## TURBOGENERATOR SHAFT OIL SEALS TYPES

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**Introduction.** The hydrogen-cooled turbogenerator (TG) is fully sealed, except for the outlets of the rotating shaft. The shaft is sealed with special oil seals that do not release hydrogen from the generator housing. It should be noted that the shaft sealing operates under rather severe conditions: at the generator rotation speed of 3000 rpm, the circumferential speed at the sealing site reaches 300-400 km/h. The basic design principle of most modern turbogenerator shaft seals is separation of hydrogen and air by oil flow, the pressure of which is slightly higher than the pressure of hydrogen.

**The purpose of work.** Determination of the structure and composition of the oil seals elements.

**The results of the research.** To prevent hydrogen leakage from the generator housing or compensator in the places where the rotor shaft passes through the end covers, oil seals of the ring or end type are used [1]. An annular seal (Fig. 1) consists of a liner 2 covering the shaft 1 and a housing 3. Oil enters the gap between the liner and the shaft and is divided into two parts: towards hydrogen and towards air. The oil going to the hydrogen side prevents hydrogen leakage from the machine body through the gap between the shaft and the liner.

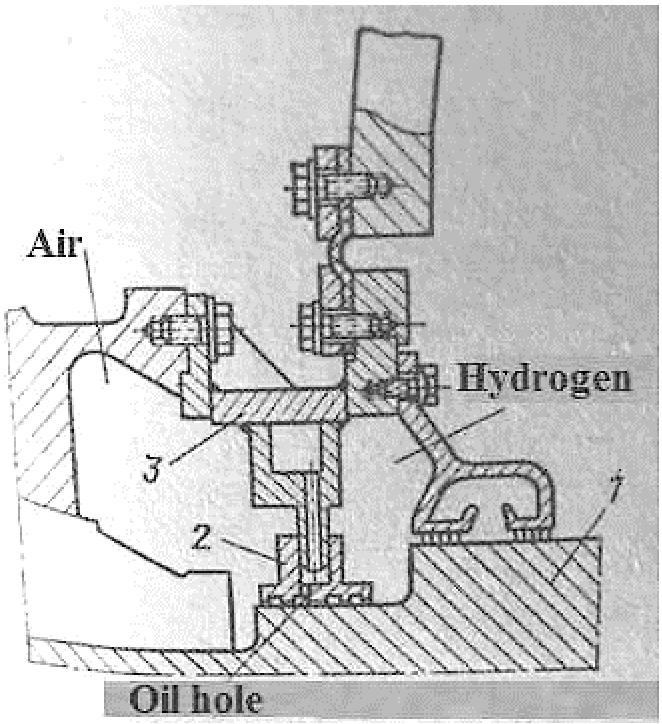


Figure 1 – Annular seal

The main advantage of annular seals is that in case of a short-term interruption of oil supply, they are usually not damaged. Melting of their liners, if it does happen, usually does not cause damage to the working surface of the shaft. However, due to the large gap between the liner and the shaft (0.3-0.4 mm) in the previously produced designs, the oil flow towards hydrogen reached 40-60 l/min. The air present in the oil was released from it, which reduced the purity of hydrogen. This necessitated the use of a vacuum unit to purify the oil supplied to the seal from air.

The mechanical seal (Fig. 2) has a liner 2 pressed against the pressure disc 1 on the rotor shaft. As in the annular seal, the oil entering the annular groove on the working surface of the liner is divided into two parts. The larger part is directed towards the air, providing lubrication of the rubbing surfaces, the smaller part - towards the hydrogen, preventing the escape of hydrogen through the gap between the liner and the disc, since the oil pressure in the annular groove is greater than the hydrogen pressure in the stator by 0.03-0.09 MPa. A smaller part of the oil also provides lubrication of the inner locking belt of the liner. Oil flow towards hydrogen due to the small gap between the liner and the disc, which is determined only by the thickness of the oil film, is small (3-5 l/min). This is the main advantage of the mechanical seal in comparison with the annular one, which makes it possible to refuse from the oil cleaning unit [2].

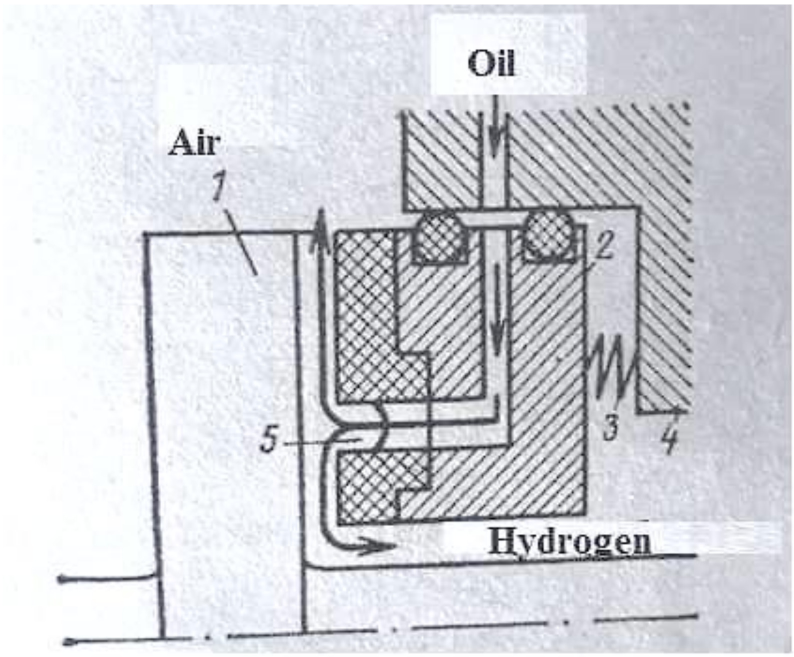


Figure 2 – Mechanical seal

Mechanical seals are divided into types depending on the way of generating forces pressing the insert to the disc, as well as on the number of autonomous oil chambers. According to the way of generating forces pressing the liner to the disc, all mechanical seals can be basically divided into four types (Fig. 3).

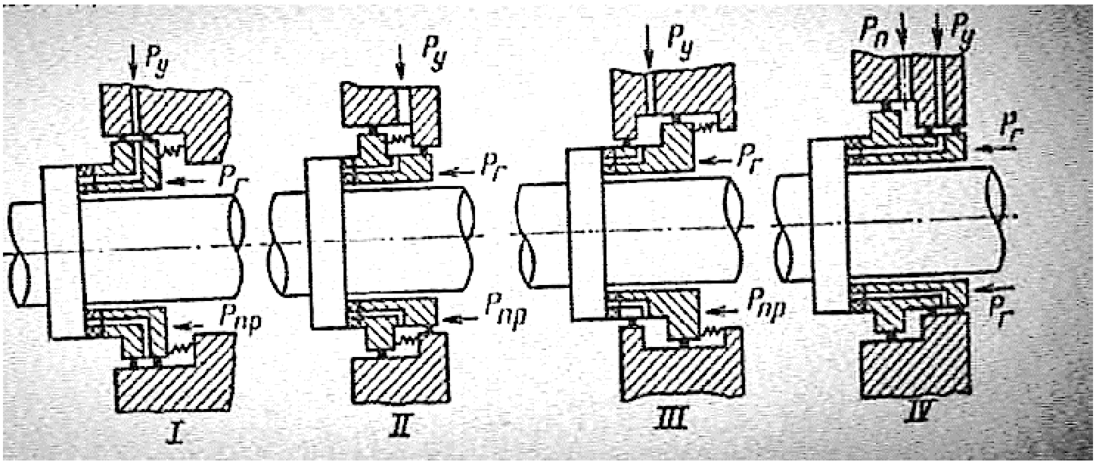


Figure 3 – Mechanical seals

Table 1 shows the ways of creating forces pressing the tab to the pressure plate and indicates in which turbogenerator each of the four types of seals is used [3].

Table 1 – Ways to create forces on the liner

| Type of seal | Ways to create efforts | Turbogenerators in which are used |
|---|---|---|
| I | Gas and spring pressure | ТГВ-200, ТГВ-200М, ТГВ-300, ТВФ-120-2 |
| II | Gas, spring and sealing oil pressure | ТГВ-165-2, ТГВ-200-2 |
| III | By gas and spring pressure. Sealing oil presses the insert from the disc | ТВФ-60-2, ТВФ-100-2 early issues |
| IV | Gas pressure and oil pressing down | ТВВ-200-2, ТВВ-200-2А, ТВВ-320-2 |

The reliability of the seals is greatly influenced by the nature of the change in the force pressing the liner to the disc, depending on the decrease in oil pressure, when due to the deterioration of lubrication, the intensity of the seal operation sharply increases. In seals of type I at emergency reduction of oil pressure, the force pressing the liner remains at the same high level, and in seals of type III it even increases. The nature of the change in force by the number of autonomous oil chambers per liner determines the requirements for the reliability of the oil supply circuit and, in particular, the permissible duration of interruptions in the supply of oil.

According to the number of autonomous oil chambers, seals are divided into single-chamber, or single-flow, and double-chamber, or double-flow. In a single-flow seal, one design of which is shown in Fig. 4, the liner is pressed against the disc by springs and hydrogen pressure on its back side. The pressure of the sealing lubricant has no effect on the clamping force. The sealing grease chamber between the housing and the liner is sealed with an oil-resistant rubber cord.

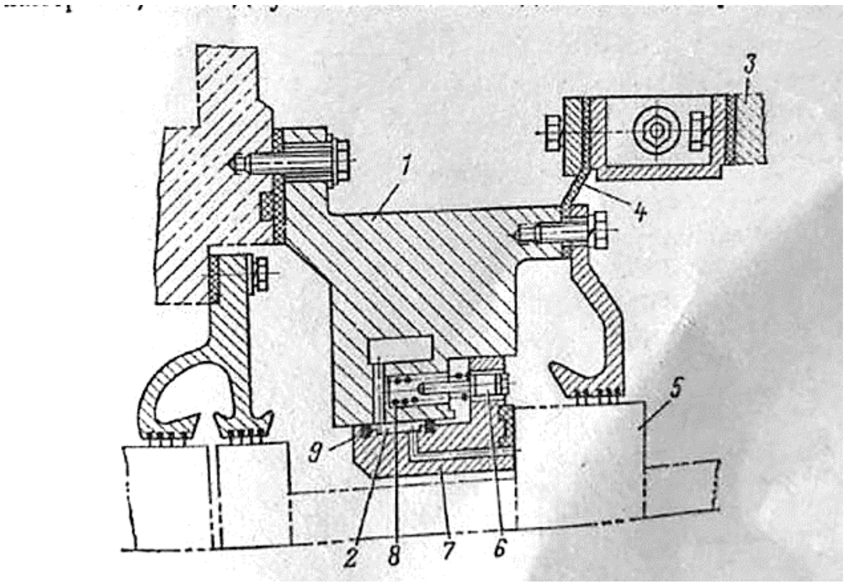


Figure 4 – Single-flow seals: 1 - seal housing; 2 - sealing oil chamber; 3 - support bearing housing; 4 - plastic diaphragm; 5 - thrust plate on rotor shaft; 6 - adjusting screw; 7 - tab; 8 - spring; 9 - sealing rubber cord.

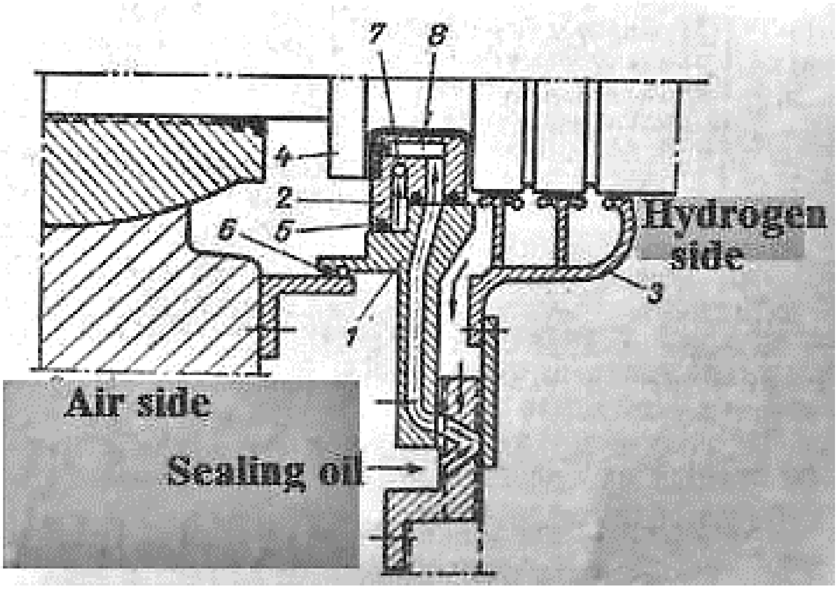


Figure 5 – Double-flow seal: 1 - housing; 2 - tab; 3 - oil traps; 4 - thrust plate on the rotor shaft; 5 - rubber O-rings; 6 - rubber gasket; 7 - pressing grease chamber; 5 - sealing grease chamber.

In two-flow seals (Fig. 5), the insert is pressed to the disc not by springs, which are not present in this seal, but by the force from the pressure of the pressing oil in chamber 7 and from the pressure of hydrogen in the generator on the back of the insert. Sealing oil enters the working surface of the liner through chamber 8. The advantage of dual-flow seals is the ability to adjust the force pressing the insert to the disc by changing the pressure of the pressing oil, that is, without disassembly.

The working surface of the end liner (Fig. 6), which is made of babbite, has wedge surfaces 1, surfaces without slope 2, inner locking belt 3, outer belt 4, radial grooves 5, annular groove 6 and lubricating holes 7. At low speeds, no pressure is generated in the oil wedges.

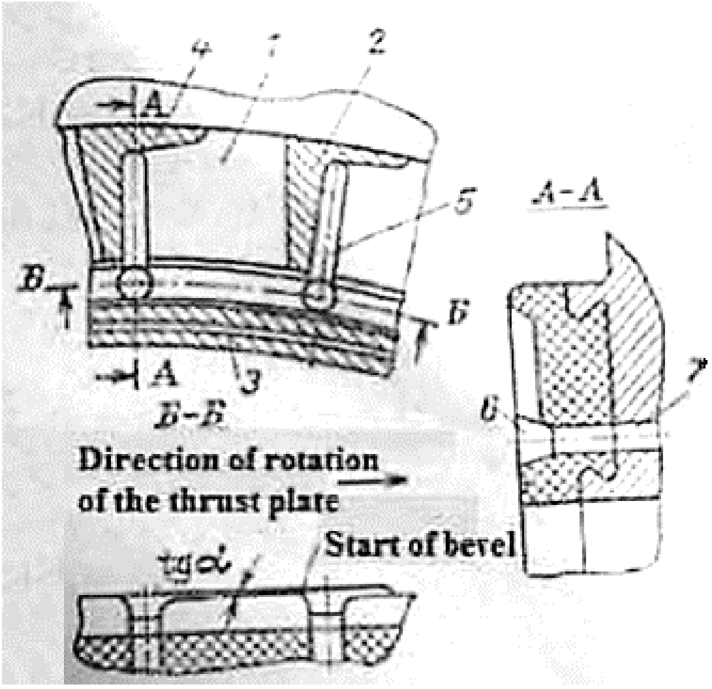


Figure 6 – Processing of the working surface of the end liner

**Conclusions.** All the forces that press the liner against the disc are absorbed by the surfaces without slopes and belts. Only at rotational speeds above 2000 rpm, the pressing force is absorbed by the entire bearing surface, and at the nominal rotational speed, most of this force is absorbed by the wedge surfaces. Thus, the liner is most stressed at speeds below 2000 rpm and especially at 400-500 rpm. This requires increased attention to the operation of the seals during stoppage and especially during start-up of the machine: it is necessary to monitor the temperature, pressure of lubricant and hydrogen, to prevent interruptions in the lubricant supply.

**References**


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