International Journal of Control and Automation
Vol. 9, No. 5 (2016), pp. 79-90
http://dx.doi.org/10.14257/ijca.2016.9.5.08

The Theoretical and Experimental Research on Combination Rotary Sealing of Ferrofluid and Magnetic Grease

Jun Ji[1], Feifei Xing[2], Jun Du[1], Ning Shi[1]

[1]Beijing Polytechnic,
[2]Beijing Machinery and Electricity Institute, High-Tech Co.Ltd
ji_jun2000@sina.com

Abstract

In this paper the combination rotary sealing of ferrofluid and magnetic grease was used for high power electric motor of certain equipment. The pressure capability and the similar friction power formula of it was derived theoretically, that the viscous dissipation power of magnetic grease sealing is approximate proportional to the $(2n+1)$ *th rotational speed. The experiments were carried out to verify the combination rotary sealing was theoretically and experimentally feasible. This structure device has been put into practical use.*

***Keywords**: Ferrofluid, Magnetic grease, Sealing, Large gap*

1. Introduction

Ferrofluid sealing is a new form of sealing method [1], it relies on the liquid material to fill the gap to achieve sealing function [2-3]. Magnetic grease is a special magnetic fluid, it's based liquid is high viscosity and non-Newtonian fluid--seal grease, and it has the advantage of higher pressure capability and better compatibility with liquid medium, but it also with the high power consumption and high heat generated during working, this seriously affects the sealing life.

The principle of the magnetic grease sealing and the Newtonian ferrofluid sealing is almost the same, but the based liquid used in the magnetic grease is sealed grease, which has a good sealing effect, and the sealing capacity of the grease mainly relies on the yield stress, it dues to the presence of grease inside the three-dimensional network skeleton structure of the solid soap thickener fibers, when it is formed by a small force, the performance characteristics is the solid style, and when the shear stress reaches the yield stress, the grease destruction of the internal skeleton structure is broke, the grease begins to flow. When the magnetic particles are uniformly dispersed in the grease, it will form a new equilibrium, the coupling agent of magnetic particles and the thickener particles makes the internal structure of the magnetic grease becomes more complex, the sealing mechanism will be more mysterious [4-5].

Compared with the Newtonian ferrofluid, the magnetic grease sealing has higher powder filling rate, higher saturation magnetization, and the viscosity increases significantly, the larger shearing force during the rotating, the viscosity-temperature effect is more significant.

This paper solved the cooling medium freon (F113) sealing problem of the high power motor of some military equipment, this high-power motor(the power level is MW rated) which has large shaft size works in a large transverse and axial vibration environments and the sealing structure requires the adaptation to follow the vibration, the sealing gap reaches to 0.7-0.8mm, but the general gap of ferrofluid sealing is about 0.1 ~ 0.3mm[3], this size of the gap has a great influence on the ability of the sealing. It is also required to consider the damp, mildew, salt spray, corrosive media and other necessary factors.

ISSN: 2005-4297 IJCA

2.The Theoretical Study on the Combination Rotary Sealing of Ferrofluid and Magnetic Grease

2.1. The Rheological Properties of Seal Grease

During the sealing process, the current state of the seal grease between the sealing gap(assumed to be the rectangular teeth) is called a plug flow. When the outside pressure exceeds the yield stress caused by the resistance it begins to flow, it has

$$F_{out} = \Delta p\pi(\delta/2)^2 = (p_1 - p_2)\pi(\delta/2)^2 .$$

The resistance caused by yield stress: $F_{res} = \tau_y \pi\delta l$, where τ_y is yield stress. When the seal grease begins to deform, the driving force exceeds the yield stress only at the radius is $\delta/2$,so only at this point the seal grease begins to flow, as $F_{out} \propto (\delta/2)^2$, $F_{res} \propto \delta/2$, the seal grease will stay there under the yield stress when the external pressure is not increased. The flow core and gradient zone of the sealing grease is shown in Figure 1, and the flow state process of the sealing grease is shown in the Figure 2.

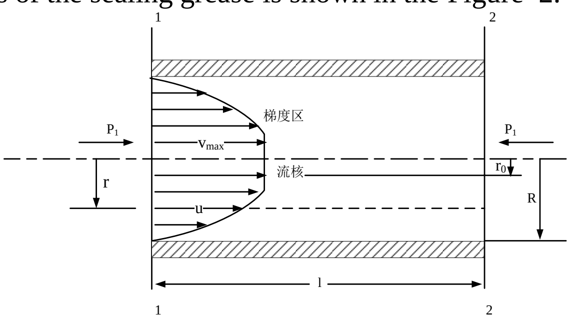


Figure 1. The Flow Core and Gradient Zone of the Sealing Grease

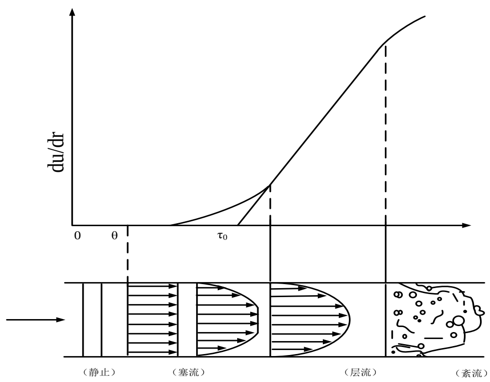


Figure 2. The Flow State Process of the Sealing Grease

The seal grease we use here is a Herschel-Bulkley model, it has $\tau = \tau_y + \iota\dot{r}^n$, where τ_y is yield stress, ι is consistency coefficient, $\dot{\gamma}$ is shear rate, n is flow index.

2.2. The Calculation of the Pressure Capability of Ferrofluid and Magnetic Grease Combination Rotary Sealing

For multi-stage sealing ability, we assume the pressure is approximately equal to each level, so the total pressure capability equation of the ferrofluid and magnetic grease combination sealing can be approximated written as

$$\Delta p = n_1 \mu_0 M_{s,1}(H_{\max,1} - H_{\min,1}) + n_2 \mu_0 M_{s,2}(H_{\max,2} - H_{\min,2}) + n_2 \tau_y ,$$

where n_1 and n_2 is ferrofluid and magnetic grease sealing series respectively, μ_0 is the permeability of vacuum, $M_{s,1}$ and $M_{s,2}$ is the saturation magnetization of ferrofluid and magnetic grease respectively, τ_y is the yield stress of the seal grease.

The sealing capability is the effect of the magnetic field and the joint action motion state of ferrofluid and magnetic grease when the shaft rotates, and ferrofluid and magnetic grease "negative effect" effect is conspicuous, mainly dues to the shaft diameter and speed[6]. Under above assumptions, and assume the viscosity of ferrofluid and magnetic grease is a constant, steady in-compressible, uniform temperature and for a laminar flow [7], the rotating sealing pressure formula can be obtained as :

$$\Delta p = n_1 \mu_0 M_{s,1}(H_{\max,1} - H_{\min,1}) + n_2 \mu_0 M_{s,2}(H_{\max,2} - H_{\min,2}) + n_2 \tau_y - \frac{1}{2}(n_1 \rho_1 + n_2 \rho_2) v_0^2 \frac{\delta}{R}$$

where ρ_1 and ρ_2 is the density of ferrofluid and magnetic grease respectively.

2.3. The Calculation of Friction Power Consumption of Magnetic Grease Rotary Sealing.

Compared with other sealing equipments, the drag torque and power consumption of Newtonian ferrofluid sealing is minimal [8]. The required shaft power in ferrofluid sealing is $p_1 = M_1 \omega L = \dfrac{4\pi\eta L \omega^2 r_1^2 r_2^2}{r_2^2 - r_1^2}$, where L is the length of the shaft. From the formula, we can see that the basic loss of viscosity is approximately proportional to the square of the speed.

For magnetic grease, its base carrier liquid is non-Newtonian fluid-sealed grease, which thixotropic and shear thinning characteristics and rheological properties are more complex than Newtonian fluid, the great viscosity will result in the large viscous drag of rotation sealed. The following power consumption formula of the magnetic grease rotating seal is derived in Cartesian coordinates, assuming that:

(i)$V_z = 0, V_r = 0, V$ is the velocity component along the surface direction of the shaft.

(ii)For the force of the magnetic grease field has:

$$f_r = \frac{\partial}{\partial r}\int_0^B MdB,\ f_z = \frac{\partial}{\partial z}\int_0^B MdB,\ f = 0.$$

(iii) $dV/dt = 0$

(iv)There is $V = V_0(1 - \dfrac{x}{\delta})$ at the height of the sealing layer

The magnetic grease in the seal gap is corresponded with Herschel-Bulkley model rheological equation: $\tau = \tau_y + \iota\dot{\gamma}^n$, where the τ_y is yield stress, the ι, n is the consistency coefficient and flow index of the fluid, $\dot{\gamma}$ is the shear rate, we can get :

$$f(\tau) = [(\tau - \tau_y)/\iota]^{\frac{1}{n}} . \tag{1}$$

To ignore the wall slippage of magnetic grease, take above equation into the following formula: $\dfrac{4V}{\delta} = \dfrac{4}{\tau_w^3}\int_0^{\tau_w} \tau^2 f(\tau) d\tau$, where the $\dfrac{4V}{\delta}$ is nominal rate cut of the total flow, the $\dfrac{4}{\tau_w^3}\int_0^{\tau_w} \tau^2 f(\tau) d\tau$ is no wall slippage shear nominal rate, τ_w is the shear stress of axial plane. Integrate the formula (1), we can get:

$$\frac{4V}{\delta}=(\frac{\tau_w-\tau_y}{\iota})^{\frac{1}{n}}4n(1-\frac{\tau_y}{\tau_w})[\frac{1}{n+1}(\frac{\tau_y}{\tau_w})^2+\frac{2}{2n+1}(\frac{\tau_y}{\tau_w})(1-\frac{\tau_y}{\tau_w})+\frac{1}{3n+1}(1-\frac{\tau_y}{\tau_w})^2] \quad (2)$$

Formula(2)can be transformed into the following form:

$$\frac{4V}{\delta}=\frac{4n}{3n+1}(\frac{\tau_w}{\iota})^{\frac{1}{n}}(1-\frac{\tau_y}{\tau_w})^{\frac{n+1}{n}}[1+\frac{2n}{2n+1}\frac{\tau_y}{\tau_w}+\frac{2n^2}{(n+1)(2n+1)}(\frac{\tau_y}{\tau_w})^2]. \quad (3)$$

Let

$$g(\frac{\tau_y}{\tau_w})=(1-\frac{\tau_y}{\tau_w})^{\frac{n+1}{n}}[1+\frac{2n}{2n+1}\frac{\tau_y}{\tau_w}+\frac{2n^2}{(n+1)(2n+1)}(\frac{\tau_y}{\tau_w})^2], \quad (4)$$

and let:

$$g(\frac{\tau_y}{\tau_w})=[\sqrt{a(\frac{\tau_y}{\tau_w})^2+b(\frac{\tau_y}{\tau_w})+c}-d]^{\frac{1}{n}}, \quad (5)$$

where a,b,c,d are undetermined coefficient,
The formula (4) can be transformed into:

$$g(1-\frac{\tau_y}{\tau_w})=(1-\frac{\tau_y}{\tau_w})^{\frac{n+1}{n}}[1-\frac{2n}{2n+1}(1-\frac{\tau_y}{\tau_w})+\frac{2n^2}{(2n+1)(3n+1)}(1-\frac{\tau_y}{\tau_w})^2].$$

The formula (5) can be transformed into the following equation:

$$g(1-\frac{\tau_y}{\tau_w})=[b'(\sqrt{a'^2+(1-\frac{\tau_y}{\tau_w})^2}-a')]^{\frac{1}{n}},$$

which a',b' is undetermined coefficient, the boundary conditions is g(0)=1,g(1)=0; and $g'(0)=0, g'(1)=-\frac{3n+1}{n(2n+1)}$, we can get: $a'=\frac{n}{\sqrt{(3n+1)(n+1)}}; b'=\sqrt{\frac{3n+1}{n+1}}$, so formula (4) can be transformed into :

$$\frac{4V}{\delta}=\frac{4n}{3n+1}(\frac{\tau_w}{\iota})^{\frac{1}{n}}[b'(\sqrt{a'^2+(1-\frac{\tau_y}{\tau_w})^2}-a')]^{\frac{1}{n}}. \quad (6)$$

Because the boundary conditions only satisfy the first derivative, the formula(6) is the approximate solution of formula(3) . Transform above equation into a quadratic equation :

$$\tau_w^2-(\tau_y+\frac{a}{b}\dot{\gamma}_n)\tau_w+\tau_y^2-\frac{\dot{\gamma}_n^2}{b^2}=0,$$

where $\dot{\gamma}_n=\iota(\frac{3n+1}{4n}\frac{4V}{\delta})^n$,solving the quadratic equation is :

$$\tau_w=\tau_y+\frac{a}{b}\dot{\gamma}_n+\sqrt{\frac{a^2+1}{b^2}\dot{\gamma}_n^2+\frac{2a}{b}\tau_y\dot{\gamma}_n},$$

Take the a',b' and $\dot{\gamma}_n$ into above equation, we can get:

$$\tau_w = \tau_y + \iota'(\frac{4V}{\delta})^n \frac{n}{3n+1}\sqrt{(\frac{2n+1}{3n+1})^2 \iota'^2 (\frac{4V}{\delta})^{2n} + \frac{2n}{3n+1}\tau_y \iota'(\frac{4V}{\delta})^n}$$

where $\iota' = \iota(\frac{3n+1}{4n})^n, V = \frac{V_0}{2}$, this is the approximate solution of formula (3), so there has: $\tau = \tau_{r\varphi} + \tau_y + \iota'(\frac{4V}{\delta})^n \frac{n}{3n+1}\sqrt{(\frac{2n+1}{3n+1})^2 \iota'^2 (\frac{4V}{\delta})^{2n} + \frac{2n}{3n+1}\tau_y \iota'(\frac{4V}{\delta})^n}$

$$= \frac{2\eta\omega r_2^2}{r_2^2 - r_1^2} + \tau_y + \iota'(\frac{2V_0}{r_2 - r_1})^n \frac{n}{3n+1}\sqrt{(\frac{2n+1}{3n+1})^2 \iota'^2 (\frac{2V_0}{r_2 - r_1})^{2n} + \frac{2n}{3n+1}\tau_y \iota'(\frac{2V_0}{r_2 - r_1})^n}$$

$$= \frac{2\eta\omega r_2^2}{r_2^2 - r_1^2} + \tau_y + \iota'(\frac{2r_1\omega}{r_2 - r_1})^n \frac{n}{3n+1}\sqrt{(\frac{2n+1}{3n+1})^2 \iota'^2 (\frac{2r_1\omega}{r_2 - r_1})^{2n} + \frac{2n}{3n+1}\tau_y \iota'(\frac{2r_1\omega}{r_2 - r_1})^n}\ .$$

Thus, the torque acting on the length unit of the cylinder is :

$$M = 2\pi r_1 \cdot 1 \cdot \tau \cdot r_1$$
$$= 4\pi r_1^2 [\frac{2\eta_1\omega r_2^2}{r_2^2 - r_1^2} + \tau_y + \iota'(\frac{2r_1\omega}{r_2 - r_1})^n \frac{n}{3n+1}\sqrt{(\frac{2n+1}{3n+1})^2 \iota'^2 (\frac{2r_1\omega}{r_2 - r_1})^{2n} + \frac{2n}{3n+1}\tau_y \iota'(\frac{2r_1\omega}{r_2 - r_1})^n}\ ]$$

Therefore, the power consumption of magnetic grease rotating seal is :

$$p = M\omega L$$
$$= 4\pi\omega L r_1^2 [\frac{2\eta_1\omega r_2^2}{r_2^2 - r_1^2} + \tau_y + \iota'(\frac{2r_1\omega}{r_2 - r_1})^n \frac{n}{3n+1}\sqrt{(\frac{2n+1}{3n+1})^2 \iota'^2 (\frac{2r_1\omega}{r_2 - r_1})^{2n} + \frac{2n}{3n+1}\tau_y \iota'(\frac{2r_1\omega}{r_2 - r_1})^n}\ ]$$

where $\iota' = \iota(\frac{3n+1}{4n})^n$, η_1 is the dynamic viscosity of magnetic grease, n is the flow index, ι is the consistency coefficient, τ_y is the yield stress.

From above analysis we can see, the highest order index of ω is $2n+1$, the viscous dissipation power of magnetic grease sealing is approximate proportional to the $(2n+1)$th rotational speed, n is the flow index of magnetic grease, that is the reason why magnetic grease sealing has so large friction heat during working.

3. Experimental Study on the Large Gap Combination Rotary Sealing of Ferrofluid and Magnetic Grease

3.1. Experimental Study on the Compatibility of Cooling Medium Freon of Power Motors Sealing

Equipment for a motor to be solved which needs cooling with F113 (chemical formula is $C_2Cl_3F_3$). F113 has gas phase and liquid phase operation state, considering the state of the liquid limitation with all the relevant materials and the compatibility conditions.

For rotary sealing experiment, we used the ester-based and oil-based ferrofluid respectively, corresponding the base carrier liquid was 2-ethylhexyl and 2# oil, the surfactant was oleic acid $C_{18}H_{34}O_2$, the magnetic powder particles were Fe_3O_4 . Through the experiments, the F113 with ester-based ferrofluid, 2-ethylhexyl ester, oil-based ferrofluid and 2# oil had dissolved mutual phenomena; the F113 diluted the Newtonian ferrofluid interface, which had impacted the sealing life. Another compatibility tests of F113 and sealing grease, magnetic grease, the magnetic grease and carrier fluid of ferrofluid showed that magnetic grease had no affinity compatibility reaction with above media liquid, and the magnetic grease surface tension was so large that the sealing can achieve "zero leakage."

Experiments also were taken at the damp, mildew, salt spray environments, the performance of magnetic grease and ferrofluid was stable in those situations.

3.2. The Structure Design of Large Gap Combination Rotary Sealing of Ferrofluid and Magnetic Grease

In the multi-stage sealing, the process of rectangular teeth is simple and the performance can be easily assured, the trapezoidal tooth has better magnetic effect [9]. Through comparative analysis, the triangular shape of teeth structure was used in this paper, because it has obvious poly magnetic effects in the pole tip and also has less reluctance effective, the use of a plurality of magnets made the magnetic field gradient in the sealing gap significantly increase, thereby, the above factors greatly improve the pressure resistance of the sealing.

As shown in Figure 3, it was designed a large gap combination rotary sealing structure of ferrofluid and magnetic grease in this paper which sealing gap was 0.7 mm, the sealing device comprised a sealedchamber, permanent magnet poles and so on.

The sealing structure used 5 permanent magnets; each permanent magnet corresponding a pair of pole pieces that constituted the magnetic circuit, the number of triangular tooth was 12, each tooth were the same size, forming a 5-slot and 6-pole 12 tooth sealing structure.

The chamber housing was 304 non-magnetic materials; permanent magnet materials were NdFeB, which had large energy product and it would generate high-intensity magnetic field, the material of magnetic pole was electric iron. The shaft material was 1Cr13, we obtained different sealing gap by machining the center axis. Magnetic pole and non-magnetic shell was made of "O" ring sealing.Each part of the structure size and the photo was shown in Figure 3.

Magnetic grease can achieve "zero leak", but in the rotation state it would make large amount of heat, therefore it cannot fully rely on magnetic grease to seal F113, in order to avoid the complexity of using water-cooling device, we used the combination sealing of ferrofluid and magnetic grease.

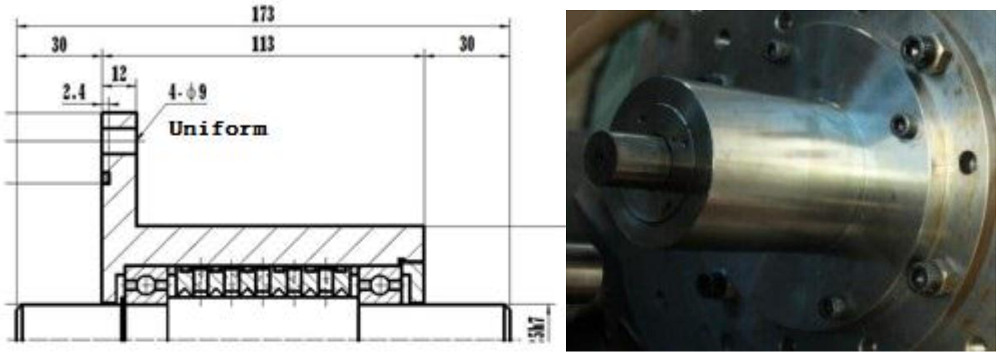


Figure 3. The Large Gap Magnetic Fluid Sealing Structure

The experimental station photo is shown in Figure 4. We used helium leak detectors to assess the leak rate of large gap sealing structure. The vacuum degree of ZQJ-230E-type helium leak detector can reach to $5\times10^{-3}Pa$, the minimum measuring of leakage rate was $10^{-10}Pa\cdot m^3/s$, the response time was less than $5s$.

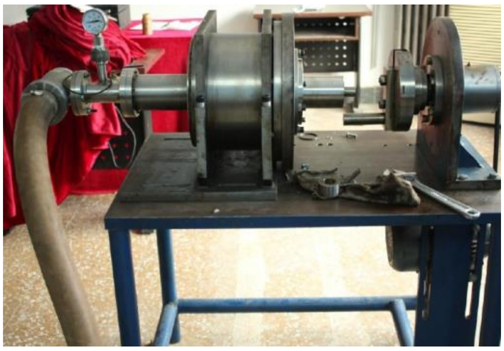


Figure 4. The Sealing Device and the Photo of the Experimental Bench

4.3. The Pressure Capability Experiments of Ferrofluid Large Gap Rotary Sealing

We first verify the pressure capability of ferrofluid large gap static sealing.

(1). The pressure test of ferrofluid large gap static sealing.

Ester-based ferrofluid was used in laboratory, each of the permanent ring injected about 5ml ferrofluid. First, open the valve, so that the high-pressure helium gas slowly went through the valve to sealing portion, about 0.02 ~ 0.03MPa helium filled at first, and then it was filled with 0.02MPa nitrogen every one or two minutes. We determined whether the sealing was destructed and read the barometer by helium leak detectors when sealing damaged.

Experimental data can be seen that the pressure decreased gradually as increasing the seal gap, the single-stage pressure capacity could meet18KPa even the maximum gap was 0.7mm. In addition, the large gap ferrofluid sealing also had self-healing properties.

According to formula calculation, the pressure calculated and experimented values were compared in the Figure 6 when the sealing gap from 0.3 to 0.7mm.

(2). The self-healing property tests of ferrofluid large gap sealing

We took the gap of 0.6mm and 0.7mm sealing for example to repeat the filling pressure experiments. When the ferrofluid sealing broke maximum pressure value(destruction), stopped inflating, the gauge would slow down to a certain value and then stayed stable; after a long time, continue making pneumatic compression to seal chamber, the process was repeated three times to determine final steady pressure. The data obtained was shown in Figure 7(a) and Figure 7(b) below:

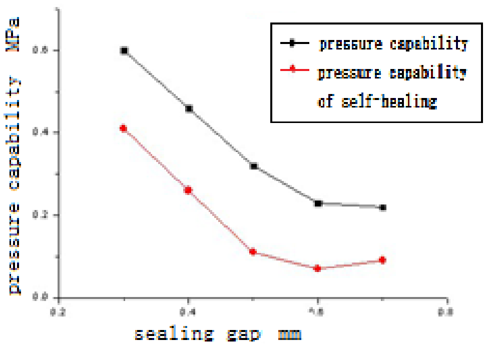


Figure 5. The Relationship Curve of Pressure Capability and Sealing Gap

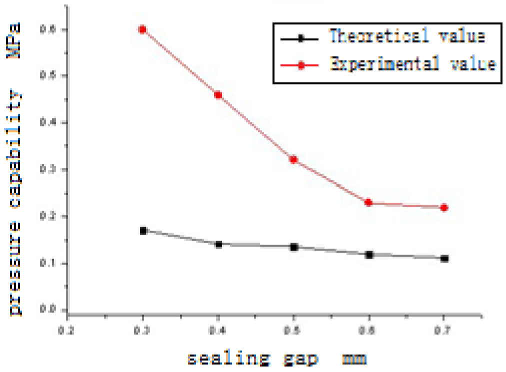


Figure 6. The Comparison of Theoretical and Experimental Values of the Pressure Capability

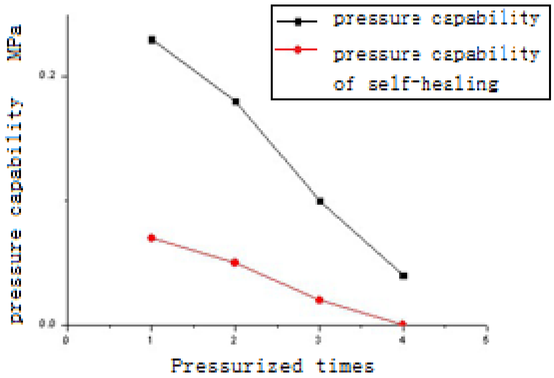


Figure 7(a). Ferrofluid Pressure Capability of 0.6mm Gap

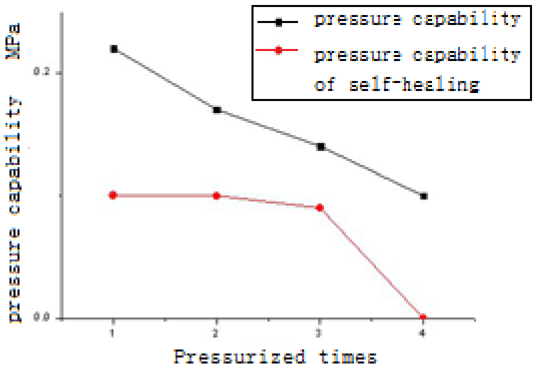


Figure 7(b). Ferrofluid Pressure Capability of 0.7mm Gap

When the sealing pressure exceeded the maximum (damage) value, the gas molecules would break ferrofluid by way of diffusion, gas instantaneous leaked, sealing was instant healed and pressure transmission was completed, pressure values stabilized at a certain value. Meanwhile, the self-healing properties of ferrofluid sealing would take some time to return.

(3). The experiments of leakage rate of ferrofluid large gap sealing

In order to test the ferrofluid sealing leakage rates, we used suction gun in the experimental stage under different pressures and the gaps were 0.3, 0.4, 0.5, 0.6, 0.7

respectively, the unit was mm, the initial vacuum degree was 2Pa, initial leakage rate was 1.0×10^{-8}.

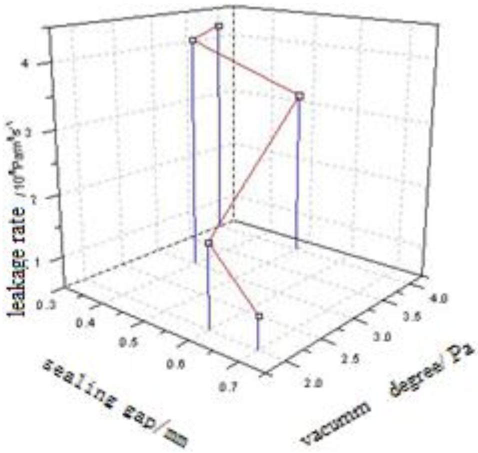


Figure 8. The Leakage Rate and Vacuum Degree of Maximum Pressure Under Different Large Gap

The experimental data was shown in Figure 8, we can get that with the increase of sealing gap, the measured leakage rate did not increase, and it declined with the increase of vacuum degree of helium leak detector. Therefore, it could be seen as a "zero"leakage at this case.

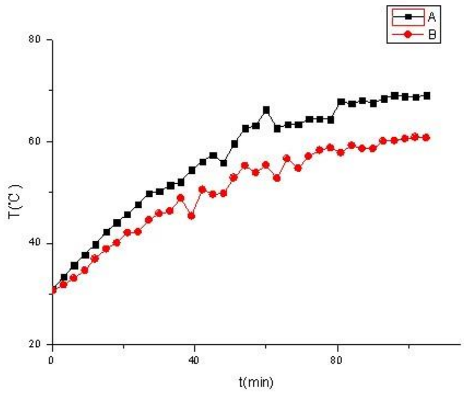
(4). The pressure capacity experiments of the ferrofluid large gap sealing under different speed.

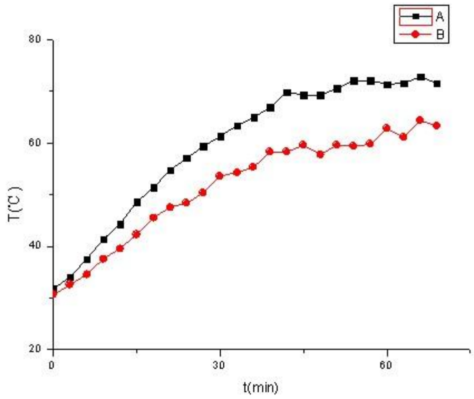
We used the 0.7mm sealing gap and the initial pressure was 0.8atm, using SB008iF-4 inverter to make Y90S-6 three-phase asynchronous motor producing output speed, and measured seal pressure changed under different speeds. Each speed continuous operation was about 1 hour.

During the experiments, the ferrofluid pressure capability remained at 0.8atm under the driving speed 200rpm, 300rpm, 400rpm, 500rpm, 1000rpm, 1500rpm respectively; it was consistent with the theoretical analysis when the line speed was less than 20m/s, the centrifugal force effect was small.

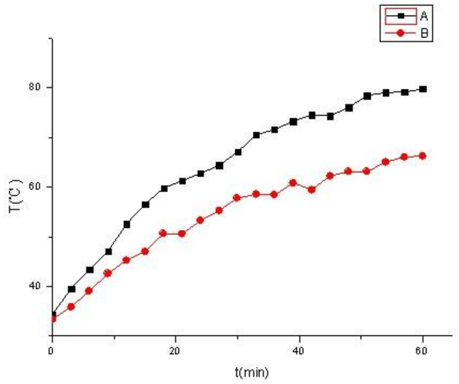
(5). The temperature experimental study on ferrofluid and magnetic grease combination rotary sealing

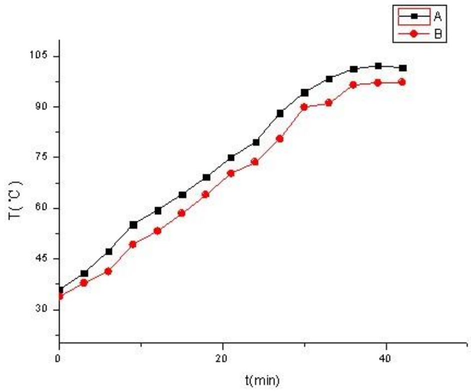
Injected the sealing device with ester-based ferrofluid, and near the magnetic poles of the F113 side with magnetic grease. First, started the motor, adjusted the driven to get the appropriate speed, so that the rotary sealing got into a stable status after three minutes. Used the contact type thermometer in the laboratory, and then getting the value once every three minutes, two sets of data corresponding to each value. First, used the probe to touch the surface of the shaft close the side of the sealing chamber, read the data after about five seconds stable; then the probe touched the shell close the side of the sealing chamber, read the data after about five seconds. As in Figure 9, curve A was temperature approximation of the shaft surface in the sealing gap, and B represented the temperature approximation of pole teeth. The temperature curves at different speeds were shown in Figure 9 and Figure 10. Since the gap between the rotary shaft and the magnet was filled with air, which had low thermal conductivity, therefore the temperature of the permanent magnets was lower.





(a) Rotary Speed was 300rpm **(b) Rotary Speed was 400rpm**





(c) Rotary Speed was 500rpm **(d) Rotary Speed was 600rpm**

A: temperature of shaft surface B: temperature of pole teeth

Figure 9. The Temperature Curve of Magnetic Grease Sealing with 300,400,500 and 600(Rpm) Speed

The following information can be get from the above Figure 9:

(1) The combination rotary sealing of ferrofluid and magnetic grease worked at different speeds, the temperature of the surface of the shaft and pole teeth increased rapidly in the frist few minutes, it turned linearly increasing trend as time changes substantially, and when the temperature of two points reached to a certain value it remained stable and the change rate was small.

(2) By way of curve comparison, the stabilization time of temperature of the ferrofluid and magnetic grease combination rotary sealing gradually shortened as the speed increased. As the speed increased, the temperature difference between the shaft surface and pole piece surface decreased uninterrupted, which was due to the increased friction loss generated by the magnetic grease.

(3) The temperature of shaft surface was always higher than the pole teeth surface, which was due to the high-speed rotation of the shaft, the magnetic grease was the high-viscosity and Non-Newtonian fluid, the friction with the shaft rotation produced large amount of heat relatively, according to the previous conclusion that the viscous dissipation power of magnetic grease sealing was approximate proportional to the $(2n+1)$ th rotational speed, n was the flow index of magnetic grease.

(4) Experiments also needed to consider the bearing heating, the sealing medium F113 has cooling effect, and the designation of sealing structure and the materials of the sealing fully met the requirements of the leak rate, pressure capability and heat-generated.

4. Conclusions

In this paper, it was proposed a combination rotary sealing of ferrofluid and magnetic grease to solve the problem of high-power motor sealing. The pressure capability and the similar friction power formula of it was derived theoretically, that the viscous dissipation power of magnetic grease sealing is approximate proportional to the $(2n+1)$ th rotational speed. It designed the structure of the device which had triangular shape of teeth was suitable for sealing the cooling medium F113 of high-power motor, it was taken some related experiments to test the feasibility of the theoretical analysis and methods.

References

[1] L. Wen, R. Zhou and C. Luo, "Development and application overview of magnetic fluid sealing technology[J]", Lubrication Engineering, vol. 6, (2002), pp. 86-90.
[2] M. Zhao, J.Z. Zou and J.H. Hu, "An analysis on the magnetic fluid seal capacity [J]", Journal of Magnetism and Magnetic Materials, vol. 303, **(2006)**, pp. 428-431.
[3] D. Mayer, "Futrue of electrotechnics: Ferrofluids [J]", Advances in electrical and electronic engineering, vol. 7, no. 1, **(2008)**, pp. 9-14.
[4] F. Xing and D. Li, "The Design and Experimental Study of Vacuum Coating Machine Biaxial Magnetic Fluid Seal", Chinese Journal of Vacuum Science and Technology [J], vol. 31, no. (3), **(2011)**, pp. 362-367.
[5] D. Li, "The Theory and Application of Magnetic Fluids[M]", Beijing. Science Press, **(2005)**, pp.31-36.
[6] D. Li, "Theory and Application of Magnetic Fluid Sealing [M]", Beijing. Science Press, **(2010)**, pp.191-244.
[7] G. Xu and Y. Zhang, "Rheological and transport properties of grease [M]", Beijing. China University of Mining and Technology Press, **(2005)**, pp.68-139.
[8] B. Yang, "Synthesis and properties of magnetic nanoparticles and magnetic fluids [M]", Taipei. National Taipei University of Technology, **(2009)**, pp.12-22.
[9] N. B. Vargaftik, "Handbook on Thermophysical properties of gases and liquids[M]", Nauka Moscow, **(1992)**.