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A method of diagnosing labyrinth seals in fluid-flow machines

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ABSTRACT

Steam turbines constitute fluid flow machines which are used for driving engines of power plants, merchant and naval ships. They are commonly applied in power industry to driving electric generators. One of the impotant elements which affect efficiency of steam turbines used in power industry and for ship propulsion is state of labyrinth sealings whose aim is to minimize losses associated with steam leakage within turbine casing. Until now to assess state of labyrinth sealings has been only possible after stopping the turbine and its dismantling in order to determine values of clearances in the sealings. This paper presents a method which makes it possible to assess state of labyrinth sealings without the necessity of stopping and dismantling the machine. This is the method which allows to assess on-line state of machine sealings during its operation.

Keywords: labyrinth sealings, steam turbines, sealing diagnostics

INTRODUCTION

Steam turbines are fluid-flow machines which are used for driving power machines, and merchant and battle ships. In power engineering they are commonly used for driving electric current generators. The efficiency of those machines strongly depends on numerous factors, the main of which are the initial steam pressure and temperature [3]. That is why higher and higher initial steam parameters are used in thermal cycles with steam turbines, both stationary and used for driving ships, to obtain higher efficiency and, consequently, to reduce the costs of electric current production or ship maintenance [3].

On the other hand, increasing initial steam parameters in the turbine leads to serious problems with sealing the turbine casing. Usually this problem is solved using systems of labyrinth seals to reduce the steam leakage [1]. Additionally, some turbine producers introduce a double turbine casing in the regions of extremely high parameters of the steam. The use of this construction reduces stresses in the casing and temperature gradients in casing material. However, both the inner casing and the outer casing are still to be sealed in a proper way to eliminate excessive steam leakages. For this purpose labyrinth seals are used in the both casings [3].

METHOD OF DIAGNOSING THE SEALINGS

Fig. 1 shows schematically a supply system for a 215 MW turbine, along with the inner and outer casings.

Thus, the use of extremely high initial steam parameters in the thermal cycle has an impact on turbine construction.

The condition of the labyrinth seals in a fluid-flow machine considerably affects parameters of its performance. Therefore the information on the current state of these seals is of high importance from two mutually related points of view, which are [3]:

- ✶ possible efficiency losses caused by poor condition of the seals
- ✶ ability to plan repair actions.

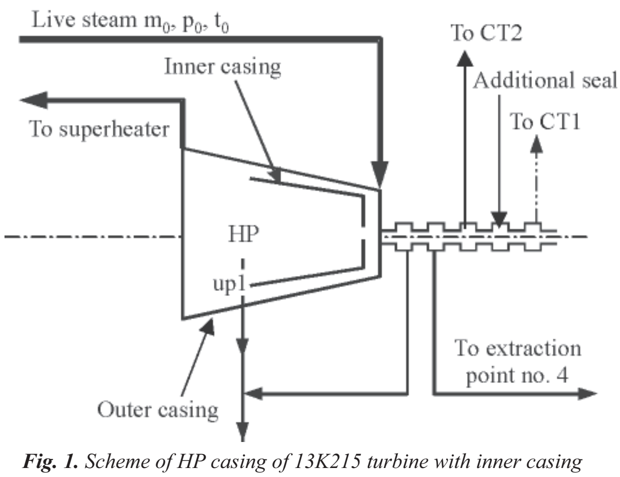

Fig. 1. *Scheme of HP casing of 13K215 turbine with inner casing*

In the past, assessing the state of such a seal was connected with stopping the machine and opening it to do necessary geometry measurements, which in case of seals situated inside the turbine (for instance in the turbine with the inner casing) required uncovering both the outer and inner casing. This, in practice, was only possible during large machine overhauls. Current information on the condition of the seal of interest during machine operation makes it possible to plan the date and range of the future machine repair.

The basic geometrical parameter used for describing the size of clearances in the labyrinth seal is the averaged clearance, defined as [1, 2]:

$$s = \sqrt{\frac{z}{\sum_{i=1}^{z} \frac{1}{s_i^2}}} \tag{1}$$

where:
s – averaged seal clearance
s_i – sizes of clearances in successive gaps of the seal
z – number of teeth in the seal.

The mass flow rate through the seal can be modelled using the equation in which the scale of leakage is related to the averaged clearance. The basic equation describing the mass flow rate through the seal is that proposed by Stodola in the form [1, 2]:

$$m = \mu \rho_1 \frac{F}{\sqrt{z}} \sqrt{\frac{p_1}{\rho_1}[1-(p_2/p_1)^2]} \tag{2}$$

where:
m – mass flow rate through the seal
μ – flow coefficient for the seal
ρ_1 – density of the medium at seal inlet
p_1 – pressure in front of the seal
p_2 – pressure behind the seal
$F = \pi Ds$ – flow area inside the seal (corresponding to the parameters)
z – number of teeth in the seal
D – seal diameter.

The above formula relates the averaged clearance to the mass flow rate through the seal. These two quantities are closely coupled by the above equation and cannot be separated from each other in this system. As a result, pressure or temperature measurements performed in front of and behind the seal do not provide opportunities for unique assessment of the mass flow rate through the seal in the situation when the averaged clearance **s** is not known. Determining losses attributed to the leakage flow through the seal requires the information on the geometry of the seal. During turbine operation, when the turbine casing is closed, we do not have the information of the current condition of the seal, as the sizes of clearances in successive gaps can change with time. A single mass flow rate equation does not provide opportunities for determining the mass flow rate when the scale **s** of the clearance changes during machine operation. This can be described using the following equation:

$$m = f(p_1, p_2, \rho_1, s, z) \tag{3}$$

where:
m – mass flow rate through the seal
p_1 – pressure in front of the seal
p_2 – pressure behind the seal
ρ_1 – steam density in front of the seal
s – averaged clearance in the seal
z – number of teeth in the seal.

The role of the averaged clearance s is of high importance when assessing the scale of power losses in the turbine. This can be seen in Fig. 2 showing the diagram of power loss recorded in the 215 MW turbine HP casing as a function of the averaged clearance **s** in the labyrinth seal situated in the inner casing. Increasing the size of the clearance in the inner casing leads to the increase of the volume of steam flowing through the seal, which in turn decreases the mass flow rate of the medium flowing through the turbine blade system. The calculations shown in Fig. 2 were done assuming the nominal averaged clearance as equal to s = 0.7 mm.

To sum up, the above discussed system is incalculable - its makes the calculation of the mass flow rate through the seal impossible when the averaged clearance **s** is not known.

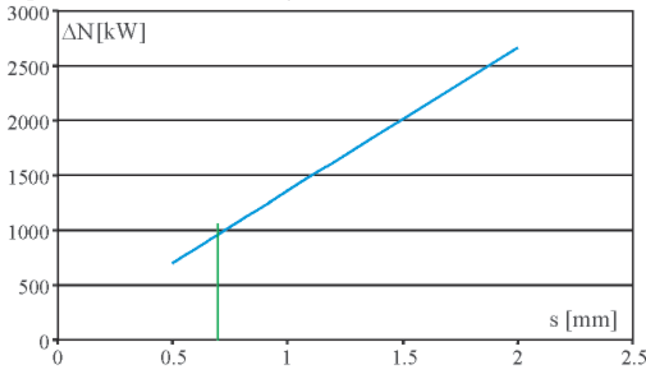

Fig. 2. *Power loss in 13K215 turbine HP casing vs. averaged clearance*

The solution presented in the article provides opportunities for on-line diagnostics of those seals, without interfering into the mechanical system of the machine. It also does not require stopping the machine to do clearance geometry measurements in the seal [4].

The essence of the new solution consists in introducing a disturbance in the flow of steam through the seal in such a way that the changes in pressure distribution in the seal which are related to this disturbance describe the current seal clearance geometry. This disturbance can have a form of an extraction system installed in the seal, through which a small but well-known volume of steam is extracted [4].

In the static element of the seal, at a selected distance from its end at least one opening is made to communicate this area with the space behind the seal, and at least one opening for measuring the pressure in the extraction area, see Fig. 3 and 4. The geometry of the communication openings and their location along the seal length are formed in such a way that the extraction flow disturbs the flow through the seal in a way sufficiently large to obtain satisfying resolution of the method, and, on the other hand, the volume of the flow leaving the seal in this way is sufficiently small to make resultant efficiency loss increase negligible [4].

Fig. 3 shows a longitudinal section of the labyrinth seal, while Fig. 4 presents its cross section.

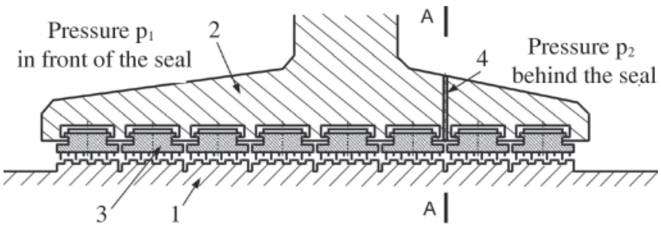

Fig. 3. *Longitudinal section of the seal with diagnostic system*

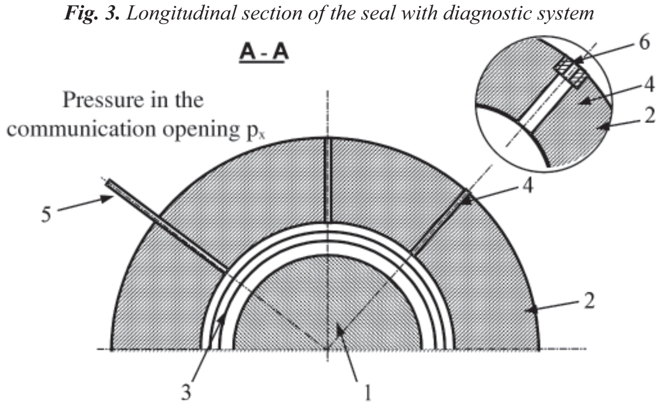

Fig. 4. *Cross section of the seal with diagnostic system*

The labyrinth seal 3 (having the form of a series of sealing segments) is situated between the shaft 1 and the static element 2 (gland clamping ring). In the clamping ring 2 a number of communication openings 4 are made to connect the inside of the seal 3 with the space behind it (the element disturbing the flow though the seal), along with the pulse opening 5 situated in the plane of communication openings 4 for measuring the pressure in the flow disturbance area.

The role of the communication openings 4 is to execute the throttling process, which can be obtained by designing relevant geometry of the opening, or, for instance, by introducing a reducing pipe 6. The locations of the communication openings (steam extraction) 4 and the pulse opening 5 are optimised using thermodynamic and flow calculations until the maximum diagnosing sensitivity/accuracy is obtained.

The first segment of the seal, marked A, is the segment between the seal inlet and the plane of communication openings (section A-A). The second segment, marked B, is the segment between the plane of openings and the seal exit.

For the seal with an extraction system (4, 6), defined as in Figs. 3 and 4, the set of equations can be written in the following form:

$$\begin{gathered} m_A = f_1(p_1, h_1, p_x, s, z_A) \\ m_B = f(p_x, h_1, p_2, s, z_B) \\ m = m_B - m_C \end{gathered} \tag{4}$$

where:

m_A – mass flow rate through seal segment A
m_B – mass flow rate through seal segment B
m_C – mass flow rate through communication openings in the seal
p_1 – pressure in front of the seal
p_2 – pressure behind the seal
p_x – pressure in the communication opening
h_1 – enthalpy of the medium in front of the seal
s – averaged clearance in the seal
z_A – number of teeth in seal segment A
z_B – number of teeth in seal segment B
Φ – diameter of nozzles in communication openings.

An essential assumption of the entire methodology is that the nozzle diameter Φ is known. As a consequence, the mass flow rate m_C through the nozzle is known as well.

$$m_C = f_3(p_x, h_1, p_2, \Phi) \tag{5}$$

When the quantities p_1, p_2, p_X, h_1 and Φ are known, we have a set of three equations with three unknowns: m_A, m_B, and s.

This set of equations can be solved, thus allowing us to determine the size of clearance s and the volumes of leakage flows in segments A and B. We assume that the nozzle geometry does not change during seal operation. Both the communication openings and the nozzles installed in them can be distributed along the entire perimeter of the seal, composing a system of openings with nozzles. In that case selecting the parameter Φ (diameter of a single nozzle) should take into account the number of communication openings over the entire perimeter.

A limiting case is the situation in which the sum of areas of the nozzles installed in the communication openings tends to zero. In this case, when m_C tends to zero, we get a seal which can be described using one equation only. As a consequence, diagnosing the state of this seal is not possible anymore.

The second limiting case is the situation in which the total area of the nozzles increases to infinity. In that case the mass flow rate through the seal segment B tends to zero. As a consequence, like in the first limiting case, diagnosing the state of the seal is impossible, as we have only one equation for the seal segment A.

Thus we can write two conditions for which the seal state diagnosis cannot be done:

$$m_C = 0 \quad \text{or} \quad m_B = 0 \tag{6}$$

In this situation, the size of the nozzles is the parameter responsible for the sensitivity of the entire methodology. Introducing the additional flow channel in the form of communication openings increases the mass flow rate of the medium flowing through the first seal segment. Selecting the size of the nozzles, i.e. the area through which the flow m_C affects, on the one hand, the ability to diagnose the state of the seal, and, on the other hand, the total volume m_A of the medium flowing through the entire seal. Increasing the nozzle area corresponds to increasing the mass flow rate m_C and the resultant mass flow rate m_A. This, in turn, will decrease the volume of the medium flowing through the turbine blade system. Therefore the basic criterion in selecting the size of the nozzles is optimisation of the system from the point of view of two aspects, the first of which is diagnosing sensitivity, and the second - extra power loss generated by the increased mass flow rate m_A of the medium flowing through the entire seal.

Practical realisation of the labyrinth seal diagnosis consists in pressure measurements in front of and behind the seal, and in the pulse opening(s). For the changing clearance **s** we will have the changing pressure p_x at the point x in the seal. This fact makes it possible to define a diagnostic parameter indicating the state of the seal and the size of the averaged clearance **s**.

Based on three measured pressures p_1, p_2, p_x a diagnostic variable P_{diag} can be defined, which can be considered a parameter indicating the level of seal degradation in time:

$$P_{diag} = (p_x - p_2)/(p_1 - p_2) \tag{7}$$

Then the recorded change of this parameter is compared to its reference value obtained during initial calibration of the diagnostic system, with possible corrections taking into account referential thermodynamic parameters. This correction compensates pressure and temperature changes of the medium in front of the seal and pressures behind the seal when the conditions of turbine operation change.

This diagnostic system was implemented in one of Polish power plants on the seal in the inner HP casing of the 105 MW turbine [4]. The theoretical distribution of the diagnostic parameter, calculated for nominal conditions of turbine operation as a function of the averaged seal clearance **s**, is given in Fig. 5 [4]. The size of the averaged clearance was assumed as equal to s = 0.6 mm.

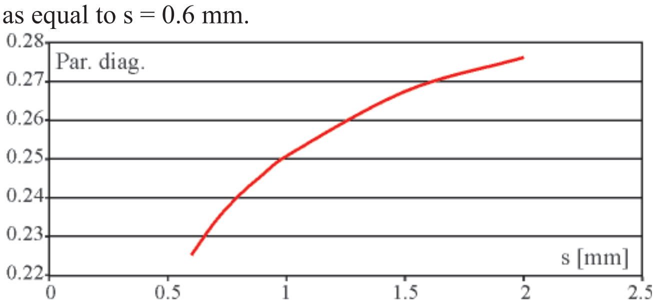

Fig. 5. *Diagnostic parameter vs. averaged clearance*

During the initial start-ups of the turbine after its overhaul, the system was calibrated and the value of the diagnostic parameter was determined as equal to 0.223 for the nominal mass flow rate 450 t/h of the steam flowing through the turbine (this value is presented in Fig. 6). Another value of the diagnostic parameter from Fig. 5 which was expected by the theory was equal to 0.225.

A few weeks after the start of the diagnostic system the failure of the generator took place. It caused rapid shutdown of the turbine, which resulted, as a consequence, in relatively large eccentricity of the shaft. This, in turn, led to the failure of the inner gland in the HP turbine.

The data obtained from turbine operation records included:

- ❖ mass flow rate of the steam entering the turbine
- ❖ live steam pressure
- ❖ steam temperature in front of the gland
- ❖ pressures in front of the gland
- ❖ pressure behind the gland
- ❖ pressure in the pulse opening.

Comparing the data obtained directly after installing the diagnostic system and directly after the failure have made it possible to assess the scale of damage, and, at the same time, confirm the efficiency of the here presented method (Fig. 6).

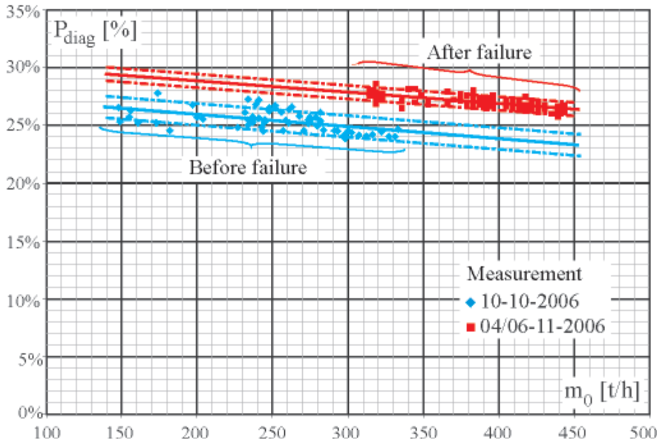

Fig. 6. *Diagnostic parameter of the gland before and after failure*

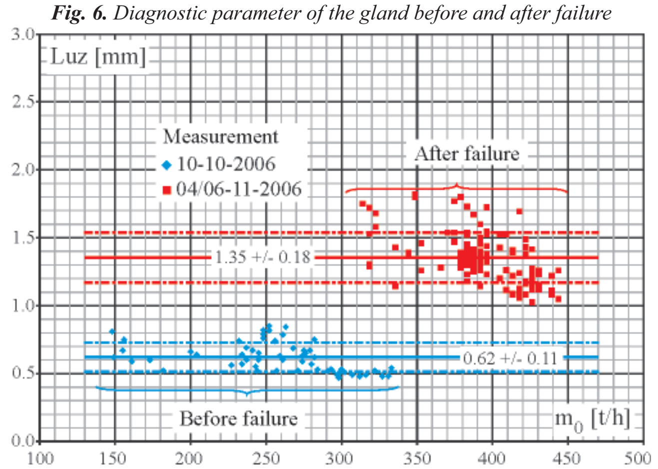

Fig. 7. *Averaged clearances in the gland before and after failure*

Processing of the obtained data has made it possible to reconstruct the averaged clearances in the gland before and after the failure, Fig. 7. In the both figures, Figs. 6 and 7, the quantities are presented as functions of the mass flow rate m_0 of the live steam entering the turbine.

As results from Fig. 7, the failure has led to the increase of averaged clearances in the gland from the nominal value of 0.6 mm up to about 1.35 mm, i.e. by about 0.7 to 0.8 mm. The average standard deviations of the spread of results are marked in the figure.

Such an increase of clearance size in the gland results in power loss by an order of 250-300 kW in this turbine, which gives the loss of an order of 2000 MWh per year (assuming each day of turbine operation at nearly nominal load).

Introducing the above described seal diagnostics methodology requires drilling communication openings in the gland casing and installing nozzles in those openings. Additionally, a duct for pressure measurements in the plane of communication openings is to be prepared. As mentioned before, all this increases the leakage flow through the entire gland. However, this increase is very small. The volume of the leakage flow through the seal in the 105 MW turbine is approximately equal to 2.2 kg/s for the case without diagnostic system, while installing it increases the leakage flow to m_A = 2.4 kg/s. The volume of the medium flowing though the nozzles is equal to about 0.2 kg/s.

CONCLUSIONS

- ❍ The essence of the here presented method is introducing a disturbance to the flow through the seal. This disturbance has a form of additional flow of steam through nozzles in communication openings. Since the nozzle does not come into direct contact with rotating elements of the turbine, its geometry does not change during turbine operation.
- ❍ Changes of the averaged clearance s in the seals, caused by possible contact between the shaft and the turbine casing, lead to differences in pressure distributions in the seal in which the extraction system was installed. When the mass flow rate of the steam through the seal is constant, i.e. without steam extraction, the pressure distribution in the seal is constant for different averaged clearances. In that case pressure measurements inside the seal do not provide opportunities for its diagnosis. The role of the steam extraction is to change the pressure distribution in the seal.
- ❍ Measurements of three pressures: in front of the seal, behind the seal, and in the plane of communication openings with nozzles, makes it possible to determine the diagnostic parameter. As a consequence, the current size s of the clearance can be assessed on-line during machine operation, without disassembling it. A necessary component of this diagnostic system is performing the three above mentioned pressure measurements. In the version implemented in the 105 MW turbine the diagnostic parameter was introduced to the controlling and measuring system of the block. Its value can be controlled during normal operation of the block, thus providing opportunities for the assessment of the current condition of the inner turbine seal [4].
- ❍ The method and principle of the above diagnostics were submitted by the authors to the RP Patent Office [5].

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