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DIFFERENCES AND CHALLENGES OF RESONATORS/ABSORBERS FOR VARIOUS APPLICATIONS

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ABSTRACT

As the world leader in resonators for charge air ducts in combustion engines, Umfotec develops resonators made from a range of different materials. Low pressure drops, prevention of sooting due to oil content and excellent noise attenuation make Helmholtz resonators a well-established approach in this application field. The prevention of sooting is achieved by adequate dimensioning of the resonator openings - hence one of the biggest challenges is to maximise transmission loss values over a broad frequency range without producing unwanted flow noise, especially tonal noise.

Sooting plays a minor role in ventilation or HVAC applications, allowing the use of acoustic absorber materials as well as resonators. Broadband noise reduction at higher frequencies is achieved by absorbers, whilst for narrowband noise reduction, especially at lower frequencies, resonators are employed. Combining both enables sound tuning and shaping.

This paper reviews the specific properties and design challenges of the different applications. While transmission loss simulation of resonators in charge air ducts is highly precise and practicable, flow-induced noise can only be predicted for academic cases with simple geometries using high computer processing capacities. For HVAC applications, the precise calculation of attenuation levels by absorber/resonator combinations remains highly challenging and inaccurate.

Keywords: Flow Noise, Resonators, HVAC-Applications
I-INCE Classification of Subject Number: 34; 35; 76

1. INTRODUCTION

Flow noise is now a topic that is continuing to grow in importance. One reason for this is that car interiors are becoming quieter, and electric cars in particular no longer mask ventilation noise, like the noise generated by combustion engines.

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For turbo-charged engines, resonators based on the Helmholtz–Principle (1) are practically standard. The advantage of these resonators is their ability to greatly reduce flow noise. This leads to very low pressure drops with an excellent and continuous live time functionality, even when oil content of the charged air is high. However, charge air resonators do tend to generate flow-induced noise, especially when they are very efficient in terms of noise reduction (2). Because of the position of charge air resonators in the engine compartment, an occasional lower level of flow-induced noise is acceptable when it is inaudible within the vehicle interior.

For HVAC (Heating, Ventilation, Air Conditioning), flow-induced noise must be handled very carefully as it occurs directly in the passenger cabin. Further sound tuning and noise perception play an important role (10). In comparison with the charge air application, where the air flow also contains oil, HVAC air is "clean". Therefore sooting plays a negligible role. Other approaches to reduce flow noise become possible, for example absorbers constructed using porous materials such as foam and fleece, with or without a perforated surface. However, absorption only is insufficient, as higher attenuation levels are usually reached just for frequencies above 2000 Hz. Furthermore, sound tuning becomes more difficult, as attenuation is more frequency-related. By combining the different physical principles of absorbing and resonance related damping, two positive effects can be achieved (3), (11). Highly specified good noise attenuation at lower frequencies from 700 – 3000 Hz and a broadband sound absorption for higher frequencies are both possible.

1.1. Differences between resonators on the charge air side or in the cooling circuit and resabtors

Resonators for the charge air side or cooling circuit are usually built concentrically around the duct (2), (13). This is possible because, in most cases, the ducts are circular and the existing package for the resonator is positioned around this duct. With this kind of concentric charge air resonator design, good damping penetration of the entire duct diameter is given. Insertion loss values exceeding 35 dB (depending on diameter and frequency range) or higher can be achieved (see Figure 1, image on left). In order to achieve these insertion loss values at a broader frequency range, multi-chamber resonators are used.

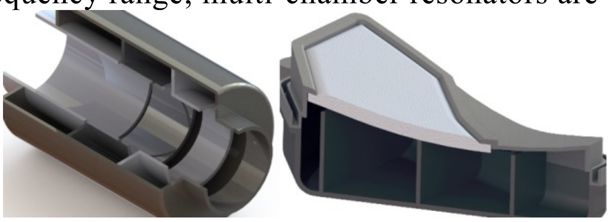

Figure 1 – Example of a three-chamber charge air resonator and a two-chamber (tightless) channel-vent resonator

For HVAC applications (not including the cooling circuit), the ducts are generally more rectangular and the package for the absorbers is more concentrated on one or two sides rather than all four sides. Furthermore, the diameter of these ducts is larger due to the desired lower flow velocity. A larger diameter and only one-sided or two-sided attenuation applications are the reasons for a reduced penetration of the entire diameter and the subsequent insertion loss values (see Figure 1, image on right).

As already indicated, sooting plays a minor role for HVAC and therefore applications for the reduction of flow noise can be achieved using different materials and physical principles. Pure absorber materials like fleece and foams can be combined with Helmholtz resonators. The marriage of these two principles will be referred to as resabtors (**Res**ona**tors** and **Ab**sorbers). Figure 2 shows an example of a resabtor. As you can see, is in this case the surface of the absorbing material perforated.

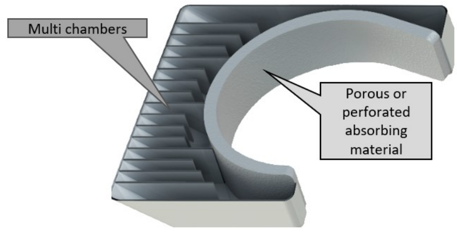

Figure 2 –Resabtor – perforated material with good absorbing properties and the multi-chambers of the Helmholtz resonator (5), (18).

This perforation allows or improves communication with the Helmholtz chambers and the attenuation level (12). The special design means these resabtors are compact and can be installed in small installation spaces.

1.2. Flow noise attenuation – downstream effects

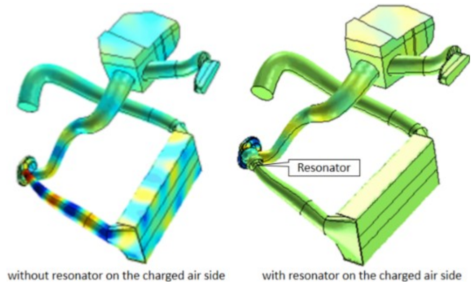

Figure 3 – Simulation (airflow routing in a combustion engine) of the sound propagation of a 2000 Hz sine wave

Figure 3 shows simulated airflow routing in a turbo-charged combustion engine. In principle, this simulation can be employed for other applications such as HVAC systems or cooling circuits. The image on the left shows the flow noise propagation *without* a resonator, and the image on the right, *with* a resonator in the airflow routing. Since the noise propagation in the ducts is nearly undampened, the noise generated by charging the air is propagating towards the duct (7). This noise can radiate through the walls of the ducts or the charge air cooler into the engine compartment and then into the interior of the passenger cabin, where it is perceived as a disturbance noise. By installing a resonator as closely as possible to the noise source – in this case the compressor – the noise can be significantly reduced, and the downstream flow becomes free of this.

A similar behaviour can also be observed for cooling circuits and HVAC systems.

2. TYPICAL SPECIFICATIONS FOR RESONATORS (CHARGE AIR SIDE OR COOLING CIRCUIT) VERSUS HVAC RESABTORS

Table 1 contains typical specifications for resonators and resabtors. The two resonator applications indicated are exposed to overpressure that influences the resonator design. According to the applications, the main frequency range that requires noise attenuation varies from 200 to 10,000 Hz. Resabtors in particular are worth mentioning here. Two frequency ranges must be distinguished for resabtors. For the lower frequency range - normally between 700 and 3000 Hz - the Helmholtz principle (damping) is employed and absorption is used for the higher frequency range approximately from 2000 to 10,000 Hz. Between 2000 and 3000 Hz both physical principles operate together (damping and absorbing).

Table 1: Typical requirements for resonators or resabtors

		Resonators for turbo charges engines	Resonators for the cooling circuit	Resabtors for HVAC	
Positioning of the chambers		concentrically around the duct		One or two sided	
max. Temperature		up to 400	< 100	< 70	°Celcius
Charged air Pressure		4	12	1	bar
Air flow velocity		< 0.3	< 0.3		mach
Working frequency range					
	main	1 000 - 4 000	100 - 700	700 -3.000	Hz
	high frequency	5 000 - 10 000		3.000 - 10.000	Hz
Wanted noise reduction (Tranmission or insertion loss)					
	main	20-35	~20	~10	dB
	high frequency	15-20		~20	dB
Pressure drop		< 50 mbar	< 10 mbar	< 50mbar	
Remark		sooting to be prevented	sooting to be prevented		

3. RESONATORS – RESABTORS FOR COOLING CIRCUITS/ THE CHARGE AIR SIDE/ HVAC AND VENTILIATION

3.1. Function of resonators

Helmholtz resonators on the charge air side have some very specific properties compared with other resonator applications. Firstly, the overpressure, which means that with multi-chamber resonators, special measures must be taken to prevent "acoustic not tight" chambers. Secondly, the frequency range above 1000 Hz, which means that precise numerical simulations are necessary to work out the best solution.

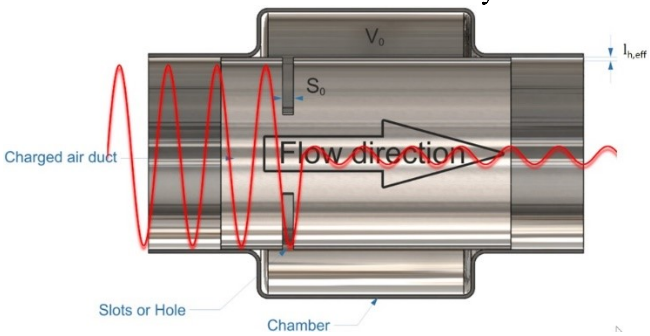

Figure 3 - Mechanical model for a Helmholtz resonator

Figure 3 contains a mechanical model of a single-chamber resonator. The resonator chamber is positioned circumferentially around the duct. This chamber is connected to the duct via slots or holes. According to equation (1), the resonance of this mechanical model can be calculated as follows:

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{S_0}{V_0 l_{h,eff}}} \quad \text{with } l_{h,eff} \approx l_h + 2 \cdot \Delta l_h \tag{1}$$

V_0 is the chamber volume, l_h corresponds to the depth of a slot and S_0 corresponds to the width of a slot.

The Helmholtz frequency is very easy to understand when a comparison with an absorber is carried out. The absorber spring corresponds to the air spring in the chamber. Thus the chamber volume (V_0) is decisive for the spring. The mass refers to the air mass which can be calculated from the volume in the slots ($l_{h,\,eff}$, S_0). Equation 1 shows the calculation of the resonance frequency of a Helmholtz resonator.

However, the resonance frequency is just one value and the decisive factor is the attenuation of noise inside the duct. Usually the transmission loss value (TL) is used as a measure for the effect of a Helmholtz resonator. The TL indicates noise attenuation when passing through the resonator by comparing sound power levels at inlet and outlet. For example, a TL value of 35 dB at a specified frequency means that the noise at this frequency can be reduced by 35 dB. However, as you can see in Figure 4, the single-chamber Helmholtz resonator operates across a very narrow band only.

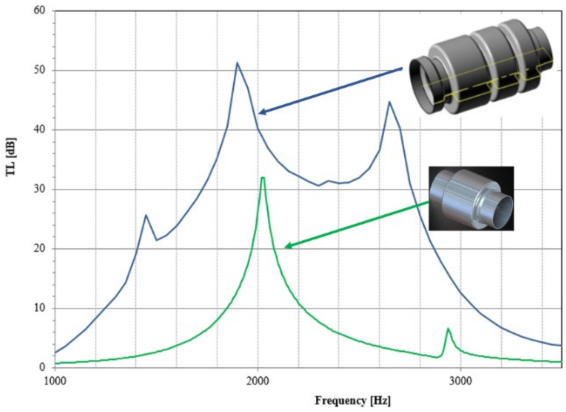

Figure 4 - Transmission loss curve of a single and a multi-chamber resonator

Figure 4 also shows a multi-chamber resonator with its corresponding transmission loss curve (TL). Broadband noise attenuation can be achieved with this type of multi-chamber resonator. Each chamber operates within a special frequency range. In most frequency ranges, many chambers interact with the noise, consequently the transmission loss value is increased.

3.2. Function of resabtors and initial approaches to evaluate and simulate resabtors

Figure 5 shows a resabtor. Instead of holes or slots, a perforated or porous material with absorbing properties is used. This material also acts as the separation between the Helmholtz chamber and the duct or rectangular channel.

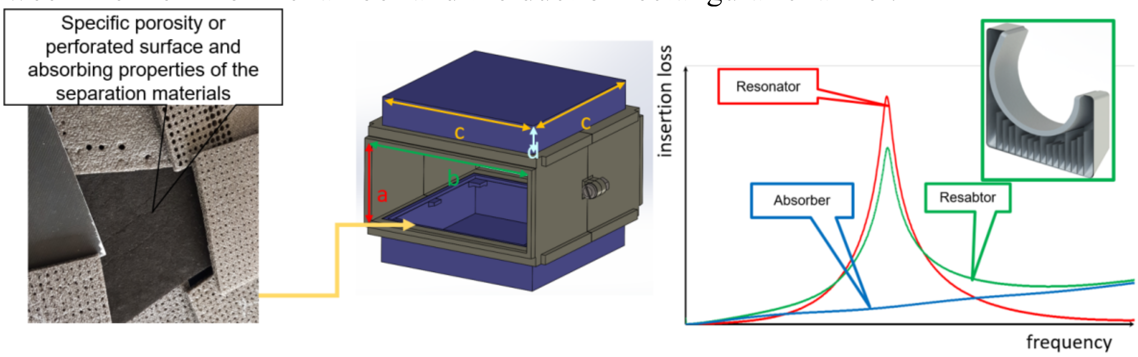

Figure 5 – Resabtor – evaluation principle for Resabtors and test bench application

This facilitates the marriage of the two principles – broadband noise reduction by an absorber and the narrowband but very powerful noise reduction via the Helmholtz principle (see Figure 6).

It is clear to see how difficult it is to simulate these systems. Precise parameters for the mechanical model are essential and a rapid and reproducible calculation is a must in order to determine the optimum for any given application. Whereas the effect of a Helmholtz resonator can precisely be calculated by acoustic FEM or in first approximation even with 1D simulation models, absorption remains a challenge in simulation work. Due to the small-scale geometries causing the damping effect of the absorber material direct noise calculations become impossible. A way to circumvent this problem is to model complex sound transitions with an acoustic transfer relation admittance matrix (15).

$$\begin{pmatrix} v_{n1} \\ v_{n2} \end{pmatrix} = \begin{pmatrix} \alpha_1 & \alpha_2 \\ \alpha_4 & \alpha_5 \end{pmatrix} \cdot \begin{pmatrix} p_1 \\ p_2 \end{pmatrix} + \begin{pmatrix} \alpha_3 \\ \alpha_6 \end{pmatrix}$$ Acoustic Transfer Relation Admittance Matrix (2)

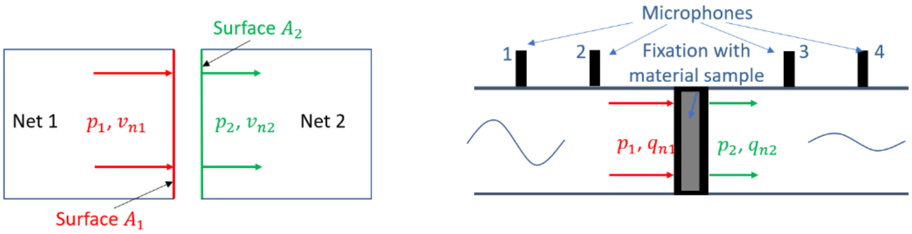

Figure 6 – Acoustic Transfer Relation Admittance Matrix – for Resabtor simulation

Figure 6 shows the equation for the acoustic transfer relation admittance matrix. This equation shows the correlation between pressure and sound particle velocity from the incoming surface A_1 with (p_1, v_{n1}) and the outgoing surface A_2 with (p_2,v_{n2}).

The admittance matrix can be identified on a test bench (14), as is shown by the diagram on the right side of Figure 6. With this and some further adjustments, the simulation and behavioural prediction of resabtors is possible. The simulation accuracy is of course highly dependent on the quality of the parameter identification.

3.3. Resonator design and construction

Figure 7 shows various serial resonators made of stainless steel, aluminium or temperature-resistant plastic materials. In most cases, the resonator is placed concentrically around the duct. The image on the right of Figure 7 shows a resonator with an asymmetrical placement of the chambers around the duct. This is sometimes necessary to prevent the collection of air-borne oil (charge air of combustion engines or in cooling circuits) (13).

Figure 7 – Resonators (4) on charge air side (left) or for cooling circuit (13) (right)

In the case of multi-chamber resonators, which are a standard application for resonators on the charge air side, the chambers must be tight to each other, or the design must be based on a patented principle by Umfotec (8). The tendency of chambers to cease to be "acoustic"-tight rises in line with overpressure, or at higher temperatures, due to material expansion (2), (3). If this tightness/seal cannot be guaranteed, the attenuation level is reduced significantly, even with very small gaps/leaks. The reason for this is that the air pressure vibration in the chamber, which can only occur with acoustic-tight chambers, is reduced even by small gaps.

In Figure 8, the influence of acoustic not tight chambers is evaluated for a two-chamber resonator. Even the smallest gaps result in a significant loss of noise attenuation.

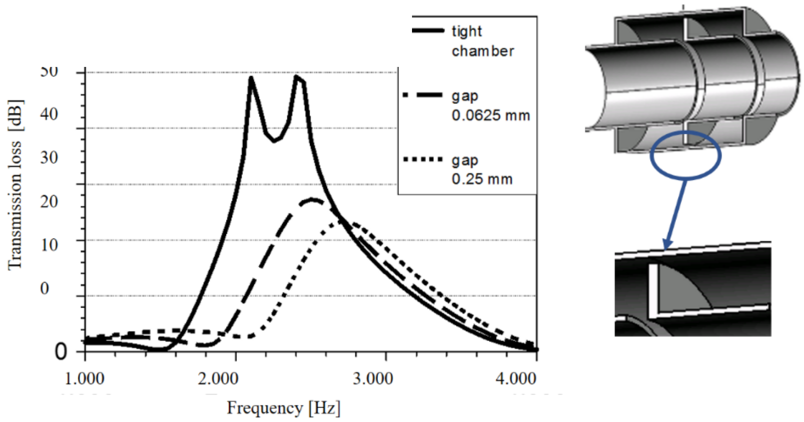

Figure 8 - Influence of acoustic not tight chamber separation on the transmission loss

3.4. Application possibilities and the construction of RESABTORS

As explained in Chapter 1.1, resabtors can be used in HVAC applications. For these applications, due to design features, rectangular ducts are more common. Furthermore, the location of resabtors on all four sides of the duct is usually not possible due to the available installation space.

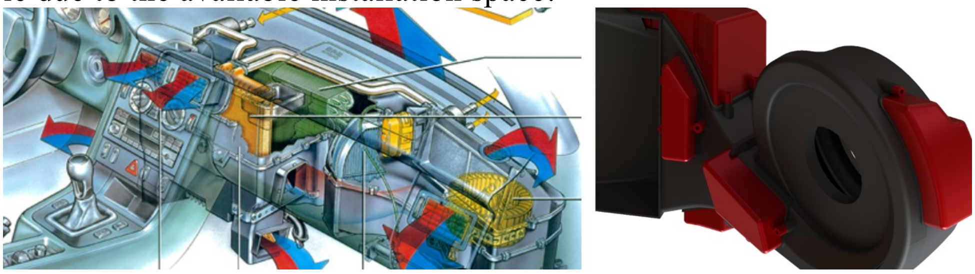

Figure 9 Multiple application possibilities for resabtors at the HVAC module or at the vents (Google)

The image on the left of Figure 9 shows the venting within a car interior and on the right, an HVAC module with resabtors (red). Wherever unwanted noise occurs, resabtors can be used. If there is no further noise source downstream, the noise in the air flow is significantly reduced. Due to fewer temperature and pressure constraints, the use of plastic materials is preferred. Therefore resabtors should be integrated within the housing of the HVAC module or as an extension of the ventilation channels.

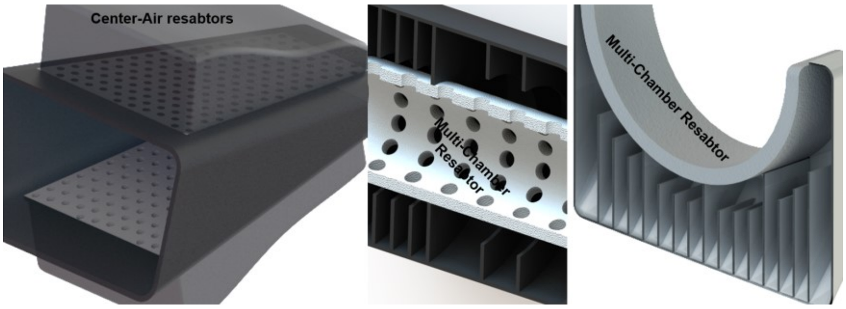

Figure 10 Resabtor applications. Image in centre shows the multi-chamber design.

The use of plastic materials is typical of these applications due to fewer temperature constraints. A resabtor can often be integrated within the housing of an HVAC module or in a section of the ventilation channels. If integration is possible, the cost of resabtors is mainly driven by the separation material between the chambers and the air-guiding. The number of chambers and the choice of separation material will depend on the desired noise reduction level and its specific respective desired frequency range.

4. FLOW-INDUCED NOISE – CHALLENGES AND SOLUTIONS

4.1. General discussion of flow-induced noise for different applications

Flow-induced noise is always undesired as it constitutes an additional noise source. In particular, the equipment used to reduce noise in the air flow should have a zero or very low tendency to generate flow-induced noise. Dr. Ralf Buck has given several presentations on the various mechanisms of flow-induced noise and their occurrence in resonators for the charge air duct e.g. (2), (16). In these talks, the reason for the generation of flow-induced noise was narrowed down to three different highly descriptive mechanisms (also see Figure 11).

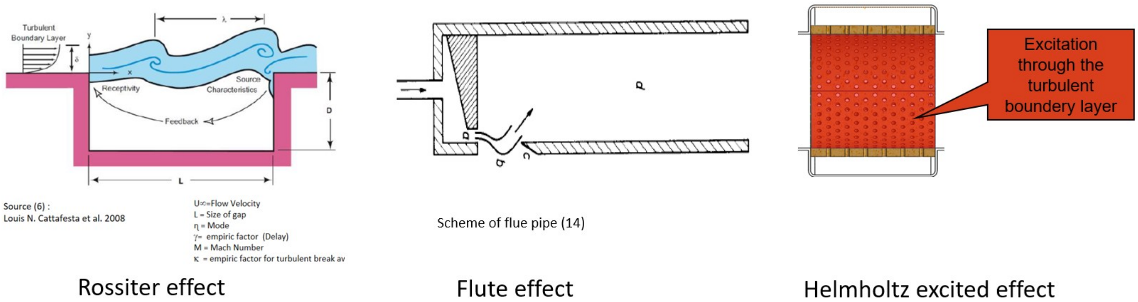

Figure 11 Three mechanisms of flow-induced noise.

The first theory was the Rossiter effect (6) or cavity noise effect based on an existing turbulent layer close to the walls of a tube. When this turbulent flow hits a sudden change, such as a hole or a slot, the flow begins to generate coherent vortices propagating from the leading to the rear edge of the slot, where they are repelled and start to build up a self-exciting feedback mechanism. As a second mechanism, the author stated the flute effect (9), which is an oscillation of the air flow into and out of the chamber. As a third theory, the Helmholtz excited whistling effect was mentioned, which also considers a turbulent boundary layer. This turbulent layer can be regarded as a broadband noise source, therefore the resonator begins to interact and produces unwanted noise close to the Helmholtz resonance.

Further discussion must be held on the sensitivity of flow-induced noise for different applications. For resonators used for cooling circuits or the charge air side, the sensitivity of flow-induced noise is not as significant as it is for HVAC or ventilation applications, as the resonators are usually located in the engine compartment, where an additional firewall is in place between the resonator and the vehicle interior. It can be shown (see next chapter) that the complete prevention of flow-induced noise is difficult or expensive, as the slots must be dimensioned to a sufficiently large size to prevent sooting. Depending on the resonator design, flow-induced noise has an extremely tonal character and often arises close to the Helmholtz resonance. In most cases, the sound pressure level (SPL) of noise interference in the ducts is major and the resonator reduces the SPL significantly, even when flow-induced noise is generated.

HVAC and ventilation applications are even more sensitive to flow-induced noise. The reason for this is their immediate installation position in the passenger compartment. The sound pressure level of unwanted noise for such applications is also lower. Flow-induced noise is easy to detect or hear. Due to Resabtor build-up (perforated or air permeable), flow-induced noise can be reduced to Helmholtz-excited whistling. As discussed in (23) the turbulent boundary layer that represents a broadband noise source excites the resabtor at its resonance frequency. Depending on the flow speed and the noise in the flow stream, this effect could be a combination of noise reduction and noise generation. Solutions in this case must affect the turbulent boundary layer. Due to the high flow noise sensitivity in HVAC applications broadband flow noise sources must be also considered as potentially disturbing. Especially the added wall roughness by the resabtor material can gain unwanted pressure drop and amplifies the turbulence of the boundary layer. Hence the surface of the resabtor material has to be treated with care.

4.2. Test benches for the generation and evaluation of flow-induced noise

The image on the left of Figure 12 shows a test bench with which flow induced noise can be evaluated. With this test bench, a straight laminar flow up to a mass flow rate of approximately 600 kg/h (depending on the duct diameter) can be achieved. Further noise can enter the flow via the loudspeakers.

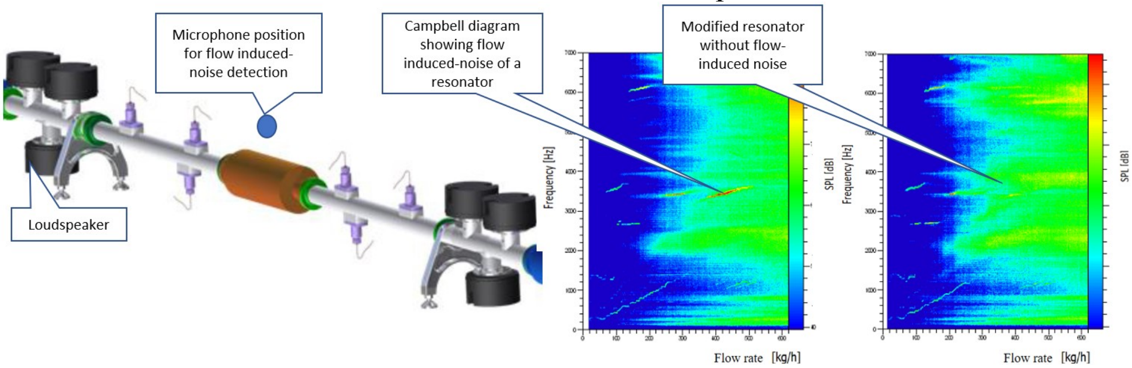

Figure 12 – Flow test bench and Campbell diagrams with a resonator, with and without flow-induced noise

To measure the flow-induced noise of resonators, the sound pressure level (SPL) measurement outside the outer shell of the resonator at a distance of 100 mm is the standard approach. The two Campbell diagrams on the right above show the SPL. The Campbell diagram on the left shows the flow-induced noise at a mass flow rate of 400-600 kg/h at 3500 Hz. The same resonator with the same TL behaviour as before, when slightly modified, prevents flow-induced noise (right side, Figure 9).

As already mentioned for HVAC applications, the ducts are rather rectangular and the noise propagating towards the ducts is lower than it is for resonator applications. It is still possible to use the test bench shown in Figure 12, but adjustments need to be carried out – see image on left in Figure 1§.

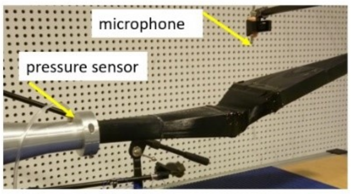

Figure 13 – Adjustments for the evaluation of flow-induced noise of resabtors for HVAC applications

The image on the right above shows a further possibility for superposing flow and noise. In this case, the flow-induced noise evaluation is performed in the proximity of the final application. This is sometimes necessary, as even extremely low levels of flow-induced noise may be audible/perceptible in the interior, and the evaluation of the noise outside the resabtors may not be sensitive enough.

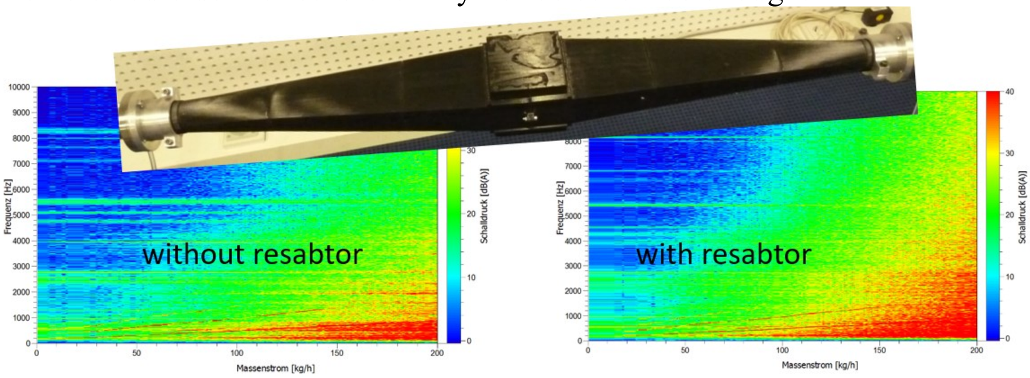

Figure 14– Flow-induced noise – flow only – of a resabtor

Figure 14 depicts a flow-induced noise measurement. Resabtors generate broadband noise at a very low level only. Due to the novelty of this type of part in this kind of application, Umfotec has recently begun to establish a new test bench and evaluation procedure. Future presentations will provide further detailed information on these evaluations.

5. SIMULATION RESULTS

As mentioned before simulating resabtors and predicting the potential noise attenuation level (insertion loss) is difficult. Precise material identification is necessary to combine the absorption and damping of the Helmholtz chamber. After several trials and failures, Umfotec is finally able to predict the behaviour of these parts, even when simulation accuracy is not quite as precise as it is for resonators. One of the reasons could be the design and construction of resabtors due to their use of perforated or porous materials, which typically have a certain inhomogenity in their geometric structure.

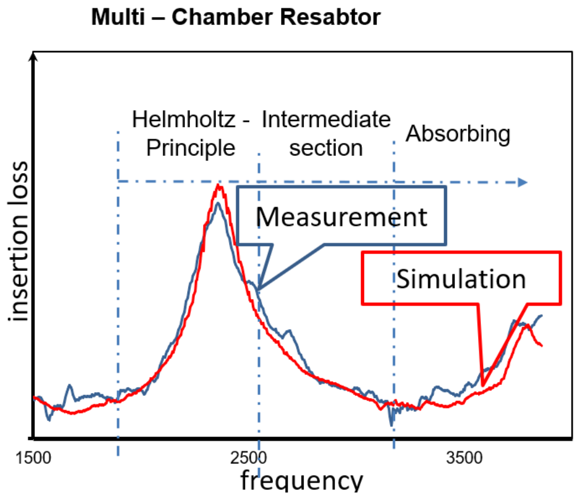

Figure 15 – Multi-chamber resabtors – A comparison of measurement and simulation

Figure 15 depicts the simulation and measurement of a multi-chamber resabtor. The objective of this application was to improve the noise reduction level, especially in the frequency range between 1200 and 2000 Hz, without losing the absorbing effect above 2500 Hz. Similarly good results were achieved after integrating this resabtor into the HVAC module.

The simulation of flow-induced noise is more complex. This simulation must be carried out in the time domain and the interaction between the flow noise and the Helmholtz effect is hard to predict.

Figure 16 shows near-field sound pressure measurements (100 mm distance from the resonator surface) and structure-borne measurements (on the outer shell of the resonator) on the flow test bench (2), (20). The two top Campbell diagrams show the original resonator. The stepped, flow-induced noise in both the near field and on the surface of the resonator are obvious. This stepped behaviour is typical of highly effective resonators. In the two Campbell diagrams below, the slots were covered slightly in the chamber by mesh wire. It is good to see that the tonal component flow-induced noise no longer exists and the problem of this tonal flow-induced noise is completely solved, whilst transmission loss behaviour remains unchanged. This solution was analysed and designed using simulation in the time domain with a single-cavity resonator.

As resabtors are constructed using a perforated or porous material to separate between the chamber and the flow, they behave as per the solution presented for resonators (Figure 16). As mentioned earlier, the sensitivity for flow-induced noise of resabtor applications is high. Flow-induced noise caused, for example, by surface roughness, must be considered and calls for evaluation. Simulation and prediction of flow induced noise by resabtors becomes even more challenging and has not been carried out successfully so far.

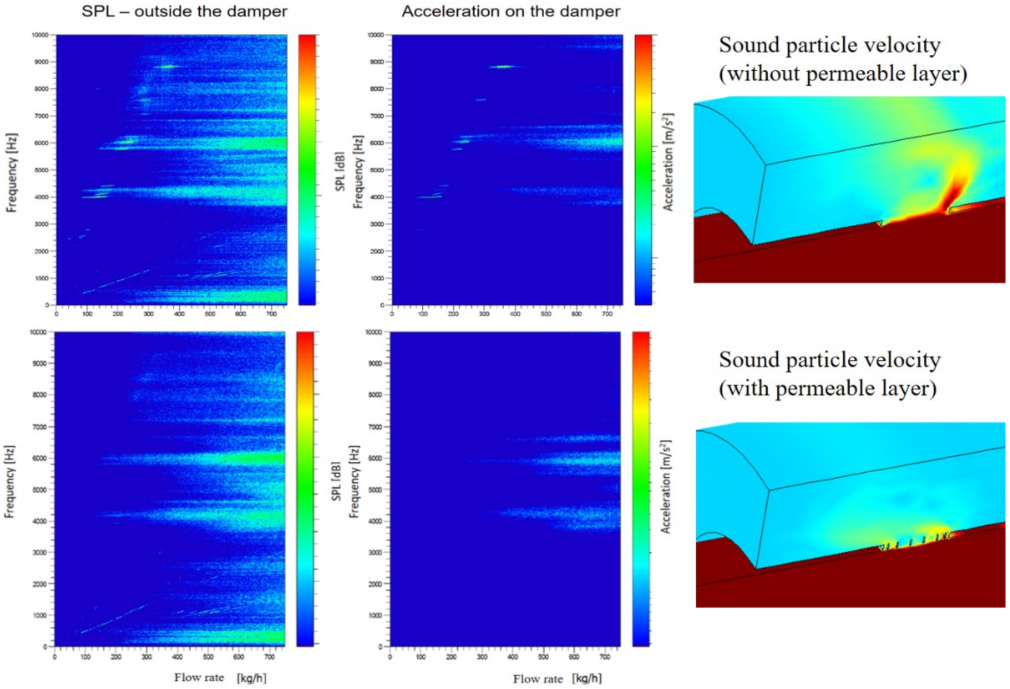

Figure 16 - Preventing flow-induced noise without influence on TL behaviour

6. SUMMARY

This paper introduces resabtors. A resabtor is a combination of a **Re**sonator based on the Helmholtz principle and an **Ab**sorber. At lower frequencies, e.g. 700 to 3000 Hz, the Helmholtz principle enables good insertion loss values to be achieved. Special tuning according to the noise problems of each application is possible. Good insertion loss values can also be achieved using special absorbing materials above 2000 Hz. Thus resabtors do offer good broadband noise attenuation capabilities above 700 Hz. Due to the design and construction of resabtors, HVAC applications are ideal, as sooting plays a minor role and a broadband noise reduction is desired.

The development of resonators for the charged air duct based on the Helmholtz principle started around 20 years ago. The simulation of this kind of device in terms of the prediction of noise attenuation is well-established and precise. For resabtors, predicting noise attenuation using simulation is challenging. Two different physical principles must be combined and identification of the exact material properties is necessary. The approaches employed to identify the material properties have been demonstrated. Using the “Acoustic Transfer Relation Admittance Matrix”, it is possible to predict noise attenuation.

Flow-induced noise caused by resonators and resabtors were discussed. The authors showed that flow-induced noise for resonators is based on three theories. Each needs to be prevented. Due to the design of resabtors, only the Helmholtz excited flow-induced noise theory is relevant. Resabtor applications in particular are highly sensitive to flow-induced noise due to the lower noise level in relation to flow rate. Given the novelty of these parts, the rating of flow-induced noise must be adjusted and developed and this is something which is currently ongoing.

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