УДК 621.224(075.8)

V.S. Obretenov, Ph.D.
Technical Universitu of Sofia
(Sofia, Bulgaria, E-mail: v_obretenov@tu-sofia.bg)

MODERNIZATION OF A FRANCIS WATER TURBINE

This study presents the results from the implementation of a design for creation of new runner of the Francis turbine installed at Assenitsa 1 HPP. The new runner design has been worked out on the grounds of analysis of the main external parameters of the turbine and its structure. With view to the specific conditions one has adopted a plan for calculation of the runner based on the consecutive application of the task related to the synthesis of the blades and analysis of the stream through them.
The results of the test made with turbine new runner show that the measured turbine efficiency values under the alteration of the generated power within the interval P=1110÷ 3630kW do comply with the modern requirements to water turbines of this type and velocityy. Increasing of maximal turbine power is with 34 percent. It should be noted that the optimal operation mode of the turbine corresponds quite accurately to the expected values as per the technical conditions.

Introduction

This work contains and presents the results from a research, directed at the modernization of the stream part of Turbine No.2 of the Assenitsa 1 HPP. This hydro power plant is located in the area of the city of Assenovgrad (Bulgaria), along the valley of the Chaya river. Three hydro-generators have been installed in it and each one is made up of a Francis type water turbine and synchronized electricity generator. The water turbines have horizontally placed axles and have been designed and manufactured by the Italian “San Giorgio” company during the 40-s of the last century. They have been launched into exploitation in 1951. Two of the generators have a power of 2 850 kW while the third has a power of 1 100 kW (according to their passport data). The objects of the current modernization will be the stream part in the zone of the runner of the turbine for the hydro-generator No.2.

I. Calculation scheme

According to the technical task, the calculated values of the basic parameters of the turbine are: head H = 70 m; discharge Q = 5 m^3/s; frequency of rotation n = 600 min^{-1}. To calculate the blade system of the runner a calculation scheme has been employed, which is based on the consecutive application of the problem for the synthesis of the blade’s surfaces and analysis of the stream through the grid of profiles [1,2].
In view of the restrictions which are enforced by the requirement for geometrical compatibility of the new runner and the guide vanes with the existing stream part, the calculation scheme includes two stages:

- Defining the optimal parameters of the turbine’s blade system;
- Parametric optimization in case the said optimal values do not conform to the requirement for geometrical compatibility.

The dimension setting of the meridional projection is performed by employing the method [3]. The setting of the profiles of the runner’s blades is performed in the conditions of potential meridional flow in the iterative correction scheme, based on accumulated experience in the process of solving similar problems. The optimization scheme is based on a planned digital experiment with three directing parameters: maximum blade diameter along the external meridional contour and the angle of development of the blade in the so-called horizontal projection. A symmetric plan has been used [4], bearing the number of the points of the carrier N=14. The values of the basic levels and variation intervals are defined on the basis of the data in the specialized literature. They are shown in Table 1.

Table 1. Values of the parameters

Parameter	Min	Base	Max	Δ
D_{1e}, mm	900	940	980	40
D_{2e}, mm	900	920	940	40
φ_p, deg	50	55	60	10
Code	-1	0	1	-

The investigations of the characteristics of the developed variants is performed by means of an algorithm and a program which have been used in practice a number of times [2,6,7]. The targeted functions, in this case, are the hydraulic losses in the runner. These are defined as the sum of the circulatory and profile losses: $h_s = h_p + h_c$.

The algorithm and the programs used for the solution of the problem related to the analysis of the flow through the blade system give the opportunity to closely follow the changes in the pressure along and on the flow-through profiles which gives the opportunity to define its minimal value p_{min}, respectively the cavity characteristics of the runner.

The regression model is a polynomial of the second degree. An evaluation is made of the importance of the regression coefficients and the model's adequacy. The optimization problems is solved with the so-called complex model of M.Box [5], which is suitable in this case due to the non-linearity of the model and the availability of restrictions to the factor area. The total minimum of the task function is defined after a set of calculations given different starting points and comparing the results obtained.

After defining the optimal values of the controlling parameters, the blade system of the optimal variant is synthesized, the flow-through of that system is being investigated and if the pressure, processed by it, conforms to the task assigned, the constructive documentation may then be prepared. Otherwise it will be necessary to reprocess the basic levels and/or the intervals in the variations of the controlling parameters and from there – to repeat the procedure.

All calculations have been performed by means of the computer program system FRTAD. With its help, and given initially set discharge values and values of the head, the optimal form of the blade system of the runner of any Francis turbine may be defined. The organization of the program system allows the preparation of data files which may be used immediately for the purpose of automatic construction of the runner and its actual manufacture.

II. Results from the calculation

In accordance with the above is synthesized meridional projection of the new runner AS01. Figure 1 shows the meridional projection of the new and the original runner. It can thus be seen, that the internal meridional contour has been preserved, but the form and the size of the outer contour have been changed as well as the position of the input and output edge of the blade. The decrease in the maximum outer diameter along the output edge (in relation to the original runner), improves the

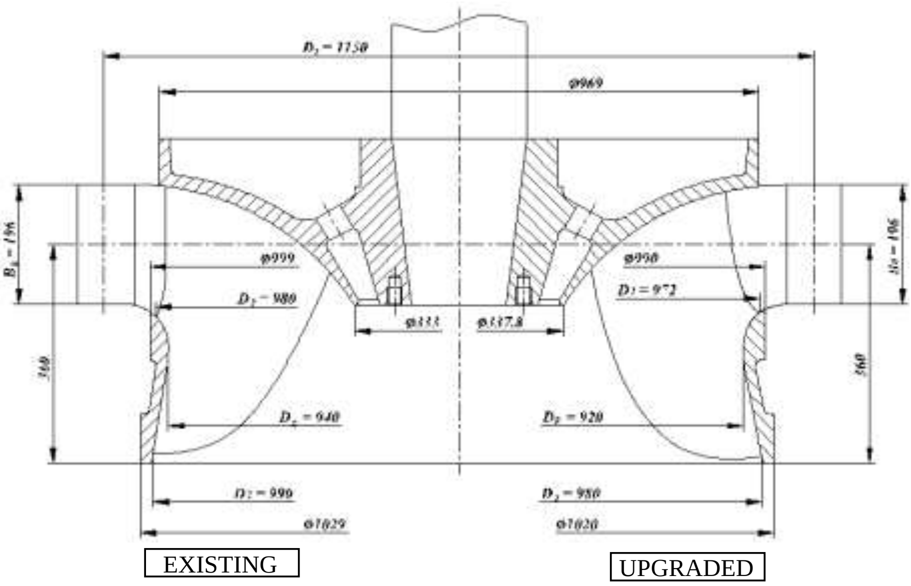

Fig.1. Comparison of the new and original runners (basic dimensions)

energy characteristics of the blade system [1], while the cross section at the input edge decreases the meridional speed within that zone and in particular in the peripheral section where energy transfer is most intensive.

The meridional and orthogonal projections of the runner blade have been shown in Figure 2 (pressure side). Figure 3 shows the profile and distribution of the relative velocity $w^{*}=w/c_m$ and relative pressure $p^{*}=p/|p_{min}|$ for the middle flow surface.

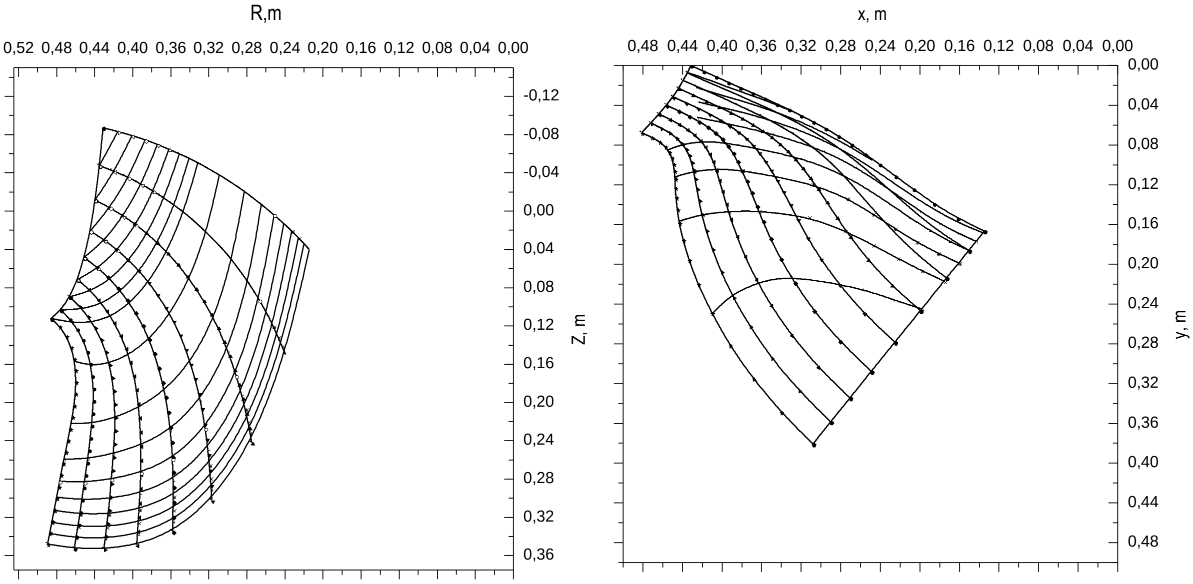

Fig. 2. Runner blade (meridional and orthogonal projection)

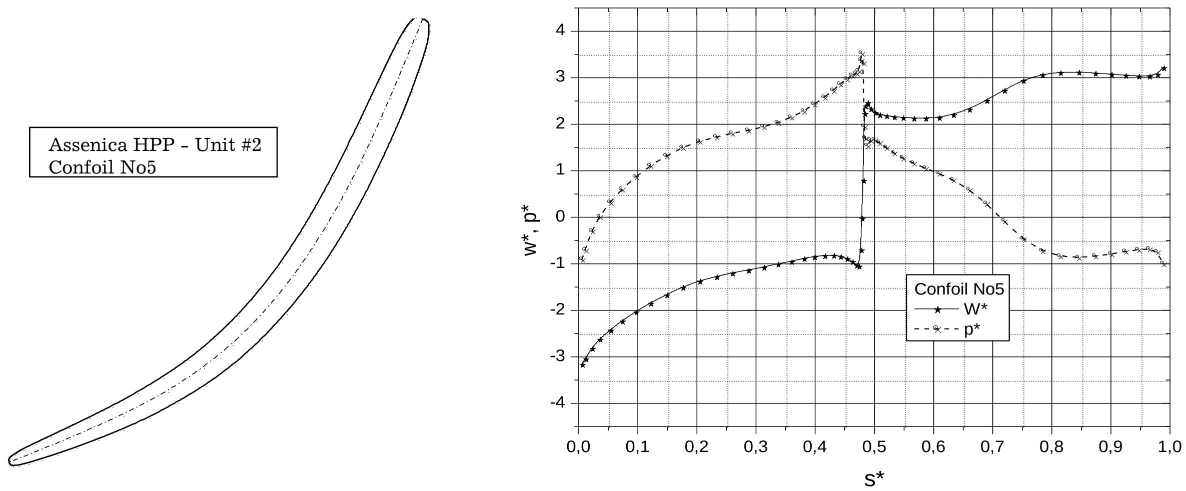

Fig. 3. Middle profile and pressure/velocities distribution

III. Experimental research

The measurement of the external parameters of the turbine has been performed in conformity to the provisions of Standard 41 of the IEC [8], where the values of the efficiency coefficient have been defined using the thermodynamic method. The major results from the testing of the turbine in various working modes have been presented in Table 2. In that table,

$n_{11} = \frac{nD_1}{\sqrt{H}}$, $Q_{11} = \frac{Q}{D_1^2 \sqrt{H}}$, $n_s = \frac{n\sqrt{P}}{H^{5/4}}$ is used to mark the reduce values of frequency of rotation and flow and the specific frequency of rotation, while a_0 is used to indicate the relative opening of the guide vanes (relation of the actual to the maximum opening). $P_{ef,}$ Q, H and η are used to mark the generator power, the effective (of the turbine's shaft) power, the flow, head and efficiency of the turbine.

Table 2. Test results

№	P_g	P_{ef}	Q	H	η	n_{11}	Q_{11}	n_s	a_0
	MW		m^3/s	*m*	-	min^{-1}	m^3/s	min^{-1}	%
1.	1.032	1.093	1.9870	69.6	0.8064	70.5	0.2480	95.9	31
2.	1.464	1.535	2.6712	69.3	0.8452	70.6	0.3340	114.7	40
3.	1.944	2.031	3.3957	69.1	0.8830	70.7	0.4254	132.8	52
4.	2.544	2.646	4.3125	68.5	0.9131	71.0	0.5424	153.5	66
5.	2.916	3.029	4.9110	68.3	0.9207	71.1	0.6186	164.9	76
6.	3.192	3.315	5.4711	67.8	0.9114	71.4	0.6917	174.1	87
7.	3.492	3.630	6.1433	67.2	0.8969	71.7	0.7803	184.3	99

The results obtained after the energy testing of the Francis turbine No.2 at Assenitsa 1 HPP serve as sufficient ground to draw the following, more important conclusions:

1. The values measured for the efficiency of the turbine when measuring the generator power within the interval $P=1000 \div 3500\ kW$ conform to the contemporary requirements toward water turbines of such type and with such outer parameters (discharge, head and specific speed coefficient). The maximum value of the efficiency of the turbine under head $H=68.3m$ is $\eta_{max}=92.07\%$. Figure 4 shows the relation $\eta = f(P_{ef})$ of the tested turbine. The broken line marks the field of errors from the measurements.

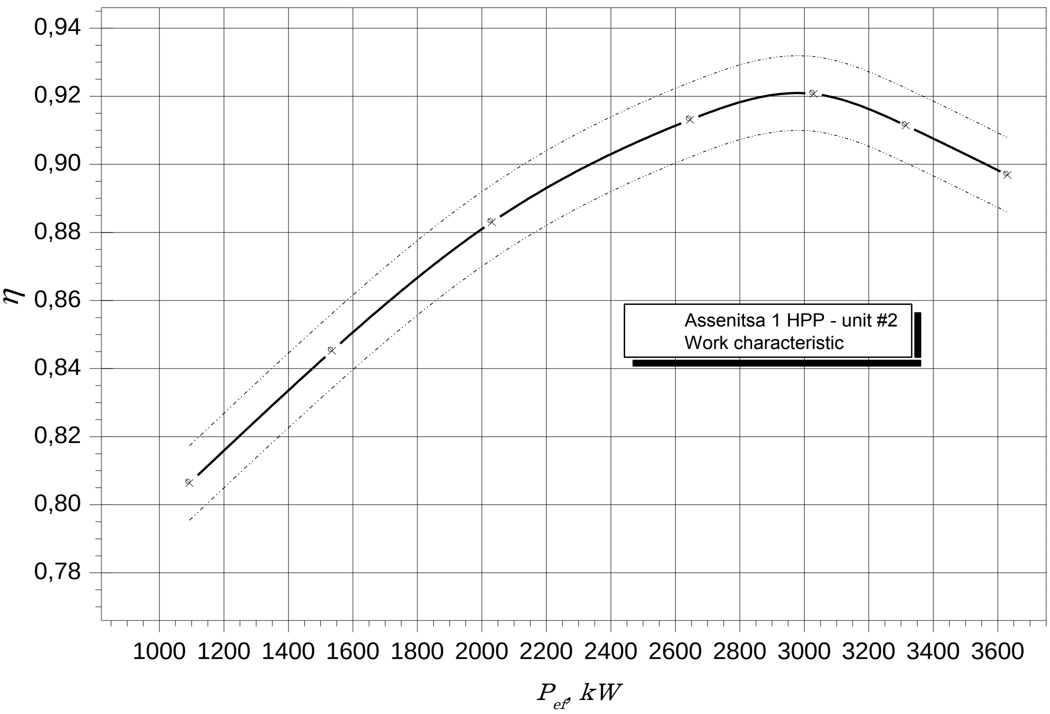

Fig. 4. Working characteristics

2. Of interest also is the comparison of the working characteristics obtained as a result of the measurements performed on Turbine No. 1 (with the same calculated parameters), obtained after the measurements made before that. It must be noted, that the conditions are practically the same.
Figure 5 shows the comparison of the working characteristics of the two turbines. It must be born in mind that the measurements made on Turbine No.2 come immediately after the modernization of the stream part.
It can be seen, that as a result of the performed modernization, the efficiency of the working process is substantially increased within the entire exploitation range. In the optimal working mode (P_g≅2300kW), turbine No.2 operates with around 6% higher efficiency coefficient while in its optimal mode turbine – by nearly 10% higher efficiency coefficient than turbine No. 1.

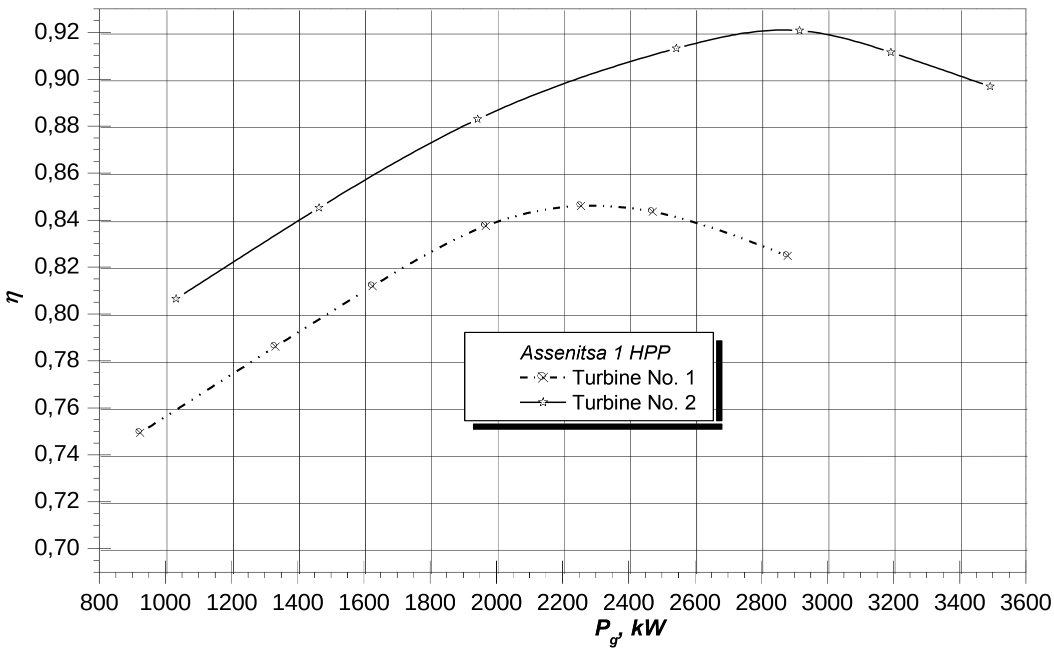

Fig. 5. Comparison of working characterictics

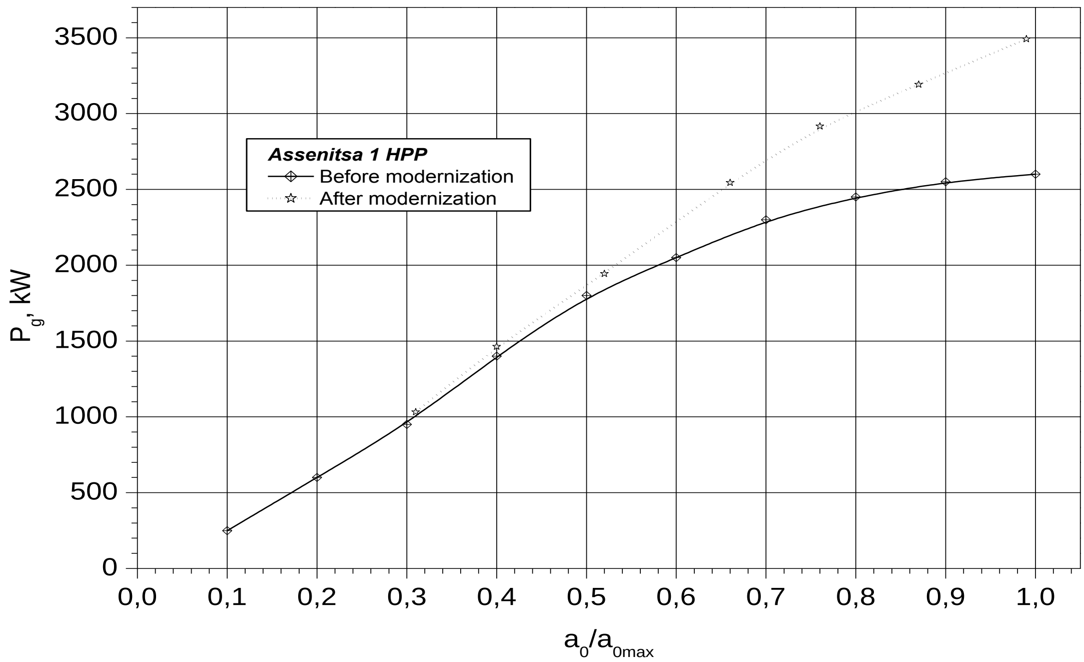

Fig. 6. Comparison of the power

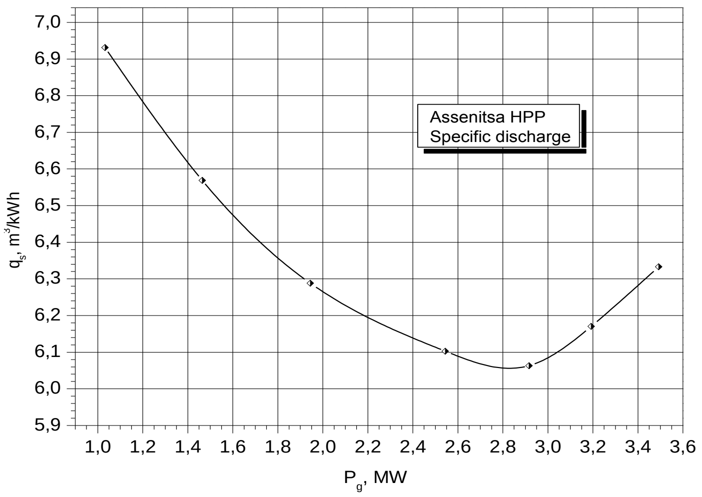

Fig. 7. Specific discharge

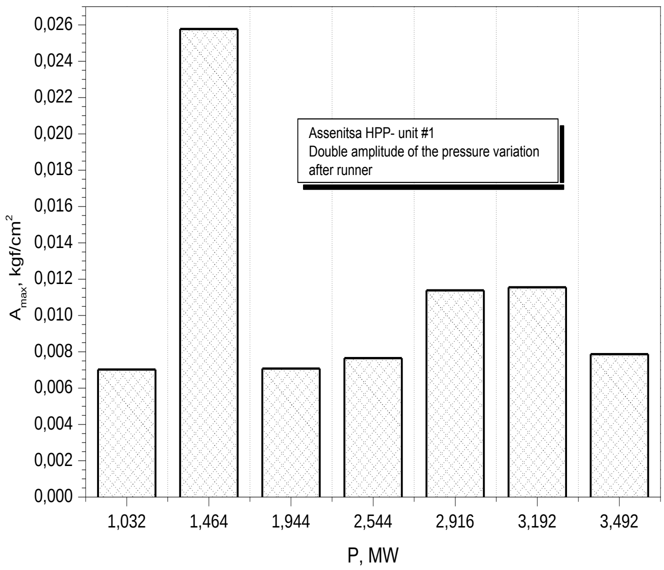

Fig. 8. Oscillation amplitude change of pressure after the runner

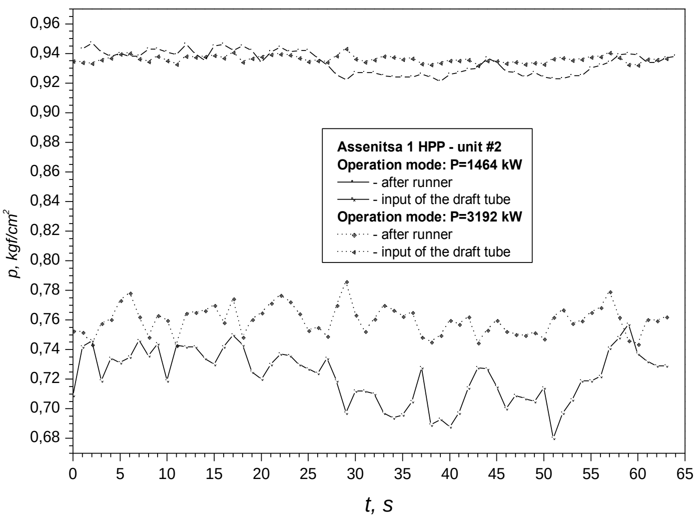

Fig. 9. Pressure variation

Figure 6 shows a comparison between the generator power from the opening of the guide vanes prior to and after the modernization. The increase in the power of the hydro-aggregate, after the completed modernization, is obvious (the generator is also modernized and increased its capacity). One may see that after a relative opening of the guide vanes by 30% prior to and after the modernization the power readings are nearly equal. In case of openings to more than 60% (basic exploitation modes), the power of the modernized turbine is visibly greater.

The working characteristics of the turbine (fig. 1) indicate that it operates with values of efficiency coefficient of over 90% when loaded to over 2300 kW. The efficiency of the operation of a given hydro generator may be defined also by analyzing the values of the so-called specific discharge q_s (volume of water, necessary for the production of 1 kWh of energy). The data from these calculations has been presented in Figure 7. The data indicates that within the interval of active generator power readings (1800 ÷ 3500 kW) the values of specific discharge keep their comparatively low levels (up to 4.7% greater than the minimum). Even under minimum load, the growth in the value of the specific discharge is by 11.4%.

3. The exploitation range of a given water turbine is defined not only by its energy related characteristics but also by its cavity characteristics. In order to evaluate the degree of pressure fall after the runner of the tested turbine, measurements of the pressure have been made within this zone. The measurement of the pressure after the runner is an indication of the processes that occur in the runner and in particular the effects of the vortex core on the cavity and vibration status of the turbine.

It is obvious that the amplitude of the pressure oscillation visibly decreases in the area, which is optimal from an energy viewpoint, of the operation of the turbine

and is greatest at loads around the 1500 kW area. Figure 8 shows the variation of the double amplitude of the pressure for the modes in which the exterior parameters of the turbine have been measured, while Figure 9 shows the record made from the measurement of absolute pressure at modes No.2 (load 1464 kW) and No. 6 (load 3192 kW) in two measurement cross-sections: after the runner and at the entrance of the draft tube.
It is obvious that under loads of 1500 kW the pressure oscillations have the highest amplitude. One may draw the conclusion that the optimal zone (with the lowest amplitudes) coincides with the optimal zone from an energy generation viewpoint.

Conclusion
The basic results, obtained as a result of the investigation performed, are expressed as follows:
1. On the basis of the results from the digital calculations, a new blade system has been developed for the runner of the Francis Turbine No.2 at Assenitsa 1 HPP. An improved version of the specialized software has been used; it has been applied on numerous occasions in the process of solving problems of a similar nature.
2. The results from the measurements indicate considerable improvement in the energy generating characteristics of the turbine after the performed modernization of the stream part. The values of the efficiency have grown within the entire exploitation range by an average of 5 ÷ 10%. This, together with the improved flow-through capacity of the blade system of the runner, have permitted an increase in the power of the hydro-generator from 2 600 kW to 3 492 kW.
3. On the basis of the results from the measurements, the values of the specific discharge during the various working modes have been calculated. Data analysis indicates that the optimal mode of operation from the point of view of minimal water consumption rates is under loads of around 2900 kW.
4. The measured values of the pressure after the output of the runner are considerably greater than the critical.
5. The assessment of the efficiency of the completed modernization indicates, that for a period of three years, profits from additional electricity sales will amount to around 300 000 EURO.

Literature
1. *Obretenov V*. Water turbines. Ekoprogress, Sofia, 2008.
2. *Obretenov V*. Modernization of Francis water turbine. Journal of the mechanical behaviour of materials, v.11, No5, pp. 365 - 372. Freund publishing house, London, England, 2000.
3. *Obretenov V*. Meridional projection of Francis turbines runners. Proceedings of VMEI “Lenin”, v.38, book 3, Technika, Sofia, 1987.
4. *Vuchkov I., Ch. Yonchev*. Planning and analysis of experiments in testing the properties of blends and alloys. Technika, Sofia, 1979.
5. *Obretenov V., G.Djambazov*. Computer-aided design of Francis turbine runners. Proceedings of the Hydroturbo'89 conference, v.2, pp. 191-198. Brno, 1989.
6. *Obretenov V., M.Al-Quadi*. Model tests of Francis turbine runner of medium speed. AGEnergy`99 conference, Athens, Greece, 1999.
7. *Obretenov V., G.Djambazov*. Modernization of Water Turbine from a small Water Power Station. Proceedings of the 5th International Energy Conference ENERGEX'93, Seoul, Korea.
8. *Field acceptance tests* to determine the hydraulic performance of hydraulic turbines, storage pumps and pump-turbines. IEC 60041 Ed. 3.0 b:1991.