Pump-turbine characteristics for analysis of unsteady flows

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Abstract. Hydraulic characteristics (head, moment, discharge, speed of rotation) of nine radial pump-turbines with different specific speeds are firstly compared among themselves and then analyzed as applied for calculation of unsteady flows and rapid transients. As known, in liquid filled systems a pump-turbine hydraulic characteristic is one of the important boundary conditions. The new sets of four-quadrant data (curves covering all four major zones of operation) are presented in a non-dimensional form of unit discharge, unit torque and unit speed, and then in Suter-curves form. The analysis is focused on differences in nine data sets for a range of specific speeds of pump-turbines. Furthermore, numerical results for a full unsteady flow system were obtained using the method of characteristics with boundary condition of various pump-turbine characteristics. Computed values are compared to the available measured data. Data presented in this paper are of great importance for all engineers who studied the calculations of transient processes in plants with pump-turbines.

1. Introduction

For all calculations of transients of the systems with turbomachines, the four quadrant curves of the machine are essential. Nearly as a rule, the curves stated in the books of Stepanoff [1], and Donsky [2] are used, where the curves are given only for three specific speeds nq=25, nq=147, nq=261 (one radial, one semi-axial and one axial turbomachine) [1-5, 7, 10-11]. Designers of the systems use the curve closest to the analyzed machine, even without interpolation. Of course that many approximations are used in fluid transient calculations, but nearly all of them have been studied in the meantime (non-stationary friction, speed of sound, fluid-structure interaction, etc.), but as far as we are aware of, no thorough analysis of specific speed variation (and its influence) has been published.

This paper provides **nine new sets** of data of hydraulic characteristics of radial pump-turbines with different specific speeds, and then transformed into Suter-curves form.

In many design situations the curves of complete pump characteristics are not available from the manufacturer, so in computer programs for simulations one must complete the W_h and W_m arrays from other available test data [5, 7, 10-11]. These curves tend to have similar shapes for the same specific speeds and then the curves may be extended by comparison with data for other specific speeds. This is an uncertain procedure, and the results of transient studies using such a data must be taken with precaution.

1.1. The four-quadrant characteristics

Are required for transient simulations in both pump and turbine regions of operation, as well as in transitory regimes. Reverse flow and reverse rotation are also possible for storage pumps. However, for pump-turbines, steady-state operations in pump and turbine modes are definitely required.

Complete operating characteristics of pumps and pump-turbines with various specific speeds are not readily available. In general, the manufacturer supplies the head, brake horsepower, and the efficiency plotted against the discharge for the normal speed of operation. From these data, the characteristics of normal pump operation may be determined. However, it is necessary to have the complete characteristics of a pump and pump-turbines in order to determine the operation for all possible steady state conditions, or to determine transient conditions for normal or abnormal operations. The complete characteristics of a pump and pump-turbines consist of the zones of pump operation, turbine operation, and energy dissipation. These zones can be plotted as families of speed-ratio curves and torque-ratio curves on a (flow ratio-head ratio) coordinate system. In this form it is convenient to determine the transient water-hammer effects by graphical procedures. In addition, it is possible to determine by inspection the steady-state conditions existing at different heads or speeds under normal and abnormal conditions of operation. Moreover, examples of the transient effects which can be determined by the use of these characteristics include the water hammer in the suction and discharge lines for normal or abnormal starting or stopping of the pumps turbines, pump turbine speed, flow through the pump turbine. It is necessary to have the complete characteristic of a pump and pump turbines of approximately the same specific speed as that being studied. In most cases this is not available, and the question has often been raised as to the error which is introduced in the transient studies by the use of improper pump and pump turbine characteristics. This paper presents nine complete radial pump turbines characteristics covering a wide range of specific speeds.

Typically, model studies have to be conducted for pump-turbines over a wide range of machine operating regimes defined by various wicket gate (guide vane) openings. The model data are generally represented in terms of non-dimensional parameters in four quadrants, with the axes being the specific speed and specific flow. The same data are sometimes correlated in terms of head coefficient and flow coefficient.

Figure 1 depicts eight zones of possible operation of a pump or pump-turbine in four quadrants. As listed in Table 1: Quadrant I - contains normal pumping via zone A, while normal turbining is zone G in Quadrant III. In a model test stand the machine can be made to operate in all four quadrants, even with those zones with negative head. For a prototype unit operating as a pump, loss of power will mean that the pump starts in zone A (Quadrant I), enters zone H (Quadrant IV) as flow reverses, then zone G (Quadrant III) as the machine reverses rotation, during which time the wicket gates are normally closing. If the transient persists until the pump passes zero torque (runaway) before the gates are closed, then the unit enters so called braking regime, designated as zone F. If the servomotor was to fail, or if the pump-turbine was initially generating (zone G in Quadrant III), there would be a possibility for the machine to enter the zone E (Quadrant II), called reverse pumping [6].

As shown in Table 1 zones B, C, and D correspond to negative head, a condition only possible to be contrived in the laboratory model. Consequently, model data are typically only reported for positive heads, represented by zones A, H, G, F, and E. In order to compute speed change during the transient regime, not only head-flow data are needed, but also torque.

Table1. Definition of zones and quadrants.

Quadrant	Zone	Mode	Energy	Rotation	Flow	Head	Torque
I	A	Normal Pumping	Productive	N > 0	Q > 0	H > 0	T > 0
I	B	Energy Dissipation	Dissipative	N > 0	Q > 0	H < 0	T > 0
I	C	Reverse Turbining	Productive	N > 0	Q > 0	H < 0	T < 0
II	D	Energy Dissipation	Dissipative	N < 0	Q > 0	H > 0	T < 0
II	E	Reverse Pumping	Productive	N < 0	Q > 0	H > 0	T < 0
III	F	Braking	Dissipative	N < 0	Q < 0	H > 0	T < 0
III	G	Normal Turbining	Productive	N < 0	Q < 0	H > 0	T > 0
IV	H	Energy Dissipation	Dissipative	N > 0	Q < 0	H > 0	T > 0

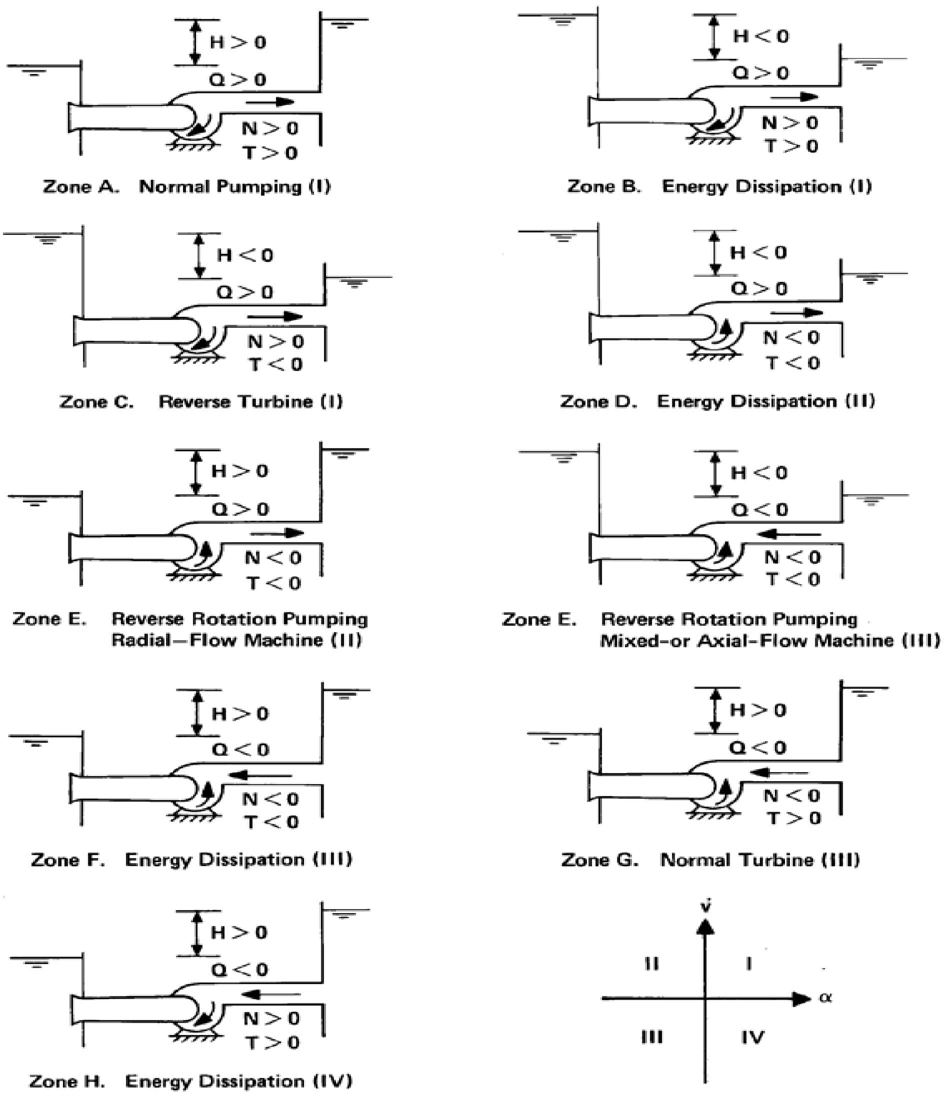

Figure 1. Definition of zones of variables for four-quadrants.

1.2. The analysis of hydraulic transients

Needs four-quadrant curves for numerical simulations in pumped storage installations, since the real potential for the occurrence of water hammer transients exists mainly due to disconnections from the electrical grid. For preliminary analyses, the three worst scenarios are: (a) full load rejection of all operating units, (b) pump power failure, and (c) load acceptance. Additionally for final analyses, more transient cases may be warranted: full load rejection with guide vane closure, full load rejection with main inlet valve closure and locked guide vanes, full load acceptance from speed-no-load, pump power failure with guide vane closure, pump power failure with main inlet valve closure, pump start-up from synchronization, normal turbine shutdown, normal pump shutdown, reduction of generating load, increase in generating load, load rejection followed by load acceptance, load acceptance followed by load rejection, load acceptance followed by normal shutdown, malfunction of governor cushioning stroke, resulting in higher closure rate over entire stroke, guide vane closure from speed-no-load, governor stability, resonance phenomena [6].

Considering the specifics of a reversible pump-turbine, especially with regard to their behavior in the transitional working area, the behavior of the pump-turbine during emergency unloading of two generators in Pumped-Storage Plant "Bajina Basta" is analyzed in this paper. Non-dimensional four quadrant characteristic diagram of the reversible pump-turbine presented in this paper can be used for hydraulic calculations in the transitional work area for types of pump-turbines in the domain of specific speeds *nq*=24.8; *nq*=27; *nq*=28.6; *nq*=38; *nq*=41.6; *nq*=50; *nq*=54.7; *nq*=56; *nq*=85.05. Also the accompanying dynamic phenomena that occur in these transient operating regimes (such as the appearance of guide vanes apparatus vibrations, runaway of the runner, cavitation occurrence, water hammer, etc.) are described in detail. The diagram shows the results obtained by calculation of water hammer for one concrete example and provides an analysis of the flow curves. Described transient operating regime of pumping hydraulic turbine can be traced in its four quadrant characteristics diagram.

2. The procedure for calculation of Suter-curves

The four-quadrant reversible pump-turbine characteristics curves are defined according to the parameters [3]:

$$n_q = \frac{N\sqrt{Q}}{H^{3/4}} \text{ - specific speed} \tag{1}$$

$$N_{11} = \frac{ND_1}{\sqrt{H}} \text{ - unit speed} \tag{2}$$

$$Q_{11} = \frac{Q}{D_1^2\sqrt{H}} \text{ - unit flow} \tag{3}$$

$$T_{11} = \frac{T}{D_1^3\sqrt{H}} \text{ - unit torque} \tag{4}$$

$$H = \frac{N^2 D_1^3}{N_{11}} \text{ - head} \tag{5}$$

$$Q = \frac{N\ D_1^3 Q_{11}}{N_{11}} \text{ - flow} \tag{6}$$

$$T = \frac{N^2 D_1^5 T_{11}}{N_{11}^2} \text{ - torque} \tag{7}$$

Figure 2 shows diagrams with relative unit sizes of flow, torque and number of revolutions for different openings of the wicket gate apparatus for pump-turbine of type *nq*=27. Four-quadrants are shown for all operating modes that can occur with the pump-turbine, as follows: Quadrant I - pump drive, electric machine operates as a motor; Quadrant II - the runner rotates in the direction of the pump, and the water flows into the turbine, electric machine operates as a generator, this operation represents a braking of the aggregate; Quadrant III - turbine drive, the electric machine operates as a generator; Quadrant IV - the runner rotates in the direction of the turbine and the water flows into the pump, the electrical machine operates as a motor [8].

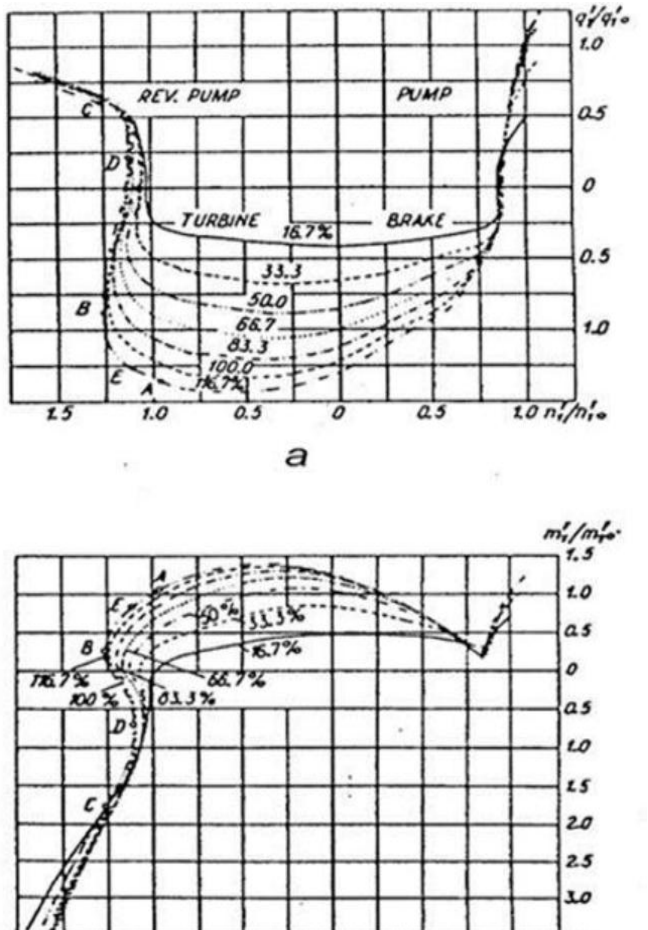

Figure 2. Pumped-Storage Plant Bajina Basta pump-turbine –
a) Flow characteristic curve, b) Torque characteristic curve.

Suter [3] produced a dimensionless representation of four-quadrant curves which is very well suited to computer use. If dimensionless variables W_h and W_m are used where W_h is given by

$$W_h = sign(H)\sqrt{\frac{H/H^*}{(N/N^*)^2 + (Q/Q^*)^2}} \tag{8}$$

and W_m by

$$W_m = sign(T)\sqrt{\frac{T/T^*}{(N/N^*)^2 + (Q/Q^*)^2}} \tag{9}$$

where affix * denotes steady state pumping conditions, H is the head across the pump turbine, Q is the flow, N the rotational speed in rev min^{-1}, T is the torque on the runner shaft, then graphs of *Wh* and *Wm* can be constructed against θ, where θ is given by [7]

$$\theta = \arctan\left(\frac{N}{N^*} \cdot \frac{Q^*}{Q}\right) \tag{10}$$

Two appropriate resulting graphs, like ones shown in Figures 3 and 4, provide a complete representation of the very complex complete set of characteristic curves of a pump-turbine over its four quadrants of operation. From these figures, it is relatively simple to read data and to put them into computer, by interpolation a particular value of *Wm* or *Wh* corresponding to a particular θ. From the equations (8) and (9) these values of *Wm* and *Wh* can be converted into T and H values. The torque value T can be used to calculate the running speed of the pump turbine at a Δt time period later and the value H, i.e. the head across the pump-turbine, can be used in the boundary condition calculation. Very few manufacturers of pump and pump-turbines are prepared to supply the four quadrant characteristics of their pumps. It is most unlikely that anything other than the normal $H \sim Q$, $P \sim Q$ curves of the pump-turbines in normal pumping mode or normal turbining mode will be available, so it is necessary to devise a reasonable approximation to the Suter curve for the other seven modes of pump turbine operation [7].

3. Head and torque characteristics of radial pump-turbines with different specific speeds – nq=24.8, nq=27, nq=28.6, nq=38, nq=41.6, nq=50, nq=56, nq=54.7 and nq=85.05

For the given nine radial pump-turbines with four quadrants of operation known, calculation of Suter curves *Wh* and *Wm* for various openings blades of wicket gate apparatus was done using the standard procedure for calculating of Suter curves.

Two graphs at Figure 3 and Figure 4 provide a complete representation of complete set of head and torque characteristics curves of the analyzed radial pump-turbines (*nq*=24.8; *nq*=27; *nq*=28.6; *nq*=38; *nq*=41.6; *nq*=50; *nq*=54.7; *nq*=56; *nq*=85.05), over four quadrants of their operation, for openings of wicket gate blades apparatus closest to the optimum operating point of the pump-turbine.

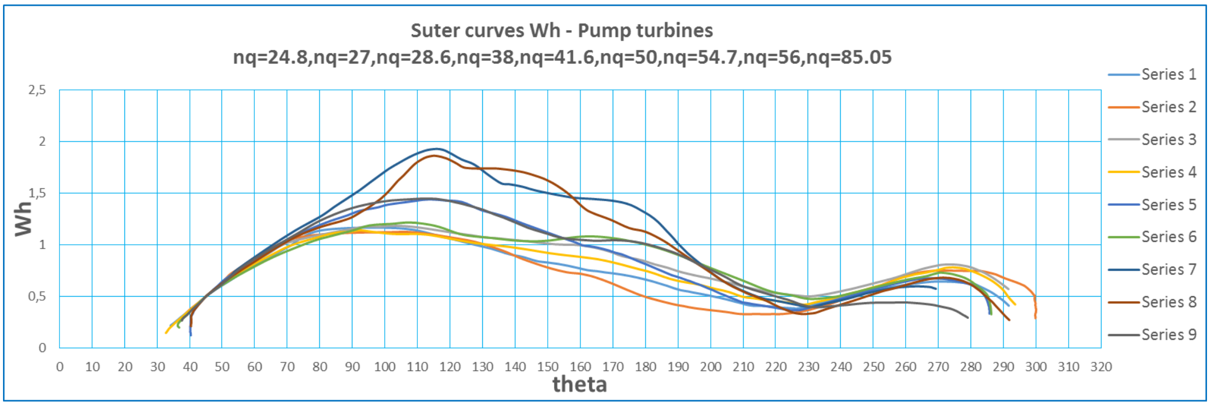

Figure 3. Suter curves *Wh* – Radial pump turbines (*nq*=24.8; *nq*=27; *nq*=28.6; *nq*=38; *nq*=41.6; *nq*=50; *nq*=54.7; *nq*=56; *nq*=85.05).

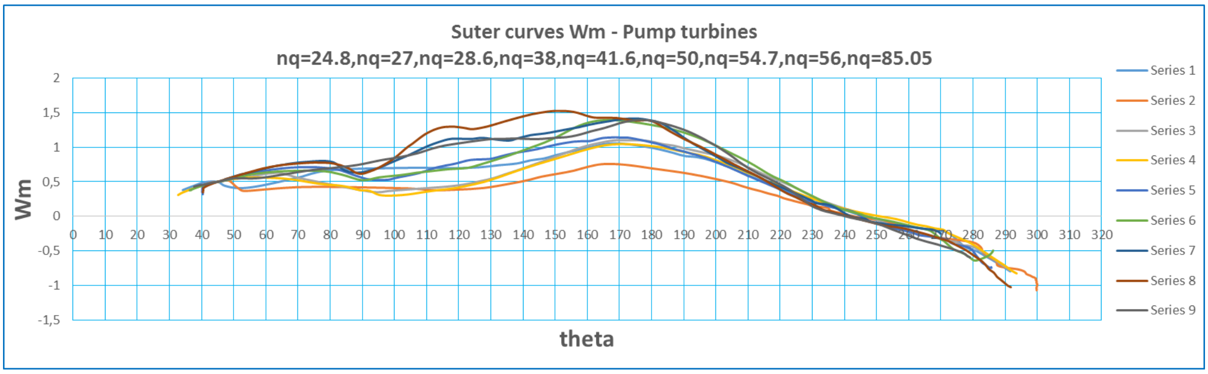

Figure 4. Suter curves *Wm* – Radial pump-turbines (*nq*=24.8; *nq*=27; *nq*=28.6; *nq*=38; *nq*=41.6; *nq*=50; *nq*=54.7; *nq*=56; *nq*=85.05).

Series 1: *nq*=24.8 - PSP "Bad Creek"; Series 2: *nq*=27 - PSP "Bajina Basta"; Series 3: *nq*=28.6 - PSP "Yards Creek"; Series 4: *nq*=38 - Pump turbine China; Series 5: *nq*=41.6 - Pump turbine Vienna Austria; Series 6: *nq*=50 - Pump turbine China; Series 7: *nq*=54.7 - Pump turbine RONT Russia; Series 8: *nq*=56 - Pump turbine China; Series 9: *nq*=85.05 - Pump turbine RONT Russia.

4. Pumped-Storage Plant "Bajina Basta" - computed values for unsteady flow compared to the available measured data

The pumped-storage plant "Bajina Basta" in Serbia (former Yugoslavia) has two pump-turbines configured to form a hydraulic loop. This arrangement was motivated by cost savings, but in the long run was mistake, since ideally each pump-turbine should have its own hydraulic waterway in spite of the higher initial price, so now the plant is endangered by hydraulic transients. Transient analyses were exercised at all design levels of Bajina Basta, during feasibility studies, general design and detailed design. High-head reversible pump-turbines (low specific speed) have an unusual characteristic in turbine operating modes: unlike typical behaviour, beyond the runaway zone any decrease in speed reduces the discharge [8].

Field tests were carried out in October 1982 [12].

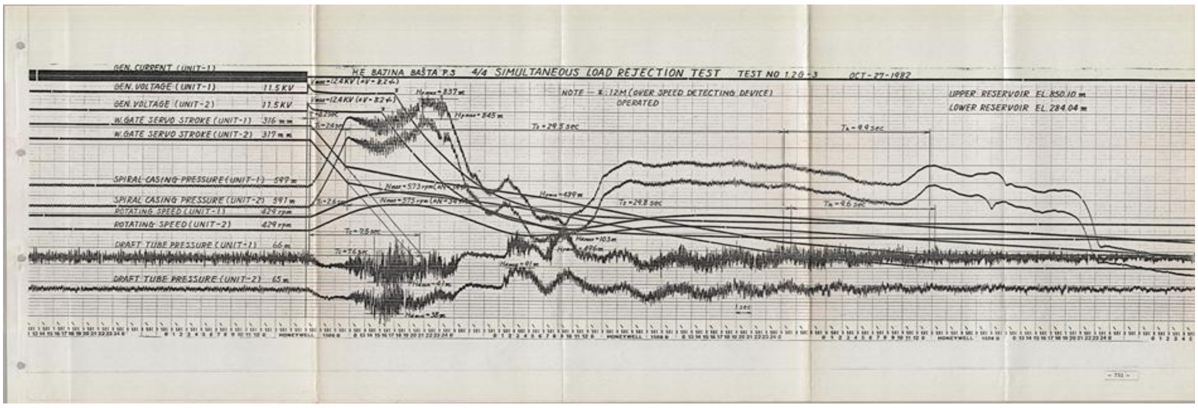

Figure 5. Pumped storage plant "Bajina Basta". Simultaneous load rejection from full capacity Unit 1 (281 MW) and Unit 2 (284 MW), copy of originally measured strip chart in 1982.

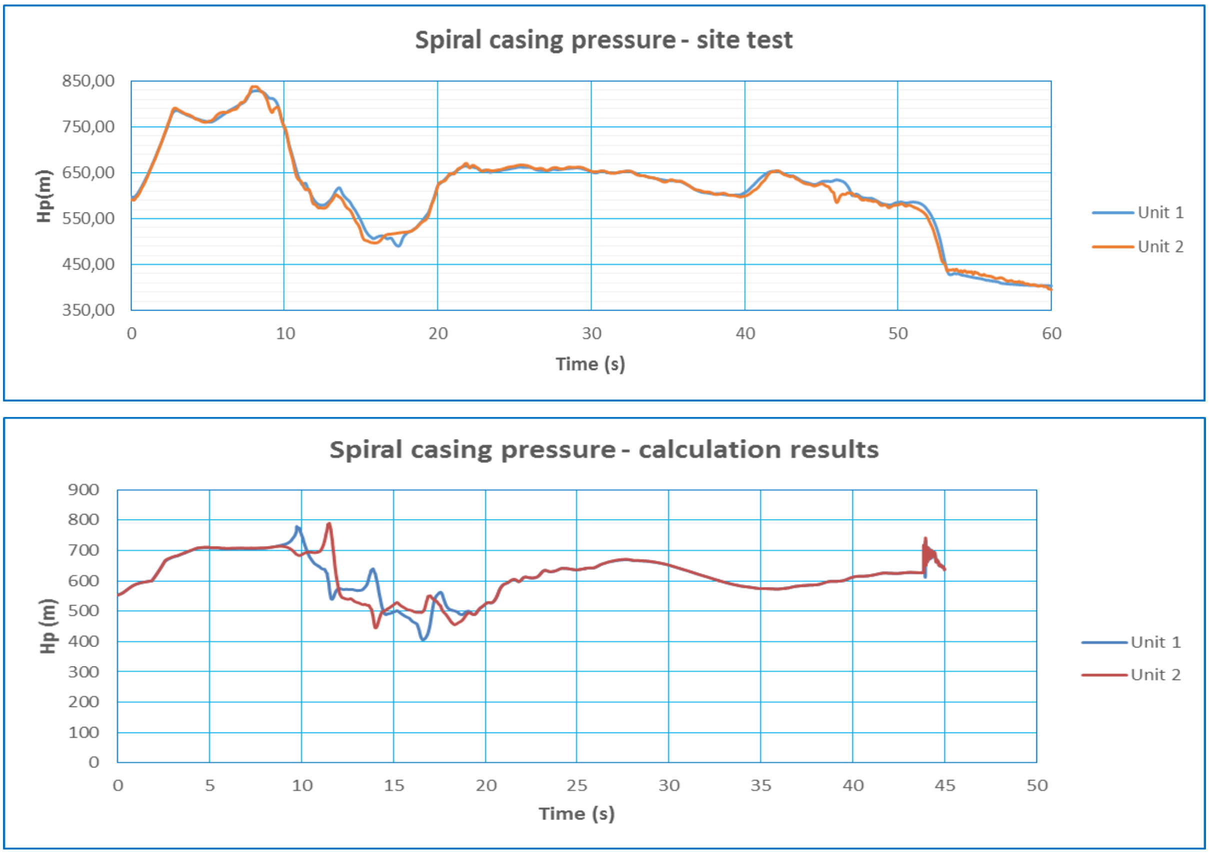

Figure 6. Pumped storage plant "Bajina Basta". Unit 1 (281 MW) and Unit 2 (284 MW) load rejection, spiral casing pressure – site test and calculation results.

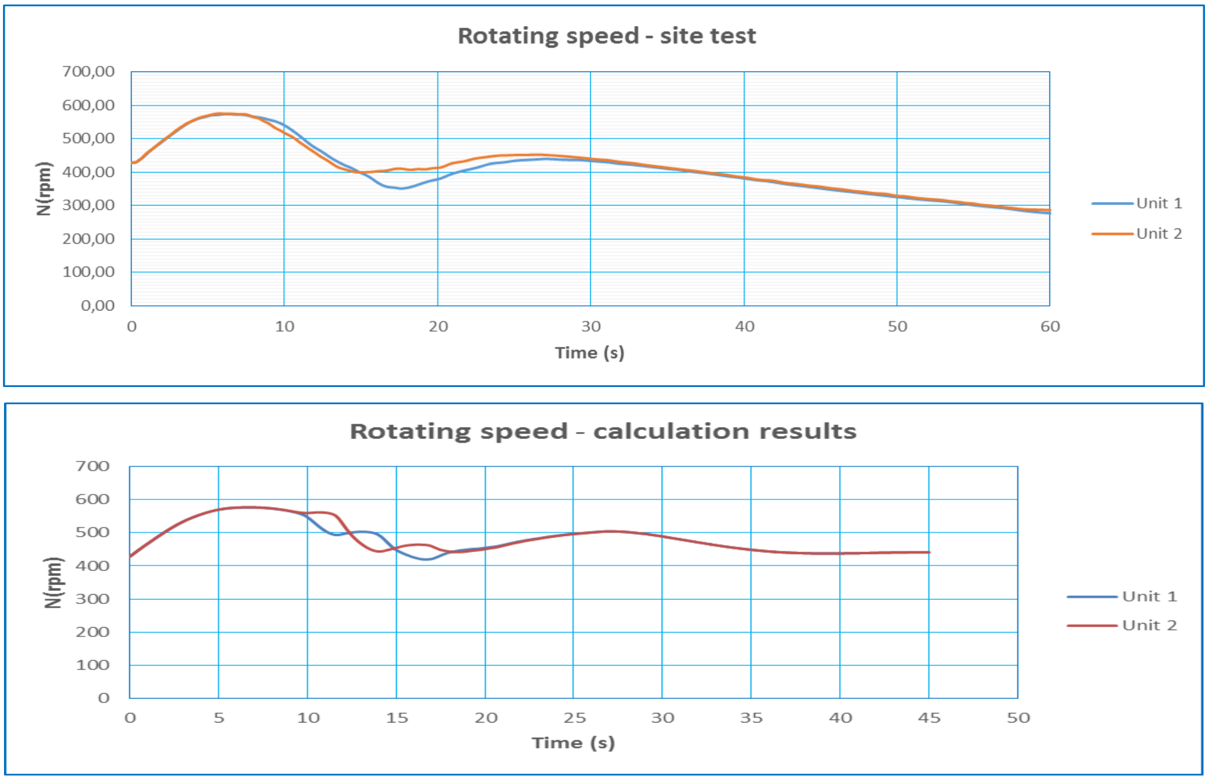

Figure 7. Pumped storage plant "Bajina Basta". Unit 1 (281 MW) and Unit 2 (284 MW) load rejection, rotating speed – site test and calculation results.

Figure 8. Pumped storage plant "Bajina Basta". Unit 1 (281 MW) and Unit 2 (284 MW) load rejection, wicket gate servo stroke – site test and calculation results.

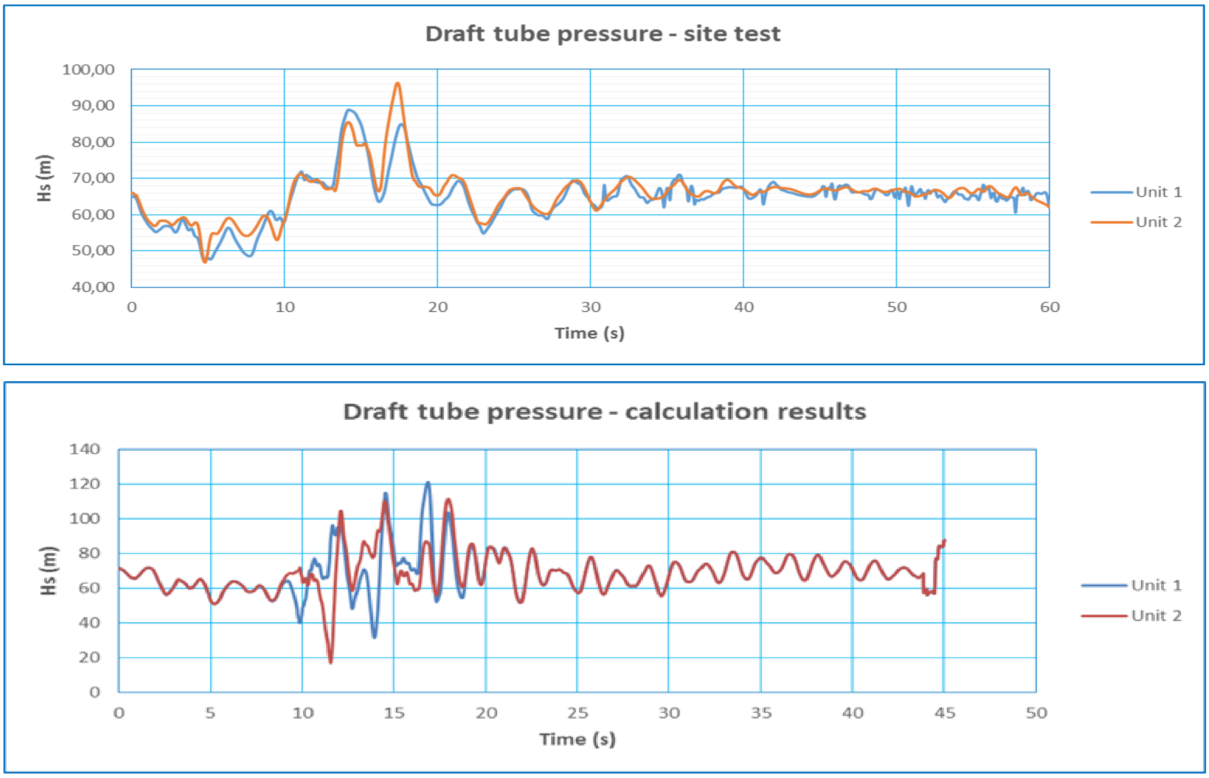

Figure 9. Pumped storage plant "Bajina Basta". Unit 1 (281 MW) and Unit 2 (284 MW) load rejection, draft tube pressure – site test and calculation results.

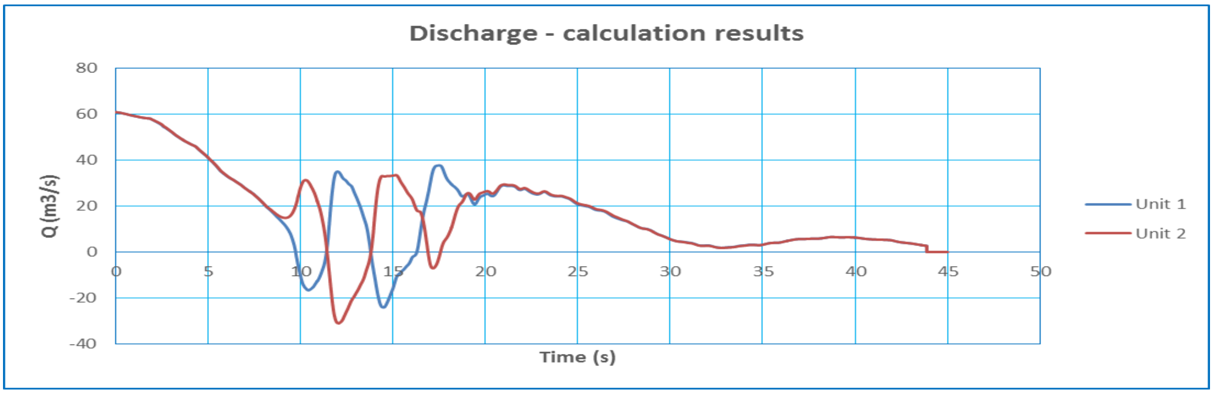

Figure 10. Pumped storage plant "Bajina Basta". Unit 1 (281 MW) and Unit 2 (284 MW) load rejection, discharge calculation results.

When both units 1 & 2 operated as turbines, the testing procedure was to make Unit 1 load rejection (281 MW) and its wicket gates closed down rapidly, and Unit 2 to drop-out (284 MW) and its wicket gates closed down rapidly. During load rejection of units 1 & 2, the discharge could not be measured. Figure 5-10 display the changes of main parameters: spiral casing pressure *Hp*, rotating speed *N*, wicket gate servo stroke *WGSS*, draft tube pressure *Hs*, discharge *Q*, with comparisons of site tests and calculation results. The wicket-gates remained open for some 0.3-0.4 s and then started to close down as prescribed, first rapidly and later slowly.

The agreement between measured and computed values is quite good, although no calibration was carried out. However, some details are clearly visible: Figure 6 show big difference in measured and calculated maximum pressure, measured much greater; wave velocity used for computation is slightly greater than the real one, comparing peaks in Figure 6 and 7; Full-sized prototype machine is different from the model, it is relatively stronger and more efficient, comparing peaks of maximum rotational speed in Figure 7; from Figure 5 made diagrams - spiral casing pressure site test at Figure 6, rotating speed site test at Figure 7, wicket gate servo strokesite test Figure 8, draft tube pressure site test Figure 9, shows that the transient traces at full load rejection are different; Physical instability is the reason for waviness of all lines; Different maximum peaks in the spiral casing pressure and minimums in draft tube pressure are evident in Figure 6 and 9; Comparing site tests and calculation results in Figure 6 and 7 it is clear that mathematical instability add more instability as the result of uncertainty and inaccuracy [8].

5. Conclusion

This paper presents 9 new sets of data for the complete head and torque characteristics for a wide range of radial pump-turbines (*nq*=24.8; *nq*=27; *nq*=28.6; *nq*=38; *nq*=41.6; *nq*=50; *nq*=54.7; *nq*=56; *nq*=85.05). It represents an additional value on the currently available data-base in general literature. Also, this paper presents comparison of site tests and calculation results of changes of parameters: spiral casing pressure, rotating speed, wicket gate servo stroke, draft tube pressure, discharge, for the cases of Unit 1 and Unit 2 load rejection, Unit 1 at 281 MW and Unit 2 at 284 MW, when wicket gates closed down rapidly. Calculation results were obtained with numerical model developed to simulate transient processes in the high head pumped storage hydroelectric plants.

6. Nomenclature

D_1	-	[m]	-	the runner inlet largest diameter in the pump operating regime (draft tube facing section)	nq	-	[-]	-	specific speed
N_{11}	-	[-]	-	unit speed	N	-	[min^{-1}]	-	number of revolutions
Q_{11}	-	[-]	-	unit discharge (flow)	Q	-	[m^3/s]	-	discharge
H	-	[m]	-	head	P	-	[MW]	-	power
T_{11}	-	[-]	-	unit torque	T	-	[Nm]	-	torque
Wh	-	[-]	-	head characteristics	Wm	-	[-]	-	torque characteristics
θ	-	[°]	-	angle	$WGSS$	-	[mm]	-	wicket gate servo stroke
Hp	-	[m]	-	pressures in the spiral casing	Hs	-	[m]	-	pressures in the draft tube
tb	-	[s]	-	time braking	Tc	-	[s]	-	time closing

7. References

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