Research Journal of Applied Sciences 9 (11): 707-710, 2014
ISSN: 1815-932X

Face Gas-Dynamic Seal of High-Speed Compressor

Sergei V. Falaleev and Petr V. Bondarchuk
Samara State Aerospace University, 34, Moskovskoe Shosse, 443086 Samara, Russia

Abstract: Sealing systems of turbo-machines determine possibilities of intensification of working processes in it. Facegas-dynamic seal are a standard technology solution for compressors. The aim of a present research is modernization of face gas-dynamic seal structures and research of processes in it. In the present studies ways of seal structure modernization are formulated. Results of experimental research of seal thermal condition are presented. It is found that in the seal a laminar gas flow is realized.

Key words: Face gas-dynamicseal, compressor, leakage, friction pair, thermal condition

INTRODUCTION

Face Gas-Dynamic Seals (FGDS) are a standard type of rotor supports seals for compressors (Novikov, 2014; Balyakin *et al*., 2002; Zhdanov *et al*., 2013; Kuz'michev *et al*., 2014). A maximal widespread these seals have in pumps for natural gas and turbo-machines for chemical industry. A long-term experience of exploitation of FGDS in gas industry shows that using of high-cost seals with gas lubrication (which required high technology for its manufacturing) gives a large economic effect during exploitation. Sealing units with FGDS are more expensive than usual, however, costs for exploitation reduce much more. A complex of questions for seal design requires a large calculation research (Falaleev *et al*., 2006, 2001, 2014; Rosseev *et al*., 2001; Belousov *et al*., 2007; Chegodaev and Falaleev, 1985). All these problems are interconnected. Methodic basement of their complex solution is presented by Belousov *et al*. (2009) and Belousov and Falaleev (1989). A development of face gas-dynamic seals is a high-technology and labor-intensive process (Vinogradov, 2014; Falaleev and Zhizhkin, 1996; Falaleev and Vinogradov, 2014; Bondarchuk and Falaleev, 2014).

Heat appears in FGDS on areas on of friction of gas in a gap between rings of friction pair and because a gas in cavities is heated up by details rotate in these cavities. Most of researcher note that all heat appeared in the gap is taken away to walls by heat conductivity (Novikov, 2014; Falaleev, 2014; Shabliy and Cherniaev 2014). Theoretical and experimental researches (Sukhomlinov *et al*., 2004) confirma reliability of isothermal theory of gas flow in a thin gap of gas bearing. Therefore, it is interesting to find temperature of rings of the friction pair and temperature of gas in an entrance to FGDS in a gap and in exit from FGDS.

RESEARCHED STRUCTURE OF FACE GAS-DYNAMIC SEAL

A sealing unit developed for booster with rotor speed 25000 rpm is presented at Fig. 1.

The seal unit contains of labyrinth 4, face gas-dynamicseal 6, screw-groove seal 12 and oil deflector ring 13 placedin series. Its destination is to divide gas (A) and oil (B) cavities. The seal is a united object and has modulus structure. Rotating details are fixed on a sleeve 5. The gas goes from an exit of compressor across final filter by channel in compressor frame to the cavity (D). Part of this gas leaks across face seal to a cavity (D), all other gas across labyrinth 4 returns back to centrifugal stage. This gas cuts off a dirty gas and cools rings of friction pairs 6 and 8. Gas leak a gas across face seal go to the cavity (D) and after it across different seals to the oil cavity (B). It is possible to judge about capacity for work of face seal by pressure in the cavity (D). Oil deflector ring 13 and screw-groove seals 12 divide the cavity (D) and oil

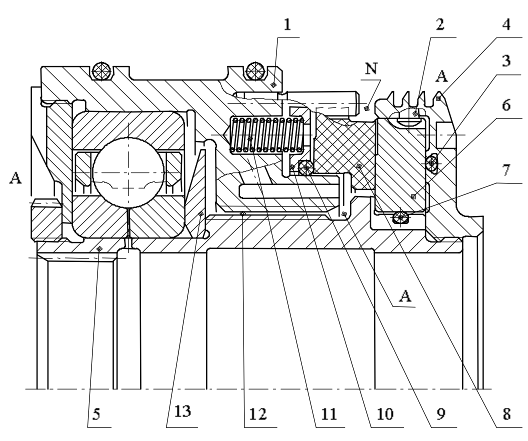

Fig. 1: Seal of booster

Corresponding Author: Sergei V. Falaleev, Samara State Aerospace University, 34, Moskovskoe Shosse, 443086 Samara, Russia

cavity. A stage of face seal has a sealing ring 8 movable in axial direction and fixed against rotation by sprigs entered into holes in a frame 1. This ring is installed hermetically by secondary rubber seal 9 in the frame 1 on elastic elements 11. These condary rubber seal 9 is placed between sealing ring 8, frame 1 and ring 10. Ring 8 forms a sealing gap with rotating ring 6 which is installed elastically on a sleeve on a rubber ring 7. Ring 6 is fixed on as haft 5 by labyrinth sleeve 4. On a root surface of rotating ring 6 there are spiral grooves which directed to high pressure cavity and placed in direction of shaft rotation. The labyrinth sleeve 4 protects the frame against a possible destruction of hard-metal ring. On an outer surface hard-metal ring there are hollows in which two sprigs are placed for transmission of rotation moment.

Complex of questions arising during a designing of face seals with gas lubrication and demanding an extensive computational research is shown on Fig. 2. Within the limits of a first method to be considered it is possible to name these questions conditionally as "internal" (since for its solution are used parameters inside of seal unit only). Especially, it is necessary to allocate following problems:

- Influence of flow structure of the sealing environment in a cavity before seal (cavity forms before seal) on a thermal condition of seal
- Influence of seal ring geometry on values of the thermal and power deformations arising in rings
- Influence of ring deformations on the form and value of a sealing gap
- Influence of the form and value of a sealing gap on pressure profile in a seal clearance
- Influence of a pressure distribution in a seal clearance on a sealing ring deformation value

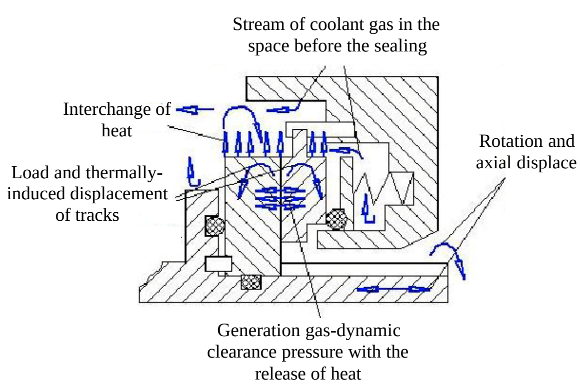

Fig. 2: Main problems solved at the seal designing as friction pair unit

DEVELOPMENT OF PROPOSITION FOR TGDS MODERNIZATION

JohnCrane company recommends that expenditure of buffer gas enough for cooling of the first stage of FGDS and cutting off the buffer zone from the mail flow of "dirty" gas should be 50 g sec^{-1}. Calculation researches carry out in the present study shows that much less expenditure, 25 g sec^{-1} is enough. During design of FGDS for booster compressor we were forced to set the buffer gas expenditure as 5 g sec^{-1}. Because compressor is high-speed and an outer diameter of friction pair is about 100 mm, a power of friction in a sealing gap is significantly more than for usual FGDS for gas compressors. A reliable research of FGDS was obtained by optimization of buffer gas flow in a cavity before the seal and by significant reducing of assembly gap in a separating labyrinth with installing a sleeve made of Teflon above the labyrinth.

A necessity of reserve stage in FGDS is a disputable question. There were some cases of destruction of first sealing stage during development of FGDS for compressor N370-18-1 (Falaleev, 2014). When a safety valve worked, a pressure between stages was not >0.4 MPa. There fore first FGDS for compressor N370-18-1 has on its reserve stage a "narrow" friction pair which need less surface for installation and less cooling. During design of FGDS for compressor N16-76-1.44 a lack of assembly space forced to reduce maximally a length of rotor sleeve of FGDS and to place a labyrinth with a barrier air into a space between stator of reserve stage of FGDS and compressor shaft that is to move the labyrinth into FGDS. During design of booster mentioned above, a distance between bearing and gas labyrinth which confines a flow part was 65 mm. It was no space for a second stage after a placement of main stage of FGDS. It was designed to place an oil deflection ring and screw-groove seal between the stage o FGDS and bearing. This structure works as a reserve stage in a seal unit. During a research the oil is driven away by screw-groove seal and leakages across FGDS. During starting and stopping of compressor an expulsion of screw-groove seal by gas is provided. If FGDS is destructed, a pressure in creases in a cavity between it and screw-groove seal and automated mechanism will research for a stopping of compressor. Tests of this booster took place successfully; it confirms an aptitude of these solutions. An obtained experience shows a possibility of modernization of reserve stage of FGDS. It allows reducing of FGDS costand assembly space.

Thus an accurate design and realization of the most effective of proposed solutions allows reducing the cost of developing FGDS, costing of its exploitation and repairing.

RESEARCH OF THERMAL CONDITION OF SEAL

Experimental research of thermal condition of FGDS was under taken for exact modeling of seal working in compressor. Temperature was measured by non-contact way with hot-wire air flow meter on a shaft of seal, on a frame of FGDS and on a rotated ring of first stage across a gap between shaft and frame. To measure a thermal condition of graphitering it was prepared by placement into graphite a miniature temperature sensor with size 1.7×2.1 mm on a distance 5 mm from a working surface. A measurement of temperature in a leakage flow from first stage was provided by analogous sensor placed on a frame of first stage at an exit from working gap.

Obtained results of temperature measurement of FGDS details for difference of air pressure on the seal 6 MPa are presented on Fig. 3.

As a result of research the temperature of shaft, frame and rings of friction pair of FGDS for different operating modes of dynamic stand was obtained. It is found that temperature of graphitering and hard-metalring are equal during its research. For short-time test its temperature was 39°C. This value is realized for long-time test for an intensive cooling of FGDS. It takes place during a work of seal amounting to compressor with buffer gas flow which is cooling a friction pair. It is obtained that during a research of FGDS amounting on the dynamic stand a significant heating of an air in a buffer cavity take place of FGDS details rotation. For example an air was heated to 95°C during 1.5 h. A maximal difference of air temperature in the buffer cavity and air leakage across FGDS was 22°C. Tests show that for FGDS calculation it is possible to consider the flow mode in a gap of FGDS as isothermal with a temperature equal to temperature of rings of friction pair. During a work of seal a cooling of gas flowing from a working gap on 20-22°C takes place. It is possible to explain it that at exit from sealing gap the gas has a velocity of sound. A crossing of this velocity connects with temperature decreasing.

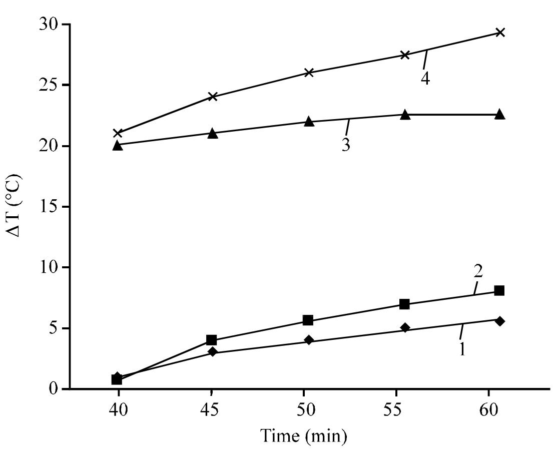

Fig. 3: Dependency of temperature difference on time for long-time testing. 1: difference of temperature of graphitering and gas fedinto seal; 2: difference of temperature of leakages from first stage and in a chamber after the first stage; 3: difference of temperature of gas fed into the seal and leakages from a second stage; 4: difference of temperature of graphitering and leakages from the first stage

Forlong-time test a temperature of graphitering exceeds a temperature of air in a buffer cavity not >6°C. It means that the temperature of friction pair during it work in compressor will be determine by temperature of buffer gas mostly.

CONCLUSION

Face gas-dynamic seals are a perspective type of seal for different turbo-machines. However, a modernization of its structure is necessary for reducing its cost and providing of high reliability. One of ways for it is a combination of functions of reserve seal stage and barrier seal. Undertaken experimental researches show that it is possible for these seals calculation to consider a laminar gas flow. A heating of friction pairs is insignificant and not exceed 30-40°C. An especial attention should be given to gas temperature in the buffer cavity before the seal.

ACKNOWLEDGEMENTS

This research was supported by the Ministry of education and science of the Russian Federation in the framework of the implementation of the Program of increasing the competitiveness of SSAU among the world's leading Scientific and Educational Centers for 2013-2020 years.

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