Tribology International 47 (2012) 134–141

Contents lists available at SciVerse ScienceDirect

Tribology International

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

Pressure dam influence on the performance of gas face seals

Marco Tulio C. Faria [a,*], Wederley Mendes Miranda [b]

[a] Department of Mechanical Engineering, Federal University of Minas Gerais, Av. Antonio Carlos, 6627, 31270-901 Belo Horizonte, MG, Brazil
[b] Graduate Program in Mechanical Engineering, Federal University of Minas Gerais, Brazil

ARTICLE INFO

Article history:
Received 31 March 2011
Received in revised form
14 October 2011
Accepted 18 October 2011
Available online 25 October 2011

Keywords:
Seals
Finite element modeling
Gas seals
Face seals

ABSTRACT

The introduction of grooves, micropores, and other forms of surface geometric modifications onto the mating seal plates has been performed to enhance the gas face seal performance. Reducing fluid leakage through the seal surface has been the main motivation for the development of efficient sealing technology for industrial turbomachinery. Pressure dams are generally etched on the seal inner and outer radii to improve even more the seal capability of reducing the gas leakage to atmosphere. This paper presents a finite element analysis carried out to determine the opening force, the flow leakage and the dynamic force coefficients of flat gas face seals with pressure dams operating under stringent conditions. Several curves of steady-state and dynamic seal performance characteristics depict the influence of the pressure dam position and width on the seal performance and efficiency.

1. Introduction

The search for more efficient and safer sealing systems for high-performance turbomachinery employed in the oil, petrochemical, chemical, and general process industries has been the main motivation for the development of new techniques and procedures for mechanical seal analysis and design [1,2]. Current environmental regulations on hazardous fluid emissions in industrial plants have cut down the admissible levels of fluid leakage into the atmosphere, forcing the seal manufacturers to invest in new sealing technologies. Within all novel and efficient sealing technologies that have been recently used for seal design, the gas sealing technology deserves to be highlighted [3,4]. The application of gas lubrication technology in end seals has permitted the development of zero emission noncontacting face seals for industrial pumps [5].

Gas lubricated face seals meet satisfactorily the requirements of low leakage rate, wear free operation and low power consumption for industrial compressors, pumps, agitators and other turbomachines [4]. Grooves, pockets, steps, and textured pores have been etched on either of the seal mating plates to improve the gas face seal performance [6–9]. All these typical gas seal plate geometries possess an unprofiled area, the sealing dam, which can be located at either the seal inner or outer radius [10]. This pressure dam is introduced into the seal face to reduce leakage and increase axial stiffness. The pressure dam location determines the seal radial flow direction. Outer dams are generally employed for outward radial flow within the seal plates and inner dams for inward radial flow.

Most of the studies performed on gas face seals deal with the analysis of the influence of the seal-mating plate patterns, which generally include various types of grooves, face waviness, radial taper, plate misalignment, and face texture, on the seal performance [2,6,7,10–13]. In the analyses of grooved face seals, the main concern has been the parametric study of the influence of the groove geometry, in conjunction with or without the sealing dam, on the seal behavior. The influence of the pressure dam geometry on the performance of gas face seals is still an important subject that has not been well exploited in the technical literature.

In order to bring some insights into the influence of the pressure dam geometry on the gas lubricated face seal behavior, this paper presents a finite element analysis of gas face seal with inner and outer pressure dams. The classical Reynolds equation for compressible fluids is the basis for the finite element (FEM) procedure developed in this work. A perturbation method is applied on the Reynolds equation to render the zeroth- and first-order lubrication equations, which are employed to estimate some steady-state and dynamic seal performance characteristics. The FEM procedure is based on the Galerkin weighted residual method [14]. The gas flow through the seal is assumed axisymmetric. In the analysis, the seal mating plates are made flat and smooth to allow the exclusive evaluation of the pressure dam geometric parameters on the seal performance. Seal performance characteristics, such as seal opening force, flow leakage, and frequency-dependent stiffness and damping force coefficients, are computed in relation to some seal parameters, such as dam

* Corresponding author. Tel.: +55 31 3409 4925; fax: +55 31 3409 4823.
E-mail addresses: mtcdf@uol.com.br (M.T.C. Faria), wederley@yahoo.com.br (W.M. Miranda).

0301-679X/$ - see front matter
doi:10.1016/j.triboint.2011.10.011

Nomenclature

CR	seal clearance ratio
C_{zz}	dimensionless axial damping
F_z	dimensionless seal opening force
h	axial sealing gap (m)
h_2	minimum sealing gap (m)
h_1	sealing gap of the unprofiled area (m)
H	dimensionless sealing gap
H_0	dimensionless sealing gap at the equilibrium position
K_{zz}	dimensionless axial stiffness
l	total seal extent (m)
l_1	seal extent of the unprofiled area (m)
l_2	dam extent (m)
L_e	finite element radial length (m)
m_e	dimensionless finite element zero-order mass flow rate
m_1^e	dimensionless finite element first-order mass flow rate
p	gas–fluid pressure (Pa)
p_i	seal inner radius pressure (Pa)
p_o	seal outer radius pressure (Pa)
P	dimensionless presssure
P_0	dimensionless zeroth-order pressure
P_1	dimensionless first-order pressure
P_i	dimensionless pressure at the seal inner radius
P_o	dimensionless pressure at the seal outer radius
Q	dimensionless flow leakage rate
q_m	dimensionless mass flow rate
r_i	seal inner radius (m)
r_s	seal groove-ridge boundary radius (m)
r_o	seal outer radius (m)
R	dimensionless radius
s	local coordinate system
SRW	seal radial width ratio
t	time (s)
Z	dimensionless seal mechanical impedance
δ	seal dam extent ratio
ΔZ	dimensionless axial perturbation
μ	fluid viscosity (Pa s)
τ	dimensionless time
σ	frequency or squeeze number
ψ_i^e	shape functions for a finite element e
ω	excitation frequency (rad/s)
Ω	rotational speed (rad/s)

extent, clearance ratio and seal width. The model validation is performed by comparing the finite element predictions with approximated analytical solutions for gas-lubricated flat mechanical face seals.

2. Gas face seal parameters and equations

To illustrate the basic geometric configuration of gas face seals, Fig. 1 shows schematic drawings of the seal plate with pressure dam. In Fig. 1, the pressure dam heights are intentionally exaggerated. The polar coordinates are represented by (r,θ). A schematic view of the geometry of a flat mechanical face seal with outer pressure dam is depicted in Fig. 2. The sealing dam can be located at either the seal inner radius or outer radius, depending on the radial pressure ratio.

The inner, groove-ridge boundary, and outer radii are given by r_i, r_s, and r_o, respectively. The pressure dam extent is represented by l_2 and the seal extent by l_1. The seal length is $l=l_1+l_2$. The sealing axial gaps at the dam and at the seal are expressed by h_2 and h_1, respectively. Generally the stepped face is rotating, while the flat face is non-rotating.

The lubricant flow in the face seal is described by the Reynolds equation for an isothermal, isoviscous, ideal gas [11]

$$\frac{1}{r}\frac{\partial}{\partial\theta}\left(\frac{ph^3}{12\mu}\frac{1}{r}\frac{\partial p}{\partial\theta}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{ph^3}{12\mu}\frac{\partial p}{\partial r}\right)=\frac{\Omega r}{2}\frac{\partial}{\partial\theta}(ph)+\frac{\partial}{\partial t}(ph) \quad (1)$$

where p describes the pressure field over the seal, h is the sealing gap, μ is the fluid viscosity, and Ω is the rotational speed of the

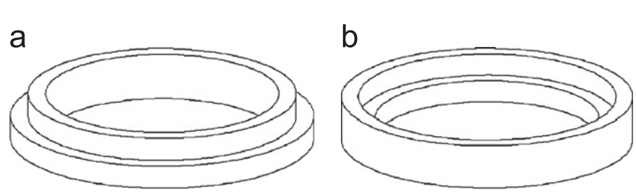

Fig. 1. Schematic drawings of seal plates with pressure dam: (a) inner seal dam and (b) outer seal dam.

seal. The boundary conditions for the system are given by $p(r_i)=p_i$ and $p(r_o)=p_o$. The pressures p_i and p_o are the inner and the outer pressures acting on the inner and the outer seal radii, respectively.

Due to the axi-symmetric nature of the gas flow, the pressure field is independent of the circumferential coordinate θ. Thus, Eq. (1) simplifies to the following form:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{ph^3}{12\mu}\frac{\partial p}{\partial r}\right)=\frac{\partial}{\partial t}(ph) \quad (2)$$

A dimensionless form of Eq. (2) is expressed as

$$\frac{1}{R}\frac{\partial}{\partial R}\left(RH^3P\frac{\partial P}{\partial R}\right)=\sigma\frac{\partial}{\partial\tau}(PH) \quad (3)$$

where the dimensionless variables are given by $R=r/r_o$, $P=p/p_a$, $H=h/h_{min}$, $\tau=t\omega$, and $\sigma=(12\mu\omega/p_a)(r_o/h_2)^2$. In these variables, ω is the axial excitation frequency, p_a is the ambient pressure, and h_{min} is the minimum value of the seal gap. σ is the frequency or squeeze number.

Seal back springs are generally used to close the gap between the mating plates in order to restrict the fluid passage. These back springs are used to compensate the hydrodynamic and hydrostatic action of the process fluid within the gap, which tends to separate the seal mating plates.

3. Seal governing equations

A linear perturbation procedure is applied in Eq. (3) to render the zeroth- and first-order seal lubrication equations. The rotating face seal is usually subjected to small axial perturbations from an equilibrium position. The effect of these small axial motions is to cause small perturbations in the pressure field. The expressions for the dimensionless perturbed axial gap and pressure field generated by dimensionless axial perturbations ΔZ are given in the following form:

$$H(R,\tau)=H_0(R)+\Delta Ze^{i\tau} \quad (4)$$