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Tridimensional analysis of a Turbulent Flow through an Eccentric Short Labyrinth Seal

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ABSTRACT: Labyrinth seals are widely used to limit leakage flow between rotating and stationary parts of turbo machines. However, these elements often generate driving forces that may increase the unstable vibration of the rotor. Thus, an accurate prediction of the static and dynamic behavior of labyrinth seals is more required to improve turbomachines performance and design. In this paper, a numerical model based on CFD computation has been developed to predict the flow characteristics through an eccentric short labyrinth seal with four teeth fixed on the rotor. The realizable k-ε and k-ω SST turbulent models have been separately used in this computational model to compare predictions to experiments for the complex turbulent flow field within the seal. The pressure distribution around the seal is calculated in each cavity and the obtained results show that the k-ω SST turbulent model predictions are better than those of the realizable k-ε model. The Pressure contours and its distribution along the seal are also presented. Additionally, a parametric study of the circumferential velocity distribution assessed the use of bi-dimensional models to predict rotor dynamic characteristics of this kind of seals. Furthermore, influences of pressure ratio and inlet swirl on the leakage flow through the seal have been studied in this paper.

KEYWORDS: Labyrinth seal, Rotor dynamic, leakage, CFD, eccentricity, inlet swirl, whirl frequency.

1 INTRODUCTION

Labyrinth seals are mechanical devices generally integrated in rotor-stator clearances to minimize secondary flows in turbomachines including gas turbines, turbo pumps and compressors. Figure 1 illustrates three different sites of labyrinth seals (shaft seal, eye seal and balance drum seal) in a centrifugal compressor. The complex working flow passing through labyrinth seals has been the subject of many worldwide studies in the last three decades, but it needs additional investigations to be more understood and accurately modelled.

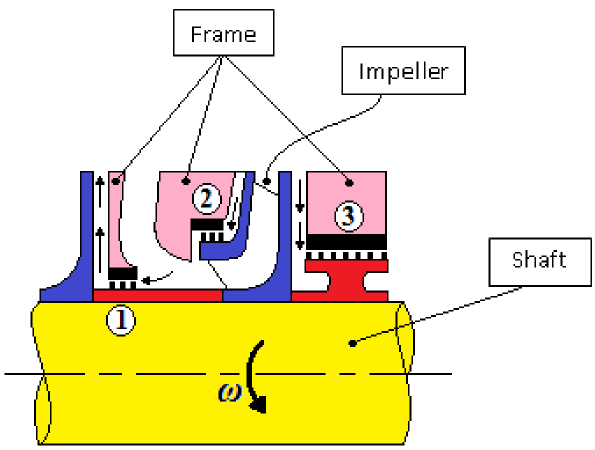

Fig. 1. Labyrinth seal sites in a compressor

The current trend in turbo machinery design requires high power and more compact machines with higher efficiencies to satisfy accentuated demand for higher rotor speeds, higher pressures and tighter clearances. This goal can't be achieved without more accurate determination of internal leakage flow and rotor dynamic forces for this kind of machines. These characteristics are generally predicted using two known approaches to solve the Navier-Stokes equations in these seals. The first way is developed codes based on the bulk flow theory and the second way is methods using computational fluid dynamics "CFD" [1, 2].

The bulk flow models were developed in the early 1980s, and these models continue to be used in the industry [3]. Several authors have developed bulk-flow approaches to predict dynamic characteristics of labyrinth seals including Iwatsubo [4], Childs and Scharrer [5, 6] and Kirk [7]. Due to the complex geometry of labyrinth seals, the bulk-flow method needs to simplify the physical problem models and governing equations as well to obtain approximate and quick results. Therefore, this approach yields good results for plain annular seals but poor predictions when recirculation is present in the flow field [8]. To improve the bulk flow model, multiple control volume techniques have been used that divide the geometry of the seal [9]. These techniques associated dominant flow behavior into different control volumes which are then linked by appropriate boundary conditions. However, a priori knowledge of the flow required parameters is not always known and the interface conditions change for different seal operating conditions. Furthermore, these models require some empirical relationships such as Hirs and Moody friction factor relationships to quantify shear stress in a developed turbulent pipe flow [10, 11]. These empirical coefficients have been the subject of many investigations [12, 13] to formulate an accurate model for these coefficients. However, it is difficult to capture the full nature of the friction factor through bulk flow models without an experimental measurement for each seal [14].

Unlike bulk flow method, computational fluid dynamic "CFD" does not rely on empirical wall and interface constants that may change for varying applications and geometry. In addition to this, the exact geometry of the seal may be modelled allowing optimization of the teeth profile. However, the obvious drawback of CFD compared to bulk-flow is increased computational requirement. The present research attempts to calculate the leakage flow and rotor dynamic forces through an eccentric short labyrinth gas seal based on three-dimensional CFD techniques solving the general Reynolds Averaged Navier-Stokes equations along with appropriate turbulence model.

2 GEOMETRY AND CFD MODEL OF THE SEAL

2.1 SEAL GEOMETRY

The labyrinth seal object of this study is supposed to be not axisymmetric. The eccentricity ratio denoted ε is defined as the ratio of the seal eccentricity (distance between rotor and stator centers) to the seal radial clearance. The seal has four teeth fixed on the rotor lateral surface. These teeth are represented by cavities in the computational domain. The seal working fluid is air. The 2-D seal geometry is shown in figure 2 and a cut section of the 3-D fluid computational domain is shown in figure 3.

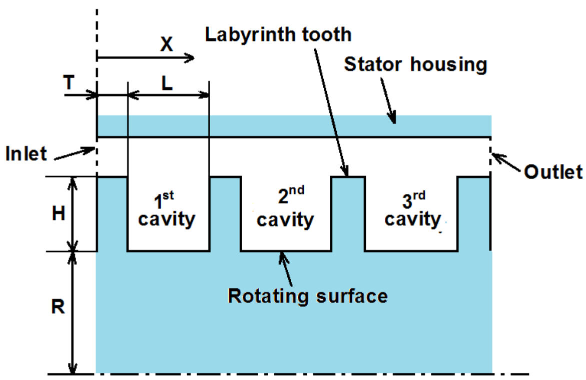

Fig. 2. Labyrinth seal geometry

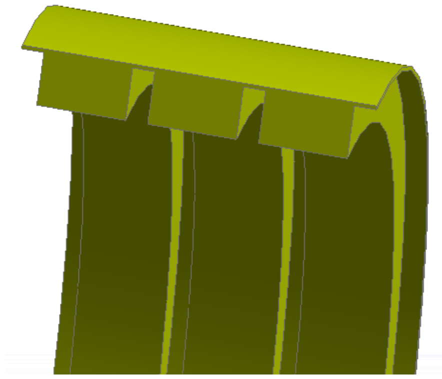

Fig. 3. A cut section of the 3-D short labyrinth seal model

The geometrical dimensions and operating conditions of the seal are shown in table 1. The negative signs of the rotating speed and the inlet swirl velocity indicate that the rotor turns in the clockwise direction as per the angle sign convention shown in figure 4. A positive radial force is a centering force while a negative radial force is a decentering one. A positive tangential force in a forward whirl force while a negative tangential force is a backward whirl force.

Table 1. Calculated conditions of the seal

Number of labyrinth cavities	3
Tooth width, L	12.7 mm
Tooth thickness, T	3.18 mm
Tooth height, H	7.94 mm
Mean clearance width, C	0.949 mm
Eccentricity ratio, ε	43 %
Rotor radius, R	93.66 mm
Rotating speed, ω	-2025 rpm
Inlet pressure, Pin	1.1077 MPa
Outlet pressure, Pout	1.033 MPa
Inlet swirl velocity, Win	-49.8 m/s

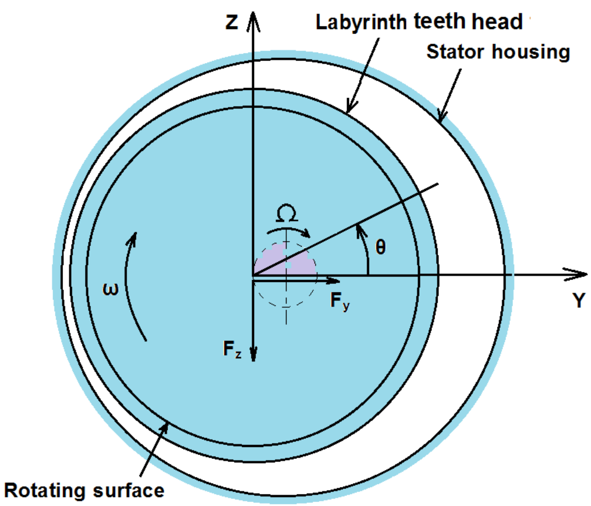

Fig. 4. Peripheral angle and fluid forces sign convention in the seal

2.2 MESHING

For the given seal geometry, an appropriate mesh is required to describe correctly the flow within the seal. Hexahedral mesh elements were used to create three dimensional structured meshes in the entire domain. An adequate mesh refinement is allowed to the clearance area and boundary layers to accurately calculate pressure distribution along and around the seal and viscous fluid forces at the wall surfaces. Fig. 5 shows a cut section of generated computational grids in the 3-D computational domain.

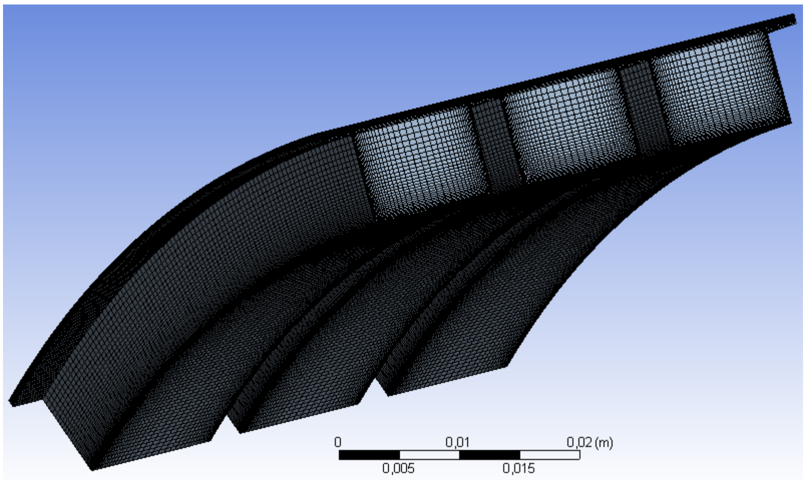

Fig. 5. A cut section of the 3-D Mesh used for the fluid labyrinth seal domain

2.3 FRAME MOTION TRANSFER

Observing the motion of rotor-seal system from a stationary frame, the rotor is spinning at the speed ω while also whirling at the speed Ω at the same time, which means that the location of rotor and thus the shape of mesh are changing all the time. So it is actually a transient problem involved with mesh moving. To avoid a transient analysis and moving mesh, a rotating frame with the speed Ω was applied as shown in figure 6. In the rotating frame, the rotor itself spins at the speed (ω-Ω), while the stator spins at the speed Ω in the opposite direction to the frame. Thus it becomes a steady state problem and there is no mesh moving. Viewing the same motions from the stationary frame, the rotor is actually spinning at the speed ω and whirling at the speed Ω, while the stator is at rest. The rotor surface moves with, against, or not at all relative to the whirling journal, depending on the whirl frequency ratio (WFR) defined as the ratio of rotor whirl to rotor spin. A WFR equal to unity is termed synchronous whirl where the rotor is whirling at the same frequency it rotates. A WFR of zero indicates a static displacement of the rotor which then simply spins.

The fluid driving forces exerted on the rotor can be obtained at by integration of pressure along and around the seal rotor surface. These driving forces act on the normal and transverse directions to the eccentric displacement as illustrated in figure 6. Rotor dynamic instability occurs when the forward driving forces exceed the resisting dissipation forces, which leads to self-excitation of the first whirling mode of the rotor [15].

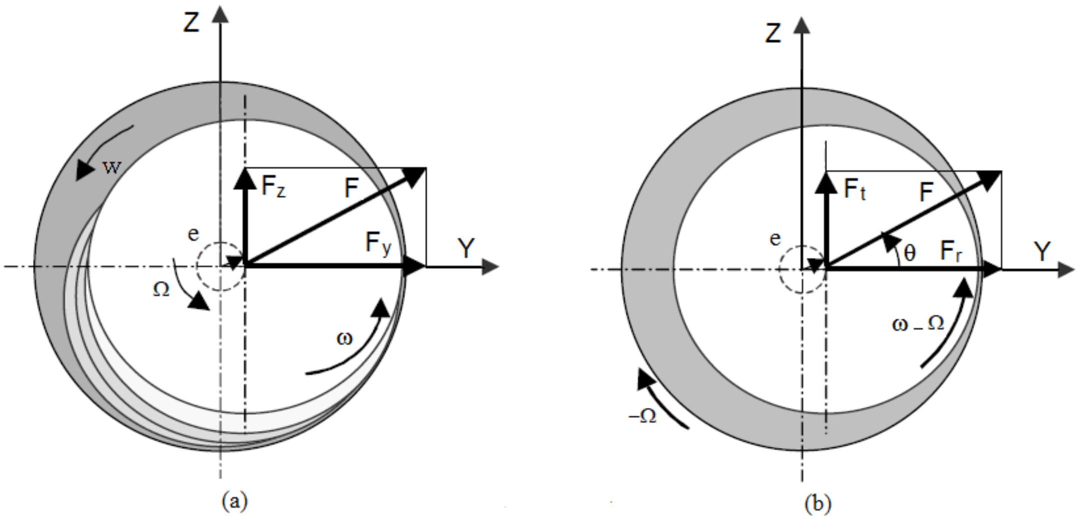

Fig. 6. Frame motion transfer from stationary (a) to rotating (b)

3 Results and discussions

The developed model has been solved in the given eccentric short labyrinth seal respecting the boundary conditions summarized in table1. Two turbulent models have been used for comparison. The realizable k-ε model and the k-ω SST turbulent model both considered more efficient than the standard k-ε model because of their generally reasonable results. This comparison will allow choosing the more appropriate model providing more accurate results. The pressure has been locally calculated in each cavity of the seal. Figure 7 shows theoretical and experimental static pressure distribution in the circumferential direction of the seal. Both of the two model predictions have been compared to experimental results of Rajakumar and Sisto [16]. Generally, it can be easily seen that pressure predictions are in good agreement with measurements in the seal cavities. Additionally, it is shown that the k-ω SST turbulent model provides overall better pressure prediction than the realizable k-ε model

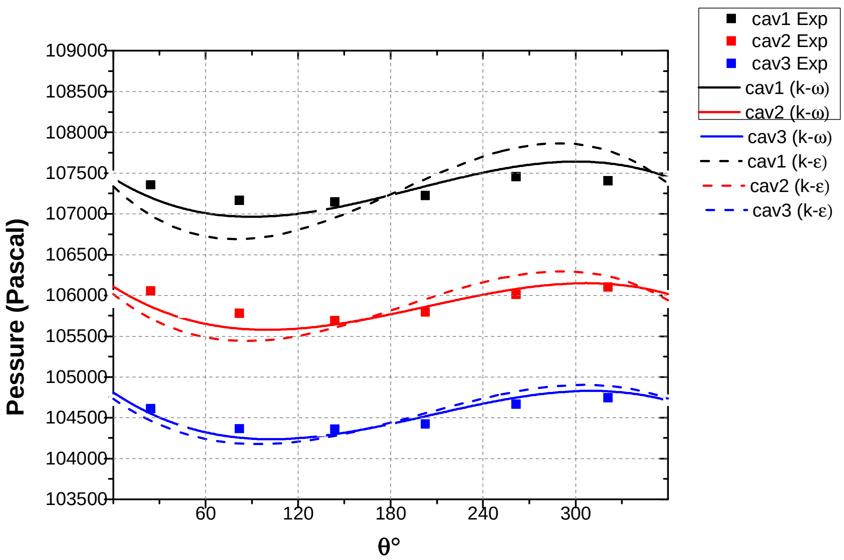

Fig. 7. Pressure distribution in the circumferential direction at the three cavities of the seal

Figure 8 shows a comparison of experiments and CFD predictions using k-ω SST turbulent model for static pressure distribution along the axial position of the seal. One can easily see that predictions are in good agreement with

Rajakumar and Sisto measurements [16]. Also, this figure shows that the pressure drop occurs from the inlet pressure at left to the outlet pressure at right, and the pressure is almost equal in the same tooth cavity interior.

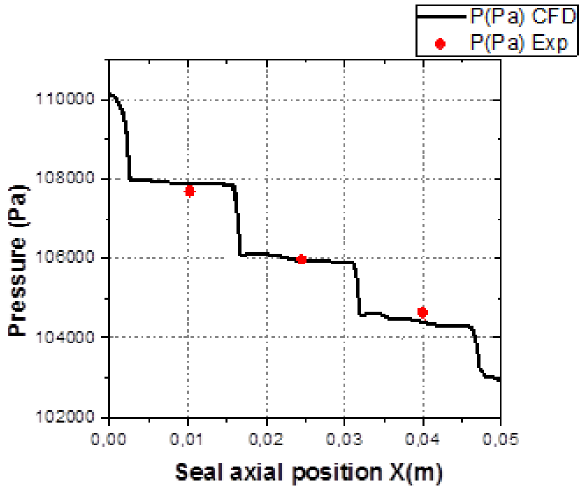

Fig. 8. Pressure distribution in the axial direction of the seal

Figure 9 shows contours of the static pressure in an axial plane of the seal. The pressure drop starts at the seal inlet and continues at each tooth throttling to rich the outlet pressure at the seal exit. The quasi same color in each cavity interior confirms that pressure is quasi constant in each cavity interior. Furthermore, this pressure distribution shows that pressure drop mainly occurs in the left zone of each cavity at the labyrinth tooth throttling. So, accurate pressure calculation at the tooth throttling is very important to achieve accurate results of the seal characteristics.

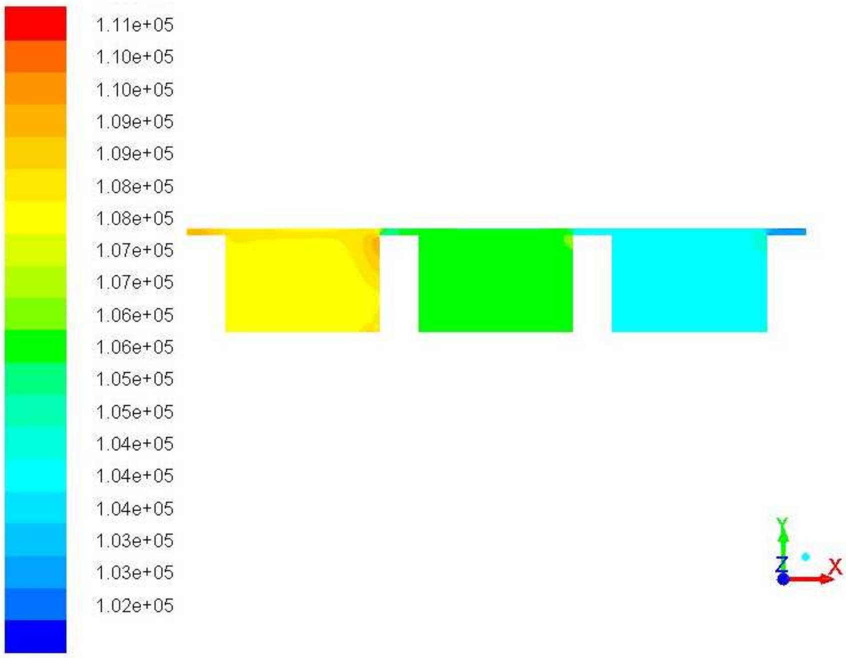

Fig. 9. Pressure contours in the XY plane of the seal

Figure 10 shows the velocity vectors in an axial plane of the seal. The high pressure drop occurs in the first cavity where a strong flow jet is generated making the flow more turbulent in this cavity. We note the presence of recirculation zones in the seal cavities. These vortexes act as brakes to stop the axial velocity of the flow through the seal and therefore to reduce the leakage flow.

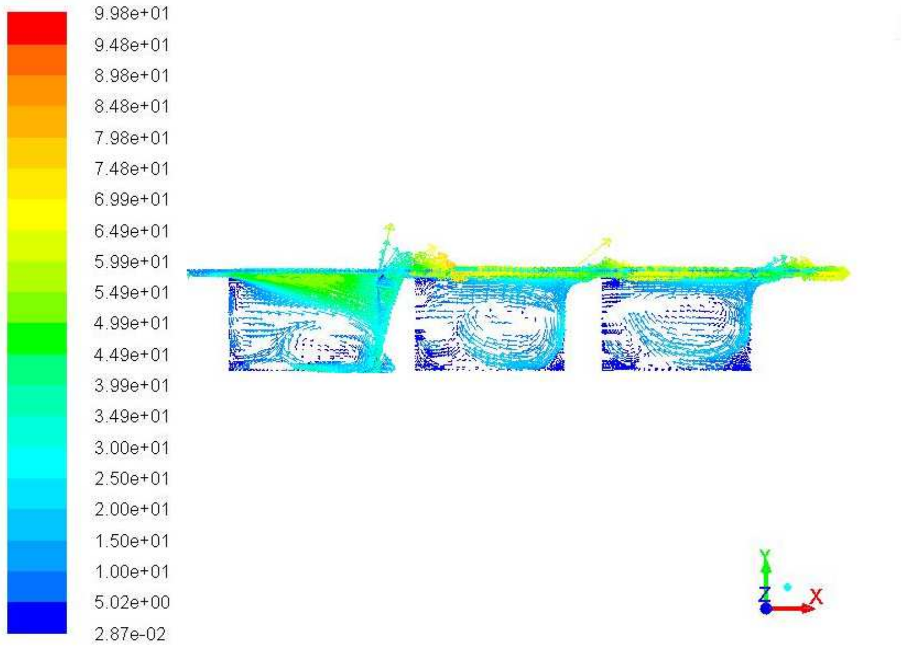

Fig. 10. Velocity vectors in the XY plane of the seal

Figure 11 shows the circumferential velocity distribution in a radial plane at mid-cavity with the inlet swirl as a parameter. It is shown that the circumferential velocity decreases, in the radial direction of the seal, from the shaft speed (W/ω = 1) at the lateral surface of the rotor to zero at the lateral inner surface of the frame. For the non-inlet swirl case ($W_{in}/R\omega = 0$), it can be easily seen that a very slight variation of the circumferential velocity is noted in the radial position at the center of the cavity and this velocity can be considered uniform in the mid-cavity. Furthermore, its mean value is about 60% of the shaft speed. However, this variation becomes important for high inlet swirls. It can be stated that when the inlet swirl exceeds 50% of the shaft speed, the circumferential velocity can't be considered uniform at the mid-cavity, especially when bi-dimensional computational models are used to predict rotordynamic characteristics for this kind of seals. This result confirms previous obtained results [17] when a tridimensional axisymmetric code based on Lagrangian-Eulerian method has been used to simulate an incompressible flow through a straight labyrinth seal with teeth fixed on the rotor.

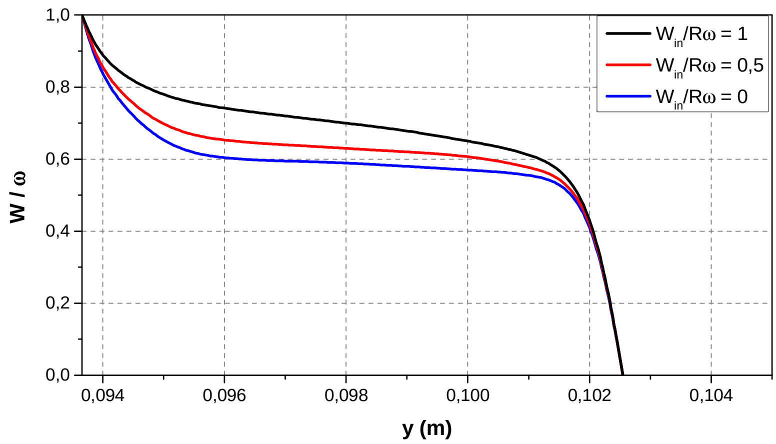

Fig. 11. Circumferential velocity profile in a radial plane at mid-cavity with the non-dimensional inlet swirl as a parameter

Figure 12 represents leakage flow versus the pressure ratio with the inlet swirl as a parameter. This figure shows that the leakage decreases with increasing pressure ratio. Additionally, it is shown that leakage flow decreases very lightly with increasing inlet swirl to the point that we can consider that the leakage through the seal is practically not influenced by the inlet swirl.

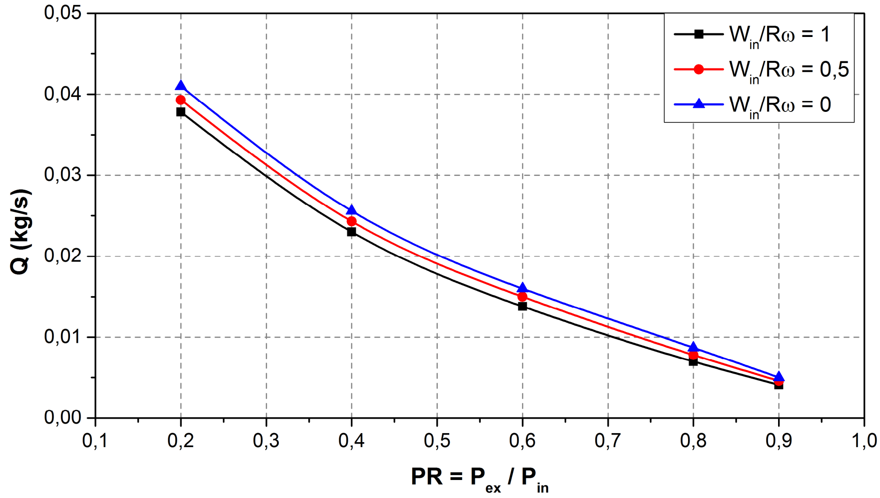

Fig. 12. Leakage flow versus pressure ratio with the non-dimensional inlet swirl as a parameter

4 CONCLUSION

A model to predict and analyze leakage and rotor dynamic characteristics of a turbulent flow through an eccentric short labyrinth seal has been developed based on CFD calculation. Two turbulent models have been used and compared. The k-ω SST model provides more accurate predictions than the realizable k-ε model for this kind of seals. Predictions of the pressure distribution along and around the seal are in good agreement with measurements. The pressure is almost quasi equal in the same tooth cavity interior and pressure drop mainly occurs in the left zone of each cavity at the labyrinth tooth throttling. Recirculation zones are shown in the center of each cavity and they act to reduce the leakage flow in the seal. when the inlet swirl exceeds 50% of the shaft speed, the circumferential velocity can’t be considered uniform at the mid-cavity of the seal and bi-dimensional computational models can’t be used to predict correctly and accurately rotor dynamic characteristics for this kind of seals.The leakage flow through the seal decreases with increasing pressure ratio but inlet swirl has practically no important influence on the seal leakage.

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