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Dynamic gear wheel simulations using multibody dynamics

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Summary. Simulation of the gear train is an important part of the dynamic simulation of the power train of a medium speed diesel engine. In this paper, the advantages of dynamic gear wheel simulation as a part of the flexible multibody simulation of a complete power train are described. The simulation is performed using AVL EXCITE Power Unit.

Key words: flexible multibody dynamics, power train, gear train

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Introduction

The power train of a medium speed diesel engine consist of a crankshaft, connecting rods, intermediate gears, and cam shafts. Flexible multibody dynamics of the whole power train is used to reliably capture the dynamics of the system (Figure 1). The important part of the power train simulation is to accurately calculate the dynamics of the gear train from the crankshaft to cam shafts through intermediate gears. Gear train dynamics affect the torsional vibration properties of the camshafts and crankshafts. An accurate simulation of the torsional vibrations of the power train is needed to avoid possible resonances within the speed range of the engine. Gear mesh forces are usually non-constant and include high impacts due to the torsional vibrations of the shafts and fluctuating loadings. These impacts are causing high frequency vibrations to the shafts and engine structure. The consequence of high frequency vibrations of the structures can be structure-borne noise and microscopic sliding in bolted contacts which may lead to fretting damage [8, 9]. Also dynamic loads to the shaft bearings, caused by gear mesh forces, need to be taken into account when designing the bearings.

The Wärtsilä Virtual engine concept is developed to simulate engine dynamics as accurately as possibly by combining the state-of-the-art simulation methods with field experience [2, 7]. The flexible multibody dynamics software AVL EXCITE Power Unit is applied to the power train simulation. The boundary conditions for the simulation are defined by measurements on laboratory engines, and the load profiles are derived from field engines.

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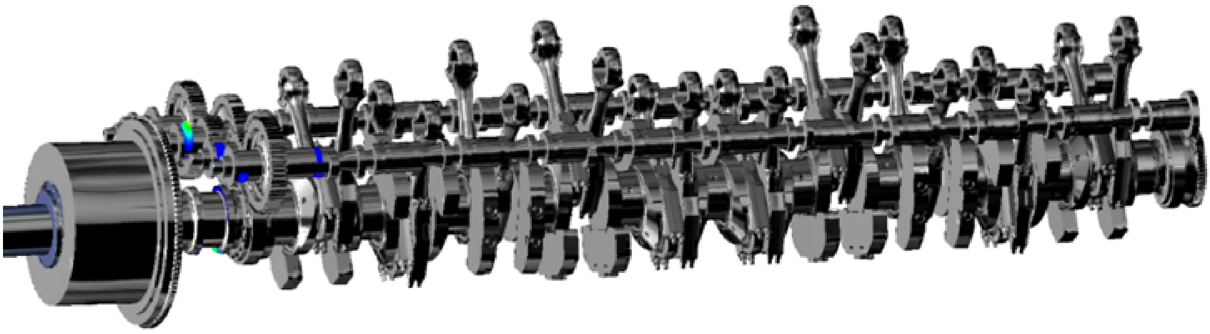

Figure 1. Multi body simulation model of power train of medium speed diesel engine

Flexible multibody simulation

Flexible multibody dynamics are a way to simulate complex systems consisting of flexible bodies and the interaction between them. Interaction between bodies can be modeled with detailed models as well as by simple pre-calculated non-linear spring-damper elements. The simulation is done in a time domain, which enables to capture the non-linear and transient behavior of the system.

Computation of global motion and deformation of bodies is done by solving the equation of motion

$$\boldsymbol{M} \cdot \ddot{q} + \boldsymbol{D} \cdot \dot{q} + \boldsymbol{K} \cdot q = f^{ext} + p^{*}, \tag{1}$$

where $\boldsymbol{M}$, $\boldsymbol{D}$ and $\boldsymbol{K}$ are mass, damping and stiffness matrices and q is displacement vector. External force vector f^{ext} is sum of joint forces and external loads. The non-linear inertia terms p^{*} consider the inertia components resulting from the global motions of the body. Floating frame of reference formulation is used to separate the global motions of the body from the deformation of the body. [4]

The complete multi-body dynamics model of the power train includes the main components as flexible bodies. The sliding bearings are modeled using hydrodynamic joints or non-linear spring-damper elements. For the interaction between the shafts, simple or more advanced gear joints are used. The level of detail in the model is balanced between the wanted accuracy and simulation time. An illustrative example of bodies, joints and loadings needed for the steady state simulation of crank train is shown in Figure 2. In the example linear elastic bodies from condensed finite element model with global motion(CON6) and without global motion(SMOT) are used. Also rigid body with global motion(RI3D) is used. [10]

Elasto-hydrodynamic (EHD) joints are used to simulate the dynamic behaviour of the slider bearings in a power train system in a very detailed way. An EHD joint allows the shaft to translate and rotate within the clearance of the bearing, and it also takes into account the flexibility of the bearing housing. These features are crucial to calculate the change of radial clearance and misalignment of gear pairs in the flexible multibody dynamics. [1]

Modeling of gear mesh dynamics in a multibody dynamics environment can be done with a wide variety of possible approaches from the 1D torsional spring-damper element to the contact-based model of mating flanks. The advanced cylindrical gear joint is the most comprehensive feature in Excite Power Unit to simulate all the necessary physics of the gear pair. The calculation of meshing stiffness is done on the basis of real geometrical

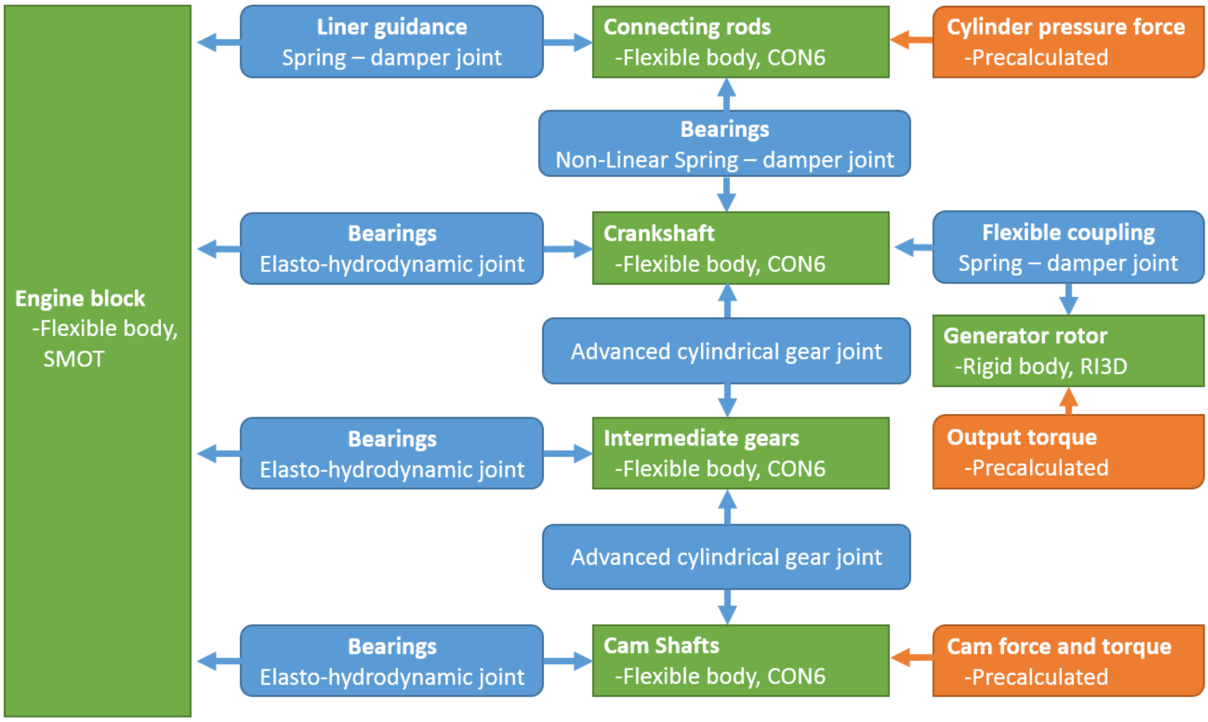

Figure 2. Bodies, joints and loads for steady state multibody dynamics simulation of crank train

properties and relative movement of gear pair. The calculation model can take into account instantaneous changes in meshing stiffness due to tilting and misalignment of gears, the number of flanks in contact and the deformation of the gear wheel and tooths. Calculation of contact and backlash damping forces is done on the basis of oil viscosity, surface roughness and gear flank curvature. [5]

Results

The main results from the dynamic simulation of gear train dynamics are the normal force, damping force and frictional force between flanks separately. Other useful results of the advanced gear joint are the torque variation, dynamic transmission error, power loss, and changes in the operational distance between the gears. Gear joint results can be used for dimensioning of the gear pairs. The old method to design, by using nominal loads multiplied by dynamic factor and other estimated factors, can be replaced by dimensioning by actual loads. This can prevent over-designing. Local root stresses and actual flank contact pressure including mesh modifications and corrections are calculated internally by the software. These values can be used directly for fatigue life analysis.[5]

Structure-borne noise excited by gearwheels can be analyzed if the surrounding structure is included in the simulation model. Gear wheels are causing wide frequency range noise called rattle and narrow banded noise called whining. Rattle noise is caused by the loss of contact between flanks and backlash impacts. Periodic fluctuations in the gear mesh is causing the whining noise mainly due to geometrical imperfections and the change of the number of flanks in simultaneous contact. [3, 11]

Conclusion

The results from a properly made simulation model of the whole crank train correspond accurately with measurements made on the engine. Power train simulation models are validated by torsional vibrations measurements from crank shaft and camshafts and by acceleration measurements from crankshaft counter weights [6].

Figure 3 compares a measurement data and steady state simulation results from the last two four-stroke cycles after the transient error from the beginning of the simulation has faded out. The torque results reveal that the torsional natural frequencies of the whole power train system are correct. The strength of the gear wheel impacts can be seen from the high frequency acceleration results. The simulation and measurements would not match so well if the gear wheels were excluded from the simulation model.

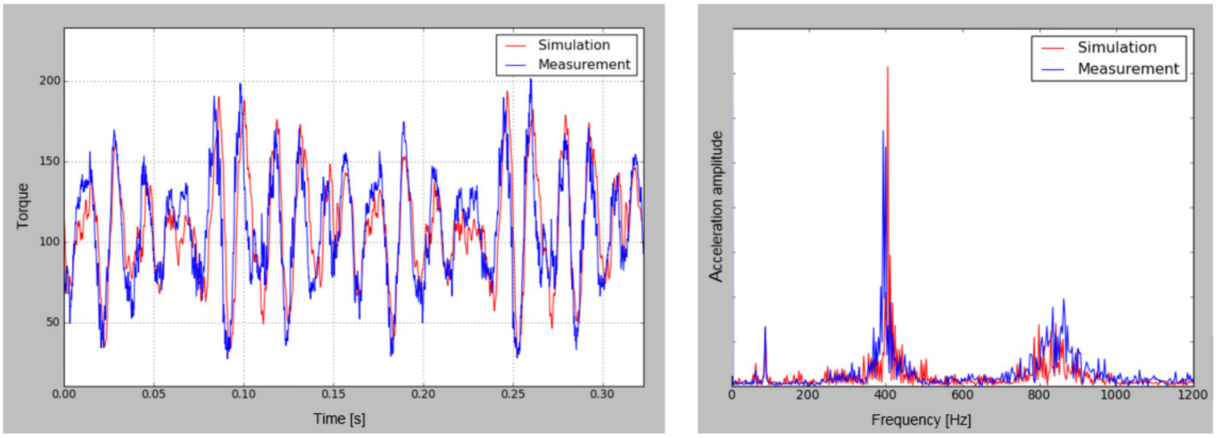

Figure 3. Torque and acceleration measurements from crankshaft compared to simulation results

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