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Effect of a rib on rim seal performance☆,☆☆

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ABSTRACT

The effect of a rib installed in a rim seal on sealing effectiveness was investigated using three-dimensional Reynolds-averaged Navier–Stokes analysis. A parametric study was performed for a single square rib installed on the surface on the rotor side. A total of 12 cases were tested with different heights, widths, and locations of the rib. The Spalart–Allmaras model was adopted as a turbulence closure model. The Reynolds number based on the axial chord of blade was 500,000 at the mainstream outlet. Ratio of the mainstream outlet velocity to mainstream inlet velocity was kept constant at 3.10. The sealing effectiveness was selected as the performance parameter. Installation of a rib in the rim seal enhances sealing performance regardless of the dimensions and location of the rib, although the effects are relatively small in some cases where the rib is located far from the surface of the rim seal on the rotor side.

1. Introduction

One of the most important problems in designing gas turbine components is avoiding the hot gas ingestion from the mainstream to the cavity between the rotor and stator disks which is called the rim seal. The rim seal is widely used to prevent the hot gas ingression and protect the turbine blade disk from the thermal degradation. Among the many parameters related to the sealing performance, the rim seal configuration is a major factor that determines the sealing performance of the rim seal.

A few researchers have studied the effects of rim seal geometries on the sealing performance in the last three decades. Phadke and Owen [1] conducted an experimental work to test various rim seal geometries, i.e., one axial and four radial rim seals. Their study presented that the radial rim seal shapes show better sealing performance than the axial one. The sealing effectiveness of four different radial rim seal configurations was evaluated by Graber et al. [2] experimentally. They found that there were optimum specific height and width of a radial rim seal which gave the best sealing performance. Chew et al. [3] evaluated the changes in the pressure distribution in the mainstream and the hot gas ingression zone for different rim seal geometries. Bohn and Wolff [4] performed an experimental study to investigate the effect of rim seal shapes on the ingested mass flow rate from the mainstream to the rim seal. Their results revealed that the mass flow rate of the ingression flow depends strongly on the rim seal configuration, and a specific rim seal geometry led to the reduction of hot gas ingestion into the rim seal. Sangan et al. [5,6] experimentally studied the sealing performances of two types of rim seal, i.e., single and double radial rim seals, under the stationary and rotating conditions. They found that the rotating cases show better sealing performances than the stationary cases. And, the rotating double radial rim seal presented the best sealing effectiveness among the tested cases. Popović and Hodson [7] numerically investigated the axial and radial seal clearances of a rim seal. They reported that increases in axial and radial seal clearances enhanced the sealing performance of the rim seal, slightly. Sangan et al. [8] compared the sealing performances among six types of rim seal experimentally. They concluded that highest sealing effectiveness is obtained by the double-radial seal.

As mentioned above, the rim seal configuration significantly affects the sealing effectiveness. However, there have been few reports on the effects of tabulators such ribs and cavities on the sealing performance so far. The objective of this study is to investigate the effect of a rib installed in a rim seal on sealing effectiveness with a parametric study using three-dimensional Reynolds-averaged Navier–Stokes (RANS) analysis.

2. Numerical analysis

Fig. 1 shows the geometries of blade and rim seal, which was investigated by previous works [9–11]. This geometry of rim seal, which was regarded as reference geometry in this work, was originally tested by Popović and Hodson [7,12]. The computational domain consists of the mainstream passage, rim seal, and coolant channel. The axial chord of blade (C) is 130.3 mm, and the ratio of pitch to axial chord (pi/C) is 1.59. Ratio of the distance between the mainstream inlet and outlet of the computational domain to axial chord (S_{out}/C) is 3.98, and ratio of

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Nomenclature

C	Axial chord of blade, m
LF	Leakage fraction, km/s
$\dot{m}$	Mass flow rate, kg/s
pi	Pitch, m
S_l	Streamwise distance from the mainstream inlet to leading edge of blade, m
S_{out}	Streamwise distance from the mainstream inlet to mainstream outlet, m
T	Temperature, K
V	Velocity, m/s
$V_{c,in}$	Rim seal inlet velocity, m/s

Greek symbols

η	Sealing effectiveness
ρ	Density, kg/m^3

Subscripts

baseline	Domain without rim seal
c	Rim seal
h	Mainstream
in	Inlet
out	Outlet

the distance between the mainstream inlet and leading edge of blade to axial chord (S_l/C) is 2.00.

In the present work, steady RANS equations were solved to analyze the flow and heat transfer using ANSYS CFX 11.0 [13]. The one-equation Spalart–Allmaras model [14] was adopted as a turbulence closure model. This turbulence model has been known to give good predictions for the boundary layers subject to adverse pressure gradients [15]. The control planes, planes A, B, and C, were located at $z = -0.20C$, $-0.11C$, and $-0.06C$, respectively, as presented in Fig. 1(c).

The working fluid was regarded as the ideal gas, and the Reynolds number based on the axial chord of blade was 500,000 at the mainstream outlet. Ratio of the mainstream outlet velocity to mainstream inlet velocity ($V_{h,out}/V_{h,in}$) was kept constant at 3.10. The temperatures of the mainstream and cooling air were fixed as 321.75 K ($T_{h,in}$) and 301.75 K ($T_{c,in}$), respectively. Mass flow rates in the mainstream passage and coolant channel were determined by the leakage fraction (LF) and the coolant channel inlet velocity ($V_{c,in}$). LF is defined in terms of mass flow rate ratio as follows [7,12]:

$$LF = \frac{\Delta \dot{m}_c}{\dot{m}_h} = \frac{\dot{m}_{c,in} - \dot{m}_{c,out}}{\dot{m}_h} \times 100. \quad (1)$$

Here, $\dot{m}_h$ and $\dot{m}_c$ indicate the mass flow rates in the mainstream and coolant channel, respectively.

To predict the complex flow structure and heat transfer characteristics in the region where the interaction between coolant and mainstream hot gas was expected to occur, the high mesh density was adopted in the region to satisfy y^+ less than 2.0. A grid dependency test was performed in a previous work [9] to find the optimum number of computational grids in the range of 2,280,000 to 5,100,000. Through the test, 3,790,000 was selected as the optimal number of grids in the computational domain (Fig. 1(c)). In the previous works [9–11], parametric studies of various rim seal configurations were performed to evaluate the sealing effectiveness, and the numerical results, which were predicted by RANS analysis with the same numerical methods used in this study, presented good agreement with the experimental data for the area-averaged sealing effectiveness on plane-A. The root mean square (RMS) relative residual values were maintained at less than 1.0e−4 as convergence criterion. A personal domain with an Intel i7 3.6 GHz CPU was used for the computational analysis.

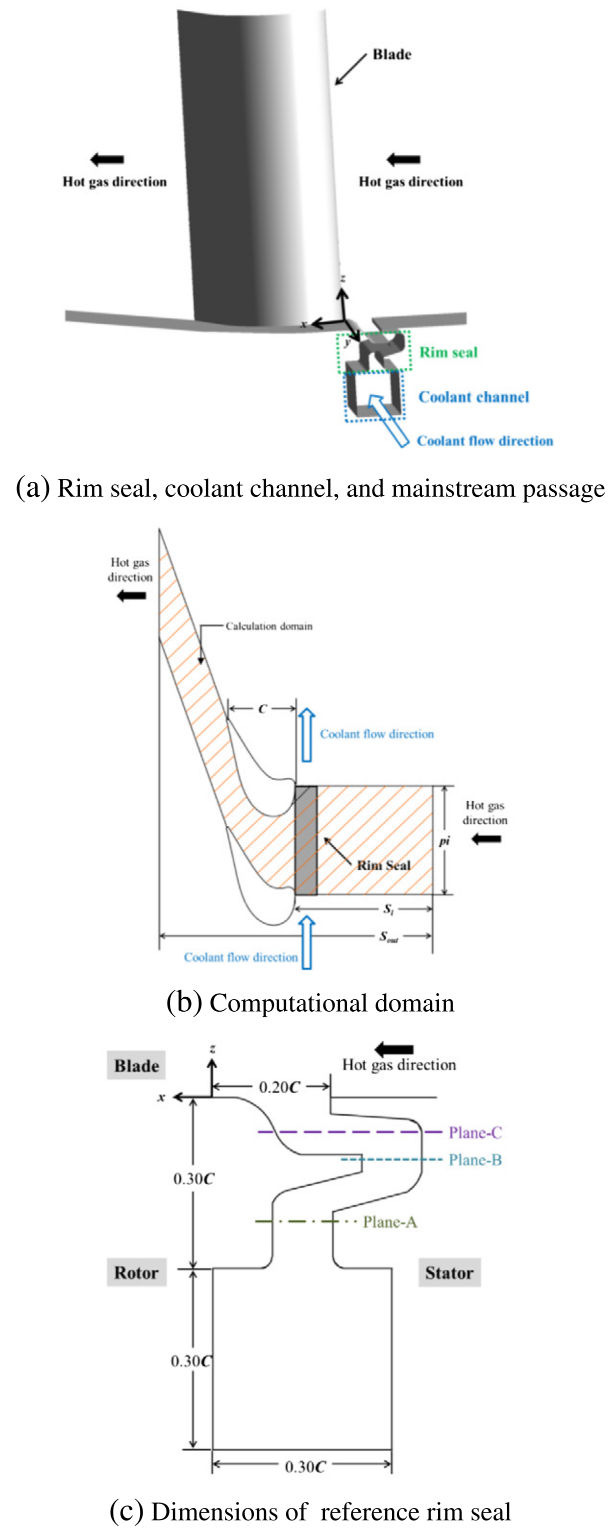

Fig. 1. Computational domain and reference rim seal; (a) rim seal, coolant channel, and mainstream passage, (b) computational domain, and (c) dimensions of reference rim seal.