

Numerical investigation on leakage characteristics of labyrinth seal for cavitation flow in the liquid hydrogen piston pump

Hongyu Ren^{1,2}, Wei Wu^{1,2}, Shaoqi Yang¹ and Xiujuan Xie^{1,*}

¹Key Laboratory of Cryogenic Science and Technology, Technical Institute of Physics and Chemistry, Chinese Academy of Sciences, Beijing 100190, China

²University of Chinese Academy of Sciences, Beijing 100049, China

*Corresponding author. E-mail: xiexujuan@mail.ipc.ac.cn

Abstract

Based on the thermodynamically modified Zwart cavitation model, a two-dimensional theoretical cavitation flow model of the labyrinth seal between the piston and cylinder in a liquid hydrogen piston pump is established and verified. The influence of cavitation on the pressure distribution along the clearance seal, which reduces the leakage rate, is analyzed. The influence of input pressure and degree of subcooling on the cavitation flow is discussed. Higher inlet pressure and degree of subcooling can diminish the sealing effect between cavitation and liquid-phase flows. Furthermore, the leakage rates for clearance and labyrinth seals with 20- to 60- μm sealing gap are compared. When the sealing gap is no less than 30 μm , the labyrinth sealing effect is superior to clearance seal. Finally, the leakage rates of labyrinth seals with three cavity structures—square, curve, and triangle—are simulated. The sealing effects are ordered as follows: square labyrinth cavity, curved labyrinth cavity, and triangular labyrinth cavity. The sealing effect is preferred when the optimal length-to-width ratio of the square cavity is around 5.0. The studies are further helpful to decrease the leakage rate of labyrinth seals and enhance volumetric efficiency in liquid hydrogen piston pump.

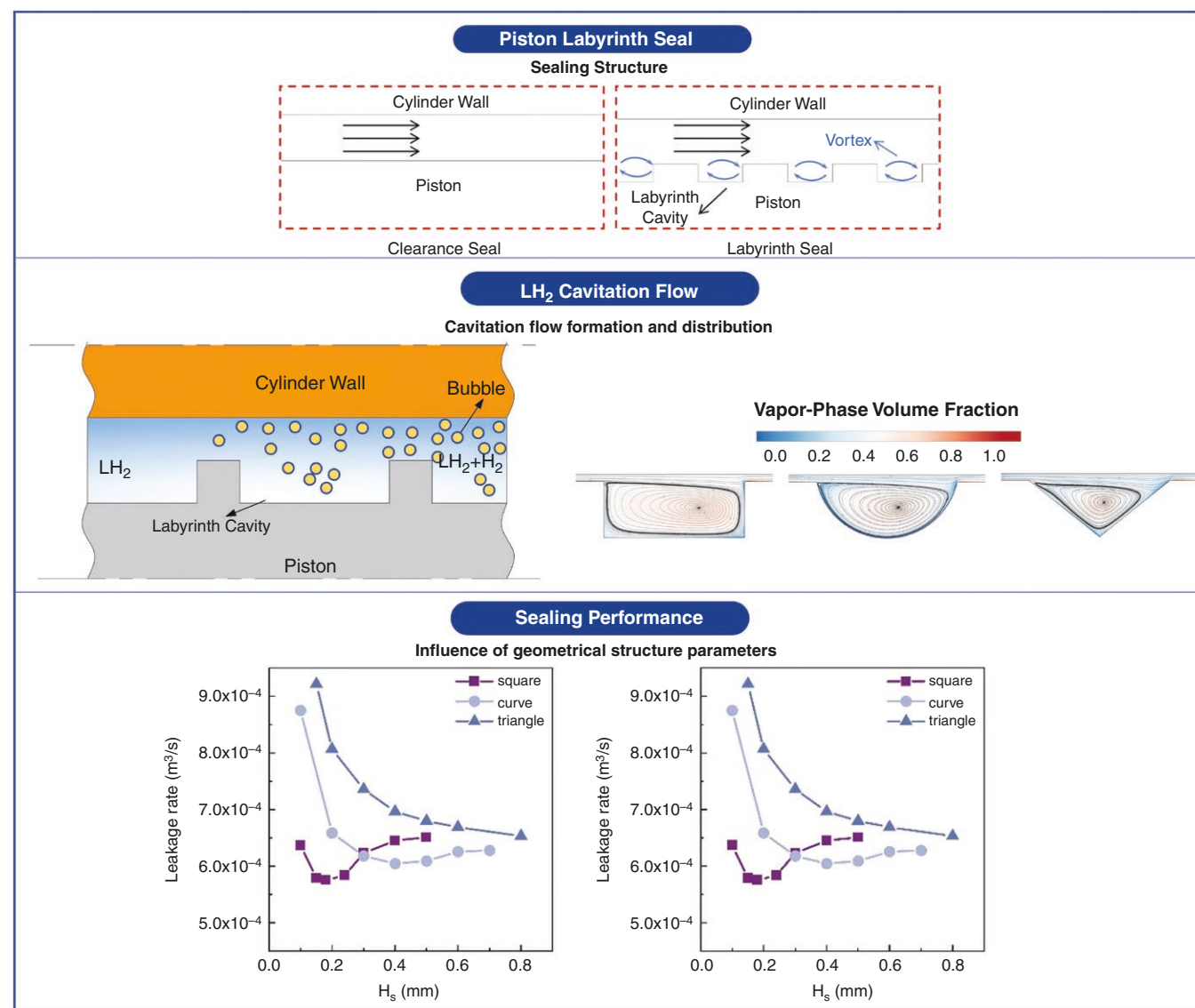
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Graphical abstract



Keywords: labyrinth seal; cavitation flow; leakage characteristics; liquid hydrogen; piston pump

1. Introduction

Hydrogen energy is a renewable, high-calorific-value, and nonpolluting fuel that offers a good alternative to traditional fossil fuels [1–4]. The liquid hydrogen refueling station, which consumes low power, is expected to be one of the ideal storage and refueling methods for hydrogen energy [5]. The liquid hydrogen piston pump is a key component of the hydrogen refueling station system, which offers several advantages, including low investment, low power consumption, and high flow rate [6, 7].

High-pressure liquid hydrogen piston pumps compress liquid hydrogen from atmospheric pressure to a supercritical state [8]. Low-pressure liquid hydrogen piston pumps are used for transferring liquid hydrogen into containers, liquefiers, or to end users. Some two-stage liquid hydrogen piston pumps can directly compress stored liquid hydrogen into a supercritical state for refueling [9]. Enhancing the volumetric efficiency of liquid hydrogen piston pumps is the focus of research. It is necessary to investigate the sealing performance, which will significantly affect the volumetric

efficiency or lead to working failure due to liquid leakage of liquid hydrogen piston pumps. For a low-pressure liquid hydrogen piston pump, the operating pressure is generally below 1 MPa. The piston seal can be in the form of a sealing ring or a clearance seal. The clearance seal, as a non-contact sealing form, has the advantages of a simple structure, low wear, allowance for thermal deformation, unlimited fluid temperature, and less friction heat generation [10]. To reduce the leakage rate in the clearance, labyrinth seals can be used effectively due to the throttle effect in teeth clearances and the dissipation effect in cavities [11]. Additionally, the labyrinth seal opens many annular grooves, which can balance the circumferential pressure distribution of the fluid in the clearance and reduce the magnitude of eccentricity.

Current studies on labyrinth seals mainly use theoretical analysis [12–15], numerical simulation [15–17], and experimental measurement methods. Theoretical research has developed a series of empirical leakage equations, which are still widely used today. With the continuous advancement of computational fluid

dynamics (CFD) technology, numerical simulations of labyrinth seals have become more prevalent, which has largely concentrated on ambient temperature conditions with limited attention paid to other factors. The difference between cryogenic fluids and ambient temperature fluids is primarily because of the thermal sensitivity of cryogenic fluids, their compressibility, and the potential for two-phase flow [5]. It is particularly important to research labyrinth seals under low temperatures. Previous research on cryogenic labyrinth seals has primarily relied on experiments [18] and theoretical analysis [19, 20], which did not take into account potential cavitation resulting in two-phase flow. When the local pressure within the labyrinth seal clearance is less than the local saturation pressure of liquid hydrogen, the flow will transition from a liquid flow to a cavitation flow [21]. As a result, liquid hydrogen no longer exists in the liquid phase but becomes a mixture of two phases, liquid and vapor, as shown in Fig. 1. Differing from those of a pure liquid, cavitation flows significantly alter the pressure distribution in the sealed area and affect the sealing effectiveness, so the classic leakage equations are no longer applicable. Unlike the condensation and evaporation model, the mass transfer equation of the cavitation model is driven by pressure differences rather than temperature differences [2]. In the gaps of the labyrinth, predicting pressure distribution is notably easier than predicting temperature distribution. The cavitation model is more suitable for this type of sealing situation, making it a critical aspect of research on cryogenic fluid seals.

Various models have been introduced to describe cavitation behavior under the isothermal assumption, including Schnerr-Sauer and Zwart models [22, 23], which have been widely applied. Due to the thermal sensitivity of the fluid at low temperatures, the isothermal assumption in these models is no longer reasonable, and the study of cavitation modeling at low temperatures is necessary. Pang et al. [22] applied a thermodynamic correction method to the Zwart model. This involved correcting the saturation pressure based on the theories of thermal equilibrium between the cavitation and the B-factor theory. They also incorporated the thermodynamic equilibrium assumption in the cavitation process, which was validated through hydrofoil experiments using liquid nitrogen and liquid hydrogen. Rodio et al. [24] enhanced the Schnerr-Sauer model by incorporating a heat source term and a convection heat transfer coefficient model into the control equation, thus accounting for the impact of heat effects. Huang et al. [25] improved the Zwart model by correcting the bubble radius thermodynamically, based on single-bubble thermal equilibrium and Fourier's law of temperature boundary layer. This correction verifies the model's reliability when using liquid hydrogen as a working medium. Zhu et al. [23] discovered a unique cavitation detachment mechanism, known as the partial detachment pattern, which explains the intricate dynamic

mechanism of cavitation clouds in liquid hydrogen. Sun et al. [26] analyzed the impact of the evaporation–condensation coefficient on the Zwart model through simulation calculations and found that setting C_{vap} and C_{cond} to 5 and 0.001, respectively, adapts to cryogenic conditions compared with experiments. Huang et al. [27] proposed the filter-based density correction model (FBDCM), which combines the filter-based model (FBM) and the density correction model (DCM) to correct turbulence viscosity for improved prediction of cavitation phenomena. Recently, some scholars have applied thermodynamically modified cavitation models to numerically simulate cryogenic labyrinth seals. Han et al. [10] developed a two-dimensional theoretical model for the axial labyrinth seal of a liquid nitrogen rotor system. They calculated the cavitation flow leakage rate via simulation. Liu et al. [28] investigated the deformation of the labyrinth seal under low-temperature and liquid nitrogen fluid loads. They conducted fluid–solid coupled simulations and compared the results with experimental data to determine the actual leakage quantity.

The above studies have mainly focused on rotating machines, whereas liquid hydrogen piston pumps have different drive forms and labyrinth seal structures compared to those of rotating machines, resulting in significant differences in gap flow and cavitation characteristics. Currently, there is a relatively lack of research on the leakage characteristics and cavitation phenomenon of liquid hydrogen piston pumps with labyrinth seals. Therefore, this paper focuses on the leakage characteristics of the labyrinth seal for cavitation flow in the liquid hydrogen piston pump. A two-dimensional theoretical model of cavitation flow in labyrinth seals of liquid hydrogen piston pumps has been established. The Zwart cavitation model, which is based on the B-factor thermodynamic modification, has been considered and verified. This analysis examines the impact of cavitation on the flow in the sealing region, as well as the effects of inlet pressure and degree of subcooling in the cylinder on the cavitation leakage rate. The leakage rate for different sealing gaps and three labyrinth structures (square, curve, and triangle) is compared, and it is determined that the square labyrinth cavity is preferred with an optimal length-to-height ratio. The experiments conducted in this study are important for optimizing the dimensions of labyrinth seal structures to decrease the leakage rate of liquid hydrogen piston labyrinth seals and enhance volumetric efficiency through quantitative analysis.

2. Theoretical model

2.1 Numerical model

The two-dimensional homogeneous mixture Navier–Stokes equations, which include continuity, momentum, and energy equations, serve as the controlling equations. The physical parameters are two-phase flow variables, and the specific equations are as follows:

$$\frac{\partial}{\partial t}(\rho_m) + \nabla \cdot (\rho_m \vec{u}) = 0 \quad (1)$$

$$\frac{\partial}{\partial t}(\rho_m \vec{u}) + \nabla \cdot (\rho_m \vec{u} \vec{u}) = -\nabla p + \nabla \cdot ((\mu_m + \mu_t) \cdot (\nabla \vec{u} + \nabla \vec{u}^T)) + \vec{F} \quad (2)$$

$$\frac{\partial}{\partial t}(\rho_m \left(h_m + \frac{u^2}{2} \right)) + \nabla \cdot \left(\rho_m \vec{u} \left(h_m + \frac{u^2}{2} \right) \right) - \frac{\partial p}{\partial t} = \nabla \cdot (k_{eff} \nabla T) + S_E \quad (3)$$

$$\frac{\partial}{\partial t}(\alpha_v \rho_v) + \nabla \cdot (\alpha_v \rho_v \vec{u}) = \dot{m}^+ - \dot{m}^- \quad (4)$$

The mixture's physical property parameters are as follows:

$$\rho_m = \rho_l (1 - \alpha_v) + \rho_v \alpha_v \quad (5)$$

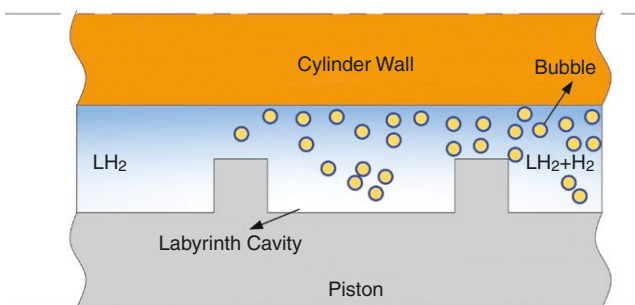


Figure 1. Cavitation phenomenon in the cryogenic labyrinth seal.

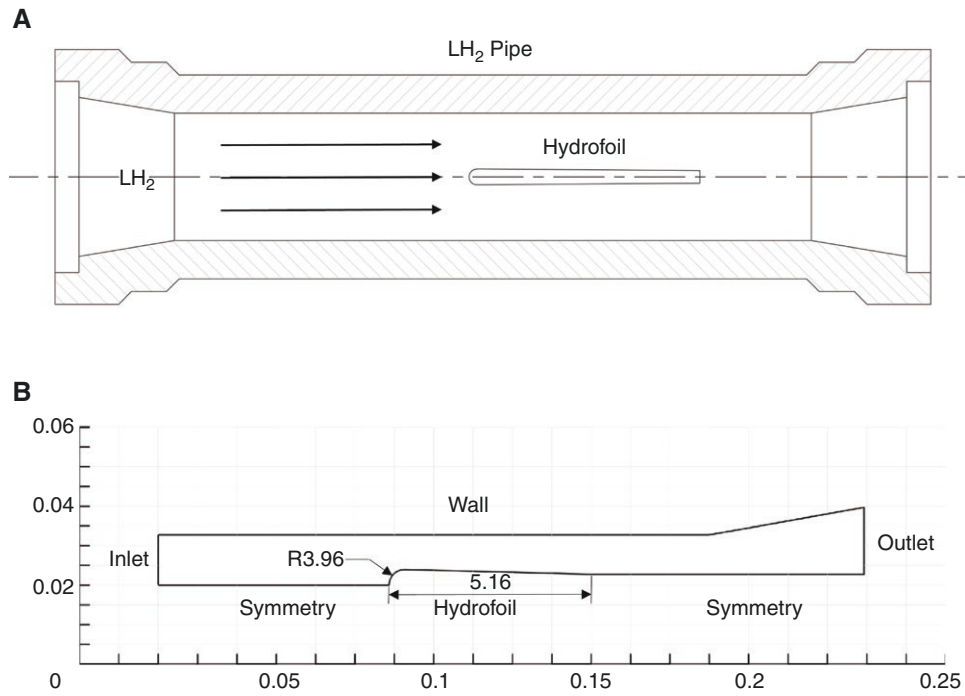


Figure 2. (a) Hord experimental setup and (b) computational fluid domain.

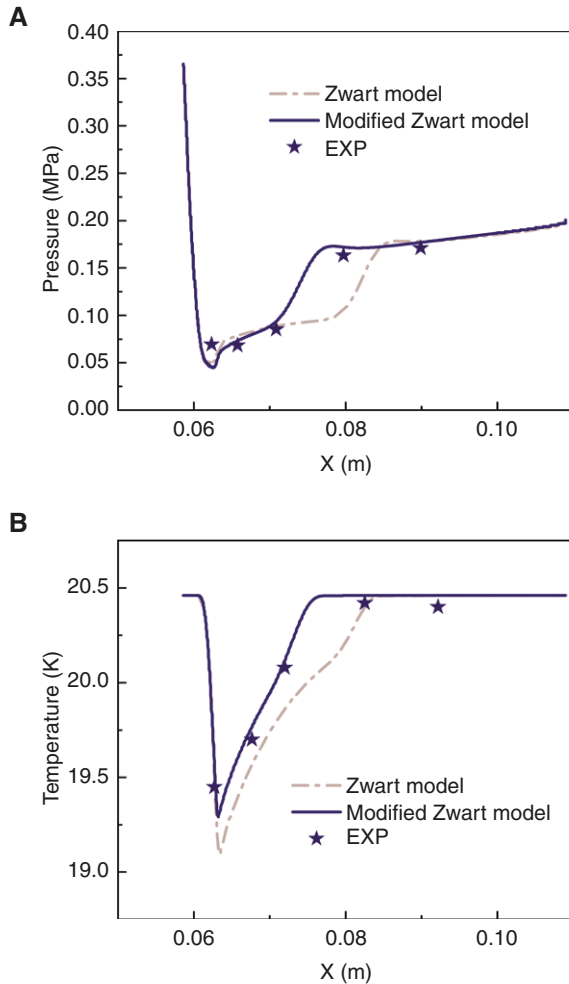


Figure 3. (a) Pressure distribution and (b) temperature distribution on the hydrofoil surface in comparison with the experiments.

$$\mu_m = \mu_l (1 - \alpha_v) + \mu_v \alpha_v \quad (6)$$

$$h_m = \frac{\rho_l h_l (1 - \alpha_v) + \rho_v h_v \alpha_v}{\rho_m} \quad (7)$$

$$k_{eff} = k_l (1 - \alpha_v) + k_v \alpha_v + k_t \quad (8)$$

Where, the subscripts v , l , m , and t stand for gas, liquid, mixture, and turbulence, respectively. p is the pressure; $\vec{u}$ is the velocity vector; $\vec{F}$ is the volumetric force; α_v is the vapor phase volume fraction; ρ_m , μ_m , and h_m are the mixture density, viscosity and enthalpy, respectively; μ_t is the turbulent viscosity; k_{eff} is the effective thermal conductivity; the subscripts v and l are the vapor and liquid phases, respectively; $\dot{m}^+$ and $\dot{m}^-$ are the evaporating and condensing masses, respectively. The source term in the energy equation represents the thermal effect caused by mass transfer between the phases due to cavitation. In the energy equation, $S_E = \dot{m}L = (\dot{m}^+ - \dot{m}^-)L$, where L is the latent heat of liquid hydrogen.

The cavitation flow model used is a modified Zwart model, taking into account the B-factor thermodynamic effect. The source terms for evaporation and condensation in the mass transfer equation can be expressed using the following equations:

$$p_v \geq p, \dot{m}^+ = C_{vap} \frac{3\rho_v \alpha_{nuc} (1 - \alpha_v)}{R_B} \sqrt{\frac{2(p_v(T) + p_L - p)}{3\rho_l}} \quad (9)$$

$$p \geq p_v, \dot{m}^- = C_{cond} \frac{3\alpha_v \rho_v}{R_B} \sqrt{\frac{2(p - p_L - p_v(T))}{3\rho_l}} \quad (10)$$

$$P_L = -\Delta T \frac{dP_v(T)}{dT} = -\frac{\alpha_v}{1 - \alpha_v} \frac{\rho_v L}{\rho_l C_{pl}} \frac{dP_v(T)}{dT} \quad (11)$$

C_{vap} and C_{cond} are empirical coefficients, which are taken as 5 and 0.001 under liquid hydrogen conditions, respectively. R_B is the spherical bubble radius, which typically taken as 1×10^{-6} ; $\alpha_{nuc} = 5 \times 10^{-4}$ is volume fraction of the gas core point. P_L is the saturation pressure correction term based on thermodynamic

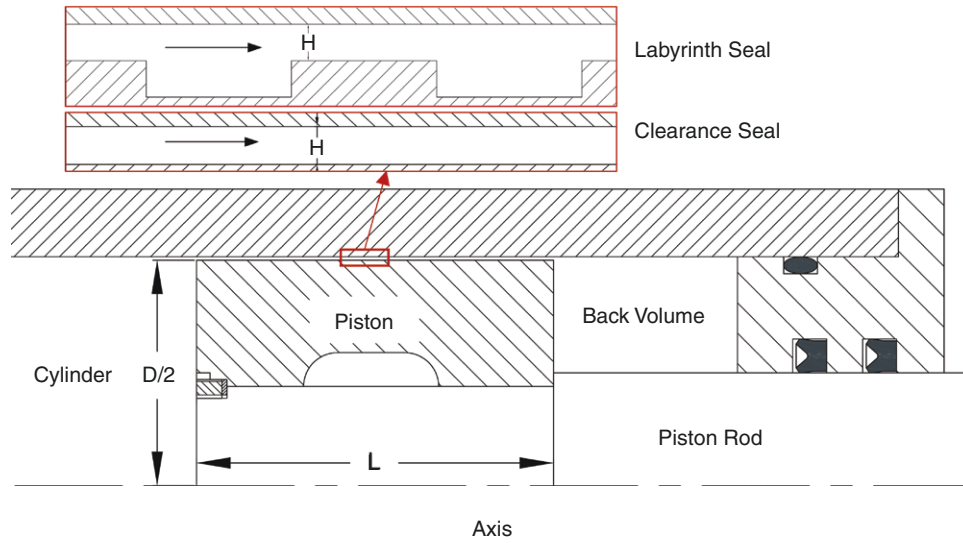


Figure 4. Schematic diagram of the liquid hydrogen piston pump structure.

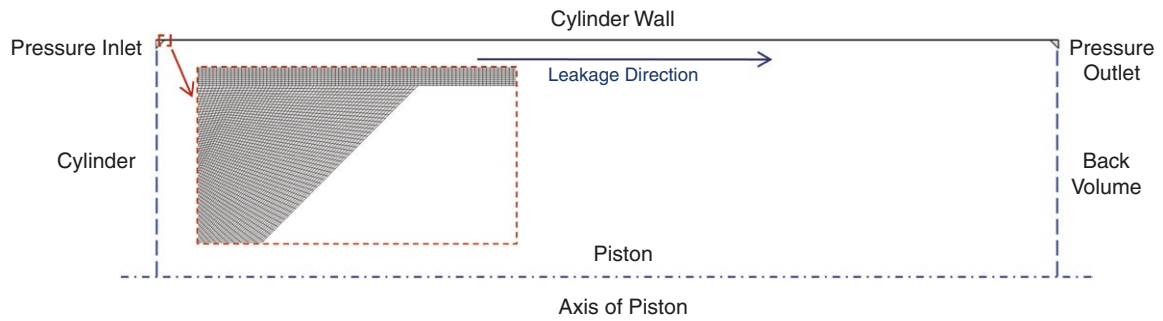


Figure 5. Computational domain and boundary conditions of clearance seal in the liquid hydrogen piston pump.

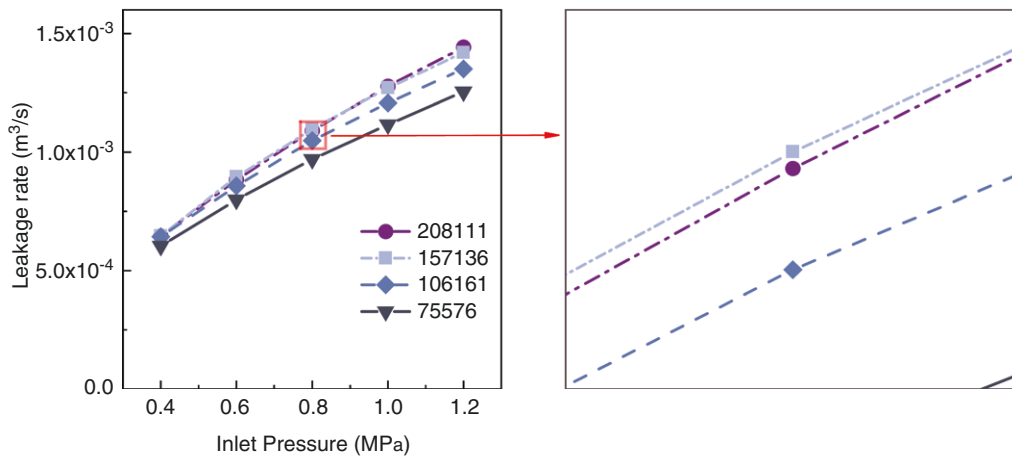


Figure 6. Independence validation of mesh size.

correction. The change of the saturation pressure with temperature can be fitted by the actual data.

The turbulence model adopts $k-\epsilon$ renormalization group (RNG) model. The effect of vortex on turbulence is included in the RNG model, enhancing accuracy for vortex flows. While the standard $k-\epsilon$ model is a high-Reynolds number model, the RNG theory provides an analytically derived differential formula for effective viscosity that accounts for low-Reynolds number effects. The RNG

model is therefore a more reliable alternative to the standard $k-\epsilon$ model when dealing with labyrinth cavities with a large number of eddy currents. The specific $k-\epsilon$ RNG model is as follows:

$$\frac{\partial}{\partial t}(\rho k) + \nabla \cdot (\rho k \vec{u}) = \nabla \cdot (\alpha_k \mu_{eff} \nabla k) + G_k + G_b - Y_M - \rho \epsilon \quad (12)$$

$$\frac{\partial}{\partial t}(\rho \epsilon) + \nabla \cdot (\rho \epsilon \vec{u}) = \nabla \cdot (\alpha_\epsilon \mu_{eff} \nabla \epsilon) + C_{1\epsilon} \frac{\epsilon}{k} (G_k + C_{3\epsilon} G_b) - C_{2\epsilon} \rho \frac{\epsilon^2}{k} - R_\epsilon \quad (13)$$

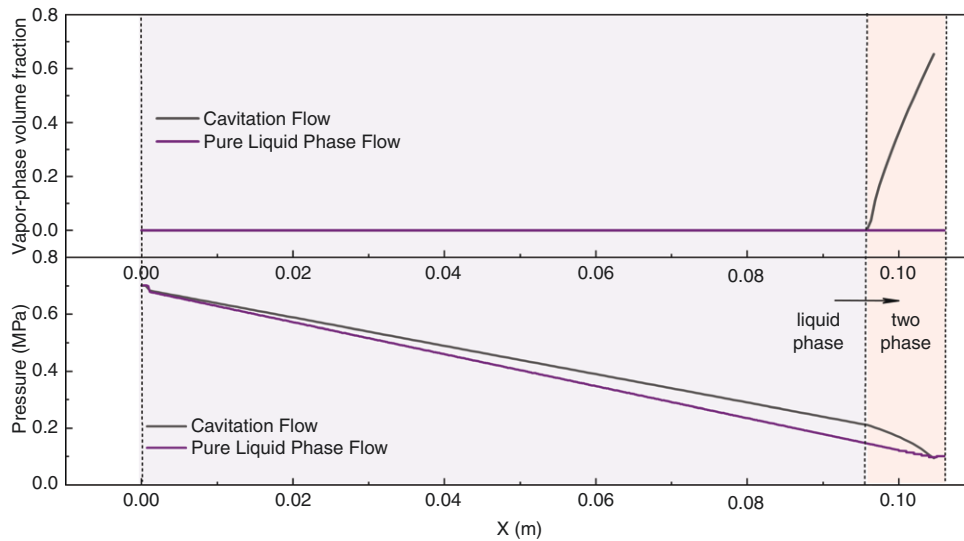


Figure 7. The axial pressure and axis vapor-phase volume fraction distribution in clearance seal region.

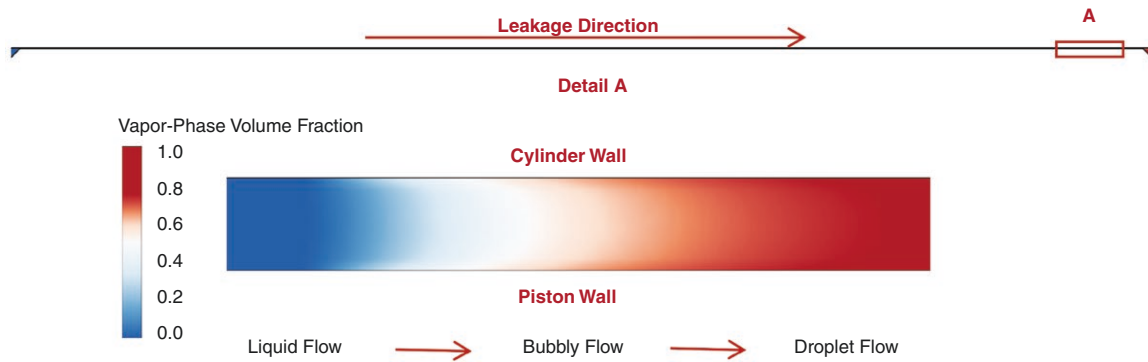


Figure 8. The clouds of vapor-phase volume fraction in the clearance seal region.

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (14)$$

Where k represents the turbulent kinetic energy, ε denotes the turbulent dissipation rate, $C_\mu = 0.0845$, $C_{1\varepsilon} = 1.42$, and $C_{2\varepsilon} = 1.68$, and the remaining parameters are calculated automatically via Fluent software. To comply with the requirements of the turbulence model, it is necessary to ensure that the height of the first layer of the grid near the wall is such that y^+ is maintained between 30 and 100.

2.2 Model validation

This paper verifies the reliability of the Zwart model based on the B-factor thermodynamic modification under liquid hydrogen conditions by using the Hord liquid hydrogen cavitation experimental data [29]. The Hord's experiment utilized a rounded tilting airfoil with an overall chord length of 63.5 mm, a rounded radius of 3.96 mm, and a pipe diameter of 25.4 mm. Five pressure and temperature sensors were arranged on the upper surface of the hydrofoil, respectively. A two-dimensional structural grid was utilized to construct the computational domain of the hydrofoil cavitation experiment. The hydrofoil is a symmetric structure, and only the upper part of the hydrofoil is considered as the computational domain. Figure 2 illustrates the configuration of the Hord experimental setup and the computational fluid domain.

This study uses the commercial software FLUENT for simulating the flow field. The turbulence model used is the $k-\varepsilon$ RNG model. The two-phase flow model employs the mixture model with liquid hydrogen as the primary phase and hydrogen as the secondary phase. The cavitation model is applied using the Zwart model based on the B-factor thermodynamic correction. The experimental results for working condition 248C were used for verification. The inlet conditions were a velocity inlet set to 51.2 m/s, a temperature set to 20.46 K, and a volume fraction of inlet hydrogen set to 0. The outlet pressure is set as 0.155 MPa. The wall and the hydrofoil are set to no-slip and adiabatic wall. Figure 3 presents the simulated results of pressure and temperature distribution on the hydrofoil surface in comparison with the experiments. The data simulated using the modified Zwart cavitation model (MZGB) are in good agreement with Hord's experimental data. This is a demonstration of the reliability of the MZGB under the conditions of cryogenic liquid hydrogen.

2.3 Physical model and mesh validation

Figure 4 shows the schematic diagram of the liquid hydrogen piston pump structure. The piston relies on the piston rod driving for the reciprocating motion to pump liquid hydrogen into the cylinder. The piston approximately measures 106 mm in length (L) and 133 mm in diameter (D). Figure 4 shows the flow path for clearance seal and labyrinth seal. The channel for clearance seal

is formed between the cylinder wall and the piston wall, which connects the cylinder and the back volume. As there is a pressure difference between the cylinder and back volume, the fluid flows through the gap from the cylinder to the back volume. To achieve labyrinth seal of the liquid hydrogen piston pump, an annular labyrinth groove must be processed in the piston.

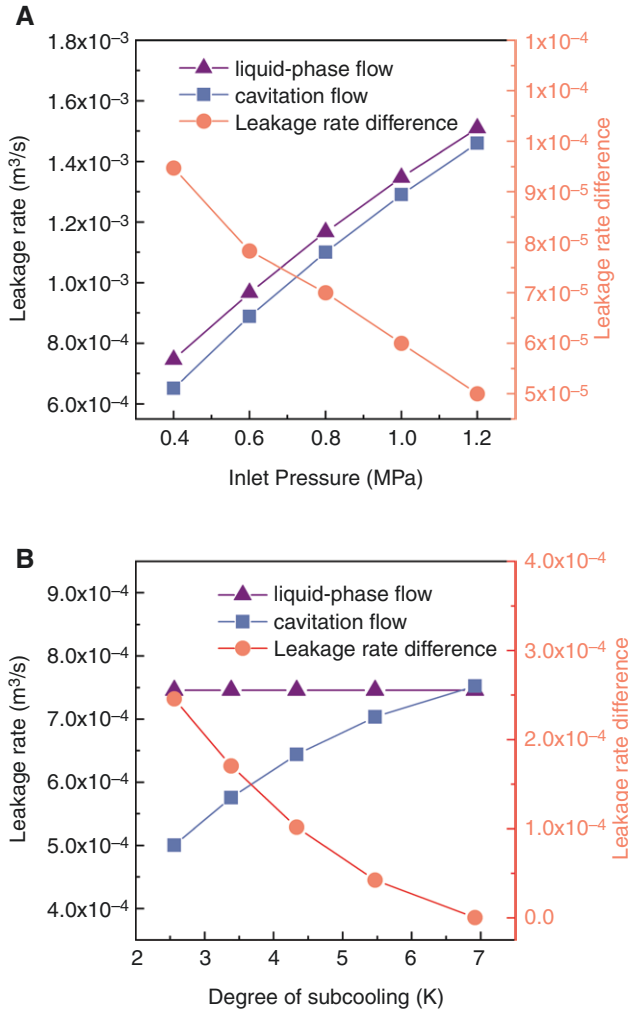


Figure 9. Leakage rate of cavitation and liquid-phase flows under different (a) inlet pressures and (b) degrees of subcooling.

The sealing gap H is four orders of magnitude smaller than the diameter and length of the piston. Thus, the fluid computational domain can be simplified to a two-dimensional model, which is used here for the clearance seal. Figure 5 shows the computational domain and boundary conditions of clearance seal in the liquid hydrogen piston pump. The domain comprises a quadrilateral structural grid, with $50 \mu\text{m}$ H . The inlet and outlet pressures are shown in Fig. 5. The cylinder wall is a static wall surface, and the piston wall is a dynamic wall surface.

Mesh size independence validation was performed to ensure that the influence of mesh size and quality on the simulated results is minimal. Simulations were conducted using the given boundary conditions and mesh quantities of 75 576, 106 161, 157 136, and 208 111. The structural factors of the liquid hydrogen pump result in a pressure in the back volume chamber that is close to atmospheric pressure. This pump is designed to operate at a pressure of 6 bar. Furthermore, the remaining subcooling will be consumed gradually in the process of liquid hydrogen transportation. The temperature of the liquid hydrogen to enter the compression chamber is 22–24 K. The inlet and outlet pressures are set as 0.6 and 0 MPa, respectively. The inlet temperature is 22.91 K. The reference pressure was 0.101325 MPa. The piston wall had a no-slip adiabatic moving wall with a velocity of -1 m/s , and the cylinder wall was a no-slip adiabatic static wall. The implementation of robust insulation measures will ensure that the total cooling leakage from the pump body is below 10 W/m^2 . For high-flow liquid hydrogen pumps, this level of cooling leakage has a relatively minor impact on the temperature distribution, so the wall is set for the adiabatic wall. Liquid hydrogen is used as the work medium, and the settings of two-phase flow are the same as those used in the validation of the cavitation model, with the gravitational force ignored. Figure 6 shows the simulated results for the independence validation of the mesh size. In this study, the sealing leakage rate is defined as the volume leakage rate per unit of circumferential length. The leakage rate of the clearance seal increases with increasing inlet pressure. Additionally, the simulation results of the leakage rate become more accurate as the number of grids increases until it stabilizes. For the simulation, a mesh with 157 136 nodes was chosen to balance computational accuracy and efficiency.

By default, subsequent simulations use an inlet pressure of 0.6 MPa, a temperature of 22.91 K, and an outlet pressure of 0 MPa. The default simulation is two-phase flow. The sealing gap H is set to $50 \mu\text{m}$ unless specified otherwise.

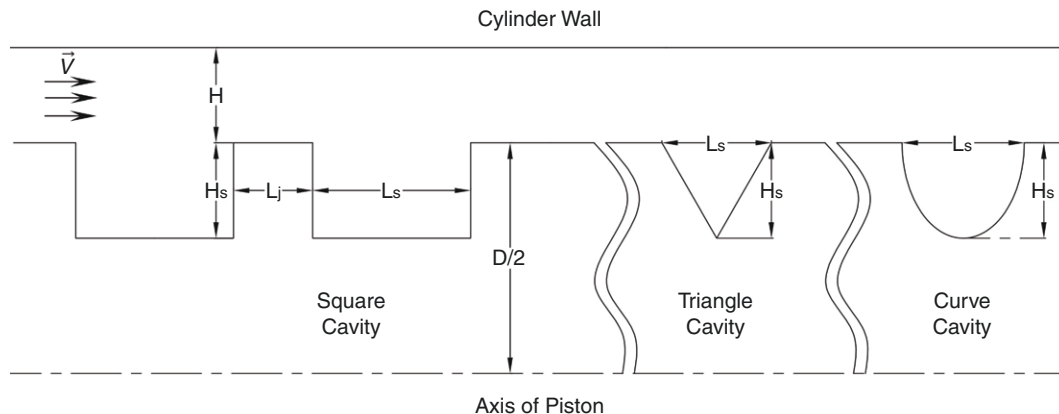


Figure 10. Geometric structure of labyrinth seal with square, triangular, and curved cavities.

3. Simulated results and discussions

3.1 The effect of cavitation flow

To verify the impact of cavitation flow on the piston clearance seal, this study simulates the pure liquid-phase flow and cavi-

tation flow. Figure 7 shows the simulated results of the axial pressure and vapor-phase volume fraction distribution in the clearance seal region. The pressure in the clearance seal decreases continuously from the inlet to the outlet. Cavitation occurs when the pressure decreases to the local saturation pressure, causing a change from liquid-phase flow to two-phase flow. The presence of cavitation flow significantly alters the pressure distribution in the clearance seal. In addition, Figure 7 shows that the pressure distribution in cavitation flow is different from that in pure liquid-phase flow. In cavitation flow, liquid hydrogen enters through the inlet and undergoes a phase change from liquid phase to two-phase when the local pressure drops below the saturation pressure. The pressure distribution in cavitation flow has an obvious transition where the phase transition occurs and is nearly segmented linear. The flow in the liquid-phase part and the two-phase part must satisfy the mass conservation and continuity equation. However, the density of the liquid-phase part is greater than that of the two-phase part, so the pressure gradient in the liquid-phase part must be significantly smaller than that in the two-phase part. The absolute value of the pressure gradient in the seal clearance is reduced from 5.44 MPa/m for pure liquid-phase flow to 4.853 MPa/m for the liquid portion of the cavitation flow. This results in a reduction in leakage rate for cavitation flow.

Figure 8 shows the clouds of vapor-phase volume fraction in the clearance seal region. Cavitation initially occurs at the surface of the cylinder and piston walls, subsequently propagating to the interior. This phenomenon is attributed to the lower pressure observed near the walls, which results in the formation of a two-phase cross-section. When cavitation occurs, the volume fraction of the vapor phase increases, and the flow transitions from a pure liquid to a bubbly flow. As the pressure decreases further and the volume fraction of the vapor phase increases further, the bubble flow transitions to droplet flow.

3.2 The effect of the inlet boundary conditions on leakage rate

The impact of inlet boundary conditions on cavitation flow in the clearance seal is significant, as discussed in this section. Figure 9a and b show the leakage rate of cavitation and liquid-phase flows under different inlet pressures and degrees of subcooling. The figure illustrates that as the inlet pressure increases, the difference between the leakage rate of liquid-phase flow and cavitation flow decreases. Additionally, the leakage rate of cavitation flow is

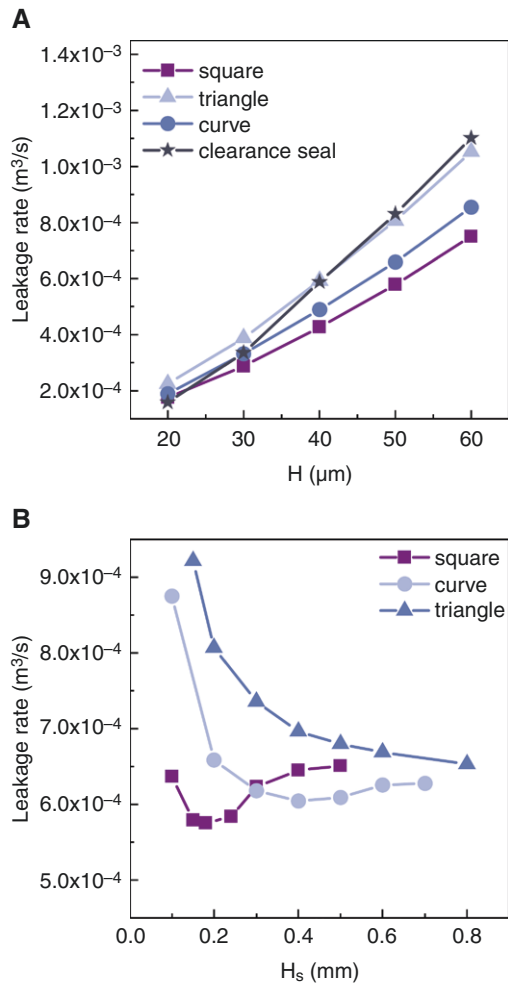


Figure 11. (a) Leakage rate versus the sealing gap H of labyrinth and clearance seals. (b) Leakage rate versus cavity depth H_s in labyrinth seals with square, triangular, and curved cavities.

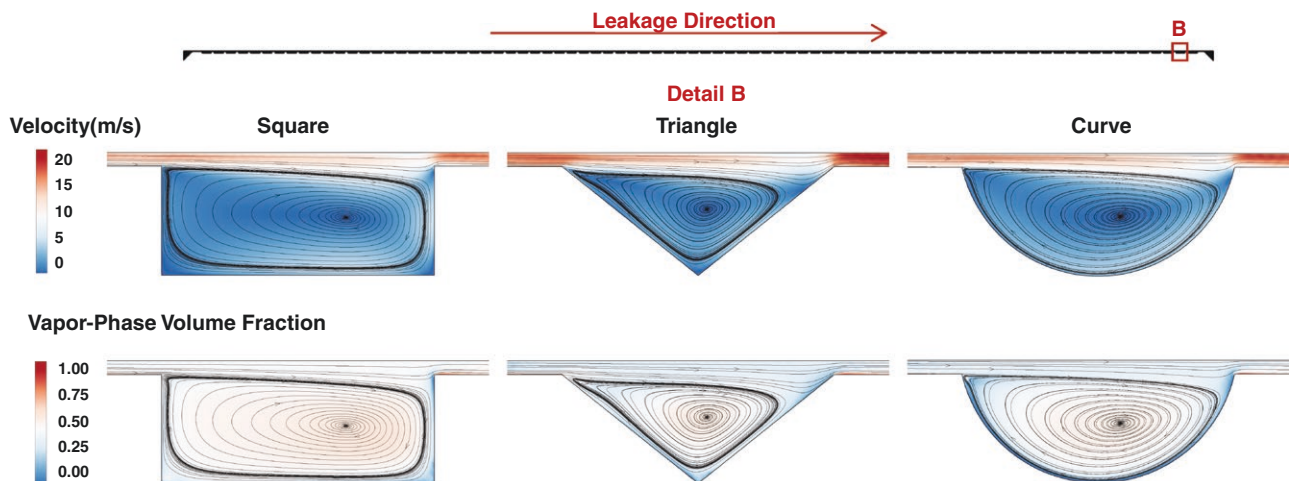


Figure 12. Velocity clouds and vapor-phase volume fraction in the labyrinth seals with square, triangular, and curved cavities.

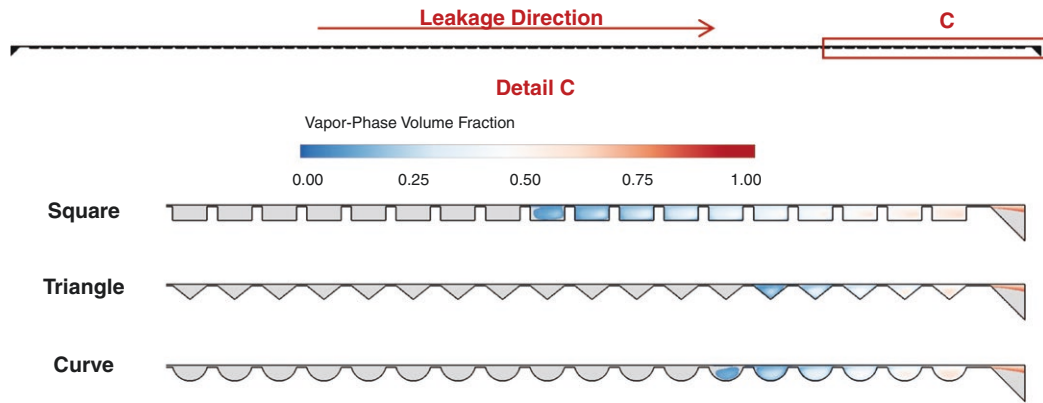


Figure 13. Cloud distribution of the axis vapor-phase volume fraction in labyrinth seals with the square, triangular, and curved cavities.

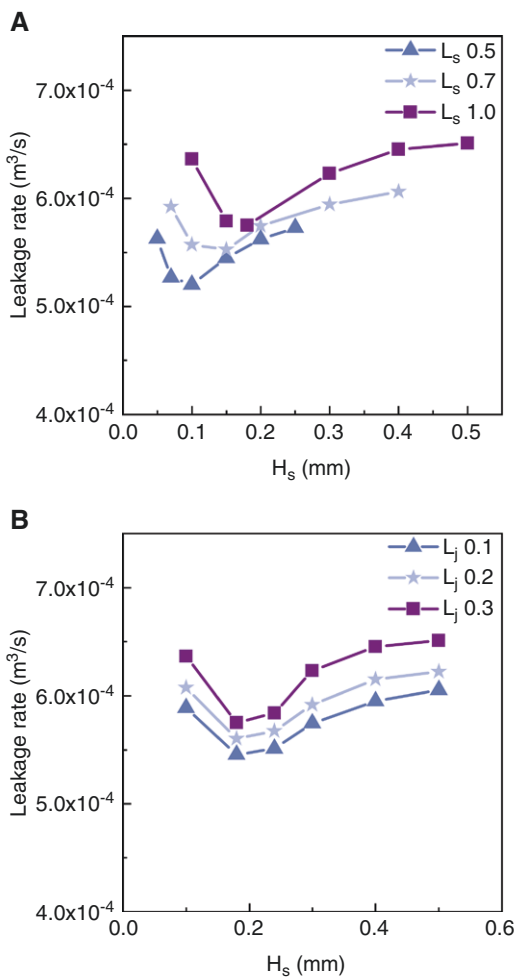


Figure 14. Leakage rate at various H_s with (a) square cavity lengths (L_s) of 0.5, 0.7, and 1 mm and (b) square cavity interval (L_j) of 0.1, 0.2, and 0.3 mm.

smaller than that of liquid-phase flow, and both increase as the inlet pressure increases. The inlet degree of subcooling is the difference between the saturation and the actual temperature in the cylinder at the current pressure. A higher degree of subcooling indicates a lower local saturation pressure. The inlet pressure is maintained at a gauge pressure of 0.4 MPa, while the effect of the inlet degree of subcooling is observed by changing the inlet

temperature. At different degrees of subcooling, Fig. 9b illustrates the difference in leakage rate between pure liquid and cavitation flow. The leakage rate of cavitation flow increases as degree of subcooling increases and local saturation pressure decreases. Inlet degree of subcooling has minimal effect on liquid-phase flow. This is because that alterations in physical characteristics resulting from temperature fluctuations are not substantial for liquid hydrogen and exert a negligible influence on liquid-phase flow.

3.3 The effect of cavity structures and dimensions of labyrinth seal on leakage rate

The labyrinth seal differs from the ordinary clearance seal, which contains many small cavities in the piston or cylinder wall. Fluid circulates within these cavities, creating eddy currents and friction with the wall, which intensifies the dissipation of kinetic energy of the fluid and reduces the leakage rate. Figure 10 shows the geometric structure of labyrinth seal with square, triangular, and curved cavities, which would have a significant impact on the sealing effect. The depth, interval, and width of the cavity are denoted by H_s , L_j , and L_s , respectively.

The geometrical parameters of the labyrinth seal were standardized as follows: $H_s = 0.2$ mm, $L_j = 0.3$ mm, and $L_s = 1$ mm. Figure 11a illustrates the leakage rate versus the sealing gap H of labyrinth and clearance seals under identical operating conditions. For H above 30 μ m, square and curved cavities provide better sealing than clearance seal, resulting in smaller leakage rate. When H is reduced to 20 μ m, the clearance seal has smaller leakage rate, which is hardly implemented in the engineering field.

Figure 11b illustrates the leakage rate varying with cavity depth H_s in labyrinth seals with square, triangular, and curved cavities. The simulated geometric structure parameter L_j was fixed at 0.3 mm and L_s at 1 mm. The square cavity has the smallest leakage rate at H_s of approximately 0.18 mm, with leakage rate increasing as H_s decreases or increases. The leakage rate of the triangular cavity decreases as H_s increases and eventually stabilizes. Curved cavity has the similar trend as square cavity. An ideal length-to-height ratio (L_s/H_s) is existed to minimize the leakage rate in three different cavities. Labyrinth seals with square cavity have slightly better sealing performance than curved cavity, while triangular cavity is the least effective.

Figure 12 displays the velocity clouds and fluid traces in the labyrinth seals with square, triangular, and curved cavities. Vortices are formed in three cavity shapes, and this vortex dissipation causes a significant reduction in fluid kinetic energy,

resulting in decreased leakage rate. In comparison to the other two shapes, the square cavity exhibits a more pronounced vortex development and higher resistance to flow, which results in lower local pressure in the vortex and higher volume fraction of the vapor phase. Figure 13 displays the cloud distribution of the axis vapor-phase volume fraction in the labyrinth seals with square, triangular, and curved cavities. It is evident that the cavitation position of the square cavity is closer to the entrance. The reason is the local pressure inside the square cavity is lower than that of the triangular and curved cavities, making it more susceptible to cavitation.

3.4 The optimization of dimensions of square cavity

Optimizing the dimensions of the labyrinth seal is crucial to enhance the sealing performance of the piston. Figure 14a illustrates the leakage rate at different H_s for square cavity lengths (L_s) of 0.5, 0.7, and 1 mm. It is evident that as L_s decreases, the cavity becomes denser, resulting in relatively smaller leakage rate. Additionally, square cavity has an optimal length-to-height ratio, which is approximately 5.0 with $L_s = 0.5$ mm and $H_s = 0.1$ mm, respectively, as shown in Fig. 14a. Figure 14b presents the leakage rate at various H_s for square cavity interval (L_s) of 0.1, 0.2, and 0.3 mm. The effect of L_s on the sealing performance of the square labyrinth is similar to that of L_s , which alters the leakage rate by modifying the density of the square cavity. As depicted in Fig. 14b, as L_s decreases, the square cavity becomes denser, resulting in a continuous decrease in the leakage rate.

4. Conclusions

This study establishes a two-dimensional cavitation flow model for labyrinth seals. The effects of cavitation flow, cavity configuration, and dimensions on the leakage characteristics of labyrinth seals are studied based on CFD, and the results of the study show that:

1. Cavitation flow would exist in the flow path connecting the cylinder and the back volume of liquid hydrogen piston pumps. The boundary conditions of high inlet pressure and high degree of subcooling can reduce the level of cavitation in the clearance seal.
2. The comparison results of different sealing gaps and cavity structures show that the smaller sealing gaps lead to the less leakage rate. When the sealing gap is smaller than 30 μm , the labyrinth seal performs worse than the clearance seal. The labyrinth seal with square cavity offers the optimal sealing performance, followed by the curved cavity, and then the triangular cavity.
3. The optimal length-to-height ratio of about 5.0 is obtained to achieve the minimum leakage rate by using labyrinth seal with square cavity. The smaller the length and interval of the square cavity, the better the sealing effect.

The above conclusions contribute to quantitatively analyze and optimize the structures and dimensions of labyrinth seal to reduce the leakage rate, and then improve the volumetric efficiency of liquid hydrogen piston pumps.

Nomenclature

α : Volume fraction

α_{nuc} : Volume fraction of the gas core

Δ : Minimum mesh size, m

m : Mass transport item, $\text{kg}/(\text{m}^3\cdot\text{s})$

k_{eff} : Thermal conductivity, $\text{W}/(\text{m}\cdot\text{K})$

μ : Kinetic viscosity, $\text{Pa}\cdot\text{s}$

ρ : Density, kg/m^3

ϵ : Turbulent dissipation, m^2/s^3

C_{cond} : Condensation empirical coefficient

C_{vap} : Evaporation empirical coefficient

C_p : Thermal capacity, $\text{J}/(\text{kg}\cdot\text{K})$

k : Turbulence kinetic energy, m^2/s^2

L : Latent heat, J/kg

p : Pressure, Pa

p_r : Thermodynamic pressure correction term, Pa

p_s : Saturation pressure, Pa

R_b : Spherical bubble radius

T : Temperature, K

t : Time, s

u : Velocity, m/s

Author contributions

Hongyu Ren (Formal analysis [lead], Investigation [lead], Software [lead], Writing—original draft [lead]), Wei Wu (Resources [equal], Writing—review & editing [equal]), Shaoqi Yang (Writing—review & editing [equal]), and Xiujian Xie (Writing—review & editing [lead])

Conflict of interest statement

None declared.

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Data availability

The data underlying this article will be shared upon reasonable request to the corresponding author.

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