Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-46088

EXPERIMENTAL CHARACTERIZATION OF THE INFLUENCE OF AUXILIARY DEVICES ON THE NOISE GENERATED BY INDUSTRIAL CENTRIFUGAL FAN AND CORRELATION TO THE GEOMETRICAL AND FLUID DYNAMIC PARAMETERS

M. Cadorin, M. Pinelli
Dipartimento di Ingegneria (ENDIF)
Università di Ferrara
Ferrara, Italy

E. Podeschi, F. Pompoli
MechLav
Università di Ferrara
Ferrara, Italy

A. Zanardi
MZ Aspiratori S.p.A.
Budrio (BO), Italy

ABSTRACT

In recent years, the aerodynamic noise generated by centrifugal fans is receiving increasing attention because of strict environmental noise level restrictions and customer demands. The noise generated by fans is due to aerodynamic sources and to other several sources, such as, for instance, by the fan drive, by bearings and gearing, and, when present, by the inverter. Additional noise sources can be also due to structural resonance effects induced by periodic forces associated with the blade passing frequency or vortex shedding. Usually, these additional noise sources are dominated by aerodynamic noise generated by the fan, in particular when the intake and outlet of the fan are free. On the other side, if fan intake and outlet are ducted, the additional sources can relevantly contribute to overall sound generation.

In this paper, an experimental characterization of the noise generated by industrial centrifugal fans when both inlet and outlet are ducted is presented. To do this, an experimental facility has been design and set up, and the sound power measured by means of the procedures outlined in the ISO 3746 international standard. A number of different type of centrifugal fan (straight-, forward- and backward blade) in different working conditions were tested, resulting in 133 different runs. These amount of data were then processed and a general formula for fan noise estimation obtained as a function of the geometrical and fluid dynamic parameters is derived. Moreover, specific coefficients with respect to blade geometry for the determination of the A-weighted frequency spectrum are presented. Finally, auxiliary devices or other features, such as inverter, thickness of the casing, acoustic insulation, electric motor shaft, are analyzed and some general rules to estimate their influence on sound power level quantified.

NOMENCLATURE

BPF	blade passing frequency
BC	backward-curved blade
C	constant
D	diameter
FC	forward-curved blade
H	head
η	efficiency
K	correction factor
L	sound level
n	rotational speed
p	pressure
Q	volume flow rate
RS	radial-straight blade
s	measurement surface area
z	number

Subscripts

0	reference value
1	correction for background noise
2	correction for reflected or absorbed sound
A	weighted A
blade	referred to fan blade
s	specific
W	power level
p	pressure level

INTRODUCTION

Fans can be classified into three distinctive types [1,2]: axial-flow fans, radial-flow fans and mixed-flow fans.

Among them, radial-flow fans are characterized by flow through the rotor, which is radial outward. The principal parts of a centrifugal fan are the impeller (rotor) and the volute (stator). The volute is the stationary element that guides the air before and after the rotor. The impeller is the rotating element that transfers energy to the fluid. The energy transfer is performed by the blades. Blades are generally classified with regards to relationship between the blade outlet angle and the tangential direction. This includes backward-curved (BC), radial/straight (RS) and forward-curved (FC) blades.

In industrial applications, centrifugal fans are employed in a wide range of situations (process industry, manufacturing, etc.) and can be produced in various sizes, ranging from few-kW to multi-MW. As concerns centrifugal fan noise, a large number of investigations have been focused on fan design optimization and sound reduction control. Neise [3,4] presents a large review of studies regarding noise in centrifugal fans based on measurement techniques. Marsh [5] describes new proposals for measuring the sound-power levels of industrial equipment in the oil and gas industry. More recently, numerical and experimental studies are focused on the determination of noise sources and on their numerical modeling. Velarde-Suárez et al. [6,7] presented experimental studies about the aerodynamic tonal noise sources in industrial centrifugal fan with backward- and forward curved blades.

On the other side, less studies exists on the determination of correlation to estimate sound power levels of centrifugal fans. This kind of correlations are proposed by different authors (see for instance [2, 8, 9, 10]) and are based on the organization of experimental data, either deducted from literature or obtained by means of ad hoc test campaign. Moreover, this correlation are usually referred to established test conditions, i.e. they are referred to the noise generated when either the fan inlet or the fan outlet (or both of them) are open.

In many industrial applications, however, fully ducted fans can be found in practice. In these applications, the centrifugal fan (impeller and volute) is inside the industrial facility while the inlet and the outlet are outside. Hence, the noise sourcing from the fan is emitted into the ambient through the casing and not directly from the inlet and/or the outlet. Moreover, in ducted configuration, other several sources, such as the fan drive, bearings and gearing, inverter, etc, can relevantly contribute to overall sound generation.

In this paper, an experimental characterization of the noise generated by industrial centrifugal fans when both inlet and outlet are ducted is presented. To do this, a dedicated experimental facility has been design and set up at the Research and Development center of MZ Aspiratori (Budrio). Acoustic measurements were executed into a wide industrial warehouse with a volume of 8640 m^3.

Test environments suitable to the usual international standard for fan testing (ISO 13347) [11,12] are specified in detail in ISO 3744. Unfortunately, the warehouse doesn't satisfy the environment correction factor value indicated by ISO 3744 [13]. Therefore, the sound power is measured by means of the procedures outlined in the ISO 3746 [14] international standard.

A number of different type of industrial centrifugal fan (straight-, forward- and backward blade) in different working conditions were tested, resulting in 133 different runs. These amount of data were then processed and general formula for fan noise estimation obtained as a function of the geometrical and fluid dynamic parameters are derived. Moreover, specific coefficients with respect to blade geometry for the determination of the A-weighted frequency spectrum are presented. Finally, auxiliary devices or other features, such as inverter, thickness of the casing, acoustic insulation, electric motor shaft, are analyzed and some general rules to account for their influence on sound power level is quantified.

EXPERIMENTAL APPARATUS

The acoustic fans characterization was performed into the Research and Development center of MZ Aspiratori (Budrio), where a circuit for sound power and fluid dynamic condition measurement was built. Every test condition was measured in accordance with ISO 3746 standard. Overall, 12 different models of centrifugal fans for 133 operative condition were tested. The tested fans are resembled in Tab. 1. As can be seen, the data include tests with BC, RS and FC impellers, and cover a quite wide range of diameters *D*, (470−800) mm, power *P*, (3−37) kW, rotational speed *n*, (990−2930) rpm, volume flow rate *Q*, (725−43000) m^3/h and head *H*, (282−4000) Pa.

Table 1 – Centrifugal fans tested

Model	Blade Type	Blade N°	Impeller Diameter	Nominal Power	Rotation speed	Volume flow rate	H
			[mm]	[kW]	[r.p.m.]	[m³/s]	[Pa]
RL 900	BC	16	870	37	990 - 1475	15000 - 43000	640 - 2730
RL 800	BC	16	770	18.5	990 - 1475	9800 - 30500	282 - 2257
RM 800	BC	8	800	30	1240 - 1770	10700 - 23500	945 - 3062
GR 800	BC	12	800	30	1240 - 1770	5500 - 23000	280 - 2700
RL 500	BC	12	500	18.5	1465 - 2930	5000 - 18400	286 - 3131
RM 500	BC	6	500	15	1465 - 2930	2620 - 10500	395 - 3620
GR 500	BC	10	500	11	1934 - 2930	2150 - 6940	728 - 3346
VM 500	BC	12	500	7.5	1930 - 2925	1150 - 4290	580 - 3730
ZB 500	FC	9	500	7.5	1930 - 2925	1000 - 3890	680 - 4000
VP 500	FC	10	500	3	2880	725 - 1090	3160 - 3900
ZC 500R	RS	6	470	15	1934 - 2930	2520 - 7250	430 - 2879
ZM 500R	RS	6	470	15	1465 - 2930	2400 - 7700	550 - 3132

BC: backward-curved blades, **FC**: forward-curved blades, **RS**: radial/straight blades, **H**: head fan

Test-rig. The test circuit used for measurement is sketched in Fig. 1 and Fig. 2. As can be seen, both inlet and outlet of the fans are ducted with an acoustically insulated pipe. The system exit is located outside the industrial warehouse. In this way, measurement of noise emitted from the exit of outlet duct is avoided.

All fans are inserted in the circuit through flexible joints (without insulation) to avoid vibration transmission between fan and circuit. For a correct positioning among suction and discharge section of fans and duct symmetry axis, small size machines were installed on a height-adjustable reflecting plane, whereas big size machines have adjustable feet. The fan flow rate regulation was carried out through a throttling valve mounted at the fan discharge; the rotational speed was controlled by an inverter that allows motor power frequency control. During each test the following parameters were recorded:

- mass flow rate using a calibrated nozzle in the inlet duct. The nozzle has been designed in accordance with the ISO 5167 International Standard. This kind of device has been chosen since it is fluid dynamically shaped but at the same time it requires less stringent installation requirement. The nozzle has been placed outside the test room to minimize any eventual influence on the fan noise;
- differential average static pressure at suction and discharge sections for the head measurement;
- ambient pressure, temperature and relative humidity;
- rotational speed of the electric motor;
- electric motor power absorption.

Acoustic characterization. The measured sound power includes two contributions: noise emitted by fan casing and electric motor noise, either with or without the use of the inverter.

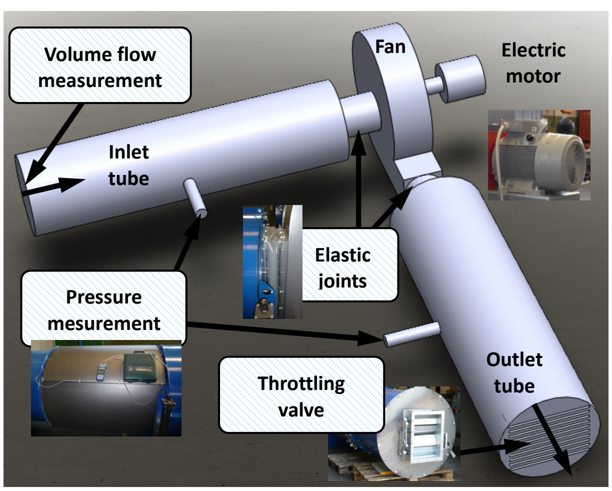


Figure 1 – Test circuit.

Figure 2 – Setup for acoustic measures of fans.

The acoustic power measurement was carried out according to ISO 3746 using 5 microphones with a distance equal to 2 m from the sound source reference box and by adjusting the sound level measured trough the K_{1A} and K_{2A} coefficients. K_{1A} and K_{2A} are expressed in dB(A) and they are used to account for influence respectively of background noise and reverberation. In accordance with the Standard, K_{1A} value is calculated by the difference between energy sum of measured and background sound pressure level, while K_{2A} value is calculated only for 1 kHz octave. Background noise was measured in the 5 standard locations to calculate the coefficient K_{1A}. The results of the measurements showed that the background average sound level can be considered negligible. The environment correction factor (K_{2A}) was evaluated through the reverberation time measurement performed with an impulse technique (exponential sine-sweep).

For each test condition, sound pressure levels in the 5 microphone positions were measured for 60 seconds. The sound power level is calculated by means of the following equation:

$$L_{\mathrm{WA}} = \overline{L_{\mathrm{pfA}}} + 10\log_{10}\frac{s}{s_0} \tag{1}$$

where s is the area of the measurement surface and s_0 is a reference area set equal to 1 m^2 and

$$\overline{L_{\mathrm{pfA}}} = \overline{L'_{\mathrm{pA}}} - K_{1\mathrm{A}} - K_{2\mathrm{A}} \tag{2}$$

In Eq. (2), $\overline{L'_{\mathrm{pA}}}$ is the pressure level averaged over the measurement surface calculated as:

$$\overline{L'_{\mathrm{PA}}} = 10\log_{10}\left[\frac{1}{5}\sum_{i=1}^{5}10^{0.1L'_{\mathrm{pA}i}}\right] \tag{3}$$

where, in turn, $L'_{\mathrm{pA}i}$ is the pressure level measured at the ith microphone position.

For sources which radiates steady broad-band noise, calculations have been made in accordance with the international standard ISO 3746 result in standard deviations of reproducibility of the A-weighted sound power level equal to or less than 3 dB(A) (since K_{2A} < 5 dB(A)).

EXPERIMENTAL RESULTS

The acoustic behavior of each tested fan has been studied by measuring the sound power emitted by different operating conditions. Flow rate was regulated by means of the throttling valve positioned at the exit of the test circuit, while the rotational speed was varied using the inverter.

In Fig. 3, noise measurements related to one of the fan tested (RL 900) at a fixed rotational speed and for different operating point along its characteristic curve are reported. It can be noticed that the sound power emitted by the fans is directly proportional to the air flow rate. This behavior is due to the aerodynamic nature of the acoustic source whose intensity increases with the airflow velocity, since it is related to the presence of turbulence within the stator ducts and in blade channels.

The possibility of varying the rotational speed by means of the inverter allows, not only to test a greater number of operating points, but also to verify the presence of blade passing frequency (BPF) increment effect in the spectrum. This phenomenon can arise for radial fans because of the blades radial sliding by the volute tongue and determines the raising of the sound power level in correspondence of the octave band containing the BPF, defined as:

$$\mathrm{BPF} = \frac{n\,z}{60} \qquad (1)$$

However, in the runs reported in Fig. 4, which is still referred to the sound power emitted by the RL 900 fan but at different rotational speed, the siren effect is not present, and this effect was not detected for most of the fan tested.

In addition to the measurements carried out for acoustic characterization of fans in standard operation conditions, further tests were realized in order to evaluate the influence of the use of the typical devices installed on a fan. In particular, the auxiliary devices taken into considerations were: the inverter, the casing insulation, the casing thickness, the connection coupling insulation, the impeller-shaft seal gasket.

Inverter. The inverter is used to vary fan rotation speed and its employment increase, regardless of the value of the speed to achieve, the noise level at high frequency in octave 4 kHz,

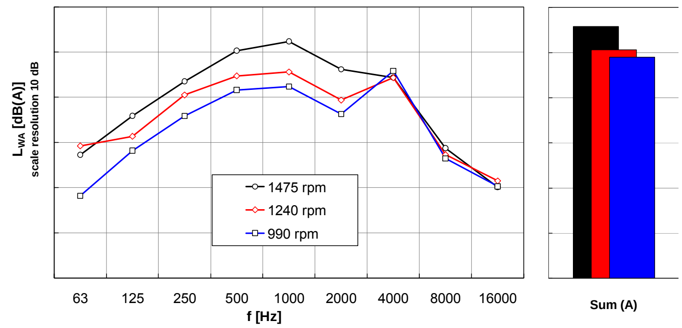


Figure 4 – 1/1 octave band spectrum of noise emission: RL 900 fan at different rotation speed (regulation valve opening: 35 %)

8 kHz and 16 kHz (Fig. 4). In order to consider this effect, the average sound power level of inverter was calculated as the average energy difference between measurements with and without the use of inverter (Fig. 5). So the presence of the inverter can be considered negligible up to 2 kHz, while at 4 kHz, 8 kHz and at 16 kHz the sound power emitted by fan must corrected with inverter effect. In first approximation, it is possible to underline that these values are independent of velocity, frequency and electric power and are the same for all fans model.

Casing insulation. The effect of the insulation of the metal casing (constituted by two metal sheets and one rock wool panel) causes a decrease of sound power level for all 1/1 octave bands (Fig. 6). At frequencies higher than 2000 Hz, in correspondence of which the insulation should have the most significant effect, a very small decrease in the sound power level can be noticed. This behavior highlights the presence of an acoustic source, located outside the metal casing of the fan, that for frequencies above 2000 Hz emits a sound power level comparable to that of the fan itself.

The electric motor is the only acoustic source presents in the test site that has this feature. In general, the tests show that the use of an insulated metal box leads to a decrease in total sound power level that can be estimated in the range (5-6) dB(A).

Casing thickness. The use of a thicker casing (3 mm thickness instead 1.5 mm) showed no significant changes in the sound power level emitted by the fan (Fig. 7).

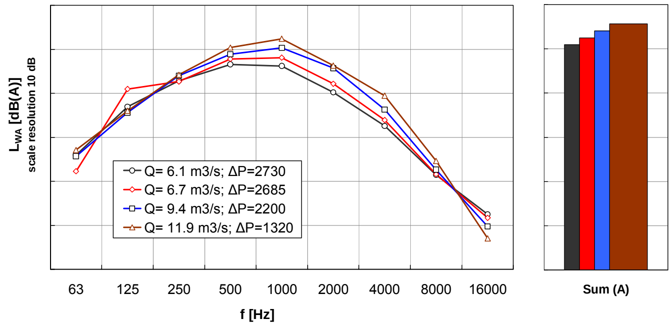


Figure 3 – 1/1 octave band spectrum of noise emission: RL 900 fan in different operating point (n = 1475 rpm)

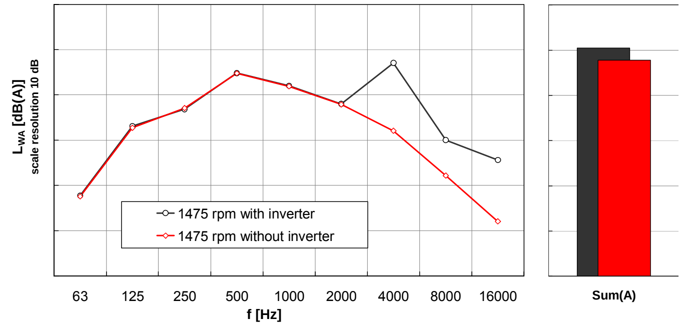


Figure 5 –1/1 octave band spectrum of noise emission: GR 800 fan with and without inverter ($Q = 0.95\ m^3/s$, $\Delta p = 2280$ Pa)

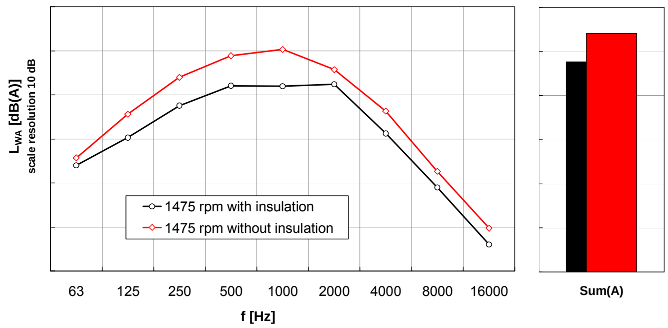


Figure 6 –1/1 octave band spectrum of noise emission: RL 900 fan, with and without insulation (Q = 9.4 m^3/s, Δp = 2200 Pa)

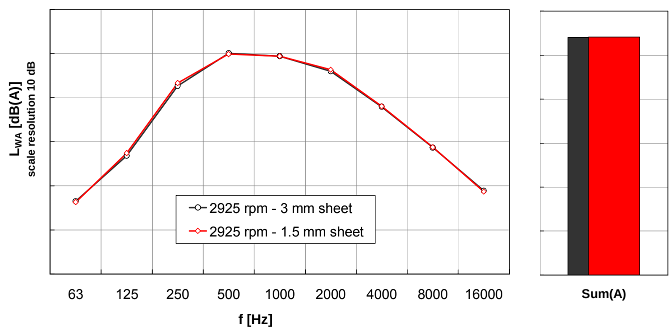


Figure 7 –1/1 octave band spectrum of noise emission: ZB 500 fan with different thickness sheet (Q = 0.95 m^3/s, Δp = 2280 Pa)

Connection coupling insulation. The insulation of flexible connection couplings constituted by a multilayer package consisting of rock wool and metal layers (Fig. 8), causes a decrease in the total sound power level of about 1-3 dB(A), with respect to the configuration characterized by non-insulated flexible couplings. Through a frequency analysis, it can be noticed that the most efficient insulation is obtained for 500, 1000 and 2000 Hz 1/1 octave band, typical frequencies of the aerodynamic noise (Fig. 9).

Impeller-shaft seal gasket. The use of a seal gasket between the shaft and the fan case determines a reduction in sound power level equal to 5 dB(A) with respect to the no seal configuration (Fig. 10). The frequencies most affected by the presence of this device are once again those directly attributable to the aerodynamic phenomena (500, 1000, 2000 Hz). This behavior confirms that, above 2000 Hz, all of the benefits

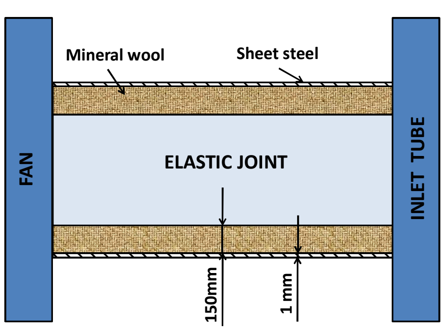


Figure 8 –Sketch of the connection coupling insulation.

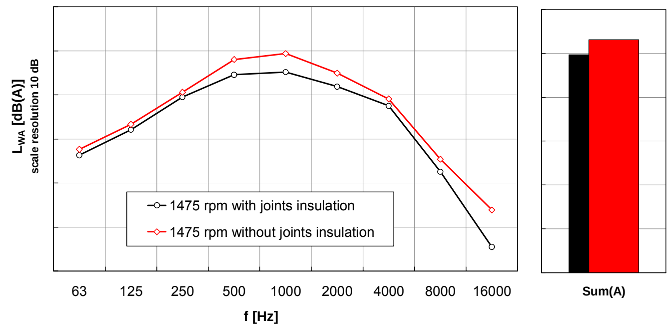


Figure 9 –1/1 octave band spectrum of noise emission: RL 900 fan, with and without joints insulation (Q = 12.2 m^3/s, Δp = 1780 Pa)

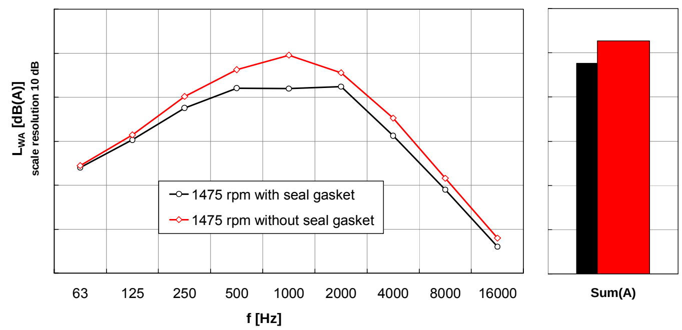


Figure 10 –1/1 octave band spectrum of noise emission: RL 900 fan, with and without drive shaft seal gasket (Q = 9.3 m^3/s, Δp = 2430 Pa)

achievable by the use of the accessories described are penalized by the presence of the electric motor. Hence, a better improvement can be achieved only thought a sound insulation of the motor.

CORRELATION BETWEEN SOUND POWER AND FLUID DYNAMIC PARAMETERS

The acoustic measurements have been exploited in order to study a formulation capable to calculate the sound power emitted not only by the tested fans but also by fans (having scaled size and similar models) operating at different flow conditions. The formulations proposed below present a structure supported by literature with regard to fluid dynamics noise generated within inlet and outlet ducts [8-10,15].

Six coefficient correlation. The first formulation, reported in Eq. (5),

$$L_{\mathrm{WA}} = \frac{C_1}{D_{\mathrm{S}}^{C_2}} + C_3 \log_{10}(Q) + C_4 \log_{10}(\Delta p) + + C_5 \log_{10}\frac{(1-\eta)}{100} + C_6 \tag{5}$$

allows the calculation of the sound power level in dB(A) and it is dependent on the specific diameter (D_S, [-]), the volume flow rate (Q [m^3/s]), the head (Δp [Pa]) and the efficiency (η [%]).

The previous formula shows the presence of six constants, calculated by minimizing the error resulting from the calculation of sound power with respect to the experimental measurements.

The coefficients C_1, C_2, C_3 and C_4 are the usual ones that can be found in other referenced correlation. In the formulation proposed, however, 2 additional terms are introduced:

- C_5 takes into account the dependence from the efficiency of the fan. In this manner, it is possible to overcome the limitation of the other correlation, which instead are referred to the sound generated at fan peak efficiency only. Hence, also the constants C_1, C_2, C_3 and C_4 are different from their values that can be found in [10, 15];
- C_6 is dependent on impeller blade type (BC, RS or FC).

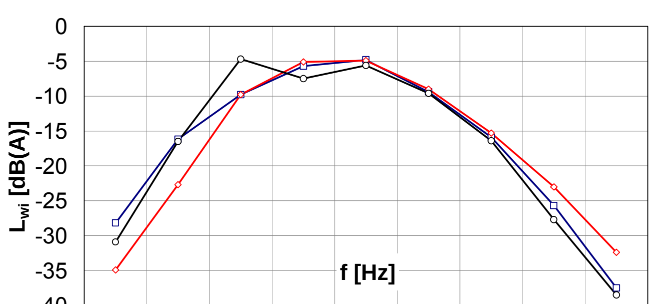
For reasons of confidentiality, the values of the six coefficients may not be published. To assess the sound power spectrum, a frequency characteristic curve has been determined for each type of blade by the calculation of an average value of the measured spectra. In this way, it was possible to determine the values of B_i (Figure 11) which, algebraically added to the total power in dB(A), allow to obtain the sound power emitted for each octave band:

$$L_{W_i} = L_{WA} + B_i \tag{6}$$

The presented correlation allows the assessment of the overall sound power with an average error equal to 1.9 dB(A) with respect to the experimental measurements performed in thus work, while in the 1/1 octave band spectrum the average error is (2-3) dB(A).

Figure 12 shows a comparison among experimental measurements and the sound power level calculated with the proposed formula and with a correlation reported in literature [10]. The literature equation is referred to the sound power emitted through the fan casing considering both intake and discharge fully ducted. The comparison is referred to two representative fans among the tested ones, in two different working conditions. It is possible to notice that the correlation proposed for fans allows to calculate the overall level and the 1/1 octave band spectrum of emitted noise in a more accurate way than the literature correlation.

Moreover, corrections in sound power calculations are proposed when above described accessories are used.



	63	125	250	500	1000	2000	4000	8000	16000
B	-28.2	-16.2	-9.8	-5.7	-4.8	-9.4	-15.9	-25.7	-37.5
F	-34.9	-22.7	-9.8	-5.1	-4.9	-9	-15.3	-23	-32.4
S	-30.9	-16.5	-4.7	-7.5	-5.6	-9.6	-16.4	-27.7	-38.5

Figure 11 – B_i coefficient for frequency spectrum determination.

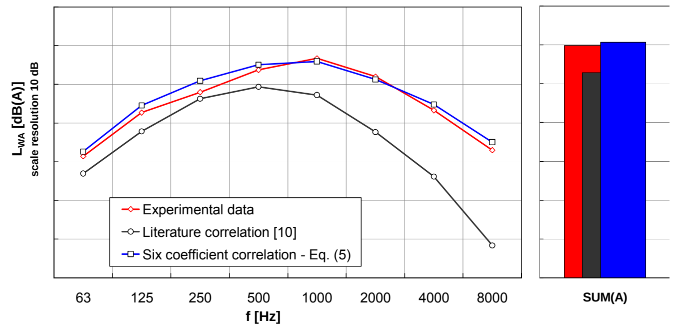


Figure 12 – Experimental results, proposed formulation and literature correlation [10] comparison (RL 500, Q = 3.4 m^3/s, Δp = 1290 Pa)

The insulation contribution and the seal gasket are simply algebraically added to the total value of sound power, while using the inverter, the characteristic values at 4000-8000 and 16000 Hz octave bands are energetically added to values obtained by Equations (5) and (6). From the analysis of the results reported in Figure 13 and 14, it can be noticed that the proposed corrections allow to obtain values of total sound power and frequency trends close to experimental ones.

Four coefficient correlation. Since the fan efficiency is usually not reported in data provided by fans manufacturers, a four coefficient correlation referred to peak efficiency working points - reported in Eq. (7) - is tested.

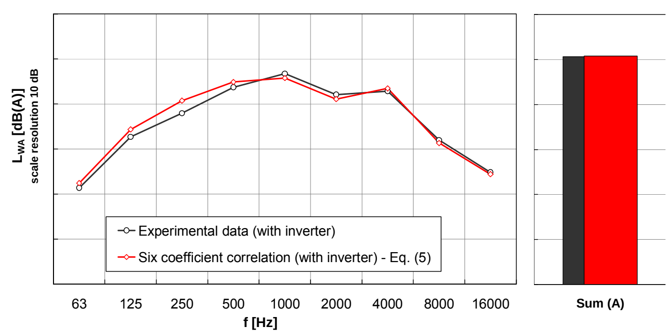


Figure 13 – Comparison between experimental data and six coefficient correlation results with inverter correction (RL 500, Q = 3.4 m^3/s, Δp = 1290 Pa)

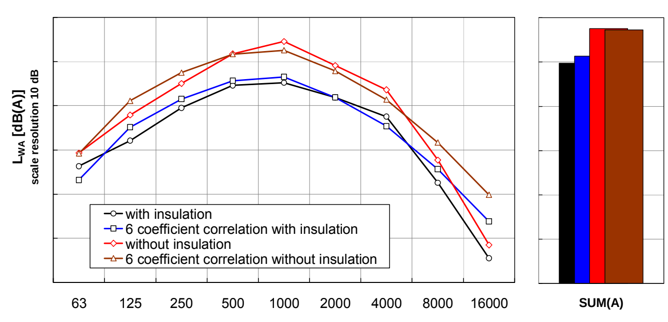


Figure 14 – Comparison between experimental data and six coefficient correlation results with and without insulation correction (RL 900, Q = 11.9 m^3/s, Δp = 1320 Pa)

The form of Eq. (7) can be usually found in literature: it can be noticed that the terms referred to fan performance (and its related constant C_5) and the constant dependent on the particular type of blade (C_6) are not considered.

$$L_{WA} = \frac{C_1}{D_S^{C_2}} + C_3 \log_{10}(Q) + C_4 \log_{10}(\Delta p) \qquad (7)$$

This formulation allows the assessment of the overall sound power with an average error of 2.1 dB(A) in comparison to experimental measurements performed at maximum efficiency working points, while for the 1/1 octaves spectrum the average error is 3-4 dB(A). Figures 15 and 16 show that the four coefficients correlation can accurately calculate the sound power emitted only for fan working points at maximum efficiency, while when the operation point moves away from maximum efficiency condition the error significant increases.

CONCLUSIONS

In the present paper, a procedure for the calculation of noise emitted by fully-ducted centrifugal fans is proposed. The correlations derive from those found in literature referred to the aerodynamic noise produced inside the intake and discharge ducts and was calculated through minimization of the error between obtained results and experimental measurements (obtained on 12 fans models for a total of 133 different operating conditions). The formulation allows the calculation of the octave spectrum sound power as a function of fan type, size, Fluid dynamic and operation parameters, with an average error of 1.9 dB(A) referred to the total sound power. The influence of different configurations, such as fan casing sheet thickness, casing insulation, insulation couplings for pipe fitting and the effect of the presence of different accessories that can be considered as additional noise sources (i.e. the inverter) were also evaluated. The correlations suggested allow the calculation of sound power with a good accuracy.

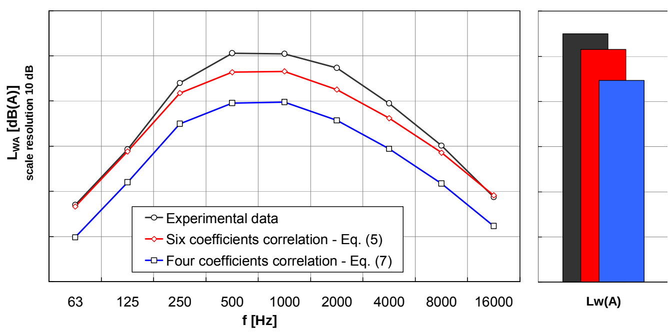


Figure 15 – Comparison between experimental data and the proposed correlation at minimum efficiency operating point (ZB 500, Q = 1.08 m^3/s, Δp = 1510 Pa)

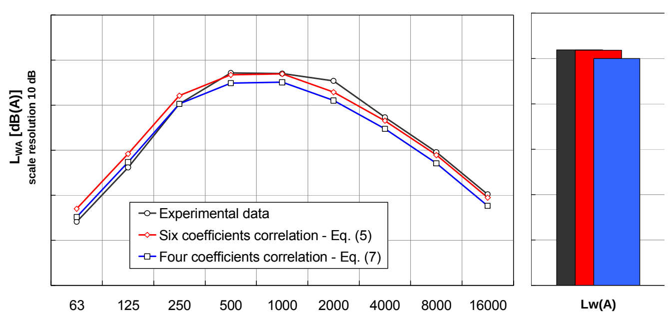


Figure 16 – Comparison between experimental measurement and the 2 proposed correlation, maximum efficiency operating point (ZB 500, Q = 0.77 m^3/s, Δp = 3150 Pa)

ACKNOWLEDGMENTS

This work has been developed within the Laboratory of Advanced Mechanics (MECH-LAV) of Ferrara Technopole, realized through the contribution of Regione Emilia Romagna - Assessorato Attivita' Produttive, Sviluppo Economico, Piano telematico - POR-FESR 2007-2013, Attività I.1.1.

REFERENCES

[1] Daly, B.B., 1960, Woods Practical Guide to Fan Engineering, Tiptrace, Essex, UK.
[2] B. Eck, Fans, Pergamon Press, New York, 1973.
[3] Neise, W., 1976, Noise reduction in centrifugal fans: a literature survey, J. Sound and Vibration, 45, pp. 375–403.
[4] Neise, W., 1982, Review of noise reduction methods for centrifugal fans, J. Eng. Ind., 104, pp. 151–161.
[5] Marsh, K.J., 1981, Determination of sound-power levels for industrial purposes, App. Acoustics, 14, pp. 113-127.
[6] S. Velarde-Suárez, C. Santolaria-Morros, R. Ballesteros-Tajadura, 1999, Experimental study on the aeroacoustic behavior of a forward curved centrifugal fan, ASME J. Fluids Eng., 121, pp. 276–281.
[7] Velarde-Suárez S, Ballesteros-Tajadura R, Hurtado-Cruz JP, Santolaria-Morros C., 2006, Experimental determination of the tonal noise sources in centrifugal fan, J. Sound and Vibration, pp. 781–96.
[8] Wright, T., 1999, Fluid machinery-Performance, analysis and design, CRC Press.
[9] ESDU, 2007, A guide to fan selection and performance, ESDU-79037.
[10] Nuovo Colombo, Manuale dell'ingegnere, Volume 2, 2003, 84a edizione Cyril M. Harris, Handbook of Acoustical Measurements and noise control, 1998
[11] ISO 13347-1:2004, Industrial fans-Determination of fan sound power levels under standardized laboratory conditions-General overview
[12] ISO 13347-3:2004, Industrial fans-Determination of fan sound power levels under standardized laboratory conditions-Enveloping surface method
[13] ISO 3744:1994, Acoustics -- Determination of sound power levels and sound energy levels of noise sources using sound pressure -- Engineering methods for an essentially free field over a reflecting plane
[14] ISO 3746:1995, Acoustics - Determination of sound power levels of noise sources using sound pressure - Survey method using an enveloping measurement surface over a reflecting plane
[15] ASHRAE, 1993, Handbook-Fundamentals9

