Tribology International 102 (2016) 333–347

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

The influence of real gas effects on the performance of supercritical CO_2 dry gas seals

Z.M. Fairuz [a,*], Ingo Jahn [b]

[a] Queensland Geothermal Energy Centre of Excellence, School of Mechanical and Mining Engineering, The University of Queensland, 4072 St. Lucia, Queensland, Australia

[b] Centre for Hypersonics, School of Mechanical and Mining Engineering, The University of Queensland, 4072 St. Lucia, Queensland, Australia

ARTICLE INFO

Article history:
Received 17 February 2016
Received in revised form
29 April 2016
Accepted 25 May 2016
Available online 27 May 2016

Keywords:
Supercritical carbon dioxide
Dry gas seal
Real gas
Brayton cycle

ABSTRACT

The current work investigates dry gas seals operation with supercritical CO_2 (sCO_2) at two operating conditions using CFD. One close and one far from the critical point. At the operating condition far from the critical point a maximum change of 1.7% in pressure, 0.4% in temperature and 1.1% in density are observed. Contrary, closer to the critical point a maximum change of 6.5% in pressure, 6.7% in temperature and 39.5% in density are observed. These changes also influence opening force and leakage rate. Far from the critical point the maximum changes are 0.7% and 3.1%, whereas close to the critical point maximum changes of 3.4% and 10.3% are observed. The centrifugal effect plays an important role when operating with dense gases.

1. Introduction

Dry gas seals were introduced in the late 1960s [1]. The use of dry gas seals in high performance turbomachinery system as a new kind of noncontact mechanical seal, such as in centrifugal compressors, turbine, turbojet engines, turbofans has increased dramatically over the past 20 years, replacing traditional oil film seals in most applications [2,3]. A dry gas seal performs well in terms of less friction and wear, lower power consumption, lower leakage, longer service life, improved operational safety and rotor system stability as well as reliability compared to wet seal systems [1,4–11]. In other word, despite alternative sealing technologies being available and having been installed with varying degrees of success for certain applications, they are often limited in speed, operating conditions and pressure. Moreover, with changes in the operating condition, they are prone to problems such as high leakage and relatively short periods of operation [1,12]. Therefore, today dry gas seals are the preferred choice and are used in a wide variety of industries despite their increase mechanical complexity compared to other seal types.

Although many surface groove patterns have been developed, the spiral groove geometry is still the most widely used in mechanical seal applications today [6]. For many years, research and early performance analysis used narrow spiral groove theory, proposed by Vohr and Pan [7], Muijderman [13], Malanoski and Pan [14] and Smalley [15]. However, this theory only provides a one-dimensional solution, while ignoring the complicated geometrical boundary shape [8]. With the development of computers, more accurate numerical models have been employed to analyse performance characteristics. Numerical treatments based on the Reynolds equation that utilize the finite difference method, finite element method, and boundary element method have been employed [5,16]. Recently, Thatte and Zheng [9] used Reynolds equation for high pressure CO_2 compressible flow and implement an iterative solution procedure to solve the governing equations using Matlab. However, the solutions of the Reynolds equation supply only two-dimensional results.

The solutions of both the narrow spiral groove theory and the Reynolds equation are useful for performance estimations and preliminary designs, but are not accurate in elucidating the three dimensional flow dynamics in the gap between two seal faces [4]. Nor can it capture non-traditional velocity profiles that arise between the rotor and stator as a result of large centrifugal and inertia forces caused by the increased gas density [17,18]. Therefore Computational Fluid Dynamics (CFD), which incorporates real gas models is required to accurately predict the fluid behaviour in the dry gas seal operating with dense fluids such as supercritical CO_2.

Some studies are available in literature, which deal with three dimensional Reynolds Averaged Navier Stroke equation in elucidating the flow field in the gap between two seal faces using ideal

* Corresponding author.
E-mail addresses: zmfairuz@gmail.com (Z.M. Fairuz), i.jahn@uq.edu.au (I. Jahn).

http://dx.doi.org/10.1016/j.triboint.2016.05.038
0301-679X/

Nomenclature

a	flow factor
A	area, Helmholtz free energy,
F_o, F_c, F_p	opening force, centrifugal force and pressure force
h, h_g	film thickness or operating gap, groove depth
H	enthalpy
n	groove number
P	pressure
P_g	pressure at the end of the groove's wall
$\dot{m}$	leakage rate, mass flow rate
r, r_i , r_o	radius, inner and outer radius
Re_c, Re_p	radial Reynolds number, tangential Reynolds number
T	temperature
T_θ	friction torque or windage loss
U_r	velocity due to the moving frame, whirl velocity magnitude
$\boldsymbol{V}$, $\boldsymbol{V_r}$, $\boldsymbol{U_r}$	absolute velocity, relative velocity and whirl velocity vectors
V_r	relative velocity magnitude
y	vertical distance to nearest wall
y^+	non-dimensional wall distance for a wall-bounded flow
Z	compressibility factor
α	spiral angle
β	ratio of groove to land
θ	angle
ℓ_l, ℓ_g	land arc length, groove arc length
μ	dynamic viscosity
ρ	density
$\boldsymbol{\tau}$	stress tensor
τ_w	wall shear stress
ϕ	flow coefficient, reduced Helmholtz free energy
ϕ^o, ϕ^r	flow coefficient for the ideal gas contribution
ω, $\boldsymbol{\omega}$	rotational speed, angular velocity vector.

gas model [4,8,10,11,18]. Heshun et al. [8] investigate the distribution of the hydrodynamic gas film pressure, opening forces and leakage rate in the gap of the spiral grooved dry gas seals at different face clearances. Shahin et al. [10] studied gas seal performances using both spiral and different herringbone shape groove configurations at different rotational speed in forward and reverse rotation conditions. Jing et al. [19] studies the flow on spiral groove dry gas seal in the laminar and turbulent flow regime. Shahin et al. [11] performed studies with constant depth grooves and with different taper grooves. Bing et al. [4] used both direct numerical simulation (DNS) and Reynolds-averaged Navier-Stokes (RANS) to study the three dimensional flow dynamics in the spiral-groove dry gas seal. Hong et al. [6] performed comprehensive analysis to resolve the problem of simulating the complex conjugate heat transfer within the seal. While insightful, these works do not address performance with highly dense supercritical fluids.

The supercritical CO_2 Brayton cycle is being actively developed for potential application in a wide range of energy conversion applications [20,21]. Dry gas seal, once mature for operation with supercritical CO_2 have been recommended for these applications [21] as they are more reliable, cost efficient, safer and offer the lowest leakage of any type of seals that can be applied in the harsh supercritical CO_2 environment [1,12]. So far, no design guideline for dry gas seals are available for supercritical fluid Brayton Cycles [12,21]. One of the key challenges in designing a seal for this application are the highly non-linear fluid and thermodynamic behaviour of the working fluid within the dry gas seal. These can have a major influence on dry gas seal performance. New insight into fluid dynamics within these seals is required in order to design efficient and robust seals for operation with supercritical CO_2.

Furthermore, the studies in literature on the supercritical fluid dry gas seals are still missing a clear descriptions and characterization of how the real gas properties and high density of the supercritical fluid affect the performance and operability of dry gas seals at operating points typical for Brayton power cycle or any other turbomachinery application. The present work carries out a numerical investigation to understand how supercritical CO_2 influence performances of dry gas seals and to provide new insight into how to design dry gas seals for supercritical CO_2 sealing applications. Within the Brayton Cycle there are two possible sealing locations at either high or low operating temperatures [8,17] (turbine and compressor seals) where dry gas seal can be selected. Hence, the present study investigates two inlet conditions one close to the critical point and one far from the critical point. At both conditions, seals operating with CO_2 (real fluid) and air (ideal gas) are compared to highlight the differences.

2. Numerical model

This section describes the numerical model used to analyse the fluid mechanics in the dry gas seals operating with supercritical CO_2.

2.1. Fluid governing equations

The fluid is simulated using the Reynolds Averaged Navier Stokes (RANS) equations, applied to a rotating system. For the single rotating reference frame simulation, the governing equations based on relative velocity are given by:

- Mass conservation

$$\nabla \bullet (\rho \boldsymbol{V_r}) = 0 \tag{1}$$

- Conservation of momentum

$$\nabla \cdot (\rho \boldsymbol{V_r V_r}) + \rho (2\boldsymbol{\omega} \times \boldsymbol{V_r} + \boldsymbol{\omega} \times \boldsymbol{\omega} \times \boldsymbol{r}) = -\nabla P + \nabla \cdot \boldsymbol{\tau} \tag{2}$$

- Conservation of energy

$$\nabla \cdot (\rho H \boldsymbol{V_r}) + \nabla \cdot \left(\rho \frac{1}{2} V_r{}^2 \boldsymbol{V_r}\right) + \nabla \cdot \left(\rho \frac{1}{2} U_r{}^2 \boldsymbol{V_r}\right) = k \nabla T + \nabla \cdot (\boldsymbol{\tau} \cdot \boldsymbol{V_r}) \tag{3}$$

where $\boldsymbol{V_r}$ is defined as:

$$\boldsymbol{V_r} = \boldsymbol{V} - \boldsymbol{U_r} \tag{4}$$

$$\boldsymbol{U_r} = \boldsymbol{\omega} \times \boldsymbol{r} \tag{5}$$

The numerical simulations are performed using the CFD package Ansys Fluent version 16 [23].

2.2. Equation of state

Two equations of state are used for the simulations based on operating fluid and operating conditions. For ideal gas simulations