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REGULATION OF FLOW RATE GENERATED BY CROSS FLOW FAN WITH USE OF INTERNAL VANE. EXPERIMENTAL AND NUMERICAL RESULTS

REGULACJA STRUMIENIA OBJĘTOŚCI WYTWARZANEGO PRZEZ WENTYLATOR POPRZECZNY. WYNIKI EKSPERYMENTALNE I NUMERYCZNE

A b s t r a c t

In this paper the problem of volumetric flow rate regulation is considered. The change of angular position of an element mounted axially inside of the cross flow fan impeller creates different aerodynamic performances: $\psi_t = f(\varphi)$, $\psi_s = f(\varphi)$ at the same rotational speed and in consequence the qualitative diversification of fan operation is obtained. It gives possibility to use a cross flow fan for several applications depending on an actual requirement respecting to volumetric flow rate or pressure coefficient. The shape of inner element as well as its angular position have an important influence on the flow structures so a method of volumetric flow rate control without necessity of rotational speed change is proposed. This way of cross flow fan flow rate control, commonly made with use of frequency converter, seems to be more effective for reduction of an energy consumption and gives an evident economic advantage.

Keywords: *cross flow fan*, *volumetric flow rate control*, *internal vane*

S t r e s z c z e n i e

W artykule rozpatrywany jest problem regulacji strumienia objętości. Zmiana kątowego położenia elementu zamontowanego osiowo w wirniku wentylatora poprzecznego powoduje uzyskanie różnych charakterystyk aerodynamicznych: $\psi_t = f(\varphi)$, $\psi_s = f(\varphi)$ przy tej samej częstości obrotów i w konsekwencji jakościową dywersyfikację działania wentylatora. Stwarza to możliwość szerszego zastosowania wentylatora poprzecznego w zależności od aktualnych wymagań dotyczących wielkości strumienia objętości i ciśnienia. Kształt wewnętrznego elementu oraz kątowego położenia maja istotny wpływ na strukturę przepływu stąd zaproponowano metodę kontroli wielkości strumienia objętości bez konieczności zmiany częstości obrotów. Ten sposób regulacji strumienia objętości, zwykle wykonywanego przy użyciu przetwornicy częstotliwości, wydaje się bardziej efektywnym w redukcji zużycia energii i ekonomicznie korzystnym.

Słowa kluczowe: *wentylator poprzeczny*, *regulacja strumienia objętości*, *łopatka wewnętrzna*

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Nomenclature

D – impeller diameter [m]
L – impeller length [m]
n – rotational speed [s^{-1}]
Δp – pressure [Pa]
u – circumferential velocity [m s^{-1}]
$\dot{V}$ – volumetric flow rate [m^3 s^{-1}]
α – angular position of vane
ρ – fluid density [kg m^{-3}]
φ – $\dfrac{\dot{V}}{D_2 \cdot L \cdot u_2}$ – flow rate coefficient [–]
ψ – $\dfrac{2 \cdot \Delta p}{\rho \cdot u_2^2}$ – pressure coefficient [–]

indexes:
s – static
t – total
2 – outer

1. Introduction

The fan dimensional or dimensionless performances may undergo a change as a result of control process. The aim of control is to fulfill the requirements of varying in time resistance of network or productivity at possible high efficiency. It can be obtained using the different ways of timing gear control in the fan or in its drive. Fan control is usually realized by change of an impeller rotational speed as well as varying of setting in motion impeller blades or mounting the guide vanes (axial and radial fans), and not only one curve but the curve families are obtained covering the wider range of fan operation.

Fans operate at varying conditions established by received networks described by resistance-drag- curves or at constant volumetric flow rate $(\dot{V} = \text{const})$ as well as at constant total pressure (Δp_t = const) so it is necessary to provide an acceptable efficiency in the whole range of fan control.

The range of volumetric flow rate control (RVFRC) is defined as a ratio of difference between maximal and minimal values of volumetric flow rate to maximal value of volumetric flow rate at the known (prescribed) resistance curve.

The most popular way for changing volumetric flow rate is use of frequency converters. Increase of the rotational speed of impeller causes enlargement of volumetric flow rate, but in some cases so high values are not required like for example in the air conditioning systems or drying rooms. There are more useful rather uniform velocity distributions in the discharge region of fan than high values of volumetric flow rate as it can be seen in hospital operating room where some parts specially near an operating table requires the laminar flow of very clean air [1].

2. Cross flow fan with internal vane

The cross flow fan sometimes called tangential fan or peripheral fan belongs to a unique type of turbo-machine with completely different operation from axial or centrifugal fans.

It consists of a long impeller with forward curved blades and casing which let the air cross twice a blading creating two steps of compression. Geometric design of fan casing (identified as a rear wall, vortex wall and ends-walls) has an important influence on the operating conditions and sometimes is difficult to parameterize. Usually the area of internal flow inside of cross flow fan has been divided into three regions: inlet, interior of impeller and outlet. The most interesting but complicated to mathematical as well as to physical description is impeller interior divided into two regions: eccentric vortex region, called the recirculation region, characterized by closed streamlines, and throughflow region, a main asymmetrical flow.

The complexity of flow conditions: the unsteady and highly turbulent nature of twice accelerated fluid, cyclic and variable working conditions of blades as well as the continuous crossing of the main eccentric vortex by the blading are the reasons of a lack of generally accepted method for cross flow fan design.

The change of operating parameters of a cross flow fan could be realized by mounting immobile or mobile elements, having different shape and localization, in inlet or outlet region of fan as well as inside of the impeller. The immobile one- or multi- blades guide vanes in shape assuring the uniform stream velocity distribution at outflow from the first stage and at inflow on the second stage of impeller were presented in papers [2, 3].

The results of experimental and theoretical methods to perform quantitative and qualitative estimation of the flow structures and flow phenomena for different shapes of inner rotated vane were published in [4]. Investigations of several models having different impeller diameters or lengths and different shapes of casing have created possibility to observe some relationships between geometric parameters and its influence on flow structure as well as on cross flow fan performances.

3. Influence of internal vane angular position on volumetric flow rate – qualitative results obtained from water visualization

Flow visualization, carried out in water tank with the tracers in form of 2 mm diameter polystyrene balls allowed for a quantitative estimation of some flow phenomena effected on flow structure inside the cross flow fan. Using different shapes of an internal vane and changing their angular positions one could observe an important influence on flow structure. In the case of so called "moonlike" vane mounted inside of cross flow fan constructed with symmetric shape of inflow and outflow parts of spiral casing the fluid stream has changed its flow direction about 180º at the constant impeller rotational speed. So the reverse flow resulted from casing symmetry and an adequate angular position of internal vane has occurred. This phenomena could be applied in ventilating systems where for very quick the change of air in room (flow of air stream inside or outside) has been realized by change of angular position of vane located inside of a cross flow fan operating with the same rotational speed (no necessity to use converter, more electric power or using another type of fan). The angular position of "drop-like" vane orders the inflow on blading in suction region, creates two streams on both sides of vane forming the throughflow, which takes full advantage of length of discharge arc flowing into the outlet region of cross flow fan. In this region the velocity of flow has the highest values in comparison to other angular positions (detected by the length of flow lines of tracers) what is the synonymous with the greatest flow rate. Analyzing the flow structures for the different angular positions of vane it has been observed that the flow rate can be easy regulated using an internal element mounted axially inside of impeller [4].

4. Quantitative results of flat plate vane influence on volumetric flow rate

The best qualitative and quantitative results have been obtained for rotated internal vane in the shape of flat plate. The influence of an angular position of vane on the flow rate has been analyzed on the basis of dimensionless performances: total ψ_t and static ψ_s pressure coefficients in function of flow coefficient φ.

Cross flow fan with casing profiled in its part of outlet in form adapted to co-operation with a heat exchanger had the basic dimensions: outer diameter of impeller D_2 = 60 mm and length L = 180 mm. Analyzing curves $\psi_t = f(\varphi)$ and $\psi_s = f(\varphi)$ presented in Fig, 1a, b, three different ranges of cross flow fan operation determined by the ranges of vane angular position change are observed.

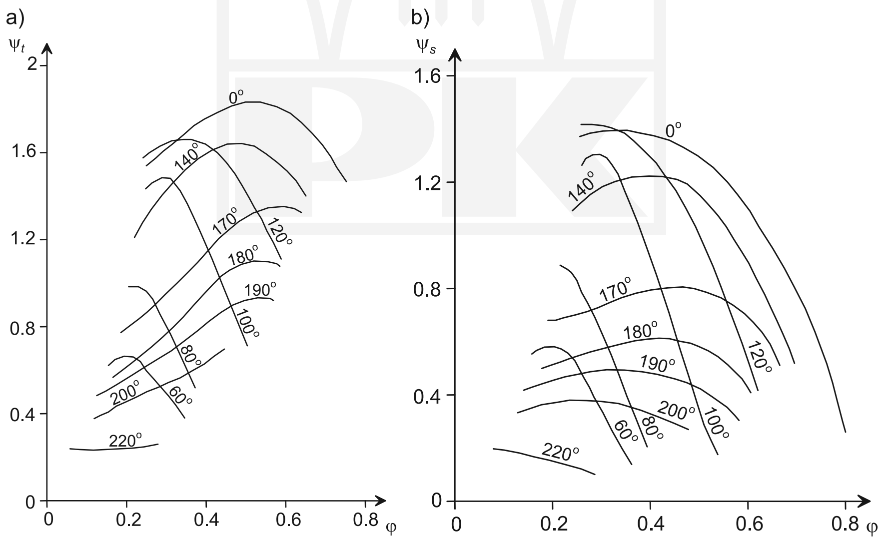

Fig. 1. Total (a) and static (b) pressure coefficients versus flow coefficient for different angular position of internal vane, $n = 37.5\ s^{-1}$

Rys. 1. Wskaźniki spiętrzenia całkowitego(a) i statycznego (b) w funkcji wskaźnika przepływu dla różnych położeń kątowych elementu wewnętrznego, $n = 37.5\ s^{-1}$

In the first range α = 60º–120º the relatively great increase of total and static pressure coefficients is observed in rather small changes of flow coefficient values – steep curves. The second range of α = 140º–220º characterized by more flat runs of curves shows that flow rate varies in the widest range of values. In the third range α = 320º–360º (0º) the highest values of total and static pressure coefficient were obtained and only a treble increase in flow coefficient. The influence of angular position α on the values of dimensionless coefficients and in consequence on different aerodynamic performances obtained for one model of cross flow fan at constant rotational speed n =37.5 s^{-1} are presented in Table 1.

Table 1

Variation in dimensionless coefficients values at different ranges of α

Range of α	Multiplicity of ψ_t	Multiplicity of ψ_s	Multiplicity of φ	Notes
60º–120º	~4	14	~3	steep curves
140º–220º	5	12	**10**	flat curves
320º–360º	1.5	9	3	stable performances

The complex comparison between the flow rate coefficients obtained for angular positions of internal vane: α = 210º and α = 350º is very difficult to make because some of total and static pressure coefficients are located in different ranges of values, Fig. 2.

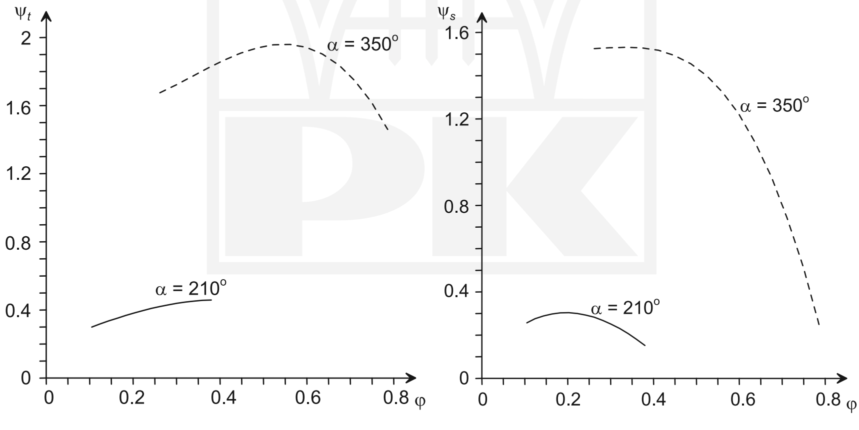

Fig. 2. Comparison between performances at angular position of internal vane: α = 210º and α = 350º

Rys. 2. Porównanie charakterystyk dla położenia kątowego łopatki: α = 210º i α = 350º

In table 2 the results of flow rate coefficients selected for similar values of static pressure coefficient during are presented.

It is worth to notice, that for almost the same values of static pressure coefficient (3.6% variation) the flow rate coefficients have increased about 648%. This case shows possibility of volumetric flow stream regulation.

Table 2

Selected values of ψ_s and φ for different angular position of internal vane

α	ψ_s	φ
210	0.257	0,105
350	0.248	0.786

5. Numerical simulation of flow inside the cross flow fan incorporating a flat vane

To numerical simulation of flow in cross flow fan program Flo^{++} based on finite volume method (FVM) has been used. The results of numerical calculation were verified by experimental data obtained for model of fan having the same geometry and basic impeller dimension: outer diameter D_2 = 100 mm, length L = 450 mm. Construction of numerical model of flow as well as Flo^{++} program used to solve the unsteady, two-dimensional, incompressible flow with small velocities (Ma < 0.3) are in details described in [5].

Cross section of the fan model (Fig. 3) with circle being a sliding edge between unmoving casing and moving impeller (use of the sliding mesh technique for interaction relying upon the interpolation of the calculated values at contact cells in following time step) divided into 19 casing blocks (a) and impeller block with an internal rotated flat vane (b) is shown in Fig. 3 [6].

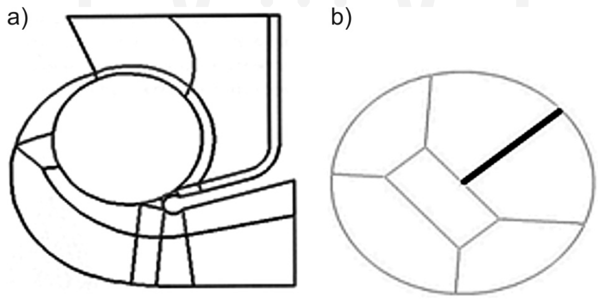

Fig. 3. Cross section of fan model with internal flat vane

Rys. 3. Przekrój poprzeczny modelu wentylatora z wewnętrzną płaską łopatką

The influence of the angular position of internal vane on the flow structure at the same rotational speed could be observed in Fig. 4 where some selected numerical results in form of the vector velocity graphs for different angles $\alpha = 0^o$; 50^o ;120^o at $n = 24.16\ s^{-1}$. For the angular position $\alpha = 0^o$ on the both sides of the vane the dead zones are observed.

The throughflow velocity has rather low values and at inlet region (suction) as well as on a greater part of inlet arc of blading flow with small velocities has been realized (Fig. 4a). Different flow structure is shown in Fig. 4b where for the vane angular position $\alpha = 50^o$ the eccentric vortex with the center near blading close to stabilizer appears and only on one side of vane the dead zone exists. The fluid stream flows with higher velocities which values have significant increased at the angular vane position described by $\alpha = 120°$ (Fig. 4c). For this vane position the vortex center has moved towards rear wall (opposite direction as during throttling) and let the fluid stream occupy a larger part of outlet arc of blading giving possibility to generate a greater volumetric flow.

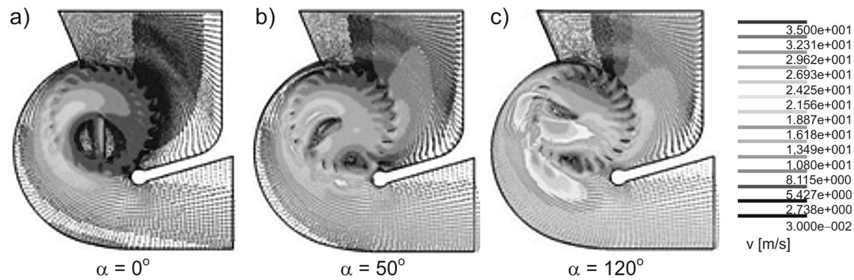

Fig. 4. Vector velocity field for different angular position of internal vane

Rys. 4. Pole wektorowe prędkości dla różnych kątowych położeń łopatki wewnętrznej

Some selected theoretically determined dimensionless aerodynamic performances for the cross flow fan for different α mentioned above at $n = 37.5\ s^{-1}$, are presented in Fig. 5.

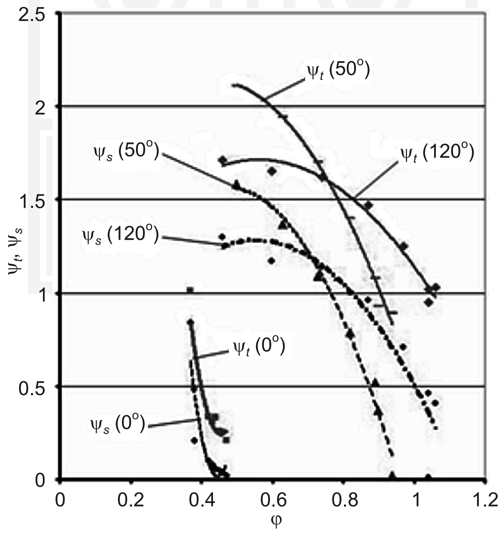

Fig. 5. Total ψ_t and static ψ_s pressure coefficient versus flow coefficient φ [6]

Rys. 5. Wskaźnik spiętrzenia całkowitego ψ_s i statycznego ψ_s w funkcji wskaźnika przepływu φ [6]

The shape of curves $\psi_t = f(\varphi)$, $\psi_s = f(\varphi)$ obtained for different angular position α of internal flat vane are comparable to results of experiments carried out for cross flow fans with different ratios of impeller length to diameter: $L/D_2 = 4.5$ and $L/D_2 = 3$ (see Fig. 1).

Analyzing the curves shown in Fig. 5 obtained numerically for two angular position of rotated vane ($\alpha = 0^{\circ}$ and $\alpha = 120^{\circ}$) one can observe almost threefold increase of flow rate coefficients for the same value of pressure coefficient.

6. Conclusions

Some selected results of experimental and numerical calculations show that the vane mounted axially inside of the cross flow fan impeller can be a controllable element used for changing the volumetric flow rate. Using for analysis the experimental curves: static pressure versus volumetric flow rate obtained at the different rotational speeds in range $n = 12.5–27.3\ s^{-1}$ published in [4] for CFF ($L/D_2 = 4.5$) the sentence mentioned above can find a confirmation. The results indicate that the lowest value of volumetric flow rate obtained at $n = 12.5\ s^{-1}$ has been tenfold less than the highest one obtained at $n = 27.3\ s^{-1}$. For such increase of rotational speed the required power (energy) consumption is equal to $[27.3/12.5]^3 = 10.36$.

It's worth to notice that the same increase in volumetric flow rate (about tenfold) has been obtained by changing the angular position of internal flat vane from $\alpha = 140^{\circ}$ to $\alpha = 220^{\circ}$.

This show that a method of volumetric flow rate control with use of internal vane instead of frequency converter seems to be more simple and more economic proposal.

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