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Multi-objective optimization on dimple shapes for gas face seals

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ABSTRACT

The shape of surface texture has a significant effect on the performance of face seals. Many studies found that the textures designed for improving load carrying capacity tend to increase leakage at the same time. Therefore, a multi-objective optimization approach is presented to optimize the dimple shapes with freedom edges. It has been found the dimples with asymmetric “V” shape offer better performance in terms of load carrying capacity and leakage. Moreover, the optimal shapes are compared with four kinds of optimal regular shapes under different rotating speeds. The results show that the superiority of shape optimization is more obvious in conditions with high speed.

1. Introduction

There is an increasing demand for reliable and durable face seals with low leakage and low friction under high speed, high temperature, and complex working conditions. Surface texturing has been proven to be an effective means to improve the tribological performance of sliding surfaces because of the hydrodynamic effect under full or mixed lubrication and lubricant reservoir effect under staved lubrication conditions [1–5].

The patterns of grooves and dimples are common types of surface texture for face seals [1–15]. They are typical representatives of connected texture and disconnected texture, respectively. Grooves such as spiral grooves [10–13] and T-shape grooves [9] have been widely used in face seals due to the pumping effect and hydrodynamic effect. Dimples were also proven to reduce friction or improve load carrying capacity under different conditions [3–7,14–18], meanwhile, they were expected to obtain a better hydrodynamic effect due to the disconnected structure [7,15]. Nakano et al. [7] found that cast iron surfaces with dimples had lower friction than the surfaces with grooves or meshes under lubricated conditions. Shi et al. [14] found that elliptical dimples with a high area density can obtain a higher load carrying capacity and higher gas film stiffness than grooves for gas lubrication. With further research, dimple shape has become a hot factor in the optimization of surface textures. Lu and Khonsari [2] reported that bushing with elliptical dimples has a lower friction coefficient than that with circular dimples under mixed lubrication. Uddin et al. [19] found that square shape dimples with a single wedge-bottom profile offered better tribological performance than triangular, chevron, circular and elliptical shapes. In view of the above studies, a conclusion can be drawn that the dimple shape is an essential factor to improve the tribological performance. Therefore, beyond the above regular shapes, are there any other shapes of dimples which can provide a better impact on the performance of sliding surface? Shen and Khonsari [20,21] conducted numerical optimization for complex dimple shapes using a sequential quadratic programming (SQP) algorithm. The optimal shape which can produce the maximum load carrying capacity was obtained by changing the design variables from an arbitrary shape. This work provides a possibility for further optimization of texture shapes.

Leakage is another important factor for gas face seals. However, it is difficult to improve the load carrying capacity or opening force and simultaneously reduce the leakage through surface texturing [8,10,22]. The clearance of seal rings increases with the increasing opening force, which leads to an undesirable increase of leakage [8]. In order to ease the case, the seal with double-row spiral grooves was presented where one row can pump the leaked medium back to the sealed space. But such a seal tends to have a complicated structure, needs larger installation space and can only be applied under low pressure differences [8]. Moreover, the load carrying capacity and the leakage rate were analyzed independently in most of the studies. It is difficult to obtain a better combination performance objectively by a single objective analysis.

This study aims to provide a multi-objective optimization approach specially for conflicting objectives i.e., load carrying capacity and leakage rate, to optimize the shape of dimples on gas face seals. The models of dimples and multi-objective optimization problem are established where the dimples have an arbitrary shape on a certain

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Nomenclature

h	gas film thickness
F_t	non-dominated front
h_0	sealing clearance
M	average molar mass of gas
h_g	groove depth
P_0	initial population
l	inner diameter of dimple region of type *a*
P_t	population at N_{th} iteration
m	number of design variables
Q	leakage rate
n_r	rotating speed
Q_t	population obtained by crossover and mutation
p	gas film pressure
R	gas constant
p_a	atmospheric pressure
R_t	union of P_t and Q_t
r	coordinate in the radial direction
T	thermodynamic temperature
r_I	inner diameter of sealing ring
ω	angular velocity
r_O	outer diameter of sealing ring
W	load carrying capacity
α	small angle of dimple region of type *b*
$\overline{R}$	dimensionless coordinate in the radial direction
β	large angle of dimple region of type *b*
$\overline{\vartheta}$	dimensionless coordinate in the circumferential direction
Θ	coordinate in the circumferential direction
$\overline{H}$	dimensionless gas film thickness
γ	gas pressure to density ratio
$\overline{P}$	dimensionless gas film pressure
δ	angle of computing domain
$\overline{Q}$	dimensionless leakage rate
ρ	gas density
$\overline{W}$	dimensionless load carrying capacity
μ	dynamic viscosity of gas
NSGA-II	elitist non-dominated sorting genetic algorithm

constraint. Then, the optimal dimple shapes and the Pareto-optimal sets are obtained using the elitist non-dominated sorting genetic algorithm (NSGA-II). Furthermore, the optimal shapes are compared with the optimal regular shapes, including circle, ellipse, square and triangle, under different rotating speeds.

2. Optimization method

2.1. Physical model and governing equation

The face seal is composed of two sealing rings. Fig. 1 shows the physical model, where r_I and r_O are the inner radius and outer radius of the rings, respectively. The sealing faces are separated by a layer of gas film with the thickness of h_0. n_r is the rotating speed, and h_g is the depth of dimples. In this study, 24 dimples are uniformly distributed on the stationary ring face. A sector unit cell containing one dimple is considered as the computing domain, and δ is the angle of computational domain.

In order to optimize the shape of the dimple from arbitrary geometry, there are 2n control points for one dimple which is formed by connecting the adjacent points. In this study, the shapes of dimples include two types: type *a* and type *b*, as shown in Fig. 2. For type *a*, the control points of *n-i*th and 2*n-i*th ($i = 0,1, \ldots n\text{-}1$) are located at the same radius, and they are free in the circumferential direction under a certain constraint which is shown in the multi-objective optimization model. The coordinates of these points are shown in Table 1. Based on this principle, 2n + 2 design variables are required, and they are $\theta_1, \theta_2, \ldots\theta_n, \theta_1{}^a, \theta_2{}^a, \ldots\theta_n{}^a, l, L$. For type *b*, the regularity is similar to that of type *a*, but the points of *n-i*th and 2*n-i*th ($i = 0,1, \ldots n\text{-}1$) are located at the same angle and they are free in the radial direction.

The number of design variables $m = 2n+2$ is an important factor for the computing accuracy and efficiency. Generally, the more the design variables, the smoother the edges of optimal shapes, however, the longer the computing time. Take type *a* as an example, the number of design variables m is studied in this study. It is found that when m is increased from 8 to 16, the optimal shapes have similar geometries and the edges become smoother and smoother, meanwhile, the computing times increase at an increasing rate. For $m = 14$, the optimized shapes have relatively smooth edges, and the computing time is 247 h which is 32% smaller than the computing time (367 h) of $m = 16$. Based on the computing time and shape accuracy, m = 14 is chosen in this study.

The two-dimensional steady-state Reynolds equation in the polar coordinates is employed to analyze the gas film pressure distribution $p(r,\theta)$. The equation can be expressed as:

$$\frac{\partial}{\partial r}\left(\frac{\rho r h^3}{\mu}\frac{\partial p}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial \theta}\left(\frac{\rho h^3}{\mu}\frac{\partial p}{\partial \theta}\right) = 6\omega r\frac{\partial(\rho h)}{\partial \theta} \tag{1}$$

where r and θ are the polar coordinates in the radial and circumferential directions, respectively; μ is the dynamic viscosity of gas, which is assumed as a constant in this study; ρ is the gas density; p is the gas pressure; h is the gas film thickness which is equal to h_0+h_g in the dimple region and equal to h_0 beyond the dimple region; and ω is the angular velocity which is equal to $2\pi n_r/60$.

Reynolds number, defined as $Re = \frac{\rho v h}{\mu}$ where v is the peripheral velocity of the seal face, is a dimensionless number that characterizes the state of fluid. Generally, the flow can be treated as laminar when $Re < 1000$ [23]. In this study, an example with the rotating speed of 10000 rpm is mainly studied, where the maximum Re is not more than 350, so laminar flow is assumed in this model.

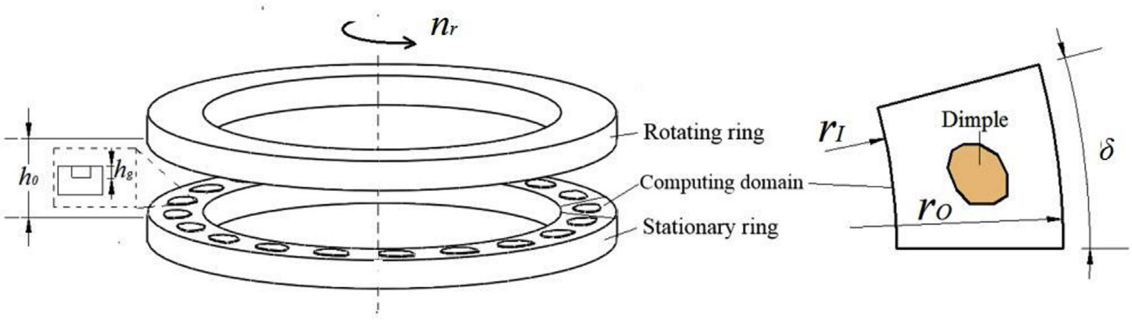

Fig. 1. Physical model of the face seal.