Development of a Model and Experimental Study of Thermal Processes in a Ferrofluid Sealer

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Abstract. The aim of the work is to create an interconnected numerical model of the magnetic, hydrodynamic and temperature fields of a ferrofluid sealer and to analyze thermal processes occurring in high-speed seals. This goal is achieved by selecting the necessary equations, boundary conditions, assumptions and physical properties of the magnetic fluid when building the numerical model of the sealer's working gap, verification of the developed model by the results of the physical experiment. The important results of the work are the obtained and analyzed data on the influence both of physical properties and the geometry of the working gap of the ferrofluid sealer on the heating of the ferrofluid. With a shaft radius of 140 mm and a linear velocity at the shaft surface of 25 m/s due to viscous heating the ferrofluid temperature exceeding the ambient temperature can reach values up to 80 degrees and higher, it has been shown. The use of the equation proposed by V.E. Fertman to determine the thermal conductivity of ferrofluid and the mixing rule to determine its heat capacity allows us to describe with sufficient accuracy for engineering calculations the thermophysical properties of concentrated ferrofluids, it was shown. The significance of the results consists in the possibility of using the developed numerical model in the study of interrelated physical processes in the working gap of the ferrofluid sealer of rotating shafts. The physical and concentration parameters of the synthetic oil-based magnetic fluid given in the paper and the results of its test operation as part of a ferrofluid seal can be used to verify the results of newly developed models of ferrofluid devices.
Keywords: ferrofluid, ferrofluid sealer, viscous heating, numerical simulation.

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Elaborarea unui model și studiu experimental al proceselor termice într-un etanșant magnetofluidic

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Rezumat. Scopul lucrării este de a crea un model numeric interconectat magnetic, hidrodinamic și de temperatură ale cîmpurilor etanșantului fluidomagnetic și de a analiza procesele termice care au loc în etanșările de mare viteză. Scopul este atins prin alegerea ecuațiilor necesare, condițiilor la limită, ipotezelor și proprietăților fizice ale fluidului magnetic în construirea unui model numeric al golului de lucru al etanșării, verificând modelul elaborat pe baza rezultatelor unui experiment fizic. Rezultate importante ale lucrării sunt datele obținute și analizate privind efectul asupra vâscoasei, a încălzirii fluidului magnetic atât proprietățile sale fizice, cât și al geometriei spațiului de lucru al sigilantului cu fluid magnetic. Se arată că cu o rază a arborelui de 140 mm și o viteză liniară pe suprafața arborelui de 25 m/s, datorită încălzirii vâscoase, excesul de temperatură a fluidului magnetic față de temperatura ambiantă poate atinge valori de până la 80 de grade și mai mult. Se arată că utilizarea ecuației propuse de V.E. Fertman și pentru a determina capacitatea sa de căldură, regula de amestecare face posibilă descrierea proprietăților termofizice ale MF-urilor concentrate cu suficientă precizie pentru calculele de inginerie. Semnificația rezultatelor constă în posibilitatea utilizării modelului numeric elaborat în studiul proceselor fizice interconectate în spațiul de lucru al unei etanșări fluide magnetice a arborilor rotativi. Parametrii fizici și de concentrație ai unui fluid magnetic pe bază de ulei sintetic, indicați în lucrare, rezultatele testării acestuia în timpul funcționării ca parte a unui sigiliu de fluid magnetic pot fi utilizați pentru a verifica rezultatele modelelor nou elaborate a dispozitivelor cu fluid magnetic.
Civinte-cheie: fluid magnetic, etanșant fluidomagnetic, încălzire vâscoasă, simulare numerică.

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Разработка модели и экспериментальное исследование тепловых процессов в магнитожидкостном герметизаторе

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Аннотация. Целью работы является создание взаимосвязанной численной модели магнитного, гидродинамического и температурного полей магнитожидкостного герметизатора и анализ тепловых процессов, протекающих в высокоскоростных уплотнениях. Поставленная цель достигается за счёт выбора необходимых уравнений, граничных условий, допущений и физических свойств магнитной жидкости при построении численной модели рабочего зазора герметизатора, верификации разработанной модели по результатам физического эксперимента. Важными результатами работы являются полученные и проанализированные данные о влиянии на вязкостный разогрев магнитной жидкости как её физических свойств, так и геометрии рабочего зазора магнитожидкостного герметизатора. Показано, что при радиусе вала в 140 мм и линейной скорости на поверхности вала 25 м/с вследствие вязкостного разогрева превышение температуры магнитной жидкости над температурой окружающей среды может достигать значений до 80 градусов и выше. Показано, что использование для определения теплопроводности магнитной жидкости уравнения, предложенного В.Е. Фертманом, а для определения её теплоёмкости правила смешения позволяет с достаточной для инженерных расчётов точностью описать теплофизические свойства концентрированных жидкостей. Значимость результатов состоит в возможности использования разработанной численной модели при исследовании взаимосвязанных физических процессов в рабочем зазоре магнитожидкостного герметизатора вращающихся валов. Приведённые в работе физические и концентрационные параметры магнитной жидкости на основе синтетического масла, результаты её испытания при работе в составе магнитожидкостного уплотнения могут быть использованы для верификации результатов вновь разрабатываемых моделей магнитожидкостных устройств.

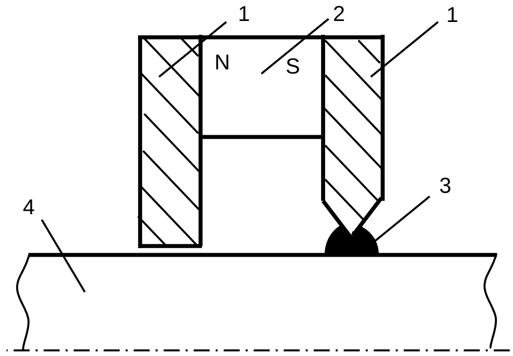
***Ключевые слова*:** магнитная жидкость, магнитожидкостный герметизатор, вязкостный разогрев, численное моделирование.

INTRODUCTION

Ferrofluids (FFs) are colloidal solutions of single-domain magnetic particles with typical sizes of about 10 nm, dispersed in a carrier liquid and stabilized by a surfactant that prevents irreversible agglomeration of particles under the influence of magnetic forces [1]. Since the 1960s of the twentieth century, when these materials were first synthesized, the areas of their technological application have continued to expand, thanks to the unique combination of fluidity and interaction with an external magnetic field.

The technology of ferrofluid sealing of rotating shafts of electric motors is a developing and promising area of application for FFs. Due to the relatively simple construction and the possibility of making it in small sizes, the installation of ferrofluid seals (FF seals) instead of bearing covers in electric motors ensures guaranteed separation of the internal volume of the electric motor from aggressive or explosive external environment with minimal friction and absence of mechanical wear of the surface of the shaft, which is typical for seals of traditional designs [2, 3, 4, 5, 6].

A typical structure of a rotary FF seal is shown in Fig. 1. A small volume of ferrofluid is retained in the annular area between the rotating shaft and the cylindrical magnetic pole surrounding it due to the additional volumetric magnetic force.



1 – pole, 2 – magnet, 3 – ferrofluid, 4 – shaft.

Fig. 1. FF seal of a typical design.

The main factor restraining the application of the FF seals on powerful machines with large diameter rotating shafts is the linear velocity increase on the surface of the shaft and the centrifugal forces and destructive effects associated with the viscous heating that occurs in the FF layer [7, 8].

With the development of numerical calculation systems for physical fields, the interest of researchers studying theoretical and practical aspects of ferrohydrodynamics is shifting towards creating models for a comprehensive assessment of the interrelated physical processes.

Among the series of works dedicated to the relationship between magnetic and hydrodynamic fields, it is worth noting the work of Kazakov Y.B. [9], which determines the shape of the ferrofluid in the gap of the FF seal using an optimization procedure, whereby the surface elements of the FF are ensured a magnetic induction value determined by pressure and temperature gradients. Rodionov A.V. with co-authors in [10, 11] define the boundaries of the ferrofluid plug in the gap of the FF seal based on the lines of equal induction obtained from numerical calculation of the magnetic field, and by using the method of finite volumes, they study the hydrodynamic processes in this volume of FF during the rotation of the shaft, assuming these boundaries are invariant. Yibiao Chen with co-authors in [12] find the initial boundaries of the FF plug in a similar way, and consider the deformation of the FF surface under the action of the applied pressure gradient and centrifugal forces when numerically calculating the hydrodynamic processes.

There are practically no works dedicated to the analysis of the thermal state and the creation of models for calculating the temperature field of the FF seals to date. The heterogeneity of the phase composition of the FF and the difference in the thermophysical properties of the magnetic phase and the liquid base (reaching two or more orders of magnitude) determine a significant dependence of the thermophysical properties of FF, such as heat capacity and thermal conductivity, on the concentration of the solid phase, and for concentrated liquids, on the intensity of the magnetic field if its vector is parallel to the heat flow [13, 14, 15, 16, 17].

Therefore, the creation of verified models for analyzing the thermal state of FF seals is an urgent scientific task.

METHODS FOR RESEARCH

The hydrodynamic calculation of the FF seal is based on the Navier-Stokes equation.

The effect of the magnetic field on the hydrodynamic processes is considered through an additional volume force acting on the FF in the gap and preset by the following equation:

$$\bar{F} = \mu_0 M \nabla \bar{H},$$

where M – is the magnetization of the FF; H – is the magnetic field intensity, and μ_0 – is the magnetic permeability of vacuum.

The magnetic field calculation is carried out according to the equations:

$$\nabla \times \overline{H} = \overline{J},\ \overline{B} = \nabla \times \overline{A},\ \overline{B} = \mu \overline{H},$$

where A – is the vector magnetic potential, B – is the magnetic induction vector, μ – is the magnetic permeability of the material.

The magnetic properties of steel and FF are specified by the corresponding magnetization curves. FF based on synthetic oil is chosen for the study, whose physical properties are summarized in Table 1.

Table 1

Physical properties of the FF and its components

Density at 293 K		1,301 g/cm^3	
Volume fraction of magnetic phase		8,89 %	
Volume fraction of surfactant		2 %	
Mass fraction of magnetic phase		35,19 %	
Mass fraction of surfactant		1,38 %	
Specific heat capacity of magnetic phase		603,5 J/(kg ·K)	
Specific heat capacity of surfactant		1848 J/(kg ·K)	
Thermal conductivity of magnetic phase		10 W/(m·K)	
Thermal conductivity of surfactant		0,231 W/(m·K)	
Plastic viscosity of FF		**Magnetization curve of the FF**	
T, K	η, Pa· s	H, A/m	M, A/m
253	36,7	0	0
263	13,7	13136	21014
273	5,78	49363	29625
283	2,59	78423	31896
293	1,3	110668	32962
303	0,724	143312	33673
313	0,457	236066	34226
323	0,302	355294	34365
Thermal conductivity of the base		**Specific heat capacity of the base**	
T, K	λ_0, W/(m· K)	T, K	C_O, J/(kg· K)
293	0,13	293	1930
313	0,128	313	1980
339	0,126	339	2030
373	0,123	373	2128
413	0,12		
433	0,118		

The problem of conjugate heat transfer is solved both in the area of the FF and in the surrounding elements of the FF seal. Heat transfer in the FF occurs by convection and thermal conductivity, while in the solid bodies, heat transfer occurs only due to thermal conductivity. The temperature field is continuous when transitioning from the FF to the elements of the casing. Heat is generated due to the viscous heating of the FF during its shear flow following the rotating shaft. The temperature field is calculated based on the heat balance equation:

$$\rho C_p \frac{\partial T}{\partial t} + \nabla \cdot (-k \nabla T) + \rho C_p u \cdot \nabla T = \\ = \eta [\nabla u + (\nabla u)^T - \frac{2}{3} \cdot \eta \cdot (\nabla \cdot u) I)] : \nabla u ,$$

where ρ – is the density, C – is the specific heat, T – is the temperature, k – is the thermal conductivity, η – is the viscosity, and u – is the velocity. The right-hand side of the equation represents the heat source due to the viscous dissipation of energy.

Given that the bases of the FF are usually operational at temperatures significantly different from the phase transition temperatures of its components, it is recommended to use the rule of mixtures when determining the heat capacity, according to which [18]:

$$C_{MЖ} = c_m^o \cdot C_p^o + c_m^c \cdot C_p^c + c_m^m \cdot C_p^m , \qquad (1)$$

where c_m^o, c_m^c, c_m^m – are the mass fractions of the base, stabilizer, and magnetic phase, respectively; C_p^o, C_p^c, C_p^m – are the corresponding heat capacities.

To determine the thermal conductivity of concentrated FFs, formula [19] is recommended:

$$lg(\lambda) = (1 - c_V) \cdot lg(\lambda_O) + c_V \cdot lg(\lambda_m), \qquad (2)$$

where c_v – is the volume fraction of the magnetic phase, λ_O – is the thermal conductivity of the base, λ_m – is the thermal conductivity of the magnetic phase.

To verify the results obtained by numerical modeling, an experimental bench was developed, shown in Fig. 2. An asynchronous motor is rotated by the electromagnet shaft. The motor rotation frequency is regulated by a frequency converter within the range of 0 to 3500 rpm.

The power consumed by the asynchronous motor is displayed on a power meter. Based on the power meter readings, taking into account the losses in the electric motor and electromagnet, an estimation of the power liberated in the FF plug due to viscous friction is made.

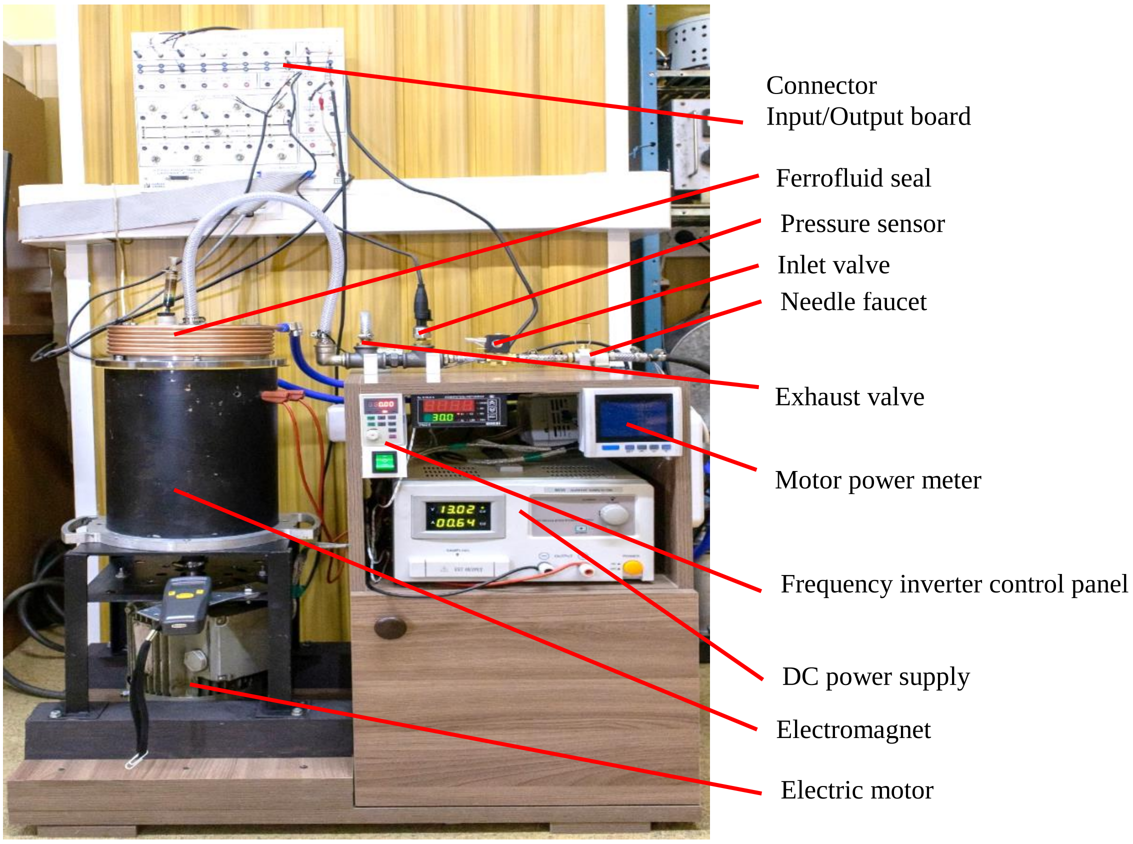
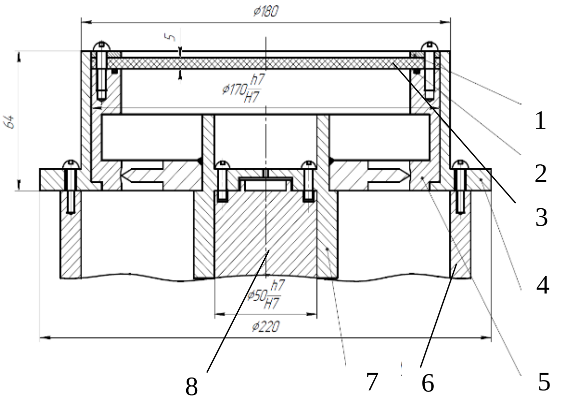


Fig. 2. Experimental bench.

The FF seal is installed on the top of the electromagnet (Fig. 3) via a 2-mm thick non-magnetic gasket, with a possibility of substitution the stationary pole attachment and the shaft for those required by the test program. The stationary pole attachment and the seal shaft form an FF seal working gap filled with FF.



1 – pressure ring, 2 – sealing ring, 3 – polycarbonate cover, 4 – FF seal housing, 5 – fixed pole attachment, 6 – solenoid housing, 7 –FF seal shaft, 8 – solenoid shaft.

Fig. 3. Design of the ferrofluid sealer.

To directly monitor the FF temperature and experimental bench components, an infrared thermal imager UNI-T UTi260A was used. Since temperature measurement on shiny metallic surfaces is difficult, a black heat-resistant paint was applied onto them (Fig. 4).

1 – surface of the pole attachment, 2 – surface of the tooth on the FF seal shaft.

Fig.4. Heat-resistant paint application locations for thermal monitoring of the FF seal temperature.

RESULTS

The temperature distribution in the FF seal working gap after 10 seconds of rotation with a linear velocity of 5 m/s obtained from the numerical simulation is shown in Fig. 5. It can be seen that apart from small areas being in contact with cold metallic surfaces, the temperature distribution in the FF plug is sufficiently uniform, and thermal imaging temperature measurement on the surface of the FF allows a reliable estimation of the temperature inside it.

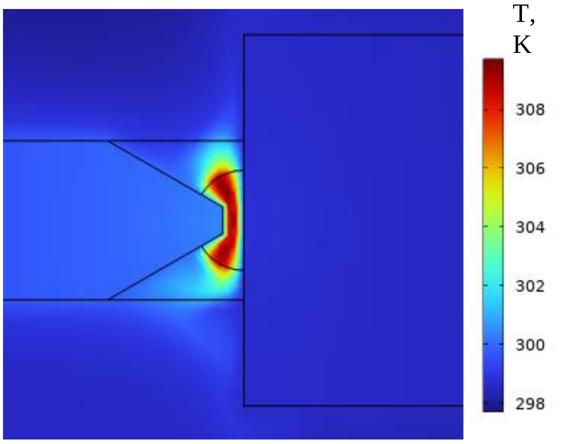


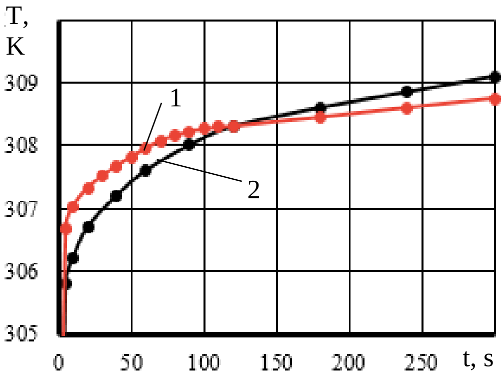
Fig. 5. Temperature distribution in the FF seal gap.

Comparison of the calculation results and the physical experiment presented in Fig. 6 shows their agreement with satisfactory accuracy for engineering calculations. The discrepancy of the curves at the initial stage is also related to the fact that in the physical experiment, unlike the mathematical one, the rotation frequency of the FF seal shaft increases not instantaneously, but gradually reaches the steady-state value over 5 seconds, and the heat release power in the FF is less than in the model.

The discrepancy in the dependencies shown in Fig. 6 after the 120th second is related to the complexity of accounting for the thermal state of technological gaps separating individual parts of the Ferrofluideal device from each other, which have a noticeable effect on its thermal conductivity.

Since such gaps are absent in the model, the thermal conductivity of the steel sections of the FF seal is higher than in the experimental device, and over time, this leads to a slowing down of the rate of temperature increase in the FF due to a more intensive heat dissipation to the metallic elements.

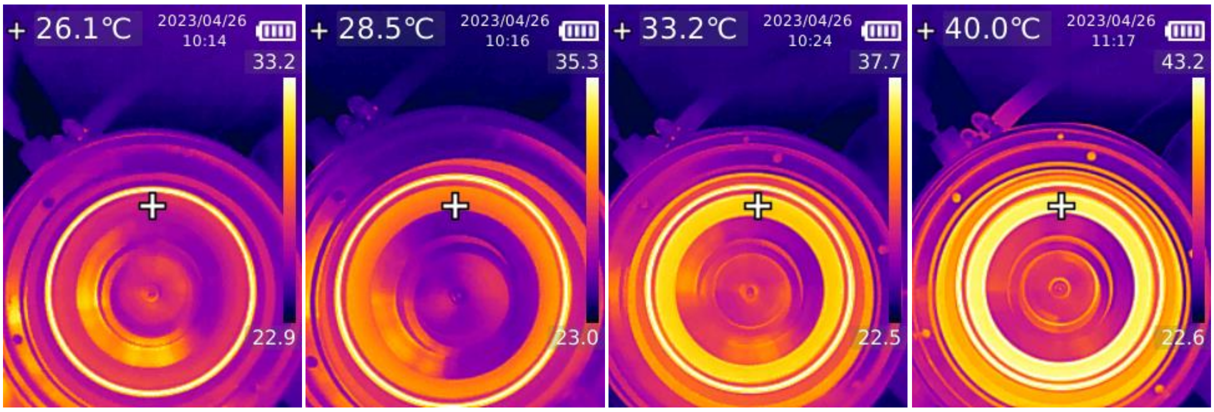
Figures 7 and 8 show the temperature distribution change in the FF seal when the shaft is rotated at speeds of 5 and 25 m/s, respectively, with a magnetic field induction of 1.2 T and a gap between the shaft and the pole piece of 0.5 mm.



1–- simulation results,
2 – results of the physical experiment.

Fig. 6. Dynamics of ferrofluid heating. The linear velocity is 5 m/s.

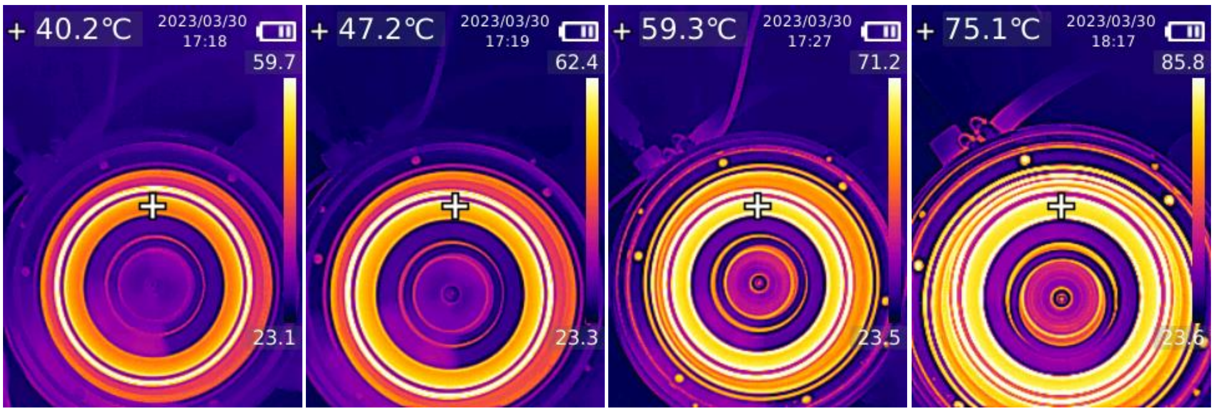
Prior to the drive motor being turned on, the FF and FF seal had a room temperature of 296 K. The FF, as the main source of heat, has a maximum temperature equal to the upper limit of the thermal scale in these images. The temperature in the upper left corner of the image is equal to the temperature of the toothed shaft of the FF seal in close proximity to the FF plug. An analysis of the temperature distribution shows that in the first 10 seconds after the start of shaft rotation, the FF temperature in the gap sharply increases. For a speed of 5 m/s, the temperature increases by 10.2 K, and for 25 m/s - by 36.7 K. At the same time, the FF viscosity sharply decreases, fluid friction decreases, and the dynamics of heating significantly reduces.



a) b) c) d)

a – 10 seconds, b – 2 minutes, c – 10 minutes, d – 60 minutes.

Fig. 7. Thermal state of the FF seal. The linear velocity is 5 m/s.



a) b) c) d)

a – 10 seconds, b – 2 minutes, c – 10 minutes, d – 60 minutes.

Fig. 8. Thermal state of the FF seal. The linear velocity is 25 m/s.

The viscous dissipation of energy in the FF plug of high-speed seals leads not only to a significant heating of the FF during operation, but also increases the overall losses in the electric motor. Figure 9 shows the change in dissipated power and temperature of the FF at a linear speed on the surface of the single-toothed FF seal shaft of 25 m/s. The increase in the temperature of the FF in the first few seconds after the start of the rotation of the shaft leads to a sharp decrease in viscosity and, as a result, a decrease in the resistance torque of the FF seal and the dissipated power. The rate of change in temperature and power is determined by the temperature dependence of the viscosity of the FF.

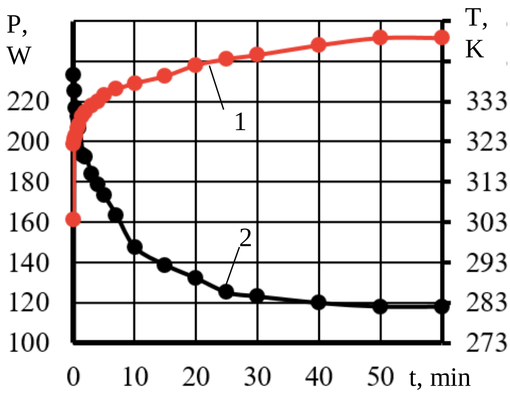
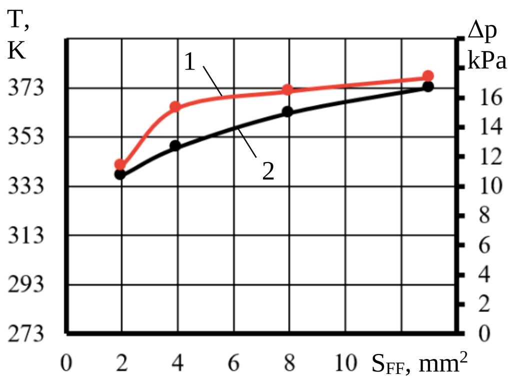


Fig. 9. Dependence of friction losses and temperature of FF in the seal.

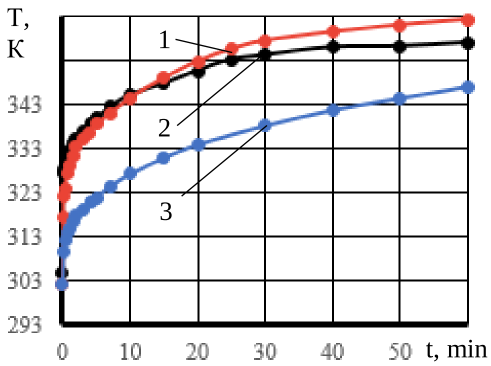
The choice of the FF necessary volume for filling the FF seal can significantly affect the performance of high-speed seals. Fig. 10 shows the experimentally obtained change in the critical pressure of the FF seal and the steady-state temperature of the FF depending on the cross-sectional area of the ferrofluid plug, directly related to the volume of the filled fluid. It can be seen that the temperature increases almost linearly with an increase in the FF volume, while the pressure drop maintained by the FF seal significantly increases with an increase in the cross-sectional area of the FF plug from 2 to 4 mm. However, the FF further addition leads to a less significant increase in the pressure drop.



1 – critical pressure, 2 – temperature.

Fig. 10. Change of FF pressure and temperature depending on the seal filling volume.

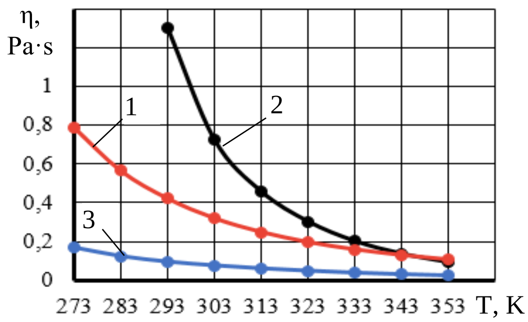
Figure 11 shows the change in the FF temperature over time when using synthetic oil, polyethylsiloxane and polymethylsiloxane liquids, as the carrier fluid.



1 – FF based on polyethylsiloxane liquid, 2 – FF based on synthetic oil, 3 – FF based on polymethylsiloxane liquid.

Fig. 11. Change of the FF temperature on different carrier fluids.

Figure 12 demonstrates the temperature dependences of viscosity for each of these FFs. All fluids have a saturation magnetization of 40 kA/m. The surface velocity of the shaft is 25 m/s.



1 – FF based on polyethylsiloxane liquid, 2 – FF based on synthetic oil, 3 – FF based on polymethylsiloxane liquid.

Fig. 12. Change of FF viscosity on different carrier fluids.

Analyzing the dependencies shown in Fig. 11 and Fig. 12, it can be concluded that the viscosity of the FF at high temperatures has a determining effect on the heating of the FF during operation in high-speed FF seal. For example, the FF based on synthetic oil has a viscosity by 3.1 times higher than that of the FF based on polyethylsiloxane at room temperature, but due to a much more pronounced temperature dependence, after just 10 minutes of operation of the FF seal, the temperature of the polyethylsiloxane-based liquid starts to exceed the temperature of the FF based on synthetic oil, and after 60 minutes of operation, this difference reaches 5.6 K. The use of a much less viscous FF based on polymethylsiloxane allows for a significant reduction in the heating of the FF seal, especially when operating in a repeat-short-term mode.

CONCLUSIONS

The developed numerical mathematical model for the integrated calculation of magnetic, hydrodynamic and thermal processes allows us to estimate the magnitude and rate of the viscosity heating of the FF by a rotating shaft in high-speed FF seals.

Comparison of the numerical calculation results with the physical experiment data shows that the use of equation (1) to determine the thermal conductivity of FF and expression (2) to determine the specific heat of FF allows describing the thermal properties of concentrated FFs with sufficient accuracy for the engineering calculations.

Experimental data show that the degree of viscosity decrease with increasing temperature has a significant influence on the magnitude of the viscosity heating of the FF plug during prolonged operation of the FF seal. The FF proper selection can significantly reduce overheating and increase the lifespan of the FF seal.

At a linear velocity of 25 m/s on the shaft surface, the temperature increase of the FF due to viscosity heating alone can reach values exceeding 80 degrees, which, in combination with the temperature increase from the operating equipment, can lead to the overheating and boiling of the FF.

Often, the FF seals are installed on industrial equipment, particularly electric motors, solely to ensure separation of the external explosive or fire hazardous environment from the internal volume. In this case, the seal only holds a small pressure differential, which is related to the heating of the air inside the operating device. In this case, reducing the volume fraction and, consequently, the viscosity and saturation magnetization of the FF appears to be an effective way to reduce the viscosity heating of the FF.

Developing a methodology for calculating the optimal volume of FF that satisfies conflicting requirements for maximum pressure differential and minimization of viscosity heating of the FF seal is a relevant scientific problem.

ACKNOWLEDGEMENT

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