PRACTICAL EFFECT OF USING ACOUSTICALLY UN-TREATED TEST DUCTS ON THE NON-MEASUREMENT SIDE, WHEN MEASURING FAN SOUND POWER LEVELS

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SUMMARY

An experimental campaign was intended to validate the sound-power level ratings for a new range of centrifugal fans. These ratings were based on measurements in accordance with the AMCA 300 standard and extrapolations with the method provided in ISO 13348. The generalized sound power level spectra measured on the inlet-side of a single fan, at different speed values and in the same standardized installation type B, were compared, to identify the optimum speed-scaling factor. A satisfactory correlation could be achieved across the sound frequency spectrum, except at the blade-passing frequency (BPF) and some of its harmonics, which are subject to wide variations, attributable to the use of acoustically un-treated duct on the non-measurement side.

INTRODUCTION

In the process to develop the catalogue ratings of a new range of double-inlet backward-curved centrifugal fans, the fan testing laboratory of Regal Beloit Italy S.p.A. performed a series of tests with the intention to validate the chosen calculation methods.

These calculation methods were used to convert a limited number of measurements, of the fan air performance and of its sound power levels, into a complete set of ratings, covering every possible duty point within the useful range of the fan speed and flow coefficient.

The sound power ratings of these fans were based on sound power measurements, carried out on each individual fan size in accordance to the AMCA 300-96 standard, and then calculated using the method provided in the ISO 13348:2007 standard, with some slight modifications for ease of numerical application.

This method entails an experimental determination of the multiplier C of the logarithmic addendum, which provides the increase of the calculated fan sound power levels $L_{Wf}(\varphi,\chi,N)$, as a function of the fan speed N, over the values of the generalized sound power spectrum $L_g(\varphi,\chi)$, itself a function of the flow coefficient φ and of the non-dimensional frequency χ .

TEST SETUP AND MEASUREMENTS

A single fan, size 250 mm, was installed on an AMCA 210-99 Fig.12 test chamber, in a standardized installation type B, i.e. with "open inlets and discharge connected to a duct", using a "short outlet duct" between the fan and the test chamber, which was used to adjust the fan operating point, and to determine it, measuring the fan volume-flow rate and static pressure.

The "short outlet duct" is a piece of straight duct with a section identical to the shape of the fan outlet, a length of 2.5 equivalent diameters, and is used to let the velocity profile and the static pressure settle across the duct, removing the peculiar disturbance which is found immediately downstream of the fan outlet, and reach a velocity profile and pressure values which are representative of the fully-developed flow inside a semi-infinite duct.

In the case of the duct used for these test runs, the internal section was a square 322 x 322 mm, and the length was 0.91 m, terminating with an abrupt transition to the measurement chamber, and without any sound-adsorbing material incorporated in the design.

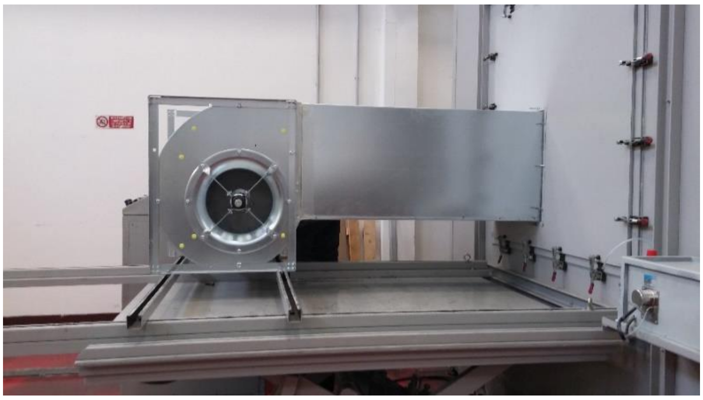

Figure 1: typical centrifugal fan "type B" installation for testing, with short outlet duct

For better acoustical separation, between the fan inlet and outlet sides, the outlet duct used in this test was made with 25 mm panels of high-density fiberboard, instead of the thin steel plate normally used for such ducts.

To avoid any interference with the measurement of the fan noise, from the noise of the motor, this was encapsulated inside an acoustically insulated enclosure. The fan speed was selected using pulley and belt drives, to prevent possible spurious noise from a VSD-driven motor.

The fan, the outlet duct and most of the AMCA test chamber were installed inside a reverberating room, as specified in AMCA 300-96 and ISO 13347-2:2004, and the fan noise was measured inside the reverberating rom, on the suction side, using an oscillating microphone boom, a single 12.7 mm pre-amplified measurement microphone and a third-octave frequency analyzer, with a useful frequency range extending from the 50 Hz third-octave band, up to the 20 kHz third-octave band, thus covering a span of nine full-octave bands.

The fan noise was measured at 4 different speed values: 2000 rpm, 2650 rpm, 3600 rpm and 5150 rpm, with the three lower speed values representing, respectively, 39 %, 51 % and 70 % of the maximum speed.

To improve the repeatability of the results, two test runs were made at each speed, each one with measurements at ten different values of the flow coefficient φ: the first run with values in increasing order, and the second in decreasing order. The average of the two measurements at each flow-coefficient was used for every successive calculation.

COMPUTATION OF THE GENERALIZED SPECTUM

The goal of this computation is to predict the octave-band sound power levels of a fan, at the eight standard center-frequencies, from 63 Hz to 8 kHz, and at any arbitrary speed (=rotation frequency) within the fan useful range.

The equation given in ISO 13348:2007 clause 7.2.3.2, to calculate the fan octave-band sound power level L_{wfc} as a function of the generalized sound power level L_g ,

$$L_{Wfc} = L_g + 10 \cdot (6 + a) \cdot lg(n) + 10 \cdot (8 + 2a + b) \cdot lg(D_r)$$

can be reversed to calculate L_g, as well as a and b, from multiple measurements of the sound power spectra of geometrically similar fans, having nominal diameter D_r [m] and rotation frequency n [Hz].

To speed-up the computation process, so that it can be carried out repeatedly in real time, even with limited computing power, this equation can be simplified as

$$L_{Wfc} = L_g + C \cdot lg(n) + D \cdot lg(D_r)$$

and then again, after defining a "partly-generalized sound power level", L_{gD} , as

$$L_{gD} = L_g + D \cdot lg(D_r)$$

L_{Wfc} becomes

$$L_{Wfc} = L_{gD}(\varphi, \chi, D_r) + C \cdot lg(n)$$

where the non-dimensional frequency χ is the decimal logarithm of the ratio, between the sound frequency f and the fan rotational frequency f_r :

$$\chi = log_{10}(f/f_r)$$

and then, reversing the equation,

$$L_{gD}(\varphi, \chi, D_r) = L_{Wfm} - C \cdot lg(n)$$

where, to better represent the effect of any dominant pure tone in the sound spectrum, the function L_{wfm} , representing the *measured* sound power spectrum, is not a simple linear interpolation of the 9 octave-band sound power levels (including the 16 kHz band), but the linear interpolation of the 27 values of the octave band sound power levels centered on the 27 standard third-octave band center-frequencies, from 50 Hz up to 20 kHz.

This approach implicitly defines a function L_{gD} which is no longer independent from the fan size, but only (in theory) from the fan speed, and which must be calculated separately from the measurement of each fan size. On the other side, if determined by measuring each fan size separately, L_{gD} is no longer affected by the errors arising from any deviation of the fan geometry from perfect similarity.

In principle, the new multiplier C could also be a function of the flow coefficient φ and of the non-dimensional frequency χ, but the test demonstrated that a single value of the multiplier, not depending from φ or χ, can provide a good correlation of the measured spectra for a low-pressure centrifugal fan.

The actual value of C was determined with a numerical approach, minimizing the sum of the deviations squared.

The value calculated for C on this backward-curved fan was so much close to 45 that this rounded-off value was used for all the following calculations.

Once again, to save computing time, instead of one of the different conventional definitions of the flow coefficient, e.g. the definition according to DIN 24163-1:1985,

$$\varphi = \frac{4 \cdot q_v}{\pi^2 \cdot {D_r}^3 \cdot n \cdot z}$$

a simplified definition $\varphi*$ is used:

$$\varphi^* = \frac{q_v}{N \cdot {D_r}^3}$$

with q_v in m³/h, N in rpm and D_r in m.

The 4 diagrams in *Annex A* show the partly-generalized sound spectra L_{gD} at the 4 test speeds, for 4 different values of the quantity φ*:

- 28 (deep stall),
- 58 (best-efficiency condition),
- 73 (1.25 times the best-efficiency volume flow-rate)
- 99 (1.7 times the best-efficiency volume flow-rate).

The most significant values of the non-dimensional frequency χ, used as abscissa in the diagrams, are shown in the following table:

		χ =
Rotation frequency (fr)	f = fr	0
2 x fr	f/fr=2	0.3010
3 x fr	f/fr=3	0.4771
4 x fr	f/fr=4	0.6021
1/3 x BPF	f/fr=3.666	0.5643
1/2 x BPF	f/fr=5.5	0.7404
Blade-passing frequency (BPF)	f/fr=11	1.0414
2 x BPF	f/fr=22	1.3424
3 x BPF	f/fr=33	1.5185
4 x BPF	f/fr=44	1.6435

Table 1: Most-significant values of χ for a fan having 11 impeller blades

In theory, the generalized sound power spectra, measured at different speeds but at the same flow coefficient, should collapse into a single function.

As shown in the diagrams of the measured values, in Annex A, notable deviations can be seen for values of χ close to 1.041 and, to a smaller extent, to 1.518, which, for an impeller having 11 blades, are, respectively, the values corresponding to the BPF fundamental harmonic and to its third-order harmonic. These are the dominant pure-tone contents in the sound power spectrum. These deviations cannot be corrected by adopting a different value of the speed-scaling factor C, in the BPF range.

The same curves do not show large deviations around the values of χ corresponding to the BPF harmonics of order 2 and 4, possibly because the two opposite single-inlet impellers are offset by half a blade step, with the result that the even harmonics of the BPF tend to cancel.

The diagram in figure 2 shows, instead, the height ($\Delta L_{gD\text{-}BPF}$) of the peak in the L_{gD} function, at the value of the non-dimensional frequency χ closest to the value 1.041 (corresponding to the fundamental blade-passing frequency of an 11-bladed impeller) as a function of the ratio between the length L_D of the outlet-side duct (fixed at 0.91 m) and the BPF quarter-wavelength, for 10 different values of the (simplified) flow coefficient φ*.

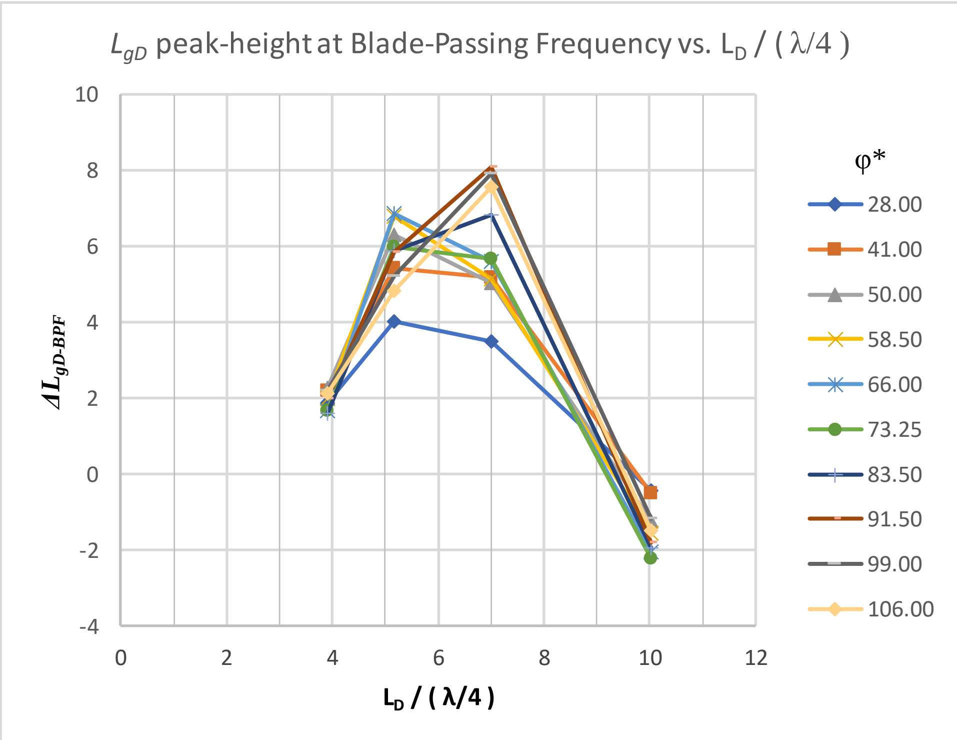

Figure 2: Height of the L_g peak, above the average of the nearby octaves, with different BPF wavelengths

The height of the peak, $\Delta L_{gD\text{-}BPF}$, in the generalized sound-power spectrum, at the blade-passing frequency, is computed as the value of L_{gD} at the measured value of χ which is closest to 1.041, and which includes any contribution from a pure tone at the BPF, less the arithmetic average of L_{gD} at χ values corresponding to sound frequencies one octave band above and one below the BPF, to avoid those "octave band" generalized sound power levels which include a contribution from such a pure tone. It is essentially the difference between the value of L_{gD} at the BPF peak, less the average of L_{gD} at two different χ values, the first being two third-octave steps above and the other two steps below 1.041.

The 4 test speeds (2000, 2650, 3600 and 5150 rpm) generate progressively higher blade-passing frequencies and smaller wavelengths, so that the ratio, between the length of the duct (which is fixed) and a quarter of the BPF wavelength, increases, at the 4 speeds, from 3.9 to 5.16, 7.02 and 10.04.

The calculation of these numbers assumes that the effect on the wavelength of the air speed, along the outlet duct, be negligible. This assumption is acceptable, but not completely correct, because the airspeed in the duct, at the maximum fan speed and maximum flow coefficient, is close to 22 m/s, which is as much 6.5 % of the speed of sound, in air at 20 °C.

The larger peak heights appear when the outlet duct length is close to an odd number of quarter-wavelengths of the blade-passing frequency, while lower peaks are measured when the duct length is close to an even number of quarter-wavelengths. This observation is largely independent from the fan flow coefficient.

CONCLUSIONS

The variation in the amplitude of the BPF peak, shown in figure 2, is clearly not a random effect.

The largest amplitude of the BPF peaks was recorded when the length of the outlet duct was close to an odd number of quarter wave-lengths, and lower peaks were measured when the length of the outlet duct was close to an even number of quarter wave-lengths.

There are too few steps, in the sequence of the measurement speeds and of the ratios, between the duct length and the BPF quarter-wavelength, to provide a really clear demonstration, but the results of these noise measurements on the fan inlet-side are correlated with the tuning of the BPF tone with the outlet-duct length.

A fan may have an inlet and an outlet, regarding the airflow, but, as a sound source, it has effectively two opposite outlets.

The changing acoustic impedance of the duct and chamber, on the discharge or “non-measurement” side of the fan, is altering its emissions, as a noise source, on its opposite side, and, as a consequence, the sound power level measurements being made on the suction side.

At the blade-passing frequency of a fan, generating a dominant pure tone in the fan sound-power spectrum, this effect may produce differences of as much as 10 dB between the lowest and largest peak amplitudes.

An attempt has been made, in the ISO 13347 series of standards, to introduce some degree of standardization of the acoustic properties of the ducts on the non-measurement side, by introducing the “simplified anechoic terminations”, to reduce the measurement uncertainty created by this effect.

This technology has not yet won a large support within the industry, especially among the manufacturers of centrifugal fans, which are most-often tested using rectangular-section test ducts. This because the use of the round-section simplified anechoic terminations not only requires an extensive collection of differently sized terminators, but also the use of a collection of transition pieces and the extension of the overall length of the test ducts.

Despite all the practical obstacles to the adoption of the simplified anechoic terminators, this technology is well worth further investigations, because it has the potential of providing a significant improvement in the reliability of the fan noise ratings, at those critical frequencies which include significant pure tones, like at the blade-passing frequency and its harmonics.

BIBLIOGRAPHY

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[3] ISO 5801:2017 – *Fans — Performance testing using standardized airways*

[4] ANSI/AMCA Standard 300-96, now superseded by ANSI/AMCA 300-14 – *Reverberant Room Method for Sound Testing of Fans*

[5] ANSI/AMCA Standard 210-99, now superseded by ANSI/AMCA 210-16 / ASHRAE 51-16 – *Laboratory Methods of Testing Fans for Certified Aerodynamic Performance Rating*

ANNEX A

The following diagrams are examples of the generalized sound-power spectra for 4 different values of the flow coefficient.

Instead of one of the different conventional definitions of the flow coefficient, a simplified definition is used for the flow coefficient φ*:

$$\varphi^* = \frac{q_v}{n \cdot {D_r}^3}$$

with q_v in m^3/h, N in rpm and D_r in m.

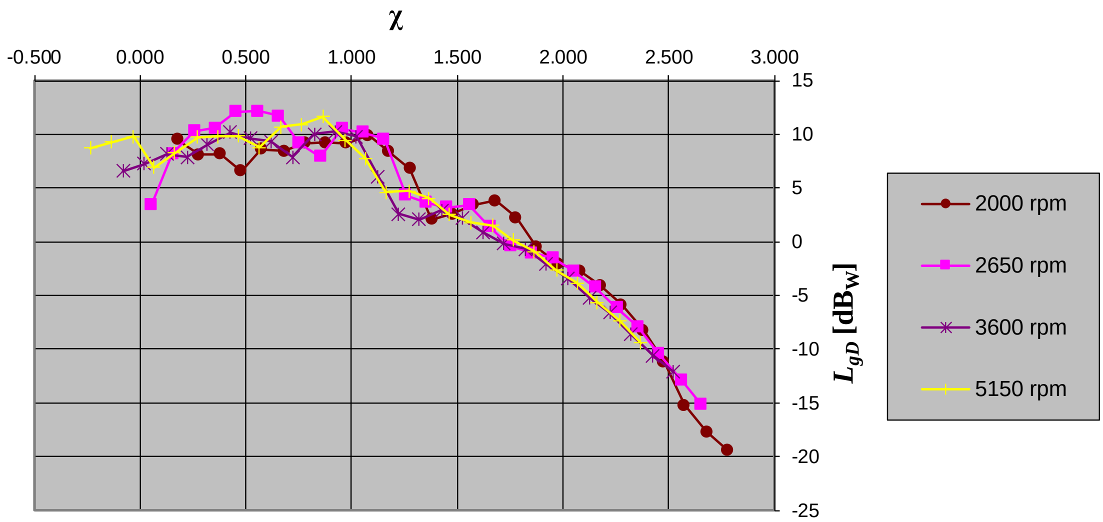

Figure A.1: Partly generalized sound-power levels for φ = 28 (deep stall).*

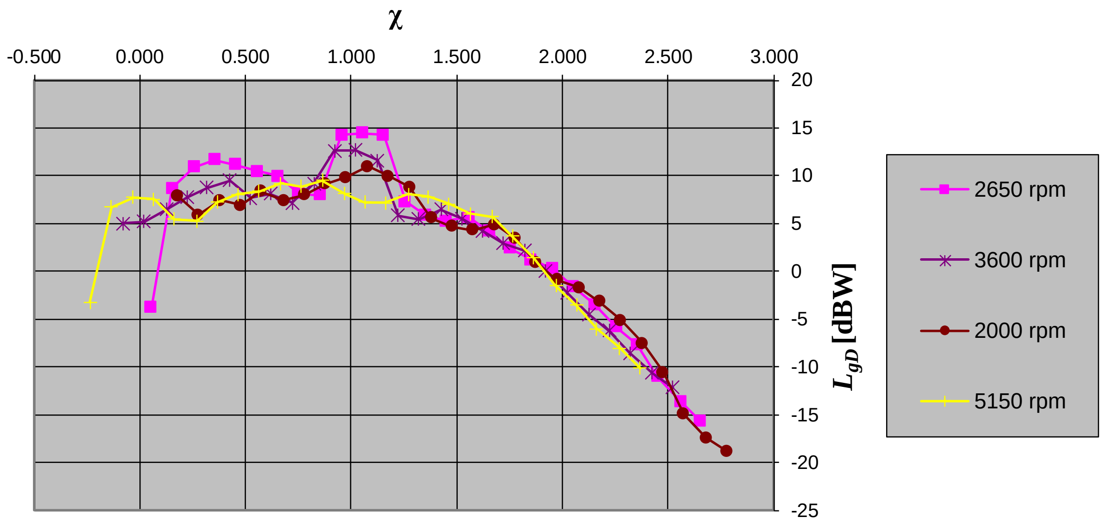

Figure A.2: Partly generalized sound-power levels for φ = 58 (best-efficiency volume f. r.).*

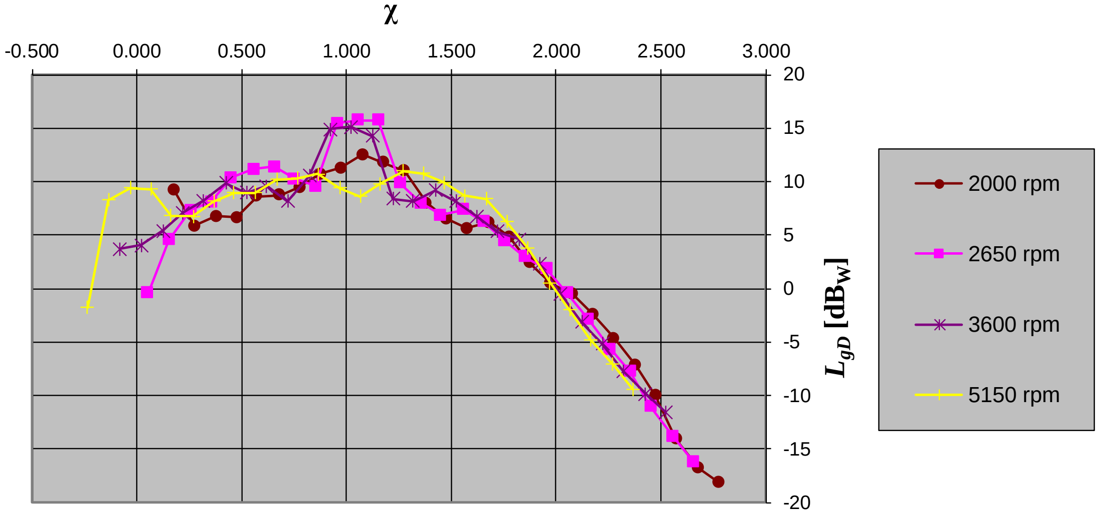

Figure A.3: Partly generalized sound-power levels for φ = 73 (1.25 times the best-efficiency volume f. r.).*

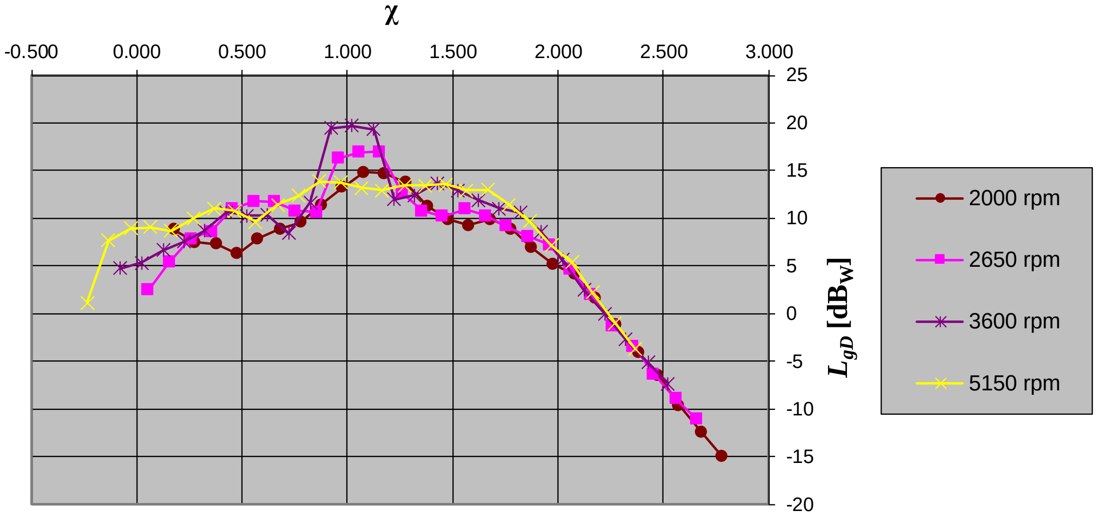

Figure A.4: Partly generalized sound-power levels for φ = 99 (1.7 times the best-efficiency volume f. r.).*