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Influence of Geometry of Channel on the Flow Noise Parameters

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Acoustic emission through duct walls is an important problem in engineering acoustics.This phenomenon most commonly occurs in heating, ventilating and air-conditioning (HVAC) and other gas flow ducting (large industrial silencers). Many works focus on elaboration of more exact description of the acoustic field phenomena reflecting the real conditions in which these appliances operate. As a standard, circle or rectangular ducts are used in ventilation systems. However, technical conditions during the installation of the HVAC system, due to the limitation of the assembly space, require often the use of channels with other geometries. This paper presents aeroacoustical parameters of three most common cross-sectional shapes of air-moving ductwork. The rectangular, square with roundedcorners and circular ducts were studied. The "natural" duct attenuation, which is a consequence of duct shape or noise breakout and involves a diminution of the internally propagated sound power was observed. Natural duct attenuation can be a useful way of reducing sound power levels in long runs of duct.

Keywords: HVAC, noise emission, shape duct, insertion loss.

1. Introduction

The flow in long ducts is a basic problem in fluid mechanics. Liquid or gas flow through ducts is commonly used in heating and cooling applications and fluid distribution net-works. The fluid in such applications is usually forced to flow by a fan or pump through a flow section. Currently most commonly described and used are phenomenons which occurs in circular pipes. The theory of fluid flow is reasonably well understood but theoretical solutions are obtained only for a few simple cases such as fully developed laminar flow in a circular pipe [1].

Fluid flow is classified as external and internal, depending on whether the fluid is forced to flow over a surface or in a conduit. Internal and external flows exhibit very different characteristics. The fluid velocity in a pipe changes from zero at the surface because of the no-slip condition to a maximum at the pipe center. In case of the fluid flow, it is convenient to work with an average velocity, which remains constant in incompressible flow when the cross-sectional area of the pipe is constant. The average velocity in heating and cooling applications may change somewhat because of changes in density with temperature. During flow of fluids the noise is also generated.

There are three issues associated with the sound propagation in ducts that affect methodology of measurements, the characteristics of acoustical energy in ducts, end reflections, and turbulence in ducts. The first issue of acoustical characteristics depends on the dimensions of the duct and the frequencies being measured. At lower frequencies with large wavelengths, only plane waves propagate in a duct and a simple relationship can be shown between sound pressure and sound power. At high frequencies with shorter wavelengths, plane modes and higher order modes can exist. This means that sound is propagating not only parallel to the axis of the duct but also in various angles due to reflections from the wall of the duct. These modes cause variations in the sound pressure level at particular locations in a cross-sectional area of the duct. The second issue when taking in-duct measurements is end reflection factors due to duct termination. An opening at the discharge of a duct can create end reflections that send a sound wave back into the duct against the airflow because of an impedance mismatch. The reflections can cause interference and generate standing waves that further complicate the patterns of sound energy being transmitted through each element of the duct system. Such standing waves in the duct can cause inaccuracies of measurements of noise in duct.The third issue with in-duct measurements is turbulence caused by the movement of air in the duct. Turbulence can be caused by obstructions to the flow and other changes in pressure. The resulting turbulent eddies have flow that may not be parallel to the axis of the duct. The turbulent fluctuations in pressure can not be differentiated by a microphone measuring the pressure changes associated with acoustical energy. These pressure fluctuations affect random frequencies of measurements taken in such a condition [2].

Turbulence and the vortices, which are formed during the fluid flow, are the cause of the flow noise. In literature this issue is known as aerodynamic noise, as well as aeroacoustics. The turbulence is usually formed by interaction of fluid motion with walls of duct or by the instability of free shear layers separating a high speed flow from a stationary environment. According to studies of the mechanisms of aerodynamic sound there are two principal source types in free vortical flows: a quadrupole, whose strength is determined by the unsteady Reynolds stress and a dipole, which is important when mean mass density variations occur within the source region.The theory of aerodynamic sound was developed by Lighthill (1), who reformulated the Navier-Stokes equation into an exact, inhomogeneous, differential, wave equation which is valid only within the turbulent region [3].He found an exact analogy between the production of sound by turbulence in a fluid whose mean pressure (p_0), density (ρ_0) and sound speed (c_0) are at large distances from the source flow, and that produced in an ideal, stationary acoustic medium forced by the stress distribution definied by Lighthill stress tensor T_{ij}(2), associated with Reynolds stress ($\rho\nu_i\nu_j$) and viscous stress (σ_{ij}), according to forms [4]:

$$\left(\frac{1}{c_0^2}\frac{\partial^2}{\partial t^2} - \nabla^2\right)\left[c_0^2\left(\rho - \rho_0\right)\right] = \frac{\partial^2 T_{ij}}{\partial x_i \partial x_j} \tag{1}$$

$$T_{ij} = \rho\nu_i\nu_j + \left(\left(p - p_0\right) - c_0^2\left(\rho - \rho_0\right)\right)\delta_{ij} - \sigma_{ij} \tag{2}$$

Most unsteady flows of technological interest are of high Reynolds number and turbulent. The acoustic radiation is a by-product of these motion. The turbulence is usually produced by fluid motion relative to solid boundaries or by the instability of free shear layers separating a high speed flow from a stationary environment. The study of flow duct noise problems includes:

- source identification and characterization - sound is generated by the fan, so-called fan noise;

- acoustic energy propagation along flow ducts, including the effect of discontinuities and various forms of nonlinear behavior - sound is generated by the flow along variable cross area of duct (e.g. circle, square, elliptic), bending or expanding parts of the duct;

- sound radiation from flow intakes or discharges and the radiation of sound from the duct walls - sound is generated by the flow near openings and also through the walls and depends on the type of material.

These relationships between "noise problems" have an influence on the total noise generated during the flow. It determines the value of the sound power level of the duct system along with the additional equipment, that is important for designers of HVAC systems. There are experimental and theoretical works about acoustic power level generated by the air flow in duct systems from different material [5, 6, 7], for straight and bending duct with some openings [8, 9, 10], for duct with different cross section [11, 12].

The purpose of this article is to clarify the effect of the geometry of the ducts with different cross sections on the aerodynamic sound. We have performed experiments of aerodynamic sounds generated by the flow in the duct with the circle, square and square with rounded corners cross-section.

2. Experimental setup

The three type of ducts was use in these studies; rectangular, circle and oval (in corners were arcs with radius 100mm) – Table 1. The ducts had the same cross section area. All channels were 1500mm long and made of 3mm sheet metal (type St3). Ducts have been sanded and varnished with three layer of paint.

The experiments were performed on the specific test stand constructed according to two standard PN-EN 3741:2011 "*Determination of sound power levels of noise sources using sound pressure–Precision methods for reverberation rooms*" [13] and PN-EN 7235:2009 "*Acoustics. Laboratory measurement procedures for ducted silencers and air-terminal units. Insertion loss, flow noise and total pressure loss*" [14] – Fig. 1.

Table 1 Geometry and characteristics parameters of studied ducts

Type of duct	Geometry parameters [mm]	Crosssectional area [m^2]	Length [mm]	Thickness [mm]
rectangular	790×495	0,3911	1500	3
circle	Ø 706	0,3913	1500	3
oval	500×800 with r=100	0,3914	1500	3

Construction of the test stand allows to determine sound noise level of devices operating in the flow together with their attenuation effect by use of precise methods.

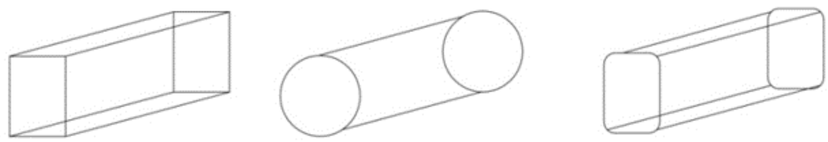

Figure 1 Geometry of studied ducts

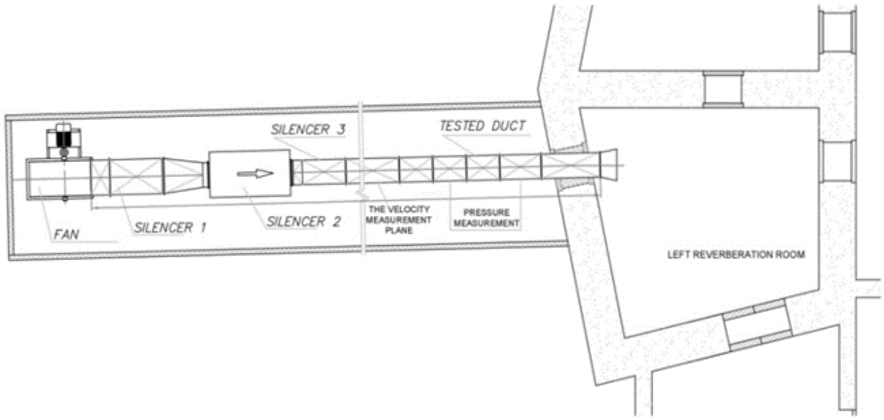

Figure 2 Construction of test stand with outflow to the reverberation room (according with PN-EN ISO 3741 and PN-EN ISO 7235)

In this method sound pressure level was measured in the reverberation and afterwards the sound power level of a noise source was determined.The reverberation chamber has got a volume of 237 m^3 and total restriction area of 231,5 m^2 with all out-of-parallel planes. The stand test is build of the following elements: radial fan, system of 3 absorption silencers, loudspeaker chamber, straight channel with a cross-section 700×700 and diffuser at the end with outlet cross-section 840×840. Total length of stand is 30 m.

A procedure for determination of a sound pressure level is measurement in reverberation chamber (outlet of channel), on the height 4m, by using rotating microphone boom Brűel & Kjear type 3923. Sound pressure levels are measured in nine discrete microphone positions with an integration time of 30 s for each microphone position and are measured in 1/3 octave bands 100-10000 Hz. The B&K 1 microphone, model no. 4146 was used and data were collected using a two-channels B&K analyzer 2144. Microphone was calibrated before commencing the acoustic test.

For studied ducts insertion losses were designated in dependence on the flow speed of air. Flow velocity was determined using the so called arithmetical calculations method described in the norm PN-ISO 5221:1994 ,,Distribution and division of air – Measurement procedures for airflow in the duct". The mean velocity in the channel was determined using log-Chebyshev method. Acoustic measurements were taken at three flow velocities, $\nu_1 = 6$ m/s; $\nu_2 = 9$ m/s; $\nu_3 = 12$ m/s. Ambient pressure was measured by means of a pitot-static tube with a pressure difference converter. Also the static pressure and temperature in the ducts were measured. Measurement of these parameters were performed using pressure transducers, temperature and humidity sensors and recorded and processed by the data acquisition station - SAD-2, equipped with the ADAM modules 4000+ [15]. Readings were taken by PC computer using ADAMView programme with Visual Basic application connected with SAD-2 station.

3. Results and discussion

3.1. *Aeroacoustical parameters*

The measurement results are compared below in charts and tables. As was mentioned in the introduction, the intention of this study is term the acoustical parameters (sound power level) of studied ducts with different cross area.

The sound power level of the unit shall be calculated in each one-third octave frequency band from the sound pressure level measured in the reverberation room as described in the comparison method of ISO 3741. The measured sound pressure levels are compared with the sound pressure levels produced in the same room by a reference sound source of known sound power output.

Measurement and calculation procedures are given for both a direct method and a comparison method of determining the sound power level. Calculation of sound power from sound pressure measurements is based on the premise that, for a source emitting a given sound power in the reverberation test room, the mean-square sound pressure averaged in space and time is directly proportional to the sound power of the flow noise and depends only on the acoustical and geometric properties of the room and on the physical constants of air, according to formula (3):

$$L_w = \overline{L_p} + D_{td} + C \tag{3}$$

where:

D_{td} – transmission loss at the end of a duct attached to the reverberation room (the measuring stand test is connected to a reverberation room via a transmission duct;the open end of this transmission duct is located in a lateral wall of the reverberation room; diffuser at the end with outlet cross-section 840×840). For the test stand $D_{td} = 0$;

C – a difference in sound power levels radiated into the reverberation room and sound pressure averaged inside the reverberation room determined according to PN-EN ISO 3741 - direct method;

$\overline{L_p}$ – mean sound pressure level in third - octave bands, determined according to PN-EN ISO 3741 [13], but not corrected with regard to ambient noise, spatially averaged in measurement time (4):

$$\overline{L}_{p,eq,T} = 10 \times \log\left[\frac{1}{T}\int_0^T \frac{p^2(t)}{p_0^2}dt\right], dB \tag{4}$$

where in general the subscripts "eq" and "T" are omitted since time-averaged sound pressure levels are necessarily determined over a certain measurement time interval.

Figure 3 displays the sound power level with A-correction over a frequency range going from 100 Hz to 10000 Hz for studied ducts for two different flow velocities. Two conclusions can be drawn from these graphs. First we can see that the value of the sound power level increase with increasing air flow velocity. Second, there isn't significant differences between the spectra for studied ducts at 6 m/s flow velocity. At this velocity, spectra of sound power level are similar, independent on geometry of ducts. Only at 250 Hz, the sound power levels for a rectangular and oval duct are 2 dB higher than for a circular. The same single-number value of the sound power level with A-correction are also obtained (Table 2).

However, at higher airflow velocities, differences in the sound power spectra of the studied ducts were observed. As we see at Fig. 3, the higher level of acoustic power is obtained for circular duct in the frequency 315 Hz; 400 Hz and between 1000-10000 Hz than rectangular and oval ducts. As shown on this chart that the oval duct is characterized by the lowest (an average of about 1-1,5 dB) sound power level in the range frequencies between 100-200 Hz and 315-10000 Hz from all the studied ducts. Single-number value of the sound power level with A-correction for oval duct is 63,6 dB, and it is lower 1,1 dB from value for rectangular duct and 1,4 from value for circular duct. The 1/3 octave band frequency of noise power level are collected in Table 3. As can be seen from Table 3 for oval duct only in 250 Hz the sound power level is higher than in the same frequency for circular duct. It is not easy to explain.

Channel acoustics is inherently associated with flow in the channel. Flow in rectangular duct is characterized by the existence of secondary flows (Prandtl's flow of the second kind) which are driven by the turbulent motion. The problem of secondary flows has attracted much attention due to the significance in engineering practice. Secondary flows are a mean flow in the transverse plane superimposed upon the axial mean flow. They are generated and maintained by:

Table 2 Single-number value of the sound power level with A-correction dependent on the flow velocities

velocity [m/s]	rectangular	circle	oval
	L_{WA} [dB]	L_{WA} [dB]	L_{WA} [dB]
6	44,5	44,5	44,7
9	56,3	56,2	55,8
12	64,7	65,0	63,6

Table 3 The 1/3 values of sound power level with A-correction dependent on flow velocities for studied ducts

f [Hz]	Velocity in duct								
	v_3=12 m/s			v_2=9 m/s			v_1=6 m/s		
	rectang.	circle	oval	rectang.	circle	oval	rectang.	circle	oval
	LwA [dB]								
100	48,5	48,4	47,4	42,2	41,2	41,7	32,2	31,3	32,8
125	50,0	50,9	50,2	42,8	43,4	45,7	32,7	33,6	34,6
160	51,5	52,5	51,1	44,3	44,6	44,6	35,0	35,4	35,8
200	53,1	52,7	51,5	46,0	45,4	44,9	36,7	35,7	35,9
250	54,4	52,6	53,8	47,2	45,5	47,9	40,3	38,2	40,2
315	57,7	59,4	57	49,7	51,3	49,9	39,6	40,8	40,3
400	58,9	59,5	58,1	51,2	51,3	50,9	40,6	40,9	40,9
500	58,2	58,6	57,1	50,4	50,4	49,7	39,4	39,5	39,4
630	58,4	58,4	57,1	50,5	50,3	49,7	39,3	39,3	39,2
800	57,3	57,3	56,3	49,3	48,9	48,9	37,2	37,0	37,5
1000	54,5	54,7	53,3	46,6	46,4	45,9	33,8	33,7	33,9
1250	54,7	54,9	53,7	46,4	46,1	45,7	31,8	31,9	32,2
1600	53,0	53,3	51,8	43,3	43,1	42,6	27,7	27,9	28,0
2000	51,5	51,9	50,2	41,3	41,1	40,6	24,9	25,1	25,2
2500	48,8	49,2	47,6	38,3	38,0	37,6	21,4	22,0	21,7
3150	46,0	46,5	44,7	34,9	34,7	34,1	19,8	21,5	20,2
4000	44,1	44,5	42,8	32,8	32,6	32,2	21,1	21,8	22,0
5000	39,4	39,8	38,5	29,0	28,7	28,9	21,1	21,0	21,7
6300	36,6	37,0	35,8	28,8	27,8	27,9	22,2	22,4	22,7
8000	32,8	32,9	31,7	22,6	22,3	22,2	17,8	18,0	17,2
10000	29,2	29,5	27,9	20,8	21,1	20,2	17,4	17,6	16,5

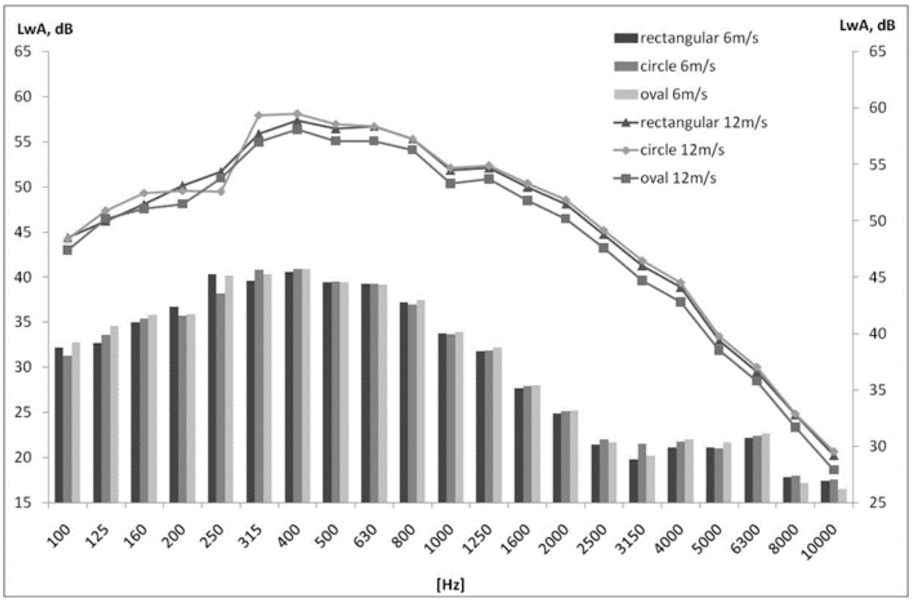

Figure 3 The 1/3 sound power level with A-correction of studied ducts at 6 m/s (bar graph) and at the 12 m/s (line graph)

- pressure driven and found in curved passages - this kind of secondary flow is dissipated within a straight circular duct;
- turbulence driven and found in non-circular straight ducts - this kind of secondary flow, encountered in non-circular ducts, is present also under fully developed conditions, and is caused by turbulence.

Prandtl formally separated these two categories into what is now know as secondary motions of Prandtl's first and second kind, respectively. The secondary flows are relatively weak, but its effect on the transport of momentum is quite significant [16, 17, 18]. Secondary flow can affect the acoustics of the duct. According to this theory, very long circular duct with laminar flow should be characterized by the lowest sound power levels what in this case is not completely unambiguous. Flow simulations by using spectral-element method (SEM) in a rectangular channel with different angles of rounding showed that rounding radius may prevent inhomogeneous interactions between vortex in corner because hinder interaction between turbulent ejections in the corner and leading to the lowest cross-flow [11, 12]. This effect may also affect the lower acoustic of the oval duct.

3.2. *Insertion loss*

Standard PN-EN ISO 7235 [14] allows determine the insertion loss D_e of the measurement object – duct, which is defined as dynamic insertion loss because depends on flow conditions. Insertion loss is the difference between the sound power or intensity levels measured in the same points of the duct work before and after the insertion of the studied object (for example silencers described in PN-EN ISO 7235 standard or other object, like studied ducts). The insertion loss depends on the flow velocity, so it's very important to determine the acoustic characteristics (attenuation) of the tested ducts. The insertion loss $D_e$in dB is determined according to the following equation:

$$D_e = L_{WII} - L_{WI} \tag{5}$$

where:

L_{WII} – sound power emitted into the connected reverberation room with inserted measuring object;

L_{WI} – sound power emitted into the connected reverberation room after replacing the measuring object with the substitution duct (in our studies substitution duct was the duct with cross area like stand test).

Table 4 Insertion loss D_e of studied ducts dependent on the flow velocities

f [Hz]	rectangular			circle			oval		
	12 m/s	9 m/s	6 m/s	12 m/s	9 m/s	6 m/s	12 m/s	9 m/s	6 m/s
100	-1,7	-1,4	-1,7	-1,1	-0,7	-0,6	-1,5	-1,8	-1,8
125	0,6	0,8	1,2	-1,0	-0,7	-0,5	-1,4	-0,8	-0,6
160	2,0	1,1	0,9	0,7	-0,3	-0,5	1,4	0,6	0,3
200	-0,2	-1,2	-1,3	0,2	-0,6	-0,6	0,1	-0,5	-0,6
250	0,3	-0,6	-0,4	-0,3	-0,8	-0,8	-0,1	-0,9	-0,7
315	0,1	-0,8	-0,8	-0,3	-1,0	-1,0	0,3	-0,4	-0,5
400	0,5	-0,1	0,0	-0,4	-0,7	-0,7	0,4	-0,2	0,1
500	0,1	0,1	0,0	1,1	1,2	1,2	0,3	0,2	0,3
630	-0,5	-0,6	-0,6	-0,2	-0,3	-0,3	0,6	0,4	0,5
800	0,4	0,2	0,4	0,6	0,6	0,7	0,5	0,3	0,4
1000	-0,2	0,4	0,4	-0,2	0,4	0,2	0,0	0,5	0,6
1250	-0,5	-0,7	-0,7	0,1	-0,1	-0,1	0,6	0,3	0,4
1600	0,2	-0,8	-1,0	0,6	-0,1	-0,3	1,0	-0,1	-0,1
2000	-0,3	0,2	0,1	-0,3	0,2	0,1	-0,7	-0,3	-0,3
2500	-0,2	-0,2	-0,1	0,0	-0,3	-0,3	0,2	0,1	0,2
3150	-0,8	-0,4	-0,4	-0,4	0,0	0,1	-0,6	-0,4	-0,1
4000	-0,6	0,1	0,0	-0,7	0,3	0,4	-0,7	-0,2	-0,1
5000	-0,4	-0,2	-0,2	-0,4	0,1	0,3	-0,2	-0,1	-0,1
6300	-1,2	-0,2	-0,6	-1,0	0,1	-0,1	-1,3	-0,4	-0,4
8000	-1,9	-0,3	-0,8	-1,6	0,2	-0,1	-2,1	-0,5	-0,7
10000	-2,0	0,0	0,1	-2,0	0,4	0,4	-2,6	-0,5	-0,3
Single-number insertion loss	**-0,2**	**-0,1**	**-0,2**	**0,0**	**0,2**	**0,2**	**0,1**	**0,1**	**0,2**

The insertion loss tells us about the damping of noise by tested object. If the tested object has got higher value of insertion loss it better muffles noise. As can be seen from Table 4, the damping effectiveness of the studied ducts is in range from 0,1 to 2 dB depending on the frequency. It is interesting that the oval channel has a positive insertion loss value in the range of 160 Hz; 400-1250 Hz; 2500 Hz independent on velocity of air and it is rather wide range of frequency. However for rectangular duct positive insertion loss is more scattered in the frequency range, as well as for the circular duct, for which they are additionally dependent on the velocity of the air through the channel. It suggests that oval ducts is acoustically more efficient than duct with another cross-sections, but of course this effect dependent on the quality of manufacture.

4. Conclusions

Engineers can minimize the sound in the HAVC system through good design practice and minimalization of occupied space by ducts. If there is possibility we should use long straight duct with the same cross-section, without any sharp edge and with special constructed mufflers behind the fan. The another solution is use flexible duct which is an excellent attenuation element. Sound will be reduced when appropriate fan speed controllers are used to reduce fan rpm rather than using mechanical devices to restrict airflow. HVAC systems operating at low supply air static pressure will also reduce the generated sound level. This will also provide more energy efficient operation and allow the central fan to be downsized. However, the same channels in the HVAC installation can already be noise suppression elements. For this purpose, their acoustic sound power level and insertion loss should be determined accordance with ISO standards.

In this work, the sound power level in 1/3 octave band frequency and as a single-number value of rectangular, round and oval channels was determined at 6 m/s, 9 m/s and 12 m/s. Additionally insertion loss of studied ducts was determined also in these velocities. The studies were done on the test stand constructed according to PN-EN 3741:2011 and PN-EN 7235:2009 standards, which describe the direct methods of acoustic measurements by using the reverberation chamber. As it turned out the duct with round corner has got lower value of sound power level than rest of ducts.Also the single-number value of insertion loss is positive for oval duct at all tested velocities. Our studies suggests that the oval duct is characterized by lower sound power levels and has got greater attenuation than a rectangular or even circular duct, which as a acoustic object seems the most suitable for use. For rectangular duct turbulent flow along streamwise corners is characterized by the appearance of secondary motions of Prandtl's second kind may be also responsible for it acoustical parameters. The secondary flow refers to the in-plane mean cross-flow perpendicular to the streamwise direction. These motions could have a significant influence on the turbulence and acoustic of rectangular duct.

Conducting additional flow tests, such as observation of the flow in the studied duct with a fast camera - for example observations of air flow with smoke in the duct towards transverse direction (through the transparent wall) and in the opposite direction to the flow (at the end of duct) – should be made. It will allows to describe and characterize the types of generated flows, which also influence their acoustics.

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