Design of 5 kW Radial Type Centrifugal Blower (Impeller)

AYE MYA THANDAR[1], PHYOE MIN THAN[2], NANG SEIN MYA[3], PA PA MINN[4]
[1] *Department of Mechanical Engineering, Technological University (Magway), Myanmar*
[2] *Department of Mechanical Engineering, Technological University (Hmawbi), Myanmar*
[3] *Department of Mechanical Engineering, Technological University (Myeik), Myanmar*
[4] *Department of Mechanical Engineering, Technological University (Panglong), Myanmar*

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. In this research paper is attempted to design a single stage radial type centrifugal blower for used in required industrial area. In this thesis, the bower is designed to provide low volume, high pressure air for cooling, ventilating and exhaust system that handle dust, materials or corrosive fumes. This centrifugal blower can produce 20 in of w.g(0.7396 psi) and 2l2l ft^3/min (60 m^3/min; of air at 60°Fand design speed 3450 rpm. In this research, the blower is designed air horsepower is 7 a. hp (5.2 kW) and the brake horsepower is 10 b. hp (7.5 kW). After choosing the design data from catalogue, the results of impeller inlet and outlet dimensions and vane angle could be obtained.

Indexed Terms: Centrifugal blowers, fans, Design, power, Vane

I. INTRODUCTION

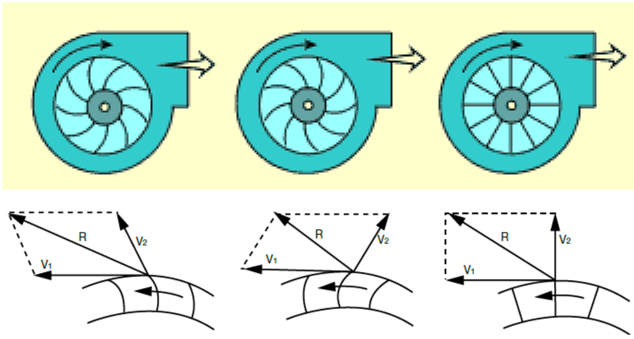
Machine that works on a flowing fluid is called a pump, blower, or compressor. A large number of blowers for high pressure applications are the centrifugal type. It consists of an impeller which has fixed between the inner and outer diameters the functions of blower are to increase the pressure of the air, to provide conditions favorable for combustion and expansion of the hot gases through the engine.

Centrifugal blowers are fundamentally high speed machines. The recent advances in steam turbine, electric motor, and high speed gearing design have greatly increased in their usage and application. A centrifugal pump or blower consists essential of one or more impellers equipped with vanes, mounted on a rotating shaft and enclosed by a casing. Fluid enters the impeller axially near the shaft and has energy. Kinetic and potential, imparted to it by the vanes. As the fluid leaves the impeller at a relatively high velocity, it is collected in a volute or series of differing passage with transforms the kinetic energy into pressures. This is of course, accompanied decrease in the velocity. After the conversion is accomplished the fluid is discharged from the machine. All rot dynamic machines have a rotating part called the impeller, through which the fluid flow is continuous. The direction of fluid in relation to the plane of impeller rotation distinguishes different classes of roto dynamic machines. One possibility is for the flow to be perpendicular to the impeller and, hence, along its axis of rotation. Machines of this kind are called axial flow machines. In centrifugal machines (sometime called radial flow), although the fluid approaches the impeller axially, it tums at the machine's inlet so that the flow through the impeller is in the plane of the impeller rotation. Mixed flow machine constitute a third category. They derive their name from the fact that the flow through their impellers is partly axial and partly radial. Radial flow impellers are normally provided with a front shroud and back shroud being called closed type. Impeller are sometimes built without the front shroud and called open type which is suitable for handling solid suspended liquids.

II. DIFFERENT TYPE OF BLOWER

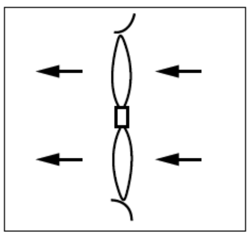
Centrifugal blower may be classified into three basic types according to blade configuration:
1. Forward curve
2. Backward inclined
3. Radial or straight blade

Each type has its own application range and limits. Modifications of these basic types include radial tip, mixed flow, and tangential flow.



1.a 1.b 1.c
Forward Curve Backward Curve Radial

Fig 1. Wheel Vector Diagrams



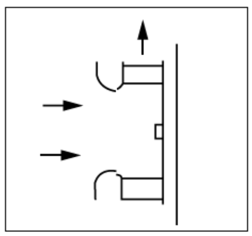


Fig.2a: Axial Flow Fig. 2b. Centrifugal Flow

The tip speed required to produce the required air particle velocity varies substantially with the type of blade used. Fig 1a, 1b and 1c show vector diagrams of forces in forward curve, backward curve, and radial blade impellers, respectively. Vector V_1represents the rotational or tangential velocity, and V_2 represents the radial velocity of the airflow between the blades with respect to the various blade shapes.

Vector R represents the resultant velocity for each of these blade shapes. Note that R for the forward curve impeller is the largest with the backward inclined impeller the smallest, while the radial blade fan lies somewhere in between. This relationship is best illustrated in Fig 3, which shows a typical tip speed/static pressure relationship for various types of centrifugal blower.

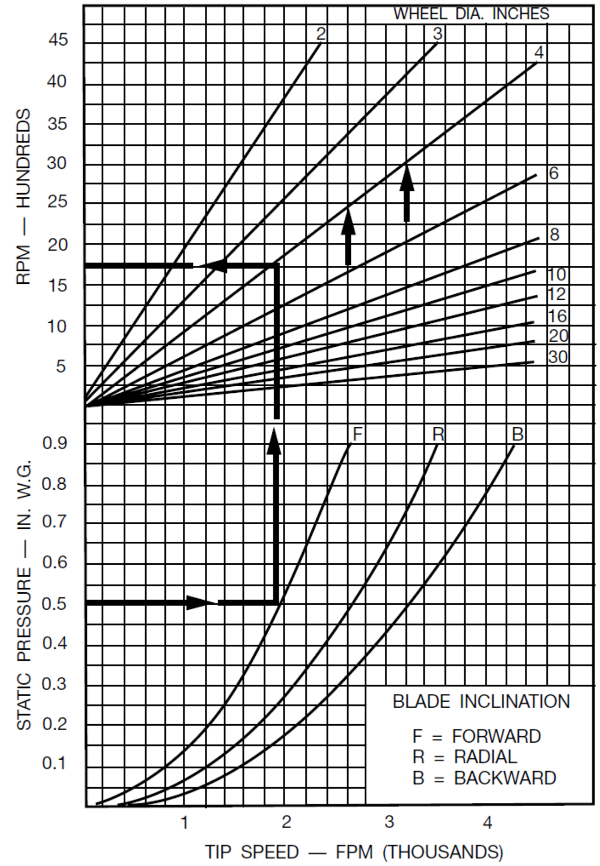


Figure 3. Tip Speed/Static Pressure Relationship

Forward Curve Fans

These fans are sometimes known as volume, squirrel cage, or sirocco blowers. The impeller blades are small and numerous with a pronounced curvature and short chord length. The concave blade curvature faces the direction of rotation. These fans operate at relatively low speeds and pressures which permits light construction of the impeller, shaft, bearings and housing.

Backward Inclined Fans

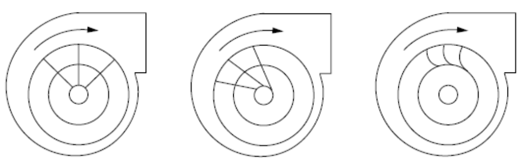
These are sometimes called load limiting or nonover loading fans. The impeller blades are larger andheavier than forward curve blades, usually number fromeight to twelve, and are inclined away from the directionof rotation. They are standardly offered in three bladeshapes:

1. Flat single thickness
2. Curved single thickness
3. Curved airfoil

Radial Blade Blower

Steel plate and paddle wheel are two of the commonnames for radial blade fans. The impeller blades aregenerally narrower, deeper and heavier than forwardcurve and backward inclined blades. A radial bladeimpeller usually comprises six to twelve equally spacedflat blades extending radially from the center of the hub.

These impellers are generally of simple design that lendsitself to rugged construction and offers a minimum ofledges, etc., for the accumulation of dust or stickymaterials.There are more variations of the radial blade fansthan the forward curve and backward inclined types.Three of the more common impellers are illustrated inFig. 4.



4a.	4b.	4c
OW/BW	AW	RT
Radial Blade	Air Wheel	Radial Tip

Fig.4. Common Radial Blade Impellers

III. DESIGN PROCEDURE OF RADIAL TYPE CENTRIFUGAL BLOWER (IMPELLER)

Design Specification of Radial Type Centrifugal Blower

This research work is based on an industrial requirement for Twin city Blower Company, Aerovent. Radial blades have lower unit blade stress for a given diameter and rotationalspeed hence lighter in weight. There is equal energy conversion in impeller anddiffuser so it gives high-pressure ratio with good efficiency [9, 20]. Looking to theserealization and facts radial blade fan is selected for this study.The input parameters for the design of radial tipped centrifugal blower are as following,

Flow Discharge Q = 2121 ft^3/min
Static Suction Pressure = 14.7 psi(absolute)
Static Delivery Pressure = 0.7396psi (gauge)
Static Pressure Gradient ΔPs = 981.2 Pa
Speed of impeller rotation N = 3450 rpm
Air Density = 1.165 kg/m^3
Outlet Blade Angle β_2 = 90°
Suction Temperature T_s= 60°F= (60 + 460)=520°R
Atmospheric Pressure P_{atm} = 1.01325 × 10^5 Pa
Atmospheric Temperature T_{atm} = 30°C = 303° K

These parameters are kept identical for each design methodology prescribedhere in.

Design of Radial Type Centrifugal blower as per Fundamental Concepts

This design procedure is based on the fundamental principles of fluid flowwith continuity and energy equations. The design follows the path from suction todischarge. To accelerate the flow at impeller inlet, converging section is designed after inlet duct. Energy balance is established at fan inlet, intermediate stage ofimpeller and outlet stage of volute/scroll casing.During this process stage velocity, pressure and discharge at different stagesare calculated. Flat front and back shrouds are selected for ease of impellerfabrication. Design procedure and calculations for above referred input parameters are presented below:

1) *Overall pressure ratio*

$$\varepsilon_p = \frac{14.7+0.7396}{14.7} = 1.0503$$

$$\varepsilon_p^{\,0.283} - 1 = 0.0139$$

2) *Total adiabatic head*

$$H_{ad} = \frac{RT_a}{0.283}\left[\varepsilon_p^{\,0.283} - 1\right]$$

$$H_{ad} = 1362 \text{ ft}$$

3) *Specific weight of air*

$$\gamma_a = \frac{P_a}{RT_a}$$

$$\gamma_a = 0.0763\ lb/ft^3$$

Weight flow; $w = \frac{Q\gamma_a}{60}$

$$w = 2.6972\ lb/sec$$

4) *Adiabatic air horsepower*

$$\text{a.hp} = \frac{wH_{ad}}{550}$$

$$\text{a.hp} = 7\text{hp}$$

IV. IMPELLER INLET DIMENSIONS AND VANE ANGLE

The velocity in the pipes for the design condition is usually between 60 and100 ft/sec; whereas the velocity at the flanges generally lies in the range from 100 to200 ft/sec.

Therefore, the velocity through the impeller eye is assumed as 100 ft/secvelocity head;

$$H = \frac{V_0^2}{2g}$$

$$H = 155.279\ ft$$

$$\varepsilon_p{}^{0.283} - 1 = \frac{0.283\ H}{RT_a}$$

$$\varepsilon_p{}^{0.283} - 1 = 0.00158$$

$$\varepsilon_p = 1.00559$$

$$P_o = \frac{P_a}{\varepsilon_p} = 14.7 \text{ psi}$$

$$T_o = \frac{T_a}{\varepsilon_p{}^{0.283}} = 215°R$$

5) The specific weight of the air in the impeller eye

$$\gamma_0 = \frac{P_0}{RT_0}$$

$$\gamma_0 = 0.0759\ lb/ft^3$$

6) Volume flow through impeller eye

$$Q_0 = \frac{w}{\gamma_0}$$

$$Q_0 = 35.5\ ft^3/sec$$

7) Air horsepower, $a.hp = 7hp$

$\eta_{overall} = 70\%$

8) Brake horse power, $b.hp = \frac{a.hp}{\eta_{overall}}$

$$b.hp = 10\ hp$$

9) Torque $T = \frac{6300 \times b.hp}{n}$

Torque T =182.608 lb-in

Allowed shear stress, $S_s = 300\ psi$

10) Shaft diameter,d

$$D_s = \sqrt[3]{\frac{16 \times T}{\pi S_s}} = 1.5 \text{ in}$$

The shaft diameter D_s, is based upon the critical speed and deflection. It will beamply strong in torsion and bending if it is made 1.5 in, in diameter. The hub diameterD_s ma then be made 2.5 in. The impeller eye diameter:

$$D_0 = \sqrt{\frac{4}{\pi} \times \frac{144 Q_0}{V_0} + D_H^2}$$

$D_0 =$ 8 in

So, $D_0 =$ 8 in may be used.

The vane inlet diameter D_1 may diameter.So, D1: 8.25 in may be selected.

Inlet tip speed: $U_1 = \frac{\pi D n}{720}$

$$U_1 = \frac{\pi \times 8.25 \times 3450}{720}$$

$U_1 = 124.191$ ft/sec

The inlet velocity is assumed to be radial, i.e $V_1 = V_{r1}$. And is made equally V_0 is 100 ft/sec. the inlet and outlet blade angle of impeller as shown in Fig 4.

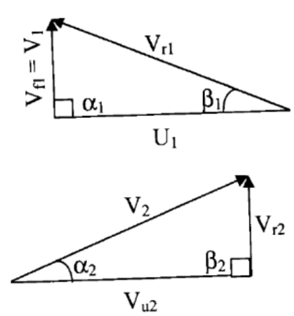


Fig.5. Inlet and Outlet Velocity Diagram of the Impeller

Impeller Blade Inlet angle, $\tan \beta_1 = \frac{V_1}{U_1}$

Impeller Blade Inlet angle, $\beta_1 = 38°$

The relative inlet velocity, $V_1 = \sqrt{U_1^2 + V_1^2}$

$$V_1 = 159.447\ ft/sec$$

In calculating the impeller areas, the flow must be increased because ofleakage past the impeller. This leakage may be assumed to be about 72 percent of theflow, subject to connection after the impeller dimensions have been established.

Impeller inlet area; $A_1 = \frac{1.072 \times Q_0 \times 144}{V_1}$

$A_1 =$ 54.801 in^2

Assuming the vane has constant thickness t: 0.25 in the inlet vane thicknessfactor ε_1: 0.85.

The impeller inlet width is

$$b_1 = \frac{A_1}{\pi D_1 \varepsilon_1}$$

$b_1 = 2.5$ in

V. SELECTION OF THE VANE DISCHARGE ANGLE

The vane outlet angle β_2 is usually made larger than the inlet β_1. The outlet vane angle, β_2 ma be selected within fairly wide limits. It is made slightly larger than the inlet angle to obtain a smooth, continuous passage. The outlet angle, β_2 cantheoretically be selected freely within a wide range. An angle β_2>90°leads tobackwards curved blades, β_2=90° means radially ending blades and β_2< 90° meansforward curved blades.

Effectsofimpellerdischargesontheoreticalheadandoutletvelocitydiagramsfor various vane angles, for radial blade design, the graph is straight line the flow rateand head constant and the power is also constant' In this design to get the powerconsumptionconstant,toreduceoreliminatethebendingstresstopreventthedamage of impeller and to maintain the blower easily, the vanes discharges angle β_2 ischosen 90°, radial blade design.

VI. IMPELLER OUTLET VANE ANGLE AND DIMENSIONS

The outside diameter of the impeller assuming the value of K′. The overallpressure coefficient K′ may be between 0.5 and 0.65.

Take K′ = 0.6.

$$D_2 = 1300 \times \frac{\sqrt{H}}{n\sqrt{K'}}$$

$D_2 =$ 18 in

The outlet vane angle β_2 is 90°.

The number of blade: Z

$$Z = \frac{D_2 + D_1}{D_2 - D_1} \sin\frac{\beta_1 + \beta_2}{2}$$

$Z = 15$ vanes

Therefore, the vane outlet angle β_2 is 90°, 15 vanes and a radial outlet velocity V_{r2} is 63ft/sec.

The impeller tip speed: $U_2 = \frac{\pi D_2 n}{720}$

$$U_1 = \frac{\pi \times 18 \times 3450}{720}$$

$U_1 = 270.962$ ft/sec

The circulatory flow effect reduces the tangential component, $W_z = U_2 \frac{\pi \sin \beta_2}{z}$

$W_z = 56.75\ ft/sec$

The tangential component of V_2 based upon a finite number of vanes, $V'_{u2} = V_{u2} + W_z$

$V'_{u2} = 270.962 - 56.75$

$V'_{u2} = 214.212\ ft/sec$

The absolute outlet velocity, $V_2 = \sqrt{V_{r2}^2 + V_{u2}^2}$

$V_2 = 278.189\ ft/sec$

Theabsolute outlet velocity with a pilot tube,

$$V'_2 = \sqrt{V_{r2}^2 + V'^2_{u2}}$$

$V'_2 = 223.284\ ft/sec$

The absolute outlet angle, $\tan \alpha'_2 = \frac{V_{r2}}{V'_{u2}}$

$\alpha'_2 =$ 16°

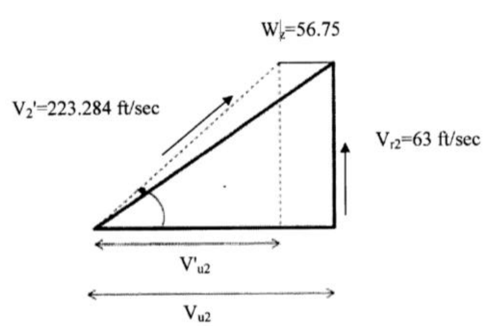


Fig.6.Virtual and Actual Outlet Velocity Diagram

The virtual and actual outlet velocity diagrams are as shown inFig.5. The virtual pressure head developed in the impeller

$$H_{vira} = \frac{1}{g} U_2^2$$

$H_{vira} =$ 2280 ft

It may be assumed that, owing to the circulation flow, friction, and turbulence in the impeller, l5 percent of this head is lost. Hence the effective head is

H = 0.85 × 2280 = 1938.116

$$\varepsilon_p{}^{0.283} - 1 = \frac{0.283\,H}{R T_a}$$

$$\varepsilon_p{}^{0.283} - 1 = 0.0198$$

ε_p=1.0717

The impeller outlet pressure,

P_2= 0.0717×14.6= 16 psi

The impeller outlet temperature,

$T_2 = T_0 \left(\varepsilon_p{}^{0.283}\right)$= 519 ×1.0198 = 529 °R

The outlet specific weight:

$$\gamma_2 = \frac{P_2}{R T_2}$$

$$\gamma_2 = 0.08165\ lb/ft^3$$

Table 1. Result Data of Impeller

Name	Symbol	Result
Shaft Diameter	D_s	1.5in
Hub Diameter	D_H	2.5in
Eye diameter	D_0	8 in
Eye velocity	V_0	100 ft/sec
Flow through eye	Q_0	35.5 ft^3/sec
Vane inlet diameter	D_1	8.25 in
Velocity at vane inlet	V_1	100 ft/sec
Impeller inlet width	b_1	2.5 in
Inlet vane angle	β_1	38°
Inlet vane thickness	ε_1	0.85 in
Number of vanes	z	15
Inlet tipspeed	U_1	124.191 ft/sec
Outside dia. of impeller	D_2	18 in
Impeller outlet tipspeed	U_2	270.962 ft/sec
Tangential component of outlet vel;	V_{u2}	214.212 ft/sec
Impeller outlet width	b_2	1.5 in
Outlet vane thickness	ε_2	0.93 in
Vane outlet angle	β_2	90°
Absolute vel;	V'_2	223.284 ft/sec
Absolute outlet angle:	α_2'	16°

Fig.7. Three-D Model of Radial type Centrifugal Compressor Impeller

VII. CONCLUSION

Blower can vary in size from a few feet to tensoffeet in diameter, depending on their application. Centrifugal blower are very useful in many industries, factories and many engineering fields. Although the centrifugal blowers are existing machines in many respective fields, we must be understood in its detail design, construction, function, application and basic theoretical knowledge. The design blower can be developed a pressure of 15.4396 psi (106.452 kPa) and deliver 2l2t ft^3/min (60 m^3/mim) of air at 3450rpm. The designed impeller has B25 in (209.55 mm) inlet diameter, 18 in (457.2 mm) outlet diameter, 380 inlet vane angle and 90°outlet vane angle. The number of vanes is 15blades. And then, the inlet width and outlet width are 2.5 in (63.5mm) and 1.5 in (38.1mm) respectively. The diameter of discharge flange is 6 in (152.4mm). So, this research would be very useful and applicable for the vane shape of designing blower that is widely used in industries.

REFERENCES

[1] Pump Handbook, Igor J.Karassik.Third Edition, 2001.

[2] Centrifugal Blower Ebara Corporation 1998'

[3] Fan Performance and Selection' ASHARE Handbook' 1992'

[4] Centrifugal Pumps and Blowers. AUSTIN H. CHURCH. 1972

[5] Energy Efficiency Guide for Industry in Asia' 2006'

[6] Jang, C. M., and Yang, S. H., 2009, "Performance Characteristics on Turbo Blowers Connected in Serial and Controlled by Inverter," The 10th Asian International Conference on Fluid Machnery, Malaysia, pp. 685–695.

[7] Lin, S. C., and Huang, C. L., 2002, "An Integrated Experimental and Numerical Study of Forward-Curved Centrifugal Fan," Experimental Thermal and Fluid Science, Vol. 26, No. 5, pp. 421–434.

[8] Yu, Z., Li, S., He, W., Wang, W., Huang, K., and Zhu, Z., 2005,"Numerical Simulation of Flow Field for a Whole Centrifugal Fan and Analysis of the Effects of Blade Inlet Angle and Impeller Gap," HVAC&R Research, Vol. 11, No. 2, pp. 263–283.

[9] Zhang, J.; Chu, W.; Zhang, H.; Wu, Y.; Dong, X. Numerical and experimental investigations of the unsteady aerodynamics and aero-acoustics characteristics of a backward curved blade centrifugal fan. *Appl. Acoust.***2016**, *110*, 256–267.

[10] Jiang, Y.Y.; Yoshimura, S.; Imai, R.; Katsura, H.; Yoshida, T.; Kato, C. Quantitative evaluation of flow induced structural vibration and noise in turbo machinery by full-scale weakly coupled simulation. *J. Fluids Struct.* **2007**, *23*, 531–544.

[11] Jie Jina Ying Fan, Wei Han, Jiaxin Hu, 2012. "Design and Analysis on Hydraulic Model of The Ultra - low Specific-speed Centrifugal Pump", International Conference on Advances in Computational Modeling and Simulation, 31: 110-114.

[12] Li Chunx, Wang Song Ling and JiaYakui, 2009. "The performance of a centrifugal fan with enlarged impeller", Journal of Sound and Vibration, 4(8): 87.

[13] Shojaeefard, M.H., M. Tahani, M.B. Ehghaghi, M.A. Fallahian, M. Beglari, 2012. "Numerical study of the effects of some geometric characteristics of a centrifugal pump impeller that pumps a viscous fluid", Computers & Fluids, 60: 61-70.

[14] Pham Ngoc Son, E. Jaewon Kim, Y. Ahn, 2011. "Effects of bell mouth geometries on the flow rate of centrifugal blowers", Journal of Mechanical Science and Technology, 25(9): 2267-2276.

[15] Singh, O.P., 2012. "Parametric study of centrifugal fan performance", International Journal of Advances in Engineering and Technology, 3(2): 33.

[16] Sun-Sheng Yang, Shahram Derakhshan, Fan-Yu Kong, 2012. "Theoretical, numerical and experimental prediction of pump as turbine performance", Renewable Energy, 48: 507-513.

[17] Taguchi, G., 1992. "Taguchi Methods - Research and Development", ASI Press, Dearborn, MI.