

An Analysis of Floating Piston Pin

2011-01-1407

Published
04/12/2011Fanghui Shi
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doi:10.4271/2011-01-1407

ABSTRACT

Presented in the paper is a comprehensive analysis for floating piston pin. It is more challenging because it is a special type of journal bearing where the rotation of the journal is coupled with the friction between the journal and the bearing. In this analysis, the multi-degree freedom mass-conserving mixed-EHD equations are solved to determine the coupled pin rotation and friction. Other bearing characteristics, such as minimum film thickness, pin secondary motions in both connecting-rod small-end bearing and piston pin-boss bearing, power loss etc are also determined. The mechanism for floating pin to have better scuffing resistance is discovered. The theoretical and numerical model is implemented in the GM internal software FLARE (Friction and Lubrication Analysis for Reciprocating Engines).

INTRODUCTION

The piston pin is an important element in a reciprocating engine. [Figure 1](#) shows a schematic drawing of a piston pin joint. It transfers the reciprocating motion and combustion force on the piston through the swing motion of the connecting rod down to the rotating motion of the crankshaft. With high load, small bearing load-carrying area and starved lubrication, the piston pin may operate under severe lubricating conditions. There are two types of piston pin configurations depending on the engine specific output. In the fixed pin configuration the pin is press-fit in the connecting rod small end bearing and clearance-fit in the piston pin-boss so the pin rotates together with the connecting rod. In the floating pin configuration the pin is clearance-fit in both bearings such that the pin is free to rotate depending on the friction on the two bearing surfaces. Floating piston pins are regarded to be superior to fixed pins in terms of lubrication,

scuff resistance, and noise generation, and are being implemented in more and more engine designs. A thorough understanding of floating pin interaction with both bearings and the remainder of the piston system is crucial to improve the pin-joint design from the friction, surface damage and noise perspective. A literature survey uncovered very limited work [1,2,3] for either fixed or floating pins. In this paper, a comprehensive theoretical and numerical model for floating pin is presented and some results are discussed. During the course of simulation, the mechanism explaining floating pin's superior scuff resistance vs. fixed pins is revealed. The results also suggest that inclusion of piston pin boss frictional moment in piston skirt analysis can yield more accurate prediction of piston secondary motions.

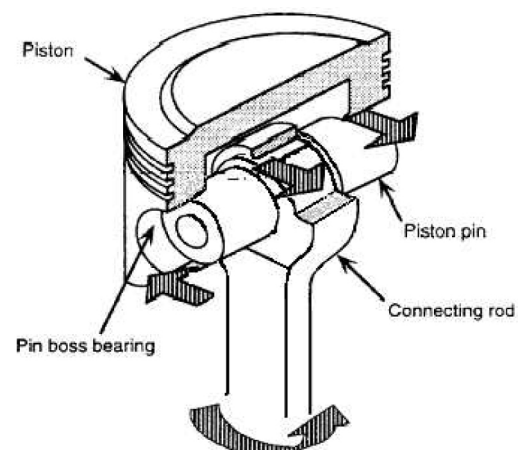


Figure 1. Piston pin joint

MODEL

The floating piston pin bearing basically has two coupled journal bearings (taking into account symmetry, the left is

half of the connecting rod small end bearing, the right is the piston pin boss bearing), as shown in the theoretical model in [figure 2](#). The couplings are through the rotation and elastic deformation of the pin. The bearing loads and connecting rod swing motion are calculated from the dynamics analysis for the piston crank-slider mechanism. Unknowns that need to be solved are the secondary motions, hydrodynamic and contact interactions between the pin and the two bearings. A multi-degree of freedom system has been developed to analyze the lubrication problems for rotating and reciprocating engine parts. In this system, the floating-pin joint is treated as a single five DOF bearing system that has a full bearing groove that separate the connecting rod small end bearing from the piston pin-boss bearing. As shown in [figure 2](#), the five DOFs are four translational bearing to pin relative motions and one rotational pin motion. In addition to the 5DOF rigid body motions, the bearing surfaces and pin surfaces are assumed elastic. Pin bending and ovalization is believed to be an important contributor to the local contact concentration, and thus friction and surface damage. Another crucial effect in pin bearing modeling is the oil starvation or cavitation, since most pin bearings do not have either holes or grooves that are present in most other engine bearings. Without oil supply from the holes and grooves in the sliding direction, the pin-bearing can only get oil in the axial direction through suction effect which is caused by the sub-ambient pressure development when the two surfaces separate. The oil flow is then balanced (assuming periodicity of an engine cycle is reached) by the side leakage due to the squeezing effect.

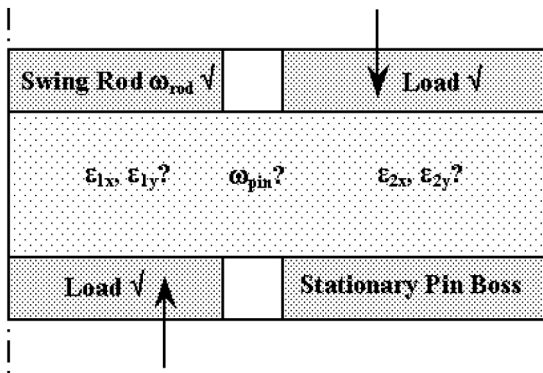


Figure 2. theoretical model

GOVERNING EQUATIONS

In order to solve simultaneously the hydrodynamic pressure in the full film region, the reduced lubricant density in the cavitation region, the surface deformation and the secondary motions of the journal, these unknowns need to be assembled in a single nonlinear system of equations. The following sections provide mathematical and numerical formulations needed to accomplish this goal.

CAVITATION

As described in Shi and Salant [4], the fluid behavior for starvation/cavitation problem is governed by the Reynolds equation for compressible fluids and is constrained by a set of physical conditions. The set of physical constraints are that in the full film region, fluid endures hydrodynamic pressure higher than the known cavitation pressure and is not compressible (known density) while in the cavitation region the fluid is under the known cavitation pressure and has a reduced density (fluid becomes fluid/air mixture). The mathematical equation is simply written as,

$$\nabla \cdot \left(\frac{h^3}{\eta} \nabla p \right) = 6 \nabla \cdot (\mathbf{u} \rho h) + 12 \frac{\partial \rho h}{\partial t} \quad (1)$$

with the constraints,

$$\rho = \rho_{ncav} \quad (x, y) \notin \Omega_{cav} \quad \text{and} \quad \rho < \rho_{ncav} \quad (x, y) \subseteq \Omega_{cav} \quad (2)$$

$$p > p_{cav} \quad (x, y) \notin \Omega_{cav} \quad \text{and} \quad p = p_{cav} \quad (x, y) \subseteq \Omega_{cav} \quad (3)$$

where ρ_{ncav} and p_{cav} are known dimensionless constants,

$$\rho_{ncav} = 1 \quad (4)$$

$$p_{cav} = 0 \quad (5)$$

The film thickness equation and the load balance equation can be written, in matrix form, as,

$$h = c + LP + F\varepsilon \quad (6)$$

$$Jp + M\ddot{\varepsilon} = W \quad (7)$$

where L is the equivalent compliance operator of both the bearing and pin surfaces to account for EHD effect, F is the secondary motion operator that corresponds rigid body bearing to pin relative motions, J is the load operator, M is the mass matrix, and ε is the 4DOF bearing secondary motion vector, defined as,

$$J = F = \begin{Bmatrix} \cos \theta \\ \sin \theta \\ \cos \theta \\ \sin \theta \end{Bmatrix} \quad M = \begin{bmatrix} m_1 & 0 & 0 & 0 \\ 0 & m_1 & 0 & 0 \\ 0 & 0 & m_2 & 0 \\ 0 & 0 & 0 & m_2 \end{bmatrix} \quad W = \begin{Bmatrix} f_x \\ f_y \\ -\tilde{f}_x \\ -\tilde{f}_y \end{Bmatrix} \quad (8)$$