Study on the Influence of High Temperature and High Speed Working Conditions on the Leakage Characteristics of Flexible Bristle Brush Seal

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Abstract. Compared with the metal bristles, the flexible bristles can reduce the heat and wear caused by the friction between the bristles bundle and the rotor, and improve the working stability and working life of the brush seal. In this paper, the L-shaped numerical mode is established to analyze leakage characteristics of flexible bristle brush seal. The flow field of brush seal is analyzed, and the effects of differential pressure, rotating speed and temperature on the leakage rate were studied as well. The leakage rate of the flexible bristles brush seal under different pressure and rotating speed was measured and compared with the numerical results. The results show that: Pressure drop mainly occurred in bristles pack; Small pressure difference, high temperature and high speed can improve the sealing performance, which, shows that the flexible bristles brush seal can well adapt to high temperature and high speed conditions; The leakage rate after running-in is slightly lower than that before running-in, and has better sealing performance, and the brush bristles arrangement is complete and tight after test, indicating that the flexible bristle brush seal has the ability of lasting and stable seal; The error between experimental results and numerical simulation results is less than 10%, and the variation trend is consistent, confirming the accuracy of the leakage rate obtained by numerical simulation.

Keywords. Flexible bristle brush seal, Porous media, Experimental investigation, Leakage characteristics

1. Introduction
Sealing technologies commonly used in rotating machinery such as fans and aircraft engines include brush seals, dry gas seals, floating ring seals, etc. [1-5] Brush seal is a flexible contact type dynamic seal device. As a new type of seal, it has developed rapidly in recent years. The flexible bristle of the brush seal can well adapt to the radial runout of the rotor, which can significantly improve the sealing performance and reduce leakage to 10%~20% of the traditional labyrinth seal [6].

The bristle materials used in brush seal are mostly cobalt-based and nickel-based high-temperature superalloy materials [6-9]. High-intensity friction between the metal bristle and the rotor will increase the temperature in the seal cavity and cause serious bristle wear, thus reduce the stability and operating life of brush seals. Brush seals are often applied in the conditions of high temperature and high speed. With the development of sealing technology, brush seals have also begun to choose some non-metallic bristle materials, such as aramid fibers [10-11]. Non-metallic bristle can reduce the overall weight of the brush seal, the temperature rise of the seal cavity, and friction and wear of the bristles, so it can meet the high requirements of modern engineering technology better. Therefore, it is particularly important to study the leakage flow characteristics and mechanism of flexible bristle brush seal under the conditions of high temperature and high speed.

Many scholars have done a lot of research on the leakage flow characteristics of brush seals. Bayley and Long [12] used a linear Darcian model of porous media which only considered the viscosity
resistance of the bristle tow to the fluid, and numerically predicted the static leakage of the brush seal. Dogu et al. [13] applied the integral porous media model to numerically predict the leakage of brush seals with clearance and without clearance. Sun Dan et al. [14] numerically predicted the relationship between leakage and pressure ratio and velocity by using a fluid-solid coupling model considering the deformation of the bristle, and he verified it by experiments. Li Shuangxi et al. [15] comprehensively studied the leakage flow characteristics of end face brush seals by using numerical prediction of porous media model and combining it with experimental verification. Zhang yuanqiao et al. [16] used the porous media model to numerically analyze the leakage flow characteristics of brush seal with different back plate structures.

The predecessors mainly studied metal bristle brush seals, but they paid less attention to non-metallic bristle materials, and rarely considered the working conditions of high temperature and high speed of brush seal. In this paper, a carbon fiber flexible bristle material is selected, and the L-shaped numerical model of the leakage flow characteristics of the brush seal is established based on the modified RANS equation and energy equation which solve the Non-Darcian porous media model. This paper analyzes the effects of pressure difference, temperature and velocity on the leakage flow characteristics of carbon fiber brush seals. The leakage rates of carbon fiber brush seals under different pressure differences and rotational speeds are measured experimentally. The experimental results are compared with numerical simulation results to analyze the leakage performance of carbon fiber brush seals under the conditions of high temperature and high speed to meet the requirements of practical engineering applications better.

2. Structure and working principle
As is shown in Figure 1, the brush seal is composed of a front plate, a back plate, and a bristle pack. The front plate is mainly used to fix and protect the bristle pack, and the back plate is used to support the bristle pack. The two fit the bristle pack by interference fitting. The bristle contacts with the rotor to form a sealed interface. The irregularly arranged bristle generate irregular flow channels, causing the fluid flowing through the area of the bristle pack to flow irregularly, increasing the flow resistance, generating a self-sealing effect, and achieving a sealing effect.

Fig.1 Schematic diagram of brush seal

The interference bristles will rub against the rotor, and the heat generated by the friction will increase the temperature in the seal cavity, affecting the stability of the brush seal operation. At the same time, the bristle wear caused by the friction between the bristle and the rotor will seriously reduce the operating life of the brush seal.
Figure 2 is a physical map of carbon fiber brush seal. The bristle is carbon fiber material. The carbon fiber bristle has the characteristics of high thermal conductivity, wear resistance and high temperature resistance. The flexible carbon fiber bristle can reduce the radial contact force with the rotor and reduce friction heat and friction wear. The flexible carbon fiber bristle has a strong recovery ability after deformation, which can well adapt to the radial runout of the rotor, alleviate the hysteresis effect and reduce leakage.

3. Numerical method and calculation model

3.1 Porous Media Model
The small diameter of bristle in brush seals and the uncertainty of the bristle arrangement make it difficult to solve the flow field inside the bristle pack. In the numerical study of the leakage characteristics of brush seals, the most widely used and reliable method is to treat the bristle pack area into a porous medium to simplify the flow of fluid inside the bristle pack. The upstream and downstream fluid in the bristle pack area is a compressible turbulent flow. The bristle pack area is processed into anisotropic materials, and the momentum equation and the continuity equation are calculated simultaneously. As shown in formulas (1) and (2):

$$\frac{\partial p}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0 \tag{1}$$

$$\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = \frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_{ij}} \tag{2}$$

When the fluid flows through the bristle pack, it will be hindered by the bristle, causing a pressure drop in the fluid, and the density of the gas in the bristle pack will also change. Therefore, the fluid flowing inside the bristle pack is treated as an ideal compressible gas, and the state equation of ideal gas is as follows:

$$p = \rho RT \tag{3}$$

When the fluid flows through the bristle pack, the resistance of the bristle to the fluid is treated as an additional momentum source term. The momentum equation is amended as follows:

$$\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = \frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_{ij}} + F_i \tag{4}$$

In the equation: $F_i = (\Sigma_{j=1}^2 D_{ij} \mu_j + \Sigma_{j=1}^2 C_{ij} \frac{1}{2} \rho |v_j| v_j)$. $F_i$ is the momentum source term; $D_{ij}$ is the viscous resistance coefficient matrix in the bristle pack region; $C_{ij}$ is the inertial resistance coefficient matrix in the bristle pack region.

The viscous resistance coefficient $\alpha_i$ and the inertia resistance coefficient $\beta_i$ can be measured through experiments or can be obtained through the following empirical formulas [17]:
\[ \alpha_i = \frac{150 (1-\varepsilon)^2}{d^2 \varepsilon^3} \]  

(5)

\[ \beta_i = \frac{7}{2d} \varepsilon^3 \]  

(6)

In the two equations, \( d \) is the diameter of the bristle; \( \varepsilon \) is the porosity of the bristle pack.

Porosity is defined as the ratio of the void volume in the porous medium to the total volume of the porous medium. The porosity of the bristle pack \( \varepsilon \) can be defined by the bristle arrangement density \( N \).

When the bristle arrangement does not have a radial tilt angle:

\[ \varepsilon = 1 - \frac{\pi d^2 N}{4L} \]  

(7)

In the equation, \( d \) is the diameter of the bristle; \( N \) is the bristle arrangement density; \( L \) is the thickness of bristle pack.

3.2 Calculation Model

Figure 3 is the structural diagram of the L-shaped carbon fiber brush seal. It can be seen that the carbon fiber brush seal and its internal fluid are symmetrical in the circumferential direction. Therefore, the calculation model is simplified by selecting the arc of 1° in the circumferential direction to reduce computing time.

![Fig.3 Structural diagram of L-shaped carbon fiber brush seal](image)

The numerical calculation model of flexible bristle brush seal is established, as shown in Figure 4 (a). The model includes a common fluid region and a bristle pack region treated as a porous medium. There is a retracting effect when the fluid enters and exits the brush seal, and a part of the fluid domain needs to be added at the entrance and exit of the brush seal. The calculation model is meshed, as shown in Figure 4 (b). The area framed by red wire is the bristle area. Through mesh independence verification, as shown in Table 1, the optimal number of grids is determined to be 259470.

| Number of mesh | Leakage rate/g·s\(^{-1}\) |
|---------------|--------------------------|
| 14777         | 0.24                     |
| 31203         | 0.253                    |
| 127470        | 0.367                    |
| 259407        | 0.375                    |
| 595150        | 0.375                    |
| 2001700       | 0.376                    |
The values of the geometric parameters for the numerical calculation of the brush seal and the attribute of the bristle material are shown in Table 2, where the bristle interference is reflected by the bristle length.

### Table 2: Main structural dimensions

| Classification                  | Value/mm |
|--------------------------------|----------|
| Front plate clearance \( h_f \) | 2.5      |
| Back plate clearance \( h_b \)  | 0.8-1.4  |
| Bristle diameter \( d \)        | 0.006    |
| Bristle pack thickness \( B \)  | 1.0      |
| Bristle length (interference) \( s \) | 1.0 |
| Sealing diameter \( D \)        | 80.6     |

Numerical calculation and the bristle material properties of the test piece are shown in Table 3.

### Table 3: Carbon fiber bristle material properties

| Classification                        | Value          |
|---------------------------------------|----------------|
| Thermal conductivity \( \lambda \)/W \( \cdot \) (m \( \cdot \) k)\(^{-1} \) | 35.127         |
| Density \( \rho \)/g \( \cdot \) cm\(^{-3} \)               | 1.802          |
| Specific heat capacity \( C \)/J \( \cdot \) (kg \( \cdot \) k)\(^{-1} \) | 753.44         |
| Tensile modulus \( M \)/GPa            | 226            |
| Coefficient of thermal expansion \( \gamma \)/k \(^{-1} \)   | -0.56\times10^{-6} |

#### 3.3 Boundary setting

In this section, Fluent software is used for analysis and calculation, and the standard k-\( \varepsilon \) equation turbulence model is used to simulate ordinary fluid domains. The bristle pack adopts the laminar flow model and the porous medium model. The whole model is calculated by the simple algorithm, and the ideal compressible air is selected as the sealing medium. The wall surface in contact with the rotor is set as a rotation boundary. The wall surface rotating in the circumferential direction of the rotor is set as a periodic boundary. The remaining wall surfaces are set as non-slipping boundary. Table 4 shows the specific values of the boundary conditions of numerical simulation.

### Table 4: Boundary condition

| Classification            | Value   |
|---------------------------|---------|
| Inlet pressure \( P_i \)/MPa | 0-1.0   |
| Outlet pressure \( P_o \)/MPa | 0       |
3.4 Numerical Analysis Results

3.4.1 Flow Field Analysis Results. The structural parameters and working condition parameters of flow field analysis numerical model of the carbon fiber brush seal are shown in Table 5.

| Classification                          | Value   |
|----------------------------------------|---------|
| Front plate clearance $h_f$/mm         | 2.5     |
| Back plate clearance $h_b$/mm          | 1.5     |
| Bristle diameter $d$/mm                | 0.006   |
| Bristle thickness $B$/mm               | 1.0     |
| Bristle length (interference) $s$      | 1.0     |
| Inlet pressure $P_j$/MPa               | 0.3     |
| Outlet pressure $P_c$/MPa              | 0       |
| Rotor speed $n/r\cdot \text{min}^{-1}$ | 10000   |

The velocity nephogram of the fluid is shown in Figure 5. The velocity of the fluid hardly changes before entering the bristle pack region, and increases gradually after entering the bristle pack region. The velocity increases to the maximum at the exit of the back plate. After the fluid enters the bristle pack region, the flow channel suddenly narrows, which gradually increases the flow velocity. When the fluid reaches the exit of the back plate, the flow channel and flow direction suddenly change, further accelerating the fluid flow and making the flow velocity reach the maximum. The friction between the bristle and the rotor generate heat, which increases the temperature and accelerate the flow of fluid as well.

Figure 6 is a part of the velocity vector map of the fluid. When the fluid passes through the bristle pack, the flow velocity and direction of the fluid is deflected due to the sudden narrowing of the flow channel in the bristle pack, and the radial velocity increases rapidly. At this time, the radial force that the bristle pack receives increases, pressing the bristle toward the rotor, increasing the friction and wear of the bristle and the rotor, and causing the brush seal to have a “blow down effect”. At the same time, the porosity of the bristle pack also decreases, reducing the leakage rate of the brush seal.
It can be seen from the pressure nephogram in Figure 7, that the overall pressure decreases along the fluid flow direction. Because the bristle pack area has a large resistance to the fluid flow, there is a significant pressure drop. There is no flow resistance in the inlet area and the outlet area, so there is no significant pressure drop. This shows that the characteristics of the bristle pack have a very important influence on the sealing performance.

3.4.2 Leakage rate analysis results. Figure 8 is the varying curves of the leakage rate of brush seals with four different back plate gaps with pressure difference. As the pressure difference increases, the leakage rates increase approximately linearly. When the pressure difference increases from 0.05 MPa to 0.7 MPa, the curve of the brush seal leakage rate with a back plate gap of 0.8 mm varies significantly less than that of the brush seal with a back plate gap of 1.4 mm. It indicates that for the brush seal with a wider back plate gap, the leakage rate will increase faster when pressure difference increases. Reducing the pressure difference at both ends of the brush seal is an important measure to improve the sealing performance of the brush seal. Where conditions permit, multi-level brush seals can be arranged to share the pressure difference at different levels, so that each level of the brush seal is in a differential environment with lower pressure.

Figure 9 is the varying curves of the leakage rate of brush seals with three different back plate gaps with temperature. As the temperature increases, the leakage rates decrease approximately parabolically, because when the temperature increases, the gas density decreases and the viscosity increases, which increases the flow resistance in the bristle pack area and gradually reduces the leakage rate. When the temperature continues to rise, the density change and viscosity change of the gas will reach the peak, the flow resistance in the bristle pack area will no longer change, and the leakage rate will stabilize.
When the temperature increases from 200 K to 900 K, the curve of the brush seal leakage rate with a back plate gap of 0.8 mm changes significantly less than that of the brush seal with a back plate gap of 1.2 mm, indicating that for a brush seal with a wide back plate gap, the effect of the temperature change on the leakage rate will be greater.

![Graph showing the effect of rotor speed on leakage rate.](image)

**Fig. 10** Effect of rotor speed on leakage rate

Figure 10 is the changing curve of the leakage rate with the speed under two pressure differences. With the increase of the rotor speed, the leakage rates both decrease slightly, and finally stabilize, and the degree of the decline is not obvious. Increasing the rotor speed can reduce the leakage rate of the brush seal, but it has little effect.

**4. Experimental research**

**4.1 Test device.**

Figure 11 shows a rotating test bench of the brush seal, which is mainly composed of a sealed chamber system, a power system, a gas path system, and a measurement system. The sealed chamber system includes chamber on low-pressure side, carbon fiber brush seal, etc. The power system includes inverter, motor, rotor, etc. The gas system includes compressor, pressure regulator, gas pipeline, etc. The measurement system includes pressure gauge, flowmeter, temperature thermocouple, etc.

![Test bench diagram](image)

**Fig. 11** Brush seal test bench

**4.2 Test process**

The compressor presses the air into the surge tank, and the pressure regulator is adjusted to the required pressure, and the high-pressure gas is injected into the sealed test chamber (high-pressure side) through the air inlet pipe, and the high-pressure gas leaks to the sealed leakage chamber (low-pressure side).
through the bristle pack. The leaked gas is discharged into the atmosphere through the exhaust pipe connected to the flowmeter, and the leakage gas is measured by the flowmeter. The test chamber and the leakage chamber are connected with a pressure gauge and a temperature thermocouple to measure the inlet and outlet pressure and temperature. In dynamic tests, the inverter can be adjusted to change the frequency, and then the required motor speed is obtained.

4.3 Test piece
The test piece for this test is a carbon fiber brush seal designed and processed independently. The front plate and back plate are made of S30408 stainless steel, and the bristle is made of carbon fiber, as shown in Figure 12.

![Fig.12 Brush seal test piece](image)

The structural parameters of the carbon fiber brush seal for the test are shown in Table 6. The test measured the leakage rates of carbon fiber brush seals before and after running-in, and analyzed the influence of pressure difference and speed on the leakage of carbon fiber brush seals before and after running-in.

| Classification                        | Value /mm |
|---------------------------------------|-----------|
| Front and back plate outer diameter $D_1$ | 93.0      |
| Front and back plate inside diameter $D_1/D_2$ | 82.2      |
| Front and back plate clearance $h_d/h_b$ | 0.8       |
| Brush seal thickness                  | 4.5       |
| Bristle length (interference) $s$    | 1.2       |
| Bristle pack thickness $B$            | 0.6       |

4.4 Test results

![Fig.13 Effect of pressure difference on static leakage rate](image)

![Fig.14 Effect of rotational speed on leakage rate](image)
Figure 13 is the changing curve of the leakage rate with the pressure difference under the static condition. The trend of leakage rate of brush seals before and after the running-in is similar to that in numerical simulation, and both increase approximately linearly with the increase of the pressure difference.

Figure 14 is the varying curve of the leakage rate with the rotor speed under the pressure difference of 30 kPa. The change of the leakage rate before and after the running-in is approximately consistent with the numerical simulation, and both gradually decrease with the increase of the rotor speed, and finally stabilize.

In Figures 13 and 14, due to factors such as machining accuracy and mounting errors of the test pieces, the test values before and after the running-in are slightly larger than the simulated values. But the error remains within 10%, which proves the correctness of the theoretical simulation. Before the running-in of brush seals, the bristle does not adhere closely to the surface of the rotor, causing local gaps, resulting in a lower leakage rate and better sealing performance after running-in than before running-in, indicating that the flexible bristle brush seal has the potential for durable and stable sealing. However, before the pressure difference of 40 kPa in Figure 13 and the rotor speed of 5000 r·min⁻¹ in Figure 14, the leakage rate after running-in is higher than that before running-in. This was caused by the fact that the brush seal was not reassembled and the deformed bristle was not recovered in time during the test after running-in.

\[ \text{Fig. 15 Brush seal} \]

\[ \text{Fig. 15 Brush seal after test} \]

It can be seen from figure 15 that the bristle arrangement after the test is complete and tight, and there is no serious wear on the rotor surface, indicating that the brush seal has the ability to maintain good sealing performance for a long time.

5. Conclusion

1. The flow velocity of the fluid is low before entering the bristle pack region, and gradually increases after entering the bristle pack region, and the flow velocity reaches the maximum at the exit of the back plate. The pressure distribution in the upstream and downstream fluid regions is uniform, and the pressure drop mainly occurs in the bristle pack regions, which indicates that the bristle characteristics play an important role in sealing performance.

2. Small pressure difference, high temperature, and high rotor speed conditions can improve the sealing performance. The effect of rotor speed is lower, indicating that the flexible bristle brush seal can well adapt to high temperature and high rotor speed conditions and meet the requirements of practical engineering applications.

3. The test shows that the leakage rate after running-in is slightly smaller than that before running-in, and the sealing performance is better. The bristle arrangement is complete and tight after the test, which indicates that the flexible bristle brush seal has the ability of durable and stable sealing.

4. The variation trend of the measured leakage rate with pressure difference and rotor speed is the same as the numerical simulation result. The error between the test result and the numerical simulation result is less than 10%, and the changing trend is approximately the same, which fully confirms the accuracy of the leakage rate change law obtained by numerical simulation.
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