TECHNICAL TRANSACTIONS | CZASOPISMO TECHNICZNE
MECHANICS | MECHANIKA
2-M/2015

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INFLUENCE OF DISCHARGE RESISTANCE ON INTERNAL FLOW STRUCTURE AND PERFORMANCES OF THE CROSS FLOW FAN AND THE UNIT INCORPORATING HEAT EXCHANGER

WPŁYW OPORU W STREFIE TŁOCZENIA NA WEWNĘTRZNĄ STRUKURĘ PRZEPŁYWU I CHARAKTERYSTYKI PRACY WENTYLATORA POPRZECZNEGO ORAZ UKŁADU ZAWIERAJĄCEGO WYMIENNIK CIEPŁA

Abstract

In this paper some selected problems dealing with an influence of different throttling conditions on the flow structure inside a cross flow fan working in a unit incorporating a heat exchanger as well as on aerodynamic performances of a cross flow fan and unit have been studied. Different experimental and numerical methods have been used to verify and confirm selected flow phenomena. The local pressure loss as an effect of a heat exchanger resistance has been estimated taking into consideration some experimental results confirmed by the Finite Volume Method computation.

Keywords: *cross flow fan, heat exchanger, performances, flow structure*

Streszczenie

W artykule przedstawiono wybrane problemy związane z wpływem zróżnicowanych warunków dławienia na strukturę przepływu w wentylatorze poprzecznym pracującym w układzie z wymiennikiem ciepła. Użyto eksperymentalnych i numerycznych metod w celu weryfikacji występujących zjawisk przepływowych. Analizując wybrane wyniki doświadczalne potwierdzone obliczeniami numerycznymi, podjęto próbę oszacowania lokalnej straty ciśnienia jako efektu oporu stawianego przez wymiennik ciepła w strefie tłoczenia wentylatora poprzecznego.

Słowa kluczowe: *wentylator poprzeczny, wymiennik ciepła, charakterystyki pracy, struktura przepływu*

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1. Introduction

A relatively wide range ofa cross flow fan application causes different requirements respecting the interior flow structure. Particularly such features as: generation of lower noise level in comparison to other kinds of fans and uniform velocity distribution at the outlet fan cross section are usually utilized in the ventilating and air conditional systems. The rectangular cross-sectional area of the fan inlet and outlet (Fig. 1) allows for a direct connection to the other device, for example, to a heat exchanger what significantly reduces or sometimes eliminates the loss of local pressure resulting in higher efficiency.

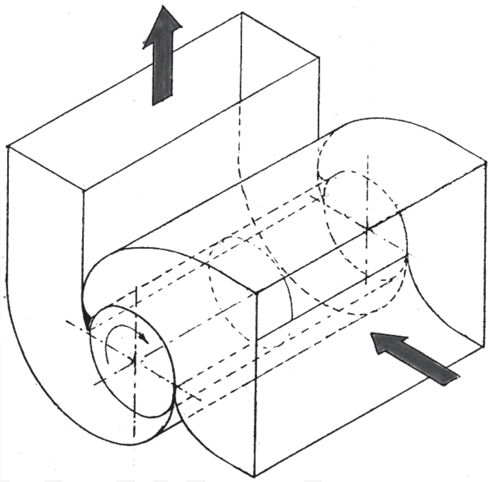


Fig. 1. Cross flow fan

In a cross flow fan the fluid flows perpendicularly to the axis of rotation crossing twice the blading, generating two steps of compression and creating higher values of pressure coefficient ψ in comparison to other types of fans having the same diameter.

This two-stage flow machine characterizes a rather complex flow structure caused by the constructional compromise of the blade passage shape fulfilling two different roles as an inlet and outlet part of blading. The velocity field inside the cross flow fan is usually divided into three different regions: the fan inlet, impeller interior with blading and fan outlet.

The flow structure inside the impeller is a function of several geometric parameters and flow conditions. Correlation between the selected elements of the casing and the impeller's geometry as well as their effect on the cross flow fan's performance has been studied and published earlier but in literature there is lack of experimental or theoretical results describing the relation between the flow structure or flow phenomena inside a cross flow fan and heat exchanger being a discharge resistance and creating one unit.

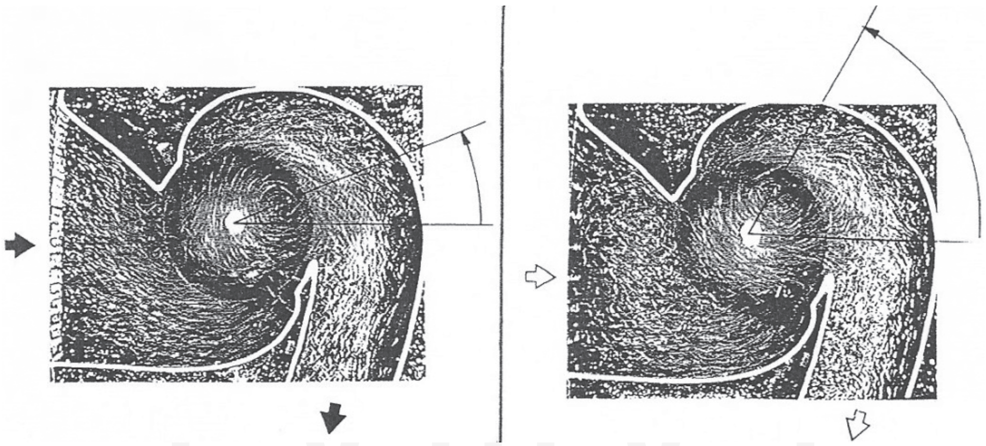
2. Some selected experimental and numerical results of flow structure changes inside the cross flow fan as a result of throttling

Several applications of the cross flow fan have required some knowledge about the flow structure inside this flow machine which is changing as a result of varying flow conditions caused, for example, by throttling realized at the inlet or outlet fan cross section.

2.1. Flow visualization in water tank

Flow visualization is one of the effective methods allowing to identify and better understand some flow phenomena as well as to observe their arising and evolution in time.

This method gives quick and visible information with reference to casing and blading design parameters and it allows to analyze and find the relationship between the internal flow and performance curves. The selected results of water visualization with the of use polystyrene particles as tracers (the same density of polystyrene and water at the measurement temperature give an effect of suspension during the revolution of impeller) are presented in figures below. The visible influence on the flow structure has a change of flow conditions observed in Fig. 2 caused by varying resistance at the inlet part of the cross flow fan.



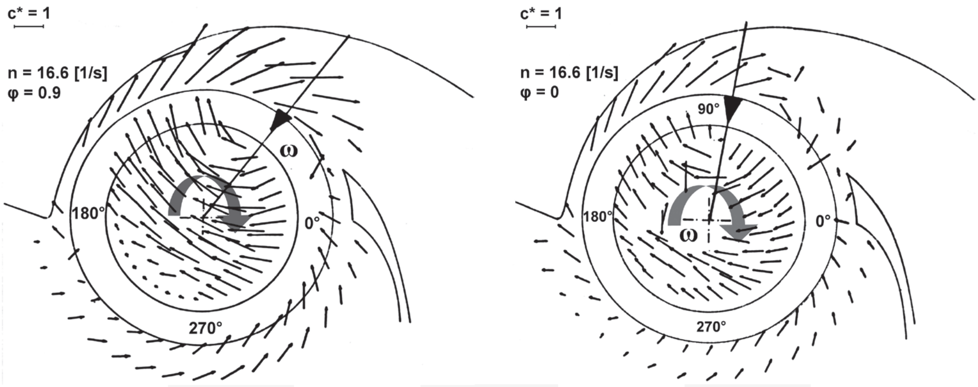
“free” flow “throttled” flow

Fig. 2. Influence of throttling on flow structure and localization of eccentric vortex

As a consequence of throttling realized at the cross section of the fan inlet a movement of eccentric vortex in the direction of the rear wall as well as its magnitude increase are noticeable. In the “throttled flow” the reverse stream observed near the tongue (stabilizer) has crossed the blade ring and flowed into the impeller’s interior. This caused the main increase in the eccentric vortex’s size occupying, in this case, the greater part of blading. Recirculation and dead zones appearing as a result of very low volumetric flow rate as well as the eccentric vortex effected in the main reduction of a cross flow fan’s efficiency [1].

2.2. Measurements of velocity field in the air – effect of throttling

Analyzing the velocity distribution obtained in different throttling conditions of the air flow for the most interesting area – the impeller’s interior (Fig. 3) a similar flow structure is observed. The different regions of the interior flow determined by the vector velocity field: throughflow, eccentric vortex near the vortex stabilizer wall, the zone of the highest values of velocity at the outlet part of the impeller and recirculation zone at the inlet part of the impeller are visible. This recirculation region aroused from the stream crossing the blade passages under no adequate angles can be reduced by using different shape of the casing element at the inlet.



“free” flow “throttled” flow

Fig 3. Absolute velocity distribution for “free” and “throttled” flows, $n = 16.6\ \mathrm{s}^{-1}$

The results of the vector velocity field measurements indicate the same tendency of the eccentric vortex movement towards the rear wall in the situation of throttling of a cross flow fan realized at suction as it was showed earlier in Fig. 2.

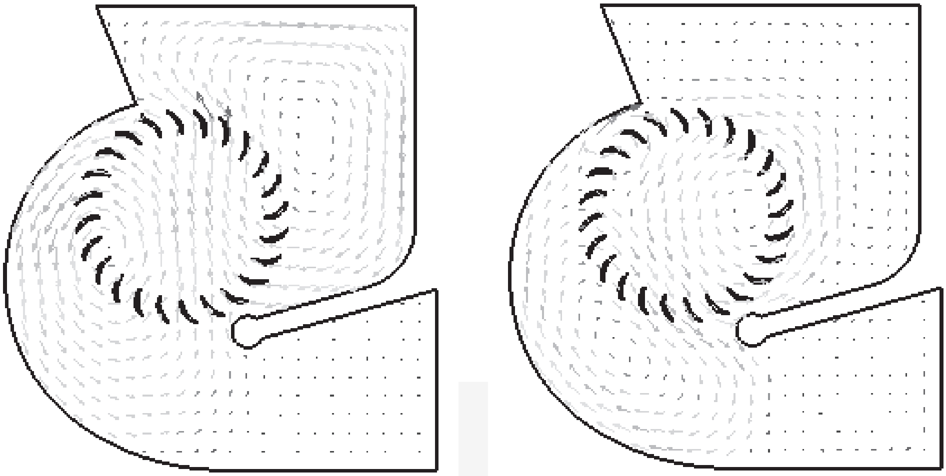
The analogy between flow parameters measured and calculated in two different media (air and water) allows to use a simple formula for local velocity estimations in the region where the direct measurements are impossible or complicated and require very special and expensive equipment, for example inside the rotating blade passages [1].

2.3. Numerical calculations

Qualitative results of flow visualization in water as well as quantitative measurements of the velocity field carried out in the air may be treated as an experimental verification of the numerical flow simulations. Most commercial computational fluid dynamics packages are based on the Finite Volume Method. This method implemented in the STAR-CD software has been used to compute two-dimensional incompressible flow of the viscous fluid with a standard Reynolds k–ε turbulence model in the cross flow fan. The computational area was divided into eighteen casing blocks and one impeller block. The geometry of the tested cross flow fan was drawn in Microstation/J with a accuracy to a millionth part of the main unit. The tetrahedral volume elements (cells) with different density concentration have been used for building the grid. At the outlet area of the fan the blocks with the greatest number of cells were located because of the necessity of high accurate readability of the calculated values of pressure and velocity

Some selected results of numerical flow simulation in a cross flow fan indicating similarity to the results obtained experimentally are presented below. In Fig.4 the vector velocity fields obtained for two different flow conditions determined by variation of throttling realized at the outlet zone of the cross flow fan at a rotational speed of $n = 24.17\ \mathrm{s}^{-1}$ are presented [2].

Throttling has an important influence on the flow structure causing the displacement of recirculation zones as well as the eccentric vortex, which is visible analyzing the vector’s velocity distribution.



“free” flow “throttled” flow

Fig. 4. Velocity vector distribution in two different flow conditions – numerical calculation

In numerical simulations the throttling was realized at cross flow fan discharge, so the eccentric vortex center has been moved in the direction of the vortex’s stabilizer wall. Numerical results confirmed the localization of the eccentric vortex near blading, an increase of its size and its center displacement during different flow conditions.

3. Cross flow fan performances at varied working conditions

The flow phenomena described above have to be taken into account considering the cooperation of the cross flow fan with the heat exchanger, treated as a discharge resistance. This kind of fan is very sensitive and even a small change of geometric parameters as well as the variation of flow conditions’ effect on the flow structure. The results of the study carried out for the air room conditioner incorporating a cross flow fan indicating the factors that influenced the fan’s performance of the unit were presented in [3].

The quantitative estimation of the flow structure’s influence on the cross flow fan working is analyzed based on the aerodynamics and efficiency curves obtained for different flow conditions. The comparison has been made for the performances obtained for a cross flow fan working separately and for the same cross flow fan working as an element of the unit containing a heat exchanger [4].

The relation between the total pressure coefficient and the flow coefficient in the form of curves: for the cross flow fan working separately (it means that the air is discharged straight to the atmosphere so static pressure at the outlet is assumed zero) $\psi_t = f(\varphi)$ and being an element of the unit containing a heat exchanger $\psi_t^* = f(\varphi)$, measured at two rotational speeds: $n = 15\ s^{-1}$ and $n = 20\ s^{-1}$ are shown in Fig. 5. The experimental results have confirmed the two expected phenomena: the higher values of the total pressure coefficient obtained at a higher rotational speed and the lower values of the total pressure coefficient obtained for a cross flow fan working as an element of unit ψ_t^* for both rotational speeds. The difference of the

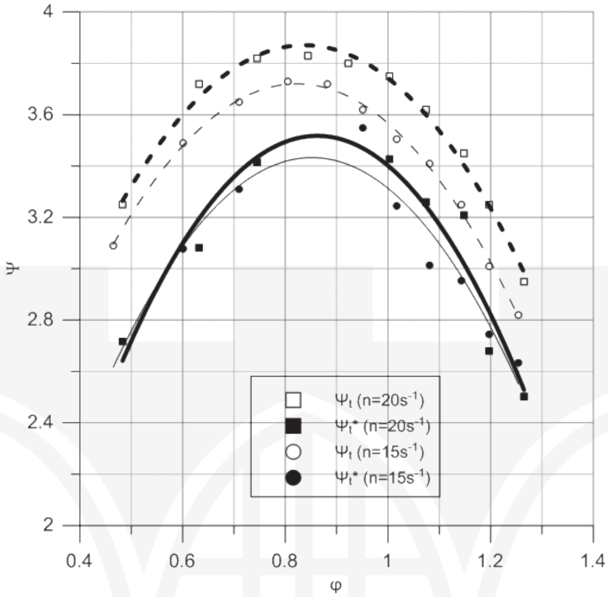


Fig. 5. Changes of total pressure coefficient for cross flow fan working separately ψ_t and aselement of unit ψ_t* at different rotational speed: $n = 20$ s^{-1} and $n = 15$ s^{-1}

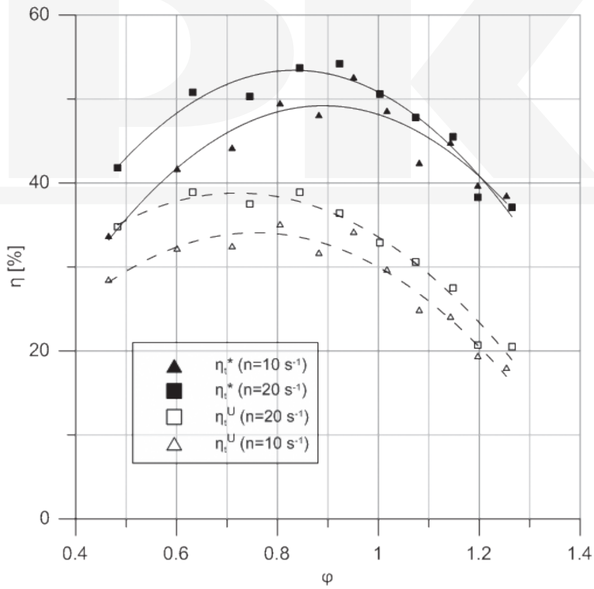


Fig. 6. Total efficiency vs. flow coefficient at rotational speed $n = 20$ s^{-1} and $n = 10$ s^{-1}

total pressure coefficient varying in the range between 8% a 10% may be treated as the consequence of the heat exchanger resistance.

Graphs of total efficiency of the cross flow fan working as an element of unit η^* and efficiency of unit η^u (dotted lines) in the function of a flow coefficient for two rotational speeds: $n = 20\ s^{-1}$ and $n = 10\ s^{-1}$ are shown in Fig. 6. The difference between the efficiency of a cross flow fan in the unit and the total efficiency of the unit about 7% to 18% in the considered range of φ, may be treated as "negative efficiency" of the heat exchanger located at the outlet section of the cross flow fan.

Resistance having an important influence on flow structure changes (see Fig. 4) especially in an increase of the eccentric vortex and exclusion of impeller blading from real work, causes a visible reduction in efficiency.

The reduction in efficiency is the consequence of resistance which has visible changes in the flow structure

4. Effect of heat exchanger as discharge resistance

Quantitative estimation of the local pressure losses caused by the connection between the cross flow fan and heat exchanger has been performed for adifferent medium temperature: 20°C, 35°C, 40°C, 45°C and 50°C. The ratio of the experimental results of dynamic pressure as difference between the total and static pressure at the cross section before (A_1) and behind (A_2) heat exchanger obtained from measurements $p_d^{h(m)}$ and calculated $p_d^{h(c)}$ using the simplified formula:

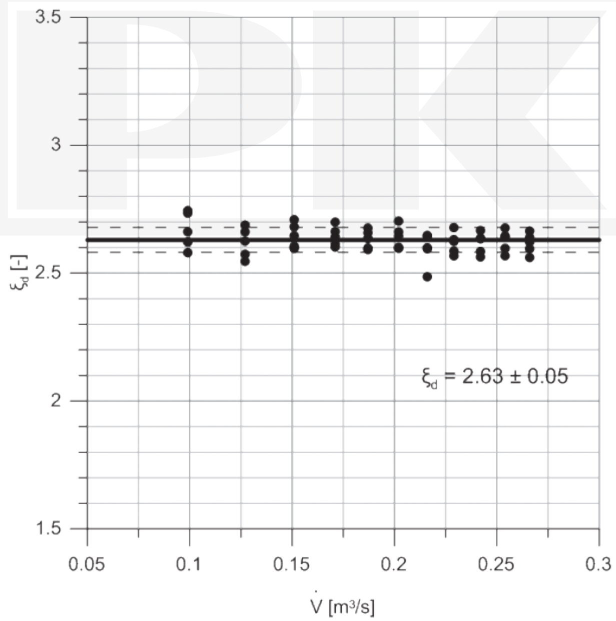


Fig. 7. Dimensionless coefficient ξ_d vs volumetric flow rate for different medium temperature

$$p_d^h = \frac{1}{2}\rho^2 \dot{V}^2 \left[\frac{1}{\rho_3 A_3^2} - \frac{1}{\rho_2 A_2^2} \right] \tag{4}$$

is determined as "dimensionless coefficient of dynamic pressure" ξ_d. The graph $\xi_d = f(\dot{V})$ is presented in Fig. 7.

Independence of ξ_d from the medium temperature in the range of 20°C÷50°C and an almost constant value 2.63 indicate a local dynamic pressure loss being the effect of the heat exchanger's resistance located at the outlet's cross section of the cross flow fan.

5. Conclusions

The investigations carried out in different media using different measurement methods: flow visualization in water and measurements of vector velocity fields in the air as well as the numerical flow simulation showed an important influence of throttling on the flow structure in a cross flow fan. Local pressure loss has been estimated in the form of a dimensionless coefficient of dynamic pressure ξ_d.

Nomenclature

D_2	–	outer diameter of impeller, m
L	–	length of impeller, m
u_2	–	tangential velocity at outer diameter, m/s
n	–	rotational speed, 1/s
p	–	pressure, Pa
$\dot{V}$	–	volumetric flow rate, m^3/s
ρ	–	fluid density, kg/m^3
$\psi = \frac{2\Delta p}{\rho u_2^2}$	–	pressure coefficient
$\varphi = \frac{\dot{V}}{D_2 L u_2}$	–	flow coefficient
$\eta = \frac{\Delta p \cdot \dot{V}}{P}$	–	efficiency

Subscripts: s – static, t – total, d – dynamic

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