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Design Calculation of Single-Stage Radial Type Centrifugal Blower for Rice Mill

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Abstract: This journal is design calculation of the single-stage centrifugal blower. The two main components of a centrifugal blower are the impeller and the casing. The impeller is a rotating component and the casing is a stationary component. In centrifugal blower, air enters axially through the impeller eyes and air exists radially. The blower casing is to direct the gas to the impeller and leads it away at a higher pressure. Before the gas leaves the casing its velocity is reduced and partially converted into pressure by diffuser action. In this journal, air flow rate is 63m3/min and the motor speed is 3500rpm. The shaft diameter and hub diameter are 24mm and 58mm. The impeller inlet width and outlet width are 63.2mm and 29.8mm. Impeller inlet vane angle is 24.9˚. Impeller inlet diameter and outlet diameter are 270mm and 583mm.

Keywords: Radial Type Centrifugal Blower, Delivery Flow, Discharge Pressure, Speed, Impeller, Volute Casing, and Temperature.

I. INTRODUCTION

Machines that remove work from the flowing fluid are known as engines, turbines, water wheels, windmills. A machine that does work on a flowing fluid is called a pump, blower, or compressor. The majority of all pumps, blowers and compressors may be classified as positive-displacement type and dynamic type. Positive-displacement type consists of reciprocating and rotary pumps, blowers and compressors. This type does not permit free flow of fluid through the pump or blower except for leakage past close-fitting parts. Dynamic type consists of centrifugal (radial flow), mixed flow and axial flow pumps or blowers. In dynamic machines, there is a free passage of fluid between the inlet and outlet of the machine without any intermittent 'sealing' taking place. Centrifugal blower consists of an impeller with small blades on the circumference, a shroud to direct and control the airflow into the centre of the impeller and out at the periphery. The blades moves the air by centrifugal force and throwing it out thus creating suction inside the impeller and suction duct. The pressure rise and flow rate in centrifugal blowers depend on the peripheral speed of impeller and blade angles.

The blower can therefore be described as a device, which converts 'driver ' energy to kinetic energy in air by accelerating it to the outer rim of a revolving device known as an impeller. The impeller is always placed directly onto the shaft of the suction motor. Air enters the impeller axially through the inlet nozzle which provides slight acceleration to the air before its entry to the impeller. The action of the impeller swings the air from a smaller to a larger radius and delivers the air at a high pressure and velocity to the casing. The centrifugal energy also contributes to the stage pressure rise. The flow from the impeller blades is collected by a spirally-shaped casing known as scroll or volute. It delivers the air to the exit of the blower. The scroll casing can further increase the static pressure of air. The outlet passage after the scroll can also take the form of a conical diffuser. Centrifugal blowers are fundamentally high speed machines (compared with the reciprocating rotary or displacement type). The recent advances in stream turbine, electric motor, and high speed gearing design have greatly increased their use and application. Centrifugal blowers are used in many applications such as for high pressure air, chemical plants, blast furnace, sewage aeration blowers, biogas application, air plain supercharger, farm machinery and other many engineering fields.

II. DESIGN OF CENTRIFUGAL BLOWER

A. Velocity Diagrams

For a fluid flowing through a rotating impeller, U is the velocity of a point on the impeller relative to the ground, V is the absolute velocity of a fluid particle flowing through the impeller relative to the ground, and is the velocity of a fluid particle relative to the impeller. The angle between V and U is called α, the angle between V_r and U extended is β and it is angle made by tangent to the impeller vane and a line in the direction of the vane. These angles are shown in Fig. 1. as well as V_r is the radial component of the absolute velocity V.

A. Design of impeller

The blower design is analyzed single stage centrifugal blower. Impeller is designed on the basic of design flow rate and rotational speed.

air flow rate, $Q=1.05\ m^3/s$
rotational speed, N=3500 rpm
inlet air pressure, P_i=101353 Pa (absolute)
inlet air temperature, $T_i=30^\circ C$
discharge air pressure, Pd=9000 Pa (gauge)
gravitational acceleration, $g=9.81\ m/s^2$
air constant, R=287 J/kg K

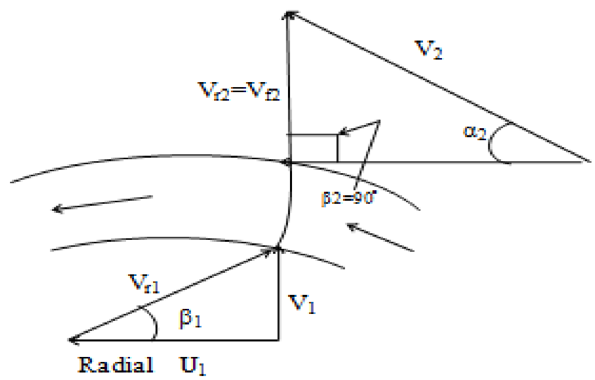

Figure1. Inlet and outlet velocity diagrams of the impeller.

Overall pressure ratio:

$$\varepsilon_p = \frac{P_d}{P_i} \tag{1}$$

Where P_d and P_i are discharge air pressure and inlet air pressure. Total adiabatic head:

$$H_{ad} = \frac{1}{g} \times \frac{RT_i}{0.287} \times \left(\varepsilon_p^{0.287} - 1\right) \tag{2}$$

and then, the weight flow of gas is;

$$w = \frac{Q\rho_i}{60} \tag{3}$$

Where, ρ_i is design of air and it is expressed by

$$\rho_i = \frac{P_i}{RT_i} \tag{4}$$

Thus, the adiabatic horsepower is determined by,

$$a.hp = \frac{wH_{ad}}{746} \tag{5}$$

The shaft diameter at the hub section:

$$D_S = 3\sqrt{\frac{16T}{\pi S_S}} \tag{6}$$

Where, T is the torsional moment and it can be estimated by,

$$T = \frac{60 \times b.hp}{2\pi n} \tag{7}$$

The value of Ss is chosen as.

The velocity at the impeller eye V_0 is slightly greater than the velocity at the suction flange and the suction flange velocity is depending on the standard pipe size. And then, the velocity head of impeller eye is;

$$H_0 = \frac{V_0^2}{2g} \tag{8}$$

For the pressure ratio between impeller inlet and impeller eye,

$$\varepsilon_p^{0.287} - 1 = \frac{0.287 \times H}{RT_i} \tag{9}$$

The pressure at impeller eye is:

$$P_0 = \frac{P_i}{\varepsilon_{p_{i-0}}} \tag{10}$$

The temperature at impeller eye is:

$$T_0 = \frac{T_i}{\varepsilon_{p_{i-0}}^{0.283}} \tag{11}$$

The density of the impeller eye is:

$$\rho_0 = \frac{P_0}{RT_0} \tag{12}$$

Volume flow through impeller eye is;

$$Q_0 = \frac{w}{\rho_0} \tag{13}$$

The hub diameter;

$$D_H = D_S + (19.05 \text{ to } 50.88) \tag{14}$$

The inlet velocity through the impeller eye V_0 is made slightly greater than the suction flange velocity.

The impeller eye diameter:

$$D_0 = \sqrt{\frac{4}{\pi} \times \frac{Q_0}{V_0} + D_H^2} \tag{15}$$

The vane inlet diameter D_1 can be made slightly greater than the eye diameter D_0. The impeller inlet speed;

$$U_1 = \frac{\pi D_1 n}{60} \tag{16}$$

The absolute velocity at the impeller inlet V_1 is assumed to be radial and V_1 is slightly greater than V_0.
The tangent of the inlet angle;

$$\tan\beta_1 = \frac{V_1}{U_1} \tag{17}$$

Which, may be increased somewhat to care for the contraction of the gas stream as it enters the vane passages. Relative inlet velocity:

$$V_{r1} = \sqrt{U_1^2 + V_1^2} \tag{18}$$

The inlet area of the impeller:

$$A_1 = \frac{Q_0}{V_1} \tag{19}$$

Impeller inlet width:

$$b_1 = \frac{A_1}{\pi D_1 \in_1} \quad (20)$$

Where, the inlet vane thickness factor $\in_1$ (assume 0.85 to 0.95).

$$D_2 = \frac{60 \times \sqrt{H_{ad} g}}{\pi n \sqrt{K'}} \quad (21)$$

Where, K' is the pressure coefficient which has a value between 0.5 and 0.65 depending on the type of impeller.
The outlet vane angle of impeller is 90°.
Blade number:

$$Z = 6.5 \times \frac{D_2 + D_1}{D_2 - D_1} \times \sin\frac{\beta_1 + \beta_2}{2} \quad (22)$$

But, the usual number of vanes varies between 15 and 30 in most blowers. A greater number will reduce the circulatory flow effect but will increase the friction.
The impeller tip speed at the outlet:

$$U_2 = \frac{\pi D_2 n}{60} \quad (23)$$

$$W_z = U_2 \frac{\pi \sin\beta_2}{z} \quad (24)$$

The radial component of the outlet gas velocity V_{r2} is made less than the inlet absolute velocity V_1.

$$U_2' = U_2 - W_z \quad (25)$$

$$V_2 = \sqrt{V_{r_2}^2 + U_2^2} \quad (26)$$

$$V_2' = \sqrt{V_{r_2}^2 + U_2'} \quad (27)$$

Absolute outlet angle at impeller:

$$\tan\alpha_2' = \frac{V_{r_2}}{U_2} \quad (28)$$

Virtual pressure head:

$$H_{vir.\infty.p} = \frac{1}{2g}\left(U_2^2 - U_1^2 + V_{r1}^2 - V_{r2}^2\right) \quad (29)$$

The effective head is;

$$H_{eff} = \eta_{overall} \times H_{vir.\infty.p} \quad (30)$$

For the pressure ratio between impeller eye and outlet, the relative equation is;

$$\varepsilon_p^{0.287} - 1 = \frac{0.287 \times H_{eff}}{RT_0} \quad (32)$$

Thus, impeller outlet pressure is;

$$P_2 = \varepsilon_p \times P_0 \quad (33)$$

The friction and turbulence losses with be transformed into heat which raises the temperature of the gas. The outlet temperature can be based upon the adiabatic head in the impeller neglecting losses.

$$\varepsilon_p^{0.287} - 1 = \frac{0.287 \times H_{vir.\infty.p}}{RT_0} \quad (34)$$

Then, the impeller outlet temperature is;

$$T_2 = T_0 \times \varepsilon_p^{0.287} \quad (35)$$

The outlet density is;

$$\rho_2 = \frac{P_2}{RT_2} \quad (36)$$

Thus, the flow leaving the impeller is;

$$Q_2 = \frac{w}{\rho_2} \quad (37)$$

Where, Q_2 is the flow leaving the impeller. Assuming the vane thickness is constant. The outlet vane thickness factor is;

$$\in_2 = \frac{\pi D_2 - \frac{Zt}{\sin\beta_2}}{\pi D_2} \quad (38)$$

Where, t is the blade thickness which the vane is chosen as 3.175mm. Thus, the outlet area of the impeller:

$$A_2 = \frac{Q_2}{V_{r2}} \quad (39)$$

The impeller outlet width:

$$b_2 = \frac{A_2}{\pi \times D_2 \in_2} \quad (40)$$

B. Design of volute casing

The purpose of the volute as outline as outlined previously is to convert the velocity head of the gas leaving the impeller as efficiently as possible. The gas in the volute has very nearly the spiral flow in RV_R=C = a constant, where C is determined from the relationship $R_2 V_{u_2}' = C$ for a given stage. It may be assumed that the flow from the impeller is uniform about its periphery, so the flow past any section of the volute is ϕ/360 of the total, where ϕ is the angle in degrees measured from the theoretical tongue of the volute as shown in Fig. 2 and 3.

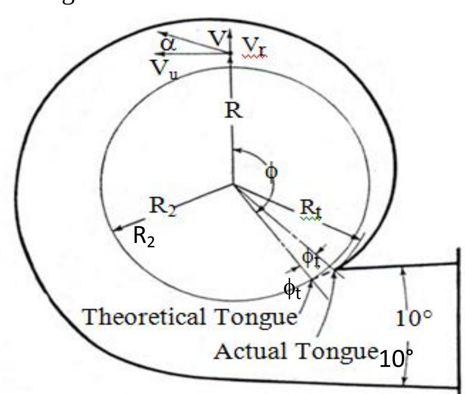

Figure 2. Elevation of Volute.

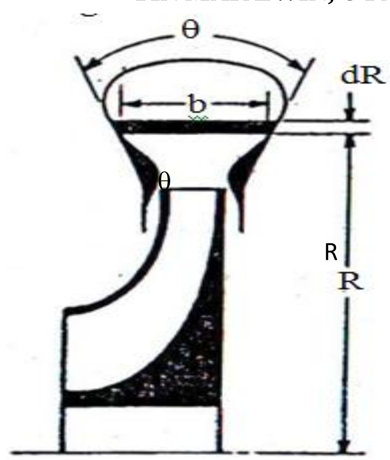

Figure 3. Section Through Volute [4].

In determining the cross sectional area of the volute at any point (see fig 4), the problem consists in finding the area of the section that will pass the volume $Q_\phi/360$ with a velocity $V_u = C/R$. It should be noted that the volute of Q used is that of delivered flow. It does not include the leakage flow which has now split off from the total impeller flow and returned to the suction through the wearing rings.

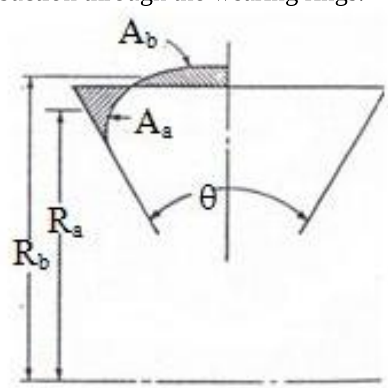

Figure 4. Volute passage cross section[4].

If friction is neglected, the flow through the differential section

$$dQ_\phi = dAV_u = b\, d\, RV_u \tag{41}$$

But $V_u = C/R$, hence $dQ_\phi = b\, dR\, C/R$, and the total flow past the section becomes

$$Q_\phi = \int_{R_2}^{R\phi} dQ = C\int_{R_2}^{R\phi} b\frac{dR}{R} \tag{42}$$

Where, R_ϕ is the outer radius of a section at ϕ from the theoretical tongue. Substituting for Q_ϕ the term $\phi Q/360$ there results

$$\phi^\circ = \frac{360C}{Q}\int_{R_2}^{R\phi} b\frac{dR}{R} = \frac{360R_2V'_{u_2}}{Q}\int_{R_2}^{R\phi} b\frac{dR}{R} \tag{43}$$

To avoid shock losses, the tongue angle should be made the same as the absolute outlet angle α'_2 of the water leaving the impeller. The radius R_t at which the tongue starts should be 5 to 10 percent greater than the outside radius of the impeller to avoid turbulence and noisiness and to give the velocities of the water leaving the impeller a chance to equalize before coming into contact with the tongue.

$$\phi_{t=} \frac{132\mathrm{Log}_{10}\frac{R}{R_2}}{\tan\alpha'_2} \tag{44}$$

For the tongue radius R= R_t tongue angle Tongue angle:

$$\phi_t = \frac{132\mathrm{Log}_{10}\frac{R}{R_2}}{\tan\alpha'_2} \tag{45}$$

IV. CALCULATED RESULT OF RADIAL TYPE CENTRIFUGAL BLOWER DESIGN

The calculated results for both impeller and casing design of radial type centrifugal blower are clearly expressed in Table I. Moreover, detail drawing of impeller and volute casing designs are also shown in following figures 5 and 6.

TABLE I: Result Data of Radial Type Impeller

No	Descriptions	Value	Units
1	Shaft diameter, Ds	24	mm
2	Hub diameter, D_H	58	mm
3	Eye diameter, D_0	243	mm
4	Eye velocity, V_0	21	m/sec
5	Flow through eye, Q_0	1.05	m^3/sec
6	Vane inlet diameter, D_1	270	mm
7	Velocity at vane inlet, V_1	23	m/sec
8	Impeller inlet width, b_1	63.2	mm
9	Inlet vane thickness factor, $\in_1$	.85	-
10	Inlet vane angle, β_1	24.9	degree
11	Number of vanes, Z	15	
12	Impeller inlet tip speed, U_1	49.48	m/sec
13	Outside diameter of impeller, D_2	583	mm
14	Radial outlet velocity, V_{r2}	20	m/sec
15	Impeller width at outlet, b_2	29.8	mm
16	Outlet vane thickness correction factor, $\in_2$	.95	-
17	Impeller outlet tip speed, U_2	106.84	m/sec
18	Vane outlet angle, β_2	degree	90

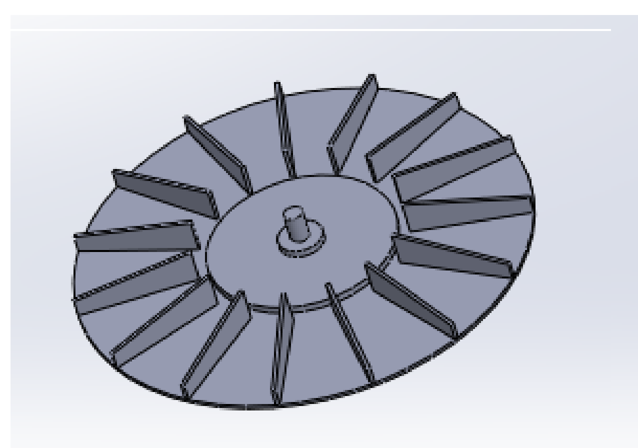

Figure 5. 3D Drawing of Impeller Blade.

TABLE II: Reselt Data Of Volute Casing

R (mm)	ΔR (mm)	Ravg (mm)	b (mm)	$\frac{b\Delta R}{R_{avg}}$	Δϕ	ϕ
291.5			52.15	0	0	0
311.5	20	301.5	63.69	4.22	35.95	35.95
331.5	20	321.5	86.79	5.39	45.92	81.87
351.5	20	341.5	109.88	6.43	54.78	136.65
371.5	20	361.5	132.98	7.36	62.71	199.36
391.5	20	381.5	156.07	8.18	69.69	269.05
411.5	20	401.5	179.17	8.92	75.99	344.03
431.5	20	421.5	202.26	9.59	81.71	426.74

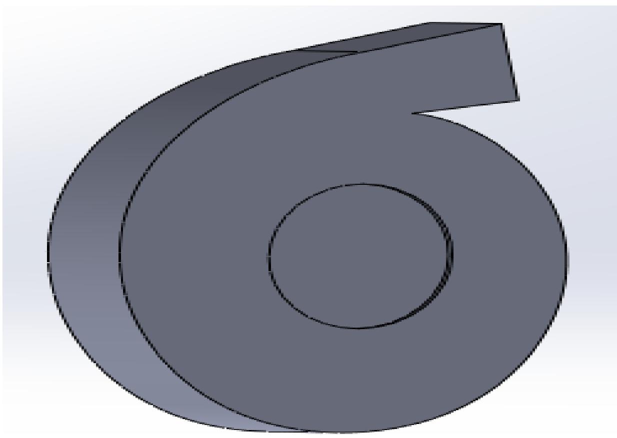

Figure 6. 3D Drawing of Volute Casing.

V. DISCUSSIONS AND CONCLUSION

The study is attempted to design single-state radial type centrifugal blower for rice mill. Modern blowers have a wide variety of application, e.g. refrigeration and air conditioning systems, pipeline transport of natural gas, petroleum refineries, gas turbine systems, farm machinery, and in many various industrial, manufacturing and building processes. Blower can vary in size from a few feet to tens of feet in diameter, depending on their application. In this journal, the radial type centrifugal blower is designed for 101.353kPa (absolute) pressure and 63m^3/min of flow rate. The rotational speed is 3500 rpm. The calculated impeller design has inlet diameter of 270mm and impeller outside diameter of 583mm. And then, the inlet width and the outlet width are 63.2mm and 29.8mm. The design volute is calculated by depending on the impeller outlet diameter and impeller outlet width. According to the result of design, impeller blade shape and volute casing are drawn by using SolidWorks software.

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