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Measured and non-free vortex design results of axial flow fans †

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Abstract

The research is about the determination of developing flow pattern in the axial flow fan impeller. A measuring method and a sizing method of axial flow fans used in a research project have been studied. In this way it has been determined the spatial distribution of static pressure directly behind the impeller as well as the spatial distribution and direction of total pressure in a fixed system to the blade channel. There is an opportunity to identify the high loss region in the impeller: to determine the pressure loss and the hydraulic efficiency at each point. The article shows the results of comparison between the measurement data and the sizing results in the case of one in the agricultural-field utilized cooling fan. The impeller was designed by the help of the fore slanted blades in circumference direction as well as the non-free vortex theory. The basic theories of the design method of the blades and the realization of measurement results are analyzed, too.

Keywords: Axial flow fan; Pressure field

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1. Introduction

A Hungarian research result of the widespread axial fans at the agriculture engineering has been presented in this article. These types of fan have been analyzed in this research. There are several ways to improve the accuracy of the design method. One method for the sizing of this fan was developed by Bencze and Szlivka (e.g. [2]), which helped the manufacturer to make several successful product-families. The axial flow impeller, e.g., standardized by Szlivka (e.g. [6]) was developed by the help of fore slanted blades in circumferential direction as well as the theory of non-free vortex. Indeed, there are inaccuracies in the sizing and there are some factors, too, which can be set only with multi-year practices. One of these factors is the hydraulic efficiency, which shows the hydraulic energy loss percentage. This work presents some measurement results as well as calculation results in the case of an agricultural cooling fan.

2. The design theory

In this section is shown the basic design theory worked out by (e.g. Bencze and Szlivka, [2]) used constant hydraulic efficiency and a product of it. It was measured the global characteristics and the microstructure. The assumption of the theory and the results of the measurement are compared.

There are several approaching solutions in the literature for determination of characteristics of the axial flow fans. One of them, which is the basic of our design method is the (e.g. Somlyódy, [5]) method, where a differential equation was solved. The vorticity is not constant along the radius, it is so called non-free vortex design.

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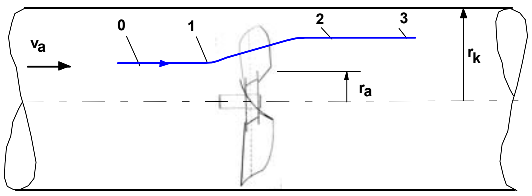


Fig. 1. Flow areas assembled from cylindrical and conical areas.

It was assumed that the flow-areas were the combination of rolls with "r" radius and cone areas with narrow open angle, respectively (see Fig. 1.).

The next step is to solve a differential equation (it is coming from the Bernoulli's equation with a loss term) that was written at a given radius and contains both the flow coefficient (φ) and total pressure rise coefficient (ψ).

$$2\varphi_3 \cdot \frac{\partial \varphi_3}{\partial R_3} - 2\varphi_0 \cdot \frac{\partial \varphi_0}{\partial R_0} \cdot \frac{\partial R_0}{\partial R_3} = \frac{\partial \psi_{i\infty}}{\partial R_3} \cdot \left(1 - \frac{\psi_{i\infty}}{2R_3^2}\right) - \frac{\partial \psi'_{0-3}}{\partial R_3}$$

If there is constant incoming velocity so the inlet flow is uniform, (φ_0) so

$$2\varphi_0 \cdot \frac{\partial \varphi_0}{\partial R_0} \cdot \frac{\partial R_0}{\partial R_3} = 0$$

The differential equation form is the next:

$$2\varphi_3 \cdot \frac{\partial \varphi_3}{\partial R_3} = \frac{\partial \psi_{i\infty}}{\partial R_3} \cdot \left(1 - \frac{\psi_{i\infty}}{2R_3^2}\right) - \eta_h \cdot \frac{\partial \psi_{i\infty}}{\partial R_3} - \frac{\partial \eta_h}{\partial R_3} \cdot \partial \psi_{i\infty}$$

If the hydraulic efficiency is constant, the last term disappears from the equation. (e.g. Bencze and Szlivka, [2]) used that form for sizing axial flow fans.

3. Experimental setup

By the help of the presented method with the applied pressure and velocity measurement technology there is the possibility to make relatively cheap and quick diagnostics on the developed as well as on the available axial flow fan. Beyond the general characteristics of volume flow rate versus pressure and efficiency it can be got information about the blades. By this method it can be achieved such measurements over the whole work range of axial flow fan, by which the expected blade runner attitude could be determined.

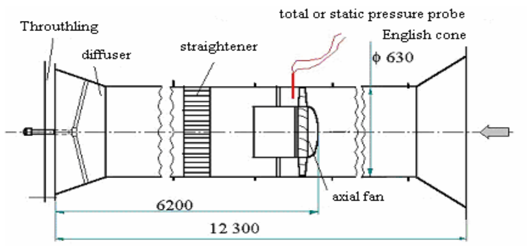


Fig. 2. Build-up of the equipment.

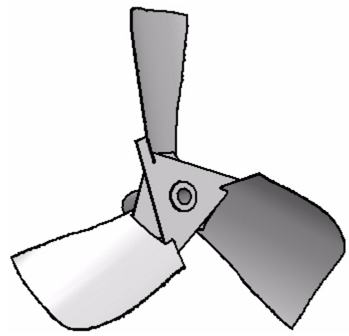


Fig. 3. The measured impeller.

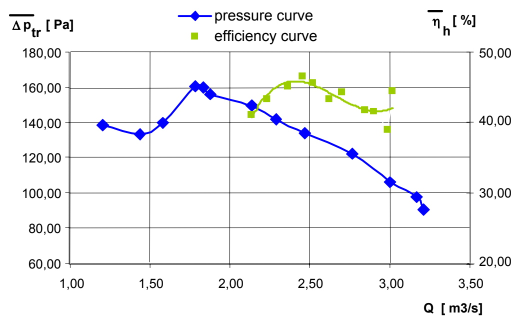


Fig. 4. Pressure and efficiency curves.

3.1 Measure of the characteristic curve

The impeller was built in a channel at the end of the circle channel there is a throttle plate to change the properties of the work point, so that the fan characteristics can be measured. A schematic figure of the experimental setup can be seen in Fig. 2, and the measured impeller is shown on the Fig. 3.

Fig. 4 shows the results of the characteristic curve of the fan. This characteristic curve has high importance during the measurements because it helps to select the target points for blade-to-blade analysis (see 3.2). The measurements were carried out at the design global flow coefficient ($\overline{\varphi} = 0,3$).

The measurement results of the efficiency can be seen in Fig.4, where the maximum of the total efficiency of the impeller is 47 %.

3.2 Blade-to-blade total pressure and velocity measurement

Beyond the general characteristics of volume flow rate and pressure efficiency information can be collected about the blade runners. By this method we can achieve such measurements over the whole work range of axial flow fan, by which the expected blade runner attitude could be determined. Using these measuring method, the fine structure distribution of pressure in addition to the velocity field growing around the environment of the impeller can be measured. However it has only used this method so far for the measuring of two velocity components. According to the experiences of LDA measures (e.g. Corsini, [3]) this gives a good approach because the radial velocity is small in comparison with the other two components. To measure the pressure and velocities only on the outer side of the impeller seemed to be adequate where pressure sensors were enough for measurements. The static pressure was measured by the Sher-disk and the total pressure was indicated by the Pitot-tube or the cylindrical probe. (The Pitot-tube has given better results when the blade number was over 10 and the cylindrical probe was used if the blade number was less than 6. The “better results” mean smaller fluctuation of the results.).

Theoretically the static and total pressure probes would have been placed behind the impeller at the same time at two different places but for easier handling and the minor influence of the flow only one probe has been used in the same time. So, the static and total pressure measures were made in two different times. Because of the difference between the timing of measures, the fan parameters (the same rpm and throttling position) had to be calibrated very accurately for the measures following each other and the spatial positions of the measuring positions had to be saved very accurately too, in order to use coherent total-, and static pressures can be used properly in the determination of other parameters. The basic assumption was that the radial velocity component is much less than the other two velocities

The velocity field behind the blade runner changes periodically according to the passing of blades. The frequency of change can be calculated as the number of blades times with the revolution per second, Fig.6. The periodical pressure problem is discussed by (e.g. Ainsworth, [1]).

There is no problem to measure the static pressure if the velocity in radial direction is small and Ser-disc is set nearly parallel with the flow direction. The bigger problem is at the total pressure measuring. In a certain point (at certain radius and angle compared to the blading) it can be measure the total pressure in that case if the hole of total pressure probe (in case of Pitot-tube and cylinder probe too), is exactly in front of the instantaneous velocity. Downstream the fan blading the direction and magnitude of velocity change during the rotating of the impeller the input direction and magnitude of flow at the total pressure probe change continuously too. The total pressure probe measures different pressures depending on the position of the rotating impeller. On the Fig. 6 it can bee seen a pressure form depending on the tangential coordinate. There is mounted an angle sensor, the whole 360 degree is divided into 1024 parts. So the

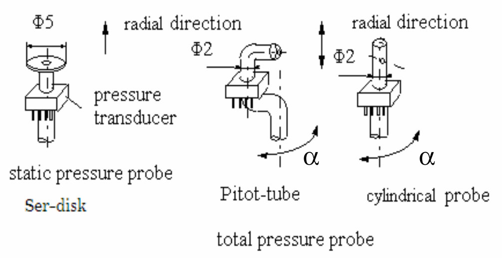


Fig. 5. Pressure sensors.

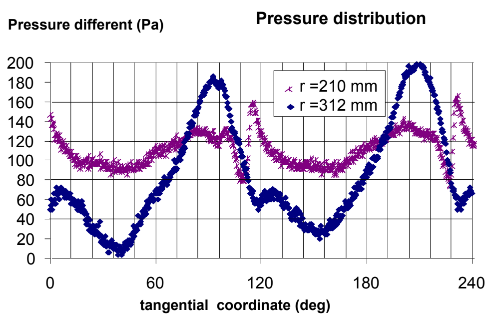


Fig. 6. Measured pressure distribution.

data logger can detect the pressure during one revolution 1024 times. On the Fig. 6 it can be seen 240 degree almost $2 \cdot 1024/3 \approx 680$ pressure date in one curve. (The impeller had three blades, it can be seen the 120 degree period.) The curves showed by the Fig. 6 were detected on difference radius r = 312; 210 mm. Let choose one point from the 1024 points. In the first revolution can measure a pressure different with the totals pressure probe. In the next revolution the measured pressure will be different because of the turbulence. In the third revolution will be a third measured magnitude of the pressure different. It was calculated the average pressure in a given position. Every point of the pressure curves shown by the Fig. 6 is an average of approximately 150 revolution data.

3.3 Determination of total pressure magnitude and direction

At a given radius the total pressure probe can be rotated around it's own axis. The turning angle around the own axis of total pressure probe is marked with α (see Fig. 5). The angle α is defined as the angle of the total pressure probe measured form the direction of axis of fan. The $\alpha = 0^{\circ\,0}$ if the direction of total pressure probe is parallel to the axis of pipe and points in the face of the flow.). If we turn the total pressure probe around along it's axis in another position then a different curve will have plotted along the circumference. For example the Fig. 6 was measured when $\alpha = 0^{\circ}$. The pressure form is determined by the total pressure probe position, α. It was measured at $\alpha_0 = 0^0, \alpha_1 = 10^0, \alpha_2 = 20^0$ .... $\alpha_{80} = 80^0, \alpha_{90} = 90^0$, ten different positions the same curves like in Fig.6. (But the speed and radius were the same) From the ten different curves for every tangential coordinate it can be selected the biggest pressure value $\alpha_{\max}$ too, at a certain geometric point. At the selection of the maximum a regression has been used in order to reduce the imprecision and the maximum and it's place have been chosen from the curve of regression. (So it can be got 1024 maximum pressure-it is the total pressure- and 1024 $\alpha_{\max}$ they are the directions of the velocity.)

The gained direction gave the direction of absolute flow. The obtained flow directions correspond very well with the directions getting from LDA results (e.g. Corsini [3]). We have done measures with Pitot-tube or Cylinder probe too. There weren't significant difference regarding neither the direction nor the volume. Sometimes the fluctuations were different)

Of course the radial velocity component can't be determined by this method. Same measures have been done with using of Pitot-tube and cylinder probe, too. There weren't significant difference regarding neither the direction nor the volume.

The total pressures and the velocity directions can detected at different radius. It was measured fourteen different radius, the step was 15 mm. The static pressure was also lodged in the same points. From the total and static pressure it can be calculated the maximum velocity magnitude, too.

3.4. About the accuracy of the method

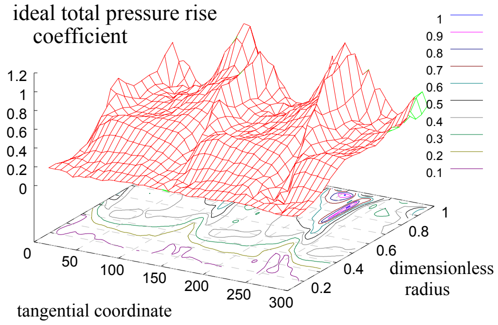
After measuring the blade-to-blade structure it is known the velocity magnitude and direction down stream in each point. It can be determine the axial and tangential components of the velocities. From the axial components it can be calculated the volume flow rate delivered by the fan. But the volume flow rate was measured by the English con (it is a standardized volume flow rate meter), too. It was a very good agreement. The maximum difference was 7,5%. The volume flow rate measured by the English cone was smaller. (See more details in e.g. Molnár and Szlivka [4]).

The blade-to-blade structure can be measured some different working point of the fan. In this paper we have shown any results in the working point, where $\varphi = 0{,}3$ it was the magnitude of the design global flow coefficient and the measured average $\psi = 0{,}152$ (Fig. 4).

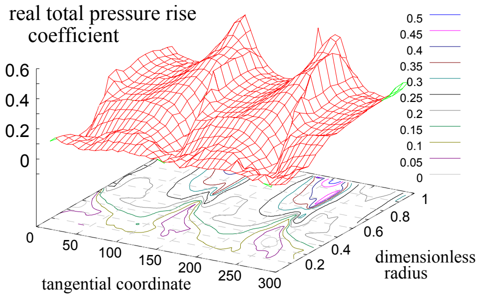
4. Results

4.1 The pressure distribution near the impeller

On the Fig.7 a result of total pressure field can be seen respect to radius and the tangential coordinate. The ideal total pressure rise is also a measured result, but from the Euler's turbine equation. The real total pressure rise coefficient means the difference the measured total pressure behind and in front of the impeller in dimensionless form. During our measures we recorded the measuring results of all of the blade channels and from these we published results belonging to more than one blade channel which corresponds to 300° central angle.



(a)



(b)

Fig. 7. Ideal and real total pressure rise coefficient distribution.

4.2 The pressure distribution along the radius

A very important assumption of this design method is the variation of the pressure along the radius. In this research the dimensionless pressure distribution was assumed to be a power function e.g., Fig. 9 The maximum is at the 85% of the casing radius. In case of the non-free vortex method this value varies along the radius in every target points. In case of the constant circulation method the pressure is constant along the radius. But it may vary if the properties of the work point changes. The real values of the measured pressure number distribution can be calculated from the difference of the total pressure values in front of and behind the impeller. The measured and designed real total pressure distributions have a good agreement if the radius is not on the hub or not on the tip of the blades e.g. Fig. 9.
(e.g. Bencze and Szlivka [2]) used constant hydraulic efficiency in the theory. From the blade-to-blade measured results seem to be a function along the radius shown in Fig. 10.

The average value 49.2%. The average efficiency from the global parameter curve, (see Fig.3) is almost the same (47%). The measured efficiency and total

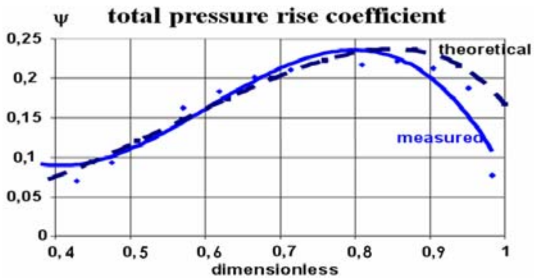


Fig. 8. Total pressure rise coefficient functions.

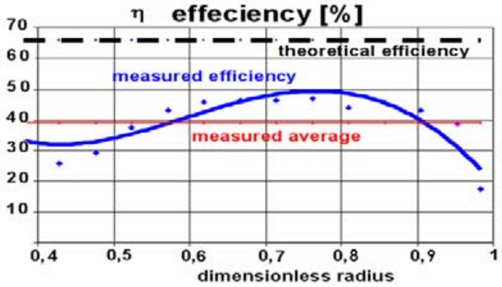


Fig. 9. Efficiency function.

pressure rise curves show discrepancy at the tip of the blades and at the hub. To take the discrepancy into consideration in the design method will be our next step in the project.

5. Conclusion

Testing the impeller of an axial flow fan (designed by the Bencze-Szlivka method [2]) was carried out in this work. Blades were made by the help of the fore slanted technology and the non-free vortex design method. Distribution of pressure number varied according to the initial design functions even in work points that differ from the initial one. The hydraulic efficiency is not constant along the radius and the parameters vary along the circumference, too. Variation of hydraulic efficiency will be taken into consideration at the design method of fans.

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Nomenclature

η_h : [–] Local efficiency (ratio of real and ideal local total pressure rise)
$\bar{\varphi}$: [–] Design global flow coefficient

Q	:	[m^3/s] Volume flow rate
$\overline{\psi}$	:	[–] Design global ideal total pressure rise coefficient
$\psi_{tid} = \frac{2 \cdot R \cdot \Delta v_u}{u_k}$	:	[–] Local pitchwise-averaged ideal pressure coefficient
$\psi'_{03} = \frac{\Delta p'_{03}}{\frac{\rho}{2} \cdot u_k^2}$	:	[–] Local pressure loss coefficient
$\varphi = \frac{v_a}{u_k}$	:	[–] Local pitchwise-averaged axial flow coefficient
u	:	[m/s] Circumferential velocity (r ω)
u_k	:	[m/s] Reference velocity (r_k ω)
R	:	r/r_k [–] Dimensionless radius
v	:	[m/s] Absolute velocity
ω	:	[1/s] Angular velocity
Δv_u	:	[m/s] Circumferential component of the absolute velocity
Δp_t	:	[Pa] The real total pressure rise

Subscripts and Superscripts

a	:	*Axial*
k	:	Casing (radius, velocity etc.)
0	:	Rotor inlet plane
3	:	Rotor outlet plane
„id" or "i"	:	Ideal
r	:	Real
t	:	Total
$\overline{\varphi}$	:	(over lined) Average
∞	:	Infinite blade number

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