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Design of Gas-Static Bearing – Statement of Problem

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Abstract: In this paper we consider the problem of designing the gas bearing with forced gas supply under the excessive pressure into the gap between the stator and the rotor. Such bearings were called gas-static bearings (GSB). Over the last few decades a number of countries are actively developing gas-lubricated bearings [1]. Sufficient experience of creating such devices in small sizes was obtained. The analysis of the available scientific and technical literature on the design of gas-static bearings was made. Performed a brief comparison of gas-static, gas-dynamic and hybrid bearings. The method of calculating the bearing capability of GSBs was developed. Treated the task of designing the gas-static bearings, intended for high speed rotation. The purpose of parametric studies of gas bearings' design is to identify the typical dependency of GSB load capacity on the pressure, temperature and flow of the working fluid. The basic concepts of gas bearings' non-stationary modes: oscillatory and transient are given. Brief information on the optimization of the shape of the gas bearing's supporting surface is detailed.

Key words: Gas lubricating · Gas-static bearing · Hybrid gas bearing · Critical frequency · The Reynolds equation · Velocity-fractional vortex · Pneumatic hammer mode · Gas consumption fluctuations

INTRODUCTION

In this paper we study the gas-static bearings (GSB) that uses a forced supply of the compressed air as the working fluid in a grease layer between the stator and the rotor under excessive pressure [1]. An example of such a construction is given in Fig. 1.

A possible way to improve the reliability and resource of rotating machines is the usage of various designs of contactless supports, which provide the rotation and attitude stabilization to the rotors without direct contact of metal parts [2]. Lack of contact on the working modes and hence, the contact friction, absence of necessity for lubrication, the ability to provide higher speed rotation, the relative simplicity and low weight of construction makes such support structures very attractive for those branches of technics, that require a long and reliable operation without maintenance, lightness and compactness [3].

Creating these bearings will allow to radically simplify the transmission of certain devices (gas-turbine engines, compressors, pumps, fans), eliminating the lubrication system [4]. In some cases, non-contact bearings are the only available technical solution. For example, space nuclear power plants that use a closed cycle gas turbines.

The cost of the oil system and a standard transmission, which uses rolling-contact bearing in typical Auxiliary power unit (APU) and gas turbine power plant (GTPP), is up to 25 % of the engine's cost. The usage of oil restricts the operation in certain climatic conditions. Oil-free transmissions are free from the disadvantages, mentioned above and can be used everywhere, if they overcome the known technical problems - limited weight capacity and overloads [5].

Main advantages of gas bearing, comparing to the ball bearing, electromagnetic and hydrostatic bearings include the following [6]:

- Able to operate at both high and low temperatures and humidity;
- Provide the long-life and reliable functioning of friction assemblies;
- Are resistant against radiation exposure;
- Does not pollute the environment;
- Greatly simplify the design of the friction unit;
- Provide low level of vibration and noise.

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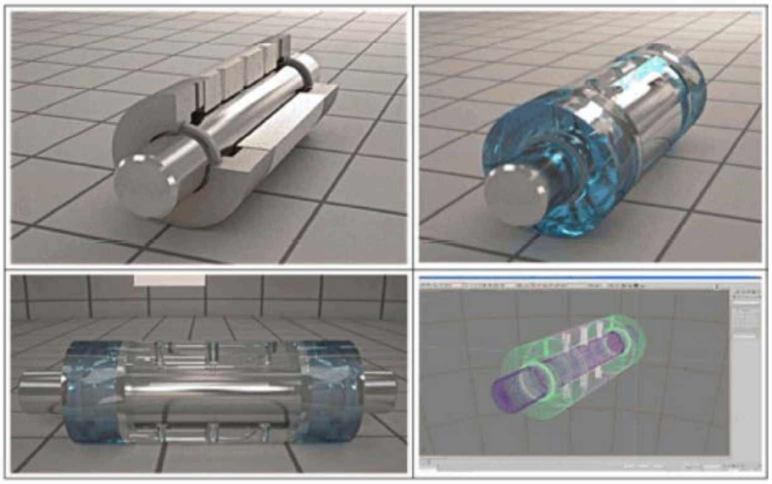


Fig. 1: Example of gas-static cylindrical bearing's design with supplying gas in the gap between a stator and the rotor through 6 simple holes

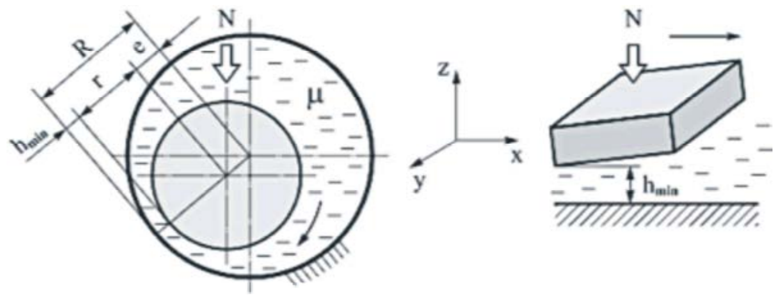


Fig. 2: Scheme of sleeve bearings' hydrodynamic. Basic concepts

Electromagnetic bearings require high-speed electronic control systems, are sensitive to the quality of power supply and electromagnetic interference, have unstable properties at the ambient temperature changes. Their rigidity is several times lower than of the gas-lubricated bearings [7].

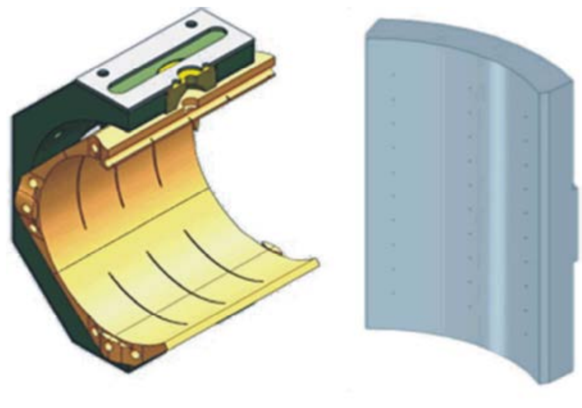
Statement of the problem regarding the design of an adjustable gas-static bearings.

For more than one hundred years of the Theory of gas lubrication development terminology of this domain was formed [8]. Consider the basic concepts. Gas gap (lubricating layer) - space between the stator and the rotor. Without the load in the radial bearing the size of the gap H is equal to its average value of h (Fig. 2) Radial load on the shaft N causes displacement of its axis from the axis of symmetry in the direction of the applied force (in this case downwards). As a result, viscous flow with velocity profile of V flows through the annular gap of variable section. If the GSB is full-wrap (Fig. 2), the interaction of the viscous flow with the variable section leads to formation of the pressure gradient on the walls, which causes the displacement of axis by an angle of [beta] from the radial load vector.

In the same time this gas gap has a minimum h_{min} and maximum h_{max} size. An absolute displacement of the shaft axis equals to the eccentricity of e. Due to the thinness of the gas layer, it is allowed to "unfold" the radial bearing into the flat toe for the purposes of analysis (Fig. 2 right).

Stiffness of the bearing is equivalent to the concept of the spring rigidity. In a wide range of h the decrease in the average thickness of the gas layer is proportional to the load. The proportionality coefficient k is called the stiffness of the gas layer.

The square-cube paradox. Gas-bearing constructions usually cannot be scaled because when the linear dimensions change, the pressure forces, which determine load capacity, are proportional to the square of them and mass forces – to the cube of the specific linear dimension. It is impossible to scale the small successful constructions. That is why small GSBs with a shaft diameter of a few millimeters and successfully working at speeds up to 300,000 rpm are so widespread. Increasing the diameter makes it necessary to set a much larger pressure in the gap and, as a consequence, a greater expense of working fluid [10]. This affects the efficiency.



a) b)

Fig. 3: Examples of the working fluid supply system into the gap between the stator and the rotor
a) the slotted nozzles, b) a system of cylindrical holes

For a given load capacity of GSBs and formulated technical requirements for its construction is usually necessary to minimize the consumption of the working fluid and get the maximum stiffness of the lubricating layer. This is achieved by optimizing the system of supplying the working fluid and careful design of the stator's supporting surface profile.

While designing GSB, as well as calculating the stiffness of the lubricating layer it is necessary to be able to define the optimal values of the basic geometric parameters, such as:

- The ratio of the length and width of the pad;
- Lubricating layer supplying holes location;
- The arrangement of the flat nozzles in or against the rotation direction, the distance between the nozzles and the between the edge nozzles and the end to the pad.

For a given pad geometry the purpose of parametric optimization is to identify the typical dependency of pad's load capacity on the pressure, temperature and flow of the working fluid. The working fluid supplying system may be composed of different combinations of supply canals, grooves (undercuts) and cavities (Fig. 3).

The solution of the variational problem on maximizing the functional $J = F_{fr} / F$ (where F - bearing capacity of the profile and F_{tp} - friction magnitude), first obtained by Rayleigh [11] shows that the optimal profile of the gas layer is described by a piecewise continuous function with a step change in the thickness of the layer. The pressure distribution in the gap shown in Figure 4 and 5 illustrates this conclusion.

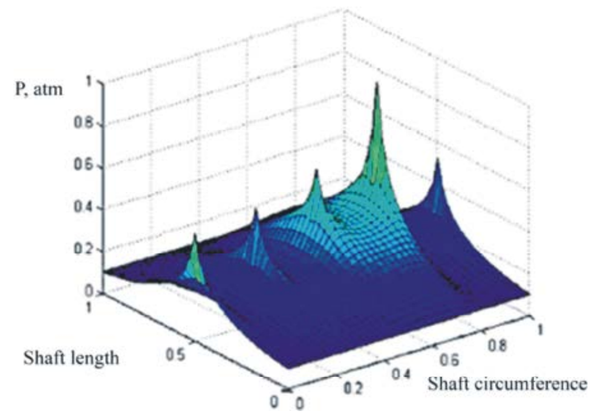


Fig. 4: Pressure distribution in the gap with the gas supplying through four simple hole

Figure 4 shows the pressure distribution in the gap when the gas is supplied under pressure through the four holes.

Bursts on the graph at a value of the azimuthal coordinate 0 and 1 correspond to the same hole. Due to the eccentricity the magnitude of the pressure peaks is different for the different holes. Obviously, this scheme is effective only when the number of holes is large as the pressure on the shaft decreases sharply with the distance from the holes growing.

Figure 5 shows the pressure distribution on the shaft when supplying gas through the four lubricating holes with pockets.

It is evident that in this case, the pressure peaks at the shaft is smaller than when the grease is supplied through simple holes and the resultant of the pressure forces is significantly higher. This confirms the appropriateness of the pockets (undercuts) usage to increase the load capacity of GSB. It is also clear that while supplying the working fluid through the holes their number should be the as high as possible. This shows the viability of using flat nozzles instead of holes.

It is experimentally shown [12] that the greatest stiffness of the lubricating layer and the smallest gas consumption at a given load capacity is achieved when the air is supplied in the gap through the porous surface and the diameters of the supply holes are as minimal as possible. The main problem of the GSB porous inserts - non-regular features in use. Therefore, it is necessary to find other forms of optimal nozzle to lead the working fluid into the gap.

Non-stationary motions of the rotor mounted on the gas bearings are treated separately from the optimization problem of GSB's carrying properties. All non-stationary regimes are divided into three types: the plane-parallel

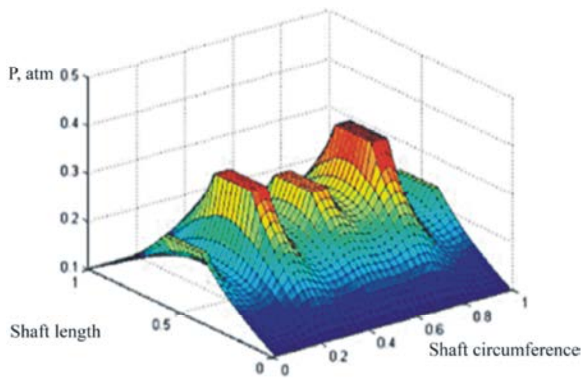


Fig. 5: Pressure distribution in the gap with grease supplying through 4 holes with pockets

displacement, the cylindrical rotation of the rotor's center of mass and the cone precession. There also is a phenomenon, known as "half-speed vortex". It appears due to the stability loss of the lubricating layer, located between two moving cylindrical surfaces under the influence of viscous friction forces. This phenomenon is typical for full-scope bearing, but at higher speeds arises in the GSB as well. There is also another type of oscillation, called "pneumatic hammer", resulting from the instant difference between gas supplied to the gap through the nozzles and flowing into the environment through the sides of the bearing.

CONCLUSION

Air bearings with forced working fluid supply into the gap between the surface of the stator and rotor (gas-static) which were discussed in this paper are intended for advanced oil-free transmissions of gas-turbine units, meant for aerospace field of usage. Rotor supports are formed by two radial GSBs and one (possibly two) axial GSB. Raising the pressure, at which the working fluid is supplied, increases load capacity and stiffness of the compressed layer and suppresses vibration. In the meantime this increases the consumption of the working fluid and reduces efficiency. The task of designing is to achieve minimum consumption of the working fluid for the given load capacity under severe restrictions in terms of vibration, dust loading of the working fluid, temperature and humidity.

Findings: The problem of designing GSB was considered. Difficulties in scaling leads to the necessity for optimizing the GSB design to meet specific technical requirements. Optimization of static load capacity and the air consumption is conducted separately from the analysis of non-stationary characteristics.

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REFERENCES

1. Bulat, P.V. and V.N. Uskov, 2012. About the survey of oscillatory motion of the gas suspended rotor for turbo-refrigerating machines and expanders. Part I. Statement of the problem. Vestnik of International Academy of Refrigeration, 3: 3-7.
2. Temis, Y.M., M.U. Temis, A.M. Egorov, V.V. Gavrilov and V.N. Ogorodov, 2012. Computational and experimental research for the development of advanced "dry" MGTE with the rotors on the gas-dynamic bearings. 12th International Salon "Engines-2012", Association Of Aero Engines Manufacturers.
3. Gavrilov, V.V., 2004. Problems and technology for creating gas bearings for perspective oil-free turbomachinery systems. Analytical review #415. Scientific works of the Central Institute of Aviation Motors, P.I. Baranov Central Institute of Aviation Motor Development, pp: 54.
4. Ravikovich, Y.A., 1998. The construction and design of sliding bearings with gas lubrication units for AE and PP. State University of Aerospace Technology: Textbook, pp: 56.
5. Bulat, P.V., O.S. Smirnova and N.V. Prodan, 2013. Application of the guided gaso- and hydrostatical bearings in the turbopump aggregates of multiple combined LRE. Fundamental Research, 4(2): 335-339.
6. Gavrilov, V.V., 2009. Problems of making auxiliary gas turbine engines with gas bearing rotors. Samara State Aerospace University Vestnik, Aviation, Rocket and Space Technology, 1: 41-55.
7. Beschastnykh, V.B. and Y.A. Ravikovich, 2010. Gas bearing for heavy gas turbine rotor. Experience in the design and implementation. Vestnik Moskovskogo Aviatsionnogo Instituta, 3: 84-94.
8. Leonov, V.P., 2004. Turbomachine rotor suspension system. The Bauman Moscow State Technical University, pp: 44.
9. Constantinescu, V.N., 1968. Gas lubrication. Mashinostroenie, pp: 712.

10. Beschastnykh, V.N., Y.A. Ravikovich and A.N. Sokolov, 2009. Evaluation of static bearing strength for a tilting-pad hydrostatic gas bearing. Vestnik Moskovskogo Aviatsionnogo Instituta, 1: 84-94.
11. Lord Rayleigh. Notes on the theory of lubrication. Philosophical Magazine, 1918, 35(1): 1-12.
12. Sheinberg, S.A., 1969. Gas-lubricated sliding supports. Mashinostroenie, pp: 336.