

Development and Application of Ring-Pack Model Integrating Global and Local Processes. Part 2: Ring-Liner Lubrication

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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 second part of this model: simulating oil transport around the ring pack system (two compression rings and one twin-land oil control ring (TLOC)) through the ring-liner interfaces by solving the oil film thickness on the liner. The ring dynamics model in Part 1 calculates the inter-ring gas pressure and the ring dynamic twist which are used in the ring-liner lubrication model as boundary conditions. Therefore, only in-plane conformability is calculated to obtain the oil film thickness on the liner. Both global process, namely, the structural response of the rings to bore distortion and piston tilt, and local processes, namely, bridging and oil-lube interaction, are considered. The model was applied to a passenger car engine. It was found that a net oil scraping by the top ring, as a potential oil consumption source, only occurs with a combination of high bore distortion and local oil bridging. The effect of fuel deposition on the oil film was also investigated.

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INTRODUCTION

One of the main limiting factors for reducing ring tensions to minimize friction loss is the bore distortion. Recent trend for increasing energy density brings greater thermal stress while demanding for reduction of the dimensions of engine blocks. Both present greater challenge to control the bore distortion and its temperature. From the piston ring side, one can see gradual reduction of the ring dimensions to reduce the stiffness of the rings so that the same conformability can be achieved with less ring tension. Yet, making smaller rings brings bigger challenges to manufacturing cost, consistency, and assembly. Therefore, it is of great practical interest to gain a better understanding on how the net oil transport to the combustion chamber can be generated from the interaction of the ring pack with a distorted bore.

While one can safely assume the oil supply to the oil control ring during a down-stroke is sufficient, the oil/fuel supply to the top two rings can come from different sources. In most part of the areas the top two rings travel, the oil film thickness on the liner is controlled by the oil control ring. However, as discovered by Przesmitzki et al. [1] [2] [3] [4], the oil can flow from the piston lands back to the liner in the local areas with sufficiently large oil accumulation. As defined in [2], this process is called bridging. Depending on the timing and location of the bridging, the bridged oil may first interact with the top two rings. As the top two rings in general are less conformable to a distorted bore than the oil control ring when high gas pressures are

not present, more oil may pass the top two rings and may be scraped later when the top two rings come back with an assist of raised cylinder pressure [5]. Thus the local oil flow from the piston regions above the oil control ring may become a source of the oil consumption even though the oil control ring itself has a proper oil control traveling down-strokes.

Another local deviation of the oil content from the oil film left by the oil control ring is the addition of the fuel, which is one of the sources for fuel dilution and local excessive wear on the liner.

A lot of research work has been done to study ring-liner lubrication and ring-liner conformability. Richardson and Borman did both theoretical and experimental study of the oil film for application to the piston ring lubrication [6]. Tian et al. built a 2D model to study ring-liner lubrication based on the Reynolds equations [5] [7]. In complement to Tian's 2D model, Li and Chen developed a deterministic method to calculate the hydrodynamic behavior between the ring and the liner [8] [9]. Liu et al. built a complete 3D model with the combination of ring dynamics, gas flow and lubrication, utilizing the straight beam finite element method [10] [11]. Mittler et al. brought another 3D model studying ring dynamics and gas flow [12]. Cheng et al. studied the ring-bore contact and ring-groove contact based on his 3D model [13] [14]. Lyubarskyy and Bartel built a 2D CFD-model to study ring dynamics, oil transport and friction [15]. Baelen 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 [16] [17]. Picard applied the same strategy in rotary engines [18] [19] [20] [21]. 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.

The objective of this model is to integrate all levels of interaction between the ring pack and the liner with considerations of bore distortion, piston tilt due to secondary motion, land pressures, ring local dynamic twist, and local additional oil/fuel supply. A similar approach to [22] was adapted to only resolving the in-plane interaction between the top two rings and the cylinder bore with gas pressures and ring local dynamic twist calculated from the model presented in Part I. Doing so preserves respective physics for these two interactions and allows efficient computation, for instance, for the sensitivities of local oil/fuel supplies.

A curved beam finite element method was utilized in this model to integrate both global and local processes at their proper length scales [16]. This method separates ring structural mesh from contact grid so that local oil transport through the ring-liner interfaces can be studied at the same time preserving the accuracy of structural response. Unlike the ring dynamics model in Part I, a precise estimation of the ring-liner clearance is required to predict the oil film thickness on the liner. Thus, both hydrodynamic pressure and contact pressure are included in the ring-liner interaction. This model was applied to a passenger car engine and the effects of bore distortion combined with bridging, and lube-fuel dilution were studied.

MODEL DEVELOPMENT

Stiffness, Mass and Load Finite Element Matrices for Ring-Liner Lubrication Model

Since the ring-liner lubrication model only solves in-plane conformability, only terms related to the radial deformation are left in the mass $[M]^{(e)}$, stiffness $[K]^{(e)}$ and load $\{F_{ext}\}^{(e)}$ matrices.

$$M_{ij}^{(e)} = L_e \int_0^1 \rho A N_i N_j d\eta \quad \text{for } i, j \in \{1, 2, 3, 4, 5, 6\} \quad (1)$$

$$K_{ij}^{(e)} = \frac{L_e}{R^4} \int_0^1 E I_{zz} (N_i + N_i'') (N_j + N_j'') d\eta \quad \text{for } i, j \in \{1, 2, 3, 4, 5, 6\} \quad (2)$$

$$F_{ext,i}^{(e)} = L_e \int_0^1 f_r N_i d\eta \quad \text{for } i \in \{1, 2, 3, 4, 5, 6\} \quad (3)$$

ρ is the density of the ring. A is the area of the cross-section. E is the Young's modulus of the ring. f_r is the resultant force in the radial direction and N_i ($i \in \{1, 2, 3, 4, 5, 6\}$) are the shape functions [16]. The ring neutral axis displacement vector also reduces to the terms in the radial direction only:

$$\{u\}^{(e)} = \{y_1, y_1', y_1'', y_2, y_2', y_2''\} \quad (4)$$

The mass, stiffness and load matrices of each element are assembled to obtain the global ones $[M]$, $[K]$ and $\{F_{ext}\}$. The global displacement vector $\{u\}$ is the assembly of the nodal displacement vectors. The detail is in Baelden's thesis [17]. Then the governing equations become:

$$[M]\{\ddot{u}\} + [K]\{u\} = \{F_{ext}\} - \{F_{initial}\} \quad (5)$$

$\{\ddot{u}\}$ is the second order derivative of the displacement vector to the time. $\{F_{initial}\}$ is the preload or initial force which was discussed in the first part of this paper.

Force Analysis of Ring-Liner Lubrication

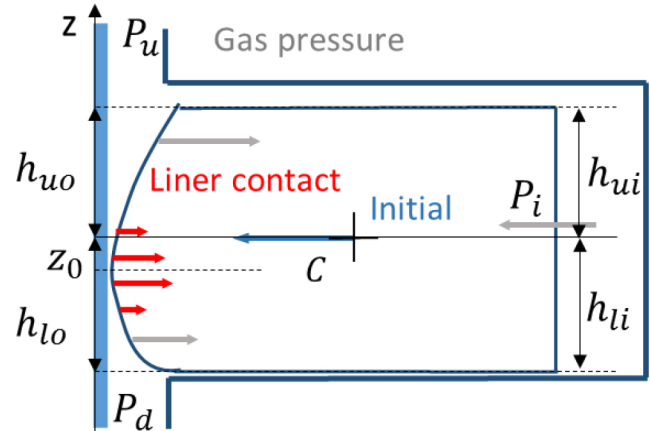


Figure 1. Force Analysis in the Radial Direction

When the rings are assembled into the piston and are running under different engine environments, the expected forces in the radial direction are shown in Figure 1. z_0 is the axial position of the minimum clearance between the ring and the liner with respect to the centroid (positive direction is pointing upwards). Gas pressure changes with time around the piston rings, especially for the top two rings. Gas from the piston upper land and the lower land pushes the rings inwards at the ring-liner interfaces. Gas inside the ring grooves acts on the rings as normal load and pushes the rings to the liner.

$$f_{r,gas} = P_i(h_{ui} + h_{li}) - P_u(h_{uo} - z_0) - P_d(h_{l0} + z_0) \quad (6)$$

Second is the ring initial force. For the top two rings, the initial force comes from the rings' strain energy when they are closed from their free shapes. After the top two rings are closed and are assembled into the piston, they tend to expand and push themselves to the liner. Part I of this paper provides the equations to calculate the initial force and only the ones in the radial direction are needed in the ring-liner lubrication model.

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

For the oil control ring, the initial force refers to the spring force of the oil control ring spring, F_p , which is an input from the ring design. This spring force is designed to be uniform along the circumferential direction in order to guarantee good conformability between the oil control ring and the liner.

Contact force from the liner is composed of two parts. The first one is dry contact force which is calculated using the simplified Greenwood-Tripp model [23] [24] and the second one is hydrodynamic pressure which can be calculated by the correlations derived by Chen, Li and Liu [9] [25] [26].

Hydrodynamic Pressure Correlations

When the piston rings slide over the liner with lubricant, hydrodynamic pressure is generated in the oil film which can be estimated by Reynolds equation. Conventional lubrication models calculate the hydrodynamic pressure without consideration of surface roughness. However, this assumption may cause problems when predicting hydrodynamic pressure between the piston rings and the liner. The twin-land oil control ring running surface is parallel to the liner surface and if without consideration of roughness, solving Reynolds equation would predict no generation of hydrodynamic pressure. Thus, roughness geometry cannot be neglected and is the only contributor to hydrodynamic pressure generation in this case. The twin-land oil control ring is designed in this way to control the oil film thickness on the liner down to a level of surface roughness. As for the top two rings, even though they have ring profile which could help generate hydrodynamic pressure, their oil supply in most of the time is determined by the oil control ring. When the supplied oil film thickness is limited to the level of surface roughness, it is not surprising that the oil film thickness between the top two rings and the liner is also at the level of surface roughness. Therefore, the roughness geometry is still an important factor to the hydrodynamic pressure generation between the top two rings and the liner. In the calculation of the ring-liner lubrication, the oil consumption of the oil film on the liner is not considered because the effect is negligible. For a typical 4-cylinder 2L engine at 2000rpm/full-load, the oil consumption is roughly 10g/hr (2.5g/hr-cylinder). This number is equivalent to 2nm/cycle-cylinder of oil consumed from the liner which is approximately 1% of the oil film thickness on the liner.

Deterministic Correlations of Twin-Land Oil Control Ring

A deterministic model solving Reynolds equation of the sliding contact between the OCR and the cylinder liner at the scale of liner surface asperity has been developed by Li and Chen [9] [25]. It considers the effect of the liner roughness on hydrodynamic pressure generation in the oil film and cavitation behind surface asperity. The oil supply is assumed to be fully-flooded and the ambient pressure is assumed to be atmospheric pressure. In this model OCR surface is considered smooth which is to say its roughness is considered much smaller than the liner roughness, as shown in Figure 2.

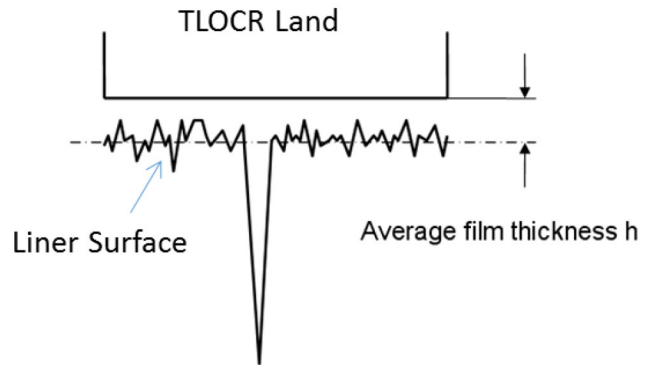


Figure 2. A Flat Ring on a Rough Surface

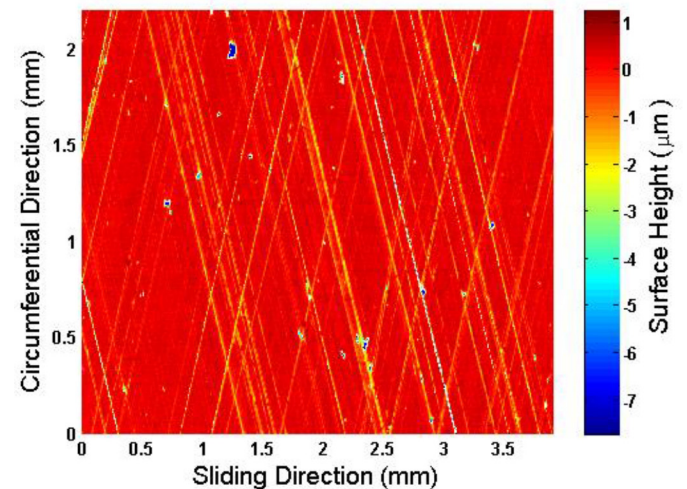


Figure 3. Liner Finish Measurement [25]

Figure 3 shows the 3D deterministic measurement of a liner finish. A surface measurement patch usually has a size of 3mm x 3mm and the land width of OCR varies from 0.06mm to 0.3mm. Quasi-steady state simulation results on the 3D deterministic measurement for a range of OCR-liner clearances are used to build a correlation between the ring-liner clearance h and the average hydrodynamic pressure P_h . Here, quasi-steady state simulation results refer to average hydrodynamic behavior between the ring and the liner with a given 3D surface measurement, ring-liner clearance h , ring sliding speed V and oil dynamic viscosity μ .

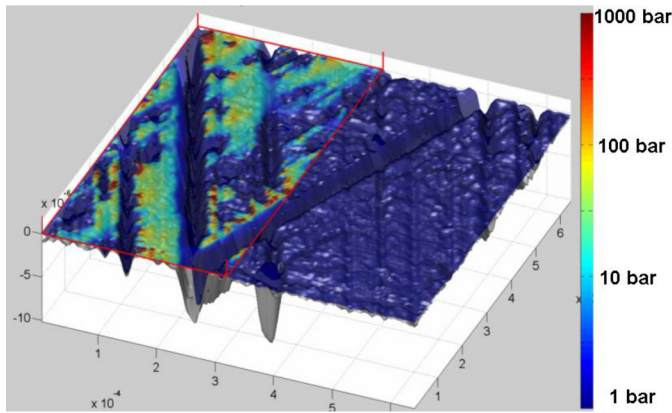


Figure 4. Hydrodynamic Pressure between the Ring Face and the Liner Finish [25]

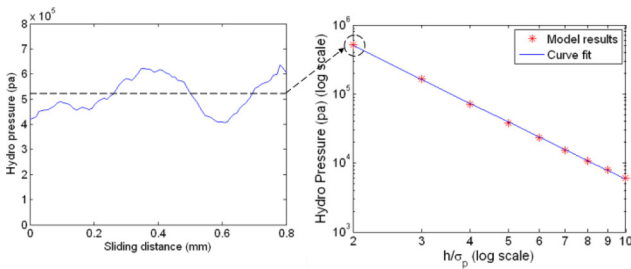


Figure 5. Hydrodynamic Pressure Generation of a Ring Face Sliding at a Constant Clearance [25]

The inter-asperity hydrodynamic pressure is deterministic at given positions of the liner surface patch (Figure 4). The hydrodynamic behavior varies due to the change of surface micro-geometry. Then, after taking the average over the entire ring sliding distance (or time steps), an average hydrodynamic pressure with the given surface measurement, ring land width, ring-liner clearance, sliding speed and oil dynamic viscosity is obtained. Repeating this process over a range of ring-liner clearances (Figure 5), the dependency of the average hydrodynamic pressure P_h on the oil film thickness h is derived.

$$P_h = P_0 \frac{\mu V}{(\mu V)_0} \left(\frac{h}{\sigma_p} \right)^{-a} \quad (8)$$

where P_0 and a are correlation coefficients and μ is oil viscosity, V is the sliding speed and $(\mu V)_0$ is the reference value of μV used in the simulation, h is the ring-liner clearance and σ_p is the plateau roughness. This correlation is going to be used in the ring-liner lubrication model to calculate the hydrodynamic pressure between the oil control ring and the liner.

Deterministic Correlations of Top Two Rings under Partially Flooded Boundary Condition

A deterministic model solving hydrodynamic pressure between the top two rings and the liner has been developed by Chen [25]. This model takes into consideration of surface roughness and ring profile but not the squeezing effect. However, unlike the OCR model, the top two rings deterministic model does not assume sufficient oil supply. Instead, it needs to use the oil film thickness result from the OCR

model as an input to determine how much oil is fed into the top two rings. Chen has also developed a correlation including the hydrodynamic pressure of the top-two-ring, the minimum clearance between the top-two-ring and the liner and the OCR ring-liner clearance (the definitions are shown in Figure 6).

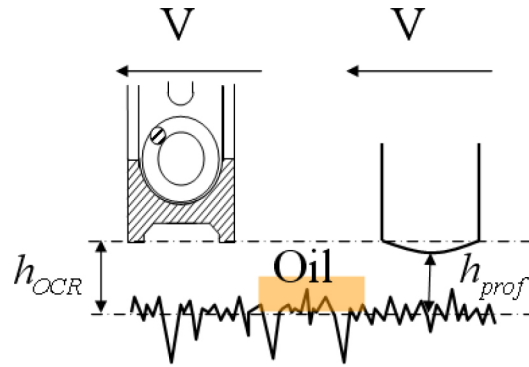


Figure 6. the OCR Ring-Liner Clearance and the Top Two Ring Minimum Ring-Liner Clearance

$$P_{hydro} = \left(\frac{h_{OCR}}{h_{prof}} \right)^{K_p} \frac{\mu V}{(\mu V)_0} (a_p P_{0,OCR}) \left(\frac{h_{prof}}{\sigma_p} \right)^{-K_{OCR}} \quad (9)$$

where $P_{0,OCR}$ and K_{OCR} are coefficients from the OCR deterministic correlation, μ and V represent oil dynamic viscosity and ring sliding speed. $(\mu V)_0$ is the reference value of μV in the simulation. h_{OCR} is the OCR ring-liner clearance and h_{prof} is the minimum clearance between the top-two-ring and the liner. The term $\left(\frac{h_{OCR}}{h_{prof}} \right)^{K_p}$ represents the filling effect. Since the rate of filling effect changes with oil supply, K_p is not a constant and correlated as a linear function of $\frac{h_{OCR}}{\sigma_p}$ (first order approximation). The term a_p captures the ring profile effect and given a liner finish and a ring profile, a_p will be a constant.

In Chen's thesis, this correlation is also called the single phase correlation since it considers oil vapor and cavitation effect but does not include the gas pressure effect. However, inside internal combustion engines, especially in the compression and expansion strokes, the gas pressure is an important boundary condition to determine the hydrodynamic pressure in the oil film. Liu has developed a multiphase correlation (10) which depends on Li's multiphase deterministic model with the consideration of gas pressure effect [27] [26].

$$P_{multiphase} = P_{single\ phase} + P_{static} \quad (10)$$

$P_{single\ phase}$ is the single phase correlation shown above and P_{static} is the gas hydrostatic pressure which captures the effect of boundary gas pressure. P_{static} is calculated by solving Reynolds equation on gas phase with the consideration of density dependency of pressure. Figure 7 shows the results of gas hydrostatic pressure with different ring-liner clearances. The boundary pressure is set to be 5atm and 30atm. h is the minimum clearance between the profiled ring and the liner. σ_p refers to the plateau roughness. Looking at the results, one may argue that the gas hydrostatic pressure is similar to water.

function especially when the minimum clearance is small. As a result, in the ring liner lubrication model, the calculation of P_{static} is simplified by a step function in (6).

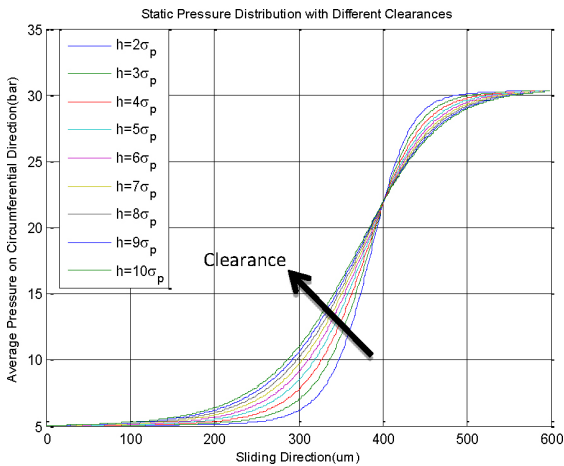


Figure 7. Gas Hydrostatic Pressure

Correlations of Top Two Rings under Fully-flooded Boundary Condition

The oil supply to the top two rings does not always come from the oil film left by the OCR. Sometimes around the top dead center or the bottom dead center, bridging may happen and it will bring additional oil from the piston to the liner [1] [2] [3] [4]. This extra amount of oil by-passes the OCR and is directly fed into the top two rings. Thus, the boundary condition of the top two rings will become fully flooded. The order of magnitude of the oil film thickness that flows to the liner due to bridging is around $10\mu\text{m}$, which exceeds the correlating range of the top-two-ring deterministic correlation. The large oil film thickness also implies that the change of inter-asperity geometry ($\sim 0.1\mu\text{m}$) is not important anymore. Then Reynolds Equation on a given ring profile and a smooth liner can be applied directly. The geometry is shown in Figure 8.

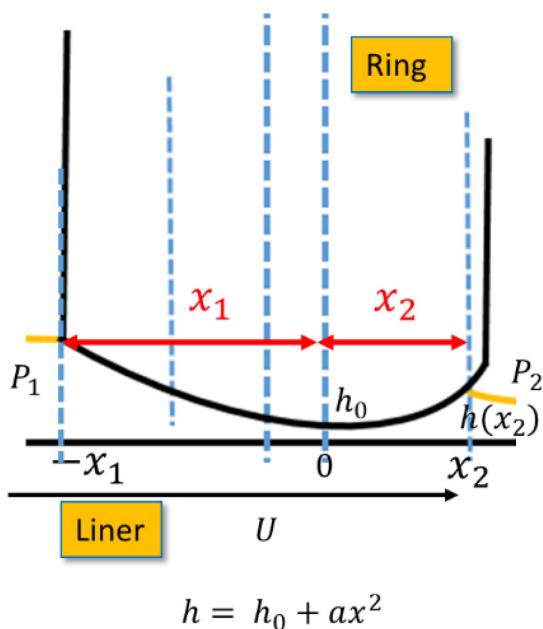


Figure 8. Schematic for Ring Profile and Liner

The oil supply is sufficient at the leading edge (x_1 , which is an input to the model). x_2 refers to the end of the wetting area and at this point, a boundary condition of $\frac{\partial P}{\partial x}|_{x_2} = 0$ is applied. x_2 is dependent on the leading edge length x_1 and the minimum ring-liner clearance h_0 . The ambient pressure is the atmospheric pressure and similar to the top-two-ring deterministic correlation, the equation (6) is used to calculate the boundary gas pressure force on the ring and it is not included in the hydrodynamic pressure calculation. Another assumption is applied here that compared to the Couette flow, the oil squeezing effect is small and neglected. Thus, the steady state Reynolds Equation is going to be used. Since $\frac{\partial P}{\partial x}|_{x_2} = 0$, the steady state Reynolds Equation can be written as:

$$\frac{dP}{dx} = \frac{6\mu U}{h^3} (h - h(x_2)) \quad (11)$$

Applying the ambient pressure boundary condition, the following equation can be derived.

$$0 = \int_{-x_1}^{x_2} \frac{dP}{dx} dx = \int_{-x_1}^{x_2} \frac{6\mu U}{h^3} (h - h(x_2)) dx \quad (12)$$

The dependency of x_2 on x_1 is obtained

$$h(x_2) = \frac{\int_{-x_1}^{x_2} \frac{dx}{h^2}}{\int_{-x_1}^{x_2} \frac{dx}{h^3}} \quad (13)$$

The hydrodynamic force per unit length ($f_{hydrodynamic}$) equals $f_{hydrodynamic} = \int_{-x_1}^{x_2} P dx$. Integrating by part, this equation can be written as

$$f_{hydrodynamic} = \int_{-x_1}^{x_2} x \frac{dP}{dx} dx = \int_{-x_1}^{x_2} x \frac{6\mu U}{h^3} (h - h(x_2)) dx \quad (14)$$

By solving (13) and (14) numerically, the results shown below in Figure 9 is obtained. $f_{hydrodynamic}'$ is the normalized hydrodynamic pressure force $\frac{6\mu U}{ah_0}$ unit length. $f_{hydrodynamic}$ equals to the multiplication of this term by ah_0 :

$$f_{hydrodynamic} = \overline{f_{hydrodynamic}'} * \frac{6\mu U}{ah_0} \quad (15)$$