DYNAMIC CHARACTERISTICS OF ROTATION TRANSMISSION DEVICE IN HIGH AMBIENT HYDROSTATIC PRESSURE CONDITIONS

Anton Kuzmin, Valery Popov & Sergei Stazhkov

This Publication has to be referred as: Kuzmin, A[nton]; Popov, V[alery] & Stazhkov, S[ergei] (2018). Dynamic Characteristics of Rotation Transmission Device in High Ambient Hydrostatic Pressure Conditions, Proceedings of the 29th DAAAM International Symposium, pp.0698-0702, B. Katalinic (Ed.), Published by DAAAM International, ISBN 978-3-902734-20-4, ISSN 1726-9679, Vienna, Austria
DOI: 10.2507/29th.daaam.proceedings.100

Abstract

Rotation Transmission Device is designed for deep-sea equipment. It can be applied for different causes, such as ocean soil sampling. The device is unique as it transfers hydrostatic energy into rotary motion in high ambient pressure conditions. As the Rotation Transmission Device operates at high rotational speed, it is important to research its dynamics. In present work a sequence of calculations of the device dynamics is shown. It is proven that the mechanism design provides proper operation of the device.

Keywords: Deep-sea sampling; hydraulics; machinery; power unit; dynamics

1. Introduction

The World Ocean research is one of the most promising, however, one of the most difficult problems of modern science. The amount of data about its resources at the depth of more than 1500-2000 meters is extremely limited by now. However, there is a great interest in deep-sea coring and sampling for various biological and geological studies about exploring the World Ocean bottom [1], [2], [3]. Deep seas microorganisms and ecological system researches are also among modern most topical spheres [4], [5], [6].

To explore and develop these depths special equipment for soil sampling needs designing. This equipment should also be capable of coring hard rock. A perspective line in such kind of machinery is an application of hydrostatic energy. In addition, there is a challenge to transmit rotary motion to an environment with high ambient pressure.

To solve the problem of deep sea coring, applying the hydrostatic energy of an ocean, the rotation transmission device in high ambient hydrostatic pressure conditions (RTD) was designed. Its structural scheme and prototype were described in articles [7] and [8]. However, dynamic characteristics were not described in these works.

The purpose of this article is to examine the RTD dynamics and to prove its operability at an operating speed range. To achieve these goals forces and torques should be defined, motion expressions should be solved and its results should be examined.

2. Forces and torques

The RTD includes a gap seal, formed with the floating bushing, pressed to the front surface of the housing by the spherical washer.

Fig. 1. Forces applied to the bushing

To get a full description of physical processes in this seal and to define its characteristics the following problems need solving:

- a gap seal performance in a stationary mode analysis – to define static characteristics, such as locking fluid flow rate and temperature, energy losses caused by friction, a hydrodynamic force;
- a non-stationary operating mode of the “shaft – floating bushing” system study. This part of work includes forced oscillation research, determination of resonant frequencies etc.

The first problem was issued in works [1] and [2]. To solve the second question all forces and torques, applied to the floating bushing, must be defined. The next step is to write an equations system of its motion and to solve it. In general case, there are four groups of forces and torques that act on the floating bushing:

1. Forces and torques, caused by pressure in the lubricating layer.
2. Forces and torques caused by friction.
3. Reaction forces of the spherical washer.
4. Inertial forces and torques.

The force scheme is presented on picture 1. The list of forces, applied to the floating bushing:

- hydrodynamic force $\bar{F}$;
- locking fluid pressure forces acting on the bushing side surface $\bar{P}_2$, $\bar{P}_3$;
- pressure of the floating bushing frontal surface force $\bar{P}_1$;
- spherical washer reaction to a hydrodynamic forces impact $\bar{R}_F$;
- spherical washer reaction to the frontal surface of the bushing pressure force $\bar{R}_{P_1}$;
- gravity force $\bar{P}$.

To describe the process of the forces impact on the bushing, it is reasonable to assume that the hydrodynamic force is applied to the center of the spherical surface C .

The forces $\bar{P}_2$, $\bar{P}_3$ are balancing each other and can be neglected in further model.

The hydrodynamic force $\bar{F}$ and the reaction $\bar{R}_F$, which is equal to the friction force on the frontal surface of the housing $\bar{F}_{Tp}$, are combined into a resultant force $\bar{F} - \bar{R}_F$; also a torque, aiming to rotate the floating bushing about the point A – M_{onp} .

The pressure of the floating bushing frontal surface force generates the torque $\bar{M}_{ct}$, which is balancing M_{onp} . herewith the force is balanced by the reactions R_{P_1} . Consequently, according to the presented force scheme, all the torques, applied to the bushing, are mutually balanced. The only force acting on the floating bushing is the force, equal to the difference of the hydrodynamic force and the reaction $\bar{R}_F$, which is defined by the friction force between the frontal surface of the housing and the spherical washer. Wherein the bushing is moving translational towards a direction of the force $\bar{F}$.

A tilt of the floating bushing on the shaft will cause a sealing gap pressure redistribution. This process will cause a displacement of the hydrodynamic force application point relative to the center of the spherical surface, which leads to adding up the torque $\bar{M}_{ct}$ to the force pattern, tending to make the bushing coaxial with the shaft.

This process, in its turn, is reacted by the spherical surface friction torque $\bar{M}_{\text{tp.c}\phi}$. The floating bushing is also affected by a liquid friction torque, that is balanced by the spherical surface friction torque. The gravity force $\bar{P}$ is applied to the bushing center of mass. Its impact is similar to the hydrodynamic force impact, displaced relatively to the sphere center. However, it can be neglected as its value order is several times lower than other forces orders.

As it follows from the performed analysis of torques and forces, a match of the spherical surface center and the hydrodynamic force application point at the coaxial position of the shaft and the bushing axis is a preferred condition and should be taken into consideration while designing.

3. Mathematical model and calculations

The first group of forces can be determined out of the pressure distribution of an operating liquid in the sealing gap between the shaft and the floating bushing. The pressure distribution in the lubricating layer can be found by solving the Reynolds equation [7]. A resultant hydrodynamic force can be calculated as:

Use the hidden table with two columns for alignment, as shown in example:

$$\begin{cases} F_{gx_1} = \int_0^{2\pi R_B} \int_{Z_0}^{Z_0+L} P_{x_1}(x, z) dx dz \\ F_{gy_1} = \int_0^{2\pi R_B} \int_{Z_0}^{Z_0+L} P_{y_1}(x, z) dx dz \end{cases} \quad (1)$$

$P_{x_1}(x, z) = P(x, z) \cdot \sin(\frac{x}{R_B})$ – a pressure force projection on the axis X_1 (fixed coordinate system);

X – curvilinear coordinate, used for lubricating layer pressure calculations;

$P_{y_1}(x, z) = P(x, z) \cdot \cos(\frac{x}{R_B})$ – a specific pressure force projection on the axis y_1 (fixed coordinate system).

Expressions for torque components, which is caused by the uneven pressure distribution, can be defined as:

$$\begin{cases} M_{gx_1} = \int_0^{2\pi R_B} \int_{Z_0}^{Z_0+L} P_{y_1}(x, z) \cdot z dx dz \\ M_{gy_1} = \int_0^{2\pi R_B} \int_{Z_0}^{Z_0+L} P_{x_1}(x, z) \cdot z dx dz \end{cases} \quad (2)$$

Friction forces and torques of the lubricating layer between the shaft and the bushing values can be found by applying the Newton hypothesis and liquid velocities on the shaft surface values, which were calculated by the method, performed in the work [7]. After all the external torques and forces have been determined, it is possible to obtain the floating bushing motion expressions.

The floating bushing with the spherical washer has 5 freedom degrees and the following possible motions: three rotary motions about the center of the spherical surface and two translator motions in directions, perpendicular to the shaft rotation axis. However, as it follows from comparison of the liquid friction torque value between the shaft and the bushing to the friction torque on the spherical surface, the liquid friction torque is two orders smaller than the other; therefore, there is no bushing rotation about its axis. Let us introduce a coordinate system OXYZ, bound to the shaft with a reference point at the center of the spherical surface of the bushing, and $O_2X_2Y_2Z_2$, bound to the bushing. Moments of inertia of the floating bushing : $J_{x_2y_2} = J_{x_2z_2} = J_{y_2z_2} = 0$

The floating bushing tilt angle:

$$\Theta = \arctg \sqrt{tg^2 \alpha + tg^2 \beta} \quad (3)$$

α and β are tilt angles relative to the X_2 and Y_2

The motion expressions system:

$$\left\{ \begin{array}{l} \frac{dv_x}{dt} = (F_{g_{x_1}} - F_{x_{tp}\Sigma}) * \frac{1}{m_{BT}} \\ \frac{dv_y}{dt} = (F_{g_{y_1}} - F_{y_{tp}\Sigma}) * \frac{1}{m_{BT}} \\ \frac{dx}{dt} = v_x \\ \frac{dy}{dt} = v_y \\ \frac{d\omega_{x_2}}{dt} = \frac{1}{J_{x_2}} (M_{g_x} - M_{ж.т.х.} - M_{т.сф.х}) \\ \frac{d\omega_{y_2}}{dt} = \frac{1}{J_{y_2}} (M_{g_y} - M_{ж.т.х.} - M_{т.сф.у}) \\ \frac{d\alpha}{dt} = \omega_{x_2} \\ \frac{d\beta}{dt} = \omega_{y_2} \end{array} \right. \quad (4)$$

v , ω – linear and angular velocities of the bushing. F_{∂} , M_{∂} , F_{tp} , $M_{ж.т.}$, $M_{т.сф.}$ – driving force, driving torque, friction force, liquid friction torque and bushing spherical surface friction torque. J – bushing moment of inertia.

The hydrodynamic force and torque abruptly change their values depending on coordinates and velocities of the solid surfaces. The system of the motion equations is rigid. In common these facts make it rather difficult to solve and requires significant time costs. A possible way of calculations is applying Gear method. A result is shown on Fig.2. The relative eccentricity equals 0.125, the gap height is 40 micron, the shaft diameter is 30 mm and its rotation frequency equals 20 1/sec. As it can be found out of the graph, steady-state oscillations of the bushing start after 2 shaft revolutions, while the bushing oscillations amplitude is smaller than the shaft oscillations amplitude, which means a proper operation of the RTD. A condition, that restricts the bushing oscillations amplitude, is remaining the lubricating layer of 3 – 5 micron height, therefore, the bushing oscillations amplitude cannot exceed the shaft oscillations amplitude for more than the gap height between them.

Fig. 2. Floating bushing and shaft oscillations

4. Simplified calculations for RTD design

As to solve rigid systems of ordinary differential equations requires developing of special calculation methods, to simplify the task it is reasonable to make a preliminary estimate of the bushing dynamic stability. The RTD structure can be shown as an oscillatory system with one degree of freedom, oscillation theory methods can be applied, and to use a linearization of the hydrodynamic forces as it is shown in [3]. Results of this preliminary calculations should be the answer if it is worthwhile to do further steps. The natural frequency:

$$\omega_0 = \sqrt{\frac{k}{m}} \quad (5)$$

k – the liquid layer stiffness coefficient. m – the bushing mass.

As it is shown in [9], the liquid layer stiffness coefficient of a cylindrical gap equals:

$$k = \frac{4Er l}{h_0} \quad (6)$$

E – the liquid bulk modulus; l – the gap length; r – the cylindrical surface radius; h_0 – the gap height. As it is shown in the expression (4), the stiffness coefficient value is mostly determined by the locking fluid bulk modulus E and the gap height h_0 . The bulk modulus value for mineral oils usually ranges from 1350 to 1750 MPa, while the gap height in the RTD ranges from 0.01 to 0.04 mm. At these parameters the stiffness coefficient equals $1,75 \cdot 10^{11} \div 6,8 \cdot 10^{11}$ N/m. Supposing the floating bushing mass to be about 1 kg, the required natural frequency ranges from 3200 to 8200 1/sec.

Resonant and near-resonant frequency ranges are dangerous for a proper operation. In machinery such ranges do not exceed 200 1/sec. Consequently, the resonant range of the RTD is far from the performed range. This fact can also be concluded from the presented calculations. The algorithm of calculations to evaluate the dynamic stability:

- to determine the RTD constructive characteristics;
- to apply expressions (5) and (6);
- if the natural frequency significantly exceeds the shaft maximal frequency, no more further calculations are needed;
- in the other case, the motion equation is solved by Gear method.

5. Conclusion

In present work, the dynamics of the RTD was researched. It was shown, that the device is able to work at the operating speed range. To achieve these results the following steps were made:

- forces and torques, applied to the floating bushing, were analyzed;
- forces and torques caused by hydromechanics processes in the gap between the bushing and the shaft were examined;
- motion expressions for the floating bushing were obtained and calculated applying Gear method;
- the bushing and the shaft oscillations were compared and it was shown that the presented design of the RTD allows it to operate at the speed range that is required for these types of machines;
- simplified technique to verify the RTD design for its dynamic characteristics was presented.

The next step is creating a full mathematical model of the Rotation Transmission Device. This model should describe both statics and dynamics of the device. It has to be enough for design of a pilot sample. The step after is to calculate all the parameters of the RTD using this full model. The RTD allows applying hydrostatic energy in deep-sea equipment. An implementation of the RTD can cause a significant progress in deep-sea researches.

6. References

- [1] Brandt, A., Gutt, J., Hildebrandt, M., Pawlowski, J., Schwendner, J., Soltwedel, T., & Thomsen, L. (2016). Cutting the Umbilical: New Technological Perspectives in Benthic Deep-Sea Research. *Journal of Marine Science and Engineering*, 4(2), 36. doi:10.3390/jmse4020036
- [2] Ebbe, B.; Billett, D.S.M.; Brandt, A.; Ellingsen, K.; Glover, A.; Keller, S.; Malyutina, M.; Martinez Arbizu, P.; Molodtsova, T.; Rex, M.; et al. (2010) Diversity of abyssal marine life. In *Life in the World's Oceans*; McIntyre, A.D., Ed.; Blackwell Publishing Ltd.: Oxford, UK; Chapter 8; p. 361.
- [3] VanDover, C.L.; Aronson, J.; Pendleton, L.; Smith, S.; Arnaud-Haond, S.; Moreno-Mateos, D.; Barbier, E.; Billett, D.; Bowers, K.; Danovaro, R.; et al. (2014) Ecological restoration in the deep sea: Desiderata. *Mar. Policy*, 44, 98–106.
- [4] Purser, A.; Thomsen, L.; Barnes, C.; Best, M.; Chapman, R.; Hofbauer, M.; Wagner, H. (2013) Temporal and spatial benthic data collection via an internet operated Deep Sea Crawler. *Methods Oceanogr.* 5, 1–18.
- [5] Smith, C.R.; De Leo, F.C.; Bernardino, A.F.; Sweetman, A.K.; Martinez Arbizu, P. (2008) Abyssal food limitation, ecosystem structure and climate change. *Trend Ecol. Evol.* 23, 518–528.
- [6] Lecroq, B.; Lejzerowicz, F.; Christen, R.; Esling, P.; Baerlocher, L.; Osteras, M.; Farinelli, L.; Pawlowski, J. (2011) Ultra-deep sequencing of foraminiferal microbarcodes unveils hidden richness of early monothalamous lineages in deep-sea sediments. *Proc. Natl. Acad. Sci. USA* 108, 13177–13182.
- [7] Kuzmin, A[nton]; Popov, V[alery] & Stazhkov, S[ergey] (2016). Rotation Transmission Device in High Ambient Hydrostatic Pressure Conditions., Chapter 22 in *DAAAM International Scientific Book 2016*, pp.239-248, B. Katalinic (Ed.), Published by DAAAM International, ISBN 978-3-902734-09-9, ISSN 1726-9687, Vienna, Austria DOI: 10.2507/daaam.scibook.2016.22
- [8] Kuzmin, A[nton]; Popov, V[alery] & Stazhkov, S[ergey] (2017), Experimental Research of Rotation Transmiission Device in High Ambient Hydrostatic Pressure Conditions. *Proceedings of the 28 th DAAAM International Symposium*, pp.617-620, Viena, Austria, 2017.
- [9] Марцинковский В.А. (1980) Бесконтактные уплотнения роторных машин. (Marcinkowsky W.A. (1980) Noncontacting seals of rotary machines) Moscow, Машиностроение., p.200