FLOW RATE MEASUREMENT USING THE PRESSURE-TIME METHOD IN A HYDROPOWER PLANT CURVED PENSTOCK

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Abstract

One of the basic flow rate measurement methods applied in hydropower plants and recommended by the International Standard IEC 41: 1991 is the pressure-time method, also called Gibson method. The method consists in determining the flow rate (discharge) by integration of the recorded time course of pressure difference variations between two cross-sections of the hydropower plant penstock. The accuracy of measurement depends on numerous factors and - according to the international standard - generally is confined within the range of 1.5-2.3 %. Following the classical approach, Gibson method applicability is limited to straight cylindrical pipelines with constant diameters. However, the IEC 41: 1991 International Standard does not exclude application of this method to more complex geometries, i.e. curved pipeline (with elbows). It is obvious that a curved pipeline causes deformation of the uniform velocity field in pipeline cross-sections, which subsequently causes aggravation of the accuracy of Gibson method flow rate measurement results.

The influence of a curved penstock application on flow rate measurements by means of the Gibson method is discussed in this paper. The CFD solver *Fluent* has been used for this purpose. Computations have been carried out in order to find, the so called equivalent value of the geometric pipe factor F required when using the Gibson method. An example of the Gibson method application to a complex geometry (two elbows in a penstock) is presented. The systematic uncertainty (error) caused by neglecting the effect of the elbows on velocity field deformation has been estimated.

INTRODUCTION

The current-meter, pressure-time (Gibson), tracer and ultrasonic techniques belong to the primary methods of discharge measurement through hydraulic machinery [IEC 41:1991]. The first three ones are traditional methods, the fourth one, however, is recently the subject of numerous research works focusing on its progress and accuracy. Nowadays the ultrasonic method is ever more frequently applied in hydraulic flow systems, mainly, because of its capability for continuous flow rate monitoring. However, the basic flow measurement methods in hydraulic machines efficiency tests are still the current-meter and pressure-time methods. Moreover, the current-meter method, very frequently applied in hydropower plants, has been lately substituted by the pressure-time method in power plants of medium and high heads. This is mainly the result of a number of advantages over the current-meter method, for instance lower costs of using the pressure-time method, which is related to the development of computer techniques that simplify the process of data acquisition and processing and are more likely to provide higher accuracy[1].

The classical approach to the pressure-time method application is limited to straight pipelines with constant diameters. However, the International Standard IEC 41: 1991 does not exclude application of this method to more complex geometries, i.e. curved pipeline (with elbows). It is obvious that a curved pipeline causes deformation of the velocity field in pipeline cross-sections which subsequently causes aggravation of the accuracy of Gibson method flow rate measurement results. In this paper a special numerical procedure is proposed for considering the influence of a penstock elbow (or elbows) on the pressure-time method results. The procedure is based on the CFD simulation using a commercial software. Its application can give possibility of improving flow measurements results achieved from the pressure-time method. As an example, the measurements of the flow rate through a 180 MW hydraulic turbine are described including a discussion of the particular conditions of a penstock containing two pipeline elbows.

[1] The increased accuracy of the devices used for pressure measurements and the use of computer techniques for collecting recorded data and their numerical processing make this method more attractive than dated versions which employed optical techniques to record pressure changes combined with manual graphics.

THEORETICAL PRINCIPLES OF THE PRESSURE-TIME METHOD

Preliminary remarks

The pressure-time method utilises the effect of liquid flow transients (water hammer phenomenon) in a pipeline [IEC 41:1991, Gibson 1923, 1960, Troskolanski 1960]. The method consists in measuring a static pressure difference which occurs between two cross-sections of a pipeline as a result of a momentum change. This condition is induced when the liquid flow in a pipeline is stopped using a cut-off device, for instance a turbine wicket gate. The flow rate is determined by integrating, within a proper time interval, the measured pressure difference time-variation caused by the water hammer phenomenon.

Before discussing the theoretical principles of the pressure-time method, it is proper to present some preliminary remarks. The pressure-time method can be used in cases in which the liquid density change and the pipeline wall deformation resulting from the pressure increase caused by stopping the stream of liquid are negligibly small. On the one hand, the objects of interest are rather non-elastic pipes, for instance steel or concrete pipelines (penstocks), and incompressible liquids, such as water, for instance. On the other hand side, pressure rises caused by the stopped stream of liquid in a pipeline should be relatively small – smaller than the possible maximum values, which are observed in the conditions of a so-called "simple water hammer" caused by a very fast closure of cut-off devices – i.e. in the time shorter than that of the pressure wave passage along a pipeline. In other words, when this method is used, the closure duration for the devices cutting-off the flow should be at least several times longer than the wave passage time.

Mathematical relationships

In order to derive a relationship for computing the volumetric flow rate Q let us consider a pipeline with the flow section area A that may change along its length - Fig.1. Let us assume that the water stream is stopped by a cut-off device. Taking into account one pipe segment of length L, limited between cross-sections 1-1 and 2-2, we assume that the velocity and pressure distributions in cross-sections of this segment are constant. Also it is assumed that the fluid density and the flow section area do not change due to the water hammer effect.

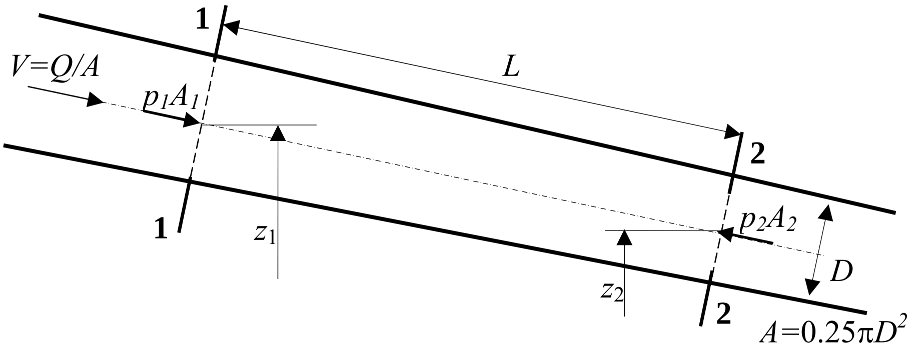

Fig. 1. Segment of a pipeline with marks needed to explain the theoretical basis of the pressure-time method.

According to these assumptions, the relationship between the parameters of the one-dimensional unsteady flow in two selected cross-sections of a pipeline can be described using the well known from the literature [Cengel & Cimbala 2006] energy balance equation:

$$\alpha_1 \frac{\rho Q^2}{2A_1^2} + p_1 + \rho g z_1 = \alpha_2 \frac{\rho Q^2}{2A_2^2} + p_2 + \rho g z_2 + \Delta P_f + \rho \frac{dQ}{dt} \int_0^L \frac{dx}{A(x)} \tag{1}$$

where ρ means liquid density, p_1 and p_2 present static pressures in pipeline sections 1-1 and 2-2, respectively (see Fig. 1), z_1 and z_2 are elevations of 1-1 and 2-2 hydrometric pipeline section weight centres, α_1, α_2 are the Coriolis coefficients[2] (kinetic energy correction coefficients) for 1-1 and 2-2 sections, respectively, Q is the flow rate (discharge), g means gravity acceleration and, finally, ΔP_f is the pressure drop caused by friction losses between
1-1 and 2-2 sections.

Let us introduce the following quantities to Eq. 1:

- Static pressure difference between 2-2 and 1-1 pipeline sections related to the reference level:

[2] The value of the Coriolis coefficient for fully developed turbulent flow in the pipeline is within the limits from 1.04 to 1.11 [Cengel & Cimbala 2006].

$$\Delta p = p_2 + \rho g z_2 - p_1 - \rho g z_1 , \tag{2}$$

- Dynamic pressure difference between 2-2 and 1-1 pipeline sections:

$$\Delta p_d = \alpha_2 \frac{\rho Q^2}{2A_2^2} - \alpha_1 \frac{\rho Q^2}{2A_1^2} , \tag{3}$$

- Geometrical factor of the examined pipeline segment of L length:

$$F = \int_0^L \frac{dx}{A(x)} . \tag{4}$$

Then, we get the differential equation in the form:

$$\rho F \frac{dQ}{dt} = -\Delta p - \Delta p_d - \Delta P_f \tag{5}$$

The left hand side term[3] of equation (5) is the unsteady term which takes into account the history of the volumetric flow rate variation $Q = V A$, recorded during the flow transients course. So, this term represents the effect of fluid inertia in the examined pipeline segment.

After integrating equation (5) over the time interval (t_0, t_k), in which the flow conditions change from initial to the final ones, we obtain the flow rate difference between these conditions. If we assume that we already know the flow rate value in the final conditions (q_k), i.e. after the cut-off device has been closed, we get the following formula for the volumetric flow rate under initial conditions (before the water flow stoppage was initiated):

$$Q_0 = \frac{1}{\rho F} \int_{t_0}^{t_k} \left(\Delta p(t) + \Delta p_d(t) + \Delta P_f(t) \right) dt + q_k \tag{6}$$

The flow rate in the final conditions (q_k), if different from zero due to leakage in the closing device, has to be measured or assessed using a separate method.
The above integral formula reveals that in order to determine the flow rate Q_0, the pressure drop ΔP_f caused by hydraulic loss in the examined pipeline segment and the dynamic pressure difference Δp_d in the hydrometric sections of the pipeline should be extracted from the measured static pressure difference Δp between these sections. The values of the last two quantities should be calculated using their dependence on the flow rate square.

We should be aware of differences between the real flow in pipelines and its theoretical model which should not be too excessive and should not result in significant errors in the flow rate evaluation. Besides those differences, other main sources of inaccuracy of the considered flow measurement method include the inaccuracy of the measuring devices used [Adamkowski & Janicki, 2007], the numerical calculations applied and determination of the F factor. The problem of determining the F factor (Eq. 4) in case of the pressure-time method applied to a curved pipeline measuring segment is the main topic of this paper.

PROPOSAL FOR THE PROBLEM SOLUTION

The geometrical factor F (Eq. 4) for a pipeline segment of L length consisting of K sub-segments with different dimensions is defined by the following formula:

$$F = \int_0^L \frac{dx}{A(x)} = \sum_{k=1}^{k=K} \frac{l_k}{A_k} , \quad \text{with} \quad \sum_{k=1}^{i=K} l_k = L \tag{7}$$

and l_k and A_k denoting the length and internal cross-sectional area of the k-th sub-segment, respectively.

The value of geometrical factor F, as determined from Eq. (7), is generally correct for a straight pipeline segment with no irregularities. It does not account for changing the velocity profiles in a curved pipe flow element. Therefore, in order to take into account the influence of the irregular shape of a considered flow element on the Gibson method results, the authors of this paper propose the following procedure of calculation:

Step 1: Determine the boundary conditions (geometry of a considered pipeline flow system, discharge Q_j, etc.) and assume the computational control space – Fig. 2.

Step 2: Divide the computational control flow space on n numerical cross-sections (sub-cross-sections) which are perpendicular to the axis of the considered i-th ($i = 1, 2, \dots, n$) pipe element – Fig. 2.

[3] For steady flows this term is equal to zero and then equation (1) takes the form of the Bernoulli equation for the real liquid flow .

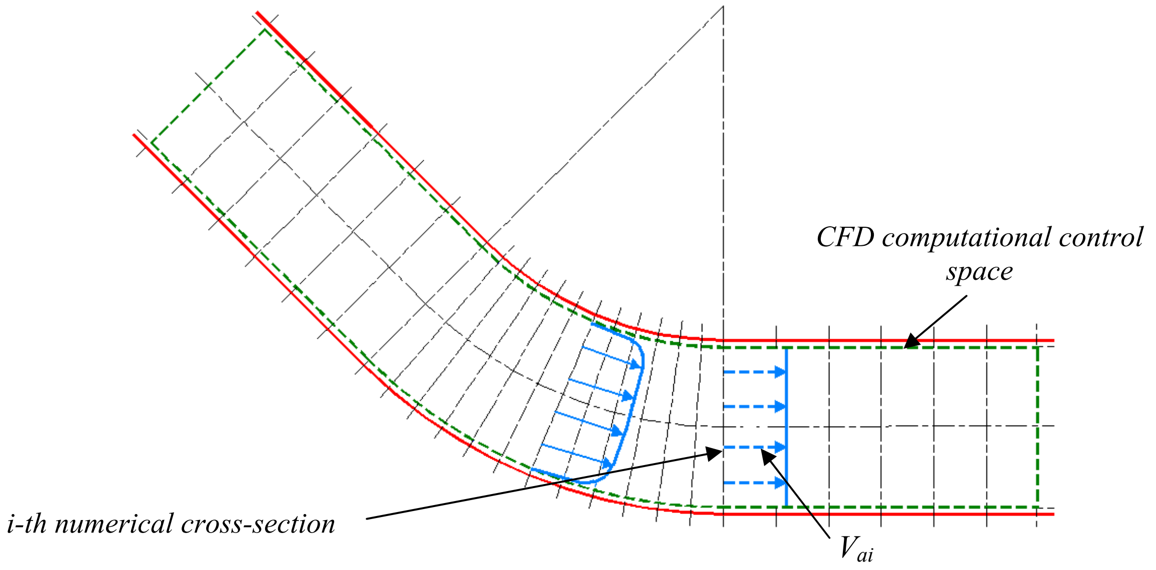

Fig. 2. A pipeline elbow with marked computational space.

Step 3: Simulate velocity distributions (velocity field $V(x,y,z)$) in the flow elements of the considered pipeline within the frame of the computational control space by means a relevant computer software (for instance *Fluent*).

Step 4: Compute average values of flow (normal) velocity V_{ai} for each *i-th* numerical cross-section from the previously derived *CFD* results (step 3) and the assumption of equal kinetic energy resulting from the simulated and the uniform flow velocity distribution:

$$\dot{e}_{k\,CFD\,i} = \dot{e}_{k\,a\,i}\,; \quad \rho = const. \tag{8}$$

$$\dot{e}_{k\,CFD\,i} = \iint_{A_i} \frac{1}{2} V_i^2 \left[\rho V_i \, dA\right] = \frac{1}{2} \rho \iint_{A_i} V_i^3 \, dA \tag{9}$$

$$\dot{e}_{k\,a\,i} = \frac{1}{2} \dot{m} V_{ai}^2 = \frac{1}{2} \rho A_i V_{ai}^3\,; \qquad \dot{m} = \rho V_{ai} A_i \tag{10}$$

$$\Downarrow$$

$$V_{ai} = \left(\frac{\iint_{A_i} V_i^3 dA}{A_i} \right)^{1/3} \tag{11}$$

where V_i denotes the flow velocity axial component (perpendicular to the *i-th* cross-section).

Step 5: Basing on the continuity equation $Q_j = V_{ai} A_{ei} = const$ compute the equivalent value of cross-sectional area A_{ei} for each numerical cross- section ($i = 1, 2, \ldots, n$):

$$A_{ei} = \frac{Q_j}{V_{ai}}, \quad i = 1, 2, \ldots\ldots\ldots, n \tag{12}$$

Step 6: For the considered flow rate Q_j through the analyzed pipe element, compute the equivalent value of the factor F_{eQj} from the following formula:

$$F_{eQj} = \sum_{i=1}^{n-1} \frac{l_{i \to i+1}}{0.5(A_{e\,i} + A_{e\,i+1})} \tag{13}$$

where $l_{i \to i+1}$ is the distance between the average velocity centers of numerical cross sections i and i+1, A_{ei} and $A_{e\,i+1}$ – equivalent areas of computational cross sections, i and i+1, respectively.

The above computation should be conducted for several discharge values (Q_j, $j = 1, 2, ..., m$) from the whole scope of its change ($0 < Q_j < Q_{max}$). The average value of equivalent F factor, calculated as follows:

$$F_e = \frac{1}{m}\sum_{j=1}^{m} F_{eQj} \quad , \tag{14}$$

is recommended to be used in the pressure-time method.

In the calculations presented above, it has been assumed that the changes in the velocity profiles are the same during steady and unsteady (transient) flow conditions. This assumption is close to reality for not very fast closure of turbine wicket gates during Gibson method tests. Practically, such conditions occur in all water turbines, because it is necessary to protect turbine flow systems against the water hammer destructive effect.

Taking the equivalent value of F_e instead of the value F calculated directly from the pipeline segment geometry, it is possible to increase Gibson method accuracy in cases when pipelines are curved.

EXAMPLE

Flow measurement conditions

The example refers to the flow rate measurements using the pressure-time method in the turbine penstock with two elbows - Fig. 3. The flow rate was measured using the pressure-time method in the version based on separate pressure measurements in two hydrometric sections. In each of these sections, four pressure taps were installed and connected by impulse tubes to the manifold and pressure transducer. A typical manifold was used in the lower penstock section 2-2. The whole system of pressure collection and measurement in this section was prepared using the access from the outside of the penstock.

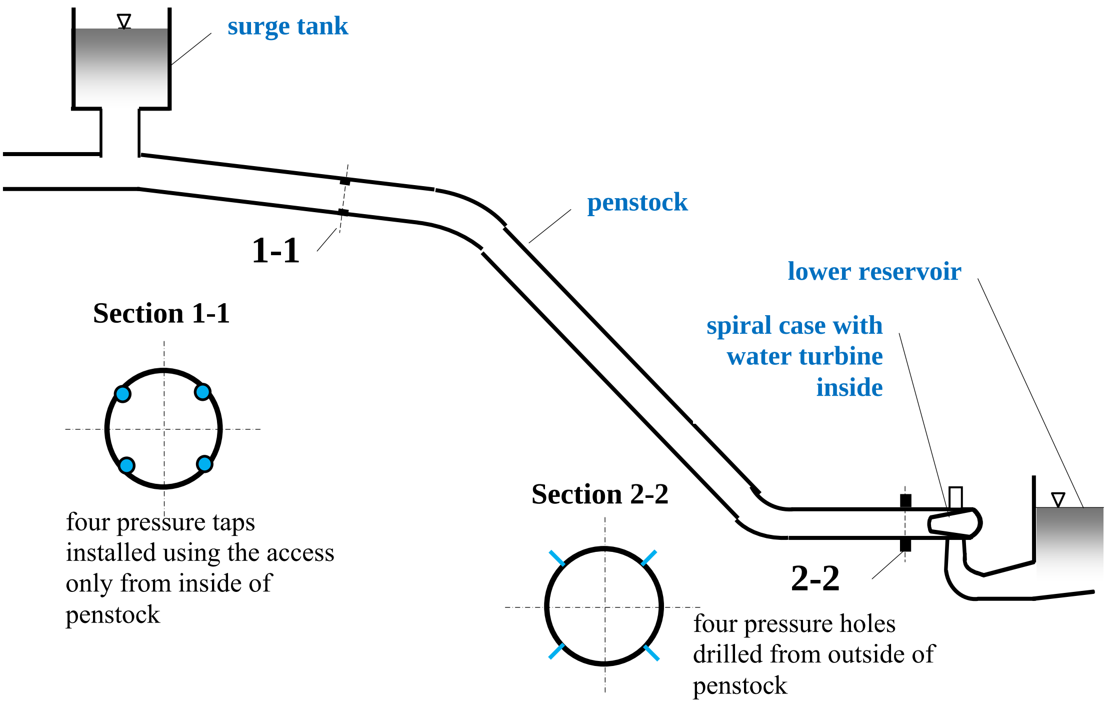

Fig. 3. A scheme of the tested turbine flow system.

Tab. 1. Geometrical data of the penstock segment measured between cross-sections 1-1 and 2-2.

Segment name	Segment length	Diameter	Cross-sectional area	Geometrical Factor
	L **[m]**	***D*** **[m]**	***A*** **[m^2]**	***F= L/A*** **[m^{-1}]**
Cylinder no. 1	14.390	6.5	33.18315	0.43365383
Elbow no. 1 (R= 23894mm)	16.681	6.5	33.18315	0.50269968
Cylinder no. 2	68.940	6.5	33.18315	2.07756045
Elbow no. 2 (R= 25800mm)	20.263	6.5	33.18315	0.61064946
Cylinder no. 3	19.440	6.5	33.18315	0.58583950
Total:	**139.714**		**Total:**	**4.21040292**

Since there was no access from outside to the upper section 1-1 (the penstock was buried in a rock mass and surrounded by concrete), a special instrumentation for pressure reception and measurement was installed inside the penstock, using a manhole for access [Kubiak & others 2005, Sierra & others 2006, Adamkowski & others 2007, Urguiza & others 2007] – see Fig. 4 and 5. Figure 5a) shows one flat bar with a pressure tap. The flat bar was welded to the penstock wall (shell), along the flow direction. Four pressure holes were connected, using copper tubes, to the manifold (Fig. 5b)), inside which the absolute pressure transducer of 0.1 precision class was mounted hermetically. It is worth stressing that preparing the pressure measuring system inside a penstock of 6 m in diameter and inclined by an angle of 40^0 degrees was an extremely difficult task.

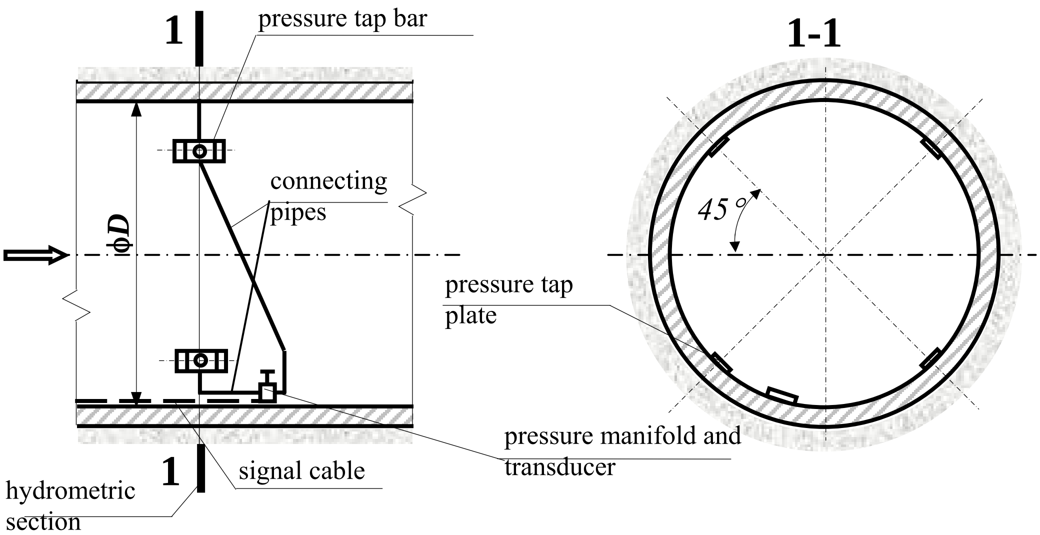

Fig. 4. Distribution of pressure tap bars in 1-1 section and their connection to hermetic manifold with absolute pressure transducer installed inside.

The length of penstock sub-segments between cross-sections 1-1 and 2-2 was determined by direct measurement. The penstock diameters were read from the technical drawings and experimentally verified. The resulting penstock geometry data are presented in Table 1. These data enable determining the *F* geometrical factor, needed in the Gibson method application, without taking into account the velocity profile changes caused by two elbows in the analyzed penstock segment.

a)

b)

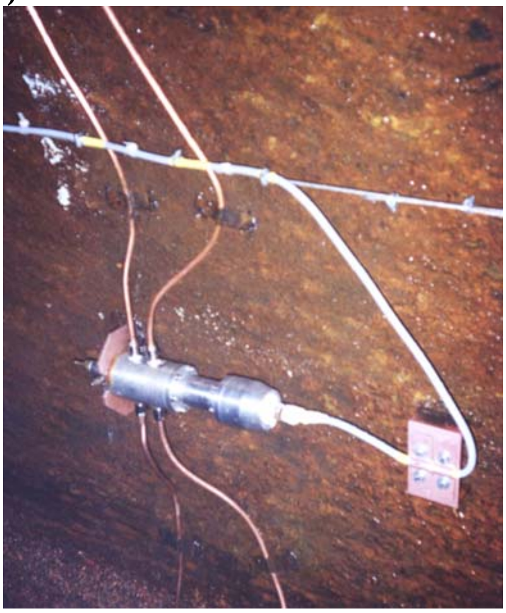

Fig. 5. a) View of a pressure tap bar used in measuring section 1-1; b) the manifold installed inside penstock (section 1-1) with the mounted absolute pressure transducer.

Flow simulation

The flow simulation was conducted using the *Fluent 6.3* commercial software (solver). This software employs a cell-centered finite-volume method.

The flow was simulated by solving the Reynolds Averaged Navier-Stokes equations in steady-state in the full three-dimensional coordinate system.

The geometry of the considered penstock segment was taken from Table 1 and the available design drawings. A computational grid that represents the geometry was created using the *Gambit 2.4* commercial software. The flow boundary conditions were assumed from the operation conditions of the tested hydraulic turbine.

The *k-ω* Shear Stress Transport (SST) turbulence model was chosen for computations of the analyzed cases. This turbulence model is very suitable for non-oblique (parallel) flows and also for flow without stagnation regions and strong accelerations. Otherwise, it produces too large turbulence.

Six values of mass flow (or discharge) from the entire range of the flow rate change (up to 200 000 kg/s - ~200 m^3/s) were taken to computations. Table 2 presents flow rate values assumed for simulation, the related Reynolds numbers *Re* and turbulence intensities *I*. The turbulence intensity is required to determine the boundary conditions at the inlet and outlet cross-sections. It was defined by the following empirical formula [Fluent 6.2 Documentation]:

$$I = 0.16\,\mathrm{Re}^{-0.125} \tag{15}$$

Tab. 2. Reynolds numbers and turbulence intensities at the inlet and outlet penstock cross-section for all simulated flow cases.

Mass flow rate	Inlet cross-section		Outlet cross-section	
	Re	*I*	*Re*	*I*
[kg/s]	[-]	[%]	[-]	[%]
30000	3435617	2.438	5123288	2.320
60000	6871233	2.236	10243502	2.127
95000	10875176	2.111	16219215	2.008
130000	14883530	2.030	22194254	1.931
170000	19462278	1.963	29023256	1.867
200000	22904136	1.923	34155236	1.830

For the considered cases, the following fluid data were assumed for water in 10°C temperature:

- density $\rho = 999.7$ kg/m^3,
- dynamic viscosity $\mu = 0.001308$ Pa·s.

The numerical grids were created with about 1.8 million of hexahedral elements. On the basis of preliminary analysis it was stated that the density of the created grids is satisfactory for the chosen turbulence model and for analyzed flow cases.
The *F* factor was calculated between the 1-1 and 2-2 cross-sections (Fig. 3). In order to determine the velocity profile with boundary layer at the 1-1 cross-section the inflow segment was created. The virtual outflow segment was also created downstream the 2-2 cross-section for reducing the effect of outlet section on the simulated back field flow. The flow boundary conditions at the inlet section were assumed in the form of uniform velocity distribution (with no boundary layer) and the flow boundary conditions at the outlet section were determined by the given turbulence intensity and pressure level.
To obtain the equivalent value of *F* factor, the penstock segment between 1-1 and 2-2 cross-sections was divided numerically into subsegments (Fig. 6) for which the average velocities were calculated. It is worthwhile to stress that each elbow was divided with higher density than the straight penstock part.

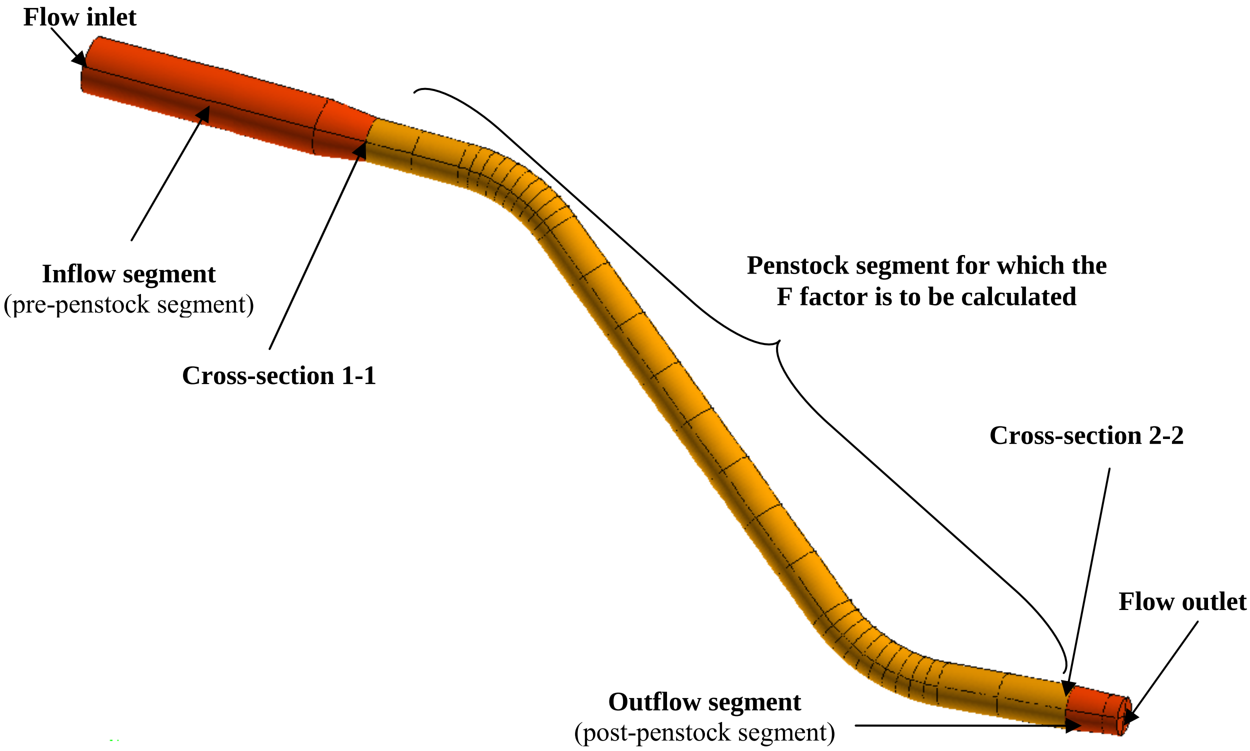

Fig. 6. The entire geometry domain created to calculations. There is presented domain where numerical Gibson's factors *F* were calculated (between cross-sections 1-1 and 2-2) and in- and outflow segments as well.

In order to obtain satisfactory numerical accuracy, the second-order upwind discretization scheme was used for computations. An example of a cross-section with the created grid is shown in Fig. 7.

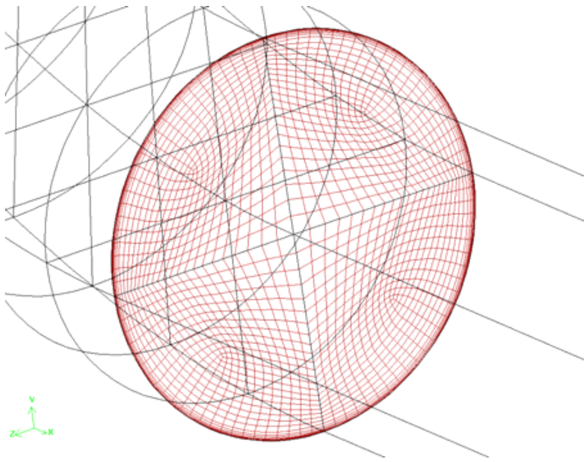

Fig. 7. Grid example in a cross-section.

Moreover, Fig. 8 and Fig. 9 present the numerically simulated velocity distribution patterns in the cross-sections between sub-segments in the considered elbows for the largest assumed discharge ~200 m^3/s (200000 kg/s).

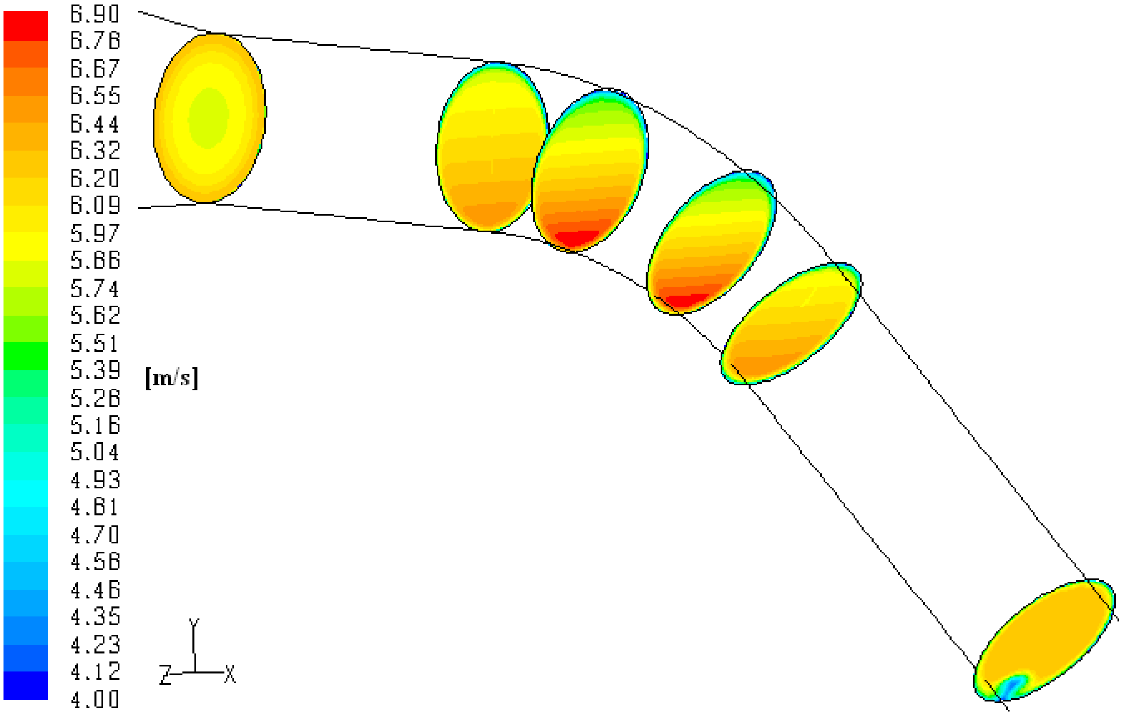

Fig. 8. The velocity magnitude distribution in the penstock cross-sections within the elbow no. 1 and cylindrical conduits upstream and downstream for mass flow 200000 kg/s (Q = ~200 m^3/s).

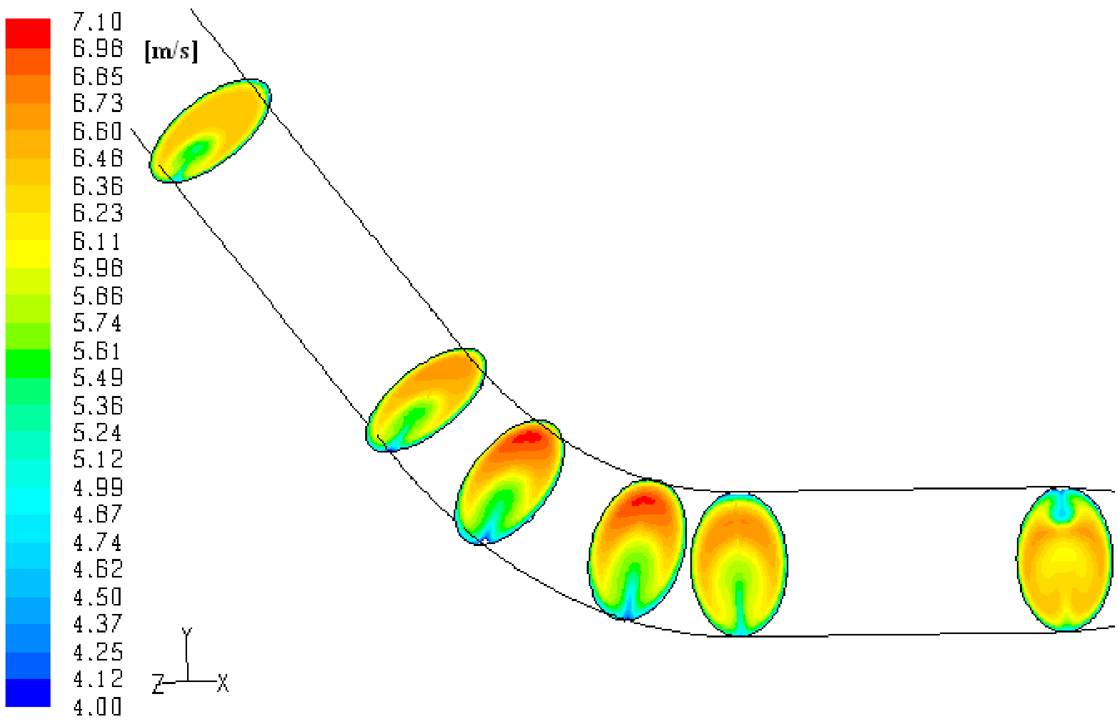

Fig. 9. The velocity magnitude distributions in the penstock cross-sections within the elbow no. 2 and cylindrical conduits upstream and downstream for mass flow 200000 kg/s (Q = ~200 m^3/s).

The simulation results obtained can be generally characterized as follows. The inflow to the first elbow (elbow no. 1) is uniform - Fig. 8. Elbow no. 1 introduces non-uniformity (disturbance) in the flow pattern which propagates along the straight penstock segment with smaller intensity to the second elbow (elbow no. 2), introducing some additional disturbance to the flow field - Fig. 9.

The obtained numerical results were used to calculate the equivalent factor F_e (according to the steps 4-6 in the procedure presented above). In order to present the results, a deviation factor Δf was introduced. It is a relative difference between the equivalent factor F_e and the geometrical factor F, calculated as follows:

$$\Delta f = \frac{F_e - F}{F} . \tag{16}$$

The quantity Δf, determined as a function of discharge, is presented in Fig. 10. The average value of Δf is about 0.45 %. The value Δf = 0.45 % was taken to correct the flow rate values following from the preliminary calculation, in the efficiency tests of the considered hydraulic turbine. It is worthwhile to notice that the values of Δf deviation factor are positive and do not change substantially versus flow rate.

It can be concluded that disregarding this correction may result in overestimation of the discharge value by ca. 0.45 % and consequently - too small value of the measured hydraulic turbine efficiency.

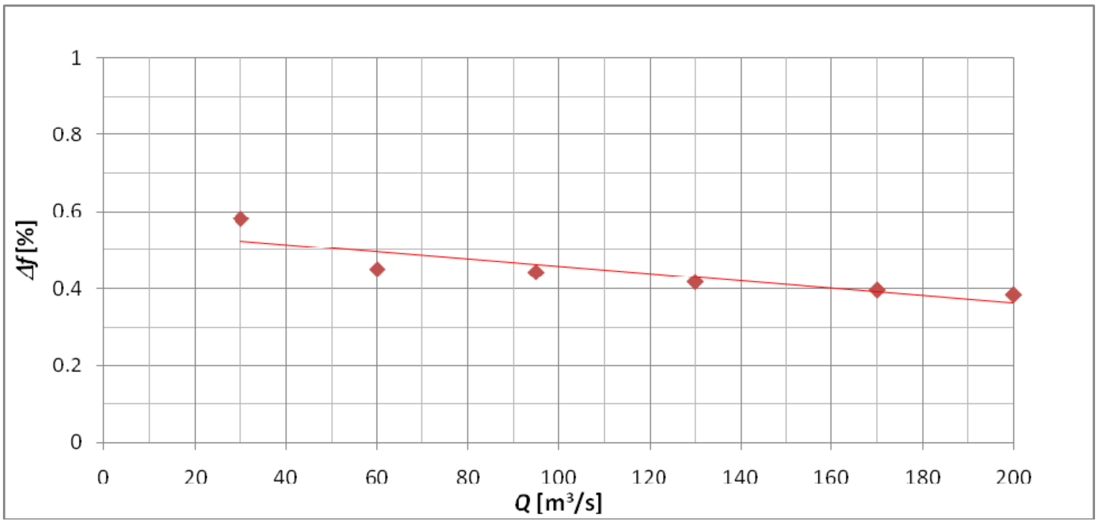

Fig. 10. The values of *Δf* deviation factor determined for the assumed flow rate values.

Experimental results

The experiment, described below, was conducted at several power values (ranging between 80 MW and 190 MW).
After setting the electric power value, constant guide vane opening was kept for about 240 s in order to stabilize the flow conditions in the entire hydraulic system. After this time period, the hydraulic turbine parameters were recorded for at least 180 s. Then, the flow rate measurement was performed using the Gibson method. For this purpose the flow was cut-off using the turbine guide vanes. The speed of the closure was similar to that applied in emergency turbine stoppage cases. Following the Gibson method requirements, the generator was not disconnected from the electric grid even after the guide vanes were in closed position and for a time period of about 60 s required for stabilization. It caused absorption of small power from the electric grid. After this procedure the turbine guide vanes were opened and measurement at another test point was performed.
In order to measure the pressure at cross-sections 1-1 and 2-2 semiconductor pressure transducers with technical data as described below were applied:

- measurement range:	upper cross-section (1-1) (0÷1100) kPa absolute,
	lower cross-section (2-2) (0÷1600) kPa relative.
- accuracy class (basic error):	0.1 %.
- linear error:	0.04 %.

Flow rate values were computed using the *GIB-ADAM* code developed in the Szewalski Fluid-Flow Machinery Institute in Gdansk of the Polish Academy of Sciences [Adamkowski & Waberska 2005]. The input data

consists in the recorded time-variations of pressure difference between 1-1 and 2-2 penstock cross-sections and factor *F* calculated basing on the geometry measurement and the numerical simulation.
Fig. 11 shows the time-histories of (1) the recorded wicket gate closure, (2) the static pressures measured in the sections 1-1 and 2-2, (3) the static pressure difference resulting from (2), and (4) the flow rate calculated using the *GIB-ADAM* code. The frequency of the recorded pressure difference time-variations was 500 Hz. The input data files to calculations using *GIB-ADAM* code were prepared in an ASCII format applying the sampling frequency of 100 Hz.

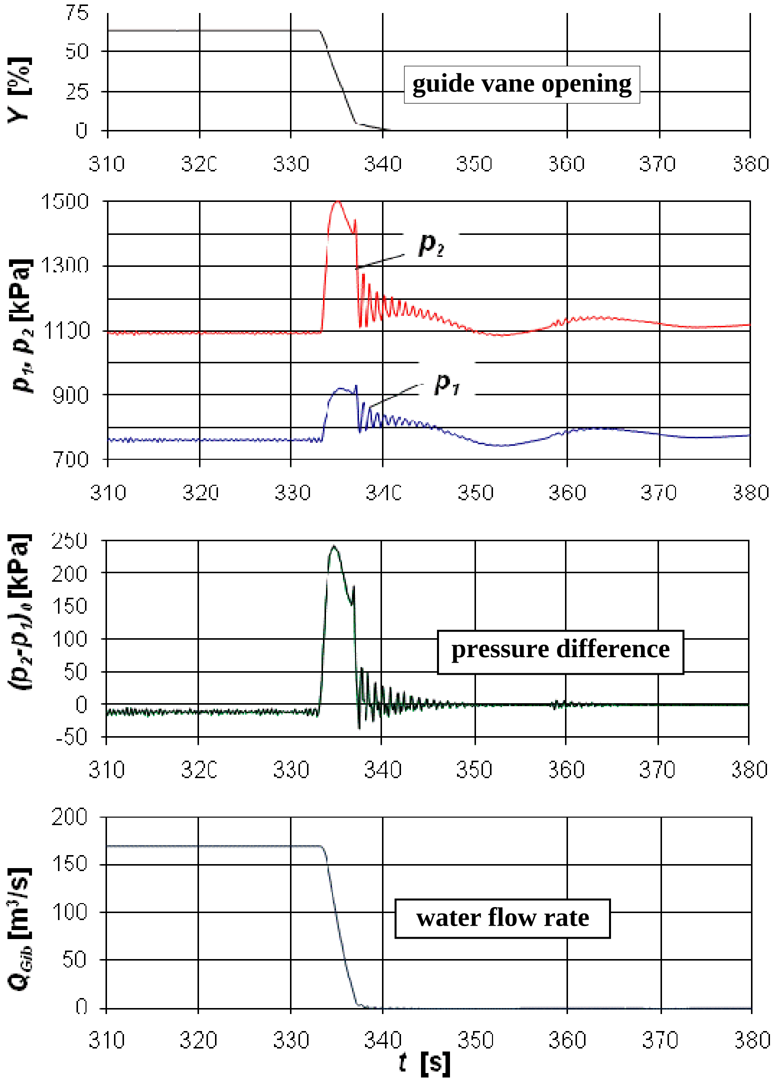

Fig. 11. Pressure changes measured in the turbine penstock measuring sections and the flow rate determined on this basis.

The leakage through the closed guide vanes was calculated from the rate of water level fall in the cylindrical penstock segment (the cut-off valve was open). The obtained leakage (at low head) was recalculated to the values corresponding to the flow cut-off process (as described above), using the formula:

$$q_k = q_{k-mesure}\left(\frac{p_t - p_m}{p_{t-meaure} - p_{m-mesure}}\right)^{0.5} \tag{17}$$

where q_k denotes the leakage flow rate through the closed turbine wicket gates after their closure during the Gibson test, p_t and p_m are static pressures in the spiral case and between the turbine wicket gates and the runner, respectively, $q_{k\text{-}measure}$ is the flow rate through the closed turbine wicket gates during leakage test measured at static pressures $p_{t\text{-}measure}$ and $p_{m\text{-}measure}$, in the spiral case and between turbine wicket gate and runner respectively.

Basing on the results following from the executed tests, the efficiency characteristics were derived. For instance, two efficiency curves are presented in Fig. 12. The first curve represents efficiency calculated using the geometrical factor *F* without correction. The second one represents efficiency calculated using the equivalent value of factor $F = F_e$ obtained according to the procedure proposed above. The correction shows that the *F* factor increase by 0.45 % which means decreasing of the flow rate and increasing the hydraulic turbine efficiency.

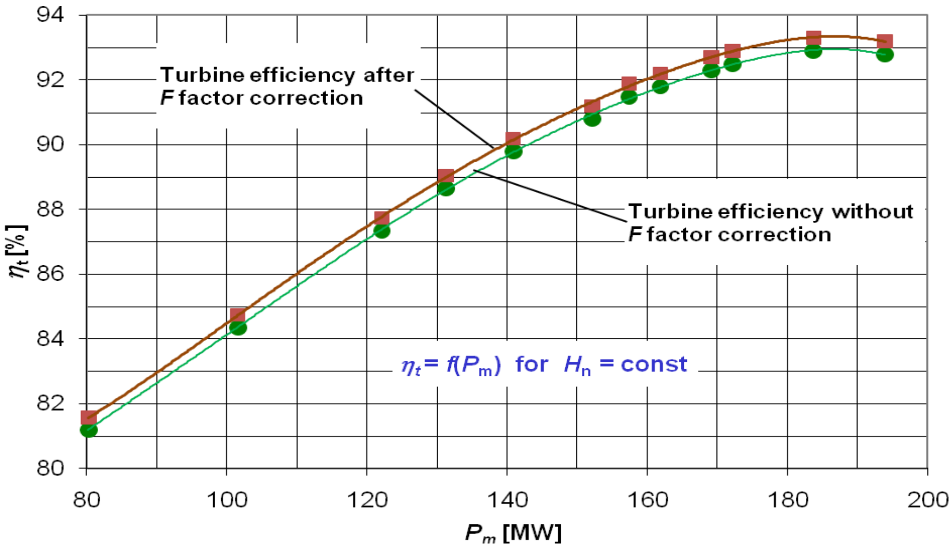

Fig. 12. Turbine efficiency versus mechanical power determined for the *F* factor calculated directly from the penstock geometry and for the *F* factor corrected by the developed procedure.

CONCLUSIONS

The flow rate through a 180 MW nominal power water turbine was measured by means of the pressure-time method (Gibson method). The application of this method concerned a large hydropower penstock with two elbows (diameter of 6 m). The curved liquid streams in the elbows disturb the regular flow pattern that could exist otherwise in the straight penstock segment. The irregular distribution of flow velocities directly affects the value of discharge measured by means of the Gibson method. A special numerical procedure has been developed for considering the influence of a penstock elbow (or elbows) on change of the *F* factor value, and, consequently, on the discharge values determined using the Gibson method. Basing on the CFD (Computational Fluid Dynamics) simulation by means of the *Fluent* solver, this procedure allows to calculate the equivalent value of *F* factor for a penstock segment with elbow (or elbows) which improves the discharge measurement accuracy. Numerical calculations have been executed for several values of discharge from the whole range of its variation in order to estimate the average value of equivalent *F* factor. It is worth saying that the *F* factor values, obtained on the base of the CFD simulation, only slightly decrease with the Reynolds number (flow rate). This feature can be used to justify correctness of the flow rate measurement result correction using the procedure presented in this paper.

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