LOW PRESSURE CENTRIFUGAL FANS OPERATING WITH DIFFERENT AIR TEMPERATURES

by

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For normal operating conditions of the centrifugal fan, the manufacturers provide operating curves (total pressure of the fan, static pressure of the fan, power, and efficiency) for standard air conditions. However, when a fan operates with air at different, usually elevated temperatures, their operating curves can significantly differ. As a result, the fan efficiency decreases, and the energy efficiency of the fan may be reduced. Fan operation in such air conditions also affects its acoustic characteristics. This paper investigates two types of centrifugal fans: with forward curved and backward inclined blades. The main goal was to determine how their operating and acoustic characteristics change with the change in air temperature. For this analysis, low pressure centrifugal fans are used, for which we already had measured operating and acoustic characteristics. The influence of the type of impeller and the curvature of the blades play an important role in the operation of the fan. For numerical investigation, ANSYS CFX software was used, and obtained results show excellent agreement with experimental measurements.

Key words: *centrifugal fan, forward and backward blades, air temperature, aerodynamic and acoustic characteristics, efficiency*

Introduction

Centrifugal fans have significant industrial application. The most common use of centrifugal fans is in various industrial processes, such as ventilation and air circulation, drying, and cooling. Also, they are largely used in low pressure pneumatic conveying systems, in air supply systems for combustion purposes, for industrial cooling, hot air exhaust purposes, *etc*.

As a type of radial turbomachines, centrifugal fans operate on a principle of transforming the mechanical energy of the fan runner blades into the energy of fluid flow, which consequently increases the gas (air) pressure from the inlet to the outlet cross-section of the fan. They are used for higher flow-rates and for a higher increase in air pressure. The most important fan operating parameters are volume flow-rate, fan pressure (total and static), fan power, fan efficiency (total and static), and their functional dependences on volume flow-rate are known as fan operating characteristics or fan operating curves. The fan operating point is obtained at the intersection of the fan pressure curve and system curve. When a fan operates with the highest efficiency, then it operates in optimal operating mode, called best efficiency

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point (BEP). Achieving the BEP is very important for the energy efficiency of the entire system. Very often a fan operates with a larger range of volume flow-rates, when it is required to operate with efficiency up to 80% of the maximal fan efficiency (so called economical fan operation).

Fan manufacturers provide fan operating curves that are obtained (measured) for standard air conditions, usually for operating air temperatures of 20 °C, pressure 101325 Pa (1 atm), and relative humidity of 50%, when the air density is 1.2 kg/m^3 [1]. Low pressure fans are considered to achieve total pressure less than 1 kPa, therefore the air density can vary less than 0.7%, when air can be considered as incompressible fluid. But, if the temperature varies significantly, the density will also change, and this will have an effect on all other operating parameters. Fan curves should be recalculated, considering other influences of the fan components as well. In addition to achieving the desired performance characteristics of a fan, when choosing a fan, it is also important to pay attention to the acoustic characteristics. The fan operation is always accompanied by noise, which may be caused by various reasons, and, in some cases, can limit the use of the fan.

Sources of aerodynamic noise are pressure pulsations, which follow the air-flow in the fan, while sources of mechanical noise are mechanical oscillations of structural elements of the fan [1]. Mechanical noise can also be caused by unbalanced fan runner, oscillation of the bearings, shrinkage of the fan case during the operation, *etc*. The level of the mechanical noise can be reduced by different construction measures [2], but such measures generally make the whole system even more expensive. The aerodynamic noise of the fan can be divided into the following causes [1]:
– vortex noise – caused by pressure pulsations, when a vortex is formed and separated from surfaces, and then these vortexes encountered solid boundary surface of the fan,
– noise of the boundary layers – caused by pressure pulsations in turbulent boundary layers, near the solid surfaces,
– noise of the un-stationary flow – due to the finite number of blades and unsteady air-flow,
– noise of turbulent flow – caused by pressure pulsation due to the transverse transfer of the vortex from the vortex trails behind the blades to the main air-flow, and
– noise of unstable fan operation – caused by pressure pulsations in instable operation regime of the fan (if fan operates in such an operating regime).

An attempt at acoustic optimization of centrifugal fans using CFD technics and FLUENT software, indicates the possibility of minimizing the fan noise, compering the overall sound power level of different fan geometry [3]. Experimental and numerical investigation of acoustic characteristics of centrifugal fans operating with standard air conditions, confirming that centrifugal fan with backward impeller generates less noise and high discharge compared to radial and forward impellers [4]. The highest noise level is expected when operating fans with radial runner blades. Slightly lower noise level is obtained in forward inclined and forward curved blades, while better acoustic performances are expected with backward inclined and backward curved blades. Also airfoiled blades have better acoustic characteristics compared to non-profiled blades.

Many investigations have been conducted on the influence of centrifugal fan geometry on its aerodynamic and acoustic characteristics, particularly the effect related to irregular blade spacing, blades number, and radial distance between the impeller periphery and the volute tongue [5, 6]. Sometimes fan noise can be the result of non-stationary regimes, such as surge and rotating stall, and these regimes must be avoided [7, 8]. Some numerical research has resulted in proposed new methods for reducing the fan noise [9]. A large number of experimental research, both aerodynamic and acoustic operating parameters [10-13], have been enabled further analysis of these machines, as well as validation of numerical results.

Determination of performance of axial-flow low pressure fans, operating with hot air, indicates the necessity of correction of fan operating parameters, when it operates with higher air temperatures [14]. Recalculation of fan curves using the affinity laws shows certain deviations, and this analysis also should be done for centrifugal fans. In this paper, the possibilities to numerically determine the performance and acoustic characteristic of two different centrifugal fans, operating with higher air temperatures are considered.

Performance characteristics of low pressure centrifugal fans

Performance characteristics of fan are obtained as a functional dependence of operating parameters on volumetric flow-rate. The volume flow-rate of the fan is determined in the inlet cross-section [1]:

$$Q = Q_I = c_I A_I \tag{1}$$

where c_I [ms^{-1}] is the air velocity in the inlet cross-section and A_I [m^2] – the size of the inlet cross-section.

The total pressure of the fan, Δp_t [Pa] can be calculated as the difference between the total pressures in the outlet (II) and inlet (I) cross-sections, or as the sum of the static, p_s, and dynamic, p_d, fan pressure [1]:

$$\Delta p_\mathrm{t} = p_{\mathrm{t}.II} - p_{\mathrm{t}.I} = p_\mathrm{s} + p_\mathrm{d} \tag{2}$$

where the dynamic pressure of the fan is:

$$p_\mathrm{d} = p_{\mathrm{d}.II} = \frac{1}{2}\rho_{II} c_{II}^2 \tag{3}$$

where c_{II} is the air velocity in the outlet cross-section, for air density in the outlet cross-section of the fan, ρ_{II}.

Static pressure is obtained as the difference between total and dynamic pressure of the fan:

$$p_\mathrm{s} = \Delta p_\mathrm{t} - p_\mathrm{d} \tag{4}$$

Low pressure centrifugal fans are operated with small Mach numbers (Ma < 0.2). For the low pressure and the mid-pressure fans ($\Delta p_\mathrm{t} \leq 3$ kPa), the density of the air does not change significantly, therefore $\rho_I = \rho_{II}$ [1].

The power of the fan, N, is the value obtained by the motor, while effective power of low pressure fan, N_ef, can be obtained using the following equation:

$$N_\mathrm{ef} = Q\Delta p_\mathrm{t} \tag{5}$$

The fan efficiency represents the ratio of effectively utilized fan power and the invested power:

$$\eta = \frac{N_\mathrm{ef}}{N} = \frac{\beta Q \Delta p_\mathrm{t}}{N} \tag{6}$$

where β is the coefficient that takes into account gas compressibility:

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

and for low pressure fans can be taken $\beta = 1$.

Similarly, the static efficiency can be determined using:

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

Operating parameters of the fan as a functional dependence on the volume flow-rate define the fan operating curves. Manufacturers usually provide performance characteristics for normal (standard) fan operating conditions (NTP – normal temperature and pressure conditions: t = 20 °C, p = 1 atm = 101325 Pa, ρ = 1.2 kg/m^3). If a fan operates with different air conditions, the operating curves should be recalculated for the existing fan operation conditions, using affinity laws.

Recalculation of fan operating parameters using the affinity laws

For the large values of Reynolds numbers, which is the case in centrifugal fans, the affinity lows allowed to (re)calculate operating curves for different air temperature. Since the value of air density changes with the temperature, the equations for calculating volume flow-rate and total pressure of the fan are [1]:

$$Q' = Q, \quad \Delta p_t' = \Delta p_t \frac{\rho_I'}{\rho_I}\frac{\eta_h'}{\eta_h} \tag{8}$$

For low pressure fans, where air can be treated as incompressible fluid the overall efficiency is:

$$\eta = \eta_h \eta_Q \eta_m \tag{9}$$

where the volumetric and mechanical efficiency can be obtained:

$$\eta_Q = \frac{Q}{Q_k} \text{ and } \eta_m = \frac{N_k}{N} \tag{10}$$

Fan efficiency is calculated by multiplying the values of mechanical, volumetric, and hydraulic efficiency, *i.e.* $\eta = \eta_m \eta_Q \eta_h$. The new efficiency is the function of previous efficiency value, $\eta' = \mathrm{f}(\eta)$, and there is no universal equation for its determination. The ratio η_h'/η_h in eq. (8) is the function of Reynolds number, for the given temperature condition [14]. If Δp_k is the pressure rise in the fan runner, hydraulic efficiency is:

$$\eta_h = \frac{\Delta p_t}{\Delta p_k} \tag{11}$$

If $\eta_Q = \eta'_Q$, $\eta_m = \eta'_m$ can be assumed, then the ratio of hydraulic efficiencies is equal to the ratio of total efficiency: $\eta'_h/\eta_h = \eta'/\eta$. When Reynolds number has a high value, such is the case in the centrifugal fans, the fan efficiency depends on the Reynolds number. Figure 1 shows the influence of Reynolds number on fan efficiency [10].

Reynolds number, as the function of kinematic viscosity, changes with temperature. Therefore, with temperature increasing, Reynolds number decreases. For the investigated centrifugal fans when air temperature increases from 25 °C to 300 °C, calculated Reynolds number changes: $\mathrm{Re}_u = 5.91 \cdot 10^5$-$1.93 \cdot 10^5$ for the centrifugal fan with forward curved blades and $\mathrm{Re}_u = 8.38 \cdot 10^5$-$2.73 \cdot 10^5$ for the centrifugal fan with backward inclined blades. Equation for calculating the value of fan efficiency reduction with a decrease of Reynolds number, can be written [1]:

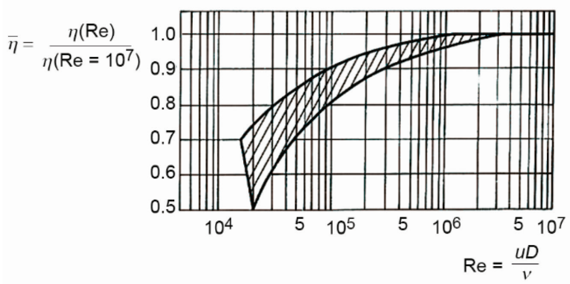

Figure 1. The fan efficiency dependence of the Reynolds number

$$\bar{\eta} = 1 - \frac{480}{\mathrm{Re}^{0,7}}, \quad \text{for} \quad \mathrm{Re} \geq 5 \cdot 10^4 \tag{12}$$

Using the relation $\eta' = \eta[\bar{\eta}'(\mathrm{Re}')/\bar{\eta}(\mathrm{Re})]$, the equation (11), becomes:

$$\eta' = \eta \frac{1 - \dfrac{480}{\mathrm{Re}'^{0,7}}}{1 - \dfrac{480}{\mathrm{Re}^{0,7}}} \tag{13}$$

There is also an equation for obtaining the efficiency correction given by Ackert [1], which can be used for both fan type, centrifugal and axial flow fans, but only for BEP and for the area nearby:

$$\eta' = 1 - \frac{1-\eta}{2}\left[1 + \left(\frac{\mathrm{Re}}{\mathrm{Re}'}\right)^{0,2}\right] \tag{14}$$

After obtaining numerical simulation results, all proposed equations are used and compared in the investigated case of centrifugal fan with forward curved blades.

Numerical simulations of low pressure centrifugal fans

Two centrifugal low pressure fans are investigated, the first one is a centrifugal fan with forward curved blades, and its geometry and performance and acoustic curves are presented in fig. 2 [10].

Operating curves are presented in the non-dimensional form, where non-dimensional operating parameters: flow coefficient φ, pressure and static pressure coefficients (ψ and ψ_s), power coefficient λ and specific speed n_q, for low pressure fans ($\beta = 1$) are defined:

$$\varphi = \frac{Q}{Au} = \frac{4Q}{D^2 \pi u}, \quad \psi = \frac{2\Delta p_t}{\rho_I u^2}, \quad \psi_s = \frac{2p_s}{\rho_I u^2}, \quad \lambda = \frac{2N}{\rho_I A u^2} = \frac{\varphi\psi}{\eta}, \quad n_q = 157.8\frac{\varphi^{0,5}}{\psi^{0,75}} \tag{15}$$

The fan outer diameter is D = 0.5 m, the fan width is 350 mm (fan runner width is 250 mm). The fan runner has circular-arched, non-profiled blades. The number of blades is $z = 32$. Blade angles are $\beta_I = 90^\circ$ and $\beta_I = 165^\circ$. Dimensions of the spiral casing are also given in fig. 2. Operating characteristics of the centrifugal fan with forward curved blades, for air temperature 25 °C and the BEP, are Q = 1979 m^3/s, Δp_t = 577 Pa, η = 0.73, n = 700 rpm (n = 11.67 s^{-1}), *i.e.* φ = 0.55; ψ = 2.9, and n_q = 52.7.

The geometry of the second low pressure centrifugal fan with backward inclined blades and its performance and acoustic characteristics are given in fig. 3. The fan outer diameter is D = 0.5 m, while the fan width is 300 mm. The number of blades is z = 12. Blade angles are given in fig. 3. Best efficient point, for air temperature 25 °C, is defined with following values: Q = 0.96 m^3/s, Δp_t = 334 Pa, η = 0.78, n = 1000 rpm (n = 16.67 s^{-1}), *i.e.* φ = 0.187; ψ = 0.82, and n_q = 79.2.

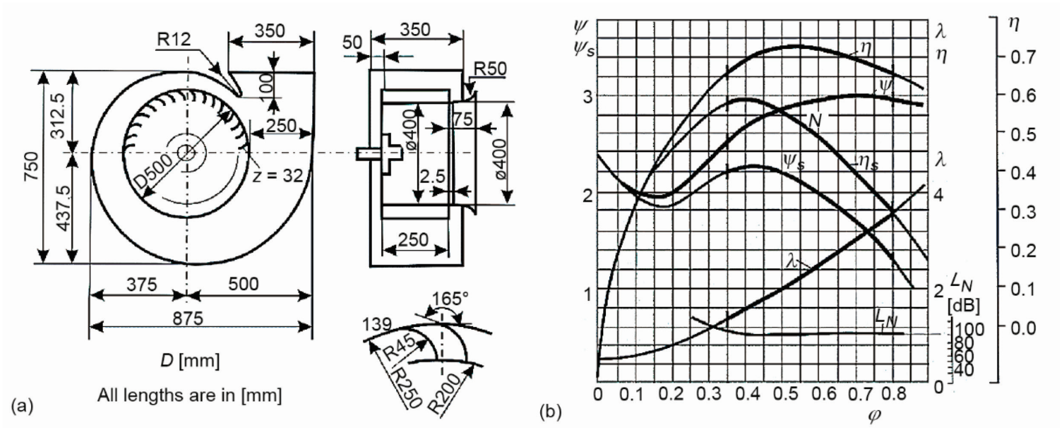

Figure 2. Geometry of the centrifugal fan with forward curved blades and its operating curves

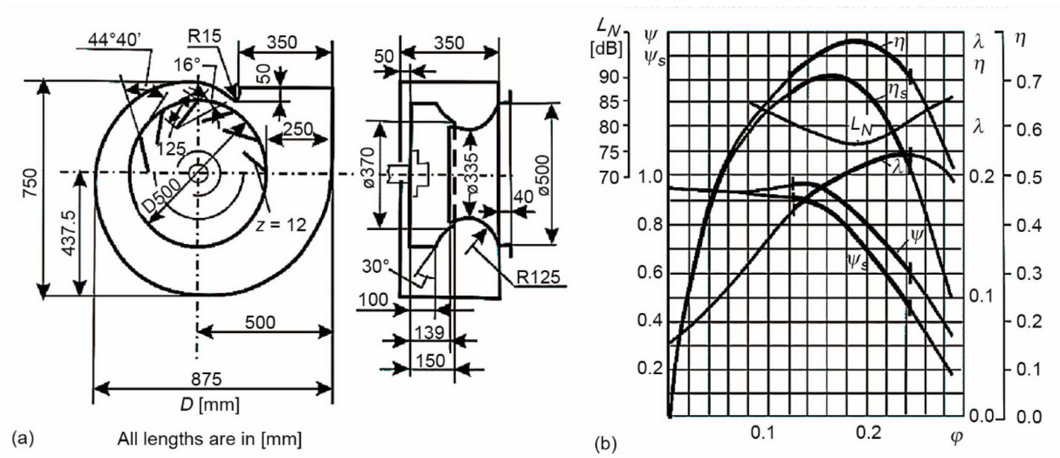

Figure 3. Geometry of the centrifugal fan with backward inclined blades and its operating curves

According to geometry of fans, figs. 2 and 3, the numerical model of both fans consists of two domains: fan runner and the volute casing, which are discretized using ICEM CFD 19.0. Both meshes are unstructured.

The mesh of the centrifugal fan with forward curved blades consists of 1300130 nodes and 5974688 elements, of which they are 4741938 tetrahedral, 939 pyramidal and 1231811 wedges, fig. 4(a). The density of the discretization mesh is higher near the wall region and around blades.

The mesh of the centrifugal fan with backward inclined blades consists of 1300130 nodes and 4611793 elements (3388366 tetrahedral, 1357 pyramidal, and 122070 wedges), from which the impeller mesh contains about 1 million elements, fig. 4(b).

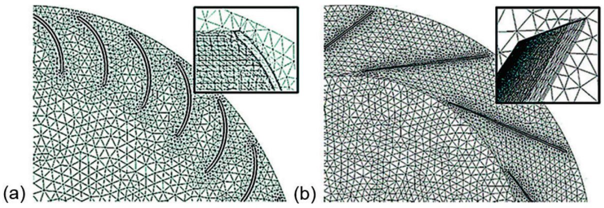

Figure 4. Centrifugal fan mash impellers; (a) with forward curved blades and (b) with backward inclined blades

The maximal value of y^+ in the fan runner with forward curved blades is 11.3, while the average value $y^+ = 9.8$, and for the centrifugal fan with backward inclined blades the average value of $y^+ \approx 11$. These values are in accordance with the recommendations given in the literature [15, 16].

The centrifugal fans are numerically simulated using ANSYS 19.0. It was used k-ε turbulent model, which is commonly used for numerical simulation of turbomachinery. Numerical models have two domains, stationary and rotating, which are connected by a frozen rotor interface. Boundary conditions are atmospheric pressure in the inlet and mass-flow rate in the outlet. For numerical interpolation was used the high-resolution scheme. The convergence criteria were that root mean square values of the equation residuals are 10^{-4}, which is common in centrifugal turbomachines with a spiral casing.

The grid independent test was obtained for the BEP for three different discretization meshes, shown in tab. 1 (fan with forward curved blades for $\varphi = 0.55$ and $\psi = 2.9$, and with backward inclined blades for $\varphi = 0.19$ and $\psi = 0.82$). Pressure coefficient and efficiency were compared for three different meshes.

Table 1. Grid independence test results

With forward curved blades					Fan with backward inclined blades				
Number of mesh elements	ψ	Relative error, ψ [%]	η	Relative error, η [%]	Number of mesh elements	ψ	Relative error, ψ [%]	η	Relative error, η [%]
≈ 5000000	2.814	3	0.796	6.8	≈ 1850000	0.78	5.1	0.736	6.2
≈ 6000000	2.875	0.9	0.734	1.5	≈ 4600000	0.79	3.4	0.814	3.8
≈ 8500000	2.895	0.2	0.747	0.3	≈ 6000000	0.8	2.8	0.807	3.0

Due to good agreement with experimental results, and less time required for conducting a series of numerical simulations, for the fan with forward curved blades was chosen the mesh with approximately $6\cdot10^6$ elements, and for the fan with backward inclined blades the mesh with approximately $4.6\cdot10^6$ elements.

Furthermore the validation on numerical results are obtained for selected meshes, fig. 5, comparing the total pressure curves of the fan, operating with air temperature of 25 °C with curves obtained by measurements, which are given in figs. 2 and 3 [10].

The relative error of the total pressure coefficient in the BEP is 0.9% and 4%, respectively, for both centrifugal fans with forward curved and backward inclined blades, and the relative error of the static pressure coefficient is 1.1% and 4.9%. The mean relative deviation of total pressure coefficient is around 1.6% and less than 3%, respectively, and for static pressure coefficient is less than 4% for both investigated fans.

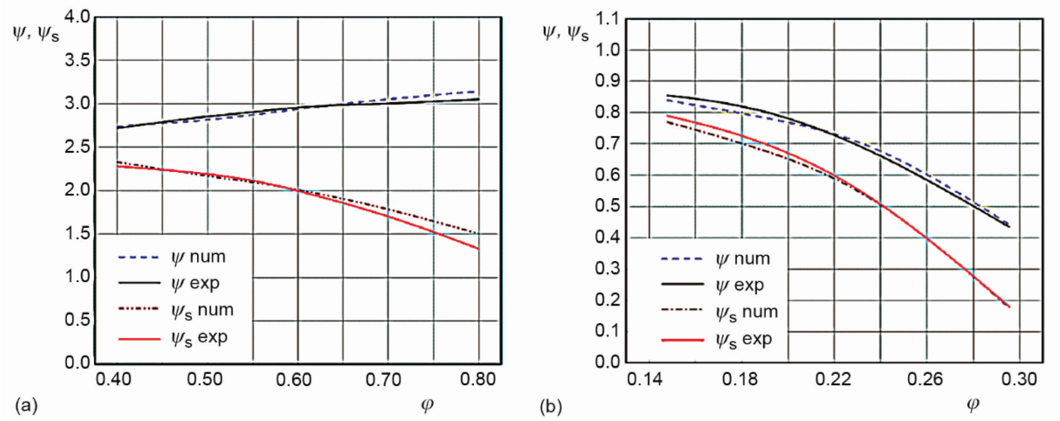

Figure 5. Operating curves of the centrifugal fan (ψ and ψ_s); (a) with forward curved blades and (b) with backward inclined blades

Total and static pressure curves for different air temperatures

The low pressure centrifugal fans operate with different air temperatures (25 °C, 50 °C, 100 °C, 200 °C, 300 °C). Operating pressure curves obtained by numerical simulations are presented in fig. 6, for the economical area of fan operation, where fan efficiency is high, according to the recommendations $\eta > 0.8\,\eta_{\max}$.

Using affinity laws, eq. (8), pressure curves are calculated for different air temperatures, and compared with pressure curves obtained by numerical simulations of flow in centrifugal fans, fig. 6.

There is a great similarity of the pressure curves (total and static) obtained by calculating using affinity laws and numerically. However, there are areas of larger deviation of pressure curves obtained numerically and by recalculation using affinity low. The fan with forward curved blades shows greater deviation of obtained results for higher flow-rates, for total pressure of the fan max 8.2% and for static fan pressure up to 13.9%. The fan with backward inclined blades shows larger deviation of obtained values for total and static pressure of the fan up to 18.8%, both for the lower and higher flow-rates. When calculating operating parameters of fans for different air temperatures, certain corrections of the results obtained by affinity laws should be made, especially if fan does not operate in optimal regimes.

Figure 7 shows specific energy-mass flow-rate curves obtained for different air temperatures for both low pressure centrifugal fans, for the economical area of fan operation.

Efficiency of the low pressure centrifugal fans for different air temperatures

Numerical analysis of results enables the calculation of hydraulic efficiency for fan while operating in the considered range of volume flow-rates. The hydraulic efficiency curves obtained numerically for the low pressure centrifugal fan operating with different air temperatures are shown in fig. 8.

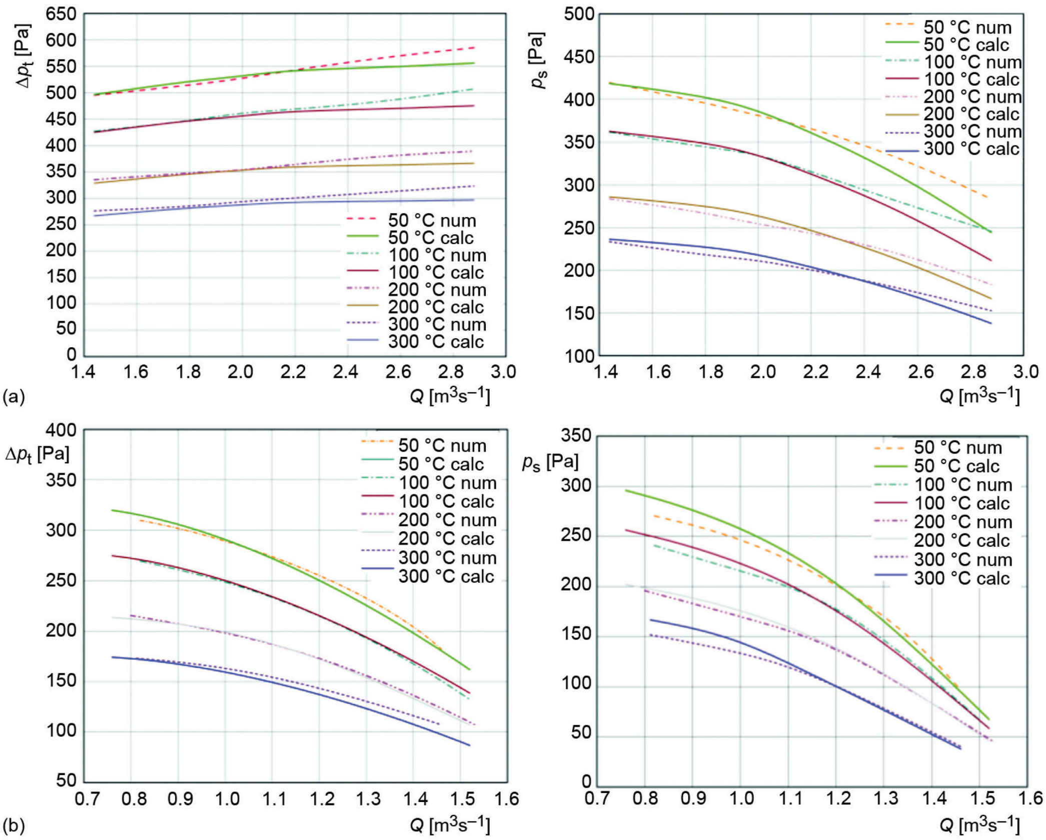

Figure 6. Numerical results and recalculated pressure curves when fans operate with different air temperatures; (a) fan with forward curved blades and (b) fan with backward inclined blades

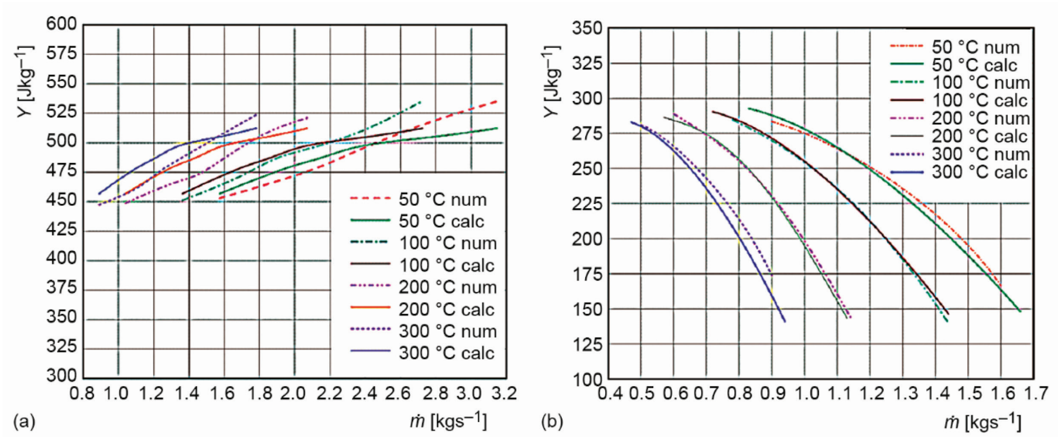

Figure 7. Specific energy mass-flow rate curves obtained numerically and by affinity laws for different air temperatures; (a) fan with forward curved blades and (b) fan with backward inclined blades

The best matching results of efficiency values are obtained for regimes close to the BEP. With an increase in air temperature, the relative difference in efficiency values in BEP is up to 3% for the fan with forward curved blades, and for a fan with backward inclined blades up to 4%. Therefore, when low pressure fans operate with different air temperatures, there is no significant efficiency drop, but it is noticeable that the increase in air temperature leads to a slight decrease in fan efficiency. It should be underlined that the change in air temperature affects efficiency less than the total numerical error. For different air temperatures, it is necessary to make certain corrections. Here we will refer to the previously given eqs. (13) and (14). Equations (13) and (14) estimate the efficiency of the centrifugal fan. Calculated efficiency values are shown in tab. 2 for the fan with forward curved blades and in tab. 3 for the fan with backward inclined blades.

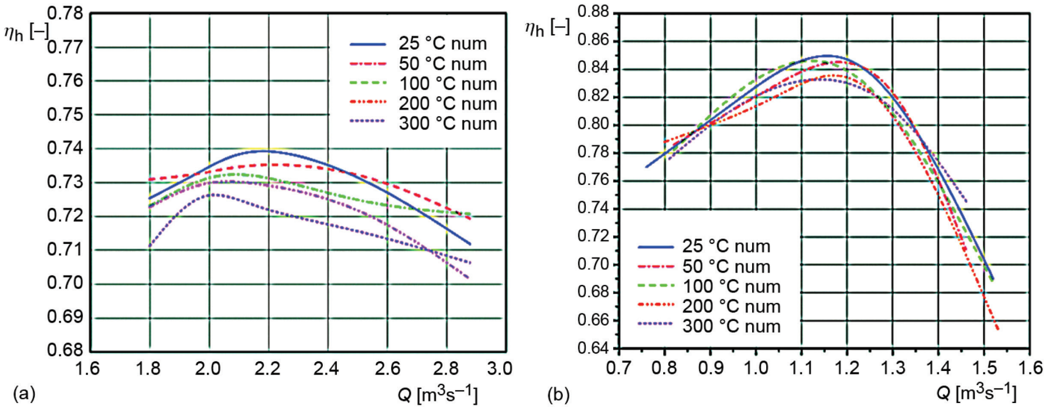

Figure 8. Hydraulic efficiency of centrifugal fans operating with different air temperatures; (a) fan with forward curved blades and (b) fan with backward inclined blades

Table 2. Efficiency correction for the different air temperatures for centrifugal fan with forward curved blades

t [°C]	Density [kgm^{-3}]	Kinematic viscosity [m^{2}s^{-1}]	Reynolds number [–]	Efficiency [–]	Efficiency corrected using eq. (13) [–]	Efficiency corrected using eq. (14) [–]
25	1.185	0.00001552	590528.4	0.742	–	–
50	1.093	0.00001788	512583.9	0.736	0.738	0.738
100	0.947	0.00002297	398998.7	0.733	0.731	0.731
200	0.745	0.00003447	265883.4	0.731	0.717	0.720
300	0.616	0.00004754	192785	0.730	0.705	0.710

Table 3. Efficiency correction for different air temperatures, for centrifugal fan with backward inclined blades

t [°C]	Density [kgm^{-3}]	Kinematic viscosity [m^{2}s^{-1}]	Reynolds number [–]	Efficiency [–]	Efficiency corrected using eq. (13) [–]	Efficiency corrected using eq. (14) [–]
25	1.185	0.00001552	837628,9	0.842	–	–
50	1.0932	0.00001788	727069,4	0.839	0.839	0.840
100	0.9467	0.00002297	565955,6	0.836	0.833	0.836
200	0.7466	0.00003447	377139,5	0.822	0.820	0.828
300	0.616	0.00004754	273453.9	0.830	0.808	0.824

As can be seen from tabs. 2 and 3, both eqs. (13) and (14), show good agreement with the values of efficiency obtained by numerical simulation, for the BEP and for all temperatures considered in this study. For temperatures 50 °C and 100 °C, for both fans, matching the obtained efficiencies is up to 0.4%, while for the higher temperatures (200 °C, 300 °C) obtained efficiency values differ more, but still only up to 3.4%.

Acoustic characteristics of centrifugal fans

There are three types of fan noise [17]: aerodynamic, electromagnetic, and mechanical noise. Interaction of air-flow with the fan runner and other fan parts create air pulsations, which produce aerodynamic noise. Depending on the cause, aerodynamic noise can be [1]: vortex shedding noise, noise of boundary-layers, non-stationary flow noise, turbulent flow noise, and noise of unstable fan operation. Aerodynamic noise cannot be reduced by simple construction measures, and therefore must be prevented or reduced at the design stage, by designing runner blades and other fan parts that would create less noise during fan operation.

Fan noise issue needs to be treated firstly theoretically and experimentally, and nowadays using CFD technics. Some numerical simulations software, like ANSYS CFX provides good noise analysis, and this has been used in this fan investigation.

The intensity of sound waves, I_{zv}, is the power of sound waves per unit area, which is normal to their propagation, $I_{zv} = N_{zv}/A$. The human ear registers sound waves from $I_z = 10^{-12}$ W/m^2 (on the threshold of audibility) to 10 W/m^2 (at the limit of pain). The sound power level (SPL) [dB], is:

$$L_N = 10\log\frac{N_{zv}}{N_{zv.0}} = 10\log N_{zv} + 120 \ [\text{dB}] \tag{16}$$

where the power of the sound wave on the threshold of audibility $N_{zv.0} = 10^{-12}$ W.

The total sound power level for multiple sound sources can be calculated as the sum of sound power levels:

$$L_{N.\Sigma} = 10\log\left(\sum_{i=1}^{n}\frac{N_{zv.i}}{N_{zv.0}}\right) = 10\log\left(\sum_{i=1}^{n}10^{0,1\,L_{N,i}}\right) \tag{17}$$

where there are i-sound sources (i = 1,2, ..., n).

Aerodynamic and acoustic curves of low pressure centrifugal fans are given in figs. 2 and 3.

On the other hand, for the numerical simulation analysis, the Praudman noise source model [18, 19] was used, to calculate and predict acoustic power per unit volume and determine the noise of turbulent flow. This model is incorporated in ANSYS 19.0 RANS simulations that we used. The Praudman magnitude can be calculated using:

$$AP = \alpha\rho_o\frac{u^3}{l}\frac{u^5}{a_o^5} \tag{18}$$

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

There are several equations in literature for estimation of sound, power level of fans. ASHRAE Handbook [15] proposes:

$$L_N = K_N + 10\log_{10} Q + 20\log_{10} p_t + BFI + C_N \tag{19}$$

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

Well-known web site www.engineeringtoolbox.com provides an expression for calculating the sound power level using the operating parameters of the fan, such as volume flow-rate and static pressure of the fan:

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

In the recognized engineering handbook [20] Recknagel, Sprenger, and Honmann suggested three equations for determining the sound power level of centrifugal fans:

$$L_N = 25 \pm 4 + 10\log_{10} Q + 20\log_{10} p_s \tag{21}$$

$$L_N = 80 + 10\log_{10} p_s \tag{22}$$

$$L_N = 77 + 10\log_{10} N + 20\log_{10} p_s \tag{23}$$

where Q [m^3h^{-1}] is the volumetric flow air of the fan, p_s [mmH_2O] – the static pressure of fan, and N [kW] – the power of the fan.

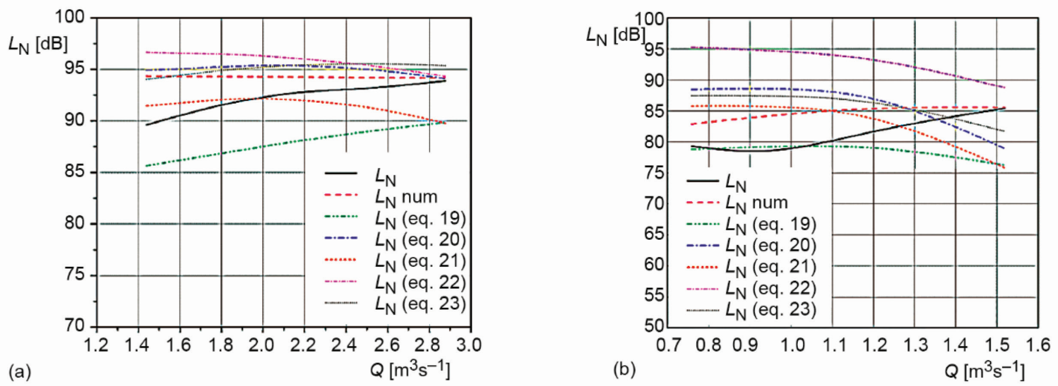

Figure 9. The sound power level of the centrifugal fan, obtained by different methods: numerical, experimental, and using eqs. (19) and (20); (a) fan with forward curved blades and (b) fan with backward inclined blades

A comparison of the results obtained by numerical simulations, the measured values of sound power level and using proposed equations is shown in fig. 9. Numerical results for the fan with forward curved blades obtained mean relative deviation of sound power level of only 2.2%, and the maximal deviation of values is around 5%, for the lowest volume flow-rate value. For this type of fan, the eqs. (21) and (20) obtained better agreement, while other equations do not suit the measured sound power level curve. Numerical results of the fan with backward inclined blades also have relatively good matching with measured curve, obtaining maximal deviation of results around 7%. From the equations presented in the paper, the best matching shows eq. (19), also eq. (21). In both cases, eq. (22) performed the worst, especially for smaller volume flow-rates.

Numerical results for the second case of low pressured centrifugal fan with backward inclined blades, show only a 1.2% deviation of sound power level in the BEP, while there is 8.7% deviation for the larger values of flow-rates. Numerical results give us a good agreement with eq. (19), while eq. (20) is useful only for higher flow-rates.

There is not much data on sound power level values of fans that operate with higher air temperatures than standard. Numerical simulation of flow and ANSYS software enables us to obtain values of sound power levels and overall sound power levels of fan operating with different air temperature, fig. 10. The overall sound power levels of the centrifugal fan with forward curved blades slightly decrease with increasing air temperature. The differences are small, up to 1.5% for the centrifugal fan with forward curved blade, and up to 6% for the centrifugal fan with backward inclined blades.

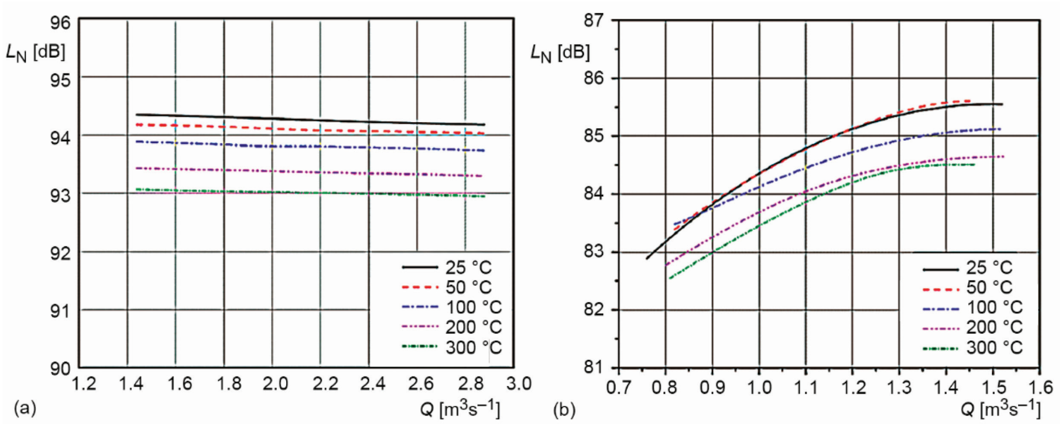

Figure 10. Overall sound power level of the centrifugal fan; (a) with forward curved blades and (b) fan with backward inclined blades, operating with different air temperatures

An important data, which is obtained using numerical flow simulations, is the change of sound power level with frequency. The dependence of sound power level on frequency, for the BEP for several air temperatures, is shown in fig. 11.

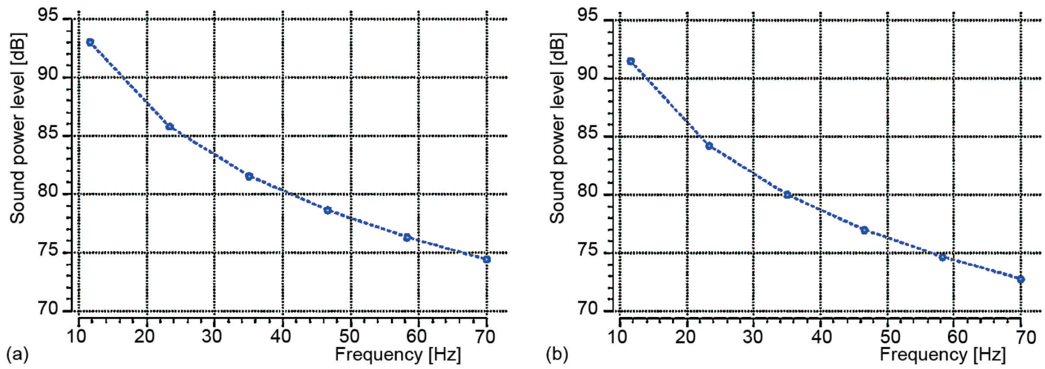

Figure 11. Sound power level for different frequency (BEP of the fan with forward curved blades); (a) 25 °C and (b) 300 °C

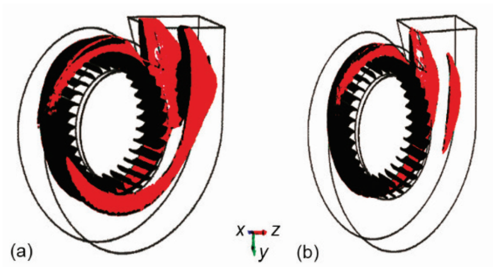

Figure 12. Isosurface at 80% of quadrupole source strength; (a) air temperature 25 °C and (b) 300 °C

Visualization of fan acoustic parameters

The ANSYS CFX POST allows the visualization of performance and acoustic parameters in the centrifugal fan numerical domain. The ANSYS CFX POST also provides the possibility of visualizing the acoustic operating parameters, according to different acoustic sources, such as monopole, dipole and quadrupole noise sources. Proudman sound power evaluated on the entire flow domain enables visualization of isosurface which are responsible for the noise generation. Figure 12 shows the Proundman sound power level [dB], in the plane of the impeller backplate and in the impeller blades, for the case of centrifugal fan with forward curved blades and temperatures 25 °C and 300 °C.

Conclusions

Operating and acoustic characteristics of low pressure centrifugal fans with forward curved blades and backward inclined blades were investigated in the paper. Numerical results show good agreement with the results obtained experimentally [11]. The results for total pressure vary up to 4% and static pressure up to 6%, for both centrifugal fans operating with air temperature 25 °C. When centrifugal fans operate with different air temperatures (50 °C, 100 °C, 200 °C, and 300 °C) pressure curves drop with increasing air temperatures. Total and static pressure curves, which are obtained numerically, differ from operating curves obtained by calculation, using affinity laws. The fan with forward curved blades shows greater deviation of obtained results for higher flow-rates, while the fan with backward inclined blades shows larger deviation of obtained results for lower flow-rates. The paper shows significant differences in the obtained calculated values of the pressure curves, compared to the numerical results, especially in the area of smaller flows, where Reynolds criterium for using affinity low can be compromised. The hydraulic efficiency of both low pressure centrifugal fans slightly decreases with increasing air temperature (up to 3%). Such a low drop in efficiency at higher temperatures indicate that it can be assumed that the change in air temperature affects hydraulic efficiency less than numerical uncertainties.

Acoustic characteristics of low pressure centrifugal fans are investigated primarily using sound power level values, obtained numerically as overall sound power level of fan. The difference between numerical and experimental values are up to 5% for the centrifugal fan with forward curved blade, while for the centrifugal fan with backward inclined blades up to 7%. There are no significant changes in sound power level values when centrifugal low pressure fans operate with higher air temperatures, but it is noticeable that with increasing the air temperature, the sound power level decreases. Numerical simulation using ANSYS CFX software allows the visualization of aerodynamic and acoustic parameters, making it easier to identify the problematic area in the fan domain. In conducted numerical simulations it is noticeable that, besides the fan runner, the shape of the volute casing, particularly the tongue, has a significant role in generating fan noise.

There are a few empirical equations, eqs. (19)-(23), which are recommended in various literature for the estimation of sound power level of fans. Therefore, obtained results are compared with numerical results, and the conclusion is that not all the proposed equations are equally good for evaluating the sound power levels of every kind of centrifugal fan. Generally, for different centrifugal fans, for both analyzed fans, eq. (21) shows the smallest deviations compared to the measured, although it does not follow the trend of the curve in the entire volume flow range. Equation (21) shows good agreement for the centrifugal fan with forward curved blade, while eq. (19) shows even better agreement for the centrifugal fan with backward inclined blades.

Further research directions of low pressure centrifugal fans are certainly in increasing the fan efficiency and the noise reduction by optimizing the geometry of the fan. Besides the geometry of the fan runner, as a crucial part of the fan, special attention should also be paid to the shape of the volute casing, particularly the tongue.

Nomenclature

D – diameter, [m]
c – velocity, [ms^{-1}]
c_s – speed of the sound, [m/s]
L_N – sound power level, (SPL) [dB]
$L_{N,\Sigma}$ – total sound power level, [dB]
I_{zv} – intensity of sound waves, [Wm^{-2}]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
Ma – mach number, (= c/c_s), [–]
n – rotational speed, [s^{-1}, min^{-1} = rpm]
N – power of the fan, [W].
N_k – power of the fan runner, [W]
p – pressure, [Pa]
Δp_k – pressure of the fan runner, [Pa]
Q – volume flow-rate, [m^3s^{-1}]
Re_u – Reynolds number, (= uD/ν)
t – temperature [°C]
u – circular velocity, $u = D\pi n$ [ms^{-1}]

Greek symbols

β – coefficient that takes into account gas compressibility [–]
ρ – density, [kgm^{-3}]
ν – kinematic viscosity, [m^2s^{-1}]
η – efficiency, [–]

Subscripts

h – hydraulic
m – mechanical
Q – volumetric

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