CAM CHARACTERISTICS OF THE KAPLAN TURBINE DETERMINED BY EFFICIENCY AND BEARING VIBRATIONS

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ABSTRACT

River-off hydropower plant that has been in operation for about 30 years has to be revitalized. Installed Kaplan turbines, with a huge runner diameter of 9.5 m, had the highest unit power output of that type in the world. Construction of the next stage improved plant cavitation parameters and allowed further increase of the unit discharge and rated unit output respectively. The existing CAM relationship was determined based on hydraulic model tests. The prototype/model length ratio was so high ($\lambda_L = \frac{D_p}{D_m} \approx 20$) and all similarity conditions can not be fulfilled. On the other hand, the CAM combination is dependent on the head, velocity and rotational speed. Because of that, serious field tests were performed. Despite big troubles associated with: the flow measurements ranging from 100 to 840 m$^3$/s and in huge cross-section areas, the turbine power output measurements through generator output, the variation of the turbine net head due to electricity consumption limitations, etc., high measurement accuracy and repetition of measurement results were obtained.

Hydraulic and energy turbine characteristics were tested and efficiency determined at the head close to rated one on two units respectively. The CAM relation was also tested based on turbine bearing vibrations.

KEY WORDS

Kaplan turbine, measurements, discharge, efficiency, vibrations.

NOMENCLATURE

$\rho$ (kg/m$^3$) - water density,
$Q$ (m$^3$/s) discharge,
$H$ (m) net head,
$\eta$ - efficiency,
$g$ (m/s$^2$) - gravity acceleration,
$P$ (MW) – turbine power output

Subscripts and Superscripts

$R$ – right, $L$ - left

ABBREVIATIONS

HPP – hydroman power plant

1. INTRODUCTION

Field tests, as stated in the heading, were carried out on Djerdap I (Iron Gate I) hydropower plant over several past years, [2, 3]. When it was constructed, this hydropower plant had the biggest Kaplan-type turbines in the world, Fig. 1.

Twelve units were installed: 6 on Yugoslav bank of the Danube and 6 on Romanian bank. Basic data are given in Fig. 1, together with longitudinal cross-section of the plant and water levels.

During the operation, units proved to be capable of accomplishing higher power output than anticipated, especially after the construction of a downstream hydropower plant Djerdap II and reduced cavitation limitations. Therefore, after serious tests and control, the rated parameters were increased twice. First, power
output and discharge per unit were increased from 175 MW and 725 m³/s to 190 MW and 800 m³/s.

Today, after about 30-year operation, a thorough revitalization of the entire plant is necessary along with a reconstruction or total replacement of some parts. A set of measurements on unit A1 was carried out during 1997-1999 [2, 8, 10], and on unit A3 during 2003 [3]. The results related to the determination of optimal CAM relationship that defines interdependence of guide vanes opening and runner blades inclinations for various heads are presented in the paper. In addition, radial vibrations of turbine bearing in flow direction results obtained during field tests [2] are shown in this paper.

2. CONDITION FOR CAM RELATION TESTS

This time it was certainly possible to test CAM relation for heads corresponding to the present upper and lower water levels and it is close to the rated head. Prior to turbine manufacture, CAM relation was determined by hydraulic model tests carried out in the laboratory of turbine supplier LMZ 1968/69. Due to a huge turbine size (runner diameter is 9.5 m) and significant discharge capacity, resulting from high specific speed, model dimensions had to be reduced, so that a runner blade model diameter was only 460 mm, head being 3 m. So the length scale ratio was \( \lambda_L = 20.65 \). According to thus obtained data a three-dimensional crankshaft had positioned for adjusting CAM relation, depending on head and required power output.

3. METHOD OF MEASUREMENT

In order to obtain CAM diagrams a set of quantities had to be defined and thereafter used for computed the data on discharge, net head, turbine power output and efficiency. All these data had been indirectly defined by measurement of other physical values.

Due to specific conditions, all measurements could not be made according to IEC documents, [5, 7]. Of all data, the flow measurements seemed to be the most complicated. Net head and turbine power-output were determined with less difficulty. Data concerning these measured values will be further described.

**Flow measurements** - On such a huge plant that has no sections with parallel streamlines, nor has it cross-sections where speed orientations would be known, it was impossible to arrange hydrometric wings to measure local distribution of velocities. Also, it was impossible to apply any other, standard-prescribed, method for flow measurements, [7]. This problem was taken into account while plant was being designed, so the plan was to drill holes on spiral casing for pressure taps. Thus, flow measurements were performed by the Winter-Kennedy inertia method. Appropriate taps were also drilled on a model spiral casing and pressure difference dependency them of the flow were calibrated. The measuring cross-section was at 55° from inlet opening, see Fig. 2.

\[
Q = k_1 \cdot (\Delta h_{\text{in}})^2
\]  

(1)

where flow \( Q \) is in m³/s, pressure difference between taps 1 and 3, \( \Delta h_{\text{in}} \) is in mWC, \( k_1, k_2 \) are flow constants.

It should be pointed out that it is irrelevant whether coefficients \( k_1, k_2 \) are accurately determined for the most favorable combination of guide vanes opening and runner blades inclination, because the entire procedure is based on relative flow changes determination. However, successful application requires that the value of coefficients \( k_1, k_2 \) do not change over the entire flow range. But it turned out that this was not the case in small flows, lower than one-fourth rated, therefore...
the application of the method is uncertain in that range. Yet, the procedure applied is justifiable, for the plant does not operate when flows are lower than one-third of maximum. Thus, all significant operating regimes are accomplished at constant value of coefficients $k_1$, $k_2$, as determined by these measurements.

$$H = z_1 - z_2 - \frac{\Delta H_L}{1 + \sqrt{\frac{\Delta H_R}{\Delta H_L}}} - \frac{\Delta H_R}{1 + \sqrt{\frac{\Delta H_R}{\Delta H_L}}} + \frac{2 \cdot Q}{g \cdot A_1^2} \left[ \frac{1}{1 + (\Delta H_R / \Delta H_L)^{3/2}} \right] + \frac{2 \cdot Q}{g \cdot A_1^2} \left[ \frac{1}{1 + (\Delta H_R / \Delta H_L)^{3/2}} \right] - \frac{2 \cdot Q}{g \cdot A_1^2} \left[ \frac{1}{4 \left( \frac{A_1}{A_2} \right)^2} \right] \quad (2)$$

In the above expressions $z_1$ and $z_2$ are levels of upper and lower water reservoir, $A_1$ and $A_2$ are inlet and outlet cross-section areas in m$^2$.

**Turbine power output** – Turbine power output was also computed by addition of power losses in both main and excitation generators, $\Sigma P_{Gl}$ and mechanical losses $\Sigma P_{ml}$ to a measured generator output $P_G$. The formulas for calculating losses were obtained by generator tests done previously. Losses in generator are divided into losses in copper, iron and excitation. Mechanical losses are in a thrust generator bearing, radial turbine bearing, radial generator bearing and ventilation losses in the generator. Thus, turbine power output is:

$$P = P_G + \sum P_{Gl} + \sum P_{ml} \quad (3)$$

**Turbine efficiency** – Turbine efficiency is determined upon the expression for power output:

$$P_I = \rho \cdot Q \cdot g \cdot H \cdot \eta \quad (4)$$

**PROPELLER CHARACTERISTICS OF UNIT 1 AND UNIT 3**

Unit 1 CAM relation was tested by energy and vibration characteristics [1,2], and on unit 3 only energy characteristics were determined [3].

To determine the curves for turbine efficiency ($\eta$) dependence of discharge ($Q$) and a corresponding combination of guide vanes opening ($a$) and runner blades angle ($\phi$) at the existing head, first a number of operating regimes with a retained CAM relation were registered [1].

The CAM relation was broken-off at the second stage of tests, and a series of propeller operating regimes with constant runner blade inclinations ($\phi = -10^\circ, -5^\circ, 0^\circ, 10^\circ, 15^\circ, 17^\circ$) was tested. A number of operating regimes were registered (usually seven to ten) for each inclination $\phi$. On the basis of data analyzed, efficiency curves were drawn, depending on discharge for each $\phi$, i.e. curves $Q$-$\eta$ for $\phi=$const., see Fig. 4 and Fig. 5. By drawing envelope curves around efficiency curves, optimal efficiency values were determined, while vertical lines, drawn through envelope curve contact points with propeller curves to the section with guide vanes inclination change curves for different runner blade angles $\phi$, determine optimal CAM characteristic. A curve was drawn through points thus obtained, defining optimal CAM combination, see lower part in Fig. 4 and Fig. 5 [5, 7].

In addition, the CAM relation on unit 1 was also tested based on turbine bearing vibrations in two radial directions: dam axis direction ($x$ - axis) and in flow direction ($y$ - axis). In this paper only radial vibrations
of turbine bearing in flow direction are presented. It is evident from Fig. 6 that vibration minima on propeller regimes do not correspond to combination regimes. Exception occurs at runner blade angle $\phi = 5^\circ$ (optimal angle).

Expected vibration minima’s match with corresponding optimal combination of guide vanes opening and runner blades angle is not completely fulfilled. Deflection for certain regimes can be seen in Fig. 6 ($\phi = -5^\circ, 0^\circ, 15^\circ$). This appearance can be explained by taking into account vibrations source. Beside excitation that originated in runner itself, vibrations can be result of other excitations in system (mechanical, electrical, etc.). Measured radial vibrations of turbine bearing (in flow direction) are the result of superposition of few different excitations in system.

4. COMPARISON OF RESULTS

The optimal CAM relationship ($a, \phi$) was determined by indirect measurements of the guide vanes opening ($a$) and runner blades angle ($\phi$) at tested operating regimes, as presented graphically in Fig. 4.

Amplitude of main harmonics related to discharge for certain propeller regimes are shown in the lower part of Fig. 6. It can be seen that harmonics minimas for different hydraulic excitations mainly correspond to the appropriate regime optimas. Frequencies of the first, the second and the third harmonic, for $z=6$ runner blades and rotational speed $\omega = 1.2 \text{ s}^{-1}$ are: $f_1 = 7 \text{ Hz}, f_2 = 14 \text{ Hz}, f_3 = 21 \text{ Hz}$. 
The tests of existing CAM relation, i.e. for establishing new relations, had been carried out by on-site measurements. Test procedures were described, and results are presented in Fig. 8 [1,2].

To compare both results, reduction to identical heads should be done. That was the reason for comparing net heads at measurements with preserved CAM relation to heads for appropriate flows in tests with broken-off CAM relation. So, satisfactory agreement was found, though both measurements had been made at heads declining with flow increase. Calculations of data to constant net head were neither possible nor justifiable, because the unit had been operating at constant frequency, and the data obtained provide a reliable basis for calculations, concerning relatively small losses.

To make comparison easier, Fig. 7 shows both efficiency curves, obtained in the existing CAM relation or looked for in broken-off relationship. In Fig. 8 CAM characteristics was drawn, and points corresponding optimal CAM relation were drawn in too. The agreement was so high that two lines could not be drawn.

Field tests on unit 1 were conducted in 1997 [1,2] (net head \( H_n = 27.4\text{m} \)) and on unit 3 in 2003 [3] (net head \( H_n = 27.8\text{m} \)). It can be noticed that unit 3 efficiency is cca 1% higher than efficiency of unit 1. It should be emphasized that unit 1 is placed on right river bank where the water inflow is inconvenient. Also discharge measurement accuracy (index method) is relatively low to make general conclusions.

Tests of both units showed insufficient reliability of Winter-Kennedy inertia method for discharge measurement in wide range of its values, especially for low discharges, that is for low runner blade inclination.

5. CONCLUSION

A remarkable agreement is evident from efficiency curves, Fig. 4., except in the range of discharge below 350 m³/s. Minimum difference in efficiencies for flows beyond 350 m³/s to the highest ones is primarily conditioned by measurement errors. And yet, it should be born in mind that such agreement justifies the application of a relative method, therefore the obtained efficiency values should be taken like that.

However, the ideal agreement of data on optimal CAM characteristic confirmed by model and on site measurements, the ultimate conclusion can not be drawn. We could conclude that it is necessary to change or correct the existing combination based on measurements at a number of different head values, including the highest and lowest ones. This certainly requires a longer period of time, may be longer than a year.

The said conclusion suggests another one, and it concerning mutual positions of turbines. It is beyond dispute that water inflow and, to an extent, flow conditions are slightly different for each of them. Also, the effect of power output distribution should not be neglected, when inflows to adjacent units are different. To explore these phenomena more extensively, measurements should be made on all units. This is confirmed by testing two of six units. It should be mentioned that the net heads during these measurements varied for about 2%, what is in accordance with the general conclusions.

It was not only that good CAM characteristic was confirmed, but also characteristics were determined at power outputs and flows far beyond boundary quantities as defined by a primary contract. To draw the final conclusion, it is necessary to consider other aspects of unit safety, reliability of hydromechanical equipment, control characteristics and technical resources of the whole plant.
ACKNOWLEDGEMENTS

The research project in which the investigations has been carried out is supported by Electricity Board of Serbia – Dept of Research & Investment, Belgrade and Electricity Board of Serbia – HPP Iron Gate, Belgrade.

REFERENCES


