Performance evaluation of bidirectional dry gas seals with special groove geometry

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Additional Information:

• This is an Accepted Manuscript of an article published by Taylor & Francis in Tribology Transactions on 4th April 2016, available online: http://dx.doi.org/10.1080/10402004.2016.1146380

Metadata Record: https://dspace.lboro.ac.uk/2134/20879

Version: Accepted for publication

Publisher: © Society of Tribologists and Lubrication Engineers. Published by Taylor & Francis.

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Performance Evaluation of Bidirectional Dry Gas Seals with Special Groove Geometry

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Abstract

There are a very few studies of bidirectional gas seals particularly those with certain profiles used in the industry. Parametric study of performance of bidirectional dry gas seals under a set of operating conditions is presented. The expounded approaches uses solution of 3D Navier-Stokes momentum and continuity equations for various forms of grooved gas seals, particular for the trapezoidal shape variety for which there has been a particular dearth of in-depth analysis. It is shown that such groove geometries enhance the load carrying capacity of the seal through increased hydrodynamic lift. This is as the result of enhanced localised wedge flow particularly with a reduced seal gap. Therefore, there is the opportunity of gap minimisation, whilst reducing leakage rate and power loss. For given operating loading, kinematic and thermal conditions as well as seal geometry and topography, the operating minimum film thickness may be considered as the main design parameter.

Keywords: Dry grooved gas seal; Bidirectional; Hydrodynamic lift; Power loss; Leakage

Nomenclature

\begin{itemize}
\item $A_s$: Total peripheral area at the inner and outer rims of contact
\item $c$: Speed of sound
\item $c_p$: Specific heat capacity at constant pressure
\item $D$: Inverse Knudsen number
\item $d$: Circumferential width of the mid-groove section at outer seal rim
\item $F_o$: Opening force (hydrodynamic lift)
\item $f$: Friction
\item $h$: Film thickness
\item $h_0$: Minimum film thickness
\item $h_g$: Groove depth
\item $h_{g1}, h_{g2}$: Relative depths at the triangular and middle trunk parts of the grooves
\item $i$: Index
\item $Kn$: Knudsen number
\end{itemize}
Ma  Mach number
\( \dot{m} \)  Mass flow rate
N  Rotational speed (RPM)
\( n_g \)  Number of grooves
\( P_l \)  Power loss
p  Pressure
\( p_{in}, p_{out} \)  Pressure at the inlet (outer radius) and outlet (inner radius)
\( \dot{Q}_l \)  Volumetric leakage flow rate
R  Specific gas constant
\( R_0 \)  Ratio of lower groove width to the total groove width in radial direction
Re  Reynolds number
r  Radius
\( r_i, r_o \)  Inner and outer radii of the seal
\( r_g \)  Groove root radius
\( r_{g1}, r_{g2} \)  Radii of the bottom rim of the lower and upper seal groove sections
\( r_m \)  Mean radius
r, \( \varphi \)  Radial and circumferential (polar) coordinates
\( \hat{r}, \hat{\varphi} \)  Unit vectors in the radial and circumferential directions
T  Temperature
v  Velocity

Greek symbols
\( \alpha \)  Spiral groove angle
\( \alpha_1, \alpha_2 \)  Spiral groove angle at the upper and bottom seal groove sections
\( \gamma \)  Heat capacity ratio
\( \eta \)  Dynamic viscosity of gas
\( \Delta_g \)  Angle of an annular sector comprising one complete groove
\( \theta \)  Angle of an annular sector used as computational domain
\( \theta_0 \)  Groove width to non-grooved surface width ratio in circumferential direction
\( \lambda \)  Molecular mean free path
\( \lambda_s \)  Strubeck film ratio
\( \rho \)  Gas density
\( \sigma \)  Standard deviation of surface roughness
\( \tau \)  Shear stress
\( \varphi \)  Circumferential direction
\( \omega \)  Angular speed

Subscripts
\( g \)  Groove
L  Left-hand side
1. Introduction

Unlike the unidirectional gas seals, the bidirectional seals are effective in applications where there is a possibility of change in the direction of rotation of the shaft for example as the result of failure of the process check valves [1]. Additionally, the bidirectional gas seals have the advantage of reducing the assembly errors in installation and maintenances [1].

The rapid development of dry gas seals and their application in oil and gas industry is owed largely to their excellent performance, which has enabled them as the replacement for the traditional floating oil ring seals, thus enhancing the sealing performance of compressors, reducing the adverse environmental effects of potential leakage of sealing oil [2]. There is also improved rotor system stability.

Dry gas seals offer significant advantages over conventional mechanical seals. With use of various etched grooves over the seal surface, the gas seal is enabled to maintain a very thin gap between its seat and face, which helps in minimisation of leakage, whilst reducing the wear rate. In addition, the use of tandem and triple sealing combinations with inert buffer gases allow application of dry gas seals in machinery which handle highly volatile fluids with negligible leakage to the environment [3].

There have been a number of interesting researches on the performance of gas seals. However, majority of these have dealt with unidirectional spiral groove seals. They include the works reported in [3-10]. In a pioneering work by Gabriel [4], spiral groove gas seals were studied. The stiffness and leakage from spiral groove gas seals were studied by Salant and Homiller in [5]. They identified the primary parameters in the study of the gas seals and showed the effectiveness of the spiral grooves in providing stiffness values which were comparable with those of hydrostatic seals. The work of Ruan [6] developed a finite element
analysis method for the study of spiral groove gas seals incorporating slip flow conditions. It was found that the slip flow can significantly affect the seal performance significantly at low speed and low pressure conditions. More recent studies including [7-9] have used CFD methods to study the gas seal performances at various operating conditions. Those studies include further details on the temperature and pressure distribution inside the grooves, which enables one to focus on optimisation of the groove geometry, for instance. In addition, the creep rotation of the seal ring with and without spiral grooves and associated dynamic model is studied extensively in [10].

Inclusion of asymmetric spiral grooves on the seal face induces hydrodynamic lift between the mating surfaces with better reliability [1]. However, in certain applications, where the direction of driving shaft may reverse and result in undesired direct contact between seal faces. In such circumstances gas seals with bidirectional symmetric surface features are an obvious choice.

Over the years, several different bidirectional groove designs have been suggested, including radial/parallel grooves [11], bidirectional tapered step grooves [12], three-row spiral grooves [13] and T-shape grooves [14]. The seals with bidirectional grooves of square and/or rectangular geometries are relatively easy to manufacture, although they may need a larger number of grooves to generate the desired load carrying capacity. Nevertheless, the advances in manufacturing processes such as using laser technologies have promoted use of more complex groove geometries.

Goldswain and Do Boer [15] patented a trapezoidal design for bidirectional grooves. Despite of the shape or form of a specific seal face, optimisation of seal design requires a profound physical understanding of the seal behaviour. A review of the available literature shows no specific study of hydrodynamic performance of such bidirectional seals with the above-mentioned specific design of the groove geometry. A detailed research on the performance of such seals is deemed essential for wider potential acceptance.

This paper investigates the hydrodynamic performance of trapezoidal bidirectional grooves (Figure 1) for a dry gas seal with the assumption of a steady, laminar, isothermal but compressible flow through the contact. For such think films the Reynolds equation, which is a simplified form of Navier-Stokes equations is commonly used for the numerical analysis purpose. However, the numerical analysis is carried out using the CFD module of COMSOL without adding to the computational costs significantly. Thus a more accurate set of results could be expected from the current analysis. It also enables the inclusion of the appropriate boundary conditions, as well as taking into account the variations of gas density with pressure for which, an ideal gas model is used.

2. Numerical Approach
Various models with pertinent boundary conditions can be used for the study of gas seals, depending on the design and operating conditions. The choice of a given model and associated boundary conditions is established, based on the operating conditions with regard to temperature, contact pressure and the seal gap, all of which are included in and the Knudsen number or its inverse Knudsen. The Knudsen number is defined as [16]:

\[
\text{Kn} = \frac{\lambda}{h} = \frac{\eta}{p h} \sqrt{\frac{\pi R T}{2}}
\]  

(1)

where, \(\lambda\) is the molecular mean free path, \(h\) is the film thickness, \(p\) is pressure, \(T\) is operating temperature and \(\eta\) is the dynamic viscosity of the gas. In addition, the inverse Knudsen number is [16] defined as:

\[
D = \frac{\sqrt{\pi}}{2 \text{Kn}}
\]  

(2)

Table 1 lists the flow regimes which are commonly considered for various ranges of the Knudsen number [17].

<table>
<thead>
<tr>
<th>Range</th>
<th>Flow type</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\text{Kn} &lt; 0.01)</td>
<td>Continuum flow with no slip boundary conditions</td>
</tr>
<tr>
<td>(0.01 \leq \text{Kn} &lt; 0.1)</td>
<td>Continuum flow with slip boundary conditions</td>
</tr>
<tr>
<td>(0.1 \leq \text{Kn} &lt; 10)</td>
<td>Transitional flow between continuum and molecular flow</td>
</tr>
<tr>
<td>(\text{Kn} &gt; 10)</td>
<td>Free molecular flow</td>
</tr>
</tbody>
</table>

Typical pressures assumed in the study of the gas seals usually are in the range of few MPa. For instance, relatively higher operating pressures of 2.1MPa [11] and 4.5MPa [4] as well as low pressures of around 0.3MPa [7] are reported in the literature although higher pressures of up to 40-55MPa can be encountered in current day applications. Depending on the operating temperature and film thickness, these would result in Knudsen numbers that can be higher than the given range for continuum flow with no slip conditions. It is noted that a typical temperature of 300K is assumed in the analyses conducted in [3-4, 11]. For these conditions, continuum flow with slip boundary condition may be assumed, for which the common form of Reynolds equation, for instance, cannot be generally used. This is because the molecular mean free path is no-negligible when compared with the seal clearance [18]. Under such conditions, the effect of slip flow should be taken into account.

The current analysis deals with conditions pertaining to lower pressures and tighter
clearances. These operating conditions represent harsher working conditions for gas seals, thus requiring use of appropriate models with suitable boundary conditions.

3. Geometry of Trapezoidal Bidirectional Grooves

Figure 1 shows the studied seal configuration with the aforementioned groove shapes. A more detailed set of geometrical specifications are also shown in Figure 2 in both radial and cross-sectional views. Figure 2b shows the rotating groove face and the flat stationary mating surface. The repetitive pattern of the groove and flat land areas in the circumferential direction allows the application of symmetric boundary conditions on an annular sector of the seal. Therefore, the annular sector, shown in Figure 2a, constitutes the computational domain. The local radius of the upper rim of the two triangular regions shown in Figure 2a is defined as follow:

\[ r = r_{g,i} \exp(\theta \tan \alpha_i), \quad \text{where } i = 1,2 \]  

(3)

The subscripts 1 and 2 represent the top and bottom grooves in Figure 2a, respectively. The parameter; \( r_g \) denotes the radius of the lower rim of the grooves. The angle subtending the radial line crossing each rim of grooves to the perpendicular determines the spiral angle of each groove, \( \alpha_i \). For the lower branch of the groove, this is given by (Figure 2a):

\[ \tan \alpha_1 = \frac{\ln(r_o/r_{g,1})}{\Delta_g/2 - \cos^{-1}(d/r_o)} \]  

(4)

and for the upper branch, the spiral angle becomes:

\[ \tan \alpha_2 = \frac{\ln(r_{g,2}/r_{g,1})}{\Delta_g/2 - \cos^{-1}(d/r_{g,1})} \]  

(5)

where, \( d \) is the circumferential width of the middle section of the groove at the outer rim of the seal and \( \Delta_g \) is the angle of a sector comprising a single groove’s circumferential width.
Figure 1: Schematics of the gas seal face with trapezoidal shape grooves

Figure 2: Detailed geometry of a sector of the gas seal comprising the groove shape details

(a) An annular sector of the seal face  (b) Side view (cross-section A-A)

The ratio of the upper groove width over the total groove width in the radial direction (an indication of the relative size of upper and lower groove branches) is defined as:

\[ R_0 = \frac{r_{g,1} - r_{g,2}}{r_o - r_{g,2}} \]  

(6)
A similar ratio can be defined in the circumferential direction. This is usually defined in the form of the ratio of groove width in the circumferential direction to the circumferential width of the non-grooved part as:

$$\theta_0 = \frac{\Delta g}{\theta - \Delta g} \quad (7)$$

The depth of each groove branch is also shown in Figure 2b. Referring to Figure 2b, the film thickness for the entire seal contact under steady state condition can be stated as:

$$h(r, \theta) = \begin{cases} 
  h_0 & \text{over non-grooved area (land)} \\
  h_0 + h_{g,1}(r, \varphi) & \text{over triangular part of groove} \\
  h_0 + h_{g,1}(r, \varphi) + h_{g,2}(r, \varphi) & \text{over central part of groove}
\end{cases} \quad (8)$$

### 4. Numerical Model

#### 4.1 Modelling assumptions

For numerical analysis for the current study, the following assumptions are made:

1. **Steady-state conditions:** In general, the gas seals operate in machinery such as compressors under relatively steady conditions (constant speed and load). In practice, however, in some cases transient conditions may ensue particularly with changes in pumping rates, altering the speed of compressors. However, for majority of the cases and for a large portion of their working lives, the assumption of steady state condition can be held as valid.

2. **The flow is considered to be isothermal:** The isothermal assumption can be considered as valid for sufficient a film thickness in excess of a critical value which would lead to a mixed regime of lubrication. This is based on the Stribeck’s film ratio of:

$$\lambda_s = \frac{h}{\sigma} > 3 \quad (9)$$

where, $\sigma$ is the composite surface roughness of seal faces, and is defined as:

$$\sigma = \sqrt{\sum_{i=1}^{2} \sigma_i^2} \quad (10)$$

in which, $\sigma_1$ and $\sigma_2$ represent the root mean square roughness value of the stationary and rotating seal faces, respectively. No direct asperity contact is expected to occur during the operation of the seal in the current analysis. This means that seal friction is considered to be entirely due to viscous friction of the film. Under these conditions relatively small
local changes to operating seal temperature would be expected to occur. However, the gas properties used in the analysis are calculated at the typical operating temperatures. Nevertheless, a thorough study would need to incorporate solution of energy equation for both fluid and solid bodies in the analysis, particularly for transient conditions (such as in stop-start or application of shock loads) or when the asperities on the opposite seal surfaces may come into the contact causing hot-spotting.

3. Iso-viscous conditions: the viscosity of gases can alter significantly with temperature. However, their viscosity is rather insensitive to changes in pressure as shown by Maxwell using the kinetic theory of gases. In addition, the viscosity of the gas used in the current analysis is considered to be at the given operating temperature. Therefore, the effect of temperature in the gas viscosity is taken into account.

4. An ideal gas model is considered: In general, the behaviour of the real gases deviate from that predicted by the ideal gas law, particularly at high temperatures and/or pressures. However, considering a more complex gas law necessitates a thorough thermodynamic analysis which is beyond the focus of the current study. The ideal gas law is a commonly made assumption in all studies dealing with tribology of dry gas seals [3-6, 11]. A more comprehensive analysis may need to include a more sophisticated alternative for the common ideal gas model.

5. Laminar gas flow: The validity of assuming a laminar flow depends largely on the operating conditions. Reynolds number is a good indication of the validity of such an assumption. Nevertheless, the results from investigations based on laminar flow can still be used as a “good” primary indication. For the conditions investigated here the Reynolds number is defined as:

\[
\text{Re} = \frac{\rho vr_{m} \omega h}{\eta}
\]  

(11)

In the equation above the tangential velocity at the mid seal radius and the film thickness are considered to be the characteristic speed and length respectively. The Reynolds number remains below 4000 in the current analyses. This indicate that the conditions studied here fall largely in the realms of laminar and to some extent transitional flow. For internal flow (flow in confined environment) conditions, a Reynolds number around 2100 to approximately 4000 is an indicator of occurrence of transitional flow regime. Shahin et al [9] have shown that in the study of gas seals the laminar flow model provides a better agreement with the experimental results than those obtained using turbulent flow models such as the K-ε or LES, particularly in the transitional flow regime.

6. The flow is compressible: The density of gas is a function of temperature and pressure. These are taken into account by replacing the density in the Navier-Stokes momentum and continuity equations from the ideal gas law in terms of pressure and temperature. The reason for considering a compressible flow is that the Mach number as stated below
varies between 0.05 and 0.43 for the investigated conditions in this paper:

$$\text{Ma} = \frac{\tau m_o}{c},$$

where $c = \sqrt{\gamma RT}$ \hspace{1cm} (12)

The upper value (i.e. Ma=0.43) is higher than the usual threshold Mach number of 0.2~0.3, beyond which the compressibility effect becomes important.

7. Flat and smooth seal faces: For conditions described in the current analysis, it is assumed that mixed and boundary regimes of lubrication do not ensue (as discussed in assumption 1). Therefore, any asperity interactions are ignored. In practice, the effect of surface roughness on flow dynamics particularly at low film ratios cannot be ignored. In such cases, roughness flow factors can be determined, using the approach reported by Patir and Cheng [19-20].

8. Rigid seal faces: The generated gas pressures are considered to be insufficient to cause any appreciable contact deformation. In addition, the thermo-elastic deformations arising from any non-uniform temperature distribution are also neglected. This is a reasonable assumption particularly with no direct contact of mating surfaces. Furthermore as the seal contact faces are assumed to perfectly flat there is no direct localised contacts which can occur in practice due to seal face waviness.

9. No misalignment: In real applications seal faces may be subject to small misaligned contact. To take this issue into account, transient inertial dynamics of the system, comprising the seal-rotor configuration would be required.

10. Seal rings are nominally flat and aligned in parallel: In practice the seal surfaces might be tapered or contain waviness originated from manufacturing processes or more importantly as a result of thermo-elastic deformations. Such effects are not included in the current study.

4.2 Governing equations

For a compressible and iso-viscous fluid, the governing Navier-Stokes momentum and continuity equations under steady state conditions, with no body force can be written as:

$$\nabla \cdot (\rho \vec{V}) = 0$$ \hspace{1cm} (13)

$$\rho (\vec{V} \cdot \nabla) \vec{V} = -\vec{V} p + \eta \nabla^2 \vec{V}$$ \hspace{1cm} (14)

where, in cylindrical coordinates the gradient, convective, vector Laplacian and ordinary Laplace operators take the following forms:

$$\vec{V} \equiv \left( \frac{\partial}{\partial r} + \frac{1}{r} \right) \vec{r} + \frac{1}{r} \frac{\partial}{\partial \theta} \vec{\theta} + \frac{\partial}{\partial z} \vec{z}$$ \hspace{1cm} (15)
\[ \vec{V} \cdot \vec{\nabla} \equiv V_r \frac{\partial}{\partial r} + V_{\theta} \frac{1}{r} \frac{\partial}{\partial \theta} + V_z \frac{\partial}{\partial z} \]  
(16)

\[ \vec{\nabla}^2 \vec{V} \equiv \left( \nabla^2 V_r - \frac{1}{r^2} V_r - \frac{2}{r^2} \frac{\partial V_{\theta}}{\partial \theta} \right) \hat{r} + \left( \nabla^2 V_{\theta} - \frac{1}{r^2} V_{\theta} + \frac{2}{r^2} \frac{\partial V_r}{\partial \theta} \right) \hat{\theta} + \nabla^2 V_z \hat{z} \]  
(17)

\[ \nabla^2 \equiv \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2}{\partial \theta^2} + \frac{\partial^2}{\partial z^2} \]  
(18)

For gas density the ideal gas law provides:

\[ \rho = \frac{p}{RT} \]  
(19)

In the study of gas seals, if the Knudsen number; \( Kn > 0.01 \), then slip boundary conditions should be taken into account.

For the cases studied in this paper, the Knudsen number is below 0.01; and hence, the slip boundary conditions are not required. This is mainly due to the higher demanded sealing pressure and the existence of a closing force which determines the minimum contact film thickness.

**4.3 Boundary conditions**

The pressure boundary conditions over the inner and outer seal radii in the circumferential direction are considered to remain constant as (see Figure 3):

\[ p(r_i, \varphi) = p_{out}, \quad \text{and} \quad p(r_o, \varphi) = p_{in} \]  
(20)

As mentioned earlier, the repetitive pattern of the groove and flat land areas in the circumferential direction allows for considering only an annular sector with a single groove unit, which is assumed to be repeated along the ring face in the circumferential direction. Therefore, the annular sector, shown in Figure 3, constitutes the computational domain. Periodic boundary conditions are considered at the two lateral extremities of the annular computational seal sector. Implementation this condition requires that the pressure gradient at any radial position in the circumferential direction to diminish. Furthermore, mass flow continuity into and out of the annular computational sector should be assured. Therefore:

\[ \frac{\partial p}{\partial \varphi} \bigg|_{(r,\varphi_L)} = \frac{\partial p}{\partial \varphi} \bigg|_{(r,\varphi_R)} = 0 \]  
(21)

\[ \dot{m}(r, \varphi_L) = \dot{m}(r, \varphi_R) \]  
(22)
The mass flow rate per unit length at any given radius over either the left or the right-hand side boundaries of the annular sector (Figure 3) becomes:

\[
\dot{m}(r, \varphi_i) = \frac{\rho h^3}{12\eta r} \frac{\partial p}{\partial \varphi_i} + \frac{\rho r \omega h}{2}
\]

(23)

![Figure 3: The annular computational sector and boundary conditions](image)

In the current analysis, the face seal, with the groove feature is considered to be the rotating surface with its counterpart remaining stationary.

### 4.4 Performance parameters

The lift or opening force is calculated through integration of conjunctural pressure distribution over the contact area as:

\[
F_o = n_g \int_{r_l}^{r_o} \int_{\varphi_L}^{\varphi_R} p r d\varphi dr
\]

(24)

The pressure distribution in the contact is obtained by solving the governing equations described in Section 4.2 accompanied by the boundary conditions described in Section 4.3 providing that the minimum film thickness, \( h_0 \), is known a priory. To find exact value for the minimum film thickness, an iterative method is used by assuming an initial value for \( h_0 \). In this method, the balance between the produced load carrying capacity in the contact (opening force) and the applied load (closing force) will determine whether the assumed value for the minimum film thickness was correct. Otherwise, a new value for the minimum film thickness is sought and the process continues until a balance between the opening and closing forces achieved.
The volumetric flow rate of leakage is determined through integration of flow flux around the computational domain as:

\[ \dot{Q}_l = \oint \vec{v} \, d\vec{A}_s \quad (25) \]

where, \( A_s \) is the total peripheral area around the edges of the computation domain and \( v \) is the velocity at the boundary.

In the absence of any direct boundary interactions, the generated friction is due to the viscous shear of the film, thus:

\[ f = n_g \int_{\tau_o}^{\tau_R} \int_{\phi_L}^{\phi_R} |\vec{\tau}_v| r d\phi \, dr \quad (26) \]

where, viscous shear is:

\[ \vec{\tau}_v = \tau_{v,r} \hat{\phi} + \tau_{v,\phi} \hat{\phi} \quad (27) \]

A key measure of performance is the minimisation of power loss, obtained as:

\[ P_l = n_g \int_{\tau_o}^{\tau_R} \int_{\phi_L}^{\phi_R} |\vec{\tau}_v| \omega r^2 d\phi \, dr \quad (28) \]

**4.5 Method of solution**

The governing equations of motion and associated boundary conditions, described above are set in the environment of COMSOL v.5.0. The geometry of the seal is considered to be similar to those encountered in the study of unidirectional groove seals. This also provides an opportunity to compare the results of unidirectional and bidirectional grooves implemented in similar seal configurations. The groove shapes are then implemented on the seal surface, based on the available physical data. The computational domain including a single groove as described in Section 4.3 is the set up and meshed using a mixture of quadrilateral and tetrahedral elements. The computational domain comprises a total of 38,000 meshing elements.

A set of mesh-dependency numerical simulations were carried out in order to determine the minimal variation in predictions with mesh density and quality. Before proceeding to the specific analyses of the trapezoidal shape gas seals, the numerical methodology is validated against available results in the literature. This ensures the validity of the approach expounded here.
5. Validation of the Numerical Method

Two published cases are chosen for the purpose of validation of the modelling approach expounded here. These are a unidirectional spiral groove face seal analysis, reported by Gabriel [4], and a bidirectional radial groove face seal presented by Basu [11].

5.1 Validation for a unidirectional spiral groove seal

The spiral groove seal geometry, considered by Gabriel [4] is shown in Figure 4.

![Diagram of a gas seal with spiral grooves studied by Gabriel [4]](image)

The geometrical data as well as the associated working conditions are listed in Table 2.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner radius, $r_i$</td>
<td>58.24</td>
<td>mm</td>
</tr>
<tr>
<td>Outer radius, $r_o$</td>
<td>77.78</td>
<td>mm</td>
</tr>
<tr>
<td>Groove root radius, $r_g$</td>
<td>69</td>
<td>mm</td>
</tr>
<tr>
<td>Spiral groove angle, $\alpha$</td>
<td>15</td>
<td>°</td>
</tr>
<tr>
<td>Groove depth, $h_g$</td>
<td>5</td>
<td>µm</td>
</tr>
<tr>
<td>Number of grooves, $n_g$</td>
<td>12</td>
<td>-</td>
</tr>
<tr>
<td>Pressure at the inner radius, $p_{out}$</td>
<td>101.3</td>
<td>kPa</td>
</tr>
<tr>
<td>Pressure at the outer radius, $p_{in}$</td>
<td>4585.2</td>
<td>kPa</td>
</tr>
<tr>
<td>Angular speed, $\omega$</td>
<td>1087.1</td>
<td>rad/s</td>
</tr>
<tr>
<td>Operating temperature, $T$</td>
<td>303</td>
<td>K</td>
</tr>
</tbody>
</table>
Other studies have also compared their results with those of Gabriel [4]. These include the work of Wang et al [21], using a DNS method, Xu et al [22] using ANSYS-FLUENT and Wang et al [8] using CFX computational fluid dynamics. Figure 5 provides a comparison of results of the current study with those of the aforementioned authors.

![Figure 5](image)

**Figure 5:** Comparison of open forces at different minimum values of film thickness for the spiral groove seal (Table 2)

It can be seen that the results of the current analysis follow a similar trend to the other published results and reside in the same range. In particular, good correlation is noted between the current predictions and those of Wang et al [8, 21]. In particular at higher film thickness values, the opening force values from different analyses converge.

For the current analysis a computational domain of 76,000 meshing elements with at least 10 elements across the film thickness at the flat face non-grooved areas, where minimum film thickness occurs, were employed. This mesh density was chosen after a rigorous mesh sensitivity analysis of the results.

### 5.2 Validation for a bidirectional groove seal

The geometry of the radial groove face seal studied by Basu [11] is shown in Figure 6.
Table 3: Geometrical and operational data for radial groove face seal gas seal [11]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner radius, ( r_i )</td>
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<td>mm</td>
</tr>
<tr>
<td>Outer radius, ( r_o )</td>
<td>61.5</td>
<td>mm</td>
</tr>
<tr>
<td>Groove root radius, ( r_g )</td>
<td>55.0</td>
<td>mm</td>
</tr>
<tr>
<td>Number of grooves, ( n_g )</td>
<td>72</td>
<td>-</td>
</tr>
<tr>
<td>Minimum gap, ( h_0 )</td>
<td>1~7 µm</td>
<td></td>
</tr>
<tr>
<td>Dynamic viscosity, ( \eta )</td>
<td>( 2 \times 10^{-5} ) Pa.s</td>
<td></td>
</tr>
<tr>
<td>Pressure at the inner radius, ( p_{out} )</td>
<td>100 kPa</td>
<td></td>
</tr>
<tr>
<td>Pressure at the outer radius, ( p_{in} )</td>
<td>2100 kPa</td>
<td></td>
</tr>
<tr>
<td>Rotational speed, ( N )</td>
<td>10,000 RPM</td>
<td></td>
</tr>
<tr>
<td>Operating temperature, ( T )</td>
<td>300 K</td>
<td></td>
</tr>
</tbody>
</table>

In Basu [11], an FEM approach is employed. Figure 7 provides a comparison of the results of the current study and that of Basu [11] for the opening forces at different minimum film thickness values. In general, the trends are very similar. Any differences in predicted values are less than 8.7% utmost. However, Basu [11] did not state all the parameter values, such as groove depth and groove land-to-width ratio. Representative values for these parameters were assumed in the current analysis.

The conformity of the predictions of the current analysis to those of Wang et al [8, 21] and Basu [11] for unidirectional and bidirectional seals provide for satisfactory validation of modelling methodology and boundary conditions of the current study.
Figure 7: Comparison of open forces at different minimum film thickness values for a radial groove face seal

6. Results and Discussion for the Trapezoidal Shape Bidirectional Groove Seals

With the validated method, an investigation of tribological conditions for the case of a trapezoidal shape bidirectional groove seal in undertaken. The geometry of this seal, operational parameters, thermos-mechanical properties and the gas properties are provided in Tables 4, 5 and 6, respectively.

Table 4: Seal geometry

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner radius, $r_i$</td>
<td>58.42</td>
<td>mm</td>
</tr>
<tr>
<td>Outer radius, $r_o$</td>
<td>77.78</td>
<td>mm</td>
</tr>
<tr>
<td>Number of grooves, $n_g$</td>
<td>12</td>
<td>-</td>
</tr>
<tr>
<td>Grooved area angle, $\Delta_g$</td>
<td>15</td>
<td>°</td>
</tr>
<tr>
<td>Bottom rim radius of lower groove, $r_{g,2}$</td>
<td>69</td>
<td>mm</td>
</tr>
<tr>
<td>Ratio of lower groove width in radial direction, $R_0$</td>
<td>1/3, 1/2 and 2/3</td>
<td>mm</td>
</tr>
<tr>
<td>Triangular part depth, $h_{g,1}$</td>
<td>1, 2, 3, 4, and 5</td>
<td>μm</td>
</tr>
<tr>
<td>Middle trunk groove depth, $h_{g,2}$</td>
<td>6, 7, 8, 9, 10, 11 and 12</td>
<td>μm</td>
</tr>
<tr>
<td>Middle trunk groove width, $d$</td>
<td>4, 6, 8 and 10</td>
<td>mm</td>
</tr>
</tbody>
</table>

Table 5: Operational conditions
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum film thickness, $h_0$</td>
<td>2, 3, 4, 5, 6 and 7</td>
<td>µm</td>
</tr>
<tr>
<td>Inlet pressure (at outer radius), $p_{in}$</td>
<td>0.2, 4.5</td>
<td>MPa</td>
</tr>
<tr>
<td>Outlet pressure (at inner radius), $p_{out}$</td>
<td>0.101</td>
<td>MPa</td>
</tr>
<tr>
<td>Rotor speed, $N$</td>
<td>3,000–18,000</td>
<td>RPM</td>
</tr>
<tr>
<td>Operating temperature, $T$</td>
<td>223, 300 and 423</td>
<td>K</td>
</tr>
</tbody>
</table>

Table 6: Thermo-mechanical and gas properties

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas type</td>
<td>Air</td>
<td>-</td>
</tr>
<tr>
<td>Dynamic viscosity, $\eta \times 10^{-5}$</td>
<td>1.47, 1.85 and 2.39</td>
<td>Pa.s</td>
</tr>
<tr>
<td>Specific heat capacity, $c_p$</td>
<td>1006, 1006 and 1017.5</td>
<td>J/kgK</td>
</tr>
<tr>
<td>Specific gas constant, $R$</td>
<td>287.03</td>
<td>J/kgK</td>
</tr>
<tr>
<td>Heat capacity ratio, $\gamma$</td>
<td>1.4046, 1.4017 and 1.3942</td>
<td>-</td>
</tr>
</tbody>
</table>

6.1 Meshing data and convergence criterion

The computational domain is covered by a mesh of quadrilateral and tetrahedral elements. The former mainly covers the flat non-grooved areas, as well as the middle trunk of the groove, whilst the latter of mainly a triangular shape covers the branches and the relevant areas around the groove edges. Figure 8 shows an example of the mesh quality and distribution.

![Figure 8: An example of the produced computational mesh](image)

Figure 8 shows the variation of the opening force. It also shows the computation time with mesh refinement. The opening force reduces to less than 0.004% as the number of mesh elements increase from 38,000 to 58,000, whilst the computation time increases by almost three folds.
6.2 Local pressure distribution

Typical conjunctional pressure distribution at two different minimum film thickness values of 2 and 5µm are shown in Figure 10a and b. These results are for the inlet pressure of $p_{in} = 4.5$ MPa, temperature of $T = 300$ K, and shaft speed of $N = 10,380$ RPM CW, $h_{g1} = h_{g2} = 5\mu$m.
A more detailed representation of pressure distribution is shown in Figure 11a and b, where the variations of pressure over the seal face in the radial direction at different circumferential locations are depicted for the two studied minimum film thickness values. The positions of these locations are also shown in the Figure.
Figure 11: Pressure distribution in radial direction at different circumferential cross-sections

In both cases, the pressure at $\varphi = -5^\circ$ is higher than that at $\varphi = 5^\circ$, because of the sense of rotation. With clockwise rotation the hydrodynamic wedge effect is more significant at the left branch of the groove.

The important point to note is the drop in the effectiveness or sensitivity of the pressure distribution to a higher film thickness. This can be seen by comparing the results in Figure 11b with those in 11a.
6.3 Gas seal performance under different operating conditions

In the study of gas seals the important measures of performance for a given seal design under certain intended application are mainly the leakage and operational non-contacting gap. This means a delicate balance should be maintained between the forces which arise from the build-up of hydrostatic/dynamic conjunctional pressure and the applied closing seal force. This is to ensure safe and reliable operation of the machinery. In addition to the parameters such as inherent unbalance, shaft misalignments, manufacturing imperfections, etc., there are three main categories of parameters that can influence performance of a gas seal. These include: operating parameters (such as rotational speed, pressures at the inner and outer radii of the contact, environmental temperature, sealing gas, etc.), geometrical design parameters (such as seal size, groove shape and various associated geometrical parameters, number of grooves and their distribution and location, etc.) and thermo-mechanical properties of the seal materials. Although some of these parameters are dictated through the requirements of application such as the pressures at the boundaries, there are also parameters that can be effectively controlled.

The sealing pressure and type of fluid are determined by the intended application. The inlet gas temperature is a function of environmental (climate) conditions. The choice of three different temperatures in this study reflects on those variable conditions. However, the operating temperature is affected by the build-up of heat in the contact due to shear gas forces and that between the gas and the seal surfaces. Ideally, the maintenance of a reliable gap between the contacting surfaces is desired, using a minimum input energy into the system. This gap should also allow sufficient amount of gas to flow through the contact to generate the required hydrodynamic lift which would balance the closing force. It is, therefore, interesting to observe the variation in the performance measures of the seal at various gap sizes for a given applied closing force.

Figure 12 demonstrates the minimum gap between the gas seal surfaces as the rotational speed of the shaft varies. At each speed the minimum gap is computed based on a force balance between the closing force of 6645N and the generated contact hydrodynamic lift. The figure also shows that as the operating temperature rises, the minimum film thickness increases mainly due to an increase in the gas viscosity, in contrast to what would be expected of liquids.
Figure 12: Variation of minimum gap with shaft rotational speed at different operating temperature.

Figure 13 illustrates the corresponding power loss and gas leakage from the contact at the minimum film thickness values shown in Figure 12. The figure shows that as the operating temperature and speed rise, both leakage and power loss increase accordingly. It is important to note that the variation in the minimum film thickness is restricted to the condition that the resultant opening force (hydrodynamic lift) in the contact equates the closing force of 6645N. This figure also indicates that at lower speeds the sensitivity of performance parameters to generated temperature is reduced.

Figure 13 indicates that the minimum possible gap would reduce leakage as well as power loss. However, in practice the gap between seal faces can be reduced to the extent that no direct contact of seal faces occurs. This is clearly a function of the topography of seal faces. When direct interaction of counter face asperities occur some of the contact load is carried by them. This results in a mixed hydrodynamic regime of lubrication which increases friction and thus, power loss. The Striebeck parameter, \( \lambda_s \), which is the ratio of the gap or film thickness to the composite standard deviation of roughness of both surfaces is used to determine the onset of mixed regime of lubrication. It is usually assumed that the Striebeck ratio of \( \lambda_s \approx 3 \) is the critical value below which indicates the onset of mixed regime of lubrication. For the current analysis, the composite RMS roughness was measured to be \( \sigma = 0.63 \mu m \). This corresponds to the critical minimum film thickness of \( h_c = 1.9 \mu m \) for the current study. This limiting boundary is shown in Figure 13 and determines the limit to which the minimum gap may be reduced.
Therefore, the minimum film thickness is an important parameter for the design of gas seals. Consequently, for varying operating range (i.e. rotational speed and sealing pressure), a system of variable closing force may be devised to maintain an optimum minimum film thickness, which determines the power losses and the leakage rate. This minimum film thickness should also guard against direct interaction of seal faces, thus endure functional durability of the surfaces.

Figure 13: Typical characteristic curves for a gas seal performance
7. Conclusions

The performance of trapezoidal-shaped bidirectional groove dry gas seals is presented. Using the available computational tools the 3D Navier-Stokes momentum and continuity equations are solved. It is noted that under given operating conditions, the Knudsen number falls outside the slip boundary flow condition. Therefore, in the studied applications, continuum flow regime prevails.

The results for conjunctional pressure distribution in the bidirectional groove geometry are presented, showing that the build-up of pressure at the groove is influenced by the minimum gap between the two seal surfaces. A lower gap enhances the influence of groove upon hydrodynamic lift, thus the load carrying capacity. It is also shown that any rise in temperature and speed increases the leakage and power loss form the contact.

It is concluded that the closing force should ideally be determined by the minimum seal gap, thus minimising leakage and reducing the power loss. However, there is a practical limit in reducing the film thickness, governed by the topography of contacting surfaces. The approach is to reduce the chance of direct boundary interactions, thus friction and wear.

Acknowledgments

The first author wishes to express her gratitude to the Chinese Scholarship Council (CSC) for providing her with the opportunity to carry out this research at Loughborough University, UK.

Thanks are also due to John Crane Ltd. for providing representative data and technical support.

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