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Characterization of centrifugal fan noise generated in residential HVAC systems using in-duct measurements

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ABSTRACT

The acoustic characterization of a centrifugal fan in a residential HVAC system is carried out through the determination of the total source spectrum of the sound generated by the fan under varying operating conditions. The most commonly used characterization technique, described by ISO 5136, requires an anechoic termination and the possibility to insert a microphone in the duct. To avoid the large dependence on the efficiency of the anechoic termination, an active one-port characterization using flush mounted microphones is adopted in this research. One of the major advantages of this approach is the fact that the characterization is carried out independent of downstream reflections, thus not requiring the construction of an anechoic termination. At first, the passive property of the fan, i.e. the reflection coefficient of the propagating waves at the inlet of the fan, is determined. Subsequently, the downstream reflection component of the test rig is characterized. Finally, both passive models are combined to compute the active source spectrum describing all noise generation and dissipation phenomena of the fan. The acoustic characterization is compared to existing scaling laws, such as described e.g. in VDI 2081.

Keywords: Fan noise, Scaling laws

1. INTRODUCTION

Noise pollution due to ventilation systems is an important concern, as the most frequently reported issues with HVAC systems are linked to excessive noise emissions. The noise nuisance often results in the misuse of ventilation systems, causing poor indoor air quality and a loss in energy efficiency (1). Through an accurate prediction of the acoustic field generated by HVAC systems, noise pollution can be avoided and innovative noise mitigation approaches can be assessed.

To allow the calculation or prediction of the acoustic field generated by residential ventilation systems, knowledge of the total source spectrum of the sound generated by a centrifugal fan is of great importance. Centrifugal fans are commonly used in residential ventilation systems C (only mechanically discharges used indoor air) and systems D (mechanically supplies fresh outdoor air and discharges used indoor air). Using the source spectrum, the sound generating mechanisms of a centrifugal fan can be investigated in further detail and efficient sound attenuation devices can be designed (2).

General guidelines and reference spectra for the sound generated by fans can be found in e.g. VDI 2081 (3). According to these guidelines, the emitted sound power level of a fan is determined from a constant value, the specific sound power level that depends on the assembly type of the fan, and the operational point of the fan, determined by the volumetric flow rate through the fan and the total pressure drop across the fan. The source spectrum of a fan can thus be related to the flow rate and pressure drop across the fan by a scaling law. The octave sound power spectrum of a fan is estimated with a standard octave sound power spectrum, through superimposing the sound power level output

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of a fan with a discrete spectrum whose strength depends on the type of fan. However, the available data on residential centrifugal fans is limited, as most reference spectra and guidelines concern more powerful fans of different assembly types. Current guidelines on the acoustic characterization of fans stated in e.g. ISO 5136 (4) describe characterization techniques for fans with duct diameters between 0,15 and 2 meters. These techniques are difficult to apply on smaller residential centrifugal fans and require the use of an anechoic termination.

Centrifugal fans have at least one inlet and one outlet. In residential ventilation systems C, the inlet is connected to the extraction grilles in the building. Consequently, only the sound generated in and transported through the inlet ducts is of interest, and the acoustic load of the fan is only relevant at the inlet of the fan. If the inlet and the outlet of the fan are acoustically uncoupled from each other, the source can be considered as a one-port source for a plane wave, linear and time-invariant system (2). The acoustic characterization of a centrifugal fan can now be carried out through the determination of the passive property of the fan, i.e. the reflection coefficient of the propagating waves at the inlet of the fan. Subsequently, the downstream reflection component of the test rig is characterized. Both passive models are combined to compute the active source spectrum describing all noise generation and dissipation phenomena of the fan, eliminating the need for an anechoic termination. The proposed active one-port characterization is therefore an improved and better suited characterization technique for residential centrifugal fans.

Through the measurement of the total source spectrum of a fan under varying operating conditions, scaling laws that relate the operating conditions to the sound generated by the fan can be determined. These scaling laws can be used to predict the source spectra of fans for a large range of operating conditions. The experimentally determined scaling laws are compared to already existing scaling laws.

This paper describes two different types of centrifugal fans. Subsequently, the active one-port characterization method is discussed, followed by a description of the measurement set-up used. Thereafter, the determined source spectra of the sound generated by centrifugal fans are discussed. Finally, a scaling law that relates the operational point of the fan to the sound generated by the centrifugal fan is proposed and compared to existing scaling laws.

2. DESCRIPTION OF THE FANS

Two types of centrifugal fans are acoustically characterized, a constant pressure fan and a constant power fan, both depicted in figure 1. When changing the aerodynamic load of the piping system, the constant pressure fan tries to keep the pressure across the fan constant. The pressure can only be adjusted by changing the settings of the fan, adjusting the rotational speed of the fan. The constant power fan aims to keep the active power constant when the aerodynamic load of the system changes. Both fans are aerodynamically characterized through the determined fan curves that relate the pressure increase across the fan to the volumetric flow rate through the fan, shown in figure 1. These curves are determined for all possible settings of the fans. When the flow rate through the fan is known, the operational point of the fan can be determined using the fan curves. The diameter of the ducts connected to the centrifugal fans is 0.125 meter. Figure 1 shows that all inlet ducts are connected to a manifold, from which the air enters into the centrifugal fan.

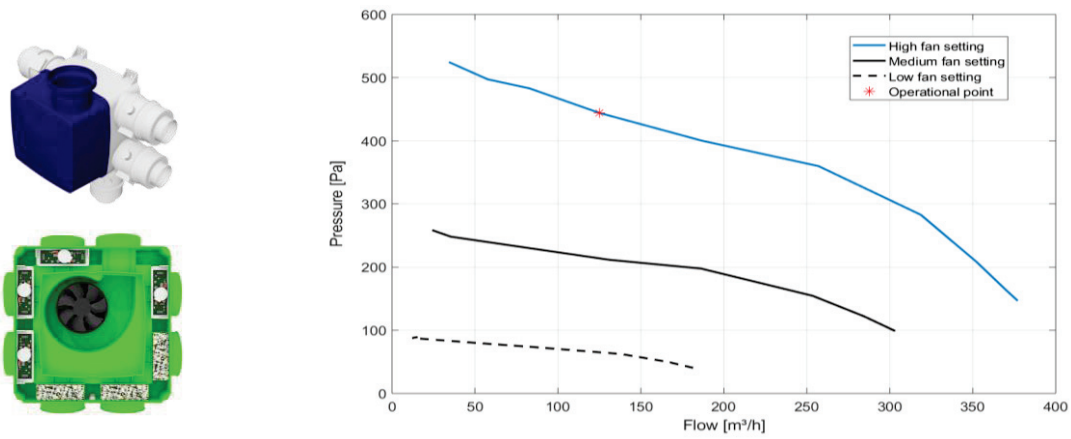

Figure 1 – The constant pressure centrifugal fan (above), the constant power centrifugal fan (below) and fan curves showing a possible operational point of a centrifugal fan (right)

3. THE MEASUREMENT METHOD

3.1 Passive one-port characterization

Previous measurements, where the centrifugal fan was considered a two-port source, determined that the transmission of sound between the inlet and outlet of the fan can be neglected. The inlet and outlet of the centrifugal fan are thus acoustically uncoupled from each other. As only the sound generated at the inlet of the fan is of interest in a ventilation system C, the centrifugal fan is considered an acoustic one-port source. An acoustic one-port source, as depicted in figure 2, can be described by equations relating the upstream and downstream propagating waves. Only plane waves are considered, constricting the frequency range to below the cut-off frequency of the first transversal mode. When extending the characterization to higher frequencies, a multi-port model is needed. In a duct with a diameter of 0,125 meter, the cut-off frequency of the first transversal mode is 1607 Hz.

The acoustic field in the inlet duct of the fan can now be described by two variables, the complex amplitudes p_i^+ and p_i^- of the upstream and downstream propagating pressure waves in the duct, as shown in figure 2. These amplitudes can't be measured directly and are computed from a measurable physical quantity, the acoustic pressure fluctuations p_i' due to an external excitation. These fluctuations are measured at four distinct axial positions in the duct. The decomposition method is explained in detail by Munjal (5). The iterative plane wave decomposition method used starts with a multiple microphone decomposition method (MMM). The MMM is executed with the measurements from four flush mounted microphones in the inlet duct, resulting in an overdetermined system of equations. The equations used in this characterization method are reformulated in terms of transfer functions with a reference signal correlated to the excitation signal, to suppress both the influence of measurement errors and the influence of pressure fluctuations uncorrelated to the excitation signal. With the initial values of the complex wave amplitudes calculated with the MMM, the final microphone spacing is obtained, the mean flow velocity and speed of sound are updated and a frequency dependent attenuation coefficient is computed. This procedure is repeated until all these parameters converge. This method is described by Denayer et al. (6).

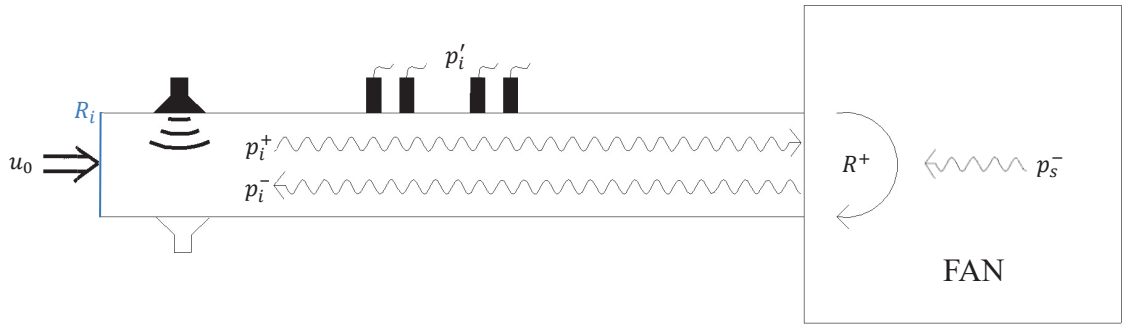

Figure 2 – Schematic overview of the experimental configuration for the characterization of an acoustic one-port source

The passive one-port model uses a reflection coefficient of the fan R^+ to describe the acoustic behavior of the fan in frequency-domain. This coefficient relates upstream and downstream propagating pressure waves in the inlet of the duct, and describes all linear sound attenuation and amplification mechanisms:

$$p_i^- = R^+ \cdot p_i^+. \tag{1}$$

The reflection coefficient is determined for various operating conditions of the fan, to determine if the reflection coefficient depends on the working point of the fan. The upstream termination of the test rig can also be represented by a reflection coefficient R_i, as there are no noise sources present at the termination of the measurement setup:

$$p_i^+ = R_i \cdot p_i^-. \tag{2}$$

Both reflection coefficients are determined with at least two independent measurements with a different flow rate, using a dominant excitation upstream of the microphone array in the inlet duct. The dominant excitation source is uncorrelated to the sound generated by the fan, allowing to neglect the source spectrum of the fan when determining the reflection coefficients. The passive one-port

characterization is reformulated in terms of transfer functions with the excitation signal to ensure that the determination is free from aerodynamic contamination.

3.2 Active one-port characterization

The passive characterization does not consider the aerodynamic noise generation inside the fan. The source term p_s^- represents the non-linear noise generation and dissipation phenomena due to the fan. The passive one-port model stated in equation (1) can now be extended to:

$$p_i^- = R^+ \cdot p_i^+ + p_s^-. \tag{3}$$

The active source vector can now be computed by applying the plane wave decomposition method (5) to the acoustic pressure measured without external excitation and with:

$$p_s^- = p_i^- - R^+ \cdot p_i^+. \tag{4}$$

However, a reference signal correlated with the source spectrum of the fan is not available. Since no external excitation is used, the active one-port model can't be reformulated in terms of transfer functions. However, by inserting equation (2) for the reflection coefficient of the termination of the test rig in equation (4), both passive models are combined and the active source spectrum can be expressed in terms of the upstream propagating waves:

$$p_s^- = (1 - R^+ \cdot R_i) \cdot p_i^-. \tag{5}$$

This greatly simplifies the analysis and additionally simplifies the plane wave decomposition from the measured pressure spectra. As a result, the influence of measurement errors is minimalized. As centrifugal fans generate broadband noise, the source spectrum is determined in a statistical description, the cross-spectral source term G_s defined as:

$$G_s = p_s^- \cdot p_s^{-*}. \tag{6}$$

This characterization method enables the characterization of the sound generated by a centrifugal fan independent of downstream reflections, thus not requiring the construction of an anechoic termination at the inlet duct of the fan.

4. MEASUREMENT SET-UP

The acoustic characterization of a centrifugal fan is executed in a semi-anechoic room to eliminate possible noise disturbances. The aerodynamic load of the piping system is changed by changing the position of a butterfly valve. This valve is situated in the outlet duct of the fan. To measure the velocity of the air flow through the fan, an L-type pitot tube from Sensing Precision is mounted on the outlet duct of the fan. The flow velocity is measured with an accuracy of ±2.5% of the reading.

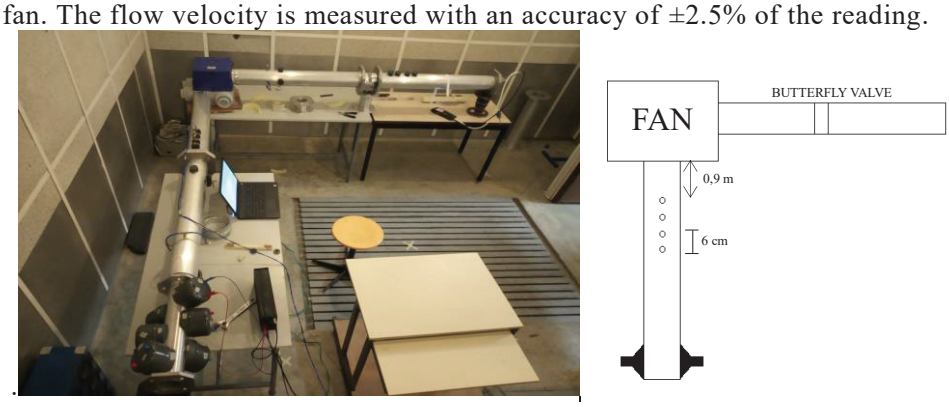

Figure 3 – Picture and overview of the measurement setup for the characterization of a centrifugal fan

The pressure spectra inside the inlet duct is measured using four flush-mounted ¼" microphones (type PCB 378C10), with a distance of 6 cm, 8 cm and 6 cm between two microphones. The distance between the fan and the closest microphone is 0,9 meter. During the passive characterization, the external excitation signal, a stepped sine signal, is sent to a loudspeaker array after amplification. The frequency range of this signal is 200-1600 Hz, with a frequency step of 50 Hz. No external excitation is used during the active characterization. The output of the microphones, as well as the excitation signal as reference signal, are sent to a Supervisory Control And Data Acquisition System (SCADAS) via coax cables. The results are exported using Siemens LMS Test.Lab 17 and processed with MATLAB.

5. RESULTS

5.1 Passive characterization

The reflection coefficient of the propagating waves at the inlet of a centrifugal fan is determined for different settings of the fan, for different rotational speeds of the fan and for the two different types of fans. Results are shown in figure 4.

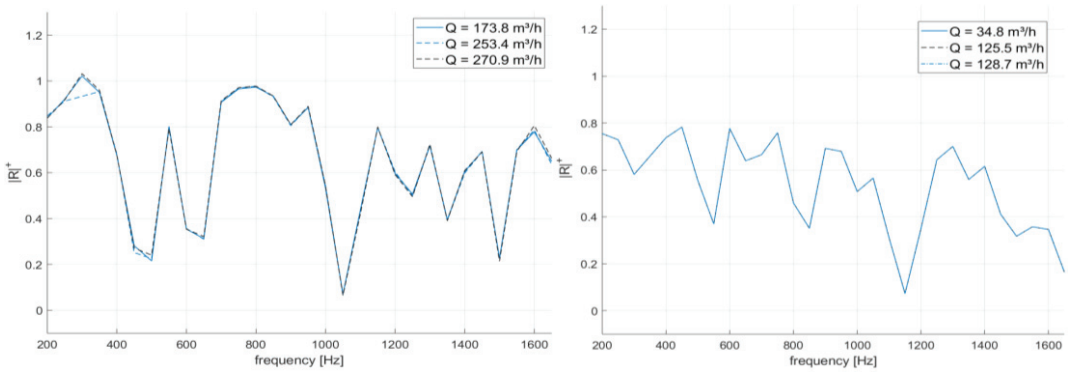

Figure 4 - The reflection coefficient R^+ of the propagating waves at the inlet of a constant pressure centrifugal fan (left) and a constant power centrifugal fan (right)

The reflection coefficient of a centrifugal fan is hence independent of the setting and the rotational speed of the fan, as the coefficient remains constant when the operating conditions of the fan change. The reflection coefficient is frequency dependent, as the collector manifold directly coupled to the fan can cause resonances dependent on the shape of the cavity. Subsequently, the downstream reflection component of the test rig or the reflection coefficient of the termination of the inlet duct, is determined. The reflection coefficient of the termination of the inlet duct has the same trend as the reflection coefficient of an unflanged finite duct (7). At the termination of the duct, the loudspeaker array causes a discontinuity in the diameter of the duct. The reduction in area results in a reflection of the incoming waves.

5.2 Active characterization

The active source spectra of the sound generated by two different types of centrifugal fans under varying operating conditions are determined. To be able to determine the narrowband source spectrum, the reflection coefficients computed in the passive characterization are interpolated over the frequency domain with a frequency step of one hertz. The operating conditions of the fans are altered by both changing the aerodynamic load of the piping system and by changing the setting or conditions of the fan itself. The characterization results in a narrowband source spectrum on a linear frequency scale, that describes the broadband sound generated by centrifugal fans in detail, as shown in figure 5 and figure 6.

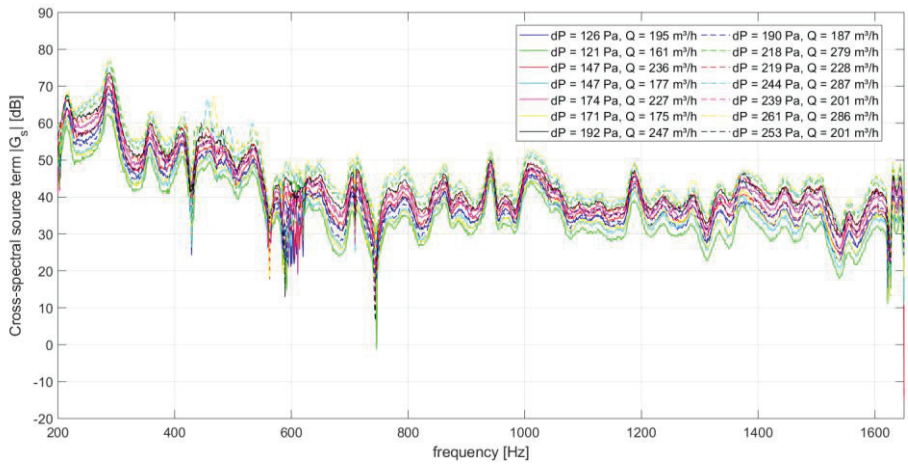

Figure 5 – The measured total source spectrum (*re* 20 µPa) of the sound generated by a constant pressure centrifugal fan under varying operating conditions

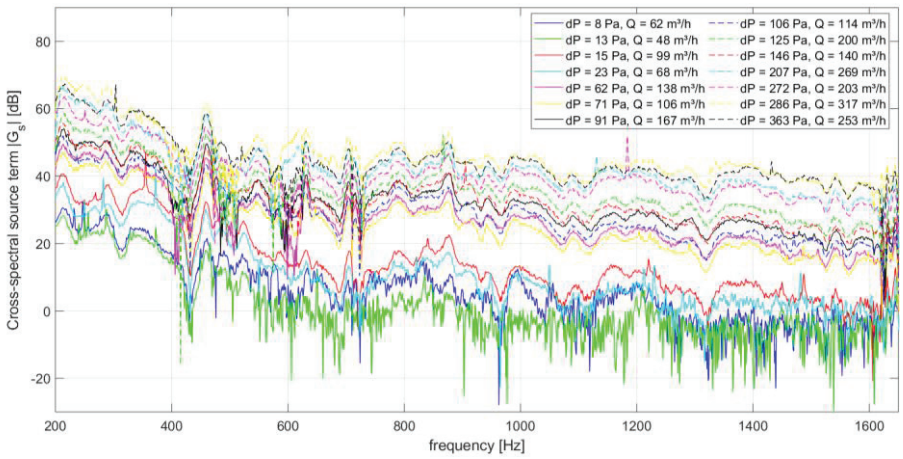

Figure 6 – The measured total source spectrum (*re* 20 µPa) of the sound generated by a constant power centrifugal fan under varying operating conditions

The sound generated by the fans has a global broadband character. The shape of the source spectra agree with source spectra of fans determined in previous studies, as there is a steep decay in the magnitude of the spectra at low frequencies, and a slow decay at higher frequencies. The sound generated by centrifugal fans is strongly influenced by various installation effects, e.g. the mechanical design of the fan and the design of the ventilation system. These installation effects result in peaks in the source spectrum at certain frequencies. When the operating conditions of the fan change, the position of the peaks caused by these installation effects remain constant. The installation effects are not taken into account in guidelines and reference spectra. The source data is sensitive to the operating conditions and the inlet flow conditions of the fan, as the source spectrum of a fan shifts when the conditions are changed. The blade frequency of the fan is not present in the source spectra, as there is no peak in the source spectrum of which the frequency changes when the rotational speed of the fan changes. The tonal peaks present on the source spectra are hence not related to the rotational speed of the fan.

6. SCALING LAWS

6.1 Existing scaling laws

Beranek (8) states that the overall sound power output L_W of a fan can be calculated with equation (7), where $\dot{V}$ is the volume rate of the flow, p is the total pressure increase and L_{WSM} is the specific sound power level of the fan. The value of L_{WSM} is constant, and depends on the assembly type of the fan. For a centrifugal fan this is equal to 34 dB according to the VDI (3).

$$L_W = L_{WSM} + 10\log_{10}\dot{V} + 20\log_{10} p \tag{7}$$

The overall scaled sound power level can thus be calculated with equation (8), this scaling is not proportional to the aerodynamic power of the fan and is shown in figure 7. The cross-spectral source term is converted to the sound power level with equation (9).

$$L_{WSM} = L_{W,exp} - 10\log_{10}\dot{V} - 20\log_{10} p \tag{8}$$

$$\text{with } L_{W,exp} = 10\log_{10} G_{ss} - 3\ dB + 10\log_{10} A \tag{9}$$

The VDI estimates the noise spectrum of a fan by superimposing the overall sound power level on a discrete spectrum $\Delta L_{W,Okt}$ whose strength depends on the type of fan. This predicted trend of the sound spectrum generated by a fan is also shown in figure 7.

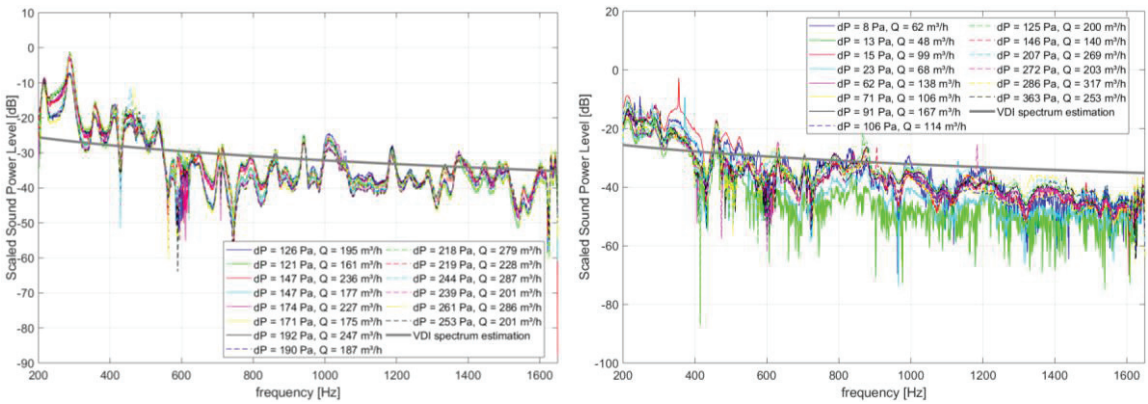

Figure 7 – The scaled total source spectrum according to Beranek of a constant pressure centrifugal fan (left) and a constant power centrifugal fan (right)

The relation between the generated sound spectrum and the operating conditions of the fan is accurately predicted by the VDI. However, the L_{WSM} determined for centrifugal fans seems to be a gross overestimation. To compare the overall sound power level generated by a fan to the estimated value by the VDI, measurements of the entire frequency scale of the source spectrum should be conducted. The estimated trend of the frequency spectrum of the sound generated by centrifugal fans is accurate, but does not include the influence of installation effect on the spectrum. These can however not be neglected.

6.2 Improved scaling law

As previously stated, the shape of the source spectra of centrifugal fans does not depend on the rotational speed of the fan as there are no blade frequency peaks present in the source spectra. However, the magnitude of the frequency spectrum does depend on the pressure increase across the fan and the volumetric flow rate through the fan. Therefore scaling laws can be determined that relate the magnitude of the spectrum to the operational point of the fan. Different existing scaling laws stated in literature, e.g. the scaling laws determined by Bies et al. (9), are applied on the measurements of the source spectra under varying operational conditions. It is determined that the sound generated by a centrifugal fan relates to the aerodynamic power delivered by the fan, as applying the scaling law stated in equation (10) aligns the source spectra with the least deviation. The source spectra scaled to the pressure across the fan p and the flow velocity through the fan u with formula (10) are shown in figure 8 and 9. The scaling law determined is also related to the flow velocity and the horsepower needed to drive the fan as stated in equation (11).

$$L_{WSM,improved} = L_{W,exp} - 10\log_{10} u^3 - 10\log_{10} p^{\frac{3}{2}} \tag{10}$$

$$L_{WSM,improved} \approx 10\log_{10} u^6 \approx 20\log_{10} hp \tag{11}$$

If the $L_{WSM,improved}$ of a fan is known, the source spectrum generated by the fan can be predicted under all possible working conditions. This prediction can be used to design silencers with the source-load interaction taken into account (2).

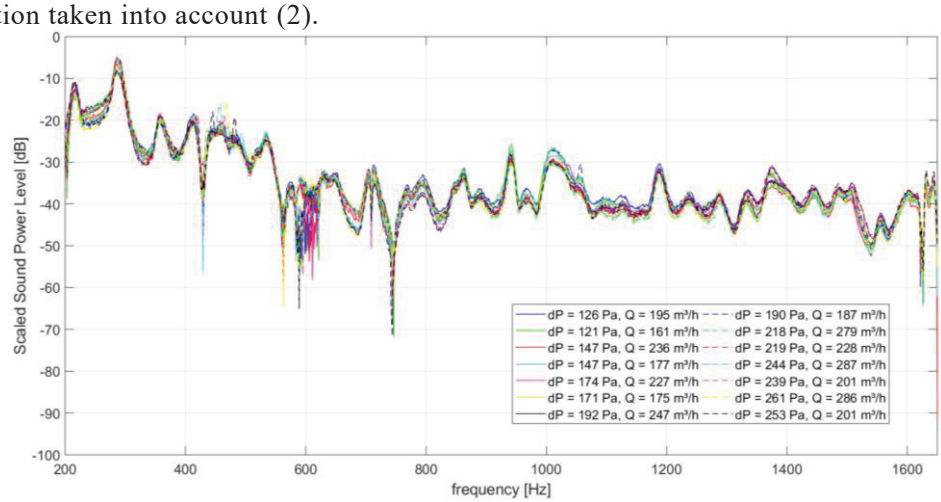

Figure 8 – The scaled total source spectrum of the sound generated by a constant pressure centrifugal fan

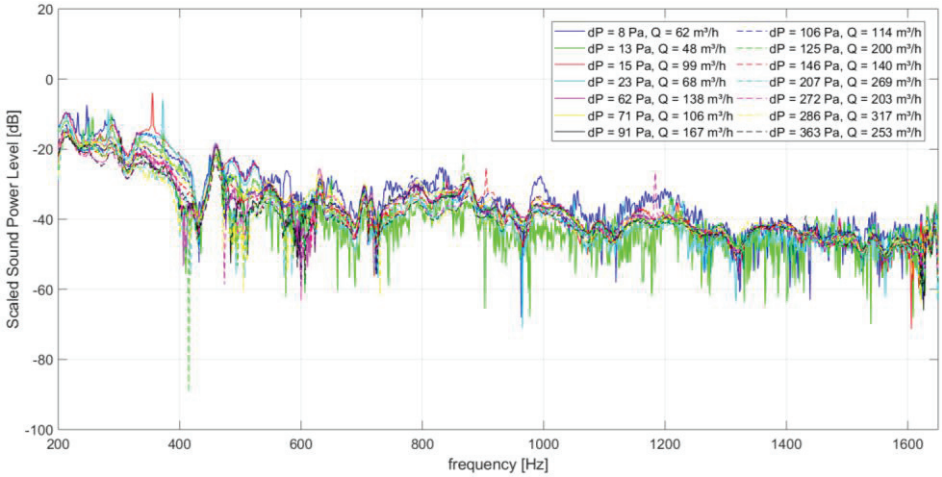

Figure 9 – The scaled total source spectrum of the sound generated by a constant power centrifugal fan

CONCLUSIONS

Two types of centrifugal fans are acoustically characterized by determining the source spectrum of the fans under varying operating conditions. The measurement method consists of two steps. First, the reflection coefficient of the incoming waves at the inlet of the fan is determined using an external excitation. Additionally, the downstream reflection component of the termination of the inlet duct is characterized. This eliminates the need for an anechoic termination to characterize the fan. Secondly, both passive models are combined to compute the active source spectrum describing all noise generation and dissipation phenomena of the fan, as well as the installation effect due to the mechanical design of the fan. The sound generated by a centrifugal fan largely depends on the working point of the fan. The source spectra shifts proportionally to the aerodynamic power generated by the fan. The measured source spectra of the fans are scaled to the pressure increase across the fan and the flow rate through the fan. The scaling to u^3 and $p^{3/2}$ aligns the source spectra measured under varying operational conditions and results in a more accurate specific sound power level of the fan compared to existing scaling laws.

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