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Hydro turbine prototype testing and generation of performance curves: Fully automated approach

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ABSTRACT

This paper presents a technology that can accelerate the development of hydro turbines by fully automating the initial testing process of prototype turbine models and automatically converting the acquired data into efficiency hill charts that allow straight forward comparison of prototypes' performance. The testing procedure of both reaction and impulse turbines is illustrated using models of Francis and Pelton turbines respectively. For the development of an appropriate hill chart containing no less than 780 points the average duration of the fully automated test is 4 h while the acquired data files can be processed into descriptive standard efficiency hill charts within less than a minute. These hill charts can then be used in research and development to quickly evaluate and compare the performance of initial turbine prototype designs before proceeding to much lengthier and more expensive development stage of the chosen design.

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1. Introduction

In a number of countries around the world including the UK, the constant increase in fossil fuel energy prices together with a need to improve their energy security by reducing the dependency on imported fuel supplies is boosting the development of renewable energy technologies [1–3]. The UK has an estimated untapped green-field small scale (capped to 10 MW) hydropower capacity of 1.5 GWs [4]. This paper describes the automation of a manual turbine prototype testing facility (Fig. 1) manufactured by Gilbert Gilkes & Gordon Ltd to enable very fast data acquisition and processing into turbine efficiency hill charts.

Ability to quickly assess the performance of a prototype turbine at the initial stage of development is very important before moving towards more accurate but much more time consuming and expensive development phase. Depending on the conditions of a particular application, different types of turbines are used [5] and therefore different issues are to be addressed. Even though for modelling of reaction turbines Computational Fluid Dynamics has reached a feasible stage [6], numerical modelling of impulse turbines (like Pelton [7] or Turgo [8]) is still a challenge. When modelling a full geometry, time durations of up to 5 days per simulation of one data point are reported [9,10]. Alternatively to reduce the time duration severe simplifications of geometry [11–15] or turbine working principle [16–19] are implied. On the other hand, experimental model tests that use runner dimensions and flow conditions that allow full scalability and ensure very high accuracy are usually performed only as the last stage of development because of its complexity and cost. That is why quickly pre-testing of prototype turbine models might aid the research and development process overall.

2. Background

The guaranteed efficiency of a turbine (Eq. (1)) as defined by the International Code for Model Acceptance Tests IEC 60193:1999 [20] is the ratio of the mechanical power provided by a shaft of a turbine (Eq. (2)) to the power generator divided by the hydraulic power (Eq. (3)):

$$\eta = \frac{P_{\mathrm{m}}}{P_{\mathrm{h}}} \tag{1}$$

where η is the guaranteed efficiency of a turbine, P_{m} is the mechanical power [J], P_{h} is the hydraulic power [J],

$$P_{\mathrm{m}} = \omega \cdot T \tag{2}$$

where ω is the rotational speed of a turbine shaft [rad/s], T is the torque provided by the turbine shaft [Nm],

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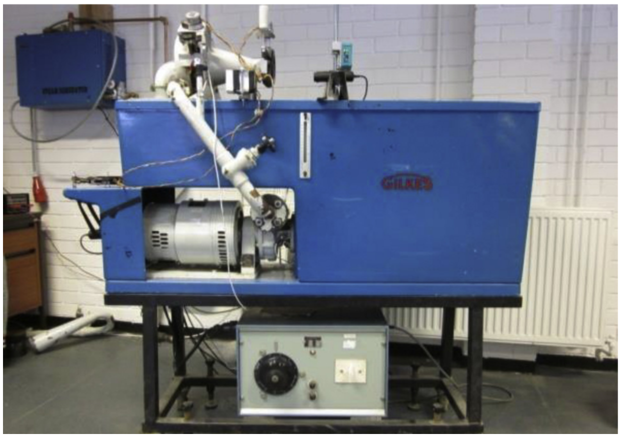

Fig. 1. Fully automated turbine prototype tester.

$$P_h = \rho Q \cdot g \Delta H \tag{3}$$

where ρ is the density of water [kg/m^3], Q is the volumetric flow rate [m^3/s], g is the gravitational acceleration constant taken as 9.81 m/s^2, ΔH is the net pressure head [m].

Calculating the mechanical power provided by the shaft is trivial, however finding the hydraulic power is more complicated as the net pressure head ΔH consists of more than one component [21], i.e. the gross pressure head, the head of pressure loss in a penstock and the velocity head. Moreover, the components differ when calculating the net head for impulse or reaction type turbines. Fig. 2 presents general schematics of a hydropower plant.

First of all, it is important to understand how the gross pressure head is measured in each case. The gross pressure head for reaction turbines is simply the difference between the upstream and the downstream water levels. However, for impulse turbines the gross pressure head is measured as the distance between the upstream water level and the level of a jet impact point which is always higher than the downstream water level. Equations (4) and (5) show how the net pressure head is calculated for impulse and reaction turbines respectively and what terms are important for each of them:

$$\text{Impulse turbines } \Delta H = H_G - H_L \tag{4}$$

$$\text{Reaction turbines } \Delta H = H_G - H_L - H_V \tag{5}$$

where H_G is the gross pressure head [m], H_L is the friction loss head [m] and H_V is the velocity head [m].

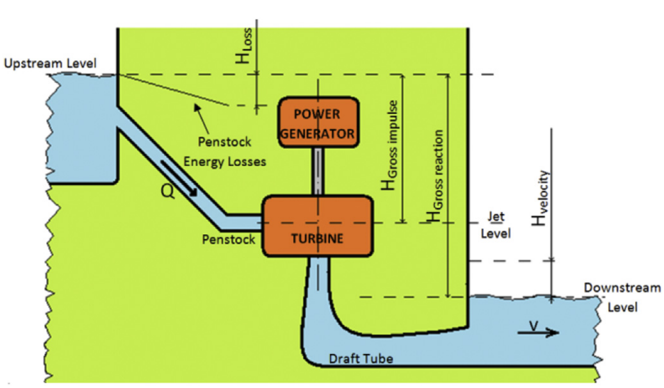

Fig. 2. Schematics of a hydropower station.

Velocity pressure head is used only when calculating the net pressure head on reaction turbines as the downstream water flow does not affect the performance of the impulse turbines. Physically the velocity head and the friction loss head are not distances measured from water levels; however they can be converted into adequate quantities expressed in metres (Eq. (6)) and then for the sake of convenience sketched on the diagram as shown in Fig. 2.

$$H_V = \frac{v^2}{2g} \tag{6}$$

where v is the mean velocity of downstream water flow [m/s].

3. Affinity laws

When designing a hydropower station it is possible to calculate the performance of a known turbine design if the performance of its model is known. Performance scaling can be done by using the affinity or so called similarity laws [22–24]. The affinity laws mathematically relate the same turbine at different speeds or geometrically similar turbines at the same speed. Equations (7)–(9) show the relationships when the diameter of the runner is kept constant, whereas Equations (10)–(12) are used when the rotational speed is constant:

$$\text{if } D = \text{const. } \frac{Q_1}{Q_2} = \frac{n_1}{n_2} \tag{7}$$

$$\frac{\Delta H_1}{\Delta H_2} = \left(\frac{n_1}{n_2}\right)^2 \tag{8}$$

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{9}$$

where n is the rotational speed [rpm],

$$\text{if } n = \text{const. } \frac{Q_1}{Q_2} = \frac{D_1}{D_2} \tag{10}$$

$$\frac{\Delta H_1}{\Delta H_2} = \left(\frac{D_1}{D_2}\right)^2 \tag{11}$$

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^3 \tag{12}$$

where D is the runner diameter [m].

In general, these laws are expressed as:

$$\frac{Q_t}{Q_m} = \frac{\sqrt{\Delta H_t}}{\sqrt{\Delta H_m}} \cdot \frac{D_t^2}{D_m^2} \tag{13}$$

$$\frac{n_t}{n_m} = \frac{\sqrt{\Delta H_t}}{\sqrt{\Delta H_m}} \cdot \frac{D_m}{D_t} \tag{14}$$

where indexes t correspond to the industrial turbine and m to the laboratory model. Therefore by choosing $\Delta H_t = 1$ m and $D_t = 1$ m and rearranging Equations (13) and (14) to express Q_m and n_m, the formulae to calculate quantities known as the unit speed n_{11} [rpm] and the unit (specific) discharge Q_{11} [m^3/s] are derived:

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