EFFECT OF STATOR RUB GROOVES ON LEAKAGE PERFORMANCE OF A STEPPED LABYRINTH SEAL

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ABSTRACT

With the numerical methods and experimental tests, the effect of stator wear on leakage performance of a labyrinth seal was investigated, and also, the leakage characteristics of the labyrinth seal with stator rub-groove were compared with those of original designs. At four clearances, nine pressure ratios and three groove width to depth ratios, the leakage rates and flow patterns inside the stepped labyrinth seals were numerically obtained. The results show that, in the wear condition, the deviation between the predicted leakage rate and measured value is within 5.58%. As the increase of clearance and stator wear level, the leakage rate is increased rapidly. If the relative rub-groove depth is fixed at 2.5, compared with the original designs, the leakage rates for the wear case at four clearances (1.2mm, 1.7mm, 2.2mm and 2.7mm) are increased by 79.6%, 78.9%, 80.9% and 90.6%, respectively. With the same geometry of rub-groove, the leakage rate is increased with the increase of pressure ratio. However, the relative discharge coefficient is almost kept constant. The existence of rub-groove in the labyrinth seal stator part changes the flow area near the clearance gap, thus the flow pattern inside the seal chamber are altered in contrast to the original design. Correspondingly, the leakage rate in the stepped labyrinth seal is affected by rub-groove geometry, significantly.

INTRODUCTION

The stepped labyrinth seal is widely used in gas turbine engine to control leakage flow in clearance involved with rotating and stationary parts. Due to the inevitable vibration, centrifugal force, thermal expansion and rotor misalignment, rub damage is commonly occurred in the stator part of labyrinth seal (see Fig.1), especially in the transition operation conditions, for example, starting, shutting down or varying load process. Since the leakage flow in the seal is an important influence factor of the turbomachinery efficiency [1], cooling quality [2], rotordynamics [3], and also heat transfer characteristic [4], investigations of the leakage variations in the stepped labyrinth seal with different stator rub grooves are quite useful for the stepped labyrinth seal design and performance evaluation.

To understand the effect of rub groove on leakage performance of the labyrinth seal, a number of researchers have investigated the rub damages in the seal stator under various conditions. Delebarre et al [5] measured the rub groove profiles in a labyrinth seal stator at different rotating speeds, it shows that the incursion speed has a pronounced effect on the rub groove geometry and contact force. Dogu et al [6] carried out a series of numerical simulations to investigate the effect of rub groove shape on the leakage characteristic of labyrinth seal. The results show that the discharge coefficient of the labyrinth seal is increased significantly after stator rubbing. Among different rub groove geometries, the leakage rate increases with triangular, rectangular, acute trapezoidal, isosceles trapezoidal and elliptical profiles successively. Zimmermann et al [7]
experimentally investigated the effect of stator rub on
discharge behaviors in the straight-through and stepped
labyrinth seals. It shows that the leakage rate is increased
significantly when rectangular rub-groove is occurred,
especially for the stepped labyrinth seal. Rhode et al [8-11]
analyzed the discharge behaviors in labyrinth seals with
different geometrical dimensions by numerical simulations.
It indicates that the leakage performance of labyrinth seal
with small clearance gap, large pitch length and moderate
step height is much sensitive to the dimensions of stator rub-
groove. Ambrosia et al’s [12] numerical results show that the
downstream rub-groove wall angle has a pronounced effect
on leakage rate and flow pattern in the stepped labyrinth seal.
Denecke et al [13] experimentally investigated the influence
of stator rub groove on labyrinth seal leakage by laser light-
sheet visualization methods. It shows that the stator rub
groove changes the carry-over effect in the labyrinth seal
thus affects the discharge behavior in the labyrinth seal.

From the above research, it is evident that the rub groove
profile and dimensions have pronounced effect on the
discharge behavior in the labyrinth seal. However, the
previous studies mainly focused on the effect of rub groove
profiles on the leakage rate in the labyrinth seal. A few
studies were concentrating on the effect of rub groove
dimensions on the leakage rate in different clearance gaps
and pressure ratios. Therefore, in this study, the experimental
tests and numerical predictions are combined to investigate
the leakage characteristic of the stepped labyrinth seal with
different degrees of rub damages at different conditions.

EXPERIMENTAL SETUP

The leakage experiments for the stepped labyrinth seal
with stator rub groove were carried out in the
Turbo machinery Laboratory at Xi’an Jiaotong University.
Fig.2 shows the geometrical parameters for the tested
labyrinth seal in the design case and wear case. The tooth
thickness is defined as \( b \), and the clearance gap is defined as
\( s \). For the rub groove, the width and depth of rub groove are
defined as \( b_N \) and \( t_N \).

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NUMERICAL METHODS

Geometrical model and meshes

In accordance with the experimental tests, the research
objective for the present study is the same with the tested
labyrinth seal, as shown in Fig.2. The geometrical
dimensions are listed in Table 1.

The maximum pressure of the air can be reached up to
0.7MPa, and the maximum mass flow rate is 0.2kg/s. The
length of the test section is 240mm, and the width is 300mm.
In the experimental tests, the clearance gap is kept at 1.2mm.
The groove width to depth ratio \( b_N / t_N \) is 10. Four
different relative groove depths, i.e. \( (t_N + s) / s \) is 1.5,
2.0, 2.5, and 3.0, are considered. A QWLJ-080 type turbine
rotor flow meter is used to measure the flow rate through the
test section. The averaged uncertainty of the flow meter is
\( \pm 1\% \). If the measured mass flow rate is lower than 0.02kg/s,
the uncertainty is \( \pm 2\% \). The inlet total pressure of the test
section is measured with a total pressure probe. The
uncertainty of piezoelectric transducer for the probe is 0.7‰.
The inlet temperature of the test section is measured with a
K-type thermo-couple. The uncertainty of the thermocouple
is \( \pm 1.5^\circC \).
factor of the mesh near the wall is set to be 1.2, and the wall adjacent cell distance is set to be 0.01mm.

Table 1 Geometrical dimensions of the stepped labyrinth seal

<table>
<thead>
<tr>
<th>parameter</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>pitch</td>
<td>30mm</td>
</tr>
<tr>
<td>tooth height</td>
<td>15mm</td>
</tr>
<tr>
<td>tooth thickness</td>
<td>1.5mm</td>
</tr>
<tr>
<td>seal length</td>
<td>320mm</td>
</tr>
<tr>
<td>step height</td>
<td>3.9mm</td>
</tr>
<tr>
<td>(s)</td>
<td>1.2mm, 1.7mm, 2.2mm, 2.7mm</td>
</tr>
<tr>
<td>(b_N / t_N)</td>
<td>2.5, 5.0, 10.0</td>
</tr>
<tr>
<td>((t_N + s) / s)</td>
<td>1.5, 2.0, 2.5, 3.0</td>
</tr>
</tbody>
</table>

Solution methods and boundary conditions

In this study, the commercial software ANSYS CFX 11.0 was adopted to solve the compressible 3D RANS equations for the stepped labyrinth seals. The air ideal gas model is adopted to simulate the property of the fluid in the labyrinth seal. For the discretizations of the advection term, a high resolution scheme \[14\] is adopted. To solve the turbulence flow information, the standard k-\(\varepsilon\) turbulence model combined with scalable wall function treatment \[14\] is used.

To comply with the experimental tests, the outlet static pressure is set to be the atmospheric condition, and the inlet total pressure is specified to be 1.02-1.55 times of the outlet static pressure depending on the pressure ratio. The inlet turbulence intensity is set to be 5%. Since there is no heat transfer involved in the present experimental tests, the adiabatic wall boundaries are assumed for all walls. As the root mean square residuals for the continuity equation, momentum equations, energy equation and turbulence equations are less than \(10^{-5}\), the iterations are considered to be converged.

Definitions

The pressure ratio is defined as

\[
\pi = \frac{p_{i,0}}{p_{s,1}} \tag{1}
\]

where, \(p_{i,0}\) and \(p_{s,1}\) are the inlet total pressure and outlet static pressure, respectively.

The discharge coefficient of the labyrinth seal is defined as

\[
C_D = \frac{\dot{m}_{\text{real}}}{\dot{m}_{id}} \tag{2}
\]

The ideal mass flow rate \(\dot{m}_{id}\) in the labyrinth seal is calculated by the Martin equation \[15\]. The realistic mass flow rate \(\dot{m}_{\text{real}}\) is derived with the CFD prediction.

\[
\dot{m}_{id} = \frac{Ap_{i,0}}{\sqrt{RT_{i,0}}} \sqrt{1 - \pi^{-2}} \left( n + \ln(\pi) \right) \tag{3}
\]

The relative discharge coefficient \(\xi\) is defined as

\[
\xi = \frac{C_{D,\text{nom w/rub-grooves}}}{C_{D w/o rub-grooves}} \tag{4}
\]

where, \(C_{D,\text{nom w/rub-grooves}}\) is the discharge coefficient for the labyrinth seal with rub groove, and \(C_{D w/o rub-grooves}\) is the discharge coefficient for the labyrinth seal without rub groove (design case).

Grid independency analysis

Table 2 Predicted leakage rates with four different mesh densities \((C_r=2.2\text{mm}, \ b_N / t_N = 5, (t_N + s) / s = 2)\)

<table>
<thead>
<tr>
<th>Nodes number</th>
<th>Leakage rate (kg/s)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>350k</td>
<td>0.1741</td>
<td>-2.03</td>
</tr>
<tr>
<td>700k</td>
<td>0.1757</td>
<td>-1.13</td>
</tr>
<tr>
<td>1.4million</td>
<td>0.1768</td>
<td>-0.51</td>
</tr>
<tr>
<td>2.8million</td>
<td>0.1777</td>
<td>-</td>
</tr>
</tbody>
</table>

The mesh density may have a significant influence on the numerical accuracy due to the discretization errors involved in the iteration process. Therefore, before the numerical simulations are performed, the effect of grid density on the predicted leakage rate in the stepped labyrinth seal is investigated. Here, four different meshes are generated in the mesh independency analysis process. The nodes numbers are selected to be 350k, 700k, 1.4million and 2.8million, respectively. The meshes are refined with the same increment ratio in three directions. Table 2 lists the predicted mass flow rates with four different mesh densities. It indicates that the predicted leakage rate is increased with
the increase of nodes. As the nodes number increases to 700k, the numerical error is about 1% compared with the extrapolation value. Therefore, in the present numerical approach, the final nodes number is selected to be 700k to balance the computational speed and numerical accuracy.

**Numerical methods validations**

To validate the present numerical methods, the predictions are also compared to the experimental data to analyze the numerical accuracy of this study. Fig.6 shows the leakage rates versus pressure ratio for the stepped labyrinth seal with rub grooves at four different cases. In general, the predicted values are in reasonable agreement with the measurements. However, for the larger pressure case (pressure ratio larger than 1.4), the predicted value is a bit lower than the measurements. As the depth of rub groove increases, the maximum deviations of the prediction from experimental data are about 5.58%, 2.24%, 2.88% and 4.98%, respectively.

To validate the repeatability of the experimental tests, the stepped labyrinth seal with the rub groove \((t_N + s)/s = 2.5\) was tested twice to see the difference between two experimental conditions. Fig.7 illustrates the test data for two experiments. It is seen that the two experiments are almost coincided to each other, and also the prediction values are agreeable with the measurements.

**RESULTS AND DISCUSSION**

![Fig.6 Leakage rate versus pressure ratio for the labyrinth seal with rub grooves \((s=1.2\text{mm}, \; b_N/t_N=10)\)](image1)

![Fig.7 Leakage rates versus pressure ratio in two experiments \((s=1.2\text{mm}, \; b_N/t_N=10, \; (t_N + s)/s = 2.5)\)](image2)

![Fig.8 Relative discharge coefficients \(\xi\) versus relative groove depth](image3)
Based on the numerical methods validations, the effects of rub groove geometries on the discharge behaviour of the labyrinth seal were investigated in detail with the CFD predictions.

Fig.8 provides the relations between relative discharge coefficient $\xi$ and relative groove depth $(t_N + s)/s$ for three $b_N/t_N$ and four clearances. It is clearly seen that most of $\xi$ increases with the increase of $(t_N + s)/s$ (degree of rub damage). At four clearances, the maximum $\xi$ are 2.62, 2.71, 2.32 and 2.20, respectively. At the fixed $(t_N + s)/s$, the leakage rate increases with the increase of groove width to depth ratio $b_N/t_N$. As $b_N/t_N$ increases from 5 to 10, the relative discharge coefficient is increased slowly.

To better understand why the relative discharge coefficients $\xi$ decreases with the increase of $(t_N + s)/s$ (degree of rub damage), Fig.9 illustrates the detailed flow fields in the seal chamber for $(t_N + s)/s = 2.5$ and $(t_N + s)/s = 3$. It is evident that, as the degree of rub damage increases (From (a) to (b)), the nominal clearance is increased, correspondingly. However, the vortex size in front of the clearance gap is increased, significantly. This flow pattern will lead to larger incidence angle $\alpha$ in the clearance. If incidence angle $\alpha$ is increased, $\dot{m} = \rho \cdot C_r \cdot V \cos \alpha$ in Eq.(5) will be decreased, correspondingly.

$$\dot{m} = \rho \cdot C_r \cdot V \cos \alpha \quad (5)$$

(a) $(t_N + s)/s = 2.5$

(b) $(t_N + s)/s = 3$

Fig.9 The flow pattern in two different rub geometries ($s=1.2\text{mm}$, $b_N/t_N = 10$)

Fig.10 plots the leakage rate versus pressure ratio for the labyrinth seals at four clearances. For different rub geometries, the leakage rate increases with the increase of pressure ratio. Note that the leakage rate increases rapidly especially at small pressure ratio condition. At a fixed pressure ratio, the leakage rate increases with the increase of sealing clearance and rub degree. At the maximum pressure ratio 1.55, compared with the original design, the leakage rates are increased by 79.6%, 78.9%, 80.9% and 90.6% for $(t_N + s)/s = 3$ case at four clearances, respectively.
Fig. 11 provides the relative discharge coefficients $\xi$ versus pressure ratio for the labyrinth seal with different rub geometries. It is interesting to note that, for each rub geometry, $\xi$ is almost constant with the pressure ratio. However, as the degree of stator wear increases ($\frac{(t_N+s)}{s}$ increases), $\xi$ is increased, correspondingly. For the small clearance gap ($s=1.2\text{mm}$), $\xi$ is increased rapidly for the larger ($\frac{(t_N+s)}{s}$ case. But as the clearance increases, $\xi$ is almost increased linearly with the increase of ($\frac{(t_N+s)}{s}$) (for example, $s=2.2\text{mm}$ and $s=2.7\text{mm}$). It is also noticed that, at the same degree of stator wear, $\xi$ is slightly increased with the increase of clearance. Another observation is that, for ($\frac{(t_N+s)}{s} = 1.5$ and $s=1.2\text{mm}$ case, the relative discharge coefficients $\xi$ is lower than 1.0. This means the leakage rate in the wear condition is lower than that in the design case.

Fig. 12 Streamlines in the labyrinth seal with different degrees of wear ($s=1.2\text{mm}$, $b_N/t_N = 2.5$)

To better understand the effect of rub damage on the flow patterns in the labyrinth seal, Fig. 12 plots the streamlines in the seal chamber with different rub geometries. Compared with the original design, the flow pattern in the wear case with ($\frac{(t_N+s)}{s} = 1.5$) rub groove changes slightly due to very small rub groove ($b = b_0$). As the rub groove increases from ($\frac{(t_N+s)}{s} = 1.5$ to ($\frac{(t_N+s)}{s} = 2.0$, two counter-rotating vortices are almost not affected by the variation of rub groove dimensions. However, as the rub groove increases from ($\frac{(t_N+s)}{s} = 2.0$ to ($\frac{(t_N+s)}{s} = 2.5$, the shapes of the two counter-rotating vortices vary significantly. Especially, the incidence angle of the leakage jet changes a lot in each
Therefore, the leakage rate is increased a lot from \((t_N + s)/s = 2.0\) to \((t_N + s)/s = 2.5\), which can also be seen from Fig.11 (a). As the rub groove increases from \((t_N + s)/s = 2.5\) to \((t_N + s)/s = 3.0\), the leakage flow in the first chamber almost impinges into tip gap with 0° incidence angle, which leads to an increase of the leakage rate in the labyrinth seal with rub groove a lot (See Fig.11 (a)).

CONCLUSIONS

In this study, the numerical methods combined with experimental tests were utilized to investigate the effect of stator rub groove on the leakage performance of labyrinth seal at four clearances, a range of pressure ratios and 12 rub dimensions. Major findings are as follows,

(1) The predicted mass flow rates in the stepped labyrinth seal with different rub geometrical dimensions are in reasonable agreement with the measurements. The maximum deviation between CFD result and test data is about 5.58%.

(2) The leakage rate in the labyrinth seal increases with the increase of clearance gap and degree of stator wear. At the maximum pressure ratio 1.55, compared with the original design, the leakage rates are increased by 79.6%, 78.9%, 80.9% and 90.6% for \((t_N + s)/s = 3\), \(b_N/t_N = 2.5\) at four clearance cases, respectively.

(3) With the same rub groove, as the pressure ratio increases, the leakage rate increases. However, at the fixed rub groove geometry, the relative discharge coefficients are almost constant with different pressure ratios.

(4) The rub groove in the stator part changes the flow area in the tip gap, thus significantly affects the flow pattern in the seal chamber, in turn affecting the leakage rate in the stepped labyrinth seal with rub groove in contrast to the original design.

NOMENCALATURE

- \(A\): area, \(m^2\)
- \(b\): tooth thickness, \(m\)
- \(b_N\): width of rub groove, \(m\)
- \(C_r\): clearance, \(m\)
- \(C_D\): discharge coefficient
- \(m\): mass flow rate, \(kg/s\)
- \(n\): tooth number
- \(p\): pressure, \(Pa\)
- \(PR\): pressure ratio
- \(t_N\): depth of rub groove, \(m\)
- \(R\): ideal gas constant, \(kJ/(kg \cdot K)\)
- \(s\): clearance, \(m\)
- \(T\): temperature, \(K\)
- \(\pi\): pressure ratio
- \(\xi\): relative discharge coefficient

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