MODELING OF PISTON RING PACK DYNAMICS

PAVEL NOVOTNÝ, VÁCLAV PÍŠTĚK, LUBOMÍR DRÁPAL

Institute of Automotive Engineering, Brno University of Technology, Technická 2, 616 69 Brno, Czech Republic,

Tel.: +420 541 142 272, Fax: +420 541 143 354, E-mail: novotny.pa@fme.vutbr.cz

SHRNUTÍ

Pístní kroužky hrají důležitou roli při mazání spalovacích motorů se značnými důsledky na opotřebení motoru, třecí ztráty a spotřebu oleje. Výpočtové řešení dynamiky pístních kroužků představuje značný problém a vyžaduje využití numerických přístupů s množstvím vhodných vstupů. V případě tohoto článku jsou použity numerické postupy pro řešení dynamiky pístních kroužků za podmínek smíšeného mazání se vstupy z virtuálního motoru a vhodných experimentů. Výpočtový algoritmus je rovněž zpracován do uživatelského prostředí a je k dispozici pro průmyslové využití. Výsledky řešení jsou prezentovány na vznětovém přeplňovaném motoru v konfiguraci s třemi pístními kroužky.

KLÍČOVÁ SLOVA: DYNAMIKA, PÍSTNÍ KROUŽEK, SMÍŠENÉ MAZÁNÍ, TŘECÍ ZTRÁTY

ABSTRACT

Piston rings play an important role in the lubricant characteristic of reciprocating engines with the consequences on engine wear, power losses and a vast amount of lubricating oil consumption. The computational solution of piston ring dynamics is a very complex problem and it requires multiple numerical approaches supplemented by suitable inputs. The paper demonstrates the principles of the numerical solution of piston ring dynamics in mixed lubrication conditions incorporating a virtual engine and experimental inputs. The simulation algorithm is incorporated into user guided interface and it is available for industry usages. The solution results are presented on a diesel turbocharged engine with three piston ring configuration.

KEYWORDS: DYNAMICS, PISTON RING, MIXED LUBRICATION, FRICTION LOSSES

1. INTRODUCTION

A common feature of all dynamic seals for linear motion is that they operate against a reciprocating counter surface, for which reason they have to be optimized in terms of sealing ability, low friction and system wear. Piston rings of reciprocating engines have several functions, mainly

- to seal the clearance between the piston and cylinder in order to retain gas pressure and to minimize blow-by,
- to ensure adequate lubricant applied on the cylinder surface to sustain high thrust and gas force loads at high speed while maintaining oil consumption at acceptable levels and
- to keep piston temperatures relatively low while transferring heat to cylinder walls and coolant.

The piston rings for current internal combustion engines have to meet all the requirements of dynamic seal for linear motion that operates under demanding thermal and chemical conditions. In short, the following requirements for piston rings can be identified [6]:

- low friction, for supporting a high power efficiency rate,
- low wear of the ring, for ensuring a long operational lifetime,
- low wear of the cylinder liner, for retaining the desired surface texture of the liner,
- emission suppression, by limiting the flow of engine oil to the combustion chamber,
- good sealing capability and low blow-by for supporting the power efficiency rate
- good resistance against thermo-mechanical fatigue, chemical attacks and hot erosion and
- reliable operational and cost effectiveness for a significant long time.

Piston rings form a ring pack, which usually consist of 2–5 rings, including at least one compression ring. The number of rings in the ring pack depends on the engine type, but usually comprises 2–4 compression rings and 0–3 oil control rings.

Reciprocating engines of passenger cars normally use three rings, two compression rings and one oil control ring. This work describes a simulation method of piston ring pack dynamics with stress on low friction losses. The algorithm includes the piston ring motions and blow-by/blowback flow models. The results are presented on Diesel engine.

2. PISTON RING FORCES

The motion of a piston ring is caused by forces and moments including:

- the static ring tension and torsion from the installation of a piston ring in the cylinder liner,
- the inertia forces and moment related to component mass and engine speed and
- surface forces, that are mainly gas pressure forces.

Figure 1 presents piston ring forces considered for a solution of piston ring dynamics.

FIGURE 1: Description of forces acting on the ring.
OBRÁZEK 1: Popis sil působících na pístní kroužek.

2.1 HYDRODYNAMIC LUBRICATION MODEL

When a gap is fully lubricated the hydrodynamic lubrication can be considered. The given problem is described by Reynolds equation, which indicates the relationship between the pressure and the film shape as a function of viscosity and velocity. The Reynolds equation can be simplified by assumptions of constant viscosity. For the analysis of lubrication in a piston ring pack, the oil film thickness between the piston ring and the cylinder wall is much smaller than the piston ring radius. It can be assumed as an infinite flat bearing. The average Reynolds equation can be simplified to an ordinary differential equation [3]

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3}{12\eta} \frac{\partial \bar{p}}{\partial x} \right) = \frac{U}{2} \frac{\partial(\rho \bar{h}_r)}{\partial x} + \frac{\partial(\rho \bar{h}_r)}{\partial t} \quad (1)$$

The h symbol presents oil film thickness, x is coordinate, t is time, $\bar{p}$ is circumferential averaged pressure, r is oil density, h is dynamic oil viscosity and U is relative velocity.

The Reynolds equation is solved numerically using Finite Difference Method (FDM). Figure 2 presents computational grid for numerical solution of 2D Reynolds equation. FDM and numerical solution details are widely presented in literature and they can be also found in [1, 2, 3].

Viscous friction forces exerting on bearing area are calculated as

FIGURE 2: Computational grid for a numerical solution of 2D Reynolds equation.
OBRÁZEK 2: Výpočtová síť pro numerické řešení 2D varianty Reynoldsovy rovnice.

$$F_{th} = \int_{x_1}^{x_2} \left(-\frac{\mu U}{h} + \frac{h}{2} \frac{\partial \bar{p}}{\partial x} \right) \cdot \pi d \cdot dx \quad (2)$$

Equation (2) is solved numerically by trapezoidal method.

Oil film thickness changes during the engine cycle. This change can be derived from the piston ring motions, oil film evaporation, etc. It affects the piston ring lubricant conditions, especially the inlet condition. Figure 3 illustrates the film thickness on the engine liner wall. For the first and the second rings, the film inlet position is the first contact point between the oil film surface and the ring face profile. For the film exit position, the oil film thickness was assumed equal to the minimum clearance between the ring surface and the engine liner wall. The oil ring is supposed to be always soaked in oil. Thus, the fully flooded condition is applied to this ring.

FIGURE 3: Oil film thickness interferences for all piston rings.
OBRÁZEK 3: Ovlivnění tloušťky oleje pístními kroužky.

2.2 BOUNDARY LUBRICATION MODEL

The pure boundary lubrication according to Greenwood and Tripp [4] is used when oil supply is insufficient. The nominal pressure can be calculated as

$$P_c = KE' F(h/\sigma) \quad (3)$$

where

$$K = \left(8\pi \frac{\sqrt{2}}{15}\right) (N\beta\sigma)^2 \left(\sqrt{\frac{\sigma}{\beta}}\right) \quad (4)$$

and

$$E' = \frac{2E_1E_2}{E_1(1-\nu_1^2) + E_2(1-\nu_2^2)} \quad (5)$$

E_1 and E_2 denotes Young's modulus of the liner and piston rings respectively, ν_1 and ν_2 are poisson numbers, s is composite RMS roughness, b is radius at asperity summit and N is number of asperities per unit area.

The boundary lubrication forces are included only if

$$h/\sigma < w \quad (6)$$

where w parameter is usually equal to 3.

Figure 4 demonstrates contact principles of two rough surfaces using assumptions of Greenwood and Tripp.

FIGURE 4: Contact of two rough surfaces.

OBRÁZEK 4: Kontakt dvou drsných povrchů.

2.3 GAS BLOW-BY/BLOW-BACK FLOW MODEL

Considering gas flows thru piston rings a piston–cylinder system is divided into several groups. The gas flow through piston ring gap is assumed to be the isentropic orifice flow. The flow between two volumes can be solved as

$$\dot{m} = A_E \rho_1 \sqrt{\kappa R T_1} \quad \text{for} \quad \left(\frac{p_1}{p_0}\right) \leq \left(\frac{2}{\kappa + 1}\right)^{\frac{\kappa}{\kappa - 1}} \quad (7)$$

or

$$\dot{m} = A_E \sqrt{\rho_0 p_0} \sqrt{\frac{2\kappa}{\kappa - 1} \left(\left(\frac{p_1}{p_0}\right)^{\frac{2}{\kappa}} - \left(\frac{p_1}{p_0}\right)^{\frac{\kappa + 1}{\kappa}} \right)} \quad \text{for} \quad \left(\frac{p_1}{p_0}\right) > \left(\frac{2}{\kappa + 1}\right)^{\frac{\kappa}{\kappa - 1}} \quad (8)$$

The symbol $\dot{m}$ in equations (7) and (8) is mass flow, A_E is efficient area R is gas constant, T is temperature, p_0 and p_1 are pressures in first volume and second volume respectively, κ is polytrophic exponent. Figure 5 presents the piston assembly gas flow model scheme.

FIGURE 5: Blow-by/Blow-back piston assembly flow model scheme.

OBRÁZEK 5: Schéma modelu profuku pístní skupiny.

2.4 PISTON RING PRETENSION

Ring pretension force influences ring dynamics with high impact on engine wear, oil consumption or power losses. Friction forces and therefore mechanical losses are highly dependent on right design of piston rings. The pretension forces can be calculated analytically or using Finite Element Method (FEM). Figure 6 presents FE models of piston rings of target engine.

FIGURE 6: FE models of piston rings.

OBRÁZEK 6: MKP modely pístních kroužků.

2.5 SOLUTION OF RING FORCE EQUILIBRIUM

All forces acting on a piston ring have to be in equilibrium. Newton-Raphson method is used for a non-linear solution together with predictor-corrector strategies. During the each time step a Jacobian matrix is evaluated. The Jacobian matrix is calculated numerically during every iteration. The force residues are iteratively minimized until defined error is reached. Newton-Raphson and predictor-corrector method descriptions can be found for example in [5].

3. SIMULATION ALGORITHM APPLICATION

Numerical solution results are presented on Diesel turbocharged four-cylinder engine. The target engine bore is 105 mm and stroke is 120 mm. The ring 1 of the target engine is a compression ring, the ring 2 is a scrapper ring and last ring is an oil control ring as they are presented on the target engine.

There are many results provided by solution algorithm. One of the most important piston ring results are ring friction losses. In general piston ring friction losses represent high portion of total engine mechanical losses. Table 1 shows piston ring power losses for an engine speed 2200 rpm.

	Ring 1 [W]	Ring 2 [W]	Oil ring [W]
Power loss	125.0	146.0	284.9

TABLE 1: Power losses of the target engine piston rings

TABULKA 1: Ztrátový výkon pístních kroužků řešeného motoru

The friction forces are highly influenced by gas flow pressure acting on ring face. Mixed lubrication conditions are especially on first piston ring near to Top Dead Center (TDC).

Calculated combustion pressures acting on ring face profiles are presented in Figure 7. The pressure values are highly influences by ring clearances and also liner deformations. Pressure acting on first ring face is very similar to combustion pressure in a combustion chamber.

Figure 8 presents calculated minimal oil film thickness of piston rings. Mainly first ring is subjected to boundary lubrication near to TDC where maximum combustion pressures are reached. TDC is positioned to 360° of crank angle in Figure 8. Oil ring is presumed to be always fully flooded.

FIGURE 7: Calculated ring face pressures.

OBRÁZEK 7: Vypočtené tlaky na povrchu pístních kroužků.

FIGURE 8: Calculated minimum oil film thickness below rings.

OBRÁZEK 8: Vypočtené minimální tloušťky olejové vrstvy pod kroužky.

FIGURE 9: Calculated friction forces.

OBRÁZEK 9: Vypočtené třecí síly.

Figure 9 shows friction forces incorporating hydrodynamic and boundary lubrication components. Mainly the first piston ring is subjected to high friction forces due to lack of lubricant together with high combustion pressure forces.

FIGURE 10: Piston ring power losses as a function of pretension tangential force.

OBRÁZEK 10: Ztrátový výkon pístních kroužků v závislosti na tangenciální předepínací síle.

4. CONCLUSION

The piston ring computational modelling can be applied for different combustion engines and it enables to compare different piston ring parameters before complicated measurements on prototype engines. The calculated values such as friction forces or minimal oil film thicknesses are important, but sometimes much more important are tendencies in some quantities. Figure 10 presents how tangential pretension forces designed by manufacturers influence piston ring power losses.

The new simulation algorithm of piston ring dynamics has been developed and incorporated into RingDyn user guided interface. The algorithm is very fast and it enables to calculate results during few second on modern PC. The RingDyn software package is ease to use and it can be used in the first stages of the internal combustion engine development.

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DESCRIPTION OF THE INGAS PROJECT

Main objective of "Integrated Gas Powertrain (InGas)" Collaborative Project is to deploy a custom designed engine integrated with specific aftertreatment systems applied to a light duty vehicle able to achieve a 10 % higher fuel conversion efficiency than that of a corresponding 2006 diesel vehicle and complying with an emission level lower than Euro 6. Additional features are advanced storage systems and vehicle architectures, as well as multi-grade fuel tolerance and fuel flexibility. To achieve the InGas targets, three main combustion technologies will be compared:

- Sub-project A1 "CNG technologies for passenger cars" will develop a natural gas car powered by a 1.4 liter displacement engine using the sequential multi-point port gas injection and following the stoichiometric approach.
- Sub-Project A2 "Turbo DI CNG engine" will develop a natural gas car powered by a 1.8 liter displacement engine using the direct gas in-cylinder injection and following a lean burn approach.
- Sub-Project A3 "Boosted lean burn gas engine" will develop a natural gas light-duty vehicle powered by a 1.9 liter displacement engine using port gas injection or low pressure direct gas injection and following ultra-lean combustions approach.

Three main enabling technologies will be compared and assessed.

- Sub-project B0 "Fuels for advanced CNG engines" will define / supply the gas mixture of the requested quality, conduct analysis and propose solutions in order to affect in a flexible way storage, combustion, aftertreatment and performance of the CNG vehicles.
- Sub-project B1 "Gas storage for passenger car CNG engine" will develop advanced gas storage and filling systems including specific components and gas sensors.
- Sub-project B2 "Aftertreatment for passenger car CNG engine" will develop an aftertreatment system for natural gas vehicles having special regards to CH_4 conversion efficiency and NO_x abatement under stoichiometric and lean combustion operations.

The project is partly subsidized from 7th Framework EU Programme. The coordinator of project is Massimo Ferrera from Centro Ricerche Fiat.

THE FOLLOWING ARTICLES DESCRIBE THE CONTRIBUTION OF PROJECT PARTNER – CZECH TECHNICAL UNIVERSITY IN PRAGUE; JOSEF BOŽEK RESEARCH CENTER TO INGAS PROJECT, IN PARTICULAR TO THE SUB-PROJECT B0 "FUELS FOR ADVANCED CNG ENGINES". >>>