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HEAT CONDUCTION PROBLEM IN NON-CONTACTING FACE SEALS

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Abstract: *A mathematical model describing the phenomenon of heat transfer in the system "sealing rings – fluid film" for non-contacting face seal was presented in this paper. The presented model was solved with the use of analytical methods for two cases of installation of the stationary ring. The first one, when the stator is completely isolated from the surrounding operating medium and the second, when heat transfer takes place via convection between the outer cylindrical ring surface and the operating factor on the processing part. The distributions of temperature fields in sealing rings were determined with the use of the method of separation of variables, which were recorded with the use of trigonometric series in the form of surface functions for the ring cross-section.*

Keywords: Mechanical seal, Non-contacting face seal, Heat transfer, Trigonometric polynomials.

1. Introduction

Mechanical seals, in particular non-contacting face seals, belong to machine elements that are characterized by a large complexity of physical phenomena. A flow of operating medium through the radial gap, heat transfer between the cooperating elements as well as fluid solid interaction occur there. What is more, in many papers one may find mathematical models describing the dynamics of the sealing rings, whose form of record is similar to the one included in the papers (Koruba et al., 2010a, 2010b; Krzysztofik & Koruba, 2012; Krzysztofik, 2012). Taking vibrations of sealing rings into consideration, attempts of controlling them (Takosoglu et al., 2009, 2012) using control schemes that are based i.a. on (Laski et al., 2015) were made. Materials used for producing operating rings as well as connecting elements made of plastics or produced with the use of the additive technology (Adamczak et al., 2014, 2015; Bochnia & Kozior, 2014; Bochnia, 2012), play a crucial role in the production of non-contacting face seals. Due to the nature of operation of non-contacting face seals, roughness of rollers of cooperating rings constitutes a crucial problem that needs to be taken into consideration. The materials for production of rings must be chosen in such a way so that one ring is produced from a material of large hardness, e.g. carborundum, tungsten carbide or stainless steel. Counter-ring, however, should be made of a soft material, e.g. impregnated graphite. Another aspects regarding the production and installation of the elements of non-contacting face seals are dimensional accuracy as well as consideration of proper shape and location tolerances (Miko & Nowakowski, 2012a, 2012b). Another phenomena which might influence the proper operation of the described type of sealing are phenomena related to heat transfer and associated thermal deformations that were presented in a detailed way in the paper (Blasiak, S. et al., 2012; Blasiak, S., 2015a). Research posts, where sensors for measuring temperature and vibrations were applied (Blasiak, M. & Kotowski, 2009), are widely described in the literature. The specificity of operation of non-contacting face seals, i.e. high angular velocity of the rotating elements, small gap of ($\sim 1 \mu$m) (Takosoglu et al., 2014), roughness of the cooperating surfaces or thermos-elastic deformations of the operating rings as well as many more other factors influence the operation of non-contacting face seals. Preparing and developing the mathematical models describing the above mentioned phenomena allow to obtain more and more detailed results of the analyses. Moreover, the choice of proper materials for sealing rings adjusted to the working conditions as well as determining the temperature value at the design stage or during bench studies allow to avoid excessive wear of the surface of sealing rings and prolong the lifetime of the cooperating elements. The subject of this paper was to identify the phenomena related to heat transfer in particular elements of non-contacting face seals.

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2. Non-contacting face seal model

The overall scheme of non-contacting face seal for two cases of installation of the stationary ring was presented in Fig. 1. Non-contacting face seal consists of two cooperating rings – stator (1), which is firmly fixed in the casing, and rotor (2), which rotates together with the shaft (6) of the rotating machine and is pressed with the spring (3) against the stator.

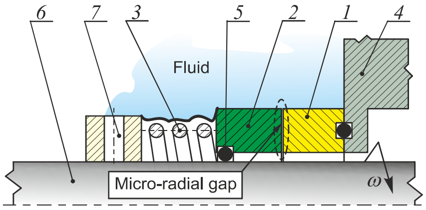

Fig. 1. Scheme of non-contacting face seal. a) stator convection, b) stator insulation; 1 – stator, 2 – rotor, 3 – spring, 4 – casing, 5 – O-ring, 6 – shaft, 7 – retainer.

The mathematical model describing the phenomenon of heat transfer was presented in papers (Blasiak, S., 2015b). As one of the boundary conditions was assumed the fact that the surfaces of operating rings (Fig.2) with no direct contact with the surrounding liquid are completely isolated. It was recorded in a general form $\frac{\partial \theta}{\partial n} = 0$, where n – means a direction normal to the surface. On the face ring surfaces (stator and rotor) limiting the radial gap, and with direct contact with the medium layer, the condition regarding the case of heat transfer via conduction is fulfilled.

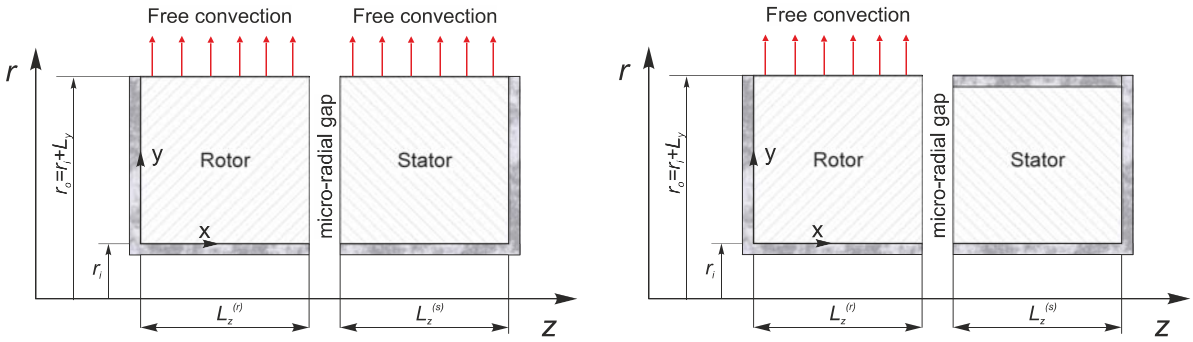

Fig. 2. Heat transfer boundary conditions, a) stator convection, b) stator insulation.

The values of heat flow on the surfaces of the element and medium under consideration are respectively:

on the stator surface: $q^s_v(x) = \lambda^s \frac{\partial \theta^s}{\partial x} = \lambda^f \frac{\partial \theta^f}{\partial x} = 0$ and $\theta^s = \theta^f$ on the rotor surface

$\lambda^f \frac{\partial \theta^f}{\partial x} = \lambda^r \frac{\partial \theta^r}{\partial x} = q^r_v(r)$ and $\theta^f = \theta^r$. It means that the maximal temperature it to be found on the border between the fluid film and stator, where there is the biggest friction of the fluid against the stationary ring surface. Thus, almost the whole heat flow is transferred by the second ring, namely the rotor. On the rotating surface of the stator and rotor (Fig.2) that contacts the surrounding medium, heat is transferred via free convection. It was generally recorded in the form: $-\lambda^i \frac{\partial \theta^i}{\partial y}\bigg|_{y=L_y} = \alpha^i \theta^i\big|_{y=L_y}$, where

$i = r, s$ rotor and stator respectively.

2.1. Analytical solution

The model of heat transfer in non-contacting face seal was solved analytically. In the first phase, the temperature distributions in sealing rings were determined by stipulating a general form of functions satisfying Laplace's equation (1), both for stationary ring (stator) as well as rotor, with accepted boundary conditions. Used in this case, the transition from the cylindrical coordinate system to the Cartesian coordinate system, considerably simplifies the analytical calculations. Appearing for the cylindrical coordinate system, Bessel functions increase the calculation time in mathematical programs.

$$\frac{\partial^2\theta}{\partial x^2}+\frac{\partial^2\theta}{\partial y^2}=0 \tag{1}$$

The presented analytical calculations were conducted similarly as in the paper (Blasiak, S., 2015b). The final formulas describing the temperature distributions in sealing rings are presented below:

in the rotor:

$$T^r=T_0+\sum_{n=1}^{\infty}\frac{-\int_0^{L_y}\bar{q}\cos(s_n y)dy}{s_n\sinh(s_n L_x)\int_0^{L_y}\cos^2(s_n y)dy}\cosh(s_n(L_x-x))\cos(s_n y) \tag{2}$$

and the in insulated stator:

$$T^s=T_0+\sum_{n=1}^{\infty}\frac{2\cdot\int_0^{L_y}\theta^f\cos\left(\frac{n\pi}{L_y}y\right)dy}{\cosh\left(\frac{n\pi}{L_y}(L_x)\right)L_y}\cosh\left(\frac{n\pi}{L_y}(L_x-x)\right)\cos\left(\frac{n\pi}{L_y}y\right)+\frac{\int_0^{L_y}(\theta^f)\mathrm{d}y}{L_y}, \tag{3}$$

as well as in the non-insulated stator:

$$T^s=T_0+\sum_{n=1}^{\infty}\frac{\int_0^{L_y}\theta^f\cos(s_n y)dy}{\cosh(s_n L_x)\int_0^{L_y}\cos^2(s_n y)dy}\cosh(s_n x)\cos(s_n y) \tag{4}$$

The above dependencies served to determine the temperature distributions in the system “fluid film - sealing rings”.

3. Numerical results

The calculations were conducted for operating and geometric parameters from the paper (Blasiak, S., 2015b). It was assumed that water T_o of $20\,^oC$ was the operating medium. In the geometry of mechanical seal it was assumed that the outer radius was 45 mm, and the inner 40 mm for both cooperating rings. It was assumed as well that the rotor rotates with the angular velocity of 500 rad/s. Thermal conductivity for the rotor of $\lambda^r=130\ (\mathrm{W/m\cdot K})$, and for the stator of $\lambda^s=15\ (\mathrm{W/m\cdot K})$ was taken into consideration. The height of the radius gap on the assumed level 1 μm .was taken into account.

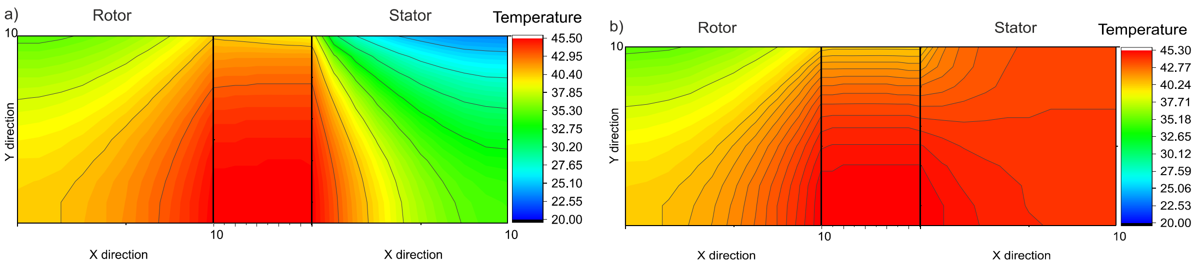

Fig. 3. Temperature distribution, a) stator convection, b) stator insulation.

On the basis of the obtained results, it might be concluded that the temperature distribution in the gap is at the same level. The temperature differences are visible only in case of stator what results from the method of its installation in the casing. For the insulated stator (Fig. 3b), the temperature is practically in the entire cross section at the level of 43 °C. For the stator (Fig. 3a), which is not insulated, the temperature on the outer radius is similar to the temperature of the working medium on the processing part. On the basis of (Fig. 3a), uneven temperature distributions in cooperating rings might be observed. It results from various physical properties of the materials that the rings are produced from. Diversification of temperature distributions in sealing rings changes the geometry of the radial gap, which results from thermos-elastic deformations.

4. Conclusions

The main task of non-contacting face seals is to keep tightness regardless of external factors. There is a tight relation between the geometry of the radial gap and the leakage. Deformations due to uneven temperature distribution in the sealing rings are main connecting factors. Developing more detailed mathematical models and determining temperature values already at the design stage provide the researchers and constructors with crucial knowledge that is essential while designing such devices. This knowledge, in turn, translates into huge reliability and longevity of the constructions.

References

Adamczak, S., Bochnia, J., & Kaczmarska, B. (2014) Estimating the uncertainty of tensile strength measurement for a photocured material produced by additive manufacturing, Metrology and Measurement Systems, 21, 3, pp. 553–560. doi:10.2478/mms-2014-0047

Adamczak, S., Bochnia, J., & Kaczmarska, B. (2015) An analysis of tensile test results to assess the innovation risk for an additive manufacturing technology, Metrology and Measurement Systems, 22, 1, pp. 127–138. doi:10.1515/mms-2015-0015

Blasiak, M., & Kotowski, R. (2009) Propagation of acoustic waves in piezoelectric crystals, Przeglad Elektrotechniczny, 85, 12, pp. 40–43.

Blasiak, S. (2015a) An analytical approach to heat transfer and thermal distortions in non-contacting face seals, International Journal of Heat and Mass Transfer, 81, pp. 90–102. doi:10.1016/j.ijheatmasstransfer.2014.10.011

Blasiak, S. (2015b) The two dimensional thermohydrodynamic analysis of a lubrication in non-contacting face seals, Journal of Thermal Science and Technology, 10, 1, pp. JTST0016–JTST0016. doi:10.1299/jtst.2015jtst0016

Blasiak, S., Kundera, C., & Bochnia, J. (2012) A Numerical Analysis of the Temperature Distributions in Face Sealing Rings, Procedia Engineering, 39, pp. 366–378. doi:10.1016/j.proeng.2012.07.040

Bochnia, J. (2012) Ideal Material Models for Engineering Calculations, Procedia Engineering, 39, 0, pp. 98–110. doi:http://dx.doi.org/10.1016/j.proeng.2012.07.013

Bochnia, J., & Kozior, T. (2014) Methods of Prototyping Process Using Modern Additive Technologies, Solid State Phenomena, 223, pp. 199–208. doi:10.4028/www.scientific.net/SSP.223.199

Koruba, Z., Dziopa, Z., & Krzysztofik, I. (2010a) Dynamics and control of a gyroscope-stabilized platform in a self-propelled anti-aircraft system, Journal of Theoretical and Applied Mechanics, 48, 1, pp. 5–26.

Koruba, Z., Dziopa, Z., & Krzysztofik, I. (2010b) Dynamics of a controlled anti-aircraft missile launcher mounted on a moveable base, Journal of Theoretical and Applied Mechanics, Vol. 48 nr, pp. 279–295.

Krzysztofik, I. (2012) The Dynamics of the Controlled Observation and Tracking Head Located on a Moving Vehicle, Solid State Phenomena, 180, pp. 313–332. doi:10.4028/www.scientific.net/SSP.180.313

Krzysztofik, I., & Koruba, Z. (2012) Model of Dynamics and Control of Tracking-Searching Head, Placed on a Moving Object, Journal of Automation and Information Sciences, 44, 5, pp. 38–47.

Laski, P. A., Takosoglu, J. E., & Blasiak, S. (2015) Design of a 3-DOF tripod electro-pneumatic parallel manipulator, Robotics and Autonomous Systems, 72, pp. 59–70. doi:10.1016/j.robot.2015.04.009

Miko, E., & Nowakowski, L. (2012a) Analysis and Verification of Surface Roughness Constitution Model After Machining Process, Procedia Engineering, 39, pp. 395–404. doi:10.1016/j.proeng.2012.07.043

Miko, E., & Nowakowski, L. (2012b) Vibrations in the Machining System of the Vertical Machining Center, Procedia Engineering, 39, pp. 405–413. doi:10.1016/j.proeng.2012.07.044

Takosoglu, J. E., Dindorf, R. F., & Laski, P. A. (2009) Rapid prototyping of fuzzy controller pneumatic servo-system, International Journal of Advanced Manufacturing Technology, 40, 3-4, pp. 349–361. doi:10.1007/s00170-007-1349-5

Takosoglu, J. E., Laski, P. A., & Blasiak, S. (2012) A fuzzy logic controller for the positioning control of an electro-pneumatic servo-drive, Proceedings of the Institution of Mechanical Engineers. Part I: Journal of Systems and Control Engineering, 226, 10, pp. 1335–1343. doi:10.1177/0959651812456498

Takosoglu, J. E., Laski, P. A., & Blasiak, S. (2014) Innovative Modular Pneumatic Valve Terminal With Self-Diagnosis, Control and Network Communications, In V. Fuis (Ed.), Engineering Mechanics 2014 (pp. 644–647). Dolejskova 5, Prague 8, 182 00, Czech Republic: Acad Sci Czech Republic, Inst Thermomechanics.