Design of the Basic Performance Characteristics Maps of an Axial Flow Turbine Stage

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ABSTRACT

Performance charts for the preliminary design of an axial turbine stage have been constructed from the three basic dimensionless performance parameters reaction ratio (R), flow coefficient (ϕ), and stage loading coefficient (ψ). The charts are completed with constant nozzle outlet flow angle lines, estimated efficiency contours.

The performance charts may be used to evaluate; the optimum values of performance, dimensionless parameters, flow coefficient, flow angles, stage efficiency for any reaction ratio.

Notation	
c	Absolute flow velocity (m/s)
h	Specific fluid enthalpy (J/kg K^o)
H/b	Aspect ratio (height to axial chord ratio)
R	Reaction ratio
s/b	Space to axial chord ratio
t_{max}/l	maximum blade thickness to actual cord ratio
U	Rotational Speed (m/s)
V	Blade velocity (m/s)
w	Relative flow velocity (m/s)
ΔW	Fluid Specific Work (W/kg)
α	Angle in the absolute system (Degree)
β	Angle in the relative system (Degree)
ε	Deflection Angle (Degree)
ϕ	Flow coefficient
η_{tt}	Total to total efficiency
ζ	Enthalpy loss coefficient
ψ	Loading Coefficient

Subscript	
1	Nozzle inlet , Rotor outlet
2	Rotor inlet, Nozzle outlet
3	Rotor outlet, Nozzle inlet
R	Rotor
N	Nozzle
O	Stagnation

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INTRODUCTION

The performance of an axial turbine stage depends on too many variables, some are related to the geometry of the stage and others are related to the physical properties of the gas, the inlet conditions and the speed of the rotor. Through the use of dimensional analysis, these variables can be grouped into few dimensionless parameters.

In general, the flow through the machine is three dimensional and turbulent. The flow may be unsymmetrical, non-uniform and unsteady depending upon the stream conditions, number of blades and hub-tip ratio.

Many computer aided methods are widely used for the solution of three dimensional through flow method problems in axial Turbine; these are stream line curvature method, matrix through flow method and computational fluid dynamic. These methods utilize numerical techniques for the solution of approximate but complex flow and energy equations [1-6].

The complex nature of the above mentioned methods necessitate the extensive use of advanced computers and much of the designer time. For this reason they are not recommended at the preliminary design stage of axial turbine. Instead the designer at the preliminary design stage can opt for much simpler approximate methods using correlation based on the analysis and experimental results of low speed flow on two dimensional cascades [7-9]. Additional experimental correlations are available to take into consideration the compressibility and three dimensional effects on the design of axial turbine.

Thus and when a new turbine design is required it is customary to conduct preliminary design studies aimed at selected and therefore quantitative assessment of the turbine dimensionless parameters which will be related to performance and cost and to identify efficiency of the solution.

Once a preliminary selection has been made then the designer can use the three-dimensional complex method for design refinement or alteration.

Published axial turbine design methods can be categorized into those giving general guide lines on the choice of vector diagram parameters to achieve optimum efficiencies and those that enable efficiencies to be predicted for a turbine of specified design. Numerous empirical and semi-empirical relations have been used to evaluate the performance of axial turbine stage [9-10]. A turbine gives its best performance while operating at its design point. However, like any other machine or system, it is also expected to operate away from the design point, therefore knowledge about its behavior at off-design operation is also necessary.

The aim of this work is to construct charts for the purpose of preliminary design and performance prediction of an axial turbine stage.

The charts include all the dimensionless parameters that are necessary for the full aerodynamic description of the turbine stage as well as estimated efficiency contours.

THEORETICAL ANALYSIS

In order to develop performance charts for the axial turbine stage, a theoretical analysis is required to relate the turbine dimensionless parameters. Empirical relations, based on cascade and other experimental results can then be used to evaluate the efficiency of the turbine stage.

During preliminary design stage of an axial turbine for a specific duty (flow rate) the designer has to select, for each turbine stage, three dimensionless parameters namely, flow coefficient (ϕ), loading coefficient (ψ), reaction ratio (R). The selection is normally based on values that are expected to give the best possible efficiency for the required duty.

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Dimensionless Parameters

Flow coefficient (ϕ) is defined as the ratio of axial velocity to rotational velocity

$$\phi = \frac{c_x}{U} \text{ .... (1)}$$

The Stage loading coefficient (ψ) is defined as

$$\psi = \frac{\Delta W}{U^2} \text{ .... (2)}$$

where $\Delta W =$ specific work done by the stage $= h_{o2} - h_{o3}$

From steady flow energy equation (assuming adiabatic flow) and momentum equation[7].

$$\Delta W = h_{o2} - h_{o3} = U(c_{y2} + c_{y3}) \text{ .... (3)}$$

From velocity traingles, Fig.(1)

$$U(w_{y2} + w_{y3}) = U(c_{y2} + c_{y3}) \text{.................... (4)}$$

Then, in terms of rotor flow angles,

$$\psi = \frac{U(w_{y2} + w_{y3})}{U^2} = \frac{c_x(\tan\beta_2 + \tan\beta_3)}{U} = \phi(\tan\beta_2 + \tan\beta_3) \text{ .... (5)}$$

And, in the terms of stator flow angles

$$\psi = \frac{U(c_{y2} + c_{y3})}{U^2} = \frac{c_x(\tan\alpha_2 + \tan\alpha_3)}{U} = \phi(\tan\alpha_2 + \tan\alpha_3) \text{.................... (6)}$$

Stage reaction ratio (R) is defined as the ratio of rotor static enthalpy rise to complete stage static enthalpy rise. Thus

$$R = \frac{h_2 - h_3}{h_1 - h_3} \text{ .... (7)}$$

From velocity triangle (Fig.1) and the fact that for an axial turbo machines where the stage mean diameter remains constant, the rotor relative stagnation enthalpy remains unchanged. Thus [7, 9]:

$$R = \frac{w_{y3} - w_{y2}}{2U} = \frac{\phi}{2}(\tan\beta_3 - \tan\beta_2) \text{ .... (8)}$$

and

$$R = 1 + \frac{(c_{y3} - c_{y2})}{2U} = 1 - \frac{\phi}{2}(\tan\alpha_3 + \tan\alpha_2) \text{ .... (9)}$$

Combining equations (5),(8) and rearranging, then

$$\tan\beta_3 = \left(\frac{\psi + 2R}{2\phi}\right) \text{ .... (10)}$$

$$\tan\beta_2 = \left(\frac{\psi - 2R}{2\phi}\right) \text{ .... (11)}$$

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Combining equations (6),(9) and rearranging, then

$$\tan\alpha_3 = \left(\frac{\psi + 2(R-1)}{2\phi}\right) \text{.................... (12)}$$

$$\tan\alpha_2 = \left(\frac{\psi - 2(R-1)}{2\phi}\right) \text{.................... (13)}$$

The deflection angle (ε) can be defined as the summation of inlet and outlet flow angles, for rotor

$$\varepsilon_R = \beta_2 + \beta_3 \text{.................... (14)}$$

for nozzle

$$\varepsilon_N = \alpha_2 + \alpha_3 \text{.................... (15)}$$

From the above equations, performance curves can be established where for a specific reaction ratio (R) and any combination of loading coefficient (ψ) and flow coefficient (ϕ), the flow angles can be evaluated.

Alternatively for any combination of either outlet flow angle for rotor (β_3) or for nozzle (α_2) and either ϕ or ψ will give rise to other three flow angles and other dimensionless parameters.

Stage Efficiency

The total to total efficiency of an axial flow turbine stage can be found from h-s diagram (Mollier diagram) for a turbine stage [7], Fig (2).

$$\eta_{tt} = \frac{h_{o1} - h_{o3}}{h_{o1} - h_{o3s}} = \left[1 + \frac{(h_3 - h_{3S}) + (h_2 - h_{2S})}{(h_1 - h_3)}\right]^{-1} \text{.................... (16)}$$

Losses in axial turbine stage which are indicated by the losses in Enthalpy can be represented by Enthalpy loss coefficient (ζ) which is based on the outlet flow velocity.

For rotor

$$\zeta_R = \frac{h_3 - h_{3S}}{\frac{1}{2} w_3^2} \text{.................... (17)}$$

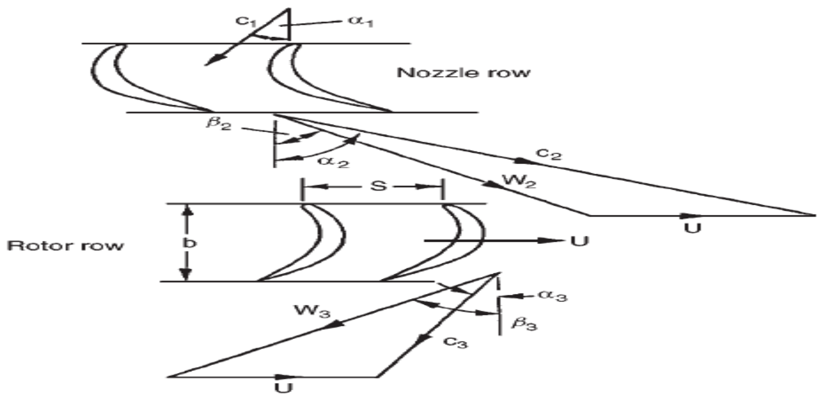

Figure 1. Velocity triangle for an axial turbine stage (at mean rotor diameter)

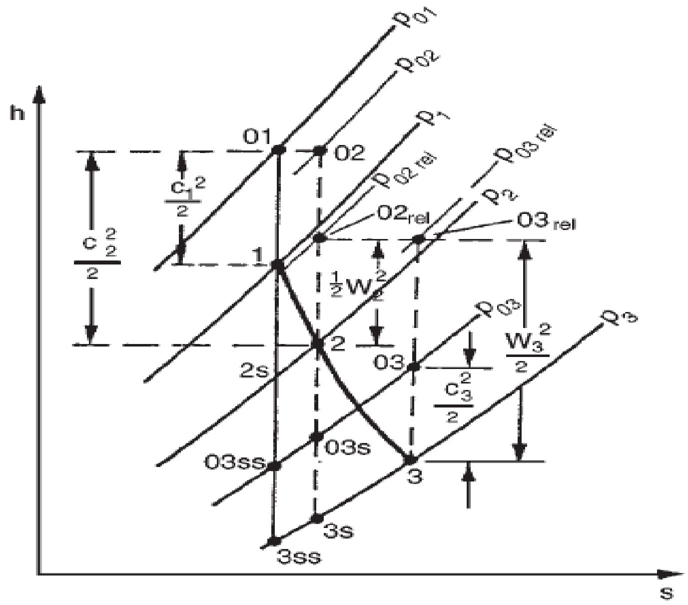

Figure 2. Mollier diagram for axial flow turbine stage

For nozzle

$$\zeta_N = \frac{h_2 - h_{2S}}{\frac{1}{2} \quad c_2^2} \text{....................} (18)$$

Substitute equations (17) and (18) into equation (16) to obtain

$$\eta_{tt} = \left[1 + \frac{\phi^2}{2\psi}\left(\frac{\zeta_R}{\cos^2\beta_3} + \frac{\zeta_N}{\cos^2\alpha_3}\right)\right]^{-1} \text{....................} (19)$$

Correlation for the evaluation of enthalpy loss coefficient will be discussed in the next section.

Loss Model

Aerodynamic losses occurring in turbine blade cascade can be grouped in the following categories:-

I. Profile Losses, these losses are associated with the growth of boundary layer on the blade profile and trailing edge thickness.

II. End Wall Losses, these include annulus, secondary, and tip clearance losses.

Various correlations are available in the literature for the evaluation of losses within the turbine blades [7, 10-13]. These correlations use empirical relations based on experimental results of cascades at low speed tests. The enthalpy loss model that will be used here, are based on the Soderberg′scorrelation. Soderberg′s suggests the following relation for Reynolds's number $\mathrm{Re} = 10^5$, aspect ratio $H/b = 3$ and the maximum blade thickness to actual cord ratio $t_{\max}/l = 0.2$:

$$\zeta = 0.04 + 0.06\left(\frac{\varepsilon}{100}\right)^2 \text{....................} (20)$$

for other values of aspect ratio the equation (20) will be

for the nozzle,

$$\overline{\zeta_N} = (1+\zeta)\left[0.993 + 0.021\left(\frac{b}{H}\right)\right] - 1 \text{....................} (21)$$

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and for the rotor,

$$\overline{\zeta_R} = (1+\zeta)\left[0.975 + 0.075\left(\frac{b}{H}\right)\right] - 1 \text{ .... (22)}$$

A further correction can be made if the Reynolds number is different than 10^5

$$\overline{\overline{\zeta}} = \left(\frac{10^5}{\text{Re}}\right)^{1/4} \overline{\zeta} \text{ .... (23)}$$

The above model was used in getting preliminary design charts.

RESULTS (DESIGN CHARTS)

Using the above analysis, performance charts which relate the dimensionless parameters (ϕ,ψ,R) and flow angels can be drawn completely with efficiency contours. The performance charts are constructed for the following selected stage geometrical parameters

- Space to axial chord ratio (s/b = 1.0),
- Aspect ratio (H/b = 3.0), and
- Maximum blade thickness to actual cord ratio ($t_{\max}/l = 0.2$).

The above are constructed to be approximately equal to their optimum or practical values.

Based on the Soderberg′scorrelation for the evaluation of losses, the complete performance charts can be established. Fig.(3.a–c) shows these charts for different reaction ratio values of 0.5, 0.4, and 0.6 respectively complete with efficiency contours. Deflection curves are not shown because their presence will complicate the already congested figures. Those figures show the following:

1. For a reaction ratio of 0.5, rotor and nozzle will have identical flow angles for the turbine stage.
2. With reaction ratio other than 0.5, the rotor and nozzle will have different flow angles.
3. For a specific reaction ratio (R) and for any selected value of (ψ,ϕ) then the charts can be used to evaluate the other parameter, all flow angles and efficiency, these parameters are extremely important for off-design operation.

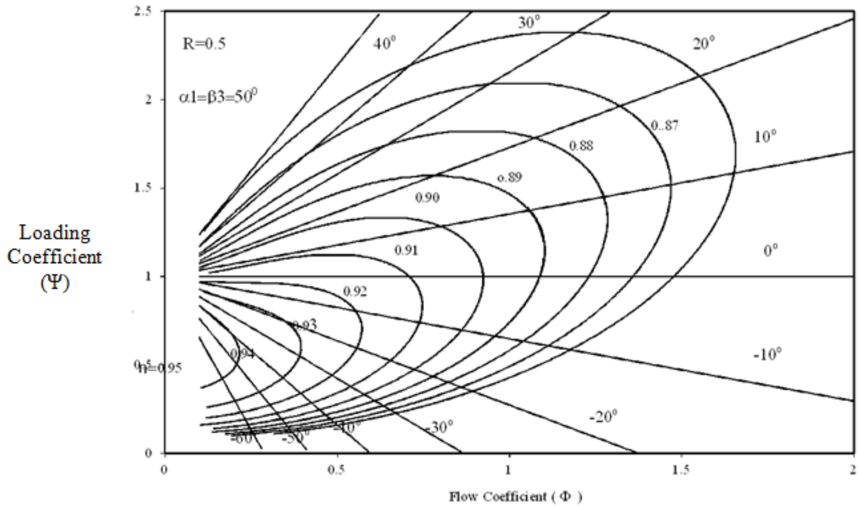

Figure 3a. Axil – Flow Turbine Performance Chart

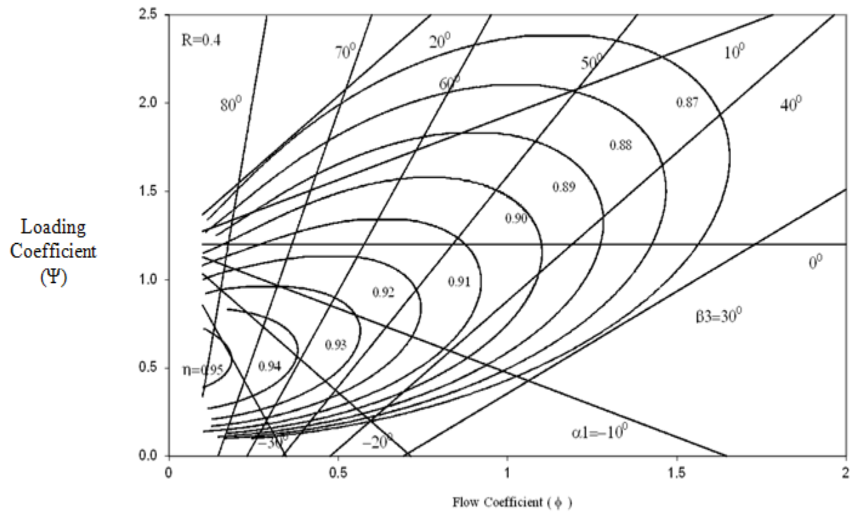

Figure 3b. Axil – Flow Turbine Performance Chart

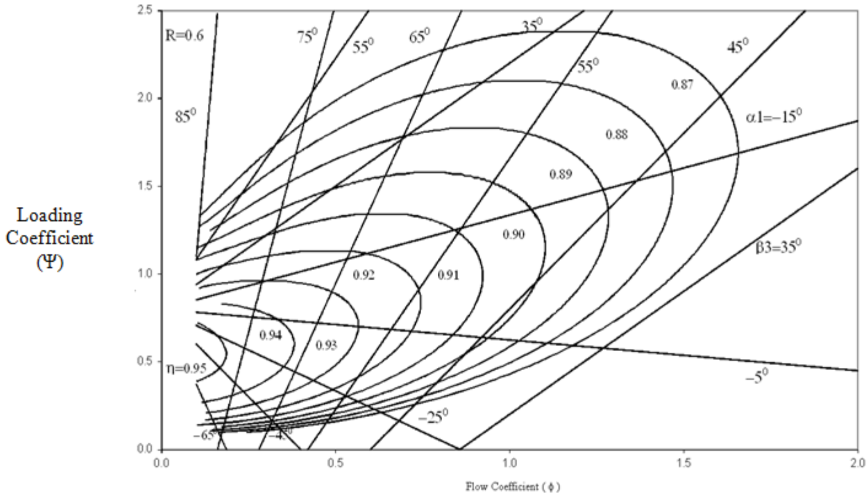

Figure 3c. Axil – Flow Turbine Performance Chart

Generally and for the optimum design conditions of an axial turbine stage, it is required to select optimum performance parameters that are expected to give best possible efficiency. Thus for different values of R, the design charts can be used to evaluate optimum parameters.

Although, the above results indicate that a reaction ratio of 0.5 should be used if higher stage efficiency and a suitable flow angles are required, the interest still remains in the effect of reaction ratio (other than 0.5) on the stage performance. This is because the design reaction ratio is normally selected at the rotor mean diameter and its value will vary along the blade height.

Hence, the results shown above in Fig.3.c has compared with some experimental and some numerical result (through flow method) []. the Fig. 4 shows that the total to total efficiency has computed based upon Soderberg′s correlation just above both experimental and numerical methods[]. However, according to this comparison the results of this paper would give good approximation and can be used for further analysis or starting points of any other numerical simulation.

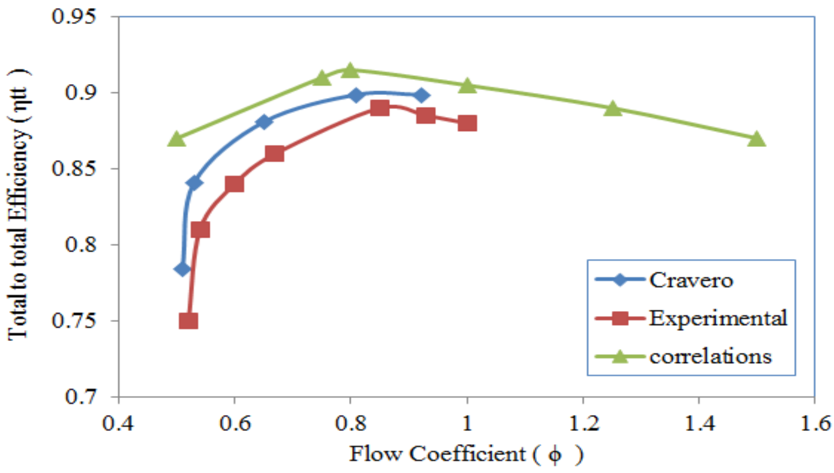

Figure 4. Experimental and Numerical Results [14] with the Paper Results

CONCLUSIONS

The following conclusions have been observed from the present work:-

1. For the preliminary design of an axial turbine stage, design charts of Fig. (2.a-e) can be used to select any combination of three parameters (ϕ,ψ,R) and to evaluate the corresponding flow angles, stage efficiency ηtt for the preliminary design of an axial turbine stage.
2. The design charts help designer to select what is the optimum parameters (stage efficiency, stage loading coefficient, flow coefficient, reaction ratio) according design requirement.
3. Obviously the optimum (design) parameters are defined as those that give maximum possible stage efficiency ηtt.
4. Using the above conclusion, a stage of reaction ratio of 0.5 has been shown to give the best design parameters because of the maximum efficiency and occur at .mean diameter and the acceptable values of flow angles
5. Using this simple correlation can give excellent results for further analysis.

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