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Performance characteristics of viscoseals in laminar and turbulent flow regimes

Dominique Souchet, Mohamed Jarray*, Aurelian Fatu

Pprime Institute, University of Poitiers, France

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ABSTRACT

In this paper, a numerical study of a viscoseal operating behavior, under laminar and turbulent regime, is presented. The finite element method is used under different hydrodynamic regimes associated to lubrication assumptions. The theoretical model, based on the classical Reynolds equation uses the approach of average value of turbulent coefficient. In order to include the film rupture and reformation, the Reynolds equation is modified using a mass conservative algorithm. A transition zone between laminar and turbulent flow is modelled.

A comparison is done between numerical and experimental results on different seals geometries and the overall comparisons are good. Next, the proposed model is used to predict viscoseal behaviors in terms of friction and seal capability for different operating conditions.

1. Introduction

The viscoseal is a hydrodynamic sealing device used in different industrial fields such as compressors and pumps. It consists, mainly, of two elements: the housing and the shaft. The sealing function is ensured by grooves designed on one or two elements. Generally, a thin fluid film or a small radial clearance is created in the gap between the two elements.

The operating principle of viscoseals consists in compensating the leakage towards one direction by hydro-dynamically generating a reverse pumping flow toward the opposite direction. This is ensured by the relative motion of a grooved surface over an ungrooved one. The viscoseal is considered as a non-contacting dynamic sealing device that have multiple advantageous characteristics as compactness, zero leakage, easy manufacturing and assembling, low wear rate and long operational lifetime.

The interest in this type of geometry started in 1928, where Rowell and Finlayson [1] conducted a theoretical analysis on screw viscosity pumps and developed an analytical model to calculate the flow in the screw pumps under laminar regime. Rowell & Finlayson neglected the effect of the groove angle and the leakage flow across the lands.

In 1949, Whipple [2] conducted the first theoretical analysis on a spiral-grooved journal bearing. He found that the pressure gradient is constant along the groove direction and that the pressure field shows a saw tooth shape in the velocity direction.

In 1959, Boon and Tal [3] published a very interesting study about the viscoseal performances in laminar regime. By supposing nil flow conditions, they proposed an analytical model able to predict the wetted area in the seal and expressed the results in terms of "sealing coefficients". Subsequent analysis was conducted by Stair [4] that detailed and used the Boon & Tal model.

Almost at the same period, Frössel [5,6] has published a series of experimental results obtained for different grooved geometries. He has especially investigated the influence of the groove angle direction on the pressure field and leakage.

A bibliographic study of previous analytic models [2,3,7,8] has been published later by Stair [9] and Wilkerson and Milligan [10]. Both studies confirm that, in laminar flow regime, Boon and Tal [3] model shows a very good agreement with the experimental results.

In a previous paper, Targaoui et al. [11] investigated the viscoseal performances in laminar regime. The authors used a mass conserving algorithm for separation theory and based there analysis on the sealing length i.e. the wetted area in the seal.

It should be mentioning that, for very high speeds or large dimensions of the sealing device, the flow is no longer laminar and the effects on the viscoseal performances could be important. Several studies [12,13], relate the different modelling of the turbulent flow in lubrication devices, from the simplest to the most complex ones. However, only a few experimental and theoretical analyses can be found in the existing literature. In 1970, Luttrull [14] studied the effect of different geometrical parameters on the sealing performance of viscoseals operating in different flow regimes: from laminar to fully turbulent. Three different seals were tested and the results clearly show a different behavior between the laminar and the turbulent regimes.

* Corresponding author.

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Nomenclature

a	ridge width (mm)
b	groove width (mm)
C	Clearance (mm)
D	Viscoseal diameter (mm)
F	Switch function
E	Universal variable
h	Film thickness (mm)
h_0	Groove depth (mm)
k_x, k_z	Turbulent coefficients
L_s	Length of the seal (mm)
L	Sealing length (mm)
R	Viscoseal Radius (mm)
$Re_h = \frac{\rho\omega Rh}{\mu}$	Reynolds number related to the local film thickness
$Re_c = \frac{\rho\omega RC}{\mu}$	Reynolds number related to the clearance
p	Pressure (MPa)
p_s	Sealing pressure (MPa)
U	Velocity (m/s)
x, y and z	Cartesian coordinate viscosity

Greek symbols

β	Groove angle (°)
ρ	Lubricant density (kg/m^3)
ρ_0	Density of fluid at the atmospheric pressure (kg/m^3)
$\chi = \frac{b}{a+b}$	Groove density
μ	Lubricant viscosity ($MPa\ s$)
Δ	Sealing coefficient
ω	Journal speed (rpm)

Acronyms

MRE	Modified Reynolds Equation

Moreover, the results clearly show a transition laminar/turbulent zone.

The purpose of the current paper is to analyze the effect of turbulent flow on the performance characteristics of different seals. The proposed approach can be considered as an extension to turbulent regimes of the study described in Ref. [11].

2. Theoretical analyses

The presented study uses the model established by Constantinescu et al. [15] that is derived from Ng and Pan [16] and is based on the average velocities. In this analysis, thermal effects are not taken into account and the flow is considered isothermal.

2.1. Geometry

In Fig. 1, the geometry of the studied viscoseal is presented. D is the diameter of the seal, h_0 the groove depth and β the groove angle. The groove density is defined by : $\chi = \frac{b}{a+b}$. One can say that $\chi = 0$ and $\chi = 1$ represent, respectively, a journal bearing with a radial clearance equal to C and (C+h_0). ω represents the journal rotational frequency.

2.2. Modified Reynolds equation

For convenience, all mathematical expressions are deduced in Cartesian coordinates. As presented in the partial cut plotted in Fig. 1, the shaft and the housing are unwrapped in the (x, z) plane. Thin film theory is considered and the inertia effects are neglected [17]. Consequently, for an incompressible viscous and Newtonian flow, the Reynolds equation is expressed in steady-state by:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{k_x}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{k_z}\frac{\partial p}{\partial z}\right) = \omega R\frac{\partial h}{\partial x} \tag{1}$$

where p and h represent the pressure and the film thickness, respectively, ω the journal speed, k_x and k_z are the frictional-flow rate parameters (Eq. (2)) in the x and z directions determined by Constantinescu [15]. They are independent of the pressure gradient. The effect of the pressure induced flow components is neglected on the turbulent structure.

$$k_x = 12 + 0.0136Re_h^{0.9} \tag{2}$$

$$k_z = 12 + 0.00432Re_h^{0.96},$$

where Re_h represents the local Reynolds number related to the film thickness.

Due to the sudden variation of the film thickness generated by the presence of grooves, cavitation phenomena can appear and highly influence the flow predictions of a visocseal. Therefore, Eq. (1) must be modified in order to take into account film rupture without losing the mass conserving. Based on the model presented in Ref. [18], a Modified Reynolds Equation (MRE) is proposed here to predict the viscoseal behavior in both laminar and turbulent flow regimes:

$$F\frac{\partial}{\partial x}\left(\frac{h^3}{k_x}\frac{\partial E}{\partial x}\right) + F\frac{\partial}{\partial z}\left(\frac{h^3}{k_z}\frac{\partial E}{\partial z}\right) = \mu\frac{\omega R}{2}\frac{\partial}{\partial x}(h + (1-F)E) \tag{3}$$

where E is a universal variable, μ the dynamic viscosity, and F a switch function that identifies the active and the non-active zone, such as:

$$\text{Active zone:}\begin{cases} E = p \\ F = 1 \end{cases} \tag{4}$$

$$\text{Non}-\text{active zone:}\begin{cases} E = r - h \\ F = 0 \end{cases} \quad \text{where: } r = \frac{\rho}{\rho_0}h \tag{5}$$

where ρ_0 and ρ denote, respectively, the lubricant density and the lubricant–gas mixture density and r represents the effective film thickness. Even if the solution of Eq. (3) can be found in previously published works [18], for completeness purpose, it will be briefly addressed in the next paragraph.

2.3. Numerical solution

The problem is discretized using four nodes quadratic linear elements. The finite element method has been chosen because it can be easily applied to film thickness discontinuities generated by the spiral grooves. The integral form of Eq. (3) defined on the domain Ω of the lubricant film is:

$$\int_\Omega\left(F\left[\frac{h^3}{k_x}\frac{\partial W}{\partial x}\left(\frac{\partial E}{\partial x}\right) + \frac{h^3}{kz}\frac{\partial W}{\partial z}\left(\frac{\partial E}{\partial z}\right)\right] - \mu\frac{\omega R}{2}\frac{\partial W}{\partial x}[h + (1 - F)E]\right)d\Omega \tag{6}$$

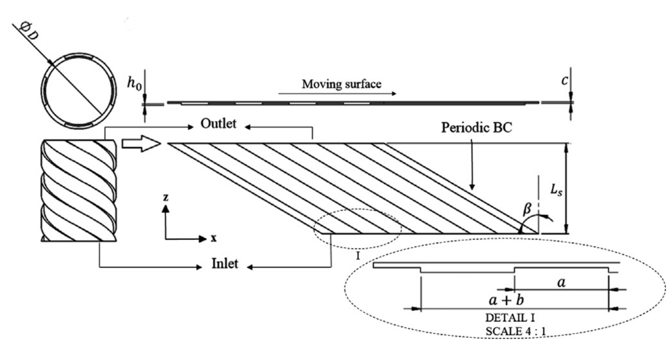

Fig. 1. Schematic representation of the studied viscoseal.