

Oil Transport Cycle Model for Rotary Engine Oil Seals

Mathieu Picard, Camille Baelden, and Tian Tian
Massachusetts Institute of Technology

Takayuki Nishino, Eiji Arai, and Hiroyuki Hidaka
Mazda Motor Corp.

ABSTRACT

The rotary engine provides high power density compared to piston engine, but one of its downside is higher oil consumption. A model of the oil seals is developed to calculate internal oil consumption (oil leakage from the crankcase through the oil seals) as a function of engine geometry and operating conditions. The deformation of the oil seals trying to conform to housing distortion is calculated to balance spring force, O-ring and groove friction, and asperity contact and hydrodynamic pressure at the interface. A control volume approach is used to track the oil over a cycle on the seals, the rotor and the housing as the seals are moving following the eccentric rotation of the rotor. The dominant cause of internal oil consumption is the non-conformability of the oil seals to the housing distortion generating net outward scraping, particularly next to the intake and exhaust port where the housing distortion valleys are deep and narrow. Simulation with housing transverse waviness shows that increasing spring force can lead to an unexpected increase in internal oil consumption. Roughness and O-ring friction can also increase significantly internal consumption for small housing distortion. Calculated internal oil consumption is on the same order of magnitude as measurement.

CITATION: Picard, M., Baelden, C., Tian, T., Nishino, T. et al., "Oil Transport Cycle Model for Rotary Engine Oil Seals," *SAE Int. J. Engines* 7(3):2014, doi:10.4271/2014-01-1664.

INTRODUCTION

The rotary engine provides a high power density and low vibration when compared to the piston engine. This makes the rotary engine a prime candidate for sports cars due to its small weight and for hybrid vehicles where packaging is critical. However, its trochoidal shape leads to a more complex system to seal the gas mixture in the different chambers and crankcase oil, as shown in Figure 1. Additionally to the oil injected to lubricate the apex, side and corner seals, oil can leak from the crankcase to the combustion chamber through the inner and outer oil seals, defined here as internal oil combustion (IOC). IOC accounts for about half the total oil consumption, which is far greater than the necessary amount of oil to lubricate the contact between the oil seals and the side housing. A better understanding of the oil transport mechanisms through the oil seals would help reduce IOC, as the oil seals are the main barrier between crankcase oil and the combustion chamber. In particular, as the O-rings are sealing the oil seal grooves, oil must pass at the seal-housing interface during inward-outward motion of the seals created by the eccentric motion of the rotor, as shown in Figure 2. Details of the rotary engine kinematics are given by Yamamoto [1].

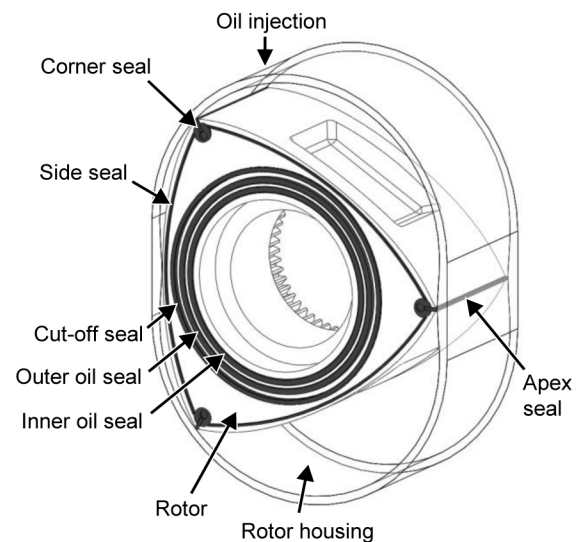


Figure 1. Rotary engine sealing system

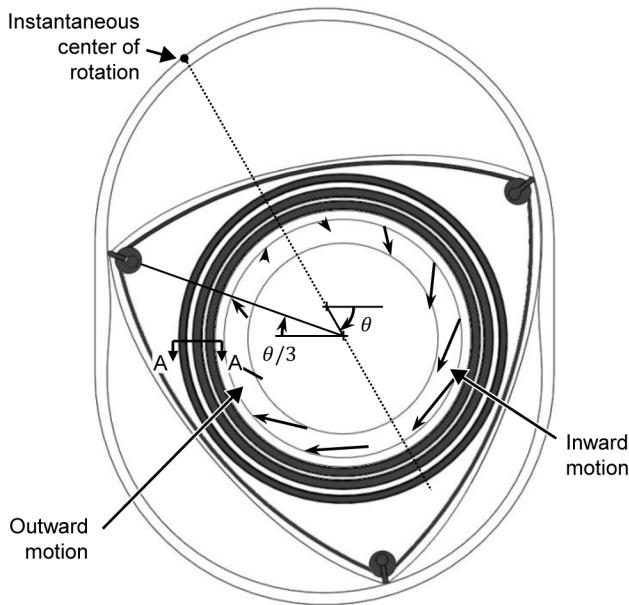


Figure 2. Side view of the engine showing inward and outward motion of the oil seals due to the eccentric motion of the rotor

Apex seal dynamics and contact have been the focus of many studies, both from experiments (e.g. [2, 3]) and modeling (e.g. [4, 5]). However, oil seals have been the focus of only a few studies. Froede [6] shows the oil consumption for three oil seal configurations from the early development of the rotary engine. Oil consumption for the best design is small, but for the worst design reaches 420 g/hr at 7000 RPM. Although, these results are from a peripheral port engine, it shows that oil seal design is critical in controlling IOC. Jones [7] and Prasse et al. [8] have also presented a few different oil seal configurations used before the modern version of the rotary engine. Finally, the oil seal configuration is roughly unchanged from the configuration shown by Muroki et al. [9]. The current oil seal configuration of the Mazda RENESIS engine [10] is shown in Figure 3. To the author's knowledge, oil seal development has mainly been empirical and no oil transport model is presented in the literature.

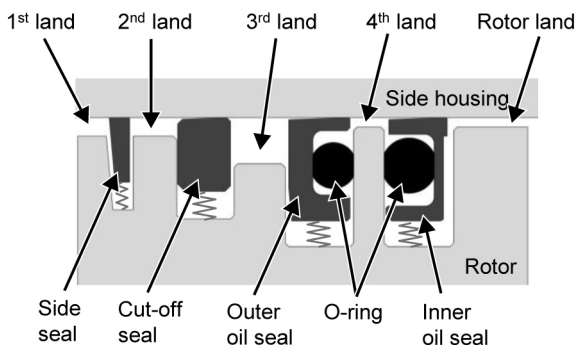


Figure 3. Section view A-A from Figure 2 showing the current oil seal configuration

This paper presents an oil transport cycle model for the oil seals of the rotary engine, including seal deformation to the distorted side housing, asperity contact and hydrodynamic pressure generation at the housing-seal interface, and the

resulting oil displacement. The goal is to predict the amount of oil that is allowed to pass the outer oil seal as a function of the engine configuration and operating conditions. As there is no mechanism to bring the oil inward after the outer oil seal, oil passed the outer oil seal is an indicator of IOC.

The model shows that the main mechanism for IOC is the difference of conformability of the seal to the side housing during inward and outward motion which generates outward scraping. Due to the eccentric motion, not only the housing distortion is important, but also its shape. This leads to the counter-intuitive result that increasing spring force behind the oil seals can lead to increasing oil consumption for some distortion shapes. A larger roughness also increases oil consumption, but significantly less than non-conformability.

MODEL DESCRIPTION

The goal of the model is to calculate oil seal performance as a function of the geometry and operating conditions. The main inputs to the model are (1) the geometry and material of the inner and outer oil seals (IOS and OOS) and their corresponding spring force, (2) the distortion and roughness of the side housing, and (3) the engine eccentricity and speed. The main model outputs are (1) oil that passed the OOS which is an indicator of internal oil consumption, (2) oil seal friction, and (3) contact pressure which is an indicator of wear. This paper focuses on internal oil consumption as the wear of the oil seals is under control, and the friction is the next step after IOC.

For every crank angle step, the model first solves the rigid-body displacement of the oil seals in the rotor plane, balancing centrifugal force and the O-ring reaction force. Second, the model solves the out-of-plane deformation to balance the spring force and the asperity contact and hydrodynamic pressure at the seal-housing clearance. The problem is approximated as quasi-static as the mass of the each seal times their relative acceleration in their groove is much smaller than the applied forces due to the O-ring which is preventing fast transition from one side of the groove to the other, as seen for the apex seal or typical piston rings. Third, once the position and deformation of the seals are known, a control volume approach is used to track the oil on the housing, seals, and rotor. The same procedure is repeated for a few engine revolutions until calculated internal oil consumption reaches a steady state. The next sub-sections describe in details the three steps of the model.

In-Plane Equilibrium

Figure 4 shows a free-body diagram of the IOS in the rotor plane. The force created by gas pressure on the seal is not included because 2nd land is connected with the intake or exhaust side port during most of the cycle so the pressure around the oil seals should remain about atmospheric. Acceleration from eccentric rotation is mainly radial, and the redistribution of the O-ring normal force. If the displacement is

too large, the seal contacts the groove. The model limits the displacement and calculates the resulting contact force. Deformation of the seal in the radial direction is neglected as its magnitude is small compared to rigid body motion and does not change the seal-housing clearance and therefore does not influence oil transport. Variation of O-ring normal force and groove reaction force influences out-of-plane equilibrium by changing the resulting friction force opposing axial motion seal rotor, as shown in the next section. Friction at the seal-housing interface is neglected in the current version of the model as it is a secondary effect, but could be included using the methodology presented in this paper.

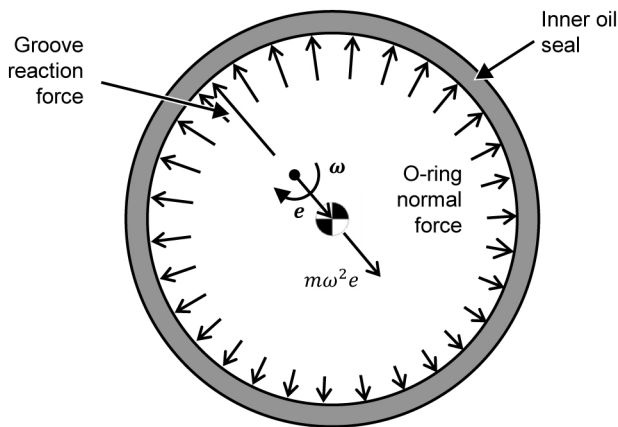


Figure 4. Forces acting on the seal in its plane

Out-of-Plane Seal Deformation

Calculating an accurate out-of-plane deformation is essential as it determines the amount of oil that is allowed to pass at the interface during inward and outward motion of the seals. As shown in Figure 5(a), the spring is pushing the seal towards the housing, trying to make the seal conform to the housing distortion. Housing distortion is an input to the oil seal cycle model obtained from finite element analysis including thermal deformation when applicable. Although not yet validated by experiments, housing distortion from finite element analysis provides an adequate deformation shape and amplitude to study its effect on IOC.

As shown in Figure 5(b), the seal deforms according to its rigidity and the resulting contact pressure from the contact with the housing. Pressure at the interface is a function of the clearance and is calculated based on asperity contact and hydrodynamic lubrication. As shown in Figure 6, O-ring and groove friction also generate out-of-plane forces on the seal due to axial motion of the seal relative to its groove. This seal-groove relative motion is created by rotor secondary motion. Rotor secondary motion is defined as motion of the rotor deviating from the designed primary motion which is the rotation of the eccentric shaft and the rotation of the rotor around the shaft at 1/3 the engine speed. Secondary motion is calculated using finite element analysis to determine displacement and deformation of the rotor and crankshaft

under engine operating conditions. The impact of secondary motion on the oil seal model is the axial motion and tilt of the rotor that creates seal-groove relative motion.

In the following subsections, calculation of the forces is first detailed and then the method for solving the seal deformation is explained.

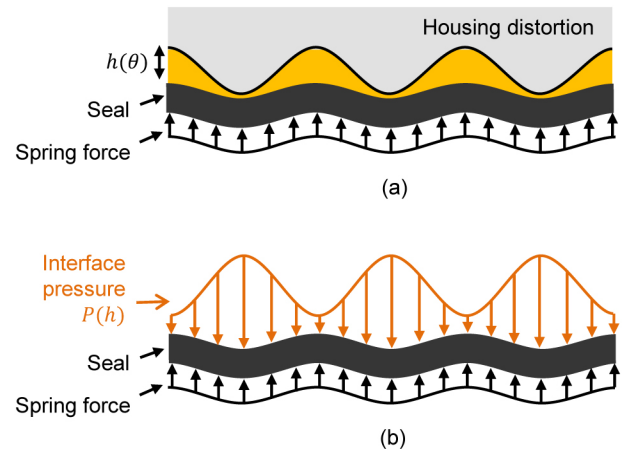


Figure 5. Schematic of the (a) seal deformation under spring force and (b) resulting interface pressure

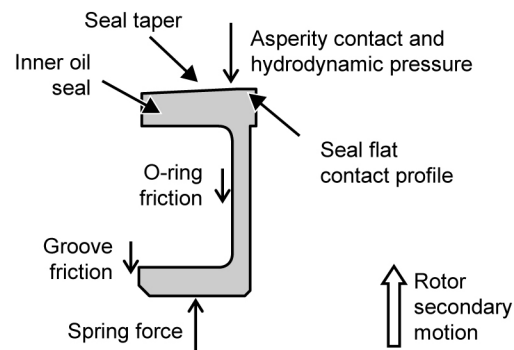


Figure 6. IOS cut-view and out-of-plane forces

Asperity Contact Pressure

Asperity contact pressure is calculated from the simplified Greenwood-Tripp model [11] proposed by Hu et al. [12]:

$$P_{contact} = \begin{cases} 0 & \text{for } h/\sigma_p \geq z \\ P_c \left(z - \frac{h_1}{\sigma_p} \right)^{K_c} & \text{for } h/\sigma_p < z \end{cases} \quad (1)$$

in which P_c , z and K_c are constant calculated from surface topography taken from Chen [13] to be 1.267×10^5 Pa, 4.8 and 6.804 respectively, h_1 is the interface clearance and σ_p is the standard deviation of roughness on the plateau area. Unless otherwise specified σ_p is taken to be $0.05 \mu\text{m}$ from housing surface measurement.

Equation and Constants Hydrodynamic Pressure

Hydrodynamic pressure is generated by microscopic and macroscopic effects. During inward motion, the macroscopic profile does not generate pressure as the running profile of the seal is flat and the taper portion is diverging. The only contribution to hydrodynamic pressure generation is roughness which generates alternate high-pressure and cavitation regions. Hydrodynamic pressure generated by roughness ($P_{hydro,r}$) under a flat macroprofile can be calculated from the correlation developed by Chen [13]:

$$P_{hydro,r} = \frac{\mu V}{\mu_0 V_0} P_h \left(\frac{h_1}{\sigma_p} \right)^{-K_h} \quad (2)$$

in which μ and μ_0 are the actual and reference viscosities, V and V_0 are the actual and reference seal velocities, and P_h and K_h are constants from the deterministic model. Unless otherwise mentioned, actual viscosity and reference viscosity are taken to be $\mu = \mu_0 = 0.005$ Pa·s, reference velocity is $V_0 = 5$ m/s, and P_h and K_h are 3.9×10^7 Pa and 2.39 respectively, calculated with Chen's deterministic model [13] for the housing surface. Roughness of the seal running surface is neglected, as it is typically smaller than the housing roughness. The correlation for the hydrodynamic pressure was developed for a piston ring moving always perpendicularly to its contact line. However, in the rotary engine, the oil seals are moving radially as well as circumferentially. This effect is neglected here as it is small compared to the effect of housing distortion, but should be considered for a seal with better conformability. The norm of the velocity is used in the correlation.

During outward motion, the macroscopic profile contributes to hydrodynamic pressure generation as the taper and housing form a converging profile, as shown in Figure 7. This effect is beneficial as it allows more oil to pass towards the crankcase by generating pressure during inward motion and separation seal and housing. In the present model, the microscopic and macroscopic effects are assumed to add linearly. Although not exact, this approximation gives the correct trend which is the goal of the model.

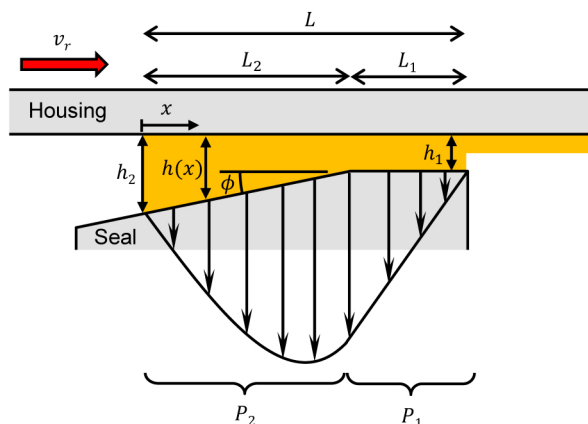


Figure 7. Pressure generated by the seal taper during outward motion

The goal is to obtain the force generated by the macroscopic profile as a function of clearance, radial velocity and seal geometry. The lubrication problem can be reduced to a 1D problem in the radial direction as variations of clearance and pressure are much smaller in the circumferential direction (out of the page in Figure 7). Variations in the radial direction scales as $1/L_2$ while variations in the circumferential direction scales as $1/R$, in which L is the length of the full-film region in the radial direction and R is the radius of the seal. This leads to variations in the radial direction on the order of 50 times larger and thus the circumferential terms in the 2D Reynolds equation can be neglected.

The flow rate (Q') through the seal-housing clearance is given by a combination of Poiseuille and Couette flows:

$$Q' = \frac{h^3}{12\mu} \left(-\frac{dP}{dx} \right) + \frac{v_r h}{2} \quad (3)$$

in which P is the hydrodynamic gauge pressure, x is the coordinate pointing inward in the radial direction as defined in Figure 7, v_r is the seal radial velocity, and h is the clearance along x calculated with small angle approximation as:

$$h(x) = \begin{cases} h_2 - \phi x & \text{for } 0 \leq x < L_2 \\ h_1 & \text{for } L_2 \leq x \leq L \end{cases} \quad (4)$$

in which h_2 is the oil thickness at the taper inlet, L_2 is the length of the full-film taper region, and ϕ is the angle of the taper (2°). Boundary conditions are zero gauge pressure at the two ends and continuity of pressure and flow rate between the two regions. Integrating, applying boundary conditions and solving for the flow rate yields:

$$Q' = \frac{v_r}{2} \frac{\int_0^L \frac{1}{h^2} dx}{\int_0^L \frac{1}{h^3} dx} = \frac{\frac{1}{h_1} - \frac{1}{h_2} + \frac{L_1 \phi}{h_1^2}}{\frac{1}{h_1^2} - \frac{1}{h_2^2} + \frac{2L_1 \phi}{h_1^3}} \quad (5)$$

in which L_1 is the length of the flat region (0.2 mm). Pressure of the flat region (P_1) is given by a linear distribution

$$P_1 = \frac{6\mu x_1}{h_1^2} \left(\frac{2Q'}{h_1} - v_r \right) \quad (6)$$

in which x_1 is the distance from inside the seal. Pressure along the converging region is sketched in Figure 7 and expressed as

$$P_2 = \frac{6\mu}{\phi} \left(\frac{1}{h} - \frac{1}{h_2} \right) \left[v_r - Q' \left(\frac{1}{h} - \frac{1}{h_2} \right) \right] \quad (7)$$