Influences of the Geometry of the Scavenge Pipe on the Air–Oil Two-Phase Flow and Heat Transfer in an Aero-Engine Bearing Chamber

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ABSTRACT: In order to improve the characteristics of the air–oil two-phase flow and heat transfer in the scavenge pipe of an aero-engine bearing chamber, this paper presents several scavenge pipes with different cross-sectional geometries, by numerically investigating the processes of the air–oil two-phase flow and heat transfer, in comparison to a circular pipe. The findings indicate that the tripetal cross-section shows the best heat-transfer effect, while the four-petal cross-section has the lowest flow resistance. Under the same working condition and the equal wetted perimeter, the tripetal cross-section has an 8.8% higher heat-transfer effect than the circular section, while the four-petal cross-section has a 28.6% lower flow resistance than the circular; under the equal cross-sectional area, the tripetal cross-section has a 9.1% higher heat-transfer effect than the circular section, while the four-petal cross-section has a 23.6% lower flow resistance than the circular; under the equal hydraulic diameter, the tripetal cross-section has a 9.2% higher heat-transfer effect than the circular section, while the four-petal cross-section has a 21.9% lower flow resistance than the circular. Taking both the heat transfer and flow resistance into consideration, the four-petal cross-section exhibits the best comprehensive performance, with the comprehensive performance coefficient decreasing with the increase of oil inlet velocity and rising with the increase of air inlet velocity.

1. INTRODUCTION

The lubrication system plays an important role in modern engines, where the lubricating oil lubricates and cools the bearings, gears, and seals. However, the continuously rising temperature of the oil may lead to spontaneous ignition and coking, which negatively affects the reliability and safety to aero engines. Generally, bearing chambers have a vent pipe at the top of the chamber and a scavenge pipe at the bottom. The vent pipe releases most of the oil from the bearing chamber. The high-temperature problems caused by the air–oil separator, the high-temperature, and the inlet velocity of lubricating oil and air. The evaluation on the distributions of flow rate, temperature, velocity, and pressure at key locations along the oil-supply pipe and the scavenge pipe is critical to the design of bearing chambers and piping systems in aero engines. The division of oil exiting the chamber through the vent and scavenge ports depends on the shaft speeds. Increase in the shaft speed leads to an increased amount of oil exiting from the vent port.

Improvements on the configurations of bearing chambers may increase the scavenge rate (the proportion through the scavenge port), by optimizing the vent port, adding scavenge ports, and improving their structures, and so forth. Porous screens introduced around the bearing would reduce the power loss considerably, by reducing droplet–droplet interactions and droplet-bearing chamber wall interactions. A further research indicates that although the screens reduce the parasitic losses and vent protrusion reduces the oil flow to the air–oil separator, the high-temperature problems caused by the oil flow distribution may lead to malfunctions in the engine’s lube system. In addition, sealing has an impact on the air–oil two-phase flow and heat-transfer performance. Compared to the vent pipe, the scavenge pipe carries away most of the oil from the bearing chamber. The high-temperature oil inevitably involves a small amount of high-pressure sealing air and forms a complex air–oil two-phase flow, which may lead to oil coking and oil fires.

The abovementioned literatures demonstrate that the air–oil two-phase flow and heat-transfer performance can be improved by modifying the bearing chamber’s configuration, sealing, and pipe structures. However, how the cross-sectional geometry of the scavenge pipe will impact on the two-phase flow and heat transfer is scarcely reported. For this reason, this paper presents several scavenge pipes with different cross-sectional geometries, and numerically investigates the characteristics of the air–oil two-phase flow and heat transfer inside the scavenge pipe. The results will lead to a better...
understanding of the underlying mechanism of the two-phase flow in aero-engine-bearing chambers.

2. MODEL & NUMERICAL METHOD

2.1. Modeling. Four different cross-sectional geometries are shown in Figure 1, with the same pipe length of $L = 200$ mm taking into account of the total mass of the aero engine and of the space requirement. Under the equal wetted perimeter, the equal cross-sectional area and the equal hydraulic diameter, respectively (Table 1), the characteristics of the air–oil two-phase flow and heat transfer inside the vertical scavenge pipe (marked in red in Figure 1a) will be discussed in detail.

Structured meshing on the abovementioned scavenge pipes is conducted by ICEM, depicted in Figure 2. The grids close to the pipe wall are refined so as to compute the local two-phase flow and heat transfer accurately. In order to test and verify the grids independence, Nusselt number on the circular pipe wall was computed under the conditions of inlet air speed $v_g = 10$ m/s, inlet oil speed $v_l = 0.5$ m/s, inlet temperature $T = 373$ K, heat flux on the pipe wall $q = 2000$ J/(m²·s), with the grid number ranging from $4 \times 10^5$ to $20 \times 10^5$ (Figure 3). It can be inferred that the optimal grid number is $12 \times 10^5$, both to save the calculation time and to ensure the simulation accuracy.

According to Kanarachos’ research experiences on the complex air–oil two-phase flow inside the scavenge pipe, the mixture model with an interphase velocity slip can simulate the flow and heat transfer in the scavenge pipe accurately, by selecting the RNG $k–\varepsilon$ two-equation turbulent model and the second-order upwind difference scheme for discretization, with the residual error of $10^{-5}$. The mixture multiphase flow model is a simplified two-phase flow model in commercial CFD software. This model allows each fluid to move at a different velocity by setting the interphase slip velocity. The standard $k–\varepsilon$ model is difficult to solve anisotropic flows in the near-
wall region, for example, the boundary layer flow on the curved wall. The RNG $k-\varepsilon$ two-equation turbulence model is developed from the standard $k-\varepsilon$ model and can be expressed as follows.19

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho ku)}{\partial x} = \frac{\partial}{\partial x_i}\left[\alpha_{i\text{eff}} \frac{\partial k}{\partial x_i}\right] + G_k + G_b - \rho \varepsilon - Y_M + S_k$$  \hspace{1cm} (1)

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho \varepsilon u)}{\partial x} = \frac{\partial}{\partial x_i}\left[\alpha_{i\text{eff}} \frac{\partial \varepsilon}{\partial x_i}\right] + C_{1\varepsilon}\frac{\varepsilon}{k} (G_k + C_A G_b) - C_{2\varepsilon}\frac{\varepsilon^2}{k}$$

$$- R_\varepsilon + S_\varepsilon$$ \hspace{1cm} (2)

The main difference between the RNG and standard $k-\varepsilon$ models lies in the additional term in the $\varepsilon$ equation given by

$$R_\varepsilon = C_{1\varepsilon}\rho \eta^3 (1 - \eta/\eta_0) \frac{\varepsilon^2}{1 + \beta \eta^3}$$  \hspace{1cm} (3)

where $C_{1\varepsilon} = 1.42, C_{2\varepsilon} = 1.68, C_\mu = 0.085, \eta = S_\eta/k, \eta_0 = 4.38, \beta = 0.012$.

Pressure drops along a vertical circular pipe at inlet oil speeds of $v_1 = 0.2$ m/s and $v_1 = 0.6$ m/s are experimentally acquired in the reference20 with the inlet air velocity ranging from 4 to 18 m/s. The numerical method in the present paper was then applied to the experiment mentioned above. The numerical results were in good agreement and showed consistent trends with the experiments, which confirmed the reliability of the numerical method in the present work (see Figure 4).

### 2.2. Physical Properties and Boundary Conditions.

The physical properties of the two fluids are listed in Table 2.21

|        | $\rho$ (kg/m$^3$) | $\nu$ (kg/m·s) | $C_p$ (J/kg·K) | $k$ (W/m·K) |
|--------|-------------------|----------------|----------------|--------------|
| Air    | 2.92              | 1.84 $\times$ $10^{-5}$ | 1005           | 0.026        |
| oil    | 900               | 0.062          | 2060           | 0.1436       |

Pressure drops along a vertical circular pipe at inlet oil speeds of $v_1 = 0.2$ m/s and $v_1 = 0.6$ m/s are experimentally acquired in the reference20 with the inlet air velocity ranging from 4 to 18 m/s. The numerical method in the present paper was then applied to the experiment mentioned above. The numerical results were in good agreement and showed consistent trends with the experiments, which confirmed the reliability of the numerical method in the present work (see Figure 4).
reduction due to the energy loss when the fluid is flowing inside the pipeline. The energy loss is caused by the friction between the flowing fluid and pipe walls, and by the collisions and momentum exchange among the fluid particles. Besides, it is also associated with the roughness and geometries of the pipe.

The total flowing pressure drop consists of the frictional pressure drop $\Delta P_f$, the local pressure drop $\Delta P_c$, the gravitational pressure drop $\Delta P_g$, and the accelerated pressure drop $\Delta P_a$, which can be expressed below

$$\Delta P = \Delta P_f + \Delta P_c + \Delta P_g + \Delta P_a$$  \hspace{1cm} (4)

Figure 7. Cross-sectional pressure distributions for different geometries: (a) “circular”, (b) “double-petal”, (c) “tripetal”, and (d) “four-petal” (equal hydraulic diameter).

Figure 8. $Nu$ and $f$ for different geometries at different benchmarks: (a) equal wetted perimeter, (b) equal cross-sectional area, and (c) equal hydraulic diameter.

The local pressure drop $\Delta P_c$ is negligible as there is no sudden shape change inside the pipeline in the present paper. In addition, the gravitational conditions of the pipes with four different cross-sectional geometries are equivalent. Moreover, the accelerated pressure drop $\Delta P_a$ arises from the great variation in the fluid velocity; yet, Figure 5 illustrates that the outlet velocities of pipes with different cross-sectional geometries are almost the same. Therefore, the impacts of the cross-sectional geometry on $\Delta P_g$ and $\Delta P_a$ can also be neglected.

As a result, the frictional pressure drop $\Delta P_f$ will be the main contributor that leads to the different pressure drops for pipes with different cross-sectional geometries. $\Delta P_f$ can be generally calculated by

$$\Delta P_f = \frac{\lambda L \rho u_b^2}{2D} = 4f \frac{L \rho u_b^2}{2D}$$  \hspace{1cm} (5)
\[ f = \frac{\Delta P D}{2 \rho L u_b^2} \]  

(6)

where \( L \) is the pipe length; \( D \) is the equivalent diameter; \( \rho \) is the fluid’s density; \( u_b \) is the fluid’s average velocity; \( \lambda \) is the friction factor; \( f \) is the Fanning friction factor which can be simply used to describe the flow resistance inside different pipes.

Nusselt number \( Nu \) on the pipe wall is defined as

\[ h = \frac{q}{T_w - T_i} \]  

(7)

\[ Nu = \frac{hD}{k_{oil}} \]  

(8)

where \( h \), \( q \), and \( T_w \) is the mean convective heat transfer coefficient, the heat flux and temperature on the pipe wall, respectively; \( T_i \) is the inlet temperature; \( k_{oil} \) is the heat conductivity coefficient of the oil; thus, \( Nu \) can be simply used to describe the heat-transfer characteristics inside different pipes.

### 3.2. Effects of the Cross-Sectional Geometries on the Two-Phase Flow and Heat Transfer

In order to examine the air–oil two-phase flow and heat-transfer characteristics in pipes with different cross-sectional geometries, the following discussions are based on the conditions of equal wetted perimeter, equal cross-sectional area, and equal hydraulic diameter, respectively. Under the equal hydraulic diameter, Figure 6 depicts the temperature distribution across the outlet section for different geometries. It can be noticed that the temperatures of “circular” and “double-petal” are relatively higher, whereas the temperature of “tripetal” is the lowest. Because of the complex cross-sectional geometries which enhance the heat-transfer performance like “ribs”, the temperatures of “tripetal” and “four-petal” are both comparatively lower.

Figure 7 illustrates the pressure distributions at different locations along the flow direction for pipes with different cross-sectional geometries. It can be noticed that the pressure drop of “tripetal” is the highest, whereas the pressure drop of “four-petal” is the lowest. The result indicates that the scavenge pipe with the “four-petal” cross-sectional shape is beneficial to the air–oil two-phase flowing.

It has been mentioned in Section 3.1 that \( Nu \) and \( f \) can be simply used to describe the heat-transfer characteristics and flow resistance inside different pipes, respectively. \( Nu \) and \( f \) for different geometries at different benchmarks are presented in Figure 8. Under the equal wetted perimeter, the tripetal cross-section has an 8.8% higher heat-transfer effect than the circular section, while the four-petal cross-section has a 28.6% lower flow resistance than the circular (Figure 8a); under the equal cross-sectional area, the tripetal cross-section has a 9.1% higher heat-transfer effect than the circular section, while the four-petal cross-section has a 23.6% lower flow resistance than the circular (Figure 8b); under the equal hydraulic diameter, the tripetal cross-section has a 9.2% higher heat-transfer effect than the circular section, while the four-petal cross-section has a 21.9% lower flow resistance than the circular (Figure 8c).

The design of the scavenge system for aero engines requires higher heat transfer and lower resistance performance. The comprehensive performance indicator \( \eta \) can be employed to

![Figure 10. Effects of the oil inlet velocity on (a) Fanning friction factor \( f \), (b) Nusselt number \( Nu \), and (c) comprehensive performance indicator \( \eta \) (equal hydraulic diameter).](image-url)
take into consideration of both the heat transfer and flow resistance, which is defined as follows.\(^{23}\)

\[ \eta = \frac{Nu/Nu_0}{(f/f_0)^{1/3}} \]  

(9)

where "0" denotes the circular cross-section. It can be noticed from Figure 9 that the comprehensive performance indicators of “double-petal”, “tripetal”, and “four-petal” are all higher than that of “circular”, with “four-petal” the best in the comprehensive performance, which provides a way to improve the flow and heat-transfer characteristics by selecting the pipes with noncircular cross-sections.

### 3.3. Effects of the Inlet Oil Velocity on the Two-Phase Flow and Heat Transfer

Under the working conditions of the inlet air velocity at 10 m/s, inlet temperature at 373 K, and heat flux on the wall at 2000 J/(m\(^2\)-s), the effects of the inlet oil velocity on the two-phase flow and heat transfer for pipes with different geometries (equal hydraulic diameter) are presented in Figure 10. Because of the higher viscosity of the lubrication oil than air, raising the inlet oil velocity means the corresponding increase in the viscosity of the air-oil two-phase flow, which results in a higher fanning friction factor \(f\) for all pipes with different cross-sectional geometries, as depicted in Figure 10a. It should be further noticed that, the flow resistance of the pipe with “tripetal” cross-section is significantly higher than the others, while the flow resistance of the pipe with “four-petal” cross-section is the lowest. In addition, it is shown in Figure 10b that the higher oil inlet velocity will reduce the heat-transfer performance for all pipes. Because of the much higher viscosity of the oil than that of the air, increase of the oil will lead to a higher viscosity of the two-phase flow, which increases the flow resistance of the two-phase flow and curbs the convective heat transfer. In addition, as the inlet velocity of the oil is much lower than that of the air, the increased inlet velocity of the oil in a small range will not raise the velocity of the two-phase flow significantly. As a result, increasing the inlet velocity of the lubricating oil has a negative impact on \(Nu\). Figure 10b also indicates that the heat-transfer performance of the pipe with “tripetal” cross-section is the best, although this geometry exhibits the highest flow resistance in Figure 10a. Taking into account of both the flow resistance and heat transfer, Figure 10c illustrates the comprehensive performance indicator \(\eta\) for pipes with different geometries at the oil inlet velocity ranging from 0.5 to 2.0 m/s. The comprehensive performance indicator \(\eta\) declines with the increasing oil inlet velocity for all the pipes with petals, which suggests that increasing the oil inlet velocity plays a negative role in the overall air–oil two-phase flow characteristics. More importantly, the pipe with “four-petal” cross-section shows the best comprehensive performance.

### 3.4. Effects of the Inlet Air Velocity on the Two-Phase Flow and Heat Transfer

Under the working conditions of the inlet oil velocity at 0.5 m/s, inlet temperature at 373 K, and heat flux on the wall at 2000 J/(m\(^2\)-s), the effects of the inlet air velocity on the two-phase flow and heat transfer for pipes with different geometries (equal hydraulic diameter) are presented in Figure 11. Because of the far lower viscosity of air than the oil, increasing the air inlet velocity will reduce the viscosity of the two-phase flow, and the two-phase flow velocity. The former effect is more significant when the air velocity is relatively low.
(<10 m/s), which reduces the Fanning friction factor \( f \). However, the continued rise in the air–oil two-phase flow velocity will increase the friction between the pipe wall and fluids, which consequently causes the curves to level off for all the pipes with different geometries, as depicted in Figure 11a. Moreover, Figure 11b indicates that the rising air inlet velocity will elevate the heat-transfer performance for all the pipes with different cross-sectional geometries. The increased two-phase velocity helps to improve the convective heat transfer. However, it should be pointed out that the amount of lubrication oil which acts as the cooling medium remains constant. As a result, the curves level off at relatively higher air inlet velocity. Taking into account both the flow resistance and heat transfer, Figure 11c illustrates the comprehensive performance indicator \( \eta \) for pipes with different geometries at the air inlet velocity ranging from 5 to 20 m/s. The comprehensive performance indicator \( \eta \) grows with the increasing air inlet velocity for all the pipes with petals, which suggests that increasing the air inlet velocity plays a positive role in the overall air–oil two-phase flow characteristics. Again, the pipe with the “four-petal” cross-section has the best comprehensive performance.

4. CONCLUSIONS

This article presents 3 types of scavenge pipes with 3 different cross-sectional geometries, that is “double-petal”, “tripetal”, and “four-petal”, aiming to improve the characteristics of the air–oil two-phase flow and heat transfer in the scavenge pipe of an aero-engine bearing chamber, in comparison to a circular pipe. The conclusions are as follows

(1) Under the same working conditions, the tripetal cross-section shows the best heat-transfer performance, while the four-petal cross-section has the lowest flow resistance, at all the benchmarks of “equal wetted perimeter”, “equal cross-sectional area”, and “equal hydraulic diameter”, as shown in Table 3.

(2) Taking both the heat transfer and flow resistance into consideration, all the 3 types of scavenge pipes with petals have a better compressive performance than the circular pipe. The pipe with four-petal cross-section exhibits the best comprehensive performance.

(3) Raising the oil inlet velocity will increase the flow resistance, and reduce the heat-transfer performance and comprehensive performance indicator \( \eta \). In comparison, raising the air inlet velocity will reduce the flow resistance, and improve the heat-transfer performance and comprehensive performance indicator \( \eta \).

Table 3. \( Nu \) and \( f \) of “Tripetal” and “Four-Petal” Compared to “Circular”

| conditions               | \( Nu_1 - Nu_0/Nu_0 \) (%) | \( f_1 - f_0/f_0 \) (%) |
|--------------------------|-----------------------------|-------------------------|
| equal wetted perimeter   | +8.8                        | −28.6                   |
| equal cross-sectional area| +9.1                        | −23.6                   |
| equal hydraulic diameter | +9.2                        | −21.9                   |

and “four-petal”, aiming to improve the characteristics of the air–oil two-phase flow and heat transfer in the scavenge pipe of an aero-engine bearing chamber, in comparison to a circular pipe. The conclusions are as follows

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**Notes**

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**NOMENCLATURE**

- \( r_1 \) circular pipe radius (mm)
- \( r_2 \) double-petal cross-section pipe radius (mm)
- \( r_3 \) tripetal cross-section pipe radius (mm)
- \( r_4 \) four-petal cross-section pipe radius (mm)
- \( v_g \) inlet air speed (m/s)
- \( v_i \) inlet oil speed (m/s)
- \( T_i \) inlet temperature (K)
- \( q \) heat flux density (J/m²·s⁻¹)
- \( \Delta P \) total flowing pressure drop (Pa)
- \( \Delta P_f \) frictional pressure drop (Pa)
- \( \Delta P_c \) local pressure drop (Pa)
- \( \Delta P_g \) the gravitational pressure drop (Pa)
- \( \Delta P_s \) accelerated pressure drop (Pa)
- \( L \) length of the pipes (m)
- \( D \) the equivalent diameter (m)
- \( u_b \) fluid’s average velocity (m/s⁻¹)
- \( f \) Fanning friction factor
- \( \varepsilon \) mean convective heat transfer coefficient (W·m⁻²·K⁻¹)
- \( T_w \) mean temperature on the pipe wall (K)
- \( \kappa_o \) heat conductivity coefficient of the oil (W·m⁻¹·K⁻¹)
- \( \text{Nusselt number} \)
- \( \mu_{f,i} \) effective turbulent viscosity (kg·m⁻¹·s⁻²)
- \( G_i \) turbulence kinetic energy due to the mean velocity gradients (J)
- \( G_b \) turbulence kinetic energy due to buoyancy (J)
- \( Y_{St} \) the fluctuating dilatation in compressible turbulence
- \( \alpha_k \) the inverse effective Prandtl numbers for \( k \)
- \( \alpha_\epsilon \) the inverse effective Prandtl numbers for \( \epsilon \)
- \( \eta \) comprehensive performance indicator
- \( \text{EWP} \) equal wetted perimeter
- \( \text{ECSA} \) equal cross-sectional area
- \( \text{EHD} \) equal hydraulic diameter

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