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Heat transfer correlations for laminar flows within a mechanical seal chamber

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Numerical investigation of conjugate heat transfer associated with laminar flow within the chamber of a mechanical seal is presented. It involves simultaneous solution of the Navier–Stokes and energy equations. The computational model takes into account the temperature distribution within the rotating and stationary rings. A series of simulation results are presented for predicting the performance of a mechanical seal assuming that the flow in the seal chamber remains laminar. Expressions are developed for predicting the convective heat transfer coefficient on the outer surfaces of the seal rings exposed to the process fluid in the seal chamber.

1. Introduction

The frictional heat generation at the interface of the rotating and stationary rings can significantly affect the mechanical seal's performance and reliability. It is, therefore, not surprising that many researchers have applied advanced experimental and analytical tools to study the nature of heat transfer in seals. For example, Li [1] assumed a constant heat input in the sealing interfaces for treating the energy equation with uniform fluid properties using the finite element method. Morariu and Pascovici [2] considered heat conduction in the seal ring, and convection by the coolant around the rings. Buck [3] provided a simplified approach for determining the seal temperature based on an analytical model that treats the seal as a fin. This method was recently extended to take into account the heat partitioning between the two seal rings [4].

In another paper, Buck [5] presented a method for estimating heat generation and temperature field using representative values of the convective heat transfer coefficient at the outer surface of the seal rings. Doane et al. [6] performed experimental measurements that primarily focused on determining the boundary conditions for the numerical computations. They obtained local and average Nusselt number for the wetted area of the stationary ring and reported measurements of the interfacial temperature distribution between the rings.

Lebeck [7] considered many of the important effects caused by the thermal environment such as thermally induced radial taper, thermally induced waviness, heat checking and hot spotting and blistering. Jang and Khonsari [8] extended the theory of thermoelastic analysis to predict the critical speed at which hot spots can occur on the surface of a seal. Salant and Cao [9] recently developed an unsteady numerical model of a mechanical seal and performed a thermal analysis using Duhamel's method to predict the performance of a mechanical seal during startup and shutdown. These publications concentrated on the thermal analysis of mechanical seals.

Research by Merati et al. [10] provided information on the turbulence flow field in a seal chamber by applying FLUENT, a commercially available software package. In their research, the temperature distribution was predicted within the stationary ring of a mechanical seal. Clark and Azibert [11] also used FLUENT to simulate the turbulence flow field in the barrier fluid domain of a dual seal in a centrifugal pump. By visualizing the flow field and performing thermal analysis of the seal rings, they proposed design features to enhance performance. More recently, Luan and Khonsari [12,13] numerically solved the laminar and turbulent flow within a seal chamber.

In the present study, the CFD analysis of laminar flows is extended to include a heat transfer analysis in mechanical seals. To this end, the appropriate governing equations are derived and numerical solution algorithms are developed to solve the Navier–Stokes (N–S) equations and the energy equation. The flow and temperature fields within the seal chamber as well as the rotating and the stationary rings are predicted simultaneously. Also, a mathematical model of heat generation at the interface

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Nomenclature

A_f	surface area of friction at seal contact face
b	mechanical seal balance ratio
c_f	constant-pressure specific heat of process fluid
c_m	constant-pressure specific heat of stationary ring
c_r	constant-pressure specific heat of rotating ring
D_p	outer diameter of the rotating ring
D_m	outer diameter of the stationary ring
D_{flush}	diameter of flush hole
E_p	heat generation along the interface between seal rings
f	friction coefficient
h	local heat transfer coefficient
$\bar{h}$	average heat transfer coefficient
H	heat generation distribution along the interface between seal rings
k_f	conductivity of water
k_m	conductivity of stationary ring
k_r	conductivity of rotating ring
Nu	local Nusselt number
$\overline{Nu}$	average Nusselt number
P	local pressure at seal contact face
P_m	average pressure at seal contact face
P_{sp}	spring pressure
q	heat flux at wetted outer surface of rotating ring and stationary ring
Q_{flush}	flow rate of flush flow
r	radial coordinate
Re_{Dp}	Reynolds number of rotating ring
Re_{Dm}	Reynolds number of stationary ring
t	time
$T(r,z)$	cross-sectional temperature in rotating ring, stationary ring and flow
T_s	temperature at outer surface of rotating ring and stationary ring
T_∞	ambient temperature
u_r	radial velocity component of seal chamber flow
u_z	axial velocity component of seal chamber flow
u_{rin}	flush in velocity
u_θ	angular velocity component
$u_{\theta in}$	rotating speed of rotating ring
V	local velocity of rotating ring
z	axial coordinate
ΔT_{AV}	average temperature difference
Δp	pressure differential
β	exponential coefficient of the kinematic viscosity function
ρ_f	density of process fluid
ρ_m	density of stationary ring
ρ_r	density of rotating ring
v_f	kinematic viscosity of process fluid
ω	shaft speed

between rotating and stationary rings—where the largest magnitude of the heat flux is known to occur—is developed. The results of this paper are restricted to laminar flows in the seal chamber that are known to occur in many applications, especially those that deal with high-viscosity process fluids such as oils. For these fluids, the viscosity tends to drop appreciably with the rise in temperature.

2. Governing equations

Fig. 1 shows the schematic of a mechanical seal. The installment includes a gland, rotating and stationary rings. The flush fluid enters into the seal chamber radially through the flush inlet port drilled into the gland. Fig. 2 shows the computation domain of interest. Making use of the axis-symmetric nature of the geometry and assuming laminar flow, the governing equations in cylindrical coordinates reduce to the following:

2.1. Conservation of mass

$$\frac{u_r}{r}+\frac{\partial u_r}{\partial r}+\frac{\partial u_z}{\partial z}=0 \tag{1}$$

2.2. Navier–Stokes equations

$$\frac{\partial u_r}{\partial t}+u_r\frac{\partial u_r}{\partial r}-\frac{u_\theta^2}{r}+u_z\frac{\partial u_r}{\partial z} = -\frac{1}{\rho_f}\frac{\partial p}{\partial r}+v_f\left(\frac{\partial^2 u_r}{\partial r^2}+\frac{1}{r}\frac{\partial u_r}{\partial r}+\frac{\partial^2 u_r}{\partial z^2}-\frac{u_r}{r^2}\right) \tag{2}$$

$$\frac{\partial u_z}{\partial t}+u_r\frac{\partial u_z}{\partial r}+u_z\frac{\partial u_z}{\partial z}=-\frac{1}{\rho_f}\frac{\partial p}{\partial z}+v_f\left(\frac{\partial^2 u_z}{\partial r^2}+\frac{1}{r}\frac{\partial u_z}{\partial r}+\frac{\partial^2 u_z}{\partial z^2}\right) \tag{3}$$

$$\frac{\partial u_\theta}{\partial t}+u_r\frac{\partial u_\theta}{\partial r}+\frac{u_r u_\theta}{r}+u_z\frac{\partial u_\theta}{\partial z}=v_f\left(\frac{\partial^2 u_\theta}{\partial r^2}+\frac{1}{r}\frac{\partial u_\theta}{\partial r}+\frac{\partial^2 u_\theta}{\partial z^2}-\frac{u_\theta}{r^2}\right) \tag{4}$$

The flow boundary conditions for the N–S equations are [12]

At the inlet boundary I1

$$u_z=0,\quad u_r=u_{rin},\quad u_\theta=0 \tag{5}$$

$$\frac{\partial p}{\partial r}=0 \tag{6}$$

where u_{rin} is the radial velocity of the flush fluid.

At the outlet boundary O1

$$\frac{\partial u_z}{\partial z}=0,\quad \frac{\partial u_r}{\partial z}=0,\quad \frac{\partial u_\theta}{\partial z}=0 \tag{7}$$

$$p=0.0 \tag{8}$$

At the surface of the stationary ring and the gland

$$u_z=0,\quad u_r=0,\quad u_\theta=0 \tag{9}$$

$$\frac{\partial p}{\partial n}=0 \tag{10}$$

where n denotes the normal to the surface, i.e., r or z.

At the surface of the rotating ring

$$u_z=0,\quad u_r=0,\quad u_\theta=u_{cin} \tag{11}$$

$$\frac{\partial p}{\partial r}=0 \tag{12}$$