

## Full Length Article

# Numerical investigation of thermo-mixed elastohydrodynamic lubrication in the first reciprocating tangential seal ring under ultra-high pressure

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## ABSTRACT

Compressors for low-density polyethylene (LDPE) production utilize metallic components to maintain sealing at high pressures of 260 MPa. This study develops a thermo-mixed elastohydrodynamic lubrication (TMEHL) theory, integrated with the Greenwood-Tripp (GT) contact model, to investigate the lubrication mechanisms of the first metal tangential packing ring (T-ring) under extreme pressure conditions. The theory incorporates frictional heat generation and the resulting temperature field throughout the entire reciprocating cycle, offering critical insights into the thermal stability of long-chain polyether lubricants. The effects of surface roughness, environmental temperature, and reciprocating velocity on lubrication performance are systematically evaluated. The results demonstrate that, even under ultra-high pressure, elevated plunger velocity generates a pronounced hydrodynamic pressure peak at the inlet region, while a local pressure reduction resembling cavitation-like behavior may occur near the outlet. Asperity contact is primarily concentrated in regions of high static contact pressure, and the interface remains in a mixed-lubrication regime throughout the cycle, although the contact load distribution varies significantly between strokes. Under severe operating conditions, the maximum oil-film temperature reaches approximately 542 K. Reducing surface roughness promotes fluid pressure and diminishes asperity load sharing, thereby mitigating temperature rise. Conversely, increasing velocity enhances fluid pressure slightly but intensifies frictional heating. An increase in environmental temperature results in an approximately linear rise in film temperature, with only a limited effect on load support. Overall, friction force is determined by the combined effects of boundary pressure, load, and velocity, whereas leakage is primarily influenced by sliding velocity and pressure gradient, becoming more significant during the outstroke. The proposed model establishes a theoretical framework for assessing metal seal lubrication performance and optimizing surface roughness, lubricant design, and operating parameters in ultra-high-pressure LDPE compressors.

## 1. Introduction

The plunger-packing friction pair serves as a key functional interface in hydraulic, pneumatic, and thermodynamic systems. It is subjected to extremely harsh operating conditions, including high-pressure cyclic loading, rapid reciprocating velocity, and significant thermo-mechanical coupling. These factors increase susceptibility to wear, scuffing, leakage and erosion failure. In ultra-high-pressure compressors, such failures can contribute to up to 40% of system downtime. In many sealing applications, the sealed working medium also functions as

a lubricant, resulting in an inherent trade-off between film buildup and leakage that requires careful optimization [1]. Salant [2] developed an early one-dimensional reciprocating mixed lubrication model for O-ring seals by integrating it with the Greenwood-Williams (GW) contact model. This approach enabled the prediction of pressure and film thickness distributions within the contact zone, as well as the calculation of engineering-focused indicators such as leakage rate and friction force. Ahmad [3,4] and Dou [5] have conducted comprehensive CFD research on the mechanisms and prevention of fluid-induced corrosion failure under high-pressure conditions.

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**Nomenclature**

|            |                                                                          |                                |                                                                      |
|------------|--------------------------------------------------------------------------|--------------------------------|----------------------------------------------------------------------|
| $c$        | Specific heat of the oil                                                 |                                |                                                                      |
| $D$        | Plunger diameter                                                         |                                |                                                                      |
| $E$        | Effective elastic modulus, $1/E = (1 - \nu_1^2)/E_1 + (1 - \nu_2^2)/E_2$ |                                |                                                                      |
| $f_c$      | Coefficient of dry friction                                              |                                |                                                                      |
| $F_f$      | Friction force on the T-ring surface                                     |                                |                                                                      |
| $h$        | Nominal film thickness                                                   |                                |                                                                      |
| $h_{sc}$   | Static contact film thickness                                            |                                |                                                                      |
| $h_s$      | Thermal contact conductance coefficient of the boundary $S_3$            |                                |                                                                      |
| $h_T$      | Mean truncated film thickness                                            |                                |                                                                      |
| $H$        | Film ratio, $H = h/\sigma$                                               |                                |                                                                      |
| $I_1$      | Influence coefficient matrix                                             |                                |                                                                      |
| $k_L, k_G$ | Thermal conductivity of the oil and the gas                              |                                |                                                                      |
| $k_p, k_s$ | Thermal conductivity of the plunger and the seal                         |                                |                                                                      |
| $k$        | Thermal conductivity                                                     |                                |                                                                      |
| $p_l, p_h$ | Pressure on the lower and higher sides around the seal                   |                                |                                                                      |
| $p_m$      | Pressure between the R-ring and T-ring                                   |                                |                                                                      |
| $p_s$      | Static contact pressure                                                  |                                |                                                                      |
| $p, p_c$   | Fluid and asperity pressure                                              |                                |                                                                      |
| $p_{cav}$  | Cavitation pressure                                                      |                                |                                                                      |
| $q$        | Flow rate                                                                |                                |                                                                      |
| $q_p, q_s$ | Local heat flow into the plunger and seal                                |                                |                                                                      |
| $x, z$     | Coordinate system                                                        |                                |                                                                      |
| $x_0, x_1$ | x-coordinate of the contact-zone boundary near and away from the origin  |                                |                                                                      |
|            |                                                                          | $T$                            | Fluid film temperature                                               |
|            |                                                                          | $T_0$                          | Reference temperature                                                |
|            |                                                                          | $T_C$                          | Temperature of the cooling system                                    |
|            |                                                                          | $T_m$                          | Averaged oil film temperature across film thickness                  |
|            |                                                                          | $T_G$                          | Gas temperature                                                      |
|            |                                                                          | $T_S$                          | Temperature of the sealing contact surface                           |
|            |                                                                          | $U$                            | Surface velocity of the plunger                                      |
|            |                                                                          | $\alpha$                       | Viscosity-pressure coefficient                                       |
|            |                                                                          | $\alpha_S$                     | The convection heat transfer coefficient of the gas                  |
|            |                                                                          | $\alpha_E$                     | The convection heat transfer coefficient of the radial ring (R-ring) |
|            |                                                                          | $\beta$                        | Viscosity-temperature coefficient                                    |
|            |                                                                          | $\beta_1$                      | Asperity height                                                      |
|            |                                                                          | $\eta_1$                       | Asperity density                                                     |
|            |                                                                          | $\theta$                       | Fluid density ratio, $\theta = \rho/\rho_L$                          |
|            |                                                                          | $\mu$                          | Lubricant viscosity                                                  |
|            |                                                                          | $\rho, \rho_L$                 | Density of the multiphase lubricant and the oil                      |
|            |                                                                          | $\nu_1, \nu_2$                 | Poisson ratio of packing and plunger                                 |
|            |                                                                          | $\phi_{xx}$                    | Pressure flow factor                                                 |
|            |                                                                          | $\phi_{scx}$                   | Shear flow factor                                                    |
|            |                                                                          | $\phi_f, \phi_{fs}, \phi_{fp}$ | Shear stress factors                                                 |
|            |                                                                          | $\Phi_m$                       | Averaged power dissipation density                                   |
|            |                                                                          | $\sigma$                       | RMS surface roughness, $\sigma = \sqrt{\sigma_1^2 + \sigma_2^2}$     |
|            |                                                                          | $\sigma_1, \sigma_2$           | Surface roughness of the packing and plunger                         |

seal materials and alters the contact state, which significantly impacts lubrication performance and the wear resistance of the sealing pair [6]. Temperature also directly affects lubricant film formation by influencing rheological properties, particularly viscosity [7]. To accommodate diverse operating conditions and modeling requirements, researchers have developed various thermal lubrication models. Finite element-based fluid-structure interaction methods have been designed by Xiang [8] to predict oil film temperature distributions. The study demonstrated that combined seal configurations can effectively improve sealing stability and leakage control under severe conditions. But the simplified formulations fail to capture temperature variations across the film thickness and along the axial direction, thereby limiting the method's applicability [9]. A recent study by Xu [10] indicates that transient remarkably affects film thickness, contact pressure, and leakage behavior, especially near stroke reversal. Results by the fully coupled TMEHL frameworks demonstrate that elevated pressure and temperature accelerate seal degradation and that the mechanical properties of elastomeric seals are strongly temperature-dependent. To enhance numerical accuracy and efficiency, a fully coupled solution strategy that integrates the Reynolds, energy, and elastic deformation equations has been proposed [11]. The approach enables the simultaneous prediction of pressure, temperature, and deformation fields under special conditions. Lubrication analyses under inclined loading conditions have shown that load inclination can induce asymmetric pressure and film thickness distributions, leading to severe local film thinning on the high-pressure side [12]. To improve computational efficiency, several approximate three-dimensional formulations have been proposed for the rapid and efficient solution of the energy equation [13,14]. Surface texturing functions as a lubricant reservoir and a trap for wear debris in practical applications. It also enhances lubrication performance under a mixed lubrication regime by generating local hydrodynamic effects and reducing asperity contact. However, the effectiveness of surface texturing is deeply dependent on texture parameters and may heighten the risk of leakage [15]. In addition, lubrication mechanisms under special operating conditions [16] and advanced numerical

methods have further broadened the scope of current TMEHL research [17,18].

The lubrication mechanisms of generalized metallic seals have been extensively studied both theoretically and experimentally. A transient TMEHL model that incorporates heat transfer and parametric surface texturing has been developed by Zhang [19] to enable coupled predictions of texture geometry, temperature distribution, and lubrication performance. These models provide valuable guidance for surface design and thermal management in cylinder liner systems with piston rings. Gaston [18], based on an aviation axial piston pump, proposed a thermally coupled model of lubrication efficiency loss, revealing the multiphysics interactions among temperature, viscosity, friction, and leakage losses. The results showed that under great pressure and velocity, thermal effects exert a strongly nonlinear, amplifying influence on lubrication behavior, becoming a key limiting factor for performance. Moreover, Jiao [20] developed a transient mixed lubrication model based on the CMFF method, accounting for coupled piston ring and film thickness, enabling fidelity predictions under predictive conditions. Results indicated that fluid-structure interaction significantly affects oil film thickness and pressure distribution, and neglecting deformation leads to underestimation of lubrication variations. Bernhard [21] developed a unified multiscale model spanning rigid body dynamics to microscale flow, allowing accurate dynamic predictions of lubrication in reciprocating compressors. The study showed that oil film thickness is strongly governed by dynamics, while heavy-pressure gas can improve loading capacity but also increase the risk of shear failure. A recent study [22] that incorporates measured piston ring geometries into elastohydrodynamic lubrication models has demonstrated that non-ideal surface profiles may either improve or deteriorate lubrication performance, while wear risk is often intensely localized. The study concluded that only three-dimensional EHL models driven by measured geometries can reliably predict engine lubrication behavior.

Compared with conventional elastic line contacts, rectangular seals operating in surface contact have a reduced possibility of generating and maintaining stable lubricant films. Previous studies have shown that

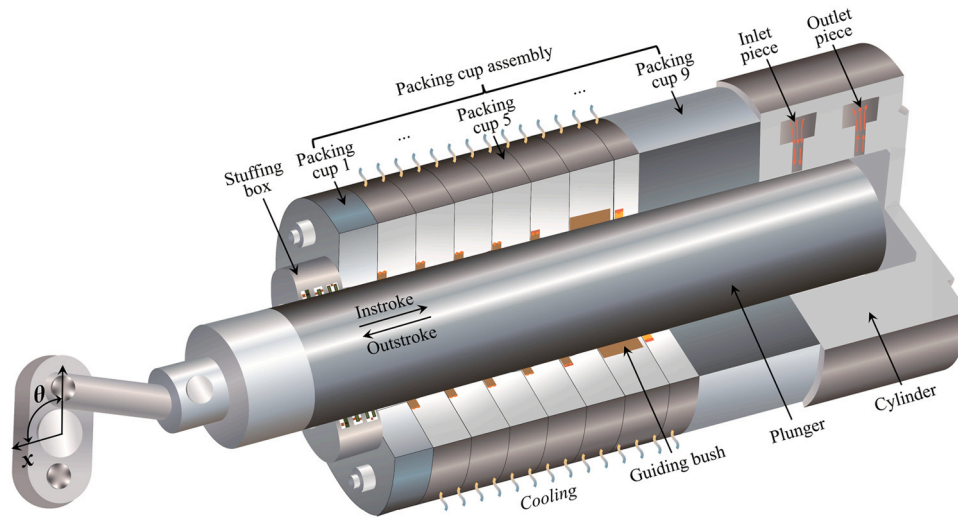


Fig. 1. Schematic representation of the cross-sectional sealing mechanism in an LDPE production compressor.

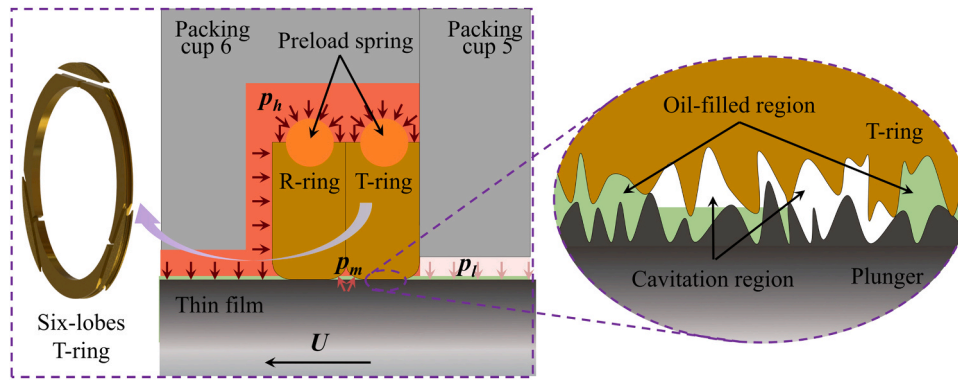


Fig. 2. Schematic diagram of the local structure and mixed lubrication in the contact region of the first packing ring set.

Table 1

Variations of gas pressure near the packings and plunger velocity with crank angle.

| Instroke |             |             |             |           | Outstroke |             |             |             |           |
|----------|-------------|-------------|-------------|-----------|-----------|-------------|-------------|-------------|-----------|
| CA, °    | $p_h$ , MPa | $p_l$ , MPa | $p_m$ , MPa | $U$ , m/s | CA, °     | $p_h$ , MPa | $p_l$ , MPa | $p_m$ , MPa | $U$ , m/s |
| 1        | 113.10      | 69.00       | 91.05       | 0.06      | 181       | 248.82      | 124.33      | 186.58      | -0.06     |
| 30       | 124.12      | 75.50       | 99.81       | 1.65      | 210       | 146.89      | 106.67      | 126.78      | -1.65     |
| 60       | 145.42      | 81.00       | 113.21      | 2.86      | 240       | 124.36      | 77.71       | 101.04      | -2.86     |
| 90       | 194.62      | 90.50       | 142.56      | 3.30      | 270       | 112.61      | 70.00       | 91.31       | -3.30     |
| 120      | 245.30      | 104.00      | 174.65      | 2.86      | 300       | 109.43      | 69.00       | 89.21       | -2.86     |
| 150      | 248.24      | 143.33      | 195.78      | 1.65      | 330       | 111.27      | 69.00       | 90.13       | -1.65     |
| 179      | 251.08      | 191.67      | 221.37      | 0.06      | 359       | 114.46      | 69.00       | 91.73       | -0.06     |

chamfered rectangular seals often exhibit localized pressure concentrations near the contact edges and that reciprocating motion significantly modifies the pressure distribution within the sealing interface [23]. The tribological performance of Glyd rings has also been identified as strongly pressure-dependent, which promotes friction, potentially resulting in higher leakage over the instroke phase [24]. Additionally, thermal lubrication models that account for heat effects have been introduced for rectangular metallic seals to enhance prediction accuracy under demanding operating conditions [25].

However, research on the thermal lubrication behavior of metallic packing seals in LDPE production hyper compressors remains limited. Tangential rings, in particular, are subjected to extremely severe contact pressures and minimal fluid film support, leading to more extreme lubrication conditions, whereas the impact of dynamic effects is comparatively minor. As a result, the lubrication mechanisms of these

sealing structures are not yet fully understood. In this study, the conventional GW asperity contact model is replaced with the GT model and integrated into a TMEHL framework that incorporates JFO cavitation boundary conditions. This model is subsequently used to analyze the lubrication performance of tangential packing seals in reciprocating ultra-high-pressure ethylene compressors.

## 2. Model establishment

### 2.1. Geometrical Model

Fig. 1 presents the schematic diagram and a photograph of an ultra-high-pressure ethylene compressor. The plunger is connected to the crankshaft on one side and to the cylinder on the other, with a packing cup assembly separating them. The packing cup assembly comprises

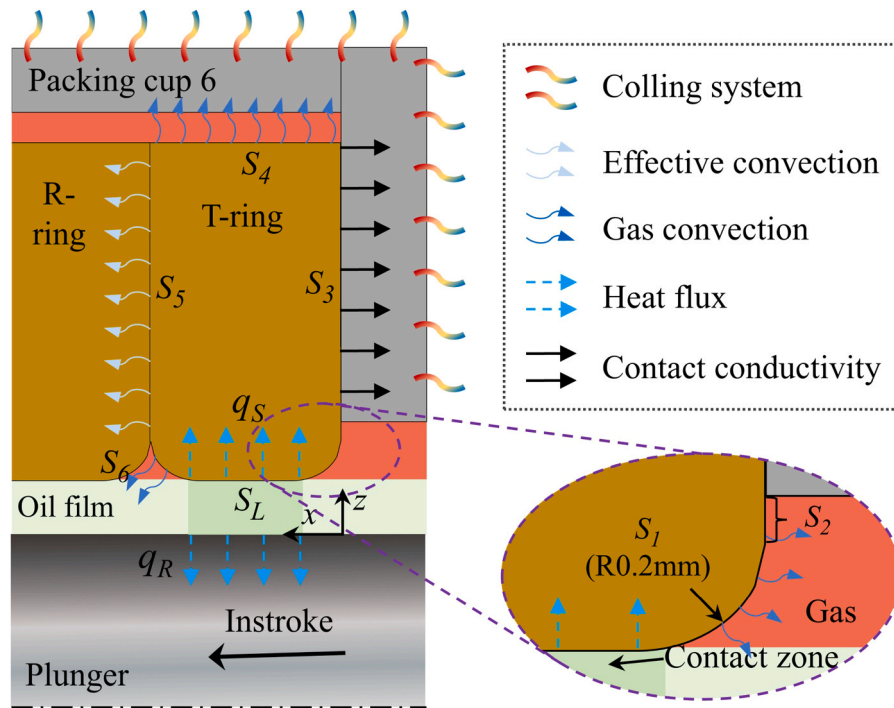


Fig. 3. Thermal boundary conditions of the T-ring.

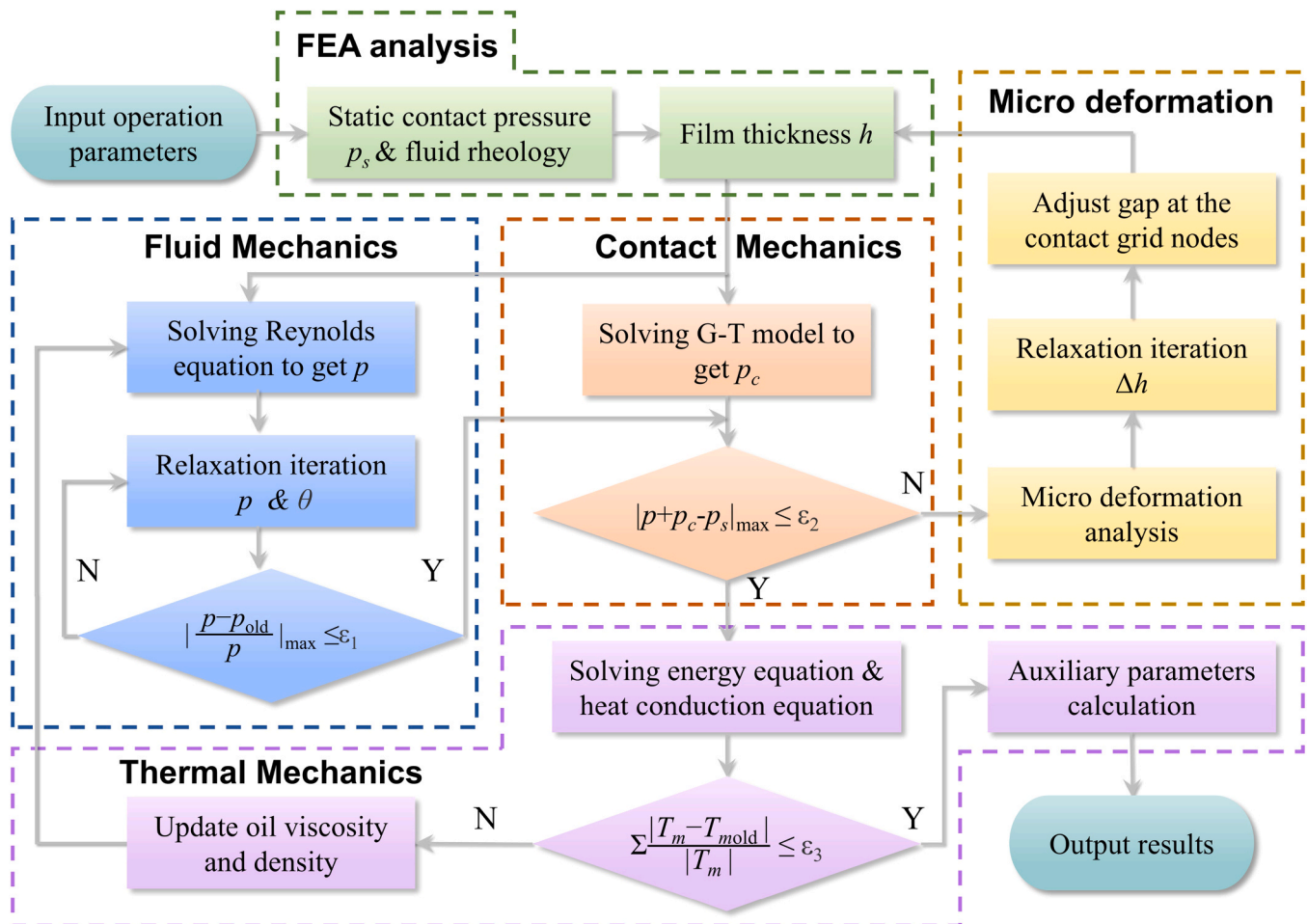


Fig. 4. Calculation flowchart.

**Table 2**  
Parameters of working conditions, geometrical and material.

| Parameters                                 | Value and unit                      |
|--------------------------------------------|-------------------------------------|
| Plunger diameter                           | 0.1 m                               |
| Stroke                                     | 0.35 m                              |
| Width of T-ring, $L$                       | 5 mm                                |
| Pressure-viscosity coefficient, $\alpha$   | $2.1 \times 10^{-8} \text{Pa}^{-1}$ |
| Viscosity-temperature coefficient, $\beta$ | $1.63 \times 10^{-2} \text{K}^{-1}$ |
| Dry friction coefficient, $f_c$            | 0.2                                 |
| Packing roughness, $\sigma_1$              | 0.1 $\mu\text{m}$                   |
| Plunger roughness, $\sigma_2$              | 0.1 $\mu\text{m}$                   |
| Asperity height, $\beta_1$                 | $1.4 \times 10^{-4} \text{m}$       |
| Asperity density, $\eta_1$                 | $2 \times 10^9 \text{m}^{-2}$       |
| Density of oil, $\rho_0$                   | 1.083 g/cm <sup>3</sup>             |
| Reference viscosity, $\mu_0$               | 1.84 mPa·s                          |
| Gas temperature, $T_G$                     | 333 K                               |
| Reference temperature, $T_0$               | 333 K                               |
| Cooling temperature, $T_c$                 | 300 K                               |
| Maximum plunger velocity, $U$              | 3.3 m/s                             |
| Cavitation pressure, $p_{cav}$             | 0.1 MPa                             |

**Table 3**  
Physical parameters of the sealing pair.

| Parameters                                               | Value and unit                           |
|----------------------------------------------------------|------------------------------------------|
| Thermal conductivity of oil, $k_l$                       | 0.123 W·m <sup>-1</sup> K <sup>-1</sup>  |
| Thermal conductivity of oil-gas mixture, $k_G$           | 0.0136 W·m <sup>-1</sup> K <sup>-1</sup> |
| Effective thermal conductivity of boundary $S_S$ , $k_E$ | 3.99 W·m <sup>-1</sup> K <sup>-1</sup>   |
| Thermal conductivity of seal, $k_S$                      | 400 W·m <sup>-1</sup> K <sup>-1</sup>    |
| Specific heat of oil, $c$                                | 2000 J·kg <sup>-1</sup> K <sup>-1</sup>  |
| Convection heat transfer coefficient of gas, $\alpha_S$  | 4 W·m <sup>-2</sup> K <sup>-1</sup>      |
| Thermal contact conductance coefficient, $h_S$           | 15000 W·m <sup>-1</sup> K <sup>-1</sup>  |

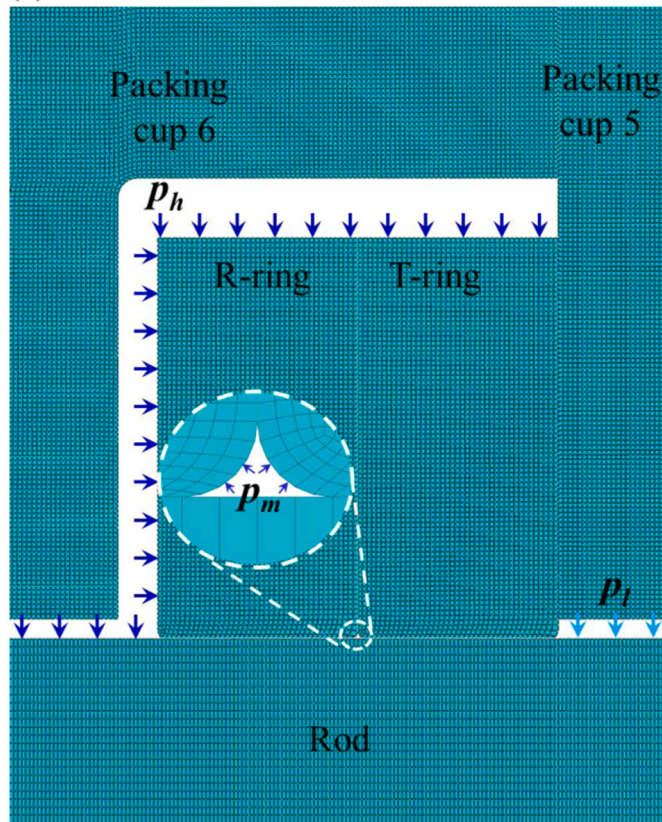
nine sets of packing cups, incorporating lubrication and cooling systems, and filled with packings. The seals in packing cup 6 operate under the most severe conditions, particularly experiencing the highest pressure at the T-ring. The lubrication performance of the seal is the focus of the present work. The T-ring consists of six identical lobes, and preload springs surround them to ensure intimate contact between the packing and the plunger surface, even with lower pressure. The packings are typically manufactured from wear-resistant high-Lead bronze. In application, ethylene streams from the cylinder and guiding bush side through the gap of the plunger and packing cup 6 into the sealing chamber and then passes through the clearances between the segments of the rings. Thereafter, it is fed into the chamfered region separating adjacent rings, and the clearance at the packing cup 5, where the pressure is reduced, continues to decrease as it propagates toward the crankshaft. As the final sealing assembly, the stuffing box lowers ethylene gas concentration to levels permitted for atmospheric release. In practical applications, a cooling system is embedded within the packing cups to regulate the environmental temperature as seals work.

Fig. 2 presents the local structure and mixed lubrication characteristics within the first T-ring sealing interfaces located between packing cups 5 and 6. In addition to direct solid contacts, an oil-filled region and a cavitation region containing a gas-oil mixture where the load-carrying effect is insignificant should be considered. High pressure,  $p_h$ , acts on the

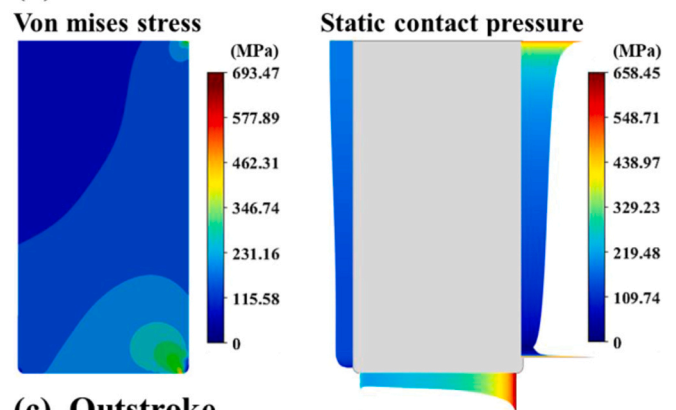
**Table 4**  
Parameters used for FEA.

| Elements    | Elastic modulus (GPa) | Poisson's ratio |
|-------------|-----------------------|-----------------|
| Rod         | 540                   | 0.3             |
| T-ring      | 125                   | 0.3             |
| Packing cup | 208                   | 0.29            |

**(a) FEA model**



**(b) Instroke**



**(c) Outstroke**

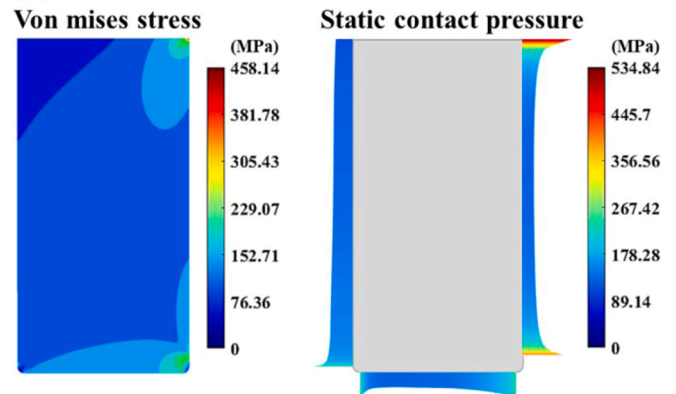


**Fig. 5.** (a) FEA model, (b) static contact pressure, instroke (CA=90°), (c) static contact pressure, outstroke (CA=270°).

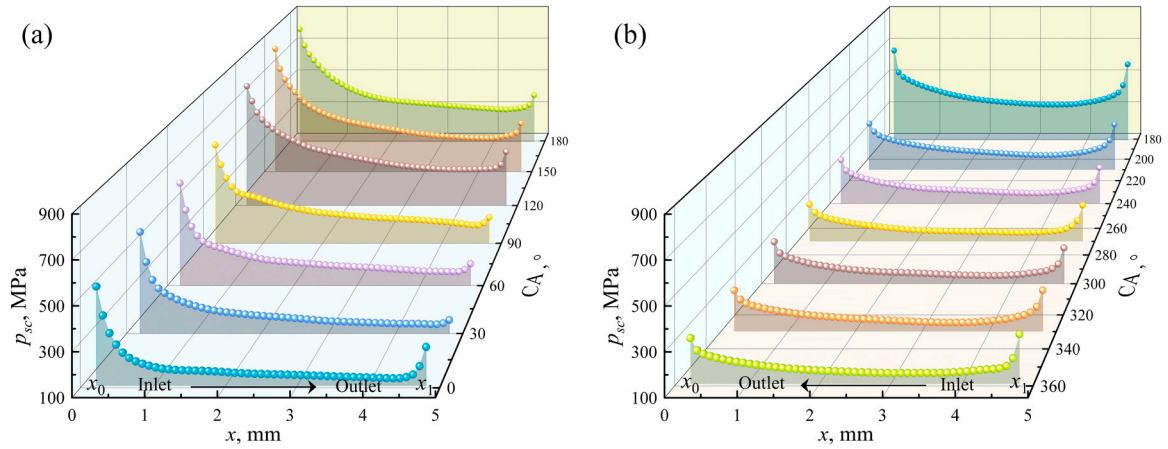


Fig. 6. Static contact pressure in the contact area at different crank angles. (a) Instroke, (b) outstroke.

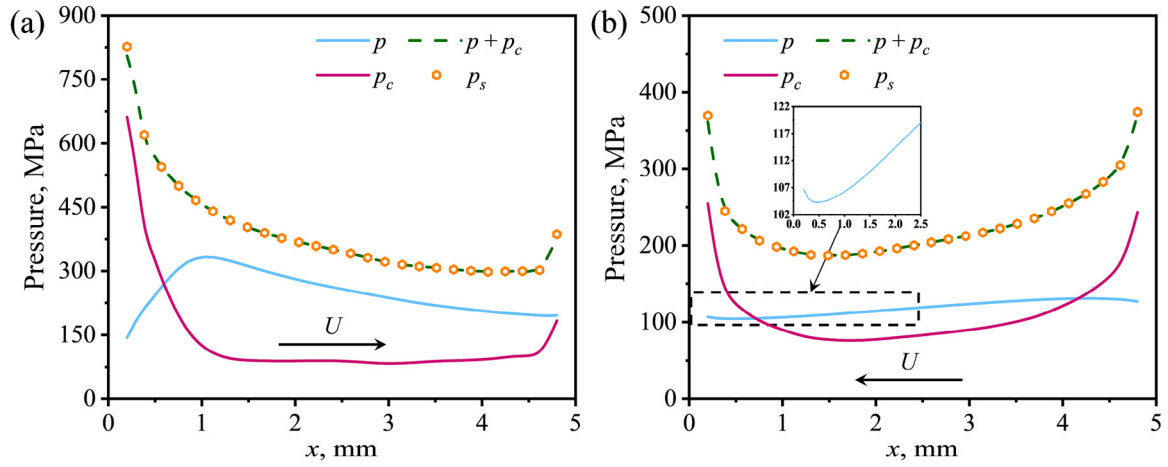


Fig. 7. Comparison between static contact pressure and calculated results. (a) CA= 90°, (b) CA= 270°.

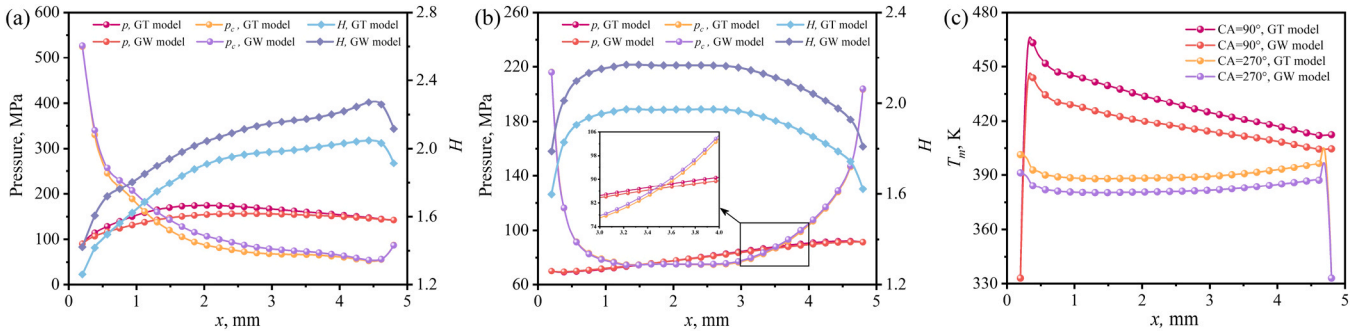


Fig. 8. Comparison of the results obtained using the GT and GW rough contact models. (a) pressure and film ratio, CA = 90°, (b) pressure and film ratio, CA = 270°, (c) average oil film temperature.

lubricating film. Assuming the isotropic fluid film pressure out of the contact region, the boundary oil pressure at origin and another side are taken as  $p_l$  and  $p_m$  respectively. The spring preload applied to the T-ring ranges from 30 to 80 kPa, which is negligible relative to the minimum packing back pressure, approximately 110 MPa. Therefore, the springs are omitted from the finite element analysis (FEA) and lubrication calculations. The T-ring is also assumed to be axisymmetric to reduce model dimensionality. Both measures are intended to enhance computational efficiency.

Table 1 presents the variations in plunger velocity and packing pressure throughout the entire reciprocating cycle. The pressures on

both sides of the packing are taken from the monograph [26], while the pressure is taken as the average of  $p_h$  and  $p_l$ . The plunger velocity,  $U$ , follows a sinusoidal profile, with a maximum value of 3.3 m/s. Across the instroke, the sealing pressure on the cylinder-side increases with crank angle (CA) and reaches nearly 250 MPa over the range of 117°–180°. Its value declines rapidly, reaching a minimum of about 110 MPa at a crank angle of approximately 275°. The value of  $p_l$  remains more stable and consistently lower than  $p_h$ , thereby preventing failure induced by slippage of the packing rings over reciprocating motion.

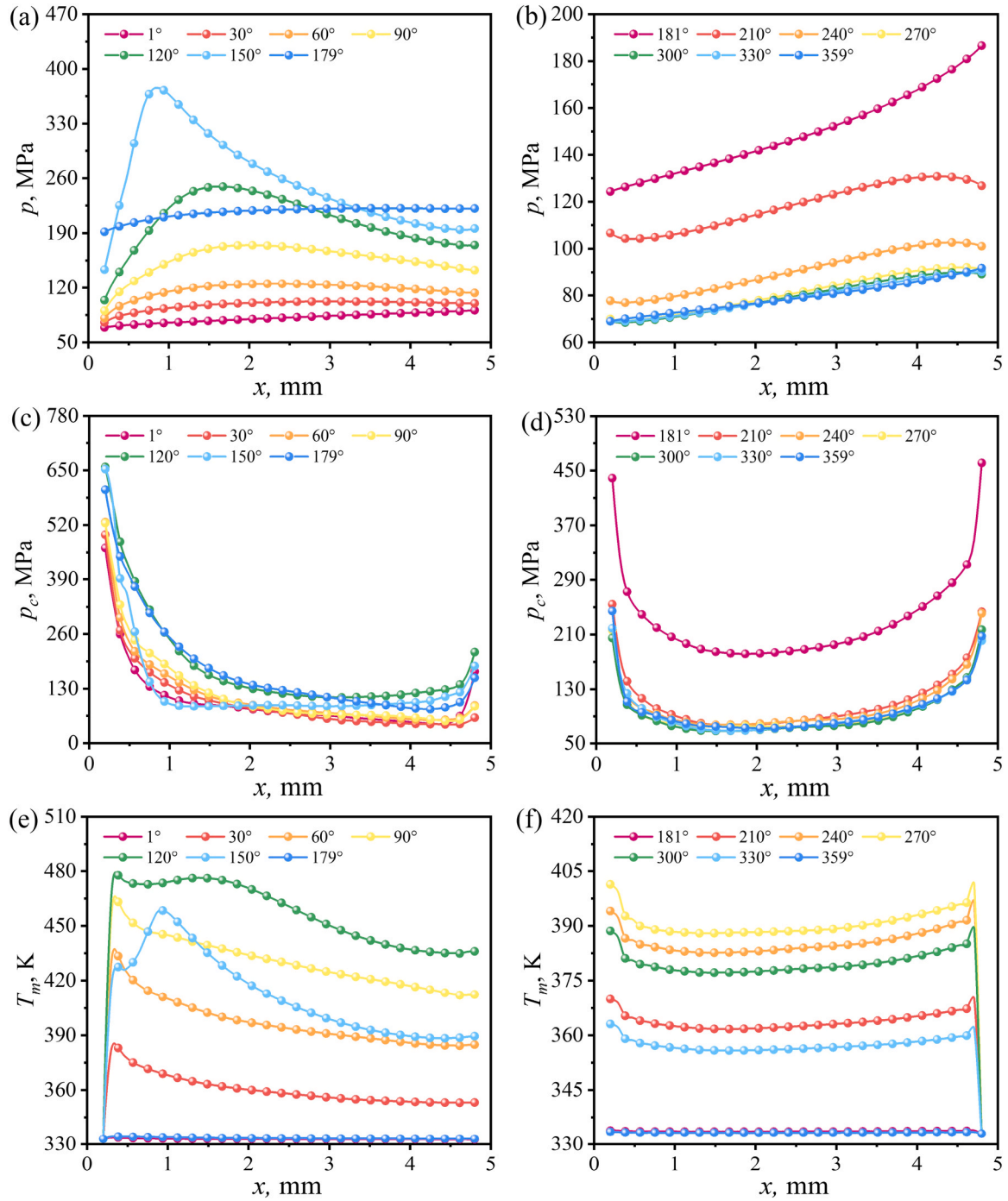


**Fig. 9.** Lubrication parameters in the contact area at different crank angles. (a) Fluid pressure, instroke, (b) fluid pressure, outstroke, (c) asperity contact pressure, instroke, (d) asperity contact pressure, outstroke, (e) average fluid temperature, instroke, (f) average fluid temperature, outstroke.

## 2.2. Fluid mechanics

The current model disregards the minor gaps between packing segments following installation and represents the packing ring as an axisymmetric structure. This approach is consistent with the axisymmetric approximation frequently used in numerical tribological analyses of engine piston ring-liner metal pairs [27–30]. A one-dimensional Reynolds equation is applied to examine oil film pressure and temperature distributions along the axial direction of the friction pair.

$$\frac{\partial}{\partial x} \left( \frac{\phi_x h^3}{12\mu} \frac{\partial p}{\partial x} \right) - U \frac{\partial}{\partial x} \left( \frac{\theta h_r}{2} \right) - \frac{1}{2} \sigma U \frac{\partial (\theta \phi_s)}{\partial x} = 0 \quad (1)$$

where the conditions for identifying the fully flooded and cavitation regions within the contact zone are defined as follows:

$$\begin{cases} p > p_{cav}, \theta = 1 & \text{in the fluid zone} \\ p = p_{cav}, 0 < \theta < 1 & \text{in the cavitation zone} \end{cases} \quad (2)$$

In the equations above,  $\phi_x$  and  $\phi_s$  denote the flow factors used to account for surface roughness effects.  $\theta = \rho/\rho_L$  is the density ratio. Assuming that the surface asperity heights follow a Gaussian distribution, the mean truncated film thickness is described in Eq. (3).

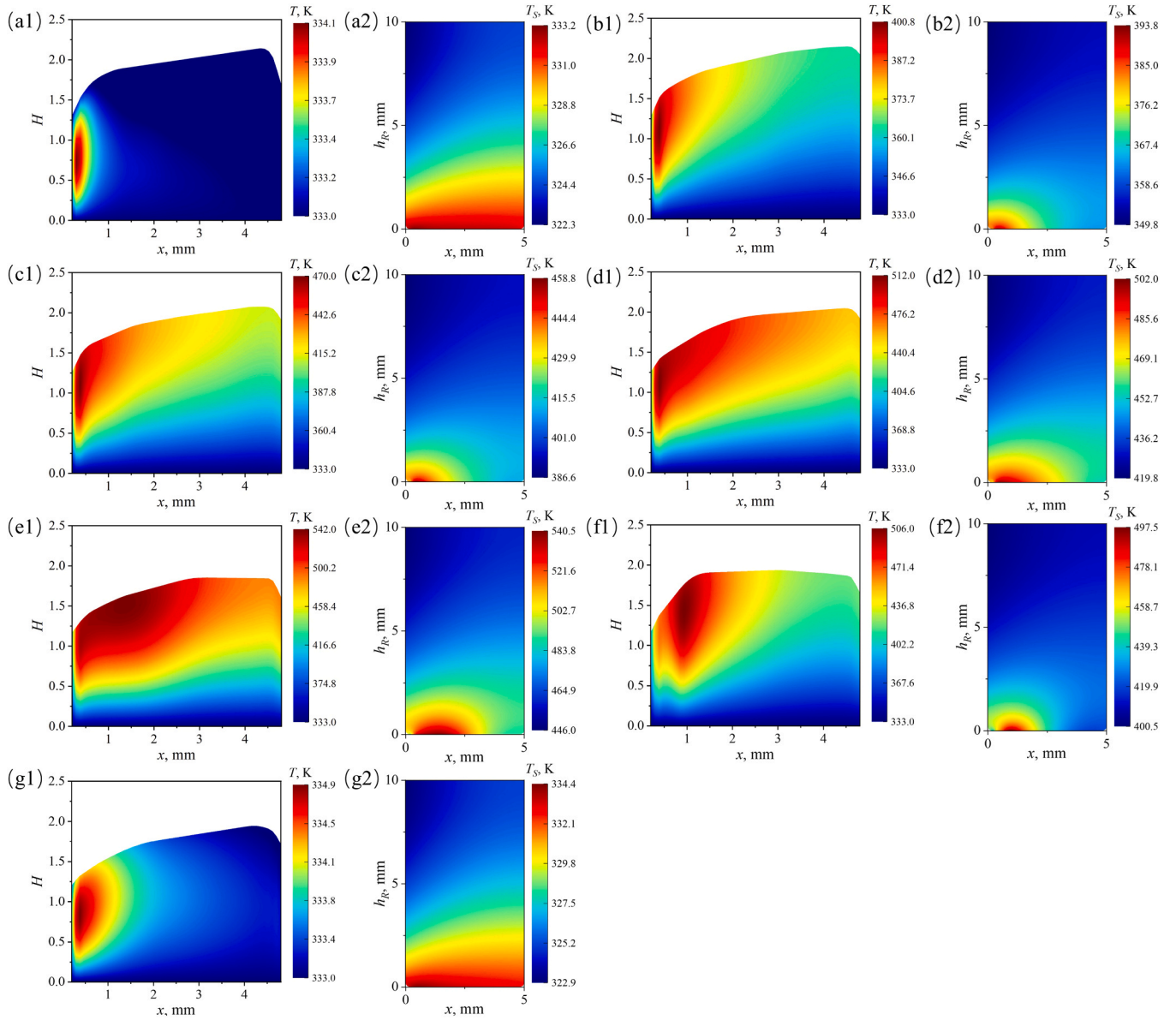


**Fig. 10.** Temperature distribution in the fluid oil film over the instroke (a1-g1) and temperature distribution of the T-ring (a2-g2). (a) CA= 1°, (b) CA= 30°, (c) CA= 60°, (d) CA= 90°, (e) CA= 120°, (f) CA= 150°, (g) CA= 179°.

$$h_T = \frac{h}{2} + \frac{h}{2} \operatorname{erf}\left(\frac{h}{\sqrt{2}}\right) + \frac{1}{\sqrt{2\pi}} \exp\left(-\frac{h^2}{2}\right) \quad (3)$$

In TMEHL analysis, lubricant viscosity and density are strongly influenced by both pressure and temperature and therefore play a crucial role in determining lubrication performance. The pressure and temperature dependency of the hydraulic oil properties are described by the following constitutive relations:

$$\mu = \mu_0 \theta \exp\left\{\left(\ln \mu_0 + 9.67\right) \times \left[-1 + \left(1 + 5.1 \times 10^{-9} p\right)^{z_0} \left(\frac{T - 138}{T_0 - 138}\right)^{-s_0}\right]\right\} \quad (4)$$

$$z_0 = \alpha / [5.1 \times 10^{-9} (\ln \mu_0 + 9.67)] \quad (5)$$

$$s_0 = \beta (T_0 - 138) / (\ln \mu_0 + 9.67) \quad (6)$$

$$\rho = \rho_0 \left[1 + \frac{6 \times 10^{-9} p}{1 + 1.7 \times 10^{-9} p} - 7 \times 10^{-4} (T - T_0)\right] \quad (7)$$

where  $\mu_0$  is the reference viscosity of the lubricant,  $T_0$  is the reference temperature,  $\alpha$  and  $\beta$  are the viscosity-pressure coefficient and the viscosity-temperature coefficient, respectively.  $\rho_0$  is the fluid density at the reference temperature. The fluid film temperature is denoted as  $T$ .

### 2.3. Asperity contact mechanics

Since the conventional GW contact model is more suitable for evaluating the contact behavior between rubber seals and metallic plungers, it is replaced in the present study by the GT model to characterize the asperity contact pressure generated by the interaction of two rough metallic surfaces.

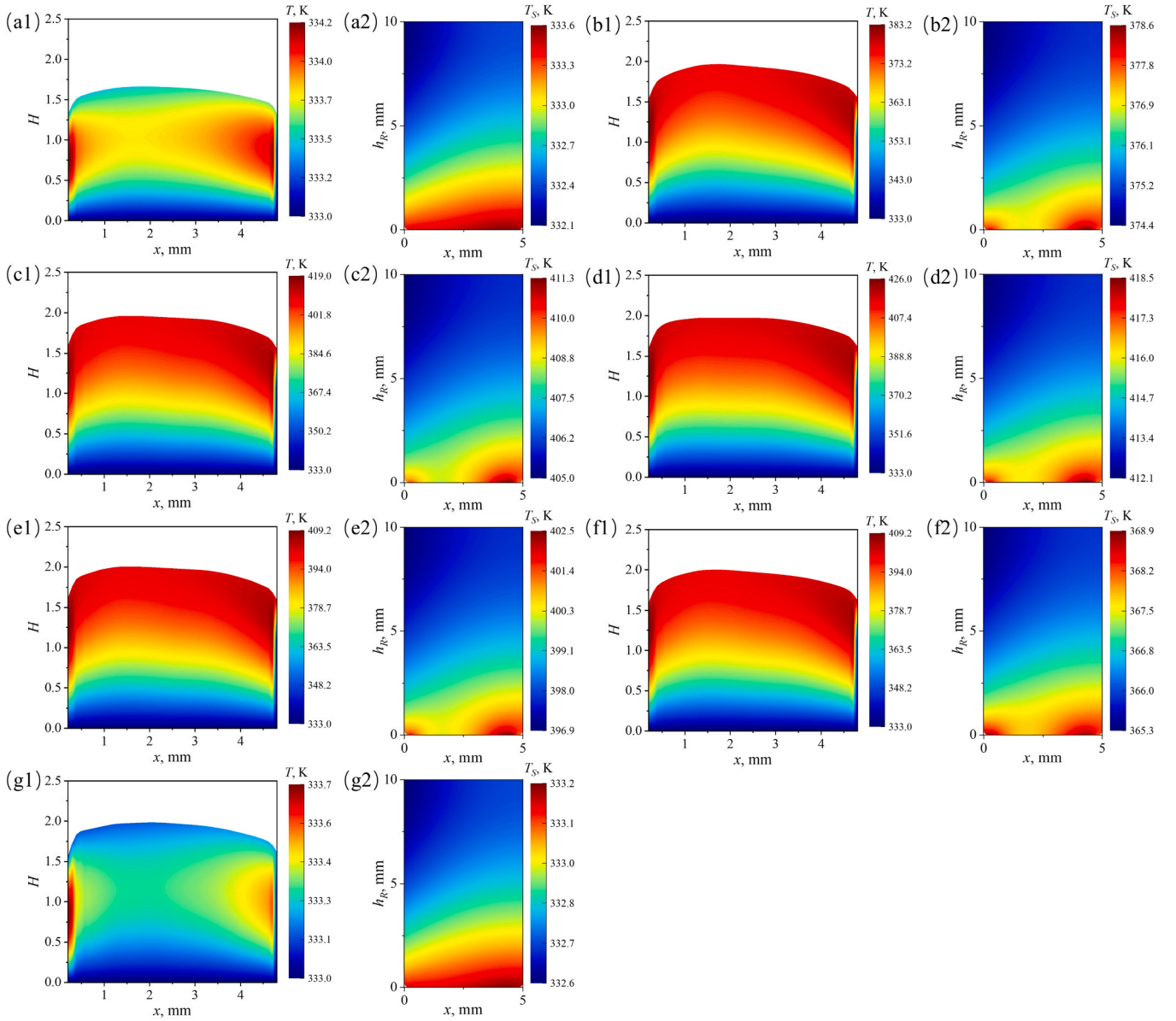


Fig. 11. Temperature distribution in the fluid oil film over the outstroke. (a1-g1) and temperature distribution of the T-ring (a2-g2). (a) CA= 181°, (b) CA= 210°, (c) CA= 240°, (d) CA= 270°, (e) CA= 300°, (f) CA= 330°, (g) CA= 359°.

$$p_c = \frac{16\sqrt{2}}{15} \pi E' \eta_1^2 \beta_1^{\frac{3}{2}} \sigma^{\frac{5}{2}} \int_H^\infty \frac{1}{\sqrt{2\pi}} e^{-\frac{z^2}{2}} (Z-H)^{\frac{5}{2}} dZ \quad (8)$$

where  $E'$  is the equivalent elastic modulus, while  $\eta_1$  and  $\beta_1$  denote the asperity density and asperity height, respectively. The static contact pressure  $p_s$  can be determined using offline finite element analysis software. Substituting  $p_s$  for  $p_c$  in Eq. (8) allows for the calculation of the integral term. Subsequently, the initial film thickness distribution is evaluated using the following fitted expression.

$$h_{sc} = \sigma \left[ -1.87425 + (2.06315 - 5.73933 \log_{10} I)^{\frac{1}{2}} \right] \quad (9)$$

Once the pressure and film thickness distributions have been determined using the iterative procedure, the leakage rate and friction force, which account for both viscous shear and asperity contact shear contributions, can be computed as in Eqs. (10–11).

$$q = \pi D \left( -\phi_x \frac{h^3}{12\mu} \frac{dp}{dx} + \frac{U}{2} \theta h_r + \frac{U}{2} \hat{\sigma} \theta \phi_s \right) \quad (10)$$

$$F_f = \pi D \int_0^L \left( \frac{-\mu U}{h} (\phi_f - \phi_s) - \phi_{fp} \frac{h}{2} \frac{dp}{dx} - \frac{fp_c U}{|U|} \right) dx \quad (11)$$

where the flow factors  $\phi_x$ ,  $\phi_f$ ,  $\phi_s$ ,  $\phi_{fp}$  and  $\phi_s$  are adopted from the previous research [26].

#### 2.4. Deformation mechanics

To address the non-convergence and numerical instability of the influence coefficient method under extreme pressure conditions, as reported in previous studies [31,32], the present work adopts an elastic deformation formulation based on line contact theory. For a contact region discretized into  $n$  nodes, the elastic deformation at the  $i$ -th node is expressed as:

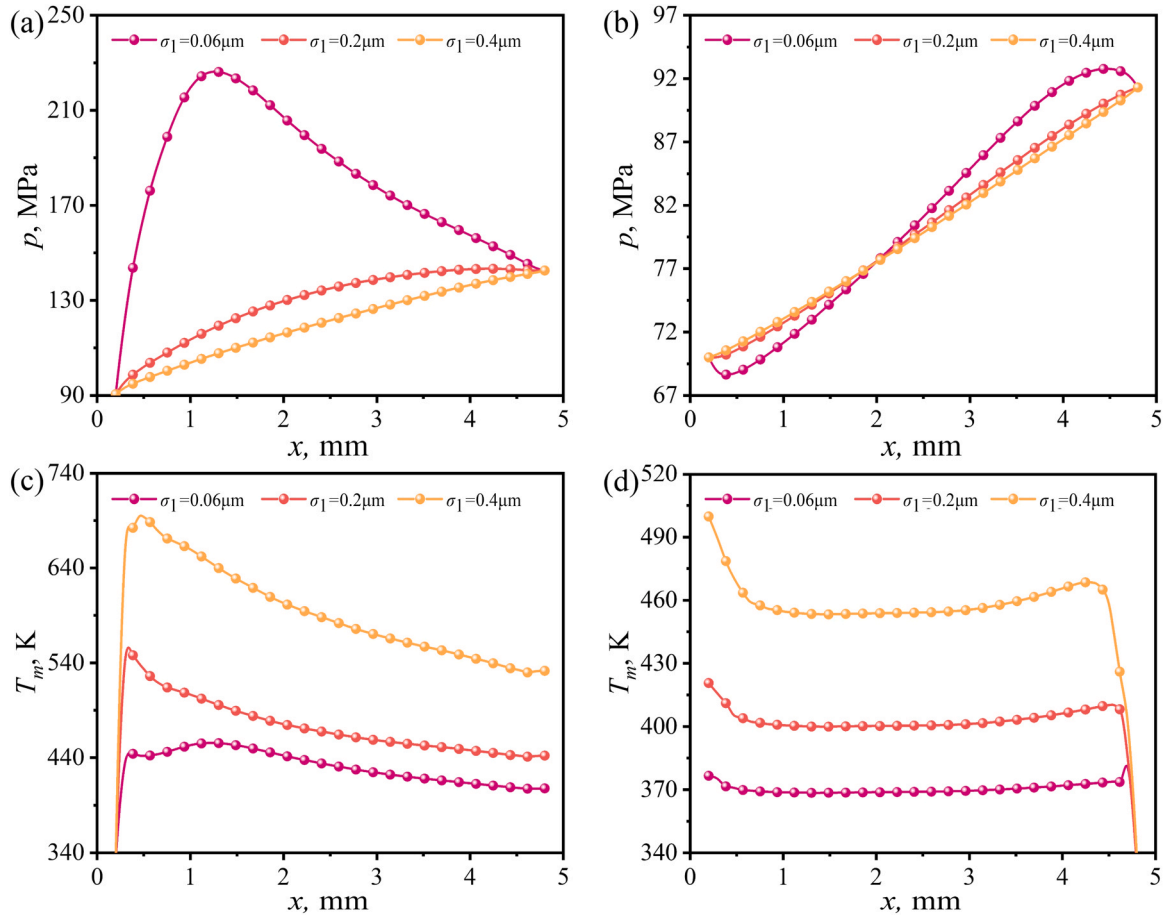


**Fig. 12.** Effect of T-ring roughness on lubrication parameters. (a) Fluid pressure, CA= 90°, (b) fluid pressure, CA= 270°, (c) average temperature, CA= 90°, (d) average temperature, CA= 270°.

$$\nu(x_i) = -\frac{2}{\pi E} \int_{x_i}^{x_i+\Delta x} p(s) \ln(s - x_i)^2 ds \quad (12)$$

By applying a unit pressure, the influence coefficient matrix is constructed from the nodal deformation responses at each location. The total elastic deformation at node  $i$  is subsequently determined as follows:

$$h_i = h_s + \sigma \sum_{k=1}^n (I_1)_{ik} [p_f(X) + p_c(X) - p_{sc}(X)] \quad (13)$$

## 2.5. Thermal mechanics

This mathematical model considers only the frictional heat generated in the mixed lubrication contact region. Some of this heat raises the lubricant film temperature, while the rest transfers to solid components by convection. Heat also dissipates through the seal surfaces and lubricant film boundaries. To determine the temperature field, a modified quasi-three-dimensional energy equation that accounts for surface roughness is used [13] to govern the fluid temperature distribution.

$$\frac{\partial}{\partial x} \left( \frac{khdT_m}{dx} \right) - \rho chu_m \frac{\partial T_m}{\partial x} + \phi_m h + f_c p_c |U| - q_s - q_p = 0 \quad (14)$$

Here,  $q_R$  and  $q_s$  represent the local heat flux transferred into the seal and the plunger, respectively. These terms can be described as follows:

$$\begin{cases} q_p = \frac{k}{h} \left[ 5T_m - \frac{3}{2}T_{SL} - \frac{7}{2}T_{PL} + \frac{h^2}{12k} \left( \frac{\mu U^2}{h^2} + \frac{h^2}{4\mu} \frac{\partial p}{\partial x} \frac{\partial p}{\partial x} - \frac{fp_c |U|}{h} \right) + \frac{h^2}{6k} U \frac{\partial p}{\partial x} \right] \\ q_s = \frac{k}{h} \left[ 5T_m - \frac{7}{2}T_{SL} - \frac{3}{2}T_{PL} - \frac{h^2}{6k} \left( \frac{\mu U^2}{h^2} + \frac{h^2}{4\mu} \frac{\partial p}{\partial x} \frac{\partial p}{\partial x} - \frac{fp_c |U|}{h} \right) - \frac{h^2}{12k} U \frac{\partial p}{\partial x} \right] \end{cases} \quad (15)$$

where  $T_{SL}$  represents the temperature at the interface between the seal and the fluid film, while  $T_{PL}$  denotes the temperature of plunger interface. The following equation calculates the oil film temperature across the thickness.

$$T = az^4 + bz^3 + cz^2 + dz + e \quad (16)$$

$$\begin{cases} a = \frac{1}{h^4} \left[ 5T_m - \frac{5}{2} \left( T_{SL} + T_{PL} \right) - \frac{5h^2}{24k} \left( \frac{2\mu U^2}{h^2} + \frac{h^2}{2\mu} \frac{\partial p}{\partial x} \frac{\partial p}{\partial x} - \frac{2fp_c |U|}{h} \right) \right] \\ b = \frac{1}{h^3} \left[ -10T_m + 5 \left( T_{SL} + T_{PL} \right) + \frac{5h^2}{6k} \left( \frac{\mu U^2}{h^2} + \frac{h^2}{4\mu} \frac{\partial p}{\partial x} \frac{\partial p}{\partial x} - \frac{fp_c |U|}{h} \right) + \frac{h^2}{3k} U \frac{\partial p}{\partial x} \right] \\ c = \frac{1}{h^2} \left[ -\frac{h^2}{2k} \left( \frac{\mu U^2}{h^2} + \frac{h^2}{4\mu} \frac{\partial p}{\partial x} \frac{\partial p}{\partial x} + U \frac{\partial p}{\partial x} - \frac{fp_c |U|}{h} \right) \right] \\ d = \frac{1}{h} \left[ 5T_m - \frac{7}{2}T_{SL} - \frac{3}{2}T_{PL} + \frac{h^2}{12k} \left( \frac{\mu U^2}{h^2} + \frac{h^2}{4\mu} \frac{\partial p}{\partial x} \frac{\partial p}{\partial x} - \frac{fp_c |U|}{h} \right) + \frac{h^2}{6k} U \frac{\partial p}{\partial x} \right] \\ e = T_{PL} \end{cases} \quad (17)$$

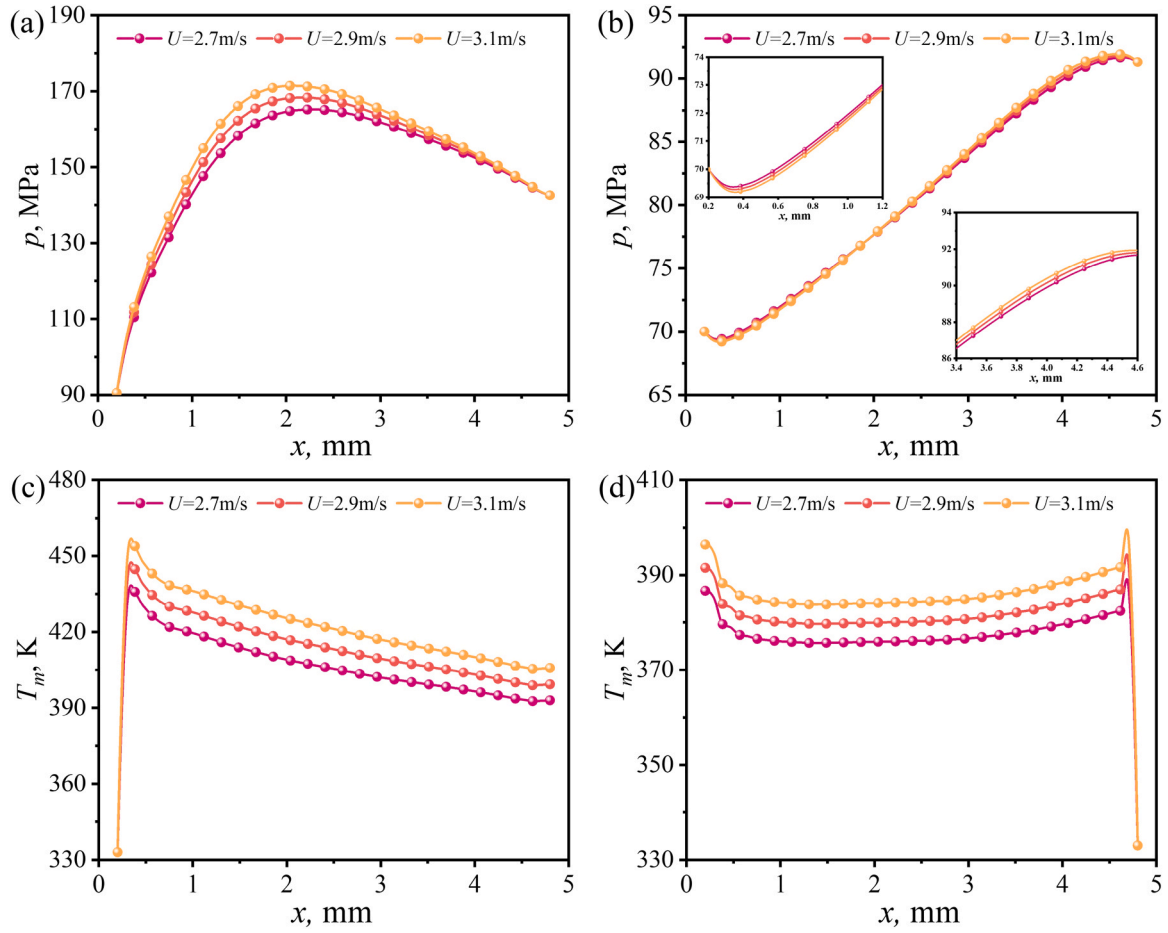


**Fig. 13.** Effect of maximum plunger velocity on lubrication parameters. (a) Fluid pressure,  $CA= 90^\circ$ , (b) fluid pressure,  $CA= 270^\circ$ , (c) average temperature,  $CA= 90^\circ$ , (d) average temperature,  $CA= 270^\circ$ .

where the thermal conductivity of the oil film,  $k$ , is determined by a cavitation index of the oil-gas mixture. It is expressed in Eq. (18). The mean fluid velocity  $u_m$  is defined by Eq. (19).

$$k = k_L \theta + (1 - \theta) k_G \quad (18)$$

$$u_m = \frac{U}{2} - \frac{h^2}{12\mu} \frac{\partial p}{\partial x} \quad (19)$$

The averaged power dissipation density function across the oil film  $\Phi_m$  can be obtained from the formula :

$$\Phi_m = \frac{\mu U^2}{h^2} + \frac{h^2}{12\mu} \frac{\partial}{\partial x} \left( \frac{\partial p}{\partial x} \right) \quad (20)$$

The heat dissipation capability of the seal ring significantly influences lubrication and friction performance in the contact region. Due to its large volume and extended exposure to hot gas, the plunger surface is generally treated as isothermal during reciprocating motion. In contrast, metallic seals have relatively small volumes, rendering their temperature distribution a critical factor for the service life of the lubrication system. Consequently, the temperature field within the seal is determined by:

$$\frac{\partial}{\partial x} \left( k_s \frac{\partial T_s}{\partial x} \right) + \frac{\partial}{\partial z} \left( k_s \frac{\partial T_s}{\partial z} \right) = 0 \quad (21)$$

## 2.6. Boundary conditions

The implementation of appropriate boundary conditions is critical to achieving reasonable simulation results. For the applied conditions

examined in this study, the velocity and pressure boundary conditions for the Reynolds equation are

$$\begin{cases} u(z=0) = U, \text{ on the plunger surface} \\ u(z=h) = 0, \text{ on the packing surface} \\ p(x=x_0) = p_m, \text{ crank gear side} \\ p(x=x_1) = p_h, \text{ cylinder side} \end{cases} \quad (22)$$

The specification of thermal boundary conditions at the lubricant film inlet and outlet is critical, since the plunger's reciprocating motion involves periodic velocity reversal. The inlet temperature is imposed using a Dirichlet boundary condition, whereas the outlet temperature is governed by a Neumann boundary condition:

$$\begin{cases} T(x=0) = T_G & \text{if } q(x=0) > 0 \\ \frac{\partial T}{\partial x} \frac{\partial T}{\partial x=0} = 0 & \text{if } q(x=0) < 0 \\ T(x=L) = T_G & \text{if } q(x=L) < 0 \\ \frac{\partial T}{\partial x} \frac{\partial T}{\partial x=L} = 0 & \text{if } q(x=L) > 0 \end{cases} \quad (23)$$

Fig. 3 presents the thermal conditions implemented for both the seal and lubricant domains. Boundaries  $S_1$ ,  $S_2$ ,  $S_4$  and  $S_6$  are primarily surrounded with gas. Thus, the temperatures are assumed to be uniform across these locations, and convective heat transfer boundary conditions are applied. As the packing cup incorporates a cooling system, boundary  $S_3$  is identified as thermal contact conductivity. For boundary  $S_5$ , the present model does not explicitly solve the temperature distribution within the R-ring. Instead, the heat transfer process is represented as a

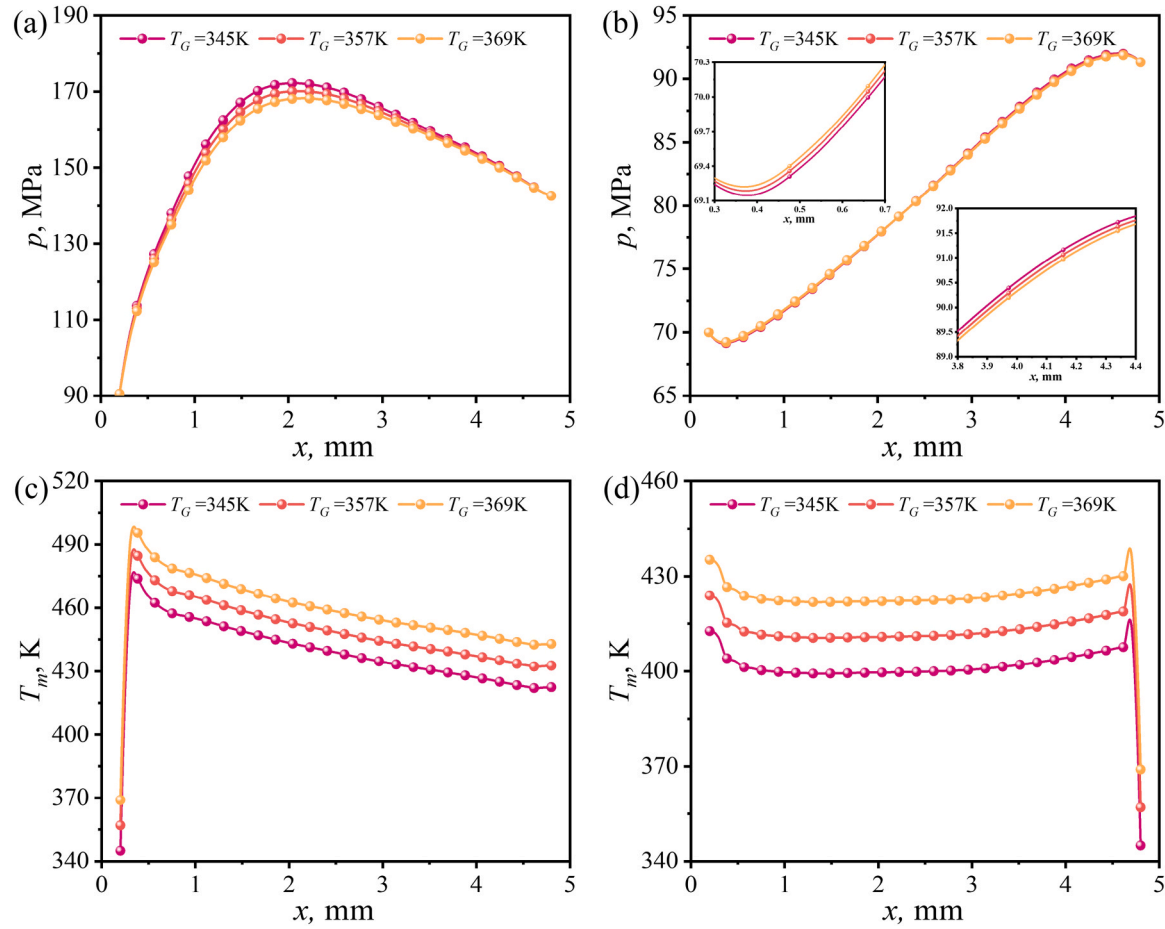


**Fig. 14.** Effect of gas temperature on lubrication parameters. (a) Fluid pressure,  $CA = 90^\circ$ , (b) fluid pressure,  $CA = 270^\circ$ , (c) average temperature,  $CA = 90^\circ$ , (d) average temperature,  $CA = 270^\circ$ .

thermal resistance between the T-ring and the gas via the R-ring. Both packing rings are composed of the same material and thus are assumed to be in perfect contact at the interface, where the temperature is treated as continuous. Consequently, the R-ring is modeled as a plane wall, and the equivalent convective heat transfer coefficient at this interface for the T-ring is given as  $\alpha_E = (1/\alpha_S + L/k)^{-1}$ , where  $L$  denotes the width of the R-ring and  $k$  represents its thermal conductivity.

$$\begin{cases} -k_{p,s} \frac{\partial T}{\partial n_{Z=0,1}} & = q_{p,s} \\ -k_S \frac{\partial T_S}{\partial n_{S_1, S_2, S_4, S_6}} & = \alpha_S(T_S - T_G) \\ -k_S \frac{\partial T_S}{\partial n_{S_3}} & = h_S(T_S - T_{CO}) \\ -k_S \frac{\partial T_S}{\partial n_{S_5}} & = \alpha_E(T_S - T_G) \end{cases} \quad (24)$$

### 3. Numerical procedure

The Reynolds equation, as presented in Eq. (1), is solved iteratively using the finite difference method. To mitigate the decrease in computational efficiency resulting from multiple nested iterative procedures, the SUPG finite element method is utilized to simultaneously solve the large linear equation system associated with the averaged energy equation and the heat conduction equation by means of the unsymmetrical multifrontal method. This approach yields the temperature distributions within both the fluid film and the sealing ring.

$$\begin{bmatrix} K^{LL} & K^{LS} \\ K^{SL} & K^{SS} \end{bmatrix} \begin{bmatrix} T_m \\ T_S \end{bmatrix} = \begin{bmatrix} F^L \\ F^S \end{bmatrix} \quad (25)$$

The terms in the temperature stiffness matrix and the load vector above are expressed as follows:

$$\begin{cases} K_{ij}^{SS} = \int_{\Omega} \nabla N_j \cdot k_S \nabla N_i d\Omega + \int_{S_1, S_2, S_4, S_6} N_i \cdot \alpha_S N_j dS + \int_{S_3} N_i \cdot h_S N_j + \int_{S_5} N_i \cdot \alpha_E N_j - \int_{S_L} \frac{7k}{2h} N_i N_j dS \\ K_{ij}^{SL} = \int_{S_L} \frac{5k}{h} N_i N_j d\Omega \\ F_i^S = \int_{S_1, S_2, S_4, S_6} T_{\infty} \cdot \alpha_S N_i dS + \int_{S_3} T_C \cdot h_S N_i + \int_{S_5} T_{\infty} \cdot \alpha_E N_i + \int_{S_L} \left[ h \left( -\frac{f_c p_{asp} |u|}{12h} - \frac{\Phi(h)}{8} + \frac{\Phi(0)}{24} \right) N_i + \frac{3k}{2h} T_R N_i \right] d\Omega \end{cases} \quad (26)$$

$$\begin{cases} K_{ij}^{LL} = \int_{\Omega_L} \left( \nabla N_j k h \nabla N_i + \rho c h N_i \nabla N_j + \frac{10k}{h} N_i N_j \right) d\Omega + \int_{\Omega_L} \tau^L u_m \nabla N_i \left[ \left( -\nabla (k h \nabla N_j) + \rho c h u_m \nabla N_j + \frac{10k}{h} N_j \right) \right] d\Omega \\ K_{ij}^{LS} = - \int_{\Omega_L} \frac{5k}{h} \left( N_i N_j + \tau^L u_m \nabla N_i N_j \right) d\Omega \\ F_i^L = - \int_{\Omega_L} \left[ h \left( \Phi_m + \frac{5f_c p_{asp} |u|}{6h} - \frac{\Phi(h) + \Phi(0)}{12} \right) + \frac{5k}{h} T_R \right] [N_i + \tau^L u_m \nabla N_i] d\Omega \end{cases} \quad (27)$$

$$\tau^L = \frac{h^{Triangle}}{2|u|} \quad (28)$$

Here,  $N$  is the interpolating function.  $h^{Triangle}$  stands for the characteristic length. The averaged power dissipation density functions at the upper and lower boundaries of the fluid film are

$$\Phi(0) = \frac{\mu u^2}{h^2} + \frac{h^2}{4\mu} \nabla p \cdot \nabla p + u \nabla p \quad (29)$$

$$\Phi(h) = \frac{\mu u^2}{h^2} + \frac{h^2}{4\mu} \nabla p \cdot \nabla p - u \nabla p \quad (30)$$

Material parameters and operating conditions in the mathematical model are strongly coupled and determine the lubrication performance of the sealing system. Fig. 4 illustrates the overall computational procedure, and the detailed iterative solution steps are detailed below:

- (1) Input the geometric configuration and material properties of the sealing system, and a finite element model of the plunger-packing friction pair is established in Abaqus software. Simulations are conducted under working conditions at various crank angles to obtain the static contact pressure distribution at the sealing interface. The initial film thickness is then determined according to Eq. (9).
- (2) In solving the Reynolds equation iteratively base on the current film thickness and temperature field, the fluid pressure is initially calculated. If  $p < p_{cav}$ , the density ratio  $\theta$  is updated according to the method proposed by Fesanghary [33]. If the error is greater than  $\varepsilon_1$ , the procedure is repeated until convergence is achieved. Model parameters are listed in Table 2.
- (3) Under identical operating conditions, the GT contact model was solved to determine the asperity contact pressure,  $p_c$ , followed by evaluation of the load balance. If the load error was greater than  $\varepsilon_2$ , the calculation advanced to the subsequent step.
- (4) Micro deformation analysis is performed to correct the local gap in the contact region. The film thickness  $h$  is updated until the load balance convergence criterion is satisfied.
- (5) The averaged energy equation and heat conduction equation are solved in a coupled manner to obtain temperature distributions in both the fluid film and sealing components. Lubricant viscosity and density are subsequently updated based on the temperature

field. The solution then returns to Step (2) for TMEHL coupling calculation until the temperature convergence criterion is satisfied. Once both the pressure and temperature fields converged, auxiliary lubrication characteristics, such as the leakage rate  $q$  and friction force  $F_f$ , are calculated. Thermal parameters of the lubricant and the T-ring are given in Table 3.

In the numerical implementation, relaxation factor of 0.3 was used in order to steadily solve fluid pressure, film thickness and temperature. Convergence tolerances for the  $\varepsilon_1$ ,  $\varepsilon_2$  and  $\varepsilon_3$  are set to  $10^{-6}$ ,  $10^{-6}$  and  $10^{-3}$ , respectively. Along the sliding direction, the contact region is discretized into 100 uniformly spaced grid points, and along the film thickness direction, into 50 grid points, with the coordinate system defined as in Fig. 3.

### 4. Results and discussion

Fig. 5(a) presents the FEA model of the sealing pair, with material properties detailed in Table 4. The gas pressure,  $p_h$ , is applied to the

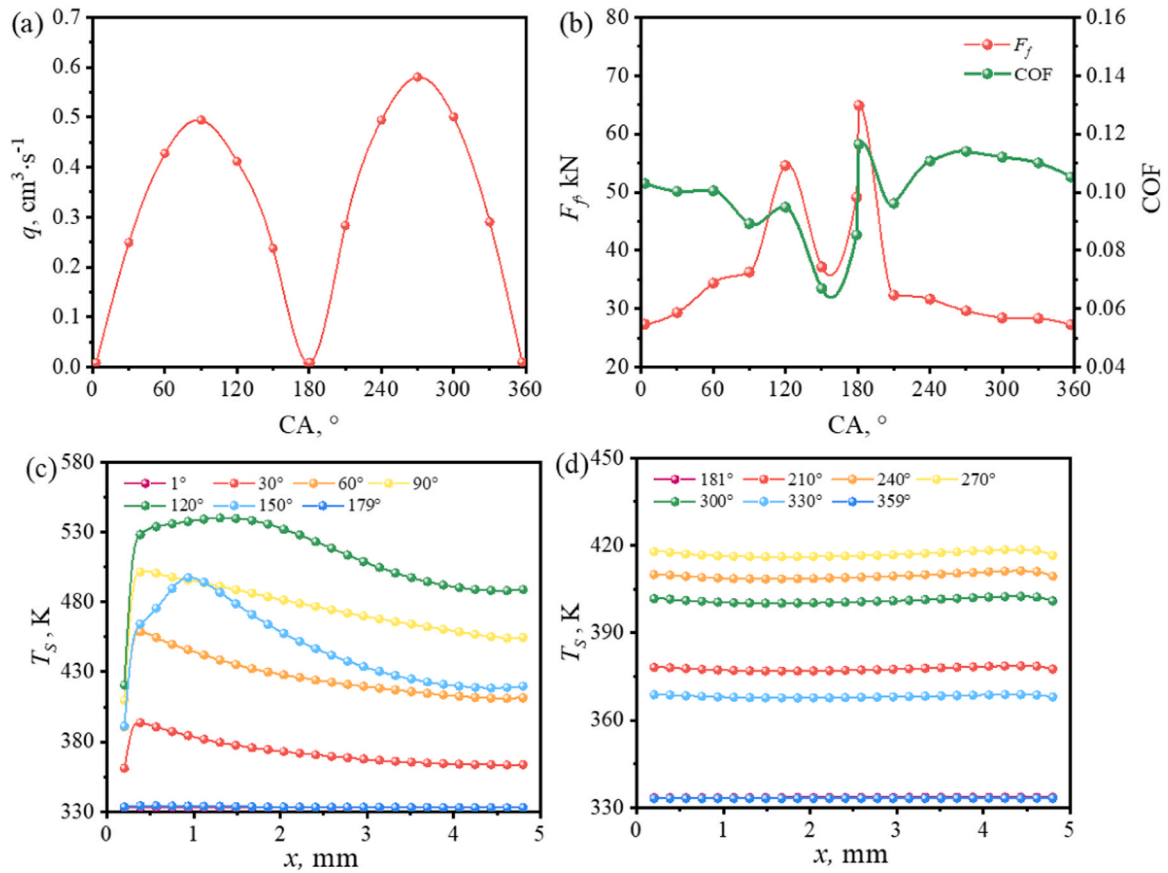


**Fig. 15.** Lubrication parameters in the contact area at different crank angles. (a) Leakage rate, (b) friction coefficient, (c) contact surface temperature of T-ring,  $CA = 90^\circ$ , (d) contact surface temperature of T-ring,  $CA = 270^\circ$ .

surfaces of the plunger and packing cup 6, while  $p_m$  is exerted on the chamfered regions between the packing rings. The low pressure,  $p_l$  acts on the clearance surface between the T-ring and packing cup 5. Table 1 lists gas pressure values at different locations, with the highest reaching 260 MPa at crank angles between  $120^\circ$  and  $180^\circ$ . The analysis also adopts the assumptions of axisymmetric seal and neglects the influence of the preload spring. All contact pairs use hard contact in the normal direction and penalty contact in the tangential direction. Static contact pressure is calculated using the implicit dynamic solver. Simulation cases were established for selected crank angles throughout the entire stroke, based on the realistic pressure and crank rotational speed.

Figs. 5(b) and 5(c) illustrate the von Mises stress contours within the T-ring and the static contact pressure distributions on the boundary surface at crank angles of  $90^\circ$  and  $270^\circ$ , respectively. The operating parameters and the simulated static contact pressures from the model served as inputs to the numerical model.

Fig. 6 presents the contact stress distributions at various crank angles. In tribological analyses, the contact zone is typically divided into inlet and outlet zones based on the contact center, with these zones interchanging as the motion reverses. At  $CA = 1^\circ$  in Fig. 6(a), the inlet boundary of the lubricating oil film is positioned near the origin of the  $x$  axis. The maximum contact pressure occurs at the inlet boundary and gradually declines toward the outlet. A distinct pressure peak can also be observed at the outlet boundary, though its magnitude is lower than that at the inlet. As the crank rotates, the static contact pressure rises remarkably due to increased packing back pressure, reaching a maximum at approximately  $CA = 120^\circ$ . Thereafter, despite a slowdown in plunger velocity, the variation in contact pressure remains minimal. As illustrated in Fig. 6(b) at  $CA = 181^\circ$ , the inlet boundary is located at

$x_1$ . The pressure peak at this position exceeds that observed in the instroke range, while the pressure at  $x_0$  goes down, resulting in comparable pressures at both locations. The average pressure in the outlet zone surpasses that in the inlet zone. Prior to  $CA = 240^\circ$ , the static contact pressure declines rapidly due to reduced packing back pressure, after which pressure variation becomes negligible.

To assess the consistency between the calculated results and the static contact pressure, comparative calculations were performed for operating conditions at crank angles of  $150^\circ$  and  $240^\circ$ , as illustrated in Fig. 7. The absolute position of the plunger remains identical at these two crank angles, enabling simultaneous consideration of both high-back-pressure and low-back-pressure gas environments. Fig. 7(a) shows that the pressure at the fluid inlet boundary is slightly lower than at the outlet boundary. Under these conditions, the plunger velocity is relatively high, and a pronounced fluid pressure peak develops in the inlet region, producing a strong hydrodynamic effect. The asperity contact pressure is primarily concentrated near the boundary regions and remains nearly uniform across most of the contact area. Fig. 7(b) indicates that the fluid inlet pressure exceeds the outlet pressure, resulting in only a relatively weak fluid pressure peak in the inlet region. Notably, at certain locations in the outlet region, the fluid pressure drops below the outlet boundary pressure. Under atmospheric pressure, cavitation is likely to occur in these regions, whereas in the compressor, the fluid pressure remains significantly greater than the air separation pressure or the saturated vapor pressure, thereby inhibiting cavitation.

In Fig. 8 the results obtained from the GW and GT contact models [34] are compared in various crank angles. For the same condition, the GW model yields a greater asperity contact force but a lower fluid pressure, as shown in Fig. 8(a) and 8(b). Also, the GW model calculates a

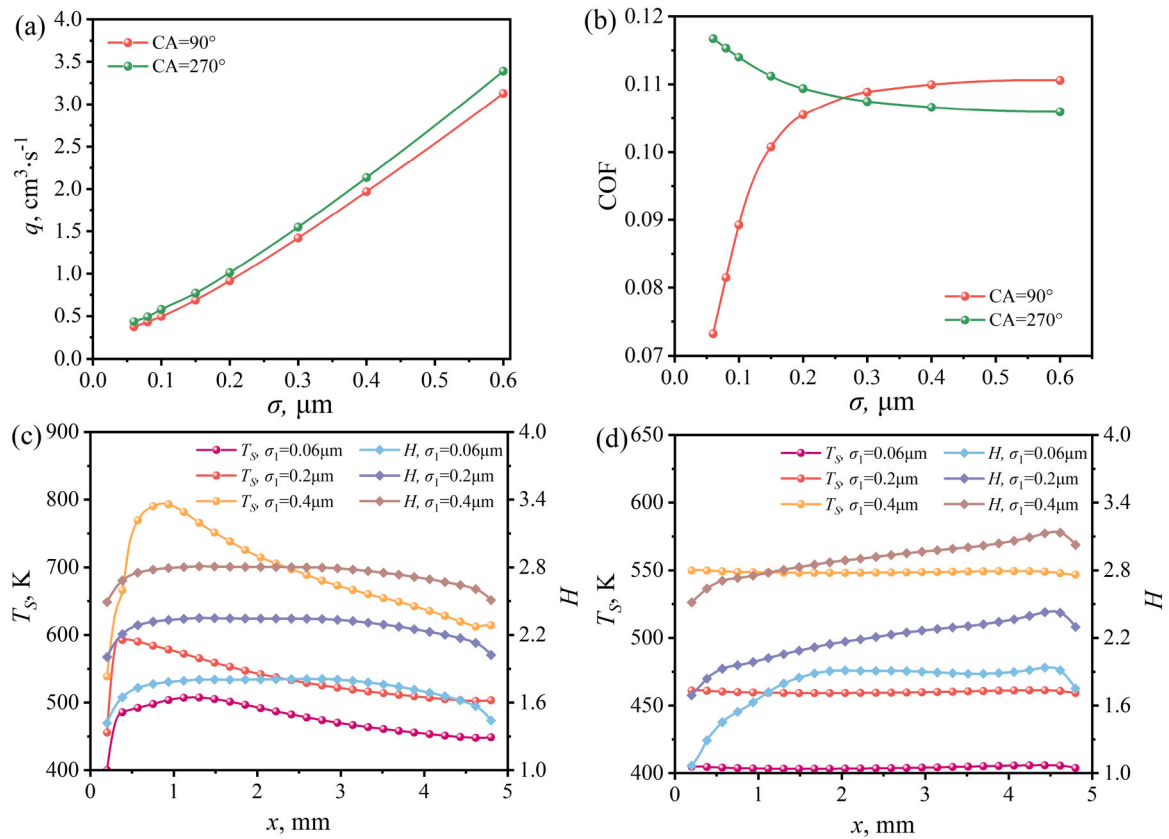


**Fig. 16.** Effect of T-ring roughness on lubrication parameters. (a) Leakage rate, (b) friction coefficient, (c) contact surface temperature of T-ring, CA= 90°, (d) contact surface temperature of T-ring, CA= 270°.

higher film thickness ratio. However, since the  $\sigma$  is different, the GT model is expected to predict a greater actual film thickness.

Fig. 8(c) demonstrates that the GW model predicts a lower average oil-film temperature. Typically, the GW model generally produces a higher asperity load than the GT model, which would result in greater frictional heating and a higher temperature rise. The heat-generation term,  $\Phi_m h + f_c p_c |U|$ , in Eq. (14) does increase when the GW model is applied. However, detailed analysis of the calculation results indicates that changes in film thickness and fluid pressure cause a more significant increase in heat transfer,  $q_s + q_p$ , from the oil film to the seal and plunger. As a result, the net heat retained in the oil film decreases, leading to a lower temperature. This temperature reduction results from the combined effects of changes in heat generation and heat dissipation when comparing the two contact models. When a single contact model is used, increased heat generation typically caused by a varying condition usually leads to a higher average oil temperature. Therefore, the lower temperature predicted by the GW model in Fig. 8(c) should be interpreted as a consequence of model comparison, rather than a general outcome of increased heat generation.

For the instroke, despite the inlet pressure typically being lower than the outlet pressure, the lubrication interface demonstrates a significant hydrodynamic effect. As illustrated in Fig. 9(a), prior to the crank angle reaching 90°, fluid pressure rises with improving plunger velocity and boundary pressure. Even as plunger velocity subsequently declines, fluid pressure continues to build primarily due to concurrent elevations in static contact pressure and boundary pressure. This enhances the fluid load-carrying capacity, resulting in a peak fluid pressure at approximately CA= 150°. As the crank angle nears 179°, static contact pressure exhibits minimal change. Although inlet pressure continues to rise, the hydrodynamic effect diminishes because of the marked reduction in plunger velocity, leading to a decrease in fluid pressure. During the outstroke, as depicted in Fig. 9(b), the gas pressure declines rapidly

within the crank-angle range of 181°–300° (see Table 1), resulting in reductions in both static contact pressure and boundary pressure. Consequently, fluid pressure diminishes substantially, although some hydrodynamic effect persists under relatively rapid-speed conditions. Subsequently, differences in fluid pressure across various crank angles gradually disappear, and the pressure distribution follows an approximately linear trend as velocity slows. Fig. 9(c) demonstrates that under instroke conditions, asperity contact pressure generally aggravates alongside static contact pressure, reaching a maximum near CA= 120°. In contrast, as shown in Fig. 9(d), asperity contact pressure under outstroke conditions remains considerably lower. Elevated contact pressure is observed only near the end of the compression stroke, while differences in contact pressure at other positions are relatively minor. Fig. 9(e) illustrates the distribution of average oil-film temperature under instroke conditions. The highest temperature occurs in the inlet region, primarily due to elevated static contact pressure that generates increased frictional heat. Overall, temperature rises with increasing plunger velocity, reaching a maximum at CA= 120°. At this crank angle, a temperature spike is observed at the inlet boundary, accompanied by a local peak within the inlet region induced by the fluid pressure peak. At CA= 150°, the temperature at the inlet boundary falls below the internal temperature peak, rendering the inlet-boundary temperature spike less prominent. Under outstroke conditions, as shown in Fig. 9(f), the relatively weak hydrodynamic effect causes temperature peaks to concentrate near the boundaries of the fluid region. The overall temperature distribution patterns at different crank angles remain similar. At constant velocity, higher static contact pressure leads to greater temperature rises.

Fig. 10 provides a detailed distribution of fluid film thickness and temperature fields within both the fluid film and the tangential ring. At the initial stage of crank rotation, as depicted in Fig. 10(a1), the temperature promotion in the fluid remains minimal and is primarily

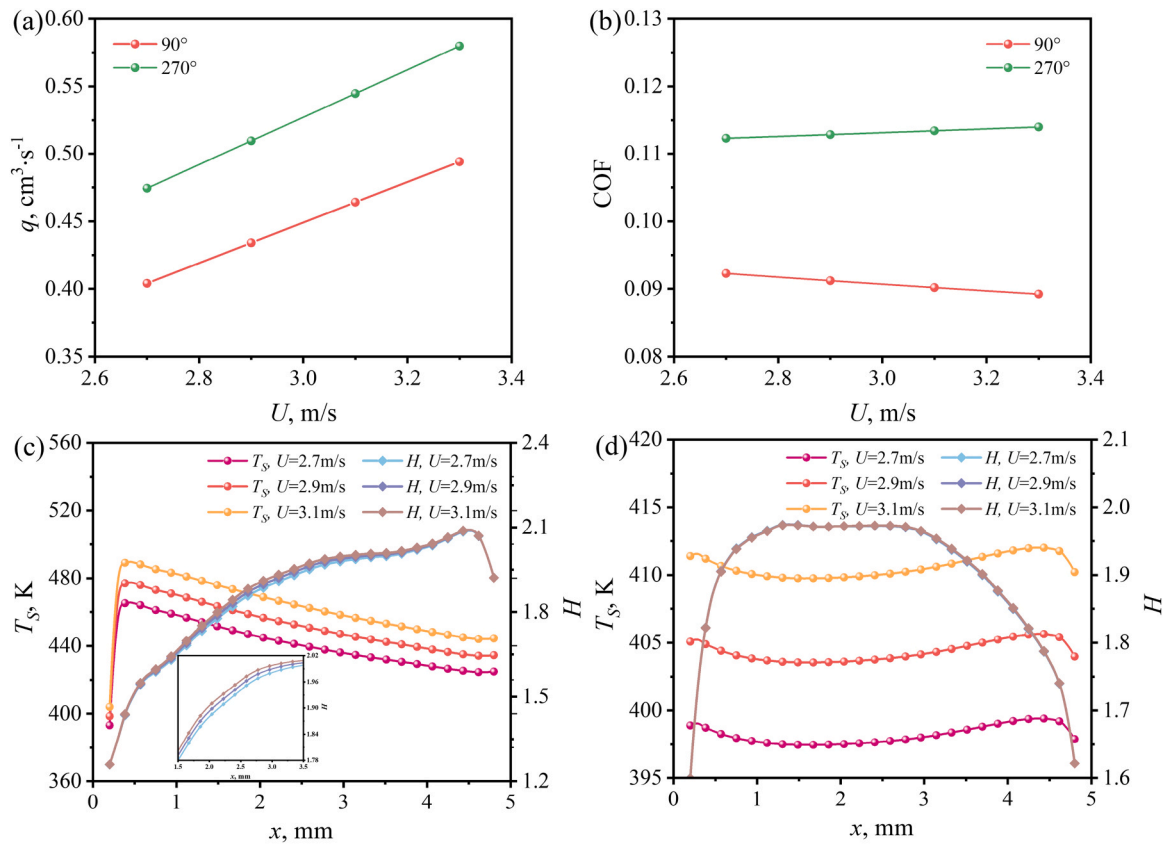


Fig. 17. Effect of maximum velocity on lubrication parameters. (a) Leakage rate, (b) friction coefficient, (c) contact surface temperature of T-ring,  $CA = 90^\circ$ , (d) contact surface temperature of T-ring,  $CA = 270^\circ$ .

concentrated near the pressure-peak region in the inlet zone. The maximum temperature rise within the fluid film occurs inside the fluid region rather than at the contact surface with the sealing ring, while the minimum film thickness is observed at the inlet boundary. For the seal, the highest temperature is found at the inlet boundary of the contact surface. As velocity and pressure increase, the temperature concentration becomes more pronounced, as shown in Fig. 10(b2) to 10(f2). The temperature declines progressively along the radial direction of the T-ring, beginning at the contact surface. Due to the cooling system within the packing cup, the surface in contact with it exhibits the lowest temperature. Notably, Fig. 10(a1) to 10(f1) demonstrate that the maximum temperature within the fluid film gradually shifts closer to the T-ring surface as rotational speed and static contact pressure increase. A comparison of Figs. 10(g) and 10(a) reveals that higher contact pressure results in elevated temperatures in both the fluid film and the sealing ring.

The heat generation characteristics over the outstroke differ significantly from those over the instroke. Fig. 9(d) shows that asperity contact pressure over the outstroke is approximately symmetric about the center of the contact zone, with slightly higher average pressure in the inlet region and a higher peak pressure at the outlet boundary. As a result, Fig. 11(a1)–11(f1) indicate that high-temperature oil covers a larger area in the inlet region of the fluid film, while the maximum temperature occurs at the outlet boundary. Fig. 11(b2)–11(f2) show two temperature peaks on the T-ring contact surface, corresponding to high-temperature regions within the contact zone, with temperature gradually decreasing along the ring-height direction. Throughout the outstroke, the minimum film thickness is observed near the end of compression, whereas the average film thickness remains greater than that observed during the instroke.

Figs. 10 and 11 demonstrate that, although the high-Lead bronze seal exhibits superior heat dissipation compared to the tungsten-carbide plunger, the temperature increase near the lubricant interface remains higher on the sealing component. This occurs because, even though the sealing  $S_3$  boundary is connected to the cooled packing cup, the seal remains stationary relative to the contact region and is subjected to continuous contact, resulting in heat accumulation near the boundary. In contrast, the plunger, due to its greater thermal mass and dynamic surface contact, dissipates heat more effectively and maintains a lower interface temperature. Additionally, across various crank angles, the temperature at the interface between the T-ring and R-ring consistently exceeds that near the packing-cup boundary.

The manufacturing process of the T-ring involves precise control of surface roughness. To assess the impact of surface roughness on lubrication performance, calculations were performed for various roughness values within the tolerance range. Figs. 12(a) and 12(b) demonstrate that lower surface roughness significantly enhances the hydrodynamic effect, which improves the fluid pressure and substantially reduces the average temperature of the lubricating film.

In the compressors, the maximum operating velocity of the plunger is typically set between 2.7 and 3.3 m/s. Figs. 13(a) and 13(b) indicate that, for both strokes, increasing velocity marginally enhances the fluid pressure. However, in the outstroke range, more rapid velocity reduces pressure in the outlet region and intensifies the cavitation effect. Under identical loading conditions, higher velocity increases the proportion of hydrodynamic load support but also results in a nearly linear increase in average fluid temperature, as demonstrated in Figs. 13(c) and 13(d). In ultra-high-pressure ethylene compressors, environmentally friendly polyether lubricants with large ethylene solubility are commonly employed. However, the long-chain molecular structures of these

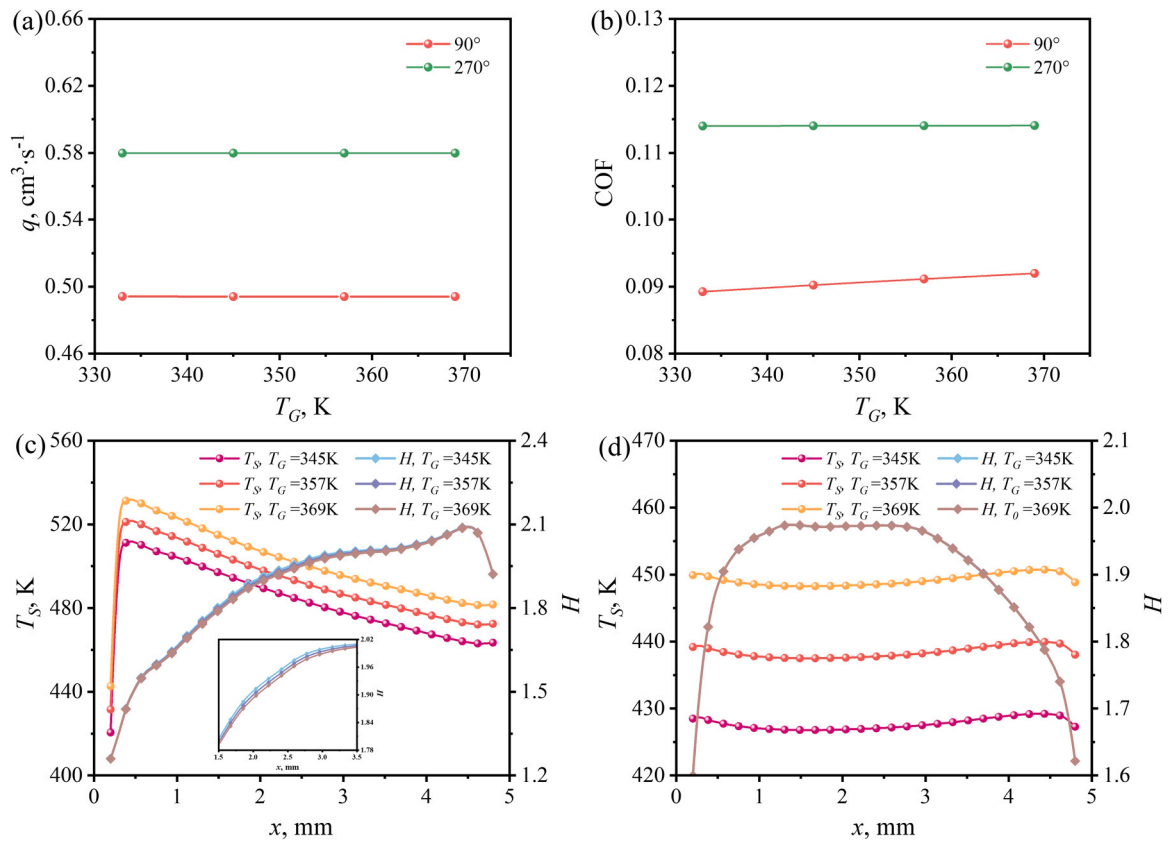


**Fig. 18.** Effect of gas temperature on lubrication parameters. (a) Leakage rate, (b) friction coefficient, (c) contact surface temperature of T-ring,  $CA = 90^\circ$ , (d) contact surface temperature of T-ring,  $CA = 270^\circ$ .

lubricants are susceptible to thermal degradation at elevated temperatures. Consequently, temperature regulation of the lubricating fluid is essential in the lubrication design process.

All over the reciprocating motion of the plunger, the maximum temperature of the ethylene gas ranges from 333 K to 369 K. Fig. 14 presents the effect of elevated gas temperature on lubrication behavior. Figs. 14(a) and 14(b) demonstrate that lower gas temperatures correspond to more notable fluid pressure, owing to the higher fluid viscosity at reduced temperatures. Specifically, Fig. 14(b) reveals that increasing temperature elevates the maximum fluid pressure while decreasing its minimum value slightly, an effect analogous to reducing surface roughness or increasing plunger velocity. Figs. 14(c) and 14(d) indicate that a uniform increase in gas temperature results in an approximately linear growth in the temperature within the fluid film.

In ultra-high-pressure ethylene compressors, ethylene acts as the working medium but does not serve as a lubricant as it does in conventional compressors. The difference eliminates the typical trade-off between maintaining a thick lubricating film and minimizing leakage. In this study, a higher leakage rate corresponds to improved lubrication and heat dissipation. Fig. 15(a) illustrates leakage behavior at different plunger positions under actual operating conditions. The leakage rate increases proportionally with absolute plunger velocity. Throughout the outstroke, plunger movement follows the pressure gradient of ethylene from the high-pressure to the crank side (Fig. 2), resulting in a larger leakage rate at equivalent velocities. The driving power required for the hyper compressors can reach several tens of gigawatts, as the crankshaft must overcome both the resistance of gas compression and the significant friction produced by multiple packing-ring groups within the packing case. Fig. 15(b) shows that the peak friction force of the T-ring in the first packing ring exceeds 60 kN. The friction force rises gradually at the initial stage of crankshaft rotation, while its variation between  $120^\circ$  and  $179^\circ$  becomes more complex. Fig. 6 demonstrates that contact

loads within this range are similar, whereas Figs. 9(a) and 9(c) reveal that a higher asperity load-sharing ratio leads to greater friction force. During the outstroke, both the friction force and the coefficient of friction (COF) generally diminish. However, the friction force at  $181^\circ$  is greater than at  $179^\circ$ . Although the back pressure near the compression end position is similar, the asperity load-sharing ratio during the outstroke remains higher, even though the boundary pressure difference is more favorable for fluid-film formation.

The metal surface temperature, which significantly influences the mechanical properties of materials in service, is analyzed separately. Figs. 15(c) and 15(d) indicate that surface temperatures are higher than the average oil-film temperatures shown in Figs. 9(e) and 9(f). Under instroke conditions, the temperature distribution follows a similar pattern. In contrast, across the outstroke, temperature concentration near the surface boundary is much less pronounced.

Analysis of Figs. 9(c), 10, and 15(b) shows that the minimum film thickness and maximum friction force throughout the instroke occur at a crank angle of  $120^\circ$ , indicating a substantial risk of lubrication failure at this position. The maximum oil temperature is observed at a crank angle of  $150^\circ$ , where the lubricant is most vulnerable to thermal failure and seals experience increased thermal deformation.

Across the entire stroke, increasing surface roughness produces an approximately linear rise in leakage rate, as shown in Fig. 16(a). In contrast, the COF displays distinct trends as illustrated in Fig. 16(b). The indicator rises with roughness within the instroke, while it goes down over the outstroke. Under high-roughness conditions, the friction coefficient in both strokes gradually stabilizes. Figs. 16(c) and 16(d) demonstrate that increasing roughness significantly elevates both the film thickness ratio and the seal surface temperature. According to Eq. (8), under identical load conditions, higher roughness substantially elevates the asperity contact force. Because the maximum oil temperature in the contact zone often coincides with the peak temperature at the

sealing surface, the high-temperature performance of the lubricant requires careful consideration as surface roughness is promoted.

Fig. 17(a) demonstrates a linear relationship between rising velocity and leakage rate. The leakage rate is higher for the duration of the outstroke, corresponding to the pressure difference across the seal. Fig. 17(b) shows that increasing rotation speed affects the COF differently at instroke and outstroke, although both relationships remain linear. Velocity contributes to the rise in fluid pressure at  $CA = 90^\circ$ , which significantly reduces asperity contact force and, in turn, lowers the friction coefficient. In contrast, over the return stroke, fluid pressure at the oil inlet increases only slightly (Fig. 13(b)), resulting in minimal reduction in asperity contact friction coefficient. However, fluid friction builds up more markedly with velocity, causing the total COF to rise. Figs. 17(c) and 17(d) show that adjusting the maximum crankshaft speed has a limited effect on film thickness but a more notable impact on fluid temperature.

Fig. 18(a) indicates that the influence of the maximum temperature within the sealing chamber on leakage rate is almost negligible under actual operating conditions, whereas the friction coefficient increases to some extent, particularly throughout the instroke, as shown in Fig. 18 (b). The temperature rise of the metal surface varies almost proportionally with the ambient temperature. Although elevated temperature slightly reduces the oil-film thickness, this effect can be considered negligible. Overall, the primary consequence of temperature promotion is the enhancement of the T-ring friction force across the instroke.

## 5. Conclusion

A thermo-mixed elastohydrodynamic lubrication model has been developed for the first metal rectangular T-ring operating with extreme ultra-high pressure. The study examines variations in tribological indicators throughout the entire stroke under realistic operating conditions and evaluates the influence of key operating parameters on lubrication performance. The principal conclusions can be summarized:

- (1) The combined influence of pressure, load, and reciprocating velocity causes the seal to operate primarily within the mixed lubrication regime over the entire instroke. Minimum film thickness is observed during the compression stage, whereas the maximum oil-film temperature, reaching 542 K near the outlet region, suggests that the thermal stability of long-chain polyether lubricants requires careful evaluation under ultra-high-pressure conditions.
- (2) Reducing surface roughness significantly elevates fluid pressure and lowers the proportion of asperity load-carrying, which is the primary contributor to temperature rise. While greater maximum velocity enhances fluid pressure, it also exacerbates heat generation. A rise in gas temperature leads to a nearly linear rise in film temperature, whereas its effect on load-carrying capacity is limited.
- (3) Friction and leakage are jointly controlled by pressure gradient and reciprocating motion. The peak friction force is 65 kN during the latter half of the instroke. The leakage rate during the outstroke is approximately 40% higher than during the instroke, as the sliding direction aligns with the pressure gradient.

It should be emphasized that the temperature field predicted by the present model requires a relatively long time to reach steady state, while actual plunger velocity varies continuously and the thermal response is comparatively slow. Consequently, the temperature distribution is inherently more complex in application. Furthermore, thermal deformation of metal sealing components at elevated temperatures can significantly affect performance, as it interacts with elastic deformation to further influence lubrication behavior. These effects should be incorporated in future numerical studies.

## CRedit authorship contribution statement

**Zhihan Fan:** Writing – original draft, Software, Methodology, Conceptualization. **Hongtao Hu:** Visualization. **Xiaoran Qiu:** Software. **Qing Ren:** Supervision, Project administration, Funding acquisition. **Abdykadyr Abidov:** Validation, Formal analysis. **Denis Kapski:** Writing – review & editing, Methodology. **Hao Liu:** Project administration, Conceptualization. **Hao Zhang:** Supervision, Project administration, Formal analysis.

## Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper

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## Data availability

Data will be made available on request.

## References

- [1] Fan Z, Wang H, Hu H, et al. Numerical investigation on lubrication of metal packing using an improved thermo-mixed elastohydrodynamic approach. *Int Commun Heat & Mass Transf* 2026. 042-002.
- [2] Salant R.F., Maser N., Yang B. Numerical model of a reciprocating hydraulic rod seal; proceedings of the Proceedings of IJTC2006 STLE/ASME International Joint Tribology Conference, San Antonio, TX, USA F,2006 [C].
- [3] Ahmad HM, Zhang J, Shirazi S, et al. Experimental and CFD analysis of erosion in plugged tees in series. *Wear* 2025;570:205956.
- [4] Ahmad H.M., Zhang J., Shirazi S.A., et al. CFD Analysis of Erosion in Plugged Tees Series: Effect of Pipe Diameter, Particle Size and Fluid Density; proceedings of the CONFERENCE 2026, F, 2026 [C]. Association for Materials Protection and Performance.
- [5] Dou X, Xiang W, Li B, et al. Design optimization of surface microstructure and guide vane for mitigating pipe erosion via CFD–DPM. *Adv Powder Technol* 2026; 37(7):105303.
- [6] Pinedo B, Hadfield M, Tzanakis I, et al. Thermal analysis and tribological investigation on TPU and NBR elastomers applied to sealing applications. *Tribology Int* 2018;127:24–36.
- [7] Chen S, Guo F, Wang W, et al. An improved mixed lubrication model for revealing the mechanism of high speed and frictional heat on sealing performance. *Tribology Int* 2024;191:109147.
- [8] Xiang C, Guo F, Liu X, et al. Thermo-elastohydrodynamic lubrication simulation of reciprocating rod seals under transient condition. *Tribology Int* 2021;153:106603.
- [9] Wang B, Li X, Peng X, et al. Thermoelastohydrodynamic mixed lubrication of combined rod seals operating at high pressures and speeds: mathematical modeling and numerical analysis. *J Tribology* 2024;146(4).
- [10] Xu M, Wang Y, Yuan G, et al. A novel coupled computational method for thermo-elastohydrodynamic lubrication problem in reciprocating seals. *Int J Heat Mass Transf* 2026;254:127641.
- [11] Yin T, Wei D, Wang T, et al. Mixed-lubrication mechanism considering thermal effect on high-pressure to reciprocating water seal. *Tribology Int* 2022;175: 107856.
- [12] Fang C, Peng Y, Zhou W, et al. Finite element modeling and simulation of mixed elastohydrodynamic lubrication for the finite line contact under tilting loads. *Tribology Int* 2023;186:108660.
- [13] Stefani F, Reborra A. Steadily loaded journal bearings: Quasi-3D mass-energy-conserving analysis. *Tribology Int* 2009;42(3):448–60.
- [14] Djamaï A, Brunetiere N, Tournerie B. Numerical modeling of thermohydrodynamic mechanical face seals. *Tribology Trans* 2010;53(3):414–25.
- [15] Yi X, Xu H, Jin G, et al. Boundary slip and lubrication mechanisms of organic friction modifiers with effect of surface moisture. *Friction* 2024;12(7):1483–98.
- [16] Wei W, Liu H, Guo C, et al. Effect of vibration on reciprocating sealing performance. *Tribology Int* 2023;178:108031.
- [17] Ischinger F, Bartel D, Brunk M, et al. Non-linear model order reduction for elastohydrodynamic lubrication simulations of polymer seals. *Tribology Int* 2019; 140:105885.
- [18] Bauer N, Rambaks A, Müller C, et al. Strategies for implementing the jakobsson-floberg-olsson cavitation model in ehl simulations of translational seals. *Int J Fluid Power* 2021:199–232.

- [19] Zhang H, Liu X, Gong J, et al. Thermohydrodynamic lubrication characteristics of piston rings in diesel engine considering transient heat transfer under the parameterized surface texture of cylinder liners. *Energies* 2023;16(24):7924.
- [20] Jiao B, Ma X, Wang Y, et al. A fluid-structure coupled transient mixed lubrication model for piston ring lubrication property analysis with CMFF method. *Int J Mech Sci* 2023;252:108377.
- [21] Fritz B, Scheichl B. Comprehensive multi-scale cylinder lubrication model for reciprocating piston compressors: From rigid-body dynamics to lubricant-flow simulation. *Tribology Int* 2023;178:108028.
- [22] Gussmagg J, Maier M, Pusterhofer M, et al. EHD simulation study on the influence of measured ICE compression piston ring profiles on the lubrication film formation. *Tribology Int* 2024;199:110015.
- [23] Prati E, Strozzi A. A study of the elastohydrodynamic problem in rectangular elastomeric seals. *J Tribology* 1984.
- [24] Yunxiang M. Research on the sealing performance of seawater hydraulic reciprocating seals under full ocean depth [D]. China; Huazhong University of Science and Technology, 2022.2022.
- [25] Nikas GK, Almond RV, Burridge G. Experimental study of leakage and friction of rectangular, elastomeric hydraulic seals for reciprocating motion from  $-54$  to  $+135$  °C and pressures from 3.4 to 34.5 MPa. *J Tribology Trans* 2014;57(5):846–65.
- [26] Patir N, Cheng H. Application of average flow model to lubrication between rough sliding surfaces. *J Lubr Technol* 1979;101(2):220–9.
- [27] Jeng Y-R. Theoretical analysis of piston-ring lubrication Part I—fully flooded lubrication. *Tribology Trans* 1992;35(4):696–706.
- [28] Xiuyi L, Jiao B, Lu X, et al. A statistical piston ring lubrication model considering the tribofilm and its effect of two-stroke marine engines. *Tribology Int* 2023;177:107996.
- [29] Ma M, Sherrington I, Smith EH, et al. Development of a detailed model for piston-ring lubrication in IC engines with circular and non-circular cylinder bores. *Tribology Int* 1997;30(11):779–88.
- [30] Li T, Ma X, Lu X, et al. Lubrication analysis for the piston ring of a two-stroke marine diesel engine taking account of the oil supply. *Int J Engine Res* 2021;22(3):949–62.
- [31] Du J. Sealing performance analysis of packing based on mixed lubrication theory [D]. Dalian Dalian Univ Technol 2022.
- [32] Li Y, Yu L, Jiang H, et al. Transient sealing characteristics of Glyd-ring in the high water-based piston pair under reciprocating pump conditions. *Tribology Int* 2024;192:109293.
- [33] Fesanghary M, Khonsari M. A modification of the switch function in the Elrod cavitation algorithm. *J Tribology* 2011;133(2).
- [34] Yang B, Salant R. Elastohydrodynamic lubrication simulation of O-ring and U-cup hydraulic seals. *Proc Inst Mech Eng Part J J Eng Tribology* 2011;225(7):603–10.