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IDENTIFYING DRYING MACHINE HEAT EXCHANGER NOISE

UDC 534.6

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Abstract. *To carry out this research, we set up a measurement station for the analysis of aerodynamic noise induced by the drying machine heat exchanger. The adequate velocity field of the air flow was simulated, while eliminating the noise of the built-in fan to the greatest extent possible.*
All measurements were taken in the anechoic chamber. The integral and local aerodynamic and acoustic characteristics of heat exchangers were analyzed. The measurement of local aerodynamic characteristics comprised the measurement of the velocity fields with the hot wire anemometer on the heat exchangers output plane. The possibility of the power spectra estimation during the design of energetic properties of the heat exchanger is discussed.

Key Words: *Noise, drying machine, heat exchanger, aeroacustics measurement*

1. INTRODUCTION

An efficient selection of a heat exchanger type is desirous during the design of a clothes dryer appliance. The method for the estimation of the sound power level and power spectra, based on measurements provided for the reduction of the energy consumption of the drying machine is presented.

Energy consumption and noise emission are the most important functional characteristics of household clothes drying machines, on the basis of which customers make their purchase decisions [1]. The heat exchanger is one of the most important components of a clothes drying machine, with a marked importance of the energy characteristics. During the design of the clothes drying machine, primary concern is oriented towards the energy efficient design of the primary and secondary flow channels, the proper selection of fans and the heat exchanger. The primary flow is the air flow passing through the clothes drum and heat exchanger, and the secondary flow is used for cooling down the heat exchanger and allowing the condensation process. For the design of the heat transfer characteristics

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of the heat exchanger, it is advantageous to measure local instantaneous velocity in order to achieve efficient condensation during the complete drying process. We also use the local instantaneous velocity fluctuations as an approximate measure of acoustic acceptance of the heat exchanger during the design process.

The measurements showed that the difference between various types of heat exchangers exceeded 5 dB (A) in acoustic power when installed in the appliance, resulting in typical deviations in the classification of drying machines, which need to be considered in the further development of the machine. The noise level depends on the geometry of the channels and aerodynamic characteristics of air flow in the heat exchanger flow channels. Heat exchanger manufacturers have no reliable characteristics concerning the acoustic properties of heat exchangers. The most pronounced noise source of the heat exchanger is the airflow at the outlet side of the secondary circuit, which is discharged directly into the environment.

To estimate the amount of sound generated by the turbulent motion, information is required concerning the mean values of certain fluctuating quantities in accordance with the Lighthill theory. Lighthill's theory is based on an acoustic analogy whereby the exact Navier-Stokes equations for the fluid flow are rearranged to form an inhomogeneous wave equation for the fluctuating fluid density. The forcing function on the right-hand side represents a distribution of acoustic sources in the ambient flow at rest, replacing the complete fluid flow. In Lighthill's theory $\partial^2 T_{ij} / \partial x_i \partial x_j$ we find the strength of the acoustic sources per unit volume, and T_{ij} is Lighthill's instantaneous applied acoustic strength tensor. The central property of Lighthill's theory is the evaluation of the fourth order two-point space-retarded time covariance of $\partial^2 T_{ij} / \partial t^2$ and its distribution throughout the given flow field. The acoustic power radiated to the far field is found by the integration of this space-retarded time covariance over the entire volume flow.

It is often convenient to approximate certain types of turbulent flow as if they were isotropic, since isotropic turbulence has no preferred direction and requires a minimum number of quantities to describe its characteristics [2]. The estimation of sound radiation from isotropic turbulence is an important step in the application of Lighthill's theory. For the total acoustic power $\overline{p}$ in the far field, Proudman found [3]

$$\overline{p} = \alpha \rho \frac{u^5}{c^5} \varepsilon \,. \tag{1}$$

Here the $u^2 = 2K / 3$, K is the turbulent kinetic energy, c is the speed of sound, u is the velocity flow, ρ is the density in the far field, ε is the dissipation rate of the isotropic turbulence and the presumably universal proportionality constant α the Proudman constant, was expressed in terms of the spatial correlation function of the turbulence.

The measured noise power spectra will be compared to the results of Rubinstein and Zhou for the sweeping hypothesis of time correlations [4,5]. The frequency spectrum of the sound radiated by isotropic turbulence is found to scale as $\omega^{-4/3}$ (recent result due to Lilley) at high frequencies, while dimensional analysis based on Kolmogorov scaling predicts the dependence $\omega^{-7/2}$ instead [6].

Digital computers are in many cases useful in predicting sound propagation in flowing fluid media [7]. The turbulence models in use by the turbulence community are, in general, not applicable for the prediction of radiated noise, since they model only average or mean characteristics and not the instantaneous properties of the turbulent flow [8]. The DNS computation has the advantage of being free of any assumption of modeling. Unfortunately, computing the fourth time derivative of the Lighthill stress tensor requires high orders of resolution in both space and time. The DNS results are also restricted to moderate Reynolds numbers. The DNS modeling of turbulent flow in the heat exchanger is due to its complicated geometry, too complicated to be feasible for the design of the clothes dryer heat exchanger.

In continuation, the measurement procedure and results will be presented. Two type A heat exchangers and one type B heat exchanger were analyzed. The channels of both type A heat exchangers are made of fins transformed from sheet metal straps, whereas type B is provided with channels based on extruded composed profiles (Fig. 1). Both type A heat exchangers have the same geometry, but they differ in the procedure of fin production and their attachment to the structure of the heat exchanger.

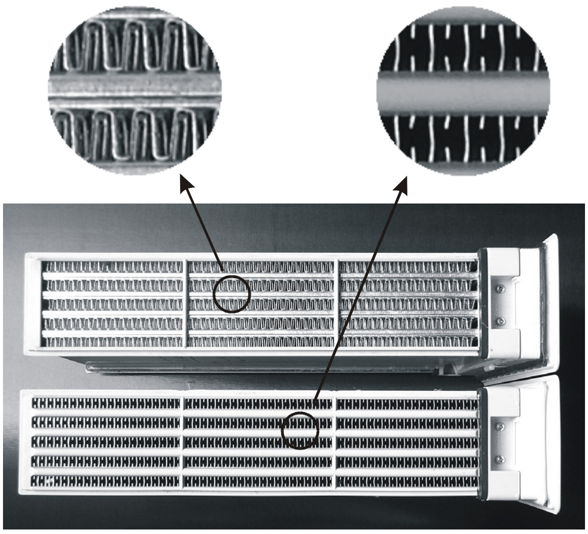

Fig. 1. Type A heat exchanger (above) and type B heat exchanger (below)

2. The Measuring Station

In order to analyze the characteristics of various heat exchangers, a measurement station was constructed. The measuring station allows the measuring of aerodynamic and acoustic parameters of clothes drying machine heat exchangers in both the primary and secondary flow. The measurements were performed at the outlet side of the secondary air circuit.

The measuring station (Fig. 2) is composed of four main units: a fan as the flow generator, a flow measuring unit, an attenuating and damping unit and a measuring unit.

The fan is of radial type and is fitted with a frequency inverter allowing the regulation of the heat exchanger operational point according to the selected volume flow. The flow measuring unit contains an orifice plate, fitted into the flow system in accordance with the DIN

1952 standard [9]. The length of the straight part of the pipe in front of the orifice plate is 10 *d* and behind the orifice plate 3 *d*. The pipe diameter is *d* = 150 mm. A four-arm flow guide with the length of 2 *d* behind the fan was used to eliminate vortices of the fan.

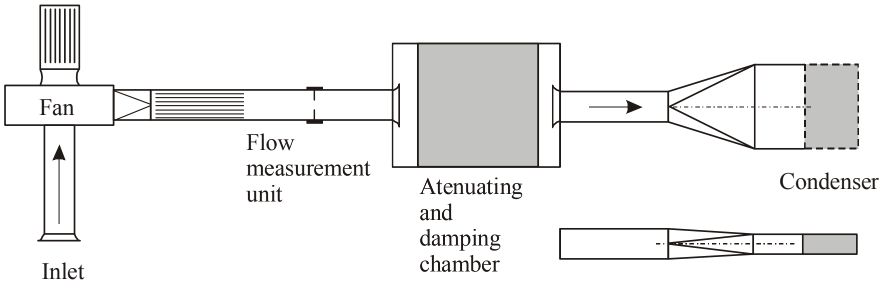

Fig. 2. Measuring station diagram

The third unit of the measuring station is the attenuating and damping chamber with a length of 1000 mm and a diameter of 600 mm. The flow slows down in the attenuating chamber. The attenuating chamber contains an 800 mm-thick glass wool layer to ensure the reduction of noise, generated by the fan.

The measuring unit is composed of a 10 *d* long pipe, a transition element and the element containing the built-in heat exchanger. The transition element allows the transition from the round cross-section to the square cross-section and is executed with small angles to prevent flow separation. The measuring unit is coated in felt, preventing sound transmission from the measuring unit structure into the environment. The measurement unit was placed in the anechoic chamber.

For an accurate selection of the heat exchanger working point, a fully gas-impermeable system was provided on the part from the orifice plate to the heat exchanger.

3. The Experiment

3.1 Measurement of the acoustic characteristics

Measurements of the heat exchangers acoustic characteristics were performed in the semi-anechoic room with a free space volume of 220 m^3 (7.8 m × 6.7 m × 4.2 m). Its walls and ceiling are coated with a sound absorption cover in the form of polyurethane foam wedges. The wedges are 80 cm long. There is also a 5 cm layer of air between the wedges and the walls. The frequency range of the room is from 100 to 20.000 Hz, the noise level in the most unfavorable circumstances is 13 dB (A).

To carry out source acoustic power measurements according to the absolute method, a measuring system was used consisting of 10 Falcon TM Range ½", type 4189 (B&K) microphones and 10 DeltaTron-Type 2671 microphone pre-amplifiers, connected via three BNC 2149 connection modules, linked to the DSA interfaces – NI 4552 dynamic signal analyzers. The temperature, humidity and ambient pressure sensors are connected to a TC 2190 module, connected to a NI 4351 interface with a coaxial cable.

Acoustic pressure measurements were carried out in 10 positions on the surface of the hemisphere with a 1 m radius as a function of flow through the heat exchanger (Fig. 3). The locations of the microphones for acoustic pressure measurements are presented in Table 1. An outlet funnel was fitted to the heat exchanger outlet part and adjusted above the reflecting surface in the anechoic chamber.

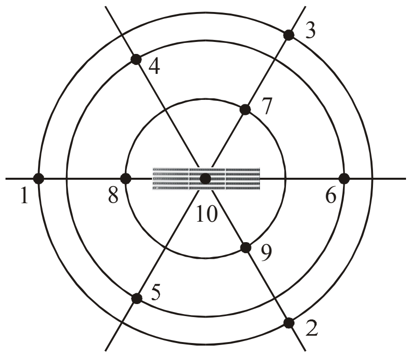

Fig. 3. Microphone positions over a heat exchanger on the reflecting plane

Table 1. Locations of the microphones

Mic.	x [m]	y [m]	z [m]
1	–0,99	0,00	0,15
2	0,50	–0,86	0,15
3	0,50	0,86	0,15
4	–0,45	0,77	0,45
5	–0,45	–0,77	0,45
6	0,89	0,00	0,45
7	0,33	0,57	0,75
8	–0,66	0,00	0,75
9	–0,33	–0,57	0,75

$$L_p = 10 \cdot \log(10^{0,1L_C} - 10^{0,1L_B}) \tag{2}$$

where L_{p} is the corrected value, L_{C} the measured value and L_{B} the background noise level. The acoustic power level was calculated on the basis of the average sound pressure level on the measuring surface:

$$L_W = \overline{L}_p + 10 \cdot \log_{10}\left(\frac{S}{S_0}\right) + C \qquad C = -10 \cdot \log_{10}\left(\sqrt{\frac{293}{273+t} \cdot \frac{p}{1000}}\right) \tag{3}$$

where $\overline{L}_p$ is the calculated average sound pressure level, r is the hemisphere radius, $S = 2 \cdot \pi \cdot r^2$ the hemisphere surface, t the operating ambient temperature in °C, p the ambient air pressure in mbar and C the correction factor, taking account of the barometric pressure and temperature t in the chamber.

The sound intensity measurements of heat exchanger acoustic characteristics were performed with a two-channel analyzer with a sound intensity probe in 21 measurement points, based on the method described by Crocker [10].

The acoustic efficiency η is defined as a relation between the acoustic flow and mechanical power:

$$\eta = W / W_{mech} \tag{4}$$

where W_{mech} is the mechanical or aerodynamic power of the flow through the heat exchanger.

The mechanical energy of the flow is:

$$W_{meh} = \frac{\rho \dot{V}^3}{2S^2} \tag{5}$$

where $\dot{V}$ = the gas flow volume, ρ = the density of gas, and S is the outlet transverse surface. The empirical estimate for the acoustic power level is

$$L_w = 120 \text{ dB} + 10 \lg \eta\, W_{mech} / W_0 \tag{6}$$

where $W_0 = 1$ W.

All measurement equipment was calibrated using the appropriate devices.

3.2 Measurements of the aerodynamic properties of the flow

Beside the measurements of the heat exchangers' acoustic properties, measurements of the aerodynamic properties of the turbulent flow at the outlet side of the secondary circuit of the heat exchanger were also performed.

Velocity field measurements were performed with the aid of a traversing method with a single-component hot-wire probe Dantec type 55P11 and a Dantec MiniCTA anemometer, which was connected to the data acquisition board in a personal computer that captured the data with a sampling rate of 50000 Hz. Total acquisition time was 4 s. The velocity was filtered using a low pass filter with a cutoff frequency of 10 kHz. The estimated uncertainty of the velocity measurements was ±3% of the average value. The velocity measurements were performed in the middle of the heat exchangers' outlet plane at a distance of 3 mm from the heat exchanger. Power spectra were calculated according to the Eq. 7 [2].

$$u(f) = \int_{-\infty}^{+\infty} u(t) e^{-i2\pi ft} dt\,. \tag{7}$$

4. Results

Fig. 4 shows the total acoustic power level of the flow through the heat exchanger for three heat exchanger types A1, A2, and B. We noticed substantial differences in the noise levels between individual heat exchangers. Heat exchanger A1 achieves a total acoustic power of 2.4 μW (63.9 dB) at a flow of 157 m^3/h. At the same flow rate, however, the acoustic power of the B heat exchanger is only 6.3 nW (27.4 dB).

The sound power spectra analysis of the loudest heat exchanger A1 at a flow rate of 125 m^3/h shows a marked tone of acoustic power at a frequency of 800 Hz (Fig. 5), which increases to 1000 Hz at a maximum flow rate. The power spectra of the total radiated acoustic power scales is $\omega^{-4/3}$, which is the scaling derived from the sweeping hypothesis of time correlations by Rubinstein and Zhou [5]. Here, peaks are superimposed on the basic spectra scaling, depending on the volume flow of the heat exchanger. The intensity of the superimposed peaks increases with the increasing volume flow through the heat exchanger.

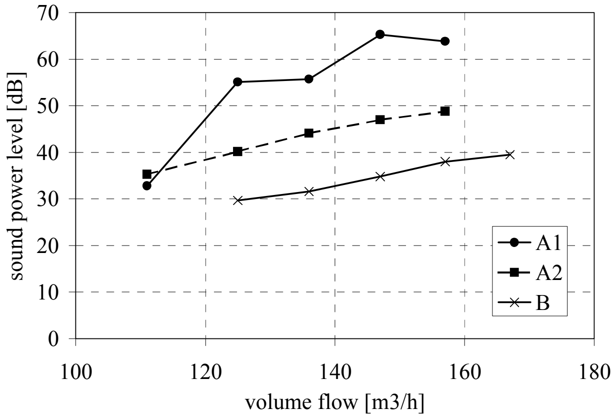

Fig. 4. Total sound power level as the function of flow through the heat exchanger

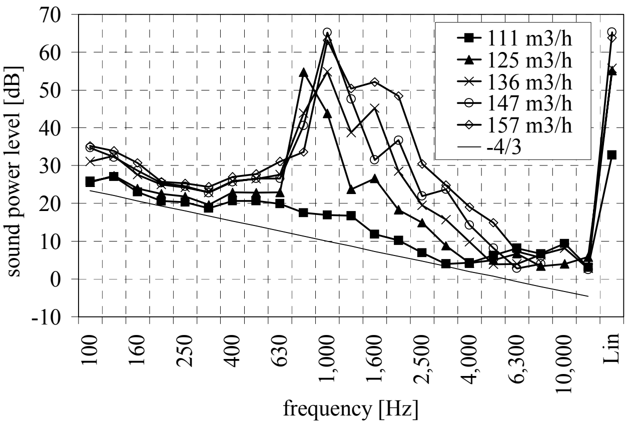

Fig. 5. Sound power level spectra - heat exchanger A1. The -4/3 straight line represents the scaling of the sound power spectra radiated by isotropic turbulence [6]

Similar conclusions may also be reached from the frequency characteristic of the sound pressure level (Fig. 6). The maximal detected sound pressure level was measured using microphone 6 at the frequency of 1000 Hz, amounting to 60.8 dB.

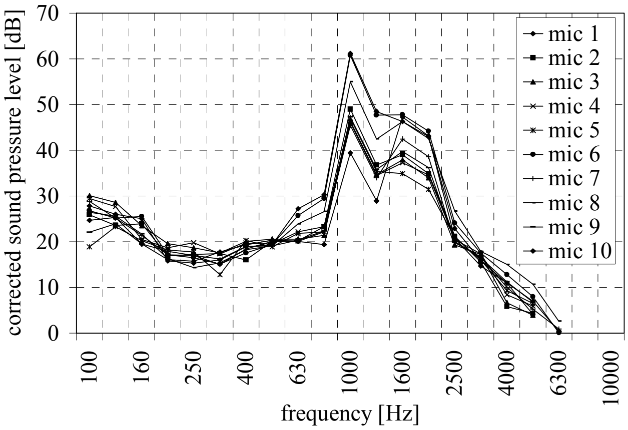

Fig. 6. Corrected sound pressure level at r =1m in accordance with Table 1, heat exchanger A1, and a volume flow of 157 m^3/h

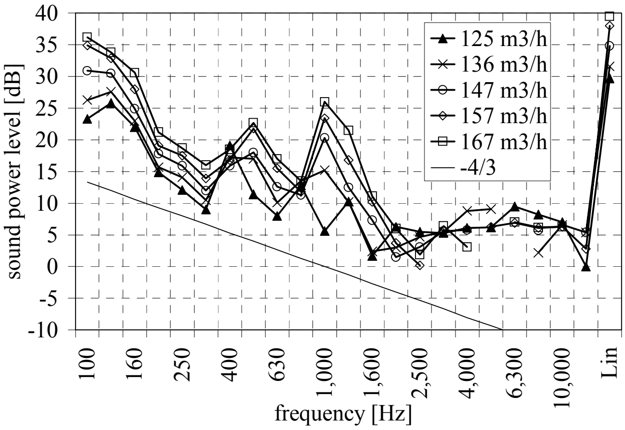

Fig. 7. Sound power level spectra - heat exchanger B. The −4/3 straight line represents the scaling of the sound power spectra radiated by isotropic turbulence [6]

Heat exchanger B shows, due to a different flow channel design, lower acoustic power levels than heat exchanger A1 (Fig. 7). The frequency characteristic demonstrates a tone component at 1000 Hz, which is nearly inaudible due to the low level. The power spectra of the total radiated acoustic power is again found to scale as $\omega^{-4/3}$, but the peaks superimposed to the basic spectra scaling are much less pronounced than in the case of the heat exchanger A1.

By measuring the acoustic intensity we visualized the flow of sound energy in A2 and B heat exchangers, at a flow rate of 125 m^3/h (Figure 8). The acoustic power level through the frontal plane, calculated by the sound intensity method, amounts to 38.2 dB for the A2 heat exchanger and 30.6 dB for the B heat exchanger. If this is compared to the absolute method of determining the total acoustic power, accounting for 40.2 dB and

29.7 dB respectively, we may conclude that the sound intensity method shows slightly different results. The radiation map in the B heat exchanger demonstrates a marked density of sound energy flow, concentrated in the central part of the outlet surface, resulting from the flow channel design. Fig. 9 shows the acoustic efficiency as a function of flow in different types of heat exchangers.

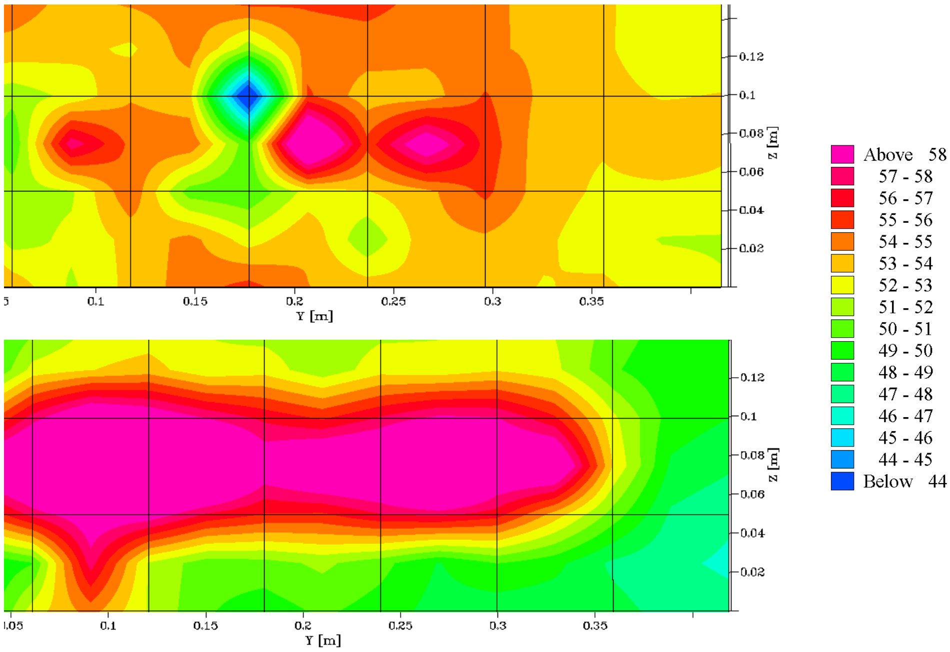

Fig. 8. Sound intensity level in the A2 heat exchanger (above) and B heat exchanger (below). The flow rate is 125 m^3/h

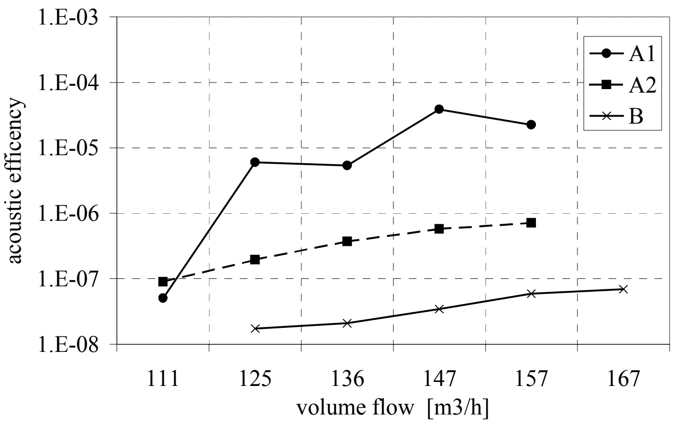

Fig. 9. Acoustic efficiency of different heat exchangers

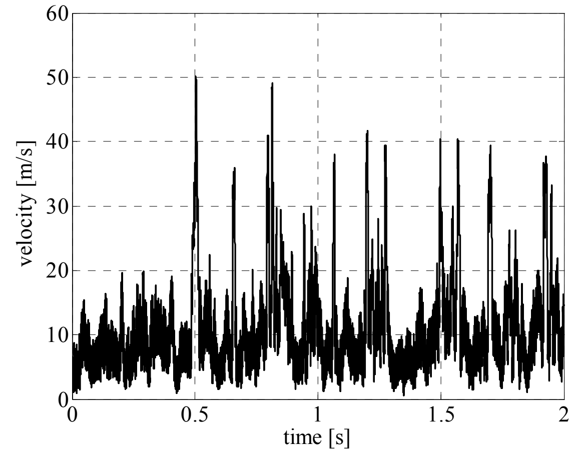

Fig. 10. Velocity fluctuations - heat exchanger A1

The results in the measurement of the velocity fluctuations are shown in figures 10 and 12. Velocity power spectra are shown in figures 11 and 13. The power spectra of the heat exchanger A shows a marked peak of fluctuations at the frequencies from 1000 to 1500 Hz, and the spectra correspond to −5/3 turbulence decay law from 100 Hz up to 10 kHz, where the filtering was applied [2]. The power spectra of the heat exchanger B, however, does not show such a peak of fluctuations, but again corresponds to a −5/3 turbulence decay law.

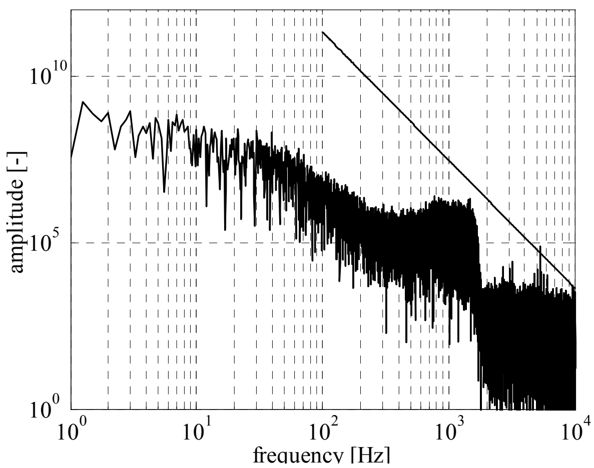

Fig. 11. Power spectra of velocity fluctuations - heat exchanger A1.
The straight line represents −5/3 turbulence decay law.

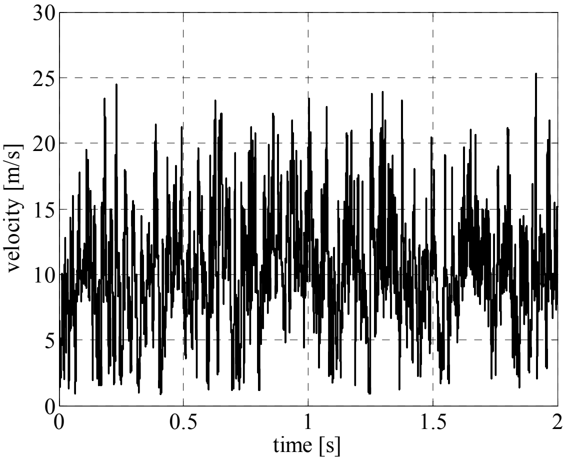

Fig. 12. Velocity fluctuations - heat exchanger B

We have found that the power spectra of velocity fluctuations in heat exchangers A1 and B both express similar behavior, except at the frequency range from approximately 500 to 1500 Hz, where a peak of fluctuations is present for heat exchanger A1. We assume that this peak of velocity fluctuations is responsible for high levels of radiated sound power level, as shown in Fig. 4 and Fig. 5. Since the noise in the case of heat exchanger A1 is presumably induced by velocity fluctuations at the outlet of the heat exchanger, measurement of velocity fluctuations of an optional heat exchanger being tested for the determination of its energetic properties, can be used also as a measure of acoustic acceptance. To establish energetic properties requires the measurement of local velocity fluctuations, which must be approximately the same intensity over the entire cross section of the secondary flow to enable proper condensation over a heat exchanger's entire volume.

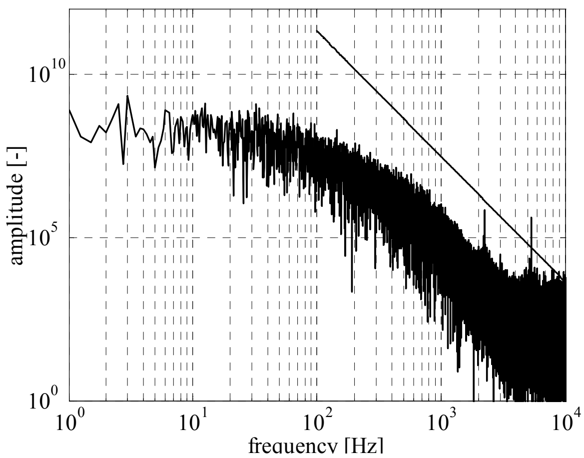

Fig. 13. Power spectra of velocity fluctuations - heat exchanger B. The straight line represents -5/3 turbulence decay law.

For an improved version of the heat exchanger, given that its energetic characteristics are acceptable, we propose a design without heavily expressed peaks in velocity fluctuation power spectra, as this is the case with heat exchanger B.

As far as further research is concerned, the determination of an inverse correlation of the time of motions of any given spatial scale should be analyzed in connection with the frequency distribution of acoustic energy of an arbitrary heat exchanger [11].

5. CONCLUSIONS

Development trends in large household appliances have been increasingly oriented toward noise reduction. In the case of a clothes dryer machine we estimate the interdependence between the noise sources of the secondary circuit at the outlet side of the heat exchanger and the aerodynamic characteristics of the heat exchanger.

Total sound power levels differ up to 30 dB among different heat exchangers and generally increase with the volume flow rate. The maximum sound power of the heat exchanger A1 is emitted at the frequency interval from 1000 to 2500 Hz, which corresponds to the peak in the power spectra of the velocity fluctuations at the outlet of the heat exchanger. Such behavior was not detected with the heat exchanger B. According to this, the measurement of velocity fluctuations can be used as a measure of acoustic acceptance of the heat exchanger during the design of its energetic properties, which include measurements of local velocity fluctuations.

In the heat exchanger B, the sound energy flow density was identified, concentrated in the central part of the outlet surface, resulting from the flow channel design.

For a future selection of heat exchangers for clothes dryers', designers will have to focus on energy efficiency as well as adequate acoustical characteristics of heat exchangers.

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IDENTIFIKACIJA AERODINAMIČNE BUKE TOPLOTNOG IZMENJIVAČA MAŠINE ZA SUŠENJE RUBLJA

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Za istraživanje je upotrebljen sistem za merenje i analizu aerodinamke buke toplotnog izmenjivača mašine za sušenje rublja. Simulirano je polje brzine vazdušnog toka uz eliminisanje, u najvećoj mogućoj meri, buke ugrađenog ventilatora. Merenje su izvodena u anehoičnoj sobi. Analizirale su integralne, lokalne aerodinamične i akustične karakteristike kondenzatora. Lokalne aerodinamične karakteristike su analizirane merenjem polja brzine sa «hot wire» anemometerom na izlaznoj površini kondenzatora. Razmatrana je mogućnost estimacije spektra snage u toku projektovanja energetskih osobina toplotnog izmenjivača.

Ključne reči: *buka, mašina za sušenje, toplotni izmenjivač, aeroakustična merenja*