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OPERTATING AND ACOUSTIC CHARACTERISTICS OF CENTRIFUGAL FANS WITH INCLINED BLADES

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Abstract. *Considering the large use of fans today for various purposes, it is very important that the fans run in an economical operation regime. The economical operation of the fan implies high fan efficiency, above 90% of the maximum fan efficiency. The geometry of the fan impeller, particularly geometry of the fan blades, and also impeller dimensions, such as impeller width, affects the size of the flow area of the fan, and therefore operating characteristics of the fan. The optimal performance of a fan should also provide the best acoustic characteristics, which can be as important as its performance curves, and it can be a limiting factor for their application. However, the main issue is the impossibility of determining a more accurate estimation of the fan noise level during the fan designing process. With the development of CFD techniques it is now possible to determine fan performance curves and acoustic parameters, and to use numerical results in order to optimize its geometric parameters. In this analysis, the numerical investigation of three similar, but different centrifugal fans, which operate with a similar range of air flow rates, was performed using Ansys CFX 19.0. The obtained numerical results were analyzed in the paper.*

Key words: *Centrifugal fans, Impeller width, Impeller blades, Operating parameters, Acoustic characteristics.*

1. INTRODUCTION

Centrifugal fans, also known as blowers, are largely used in technical practice, such as industrial applications, HVAC systems, pneumatic conveying etc. Proper selection of fans, as well as their efficient operation significantly affect the reduction of electricity consumption. Economic fan operation is when fan obtained efficiency greater than 90% of the maximum efficiency of the fan ($\eta \geq 0{,}9\,\eta_{max}$). In addition to the good efficiency of the fan the optimal operating flow rates should also enable lower noise levels. Acoustic characteristics of fans are particularly important when fan noise affects the human working and living comfort. In such cases acoustic characteristics of fan can reduce their application. Therefore, this paper investigates the dependence of the centrifugal fan

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operating characteristics on some constructive parameters. Three similar low-pressure centrifugal fans are used for two different dimensions of impeller width, and the third fan has also profiled impeller blades. The research was obtained using CFD methods, which showed a satisfactory agreement with operating curves of the tested fans (total and static pressure curves, fan efficiency and power). An analysis of the obtained operating parameters of the fans was performed, as well as an analysis of their acoustic characteristics.

Improving fan efficiency has always been a goal for many researchers, whether experimental or numerical methods are used [1-3].

Changing the inclination angle of cylindrical blades in low-pressure centrifugal fans with backward curved blades gives a large impact on operating characteristics of fan [4].

The influence of the blade outlet angle on the flow field and pressure pulsation in different centrifugal fans were investigated, showing significance regarding the design of the pneumatic performance and noise reduction performance of centrifugal fans, and giving some recommendation regarding the influence of the outlet angle on the fan efficiency [5].

Effect of various attack angle on flow characteristics of centrifugal fan led to conclusions regarding the influence of angle of attack on maximum fan efficiency and other fan operating parameters [6].

When a fan operates with high efficiency (around best efficiency point), this is also reflected in the better acoustic characteristics of the fan.

Aerodynamic noise in fans is caused by pressure pulsations in the fan, and it depends on the air flow flowing through the fan domain. Aerodynamic noise includes the following types: vortex noise, boundary layer noise, stationary flow noise, turbulent flow noise and noise of unstable operation of the fan [7]. Therefore, it can be prevented and reduced by adequate construction of the fan, especially the fan impeller. On the other hand, mechanical noise can be reduced by different construction methods.

Several experimental and numerical investigations of acoustic characteristics of centrifugal fans show that backward fan impeller generates less noise than other fan types [8-10]. The most favorable variant certainly is impeller with backward inclined and backward curved blades, preferably with airfoiled blades, while forward inclined and forward curved blades have worse acoustic characteristics. Fan impellers with radial blades have the worst operating and acoustic performances of all fan types, but they are widely used because of the simplicity of their design and production.

Numerical research today deals with operating characteristics of fan, when fan operates in stationary operating conditions, and also can investigate various phenomena that occur in non-stationary operating regimes [11-13]. Also, non-stationary fan operating regimes generate additional noise and increase sound power level of a fan, therefore such operating modes must be avoided at all costs.

The possibility to easily change fan geometry (such as impeller blades, geometry parameters of impeller and/or volute casing), as well as different flow parameters of the gas (such as the type of gas, the gas temperature etc.), gives the numerical approach numerous advantages. But still, experimental measurements are necessary, for the validation of the numerical results, and for the final testing of the fan operating characteristics.

Similar constructions of three different low-pressure centrifugal fans with backward inclined blades are investigated in this research.

2. OPERATING AND ACOUSTIC CHARACTERISTICS OF FANS

Fan operating and acoustic characteristics are functional dependence of operating and acoustic parameters on the volume flow rate of the fan. Volume flow rate is determined in the inlet cross-section of the fan (Q_I).

1.1 Operating curves of low-pressure fans

Fan operating curves are given by the manufacturer for normal temperature and pressure condition of the air (t=20°C, p=1 atm=101325 Pa).

Total (Δp_t), dynamic (p_d) and static (p_s) pressure of the fan are defined as follows:

$$\Delta p_t = p_{t.II} - p_{t.I} \tag{1}$$

$$p_d = p_{d.II} = \frac{1}{2}\rho_{II} c_{II}^2 \tag{2}$$

$$p_s = \Delta p_t - p_d \tag{3}$$

where $p_{t.I}$ and $p_{t.II}$ are total pressures in the inlet and outlet of the fan [Pa], $p_{d.II}$ is dynamic pressure in the fan outlet, ρ_{II} - air density in the fan outlet [kg/m^3] and c_{II} - air velocity in the fan outlet [m/s].

For low pressure fans (Δp_t≤3 kPa), air can be considered incompressible (with constant air density, $\rho_I=\rho_{II}$).

Besides fan pressure characteristics, significant operating parameter is also the fan power, which has to be provided by the motor, usually an electric motor (N) and effective power of the fan (N_{ef} [W])

$$N_{ef} = Q \cdot \Delta p_t \tag{4}$$

Therefore, fan efficiency can be determined using the following equation:

$$\eta = \frac{N_{ef}}{N} = \frac{\beta \cdot Q \cdot \Delta p_t}{N} \tag{5}$$

where β is a compressibility factor which defines the air compressibility:

$$\beta = \frac{\kappa}{\kappa - 1} \frac{\Pi_t^{\frac{\kappa-1}{\kappa}} - 1}{\Pi_t - 1} \tag{6}$$

and for low-pressure fans β=1.

When fan operates with static pressure curve (for fan with suction ducts), the static efficiency can be calculated as:

$$\eta_s = \frac{\beta \cdot Q \cdot p_s}{N} \tag{7}$$

Flow and pressure coefficient, and also power coefficient, as dimensionless quantities are calculated using following formula:

$$\varphi = \frac{4 \cdot Q}{D^2 \pi}, \quad \psi = \frac{4 \cdot \beta \cdot \Delta p_t}{\rho_I u^2}, \quad \lambda = \frac{\varphi \psi}{\eta} \tag{8}$$

where D – outlet diameter of the fan [m], u - circumferential velocity on the fan outlet [m/s].

1.2 Acoustic characteristics of fans

Aerodynamic, electromagnetic and mechanical noise are the three types of fan noise [14]. Aerodynamic noise is created in the fan by the air pulsations, which occur in the fan runner as well as in the stationary parts of the fan (inlet cone and volute casing). Aerodynamic noise cannot be easily reduced by simple constructive solutions, it should be prevented during the design and manufacture of the fan.

The intensity of sound waves (I_s) is the ratio of the sound wave power to the surface normal to the propagation of those waves ($I_s=N_s/A$) [7]. The human ear registers intensity of sound waves from $I_s=10^{-12}$ W/m^2 (on the threshold of audibility) to 10 W/m^2 (at the limit of pain).

For estimation of sound power level (SPL or L_N [dB]) there is an equation [7]:

$$L_N = 10log\frac{N_s}{N_{s.0}} = 10logN_s + 120 \tag{9}$$

where, N_s is the power of the sound wave and $N_{s.0}$ is the power of the sound wave on the threshold of audibility.

The total sound power level for multiple sound sources (i=1,2, ..., n) can be calculated as the sum of sound power levels of all sources, and it can be presented as:

$$L_{N.\Sigma} = 10log\left(\sum_{i=1}^{N}\frac{N_{s.i}}{N_{s.0}}\right) = 10log\left(\sum_{i=1}^{N}10^{0,1L_{N.i}}\right) \tag{10}$$

Numerical simulation of flow in fans offers an acoustic analysis and the estimation of sound power level using the Proudman noise source model [15].

ANSYS software incorporates this model, where the Proudman magnitude formula (AP [W/m^3]) can be calculated using the equation:

$$AP = \alpha\rho_o\frac{u^3}{l}\frac{u^5}{\alpha_o^5} \tag{11}$$

where, u - turbulent velocity, l - length scales, α_o – the speed of the sound and α – a model constant.

For quick estimation of sound power level of fans, there are many empirical equations in the literature.

When it comes to obtaining sound power level of centrifugal fans, some of the empirical formulas stand out, showing a very good agreement with the actual quantities obtained by measurements.

One of them is formula proposed by ASHRAE [16]:

$$L_N = K_N + 10log_{10}Q + 20log_{10}p_t + BFI + C_N \tag{12}$$

where: L_N – sound power level [dB], K_N – specific sound power level depending on the fan type, Q – volume flow rate [cfm], p_t – total pressure [inches of H_2O], BFI – blade frequency increment, which is the correction for pure tone produced by the blade passing frequency (BFI=number of blades x rpm/60 [Hz]), C_N – efficiency correction which can be calculated with formula: $C_N=10+10log10((1-\eta)/\eta)$.

Recknagel, Sprenger and Hönmann [17] suggested three formulas for determining the sound power level of centrifugal fans, but the best results we obtained by following one:

$$L_N = 25 \pm 4 + 10log_{10}Q + 20log_{10}p_s \tag{13}$$

where Q [m^3/h] is the volumetric flow rate of the fan, p_s [mmH_2O] is static pressure of the fan and N [kW] is the fan power.

Some of the engineering web portals, such as engineeringtoolbox.com offer the formula for calculation of fan sound pressure level:

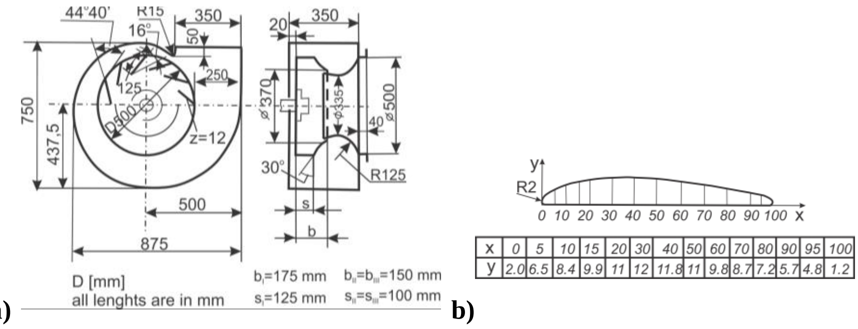
$$L_N = 40 + 10log_{10}Q + 20log_{10}p_s \tag{14}$$

which also gives good values of sound pressure level.

Graphs of the sound power level curves obtained using previous formulas will be shown and analyzed later on.

3. Numerical simulation of low-pressure characteristics of fans with inclined blades

For obtaining numerical analysis of low-pressure centrifugal fans with backward inclined blades, three centrifugal fans were used: two with non-profiled blades (with different impeller width) and one with profiled blades. The geometric parameters of centrifugal fans are similar for all fans, given in Fig.1a.



x	0	5	10	15	20	30	40	50	60	70	80	90	95	100
y	2.0	6.5	8.4	9.9	11	12	11.8	11	9.8	8.7	7.2	5.7	4.8	1.2

Fig. 1 a) Geometry of the centrifugal fans (I, II, III) with backward inclined blades; b) Blade profile of fan III

Fan I and II (III) have a different impeller width: the impeller width of fan I is b_I=175 mm, while the impeller width of the fan II and III is b_{II}=150 mm (Fig.1). The fan outer diameter is D=0,5 m. All centrifugal fans have 12 blades each and blades are straight and inclined, 125 mm long. Fan III has profiled blades, which geometry is given in Fig.1b.

All impellers rotate with rotational speed n=1200 rev/min operating with best efficiency points:

- for fan I: Q=1,42 m^3/s, Δp_t=520 Pa (i.e., φ=0,23, ψ=0,89) η=0,81,
- for fan II: Q=0,96 m^3/s, Δp_t==334 Pa (i.e., φ=0,187, ψ=0,82) η=0,78,
- for fan III: Q=0,96 m^3/s, Δp_t==334 Pa (i.e., φ=0,18, ψ=0,764) η=0,82.

Numerical models of fans consist of two domains: rotating impeller and stationary volute casing. All meshes are unstructured, generated using ICEM CFD 19.0. Meshes consist of approximately 4,9 million, 4,6 million and 4,3 million elements, respectively for fans I, II and III which are mainly tetrahedral, with pyramidal and wedges elements (Fig.2).

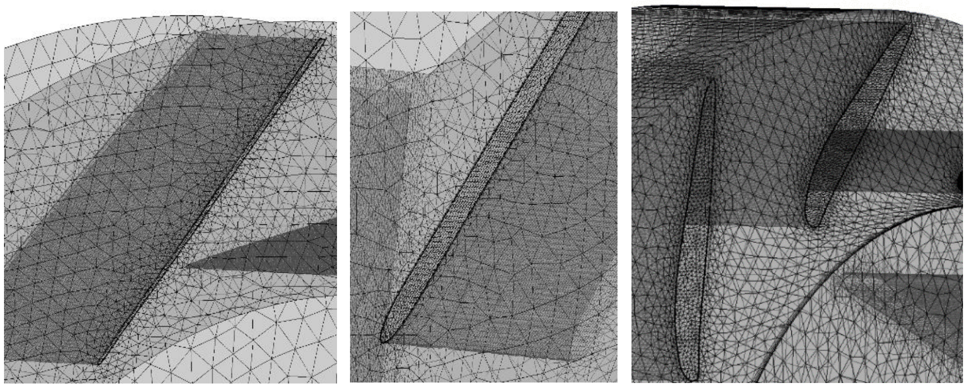


Fig. 2 Meshes of centrifugal fans.

Numerical simulations are conducted using ANSYS 19.0. It was used k-ε turbulent model, that is common for application in turbomachinery [18]. Rotating and stationary parts are connected by frozen rotor stationary interface. For the inlet boundary condition is given an atmospheric pressure and for the outlet boundary condition mass flow rates.

The high-resolution scheme was used for numerical interpolation, and the convergence criteria were that root mean square values of the equation residuals are 10^{-4}.

3.1 Grid independence test

In order to establish the best discretization mesh, from the aspect of the accuracy of the obtained results, but also the time required for numerical simulations, the grid independence test was conducted. The results are presented in Table 1.

Table 1 Grid independence test

Fan I					Fan II					Fan III				
No of elem.	ψ	Rel. error [%]	ψ_s	Rel. error [%]	No of elem.	ψ	Rel. error [%]	ψ_s	Rel. error [%]	No of elem.	ψ	Rel. error [%]	ψ_s	Rel. error [%]
$2.1\cdot10^6$	0.944	3.7	0.773	3.0	$1.9\cdot10^6$	0.78	5.1	0.678	5.8	$1.9\cdot10^6$	0.768	2.3	0.694	0.1
$4.9\cdot10^6$	0.932	2.4	0.766	2.1	$4.6\cdot10^6$	0.79	3.4	0.685	4.8	$4.3\cdot10^6$	0.754	0.4	0.695	0.06
$5.2\cdot10^6$	0.925	1.7	0.759	1.2	$6.0\cdot10^6$	0.80	2.8	0.688	4.5	$7.6\cdot10^6$	0.749	0.2	0.698	0.02

Numerical meshes consist of approximately 4.9 million elements for fan I, 4.6 million elements for fan II and 4.3 million elements for fan III were chosen, due to excellent numerical results considering the time required for obtaining series of numerical simulations, conducted for a range of different flow rates.

3.2 Validation of numerical results

For validation of numerical results, total and static pressure curves were compared with results obtained by measurements [8, 9], presented in Fig.3. For all pressure curves it is noticeable that there is a very good agreement of compared curves, especially for the best efficient point (BEP), where total pressure is up to 3.4%, while static pressure does not exceed 4.8%. The largest deviations of results are in the area of the lowest and highest flow rates.

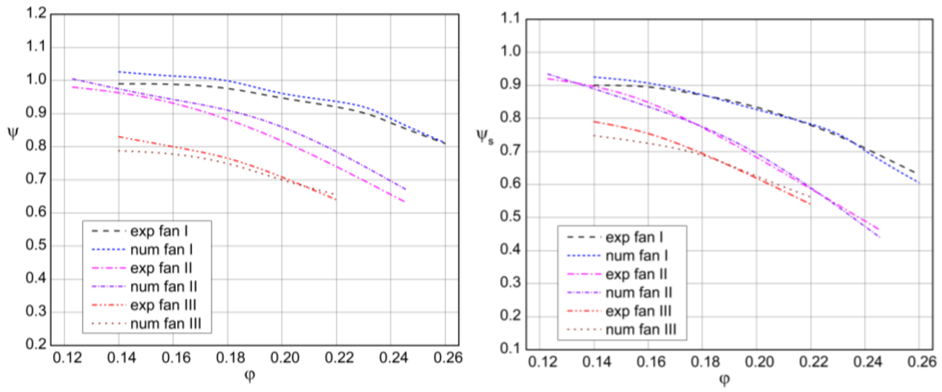


Fig. 3 Total and static pressure coefficients

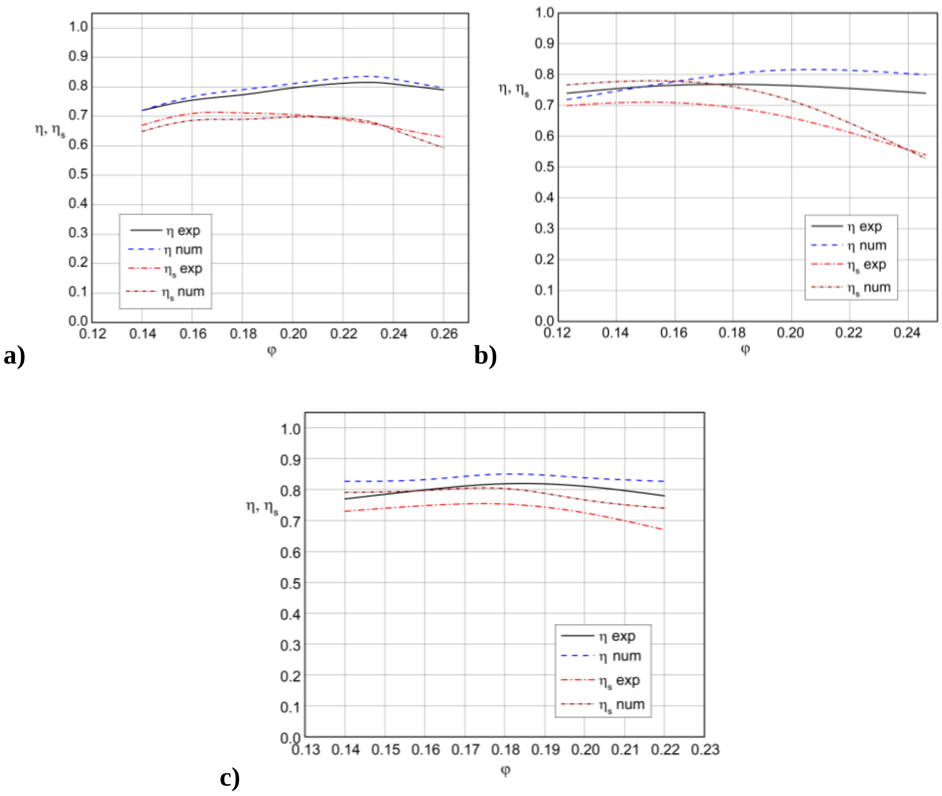


Fig. 4 Efficiency comparison for fans a) I, b) II and c) III

The comparison of the results of the numerically obtained efficiency for all fans is shown in Fig.4.

The numerical simulations obtained higher efficiency values than they actually are. For fan I numerically obtained efficiency curves have relative deviations of values up to 3.5%, except for the maximum flow rate values, where relative error of static efficiency is approximately 5.8%. This represents good numerical results, especially for such complex flows that exist in centrifugal fans. Fan II shows larger deviations of numerical efficiency, up to 10%. Total efficiency of fan III, which has profiled blades is up to 7,4%, while static efficiency of fan III has larger deviations of numerical results, up to 10% for higher flow rates.

4. Analysis of Numerical Results

Operating flow rate range for both fans are similar, but it is obvious that the pressure curves (i.e., pressure coefficient) differ significantly, especially for higher flow rates (Fig.3), due to the difference in the geometry of the impeller.

Also, there is a noticeable difference in centrifugal fan efficiency values. Wider impeller has the highest values of total and static pressure. On the other hand, the impeller with profiled blades obtained the highest efficiency values, while operating with lowest pressure values. The largest relative error is obtained for fan II when operated with high flow rates.

Comparison of all efficiency curves is given in Fig.5.

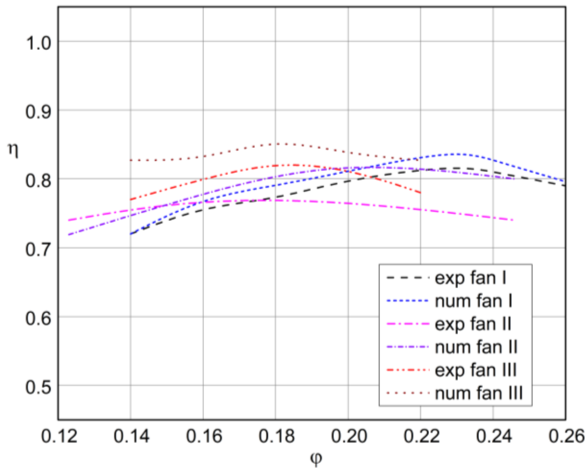


Fig. 5 Efficiency comparison for fans I, II and III

Power coefficients are presented in Fig.6, showing very good agreement. For fan I numerical results have relative error less than 1% (for lowest flow rate up to 3.6%), for fan II up to 5.5%, while fan III has larger deviation of power coefficients which is up to 7.4%.

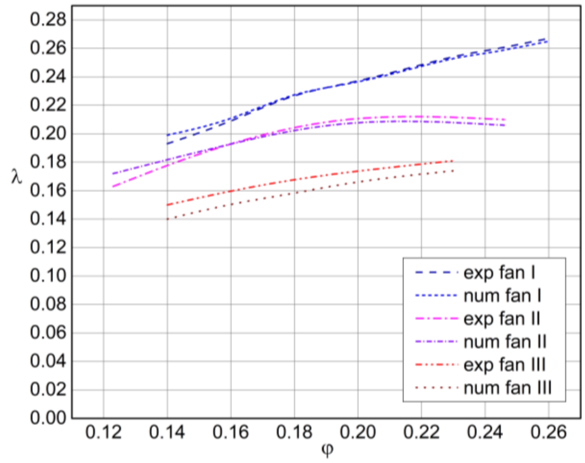


Fig. 6 Power coefficient curves of centrifugal fans I, II and III (n=1200 rev/min)

The acoustic characteristics of centrifugal fans that are considered in the paper, are also numerically obtained and they have relatively good agreement with measured values, which is up to 9% for fan I, for fan II up to 2.5% and for fan III up to 7% (Fig.7). Numerical values obtained by the ANSYS software uses the Proudman noise source model for estimation sound power level.

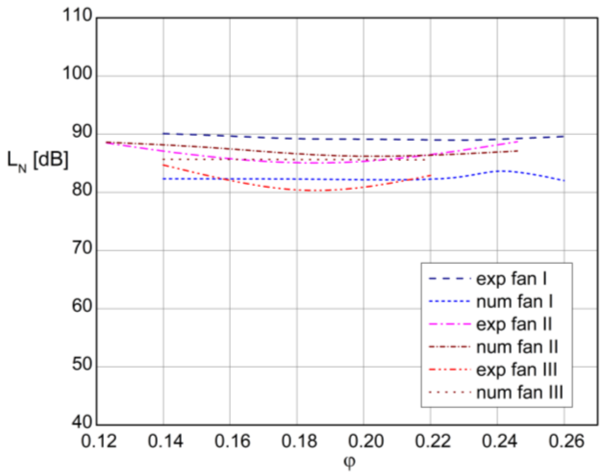


Fig.7 Numerically obtained SPL of fans.

The comparison of numerical results and results obtained by empirical formula used in technical practice, given by equations (12), (13) and (14), is presented in Fig.8, Fig.9 and Fig.10, for fans I, II and III respectively.

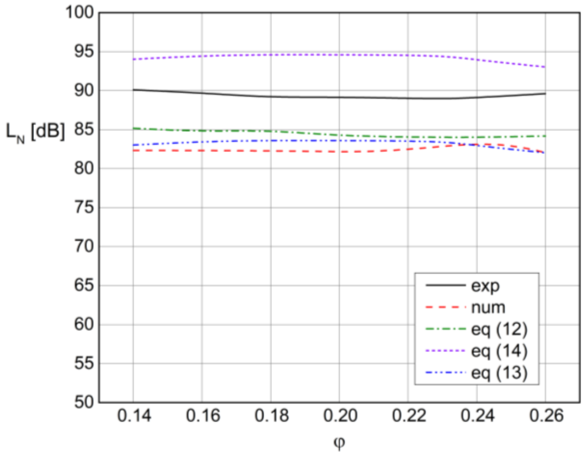


Fig. 8 Comparison of SPL for the fan I

Fig.8 shows a larger difference between obtained values and measured one, for fan I. Equations (12) and (13) give results similar to numerical ones, while equation (14) obtains the highest value of sound power level.

A better agreement of sound power level was obtained for fan II (Fig.9), where only results obtained by equation (13) show larger deviations of results. Equation (12) shows the best agreement with experimental and numerical results.

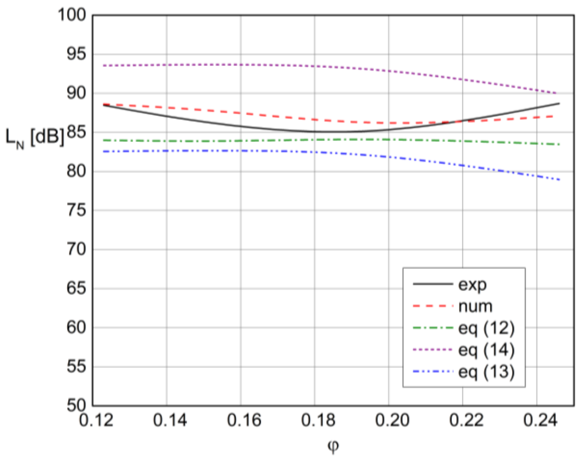


Fig. 9 Comparison of SPL for the fan II

For Fan III equations (12) and (13) show the best agreement with experimental results, as shown in Fig.10. Using the equation (14) much higher values of sound power level are calculated than by other equations (Fig.10). Therefore, this equation should be taken with a lot of caution.

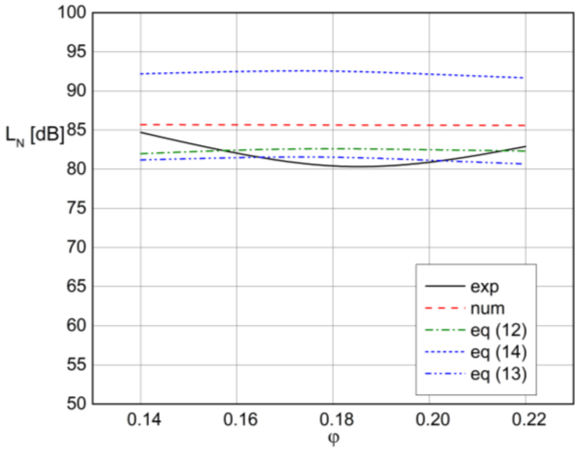


Fig. 10 Comparison of SPL for the fan III

For all investigated fans the best results of sound power level are calculated using equation (12), which was proposed by ASHRAE [16].

Ansys 19.0 can provide visualization of noise sources, derived from Ffowcs Williams and Hawkings (FW-H) equations. These sources can be compared with each other and with the broadband noise to determine the dominant noise source in the design.

Monopole source is connected to the movement of the source surface (self-noise), and for these centrifugal fans monopole source strength is not very high (Fig.11a and Fig.12a). Dipole source describes the interaction between the fluid and the surface of the source and in Fig.11b and Fig.12b it is noticeable higher rate of the dipole source strength on some parts of the impeller blades. Quadrupole source (Fig.11c and Fig.12c) is related to the turbulence fluctuation levels of the fluid, showing the critical places in the fan impeller, where an increased noise level can take a place.

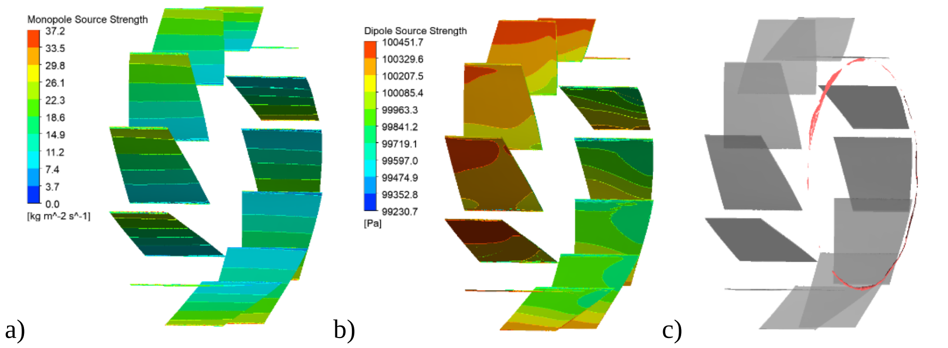


Fig. 11 Noise sources for the fan II (φ=0.2): a) monopole source, b) dipole source and c) quadrupole source

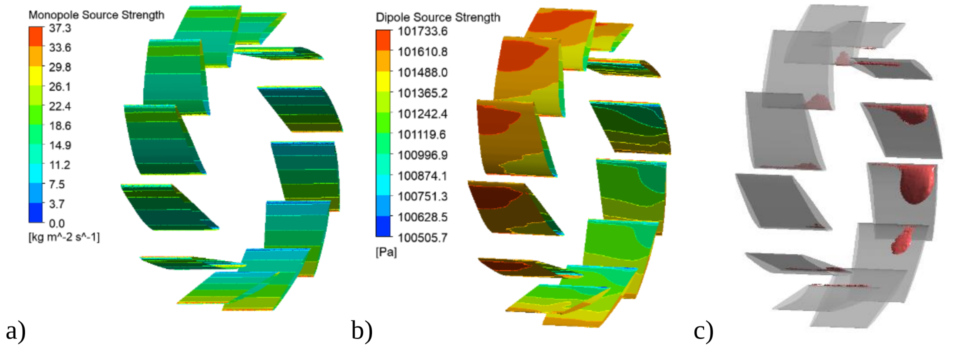


Fig. 12 Noise sources for the fan III (φ=0.2): a) monopole source, b) dipole source and c) quadrupole source

5. Conclusion

Numerical simulation of centrifugal fans offers the possibility of obtaining comparatively good results, which enables analyzing the air flow in the centrifugal fans and its operating and acoustic characteristics. Thereby it is possible to determine to what extent a fan construction corresponds to the operating parameters of the fan. All numerical results for all presented cases are in good agreement with the measured values.

Any change made in the construction of the fan impeller, whether its blades or other parts of the fan, changes the operating curves of the fan. This is shown on the example of fans with different impeller widths, as well as fans with non-profiled and profiled impeller blades.

Increasing the width of the fan impeller leads to an increase of the pressure curves in the flow range for the economic fan operation. Also, fan impellers with a larger impeller width have a lower value of efficiency. For smaller fan width, maximum efficiency shifts towards lower flow rates. Maximum relative error of fan efficiency obtained numerically does not exceed 10%. The total pressure and total pressure coefficient do not exceed 3,4% of measured results, while the static pressure and static pressure coefficient goes up to 4,8% of the measured values.

An increase of the fan impeller width leads to power coefficient increases up to 25% for the larger air flow rates. Profiled blades lower the flow domain and flow rate, lowering the power (coefficient) of the fan.

Numerical estimation of acoustic characteristic of fan, which is sound power level of fan, show certain deviations compared to measured values.

For all centrifugal fans which were discussed in the paper, the equation (12) proposed by ASHRAE gives the best results, while equation (14) obtains the least accurate results.

Numerical simulations of flow in low-pressure centrifugal fans are a useful tool for analyzing the effect of any geometry changes on the fan performances. It provides us with plenty of useful information concerning all physical quantities, including their visualization in the flow domain, which allows us to spot potentially problematic areas in the fan.

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