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DEVELOPMENT OF COMPUTATIONAL MODEL OF PISTON DYNAMICS BEHAVIOUR

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Abstract: *Sooner or later, the development process of each technical product, inevitably, reaches a phase when several other design proposals are being compared. But based on what is the best design chosen? Engineers all around the world have a challenging task to find the best compromise among numerous aspects—e.g. manufacturing, cost, functionality, reliability etc.—especially in contemporary phenomenon of endless minimization of all kinds of losses. To estimate the operational lifetime characteristics, i.e. wear, friction, noise and vibration and others, test benches are widely used. Most of the time, they offer accurate results, but the insight in to the physics is lost. On the other hand, simulation tools can clarify the details during the product's operational cycle but the disadvantage lays within the adequacy of the algorithms used. Therefore, computational models are often verified by the test results. Only then their full potential can be exploited, providing fast and reliable results with translucent insights into the true physics behind it. The following paper presents the development of one such computational model focused on the investigation of piston dynamics behavior.*

Keywords: Piston, Tribology, Mechanical loss, Multibody dynamics, Elastohydrodynamic lubrication

1. Introduction

The piston of internal-combustion engine (ICE) is primarily designed to transfer released fuel energy into the mechanical work as a rotational motion of the crankshaft. This way piston, together with piston rings, has an additional task to seal the combustion chamber (in order to prevent the exhaust gas leakage into the crankcase) and to dissipate the released heat energy into the liner. In addition, the piston crown shape is designed to enhance the air/fuel mixture creation.

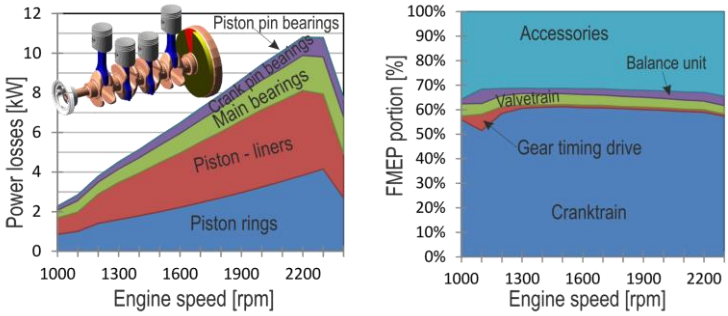

Fig. 1: Mechanical loss contribution in a 4.2L diesel engine (Novotný et al., 2010)

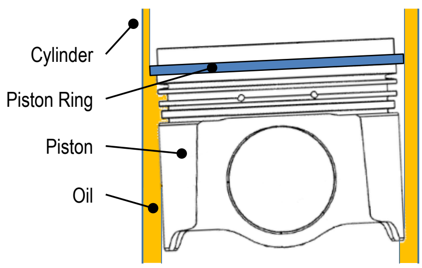

Fig. 2: Piston/Liner Interaction

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The piston has to fulfill requirements such as structural strength, adaptability to operating conditions, low friction, low wear, seizure resistance and simultaneous running smoothness, low weight with sufficient shape stability, low oil consumption, and low pollutant emissions values (Mahle GmbH, 2012). The piston group is the main contributor to the overall ICE mechanical loss as depicted in Fig. 1.

2. Theoretical background

The piston has to withstand very high thermal and mechanical loads during its demanding utilization. It is guided by piston skirt/liner interaction via lubrication oil – journal bearing (Fig. 2).

2.1. Hydrodynamic and asperity contact model

Oil response is mathematically represented by the Reynolds equation (RE) in the following form:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p_H}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\eta}\frac{\partial p_H}{\partial y}\right)-\frac{\partial\left(u\rho h\right)}{\partial x}-\frac{\partial\left(\rho h\right)}{\partial t}=0 \tag{1}$$

where x is direction of liner rotational axis, y is circumferential direction, h is oil film thickness, p_H is elastohydrodynamic (EHD) pressure, u is relative sliding velocity, η is oil dynamic viscosity and ρ is oil density. Oil film thickness consists of the following contributors:

$$h = h_{rigid} + h_{thermal}^{piston} + h_{elastic}^{piston} + h_{thermal}^{liner} + h_{elastic}^{liner} \tag{2}$$

where h_{rigid} is the clearance between the piston (undeformed with specific ovality and contour shape) and the liner (deformed due to the assembling – head bolts, gasket, etc.), $h_{thermal}^{piston}$ and $h_{thermal}^{liner}$ are the thermal expansion of the piston and the liner, respectively; $h_{elastic}^{piston}$ and $h_{elastic}^{liner}$ are the elastic deformation of the piston and the liner, respectively.

The asperity contact pressure may be calculated by the Greenwood and Tripp (1970) as:

$$p_c = \frac{8\pi}{5}\left(\eta\beta\sigma\right)KF_{5/2}\left(\frac{h}{\sigma}\right) \tag{3}$$

where p_c is the pressure caused by the contact of surface solids, β is the radius of curvature at asperity peak, σ is the standard deviation of the sum of the summit heights, and $F_{5/2}$ is the statistical function for Gaussian distribution of the summit heights.

Oil film thickness h is affecting the hydrodynamic and asperity contact pressure and vice versa (oil density and viscosity may be pressure dependent as well). Therefore, Gauss-Seidel solver enhanced by the Successive Over Relaxation (SOR) method with overrelaxation parameter is used.

2.2. Elastic deformation

To be able to calculate the overall oil film thickness h, elastic deformations have to be known. To do so, Multibody Dynamics (MBD) software, namely MSC Adams (Chapter 3.1.), can be augmented by the flexibility of simulated bodies. In this case, the elastic deformation is assessed as the combination of multiple mode shapes – modal superposition. It has to be kept in mind that this approach assumes only small linear deformations relative to a local reference. The number and type of chosen modal shapes determines the accuracy of calculated deformed shape. To be able to detect correct deformations, the Craig-Bampton method is applied (MSC Software, 2015).

2.3. Thermal load

To predict the piston behavior accurately, the temperature distribution has to be estimated since it affects not only mechanical properties but also thermally deformed shapes – $h_{thermal}^{piston}$ and $h_{thermal}^{liner}$ in eq. (2). For this purpose Computational Fluid Dynamics (CFD) simulation can be used. Unfortunately, this method is very sensible to the input variables which are not always known. Therefore, a much more industrially used approach is preferred for the piston temperature. It is the calibration of thermal Finite Element Analysis (FEA) to the measured peak temperatures at the piston's specific locations. This may be done by use of templugs inserted into the measurement locations of the piston. The temperature is estimated by the decrease in the templug hardness (Mahle GmbH, 2012). The liner does not perform large body motion as the piston does, therefore the temperature distribution may be measured by the thermocouples placed in the specific distances across its thickness.

3. Simulation strategy

Commercial MBD software (MSC Adams) is used for the solution of the piston/liner model. It solves the force balance and the equation of motion of the whole system with respect to the nonlinear system response, i.e. force/motion two-way dependence arisen from the RE eq. (1). The EHD pressure is calculated simultaneously in the user-written subroutine in Fortran programming language. The solver/subroutine exchange is clear from Fig. 3.

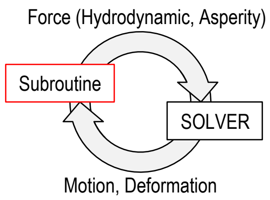

Fig. 3: MBD/subroutine exchange

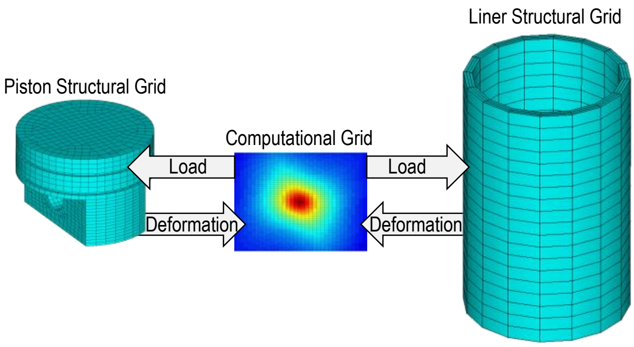

Fig. 4: Mapping process with different structural and computational grids

3.1. Mapping

For the HD and asperity contact solution (Chapter 2.1.) the computational grid is fixed to the piston. Since this grid simulates the piston/liner interaction, the calculated load distribution has to be mapped onto the liner surface. Furthermore, the liner deformation response has to mapped back onto the computational grid (Fig. 4). There are two kinds of mapping: the load and the deformation mapping, each having different requirements.

The load mapping has to preserve force and momentum static equilibrium between the *source* and the *target* – the *source* and the *target* load has to lead to the same body dynamic response. The *source* pressure distribution is transferred by the bilinear shape functions into the analytical form. Then it is analytically integrated per each of *target* cells – *target* force distribution. This approach satisfies the previously mentioned requirements.

For the purpose of good numerical stability, the deformation mapping has to lead to the smooth deformation shape. The 2D cubic Hermit spline is created from the *source* deformation and the *target* deformations are estimated as a functional values of this cubic spline.

Further investigations lead to the fact that computational grid has to be much finer than the structural one. Very fine structural grid is causing long computational times, because it is a major contributor of Degree of Freedom (DOF) in the simulation. In order to avoid that, different structural and computational grids are used on the piston – load and deformation mapping is required.

4. Example of results

The simulation of 4-stroke Spark Ignited (SI) single cylinder engine with bore of 86 mm and stroke of 86 mm is used. The nominal radial clearance is set to 50 μm, skirt profile and piston dimensions are taken from McNally (2000). Only piston-side EHD simulation is performed, with fully-flooded conditions, constant oil density and viscosity. Piston secondary motion is one of the most significant piston behaviors which is often experimentally measured. It affects engine optimization parameters like noise, friction loss, wear, etc.

Graph in Fig. 5 shows the piston secondary motion in a firing engine at 1000 rpm. The major portion of the side force is caused by the gas pressure acting on the piston crown. At the Top Dead Centre (TDC) the piston changes position from one side to the other very rapidly – piston slap. The biggest impact of the piston's elastic deformation is during the expansion stroke, where the higher loads are present. Overall piston motion history is smooth and damped – effect of the fully-flooded oil conditions.

Some authors introduce piston flexibility only as the stiffness matrix of the piston skirt nodes. However, looking at the piston deformation in Fig. 6, the piston pin bosses are deformed as well and significantly affect the piston skirt shape.

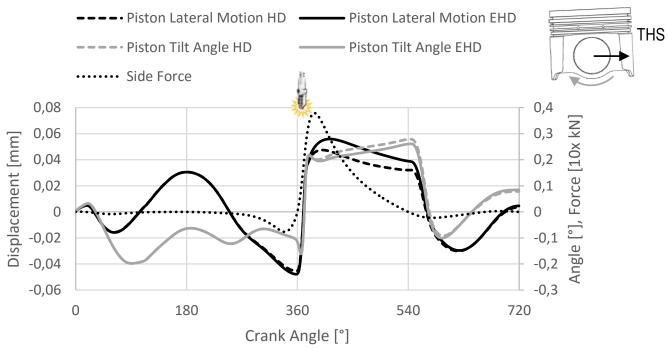

Fig. 5: Piston secondary motion by HD and EHD solution

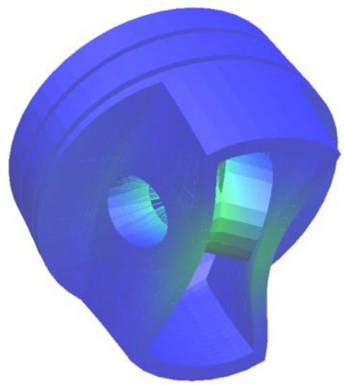

Fig. 6: Piston deformation due to the flexibility of pin bosses

5. Conclusion

The presented computational model is combining advantages of two approaches. Firstly, it uses commercial MBD software which is very stable and meant to simulate varied set of dynamic analyses. Secondly, developed computational model augments commercial MBD to be able to catch the most of the major physical processes acting on the piston during the operational cycle.

However, there are some difficulties. Current HD solver in user-written subroutine (SOR) does not seem to be the most efficient when finer computational grids are applied – multigrid might be needed. Too many time consuming mapping processes are calculated during each of the MBD solver iterations – optimization is inevitable.

The usage of the flexible liner incorporated into the cylinder block would allow to investigate the engine block noise and vibration caused by the piston motion – piston shape optimization in terms of NVH.

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