

Crankcase and Crankshaft Coupled Structural Analysis Based on Hybrid Dynamic Simulation

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ABSTRACT

This paper presents the comparison of two different approaches for crankcase structural analysis. The first approach is a conventional quasi-static simulation, which will not be detailed in this work and the second approach involves determining the dynamic loading generated by the crankshaft torsional, flexural and axial vibrations on the crankcase. The accuracy of this approach consists in the development of a robust mathematical model that can couple the dynamic characteristics of the crankshaft and the crankcase, representing realistically the interaction between both components. The methodology to evaluate these dynamic responses is referred to as hybrid simulation, which consists of the solution of the dynamics of an E-MBS (Elastic Multi Body System) coupled with consecutive FEA (Finite Element Analysis).

For this study, an in-line 6-cylinder diesel engine was selected, where the crankcase design has to be revised to withstand higher loads due to new calibration targets by keeping design changes at a minimum and taking mass and cost savings into account. As result, the crankcase fatigue design margins are presented showing clearly the benefits of the adopted methodology, which enabled fulfilling the objectives of the project through small changes in the design. The crankshaft dynamics stresses and safety margins are also presented in this paper.

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INTRODUCTION

Considering the trends for internal combustion engines and specifically talking about diesel engines for commercial vehicles, it is clear that while the power output increases for every new application, the mass, fuel consumption and engine final cost must develop in the opposite direction. From these points of view, one relevant aspect is that the downsizing of the engines is an actual trend [1].

The engine considered in this study was calibrated to achieve Euro-V legislation targets without significant changes in basic components utilized in previous applications. This new engine performance presents 6.5% higher power output and 18% higher torque.

Taking into account these input variables, the crankshaft and crankcase durability were calculated considering conventional methodologies which are accepted worldwide by many engine manufacturers, i.e. dynamic simulations for

the crankshaft and quasi-static simulation for the crankcase at main bearings region.

After the end of the calibration phase and after some durability tests at the dynamometer, some cracks occurred in the block at the flywheel side in a fillet area, close to the main bearing cap. The first conclusion from the failure analysis indicated some possible unbalance of the propeller shaft at the test bed, but a deeper analysis of the problem highlighted that the bending vibrations of the crankshaft was the root cause of the crankcase failures.

There are some possibilities to reduce crankshaft flexural vibrations by increasing its bending stiffness. One effective design change is to increase the overlap of the crankshaft by modifying the crankpin and/or journal diameters. However, this proposal has many side impacts on the design affecting many other components, including the crankcase, which could cause increased engine mass and final product costs.

Since reducing the component's dynamic response by changing the overlap is prohibitive, the adopted solution to

solve this issue was the introduction of a steel plate ladder frame at the bottom of the crankcase, to increase the stiffness of the main bearings and by this way, reducing the stress amplitudes at the fillet area. To certify the effectiveness of this proposal, hybrid simulations were performed for the current crankcase and for the component with the ladder frame.

As mentioned before, the increase of the engine output in combination with the lightweight design of the components results in new physical effects that cannot be detected by conventional calculation methods. In this specific case, even with almost the same PCP (Peak Cylinder Pressure) the dynamic response of the crankshaft (in bending) presents higher amplitudes that introduce substantially higher forces on the main bearings, overcoming the fatigue strength of the crankcase material.

For torsional vibration analysis, the conventional mathematical model depicted in reference [2] delivers accurate output results for angular amplitudes, dynamic torques, dissipated power of the TVD (Torsional Vibration Damper) and shear stresses at rubber dampers. In case of the crankshaft stress calculations, this model is limited and shall be used with caution, mainly due to the strong coupling between bending and torsional vibrations, which is common in modern engines.

Reference [3] describes the current methodologies considered for crankshafts simulation. A comparison between conventional approaches and hybrid simulation (E-MBS / FEM) is presented and it is shown that, a crankshaft, which is considered to be borderline or critical by conventional methods, can be proven to be oversized or safe through hybrid simulations.

For any trouble shooting or optimization attempt it is very important to be able to understand the local dynamics in the critical regions and to survey the possible triggering mechanisms. In that aspect elastic multi body simulations and hybrid methodologies are indispensable as they offer a realistic approach, which is able to build up most of the dynamic aspects accurately and offers an unlimited amount of measurement possibilities to analyze those dynamics.

The simulated in-line 6 cylinder engine has the following data:

- Power output: 243 kW @ 2200 rpm
- Maximum torque: 1300 Nm @ 1200 - 1600 rpm
- Displacement: 7.2 liters
- Peak cylinder pressure: 180 bar

THEORETICAL MODELING

Basically, the hybrid simulation is divided into two steps. Initially, the responses of the components to the excitation generated by the combustion and inertial forces of rotating and reciprocating masses are computed through the elastic multi body simulation. At this point crankshaft stresses can be calculated taking into account torsional vibrations,

bending and axial vibrations, gyroscopic effects, dynamic unbalance, flywheel tumbling, etc.

As next step, the main bearing dynamic loadings are exported to an implicit nonlinear FE model, which will calculate the total stresses on the crankcase for a durability analysis.

Hybrid Model for Dynamic Simulation of the Crankshaft & Crankcase

The FE model for the hybrid simulation is created with 2nd order elements for the crankcase, crankshaft, main bearings, flywheel and pulley, resulting in a model with 3.5 million nodes. Thus, it is possible to determine accurately the strain/stress variations in time domain for any element of the model. On the other hand, the simulation is linear and thus, contacts or other nonlinearities cannot be considered in the model.

Figure 1 shows the methodology for hybrid simulation

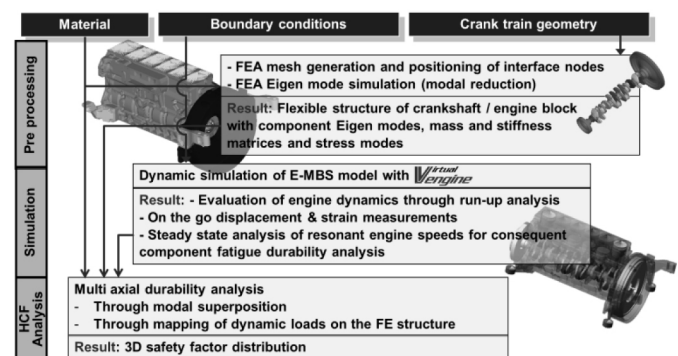


Figure 1. Hybrid simulation methodology

The E-MBS model consists of the flexible bodies for crankshaft, flywheel and engine block prepared according to well established Craig-Bampton method [8]. All the other bodies are modeled as rigid bodies.

A linear torsional vibration damper model is created based on the data from the viscous damper supplier that is installed on the engine.

The coupling between crankshaft and crankcase is modeled by 3D hydrodynamic bearings which consider the elasticity of the components. This is a well established and thoroughly proven model, which has been available in *FEV Virtual Engine* since a long time. It bases on the Buntenschoen method, however in addition accounts for the bearing tilting through an extension in the hydrodynamic theory. Before the simulations Reynolds equations are numerically solved for bearing diameter/length ratios from 0.75 to 8.0 to create predefined tables for data interpolation. The oil viscosity and supply pressure are also included in the mathematical model. For more information on the theoretical models please refer to references [6] and [7].

During the multi body simulation, main bearing forces are transmitted to the crankshaft and engine block through interface nodes on the flexible bodies. Each main bearing

consists of 12 nodes on bearing shells and 3 nodes at each journal as shown in Figure 2. This consideration allows the simulation of the misalignments and eccentricities between the pin and bore due to the dynamic deformations of the components. It is important to highlight that only compressive forces are taken into account for the load interaction between both components.

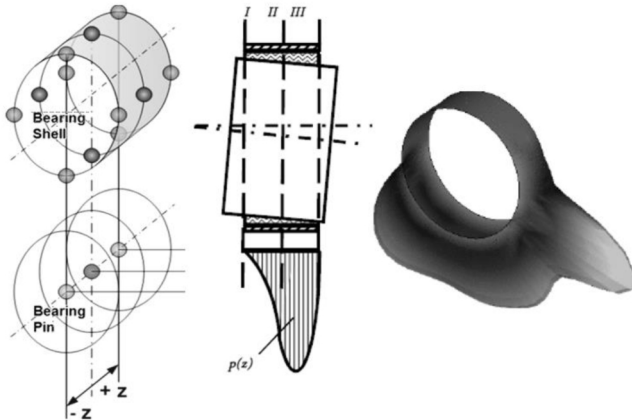


Figure 2. Model of 3D hydrodynamic flexible bearings

On the go measurements were carried out at several locations in order to better understand local dynamics and to carry out comparisons, on one hand between measurements and simulations, on the other hand between current engine and the design with ladder frame. Measurements can include flywheel tumbling (or wobble), strain at critical areas, bearing tilting, etc. Transient rpm sweep analyses are carried out as a first step in order to:

1. Determine the resonant engine speeds for which consequent engine block and crankshaft durability analysis are to be carried out.
2. Inspect local dynamics and phenomena causing bulkhead failures.
3. Verify the E-MBS model by comparing measurement and simulation results.
4. Compare engine dynamics of the variants without and with ladder frame.

The resonant and rated engine speeds are further investigated through steady state analysis to provide the necessary input data for component durability analysis.

Crankshaft Durability Analysis

Crankshaft durability analysis is carried out with the commercial fatigue software *FEMFAT*. Figure 3 depicts the methodology to determine the 3D crankshaft dynamic stress distribution.

From steady state simulations with *FEV Virtual Engine* “modal participation factors” are exported to the fatigue software. Those factors describe participation of each stress

and deformation mode to the global stress and deformation state of the crankshaft at any crank angle throughout an engine cycle.

Stress synthesis - superposition of modal participation factors and modal stresses - is carried out in the fatigue software, which results in the 3D stress data in time domain over one engine cycle.

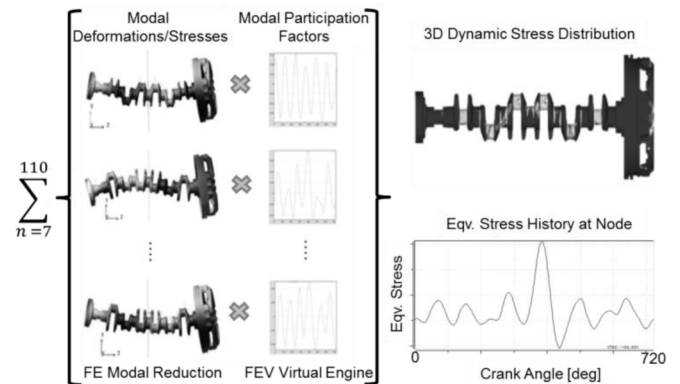


Figure 3. Stress Synthesis in FEMFAT

Another important aspect of durability simulations is the material model, which in this case was prepared based on fatigue test data. Material properties of the crankshaft are included in Table 1. This data is given at a survival probability of 95% and was acquired through fully reversed bending fatigue test on the crankshaft throws. Hence, it includes all the statistical and geometrical correction / influence factors. Therefore in calculating the local Haigh diagram at nodes, only the influence of mean stress, stress gradient and selected survival probability are considered.

Table 1. Crankshaft Fatigue Properties

Material	SAE 1548
E	206800 MPa
ν	0.29
UTS	1100 MPa
$\sigma_{bw} (*)$	758 MPa

(*) 95% survival probability for induction hardened regions.

A material is created based on the experimentally determined bending fatigue strength and material classification utilizing the *FEMFAT* material generator with FKM principles. In order to create the material model with the bending fatigue strength given in Table 1, the material UTS input is varied until the achievement of the correct bending fatigue strength. This approach results in a material model with an UTS different than the actual material, introducing only small inaccuracies at high mean stresses. However, this consideration results in negligible effects for a component like the crankshaft, which undergoes low levels of mean stresses (approximately 150 MPa). The resulting Haigh