Mechanical Simulation in the Engine Development Process: Part I = The Crank Train Subsystem

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Summary

The continuously increasing requirements regarding product quality while controlling the cost combined with the increase in computer hardware and software performance leads to a fast expanding usage of simulation techniques. In the mid term the objective is to substitute a hardware prototype with a virtual prototype.

In order to achieve this goal, it is required that the thermo-dynamic processes as well as the engine mechanics are considered to the highest level of accuracy. To support the different calculation disciplines several numerical simulation methods such as the finite element method (FEM), the multi body dynamics simulation method (MBS) and the computational fluid dynamics simulation method (CFD) have been proven. The tools, which utilize these methods, are so called multi purpose software packages designed to be applied to a variety of engineering problems. With regard to the engine mechanics calculation the combination of FEM with MBS through modal condensation appears to be the most effective approach. With that, it is possible to accurately predict engine component behavior as well as their interaction with other components and systems.

When analyzing the current use of CAE methods in the engine development process, it becomes apparent that simulation models have different levels of refinement according to the development phases. Very often these models are not compatible in terms of modeling method and do not share the same data model. Also, the data exchange with suppliers, which should be involved very early in today’s processes, is very challenging.

Additionally, the engineers have to have in depth knowledge of numerical methods beside their engine expertise, which leads to long adoption periods as well as an overall error prone process. The implementation of engine specific simulation modules in existing multi purpose software packages presents a solution to these problems. With this approach an interface between the development engineer and the powerful numerical simulation program has been created.

The requirements mentioned above were the corner stones for the development of the engine mechanics software: **ADAMS/Engine powered by FEV**. This paper contains an exemplary description of the major capabilities of this software. In particular it focuses on the crank train, the valve train and the timing mechanism.

1 Introduction: CAE in the Engine Development Process

There are a number of reasons to use CAE methods in the modern engine development process. Foremost the need to cut cost increases the usage of virtual testing versus costly hardware testing.

Another important argument is the fact that certain characteristics of the system cannot always be measured without influencing the system. In a first step the characteristics, which can be measured with relative low effort, may be used to correlate and to verify the validity of the calculation models. In return these models deliver a large number of characteristics who’s measurement would be too time consuming and costly in the frame of a regular development project.

Since hardware prototypes are not available early in the design process especially in the „pre-prototype phase“, the usage of virtual prototypes is required and represents the most important scenario. During this period the results from calculations and the engineering experience of the designers are the only basis for design decisions, which influence the entire design process until production [1]. The more innovative the design is the less engineering expertise is available; hence the importance of the calculation increases dramatically in the concept phase. During the last years the significance of new concepts with respect to the engine mechanics has reached a new level as shown by the introduction of engines with variable compression ratio and valve trains with variable timing.

According to the application it is possible to distinguish between predictive and validated simulation models. The predictive models should deliver results based on physical input parameters early in the process, when no hardware prototype is available. The validated models are first validated against measured data using data deviating from the physical input data. These models are usually used in the detailed design phase, such that the requirements on model accuracy are much higher. In general, models with increasing levels of refinement are necessary during the development process [2].

Depending on the need to consider the interaction between models a modular approach is required. Component level analysis as well as subsystem and full system analysis must be possible. The modularity is also important to support the communication between the different areas of responsibility in the development process and to enable the focused analysis in these areas. The trend to out-source more of that development responsibility to suppliers further underlines this demand. The following main requirements for CAE software in the engine development process can be derived from the points mentioned above:

- Open and extensible
- Easy to use
- Multiple levels of refinement
- Modular

The common practice is to use different simulation tools during the different stages of the development process. While simple engineering judgment tools are used during the concept phase more complex programs are used towards the end of the design cycle. The data exchange between the tools is particularly difficult since no standard has been declared.

During the development of the engine simulation software ADAMS/Engine powered by FEV all the requirements mentioned above have been considered. In particular the focus was on the entire process rather then on partial areas of the process.

The conflict between the requirement for extendibility and openness on the one side and user friendliness on the other was solved by separating topology and data. While standard users can make quick and simple changes to the data then applied to existing templates, expert users can develop entirely new concepts quickly by utilizing the predefined components and templates. In this way the capabilities of special purpose software and multi purpose software have been ideally combined.

The existence of the clearly defined data model also enables the exchange of data between the different internal and external participants very efficiently. Through the usage of „communicators“ it is possible to combine a number of subsystems such that the modularity of the models is guaranteed. This allows the user to focus on the component analysis, subsystem and full system analysis using the models in the one environment.

Attention has been paid to the implementation of different modeling approaches for components representing different levels of refinement. The modeling approach can be changed quickly without changing the topology model. Following the usage of ADAMS/Engine powered by FEV in the engine development process is described with examples.

2 Application Examples

2.1 Crank Train Layout

A 3.5-liter V6 60 degree gasoline engine will be used as an example. The concept/layout phase includes the following steps:

- Balancing of the free forces and moments
- Minimization of the internal bending and torsional stresses
- Study of the hydrodynamic bearings

In the case of V-engines it is possible to use the crankshaft counterweights not only to minimize the bending moments but also for the reduction of the inertial and transitional forces. This means that the layout of the crank train has to be done in two steps.

2.1.1 Model Topology

The topology of the exemplary engine can be described as follows:

- 6 cylinder
- 4 main bearings
- 2 crank pins per throw
- Crank pins connected via intermediate web

Since this is a commonly used layout a predefined template can be used. The balancing shaft as well as the four mounts can be deactivated individually.

A subsystem based on the topology stored in the template is created. The first step is to define the global data of the engine such as bore, stroke, bank angle, firing order, etc. Additionally the data for the most important engine components have to be set. Especially important are the masses and inertia properties of the moving parts as well as the pressure maps of the cylinders. The counter weights of the crankshaft will initially be deactivated for the mass balancing calculations.

To complete the model, the subsystem is assembled with a virtual test rig, which allows the user to define the driving function. Figure 1 shows the basic engine assembly.

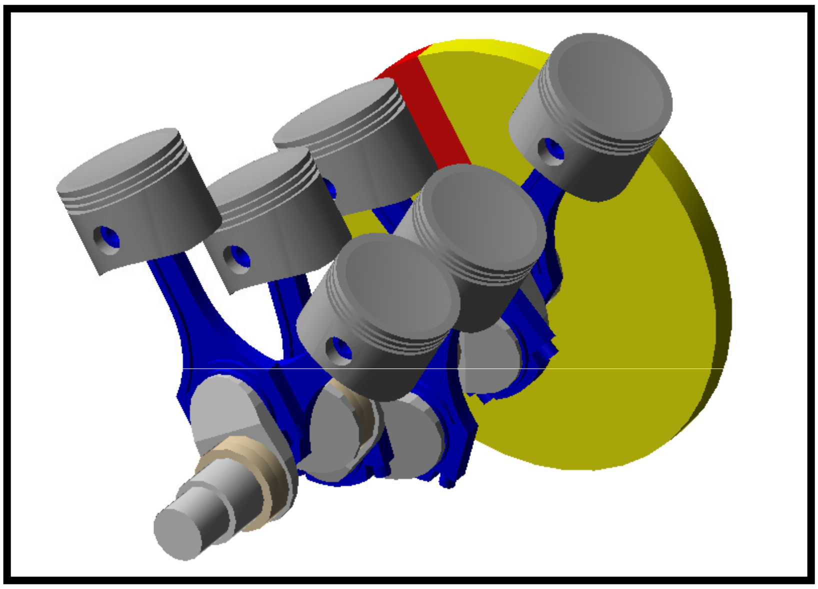

Figure. 1: Basic engine assembly

2.1.2 Balancing of the free Forces and Moments

All connections are done via constraints and the crankshaft is driven with a constant rpm. The engine block is fixed to ground, such that the forces and moments in the constraint are part of the results from this simulation. Usually the geometric center of the crank train is chosen as a reference system for the calculation of these forces. In a subsequent analysis of the results the amplitude and phase of the free rotating and oscillating forces and moments are determined with respect to the engine orders, Table 1 shows the free forces and moments of the example engine for the first four engine orders. Table 2 shows the amplitudes and phases, which can directly be used to derive corrective design changes.

Engine Order	rotating force [N] lateral axes	oscillating force [N] lateral axes	rotating torque [N mm] lateral axes	oscillating torque [N mm] lateral axes	oscillating torque [N mm] rotation axis
1.0	0	0	2714756.70	0	0
2.0	0	0	-215022.04	0	0
3.0	0	0	0	0	507579.03
4.0	0	0	4381.60	0	0

Tab. 1: Free forces and moments of the example engine

Engine Order	rotating stat. amp. [kg mm] phase [deg] lateral axes	oscillating stat. amp. [kg mm] phase [deg] lateral axes	rotating dyn. amp. [kg mm^2] phase [deg] lateral axes	oscillating dyn. amp. [kg mm^2] phase [deg] lateral axes	oscillating dyn. amp. [kg mm^2] phase [deg] rotation axis
1.0	0 0	0 0	2.750624e+004 180.0	0 0	0 0
2.0	0 0	0 0	-5.446572e+002 0	0 0	0 0
3.0	0 0	0 0	0 0	0 0	5.714279e+002 0
4.0	0 0	0 0	2.774682 0	0 0	0 0

Tab. 2: Static and dynamic amplitudes and phases of the example engine

The oscillating values always have positive amplitudes whereas the sign of the amplitude of the rotating loads determines the rotational direction of the moment. For the example engine, this means that there is a rotating moment for the first engine order in the direction of rotation of the crankshaft and a rotating moment for the second engine order in the opposite direction. The first can be completely balanced through counter weights. The balancing of the second requires a balancing shaft. The moment of the fourth engine order does not require any balancing due to its small amplitude. The oscillating moment for the third engine order is the reaction to the resistance torque and usually remains unbalanced.

The use of counter weights and a balancing shaft are therefore meaningful measures to take. The masses as well as their position can be derived from the phases and amplitudes directly. Since the original template already contained a balancing shaft, it only has to be activated and set with the appropriate values.

The pitching and rolling reaction moments for the design without balancing, balancing of the first order and balancing of the second order is compared as shown in figure 2 in order to verify the balancing of the example engine. The dominant first order moment is entirely eliminated as expected through the introduction of the counter weights and similarly the second order moment through the balancing shaft. The remaining fourth order moments have small amplitudes, which hardly show, such that the free forces of the crank train can be considered well balanced.

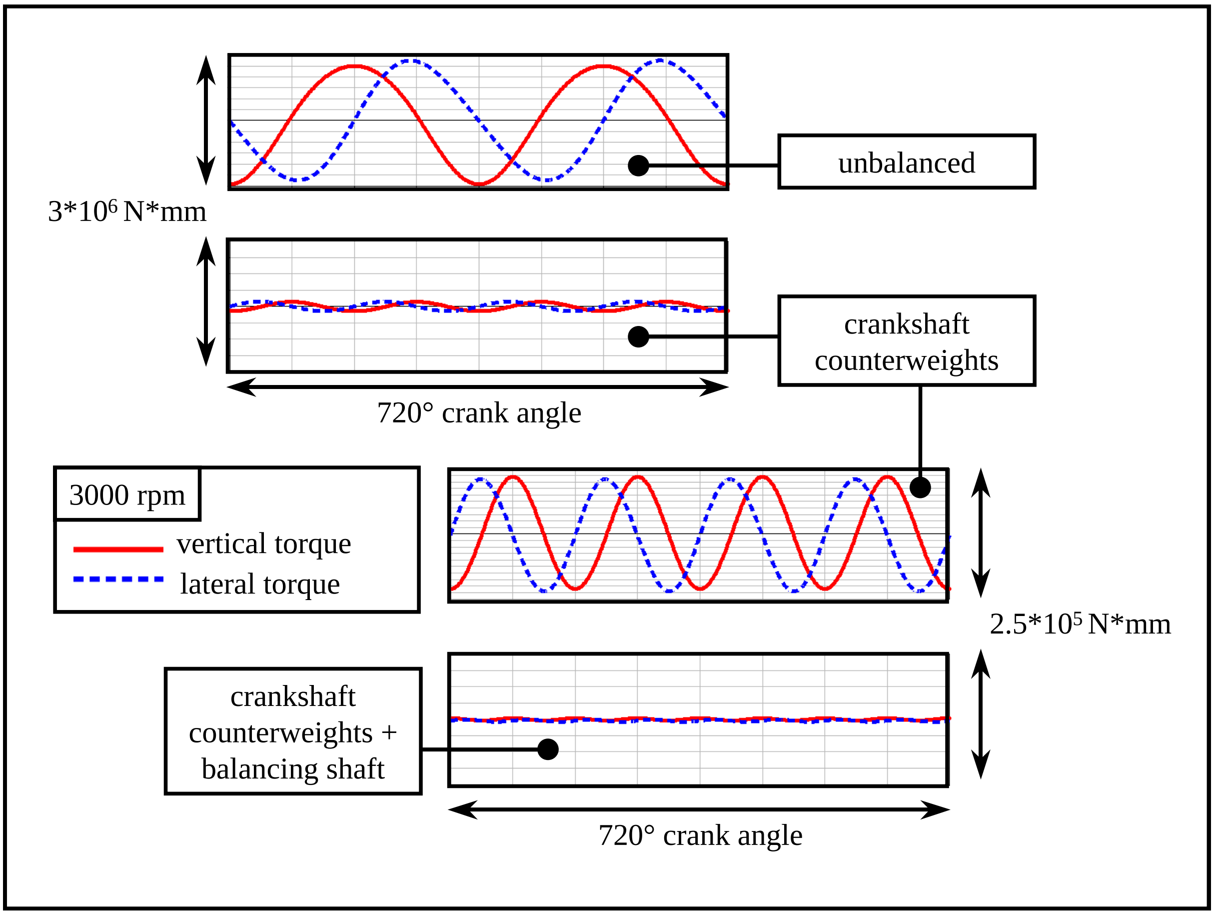

Figure 2: Mass moments in the engine reference system

For the detailed study of the influence of the free forces on the position and characteristic of the engine mounts it is possible to deactivate the rigid connection of the engine block to ground and to activate the engine mounts contained in the model.

2.1.3 Calculation of the Crankshaft torsion Stress

To compute the torsional loads in the crankshaft additional compliance is added to the model. This is possible without changing the topology of the model by using a different modeling option on the crankshaft element. The already defined model data are inherited to this model and the additional data for the stiffness and damping of the shaft is automatically calculated based on geometry and material data. Two additional rotational degrees of freedom are added to the test rig to allow the vibration of the shaft.

Without high fidelity FE models of the crankshaft it is not possible to determine the component stresses to the highest level of accuracy, but it is noteworthy that even with the rigid body based models with compliant connections it is possible to make comparative analysis very well. Also, the investigation of rotational vibrations and the effects of a torsional vibration damper with respect to the stresses can be performed with this model since it is possible to easily activate such a damper. In the example engine a rubber damper is selected.

Figure 3 shows the torsional compliant crank train model with a balanced crankshaft, balancing shaft, engine mounts and vibration damper as well as the adjusted test rig.

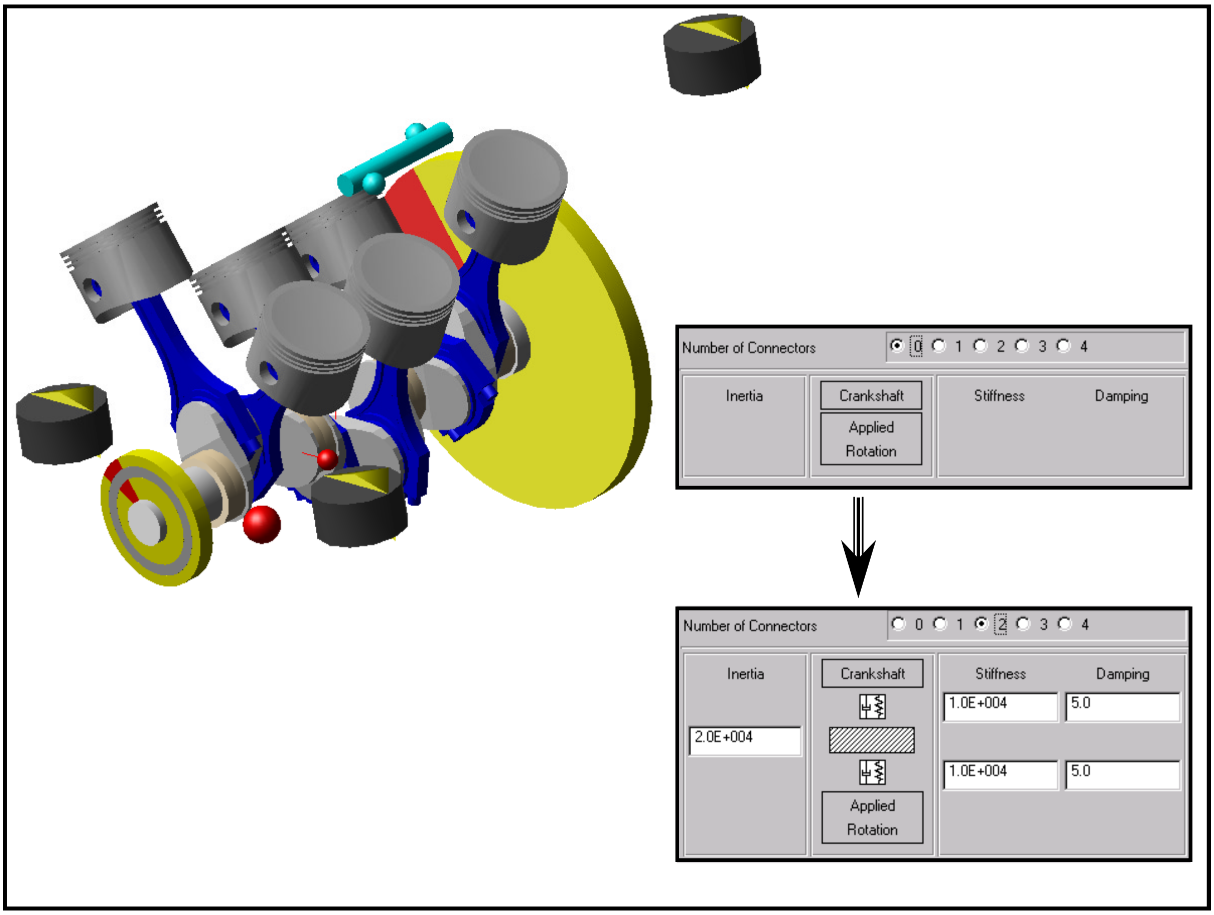

Figure 3: Torsional compliant model

The critical engine rpm is calculated in an rpm sweep analysis by monitoring the nominal stress [4] (= torsional stress at the main pins). The highest stress occurs at 6000 rpm, where we find a resonance excited by the sixth engine order.

The rubber damper is then designed around this operating point. Following a steady state analysis at 6000 rpm with and without a damper under full throttle is performed. The crankshaft element has a number of predefined measurement points at the main pins monitoring the nominal stress. Figure 4 shows stress results for a full engine cycle with and without the damper.

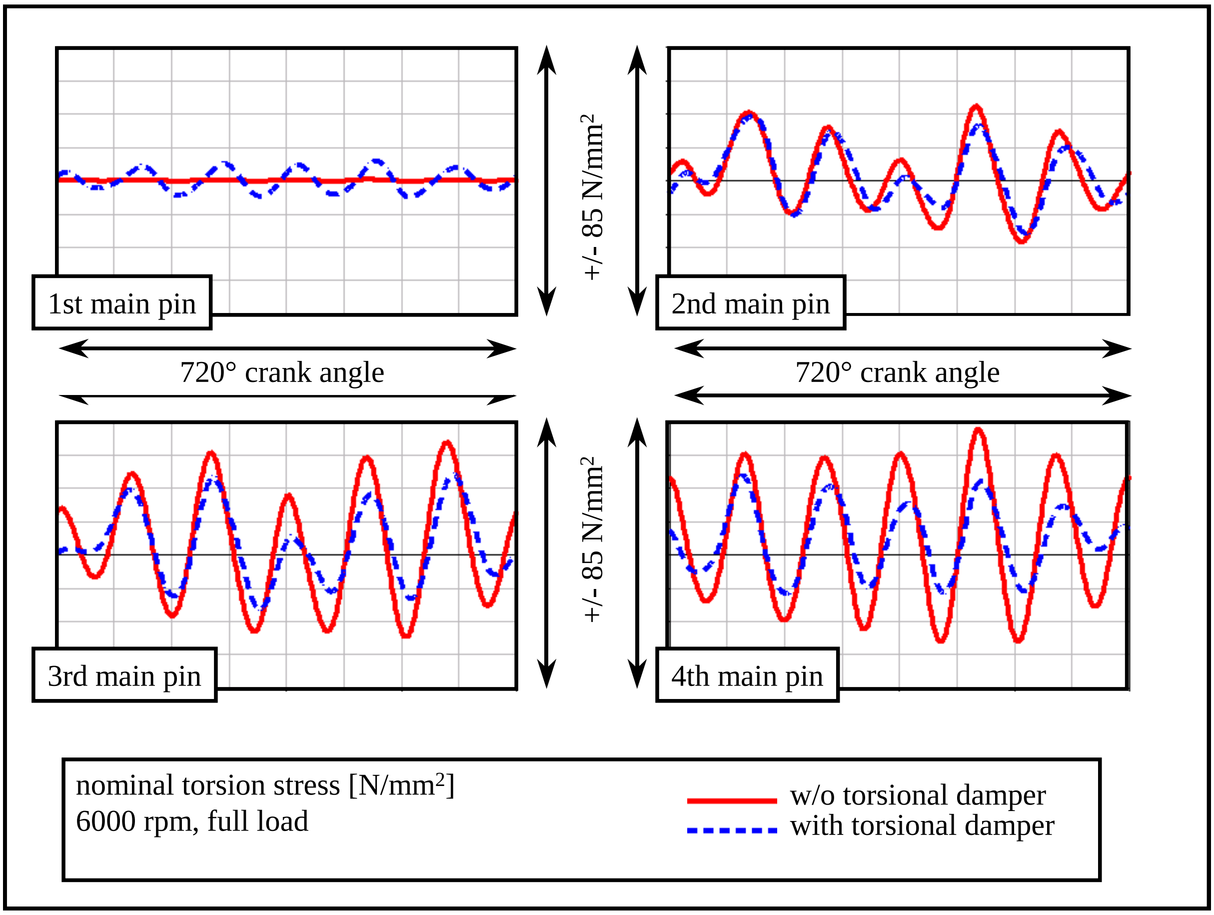

Figure 4: Influence of the torsional vibration damper on the nominal torsion stress

As expected, the stress amplitudes are higher at the flywheel end of the crankshaft. The torsional vibration damper has its biggest effect on the most critical main pin number four. The maximum amplitude is lowered by 40%. At the second and third main pin the stresses are also reduced noticeably. Only at the free end of the crankshaft the stress increases but still remains below the other main pins. This can be explained with the fact that in order for the damper to function it introduces a moment into the crankshaft, which otherwise does not exist.

2.1.4 Optimization of the internal Mass Balancing

During the first balancing step as described in 2.1.2 counter weights were used to eliminate the rotating first order moments. The internal mass balancing was not considered at that point. To reduce the bending moments in the crankshaft and the bearing loads it is required that the counter weights are further refined. The resulting deviation moment as determined in the previous calculation has to be a boundary condition in this process.

The basis for the refinement of the counter weight design in order to minimize the bending moments of the crankshaft is, to achieve a relatively high balancing rate with respect to each cylinder. Other influences such as the ventilation of the crankcase have to be considered as well.

For the calculation of the bending stresses and bearing loads it is required to increase the level of refinement such that beam elements are connecting the parts representing the main pin, web, and crank pin. The stiffness and damping of the beams are calculated based on the geometry and material data of the crankshaft.

It is also recommended to increase the refinement level with respect to the connection between crankshaft and block. This means that the constrained based connections are replaced against hydrodynamic bearing connections.

Figure 5 shows on the left side the influence of the optimized counter weights on the nominal bending stresses in the six main webs. It becomes obvious that the crankshaft with optimized counter weights has significantly lower bending stress amplitudes in particular in the outside webs.

Another objective of the balancing is to ensure an even loading of the bearings as a result of the combined gas and mass forces. On the right side of figure 5 the bearings loads are shown. The improvement is even more apparent with respect to the bearing loads especially for the 2 center bearings. Here the loads have been reduced by 50%. The modified balancing of the crankshaft has significantly improved the loading of the crankshaft and the bearings.

Since the modified counter weights may have changed the torsional behavior of the crankshaft as previously calculated, the steps described in 2.1.3 to determine the torsional stresses should be repeated. In the case of the example engine the effect can be considered as irrelevant.

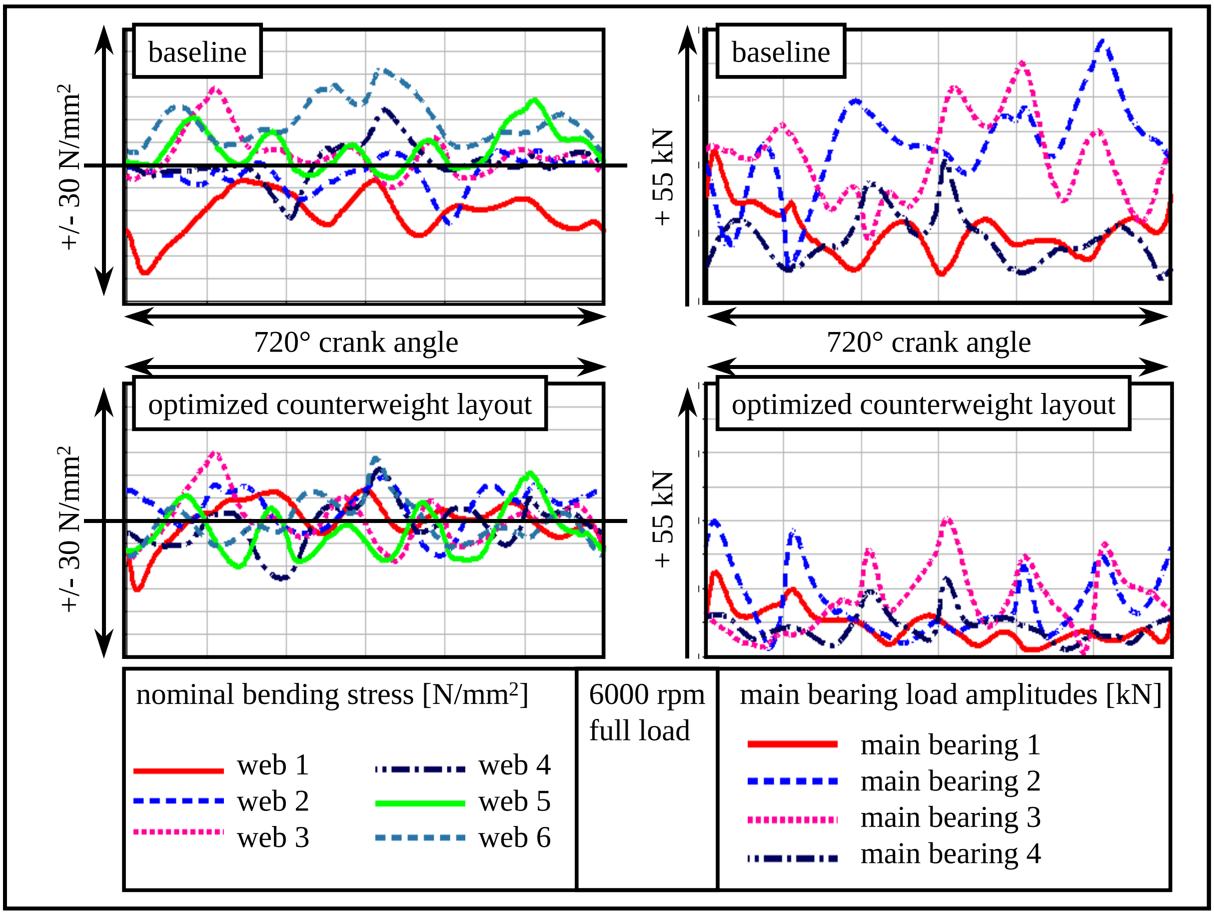

Figure 5: Optimized counter weights: Bending stresses and bearing loads

2.1.5 Hydrodynamic Bearings

Since the hydrodynamic bearings have been used in the previous step a number of characteristics of the bearing are readily available.

Although the pin orbital movement is not perfectly accurate using this rigid body based approach with beams, it is possible to draw comparative conclusions. Additionally there is quite some expertise available with regard to this approach, such that these results can be used in the engineering process.

Especially the orbital movement of the pin allows for a deeper understanding of the hydrodynamic behavior of the bearing. Figure 6 shows the pin orbital curves of all four bearings for the previously described counter weight variations. With the original counter weight layout as described in 2.1.2 the orbital curves have a circular characteristic, which already indicates a bad balancing of the internal moments. In the optimized design resulting from 2.1.4 the orbital curves show a qualitatively different characteristic with much lower amplitudes, which means that the remaining oil film thickness is higher such that the safety against failure is increased. This can also be derived from the plots on the right side of figure 6.

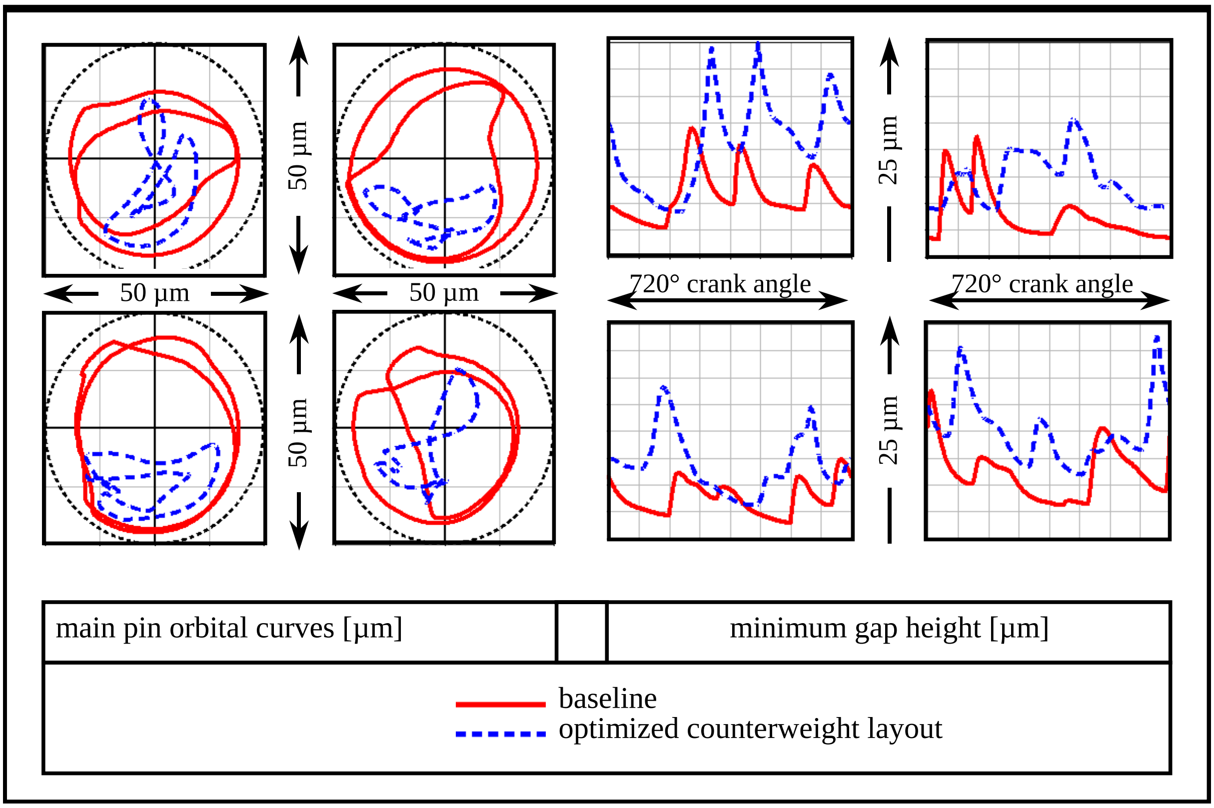

Figure 6: Optimized counter weights: Influence on the hydrodynamic bearings

3 Conclusion

In conclusion it can be said, that within the same model and with a limited amount of input data it is possible to determine the mechanical characteristic of the crank train early in the development process. Without switching the software package the model can be refined to the level appropriate for the task at hand. The integration of flexible structures is feasible and represents the next higher level of refinement.

4 Literature

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