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Flow control in Banki turbines

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Abstract

Cross flow micro turbines are environmentally friendly equipment. A low head Banki turbine is studied in this paper numerically and experimentally in order to identify the reasons why the turbine efficiency is poor at low rotational speeds. A procedure that consists in dividing the flow domain into different zones is proposed for clarifying the main issues. The turbine efficiency obtained experimentally is explained based on a CFD analysis that highlights the influence of recirculation phenomena and shock losses produced when the water jet impinges on the shaft. This analysis is also used to propose a device for internal flow control.

Keywords: Banki turbine; fluid flow control; hydro system; numerical simulation; renewable energy sources

1. Introduction

Hydropower is a renewable, low polluting energy source that produces electrical energy with efficient equipment, requires cheap operating and maintenance costs and offers reliable and flexible operation. Production in large hydropower plants needs large investments and can cause negative social and environmental effects, including interference with fish migration. By harvesting low hydro potential of rivers, flooding of large land areas and dramatic changes of aquatic ecosystems can be avoided. Besides, small reservoirs and run of the river hydro systems are sustainable development options, provided that the hydropower system is well designed and constructed.

The definition of small hydropower varies around the world. International Energy Agency (IEA) states that it

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should be used for up to 10 MW [1]. A classification by subdivisions mentions that mini hydro refers to hydroelectric power plants below 2 MW, micro-hydro below 500 kW, and pico-hydro below 10 kW [2].

In the main scenario, World Energy Outlook 2016, published by IEA, forecasts that about 60% of all new power generation capacity by 2040 would come from renewable sources. The Directive 2009/28/EC of the European Parliament and of the Council, from 23 April 2009, on the promotion of the use of energy from renewable sources [3] encourages electricity generation from local energy sources. Implementation of EU energy policies, irrespective of the size of producer, brings into focus pico-hydro turbines as an alternative source of renewable energy to be used in remote locations as standalone installations or as part of hybrid solar-wind-hydro systems.

Banki turbines are recommended in these cases, due to their advantages: flat efficiency curve under varying load, self cleaning capability, simple and cheap construction. In a Banki turbine, the water passes through the blade channels twice, first from outside to inside and then back to outside, following a roughly transverse path inside the runner. Hence, the turbine is considered to be a two stage one and is called cross flow turbine. Its runner is foreseen with blades only at its rim, having a bladeless region inside. Since 1922, when Donat Banki published his patent [4], many studies proposed different construction designs of cross flow turbines.

This paper aims to analyze by numerical simulations correlated with experimental results the cause of poor efficiency of Banki turbines operating at low rotational speeds. In the first part of the paper, after a brief presentation of the state of the art, experimental data and results of numerical simulations are presented. They were obtained on a Banki turbine designed and tested at low head. The numerical simulations were validated with the experimental data. Using the numerical results, causes of the decrease in efficiency as the rotational speed decreases are analyzed. In the last part of the paper, in order to mitigate recirculation phenomena and undesirable spread of the flow inside the internal zone of the runner, the shape of a control device is proposed based on the insight into the flow offered by the numerical simulations.

2. Theoretical framework

Environmentally friendly hydropower systems can be implemented through pico-turbines that do not require high heads to operate. Cross flow turbines are a good choice for small hydropower, but they are usually designed to work at heads of 2 m < H < 200 m. Turbine efficiency was always a primary topic for researchers looking for the best solution and some of them focused on low head operating conditions.

Mockmore and Merryfield [5] made experimental tests on a Banki turbine, having the diameter D = 332 mm and the length L = 305 mm, that was discharging from the nozzle at atmospheric pressure. Their turbine reached a maximum efficiency of 68% when operating with the flowrate Q = 62 l/s at the head H = 5 m.

Durgin and Fay [6] constructed a small turbine with a plexiglas pressure casing and an open ended runner. The maximum efficiency was of 61% in this case. For other design solutions, the tests performed by Van Dixhorn [7] lead to maximum efficiencies in the range 60...70%, at very low heads, 1 m < H < 2.6 m.

Aziz and Desai [8] studied 27 cross flow pico-turbines at the Hydraulic Laboratory from Clemson University. The maximum efficiency values were in a range from 47% to 76%, depending on number of blades, ratio between internal and external diameter, and angle of attack. By combining the results of their study, Desai et al. [9] proposed prototypes expected to reach a maximum efficiency of up to 84%.

A research conducted by Olgun [10] on 4 cross flow turbines revealed that the turbines can reach an efficiency of 72% efficiency at a head of H = 8 m. During experimental studies performed at Norwegian University of Science and Technology, Walseth [11] obtained an efficiency of 75% at a head of H = 5 m.

In the last years, experimental studies were accompanied by numerical studies which confirm that a low head restricts the turbine efficiency to a moderate value. For instance, Andrade et al. [12] carried out numerical studies on a Banki turbine and the highest efficiency obtained was of 75%.

3. Flow control methods

Cross flow turbines are less performing compared to other types of turbines, due to their design that makes the water cross the runner blade passages twice. During the first pass, in the first stage of energy transfer, the efficiency is good, because the angle of attack at the entry of the blade passage can be controlled by properly designing the nozzle and the curvature of the blades. After the first pass, the water flow through the inner part of the runner can exhibit chaotic trajectories and the angle of attack at the second entry into the blade passage is difficult to be foreseen. This leads to shock losses and, consequently, to a poor energy transfer during the second pass.

Previous studies, developed by researchers in order to control the flow in the inner zone as much as possible, proposed different internal devices to be installed there. A patent authored by Kuenzel [13] presents a water turbine with an internal channel that captures the flow at the exit of the first pass and conducts it through a guide channel to the blades where the second pass starts. Tsutomu et al. [14] proposed to insert a regulating plate on the shaft of the cross flow turbine to improve the angle of attack at the entry of the second pass. Another interesting solution developed by Takashi [15], in order to reach a constant direction at the exit of the first pass and at the entry of the second pass, is to install a fixed guide vane inside the inner zone, concentric to the runner blading.

In the '90s, several researchers at Clemson University [16, 17] proposed and tested interior guide tubes that improved the efficiency with up to 5%, by better directing the water flow exiting the blades after the first pass towards the second pass. Their studies also highlighted the importance of the losses caused by the impact of the water flow on the shaft.

To overcome the problem of impact losses, Sinagra et al. [18, 19] proposed a cross flow turbine without shaft. They performed numerical and experimental tests on prototypes, with and without shaft. According to their results, a 5% increase in efficiency is possible by eliminating the impact losses caused by the shaft.

4. Results of experimental and numerical investigations

A classic Banki pico-turbine designed for low heads was tested at Department of Fluid Mechanics, Fluid Machines and Drives from Technical University Gheorghe Asachi of Iasi. The external diameter of the runner is D_{ext} = 382 mm, the internal diameter is D_{int} = 252 mm, and the diameter of the shaft is D_{shaft} = 40 mm. The runner has a width of 462 mm. Experimental results obtained for different discharges at the head H = 1.2 m are presented in Fig. 1. It can be noticed that the best turbine efficiency, that can reach the value $\eta = 58\%$, is attained roughly at the same speed n = 125 rpm irrespective of discharge. For rotational speeds lower than 100 rpm, the efficiency drops below 50% even at the maximum discharge.

To better understand the shape of the performance curves presented in Fig. 1, numerical simulations were performed with ANSYS Fluent for the air-water two-phase turbulent flow inside the turbine. A two-dimensional geometry that represents a cross-section of the turbine was studied. The flow domain was split into three subdomains – the nozzle, the runner, and the draft tube beneath the runner – separated by interfaces that allow the transport of all flow quantities. The runner movement was simulated with the sliding mesh technique. The numerical analysis used the k-ω SST turbulence model. A constant water mass flow rate of 502.2 kg/(s m) (corresponding to a discharge of 232 l/s distributed over the entire runner width) was imposed at nozzle inlet. At draft tube outlet, a total pressure of 0 Pa was prescribed, since there the pressure equals the atmospheric pressure. At all solid walls, the usual no-slip condition was used. The flow equations together with the boundary conditions were integrated in time and space with the Finite Volume Method and the water-air interface was tracked with the Volume of Fluid Method. The flow equations together with the boundary conditions were integrated in time and space with the Finite Volume Method and the water-air interface was tracked with the Volume of Fluid Method.

The simulations were validated with the experimental data. Although the efficiency is of main interest here, it is rather difficult to compare the efficiencies obtained experimentally and numerically since the experiment provides the total efficiency of the turbine while the simulations can provide only the hydraulic efficiency of the runner, without properly accounting for the mechanical and volumetric efficiencies. While it is relatively easy to estimate the mechanical efficiency, it is very difficult to assess the volumetric efficiency. Therefore, instead of the efficiencies, the shaft torques were compared and the comparison is presented in Fig. 2. It can be seen that the agreement between experiment and simulation is fairly good, considering that the numerical simulations do not account for mechanical and volumetric losses.

Numerical results obtained for the discharge Q = 232 l/s and the head H = 1.2 m are presented in Fig. 3 for two rotational speeds: 50 rpm and 125 rpm. For a good interpretation of the results, a separation of the fluid flow domain in different zones is proposed in Fig. 3a. The first zone, denoted (1), corresponds to the first stage of energy transfer inside the blade passages that receive water from the nozzle and discharge it into the free flow inner zone. Zone (2) corresponds to the second stage, where the energy transfer takes place inside the blade passages that receive water from the free flow inner region and discharge it outside the turbine. In the free flow inner region, there is a zone (3) where the water has favorable trajectories and several zones where the water spreads along undesirable directions. Such a zone is (4), generated by a solid obstacle: the shaft. Other zones are characterized by recirculation phenomena: zone (5) inside the inner part and zone (6) inside blade passages that go from the first to the second pass.

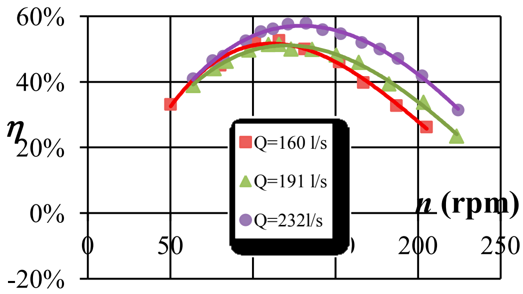


Fig. 1. Experimental results.

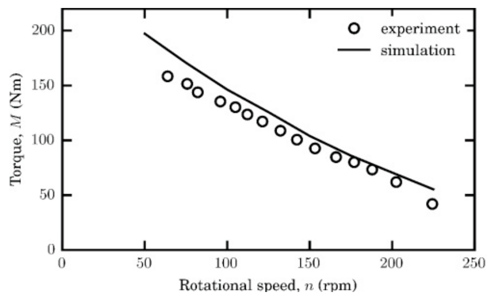
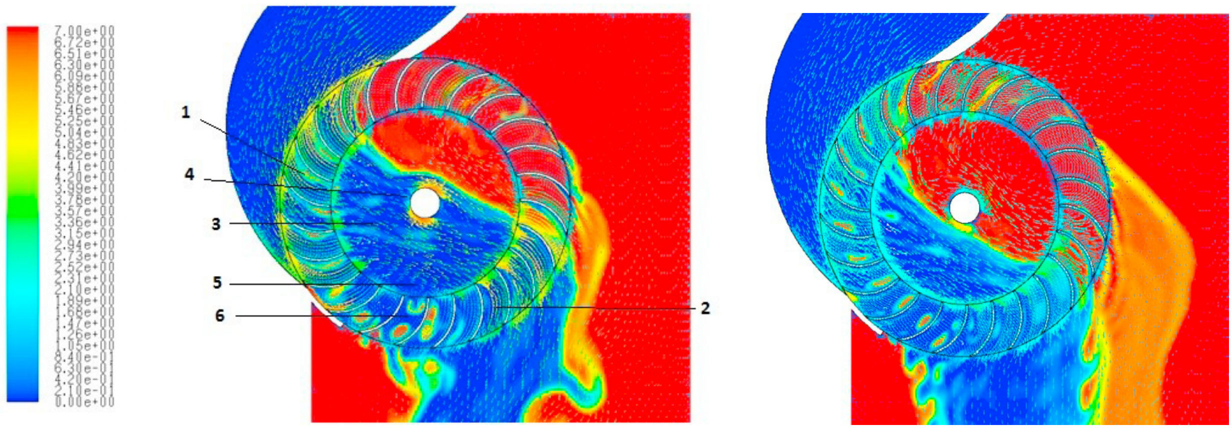


Fig. 2. Comparison between measured torques and torques obtained by numerical simulations.

The results of the numerical simulations reveal several causes of the poor turbine efficiency at low rotational speeds. In Fig. 3a, it can be noticed that at n = 50 rpm the water impinges on the shaft, which means that the momentum decreases and the flow paths are adversely diverted. Moreover, the new trajectories of the water particles interfere with trajectories of neighboring particles. In Fig. 3b, at the best efficiency point n = 125 rpm, there is no interaction between the shaft and the zone filled by the water. A further analysis of Fig. 3a versus Fig. 3b highlights another important issue: recirculation phenomena occur at the rotational speed n = 50 rpm – in zone (5) –, while at the rotational speed n = 125 rpm, the recirculation is no more present.



a) n = 50 rpm b) n = 125 rpm

Fig. 3. Air-water volume fractions and velocity fields inside the Banki turbine at two speeds: a) n = 50 rpm and b) n = 125 rpm. The background color indicates the air-water volume fraction (blue for water, red for air). The legend shows velocity values.

5. Proposed solution

Based on the results of the numerical simulations presented in Section 4, shapes of internal guiding devices are proposed in this section. In Fig. 4, the proposed devices, denoted A and B, are presented. Their shape is in accordance with the velocity fields obtained numerically.

The momentum generated by the flow determines the position of the mobile device A, that can roll around the shaft. Device A is impermeable and fills all the upper space of the internal zone, blocking the access of the flow to the upper zone and forcing it to move straight to zone 2. Device B obstructs the access of the water to the recirculation zone (5) and to the blade passages in zone (6), where, as it can be observed in Fig. 3a, an undesirable water-air mixture generates difficulties during the second pass.

The contour of device A can be divided into a surface that faces the blades and a surface that faces the inner zone of the runner. The surface A1-A4-A3 is part of a cylindrical sector separated from the blades by a narrow space. To force the flow to follow a favorable path, the shape of the other surface of device A, denoted A1-A2-A3, has to be in accordance to the form of the streamlines at the best efficiency point. To avoid shock losses, the surface A1-A2-A3 encloses the shaft.

Device B has a convex shape designed to avoid the recirculation in zone (5) and to better direct the water from the bottom side of the inner zone towards the blade area (2), thus improving the chances for a more efficient energy transfer in that zone.

The path to be followed by the flow in the Banki turbine having the proposed devices installed is as follows: the water flows through the blade passages (1), where the first stage of energy transfer takes place, then it reaches zone (3) bounded above by device A and below by device B, and finally it exits through the blade passages in zone (2), where the energy transfer of the second stage occurs.

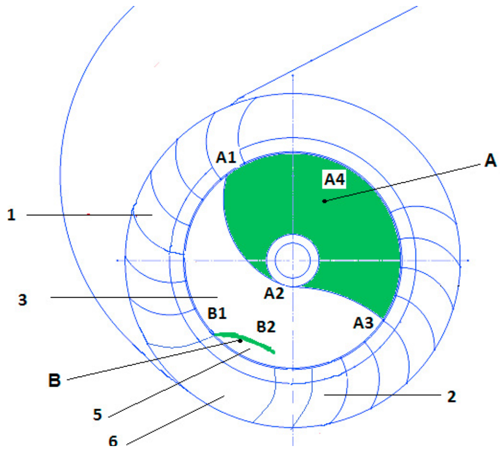


Fig. 4. Devices A and B for flow control.

6. Conclusion

The results of the experimental and numerical study presented in this work prove that an important issue of Banki turbines derives from the fact that, after the first pass between the runner blades, the water flows uncontrolled through the bladeless internal region of the runner. Based on the analysis of the velocity fields obtained by numerical simulations performed with ANSYS Fluent, zones of the flow domain that are responsible for the decrease in turbine efficiency were identified.

The numerical simulations indicate that the water interacts negatively with the runner shaft and recirculation phenomena occur when the rotational speed is low. These operating conditions correspond to the low efficiency obtained by experimental tests on a Banki turbine, at the rotational speed n = 50 rpm. The numerical simulations performed at the best efficiency point, found by experimental tests to be reached at the rotational speed n = 125 rpm, confirm that, in this case, the aforementioned phenomena do not occur anymore.

To mitigate recirculation phenomena and eliminate shock losses produced by the impact of the water jet on the shaft, the usage of an internal guiding device was proposed in this paper, as a preliminary result of an ongoing research. The flow field resulted from a CFD analysis offered guidelines for properly designing the device. Further numerical and experimental work must be carried out to assess the effectiveness of the proposed device and to fine tune its geometry so that it remains efficient over an as wide as possible range of operating conditions.

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