NOISE CONTROL FOR COOLING FANS ON HEAVY VEHICLES

Sabry ALLAM[1], Mats ÅBOM[1] and Lars WESTER[2]

[1]*KTH - Competence Centre for Gas Exchange (CCGEx),*
The Marcus Wallenberg Laboratory (MWL), SE-100 44 Stockholm, Sweden.

[2]*Sontech AB, SE- 196 37 Kungsängen, Sweden*

SUMMARY

In this paper two different objects for fan passive noise control have been examined both experimentally and theoretically; the heat exchanger and inlet/outlet parallel baffle silencers based on micro-perforated plates. Throughout this paper two measurement methods were used. The ISO 15186-1:2000 to test the acoustic transmission for a diffuse field and also a sample cut from each heat exchanger type and put in a duct and measured using the two microphone method (TMM). Based on an anisotropic equivalent fluid model a theoretical model for the heat exchanger acoustic transmission is developed and validated. Empirical formulas for the new type of baffle silencers based on micro-perforated plates, which can add damping up 10-20 dB in the frequency range of interest, are also presented.

INTRODUCTION

An important subsystem in surface transportation vehicles; rail bound, automotive and heavy duty is the engine cooling module. For this reason the European Commission is funding the Efficient Cooling Systems for Quieter Surface Transport (ECOQUEST) project and the overall technical objectives are innovative contributions towards reduced noise emission and energy efficient cooling modules. The three main objectives in the project concern; implement an integrated simulation platform to gather modeling tools for noise, near-field acoustic scattering and propagation towards listeners in the far field, develop thermally and acoustically optimal cooling modules, and development of innovative fan designs and new passive noise control solutions.

Within this project two different main objects for fan passive noise control have been examined both experimentally and theoretically; the heat exchanger and inlet parallel baffle silencers.

For the first object seven heat exchangers were experimentally assessed, using a modified version of ISO 15186-1:2000 [1], to test the acoustic transmission for a diffuse field. In addition a sample from each heat exchanger type was also cut out and tested by measuring the acoustic two-port [2] in a duct, i.e., the transmission and reflection at normal incidence were determined. Theoretically, the basic configuration is assumed to be a matrix of parallel and rectangular narrow channels. The developed model is based on a so called equivalent fluid for an anisotropic medium. It is mainly dependent on the heat exchanger geometry combined with the Kirchhoff model for thermo-viscous wave propagation in narrow tubes. This model is a continuation of earlier work by Yan and Åbom presented in Ref. [3].

In order to reduce the transmission through heat-exchangers they can be fitted with parallel baffle silencers. In ECOQUEST a new type of such silencers using Micro Perforated Plates (MPP:s) have been designed and tested. Results from this work are presented showing that such MPP baffle silencers can provide up 10-20 dB added damping in the frequency range of interest.

THEORETICAL CHARACTERIZATION OF HEAT EXCHANGERS

Common appliances containing a heat exchanger include air conditioners, refrigerators, and space heaters. Perhaps the most commonly known heat exchanger is a car radiator, which cools the hot radiator fluid by taking advantage of airflow over the surface of the radiator.

There are also four different designs of heat exchangers: shell and tube, plate, regenerative, and intermediate fluid or solid. The most typical type of heat exchanger is the shell and tube design. This heat exchanger has multiple finned tubes. One of the fluids runs through the tubes while the other fluid runs over them, causing it to be heated or cooled. In the plate heat exchanger, the fluid flows through baffles. This causes the fluids to be separated by plates with a large surface area. This type of heat exchanger is typically more efficient than the shell and tube design [4].

The regenerative heat exchanger takes advantage of the heat from a specific process in order to heat the fluid used in the same process. These heat exchangers can be made with the shell and tube design or the plate design. The intermediate fluid or solid heat exchanger uses the fluids or solids within it to hold heat and move it to the other side in order to be released. This method is commonly used to cool gases while removing impurities at the same time.

In this study, the parallel plates heat exchanger (the second design) is used, which is different from the types that are used in previous acoustical studies as example [5-9], and due to the complexity in the configuration of how the plates and water tubes are intermixed , few relevant references can be found [10-12]. These references are treating sound in narrow channels but with a different application - diesel particulate traps for IC-engines.

Sound Transmission in a Heat Exchanger

It is assumed that the heat-exchanger fine structure is much smaller than the acoustic wavelength making a continuous medium model possible. A general model is then a homogenous anisotropic medium with a negligible mean flow. . The linearized equations of motion along the three principal Cartesian axis x_n for such a medium are [3]:

$$\rho_n i\omega v_n = -\frac{\partial p}{\partial x_n}, \qquad n = 1,2,3 \tag{1}$$

and the equation of conservation of mass is

$$\frac{i\omega p\sigma}{c_h^2} + \rho_0 \nabla \cdot \bar{v} = 0. \tag{2}$$

Here, ρ_n and c_h represent the density and speed of sound (in the heat exchanger), p, v denote the pressure and particle velocity of the gas medium respectively, ρ_0 is the equilibrium gas density and σ denotes the porosity of the heat-exchanger. The effective density ρ_n can be related the structure and flow resistance of the medium

$$\rho_n = \rho_0 \chi_n + \frac{r_n}{i\omega}, \tag{3}$$

where χ_n is the structure factor in n direction and is calculated according to reference [13] and r_n is the specific flow resistance. By assuming an incident plane wave with a harmonic time dependence $e^{i\omega t}$ (omitted):

$$p_i = \hat{p}_i e^{-ik_{1,i}x_1 - ik_{2,i}x_2 - ik_{3,i}x_3} \tag{4}$$

The assumption of homogeneity in space implies that the principle of matching of wave numbers along the plane inlet and outlet surfaces at $x_1 = 0, L$ applies where L is the length (thickness) of the heat exchanger in the x_1 direction, also, $k_{1,i}$, $k_{2,i}$, $k_{3,i}$ are the wave number in the x_1, x_2 and x_3 directions, respectively at the inlet side. Combining equation (1) and (2), together with equation (3), now gives:

$$\xi_1^{-1}\frac{\partial^2 p}{\partial x_1^2} + \xi_2^{-1}\frac{\partial^2 p}{\partial x_2^2} + \xi_3^{-1}\frac{\partial^2 p}{\partial x_3^2} + \frac{\sigma\omega^2}{c_h^2} p = 0, \tag{5}$$

where

$$\xi_n = \rho_n / \rho_0 = \chi_n + \frac{r_n}{i\omega\rho_0} \tag{6}$$

The dimensionless quantity ξ_n is essentially the ratio of the effective density to the air density. Using equation (5) with equation (4), dropping the term $\hat{p}_{i,n}$, observing that the wave numbers in x_2 and x_3 are unchanged from the incident wave (Snell´s law), yields

$$\frac{k_{1,h}^2}{\xi_1} + \frac{k_{2,i}^2}{\xi_2} + \frac{k_{3,i}^2}{\xi_3} = \frac{\sigma\omega^2}{c_h^2} \tag{7}$$

where $k_{1,h}$ can be computed for a given incident wave and the data from the heat exchanger. The wave numbers over the heat-exchanger surface can be expressed as

$$k_{2,i} = k_0 \sin\theta \cos\phi \qquad k_{3,i} = k_0 \sin\theta \sin\phi \tag{8}$$

where the angles refer to spherical co-ordinates and the speed of sound c_h can be calculated according to [12,14]

$$c_h = c_0 \frac{\left(1 - F(s)\right)^{1/2}}{\left(1 + (\gamma - 1)F(\zeta \cdot s)\right)^{1/2}}, \; F(s) = \frac{2}{s\sqrt{-i}} \frac{J_1(s\sqrt{-i})}{J_0(s\sqrt{-i})}, \; s = a\sqrt{\frac{\rho_0\omega}{\mu}}, \text{ and } \rho_n = \frac{\rho_0}{1 - F(s)} \tag{9}$$

where c_0 is the adiabatic sound speed, ζ represents the Prandtl number, J_0 and J_1 are Bessel functions, s is the shear wave number, μ is the dynamic viscosity and a is the hydraulic radius.

The specific flow resistance r_n is calculated in accordance with equation (6) and equation (9) with mean flow effects added based on the measured pressure drop (Pressure drop, $\Delta p = R_1 U_1 + R_2 U_1^2$, where U_1 is average flow speed as across the heat exchanger) [3] as

$$r_1 = i\omega\rho_0\left(\frac{1}{1-F(s_1)} - \chi_1\right) + \frac{2R_2 U}{L} \quad \text{and} \quad r_n = i\omega\rho_0\left(\frac{1}{1-F(s_n)} - \chi_n\right), \quad n = 2,3. \tag{10}$$

Sound Transmission Calculation

Knowing that the sound transmission through the heat exchanger is most effectively handled using a transfer-matrix approach [15]. So, if only plane waves exist in a system with just two openings the system can be described as an acoustic two-port matrix. The most commonly used model is developed by using acoustic pressure p and velocity v_1 to represent the input and output state vector. In this case we also take advantage of the fact that 3D plane waves in the problem share the factor $e^{-ik_{2,i}x_2 - ik_{3,i}x_3}$ in the entire system. This together with the boundary conditions at $x_1 = 0, L$ and the continuity of p and v_1, implies that we only have to analyze the reflection and propagation in the x_1 direction. A transfer matrix suited for this problem can be defined by:

$$\begin{pmatrix} \hat{p} \\ \hat{v}_1 \end{pmatrix}_{x_1=0} = \begin{pmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{pmatrix} \begin{pmatrix} \hat{p} \\ \hat{v}_1 \end{pmatrix}_{x_1=L}, \tag{11}$$

where **T** is given by [3]:

$$T = \begin{pmatrix} \cos(k_{1,h}L) & \dfrac{i\rho_1\omega}{k_{1,h}}\sin(k_{1,h}L) \\ \dfrac{ik_{1,h}}{\rho_1\omega}\sin(k_{1,h}L) & \cos(k_{1,h}L) \end{pmatrix}. \tag{12}$$

To obtain the acoustic transmission the boundary conditions and the plan wave relationships on the inlet and outlet sides imply that:

$$\begin{cases} \hat{p}_{x_1=0} = \hat{p}_i + \hat{p}_r = \hat{p}_i(1+r) \\ \hat{v}_{1,x_1=0} = \hat{p}_i(1-r)\dfrac{k_{1,i}}{\rho_0\omega} \end{cases} \tag{13}$$

$$\begin{cases} \hat{p}_{x_1=L} = \hat{p}_t = \tau\cdot\hat{p}_i \\ \hat{v}_{1,x_1=L} = \hat{p}_t\dfrac{k_{1,t}}{\rho_0\omega} = \tau\cdot\hat{p}_i\dfrac{k_{1,i}}{\rho_0\omega} \end{cases} \tag{14}$$

where the amplitudes of the reflected and the transmitted fields are related to the incident wave with a transmission coefficient τ ($\tau = \hat{p}_t / \hat{p}_i$) and a reflection coefficient r ($r = \hat{p}_r / \hat{p}_i$). By substituting equations (13) and (14) into equation (11), it yields

$$\begin{cases} 1 + r = \tau \cos(k_{1,h} L) + i\tau \dfrac{\xi_1 k_{1,i}}{k_{1,n}} \sin(k_{1,h} L) \\ 1 - r = i\tau \dfrac{k_{1,h}}{\xi_1 k_{1,i}} \sin(k_{1,h} L) + \tau \cos(k_{1,h} L) \end{cases} \tag{15}$$

By evaluating equation (15), the transmission and reflection coefficients can be obtained as

$$\tau = \frac{1}{\cos(k_{1,h} L) + \frac{i}{2}(\frac{\xi_1 k_{1,i}}{k_{1,h}} + \frac{k_{1,h}}{\xi_1 k_{1,i}}) \sin(k_{1,h} L)} \tag{16}$$

$$r = \frac{\frac{i}{2}(\frac{\xi_1 k_{1,i}}{k_{1,h}} - \frac{k_{1,h}}{\xi_1 k_{1,i}}) \sin(k_{1,h} L)}{\cos(k_{1,h} L) + \frac{i}{2}(\frac{\xi_1 k_{1,i}}{k_{1,h}} + \frac{k_{1,h}}{\xi_1 k_{1,i}}) \sin(k_{1,h} L)} \tag{17}$$

The wave number $k_{1,h}$ can be obtained from equation (7):

$$k_{1,h} = \sqrt{\left(\frac{\sigma \omega^2}{c_h^2} - \frac{k_{2,i}^2}{\xi_2} - \frac{k_{3,i}^2}{\xi_3} \right) \cdot \xi_1}, \tag{18}$$

For the heat exchangers studied here x_2 is along the direction where there is a complete blockage as depicted in Figure 2, then ξ_2 can be set as infinite and k_2 drops out from equation (18). It then gives:

$$k_{1,h} = \sqrt{\left(\frac{\sigma \omega^2}{c_h^2} - \frac{k_{3,i}^2}{\xi_3} \right) \cdot \xi_1} \tag{19}$$

where $\mathrm{Im}[k_{1,h}] < 0$. For a given incident wave and heat exchange data, equations (18) and (19) can be used to compute $\tau(\bar{n}_i)$ and $r(\bar{n}_i)$. The corresponding absorption coefficient α is obtained from the value of *r* as:

$$\alpha = 1 - |r|^2. \tag{20}$$

The noise reduction index (*NRI*) or transmission loss *TL* for a certain incident angle (θ, φ) becomes

$$TL(\bar{n}_i) = 10 \log \frac{1}{|\tau(\bar{n}_i)|^2}. \tag{21}$$

where $\bar{n}_i$ is the unit vector along the incidence direction.

By averaging equation (16) over all possible incident angles, the transmission loss in a diffuse field is obtained [16]:

$$\langle TL \rangle = -10 \log \langle |\tau|^2 \rangle. \tag{22}$$

where

$$\left\langle \left| \tau(\bar{n}_i) \right|^2 \right\rangle_{all\ angles} = \frac{1}{2\pi} \int_{\theta=0}^{\pi/2} \int_{\varphi=0}^{2\pi} \left| \tau(\theta,\varphi) \right|^2 \sin 2\theta \cdot d\theta d\varphi, \qquad (23)$$

A similar treatment for the absorption coefficient in a diffuse field yields

$$\left\langle \alpha \right\rangle_{all\ angles} = \frac{1}{2\pi} \int_{\theta=0}^{\pi/2} \int_{\varphi=0}^{2\pi} \alpha(\theta,\varphi) \sin 2\theta \cdot d\theta d\varphi \qquad (24)$$

EXPERIMENTAL CHARACTERIZATION OF HEAT EXCHANGERS

ISO 15186 Experimental Procedure

Since the measurements were to be carried out with a mean flow through the heat exchanger it was not possible to fully apply a standard ISO procedure. Thus a slightly modified method described below was used for these measurements.

Measurement Procedures

Step 1: In this step there was no mean flow. A sound source emitting white noise was mounted in the reverberation room and the sound reduction index can be measured with ISO 15186-1:2000:

$$R_I = L_{pI} - 6 - (L_{In} + 10\log(S_m / S)) \qquad (25)$$

where L_{pI} is the sound pressure level measured by a rotating microphone in the reverberation room and L_{In} is the sound intensity level obtained by scanning the surface of the heat exchanger with an intensity probe in the receiving anechoic room. The whole setup can be seen in Figure 1a and for the present test case the areas S_m and S are equal. .

Step 2: Also in this step there was no mean flow. An inverse measurement compared to the first step was set up in order to give a reference value. Two larger sound sources were arranged in the anechoic room and generate white noise. Three microphones were put in front of the heat exchanger in the anechoic room. Then a second sound reduction index can be calculated as:

$$R_{\mathrm{II}} = L_{pa\mathrm{II}} - L_{pr\mathrm{II}} \qquad (26)$$

where $L_{pa\mathrm{II}}$ signifies the averaged sound pressure level from three microphones in front of the heat exchanger in the anechoic room and $L_{pr\mathrm{II}}$ is the sound pressure level measured by the rotating microphone in the reverberation room as shown in Figures 1b and c.

With these two values of sound reduction index R_{I} and R_{II} a correction factor can be introduced as:

$$A = R_{\mathrm{II}} - R_{\mathrm{I}} \qquad (27)$$

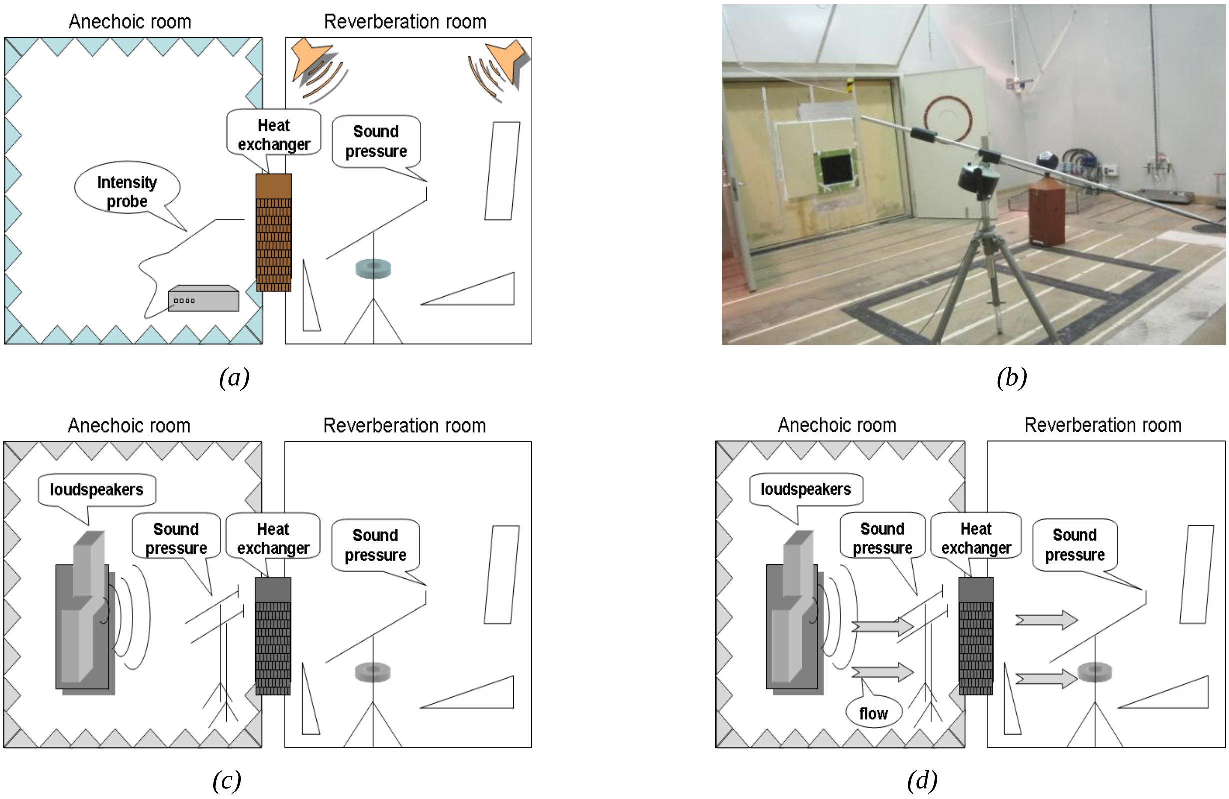


Figure 1: Measurement Setup used with Modified ISO Standard (15186-1:2000) procedure.

Step 3: The configuration used in step 2 was kept but now a steady mean flow from the anechoic room (which can be pressurized via fans) was added, see Figure 1d. Six different speeds of flow ranging from 2.5 m/s to 15 m/s were used to compare the acoustic characteristics of the heat exchanger for different flow speeds. In order to analyze the influence of self-generated sound by flow, the measurement was deployed in two situations for each flow speed: firstly, with the two sound sources off and just the flow creating flow induced noise and secondly, with the two sound sources on and with an unchanged flow speed. The data in the second measurement were then corrected for the flow noise contribution before a sound reduction index was computed as:

$$R_{\mathrm{III}} = L_{pa\mathrm{III}} - L_{pr\mathrm{III}} \tag{28}$$

where $L_{pa\mathrm{III}}$ and $L_{pr\mathrm{III}}$ are the sound pressure levels in the anechoic room and reverberation room respectively (corrected for flow noise).

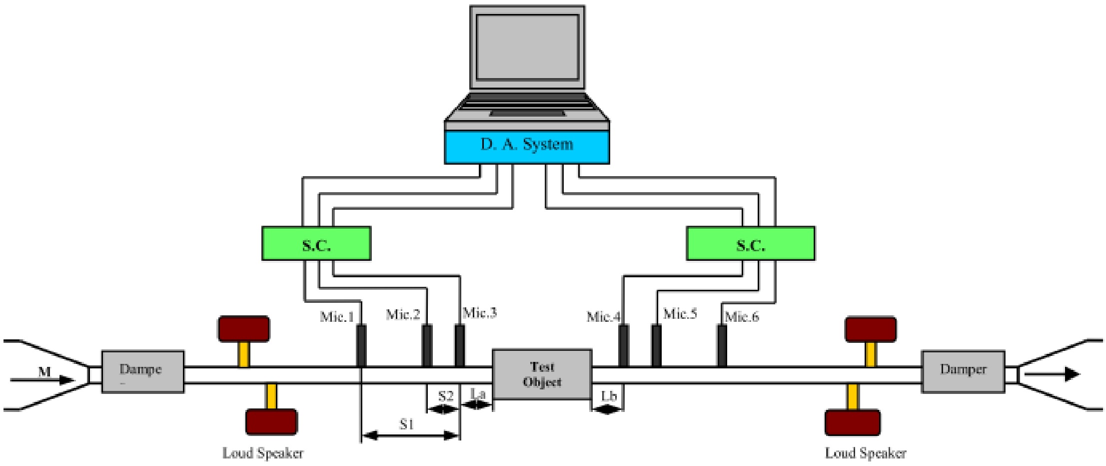
Step 4: We assume that no matter how large the flow speed is, the correction factor A obtained in the second step is unchanged. Then the final "ISO corrected" sound transmission loss of the heat exchanger for different speeds can be calculated as:

$$R_{\mathrm{IV}} = R_{\mathrm{III}} - A. \tag{29}$$

Besides the acoustic data the pressure drop as a function of the average flow speed through the heat exchanger was also measured. The pressure drop across the heat exchanger was measured by an electrical manometer and the averaged flow speed was obtained using hot-wire anemometry by scanning the surface of the heat exchanger in the reverberation room.

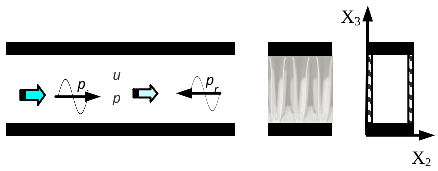
Two-Port Experimental Procedure

Experiments were carried out at room temperature using the flow acoustic test facility at the Marcus Wallenberg Laboratory for Sound and Vibration research (MWL) at KTH. The test duct used during the experiments consisted of a standard steel-pipe with a wall thickness of 3 mm, duct inner diameter d_i=91 mm and overall length of around 7 meters. Four loudspeakers were used as external acoustic sources, and they were divided equally between the upstream and downstream side as shown in Figure 2. The distances between the loudspeakers were chosen to avoid any pressure minima at the source position. Six flush mounted condenser microphones (B&K 4938) were used, three upstream and three downstream of the test object for the plane wave decomposition, the microphone separations are chosen to fulfill the frequency ranges of interest. All measurements are performed using the source switching technique [2] in order to determine the complete two-port data. From these data the transmission loss at normal incidence could be computed and also the reflection coefficient. The flow speed was measured upstream of the test section using a small pitot-tube connected to an electronic manometer at a distance of 1000 mm from the upstream loudspeakers section.



(a) Measurement configuration for plane wave decomposition at MWL.

(b) Samples of the used test object.

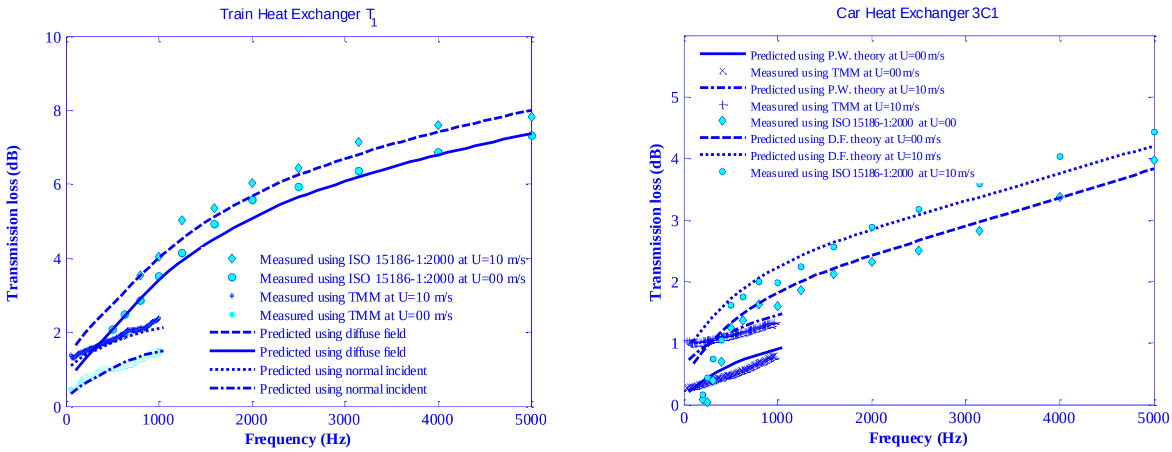


(c) Structure of the cell shape.

Figure 2. Measurement set up used during the Two - Port Experimental procedure.

HEAT EXCHANGERS-RESULTS AND DISCUSSIONS

The proposed model presented in the previous section is used to calculate the sound transmission loss in heat exchangers for both a diffuse sound field and it is also used for normal incident sound (where θ =0 , and φ=0). Figure 3 shows the transmission loss for a sample of a train and an automotive heat exchanger at two different flow speeds (U=0, and 10 m/s). It can be seen from the results that the model gives a good agreement with the measured results in both diffuse and normal incident sound field. It can be also noted that sound transmission loss of the heat exchanger is very small especially at low frequency which calls for the use of added damping, i.e., via the use of baffle silencers as will be discussed below. Another alternative not addressed here would be to modify the inner structure, e.g., the flow resistivity in the different directions, of heat exchangers in order to maximize the acoustic absorption and damping.



(a) Dimensions (430 × 400 × 140) mm , σ=64%. *(b) Dimensions (680 × 470 × 18) mm, σ=69%*

Figure 3. Transmission loss in one third octave band versus frequency.

PARALLEL BAFFLE SILENCERS MADE OF MICRO-PERFOTATED PLATES

Dissipative silencers are commonly used in HVAC ducts to attenuate broadband noise emanating from an air moving device such as a fan. HVAC ducts commonly have a rectangular cross section and a silencer made up of a number of parallel splitters. Each splitter normally consists of a bulk reacting porous material separated from the airway by a thin, perforated, metal sheet. Each perforated sheet is joined to metallic fairing at either end of the splitter, see Figure4. This helps to maintain the dimensional stability of a splitter, but also to channel airflow between each splitter, lowering the static air pressure loss across the silencer [17-21].

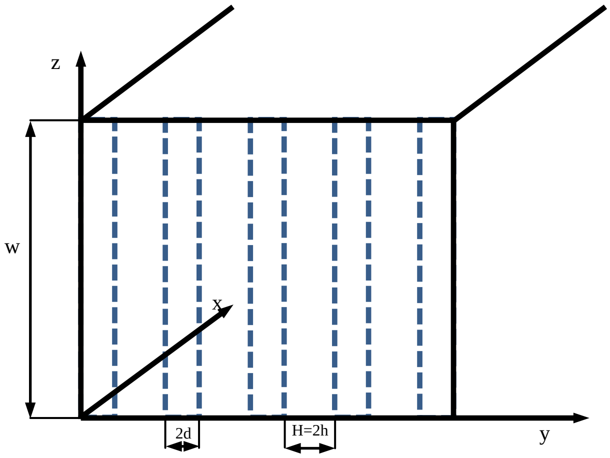


Figure 4: The structure of parallel baffles muffler air channel width 2h, baffle thickness 2d.

Micro perforated panel (MPP) absorbers, which are made of metal have the potential to be used instead of porous materials in dissipative mufflers, which not only can save weight but also offer a non-fibrous alternative. This helps to avoid the harmful effects of some fibrous materials on health, especially when they are applied in heating, ventilation, and air-conditioning HVAC systems.

Furthermore, since MPP absorbers have a large steady flow resistance they can be used as acoustically absorbing guide vanes at duct bends or in fans [22]. Also the effects of flow, high temperatures, and high sound levels on the acoustic impedance of the MPP absorber with circular and slit-shaped holes are presented in Ref. [22]. So, in this study the inlet parallel baffles muffler is built based on slit-shaped holes MPP absorber as shown in Figure 5 (a), and the internal structure of the baffles is built with or without rigid inner walls as shown in Figure 5 (b). The inner wall was chosen to create wave fields inside the MPP with resonances in the mid frequency to increase the damping in this range. The measurements were only performed using the modified ISO 15186-1:2000, which is previously presented, and the installation in the wall is shown in Figure 6. The baffle dimension (B=0.05 × 1 × 0.5 m) was kept constant while the air spacing between two baffles (H=2h) was varied according to number of baffles within the total width 1 m.

(a) Photo of the MPP used with slits (Acustimet™). (b) MPP baffle with rigid inner walls and spacing 165× 241 mm.

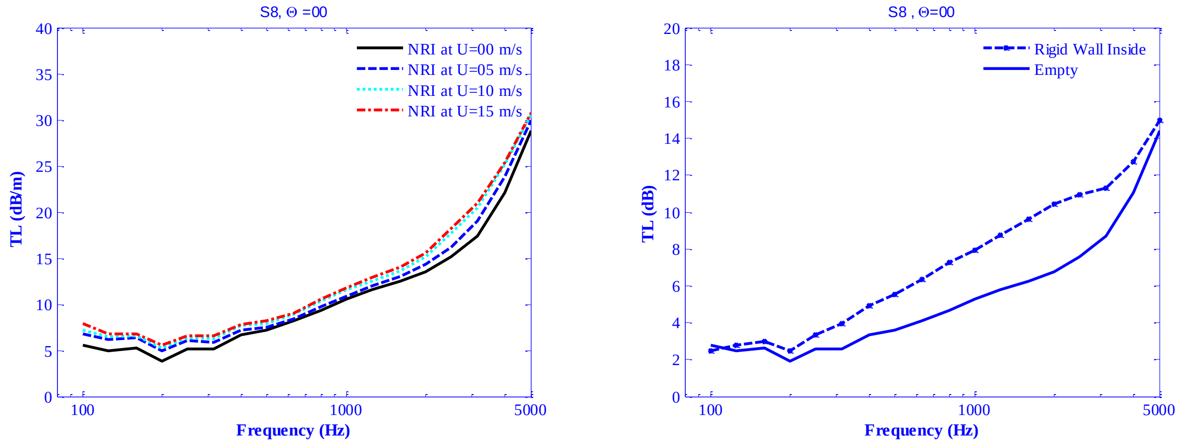
Figure 5: Structure of a parallel baffles silencer with inner walls.

(a) Reverberant room side. *(b) Anechoic room side.*

Figure 6: Fixing the Test object in the wall between the two rooms.

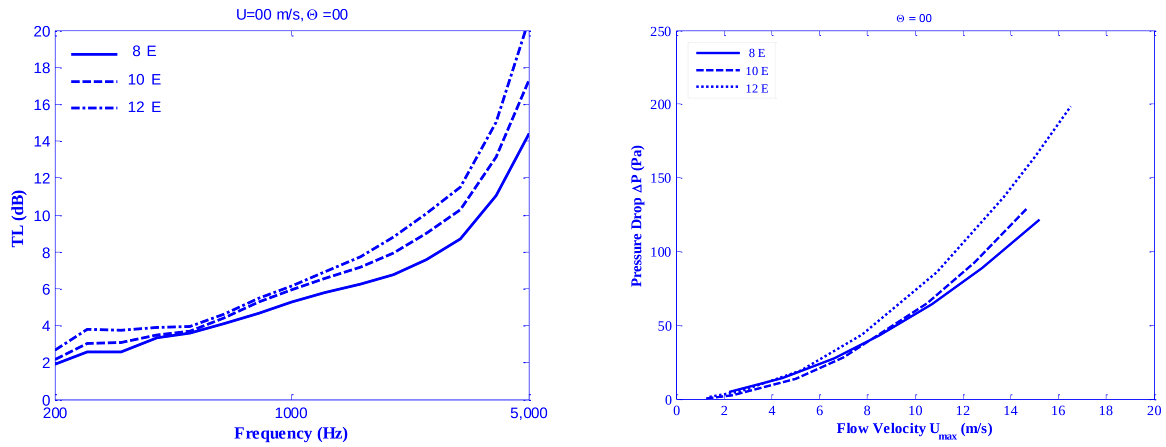
MPP BAFFLE SILENCERS - RESULTS AND DISCUSSIONS

As shown in Figure 7 (a), the Transmission loss (TL) increases with the frequency and there is also some small flow effect (< 2 dB up to 15 m/s). As expected using a baffle with inner walls increase the noise reduction in the mid frequency range, this behavior is found in all tested parallel absorbers. The TL and pressure drop increase with number of baffles as shown in 8 and also with the inclination angle as shown in Figure 9.



(a) Effect of different flow speed. *(b) Effect of inner rigid walls.*

Figure 7. Transmission loss (TL) versus frequency for an 8 MPP baffle silencer (2h=85.4 mm).



(a) Noise reduction versus frequency. *(b) Pressure drop versus flow Speed.*

Figure 8. Effect of Number of baffles on transmission loss and pressure drop.

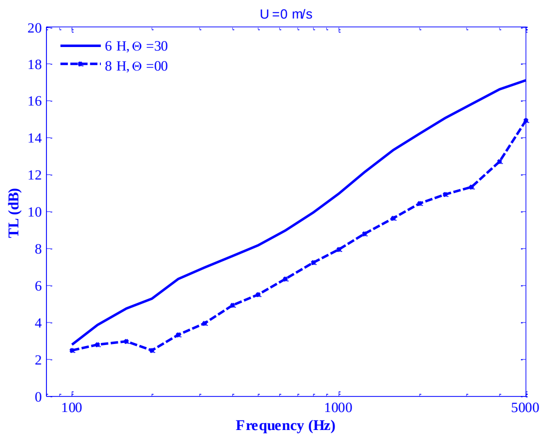


Figure 9. Effect of Angle on transmission loss.

CONCLUSIONS AND FUTURE WORK

A model for sound transmission through heat exchangers has been presented. The model is based on an assumed anisotropic equivalent fluid system. This type of model is valid as long as the characteristic length scale of the heat exchanger inner structure is much smaller than the wavelength. The data needed for the model can be obtained from the geometry of a given heat exchanger. To obtain the losses in the model either pressure drop data can be used or for the parallel plate type of heat exchangers, the losses can be estimated using the Kirchhoff model for sound in narrow tubes. The model gives a good agreement with the measured results

A new type of micro-perforated parallel baffle type of silencer has also been proposed. The transmission loss and flow-generated sound have been investigated for seven prototypes of parallel baffle mufflers. It is found that the transmission loss can be enlarged by increasing the number of baffles, adding inner rigid walls inside the baffle and using inclined baffles. The inclined baffles mufflers with inner rigid walls have the largest transmission loss. The inclination increases the high frequency performance while the inner rigid walls give an increase for the mid frequency range. The flow-generated sound is affected by the number of baffles and the inclination angle. Increasing the number of baffles and increasing the inclination angle will increase the flow noise.

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