The Design of a Closed-Type-Impeller Blower for a 500kg Capacity Rotary Furnace

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Abstract

The blower was specifically designed to ensure a near-complete standardization in the design of EMR500. The suction conditions and other application data are used to calculate parameters such as the suction specific speed, specific speed, required shaft power, required impeller dimension and volute dimensions. The impeller was made to be of the closed type because of the level of pressure needed most especially when the rotary furnace is incorporated with a heat exchanger. 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.

Keywords: Specific speed, shaft power, impeller, volute casing, suction eye.

Introduction

Quite a large number of rotary furnaces are found to perform below the expected capacity, most especially in terms of thermal efficiency, mainly because of non-standardization of designs (Oyelami and Adejuyigbe 2006). It is in this regard that this design was done in order to standardize the design of 500kg capacity rotary furnace system.

The principle involved in the design of a blower is similar in virtually every important aspect as that of a centrifugal pump except for the fact that the term “centrifugal pump” is often associated with liquid as its working fluid while the blower is meant to work on air (Edward 1995). The blower can therefore be described as a device, which converts ‘driver’ energy to kinetic energy in a fluid by accelerating it to the outer rim of a revolving device known as an 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 key idea here is that the energy created is kinetic energy. The amount of energy given to the fluid corresponds to the velocity at the edge or vane tip of the impeller. Addison (1995) established that the faster the impeller revolves or the bigger the impeller is, the higher will be the velocity of the fluid at the vane tip and the greater the energy imparted to the fluid.

Fig. 1. Sectional drawing of a blower.

Fig. 1 illustrates a cross section of the blower designed. Fluid enters the inlet port at the center of the rotating impeller, or the
suction eye. As the impeller spins in a counterclockwise direction, it thrusts the fluid outward radially, causing centrifugal acceleration. As it does this, it creates a vacuum in its wake, drawing even more fluid into the inlet. Centrifugal acceleration creates energy proportional to the speed of the impeller (Csanady 1981). The faster the impeller rotates, the faster the fluid movement and the stronger its force. Impellers are the rotating blades that actually move the fluid. They are connected to the drive shaft that rotates within the blower casing. The impeller is designed to impart a whirling or motion to the air in the blower.

**Impeller Diameter and RPM**

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 (Dryden 1982). A wider diameter impeller accelerates the fluid to a final exit velocity greater than the proportional increase in the diameter. However Horlock (1984) established that the wider the diameter of the impeller, the greater the possibility of the blower vibration due to possible out-of-balance mass as a result of the expected high speed of rotation.

**The Impeller**

Impeller is the most important part of the blower components because of the fact that its performance inadvertently determines the blower’s performance. An impeller is essentially a disk shaped structure with vanes that create the actual suction in a blower. The impeller is always placed directly onto the shaft of the electric motor so that it spins at a very high speed. The effects of centrifugal force acting upon the spinning air within the impeller create the suction. As the impeller rotates, Von Cube and Steimle (1981) confirms that the spinning air moves outward away from the hub, creating a partial vacuum which causes more air to flow into the impeller. The most important impeller parameters can be grouped into three categories:

- Geometrical Parameters: Tip diameter, hub diameter and tip width;
- Operating conditions: Inlet total pressure, inlet total temperature and fluid density;
- Performance characteristics: mass flow parameter, pressure ratio and specific speed.

**Technical Performance Data**

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Figure 2 above shows the velocity triangle of the air flow leaving the impeller. Since there are no inlet guide vanes, the entering flow has no tangential component of motion. The entering flow is therefore in radial direction, and $v_{r1}$, which is the radial component of the absolute velocity, is the same as the inlet velocity, $V_1$. That is, $v_{r1} = V_1$.

It is assumed that the flow is completely guided by the blades and that the flow angles coincide with the blade angles. Also:

Inlet Vane Angle $\beta_1 = 29^\circ$;
Outlet Vane Angle $\beta_2 = 31^\circ$;
Volute Inside radius $r_1=0.08$m;
Volute Outside radius $r_2=0.335$m;
Vane Width at the Suction Eye $b_1=0.0355$m;
Vane Width at the tapered end $b_2=0.024$m;
Rotational Speed $N = 3,430$rpm.
The Linear Speed at the inlet is
\[ U_1 = r_1 \omega = \frac{2\pi \times 3430 \times 0.08}{60} = 28.74 \text{ m/s} \; ; \]
\[ V_1 = v_{1r} = U_1 \tan \beta_1 = 28.74 \times 29 = 159.3 \text{ m/s} \; . \]

The expected flow rate is
\[ \dot{Q} = 2\pi r_1 v_{1r} = 2\pi \times 0.08 \times 0.0355 \times 15.93 = 0.2842 \text{ m}^3/\text{s} \; . \]

Applying continuity concept at the blower discharge,
\[ V_3 = \frac{\dot{Q}}{A_3} = \frac{0.2842}{0.134 \times 0.134} = 15.83 \text{ m/s} \; . \]

Applying Bernoulli equation between the room and blower discharge at 3,
\[ \frac{P_{atm}}{\rho g} + (\Delta h)_1 \rightarrow 2 = \frac{P_{atm}}{\rho g} + \frac{V_3^2}{2g} \; , \]
where:
- \( V_3 \) is the discharge velocity;
- \( \dot{Q} \) = Fluid flow rate;
- \( A_3 \) = Cross-sectional area at the discharge end;
- \( P_{atm} \) = Atmospheric pressure;
- \( (\Delta h)_{1 \rightarrow 2} \) = The total head change;
- \( \rho \) = Air density;
- \( g \) = acceleration due to gravity.

The total head change imparted to the flow by the blower is then
\[ (\Delta h)_{1 \rightarrow 2} = \frac{V_3^2}{2g} = \frac{15.83^2}{2 \times 9.81} = 12.77 \text{ m} \; . \]

But \( (\Delta h)_{1 \rightarrow 2} = \frac{U_2 v_{\theta 2} - U_1 v_{\theta 1}}{g} \) (Alan M. 1979),
where \( v_{\theta} \) is the tangential component of the fluid absolute velocity.

Since the flow enters radially, \( v_{\theta 1} = 0 \),
\[ v_{\theta 2} = \frac{(\Delta h)_{1 \rightarrow 2}}{U_2} \; , \]
where
\[ U_2 = r_2 \omega = \frac{2\pi N r_2}{60} = \frac{2\pi \times 3430}{60} \times 0.335 = 120.33 \text{ m/s} \; , \]
\[ v_{r 2} = \frac{\dot{Q}}{2\pi r_2 b_2} = \frac{0.2842}{2\pi \times 0.335 \times 0.024} = 5.625 \text{ m/s} \; , \]
\[ \tan \beta_2 = \frac{v_{r 2}}{U_2 - v_{\theta 2}} \; , \]
\[ v_{\theta 2} = U_2 - \frac{v_{r 2}}{\tan \beta_2} = 120.33 - \frac{5.625}{\tan 31} = 110.97 \text{ m/s} \; , \]
\[ V_2 = \sqrt{v_{\theta 2}^2 + v_{r 2}^2} = \sqrt{110.97^2 + 120.33^2} = 163.69 \text{ m/s} \;
\]
The power input to the blower is
\[ P = \rho \dot{Q} g h \; . \]
\[ = 1.239 \times 0.2842 \times 9.81 \times 12.77 = 44.1 \text{ kW} \; . \]

3D Drawings of the Blower Components
The three dimensional drawings of the major components of the blower designed are as shown in Figs. 3 – 17 below:
Fig. 7. Discharge flange.

Fig. 8. Volute casing sub-assembly.

Fig. 9. Electric motor/impeller coupler.

Fig. 10. Open impeller.

Fig. 11. Impeller front cover.

Fig. 12. Closed impeller sub-assembly.

Fig. 13. Blower stand sub-assembly.

Fig. 14. S/W isometric view of blower assembly.
Performance Characteristics

Two major means of measuring the performance of a blower are in the measurements of its efficiencies vis-à-vis some parameters and also calculating the impeller specific speed, which determines the suction rate.

Mechanical Efficiency vs. Flow Coefficients

Graphs of the mechanical efficiency versus the flow coefficient for various speeds (rpm) of the impeller of the blower are as shown in Fig. 18.

The mechanical efficiency at a particular flow coefficient is noted to be higher for lower rpm because of the tendency for more instability at higher rpm.
Pressure vs. Flow Coefficients

Fig. 19 shows the graphs of pressure coefficients against the flow coefficients for various revolutions of the impeller.

The flow and pressure coefficients are dependent on blades angles and blade widths (Anon. 2006; and Anon. 2006b).

![Graph of Pressure Coefficient vs Flow Coefficient for Different RPMs]

Fig. 19. Variation of pressure coefficient with flow coefficient for different RPMs.

Impeller Specific Speed

The specific speed of the impeller is calculated from

\[ N_s = \frac{n \sqrt{Q}}{H^{3/4}}, \]

where: \( N_s \) = specific speed; \( n \) = impeller speed in rpm; \( Q \) = volumetric flow rate in gpm; \( H \) = head in ft (Anon. 2005). Here:

\[ Q \text{ in gpm} = Q \text{ (in m}^3/\text{min}) \times \rho \text{ (in g/m}^3) = 0.2842 \times 60 \times 1239 = 21127 \text{gpm}, \]

\[ N_s = \frac{3430 \times \sqrt{21127}}{12.77 \times 3.281} = 11899 \approx 11900. \]

The value falls within the typical range of \( N_s \) for this type of impeller which is 11,000 to 18,000.

Sources of Blower Failures/Poor Performance

There are generally four reasons why a blower will not perform optimally. These reasons can be categorized into four. They are Vibration, Lack of Performance, Excessive Noise and Premature Component Failure (Edward 1995).

Vibration

There are generally eight sources of vibration in a blower. These are summarized thus:

1. Material build-up on the wheel.
2. Loose mounting setscrews, bearings, bolts, or couplings.
3. Misalignment or excessive wear of belts coupling or bearings.
5. Material build-up on the wheel.
6. Excessive system pressure or restriction of airflow due to closed dampers.
7. Inadequate structural support or mounting.
8. Externally transmitted vibration.

Lack of Performance

The general reasons for lack of performance of a blower are:

1. Incorrect system design calculation or testing procedures.
2. Incorrect blower rpm.
3. Blower wheel rotating in wrong direction - check motor leads.
4. Improper wheel to inlet cone clearance.
5. Inlet or discharge air leaks, clogged filters, coils or damper settings.
6. System effect due to improper inlet or discharge connection.

Excessive Noise

The various causes of noise during a blower’s operation are:

1. Fan operating near “stall” due to incorrect system design or installation.
2. Vibration originating elsewhere in the system.
3. System resonance or pulsation.
4. Improper location or orientation of fan intake and discharge.
5. Inadequate or faulty design of supporting structures.
7. Loose accessories or components.
8. Worn bearings.

**Premature Component Failure**

Component failure will occur due to any of the following causes:
1. Abrasion or corrosion of internal fan components.
2. Vibration due to impeller out of balance.
3. Lack of lubrication of bearings.
4. Misalignment or power transmission components or bearings.
5. Bearing failure from incorrect or contaminated lubricant or grounding through the bearings while arc welding.
6. Extreme ambient or air-stream temperatures.

**Conclusion**

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. The development of this design will ensure a near-complete standardization of the 500kg capacity rotary furnace setup by ensuring uniformity in design since the blowers hitherto used are of various designs and most of them needed repair/modifications before they can perform satisfactorily.

**References**


