CHALLENGES AND OPPORTUNITIES FOR FLOW NOISE PREDICTION IN HVAC SYSTEMS

Oscar KÅREKULL[1,2], Gunilla EFRAIMSSON[3],
Mats ÅBOM[2]

[1] *Marcus Wallenberg Laboratory, KTH, Stockholm, Sweden*

[2] *Fläkt Woods, 551 84 Jönköping, Sweden*

[3] *Aeronautical and Vehicle Engineering, KTH, Stockholm, Sweden*

SUMMARY

This paper investigates the possibilities of noise prediction in Heating Ventilation and Air Conditioning (HVAC) systems using semi empirical scaling laws. An approach is presented where the general noise reference spectra are combined with Reynolds Average Navier Stokes (RANS) simulations. Focus is at applying the suggested noise prediction approach to common HVAC components but also to discuss the properties of the prediction model, e.g. radiation characteristics and chosen reference spectra. A model is presented, using a momentum flux assumption of the noise sources, which is validated by a range of HVAC components of both high and low pressure loss.

INTRODUCTION

The Heating, Ventilating and Air Conditioning (HVAC) system is supposed to deliver a healthy indoor climate where apart from temperature, air quality and air velocity, a low noise level is crucial. A challenge of HVAC components is to deliver a low noise level in combination with high energy efficiency. In addition, on a global market, products need to be developed for a wide range of conditions. For example a warm climate has a higher cooling demand compared to the airflow than in a cooler climate. Different means of limiting airflow needs to be introduced into products and the noise generating conditions will be tougher to control. For more flexible product solutions, a better understanding and the ability to predict the acoustic properties is desirable.

Traditionally the focus of the HVAC system flow noise has been the sound generation from fans, while the fan historically has been the main noise source of the system. Due to the increased demands of energy efficient solutions, larger fans rotating at lower speeds are nowadays a practise. Lower speed decreases the fan noise level and increases the importance of flow generated noise in other system components, e.g. dampers, bends and air terminal devices. The frequency contents of the HVAC flow noise can overall be divided into fan related low frequencies and mid and high frequencies from flow noise of other system components.

General guidelines and reference spectra for air conditioning systems can be found in e.g. ASHREA Handbook for HVAC Applications [1] or VDI 2081 Noise generation and noise reduction in air-conditioning systems [2]. Optionally, the noise can be predicted by semi empirical scaling laws [3] where the component flow characteristics can be gained from Computational Fluid Dynamics (CFD) simulations [4]. A number of publications [5-12], summarized in table 3, have evaluated the semi empirical scaling laws and their generality by introducing measurement data of different geometries. Kårekull et. al. recently reviewed and presented these publications including some new data in a consistent way [4].

This paper describes the application of a noise prediction approach, to a selected number of HVAC applications, according to a recent formulation of the semi empirical scaling laws where the dipole force scale to the momentum flux [13]. First the prediction model is presented and the measurement setup for the evaluation is described. Secondly, the prediction model is applied to measurement data of different air terminal devices and baffle silencers. Finally the opportunities and challenges of the proposed noise prediction approach as a tool for HVAC system noise design is discussed.

NOISE PREDICTION APPROACH

The semi empirical scaling laws, originally suggested by Nelson and Morfey [3] in 1981, are an option for noise predictions to avoid a time consuming fully resolved simulation or a full scale measurement. The scaling laws can be seen as a combination of generalized noise measurement data and component flow characteristics. The model is valid for low Mach number flows and considers only dipole sound sources related to the flow separation. The effect of e.g. whistling or disturbed inflow is not included into the model. The sound power up or downstream of a constriction, is defined as the product of a force, F, and a function describing the radiation properties, R, given by

$$W_D = R(He)\, S_{FF}(St) \tag{1}$$

where S_{FF} is the force auto-spectrum as a function of the Strouhal number (St) and R is the radiation resistance for an infinite duct as a function of the Helmholtz number (He) [4]. With the assumption that the force auto-spectrum can be split into a frequency independent mean force part, $\overline{F}$, and a source strength spectrum part, K^2, [3] the sound power can be described by

$$W_D = R(He)\, \overline{F}^2 K^2(St). \tag{2}$$

The mean force, as originally proposed by Nelson and Morfey [3], can be determined from the component pressure drop by

$$\overline{F} = A\, \Delta P \tag{3}$$

where ΔP is the stagnation pressure drop of the constriction and A is the duct cross section area. Alternatively, to include constrictions of high pressure loss as recently suggested in [13], the force can be assumed to scale with the momentum flux which yields

$$\overline{F} = \frac{\rho_0 A^2 U^2}{A_{vc}} \tag{4}$$

where A_{vc} is the area at vena contracta, U is the flow velocity and ρ_0 is the air density in the duct. Oldham and Ukpoho [5] derived A_{vc} for an in-duct orifice and argued for the validity of this A_{vc} for a wider range of constrictions. A_{vc} is given by

$$A_{vc} = \frac{A}{1+\sqrt{C_L}} \tag{5}$$

where C_L is the pressure loss coefficient which can be found for the most common geometries in handbooks by e.g. Idelchick [14] or Blevins [15]. Alternatively, C_L can be determined from measurements or simulations by

$$C_L = \frac{\Delta P}{0.5\rho_0 U^2}. \tag{6}$$

For the evaluation of HVAC components, the reference spectrum K^2 is here, using the momentum flux force definition and the definition of A_{vc} from Eq (5), given by

$$K^2 = \frac{W_D A_{vc}^2}{R(He)\rho_0^2 A^4 U^4}. \tag{7}$$

The radiation resistance, R, will depend on the modes of the duct. For frequencies below the first cut-on mode of the duct the radiation resistance yields

$$R = \frac{1}{2A\rho_0 c_0} \tag{8}$$

where the first mode of the duct, i.e. the wave number k_0 , can be determined e.g. as described in [4]. Above cut on, all propagating modes needs to be considered and a summation of them will return the generated sound power as a function of the Helmholz number. For $k>k_0$ the radiation resistance for a circular duct, rewritten from [5], and for a rectangular duct, rewritten from [3], are given by

$$R_{rect.} = \frac{k^2\left(1+(3\pi/4k)(a+b)/A\right)}{12\pi\rho_0 c_0} \tag{9}$$

$$R_{circ.} = \frac{k^2\left(1+(3\pi/4rk)\right)}{12\pi\rho_0 c_0} \tag{10}$$

where k is the wave number, c_0 is the speed of sound in air, a and b the dimensions of a rectangular duct and r is the radius of a circular duct.

An end duct constriction can be seen as an in duct constriction radiating into a duct of infinite diameter for which Eq (9) and (10) will, as recently suggested in [13], approach

$$R = \frac{k^2}{12\pi\rho_0 c_0}. \tag{11}$$

Eq (11) will here be used as the radiation resistance for end duct geometries, e.g. air terminal devices.

When using the semi empirical scaling law, the frequency is traditionally scaled by the Strouhal number given by

$$St = \frac{f\,d_c}{U_c} \tag{12}$$

where f_c is the band centre frequency, d_c is a characteristic dimension and U_c is a characteristic velocity in the constriction geometry. In a complex geometry, the dimension selection may not be straight forward. In those cases d_c and U_c can be gained from the pressure loss coefficient [4] and the Strouhal number is given by

$$St = \frac{f_c \sigma^{1.5} \sqrt{\frac{4A}{\pi}}}{U} \tag{13}$$

where σ is the openness of the component i.e. A_{vc}/A. The parameters for the Strouhal number can also be obtained via pressure drop calculations using Reynolds Average Navier Stokes (RANS) simulations [4].

The semi empirical scaling law, i.e. Eqs. (7) and (13), will be used for the evaluation of the noise prediction for HVAC air terminal devices and baffle silencers. It can be noted that the proposed scaling is an extension of the original scaling suggested by Nelson&Morfey [3], which is valid for larger pressure loss coefficients. A more detailed description of the presented noise prediction approach can be found in [4] and [13].

MEASUREMENT SETUP

The measurements were conducted at the Fläkt Woods laboratory in Jönköping, Sweden. A sketch of the measurement setup is presented in Fig 1. A fan followed by silencers, positioned outside of the reverberation chamber, produced the air flow in duct. The airflow was determined by pressure drop measurement, over a calibrated nozzle, according to ISO 5167 [16]. Sound power measurements were conducted in a reverberation chamber according to ISO 3741 [17] and the sound absorption of the chamber was determined by a reference noise source, calibrated according to ISO 6926 [18]. The sound power levels were determined in third octave bands using a frequency range of 200 Hz to 4 kHz. The band levels were corrected for the effect of duct end reflections according to ISO 5135 [19].

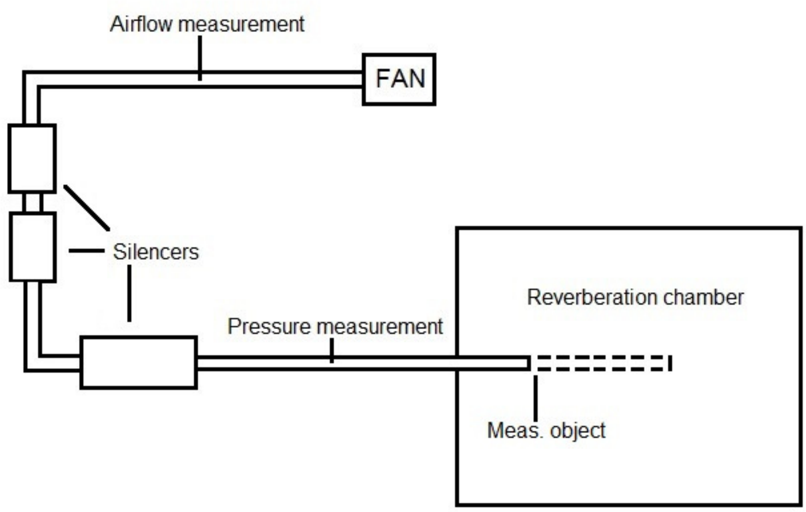

Figure 1: Measurement setup

AIR TERMINAL DEVICES

The flow noise generated by an air terminal device can be the dominating noise source in a ventilated room. A part of the HVAC system design process is to validate that the specified maximum noise level is met. The possibility of noise prediction of an ATD, i.e. a duct end constriction with similarities to orifice geometries, using the semi empirical scaling law, described above in Eqs. (7) and (13), is here evaluated for three air terminal devices. The ATDs, presented in Fig 2, are manufactured by Fläkt Woods under the names STI, VST and STQA.

Measurements, using the previously described measurement setup, were conducted for duct diameters and product sizes equivalent to duct diameters of 125 and 160 mm for STI and VST but only for 125 mm for STQA. In addition, the pressure loss of the chosen air terminal devices are variable and an average of two different pressure loss settings is included, i.e. C_L corresponding to approximately 8 and 17 for STI and VST and to 17 for STQA. Flow velocities in the duct were between 1 and 4 m/s.

Figure 2: Air terminal devices: VST to the left, STI in the middle and STQA to the right

The dimensionless source strength spectra for the air terminal devices are presented in Fig 3 and compared to the general orifice spectrum suggested in [4] and given by

$$K^2(St) = 65 \qquad St < 1 \qquad (14)$$

$$K^2(St) = 65 - 28\log(St) \qquad St > 1\,. \qquad (15)$$

As seen from Fig 3 the proposed general orifice scaling from Ref. [4] gives a quite good collapse also for different air terminal devices. There is a small tendency to over predict the actual levels with a few dB but the results are still very encouraging. The lower levels of e.g. STI may be the result of a design for low noise generation which would also argue for a lower level than the general orifice case. Many HVAC air terminal devices and chilled beams consist of combinations of orifice like geometries and an opportunity to predict noise from these geometries with a satisfying accuracy seems possible with the proposed noise prediction approach.

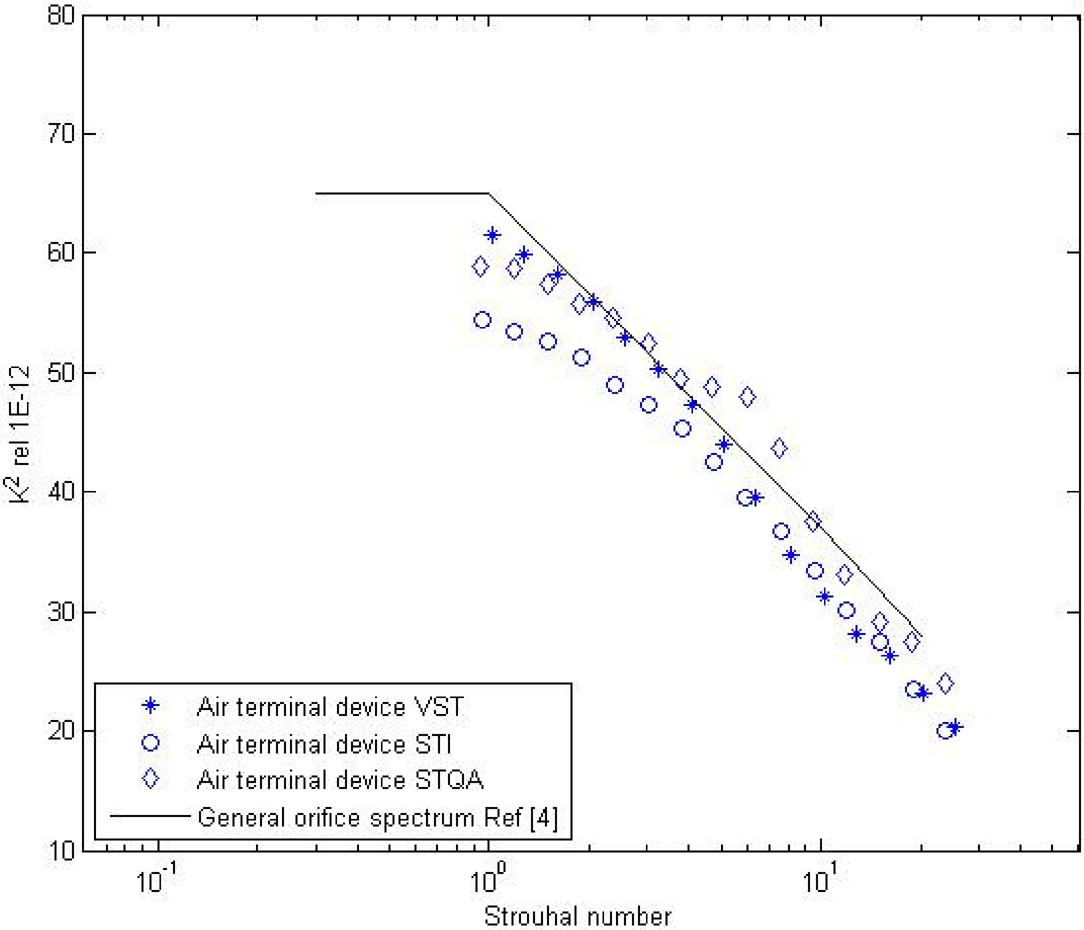

Figure 3: Dimensionless source strength spectra for air terminal devices

BAFFLE SILENCERS

Flow noise estimation of silencers with baffles is, like air terminal devices, a common application for flow noise generation in HVAC systems. The width and height of a silencer can be in the range of several meters, for e.g. rectangular silencers, down to diameters less than a quarter of a meter for e.g. circular silencers. To adjust the airflow to fit the measurement facilities, a circular silencer with an outer diameter of 335 mm, an inner diameter of 315 mm with one centered baffle approx. 100 mm thick and the full duct width wide, was chosen. The silencers, presented in Fig 4, are manufactured by Fläkt Woods under the name BDER-35-315 and designed to fit a circular duct of 315 mm diameter. Silencer walls and the baffles are filled with sound absorbing glass wool. Two lengths of silencers, 0.5 m and 1.2 m, were chosen to evaluate the effect of the baffle length. The chosen silencer model has one baffle end with an end geometry of rectangular shape and one of a round shape i.e. either the inlet shape is round and the outlet shape is rectangular or vice versa. The flow noise generation was measured for flow directions entering the rectangular end and entering the round end respectively. An overview of the test cases is presented in Table 1. Duct flow velocities between 9-15 m/s were considered.

Table 1: Tested silencer versions

	Length	**Inlet baffle shape**	**Outlet baffle shape**
Short silencer, rect. outlet	0.5 m	Round	Rectangular
Short silencer, round outlet	0.5 m	Rectangular	Round
Long silencer, rect. outlet	1.2 m	Round	Rectangular
Long silencer, round outlet	1.2 m	Rectangular	Round

Figure 4: Photo of long and short silencer. Both rectangular and round baffle end shapes are seen

The dimensionless source strength spectra for the silencer test cases are presented in Fig 5. All spectra collapse well to the general orifice spectrum in Eq (15). In addition to the low frequency uncertainty of the reverberation chamber method [17], installation effects and geometry dependence can explain the deviations for low Strouhal numbers, see [4]. Comparing the spectra of the short silencer to the long silencer, the longer silencer is at a general lower level. The fact that the noise generation is not at one distinct duct cross section can be a possible explanation. In comparison to an orifice, which has a distinct flow separation point, a baffle silencer is, from a flow noise generation perspective, a more complex component. Not only is absorption introduced into the geometry but also the noise generation is, at least, divided into both inlet, along the perforated walls and outlet end of the baffles. In the longer silencer, the flow noise generated at the inlet and along the perforated walls is absorbed to a higher degree by the silencer than in the short silencer. A correct scaling would only consider the pressure drop related to the measured downstream noise generation.

For the long silencer, no difference is seen between the spectra for the flow directions entering the rectangular or the round shape baffle end. Since the flow noise generated at the front end, as previously discussed, may be absorbed by the silencer one may conclude that the end geometry is not sensitive, from a purely noise generation perspective, to the two different shapes. For the short silencer a difference at low frequencies is detected for the two end shapes. Since the end geometry is indicated not to have an effect on the noise generation, a higher noise generation is indicated for the rectangular inlet shape than the round shape. This is in accordance with the product instruction to use the round shaped baffle end as inlet.

The spectra for baffle silencers collapse well to the general orifice spectrum. The inclusion of baffle silencers in to a system design will be further discussed in the next section.

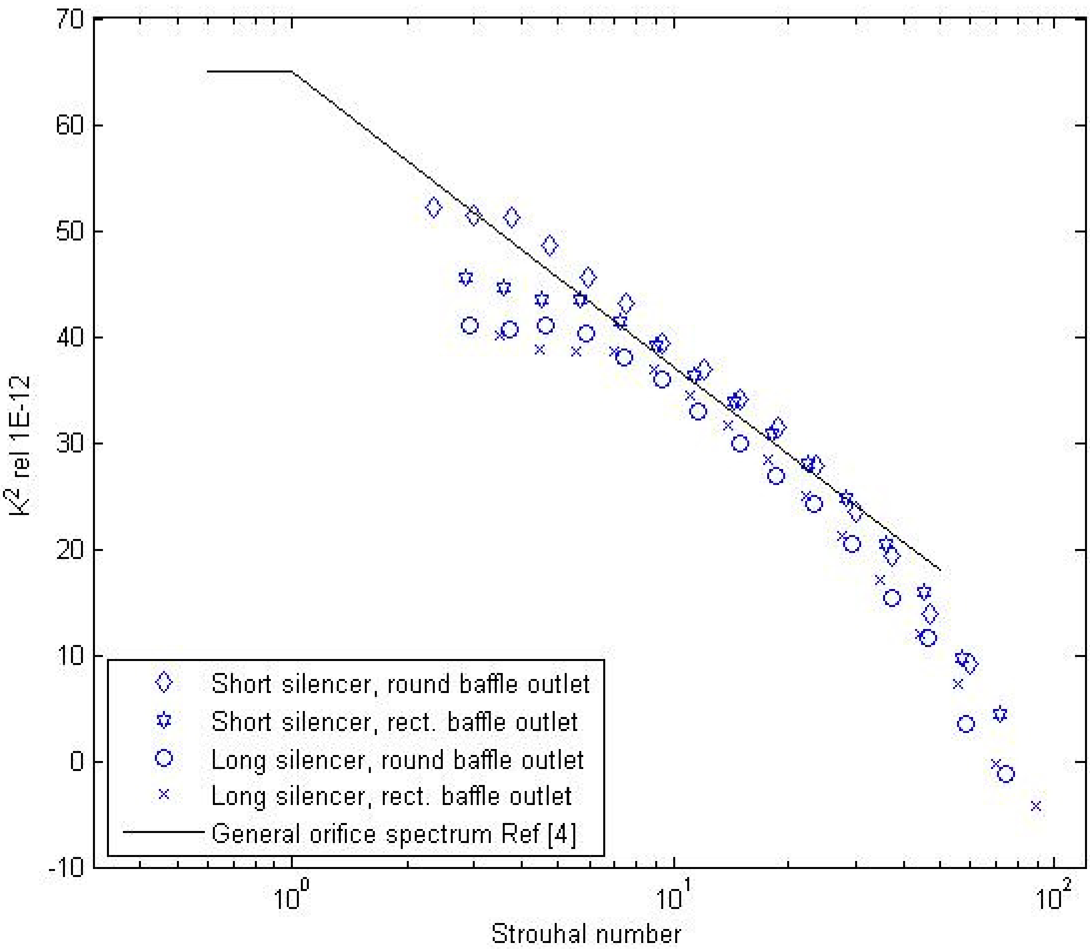

Figure 5: Dimensionless source strength spectra for silencers

GENERAL FLOW NOISE PREDICTION FOR HVAC SYSTEM COMPONENTS

When designing a HVAC system, potential flow noise generation needs to be estimated since measurement data is not available until the product and duty point is specified. It would be beneficial to avoid system changes at a later design stage, by early identify the dominating noise sources, from purely the flow characteristic information. A general and precise noise prediction for a wide range of flow duct constriction devices and geometries is not possible, using the semi empirical scaling laws, due to the variations in flow separation properties. Still, the opportunities of the suggested noise prediction approach as an approximative system design tool will be evaluated using both already published data and the here presented measurement data.

The measurement results, used in the evaluation, are presented in table 2. In addition to the air terminal devices and the short and long silencers, measurement data of a circular orifice of high pressure loss [13] and a circular bend is introduced. The published results consist of data for orifices, bends and dampers as presented in table 3. Details of the specific geometries and flow characteristics can be found in [4]. The reference spectra are presented in third octave bands as a function of the Strouhal number. When the original published data is given in octave bands a scaling factor of 10log(1/3) has been subtracted from the octave band level assuming an equal level in each band.

Table 2: Measurement data

Geometry	Flow vel. [m/s]	Duct type	Duct size [m]
Air terminal device, STI	1-4	Circ.	0.125 & 0.160
Air terminal device, VST	1-4	Circ.	0.125 & 0.160
Air terminal device, STQA	1-4	Circ.	0.125
Short silencer, length 0.5m, round baffle shape at inlet and rect. shape at outlet	6-14	Circ.	0.315
Long silencer, length 1.2m, round baffle shape at inlet and rect. shape at outlet	6-14	Circ.	0.315
Bend, r/D=1.2	9-15	Circ.	0.315
Orifice [13], d=30mm	0.1-0.3	Circ.	0.315

Table 3: Published data

Geometry	Author	Flow vel. [m/s]	Duct type	Duct size [m]
Orifices, rectangular	Nelson and Morfey [3]	2.5-27	Rect.	0.3
Orifices, circular	Oldham and Ukpoho [5]	8-18	Circ.	0.3
Orifice, circular	Allam and Åbom [10]	15-34	Circ.	0.057
Orifices, Rectangular	Mak et al [12]	10-35	Rect.	0.1
Bend, miter	Waddington and Oldham [6]	7-22	Rect.	0.4 & 0.6
Bends, miter & r/D=0.5	Gijrath et al [8]	34-120	Circ.	0.043
Damper, circular	Oldham and Ukpoho [5]	10-25	Circ.	0.3
Damper, rectangular	Ingard [11]	11-13	Rect.	0.61

The dimensionless source strength spectra, for the data in table 2 and 3, are presented in Fig 6. A shift from a constant level, at low Strouhal numbers, to a constant inclination at higher is visible in most results. Where low enough Strouhal numbers were not measured, the inclination shift might not be visible [4]. At what exact Strouhal number the inclination shift occurs at is not established. A potential parameter of importance is the duct diameter. Where the duct diameters, as presented in table 3, are less than 0.1 m the inclination shift is below Strouhal equal to one. For duct diameters larger than 0.1 m the inclination shift is around or above Strouhal one. The highest Strouhal number of the inclination shift is also demonstrated by the largest duct diameter i.e. 0.61 m for the damper by Ingard [11]. A more detailed selection of the Strouhal number may be possible e.g. by the maximum velocity and characteristic dimension from RANS simulations as suggested by Jong and Golliard [20].

A maximum and minimum spectrum can be determined from the data above the inclination shift using the inclination from the best fit calculation in [4]. The resulting average spectrum using an inclination shift at Strouhal one, is given by

$$K^2(St) = 60 \qquad St < 1 \tag{16}$$

$$K^2(St) = 60 - 28\log(St) \qquad St > 1. \tag{17}$$

Assuming normally distributed data, the standard deviation can be determined to 2-3 dB. Below the inclination shift the standard deviation is estimated to 5-6dB. For the described HVAC components, an opportunity to predict the flow noise generation using the semi empirical scaling laws, i.e. Eqs (7) and (13), is suggested. The approach may also be useful as a comparative tool for product development. The component flow characteristics can be gained from RANS simulations which enables a noise prediction without the need of measurements or fully resolved simulations.

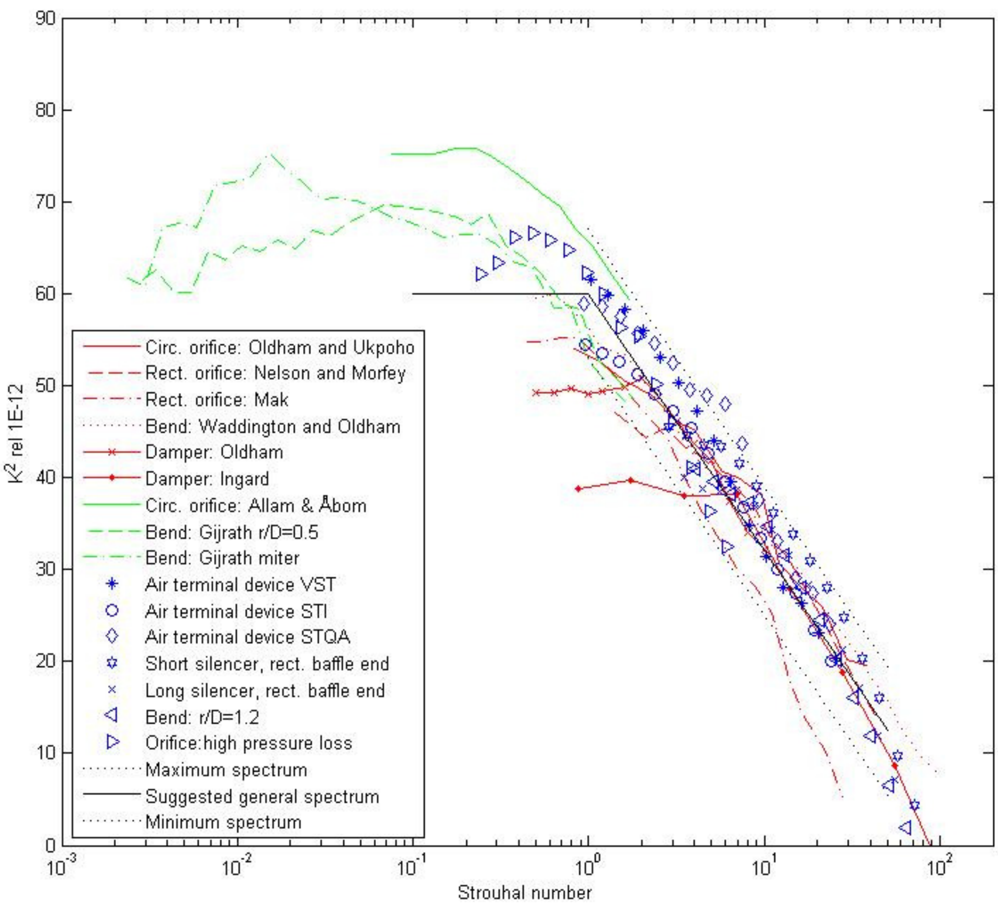

Figure 6: Dimensionless source strength spectra for measurements with duct diameters larger than 0.1m (blue), published data with duct diameters larger than 0.1m (red), published data with duct diameters less than 0.1m (green),

CONCLUSIONS

The use of semi empirical scaling laws for HVAC applications has been investigated and the original model by Nelson and Morfey has been generalized to cases with higher loss coefficients, see Eq. (7). The results indicate that a noise prediction without the exact details of the geometry, but only the pressure drop, the duct flow velocity and the duct dimensions, is possible for a number of geometries. The noise prediction approach is applicable to geometries both inside ducts and at end of ducts e.g. air terminal devices. Noise prediction of silencers with baffles is applicable but will introduce larger uncertainties since the noise generation is not at a distinct duct cross section and absorption is included in the component. To determine the accuracy of the presented general noise prediction approach, further investigations are needed. The dependency of turbulence levels and changed mean velocity profile, compared to the constriction inflow characteristics for a straight flow duct, is also an important issue for future work.

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BIBLIOGRAPHY

[1] ASHREA Handbook - *HVAC Applications* (SI), **2007**, 47.7-47.10

[2] VDI 2081 Part 1, *Noise generation and noise reduction in air-conditioning systems*, **2001**, 26-36

[3] Nelson P. A. and Morfey C. L. *Aerodynamic sound prediction in low speed flow ducts*, J. Sound and Vib. **1981**; 79: 263-289

[4] Kårekull O., Efraimsson G. and Åbom M. *Prediction model of flow duct constriction noise*. Applied Acoustics **2014**; 82:45-52 (Open access)

[5] Oldham D.J. and Ukpoho A.U. *A pressure-based technique for predicting regenerated noise levels in ventilation systems*, J. Sound and Vib. **1990**; 140: 259-272.

[6] Waddington D. C. and Oldham D.J. *Generalized flow noise prediction curves for air duct elements*, J. Sound and Vib **1999**; 222: 163-169.

[7] Oldham D.J. and Waddington D.C. *The prediction of airflow-generated noise in ducts from considerations of similarity*, J. Sound and Vib **2001**; 248(4): 780-787.

[8] Gijrath J.W.M., Verhaar B.T. and Bruggeman J.C. *Prediction model for broadband noise in bends*, Delft, TNO Institute of Applied Physics; **1999**

[9] Nygård S. *Modeling of low frequency sound in duct networks*, Licentiate Thesis, Department of Aeronautical and Vehicle engineering, Royal Institute of Technology, Stockholm, Sweden; **2000**

[10] Allam S. and Åbom M. Aero-acoustic *Study of Single and Double Diaphragm Orifices under subsonic conditions*, Royal Institute of Technology, Stockholm, Sweden; **2005**

[11] Ingard U., Oppenheim A. and Hirshorn M., *Noise generation in ducts*, ASHREA Transactions **1968** 74(1) V1.1-V1.10

[12] Mak C.M.,Wu J, Ye C, Yang J , *Flow noise from spoilers in ducts*, J. Acoust. Soc. Am. **2009** 125 (6), 3756-65

[13] Kårekull O., Efraimsson G. and Åbom M. *Revisiting the Nelson-Morfey scaling law for flow noise from duct constrictions*, Submitted to the Journal of Sound and Vibration

[14] Idelchik, I. E., *Handbook of Hydraulic Resistance*, 2nd ed., Springer Verlag, Berlin, **1986**

[15] Blevins R. D., *Applied Fluid Dynamics Handbook*, Van Nostrand Reinhold, New York, **1984**

[16] ISO 5167 *Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full*

[17] ISO 3741 *Acoustics -- Determination of sound power levels and sound energy levels of noise sources using sound pressure -- Precision methods for reverberation test rooms*

[18] ISO 6926 *Acoustics -- Requirements for the performance and calibration of reference sound sources used for the determination of sound power levels*

[19] ISO 5135 *Acoustics - Determination of sound power levels of noise from air-terminal devices, air-terminal units, dampers and valves by measurement in a reverberation room*

[20] Jong A. and Golliard J. *Towards a universal scaling for broadband turbulent noise in internal flow devices*, IMECE2013-66723; **2013**