EFFECTS OF PRESSURE RATIO AND ROTATIONAL SPEED ON THE SEALING EFFECTIVENESS OF TURBINE RADIAL RIM SEAL

Zhigang LI
Institute of Turbomachinery, Xi’an Jiaotong University
zhigangli@mail.xjtu.edu.cn
Xi’an, China

Jun LI *
Institute of Turbomachinery, Xi’an Jiaotong University
*corresponding author: junil@mail.xjtu.edu.cn
Xi’an, China

Qing GAO
Energy Saving Center, Xi’an Thermal Power Research Institute Company Limited
leopard.gao@stu.xjtu.edu.cn
Xi’an, China

Zhenping FENG
Institute of Turbomachinery, Xi’an Jiaotong University
zpcheng@mail.xjtu.edu.cn
Xi’an, China

ABSTRACT
Effects of pressure ratio and rotational speed on the sealing effectiveness of turbine radial rim seal were numerically investigated in this work. Three-dimensional Unsteady Reynolds-Averaged Navier-Stokes (URANS) and SST turbulent model was used to study the sealing effectiveness of radial rim seal with consideration of the stator and rotor blade interactions. The numerical results were agreeing well with the experimental data. Four different pressure ratios with fixed rotational speed and four rotational speeds with fixed pressure ratio were utilized to analyze the sealing effectiveness of radial rim seal. The numerical results show that the sealing effectiveness decreases with increase of the pressure ratio. In addition, the sealing effectiveness increases with increase of the rotational speed. The variation of the pressure ratio and rotational speed changes the incidence on the rotor blade and circumferential pressure distribution at the rim seal clearance. This flow behaviour results in the variation of the sealing effectiveness. The unsteady time-averaged and unsteady flow pattern were illustrated to investigate the fluid dynamics and sealing effectiveness of radial rim seal.

NOMENCLATURE
b  radius of rim seal [mm]
c₀  tracer gas concentration at sealing flow inlet
cₐ  tracer gas concentration at main annulus flow inlet
cₛ  tracer gas concentration at determined points on the stator surface
Cₘᵢ  axial gap of rim seal outer fin [mm]
Cₒᵣ  relative gap between the inner and outer fin [mm]

Cₚ  static pressure coefficient
Cₚₚ  non-dimensional rim seal flow rate [m/(μR₀)]
Cₚₚ₀  non-dimensional rim seal flow rate [m₀/(μR₀)]
h  mainstream passage height [mm]
h₁  width of rim seal axial fin [mm]
h₂  width of rim seal radial inner fin [mm]
l  length of rim seal radial inner fin [mm]
m  cooling purge flow rate [kg/m²]
m₀  cooling purge flow rate [kg/m²]
n  designed rotational speed [rpm]
P₀  designed inlet total pressure [Pa]
Pₜ₀  outlet static pressure [Pa]
R₀  cavity outer radius [mm]
T₀  inlet total temperature [K]
U  fluid velocity [m/s]
μ  dynamic viscosity
εₛ  sealing effectiveness
ρ  density [kg/m³]
Scᵣ  turbulence Schmidt number

INTRODUCTION
The secondary air system plays a significant role on the energy transfer efficiency and output power of modern gas turbines. Rim seal is one of the important components of the secondary air system to protect the turbine blade and cooling the wheel-space disc. Rim seals installed at the periphery of the hub interface between the stationary and rotational
components of gas turbines. Well-designed rim seal can effectively prevent hot gas ingestion into the wheel space cavity, which represents an unavoidable cavity between stator-disc and rotor-disc of a turbine, to avoid the phenomenon of disc overheating. Hence, it is very meaningful to give a better understanding on flow characteristics and sealing effectiveness of rim seals [1].

Experimental measurements and numerical simulations, as well as theoretical analysis have been used to investigate the sealing performance of rim seals. Phadke and Owen [2-4] experimentally measured the minimum sealing flow with different rim seal geometries under both circumferential pressure symmetry as well as pressure asymmetry. The results show that asymmetry of circumferential pressure has a significant influence on the minimum sealing flow. Owen [5, 6] developed the orifice model to predict the mainstream ingestion through the turbine rim seal for the rotationally induced ingress and externally induced and combined ingress based on the experimental data.

With the purpose of understanding the circumferential pressure asymmetry on effectiveness of rim seal, nowadays many researchers carried out experiments with turbine test rig including vane and blade in the annulus. Gentilhomme et al. [7] investigated experimentally and numerically the hot gas ingestion through axial rim seal on a single turbine stage. The pressure and gas concentration measurements were used to analyze the sealing performance of rim seals. They concluded that the highly swirling annulus air results in the ingestion degree. The rim seal purge flow interaction with main annulus flow was numerically investigated by Cao et al. [21]. The ingestion of the highly swirling annulus flow results in the increase vortex strength within the cavity according to the experimental data. Teuber et al. [22] conducted the fluid dynamics of the turbine axial and radial rim seals at the low Mach numbers using unsteady numerical simulations. They concluded that the peak-to-trough pressure difference in the main annulus which is the driving mechanism for ingestion. Wang et al. [23] studied the hot gas ingestion physics of the radial rim seal using a 360° time-dependent numerical approach. The obtained results show that the irregular circumferential pressure distribution which is produced by the interaction between the vane and rotor blade provides the driving forces for ingestion. Lalwani et al. [24] studied the ingestion behaviour of turbine rim seal using steady numerical simulation. Li et al. [25, 26] developed the unsteady numerical method to obtain the sealing effectiveness of rim seals using the additional scalar variable approach. The main flow and rim seal purge flow transported with the wheel-space cavity was simulated. The sealing effectiveness of four different rim seals and new invented honeycomb structure were compared. The high sealing performance of the designed honeycomb rim seal was demonstrated.

Above research works focus on the geometrical structures and sealing flow rates on the sealing effectiveness of rim seals using steady and unsteady experimental measurements and numerical simulations. A gas turbine always operates at off-design conditions. The sealing effectiveness of rim seals at different pressure ratios of main annulus and rotational speeds of rotational disc would influence the gas turbine performance. It is very important to understand the ingestion mechanisms of rim seal at off-designed operational conditions. In this paper, effects of the pressure ratios and rotational speeds on the sealing effectiveness of turbine radial rim seals were numerically investigated using Unsteady Reynolds-Averaged Navier-Stokes (URANS) and additional passive tracer method solutions. The accuracy of
the used numerical approach was demonstrated by comparison to the experimental data. The physical mechanism of the sealing performance of the radial rim seal at different pressure ratios and rotational speeds were illustrated based on the time-averaged and unsteady flow field analysis.

**COMPUTATIONAL MODEL AND METHOD**

**Numerical Method**

In experiment, the study on sealing effectiveness of the rim seal mostly adopts a tracer gas concentration method. Tracer gas such as CO₂ was seeded into the wheel-space cavity inlet and then ingress into the main annulus. The tracer gas density was collected at determined points at the stator disc surface to evaluate the level of mainstream ingestion.

Based on the concentration measurement, the sealing effectiveness of the rim seal was defined as:

$$ e_c = \frac{c_a - c_0}{c_a - c_u} $$  \hspace{1cm} (1)

where $e_c$ is sealing effectiveness, while $c_0$ is tracer gas concentration at sealing flow inlet, $c_a$ is tracer gas concentration at main annulus flow inlet and $c_u$ is tracer gas concentration at determined points on the stator surface.

In this work to simulate the turbulent transportation and diffusion of tracer gas, the additional scalar variable was adopted to obtain the distribution of tracer gas concentration. The general form of the turbulent transportation and diffusion equation for additional scalar variable can be represented as following expression.

$$ \frac{\partial (\rho \phi)}{\partial t} + \nabla \cdot (\rho U \phi) = div \left( \left( \rho D_{\phi} + \frac{\mu_t}{Sc_i} \right) \nabla \phi \right) + S_{\phi} $$  \hspace{1cm} (2)

where $U$ is the fluid velocity, $\rho$ is the density, $\phi$ is the conserved quantity per unit volume, or concentration, $\phi = \phi/\rho$ is the conserved quantity per unit mass, $S_{\phi}$ is a volumetric source term, with units of conserved quantity per unit volume per unit time, $D_{\phi}$ is the kinematic diffusivity for the scalar, $Sc_i$ is the turbulence Schmidt number, $\mu_t$ is the turbulence viscosity.

The concentration value of the additional scalar variable is set as 1 at the sealing flow inlet and 0 at the annulus flow inlet. Due to the mixing with ingress gas and egress sealing air, the concentration of the additional scalar variable falls into between 0 and 1 in the wheel-space.

To demonstrate the accuracy and reliability of the additional scalar variable approach for predicting the sealing effectiveness of the rim seal, the research single turbine rim seal was utilized [27]. The comparison of the sealing effectiveness between the experimental data and numerical results for steady and unsteady computation is given in Fig. 1. The sealing effectiveness is over-predicted using steady computation, but the predicted values with unsteady computation are consistent with the experimental data. The sealing effectiveness is overpredicted by about 36% for steady computation at purge flow rate $c_{i0} = 1250$ and mismatch is only 6% for unsteady computation. Therefore, unsteady computation coupled with the URANS and additional scalar variable method can accurate predict sealing performance of rim seal.

![Fig. 1 Sealing effectiveness of rim seal between the numerical results and experimental data](image)

**Computational Model**

The effects of the pressure ratios and rotational speeds on the sealing effectiveness of the turbine radial rim seal were conducted and discussed. Fig. 2 illustrates the turbine radial rim seal. Table 1 lists the geometrical parameters of the studied radial rim seal. The sealing flow injects into wheel-space cavity at the bottom of the cavity. In this paper’s research, the profile of the vane and rotor blade of turbine stage is adopted from Ref. [28]. The computational domain of the turbine stage has one vane and rotor blade in this unsteady calculation.

![Fig. 2 Computational model of turbine radial rim seal](image)

<table>
<thead>
<tr>
<th>Table 1 Main geometrical parameters of radial rim seal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geometrical parameter</td>
</tr>
<tr>
<td>mainstream passage height $h$</td>
</tr>
<tr>
<td>radius of rim seal $b$</td>
</tr>
<tr>
<td>axial gap of rim seal outer fin $C_e$</td>
</tr>
<tr>
<td>relative gap between the inner and outer fin $C_{i0}$</td>
</tr>
<tr>
<td>radial gap of rim seal inner fin $C_{in}$</td>
</tr>
<tr>
<td>width of rim seal axial fin $H_1$</td>
</tr>
<tr>
<td>length of rim seal axial inner fin $L$</td>
</tr>
<tr>
<td>width of rim seal radial inner fin $H_2$</td>
</tr>
</tbody>
</table>

Figure 3 illustrates the multi-block structured grid of the computational domain including the turbine stage and radial rim seal. The computational mesh of the turbine stage passage was generated using NUMECA-IGG. An O-grid was
adopted around the turbine blade and H-grid was applied to the other part in turbine stage passage. The total number of computational grid cell is 1,860,000. In addition, by means of ANSYS-CFX, computational grid is refined near the wall to be ensured that $y^+ < 1$ to meet the requirement of SST turbulent model.

![Fig. 3 Computational grid of turbine radial rim seal](image)

Table 2 Computational boundary conditions and numerical method

<table>
<thead>
<tr>
<th>Designed inlet total pressure $P_0$</th>
<th>123000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outlet total pressure $P_{out}$</td>
<td>101325</td>
</tr>
<tr>
<td>Designed rotational speed $n_{rpm}$</td>
<td>3600</td>
</tr>
<tr>
<td>Rim seal flow rate $C_w$</td>
<td>5170</td>
</tr>
<tr>
<td>Fluid</td>
<td>Air (ideal gas)</td>
</tr>
<tr>
<td>Discretization scheme</td>
<td>High resolution</td>
</tr>
<tr>
<td>Computational method</td>
<td>Time marching</td>
</tr>
<tr>
<td>Turbulence model</td>
<td>SST</td>
</tr>
<tr>
<td>Rotor-stator interface</td>
<td>Frozen Rotor (steady)</td>
</tr>
<tr>
<td>Rotor-stator interface</td>
<td>Transient rotor stator (unsteady)</td>
</tr>
<tr>
<td>Wall properties</td>
<td>Adiabatic, smooth surface</td>
</tr>
</tbody>
</table>

The sealing effectiveness and flow field of the turbine radial rim seals were computed using three-dimensional URANS solution and SST turbulent model combined with an additional scalar variable method by means of the ANSYS-CFX. In this paper, the rim seal effectiveness was assessed by means of the concentration sealing effectiveness $e_c$ defined by Eq. (1).

The high resolution was used for the advection terms and the second order backward Euler scheme was used for the transient calculations. The time step was set as $2.777 \times 10^{-5} \, s$, corresponding to 20 time steps for one blade passing period. Firstly, steady simulation was conducted with the domain interface modelled by frozen rotor and the steady results were used as the initial flow field of the unsteady prediction. The convergence criteria is based on root mean square residuals (RMS) for momentum, mass and energy equations, and turbulence equations reached (or were even lower than) $10^{-5}$ and when the imbalance of all properties through the whole domain is less than 0.1 %.

The detailed numerical approaches in the current study are listed in Table 2. In the calculation, the turbulent intensity of 5% was defined at the turbine annulus inlet. Total pressure was specified at the turbine annulus inlet. The averaged static pressure was set at the annulus outlet. Mass flow rate was prescribed at the sealing flow inlet. In addition, the static and rotating walls were defined to be adiabatic. In the present study, the mass flow rates with $C_w = 5170$ was specified at the sealing flow inlet. The designed flow condition of the turbine stage was operated at the inlet total pressure of 123000Pa and rotational speed of 3600rpm.

RESULTS AND DISCUSSION

Four different pressure ratios and rotational speeds were used to analyze the operational conditions on the sealing effectiveness of turbine radial rim seal. At first, pressure ratio with 1.1, 1.2 (designed flow condition), 1.3 and 1.4 was calculated at fixed rotational speed of 3600rpm. Secondly, rotational speed with 2800rpm, 3200rpm, 3600rpm (designed flow condition) and 4000rpm at fixed pressure ratio of 1.2 were then conducted.

Pressure Ratio Effect

Figure 4 illustrates the time-averaged sealing effectiveness contours distribution on the meridional plane of disc at different pressure ratios. Most of the region in the disc cavity are well protected by the sealing flow when the pressure ratio equals 1.1. The hot gas ingress only occurs at the clearance of the rim seal outer fin and does not penetrate into the disc due to the sealing flow purge effect. The ingress passes through the rim seal outer fin when the pressure ratio is 1.2. The most of regions in the disc cavity is well protected in this flow condition. When the pressure ratio increases to 1.3, the ingress is filled in the clearance of rim seal and intrudes in the clearance of the rim seal radial inner fin. The sealing effectiveness near the stationary disc obviously decreases. When the pressure ratio obtains 1.4, the ingress penetrates into the clearance of the rim seal radial inner fin. The sealing effectiveness in the center of the turbine disc decreases at a certain extent.

Figure 5 compares the sealing effectiveness distribution along the radial direction for the stationary and rotational disc at different pressure ratios. At the stationary disc, the sealing effectiveness decreases with increase of the pressure ratio. However, the variation of the sealing effectiveness at the rotational disc is very limited with the increase of the pressure ratio. The radial rim seal realizes the effectively cooling function at the sealing flow rate at the pressure ratio of 1.1. The variation gradient of the sealing effectiveness at the rotational disc is small and equals to 1.0. There only exists the rapidly decrease at the rim seal outer fin gap. The larger decreases gradient of the sealing effectiveness within $0.9 < r/b < 1.0$ is observed because of the mix of the ingress and sealing flow when the pressure ratio equals 1.4.
Fig. 4 Time-averaged sealing effectiveness contours distribution on the meridional plane of the disc at different pressure ratios

(b) \( PR = 1.2 \)

(c) \( PR = 1.3 \)

(d) \( PR = 1.4 \)

Fig. 5 Sealing effectiveness distribution along the radial direction at different pressure ratios

Fig. 6 Comparison of the sealing effectiveness among different pressure ratios

Figure 6 shows the effect of the pressure ratio on the sealing effectiveness of radial rim seal which from the evaluation plane result. The sealing effectiveness of radial rim seal reduces with the pressure ratio increases. The sealing effectiveness obtains 0.97 when the pressure ratio is 1.1. The sealing effectiveness decreases to 0.62 when the pressure ratio reaches 1.4. The decreasing rate of the sealing effectiveness increases with the increase of the pressure ratio.

Fig. 7 Variation of the inlet velocity triangle of rotor blade at different pressure ratios

To reveal the physical mechanisms of the variation of sealing effectiveness of rim seal versus pressure ratio, Fig. 7 gives the variation of the inlet velocity triangle of the rotor blade at different pressure ratios. The exit velocity of the vane increases as the pressure ratio increases. The incidence
on the rotor blade changes from the negative into positive as
the pressure ratio increases. This flow behaviour changes the
pressure distribution of the potential field near the leading
edge of the rotor blade. Thus, the sealing effectiveness of the
rim seal is also varied.

Figure 8 gives static pressure coefficients distribution
along circumferential direction at rim seal clearance. The
static pressure coefficient is defined as equation (3).

\[
C_p = \frac{P_r - P_{a}}{0.5 \rho \Omega^2 b^2}
\]  

(3)

where \( P_r \) is the monitored point pressure, \( P_{a} \) is the
circumferential averaged pressure corresponding the
monitored position, \( \Omega \) is the rotational speed of disc, \( b \) is
the radius of the rim seal and \( \rho \) is the density of the fluid.

The circumferential pressure difference is the main driver
of the hot gas ingestion. The peak-to-trough pressure
difference along the circumferential direction increases with
the pressure ratio increases. This flow pattern promotes the
ingestion and reduces the sealing effectiveness of rim seal.

![Pressure coefficients distribution along circumferential direction on rim seal clearance at different pressure ratios](image)

**Fig. 8** Pressure coefficients distribution along circumferential direction on rim seal clearance at different pressure ratios

**Rotational Speed Effect**

Figure 9 shows the time-averaged sealing effectiveness
contours distribution on the meridional plane of disc at
different rotational speeds. The ingress degree decreases with
the increase of the rotational speed. The ingestion penetrates
into clearance of the rim seal inner fin and mixes with the
sealing flow when the rotational speed is 2800rpm. The
ingestion is restricted in the region of the outer and inner fin
when the rotational speed equals 3200rpm. In this case, the
ingestion degree is weakened and the sealing effectiveness in
the disc cavity improves. The sealing effectiveness in the
disc cavity increases obviously when the rotational speed
obtains 3600rpm. The ingress is prevented in the clearance of
the rim seal outer fin. Furthermore, no ingestion phenomenon
in the disc cavity is observed when the rotational speed
obtains 4000rpm.

![Time-averaged sealing effectiveness contours distribution on the meridional plane of the disc at different rotational speeds](image)

**Fig. 9** Time-averaged sealing effectiveness contours distribution on the meridional plane of the disc at different rotational speeds

Figure 10 compares the sealing effectiveness of the
stationary and rotational disc wall along the radial direction
at different rotational speeds. The obvious variation of the
sealing effectiveness on the stationary disc wall with the
rotational speeds is captured. The rotational disc wall is well
protected by the sealing flow for different rotational speeds.
Fig. 11 gives the sealing effectiveness at specified plane of
rim seal for different rotational speeds. The sealing
effectiveness increases with the rotational speeds increases
according to Fig. 11. At the fixed pressure ratio flow
condition, the incidence of the rotor blade changes from the
positive to negative with the increases of the rotational speed
as shown in Fig. 12.
Figure 13 Pressure coefficients distribution along circumferential direction on rim seal clearance at different rotational speeds

Figure 13 illustrates the pressure coefficients distribution along circumferential direction on rim seal clearance at different rotational speeds. The maximum and minimum static pressure gradually decreases with the rotational speeds increases. The static pressure coefficient difference along the circumferential direction equals 0.6 with rotational speed of 2800rpm. However, this value reduces to the 0.3 when the rotational speed obtains 4000rpm. The increase of the rotational speed results in reduction of the non-uniformity of the pressure distribution along the circumferential direction, as well as the peak-to-trough value. This flow pattern decreases the driver for the ingestion from the rim seal and improves the sealing effectiveness.

**Unsteady Interaction Effect**

The incidence of the rotor blade changes with the variation of the pressure ratio and rotational speed. Thus the circumferential pressure asymmetries distribution in the main annulus directly influences the ingestion degree and correspondingly the sealing effectiveness of rim seal. Furthermore, the stationary vane and rotating blade creates an unsteady pressure distribution on the main annulus near the rim seal clearance. Ingress occurs in the circumferential clearance area where the pressure in the annulus is higher than that in the wheel-space disc cavity. At the same time, egress of the sealing flow occurs in those regions where this pressure difference is reversed.

The unsteady flow field and sealing effectiveness of rim seal was discussed at the pressure ratio 1.2 and rotational speed of 3600rpm. Figure 14 illustrates the unsteady radial velocity contours distribution on rim seal gap at different relative stator-rotor positions. When the wake position of the vane impacts on the suction surface of the downstream rotor blade at \( T_1 \) and \( T_2 \) as shown in Fig. 14(a, b), the smaller radial velocity at the rim seal clearance is observed. With the rotor blade rotating, when the wake position of the vane locates at the leading edge of the downstream rotor blade, the radial velocity at the axial clearance of the rim seal obtains the maximum according to Fig. 14(c). When the regions of the wake impacts on the pressure surface of the downstream rotor blade at \( T_3 \), the radial velocity at the axial clearance decreases, but is higher than that of the \( T_1 \) and \( T_2 \) cases.
Fig. 14 Velocity contours in the rim seal gap at different stator-rotor positions

Fig. 15 Sealing effectiveness contours distribution on the meridional plane of the disc at different relative stator-rotor positions

Figure 15 shows the sealing effectiveness contours distribution on the meridional plane of disc cavity at different relative stator-rotor positions. At $T_1$ and $T_2$ times, low sealing effectiveness area mostly focuses at the clearance $C_{ax}$ and $C_{io}$ of rim seal. At $T_3$ and $T_4$ times, the ingress penetrates into clearance $C_{in}$ of rim seal. Fig. 16 gives the static pressure coefficients distribution along circumferential direction on rim seal clearance at different relative stator-rotor positions. The peak-to-trough pressure
value at $T_3$ equals about 0.58, however, the peak-to-trough pressure difference value at $T_1$ is about 0.35. The interaction degree between the stationary vane and rotating blade changes with the variation of the relative stator-rotor positions. This flow behaviour results in the variation of the circumferential pressure distribution on the clearance of rim seal and correspondingly the ingress degree. Therefore, the sealing effectiveness of rim seal changes with the variation of the pressure ratio and rotational speed, as well as the relative stator-rotor positions.

Fig. 16 Pressure coefficients distribution along circumferential direction on rim seal clearance at different relative stator-rotor positions

CONCLUSIONS
Effects of the pressure ratio and rotational speed on the sealing effectiveness of turbine radial rim seal were numerically investigated using URANS and SST turbulent model coupled with an additional passive tracer method solutions.

The sealing effectiveness of the turbine radial rim seal decreases from 0.983 to 0.6 at the evaluation plane as the pressure ratio increases from 1.1 to 1.4. The variation of the pressure ratio leads to change the potential field near the leading edge of the rotor blade and correspondingly influence the ingestion behaviour. Thus, the sealing effectiveness of the rim seal is also varied. The incidence of the rotor blade changes from the negative to positive with the pressure ratio varies 1.1 to 1.4. The peak-to-trough pressure difference along the circumferential direction increases as the pressure ratio increases. This flow pattern strengthens the ingestion degree and correspondingly the sealing effectiveness decreases.

The sealing effectiveness of the turbine radial rim seal increases from 0.6 to 0.97 at the evaluation plane as the rotational speed increases from 2800rpm to 4000rpm. The incidence of the rotor blade changes from the positive to negative as the rotational speed increases. The non-uniformity of the circumferential pressure distribution at the clearance of rim seal decreases and correspondingly the sealing effectiveness improves with the rotational speed increases.

The relative stator-rotor positions results in the interaction degree between the vane wake and blade leading edge potential field. This unsteady flow pattern leads to the variation of the sealing effectiveness of rim seal. As the vane wake impacts on the leading edge of the downstream blade, the ingestion is strengthened. The sealing effectiveness of rim seal correspondingly decreases.

ACKNOWLEDGMENTS
This study has been funded by the National Natural Science Foundation (Grant No. 51376144) and Fundamental Research Funds for the Central Universities of China.

REFERENCES


