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Advanced Computational Analysis of Connecting Rod of an Aircraft Engine

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Abstract: Development of internal combustion engine's components is based on the use of the advanced computational models in order to compare and verify the individual design proposals. Connecting rod, which performs a general planar motion, is exposed to the gas pressure forces, inertia, contacts and hydrodynamic pressure during the engine operation cycle. To incorporate all these aspects, Finite Element Method (FEM) is extended by Finite Difference Method (FDM) simulating a slide bearing of connecting rod's eye. It includes different properties of lubricating oil (pressure and temperature dependent viscosity and density) and elastic deformations, so the pressure distribution in an oil film can be evaluated. The computational process finishes with an estimation of the endurance safety factor of the connecting rod. The four-cylinder inverted aircraft engine is used as an example.

Introduction

Countless number of structural analyses of connecting rods are available among the engineers. Widely popular commercial software usually provide the life prediction tools for mechanically loaded parts, e.g., by the force, acceleration, rotary speed, etc. But only few are able to compute pressure distribution in the lubricant oil film with or without elastic influences – elastohydrodynamic (EHD) or hydrodynamic (HD) theory.

Theoretical background and numerical methods

In many cases oil film thickness shape is determined exclusively by the rigid geometry of the running surfaces. Other situation occurs when hydrodynamic pressure is causing the active surfaces to deform, changing the oil film thickness shape and affecting the pressure distribution reversely. It happens by means of high hydrodynamic pressure (gear teeth, roller bearings, etc.) or by means of high flexibility of contact regions (connecting rod's eyes).

Oil film thickness in EHD theory is calculated as follows:

$$\boldsymbol{h}(x,z) = \boldsymbol{h}_{rigid} + \boldsymbol{K}_{eye}^{-1}\left(\boldsymbol{p}_{hydro} + \boldsymbol{p}_{rough}\right) + \boldsymbol{K}_{pin}^{-1}\left(\boldsymbol{p}_{hydro} + \boldsymbol{p}_{rough}\right) \tag{1}$$

where $\boldsymbol{h}_{rigid}$ is the external deformation e.g. due to the bolt pretension, $\boldsymbol{K}_{eye}$ and $\boldsymbol{K}_{pin}$ are the stiffness matrices of the connecting rod and the pin, respectively, $\boldsymbol{p}_{hydro}$ is the hydrodynamic pressure and $\boldsymbol{p}_{rough}$ is the pressure caused by the contact of the surface solids. The asperity contact is simulated with Greenwood and Trip [1].

Since EHD lubrication is a Fluid-Structure-Interaction (FSI) problem, Reynolds differential equation is used. It is rewritten using dimensionless variables to improve the stability properties of numerical solution. This equation is non-linear due to several phenomena like cavitation, density and viscosity dependence on the oil pressure and temperature. Viscosity-pressure relation can be expressed by Barun, or more accurate Roeland [2] as an exponential – oil viscosity η can vary very widely in the contact region (from 10 to 10^{15} times higher than the nominal viscosity η_0 [3]). This is the cause of one of the main problems in the simultaneous numerical solution of discretized

Reynolds equation and discretized oil film thickness equation. Therefore, Gauss-Seidel solver enhanced by the Successive Over Relaxation (SOR) method with underrelaxation parameter is used.

The last mandatory equation, that needs to be solved, is the force balance condition – the integral over the pressure in the oil film and the asperity pressure balance the externally applied load. In this case, Newton-Raphson Algorithm (NRA) is used:

$$\overline{\boldsymbol{X}} = \widetilde{\boldsymbol{X}} - \omega_F \boldsymbol{J}^{-1} \boldsymbol{F}(\widetilde{\boldsymbol{X}}) \tag{2}$$

where $\boldsymbol{X}$ is matrix of coordinates (overline and tilde stand for current and previous iteration, respectively), ω_F is line search parameter, $\boldsymbol{F}(\boldsymbol{X})$ is force equilibrium function and $\boldsymbol{J}$ is Jacobian.

Methods and procedures

Two computational models were used, both simulating the connecting rod quasistaticly (no damping). In first, *conventional FEA*, oil film pressure was replaced by the compression-only link elements with parabolic stiffness distribution. The crank and the piston pin were considered as rigid. Second, *FEA augmented by EHD solution*, included only a half model constrained in a cut. Fluid-structure interaction was modelled by EHD pressure applied. EHD pressure distribution was an output from algorithm described in the previous section.

Results and conclusion

Endurance safety factor serves for the results comparison, since it takes the whole time-history loading into account. The major differences are in the connecting rod's bearing as expected (Fig. 1) – the EHD oil film pressure distribution leads to the lower value of the safety factor. EHD solution is mandatory for an adequate stress distribution in the connecting rod's bearings.

Other areas show difference up to 5 % (except the regions close to the constraint boundary condition according to Saint-Venant's principle). Hence, the *conventional FEA* model can be used to accurately investigate regions different than bearings.

The *FEA augmented by EHD solution* replaces displacement constraints in the pin centers by the precomputed oil film pressure. The simulation of the whole connecting rod assembly, including EHD solution of both bearings, requires the Weak spring boundary conditions to enforce the numerical stability.

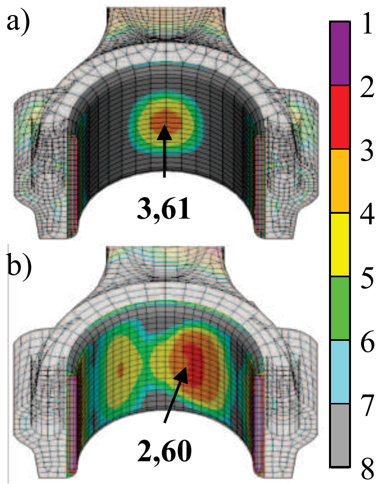

Fig. 1: Endurance safety factor: a) conventional FEA; b) FEA augmented by EHD solution.

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