Evaluation of the Performance of a Developed Centrifugal Fan for Chemicals Application in Orchard Farm

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Abstract
A centrifugal fan for chemical application in orchard farm, producing air stream with a maximum value of 9.81 m/s was developed and its performance evaluated in the research. The fan performance was evaluated with three parameters, namely: linear velocity (air velocity), fan output power and mechanical efficiency. The fan input power or the shaft power was obtained at different sheave diameters of 110, 130, 150 and 190 mm using established relationships. The fan parameters obtained were subjected to analysis of variance in order to determine if there were significant differences between these parameters at the selected sheave diameters at 95 % confidence level. Findings of the research revealed that the fan input power ranged from 1.2039 to 3.16 kW; the analytical fan output power ranges from 0.0043 to 0.0161 kW; the air velocity ranged from 7.73 to 9.81 m/s; while the airflow rates ranged from 0.0043 to 0.0161 m^3/s. The developed centrifugal fan has a maximum of 68.42 % mechanical efficiency, and capable of delivering air stream at a velocity of 9.81 m/s. The calculated parameters were significant at 95 % confidence level at different sheave diameters. The centrifugal fan was found suitable for chemical application in orchard protection operations.

Key Words: *Air flow velocity, Centrifugal fan, fan output power, Mechanical efficiency*

1.0 Introduction

Centrifugal fans (blowers), axial fan and compressors are all equipment that move air. They are differentiated only by the method adopted as well as their level of involvement. The American Society of Mechanical Engineers (ASME) uses specific ratio, that is, the ratio of discharge pressure to suction pressure, to classify the air moving devices into fans, blowers and compressor (Cevik, 2010). UNEP (2006) described centrifugal fan as a mechanical device for moving air or other gases, by increasing the speed of air stream through relative increase in the speed of the rotating impellers. Centrifugal fans are by far the most prevalent type of fan used in the Heating, Ventilation and Air Conditioning (HVAC) industry today. They are often cheaper and simpler in construction than axial fans (Singh *et al.,* 2011). They are used in transporting gas or materials and in ventilation system for buildings. They are also well-suited for industrial processes and air pollution control systems.

Spraying of crops can be viewed in multiple ways, which changes according to the changes in the ranking of some values placed on them by people, among which spraying efficiency, costs and environment impact are important issues (Osterman *et al.*, 2014). However, from the engineering point of view, these issues consist of proper coverage of tree canopy, reduction of pesticides consumption, drift and other pesticide losses into the soil, air and water (Doruchowski *et al.,* 2011). Orchard is an intentional plantation of trees or shrubs that is maintained for food production. An orchard is a type of farm where fruits and nuts are grown on trees and shrubs. Examples of orchard fruits are apples, pears, oranges, bananas and cherries. Examples of orchard nuts are pecans, walnuts, and almonds. Orchards comprise fruit- or nut-producing trees, which are generally grown for commercial production. Pesticide application in orchards is mostly carried out using axial fan air-assisted sprayers. These machines are often provided with few options for adjusting the air flow rate and the vertical spray liquid profile (Pergher, 2006).

The consequence is that large amounts of air are employed to convey the droplets from the sprayer to the top of the plants. The air flux generated by an axial fan strikes with the same magnitude the spray jets from all nozzles, therefore the liquid sprayed towards the lower parts of the canopies. The resultant effect is that, on one hand, the air velocity which hits the leaves closer to the sprayer is very high, even excessive. This fact increases the risk that leaves are turned according to the air flux direction, and a consistent fraction of spray overpasses the canopy and generates drift (Cross *et al*., 2003). On the other hand, the air velocity directed towards the top of plants is generally not sufficient to guarantee an even coverage, and an adequate spray penetration into the higher parts of tree canopy, especially when the latter is very dense (Cross *et al*., 2003). The aim of this study is to evaluate the performance of a developed centrifugal fan capable of creating an artificial hurricane of 9.81 m/s which can address the observed problems, for use in orchard farm protection.

1.1 Objectives of the research

The objectives of the research were to:

i. Determine the air power or output power of generated air stream with the speed of 9.81 m/s.
ii. Design and fabricate the centrifugal fan according to the specification in (i) above.
iii. Determine the input power or the shaft power of the centrifugal fan that would produce the computed output power.
iv. Evaluate the performance of the developed centrifugal fan (that is, the mechanical efficiency of the fan).

2.0 Materials and Methods

2.1 Research details

A centrifugal fan for possible use on an air blast assisted tree crop sprayer, was developed and evaluated. The air power or output power of the developed air stream was determined from the given speed of 9.81 m/s, and the input or shaft power of the centrifugal fan was determined by the output power that was computed. The fan was designed based on the mathematical expression that was generated from literature.

2.2 Materials

The materials that were used for the construction of the centrifugal fan are as follows:

i. Galvanized steel pipe: The fan duct was fabricated from galvanized steel pipe of length 865 mm, diameter 275 mm and thickness 5 mm.
ii. Stainless steel sheet metal: The fan blade was fabricated from stainless steel sheet metal with blade diameter of 263 mm, hub diameter of 87.7 mm and thickness of 1.5 mm
iii. Angular pipe: The frame that accommodates the fan was made from angular pipe of thickness 6 mm.
iv. Solid Shaft with 15 mm shaft diameter was used to transmit power from the shaft of the electric motor to the fan shaft.
v. Galvanized steel sheet: Galvanized sheet metal of 2 mm thickness was used to construct the housing of the fan.
vi. Angular Iron: Angular iron of 5 mm thickness was used to construct the seat for the electric motor.
vii. Welding electrode: 12 gauge welding electrode was used to weld the different parts together.
viii. Bolt and nuts: 5 pairs of bolts and nuts were used to fasten each component parts of the machine to one another.
ix. Sheaves: Different sizes of sheaves were used to transmit power from the electric motor to the fan.

Equipment that were used in the fabrication of the fan are:

i. Tachometer: Tachometer of model 371 that is capable of measuring speed from 0-9999 rpm was used to test the fan.
ii. Gas cutting machine
iii. Grinding machine.
iv. Welding equipment.
v. Drilling machine.
vi. Sledge hammer.
vii. Bench vice.

2.3 Centrifugal fan design

In designing the centrifugal fan, the following parameters were calculated and used:

2.3.1 Air density

The density of the air was determined using equation 1 given by BEE (2010). The density is dependent on altitude and temperature. Air is the same as gas.

$$\textit{Air or gas density } (\gamma) = \frac{273 \times 1.293}{273 + t^{o}C} \qquad 1$$

Where:

$t°C$ − temperature of gas/air at site condition, taken as 20°C

$$(\gamma) = \frac{273 \times 1.293}{273+20}$$

$(\gamma) = 1.20 \text{ kg/m}^3$

Density of air = 1.20kg/m^3

2.4 Velocity of airstream

Velocity of the airstream is given by

V_{as} = velocity of airstream, m/s, given from the objective as 9.81 m/s (2)

2.5 Air dynamic pressure

The air dynamic pressure was determined from equation 3 as given by Ananthemarayanam (2015):

$$P_d = ½(\rho_{air})(V^2) \quad (3)$$

$$= \frac{1}{2}\,(1.20\ \times\ 9.81^2)$$

$$P_d = 57.74\ \text{N/m}^2$$

Where:

P_d = air dynamic pressure, Pa (or N/m^2)

ρ_{air} = air density, kg/m^3

V = velocity of air stream, m/s

2.6 Diameter of fan wheel

The diameter of the fan wheel is represented by D_{fw} cm. Six values of the fan wheel diameters were selected, which were: 23 cm, 34 cm, 51 cm, 76 cm, 112 cm and 167 cm. The fan wheel with 23 cm diameter will be used to produce the required fan velocity of 9.81 m/s.

2.7Area of fan inlet

The area of the fan inlet was computed using equation 4 as given by Ananthemarayanam (2015):

$$A_{inlet} = \pi(D_{fw}^2)/4 \quad (4)$$

$$= \frac{3.142\ X\ 0.23^2}{4}$$

$$A_{inlet} = 0.041548\ \text{m}^2$$

Where,

A_{inlet} = area of fan inlet, m^2

D_{fw} = diameter of fan wheel, cm (or m)

2.8 Airflow rate

The airflow rate was computed with equation 5 as given by Ananthemarayanam (2015).

$$Q_{air} = A_{inlet}\ V_{as} \quad (5)$$

$$Q_{air} = 0.041548\ X\ 9.81$$

$$Q_{air} = 0.40758\ \text{m}^3/\text{s}$$

Where:

Q_{air} = airflow rate (m^3/s)

2.9 Fan output power or air power

The fan output power or air power was computed using equation 6 as given by Ananthemarayanam (2015):

$$P_{out} = (P_d)(Q_{air}) \quad (6)$$

Where:

$P_{out,}$ = Fan output power or air power, kW
P_d= air dynamic pressure, N/m^2
Q_{air}= airflow rate, m^3/s
$P_d = ½(\rho_{air})(V^2)$
P_d= 0.5 × 1.20 × 9.81^2 = 57.74N/m^2
Q_{air} = 0.40758 m^3/s
P_{out}= 57.74 × 0.40758 = 23.53W
P_{out} = 0.02353 kW or 0.0315 hp

The required fan output power, P_{out} is 0.02353 kW or 0.0315 hp.
Fan input power (power of the electric motor) supplied to the shaft at varying sheave diameter was selected to be 0.02353 kW (0.0315 hp).
The P_{out} supplied to the sheave shaft, becomes the input power for the fan to determine the fan output power.

2.10 Brake horse power

The brake horse power was computed from equation 7 as given by Ananthemarayanam (2015),

$$\textit{Brake horse power (bhp)} = \frac{P_{out}}{\eta_{total}} = \frac{0.02353\ kW}{0.45} = 0.0523\text{ kW} = 0.07\text{ hp} \qquad 7$$

Where:
Assumed total efficiency, η_{total} = 45%
Air dynamic pressure = 57.74 N/m^2 = 0.5774kN/m^2

From the American Blower Co. Table A, 9 in (23 cm) diameter fan running at 2278 rpm delivers 1880 cfm (0.88726 m^3/s)(Q_1) of air, developing a static pressure(SP_1) of 3.0 in(76.2 mm) water gage and consumes 2.03 HP(1.514 kW) bhp. If the airflow rate is increased to 1.86964 m^3/s (that is, 3961.61 cfm)(Q_2), increase static pressure (SP_2) and fan speed (N_2) were determined.

2.11 Determination of the fan speed

The fan speed was determined from equation 8 as given by Severns and Fellows (1998).

$$\text{Fan speed } (N_2) = \frac{Q_2}{Q_1} \text{ x } N_1 = \frac{1.86964}{0.88726} \times 2278 \qquad 8$$

Where:
N_2 = fan speed (rpm)
N_1 = initial or output fan speed (rpm)
Q_2 = increased in air flow rate (m^3), given as 1.86964 m^3
Q_1 = air flow rate (m^3/s), given as 0.88726 m^3/s
Fan speed = 4800.21 say 4800 rpm.

2.12 Fan static pressure

The fan static pressure was determined using equation 9 as given by Ananthemarayanam (2015),

$$\textit{Static Pressure } (SP_2) = SP_1 \text{ x } \frac{N_2^2}{N_1^2} \qquad 9$$

Where:
SP_2 = final static pressure (mm)

SP_1 = Initial static pressure (mm), given as 76.2 mm from American Blower CO. table A

N_1 = initial or output fan speed (rpm), given as 2278 rpm from American Blower CO. table A

N_2 = fan speed (rpm), 4800 rpm

$$\text{Static Pressure} = 76.2 \text{ x} \frac{4800^2}{2278^2}$$

$$= 76.2 \text{ x} \frac{23040000}{5189284}$$

$$= 76.2 \text{ x } 4.43991$$

Static pressure = 338.32 mm(13.31 inches of water).

2.13 Diameter of the fan wheel

The diameter of the fan wheel was determined using equation 10 as given by Ananthemarayanam (2015).

$$\textit{Diameter of fan wheel } (d_2) = d_1 \text{x} \left(\frac{Q_2}{Q_1}\right)^{1/3} \qquad 10$$

Where:

d_2 = diameter of fan wheel (cm)

d_1 = diameter of fan wheel (cm), obtained from American blower Co. table as 23 cm

Q_2 = finalair flow rate (m^3), obtained from American blower Co. table as 1.86964 m^3

Q_1 = Initial air flow rate (m^3/s), obtained from American blower Co. Table as 0.88726 m^3/s

$$\text{Diameter of the fan wheel} = 23 \text{ x} \left(\frac{1.86964}{0.88726}\right)^{1/3} = 23 \text{ x } 1.282042324$$

Diameter of fan wheel = 29.486 say 30 cm.

2.14 Key design

The key was designed using the relationship given by Uffe *et al.* (1983) in equation 11

Shearing stress in the key is

$$\tau = {}^{F}/_{bL} \qquad 11$$

By introducing torque in the equation 11 through the expression in equation 13, a more useful equation of shearing stress in the key was obtained in equation

$$F = {}^{2T}/_{D} \qquad 12$$

$$\tau = {}^{2T}/_{bLD} \qquad 13$$

The compressive stress, σ on the sides of the key (and on the keyway walls) is given by:

$$\sigma = {}^{4T}/_{tLD} \qquad 14$$

Where:

T = Torque transmitted by the shaft (N.mm)

F = Tangential force acting at the circumference of the shaft (N)

d = Diameter of shaft (mm)

L = Length of key (mm)
b = width of key (mm)
t = thickness of key (mm) and
τ and σ = Shear and crushing (compressive) stresses for the material of key (Pa)

Thus, the length and width of the key were determined from equations 15 and 16 respectively:

$$L = 1.5D \quad (15)$$

L = 1.5 x 15
L = 22.5 mm

$$b = 0.25D \quad (16)$$

b = 0.25 x 15
b =3.75 mm
Where:
L = length of the key, mm
b = width of the key, mm
D = diameter of the shaft, mm

Table 1 presents the standard metric sizes of square keys.

2.15 Fan input /shaft power

The fan input power at different sheave diameter was determined from equation 17 as given by Severns and Fellows (1998)

$$kW_2 = kW_1 \text{ x } \frac{N_2^3}{N_1^3} \quad (17)$$

Where;
kW_2 = input/shaft power, kW
kW_1 = power of electric motor, kW
N_1= speed of the electric motor, rpm
N_2 = speed of the selected sheave, rpm

Table 1 Standard metric sizes of square keys

Standard metric sizes of square keys	
Shaft size (D, mm)	Key size (b, mm)
12-15	3
15-20	4
20-30	6
30-40	8
40-50	10
50-60	14
60-70	16
70-80	18
80-90	20
90-100	24

Source: (Uffe *et al*., 1983)

2.16 Research procedure

2.16.1 Fabrication of the centrifugal fan

Centrifugal fan blade of diameter 23 cm was procured from the new market Kontagora, Nigeria. Metal sheet of 2 mm thickness was cut into different sizes and welded together to produce the fan casing. Angular pipe of thickness 6 mm was cut into sizes and used to produce the frame upon which the entire fan sit on. Angular pipe of 1.5 inches was cut with hacksaw and welded together with welding machine to produce the engine seat upon which the engine that will power the machine will sit on. Shaft of diameter 15 mm thickness was used to suspend the fan in between the fan and its casing, and also used to transmit power from engine motor to the fan. The suspended shaft was held by a pair of bearing at both ends for ease of rotation. Five pairs of bolts and nuts were used to hold each part together.Plate 2 shows the pictorial view of the fabricated fan for use on air – blast generator, while Figure 1 shows the exploded view of the fan.

Plate 2 Pictorial view of the fabricated centrifugal fan.

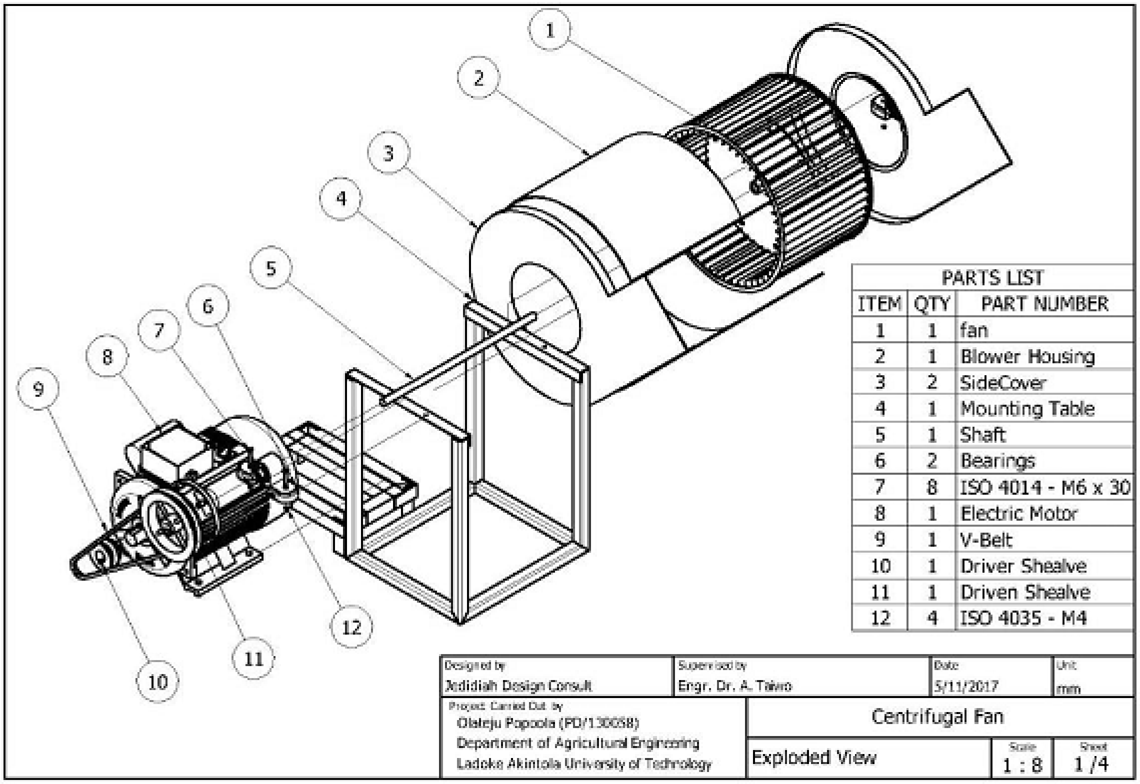

Figure 1 Exploded View of the Centrifugal Fan

2.16.2 Performance evaluation of the centrifugal fan

The fabricated centrifugal fan was evaluated in terms of the air power generated using different sheave diameters. The mechanical efficiency of the centrifugal fan was also determined.

2.16.3 Procedure for data collection

The fabricated centrifugal fan was connected to a power source in the Agricultural Education Department Workshop of the Federal College of Education, Kontagora, Nigeria. Four sheave diameters were used for the evaluation (110 mm, 130 mm, 150 mm and 190 mm). The fan speed was measured using a digital tachometer of model 371, capable of measuring between 0 - 9999 rpm. At each diameter of the fan sheave, the tachometer was switched on and placed to measure speed of the fan. The different sheave diameters were further used to determine the respective air power or output power using the relationship in the equation 6.

From the value of air speed recorded for each sheave diameter, the linear velocity of the hurricane was determined using equation 18:

$$V = \omega r \qquad 18$$

$$V = \frac{2\pi}{60} n\, r = \pi D n / 60$$

Where:

V = linear air velocity, m/s

ω = angular velocity, rad/sec
n = fan rotational speed, rpm
r = radius of sheave, mm
D= sheave diameter, mm

2.16.4 Mechanical efficiency

This is a measure of the effectiveness with which a machine transformed the energy/power input to the device into an output energy/power.

The efficiency of the centrifugal fan that was developed was calculated using equation 19 as follows for each of the sheaves:

$$\text{Mechanical Efficiency} = \frac{power\ output}{power\ input} \times 100\ \% \qquad \mathbf{19}$$

Where;
Power input = 0.02353 kW
Power output calculated from equation 3.19 for each sheave.
$Pout = P_d \times Q\ (kW)$
Where;
P_d is the air dynamic pressure, N/m^2
Q is the airflow rate (Area x air velocity), m^3/s.

2.16.5 Power calculation using excel software

The calculated parameters were computed into an excel software, and the fan diameter of 17 cm that produced the required fan velocity of 9.81 m/s was selected. The power output was also calculated from the excel package. The calculated parameters include the fan input power or the shaft power at different sheave diameters and their corresponding speeds in rpm, the airflow rates in m^3/s, and the air velocities in m/s, all determined at the different sheave diameters.

2.17 Statistical analysis

Data obtained on various parameters were analyzed using SPSS (Levesque, 2007). Standard Error of Means (SEM) was used to separate the means where there are statistical significant differences.

3.0 Results and discussion

3.1 Fan output power

The fan output power of the artificial hurricane at a velocity of 9.81 m/s has been determined following established relationship on equations 6. The following parameters were determined to obtain the fan output power of 0.02353 kW (0.0315 hp). These parameters include; the area of the fan inlet of 23 cm (0.23m) was 0.04155 m^3, the air density was 1.20 kg/m^3, the dynamics pressure was 57.74 N/m^2, the airflow rate was determined to be 0.40758 m^3/s.

It was discovered that, the fan output or air power increases as the diameter of the sheave increases, and also, the airflow rate increases. The highest fan output or air power of 0.016 kW was obtained at the sheave diameter of 190 mm. Though, the fan speed (rpm) decreases as increase in the sheave diameter, but the air velocity of 9.81 m/s was attained with the sheave diameter of 190 mm which was the aim of the project.

3.2 Fan input power/shaft power

Table 3 shows the results of the calculated shaft power (at different sheave diameter and the corresponding speeds in rpm), the air velocities, and the airflow rates at different sheave diameters. The electric motor sheave (pulley) diameter, the electric motor speed and the electric motor power used as the input power were 0.06 m, 1440 rpm and 3.75 kW respectively.

From Table 2, the least sheave diameter of 110 mm produced shaft power of 3.16 kW which was the highest while the largest sheave diameter of 190 mm produced the least shaft power of 1.2039 kW. The shaft power decreases with increase in sheave diameter. Similarly, the sheave rotational speed measured using the tachometer decreases with increase in sheave diameter with the highest of 1360 rpm at 110 mm diameter and the least of 986 rpm obtained using 190 mm diameter.

The highest air velocity of 9.81 m/s was obtained when a sheave diameter of 190 mm was used and a shaft power of 1.2039 kW. Consequently, the lowest air velocity of 7.73 m/s was obtained from a sheave diameter of 130 mm producing a shaft power of 1.8411 kW.

Analytically, the maximum fan output power of 0.016 kW was obtained at an air flow rate of 0.2786 m^3/s from a sheave diameter of 190 mm. while the least sheave diameter of 110 mm produced the lowest fan output power of 0.0043 kW at an air flow rate of 0.0744 m^3/s. This also implies that the airflow rate is directly proportional to the sheave size.

Tables 3, 4 and 5 show the analysis of variance (ANOVA) of the sheave diameter, air power and mechanical efficiency. There is a significant difference between the sheave diameter and the air power at 95 % confidence level. Similarly, there was significant difference between the air power and the air velocity at 95 % confidence level.

3.0 Discussion

The highest air velocity of 9.81 m/s was obtained when a sheave diameter of 190 mm was used and a shaft power of 1.2039 kW. Consequently, the lowest air velocity of 7.73 m/s was obtained from a sheave diameter of 130 mm producing a shaft power of 1.8411 kW. This result can be compared with similar one by Ozigis *et al* (2015), for the developed centrifugal fan for a fluidized bed combustor which obtained fan output power of 1.61 kW and outlet air velocity of 6.85 m/s.

Table 4 shows the analysis of variance of the fan power. P value of 0.001 indicates that fan diameter is a significant model term. The R- square value 97.22% shows that fan sheave diameter has high effect on the air power produced from the machine. Table 5 shows the ANOVA test result for the air velocity of the fan. Like the fan power, the P value of 0.001 reflects the significance of the model term (fan diameter) on air velocity. It also indicates that varying air characteristics can be done by varying fan sheave diameter.

Table 2 Results of calculated performance parameters of the developed centrifugal fan

Electric motor Dia. (m)	Sheave Dia. (m)	Electric motor speed, rpm	Sheave speed, rpm	Electric motor power, kW	Fan Input power (analytical), kW	Area of sheaves (m^2)	Air velocity, m/s	Airflow rate, Q, m^3/s	Fan output power, kW	Mechanical Efficiency (%)
0.06	0.11	1440	1360	3.75	0.02353	0.0095	7.83	0.0744	0.0043	18.27
0.06	0.13	1440	1136	3.75	0.02353	0.0133	7.73	0.1030	0.0060	25.50
0.06	0.15	1440	1107	3.75	0.02353	0.0177	8.69	0.1540	0.0089	37.82
0.06	0.19	1440	986	3.75	0.02353	0.0284	9.81	0.2786	0.0161	68.42

Table 3 Analysis of variance (ANOVA) for air power (kW)

Source	DF	SS	MS	F	P
Fan sheaves diameter	3	1.42318	0.47439	46.68	0.001
Error	4	0.04065	0.01016		
Total	7	1.46383			

S = 0.100809 R-Sq = 97.22 % R-Sq (adj) = 95.14 %

Table 4 Analysis of variance (ANOVA) for air velocity (m/s)

Source	**DF**	**SS**	**MS**	**F**	**P**
Fan sheaves diameter	3	5.5620	1.8540	92.70	0.00
Error	4	0.0800	0.0200		
Total	7	5.6420			

S = 0.100809 R-Sq = 97.22 % R-Sq (adj) = 95.14 %

Also, for the mechanical efficiency of the developed centrifugal fan for chemical application, the results of the ANOVA test indicated the machine performance efficiency is acceptable at R-square value of 97.22% (Table 6). From the ANOVA results, it can be observed that there is a significant difference between the sheave diameter and the air power, between the air power and the air velocity at 95 % confidence level. Also, at 95% confidence level, there was significant difference between the air power and mechanical efficiency.

Table 5 Analysis of variance (ANOVA) for mechanical efficiency (%)

Source	DF	SS	MS	F	P
Fan sheaves diameter	3	1.44135	0.48045	67.20	0.001
Error	4	0.02859	0.00715		
Total	7	1.46994			

S = 0.100809 R-Sq = 97.22 % R-Sq (adj) = 95.14 %

5.0 Conclusion

The following conclusions were drawn from the research:

i. An air power of 0.02353 kW has been determined for a generated airstream by the fan at an air velocity of 9.81 m/s.
ii. The fan input power (shaft power) generated and the corresponding air velocities using different sheave have been determined. The larger the diameter of the sheave, the lower the fan speed in rpm and the higher the linear air velocity.
iii. A centrifugal fan with an air power of 0.02353 kW and 9.81 m/s air velocity has been designed and fabricated.

iv. A maximum mechanical efficiency of 68.42 % was achieved with the fabricated centrifugal fan.

6.0 Recommendations

The following recommendations are pertinent sequel to the findings of the research:

i. It is recommended that, a nozzle should be incorporated to the fan, so as to aid easy delivery of air to sprayers and others.
ii. Further studies or modifications should be done to improve or increase the fan efficiency.

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