

DESIGN & FLOW FEATURES OF AXIAL FAN USED IN AN AIR COOLED HEAT EXCHANGER BY EXPERIMENTAL ANALYSIS

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ABSTRACT

A heat exchanger is a piece of equipment built for efficient heat transfer from one medium to another. The media may be separated by a solid wall, so that they never mix, or they may be in direct contact. They are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, natural gas processing, and sewage treatment. The role of air flow plays vital role in heat exchanger. Presently heat exchangers use the air flow rate of 5-6 m³/s which gives less efficiency. Therefore the design of more than 7 m³/s air flow rate in the heat exchanger is done to maintain better efficiency. The required parameters of air flow fan are air flow, operating temperature, static pressure, Nominal fan speed, External diameter.

The modification in existing fan was done and desired air flow optimum velocity was achieved 15.23 m/s at circular duct and 6.675 m/s to archive required parameters.

Keyword: Heat Exchanger, Axial Flow Fans, 1 Tube-Axial Fan, Petroleum Refineries.

I INTRODUCTION

There are two general classifications of fans: the centrifugal or radial flow fan (see ED-2400) and the propeller or axial flow fan. In the broadest sense, what sets them apart is how the air passes through the impeller. The propeller or axial flow fan propels the air in an axial direction (Figure 1.1) with a swirling tangential motion created by the rotating impeller blades. In a centrifugal fan the air enters the impeller axially and is accelerated by the blades and discharged radially (Figure 1.2).

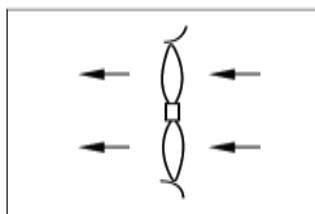


Figure 1.1 Axial Flow

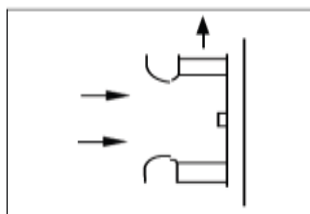


Figure 1.2 Centrifugal Flow

1.1 Axial Flow Fan: Axial flow fan is shown on fig 1.3

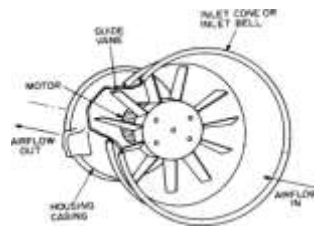


Figure 1.3: Axial flow fan

1.2 Axial Flow Fan Types

Propeller Fans: Sometimes called as the panel fans, propeller fans are the lightest, least expensive and most commonly used fans. These fans normally consist of a flat frame or housing to be mounted in a wall or in a partition to exhaust air from a building. This exhausted air has to be replaced by fresh air, coming in through other openings. If these openings are large enough, the suction pressure needed is small. The propeller fans, therefore, are designed to operate in the range near free delivery, to move large air volumes against low static pressures. These fans can be built both direct drive and belt drive (Figure 1.4 and Figure 1.5).



Figure 1.4: Propeller fan with direct drive



Figure 1.5: Propeller fan with belt drive

1.2.1 Tube-axial Fans

A tube-axial fan is a glorified type of propeller fan with a cylindrical housing about one diameter long, containing a motor support, a motor and a fan wheel. The motor can be located either on upstream or downstream of the fan wheel. The fan wheel of a tube-axial fan can be similar to that of a propeller fan. It often has a medium sized hub diameter,



Figure 1.6: Tube-Axial Fan

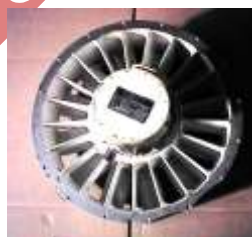


Figure 1.7: Vane Axial Fan



Figure 1.8: Single -Stage Axial Flow Fan

1.3 Performance of Axial Flow Fans

Figure 1.9 shows the shape of a typical pressure versus flow rate curve. Starting from the free delivery, the pressure value rises to a peak value. This is the good operating range for an axial flow fan. As the air volume decreases due to increasing restrictions, the axial air velocity decreases as well, resulting in an increased angle of attack and increased lift coefficients (the phenomena can be understood better in Chapter 2). The increase in the lift coefficient is responsible for the increase in the pressure. After the maximum lift angle is reached, the

flow can no longer follow the upper contour of the blades, thus separate from the surface of the blade. Separated flow results in a decrease in lift coefficient, thus a decrease in pressure occurs.

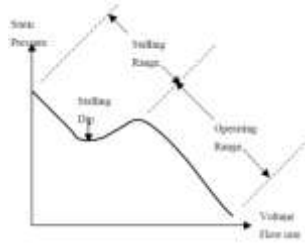


Figure 1.9: Static Pressure vs. Volume flow rate of an axial fan [24].

1.4 Heat exchangers

Heat exchanger geometry and area are constant by definition in the fixed plant simulations. However, changes to fluid flow rates will affect the fluid flow regime, equipment operating pressures, and temperature differences throughout each heat exchanger; which in turn will affect fluid physical properties. Changes in the fluid flow regime were assumed to have a greater affect on the overall heat transfer coefficient than changes in fluid properties (viscosity, density, heat capacity, and thermal conductivity).

II LITERATURE REVIEW

Alireza Falahat[ISSN: 2045-7057] et al [2011] In this study an attempt was made to find the best the best angle of attack and rotational velocity of a flat blade at a fixed hub to tip ratio for a maximum flow coefficient in an axial fan in a steady and turbulent conditions. In this study the blade angles are varied from 30 to 70 degrees and the rotational velocity is varied from 50 to 200 rad/sec for a number of blades from 2 to 6, at a fixed hub to tip ratio. The numerical and experimental results show that, the maximum flow coefficient is achieved at the blade angle of attack of between 45 to 55 degrees when the number of blades was equal to 4 at most rotational velocities. The numerical results show that as the rotational velocity increased, the flow coefficient increased but at very high rotational velocities the flow coefficient remained constant.

Oday I. Abdullah, Josef Schlattmann, et al [2012] In this paper the finite element method has been used to determine the stresses and deformations of an axial fan blade. Three dimensional, finite element programs have been developed using eight-node super parametric shell element as a discretization element for the blade structure. All the formulations and computations are coded in Fortran-77. This work was achieved by modeling the fan blade as a rotating shell. The investigation covers the effect of centrifugal forces on stresses and deformations of rotating fan blades. Extensive analysis has been done for various values, speed of rotation, thickness, skew angle, and the effect of the curvature on the stress and deformation. The numerical results have shown a good agreement compared with the available investigations using other methods.

Mahajan Vandana N.,Shekhawat Sanjay P et al [2012], In the present paper CFD based investigation has been reported in order to study the effect of change in speed of fan on velocity, pressure, and mass flow rate of axial flow fan. It has been observed that there is a significant change in mass flow rate, velocity of rotor and guide or stator vanes as the speed of fan is varied. As the performance of fan is directly dependent on mass flow output. So there should be a moderate velocity and pressure profile as all these parameters are co-related. In order to predict the mass flow output, velocity and pressure on stator and rotor section ,analysis is done using a

software ansys12.and to create a general idea about an axial flow fan a model is created using a modeling software catia v5.

III MATERIAL SELECTION & EXPERIMENTAL SETUP

3.1 Material Selection

In air cooled heat exchanger as they exposed to changing climate condition, the control of the air in the cooler is relevant. The problem exists in the fan provided in the heat exchanger available in Patel Heat Exchanger, Pvt. Limited, Govindpura, and Bhopal. It was given the air flow at the rate $5.944 \text{ m}^3/\text{s}$ where as it was required for efficient cooling at the rate $7 \text{ m}^3/\text{s}$ is above. The design of the fan was done and after proper fabrication the fan was tested for the desired rate of air flow. The following design considerations were taken into account to modify the design of the fan.

1. External diameter (d_2) of the axial flow fan
2. Internal diameter (d_1) of the axial flow fan.
3. Rotational speed (n).
4. Angle of Blade at enter (β°_1) & discharge (β°_2).
5. No. of Blade (z).
6. Shape & aerofoil section of the Blade.

In the design of single stage axial flow fan, the speed is mainly adapted to the choice of the driving machine, therefore especially for electrical driver, the speed are generally taken as 1400-1500 rpm. On the basis of quantities, I_p , n , the peripheral speed, the external fan diameter, the meridional speed and coefficient is calculated in the following

3.2 Design of axial flow fan (Modified design)

Table 3.1 Requirement of Axial Flow Fan & Input Parameter

S. No.	Require	Quantity
1.	Air Flow (v)	$7 \text{ m}^3/\text{Sec}$
2.	Operating Temperature	30° to 50°
3.	Static Pressure	30 mmwg
4.	Nominal fan Speed(N)	1450 rpm
5.	External Diameter(d_2)	0.8m
6.	External Diameter(d_2)	25 m

We take External Diameter d_2

$$d_2 = \sqrt[3]{\left(\frac{240V}{\pi^2 \phi n}\right)} \psi$$

Then we find Value ϕ

$$d_2 = \frac{60}{\pi n} \sqrt{(2\Delta P) / \sqrt{(p)}}$$

We find Value this Equation ψ

Take Equation of ψ

$$\psi = \frac{1}{\sigma^2 \delta^2}$$

Then we find the value of

$$\sigma = \frac{2.773}{\delta}$$

And Equation of

$$\phi = \frac{1}{\sigma \delta^3}$$

Equating both value ψ and ϕ

Then Find Value σ & δ

After the value speed coefficient and diameter coefficient is calculated, specific diameter r is read from the graph given in Figure

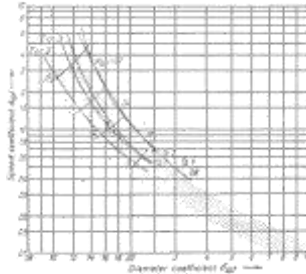


Fig. 3.1 Optimum curves with minimal hub ratios for different methods of Installation for impellers with low characteristics

Specific speed coefficient

The meridional speed is obtained from

$$c_m = V \left(\frac{\pi}{4} \right) (d_2^2 - d_1^2)$$

$$\mu_2 = \pi d_2 n / 60$$

$$\phi' = c_m / \mu_2$$

$$c_u = \frac{\Delta P}{\rho u \eta}$$

Then,

For t/l , the best value of which range selects the first value are being at the tip of the blade. Thus we have take number of blade Z

$$Z = (4\pi / 1.5) [\sin \beta_2 / \{1 - (r_1/r_2)\}]$$

Then obtain value of β_2 then we take value graph by Eck then we obtain value β_1

Then we obtain value s/l by fig. Weining Draw Velocity triangle

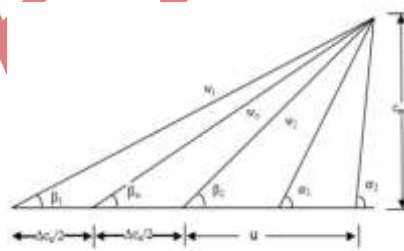


Fig.3.2: Velocity Diagram with Pre-rotation; Axial Outlet.

$$c_m = \omega_2 \sin \beta_2$$

Then we obtain value of w_2 in m/s and $u = \omega_2 \cos \beta_2$

Then we obtain value of u General relationship angle

Then we find value of β_∞

And

$$\cot \beta_\infty = \frac{1}{2} (\cot \beta_1 + \cot \beta_2)$$

$$c_m = \omega_\infty \sin \beta_\infty$$

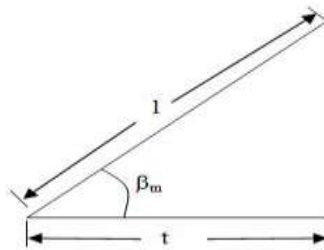


Fig.3.3: Distance of Tip and Length of Chord

We find the value of $\omega_\infty \quad v_\infty / 2 = (\beta_2 - \beta_1)/2 \quad \beta_m = (\beta_1 + \beta_2)/2$

And value of t/l

Then we take value of by graph $t=l \cos \beta_m$ then take l

And we take a value of μ

And $(1-\mu)/\mu =$ value of this equation

And Value

$$V = (v_\infty/2) \{ (1-\mu)/\mu \}$$

We obtain value v

$$\beta_1' = \beta_1 - v$$

$$\beta_2' = \beta_2 - v$$

$\Delta \beta = \Delta \beta_1 (l/t)^2$ according to Weining graph determining the increase of angle by profiling s , profile thickness, l chord length

$\Delta \beta =$ value of this equation

We take value of

$$\beta_m + \Delta \beta = \text{value of this equation}$$

3.3 Calculation of Axial Flow Fan

We take External Diameter $d_2 = 0.795$

$$d_2 = \sqrt[3]{\left(\frac{240V}{\pi^2 \phi n} \right)}$$

Then we find Value ϕ

$$0.795 = \sqrt[2]{\left(\frac{240 \times 7}{\pi^2 \phi \times 1450} \right)}$$

Then we find Value $\phi = 0.22$

$$d_2 = \frac{60}{\pi n} \sqrt{(2\Delta P)} / \sqrt{(\psi p)}$$

$$0.795 = \frac{60}{\pi \times 1450} \sqrt{(2 \times 30)} / \sqrt{(\psi \times 1/8)}$$

We find Value this Equation $\psi = 0.130$

Take Equation of ψ

$$\psi = \frac{1}{\sigma^2 \delta^2}$$

$$0.130 = \frac{1}{\sigma^2 \delta^2}$$

Then we find the value of

$$\sigma = \frac{2.773}{\delta}$$

And Equation of ϕ

$$\varphi = \frac{1}{\sigma \delta^2}$$

$$0.22 = \frac{1}{\sigma \delta^2}$$

Equating both value ψ and ϕ

Then Find Value $\sigma = 2.16$ & $\delta = 1.28$

After the value speed coefficient and diameter coefficient read from the graph given in Figure by Eck

Specific speed coefficient $\sigma = 0.379 n_{\text{sec}} (V^{1/2} / H^{3/4})$

$d_2 = 795 \text{ mm}$, & $d_1 = 795 \times 28 = 225 \text{ mm}$, $\eta = 73\%$

The meridional speed is obtained from $c_m = V / (\pi/4 (d_2^2 - d_1^2))$

$$c_m = 15.122 \text{ m/sec}$$

$$u_2 = \pi d_2 n / 60$$

$$u_2 = 60.73 \text{ m/sec}$$

$$\varphi' = c_m / u_2$$

$$\varphi' = 0.24$$

Then,

$$c_u = \frac{\Delta p}{\rho u \eta}$$

$$c_u = \frac{328.76}{u}$$

For t/l , the best value of which range selects the first value are being at the tip of the blade. Thus we have take number of blade $Z = 9$.

$$Z = (4\pi / 1.5) [\sin \beta_2 / \{1 - (r_1/r_2)\}]$$

Then find value $\beta_2 = 50.54^\circ \sim 50^\circ$

Then we take fig. blade pitch according to Zweifel

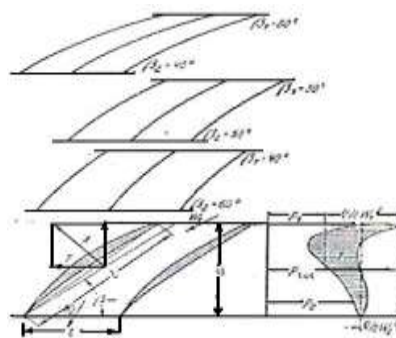


Fig.3.4: Blade Cascades with Optimum Pitch According to Zweifel.

Take according to Zweifel $\beta_1 = 30^\circ$

Then we obtain value $s/l = 0.066$ by fig. effect of Tip clearance of Weining

Draw Velocity triangle

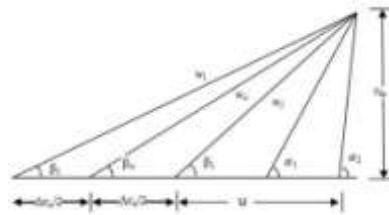


Fig.3.5: Velocity Diagram with Pre- Rotation; Axial Outlet.

$$c_m = \omega_2 \sin \beta_2$$

$$\omega_2 = 19.74 \text{ m/sec}$$

Then we obtain value of w_2 and

$$u = \omega_2 \cos \beta_2$$

Then we obtain value of General relationship angle Then we find value of And $u = 12.68 \text{ m/s}$

$$\cot \beta_\infty = \frac{1}{2} (\cot \beta_1 + \cot \beta_2)$$

$$\beta_\infty = 37.87^\circ$$

$$c_m = \omega_\infty \sin \beta_\infty \quad \omega_\infty = 24.63 \text{ m/s}$$

$$v_\infty / 2 = (\beta_2 - \beta_1) / 2 = 10^\circ \quad \beta_m = (\beta_1 + \beta_2) / 2 = 40^\circ$$

And value of t/l

Then we take value of by graph

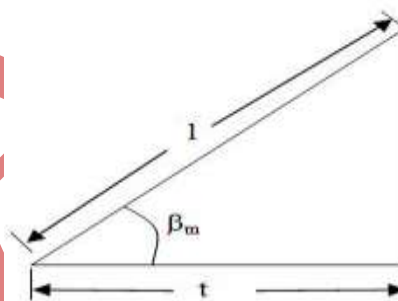


Fig.3.6: A Graph Between Distance of Tip And Length Of Chord

$$t = l \cos \beta_m \text{ then take } l = 0.085 \text{ m}$$

And we take a value of $\mu = 0.61$

And

$$(1 - \mu) / \mu = 0.639$$

And Value

$$V = (V_\infty / 2) \{ (1 - \mu) / \mu \}$$

$$V = 6.393$$

$$\beta_1' = \beta_1 - V$$

$$\beta_1' = 23.607^\circ$$

$$\beta_2' = \beta_2 - V$$

$$\beta_2' = 56.393^\circ$$

$\Delta \beta = \Delta \beta_1 (l/t)^2$ according to Weining graph determining the increase of angle by profiling s , profile thickness, l chord length

$$\Delta \beta = 2.39^\circ$$

We take value of

$$\beta_m + \Delta \beta = 42.39^\circ$$

3.4 Design Data of Impeller of Existing Fan and Modified Fan

Table3.2: Design data of the fan

Parameter	Value of Existing Design	Value of Modified Design
d_2	775 mm	795 mm
d_1	230mm	225 mm
β_1°	25	30
β_2°	45	50
L	85mm	85 mm
μ	0.54	0.61
v	8.51	6.393
	0.130	0.130
	0.22	0.22
C_m	16.13	15.32
	2.16	2.16
z	9	9

3.5 Experimental Setup

To evaluate the air flow velocity with the standard measurement criterions by the help of Anemometer, test setup was developed as shown in fig (3.7) & fig (3.8). A circular duct was placed at the fan side to evaluate the inlet air velocity. A rectangular duct was fitted at the exit end of the heat exchanger to find the velocity of the air leaving the heat exchanger. Duct Transverse method for air flow measurements for circular duct and rectangular duct was adopted.

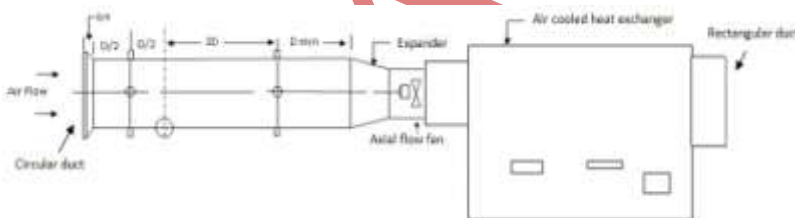


Fig 3.7: Block diagram of Experimental setup



Fig.3.8: Experimental setup

Several test locations for the evaluation of velocity were set along the strings as marked as A, B and C in the fig. (2). Anemometers have traditionally been employed for air duct balancing. This cumbersome task requires performing a traverse of the opening, measuring and manually recording the velocity

IV RESULT AND DISCUSSION

The fan of heat exchanger was designed and with modified dimension, the fan was fabricated and installed in heat exchanger. The modified fan was tested and the details of testing as given below:-

4.1 Testing of fan velocity at different string A, B and C.

The experimental set up was developed to test the air flow velocity of the both the type of duct is circular duct at inlet and rectangular duct at the outlet. To take the data statistically, circular duct was divided in

there to in to three parts string A, B and C and each string is divided in six parts to the observation efficiency and statistically. The observation of three string A, B and C of air flow velocity have taken are presented in table no. 4.1

Table 4.1: Air Flow Measurement On The Face Of the Duct By Duct Transverse Method For Circular Duct.

Measuring Point	Velocity (m/s)					
	String A		String B		String C	
	Existing Fan	Modified Fan	Existing Fan	Modified Fan	Existing Fan	Modified Fan
1	12.1	14	12.2	14	13.2	15
2	13	16.2	12.6	15.3	13.5	16.4
3	12.5	15.1	11.6	14.8	12.3	15.6
4	12.8	15.5	12.4	15.1	12.5	15.2
5	12.8	16	12	15	14.5	16.5
6	11.2	15	11	13.2	13.2	15.2

4.2 Calculation of Existing Fan

Statics Pressure (Δp)= 20 mmwg

Total velocity of duct =212.5.9m/s For average velocity =11.82 m/s

Air flow of the circular duct= A V F Where $F=1 = 5.944 \text{ m}^3/\text{s}$ Say $6 \text{ m}^3/\text{s}$

4.3 Calculation of Modified Fan

Statics Pressure (Δp) = 20 mmwg

Total velocity of duct =278.3 m/s For average velocity =15.5m/s

Air flow of the circular duct = A V F = $7.69 \text{ m}^3/\text{s}$

The data show that in string A, the observation of air velocity at point 1 and 6 which are closed to the end of the duct varies from 12.1 m/s and 11.2 m/s in existing fan respectively, where as the point 1 and 6 in modified fan, the air flow velocity varies from 14 m/s and 15 m/s respectively. The velocity at the point 2 and 5 in the existing fan was observed 13 m/s and 12 m/s respectively where as in the modified fan, the air flow velocity at point 3 and 4 points the modified fan where as the air velocity in 2 and 4 points in modified fan was observed at the rate of 15.1 m/s and 15.5 m/s respectively, the average air flow velocity in the string A was observed in the existing fan the rate of 12.4 m/s where as the average air flow velocity in the modified fan was observed at the rate of 15.26 m/s which is desired one.

The string is also divided in six parts take the data statically to represent all the point of the duct and compared the air flow velocity of the duct of the existing fan and modified fan. The data were taken 1, 2,3,4,5 and 6 points. The point 1 and 6 are varying close to the end of the duct where as the point 2 and 5 are found in the

middle of the duct. The air in the existing fan was observed 12.2 m/s and 11 m/s respectively. Where as the at the point 1 and 6 is the modified fan was observed 14 m/s and 13.2 m/s respectively.

The air flow rate at the point 2 and 5 in existing fan and modified fan was recorded as 13.6 m/s and 12 m/s in existing fan and 15.3 m/s and 15 m/s in modified fan. Similarly the air flow velocity rate in existing and modified fan at point 3 and 4 was found 11.6 m/s, 12.4 m/s and 14.8, 15.1 m/s respectively.

The average velocity in string in the string B at all point the point in existing and modified fan are found 11.96 m/s and 14.56 m/s respectively.

The data of string C show that the observation of air velocity at measuring point 1 and 6. Which are close to the end of the duct varies from 14.2 m/s and 13.2 m/s in existing fan respectively. Where as at point 1 and 6 in modified fan, the air flow varies from 15 m/s and 15.2 m/s respectively. The air flow rate at point 2 and 5 in existing fan and modified fan was observed as 13.5 m/s and 14.5 m/s in existing fan and 16.4 m/s and 16.5 m/s in modified fan. Similarly, the air flow velocity rate in existing and modified fan at point 3 and 4 was found 12.3 m/s, 12.5 m/s and 15.6 m/s and 15.2 m/s respectively.

The average velocity in string at all point the in existing and modified fan are found 13.2m/s and 15.6 m/s respectively.

The graphical optimization of value of air flow velocity was obtained of string A, B, and C of existing and modified fans.

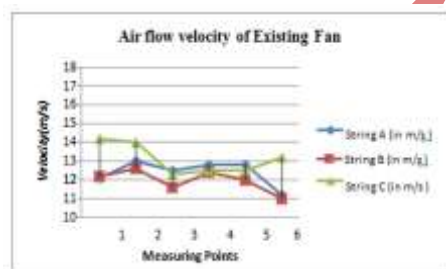


Fig. 4.1 : Velocity at different measuring point at circular duct of Existing fan .

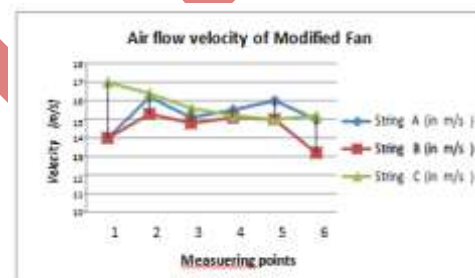


Fig. 4.2: Velocity at different measuring point circular duct of Modified fan.

Showered that graph of modified fan the graph of modified fan of string A , B and C were imposed in one graph and it indicate that at point 3 and 4 all the three graph are meeting. Therefore it may be predicted that highest average velocity at point 3 and 4 is about 13 m/s.

If showed that modified fan has given average highest velocity at the range of 16 m/s in comparison to the average highest velocity in existing fan is 13 m/s.

Testing of rectangular duct placed at outlet for testing of rectangular duct, the duct transversing method was used. The duct was divided horizontally to the Twelve parts is from C1 to C12 and in four parts vertically is R1 to get the result efficiently and may be verified statically.

Table 4.2: Air flow measurement on the face of the Duct by Duct Transverse method for rectangular duct of Existing Fan.

Test point	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12

R1	2.2	2	2.1	5	5.1	3	2.5	4.5	4.2	2.8	2.2	3.1
R2	1.4	2.5	2.8	5.5	3	1.5	0.4	1.3	2.9	2.8	1.5	2.5
R3	2.3	2.2	2.3	3.5	2.6	1	1.5	2.7	3.5	3.2	2.2	2.3
R4	3.1	3	2.9	2.5	5	4	4.3	4.7	6.5	2.1	1.8	2

Avg. Velocity of Exiting fan at rectangular section = $140/48$
= 2.91 m/s

Air flow of the Rectangular duct = $A V F$ Where $F=1$
= $1.82 \times 2.91 = 5.3 \text{ m}^3/\text{s}$

Table 4.3: Air flow measurement on the face of the Duct by Duct Transverse method for rectangular duct of Modified Fan.

Test point	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12
R1	2.9	2.4	2.8	6.2	6.2	4.2	3	5.1	5.1	3.4	2.8	3.6
R2	1.7	3.2	3.2	6	3.9	1.8	0.5	2.8	4.2	4.5	2.5	3.3
R3	2.7	2.7	3.3	4.9	3.6	1.6	1.5	3.4	4.6	4.1	2.8	2.8
R4	3.9	3.5	3.8	5.3	5.3	4.3	5.4	5.8	7	3.2	2.4	2.7

Avg. Velocity of Modified fan at rectangular section = $175.2/48$
= 3.65 m/s

Air flow of the Rectangular duct = $A V F$ Where $F=1$
= 1.82×3.65
= $6.64 \text{ m}^3/\text{s}$

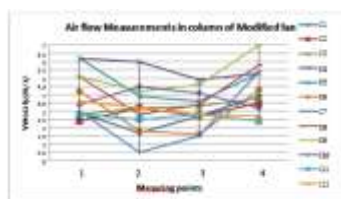


Fig.4.3: Velocity at different measuring point at rectangular duct in column wise of Modified fan.

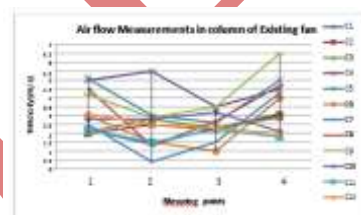


Fig.4.4: Velocity at different measuring point at rectangular duct in column wise of Existing fan.

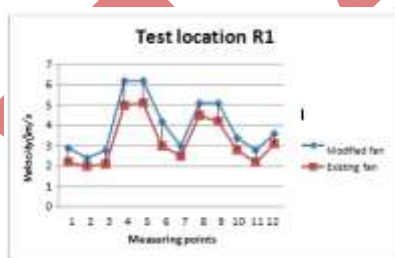


Fig.4.5: Velocity at different measuring point at row R1.



R2.

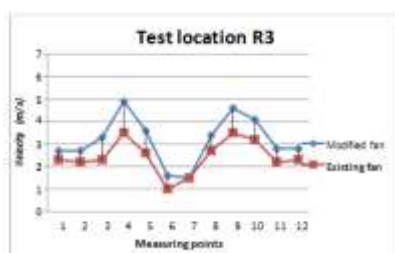


Fig.4.7: Velocity at different measuring point at row R3

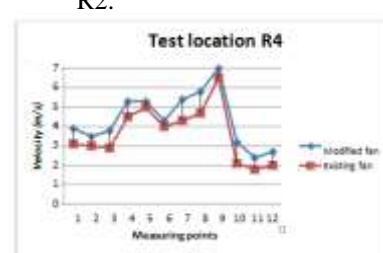


Fig.4.8: Velocity at different measuring point at row R4

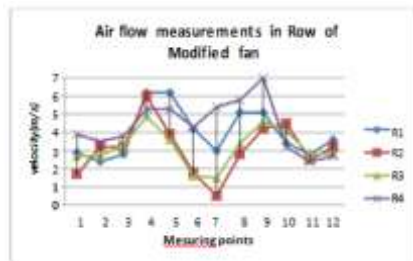


Fig.4.9: Velocity at different measuring point at rectangular duct in row wise of Modified fan

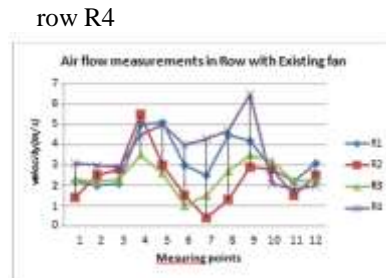


Fig.4.10: Velocity at different measuring point at rectangular duct in row wise of Existing fan.

Test was conducted to take observation on rectangular duct of modified. The data from C1 to C12 at R1 varied from 2.4 m/s to 6.2 m/s and 1.7 m/s to 6 m/s. The value of air flow velocity from C1 to C12 at R3 and R4 varied from 1.5 m/s to 4.9 m/s where as at R4 varied from 2.4 m/s to 7 m/s. It show that maximum value of air flow velocity obtained at C4 and C5 with corresponding to R1 and R4.

Similarly the test was conducted to take observation rectangular duct of modified fan. The data showed that from C1 to C12 at R1 point, the air flow velocity varied from 2 m/s to 5.1 m/s . Where 1 m/s to 3.2 m/s and 2 m/s to 5 m/s respectively.

The maximum value of air flow velocity at existing fan was obtained at C4 and C5 at corresponding to R1 is 5 to 5.1 m/s.

The data show that the maximum air flow velocity is obtained in the middle of the rectangular duct in Existing fan as well as modified fan.

V CONCLUSION & FUTURE SCOPE

The data of air flow velocity in terms of bar diagram is presented in fig. 5.3m .The result showed that the maximum volume of air flow rate is obtained in existing, required and modified fan are $5.994 \text{ m}^3/\text{s}$, $7 \text{ m}^3/\text{s}$ and $7.65 \text{ m}^3/\text{s}$ respectively. The testing result showed that the cooling effect was obtained maximum with modified fan air flow volume $7 \text{ m}^3/\text{s}$.

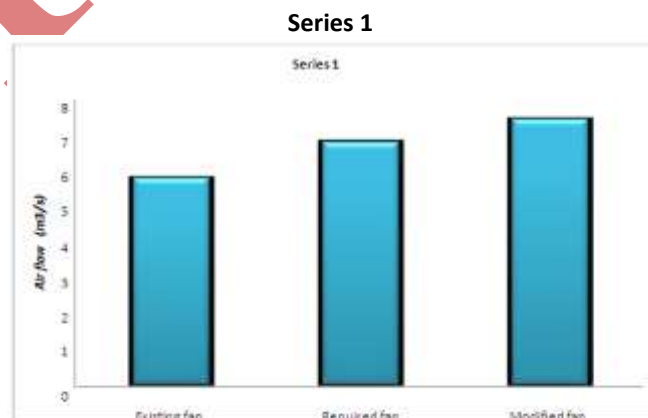


Fig .5.1: A bar chart of Air flow volume of Existing fan, required fan and modified fan.

5.1 Future Scope

In the present distribution work, the detailed design procedure for an axial flow fan of heat exchanger application has been presented experimental studies have also been performed for evaluating the performance of the axial flow fan and necessary modification for required air flow velocity is also carried out. The testing of the modified axial flow fan was done and desired air flow rate was obtained.

In future some mathematical model may be developed for carrying out the performance evaluation of the axial flow fan and the result may be validated form the present experimental results. The optimization of axial air flow fan parameter may be done for the industrial use.

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