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Computational analysis of part-load flow control for crossflow hydro-turbines

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ABSTRACT

Crossflow turbines are commonly used in low-head small hydro systems, typically less than 50 m, and produce power up to a few hundred kWs. A crossflow turbine is a unique type of hydro-turbine in that the flow passes twice through the runner. Therefore, the power is extracted in two stages. These turbines often employ a guide vane in the nozzle for controlling the flow entering the runner. However, a guide vane significantly reduces the quality of the entry flow by splitting it into two jets and producing non-uniform entry flow angles which can cause a serious loss in turbine efficiency. Adhikari and Wood (2017) and Adhikari and Wood (2018) showed that well designed crossflow turbines without a guide vane can achieve full-load efficiencies of over 90%. This leaves open the issue of maintaining high efficiency at part-load, defined as reduced flow at constant head. As for other hydro-turbine types, efficient part-load operation cannot be achieved solely by employing power electronics to reduce the shaft speed as the flow decreases. This paper analyzes a slider or "Cink" control device at the entry to the runner to reduce the entry arc, i.e. the angular extent of the runner entry, as the flow rate decreases, (Sinagra et al., 2014). We show that a properly operated slider will maintain the radial and azimuthal entry velocities at their full-load values and therefore will not change the optimum shaft speed as the flow rate changes. This simplifies the power electronics required for the turbine generator. Three-dimensional Reynolds-Averaged Navier-Stokes simulations of a 0.53 kW turbine with 88% full-load efficiency showed that the slider maintains high efficiency at part-flow conditions. We conclude that the slider shows significant advantages over a guide vane as the primary means of crossflow turbine control at part-load.

Introduction

Hydropower is the most efficient means of generating reliable renewable energy in many parts of the world where good hydro resources are available, and is expected to continue to be an important part of the sustainable energy systems in the future. Small-scale hydropower systems, which typically operate at head, $H < 50$ m, and produce power from a few kWs to a few hundred kWs, are popular in many remote locations around the world (Hirmer & Cruickshank, 2014; Paish, 2002), especially in those areas where grid extension is too expensive. Such systems improve the living standards of rural people and support development services such as education and healthcare. The primary obstacles to the further adoption of small-scale systems are cost and long-term sustainability. Many small hydropower systems employ crossflow turbines, for example in Nepal where they are also manufactured. These turbines are extremely simple to design and manufacture locally at low cost but suffer from lower efficiency. Current turbine designs have about 70–85% efficiency (Acharya, Kim, Thapa, & Lee, 2015; Choi, Lim, Kim, & Lee, 2008; Sammartano, Morreale, Sinagra, & Tucciarelli, 2016; Sinagra, Sammartano, Aricò, Collura, & Tucciarelli, 2014), whereas the most efficient turbine designs, such as Francis and Pelton, can easily achieve 90% efficiency (Dixon & Hall, 2013; Elbatran, Yaakob, Ahmed, & Shabara, 2015). Improvements in the maximum efficiency would lower the cost of crossflow turbines and improve overall sustainability. Desai (1993), Totapally and Aziz (1994), Adhikari and Wood (2017) and Adhikari and Wood (2018) showed that full-load efficiency of 90% could be obtained from well designed crossflow turbines without a guide vane. This leaves open the issue of maintaining high efficiency at part-load, defined as reduced flow at constant head. As for other hydro-turbine types, efficient part-load operation cannot be achieved solely by employing power electronics to reduce the shaft speed as the flow decreases. Most crossflow turbines employ a guide vane in the nozzle for controlling the flow entering the runner as shown in Fig. 1. However, a guide vane significantly reduces the quality of the entry flow by splitting it into two jets and producing

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Nomenclature

β_1	runner entry flow angle (rad or degrees)
β_{1b}	outer blade angle (rad or degrees)
H	turbine head (m)
h_0	nozzle throat (m)
$h(\theta)$	nozzle rear wall from the runner tip, $R(\theta) - R_1$, (m)
N_b	number of blades
Q	flow rate through turbine (lps)
Q_{max}	maximum(design) flow rate (lps)
$R(\theta)$	radius of nozzle rear-wall (m)
R_1	outer radius of the runner (m)
R_2	inner radius of the runner (m)
t	blade thickness (mm)
u_0	nozzle inlet velocity (m/s)
u_t	total velocity at runner entry (m/s)
u_r	radial velocity (m/s)
W	nozzle and runner width (m)
$\dot{W}$	turbine power (kW)
ω	runner angular speed (m/s)
θ_s	entry arc (rad or degrees)
η	turbine efficiency (%)

non-uniform entry flow angles which can cause a serious loss in turbine efficiency. Current literature presented in Adhikari and Wood (2018) revealed that turbines designed with a guide vane have a maximum efficiency of less than 85%.

In this paper, we present a computational analysis of a slider or "Cink" control device in the nozzle to reduce the entry arc θ_s – the angular extent of the runner entry – as the flow rate decreases (Sinagra et al., 2014). To assess the performance of slider control, a systematic study was conducted on the 88% efficient turbine using the 2-D analytical model of Adhikari and Wood (2017) for setting the slider position as the flow rate, Q, varied. Three-dimensional Reynolds-averaged Navier-Stokes (RANS) simulations were used for evaluating the influence of flow control on turbine performance.

As shown in Fig. 1, the crossflow turbine comprises two main components, a stationary nozzle and a rotating runner. The nozzle accelerates the inlet flow and directs it at the runner entry at angle β_1. For maximum efficiency, the nozzle should also convert all the available head, H, into kinetic energy at entry to the runner, (Adhikari & Wood, 2017) , which may be difficult to achieve as Q falls from the maximum value, Q_{max}. In this study, we will use "inlet" to describe the flow anywhere in the nozzle and "entry" for the flow as it passes from the inlet to the runner. A unique feature of the crossflow turbine is that flow passes twice through the rotating runner. The flow enters the so-called the first stage, traverses the central air-space region, and exits through the second stage. Often the first stage does not convert all the available energy into power (Adhikari & Wood, 2018; Choi et al., 2008). For maximum efficiency, β_1 should match the outer blade angle, β_{1b}, of the runner taking into account the transfer from stationary to rotating co-ordinates. Uniformity of the entry flow is also important because it directly affects the performance of the runner, which, in turn, must be designed to extract the maximum amount of angular momentum (Adhikari & Wood, 2017). Adhikari and Wood (2017) derived analytical expressions for the nozzle shape and the radial and angular velocities at runner entry. The design of the runner to match these entry velocities and to achieve high efficiency is described in Adhikari and Wood (2018) .

During part-flow operation without flow control, i.e. unchanged H at reduced Q, the total velocity and β_1 of the flow entering the

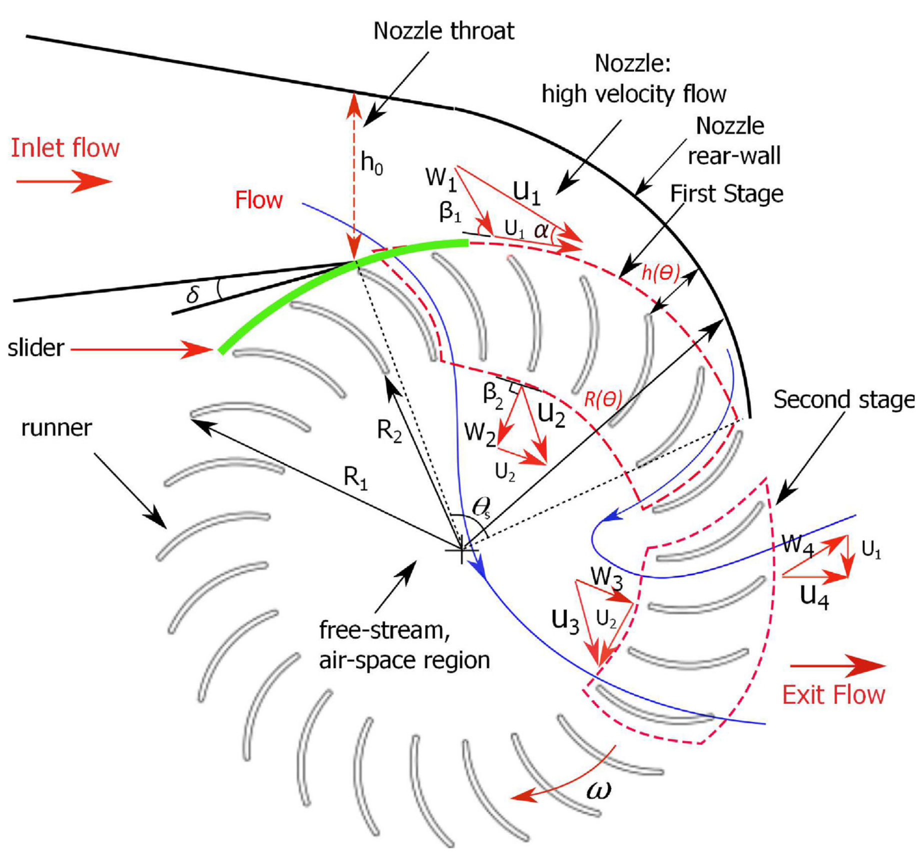

Fig. 1. Schematic illustration of key geometric parameters of crossflow turbine with a slider mechanism indicated in green. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)