III International Symposium
Contemporary Problems of Fluid Mechanics
May 12-13th, 2011.
University of Belgrade,
Faculty of Mechanical Engineering, Serbia

Hydraulic Tests of the Bulb Turbine Unit at the Hydropower Plant "Djerdap 2"

Miroslav Benišek, Ivan Božić, Đorđe Čantrak and Dejan Ilić

Department of Hydraulic Machinery and Energy Systems, University of Belgrade, Republic of Serbia
Corresponding author: mbenisek@mas.bg.ac.rs

Abstract

Authors present some results of the bulb turbine complex commissioning tests of the unit No 9 at the additional Hydropower plant (HPP) "Djerdap 2". The methods and conditions under which the tests were performed are given. A brief analysis of the results is also given. Tests were performed at three different average heads: H_r = 6.62m, H_r = 7.36m and H_r = 11.77m. This investigation is of great significance for the reliable work of HPP "Djerdap 2".

Keywords: HPP, bulb hydraulic turbine, hydraulics, measurements.

1. Introduction

Hydropower plant (HPP) "Djerdap 2" is located at the river Danube, 82 km downstream the HPP "Đerdap I". Serbian part of the HPP is equipped with ten bulb double regulated four blade propeller turbine. Bulb hydropower unit includes double regulated four blade propeller turbine of the diameter D=7.5m connected with the shaft to the synchronous electrical generator, in the bulb, with rotor and stator water cooling and with thyristor exciter system and additional unit systems.

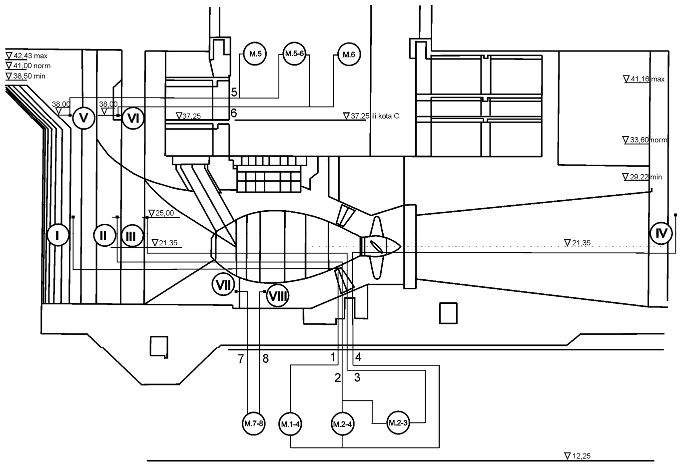
Complex commissioning tests of the bulb hydropower units in the HPP "Djerdap 2" units have been performed on the unit No 5 [2], [8], around twenty years ago. These measurements have also been conducted by the Faculty of Mechanical Engineering University of Belgrade, Serbia. In the last few years hydraulic tests of the additional unit No 9 on three different heads of average values: H_r = 6.62 m, H_r = 7.36 m and H_r = 11.77 m [3] have been performed.

Hydraulic tests of the bulb hydropower unit No 9, additional unit of the HPP "Djerdap 2" have been performed by the researchers group Center for hydraulic machinery and energy systems Faculty of Mechanical Engineering University of Belgrade in collaboration with the Institutes "Lola" and "Nikola Tesla" from Belgrade.

2. Bulb Turbine Test Methodology

Measurement methodology of the hydraulic and energy tests of the bulb hydropower unit No 9 during commissioning tests of the additional HPP "Djerdap 2" is based on the IEC standards [9], on the experience of author and coauthors and research performed in the Center for Hydraulic Machinery and Energy Systems Faculty of Mechanical Engineering University of Belgrade.

Scheme presented in the Fig. 1 gives instrument connections at the bulb hydropower unit No 9, additional HPP "Djerdap 2". Determination of the necessary physical constants, description of the methodology and methods follow.



FIigure 1. Instrument connection scheme at the bulb hydropower unit No 9, additional HPP “Djerdap 2”.

2.1. Physical constants determination

Following formulas are used for defining necessary physical constants:

- acceleration due to gravity:
  $g = 9.80617 \cdot \left(1 - 2.64 \cdot 10^{-3} \cos 2\phi_G + 7 \cdot 10^{-6} \cos^2 2\phi_G\right) - 3.086 \cdot 10^{-6} z_G,$
- water density:
  $\rho = 1000.1800014 + 0.00842284\theta - 0.00528570\theta^2,$
- mercury density:
  $\rho_{Hg} = 13595{,}08 - 2{,}47\theta + 0{,}9693410^{-3}\theta^3,$
- water viscosity:
  $\nu = e^{\left(-16{,}921 + 396{,}13/(107{,}41+\theta)\right)}.$

Constants used in these expressions are defined in the Tab. 1. Density values, kinematic viscosity and specific heat capacity for oil M.30 (Tb – A.29) have been read from the diagrams specified in the [3].

Table 1. Constants in the expressions for physical constants determination

Physical constant:	Sign:	Value:
Altitude of the machine building floor	z_G	32.75 m
HPP “Djerdap 2” latitude	φ_G	44º20’0”
Water temperature	θ	[ºC]
Oil temperature M.30	θ_u	[ºC]

Acceleration due to gravity calculated according to the expression given in the Table 1. is g=9.8055 $\mathrm{m/s^2}$.

2.2. Water level and pressure drop measurements

Water level and pressure drop measurements have been conducted in the various positions (Fig. 1):

- water level in front of the trash rack,
- water level behind the trash rack,
- pressure drop in the trash rack and its loss,
- pressure drop and quick stop log slot,
- pressure difference measured in the sections 1 and 4,
- pressure difference measured in the sections 2 and 4.

2.3. Volumetric flow rate measurements

Measurement of the volumetric flow rate through the turbine is based on the Winter-Kennedy (W-K) method, by measuring pressure difference on the under bulb support by use of manometer M.7-8. Connection scheme is given in the Fig. 2.
Volumetric flow rate determination, according to the W-K formula, follows:

$$Q = K'\sqrt{\Delta p_{7-8}} = K_p\sqrt{\left(1-\frac{\rho_v}{\rho}\right)\Delta h_{7-8}}. \tag{1}$$

Constant K_p is defined by use of the Index test.

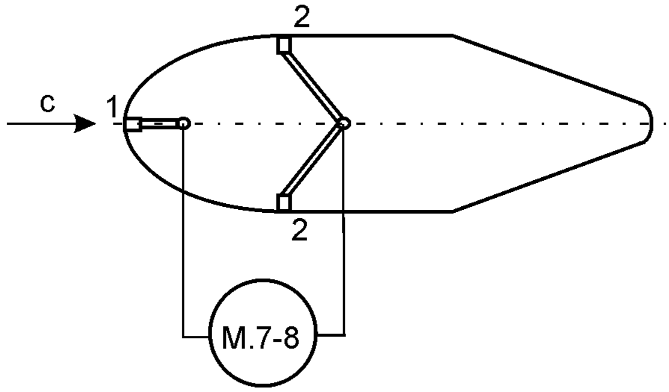


Figure 2. Horizontal Section of the Under Bulb Support – Scheme of the Manometer M.7-8 connection.

2.4. Head determination

Turbine head is defined according to the IEC 41 1991-11 norms [9]. Reference sections are: turbine inlet A-A is the averaged section of the slot of the upstream quick gate, and turbine outlet B-B is the section which goes through the measuring position IV (Fig.1.).

Turbine net head is: $H = \frac{p_A - p_B}{\rho g} + z_A - z_B + \frac{c_A^2 - c_B^2}{2g}$. Applying of everything previously said and including basic equations of the fluid mechanics, net head is defined as following:

$$H = \left(z_2 - z_4 - \frac{z_2 - z_3}{2}\right) + \frac{Q^2}{2g}\left(\frac{1}{A_A^2} - \frac{1}{A_B^2}\right), \tag{2}$$

where areas are: $A_A = 241.79\text{m}^2$ and $A_B = 165.12\text{m}^2$ precisely measured.

Turbine gross head determination is defined as the difference between the water levels in sections I and IV.

2.5. Determination of the power loss in the turbine bearings

Mechanical losses in the turbine bearings axial A and radial R3, increase the oil and casing inner energy. This implies that bearing power loss ΔP_g could be calculated in the following manner: $\Delta P_{gi} = c_{pu} Q_u \rho_u (\theta_i - \theta_u) + q$, where c_{pu} (J/kgK) is the oil specific heat capacity by *p=const.*.

Oil specific heat capacity values c_{pu} has been determined for the oil average temperatures $\theta_{um} = 0.5(\theta_i + \theta_u)$, where θ_u and θ_i $(^\circ\text{C})$ are oil temperatures at the bearing inlet and outlet (Fig. 3.); ρ_u (kg/m^3)- oil density for the

temperature θ_{um}; Q_u m^3/s - oil flow rate through the bearing. Oil volumetric flow rates Q_{ui} through turbine axial bearing and radial bearing R3, as well as oil flow rate through the generator bearings are measured by use of nonstandard orifice plate. Oil flow rates are based on the formula: $Q_{ui} = \alpha_{Bi} A_{\alpha i}\sqrt{2\Delta p_i / \rho_u}$, where: index *i*- denotes either axial A or radial R3 turbine bearing; $\alpha_{Bi}[-]$- discharge coefficient.

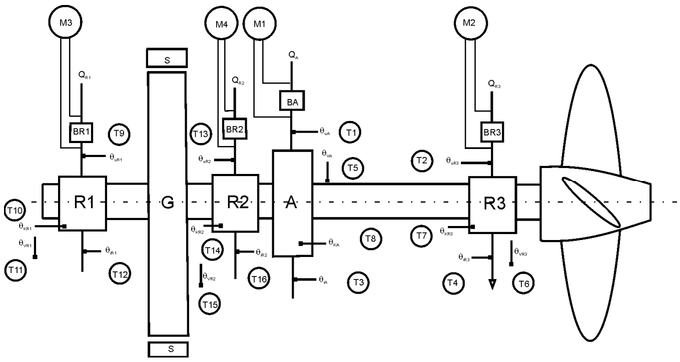


Figure 3. Connection Scheme of the Bearing Losses Measurements.

Nonstandard orifice plates, geometry specified in [3], have beein calibrated, and used during the measurements on the heads H_r = 6.62 m and H_r = 11.77 m. For head measurements H_r = 7.36 m orifice plates have been replaced, for oil flow measurements on the bearings R1, R2 and R3, with the standard ones with specified geometry in [3].

Quantity q, existing in the expression for ΔP_{gi}, presents heat exchange power from bearing casing to the air. It is calculated by the McAdams formulae: $q_i = \alpha_i^{'} A_{ki}\left(\theta_{ki} - \theta_{vi}\right)$, where the coefficient of heat transfer ($\alpha_i^{'}$) is calculated by use of McAdamsa equations. In this formulas exist appropriate nondimensional quantities, which characterize heat exchange flow. Calculated values of q_i are usually small compared to the values of ΔP_{gi} and could be neglected, what is taken into consideration during calculations.

2.6. Turbine power measurements

Power on the turbine shaft, between the axial bearing A and radial generator bearing R2 is turbine power P_T. It is calculated with the following expression: $P_T = P_M + P_{gS} + P_{gmG}$.

Stator power loss P_{gS} is defined with the: $P_{gS} = P_{Fe} + P_{Cu}$, where: P_{Fe} (kW) - iron power losses $P_{Fe} = P_{FeN}\left(U/U_N\right)^2$, P_{FeN} (kW) - power losses under the nominal voltage, U (V) – measured voltage, $U_N = 6300$ (V) – nominal voltage, $P_{Cu} = P_{KS}$ (kW) – copper coils power loss. It is calculated in the next manner $P_{KS} = P_{KSn}\left(I/I_n\right)^2$, where: $P_{KSn} = 344.17$ (kW) - power loss by nominal stator current.

Generator mechanical power losses P_{gmG} could be expressed in the next way: $P_{gmG} = P_{gv} + P_{gR}$, where: P_{gv} (kW) - fan power losses and $P_{gR} = P_{gR1} + P_{gR2}$ (kW) - power losses in the generator bearings R1 and R2.

2.7. Turbine shaft power determination

Turbine shaft power P_i, mechanical power transmited through the coupling of the runner and shaft, is determined on the basis of: $P_i = P_T + P_{gA} + P_{gR3} = P_T + P_{gmT}$, where: P_T (kW) - turbine power; P_{gA} and P_{gR3} (kW) - power losses in the axial (A) and radial (R3) bearings, respectively (determined with the above mentioned expressions); P_{gmT} (kW) total mechanical losses power in the axial (A) and radial (R3) bearing.

2.8. Turbine hydraulic power determination

Turbine hydraulic power is determined on the basis of the expression: $P_h = \rho g H Q$, where: $\rho\left(\mathrm{kg/m^3}\right)$ is water density in the turbine, defined with the expression given in Table 1. for the river Danube temperature, $g = 9.8055\ \mathrm{m/s^2}$; H (m) - turbine net head.

2.9. Turbine hydraulic efficiency coefficient determination

Turbine hydraulic efficiency coefficient, in the case of bulb turbines, is defined as the ratio of the inner power P_i and hydraulic power P_h : $\eta_h = P_i / P_h$. It should be noted that volumetric efficiency coefficient is included in the hydraulic efficiency coefficient (η_h) in the next manner: $\eta_h = \eta'_h \eta_Q$.

2.10. Runner angle and guide vane angle measurements

Existing functional relations $\beta = f(Y_{OK})$ and $\alpha_{SA} = f(Y_{SA})$, where: $\beta(°)$ - runner angle opening, $Y_{OK}(\mathrm{mm})$ - runner servomotor displacement, $\alpha_{SA}(°)$ - guide vane angle opening and $Y_{SA}(\mathrm{mm})$ - guide vane servomotor displacement, have been first defined in the commisioning tests [3]. On the basis of these relations appropriate values of the runner angle and guide vane angle openings, have been afterwards determined.

2.11. Calibration of the turbine flowmeter – defining value of the flow meter discharge coefficient K_p

Flowmeter, under bulb support, discharge coefficient K_p, is determined bz use of index test [9] on the basis of the expression: $K_p = P_i \Big/ \rho_v g H \eta_{h\max} \sqrt{\Delta h_{7-8}\left(1 - \dfrac{\rho_v}{\rho}\right)}$, where: P_i (kW) - turbine shaft power; H (m) - turbine net head; Δh_{7-8} (m)- manometer M7-8 readings and $\eta_{h\max}$ - maximum efficiency coefficient for chosen propeller and a head from propeller hillchart obtained by recalculation of the model efficency coefficients on the prototype. Propeller characteristics are given in the Report of existing turbine tests at the HPP "Djerdap 2", Laboratory for Hydraulic Machinery (LMH), EPFL, Lausanne, Switzerland [10]. Ccoefficient K_p has been detrmined by taking into consideration highest efficency coefficients for the measured propellers on the heads: H_r = 6.62 m, H_r = 7.36 m and H_r = 11.77 m. These values are given in the Fig. 4. Average value of the discharge coefficient is $K_{psr} = 337.24$. Relative deviations f_{Kp} of the measured valuea K_p are calculated on the basis of $f_{Kp} = \left(K_p - K_{psr}\right) \cdot K_{psr} \cdot 100\%$. Highest relative error is **2.57**%.

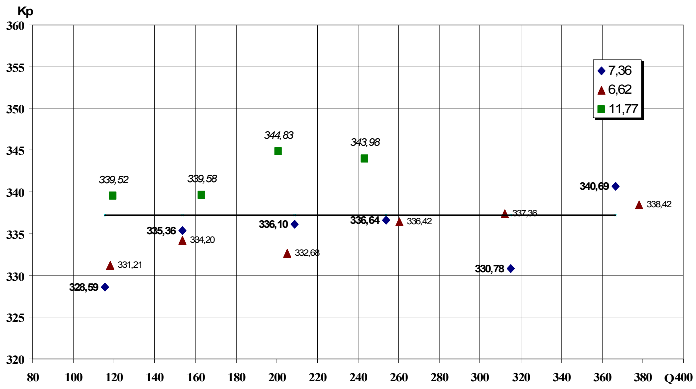


Figure 4. Discharge coefficient K_p determined by use of the index test.

3. Determination of the Optimum Combinatory Link

Turbine propeller characteristics have been measured in order to define optimal combinatory links ($\alpha_{SA} = f(\beta)$ for $\eta_{h\max}$ for various parametric values *H=const.*). Guide vane angle openings α_{SA} have been varied for constant values of runner angle opening β . Point of the maximum efficiency coefficient is, in such way, included into consideration. Values of β , depending on the net head, have been chosen in the interval of the guaranteed turbine characteristics: $\beta = -10°, -5°, 0°, +5°, +10°$ and $+15°$

All necessary values, defined by the programme, have been measured for each value of β , after achieving stationary regime. On the basis of the measured values needed values have been calculated. It was very difficult to keep net head constant during measurements. Recalculation of the measured results on the same referent head H_r has been used. Referent head, for a measuring, done for almost the same head, is: :

$$H_r = \frac{1}{n}\sum_{i=1}^{n} H_i \,, \tag{3}$$

where: n -number of measurements in one seria, H_i-measured heads in on measuring seria.

All calculated values of flow Q and efficiency coefficients η_h for various net heads H_i, of one measuring interval, have been recalculated to the referent net head H_r=*const* for measured values of the runner angle opening β and guide vane angle opening α_{SA}.

Recalculated values of Q_p and η are determined as follows:

$$Q_r = Q + \Delta Q = Q + \left(\frac{\partial Q}{\partial H}\right)_{H_r} (H_r - H) \tag{4}$$

$$\eta_r = \eta + \Delta\eta = \eta + \left(\frac{\partial \eta}{\partial H}\right)_{H_r} (H_r - H) \tag{5}$$

where: $\left({}^{\partial Q}\!/\!_{\partial H}\right)_{H_r}$, $\left({}^{\partial \eta}\!/\!_{\partial H}\right)_{H_r}$ - are flow gradient and efficiency coefficient gradient for the measured value α_{SA}=*const*, β=*const.* in the point H_r=*const.* Gradient values have been calculated on the basis of the bulb turbine HPP Djerdap 2 PRKT-750-01, obtained by scale up calculation from the model to the original turbine [6].

For values H_r=*const* (three heads) propeller diagrams $\eta_r = f_p(Q_r)$ for β=*const* have been drawn. Combinatory link $(\eta_r = f_k(Q_r))$ for defined head, is obtained, after drawing an envelope of the propeller curves $(\eta_r = f_p(Q_r))$. Combinatory link $(\alpha_{SA} = f_k(Q_r))$ is obtained by connecting points on the propeller curves $(\alpha_{SA} = f_p(Q_r))$, which correspond to the tangential points of the envelope with propeller $(\eta_r = f_p(Q_r))$.

Optimal guide vane angle openings are obtained from propeller and combinatory characteristics for referent heads and curves $\alpha_{SA} = f(Q)$ [1]. Combinatory links for constant referent heads H_r = 6.62 m, H_r = 7.36 m and H_r = 11.77 m are given in Fig. 5.

These combinatory links are:

- obtained by measurements presented in this paper,
- obtained by scale up calculation of the model tests performed in LMZ, Sankt Petersburg, Russia and
- obtained by scale up calculation of the model tests performed in LMH, Laussane, Switzerland.

Combinatory links of β and α_{SA} for measured heads H_r = 6.62 m, H_r = 7.36 m and H_r = 11.77 m is given in the Table 2.

Table 2. Combinatory links for the Hydropower Unit No 9 for measured heads H_r.

β[°]	H_r = 6.62 m	H_r = 7.36 m	H_r = 11.77 m
	α_{SA} [°]	α_{SA}[°]	α_{SA}[°]
-10	39.0	37.0	30.0
-5	48.0	45.5	37.0
0	55.0	52.0	42.0
5	61.5	58.5	47.0
10	68.0	64.5	
15	75	72.5	

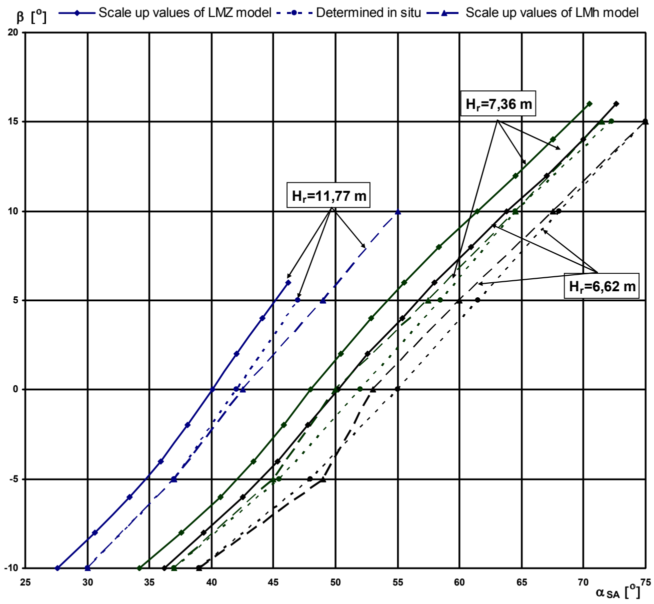


Figure 5. Optimal combinatory link (β=f(α_{SA})) for heads H_r=6.62 m, H_r=7.36 m and H_r =11.77.

4. Conclusion

PERFORMED MEASUREMENTS AND CONDUCTED ANALYZE OF THE ACHIEVED RESULTS OFFER FOLLOWING CONCLUSIONS:

- Measurements have been performed for three average net heads H_r = 6.62 m, H_r = 7.36 m and H_r = 11.77 m.
- Keeping head constant during measurements, in longer time interval, was almost impossible. Obtained measured results were, consequently, recalculated on the average head by the procedure described in the Chapter 3.
- Discharge coefficient K_p was determined by use of index test, where, for propeller maximum efficiency coefficients were taken scaled up values of efficiency coefficients from model tests performed in LMH to prototype [10]. Average value $K_{psr} = 337.24$ is calculated for all discharge coefficients K_p determined for all measured propellers. Maximum relative error was 2.57%.
- In-situ measured combinatory links of the hydropower unit No 9 have good agreement with measured combinatory links obtained in the laboratory LMH [10]. Maximum deviation is $2°$.
- Measured unit No 9 powers are smaller then scaled up ones for the same runner and guide vane angle openings for almost 12.5% comparing to the LMH results. This is explained by comparing measured flow values and scaled up results from the model tests on the prototype. Turbine unit No 9 has smaller discharge capabilities for the same heads H_r comparing to the appropriate values of LMH and LMZ. Smaller power is, consequently, achieved. Unit No 9 should work with greater runner and guide vane angle openings, after measurements defined combinatory link, in order to achieve flows and powers obtained by scale up calculations of model test results in LMH laboratory.

Acknowledgment

This work was partially financed by the company “Hydro Power Plants Djerdap” and partially funded by the grants from the Ministry of Science and Technological Development, Republic of Serbia Project No. TR 35046.

References

[1] Benišek M., Ignjatović B., Vušković I. (1982) *Efficiency scale-up for tube turbines at the operating point of best efficiency and outside the point of best efficiency,* Symposium IAHR, 13-17. September, Amsterdam.

[2] Benišek M., Nedeljković M. et al. (1989) *Commissioning and complex tests of the hydropower bulb turbine unit No 5 in HPP "Djerdap 2",* Hydraulic tests of the bulb turbine, Faculty of Mechanical Engineering, Belgrade (in Serbian).

[3] Benišek M., Božić I., Ilić D., Čantrak Đ. et al. (2006) *Commissioning hydraulic tests of the hydropower bulb turbine Unit No 9 HPP "Djerdap 2",* Hydraulic tests of the bulb turbine, Faculty of Mechanical Engineering, Belgrade (in Serbian).

[4] Bowman A.H., Schoonover. M.R. (1967) *Procedure for high precision density determinations by hydrostatic weight,* Journal of Research of the National Bureau of Standards, Vol. 71.C, No.3, July 1967, 179-198.

[5] Brand L.F. (1984) *Die Messeninrichtungen der Hydraulischen Versuchsanstalt "Brunnenmuhle",* Voith Forschung und Konstruktion, Heft 30, auf. 7.1. (in German)

[6] Vušković I., Benišek M., Nedeljković M. (1983) *Hydraulic characteristics of the bulb turbine units for HPP"Djerdap 2"; 5. Flow propeller and flow combinatory characteristics of the original bulb turbine HPP "Djerdap 2" of runner diameter D_1=7.5m for heads H=1.5-12.5m for working conditions t_V=20˚C and n=62,5min^{-1},* Faculty of Mechanical Engineering, Belgrade (in Serbian).

[7] *Chemical Engineering Handbook* (1963) McGraw-Hill Book Co., New York.

[8] Ignjatović B., Bogdanović S., Benišek M., Albijanić R. (1988) *Complex commissioning tests of the bulb units of the HPP "Djerdap 2",* Lola reports, UDK 621.224:627.84/.88., pp. 4555-4578. (in Serbian).

[9] IEC 41 1991-11 *Field acceptance tests to determine the hydraulic performance of hydraulic turbines, storage pumps and pump turbines*.

[10] LMH-EPFL *Porcile de Fier II, Bulb Turbines Comparative Model Tests*, Final Report No. 472/473.