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LABYRINTH SEALS - A LITERATURE REVIEW

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Abstract. *Gaps between stationary and rotating parts are inevitable in turbomachines. The leakage flow between the rotor-stator gaps of the high pressure region to the low pressure region can be reduced using seals. The most commonly used seals or seals in turbines are those of the brush seal type, seal oring, grooves, retainer, felt, gaskets, mechanical face seal and labyrinth. The labyrinth seals found in the literature are classified as plane, straight-through, staggered, stepped converged and stepped diverged. This paper presents a systematic approach to the different types of labyrinths and mathematical models that describe the flow and the head loss through these labyrinths.*

Keywords: *Labyrinth seal, volumetric flow, head loss, turbines.*

1. INTRODUCTION

Gaps between stationary and rotating parts are inevitable in turbomachines. Large openings can lead to instability of flow and reduced efficiency of the system, while very small gaps can result in collision between rotor and stator. The sealing system in the turbomachine is very important as it increases the output power, the efficiency of the equipment and the lifetime [1].

The seals are widely used in turbomachinery to reduce the leakage flow from high pressure region, through the rotor-stator gaps, to low pressure region [2].

Among the types of sealing, there is the labyrinth type. The labyrinth seal minimizes unwanted leaks between stationary parts and moving parts. They include simple advantages such as reliability, high temperature resistance and wide range of operation in terms of pressure ratio. This type of seal is a static seal consisting of a series of cavities connected by small adjusted clearances, which are intended to promote various expansions of the sealed fluid. Therefore, pressure reduction is achieved by balancing the forces between the sides, ensuring that the fluid does not permeate the opposing sealed cavity [3].

The labyrinth seal can be used on steam, gas or hydraulic turbines. In the specialized literature are found the labyrinths of the types: straight-through, staggered, stepped converged, and stepped diverged [4].

This paper presents a systematic approach to the different types of labyrinths and mathematical models that describe the flow and the head loss through these labyrinths.

2. HIDRAULICS TURBINES

The hydraulic turbine transforms the hydraulic energy into mechanical energy, and later transformed into electrical energy by the generator. Basically, a turbine is composed of two components – rotor and the guide vanes. The rotor is a movable component, on which acts the water adduced by the guide vanes. It has blades or shells on which the flowing water acts, and the forces resulting from the water velocity give rise to a torque, which gives the axis the power and the desired movement. Already the guide vanes, fixed component of the turbine that exerts on the water a directing action, conducting the fluid to the rotor. It is also its function, in addition to transforming the energy of water pressure into kinetic energy, to regulate the flow flowing to the rotor [5].

Hydraulic turbines, whose origin dates back to 1820s, have evolved in an impressive way, as a few dozen steam horses (c.v.) supplied by the primitive models have now reached large powers. A French engineer, Benoit Fourneyron, developed the first commercially successful hydraulic turbine. Later Fourneyron built turbines for industrial purposes at achieved a speed of 2300 rpm, developing about 50 kW at an efficiency of over 80%. [6]

3. HYDRAULIC TURBINES CLASSIFICATION

a) According to the type of energy at inlet

Impulse turbines: The water upon the rotor has almost exclusively kinetic energy. There is no difference in pressure between the top and bottom of the rotor, where the pressure is atmospheric. The rotor is not in contact with the lower water level because the rotor is suspended [5].

Reaction turbines: The water entering the rotor is provided with kinetic energy and pressure energy. There is a pressure difference between the top and bottom of the rotor. The rotor is in contact with the lower water level (escape channel) through the suction pipe [5].

b) According to the direction of flow through runner

Radial turbine: It is one in which the liquid particle in its action on the receiver remains approximately on a plane normal to the axis of the turbine [6].

Axial turbines: The liquid particles run through trajectories contained in cylindrical surfaces of revolution around the axis of the turbine [6].

Tangential Turbine: The water is designed in the form of a jet over a limited number of receiver blades [6].

Table 1 presents the comparative classification of hydraulic turbines.

Table 1. Comparison of turbine classifications.

Classification	Tipo	Observação
Radial	Girard, Fourneyron, Francis (slow)	Centrifuge or exterior Centripetal or interior
Axial	Jonval, Fontaine, Propeller Kaplan Tubular, bulb, Straflo	Obsolete - - -
Mist	Francis (normal and fast)	-
	Deriaz	Similar to Rapid Francis, however, the blades of the receiver are orientable in a manner similar to that of Kaplan
Tangential	Girard Schwamkrug Zuppinger Michel Banki Pelton	- Obsolete Obsolete - - The most used

4. LABYRINTHS AND SEALANTS

In hydraulic drive machines, labyrinths and seals are parts or devices intended to achieve watertightness or, in other words, allow the smallest leakage possible [6]. The labyrinths are the interstices formed by the pairs of fixed and movable wear rings which are positioned so that they have the non-contact sealing function, causing the rings to have the smallest distance between the rotating and fixed part of the turbine rotor, resulting so in a small flow [7].

The leakage occurs at the edge of the Francis rotors, between the rotor inlet and the suction pipe, between the rotor and an upper cover. The seal is obtained in a small value for a radial clearance (between 0.5 to 2 mm) or by using labyrinths or baffles [6].

According to the fall of available raw water, the labyrinths are modified, having as example the Figure 1. In Figure 2, a type of labyrinth seal for Francis turbine is presented.

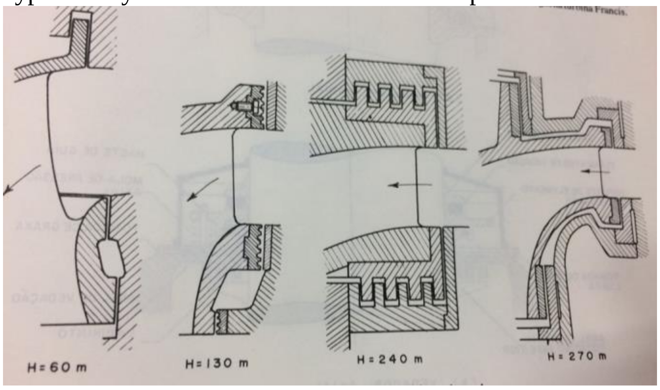


Figure 1. Rotor sealing gaskets and Francis turbine housing for different heads (H). [6]

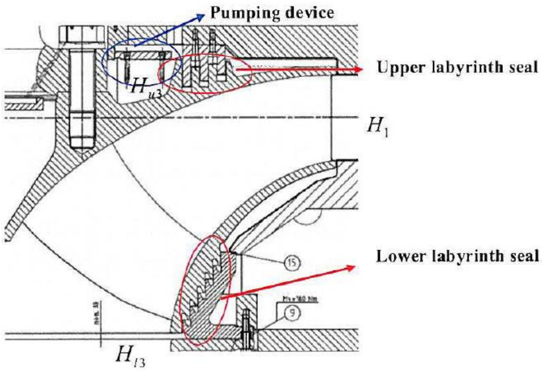


Figure 2. Schematic view of upper and lower labyrinth seals for high head Francis turbine. [12].

The sealing between the shaft and the turbine housing is obtained by means of seals, which are labyrinth devices, sealing rings and wear gloves, housed in cases cooled by circulating water.

Labyrinths of type straight-through, staggered, stepped converged and stepped diverged are found in specialized literature [4]. Figures 3 to 6 show the aforementioned labyrinths.

Figure 3. Straight-through [4].

Figure 4. Staggered [4].

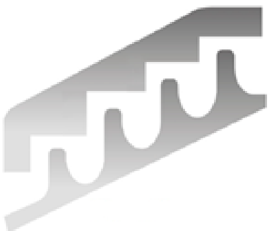


Figure 5. Stepped converged [4].

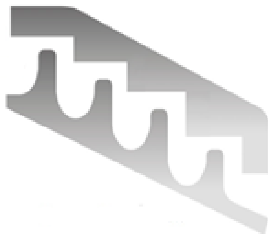


Figure 6. Stepped diverged [4].

The intercalated configuration, as well as the convergent and divergent configuration, allows greater control of leakage from one side of the labyrinths to the other [4].

The design of a labyrinth seal is usually performed by determining the numbers of teeth and cavities, as well as the sizing of the teeth length [4].

The main geometric parameters of interest are: the thickness of the tooth (t_l), the step (l), the height of the tooth (h), radial clearance (c), angle of inclination of the tooth (β), internal radius (r_i), external radius (r_0) and the tooth radius (r_{dente}). Figure 7 shows the parameters for a direct-type labyrinth. The flow conditions also have a strong influence on the performance of a labyrinth seal [4].

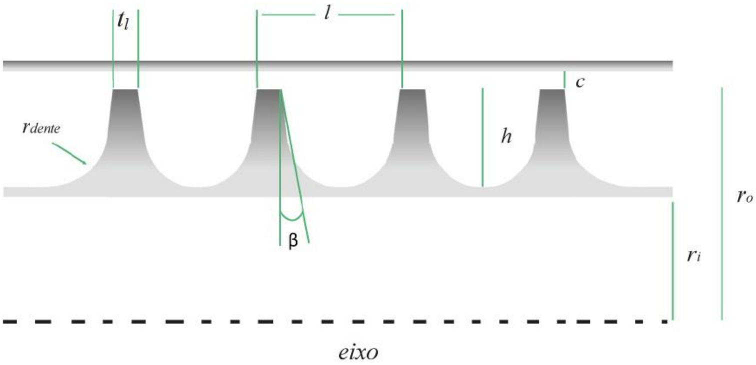


Figure 7. Main geometric parameters of a direct-type labyrinth seal.

Simulations direct labyrinth type for use in gas turbines have been proposed having the purpose of investigating the influence of variation on the geometry of mass airflow, performing geometric modifications cavity, tooth length and adding grooves. The results found by Freitas [4] for different labyrinth geometries are showed in Fig. 10 to 17.

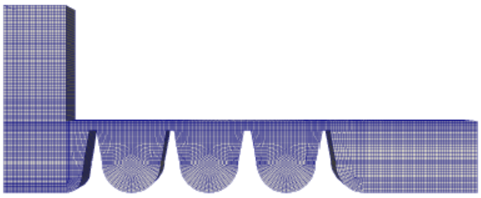


Figure 10. Case 1.

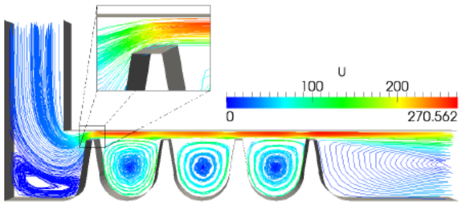


Figure 11. Streamlines case 1.

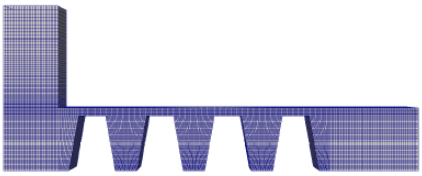


Figure 12. Case 2.

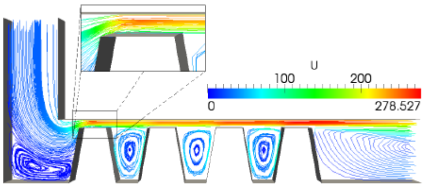


Figure 13. Streamlines case 2.

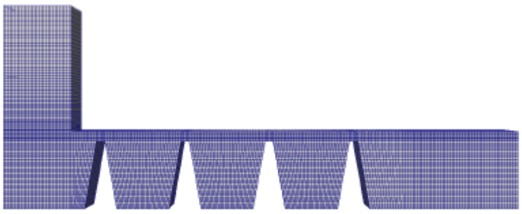


Figure 14. Case 3.

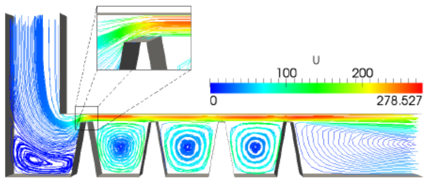


Figure 15. Streamlines case 3.

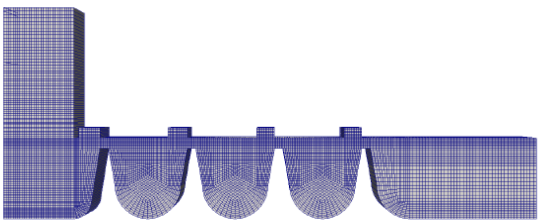


Figure 16. Case 4.

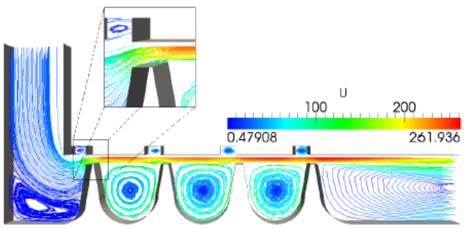


Figure 17. Streamlines case 4.

Case 1 had the length of the tooth reduced, case 2 had the cavity altered, having the straight bottom, case 3 was combined the two previous alterations, having then a labyrinth of square cavity with reduced tooth length and, finally, case 4 had the inclusion of grooves upstream of the tooth.

In all the results, there is an eddy of the fluid just at the entrance of the labyrinth. This fact is due to the existing 90º curve. As the tooth length is reduced, an increase in the area of fluid expansion is observed, obtaining a reduction in the discharge coefficient (Figure 15). The labyrinth with circular cavity presented a minor eddy, but the fluid velocity remained similar to the other geometries. The grooves (Figures 17 and 18) allowed a contraction of the fluid, resulting in a drop in the discharge coefficient.

Expansion of the fluid followed by a contraction increases the loss of charge in the labyrinth [4].

5. MATEMATICAL MODELLING

The leakage flow "Q_f" through a labyrinth is given by Eq. (1). The average speed "c_L" is given by Eq. (2) [8].

$$Q_f = c_L . A_L \tag{1}$$

$$c_L = \mu \sqrt{\frac{2\Delta p_L}{\rho}} \tag{2}$$

In these equations A_L is the annular area of the flow, μ is the empirical coefficient that depends on the geometry of the labyrinth, on the friction factor, and on the contraction and expansion coefficients, Δp_L is the total pressures loss and ρ is the density.

5.1 Straight-through labyrinth

The total pressures loss is given by Eq. (3), the mean speed "c_L" by Eq. (4), resulting in Eq. (5). The inlet and outlet loss coefficient (ζ_E and ζ_S) depends on the geometry of the labyrinth. If it is rounded or chamfered the inlet loss coefficients is: $0{,}4 \leq \zeta_E \leq 0{,}5$ [11]. The outlet loss coefficients is $\zeta_E \approx 1$ for any geometry [11]. Some author adopt $\zeta_E = 0{,}5$ e $\zeta_E = 1$ [9] e [10].

$$\Delta p_L = \rho\left(\lambda \frac{L_L}{2S}\frac{c_L^2}{2} + \zeta_E \frac{c_L^2}{2} + \zeta_S \frac{c_L^2}{2}\right) \tag{3}$$

$$c_L = \frac{1}{\sqrt{\lambda \frac{L_L}{2s} + (\zeta_E + \zeta_S)}} \sqrt{\frac{2\Delta p_L}{\rho}} \tag{4}$$

$$\mu = \frac{1}{\sqrt{\lambda \frac{L_L}{2s} + (\zeta_E + \zeta_S)}} \tag{5}$$

In these equations λ is the Darcy friction factor, L_L is the length of the labyrinth and s the width of the labyrinth.

5.2 Staggered labyrinth

It should be added for each slot (total number of slots "N_{RL}"), a loss of pressure equal to $(\rho c_L^2)/2$ and it results in the Eq. (6) [9].

$$\mu = \frac{1}{\sqrt{\lambda \frac{L_L}{2s} + \left(\zeta_E + \zeta_S\right) + N_{RL}}} \tag{6}$$

5.3 Stepped labyrinth

It is recommended the Eq. (7) for the stepped labyrinth [9].

$$\mu = \frac{1}{\sqrt{1{,}5 + 0{,}02\frac{L\prime}{s\prime} + 0{,}02\frac{L\prime}{s\prime\prime}\left(\frac{s\prime}{s\prime\prime}\right)^2}} \tag{7}$$

L_L is the length of the labyrinth and s the width of the labyrinth.

6. RESULTS

6.1 Labyrinth

Simulations direct labyrinth type for use in gas turbines have been proposed having the purpose of investigating the influence of variation on the geometry of mass airflow, performing geometric modifications cavity, tooth length and adding grooves. The results found by Freitas [4] for different labyrinth geometries are showed in Fig. 8 to 15.

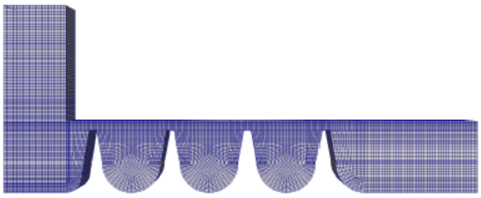


Figure 8. Case 1.

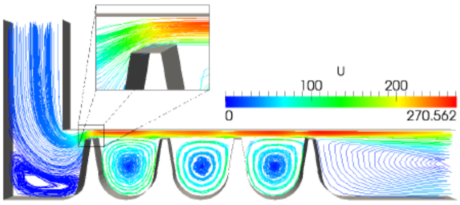


Figure 9. Streamlines case 1.

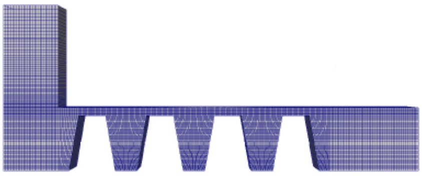


Figure 10. Case 2.

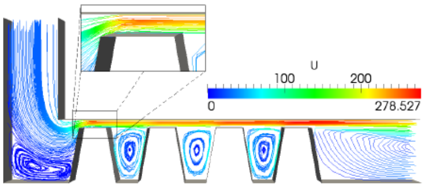


Figure 11. Streamlines case 2.

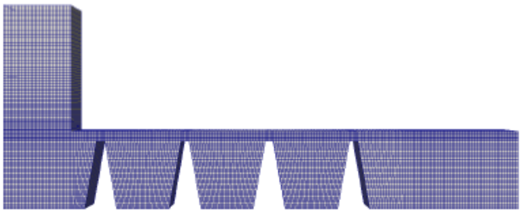


Figure 12. Case 3.

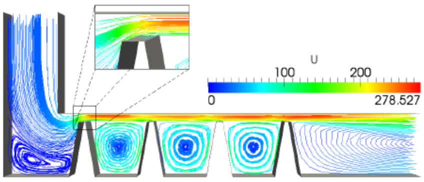


Figure 13. Streamlines case 3.

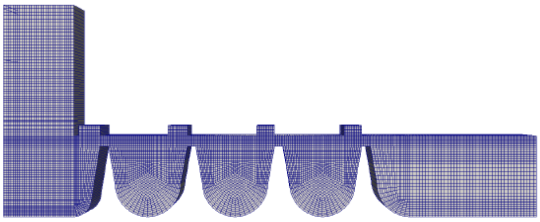
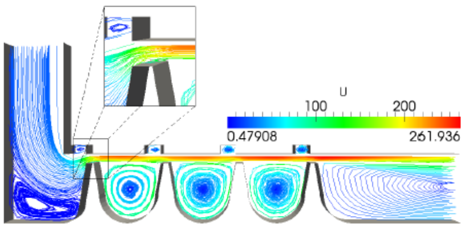


Figure 14. Case 4.



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In all the results, there is an eddy of the fluid just at the entrance of the labyrinth. This fact is due to the existing 90º curve. As the tooth length is reduced, an increase in the area of fluid expansion is observed, obtaining a reduction in the discharge coefficient (Figure 13). The labyrinth with circular cavity presented a minor eddy, but the fluid velocity remained similar to the other geometries. The grooves (Figures 14 and 15) allowed a contraction of the fluid, resulting in a drop in the discharge coefficient.

Expansion of the fluid followed by a contraction increases the loss of charge in the labyrinth [4].

6.2 Validação dos modelos matemáticos

A comparative test was performed between the equations presented and results measured at the Åbjøra power hydroelectric plant, located inside the mountain west of Lake Aurdalsfjord in Nord-Aurdal Municipality in Norway, recording the data presented in Table 2 [12].

Table 2. Comparison of the theoretical calculation and the measurements.

Q	η	ΔQ_{upper}	ΔQ_{lower}	$\Delta Q_{upper}/Q_{calculation}$	$\Delta Q_{lower}/Q$	$\Delta Q_{upper}/Q_{measurements}$
12,4	0,931	0,0290	0,0734	0,23%	0,59%	0,22%
15,7	0,95	0,0308	0,0769	0,20%	0,49%	0,19%
17,3	0,955	0,0318	0,078	0,18%	0,46%	0,17%
19,2	0,958	0,0329	0,0811	0,17%	0,42%	0,22%
19,7	0,959	0,0332	0,0817	0,17%	0,41%	0,23%
21,0	0,958	0,0340	0,0834	0,16%	0,40%	0,20%
22,1	0,957	0,0347	0,0848	0,16%	0,38%	0,15%
22,3	0,957	0,0348	0,0850	0,16%	0,38%	0,14%
23,3	0,951	0,0356	0,0865	0,15%	0,37%	0,13%
24,7	0,953	0,0362	0,0879	0,15%	0,36%	0,13%

The results are satisfactory and the difference between the values obtained from the equations and by the measurement are lower than 0,6%.

7. CONCLUSION

In order to reduce turbomachinery losses and consequently increase output power and efficiency, labyrinth seals can be used.

The variation of the geometry directly alters the analysis parameters, as a function of the change of air / water flow through it.

The choice of labyrinth type will depend on each application and for each labyrinth the geometrical parameters of interest as well as the flow conditions should be studied.

As future work, it is suggested bench tests to obtain coefficients of losses in the inlet and outlet "ζ_E e ζ_s" [9-11].

8. REFERENCES

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