MODERNIZATION OF FRANCIS WATER TURBINE

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ABSTRACT

The following paper presents the results from modernization of a Francis turbine at the low-power capacity hydropower station "Bjal Izvor," Bulgaria. The reviewed turbine is designed and manufactured by CKD Blansko Co., Czechoslovakia, in 1952. At the discussed hydropower station are installed two turbines of the above-mentioned type (unit No1 and No3).

A project for modernization of the runner and the guide vane has been executed. It was based on the results from the accomplished diagnostics, and the analysis of the characteristics and the construction of Hydro-aggregate No3 Turbine. A brief description of the employed methods and computer programs is provided along with a review of some technological aspects. The results from an experimental study of the characteristics of the modernized-turbine are presented as well.

1. INTRODUCTION

One of the major issues to arise in contemporary hydropower engineering is modernizing machinery equipment of hydropower stations (HPS) in view of increasing their efficiency. It is well known that it is possible to increase significantly the production of energy at operating HPS without unreasonable investment of capital. Resolving the above-mentioned problem is of enormous importance for countries with limited energy resources.

The paper hereby reviews opportunities for modernization of the Francis-turbine stream part of Hydro-aggregate No3 at "Bjal Izvor" HPS, Bulgaria.

2. ANALYSIS OF THE STREAM PART OF THE TURBINE

The analysis of the geometry of the stream part, and the exploitation characteristics of the turbine, afforded opportunity for drawing the following conclusions:

1. In the process of designing the turbine is used model F42, manufactured by CKD Blansko Co. [1]. With proper consideration of the particular values of total flow rate ($Q = 12m^3/s$) and head ($H = 18m$) of the power station, implementation of high-speed Francis turbines (the specific speed is $n_s = 275$) did not prove to be the best solution.

2. The data from the exploitation of the turbine show that, in the initial period, the maximum power of the hydro-aggregate was 560 kW, while immediately prior to commencing work on this project, the measured power was estimated to be 500 kW. The analysis demonstrated that the major reason for this problem is defects found in the stream part of the turbine. It is pertinent to note that, in 1957, a new one with Russian documentation replaced the runner of the turbine. In addition, as a result of continuous contact with heavily polluted waters, the stream part in the area of the runner and of the guide vane is worn out, which, in effect, leads to significant increase in the volume and the hydraulic losses.

3. Increase in the effective power of the turbine with significantly reduced consumption of water can be achieved through modification of the blade system of the runner. This, in effect, requires that the operating runner be replaced by a new one that will transform more efficiently the available pressure. The need for coordination in the operating process of the runner and the adjacent details of the stream part of all reaction turbines often demands modification of the geometry of the guide vane.

3. RUNNER

The following passage reviews the specifics in designing and manufacturing the new runner.

3.1. Selection of the principal parameters in the process of computation

On the basis of the accomplished analysis of the operational conditions and the performed measurements of the turbines of unit No1 and No3 at HPS "Bjal Izvor", the following values for calculation of the new runner were accepted: head H=17m, flow rate $Q=4.2\ m^3/s$, frequency of rotation $n=428.6\ min^{-1}$.

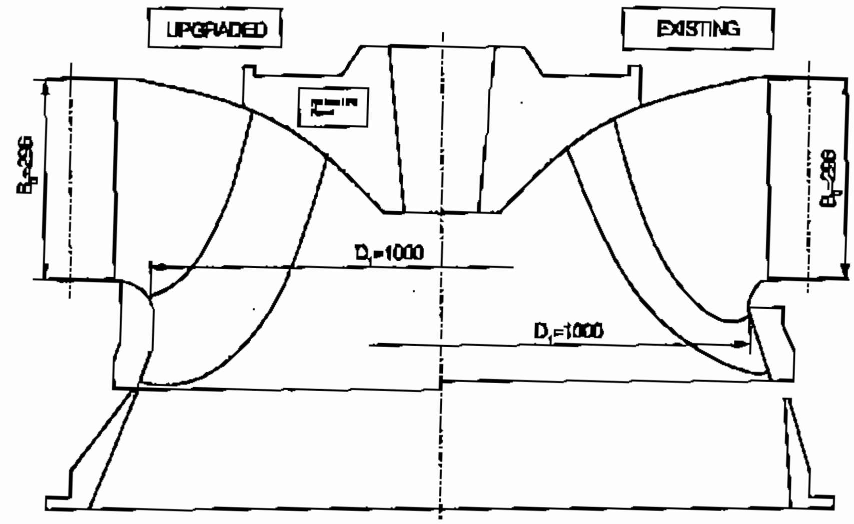

Fig.1: Comparison of the meridian projection of the runner area between the old and the modernized turbines.

3.2. Dimensioning of the meridian projection

Dimensioning of the meridian projection is accomplished in accordance with method [2]. The size of the shaft and the hub limit, to a large extent, the opportunities for correcting the inner meridian contour. Figure 1 demonstrates a comparison of the meridian projection of the runner area between the old and the modernized turbines.

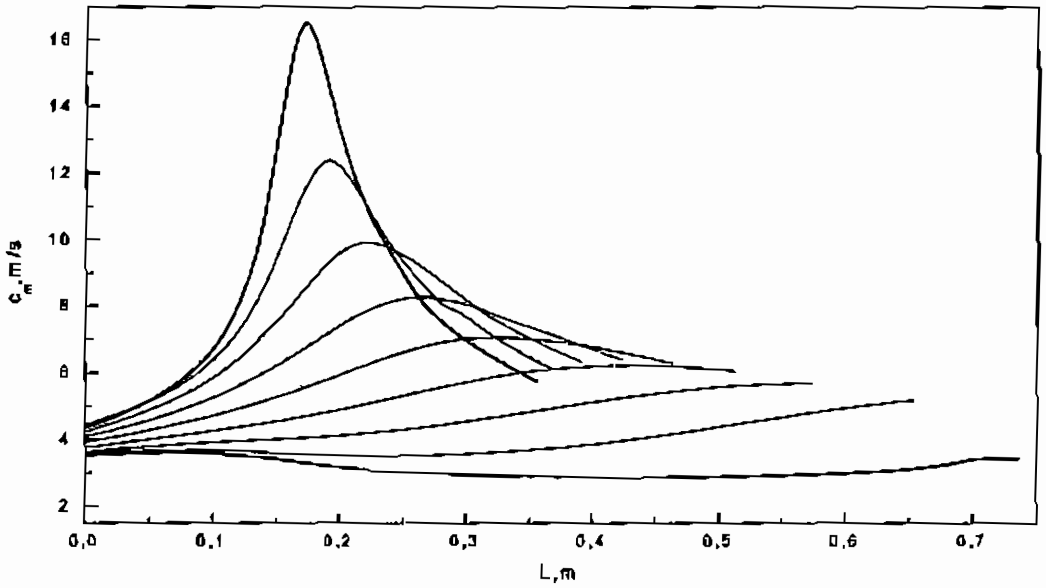

Fig. 2: Meridional velocity distribution

3.3. Calculation of the meridian flow

The calculation of the meridian flow is performed on the condition that the latter is axis-symmetrical and potential. The meridional velocity distribution along the current lines is shown on Figure 2.

3.4. Profiling the runner blades

A parametrical scheme for optimization based on the consecutive application of the so-called inverse (synthesis of the blade system), and direct (analysis of the flow through it) hydrodynamic problems, is applied [3]. The employed methods and

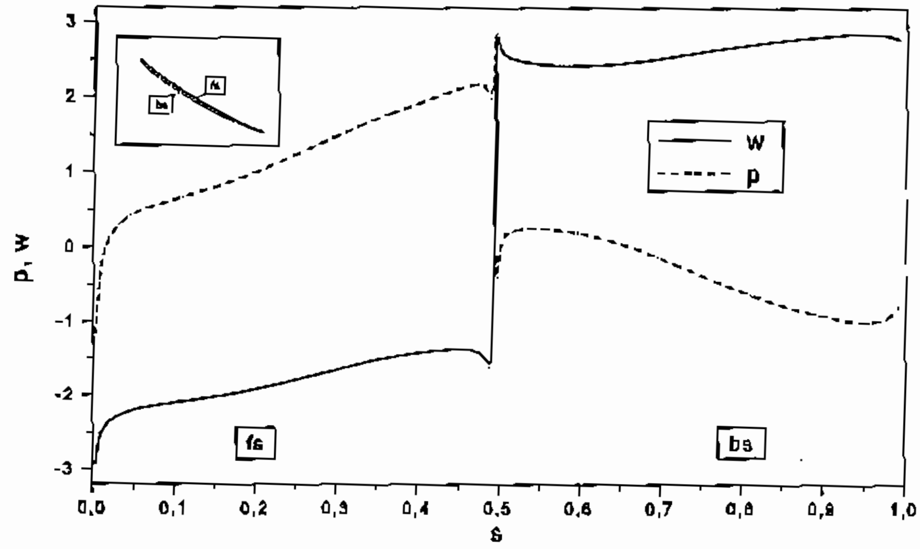

Fig.3: Distribution of the relative velocity and the pressure

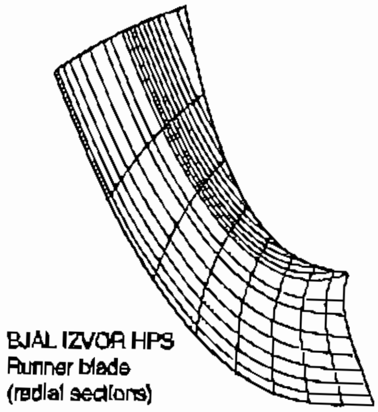

Fig.4: Meridian projection of the blade and the radial sections

computer programs have been applied on a number of occasions to dimensioning runners of Francis turbines with various specific speeds [3,4]. Figure 3 demonstrates the profile of the blade for the middle section, and the distribution of the relative velocity and the pressure to the front side and the backside of the profile. The meridian projection of the blade and the respective radial sections are shown on Figure 4. Table 1 presents some calculation results: the values of the meridian velocities c_m; the circulations $\Gamma = 2\pi c_u r$ of the blade inlet and outlet for each current line; the produced blade head H_t; the relative profile h_p and circulational h_c losses.

3.5. Strength Test

In the case of the new runner, the geometry of the hub remains practically unchanged, but the shape of the crown is modified. A test for bending of the blades is conducted according to method [5]. The maximum thickness of the blade is $\delta = 8mm$.

Table 1.

Stream line	1	2	3	4	5	6	7	8	9
H_t, m	16.4	16.6	15.3	16.6	16.7	17.4	17.5	16.3	15.7
h_p, %	3.02	2.45	2.16	1.83	1.55	1.29	1.09	0.85	0.81
h_c, %	0.44	0.42	0.35	0.30	0.29	0.25	0.20	0.15	0.14
Inlet edge									
Γ_1 m^2/s	25.13	25.13	25.13	25.13	25.13	25.13	25.13	24.82	24.50
C_m, m/s	10.97	8.67	8.64	7.57	6.63	5.62	4.53	3.46	2.70
Outlet edge									
Γ_2 m^2/s	2.58	20.39	4.21	2.32	2.14	1.32	1.07	2.83	3.14
C_m, m^2/s	14.20	15.30	17.10	18.00	19.30	20.10	19.80	19.10	18.20

In the section at the hub, the bending pressure is $\sigma = 241.7.10^6\ Pa$. The runner blades are produced from alloyed steel. It has the following chemical composition: C – 0.12%; Cr – 18%; Ni – 10%; Ti – 0.8%; Mn – 2.0%; Si – 0.8%. The basic mechanical characteristics of the steel are: strength of extension 529MPa; relative extension: 38%; impact elasticity: 490 kJ/m^2. This steel has high corrosive and cavitational resistance as well as good welding qualities. A strength test of the two discs of the runner has been conducted as well (hub and crown).

The runner is composite, i.e. the hub, the crown, and the runner blades are all manufactured separately. The runner blades are stamped of sheet iron; for the latter purpose special technological equipment is designed. A model of the blade surface is designed to this end by implementing the radial sections (fig. 4). On the one hand, this model is necessary for manufacturing a device for bending the blades; on the other, it is used to warrant the required accuracy in fastening the blades to the hub. The model and the device could be used on multiple occasions in case there is a need of producing a new runner. This technology has been applied many times in the implementation of other projects.

The modifications in the dimensions and the shape of the runner crown demanded refashioning of the back cover and the crank at the turbine run-out.

4. GUIDE VANE

A normalized guiding blade from the manufacturing company is used in the guide vane of the original turbine. It is known that the guiding blades directly influence the energy losses (hydraulic losses from friction and instances of vortex formation); on the other hand, they affect the hydraulic losses in the adjacent elements (spiral case and runner). The analysis demonstrated that there is a discrepancy between the flow angle, formed by the spiral case, and the inlet angle of the guiding blades during opening, corresponding to the nominal flow. This is one of the major reasons for the displacement of the optimal operating conditions of the turbine as compared to the planned values.

The available opportunities for implementation of several variants of profiles for the guiding blade have been considered. All of these have shorter length of the chord, which is beneficial with regard to the impact of unequal spacing of the flow at the guide-vane outlet on the performance of the runner. The final variant is a modification of the normalized asymmetrical profile of LMZ, Russia [6]. In view of maintaining the circulation of the guide vane, it is usually recommended for high- specific speed Francis turbines not to have the inlet edge of the runner shifted away from the guiding blades. In the particular case, this requirement is satisfied- the inlet edge of the new runner is removed outwards as compared with the previous one (Fig.1).

5. RECONSTRUCTION OF THE TURBINE COVER AND CRANK

The modifications in the dimensions and the shape of the runner crown demanded refashioning of the back cover and the crank at the turbine run-out (the principal diameter decreased with 42 mm; the crown height increased with 35 mm - fig.1). The design solution was directed toward optimal implementation of the details of the existing construction. The configuration of the jacketed rings was modified in accordance with the requirements of the new runner, with ensured radial clearance $\delta = 0.8 mm$. The construction of the rings was seriously obstructed by the numerous corrections of the cover at previous repairs.

The dimensions and shape of the transition body were modified for two reasons: it was heavily worn-out, and there was a necessity for ensuring normal conditions for the flow past the outlet of the runner. The latter requirement is essential, especially with regard to high-specific speed Francis turbines. Since it was not possible for us to correct the inlet part of the crank, the angle of the meridian contour in this area was increased with approximately 2°.

6. EXPERIMENTAL INVESTIGATIONS

After the modernized-turbine hydro-aggregate operated for 1200 hours, a set of measurements at various openings for the guide vane and head H=16.5m has been conducted. The results are shown on Figure 5. A comparison with analogous characteristic of Hydro-aggregate No1 is provided as well. It can be observed that at operating conditions with relative opening of the guide vane $a_0^* = a_0 / a_{0max}$?0.7, the power of the modernized turbine hydro-aggregate increases with 60÷80 kW. For the optimal operating conditions, the relative increase of power is 16%.

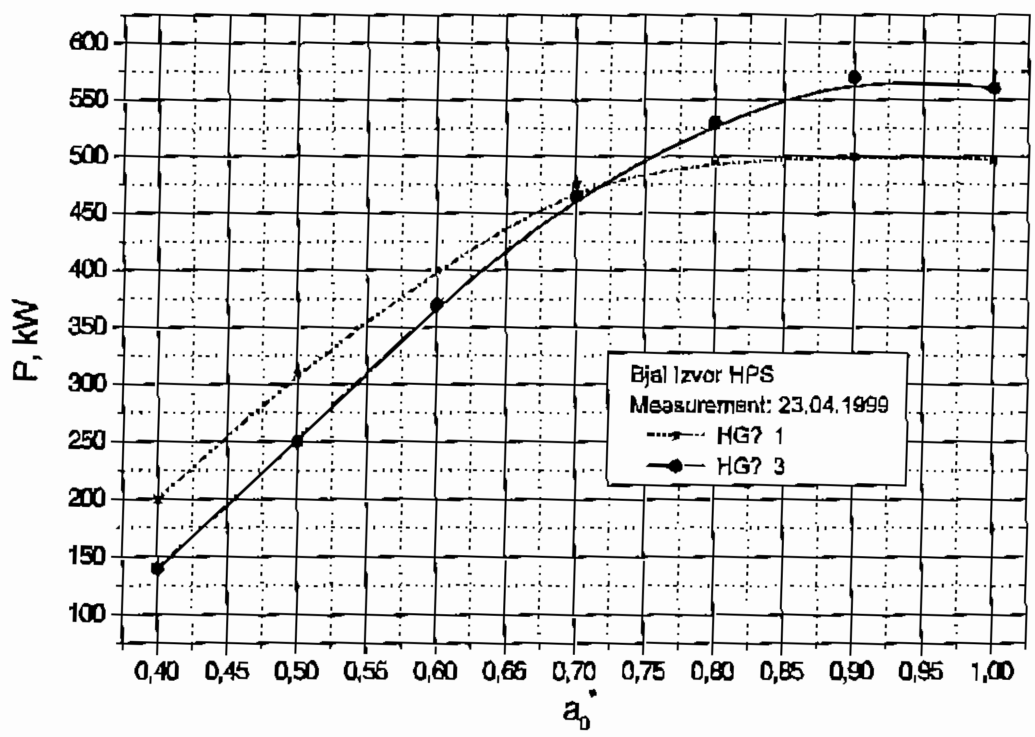

Fig.5: Working characteristics

CONCLUSION

1. A project for modernization of the elements of the stream part of Francis water turbine with high specific speed is designed and implemented on the basis of numerical research.

2. The results from a yearlong exploitation of the modernized hydro-aggregate show an increase in power, respectively in produced electrical energy, of approximately 16%. The calculations demonstrate that within the above-mentioned period, with proper consideration of the specific conditions in Bulgaria, some 60% of the invested capital has practically been returned.

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