Effects of Pressure Ratio and Rotational Speed on Leakage Flow and Cavity Pressure in the Staggered Labyrinth Seal

Zhigang Li
Jun Li
Xin Yan
Zhenping Feng

Institute of Turbomachinery, Xi’an Jiaotong University, Xi’an 710049, China

Effects of pressure ratio and rotational speed on the leakage flow and cavity pressure characteristics of the rotating staggered labyrinth seal were investigated by means of experimental measurements and numerical simulations. The rotating seal test rig with turbine flowmeter and pressure measuring instruments was utilized to investigate the leakage flow of the staggered labyrinth seal at eight pressure ratios and five rotational speeds. The repeatability of the experimental data was demonstrated by three measurements at different pressure ratios and fixed rotational speed. The three-dimensional Reynolds-averaged Navier–Stokes equations and standard k-ε turbulent model were also applied to study the leakage flow characteristics of the staggered labyrinth seal under stationary and experimental conditions. The validation of the numerical approach was verified through comparison of the experimental data. The detailed flow field in the staggered labyrinth seal was illustrated according to the numerical simulations. The experimental and numerical results show that the leakage flow coefficient increases with increasing pressure ratio at the fixed rotational speed and is more sensitive to the smaller pressure ratio. The influence of rotational speed on the leakage flow coefficient is not obvious in the present rotational speed limitations. The cavity pressure coefficient in the staggered labyrinth seal decreases and is significantly influenced by the cavity structure along the flow direction. [DOI: 10.1115/1.4003788]

Keywords: staggered labyrinth seal, leakage flow, experimental measurement, numerical simulation

1 Introduction

The reduction in leakage flows is a cost-effective method to improve the aerodynamic efficiency of turbomachinery as well as a main goal in the design of current and future turbomachinery. Rotating seals serve a significant purpose to enhance turbomachinery performance, as it can effectively control gas leakage from high pressure regions to low pressure regions. In order to match the discharge behavior of seals to require in a given design of turbomachinery, it is of particular importance to correctly predict the leakage of rotating seals in different working conditions. Due to the benefits of low costs, long life-time, simple design, low maintenance, and high temperature capability, labyrinth seals are widely used in steam turbines, gas turbines, and other applications where transient rubbing contact is likely [1].

Stocker et al. [2] utilized experimental technique to determine the static and dynamic performances of the labyrinth seal using solid, abrasive, and honeycomb lands. Wittig et al. [3] investigated the effects of pressure ratio and Reynolds number on straight-through labyrinth seals by measuring leakage rates of geometrically similar, 2D plane models at different sizes. Rhode and Adams [4] applied experimental method and 2D numerical method to explore the effects of the size of wear-in rub grooves for straight-through and stepped labyrinth seals. The relationships among the friction coefficients, the leakage Reynolds number, the groove depth and width, and the prerub radial clearance were examined in their works. Willenberg et al. [5] investigated the flow characteristics in labyrinth seals using experimental measurement and numerical method. Different types of labyrinth seals and several geometrical parameters such as seal clearance, step height, or rotor configuration were considered in their works.

Based on investigations of Paolillo et al. [6], the influence of rotational speed on the leakage rate does not appear at lower speed (less than 15,000 rpm). As the swirl speed of the air approaches the axial through-flow velocity of the air, rotational dependency is reached. The laminar incompressible flow model and the performance of a number of seals of various shapes for both stationary and rotating conditions were numerically investigated [7]. The numerical results showed that the shaft rotation has little effect on the leakage from the grooved shaft and grooved casing seals. Pychynski et al. [8] predicted the discharge coefficients of labyrinth seals using a database method. But, the database method needs large number of experimental data and numerical results. The published rotating seal experimental measurement for the labyrinth seal is performed to obtain the leakage flow rate. The cavity pressure variations of labyrinth seals at designed and off-designed flow conditions are seldom reported. To investigate the leakage flow characteristics of the staggered labyrinth seal for turbomachinery, the rotating seal experimental test rig and three-dimensional numerical approach were utilized to study the effect of the rotational speed and pressure ratio on the leakage flow of the staggered labyrinth seal as well as the cavity pressure variation.

2 Experimental Setup

The present experiments were conducted on a rotating seal test facility, which is illustrated in Fig. 1. The test facility consists of the pressurized air supply system, labyrinth seal test rig, and measurement system. The air was supplied by a compressor at a maximum pressure of 0.7 MPa. The clean and dry pressurized air was obtained through the oil and water extractor instrument. An inlet valve and a bypass valve served to limit the maximum pressure up to 0.4 MPa at the turbine flowmeter. The outlet pressure and temperature depend on the atmosphere boundary conditions, while the inlet pressure was varied with flow control valve downstream of the turbine flowmeter. The pressure ratio is defined by the inlet total pressure divided by the outlet static pressure. The pressure ratio across the experimental staggered labyrinth seal could be controlled from 1.05 to 1.4 with valve downstream of the turbine flowmeter. The pressurized air enters the pressure chamber from the top and exhausts to the atmosphere through the staggered labyrinth seal clearance. The rotating disk with the 400 mm diameter of the staggered labyrinth seal is mounted on a cantilevered shaft. The shaft is supported by two bearings and directly driven by a variable frequency drive (VFD).

2.1 Experimental Labyrinth Seal. The main geometrical parameters of the experimental staggered labyrinth seal are determined in Fig. 2. The rotating component carries three steps and represents the rotor of the staggered labyrinth seal. The stationary part represents the stator and is equipped with ten staggered seal teeth. There are two short teeth between two adjacent long teeth of
the staggered labyrinth seal. The labyrinth seal stator is divided into six circumferential segments. To limit the axial and radial movements in the test rig, the seal stator rests on the upper surface of the seal stator holder and is held in place by the adjustable bolt. In order to eliminate any radial leakage at the interfaces where the segments meet, the interface surfaces are machined closely and treated with a sealing compound. In order to measure the static cavity pressure, the pressure transducer hole of nine cavity pressure taps is drilled on the seal stator, as shown in Fig. 2.

2.2 Experimental Instrumentations. The flow rate through the experimental staggered labyrinth seal was measured using a Rotork QWLJ-050 turbine flowmeter, which allows for the pressure and temperature compensation. The turbine flowmeter is designed with a maximum uncertainty of ±1%. The inlet total pressure and cavity static pressures were measured using a Rosemont 3051T static pressure transducer, which is connected to the total pressure probe in the inlet air pipe, and cavity pressure taps are installed in the seal using Nylon tubing. The maximum uncertainty of the pressure transducer equals ±0.075%. An armored thermocouple (NiCr–Cu, type T) was used to measure the inlet air temperature with an uncertainty of ±0.75%. The variation of the rotational speed was adjusted by means of frequency converter.

The analysis of the leakage flow rate is highly sensitive to the clearance of the staggered labyrinth seal. Feeler gauge was used to measure the tooth tip clearances with an uncertainty of ±0.02 mm. In order to ensure that the clearance of the staggered labyrinth seal is the same for all circumferential segments, 12 clearance measure points are equally distributed along the circumference. Adjustable bolts were used to adjust the position of every circumferential segment of the staggered labyrinth seal referring to the clearance values in 12 measure points until the maximum difference between 12 clearance values is less than 0.02 mm. Considering a maximum uncertainty of measuring devices, uncertainties in the estimation of the pressure ratios are ±0.15%, those of the leakage flow coefficients are ±4.045%, and those of the pressure coefficients are ±0.3%.

![Fig. 1 Schematic view of the rotating seal test system](image)

![Fig. 2 Geometrical profile and parameters of the experimental labyrinth seal](image)

\[
H/R = 0.0225, h/R = 0.0015, b_1/R = 0.05, b_2/R = 0.0275, t_1/R = 0.0775, t_2/R = 0.02, t_3/R = 0.0325, L_1/R = 0.3, L_2/R = 0.058, W_1/R = 0.012, W_2/R = 0.004, W_3/R = 0.008, W_4/R = 0.004, s/R = 0.0039, R = 200 \text{ mm}
\]
Table 1 Boundary conditions and numerical methods

<table>
<thead>
<tr>
<th>Condition</th>
<th>Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet total pressure</td>
<td>101.01–134.68 kPa</td>
</tr>
<tr>
<td>Inlet total temperature</td>
<td>292.31–298.56 K</td>
</tr>
<tr>
<td>Outlet static pressure</td>
<td>96.2 kPa</td>
</tr>
<tr>
<td>Computational scheme</td>
<td>Time marching</td>
</tr>
<tr>
<td>Turbulence model</td>
<td>k-ε, scalable log</td>
</tr>
<tr>
<td>Fluid</td>
<td>Air (idea gas)</td>
</tr>
<tr>
<td>Wall properties</td>
<td>Adiabatic, smooth</td>
</tr>
</tbody>
</table>

3 Numerical Approach

To investigate the leakage flow field of the experimental staggered labyrinth seal, a commercial computational fluid dynamic (CFD) software ANSYS CFX11.0 was used. The detailed boundary conditions and numerical methods are listed in Table 1.

According to Refs. [9,10], the dimensionless flow coefficient $\phi$ and pressure coefficient $\psi$ are expressed by

$$\phi = \frac{m \cdot \sqrt{2 \cdot R \cdot T_{tot,in}}}{(A \cdot P_{tot,in})}$$

and

$$\psi = \frac{(P_i - P_{stat,out})}{(P_{tot,in} - P_{static})}$$

where $A = \pi \cdot [(R+s)^2 - R^2]$ denotes the annular area of the seal clearance, $P_i$ is the pressure at cavity $i$, $P_{tot,in}$ and $T_{tot,in}$ mean the inlet total pressure and temperature, respectively, and $P_{stat,out}$ is the outlet static pressure whereby $R = 287.2 \ J/(kg \ K)$ is the gas constant. The labyrinth seal inlet total pressures are made dimensionless through pressure ratio $\pi$. The velocities are made dimensionless by the velocity ratio $U/C_{av}$, which is defined as the ratio of the circumferential velocity of the rotor surface $U$ to the averaged axial velocity $C_{av}$ through the tooth tip gap of the labyrinth seal, as introduced by Paolillo et al. [6].

4 Results and Discussions

The seal gap clearance is adjusted to 0.78 mm during assemble and kept constant for all tests. The test starts first with adjusting the rotational speed by the frequency converter. After achieving the rotational speed of interest, the seals’ inlet pressure was varied using the flow control valve downstream of the turbine flowmeter for the predetermined pressure ratios. There were 40 test conditions, consisting of five rotational speeds (0 rpm, 300 rpm, 600 rpm, 900 rpm, and 1200 rpm) and eight pressure ratios (1.05, 1.1, 1.15, 1.2, 1.25, 1.3, 1.35, and 1.4) for a fixed clearance of 0.78 mm and a seal exit pressure of 96.2 kPa. For all experimental conditions, inlet total pressure, and temperature, leakage flow rate, cavity pressure, and rotational speed readings are recorded.

4.1 Repeatability Analysis. For the same operating condition with the pressure ratio and the rotational speed of interest, measurements of the leakage flow rate and cavity pressure were repeated three times. The maximum scatter of the experimental data equals 1.1%. The three times experimental measurements are named as test 1, test 2, and test 3, respectively. Tests 2 and 3 of the experiment were simulated numerically. Due to the limitation of control valve instrument sensitivity, the achieved inlet total pressures depart from the interest values slightly for the three times experimental measurements. Consequently, the boundary conditions of the numerical simulation tests 2 and 3 differ from each other slightly. Table 2 gives the detailed seal inlet boundary conditions and leakage results for CFD tests 2 and 3.

The comparison of the leakage flow rate between the experimental data of three tests and numerical results for the labyrinth seal is given in Fig. 3(a). The comparison of the cavity pressure in
the staggered labyrinth seal chambers is illustrated in Fig. 3(b). As shown in Fig. 3, the differences of experimental data obtained in three tests were negligible, and the maximum differences are 1.1% and 0.5% for the leakage flow rate and cavity pressure, respectively. The present numerical results are in a good agreement with experimental data, and the maximum differences are 2.3% and 3.2% for the leakage flow rate and cavity pressure, respectively. Similar results were obtained from tests on the labyrinth seal at other pressure ratio and rotational speed conditions.

4.2 Effects of Pressure Ratio. The dependence of the leakage flow coefficient upon the pressure ratio of the staggered labyrinth seal in the case of fixed clearance size is shown in Table 3. The leakage flow coefficient increases with increasing pressure ratio. For each pressure ratio up to $\pi = 1.2$, the measured leakage flow coefficients and the numerically obtained values agree very well. With increasing $\pi$, the numerical results deviate slightly from those obtained experimentally, and the maximum relative deviation is equal to 2.5%.

4.3 Effects of Rotational Speed. The measured leakage rates and numerical results for the labyrinth seal are plotted as the corresponding dimensionless number $\phi$ versus the velocity ratio $U/C_{ax}$, which indicates the rotational speed and takes the seal rotor radius into account, as shown in Fig. 4. With increasing $U/C_{ax}$, the leakage flow coefficient remains approximately constant and is consistent with the published results by Scherer et al. [10] and Paolillo et al. [6]. A critical velocity ratio of $U/C_{ax} = 1.0$ was introduced by Scherer et al. [10] and Paolillo et al. [6]. Below this critical value, a rotational influence on the leakage flow coefficient does not appear. Beyond this critical value, the leakage flow coefficient can be reduced significantly. In addition, a good agreement between the numerical predictions and the measured values is observed, and the maximum relative deviation is 2.7%, as shown in Fig. 4.

4.4 Cavity Pressure. The sealing effect of labyrinth seals is achieved by repeated throttling of the leakage fluid through narrow gaps between the rotating and stationary components [1,2]. The effectiveness of dissipation of kinetic energy between throttles determines the leakage flow rate through the labyrinth seal. In the present paper, mean cavity pressure measurements were made to investigate the dissipation mechanisms of kinetic energy in the labyrinth seal. The measurement positions and index number cavities 1–9 are shown in Fig. 2.

Figure 5 shows the variation of mean cavity pressure (in terms

![Image of Table 3](http://asmedigitalcollection.asme.org/)

<table>
<thead>
<tr>
<th>$\pi$</th>
<th>CFD</th>
<th>$\phi$</th>
<th>Experiment</th>
<th>$\pi$</th>
<th>CFD</th>
<th>$\phi$</th>
<th>Experiment</th>
<th>$\pi$</th>
<th>CFD</th>
<th>$\phi$</th>
<th>Experiment</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.044</td>
<td>0.075</td>
<td>0.076</td>
<td></td>
<td>1.050</td>
<td>0.075</td>
<td>0.075</td>
<td></td>
<td>1.046</td>
<td>0.076</td>
<td>0.075</td>
<td></td>
</tr>
<tr>
<td>1.095</td>
<td>0.103</td>
<td>0.105</td>
<td></td>
<td>1.100</td>
<td>0.103</td>
<td>0.103</td>
<td></td>
<td>1.096</td>
<td>0.103</td>
<td>0.103</td>
<td></td>
</tr>
<tr>
<td>1.145</td>
<td>0.122</td>
<td>0.124</td>
<td></td>
<td>1.151</td>
<td>0.122</td>
<td>0.123</td>
<td></td>
<td>1.146</td>
<td>0.122</td>
<td>0.124</td>
<td></td>
</tr>
<tr>
<td>1.194</td>
<td>0.136</td>
<td>0.140</td>
<td></td>
<td>1.201</td>
<td>0.136</td>
<td>0.138</td>
<td></td>
<td>1.196</td>
<td>0.137</td>
<td>0.139</td>
<td></td>
</tr>
<tr>
<td>1.243</td>
<td>0.148</td>
<td>0.152</td>
<td></td>
<td>1.250</td>
<td>0.147</td>
<td>0.150</td>
<td></td>
<td>1.245</td>
<td>0.148</td>
<td>0.150</td>
<td></td>
</tr>
<tr>
<td>1.295</td>
<td>0.158</td>
<td>0.162</td>
<td></td>
<td>1.301</td>
<td>0.157</td>
<td>0.160</td>
<td></td>
<td>1.296</td>
<td>0.158</td>
<td>0.160</td>
<td></td>
</tr>
<tr>
<td>1.344</td>
<td>0.165</td>
<td>0.170</td>
<td></td>
<td>1.351</td>
<td>0.165</td>
<td>0.169</td>
<td></td>
<td>1.346</td>
<td>0.166</td>
<td>0.170</td>
<td></td>
</tr>
<tr>
<td>1.394</td>
<td>0.172</td>
<td>0.178</td>
<td></td>
<td>1.402</td>
<td>0.172</td>
<td>0.176</td>
<td></td>
<td>1.396</td>
<td>0.172</td>
<td>0.176</td>
<td></td>
</tr>
</tbody>
</table>

![Image of Table 3](http://asmedigitalcollection.asme.org/)

![Fig. 4 Leakage flow rate with different velocity ratios](http://asmedigitalcollection.asme.org/)

![Fig. 5 Static pressure in cavities of labyrinth seal](http://asmedigitalcollection.asme.org/)
of the dimensionless pressure coefficient along the flow direction from the seal inlet to the exit. For the same seal cavity and constant rotational speed, the cavity pressure coefficient shows a slight increase as the pressure ratio increases. The cavity pressure coefficients at different rotational speeds are nearly the same. In Fig. 5, except for cavity 8, the same regressive behavior of the cavity pressure coefficients along the flow direction is shown for the present experimental and numerical results. This regressive behavior can be supported by a simple theoretical analysis, which assumes an isentropic expansion across the tooth tip gap and an isobaric dissipation within the labyrinth cavities. Except for cavity 1, the other cavity pressure coefficients numerically obtained are all overpredicted. The maximum relative deviation of the cavity pressure coefficients is 22%, presented in cavity 5. The maximum measurement error results for the lowest cavity pressure in smallest cavities 2, 5, and 8 because of the pressure rise through the downstream tooth tip gap of cavity 8. It is important to note that the maximum relative of the dimensional cavity pressure is less than 3.2%, which is presented in Fig. 3(b). Therefore, the measured cavity pressure and the numerical obtained values agree well.

As shown in Fig. 5, the pressure coefficient drops through the downstream tooth tip gaps of cavities 2, 5, and 8 are all clearly smaller than those through the other tooth tip gaps. This phenomenon can be explained by the seal geometry and the velocity field, as shown in Fig. 6. The flow pattern shows a distinct dependence on the cavity geometry construction. There are three kinds of cavity geometry construction, and this results in three different flow patterns. For cavities 1–3, the fluid passes the first tooth tip gap and impinges on the step or tooth wall. The shapes of these two recirculation zones are similar, but the size of the recirculation in cavity 1 is larger. The jet passes the tooth tip gap above the seal step and rips into cavity 3. Inside the cavity, the jet is slightly deflected toward the bottom of the labyrinth chamber then impinges onto the tooth wall and separates two counter-rotating different sized recirculation zones.

The pressure data in Fig. 5 indicate that the geometry construction of cavities 1 and 3 is more effective than that of cavity 2 in dropping the gas pressure. This indicates that these geometries of cavities 1 and 3 are more effective in reducing the leakage through the seal. This phenomenon is illustrated with the velocity vector distribution inside the seal according to Fig. 6. The intensity of recirculation inside cavities 1 and 3 is stronger than that of cavity 2. The larger viscous shear stress is generated inside cavities 1 and 3 to dissipate the leakage jet kinetic energy more efficiently. For cavities 1 and 3, the main flow impinges on the step or tooth wall and is deflected by almost 90 deg toward top or bottom. The main fluid kinetic energy converted from the fluid pressure energy is dissipated into heat by turbulence. For cavity 2, the main flow direction does not deflect due to the “straight-through” seal geometry (the adjacent tooth gaps are located at the same radial height), but directly enters the subsequent tooth gap by kinetic energy carryover. Therefore, the main fluid kinetic energy is not dissipated sufficiently inside cavity 2.

5 Conclusions

The results of a combined experimental and numerical investigation on leakage flow and mean cavity pressure characteristics of the staggered labyrinth seal were conducted in this paper. The flow coefficients show a continuous increase with increasing pressure ratio. In cases of low rotational speed where the \( U/C_{ax} < 1.0 \) was the limitation of the present experimental setup, the influence of the rotational speed on the flow coefficient is negligible. A significant influence of the cavity geometry on the leakage flow field and cavity pressure drop between cavities was observed. Compared with the straight-through cavity geometry, the cavity geometry with the seal step at the downstream or upstream is more effective in reducing the gas pressure. This indicates that the labyrinth seal with steps is more effective than the straight-through labyrinth seal in reducing the leakage flow rate through the seal.

Acknowledgment

The authors are grateful for Project No. 50976083 supported by the National Natural Science Foundation of China.

References


