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Study on Heat Transfer Characteristics of Brush-Type Sealing Labyrinth

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Abstract

This study addresses the significant deviation between actual heat consumption rates and design values caused by inter-stage air leakage in a power plant steam turbine's staggered-tooth labyrinth seal system. Focusing on retrofitting existing labyrinth seals, we implemented brush seals at specific stator sealing tooth positions based on field operational data. The proposed hybrid system demonstrates that maximum temperature serves as the primary thermal performance metric for evaluating structural and operational parameters. Key findings include: 1) Positive correlations exist between brush wire diameter, bundle thickness, brush ring inner diameter, temperature, and rotational speed with maximum temperature, while negative correlations are observed for brush arrangement angle, front/rear plate height, brush ring outer diameter, sealing clearance, and pressure differential. Labyrinth tooth clearance has minimal impact. 2) Temperature variations primarily result from frictional heat generation and leakage variations caused by parameter adjustments.

Keywords

Brush sealing; Labyrinth sealing; Interlocking teeth; Heat transfer characteristics.

1. INTRODUCTION

China's current energy landscape is characterized by natural gas scarcity, limited oil reserves, and relatively abundant coal reserves. Thermal power generation has long dominated the nation's electricity production structure. As the core component of thermal power generation, steam turbines convert thermal energy into mechanical energy. Enhancing their operational efficiency is crucial for improving the overall energy utilization efficiency of the thermal cycle system, reducing emissions, and ensuring operational safety. The sealing structure is a critical part of steam turbines. To prevent vibrations, friction, and other failures between the rotor and stator, a certain clearance is typically maintained. At China's current technological level, this clearance is approximately 0.8mm.The presence of sealing gaps is the root cause of medium leakage, which not only reduces the working fluid but also decreases the overall efficiency of the steam turbine. Studies indicate that leakage at this location can account for 22%-33% of the turbine's total losses [1, 2]. To minimize leakage losses and enhance the efficiency of the steam turbine unit, the primary technical solution currently involves optimizing and enhancing the sealing structure.While damping seals and louver seals are used in steam turbines, labyrinth seals have gained wider adoption due to their comprehensive advantages in reliability, maintenance convenience, and cost-effectiveness [3]. Compared to traditional louver seals, brush seals reduce leakage by 50%-90% [4, 5]. When used as a replacement in critical components, they enhance engine power by 1%-3% while cutting fuel consumption by 3%-5% [6]. Beyond sealing performance, brush seals also offer lightweight design, low wear, simple structure, and high safety [7].

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2. CORRELATION STUDY

In the field of heat transfer numerical research, Atkinson et al. [8] demonstrated that the frictional heat generation at the brush-rotor interface significantly influences the wear rate and leakage volume of seals. Hendricks et al. [9] pioneered the theoretical calculation method for frictional heat in brush seals. Owen et al. [10] further refined the frictional heat model, providing a comprehensive theoretical analysis of heat transfer processes in brush seals, rotating shafts, and leakage airflow systems.Demiroglu et al. [11] developed a closed system of equations to precisely calculate frictional heat generation. Meanwhile, infrared thermal imaging technology was employed to visualize the temperature field in key regions of the rotor surface and rear baffle. Additionally, a finite element model was utilized to numerically calculate the heat flux density on the rotor surface.Based on the needle-rib structure model, Dogu et al. [12] analyzed the heat transfer mechanism of brush seals and conducted numerical studies on temperature field distribution characteristics using a porous medium model.Qiu Bo et al. [13-14] conducted numerical simulations on the thermal performance of brush seals using a porous medium model, and established a dual-energy equation model for brush seals. Chai Baotong et al. [15] systematically investigated the effects of installation interference, pressure differentials, and rotational speeds on the thermal performance of brush seals through the porous medium model.Zhang Yuanqiao et al. [16] By integrating CFD and FEM models, this study conducted a multiphysics coupling analysis of heat transfer characteristics in brush seals, treating leakage flow parameters as coupling variables between brush airflow loads and frictional heat generation. Based on a three-dimensional porous medium model and nonlinear contact algorithms in finite element analysis, Ma Dengqian et al. [17] developed a model for brush deformation and frictional heat generation. Their research demonstrated that slotted rear baffles effectively reduce the system's maximum operating temperature.Sun Dan et al. [18] conducted in-depth research on the impact of pressure differential, rotational speed, interference fit, and heat flux density on the temperature extremes of brush filaments using a fluid-solid coupling model. Zhang Guoqiang et al. [19] performed numerical simulations on the heat transfer characteristics of a two-stage series brush seal, systematically analyzing how parameters such as pressure differential, rotational speed, and interference fit influence the temperature extremes of the brush filaments.

3. GEOMETRIC MODEL AND PARAMETERS

3.1. Geometric Model

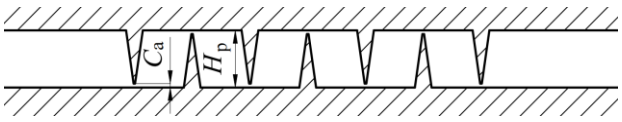


Figure 1. Schematic diagram of the staggered tooth labyrinth seal

Figure 1 illustrates the structural diagram of an interlocking tooth labyrinth seal for a steam turbine's low-pressure cylinder shaft end. At the stator seal tooth position, the sealing tooth is replaced with a brush seal, where the brush wire bundle and baffle are welded together. The front and rear baffles are welded to the labyrinth seal ring. The modified structure is shown in Figure 2 (which replaces the first upstream labyrinth tooth with a brush seal). The interlocking tooth labyrinth-brush seal retains the original installation dimensions and does not affect the positioning of other components. This upgrade directly replaces the original interlocking tooth labyrinth seal while enhancing sealing performance.

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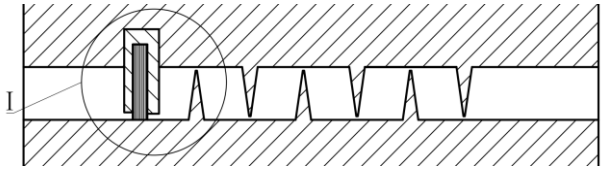


Figure 2. Overall schematic diagram of the staggered tooth labyrinth-brush seal

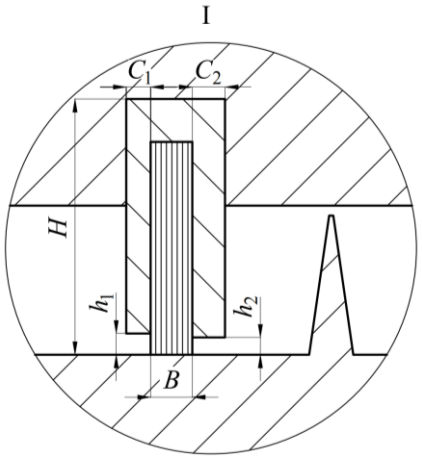


Figure 3. Diagram of the local structure of the brush seal

3.2. Parameters

Table 1. Structural parameters of staggered tooth labyrinth-brush seal

Parameter	WIHS
Labyrinthine teeth and rotor space (Ca /mm)	0.5
Cavity height (Hp /mm)	7
labyrinthine tooth number (n1 /mm)	7
rotor diameter (D /mm)	150
outer diameter of brush ring (Dob /mm)	170
inner diameter of brush ring (Dib /mm)	150
Brush diameter (d /mm)	0.076
front baffle height (h1 /mm)	1
rear baffle height (h2 /mm)	0.8
Brush arrangement angle (β /°)	45°
Brush bundle thickness (B /mm)	2
Front baffle thickness (C1 /mm)	1.1
rear baffle thickness (C2 /mm)	1.5
shrink range (δr /mm)	0.2
Brush height (H /mm)	10

The geometric parameters of brush wires significantly influence the overall performance of sealing components. The brush wire diameter, as a critical design factor, directly affects both the sealing performance of the structure and the stiffness of the brush wires. Experimental results indicate that excessively long brush wire lengths lead to increased leakage and uneven contact pressure distribution. In this study, a brush wire length of 10mm was selected. The optimal range for the tilt angle is typically controlled within 30-60 degrees, and the design adopts a 45-degree tilt angle for the brush wires.

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The parameters of the sealing structure are shown in Table 2, where the inlet and outlet pressures are both relative pressures.

Table 2. Condition parameters

Parameter	Numerical value
Rotational Speed (rad/min)	10000
Pressure Inlet (Mpa)	10
Pressure Outlet (Mpa)	0.1
Exit temperature(℃)	15

4. HEAT TRANSFER CONTROL EQUATION

Because the sealing structure of this paper has no radial dynamic displacement or thermal expansion deformation of the rotor, the sliding friction heat effect between the working end of the brush and the surface of the rotating shaft is the only thing to be considered in the stable operation state.

In the study of the heat transfer characteristics of brush seal friction, the frictional heat of the brush bundle and the rotor mainly depends on the frictional force and the contact line velocity of the frictional surfaces produced by the high-speed friction. According to this, the expression of the frictional heat Q can be obtained:

$$Q = F_f \cdot V = \mu_1 \cdot F_n \cdot V \tag{1}$$

Where F_f is the contact friction force; F_n is the normal contact force between the brush bundle and the rotor; μ_1 is the friction coefficient between the brush bundle and the rotor, set to 0.3; V is the relative contact line velocity of the friction surfaces.

The contact wire velocity between the brush tip D and the rotor surface N can be expressed as:

$$V = \frac{2\pi N}{60}\frac{D}{2} \tag{2}$$

Where D is the rotor diameter, D_i is the inner diameter of the brush ring, and δ_r is the interference fit of the brush wire.

In zero clearance fit, $D=D_i$; in interference fit, $D=D_i+\delta_r$. As analyzed in reference [20], the normal contact force F_{ni} between the brush wire's free end and the rotor during interference fit can be calculated using the following parameters: brush wire diameter d, arrangement angle β, length l, elastic modulus E of the material, and interference fit clearance δ_r. The derived expression is:

$$F_{ni} = \frac{3\pi}{64}\frac{Ed^4}{l^3 \sin^2\beta}\delta_r \tag{3}$$

To facilitate the representation, the equivalent stiffness of the brush is introduced to indicate the pressure change when the brush tip undergoes radial displacement. The expression for the equivalent stiffness is as follows:

$$K_{eff} = \frac{F_{ni}}{\delta_r} = \frac{3\pi}{64}\frac{Ed^4}{l^3 \sin^2\beta} \tag{4}$$

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The contact area S between the brush bundle and the rotor is related to the rotor diameter D and the brush bundle thickness B. The expression for the contact area is:

$$S = B \bullet \pi \bullet D \tag{5}$$

Heat flux density of friction:

$$q = Q / S \tag{6}$$

The heat transfer analysis model for brush seals, developed within the theoretical framework of porous media, is fundamentally based on the use of equivalent thermal conductivity parameters to comprehensively represent the heat transfer interactions between brush fibers and pores. This parameter holistically captures the thermal conduction dynamics between the solid phase (brush fibers) and gas phase (pores) within the brush bundle medium region. The energy conservation equation can be expressed as [21]:

$$\frac{\partial\left(\rho c_{\mathrm{p}} u_{\mathrm{i}} T\right)}{\partial x_{\mathrm{i}}} = \frac{\partial}{\partial x_{\mathrm{i}}}\left(k_{\mathrm{eff}} \frac{\partial T}{\partial x_{\mathrm{j}}}\right) + S_{\mathrm{T}} \tag{7}$$

In the formula: x_i denotes the coordinate direction; μ_i represents the velocity in the x_i direction; x_j is the viscous stress tensor of the gas; c_p is the specific constant pressure heat capacity of the gas; k_{eff} is the effective thermal conductivity of the porous medium; S_T is the heat source term.

The effective thermal conductivity, as a key parameter characterizing the heat transfer performance of porous media, comprehensively reflects the coupled effects of thermal conduction between the solid skeleton and pore spaces within the medium. When the isotropic assumption is satisfied, the thermal conductivity of the medium exhibits uniform characteristics across different spatial dimensions (radial, axial, and circumferential). In this case, the equivalent thermal conductivity can be characterized using a scalar form. Its mathematical expression can be constructed as follows:

$$k_{\mathrm{eff}} = \varepsilon_{\mathrm{v}} k_{\mathrm{f}} + \left(1 - \varepsilon_{\mathrm{v}}\right) k_{\mathrm{s}} \tag{8}$$

In the formula: k_{eff} is the volume average of the solid thermal conductivity in the brush bundle porous medium region and the fluid thermal conductivity in the voids; ε_v is the bulk porosity of the brush bundle; k_f is the thermal conductivity of the fluid; k_s is the thermal conductivity of the brush solid.

The porous medium model in this study employs an anisotropic framework, where the effective thermal conductivity varies between axial and radial directions. When heat transfers along the axial and circumferential paths of the brush fibers, it is equivalent to a series connection of the thermal conductivity of the fibers and their interstitial spaces. For radial heat transfer, the system behaves as a parallel connection of these thermal conductivities. In this configuration, the effective thermal conductivity is no longer a scalar quantity.

5. INFLUENCE OF STRUCTURAL PARAMETERS ON HEAT TRANSFER CHARACTERISTICS

The evaluation of heat transfer performance in brush seals involves multiple metrics. When temperature differentials exist between the sealing structure's front and rear, key indicators include heat flux, heat dissipation ratio, and temperature drop across the brush bundle. In cases

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of internal heat generation within the seal structure, the peak temperature typically serves as the primary metric. Appropriate evaluation criteria should be selected based on specific application scenarios and research objectives. Given the internal heat generation in this sealing structure, the peak temperature between the brush bundle and rotor is adopted as the evaluation standard to demonstrate the variation curves of the interdigitated tooth labyrinth-brush seal under different structural parameters, operating conditions, and installation configurations.

The following figure shows the influence of the parameters such as the diameter of the brush, the height of the front baffle, the height of the rear baffle, the thickness of the brush bundle, and the arrangement angle of the brush on the heat transfer characteristics of the interlocking tooth labyrinth-brush seal under the pressure difference of 0.05 Mpa, 0.1 Mpa, 0.2 Mpa, and 0.3 Mpa.

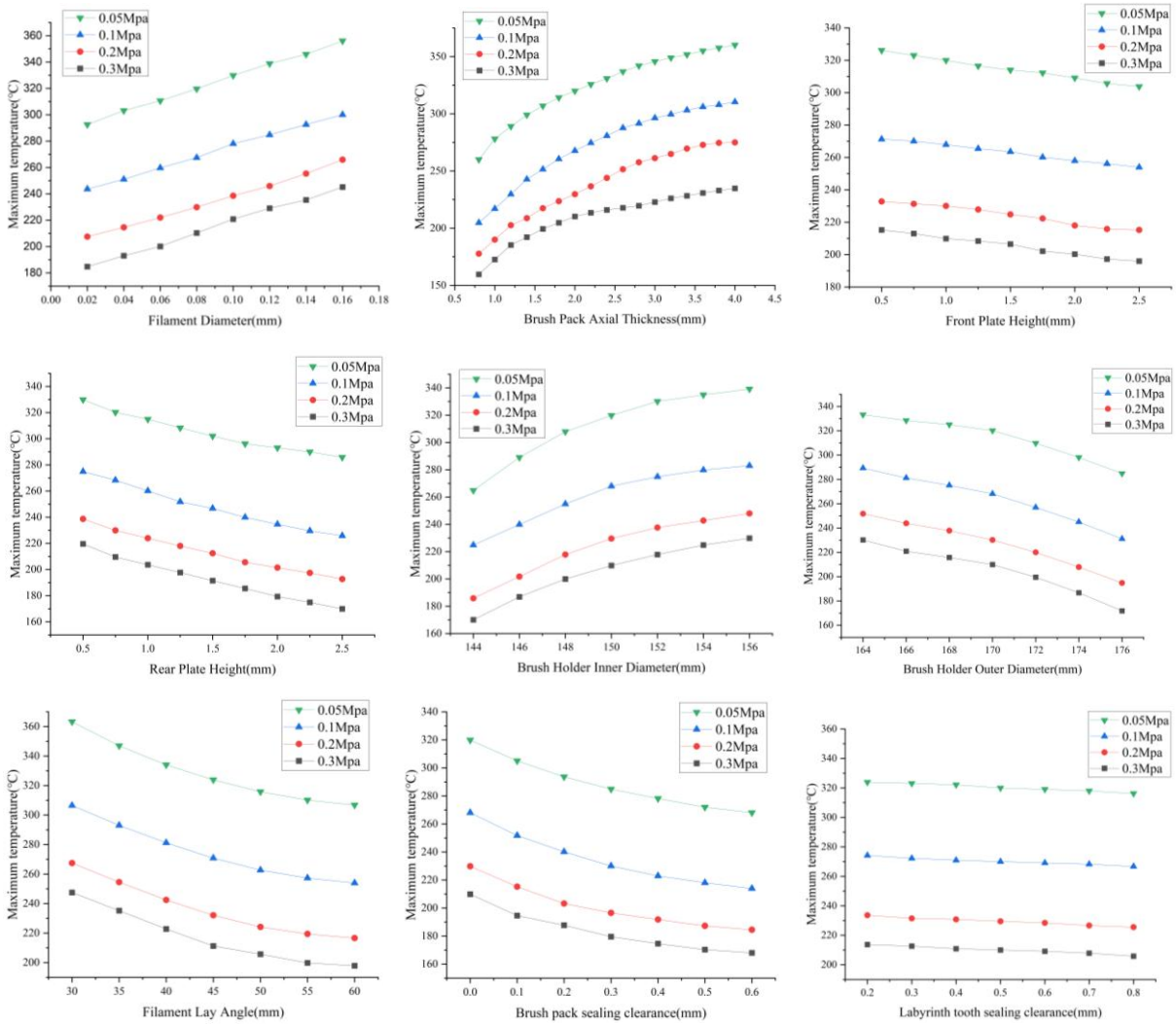


Figure 4. Effect of structural parameters on maximum temperature

The maximum temperature increases with the growth of brush diameter, exhibiting a nearly linear relationship. According to the heat transfer control equation, the frictional heat and the normal pressure between the brush and rotor are directly related: the brush diameter directly affects the normal pressure at the brush's free end contacting the rotor, thereby influencing the generated frictional heat. When the brush diameter remains constant, the maximum temperature decreases with increasing pressure differentials. This occurs because higher pressure differentials lead to greater fluid leakage, which carries away more heat, ultimately resulting in lower maximum temperatures.

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The maximum temperature increases with brush bundle thickness, as thicker bundles reduce leakage and generate more frictional heat. However, after reaching approximately 3.0mm, the temperature rise rate slows down, indicating that beyond this point, the thickness of the brush bundle has minimal impact on the maximum temperature of the interdigitated tooth labyrinth-brush seal.

The maximum temperature decreases with the increase of the height of the front or rear plate, and the relationship is approximately linear. Because the leakage increases with the increase of the height of the baffle, but the frictional heat is not obviously changed, the more fluid is leaked, the more heat is taken away, and the maximum temperature will decrease.

The maximum temperature increases with the enlargement of the brush ring's inner diameter. As the inner diameter increases, both the leakage rate and the length of the brush hairs decrease. According to the friction heat formula, a shorter brush hair length results in higher friction heat, which explains the temperature rise. When the inner diameter approaches 152mm, the temperature increase rate slows down, indicating that the impact of the inner diameter on the maximum temperature of the interlocking tooth labyrinth-brush seal diminishes at this point.

The maximum temperature decreases as the outer diameter of the brush ring increases. This occurs because larger outer diameters lead to increased leakage and longer brush hairs. According to the friction heat formula, longer brush hairs result in reduced friction heat, which explains the temperature drop. When the outer diameter approaches 170mm, the rate of temperature decrease accelerates, indicating that beyond this point, the outer diameter becomes a more significant factor affecting the maximum temperature of the interlocking tooth labyrinth-brush seal.

The maximum temperature decreases as the brush arrangement angle increases. This occurs because the frictional heat generated by sealing diminishes with increased brush arrangement angles, where the direct impact on thermal generation outweighs the temperature rise caused by leakage volume. Consequently, the maximum temperature declines with increasing brush arrangement angles. When the angle approaches 50 degrees, the rate of temperature reduction slows down, indicating that beyond this threshold, the influence of brush arrangement angle on the maximum temperature of the interlocking tooth labyrinth-brush seal diminishes.

The maximum temperature decreases as the brush bundle sealing gap increases. This occurs because larger gaps result in greater leakage, which carries away more heat from the fluid, thereby lowering the peak temperature. When the sealing gap approaches 0.3mm, the rate of temperature reduction slows down, indicating that the gap's influence on the maximum temperature of the interdigitated tooth labyrinth-brush seal diminishes at this point.

With the increase of the labyrinth seal clearance, the maximum temperature decreases slightly, but the overall change is not significant. This is because as the labyrinth seal clearance increases, the leakage volume increases slightly, without affecting the magnitude of frictional heat generation, thus the maximum temperature increases slightly.

6. INFLUENCE OF OPERATING PARAMETERS ON HEAT TRANSFER CHARACTERISTICS

The following figure shows the effects of the parameters of pressure difference, speed and ambient temperature on the heat transfer characteristics of the interlocking tooth labyrinth-brush seal under the height of the rear plate of 0.75mm, 1mm, 1.25mm and 1.5mm.

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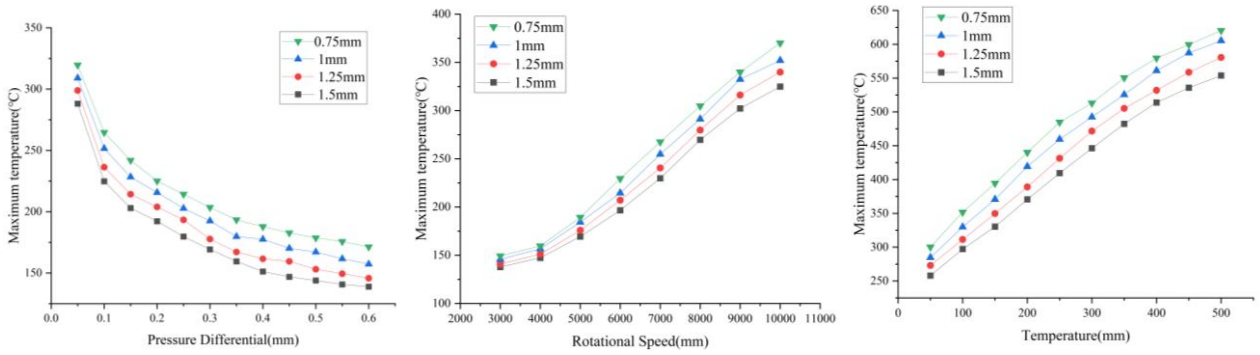


Figure 5. Effect of working parameters on maximum temperature

The maximum temperature decreases with the increase of pressure difference. This is because the pressure difference serves as the primary driving force for fluid flow. As the pressure difference increases, the leakage rate significantly rises, and the heat carried away by the leaking fluid becomes more substantial, resulting in a decrease in the maximum temperature with increasing pressure difference.

The maximum temperature increases with rotational speed. As the heat transfer formula indicates, the shaft speed directly affects the generated frictional heat, resulting in a significant rise in the maximum temperature when speed increases. When the speed is below 5000 rad/min, the temperature increases more slowly with speed. However, once the speed exceeds 5000 rad/min, the temperature rises rapidly and linearly.

The maximum temperature increases with rising ambient temperature. This occurs because the inlet temperature is higher, meaning the inlet fluid temperature is elevated. Changes in ambient temperature do not affect the amount of frictional heat generated. The higher the fluid temperature, the less heat is exchanged between the fluid and the maximum temperature point. Simultaneously, increased ambient temperature reduces leakage. These two factors collectively contribute to the influence of ambient temperature on the maximum temperature.

7. CONCLUSION

This study systematically investigates the heat transfer characteristics of interlocking tooth labyrinth-brush seals from two primary perspectives: structural parameters and operating conditions. Through the development of a rational numerical model and experimental validation, it establishes the highest temperature as the key metric for evaluating thermal performance. The research quantitatively delineates how varying structural parameters and operating conditions influence the sealing system's heat transfer behavior. The findings demonstrate that:

(1) The effects of brush diameter, brush bundle thickness, brush ring inner diameter, ambient temperature, and rotational speed on the maximum temperature are positive feedback; the effects of brush arrangement angle, front plate height, rear plate height, brush ring outer diameter, sealing gap, and pressure difference on the maximum temperature are negative feedback; the effect of labyrinth seal gap on the maximum temperature is not significant.

(2) The change of the maximum temperature is mainly caused by two aspects: one is the change of the structure parameters or working condition parameters directly leads to the change of the frictional heat; the other is the change of the structure parameters or working condition parameters leads to the increase or decrease of the leakage of the seal, which indirectly causes the change of the temperature.

The results not only deepen the understanding of the thermal management mechanism of the interlocking tooth labyrinth-brush seal, but also provide an important theoretical basis and data support for the optimization design and engineering application of this kind of seal structure.

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