Acoustical Characterization and Optimization of a Kitchen Hood

Alexandre Santos Cunha Medeiros Pacheco
alexandrecunhapacheco@gmail.com

Instituto Superior Técnico, Lisboa, Portugal

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Abstract

This work is focused on the acoustical characterization and optimization of a kitchen range hood, specifically of the sirocco fan contained within. First of all, it was done a careful assessment of the noise sources of the range hood by analysis of the noise spectra of its different components; it was concluded that the spectra of the housed fan is mainly broadband with just a significant tonal component at twice line frequency due to motor vibrations. For the improvement of the sound emissions, the effects of some noise control methods were tested; in particular, the results show that noise reductions on the housed fan between 4-7 dB or 3-4 dB(A) are achieved by only modifying the volute geometry. Finally, the impeller aerodynamic noise was numerically predicted by using hybrid computational acoustics techniques, where the computational fluid dynamics (CFD) and the Farassats 1A formulation were sequentially used. Through CFD, the aerodynamic loadings on the impeller in the presence of the original and redesigned volutes were obtained; it was verified that the sound spectra radiated by the impeller alone was nearly coincident for both cases, leading to the conclusion that the actual noise reduction seen by the experiments is due to improvements of the flow conditions near the volute inner walls, rather than the blades; it was also observed that the impeller aerodynamic noise is mainly originated by dipolar sources (loading noise), being the thickness noise (monopolar sources) negligible.

Keywords: Noise reduction, Centrifugal fan, Sirocco, Spectral analysis, volute, Numerical simulation

1. Introduction

The work reported here is inserted in a joint work of research, development and optimization of a kitchen range hood, within a project in collaboration with a specific company of the sector.

In order to modern industries make their products comply with noise regulations and succeed in market competition, the use of noise reduction methods during the design process or the acoustical improvement of existing products is a subject of major interest. Hence, the study here presented is focused on the acoustical characterization and optimization of the range hood and, more specifically, of the contained centrifugal responsible for the air exhaustion. The main objectives are: first, a careful assessment of the noise sources of the different parts that make the range hood; secondly, the mitigation of the acoustic emissions through the implementation of existing noise control methods; and finally the development of a numerical code to predict the impeller aerodynamic noise.

Centrifugal fans are widely used in various fields of engineering; they can be found in numerous applications such as in ventilation and air-conditioning systems [1, 2, 3] and refrigeration systems [4]. The primary deficiencies of centrifugal fans are the lower efficiency and higher noise levels than those of axial fans [5]. The fan here studied is a sirocco type fan, also known as squirrel-cage fan; this fan type is usually used in applications requiring small size, relatively high flow rate and low cost [6].

Concerning the generation mechanisms on centrifugal fans, the noise could have an aerodynamic or non-aerodynamic origin, such as the vibration-induced noise. Regarding the aerodynamic one, it can be formally divided into two components [7]: the broadband noise and the tonal noise component. The broadband noise is generated by the turbulent flow acting on the solid surfaces of the impeller blades and the fan casing, since the fan operates in a spatially non-uniform flow field. The tonal component is associated with the flow interaction between the rotating impeller blades and the fan volute; a small concentrative region in the vicinity of the volute tongue is the predominant source of tonal noise [5]. However, the spectrum of sirocco fans is mainly broadband with a weak tonal component [8] being the emissions at blade passing frequency (BPF) sometimes undetectable.

The aerodynamic sound field radiated can also be considered to be generated by three source distribu-
tions: monopolar, dipolar and quadrupolar. Among them, the dipolar sources resultant from the interactions between turbulent flow and the solid surfaces are the ones that mainly contribute to the noise generated by fans [9].

With respect to the noise control mechanisms of centrifugal fans, some methods are based on modifications of the volute geometry, either on the volute tongue [7, 6] in order to minimize the pressure fluctuations in this zone or on the volute curvature [10] to improve the flow conditions. Increasing cut-off clearance is also a widely used method to reduce the pressure fluctuations in the volute tongue and therefore the sound radiated at BPF [7, 11]. Other methods are focused on inducing a phase difference on the pressures along the volute tongue or impeller blades: this could be done by inclining the cut-off edge or blades [7, 12] or, in the case of double inlet impellers, by staggering the two rotor halves so that the blades of one rotor half lie half-way between those of the other half [7, 13]. Other researchers also focused their attention on the fan vibration; Eck [14] suggests the introduction of plates made of flexible materials with high internal absorption below the resting surfaces of the fan to attenuate the vibrational originated noise. Note however that the combined effect of two or more noise control methods is in general not equal to the sum of the individual reductions [7].

Nevertheless, the prediction of noise generated by centrifugal fans is complex and, as a consequence, many researchers focus their works on parametric experimental studies of different models [15, 6, 5], requiring to test a large number of physical prototypes in a trial-and-error process; this can be highly time consuming and expensive and therefore, numerical studies on sound radiation can be more advantageous than experimental works.

Kim et al. [16], Heo et al. [4], Khelladi et al. [17] and Tang [18] studied fan noise radiation through a CAA hybrid approach in which the computational domain is split into two different regions: first Unsteady flow fields are obtained by CFD; then, the aeroacoustic sources are extracted; finally, the aerodynamically generated sound is modeled by using the predicted flow field data through acoustic analogy. Among the commonly used analogies is the Ffowcs Williams-Hawkins (FW-H) [19] equation and related formulations, such as the Farassats Formulations 1 and 1A [20], because of their relative low computation cost and robustness [21].

2. Characterization of the range hood

The range hood in study is a specific model of a Teka® kitchen range hood (Fig. 1) and it is composed by a parallelepiped hollow box and two grease filters. The range hood’s main dimensions are as follows: 600 mm width; 413 mm depth; 181 mm height and 120 mm exit pipe’s inner diameter.

![Figure 1: Studied range hood with transparent front wall for inside visualization.](image1)

The fan used to create the air flow is a double inlet centrifugal sirocco fan with 50 impeller blades separated by a central plate (Fig. 2).

![Figure 2: Volute and impeller.](image2)

The impeller is contained within a scroll casing, called volute, that redirects the fluid to the vertical outlet discharge duct (Fig. 2(b)). Figure 2(a) shows a sketch of the volute and the adopted coordinate system. Note that the z-coordinate is zero at the fan’s middle meridian section; the azimuth ψ is measured from the x-axis in the counter-clockwise direction, the volute angle 𝛼_v is the aperture angle of the scroll, d is the cut-off clearance and 𝜃_c is the cut-off clearance angle.
sition (P1) makes the rotor spin at 1300 rpm and the second one (P2) makes it spin at 2000 rpm.

3. Experimental measurements
Several experimental measurements were made to the original range-hood and its components in order to identify the different sources of noise. Thus, several sets of measurements were performed to the motor, motor and casing, unhoused impeller, housed impeller with and without the outlet duct and the whole range hood without the outlet duct.

All the experiments are performed for the two range hood operating positions. The presented measurements are made by positioning the microphone at the fan’s middle section along the negative side of the x-axis. For a matter of convenience, the microphone positions presented on the graphs refer to the distance along the x-axis between the microphone and the most upfront side of the concerning source of noise (Fig. 3).

![Figure 3: Way in which the distance between each component and the microphone was measured for the different sets of measurements.](image)

The environment where all the noise measurements take place is also a subject of major importance. This place must on one hand be insulated as much as possible from exterior sources of noise and on the other hand should also be suitable to conduct experiments in nominally free field conditions. This way, all the experimental measurements have been carried out in the anechoic chamber of the IST’s Aerospace Laboratory. The device used to measure the sound pressure level was the Brüel & Kjær Type 2250 sound meter level, provided by IST.

Care was taken in order to place both the microphone and the component being measured far away from the side walls in order to avoid interferences and assure the free field condition. One also had to make sure that the concerning part being measured was not transmitting significant vibrations to the support structure and therefore spare sponges used to make the wedges of the anechoic chamber were placed halfway between the sponges of wedges of the floor in such a way that a horizontal support for the equipment being measured is created (Fig. 4). The microphone is also placed on sponges lying on the top of the anechoic wedges, which allows an easy transfer to a new microphone position during the measurements.

![Figure 4: Experimental setup of the motor and casing assembled.](image)

Each set of measurements are going to be presented in the frequency domain band, more specifically in 1/3 octave bands. For each case, the overall sound pressure level (OASPL) will also be shown in dB and dB(A) from 20 Hz to 20000 Hz, which is the frequency range for which the human ear can detect harmonic pressure fluctuations. The background noise was measured at the middle of the anechoic chamber at a height that is equal to the one of the range-hood’s part being measured. The overall OASPL of the background noise is typically much lower than the noise measured in each set of experiments, which guarantees the reliability of the measurements.

The measurements were taken at 10, 20, 40 and 80 cm away from each component part. The results at 40 cm for the two motor speeds are shown in Fig. 5.

3.1. Motor
The results of the instrument-measured sound levels of the motor shown in Fig. 5 reveal a major spectral peak in the 100 Hz 1/3 octave band. This peak frequency is unaffected by the motor speed, thus suggesting that its cause is not mechanical.

In order to identify the nature of this peak one needs to explore a bit more the electromagnetic forces on an AC powered motor. When a current-carrying wire is exposed to a magnetic field, it will experience a force \( F_e \), that is proportional to the 2nd power of the current intensity passing through that wire. Since in a normal AC power system, the current varies sinusoidally at a certain angular fre-
Figure 5: Sound spectra measured and OASPL measured for each case 40 cm away from the concerning component at motor speeds P1 (top) and P2 (bottom).

\[ F_e \propto \cos^2 \omega t = \frac{1}{2} + \frac{1}{2} \cos 2\omega t. \]  

(1)

Hence, it is verified that the force is proportional to a constant plus an harmonic part whose frequency is twice the line frequency, which in Portugal is \(2 \times 50 \text{ Hz} = 100 \text{ Hz}\).

3.2. Motor and volute
The most prominent noise source of the assembly 'motor and volute' is as well at 100 Hz due to the electromagnetic vibrations on the AC motor, which are amplified by the propagation of the vibrations from the motor to the volute. It is noted as well a significant increase of the noise radiated for frequencies in the range 500-1600 Hz with respect to the measurements made to the motor.

3.3. Unhoused impeller
Finding a suitable basis to properly support the motor with negligible vibrational originated noise was too challenging and therefore the experiments for the unhoused impeller were conducted with a second person holding the motor (Fig. 6).

The sound pressure level of the unhoused impeller is higher at low frequencies and roughly decreases towards higher frequencies, without any peak in the audible range. The tonal part is so weak for both rotational speeds that is impossible to visualize any peak at the blade passing frequencies \(BPF_{P1} = 1083 \text{ Hz}\) and \(BPF_{P2} = 1667 \text{ Hz}\).
3.4. Housed impeller

As expected, the sound spectra of the housed impeller is mainly broadband with the sound pressure level spread out through a wide range of frequencies; the emissions of tonal sound at BPF are completely masked by the adjacent frequencies as well as what happened for the unhoused impeller case. Great broadband levels at low frequencies can be also seen in the spectra; this phenomena was investigated by Fehse and Neise [10] and they concluded that the broadband components at low frequencies in the range 20-200 Hz are generated by classical flow separation regions located on the blades suction sides and the impeller casing.

In addition, it is noteworthy the appearance of a small peak at the 1250 Hz center band frequency for both P1 and P2. The fact that the peak prevails at the same frequency independently of the rotational speed of the impeller suggests that its cause is not aerodynamic.

Moreland [23] studied the presence of well defined frequency regions in a housed impeller of a FC centrifugal fan (no information is given about the rotor dimensions, the number of impeller blades and the blade geometry). He proposed that the explanation to this phenomena could be related to an acoustical resonance of the housing, it is also advocated that the first resonance frequency can be theoretically found assuming that the housing behaves as a Helmholtz resonator under the condition that the wavelength associated is greater than any of the housing dimensions. The Helmholtz resonance \( f_H \) for a double inlet blower is calculated as follows:

\[
    f_H = \frac{c}{2\pi} \left[ \frac{L_i S_o + 2L_o S_i}{L_i L_o V_{ol}} \right]^{1/2},
\]

where \( L_i \) is the effective inlet length, \( L_o \) is the effective outlet length, \( S_i \) is the inlet area, \( S_o \) is the outlet area and \( V_{ol} \) is the volume of the housing.

Applying Eq. 2 to the fan in study one determines that the Helmholtz resonance frequency is \( f_0 = 1144 \) Hz. This frequency is located in the frequency band in which the peak appears suggesting that its origin could actually be an acoustical resonance. Further, the associated wavelength \( \lambda = c/f_0 = 0.3 \) m is greater than all the casing dimensions which is in agreement with the initial hypothesis. Nevertheless, in order to prove that the appearance of the peak at 1250 Hz is indeed due to an acoustical resonance of the housing, one should acoustically excite the volute with a loudspeaker and check at which frequencies it resonates.

Finally, it is noted a clear reduction of the OASPL in dB in comparison with the unhoused impeller case. This could be related to noise reflections, scattering effects or even to local pressure cancellations of opposite sources between the impeller blades and volute walls. In addition, the impeller has greater rotational speeds when it is unhoused, which could also contribute to an increase of the noise radiated.

3.5. Housed impeller with the outlet duct

The attachment of the duct leads to an increase of the sound radiated, specially for frequencies in the range 80-800 Hz, which could be associated to turbulent flow passing through the duct walls or vibrations due to a poor guidance of the flow.

It can be observed a small peak on the noise spectra at the 1250 Hz centre frequency as well as for the case of the housed impeller without the discharge duct. However, calculating the first resonance frequency through Eq. 2 does not lead to any conclusion since the associated wavelength is smaller than the duct length and so the structure can not be considered a Helmholtz resonator.

It is also observed that the attachment of the duct introduces noticeable changes on the noise spectrum with special emphasis for the appearance of two peaks, one at 200 Hz and the other one at 400 Hz. The fact that these peaks are independent of the motor speed indicates that their origin is not aerodynamic. Therefore, their appearance could be related to acoustics resonances inside the duct; however, to accurately analyse the effect of these one should use a narrower band width.

Another effect that can be noted is that after the introduction of the duct it can be verified a reduction of the noise emitted for frequencies less or equal than 80 Hz, which could be related to the end-reflection losses that are, indeed, more pronounced for lower frequencies [24].

3.6. Whole range hood without outlet duct

Comparing the current case to the one without the exterior box it can be seen an increase of the OASPL for both motor speeds. Two important peaks can be spotted on the spectra: a dominant one again at 100 Hz and another one at 200 Hz. Above this last frequency the SPL roughly decreases towards higher frequencies for both motor positions. Below 200 Hz it can be also seen the appearance of a small peak at the 31.5 Hz for P2, which is also present for the housed impeller case; the fact that it only appears at one of the motor operating positions suggests that it has an aerodynamic cause.

4. Tested control methods

4.1. Isolation of vibrations

The attention is going to be focused on insulating the vibrations from the different range hood’s parts. The first study will be performed on the propagation of the motor vibrations, specially in an attempt to reduce the peak at twice line frequency. In order to do so, anti-vibrational plates made of flexible materials with considerable internal absorption
were put in between the motor and the casing (Fig. 7). Three materials were chosen for this purpose: rubber (thickness = 2 mm), sponge (thickness = 2 mm) and styrofoam (thickness = 9 mm).

(a) Rubber  
(b) Sponge  
(c) Styrofoam  

Figure 7: Placement of the chosen anti-vibrational plates on the motor.

The noise measurements were made for the housed impeller at both motor speeds. The obtained sound spectra and OASPL did not suffer any significant change and therefore no improvement on the noise radiated was achieved. Since one failed to reduce the propagation of the vibrations at 100 Hz at the origin it was tested the effect of insulating vibrations from the volute to the exterior box by placing rubber plates at the mountings (Fig. 8).

Nevertheless, this also did not have any relevant impact at the noise spectra at, with even a slight increase of the OASPL around 1 dB for P1 and P2.

(a) Rubber plates.  
(b) Sponge plates.  
(c) Styrofoam plates.

Figure 8: Placement of the rubber plates on the casing mountings with the exterior box.

4.2. Staggering blades

It is here tested the effect of staggering the blades so that the blades from one rotor half lay halfway from the blades of the other half (Fig. 9).

As expected, the spectra shape of the housed fan with the original and staggered blades is similar, since staggering the blades is known to only have an effect at the BPF tones, which are not even visible in the case in study. However, the experiments to the housed fan with the blades staggered showed a slight decrease of the noise emissions over a wide range of frequencies, in comparison with the original impeller design. This drop was more obvious for frequencies below 2000 Hz, with A-weighted OASPL reductions up to 2 dB(A) for the measured distances.

4.3. Changing volute geometry

Finally, one proceeded to modification of the volute geometry in order to improve the flow conditions in general and therefore increase the overall efficiency and reduce the noise generated. Through CFD analysis, that will be explained with more detail in section 5, it is evident that the total pressure near the volute tongue is substantially high 10(a), since the flow runs to the side wall away from the volute tongue. This leads to the formation of a leakage flow region in the vicinity of the volute tongue and therefore to the generation of acoustic noise. Hence, the volute tongue geometry will be object of modifications in an attempt to reduce the high pressure fluctuations at this zone.

(a) Original volute.  
(b) Redesigned volute.

Figure 10: Total pressure on middle meridian section of the volute, at motor speed P2.

Likewise, it is also recognized an increased total pressure along the side walls of the volute and outlet duct. Thus, the volute radius of curvature will be increased so that a better guidance of the flow is accomplished. On a study performed by Kitadume et al. [2] it is proposed a concept of casing design in such a way that the mean velocity in the cross section is the same at any azimuth angle at the design condition of the fan. The expression proposed to
determine the curvature radius of the volute \((R_v)\) is:

\[
R_v = (R_2 + d) \exp (\psi \tan \alpha_v),
\]

(3)

where the variables presented can be visualised in Fig. 2(a); also note that \(\theta_c \leq \psi \leq 360\).

For the redesigned volute it is going to be used a volute angle \(\alpha_v = 5\), since this angle was found to be the optimum one for a 40 bladed sirocco fan with similar characteristics as the one being here studied [25]. The cut-off clearance was not changed.

![Figure 11: Sketch of the new casing design](image)

The volute tongue redesign was based on trial and error. After several CFD simulations performed in order to avoid flow separation at this zone it was possible to obtain the final volute tongue geometry. A sketch of the new volute geometry is shown in Fig. 11.

Comparing the total pressure distribution for the case of the original volute (Fig. 10(a)) and redesigned one (Fig.10(b)) it is observed a significant reduction on the total pressure along the side walls of the volute and outlet duct, as a result of a better guidance of the flow. Further, the volute tongue redesign made possible to substantially decrease the pressure fluctuations near this zone and consequently downstream near the duct inner wall due to a more uniform flow passing though the tongue.

New acoustic experiments were carried out for the redesigned volute in the anechoic chamber, as previously described in section 3. A reduction on the OASPL around 4-7 dB or 3-4 dB(a) was obtained comparing to the original the housed impeller without the outlet duct 12; this reduction is more noticeable for frequencies less or equal than 200 Hz.

In addition, these changes on the volute geometry also led to an increase in the overall efficiency of the blower around 42% [22]. Hence, it seems quite reasonable to assume that both modifications on the volute tongue and volute spiral geometries had a positive effect on the efficiency and acoustic emissions. Still, testing both changes separately could give more information about how each parameter is affected by each modification.

5. Computational Acoustics

This section is dedicated to the computational prediction of the aerodynamic noise generation of the rotating impeller. For that purpose one choose a hybrid CFD/CAA method. In this method, the computational domain is split into different parts: first a dedicated CFD tool in which the aerodynamic forces on the fan blades over the circumferential direction are calculated and secondly, the acoustical sources obtained by CFD are provided to the acoustic solver which calculates the acoustical propagation.

The numerical simulations have been carried out with the commercial program FloEFD [26], which is integrated in SolidWorks® Flow Simulation. This solver solves the governing equations with a discrete numerical technique based on the finite volume (FV) method and uses a \(k-\varepsilon\) turbulence model. It was used a local sliding rotating mesh with 4.2M cells (cell centred) on the rotative domain region.

For the acoustical solver, a software called FAST® (Farassat 1A Acoustic Solver Tool) developed by Ana Vieira [27] was used. As the name itself indicates, this acoustic tool is based in the Formulation 1A of Farassat [20]; it was developed in C language and is capable of predicting two important types of noise: the loading and the thickness noise.

Given that the fan in study rotates at velocities well below subsonic, the acoustic radiation efficiency of the thickness is low [8] and is not important to the noise prediction of these type of fan; thus, the radial component of the velocity is going to be neglected. In addition, since SolidWorks® cannot give the distribution of the forces on the blades it is assumed that the pressure distribution is constant in the spanwise direction and equal to the mean force applied divided by the span.

It was verified that minimum time step for which FAST® could run is \(\Delta t = 0.00025\) and therefore, this value is going to be used on the presented simulations; the number of iterations chosen was 10000 to assure the convergence of the results; finally, the number of spanwise and airfoil points used in the simulations is respectively 40 and 23.

First, the aerodynamic noise radiated by the blades for the housed impeller case is computed. For that purpose, the average aerodynamic forces
Figure 12: Sound spectra of the housed impeller with the original and redesigned volutes with and without the outlet duct, 40 cm away from the volute.

Figure 13: Comparison between the computed aerodynamic noise spectra and the experimental measured noise for the housed impeller, 40 cm away from the volute at motor position P1.

along the azimuthal direction of a given rotating blade were obtained through CFD simulations with a time step of 0.0002 s for the housed impeller (with the original and redesigned volutes).

The sound spectra computed by FAST® (Fig.13) for the impeller aerodynamic noise with the two different volute designs is nearly coincident. This fact leads to conclude that the actual noise reduction verified by the experiments is due to the improvement of the flow conditions on the volute inner walls, rather than on the blades that actually did not suffer any geometrical modification.

Furthermore, FAST® outputs show that the loading noise is clearly dominant compared to the thickness noise at all frequencies; the thickness noise OASPL is as low as -45 dB for the case presented, which is in agreement with the assumptions previously made.

The computed outputs are, as expected, significantly different from the measured spectra of the housed impeller. This is due to the fact that the numerical model only takes into account the aerodynamic noise generated by the impeller and the noise constituents of a housed fan include both the aerodynamic noise generated on all the solid surfaces (which includes both the impeller blades and the volute inner walls) and the vibroacoustic components. In addition, Farassat’s 1A formulation is based on free-space Green’s function and so the presence of the volute is not considered as well as phenomena as reflections, diffractions and scattering.

6. Conclusions
The first objective of the present work was to do an acoustical characterization of the range hood. For that, several experimental measurements were conducted to different parts of the range hood.

It is concluded that the noise spectra of the unhoused impeller is higher at low frequencies and then roughly decreases towards higher frequencies; the spectra of the housed impeller with and without the outlet duct as well as the spectra of the complete range hood assembled is mainly broadband, being the tonal component at BPFs completely masked by the adjacent frequencies.

In all the cases, except for the unhoused impeller one, there are the appearance of noise peaks at well-defined frequencies that are independent of the im-
peller rotational speed. The most prominent one is at twice line frequency (100 Hz) and is related to vibrations originated by electromagnetic forces in the motor that then propagate and get significantly amplified when the motor is assembled with the remaining range hood’s components.

Then, several noise control techniques were studied. Three control methods were applied to the current blower: staggering the blades so that the blades from one side lay halfway the blades from the opposite side; isolating vibration with the use of damping materials between mountings and finally changing the volute geometry.

The experiments to the housed fan with the blades staggered showed a slight decrease of the noise emissions over a wide range of frequencies, in comparison with the original impeller design; this drop was more obvious for frequencies below 2000 Hz, except at twice line frequency (100 Hz), with OASPL reductions up to 2 dB(A). The introduction of different flexible materials with high internal absorption in between the motor-volute and volute-box mountings did not have any significant effect neither at the noise spectra shape at audible range frequencies nor at the overall sound radiated. Changing the volute geometry by increasing its curvature radius and redesigning its tongue proved to be an effective noise control method, with OASPL reductions of the housed impeller around 3 dB(A) and 3-4 dB(A), respectively for the lowest and highest impeller rotational speeds.

The developed numerical analysis was made using hybrid computational aeroacoustic techniques in which the computational fluid dynamics and an acoustic analogy based on the Formulation 1A of Farassat were sequentially used. It was computed the aerodynamic impeller noise for two cases: the housed impeller with the original and redesigned volutes. The computed noise spectra for both cases is nearly the same, leading to the conclusion that the actual noise reduction verified by the experiments for the redesigned volute is due to improvements of the flow conditions on the volute inner walls. In addition, the computed contribution of the dipolar sources to the overall noise radiated is large compared to the monopolar ones, which is in agreement with the assumptions made.

For future work it would be interesting to explore more about the existing possibilities to reduce the sound radiation at twice line frequency, since it is the one that contributes the most to the OASPL measured in dB; for instance, correctly understand if the physical phenomena behind the vibration at this frequency is an internal fault in the motor electrical system. If it is, consider the hypothesis of changing the motor model and, if not, conduct a detailed study in order redesign the motor supports with the volute in such a way that the motor vibrations do not suffer such amplification.

Finally, some modifications can be carried out in the acoustic tool to take into account the noise generated by the flow passing through the volute inner walls and simulate scattering effects, which could be done by the use of a Boundary Element Method (BEM).

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