Numerical Investigation on Leakage Characteristics of a Novel Honeycomb Seal with Wall Holes

Huzhi Du, Yinghou Jiao[✉], Xiang Zhang, and Renwei Che

School of Mechanical Electronic Engineering, Harbin Institute of Technology, Harbin 150001, Heilongjiang Province, China

`jiaoyh@hit.edu.cn`

Abstract. Honeycomb seals are a critical component to reduce leakage flow and improve system stability for turbomachines. In this work, a novel single-wall-hole-honeycomb seal (S-WHHCS) is proposed, which is built by the traditional honeycomb seal (HCS) with single hole drilling in the honeycomb sidewall. The computational fluid dynamics (CFD) method was used to investigate the leakage characteristics of the S-WHHCS. A series of working conditions of rotation speeds and operating pressures were considered in simulations. The influences of the operating parameters on the leakage characteristics of the S-WHHCS were studied and analysed. The mechanism of the leakage reduction effect of the S-WHHCS was revealed compared with a typical HCS. Numerical results show that the newly added hole influences the pressure drop and turbulence kinetic energy distribution for the seals. Moreover, the numerical results also show that the leakage rate of the S-WHHCS can be reduced in a wide range of operating conditions compared with the typical HCS. The current studies on single-wall-hole-honeycomb seals provide an option to enhance honeycomb seal construction.

Keywords: Honeycomb seal · Leakage characteristics · honeycomb wall hole · Numerical simulation

1 Introduction

Seal structures are widely used in turbine machinery to restrict leakage flow through rotor–stator clearances [1]. A successful seal structure can significantly improve the stability and efficiency of rotating machinery [2]. There are various annular seal types used in turbomachinery, such as the labyrinth seal (LS) [3], finger seal, honeycomb seal [4], and scallop seal [5]. As a typical representative of damping seals with outstanding performance in both improving the dynamic characteristics and restricting the leakage flow of rotor systems, Honeycomb seals have been extensively used in turbomachinery in recent years [6].

The operating parameters and the structure of the HCS have a significant effect on the sealing and dynamic performance. Childs [7] conducted a series of studies on the leakage characteristics and rotordynamic characteristics of honeycomb seals, focusing

on the effect of seal clearance and geometrical parameters of the honeycomb cavity. The results show that the honeycomb hole structure has a significant impact on the leakage characteristics. Yan [8] and Li [9] investigated the effects of the pressure ratio, honeycomb diameter and depth on the leakage characteristics of honeycomb seals and proposed an optimal hole diameter and depth to achieve minimum leakage. Zhang [10] tested the leakage characteristics of the interlaced hole honeycomb seal (IHHCS) and the noninterlaced hole honeycomb seal (NIHHCS) by the CFD method. The study shows that the IHHCS possesses a better leakage performance than the NIHHCS. Jiang [11] proposed a novel hole-pattern damping seal with dovetail-like diversion grooves (D-HPDS). The results show that the leakage rate and the sealing performance of D-HPDS are significantly better than those of traditional hole-pattern damping seals.

This paper proposes a new type of single-wall-hole-honeycomb seal (S-WHHCS). The three-dimensional model of annular seals used in the HCS and S-WHHCS was established. The leakage characteristics of the HCS and S-WHHCS in different situations were compared and analysed by ANSYS Fluent, the leakage reduction mechanism of the S-WHHCS was revealed, and the influence of operation parameters on the leakage rate of the S-WHHCS was investigated.

2   Numerical Models and Calculation Method

2.1   Geometrical Model of the Single-Wall-Wole-Honeycomb Seal

The new kind of seal structure designed in this paper is based on the traditional HCS, as shown in Fig. 1a. The design values of the HCS are the same as the model established by Zhang's research [10].

On this basis, a new type of honeycomb damping seal structure named the single-wall-hole-honeycomb seal (S-WHHCS), which is shown in Fig. 1b, was proposed by drilling a single hole in the center of the right hole wall with an axial honeycomb cavity number of 14. The geometric parameters used in the HCS and S-WHHCS are listed in Table 1. The seal length L, hole depth H, wall thickness t, hole width w, rotor radius R, Radial clearance C_r and Axial hole numbers are all same for both seals, with the exception of some central holes with a diameter of 0.5 mm. The length of the outlet was extended appropriately for the full development of turbulence.

2.2   Numerical Method and Grid Meshing

In this work, the commercial software ANSYS Fluent was employed to simulate the three-dimensional compressible flow in the seals. Table 2 lists the working parameters and boundary conditions in this paper. Many studies have proven the reliability of the standard $k-\varepsilon$ turbulence model in simulating the flow characteristics in seals [12, 13]. Therefore, the standard k-ε turbulence model with a scalable wall function is adopted. The working substance is set as the ideal air gas, and the SIMPLE method is used to solve the pressure distribution in the sealing gap. The inlet and outlet boundary conditions are set to the inlet total pressure boundary and outlet static pressure boundary, respectively. The calculation is considered to meet requirements when all the residuals of the turbulence
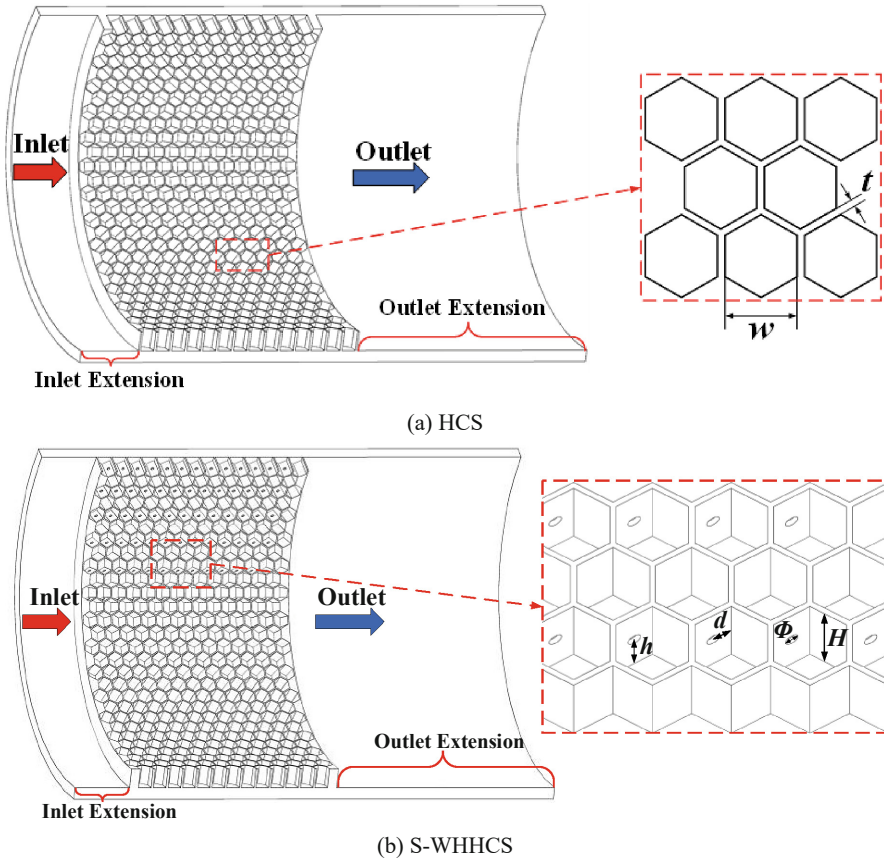
(a) HCS

(b) S-WHHCS

**Fig. 1.** Schematic geometry of the HCS and S-WHHCS.

equation, continuity equation, and momentum equation are lower than 10^{-6}, and the relative error between inlet and outlet mass flow rate is less than 0.01%.

The full three-dimensional models of HCS and S-WHHCS were meshed through ICEM with sensible approaches. The meshing details of the seals are shown in Fig. 2. Since the calculations in this paper belong to hydrodynamics, the grid independence was verified by drawing different numbers of meshes. The grid independence verification was calculated using the parameters in Table 2. ($P_{in} = 0.5$ Mpa, $\omega = 5000$ rpm), and the results are shown in Fig. 3. The number of the grids adopted in the HCS are 3.61, 5.02, 5.80 and 9.59 million, which means that the maximum volume of control volumes decreased from 0.0382 mm^3 to 0.0258 mm^3. The number of the grids adopted in the S-WHHCS are 4.61, 6.13 and 9.93 million, which is more than the HCS. As wall holes are added, the meshes around the holes are correspondingly encrypted. The leakage flow rate sharply declines with increasing grid number from 3.61 to 5.02 million for the HCS and 4.61 to 6.13 million for the S-WHHCS. However, the leakage flow rate only

decreased 0.47% and 0.36% when the number of the grids increased from 5.8 to 9.6 million for the HCS and 6.13 to 9.93 million for the S-WHHCS.

Table 1. Geometric parameters of HCS and S-WHHCS.

Parameters of two seals	Value
Seals length, L/mm	38.0
Inlet extension, L_i/mm	10.0
Outlet extension, L_o/mm	40.0
Rotor radius, R/mm	30.0
Radial clearance, C_r/mm	0.20
Hole depth, H/mm	3.30
Hole width, w/mm	2.50
Wall thickness, t/mm	0.20
Axial hole numbers, N_1	14/13
Hole diameter of S-WWHCS, ϕ/mm	0.50
Distance between hole and bottom, h/mm	1.65
Distance between hole and side, d/mm	0.72
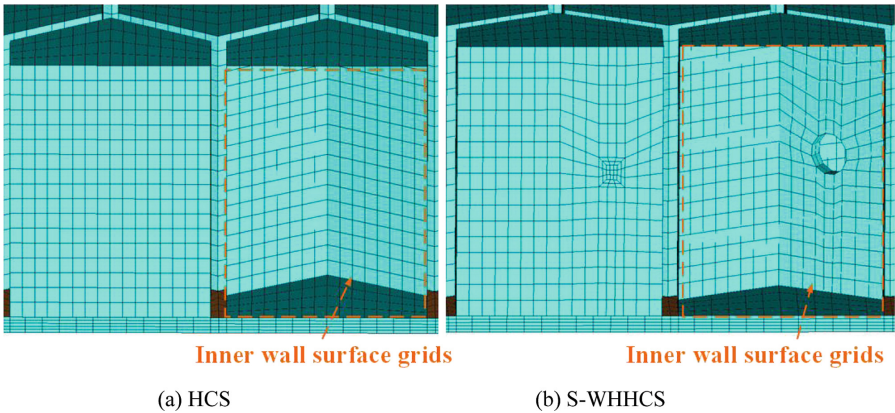
Inner wall surface grids   Inner wall surface grids

(a) HCS   (b) S-WHHCS

Fig. 2. Mesh details of two different honeycomb seals.

3 Results and Discussion

In this paper, a range of numerical simulations were carried out based on the working conditions, which are already shown in Table 2. The influence of the pressure and rotational speed on the leakage rate of the HCS and S-WHHCS is discussed. The pressure
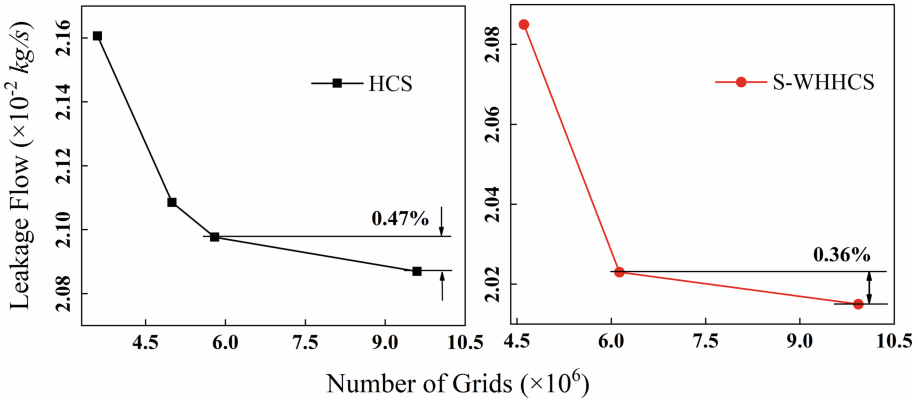
Fig. 3. Grid independence verification ($P_{in} = 0.5$ Mpa, $\omega = 5000$ rpm)

Table 2. The working parameters and boundary conditions.

Parameters	Value
Working substance	Air (Ideal gas)
Turbulence model	Standard $k - \varepsilon$
Wall properties	Adiabatic, smooth wall
Inlet pressure, P_{in}/MPa	0.2, 0.3, 0.5, 0.8
Outlet pressure, P_{out}/MPa	0.1
Rotating speed, ω/rpm	0, 1000, 3000, 5000, 7000, 10000

field and the turbulence kinetic energy of the HCS and S-WHHCS are obtained, and their leakage characteristics are compared and analysed.

Figure 4 displays the leakage flow rate of the HCS and S-WHHCS at different rotating speeds and operating pressures. The leakage flow rates of both the HCS and S-WHHCS increase with increasing operating pressure. With the same operating pressure, the leakage flow rate of the S-WHHCS does not change significantly when the rotational speed is increased. This indicates that the leakage flow rate of S-WHHCS is insensitive to the growth of the rotating speed. However, the leakage flow rate of the HCS declined as the rotating speed growing, which means that the HCS is more sensitive to the variation in rotational speed than the S-WHHCS. Meanwhile, it can be seen that the leakage flow rate of the HCS is always larger than that of the S-WHHCS, regardless of the operating pressure and rotating speed. This phenomenon demonstrates that the S-WHHCS has a stronger capacity for fluid leakage control and exhibits excellent sealing performance over the HCS under a wide range of operating conditions.

To investigate the sealing performance of the S-WHHCS in detail, the influence of operating conditions on the flow leakage rate was plotted, as shown in Fig. 5. It is apparent that the S-WHHCS has a certain degree of leakage reduction effect when the operating

pressure ranges from 0.2 MPa to 0.8 MPa, and the leakage reduction amplitude increases as the operating pressure increases. What's more, the reduction ratio of flow leakage shows a declining trend with increasing rotation speed. This shows that the S-WHHCS is more appropriate for application in high operating pressure and low rotating speed conditions, where the sealing performance of the S-WHHCS can be better exploited.

Figure 6 shows the turbulence kinetic energy along the axial direction for the HCS and S-WHHCS when the inlet pressure is 0.3 MPa and the rotating speed is 3000 rpm. It is obvious that the turbulence kinetic energy of both seals reaches its maximum at the exit, which indicates that the turbulent dissipation effect occurs when the fluid flows through the seals. At the midstream and downstream of the seals, the turbulent kinetic energy growth rate of the S-WHHCS is higher than that of the HCS. At the outlet, the S-WHHCS has a significantly larger peak turbulent kinetic energy, while the HCS has a more uniform turbulent kinetic energy distribution. These phenomena indicate that the holes in the honeycomb wall have a significant effect on the turbulent kinetic energy distribution of the seal and the S-WHHCS is more helpful in dissipating the energy of the fluid.
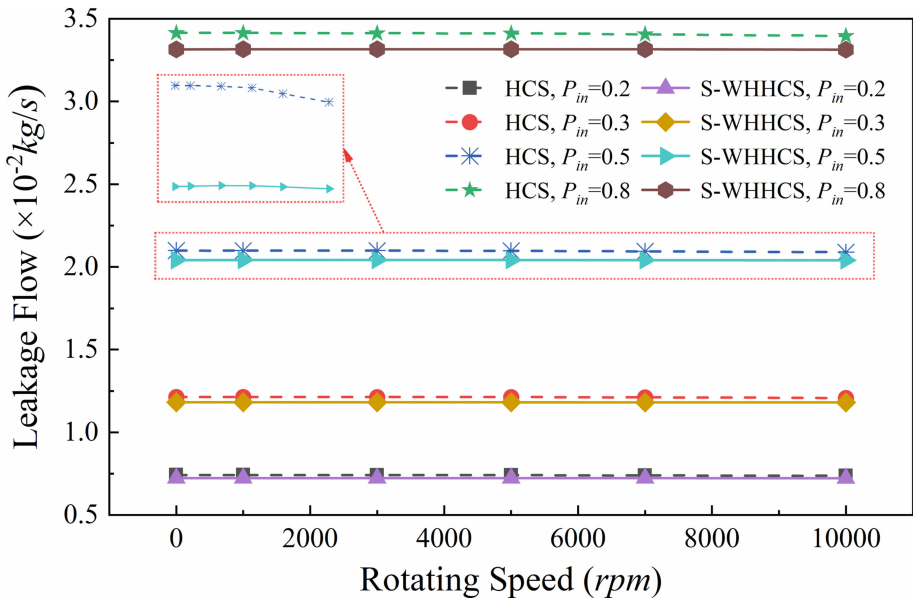

Fig. 4. Flow leakages of the HCS and S-WHHCS under different operating conditions.

To reveal the flow leakage reduction mechanism of the S-WHHCS, the static pressure distribution in the neutral plane for two seals ($P_{in} = 0.3$ Mpa, $\omega = 3000$ rpm) is drawn in Fig. 7. In each honeycomb cavity, the value and the increase in the amplitude of the static pressure in the S-WHHCS are greater than those in the HCS, which means that the vortex dissipation in the cavity of the S-WHHCS is more pronounced. When the fluid flows out of the cavities, the convergent gas compressed in the sealing gap due to the influence of fluid inertia, leads to a sudden decrease in static pressure. It is clear that
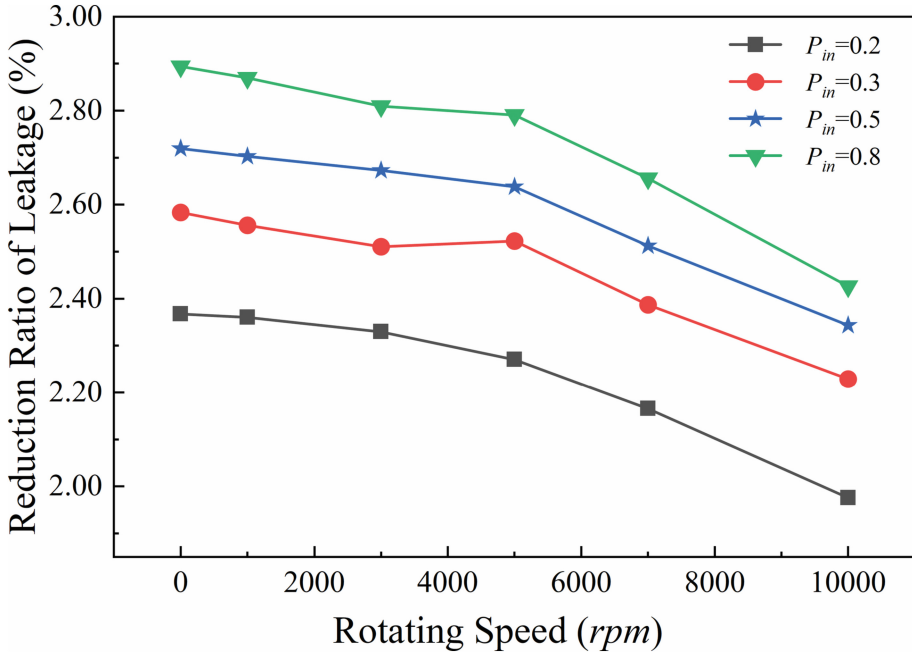
Fig. 5. Influence of operating conditions on the flow leakage rate.

the static pressure drop in the sealing gap is greater for S-WHHCS, indicating a more obvious throttling contraction effect. In conclusion, the S-WHHCS has stronger vortex dissipation strength and outlet throttling shrinkage effect than the HCS, which is the main reason why the S-WHHCS exhibits excellent leakage control capacity.
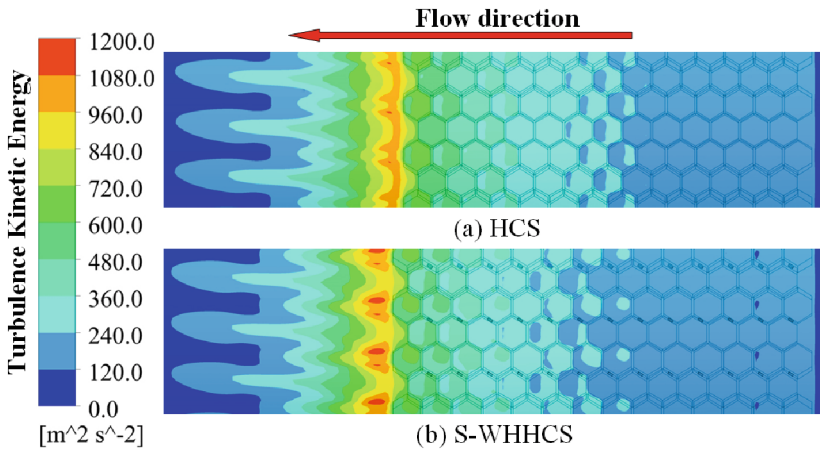

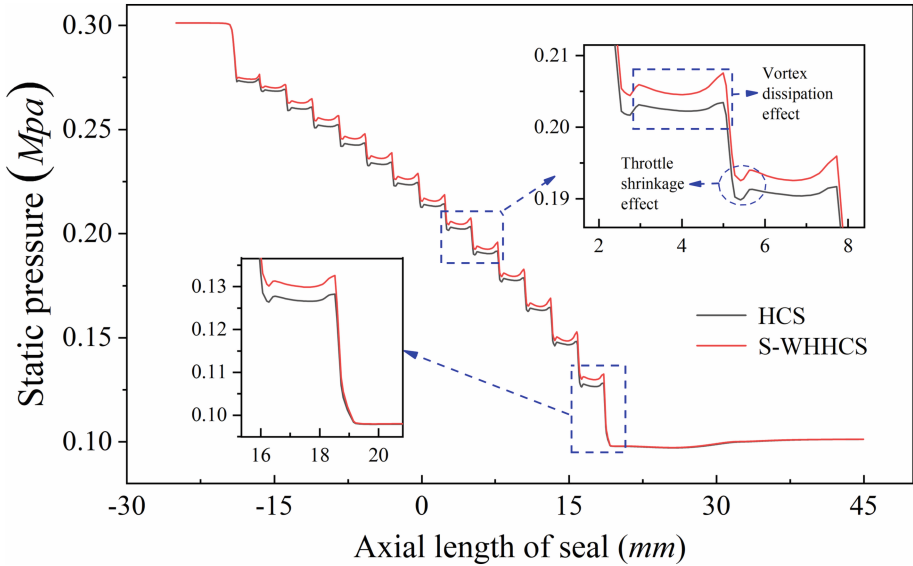**Fig. 6.** The turbulence kinetic energy along the axial direction for the two kinds of seals.

Fig. 7. The static pressure distribution in the meridian plane for two types of seals.

4 Conclusions

In this paper, a kind of single wall hole honeycomb seal is proposed, and its leakage characteristics are studied under different operating conditions by a numerical method.

The results show that the flow leakage of the HCS and S-WHHCS shows a positive correlation with the static pressure changes. The S-WHHCS is insensitive to rotating speed changes, and it can reduce the flow leakage rate under a large range of working conditions. Meanwhile, the S-WHHCS has the best leakage reduction effect under high pressure and low speed situations compared to the HCS.

The hole in the honeycomb walls has significant effects on the turbulence kinetic energy distribution and pressure drop. Compared with the HCS, the turbulent kinetic energy in the S-WHHCS is more irregularly distributed and has larger peaks. Moreover, the S-WHHCS has a higher intracavity static pressure and outlet pressure drop, which results in a stronger vortex dissipation intensity and outlet throttling contraction effect. This is the main reason why S-WHHCS has superior leakage control ability.

References

1. Lakshminarayana, B.: Fluid Dynamics and Heat Transfer of Turbomachinery, pp. 339–347. Wiley, New York (1996)
2. Yucel, U., Kazakia, J.Y.: Analytical prediction techniques for axisymmetric flow in gas labyrinth seals. J. Eng. Gas Turbines Power-Trans. Asme. **123**(1), 255–257 (2001)
3. Gamal, A.J.M., Vance, J.M.: Labyrinth seal leakage tests: tooth profile, tooth thickness, and eccentricity effects. J. Eng. Gas Turbines Power **130**(1), 012510 (2008)
4. K.K. Nielsen, K. Jnck, H. Underbakke: Hole-pattern and honeycomb seal rotordynamic forces: validation of CFD-based prediction techniques. J. Eng. Gas Turbines Power 134, 1e10 (2012). https://doi.org/10.1115/1.4007344
5. Zahorulko, A.V., Lee, Y.B.: Computational analysis for scallop seals with sickle grooves, part II: rotordynamic characteristics. Mech. Syst. Signal Process. **147**, 107154 (2021). https://doi.org/10.1016/j.ymssp.2020.107154
6. Lisyanskii, A.S., Gribin, V.G., Sakhnin, Yu.A., Fat'kov, O.V., Gorlitsyn, K.V., Ushinin, S.V.: Practical experience with the introduction of honeycomb shroud seals on 250–800 MW supercritical pressure units. Power Technol. Eng. **47**(6), 440–445 (2014)
7. Childs, D.W., Kim, C.H.: Analysis and testing for rotordynamic coefficients of turbulent annular seals with different, directionally homogeneous sur-face-roughness treatment for rotor and stator elements. ASME J. Tribol. **107**(3), 296–306 (1985)
8. Yan, X., Li, J., Feng, Z.: Effects of inlet Preswirl and cell diameter and depth on honeycomb seal characteristics. ASME J. Eng. Gas Turbines Power **132**(12), 122506 (2010)
9. Li, J., Kong, S., Yan, X., et al.: Numerical investigations on leakage performance of the rotating labyrinth honeycomb seal. ASME J. Eng. Gas Turbines Power **132**(6), 062501 (2010). https://doi.org/10.1115/1.4000091
10. Zhang, W., Gu, C., Yang, X., et al.: Effect of hole arrangement patterns on the leakage and rotordynamic characteristics of the honeycomb seal. Propul. Power Res. **11**(2), 181–195 (2022). https://doi.org/10.1016/j.jppr.2022.03.002
11. Jiang, J., Zhang, X., Zhao, W., et al.: Leakage characteristics of a novel hole-pattern damping seal with dovetail-like diversion grooves: numerical simulation and experiment. J. Eng. Gas Turbines Power **144**(7), 071004 (2022)
12. Zhang, X., Jiang, J., Peng, X., Li, J.: Leakage and rotordynamic characteristics of labyrinth seal and hole-pattern damping seal with special-shaped 3D cavity. Ind. Lubr. Tribol. **73**(2), 396–403 (2021). https://doi.org/10.1108/ILT-07-2020-0262
13. Li, Z., Li, J., Feng, Z.: Numerical investigations on the leakage and rotordynamic characteristics of pocket damper seals—Part I: effects of pressure ratio, rotational speed, and inlet Preswirl. J. Eng. Gas Turbines Power **137**(3), 032503.1-032503.15 (2015). https://doi.org/10.1115/1.4028373