FACTA UNIVERSITATIS
Series: **Working and Living Environmental Protection** Vol. 10, N° 1, 2013, pp. 13 - 26

SOUND INTENSITY MEASUREMENT AS A DIAGNOSTIC TOOL FOR THE NOISE REDUCTION OF DOMESTIC REFRIGERATORS

UDC 534.61:613.644:64.06

Nikola Holeček[1], Momir Praščević[2]

[1]Gorenje, d.d., Environmental protection College, Velenje, Slovenia, EU
E-mail: nikola.holecek@gorenje.si
[2]University of Niš, Faculty of Occupational Safety of Niš, Serbia

Abstract. *In addition to highly complex functional features and low energy consumption, the silent operation of a domestic appliance is an essential design component of a modern integrated refrigerator and freezer. This is because noise emission is an extremely important differentiating factor when choosing a new appliance; therefore, manufacturers of exceptionally silent appliances usually advertise their low noise emission using terms such as "super silent operation". This article presents a procedure to obtain the desired noise level for such an appliance. All measurements were conducted in an acoustics laboratory, under free-sound-field conditions. The level of noise in the different development phases was represented by the total sound-power level, the frequency spectrum and the use of sound imaging analysis. The critical sources of internal noise were determined based on experimental results and based on this the necessary design modifications to reduce the sound-power level from 57.6 dB (A) to an acceptable 44.5 dB (A) were made.*

Key words: *noise source, sound intensity measurements, noise source location, noise reduction, domestic appliance*

1. INTRODUCTION

To meet the needs of the most demanding appliance markets, large built-in **I**ntegrated **R**efrigerator and **F**reezer (IRF) units with a width of 0.75 m, height of 2 m and weighing over 230 kg have been developed. Considering the size and weight of the appliance, all the vital parts of the refrigeration system's components i.e., the compressor compartment containing two compressors, a dynamic condenser and either one or two fans are located in a pull-out drawer at the bottom of the appliance accessible from the front side, so that they can be readily accessed without having to move the appliance.

Received June 10, 2013

The upper compartment of the appliance is intended for refrigeration (fresh food, FF), the middle compartment is a freezer (bottom freezer BF) with an automatic ice maker, while the bottom compartment is a multifunctional drawer (convertible drawer, CD), which can be used either as a freezer, a refrigerator or a wine storage unit. Such a wide range of functions requires a working temperature of -21.1 to +12.8 °C and in order to achieve this, an innovative cooling system was developed.

The appliance's features control the thermal loads within the unit by varying the cooling volume flow by means of hatches, fans and by varying the cooling power of the compressors. For the purpose of cooling two separately adjustable compartments, a multi-channel system is installed to distribute and inject air into the interior, ensuring a homogenous temperature field within the internal spaces. The majority of these design solutions were first verified by means of numeric simulations which, to some extent, provided the answers to the questions regarding how to ensure that an appropriate temperature field exists in each of the individual compartments.

Here we present our comments on selecting appropriate design concepts. A design with the lowest and continuous airflow velocity and acceleration should give the best results in terms of acoustic properties. Continuous operation in general produces less noise than intermittent operation. For the given operating mode, the noise of the appliance can be reduced by making the correct design choices. Materials, shape, position, number of elements, dimensions, structure and how these elements are connected can all have a significant impact on noise emissions and if chosen correctly, can significantly reduce vibrations and sound propagation. In this paper we present a development procedure. This development procedure included noise measurements, data analysis and replacement of parts, representing important noise sources, leading to a significant reduction in acoustic emissions.

2. Analysis of Noise Sources

In order to reduce noise emissions, we adopt a methodology of identifying and eliminating the critical sources of noise in the earliest possible stages of the design phase.

The first step involves determining the main noise sources within the appliance and making a priority list or plan. Once the main sources have been determined, they need to be characterized and a more detailed analysis of the actual noise mechanisms conducted. The next step requires an analysis and a description of how noise propagates out from its source and how it is then transmitted through the structure onto propagation surfaces. This is then followed by an analysis of how it propagates from these surfaces and then determining their contributions to the sound pressure level at inlet points. Finally, an estimate can be made of which combination of noise abatement measures is the optimal one, [1]. Using an appliance prototype, we have the opportunity to verify, with actual measurements, the operation of the actual appliance design.

The active components that generate noise in an IRF appliance are the following: the centrifugal fan in the FF-compartment, the axial propeller fan in the BF-compartment, and the axial fan and two compressors in the compressor compartment (Fig. 1).

Fan noise depends on fan type: radial or axial, and on the operating conditions. If the impeller is well balanced and fixed in place to prevent any transmission of vibrations, then the generated noise changes in relation to the operating settings. Fans are usually designed so that they emit the least amount of noise at their optimal operational setting, while any deviation from this usually increases the amount of noise being generated. Volume flow rates below optimal induce vortices and flow separations that lead to an increase in pressure pulsations, which are then reflected as broadband noise at frequencies lower than the rotational frequency of the impeller. At volume flow rates above optimal, broadband noise is generated [2].

Yet another significant noise source are the stationary channels and the compressor compartment grille where it was not possible to achieve optimal flow conditions during the design process due to technical and spatial limitations. In places where the channel cross-section is either enlarged or diminished, flow separation or recirculation can occur, leading to pressure and flow velocity pulsations and to the generation of turbulent noise, [3].

Fig. 1 Elements of a compressor compartment (unit) – basic design: dynamic condenser (1), axial fan (2), two 'variable speed drive' (VSD) compressors (3), air inlet grille (4), air outlet grille (5)

3. METHODOLOGY DESCRIPTION

Reduction of noise generation in domestic appliances may only be achieved after sufficient data concerning noise sources and noise transfer routes is obtained. This information is also dependent on the frequency interval. Source analysis can be performed based on the spectrum of radiated sound combined with knowledge of the potential internal noise sources, rotational speed, and the number of blades on the fan etc. Peaks in the emitted noise pressure spectra can be attributed to selected relevant noise sources.

In the cases where information from emitted noise spectra does not bear information about the noise sources, the method of (reversed) elimination of noise sources is available. At every step, we determined the total energy flow (sound power) of the appliance and record the sound spectra, [4]. Partial contributions are also analyzed by means of visualizing the sound field i.e., creating a sound image, to establish the actual point from which a sound originates [5-9]. Although this can be viewed as a complex problem, it is feasible by using measurement techniques for noise source identification (NSI). In this study, we measured the density of energy flow of the sound waves (sound intensity) and used the NSL 7681 B&K (noise source location) program for sound imaging processing.

4. Description of Used Measurement Procedures

As mentioned in section 2, engineering methods were used in the process of noise reduction in the IRF appliance. We conducted measurements to determine both the sound power level and to visualize the sound field surrounding the unit.

4.1. Measurement of the sound power level

Measurements of the sound power level required an appropriate acoustic environment in the form of a free sound field which allows sound waves to propagate freely without reflections. For laboratory testing, we used a semi-anechoic chamber to ensure good repeatability and low measurement uncertainty. To provide a free sound field above the reflective surface, the testing chamber must:

- be of an appropriate size,
- have a high sound absorption capacity across the entire monitored frequency range,
- be without acoustically reflective surfaces and obstacles with the exception of the floor in a semi-anechoic chamber,
- have an adequately low level of background noise.

Our semi anechoic chamber had a free-space volume of 220 m^3 (7.8 m × 6.7 m × 4.2 m). Its walls and ceiling are coated with sound-absorbing polyurethane foam wedges. The wedges are 80 cm long and a 5-cm air gap exists between the wedges and the walls (the Helmholtz resonator). The total length of the wedges and the air gap was $\lambda/4$, where λ is the wavelength of the sound corresponding to the frequency center of the lowest monitored frequency band, [10]. The shape and dimensions of the wedges were determined based on the measurements in Kundt's tube. To prevent the absorption coefficient, which was at least 0.99, from being dependent on the wave's angle of incidence, the wedges were installed mutually rotated by 90^o. All the measurements in the anechoic chamber were made according to ISO 3745 [11]. They reveal that the lower frequency limit is 100 Hz, the background A-weighted sound pressure level is 13 dB and the discrepancy from legality "1/R" is between 90 Hz to 10 kHz, which is admissible according to the standard [11].

All the objects and measuring equipment, except the sensors and microphones, were located outside the chamber. Hollow tubes for cable access were filled with sound absorbing material to prevent the transmission of vibrations.

To make source acoustic-power measurements according to the absolute method, a measuring system which consisted of 20 Falcon TM Range ½"-type 4189 (Brüel & Kjær) microphones and 20 Delta Tron-Type 2671 microphone pre-amplifiers was used. To determine the sound power level, we used the PULSE measuring system manufactured by B&K, and for the analysis a standard multichannel real-time analyzer 3560D. The measuring of the power level in the free sound field above the reflective surface in a semi-anechoic chamber was conducted in compliance with the requirements of the international standard ISO 3745:2003 [11]. The sound pressure level was measured simultaneously at 20 measuring points spirally distributed around a virtual hemisphere. The entire measuring procedure was conducted using the PULSE program with the installed application 7771 for sound power level measurement. The results were recorded and transferred via an ActiveX link into a standard Excel spreadsheet. All the measuring equipment was calibrated using a piston phone and a sound-intensity calibrator with a valid certificate of calibration. All the measuring system elements, including microphones and cables, complied with Instrument Class 1 in accordance with IEC 61672-1:2002 [8]. The filters used also complied with Instrument Class 1 in accordance with IEC 61260:1995 [12, 13].

4.2 Measurements of sound intensity

By measuring sound intensity, our aim was to visualize the density of the energy flow of the sound waves on all five sides of the IFR unit. To that purpose, we selected 86 measurement points. The measurement plane was 60 mm away from the appliance's surfaces. The measurements were conducted at discrete points in the near field of the whole unit. Afterwards we measured only the vector field of sound intensity at the front side of the compressor compartment, where most of the noise sources are located. Sound-intensity measurements of the IRF acoustic characteristics were performed using a two-channel analyzer with a sound-intensity probe placed at a discrete distance and using the Noise Source Location program. The purpose of this program is to help locate the source of the noise emitted from the IRF unit [14, 15]. The sound-intensity probe was positioned by a computer-controlled traversing system. Since the absolute precision of the Class I sound power measuring method had been used for the precise calculation of energy flow, we also used it for visualization i.e., sound imaging, which is the main advantage of this technique.

5 Measurement Results

5.1. IRF unit during operation with different internal noise sources

To identify the noise originating from the compressors situated in the compressor compartment, we first measured the sound pressure level at a distance of 1 m from the compartment (Fig. 2). However, it was later realized that a higher level of noise was generated by the centrifugal fan located in the FF compartment. Fig. 2 shows the frequency spectrum of the sound pressure on the front side of the compressor unit compartment (basic design). The peaks in the spectrum are the harmonics associated with the rotating frequency of the fan and the frequencies that are generated on the blades of the axial fan in the compressor compartment.

In a follow-up, we conducted measurements of the total sound power level of the IRF unit under the following operating conditions:

- only the propeller fan in the BF-unit,
- only the cooling compressors,
- only the axial fan in the compressor unit,
- only the centrifugal fan in the FF-unit.

The results are shown in Table 1.

The results in Table 1 are presented as total sound-power level (A-weighted). Table 1 reveals the reasons for the increase in the IRF unit noise level. The fan operating in the BF unit (Table1 - 5a) and the operation of both compressors (Table 1 - 5b) is not responsible for increased IRF unit noise level. Instead the dominant noise sources are (1) the centrifugal fan in the FF compartment, which emits a sound-power level of 56.7 dBA (Table 1- 5d) and (2) the axial fan in the compressor compartment, which produces a sound-power level of 50.2 dBA.

The sound power frequency spectrum of the centrifugal fan in the FF-unit and the entire system in operation are shown in Fig. 3 and Fig. 4, respectively.

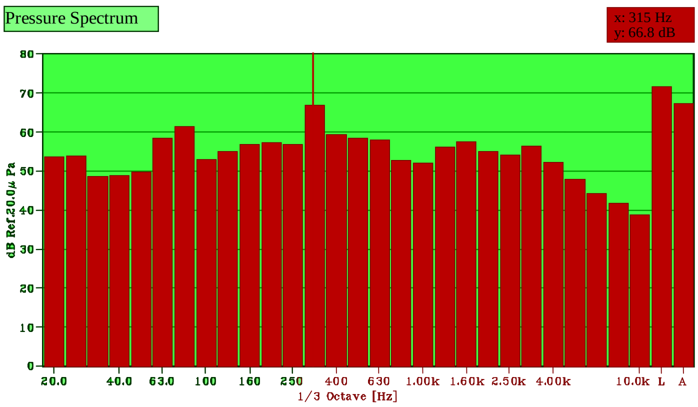


Fig. 2 Frequency spectrum of the sound pressure of an IRF unit recorded for the preliminary characterization of internal sound generators at 1 m in front of the compressor compartment.

Table 1 The total sound power level (A-weighted) of the IRF unit, operation with different internal noise sources.

Operation of the appliance	L_{WA} (dBA)
a) only the propeller fan in the BF-unit	41.1
b) both compressors (variable speed drive, VSD) in the compressor unit	40.7
c) the axial fan in the compressor unit	50.2
d) the centrifugal fan in the FF-unit	56.7
e) the entire system in operation	57.6

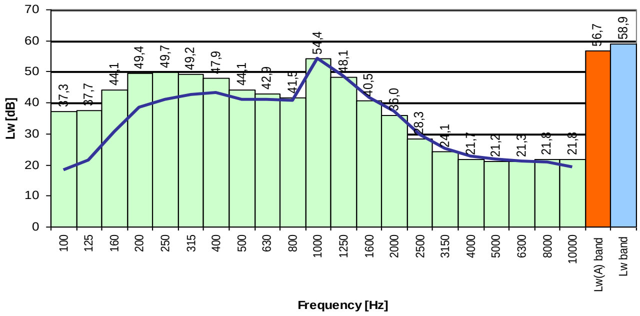


Fig. 3 The frequency spectrum of the sound power of the centrifugal fan in the FF-unit

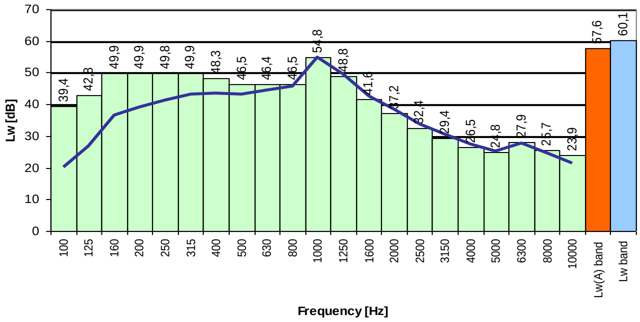


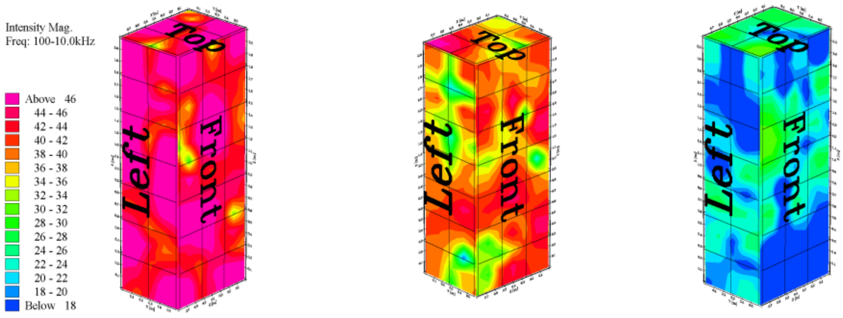
Fig. 4 The frequency spectrum of the sound power of the entire system in operation

The dominant frequency occurs at 1000 Hz, due to the rotating fan blades, with additional broadband turbulent fan noise observed at 200 Hz to 1600 Hz (Fig. 3 and Fig. 4). Therefore, the increased sound-power level at the rear surface of the IRF unit is related to the aerodynamic noise generation by the centrifugal fan situated at the rear of the unit.

5.2. Replacement of individual active components

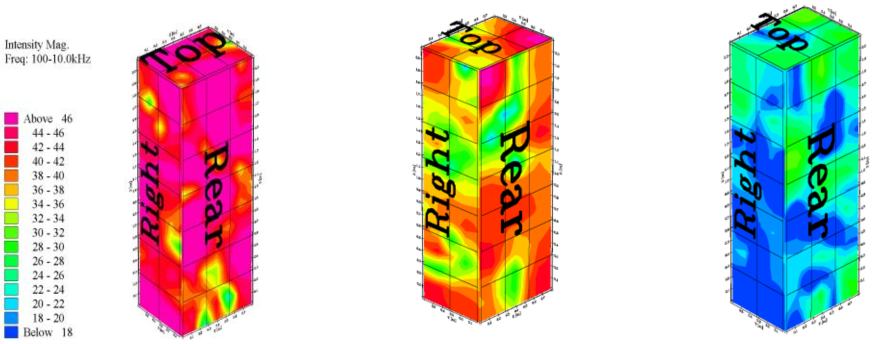
The results presented in Section 5.1 reveal the dominant sources of noise in the IRF unit. In the follow-up, we confirmed this by measuring the sound intensity and creating a sound image, which is shown in Fig. 5 and Fig. 6, respectively. A sound image was recorded in the following three cases:

a) all the internal sources of noise are in operation (axial fans in the compressor compartment, the centrifugal fan in the FF compartment and both compressors),
b) all the elements are in operation, except the centrifugal fan in the FF-compartment,
c) all the elements are in operation, except the centrifugal fan in the FF-compartment and the axial fan in the compressor compartment.



a) b) c)

Fig. 5 Sound intensity field image of the IFR, left front view: (a) all elements in operation, (b) centrifugal fan in FF-compartment switched off, (c) centrifugal fan in the FF-compartment and fan in the compressor compartment switched off.



a) b) c)

Fig. 6 Sound intensity field image of the IFR unit, right rear view: (a) all elements in operation, (b) centrifugal fan in FF-compartment switched off, (c) centrifugal fan in the FF-compartment and axial fan in the compressor compartment switched off.

The maximum density of the acoustic energy is in the region of the centrifugal blower, shown on the left side of Fig. 5a and Fig. 6a. The images clearly show how the noise is generated by the centrifugal fan and at the elements of the connecting flow tracts where local flow velocity is high.

The second important noise source is the axial fan in the compressor unit, which is positioned on an extractable holder (Fig. 5b and Fig 6b). The same conclusions are obtained when the sound-power level and sound-power level spectra are analyzed (Table 1c).

The structural noise was assumed to be of minor importance. 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 [10].

The measurements show that to radically reduce the noise level, the centrifugal fan in the FF-unit should be replaced by a more appropriate design (Fig. 7), and the axial fan in the compressor compartment should also be replaced with two axial fans to allow a more efficient flow-through of the air-field in the compressor compartment, and thus a lower level of sound emission. This was also confirmed by measuring the vector field of sound intensity at the front side of the compressor compartment (Fig. 9).

Modifications of the IRF unit were performed in three steps. First, a centrifugal fan in the FF-unit was replaced with a more appropriate model, second, the fan in the compressor compartment was replaced with two fans and third, the rotational speed of these two fans was reduced.

The optimized design of the compressor unit is shown in Fig. 8.

The measurements were repeated. In each step, we determined the total sound-power level and corresponding frequency spectra under the following operating conditions:

1. operation of an unmodified IRF unit,
2. operation of the IRF unit with a new centrifugal fan installed in the FF-unit,
3. operation of the IRF unit with a new centrifugal fan installed in the FF-unit and two axial fans installed in the compressor compartment,
4. operation of the IRF unit with a new centrifugal fan installed in the FF-unit and two axial fans installed in the compressor compartment, operation with reduced rotational speed.

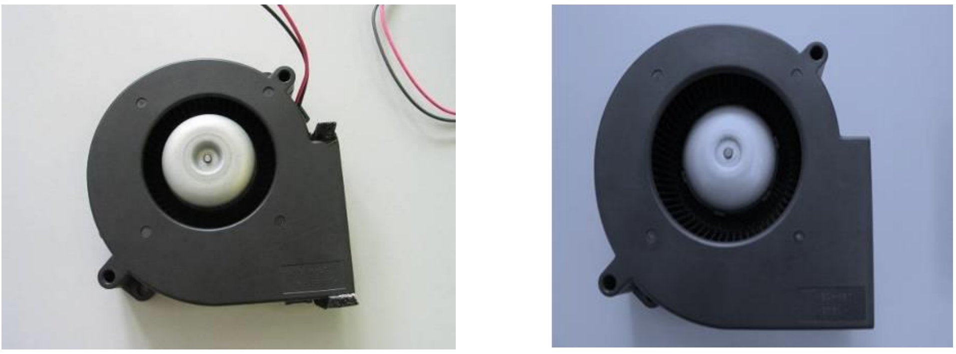


Fig. 7 Centrifugal fan in FF compartment: original (left) and new (right)

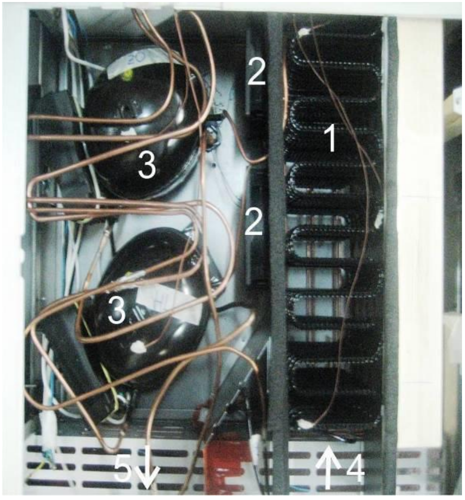


Fig. 8 Elements of the compressor unit (optimized design): (1) condenser, (2) two axial fans, (3) two variable speed drive (VSD) compressors, (4) inlet of air flow, (5) outlet of air flow

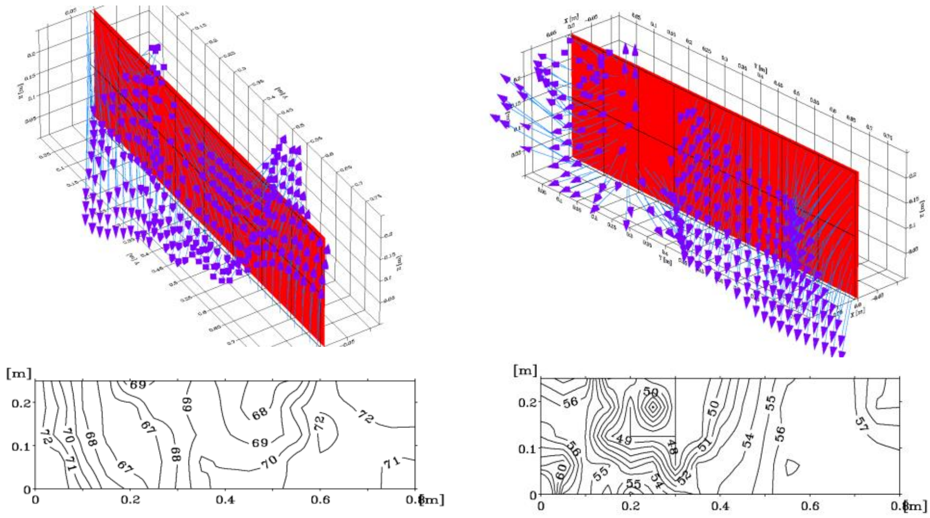


Fig. 9 Vector and surface representation of sound intensity at the front side of the compressor compartment: the left side indicating the initial situation, the right side showing the situation after modification (installation of two axial fans); frequencies from 200 Hz to 10 kHz are shown.

Table 2 shows the obtained results. The reduction of rotational speed was justified with an adequate cooling power and distribution of temperature. Rotational speed was reduced from 2150 to 1800 rpm.

Fig. 10 to Fig. 12 show the frequency spectrum of the sound power of the entire system in operation after modifications of the IRF unit.

Table 2 The total sound power-level (A-weighted) of the modified IRF unit, operation with different noise sources.

Operation of the appliance	L_{WA} (dBA)
1. Operation of an unmodified IRF unit	57.6
2. Replacement of the existing centrifugal fan in the FF-unit with a new centrifugal fan	51.1
3. The axial fan in the compressor chamber replaced with two axial fans	46.1
4. Reduction of the rotational speed of both axial fans in the compressor compartment	44.5

Acoustic measurements of individual elements (as described in Section 5.1) have shown that some internal noise sources require either modification or replacement. In addition, the total sound-power level significantly exceeded the expected and desired values and measurements of sound intensity and sound imaging during the three operating regimes, showing how partial noise sources are dominant.

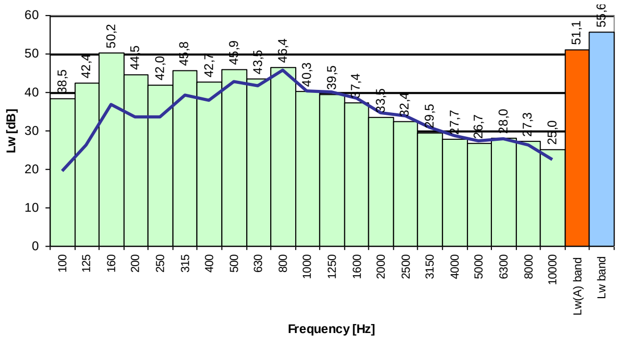


Fig. 10 Frequency spectrum of the sound power of the entire system in operation after replacement of the existing centrifugal fan in the FF-unit with a new centrifugal fan

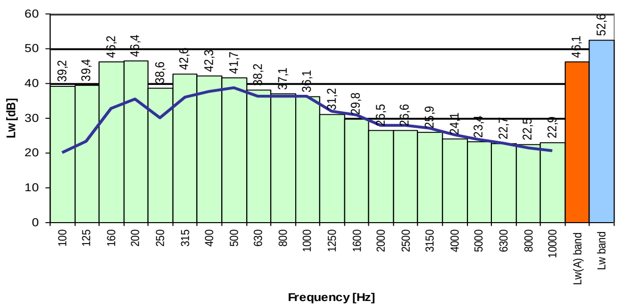


Fig. 11 Frequency spectrum of the sound power of the entire system in operation after the replacement of the axial fan in the compressor chamber with two axial fans

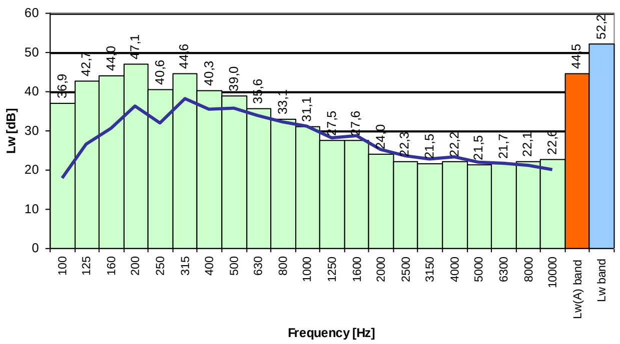


Fig. 12 Frequency spectrum of the sound power of the entire system in operation after the reduction of the rotational speed of both axial fans in the compressor compartment

The original centrifugal fan required replacing with a more appropriate one. As seen from Table 2 (case 1), the old centrifugal fan features both narrow band and broad band noise sources. The noise is generated at the installed fans 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 their rotational speed, i.e., the blade-passing frequency and its harmonics, and the broadband noise or turbulence caused by the aerodynamic fluctuations of the forces applied by the fluid on the rotor blades of the fan. This unsteady loading forms a source system resulting in the 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, [10]. Several mechanisms of noise generation coexist, without it always being possible to provide evidence of a dominant source. Therefore, to better understand broadband-noise generation mechanisms, precise knowledge of the flow behavior over the impeller is vital.

In replacing the original centrifugal fan with another more appropriate one, forward sweeping blades were selected in place of the original backward swept blades. The resultant reduction in broadband noise is due to the backward sweeping blades that produce less tip vortex interaction with neighboring blades. Reduction of the narrow band noise sources was achieved because of the angled leading end trailing edge of the fan blades. Improved aerodynamics also means that the new fan rotates more slowly. The result is a reduction in the sound-power level from 57.6 dB(A) to 51.1 dB(A). Furthermore, the centrifugal fan in the BF-unit no longer contributes significantly to the total sound-power level of the IRF unit. The same applies for the refrigerator compressors. In our further noise reduction process, the total appliance noise was mainly dependent on the noise generated by the fan in the compressor compartment. However, the reduction of the noise emitted from the compressor unit required additional research work, which is described in the work of Holeček et al. [17].

After the appropriate modification/replacement and installation of the two axial fans, we achieved a sound-power level of 46.1 dB (A).

The frequency spectrum of the sound pressure at the front side of the compressor unit of the modified (optimized) design shows an improvement in 1/3 of the octave for about 5 dB of the sound pressure level (Fig. 13).

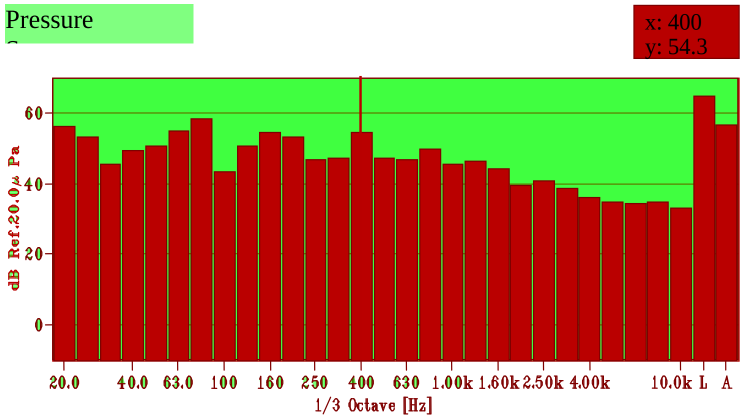


Fig. 13 Sound pressure level spectra of the compressor unit modified design

This last measurement shows that we can achieve a sound-power level of 44.5 dB(A) if we reduce the rotational speed of the compressor compartment fans from 2150 rpm to 1800 rpm. We verified whether such a modification would still acceptable according to the required cooling parameters, and we obtained a positive response. Thus we had finally achieved a significant lowering of the sound-power level by 13.1 dB (A), from the initial 57.6 dB(A) to 44.5 dB(A).

6. Conclusion

Development trends of large home appliances continue to indicate the need for quieter operation. We conducted acoustic measurements on an IRF unit, with the aim of reducing its sound emissions. Under free-sound-field conditions, we determined the total and third-octave level of sound power, and by measuring sound intensity, we were able to perform sound imaging. By measuring the sound intensity on all five of the emission surfaces of the IRF unit, we located the main sources of noise emissions. From this, we were able to conclude that the increased sound intensity at all five sides of the IRF unit is related to the aerodynamic noise generated by the rotor of the centrifugal fan in the FF-compartment and to the noise generated by the turbulence in the airflow distribution channel. The reason for this was the installation of an inappropriate type of centrifugal fan in the FF-compartment. By replacing this with a more appropriate design, we managed to lower the sound-power level from 57.6 dB (A) to 51.1 dB (A).

Target noise values for the IRF unit were still lower than the achieved values. Therefore, we took the extra step of replacing the fan in the compressor compartment. Optimal flow of the air was obtained by replacing the original fan with two axial fans. This resulted in a further lowering of the noise level from 51.1 dB (A) to 46.1 dB(A). As the target value was below 45 dB (A), we took an additional measure of reducing their rotational speed from 2150 to 1800 rpm. We checked the impact of this action on the functional and energy properties of the appliance. After obtaining a positive response, we decided to implement this change and thus achieved the target value of 44.5 dB (A).

REFERENCES

1. ISO 11688–2 Acoustics, "Recommended practice for the design of low-noise machinery and equipment, Part 2: Introduction to the physics of low-noise design."
2. Beranek, L. (1988): Noise and Vibration Control. Cambridge: Institute of noise control engineering.
3. Crocker, M. J. (1997): Noise and vibration control. John Wiley &Sons, Inc.
4. Doebelin, E. (1990): Measurement system. New York: McGraw-Hill Book Company.
5. Fahy, Frank J. (1989): Sound Intensity. London: Elsevier Science Publisher Ltd.
6. M. Prascevic, D. Cvetkovic, D. Mihajlov, *A new method of the noise emission declaration based on sound intensity measurements* Facta Universitatis, Series: Working and Living Environmental Protection, Vol. 2, No. 5, 2005
7. M. Praščević, D. Cvetković, *Diagnostics of Acoustic Processes by Intensity Measurement*, Facta Universitatis, Series: Working and Living Environmental Protection, Vol. 1, No. 2, 1997
8. M. Praščević, M. Milošević, D. Cvetković, *Determination of Absorption Characteristic of Materials on Basis of Sound Intensity Measurement*, Journal de Physique IV, Colloquia C5, Vol. 4, pp. 159-162, 1994
9. M. Prascevic, D. Cvetkovic, D. Mihajlov, *Noise emission declaration by sound intensity method*, in Proc. Of Eleventh International Congress on Sound and Vibration (CD), 2004, pp. 3175-3182
10. Holeček, N. *Acoustical laboratory in Gorenje*, Invited Papers, Proc. First Congress of Slovenian Acoustical Society, Portorož, Slovenia (1998), pp. 369-374,
11. ISO 3745 2003 Acoustics - Determination of sound power levels of noise sources - Precision methods for anechoic and semi-anechoic rooms, International standard (International organization for standardization, Geneva, Switzerland, 2003).
12. IEC 61672-1:2002 Electroacoustics – Sound level meters –International Electrotechnical Commission,
13. IEC 61260:1995 Electroacoustics - Octave-band and fractional-octave-band filters International Electrotechnical Commission, CENELEC, Brussels 1995
14. S.J Yang and A.J. Ellison, *Machinery noise measurement* (Clarendon press, Oxford, 1985)
15. Bruel & Kjaer, Technical Documentation Noise Source Location Software Type 7681 (Bruel & Kjaer, 1994)
16. Nikola Holeček, Brane Širok, Marko Hočevar, Rudolf Podgornik and Rok Grudnik, *Reducing the noise emitted from domestic clothes-drying machine*, Noise Control Eng. J., 54(3), 137–145, (2006).
17. Holeček, N., Eberlinc, M., Hočevar, M., Širok, B. (2009): *Experimental investigation of noise generated by multifunctional refrigerating units.* V: The 38th International Congress and Exposition on Noise Control Engineering, Ottawa, Canada, 23.–26. August. Proceedings of inter-noise 2009: innovations in practical noise control. Washington: Institute of Noise Control Engineering.

MERENJE INTENZITETA ZVUKA KAO DIJAGNOSTIČKI ALAT ZA REDUKCIJU BUKE KUĆNIH FRIŽIDERA

Pored vrlo složenih funkcionalnih karakteristika i niske potrošnje energije, tihi rad je bitna komponenta modernog integralnog frižidera. To je zbog toga što je emisija buke izuzetno važan faktor za kupca pri izboru novog aparata. Proizvođači tihih frižidera obično reklamiraju njihovu nisku emisiju buke koristeći pojmove "super tihi rad". Ovaj rad predstavlja postupak za postizanje željenog nivoa buke za jedan takav aparat. Sva merenja izvedena su u laboratoriju za akustiku, u uslovima slobodnog zvučnog polja. Nivo buke u različitim fazama razvoja je izražen ukupnom zvučnom snagom, frekvencijskim spektrom i pomoću vizualiziranih slika zvučnog intenziteta. Na osnovu tih eksperimentalnih rezultata određeni su kritični izvori buke, što je omogućilo neophodne izmene konstrukcije frižidera za smanjenje ukupnog nivoa zvučne snage od 57 dB(A) na prihvatljivih 44,5 dB (A).

Ključne reči: *izvor buke, merenje zvučnog intenziteta, lokacija izvora buke, smanjenje buke, aparati za domaćintstvo*