STUDY OF HYDRAULIC LOSSES IN THE FRANCIS TURBINES

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ABSTRACT
The paper presents some calculation relations of the hydraulic losses in the spiral casing, stay ring and draft tube of the hydraulic Francis turbine. Numerical results will be compared to values of specific hydraulic losses found in literature, observing the evolution tendencies. By analyzing the results, conclusions will be drawn in order to establish recommended calculation relations for the hydraulic losses.

KEYWORDS

NOMENCLATURE
hp - hydraulic losses
λ - Darcy coefficient
ζ - local loss coefficient (Weissbach)
d - spiral casing characteristic diameter
v - spiral casing characteristic speed
R - spiral casing characteristic radius
α - spiral inclining angle
bo - wicked gates high
ϕ - spiral wrapping angle

SUBSCRIPTS
sc - spiral casing
i - inlet
st - stay ring
sp - spiral
ex - outside potential movement
sl - boundary layer
ms - secondary movement

1. INTRODUCTION
In the operation process of the hydraulic machines there are inevitable losses due to the friction of the fluid layers in motion, due to the friction with the solid walls and due to the local resistance such as the changes of speed direction, the changes of tube section and areas, etc. The quantitative amount of the lost energy, the irreversible change in other energy forms (ex. heat) determines the efficiency of the system. The hydraulic losses have a high amount in the energy balance of the turbo machines. A very precise quantitative evaluation of the hydraulic dissipations could permit the deepening of the energetic transfer, as well as some constructive optimizations.

Due to the geometrical complexity of the hydraulic circuit of the turbo machines, an exact mathematical model cannot be given, in order to come closer to the real situation of flow through the turbo machines. Then, the study of hydraulic losses in each one of the hydraulic circuit elements (spiral casing, stay ring, runner, wicked gates, draft tube) becomes necessary for the designer of hydraulic machines.

The first step in the determination of hydraulic losses will be assimilation of some parts of the hydraulic circuit of the machine with simple geometrical form as fix and rotative curves channels.

We have done studies of the hydraulic flows and dissipations in the hydraulic circuit of the turbo machines: spiral casing, stay ring and draft tube.

2. CALCULATIONS OF HYDRAULIC LOSSES
Usually the study of hydraulic losses in the spiral casing and stay ring are treated together. The efficiency of the energy transfer in the turbine highly depends on the structure of the output current from the spiral casing and its kinetic and energetic parameters.
The spiral casing is the first element of the hydraulic turbine and connects the adduction pipe to the other element of the turbine. Relative calculation close to the real values of hydraulic losses in the spiral casing may be used in the selection criteria during the turbine design.

The losses from the spiral casing and stay ring are composed of the longitudinal losses, local losses, and losses due to secondary movements, losses due to the section and shock variations at stay ring entry, longitudinal losses in stay ring, and losses due to section variations as well as losses trail.

In the special studies, the approach to the hydraulic losses in the spiral casing is different, starting with neglecting them up to taking into account of a great amount in the losses from the turbines -66% [4]. The losses in the spiral casing have a well define place in the energetic amount of the machine, as seen in [1], [3], [4], [9].

The calculus of the dissipations from the spiral casing is very divers. Thus, in [7], there is considered that there are only distributed losses in the spiral casing, the calculus relation being taken from the pipes.

\[ h_{pcs} = \lambda \cdot \frac{l_{sp}}{d_{sp}} \cdot \frac{v_{sp}^2}{2g} \]  \hspace{1cm} (1)

where \( l_{sp} \) – being the length of the medium path of the spiral and \( d_{sp} \) is equivalent diameter along the spiral

\[ h_{pcs} = \lambda \cdot \frac{l_{sp}}{4R_{sp}} \cdot \frac{v_{sp}^2}{2g} \]  \hspace{1cm} (2)

Which relies hypotheses of constant speed of the main current on the median sections of the spiral casing, along the spiral and the coefficient \( \lambda \) (0.01 – 0.02). Both in [1] and [2] the reference speak for the calculations of the hydraulic losses in the spiral casing (also used in the design calculations [1]).

Lasenko [8] considers that the losses distributed through the spiral casing are determined by the relations:

\[ h_{pcsl} = \frac{\lambda K^2}{2g} \cdot \frac{7(1 + \sqrt{4r_{st0} \rho_{st0}})}{C} + \frac{12 r_{st0} \rho_{st0}}{C} \left( \frac{\phi_{max}}{C} + \frac{\sqrt{4r_{st0} \rho_{st0}}}{C} \right) + \frac{2r_{st0} \rho_{st0}}{C} \left( \frac{r_{st0}}{C} + \frac{12 \phi_{max}}{C} + 16 \frac{\sqrt{4r_{st0} \rho_{st0}}}{C} \right) \]

where \( \phi_{max} \) – the winding angle of the spiral

\[ R_{st0} \] – the reference radius of the stator:

\[ K = \frac{Q}{\int_{r_{st0}}^{b} \cdot dr} \]; \( C = \frac{720 \cdot K \cdot \pi}{Q} \)  \hspace{1cm} (4)

Relation (3) takes into account the influence of the winding angle and the output speed from the spiral casing, through “\( \kappa \)”. Levin and Clermont [9] assimilate the spiral casing to the confuzor and calculates the global loss coefficient like a local resistance coefficient depends on the converging degree “\( n_0 \)” and “\( \kappa \)” confuzor angle.

\[ \zeta_c = (0,0125 n_0^2 + 0,024 n_0^3 - 0,0723 n_0^2 - 0,0044 n_0 - 0,00745) \cdot (\alpha_c^2 - 2\pi \alpha_c^2 - 10 \alpha_c) \]  \hspace{1cm} (5)

Lasenko [8] calculates the losses in the spiral casing as a sum between the distributed losses and the confuzor losses

\[ h_{pcs} = h_{pcsl} + h_{pcs} \]  \hspace{1cm} (6)

\[ h_{pcs} = \zeta_c \cdot \frac{v_{sp}^2}{2g} \]  \hspace{1cm} (7)

where \( b_0 \) – the height of the cylindrical area at the exit from the spiral casing; \( \zeta = 0,1 - 0,25 \) – the confuzor loss coefficient

In a big theoretical and experimental study of losses in the spiral casing I. Fitero [5] calculates this losses by use of a global coefficient \( \zeta_{cs} \) in the form:

\[ h_{pcs} = \zeta_{cs} \cdot \frac{v_{sp}^2}{2g} \]  \hspace{1cm} (8)

where \( b_0 \) – the height of the cylindrical area at the exit from the spiral casing; \( \zeta = 0,1 - 0,25 \) – the confuzor loss coefficient

A. Baya [3] considers the spiral casing without stator and relates the losses hydraulic to the cylindrical section from the exit of the spiral casing, with a relation like:

\[ h_{pcs} = \zeta_{cs} \cdot \frac{Q^2}{2 \pi^2 g \cdot D_{cs}^2 \cdot b_0^2 \cdot \rho_{st0}^2} \]  \hspace{1cm} (10)

where \( D_{cs} \) - the diameter specific to the exit from the spiral casing; \( \rho_{st0} \) – the obstruction factor of this sections by the stay ring blade.

After undertaking an experimental study of the loss coefficient in the prismatic channels, he recommends \( \zeta_{confuzor} = 0,235 \) which is in accordance with the results given by I. Fitero [5].

M. Tamas [11] relates the losses to the exit section from the spiral casing by use:

\[ \]
\[ h_{Pc} = \zeta_c \cdot \frac{Q^2}{2\pi^2 g \cdot D_{ecs}^2 \cdot b^2_0} \]
\[ = \zeta_c \cdot \frac{k^2_0 \cdot n^2 \cdot D^2}{2\pi^2 g \cdot \left(\frac{D_{ecs}}{D}\right)^2 \cdot \left(\frac{b_0}{D}\right)^2} \]
\[ (11) \]
where \( D \) – the diameter of the runner of the turbine
\[ \frac{D_{ecs}}{D} = 1.55 \pm 1.7 \]
\[ \frac{b_0}{D} = 0.08 \div 0.15 \]
\[ (12) \]
which can be stated for each concrete case.

Klein and Reininger [6] recommends:
\[ h_{pcs} = \zeta_c \cdot \frac{v_1^2}{2g} \]
\[ (13) \]
\[ h_{pcs} = \zeta_c \cdot n^2 \cdot D^2 \]
\[ (14) \]
from which we obtain:
\[ \lambda = 2^{\frac{11}{3}} \cdot I_1 \cdot D^\frac{1}{3} \cdot P_c^\frac{1}{3} \cdot V_{y10}^\frac{9}{10} \]
\[ V_{y10} = I_2 - I_3 \cdot D^\frac{1}{3} \cdot P_c^\frac{1}{3} \cdot V_{y10}^\frac{1}{10} \]
As to the stators it is recommended:
\[ h_{psc} = \zeta_c \cdot K \cdot Q^2 \]
\[ (15) \]

The draft tube leads the fluid to the lower reservoir, under the conditions of transforming the kinetic energy in the potential energy. The shape of the draft tube depends on the type of the turbine and on the degree of recovering the kinetic energy from the runner exit; this shape influences the hydraulic losses which mostly are greater in the draft tube than in the rotor [1]. In the draft tubes the total losses are made of friction losses, diffusing losses, and the kinetic energy losses.

In this case:
\[ h_{Pta} = h_{fr} + h_{pd} + h_{pe} \]
\[ (16) \]
\[ h_{fr} = \frac{\lambda \cdot L}{8 \cdot \frac{\theta}{2} \cdot \frac{\theta}{2} \cdot D} \cdot \frac{v_3^2 - v_5^2}{2g} \]
\[ (17) \]
\[ h_{pd} = 3.2 \cdot \left(\frac{\theta}{2}\right)^{1.25} \cdot \frac{(v_3 - v_5)^2}{2g} \]
\[ (18) \]
\[ h_{pe} = \frac{v_5^2}{2g} \]
\[ (19) \]
The values \( V_5 = f(H) \) and \( L/D = 3 \) are established [1] on bases of several economic studies related to the cost of the investment of the building of the power plant and on the losses of its corresponding kinetic energy.

The relation of calculation of the loss coefficient for the turbulent conditions is the following:
\[ \lambda = 2^{\frac{11}{3}} \cdot I_1 \cdot D^\frac{1}{3} \cdot P_c^\frac{1}{3} \cdot V_{y10}^\frac{9}{10} \]
\[ V_{y10} = I_2 - I_3 \cdot D^\frac{1}{3} \cdot P_c^\frac{1}{3} \cdot V_{y10}^\frac{1}{10} \]
where:
\[ I_1 = 0.0485 \]
\[ I_2 = 0.792 \]
\[ I_3 = 0.11 \]
\[ D = R_c \cdot P_c^{1/2} \]
\[ P_c = \frac{R}{R_c} \]
\[ R_c – curve radius \]

3. NUMERICAL RESULTS

Real evaluation of calculation relations of hydraulic losses from the spiral casing, stay ring and the draft tube is possible only after their real application because: “Only reality and experience bring out the truth” (A. Einstein). For the numerical calculation have been chosen a Francis turbine, designed and tested in the laboratories UCM Resita, and conclusions have been drawn on the calculation result by use of the above mentioned relation.

Relations (2), (13), (8) give values for the hydraulic losses in the range of (0.45 – 0.55%) from turbine head; Relations (11), (16) give values of hydraulic losses in the range (1.2 – 1.7%) from total head. Taking into consideration the stay ring losses which can be up to 0.05% of the total head its results that the losses from the spiral casing (the stay ring included too) and the draft tube can be estimated in the range of (0.5 – 1.8%) of the total turbine head.

4. CONCLUSIONS

After analyzing the relations that express the type of the hydraulic losses the following conclusions have been observed:
- the numerical results confirm the tendency found in literature.
- by considering the efficiency of the model with the optimum operating point equal with 90% the values of the hydraulic losses in spiral casing, stay ring and draft tube will represent 18 – 19% of the turbine total losses, which is in accordance with the results from [3], [11].
• It is recommended to calculate the losses in each one of these elements; any calculations which leads to losses greater than 2% of the turbine head asks a redimensioning of the corresponding hydraulic elements.

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