Predicting aerodynamic resistance of brush seals using computational fluid dynamics and a 2-D tube banks model[☆]

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ABSTRACT

Aerodynamic resistance of a brush seal was mainly studied. The velocity distribution along three specified lines was presented. By considering the pressure differential, Reynolds number and Euler number (Eu) were modified. The effect of geometric arrangements and pressure differentials on Eu and leakage were analyzed. Two correlations were fitted based on the numerical results. The results reveal the velocity distribution is almost flat, asymmetric along the specified lines. The velocity increases and decreases almost linearly at centerlines. Eu decreases gradually less with the increase of pressure differential and trends towards a fixed value. A larger Eu indicates stronger resistance but not necessarily less leakage. Finally, two fitted correlations are developed and one is exponential to the row number fits better.

1. Introduction

A brush seal, a novel type contact seal, has been widely used in recent decades. It is characterized by excellent leakage performance, low friction and rotor excursion capability. A typical brush seal consists of a back plate, a front plate and thousands of bristles, as illustrated in Fig. 1. Generally, a brush seal is used to reduce parasitic leakage between turning and stationary components of a turbomachinery.

Many literature have been focused on numerical simulation to analyze fluid flow in brush seals. One numerical model for assessing brush seals is the tube bank model, which contains in-line or staggered arranged circles. Dai and Liu developed a compact, staggered tube array numerical model and analyzed resistance which revealed an exponential relationship between energy loss and Reynolds number. In addition, the extent of resistance increased dramatically with a more compact geometry [1]. Huang et al. analyzed the fluid flow in a brush seal based on a two-dimensional (2-D) staggered tube bundle model. They argued that the sealing effect can be significantly enhanced by reducing the inter-tube spacing of the bristles for normal pressure differentials and the number of axial bristle rows [2]. Tan et al. proposed a meshing method which is able to achieve a better mesh independence and studied the separation points in tube banks models of a brush seal. They found that the separation point may move to the rear part of the bristle when the brush seal is subjected to a high pressure differential

[3]. Wang et al. analyzed the flows of two-stage brush seals by staggered tube bank model [4].

In addition to analyze flow across tube banks in the brush seal, several studies over the years have adopted tube bank models in heat exchangers. Aerodynamic or hydraulic resistance is one of the primary quantities of heat exchangers and is characterized by the total pressure drop. This is a form of energy loss partially caused by neighboring tubes and service conditions. Jin et al. performed numerical studies about the pressure drop in H-type finned tube banks with 10 rows and presented the correlation of Euler number (Eu) for the 10-row tube banks model [5]. Chen et al. analyzed the relationship between pressure drop and geometric parameters. The predictable correlations of Eu were developed based on the experimental results [6]. Studying the pressure loss of tube banks with bimetallic fins, Kuntysht et al. found that tight-finned tube banks have better thermal efficiency than a loose one [7]. The study about the pressure drop in tube banks has been conducted in terms of geometry parameters such as longitudinal and transverse pitch ratios and working conditions such as Reynolds number.

A brush seal is used for secondary system sealing performance in a gas turbine, where a higher pressure ratio can improve global efficiencies. So the pressure differential for a brush seal is a key design factor. According to the literature, brush seal leakage increases exponentially with excessive pressure loads [9]. Bristle deformation, pressure stiffening effect, and hysteresis are associated with the

[☆] Fully documented templates are available in the elsarticle package on CTAN.

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Fig. 1. Schematic diagram of a typical brush seal [8].

pressure differential across a brush seal. Zhu argued that 0.3 ~ 0.35 MPa is the maximum load for these seals [10]. By contrast, Chupp et al. reported that the typical operation limit is 0.21 MPa [11].

A review of the literature presents that almost all studies about brush seal pressure capability are based on experience. Although no model can be comprehensive enough to be applicable for all geometric arrangements, a few cases can be used to test a hypothesis. A quantity, Eu , has been used to express the aerodynamic resistance of tube banks in heat exchangers. Therefore, to analyze the aerodynamic resistance of the brush seal, Eu is adopted in the present study. In addition, the effect of parameters, such as longitudinal and transverse pitch ratios, and pressure differentials, on aerodynamic resistance are studied. Furthermore, the correlation of Eu is developed based on numerical results. The results of this study can provide a theoretical basis for measuring resistance in brush seals and serve as a reference for industrial application.

2. Model description and numerical method

2.1. Physical model

A schematic diagram of a multiple-row brush seal tube bank model is presented in Fig. 2. The geometric parameters of the tube bank model are listed in Table 1. The geometry in the present study is by referring Huang's aforementioned model [2]. They found that the 1-column model and the 6-column model are equivalent for analyzing intertube flow.

To simplify the expression, $(A, B, C)_p^N$ stands for the different cases according to the geometric parameters in Table 1. A, B and C stands for the angle θ of 30°, 45°, 60°, respectively, where subscript p is the pressure differential and superscript N is the total row number of bristles in a brush seal. For example, $A_{0.1}^{10}$ denotes the case $\theta = 30^\circ$, pressure differential 0.1 MPa and row number 10.

2.2. Governing equations and boundary conditions

Before starting a computational fluid dynamics (CFD) numerical simulation, the flow status must be assessed. Previous studies reveal different opinions about the flow status in a brush seal. Some researchers believe that the fluid flow is turbulent [3,8,12] while others believe it becomes laminar. Huang et al. adopted the laminar model in compact tube banks and the turbulent model in spacious tube banks [2,13]. Zhang et al. argued that fluid flow is turbulent in the upstream

Table 1

Geometric parameters of a brush seal based on tube banks model.

Case	$\theta/^\circ$	S_D/d	S_L/d	S_T/d	δ/mm	d/mm	N
A	30	1.1	0.953	1.1	0.007	0.07	10, 14, 18
B	45	1.1	0.778	1.556	0.007	0.07	10, 14, 18
C	60	1.1	0.55	1.905	0.007	0.07	10, 14, 18

and downstream regions but becomes laminar in the intertube due to slow fluid flow [14]. According to the states of flow around a circular cylinder in different flow regimes, the transition (from laminar to turbulent) occurs at low Reynolds number [15]. A turbulent model was adopted in this research.

The fluid was assumed to be an ideal gas in a steady state. The governing equations of this research includes those of mass, momentum and energy conservation, all of which have been previously described elsewhere. The turbulent model is presented below. The standard $k - \varepsilon$ turbulent model was used and expressed as

$$\rho \frac{Dk}{Dt} = \frac{\partial}{\partial x_i} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right] + G_k + G_b - \rho \varepsilon - Y_M \quad (1)$$

$$\rho \frac{D\varepsilon}{Dt} = \frac{\partial}{\partial x_i} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_i} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} \quad (2)$$

where G_k is the formation of turbulent kinetic energy because of the average velocity gradients, G_b is the formation of turbulent kinetic energy due to buoyancy, and Y_M is the contribution of the fluctuating dilatation on compressible turbulence to the overall dissipation rate. μ_t is the eddy viscosity, which was modeled as $\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$ with $C_\mu = 0.0845$. The standard model yields $C_{1\varepsilon} = 1.44$, $C_{2\varepsilon} = 1.92$, $C_{3\varepsilon} = 0.09$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$ [16].

To guarantee inlet uniformity, the upstream domain was extended by fifteen times of the bristle diameter for the entrance section. To reduce the effect of reverse flow, the downstream domain was extended by twenty times of the bristle diameter for the exit section.

There are four different boundary conditions in this model. At the inlet boundary, pressured air enters the computational domain with the turbulent intensity 5%. The outlet boundary pressure was set as 0.101 MPa. The bristle surface was assumed to be fixed and set as wall. The upper surface and lower surface were set as symmetric to simulate a more authentic flow domain. A detailed description of these boundary condition are presented in Table 2.

2-D steady-state numerical simulations were solved by commercial code FLUENT. A hybrid discretization scheme (the default settings in FLUENT) was used until a preliminary convergence is obtained. Then the second-order discretization scheme was adopted to gain a better solution efficiency. The residual criteria for each variable was 1.0×10^{-6} , except for continuity. By monitoring values between iterations is another criterion of the FLUENT tutorial [17]. The velocity difference, between different iteration steps at Line CD was monitored (Fig. 2).

2.3. Grid meshing and independent solution

By reviewing the previous method [3], an inflation tool was used to generate four layers of quadrilateral cells in contact with bristle

Fig. 2. A schematic configuration of a brush seal using a two-dimensional tube banks model.

Table 2
Boundary conditions of the model.

Location	Type	Axial Velocity (m/s)	Circumferential Velocity (m/s)	Pressure (MPa)
Inlet	Pressure			0.201 0.231 0.261 0.291 0.321 0.351 0.101
Outlet	Pressure			
Upper Wall	Symmetry	$\frac{\partial u}{\partial y} = 0$	0	
Bottom Wall	Symmetry	$\frac{\partial u}{\partial y} = 0$	0	
Bristle	Wall	0	0

Fig. 3. An illustration of the generated mesh.

surfaces. The domains of the upstream and downstream regions meshed into quadrilateral cells. The rest of the domains were meshed into triangular cells. Part of the mesh is presented in Fig. 3. Five different meshing from coarse to fine were generated so as to investigate the grid-independency of the solution, including approximately 2.09×10^5 , 2.53×10^5 , 3.02×10^5 , 3.39×10^5 and 3.6×10^5 cells. The velocity along Line CD was used to investigate mesh independence, because velocity changes obviously at Line CD. Generally considering the mesh independence and computational time, the grid amount of 3.02×10^5 can be viewed as grid independent.

2.4. Validation of the numerical model

To validate the reliability of the computational model and numerical method, a comparison between the numerical result and experimental or analytical results should be done. The brush seal experiments by Bayley and Long are frequently cited in the literature [2,18]. Twenty-four pressure tappings were adopted, eleven of which were expected to be under the bristle tips, in the rotor surface. The rest 13 pressure tappings were designed to accommodate the predicted axial deflection of the bristles and other uncertainty. Experiments were conducted at upstream absolute pressures ranging from 107 KPa to 380 KPa, and a downstream absolute pressure of 100 KPa. The effect of rotation was not considered [19]. Cases $A_{0.1}^{10}$ and $A_{0.19}^{10}$ were used to compare with the experimental results. Fig. 4 represents the validation of the current model. The pressure distribution along centerlines at each gap was compared with the experimental results. The centerlines are described as Line EF in Fig. 2. The X^* and P^* denotes two dimensionless variables, which are expressed as

$$X^* = \frac{x - x_u}{x_d - x_u} \quad (3)$$

$$P^* = \frac{p - p_d}{p_u - p_d} \quad (4)$$

where x_u is the x coordinate of the end of the first bristle gap centerline, and x_d is the x coordinate of the beginning of the last bristle gap centerline. Parameters p_u and p_d are the pressure upstream and downstream of the brush seal, respectively. Fig. 4 presents the numerical

Fig. 4. A comparison of pressure distribution between the numerical and experimental results.

results, which reveal a trend similar to that of experimental results. Although a slight difference between the two results, their similarity confirms the validity of the model to some extents.

3. Numerical results and discussion

3.1. Velocity distribution

Fluid velocity distribution is a basic analysis for flow in a tube banks model. Fig. 5 shows the velocity distribution along Line AB and Line CD (in Fig. 2). The x-axis represents the data counts from the post-processing software. The y-axis represents the dimensionless velocity, defined as the velocity divided by the maximum. In Line AB, the velocity is almost flat but then decreases to zero when approaching the bristle surface. In addition, the velocity is larger for a high pressure differential than a low pressure differential. In Line CD, the velocity is almost symmetrical for Case $B_{10}^{0.16}$ and Case $B_{10}^{0.1}$. However, the velocity distribution is asymmetrical for Case $B_{10}^{0.22}$.

This asymmetrical distribution is deemed as the bistable phenomenon. It can be characterized by two vortices: a narrow near-wake region at the rear part of one of the cylinders, and a wide near-wake at the rear part of the other cylinder. Fig. 6 demonstrates the scheme of two bistable flow modes after two circles in-line arranged. Rather than being triggered by the arrangement of the cylinders or other external factors, these bistable characteristics are an intrinsic feature of the flow [20]. Fig. 7 presents the velocity distribution at Line CD for Case $(A, B, C)_{14}^{0.1}$, which clearly reveals the symmetrical velocity distribution

Fig. 5. Velocity distribution at line AB and CD for Case B_{10} .

Fig. 6. Schematic diagram of two bistable flow models after two circles side by side [21].

Fig. 7. Velocity distribution at Line AB for Case (A, B, C)₁₄^{0.1}.

for Case A and Case C. In addition, this distribution contained two peaks in Case C₁₄^{0.1}. This topic of bistable phenomenon requires further attention but it is beyond the scope of the present study.

The velocity inside the bristle pack is analyzed here to calculate Reynolds number in a brush seal. For each case, the velocity distribution along the centerlines of the narrowest gap is viewed and analyzed. As illustrated in Fig. 8, the geometry from left to right is for Case A, Case B and Case C, respectively. The velocities along Line AB, Line CD and Line EF were extracted for Case A. Line GH and HI for the Case B. Line JK and LM for the Case C. Velocity varies much more significantly along these centerlines than along the centerlines of the spacious gaps, which indicates velocity along centerlines of the narrowest gaps is more representative.

Fig. 9 presents the velocity distribution along Line GH and Line HI for Case B₁₀^{0.1}, B₁₀^{0.16} and B₁₀^{0.22}. Velocity distribution along Line AB, Line CD and Line EF for Case A and Line JK and Line LM for Case C are similar to Line GH and Line HI for Case B in trend, but different in amplitude. So Case A and Case C are not presented here. At each gap, velocity increases almost linearly to a maximum value and then decreases almost linearly to a minimum value. The maximum value occurs at the points X_k (k = 0...6) at each gap. These points are at the narrowest location along the line. The velocity is larger with respect to a high pressure differential. These findings on velocity distribution are similar to those by Huang et al. [2].

Fig. 8. Schematic diagrams of the centerlines for Case A, Case B and Case C, respectively.

Fig. 9. The velocity distribution at gap centerlines for Case B₁₀.

3.2. Parameters definition

3.2.1. Reynolds number

Reynolds number is a dimensionless quantity to predict the flow pattern in fluid mechanics. In the present study, Reynolds number is used as a quantity to fit the *Eu* correlation, but not to judge whether the flow is laminar or turbulent. It is expressed as the ratio of inertial forces to viscous forces and expressed as

$$Re = \frac{\rho U d}{\mu} \quad (5)$$

where U is the velocity, and d is the characteristic length. Researchers have different opinions toward velocity selection, which is an important quantity in calculating Reynolds number. Endres and Möller proposed four different reference velocities: impinging velocity, percolation velocity, gap velocity and measured gap velocity. The percolation velocity and gap velocity are calculated by proposed formulas, and the other two are obtained by experiments [22]. Žukauskas argued that the main flow velocity should be used as the reference velocity in a model with a transverse pitch ratio, which is less than 1.25. Another widely adopted method is described as

$$\begin{cases} \frac{u_{main} S_T}{S_T - d} & 2S_D \geq S_T \\ \frac{u_{main} S_T}{2(S_T - d)} & 2S_D < S_T \end{cases} \quad (6)$$

where u_{main} is the main stream velocity. Equation (6) can be derived from the continuity equation. All the velocity in the methods above is constant, which means the kinetic energy keeps constant. This is the reason for the methods mentioned above not appropriate for brush seals. The pressure differential is converted to the kinetic energy when the fluid flows from upstream area to downstream area.

The conventional method of calculating Reynolds number is modified here to calculate Reynolds number in a brush seal. A single bristle and its nearby area are an element domain. Average velocity is adopted due to the dramatic variation of velocity, so the Reynolds number for each bristle can be expressed as

$$Re_i = \frac{\rho \bar{U} d}{\mu} \quad (7)$$

where subscript i is the bristle sequence, $\bar{U}$ is the average velocity in this gap. Reynolds number for the whole brush seal can be written as follows:

$$Re = \frac{\sum_{i=1}^N Re_i}{N} \quad (8)$$

3.2.2. Euler number(Eu)

The Eu is, a dimensionless number, introduced to analyze the relationship between pressure reduction and kinetic energy per volume. It is defined as

$$Eu = \frac{\text{pressure forces}}{\text{inertial forces}} = \frac{(\text{pressure})(\text{area})}{(\text{mass})(\text{acceleration})} = \frac{(\Delta p)L^2}{0.5(\rho L^3)(U^2/L)} = \frac{2\Delta p}{\rho U^2} \quad (9)$$

where Δp is the pressure differential between two successive rows. A constant term 0.5 is multiplied in front of the inertial forces because $0.5\rho V^2$ stands for the kinetic energy, as used by many previous researchers [23–25].

Eu is the ratio of pressure to kinetic energy. In heat exchanger industry, velocity U in the kinetic energy term is a constant which can be obtained by the methods in Section 3.2.1. Pressure reduction Δp can be measured by experiments or extracted from numerical results. But, pressure and kinetic energy selection for a brush seal are different with those for a heat exchanger. Pressure is converted to kinetic energy when the fluid flows a brush seal, indicating that velocity increases and pressure decreases simultaneously. In a brush seal, pressure reduction is a known operation parameter. To express kinetic energy increase, therefore Eu becomes

$$Eu = \frac{2\Delta p}{\rho N(U_{ex}^2 - U_{in}^2)} \quad (10)$$

where U_{ex} is the maximum velocity at Line CD, and U_{in} is the velocity along Line AB (As shown in Fig. 5, the flat section of velocity distribution along Line AB is adopted). The velocity distribution at Line CD is chosen rather than at the outlet because the kinetic energy dissipates gradually in the exit domain. Also, the overall performance of a brush seal is analyzed rather than the resistance of each bristle. Therefore, the difference of the kinetic energy for the fluid domain is adopted and the gap velocity (as mentioned in Equation (7)) is not used here.

3.3. Case studies

Figs. 10–12 demonstrate the trends of Eu and leakage of Case A, Case B and Case C, respectively. Among these three cases, Eu decreases

Fig. 10. The Eu and leakage for Case A.

Fig. 11. The Eu and leakage for Case B.

Fig. 12. The Eu and leakage for Case C.

with the increase of pressure differentials, indicating that the resistance becomes weak at a high pressure differential. Eu becomes larger if a brush seal contained more rows which indicates that adding rows to a brush seal can strengthen its resistance. Another common observation among three cases has been found is that Eu decline less gradually at a higher pressure differential than at a lower pressure differential, which can be proved by slopes. Tables 3–5 presents the slope between two successive points for Case A, Case B and Case C, respectively. There is an increase in slopes from a small pressure to a large pressure. All three cases present the same trend and this indicates that the resistant capability becomes gradually weak and approaches a fixed value with the increase of pressure differential, suggesting that the resistance of a brush seal is limited. If a brush seal is subjected to a higher pressure differential, much more pressure energy will convert into kinetic energy, suggesting that fluid flee away at a high velocity. An approximate Eu value should be chosen based on experience.

The effect of geometric arrangement on Eu is demonstrated in Table 6. Sign $>$ in Table 6 denotes the resistance is stronger than the

Table 3

The slope between two successive points for Case A.

Pressure Range (MPa)	0.1–0.13	0.13–0.16	0.16–0.19	0.19–0.22	0.22–0.25
Row number					
10Rows	–92	–58.767	–42.3	–26.933	–17.466
14Rows	–145.67	–101.13	–51.37	–34.07	–25
18Rows	–179.2	–115.33	–67.8	–48.8	–33.97

Table 4

The slope between two successive points for Case B.

Pressure Range (MPa)	0.1–0.13	0.13–0.16	0.16–0.19	0.19–0.22	0.22–0.25
Row number					
10Rows	–70.053	–60.686	–55.488	–32.55	–18.431
14Rows	–71.507	–52.897	–56.183	–39.179	–37.466
18Rows	–84.19	–60.116	–53.931	–43.58	–36.82

Table 5

The slope between two successive points for Case C.

Pressure Range (MPa)	0.1–0.13	0.13–0.16	0.16–0.19	0.19–0.22	0.22–0.25
Row number					
10Rows	–89.245	–50.840	–33.564	–24.778	–18.535
14Rows	–144.673	–76.686	–51.874	–32.853	–25.05
18Rows	–193.325	–109.901	–65.756	–45.917	–32.239

other case. As for a 10-row brush seal, Case B is the most resistant among three cases. For a 14-row brush seal, Case B has the largest resistance at the whole pressure differentials except for 0.1 MPa. For a 18-row brush seal, Case B is the most resistant among three cases at all the pressure differentials except for 0.1 MPa and 0.13 MPa. Generally, what can be concluded is that Case B is the most resistant. By comparing the Eu of Case A and Case C in Figs. 10–12, the discrepancy between Case A and Case C is far less than that between Case B and Case A or Case B and Case C in most cases, indicating that Case A and Case C present similar resistance.

It's hard to define a geometric parameter (S_T and S_L) which is most influential on Eu . Case A has the shortest S_T but the longest S_L . Case C has the shortest S_L but the longest S_T . The difference of Eu between Case A and Case C is not obvious. S_T and S_L of Case B are the compromise between Case A and Case C. And Case B is the most resistant among the three cases. Rather than find which geometric parameter is the most influential on Eu , the present paper analyzes the correlations between Eu and geometric parameters, which are presented in Section 4.

Leakage is an important criterion for judging a seal. For all cases, leakage increases almost linearly with the increase of pressure differential. Leakage of Case A is far less than that of Case B and Case C. For Case $A_{10}^{0.25}$, leakage is about 2.5×10^{-3} kg/s. For Case $B_{10}^{0.25}$ and $C_{10}^{0.25}$, leakage are approximately 6.0×10^{-3} kg/s and 6.4×10^{-3} kg/s, respectively. Case B has the largest resistance while Case A has the least leakage among three cases. The different trend between leakage and aerodynamic resistance indicates that leakage is not fully equivalent to resistance. Resistance is used to express the energy loss in a brush seal while leakage is obtained at the outlet. For a brush seal, the ideal conditions are low leakage and high energy loss.

4. Multiple regression

Based on the numerical results above, multiple regressions between

Table 6The comparison of Eu among three cases at different rows and pressure differentials.

Pressure Differential (MPa)	0.1	0.13	0.16	0.19	0.22	0.25
Row number						
10	B > C > A	B > C > A	B > C > A	B > C > A	B > C > A	B > C > A
14	A > B > C	B > A > C	B > A > C	B > A > C	B > A > C	B > A > C
18	A > C > B	A > B > C	B > A > C	B > A > C	B > A > C	B > A > C

Eu and geometric parameters are obtained. A common correlation for Eu is expressed as follows [26].

$$Eu = f\left(Re, \frac{S_T}{d}, \frac{S_L}{d}, N\right) \quad (11)$$

Žukauskas proposed that Eu across a tube banks model is proportional to the number of rows [27]. For practical calculations, the exponential correlation form is assumed

$$Eu1 = C_1 Re^{C_2} \left(\frac{S_T}{d}\right)^{C_3} \left(\frac{S_L}{d}\right)^{C_4} N \quad (12)$$

This paper proposes the correlation with a minor modification of the work by Žukauskas

$$Eu2 = C_1 Re^{C_2} \left(\frac{S_T}{d}\right)^{C_3} \left(\frac{S_L}{d}\right)^{C_4} N^{C_5} \quad (13)$$

where coefficients C_1, C_2, C_3, C_4, C_5 are determined by the method of multiple regression. The final correlations are as follows

$$Eu1 = 27.648 Re^{-0.728} \left(\frac{S_T}{d}\right)^{0.136} \left(\frac{S_L}{d}\right)^{0.29} N \quad (14)$$

$$Eu2 = 2429.923 Re^{-0.981} \left(\frac{S_T}{d}\right)^{-0.065} \left(\frac{S_L}{d}\right)^{0.094} N^{-0.165} \quad (15)$$

The residual square (R^2) can quantify how well a correlation fits the data. However, an intrinsic drawback of this method is that R^2 will always increase if a new predictor is added. The adjusted residual-square (R_{adj}^2) can be used to determine how well terms fit the data. The R_{adj}^2 is expressed as follows

$$R_{adj}^2 = 1 - \left[\frac{(1 - R^2)(n - 1)}{n - k - 1} \right] \quad (16)$$

where n is the number of points in a series of data, k is the number of independent regressors. The R_{adj}^2 is 0.985 for $Eu2$ (Equation (15)) and 0.654 for $Eu1$ (Equation (14)). Therefore, $Eu2$ fits the data better than $Eu1$.

5. Conclusion

The present study analyzed the resistance in the tube banks model of brush seals by adopting Eu , which has been widely used in the heat exchanger industry. Three different geometric arrangements containing different rows subjected to six pressure differentials were used as the objects. The Reynolds number and Euler number equations were updated according to brush seal working conditions. The present study demonstrates five important findings:

1. Velocity was a constant in the upstream region along Line AB. Velocity distribution along Line CD was symmetric for part of cases and asymmetric for the other cases. At each gap, velocity increased linearly from the beginning of the gap to the middle and then decreased linearly to the end of the gap.
2. A Reynolds number calculation was proposed based on velocity distribution along gap centerlines, which was applicable for a brush seal subjected to the pressure differential. A conventional Eu was modified to analyze its intrinsic aerodynamic resistance of a brush

seal.

3. Eu decreased less gradually with the increase of pressure differentials, indicating that there may exist a minimum Eu value for each case. Case B had the largest Eu among three cases.
4. A high resistance was not equivalent to low leakage. The Eu was interpreted by energy loss while leakage is a quantity measured at the outlet.
5. Two different fitted correlations were achieved between Eu and geometric parameters. The correlation that was exponential to the numbers of rows fit better than that which is proportional to the numbers of rows.

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Nomenclature

N	row number
p	pressure (Pa)
ρ	density of air (Kg/m^3)
S_D	diagonal pitch (mm)
S_L	longitudinal pitch (mm)
S_T	transverse pitch (mm)
d	diameter of bristle (mm)
θ	angle ($^\circ$)
δ	gap between two bristle (mm)
u_i, u_j	velocity component in the i and j direction (mm)
u_{main}	main stream velocity (mm/s)
U	velocity (mm/s)
$\bar{U}$	average velocity (mm/s)
u_{ex}	exit velocity (mm/s)
u_{in}	inlet velocity (mm/s)
Re_i	located Reynolds number
Re	Reynolds number
Eu	Euler number
$C_i (i = 1, 2, 3, 4, 5)$	coefficients
R_{adj}^2	adjusted residual square

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