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Article

Characteristics of Modified Spiral Thrust Bearing through Geometries and Dimension Modifications

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Abstract

This research focuses on the optimum design of fluid dynamic bearing (FDB) named modified spiral design. The objective is to improve the pressure and velocity distribution inside the FDB. In this paper, the current spiral design has 12 number of grooves while the modified spiral design has 24 number of grooves. Both design can be classified into two, bearing with seal and without seal. Air was chosen to replace the oil as the lubricant. Results show that the modified spiral bearing design has comparable characteristics compared to conventional spiral design in terms of pressure and velocity distribution. Modified spiral geometries also shows a tendency to replace the function of seals in FDB. This makes it possible to simplify bearing design without using any seal just by modifying its geometries based on novel modified spiral geometries. Experimental verifications also proved that the modified spiral bearing design has better air leakage control compared to the conventional ones. This phenomenon occurs in both design parameters, where when the designs are compared with respect to increase of rotational speed and air film thickness.

Keywords

fluid dynamic bearing, modified spiral geometry, pressure distribution, air leakages

Nomenclature

CFD	Computational fluid dynamics
DVD	Digital versatile disk
FDB	Fluid dynamic bearing
HDD	Hard disk drive
N	Number of groove
n	Rotational speed
NRRO	Non-repeatable runout
R	Radius
RRO	Repeatable runout

Subscript

c	Clearance
g	Gravity
h_r	The oil lubricant film thickness
k	Dynamic stiffness
m	Mass
p_a	Atmospheric pressure
p_o	Static pressure
W	Load-carrying capacity
τ_r	Functional torque values

1 Introduction

Bearing is a simple and important device used to reduce the friction between parts during relative movements. Fluid dynamic bearing (FDB) is a type of fluid film bearing that is being used in the industry right now due to its outstanding damping characteristic compared to conventional ball bearings [1]. The bearing performance characteristics can be classified into two which is static (load capacity, friction coefficient, air leakage rate, etc) and dynamic (stiffness, damping, etc). In the journal by Han et al. [2] study on the characteristics of externally pressurized air bearings state for the design of an accurate high-speed spindle with externally pressurized air bearings, the analysis of rotor-dynamic characteristics such as

bearing stiffness and damping coefficients is very important, as is the analysis of static characteristics such as bearing load capacity and required air flow rate. Other than that, Liu et al. [3] found out that the FDB improved the load capacity for miniature spindle motors and small-form-factor data storage applications. Fluid dynamic bearing is widely used in high speed or in high precision applications. The design of bearing has the function to extend bearing life in machinery, reduce friction, energy losses and wear, and minimize the maintenance expenses and downtime of machinery which usually caused by bearing failures [4]. FDB is also widely applied in the computer information storage industry in order to provide high rpm performance that is required for hard disk drives (HDD) and digital versatile disk (DVD) drives [3]. Some of the bearing was low rotational speed and some high speed. Asada et al. [5] discussed on the problems and the important design issues for high speed bearing. The two important design issues in the optimization of grooves are the prevention of bubbles ingestions and the structural design consideration in bearing that is being used for high speed rotation, specifically for oil lubricated design bearings. In the paper written by Lv et al., they stated that turbulence increased the coefficient of friction, increased the minimum minimal film thickness and decreased the transition speed from mixed-lubricant regime to hydrodynamic lubrication regime [6].

This present paper focuses on the optimum design of a fluid dynamic bearing named as modified spiral design. The objectives of this paper are to improve the bearing's pressure distribution and identify any alternative lubricant for FDB to minimize the temperature of FDB. In contrary, most researchers only emphasize in reducing non-repeatable runout (NRRO) and repeatable runout (RRO) for the optimum design of FDB [7]. Some researchers suggest on the usage of oil as the lubricant which will reduce the absorption on the surface with the aid the magnetics field [8] but Tamboli et al. developed the high speed bearings using low viscous fluid other than oil [9]. The lubricant is pushed to the inner vicinity of the groove and generates pressure to the bearing surfaces which will cause the bearing to levitate with the aid of the magnetic fields.

To obtain an insight into the operation of the bearing, the relationship between the amount of viscous fluid pumped to the center, the pump effect and the leakage must be considered. The leakage out of the bearing is a consequence of the pressure developed by the pump effect. If the grooves are too deep, the leakage will be excessive, while if the groove depth is reduced to zero the pump effect naturally ceases. If the disc is rotated faster, the spiral groove bearing will pump more air in, as a result of which the air gap will increase. This increase in the air gap will however increase the leak out of the bearing, so that the pressure in the film between the discs still just balances the weight of the upper disc. This is necessary, because the (constant) weight of the grooved disc forms the load of the bearing [10].

The authors focus on the bearings model with a diameter of 64 mm. The current conventional spiral design in this research has 12 number of grooves while the modified spiral has 24 number of grooves, shown respectively in Fig. 1. The details of the design comparison are summarized in Table 1. Both designs are classified into two, which is bearing with seal and without seal. The difference between the two designs are shown in Figs. 2 and 3.

1.1 Design and parameter

In this paper, the authors focus on FDB which is applied as a thrust hydrodynamic bearing. The modified spiral groove design with a smaller diameter was also proposed by Ibrahim et al. [11] using modification method of groove geometries with air as its lubricants. However, no experimental analysis verifications were discussed in the paper.

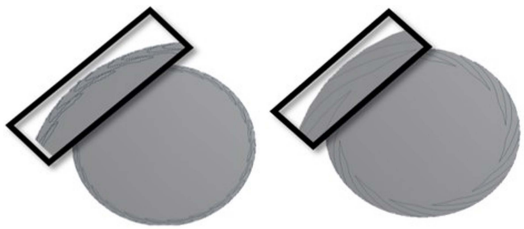


Fig. 1 Modified spiral design and conventional spiral design

Table 1 Parameter of designs for spiral groove and modified spiral groove

Parameters		Spiral groove	Modified Spiral groove
Number of groove	N[mm]	12	24
Outer radius	R_1[mm]	32	32.0
Inside radius	R_2[mm]	25.6	25.6
Seal radius	R_s[mm]	27.52	29.76
Groove depth	H_g[μm]	5.0	8.0
Rotational speed	n[rpm]	10,000	10,000

Fig. 2 Schematic cross-sectional diagram for bearing with seal design

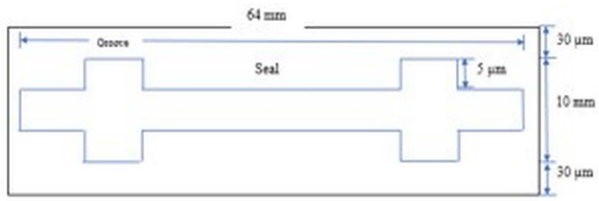


Fig. 3 Schematic cross-sectional diagram for bearing without seal design

2 Numerical and experimental methods

2.1 Numerical analysis

The numerical designs were analyzed using a computational fluid dynamics (CFD) software. The model treated air as the lubricant replacing the conventional oil lubricants. In the numerical methods, we fixed the rotational speed of shaft at 10,000 rpm. The details of initial parameter settings for CFD analysis is shown in Table 2. Design and simulation were conducted using the same software (Ansys Fluent Software).

2.2 Experimental setup

In the experimental analysis, the spiral design with seal and the modified spiral design with seal were compared.

Table 2 Initial parameters for numerical analysis

Parameter	Values
Density [kg/m^3]	1.225
Pressure [*Pa*]	101325
Temperature [*K*]	293
Velocity [m/s]	33.5
Viscosity [(kg/m). *s*]	1.7894×10^{-5}
Ratio of specific heat	1.4

Experimental device for visualization of a dry gas seal was used to measure the amount of air leaked from the fluid dynamic bearing. The experimental setup is shown in Fig. 4. The experimental analysis was carried out with varied rotational speed and air-film thickness, ranging from 0-20,000 rpm, and 18-30 μm, respectively.

3 Results and discussions

3.1 Numerical results

Figures 5 through 13 show the numerical analysis results for designs of modified spiral with seal, modified spiral without seal and spiral without seals, respectively. Comparison between Figs. 5 and 8 shows that pressure distribution has a similar pattern but the average pressure difference between the two designs is only a mere 3%. For comparison between the modified spiral and spiral design of without seals, the modified spiral gives higher pressure values than the conventional spiral design of that without seal.

The pressure distribution pattern for spiral without seal shown in Fig. 11 was almost identical with the researches previously presented by Sunami et al. and Ibrahim et al. [11, 12]. An increment of about 94% of average differential pressure distribution between conventional spiral design without seal and modified spiral design without seal can be seen between Figs. 8 and 11.

In this paper, velocity distribution and patterns are one of the characteristics that was analyzed. In order to identify how much air flow is pushed into the bearing, and how the air lubricant collides onto the groove and seal, the vector velocity analysis was chosen. Vector velocity distribution of the modified spiral was better compared to the conventional spiral due to the inversed design groove, which allows more air film to be concentrated at the center of the groove while at the same time pushing air outwards from the middle of the seal area towards outer radius. This will eventually balance the air pressure distribution to the seal smoothly.

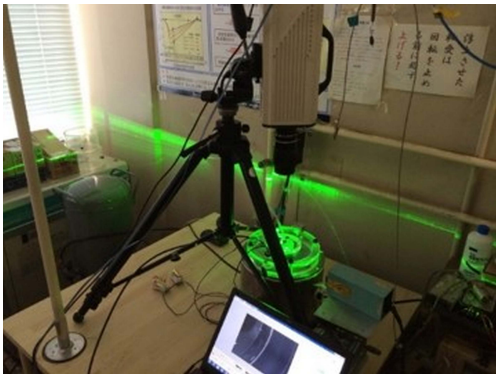


Fig. 4 Visualization experimental device

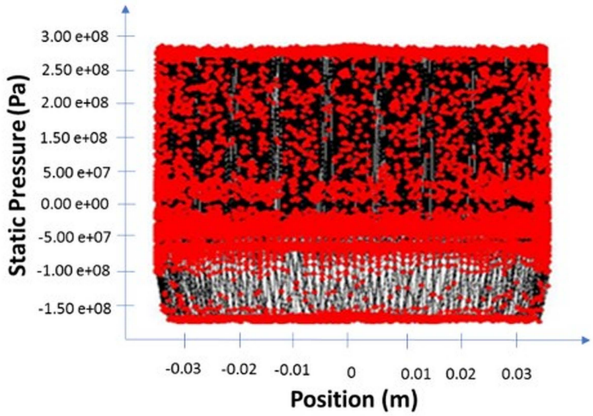


Fig. 5 Static pressure of modified spiral with seal

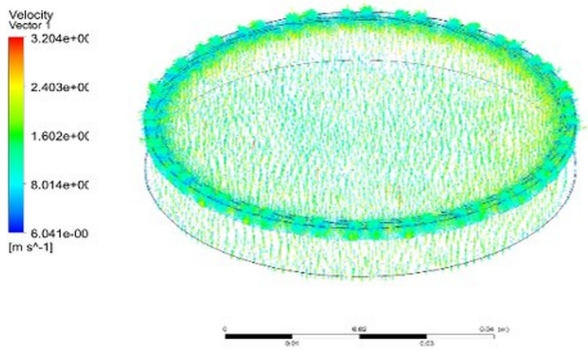


Fig. 6 Velocity vector of modified spiral with seal

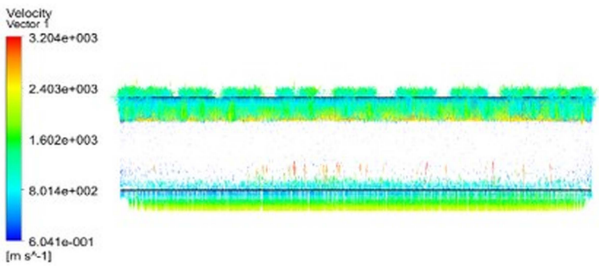


Fig. 7 Side view velocity vector of modified spiral with seal

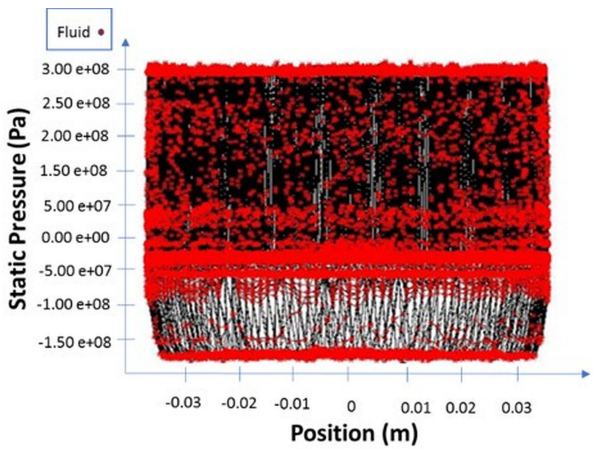


Fig. 8 Static pressure of modified spiral without seal

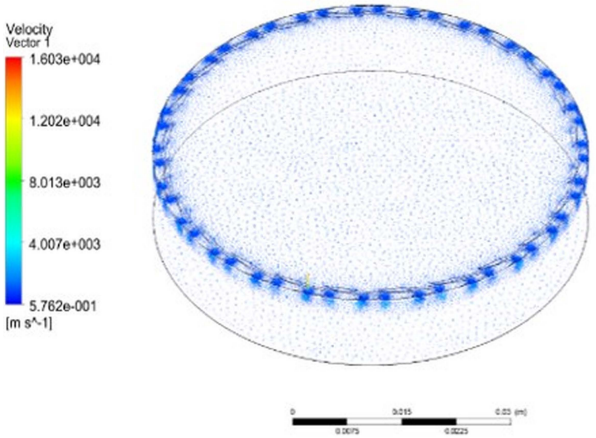


Fig. 9 Velocity vector of modified spiral without seal

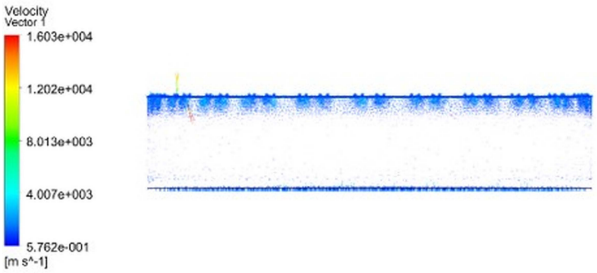


Fig. 10 Side view velocity vector of modified spiral without seal

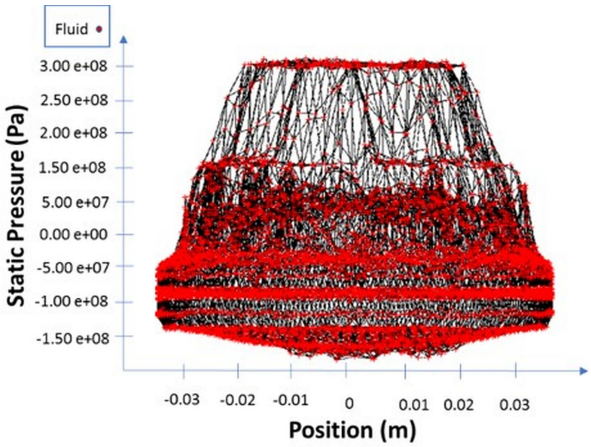


Fig. 11 Static pressure of spiral without seal

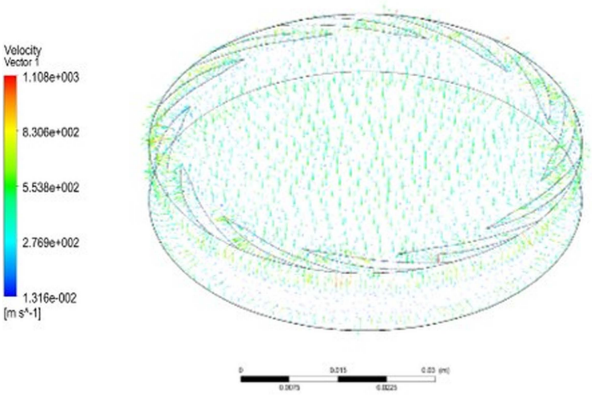


Fig. 12 Velocity vector of spiral with seal

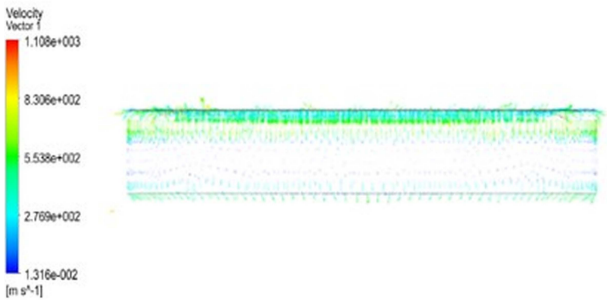


Fig. 13 Side view of velocity vector for conventional spiral without seal

For both modified spiral with seal and without seal characteristics, at the center of the bearing vicinity, the ones with seal possesses more scattered air film compared to the ones without seal of a modified spiral bearing design. The velocity vector results can be seen in Figs. 6 and 7 for modified spiral with seal, Figs. 9 and 10 for modified spiral without seal, and Figs. 12 and 13 for spiral without seal, respectively. Conventional spiral design velocity vector shows a much lower value compared to both modified spiral designs. Figures 12 and 13 shows velocity vector for spiral without seal design. In this design, the velocity was evenly distributed to the surfaces including the center of the bearing. Compared to conventional spiral design, modified spiral was focused and distributed on the groove itself. Maximum velocity in the spiral design was also lower compared to the modified spiral. It is reduced about $1.4922\times10^4\ ms^{-1}$. From this we can state that the modified spiral has more velocity.

Figures 14 and 15 shows the cross-sectional view of groove inside the bearing for pressure and velocity distributions, respectively. The figures are differentiated by the numbering of bracketed (a), (b) and (c), which resembling modified spiral with seal, modified spiral without seal and spiral without seal, respectively. From Fig. 14, the pressure distribution of the modified spiral with seal does not show significant pressure difference compared to each design. However, negative pressure can be found from the cross-sectional groove in the bottom of the surface of bearing. The negative pressure can be seen if we scale down the range of pressure. Negative pressure indicates that the pressure in the bearing is lower than the atmospheric pressure [11]. The pressure shows that both modified spiral design has lower pressure compared to the spiral design. It is in the opposite with velocity. The speed inside the groove shows different pattern and values for every design presented in this paper. The value for modified spiral is higher compared to the conventional spiral design. The velocity for modified spiral with seal increased around 280% compared to the conventional design. Table 4 shows the force and torque values for each type of design that it can support.

In the process of analyzing the data of characteristic of the bearing, the Reynolds equivalent equation was used. The Reynolds equation will be solved by using the Newton-Raphson iteration method.

Some basic assumptions are applied in this study before deriving the equation of the fluid dynamic bearing, where the flow is considered as laminar flow. The governing equations for this problem can be written as follows: where p is the pressure distribution of the bearing, r is the radius of the bearing, h the oil lubricant film thickness (which are replaced with air). Equation (1) may be integrated twice and evaluated to determine the load carrying capacity.

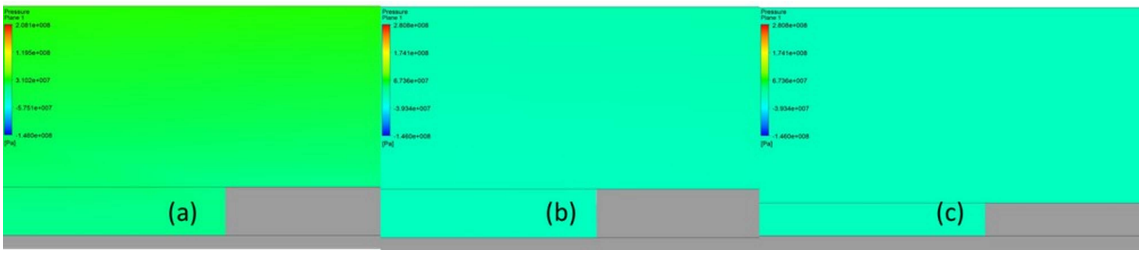


Fig. 14 Pressure profile cross sectional groove (a) modified spiral with seal (b) modified spiral without seal (c) spiral without seal

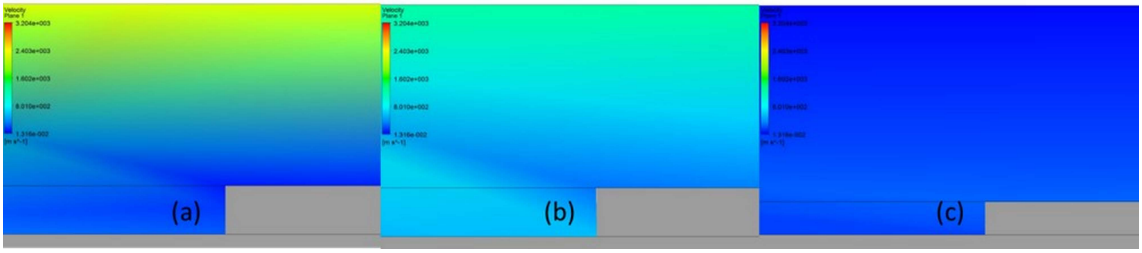


Fig. 15 Velocity profile cross sectional groove (a) modified spiral with seal (b) modified spiral without seal (c) spiral without seal

Table 3 Cross-section groove average value

	Modified Spiral with seal	Modified Spiral without seal	Spiral without seal
Pressure, Pa	-3.282×10^{6}	-0.18708×10^{6}	2.78988×10^{6}
Velocity, ms^{-1}	1050.25	1001.74	262.26

Table 4 Comparison of the force and torque that bearing can support

	Modified Spiral with seal	Modified Spiral without seal	Spiral without seal
Force, N	1.31281	1.33882	5.50643
Torque, Nm	4.4332	11.9006	0.8611

The load-carrying capacity, W

$$W = \int_0^{2\pi}\int_{r1}^{r2} \{p_o(h_{ro}) - p_a\}\, r dr d\theta \quad (1)$$

Where:
p_o = static pressure
p_a = atmospheric pressure
h_r = the oil lubricant film thickness
Where the force balanced equation;

$$W(h_r) - mg = 0$$

Functional torque values, τ_r

$$\tau_r = \int_{r2}^{r1}\int_0^{2\pi}\left[\frac{\mu r^3 cos}{h_0} - \frac{rh_o}{2}\frac{\partial P_o}{\partial\theta}\right] dr\, d\theta$$

The dynamic stiffness can be obtained from following equation.

$$k = \sqrt{k^2 + (w_f c)^2}$$

3.2 Experimental results

From the results in Fig. 16, it can be seen that the amount of leakage decreased with the increasing speed of rotor for both designs. However, the modified spiral has better characteristics in containing the air leak compared to the conventional spiral design. From these experimental results, it shows that the modified spiral bearing design has the capability to improve the air lubricated containment area or the seal function that a bearing possesses.

Figure 17 shows the relation of air leaks when the thickness of the air film lubrication is increased. The inversed modified spiral vicinity will improve the air flow in the bearing surface by balancing the amount of negative and positive pressure in the bearing region and provides less leakage when compared to the conventional spiral design. In the context of groove design, a modified spiral has better characteristics to reduce the amount of air leakages. This difference represents 32% reduction of the amount of leakages between the two bearing designs.

Figure 18 shows the visualization result. The image taken with high-speed camera upward of the test seal which is the object to be photographed. The image shows the average vectors of five rotation of rotors. Figure 18 (a) show the spiral groove while Fig. 18 (b) shows the modified spiral groove. Red circle in Fig. 18 shows the difference of amount of the high pressure between the two designs. Modified spiral design shows a more high-pressure area distribution compared to the spiral design. From these figures we can say that the high pressure in the modified spiral design has the ability to compensate the seal effect to the bearing, simplifying the bearing design.

4 Conclusions

Modified spiral design is introduced as an innovative design to replace the conventional spiral design. This paper investigates the effect of air lubrication distribution to the presence of seal in its design. The paper also shows that the effect of seal is insignificant for the comparison between modified spiral with seal and without seal. However, the comparison between spiral without seal and modified spiral without seal shows higher pressure distribution difference. A bearing that does not

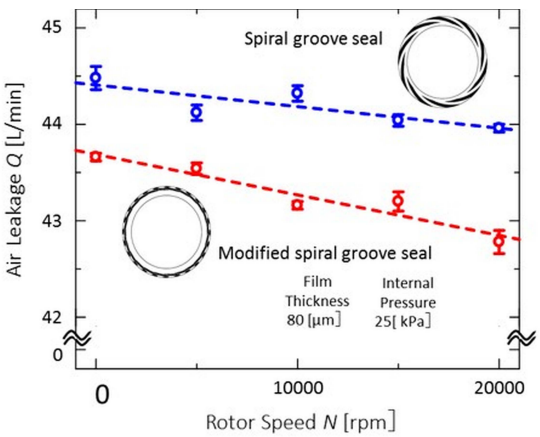


Fig. 16 Leakage vs rotor speed, N

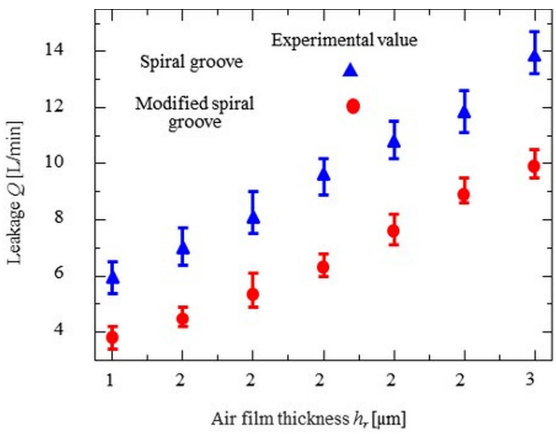
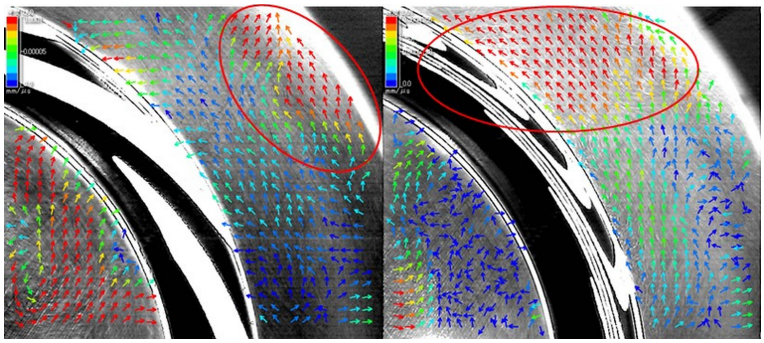


Fig. 17 Leakage vs air-film thickness



(a) Spiral groove design (b) Modified spiral groove design

Fig. 18 Visualization results

requires seal promotes simplicity in bearing design, without jeopardizing the characteristic of the FDB itself.

In addition to that, when a modified spiral with seal is compared with a modified spiral without seal the modified spiral with seal has higher concentration of air lubrication in the center of the bearing. This implies that the film could avoid the collision of bearing surfaces in the event of startup and stoppage of an air lubricated bearing. From the experimental results, we can also conclude that the modified spiral with seal give positives result to replace the conventional spiral design, in terms of air leakages control.

Acknowledgements

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