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Research Article

An experimental test set-up with the ANSI / AMCA 210-16 standard for performance analysis of axial fans

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ABSTRACT

The fact that fans are used in a large variety and wide areas today requires them to have a certain performance and quality. The test set-up to be used to determine the performance values of the fans should also be prepared and verified in accordance with standards such as ISO, ANSI / AMCA. In this study, a test set-up was designed and established in accordance with the ANSI / AMCA 210-16 standard. In order to ensure the accuracy of this test set-up, the static pressure values for different flow rates of the AXI500-5-25 model axial fan produced in a fan manufacturer company in Konya were measured, and data were obtained. The static pressure-flow rate performance curve obtained from the tests was compared with the manufacturer's catalog data. As the most important result in comparison, it was seen that the experimental and the catalog data were quite compatible. The highest difference between experimental and catalog data was obtained at the points where the flow rate is minimum and maximum. However, this difference was determined to be around 5%. With the results obtained in the study, it was determined that the accuracy of the experimental set-up designed and set up according to ANSI/AMCA 210-16 standard was ensured and it was suitable to be used for testing.

Keywords: ANSI/AMCA; Axial fan; Fan performance; Test set-up

Nomenclature

P_e	saturated vapor pressure	ρ_0	atmospheric air density
P_P	partial vapor pressure	ρ_2	air density at second plane
P_b	ambient barometric pressure	ρ_3	air density at 3rd plane
P_{s2}	static pressure at second plane	V_3	velocity at the pitot traverse plane
P_{s3}	static pressure at 3rd plane	n	number of measurements
P_v	fan velocity pressure	A_2	fan outlet area
P_{v3}	velocity pressure at 3rd plane	A_3	area at the pitot traverse plane
P_t	fan total pressure	Q	fan airflow rate at test conditions
P_{t1}	total pressure at the fan inlet	Q_3	airflow rate at the pitot traverse plane
P_{t2}	total pressure at the fan outlet	f	coefficient of friction
t_{w0}	ambient wet-bulb temperature	Re	Reynolds number
t_{d0}	ambient dry-bulb temperature	μ	Dynamic air viscosity
t_{d2}	dry-bulb temperature at second plane	D	actual diameter
t_{d3}	dry-bulb temperature at 3rd plane	D_{h3}	hydraulic diameter at 3rd plane
t_{s1}	stagnation (total) temperature at the fan inlet	L_e	equivalent length of a straightener
ρ	fan air density		

1. Introduction

Fans which are defined as a device that uses a power-driven rotating propeller to move air or gases; building HVAC (heating, ventilation and air conditioning) systems, highway and railway tunnels and parking ventilation, cement, iron and steel, glass industry, power plant, wind tunnel, and industrial ventilation. The use of fans in such a variety and wide-area required them to be of a certain quality and performance. The test set-up to be used for the performance measurements of the fans must also comply with the standards prepared by organizations such as ISO, ANSI / AMCA.

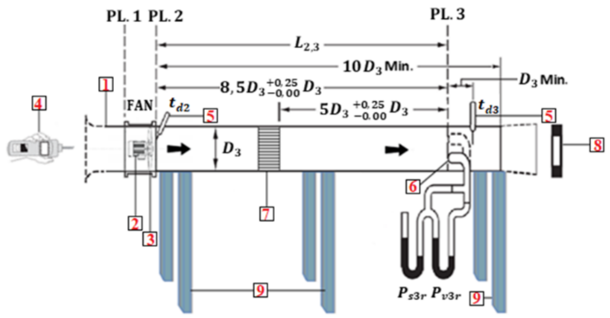
Axial fans are devices that provide a high flow rate under low-pressure conditions and do not change the air pressure much. At the axial fan wheel outlet, fluid flow occurs parallel to the shaft axis [1]. Axial fans are in the second place with a rate of 23%, after centrifugal fans, which are in the first place in the market share of industrial applications with 34% [2]. For axial fans, whose efficiency is generally around 70-80%, their efficiency is increased up to 85% with applications such as inlet cone and guide vane installation [3].

Elliot and Nadel, in the report they prepared for ACEEE (American Council for an Energy-Efficient Economy), determined that motor systems consume 50%-60% of electricity on a national basis. They stated that almost half of this consumption is distributed among the industrial sector and the rest among the residential, commercial, and public sectors. In the same report, the energy-saving potential of industrial fans and pumps in the chemical production, paper production, oil and coal products manufacturing, and mining sectors was 74%. In addition, on a national basis, with the more efficient use of energy in the entire industrial sector, 30% of energy consumption will be saved on electric motors [4]. Waide and Brunner stated in their study that fan motor systems consume approximately 20% of all electricity in the European Union. They found that these systems had become a high priority for the European Commission and EuP (EBM-Papst), and remarked that they published a draft regulation for this in 2010 approved by the Ecodesign Regulatory Committee. In the regulation, fans make up about 18.9% of the total energy consumption of electric motors, and active system improvements for fans in OECD economies are around 40% [5]. In 2018, 45.6% of electricity consumption determined as 258,232 GWh was consumed in the industrial field according to the Turkey Electricity Distribution and Consumption Statistics [6]. Considering that fans have a large share in electricity consumption in the industry based on these data, the importance of energy saving in this area becomes evident. Bleier defined AMCA test codes for fan performance tests and gave information about various laboratory test set-up [7]. Zhu et al. studied the wake flow at various tip gaps of an axial ventilation fan by experimental measurement and numerical simulation. For the fan performance test, they prepared a test installation according to the Chinese National Standard Gb1236-2000, which is equivalent to ISO5801 (1997E) [8]. Yang et al. analyzed and compared the flow mechanism of forward-curved impellers and radial impellers in a low-speed axial fan. In this study, they designed a fan test rig that performs aerodynamic performance and acoustic measurement according to the standards [9]. Cho et al. set up a test rig with a total length of 7650 mm for the fan with a casing diameter of 500 mm to investigate the three-dimensional unstable flow characteristics of CRF (Counter-Rotating Axial Flow Fan). An auxiliary fan was not used in the test set-up. A bell mouth was mounted at the inlet of the fan to reduce the energy loss due to fluid friction and flow separation at the inlet region. A conical damper is used to adjust the flow rate at the outlet of the test duct [10]. Munisamy et al. validated the CFD results of an axial fan design with experimental results. In the study, the experimental set-up and test procedure have been developed with reference to the AMCA 2010 standard [11]. Zhang, experimentally investigated the performance of the contra-rotating axial flow fan with reference to the GB / T 1236-2000 standard [12]. Castegnaro et al. designed an experimental set-up to test fans with a maximum inlet diameter of 0.8 m, according to the ISO 5801 standard. They highlighted the problems encountered in the preliminary study of the 5801 standards and made suggestions to improve the standard [13].

In this study, a test set-up in accordance with ANSI / AMCA 210-16 standard was prepared in a fan manufacturer company in Konya. The test set-up includes a fan unit, test duct connected in the same axial direction with the fan unit, a cell straightener, a throttling device, and various measurement systems. Performance parameters such as flow rate, pressure, and electricity consumption of the tube axial fan whose body material is galvanized sheet and the propeller material is aluminum were measured in the experimental set-up. The results have been verified by comparing the data obtained from the experimental set-up with the values of the fan manufacturer. Since the data obtained from the experimental set-up established according to international standards are verified, the results of future studies will contribute to the literature. Also, the results can be used for validation of numerical studies.

2. Theoretical Background

Fig. 1 shows the experimental set-up prepared in a fan manufacturer firm considering ANSI/AMCA 210-16 standards. Outlet duct set-up Pitot Traverse in Outlet Duct with Cell Straightener diagram (in standard Figure 7A) has been selected from the 16 test diagrams in the standard [14]. Accordingly, the experimental set-up consists of the test fan unit (3), the electric motor (2) connected to the fan, the test duct (1) connected in the same axial direction with the fan unit, a cell straightener (7), the throttling device (8) and various measurement systems.



1-Test Duct
2-Electric Motor
3-Test Fan
4-Tachometer
5- Thermometers
6- Pitot Tube Traverse and Manometer
7- Cell Straightener
8- Throttling Device
9- Test Stand

Fig. 1. Axial fan experimental set-up [14]

The test fan used in the experimental set-up is an axial type fan with a diameter (D) of 500 mm manufactured in a private company. Since the diameter of the test duct and the diameter of the fan case is the same, the length of the test duct is 550 cm, which is approximately 11 * D. In the experiments, the speed of the test fan was measured with a Diwu DT2234-C Laser brand tachometer (4). Thermometers (5) were placed to points shown in the experimental set-up to determine whether there is a density difference in the test environment and test duct (1) parts according to the standard.

The cell straightener (7) whose geometry and dimensions are given in Figure 2 has been placed in the position shown in Figure 1 to ensure uniform velocity distribution of the air entering the test duct.

To determine the flow rate and total pressure of the fan, a pitot tube and a manometer (6) were placed at a distance of 467.5 cm, which is $L_{2,3} = 8.5 * D$ (Fig. 1). The Pitot tube used is the Dwyer 160 series in accordance with the AMCA standard. The Pitot tube is adjusted to measure the points in the positions shown in Figure 3. Thus, the velocity, flow rate, dynamic pressure, and total pressure of the air passing through the test duct are determined. A throttling device (8) is placed at the exit of the test duct to adjust the fan flow rate.

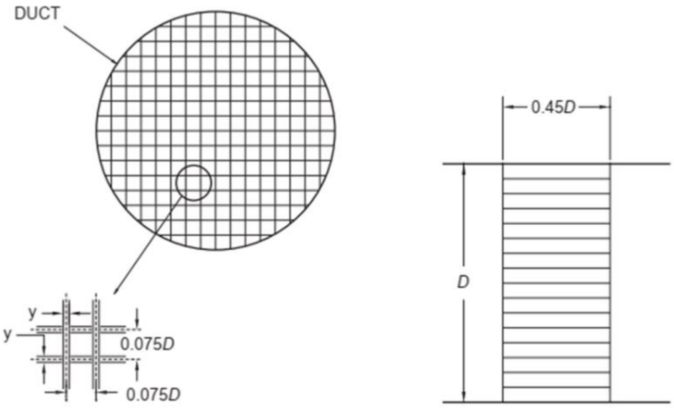


Fig. 2. Cell straightener geometry and dimensions [14]

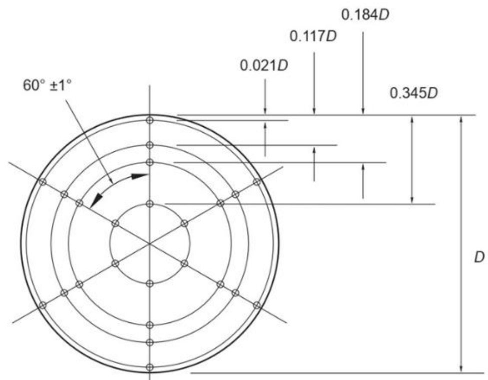


Fig. 3. Pitot tube traverse measurement points inserted into the test duct [14]

3. Data Analysis

In the study, performance parameters were measured for the axial fan coded AXI 500-5-25 produced in a company in Konya. The operating temperature of these fans with efficiency class IE2-IE3 and used with TEFC motor type varies between -20 and +50 C °. According to the fan manufacturer's data, the fan's voltage value is 380 V, the frequency is 50 Hz, the fan power is 0.55 kW, the speed is 1453 rpm and the flow rate is 6000 m3/h. The test fan is shown in Figure 4.

The experiments were repeated for different flow rates between 0 and 6000 m^3/h. To compare the performance curve of the test fan, a data table was obtained by recording the pressure values, temperatures, current, and volt values for the electric motor power and the values taken by the tachometer. With the help of the data obtained, the parameter values of the test fan are calculated from the following equations given in the AMCA 210-16 standard [14].

Fig. 4. The axial fan used for test

Firstly, the saturated vapor pressure (P_e) and the partial vapor pressure (P_P) are calculated by the following equation:

$$P_e = 3.25 \times t_{w0}^2 + 18.6\, t_{w0} + 692 \quad (1)$$

$$P_P = P_e - P_b \left(\frac{t_{d0} - t_{w0}}{1500}\right) \quad (2)$$

t_{w0}, t_{d0} and P_b in this equation denote ambient wet-bulb temperature, ambient dry-bulb temperature, and ambient barometric pressure, respectively. Air density is calculated by using the partial vapor pressure value obtained with equation (2).

$$\rho_0 = \frac{P_b - 0.378 P_P}{R(t_{d0} + 273.15)} \quad (3)$$

Here, there is no need to measure t_{d2}, t_{d3} if $|P_{s3} - P_{s2}| \leq 1$ kPa. In this case, $t_{d2} = t_{d3} = t_{d0}$, that is, the ambient temperature value is used in the calculation. Again, if $|P_{s3} - P_{s2}| \leq$ 1kPa, then $\rho_3 = \rho_2$ and it is calculated from equation (3).

$$\rho_{2,3} = \rho_0 \left[\frac{t_{d0} + 273.15}{t_{d2,3} + 273.15}\right] \left[\frac{P_{s2,3} + P_b}{P_b}\right] \quad (4)$$

Since $P_{t1} = 0$ in the output duct, it is $t_{s1} = t_{d0}$. Therefore, the fan air density is $\rho = \rho_0$. Accordingly, the dynamic pressure (P_{v3}) and velocity (V_3) in the 3rd plane are calculated from equations (5) and (6).

$$P_{v3} = \left(\frac{\sum \sqrt{P_{v3r}}}{n}\right)^2 \quad (5)$$

$$V_3 = \sqrt{2} \left(\sqrt{\frac{P_{v3}}{\rho_3}}\right) \quad (6)$$

Here, n: is the number of measurements. If there is a cross-section difference in the test duct; fan dynamic pressure is calculated from equation (7).

$$P_v = P_{v3} \left(\frac{A_3}{A_2}\right)^2 \left(\frac{\rho_3}{\rho_2}\right) \quad (7)$$

In this study, $P_v = P_{v3}$ since $A_3 = A_2$ and $\rho_3 = \rho_2$. The fan flow rate (Q_3) is obtained from equation (8) according to the cross-section in the 3rd plane, and since there is no cross-section change, $Q_3 = Q$.

$$Q_3 = V_3 \cdot A_3 \quad (8)$$

Fan operating static pressure is found from equation (9) using the pitot tube data (P_{s3}) in the 3rd plane.

$$P_{s3} = \frac{\sum P_{s3r}}{n} \quad (9)$$

Total pressure is calculated from equation (10).

$$P_t = P_{t2} - P_{t1} \quad (10)$$

Here, P_{t1} and P_{t2} are the total pressure on the 1st and 2nd plane, respectively, and $P_{t1} = 0$ since the experimental set-up inlet is open to the atmosphere. Accordingly, P_{t2}=P_t and it is calculated from equation (11).

$$P_{t2} = P_{s3} + P_{v3} + f\left(\frac{L_{2,3}}{D_{h3}} + \frac{L_e}{D_{h3}}\right) p_{v3} \quad (11)$$

In equation (11), f is the coefficient of friction and is determined from equation (12).

$$f = \frac{0.14}{Re^{0.17}} \quad (12)$$

Re is the Reynolds number and is found from equation (13).

$$Re = \frac{D_h V \rho}{\mu} \quad (13)$$

Here, the dynamic viscosity is $\mu = (17.23 + 0.048 t_d) \times 10^{-6}$. It is also taken as $\mu = 1.819 \times 10^{-5}$ Pa.s between 4°C and 40°C [14].
Expressions in parentheses for Equation (11);

$$\frac{L_{2,3}}{D_{h3}} = \frac{4.1}{0.5} = 8.2 \text{ and } \frac{L_e}{D_{h3}} = \frac{15.04}{\left[1 - 26.65\left(\frac{y}{D}\right) + 1846\left(\frac{y}{D}\right)^2\right]^{1.83}} \quad (14)$$

$$\frac{L_e}{D_{h3}} = 15.188$$

4. Results and Discussion

In this study, an air duct experimental set-up was designed and installed in accordance with ANSI/AMCA 210-16 standard. A comparison is made with the static pressure-flow performance curve in the catalog for the AXI500-5-25 model of a tube axial fan to determine the accuracy of the experimental set-up. In the context, the data were obtained by measuring the static pressure values for different flow rates.
In Figure 5, the pressure change curve is given according to different flow rates. When the given values are examined, it has been determined that the experimental data are quite compatible with the catalog data. In addition, it has been observed that as the flow rate increases up to 2800 m^3/h, the static pressure values decrease. In cases where the flow rate changes between 2800 m^3/h and 4000 m^3/h, the roll-off rate decreases. For higher flow rates, the roll-off rate appears to be similar to the first case, as it is valued in the catalog. Also, the highest difference between catalog and experimental data was obtained at the points where the flow rate is minimum and maximum. However, this difference was determined to be around 5%.
The accuracy of the experimental set-up designed according to the ANSI/AMCA 210-16 standard within the framework of the results obtained has been proven with the catalog data and it is suitable for testing purposes.

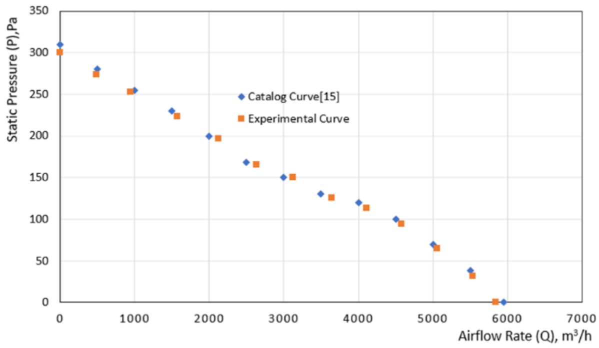


Fig. 5. Variation of static pressure with airflow rate of axial fan

5. Conclusion

In this study, the performance curve of an axial fan manufactured in Konya in the experimental test set-up prepared according to ANSI / AMCA 210-16 standard was compared with the experimental results. Thus, the accuracy of the experimental unit was investigated. The results obtained can be summarized as follows:

- It has been observed that the experimental and the catalog results are quite compatible. For this reason, it has been determined that the accuracy of the experimental unit is ensured and that the fan performance tests can be performed in accordance with the standard.
- The electrical power required to draw the efficiency curves of the axial fans manufactured in the company can made by adding the necessary measuring instruments to the experimental unit. Thus, the optimum operating range of the fans can be determined.

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