Topological Optimization of Dry Gas Seals for Improving Seal Characteristics

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Abstract

This paper describes a new optimum design approach in dry gas seal which used widely for high speed turbo machinery for minimizing air leakage. The dry gas seals are required for high quality of sealing characteristics because the large air leakage leads to decrease of efficiency and increase of clash risk on high speed and high quality turbo machinery. Hence, in the designing of seals, minimizing of air leakage is the most important thing. On the other hand, previous studies have not been conducted from the view point of optimum groove shape mounted on seal surfaces. Therefore, we applied the topological optimization method proposed by Hashimoto [1] to dry gas seal to improve the air leakage characteristics. The optimization problems are defined by reference an actual design problem and they are solved theoretically by using the proposed the optimization method. As a result, the optimized groove shape of dry gas seals are quite deferent comparison of usual spiral groove shape. Moreover, the applicability of optimized dry gas seals is verified theoretically.

Keywords: tribology dry gas seal, optimum design, topological optimization, leakage

1 Introduction

Recently, high efficiency and low environmental load are required for machine industry. For almost all of turbo machinery, low fluid leakage leads to high efficiency and low environmental load. For example, the gas turbines generate power by turbine rotation from combustion gas which elevated temperature and pressure. Low gas leakage makes it possible to high inner pressure and temperature and to realize the high efficiency combustion.

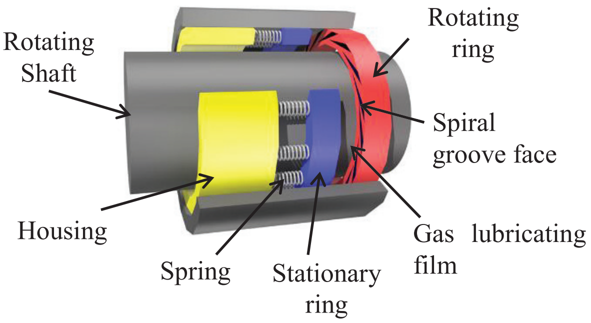


Fig. 1 Outline of Dry Gas Seals

Figure 1 shows the typical dry gas seal which consists of a rotating shaft, a rotating ring with spiral groove face, a stator, housing, and springs. As shown **Fig. 1**, the gas lubricating film and gas pressure is generated between stator ring by the effect of spiral groove and controlling the air film thickness by spring forces. The seal effect is obtained by the gas lubricating film and changing the groove design makes it seal characteristics.

Previously, many researchers [2]-[5] treated how the seal effect is obtained and how to reduce the air leakage by various ideas. However in many studies, basic groove shape was constant spiral curve and there are less research concerning optimization of groove shape. On the other hand, Hashimoto [1] tryed to topological optimization in thrust air bearings to ehnance mainly the stiffness of air lubricating film and varified the optimization effect theoretically and experimentally. Therefore, in this study the topological optimization method proposed by Hashimoto was applied to the dry gas seal design in order to minimize the amount of seal leakage in a wide range of operating conditions. Moreover, the appricavility of the optimization method on sealing effect of dry gas seal are verified theoretically.

2 Topological optimum design method

Figure 2 shows the geometry of dry gas seal. In this study, the minimizing the leakage quantity of dry gas seal is defined as objective under constant conditions of the internal pressure P_i [Pa], the seal clearance h_c [μm], the seal inner and outer radius ratio r_r, rotational speed n_s [rpm].

The outline of the topological optimization is shown in **Fig. 3**. The cubic spline interpolation function is applied so that we can change the groove shape freely. First, the bearing surface is divided equally in the axial direction as shown in the figure, and then the orbital nodes between the concentric circles and the curve of spiral grooved bearing which is defined as initial shape in our optimization are set as design point C_i (i=1~4). The groove shape of seal is expressed by interpolating the each node by the third spline function. The extent of angle change ϕ_i (i=1~4) in the θ direction from initial bearing geometry are defined as design variables, the values are changed so that intended characteristics

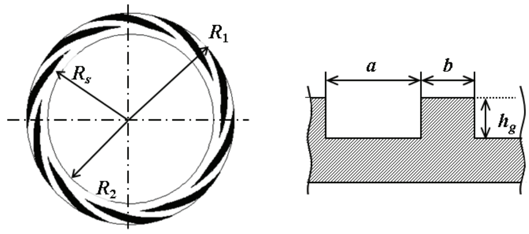


Fig. 2 Geometry of dry gas seal

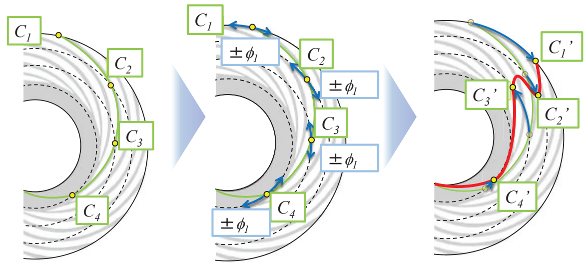


Fig. 3 Flow of groove optimization

become maximum or minimum. Simultaneously, groove depth h_g, seal radius R_s, groove width ratioα, number of grooves N are set as the design values. Hence, the design vector is defines as follows.

$$X = (\phi_i(i = 1 \sim 4), N, R_s, h_g, \alpha) \quad (1)$$

The objective function is leakage quantity q and defined by the following equation.

$$f(\boldsymbol{X}) = q \quad (2)$$

On the other hand, constraints imposed on design variables and the quantity of state are upper and lower limits for the number n of design variables shown in Eq. (5), allowable air lubricating film thickness, and the damping coefficient of air lubricating film which must not be a negative sign. These variables are defined by using the following inequality:

$$g_i(X) \le 0 (i = 1 \sim 14) \quad (3)$$

Then, the constraint functions g_i (i=1~14) are obtained as follows:

$$\left.\begin{array}{l} g_1 = \phi_{1\min} - \phi_1,\ g_2 = \phi_1 - \phi_{1\max},\ g_3 = \phi_{2\min} - \phi_2, \\ g_4 = \phi_2 - \phi_{2\max},\ g_5 = \phi_{3\min} - \phi_3,\ g_6 = \phi_3 - \phi_{3\max}, \\ g_7 = \phi_{4\min} - \phi_4,\ g_8 = \phi_4 - \phi_{4\max},\ g_9 = R_{s\min} - R_s, \\ g_{10} = R_s - R_{s\max},\ g_{11} = h_{g\min} - h_g,\ g_{12} = h_g - h_{g\max}, \\ g_{13} = \alpha_{\min} - \alpha,\ g_{14} = \alpha - \alpha_{\max} \end{array}\right\} \quad (4)$$

Here, the above static and dynamic seal characteristics are obtained by solving the Reynolds equivalent equation. The outline is shown as follows.

First, the bearing shape from r-θ polar coordinate system into a boundary-fitted coordinate system and after that modified DF method in solving the Reynolds equivalent equation. From the equilibrium of mass flow rate between air inflow and outflow in the control volume caused by the shaft rotation and the squeezing action, the following Reynolds equivalent equation is obtained.

$$Q^{\xi}_{2I} + Q^{\xi}_{1III} - Q^{\xi}_{2II} - Q^{\xi}_{1IV} + Q^{\eta}_{2I} + Q^{\eta}_{1II} - Q^{\eta}_{2III} - Q^{\eta}_{1IV} = Q^{\Gamma} \quad (5)$$

where, the mass flow rates of Q^{ξ}, Q^{η} and Q^{Γ} are given as follows.

$$Q^{\xi} = \int_{\eta_1}^{\eta_2} \rho \left(-A\frac{\partial p}{\partial \xi} + B\frac{\partial p}{\partial \eta} + D + E \right) d\eta \quad (6.1)$$

$$Q^{\eta} = \int_{\xi_1}^{\xi_2} \rho \left(B\frac{\partial p}{\partial \xi} - C\frac{\partial p}{\partial \eta} + F + G \right) d\xi \quad (6.2)$$

$$Q^{\Gamma} = \int_{\xi_1}^{\xi_2} \int_{\eta_1}^{\eta_2} \frac{\partial(\rho h)}{\partial t} |J| d\eta d\xi \quad (6.3)$$

Applying the perturbation method which assumes micro vibrations of air film in the vertical direction to equation (4), the dynamic characteristics of dry gas seal are obtained. The minimum air lubricating film thickness h and pressure p can be expressed as follows:

$$h = h_0 + \varepsilon e^{j\omega_f t} \quad (7.1)$$

$$p = p_0 + \varepsilon p_t e^{j\omega_f t} \quad (7.2)$$

where ε indicate the amplitude of small variations of the air film thickness.

Substituting Eq. (7) into Eq.(5), four equations in terms concerning about $\Delta\theta_x$, $\Delta\theta_y$ and ΔZ in orders 0 and 1 are obtained as follows:

$$F_0(p_0) = Q^{\xi}_{2I0} + Q^{\xi}_{1III0} - Q^{\xi}_{2II0} - Q^{\xi}_{1IV0} + Q^{\eta}_{2I0} + Q^{\eta}_{1II0} - Q^{\eta}_{2III0} - Q^{\eta}_{1IV0} = 0 \quad (8.1)$$

$$F_t(p_t, p_0) = Q^{\xi}_{2It} + Q^{\xi}_{1IIIt} - Q^{\xi}_{2IIt} - Q^{\xi}_{1IVt} + Q^{\eta}_{2It} + Q^{\eta}_{1IIt} - Q^{\eta}_{2IIIt} - Q^{\eta}_{1IVt} - Q^{\Gamma}_t = 0 \quad (8.2)$$

When the sequential solution is performed by the digitizing equations in increments using the Newton–Raphson iterative method, the static pressure component p_0 and the dynamic pressure component p_t are obtained. Finally the damping coefficient and the amount of air leakage can calculate by following integrations.

$$c = \int_0^{2\pi} \int_1^2 \mathrm{Im}\{-p_t\} r dr d\theta \quad (9.1)$$

Table 1 Given variables and constraints

Parameters	Values
Minimum groove depth	h_{gmin} =5[μm]
Maximum groove depth	h_{gmax} =[10μm]
Minimum seal radius to outer radius ration	R_{smin}=0.85
Maximum seal radius to outer radius ration	R_{smax}=0.95
Minimum groove width	α_{min} =0.1
Maximum groove width	α_{max} =0.9
Minimum spiral angle	β_{min}=10
Maximum spiral angle	β_{max}=20
Minimum groove number	N_{min}=8
Maximum groove number	N_{max}=24
Minimum angle amount	φ_{imin} =-π(i=1~4)
Maximum angle amount	φ_{imax} =π(i=1~4)
Outer radius	R_1=50[mm]
Inner radius	R_2=40[mm]

$$q = \int_0^{2\pi} -\frac{h_r^3}{12\eta}\frac{\partial P}{\partial r}\Big|_{R=R_2} r d\theta \tag{9.2}$$

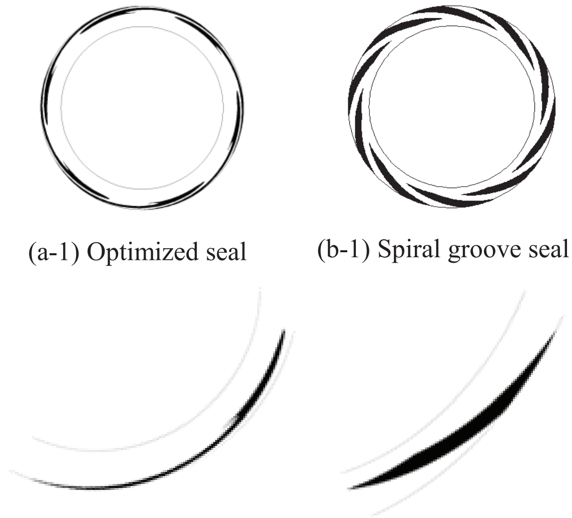
Therefore the optimization problem of dry gas seal is formulated as follows.

$$\begin{gathered} Find\ X\ to\ minimum\ f(X) \\ subjected\ to\ g_i(X) \le 0 (i = 1 \sim 16) \end{gathered} \tag{10}$$

3 Results and discussion

The constraints and constant design parameters are set specifically as shown in **Table 1**. They were defined as actual dry gas seals. The numbers of groove are set as 6, 8, 12, 16, 20, 24 non-continuously, because the number of grooves are integer. The optimum designs were conducted under each number of grooves. After all case calculations, the minimum objective function among them is selected as the optimal number of groove. The optimum calculations are conducted under the each internal pressure of P_i =0.5 to 10 [MPa] and the compressibility number which is dimensionless design parameter of Λ=10 to 1000.

Figure 4 shows the comparison of groove shape of optimized seal and that of initial spiral groove seal. From **Fig.4** (a-1) and (b-1) the grove radius area of optimized



(a-1) Optimized seal (b-1) Spiral groove seal

(a-2) Optimized seal (b-2) Spiral groove seal

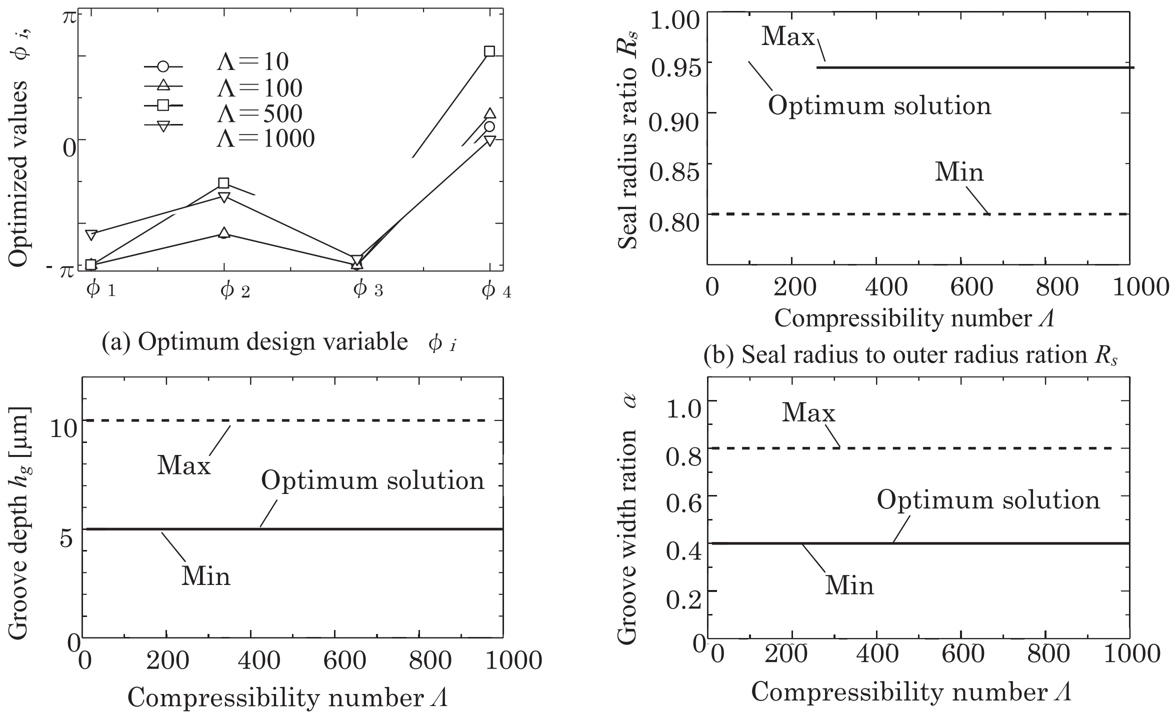
Fig.4 Seal geometry

seal is quite narrow, the groove width is thin and the number of groove decreases compared with spiral groove seal. In addition, it is found from Fig. 4(a-2) and (b-2) that the groove shape of optimized seal has bent position and quite different from spiral groove shape.

Figure 5 shows the calculation results of obtained design valuables of (a): distances from initial groove shape ϕ_i (i=1~4), (b): seal radius ratio, (c): groove depth, (d): groove width ratio against the compressibility number under internal pressure conditions of (a) P_i =0.5 [MPa]. From these results, it is found that the tendency of the optimum seal valuables is quite similar in almost all of the design conditions. In the other design valuables of seal radius ratio, groove width ratio and groove depth become maximum or minimum values of constraints are confirmed in a wide range of compressibility numbers. Therefore, from above results, the seal design for suppressing the air leakage of rotating machinery should be quite different groove shape from usual spiral groove seal and the optimized seal designs in almost all conditions are quite similar.

Figure 6 shows the comparison between the optimum designed seal and non-optimum designed spiral groove seal. In this figure, the dotted and solid lines indicate the calculation results of air leakage which is set as objective function in this study. Here, the optimized results mean the optimized at each condition. On the other hand, the plots indicate the results of optimized dry gas seal under the conditions of compressibility number of Λ=100 and inner pressure P_i=1[MPa].

From these figures, the amount of air leakage of spiral groove seal varied against the compressibility number in each inner pressure conditions. On the other hand, the air leakage results of optimized seal are almost constant against the compressibility numbers in our all calculation conditions. Especially in high area of compressibility number, amounts of gas leakage of optimized seal are suppressed effectively. This is due to pomp-out effect in outer vicinity of seal face.



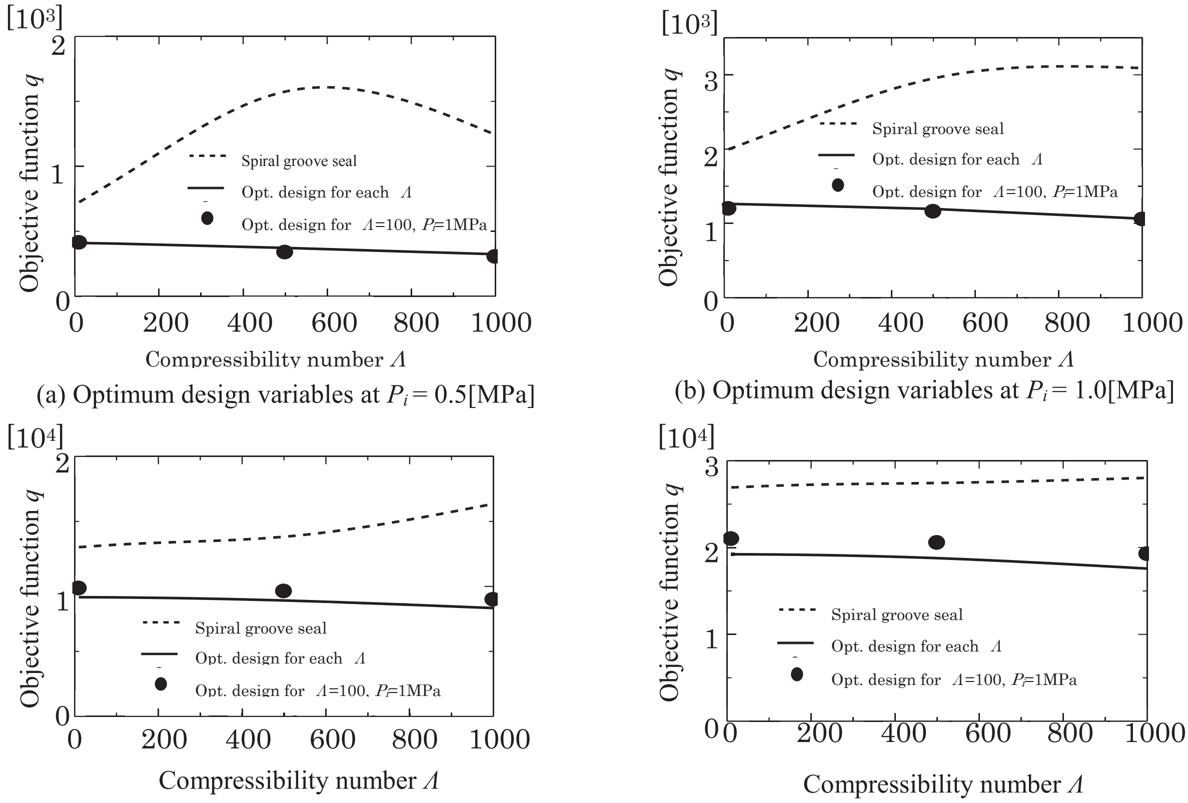
(a) Optimum design variable ϕ_i

(b) Seal radius to outer radius ration R_s

(c) Groove depth h_g

(d) Groove width ration α

Fig. 5 Optimum design variables at P_i = 0.5MPa



(a) Optimum design variables at P_i = 0.5[MPa]

(b) Optimum design variables at P_i = 1.0[MPa]

(c) Optimum design variables at P_i = 5[MPa]

(d) Optimum design variables at P_i = 10[MPa]

Fig. 6 Comparison between the optimum designed seals and the spiral groove seal

Moreover, the plots which means results of the optimized seal in case of Λ=100 and P_i=1[MPa] is almost coincide with the solid lines which means results of optimization at each compressibility numbers. This means that one case topological optimized seal have also robustness against the various design conditions.

4 Conclusions

In this study, the optimum design method is formulated for the purpose of the seal characteristic improvement of the dry gas seals. As a result, following conclusions are obtained.

1. Optimized groove shapes of dry gas seal are quite different from spiral groove shape.
2. Obtained optimized groove shapes under various compressibility numbers and inner pressures are quite similar.
3. Applying our optimization method make it possible to reduce amount of gas leakage drastically.

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