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Some Aerodynamic Considerations in the Design of Centrifugal Compressor Impellers

G. Gopalakrishnan*
**Professor Emeritus, Department of Mechanical Engineering, Dr. M.G.R Educational & Research Institute, Chennai, India*

Abstract: *A simple one dimensional approach to determine the size of the centrifugal impeller for maximum through flow has been indicated. This design procedure does not mean that the centrifugal impeller could be finally decided on the results obtained from this article.*
It must be borne in mind that this only deals with some aspects of the design and in itself is not a complete one by which the impeller could have its final shape and size. The major idea of introducing this paper is to highlight the importance of some of the parameters that go towards influencing the design of the centrifugal impeller.
The stage pressure, temperature and work output of a centrifugal machine is basically dependant on the major dimensions like the diameter of the impeller '$d2$' and its tip speed '$u2$ ', the vane outlet angle '$\beta 2$' and the flow coefficient '$\emptyset 2$' at exit '$c2m/ u2$'.
Further, the influence of the inlet vane angle '$\beta 1$' and the inlet Mach number '$M1$'of the flow seriously affect the design of the high speed centrifugal impeller. Some guidelines have been indicated.
The shape of the impeller channel passages are to be designed based on whether the compressor needs to have an energy build up in the impeller or a high pressure generation within the impeller of the compressor. In such cases, the radial diffuser at the exit of the impeller plays a very crucial role.
Since the speed at which the impeller rotates is of interest, it is absolutely necessary for the designer to pay attention to the stress calculations of the impeller rear and front shrouds.

Nomenclature:

α	*fluid angle*
β	*vane, angle*
δ	*semi cone angle, boundary layer thickness*
$\emptyset$	*flow coefficient* c_m/u_2
ρ	*fluid density*
σ	*Slip fctor*
$\aleph$, *k*	*ratio of specific heats* c_p/c_v
ω	*angular velocity*Type equation here.
a, A	*constants, annulus area, area, velocity of sound*
b, bl, B	*vane height, blade*
c	*absolute velocity*
c_p	*Specific heat at constant pressure*
c_v	*Specific heat at constant volume*$\aleph$
h	*vane height*
k	*ratio of specific heats* c_p/c_v
m, M	*mas flow, Mach no.*
n	*distance normal to streamline*
p	*static pressure*
r, R	*radius*
t_1, *t* ,*T*	*time, temperature*
V	*volume flow rate*
w,W	*relative velocity, Specific work*
X	*coordinate direction, linear length*
y, Y	*coordinate direction*
z	*coordinate direction, number of vanes*

Subscripts

_	*average*
'	*first differentiation with reference to x*

"	*second differentiation with reference to x*
bl	*blade*
h	*hub*
m	*meridional component*
0	*total*
r	*relative component*
t	*tip*
u	*tangential component*
1	*inlet of impeller*
2	*exit of impeller*

I. Introduction

A matter of timely concern is the harvesting of solar energy. In this aspect the smaller gas turbines are being sought after. The centrifugal compressor comes in as a good competitor to the axial flow machine, in that, compared to its axial counterpart, it is easier to produce and more compact. Of immediate concern is the development and design of the small compact compressors of high efficiency. Efficiencies of larger machines are of the order of 85%-95%; but however in the case of the smaller machines it becomes difficult in having matching diffusers for the highly efficient impeller.

Large amount of theoretical work have been reported widely across the globe; with both the RANS (Reynolds Navier Stokes) and CFD techniques suggesting improved design methods. The separation of the fluid on the suction side of the impeller vanes has been for long posing diffuser matching problems. Secondary flows in the impeller channels have been troubling aerodynamic calculations of the system. Splitter vanes and boundary layer fences have also been tried with little improvement.

Pfleiderer was the first perhaps, to enunciate the maintenance of the inlet angle of the impeller vanes to be 34 deg. to avoid shocks at inlet to the impeller.

Fig. 1 Centrifugal Compressor Impeller

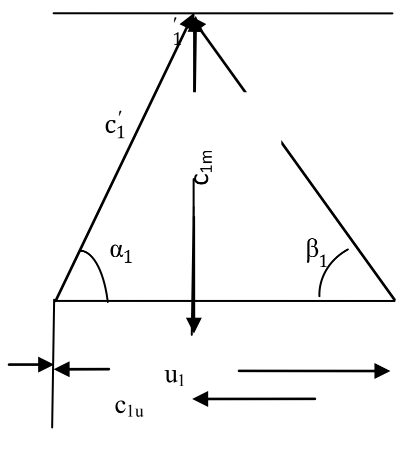

Fig. 2 Velocity Triangle at Inlet to Impeller

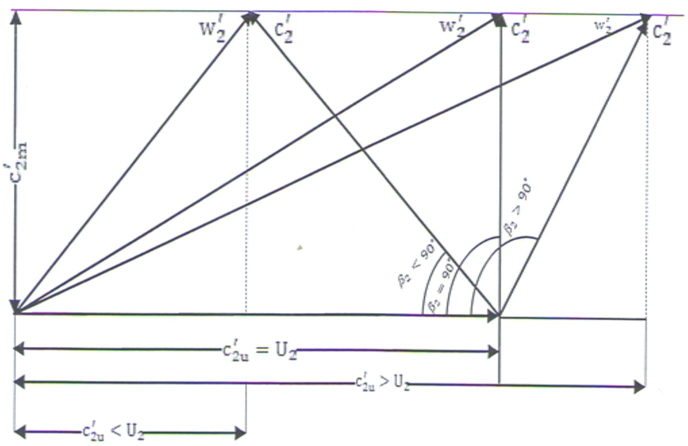

Fig. 3 Velocity Triangles at Exit of Impeller

Stanitz using relaxation methods to analyze the non-viscous two dimensional through –flow through radial and logarithmic spiral vaned impellers arrived at an equation similar to Stodola's σ

$$\sigma = 1 - \frac{0.63\pi \sin\beta_2'}{z[1 - \emptyset_2 \cot\beta_2']} \qquad (1)$$

Stanitz further observed that slip is unaffected by changes in impeller tip speed and compressibility, while for an impeller with constant cone angle, slip is only a function of the number of vanes.

Influence of Impeller Vane Exit Angle

It has been shown, that the energy transfer in a turbomachine impeller is essentially dependent on the magnitude of u_2 and c_{2u} for a given impeller tip speed, the blade specific work, W_{bl} is proportional to c_{2u} and therefore a function of β_2

$$W_{bl} = u_2 c_{2u} - u_1 c_{1u} \qquad (2)$$

In the absence of prewhirl, i.e

$\alpha_1 = 90^o$, or $c_{1u} = 0$,

$$W_{bl} = u_2 c_{2u} \qquad (3)$$

(3)

$$= u_2\,[u_2 - c_{2m} / \tan\beta_2] \qquad (4)$$

$$u_2 = \frac{c_{2m}}{2\tan\beta_2} \pm \sqrt{\left[\frac{c_{2m}}{2\tan\beta_2}\right]^2 + W_{bl}} \qquad (5)$$

more generally,

$$u_2 = \frac{c_{2m}}{2\tan\beta_2} \pm \sqrt{\left[\frac{c_{2m}}{2\tan\beta_2}\right] + W_{bl} + u_1 c_{1u}} \qquad (6)$$

As a consequence of Eqn. (4) it can be shown that the energy transfer per stage is primarily a function of u_2, c_{2m} and β_2

Fig.(4) shows the three types of impellers with different vane angles β_2'. It should be borne in mind that this β_2 is the fluid angle at vane exit. An impeller with a backward swept vane ($\beta_2' < 90^\circ$) would have a falling characteristic, a radial vane ($\beta_2' = 90^\circ$) would have a level characteristic and that with a forward swept vane ($\beta_2' > 90^\circ$) an increasing characteristic. It can also be explained with the help of Eqn (6) that for a given speed 'n', the size of the impeller would increase as β_2' decreases.

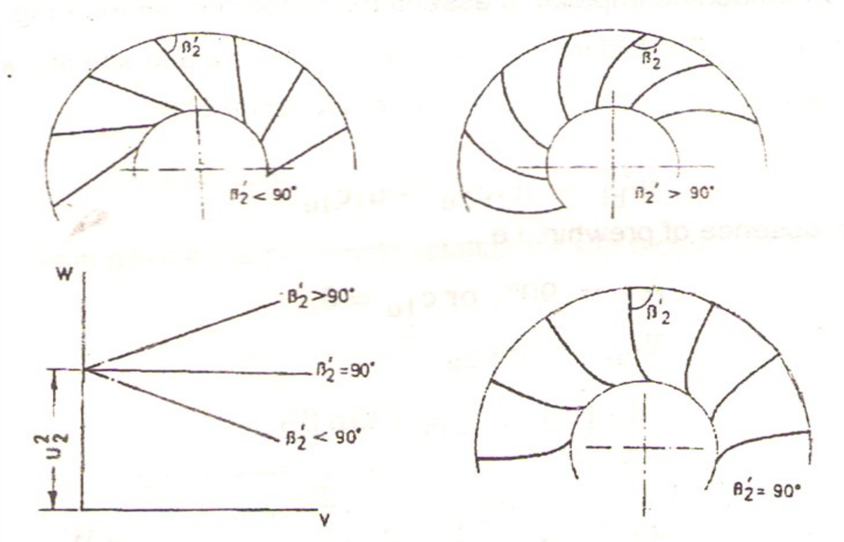

Fig. 4 Types of Vane Profiles

One Dimensional Considerations for Maximum Through –Flow

The desired maximum capacity of a centrifugal machine is mostly dependent on the maximum inlet area that could be provided. At higher speeds and capacities the performance of centrifugal impellers decay rather rapidly due to the formation of compression shocks. The inducer is the most important component that needs to be designed very carefully to avoid Mach number peaks at impeller inlet. An inducer imparts a solid body rotation to the fluid at inlet.

Assuming an uniform inlet velocity distribution namely, $c_{1m} \neq f(r)$, and the flow to be compressible, then the volume flow at inlet could be represented as

$$V = c_{1m}.\ A_1 = c_{1m}.\pi\{r_{lt}^2 - r_{lh}^2\} \tag{7}$$

$$c_{1m} = [w_{lt}^2 - \omega^2\ r_{lt}^2]^{1/2} \quad \text{when } c_{lu} = 0 \tag{8}$$

$$V = \pi[r_{lt}^2 - r_{lh}^2][w_{lt}^2 - \omega^2.r_{lt}^2]^{1/2} \tag{9}$$

Differentiating Eqn.(9) w.r.t. 'r_{1t}' and equating to zero for maximum volume rate of flow,

$$\frac{1}{\pi}\frac{\partial V}{\partial r_{1t}} = 2.r_{1t}[w_{lt}^2 - \omega^2.r_{lt}^2]^{1/2} - \frac{\omega^2 r_{1t}(r_{1t}^2 - r_{1h}^2)}{(w_{lt}^2 - \omega^2.r_{lt}^2)} = 0 \tag{10}$$

On rewriting,

$$w_{lt}^2 = \omega^2\ [\ 1.5\ r_{1t}^2 - 0.5\ r_{1h}^2] \tag{11}$$

so that, $\tan\ \beta_{1t} = \frac{c_{1m}}{u_{1t}} = 0.5\ [1 - [r_{1h}\ /\ r_{1t}]^2]^{1/2}$ (12)

on the other hand, if the flow is assumed to be compressible,

$$m = A_1.\ c_1.\ \rho_1, \text{ when } c_{1u} = 0,\ c_{1m} = c_1 \tag{13}$$

$$= A_1.\ \rho_0\ .\ a_0.\ M_1.\ \frac{\rho_{1.a_1}}{\rho_{0.a_0}} \tag{14}$$

$$= \pi.r_{1t}^2.\ [1 - \left(\frac{r_{1h}}{r_{1t}}\right)^2].\ \rho_0,\ a_0.M_1.[1 + \frac{k-1}{2}\ M_1^2]^{-\ (k+1)/2\ (k-1)} \tag{15}$$

Where

$$A_1 = \pi.r_{1t}^2.\ [1 - \left(\frac{r_{1h}}{r_{1t}}\right)^2]$$

$$\frac{\rho_{1.a_1}}{\rho_{0.a_0}} = \left[\frac{T_1}{T_0}\right]^{1/k-1} . \left[\frac{T_1}{T_0}\right]^{1/2}$$

$$= [1 + \frac{k-1}{2}\ M_1^2]^{-\ (k+1)/2\ (k-1)} \tag{16}$$

$$\frac{r_{1t}}{r_2} = \frac{M_{1rt}\cos\beta_{1t}}{u_2/a_0}\ .\ [1 + \frac{k-1}{2}\ M_1^2]^{-1/2} \tag{17}$$

$$\frac{m}{\rho_0 a_0 \pi}\left[\frac{u_2}{a_0}\right] = [1 - \{r_{1h}/\ r_{1t}\}^2]\ .\ r_2^2\{\frac{M_1[M_{1rt}.\cos\ \beta_{1t}]^2}{[1+\frac{k-1}{2}M_1^2]^{(3k-1)/2(k-1)}}\} \tag{18}$$

The mass flow is a maximum when the right hand side of Eqn.(15) becomes maximum.

Fig. (5) shows the Variation of inlet Mach no. as a function of Vane Inlet Angle and Inlet Relative Mach No

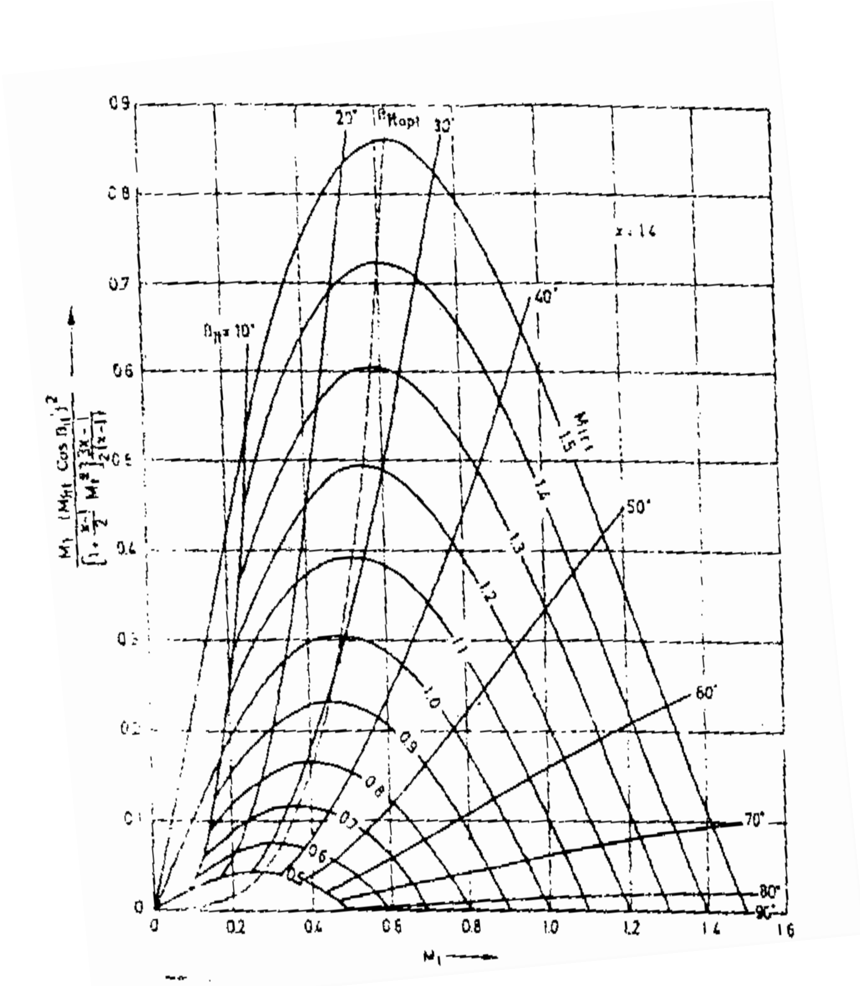

Fig. 5 Variation of inlet Mach no. as a function of Vane Inlet Angle and Inlet Relative Mach No

Stage temperature rise

The blade specific work per stage is given by

$$W_{bl} = u_2c_{2u} - u_1c_{1u}$$
$$= cp(T_{02} - T_{01}) \tag{19}$$

so that $\frac{T_{02}-T_{01}}{T_{01}} = \frac{u_2^2}{c_pT_{01}}\left[\frac{c_{2u}}{u_2} - \left[\frac{r_1}{r_2}\right]^2 \frac{c_{1u}}{u_1}\right]$ (20)

The sonic velocity referred to inlet stagnation conditions is given by $a_{01}^2 = (k-1).c_p.T_{01}$

Hence,

$$\frac{T_{02}-T_{01}}{T_{01}} = [k-1].\left[\frac{u_2}{a_{o1}}\right]^2\left[\frac{c_{2u}}{u_2} - \left[\frac{r_1}{r_2}\right]^2 \frac{c_{1u}}{u_1}\right] \tag{21}$$

with no inlet prewhirl, the stage signation temperature rise is essentially a function of the impeller tip speed and slip. The ratio [u_2 / a_{ol}] is termed the 'Mach Index' of the stage, and denoted as 'π_m'.

Mach number at impeller exit

The absolute Mach number at impeller exit can be expressed as

$$M_2^2 = \frac{c_2^2}{kRT_2}$$
$$= \frac{c_2^2}{T_{o1}}.\frac{T_{o1}}{T_{o2}}.\frac{1}{kR} \tag{20}$$
$$\frac{c_2^2}{T_{o1}} = [\emptyset_2^2 + \left[1 - \frac{\emptyset_2}{\tan\beta_2'}\right].\frac{u_2^2}{T_{o1}}$$
$$\frac{T_2}{T_{o1}} = \left[1 + \frac{\Delta T_o}{T_{o1}} - \frac{c_2^2}{2c_pT_{01}}\right] \tag{21}$$

Therefore, $M_2 = f\,[\frac{\Delta T_o}{T_{o1}}, \emptyset_2, \beta_{2,}\, c_p]$

The absolute impeller Mach number is therefore for a given media, a function of the flow co-efficient and the vane angle at exit.

Stage Pressure Rise

The stage temperature rise

$$[T_{03} - T_{01}] = \eta_1\,[T_{02} - T_{01}] \qquad (22)$$

$$[T_{02} - T_{01}] = \eta_1.\sigma.\,u_2^2 \,/\, c_p \qquad (23)$$

$$\frac{P_{02}}{P_{01}} = \left(\frac{T_{03}}{T_{01}}\right)^{\frac{K}{K-1}} \qquad (24)$$

$$= [1 + \eta_1.\sigma.\,u_2^2 \,/\, c_p\, T_{01}] \qquad (25)$$

Eqn.(25) necessarily implies that the blade tip speed of a high pressure ratio centrifugal compressor must be high and if non-radial, the vanes would be subjected to large bending stresses as a result of centrifugal forces.

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