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An improved formula for determination of secondary energy losses in the runner of Kaplan turbine

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A R T I C L E I N F O

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A B S T R A C T

Secondary losses, which occur during the energy transfer process in axial turbomachinery, have been the subject of investigation for a long time. So far, much more attention has been naturally paid to the research of the phenomenon in gas and steam turbines than in hydraulic turbines. In the paper, the profile and secondary energy losses in the Kaplan turbine runner were considered by the comprehensive numerical simulations using two turbulence models and by the available experimental data of the blade profiles' characteristics. The performed research provides a contribution to better determination of secondary losses in the engineering practice, its distributions and participations in the total turbine losses for a wide range of on-cam operating modes. According to the proposed methodology and the obtained results, the improvement of the existing formula has been done thus enabling more accurate quantifying of secondary losses in the runner of the axial hydraulic turbine.

1. Introduction

The most common contemporary trend in the power generation industry, apart from building new hydropower plants (HPPs), is the refurbishment of the existing ones with the aim of both improving their reliability, operational integrity and efficiency, and ensuring life extension and availability. Such an approach is generally reasonable due to the fact that HPPs are renewable by nature, less capital intensive, have low gestation period and are pollution free and the cleanest source of energy [1].

Kaplan turbines have long been in use for efficient energy generating process and their good hydrodynamic and energy characteristics enable them to be operated in a wide range of discharges and lower net heads. In the following decades, most of the existing various capacities units with Kaplan turbines will be modernized, allowing for improved efficiency and higher power output by retrofitting new equipment [2]. Designers must either modify the existing components or develop the new ones, both of which should be highly specific to the site operating conditions and well adapted to other existing components for which replacement cannot be economically justified [3]. Generally, the efficiency of the entire hydraulic turbine is a consequence of all of its flow passages energy losses, so the successful design of a turbine is primarily based on the knowledge of fluid flow and hydraulic loss in each of them. As a result of significant progress in the turbine design over the past decades, hydraulic losses in the spiral casing, guide vanes and runner have had rather modest variation with the operation regime [4,5]. Although each passage plays a role in the hydraulic energy transfer process, most attention in design and optimization of Kaplan turbines is paid to the well-profiling runner blades. Also, according to the cost-effective proved arguments [1], the component which is most often replaced and modified during the modernization is the turbine runner.

During the energy transfer process in the interaction between the fluid and runner blades that generates the runner torque, the reduction of the hydraulic energy occurs due to the complex real viscous fluid flow. In order to understand and predict such a reduction, it is absolutely essential to know the losses mechanisms, to find the method for their quantifying and to determine the distribution of different types of losses in the runner for a wide range of operating modes. Losses in the axial turbine runners, which are diversely categorized by the researchers, have been the subject of investigation for a long time. The literature survey shows that much more attention has been naturally paid to the research of the losses phenomena in the gas and steam turbines and not enough in the hydraulic turbines. According to the proposed losses mechanisms given in Ref. [6], as well as the presented overview of

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the loss models in axial turbomachines [7,8], total losses in the runner could be variously classified. In general, any strict classification of the runner losses is questionable because of their simultaneous occurrences, however it is very common, for axial hydraulic turbines, to separate the losses in two types only: profile and secondary losses. Profile losses are defined as the sum of losses generated in the boundary layers on blade surfaces, shock losses and mixing losses in the wake region downstream of the trailing edge. All other additional losses belong to non-profile losses, the so called secondary losses. Secondary losses in the runner of a Kaplan turbine are direct consequences of the existing radial gaps (shroud and hub tip clearances), the secondary flow in the regions between the blades (blade-to-blade passages) and the three-dimensional character of the boundary layers [9]. The previously mentioned factors cause tip leakage and endwall losses as well, both of which are particular components of secondary losses. These losses are also in close relation with the volumetric losses in the Kaplan turbine runner, but their diversification cannot be performed easily.

The phenomena of fluid flows in tip clearances and endwall regions of the runners and their effects on total energy losses in axial turbomachinery were investigated by numerous researchers [10–16 etc.]. Much less relevant results of such investigations in the field of hydraulic turbines neither exist nor are available. The losses in the tip clearance of the Kaplan turbine were first considered using the lifting line theory to a simplified two-dimensional rectilinear cascade [17]. Mutual dependences between secondary losses in the runner of a propeller hydraulic turbine and certain runner geometrical parameters for some operating conditions are given in Ref. [18] as an analytical relation. Then, some significant improvement based on the theoretical approaches was implemented in that relation [9,19]. In Ref. [20], three tip clearance configurations were investigated experimentally in a small axial hydraulic turbine to determine the effect of clearance on the efficiency and flow.

In the last two decades, with the rapid increase of the computational fluid dynamics (CFD) applications, numerical experiments have played an important role in both predicting the fluid flow and researching the energy losses in each flow passage of water turbines. The analysis [21,22] of the tip clearance flow in a Kaplan water turbine, under four different conditions (different guide vane angles and specific speeds, but for constant runner blade angle), was conducted by numerical simulations using k-ω turbulence model focusing on the mass flow, velocity components and pressure distributions and visualization of vortex core close to the tip clearance. According to the numerical researches [23,24], the leakage flow through the tip clearance can be divided into four regions of the blade flange: leading edge, low pressure regions in the pressure and the suction sides, middle region within tip clearance where the leakage flow changes direction and trailing edge vortex region. The influences of the tip clearance size on the leakage flow and tip vortex in the Kaplan turbine runner were determined by the shear stress transport (SST) turbulence model [25] and by the large eddy numerical simulations [26]. In Ref. [27], the hub leakage flow and its effect on the Kaplan turbine performance are investigated by the CFD numerical approach.

The occurrence of cavitation in the tip vortex structure of the Kaplan turbine runner with and without anti-cavitation lips has also been the subject of the CFD investigation using different turbulence models [28–31]. On the other hand, the contemporary measurement methods in experimental investigation of the tip clearance flow phenomena have been being increasingly applied recently [32–34 etc.].

Following in the paper, there is one of the approaches to the determination of total secondary losses in the runner of the Kaplan turbine combining the numerical results obtained by two turbulence models use and the existing experimental data of blade profiles' characteristics. The authors' focus was getting an improved formula for the determination of secondary losses, using the existing one as the starting point.

2. Theoretical background

Streamlines in the rotating blade-to-blade flow passage of the Kaplan turbine runner should be, theoretically, congruent. Namely, the fluid should flow following each blade surface and be balanced in the direction perpendicular to the twisted blade surfaces between the volume centripetal force along the streamlines and the pressure gradient normal to streamlines. However, from the inlet to the outlet of the runner, the spanwise (from hub to shroud) nonuniformities of both the velocity and pressure distributions take place, especially in the blade endwall regions (Fig. 1). An imbalance between the normal pressure gradient and the centripetal acceleration near the endwall is caused by the presence of radial gaps – tip clearances in the shroud and hub regions. That imbalance induces oblique flow on the pressure side of the blades disrupting uniform surfaces flow, i.e. the flow moves towards the suction side of the blade. Such generated secondary flow with the three-dimensionality of boundary layers in the endwall region has an influence on additional (secondary) loss, which together with the total profile loss increases total energy losses in the runner of the Kaplan turbine. The total blade profile loss is usually determined as the midspan loss, but the prediction and calculation of secondary losses is a very important but also a challenging task.

Generally, by neglecting the gravity influence, the well-known Bernoulli equation can be used for determining the runner total losses as a difference of the circumferentially averaged values of specific energy $e = p/\rho + 0.5w^2$ which refer to the runner inlet and outlet cross-sections in the rotating coordinate system. Therefore, the total relative runner loss δ_{LR} can be defined in relation (1) as a part of the total runner energy loss e_{LR} in the specific hydraulic energy of turbine E

$$\delta_{LR} = \frac{e_{LR}}{E} = \frac{e_1 - e_2}{E}. \quad (1)$$

Since the total runner loss is the sum of the total profile loss e_{LRp} and the total secondary (additional) loss e_{LRs} [35], the total relative runner loss can be also calculated by relation (2)

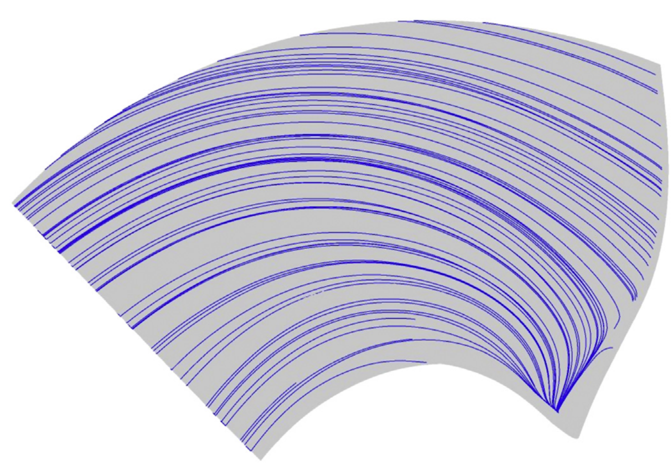

Fig. 1. Visualization of streamlines in the pressure side of the blade.