Cairo University

NUMERICAL STUDY FOR PERFORMANCE EVALUATION OF A CROSS-FLOW HYDRAULIC TURBINE

By

Mostafa Mohamed Mahmoud Badawy

A Thesis submitted to the
Faculty of Engineering at Cairo University
in Partial Fulfillment of the
Requirements for the Degree of
MASTER OF SCIENCE
in
MECHANICAL POWER ENGINEERING

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Title of Thesis:
Numerical Study for Performance Evaluation of A Cross-Flow Hydraulic Turbine

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Summary:
Mini-hydropower plants play an important role in supplying power to remote areas. This study focuses on studying the cross-flow hydraulic turbine as an outstanding example of mini-hydraulic turbines. The study is performed using CFD analysis to assess the most appropriate design parameters that can improve the cross-flow hydraulic turbine performance. The study started with reviewing the outstanding previous studies and the comparative study of the available data yielded a set of design parameters that are expected to constitute the optimum design parameters for a high-performance cross-flow turbine. From that study, it was found that a newly defined specific speed (named unified specific speed) can be used better to define the geometry of the radial section of the turbine irrespective of rotor width. The comparative study optimum design parameters were tested using a numerical model using ANSYS Fluent 15.0 software. The results showed the maximum efficiency was enhanced from 62.9% to 67.12% when applying the recommended optimum parameters for the numerically investigated turbine.

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Nomenclature

Symbol	Description	Unit
b	Nozzle width	m
B_w	Runner blade width	m
C	Absolute velocity, $C = \sqrt{2gH}$	m/s
C_{radial}	Radial chord, $C_{radial}=\frac{D_0 - D_i}{2}$	m
D_i	Runner inner diameter	m
D_m	Runner mean diameter, $D_m=\frac{D_0 + D_i}{2}$	m
D_o	Runner outer diameter	m
g	Gravitational acceleration	m^2/s
H	Total head	m
N	Rotational speed	rpm
N_s	Specific speed , $N_s = \frac{N\sqrt{P}}{H^{5/4}}$	--
N_s^*	Unified specific speed, $N_s^* = N_s / \sqrt{(b/D_o)}$	--
P	Power	Watt
Q	Volume	m^3/s
r_b	Radius of blade curvature	m
S	Pitch, $S = \frac{\pi D_m}{Z}$	m
S_0	Nozzle throat width	m
TSR	Tip speed ratio, $TSR= \frac{U}{C}$	--
U	Blade velocity, $U=\frac{\pi D_o N}{60}$	m/s
Z	Number of runner blades	--

Greek

η	Efficiency	--
α_0	Nozzle outlet angle	degree
α_1	First stage inlet flow angle	degree
α_2	First stage outlet flow angle	degree
α'_1	Second stage inlet flow angle	degree
α'_2	Second stage outlet flow angle	degree
β_1	First stage inlet blade angle	degree
β_2	First stage outlet blade angle	degree
β'_1	Second stage inlet blade angle	degree
β'_2	Second stage outlet blade angle	degree
λ	Nozzle entry arc (arc of admission)	degree
ρ	Density	kg/m^3
τ	Torque	N.m

ω	Angular velocity	rad/s
Ψ	Head coefficient, $\Psi = \frac{2gH}{(\pi D_0 N/60)^2}$	--
φ	Flow coefficient, $\varphi = \frac{Q}{(\frac{N}{60})(b)(D_0{}^2)}$	--

Abstract

Mini-hydropower plants play an important role in supplying power to remote areas. This study focuses on studying the cross-flow hydraulic turbine as an outstanding example of mini-hydraulic turbines. The study is performed using CFD analysis to assess the most appropriate design parameters that can improve the cross-flow hydraulic turbine performance.

The study started with reviewing the outstanding previous studies with the aim of comparing available data to find out how to select the optimum values of the main design parameters according to previous works.

The comparative study of the available data yielded a set of design parameters that are expected to constitute the optimum design parameters for a high-performance cross-flow turbine. From that study, it was found that a newly defined specific speed (named unified specific speed) can be used better to define the geometry of the radial section of the turbine irrespective of rotor width.

The results of the comparative study showed that the recommended range for the unified specific speed (N_s^*) is between (65) to (115), the admission arc (λ) value (90°), runner diameter ratio (0.65), pitch to radial chord ratio (S/C_{radial}) to be (0.5), tip speed ratio (TSR) to be around (0.47) and the inlet and the outlet blade angle are 30° and 87° respectively. The nozzle outlet angle (α_0) is preferred to be as small as possible and to be about 13°.

The nozzle orientation is recommended to be vertical in order to make use of gravity and the inlet guide vanes weren't recommended because it causes flow blockage and increases frictional losses. The nozzle throat width S_0 is recommended to be 0.32 of runner diameter because any further reduction led to an increase in the total head required to overcome the restriction of the nozzle throat and would affect the efficiency adversely. The use of anti-recirculation block (ARB) is recommended to reduce the losses due to flow recirculation inside the rotor cavity.

A numerical model using ANSYS Fluent 15.0 was built to study the performance of the cross-flow turbine. The model was tested using experimental and numerical data from a previous work and showed good results. The suggested numerical model was used to predict the performance of a turbine designed according to the recommended set of design parameters. The results showed improvement in performance. The maximum efficiency obtained was enhanced from 62.9% to 67.12% when applying the recommended optimum parameters for the numerically investigated turbine.

Chapter 1 INTRODUCTION

1.1 Introduction

The first cross-flow turbine was developed by an Australian mechanical engineer, Anthony George Maldon Michell who got his patent in 1903. Then the Hungarian Donát Bánki performed a series of studies on cross-flow turbines in the period between 1912 and 1919. Thus the cross-flow turbine became known as Banki turbine. In 1933 Ossberger's first patent was granted and he manufactured his turbine as a standard product. Nowadays, Ossberger's company is the leading manufacturer of this type of turbine [1]. Figure (1-1) and figure (1-2) show the cross-flow turbine and its general features.

The turbine's runner is built from two or more parallel disks connected together by a series of thin curved blades. The water leaves the nozzle in the form of a rectangular cross section jet and this jet passes through the first stage of the runner blades to the empty space in the center then passes again through the second stage runner blades.

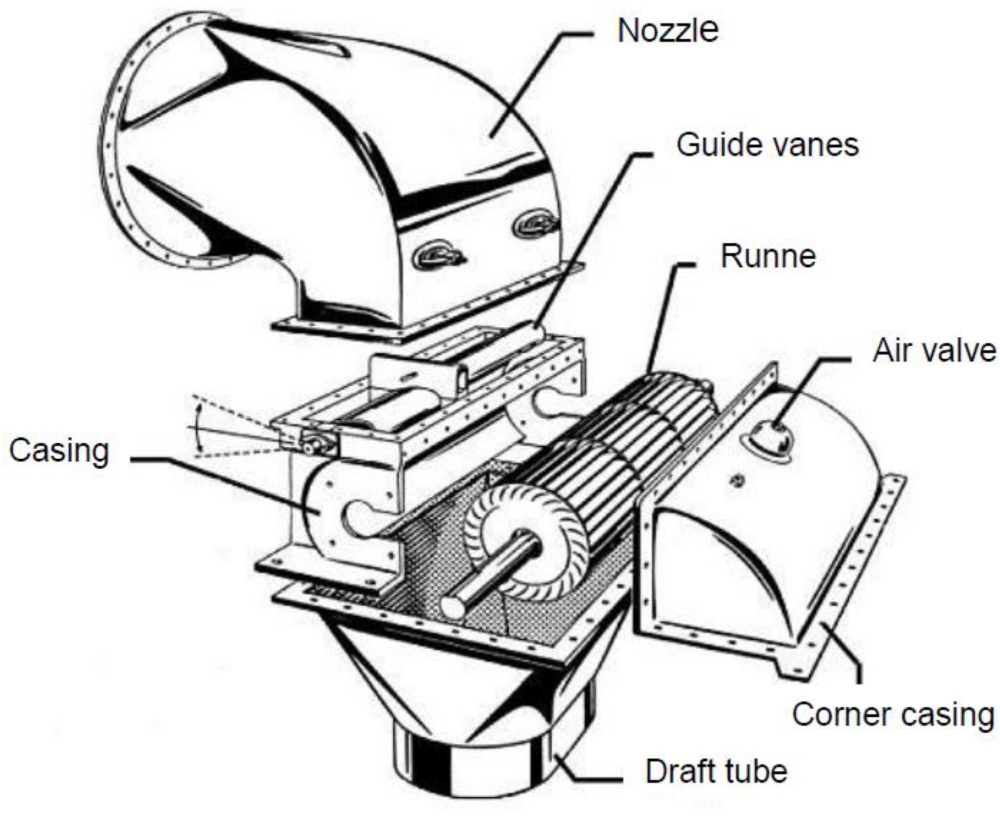

Figure 1-1 Cross-flow turbine *[2]*

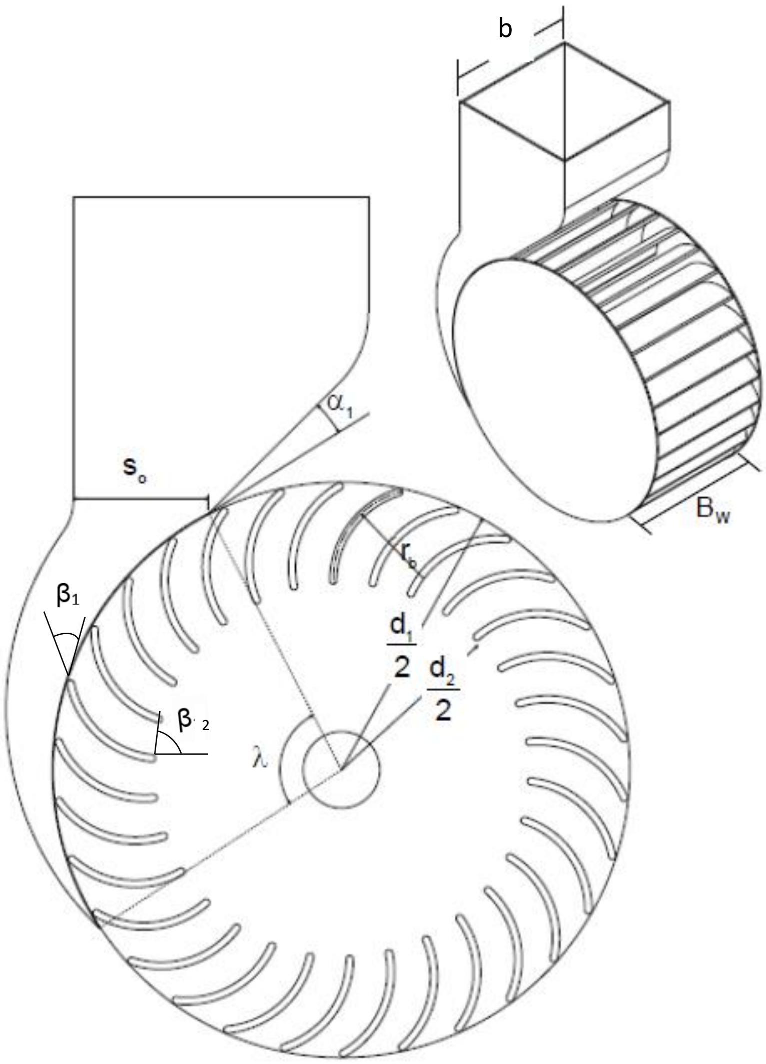

Figure 1-2 General features of cross-flow turbine *[2]*

This chapter is divided into two parts, the first part focuses on reviewing the most important experimental and numerical studies on the cross-flow turbines.

The second part of this chapter is devoted to the analysis of literature review. The purpose of this analysis is to find the most appropriate hydraulic cross flow turbine design parameters.

1.2 Literature review

Mockmore and Merryfield [3] conducted a translation of Banki's paper "Neue Wasser turbine" and showed results of a series of tests on a laboratory turbine which was built according to the specifications of Banki. In their calculations, the nozzle outlet angle was 16° and the blade inlet angle was 30° while the blade outlet angle was 90°.The actual rotational speed 270 rpm and the highest efficiency attained was 68 %.