Impeller Design of Centrifugal Blower for 40 kW Wood Chips Gasifier

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Abstract- Impeller design plays an important role in manufacturing centrifugal blowers because, without proper design, the blowers cannot function effectively. This paper presents design of the 12-blade backward-curved impeller to be used in the centrifugal blower for 40 kW wood chips gasifier. Design is mainly focused for the single-stage impeller. Required design parameters (such as impeller dimensions, blade shape, vane angle, number of blades for centrifugal blower) of the proposed impeller are calculated and the results are shown in this paper.

Index Terms- Blade, Blower, Gasifier, Impeller

I. Introduction

The application and use of centrifugal pumps and blowers today are universal. Nowadays, local made centrifugal pumps and blowers are widely used in factories, farm machinery, gasifiers and other buildings in Myanmar. Gas-pumping turbomachinery is composed at a wide variety of machines such as fans, blowers, and compressors. All of them are gas compressed and moved by dynamic action of rotating vanes of impellers which impact velocity and pressure to flow gas.

The majority of all pumps, blowers and compressors may be classified as reciprocating rotary or centrifugal. Reciprocating and rotary blowers or pumps do not permit free flow of fluid through the blower except for leakage past close-fitting parts, and are called “positive-displacement” blowers [1].

A centrifugal pump or blower consists essential of one or more impeller equipped with vanes, mounted on a rotating shaft and enclosed by a casing. Fluid enters the impeller axially near the shaft and has energy, both 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 diffusing passages which transforms the kinetic energy into pressure. This is, of course, accompanied decrease in the velocity. After the conversion is accomplished the fluid is discharged from the machine.

Centrifugal pumps and blowers are fundamentally high speed machines (compared with the reciprocating, rotary, or displacement type) [2]. The recent advances in steam turbine, electric motor, and high speed gearing design have greatly increased their use and application. As the centrifugal machines have been developed, they have had to complete with the already established reciprocating units.

II . TYPES OF BLOWER

The term blower generally used for high pressure fans or low pressure compressor. The great variety of blowers built for various applications may be reduced to a few basic hydraulic types. Every blower consists of two principal parts. The first principal is an impeller or lobed rotors, which force the gas into rotary motion by impelling action of the vanes. The second principal is casing, which directs 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. These are several methods of converting the velocity of gas issuing from the impeller.

In general, blowers can be separated into two types:

1) Turbo type machines
2) Positive-displacement type machines [3].

Turbo type machines that transfer energy from rotor to fluid via dynamic action, acceleration, deceleration, and motion in radial force field includes radial flow blowers sometimes called centrifugal blowers and axial flow blowers**.**

Positive-displacement machines type that fluid is drawn or forced into a finite space bounded by mechanical parts and is then sealed in it by some mechanical means consist of only roots blowers (lobed impeller).

III. BASIC THEORY OF BLOWER

The path and velocity of a fluid particle flowing through the impeller would appear to be quite different to an observer standing on the ground, than it would to one station inside the rotating impeller, if that were possible. The velocity of the particle relative to the ground is called absolute; the velocity relative to the impeller is called relative [4]. It is important to understand thoroughly these two types of velocities and the relationships between them.

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 v is the velocity of a fluid particle relative to the impeller.

Assuming that the flow takes place in a plane, i.e., that it is two-dimensional, and that the fluid follows the impeller vanes exactly, the inlet and outlet triangles of velocities for an impeller having backward curved vanes are shown in Figure 1.

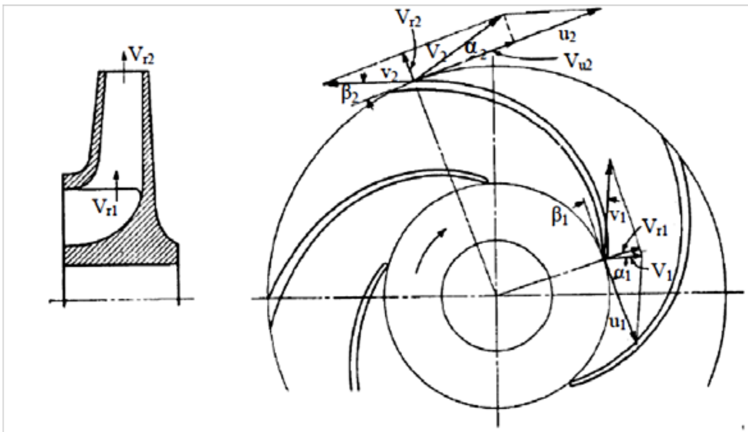

Figure 1: Inlet and Outlet Velocity Diagrams of an Impeller Having Backward-Curved Vanes [1]

The angle between V and u is called α; the angle between v and u extended (negative u) is β and it is the angle made by tangent to the impeller vane and a line in the direction of motion of the vane. These angles are shown on the Figure 2, as well as V_r which is the radial component of the absolute velocity V. The simplified inlet and outlet diagrams of these velocities for the impeller are shown in Figure 3.2. Note that V_u is the tangential component of V and equals V cosα.

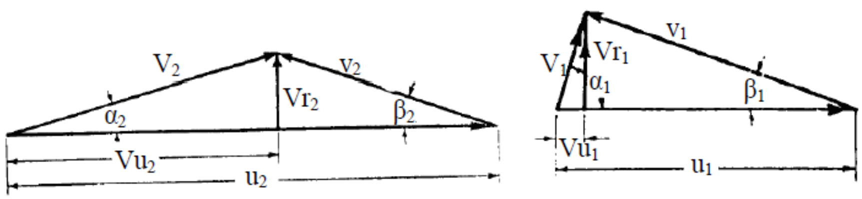

Figure 2. Virtual Inlet and Outlet Velocity Diagrams of the Impeller [1]

IV. DESIGN CALCULATION OF THE PROPOSED IMPELLER

The centrifugal blower for gasifier is to run at 2500 r.p.m. and handle 882.75 c.f.m. (25 m^3/min) of gas i.e. 14.7125 c.f .s. at 26° C (78.8° F) i.e. 538.8° R and 14.7 psi absolute with a discharge pressure of 0.3317 psi (250 mmH_2O).

Overall pressure ratio: $\epsilon_p = \dfrac{14.7 + 0.3317}{14.7} = 1.0226$

$$\epsilon_p^{0.283} - 1 = 6.3348 \times 10^{-3}$$

Total adiabatic head: $H_{ad} = \dfrac{RT_a}{0.283} \times \left(\epsilon_p^{0.283} - 1\right) = \dfrac{55.23 \times 538.8}{0.283} \times 6.3348 \times 10^{-3}$ = 666.11 ft (203.03 m)

where carbon monoxide gas constant: R = 55.23 ft lb/ lb_m°R

Specific weight of gas: $\gamma = \dfrac{P_a}{RT_a} = \dfrac{144 \times 14.7}{55.23 \times 538.8}$ = 0.0711 lb/ft^3

Weight flow: $w = \dfrac{Q\gamma}{60} = \dfrac{882.75 \times 0.0711}{60}$ = 1.0466 lb/sec

Adiabatic horsepower: $hp = \dfrac{wH_{ad}}{550} = \dfrac{1.0466 \times 666.11}{550}$ =1.27 hp

A. Impeller Inlet Dimensions and Vane Angle

Assume a velocity through the impeller eye V_0 of 65 ft/sec.

Velocity head: $\frac{V_0^2}{2g} = \frac{65^2}{2 \times 32.2} = 65.61$ ft

$$\in_p^{0.283} - 1 = \frac{0.283 \times H}{RT_a} = \frac{0.283 \times 65.61}{55.23 \times 538.8} = 6.2396 \times 10^{-4}$$

$$\in_p^{0.283} = 1.0006\ ; \qquad \in_p = 1.0022$$

$$P_0 = \frac{14.7}{\in_p} = \frac{14.7}{1.0022} = 14.67 \text{ lb/in}^2$$

$$T_0 = \frac{T_a}{\in_p^{0.283}} = \frac{538.8}{1.0006} = 538.5° \text{ R}$$

The specific weight of the gas in the impeller eye: $\gamma_0 = \frac{P_0}{RT_0} = \frac{144 \times 14.67}{55.23 \times 538.5} = 0.0710 \text{ lb/ft}^3$

Volume flow through impeller eye: $Q_0 = \frac{w}{\gamma_0} = \frac{1.0466}{0.0710} = 14.74 \text{ ft}^3\text{/sec}$

The shaft diameter Ds is based upon the critical speed and deflection. The shaft diameter Ds is made 1.5 inches. The hub diameter D_H may then be taken as 3 inches.

The impeller eye diameter: $D_0 = \sqrt{\frac{4}{\pi} \times \frac{144 \times Q_0}{V_0} + D_H^2} = \sqrt{\frac{4}{\pi} \times \frac{144 \times 14.74}{65} + 3^2} = 7.11$ in.

So D_0 = 7 in. may be used.

The vane inlet diameter D_1 may be made slightly greater than the eye diameter, so D_1 is taken as 7.5 inches.

Inlet tip speed: $u_1 = \frac{\pi D_1 N}{720} = \frac{\pi \times 7.5 \times 2500}{720} = 81.81$ ft/sec

The inlet velocity is assumed to be radial; i.e.,$V_1 = V_{r1}$, and is made slightly greater than V_0, so V_1 is 70 ft/sec.

The tangent of the inlet angle: $\tan \beta_1 = \frac{V_1}{u_1} = \frac{70}{81.81} = 0.8556$

This should be increased by about 3 percent to care for the contraction of the steam at the inlet: 1.03×0.8556 = 0.8813. Then

$$\beta_1 = \tan^{-1}(0.8813) = 41.39° \quad (41° 23')$$

Relative inlet velocity: $v_1 = \sqrt{u_1^2 + V_1^2} = \sqrt{81.81^2 + 70^2} = 107.67$ ft/sec

In calculating the impeller areas, the flow must be increased about 10 percent because of leakage past the impeller.

Impeller inlet area: $A_1 = \frac{1.1 \times Q_0 \times 144}{V_1} = \frac{1.1 \times 14.74 \times 144}{70} = 33.35 \text{ in}^2$

Assuming the vane thickness factor $\in_1$ of 0.9 and the impeller inlet width is

$$b_1 = \frac{A_1}{\pi D_1 \in_1} = \frac{33.35}{\pi \times 7.5 \times 0.9} = 1.6 \text{ in}$$

B. Impeller Outlet Dimensions and Vane Angle

The outside diameter of the impeller is found from Equation after assuming the value of K′. The overall pressure coefficient K′ may be between 0.5 and 0.65. Take the value of K′ = 0.575.

Outside diameter of impeller: $D_2 = \dfrac{1300 \times \sqrt{H}}{N\sqrt{K'}} = \dfrac{1300 \times \sqrt{666.11}}{2500 \times \sqrt{0.575}} = 17.7$ in

Therefore, D_2 is taken as 18 in.
The outlet vane angle, β_2 of 55° is assumed.

The number of blade: $Z = 6.5 \times \dfrac{D_2 + D_1}{D_2 - D_1} \times \sin\dfrac{\beta_1 + \beta_2}{2} = 6.5 \times \dfrac{18 + 7.5}{18 - 7.5} \times \sin\dfrac{41.39 + 55}{2} = 11.77$

So the number of blade may be used 12 vanes.
The radial outlet velocity V_{r2} is made less than the inlet velocity V_1 and may be taken as 45 ft/sec.

The impeller tip speed: $u_2 = \dfrac{\pi D_2 N}{720} = \dfrac{\pi \times 18 \times 2500}{720} = 196.35$ ft/sec

$$V_{u2} = u_2 - \frac{V_{r2}}{\tan\beta_2} = 196.35 - \frac{45}{\tan 55} = 164.84 \text{ ft/sec}$$

$$W_z = u_2 \times \frac{\pi \times \sin\beta_2}{12} = \frac{196.35 \times \pi \times \sin 55}{12} = 42.11 \text{ ft/sec}$$

$$V'_{u2} = V_{u2} - W_z = 164.84 - 42.11 = 122.73 \text{ ft/sec}$$

$$V_2 = \sqrt{V_{r2}^2 - V_{u2}^2} = \sqrt{45^2 + 164.84^2} = 170.87 \text{ ft/sec}$$

$$V'_2 = \sqrt{V_{r2}^2 - V'^2_{u2}} = \sqrt{45^2 + 122.73^2} = 130.72 \text{ ft/sec}$$

$$v_2 = \sqrt{V_{r2}^2 + (u_2 - V_{u2})^2} = \sqrt{45^2 + (196.35 - 164.84)^2} = 54.94 \text{ ft/sec}$$

$$\tan\alpha'_2 = \frac{V_{r2}}{V'_{u2}} = \frac{45}{122.73} = 0.3607 \quad ; \quad \alpha'_2 = 20.14° \ (20° \ 8')$$

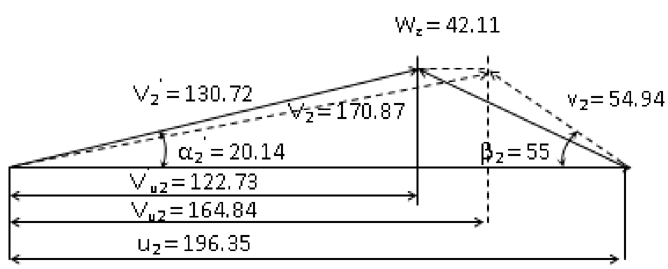

Figure 3: Outlet Velocity Diagram

The outlet velocity diagram for impeller is shown in Figure 3.
The virtual pressure head developed in the impeller is;

$$H_{vir.\infty.p} = \frac{1}{2g}\left(u_2^2 - u_1^2 + v_1^2 - v_2^2\right) = \frac{1}{2 \times 32.2} \times \left(196.35^2 - 81.81^2 + 107.67^2 - 54.94^2\right) = 627.87 \text{ ft}$$

It may be assumed that, owning to the circulatory flow, friction, and turbulence in the impeller, 10 percent of this head is lost. Hence the effective head is

$$H_{eff} = 0.9 \times 627.87 = 565.08 \text{ ft}$$

$$\epsilon_p^{0.283} - 1 = \frac{0.283 \times H_{eff}}{RT_0} = \frac{0.283 \times 565.08}{55.23 \times 538.5} = 5.3770 \times 10^{-3} \ ; \qquad \epsilon_p = 1.0191$$

Impeller outlet pressure: $P_2 = \epsilon_p \times P_0 = 1.0191 \times 14.67 = 14.95$ lb/in^2

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

$$\epsilon_p^{0.283} - 1 = \frac{0.283 \times H_{vir.\infty.p}}{RT_0} = \frac{0.283 \times 627.87}{55.23 \times 538.5} = 5.9744 \times 10^{-3}$$

$$\epsilon_p^{0.283} = 1.0060$$

The outlet temperature: $T_2 = T_0 \times \epsilon_p^{0.283} = 538.5 \times 1.0060 = 541.72° \text{ R}$

The outlet specific weight: $\gamma_2 = \frac{P_2}{RT_2} = \frac{144 \times 14.95}{55.23 \times 541.72} = 0.0720 \text{ lb/ft}^3$

The flow leaving the impeller: $Q_2 = \frac{1.1 \times w}{\gamma_2} = \frac{1.1 \times 1.0466}{0.0720} = 15.99 \text{ ft}^3\text{/sec}$

The net impeller outlet area: $A_2 = \frac{144 \times Q_2}{V_{r2}} = \frac{144 \times 15.99}{45} = 51.17 \text{ in}^2$

Assuming the vane thickness be 1/8 inches, the outlet vane thickness factor:

$$\epsilon_2 = \frac{\pi D_2 - \frac{Zt}{\sin\beta_2}}{\pi D_2} = \frac{\pi \times 18 - \frac{12 \times 0.125}{\sin 55}}{\pi \times 18} = 0.968$$

The impeller outlet width: $b_2 = \frac{A_2}{\pi \times D_2\ \epsilon_2} = \frac{51.17}{\pi \times 18 \times 0.968} = 0.935 \text{ in}$

The vane thickness factor at the inlet: $\epsilon_1 = \frac{7.5 \times \pi - \frac{12 \times 0.125}{\sin 41.39}}{7.5 \times \pi} = 0.9$

The vane thickness factor at the inlet ϵ_1 is nearly equal to the previous assumed value.

So, the impeller inlet width: $b_1 = \frac{A_1}{\pi D_1\ \epsilon_1} = \frac{33.35}{\pi \times 7.5 \times 0.9} = 1.6 \text{ in}$

Summary of calculated data for the designed impeller is stated in Table I and calculated data for blade shape is described in Table II. Then Figure 4 illustrates the complete vane design in polar coordinates. And Figure 5 depicts detail drawing of 12 blades backward-curve impeller.

Table I. Summary of Impeller Design

Data	Units	Values
Shaft diameter (D_s)	in.	1.5
Hub diameter (D_H)	in.	3
Eye diameter (D_0)	in.	7
Eye velocity (V_0)	ft/sec	65
Flow through eye (Q_0)	ft^3/sec	14.74
Vane inlet diameter (D_1)	in.	7.5
Velocity at vane inlet (V_1)	ft/sec	70
Impeller inlet width (b_1)	in.	1.6
Inlet vane angle (β_1)	degree	41.39
Inlet vane thickness factor (ϵ_1)	-	0.9
Number of vanes	-	12
Inlet tip speed (u_1)	ft/sec	81.81
Outlet diameter of impeller (D_2)	in.	18
Radial outlet velocity (V_{r2})	ft/sec	45
Impeller outlet width (b_2)	in.	0.94

Outlet vane thickness factor (ϵ_1)	-	0.97
Impeller outlet tip speed (u_2)	ft/sec	196.35
Vane outlet angle (β_2)	degree	55
Absolute outlet velocity (V_2')	ft/sec	130.72
Flow from impeller outlet (Q_2)	ft^3/sec	15.99

Table II. Data for Blade Shape

Ring	R	β	tan β	$\frac{1}{R \tan\beta}$	$\frac{1}{R \tan\beta}$	ΔR	$\frac{\Delta R}{R \tan\beta}$	Δθ°	θ°
1	3.75	41.00°	0.8693	0.307					0
					0.255	1.5	0.3825	21.92	
b	5.25	43.25°	0.9407	0.202					21.92
					0.169	1.75	0.2958	16.95	
c	7.00	46.50°	1.0538	0.136					38.87
					0.119	1.25	0.1488	8.53	
d	8.25	50.25°	1.2024	0.101					47.4
					0.090	0.75	0.0675	3.87	
2	9.00	55.00°	1.4281	0.078					51.27

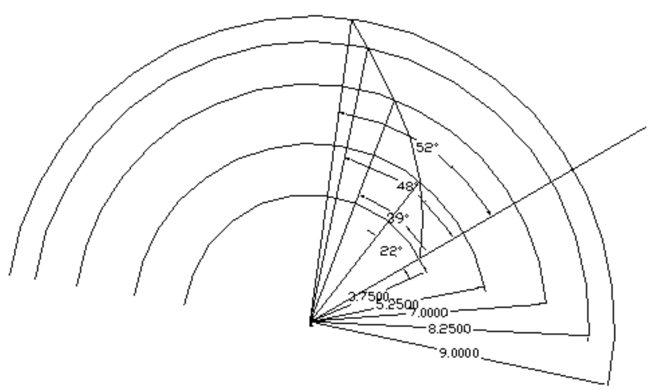

Figure 4: Vane Design Using Polar Coordinate

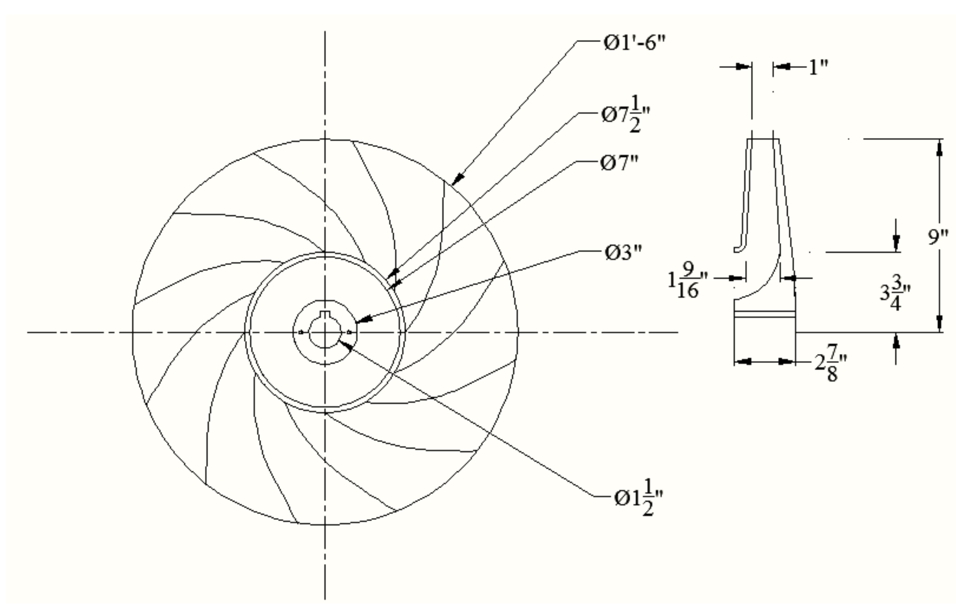

Figure 5: 12 Blades Backward-curved Impeller

V. CONCLUSION

In this paper, the single-stage impeller was designed for the centrifugal blower used in 40 kW wood-chips gasifier. The calculation of impeller consisted of the determinations of overall dimensions, such as inlet and outlet diameter, vane angle, and number of blades and blade shape. Finally, the designed impeller was a 12- blade backward-curved one. This impeller design was considered for centrifugal blowers which run at 2500 rpm. In conclusion, this impeller can be applied in the centrifugal blower for 40 kW wood-chips gasifier.

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