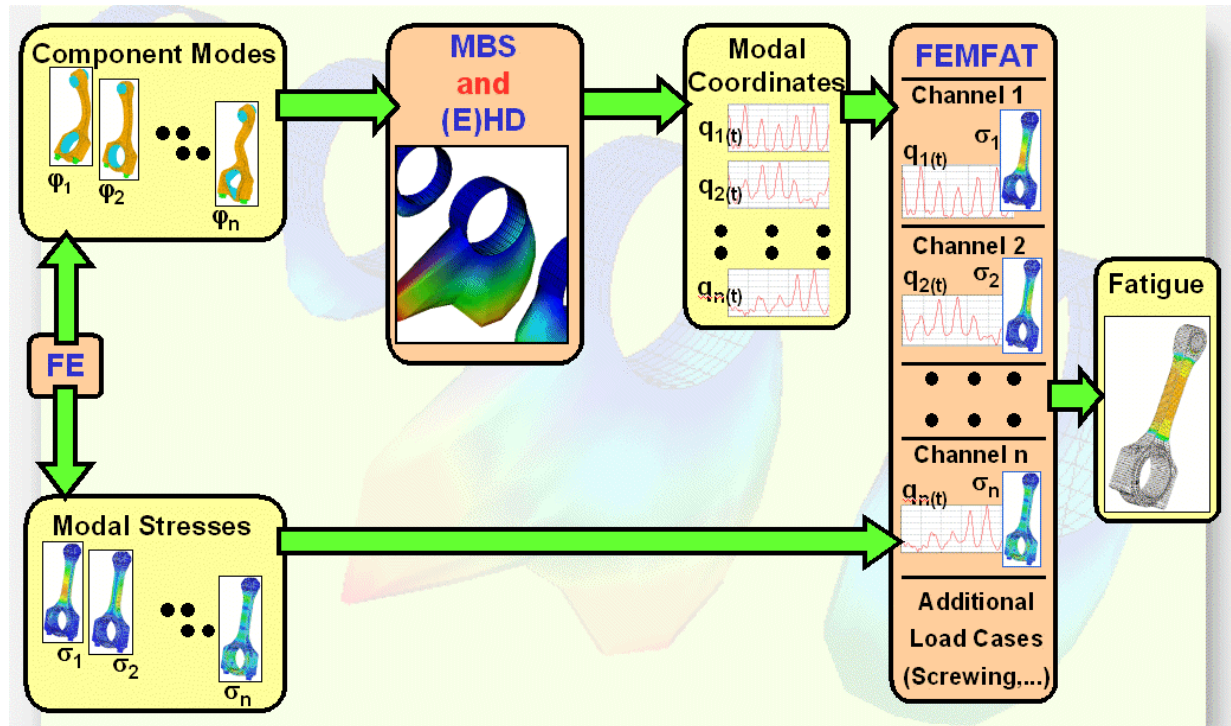


Figure 7: Proceeding of the durability analysis based on modal stresses

This fact can also be expressed mathematically [9]. If all displacements \mathbf{u} of the nodes inside a body are known, a vector $\boldsymbol{\varepsilon}$ containing the strains can be computed using:

$$\boldsymbol{\varepsilon} = \mathbf{L}^* \mathbf{u} \quad (\text{Eq. 4})$$

with \mathbf{L} denoting a matrix of linear operators. For elastic properties the linear relation between strain $\boldsymbol{\varepsilon}$ and stress $\boldsymbol{\sigma}$ is given by

$$\boldsymbol{\sigma} = \mathbf{D}^* (\boldsymbol{\varepsilon} - \boldsymbol{\varepsilon}_o) + \boldsymbol{\sigma}_o \quad (\text{Eq. 5})$$

where \mathbf{D} is the matrix of elasticity containing Youngs Modulus, $\boldsymbol{\varepsilon}_o$ a vector of intrinsic strain, $\boldsymbol{\sigma}_o$ a vector of intrinsic stress and $\boldsymbol{\sigma}$ a vector of the resulting stress. If all intrinsic contributions are zero, equation 5 can be rewritten to

$$\boldsymbol{\sigma} \cong \mathbf{D}^* \mathbf{L}^* \boldsymbol{\Phi}^* \mathbf{q} \quad (\text{Eq. 6})$$

Thus, starting from a known time-history of the Modal Coordinate \mathbf{q} the stress distribution $\boldsymbol{\sigma}$ at each time-step is given by equation 6.

Durability Analysis:

The durability analysis is performed with the Software FEMFAT, developed at the Engineering Center Steyr. The computation methods of FEMFAT range from simple durability analysis up to transient multiaxial fatigue analysis considering temperature, weldings, local plastification and so on [6]. The stress data are provided by a previous FE analysis. The appropriate time history data are determined by measurement or by simulation.

Figure 7 outlines the performed modally based fatigue lifetime prediction. The modal coordinates and the modal stresses are the input for the durability analysis. The modally based durability analysis is a multiaxial and channel-based fatigue simulation. Consequently, one channel consists of a modal stress and the time series of the corresponding modal coordinate. Additional channels can be used to consider further load cases like temperature, or prestress due to a screwing, or stress due to other loads. FEMFAT computes the resulting stress state by superposing the channels which are weighted by the modal coordinates. The example considered in this presentation has been worked out with ADAMS (MBS) and NASTRAN (FE).

It is to mention that the modally based approach provides the possibility for fatigue lifetime prediction of vibration dominated problems like components of an engine block. This is not practicable with the common quasistatic method.

Great care must be taken when modeling the flex-body interface nodes which are required to connect the flexible crank shaft (FE) to ADAMS elements (MBS). The goal of a reliable fatigue lifetime prediction is not to influence the elastic properties of the part under investigation (see figure 1-1). However, a compromise has to be made because the interface nodes have to transfer all loads applied to the flex body.

During an engine speed-up simulation run with ADAMS and the EHD subroutine the computed modal coordinates are written to a file that can be read by FEMFAT. The succeeding modally based multiaxial fatigue lifetime analysis with FEMFAT, to compute the integral endurance safety factor, is done using an initially computed file containing the modal stresses. The proceeding of this method is illustrated in figure 8. Finally, the time history of the modal coordinates can be subdivided into equidistant rotational speed regions for rpm sensitive analysis.

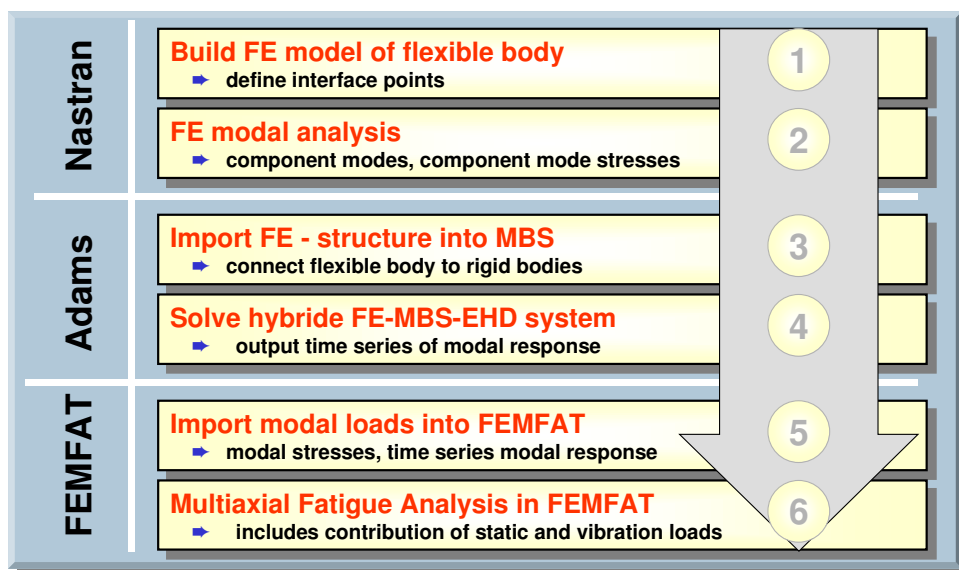


Figure 8: Principle of the durability simulation with modal stresses

Results:

The result of the modally based fatigue lifetime prediction using our dynamic 3D oil film model is shown in figure 9. It contains a picture of a computed damage situation of the crank shaft as well as the results of a rpm sensitive fatigue analysis (open cycles) taken from an instationary engine speed-up run. The computed values from stationary simulations confirm these results (absolute values not shown).

It must be emphasized that our approach of considering the intrinsic dynamics of the oil film in the lubrication gap of the crank shaft main bearings and the elasticity of the bearing housings results in a very accurate bending line of the crank shaft and thus leads to a reliable and precise fatigue life time prediction. Additionally, the modally based fatigue lifetime prediction considers dynamic effects like resonance excitation. When comparing figures 6 and 9 such a resonance excitation at about 3300 rpm is obvious.

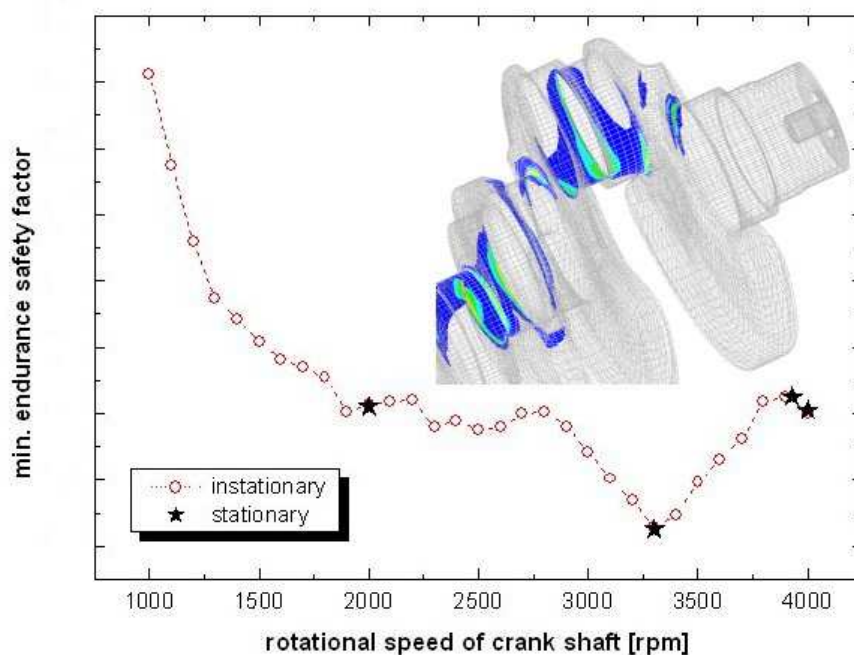


Figure 9: Fatigue results of an engine speed-up run of a V6 engine

5 Conclusion and Outlook

A full flexible crank shaft fatigue simulation was performed using modal stresses. The result regards the dynamic effects of the crank shaft and the dynamic properties of the oil film in the lubrication gap as well as the elasticity of the crank shaft main bearings. Thus, the global bending line of the crank shaft as it is represented in the simulation process is very accurate. Consequently, the performed fatigue lifetime prediction yields more accurate results than other commonly applied techniques [1].

It is to expect that the simulation hard- and software performance will increase continually. Therefore more detailed models will be used resulting in more realistic results. One of the next goals could be to consider the flexing of the bearing shells. This will lead to larger FE structures and longer integration times. However, the modally based durability analysis provides the required capabilities [2].

6 Literature

- [1] W. Witteveen
diploma thesis: Elastic Multi Body Simulation of the Crank Drive with nonlinear Oil Film Structure Interaction
Engineering Center Steyr GmbH, Steyr Daimler Puch
Steyr, Austria, 1999
- [2] P. Fischer, W. Witteveen, M. Schabasser
Integrated MBS-FE-Durability Analysis of Truck Frame Components by Modal Stresses
ADAMS User Conference 2000 / Berlin
- [3] CRAIG, R. R; BAMPTON M. C. C.
Coupling of Substructures for Dynamics Analysis
AIAA Journal, Nr. 6, 1968, pp. 1313 - 1319
- [4] ARGYRIS, J. MLEJNEK, H-P.
Dynamics of Structures.
North-Holland, ISBN 0.444.89045-9
Amsterdam, 1991
- [5] CRAIG R. R.
Structural Dynamics – An Introduction to Computer Methods.
John Wiley & Sons, ISBN 0-04499-7
New York, U.S.A, 1981.
- [6] FEMFAT 4.3 User Manual
Lebensdauer – Simulation dynamisch beanspruchter Bauteile für FE Modelle
Durability Analysis of dynamically stressed components of FE Models
Engineering Center Steyr GmbH, Steyr Daimler Puch
Steyr, Austria, 2002
- [7] M. Prandstötter
diploma thesis: Nichtlineare Gleitlagermodelle für transient belastete Finite Elemente Strukturen
Engineering Center Steyr GmbH, Steyr Daimler Puch
Steyr, Austria, 1998
- [8] O. R. Lang, W. Steinhilper
Gleitlager
Springer-Verlag, Berlin, Heidelberg, New York 1978
- [9] ZIENKIEWICZ O.C.
The Finite Element Method, Third edition
McGraw Hill Book Company, 1977, p 24
ISBN 0-07-084072-5
- [10] J. Zeischka, H. Hoshino
Time Varying Surface Loading of Flexible Bodies through MFORCE and Load Shape Functions
MDI North American User Conference 2002, Scottsdale, Arizona