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Comparative analysis of the influence of labyrinth seal configuration on leakage behavior †

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Abstract

This study analyzed the influence of configuration and clearance on the leakage behavior of labyrinth seals. Both computational fluid dynamics (CFD) and an analytical tool were used to predict the leakage flow of two different (straight and stepped) seal configurations with various clearances. The predicted results were compared with experimental data. The CFD gives a better agreement with the experimental result than the analytical model on average. In the straight seal, the dependence of the discharge coefficient on the clearance is considerable, while it is much smaller in the stepped seal. The CFD captures the entire behavior sufficiently well, but the analytical model overpredicts the clearance dependence in the stepped seal. The CFD also predicts well the influence of the flow direction on the leakage flow. The advantage of the stepped seal over the straight seal becomes more evident as the clearance gets larger. As the clearance becomes sufficiently small, the advantage of the stepped seal reduces.

***Keywords*:** Labyrinth seal; Straight; Stepped; Pressure ratio; Leakage; Number of teeth; Computational fluid dynamics

1. Introduction

Minimizing unwanted leakage between stationary and rotating parts is very important in achieving high performance of rotating machines such as gas and steam turbines. Labyrinth seals remain popular despite the recent development of several advanced sealing techniques. Their main advantages include structural simplicity, reliability, high temperature resistance, a wide operating range in terms of pressure ratio, and so on. They are widely used in various local components of gas and steam turbines, particularly in the compressor and turbine blade tip areas and the secondary air system. A labyrinth seal is a non-contacting sealing device that consists of a series of cavities connected by small clearances. The flow loses its total pressure while it sequentially experiences acceleration into the clearance due to contraction, friction through the clearance, and dissipation of kinetic energy at the cavity. This process repeats until the flow exits the final cavity.

Recent rapid improvements on the efficiency and power output of gas turbines require enhanced design of every flow component inside the engine. As the performance improvement becomes marginal, minimization of leakage flows becomes more important. Therefore, labyrinth seals are used more intensively, their clearances are more tightly designed and controlled than before, and their configurations are evolving continuously. Therefore, the requirement for an accurate leakage prediction is becoming crucial.

There are two basic types of labyrinth seal configurations: straight and stepped. The main purpose of using a stepped seal is to provide additional flow resistance. Major operating parameters are the pressure difference (or ratio) between up and down streams of a

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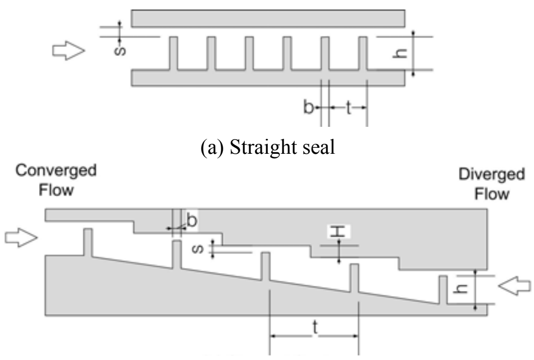
Table 1. Labyrinth seal geometric variables.

Seal type	N	s [mm]	t [mm]	b [mm]	h [mm]	H [mm]
Straight	6	0.5-2.5	12	2.5	10.5	-
Stepped	5	0.4-2.0	28	2.5	9.0	3.8



(b) Stepped seal

Fig. 1. Labyrinth seal geometry.

seal and the size of the clearance that is formed between the seal teeth of one side and the land of the other side. Based on a maximum allowable leakage flow, seal designers decide the seal type, running clearance, and the number of teeth, taking into account other design constraints such as the maximum space for seal installation. Thus, seal design is an optimization process that requires a compromise with designs of other components.

Currently, gas turbine manufacturers use analytic leakage prediction tools. Predicting leakage flow in labyrinth seals dates back to Vermes [1] in the early 1960s. Since then, other efforts have been made to accumulate test data [2-4] and to construct performance prediction tools based on experimental data [4-,6]. Despite those efforts, simple analytic models to satisfactorily predict performances of wide variants of seal configurations remain difficult to set up. Thus, the rig test is still recommended for accurately predicting seal leakage flow. Testing seal performance under real engine conditions is one of the recent research trends [7]. Computational fluid dynamics (CFD) is also increasingly used in analyzing flows inside labyrinth seals [8-12]. While numerical analysis sometimes does not give very accurate predictions of overall aerodynamic performance, it at least shows local flow fields and relative performance comparisons between different configurations.

In this work, the aerodynamic performance of two typical labyrinth seal types (straight and stepped) were compared. Numerical analysis was used to predict the flow and leakage performance of the seals. Predicted leakage behaviors according to clearance size and pressure ratio were compared with experimental data, and leakage performances of the two seal types were compared. The effect of the number of teeth was also analyzed. Leakage was predicted using a typical analytic design tool and results were compared with both the experimental data and numerical analysis results. Essentially, this paper presents a systematic performance comparison between straight and stepped seals and demonstrates the usefulness of numerical analysis.

2. Labyrinth seal

2.1 Geometry

For this study, various previous experimental works were reviewed, and the experimental work of Wittig et al. [8] was selected for a case study. They conducted an experiment for both the straight and stepped seal geometries with similar geometric parameters. They presented results for various clearance sizes. In real gas turbines, one of the two sides that form the seal space rotates. It was found that the effect of rotation is important only when the rotating speed is very high (more exactly, when the ratio between the circumferential speed of the seal arm and the flow speed is very large) [13]. Therefore, static rigs were preferred in most experiments. Recently, test data in a rotating environment have been reported [7]. Without the effect of the rotation, a 2-D rig is expected to provide nearly the same results as an axisymmetric 3-D rig does [3]. Therefore, the static rig case used in this work provided sufficient insight into the aerodynamic performance comparison between the straight and stepped seals.

Fig. 1 shows the two labyrinth seal geometries and Table 1 summarizes their dimensions. The labyrinth seal test section consists of an upper part, either straight or stepped, and a lower part with teeth. In a real engine, the upper and lower parts correspond to the stationary and rotating parts, respectively. The straight seals have six teeth and the stepped have five. The two seals have almost similar teeth dimensions. The pitch (t) of the stepped seal is larger than that of the straight seal. This is quite reasonable because axial movement in the real gas turbine may cause contact between the teeth and the step face. This is a disadvantage of the stepped seal along with a larger overall seal height. Hence, the overall size of the stepped seal is larger than that of the straight seal. The lateral (depthwise in the figure) dimension was sufficiently large to ensure two-dimensionality. Gaps (clearance, *s*) were adjusted using spacers. For the stepped seal, two flow directions were tested. The flow from the left is the converged flow and the other is the diverged flow. Details of the test can be found in the literature [8].

2.2 Performance parameters

The performance of a seal can be described by the relation between the pressure ratio and a flow parameter. The most common flow parameter is the following flow function:

$$\frac{\dot{m}\sqrt{T_{o,in}}}{A_c P_{o,in}} \tag{1}$$

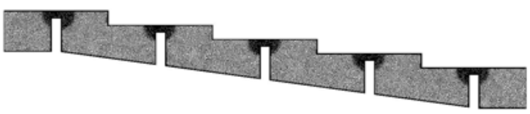
Seal performance is also described by the discharge coefficient defined as follows:

$$C_D = \frac{\dot{m}}{\dot{m}_{id}} \tag{2}$$

The ideal mass flow is calculated by

$$\dot{m}_{id} = Q_{id} \cdot \frac{P_{o,in} A_c}{\sqrt{T_{o,in}}} \tag{3}$$

where



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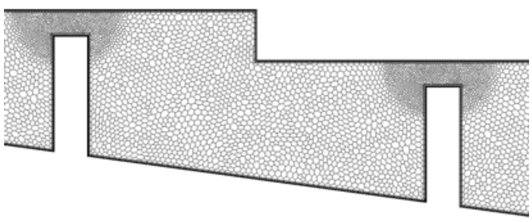


(b) Enlarged view

Fig. 2. An example of the generated meshes for the case of the stepped seal.

$$Q_{id} = \sqrt{\frac{2k}{R(k-1)}\left[\left(\frac{P_{out}}{P_{o,in}}\right)^{\frac{2}{k}} - \left(\frac{P_{out}}{P_{o,in}}\right)^{\frac{k+1}{k}}\right]} \tag{4}$$

The inlet properties are total values measured at the upstream settling chamber. The pressure ratio is defined as the ratio of the inlet total pressure to the exit static pressure ($P_{o,in}/P_{out}$).

3. Analysis

3.1 Numerical analysis

An advantage of using CFD is its capability to analyze a large number of design configurations and parameters in a relatively short period of time. Therefore, with the development of commercial codes, the use of CFD analysis has been increasing rapidly in recent years as suggested in the references [8-12]. Either two- or three-dimensional analysis has been adopted. Since the test case adopted in this work is two-dimensional and a number of different operating conditions were studied, two-dimensional analysis was used in this study.

A commercial finite volume code, STAR-CCM+ [14], was used. It was assumed that air was an ideal gas and the flow was steady and adiabatic. A realizable two-layer k-ε turbulent model was used. This model combines the realizable k-ε turbulent model with the two-layer approach. The realizable k-ε turbulent model uses equivalent kinetic energy and dissipation rate equations, but has additional flexibility of all y+ wall treatments. The two layer approach is designed to give results similar to the low y+ treatment as y+ approaches zero (viscous sublayer) and to the high y+ treatment for y+>30 (wall function layer). It gives reasonable results for intermediate meshes where the cell falls in the buffer layer. Polyhedral mesh elements were used to create unstructured meshes in the entire domain. Fig. 2 shows and an example of generated computational grids. The grid density in the clearance area was refined to locate sufficiently large number of meshes. Prism layers near the solid surface wall were set up such that maximum y+ is kept within the upper limit of the wall function treatment.

For a given geometry, the exit pressure and mass flow rate were given, and the corresponding inlet total pressure was obtained thru calculation. Since the exact inlet and exit conditions of the test seals were not given, the exit pressure was set at ambient pressure and ambient temperature was given at the inlet. Arbitrary setting of those boundary conditions did not matter significantly because calculation results would be presented in terms of dimensionless parameters (C_D and pressure ratio).

Grid dependence was checked to produce sufficiently converged solutions according to mesh size. Fig. 3 shows an example of grid dependence in the case of the straight seal, presenting a variation of pressure ratio with the number of meshes for a given mass flow rate. The selected number of meshes of this specific case is around 20,000, which gives an almost converged solution. The number of meshes ranges from 13,000 to 20,000 for the straight seal and from 19,000 to 25,000 for the stepped seal, depending on the clearance size.

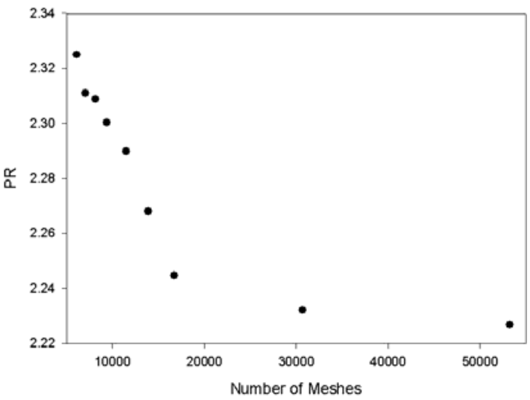


Fig. 3. An example of grid dependence of the CFD result.

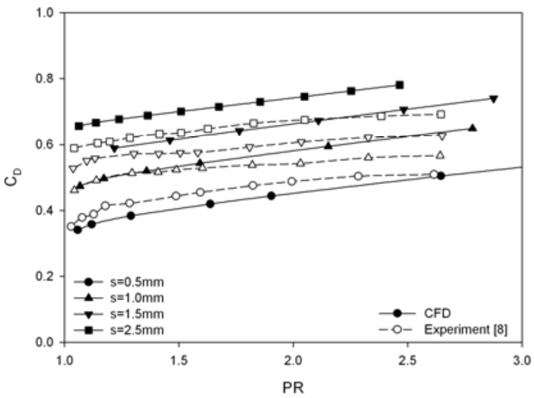


Fig. 4. Variation of discharge coefficient with pressure ratio and clearance for the straight seal.

3.2 Analytic method

There are several analytical tools to predict the leakage flow of labyrinth seals. Various models can be classified into two categories: global and knife-to-knife (teeth-to-teeth) models. The model of Vermes [1] is a typical global analysis tool, which does not model teeth-by-teeth phenomenon but calculates overall performance using semi-empirical models. Tipton et al. [5] summarized the characteristics of various available models and suggested their own knife-to-knife model. The model is coded as a computational program [15]. The model dealt with three loss mechanisms inside a seal separately: contraction, venturi and friction, and partial or full expansion. The total pressure loss for a flow passing through a tooth is calculated as follows by summing up the three loss components.

$$\Delta P_{o,in} = \Delta P_{o,c} + \Delta P_{o,vf} + \Delta P_{o,e} \tag{5}$$

$$\Delta P_{o,c} = K_c \cdot \frac{k}{2} \cdot P \cdot M^2$$

$$\Delta P_{o,vf} = K_{vf} \cdot \frac{k}{2} \cdot P \cdot M^2 \tag{6}$$

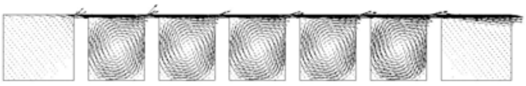
$$\Delta P_{o,e} = K_e \cdot (P_o - P)$$

The loss coefficients are functions of various geometric and flow parameters. Tipton et al. established the model based on a vast amount of available solid seal data. The data are diverse in terms of seal configuration (straight and stepped), clearance size, number of teeth, and flow direction for stepped seals (converged and diverged). The performance of each tooth (or cavity) is stacked to produce overall seal performance of a multi-cavity seal. More details of the model can be found in the above-mentioned references [5, 15].

4. Results and discussion

Fig. 4 shows the discharge coefficients calculated from the numerical analysis for the straight seal. The experimental results were also shown for comparison. For all clearance cases, the discharge coefficient increases almost linearly as the pressure ratio increases. The calculated results show strong dependence of the discharge coefficient on the clearance size. As the clearance gets larger, the discharge coefficient increases, which was also observed in the experiment. For example, the absolute leakage flow of a clearance of 2.5 mm is more than five times (gap area ratio) that of a clearance of 0.5 mm. Even though there are some quantitative discrepancies between the experimental data and the calculations, overall agreement, especially regarding dependence on the clearance size, is good.

Figs. 5 and 6 show the velocity fields for the smallest and largest clearance cases with comparable overall pressure ratios around 1.5. For a small gap, the flow fields of all cavities are very similar. However, for a big gap, the larger mass flow creates a very strong vortex in the first cavity, thus producing a large pressure drop there. Even a sensible recirculation in the gap area is predicted owing to the sudden contraction of the high velocity flow. As a result, the pressure distribution is nearly uniform (evenly divided pressure drops) along the cavities in a sufficiently small clearance case, while the pressure drop is relatively greater in the first cavity in a sufficiently large clearance case.



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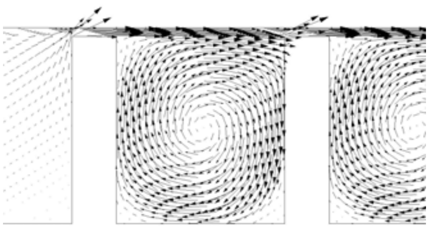
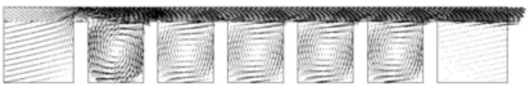


(b) Local view around the first cavity

Fig. 5. Velocity field for the straight seal with a clearance of 0.5 mm (PR=1.5).



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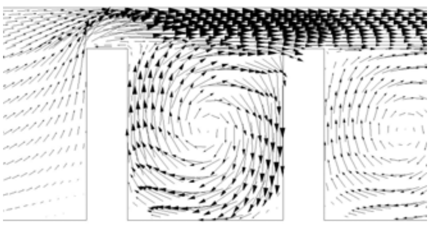


(b) Local view around the first cavity

Fig. 6. Velocity field for the straight seal with a clearance of 2.5 mm (PR=1.5).

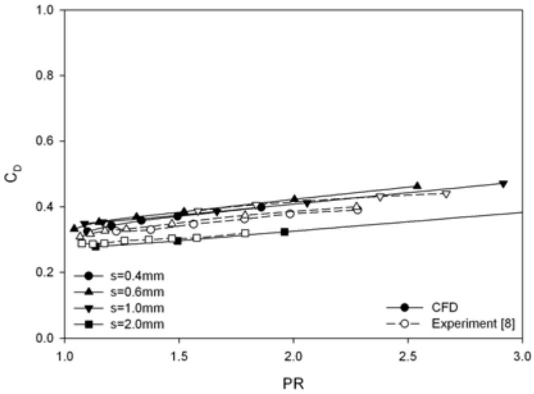
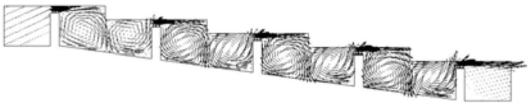


Fig. 7. Variation of discharge coefficient with pressure ratio and clearance for the stepped seal with a converged flow arrangement.



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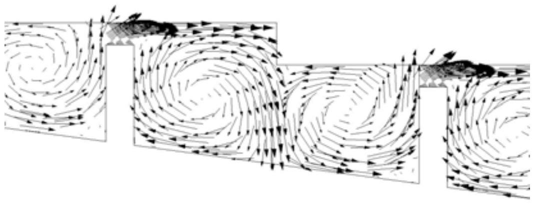

(b) Local view

Fig. 8. Velocity field for the converged stepped seal with a clearance of 2.0 mm (PR=1.5).

Fig. 7 shows the discharge coefficients for the stepped seal with a converged flow direction. In this case, the agreement between the CFD and the experiment is even better than that of the straight seal case. Both the trend and the absolute variation of the discharge coefficient with varying clearance sizes were adequately predicted by the CFD. The average discharge coefficient of the stepped seal is smaller than that of the straight seal, indicating that the leakage performance of the stepped seal is better. Also, the clearance dependence of the discharge coefficient is much smaller for the stepped seal. Fig. 8 shows the velocity field of the stepped seal with a converged flow direction. For the stepped seal, the flow through the gap hits the step wall, thus dissipating more dynamic energy inside the cavity than the straight seal. Two counter rotating flows occur inside a cavity. This complicated flow structure, which is caused by the step, incurs a larger pressure drop for a given mass flow. In other words, if the overall pressure ratio is fixed, there is less leakage flow in the stepped seal. Similar flow fields are predicted in other clearances

Fig. 9 shows the discharge coefficients for the stepped seal with a reversed (diverged) flow direction. The clearance dependence is also relatively weak. On average, however, the discharge coefficient is slightly larger than that of the converged flow. The agreement between the CFD and the experiment is good. Fig. 10 and 11 show the velocity fields. With a small gap, the flow from the gap cannot head directly towards the next gap, and two circulations occur in the right and left parts of the cavity. In a wide gap, the main flow passing through the gap directly flows to the next gap, separating the cavity into the upper and lower circulation regions.

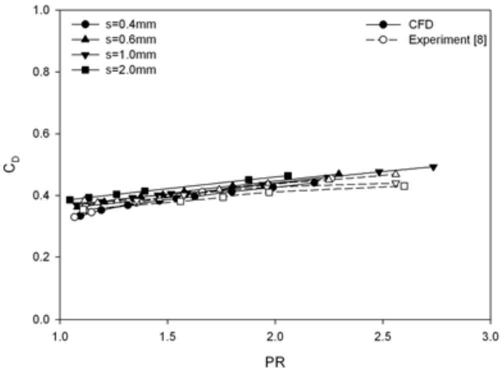

Fig. 9. Variation of discharge coefficient with pressure ratio and clearance for the stepped seal with a diverged flow arrangement.

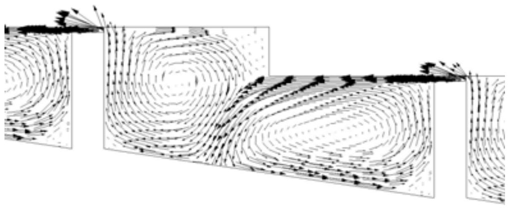

Fig. 10. Velocity field for the diverged stepped seal with a clearance of 0.4 mm (PR=1.5).

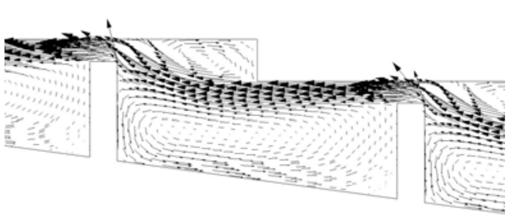

Fig. 11. Velocity field for the diverged stepped seal with a clearance of 2.0 mm (PR=1.5).

Now, the results of the analytical KTK model are discussed. Figs. 12 to 14 show the variation of the discharge coefficient with pressure ratio for the three configurations. In the straight seal, the model also predicts that the discharge coefficient increases with an increasing clearance, but the variation is slightly smaller than that of the experiment. In the stepped seal, the model predicts a considerably greater dependence of the discharge coefficient on the clearance. The model especially predicts that the converged seal allows a slightly larger leakage in comparison to the diverged seal. However, a reverse trend was observed both in the experiment and the CFD, as shown in Figs. 7 and 9. The experimental and CFD results seem to be more physically feasible because the flow hits the step wall in the converged flow arrangement, which may cause a rather large flow resistance. Thus, CFD, on the whole, gives a more accurate qualitative and quantitative leakage prediction in comparison with

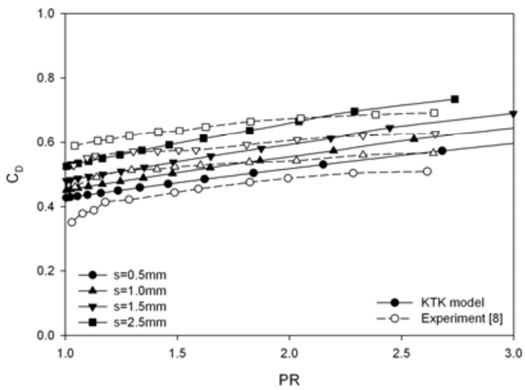

Fig. 12. Comparison between the analytical model and the experiment for the straight seal.

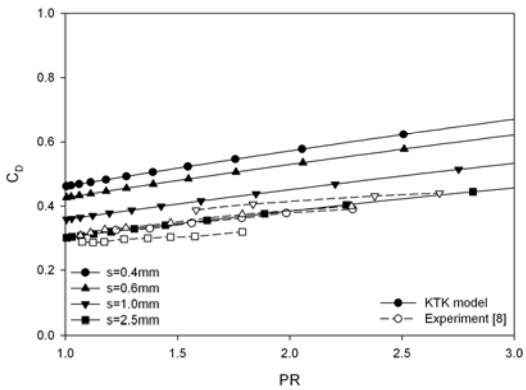

Fig. 13. Comparison between the analytical model and the experiment for the stepped seal with a converged flow arrangement.

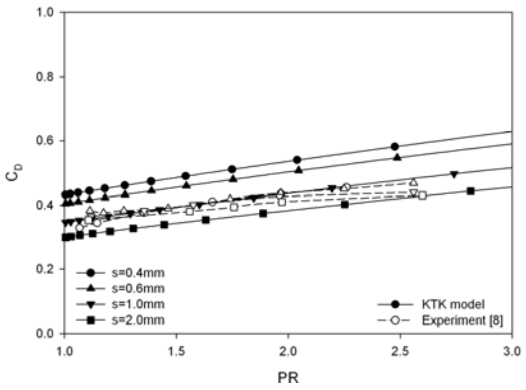

Fig. 14. Comparison between the analytical model and the experiment for the stepped seal with a diverged flow arrangement.

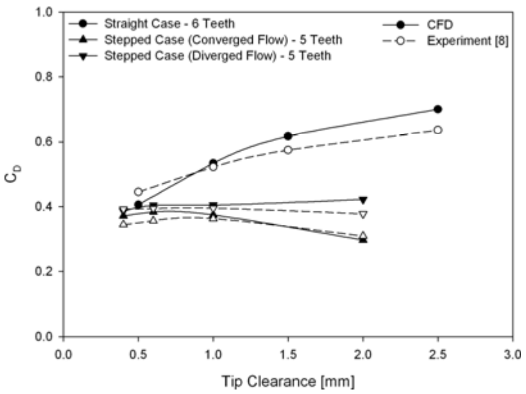

Fig. 15. Dependence of discharge coefficient on clearance.

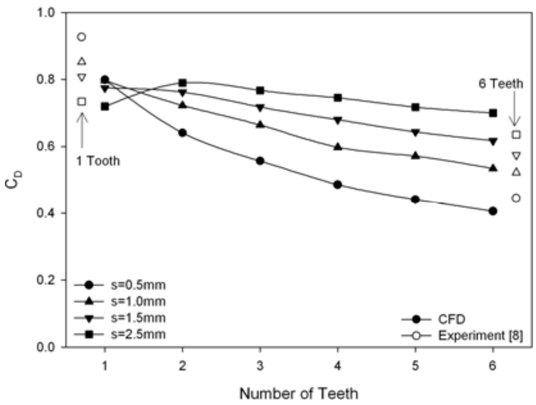

Fig. 16. Influence of the number of teeth on the discharge coefficient of the straight seal.

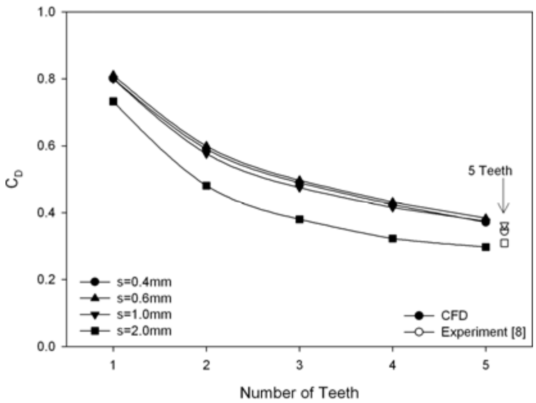

Fig. 17. Influence of the number of teeth on the discharge coefficient of the converged stepped seal.

the analytical model especially for the stepped seal.

Fig. 15 shows the dependence of discharge coefficient on clearance size for a pressure ratio of 1.5. Agreements between the experiment and the CFD are acceptable as already discussed above. From this Fig. it is clear that in the straight seal, the discharge coefficient decreases as the clearance decreases. In the stepped seal, however, a trend is not evident. When the clearance is sufficiently large, the discharge coefficient is either very insensitive to the clearance or increases slightly with a decreasing clearance in contrast to the stepped case. If the clearance is reduced sufficiently, the discharge coefficient hardly changes or decreases slightly as the clearance decreases. These trends are predicted by the CFD, but they are not captured well by the analytical model. The result of the analytical model is not shown owing to the complexity of the figure, but the agreement with experimental data is, on average, not as good as the agreement between the CFD results and experimental data. It is noted that the usefulness of the stepped seal lessens as the running clearance gets smaller. When the clearance is sufficiently large, the stepped seal allows a considerably reduced leakage. However, if the clearance is very small, the difference in the leakages between the straight and the stepped seal diminishes.

The variation of the leakage performance with the number of teeth was predicted by the CFD. Figs. 16 and 17 present the results for a pressure ratio of 1.5. Also shown are the available experimental results. The leakage generally increases as the number of teeth decreases. The clearance dependence is also generally maintained even though the number of teeth changes. However, in the straight seal, clearance dependence decreases as the number of teeth decrease; finally this is reversed for a single tooth case. The reversed dependence was also observed in the experiment and the analytical KTK model.

5. Conclusions

This work investigated the influence of basic design parameters of labyrinth seals on their leakage behaviors. The following results were obtained.

(1) The CFD predicted sufficiently well the leakage flows of different seal configurations and flow arrangements for various clearances. In the straight seal, the discharge coefficient decreases as the clearance decreases. This trend was adequately predicted by the CFD. In both the converged and diverged flows of the stepped seal, the clearance dependence was much smaller than that of the straight seal. The CFD results were also sufficiently accurate both qualitatively and quantitatively. The difference in the leakage behavior according to the flow direction in the stepped seal was well predicted by the CFD.

(2) The analytical KTK model provided acceptable prediction in the case of straight seal, but did not provide satisfactory predictions for the stepped seal. In particular, the model overpredicted the dependence of the discharge coefficient on the clearance. A partial reason for this result seems to be the fact that the range of the geometric variables of the correlations adopted in the model might did not fully cover those of the seals considered in this study. Therefore, a continuous revision of the model to account for a wider range of design variables is required.

(3) The trends of the variation of the discharge coefficient with the clearance size were quite different between the straight and the stepped seals. The advantage of the stepped seal is obvious when the clearance is large; the advantage diminishes when the clearance becomes very small. Therefore, since the marginal performance gain over the straight seal can hardly justify the increased fabrication cost, the application of the stepped seal does not seem very feasible in instances where the running clearance needs to be very small. Instead, a straight seal with more teeth (the straight seal can accommodate more teeth than the stepped seal for a given axial length) seems to be a suitable design alternative.

Nomenclature

A_c	:	Clearance area [m^2]
b	:	Teeth width [mm]
C_D	:	Discharge coefficient
H	:	Step height [mm]
h	:	Teeth height [mm]
K	:	Loss coefficient
k	:	Specific heat ratio
M	:	Mach number
$\dot{m}$	:	Mass flow rate [kg/s]
$\dot{m}_{id}$	:	Ideal mass flow rate [kg/s]
N	:	Number of teeth
P_o	:	Total pressure [kPa]
P	:	Static pressure [kPa]
PR	:	Pressure ratio, $P_{o,in}/P_{out}$
Q_{id}	:	Ideal flow function [kgK$^{0.5}$/kNs]
R	:	Gas constant [kJ/kgK]
s	:	Clearance [mm]
T_o	:	Inlet total temperature [K]
t	:	Pitch [mm]

Subscript

c	:	Contraction
e	:	Expansion
in	:	Inlet
out	:	Outlet
vf	:	Venturi and friction

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