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Numerical analysis on a new pump-out hydrodynamic mechanical seal

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ABSTRACT

A new pump-out hydrodynamic mechanical seal with self-pumping effect is presented based on centrifugal pump's working principle. Results of parametric study show that both of the opening force and leakage rate increase with increasing medium pressure, channel diameter, spiral angle and width ratio of groove to ridge; The opening force increases with decreasing groove number and length ratio of groove to dam while the leakage rate decreases slightly; There exits an optimum groove depth that obtains higher opening force along with lower leakage rate. Moreover, the new mechanical seal can be used in conditions with larger pressure difference.

1. Introduction

Non-contact mechanical seal has been developed quickly since John Cline Company proposed gas seal with spiral grooves in 1970s. Nowadays, as its typical forms, the dry gas seal and upstream pumping mechanical seal are wildly used in industries of petroleum, chemical engineering, electric power, and metallurgical to serve devices like centrifugal compressor, centrifugal pump, etc. To improve their sealing performance, many theoretical and experimental studies have been carried out that concentrate on the optimization of groove shape, size and arrangement [1–8]. Wang et al. [6], for instance, analyzed the sealing property of upstream pumping mechanical seal with five types of spiral grooves under different working conditions using FVM method. They indicated that the performances of circumferential divergent 2-ladder different deep spiral groove are better than others, with larger opening force and better stabilization, while with the same leakage. A parametric study of spiral groove gas face seals has also been conducted by Zirkelback [8], the recommended geometric parameters including groove number, spiral groove angle, groove width ratio, and seal dam extent were presented to ensure large static stiffness and damping force coefficients while still allowing for low leakage rates.

It is well accepted that both of the dry gas seal and upstream pumping mechanical seal make use of the relative rotation of seal rings to pump the sealing medium into the grooves and the hydrodynamic wedge can be formed. The opening force will be produced in the root of the grooves to reduce the wear and prolong the service life of seal faces. On the other hand, the clearance of the seal rings increases due to the produced opening force and this leads to the undesirable increase of leakage rate. In particular, for the upstream pumping mechanical seal, the medium in sealing chamber will probably be polluted by the buffer fluid from lower pressure side. Moreover, the sealing dam can be damaged easily if there are particles in the medium and the mechanical seal will lose its effectiveness rapidly. Therefore, to ease the case that a large opening force corresponds to a large leakage rate, some new structures of end face of mechanical seal are proposed, such as the US patent '*face seal with double spiral grooves*' [9], China patents '*end face seal of double loop with spiral groove*' [10] and '*self-lubricating non-contact mechanical seal with double rows of fluid dynamic pressure groove*' [11]. They adopted the concept of double-row spiral grooves that one row is applied to pump the medium to downstream while the other one to upstream. The medium pressure difference between inner and outer diameter of the seal ring can be balanced by the pumping pressure difference between the two rows. However, such seals tend to have more complicated structure, need larger installation space and only be applied under the operating conditions with relative lower medium pressure difference between inner and outer diameter of the seal ring.

Thus, to simplify structure and reduce installation space, while maintaining sealing property, a new pump-out hydrodynamic mechanical seal with self-pumping effect was put forward in authors' patent [12]. In this paper, numerical analysis is conducted to prove the effectiveness of the seal and to investigate the influences of geometric and operating parameters on sealing properties, including the opening force and leakage rate. The results could be the theoretical basis for further study and industrial application of the seal.

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Nomenclature

b	width of seal face [m]
d	circular channel diameter [m]
e	the base of natural logarithms, ≈2.718g gravitational acceleration [m s^{-2}]
h_0	sealing clearance [μm]
h	local seal clearance [m]
h_g	groove depth [μm]
H	energy head [J kg^{-1}]
K	elasticitycirculation coefficient
l	width of sealing dam [m]
μ	dynamic viscosity of sealed medium [Pa s]
n	spindle speed [rpm]
N	shaft power [W]
N_g	groove number
Δp	pressure difference of inner and outer radius of the seal face [Pa]
Δp_g	pressure difference of the two sides of the sealing dam [Pa]
p_i	inlet pressure [Pa]
p_o	medium pressure [Pa]
q	mass flow rate [kg s^{-1}]
q_L	leakage rate [m^3 s^{-1}]
Q	medium flow rate [m^3 h^{-1}]
r_g	root radius of the groove [m]
r	radius of the groove [m]
r_i, r_o	inner and outer radius of the rotating ring [m]
α	helix angle [rad]
γ	length ratio of groove to dam
θ	rotation angle [rad]
δ	width ratio of groove to ridge
ρ	density of sealing medium [kg m^{-3}]
η	pumping efficiency, $\eta=\eta_m\times\eta_h\times\eta_v$
η_m, η_h, η_v	mechanical efficiency, hydraulic efficiency and volumetric efficiency, respectively

2. End face structure and working principle of the new pump-out hydrodynamic mechanical seal

2.1. End face structure

The end face structure for rotating ring of the new pump-out hydrodynamic mechanical seal is shown schematically in Fig. 1. As can be seen, the end face is composed of two regions, named as groove region and sealing dam region, which are located in outside and inside of the face, respectively. The groove region contains spiral grooves, including two logarithmic spirals and one connection arc for each groove, and sealing weirs, defined as the sealing faces between adjacent grooves. The expression of the logarithmic spiral for groove can be written in polar coordinates [13]

$$r = r_g e^{\theta \tan\alpha} \tag{1}$$

The outlet and inlet of fluid are located at the outer diameter of the sealing face and the axial circular channels of rotating ring, respectively. The axial channels are elongated channels started from the root of spiral groove to the sealing chamber and their centers are coincident with the arc centers situated at the top of the grooves.

2.2. Working principle

As shown in Fig. 2, working face is defined as the spiral groove side that accelerate the sealing medium to a high speed, and the other side is defined as non-working face accordingly. Due to the centrifugal effects, the medium will flow to the outer diameter of the rotation ring along the tangential direction of working face and be pumped into sealing chamber. In other words, rather than is pumped into the groove as traditional seals, the medium is pumped out from the groove. During the process of flowing from the root of groove to outlet, the speed of the medium decreases due to the increase of groove cross-section area and thus the pressure energy increases, which is beneficial to separate the seal faces. Then the medium can be sealed effectively because of the high pressure area built by grooves and the resistance brought by sealing dam. On the other hand, lower pressure zone will be formed in groove root when the medium flows out and, as the existence of pressure difference, the medium in sealing chamber will be sucked into the groove through the circular channel. The medium will then be accelerated to a high speed by working face and pumped into sealing chamber as before. The cycle continues as the rotating of the seal ring, where the medium is pumped in and out sequentially in a spontaneous way. Thus the "self-pumping" effect is presented for the new pump-out hydrodynamic mechanical seal [14]. Similar to centrifugal pump, the relationship between the energy head of the new pump-out hydrodynamic mechanical seal, including fluid pressure energy and kinetic energy, and the flow rate of medium in the groove can be expressed as [15]

$$H = \frac{N\eta K}{Q\rho g} \tag{2}$$

Note that K is always less than 1 and approximate to 1 when the groove number has an infinite amount.

And likewise, the relationship between leakage rate and the pressure difference of the seal sides can be given by [16]

$$q_L = \frac{\pi\delta h_0^3 r_o \Delta p}{6\mu b} + \frac{\pi(1-\delta)h_0^3 r_g \Delta p_g}{6\mu l} \tag{3}$$

3. Numerical analysis model

3.1. Basic assumptions

The calculation of flow field is very complicated for hydrodynamic mechanical seal [17]. To simplify the computation while taking into account the seal ring structure and the intrinsic property of the sealing system, some assumptions are made as follows:

(1) The medium is taken as the continuous fluid, whose temperature and viscosity remain unchanged for the study;

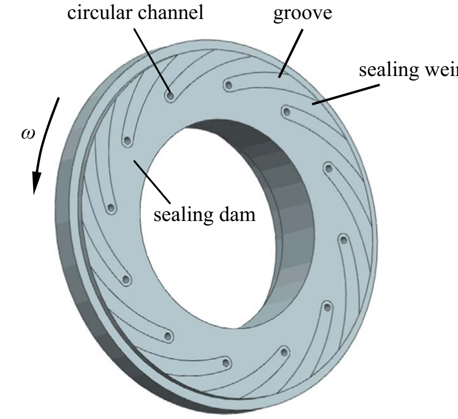

Fig. 1. Seal face structure of the pump-out hydrodynamic mechanical seal.