Proceedings of the
International Conference on Mechanical Engineering 2003
(ICME2003) 26- 28 December 2003, Dhaka, Bangladesh

ICME03-FL-12

EXPERIMENTAL AND ANALYTICAL INVESTIGATIONS OF SLIP FACTOR IN RADIAL TIPPED CENTRIFUGAL FAN.

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ABSTRACT

For proper design of centrifugal fans it is essential to estimate the slip factor correctly. Several co-relations are used for estimating the slip factor which concludes that for a given specified fan the value of slip factor is constant & is dependent of impeller geometry only. But on the basis of fewer historical evidences, approach & experience this statement has been challenged and found to be partially correct and factual evidences were proved which suggested that the slip factor not only depends on the geometry of the impeller but also on the specific speed and flow rate. Moreover the value does not remain constant at any location for the exit of air through impeller blades. Also the experimental value is found to be considerably smaller (3 -12%) than the predicted values based on various correlations due to occurrence of possible discrepancies. The objective of present study is the assessment of theoretically correlated and experimentally evaluated slip factor at varied number of blades. Here the experimental value of the slip factor for a radial tipped centrifugal blower is determined at various selected test locations around the circumference of the impeller covering the entire width of impeller at varying number of blades. Further it is compared with various theoretical correlations and differences are investigated and analyzed. A special three-hole probe is designed, developed & calibrated for measuring and sensing the local velocity.

Keywords: Concept of Slip factor, Three-hole probe, Experimental slip factor compared with various correlations.

1. INTRODUCTION

Thermal turbo machines, which uses air as the working fluid (compressible) & energy transfer is from rotor to fluid are [1]:

	Outlet pressure range.
FAN	Upto 7 kpa.
BLOWERS	7 to 240 kpa.
COMPRESSORS	Above 240 kpa.

Occurrence of losses existing in fans and blowers are:

- Impeller entry losses.
- Leakage losses.
- Impeller losses
- Diffuser losses & volute casing losses.
- Disc friction losses.

For radial tipped centrifugal blowers, the flow enters the impeller axially & leaves radially. This conversion of axial flow to radial flow takes place within meridional region (fig.1) of impeller. It is the space between two consecutive blades of impeller and losses occurring within this region leads to impeller losses.

2. LITERATURE REVIEW

2.1 Concept of Slip Factor

The concept of slip factor is implied to impeller losses. It is defined as the ratio of the actual & ideal values of the whirl components at the exit of impeller. Mathematically,

$$\mu = {V'_{W2}} / {V_{W2}} \qquad (1)$$

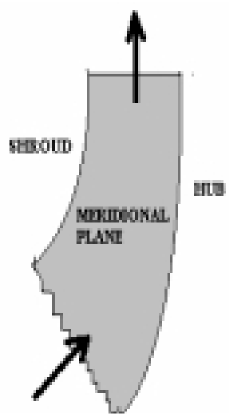

Fig.1 Meridional plane.

Therefore the slip velocity is given by,

$$V_s = V_{W2} - V'_{W2} = V_{W2}(1-\mu) \quad (2)$$

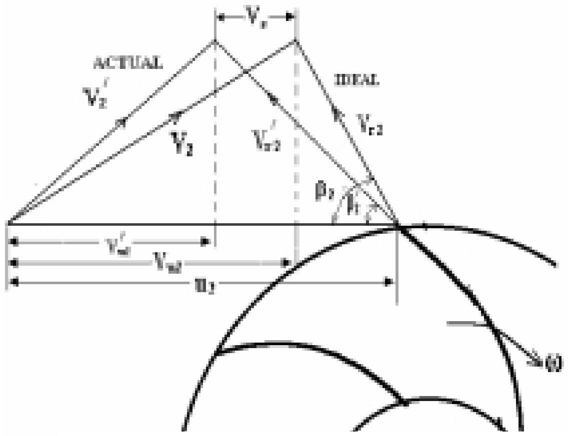

Fig.2 Ideal & Actual velocity triangles at exit.

The variation between the actual flow and assumed ideal flow exists due to [2], [3]:

i. Entry conditions.
ii. Finite thickness of blades
iii. Finite number of blades.
iv. Effect of friction.
v. Relative eddy flow.

This variation in the actual and ideal whirl components are suggested in various slip factor correlations viz: Stodola, Balje, Stanitz, Pfleiderers, Weisner, Bruno-Eck, Senoo-Nakase;etc. According to these researchers the major cause of slip factor are the ***relatives eddies*** generated within the meridional region that are dependent on the geometry of impeller.

Fewer correlations, which are used here for assessment purpose, are:

(a) ***Stodola's Equation:-*** The relative eddy is assumed to fill the entire exit section of the impeller passage. It is considered equivalent to the rotation of a cylinder of diameter d=2r at an angular velocity ω which is equal and opposite to that of the impeller.

$$\mu = 1 - \frac{\pi}{z} \frac{\sin \beta_2}{(1-\phi_2 \cot \beta_2)} \quad (3)$$

For radial tip: $\beta_2 = 90^0$ $\quad \therefore \mu = 1 - \dfrac{\pi}{z}$ (4)

The above expressions for the slip show that for a given geometry of flow the slip factor increases with the number of impeller blades. Along with this fact that the number of impeller blades is one of the governing parameters for losses and it should not be lost sight of.

(b) ***Stanitz's Equation:*** Stanitz suggests a method based on the solution of potential flow in the impeller passages for β_2 = 45^0 to 90^0. The slip velocity is found to be independent of the blade exit angle and the compressibility. This is given by

$$\mu = 1 - \frac{1.98}{z(1-\phi_2 \cot \beta_2)} \quad (5)$$

For radial tip: $\beta_2 = 90^0$ $\quad \therefore \mu = 1 - \dfrac{1.98}{z}$ (6)

(c) ***Balje's formula:*** Balje suggested an approximate formula for radial-tipped (β_2=90^0) blade impellers:

$$\mu = \left[1 + \frac{6.2}{z.n^{2/3}}\right]^{-1} \quad (7)$$

where, $\quad n = \dfrac{\text{impeller tip diameter}}{\text{eye tip diameter}}$

Thus the study of various correlations concludes ***"slip factor is function of impeller geometry alone and is not affected by flow rate."***

2.2 Historical Evidences

Above statement has been challenged and found to be partially correct.

(a) *J.F.PECK- 1951*[4] published an experimental study of the flow in a centrifugal pump. As a result of his study, he came to the conclusion that the slip factor must be a function not only of the impeller geometry, but also of the flow rate. According to him major cause for slip factor were ***relative eddies*** *as well as* ***back eddies***. Back eddies are dependent on flow rate. Unfortunately, the supporting experimental data presented by him were not immune to criticism. This has resulted in rather poor acceptance of PECK'S theory.

(b) *Sh.YEDIDIAH-1974* [5] carried out number of studies, which proved that PECK'S conclusion was really correct. He presented factual evidences, which proved that *the slip factor for a given impeller is not constant, but varies with the flow rate.* A new model of the flow through an impeller was established, which gave possible explanation to the observed discrepancies between the existing theory and reality.

(c) *S,MARUYAMA & T.KOIZUMI-1974* [6] experimentally determined slip factors for different kinds of impellers. A big discrepancy was recognized between the experimental values and the prediction based on various co-relations. They presented the experimental evidences and reasons of the practical discrepancies in the slip factors

2.3 Possible reasons & discrepancies observed for the occurrence of slip factor

a. Relative eddies.
b. Back eddies.
c. Geometry of impeller
   i. Mean blade loading.
   ii. Thickness of blade.
   iii. Finite number of blades.
d. Entry conditions.
e. Viscosity of the working fluid.
f. Effect of boundary layer growth.
g. Variation of effective blade camber line.
h. Separation of flow.
j. Friction forces on the walls of flow passages.
k. Boundary layer blockage.

3. EXPERIMENTAL SET UP

To calculate practically or experimentally the slip factor for radial tipped centrifugal fan by installing different number of blades, a Test set up was installed.

3.1 Main Requirements

Á Selection of radial tipped centrifugal fan.
Á Selection & designing of pressure probes for measuring slip factor.
Á Calibration of pressure probe. Providing traversing and sliding feed mechanism for pressure probe.
Á Selection and design of manometers required for pressure probe readings.

3.2 Specifications of fan and Conditions Set For Experiment [8]

Impeller diameter: 300 mm
Blade width: 74 mm
Max. Number of blades: 24
Type of blade: radial tipped.
Material: Acrylic.
Air delivered: 1500 m^3/ hr.
Total efficiency: 81% (at design conditions)
Nature of medium: atmospheric air.
Atmospheric conditions: Pressure: 1.01325 bar.
Temperature: 30^0C.
Electrical supply for motor:-
1 phase, ½ hp, AC motor giving 1280 rpm under load conditions.
Constant speed of motor to be maintained: 1280 rpm.

4 CALCULATIONS OF SLIP FACTOR

Practically, slip factor is defined as the ratio of actual exit whirl velocity to the theoretical exit whirl velocity. Theoretical (ideal) value of whirl velocity at exit of blade can be obtained analytically while the actual value of whirl component is obtained experimentally.

4.1 Analytical method for obtaining components of ideal velocity triangle

The main components are (fig.3):

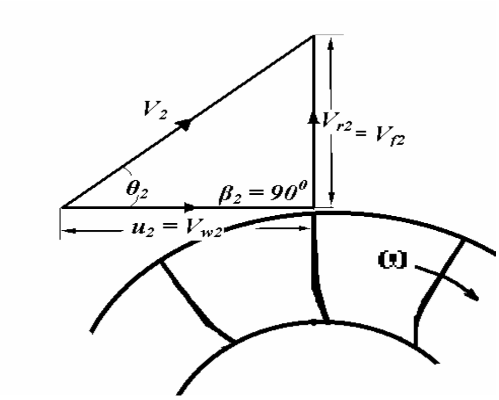

Fig.3 Ideal velocity triangle at exit.

The peripheral or the tangential velocity: $u_2 = \dfrac{\pi DN}{60}$.

Absolute velocity at the exit of blade: V_2
As the blade is radial: $б_2 = 90^0$ & $u_2 = V_{w2}$.
Relative or radial velocity of flow at the exit of blade: $V_{r2} = V_{f2}$.

Also, $V_{r2} = ű.u_2$

Now, ű = volume coefficient = $1 \big/ \sigma\delta^3$

Where, ů = speed coefficient = $\dfrac{0.379N\sqrt{Q}}{60H^{3/4}}$

& ŭ = diameter coefficient.

The inter relationship between speed coefficient and diameter coefficient is given by Cordier's theory for the selection of most favorable impeller size. From Cordier's diagram (fig.4), the value of diameter coefficient can be known if the value of speed coefficient is given and vice-versa. So, by determining the value of volume coefficient from speed coefficient and diameter coefficient, radial or relative velocity can be calculated.

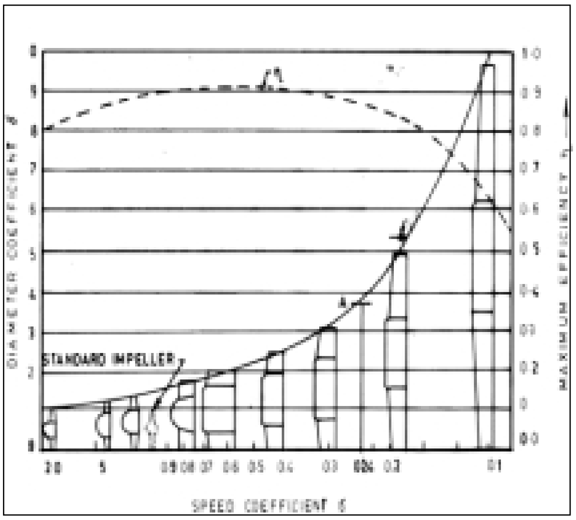

Fig. 4 Cordier's Diagram

(A–Denotes the parameters of impeller designed) Various Impellers Shown On σ - δ With Curve Of Maximum Efficiencies (Bruno Ing. Eck - 1975)

Thus, absolute velocity would be known by the expression. $V_2 = \sqrt{u_2^2 + V_{r2}^2}$ .Also, for radial tipped blades ideally, $V_{W_2} = u_2$. Thus by knowing all the components the theoretical velocity diagram can be constructed.

4.2 Experimental method for obtaining components of actual velocity triangle [9]

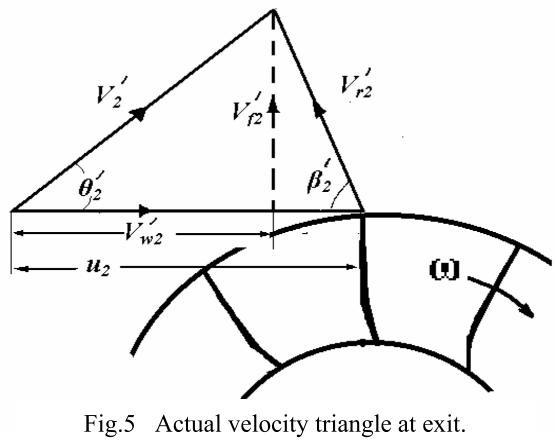

Fig.5 Actual velocity triangle at exit.

For constructing the actual velocity diagram and henceforth calculating the slip factor, actual whirl component of actual absolute velocity is to be determined which is given by, $V_{w2}' = V_2' \cos\theta_2'$.
Now, the actual absolute velocity at the exit can be calculated by, $V_2' = C_v \sqrt{\frac{2P_d}{\rho_f}}$. C_v = coefficient of velocity = 0.93 (calibration of the designed probe)
The actual absolute velocity as well as actual whirl angle is measured experimentally by determining the position of V_2' with the help of angular movement mechanism provided on the sliding fixture.
Thus, experimentally, the value of actual slip factor is obtained. As per the basic requirement for the experimentation, the probe (fig.6) must have following features: -

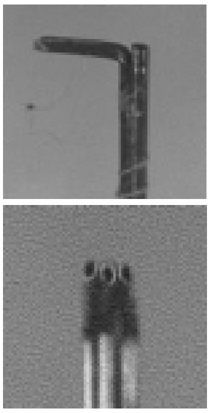

Fig.6 Views of designed three-hole probe.

ü Comparatively very small diameter tubes (0.6 to 0.8mm ID & 1 to 1.5mm OD) should be used.
ü Variation and disturbance in the flow velocity due to its insertion within the flow region must be very small or negligible.
ü Should be sensitive enough for flow pressure at any selected location.
ü Must be able to sense and determine the exact direction of flow.
ü Light in weight.
ü Highly rigid and strong for sustaining the high flow pressures within the blower.

4.3 Selection of test locations [9]

For experimentally measuring the absolute velocity, some test locations are to be predetermined along the profile of blower. The aspects that have to be taken into consideration while selecting the test-locations are as under:

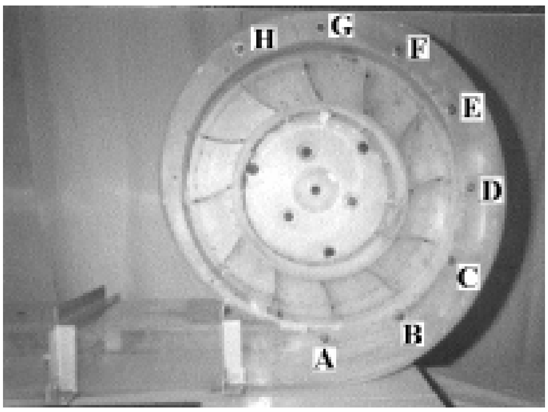

Fig.7 Selection of test location.

Table: 1 Selection of test locations.

Distance along the blade width. (mm)		0	15	30	37	45	60	75
		1	2	3	4	5	6	7
Main locations along the circumference of the impeller.		Sub location points along the blade width.						
Degree.	Location points							
0	A	A_1	A_2	A_3	A_4	A_5	A_6	A_7
30	B	B_1	B_2	B_3	B_4	B_5	B_6	B_7
60	C	C_1	C_2	C_3	C_4	C_5	C_6	C_7
90	D	D_1	D_2	D_3	D_4	D_5	D_6	D_7
120	E	E_1	E_2	E_3	E_4	E_5	E_6	E_7
150	F	F_1	F_2	F_3	F_4	F_5	F_6	F_7
180	G	G_1	G_2	G_3	G_4	G_5	G_6	G_7
210	H	H_1	H_2	H_3	H_4	H_5	H_6	H_7

No of Main Test Locations: 8.
No. Of Sub Test Locations: 56.

a. It must cover the max. circumferential region of the impeller. Because of the limitations of insertion of the probe, due to availability of flow-cross sectional region between the impeller and volute casing does restrict the selection of test location. Here some eight main test locations are selected (from A to H) (fig.7) an angular difference of 30^0 along the circumference of the impeller. These locations are drilled on the casing profile somewhat above the circumference of the impeller in order to avoid the collision of probe with the running impeller. The gap between the probe and the circumference of the impeller must as minimum as possible, of course, with prior precautions.
b. It must also cover the total width of the impeller or blade in order to study the effect of blade width on velocity profile as well as slip factor profile. Along the width some 7 sub-locations are selected with respect to the main test locations. Thus, each main test location along the circumference of running impeller there are 7 sub-locations along the width of blade.

5. OBSERVATIONS

Numbers of blades were varied to 12, 16 & 24 and probe readings were taken at all test locations along the width of the impeller. A sample reading at ***location D*** as been shown below for ***number of blades 16.***

Table: 2 Observation table for location D

Location	$H_{dynamic}$	P_d	V_2'	Slip factor
	mm of water	N/m^2	m/sec	μ
D_1	18.64	182.65	16.47	0.693
D_2	21.48	210.47	17.68	0.744
D_3	23.3	228.31	18.41	0.775
D_4	24.07	235.85	18.71	0.788
D_5	23.04	225.76	18.31	0.771
D_6	20.71	202.93	17.36	0.731
D_7	15.53	152.17	15.03	0.633

The variation of angles for actual absolute velocity [V_2'] with respect to the tangential velocity [u_2] is 35-39^0. Therefore, by taking the average value, $\boldsymbol{\theta_2' = 37^0}$. The average practical slip factor is **0.734**

6. RESULTS & DISCUSSIONS

6.1 Comparison of Average Practical Slip Factor at Different Locations

Here the practical value of slip factor is compared with some of the standard values obtained by utilizing various co-relations viz. BALJE, STODOLA & STANITZ. Further the average of all the averages of slip factors at all location is determined (fig 8) to give the final practically concluded slip factor for ***number of blades:-16.***

Table: 3 Results for all test locations.

Deg.	Location	Av. Slip factor	Prac. slip factor	Standard Values Using Co-Relations		
				Balje	Stodola	Stanitz
0	A	0.821	0.771	0.78	0.804	0.876
30	B	0.756				
60	C	0.743				
90	D	0.734				
120	E	0.735				
150	F	0.792				
180	G	0.814				
210	H	0.773				

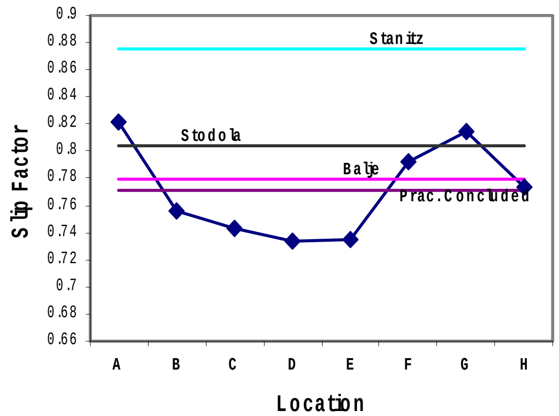

Fig.8 Slip factor vs. Location for 16 no. of blades

6.2 Actual velocity or slip factor profile obtained at the exit of impeller at all test locations along the width of impeller.

Sample profile obtained at location D for no. of blade 12, 16 & 24 is shown herewith (fig.9).

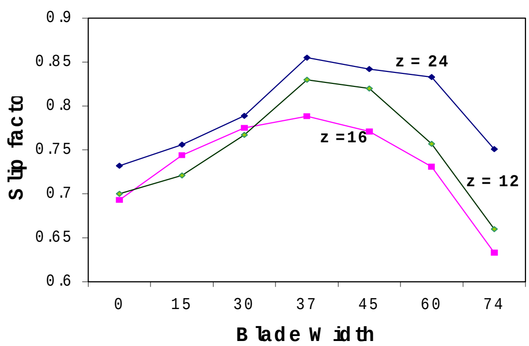

Fig.9 Slip-factor profile at location D
[No. of blades 12, 16 & 24]

6.3 Comparison of the practically concluded slip factor with various co-relations for radial tipped centrifugal blower by changing number of blades. (fig.10)

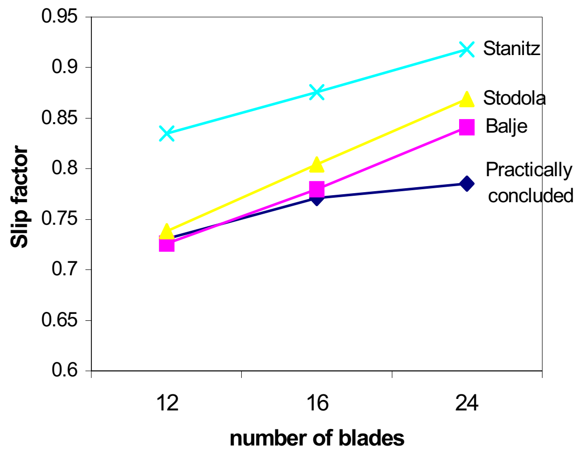

Fig.10 Slip factor vs. number of blades

6.4 Discussions

(a) As the number of blade increases, the slip factor increases.

(b) The curve of slip factor that has been obtained experimentally is less steep as compared to all the correlated curves.

(c) It indicates that if the number of blades reaches to infinity, the slip factor steeps up to unity.

(d) As the number of blades increases, the meridional region is reduces. Thus the deviation of the exit absolute velocity vector is minimized and hence can be efficiently sense by the designed probe.

(e) Again the number of blades has to be optimized, cause increase in the number of blades causes increase in the solidity ratio and hence results in the reduction of discharge. Now, high discharge is the basic need or the ultimate output of the blower, otherwise it's just an ordinary machine giving just ordinary flow of air with no blowing effect.

(f) At low speeds the slip factor would reach to unity, but at high speeds as required for high discharge, the slip factor is reduced.

(g) The experimental values obtained are somewhat lower than the correlated values that lead to certain possible causes of discrepancies which has been predicted as under:

i. ***Geometry of impeller:*** impeller designed for a large specific speed has a large inlet to exit radius ratio and the curvature of the shroud is large to turn the flow from axial to radial. Therefore unless the impeller is carefully designed the flow in meridional plane separates from the shroud.

ii. ***Blockage of flow passage*** due to the boundary layer growth.

iii. ***Variation of effective camber lines*** due to accumulation of the boundary layer on the blade suction surface.

iv. ***Frictional forces*** action on the wall of the flow passages inside the impeller.

v. ***Separation of flow*** within the flow passage: as the flow in the impeller is a retarded flow, the flow may separate from suction surface of the blades creating a large separation zone on the suction surface and the direction if flow is considerably different from the blades.

vi. ***Non uniform velocity distribution*** at the exit of impeller: the velocity profile at the exit of the impeller must be parabolic, but if it is not so, non uniformity acts because the mass averaged momentum flux is large than that of the uniform flow due to blockage.

7. CONCLUSION

Slip factor along the blade width is not constant because of boundary layer blockage & friction force along the flow walls. Slip factor along the radial circumference of impeller is not constant because the effect of volute casing has its impact.

Experimental curves & various correlated curves obtain for slip factor for varying number of blades, when compared, indicates that:

- Slip factor increases as the no. of blades increases.
- The experimental curve is less steep as compared to various correlations, because the correlated curve hasn't considered the effects of blade width, volute casing & fluid flow.
- The experimental slip factor is about 3 to 12% lesser when compared with various correlations.

Thus in a nut shell, it can be concluded that, **"Slip factor is the concept which is dependent on impeller geometry, flow rate, specific speed and various other predicted and analyzed discrepancies….!"**

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7. NOMENCLATURE

Symbol	Meaning	Unit
V_2, V_2'	Ideal & Actual absolute velocity at exit	m/sec
V_{w2}, V_{w2}'	Ideal & Actual whirl velocity at exit	m/sec
V_{f2}, V_{f2}'	Ideal & Actual flow velocity at exit	m/sec
V_{r2}, V_{r2}'	Ideal & Actual relative velocity at exit	m/sec
β_2, β_2'	Ideal & Actual blade angle at exit	Deg.
u_2	Peripheral velocity	m/sec
D	Diameter of impeller	m
N	Rotation of impeller	rpm
Q	Discharge.	m^3/s
H	Net static head.	m of air
P_d	Dynamic pr. $= gH_d(\rho_m - \rho_f)$	N/m^2
H_d	Dynamic pressure head $= (H_{stag.} - H_{static})\sin\lambda$	m of water
λ	Angle of inclination of manometer	Deg.
ρ_m	Density of manometric fluid (water) = 1000	Kg/m^3
ρ_f	Density of working fluid (air) = 1.165 at P_{atm} & temp. 30^0C	Kg/m^3
μ	Slip factor	-
z	Number of blades.	-
ϕ_2	Flow coefficient	-
ρ	Volume coefficient.	-
σ	Speed coefficient.	-
δ	Diameter coefficient.	-
C_v	Coefficient of velocity	-