PSYCHOACOUSTIC EVALUATION OF FAN NOISE

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SUMMARY

Besides the deduction of general rules concerning psychoacoustic parameters of fan noise coming from a plurality of axial and radial fan sound signals, two detailed examples of psychoacoustic investigation of fan noise are presented. Therein installation effects, varying operation points and rotational numbers are studied. Hearing comparisons are conducted in form of two methods – the semantic differential and paired comparison. The resulting values of annoyance factor are placed in a relationship to the psychoacoustic annoyance and a modified formulation. The latter is a first step in finding a psychoacoustic metric of fan noise, which gives a single-number value of annoyance by combining relevant psychoacoustic parameters.

INTRODUCTION

Besides aerodynamic properties the sound quality of fans gains increasingly in importance. In contrast to technical parameters like efficiency, fan noise is directly perceived and evaluated by human beings. Hence differences in the sound character which are not always captured by the commonly used physical parameters like sound pressure levels and even more detailed spectral analysis can lead to very different product assessment. The A-weighted sound pressure level merely serves for a coarse classification and adverse selection – for the characterization of subjective felt sound quality the A-weighted sound pressure level is often unsuitable.

It should therefore be an aim to develop a measure to characterize the sound quality of fan noise which is more oriented towards the subjective felt noise impressions and which is able to represent the sound emissions of fans in different applications more aurally-equivalent, i.e. in a manner that corresponds better to our hearing. The concerned special field of acoustics is the science of psychoacoustics.

For that reason the company ebm-papst built up a psychoacoustic lab with one moderator and eight listening work stations where hearing comparisons of recorded fan noise signals can be conducted by listening to the sound examples with loudspeakers and headphones.

In these listening tests the annoyance level and other acoustic relevant attributes are queried from a number of test persons by means of two methods: the semantic differential and the paired comparison.

Several examples of axial and radial fans in undisturbed laboratory setup and in different applications are investigated. One measurement series addresses the comparison between the acoustic (physical) and psychoacoustic (subjective) behaviour of different fans under different installation situations. In another investigation effects of varying operating point and rotational speed are studied.

The influence of dedicated acoustic phenomena like for example tonal components is investigated by means of signal manipulation.

In summary one can say that the annoyance level is decisively influenced by the objective psychoacoustic parameter loudness. Furthermore the tonality and especially in fan operating points near stall the roughness and especially the fluctuating strength can play an important role.

PSYCHOACOUSTIC PARAMETERS

Important psychoacoustic parameters and a respective short description are listed in Table 1 [1]. These perception variables ‘measure’ independently from each other different aspects of a sound signals.

Table 1: Short description of important psychoacoustic parameters

Psychoacoustic parameter	**Unit**	**Description**
Loudness N	sone	The subjective felt sound intensity (intensity sensation)
Sharpness S	acum	Relation between high- and low- frequency spectral fractions
Fluctuation Strength F	vacil	Fluctuations in the sound signal up to approx. 20 Hz modulation frequency – maximum at modulation frequency of 4 Hz, ref. signal: 1 kHz, 60 dB(SPL), m = 1, f_{mod} = 4 Hz $\Rightarrow$ 1 vacil
Roughness R	asper	Fluctuations in the sound signal between approx. 15 and 250 Hz modulation frequency – maximum at modulation frequency of 70 Hz
Tonality T	dB(penalty)	Sensation of tonal components which stick out from a broadband sound spectrum – often felt to be disturbing/annoying

The loudness N (DIN 45631/A1 [6]), the tonality T (DIN 45681 [7]) and the sharpness S (DIN 45692 [8]) have been already standardized.

Psychoacoustic metrics are formulas which combine two or more psychoacoustic parameters with the aim to describe the annoyance or the pleasantness of a noise class.

PSYCHOACOUSTIC LAB

Aiming at the investigation and the characterization of the fan noise sound quality the company ebm-papst built up a psychoacoustic lab with one moderator and eight listening work stations (see Fig. 1) [3] where hearing comparisons of recorded fan noise signals can be conducted by listening to the sound examples with loudspeakers and headphones. The room has movable wall elements for influencing the reverberation time and a window equipped with a transparent microperforated absorber to reduce reflections. Because of noise reduction issues the work station PCs are placed in a rack outside the room, a flat screen monitor instead of a beamer and a noise insulated air conditioning are installed. The used HEAD acoustics software gives the possibility to do acoustic and psychoacoustic analysis (ArtemiS Suite™) and perform hearing comparisons with a group of test persons (SQuare™).

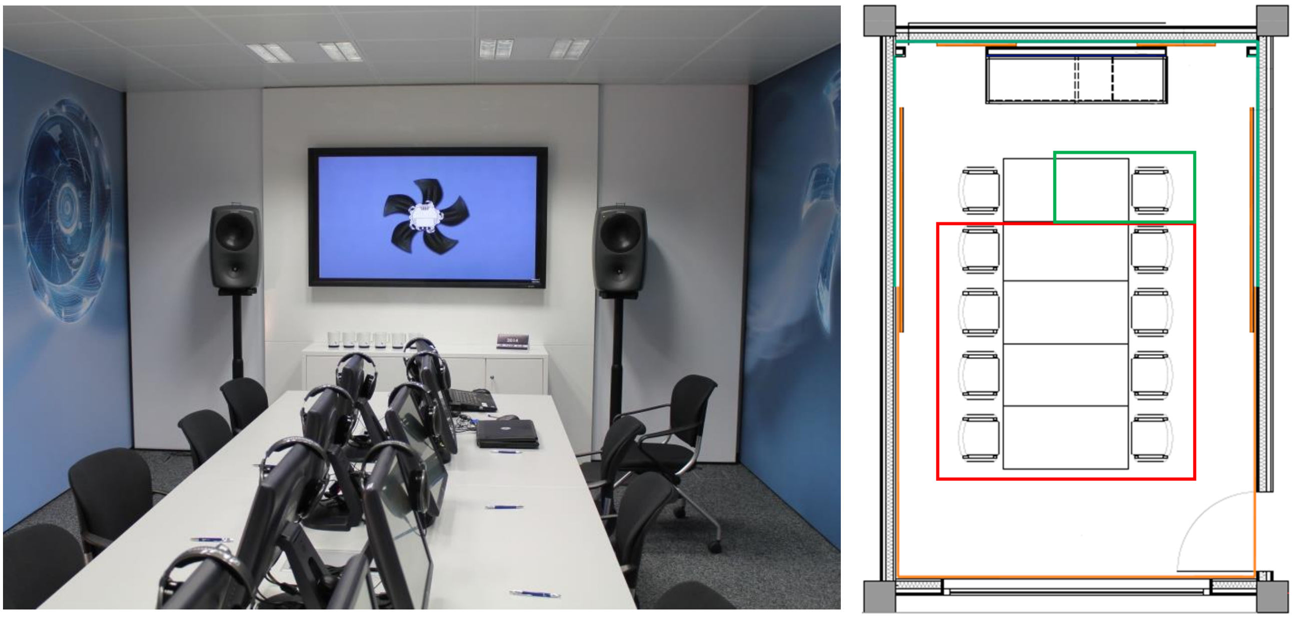

Figure 1: Picture (left) and floor plan (right) of the psychoacoustic lab at ebm-papst Mulfingen

SYSTEMATIC HEARING COMPARISONS

Hearing comparisons are used to transfer multiple dimensions of sounds and their annoyance to a single-number value. Therefore the subjective assessments need to be associated with physical and psychoacoustic parameters [2]. Eventually the different sound sensations should be integrated in a metric which specifies the annoyance or the pleasantness in a single number.

In the performed listening tests the annoyance level and other acoustic relevant attributes are queried from a number of test persons by means of two methods: while the paired comparison considers just one attribute in each case, the semantic differential addresses multiple dimensions of the sound signal. The latter serves as identification method of relevant attributes of a noise group [2]. In general several noise signals are rated by up to 30 attributes on a 7 to 9-step bipolar scale. The evaluation is carried out normally by a factor analysis where the correlations and dependencies of the different attributes are investigated. In order to reproduce the subjective perception of the sounds as precisely as possible the selection of the pairs of antonyms is of particular importance.

For the measurement series of different fans under various installation setups described later (first example, see p.5ff) 10 attributes are queried from 25 test persons on a 7-step scale. The pairs of antonyms can be classified in four groups according to Table 2 (according to [9]). The results are often presented in form of a spider chart (see Fig. 6).

Table 2: Attributes (pairs of antonyms) for the semantic differential, acc. to [9]

Quality	Evaluation	Spectral content	Time structure
• strong – weak • high quality – low quality	• quiet – loud • pleasant – annoying	• without tones – with tones • non-whooshing – whooshing • non-humming – humming • non-droning – droning • high tone – low tone	• non-fluctuating – fluctuating

The paired comparison offers the possibility to compare two sounds directly, commonly with regard to annoyance or disturbance (like in the second example, see p.8ff). This method should be preferred if the differences in the attribute specification are small, because people use it everyday in decision-making processes [2].

PSYCHOACOUSTICS OF FAN NOISE

General rules concerning psychoacoustic parameters are derived by an objective analysis of selected fan noise time signals. For that reason 52 time signals of axial fans and 58 time signals of radial fans from the ebm-papst data base are investigated [4]. The fans outer diameter ranges from 250 to 1250 mm and the rotational speed from 450 to 4000 rpm. The median absolute values of the specific psychoacoustic parameter X′ are divided by the overall sum ΣX′ and plotted vs. the critical-band rate z (see for instance [1]) with the specified resolution Δz.

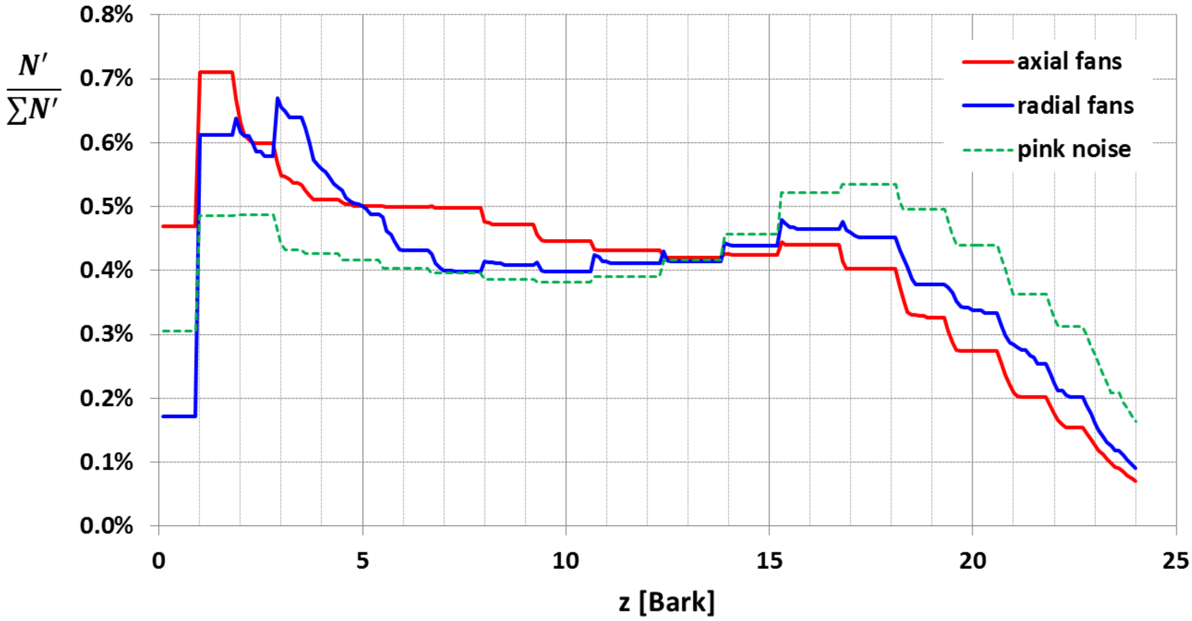

Figure 2: Median of relative specific loudness vs. critical-band rate z (Δz = 0.1 Bark) for axial (red) and radial (blue) fans, comparison to pink noise spectrum (green dashed)

Figure 2 shows the averaged curves of the relative specific loudness vs. the critical band rate z for axial and radial fans in comparison to the pink noise spectrum. According to this, the shapes are similar, but fan noise is characterized more strongly by low frequencies while pink noise is dominating in the high frequency regime greater than 15 Bark (approx. 3 kHz). Tonal components associated with the blade passing frequency (BPF) affect mainly the critical bands 2 to 4. Figure 3 shows the median curves of the relative roughness and the relative fluctuation strength again for both fan types. The averaged graphs are very similar and reveal a close match to the respective pink noise spectrum. The roughness is pronounced between 3 and 11 Bark, whereas the maximum of fluctuating strength is narrower at approximately 3 Bark. The additionally plotted three exemplary outliers show how big deviations from the median curves can be in individual cases.

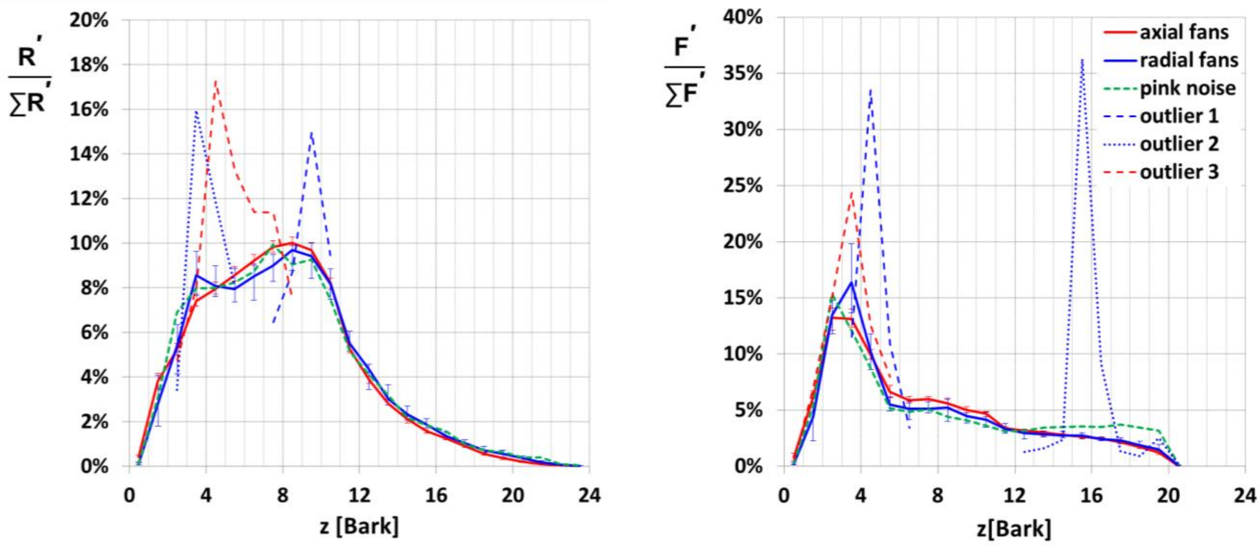

Figure 3: Median of relative roughness (left) and relative fluctuation strength (right) for axial (red) and radial (blue) fans vs. critical-band rate z (Δz = 1.0 Bark), comparison to pink noise (green dashed), three exemplary outliers

EXAMPLES OF DETAILED PSYCHOCOUSTIC INVESTIGATIONS

In the **first example** three different radial fans with backward curved blades and an outer diameter of 250 mm are investigated in four operating points under five different inflow setups [5]. Technical data of the fans are given in Table 3 (speed of best efficiency point), the chosen operation points in Table 4. The adjustment is made by variation of the rotational number. Best efficiency point for fan 1 and 3 is OP 3; fan 2 performs best in OP 2. In addition to the undisturbed inflow setup (w/o) and a heat exchanger unit (NWT) in front of the fan, three artificial but typical lateral inflow conditions representing setups were investigated: boxes of different height (N1, H1) and a diagonally metal sheet installed (N1d), see Figure 4.

Table 3: Technical data of tested fans

	Speed [rpm]	**Number of blades**
Fan 1	2015 @ OP 3	11
Fan 2	1993@ OP 2	7
Fan 3	1941@ OP 3	11

Table 4: Investigated operating points

	Volume flow [m³/h]	**Pressure rise [Pa]**
OP 1	577	260
OP 2	682	220
OP 3	767	190
OP 4	892	160

Figure 5 shows the sound pressure level Lp, the loudness N and the sharpness S for the five investigated inflow setups. The plotted values are averaged over the four selected operating points near fans best efficiency point (see Table 4). While fan 2 has the lowest levels (Lp) for straight inflow conditions (w/o, NWT), it seems to be very sensitive to lateral inflow setups, especially without diagonally metal sheet (N1, H1) which results in a clearly increased level. Fan 1 and 3 perform very similar with varying inflow configurations (with slight advantages of fan 1), in both cases the maximum levels occur with N1d. The heat exchanger (NWT) produces similar, but slightly increased values (1-2 dB) compared to the undisturbed inflow (w/o).

The question arises now is whether the sound pressure level correlates with the subjective loudness sensations. It is notable that the minimum loudness values for fan 1 and 3 occurs at the lateral inflow setup H1, whereas it is found for fan 2 in the undisturbed inflow situation (w/o) in accordance with the sound pressure level Lp. The maximum loudness for fan 1 and 3 is determined with the heat exchanger setup (NWT), where – on the contrary – fan 2 nearly reaches the minimum of configuration w/o. Another interesting finding is that the Lp maxima for fan 2 with the inflow setups N1 and H1 do not occur in the loudness diagram. In summary for the loudness N one can say that – in contrast to the sound power level Lp – a lateral inflow setup (N1, N1d, H1) has no general negative impact on the perceived loudness.

The courses of the sharpness curves are nearly identical for all of the three investigated fans. Therefore no statement can be drawn from the sharpness concerning the single radial fans, but even clearer with regard to the inflow configuration: with lateral incoming flow (N1, N1d, H1) the sharpness is reduced compared to the straight inflow (w/o, NWT). This is due to the fact that the sound levels in the low frequency regime are increased especially for lateral cases and therefore the ratio between low and high frequency spectral fractions is influenced.

Figure 6 (left) shows a spider chart as a result of the hearing comparison performed as semantic differential. The 10 attributes are queried from 25 test persons on a 7-step scale (-3 to +3). They are arranged such that the “positive” antonym is positioned at the diagram outer side (whereas “high tone – low tone” is neutral in this regard). It can be seen that for example the tone pitch is assessed clearly because the whole scale is used. On the other hand the attribute pair “quiet – loud” appears to be more difficult to evaluate probably due to the fact that no significant sound level difference is perceived although the unweighted levels differ by about 10 dB. The two sounds of fan 2 (green lines) seem to deviate somehow from the others, which are evaluated in a similar way for the majority of the attributes. Merely the sound of fan 3, w/o is perceived as low-quality, more tonal, louder and more annoying than the others. Hence a relationship between the sensation of tone pitch (where this configuration has a high value) and the subjective felt quality respectively the annoyance can be presumed. Figure 6 (right) shows the results for the important attribute “pleasant” (in pair of antonyms with annoying = -3) what often is queried as general noise evaluation in a paired comparison if only one attribute is considered. In this diagram the previously described findings become obvious: while the annoyance increases for fan 1 and 2 when putting the box N1 in front of the inlet, opposite behavior is observed for fan 3 probably due to the fact that the influence of high frequency spectral fractions is reduced or masked by the effect of N1.

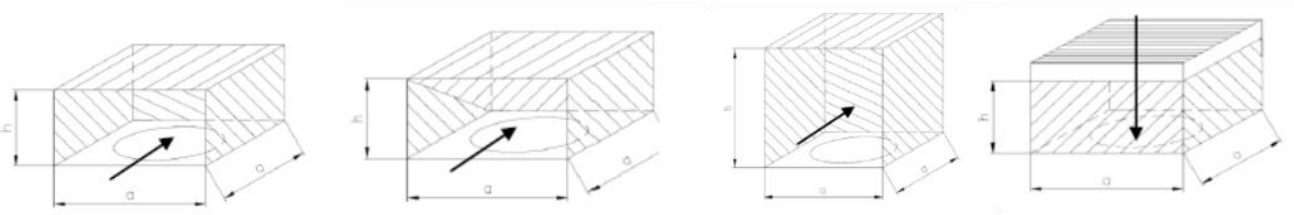

Figure 4: Different inflow setups, from left to right: N1, N1d, H1 and NWT [5]

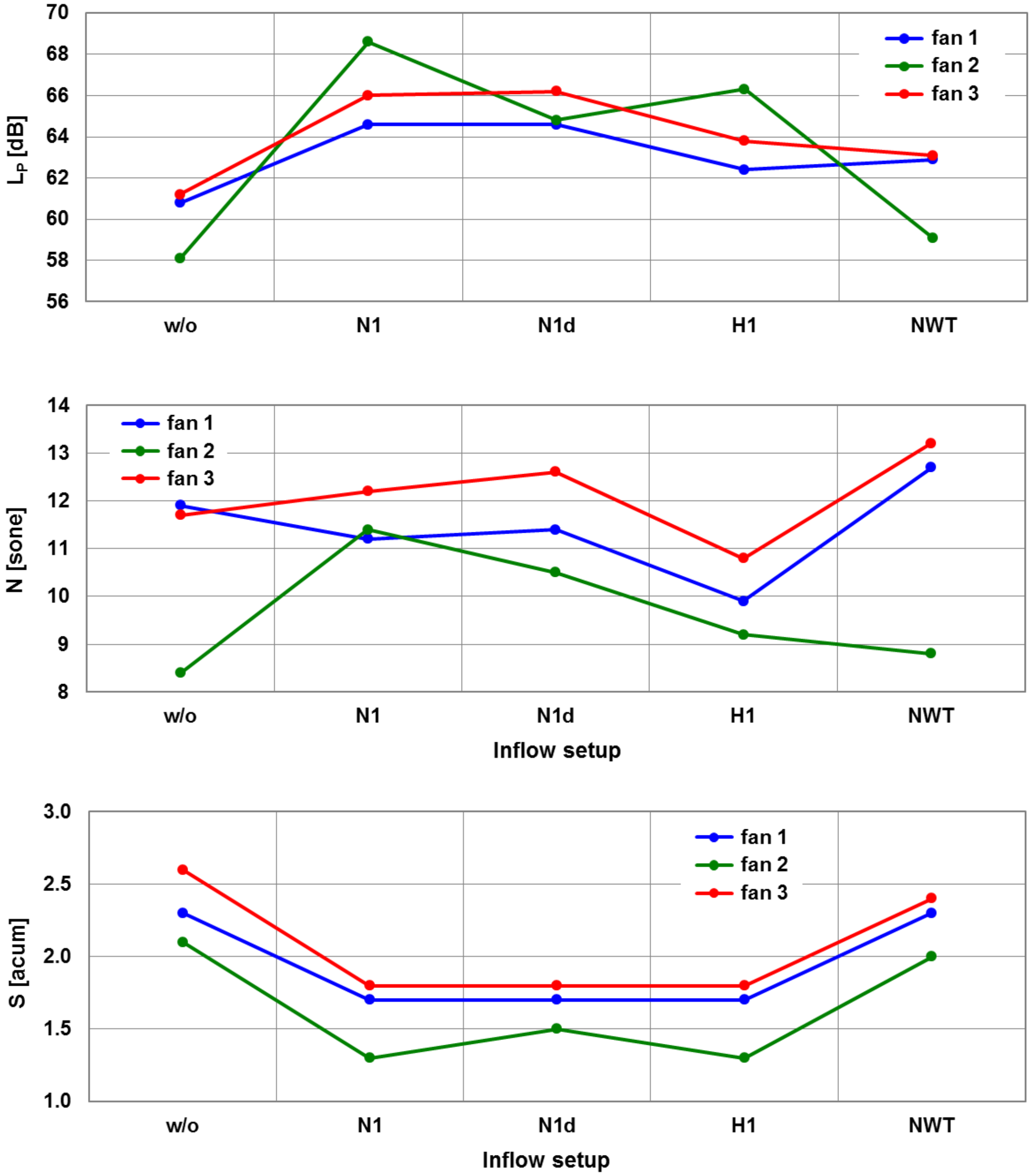

Figure 5: Sound pressure level Lp (upper), loudness N (middle) and sharpness S (lower) for the five investigated inflow setups, values averaged over four operating points near fans best efficiency point

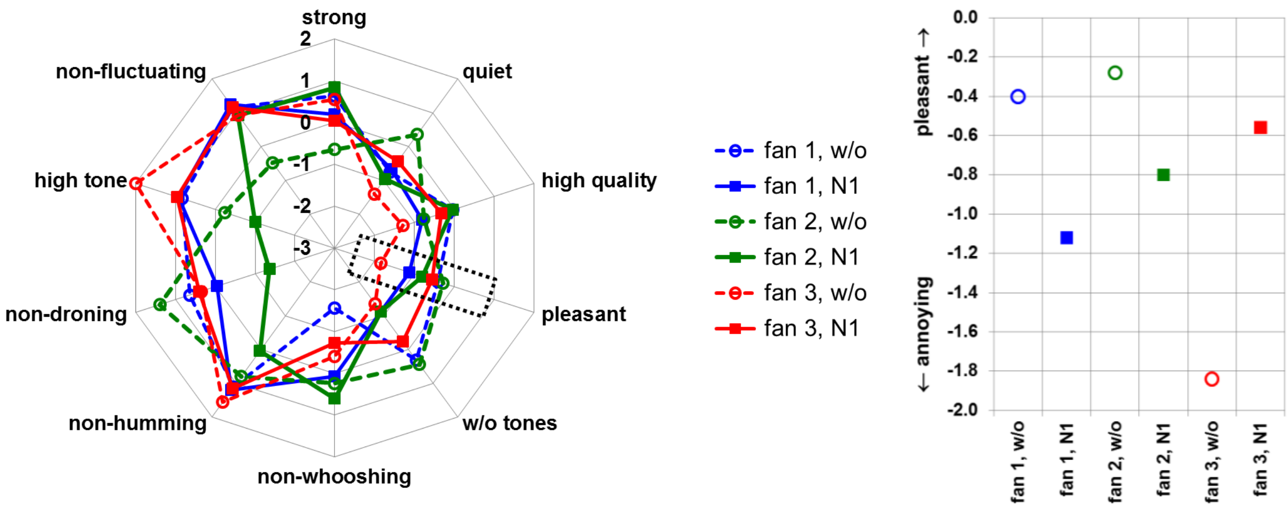

Figure 6: Spider chart of the 10 queried attributes (left) and attribute "pleasant" (right) for three investigated fans, inflow setups: undisturbed (w/o) and box (N1)

In the **second example** a radial fan with seven backward curved blades and an outer diameter of 250 mm is investigated. Complete performance curves including acoustical data are measured for seven different rotational speeds in the range between n = 1390 and 3480 rpm. Figure 7 (upper left) shows the A-weighted overall sound pressure level LpA taken from a microphone on the rotational axis (suction side, distance: 1 m) vs. the volume flow rate qv. For better comparability in the other diagrams of Figure 7 the dimensionless flow coefficient φ is used:

$$\varphi = \frac{4 \cdot qv}{\pi^2 \cdot D^3 \cdot n} \tag{1}$$

At maximum speed (n = 3480 rpm) the minimal overall sound level coincides with the best efficiency point at φ = 0.24. With decreasing speed this minimum moves towards a value of φ = 0.20 (upper middle). The curves for loudness N have similar shapes (upper right); both diagrams reveal a flattening of the graphs and a movement of the maximum value point from low to high values of the flow coefficient φ for reduced speed. The roughness curves show courses comparable to the loudness graphs (lower left), but with maxima at low values of φ for all rotational numbers. This means the fan noise becomes rougher with decreasing volume flow rate, probably due to flow separation. In this context the operating point at φ = 0.15 appears as outstanding for the roughness and for the fluctuation strength F (lower middle), here especially for high speeds. This is very likely linked to the beginning of separation – the so called "rotating stall". The sharpness S (lower right) according to DIN 45692 [8] shows a completely different course – the curves reveal a high correlation to the volume flow rate, i.e. with reduced φ the sharpness decreases and forms a sort of plateau for φ < 0.15.

Figure 8 depicts the A-weighted narrow band spectrum (left) and the specific fluctuation strength (right) for the mentioned "rotating stall" operating point at φ = 0.15 with n = 3180 rpm (cp. red line in Fig. 7, lower middle). It is well known from literature that these duty points are marked by a flow separation not in all blade channels which rotates slightly but distinctive slower than the fan (about 70-80% rpm) resulting in a more or less prominent additional peak in the noise spectrum. This peak can be seen in the original sound signal (black line) at f = 307 Hz. In this operating point it is significantly larger than the BPF tone at f = 375 Hz. The by means of band stop filter manipulated signal (green line) reveals that "rotating stall" is closely linked to fluctuating strength.

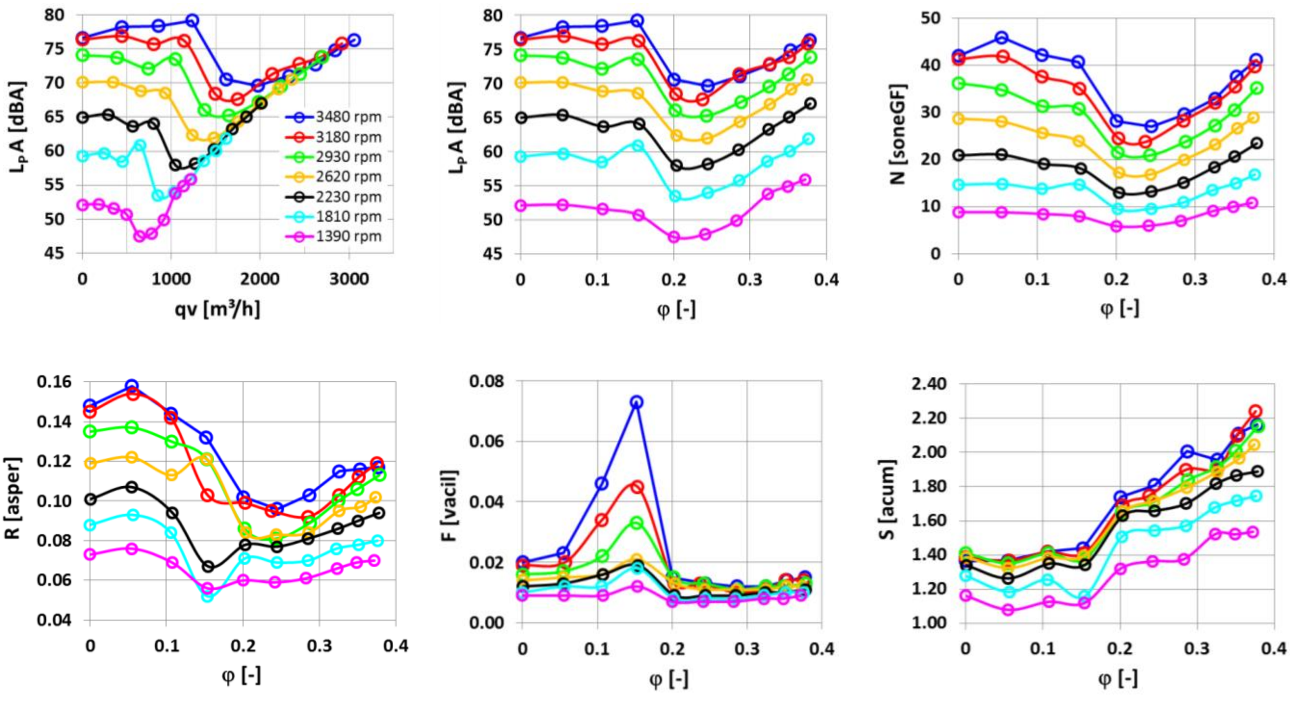

Figure 7: A-weighted overall sound pressure level LpA vs. volume flow rate qv (upper left) and flow coefficient φ (upper middle) for varying rotational speed (n = 1390 to 3480 rpm) of the investigated radial fan (D = 250 mm); corresponding psychoacoustic parameters: loudness N (upper right), roughness R (lower left), fluctuation strength F (lower middle) and sharpness S (acc. to DIN 45692 [8], lower right) vs. flow coefficient φ

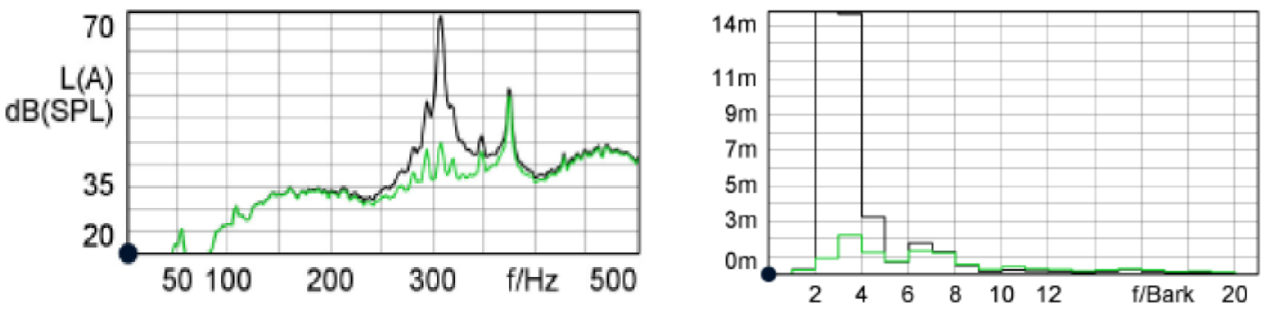

Figure 8: A-weighted narrow band spectrum (left) and specific fluctuation strength (right, in milli-vacil), φ = 0.15, n = 3180 rpm: original sound signal (black), manipulated signal by means of band stop filter, f = 307 Hz (green)

For a paired comparison with eight test persons (average age: 29.9 years) according to the method of double-sided A/B-Matrix three operation points at maximum speed are chosen. Figure 9 shows the A-weighted narrow band spectrum for the three selected operation points: $\varphi_1 = 0.29$ (left), best efficiency point $\varphi_2 = 0.24$ (middle) and “rotating stall” operating point $\varphi_3 = 0.15$ (right) at n = 3480 rpm (cp. blue lines in Fig. 7). Besides the original sound signal (blue) a manipulated signal (red; φ_x*) for every duty point is considered. In the manipulated sounds the low frequency tonal components (linked with BPF and rotating stall) are eliminated by means of band stop filters.

The result of the hearing comparison is depicted in Figure 10 in form of a boxplot. The sensation of annoyance (annoyance factor from bipolar scale) for the three selected operation points and the respective manipulated duty points (*) are plotted using statistics: the median values (red line), the interquartile range (blue rectangle) and the 1.5-times interquartile range (Whisker, black line). It can be clearly seen that the signals without tones (*) perceived as more pleasant. The stimulus at the optimum point (φ_2 = 0.24) is evaluated best. The manipulated signals are allocated to a distinct annoyance factor in two of three cases – merely at φ_3 = 0.15 slight uncertainty occurs.

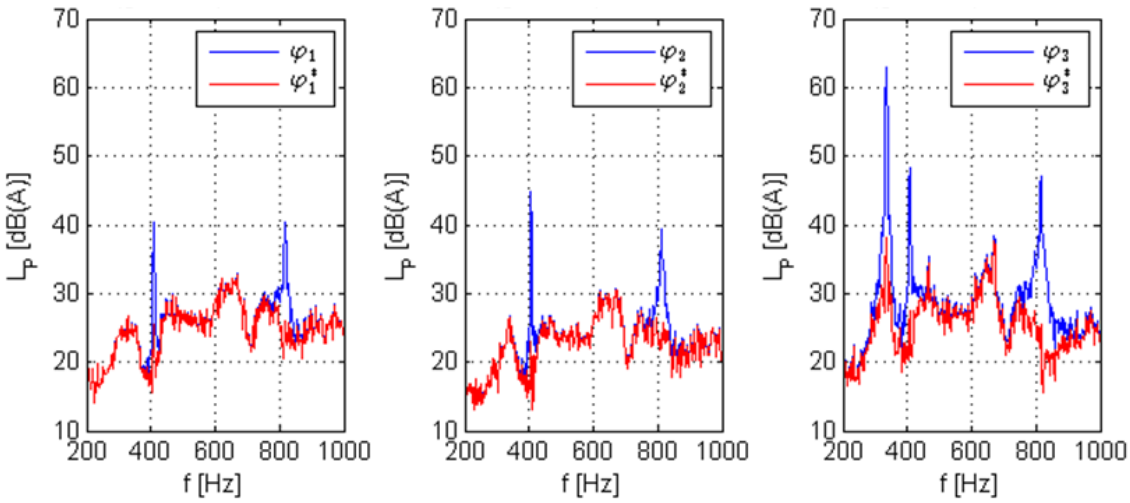

Figure 9: A-weighted narrow band spectrum for three selected duty points: φ_1 = 0.29 (left), φ_2 = 0.24 (middle) and φ_3 = 0.15 (right), n = 3480 rpm; original sound signal (blue), manipulated signal by means of band stop filters (red)

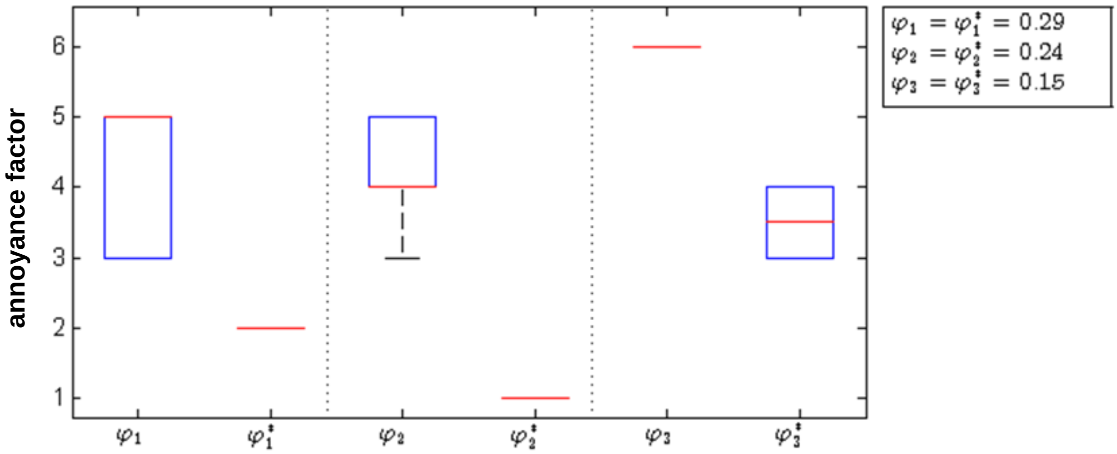

Figure 10: Sensation of annoyance (annoyance factor from bipolar scale) for three selected operation points and respective manipulated duty points (*); median values (red line), interquartile range (blue rectangle) and 1.5-times interquartile range (Whisker, black line)

PSYCHOCOUSTIC METRIC

The objective in psychoacoustics is to find a metric which allows the evaluation concerning annoyance or pleasantness of sounds of a special technical noise class without performing extensive hearing comparisons. By combining four psychoacoustic parameters Fastl and Zwicker [1] developed the psychoacoustic annoyance PA for synthetic and technical sounds:

$$PA = N_5\left(1+\sqrt{(0.25\cdot\log(N_5+10)\cdot(S-1.75))^2+\left(\frac{2.18}{N_5^{\,0.4}}\cdot(0.4\cdot F+0.6\cdot R)\right)^2}\right) \qquad (2)$$

with the percentile loudness N_5 in sone, the sharpness S in acum (with S ≥ 1.75), the fluctuation strength F in vacil and the roughness R in asper. Therein the loudness is dominating with an influence of about 80 %. The sharpness is included only for values above 1.75 acum – then, however, with a factor of approximately 30 %. Fluctuating strength and roughness play a minor role in this equation (each with about 2 %).

Because tonal components often play an important role related to fan noise, a modification of equation (2) is investigated [4] which includes the tonality – the so called modified psychoacoustic annoyance PA*:

$$PA^* = PA \cdot (1+T) \tag{3}$$

The results with the additional weighting of the tonality T shows the necessity for its consideration to account for the annoyance perception in this example (see above: second example, cp. Fig. 10). The dependency between annoyance factor and PA (see Fig. 11, lower left) which is dominated by the loudness N (dependency depicted in Fig. 11, upper left) is not so strong and clearly, resulting in a lower coefficient of determination $R^2 = 0.58$. On the contrary, when considering the tonality T (dependency shown in Fig. 11, upper right), the relationship between the annoyance factor coming from the *subjective* hearing comparison and the modified psychoacoustic annoyance PA* calculated from *objective* psychoacoustic parameters becomes significantly clearer with $R^2 = 0.87$ (see Fig. 11, lower right). Further investigations need to be done in the future to confirm and to extend these findings, especially concerning improved algorithms for taking into account the tonality.

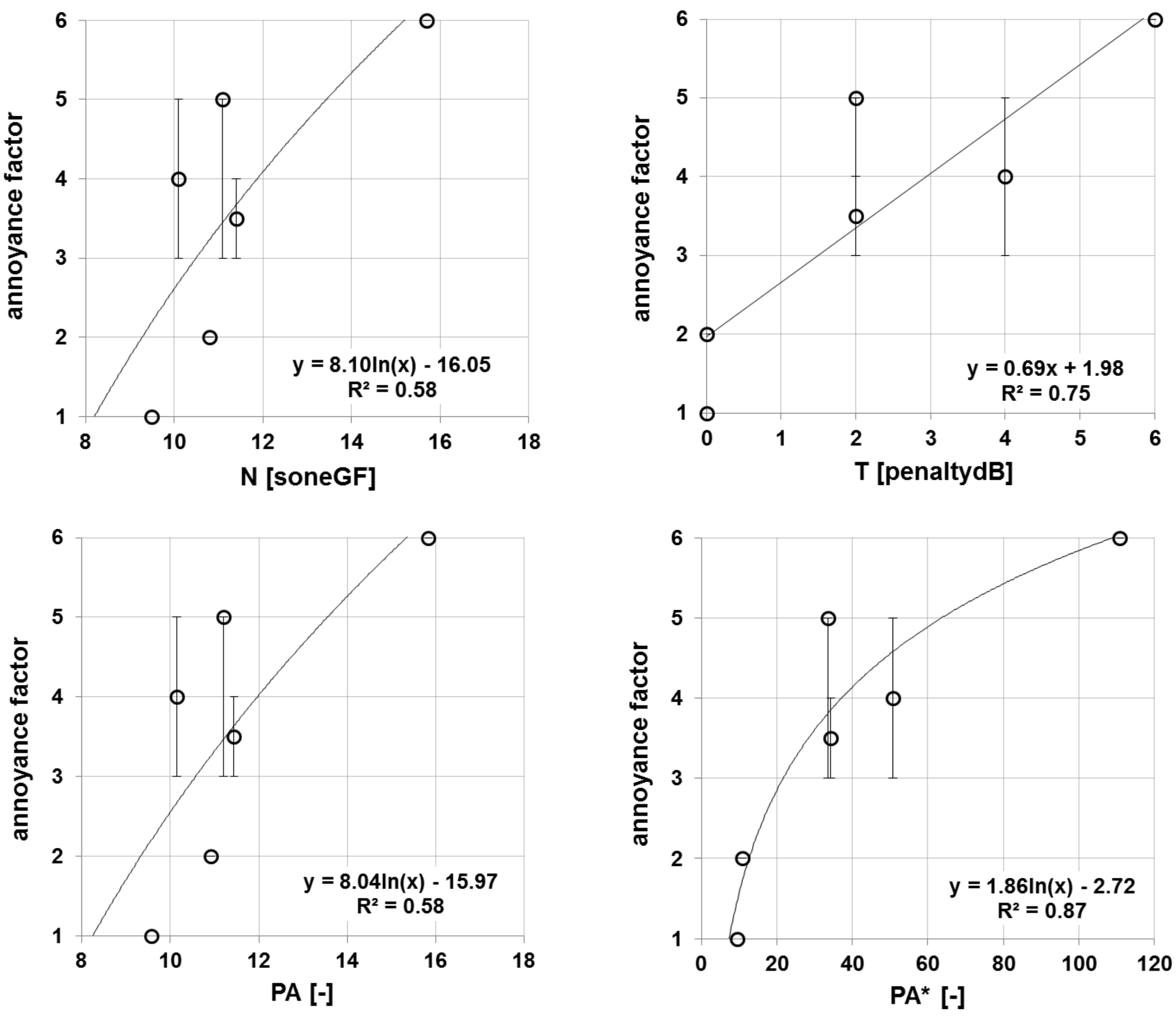

Figure 11: Loudness N (upper left), tonality T (upper right), psychoacoustic annoyance PA (lower left) and modified psychoacoustic annoyance PA (lower right) vs. annoyance factor (from paired comparison)*

CONCLUSIONS

In summary one can say that the physical parameter sound level and the related sound spectrum characterize fan noise incompletely. For a comprehensive consideration the use of psychoacoustic parameters is needed to take into account the subjective noise sensation.

The investigations of the general rules reveal that the averaged psychoacoustic spectra are similar to that of pink noise, so that the latter could be used directly or slightly modified for hearing comparisons with a reference sound suggested by Weber et al. [10]. It is shown that the psychoacoustic parameters of fan noise can be influenced clearly by varying the inflow conditions, the rotational speed and the operating point. It becomes obvious that the annoyance level is decisively associated with the objective psychoacoustic parameter loudness. Furthermore the tonality and especially in fan operating points near stall the fluctuating strength can play an important role. Hearing comparisons are conducted in form of two methods – the semantic differential and paired comparison. The resulting values of the annoyance factor are placed in a relationship to the psychoacoustic annoyance and a modified formulation. The latter is a first step in finding a psychoacoustic metric of fan noise which gives a single-number value of annoyance by combining relevant psychoacoustic parameters.

It should be a goal for the future to use these psychoacoustic parameters or combinations of them additionally to the physical ones. Therefore one should aim at establishing an international standard based on the existing norms referring to the psychoacoustics of fan noise.

BIBLIOGRAPHY

[1] H. Fastl, E. Zwicker – *Psychoacoustics – Facts and Models,* Third Edition, Springer Verlag, Berlin, **2006**

[2] K. Genuit – *Sound Engineering im Automobilbereich*, Springer Verlag, Berlin, **2010**

[3] M. Amann – *Psychoakustische Beurteilung von Ventilatorgeräuschen*, Bachelor Thesis, ebm-papst Mulfingen GmbH & Co. KG, **2013**

[4] C. Feldmann – *Psychoakustische Geräuschbeurteilung von Ventilatoren*, Master Thesis, ebm-papst Mulfingen GmbH & Co. KG, **2014**

[5] F. Meyer – *Untersuchung des Einflusses typischer Zu- und Abströmstörungen bei freilaufenden rückwärtsgekrümmten Radialventilatoren,* Bachelor Thesis, ebm-papst Mulfingen GmbH & Co. KG, **2012**

[6] DIN 45631/A1 – *Berechnung des Lautstärkepegels und der Lautheit aus dem Geräuschspektrum – Verfahren nach E. Zwicker – Änderung 1: Berechnung der Lautheit zeitvarianter Geräusche*, Beuth Verlag GmbH, Berlin, **2008**

[7] DIN 45681 – *Bestimmung der Tonhaltigkeit von Geräuschen und Ermittlung eines Tonzuschlages für die Beurteilung von Geräuschemissionen*, Beuth Verlag GmbH, Berlin, **2005**

[8] DIN 45692 – *Messtechnische Simulation der Hörempfindung Schärfe*, Beuth Verlag GmbH, Berlin, **2009**

[9] S. van de Par, D. Hülsmeier, L. Schell-Majoor, J. Rennies – *Wahrnehmung der Klangqualität von Produktgeräuschen*, Fortschritte der Akustik - DAGA 2014, Oldenburg, **2014**

[10] R. Weber, A. Ötjen, S. Töpken – *Vergleichende psychoakustische Charakterisierung von Klangunterschieden mit Referenzschallen und der Bestimmung des Punktes der subjektiven Gleichheit*, Fortschritte der Akustik - DAGA 2012, Darmstadt, **2012**