COMPUTER AIDED DESIGN OF RADIAL TIPPED CENTRIFUGAL BLOWERS AND FANS

Nitin N. Vibhakar* and S. A. Channiwala**
*Sarvajanik College of Engg. & Tech., Athwalines,
Surat - 395 001, Gujarat, India.
** S. V. R. College of Engg. & Tech., Ichhanath,
Surat - 395 007, Gujarat, India.

Abstract There exist few design methodologies for centrifugal blowers and fans in the literature. However unified design methodology particularly for radial tipped centrifugal blowers and fans is not easily traceable in the literature available. This work is an attempt to present the extract of design methodologies for radial tipped centrifugal blowers and fans in a unified way. A generalized computer program for sizing and performance evaluation of centrifugal blowers and fans is developed during the course of this work. The program is a highly versatile one and needs only speed of rotation, discharge and static pressure rise across the stage as the input parameters and gives complete design of impeller, blade profile and volute casing etc. as well as performance parameters and various losses as the output. Using this program, the generalized design and performance charts and loss distribution charts have been developed, which may be used as design and performance evaluation tool for any kind of radial tipped centrifugal blowers and fans.

Keywords: Centrifugal blowers and fans, Design, Computer program, Generalized charts.

INTRODUCTION

The pioneering design work in the unification treatment of all Turbomachines is Spannhake's "Centrifugal pumps, turbines and propellers" (1934). While better known successor is 'Wislieenus' by " Fluid Mechanics of Turbomachinery" (1945)...
{Csanady, 1964}. Professor W. J. Kearton presented his findings on influence of number of impeller blades on the pressure generated. Stodola developed the useful method to estimate slip factor.

Austin H. Church {Church ,1962} has probably made the first attempt to compile the design methodology for pumps and blowers. While Eck Bruno extended the work of Church and provided more detailed design methodology for centrifugal, axial and cross flow fans and stated that the optimum number of blades of a radial impeller can only be truly ascertained by experiments ..... { Bruno Ing. Eck , 1975} .Mrs. Bela Mishra studied the existing design methodology and attempted to present unified design methodology, for the radial tipped centrifugal blower.{Mrs. Mishra ,1997}. However the experimental validity of this design methodology was not been evidenced.

The survey of literature clearly indicates that the lacuna exists towards the availability of systematic design procedure, which has been validated through experiments. Hence an attempt is made to present unified design procedure which has been validated through experiments. The general design procedure is adopted as per the methodology outlined by Eck Bruno and streamlined by Mrs. Bela Mishra. Profile of the blade is designed as per the methodology suggested by Austin Church . The no. of blades are estimated as per the empirical formulas suggested by Eck Bruno, Pfleiaderer and Stepanoff { Bruno Ing. Eck, 1975}. After calculating the optimum no. of blades by empirical formulas, the authors had experimentally verified them in the steps of 8, 12, 16 and 24 { Channiwala S. A. , Vibhakar N. N. , 1999 }.

The speed, diameter, pressure and volume coefficients are used to calculate actual geometrical parameters. Velocity triangles for inlet and outlet conditions are established by calculating peripheral, relative and absolute velocities accompanied with flow and blade angles. Volute casing for efficient diffusion of flow is designed considering free vortex flow { Bruno Ing. Eck , 1975, Yahya S M - 1964 }.

Taking the weight flow of air per unit of time passing at a point in a blower or fan is constant when the flow is steady. The volume flow will not be constant since the specific weight varies with changes in temperature and pressure of air. The dimensions of the air passage are calculated in accordance with these variations in the volume flow. {Church Austin H -1962 **}.**

*Email: v_nitin1@rediffmail.com

DESIGN OF RADIAL TIPPED CENTRIFUGAL BLOWER

{ Church Austin H **-**1962**,** Yahya S. M.–1964, Bruno Ing. Eck - 1975**,** Mrs. Mishra –1997, Vibhakar N. N. 1998. **}**

Rated Parameters :-

The following information in respect to an electrically driven centrifugal blower used in SDS-12 texturising machine is furnished to take as sample design conditions.

a) Size : 225 to 325 mm impeller dia preferred.

b) Electric Supply Details : 3 phase induction motor with DOL starter having 2950 RPM at no load condition.

c) Air Suction Pressure At Blower Inlet:-1000 N/m^2 (gauge static pressure)

d) Air Delivery Pressure : + 150 N/m^2 static

e) Discharge Required : 25 m^3/min. = 0.417 m^3/sec

f) Nature Of Medium : Atmospheric air

g) Atmospheric Conditions :

Atmospheric Pressure (P_a) =1.01325×10^5 N/m^2

Atmospheric Temperature = 30 ^oC = 303 ^oK

h) Altitude Of The Place Where The Blower Is To Be Installed : 10 m from sea level.

I) Whether Guards Are Required : Not required.

Co-efficients :-

Speed coefficient σ can be found out by the relation,

$\sigma = (0.379 * N_1 * Q^{1/2}) / (H)^{3/4}$

Next is to find the value of diameter coefficient δ from given Cordier's diagram. This can be known if the value of speed coefficient is known and vice versa.

The pressure coefficient is the ratio of pressure head rise to the peripheral velocity head i.e.

$$\varphi = [(\Delta p) * 2] / [\rho * (u_2)^2]$$

The ratio of actual volume discharge to theoretical volume discharge is known as volume coefficient, formulated as

$\phi = Q / [(\pi/4) * (d_2)^2 * u_2]$

Impeller Outlet and Inlet Diameter (d_2 & d_1) : -

As per basic equation derived,

$$d_2 = \sqrt[3]{[240 * Q *] / [(\pi)^2 * \phi * N]}$$

For the free vortex flow, ECK Bruno has given the following relation.

$$d_1 / d_2 = 1.2 * \sqrt[3]{(\phi)}$$

The flow geometry at the entry and exit of a turbomachine stage can be described by the velocity triangles at mentioned stations.

To find blade profile, there is no specific method available which relates the effect of blade curvature to blower efficiency, although it has great influence on the same. While designing, it should be considered that the passage should be short enough to reduce the frictional loss and there should be gradual changes in cross sectional area to avoid turbulence losses. There are two suggested methods given by A.H. Church for blade profile construction.

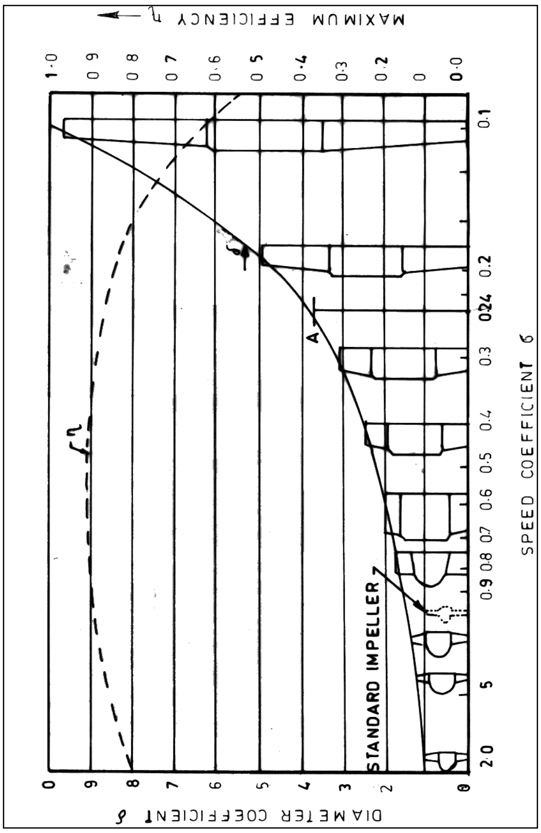

Fig.1 Cordier's Diagram

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

1. By tangent of circular arcs method. (The impeller is arbitrarily divided in to a number of concentric rings between (r_1 and r_2).
2. By polar Coordinates method.

Here we have selected the polar coordinates method for design purpose.

Design Of Volute Casing : -

For majority of centrifugal blowers, the higher cost and size that resulted by employing diffuser, overweigh its advantage. So most of single stage centrifugal impeller discharges directly into the volute casing, where some static pressure recovery can also occur.

Two most widely used methods of volute design are as;

1. Free vortex design.
2. Constant mean velocity design.

During this course of work, free vortex design is followed.

Number Of Blades : -

ECK Bruno has recommended the following relation,

$$Z = (8.5 \sin \beta_2) / (1- d_1/d_2)$$

while Pfleiaderer has recommended;

$$Z = 6.5 ((d_2+d_1)/(d_2-d_1)) * \sin ((\beta_{2} + \beta_2)/2)$$

and Stepanoff has suggested as ;

$$Z = (1/3) \beta_2$$

LOSSES, POWER AND EFFICIENCY

The various significant losses are briefly described below along with its formulation at design point as well at no flow condition and finally the total efficiency of the designed blower can be calculated.

Impeller Entry Loss :-

$dp_1 = K_1 (\rho /2) C_1^2$ (The value of K_1 = 0.15 to 0.25)

Flow Friction Loss : -

$$dp_2 = \frac{\{C_f*\rho*u_2^2) / 2 \}* (d_1 /d_2)^2 * ((1+V_x) / 2)^2 *2* [Z *l* b_m /d1*b1) +(\pi*d_2^2 / 4*d_1*b_1)* \{1-(d_1 / d_2)^2 \}]}{\pi* \sin \beta_1 *\cos^2 \beta_1}$$

The pressure loss due to retardation of flow given as ,

$dp_3 = K_3 *(\rho/2*g)* u2^2* (d_1/d_2)^2 *\{ (1- V_x^2)/\cos^2\beta_1) \}$

where C_f = 0.0040 to 0.0045 and K_3 = 0.1 to 0.2

Shock Loss :-

$d_{p4}= K_4 *(\rho/2) * (C_{x)})^2$

where ($K_4 = 0.03+ 0.6* (\beta_2/90)$) or K_4 = 0.7 - 0.9.

Volute Casing Loss :-

$d_{p5} = (\rho / 2) x (C_3^2 - C_4^2)*K_5$ (where K_5 = 0.2 to 0.3)

Disc Friction Loss :-

$P_{df} = K_6 *\rho *d_2^2* u_2^3$ (where K_6 = 1.1 to 1.2)

Clearance And Leakage Loss :-

$dQ = 3.89 * Q / u2 * (d_1/d_3)^2$

Power Required :-

$P = (((dQ + Q) (d_{p1} + d_{p2} + d_{p3} + d_{p4} + d_{p5}))/ 75) + P_{df}$

Efficiencies :-

The impeller generates the head in a centrifugal blower . The rest of the parts contribute nothing to the head but invites losses like hydraulic, mechanical and leakage. Considering all respective losses in the blower,

Hydraulic Efficiency :-

$\eta_{hyd} = \Delta P / (\Delta P + d_{p1} + d_{p2} + d_{p4} + d_{p5})$

Volumetric Efficiency :-

$\eta_{vol} = Q / Q + dQ$

Total Efficiency :-

$\eta_t = \eta_{hyd} * \eta_{vol}$

COMPUTER PROGRAM FOR THE DESIGN OF CENTRIFUGAL BLOWER AND FAN

The computer software is developed to rationalize the design procedure of the radial tipped centrifugal blower as outlined in previous design steps. The software is written in "TURBO C" language and is in the most generalized format. The same software can be modified for the design of multistage centrifugal blowers and fans. The basic data one requires while designing the blower and fan with the help of this software are - discharge, speed of the motor , the static pressure at suction and delivery sides and diameter coefficient from Cordier diagram.

The software running for rated parameters and results obtained as given below are self explanatory.

Designed Parameters

[A] Input Parameters

[i] Discharge : $Q = 0.417\ m^3 / Sec$
[ii] Speed Of Rotation : N = 2950 rpm
[iii]Static Pressure At Suction:$(P_{suc})_{Static}$ = - 1000 N/ m^2
[iv] Static Pressure At Delivery : $(P_{del})_{Static}$ = 150 N/ m^2
[v] Diameter Co-Efficient : δ = 2.7

[B] Output Parameters

[1] Static Pressure Gradient : $(\Delta P)_{Static}$ = 1150 N / m^2
[2] Inlet Atmospheric Pressure : P_{inlet} = 101325 N /m^2
[3] Room Temperature : T_1 =303 ^oK
[4] Density : (ρ) = 1.165 Kg/m^3
[5] Static Head : H_{static} = 100.609 m
[6] Speed Co-Efficient : ε = 0.379
[7] Diameter Co-Efficient : δ = 2.7
[8] Pressure Co-Efficient : φ = 0.956
[9] Volume Co-Efficient : Φ = 0.134
[10] Impeller Outlet Diameter : D_2 = 0.295 m
[11] Impeller Width At Exit : B_2 = 0.0737 m
[12] Area Of C/S At Impeller Exit : A_2 = 0.068 m^2
[13] Relative Air Angle At Exit : β_2 = 90 ^{o}
[14] Absolute Air Angle At Exit : α_2= 7.643 ^{o}
[15] Peripheral Velocity At Exit : U_2 = 45.538 m/Sec
[16] Radial Component Of Absolute Velocity At Exit : C_{f2} = 6.108 m/Sec
[17] Relative Velocity At Exit : W_2 = 6.108 m/Sec
[18] Absolute Velocity At Exit : C_2 = 45.946 m/Sec
[19] Impeller Inlet Diameter : D_1 = 0.182 m
[20] Impeller Width At Entry : B_1 = 0.0737 m^2
[21] Area Of C/S At Impeller Entry : A_1 = 0.0422 m^2
[22] Relative Air Angle At Entry : β_1 = 19.338 ^{o}
[23] Absolute Air At Angle At Entry : $\alpha_1 = 90^o$
[24] Absolute Velocity At Entry : C_1 = 9.877 m/Sec
[25] Radial Component Of Absolute Velocity At Entry : C_{f1} = 9.877 m/Sec
[26] Peripheral Velocity At Entry : U_1 =28.160 m/Sec
[27] Relative Velocity At Entry : W_1 = 29.842 m/Sec
[28] Diameter Of Volute Base Circle : D_3 = 0.305 m

[29] Tangential Comp. Of Absolute Vel. At Volute Entry : C_{theta3} = 45.538 m/Sec
[30] Volute Width : B_3 = 0.169 m
[31] Area Of Volute At Exit : A_{4max} = 0.011 m^2
[32] Absolute Velocity At The Exit Of Volute : C_{4max} = 37.927 M/Sec
[33] Number Of Blades : Z = 16
[34] Impeller Entry Losses : D_{p1} = 8.525 N/m^2
[35] Flow Friction Losses : D_{p2} = 11.935 N/m^2
[36] Retardation Losses : D_{p3} = 49.708 N/m^2
[37] Shock Losses : D_{p4} = 75.423 N/m^2
[38] Clearance Losses : dQ = 0.0932 m^3/Sec
[39] Disc Friction Losses : P_{df} = 0.01053 kW
[40] Losses In Volute Casing : D_{p5} = 74.020 N/m^2
[41] Power Required For The Blower : P =1.5043 kW
[42] Diameter Of Shaft : D_s = 0.00709 m
[43] Hydraulic Efficiency : E_{hyd} = 83.97 %
[44] Volumetric Efficiency : E_{vol} = 81.74%
[45] Total Efficiency : E_{Tot} = 68.64 %

Blade Profile Design :-

Radius	Beta	dr	Dtheta	Theta
(m)	(degree)	(m)	(degree)	(degree)
0.0912	19.1	0.0057	0.00	0.00
0.0966	26.2	0.0057	8.56	8.56
0.1023	33.3	0.0057	5.82	14.38
0.1079	40.4	0.0057	4.18	18.56
0.1136	47.5	0.0057	3.08	21.64
0.1192	54.6	0.0057	2.28	23.91
0.1249	61.6	0.0057	1.67	25.58
0.1305	68.7	0.0057	1.18	26.77
0.1362	75.8	0.0057	0.78	27.55
0.1418	82.9	0.0057	0.44	27.99
0.1475	90.0	0.0057	0.14	28.14

Volute Casing Design :-

Value of Theta degree	Volute Radius (m)	Value of Theta degree	Volute Radius (m)
0	0.1525	210	0.1876
30	0.1571	240	0.1932
60	0.1618	250	0.1990
90	0.1666	300	0.2050
120	0.1717	330	0.2112
150	0.1768	360	0.2175
180	0.1821		

RESULTS AND DISCUSSIONS

Fig. 2, Fig. 3 and Fig. 4 gives generalized charts for sizing the impeller outside and inlet diameter and impeller width of blower/fan as a function of speed with air-power (Delta P . Q) as a parameter.
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While Fig. 5 represents the picture of total pressure losses occurring in impeller and volute casing , Fig. 6 presents leakage loss in percentage of Q and Fig. 7 for total efficiency as a function of speed with air-power (Delta P * Q) as a parameter.

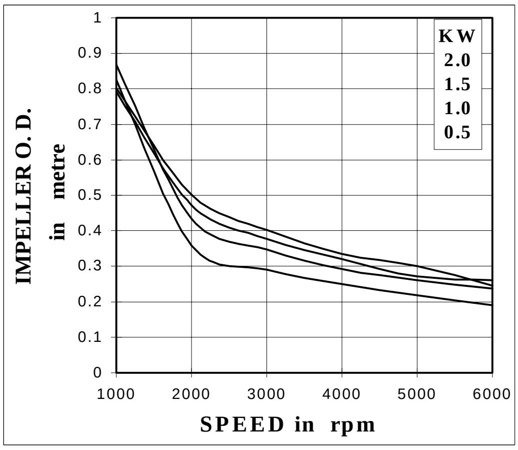

Fig. 2 Impeller O. D. Vs Speed

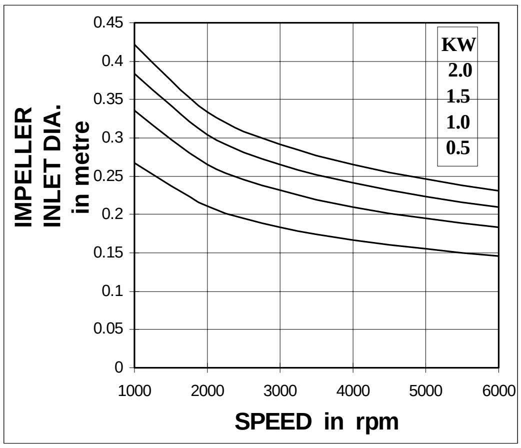

Fig. 3 Impeller I. D. Vs Speed

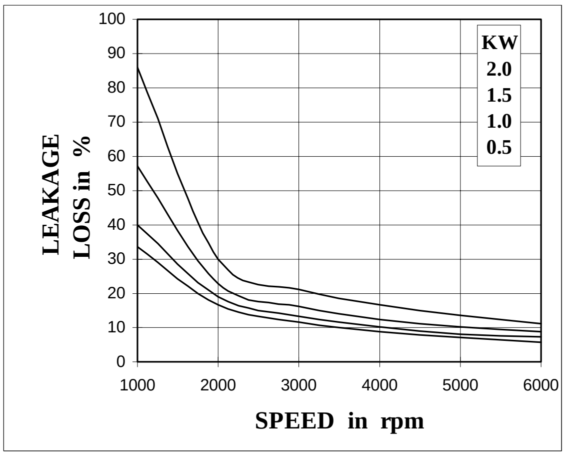

Fig. 4 Impeller Width Vs Speed

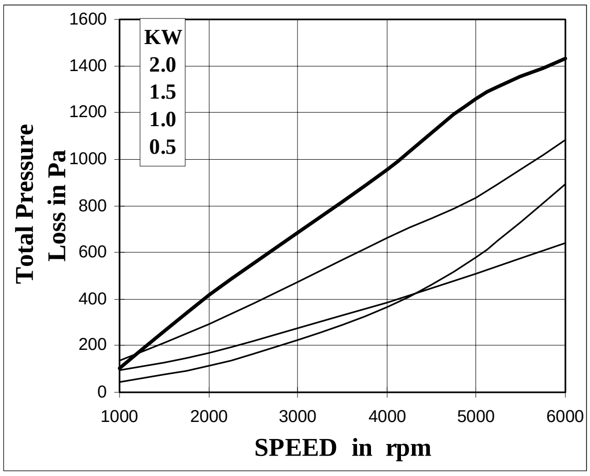

Fig. 5 Total Pressure Loss Vs Speed

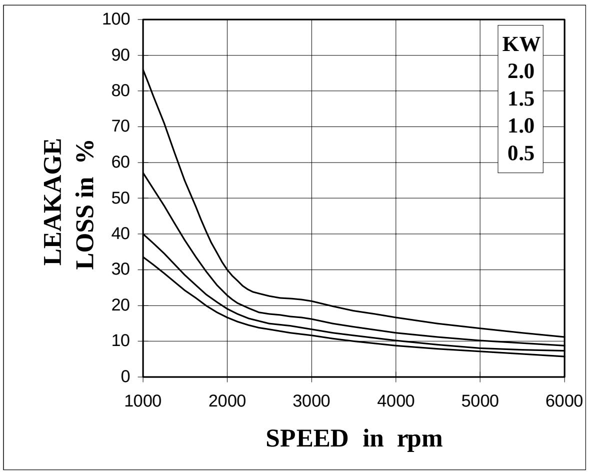

Fig. 6 Leakage Loss Vs Speed

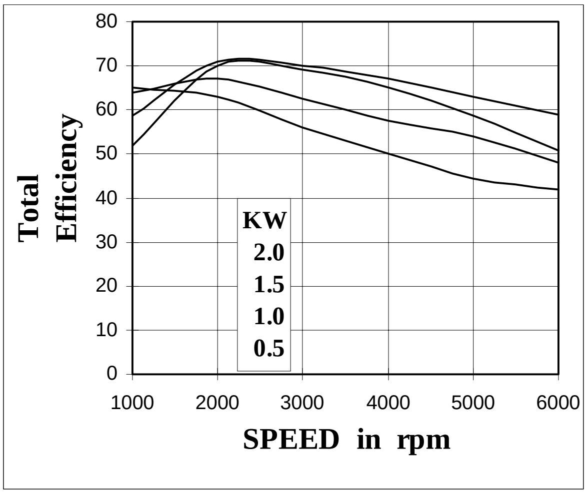

Fig. 7 Total Efficiency Vs Speed

CONCLUSIONS

The design methodology suggested by different researchers differs widely.

This work presents unified design methodology for radial tipped centrifugal blower / fan.

Presented program is extremely useful in sizing and to get generalized charts for the design & performance for radial tipped centrifugal blower or fan within the specified range.

NOMENCLATURE

α	Absolute air angle	degree
δ	Diameter Coefficient	
φ	Pressure Coefficient	
σ	Speed Coefficient	
ϕ	Volume Coefficient	
β	Relative air angle	degree
ρ	Density	kg/m^3
(P_{suc}) static	Static Pressure at suction	N/m^2
(P_{del}) static	Static Pressure at delivery	N/m^2
η_{hyd}	Hydraulic efficiency	%
$\eta_{vol.}$	Volumetric efficiency	%
η_{tot}	Total efficiency	%
A	Cross sectional Area of impeller	m^2
b	Blade width	m
C	Absolute velocity	m/sec
C_d	Coefficient of discharge	
C_f	Radial component of absolute velocity	m/sec
C_m	Mean velocity	m/sec
d	Impeller diameter	m
d_{p2}	Flow friction losses	N/m^2
d_{p3}	Retardation losses	N/m^2
d_{p1}	Impeller entry loses	N/m^2
d_{p4}	Shock losses	N/m^2
d_{p5}	Losses in volute casing	N/m^2
d_s	Diameter of shaft	m
dQ	Clearance & leakage losses	m^3/sec
g	Gravitational acceleration	m/sec^2
m	mass flow rate	m^3/sec
N	Speed of Rotation	rpm
N_{imp}	Disc friction losses	kW
P	Power required for the blower	kW
p_a	Atmospheric Pressure	N/m^2
Q	Discharge	m^3/s
T_a	Room Temperature	K
u	Peripheral Velocity	m/sec
w	Relative velocity	m/sec
Z	Number of blades	

Note: -

(a) Suffix 1 with the script indicates inlet condition at the impeller.
(b) Suffix 2 with the script indicates exit condition at the impeller.
(c) Suffix 3 with the script indicates inlet condition at volute base circle.
(d) Suffix 4 with the script indicates exit condition at volute outlet .

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