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ANALYSIS OF THE LOAD OF PELTON WATER TURBINE SHAFT AND GUIDE BEARING

Valentin Slavov Obretenov

Professor, Dept of Hydroaerodynamics and Hydraulic Machinery, Technical University of Sofia, Bulgaria

ABSTRACT

The goal of this study is to calculate the load of the guide bearing at various operation modes of Pelton water turbine, installed in Batak Hydropower Plant. The calculations have been performed in connection with the modernization of the bearing of Turbine No4 (supplied by CKD Blansko). All calculations have been accomplished through method and computer software programs for three characteristic operating conditions: start; maximum load; stop. The interaction between the runner and the jet as well as the pressure distribution on the buckets have been studied. Based on the analyses of the obtained calculation results, a number of recommendations have been made.

Keywords: Pelton Turbine, Pressure Distribution, Bearing, Jet, Bucket.

I. INTRODUCTION

Batak HPP is the first stage of the “Batashki vodnosilov pyt” cascade (in Bulgaria). It has four hydropower aggregates, each comprising a Pelton water turbine and a synchronous generator. The turbines have two nozzles and a vertical shaft. The values of the primary outlet parameters are provided in Table I.

The goal of this study is to calculate the load of the guide bearing at various operation modes of Turbine No.4 in view of the modernization of the above-mentioned bearing. The water turbine has been designed and manufactured by CKD Blansko (Czech Republic) in 1963.

TABLE I. Main parameters of the turbine

Parameter	**Value**	
	Rated	**Maximal**
Head, *H, m*	385.0	413.5
Flow, *Q,* m^3/s	3.4	3.4
Output power, P_a*, MW*	11.3	12.1
Frequency of rotation, *n,* min^{-1}	428.6	810.0
Main diameter, D_1*, mm*	1800	

II. DETERMINING THE VELOCITY AT THE OUTLET OF THE NOZZLE

The parameters of the flow at the outlet of the nozzle have a significant impact on the interaction between the jet and the buckets-hence the necessity for calculating the parameters of the flow through the nozzle. The calculations are performed in order to determine the value of the head at a maximum opening of the nozzle by means of OPTNOZ computer program [1].

The geometry of the nozzle is specified in the design documentation of the turbine. The calculations are performed under the condition that the flow is both potential and axis-symmetrical. Fig.1 shows the velocity distribution along the stream lines at the outlet section of the nozzle.The average value of the jet is determined by utilizing the data from the velocity distribution at the outlet section of the nozzle.

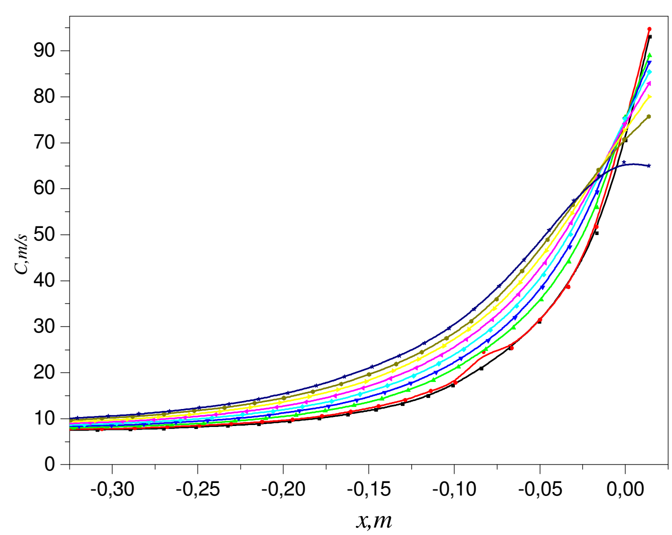

Fig. 1. Velocity distribution at the nozzle

III. ANALYSIS OF THE BUCKET STREAMLINE PROCESS

3.1. Velocity and Pressure Distribution on the Bucket

Determining the distribution of the relative velocity and the pressure at the front section of the bucket provides an opportunity both for a more accurate calculation of the bucket load and for appraising the qualities of the bucket itself.

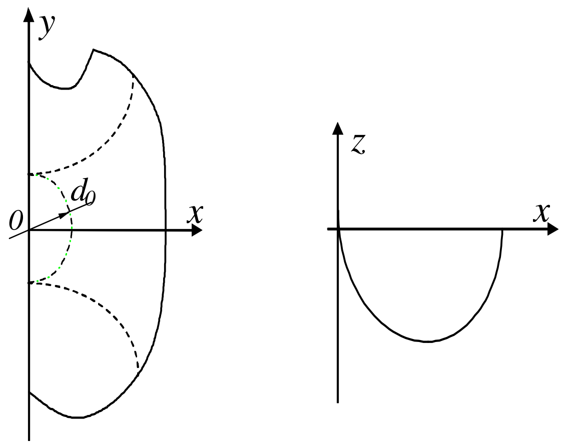

Fig. 2. Bucket design diagram

The calculations are performed in accordance with method [2] by means of PWAN computer program. Fig.2 shows the direction of the bucket co-ordinate axes, and Fig.3 demonstrates the distribution of relative pressure $p^* = \frac{p}{\rho g H}$, and of relative velocity (w) at the cross-sections (y=*const*).

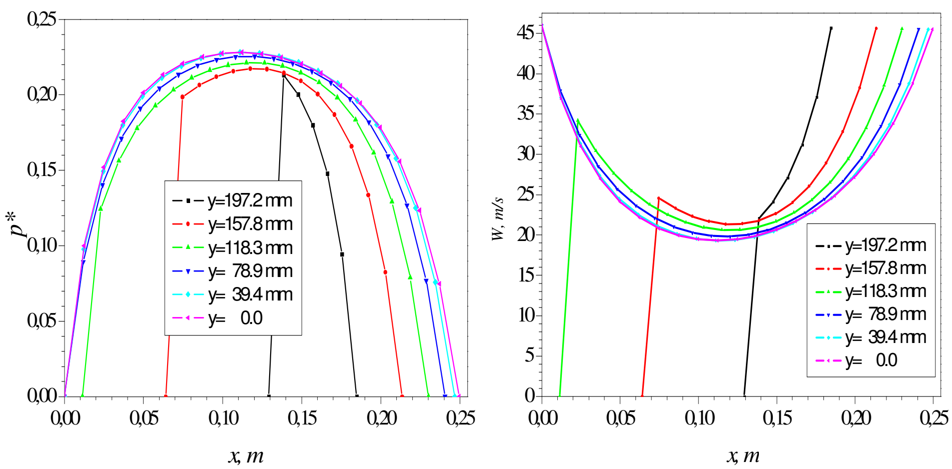

Fig. 3. Distribution of the relative pressure and velocity

3.2. Kinematics of the interaction between the jet and the bucket

In regard to the load of the wheel, the shaft, and the guide bearing of the turbine, respectively, the study of the kinematics of interaction between the jet and the buckets is especially important. Determining the angle at full injection, the timing of attack for each bucket, the trajectory of the water particles, and so forth, is particularly important. Fig.4 shows a diagram of the interaction of the bucket with the jet in accordance with the results of the calculations. Line (a) shows the interaction of the inlet edge at the incision of the bucket into the jet, and along line (b) are shown the last particles cut off from the jet that is attacking the incision edge. Line (c) shows the position of the knife, when the latter is perpendicular to the jet axis.

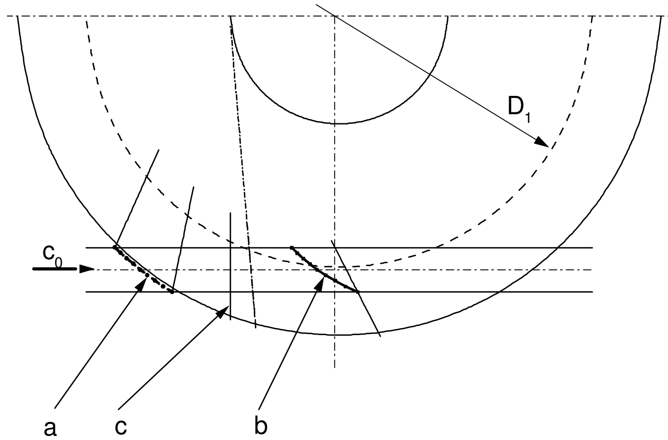

Fig. 4. Interaction of the buckets with the jet

The calculations are performed in accordance with method of [3] by means of BJET computer program. Some results (regarding the nominal and the maximum head values) are presented in Table II.

TABLE II. Results of calculations

Parameter	Value	
Head, *m*	385.00	413.50
Flow, m^3/s	1.70	1.70
Jet diameter, *mm*	160.00	157.00
Jet speed, *m/s*	84.29	87.36
Injection angles, *deg*		
• Active arc	62.72	61.09
• Perpendicular knife	-19.00	-19.78
Jet attack angles, *deg*		
• Initial	-44.28	-44.18
• Full jet action	-31.12	-31.26
• Final	18.44	16.92
Time for full attack, *ms*	8.46	7.62

IV. DETERMINING THE PRESSING FORCE OF THE JET ON THE BUCKET

4.1. Regular Mode of Operation

If the distribution of pressure at the front section of the bucket is known, the pressing force is determined according to the following expression:

$$F = \int_s p ds \,. \qquad (1)$$

In this case, the pressure distribution is known from the calculations in (III.1) and the range of the streamline part is determined according to [4]. When the inlet ridge is positioned perpendicularly to the stream axis, the integral pressing force on the bucket could be determined according to [5]:

$$F_л = \rho\, Q_л w_1 \cos\beta_1 (1 + \xi \frac{\cos\beta_2}{\cos\beta_1}), \qquad (2)$$

where $Q_л = \pi\, d_0^2 w_1 / 4$ is the discharge on the bucket (in relative movement); β_1 and β_2 are, respectively, the inlet and outlet angles of the bucket. Regarding the central cross-section ($y = 0$ - Fig.2), $\beta_1 = 15.6^0$ and $\beta_2 = 16^0$; $\xi = w_2 / w_1 = \sqrt{1 - \frac{2gh_v}{w_1^2}}$ is a coefficient that characterizes the hydraulic losses h_v on the bucket [7]; w_1, w_2 are, respectively, the relative velocities at the inlet and the outlet of the bucket.

Comparison of the values of the force obtained by implementing both methods demonstrates satisfactory correlation.

4.2. Start Mode of Operation

In this case, it is more convenient to present the expression of the force on the bucket in following form [5]:

$$F_л = \rho\, Q c_1 \frac{(\cos\alpha_1 - \psi)^2}{\cos\beta_1}\left(1 + \xi \frac{\cos\beta_2}{\cos\beta_1}\right), \qquad (3)$$

where Q is the jet discharge; α_1 is the angle between the peripheral and the absolute values of the velocity at the inlet of the bucket; $\psi = u / c_1$ is a regime parameter; $u = \pi D_1 n / 60$ is the peripheral velocity.

Considering that $\psi = 0$ for this mode of operation, the force is to be obtained as follows:

$$F_л = \rho\, Q c_1 \frac{\cos^2\alpha_1}{\cos\beta_1}\left(1 + \xi \frac{\cos\beta_2}{\cos\beta_1}\right). \qquad (3a)$$

4.3. Stop Mode of Operation

The brake nozzle is activated, wherefrom the impact of the force on the bucket rear side is obtained following expression [5,6]:

$$F_л = \rho\, Q_p c_p (1 + \psi), \qquad (4)$$

where Q_p is the discharge of the brake nozzle, and c_p is the average jet value.

V. DETERMINING THE LOAD OF THE GUIDE BEARING

5.1. Mode of Maximum Load (Mode A)

Fig.5 shows the force-load diagram. An orthogonal co-ordinate system, whose axes are directed along the milestone axes of the power station has been used. Axis x is directed along the longitudinal axis of the engine room; axis z coincides with the shaft axis of the aggregate, and axis y is perpendicular to the other two axes.

The effective force of the work of the two jets on the wheel is calculated under the following conditions:

- The jet axes are tangent to the circumference, whose diameter is equal to the main wheel diameter, and are located within the plane of the same circumference;
- Only the forces resulting from the interaction between the jet and the operation buckets are considered;
- The shaft and the bearings are non-deformable.

- Under the above conditions, the derivatives of the effective force along axes x and y is obtained as follows:

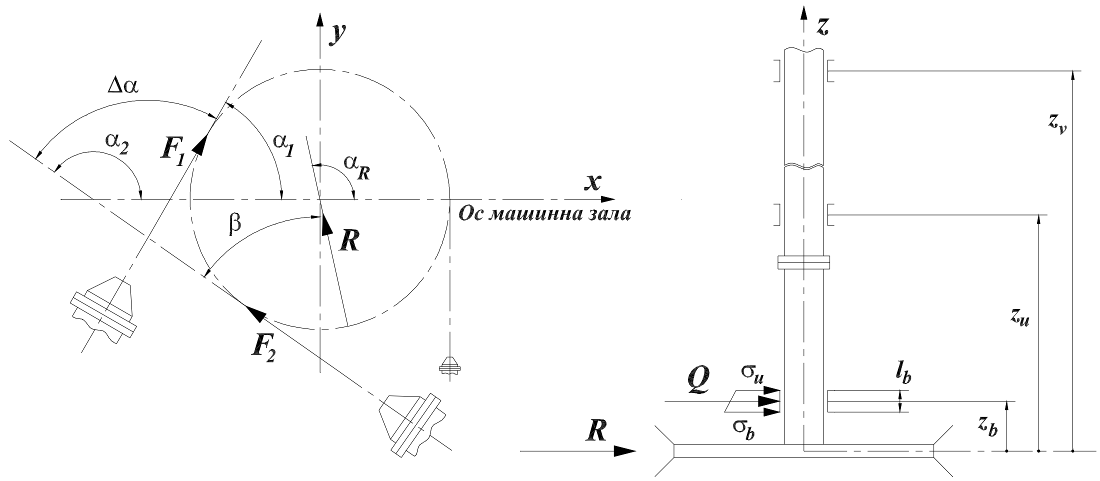

Fig. 5. Wheel and guide bearing load diagram

Under the above conditions, the derivatives of the effective force along axes x and y is obtained as follows:

$$R_x = F_1 \cos\alpha_1 + F_2 \cos\alpha_2,$$
$$R_y = F_1 \sin\alpha_1 + F_2 \sin\alpha_2, \tag{5}$$

where F_1 and F_2 are the compressive forces of the jets of both nozzles (N1 and N2). Angles α_1 and α_2 could be seen on Fig.5.

The effective force is determined according to the following expression:

$$R = \sqrt{R_x^2 + R_y^2}\,. \tag{6}$$

The direction of the effective force is determined from angle α_R (Fig.5):

$$\alpha_R = arctg\frac{R_x}{R_y}\,. \tag{7}$$

Force $\boldsymbol{T}$ that loads the turbine bearing is determined according to the position of the bearings (Fig.5):

$$T = R\frac{z_u}{z_u - z_b}\,. \tag{8}$$

TABLE III. Geometrical parameters

α_1	α_2	$\Delta\alpha$	β	z_u	z_b	z_v	l_b
deg				*mm*			
60	145	85	55	3495	600	6935	180

The values of the geometrical parameters, necessary for determining the forces in Fig.5, are provided in Table III:

5.2. Start Mode (Mode B/B1)

The turbine is started by means of nozzle N1 (the left nozzle - Fig.5). A needle opening of $a_0 \approx 0.2a_{0\max}$ is necessary to synchronize the hydroaggregate. Maximum pressure on the buckets is attained in the following two cases:

- The needle opens immediately at $a_0 \approx 0.2a_{0\max}$ (mode B1);
- The wheel remains static (emergency mode B).

In these two essentially hypothetical cases (though particularly interesting in terms of the extreme load of the bearing), the pressing force is calculated in expression (2a). The discharge at the above-mentioned opening is determined by the discharge characteristic: $Q_{a_0=0.2} = 0.58m^3/s$.

The average jet velocity at maximum value for the head at this operation mode, immediately before the buckets, is $c_0 = 87.37m/s$. If angles $\alpha_1 = 0, \beta_1 = 0, \beta_2 = 0$ (in this case the pressure is at its maximum), the pressing force of the jet in this operation mode is obtained as follows:

$$F_n = 2\rho Q c_0$$

The direction of force F_n is determined by the value of angle α_1.

The operation mode at which the hydroaggregate operates in parallel to the energy system, and at which nozzle N1 operates at opening $a_0 \approx 0.2a_{0\max}$, is designated as B1.

5.3. Stop Mode of Operation (mode C)

The force is determined according to equation (4). Fig. 5 shows the location of brake nozzle BN. It has a diameter $d_p = 45mm$ at the outlet. In determining the diameter of the brake jet, it is necessary to consider the jet contraction [7,8]. The contraction coefficient could be derived from the familiar formulae [5,7].

Thus, for example, if the formula of A.Sokolov is utilized for our purposes [5], the jet diameter is $d_{0p} = 39.6mm$. The average jet velocity is determined for the maximum value of the head (considering a lower value for the velocity coefficient): $c_0 = 85.57m/s$. Under such conditions, the discharge of the brake nozzle is obtained as follows $Q_p = 0.1054m^3/s$.

The axis of the brake jet is perpendicular to axis x: $\alpha_p = 90^o$ (Fig.5).

The calculated data are generalized in Table IV.

TABLE IV. Results for all modes

№	Mode	Force, N	
		Wheel	**Bearing**
1.	A	116893.2	141119.8
2.	B	101349.2	122354.2
3.	B1	29379.4	35946.9
4.	C	26516.0	32011.5

VI. ANALYSIS OF THE RESULTS

In dimensioning the guide bearings of the water turbines, it is necessary to introduce a parameter of specific pressure p_s as determined according to formula [4]:

$$p_s = \frac{F}{dl}, \tag{9}$$

where F is the radial force; d is the shaft diameter; $d = 600mm$; l is the length of the bearing; $l = 180mm$.

One more criterion is often utilized in the construction and exploitation process of the bearings of this type [4]:

$$p_u = \sqrt{p_s u^3}\,, \tag{10}$$

where u is the peripheral velocity, and p_s is measured in kgf / cm^2.

Table V presents the values of the bearing pressure as well as the values of parameter p_u with regard to the above mentioned operation modes.

TABLE V. Values of parameters p_s and p_u

№	Mode	p_s, *MPa*	p_u, $W^{0,5}$
1.	A	1.3067	176.80
2.	B	1.1320	164.56
3.	B1	0.3330	89.25
4.	C	0.2963	84.19

The data in Table IV and Table V suggests that the maximum loads of the shaft and of the turbine bearing are achieved at an operation mode with two nozzles, at full power.

The specific pressure in this case is $p_s = 1.3067MPa$. Generally, this is a relatively low value considering that, for example, the admissible specific-pressure value for bearing compositions of Russian water turbines (type Babit, model B83) is $|p_s| = 4MPa$ [4]. It is obvious (cf. Table V) that, in the other operation modes, the values of the specific pressure are considerably lower.

It should be heeded that the bearing is designed in the 50`s and that the bearing compositions Babit (model B16) used at the time are assigned admissible values $|p_s| = 1 \div 1.5MPa$ [4] in the specialized literature.

On the other hand, it should be heeded that the bearing has 10 sections, which means that the actual load will have higher value. Based on the data for the bearing geometry (the chord length of one of the sectors is $a = 145mm$; the bearing diameter is $d = 600mm$), the central angle of the sector for each of the sectors is obtained as follows:

$$\alpha = 2\arcsin\frac{a}{d}\,. \tag{11}$$

Considering the number of the sectors, an increase in the specific pressure with a coefficient $k = 1.287$ might be anticipated - i.e. the specific pressure of the heaviest-load mode A (Table V) will be $p_s = 1.6818MPa$. The values of criterion p_u show that the bearing is in the zone, where circulation lubrication is required [4].

It is pertinent to note that the inlet edge of the bucket is not radial and, therefore, the latter assumes perpendicular position to the jet earlier. Based on the calculations in 3.2, the value of this angle at maximum head value is $\varphi_g = 19.78^0$.

On the other hand, the study of the kinematics of the interaction between the jet and the wheel demonstrates that the active arc has value of $|\varphi_a| = 61.09^o$. The zone of impact of the effective pressure will be $|\Delta\varphi_1| = 24.4^o$ before, and $|\Delta\varphi_2| = 36.7^o$ after, the position determined by angle α_R

(Fig.5). Greater intesity of wearing-out of the bearing sectors should be anticipated within the above-mentioned zone. The other two possible zones of wearing out, in accordance with the discussion in V, will be determined by angles $\alpha_{R1} = \alpha_1$ and $\alpha_{R2} = \alpha_p$ - Fig.8 (angles α_{R1} and α_{R2} are read from the initial point of the co-ordinate system and along the positive direction of axis x - cf. angle α_R). It could be observed that angle α_{R2} determines a wearing-out zone that coincides with the wearing-out zone at a two-nozzle operation mode.

VII. CONCLUSION

An analysis of the results of the present study yields the following important conclusions:

1. Based on the performed tests, the load values for the guide bearing of water turbine No4 are determined for four modes of operation. The maximum load is achieved when both nozzles operate at full power (Mode A). The zone of the most intense load is thus determined.
2. The calculations show that the maximum value for the specific pressure of the bearing is lower than the admissible values for the bearing compositions generally utilized in hydropower turbine construction.
3. It should be heeded that the calculations are performed for an established mode of operation. Additional load resulting from failure to meet the conditions set out in 4.1 and 5.1 is not considered - for example, in case of an asymmetric attack of the jets. As a consequence, greater values of the forces are possible than those determined in V, but the latter are generally considered in equation (9). The safety margin of the specific pressure is also very significant.
4. The performed calculations demonstrate the opportunities afforded by PElTAD software package.

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