Effect of Motor Shaft Eccentricity on the Performance of a High-Speed Magnetic Fluid Sealer

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Abstract. The objective of the study is to develop recommendations for accounting during the design stage and the feasibility of operating a magnetic fluid seal with a significant eccentricity of the rotating shaft relative to stationary pole attachments. This goal is achieved through conducted experimental research, the selection of necessary equations, boundary conditions, assumptions, and physical properties of the magnetic fluid when constructing a numerical mathematical model of the working gap of the magnetic fluid seal. The most important results of the study include obtained and analyzed distributions of the magnetic field, velocity field, and pressure in the magnetic fluid, as well as the evaluation results of the impact of absolute and relative shaft misalignment in the magnetic fluid seal, centrifugal forces arising during shaft rotation, on the retained pressure drop by the seal. A significant reduction in the retained pressure drop occurs at an eccentricity of up to 40% of the working gap, and with further increases in eccentricity, the rate of pressure drop reduction slows down. The significance of the results lies in the potential utilization of the provided numerical model, as well as the outcomes of physical and mathematical experiments, in the development of a magnetic fluid seal operating with significant misalignment between the rotating shaft and the housing. The dimensionless dependencies obtained allow for consideration, during the design stage, of the reduction in retained pressure drop with shaft eccentricity, taking into account the magnitude of the working gap, magnetic induction, and linear velocity.
Keywords: magnetic fluid, magnetic fluid seal, eccentricity, pressure drop, numerical modeling.

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Influența excentricității arborelui motorului electric asupra performanței unui dispozitiv de etanșare cu fluid magnetic de mare viteză

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Abstract. Scopul lucrării este elaborarea recomandărilor cu privire la modul în care trebuie luate în considerare în faza de proiectare și posibilitatea de a opera un etanșant cu fluid magnetic cu o excentricitate semnificativă a arborelui de rotație în raport cu atașamentele stâlpilor staționari. Acest obiectiv este atins prin cercetare experimentală, selectarea ecuațiilor necesare, condițiilor la limită, ipotezelor și proprietăților fizice ale fluidului magnetic atunci când se construiește un model matematic numeric al intervalului de lucru al sigilantului fluid magnetic. Cele mai importante rezultate ale lucrării sunt distribuțiile obținute și analizate ale câmpului magnetic și ale câmpului vitezelor și presiunii în fluidul magnetic, rezultatele evaluării influenței dezalinierii absolute și relative a arborelui de etanșare a fluidului magnetic, forțele centrifuge apărute în timpul rotației arborelui pe căderea de presiune reținută de etanșant. S-a demonstrat că dispozitivul de etanșare cu fluid magnetic își păstrează eficiența și performanța chiar și la o excentricitate a arborelui rotativ de până la 80% din spațiul de lucru. O scădere semnificativă a căderii de presiune reținută are loc la o excentricitate de până la 40% din spațiul de lucru, iar cu o creștere suplimentară a excentricității, rata de scădere a presiunii reținute încetinește. Semnificația rezultatelor constă în posibilitatea utilizării modelului numeric prezentat, a rezultatelor experimentelor fizice și matematice în dezvoltarea unui dispozitiv de etanșare cu fluid magnetic care funcționează cu o dezaliniere semnificativă a arborelui rotativ și a carcasei. Dependențele adimensionale obținute fac posibilă în faza de proiectare să se țină cont de scăderea căderii de presiune reținută cu excentricitatea arborelui, ținând cont de dimensiunea spațiului de lucru, inducția magnetică și viteza liniară.
Cuvinte-cheie: fluid magnetic, etanșare fluidului magnetic, excentricitate, cădere de presiune, modelare numerică.

Влияние эксцентриситета вала электродвигателя на работоспособность высокоскоростного магнитожидкостного герметизатора

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Аннотация. Целью работы является выработка рекомендаций по способу учёта на этапе проектирования и возможности эксплуатации магнитожидкостного герметизатора со значительным эксцентриситетом вращающегося вала относительно неподвижных полюсных приставок. Поставленная цель достигается за счёт проведённого экспериментального исследования, выбора необходимых уравнения, граничных условий, допущений и физических свойств магнитной жидкости при построении численной математической модели рабочего зазора магнитожидкостного герметизатора. Наиболее важными результатами работы являются полученные и проанализированные распределения магнитного поля и поля скоростей и давления в магнитной жидкости, результаты оценки влияния абсолютной и относительной несоосности вала магнитожидкостного герметизатора, центробежных усилий, возникающих при вращении вала, на удерживаемый герметизатором перепад давления. Показано, что магнитожидкостный герметизатор сохраняет свою эффективность и работоспособность даже при эксцентриситете вращающегося вала до 80 % от величины рабочего зазора. Существенное снижение удерживаемого перепада давления происходит при эксцентриситете до 40 % от величины рабочего зазора, а при дальнейшем увеличении эксцентриситета темп снижения удерживаемого давления замедляется. При увеличении эксцентриситета до 80 % от рабочего зазора удерживаемое давление снижается на 45 % по сравнению с соосным расположением вала и полюсных приставок. Выполнение зубцов на валу даёт прирост удерживаемого перепада давления при линейной скорости на поверхности вала 25 м/с в среднем на 30 % по сравнению с конструкцией магнитожидкостного герметизатора с зубцами на неподвижных полюсных приставках. Значимость результатов состоит в возможности использования приведённой численной модели, результатов физического и математического экспериментов при разработке магнитожидкостного герметизатора, работающего со значительной несоосностью вращающегося вала и корпуса. Полученные безразмерные зависимости позволяют на этапе проектирования учесть уменьшение удерживаемого перепада давления при эксцентриситете вала с учётом величины рабочего зазора, магнитной индукции и линейной скорости.

***Ключевые слова*:** магнитная жидкость, магнитожидкостный герметизатор, эксцентриситет, перепад давления, численное моделирование.

INTRODUCTION

Sealing rotating shafts, bearings, and other mechanical systems using magnetic fluid (MF), confined in the sealing gap by a magnetic field, is a successful and simultaneously promising technical solution [1-4].

Magnetic fluids used in magnetic fluid seals are colloidal suspensions of magnetic nanoscale particles (e.g., magnetite – Fe2O3) with a diameter of approximately 10 nm in a carrier fluid (e.g., synthetic oil or silicone fluid). Particles of such sizes are in constant motion due to the thermal Brownian motion of the carrier fluid molecules, forming a stable colloidal solution. During production, a surfactant (e.g., oleic acid) is introduced into the carrier fluid, covering the surface of the magnetic particles, thereby protecting them from aggregation and settling of formed multi-particle magnetic aggregates under the influence of gravitational and magnetic forces. MFs with nanoscale particles exhibit high stability and durability of the colloidal system, with a lifespan exceeding 20 years [5-8].

Figure 1 illustrates the design of a typical MF Seal for a rotating shaft. Housing 1, made of non-magnetic material, is attached to a ring permanent magnet 2 magnetized in the axial direction.

On each side of the permanent magnet, there are pole attachments three made of ferromagnetic material with surfaces facing the ferromagnetic shaft 4, featuring teeth – concentrators of the magnetic field. A gap filled with MF 5 is present between the shaft and the pole attachments. The MF in the gap is retained by the magnetic field generated by the permanent magnet.

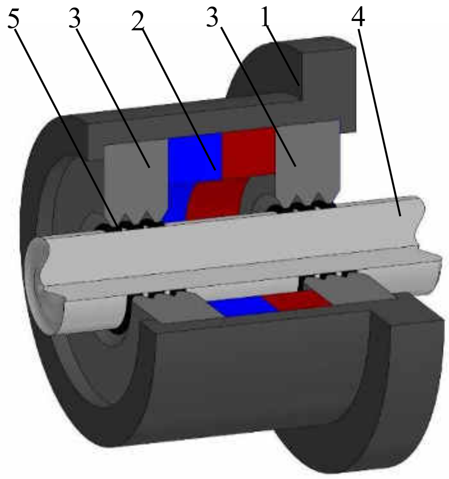
The thickness of the gap with MF typically ranges from 0.1 to 0.3 mm for small-diameter shafts and up to 0.5 mm for shafts with a diameter of 100 mm or more [9-12].

The physics of the process ensuring the sealing of MF seals differs significantly from the mechanisms observed in other types of seals.

In connection with this, there is an opinion that the application of MF seal is primarily limited to precision machinery.

This limitation is justified by the specific properties of MFs, the necessity of using high-energy permanent magnets, and the relatively high precision required in the manufacture of MF seal, along with the costliness of high-dispersion magnetic fluids.

This combination of factors is the reason why MF Seals currently do not find widespread use in electric machines.



1 – housing, 2 – ring permanent magnet, 3 – pole attachments, 4 –shaft, 5 – magnetic fluid.

Fig. 1. Magnetic fluid seal of a typical design.

Despite the mentioned challenges, the evident advantages of MF Seals, such as complete hermeticity and low rotational resistance, contribute to a growing interest in their application in demanding operational conditions. In these conditions, they are often used in combination with seals of traditional designs. MF seals has found application in chemical, biochemical, and pharmaceutical reactors, as well as in machines in the petroleum refining industry, which frequently operate under conditions involving uneven motion, oscillations and/or vibrations, high temperatures, and exposure to aggressive environments in the form of vapors and aerosols.

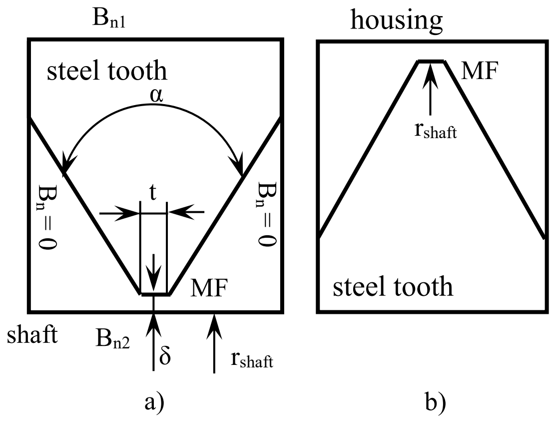
A significant consideration in designing MF seals for installation on general-purpose industrial electric motors instead of traditional bearing covers with labyrinth seals is the permissible eccentricity, the displacement of the rotation axis of the shaft relative to the axis of stationary pole attachments, and the runout of the output end of the motor shaft.

In the overwhelming majority of publications and studies, the magnetic system of MF seals is considered as a set of concentric circles. Mentions of eccentricity in publications are rare, not always leading to its numerical specification, and even less frequently being considered in mathematical models [13-17].

Determining the critical pressure of MF seals in dynamic conditions is significantly more complex than in static conditions. This complexity arises because the MF in the working gap is subject not only to magnetic forces but also to flow processes and centrifugal forces. The latter influences the position of the MF in the gap and alters the retained pressure [18-23].

RESEARCH METHODS

The inability to accurately model the redistribution of the magnetic field that occurs with the eccentricity of MF seals in a two-dimensional or axisymmetric setup necessitates the use of a three-dimensional representation, despite the significantly increased calculation time. Numerical calculation of the hydrodynamic flow in the MF seal gap, due to large gradients compared to the magnetic field, requires a much finer mesh in the computational domain. Therefore, the decision was made to perform a three-dimensional calculation of the magnetic field redistribution at one tooth pitch of the MF seal with shaft eccentricity. Based on the obtained distribution of magnetic induction, a calculation of the interacting magnetic and hydrodynamic fields was carried out in a two-dimensional setup. The computational domain with the utilized boundary conditions is depicted in Fig. 2 and consists of a stationary steel tooth mounted on the housing (Fig. 2a) or located on the shaft and rotating (Fig. 2b), both filled with magnetic fluid in the working gap.



a – tooth on the pole; b – tooth on the shaft.

Fig. 2. Investigated area of the magnetic fluid sealer.

Hydrodynamic calculation of laminar flow of viscous incompressible liquid is carried out based on the Navier-Stokes equation:

$$\rho\left[\left(\bar{v}\cdot\nabla\right)\bar{v}\right]=-\nabla p+\nabla\left(\eta\left(\nabla\bar{v}+\left(\nabla\bar{v}\right)^{T}\right)\right)+\sum\bar{F} \quad (1)$$

where $\bar{v}$, p, ρ and η are, respectively, velocity, pressure, density, and dynamic viscosity of the magnetic fluid; $\bar{F}$ is the sum of the volumetric forces acting on the MF.

As assumptions in the calculation, it is considered that the viscosity of the MF is constant and independent of the magnetic induction, shear rate, and temperature.

The magnetic fluid is retained in the seal gap due to an additional volumetric magnetic force:

$$\bar{F} = \mu_0 M \nabla \bar{H}, \tag{2}$$

where M is the magnetization of the FF; $\bar{H}$ is the magnetic field intensity, which establishes the connection between the magnetic and hydrodynamic fields in the calculation.

The gravitational force is significantly smaller than the magnetic force and is not considered in the calculation.

The calculation of the magnetic field is performed using Maxwell's equations. The magnetic properties of the tooth are defined by the magnetization curve of steel 3.

The MF is in a state of saturation, and its relative magnetic permeability is assumed to be 1.

The magnetic field is specified through the normal component of the magnetic induction vector at the boundaries, as shown in Fig. 2a.

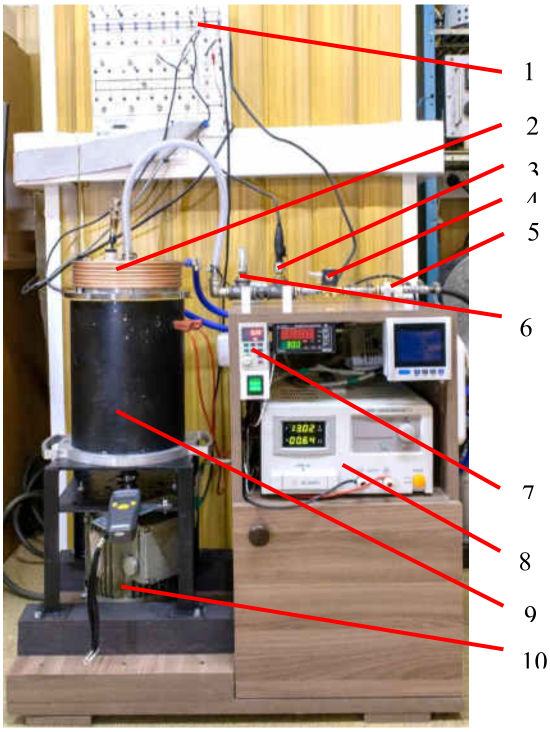
Because of the combined calculation of the hydrodynamic and magnetic fields, distributions of velocity and pressure fields in the working gap of the MF seal are obtained. The pressure distribution inside the MF follows the Bernoulli equation:

$$\rho \frac{v^2}{2} + p - \mu_0 \int_0^H M dH = const. \tag{3}$$

To verify the results obtained through numerical modeling and experimental investigation of the influence of shaft eccentricity on the performance of the MF seal, an experimental setup has been developed, as shown in Fig. 2.

An asynchronous motor rotates the shaft of an electromagnet. The motor's rotation frequency is controlled by a frequency converter.

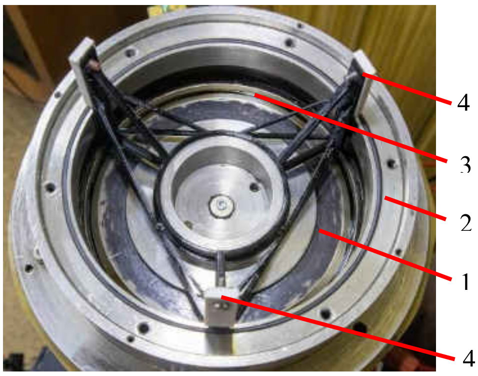
The MF seal is installed on the upper part of the electromagnet. The stationary pole attachment and the seal shaft form the working gap of the MF seal, which is filled with MF.



1 – connector Input/Output board, 2 – magnetic fluid seal, 3 – pressure sensor, 4 – inlet valve, 5 – needle faucet, 6 – exhaust valve, 7 – frequency inverter control panel, 8 – DC power supply, 9 – electromagnet, 10 - electric motor.

Fig. 3. Experimental bench.

The eccentricity of the working gap is defined by the displacement of the MF seal housing relative to the shaft. The precision of their mutual positioning is monitored by three spring-loaded probes set into the working gap with a 120-degree displacement, as shown in Fig. 4.



1 – seal shaft, 2 – pole attachment, 3 – working gap, *4 – spring-loaded probes set.*

Fig. 4. Positioning of the housing and shaft of the magnetic fluid seal.

RESULTS

The results obtained from the numerical model calculation of magnetic induction distribution, flow velocity, and pressure in the *MF* are depicted in Fig. 4. In Fig. 4a, the magnetic field pattern is shown with contour lines of equal induction, representing the location of the free surface of the MF plug when the shaft of the *MF* seal is stationary. The magnetic induction value on the shaft surface varies from 0.81 T in the minimum gap zone to 0.34 T in the inter-tooth space. Using a known formula, it is possible to analytically calculate the maximum pressure drop retained by the seal when the shaft is stationary [2, 3, 4]

$$\Delta p = M_s(B_{\max} - B_{\min}) = \\ = 40000(0.81 - 0.34) = 18,8\ kPa.$$

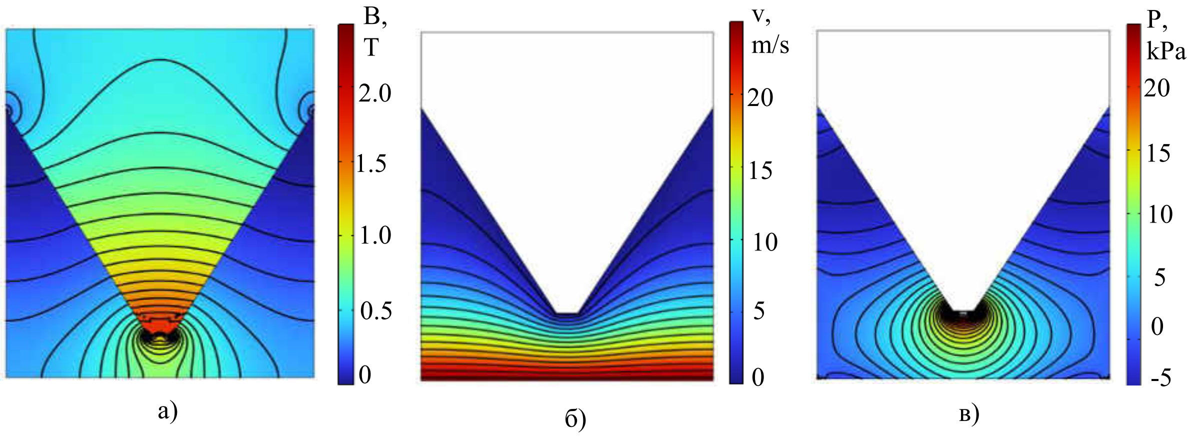
In Fig. 5b, the velocity field in the MF during the rotation of the MF seal shaft is depicted. It can be observed that under the tooth tip, in the region of the minimum gap, the velocity gradient is maximized. When installing the MF seal on two-pole alternating current electric motors with large-diameter shafts (over 150 mm) and a rotational frequency of 3000 or 3600 rpm, intense heat generation and significant heating of the MF occur due to the high velocity gradient in this zone. An effective method to reduce viscous losses in the MF is to increase the size of the working gap, leading to a reduction in the shear rate of MF layers under the tooth tip.

The volumetric magnetic force acting on the *MF* under the influence of the magnetic field creates magnetic pressure in the liquid, the distribution of which corresponds to the lines of equal induction shown in Fig. 5a.

Figure 5c shows the pressure distribution in the working gap of the *MF* seal at a linear velocity on the rotating shaft's surface equal to 25 m/s. In this pressure distribution, besides the volumetric magnetic force, centrifugal forces have a significant impact, trying to detach the *MF* from the shaft's surface. This leads to a decrease in the retained pressure drop by the seal.

In high-speed *MF* seals units, to compensate for the decrease in retained pressure due to centrifugal forces, it is necessary to increase the magnitude of the magnetic induction in the working gap, thereby enhancing the magnetic force acting on the *MF*.

The literature-recommended limit on the maximum induction in the *MF* seal gap of up to 0.8 T is associated with a significant redistribution of the magnetic phase within the liquid when the shaft is stationary and the entrainment of magnetic particles into the area of the minimum gap. This increases the initial moment of *MF* grabbing, which can be neglected in general-purpose industrial motors.



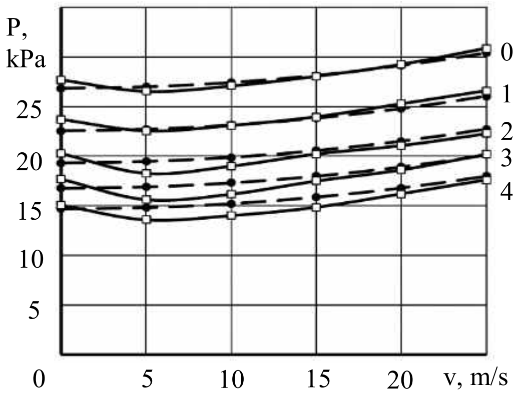
a – magnetic field, б – velocity field at v = 25 m/s, в – pressure distribution at v = 25 m/s.

Fig. 5. Results of numerical calculation of physical fields.

Figure 6 illustrates the variation in the retained pressure drop with the increase in linear velocity on the surface of the shaft for the design of the *MF* seal with teeth and concentrators of the magnetic field located on the engine shaft, as shown in Fig. 2b.

The maximum induction value in the working gap has been increased to 1.2 T. In this case, the magnetic field in the inter-tooth space has also grown to a value of 0.53 T. The retained pressure drop by the seal with a stationary shaft and no eccentricity (curve 0) is 26.8 kPa.

In *MF* seals with teeth on the shaft, as the rotation frequency increases, centrifugal forces shift the boundary of the *MF* droplet along the tooth towards the region of maximum induction, causing an increase in the retained pressure drop.



0 – eccentricity 0 mm (0 a.u.), 1 – 0.1 mm (0.2 a.u.), 2 – 0.2 mm (0.4 a.u.), 3 – 0.3 mm (0.6 a.u.), 4 – 0.4 mm (0.8 a.u.).

Fig. 6 Pressures from shaft speed. Tooth on the shaft, gap 0.5 mm.

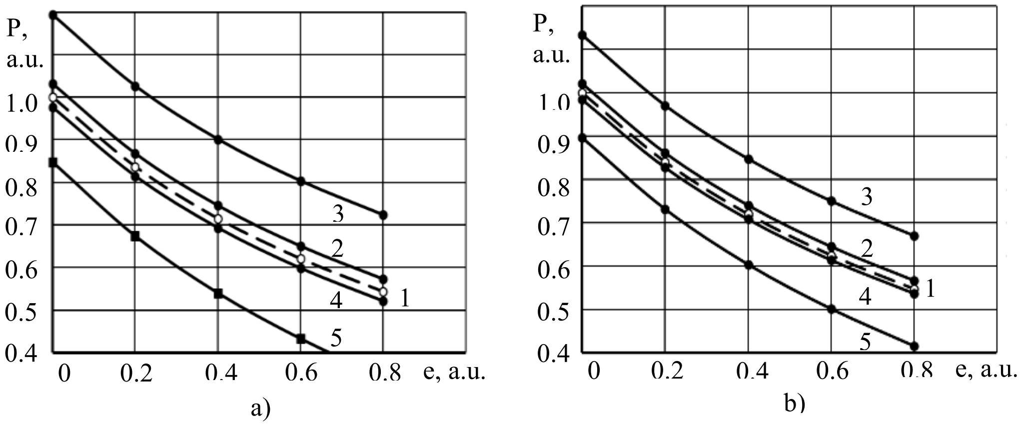
With eccentricity, the distribution of the magnetic field in the *MF* seal gap changes. The magnetic induction value increases in the area of the minimum gap, while it decreases on the side of the maximum gap. The retained pressure drop by the *MF* seal is determined by the smaller induction gradient observed in the region of the maximum gap.

Lines 1-4 in Fig. 6 show the change in the retained pressure drop of the *MF* seal with shaft eccentricity. From the analysis of the dependencies, it is evident that with eccentricity up to 0.2 mm (40% of the working gap size), there is a significant decrease in the retained pressure drop. With further increases in eccentricity, the rate of pressure drop reduction slows down, which is associated with the nonlinearity of the magnetic properties of the steel tooth and its saturation in the area of the minimum gap.

Solid lines in Fig. 6 represent the results of experimental research, while dashed lines represent the results of mathematical modeling. It can be observed that there is good agreement between the experiment and the model. However, in a real MF seal, with the initiation of shaft movement, the retained pressure drop initially decreases. This is because the non-ideal geometry of the *MF* seal tooth and the vibration from the rotating shaft reduce the stability of the MF plug, which is not considered in the model.

The dependencies in Fig. 7 illustrate the variation in the retained pressure drop of the *MF* seal with increasing relative eccentricity of the shaft. As the eccentricity increases to 80% of the gap, the retained pressure decreases by 45% compared to the concentric arrangement of the shaft and pole tips. However, despite such a significant eccentricity, the seal maintains its tightness even with a rotating shaft.

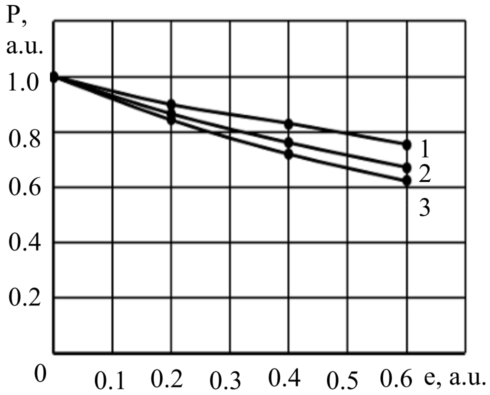
Comparing the dependencies in Fig. 7a and Fig. 7b, it can be concluded that increasing the magnitude of magnetic induction in the gap of the *MF* seal does not affect the change in the relative retained pressure drop with eccentricity. However, it reduces the pressure variation with increasing linear speed on the surface of the shaft, enhancing the stability of the seal's characteristics. At a linear speed on the shaft surface of 25 m/s, having teeth on the shaft rather than on the stationary pole tips increases the retained pressure drop by 25% at an induction of 1.2 T and by 35% at an induction of 0.8 T, proving to be an effective measure for retaining the *MF* in the gap of high-speed MF seals.



a - induction 0.8 T, b - induction 1.2 T. Gap 0.5 mm. 1 - v = 0 m/s, 2 - v = 10 m/s, tooth on the shaft, 3 - v = 25 m/s, tooth on the shaft, 4 - v = 10 m/s, tooth on the pole, 5 - v = 25 m/s, tooth on the pole.

Fig. 7 Pressure from relative eccentricity.

In Fig. 8, the reduction in retained pressure drop with an increase in the relative eccentricity of the shaft is shown for different values of the working gap of the *MF* seal.



1 – gap 0.1 mm, 2 – 0.3 mm, 3 –0.5 mm.

Fig. 8 Pressure from the relative eccentricity value for different gap values.

When striving to maintain a minimal gap in the seal at 0.1 mm and a shaft eccentricity of 60% of the gap, which is 0.06 mm in absolute terms, the retained pressure drop decreases by 23% compared to a concentric arrangement of the shaft and pole tips. If, during the design stage, the initial working gap of the *MF* seal is increased to 0.3 mm, the relative eccentricity decreases to 20%, and the retained pressure drop by the seal decreases by 12%, enhancing the stability and predictability of the *MF* seal characteristics.

CONCLUSIONS

The conducted research and analysis of the results have demonstrated that the *MF* seal maintains its efficiency and performance even with an eccentricity of the rotating shaft up to 80% of the working gap magnitude.

During the design stage of the *MF* seal, the influence of shaft eccentricity can be accurately considered based on dimensionless dependencies obtained through numerical modeling, taking into account the working gap magnitude, magnetic induction, and linear velocity at the shaft surface.

In electric machines of general industrial application, the starting torque of the rotor after prolonged downtime is not critically important. An effective method to compensate for the decrease in retained pressure drop, considering the evolving eccentricity and shaft vibrations during bearing wear, is to increase the maximum induction in the working gap of the seal.

With the increase in the diameter of the sealed shaft, permissible deviations in the geometric dimensions of electric machine components and the MF seal inevitably grow. Increasing the working gap during the design stage allows for reducing the relative eccentricity of the shaft and enhancing the stability of MF seal characteristics under the same actual misalignments. Achieving the required magnetic induction value in the MF seal gap is possible by utilizing modern high-coercivity magnets.

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