DESIGN AND DEVELOPMENT OF A BLOWER FOR DOWNDRAFT BIOMASS GASIFIER

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Abstract

Inadequate and improper supply and control of air are major causes of failures in gasifiers. Lack of adequate number of blowers as well as the use of unsuitable type of blower also lead to the total collapse of the gasification process. This research presents the design of a blower for use in the operation of a batch type downdraft gasifier, using analytical and numerical methods from the literature. The basic air requirements for gasification were considered as input parameters in order to determine the impeller design parameters. They included the geometrical parameters such as the tip diameter, hub diameter and tip width; operating conditions such as the inlet total pressure, inlet total temperature and fluid density; and performance characteristics such as mass flow parameter, pressure ratio and specific speed. The blower’s physical characteristics, geometric parameters and performance parameters were found and calculated to be β₁, β₂, U₁, V₂, as 42°, 90°, 17.52 m/s, 26.2 m/s and; d₁min, d₂, Z as 186 mm, 190 mm, and 8; while Q, P, N, η as 0.5m³/s, 221W, 2800 rpm and 85.1% respectively. To accommodate the parameters calculated and ensure optimal performance of the blower, a 0.5 HP motor was selected and used. However, as blowers are generally associated with some problems such as vibration, lack of performance, excessive noise and premature component failure, extensive tests will be carried out.

Keywords: Blower, air requirements, geometric parameters, operating conditions, performance characteristics.

1. Introduction

Blowers and fans are turbo machines widely used in present industrial and domestic life. It is important to recognize that the design of such turbo machine is an interdisciplinary process, involving aerodynamics, thermodynamics, fluid dynamics, stress analysis, vibration analysis, the selection of materials, and the requirements for manufacturing (Bleier, 2013).

The principles involved in the design of a blower is similar to 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. 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 (Adekunle et al., 2008).

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.

An extensive work has been done on gasification but low capacity batch gasifiers have not received the adequate attention. To this, should be added the fact that materials for blower construction need to demonstrate high heat resistance and ability to withstand sudden changes in both direction and temperatures. Inadequate and improper supply and control of air are other major causes of failures in a gasifiers. These failures are also due to lack of adequate as well as suitable blower which leads to the total collapse of the gasification process.

This research presents the design and development of a blower for use in the operation of a low capacity, batch type downdraft gasifier. While designing the blower, the most important impeller design parameters to be determined were grouped into three categories and used in the construction. The groups are: Geometrical Parameters (Tip diameter, hub diameter and tip width), Operating conditions (Inlet total pressure, inlet total temperature and fluid density), and Performance characteristics (mass flow parameter, pressure ratio and specific speed).

2. Materials and methods
2.1 Theoretical Design Calculations
Proper sizing of blowers involved the determinination of many system factors and prioritizing them into requirements versus preferences. Some of these factors were the static pressure that the blower must overcome, the required average air flow volume, the shape and direction of the desired air flow, space limitations, audible noise allowances, available power, efficiency, air density, and cost. Air flow and static pressure, along with available power considerations are generally the most critical for blower designs. These three address the fundamental questions on how much air is needed and what is it going to cost in system power to get it.
2.2 Theoretical Design of the Blower

Basically, the major parameters determined while designing a blade on the blower are shown in Figure 1:

![Diagram of blower components](image)

Figure 1: Descriptive view of major components and parts of a blower (Adekunle et al., 2008).

The blower provides the necessary airflow that is needed for the gasification of biomass. Blowers are usually available in AC or DC. The blower to be used should be capable enough to overcome the pressure exerted by the biomass and, subsequently, by the char. A high pressure blower is usually ideal for down-draft type gasifier reactor, while low-pressure blower is used for cross-draft type reactor. Thus the amount of air needed for gasification needs to be calculated.

This is very important in determining the size of the blower needed for the reactor to be used in gasifying biomass. As shown, this can be simply determined using the rate of consumption of the biomass fuel (FCR), the Stoichiometric air of biomass (SA), and the recommended equivalence ratio (ε) for gasifying the biomass.

On an average 1 kg of biomass produces about 2.5 m$^3$ of producer gas at standard temperature and pressure (S.T.P.). In this process it consumes about 1.5 m$^3$ of air for combustion. For complete combustion of wood about 4.5 m$^3$ of air is required. Thus biomass gasification consumes about 33% of theoretical stoichiometric ratio for wood burning (i.e. 1.485 m$^3$/hr) (Anil, 2014). On the other hand, rice husk was found to require 2.7 m$^3$ of air per hour for complete gasification, while the air density ($\rho_{a}$) is 1.26 kg/m$^3$ (Abubakar, 2014).

2.3 Blower Design Procedure

2.3.1 Design of Impeller

It was assumed that the duct size is 10% higher than impeller eye or impeller inlet diameter, therefore,

$$D_{duct} = 1.1D_{eye} = 1.1D_1$$

(1)

During 90° turning from eye inlet to impeller inlet, the eye inlet velocity remained same as absolute velocity vector at the entry of the impeller

$$V_{eye} = V_1 = V_{ml}$$

(2)
The tangential velocity component needs to be 10% higher than axial velocity component for better induction of flow (Vibhakar, 2012).

So, Inlet Tip velocity
\[ U_1 = 1.1V_1 = 1.1V_{ml} \]  
Discharge, \[ Q = \frac{\pi D_{eye}^2}{4} \times V_1 \]  
Hence \[ Q= 0.5 \text{ m}^3/\text{s} \] and the impeller rotation \[ N= 2800 \text{ rpm} \]

\[ U_1 = \frac{\pi D_1 N}{60} \]

### 2.3.2 Impeller Blade Angle
\[ \tan \beta_1 = \frac{V_s}{U_1} \] (Vibhakar, 2012)

### 2.3.3 Impeller Width at Inlet
The number of blades in a centrifugal blower can vary from 2 to 64 depending upon the application, type and size. Too few blades are unable to fully impose their geometry on the flow, whereas too many of them restrict the flow passage and lead to higher losses. Most of the efforts to determine the optimum number of blades have resulted in only empirical relations given by Vibhakar (2012) as:

\[ Q = [\pi D_1 - Zt] \times b_1 \times V_{ml} \]

### 3.4 Impeller Outlet Parameters
The Blower Power= \( \Delta P \times Q \)

Considering 10% extra (Bleier, 2013) in order to accommodate flow recirculation and impeller exit

Hydraulic losses
So 1.1x the blower power = 1.1 \times 490.6 = 539.66 W

\[ Power, P = m \times W_s \]  
\[ Euler \ power = mV_{u2}U_2 \]  
Taking \( V_{u2} = 0.8U_2 \) (A slip factor = 0.8 was assumed for radial blades) (Bleier, 2013).

The width of blade at inlet was taken to be equal to outlet blade width

\[ V'_2 = \sqrt{V_{u2}^2 + V_{m2}^2} \]  
\[ \tan \alpha'_2 = \frac{V_{m2}}{V_{u2}} \]

### 2.3.5 Design of Volute Casing
Analysing steady flow energy equation at inlet and exit:

\[ \frac{P_1}{\rho_1} + \frac{1}{2}V_1^2 + gz_1 + W_s = \frac{P_2}{\rho_2} + \frac{1}{2}V_4^2 + gz_2 \]  
Neglecting potential difference
\[ V_4^2 = \frac{-2(P_2-P_1)}{\rho_f} + V_1^2 + 2W_s \]
The width of volute casing \( b_v \) is normally 2 to 3 times the Impeller inlet blade width \( b_1 \) (Vibhakar, 2012). For this work, 2.5 times was used, hence, the incremental volute angle with respect to increase in radius of casing.

\[
V_2' = \sqrt{V_{a2}^2 + \xi_{m2}^2} \\
\tan \alpha_2' = \frac{\xi_{m2}}{V_{a2}} \\
\Delta r = (r_{at 360^0} - r_{at 0^0})
\]

The volute radii at different volute angles are given in Table 1:

<table>
<thead>
<tr>
<th>( \Theta ) in degree</th>
<th>Volute Radius ( r ) in m</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.121</td>
</tr>
<tr>
<td>60</td>
<td>0.150</td>
</tr>
<tr>
<td>120</td>
<td>0.178</td>
</tr>
<tr>
<td>180</td>
<td>0.207</td>
</tr>
<tr>
<td>240</td>
<td>0.235</td>
</tr>
<tr>
<td>300</td>
<td>0.264</td>
</tr>
<tr>
<td>360</td>
<td>0.292</td>
</tr>
</tbody>
</table>

The radius of volute Tongue, \( r_t \) was calculated from Eq. 19

\[
r_t = 1.075r_2 = 1.075 \times 0.11 = 0.11825
\]

The angle of volute tongue was obtained from Eq. 20

\[
\theta_t = \frac{132\log_{10}\left(\frac{r_t}{r_2}\right)}{\tan a_2}
\]

2.3.6 Hydraulic, Leakage and Power losses

Vibhakar (2012) gave the various losses as follows:

2.3.6.1 Leakage loss

\[
Q_l = C_d \times \pi \times D_1 \times \delta \times \sqrt{\frac{2P}{\rho}}
\]

Here, \( P = \frac{2}{3} \Delta P_3 \)

2.3.6.2 Suction pressure loss

The suction pressure loss \((dp_{suc})\) was calculated from Eq. 22

\[
dp_{suc} = \frac{1}{2} \times k_1 \times \rho \times V_{eye}^2
\]

where: \( k_1 \) is loss factor probably of the order of 0.5 to 0.8

2.3.6.3 Impeller pressure loss

The Impeller pressure loss \((dp_{imp})\) was obtained from Eq. 23

\[
dp_{imp} = k_{ii} \times \frac{1}{2} \times \rho (W_1 - W_2)^2
\]

At design point of maximum efficiency \( k_{ii} \) is in order of 0.2 – 0.3 for sheet metal blades and rather less for aerofoil section. Selecting \( k_{ii} = 0.25 \)
2.3.6.4. Volute pressure loss
The Volute pressure loss \((dp_{vc})\) was calculated from Eq. 24.
\[
dp_{vc} = k_{iii} \times \frac{1}{2} \times \rho (V_2^2 - V_4^2) \tag{24}
\]

2.3.6.5 Disc friction loss
\[
T_{df} = \pi f \rho \omega_2 \frac{r_2^5}{5} = \pi \dot{\rho} (U_1/r_2)^2 \frac{r_2^5}{5}
\]
Where \(f\) is material friction factor in order of 0.005 for mild steel sheet metal.

2.3.7 Efficiencies
2.3.7.1 Hydraulic efficiency
\[
\eta_h = \frac{\Delta P}{\Delta P + dP_{suc} + dP_{imp} + dP_{vc}} \tag{26}
\]

2.3.7.2 Volumetric efficiency
\[
\eta_{vol} = \frac{Q}{Q + Q_L} \tag{27}
\]

2.3.7.3 Total efficiency
\[
\eta_{total} = \eta_h \times \eta_{vol} \tag{28}
\]

2.3.8 Ideal shaft power required to run the blower
\[
P_{total} = \frac{P + dP_{suc} + dP_{imp} + dP_{vc} (Q + Q_L)}{\eta_{total}} + \text{Power loss due to disc friction} \tag{29}
\]

Bleier (2013) suggests the adoption of a very conservative approach allocating large safety margins, resulting in oversized blower which operate at flow rates much below their design values and, consequently, at very poor efficiency. Thus, we recommend a 0.5 HP motor to serve for the supply of the air required to the gasifier.

Therefore, Torque = \(\frac{221 \times 60}{2 \pi \times 2800} = 0.754 \text{Nm}\)

2.3.9 Shaft diameter
\[
D_s = \sqrt[3]{\frac{16 \times \text{Factor of safety}}{\pi T}} \tag{30}
\]
\[
D_s = 11.9 \text{mm}
\]

For the shaft, a 15 mm diameter mild steel rod was selected to accommodate the torsional and bending loads the shaft may be subjected to as a result of high speed rotation and weight of the blower impeller.

2.3.10 Blade profile
Blade profile is made by tangent arc (Vibhakar, 2012). When this method is used, the impeller is divided into a number of assumed concentric rings, not necessarily equally spaced between inner and outer radii. The radius \(R_b\) of the arc is defining the blade shape between inner and outer radii.
\[
R_b = \frac{r_2^2 - r_1^2}{2 [r_1 \cos \beta_1 - r_2 \cos \beta_2]} \tag{31}
\]
2.3.11 The Blower Construction
The materials used for the construction of the blower are Mild steel gauge 16 (0.9 mm) sheet, an angle iron of dimensions (25 mm x 3 mm) and a 0.5 HP, 2800 rpm electric motor obtained in the market that can give the required power output to the blower.

The blower casing was made of mild steel 190 mm diameter by 2 piece, cut and drilled a hole of 17 mm and 70 mm diameters respectively. A strip of (70 mm x 600 mm) mild steel was measured and cut to form the circumference of the casing. On the other circular plate, four holes of 7mm diameter was drilled at a square of 60 mm.

A 50 mm diameter round pipe mild steel was measured and cut to a length of 300 mm while the other end was chamfered to 60°. An angle iron of 25 mm x 3 mm x 4 pieces was cut, drilled a hole of 80 mm diameter and welded to the circular sheet.

A circular sheet mild steel of 90mm diameter was cut, marked, divided into 8 equal parts and drilled a hole of 14 mm diameter at the center. A strip of 30 mm width x 70 mm long was cut and welded to the circular sheet which formed the blades.

Finally, the components were assembled and coupled to the 0.5 HP, 2800 rpm electric motor.

The tools and equipment used for the fabrication of the blower impeller and its housing were hacksaw, bench, hammer, divider, center punch, steel rule, chisel, hand shear, hand files, scriber, mallet, work bench V-block, drilling machine, welding machine and a bench vice.

3. Results and Discussion
During the design of the blower, the following physical characterisitics and performance parameters were determined as shown in Table 2, while the proposed design’s velocity diagram is as presented in Figure 2.

![Velocity Diagram](image)

**Figure 2:** Inlet and outlet velocity diagram of the blower
Table 2: Physical Characteristics and Performance Parameters of the Designed Blower

<table>
<thead>
<tr>
<th>S/N</th>
<th>Design Parameters</th>
<th>Nomenclature</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Minimum impeller inlet diameter</td>
<td>$d_{1\text{min}}$</td>
<td>186 mm</td>
</tr>
<tr>
<td>2</td>
<td>Impeller outside diameter</td>
<td>$d_2$</td>
<td>190 mm</td>
</tr>
<tr>
<td>3</td>
<td>Eye or shroud diameter</td>
<td>$d_s$</td>
<td>100 mm</td>
</tr>
<tr>
<td>4</td>
<td>Area of shroud</td>
<td>$A_s$</td>
<td>$7.86 \times 10^{-3}$ m$^2$</td>
</tr>
<tr>
<td>5</td>
<td>Impeller inlet blade width</td>
<td>$b_1$</td>
<td>5.44 mm</td>
</tr>
<tr>
<td>6</td>
<td>Blade peripheral velocity at inlet</td>
<td>$U_1$</td>
<td>17.52 m/s</td>
</tr>
<tr>
<td>7</td>
<td>Absolute velocity at impeller inlet</td>
<td>$V_i$</td>
<td>15.92 m/s</td>
</tr>
<tr>
<td>8</td>
<td>Blade angle at impeller inlet</td>
<td>$\beta_1$</td>
<td>42°</td>
</tr>
<tr>
<td>9</td>
<td>Impeller outlet blade width</td>
<td>$b_2$</td>
<td>5.44 mm</td>
</tr>
<tr>
<td>10</td>
<td>Outlet blade velocity</td>
<td>$U_2$</td>
<td>32.72 m/s</td>
</tr>
<tr>
<td>11</td>
<td>Radial component of outlet velocity</td>
<td>$V_{r2}$</td>
<td>26.2 m/s</td>
</tr>
<tr>
<td>12</td>
<td>Blade exit angle</td>
<td>$\beta_2$</td>
<td>90°</td>
</tr>
<tr>
<td>13</td>
<td>Number of blades</td>
<td>$Z$</td>
<td>8</td>
</tr>
<tr>
<td>14</td>
<td>Slip factor</td>
<td>$\mu$</td>
<td>0.8</td>
</tr>
<tr>
<td>15</td>
<td>Actual exit velocity peripheral component due to slip</td>
<td>$V_{u2}$</td>
<td>26.2 m/s</td>
</tr>
<tr>
<td>16</td>
<td>Actual absolute exit velocity</td>
<td>$V'_2$</td>
<td>29.5 m/s</td>
</tr>
<tr>
<td>17</td>
<td>Actual relative velocity</td>
<td>$W_2'$</td>
<td>14.94 m/s</td>
</tr>
<tr>
<td>18</td>
<td>Air velocity at impeller eye</td>
<td>$V_{\text{eye}}$</td>
<td>15.92 m/s</td>
</tr>
<tr>
<td>19</td>
<td>Air density</td>
<td>$\rho$</td>
<td>1.26 kg/m$^3$</td>
</tr>
<tr>
<td>20</td>
<td>Volute tongue diameter</td>
<td>$d_3$</td>
<td>0.236 m</td>
</tr>
<tr>
<td>21</td>
<td>Volute tongue radius</td>
<td>$R_3$</td>
<td>0.118 m</td>
</tr>
<tr>
<td>22</td>
<td>Volute radius at various angles from tongue</td>
<td>$r_\theta$</td>
<td>See table 1</td>
</tr>
<tr>
<td>23</td>
<td>Volute width</td>
<td>$b_v$</td>
<td>0.136 m</td>
</tr>
<tr>
<td>24</td>
<td>Total pressure loss</td>
<td>$\Delta P_{\text{total}}$</td>
<td>122.9 Pa</td>
</tr>
<tr>
<td>25</td>
<td>Leakage across impeller inlet and casing</td>
<td>$Q_L$</td>
<td>0.0225 m$^3$/s</td>
</tr>
<tr>
<td>26</td>
<td>Radial clearance between impeller and casing</td>
<td>$\delta$</td>
<td>2 mm</td>
</tr>
<tr>
<td>27</td>
<td>Discharge coefficient</td>
<td>$C_D$</td>
<td>0.6</td>
</tr>
<tr>
<td>28</td>
<td>Volume flow rate</td>
<td>$Q$</td>
<td>0.5 m$^3$/s</td>
</tr>
<tr>
<td>29</td>
<td>Torque due to disc friction</td>
<td>$\tau$</td>
<td>0.71N/m</td>
</tr>
<tr>
<td>30</td>
<td>Power loss due to disc friction</td>
<td>$P_{\text{disc}}$</td>
<td>208 W</td>
</tr>
<tr>
<td>31</td>
<td>Hydraulic efficiency</td>
<td>$\eta_h$</td>
<td>88.8%</td>
</tr>
<tr>
<td>32</td>
<td>Volumetric efficiency</td>
<td>$\eta_v$</td>
<td>95.9%</td>
</tr>
<tr>
<td>33</td>
<td>Total efficiency</td>
<td>$\eta_T$</td>
<td>85.1%</td>
</tr>
<tr>
<td>34</td>
<td>Power required to run the blower</td>
<td>$P$</td>
<td>221 W</td>
</tr>
<tr>
<td>35</td>
<td>Power of motor selected</td>
<td></td>
<td>0.5 HP</td>
</tr>
</tbody>
</table>
However, the blowers are generally associated with some problems such as vibration, lack of performance, excessive noise and premature component failure. These limit the speed of the blower. An addition of adequate balls bearings on the shaft was done to reduce the noise.

4. Conclusion
The following conclusions were drawn at the end of this work:

i. A blower was designed and constructed for use in the operation of a downdraft gasifier, using locally available materials.

ii. The geometric parameters were determined to be: \( d_{1\text{min}}, d_2, d_s, A_s, b_1, b_2, Z \) as 186 mm, 190 mm, 100 mm, 7.86 \( \times 10^{-3} \) m\(^2\), 5.44 mm, 5.44 mm and 8 respectively.

iii. The blower physical characteristics were found to be: \( \beta_1, \beta_2, U_1, V_{r2}, V_1, V_{u2}, V_2, V_{\text{eye}} \) as 42°, 90°, 17.52 m/s, 26.2 m/s and 15.92 m/s respectively.
iv. The blower performance parameters were calculated to be: \( Q, P, N, \Delta P_{\text{total}}, \eta \) as 0.5 m\(^3\)/s, 221W, 2800 rpm and 85.1% respectively.

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