American Journal of Engineering Research (AJER)
e-ISSN: 2320-0847 p-ISSN : 2320-0936
Volume-5, Issue-10, pp-181-186
www.ajer.org
Research Paper Open Access

Design of a Centrifugal Blower for a 400kg Rotary Furnace.

Sani Malami Suleiman, Kantiok Obadiah, and Lawal Ibrahim Atiku
Department of Mechanical Engineering, College of Engineering, Kaduna polytechnic, Kaduna.

ABSTRACT: *Poor performance of a rotary furnace cannot be unconnected to failure in the design of the blower among others, This paper discuss the design of a centrifugal blower for a rotary furnace which will give the required manometric efficiency that will aid adequate combustion as required. The blower was designed to convert 'driver' energy to kinetic energy in the fluid by accelerating it to the outer rim of the revolving device known as the impeller. The impeller, driven by the blower shaft adds the velocity component to the fluid by centrifugally casting the fluid away from the impeller vane tips. The amount of energy given to the fluid corresponds to the velocity at the edge or vane tip of the impeller.*

Significance*: Centrifugal blowers are applicable in furnaces such as Rotary and cupola furnace, the efficiency of these furnaces depend on the blast rate and air delivery from a well design blower. This paper will guide to achieve this aims.*

Keywords: *Impeller, Blower, Manometric efficiency, Rotary furnace*

I. INTRODUCTION

Blowers is a contrivance which provides energy to the air, system, to assists to increase the pressure flow takes place from the low pressure side to high pressure side.

The blowers can either be classified as a rotor dynamics machines or under displacement machines [Alan, M.(1979)].. In this design a rotor dynamic (BLOWER) is taken because of its higher efficiency and less cost.

On the basis of transfer of mechanical energy, the blowers can be broadly classified as follows:

1.1 Rotor Dynamic Blowers

In rotor dynamic blowers, increase in energy level is due to a combination of centrifugal energy pressure energy and kinetic energy.

1. The energy transfer, in radial flow blowers, occurs mainly when the flow is in its radial path.
2. In axial flow blowers, the energy transfer occurs when the flow is in its axial direction.
3. The every transfer in a mixed flow blowers takes place when the flow comprises of radial as well as a axial components.

1.2 Positive Displacement Blowers

This type of blowers uses the principle of piston displacement in a cylinder, which controls the volume and pressure rise and drop of the incoming air. It works efficiently with higher pressure drops and less flow rate.

1.3 Classification Of Centrifugal Blowers

On the basis of characteristic features, the centrifugal blowers are classified as:

1. Type of casing
2. working heads
3. liquid handled
4. Number of impellers per shaft
5. Number of entrance to the impeller
6. Relative direction of flow through impeller

1.4 Components Of The Centrifugal Blower

1. **Impeller:** an impeller is a wheel (or rotor) with a series of backward curved varies or blades. It is mounted on a shaft which is usually coupled to an electric motor. And there are three types of impellers.

a. **Closed Impellers:** In this type of impeller, vanes are provided with metal cover plates en shrouds on both the sides. It provides better guidance for the fluid. This type of impellers is chosen in this design.
b. **Semi-open impellers:** in this type, the vanes have only the base plate and no crown plate. It can be used with fluids that contain debris.
c. **Open-impeller:** this type of impellers has neither the crown plate nor the bottom, plate. They are employed to blow the air that contains suspended solids.

2. Casing:

The casing is an air tight-chamber surrounding the blower impeller. It contains suction and discharge arrangements, supporting bearings and facilitates to house the rotor assembly also protects leakages. There are also three types of casing.

a. **Volute Casing:** In this casing type, the area of flow gradually increases from impeller outlet to the delivery pipe so as to reduce the velocity of flow. Thus the increase in pressure occurs in volute casing.
b. **Vortex Casing:** If a circular chamber is provided between the impeller and the volute chamber, the casing is known as vortex casing. The circular chamber is known as vortex or whirl pool chamber. The vortex chamber converts some of the kinetic energy to pressure energy. So it has more efficiency than simple involutes casing.

3. Suction Pipe:

This is the pipe or opening that connects the centre of the impeller to sump from which fluid is to be lifted.

4. Delivery Pipe (Nozzle)

This is the outlet of the blower; its main function is to help in giving higher velocity to the air at the expense of pressure drop in it.

II. METHODODLOGY

Design Analysis

This design is made based on the following assumptions for maximum efficiency

i. Air enters the impeller eye in radial direction which makes the whirls component at inlet zero.
ii. No loss due to shock at entry base on the above mentioned assumptions the design parameters are fairly estimated, these are:

- Blade inlet and outlet diameter
- Blade width
- Number of blades
- Number of blade revolutions per minute
- Manometric head
- Analysis of the various velocities during he blades operation
- The Motor horse power required
- The pressure rise across the impeller
- The overall efficiency and power developed by the blower.

2.1 Design Specification

This blower is expected to have the following ratings:

$Q = 6.77m^3/min$

$P = 2 – 10Kpa$

Where Q= volumetric flow rate

P= pressure developed

a) Blades Diameters

From the optimum performance specification it is stated that the ratio of the internal diameter to the external diameter is to fall between 0.4 to 0.7 as stated in the ASME code.

That is $0.4 < D_1/D_2 < 0.7$, for this design

The ratio D_1/D_2 = 0.65 is taken;

If D_1 = 182mm

$$D_2 = \frac{D_1}{0.65} = \frac{182}{0.65} = 280 \ mm$$

Therefore, D_1 = 182mm, D_2 = 280mm

b) Number of Blades

From ASME code, for optimum performance the number of blades is given by

$$C = \frac{8.5 \ Sin \ \beta_2}{1 - D_1/D_2}$$

Where β_2 is the outlet vane angle which has a range of $20^0 < \beta_2 < 90^0$

For this design $\beta_2 = 81^0$ and $D_1/D_2 = 0.65$

$$C = \frac{8.5 \ Sin \ \beta_2}{1 - D_1/D_2}$$

$$= \frac{8.5 \ Sin \ 81^0}{1 - 0.65}$$

$$= 24 \ \text{Blades}$$

c) Blade Width

It is given from ASME code specification that, the blade width is given by the formula. [Adejuyigbe, S.B. (2006).]

$$W = \frac{6(D_1/2)}{C + 1}$$

Where C is the number of blades

$$W = \frac{6\left(\frac{0.182}{2}\right)}{24 + 1} = 22 \ mm$$

d) Number of revolution per minute

For a centrifugal blower to deliver an air, the centrifugal head must be equal to the total head [Edward, H.S. (1995)].

Thus, $\frac{U_2^2 - U_1^2}{2g} = H_{man}$

Where U_1 and U_2 are he respective impeller velocities at inlet and outlet.

$$U_1 = \frac{\pi D_1 N}{60} \quad \text{and} \quad U_2 = \frac{\pi D_2 N}{60}$$

So,

$$U_2^2 - U_1^2 = 2gH_{man}$$

Where, H_{man} is the manometric head

$$\left[\frac{\pi D_2 N}{60}\right]^2 - \left[\frac{\pi D_1 N}{60}\right]^2 = 2 \times 9.81 \times 10$$

$$\left[\frac{\pi \times 0.280 \times N}{60}\right]^2 - \left[\frac{\pi \times 0.182 \times N}{60}\right]^2 = 196$$

$$2.1 \times 10^{-4} N^2 - 9.0 \times 10^{-5} N^2 = 196$$

$$N = 1247 \; rev / \min$$

e) The Manometric Head

This is head which the blower needs to overcome before it delivers and it is given as

$$H_{man} = \frac{U_2^2 - U_1^2}{2g}$$

$$U_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.182 \times 1247}{60} = 12 \; mls$$

$$U_2 = \frac{\pi D_2 N}{60} = \frac{\pi \times 0.280 \times 1247}{60} = 18 \; mls$$

Therefore ,

$$H_{man} = \frac{18^2 - 12^2}{2 \times 9.8} = 10 \; m$$

Analysis of the various velocities of the blades

From the velocity diagram

V_2 = absolute velocity of the fluid at outlet

V_{r2} = relative velocity of the fluid at outlet

Vf_2 = velocity of flow at outlet

Vw_2 = whirl velocity at outlet

U_2 = Velocity of the blades at outlet

β_2 = Outlet Vane angle

The corresponding inlet parameters are:

V_1, V_r, Vf_1, V_{w1}, U_1 and β_1

From the continuity equation

$$Q = \pi D_2 \beta_2 Vf_1 = \pi D_1 \beta_1 Vf_1 = 0.1128 \ m^3 / s$$

$$Vf_2 = \frac{Q}{\pi D_2 \beta_2} = \frac{0.1128}{\pi \times 0.280 \times 0.022}$$

$$= 5.8 m / s$$

$$Vf_1 = \frac{Q}{\pi D_1 \beta_1} = \frac{0.1128}{\pi \times 0.182 \times 0.022} = 9 m / s$$

These are the velocities of flow at outlet and inlet.
For the absolute velocities of the flow V1 and V2 from the velocity diagram.

$$V_2 = \frac{Vf_2}{Sin \ \phi} \quad \text{and} \quad V_1 = Vf_1 \text{ (from initial assumption)}$$

Also in this design $\phi = 30^0$, $\beta_1 = 10^0$ and $\beta_2 = 81^0$.
These are taken from the range of optimum performance from ASME code.

$$V_2 = \frac{Vf_2}{Sin \ 30^0} = \frac{5.8}{Sin \ 30^0} = 11.6 m / s$$

$V_1 = Vf_1$ by assuming radial flow at inlet.
The whirl velocities at inlet and outlet are estimated from

$V_{w_1} = 0$ From initial assumption

$$V_{w_2} = \sqrt{11.6^2 - 9^2} = 7,3 m / s$$

The relative velocities Vr_1 and Vr_2 at inlet and outlet are given from velocity triangle.

$$Vr_2 = \frac{Vf_2}{Sin \ \beta_2} = \frac{5.8}{Sin \ 81^0} = 5.9 m / s$$

$$Vr_1 = \frac{V_1}{Sin \ \beta_1} = \frac{9}{Sin \ 10^0} = 52 \ m / s$$

f) Horse power required by the motor
This is given by the formula

$$\rho = \frac{Q \gamma H_{man}}{550}$$

Where, Q = volumetric flow rate m^3/s
$\gamma = \rho g$
ρ= density of air Kg/m^3

$$P = \frac{6.77 \times 1.002 \times 9.8 \times 10}{550}$$

$$P = 1.2 hp$$

g) The pressure through the impeller

By applying the energy, equation from the entrance to exit of the impeller including the energy head.

$$\frac{P_2 - P_1}{r} = H_{man} - \frac{V_2^{\,2} - V_1^{\,2}}{2g}$$

$$P_2 - P_1 = r\left[H_{man} - \frac{V_2^{\,2} - V_1^{\,2}}{2g}\right]^2$$

$$P_2 - P_1 = 9.8 \times 1.002\left[10 - \frac{11.6^2 - 9^2}{2 \times 9.8}\right]$$

$P_2 - P_2 = 71$Pa

(h) Efficiency of the blower

The efficiency of the blower is given by

$$\int_m = \frac{\text{manometric head}}{\text{Head imparted by imepller to liquid}}$$

$$H_{man} = 10\ m$$

Head imparted by impeller =

$$\frac{V_{w2} U_2}{g} = \frac{7.3 \times 18}{9.8} = 13.4$$

$$\int_m = \frac{10}{13.4} = 74.6\%$$

This is the manometric efficiency developed by impeller

III. DICUSSIONS AND CONCLUSION

For optimum performance the ratio of internal diameter to external diameter falls between 0.4 to 0.7(ASME).The diameter of the blade is a design parameter which others depend opon. If the speed (rpm) of the impeller remains the same then the larger the impeller diameter the higher the generated head. As the diameter of the impeller is increased, the tip speed at the outer edge of the impeller increases commensurately. However, the total energy imparted to the fluid as the diameter increases goes up by the square of the diameter increase. This can be understood by the fact that the fluid’s energy is a function of its velocity and the velocity accelerates as the fluid passes through the impeller . The blower was specifically designed for rotary furnace operation (500kg capacity) but it can equally be adapted for use in operations that needed air supply system for its smooth running, most especially in combustion-related operations.

REFERENCES

[1]. Alan, M.(1979). Engineering Fluid Mechanics McGraw-Hill Int. Book Co., Auckland, New Zealand.

[2]. Dr C.O.Nwajagu(1994) , Foundry Theory and Practice Published by B. C. Publisher Ltd Nigeria Metallurgical Engineering Department Enugu state University.

[3]. D.Taylor Lyman,(1996), Metal hand book 8th edition vol.15 published by American Society of metal Ohio.

[4]. Edward, H.S. (1995). Mechanical Engineers Reference Book, 12th ed. The Bath Press, Bath, UK.

[5]. Oyelami, A.T.; and Adejuyigbe, S.B. (2006). The design of a radiation-recuperative heat exchanger for a 200kg capacity rotary furnace. AU J.T. 10(2): 101-8.

[6]. Tuton, R.K. (1984). Principles of Turbo-machinery. S & FN Spon, London, England