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Thermo-hydrodynamic characteristics of spiral groove gas face seals operating at low pressure

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ABSTRACT

Thermo-hydrodynamic behaviors of spiral groove gas face seals are investigated for the cases of low seal pressure. Pressure and temperature fields of gas film are calculated, and then influence of thermal distortion on seal performance are analyzed. It is found that spiral grooves lead to complex film temperature distributions and rotational speed results in the increase of whole film temperature. But the shear heat and pumping effect of spiral grooves have little influence on film temperature gradient along the leakage direction. The face thermal distortion, forming divergent clearance along the leakage direction, makes the opening force decrease and the leakage increase. A higher seal pressure and clearance will result in a larger thermal distortion. And the rotational speed may enhance the face thermal distortion further in some low pressure cases.

1. Introduction

Spiral groove gas face seal has been widely applied in centrifugal compressor, gas turbines and other rotating mechanical equipments, which design generally lies on the isothermal gas lubrication theory. However, with the increase of operational speed of mechanical equipments, thermal hydrodynamic and face distortion problems have been obvious, which is complex due to combination of both expansion heat and shear heat.

Pan and Sternlicht [1], Kapur and Yadav [2], Boncompain et al. [3], and Sahu et al. [4] found the friction heat generated by fluid film could distort the surface and cause a considerable degradation in load-bearing capacity on gas lubricated thrust bearing. Experimental researches by Ohishi and Matsuzaki [5] on aerostatic journal bearing show that the gas film temperature increased with the increase of rotational speed, and temperature increased about 30 °C under a maximum rotational speed of 20,000 r/min, radial clearance 20 μm, pressure 0.7 MPa with a 60 mm diameter spindle. Besides, Salehi et al. [6,7] presented the thermal characteristics of compliant foil bearing under different bearing capacity and rotational speed in theoretical and experimental ways. Further, Lee and Kim [8], San Andrés and Kim [9], and Kim and San Andrés [10] found that the rotor temperature increased with the increase of bearing capacity, rotational speed and the decrease of foil bearing clearance. In relative research works of liquid lubrication, thermal problems have been also discussed. Recently, Habchi [11] found that friction might be controlled in TEHD contacts by a suitable choice of surface coating based on thermal properties. Tala-Ighil and Fillon [12] pointed out that considering the temperature effect is more realistic for journal bearings. Migout et al. [13] numerically investigated the effect of an increase in feeding temperature on a mechanical seal operating with water, and it was illustrated that above a temperature threshold, the mechanical seal could exhibit unstable behavior.

Thermal problems of gas face seal are more complex than bearing, since there is not only shear flow induced by rotational speed but also pressure flow resulted from seal pressure. The thermal effects induced by gas pressure flow on coning angle static mechanical face seals leads to significant face distortions, which forms divergent clearance and decreases film load capability under high seal pressure conditions shown in Thomas's theoretical works [14,15]. However, the isobaric expansion coefficient of the perfect gas was used, and no equation for the relation between density and temperature was provided in the previous works.

To obtain more detail of gas film temperature in gas lubrication, a gas thermo-hydrodynamic theoretical model based on the energy equipartition principle was developed in our previous work [16]. Bai et al. [17] illustrates that thermal distortion has the same influence degree as the elastic distortion on the sealing performance of high pressure spiral groove gas face seals based on this method. Further, it has been found that spiral grooves may lead to complex temperature distribution and an obvious increase in film loading capability of thrust bearing [16] where shear flow play a main role. That is to say, face distortion can be ignored and function of spiral groove becomes significant under low seal pressure. Hence, the thermo hydrodynamic

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Nomenclature

c_p gas pressure coefficient, J mol^{-1} K^{-1}
C_p isobaric specific heat capacity, 29.1 J mol^{-1} K^{-1}
c_v specific heat at constant volume, J mol^{-1} K^{-1}
c_{s2} specific heat, J kg^{-1} K^{-1}
E_m energy of gas molecular per freedom, J/mol
F_o opening force, N
h clearance, m
h_{Deform} variation of clearance reduced from seal face deformation, m
h_{min} minimum clearance, m
h_1,h_2 seal ring thickness, m
h_p groove depth, m
i_d degree of seal gas molecular freedom
k_{c1}, k_{c2} thermal conductivity respectively of stator and rotor, W m^{-1} K^{-1}
k_{c_gas} gas thermal conductivity, W m^{-1} K^{-1}
k_{s1},k_{s2},k_{gs1}, k_{gs2} convection heat transfer coefficient, W m^{-2} K^{-1}
M_{inlet}, M_{exit} Mach number at inlet and exit of the seal, respectively
M_1 Mach number at some point in lubrication region near exit
N groove number
p gas pressure in lubrication regime, Pa
p_a standard atmospheric pressure, 1.01325 × 10^5 Pa
pi ambient pressure, Pa
p_o sealed gas pressure, Pa
p_{inlet}, p_{exit} gas pressure at inlet and exit, respectively, Pa
p_1, p_2 gas pressure at some point in lubrication region near exit and inlet, respectively, Pa
q_r flux in the cross-section of seal face, kg/s
Q leakage rate, kg/s
Q_r Prandtl number
r, z coordinates, m
r_b balance radius, m
r_i inside radius, m
r_o outside radius, m
r_p spiral radius, m
R_0 universal ideal gas constant, 8.314510 J mol^{-1} K^{-1}
T lubricating film temperature, K
T_o ambient temperature, K
T_i sealed gas temperature, K
T_{inlet} inlet temperature, K
T_s solid temperature, K
T_{s1}, T_{s2} solid surface temperature, K
β spiral angle, °
θ coordinate, rad
η bulk viscosity, Pa s
ρ gas density, mol/m^3
ρ_{inlet}, ρ_{exit} gas density at inlet and exit, respectively, mol/m^3
ρ_{s1}, ρ_{s2} ring density, kg/m^3
ρ_1, ρ_2 gas density at some point in lubrication region near exit and inlet, respectively, mol/m^3
ω rotational speed, rad/s

effect in this condition is expected to be studied, to get a better predicts of the sealing performances.

In this paper, the distribution of film temperature and the performances of spiral groove gas face seals operating at low seal pressure are investigated. Numerical model is built considering thermal and pressure boundary conditions. Then, the influences of face spiral groove on the pressure and temperature distributions are discussed. Finally, thermal distortions under different pressure, rotational speed and seal clearance are studied.

2. Theoretical model

The theoretical model has been developed and described previously [17]. It can be summarized as follows.

2.1. Governing equations

Fig. 1 presents a schematic of face seal structures, which consists of a smooth ring and a spiral grooved one, and thermal

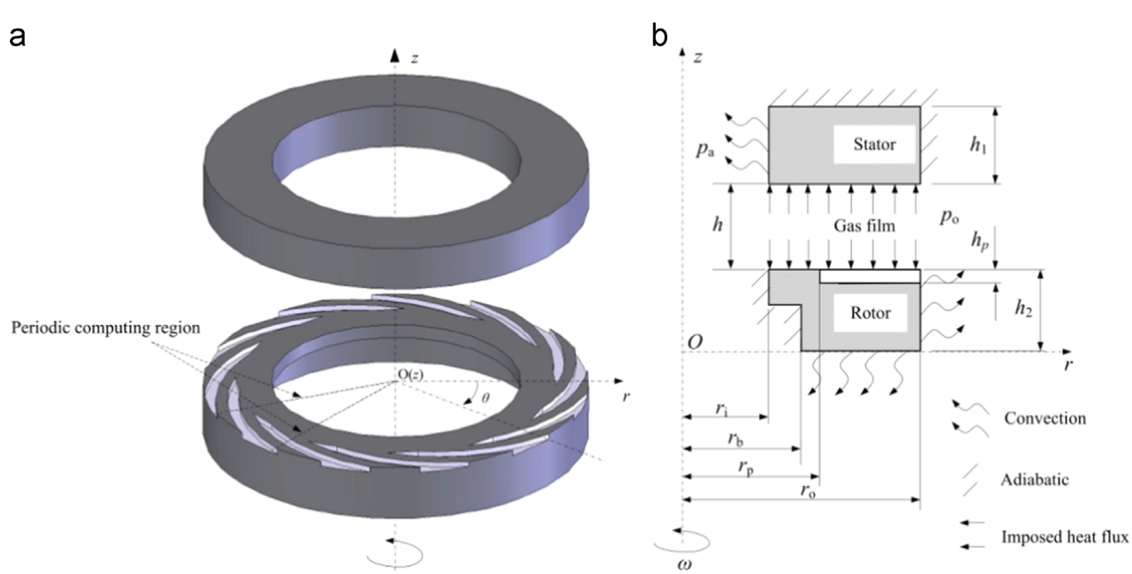

Fig. 1. Schematic of spiral grooved face seal and thermal boundary conditions.