Reducing the noise emitted from a domestic clothes-drying machine

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In recent years the development of large household appliances has increasingly focused on the reduction of noise. In this paper we report on attempts to reduce the sound-power level of an existing clothes dryer by investigating the aerodynamic noise sources. The critical points were determined on the basis of spatial sound-intensity measurements and sound-pressure-level spectra measurements. The changes to produce the modified dryer, version I, involved corrections to the secondary-flow fan's inlet region. For version II a different design of the heat exchanger was considered. For this, a measurement station was set up in order to analyze the aerodynamic properties of heat exchangers. The velocity and the acoustic properties of the heat exchangers were measured to help in the selection of the heat exchanger in accordance with the aeroacoustics theory that connects turbulent fluctuations and noise generation. After performing all the modifications, the A – weighted total sound-power level was reduced from 76.0 to 66.5 dB for left rotation and from 73.8 to 64.7 dB for right rotation. © *Institute of Noise Control Engineering*

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1 INTRODUCTION

Customers make their purchases of clothes dryers based on the price, the energy consumption, and the acoustic properties of the appliance. In this study we show how the increased demands of the market for improved noise characteristics were considered and how appropriate improvements were selected and implemented. The basic version and two modified versions of the domestic appliance were produced and analyzed in terms of their acoustic characteristics. It should be noted that the modifications to the appliance did not reduce its functional or energy characteristics.

In the following sections the condenser dryer will be described in more detail, with an emphasis on possible noise sources, followed by a description of the experimental set-up, the results and the discussion.

2 DESCRIPTION OF THE CONDENSATION CLOTHES-DRYER APPLIANCE

A condensation-type clothes dryer is designed to dry clothes in places where, for various reasons, the use of a ventilation-type dryer is not possible. There are two airflows in a condensation-type clothes dryer, the primary and the secondary, as shown in Fig. 1. The primary airflow is in a closed loop. The air from the primary fan flows through the electric heater, enters the clothes drum, and takes moisture from the clothes (up to 100% relative humidity). The primary airflow then enters a heat exchanger, where it is cooled down by the secondary airflow. As the temperature of the primary airflow is thus reduced, the condensate forms on the primary fins of the heat exchanger, and the condensed water is stored in a special container that has to be emptied after each drying. The secondary-flow fan pushes the secondary airflow in the secondary-flow channel, and after being in the heat exchanger it receives heat from the primary airflow and leaves the appliance.

The condensation-type dryer under consideration is capable of drying up to 5 kg of clothes. Its housing is made of zinced and lacquered sheet metal of thickness 0.7 to 1 mm. The outer dimensions of the dryer are 600 x 600 x 850 mm, with a 100-

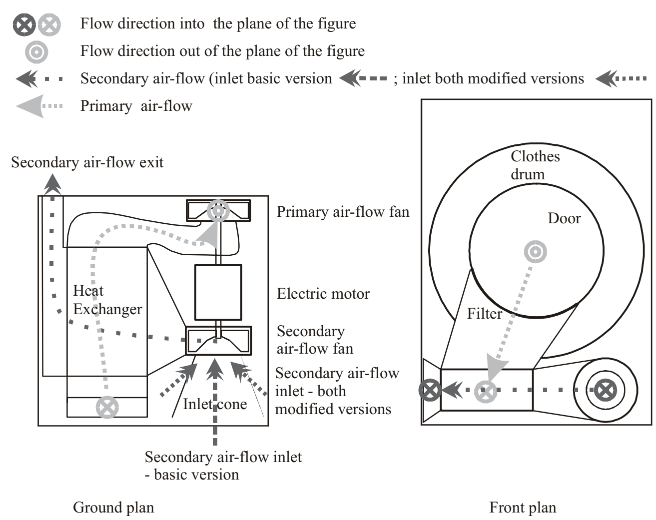

Fig. 1. Schematics of the condenser-type clothes dryer. The thick arrows show the flow direction of the primary and secondary airflows. In the region of the inlet cone (in the ground plan) the long vertical arrow shows the secondary airflow of the basic version (completely closed off in the both modified versions), and the two short arrows show the airflow of the first and second modified versions.

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litre drum made of stainless steel. Two or three ribs (depending on the model) are mounted circumferentially on the inner side of the drum to help mix the clothes during the drum's rotation. The rear drum wall is perforated and supported by ball bearings that are sealed (composite seal or felt) on the inlet-air channel in which the electrical heater is mounted. The front drum wall is supported and sealed on the outlet channel where the air filter is placed.

The heat exchanger is an aluminum-plate type, with ribs on the side of the secondary airflow. The channels of both types on the A-type heat exchanger are made of fins transformed from sheet metal strips, whereas on the B-type heat exchanger the channels are based on extruded composed profiles (Fig. 2). The two versions differ in the form of the flow channel in the direction of the secondary flow. At the entrance to the heat exchanger an additional filter is placed to prevent soiling; however, this can also cause a reduction in the heat transfer. The drum is belt driven, with the belt placed circumferentially onto the drum. A single electric motor drives both the fans and the drum.

3 AERODYNAMIC NOISE GENERATION IN THE CLOTHES DRYER

Our experimental study of the aerodynamic and acoustic parameters was designed to help remove the most significant noise sources. We estimated that the potential noise sources of the dryer are influenced by the aerodynamic properties of both the installed fans, by the design of the flow tracts (including the inlet and outlet slots), and the acoustic characteristics of the heat exchangers[1,2].

The noise is generated at the installed ventilators and at the elements of the connecting flow tracts where the local flow velocity is the highest. The aerodynamic noise generated by the fans comes from a combination of the discrete noise frequencies relating to the number of blades and the rotational speed of the machine, i.e., the blade-passing frequency and its harmonics, and the broadband noise or turbulence noise[3]. Several publications show that a significant portion of the noise is related directly to the aerodynamic fluctuations of the forces applied by the fluid on the rotor blades of the fan. This unsteady loading forms a source system creating acoustic propagation in the surrounding medium up to the listener's ear. The aerodynamic forces brought into play, steady or fluctuating, are represented by a source that is dipolar in nature[4]. Several mechanisms of noise generation coexist, without it always being possible to be able to provide evidence of a dominant source. Therefore, in order to understand the broadband-noise generation mechanisms, a precise knowledge of the flow behavior in the impeller is very important.

The installation of the secondary fan in the secondary-flow tract of the clothes dryer is such that the fan rotates in both directions. This is due to the need for the drum to rotate in both directions to prevent crushing of the clothes in the drum and because a single motor drives both fans and the drum. The secondary flow is open, and the position of the secondary-flow inlet is shown in Fig. 3. The diffuser is implemented with no guiding vanes, and because of the rotation in both directions there is no cut-off region. The absence of the cut-off region, where in centrifugal fans the noise is generated with the blade-passing frequency, implies that the noise of the secondary fan is predominantly broadband turbulent noise[5].

At the airflow outlet from the heat exchanger the flow is driven directly into the surroundings. The noise generated by the machine is thus also influenced by the selection of the heat exchanger and the muffler. Preliminary measurements have shown that the difference between the various types of heat exchangers exceeded 3 dB in the A – weighted total sound-power level, when installed in the appliance, resulting in typical deviations in drying-machine classification that need to be considered in the further development of the machine. The noise level depends on the geometry of the channels and the aerodynamic characteristics of the airflow 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 a heat exchanger is the airflow on the outlet side of the secondary circuit, which is discharged directly into the environment. In what follows we will present the measurement procedure and the results.

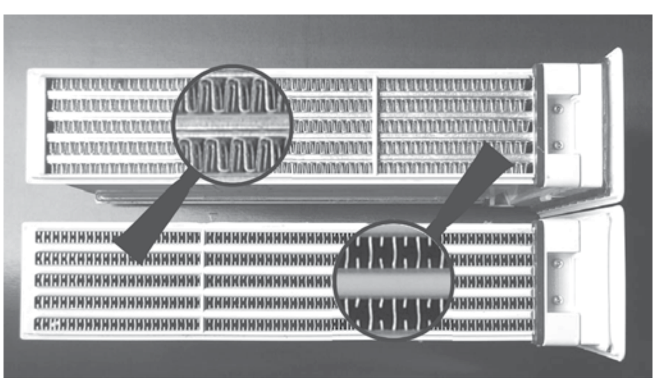

Fig. 2. Type-A heat exchanger (top) and type type-B heat exchanger (below).

Fig. 3. Inlet channel to the secondary air-flow fan, basic version (left), modified versions I and II (right). The inlet slots on the front wall of the basic version were completely closed off and for modified versions I and II replaced with slots in the inlet duct so that the modified version drew all inlet air from the inside of the cabinet.

Two types of heat exchanger were analyzed; we have designated them as A and B. The heat exchanger is one of the most important components of a clothes-drying machine, particularly in terms of the energy characteristics. During the design of a clothes-drying machine the main concern is an energy-efficient design of the primary- and secondary-flow channels, the proper selection of fans, and the choice of heat exchanger. For the design of the characteristics of the heat exchanger it is advantageous to measure the local instantaneous velocity in order to achieve efficient condensation during the complete drying process. We used the local, instantaneous velocity fluctuations as an approximate measure of the acoustic acceptance of the heat exchanger during the design process.

In the following we will use the Lighthill acoustic analogy to describe the noise production at the outflow from the heat exchanger. To estimate the amount of sound generated by turbulent motion, information is required concerning the mean values of certain fluctuating quantities in accordance with Lighthill's 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 central property of Lighthill's theory is the evaluation of the fourth-order two-point space-retarded time covariance and its distribution throughout the given flow field[6]. The acoustic power radiated to the far field is found by integrating this space-retarded time covariance over the entire flow volume.

It is often convenient to approximate certain types of turbulent flow as if it were isotropic since isotropic turbulence has no preferred direction and requires a minimum number of quantities to describe its characteristics[7]. Using this approximation the estimation of the sound radiation from isotropic turbulence is an important step in the application of Lighthill's theory. The measured noise power spectra will be compared with the results of Rubinstein and Zhou for the sweeping hypothesis of time correlations[6,8]. They found that the frequency spectrum of the sound radiated by isotropic turbulence scales is $\omega^{-4/3}$ at high frequencies, while a dimensional analysis based on $\omega^{-7/3}$ Kolmogorov scaling predicts the dependence instead[9].

Digital computers are in many cases useful for predicting sound propagation in flowing fluid media. The turbulence models in use by the turbulence community are, in general, not applicable for the prediction of radiated noise, since they model only the average or mean characteristics and not the instantaneous properties of the turbulent flow[10,11]. The DNS (Direct Numerical Simulation) computation has the advantage of being free of any modeling assumptions. 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.

4 EXPERIMENT

4.1 Measurements of the acoustic characteristics of the clothes dryer

Measurements of the acoustic characteristics of the clothes dryer were performed in an anechoic room with a free-space volume of 220 m^3 (7.8 m x 6.7 m x 4.2 m). Its walls and ceiling are coated with a sound-absorbing covering 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. Measurements according to the ISO 3745 standard were carried out for the anechoic chamber[12]. They showed that the lower limit frequency is 100 Hz, the background A-weighted sound pressure level is 13 dB and the discrepancy from legality "1/R" in the frequency range 90 Hz to 10 kHz is admissible according to the standard ISO 3745[13].

To carry out source acoustic-power measurements according to the absolute method[12] a measuring system was used consisting of 10 Falcon ™ Range ½"-type 4189 (Brüel & Kjær) microphones and 10 Delta Tron-Type 2671 microphone pre-amplifiers connected via three BNC 2149 connection modules, linked to DSA (Dynamic Signal Analyzer) interfaces – NI (National Instruments) 4552 dynamic signal analyzers. The temperature, humidity and ambient-pressure sensors were connected to a NI TC2190 module, which was connected to an NI 4351 interface with a coaxial cable.

Sound-intensity measurements of the clothes dryer's acoustic characteristics were performed using a two-channel analyzer with a sound-intensity probe at discrete measurement points and Noise Source Location program. The main purpose of the program is to help identify the location of noise emitted from clothes dryer[14,15]. The sound-intensity probe was positioned by a computer-controlled traversing system.

All the measurement equipment was calibrated using a suitable piston phone and a sound-intensity calibrator with a valid calibration certificate from the appropriate authority.

4.2 Measurements of the acoustic characteristics of the heat exchanger

In order to analyze the characteristics of various heat exchangers a measurement station was constructed. The measuring station allows measurements of the aerodynamic and acoustic parameters of the heat exchangers of a clothes-drying machine in both the primary and secondary flows. The measurements were performed at the outlet side of the secondary-air circuit[16].

The measuring station (Fig. 4) 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[16].

The fan is of a radial type and is fitted with a frequency inverter allowing regulation of the heat exchanger's 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[17]. The length of the straight part of the pipe in front of the orifice plate is 10 *d*, and behind the orifice plate it is 3 *d*. The pipe

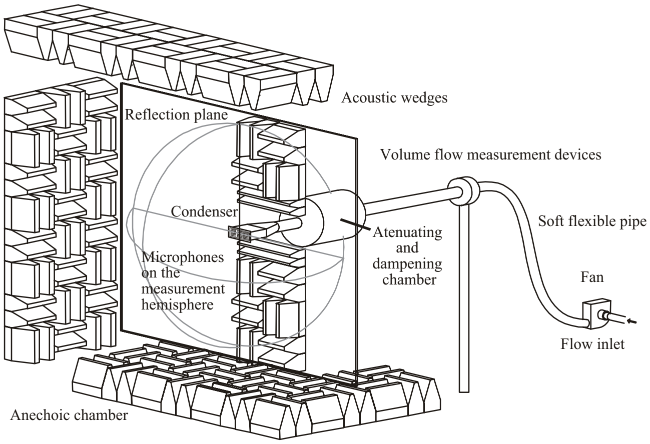

Fig. 4. Diagram of the measuring station.

diameter is $d = 150$ mm. A four-arm flow guide with a length of 2 d behind the fan was used to eliminate the vortices of the fan.

The third unit of the measuring station is an 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 that the noise generated by the fan is reduced.

The measuring unit is composed of a pipe with a length of 10 d, 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 the sound from being transmitted from the structure of the measuring unit to the environment. The measurement unit was placed in the anechoic chamber.

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

The location of the heat exchanger in the acoustic chamber was 1.5 m from the entrance shaft, 3.35 m from both side walls and 2.1 m above the ground. The acoustic pressure measurements were carried out at 10 positions on the surface of the hemisphere with a 1-m radius as a function of the flow through the heat exchanger (Fig. 4). The locations of the microphones for the acoustic pressure measurements were selected according to ISO 3745. An outlet funnel was fitted to the heat exchanger's outlet part and adjusted above the reflecting surface in the anechoic chamber.

The sound-pressure frequency characteristic (1/3 octave) was measured at every measurement point, and the sound-pressure levels were corrected in accordance with the background sound-pressure levels using Eq. 2[18],

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

where L_p is the corrected value, L_C is the measured value, and L_B is the background noise level. The background sound pressure level for each frequency band was at least 6 dB below the measured sound pressure level .

The acoustic power level was calculated on the basis of the average sound-pressure level on the measuring surface:

$$L_W = \overline{L_p} + 10 \cdot \log_{10}\left(S/S_0\right) + C \tag{3}$$

$$C = -10 \cdot \log_{10}\left(\sqrt{293/(273+\theta)} \times p/1000\right)$$

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

4.3 Measurements of the aerodynamic characteristics of the heat exchanger in the measuring station

Besides measurements of the heat exchanger's 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 the traversing method using a Dantec type 55P11 single-component hot-wire-probe 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 50,000 Hz. The 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 frequency spectra of the velocity measurements were calculated as:

$$u(f) = \int_{-\infty}^{+\infty} u(t) e^{-i2\pi ft}\, dt \ . \tag{4}$$

The velocity measurements were performed in the middle of the heat exchanger's outlet plane at a distance of 3 mm from the heat exchanger. A Hanning window was used to avoid aliasing problems.

5 RESULTS

Three different versions of the clothes dryer were analyzed. In the following they will be designated as the basic version, the modified version I and the modified version II. For the first modified version the inlet to the fan was changed, while for the second modified version a different type of heat exchanger was also selected. The modifications will be discussed in detail in the following section.

The results are presented as the total sound-power level (A-weighted), the sound-pressure frequency spectra, and the sound-intensity field. The sound-power levels for all the versions of the clothes dryers (A-weighted) are shown in Table 1. Both modified versions had reduced A-weighted total sound-power levels. The value was reduced by 9.5 dB for the left rotation and by 9.1 dB for the right rotation. This reduction is of particular importance since it appears on the label attached to the appliance, and customers make their purchasing decisions based on this information.

Table 1 - A-weighted total sound-power level of the clothes dryer variants

Clothes Dryer	A-weighted total sound-power level LWA (dB)	
	Rotation left	Rotation right
Basic Version	76.0	73.8
Modified Version I	69.3	67.3
Modified Version II	66.5	64.7

The sound-intensity fields of all three versions of the clothes dryers are shown in Figs. 5 and 6, and the sound-pressure spectra are shown in Fig. 7. The sound-power-level spectra of the heat exchangers A and B are shown in Fig. 8, and the power spectra of the velocity fluctuations are shown in Fig. 9.

6 DISCUSSION OF THE RESULTS

6.1 Basic version

The basic version of the clothes dryer was sold on the European market from 1998 to 2000. During this period the noise generated by domestic appliances has become increasingly important, and the market demanded modifications to the existing, basic version. The results of the A-weighted total sound-power-level measurements from Table 1 show high noise emissions that exceed the noise emissions of competing clothes dryers from other well-known producers.

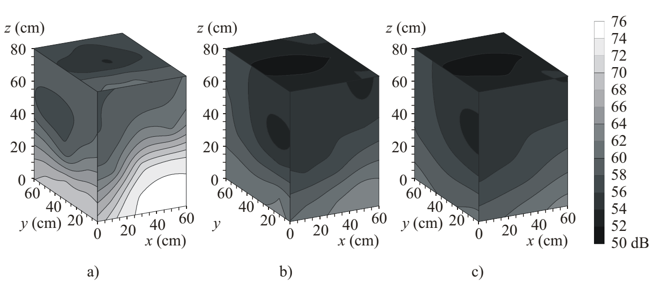

Fig. 5. Visualization of sound sound-intensity field, view from the front-left side of the appliance, basic version (a), modified version I (b), modified version II (c).

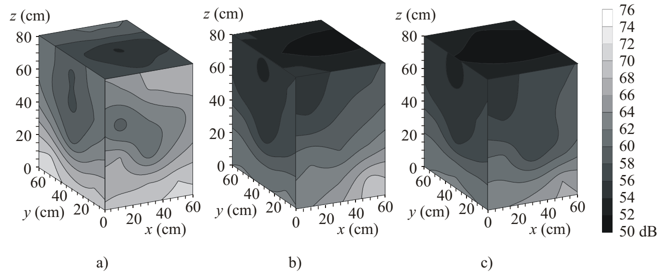

Fig. 6. Visualization of the sound-intensity field from the rear-right side of the appliance (row below), basic version (a), modified version I (b), modified version II (c).

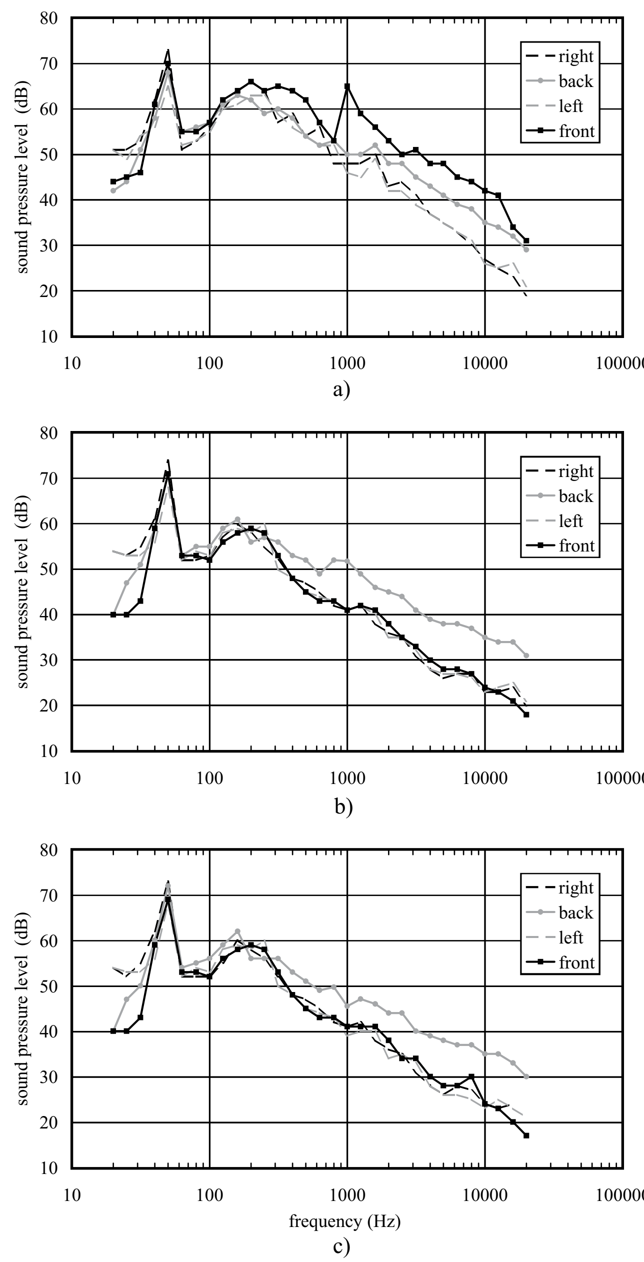

Fig. 7. Sound-pressure-level spectra of the basic version (a), modified version I (b) and modified version II (c).

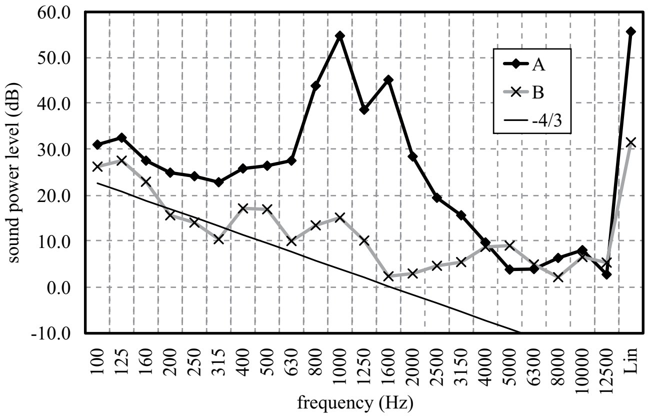

Fig. 8. Sound-power -level spectra of the heat exchangers A and B (136 m^3/h). The -4/3 straight line represents the scaling of the sound-power spectra radiated by isotropic turbulence.

The first step in the process of improving this situation was to make measurements of the sound intensity and the sound pressure on all five emitting surfaces. Using these results, the spatial positions of the peak emissions were localized, and the frequency region of the emitted noise was determined.

The maximum density of the acoustic energy flow for the basic version of the clothes dryer is in the region of the inlet grid (the inlet of the secondary-flow fan), shown in the lower-right corner of Fig. 5a. Because it is at this position that the secondary flow enters through the grid into the secondary-flow fan inlet, the increased sound power at this position is connected with the noise generation at the fan rotor and to the turbulent effects at the inlet and the outlet of the fan. It must be emphasized, however, that the sound emission through the inlet grid is freely radiated into the environment and that the additional noise generation occurred at the grid itself.

The same conclusions are obtained when the sound-pressure level spectra on four vertical clothes-dryer surfaces are analyzed. From the sound-pressure level spectra diagram in Fig. 7a, the dominant frequency is observed at 50 Hz, due to the rotating of the fan, with additional broadband turbulent fan noise observed from 200 Hz to 1600 Hz. The peak of the sound power at 50 Hz is less audible, since the sound-pressure level in Fig. 7 are not A-weighted. Besides, in the frequency region around 1 kHz the front-surface power spectra significantly deviate from the other sound-pressure level spectra of the dryer's side walls. This deviation corresponds to the multiple of the rotational speed and the number of fan blades (20 blades/fan). Therefore, one can conclude that the increased sound-power level at the front surface of the dryer is correlated with the aerodynamic noise generated by the fan rotor, and to the noise generated by the turbulent effects of the dryer's flow tract.

The structural noise was assumed to be of minor importance. The unbalance of the fans is low enough, and other rotating parts, e.g., the drum, rotate much more slowly and do not contribute to the overall noise. Since intrinsic noise sources are difficult to identify and the characterization of these sources and the transmission structure is essential, the appliance was designed as much as possible in accordance with the recommended practice for the design of low-noise machinery and equipment[19,20]. Measurements of the local velocity profile and the intensity of the turbulent velocity fluctuations in the input channel to the secondary flow fan were also performed in Širok et al, 2000[5]. These measurements show that the velocity profile in the inlet channel is homogeneous, while at the outlet of the fan it is non-symmetrical, as is usual for the outlet profile of the radial fan. The measurements also show that the velocity fluctuations in the inlet and the outlet channels exhibit turbulent behavior and that the peak of the velocity fluctuations in the inlet to the fan is below 250 Hz. The fluctuations at the outlet of the fan have two peaks: the first below 250 Hz; and the second, much smaller peak, around 1000 Hz[5].

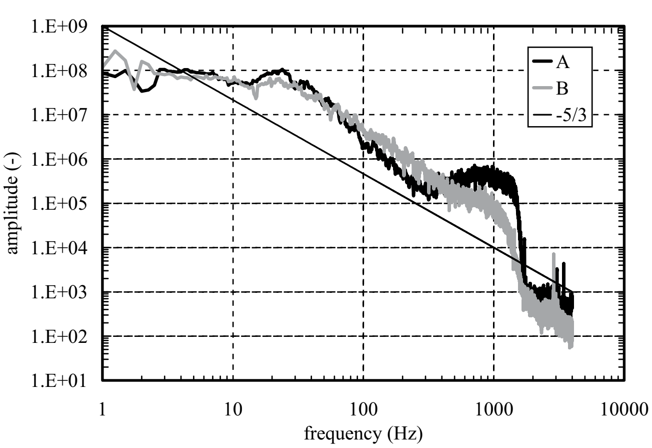

Fig. 9. Power spectra of the velocity fluctuations. The thin, straight line represents the -5/3 turbulence decay law.

6.2 Modified version I

Based on the analysis of the aerodynamic and acoustic characteristics of the basic version of the clothes dryer, modifications were made to the construction of the dryer to produce the modified version I. The secondary fan's inlet region was modified, additional inlet-guide elements were inserted into the flow tract, and the pressure channel at the fan junction was corrected.

However, the main improvement was the correction of the secondary flow fan inlet. The inlet slots on the front wall of the basic version were completely closed off and replaced with axially symmetrical slots in the inlet duct so that the modified version drew all inlet air from the inside of the cabinet (Fig. 3, right). The new slots were designed so that their cross-section was slightly larger than the cross-section of the original slots in the basic version. This was done so that the entrance to the fan in the modified version is now radial, and the flow enters the fan inlet radially; this also prevents the direct radiation of the noise into the environment. Sound-absorbing materials were also installed on the inner side of the clothes dryer's structure, where the slots were located in the inlet of the basic version. The second modification is connected with a correction to the outlet channel's geometry: the transitions of the pressure channel and the guide vane were smoothed. With a modification of the geometrical, and consequently the flow, properties the sound-power level of the modified version I of the dryer was decreased from 76.0 to 69.3 dB (A- weighted) for left rotation, and from 73.8 to 67.3 dB for right rotation, as shown in Table 1.

The sound-intensity field at the dryer's front wall, from Fig. 5b, shows a significant reduction in the sound intensity in the region of the inlet grid at the bottom right-hand part of the front side. In contrast, the sound-intensity field of the rear side in Fig. 6b shows little reduction in comparison with the basic version. The most important sources identified in the modified version I remain on the rear side of the clothes dryer, i.e., the exit of the secondary-flow channel behind the heat exchanger, and to a much smaller extent the primary fan in the primary channel, also at the rear side. The primary airflow is a closed circuit, and at the rear side of the clothes dryer the flow channels are made of relatively thick metal sheets, which explains the lower sound intensity at the location of the primary-flow fan.

The sound-pressure-level spectra in Fig. 7 also show that the rear side of the clothes dryer is the main source of the noise in the modified version I. In addition to this we see that at the front side the peak in the frequency region at around 1000 Hz is no longer present in the modified version I. Much smaller peaks in the same frequency region remain after the modifications to the rear side of the dryer.

6.3 Modified version II

After encouraging noise-reduction levels with the modified version I, further attempts were made to reduce the noise. Further modifications were performed for version II, which also includes all modifications realized in version I. An attempt was made to improve the primary and secondary fans, but unfortunately this was not possible for the following reasons.

The most significant noise source of the modified version I is the rear side of the appliance (Fig. 6b and Fig. 7b). Here, two sources were identified, the first is the outflow of the secondary flow through the heat exchanger, and the second is the fan of the primary-flow circuit. The primary circuit is closed, while the secondary circuit is open. The primary fan has to be manufactured from steel due to the proximity of the heater, and to keep production costs low the blades are straight, non-curved, non-profiled and are pressed from flat steel plate. Reduced noise emissions could be achieved with a primary fan with curved and profiled blades[21,22]. The application of curved and profiled blades would, however, require a change to the manufacturing process, and a casting process would have to be implemented. Unfortunately, this proved to be too expensive and was not accepted by the management.

We have proceeded with a further reduction of the noise generated in the secondary circuit by testing different designs of heat exchanger. The type of heat exchanger was changed, which provided a further reduction in the overall A-weighted sound power level by 2.6 dB, from 67.3 dB to 64.7 dB for the right rotation, and by 2.8 dB, from 69.3 dB to 66.5 dB for the left rotation. The control software of the clothes dryer was pre-programmed so that the fan rotates mostly in left rotation, and only from time to time is the rotation changed to prevent clogging of the laundry. For similar reasons to those mentioned above, changes to the fan of the secondary circuit were not accepted by the management.

The most important parameter of a heat exchanger is its energy efficiency. For this reason we tested a few different heat exchangers and two with equal energy characteristics were selected. From among these the most appropriate heat exchanger in terms of acoustic properties was selected. Several tests were performed to determine the interdependence of the acoustic and aerodynamic properties of the selected heat exchangers. The measurements were performed using the measuring station, as described in section 4.2.

The heat exchanger is the last element of the secondary-flow tract and is located directly before the exit of the secondary flow to the environment. It is therefore important that it screens the noise from the fan and does not cause significant noise generation due to turbulent effects in the heat exchanger's fins. New designs of heat exchangers for clothes dryers are more prone to this since heat exchangers designed to keep costs low tend to be small and so the velocity is high.

The basic version and modified version I were equipped with a type-A heat exchanger; however, this was changed to a B-type heat exchanger in the modified version II. From Fig. 6c we can see that most of the reduction was achieved in the lower rear part of the appliance. The results in Fig. 7 show the reduction of the sound-pressure level in the frequency region around 1 kHz at the rear side of the clothes dryer.

The measurements of velocity and of the acoustic properties of the heat exchangers were performed to establish a guide for the selection of the heat exchanger according to the aeroacoustics theory that connects turbulent fluctuations and noise generation, as introduced in section 3. The power spectra of the heat exchangers radiated acoustic power scales as $\omega^{-4/3}$ (Fig. 8), which corresponds to the scaling derived from the sweeping hypothesis of time correlations by Rubinstein and Zhou[6,16]. In addition to this the sound-power-level peaks are superimposed on the basic spectra scaling (Fig. 8) for the type-A heat exchanger, and are much less pronounced for the B-type heat exchanger. The intensity of the superimposed peaks also increases with increasing volume flow through the heat exchanger. Fig. 8, however, shows the volume flow that was measured for the right rotation of the fan during normal operation.

Due to a different flow-channel design the heat exchanger B shows lower sound power levels than heat exchanger A (Fig. 8). Both types of heat exchanger demonstrate a tone component at 1000 Hz, which for heat exchanger B is nearly inaudible due to its low level.

The results of the measurements of velocity fluctuations are shown in Fig. 9. The power spectrum of heat exchanger A corresponds to -5/3 of the turbulence decay law[7] in the region from 30 Hz to 100 Hz; in the frequency region from 200 Hz to 400 Hz a small dip is shown; and in the frequency region from 400 Hz to 2 kHz a pronounced peak was detected. We assume that this peak of velocity fluctuations is responsible for the high levels of the radiated sound-power level shown in Fig. 8. The power spectra of the heat exchanger B, however, does not show such a dip and peak of fluctuations, and better corresponds to the -5/3 turbulence decay law.

We have found that the power spectra of the velocity fluctuations in heat exchangers A and B do not have similar behavior in the frequency region where a slight noise reduction was achieved in version II. Since the noise in the case of heat exchanger A is presumably induced by velocity fluctuations at the outlet of the heat exchanger, a measurement of the velocity fluctuations of the optional heat exchanger under test to determine its energy properties can also be used as a measure of acoustic acceptance. Establishing the energy properties requires a measurement of the local velocity fluctuations, which must be approximately of the same intensity over the entire cross-section of the secondary flow to enable proper condensation over the heat exchanger's entire volume.

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

In future work it is important to determine the inverse correlation time of the motions of any given spatial scale in connection with the frequency distribution of the acoustic energy of an arbitrary heat exchanger[23] .

5 CONCLUSIONS

The development of large household appliances in recent years is increasingly focused on noise reduction. In the presented work we estimated the potential noise sources on particular clothes dryer that are connected to the aerodynamic characteristics of both the installed fans, the heat exchanger and to the geometry of the flow tracts. We estimated the influence of the noise sources of the secondary circuit at the outlet side of the heat exchanger and the aerodynamic characteristics of the heat exchanger.

The maximum sound intensity was identified in the region of the inlet grid of the secondary-flow fan on the lower right-hand part of the front side of the basic version of the clothes dryer. Therefore, it can be concluded that the increased sound intensity on the front side of the dryer is connected to the aerodynamic noise, generated by the fan rotor, and the noise generated by the turbulent effects of the dryer's flow tract. The correction of the secondary fan's inlet was performed in such a way that the inlet slots on the dryer's front side were replaced by axially symmetrical inlet channels, which make possible radial air suction. Therefore, the direct emission of sound into space was prevented. The improved sound characteristics are obvious from the overall sound-power level and from the distribution of the sound intensity on the dryer's front side.

Further modifications included a new heat-exchanger design. The total sound-power level is different for the various heat exchangers and generally increases with the volume flow rate. The maximum sound power of heat exchanger A is emitted in the frequency interval around 1000 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, a measurement of the velocity fluctuations can be used as a measure for acoustic acceptance of the heat exchanger during the design of its energy properties, which include measurements of the local velocity fluctuations.

After making all the modifications, the A-weighted total sound-power level was reduced from 76.0 to 66.5 dB for left rotation, and from 73.8 to 64.7 dB for right rotation. The future selection of heat exchangers for clothes dryers will have to focus not only on energy efficiency but also on the appropriate acoustic characteristics of the heat exchanger.

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