Winter-Kennedy method in hydraulic discharge measurement: Problems and Challenges

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Abstract

Winter Kennedy (WK) method is a popular way to measure the relative discharge and thus efficiency in Swedish hydropower plants. This is largely motivated by the numerous low head turbines and low cost of the method. WK is an index testing method that provides relative values of hydraulic efficiency by measuring differential pressures in one or two pairs of pressure taps in radial planes of the spiral casing. The method is described in the IEC41 standard. Despite several limitations, it is generally used to verify the increment in efficiency for refurbishment projects and sometimes for the continuous flow rate monitoring. Uncertainties in the results reaching up to 5% have been reported in different researches. Those are often attributed to a change in flow conditions after the refurbishment or in the course of time. However, a proper error analysis has not been performed yet. This paper includes a review of the available literature related to the topic to understand its problems and possible ways to investigate its limitations systematically.

Keywords: Winter-Kennedy, discharge measurement, review, limitations

1. Introduction

Discharge is the most difficult hydrodynamic parameter to assess during turbine efficiency measurement in hydropower. The knowledge of efficiency and performance are necessary for knowing the operating hill chart, fulfilling guarantees, optimizing operation [1] and moreover to confirm efficiency gain after upgrading the old plant. Swedish hydropower plants were mostly built during 1950-70s and are now undergoing major refurbishments. The number of refurbishment projects has increased due to the European Renewable Directive 2009/28/EC. The incentives for those have been further stimulated by the introduction of electric certificate system from the beginning of May 2003 and the joint Swedish-Norwegian market for electricity certificate from January 2012 [2]. The main purpose of this system is to increase the renewable energy production by 26.4 TWh in both countries by 2020 by investing in renewable energy and upgrading the old plants.

As hydropower in Sweden occupies around 41% of total electricity generation (149 TWh in 2013) [2], the electricity certificate system clearly states that hydropower refurbishments are one of the major drivers to renewable electricity. The major hydro-mechanical components during refurbishments is the replacement of runner, flow improvements and sealing of wicket gate [3]. The increment in the efficiency of old hydro turbines is therefore essential in this regard and the discharge measurement is the most challenging parameter to be measured. The shorter intakes and geometrical variation in the low head plants further complicate the discharge measurement.

Standards for the field testing and model testing of hydro turbines can be found in IEC 60041:1991 standards [4] and IEC 60193:1999 [5] respectively. Efficiency of a turbine η expressed in IEC 41 standard is calculated by:

$$\eta = P/P_h \tag{1}$$

Where P is turbine mechanical power and given by:

$$P = P_a + P_b + P_c + P_d + P_e - P_f \tag{2}$$

P_a is the generator power, P_b is the mechanical and electric losses in the generator including windage loss, P_c is the thrust bearing losses due to generator, P_d is the losses in all rotating external to the turbine and to the generator, P_e is the power supplied to any directly driven auxiliary machine and P_f is the electric power supplied to the auxiliary equipment.

Hydraulic power P_h is expressed as

$$P_h = E(\varrho Q) \pm \Delta P_h \quad (3)$$

ΔP_h is the hydraulic power correction depending on contractual definitions and local conditions. Q is the volume flow rate or discharge, ϱ is the fluid density, E is the specific hydraulic energy of the turbine and for low head the simplified relation is

$$E = \bar{g}.Z\left(1 - \frac{\varrho_a}{\bar{\varrho}}\right) + \frac{(v_1^2 - v_2^2)}{2} \quad (4)$$

Where $\bar{g}$ is local value of acceleration due to gravity, Z is the difference in elevation between two measurement points shown in figure here, $\bar{\varrho}$ given by $(\varrho_1 + \varrho_2)/2$ and ϱ_a are the density of water and air respectively, v_1 and v_2 are the mean velocities at the measurement points 1 and 2 respectively.

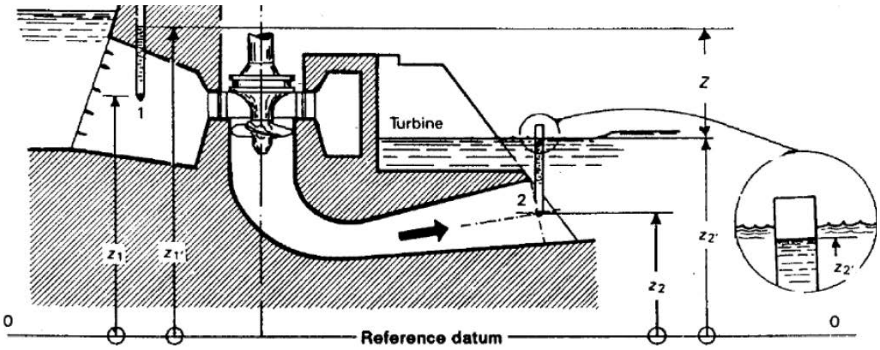

Figure 1: Measurement points for Low head turbine (Kaplan turbine) [4]

Several standards methods have been used as the absolute discharge measurement in the IEC41. Absolute measurements come with some limitations which may not be technically and economically feasible to be employed in the low head power plants. Table 1 provides information on the discharge measurement methods, its estimated cost, and development status for the low head plants (usually head under 50 m).

Table 1: Available discharge measurement method and its development status for low head plants [6]

Method	Type	Development status for low head	Estimated cost (MSEK)	Practical Uncertainty at 95% Confidence level
Winter-Kennedy	Relative	low	0.2	< ± 10% [7] [8] [9]
Pressure-time	Absolute	very low	0.2	< ± 1.4% [10]
Transit time	Absolute	average	1	< ± 0.1% [11]
Scintillation	Absolute	low	1	<± 0.5% [11]
Current meter	Absolute	Very good	1	< ± 1.2% [11]
Dilution	Absolute	Very low	0.2	< ± 3% [12] [13]
Volumetric	Absolute	Very low	0.2	< ± 1.2% [4]*
Model testing	Absolute	Very good	5	< ± 0.2% [4]**

**Uncertainty in an artificial basin according to IEC41*
*** Uncertainty could be ± 5% while scaling up from the model to prototype* [5]

There are several pros and cons of the above methods applied to the low head turbines that are described in [6]. As a quick overview, the pressure-time or Gibson method looks attractive but the understanding and experience for shorter penstock are limited. Current meters come with a higher cost, installation time and limitation like a change in flow angles due to a variable cross section of intake. Volumetric have not received much attention. Model testing could be very expensive (in order of 5 million SEK). The cost of current meters and scintillation estimated by Taylor *et al.* [14] also shows the similar as mentioned in Table 1, but higher in the case of transit time/time of flight method. Scintillation and transit time method are developing method for the low head applications. The measurement performed in Kootenay canal [15] shows transit time and scintillation predicted flow rates within about 0.1% and 0.5% respectively (also mentioned in Table 1).

As most of the turbines in the Swedish hydropower are low head machines operating below 50 m, the WK method is the most popular way for discharge measurement. The cost is very attractive with almost no downtime if the pressure sensors are already installed. However the method sometimes show large discrepancies and the fundamental understanding of the method is still limited. The present paper aims to address the theory, review of available literature and the possible ways to understand its limitations systematically.

2. Winter Kennedy method as a discharge measurement

The WK method is based on the measurement of pressure difference between 2 or 4 pressure taps located at 1 or 2 radial sections of a spiral casing. Because inconsistencies in the results are sometimes reported; they are usually not recommended for the comparative tests. The variability in the coefficient may be due to several factors such as inflow conditions, guide vane opening, design and location of taps, effect of a runner and surface roughness.

The WK method was initially described by Ireal A. Winter and A. M. Kennedy in their paper [16]. IEC 41 [4] considers this as a secondary method and can only be used as a part of field acceptance test if the method is calibrated by absolute method considered in the standard. The standard also suggested that the WK method cannot be used to check the power guarantee of the machine unless both parties agree.

2.1. Principle

The principle of this method has been extended from the flow physics in a curvilinear path and based on free vortex theory. A flow in a curved pipe is subjected to a centrifugal force and create angular momentum and causes the pressure difference between the outer and inner side of the curved pipe (or elbow). This differential pressure is used to calculate the velocities. Consider a streamline as in Figure 2 with s, n and l coordinates in flow, normal and bi-normal direction of the streamline respectively with a local radius of curvature r and tangential velocity as u_θ. Then the pressure normal to streamline is $P.ds.dl - \left(P + \frac{\partial P}{\partial n}.dn\right) ds.dl$, which equals to $-\frac{\partial P}{\partial n}.dn.ds.dl$. The Newton's second law of motion gives the force in the streamline as $\rho.dn.ds.dl.\frac{u_\theta^2}{r}$, where u_θ^2/r is the centrifugal acceleration and ρ is the fluid density. Equating the two forces the following relation is derived.

$$\frac{u_\theta^2}{r} = \frac{1}{\rho}\frac{\partial P}{\partial n} \tag{5}$$

The derivation of the Bernoulli equation yields

$$\frac{1}{\rho}\frac{\partial P}{\partial n} + u_\theta \frac{\partial u_\theta}{\partial n} = 0 \tag{6}$$

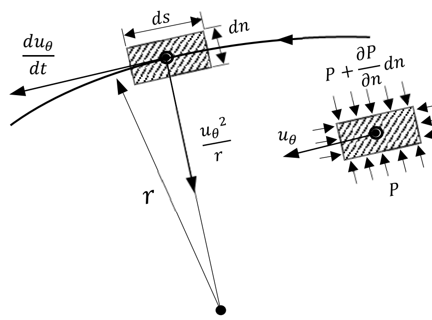

Figure 2: Streamline coordinates and forces acting

From equation (5) and (6), it gives $d(u_\theta.r) = 0$, which means the $r.u_\theta$ is constant suggesting the free vortex theory.

The derivation can also be achieved from the radial component of Navier-Stokes equation for incompressible fluid in cylindrical coordinates, given by:

$$\rho\left(\frac{\partial u_r}{\partial t} + u_r\frac{\partial u_r}{\partial r} + \frac{u_\theta}{r}\frac{\partial u_r}{\partial \theta} - \frac{u_\theta^2}{r} + u_z\frac{\partial u_r}{\partial z}\right) = -\frac{\partial P}{\partial r} + \rho g_r + \mu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_r}{\partial r}\right) - \frac{u_r}{r^2} + \frac{1}{r^2}\frac{\partial^2 u_r}{\partial \theta^2} - \frac{2}{r^2}\frac{\partial u_\theta}{\partial \theta} + \frac{\partial^2 u_r}{\partial z^2}\right] \tag{7}$$

u_r, u_θ and u_z are radial, tangential and axial velocity components respectively and g_r is the body acceleration. Considering inviscid steady flow and also assuming the negligible radial and vertical components, the equation reduces to equation (5).

Then integrating equation (5) from $r1$ to 2 ,

$$\int_{r1}^{r2} \frac{u_\theta^2}{r} dr = \frac{1}{\rho} \int_{P1}^{P2} dp \tag{8}$$

And considering u_θ constant across the considered section with area A, and volumetric flow rate $Q = u_\theta A$,

$$Q = K \times \sqrt{\Delta P}, where\ K = A \Big/ \sqrt{\rho \ln \frac{r_2}{r_1}} \tag{9}$$

IEC 41 standard mention the above equation as $Q = K \times \Delta P^n$, where the value of exponent n have a range between 0.48 and 0.52. The flow coefficient K is generally determined by model testing or calibrating against the absolute method. The differential pressure measurement is done between 1 or 2 pairs of pressure taps located at 1 or 2 radial sections of a spiral casing. IEC 41 states the outer tap to be located at the outer side of the spiral whereas the inner tap shall be located outside of the stay vanes on a flow line passing midway between the two adjacent stay vanes. The standard also recommends using the other pair of pressure taps in another radial section. The spiral with the WK method is shown in Figure 3.

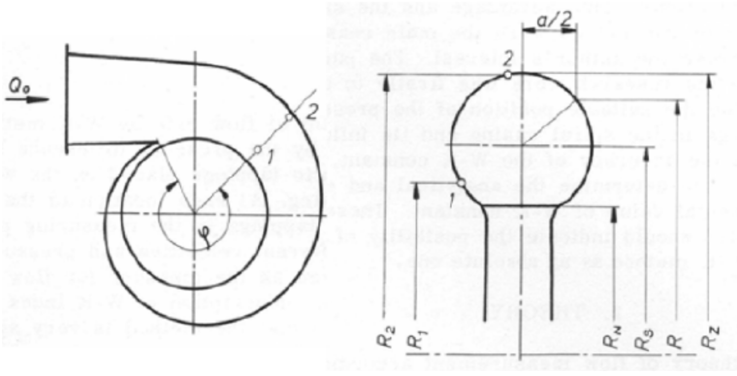

Figure 3: location of pressure taps 1 and 2 shown for WK method [4]

2.2. Practical methods and alternative considerations

The piezometer(s) are actually measuring the relative weight flow rate (product of the specific weight of fluid and discharge) of the fluid. Therefore, the pressure transducers are to be calibrated at the lab using same specific weight as that in the spiral casing and the transducer lines should be bled to remove any gas bubbles [17] [18]. The WK method applied in the field investigations have been reported by several researchers [17] [19] [8]. There are two methods of calibrating WK coefficients [17]. The first one is the single point method where the prototype flow value from a model test at the peak efficiency point is equated to the square root of the differential pressure at peak relative efficiency. The exponent n is exactly kept 0.5, and K is determined. The second method use multiple points, here the absolute flow rates are measured simultaneously with WK pressure differentials, then the plot for ΔP versus Q are curve fitted generally with the least square method. It is also common to use a log-log plot for ΔP versus Q forming a straight line in the slope-intercept form. For log-log plot, with base of 10, $\log_{10} Q = n \log_{10} \Delta P + b$ and using log identity as $\log_{10} Q = \log_{10} \Delta P^n + b$, or $10^{\log Q} = 10^{\log \Delta P^n + b}$, the relation reduces to,

$$Q = 10^b \Delta P^n \text{ Or } Q = k\Delta P^n \tag{10}$$

A new method of calibrating the above equation was derived by Sheldon [17] in which the exponent n will no longer be constant but varies with the relation $Q = k\Delta P^{n+a(\log \Delta P)}$, where n is near to 0.5 and a is the coefficient of the second order term from the calibration second order equation $Q = a(\log^2 \Delta P) + n(\log \Delta P) + b$. This non-linear is because the exponent of the differential pressure varies with the flow rate. Nicolle and Proulx [9] also made modification of the equation in the coefficient K, where K was the function of guide vane opening, but the exponent n was kept constant to square root in this case. But the modification was based on the physical results rather mathematical modification. An alternative fitting procedure if the WK method is used as transfer between absolute discharge measurements is given by [20] where the author introduces the term to remove the non-linearity error and to be used with two discharge measurement data sets. Given in the form of $Q = aD^{0.5} + b$, (instead of $Q = KD^n$) where a is proportional bias and b refers to zero offset constant which accounts for changing flow regimes as the flow approaches zero. Flow can approach zero conditions when there is flow separation near the taps.

Numerical simulation is also used to calibrate the WK coefficients, as in [21] where the experimental results from thermodynamic method and WK method have been used to validate the numerical results. The calibration coefficient *K* and exponent *n* are determined by non-linear curve fitting in numerical simulations results.

IEC60193 [5] states in the subclause 4.8.1 that "*the index test in the model can never be a substitute for an absolute discharge measurement at the prototype*". Even under favorable conditions, the uncertainty in the discharge measurement at the prototype using the calibrated *k* from the model tests can show about ± 5%. This is because the coefficient *k* is a function of flow condition, Reynolds numbers and wall roughness, which are not constant between model and prototype. On the other hand, calibrating on the prototype with some absolute measurement technique, and varying the exponent *n* poses question towards the formulation of the WK method, whereas a specific k, calibrated under certain conditions, might not be valid anymore if the conditions are changed.

3. Previous investigations and the problems with the WK method

Since WK method is widely used in refurbishment projects, it is common to compare efficiency after replacing the old runner with the new one in the hydro turbine. This is generally done by calibrating the constants of WK equation with the old runner and using the same values after refurbishment. Usually an increase in efficiency is expected but sometimes low improvement value or even negative results have been reported which question the use of the old calibration values into the new one.

The intake flow conditions for the low head plants affects the WK method. The vortex could be generated from the power plant design or inflow conditions [1] and affects the WK measurements as the flow condition is changed in the spiral case making it unreliable. The WK predicts really good in favorable conditions as in Hulaas *et al.* [10], with net head of 52 m, 14 MW vertical Francis turbine and using exponent value of 0.5, the results fits well with the absolute efficiency tests (thermodynamic and pressure-time) with difference of only -0.59 (min) and 0.11 (max) percentage point. In double regulated machine like Kaplan turbine, the off cam set up could produce deviations in the WK measurements as in Topham *et al*. [22]. Sometimes the method can have larger error around 3.7% (compared with acoustic scintillation flow meter) [19]. The authors in [9] demonstrated that the flow homology conditions cannot be always achieved, so the better way is to have larger pressure difference measurement and calibrate it. The numerical studies found that the flow distribution is changing with changing the GV openings/angles as shown in Figure 4. The pressure taps placed at different locations have shown different characteristics (see figure 4) to guide vane opening and adjacent unit operation. The result could even have up to 5.4% error. Hence, the author proposed a new method where the index constant *K* would no longer be constant and vary with GV opening. This method developed at Hydro-Quebec as described in [9] [23] [24] is also used for online flow monitoring and the experiments on 200 MW Francis turbine and 110 MW propeller turbine have shown good agreement, even when head changes.

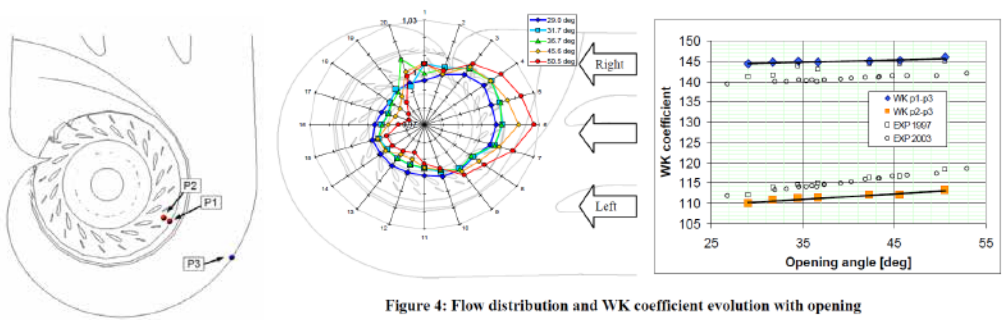

Figure 4: Influence of guide vane angle opening in WK coefficient [9]

Andersson *et al*. [7] investigated the effect caused by a well-defined skew inflow in the WK measurements. The inlet velocity profile was skewed by sieve plates and the pitot tubes were used to measure this profile. The authors reported up to 10% deviation in WK pressure measurements due to this skewness. WK differential pressure is found to have discrepancies even in two identical units. Rau and Eissner [8] investigated the WK method in two identical units and the pressure-time method was used to calibrate WK coefficients of that turbine. Although the same coefficients were used to calibrate the WK method to the other unit, however, the efficiency curve could not be reproduced even though the units were identical (0.8% discrepancy in higher loads). Here it should be noted that the identical units are difficult to obtain because there are always slight variations in the two units. It is because each unit involves some manual welding, grinding and thermal process which affect the final geometry.

Even the WK coefficients calibrated from the model test show discrepancy in the efficiency curve while applied to the prototype (1% discrepancy) at higher load (Figure 5 left). The authors recommend not to use WK method in the comparative test as the behavior is not well reproduced.

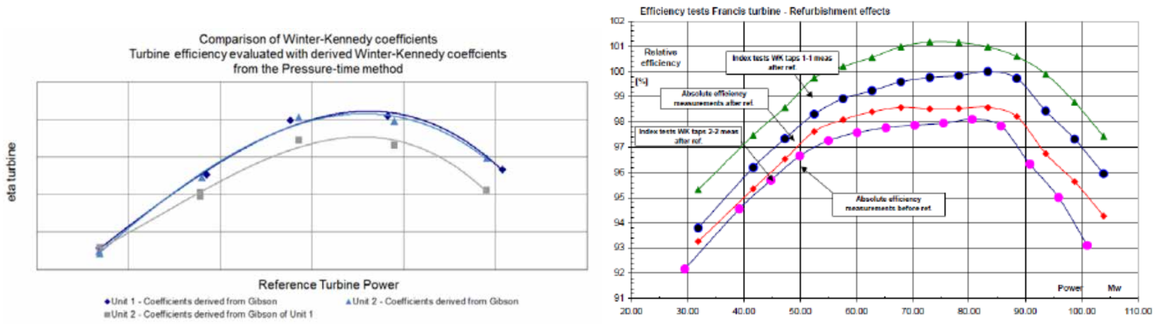

Figure 5: Winter Kennedy measurement discrepancies: *left*- WK compare with the Pressure time [8], *right*-refurbishment effects in WK measurements [25]

The two sets of WK taps installed could give different results as in Fabio and Ramdall [25]. There study shows higher differential taps resulted 4% increase in efficiency after refurbishment (but same runner design) and lower differential taps measured only 1% increment in efficiency, whereas pressure time method shows 2% increment (see Figure 5 right). The discrepancies in results have been related to the corrosion inside the spiral casing.

Several numerical investigations have been performed to analyze the flow phenomena due to intake variations [9] [26] [27] [21]. The wakes resulted from the trash racks were seen to affect both mean velocity and turbulence even at the 5 m downstream measurements and gave discrepancies in the results of different methods [26]. The Figure 6 shows the mean horizontal velocity component distribution in the middle cross section of the intake. The velocity profiles measured and calculated matched along the elevation but seen some fluctuations in the flow angle which have resulted from the turbulence produced from the racks or the sediment/debris deposited at the bottoms over the years.

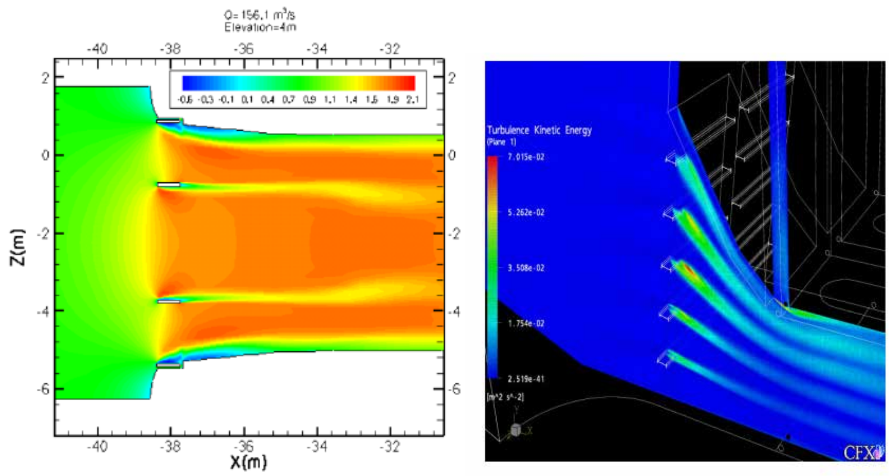

Figure 6: velocity component distribution due to trash racks [26] and turbulent kinetic energy distribution after trash rack [27]

The different results with the WK method are believed to be influence by hydraulic boundary conditions, design and location of pressure taps, surface roughness and local flow disturbances as well as air pockets in the pressure pipes. It has been observed that the WK method can give sufficient result in favorable conditions but sometimes can produce totally unreliable results. The method is widely employed in comparative tests during the refurbishments. However many studies pointed out that the method should not be used during refurbishments for comparisons since the flow conditions inside spiral casing might change.

4. Systematic error analysis for WK method

Though there have been several experimental and numerical investigations in the WK method in the past decades, systematic error analysis of this method has still not been done. The following are necessary to be taken into consideration for the systematic error analysis of this method:

4.1. Flow in a curved pipe strongly depends on inflow conditions

The formulation of WK method is based on the free vortex theory. The velocity streamlines in the spiral casing are assumed here as irrotational. In other words, the flow is moving in a circular path in such a way that the flow do not rotate in their own center but just follow a circular path. The spiral case can be considered as a curved pipe and earlier investigations [28] [29] including numerous secondary flow investigations [30] [31] [32] [33] [34] inside the curve pipe/tube have been done since the Dean's first investigations [35] [36]. Dean [35] showed the secondary flow in the cross-section plane (so-called Dean Vortices) of the pipe decreases the flow rate produced by a given pressure gradient and the streamlines pattern was found to be symmetrical inside and outside of the bend. The dean vortices are also reported in a spiral case by Mulu and Cervantes in [37] . The pattern of this streamlines creating Dean vortices could be different in higher Dean's number D as investigated by McConalogue and Srivastava [28], who showed for larger D, ($D\sim600$), radial pressure gradient opposes the distribution of centrifugal force and can create approximately uniform secondary flow. The similar results were also reported by [30] [32], among others. The shifting of maximum axial peak towards the wall increases the viscous rate of dissipation due to shear [28]. This effect reduces the flow rate in the curved pipe compared to the straight pipe with the similar configuration. The nature and strength of secondary flow are influenced by the initial inlet flow conditions [30] [38], consequently, the wall static pressure may vary. Figure 7 shows the variation of constant axial velocity for the pipe with bend radius degrees resulting from the different inlet velocity distribution.

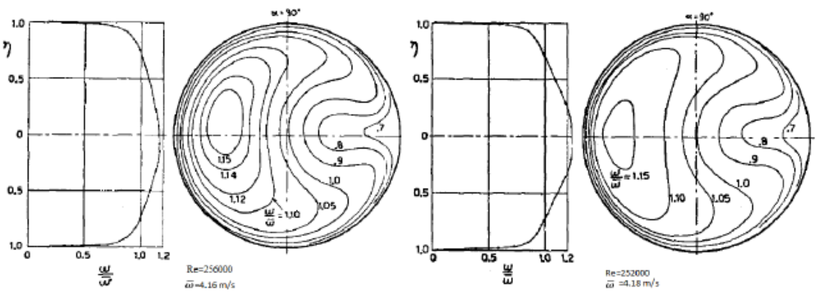

Figure 7: Effect of inlet velocity profile (secondary flow) in velocity distribution in sectional plane of curved pipe. [30]

The flow in the spiral case (for free vortex type) is axisymmetric if it is well designed. But the secondary flows can be induced near the walls and the flow can also be distorted near the inlet of stay vanes [39]. A Laser Doppler Velocimetry LDV measurement of velocity components in spiral casing was also performed by Nilsson *et al*. [40] and Mulu and Cervantes [37]. Previous LDA measurements [37] show that the tangential velocity is higher in the inner region (entrance to the stay ring) and that the turbulent intensity is higher in near wall regions. The simulations also show good agreement with measurements [41] [42]. The results reported in [37] show that the flow can be turbulent in the spiral casing when there is a bend in upstream which creates large recirculation. At the beginning of the case (at SI in Figure 8) the maximum tangential velocity is at the bottom region of the SC and it decreases toward the middle height of the guide vanes. However, in the inner section (SII in Figure 8), the maximum tangential velocity region is located somewhere between the central level of guide vane and upper level of the leading edge of guide vanes or stay vanes.

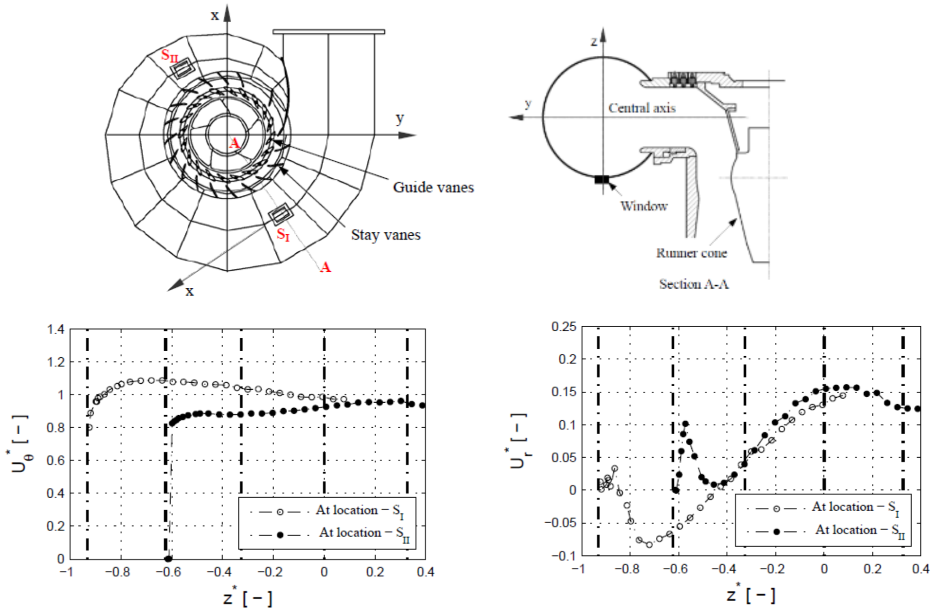

Figure 8: Tangential and radial velocity from LDA measurement from [37] in the spiral case of Kaplan turbine.

This measurement clearly illustrates that the flow is evolving in spiral case is due to the inflow conditions. The distortion of inlet velocity by implementing well-defined skew inflow condition was also confirmed by the recent study by Andersson *et al.* [7] in a spiral case, where they resulted in a deviation of WK pressure measurement that could be up to 10%. Geberkiden [43] investigation also shows a secondary flow in the penstock when there is a curved section upstream. Recent numerical studies by Nakkina *et al.* [44] conducted on several spiral casing designs also shows that the twin vortices due to secondary flow emerge as in earlier the curved pipes investigation, but their strength is decreasing from one section to another. All these investigations show that the secondary flow can emerge due to upstream conditions in the spiral case and thus influence the pressure measurement.

4.2. *Local flow disturbances or flow separation can occur due to surface roughness or inlet conditions near the pressure taps*

The advanced pressure sensors available today can accurately measure within the very low tolerance. However, the measurement can be greatly influenced by the local flow disturbances near the tap. The pressure at the wall can be influenced by the following variables [45].

$$\Pi = f\left(\frac{d_s u_\tau}{\nu}, \frac{d_s}{D}, M, \frac{l_s}{d_s}, \frac{d_c}{d_s}, \frac{\epsilon}{d_s}\right) \tag{11}$$

d_s is the diameter of the tap, u_τ is the friction velocity given by $\sqrt{\tau_\omega/\rho}$, τ_ω is the wall shear stress, ρ is the fluid density, M is the Mach number, l_s is the depth of the tap (orifice), d_c is the cavity behind the orifice, ϵ is the RMS of burrs on the edge of the tap orifice, ν is the kinematic viscosity of the fluid. The local flow at the tap can get complex with flow distortion and creation of cavity vortices (which can result in higher pressure measurement [46]). The local surface irregularity in spiral case can also be related to the surface roughness due to erosion and wear over time. Though there has been numerous researches regarding local flow disturbances due to a surface roughness in turbulence flows, it is essential to understand this phenomenon around the taps in a spiral case.

4.3. Downstream changes can influence WK measurement

The WK measurement is seen to be affected by replacing the runner. The experiment conducted by Lövgren and Cervantes [47] for low head Kaplan model turbine resulted in 2% difference in flow rate estimation using WK method. The constant for the new runner was based on the calibration constants of the old runner. Rau and Eissner [8] have also reported the discrepancy that could reach 1% at higher loads.

In some cases the pressure waves from the rotor-stator interactions could propagate into the spiral casing [48]. A bad design of spiral casing can produce asymmetrical load distribution [49] in the runner, which in turn could affect the unsteady flow phenomenon in the casing. Pressure measurements in the spiral casing of a Kaplan model turbine by Jonsson and Cervantes [50] show the runner frequency is noticed at casing as one of the dominating frequencies in the best efficiency point (as well as at high load) whereas the rotating vortex rope appears at part load. The acoustic propagation of this vortex rope in the spiral case had about ~1/5 of the measured amplitude in the draft tube.

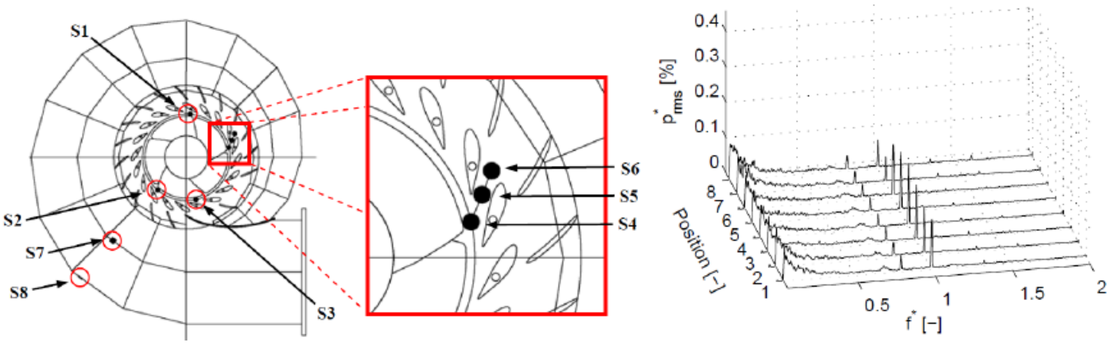

Figure 9: Amplitude spectrum at spiral case *(right)* for the sensors mounted in casing *(left)* [50]

Figure 9 shows the frequency spectrum measured at the sensors positioned in spiral casing. The frequency $1.f^*$ corresponds to the runner frequency and it is noticed that this frequency is dominant in all the sensors located in the case. The pressure amplitude is larger in the inner sensors (S1 to S6), whereas smaller at the outer wall of the case (i.e. S7 and S8 in Figure 9).

5. Summary

Winter-Kennedy is a relative method for low head hydraulic turbine discharge measurement that is very attractive because of its low cost and time requirements. The method is based on differential pressure measurement with 1 or 2 pairs of pressure transducers at the radial sections of the spiral casing. The method is very popular in low head plants since the short and sometimes complex intake geometry pose limitation to the other absolute flow measurement methods. As this method is comparatively cheap and easy to implement, the use of this method is seen promising in future too. The WK method is seen to be favorable if the calibration is performed in the prototype itself and the flow conditions are unchanged. Indexing from model test can never be a substitute for the absolute discharge measurement of the prototype.

The WK method of discharge was widely investigated in the past two decades and results have shown discrepancies. Errors up to 10% have been reported. Many researchers also mentioned that using WK coefficients calibrated with the old runner for a new runner is not recommended as the flow physics changes. The downstream influence like a change in guide vane angles, runner change or even vortex rope breakdown can also introduce flow change and wave propagation in the spiral casing. It is also interesting to mention how the secondary flow and flow separation could occur from the upstream influence. For instance, intake pipe bend, bifurcations and operation of adjacent units can all alter the velocity profile and hence the pressure distribution in the cross-sectional plane of the spiral case. Moreover, the local flow disturbances due to surface roughness, eroded inner surfaces or incorrect installation of pressure sensors can also explain some of the discrepancies shown by this method.

This literature survey deduces that it is crucial to understand the fundamental flow phenomena in spiral case. The further understanding of flow physics inside spiral casing can give the possible explanation to the error and uncertainties associated with this method. Therefore, a systematic error analysis and reporting of this method are to be developed with the following parameters under investigation:

- Velocity field and pressure measurements in selected section of spiral case near the WK pressure taps through optical measurement techniques like LDV while altering the inflow conditions. The inflow conditions should be affected by changing velocity profiles, introducing oscillating flow and at different loads (part load, best efficient point, and high load) or through some mechanism with valve opening or closing.
- Influence of roughness in local flow disturbances near the taps.
- Change in downstream geometry like runner, guide vane angle and clearance gap (for Francis type) or changing blade pitch (propeller turbine), and investigate the phase-resolved velocity measurement that can provide detail information regarding the main cause of the erroneous results.

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