DEVELOPMENT OF A ROTARY SEALING COMPONENT FEATURING MR FLUID AND PERMANENT MAGNET

LE HAI ZY ZY, VO VAN CUONG, DO QUI DUYEN, DIEP BAO TRI, NGUYEN QUOC HUNG*

Faculty of Mechanical Engineering, Industrial University of Ho Chi Minh city, HCMC, Viet Nam

**Corresponding author: nguyenquochung@iuh.edu.vn*

DOIs: https://doi.org/10.46242/jstiuh.v60i06.4630

Abstract. In this research work, a self-sealing component for rotary shaft of Magneto-rheological (MR) fluid based devices such as MR brakes and MR clutches is designed and tested by both simulation and experiment. The sealing component is composed of a permanent magnet and a magnetic core placed on a rotary shaft to replace a traditional lip-seal. After an overview of MR fluid and its applications as well as researches on sealing components based on MR fluid, a configuration of a self-sealing component for rotary shaft of a MR fluid based device is proposed. Afterwards, the design and modeling of the sealing component is then conducted based on Bingham plastic rheological model of the MR fluid and finite element analysis. Based on finite element analysis, optimal design of the sealing component is obtained. Prototypes of the sealing component are then manufactured and experimental works are then conducted. Base on experimental results, performance characteristics of the sealing component are investigated and compared with simulated results.

Keywords. Magneto-rheological fluid, Magneto-rheological seal, permanent magnet, MR brake.

1. INTRODUCTION

In recent years, there have been numerous researches on the application of magnetorheological fluid (MRF) such as MR dampers, MR brakes and clutches, MR engine mounts, MR valves, etc. Recently, there have been some researches on employing MRF in sealing to prevent leakage of working fluids. However, these researches are inadequate and quite limited, mostly proposed solutions and applied to a specific case. In addition, the previous studies have not concentrated on the calculation, optimal design and experimental verification of the fundamental features of seal based devices such as the frictional torque, heat generated during operation, life cycle as well as not implemented comparison with traditional seals. Kordonsky et al. [1] implemented experiments to test the ability of seals to use magnetorheological fluid (MRF) to prevent air leakage through the shaft gap in the pressurized chamber. The results showed that at different strengths of excited magnetic fields, the maximum working pressure without causing leakage and MRF was different. In addition, the friction torque caused by the MR fluid is quite small, which is an advantage for sealing application. Matuszewski et al. [2] have proposed several seal configurations using MRF for the underwater equipment. However, this study did not perform modeling, calculations and experiments. Urreta et al. [3] have researched and developed a method of sealing the rotating shaft of precision machines using MRF. The result also showed that the frictional torque is very small. In addition, using different MR fluids leads to different frictional torque. Hegger et al. [4] has proposed a method to prevent leakage of the MRF of an MR actuator by using a smart seal configuration. Some initial experiments show that the frictional torque generated by the MRF seal is very small and the system can work constantly for 6 months. Kubík et al. [5] have conducted an experiment to test the self-preventing leakage of MRF using a magnetic field. The experimental results show that the magnetic fields can be used to prevent leakage of the MRF through the slot between a rotating shaft and a fixed housing with a considerably small frictional torque.

In this study, we focus on new configurations, optimal design and experimental validation of sealing devices using MRF (called MRF seal). The seal is developed for cases, where MRF serves as the working fluid such as MR clutches, MR brakes, MR actuators. In the next section, configuration of the proposed MRF seal is presented, then the modeling of the seal is conducted based on Bingham plastic rheological model of the MR fluid and finite element analysis. From finite element analysis, optimal solution of the MRF seal is obtained. Prototypes of the optimized MRF seal are then fabricated and experimental works are then conducted. Base on experimental results, performance characteristics of the MRF seal are investigated and compared with simulated results.

2. CONFIGURATION AND WORKING PRINCIPLE OF THE PROPOSED MRF-BASED SEAL

Figure 1 shows the configuration with rectangular shaped poles of the proposed MRF seal. Significant geometric dimensions of the seal are also shown in the figure. The MRF seal includes a permanent magnet (axially magnetized), magnetic poles, magnetic sleeve, nonmagnetic separator and a nonmagnetic housing. It is noted that the magnetic sleeve is used when the shaft is made of nonmagnetic material. The permanent magnet is fixed on the two poles. This sub assembly is then enveloped in the nonmagnetic housing. Obviously, the permanent magnet plays an essential role in the MRF seal which creates a magnetic field with magnetic flux going across the MRF at the poles to prevent MR fluid leakage. There are several shapes of magnet can be implemented such as block, ring, cylinder, radial assembly and Halbach assembly. For simple structure and low cost, in this study a ring-shaped magnet axially magnetized is employed. The nonmagnetic separator is used to avoid magnetic going from this pole to the other pole without going across the MRF gap and to present the contact between the MRF and the magnet. The envelope is made of non-magnetic material to prevent magnetic field loss to the ambient. In the presence of magnetic field across the MRF gap, the magnetic particles attract each other which are then arranged along the magnetic flux lines. As a result, the MRF is almost solidified. Thanks to the above phenomenon, it helps preventing the leakage of MRF.

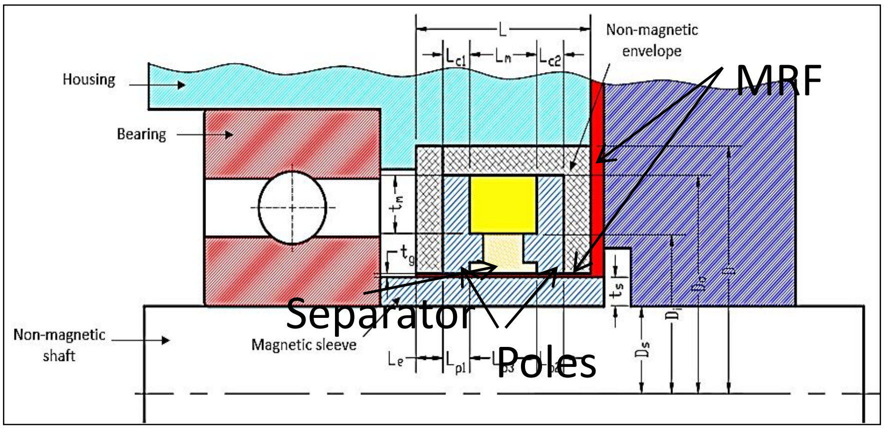


Figure 1: Configuration of the proposed MRF seal.

3. MODELING AND OPTIMIZATION OF THE MRF SEAL

3.1 Modeling of magnetic circuit

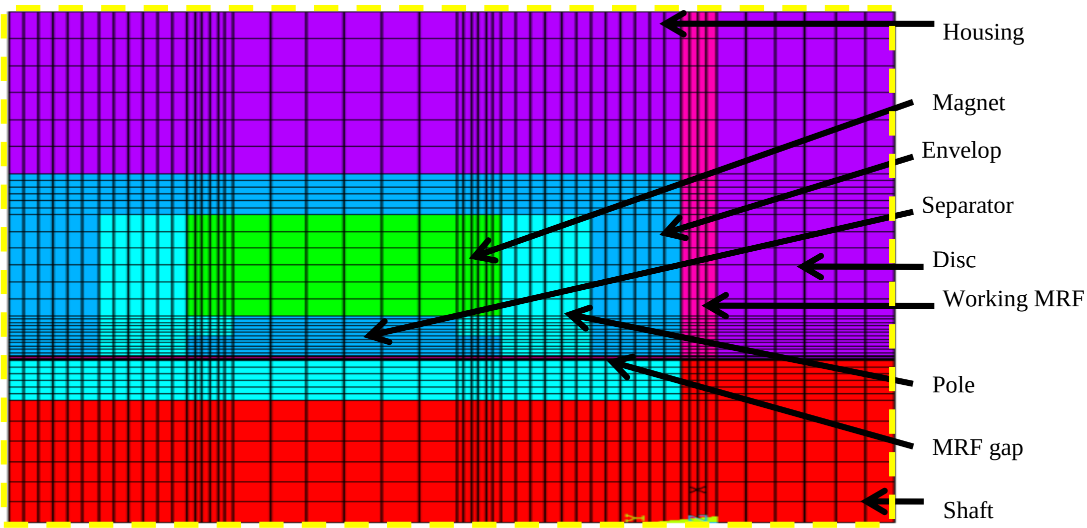


Figure 2: Finite element models for magnetic circuit analysis of the MRF seal.

The electromagnetism module integrated in ANSYS software is implemented to determine the magnetic flux density across the MRF gap in the MRF seal. Figure 2 shows the finite element model of the three MRF seals using quadrilateral element (axis-symmetry element PLANE 13) of ANSYS software. The meshing is determined by the number of elements. The outer edges of the nonmagnetic envelope are set as boundary lines of the magnetic field with parallel magnetic flux line boundary condition. The global coordinate is also implemented as the coordinate of the permanent magnet in which the y axis direction is the polarized direction of the magnet. Commercial magnetic are employed with following parameters: the material of magnet is made of NdFeB,Grade N42, plating is Ni-Cu-Ni (Nickel), axial magnetization direction, pull force is about 2.60-2.90 lbs, Brmax and Bhmax are 13,200 gauss and 42 MGOe, respectively. The magnetic property of the MRF is expressed by B-H curve which is approximately determined by the following equation [6]:

$$B = 1.91\Phi^{1.133}\left[1-\exp(-10.97\mu_0 H)+\mu_0 H\right] \quad (1)$$

Where B is the flux magnetic density (Tesla), H is the exerted magnetic flux intensity (A/m), $\mu_0 = 4\pi.10^{-7}$ *Tm/A* is the permeability of vacuum and Φ is volume fraction of the MRF.

3.2 Maximum working pressure and frictional torque of MRF-based seal

In this study, the induced yield stress (τ_y) of MRF is a function of the induced magnetic flux intensity across the MRF gap, approximately determined by experimental curve fitting as follows [6]:

$$\tau_y = 2.717\times10^5 C\Phi^{1.5239}\tanh(6.33\times10^6\ \mathrm{H}) \quad (2)$$

where τ_y is the induced yield stress (Pa), Φ and H (A/m) are the volume fraction of iron in MRF and the applied magnetic flux intensity via the MRF clearance, respectively. C has been influenced by carrier fluid such as hydrocarbon oil C=1.0 ,water C= 1.16, silicone oil C = 0.95.

The post-yield viscosity of MRF is assumed not dependent on the induced magnetic field. Equation (2) is the empirical equation for the viscosity of MRF as a function of working temperature.

$$\eta = \eta_{40}\exp\left[\frac{(1+2.43\Phi)(40-T)}{48-T}\right] \quad (3)$$

where η is the viscosity at the working temperature measure in (Pascal), η_{40} is the viscosity of the MRF at 40℃, Φ is the volume fraction of iron in MRF (%), T is the working temperature of the MRF ℃.

The maximum working pressured of the MRF seal is defined as the pressure of the working fluid at which the MRF flow starts in the MRF gap It is assumed that the pressure at the exit of the MRF seal is atmospheric pressure, and then the maximum working pressure of the seal is equal to the pressure drop of the MRF flow through the seal gap. At this starting phase of the MRF flow, the velocity of MRF in the gap is very small (almost zero), and the pressure drop due to viscosity of MRF can be neglected. Thus, the pressure drop of the MRF flow through the MRF gap of the seal (also the maximum working pressure of the seal) can be determined by [7]:

$$P_{\max} = \Delta P = c_1\frac{L_{p1}}{t_g}\tau_{y1} + c_2\frac{L_{p2}}{t_g}\tau_{y2} + c_3\frac{L_{p3}}{t_g}\tau_{y3} + 2c_4\frac{L_e}{t_g}\tau_{y4} \quad (4)$$

where L_{p1} and L_{p2} are the inner and outer pole length respectively. L_{p3}, L_e are respectively the length of the MRF duct at the nonmagnetic separator and the envelope length. c_1, c_2, c_3, c_4 are the coefficient depend on the velocity profile of the MRF flow in the gap, which is almost equal 2.0 (because of very small velocity of MRF flow).

The friction torque of the MRF seal is defined as friction torque of MRF in the gap acting on the shaft, which can be calculated by [8]:

$$T_f = T_{f1} + T_{f2} + T_{f3} + 2T_{f4} \quad (5)$$

where T_{f1}, T_{f2}, T_{f3} are respectively the frictional torque of MRF at the inner pole, the separator and the outer pole, which are determined by [8]:

$$T_{f1} = 2\pi R_s^2 L_{p1}\left(\tau_{y1} + \eta\frac{\Omega R_g}{t_g}\right) \quad (6)$$

$$T_{f2} = 2\pi R_s^2 L_{p2}\left(\tau_{y2} + \eta\frac{\Omega R_g}{t_g}\right) \quad (7)$$

$$T_{f3} = 2\pi R_s^2 L_{p3}\left(\tau_{y3} + \eta\frac{\Omega R_g}{t_g}\right) \quad (8)$$

$$T_{f4} = 2\pi R_s^2 L_{e}\left(\tau_{y4} + \eta\frac{\Omega R_g}{t_g}\right) \quad (9)$$

where R_g is the sleeve radius and τ_{y1}, τ_{y2}, τ_{y3}, τ_{y4} are respectively yield stress of the MRF in L_{p1}, L_{p2}, L_{p3}, L_e. η is the post-yield viscosity of MRF, Ω is the velocity of the shaft which is measured in rounds per minute, t_g is the MRF gap.

3.3 Optimization of the MRF-based seal

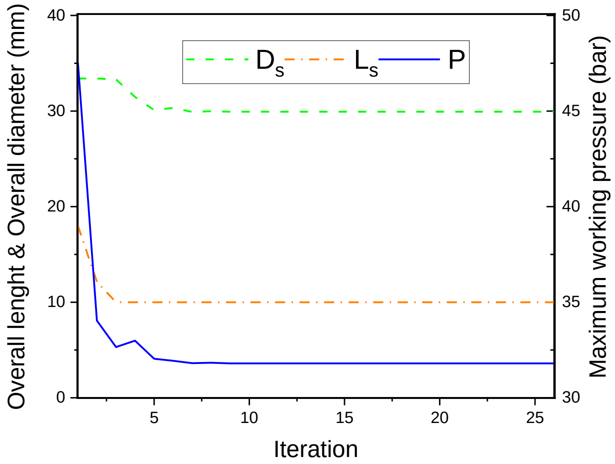
It is well-known that considering the maximum working pressure, the overall length and diameter of the seal are three most crucial in the design of the MRF based seals for replacing conventional seals or interchangeability. Thus, in the research, the optimal design objective is to maximize the working pressure whilst its dimensions such as the length and diameter were constrained to be equal or smaller than the required values. Mathematically, the optimal design problem of the MRF based seal can be stated as follows: Find optimal values of significant geometric dimensions of the MRF seal such as the pole length (L_{p1}, L_{p2}), the permanent magnet size (L_m, D_i, D_o), the sleeve thickness (t_s), the envelope length (L_e), and the core length (L_{c1}, L_{c2}) so that the working pressure determined by Equation (4) is maximized, subject to: $L \le L_{sl}$; $D \le D_{sl}$ .where, D_{sl} and L_{sl} are diameter and length of the constrained volume determined by overall size of the equivalent lip seal.

It is noted that the smaller MRF gap size, the higher pressure drop can be archived but the higher frictional torque is resulted. In addition, the manufacturing benefit and wearing problem are also important issues should be accounted. In this research, the MRF gap size is empirically chosen as 0.2 mm. The seal structure is assumed symmetric, thus ($L_{p1} = L_{p2}$) and ($L_{c1} = L_{c2}$). In order to solve the optimization problem, the first order optimization method with gradient descent algorithm integrated in the optimization toolbox of the ANSYS software is implemented. The optimization procedure can be explained as followings:

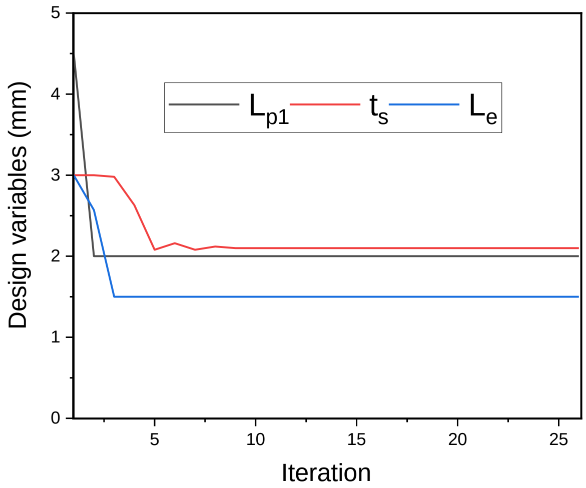
- The magnetic circuit problem of the MRF seal is solved by using APDL language with arbitrary initial values of the design variables. The average magnetic intensity across the three portions of the MRF gap is determined using path operation, in which a path is defined along the gap, the magnetic intensity across the MRF gap is then mapped on the path and average magnetic intensity is evaluated based on integration of the intensity along the path.
- The induced yield stress in the portions of the MRF gap and the post yield viscosity of the MRF are respectively evaluated based on Equations (2) and (3).
- The maximum working pressure and the frictional torque are then calculated using Equation (4) and (5).
- The entire above are coded using APDL language and saved in a notepad file, which is used during the optimization.
- From optimization toolbox, define the notepad file used in the optimization, the objective function, the design variables, the constraint functions (state variables) and the method used for solving the optimization.
- Solve the problem with initial values of the design variable.
- Execute the optimization process until the convergence is archived or the maximum iteration is reached.

4 OPTIMAL RESULTS

In this section, the optimal results of all taken into account MRF seals are obtained and some crucial discussions are presented in detail. In this study, the commercial steel S45C is made of use in the magnetic components such as the housing, sleeve and two magnetic cores and poles. The commercial MRF produced by LORD Corporation, MRF132-DG, is used as aforementioned. The stainless steel was used for nonmagnetic parts such as the envelope and the separator. Figure 3 show the optimal solution of the MRF with the constrained working space defined by D_{sl} =30 mm, L_{sl} =10 mm and D_s =10 mm, which is determined from the overall size of the commercial lip-seal Parker-62576. The convergence rate of the optimization is set by 0,1%. As shown in Figure 3(a), the convergence is achieved at the 25th iteration, at which the maximum working pressure is 26.18 bar, the overall length and outer radius are respectively 10 mm and 30 mm as constrained. The design variables of the optimal solution are shown in Figure 3(b). The optimal solution at the 25th iteration is summarized in Table 1. The magnetic density distribution of the MRF seal at the optimum is shown in Figure 4. It is noted that from the optimal result, considering the availability of commercial magnets, the actual sizes of the magnet are as following: L_m=3mm, D_i=18mm, D_o=25mm.



(a) Objective and constraints.



(b) Design variables.

Figure 3: Optimal solution of the MRF-based seal.

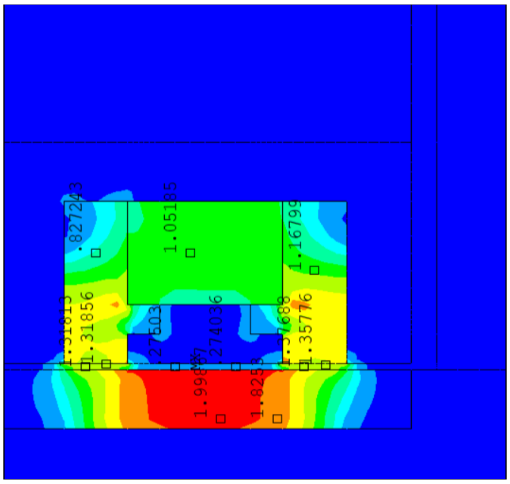


Figure 4: Magnetic flux density at the optimum.

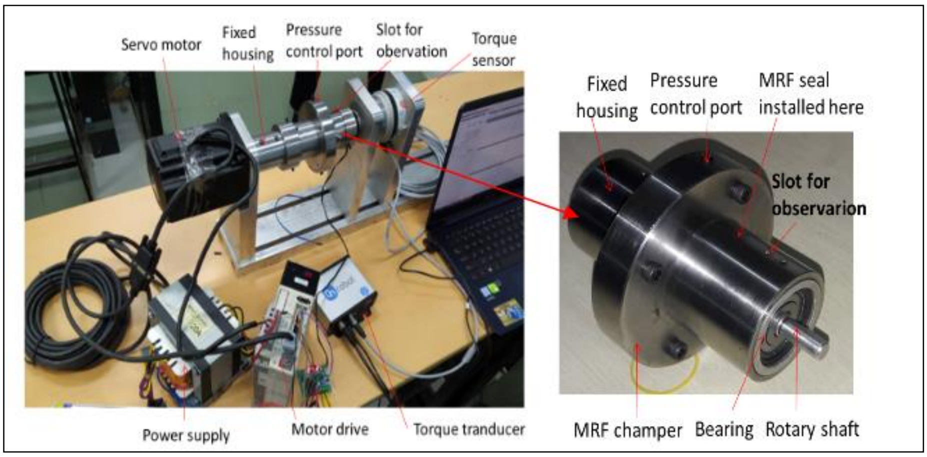
Table 1: Optimal results of the MRF seal.

Parameters	Frobnability
Overall Length: L = 10 *mm*	Max. Pressure: Δ_P = 31.83 *bar*
Overall diameter: D = 29.98 *mm*	Frictional torque: T_f = 0.091 *Nm*
Inner diameter of the magnet: D_i = 17.98 *mm*	
Outer diameter of the magnet: D_o = 24.98 *mm*	
Outer diameter of the sleeve: D_g = 14.24 *mm*	
Pole length: $L_{p1} = L_{p2}$ = 2 *mm*	
Magnet length: L_m = 3 *mm*	
Separator length: L_{p3} = 3 *mm*	
The enveloped thickness: L_e = 1.5 *mm*	
The magnet annular thickness: t_m = 3.5 *mm*	

5 EXPERIMENTAL RESULTS

5.1 Experimental set up

Figure 5 shows experimental set up to test performance of the MRF seal. A servo motor is used to drive the MRF device shaft, on which a disc is attached. The Chamber between the disc and the fixed housing is filled with MRF and pressured by a piston-cylinder system through the pressure control port. The two MRF seals are installed on both side of the shaft to prevent MRF leaking from the chamber to the bearings. A slot is machined on the housing for observation of MRF leaking. One end of the housing is fastened to the housing that fixed to the motor support. The other side of the housing is fixed to the middle support. The output shaft of the MRF device is connected to the toque sensor shaft through a mechanical coupling. The housing of the torque sensor is fixed to the outside support. The speed of the servo motor is control by a computer through the motor drive. The measured signal from the torque sensor is send to the computer through the torque transducer for evaluation.



(a) experiential set up (b) device with rotary shaft and MRF seal

Figure 5: Test bench for friction torque experiment.

5.2 Experimental results

In the first experiment, frictional torque of the MRF seal and the lip-seal are measured and compared to each other. In this experiment, the lip-seals and the MRF seals are installed at the sealing positions and the corresponding frictional torques are measured. Experimental results, in case the shaft rotates at 300 rpm, are shown in Figure 6. It is observed that the frictional torque in case of the MRF seal is around 0.13 Nm which is considerably smaller than that of the lip-seal (around 0.155 Nm). This is an advantage of MRF seals compared to lip-seals as expected.

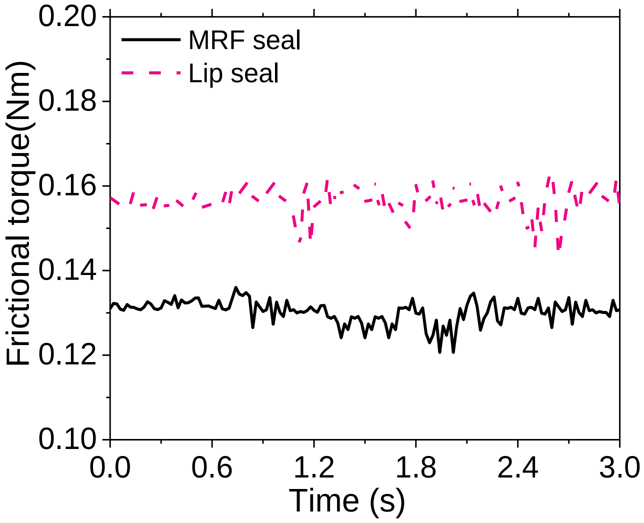


Figure 6: Experimental results on frictional torque.

In the second experiment, the maximum working pressure of the MRF seal is investigated. In this experiment, the motor is stopped and a cylinder-piston pressurized system is connected to the pressure control port on the housing. The pressure is increased gradually with 1 bar-increment step and kept each pressure step for 02 minutes. The leaking of MRF is observed at the observation slot. The experiment results show that as the pressure in the chamber reaches to 29 bar, the leaking of the MRF is observed. This is a bit smaller than the calculation (31.83 bar). The different may come from the magnetic loss, the improper inputs of material properties and the manufacturing inaccuracy. It is also noted that, for the lip-seal, the chamber pressured is often limited by 01 bar due to high frictional torque, heat problem, the wear and life time of the seal.

6 CONCLUSIONs

In this research work, a magneto-rheological fluid (MRF) based seal, called MRF seal, is proposed to replace conventional lip-seals used in MRF based devices such as MRB brakes, MR clutches and MR actuators. After an overview of MR fluid and its applications, especially the state-of-the-art MRF based seals, configuration and working principle of the proposed MRF seal was introduced. The design and modeling of the MRF seal was then conducted based on Bingham plastic rheological model of the MR fluid and finite element analysis. Based on finite element analysis, optimal design of the MRF seal was obtained using the first order method integrated in ANSYS-Mechanical APDL. From the optimal results, two prototypes of the MRF seals were fabricated for experimental works. Experimental results on frictional torque showed that the frictional torque in case of the MRF seal is significantly smaller than that of the lip seal, which are 0.13 Nm and 0.155 Nm, respectively. The maximum working pressure of the MRF prototype seals are 29 bar, which is very close to the calculated one (31.83 bar), and much higher than the working pressure of the equivalent lip-seal (1 bar). In the next research, different shapes of the poles are considered, multi-objective optimization will be conducted and more experimental works on the MRF seals such as life time as a function of shaft speed, working pressure will be performed.

ACKNOWLEDGMENT

This work was supported by research fund from Ministry of Education and Training (MOET) of Vietnam.

REFERENCES

[1] W.I. Kordonski, S.R. Gorodkin, "Magnetorheological fluid-based seal," *Journal of Intelligent Material Systems and Structures*, vol. 7, no. 5, pp.147-163, 1996. DOI: 10.1177/1045389X9600700518.

[2] L. Matuszewski, Z. Szydlo, "The application of magnetic fluids in sealing nodes designed for operation in difficult conditions and in machines used in sea environment," *Polish Maritime Research*, vol. 15, no. 3, pp.49-58, 2008. DOI: 10.2478/v10012-007-0083-0.

[3] H. Urreta, G. Aguirre, P. Kuzhir, L.N.L.d. Lacalle, "Seals based on magnetic fluids for high precision spindles of machine tools," *International journal of Precision Engineering and Manufacturing*, vol. 19, no. 4, pp.495-503, 2018. DOI: 10.1007/s12541-018-0060-9.

[4] C. Hegger, J. Maas, "Smart sealing for magnetorheological fluid actuators," *Journal of Intelligent Material Systems and Structures*, vol. 30, no. 5, pp.1-12, 2019. DOI: 10.1177/1045389X17754261.

[5] M. Kubík, D. Pavlíček, O. Macháček, Z. Strecker, J. Roupec, "A magnetorheological fluid shaft seal with low friction torque," *Smart Material and Structure*, vol. 28, no. 4, 2019. DOI:10.1088/1361-665X/ab0834.

[6] M.I. Varela-Jiménez, J. L.V. Luna, J. A. Cortes-Ramirez, G. Song, "Constitutive model for shear yield stress of magnetorheological fluid based on the concept of state transition," *Smart Material and Structers*, vol. 24, no. 4, pp. 045-039, 2015. DOI: 10.1088/0964-1726/24/4/045039.

[7] H.Q. Nguyen, Y.M. Han, S.B. Choi, N.M. Wereley, "Geometry optimization of MR valves constrained in a specific volume using the finite element method," *Smart Material and Structers*, vol. 16, no. 6, pp. 2242-2252, 2007. DOI: 10.1088/0964-1726/16/6/027.

[8] H.Q. Nguyen, S.B. Choi, "Optimal design of a novel hybrid MR brake for motorcycles considering axial and radial magnetic flux," *Smart Material and Structures*, vol. 21, no. 5, 2012. DOI: 10.1088/0964-1726/21/5/055003.

PHÁT TRIỂN MỘT THIẾT BỊ QUAY LÀM KÍN GỒM LƯU CHẤT MR VÀ NAM CHÂM VĨNH CỬU

LÊ HẢI ZY ZY, VÕ VĂN CƯƠNG, ĐỖ QUÝ DUYÊN, DIỆP BẢO TRÍ, NGUYỄN QUỐC HƯNG*

Khoa Cơ Khí, Trường đại học Công nghiệp thành phố Hồ Chí Minh, HCM, Viet Nam

** Tác giả liên hệ: nguyenquochung@iuh.edu.vn*

Tóm tắt. Trong công trình nghiên cứu này, một thiết bị tự làm kín cho trục quay của các thiết bị hoạt động dựa trên lưu chất từ biến (MR) sẽ được thiết kế và kiểm tra bằng cả mô phỏng và thực nghiệm. Thiết bị làm kín gồm có nam châm vĩnh cửu và lõi từ tính được đặt trên trục quay để thay thế cho các thiết bị chặn dầu truyền thống. Sau khi tổng hợp về lưu chất MR và các ứng dụng của nó cũng như là các nghiên cứu về các thiết bị làm kín dựa trên lưu chất MR, cấu hình của thiết bị làm kín cho trục quay của các thiết bị dựa trên lưu chất MR sẽ được đề xuất. Sau đó, thiết kế và mô hình hóa thiết bị làm kín sau đó được tiến hành dựa trên mô hình dẻo Bingham của lưu chất MR và phương pháp phần tử hữu hạn. Dựa trên phương pháp phần tử hữu hạn, thiết kế tối ưu của thiết bị làm kín đạt được. Mô hình của thiết bị làm kín sau đó sẽ được chế tạo và các công việc thực nghiệm sau đó cũng được tiến hành. Dựa trên kết quả thực nghiệm, các đặc tính hoạt động của thiết bị làm kín được kiểm tra và so sánh với kết quả mô phỏng.

Từ khóa. Lưu chất từ biến, thiết bị chặn rò rỉ từ biến, nam châm vĩnh cữu, phanh MR.

Received on: 30/09/2021

Accepted on: 01/04/2022