WATER TURBINES; CONTRIBUTIONS TO THEIR STUDY, COMPUTATION AND DESIGN

Published @ 2017 Trieste Publishing Pty Ltd

ISBN 9780649741434

Water Turbines; Contributions to Their Study, Computation and Design by S. J. Zowski

Edited by Trieste Publishing Pty Ltd.
Cover @ 2017

www.triestepublishing.com

S. J. ZOWSKI

WATER TURBINES; CONTRIBUTIONS TO THEIR STUDY, COMPUTATION AND DESIGN

Trieste

WATER TURBINES

Contributions to Their Study Computation and Design

BY

S. J. ZOWSKI

Published by
The Engineering Society
University of Michigan

THE AMERICAN HIGH SPEED RUNNERS FOR WATER TURBINES

Looking at a modern American standard runner for a water turbine one is liable to wonder why the design of such runners is considered to be one of the most difficult problems in hydraulic engineering. The forms of the runners are so natural, the buckets and their curvature so simple, that "we fail to see where the pretended difficulties are." As is usual in such cases, we forget here again that there is always a direct proportion between the simplicity of a machine and the amount of brainwork and time necessary to produce the same. A brief history of the evolution of the American turbine or a glance at the reports of the numerous tests made in the Holyoke testing flume would convince us of this fact. Indeed, the American standard runners, as they are manufactured now, represent a great amount of hard and earnest work. Hundreds of tedious and expensive experiments, with many a failure and success, had to be made—an experience of almost half a century had to be aggregated first, before this modern runner type was produed.

The aim was, first, of course, a good efficiency. But this was not all. Already in the early eighties—at a time when the European engineers still were questioning the advantages of the radial inward flow turbine— there were in this country wheels of this type, which yielded efficiencies up to 84%, acording to the tests made in Holyoke. And yet since that time remarkable progress has been made. Following the general tendency of modern engineering, the speed and capacity of the turbines had to be steadily increased. That for the turbine designer this resulted in new difficulties is evident, as high speed calls for small dimensions, while high capacity calls for large dimensions, and consequently the increase of both is possible only to a certain limit.

The purpose of this study is to show how far the American manufacturers of water turbines have come in this respect and also to compare the results which were obtained by their various runner types. The accuracy of this study is naturally limited by the accuracy of the data, which were accessible to the writer, and which were taken from the guarantees of the different concerns. But as these guarantees are based on careful tests made in the Holyoke testing flume, and as these tests are considered official in this country, also the results of the following study can be considered reliable. The comparison at least will be accurate, because, if mistakes in the testing of the wheels be made (and some engineers, especially in Europe, believe that the Holyoke tests are not quite reliable regarding the actual discharge) the same mistakes owuld be made on all runners.

NOTATION.

To get a proper basis for this study, some of the principal turbine formulae must be recalled and some new ones derived. The notation is the same, which the writer uses in his lectures on water turbines at the University of Michigan.

$H\text{-}P$ = effective power of the runner.
N = speed of the runner in R. P. M.
Q = discharge of the runner in cubic feet per second.
H = net head in feet acting upon the turbine = gross head minus all losses in head race, conduit and tail-race.
ϵ = hydraulic efficiency of the turbine.
$(1-\epsilon)\,H$ = head lost inside of turbine itself due to friction, whirls and shocks.
D_1 = mean entrance diameter of runner in feet.
B = height of guide case in feet.
α_1 = angle between entrance speed and peripheal speed at D_1.
β_1 = bucket angle at D_1.
c_1 = real entrance speed at D_1.
w_1 = relative entrance speed at D_1.
v_1 = peripheral speed at D_1.
c_r = radial entrance speed at D_1 = radial component of c_1, see Figs. 1 and 2.

SPEED.

All modern American runners are of the radial inward flow type, working with pressurehead. The definition of the "*pressurehead turbine*" or "*pressure turbine*" (so-called reaction turbine) is: "The water enters the runner and flows through the same with a certain pressurehead, as the whole available head is not turned into speed at the entrance. The real entrance speed

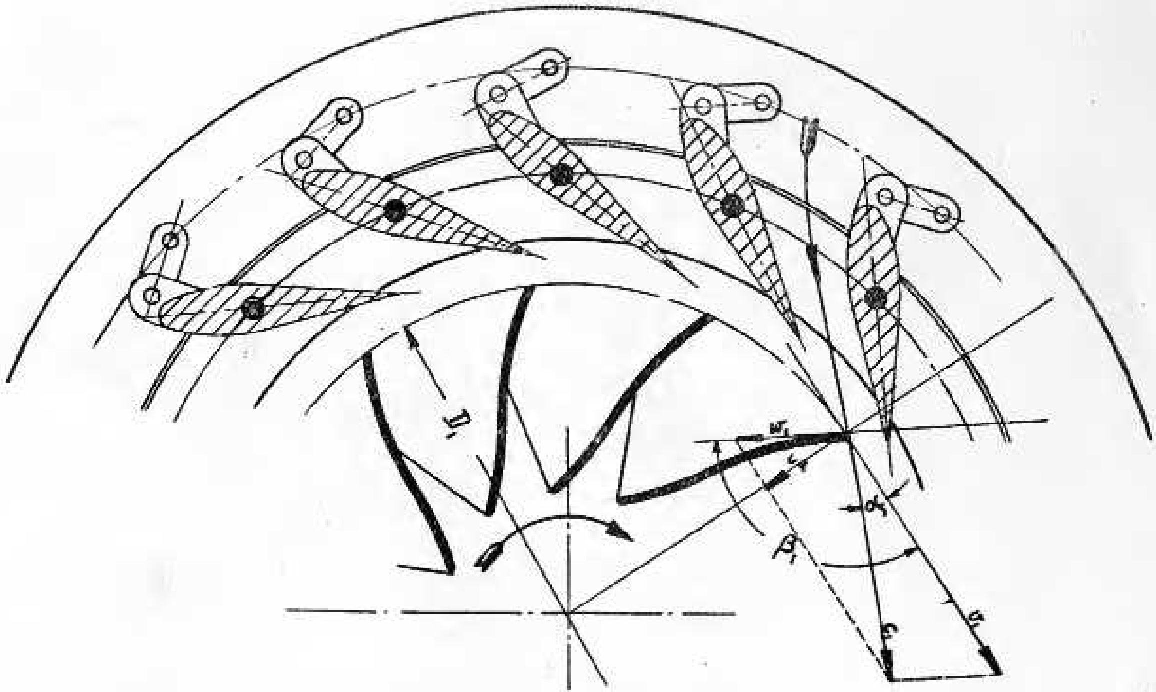

Fig. 1.

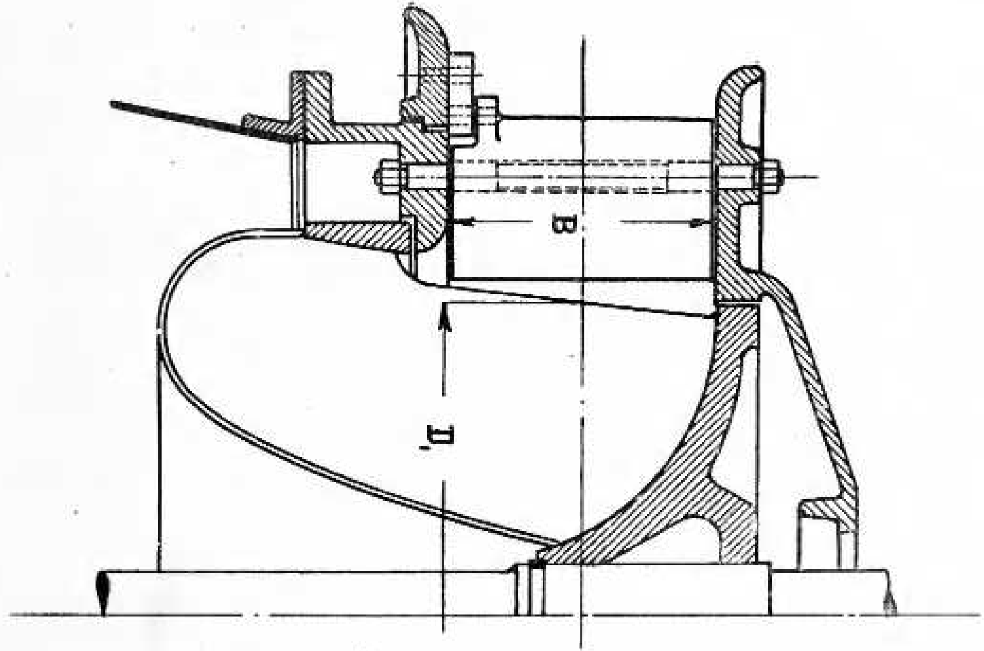

FIG. 2.

c_1 is smaller than the spouting velocity. A pressurehead is left, to be used for the acceleration of the flow of the water through the runner."

The regulation of the hydrodynamic conditions in the runner, for a flow either with or without pressurehead, is possible by the choicé of the angles β_1 and α_1.

If the entrance into the runner is *"shockless,"* and the discharge is *"perpendicular,"* (real discharge speed perpendicular to the corresponding peripheral speed) then

$$c_1 = \sqrt{\epsilon g H} \sqrt{\frac{\sin \beta_1}{\sin (\beta_1 - \alpha_1) \cos \alpha_1}} \qquad (1)$$

$$v_1 = \sqrt{\epsilon g H} \sqrt{\frac{\sin (\beta_1 - \alpha_1)}{\sin \beta_1 \ \cos \alpha_1}} \qquad (2)$$

Both c_1 and v_1 are functions of the angles α_1 and β_1 for a given head. The speed c_1 can naturally never exceed the spouting veloicty $\sqrt{2g\epsilon H}$. It would become equal to this velocity if

$$\sqrt{\frac{\sin \beta_1}{\sin (\beta_1 - \alpha_1) \cos \alpha_1}} \sqrt{\epsilon g H} = \sqrt{2 g \epsilon H}$$

or if

$$\beta_1 = 2\alpha_1$$

For all angles β_1 which are larger than $2\alpha_1$, the speed c_1 will be smaller than the spouting velocity, hence the turbine will be a pressure turbine.

For a pressureless turbine the peripheral speed would be

$$v_1 = \frac{1}{2 \cos \alpha_1} \sqrt{2 g \epsilon H}$$

This is variable only within very small limits, as $\cos\alpha_1$ varies only a little for the values of α_1 which are used in practice. Hence the peripheral speed of the pressureless turbine is practically given by the head, and consequently the speed N (R.P.M.) can be varied only by variation of the runner diameter D_1. As practical reasons restrict both the increase and decrease of D_1, the speed of a pressureless turbine is variable only within narrow limits. This is one of the main reasons why nowdays pressure turbines occupy the first place, and pressureless turbines (Impulse wheels and Schwamkrug-turbines) are used only when absolutely necessary. The speed of the pressure turbine can be varied not only by variation of the runner diameter but also, and very effectively, by variation of the angles β_1 and α_1. Combining both means, it is easy to vary the speed of a pressure turbine for a given head and capacity in the ratio of 6 : 1.

To show how the angles α_1 and β_1 affect the peripheral speed v_1 the factor

$$\sqrt{\frac{\sin(\beta_1 - \alpha_1)}{\sin \beta_1 \cos \alpha_1}}$$

of equation (2) has been represented by a series of curves. Figure 3 gives the values of this factor for a series of constant bucket angles β_1 with variable angles α_1. Figure 4 gives the same values for a series of constant angles α_1 with variable angles β_1.

For $\beta_1 = 90^\circ$ the factor

$$\sqrt{\frac{\sin(\beta_1 - \alpha_1)}{\sin \beta_1 \cos \alpha_1}} = 1$$

for all values of α_1. For all angles $\beta_1 < 90^\circ$ the value of the radical is smaller than 1; for all angles $\beta_1 > 90^\circ$ its value is larger than 1.

As a low or medium head turbine must, as a rule, be designed for a relatively high speed, all American standard runners, being built for low or medium heads, have $\beta_1 > 90^\circ$ and are *"high speed runners."* Runners with $\beta_1 = 90^\circ$ are called *"medium speed runners"* and those with $\beta_1 < 90^\circ$ are called *"low speed runners."* See Figures 5, 6, 7.

Practical reasons, as for instance the necessity of an easy, smooth and yet a short curvature of the bucket, are limiting the increase of β_1. The value of $\beta_{1max} = 135^\circ$ will represent good practice and will be found in many of the best American high speed runners. The increase of angle α_1 also increases the speed v_1 for all angles $\beta_1 > 90^\circ$. But to avoid what is called *overgating*, it is advisable not to assume too high values for α_1. Tests show that the capacity of the runner reaches its maximum at a certain gate opening. To open more, is not only useless, but even