EXPERIMENTAL RESEARCH OF ROTATION TRANSMISSION DEVICE IN HIGH AMBIENT HYDROSTATIC PRESSURE CONDITIONS

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Abstract

In this article, the results of experimental research of rotation transmission device prototype characteristics are presented. The device can be used in different types of deep-water equipment. To confirm theoretical calculations of the device characteristics a prototype was designed and produced. The scheme of the prototype is explained in the article. The experiments were conducted for three different work fluids. Leakage volume dependencies on the operating pressure and shaft rotation frequency are examined. Friction torque and breakaway torque dependencies on operating pressure are examined. Efficiency coefficient and time of establishment of thermal regime of the rotation transmission device are found.

Keywords: machinery; deep-sea mining; power unit; hydraulics

1. Introduction

The World ocean is one of the most important sources of life sustain on Earth. The knowledge about its resources shows that in future resource depletion on Earth surface can be stopped by changing mining priority from land to ocean floor. Therefore, the World ocean research and development can significantly influence on solving a number of global problems.

Despite great prospects of using of bowels of the World Ocean, society at the modern stage of technical development is mostly focused on oil and gas production at easy accessible districts near continents. In most cases modern methods of World Ocean research requires application of universal underwater engineering. Recently increased interest in the development of the seabed and the ocean floor demands equipment that is capable of solving various tasks in scientific, ecological spheres as well as constructing of underwater objects etc.

A common problem that comes up during designing of underwater objects for different purposes is to transmit rotation through strong housing into high ambient hydrostatic pressure environment, and an essential question is to use proper shaft seal. In modern industry, it is mostly common to use translatory motion to transmit energy in deep sea environment. This fact is provided inter alia by different conditions of friction: when an interior element moves translatory relative to

an outer element, contact area is changing during one operating cycle. When an interior element rotates in an outer element, there is a problem of contact stresses and constant friction area. Therefore, the problem of gap seal development is mostly solved for translatory motion, but this solutions can be not quite suitable for rotation. Wang et. al [1] decided to use a thin layer of polyetheretherketone (PEEK) as a seal material. However, while the inner element rotates with high speed relative to the outer element a covering on the surface of the inner element has a high probability to be erased. Wang, Zhi, Hu et. al [2] proved that gap seal has significant advantages over seal rings. As for application of hydraulic oil to transfer seawater pressure Selwyn and McCoy [3, 4] designed a new reciprocating hydrostatic motor.

An eccentricity of a gap seal has a large influence on leakage volume. Wieczorek and Ivantysynova examined this fact in their work [5]. To increase hydrodynamic forces that reduce the eccentricity of a piston that rotates and moves in axial direction in a bushing of axial piston pump Ivantysynova and Lasaar [6] proposed a design of a special barreled shape of the piston. This can also be not acceptable for a rotating shaft as it may cause some additional contact stresses, while variable section does not allow using minimal gap for the whole seal length. Hydrodynamic pressure can reach high values and handle external load in case of axial piston machines as it was shown in a work by Pelosi [7], though axial motion has a significant impact on the pressure that is built in the gap, therefore, this solution also may not be basic for advanced gap seal design.

To implement the task the rotation transmission device in High Ambient Hydrostatic Pressure Conditions was designed. Its constructive scheme and mathematical model for its characteristics calculation was examined in [8]. To verify the calculation methods of the rotation transmission device characteristics described in [8], to confirm its operability and to determine its characteristics experimentally its prototype was designed. A number of experiments was conducted under the following conditions: 1) ambient pressure 20 – 60 MPa, 2) rotation frequency 10 – 200 1/s, 3) torque 10 – 300 N m, 4) the research was conducted under actual conditions. The prototype was manufactured and comprehensively tested. The prototype design is shown on fig. 1.

2. The rotation transmission device prototype

The rotation transmission device prototype works in a following way:

Elastic elements 17 provide precompression of the floating bushing 7 through ground-in spherical washer 19 to the frontal surface of the partition 18 of the housing 1. Wherein tightness of the joint between the floating bushing 7 and the washer 19 and the face of the partition 18 of the housing 1.

At the impact of hydrostatic pressure the locking fluid under a pressure, equal to an ambient pressure, is supplied from a source to the primary locking chamber 8. Wherein further compression of the floating bushing 7 and spherical washer 19 to the partition 18 of the housing 1 is provided.

Face seal 5 serves for separation of the environment of the high pressure and the locking liquid. It operates at the pressure drop close to 0. The locking fluid is sealed on account of hydraulic resistance of an annular gap between the floating bushing 7 and the shaft 4.

The floating bushing 7 due to its mobility relative to the spherical surface and mobility of the spherical washer 19 in radial direction under an influence of hydrostatic and hydrodynamic forces is mounted coaxially to the shaft and traces its oscillation. Through the gap seal 7 leakages enter to the secondary locking chamber 9. From the chamber, the leakage is drained to a measuring capacitance. An axial force affecting the shaft 4, is perceived by a thrust bearing.

Fig. 1. The prototype scheme

This prototype is designed for moment transmission up to 20 kg·m at rotation frequency up to 3000 rpm and pressure value not exceeding 60 MPa. Its shaft diameter is 30 mm. The static characteristics of the rotation transmission device are volume losses (leakage), mechanical losses (friction torque, breakaway torque), and locking fluid temperature at the outlet of the device.

3. The experiment

The prototype of the rotation transmission device was tested for three types of locking fluid: spindle oil $\nu = 0,45 \text{ cm}^2/\text{s}$ vacuum oil 1 $\nu = 1,4 \text{ cm}^2/\text{s}$ and vacuum oil 2 $\nu = 3,2 \text{ cm}^2/\text{s}$. Due to the fact, that viscosity of the locking fluid has a major impact on amount of leakage, before the research a series of experiments were conducted to determine kinematic viscosity coefficient ν dependence on temperature.

Leakages were measured at two values of the radial gap between the shaft and the bushing, equal to 0.045 mm and 0.0175 mm. As thermal processes have a big impact on the amount of leakages for the small gaps and flow rates of the locking fluid, the leakages values for every pressure value were measured after establishment of stable thermal conditions – in about one or two hours after the rotation transmission device operation start. This time depends on the locking fluid pressure.

Leakage dependencies of the three oils on operating pressure for the gap of 0.045 mm is presented on Fig.2, and for the gap of 0.0175 mm – on Fig. 3. As it can be seen from the graphs, the dependencies are linear and most affected by the height of the gap between the shaft and the bushing. At the gap of 0.008 mm, vacuum oil 1 and operating pressure of 40 MPa a leakage rate of 10–15 cm^3/min was achieved.

At smaller gap height, values and pressure values of more than 7–8 MPa the bushing stopped its motion relative to the shaft, being pressed to it. The friction torque value dramatically increased. This fact can be explained by the bushing deformation, as its value becomes comparable to the gap height value and by a breakdown of the lubrication film between the bushing and the shaft.

Fig. 2. Leakage rate at the gap height of 0.045 mm

The shaft rotation frequency, changing from 50 to 3000 rpm, showed no impact on the volume of leakage. The leakage volume when the shaft was not rotating at the same operating pressure was different in different experiments and varied from the leakage volume in the case of rotating shaft (Q_H) up to 1.5 Q_H . This fact can be explained by a different eccentricity of the bushing relative to the shaft, and minimal leakage in the case of rotating shaft indirectly shows a centering of the bushing when the shaft was rotating.

The friction torque of the rotation transmission device depends on pressure and viscosity of the locking fluid. It consists of several values: a friction torque in a liquid layer between the bushing and the shaft, a friction torque in the thrust bearing, friction torques in radial bearings, friction torques in the face seals. It is worth noting that only two first components depend on the locking fluid pressure, while the working fluid viscosity has an impact only on the first component.

As the locking fluid temperature grows at a fixed pressure value, the friction torque in the rotation transmission device decreases due to locking fluid viscosity decrease and becomes constant as it comes to stationary temperature conditions. With the pressure changing, the friction torque varies from 1.2–2.6 N m.

Comparing the friction torque function of the operating pressure to the friction torque in the thrust bearing, it is worth noting, that almost all the increase of the friction torque in the rotation transmission device with the pressure changing from 0 to 35 MPa is related to the increase of the friction torque in the thrust bearing.

The device breakaway torque dependence on the operating pressure value was also examined experimentally. Results of the experiment are shown on Fig. 4.

Comparing these experimental results to the experimental data of a friction torque at the spherical surface of the bushing and a friction torque at the face surface of the spherical washer, it can be concluded, that the breakaway torque value is determined by the bushing–shaft interface before the shaft starts its rotation, but not by the bushing turn along the face surface, as the breakaway torque value is much less than the value of the friction torque at the face surface.

Fig. 3. Leakage rate at the gap height of 0.0175 mm

Fig. 4. Breakaway torque function of pressure

Experimental measurement of the locking fluid temperature at the outlet of the gap seal showed, that stationary temperature conditions is established in about an hour after the rotation transmission device starts. Wherein a liquid temperature increment at the outlet of the gap seal has a significant value and can reach up to a few dozens of degrees, which entails changes of the operating liquid viscosity 6–8 times.

The results of the experiments fully confirmed theoretical research [8]. Other investigations were done for short gaps [9], and in the present case, the gap is longer due to the spherical washer. Operability of this device was also confirmed, as well as its advanced characteristics.

4. Conclusion

The problem of transmission of rotary motion to an environment with high ambient pressure was solved. The constructive scheme and the device prototype were presented. The results of the experimental research confirmed the operability of the device and allowed to define its specifications.

The experimental static characteristics of the prototype confirmed an acceptability of the assumptions, introduced in theoretical calculations. It also showed that the difference between the values of theoretical and experimental data does not exceed 20%, what is acceptable for this kind of calculations. The experimental efficiency of the rotation transmission device at the pressure of 40 MPa was 98%.

The rotation transmission device was designed and examined theoretically and experimentally. In the future it is planned to conduct durability tests of the device.

5. References

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