Investigation on the oil transfer behaviors and the air–oil interfacial flow patterns in a ball bearing under different capillary conditions

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Abstract: Lubricant oil is crucial to the rolling bearings as the main medium of lubricating, cooling, cleaning, and so on. The oil starvation in and around the contacts is harmful to the performance and fatigue life of rolling bearings. Therefore, it is of necessity to understand the behaviors of oil transfer and the patterns of air–oil two-phase flow in bearings, especially with the influence of different capillary properties. This work established a transient air–oil two-phase flow model in a ball bearing based on computational fluid dynamics (CFD). Groups of cases are implemented to investigate the behaviors of oil transfer and air–oil flow under different capillary conditions with speed, surface tension, and viscosity. Flow patterns are classified by the morphological features of the air–oil flow. Staged phenomena are analyzed with flow patterns and reach good agreements with the observations from experiments. It is found that the oil distribution and air–oil flow behaviors in a ball bearing are strongly related to the speed and the ratio of oil viscosity and air–oil surface tension ($\mu_{oil}/\sigma$). The flow maps imply that the levels of capillary number (Ca) may be the boundaries and the critical points of flow pattern transition between the different flow patterns in bearing.

Keywords: flow pattern; oil transfer; computational fluid dynamics (CFD); bearing

1 Introduction

It has been widely accepted that lubricant is considered as a crucial component affecting the performance and fatigue life of rolling bearings. Various failures in rolling bearings have been found associated with lubrication. Oil is usually chosen as the main medium of lubrication, cooling, cleaning, and so on. Moreover, it is the basic ingredient of greases. Therefore, in the past decades, a large number of studies have worked on the properties of oil and their effects on the lubrication performance of rolling bearings.

The early works focused on the lubrication film forming in the contacts of rolling bearing. The classical theory of elastohydrodynamic lubrication (EHL) has been established by Reynolds [1], Grubin [2], Dowson and Hamrock [3–6], et al. Furthermore, the conditions where the oil supply to the contacts is insufficient drew more and more attention. The works in Refs. [7–16] have proved that oil starvation will cause a significant decrease in lubrication film thickness, which is harmful to the contact surfaces and will increase the risk of lubrication failures. Hence, it is necessary to understand the distribution and behaviors of the oil transfer in rolling bearings, which leads to the researches in three aspects as displayed in Fig. 1.

The first aspect is the oil distribution and replenishment around the contact region, as Fig. 1(a) shows. The optical observation was applied to the

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Nomenclature

| Symbol | Description |
|--------|-------------|
| $\phi_{\text{oil}}$ | Oil volume fraction |
| $\rho_{\text{oil}}$ | Density of oil |
| $\mu_{\text{oil}}$ | Viscosity of oil |
| $\phi_{\text{air}}$ | Air volume fraction |
| $\rho_{\text{air}}$ | Density of air |
| $\mu_{\text{air}}$ | Viscosity of air |
| $\rho$ | Density of mixture |
| $\mu$ | Viscosity of mixture |
| $\sigma$ | Surface tension coefficient of air–oil interface |
| $\vec{v}$ | Velocity field |
| $p$ | Pressure |
| $\omega_{\text{in}}$ | Absolute angular speed of inner ring |
| $\omega_{\text{outr}}$ | Relative angular speed of outer ring |
| $\omega_{\text{inr}}$ | Relative angular speed of inner ring |
| $d$ | Diameter of the inner ring |
| $n$ | Rotating speed of inner ring |
| $t$ | Time |
| $U$ | Reference velocity |
| $\text{Ca}$ | Capillary number |
| $\overline{T}$ | Stress tensor |
| $I$ | Unit tensor |
| $f_{\sigma}$ | Capillary force |

ball-on-disc rolling contact apparatus in the 1970s [17, 20]. The oil distribution around the contact region was described as the butterfly oil reservoir in morphology. As the oil layer splits and recovers through the oil reservoir, the behaviors of the oil reservoir will affect the starvation degree in the contact region.

Observations of the oil reservoir have been presented in the past few years [21–24]. As computational fluid dynamics (CFD) simulation has become a strong supplementary to the experiments, various models with STAR-CD, COMSOL, and OpenFOAM of the air–oil flow around the rolling point contact region were established [25–27]. These works have revealed...
that the speed, viscosity, and surface tension and slide-roll ratio will influence the behaviors of the oil reservoir. Dimensionless numbers such as capillary number (Ca) and Weber’s number (We) were found as indicators to predict the behaviors.

The second aspect of studies pays attention to the recovery of the oil layer along the raceways of the bearing, as Fig. 1(b) shows. To simulate the rebounding of the lubricant oil layer behind the outlet, Yin et al. [28] established a free thin layer flow model in 1999. Then Gershuni et al. [29] studied the oil replenishment after the oil layer passed through the contact region. By extending this model, van Zoelen et al. and Venner et al. [18, 30–32] investigated the oil distribution on the free races of bearing under centrifugal effects and provided simulations of pressure-induced lubricating film thickness decay in rolling bearings from 2008 to 2012. However, this 2D model was highly simplified and based on the assumption that the oil layer was thin and stable enough. It cannot be used to deal with the complex 3D air–oil interfacial flow problems, such as the ruptures and deformations of the meniscus and drops, etc.

The third aspect of investigation pays attention to the global oil transfer and distribution in the whole rolling bearing, as Fig. 1(c) shows. CFD simulations and experiments were applied to study the air–oil flow and distribution in rolling bearings. Several two-phase flow simulations with the VOF (volume of fluid) method based on ANSYS FLUENT have been implemented in the past decade. The oil–jet models of a ball bearing were established in Refs. [19, 33, 34] from 2014 to 2016. It was found that the oil distribution is not uniform. The positions of the nozzles and the speed will influence the air–oil flow and the cooling. Moreover, it was proved that higher temperature always appears where the oil is little, which could be coincident with the results of the experiments. Adeniyi et al. [35, 36] simulated the oil drops in the aero-engine bearing. The break-up pattern of oil drops in a bearing sector and between the cage and inner race was analyzed. Yan et al. and Ge et al. [37–41] investigated the airflow pattern in angular ball bearings based on CFD and experiments. The oil jet was optimized in consideration of the second-order vortexes in the bearing. Grooves on the inner ring were designed to improve the performance of oil–air lubrication. Liang et al. [42] observed the oil distribution and migration in a rolling bearing under low rotating speeds. The influences of different shapes and surface properties of the cage were studied and the roles of capillary force were discussed.

In summary of the works above, it could be found that there are considerable relationships between the oil distribution, the lubrication, and the cooling in the rolling bearings. Relevant optimization designs of oil properties and oil supply devices have been done, however, it is still perplexing when, where, and why starved lubrication will happen in the bearings. Therefore, finding quantitative rules and building scientific theories of oil transfer and distribution is of great benefit to the further development of high-performance ball bearings. Furthermore, the effects of the capillary–viscous flow in the bearing have not been clearly revealed and understood, especially when oil properties such as surface tension and viscosity change with the working conditions and temperatures.

As in an experiment the various factors are usually coupled with each other, it is difficult to make clear the roles of each factor. However, the CFD simulations are of advantages to gain the quantitative information about oil transfer and distribution based on single variable analysis.

In this work, a transient air–oil two-phase flow model in a ball bearing is established based on CFD with the VOF method. Groups of simulation cases with fixed oil supply are implemented to investigate the behaviors of oil transfer and the patterns of air–oil interfacial flow. Flow patterns are classified and compared with experimental results. The effects of different capillary conditions with speed, surface tension and viscosity are in discussion.

2 Model and case

2.1 CFD model of ball bearing

A three-dimensional CFD model of a typical rolling bearing is established based on the ball bearing 7008 consisting of 18 balls, as shown in Fig. 2(a). The outer ring is replaced with a steel cylindrical ring to keep the same geometry as the experiment. In consideration
of the periodicity, 1/18 sector of the bearing is taken as the computational domain, which includes the ball, the cage and the rings, as shown in Fig. 2(b). Constant gaps of 10 μm have remained between the surfaces of the ball and rings to allow the oil and air to flow through contact regions.

The rotational speed is applied to the inner ring and the outer ring keeps static. However, as there exist rotational and orbital motions of the balls, a rotational coordinate system tied with the cage and relative velocities is used here, as shown in Fig. 2(b). The surfaces of the ball, the cage and the rings are set as no-slip velocity boundaries, and two sides in the circumferential direction of the computational domain are set as periodic boundaries. Other faces are set as zero-velocity boundaries and the environment pressure is set as one atm.

2.2 Numerical methods

2.2.1 Governing equations

The oil flow in this model should comply with the conservation laws described by the Navier–Stokes equation system. Especially, to model air–oil interface and effect of air–oil surface tension, the VOF method is used here [43]. A volume fraction field of the oil phase is defined to mark the two phases and air–oil interface in the VOF model. The oil volume fraction \( \varphi_{\text{oil}} \) is written as

\[
\begin{align*}
\varphi_{\text{oil}} = 1, & \quad \text{bulk phase of oil} \\
\varphi_{\text{oil}} = 0, & \quad \text{bulk phase of air} \\
0 < \varphi_{\text{oil}} < 1, & \quad \text{mixture and interface}
\end{align*}
\]

And the volume fraction of air is defined as

\[
\varphi_{\text{air}} = 1 - \varphi_{\text{oil}}
\]

Accordingly, the density and viscosity are reformulated as

\[
\begin{align*}
\rho &= \varphi_{\text{oil}} \rho_{\text{oil}} + (1 - \varphi_{\text{oil}}) \rho_{\text{air}} \\
\mu &= \varphi_{\text{oil}} \mu_{\text{oil}} + (1 - \varphi_{\text{oil}}) \mu_{\text{air}}
\end{align*}
\]

The continuity equation is written as

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0
\]

where \( \vec{v} \) is the velocity vector. And the equation of
momentum conservation is expressed as

$$\frac{\partial (\rho \vec{v})}{\partial t} + \nabla \cdot (p \vec{v} \otimes \vec{v} - \vec{T}) = \vec{f}_a$$ \( (5) \)

where \( \vec{f}_a \) is the capillary force defined for air–oil surface tension and \( \vec{T} \) is the stress tensor for Newtonian fluid, which is defined by Stokes as

$$\vec{T} = -\left( p + \frac{2}{3} \mu \nabla \cdot \vec{v} \right) \vec{I} + \mu (\nabla \otimes \vec{v} + (\nabla \otimes \vec{v})^T)$$ \( (6) \)

where \( p \) is the static pressure and \( \vec{I} \) the unit tensor.

The volume fraction field is ruled by the transport equation as

$$\frac{\partial \varphi_{oil}}{\partial t} + \nabla \cdot (\varphi_{oil} \vec{v}) = 0$$ \( (7) \)

Then the capillary force term \( \vec{f}_a \) is defined with the continuum surface force (CSF) model formulated as [44]:

$$\vec{f}_a = -\sigma \left( \nabla \cdot \frac{\nabla \varphi_{oil}}{\nabla \varphi_{oil}} \right) \nabla \varphi_{oil}$$ \( (8) \)

where \( \sigma \) is the surface tension coefficient.

The governing equations include the continuity Eq. (4), the momentum Eq. (5), and the volume fraction transport Eq. (7). Assumptions such as incompressible, laminar and isothermal flow are used here.

2.2.2 Discretization and solver

The finite volume method (FVM) is used to discretize the governing equations. An entire hexahedral and orthogonal body-fitted mesh topology is designed as shown in Fig. 2(c) as the hexahedra elements can lessen the diffusion of the interface. The meshes near the boundary surfaces and around the contact regions are denser than that of other regions, because the oil mainly exists there and its flow behaviors are more complex and important. Based on several tests, the amount of elements is chosen as 364,704 in consideration of both the accuracy and the computational efficiency.

Considering the advantages of parallel computation, ANSYS FLUENT was used in this work [45]. Transient pressure–velocity coupling algorithm of pressure–implicit with splitting of operators (PISO) is adopted to solve the continuity equation and momentum equations iteratively. Second-order schemes are used for the discretization of momentum equations to improve accuracy. Especially, to capture sharp air–oil interface, compressive interface capturing scheme for arbitrary meshes (CICSAM) scheme is used in the discretization of volume fraction transport equation, and the time step is controlled by Courant number.

2.3 Design of cases

The properties of base oil polyalphaolefin (PAO) were used with density \( \rho_{oil} = 837 \text{ kg/m}^3 \), and typical case groups with different ratios of viscosity \( \mu_{oil} \) and air–oil surface tension \( \sigma \) are set as listed in Table 1. The air phase is set as \( \rho_{air} = 1.225 \text{ kg/m}^3, \mu_{air} = 0.018 \text{ mPa·s} \). To eliminate the differences caused by the wettability in this simulation, the oil–solid contact angle is consistently set as 1° everywhere. The initial oil distribution is set as an even oil layer with a thickness of 200 \( \mu \text{m} \) on the inner and outer raceways.

The air–oil interfacial flow in the ball bearing is a complex non-linear problem, which is sensitive to the previous conditions. The transient model is chosen and several speed stages are selected to verify that the phenomena will sustain under constant speeds. The cases are set as two types for acceleration and constant speed. The speed of inner ring \( n \) increases from zero to 1,000 r/min in one second. Then several constant cases are selected from the acceleration as Eq. (9). This could ensure the coherence of the cases because all of the initial conditions in the constant speed cases are provided by the same acceleration case. In order to get stable solutions, the simulations of constant speed cases last for at least an inner ring

| Table 1 | Properties of oil phase in different cases. |
|---------|--------------------------------------------|
| Group number | \( \mu_{oil} \) (Pa·s) | \( \sigma \) (N/m) | \( \mu_{oil}/\sigma \) (s/m) |
| 1        | 0.026                                     | 0.04          | 0.65          |
| 2        | 0.026                                     | 0.03          | 0.87          |
| 3        | 0.026                                     | 0.02          | 1.3           |
| 4        | 0.061                                     | 0.04          | 1.5           |
| 5        | 0.061                                     | 0.03          | 2.03          |
| 6        | 0.061                                     | 0.02          | 3.05          |
| 7        | 0.4                                       | 0.04          | 10            |
cycle, which includes 18 passages of ball through the computational region. The stabilization is shown in Appendix A. Workstations with 60 cores are used here. The simulation times of the cases vary from one week to one month, which depends on the different working conditions.

\[
\begin{cases}
1,000^* t, & 0 \leq t \leq t_c, \\
1,000^* t_c, & t_c \in \{0.05, 0.1, 0.2, 0.4, 0.6, 0.8, 1.0\}
\end{cases}
\]

(9)

3 Results and discussion

3.1 Global behaviors of oil transfer

3.1.1 During acceleration

3D global oil distribution when \( \mu_{\text{oil}} = 0.026 \, \text{Pa} \cdot \text{s} \) and \( \sigma = 0.03 \, \text{N/m} \) during acceleration is presented in Fig. 3 as an example of all of these groups. The oil fraction values equal to and more than 0.5 are marked as the air–oil interface and the oil bulk phase, respectively. The rolling direction, the inlet and outlet of lubricated rolling contact are indicated.

When the ball rolls over the lubricant oil layer, the oil adheres to the surface of the ball, as shown in Figs. 3(a) and 3(d). The oil surrounding both the outer and the inner contacts accumulates and forms oil reservoirs. The air–oil interface ruptures from the outlet of inner contact once the inner raceway starts rotating, and the oil layer on the raceway and the ball splits into two parts. Two oil ridges form along the edges of the inner raceway with an oil-starved wake remaining between them, as shown in Figs. 3(b) and 3(e). Then they flow through the pocket of the cage. As it accelerates, the outer oil reservoir also splits into two parts. Soon the oil adhering on the raceways spreads as oil patches, as shown in Figs. 3(c) and 3(f). At the end of the accelerating case, the oil ridges in the cage pocket also break down and spread as oil patches.

3.1.2 Under different stable speeds

As the description above, the air–oil flow behaviors and oil distribution are related to the acceleration. In order to discuss the effects of different stable speeds, the stable results from the different constant speed cases are in comparison. The cases for \( \mu_{\text{oil}} = 0.026 \, \text{Pa} \cdot \text{s} \) and \( \sigma = 0.03 \, \text{N/m} \) are displayed in Fig. 4 as an example.

The differences between the cases of various speeds are obvious. When \( n = 50 \, \text{r/min} \), the oil layer on the ball and the rings is enclosed and smooth as shown in Fig. 4(a). Nevertheless, the oil layer under other speeds ruptures to different degrees. The oil layer on the inner raceway ruptures when the speed is above 100 r/min as Fig. 4(b) shows. However, the oil layer on the outer ring will not rupture until \( n \) is greater than 200 r/min, as shown in Fig. 4(c). When the speed

![Fig. 3 3D oil distribution during acceleration when \( \mu_{\text{oil}}/\sigma = 0.87 \, \text{s/m} \).](image-url)
is a little higher, the oil ridges on the outer ring migrate to the sidebands and leave a nearly bare region on the ring as Fig. 4(d) shows. Then under 1,000 r/min, as Fig. 4(f) shows, the oil reservoir around the outer contact thoroughly splits, and the oil layer on the abutments of the inner ring starts to wave as oil ridges and rises towards the cage. As the speed increases, the oil ridges on the ball and the raceway tend to be narrow with their oil volume decreasing, while the width between the oil ridges increases, as Figs. 4(d)–4(f) show. Moreover, the wave crests of the oil occur and ridge up towards the inner surface of the cage when speed rises. These suggest that the oil transfer and distribution are directly influenced by the levels of speed.

Moreover, the path lines of air–oil two-phase flow are displayed in Fig. 5. As shown in Fig. 5(a), the oil flow near the rings and the ball follows the velocity
direction of the surfaces. The shape of the path lines around the ball and the contacts could be regarded as a perfect circle, which means that the flow remains laminar and stable in general.

However, there are characteristic vortexes flowing back in both the inlet and the outlet regions. These vortexes may be related to the little oil volume in the middle region of the raceway. Then the oil cumulates towards the two axial side regions of the rings and forms the two oil ridges. The oil adhering on the surface of the ball implements the transfer cycle between the inner ring and the outer ring.

Comparing the positions of oil ridges in Figs. 5(a) and 5(b), it could be found that the path lines around the ball expand into a wider space as the speed increases. And the oil ridges also migrate to the further position towards the side regions of the rings. This kind of oil migration finally results in the disappearance of the two oil ridges on the ball.

### 3.1.3 Under different \( \mu_{oil}/\sigma \) ratios

The effect of the ratio \( \mu_{oil}/\sigma \) on the oil flow of in bearing is considered in this section. In Fig. 6, the cases with different values of \( \mu_{oil}/\sigma \) under 200 r/min are displayed as an example. It can be found that the broken degree of the oil layer increases with the ratio \( \mu_{oil}/\sigma \) under the same speed. On the other hand, if the ratio increases, the critical speeds of the oil layer split will also increase. As shown in Figs. 6(c)–6(f), the oil migrates to the abutment of the inner raceway and

the axial sides of the outer ring with the increase of ratio \( \mu_{oil}/\sigma \), and the oil volume adhering on the ball and the mid area of the raceway decreases.

### 3.1.4 Similarities among cases

The features of air–oil flow are influenced by the levels of both the speed and the ratio \( \mu_{oil}/\sigma \). To evaluate the combined effects, the dimensionless capillary number \( Ca \) is defined as

\[
Ca = \frac{\mu_{oil} U}{\sigma} = \frac{\mu_{oil} \left( \frac{n \pi d}{60} \right)}{\sigma} = \frac{\pi \mu_{oil} (dn)}{60 \sigma}
\]

where \( d \) means the diameter of the inner ring. The reference velocity \( U \) is chosen as the product of \( dn \), which is a critical index for rolling bearings.

As shown in Fig. 7, the features of air–oil flow and distribution for the same \( Ca \) level are similar. The width of the oil-starved area increases and the coverage area of the oil layer on the raceways and the ball decreases with the increase of \( Ca \). Then the oil layer on the inner ring ridges and breaks towards the cage, and the degree of oil ridging and the volume of oil migrating onto the cage also increases with \( Ca \).

The oil migrates into the pocket and the oil layer on the ball interacts with the wall of the pocket and forms oil bridges. Under low \( Ca \) levels, the oil bridges maintain general stability. On the contrary, the oil under high \( Ca \) level would break and spread in the

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**Fig. 6** 3D oil distribution with different properties when \( n = 200 \) r/min.
pocket, and accumulate on the two side gaps between the ball and the pocket. Finally, the oil ridges on the ball would break into oil patches and disappear.

The phenomena above are based on the global morphological features of the oil layer. To get more information on the relationship between the oil distribution and Ca, it is necessary to classify the patterns of air–oil two-phase flow and oil distribution in different regions. Inspired by the previous studies about the flow patterns, a similar analysis method would be proposed below. Although flow patterns occurring around the rolling point contact regions has been described and discussed in previous studies, it is of benefit for the understanding to extend these flow pattern concepts and definitions to a more common situation and practical aspect.

3.2 Analysis of air–oil flow patterns

3.2.1 Around outer contact region

The classification and definition of the flow patterns are based on the detailed morphological features of the oil distribution and air–oil flow. The typical results of air–oil flow around the outer contact region are extracted and displayed in Fig. 8. Features and their evolution processes with the increase of speed or ratio $\mu_{oil}/\sigma$ are presented. The rupture of outlet meniscus, the concave of inlet meniscus, and the breakup of the oil reservoir in the contact region are chosen as the three key features to distinguish the flow patterns, as marked with red star. As a result, it can be classified into four flow patterns, which are named single tail flow, double tail flow, butterfly flow, and tunnel flow here.

The single tail flow means that the meniscus of the oil reservoir is enclosing and convex, and the outlet meniscus elongates and forms the oil tail, as shown in Fig. 8(a). The oil tail would diverge and form two oil ridges on the surfaces of the ring and the ball. The differences between Figs. 8(a1) and 8(a2) imply two conditions since the speed or ratio in Fig. 8(a2) is more than that in Fig. 8(a1). As the speed or the ratio increases, the volume of the oil reservoir would decrease, while the oil tail would become narrower and longer.

Then as the speed or the ratio rising, the outlet meniscus ruptures and the oil tail splits into two parts. If the inlet meniscus remains enclosing and convex, it is named as the double tail flow here, as shown in Fig. 8(b). The two oil tails would diverge and form four oil ridges on both the surfaces of the ring and the ball. In addition, there would be parched areas between the oil ridges and the oil sidebands. Comparing Figs. 8(b2) with 8(b1), it could be found that the oil volume would decrease with the increase of speed and the ratio. Since the oil reservoir deforms and breaks as shown in Fig. 8(b2), it seems that the meniscus is more unstable under conditions with high speed or high ratio.

When the oil volume supplying to the inlet decreases to some extent, the inlet meniscus concaves and the flow pattern changes to a butterfly flow, as
shown in Fig. 8(c). The oil ridges from the outlet of the contact region connect with the frontier of the inlet meniscus, and the oil reservoir is split into two parts. The difference between Figs. 8(c1) and 8(c2) implies that the width of the oil-starved wake behind the outlet would increase with the speed or the ratio. Moreover, the degree of the inlet meniscus concave would increase with the starvation degree of the wake. The oil ridges would tend to migrate to the oil sidebands and join them.

If the oil reservoir around the contact region cannot maintain its integrity, it would break into oil patches. The oil ridges on the ring would thoroughly migrate to the sidebands and disappear, and the oil adhering to the ball would accumulate to the axial ends. There would leave a wide bare or oil-starved area along the rolling contact track, which seems like a tunnel through the contact region. This pattern is named tunnel flow here, as shown in Fig. 8(d). The distance between the two sidebands and the break degree of oil patches would also increase with the speed and the ratio $\mu/\sigma$, as the comparison between Figs. 8(d1) and 8(d2) shows.

3.2.2 Around inner contact region

Analogical analyses of flow patterns around the inner contact region are conducted and displayed in Fig. 9. The inner ring is rotating and its conforming degree with the ball is more than that of the outer ring. Although it seems the flow patterns around inner contact are somewhat different from those around outer contact, their features and evolution process are quite similar. The single tail flow, double tail flow, butterfly flow and tunnel flow could also be observed, and another flow pattern named heap flow should be added here.

The single tail flow here still means that the meniscus of the oil reservoir is enclosing and smooth, though the elongation of the outlet meniscus is too short to distinguish. The oil tail would rapidly diverge into two oil ridges on the inner ring and the ball, as shown in Fig. 9(a). As the increase of speed or the ratio, the outlet meniscus also ruptures and the oil tail splits into two parts. The double tail flow occurs when the inner meniscus does not concave, as shown in Fig. 9(b). The oil ridges also exist, but there are no parched areas between the oil ridges and the oil
sidebands. The oil sidebands are connected with the oil ridges along the edges of the inner raceway.

If the oil volume supplied to the inlet is not enough, the inlet meniscus concaves towards the inner contact region, and the flow pattern transforms to a butterfly flow, as shown in Fig. 9(c). The oil reservoir is split into two parts. Comparing Figs. 9(c1) with 9(c2), it is also implied that the width of the oil-starved wake of the outlet would increase with the speed or the ratio. And furthermore, compared with the flow pattern features around outer contact, it seems the distance between the oil ridges from the inner contact region under double tail flow and butterfly flow is more than that from the outer contact region. This may be due to the difference of the conformal degree between the ball and the ring.

Nevertheless, the degree of the inlet meniscus concave would not follow the same rule as that around the outer contact region. Two kinds of situations would happen as the increase of speed and the ratio, as shown in Fig. 9. If the oil reservoir around the inner contact region loses its integrity and breaks into oil patches, and the oil ridges would thoroughly migrate to the sidebands and disappear, it would be a tunnel flow, as Fig. 9(d) shows. However, as shown in Fig. 9(e), the oil in front of the inlet would accumulate, which could be called an oil heap. This kind of flow pattern is named heap flow here. The volume of the oil heap is sizable and it would become convex to the further space from the inlet.

3.2.3 Along inner ring side regions

The behaviors of the oil layer in the axial side regions of the inner raceway, which are on the two abutments of the inner ring, are also of interest. There are two flow patterns as displayed in Fig. 10. The first one is named as layer flow, which means that the oil layer on the side regions of the inner ring could maintain its smoothness and peace as a layer. However, if the interface of the oil layer starts to wave, it will become another flow pattern, which is named as wave flow. The waves rise up as the increase of speed or the ratio $\mu_{oil}/\sigma$. If the oil wave is high enough and reach the inner surface of the cage, the crest of the wave would adhere to the inner surface of the cage and form oil bridges as shown in Fig. 10 (b2), and some volume of the oil would migrate to the cage region through the oil bridge.
3.2.4 Overall effects of flow patterns

Figure 11 displays the relationship between global air–oil flow and the local flow patterns. It can be observed that the oil ridges form oil bridges between the ball and the cage, and spread to the cylinder surfaces of the cage. The positions of the oil bridges are coincident with the oil ridges on the ball. A large proportion of the oil transferring along the radial direction is prevented by the inner cylinder surface of the cage, and the oil distribution on the cage is related to the features of wave flow on the inner ring. Moreover, geometrical conformal degree between the surfaces of the ball and the ring may cause differences. As shown in Fig. 11, the distance between the two oil ridges from the outer contact outlet is smaller, so that the oil ridges from the outer contact outlet are always in between those from the inner contact outlet. As the oil ridges from the outer contact outlet adhere on the ball surface and their positions

These imply that the oil on other parts will form an interconnected transfer system with the flow patterns, though the flow patterns only form around the contact regions and on the inner ring.
nearly do not change, it may contribute to the oil heap when they reach the inner contact inlet.

3.3 Flow pattern maps with capillary effects

3.3.1 Around outer contact region

As the flow patterns described above are closely related to the speed and the ratio \( \mu_{oil}/\sigma \), flow pattern maps are constructed based on the groups of the cases with different capillary properties. The flow patterns are arranged in the matrix with the column of speed and the row of ratio \( \mu_{oil}/\sigma \), as displayed in Fig. 12. As is mentioned above that the global oil transfer and distribution is related to the levels of Ca, the dashed lines corresponding to certain Ca values are given in Fig. 12. A quantitative law could be found that the different flow patterns have obvious boundaries as marked by the dashed line. It implies that the transition of the flow patterns is strongly related to the level of Ca. The critical Ca values of the flow pattern transition are about 0.3, 0.6, and 2.5 in this simulation.

3.3.2 Around inner contact region

The flow patterns around the inner contact region described above are also summarized in a map as shown in Fig. 13. The distribution of the flow patterns is similar to the situation around the outer contact region with the obvious boundaries, and the boundaries between the different flow patterns also coincide with the isograms of Ca. However, by comparing Fig. 12 with Fig. 13, it could be found that the critical Ca values of flow pattern transition around the inner contact region are smaller than those around the outer contact region. This may be due to the difference in the conformal degree between the ball and the ring. Moreover, the flow pattern map around the inner contact region shows that the heap flow only appears when the surface tension is high, which implies that the high surface tension may be necessary to maintain the stability of the oil heap.

3.3.3 Along inner ring side regions

The map of flow patterns in the axial side regions on the abutments of the inner ring also could be divided into two regions bounded by one Ca isogram. As shown in Fig. 14, the critical Ca value is about 1.6.

In summary, considering that the ruptures, concaves, and wave initiation of the oil layer are all related to the stability of air–oil interface, it seems that a high value of Ca is beneficial for maintaining the stability of the air–oil interface. Furthermore, as the features of global oil distribution on the same Ca level are similar, there may be some correlation between the flow patterns and the oil transfer and distribution. The combined effects of the flow patterns and their transitions may influence the global behaviors of oil transfer in the ball bearing largely.
3.4 Comparison with experiments

Although it is difficult to control the oil properties and initial conditions in experiments the same as the simulations by now, it is feasible to find and to prove the existence of relevant phenomena. The flow patterns around the contact regions could be found in the experimental results from the works of literatures. As shown in Fig. 15, the flow patterns in this simulation reach a good agreement with the 2D laser-induced fluorescence (LIF) images from both the ball-on-disc rolling contact experiments and the ball bearing experiments. The bright area and the green area in Figs. 15(a) and 15(c) stand for the oil, and their brightness implies the thickness of the oil layer. It could be found that the oil reservoir around the rolling point contact region will elongate, rupture, and concave, as the bright area in Figs. 15(a) and 15(c) shows. These phenomena correspond to the 3D oil distribution shown in Fig. 15(b). And the coincidence of the simulations and the two experiments implies that the flow patterns around the rolling point contact regions may follow the same general rule. Furthermore, the results of the experiments and the flow pattern maps in this work accord with the previous studies about the air–oil interfacial flow around the rolling point contact regions [27, 46], which have mentioned that the Ca values are critical to the flow pattern transitions. Hence, it implies that the flow patterns in this work may follow the same forming mechanism and rules as those mentioned in the relevant works.

The observation of oil flow in the angular contact ball bearing is also implemented in this work from the axial side view. As shown in Fig. 16(a1), the ball bearing is illuminated and observed from the side view.
The oil is dyed green and can be distinguished from the images. It can be observed that the oil reservoir forms on the inner raceway in Fig. 16(a2), and the oil waves on the abutments of the inner ring are obvious in Fig. 16(a3). All of these features reach good agreements with the results of the simulation, as shown in Figs. 16(b2) and 16(b3). In addition, the wave flow on the abutments in Fig. 16(a3) happens under 1,000 r/min in the experiment when the ratio \( \frac{\mu_{\text{oil}}}{\sigma} \) is about 2.9 s/m, it also well fits with the flow pattern maps of Fig. 14 from simulation. The existence of the various flow patterns predicted in simulations could be proved.

4 Conclusions

A transient air–oil two-phase flow model for a ball bearing is established based on computational fluid dynamics (CFD). Groups of simulations are implemented to investigate the behaviors of oil transfer and the patterns of air–oil interfacial flow. Flow patterns are classified and compared with experiments, and the effects of different capillary conditions are discussed. The potential influence and association between the oil transfer and distribution, flow patterns, and capillary numbers are revealed. In-depth understandings of the oil supply in ball bearings are proposed. The conclusions are as follows:

1) The behaviors of oil transfer and distribution in a ball bearing are strongly related to the levels of the speed and the ratio between oil viscosity and surface tension (\( \frac{\mu_{\text{oil}}}{\sigma} \)). Staged phenomena of oil distribution happen as the increase of speed or the ratio \( \frac{\mu_{\text{oil}}}{\sigma} \). Similar phenomena are observed as long as the capillary number (Ca) level is the same.

2) Single tail flow, double tail flow, and butterfly flow will form around both the outer and inner contact regions. Tunnel flow, heap flow, layer flow and wave flow are further found and classified. The features reach good agreements with the observations from bearing and ball-on-disc experiments.

3) The combined effects of flow patterns and their transitions influence the global behaviors of oil transfer largely in bearing. Flow pattern maps are constructed by the speed and the ratio \( \frac{\mu_{\text{oil}}}{\sigma} \) for the air–oil interfacial flow around the contact regions and on inner ring abutments.

4) It is shown that the levels of Ca may be the boundaries and the critical points of the transition between different flow patterns, which implies that the flow patterns are closely related to the value of Ca.
Appendix A

The results of the stabilization test are displayed in Fig. A1. The condition when $\sigma = 0.03 \text{ N/m}$, $\mu_{\text{oil}} = 0.061 \text{ Pa-s}$, and $n = 100 \text{ r/min}$ is selected as an example. The oil volumes around the contact and side regions are extracted. It can be found that the solution becomes stabilized after 18 passages of computation domain, which corresponds to one revolution of bearing.

Fig. A1 Results of stabilization test.

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