Boško Rašuo
Full-Professor
University of Belgrade
Faculty of Mechanical Engineering

Mirko Dinulović
Full-Professor
University of Belgrade
Faculty of Mechanical Engineering

Marta Trninić
Senior Lecturer
The Academy of Applied Technical Studies, Belgrade, Department Belgrade Polytechnic

Marina Stamenović
Professor of vocational studies
The Academy of Applied Technical Studies, Belgrade, Department Belgrade Polytechnic

Novak Milošević
Assistant
The Academy of Applied Technical Studies, Belgrade, Department Belgrade Polytechnic

Nebojša Ćurčić
Assistant
The Academy of Applied Technical Studies, Belgrade, Department Belgrade Polytechnic

A Study of Aerodynamic Noise in Air Duct Systems with Turning Vanes

Noise is becoming one of the problems in many industrial applications. In this paper the aerodynamic noise in air duct system that arises from air flow passing over a surface of turning vanes is investigated. Based on Proudman's formula using Lighthill's acoustic analogy broadband acoustic noise model can be predicted. To model the turbulent flow in an air duct $k-\varepsilon$ *turbulent model is used and required constants are obtained experimentally in a low speed wind tunnel followed by noise measurement in the vicinity of deflected airfoil. Several designs are investigated: Inner airfoil in the duct elbow, center positioned air foil, outer positioned and the overall combination of all previous cases. It was found that in order to satisfy all noise requirements and regulation, which are becoming more strict nowadays, the acoustic analysis and design must be performed in most industrial systems since the noise levels arising from the operating industrial equipment may represent occupational and health hazard.*

Keywords: *Aerodynamic noise, airfil vanes, turbulent flow.*

1. INTRODUCTION

The accurate prediction of aerodynamically generated noise in air duct systems is necessary for noise control in industrial ducts, such as HVAC systems or pipelines. The presence of singularities in air-flow ducts, such as sharp duct bends, orifices and diaphragms on one end and the presence of different types of obstacles in the duct such as flaps, splits and turning vanes on the other, results in unsteady aerodynamic flow generating propagating sound waves along the duct [1]. Acoustics analysis is related to fluid dynamics. Sounds that are technologically important in industrial applications are generated by and propagated in fluid flows. With great confidence the phenomena associated with sounds can be analyzed within the field of fluid dynamics and the governing equations for acoustics are the same as the ones governing fluid flows.

In a duct without turning vanes, in order to change the air flow to the desired direction, the impact of the working fluid must be absorbed by the duct walls. In ducts with turning vanes, the airflow direction change is gradual, consequently inducing less impact and less force on the duct walls. Turning vanes surfaces add a small amount of friction, and they can be one of the noise sources generated from the airflow passing over the turning vane active surface [2], [3].

In the present paper the aerodynamically generated noise in air duct systems with turning vanes is investigated for low Mach numbers. Noise levels for different duct configurations (position of the vanes, number of the vanes and inlet velocities) are estimated using numerical approach based on Broadband acoustic model and Lighthill's analogy. In this research a k - ε sound model is used to approximate the noise generated by turbulent flow past turning vanes in an air duct with addition of Proudman's analytical expressions to approximate the sound intensity of isotropic turbulence for low Mach numbers in terms of the kinetic energy of the turbulence and its dissipation rate.

In recent years, the regulation of noise generation in the vicinity of operating machinery is becoming increasingly stringent. Year on year the subject of noise reduction is becoming more pertinent, both from the public and from the manufacturer's point of view. Noise has been related to multiple effects on human health and secondary to noise pollution [4].

Furthermore, there are many standards regulating acceptable noise levels within different working areas and describing basic methods for determining sound power levels.

Some of the current standards related to noise regulations are: ISO 3741 to ISO 3747 (sound power level determination using sound pressure level measurements), ISO 9614-1 to ISO 9614-3 (sound power level determination using sound intensity measurements), ISO/TS 7849-1 and ISO/TS 7849-2 (sound power level determination using vibration measurements).

These standards specify methods for determining the sound power level and the accuracy, characterized by the reproducibility of the method, system design and mounting conditions, and the uncertainty associated with previously mentioned conditions. The standards mentioned above differ in their range of applications and their requirements with regard to the test

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Correspondence to: Dr Marta Trninć, Academy of Technical Applied Studies Belgrade Department Belgrade Politechnic, Katarine Ambrozić 3, 11120 Belgrade, Serbia. E-mail: mtrninc@politehnika.edu.rs
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environment since some machinery (or components) equipment and products emit high-frequency noise, which can be broad-band noise, narrow-band noise or discrete tones.

As with any occupational hazard, control technology should aim at reducing noise to acceptable levels by action on the work environment. Such action involves the implementation of any measure that will reduce noise being generated, and/or will reduce the noise transmission through the air or through the structure of the workplace. Such measures include modifications of the machinery, the workplace operations, and the layout of the workroom. In fact, the best approach for noise hazard control in the work environment, is to eliminate or reduce the hazard at its source of generation, either by direct action on the source or by its confinement.

Prior to the selection and design of control measures, noise sources must be identified and the noise produced must be carefully evaluated.

To adequately define the noise problem and set a good basis for the control strategy, the following factors should be considered:

- type of noise
- noise levels and temporal pattern
- frequency distribution
- noise sources (location, power, directivity)
- noise propagation pathways, through air or through structure
- room acoustics (reverberation)

Any noise problem may be described in terms of a source, a transmission path and a receiver. The noise control may take the form of altering any one or all of these elements. The noise source is where the vibratory mechanical energy originates, as a result of a physical phenomenon, such as mechanical shock, impacts, friction or turbulent airflow.

2. WIND TUNNEL BROADBAND ACOUSTIC MODEL

Broadband noise can be considered as a type of noise that is composed of a wide range of frequencies. Furthermore, this type of noise is nonperiodic and random, generally caused by turbulent flow passing over the vanes in an air duct.

In many practical engineering applications involving turbulent flows, noise does not have any distinct tones. The sound energy is generally distributed over a broad range of frequencies. In these circumstances, statistical turbulence quantities computed from CFD RANS equations can be utilized and Lighthill's acoustic analogy.

The mathematical model of Aerodynamic sound produced by interaction of a low Mach number fluid flow with fixed turning vanes inserted in a rectangular duct is governed by mass conservation and Navier-Stokes momentum equations [5].

$$\begin{aligned} &\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_i}\left(\rho u_i\right)=0 \\ &\frac{\partial}{\partial t}\left(\rho u_i\right)+\frac{\partial}{\partial x_i}\left(\rho u_i u_j+p_{ij}\right)=0 \end{aligned} \tag{1}$$

In previous equation u_i and u_j are the velocity components and p_{ij} is the stress tensor expressed as:

$$p_{ij}=-\sigma_{ij}+\delta_{ij}p \tag{2}$$

Where p is the static pressure of the flow filed, δ_{ij} the Kronecker delta and σ_{ij} is viscosity stress given in the following relation:

$$\sigma_{ij}=\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}-\frac{2}{3}\left(\frac{\partial u_j}{\partial x_i}\right)\cdot\delta_{ij}\right) \tag{3}$$

Equations of sound propagation are derived by use of the above mass and momentum conservation equations as:

$$\begin{aligned} &\frac{\partial^2 \rho}{\partial t^2}-a_0^2\nabla^2\rho=\frac{\partial^2}{\partial x_i \partial x_j}\cdot T_{ij} \\ &T_{ij}=\rho\cdot u_i u_j+p_{ij}-a_0^2\rho\delta_{ij} \end{aligned} \tag{4}$$

Previous equation is a hyperbolic partial differential equation. This equation describes a wave propagating at the speed of sound a_0 in a medium at rest, on which externally applied fluctuating forces are given in the form described by the right-hand side of equation (equation 4). Based on this mathematical model sound is generated by the fluid flow's fluctuating internal stresses acting on an acoustic medium at rest, and propagated at the speed of sound [6]. Using Lighthill's analogy approach it is possible to separate the analysis of aerodynamic acoustics into two steps:

- The first step is sound generation induced by fluid flow in any real continuous medium.
- The second step is sound propagation in an acoustic medium at rest, exerted by external fluctuating sources which are a function of Lighthill (turbulence) stress tensor T_{ij}, known from the first step.

The equation of sound propagation has been solved by Lighthill [7] and Curle [8] in the form:

$$\begin{aligned} &\rho'(\mathrm{x,t})=\rho(\mathrm{x,t})-\rho_0= \\ &=\frac{1}{4\pi a_0^2}\cdot\frac{\partial^2}{\partial x_i \partial x_j}\cdot\int_V\frac{T_{ij}(\mathrm{y,t}-\frac{R}{a_0})}{R}\cdot dV(y)+ \\ &+\frac{1}{4\pi a_0^2}\cdot\frac{\partial^2}{\partial x_i}\cdot\int_S\frac{l_{ij}\cdot \mathrm{p}_{ij}(\mathrm{y,t}-\frac{R}{a_0})}{R}\cdot dS(y) \end{aligned} \tag{5}$$

The acoustic observation point is denoted with x and y is the point in the flow field where sound is generated. The distance between the acoustic observation point and the point in the flow field $R=abs(x-y)$ is where sound is generated. L_j is the unit direction vector of the solid boundary, pointing toward the fluid, at time t (observation time).

In a flow filed, fluctuating stresses and the acoustic density oscillation are related in previous equation. This can be considered as conversion from the kinetic energy of fluctuating shearing motion in the flow to the acoustic energy of oscillations.

The previous equation can be used in situations when the sound is radiated into free space, the sound induced by fluid flow is weak and the fluid flow is not sensitive to the sound induced by the fluid flow.

Lighthill's acoustic analogy is therefore successfully applicable to the analysis of energy generated from subsonic flows as sound, and not to the analysis of the change in character of generated sound which is often observed in transitions to supersonic flow due to high frequency emission associated with shock waves.

There are several source models that enable you to quantify the local contribution (volume or per unit surface area) to the total acoustic power generated by the flow. Known source models are: Proudman's formula, jet noise source model, boundary layer noise source model, source terms in the linearized Euler equations and source terms in Lilley's equation.

Based on the work of Lighthill, Proudman derived analytical expressions to approximate the sound intensity of isotropic turbulence for low Mach numbers in terms of the kinetic energy of the turbulence and its dissipation rate. Based on Proudman’s analogy, the flow induced acoustic intensity can be expressed in terms of standard steady-state variables corresponding to the turbulent kinetic energy k, and dissipation rate ε, which are conventionally used in CFD. The advantage of this approach is that the flow induced noise can be approximated from steady-state CFD data and hence methods based on Proudman’s analogy are much more computationally efficient than those based on Lighthill’s analogy. The computational efficiency of flow noise predictions obtained using Proudman’s analogy makes the method extremely attractive for comparative design studies.

Proudman [9] using Lighthill's acoustic analogy, derived a formula for acoustic power generated by isotropic turbulence without mean flow. More recently, Lilley [10] rederived the formula by accounting for the retarded time difference which was neglected in Proudman's original. Both derivations yield acoustic power due to the unit volume of isotropic turbulence (in W/m^3) as

$$p_A = \alpha \rho_0 \left(\frac{u^3}{l}\right) \cdot \left(\frac{u^5}{a_0^5}\right) \tag{3}$$

where u and l are the turbulence velocity and length scales, respectively, and a_0 is the speed of sound. In Equation 3, α is a model constant expressed in terms of k and ε. Previous equation can be rewritten as

$$\begin{aligned} p_A &= \alpha_\varepsilon \cdot \rho_0 \varepsilon M_t^5 \\ M_t &= \frac{\sqrt{2k}}{a_0} \end{aligned} \tag{3}$$

The rescaled constant α_ε is initially set to 0.1 based on the calibration of Sarkar and Hussaini [11] using direct numerical simulation of isotropic turbulence.

The acoustic power can be expressed in dB, which is computed from

$$L_P = 10\log\left(\frac{P_A}{P_{ref}}\right) \tag{3}$$

where P_{ref} is the reference acoustic power (P_{ref}=10^{-12} W/m^3).

The Proudman's formula gives an approximate measure of the local contribution to total acoustic power per unit volume in a given turbulence field. Proper caution, however, should be taken when interpreting the results in view of the assumptions made in the derivation, such as high Reynolds number, small Mach number, isotropy of turbulence, and zero mean motion.

3. MODEL CALIBRATION

One of the main challenges in numerically predicting sound waves arises from the fact that sounds have much lower energy than fluid flows. This represents a great challenge to sounds computation in terms of numerically resolving sound waves, especially when one is interested in predicting sound propagation to the far field. Another challenge is in prediction of the flow characteristic in the near field that is responsible for sound generation.

To obtain the parameters for the surface acoustic sources in the proposed k - ε sound model, calibration studies are performed, using low Mach, electrically powered wind tunnel. Initially, turning vanes, in the form of NACA 2412 airfoil (constant cross-section) were manufactured using 3D printing technology. The airfoil vanes were manufacture using polylactide material (PLA), which is thermoplastic polyester (Figure 1).

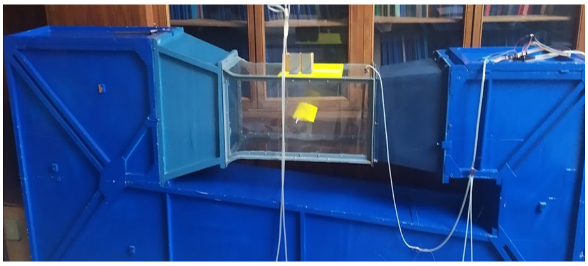


Figure 1. Wind tunnel, experimental set-up

For different inlet velocities (ranging from 5 m/s to 25 m/s with 2.5 m/s steps) static pressure at the wind tunnel outlet working area was measured using DSP pressure sensor. At the inlet the velocity was measured using standard pitot tube. After the data analysis the following constants for turbulent k - ε sound model were obtained (Table 1.) [12]:

Table 1. Empirical constants in the standard *k-ε* model

C_μ	$C_{1\varepsilon}$	$C_{2\varepsilon}$	P_{rk}	$P_{r\varepsilon}$	P_{rt}	S_{ct}
0.09	1.44	1.92	1	1.3	0.85	0.7

The experiment was performed in a subsonic wind tunnel (closed type), with electrical power motor, capable of generating an air flow up to 30 m/s in working section of the wind tunnel. At the inlet, the pitot tube was installed and noise measurement was performed for velocities in the range of 5 – 25 m/s for different positions (angles of attack) of air foil vane. The pressure was constantly monitored at the outlet using dsp pressure sensor. All sensors were connected to data acquisition unit. The acoustic power level was measured in the vicinity of deflected vane, which was located in the center of working wind tunnel area. The cross-sectional area, where touring van was located was d x h = 400 x 450 [mm]. The acoustic power level was measured using Minilyzer ML1 - Handheld Audio Analyzer which was initially calibrated using Tecpel 336A sound calibrator (Figure 2).

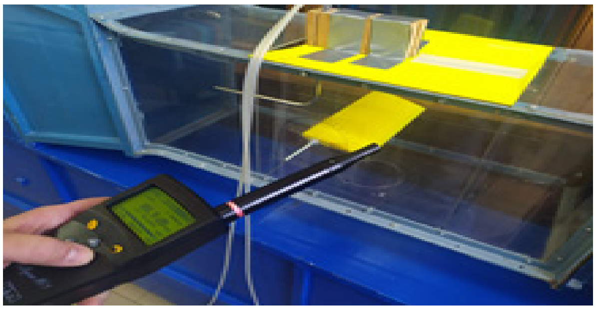


Figure 2. Broadband noise measurement

Based on experimental data, results were compared to the acoustic model generated for the inlet velocities in the range of 5-25 m/s and angles of attack of -20^0 to 20^0 in respect to incoming air flow. Acoustic power level, expressed in decibels (dB) for the angle of attack of 20^0 for the inlet velocity of 10 m/s is presented in the following figure (Figure 3).

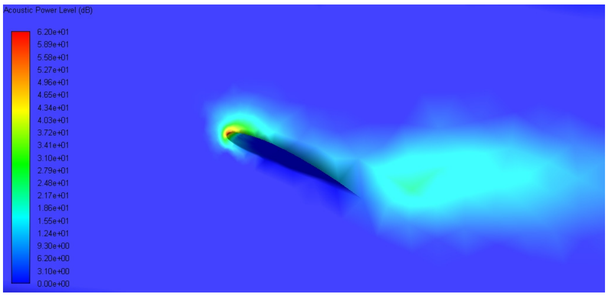


Figure 3. Deflected Airfoil vane, noise generation (10 m/s)

4. NUMERICAL ANALYSIS

Turning vanes are proven very valuable for reducing pressure losses, vibrations and increasing system efficiencies. In air duct design where airflow must change direction (mostly due to geometrical constraints) without turning vanes, the duct walls must absorb the sudden impact of the working fluid in order to reorient the airflow to the desired direction.

In the case of rectangular 90° duct elbows, a common rule of thumb for best performance and economy of manufacturing is the elbow having a center line radius equal to 1 1/2 times the duct width in the plane of the bend. When space does not permit this ratio, the radius is reduced and, conversely dynamic losses are expected to increase. To improve elbow performance thin partions called splitters or turning vanes can be installed in the elbow [13].

Turning vanes assist the airflow in making a smoother and more gradual change in direction, resulting in lesser impact, and consequently lesser force is transferred. Turning vane surfaces do add a small amount of friction, the amount of energy lost to friction from the vanes is much smaller compared to the energy lost in the impact resulting from the airflow taking change in direction.

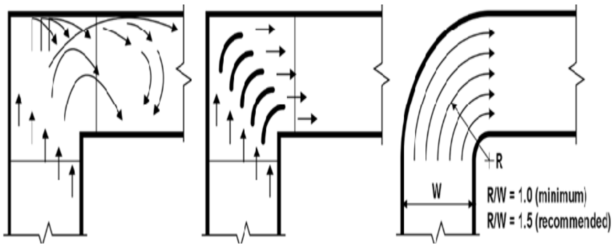


Figure 4. Flow pattern; a) 90° rectangular elbow, b) 90° rectangular elbow with turning vanes, c) 90° elbow with a moderate radius

Computation domain, used for all CFD and acoustic analyses is presented in Figure 5. As in the experimental part, Cylindrical axial flow fan type Ebmpapst model W4S200-HK04-01 is modelled as rotating boundary within static air duct. In the wind tunnel of rectangular cross section of d x h = 400 x 450 [mm] at maximal rotating speeds (50 Hz), this type of power plant is capable of generating velocities of up to 30 m/s. The rotation speed of the axial flow fan is used as the boundary condition at the air duct inlet.

The turning vanes are modelled in the shape on of the thin air foils based on research of Drela et al. [14]. Location of the turning airfoil vanes, in the zone of the air duct where the air flow changes the direction for 90^0 and they are presented in Figure 4.

To stream the air flow just after the power plant group, flat plates (streamers) are incorporated into the domain and are modelled as rectangular flat plates. In this analysis total of 3 streamers are used in all acoustic and CFD analyses.

The complete domain (3D) with all system components analysed in this study is presented in Figure 5.

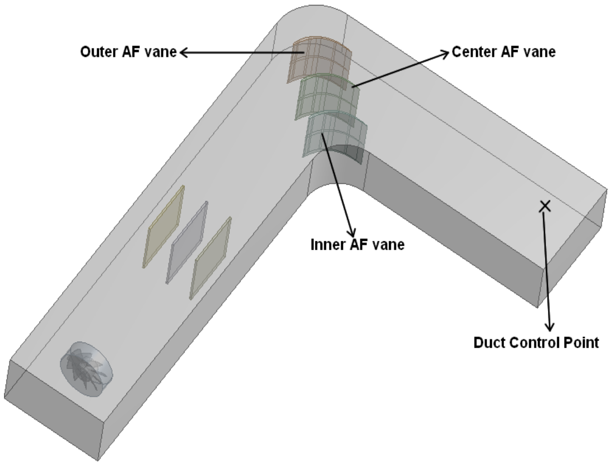


Figure 5. Computational domain (cross-section) of the air duct with the vanes in the shape of thin airfoils.

As a reference the ducts without turning vanes are initially analysed. Acoustic analysis is performed for two types of commercially available elbows for different rectangular cross sections. The first type is 90^0 degree elbow presented in Figure 7 (B), the second type is elbow with moderate radius, Figure 7 (A).

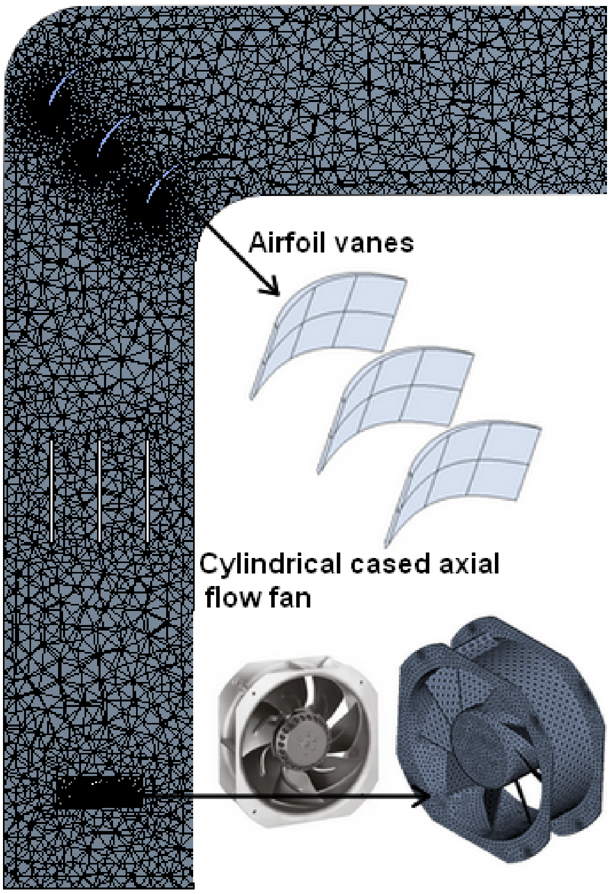


Figure 6. Computational domain (mesh cross-section) of the air duct with the vanes in the shape of thin airfoils.

Surface acoustic power level fields for the analysed elbows are presented in Figure 7, and by comparison of noise levels generated at duct surfaces it can be concluded that the elbow with moderate radius generates almost twice lesser noise compared to the elbow with straight boundaries for the same air-duct geometry and for the same flow conditions (low Mach number turbulent flows).

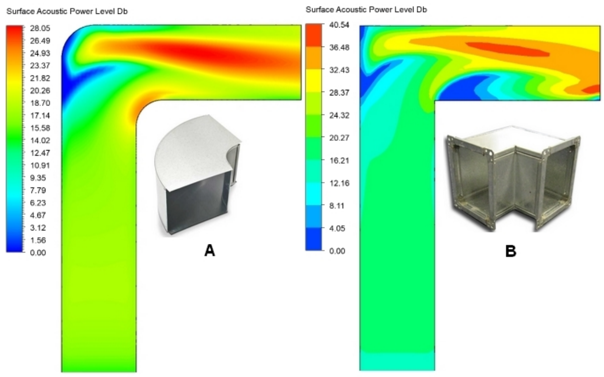


Figure 7. Surface acoustic power level field in the air duct with 90^0 elbows with moderate radius and

To investigate the effect of turning vanes in the shape of an airfoil on the noise generated in air ducts the acoustic analysis was performed for several different designs. Firstly, vanes are usually placed in places where air flow requires directional change.

Therefore, the airfoiled vanes are placed in elbow section, where it is expected the air flow to change direction for 90 degrees with minimal pressure drop at the duct outlet and generating minimal system vibrations.

Previous research has shown that, if adequately designed, vanes in the shape of air foil are more effective compared to classical vanes that are usually used in HVAC systems and are in the form of thin plates.It was assumed that several parameters influence the noise levels generated at the surfaces of air foil vanes: number of airfoils, geometrical position(s) of airfoil vanes within the air duct, angle(s) of attack(s) and type and airfoil thickness.

In this acoustic analysis. the airfoil vanes are positioned in inner elbow duct area, followed by the vane positioned at the outer elbow area and finally at the center of the elbow at different angles of attacks.

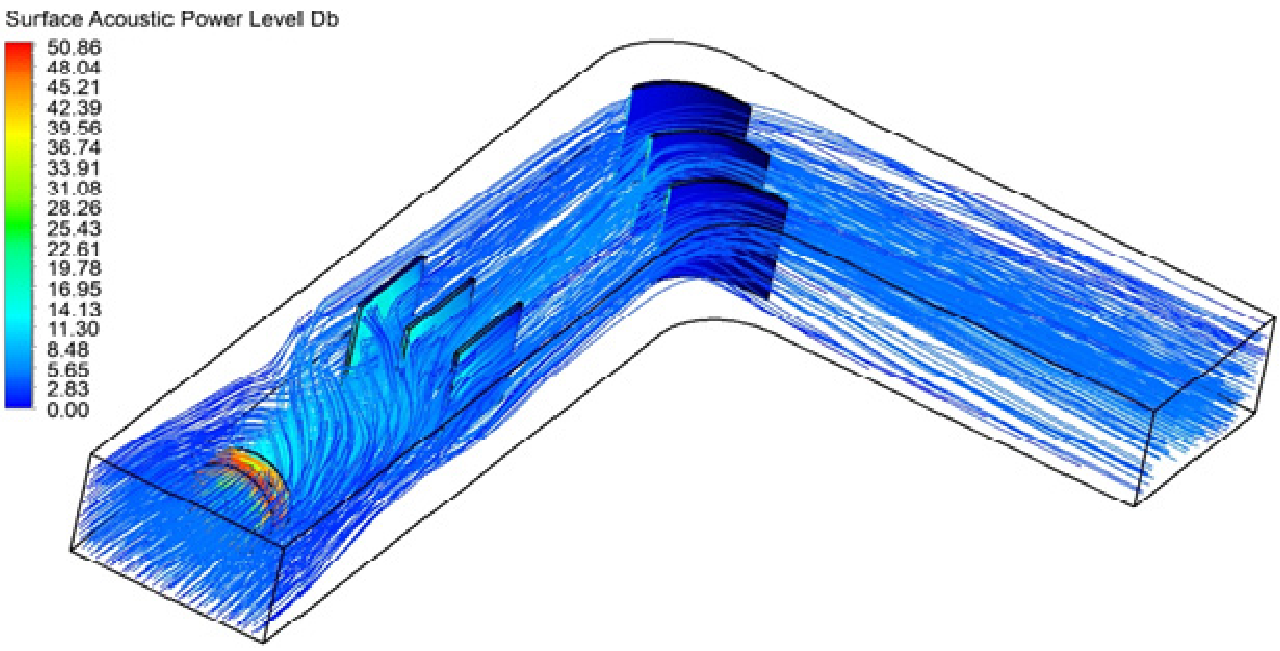
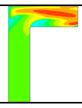
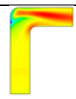
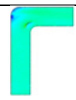
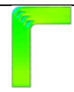
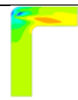
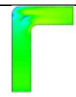
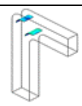
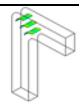
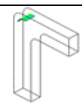
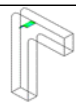


Figure 8. Surface acoustic power level and velocity fields in the air duct with 90^0 elbow with moderate radius and turning vanes in the shape of airfoil

Table 2. Empirical constants in the standard *k*-*ε* model

		Straight duct	Bend duct	Inner AF	Inner, Center and Outer AF	Outer AF	Center AF
							
Control point	10 m/s	33.11 dB	26.8 dB	17.10 dB	19.14 dB	22.23 dB	21.53 dB
	20 m/s	49.05 dB	43.24.33 dB	32.33 dB	35.49 dB	38.14 dB	37.91 dB
		-	-				
AF Leading edge	10 m/s	-	-	64.48 dB	45.99 dB	36. 19 dB	50.37 dB
	20 m/s	-	-	82.39 dB	61.22 dB	52.74 dB	66.70 dB

One of the most efficient designs, when outlet pressure drop is in question, is the design with multiple vanes in the elbow area as presented in Figure 4. One of the designs investigated in this study is the design with three airfoil vanes positioned in the elbow area (equidistant).

The surface acoustic field and the velocity field for the 10 m/s flow velocity of this case is presented in the Figure 8.

The results of acoustic analysis, for the cases analysed are presented in Table 2. Broadband noise levels are presented in decibels (dB) for the airfoil leading edge and the control point (receiver) located on the duct surface in the vicinity of the duct outlet (Figure. 5). The air flow velocity was set to 10 m/s and 20 m/s (inlet velocity) which are common velocity ranges in many industrial HVAC systems.

Being one of the noise sources, the airfoil vane surface acoustic power level, for the case of three equidistant airfoils in the elbow area is presented in the following picture (Figure 8.). It should be noted that the highest noise level arises from the airfoil leading edge.

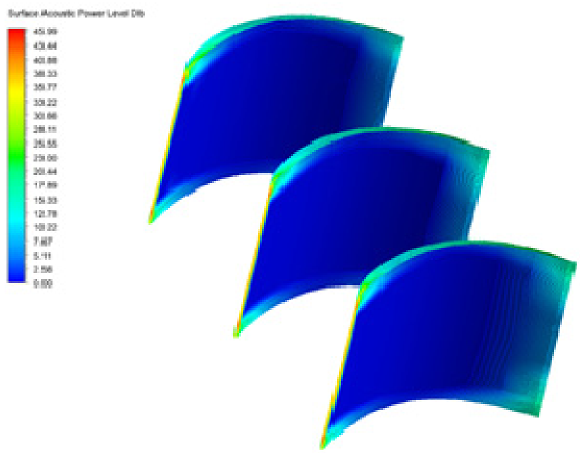


Figure 8. Surface acoustic power level on the airfoil vane

5. CONCLUSION

Noise is considered to be one of the problems in many industrial applications. During system design, operation and implementation measures that will reduce noise that arises from operating equipment must be considered. In the present analysis method for predicting aerodynamically generated broadband noise that arises from the air flow passing over the air duct vanes is presented. The following can be concluded:

Based on the Proudman's formula, using Lighthill's acoustic analogy broadband acoustic noise can be predicted. Required turbulent k - ε sound model is effective for this class of problems provided that empirical model constants are determined experimentally.

Turning vanes in the shape of an airfoil reduce noise levels for the low Mach number turbulent flows. Level of aerodynamically generated noise is affected by many factors where, position of the airfoil within the elbow duct, number of airfoils positioned, type and relative thickness of the airfoil and the airfoil angle of attack in respect to incoming air stream.

In order to satisfy all noise regulations and requirements acoustic analysis must be performed during system design phase and Lighthill's acoustic analogy broadband acoustic noise model can be used for noise predictions in HVAC systems.

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АНАЛИЗА АЕРОДИНАМИЧКЕ БУКЕ У ВЕНТИЛАЦИОНИМ КАНАЛИМА СА УМЕРИВАЧИМА ВАЗДУХА

Б. Рашуо, М. Динуловић, М. Трнинић, М. Стаменовић, Н. Милошевић, Н. Ђурчић

Бука је један од проблема у многим индустријама. У овом раду приказана је анализа аеродинамички настале буке која се јавља при струјањима ваздуха преко усмеривача ваздуха облика аеропрофила у ваздушним каналима. На основу Проудманове једначине и Лајтхилове акустичне аналогије широкопојасни ниво буке у ваздушном каналу може бити израчунат. За моделирање турбулентног модела, k – ε турбулентни модел је коришћен а потребне константе су експериментално одређене мерењем у аеротунелу за подзвучне брзине, након којих су вршена мерења буке за у непосредној близини усмеривача ваздуха облика аеропрофила. Неколико решења је анализирано, канал са унутрашњим аеропрофилом, канал са централним аеропрофилом, канал са спољашњим аеропрофилом као и комбинација претходних случајева. Анализом добијених резултата, а у циљу испуњавања свих законских прописа који се тичу индустријске буке, који постају све стриктнији, акустична анализа и пројектовање морају бити примењени у већини индустријских система јер бука представља опасност по здравље радника у радном окружењу.