Advanced Numerical Simulation Techniques for the Fatigue and NVH Optimization of Engines

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Abstract

Due to increasing government regulations concerning the emissions of vehicles (air polution and noise), the necessary reduction of oil and raw material consumption and customer requirements - sound engineering etc. - the automotive industry more and more is forced to optimize the NVH behaviour of engines and to reduce the vehicle weight. Anyway these goals always have to be tightly connected with the maintainance of the fatigue behaviour.

The use of numerical methods such as the finite element method (FEM) enables detailed information about the static and the dynamic behaviour of structures to be determined during the development stage /1/. It can be an excellent tool to analyze and optimize structures by computer simulation and therefore can help to reduce time and costs required for prototyping and to avoid numerous test series.

1 Fatigue Analysis of Crankshafts

1.1 State-of-the-Art

Static finite element analysis of single crank throws are used for the basic evaluation of the stress behaviour due to maximum gas load, mass forces, and maximum torque loading cases, Fig.1,2, considering a statically determined supporting.

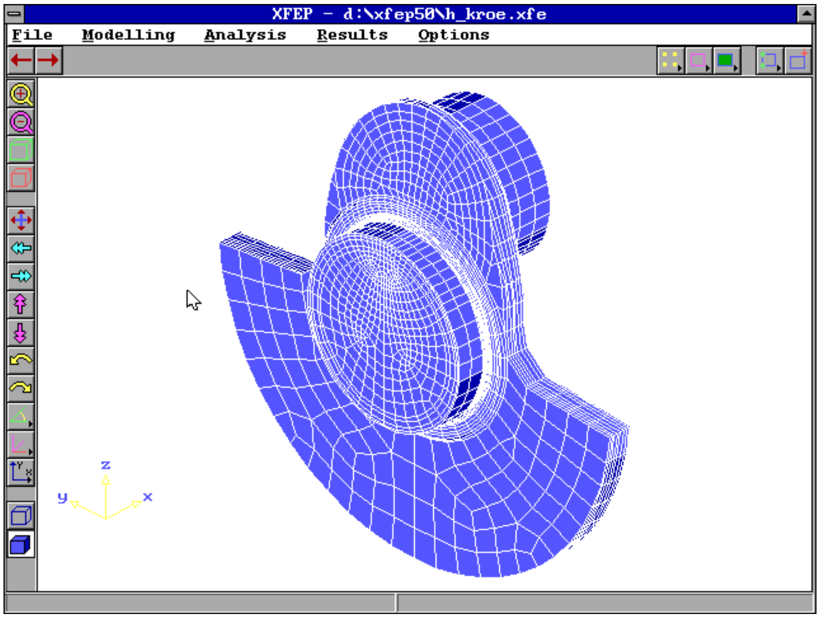

Fig.1 Solid-Element-Model of a Crankshaft Section

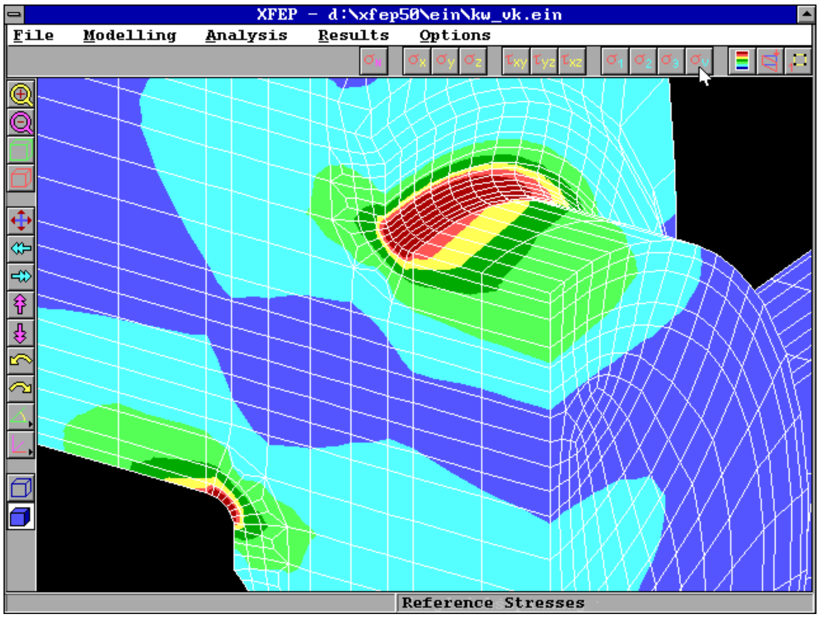

Fig.2 Stress distribution due to 'maximum gas load'

1.2 Nonlinear dynamic analysis

To approve the accuracy of those fatigue analysis K+P developed a method, Fig.3, which is based on inhouse developed software products. It enables numerical analysis of crankshaft dynamics and crankshaft loads which are very close to reality. Herein nonlinear effects in the bearings of the statically undetermined supported, rotating, crankshaft as well as crankshaft dynamic effects such as the flywheel wobbling are considered. At the moment K+P is working to create a direct coupling of the software to MSC.Nastran, MSC.Marc and MSC.Fatigue.

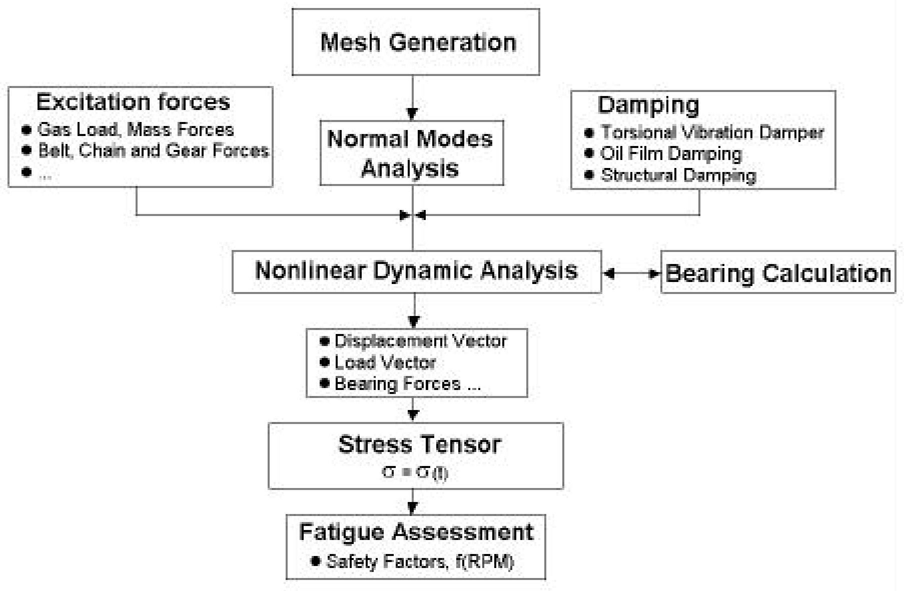

Fig.3 Flow Chart for nonlinear dynamic fatigue analysis of crankshafts

1.2.1 Crankshaft Model

For this nonlinear analysis a 3-dimensional beam-mass-model /2/ of the crankshaft, Fig.4, can be generated fully automatically based on the most important crankshaft parameters such as cylinder distance, pin diameters etc. Furthermore the masses and stiffnesses of the vibrational damper, the flywheel, pistons and connecting rods are considered.

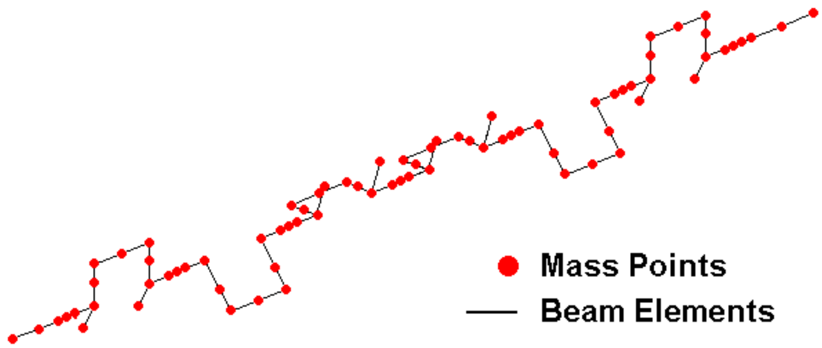

Fig.4 Beam-Mass-Model for nonlinear dynamic analysis

The beam-mass-model of the crankshaft is very convenient to simulate its actual three dimensional dynamic behaviour, especially the coupled longitudinal, bending and torsional vibrations, see Fig.5.

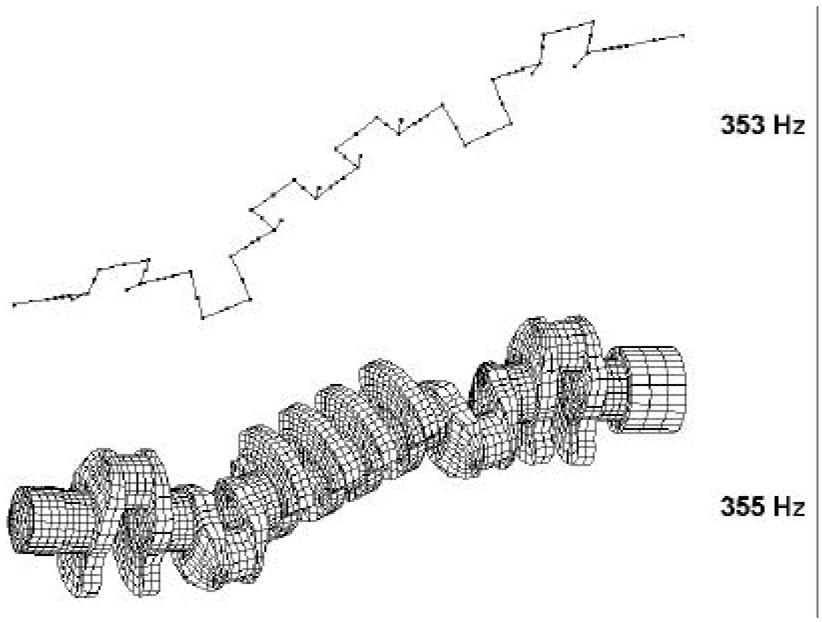

Fig.5 Natural Vibration Mode, example for the good correlation between beam-mass-model and solid-element-model

1.2.2 Oilfilm

Within those nonlinear analysis the oilfilm is considered by means of forces, acting both at the crankshaft and at the main bearing walls. In each time step those forces are calculated using convenient bearing calculations /3,4/ considering oil viscosity, clearance etc.

1.2.3 Example: ‚Fatigue analysis of a 6-cylinder-boxer-crankshaft'

The fatigue behaviour of a 6-cylinder-boxer-crankshaft of a large engine was evaluated based on nonlinear dynamic analysis for a beam mass model, Fig.6, covering the whole engine speed range.

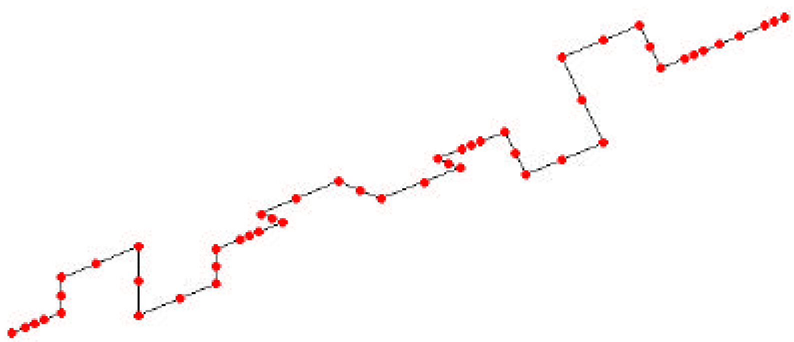

Fig.6 Beam-Mass-Model

Using this model the determination of the time dependent displacement and load vectors - forces, bending and torsional moments in the beams - of the crankshaft was the first step for the determination of its fatigue behaviour. Evaluating the load vectors a basic assessment was possible to determine the engine speed range with the most critical safety factors.

In the next step the time dependent stress tensor was calculated for the solid-element-model based on the time dependent displacement vector of the beam-mass-model.

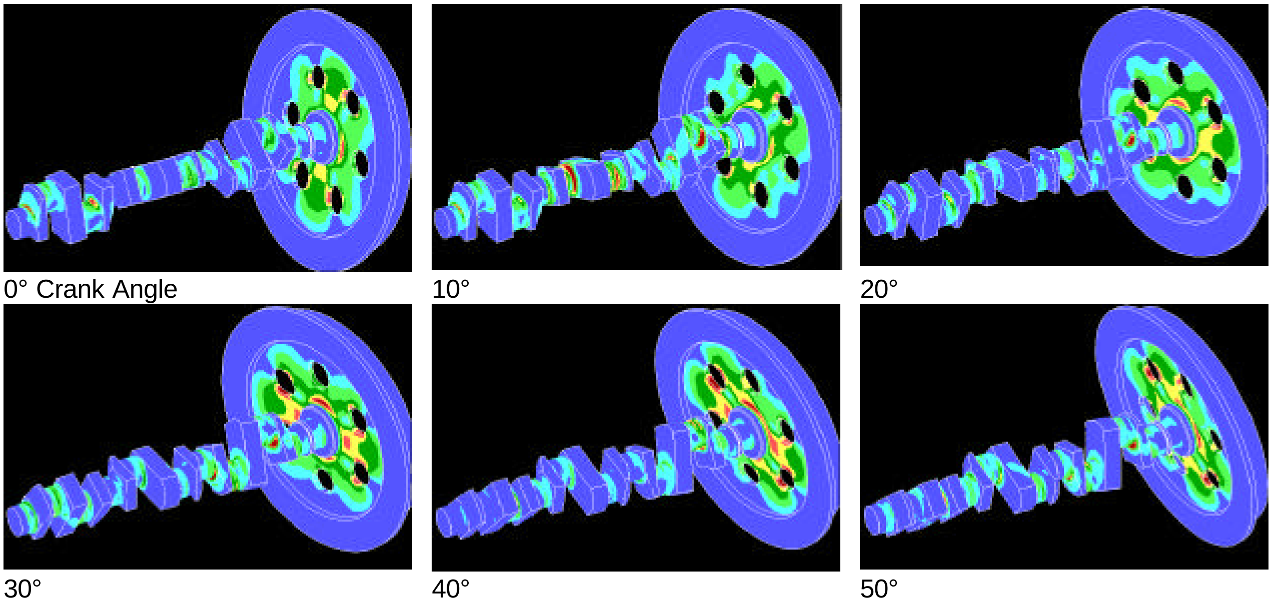

Fig.7 Time dependent stress distribution, shown for different ° crank angle

Using a K+P material data base and additional informations from the customer side, the fatigue assessment finally was done using MSC.Fatigue. Thus the safety factors could be determined for all webs of the crankshaft and the whole engine speed range, Fig.8. Herein web no. 9 is the web, whose position is near to the flywheel.

The results show a decrease of the safety factors in the range of 300 RPM, being caused by a resonance effect of the flywheel at this engine speed.

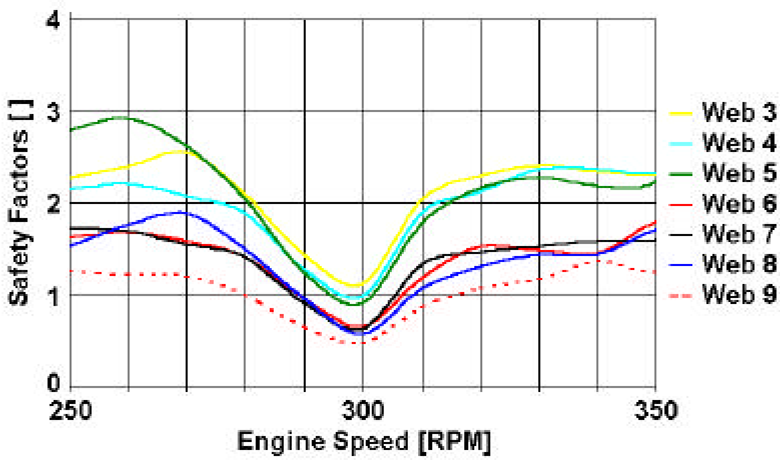

Fig.8 Safety factors over engine speed

The suggested modification of the flywheel design finally caused the fatigue behaviour of the crankshaft to be sufficient again.

2 NVH Analysis of Engines

2.1 State-of-the-art

Linear dynamic analysis are state-of-the-art for the acoustic analysis of engines by means of the finite-element-method.

Herein the assessment of the engine noise basicly is done for two different frequency ranges. The structure borne noise which is transfered to the car body via the engine mounts, occures in the frequency range below 800 Hz and is named as 'low frequency noise'. On the other hand the structure borne noise at the engine surface which is radiated to the surrounding air occures in the frequency range above 800 Hz and is named 'high frequency noise'.

2.1.1 Crankshaft and Oilfilm Models

Performing linear acoustic analysis the bending stiffnesses of the crankshaft and its masses are considered using a 1-dimensional beam-mass-model. For analysis in the range of the 'high frequency noise' usually also the masses and stiffnesses of the pistons and the connecting rods are considered.

The oilfilm is modelled as an equivalent system, representing a direct coupling between the crankshaft and the main bearing walls. This equivalent system consists of rod elements whose stiffnesses are obtained by hydrodynamic bearing calculations. For the analysis in the range of the 'low frequency noise' those stiffnesses are averaged values for a complete engine cycle. Accordingly the stiffnesses of the rod elements for 'high frequency analysis' are higher as they have to represent the oilfilm stiffness at the time of the ignition in the neighbour cylinders.

2.1.2 Normal Mode and Forced Vibration Analysis

The normal modes of an engine and its components are calculated by means of the finite-element-method both to get knowledge about its basic dynamic behaviour - e.g. for the identification of resonance effects, see Fig.9 - and to enable the explanation of phenomena occuring at forced vibration analysis.

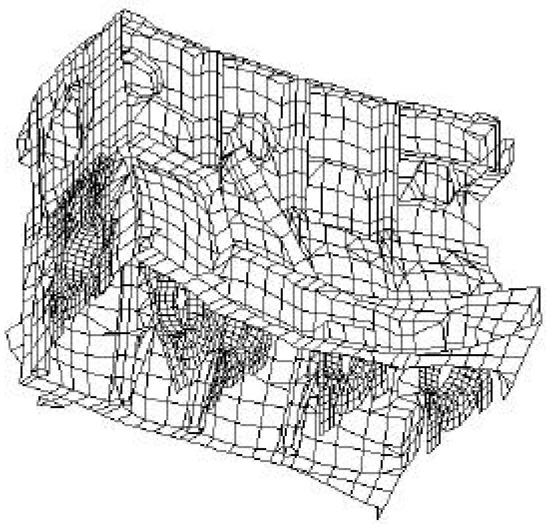

Fig.9 Normal Mode 'Crankcase and Main Bearing Wall'

Those forced vibration analysis usually are performed in the frequency range and are used for the determination of transfer functions. Herein the gas force excitation is considered.

2.2 Nonlinear acoustic analysis

Whereas linear analysis can help to achieve a basic knowledge about the dynamic behaviour of an engine, a lot of dynamic effects and excitation mechanisms can not be considered due to the nature of such analysis, see cap. 2.1.1.

To enable the consideration of actual dynamic effects of the rotating engine components - such as flywheel wobbling -, hydrodynamic bearing conditions and effects caused by piston slap, valve train, belt, chain and gear forces, K+P developed an outstanding software package for highly sophisticated acoustic analysis in close cooperation with leading automotive manufacturers. The direct coupling of these software tools to MSC.Nastran and MSC.Akusmod is one of the strategic goals for the near future.

The basic procedure for K+P’s nonlinear analysis of engine dynamics is shown in Fig.10. Herein quite a lot of actual excitation mechanisms can be considered by precalculating the dynamic behaviour of the different rotating and oscillating engine components using convenient equivalent systems.

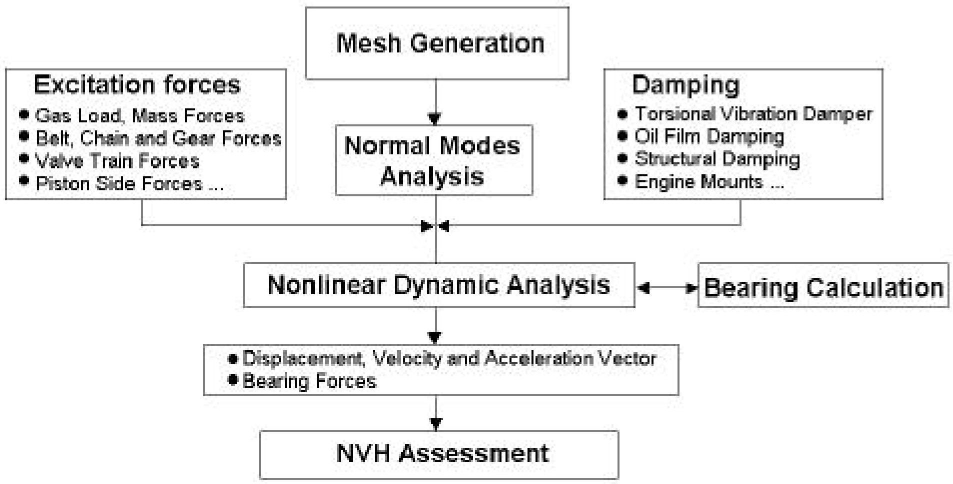

Fig.10 Flow-Chart for nonlinear NVH analysis

2.2.1 Piston Side Forces and Piston Slap

For the calculation of the piston side forces as well as for the evaluation of the impacts due to the piston slap (piston secondary motion) a software tool was developed, which enables the consideration of parameters such as excentric piston pin position, excentric gas forces and the liner contour. Furthermore the friction forces between piston and piston pin as well as between piston and piston rings can be considered.

The tool can be used for calculating both the contact forces at the bottom piston edge and the forces in the area of the ring package. Additionally also the impact amounts occuring as the piston alternates between the two liner contours can be determined. Typical results of such analysis are shown in Fig.11,12.

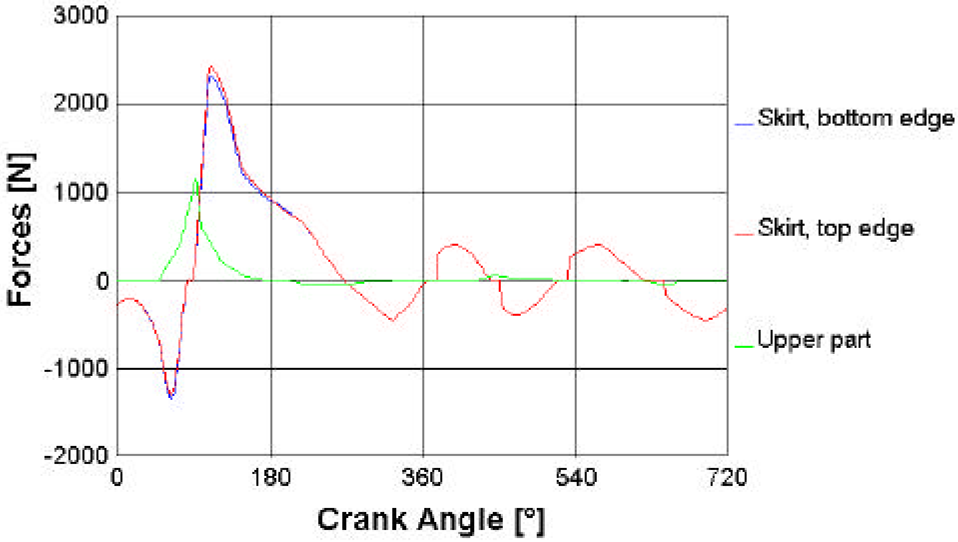

Fig. 11 Contact Forces between Piston and Liner

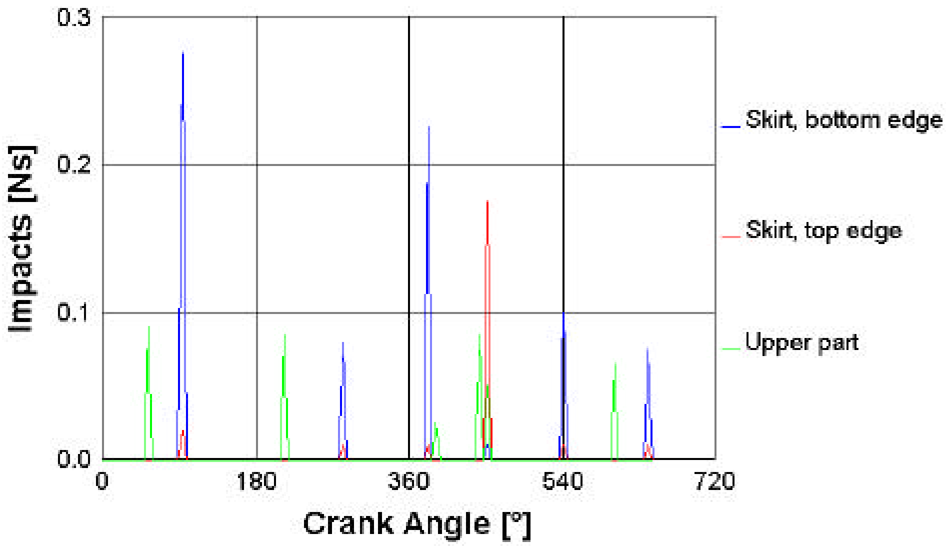

Fig. 12 Piston Impacts

2.2.2 Valve Train Forces

To achieve highly sophisticated NVH analysis of engines the excitation forces due to the valve train dynamics have to be considered. K+P determines the valve train forces - such as the contact forces between cam and follower, the contact forces between valve-springs and cylinderhead as well as the valve seating forces etc. - using an convenient equivalent system, see Fig.13.

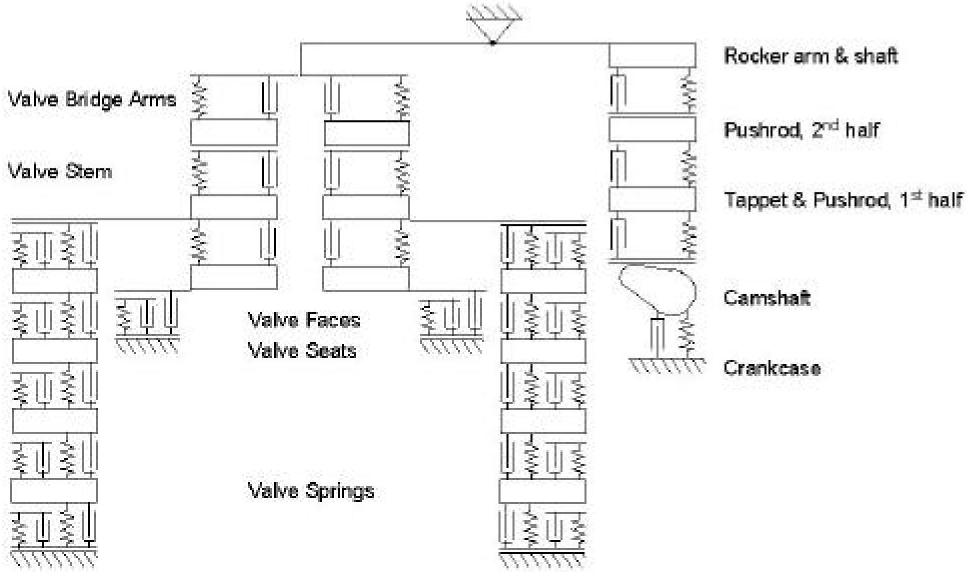

Fig. 13 Equivalent System for analysis of Valve Train Dynamics

It consists of various mass-spring-damper-elements which represent the dynamic characteristics of the individual valve train components.

The dynamics of this equivalent system is excited by the prescription of the cam lift curve. Herein the influence of the valve lash - either for a cold or a warm engine - as well as the influence of the cylinder pressure is considered. Fig.14 shows calculated forces in the contact area between cam and roller follower.

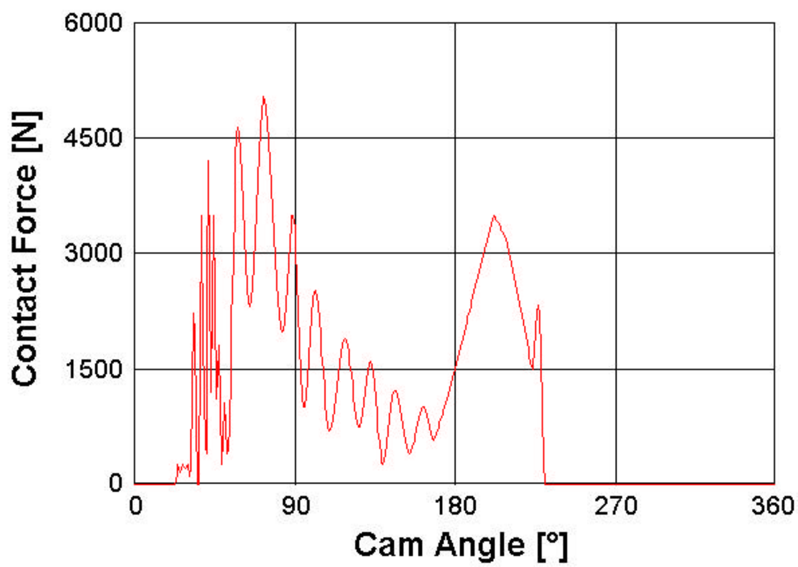

Fig. 14 Contact Force between Cam and Roller Follower

By the way the predetermination of the contact forces occuring at the individual valve train components also enables a lot of design analyses and strength calculations such as the evaluation of the camshaft bearings, strength calculations of all components and the evaluation of the valve spring design regarding to the number of coils, spring stiffness characteristics, spring material etc.

2.2.3 Belt, Chain and Gear Forces

Driving of balancing shafts and individual auxiliaries - such as pumps, generator etc. - often causes high dynamic effects and forces, which have to be considered performing NVH analysis.

This made K+P develop a convenient software tool for the calculation of those excitation mechanisms using an equivalent mass-spring-damper-system again. This tool basically enables the calculation of the torsional vibrations and the forces acting between mating gears or in the individual chains or belts, thus being also an excellent tool for gear strength calculations.

Facing the specific requirements for the NVH optimization of a 4-cylinder-inline-engine with balancing shafts K+P had to develop a special technology that enables the consideration of the interaction between the dynamics of the rotating engine components - crankshaft, oilpump shaft and balancing shafts - and the backlash alteration and the forces in the different toothings of the balancing shaft gear drive, Fig.15.

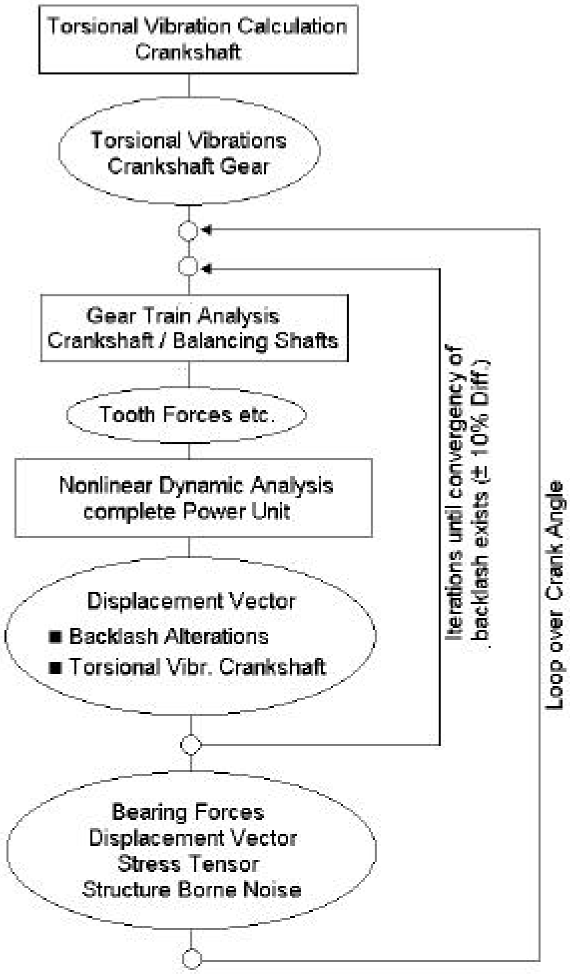

Fig.15 Flow Chart for considering the interaction between shaft dynamics and backlash alteration in toothings

In Fig.16 the corresponding equivalent system for the dynamic analysis of this gear drive is shown. Nonlinearities - backlash influence etc. - are considered.

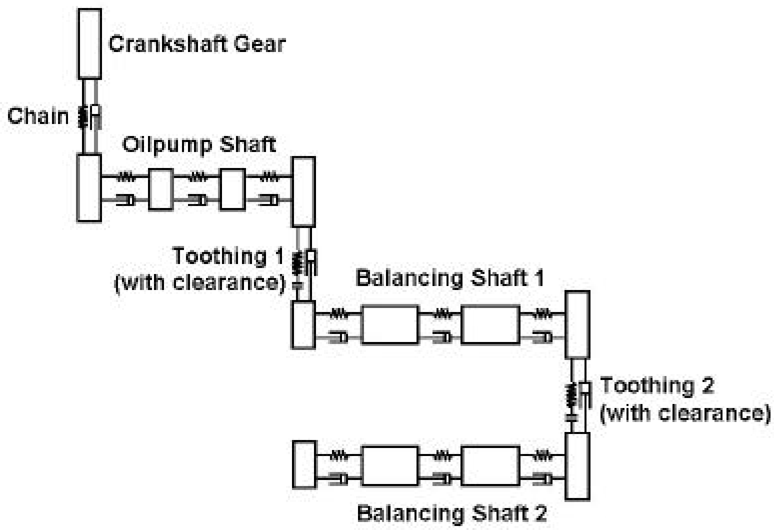

Fig.16 Equivalent system for nonlinear analysis of the balancing shaft gear drive

Herein the vibrations of the complete gear drive system are forced by the torsional vibrations of the crankshaft gear. Based on this excitation the tooth forces are calculated in each time step of the NVH analysis using the iterative technology described above, Fig.17.

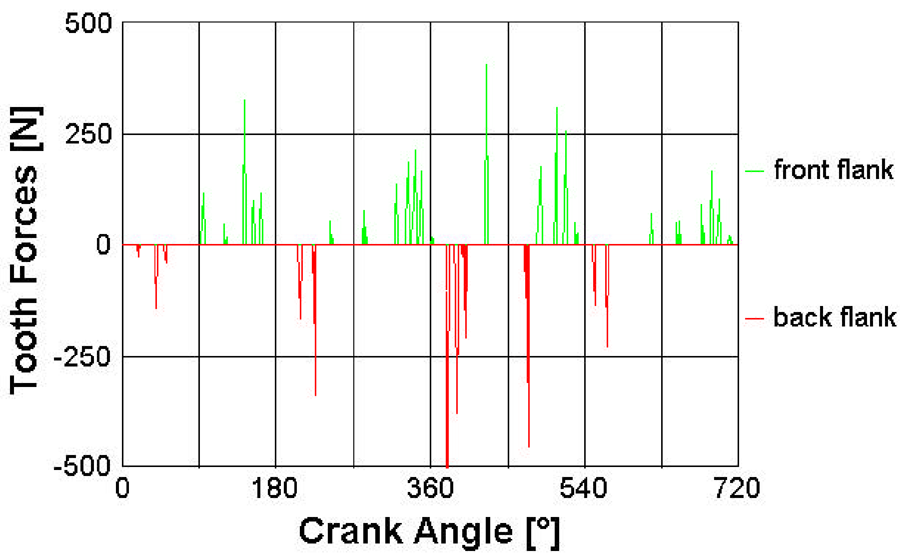

Fig.17 Tooth Forces in the primary toothing

Subsequently those tooth forces get directly applied to the corresponding nodes of the rotating shafts - crankshaft, oilpump shaft and balancing shafts - thus making the axial tooth force components be the major structural excitation forces in the area of the balancing shaft housing, Fig.18,19.

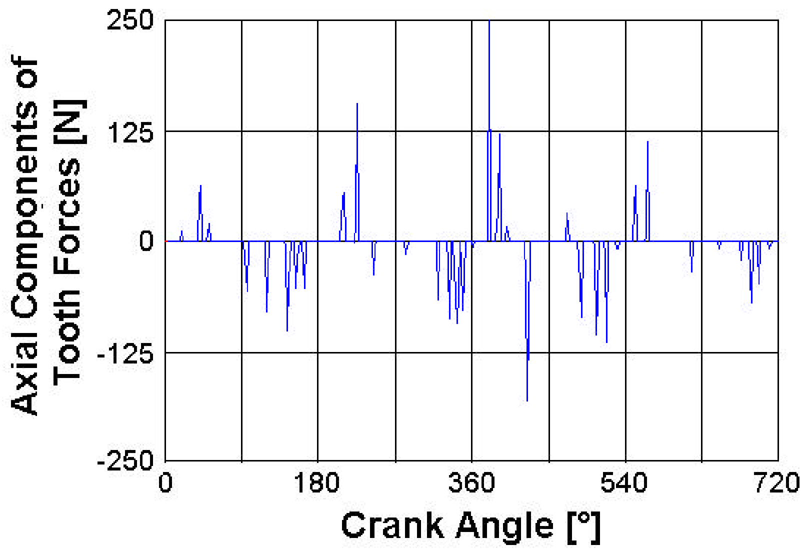

Fig.18 Axial components of tooth forces in the primary toothing

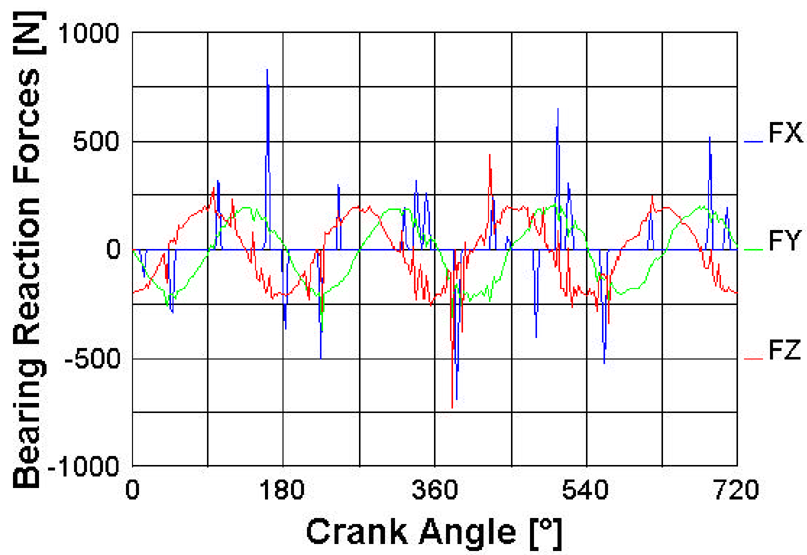

Fig.19 Reaction forces in the axial thrust bearing of the primary balancing shaft

This nonlinear NVH analysis originally was performed for a basic engine design in a wide engine speed range considering different operating temperatures. A comparison of analysis and measurement results showed a very good correlation.

Based on this knowledge some concerted measures for a further noise optimization could be found - modifications of helix angles and backlashes, stiffening of the structure etc. - and were analysed using the described nonlinear solution technique again, see Fig.20.

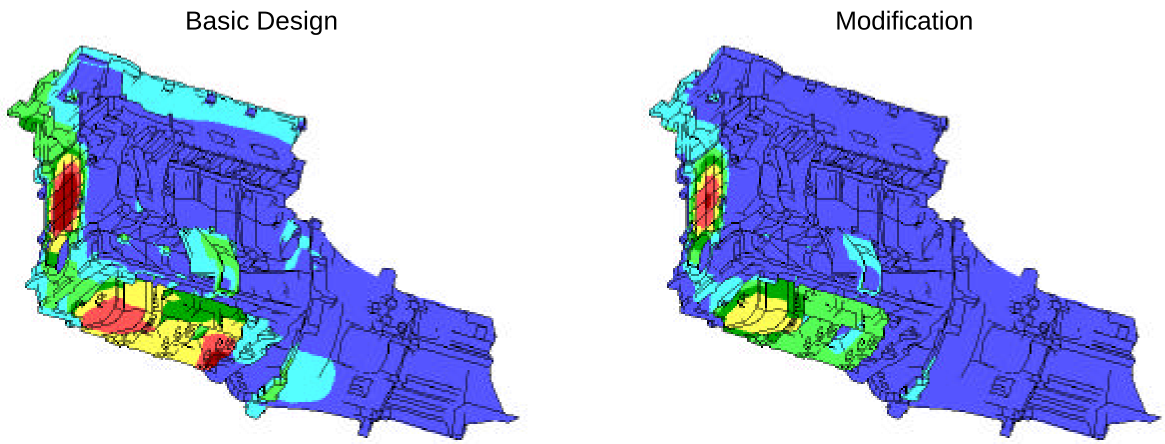

Fig.20 Integral Velocity Levels

The application of these software tools and the intended enhancements - such as its coupling to MSC.Akusmod - will enable acoustic analysis of ultimate quality and efficiency. It will offer the possibility to assess the actual influence of crankshaft dynamics, oilfilm properties and many other excitation mechanisms.

3 Conclusion

Linear static and dynamic finite element analysis can be a usable tool to get basic knowledge about the fatigue behaviour of crankshafts and the NVH behaviour of engines during the first development stage.

However for both the reduction of weight and the optimization of fatigue and NVH behaviour it is unavoidable to perform nonlinear dynamic analysis. Those analysis help to get more detailed information about the vibrations and loads due to the actual dynamic behaviour of rotating and oscillating engine components.

Special algorithms for pre- and postprocessing and advanced engineering data management tools can provide a powerful framework for analysis of ultimate quality and efficiency.

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