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Dry Gas Seal Simulation with Different Spiral Tapered Grooves

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ABSTRACT

The flow and performance analysis for a dry gas seal (DGS) with a standard and tapered spiral grooves is developed using a verified FLUENT CFD model. A comprehensive analysis is proposed to resolve the effect of modifying the seal grooves geometry on gas film pressure distribution and the seal performance. The effect of the fluid state and the rotational speed on the gas seal performance and internal flow field are studied and presented. The fluid state effect on the gas seal internal flow and performance is studied, Laminar and turbulent flow with RNG K-ε turbulence model and LES is examined for the same geometrical and operating conditions. Also different rotational speed effect on gas seal

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performance is examined for 0, 2500, 5000, 7500 and 10380 rpm. Three taper spiral grooves are studied each with three different angles, including taper grooves in the radial, circumferential and combined radial-circumferential directions. The laminar flow simulation for the dry gas seal agree well with the experimental results more than the turbulent flow simulation which overestimate the pressure distribution inside the seal. The results indicate that as the rotational speed increase the seal open force and leakage increase. The use of tapered type spiral groove causes a reduction in the seal open force and the leakage rate. Increasing the angle of radial taper groove reduces the temperature distribution inside the gas film. The Reduction in seal open force and leakage rate is higher when use combined Radial-circumferential taper more than radial and circumferential taper used.

Keywords: *Dry Gas Seal, Simulation, Spiral, Tapered, Grooves*

Introduction

Leakage is a wide spread phenomenon in the oil and gas production rotating machines. It does not only cause a waste of supplies and energy, but also threats to the safety of people around, due to flammable, explosive, corrosive, toxic working fluid. Advanced sealing technology is greatly needed to solve the problem. Dry Gas Seal (DGS) is a kind of non-contacting dynamic seal used for sealing of rotating shafts. It gets wide applications in petroleum and chemical industries, especially in centrifugal compressors and gas turbines because of high stability and low maintenance. The DGS is located on the compressor shaft sealing the flow inside the compressor from the external atmosphere. The seals are fed with a clean, heated, if necessary, and dry gas usually taken at the discharge of the compressor. [1]. The gas is forced to pass between a stationary and rotating parts. The stationary part slightly moves axially and the rotating part is a grooved ring that rotates by the compressor shaft. The shaft rotations generate an aerodynamic pumping effect due to the grooves creating a very small gap between the rotating and stationary faces. The flow generated by the pressure difference leaks between the two faces, which is then is forced to the venting system of the machine vent.

For many years, researchers have been working to develop a better understanding for DGS film flow dynamics. Bing, et al. [3], presented a CFD study for turbulence effects and methods for their evaluation which were not considered in the existing industrial designs of dry gas seal. The direct numerical simulation (DNS) and Reynolds averaged Navires Stokes (RANS) methods were utilized to predict the velocity field properties in the grooves and gas film. The presented RANS method underestimated the performance, compared with that of the DNS. Jing, et al. [4], developed a CFD study for a three dimension micro-scale flow field in spiral groove dry gas seals (S-DGS). Effect of gas flow state on the seal performance was analysed and presented under different gas film thickness. Bing and Huiqiang [5] discussed the micro scale effect on spiral groove dry gas seal performance in a numerical solution of a corrected Reynolds equation. The numerical results showed that the average pressure in the gas film and the sealed gas leakage increased due to micro scale effects. These micro scale effects, also, enlarged the open force, and significantly decreased the gas film stiffness.

Hifumi and Mitsuo [6] experimentally and numerically studied the hydrodynamics analysis of dry gas seal. They presented a comparison of Labyrinth-seal and Dry-Gas-Seal, especially for gas leakage performance. Miller, and Green, [7] developed two new methods for characterizing the properties of gas lubricated mechanical face seals. The first is based on the step jump method; the second computes the gas film frequency response using direct numerical time simulation of the pressure field for harmonic motion of the stator. Results from both methods agreed well with previously published results computed using the perturbation method. Peng et al. [8] developed empirical formulae for calculating the convection heat transfer coefficient of a stationary ring. However Brunetiere and Modolo. [9] reported that the range of validity of those empirical formulae was too small when compared to Reynolds numbers observed during different mechanical face seal operations. Moreover, some parameters, such as material property and geometry of the face seal, were not considered in those empirical formulae.

Numerical simulation was carried out by Heshun, and Cichang [10], based on the Navier-Stokes (N-S) equation, the laminar model and the SIMPLEC algorithm. Five geometric parameters were investigated emphatically includes groove depth, spiral angle, the ratio of groove width to the land width (circumferential), the ratio of dam length to the groove length (radial), and n number of groove. To obtain more details about the fluid field and high stability of dry gas seal, Weibing, et al. [12] developed a simulation with Reynolds equation of compressible perfect gas, on isothermal condition, for T shape grooved dry seal. They investigated the effect of T shape geometrical parameters such as; groove depth, and balance clearance on the seal performance they presented a theoretical base to support for engineering design and application of T-shape groove dry gas seal.

Kowalskik and Basu [13] described the analysis, design and testing of a spiral groove seal with reverse rotation capability. The spiral groove geometries were optimized for good forward rotational stiffness at design pressure and speed and acceptable reverse rotation capability at lower speed levels. The optimized parameters included number of grooves, groove angle, groove depth and land-to-groove width ratio. A comprehensive analysis method of simulating the complex thermo-flow in a dry gas seal is described by Hong, et al. [14] which was carried out by using the commercial computational fluid dynamics

(CFD) software CFX [15]. The three-dimensional velocity and pressure fields in the gas film flow and the temperature distribution within the sealing rings were investigated for three kinds of film thickness. A comparison of thermo hydrodynamic characteristics of the dry gas seal was conducted using air and helium as sealed gases. Qiu and Khonsari [16] developed a three-dimensional thermo hydrodynamic CFD model to study the characteristics of an inward pumping spiral groove mechanical seal pair using a commercial CFD software CFD-ACE. Based on their CFD model, a parametric study was conducted to evaluate the performance of the seal. They found that thermal behaviour plays an important role in the overall performance of a seal.

The above brief review reveals common shortcomings in existing studies in the literature. The studies are done for the standard design of the spiral groove dry gas seal with little attention to the thermo-flow analysis of the internal film. Furthermore these studies did not consider adapting DGS to wide variety of operating conditions. In the current study, a comprehensive analysis is proposed to resolve the effect of modifying the seal grooves geometry. Firstly, standard spiral groove DGS is modelled at different fluid state conditions and different rotation directions. Secondly, spiral groove DGS is modified by tapering the grooves, using annular grooves and tapered-spiral grooves with annular. For all studied cases, the internal flow and performance of the seal are analysed and compared with the standard DGS.

Governing Equations

In the present study, the three-dimensional, steady Reynolds-averaged Navier-Stokes equations are used. For absolute velocity formulation, the governing equations of fluid flow for a steadily moving frame may be written as follows [17]:

Mass conservation equation:

$$\nabla . \rho \vec{v}_r = 0 \tag{1}$$

Where:

$\vec{v}_r$: Relative velocity

ρ : Density

Momentum conservation equation:

$$\nabla . (\rho \vec{v}_r \vec{v}) + \rho [\vec{\omega} (\vec{v} - \vec{v}_t)] = -\nabla P + \nabla . \bar{\tau} + \vec{F} \tag{2}$$

Where:

$\vec{v}$: Absolute velocity

ω : The angle velocity of the rotator

$\vec{V}_t$: Translational frame velocity
P : pressure
$\overline{\overline{\tau}}$: Viscous stress tensor with relative velocity derivatives
$\vec{F}$: External body force

Energy conservation equation:

$$\nabla.(\rho\vec{V}_r H + p\vec{u}_r) = \nabla.\left(k\,\Delta T + \overline{\overline{\tau}}\,.\vec{v}\right) + S_h \tag{3}$$

Where:

H : Total enthalpy
$\vec{u}_r$: The velocity of the moving frame relative to the inertial reference frame
K : conductivity
T : Temperature
S_h : Volumetric heat source

The numerical simulation has been performed with different fluid states including laminar and turbulent states. The flow Reynolds number in the dry gas seal flow is defined as [16]:

$$Re = \frac{\rho V \delta}{\mu} \tag{4}$$

Where:

δ : Gas film thickness
μ : Viscosity of the fluid

Under the flow conditions and dimensions indicated in Table 1, The Reynolds number is 1147 for $\delta = 5\mu m$, therefor the gas film flow is still laminar. However, many references show the importance of turbulence while simulating the flow in DGS due to the grid scaling. Although, in the current model no scaling is used the Renormalized Navier Stokes K-ε RNG turbulence model is used to investigate its role in the new approach. The turbulent model is treated with non-equilibrium near wall treatments, with 2% turbulence intensity at flow inlet, which is compared with Large Eddy Simulation (LES). Second order discretization scheme is chosen for better accuracy. The solution residuals for continuity, momentum and energy equations have been set to be less than 10e-4. The seal performance parameters including leakage rate are monitored during solution to confirm its convergence.

The fluid is treated as an ideal gas [3], the equation of state is:

$$P = \rho RT \tag{5}$$

The Knudsen number is a dimensionless number defined as the ratio of the molecular mean free path length to a representative physical length scale;

it is commonly used to distinguish the flow micro effect:

$$K_n = \frac{\mu\sqrt{2RT_o}}{Ph} \tag{6}$$

According to Ruan [18] the flow in the present study can be treated as a continuous medium because $K_n = 0.002$ which is less than 0.01 in the gas film, even for the smallest film thickness h = 2 μm.

Model Description and Boundary Conditions

The model is used to simulate the fluid between the rotating and stationary faces of the dry gas seal. Figure 1 shows the basic geometrical parameters of the simulated domain and the spiral groove profile used. The computational domain and grid generation are created by GAMBIT preprocessor shown in Figure 2 in which the fluid domain is magnified by 1000 times in the groove depth direction for geometry and grid detail clarity only. The calculations are done without any scaling to capture the actual gas film properties between the seal faces. The sealed medium pressure is used at the inlet for the fluid as inlet boundary condition, the environmental medium pressure is used at the domain outlet as outlet boundary conditions, the rotating periodicity boundary condition is used and the wall boundary conditions with no slip velocity are employed at the stationary and rotating seal faces Ruan [18].

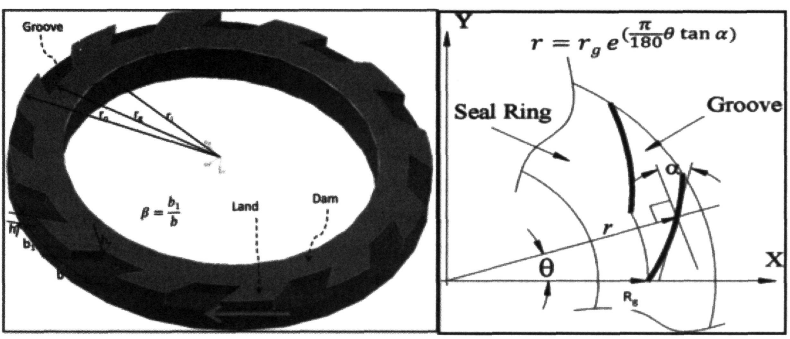


Figure 1: Standard spiral groove (DGS) geometric parameters

The single rotating reference frame model is used to model the fluid rotation. The model is validated with experiments of [11] for which the geometric and operating conditions are shown in Table 1. These geometrical parameters are used for the present study, and then modified according to the studied parameters. The hexahedral elements are used for domain meshing, using cooper method Figure 2. The geometry for the standard and modified DGS of the studied cases is show in Figure 3 and Figure 4.

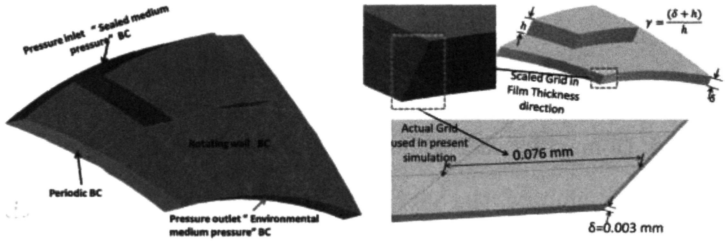


Figure 2: Standard spiral groove DGS boundary conditions and mesh of one – twelfth of the gas film

Table 1: Calculation conditions of [11] experiment for validation of the numerical method

Geometrical Parameter	Value	Operating Parameter	Value
Inner radius mm	58.42	Film depth µm	3.05
Outer radius mm	77.78	Sealed medium pressure bar	45.85 2
Groove root radius mm	69.0	Environmental medium pressure bar	1.01 3
Spiral angel (°)	15	Spin speed rpm	10 380
Ratio of groove to land	1	Inlet Temperature K	300
Grooves number	12	Dynamics viscosity g. m. s -1	0.018

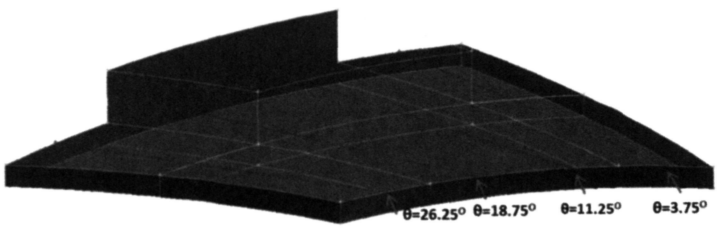


Figure 3: Standard spiral groove DGS computational domain and angular position for results

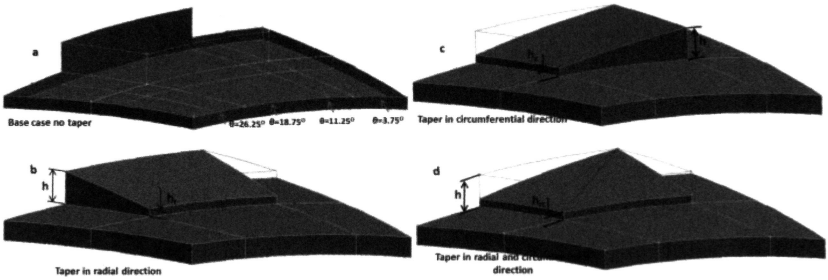


Figure 4: Domain for case without taper and cases with taper groove

Results and Discussion

Model validation and grid independence study

In order to ensure the accuracy of numerical solution, a careful check for grid independence is conducted. Three; 1.17, 1.8 and 2.173 million computational nodes are used. The pressure along the radial direction is plotted in Figure 5. For present simulation with the three mesh densities and the experimental results of [11]. The present results with 1.17 million agree well with experimental data from [11], which agrees also with the spiral groove DGS pumping theory. No significant change is noticed by using thee higher mesh densities, so the grid with 1.17 million cells is used for all cases conducted in the present simulation.

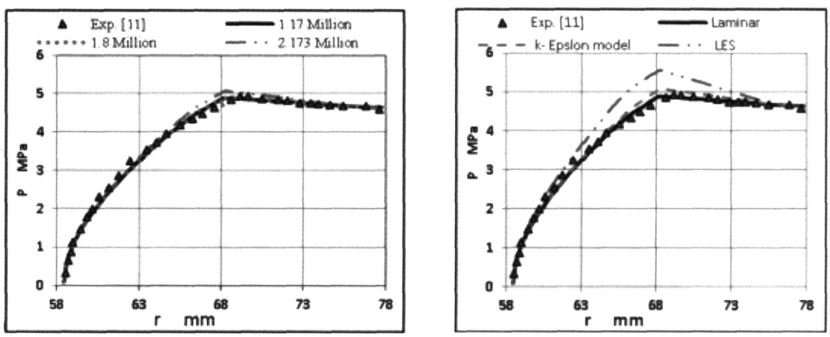


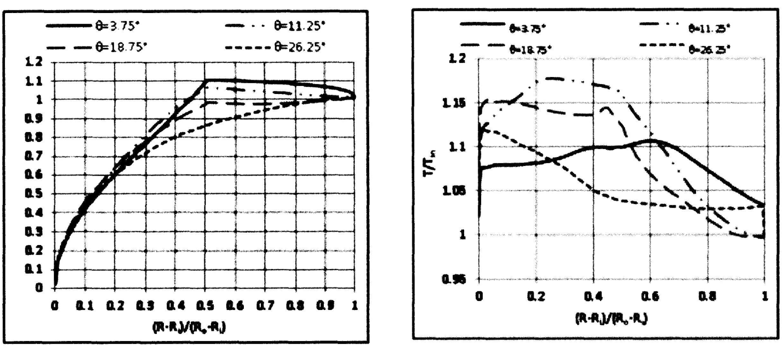
Figure 5: Standard spiral DGS static pressure distribution in radial direction θ = 11.25° with: - (a) different no. of computational nodes, (b) laminar flow and different turbulence model

Influence of fluid state for standard spiral gas seal

The influence of fluid state is done by simulating the flow as a laminar flow and as turbulent flow using different turbulence model. The pressure along the radial direction is plotted in Figure 5(b) for laminar and turbulent flow calculations The Laminar flow results agree well with experimental data from literature [11], whereas the solutions of the turbulent flow using K-ε turbulence model or large eddy simulation are less accurate. The turbulent calculation overestimates the peak value of the gas film pressure especially around the groove gross. This is because that by using turbulence models, medium gas will be seen as a compressible viscous fluid, and in fact, gas is sticky [4]. Which will make the calculated pressure at turbulent state is inevitable higher than the calculated pressure at laminar state. Based on this result, the laminar flow calculations are considered for all the studies conducted in the later sections.

Standard spiral gas seal internal flow analysis

The internal flow analyses is conducted for the static pressure and temperature at four angular positions shown in Figure 6; 3.75°, 11.25°, 18.75° and 26.25°. The relation between the normalized static pressure, with respect to inlet gas pressure, and the dimensionless radial direction position is shown in Figure 8(a); the pumping action of the spiral groove is noticed as the pressure rise approaching the inner dam. Static pressure peaks are observed for the angular positions over the groove $\theta = 11.25°$, near the dam, but with values lower than that at $\theta = 3.75°$, just downstream of the groove. The pressure develops across the groove region along the direction of rotation. The pressure variation over the land region at $\theta = 26.25°$, upstream of the groove, is minimal when compared



(a) (b)

Figure 6: Mid gas film internal flow at different circumferential positions: - (a) pressure distribution (b) temperature distribution

with those associated with the other circumferential positions, including that for the land region downstream of the groove.

The normalized temperature distribution, by the inlet temperature, along the radial direction is shown in Figure 8(b). Generally a temperature increase is noticed with the flow from the outer to inner diameter at all circumferential positions. This temperature increase has its highest value in the dam area after the groove for positions at $\theta = 11.25°$, and $\theta = 18.75°$. As the flow enter the groove, a large amount of heat is transferred through heat convection resulting in low fluid temperature. However, at the inner portion of the groove, there is only a small amount of fluid exchange between the groove and the fluid film reducing heat transfer due to convection which results in a higher temperature at the end of the groove.

Influence of rotational speed and rotation direction

To examine the influence of the rotational speed and direction on the DGS internal flow and performance, all the geometrical parameters are kept the same and the simulation is conducted with speed change from "-10380" to "+10380" rpm with a step of 5000 rpm. Normalized static pressure distribution in radial direction with different forward and reverse rotational speed is shown in Figure 7. At high forward rotational speed 10380 rpm, the pressure at the end of the spiral groove is about 8 percent higher than the inlet pressure at the seal outer diameter. This is due to the inward pumping of the fluid by the spiral grooves against the sealing dam with forward rotation. At a smaller forward

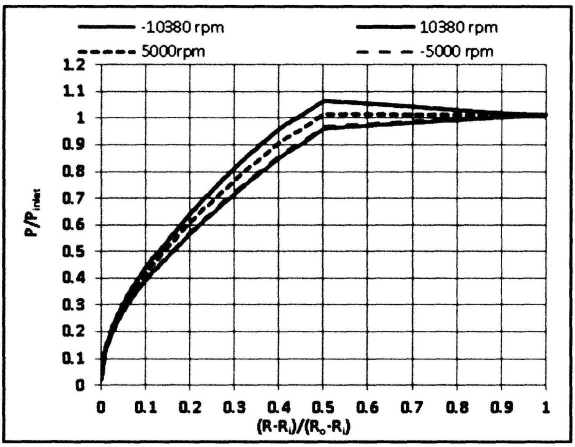


Figure 7: Normalized static pressure at $\theta = 11.25°$ with different rotational speed

rotational speed, this pumping action will be even smaller. At reverse rotation, the curve corresponding to - 5000 rpm shows that the pressure is not increased. This is due to the outward pumping tendency of the spiral grooves with reverse rotation, which creates a negative film stiffness that initiates instability and collapse. For all the reverse rotational speed the standard spiral grooves loss the inward pumping action, and inlet pressure decreases along the grooves without any pressure peaks. The same behavior is resulted in static pressure contours on the seal rotating face as shown in Figure 8.

The standard spiral groove seal performance is also affected by the rotational speed value and direction as shown in Figure 9. The lower speed reduces the open force and leakage rate. At reverse rotation, the lower speed also, reduces the seal open force due to the absence of the inward pumping effect. At this condition, the seal faces may be collapsed and seal failure is expected.

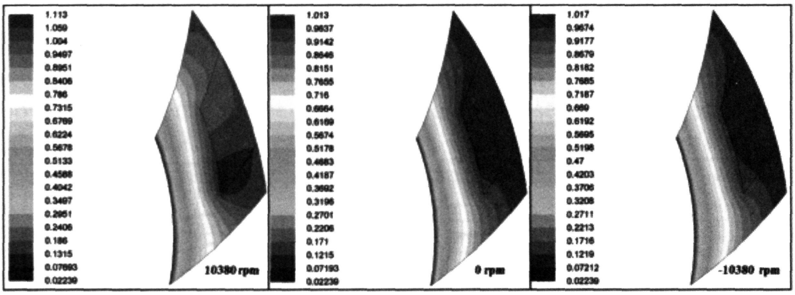
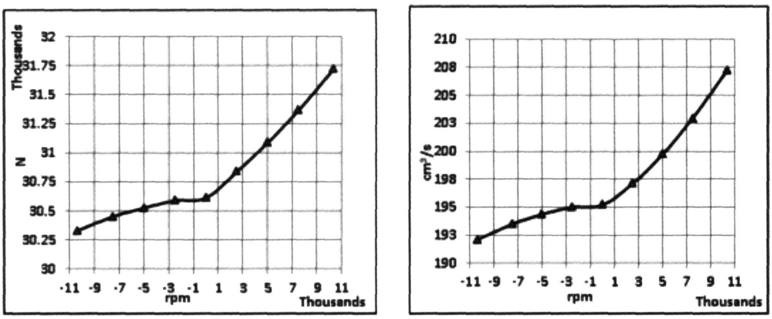


Figure 8: Normalized static pressure distribution in radial direction with different rotation direction

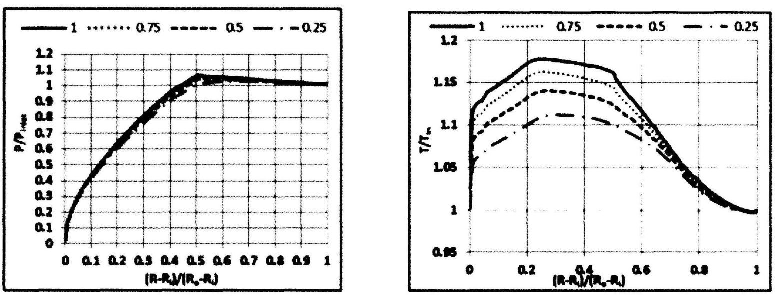


(a) (b)

Figure 9: Standard spiral DGS (a) open force with rpm, (b) leakage volume flow rate with rpm

Influence of using radial taper groove

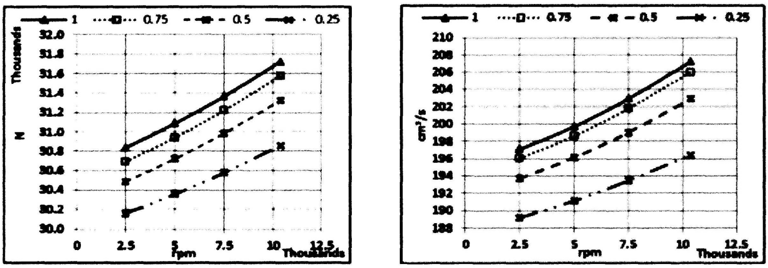
To examine the effect of using radial taper groove on the gas seal performance, the simulation is done for the gas seal with the same geometrical parameters except that the groove depth is decreased in the radial direction by ratio h_r/h, the rotating face geometry and the computational domain are shown in Figure 1. Four cases are studied with ratio 1, 0.75, 0.5 and 0.25; by decreasing this ratio the groove height at the gross is decreased. The radial taper groove has a small effect on the pressure distribution in radial direction Figure 10(a), as the taper angle increase the pressure distribution decreases and the pressure peak decreases. Figure 10(b) shows the effect of radial taper groove angle on the temperature distribution, and it is clear that the temperature decreases with increasing the taper angle especially at the groove end and seal dam area. This is thought to be due to the fluid exchange between the groove and the fluid film on top of the grooves. As the groove tapered with high angle, the area of the groove contact with the gas system is larger, and the fluid exchange between the groove and the fluid film becomes greater. Consequently heat is then carried through convective heat transfer, thus resulting in a lower fluid temperature and seal temperature.



(a) (b)

Figure 10: Radial taper with different depth reduction ration effect on (a) normalized static pressure (b) normalized temperature

Figure 11(a) illustrates how the seal open force varies with groove taper angle and speed. At low groove taper angle, the open force is high and as the groove taper angle becomes greater, the open force decreases rapidly. The leakage rate also decreases with increasing the taper groove angle Figure 11(b). This means that if one desires a low leakage rate and a low seal temperature, then high angle taper groove is required. If low friction and wear is preferred, then groove without taper or small tapper grooves should be applied.

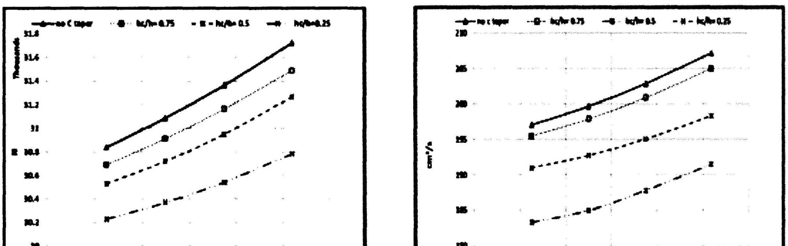


(a) (b)

Figure 11: Radial taper with different depth reduction ratio effect on (a) open force (b) leakage rate

Influence of using circumferential taper groove

In this numerical simulation the groove depth is decreased gradually in the circumferential direction with ratio h_c/h= 1, 0.75, 0.5 and 0.25 with the rotation direction as shown in Figure 6. Figure 12 show that the circumferential taper groove has a negative effect on the open force of a seal. With increasing taper angle, the open force decreases. Also the leakage rate decreases with using circumferential taper, especially at high taper angle, as a result of flow area across the groove.

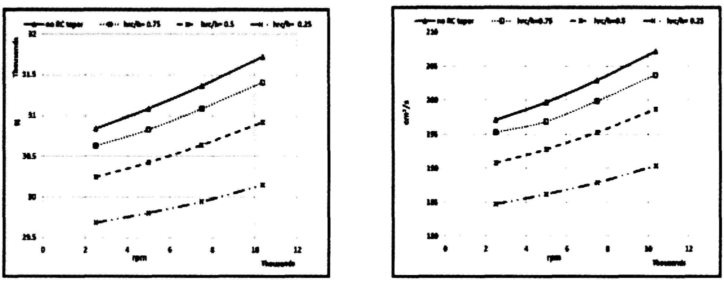


(a) (b)

Figure 12: Circumferential taper with different depth reduction ratio effect on (a) open force (b) leakage rate

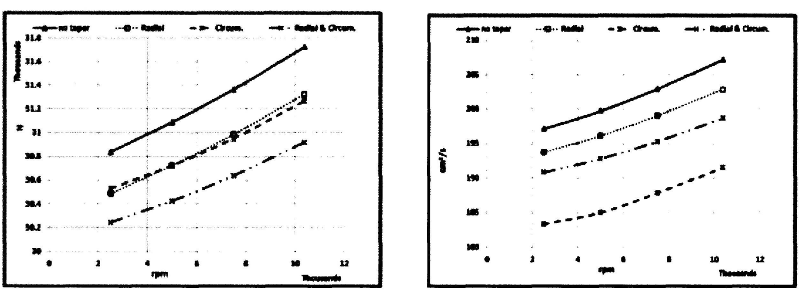
Influence of using combined radial and circumferential taper groove

To examine the effect of combined radial and circumferential taper groove on the seal performance, simulation is done for grooves with three taper angles with ratio h_{rc}/h = 1, 0.75, 0.5 and 0.25 shown in Figure 4 in both radial and circumferential direction. Figure 13(a) shows the effect of combined taper on the seal open force, as the combined taper angle increases the open force decrease. This decrease is slightly affected by the rotational speed, it can be seen that at high speed the decrease in open force is higher than that at lower speed. The leakage rate for different combined taper grooves angles is illustrated in Figure 13(b), as the taper angle increased the seal leakage decreased. As taper angle increases the friction area from groove inlet to groove end increased, that result in generating higher pressure drop across the grooves causing lower leakage rate. The performance results are drawn for the same taper angle but for different taper type. Figure 14 shows that the decrease in open force when combined taper is used is more than each of radial or circumferential taper used. The open force remains the same for radial and circumferential taper. At the same time the decrease in leakage rate for circumferential taper is much more the decrease when using radial taper grooves.



(a) (b)

Figure 13: Combined radial and circumferential taper with different depth reduction ratio effect on (a) open force (b) leakage rate



(a) (b)

Figure 14: Taper type with 0.5 reduction ratio effect on (a) open force (b) leakage rate

Conclusion

A three dimensional CFD model is developed to study the flow in a DGS with a standard spiral groove and another with advanced spiral groove, including spiral grooves with tapered-spiral grooves. The effect of the fluid state and the rotational speed on the gas seal performance and internal flow field are presented. The following conclusions can be drawn:

1. The Laminar flow simulation results agrees well with experimental data from literature whereas the solutions of the turbulent flow using K-Epsilon turbulence model or large eddy simulation overestimate the pressure distribution inside the gas seal.
2. The pumping action of the spiral groove is noticed as the pressure rise approaching the inner dam. Near the dam, static pressure peaks are observed over the groove but lower than the peak value at just downstream of the groove.
3. The pressure variation over the land region upstream of the groove is minimal comparing to those associated with the land region downstream of the groove, so the pressure distribution for standard DGS is not uniform in the circumferential direction.
4. As the rotational speed increases the pressure peak value at the groove root is increased. The seal open force and leakage rate are increased with the rotational speed increase.
5. At high forward rotational speed 10380 rpm, the pressure at the end of the spiral groove is about 11.4% higher than the inlet pressure at the seal outer diameter. On the other hand at reverse rotation, the pressure is not increased in radial direction. This is due to the outward pumping tendency of the spiral grooves with reverse rotation, which creates negative film stiffness initiating instability and collapse.
6. The radial taper groove has a small effect on the pressure distribution. However, increase the taper angle causes significant decrease in the temperature distribution inside the seal.
7. The use of tapered type spiral groove causes a reduction in the seal open force and the leakage rate.
8. The radial taper groove has a small effect on the pressure distribution and increase the taper angle causes a noticed decrease in the temperature distribution inside the seal.

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