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The Specific Features of High-Velocity Magnetic Fluid Sealing Complexes

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Abstract: Studies on the impact of magnetic and centrifugal forces in hermetic magnetic-liquid seals were presented. The results of numerical calculations on the decrease of the impact of the centrifugal force through the use of magnetic flux concentrators on a rotating shaft were shown. It has been demonstrated that there is a greater possibility for the use of hermetic magnetic-liquid seals in axially symmetrical clearances in gas-steam separation for coke gases.

Keywords: ferromagnetic, axial clearance

1 Problem statement and the analysis of the latest publications.

The factors, which are insignificant at small velocities, become rather considerable with increasing the shaft surface velocity as far as the magnetic fluid sealing complex (MFSC) is concerned. They impact both on the pressure drop restrained by the sealer and on MFS operational resource.

Primarily, MFSC hydrodynamics is complicated because centrifugal forces are added to magnetic forces to restrain magnetic fluid (MF) in the working gap, and, moreover, radial pressure gradient $\sim v^2/R$ due to fluid motion is added to axial pressure gradient resulted from magnetic volume force $\mu_f M_S \nabla \hat{I}$.

While the rotation frequency and the sealed shaft diameter are increasing the linear velocity of the shaft surface and the value of centrifugal forces are also increasing. These forces throw the fluid off the shaft surface which

causes a number of negative consequences, including the complete ejection of magnetic fluid from the working gap.

The problem of flow stability between the coaxial cylinders (Taylor Couette flow) is considered classical for hydrodynamical and magnetohydrodynamical stability. The specific feature of the MFS gap in comparison with the gap in the hydrodynamics of conventional fluid is the fact that it is of rather complicated geometrical form (it is necessary for focusing the magnetic flux and the fluid volume is limited by not only the solid but also by the deformed free boundaries. The problem is also complicated by the following factors: the probable non-uniformity of the working gap, the non-linear characteristics of magnetization and magnetological properties of the magnetic fluid itself.

Moreover, these rotational flows are also specific because the secondary laminar regime immediately occurs when the primary laminar flow loses its stability but not turbulent flow, the system of period laminar flows straps on the primary laminar flow along the shaft axis of meridian vortex, so called Taylor vortexes [1].

Before they emerge, the fluid's laminar flow is damaged because the axial wave process vortexes emerge, the velocity of strapped waves being equal to 1/3 of the velocity on the shaft surface and the third harmonic of the vibrational process emerging (mode $m = 2$) [2]. Azimuth component of the velocity vector will be divided into a series of odd harmonics and around the gap there will be the integre number of waves, the process being discrete: screw lines of the current are closed and the flow is quasilaminar.

The paper [1] shows that secondary laminar flow is unstable even in the weak magnetic field. It is in a good accord with the data referred to in [3], where they indicate that the increase in shaft rotation can take place at the modes $m = 0$ and $m = 1$ in weak magnetic fields. It should be kept in mind that it is practically impossible to determine which mode is the most unstable because the mode depends on the value of magnetic field, MF properties etc.

The experience in bench testing [4] and the author's survey for MFS service in "Ferrohydrodynamics" shows that the problems of MF ejection from the gap are due to the excess in the linear velocity in the gap of 15 m/sec.

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It was noticed visually that fine disperse MF drops are thrown off to the azimuthal direction, which testifies to the fact that the development of instable free surface causes the ejection. When the linear velocity is increasing up to 35 m/sec, it is impossible to restrain magnetic fluid in the MFSC working zone of any classical design. Primarily, it depends on the fact that during the execution of concentrators of the magnetic flux, the action of centrifugal forces on magnetic drives of the sealer coincides with the direction of magnetic induction gradient.

It is known that the issues of friction in high-velocity magnetic fluid sealing complexes have been described in detail. The friction in MFSCs is insignificant, but some researches [6, 7] etc., note that it should not be neglected because viscous friction leads to MF heating, its subsequent evaporation and desorption of the molecules of surface active substances from the particles surface. The both factors put magnetic fluid out of operation in the sealer gap and the magnetic fluid sealer itself. At the same time the analysis of the operation of more than 4000 magnetic fluid seal complexes under various operational conditions conducted by “Ferrohydrodynamica” shows that the failures, due to magnetic fluid temperature heating, have not been registered. The contradiction is explained in [8]: for high-velocity MFSCs it is important to take into account thermomagnetic convection which results in the intensive circulation flow in the meridional plane in the narrow working gap above the pole, thus reducing the fluid temperature in this region. Generally speaking, it is the positive factor, therefore special measures for additional flows should not be taken.

To increase the operational capability of high-velocity magnetic fluid seals, it is necessary to take a number of measures for restraining magnetic fluid in the working gap, excluding the impact of centrifugal forces by means of MFSC design changes.

The execution of shaft labyrinths is the most effective and widely used method in sealing technology (hydrodynamical, impeller, labyrinth, slot seals, etc.) for reducing the influence of centrifugal forces [9]. Tooth concentrators of the magnetic flux can similarly be made on the shaft but not on the magnetic circuits. In this case centrifugal forces throw magnetic fluid to the tooth top into the region of maximal magnetic field blocking the decrease in the critical pressure drop and magnetic fluid throw-off from the gap.

In [3] the author attracts the attention to the fact that there has not been obtained the criterion for classifying rotation as stable or instable for Couette flow of complex geometry of the working gap. The nature of instable flow of magnetic fluid in the MFSC is determined for each partic-

ular case because they are determined by magnetic fluid properties.

Therefore it is only direct numerical calculation that can define the flow character. The aim of the research is to analyse the interaction of magnetic and centrifugal forces in the seal working gap and consequently, to design the magnetic fluid seal complex for wider applications.

2 The research description.

It is expedient to model the hydrodynamic processes in the working gap of the complex by means of numerical methods based on field models containing numerous elements. The generalized methodological approach to the analysis of physical fields in the working gap can be selected on the basis of the method of multiphysical modelling of the set of interconnected processes of various physical nature [10].

Taking into account the main characteristics of the magnetic fluid sealing complex, the parameters of the magnetic field in the working gap are determined and the mathematical model of interconnected non-linear magnetic and hydrodynamical processes in the MFSC gap is described [11]. The model is calculated by means of the numerical method of finite elements with Comsol package [12].

The sealer is characterized by axial symmetry, therefore the field problem is determined in 2D cylindrical system of coordinates. Fig. 1 shows the fields for the analysis of the magnetic field with a variety of locations of magnetic flux concentrators and their shapes. It contains three types of magnetic materials: constant magnets 1, magnetized in the axial direction; ferromagnetic material of the poles of magnetic System 2 and rotating shaft 4; and the field of magnetic fluid 3.

The magnetic circuit shaft of 210mm dia was investigated, its rotational frequency being 3000 rev/min.

The distribution of the force lines of the magnetic field and magnetic induction are also shown in Fig. 1. Fig. 2 shows the distribution of the module of the magnetic induction vector under one concentrator of the magnetic flux (the pole) for different design circuits.

The curves of magnetic induction distribution for all the circuits concerned are of the similar wave like character. It is necessary to note that the location of teeth on the shaft enables to decrease the field drop under the last outer tooth of the pole (up to 10%), which emerges due to the end effect and bulging of the force lines of the magnetic field, and it is extremely important because it is the place where magnetic fluid is ejected.

Figure 1: The distribution of the module of the magnetic induction vector (in color) and the force lines of the vector magnetic potential for: a) – the sealer of classical design; b) – with teeth on the magnetic circuit of trapezoid form and “inner” teeth of rectangular form on the shaft; c) – with “inner” teeth of trapezoid form on the shaft; d) – with “inner” teeth of cylindrical form on the shaft; e) – with the teeth of trapezoid form on the magnetic circuit and “outer” teeth on the shaft.

Figure 2: The distribution of the module of the magnetic induction vector $|B|$ under one pole of the sealer along the midline in the gap: a) – with teeth on the magnetic circuit; b) – with teeth on the magnetic circuit of trapezoidal form and “inner” teeth of rectangular form on the shaft; c) – with “inner” teeth of trapezoidal form on the shaft; d) – with “inner” teeth of cylindrical form on the shaft.

The analysis of Fig. 2b shows the decrease of the magnetic field induction approximately by 15% in comparison with the other versions concerned.

The hydrodynamic problem of the magnetic fluid flux in the working gap was solved under the following assumptions:

Figure 3: The distribution of azimuth component of the velocity of magnetic fluid flux in the gap of the magnetic system at the surface velocity $v_j = 33i/\tilde{n}$ for: a) – the sealer of classical design; b) – with teeth on the magnetic circuit of trapezoidal form and “inner” teeth of rectangular form on the shaft; c) – with “inner” teeth of trapezoidal form on the shaft.

- the position of the free boundary of dynamic magnetic fluid coincides with that of static magnetic fluid;
- the distribution of the magnetic field in the gap of the magnetic fluid seal complex coincides with the field distribution in Fig. 1 and 2 at this position of magnetic fluid boundary.

The distribution of the azimuth component of velocity in the working gap is shown in Fig. 3. The design results show the monotonous decrease in the velocity from 33 m/sec on the shaft surface to 0 on the magnetic pole surface. The most dramatic change of the velocity takes place directly under the teeth of the magnetic system, i.e. in the fields of minimal gap.

The velocity field in the end part of the magnetic fluid complex was analysed separately (the last outer tooth of the pole). Fig. 3b and 3c show that the value of the maximum velocity in the working gap for all the versions of magnetic flux concentrator locations on the shaft is lower than for the MFSC of a conventional design (see Fig. 3a). The distribution of the azimuth component of velocity for the location of the teeth of trapezoidal and rectangular forms on the shaft is practically identical, therefore Fig. 3 only shows the distribution for one version. It can be concluded that the geometric form of the concentrators of the magnetic flux makes an insignificant effect on velocity distribution in the gap.

Another step of the research consisted in the analysis of vortex structures which were formed in the magnetic fluid of radial and azimuth components of velocity.

Fig. 4 shows vortex structures. It is evident that the structure of the highest vortex velocity takes place in the last right end region which is explained by the large of free fluid boundary. In other slot regions there are also vortex

structures but the maximum value of velocity in them is much less which is stipulated by the intensive braking of fluid due to the nearest walls of magnetic circuits, and they do not make a considerable impact on magnetic fluid ejection from the working gap.

Fig. 4 shows that directions of the actions of the vortex and magnetic induction vector are directly opposite. The comparison of the vortex structures for the MFSC of conventional design with the other versions where magnetic flux concentrators were positioned on the shaft shows the decrease of the absolute value of vortex velocity when the given design was used.

For the hydrodynamical processes in the working gap, the scheme of locating teeth both on the magnetic circuit and on the “outer” teeth of the shaft is the most expedient. However, the circuit requires high precision in implementation; besides, the experience obtained in “Ferrohydrodynamica” for MFSC mounting shows that it is impossible to carry out mounting technologically in many cases. Therefore we compared the versions shown in Fig. 4b and 4c. The version in Fig. 4c was selected. In spite of the fact that vortex structures in version of Fig. 4b had lower velocities but the decrease in magnetic induction in the gap in comparison with the version in Fig. 4b by approximately 15% was decisive for the selection of the circuit with “inner” teeth of cylindrical form on the shaft.

The circuit has been introduced for the design of end seals of a coke gas-blower instead of conventional labyrinth seals. The frequency of shaft rotation was 3000 rev/min, the excess pressure of suction – 50 Pa, the excess pressure of discharge – 8000...15000 Pa. There was the axial displacement of the shaft in operation to 2 mm. Therefore the scheme shown in Fig. 1e could not be implemented in any way.

Figure 4: The vortex structures formed by radial u_r and axial u_z components of velocity and the absolute value of velocity $\sqrt{u_r^2 + u_z^2}$, in colour for: a) – the sealer of classical design; b) – with teeth of the magnetic circuit of trapezoidal form and “inner” teeth of rectangular form on the shaft; c) – with “inner” tooth of cylindrical form on the shaft; d) – with teeth of trapezoidal form on the magnetic circuit and “outer” teeth on the shaft.

MFSC (Fig. 5) is executed as a monoblock including the labyrinth sealer and the magnetic fluid sealer, the ferromagnetic steel bush 2 is installed on shaft 1. The bush section located opposite the magnetic circuits is implemented as 35 mm long flange and 160 mm outer diameter. Ring teeth are cut on the outer surface of the flange. The tooth profile is the same as the metrical thread profile. But generally the design of the magnetic fluid sealing complex is similar to other “Ferrohydrodynamica” designs which are known as reliable and efficient.

3 Conclusions

1. The interaction of magnetic and centrifugal forces in MFSC working gap is studied by means of multiphysical modelling. The wider potentialities for seal applications at high linear forces in the gap are demonstrated.
2. It is shown that the location of the magnetic flux concentrators on the shaft results in the decrease in the magnetic field drop under the last outer tooth by 10%.
3. It is also shown that the location of magnetic flux concentrators on the shaft reduces the value of the vortex velocity in the end part of the sealer, the directions of

Figure 5: The drawing of the MFSC gas-blower.

the vortex and the magnetic induction vector are opposite each other.

4. The results of multiphysical modeling were used to design the magnetic fluid sealing complex for the coke gas-blower.

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