EXPERIMENTAL RESEARCH OF NOISE REDUCTION POSSIBILITIES OF AN AXIAL FAN WITH AN ADDITIONAL CONTRA-ROTOR

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ABSTRACT

This work refers to investigations of the noise propagation of a fan with an inlet contra-rotating rotor. The main goal was to find if there were regions of rotational frequency where the noise levels emitted by the system may be lower than expected. The measurements were performed at Lodz University of Technology in the aeroacoustic anechoic chamber. Constant rotational speed of the first rod-rotor was set-up as constant with varying rotational speed of the second rotor (the fan). The system was similar to one of the NASA earlier experiments. The analysis showed correlations between the emitted noise levels and the rotational speeds of the rotors. Next, the noise emitted only by the second rotor (the fan) was compared with the noise of two co-operating rotors. Some regions, where the noise of the fan was reduced due to the presence of the second contra-rotating rotor at the inlet, gave a base for future investigations. The issue is important especially in cases where quiet electric drives can be used and the aerodynamic noise is crucial. The presented results can be used in similar systems in modern flow machinery, aeroengines configurations, double-rotor drones, as well as, due to the simplicity of the system and its geometry, for verification of numerical methods.

KEYWORDS

FAN, CONTRA-ROTATING ROTORS, NOISE REDUCTION

NOMENCLATURE

BPF	[-]	blade passing frequency = $8 \cdot n$
c	[m/s]	flow velocity
c_0	[m/s]	flow velocity for $n = n_0$
L	[dB]	sound pressure level
n	[Hz]	fan rotational speed
n_0	[Hz]	rod-rotor rotational speed
p_{ref}	[Pa]	ref. sound pressure: 20 µPa
P	[W]	power of the drive
P_0	[W]	power of the fan drive for $n = n_0$
RPF	[-]	rod passing frequency = $3 \cdot n_0$
Tu	[-]	turbulence level
Tu_0	[-]	turbulence level for $n = n_0$
Δp	[Pa]	dynamic pressure fluctuation
Δp_0	[Pa]	dyn. press. fluctuation for $n = n_0$

INTRODUCTION

Searching for lowering noise levels and higher efficiency (lower fuel or electricity consumption) leads to the evolution of modern flow machines. To meet these requirements, a lot of experimental researches were performed, among others, at the Institute of Turbomachinery, Lodz University of Technology (Smolny et al., 1996-2011; Krysinski et al., 2000-2005; Blaszczak et al., 2001-2012; Liskiewicz et al., 2014; Kabalyk et al., 2015; Kosek et al., 2016). Additionally, all over the world new trends were introduced. Flow channels, machines geometries (especially those of vanes and blades), and flow conducting solutions started to be redesigned since new machines should be as compact as possible, be characterized by low energy losses and the use of new materials and technologies is involved (Saren et al., 2000; Rzadkowski et al., 2003; Liptak, 2009; Balicki et al., 2010; Kang, 2014; Swirydczuk, 2015). To solve these problems rarely completely new unconventional methods are proposed, basing mostly on numerical methods (Marczyk, 1999; Sondak, Dorney, 2000; Bennett, 2005; Button, 2015; Manoha, 2017), sometimes also using neural networks (Gluch et al., 2012). Anyway, they should be always completed and verified basing on detailed measurement sessions; this also applies to cases of improvements to previous construction of flow machines (Karczewski et al., 2008; Sobczak et al., 2008).

Important role in such researches plays unsteadiness of the flow at the inlet to a flow machine. It can be due to the inlet channel conditions, interaction with the preceding stages or other, sometimes unpredictable, influences. It can be also due to machine operating conditions, which can vary during exploitation. Controlling unsteadiness of the inlet flow often is also one of the ways to reduce vibrations and emitted noise by a machine (Huff, 2004; Blaszczak, 2005-2012; Watts et al., 2016). This problem is especially important for fans (Fahy, 2001; Crocker et al., 2007).

The presented study is a continuation of similar researches performed earlier in the Institute of Turbomachinery at the Mechanical Faculty of Lodz University of Technology (Poland) (for example: Krysinski et al., 1998-2003; Smolny, Blaszczak, 1997). This paper shows the analysis and discussion of the results of flow noise of an axial fan. The main assumption was to check if some particular conditions of the inlet flow may cause a situation when two contra-rotating impellers generate less noise than expected.

Phenomena associated to unsteady flows are often used in many aeronautical solutions, among them for example fans, drones (Heliara et al., 2016) or in aircraft (Heidmann, Lucci, 1997; Envia, 2010) and helicopter aeroengines (Phelps, Mineck, 1978; Greenwood et al., 2016) to improve their work. Possibility to influence on the inlet flow unsteadiness gives additional solutions to increase efficiency and thus to reduce the amount of energy used by a machine. Herein, a method of using an additional contra-rotating rotor, which gave possibility to interact with the main flow, is important. In this case, a fan was tested for different rotational speeds. From the point of view of noise emission, this solution (adding an additional rotor in front of the main rotor) may seem to be rather controversial but results in some frequency regions seem to be interesting. Additionally, it is a good simulation of classical solutions of two coaxial contra-rotating rotors (used earlier also by NASA: Heidmann et al., 1997b), especially when very simple geometries of rotors are used and such cases can be a base for numerical simulations.

Aside of obvious advantages of a system with two contra-rotating rotors, there are also some disadvantages, among them the fact that their use may not be profitable. The increase of complexities of machine structures (due to the use of individual drives) and increases of the weight and production costs, in many cases outweigh the advantages in terms of the efficiency growth and economy of the supplied energy. Anyway, the biggest issue is the emitted noise, so it is therefore important to perform many researches to check main parameters of such flows and to test the systems in order to ensure maximum use of its advantages and minimum of its disadvantages. They are subjects of some other experiments performed in the aeroacoustic anechoic chamber designed by the author (Blaszczak, 2005b) in the Aeroacoustic Laboratory of the Institute of Turbomachinery (Lodz UT). The presented researches focused mainly on possibilities of the noise reduction of the presented two-rotor system.

RESEARCH DESCRIPTION

The researches were carried on the test stand, which was prepared especially for this purpose (Figure 1). It was designed basing on one of the NASA experiments, where the first rotor was simulated by rotor with rods (Heidmann, 1997b). The main test object was an axial fan with eight straight blades. The height of every blade was 120 mm, the width was 53 mm, and the thickness was 5 mm. The diameter of the hub was also 120 mm.

Unsteadiness of the inlet flow was simulated by additional inlet coaxial contra-rotating rotor with a hub of the same diameter (120 mm) equipped with three cylindrical rods of diameters equal to 3 mm and the height of the rods was 120 mm (Figure 1*a*). The distance between the trailing edge plane of the rods and the leading edge of the fan blades was 49.9 mm (the ratio of the distance to the width of the blade = 0.94).

The rotor with cylindrical rods was chosen basing on some earlier tests (Wojciechowski, Blaszczak, 2011; Rydlewicz, Blaszczak, 2016) and numerical simulations (Sobczak, Blaszczak, 2010). Rod-rotor simulated the unsteady inlet flow by changing the relative ratio of the rotational speeds of both rotors. The drives of both rotors were controlled by independent inverters placed in the control room (aside of the anechoic chamber) to avoid electromagnetic disturbances. The schematic presentation of the system is shown in Figure 1*b*. The rotor system was covered with a narrow casing to protect the microphones against the effects of possible radial flow phenomena, which can disturbed the acoustic measurements and lead to pseudo-noise signals (Jacob, 2017ab).

The acoustic measurements were performed using three microphones placed at the distance of 1 m from the object, according to standards and other experiments (Roger, 2017). Basing on earlier measurements sessions, two main microphones (M1 and M3) were in the plane, which was perpendicular to the main axis of the flow system, and at its centre (Fig. 1c). The results obtained from both microphones were compared to check the symmetry of the sound field around the object. The last microphone was used as the reference one to compare the changes of the acoustical environment conditions during all measurement sessions. Before every measurement session all microphones were calibrated. The accuracy of the measured noise levels was estimated as 0.5 dB (the calibrator uncertainty was 0.3 dB). The acoustical signals were filtered to the acoustic range (high-pass analogue filter 20 Hz and low-pass analogue filter 20 kHz).

The flow velocity and its unsteadiness was measured using a hot-wire thermoanemometric probe connected to the constant temperature anemometer (CTA). It was tuned before every measurement session to avoid electromagnetic oscillations, by using an oscilloscope. The probe was placed 10 cm behind the fan rotor, in the middle of its blade height. The temperature of the probe was setup as 160^0C (the ambient temperature in the chamber was constant during all sessions and about 19^0C). The mean accuracy of the flow velocity measurements was about 0.05 m/s depending on the velocity value. During the presented sessions, the Reynolds numbers (based on the blade cord = blade width) were in the range $2.17 \cdot 10^4$ to $3.31 \cdot 10^4$. Mach numbers were smaller then 0.3 (about 0.1) so the flow was treated as incompressible. Additionally, the overall unsteadiness of the flow was also measured (with reference to the atmospheric pressure) at the distance of 1 m behind the fan using a dynamic pressure transducer to check overall outlet flow conditions.

The power of the electric drives was measured using two watt-meters connected into the Aaron's system (inside the control room). The accuracy of the measurement method used to determine the shaft power was 2 W. The adjustments of rotation speeds of both rotors were independent and changed by separate electronic control systems using electrical converters.

All measurement channels were connected through 16-bit/1 MHz signal analogue-to-digital converter to especially equipped computer in the control room (Blaszczak, 2005b, 2009).

The contra-rotating system was placed inside the aeroacoustic anechoic chamber (outer volume 582 m^3, inner volume 386.5 m^3, inner acoustic surface 368 m^2) at the Institute of Turbomachinery of Lodz University of Technology. The results of measurements of its acoustic characteristics can be found in earlier publications (Blaszczak et al., 2005a; Blaszczak, 2016).

a) b) c)

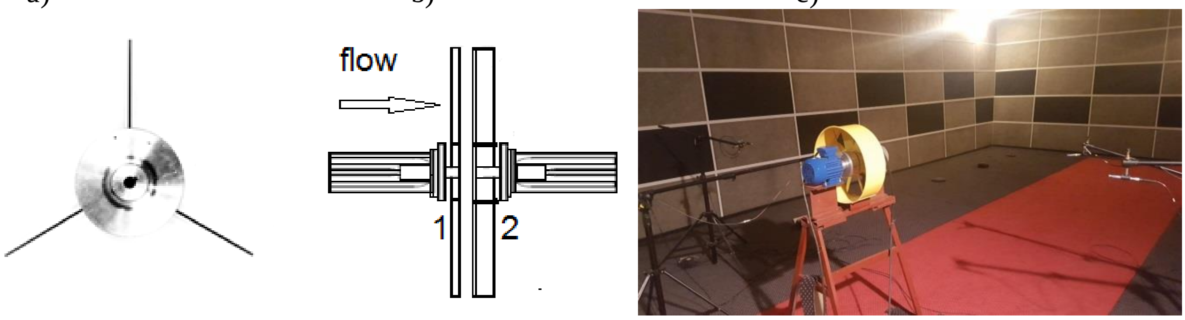

Figure 1. Test rig presentation: a) the three-rod inlet rotor; b) test rig schema: 1 – three-rod inlet rotor, 2 – eight-blade fan; c) the test rig inside the anechoic room

There were three measurement sessions. The first session included the run of both rotors (the fan and the inlet rotor). The second one was the case when only the fan was working. In the third session only the inlet rotor was tested. During the tests the inlet rotor (with cylindrical rods) was working with constant rotational speed. The rotational speed of the fan was regulated in the range below its nominal value to simulate higher load conditions. Before and after every measurement sessions the overall background signals levels were checked. For all measurement channels the sampling frequencies were chosen as 50 kHz (according to the Nyquist–Shannon sampling theorem) with time windows of 10 s (i.e. 250 000 samples from each of the measurement channels for every operating point of the fan and the rod-rotor).

RESULTS

The obtained results are presented as relative changes of the measured values for easier comparison with other data sets.

The power of the fan for both cases (with and without the inlet turbulent flow; see Figure 2) was tested in the same range of rotational speed of the fan. Here the dimensionless relative changes are presented with the reference to the value P_0 obtained for the fan rotational velocity equals to the rod-rotor rotational velocity n_0. The obtained results are similar; to compare, Figure 2 shows also both cases. It can be seen that some points are aside the trend lines (in accordance to theoretical considerations), however no significant power deviations were observed in this case of study. The results are also similar to the expected ones except some ranges where small power drops were observed. These regions were taken under more careful consideration and then checked for correlations with other flow parameters.

Comparing the graphs of the supplied power in function of the speed of rotation, it can be seen that, over whole rotational velocity range, that the power measured during operation of the two rotors is slightly greater than the power supplied during operation of the fan only. At low speeds, the differences are rather small. Within the range of the higher rotational speeds values almost coincide one with another. However, as the speed grows this difference also increases and the power increase is larger at the highest rotational speeds. Anyway, in some areas of the rotational speed range, the decrease in power to lower levels was observed, even slightly lower than the values when only the fan was running. Similar results were observed during researches on the same test-rig for different rod-to-blade ratios and other distances.

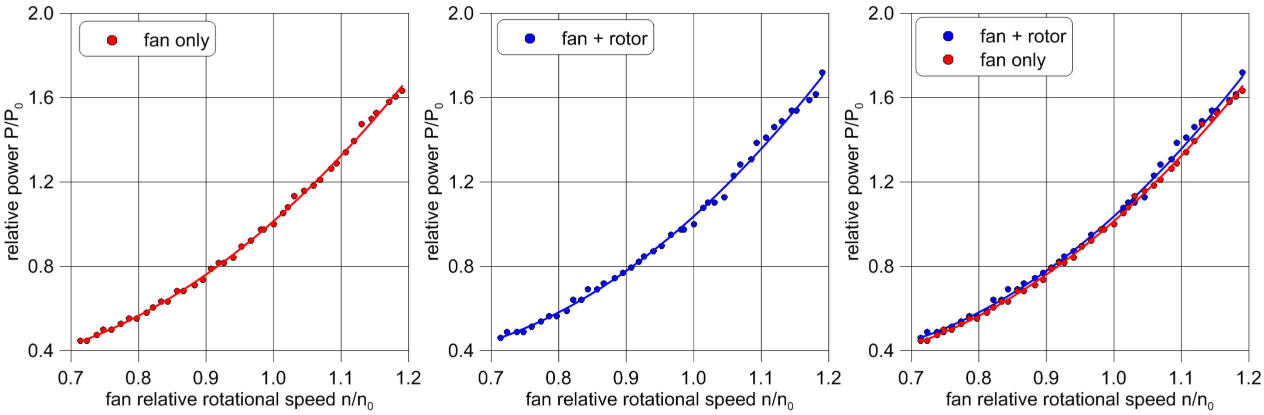

Figure 2. Comparison of individual power supplies of the fan and the rotor in function of the relative rotational speed of the fan (with and without the inlet rotor)

In Figure 3 and Figure 4 the values of the measured noise levels are presented. To show it in the logarithmic scale (in dB), according to the standards, the reference level was taken as 20 µPa. The last plots in Figure 3 and Figure 4 show comparison of noise levels in function of the rotational speed of the fan with and without the inlet rotor. The presented values are overall SPLs calculated over the entire acoustic frequency range (20 Hz up to 20 kHz).

It can be noticed that in some frequency ranges the measured noise levels deviate from the trend line and there are areas with increased and reduced noise levels. The spread of data around the trend line for the expected sound pressure level was less for the case when two rotors were contra-rotating. This means that the presence of the preceding rotor stabilized the work of the entire system. Areas deviating from the noise trend line, including its decrease, were observed. The fact that there were such areas (with a reduced level of sound emission) means the presence of working zones, in which the noise level can be reduced. More detailed analysis showed that the blade- and rod passing frequencies dominated in both cases. Tonal noises related to the Strouhal number frequencies (i.e. frequencies of the flow wakes oscillations) were of less importance. It was expected and in accordance with other researches performed for different two-rotors configurations (for example Kennedy et al., 2013, Rydlewicz et al., 2016).

The relative changes of the mean outflow velocities, turbulence levels, and dynamic pressure differences (in relation to the outlet static pressure, i.e. the atmospheric pressure) at the exit of the fan are shown in Figures 5 and 6. In Figure 5 they are presented only for the case when the fan was working and in Figure 6 for the case with the inlet contra-rotating rod-rotor. It can be observed that even for the case when only the fan was running (the case without any inflow unsteadiness) there are visible changes of the flow parameters with increasing fan rotational speed. Anyway it can be seen another evidence that presence of the inlet rotor somehow stabilizes the outlet parameters. It can be also observed that with higher values of the rotational velocity, the overall turbulence level in proximity of the fan (10 cm) is going down (also due to the higher flow velocity), to lower values, much quicker with the presence of the inlet rod-rotor. The dynamic pressure fluctuations (Figure 5 and Figure 6), measured at the distance of 1 m behind the fan exit plane, were significantly growing with higher rotational velocities and it seems that they were not strongly influenced by the inlet unsteadiness of the flow. Some zones of lower pulsations levels were also observed. Completely different image at different distances behind the fan mostly can be explained by development of the vortex structures behind the second rotor and lower flow velocities due to the big outer volume of the anechoic chamber. Anyway, lower unsteadiness directly at the exit plane of the fan looks promising since it is related to lower flow energy losses.

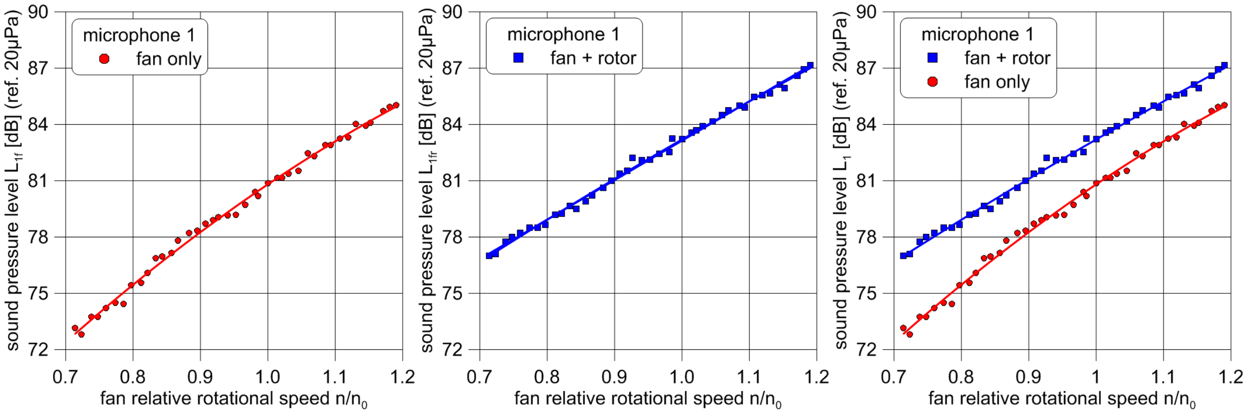

Figure 3. Microphone No. 1: comparison of noise levels (ref. 20 µPa) of the fan and the rotor in function of the relative rotational speed of the fan with and without the inlet rotor

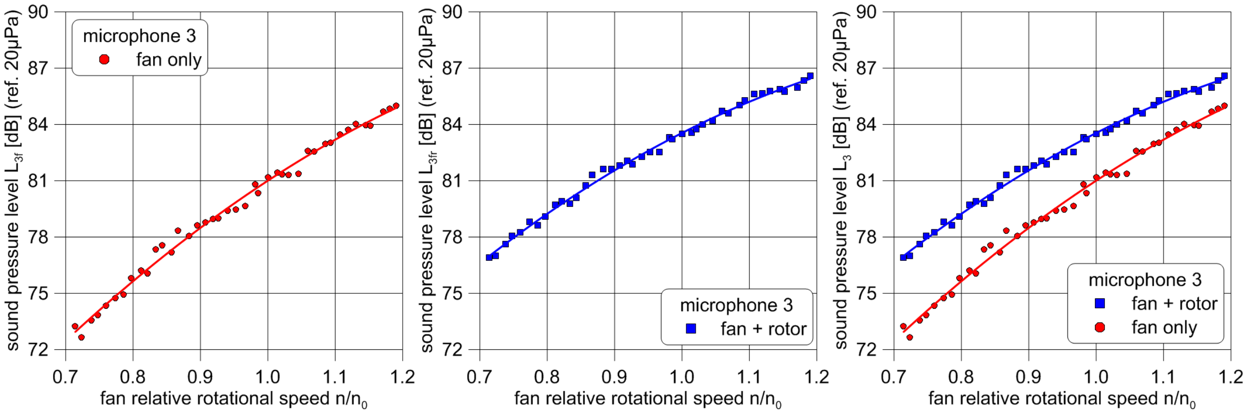

Figure 4. Microphone No. 3: comparison of noise levels (ref. 20 µPa) of the fan and the rotor in function of the relative rotational speed of the fan with and without the inlet rotor

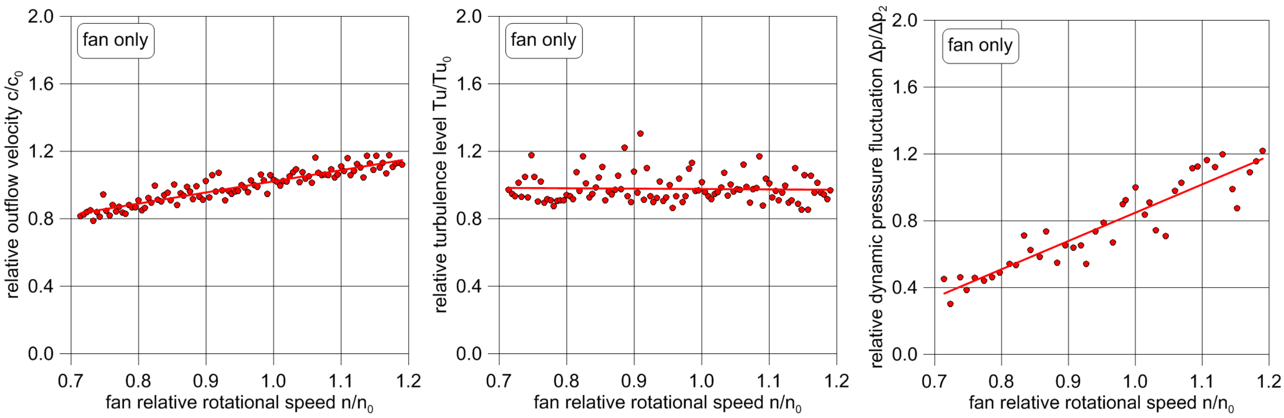

Figure 5. Changes of the relative values of the outflow velocity, turbulence levels, and dynamic pressure fluctuations in function of the relative rotational speed for the case without turbulent inlet flow

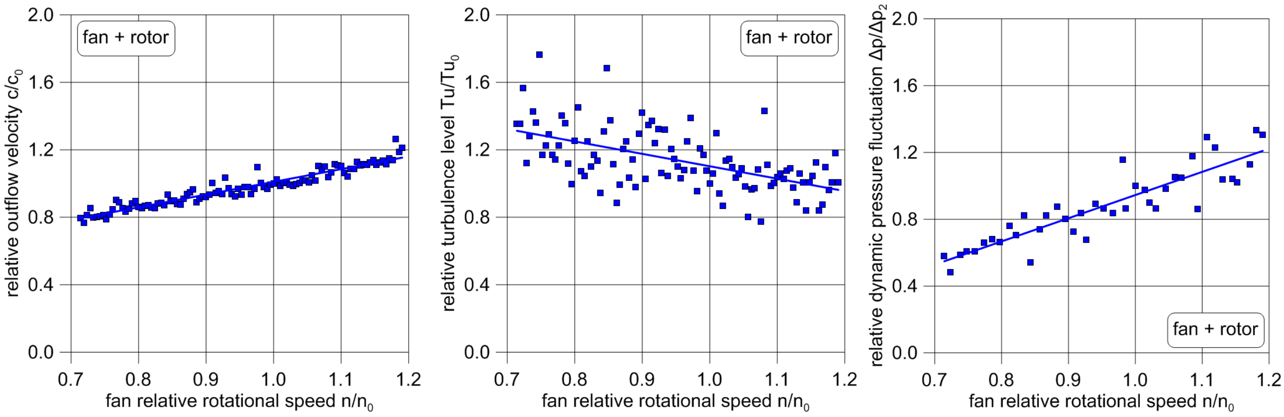

Figure 6. Changes of the relative values of the outflow velocity, turbulence levels, and dynamic pressure fluctuations in function of the relative rotational speed for the case with turbulent inlet flow

Plots from Figure 7 to Figures 12 show examples of the frequency analysis of the emitted noise for the same rotational frequency of the fan with- (Figures 9 and 12) and without the rod-rotor (Figure 8 and 11). For comparison, the noise emitted by the rod-rotor alone is also shown in the same scale (Figure 7 and 10).

In the Figures 7, 8, and 9 sound pressure levels are shown. The figures 10, 11, and 12 additionally present relative acoustic pressure (also with reference to the threshold of hearing, i.e. 20 μPa) because the peaks are better visible in such a manner. For comparison reasons, the plots are presented in the same ranges.

It can be seen that tonal noises connected to the blade passing frequencies are important. Rod passing frequency and its harmonics (Figure 7 and 10), when compared to the levels of the blade passing frequencies (Figure 8 and 11), are almost invisible, however the presence of the inlet rod-rotor influenced strongly the emitted fan noise during cooperation of two rotors (Figure 9 and 12). The noise related to the main blade passing frequency of the fan was lower, its harmonics vanished, but additional components of the noise occurred. They were due to the ratios of the rotational velocities of both rotors according to well known equations (Fahy, 2001; Crocker et al., 2007).

After having compared the charts of the presented measurement sessions it can be seen that there are some interesting regions of the working zones of the fan, which should be analysed more carefully. According to this, further, more detailed studies of the discussed flow phenomena are envisaged.

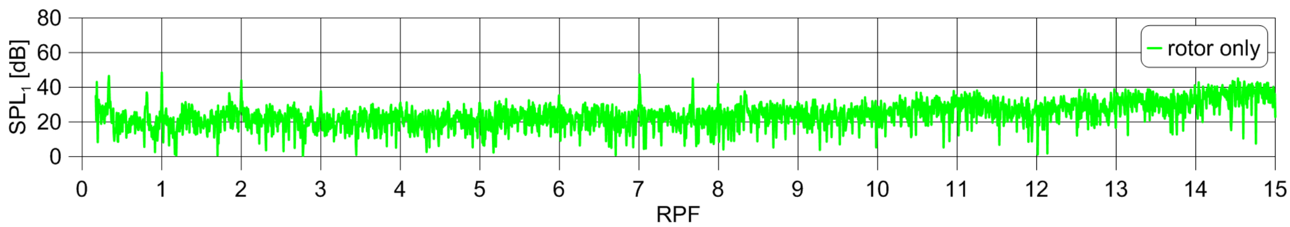

Figure 7. Sound pressure level (SPL, p_{ref} = $2\cdot10^{-5}$ Pa) in function of the rod passing frequency (RPF) for the rod-rotor only (no flow)

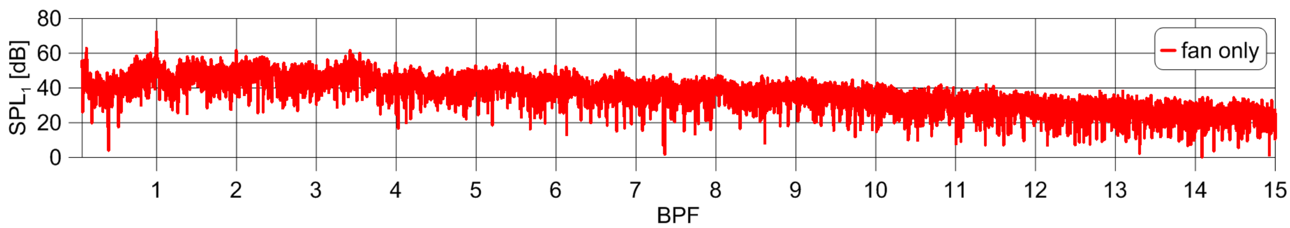

Figure 8. Sound pressure level (SPL, p_{ref} = $2\cdot10^{-5}$ Pa) in function of the blade passing frequency (BPF) for the fan only (without turbulent in-flow)

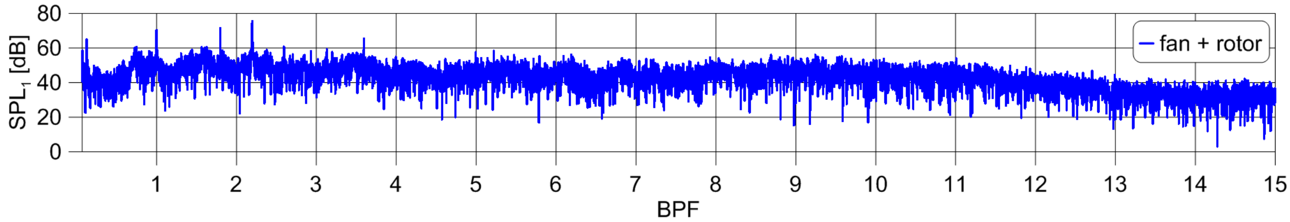

Figure 9. Sound pressure level (SPL, p_{ref} = 2·10^{-5} Pa) in function of the blade passing frequency (BPF) for the fan and rotor (with turbulent in-flow)

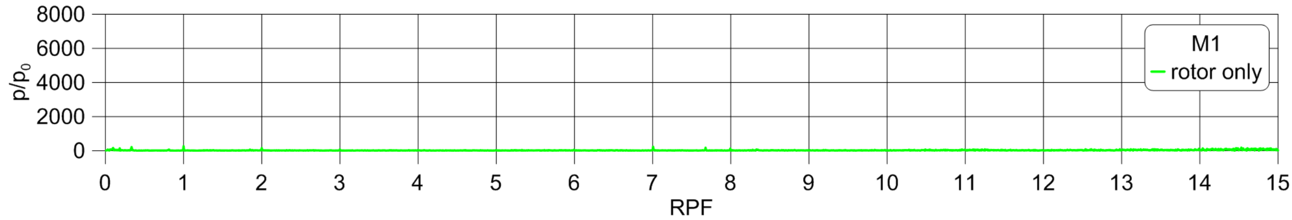

Figure 10. Relative acoustic pressure (p_{ref} = 2·10^{-5} Pa) in function of the rod passing frequency (RPF) for the rod-rotor only (no flow)

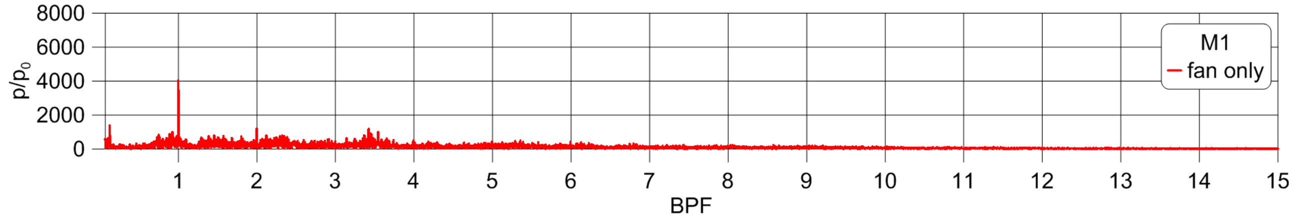

Figure 11. Relative acoustic pressure (p_{ref} = 2·10^{-5} Pa) in function of the blade passing frequency (BPF) for the fan only (without turbulent in-flow)

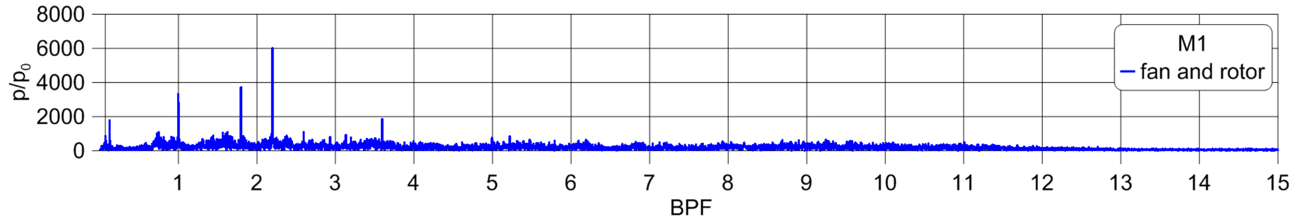

Figure 12. Relative acoustic pressure (p_{ref} = 2·10^{-5} Pa) in function of the blade passing frequency (BPF) for the fan and rotor (with turbulent in-flow)

CONCLUSIONS

In the presented studies the influence of the preceding rod-rotor on the fan performance is presented. Additionally, emitted noise levels were also compared, what is with accordance of modern challenges in technology (Directive, 2002; Rolt and Whurr, 2015; Travis, 2015; Button, 2017a].

It was found out that there are zones inside the fan operating range in which there are possibilities to improve the machine parameters and reduce the emitted noise only by controlling the relative rotational speeds of both rotors. It can lead to decrease of the supplied power, lower noise emission and improvement of the flow parameters. The presented results show possibility of tuning such a contra-rotating system of two rotors by rather simple method. This can be a base of the future analysis of such issues, not only cases of fans, but also, for example, modern drones with double rotors systems, especially in smart cities, where, in quiet zones of residence, fast delivery of small packages soon will become the norm (Button, 2017b; Zazulia, 2018). Additionally, the simplicity of the presented test rig gives possibilities to use the presented results as validations of numerical methods for other researchers. This is due to the continuous need of their verifications from the experimental work (Lyrintzis, 2003; Jaron et al., 2015; Greenwood et al., 2016).

New measurement sessions, also with completely open-rotor system cases, are planned.

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