

Development and Application of Ring-Pack Model Integrating Global and Local Processes. Part 1: Gas Pressure and Dynamic Behavior of Piston Ring Pack

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ABSTRACT

A new ring pack model has been developed based on the curved beam finite element method. This paper describes the first part of this model: simulating gas pressure in different regions above piston skirt and ring dynamic behavior of two compression rings and a twin-land oil control ring. The model allows separate grid divisions to resolve ring structure dynamics, local force/pressure generation, and gas pressure distribution. Doing so enables the model to capture both global and local processes at their proper length scales. The effects of bore distortion, piston secondary motion, and groove distortion are considered. Gas flows, gas pressure distribution in the ring pack, and ring structural dynamics are coupled with ring-groove and ring-liner interactions, and an implicit scheme is employed to ensure numerical stability. The model is applied to a passenger car engine to demonstrate its ability to predict global and local effects on ring dynamics and oil transport.

CITATION: Liu, Y. and Tian, T., "Development and Application of Ring-Pack Model Integrating Global and Local Processes. Part 1: Gas Pressure and Dynamic Behavior of Piston Ring Pack," *SAE Int. J. Engines* 10(4):2017, doi:10.4271/2017-01-1043.

INTRODUCTION

The piston ring pack system is the most critical components for sealing high-pressure gas in the combustion chamber and controlling oil consumption, which are the two major constraints to the development of internal combustion engines. While gas flows and blow-by can be mostly resolved without considering variation along the circumference (commonly called 3D effect), the oil leakage through rings may mainly occur locally [1] [2] [3].

Generally speaking, oil can flow from the crankcase to the combustion chamber through three major paths: the ring-liner interfaces, the ring-groove interfaces, and the ring gaps. There are three main elements to be resolved to predict oil leakage through different interfaces in the ring pack. First, clearances of the ring-liner interfaces, ring-groove interfaces, and the ring gaps need to be calculated, which are dependent on ring structure dynamics and the distorted geometry of the piston and the cylinder bore. Second, the gas pressure distribution in different regions is important to oil flows because it not only deforms the piston rings to adapt to the geometry of the piston and the cylinder bore but also redistributes the oil on the piston lands and inside the piston grooves. This redistribution leads to local oil accumulation and a variance of oil supply to the ring-groove interfaces and the ring-liner interfaces [4] [5]. Finally, proper oil flow models are needed to compute the oil flow rates through the ring gaps and the ring-groove interfaces, and the oil film thickness on the liner.

Oil transport and ring structure dynamics are the results of combined global and local processes with different length scales. The cylinder bore and the piston are distorted due to thermal and mechanical stresses. At structural level, the piston rings try to adapt to the distorted geometry with the help of gas pressure, inertial force and ring tension. Piston tilt or, more precisely, tilted grooves force the rings to twist to align with the angle of the groove flanks. All these above mentioned large-scale processes dictate how the force and structural deformation are distributed along the piston rings. Then, the exact local clearances are determined by the factors responsible for local pressure generation such as ring-groove pumping, bridging, ring profile and surface roughness.

Several numerical models focused on ring dynamics and lubrication have been developed. Tian developed a 2D model based on an axisymmetric assumption to estimate ring dynamics and gas flow [6] [7]. Most of the published 2D models combine the calculation of ring dynamics and gas flow and are found to be useful tools in understanding ring dynamic behavior [6] [7] [8] [9] [10] [11]. To address non-axisymmetric characteristics of the piston-ring system, Ejakov et al. simulated the dynamic twist angle of the piston ring pack using finite element analysis [12]. Hu et al. and Ma et al. put extensive effort on a 3D lubrication model coupling a detailed elastic analysis and a complementarity method to evaluate the conformability between the ring and the liner [13] [14] [15]. Lin et al. built a complete 3D model with the combination of ring dynamics

gas flow and lubrication, utilizing the straight beam finite element method [16] [17]. Mittler et al. brought another 3D model studying ring dynamics and gas flow [18] [19]. Cheng et al. studied the second ring flutter and its collapse in engine cycles and ring-liner, ring-groove contact based on his modeling work [20] [21]. Tomanik improved the criterion for ring conformability under realistic bore distortion based on Liu's 3D model [22] [23]. Baelden et al. first introduced the curved beam finite element method to model oil transport around a twin-land oil control ring (TLOCR), which combined both OCR dynamics and lubrication [24] [25]. Picard applied the same strategy in rotary engines [26] [27] [28] [29]. The curved beam finite element method has many advantages over the straight beam finite element method. The most important one is that it couples ring deformation to contact interaction which are on different length scales by separating the structural mesh and contact grid using element shape functions.

All these existing models succeed in capturing certain local and average phenomena. However, none is able to integrate all the individual processes critical to the ring pack behavior with a proper handling of their own length scales. In this work, a new model framework has been developed to integrate multi-physics at their own scales. The model starts with dividing a ring into a number of curved beam elements, on which local pressures and forces are further distributed with appropriate sub-models. The overall model is divided into two modules: 1) ring dynamics and gas flows; 2) ring-liner lubrication. This paper presents the first module focusing on the ring dynamics, gas pressure distribution, and gas flows with consideration of piston and bore distortion as well as the piston secondary motion (as shown in Figure 1). The oil flow through the ring-groove interfaces with localized oil supply boundary condition is considered, although the current oil flow model is simplified. This module also provides the environmental conditions for the second module.

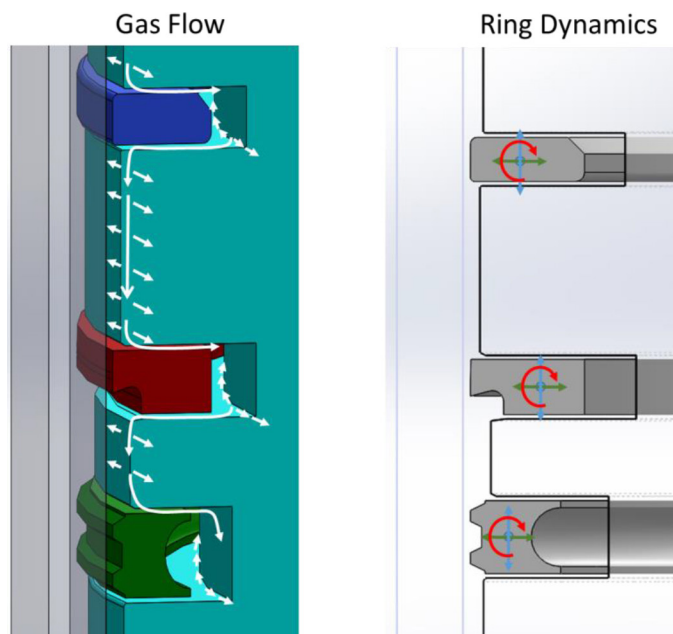


Figure 1. Overview of Gas Flow and Dynamics of Piston-Ring Assembly

In engine cycles, a variety of forces act on piston rings and they are included in the model. (a) The initial force is caused by the top two rings' strain energy change or the OCR spring force when they are closed and assembled into the piston. (b) Gas pressure changes dramatically inside the ring pack system and it applies pressure force on the piston rings. (c) The piston experiences acceleration and deceleration. As a result, when the piston is chosen as a reference frame, the inertial force needs to be applied. (d) The ring-liner interaction provides normal force and friction force on the piston rings. Since the purpose of this module is to calculate the ring dynamic behavior instead of lubrication, a simplified dry-contact model is used to calculate contact force between the top two rings and the liner which is accurate enough for dynamics calculation. (e) The ring-groove interaction provides axial forces on the piston rings, which includes asperity contact, gas pressure from the ring-groove gas flow, hydrostatic pressure and oil pressure due to squeezing. In the ring-groove interfaces, the oil film thickness is set to be constant to simplify the calculation.

Gas pressure plays an important role in the ring pack dynamic behavior but its value at the same time is dependent on the ring deformation. Thus, the ring dynamics and gas pressure calculations need to be coupled in order to obtain reasonable results. In this model, the gas flows are considered in the axial direction, the radial direction, and the circumferential direction, through the ring-groove interfaces, the ring gaps, the drain-holes of the OCR groove and the ring-liner interfaces when the rings lose contact with the liner.

In the following, the model formulation is described first. Then the results for a SI engine and the effects of groove distortion are presented.

MODEL DEVELOPMENT

Curved Beam Finite Element Model of Piston Rings

The curved beam finite element method for piston rings was developed by Baelden [24] [25], which discretizes the ring neutral axis into a number of elements and uses nodal displacements and their derivatives to interpolate its geometry. The deformation of the ring to be interpolated can be broken down into three components: the radial displacement y , the axial displacement, z , and the orientation of the cross-section α_r , as shown in Figure 2. In the reference position, the ring is centered axially in the piston groove and its radial position is such that it is in contact with the non-deformed cylinder.

The Hermite polynomial shape functions are used to interpolate all the three components. The radial and axial displacement of the ring neutral axis are interpolated using 5th order Hermite polynomial spline. This is to guarantee the continuity of the curvature of the ring which is the key variable for bending. 5th order interpolation requires nodal displacements to include the displacement and the corresponding first and second derivatives (Figure 3).

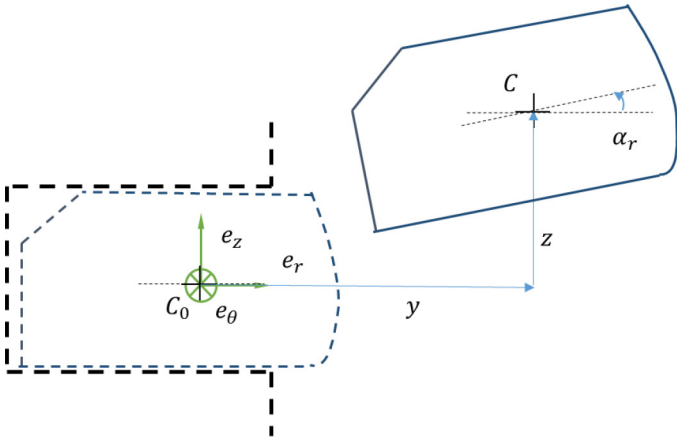


Figure 2. Ring Displacement in Three Directions

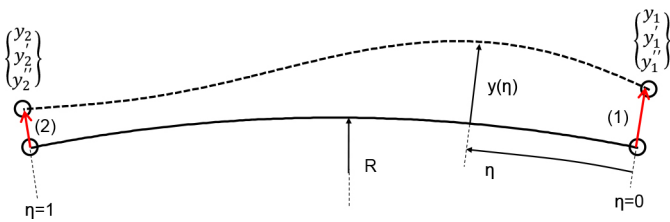


Figure 3. Spline Interpolation of Ring Radial Displacement [24]

$$y(\eta) = \sum_{k=1}^6 N_k(\eta) u_{yk} \text{ where } \{u_y\} = \{u_{y1}, \dots, u_{y6}\}^T = \{y_1, y_1', y_1'', y_2, y_2', y_2''\}^T \quad (1)$$

$$z(\eta) = \sum_{k=1}^6 N_k(\eta) u_{zk} \text{ where } \{u_z\} = \{u_{z1}, \dots, u_{z6}\}^T = \{z_1, z_1', z_1'', z_2, z_2', z_2''\}^T \quad (2)$$

The prime sign here refers to a derivation with respect to the polar angle θ and this notation will be used for the rest of this paper. There are six shape functions: $N_1, N_2, N_3, N_4, N_5, N_6$.

$$N_1 = 1 - 10\eta^3 + 15\eta^4 - 6\eta^5$$

$$N_2 = \theta_e(\eta - 6\eta^3 + 8\eta^4 - 3\eta^5)$$

$$N_3 = \theta_e^2 \left(\frac{\eta^2}{2} - 3\frac{\eta^3}{2} + 3\frac{\eta^4}{2} - \frac{\eta^5}{2} \right)$$

$$N_4 = 10\eta^3 - 15\eta^4 + 6\eta^5$$

$$N_5 = \theta_e(-4\eta^3 + 7\eta^4 - 3\eta^5)$$

$$N_6 = \theta_e^2 \left(\frac{\eta^3}{2} - \eta^4 + \frac{\eta^5}{2} \right)$$

(3)

The twist angle is interpolated using 3rd order Hermite polynomials. The twisting moment is proportional to the first order derivative of the twist angle. Thus, 3rd order polynomial interpolation is sufficient and requires ring twist and its first order derivative at two nodes.

$$\alpha_r(\eta) = \sum_{k=1}^4 N_{ak}(\eta) u_{ak} \text{ where } \{u_a\} = \{u_{a1}, \dots, u_{a4}\}^T = \{\alpha_{r1}, \alpha'_{r1}, \alpha_{r2}, \alpha'_{r2}\}^T \quad (4)$$

The curved beam finite element method uses two 5th order and one 3rd order Hermite polynomials to discretize the ring deformation which leads to 8 degrees of freedoms per node: 3 for radial displacements, 3 for axial displacements and 2 for twists. The ring element has two nodes and therefore 16 nodal displacements:

$$\{u\}^{(e)} = \{u_1, u_2, \dots, u_{16}\}^T = \{y_1, y_1', y_1'', z_1, z_1', z_1'', \alpha_1, \alpha'_1, y_2, y_2', y_2'', z_2, z_2', z_2'', \alpha_2, \alpha'_2\}^T \quad (5)$$

Ring Gap Closing with Large Displacement

When the top two rings are in their free shape, the radial displacement of their neutral axes with respect to the nominal radius is on the order of several millimeters and cannot be assumed small. Therefore, the curvature needs to be calculated by the general expression for a planar curve.

$$\kappa_{fs} = \frac{(r_{fs}^2 + 2r_{fs}'^2 - r_{fs}r_{fs}'')^2}{(r_{fs}^2 + r_{fs}'^2)^{3/2}} \quad (6)$$

κ_{fs} is the curvature of the free shape and r_{fs} is its radius. The calculation of the curvature change will become very complicated and non-linear if directly applying the above equation. In order to simplify the calculation, an intermediate state is added that with a preload the ring is closed at its nominal radius without any twist. Once the ring is closed, its neutral axis displacements with respect to its nominal radius can be simplified using the small displacement assumption. The required preload (initial force) holding the ring at this intermediate state from its free shape is obtained by applying Euler-Lagrange Equations.

$$F_{initial,ky}^{(e)} = L_e EI_{zz} \int_0^1 \left(\frac{1}{R} - \kappa_{fs} \right) \left[-\frac{\cos^2 \alpha_p}{R^2} \right] (N_{m(ky)} + N''_{m(ky)}) d\eta + L_e EI_{yy} \int_0^1 \left(\frac{1}{R} - \kappa_{fs} \right) \left[-\frac{\sin^2 \alpha_p}{R^2} \right] (N_{m(ky)} + N''_{m(ky)}) d\eta \quad (7)$$

$$F_{initial,kz}^{(e)} = L_e EI_{zz} \int_0^1 \left(\frac{1}{R} - \kappa_{fs} \right) \left(-\frac{1}{R} \right) \cos \alpha_p \sin \alpha_p \frac{N''_{m(kz)}}{R} d\eta + L_e EI_{yy} \int_0^1 \left(\frac{1}{R} - \kappa_{fs} \right) \frac{1}{R} \sin \alpha_p \cos \alpha_p \frac{N''_{m(kz)}}{R} d\eta \quad (8)$$

$$F_{initial,k_\alpha}^{(e)} = L_e E I_{zz} \int_0^1 \left(\frac{1}{R} - \kappa_{fs} \right) \left(-\frac{1}{R} \right) \cos \alpha_p \sin \alpha_p \frac{N'_{am(k_\alpha)}}{R} d\eta$$

$$+ L_e E I_{yy} \int_0^1 \left(\frac{1}{R} - \kappa_{fs} \right) \frac{1}{R} \sin \alpha_p \cos \alpha_p \frac{N'_{am(k_\alpha)}}{R} d\eta$$
(9)

R is the ring nominal radius. L_e is the length of the ring element. E is the Young's modulus of the ring. I_{zz} and I_{yy} are the two moments of inertial associated with the principal axes of the ring cross-section and α_p is the principal angle (as shown in Figure 4). k_y , k_z , and k_α are the positions of the radial displacement, axial displacement and the orientation of the cross-section in the displacement vector of the ring element (5). m is the mapping function to select the shape function corresponding to a ring element k^{th} nodal displacement.

$$k_y = [1, 2, 3, 9, 10, 11]$$
(10)

$$k_z = [4, 5, 6, 12, 13, 14]$$
(11)

$$k_\alpha = [7, 8, 15, 16]$$
(12)

$$k \rightarrow m(k):$$

$$[1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16]$$

$$\rightarrow [1, 2, 3, 1, 2, 3, 1, 2, 4, 5, 6, 4, 5, 6, 3, 4]$$
(13)

Principal frame

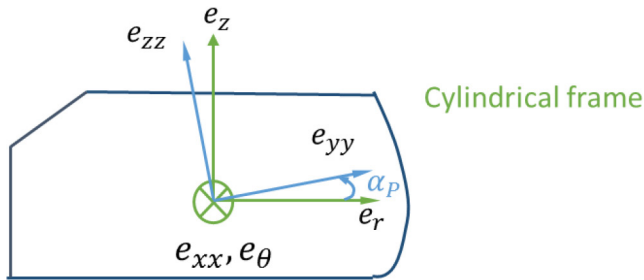


Figure 4. Principal Axes of the Ring Cross Section

Force Analysis in the Ring Dynamics Model

In engine cycles, the piston rings have different forces applied. The first is the initial force. When assembled into the piston, the top two rings are closed from their free shape and the change of curvature and torsion of their neutral axes causes internal stress. The OCR is assembled together with a spring that provides the expansion force pushing it towards the liner. Pressured gas flows around the ring pack system, which contributes to pressure force in both radial and axial directions. The ring-groove interaction is composed of dry contact force, oil pressure force due to squeezing and gas pressure force if there exists gas flow through the ring-groove interfaces. The liner provides contact force in the radial direction and friction force in the axial direction. The reference frame is fixed to the piston. Thus, the inertial force also needs to be included due to the piston acceleration and deceleration relative to the engine block. The following figure

demonstrates all the forces acting on the ring. The resultant force can be split into 3 components: the radial force f_r , the axial force f_z and the twisting moment m_θ . These three components will be used later to calculate the external load vector in the governing equations.

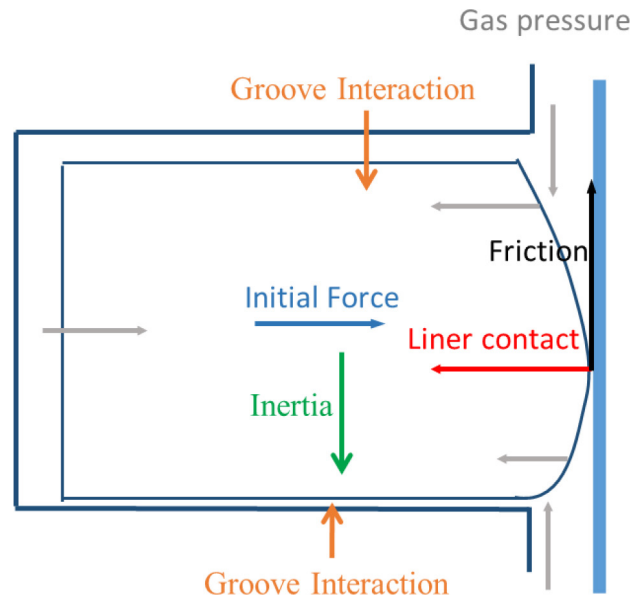


Figure 5. Force Analysis of Piston Ring

Ring-Groove Interaction

Ring-groove interaction is important in the ring dynamics model. The acting force comes from the asperity contact force, oil pressure force due to squeezing, hydrostatic force and gas pressure force. The ring-groove dry contact force can be calculated by the fitting formula in [13] based on the Greenwood & Tripp's theory [30]. When the ring is away from the groove flank and oil film, there is a pressure driven gas flow through the ring-groove clearance. The corresponding pressure distribution can be calculated based on the assumption of fully developed, quasi-steady and locally parallel flow [6]. When the ring is in contact with the oil film on the groove flanks, the gas flow is blocked and the oil pressure is generated. By neglecting the possibility of 'suction' or occurrence of 'negative pressure', the squeezing pressure can be calculated by Reynolds equation [6].

Ring-Liner Interaction

The forces acting on the ring from liner include the dry-contact force, the hydrodynamic force from the oil film between the ring and the liner and the gas pressure force. The radial force applied by the gas pressure on the ring face is calculated in a simplified manner analogous to that of ref. [7]: the ring running face is divided into two regions by the point that has the minimum ring-liner clearance; above the minimum point, the pressure is assumed to share the same magnitude with the pressure at the land region above the ring; underneath the minimum point, the pressure of the land region below the ring is assumed. The dry contact force is calculated using the simplified Greenwoods model [13] [30]. The dry contact friction force is calculated by the multiplication of the dry contact force by a constant friction coefficient. In the ring dynamics model, the hydrodynamic force is neglected for the top two rings in order to achieve better calculation speed. Neglecting the hydrodynamic force