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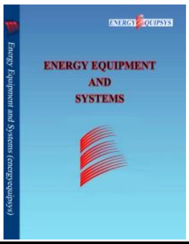

Effect of a ring type barrier and rotational speed on leakage flow of gas turbine brush seal

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ABSTRACT

This paper investigates the effect of inserting a ring type barrier on leakage flow of brush seals with different bristles clearances (the distance between bristle pack tip and rotor surface). The ring is placed on both upstream and downstream sides of the bristles. An axisymmetric CFD model is employed to calculate radial pressure distribution along backing plate, axial pressure variation on rotor surface, and leakage mass flow rate of the brush seal. Reynolds-Averaged-Navier-Stokes (RANS) together with non-Darcian porous medium approach is performed to solve the flow field. The accuracy and reliability of the model are evaluated through comparison of the numerical results and experimental data. The results show that inserting the ring is not effective for the brush seal with zero clearance, neither at upstream nor at downstream. In other cases, the downstream ring is considerably more effective than the upstream one, when the ring is tangent to the back of bristles. The greater the distance between the bristles and the ring, the less reduction in leakage flow. Also, the best performance is obtained for the ring height equal to clearance size. Moreover, the effect of rotor rotation on leakage flow is investigated. The results show a negligible decrease in brush seal leakage flow with increasing the rotational speed.

Keywords: Brush Seal, Leakage, Ring, CFD, Barrier, Rotational Speed.

1. Introduction

In recent years, circular type brush seals are increasingly demanded to reduce leakage flow in gas turbines. Figure 1 illustrates configurations of a brush seal with clearance. It normally consists of three components: front plate, bristle pack, and backing plate. The bristle pack is made of wires and is clamped at a lay angle between backing and front plates. Brush seals are used to enhance power output and efficiency in gas and steam turbines.
Ferguson [1] showed less leakage and better sealing performance of brush seals compared to conventional labyrinth seals. Bayley and

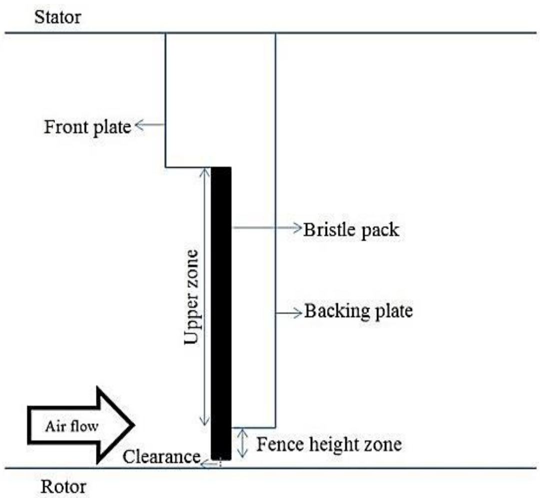

Fig.1. Brush seal configurations

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Long [2] used the Darcian porous medium approach to calculate the leakage flow of the brush seal. They compared the results to their experimental data. In their model, Darcian porous medium approach was solved and inertial terms were neglected resulting in a balance equation between viscous forces and pressure. Further investigations of fluid flow in brush seals showed better results of non-Darcian porous medium approach compared to the Darcian model. Chew et al. [3] studied a two-dimensional axisymmetric model. Added resistance forces were assumed for the bristles. They chose the resistance coefficients through calibration against calculated data. Both the inertial and viscous resistance coefficients were considered in their model. Chew and Hogg [4] applied the non-Darcian porous medium approach. They presented a one-dimensional model to show good agreement between predictions from the model and the experimental results. They utilized three experimental data sources of Bayley and Long [2], Carlile et al. [5] and O'Neill [6]. Chen et al. [7] utilized higher resistance coefficients for the bristle pack. The results were in good agreement with the experimental data at higher pressure ratios. Dogu [8] performed the bulk porous medium approach to study the flow field in bristle pack. The bristle pack was divided into two different regions; the upper zone and the fence height zone. Two spatial directions of axial and radial were assumed in the analysis. Finally, it was founded that the flow resistance for the upper zone is 20% higher than the fence height zone. Dogu and Aksit [9, 10] studied a CFD model to investigate influences of front and backing plate geometries on leakage and pressure distribution. Li et al. [11] investigated the effects of clearance sizes on labyrinth brush seal leakage performance. Pugachev and Helm [12] also employed the porous medium approach. They used the calibrated model to calculate pressure distribution and leakage of different brush seals with different bristle pack thickness sizes. Aslan-zada et al. [13] made a comparison between brush seals and conventional labyrinth seals and showed the advantages of brush seals over labyrinth seals. Huang et al. [14] used a CFD model. They employed a retaining ring on the downstream side of the brush seal located under the backing ring. It showed reduction of leakage flow in the brush seal with radial clearance. Qiu et al. [15] predicted the leakage characteristics of multi-stage brush seals using a numerical method and experimental data. They investigated the effects of sealing clearance and rotor speed on pressure distribution and leakage of the brush seal. Gresham et al. [16] presented a CFD simulation to measure the permeability of the bristle pack. Effects of pressure ratio on the permeability were studied in their model. Sun et al. [17] conducted a study based on a three-dimensional model of brush seal. They measured the leakage of the brush seal. The leakage of the brush seal, considering the effect of bristle deflection, was closer to the experimental data in their model.

To model a fluid flow in the bristle pack of brush seal, there are some limitations that need to be resolved. One of the most important issues is the short space between components especially the space between rotor and bristles tip. The gap between bristles tip and rotor (clearance) should be long enough to prevent the possible collision between them. Also, long distance between rotor and bristle pack can lead to an increase in leakage flow. In a previous work, Bahadori and Zirak [18] reported a comprehensive investigation of the effects of various brush geometricalparameters on leakage flow. They presented the leakage flow rate variations with increasing the clearance for various values of pressure ratio. As mentioned before, many studies have been conducted to reduce the leakage flow in gas turbine brush seal. However, only one of them [14] has used a barrier type ring. Therefore, there is a strong need for further study of the impact of different types of barrier rings of leakage of brush seals. In the present work, a CFD model is applied to show the effect of inserting a ring type barrier, on both sides of the bristle pack, on the leakage flow of the brush seal. Different geometries are defined to be compared to the case of brush seal without ring. The analysis is performed for various values of bristle pack clearances from rotor. In addition, effect of axial distance between bristles and the barrier ring on leakage flow is investigated for clearance of 0.5 mm. Moreover, the influence of rotor speed on leakage flow is reported. Figure 2 presents brush seal dimensions without clearance [2].

Bayley and long's paper [2] is one of a few papers with all the details of experimental pressure and leakage data. All the information for boundary conditions and geometry was presented in their study. Therefore, Bayley and long's paper is selected as a basis for the present work.

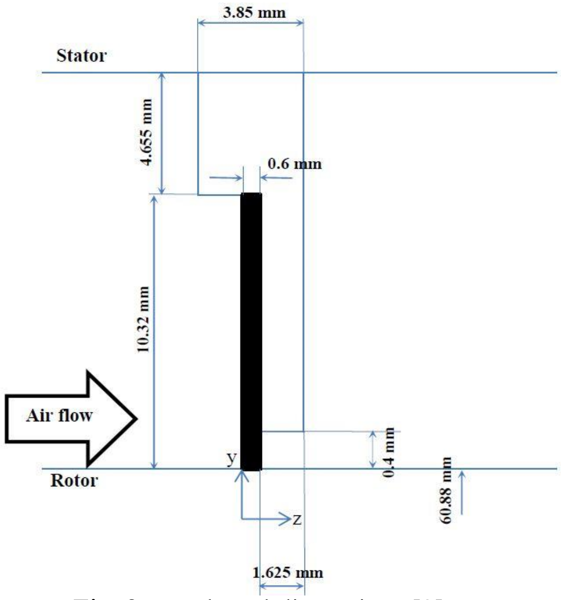


Fig. 2. Brush seal dimensions [2]

Nomenclature

a	viscous resistance coefficient (1/m^2)
A	fence height area (A)
b	inertial resistance coefficient (1/m)
d	bristle diameter (mm)
E	total energy (J)
h_{bf}	bristle free height (mm)
k	turbulent kinetic energy, thermal conductivity (m^2/s^2)
k_{eff}	effective thermal conductivity (w/mk)
l	bristle pack thickness (mm)
L	dimensionless bristle pack thickness
m	mass flow rate (kg/s)
N	number of bristles per unit length
p	static pressure (Pa)
p^*	dimensionless pressure
P	dimensionless pressure coefficient
R	gas constant (J/kgk)
Re	Reynolds number
R_p	pressure ratio
S_h	volumetric heat source (w/m^3)
T	static temperature (k)
u	velocity component (m/s)
V	average velocity (m/s)
x	spatial coordinate direction (mm)
y	radial distance from rotor surface (mm)
Y	dimensionless radial coordinate
z	axial coordinate direction (mm)
α	inertial resistance, empirical constant (kg/m^4)
β	viscous resistance, empirical constant (kg/m^3)
ε	porosity, turbulent dissipation (m^2/s^3)
μ	dynamic viscosity (Pas)
μ_t	turbulent viscosity (Pas)
ρ	density (kg/m^3)
$(\tau_{ij})_{eff}$	effective stress tensor (Pa)
$\emptyset$	bristle lay angle

Subscripts

d	downstream
i	spatial coordinate
j	spatial coordinate
r	radial coordinate
u	upstream
z	axial coordinate
θ	tangential coordinate

2. Analytical model

An analytical method was presented by Chew and Hogg [4] to calculate leakage flow through bristle pack. It considers a quadratic pressure gradient relation across the seal as:

$$-\frac{dp}{dz} = a\mu V + b\rho V^2 \tag{1}$$

where z is axial coordinate and p is the pressure. The average velocity is $V = m/\rho A$ and density of air is $\rho = \frac{p}{RT}$. Replacing these terms in Eq. (1) gives:

$$-\frac{p}{RT}dp = \frac{a\mu}{A}mdz + \frac{bm^2}{A^2}\,dz \tag{2}$$

in which A is the fence height area. a and b, are resistance coefficients. m represents the mass flow rate and μ is the dynamic viscosity of air. R is defined as gas constant and T is the air temperature. Integrating Eq. (2) from upstream pressure, p_u, to downstream pressure, p_d, corresponding to $z = 0$ to l,

$$\left(\frac{m}{A}\right)^2 + \left(\frac{a\mu}{b}\right)\left(\frac{m}{A}\right) - \frac{p_u^2 - p_d^2}{2RTlb} = 0 \tag{3}$$

Defining Reynolds number, $Re = \frac{md}{A\mu}$ and dimensionless pressure coefficient $= \frac{(p_u^2 - p_d^2)d^2}{2RT\mu^2}$, where d is bristle diameter. rearranging Eq. (3) gives:

$$Psin\emptyset/Nd = Re(bdRe + ad^2)L \quad (4)$$

Equation 4 shows a relation between parameter P and Re where, $ad^2 = \frac{80\alpha(1-\varepsilon)^2}{\varepsilon^3}$ and $bd = \frac{\beta(1-\varepsilon)}{2\varepsilon^3}$. α and β are empirical constants which according to reference [19], α =1 and β =2.32. $\varepsilon = 1 - \frac{\pi d^2 N}{4lsin\emptyset}$ is the porosity and $L = lsin\emptyset/Nd^2$ is dimensionless bristle pack thickness.
where N is number of bristles and $\emptyset$ is the bristle pack lay angle. l is defined as bristle pack thickness.

3. Numerical methodology

3.1 Governing equations

The geometry of model is divided into two parts; the bristle pack region and the upstream and downstream cavities. Outside the bristle pack region, the Navier-Stokes equations (conservation of mass and momentum) are solved. The *k*-*ε* turbulent model is utilized to calculate the Reynolds stresses. The Boussinesq hypothesis is used to evaluate the Reynolds stresses. Energy equation is shown in Eq. (8). The Navier-Stokes equations can be written as [20]:

$$\frac{\partial}{\partial x_j}(\rho \bar{u}_j) = 0 \quad (5)$$

$$\frac{\partial}{\partial x_j}(\rho \bar{u}_i \bar{u}_j) = -\frac{\overline{\partial p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\overline{\partial u_i}}{\partial x_j} + \frac{\overline{\partial u_j}}{\partial x_i} - 2/3\delta_{ij}\frac{\overline{\partial u_l}}{\partial x_l}\right)\right] + \frac{\partial}{\partial x_j}(-\rho\overline{\acute{u}_i \acute{u}_j}) \quad (6)$$

Reynolds stresses,

$$-\rho\overline{\acute{u}_i \acute{u}_j} = \mu_t\left(\frac{\overline{\partial u_j}}{\partial x_i} + \frac{\overline{\partial u_i}}{\partial x_j}\right) - \frac{2}{3}(\rho k + \mu_t\frac{\overline{\partial u_l}}{\partial x_l})\delta_{ij} \quad (7)$$

Energy equation,

$$\frac{\partial}{\partial x_j}[u_j(\rho E + p)] = \frac{\partial}{\partial x_j}\left[(k_{eff})\frac{\partial T}{\partial x_j} + (\tau_{ji})_{eff}u_i\right] + S_h \quad (8)$$

There are additional resistance forces in the bristle pack region. For bristle pack region, the following relation is used by many researchers:

$$-\frac{dp}{dx_i} = a_i\mu u_i + b_i\rho|u|u_i \quad (9)$$

a and b are viscous and inertial resistance coefficients. Equation (9) can be written in other form as:

$$-\frac{dp}{dx_i} = (\alpha_i|u| + \beta_i)u_i \quad (10)$$

The mentioned equations show the non-Darcian porous medium approach. Dogu [8] utilized Eqs. (9) and (10) to describe flow field in bristle pack. In Eq. (10), α and β are inertial and viscous resistances. Chew et al. [3] added the right hand side of Eq. (10) to the right hand side of the momentum equation, Eq. (6). Furthermore, the inertial terms on the left hand side and the viscous terms on the right hand side of Eq. (6) were neglected in their analysis which it yields a relation between resistances and pressure gradient as Eq. (10).

3.2 Resistance coefficients

Dogu [8] defined inertial and viscous resistances through calibration. According to that, together with the data in reference [14], the resistance coefficients can be given in Table 1. Here, z and r refer to axial and radial directions, respectively. If the rotor rotates, extra viscous and inertial resistance coefficients in tangential direction are added to the model. The effects of pressure ratio on resistance coefficients are ignored. Normally flow resistances for the upper zone are 20% higher than those of the fence height zone.

ANSYS-FLUENT software is used to calculate 2D axisymmetric air flow through the brush seal. For the steady compressible air flow, the ideal gas law is used. As stated above, the *k*-*ε* turbulent model is utilized to calculate the Reynolds stresses. The basis for selecting the standard *k*-*ε* turbulent model is that for brush seals, the following turbulent model has been used successfully by many researchers. The SIMPLE algorithm is used for the simulation. The porous medium approach is defined for the region of bristle pack. Up/downstream pressure and temperature are used for boundary

Table 1. Resistance coefficients [8]

Bristle pack	$a_z\ (m^{-2})$	$a_r\ (m^{-2})$	$b_z\ (m^{-1})$	$b_r\ (m^{-1})$
Fence height zone	2.51×10^{12}	5.58×10^{9}	5.81×10^{6}	7.75×10^{4}
Upper zone	3.01×10^{12}	6.69×10^{9}	6.97×10^{6}	9.30×10^{4}

conditions. The downstream static pressure is kept at 100 kPa and the upstream total pressure is increased from 125 to 400 kPa, corresponding to a pressure ratio from R_p=1.25 to 4. The air temperature is 20° C. No-slip boundary condition is selected at the walls.

3.3 Grid independency

Figure 3 indicates the results of mass flow rate for different mesh sizes (at Pressure ratio equal to =2.5). As it is observed, there are not any considerable changes in results for grid with more than 27189 nodes. Figure 4 shows its corresponding mesh layout. As it is observed, the grid type is structured and the mesh is denser at porous zones (bristles area) and near the walls.

4. Results and discussion

4.1. Validation

Figure 5 compares the results of leakage mass flow rate with those of experimental and previous mentioned analytical for various values of pressure ratios in the brush seal without ring and clearance.

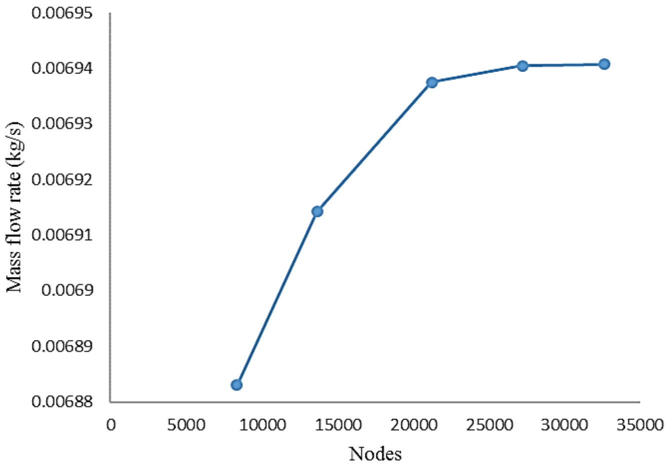


Fig. 3. mesh independence study

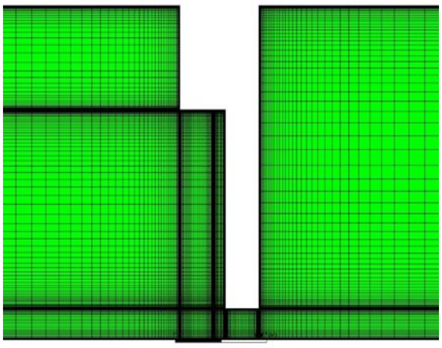


Fig. 4. Computational mesh of the brush seal

It shows a good agreement. In Fig. 6, the dimensionless pressure, defined as $p^* = \frac{p-p_d}{p_u-p_d}$, is plotted against the dimensionless radial coordinate $Y = y/h_{bf}$, where, h_{bf} is the radial distance between the bristle pack and the front plate tip. As it is shown in Fig. 6, p^* increases with increasing the pressure ratio . It is clear that the analytical method doesn't contain such a kind of calculation. Figure 7 demonstrates dimensionless pressure, p^*, versus axial coordinate, z. The difference between present CFD results and experimental data in Fig. 7 (the offset from the experimental line) is because the bristle pack is assumed to stay fixed at its position in the numerical model while it is deflected axially during the operation and tests [2, 8].

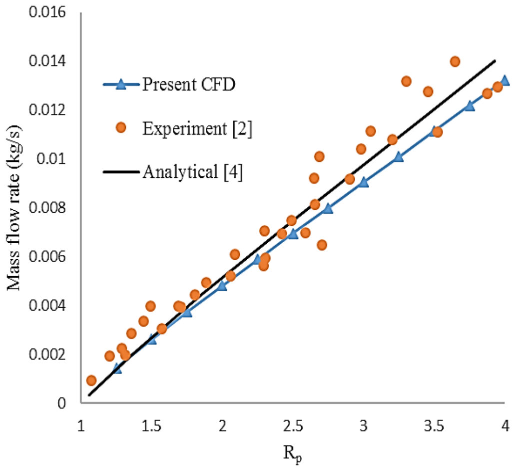


Fig. 5. Comparison of CFD results with experimental data and analytical results

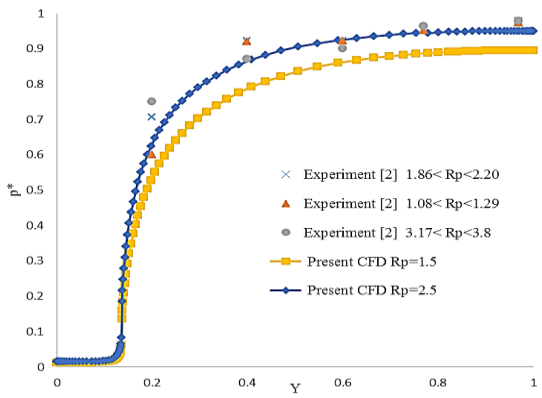


Fig. 6. Comparison of radial pressure distribution along backing ring

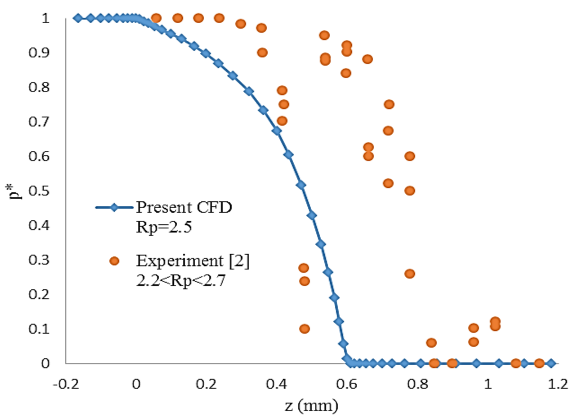


Fig. 7. Comparison of axial pressure distribution on rotor surface

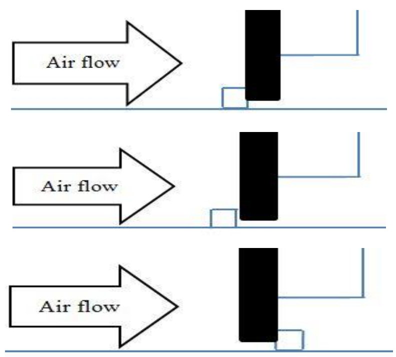
The figure states that the pressure reduction on rotor surface mostly occurs at the area where the bristles are in contact with the rotor. For more clarity, pressure distribution along the backing plate and the rotor are shown by static pressure contours for all cases in Fig. 10.

4.2. Flow field changes in the presence of barrier ring

When a ring is added to the conventional geometry of the brush seal, it blocks the flow and hence less air flow passes through the clearance between the rotor and the bristle pack tip. If the ring is employed to reduce the leakage flow of the bristles, the value of the maximum velocity decreases. The Clearance and the barrier ring dimensions are two significant factors to change the flow field, pressure distribution and velocity magnitude resulting in a change in leakage. As it will be stated, the least leakage flow rate occurs when the ring height and the clearance have the same value.

4.3. Effect of added barrier ring on leakage

Effects of adding a barrier ring on the upstream and downstream sides of the bristle pack on leakage of the brush seal with different clearance sizes are investigated in the CFD model. Three geometries, as shown in Fig. 8(a), are defined for the new design of the brush seal relative to the conventional designs, 1) brush seal with an upstream ring tangent to the bristle pack, 2) brush seal with an upstream ring with a 0.15 mm distance in front of the bristle pack, and 3) brush seal with a downstream tangent ring. The ring cross section is a square of 0.5 mm sides. The operating (test) pressure ratio is R_p=2.5 for all cases. Table 2 gives the percentage of mass flow rate decrease for various clearance sizes relative to the case of brush seal without ring. As illustrated in Fig. 8(b), the clearances are selected to cover the range of opening from zero to beyond the ring height. The ring blocks the flow and hence the flow total pressure decreases resulting in a decrease in brush seal leakage.



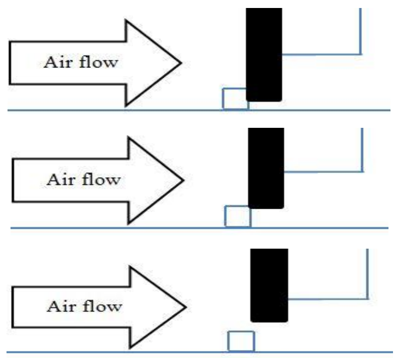


Fig. 8. Brush seal with added ring in different positions

Table 2. Percentage decrease in leakage for new designs in comparison with the brush seal without ring

Clearance (mm)	Ratio of clearance to the distance between backing plate tip and rotor	Brush seal with a tangent ring on the upstream side	Brush seal with a ring with 0.15 mm distance on the upstream side	Brush seal with a tangent ring on the downstream side
0	0	1.1	0.06	6.35
0.1	0.071	33.7	1.9	54.9
0.2	0.143	60.6	16.9	74.4
0.3	0.214	72.3	31.5	82.4
0.4	0.286	78.5	44.3	86.5
0.5	0.357	81.7	52	88.4
0.6	0.429	71.1	52.4	75.4
0.7	0.5	62.3	50.3	66
0.8	0.571	55	46.4	58.9

As it is seen:

- Adding a ring, on upstream or downstream side, causes a considerable decrease in leakage in all cases except for the case with zero clearance. In other words, there is not any gain in leakage reduction by using a barrier ring when the bristle pack is in contact with rotor.
- The maximum performance effect of the ring, on upstream or downstream side, is observed for the ring with the same height as clearance, although the leakage value in brush seal with 0.15 mm distance is almost equal for both 0.5 and 0.6 mm clearances.
- In all cases, the ring on downstream side decreases the leakage more than the upstream rings.
- The upstream tangent ring is considerably more effective in reducing the leakage compared to the ring at a distance (here, 0.15 mm) ahead of bristle pack.

Figure 9 gives a plot of mass flow rate versus clearance for the all cases. Contours of static pressure are compared in Fig. 10 for all designs with 0.5 mm clearance. Table 3 gives the results of changing the axial distance between the ring and the bristles on both upstream and downstream sides with 0.5 mm clearance. The upstream distance is considered sequentially as 0, 0.15, 0.30, and 0.45 mm. The results show the maximum decrease in leakage for 0 mm (tangent ring), and the effect of presence of the ring on leakage reduces with increasing the distance up to 0.45. For downstream ring, it is observed a very effective leakage reduction with tangent ring while for the ring with 1.625 mm distance (at the rear of backing plate), the presence of the ring almost does not affect the leakage value.

4.3. Rotational speed effects

To consider the effect of rotor speed on the leakage flow, additional viscous and inertial resistance coefficients are specified in the tangential direction. Leakage mass flow rate through the bristle pack is calculated for various rotor speeds from zero to 3000 rpm. Figure 11 shows the obtained results for four values of viscous resistance coefficients while the inertial resistance coefficient is neglected, $b_\theta = 0$. Figure 12 indicates the same results when considering various inertial resistance coefficients while viscous resistance coefficient is neglected, $a_\theta = 0$. There is a drop in leakage when rotor speed is included. The impact of rotational speed on leakage decreases in high range of viscous resistance coefficients. All the results show negligible effects of rotor speed on brush seal leakage value.

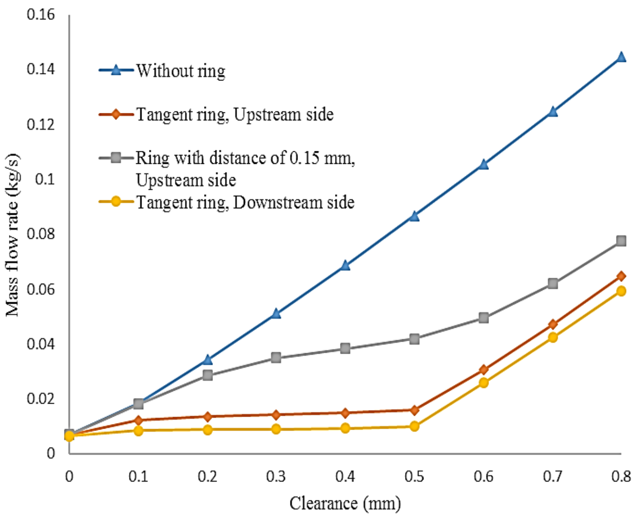
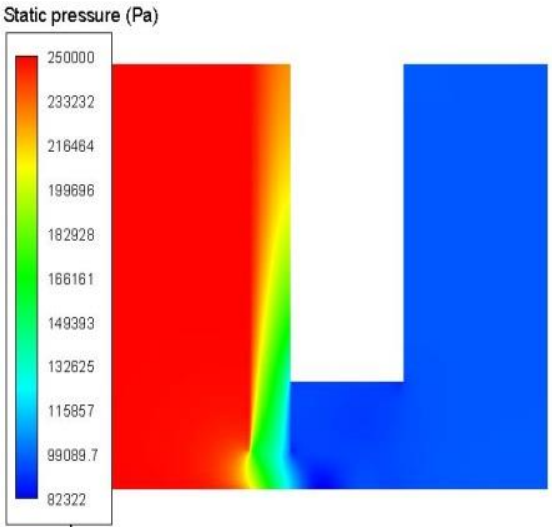


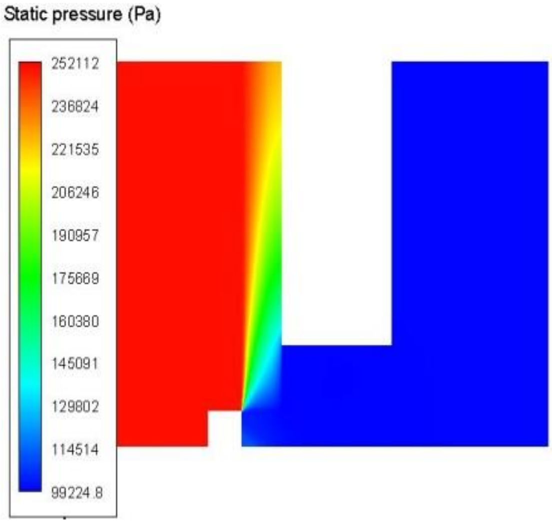
Fig. 9. Effect of employing a ring on brush seal leakage

Table 3. Percentage decrease in leakage for various axial ring positions in comparison with the brush seal without ring

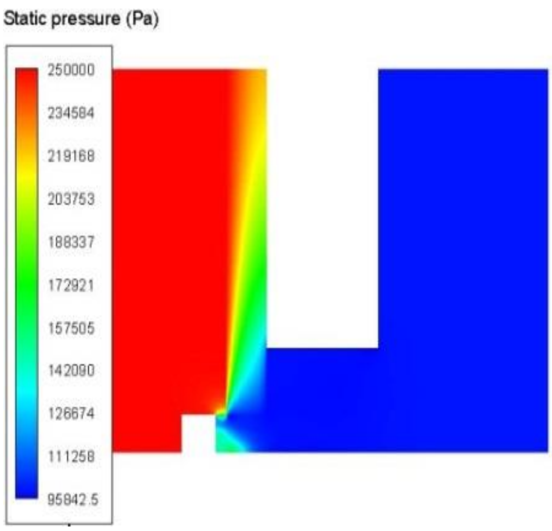
Upstream tangent ring	Upstream ring with 0.15 mm distance	Upstream ring with 0.30 mm distance	Upstream ring with 0.45 mm distance	Downstream tangent ring	Downstream ring with 1.625 mm distance
81.7	52	29.58	17.6	88.4	2.67



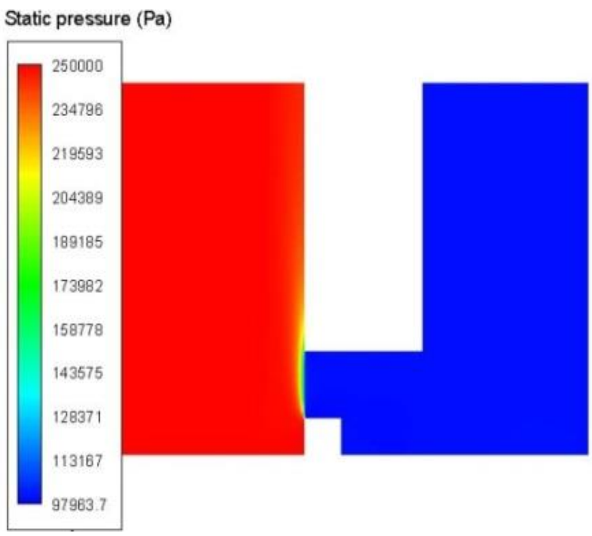
(a) Without ring



(b) Upstream tangent ring



(c) Upstream Ring with distance of 0.15 mm



(d) Downstream tangent ring

Fig. 10. Static pressure (a, b, c, d) contours of the brush seal with 0.5 mm clearance

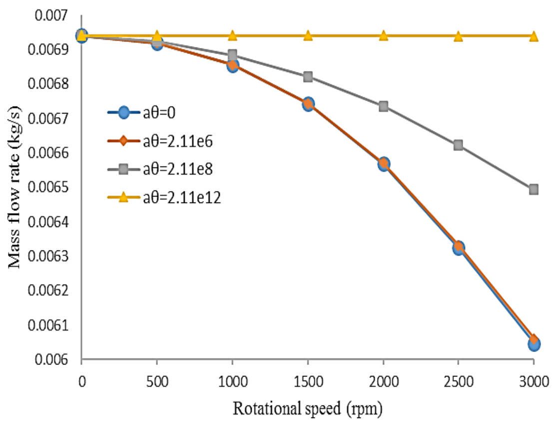


Fig. 11. Effect of rotor speed on leakage ($b_\theta = 0$)

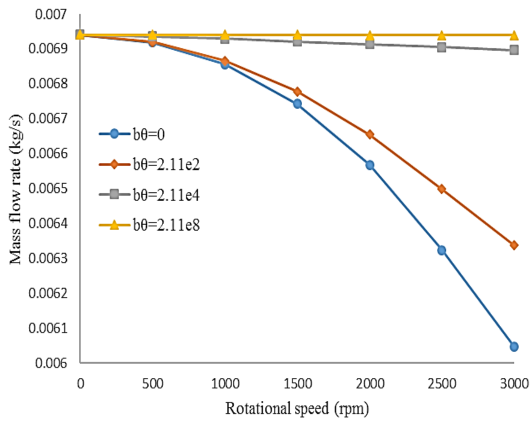


Fig. 12. Effect of rotor speed on leakage ($a_\theta = 0$)

5. Conclusions

A CFD model based on non-Darcian porous medium approach has been presented to investigate the influence of adding a barrier ring to the brush seal on leakage flow. The ring was placed on both upstream and downstream sides of the bristles. The results showed that 1) inserting the ring is not effective when the bristle pack tip is in contact with the rotor (zero clearance), neither at upstream nor at downstream, 2) in other cases, the downstream ring is considerably more effective than the upstream one, 3) the ring produces the best performance when its height is equal to the clearance, and finally, 4) the ring effectiveness is reduced when it is placed at a distance from bristles, the greater the distance between the bristles and the ring, the less reduction in leakage mass flow rate. However, this work proves adding a tangent ring specially on the downstream side of the bristles is a productive approach to decrease the leakage flow of the brush seal if the clearance is assumed to be nonzero. Moreover, the effect of rotor rotation was investigated. The results showed a slight decrease in brush seal leakage flow with increasing the rotational speed.

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