Extension of the Pressure Time Method to 3-Dimensional Flows

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Abstract

Measuring the flow rate is a complex task in determining turbine efficiency before and after refurbishment, especially for low-head machines. The pressure-time method represents a cost-effective solution for estimating the flow rate. This method is based on converting momentum into pressure during the deceleration of the liquid mass caused by closing the guide vanes or a valve.

According to the IEC-60041 standard, the pressure-time method has a total uncertainty range of ±(1.5-2.0)%. This methodology considers one-dimensional flow, and it is limited to straight pipes with uniform cross-sections, subject to certain constraints on pipe length, and fluid velocity. However, low-head hydropower plants typically consist of a short passage with variable cross-section and sometimes a bend, making the application of this method challenging. These conditions affect the method's accuracy, leading to measurement errors.

This paper presents recent developments in the pressure time method to perform well also under such conditions. To address this issue, an experimental setup was developed at Luleå University of Technology to study the pressure-time method with the presence of a reducer and a bend for conditions similar to low-head hydropower. Additionally, a novel approach is introduced where transient 3D CFD simulation is combined with the regular 1-D formulation.

Keywords: Pressure-time method, low-head hydraulic turbine, flow rate calculation, 3 CFD

1. Introduction

Hydropower, an environmentally friendly and renewable energy source, the development of which commenced in the late 19th century, has been extensively utilized and remains a key means for renewable electricity generation. However, many hydropower facilities were constructed between 50 to 70 years ago, necessitating a refurbishment for many turbines. Hence, it is imperative to compare the efficiency of these turbines before and after refurbishments to ensure the fulfilment of guarantees and optimize operational performance. Among all parameters used to measure efficiency, discharge is regarded as one of the most challenging hydrodynamic parameters, especially for low-head machines [1].

Among different discharge measurement methods, the pressure-time method (PTM) stands out as relatively inexpensive and straightforward to implement, with an uncertainty of approximately ±(1.5-2.0)% [2]. The standard PTM assumes a one-dimensional flow (1D) and is based on the pressure rise caused by the deceleration of the flow in a closed conduit, such as a penstock in hydropower plants. Pressure is measured between two cross-sections during the deceleration of the liquid mass by closing a valve or turbine wicket gates to obtain the initial flow rate. However, the PTM has limitations that make it challenging to implement in low-head machines [2]. These limitations include:

- The measuring length (L) must be at least 10 m.
- The product of average velocity and length must exceed 50 m^2/s.
- The cross-section must remain constant without any significant irregularity.
- Pressure taps should not be located at a distance less than 2 times the diameter (2×D) from any irregularity.

In previous research, the PTM has shown improvement for shorter lengths and variation in cross-section [1,3–6]. The Energy Equation (Eq. 1) has been utilized for pipes with variable cross-sections or in the presence of secondary flow [1].

$$Q = \frac{1}{\rho C}\int_0^t (\Delta P_f + \Delta P + \Delta P_d)dt \qquad \text{Eq. 1}$$

In Eq.(1), Q, Δp, Δp_f, ΔP_d, C, ρ t_f and q are the flow rate before closure, differential pressure, pressure loss due to friction, dynamic pressure variation, geometry factor, density, final limit of integration and leakage flow rate after valve closure, respectively. The dynamic pressure can be calculated from the relation $\Delta P_d = \alpha_2 \frac{\rho Q^2}{2A_2^2} - \alpha_1 \frac{\rho Q^2}{2A_1^2}$, where α represents the kinetic energy correction factor, defined as the ratio between the flux of kinetic energy calculated from the actual velocity and the flux of kinetic energy calculated from the mean velocity. For laminar flow, α is typically constant equal to 2, while it is approximately 1.05 for steady and fully developed flow [7]. However, in cases of skew profiles, a higher value of the kinetic energy coefficient is anticipated due to the deviation of the velocity profile from the developed flattened flow across the section [8]. Another parameter in the PTM equation is the geometry factor which can be calculated by Eq. (2) [9].

$$C = \int_0^L dx/A_c. \qquad \text{Eq. 2}$$

Neyestanaki et al. [10] conducted a comparative analysis of the constant and quasi-steady assumptions in estimating pressure loss due to friction and dynamic pressure variation in a pipe featuring a variable cross-section. Through the application of the quasi-steady assumption for both kinetic energy and friction factor coefficients, they successfully minimized the deviation in comparison to the reference flow meter from -0.72% to -0.42%. Ramdal et al. [11] explored the application of the PTM with the presence of bends. They contended that the introduction of two 45° bends resulted in a flow rate underestimation of around 1%. Conversely, a single 90° bend was associated with a more pronounced underestimation, exhibiting a deviation of 8.5% when compared to the reference flow meter.

The conventional 1D PTM assumes a uniform pressure distribution throughout the cross-section. However, the presence of bends can render this assumption invalid. The non-uniform pressure profile observed in bends may result in pressure measurement at the pressure taps differing from the average pressure across the cross-section leading to inaccurate flow rate estimations. Therefore, it is crucial to carefully consider the impact of bends on pressure measurement accuracy in PTM applications to ensure satisfactory performance.

Computational Fluid Dynamics (CFD) offers invaluable insights into flow characteristics that are beyond the reach of experimental measurements. Numerous CFD investigations have been done to study the transient flow phenomena encountered when employing PTM [1,12–14]. This approach yields a more comprehensive understanding of flow behaviour and enhances the accuracy of flow rate measurements.

Nevertheless, bends and reducers are common features in penstock geometry for both low and medium-head hydropower systems. In such conditions, the kinetic energy correction factor may deviate from constant values. Moreover, pressure measurements at pressure taps may deviate from the mean differential pressure at the section as a result of geometry variations. This paper aims to apply similar limitations as outlined in IEC standards, including the presence of bends and reducers. Therefore, in this paper, a combination of experimental measurements and 3D CFD simulations has been employed to assess the validity of PTM in the presence of bends and reducers.

2. Materials and Methods

2.1. Experimental setup

To broaden the scope of the IEC-60041 standard, a specialized testing apparatus was designed and constructed at LTU. Detailed information regarding the test rig geometry, location of pressure taps, and operating conditions can be found in Ref [8]. In this paper, three sets of measurements were carried out, departing from the IEC standard, as described below:

- measurement between sections VA and HA to assess the impact of a bend between two sections in the PTM accuracy
- measurement between sections HA and HB to assess the impact of a bend on the PTM with a distance shorter than the IEC recommendation
- measurement between sections HD and HE to assess the impact of the cross-section variation in the PTM

The length of the measurements considered was approximately 1 m, with additional details provided in Table 1. The locations of the pressure taps are illustrated in Figure 1.

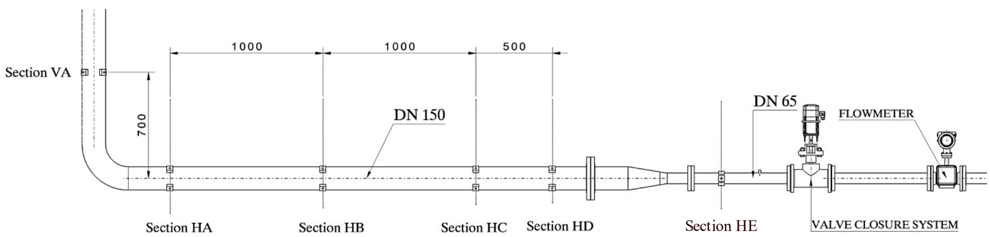

Figure 1: The location of the sections with the pressure taps

Table 1: Test cases description

Test case name	Inlet section	Outlet section	Length (m)	Distance from the elbow to Inlet section	Distance from the elbow to Inlet section	Cross-section
VA-HA	VA	HA	1	-	1.8 D	Constant
HA-HB	HA	HB	1	1.8 D	3.5 D	
HD-HE	HD	HE	1.275	-	-	Variable

The detailed calculation of measurement uncertainty is outlined in Refs [10]. Ten measurements were recorded for each case to calculate the random uncertainty in the measurement. The total uncertainty value with a 95% confidence level (U_{95}) is determined using Eq. (3) [15].

$$U_{95} = 2\sqrt{(b_r^2 + s_r^2)} \qquad \text{Eq. 3}$$

2.2. Flow rate calculation

The aim of the 3D CFD simulation is to analyze the velocity and pressure distribution across various sections of the experimental setup, replicating flow conditions akin to those in the experiment. These simulated profiles are then utilized to estimate the average pressure and kinetic energy correction factor at the measurement section, subsequently applied in the 1D PTM. To achieve this, the time-dependent flow rate derived from the 1D PTM serves as the outlet boundary condition for the 3D CFD simulation. An iterative loop that combines the 1D PTM and 3D CFD is employed to estimate the flow rate. The updated value of the transient flow rate is then used as input for the subsequent iteration of the CFD simulation. This iterative process continues until convergence is achieved, with the residual variation of the estimated flow rate serving as the convergence criteria [8].

2.2.1. **Flow rate calculation with 1D PTM**

In PTM, the flowrate is calculated in a 1D iterative loop based on Eq. (1). Pressure loss is calculated by relation $\Delta \mathrm{P_f}(t) = K\mathrm{Q}(\mathrm{t}_t)|\mathrm{Q}(\mathrm{t})|$ and constant friction factor K can be evaluated based on experimental measurements conducted before valve movement, as outlined in Eq (4) [10].

$$\mathrm{K} = \frac{-\Delta\mathrm{P}(\mathrm{t_0}) - \Delta\mathrm{P_d}(\mathrm{t_0})}{\mathrm{Q}(\mathrm{t_0})|\mathrm{Q}(\mathrm{t_0})|} \qquad \text{Eq. 4}$$

In this context, $\Delta P_d(t_0)$, $Q(t_0)$ and $\Delta P(t_0)$ represent the dynamic pressure difference, flow rate, and measured differential respectively, obtained during the steady-state condition before the initiation of valve movement. Approximately 40 seconds of measurement data were employed to calculate the coefficient K using the average pressure loss $\Delta\mathrm{P}(\mathrm{t_0})$ before the initiation of valve movement. The pressure loss and dynamic pressure variation are updated based on the obtained flow rate at each iteration. The loop will continue until the convergence of the flow rate. The methodology presented in Ref [10] has been used for the estimation of the end point of integration.

Figure 2 illustrates the transient flow rate, Q(t), obtained by the standard pressure-time method based on a differential pressure measurement between sections HA and HB. This approach involves assuming constant values for friction coefficients and the kinetic energy correction factor. Following this, the obtained time-dependent flow rate from standard PTM serves as a boundary condition for the 3D CFD simulation.

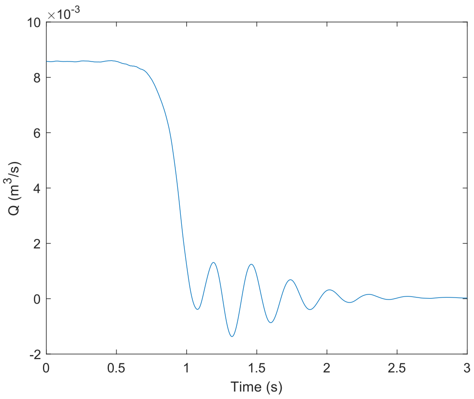

Figure 2: Transient flowrate based on 1D PTM from experimental measurement between sections HA and HB

2.3. Mathematical Modeling

The entire transient process of the PTM is simulated utilizing 3D CFD, integrating the three-dimensional geometry of the experimental setup. The governing equations for the time-dependent, incompressible, turbulent flow are outlined as follows.

$$\frac{\partial(\rho U_j)}{\partial x_j} = 0 \qquad \text{Eq. 5}$$

$$\rho\frac{\partial(U_i)}{\partial t} + \rho\frac{\partial(U_j U_i)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu\frac{\partial U_i}{\partial x_j} - \rho\overline{u_i u_j}\right) \qquad \text{Eq. 6}$$

The low Reynolds k-ω SST turbulence model developed by Menter [16] is utilized to approximate the turbulence shear stress term, $-\rho\overline{u_i u_j}$. This turbulence model, known as SST *k-ω*, is widely recognized for its improved accuracy in predicting fluid flow in the near-wall region at low Reynolds numbers. The value of y^+ is maintained at approximately 1, indicating that the near-wall flow is adequately captured. Temperature and density variation are neglected in the analysis. The coupled finite volume equations of motion are solved using ANSYS-Fluent. The pressure field is computed using the semi-implicit method for pressure-linked equations (SIMPLE) algorithm.

2.3.1. Boundary condition:

To simplify the 3D CFD simulations, a transient flow rate is imposed as an outlet boundary condition instead of explicitly modelling the valve closure. This approach is adopted to circumvent the additional computational time required for simulating the valve closure. The entire upstream geometry of the valve is included in the simulation to account for the influence of the developing flow. The flow domain considered in the simulation is depicted in Figure 3.

Throughout the simulation, the total pressure is considered as the boundary condition at the inlet. The transient flow rate *Q(t)* estimated from the 1D PTM (Figure 2) is utilized as the outlet boundary condition for the 3D CFD simulation. A converged steady state is achieved with an initial mass flow outlet to establish the initial condition.

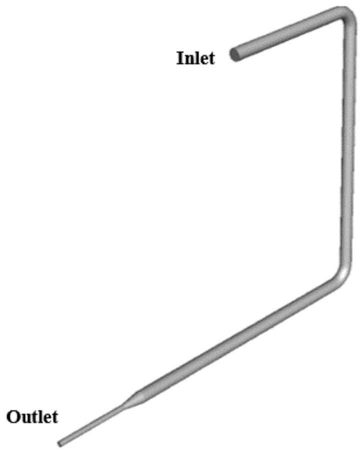

Figure 3: Fluid flow domain.

3. Results and discussion

3.1. Experimental results with 1D PTM

In Figure 4, the mean error and 95% confidence interval ($U_{95} = 2\sigma$) of the uncertainty of estimated flow rates obtained using the 1D pressure-time method are depicted for measurements between sections VA-HA , HA-HB, and HD-HE to examine the effect of the presence of bend between two measurement sections, presence of bend with distance 1.8×D and variation in cross-section on the accuracy of the PTM. Constant kinetic energy correction factor and friction factor coefficient have been considered for the estimation of flow rate with the 1D PTM.

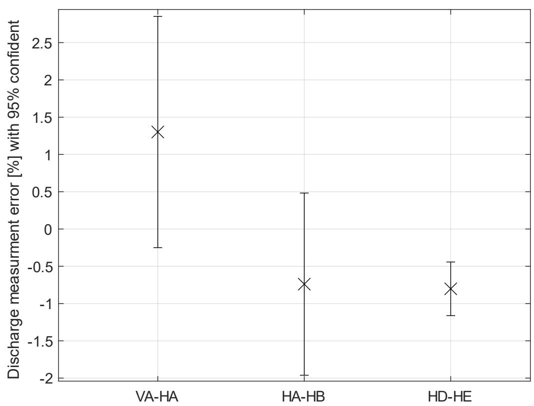

Figure 4: Flowrate error for 3 cases between sections VA- HA, HA- HB and HB-HC. The bars represent the uncertainty at the 95 % confidence

After the bend, the flow profile becomes non-uniform and developing, known as Dean vortices. This leads to a discrepancy between the differential pressure measured across the pressure taps and the average pressure within the pipe sections, resulting in deviations of the measured pressure from the mean pressure. For measurements conducted between sections VA-HA, the mean deviation of flow measurement reaches to highest uncertainty in flow measurement with 1.30% error. For measurements conducted between sections HA-HB, the mean deviation of flow measurement reaches -0.74%. For measurements where the cross-section varies (HD-HE), the presence of a reducer leads to a negative error of -0.83%. The 1D PTM is not able to clarify these deviations as there is a lack of detailed information for 3D fluid flow. The reasons for these deviations will become clearer after 3D CFD simulation.

Among all these measurements, the VA-HA section measurement exhibits the greatest random uncertainty (σ_{95}=1.56%), primarily due to transient secondary flows, which may vary across different measurements. The presence of a reducer between sections HD - HE induces a higher level of differential pressure, encompassing both viscous pressure loss and dynamic pressure variations. This elevated measured pressure leads to a reduction in the random uncertainty for this set of measurements with σ_{95}=0.34%.

3.2. Expansion of PTM by CFD

The presented methodology has been proven to converge after one iteration, as demonstrated in the author's previous paper [8]. The methodology is applied to a sample from each measurement set mentioned in Table 1. The sample has been selected to use measurements that are close to the mean deviation. The effect of applying the proposed methodology involves the variation of the dynamic pressure correction factor and pressure correction for measured differential pressure at the pressure taps. The estimated flow rate is compared to the reference flow rate function of different endpoints and presented in Figure. 5. The deviation of the pressure measurement from the pressure taps has a more significant effect on the flow rate estimation than the dynamic pressure variation for measurement between sections VA-HA and HA-HB. The deviation is changed by around +0.85% for measurement between sections HA and HB. For measurement between sections VA and HA with the highest uncertainty of random measurement, led to a change of deviation around -1.8%. For measurement between section HD-HE, the mean deviation compared to the reference flow meter is changed using CFD data around +0.8%. It also noted that with the presence of a reducer between measurement sections, the kinetic energy correction factor has the highest effect on flow measurement improvement. It is also observed that the magnitude of the change in deviation is similar for all repetitions of each case in Table 1. The difference in the flow rate estimation between the different measurements of the same case was estimated to be below ±0.05%.

The mean deviation of the 1D PTM is adjusted by the modified PTM (presented methodology) which is presented in Figure. 6. The uncertainty at 95% confidence showed just for results based on standard PTM. For cases with random uncertainty of 0.6% or less, HD-HE and HA-HB, the mean deviation after applying the methodology reached a range of ±0.15% which is in the range of systematic uncertainty of the reference electromagnetic low flowmeter. For measurement between sections VA-HB, with the highest random uncertainty of 0.78% (σ_{95}=1.56%) and higher initial mean deviation of 1.3%, after applying the methodology, the mean deviation in flow measurement reached -0.4% which is well below the random uncertainty of this set of measurements. The error introduced by the elbow and reducer is outside the 95% confidence interval, further emphasising the importance of including 3D CFD.

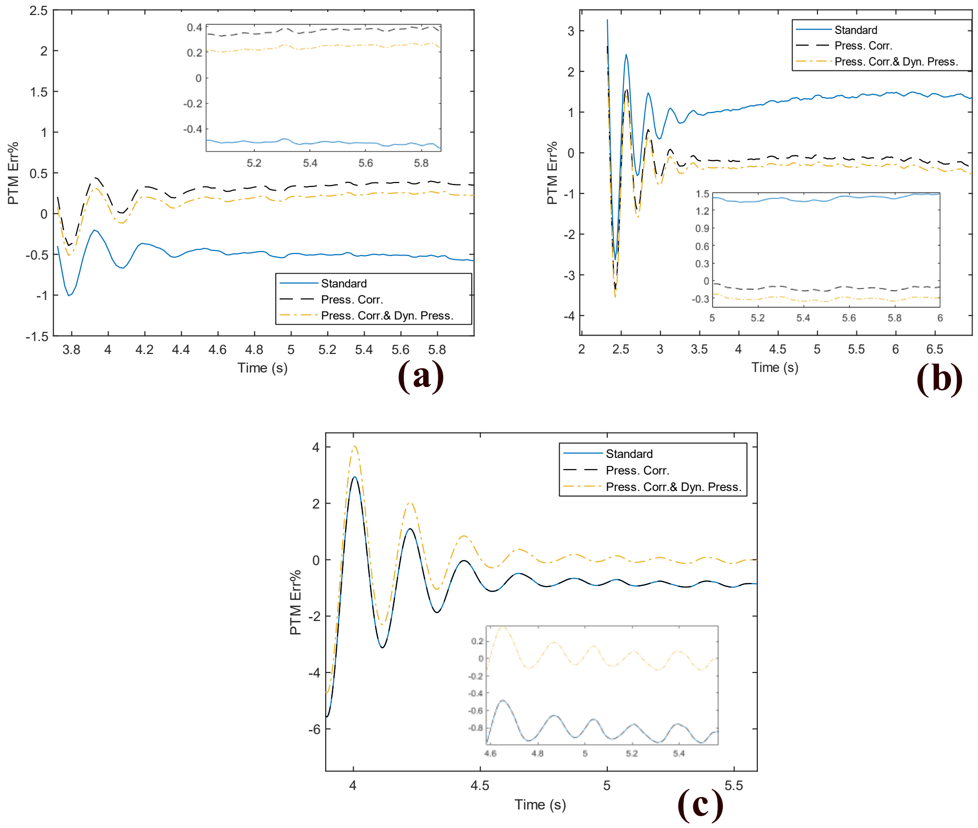

Figure. 5 Flowrate deviation compared to the reference flow meter function of endpoint for the all cases (a) HA-HB, (b) VA-HA, (c) HD-HE

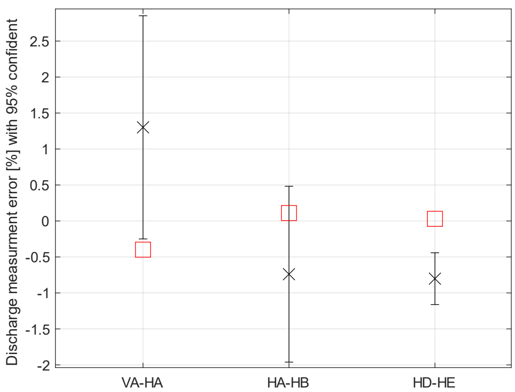

Figure. 6 Flowrate deviation for all cases; × represents the mean deviation of the experimental measurement with 1D PTM, □ represents the mean deviation with 3D CFD methodology. The bars represent the uncertainty at 95 % confidence based on standard PTM.

Conclusion

This study explores the application of the PTM beyond the limits of the IEC standard recommendations, particularly in scenarios involving a reducer and bend. Employing a hybrid approach integrating 3D CFD with the traditional 1D PTM improves method accuracy.

Initially, the 1D PTM calculates transient flow rates, serving as boundary conditions for subsequent 3D CFD simulations. Through these simulations, velocity and pressure profiles across various sections of the test rig are obtained under conditions mirroring experimental settings. Utilizing the velocity and pressure profiles, the deviation of the pressure measurement at pressure taps and kinetic energy correction factors are determined for each section, thereby refining the accuracy of the 1D PTM.

The results of this investigation underscore the methodology's capacity to elevate the precision of the method substantially. Notably, the deviation in measurements relative to the reference flow meter is mitigated to a degree commensurate with the random uncertainty inherent in measurement practices.

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