NOISE REDUCTION FOR AUTOMOTIVE RADIATOR COOLING FANS

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SUMMARY

Engine cooling fans have long been recognized as one of the major noise sources in a vehicle. As the engine and other vehicle components are made quieter, the need to reduce fan noise has become more and more urgent. To reduce fan noise in a cost-effective manner, it is necessary to incorporate the component of noise reduction into an early design stage. In this paper a detailed experimental study on an automotive vehicle cooling system is presented. The aim is to investigate the flow generated noise, characterize the heat exchanger damping properties and investigate the use of near-field noise control by micro-perforated (MPP) shrouds and tuned MPP dampers. For the tested standard automotive cooling fan system the MPP shroud gave a reduction in the range 1.5 to 4.5 dB(A) depending on the fan speed. Also the absorption on the back-side is significantly increased which can reduce the noise further. The near-field tuned MPP damper concept is also promising and gives a reduction around 3 dB(A) at the operating points.

INTRODUCTION

Background

Low-speed rotating axial and radial fans are frequently used to manage engine temperature by ensuring adequate airflow through heat exchangers (radiator), especially at low vehicle speeds or idle. An undesirable side effect of these fans is generation of flow-induced noise, which is an annoyance to the operators and passengers, and a source of community noise, especially for commercial vehicles and heavy equipment. In most cases and particularly for high mass flow configurations, the cooling fan is a major contributor to the overall noise, and in some cases dominates relative to other sources such as engine, transmission, tire, mechanical, or exhaust contributions [1]. In addition, cooling fan noise is a perceived quality issue that can affect brand image and customer satisfaction. Given the regulations and growing importance of acoustic comfort in many markets, there is high value in addressing cooling fan flow-induced noise problems as early as possible during product development.

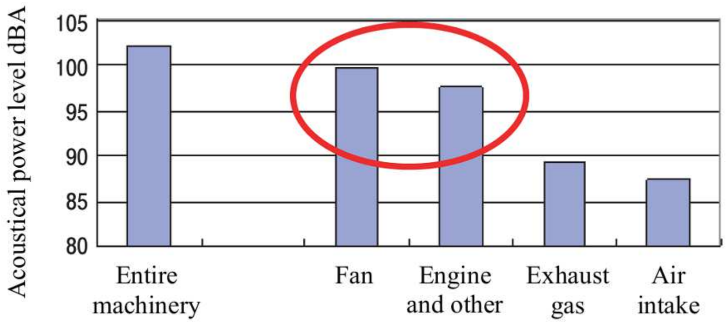


Figure 1. An analysis of contribution factors of construction machinery.

Some investigations have been published on noise generation and sound radiation from cooling systems in vehicles. Staiano [2] had investigated the engine fan noise in buses and made measurements on a transit coach operated over a specified operating cycle to determine the relation of fan usage to noise exposure levels. In Ref [3], the development of a radiator cooling fan used in automobile vehicles has been reviewed with a proposed noise control technique developed for the design of quiet marine propellers. Frid et.al. [4] made extensive investigation on diesel train engine cooling noise as part of the EU-project SILENCE. One obvious conclusion from the above works is that increased use of cooling control strategies, which also considers noise emission is a measure which can be applied more. Some recommendations have been presented by Dittrich and Zhang [5] regarding the relation between cooling fan noise and rotational speed under average conditions of constant speed, acceleration, standstill/idling and deceleration. Schulte-Werning et.al. [6] described a test made on a locomotive to find ways of reducing the noise of the cooling ventilation by several modifications such as: the change of fan blade configuration and the use of sound-absorbing surface layers and silencers.

Fan noise mechanisms

Generally, the principal noise mechanisms of low tip speed, axial flow fans can be separated into the two categories of non-rotational and rotational [7]. The non-rotational includes blade interactions with inflow distortion and turbulence, and with nearby fixtures, while the rotational noise includes laminar boundary layer vortex shedding, blade interactions with the tip clearance vortex and blade stall. The above mechanisms will generate aeroacoustic dipole noise. Concerning monopole noise related to the blade thickness it can be neglected for subsonic fans [8]. Finally, quadrupole noise due to the unsteady momentum transport in turbulent flows, is also negligible for subsonic fans [9]. Therefore as proposed by Neise [10], the dominant aeroacoustic noise source for subsonic fans comes from dipole sources. The dipole sources can be divided into two types, due to the steady rotating forces and due to the unsteady rotating forces. For a uniform inflow without disturbances, periodic pressure fluctuations at discrete harmonics exist and the corresponding tonal noise is referred to as 'Gutin-noise' [11]. In the case of low Mach-numbers the 'Gutin-noise' contribution is negligible. The non-uniform stationary inflow will create the harmonic part of the fan noise spectrum with tones at multiples of the blade passing frequency (BPF = (B number of blades) × (shaft rotational frequency)). The levels of these tones are, especially for axial fans, strongly influenced by interaction with outlet guide vanes. It is not surprising that the separation between guide vanes and blades has a large influence on the level of the tones [12], but also the circumferential periodicity plays an important role [13].

The non-uniform unsteady flow entering the fan is mainly associated with inflow turbulence and produces low frequency broad band noise.

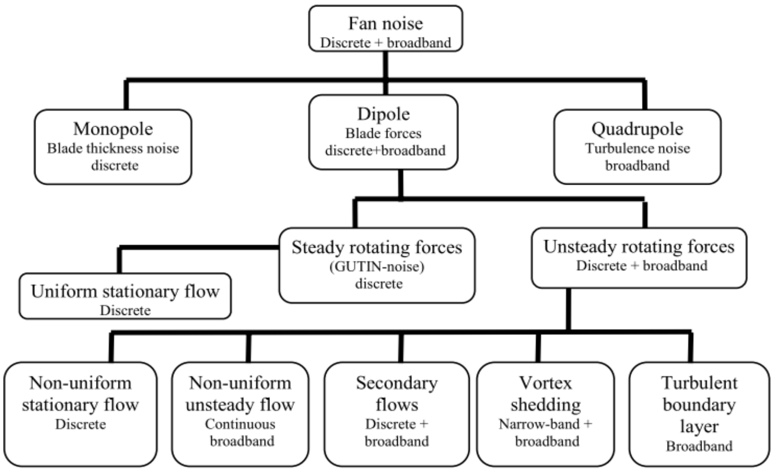


Figure 2. Overview of the aeroacoustic sound generation mechanisms for fans [10].

The last three source terms at the bottom in Figure 2 consist of several parts categorized as self-noise and concern the flow induced sound over the blades. Much attention has been paid to the research of these mechanisms, see for instance Refs [14-21]. Tip clearance noise or tip clearance vortex noise is one of the most important self-noise sources [10] and contributes to the high frequency broadband spectrum. The mechanism is simply that air is forced to flow between the high and low pressure side through the gap at the blade tip and then forms vortices propagating with the flow. The tip clearance flow has significant effects on both the acoustic and the aerodynamic properties of the fan.

Fan noise reduction methodologies

A useful starting point to reduce the noise generated from the engine cooler is to draw a source and transmission path model, which is shown in Figure 3.

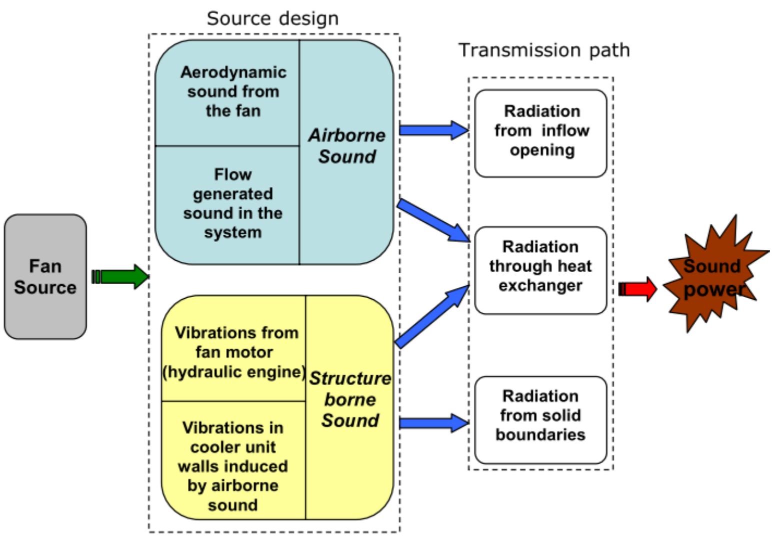


Figure 3. A source-transmission path model for the noise produced by the cooling fan unit [22].

Such a model lists the main source mechanisms and the main transmission paths for the acoustic power via various subsystems into sound power radiated to the outside. Based on this model one can then try to rank the different sources and transmission paths in order to determine a priority list for noise control measures [22].

The fan noise reduction can be achieved by: (i) noise reduction by source modifications; (ii) noise reduction by transmission path modification [11]. The noise reduction by source modifications (i) can be performed by minimizing fan RPM for a given volume flow (Low ΔP design), reversing airflow and use an axial fan in a pushing configuration [4], using axial fans with swept blades or radial fan, reducing rotor-stator interaction noise (by wake generator, etc), optimizing the number of guide vanes (Tyler-Sofrin rule), reduce tip clearance noise [4] and using an active control i.e, inserting anti-sources [22]. Most of all these proposed source modification have taken into account during the ECQUEST [23] project, the final Fan configurations are shown in Figure 6(a, b) and Figure 7(a, b). The present paper contributes to (ii) noise reduction by transmission path modification. A detailed experimental study on an automotive vehicle cooling system is presented. The aim is to characterize the heat exchanger damping properties and investigate the use of near-field noise control by so called micro-perforated plates (MPP).

MEASUREMENT PROCEDURE

General

All test objects have been tested using Flow Acoustic Test Facility at The Marcus Wallenberg Laboratory (MWL) at Kungliga Tekniska Högskolan (KTH). The test objects were mounted in the wall between the anechoic and the reverberant rooms and treated as wall elements and the measured results were performed in one third octave band.

Experimental characterization of Heat Exchangers (HE)

A sound source emitting white noise was mounted in the reverberation room and the sound transmission loss is calculated based on the measured results using the ISO 15186-1:2000 [24] as:

$$TL = L_{pI} - 6 - (L_{In} + 10\log(S_m / S)) \qquad (1)$$

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 [25]. The whole setup can be seen in Figure 4 and for the present test case the areas S_m and S are equal. The measurement with flow is performed based on the modification of ISO 15186 which is presented in Reference [26-27].

(a) *(b)* *(c)*

Figure 4: Measurement setup used with ISO standard (15186-1:2000) procedure.
(a) Reverberant room side,
(b)HE & condenser,
(c) Anechoic room side.

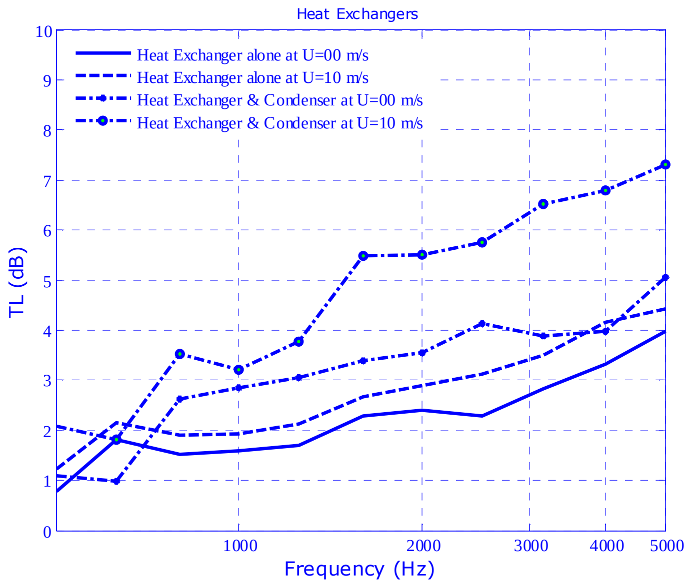


Figure 5.Acoustic characteristics of heat exchanger, (a) Measured TL using the ISO: 15186-1:2000.

Using the measured transmission loss presented in Figure 5 and based on the theoretical model which is presented by the same authors [26], the acoustic performance of exchanger can be theoretically characterized and optimized using MPP pipes instead of louvered substrates.

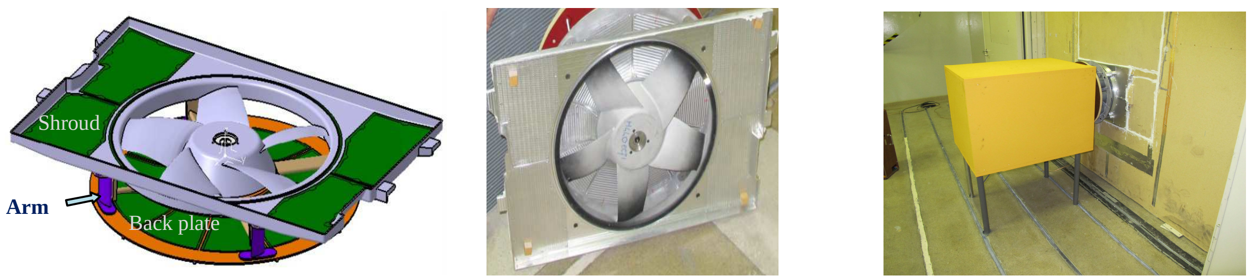
Experimental characterization of cooling fan units

The radiated sound power in one third octave band is measured using the ISO 3747 method [28] with a reference source. The sound power (L_w) is calculated based on the measured results as follow:

$$L_W = L_p + \left(L_{W,R} - L_{p,R}\right) \tag{2}$$

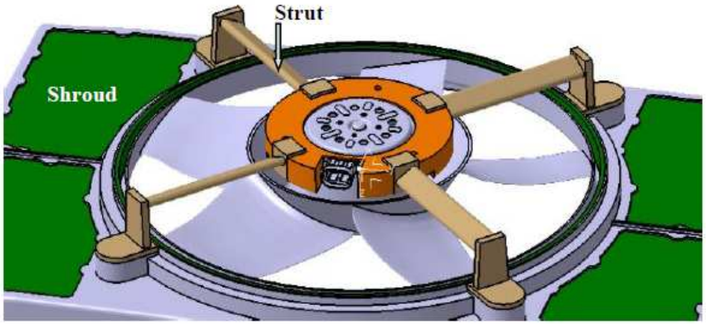
where, L_p is the average sound pressure of the source in the reverberation room, $L_{p,R}$ and $L_{W,R}$ is the sound pressure level (SPL) and sound power of the reference source. The background noise is measured and has been canceled out from the final results.

In this study, two cooling units are acoustically characterized; a five bladed semi radial fan see Figure 6 and an eight bladed axial fan as shown in Figure 7. During these studies the effect of engine block on the radiated sound power is taken into account by using a wooden block with the same engine size at 110 mm from the fan housing.



(a) ***(b)*** ***(c)***

Figure 6. Semi – radial fan. a) Constructor drawing of fan used during the experiments. Note the back plate that redirects the outflow into the radial direction. Green areas denote the shroud sections + back plate replaced by MPP, b) Modified unit with shroud and back plate made partly in MPP. c) Shows the Engine dummy block in wood: $L\times W\times H=880\times682\times627$ mm^3 standing in the reverberation room.



(a)

(b)

Figure 7. Axial fan used during the experiments. Green areas denote the shroud sections replaced by MPP. (a) Constructor drawing , (b) Photo of the unit.

The effects of shroud, back plate material and a dummy engine block which shown in Figure 6, on the acoustic source strength (radiated sound power) were measured under different controlled conditions. For the back plate two MPP arrangements were used. A single (standard) MPP plate and a double wall MPP with two standard plates separated 10 mm by a space filled with melamine foam. The cooling units were derived with electric motor and mounted between two acoustic test chambers with the receiving (downstream) side in reverberant chamber where a rotating boom was used to measure the source strength for different fan RPM´s, see Figure 6 (c). Sample of the measured results for the semi-radial fan with original and modified shroud with MPP is presented in Figure 8. Also, the effect of the MPP shroud on the axial fan radiated sound power is tested, compared with the original unite at the desired operating speeds are presented in Figure 9.

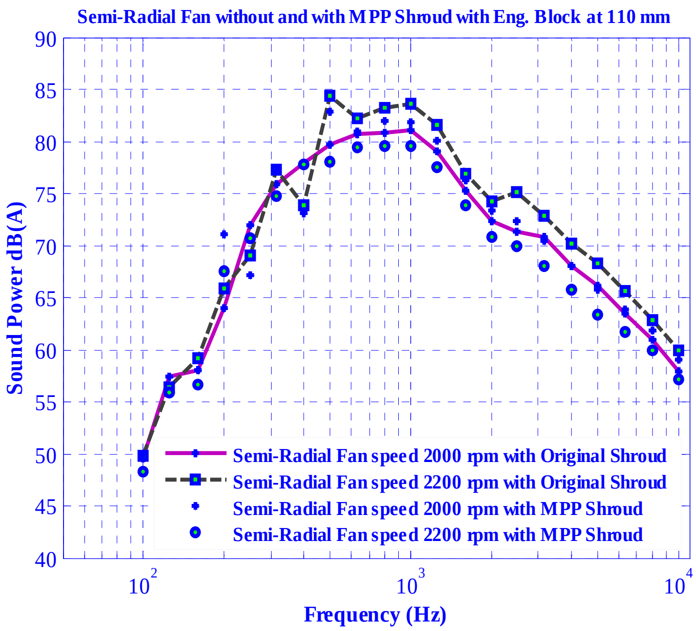


Figure 8. Sound power versus frequency for the semi-radial fan, original shroud, MPP Shroud and MPP back plate, see Figure 6a.

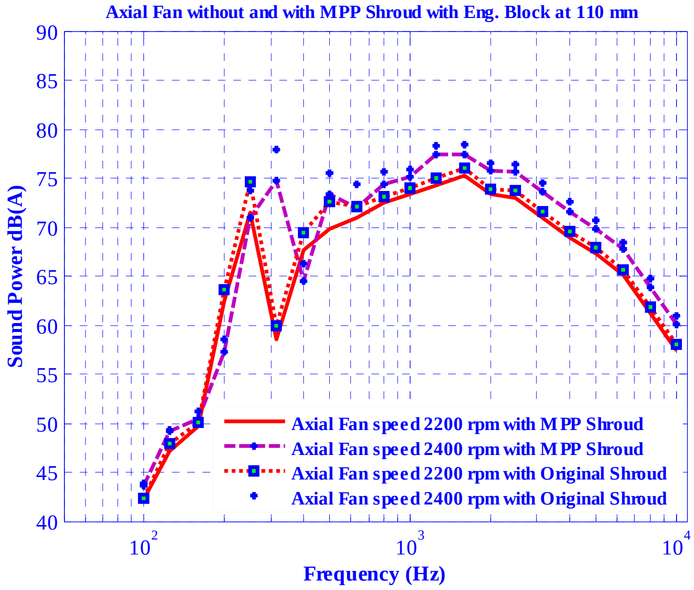


Figure 9. Sound power versus frequency for the axial fan with original, MPP shrouds and engine block.

Comparison between measured results for semi-radial and axial fans

In Figure 10, the radiated sound power of two fan prototypes are compared at two different speeds which give the same volume flow rate, and both working points are almost at the best efficiency of fans. As seen from the figure, the semi-radial fan is noisier than the axial fan at the desired operating conditions.

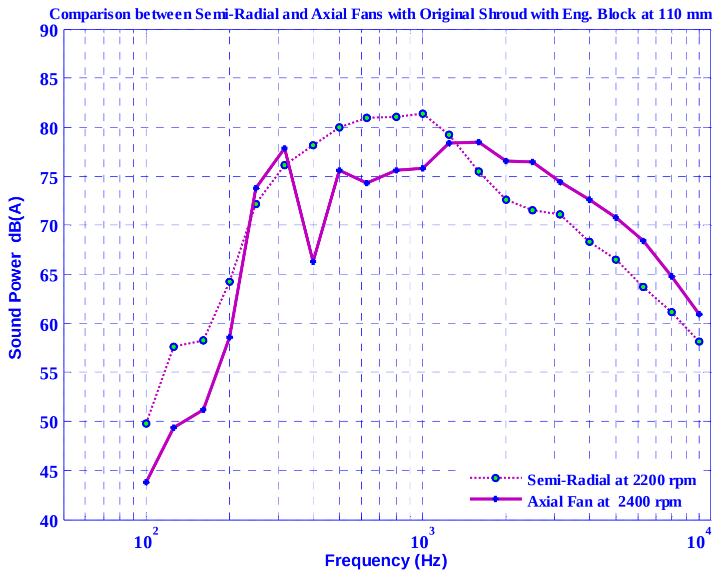


Figure 10. Comparison between semi-radial 89 dB(A) and axial fans 87.5 dB(A) at the desired operating conditions using original shroud.

The effect of using the MPP shroud on two fan prototypes are presented in Figure 11, it can be noticed that the reduction with a semi-radial fan is higher than the reduction with an axial fan, mainly due to the larger area (shroud + back plate) covered with MPP, compare green areas in Figure 6(a) and Figure 7(a). It can be seen that the radiated sound power could be reduced between (2-4) dB(A) by using a MPP shroud compared with the original metal shroud. However, it can be noted that this reduction goes down as the RPM goes up. The reason is that a MPP has a frequency limit where the imaginary part of its impedance starts to dominate. For the MPP used here, this is around 1000 Hz and for frequencies much higher than this the damping is poor. This limit can be controlled by using a MPP with smaller holes, which will push the frequency limit up in frequency. The use of a double wall MPP has a very small effect and gives no significant improvement; see Figures 10. Another alternative to handle this problem, i.e., extending the MPP frequency range, which will be discussed in the incoming section; is to use a resonant volume on the back side of the MPP.

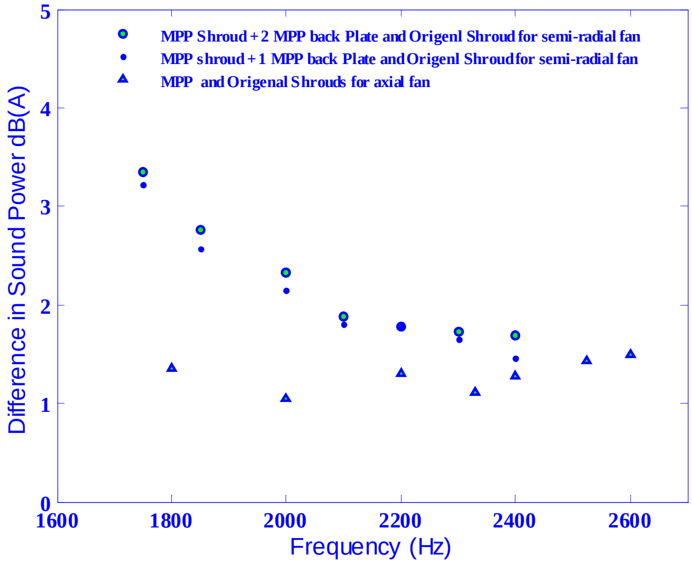


Figure 11. Difference in sound power using the original and MPP Shroud with single and double MPP back plate with engine block for semi-radial and axial fan.

Design of a tuned MPP damper

The idea of the concept is to maximize the MPP damping in a certain frequency range, i.e., around the resonance of a Quarter Wave Resonator (QWR). To test the concept the semi-radial fan back plate was fitted with QWRs as shown in Figure 12. The acoustic impedance of a MPP coupled to a resonator is composed of facing-sheet impedance (resistance and reactance) and air-cavity impedance. Here it is assumed that the cavity is locally reacting and behaves as a quarter wave resonator. This gives

$$z^* = \left(r_{MPP} + ix_{MPP}\right) - i\cot(kh), \tag{3}$$

where $z^*= Z/\rho c$ is the normalized acoustic impedance, Z is the acoustic impedance, ρc is the characteristic impedance of air, r_{MPP} is the normalized acoustic resistance, x_{MPP} is the normalized acoustic reactance, k is the wavenumber, and h is the resonator depth in m. For the MPP type used here with slit like holes it was decided to use the impedance formula presented in Ref. [29,30], which can be summarized as follow, the normalized resistance can be written as

$$r_{MPP} = \mathrm{Re}\left(\frac{j\omega t_P}{\sigma c}\left[1 - \frac{tanh\left(k_S\sqrt{j}\right)}{k_S\sqrt{j}}\right]^{-1}\right) + \frac{2\alpha R_S}{\sigma\rho\, c} + \frac{0.3M_g}{\sigma}, \tag{4}$$

and the normalized reactance can be written as

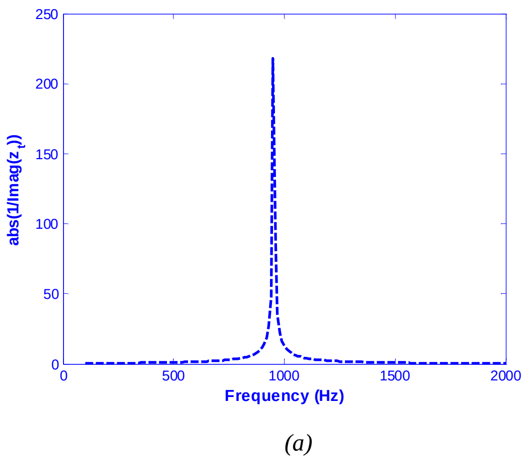
$$x_{MPP} = \mathrm{Im}\left(\frac{j\omega t}{\sigma c}\left[1-\frac{tanh\left(k_S\sqrt{j}\right)}{k_S\sqrt{j}}\right]^{-1}\right)+\frac{\delta\omega F_\delta}{\sigma c} \tag{5}$$

where, $k_s = d_h\sqrt{\omega/4\eta}$ is the Stokes number relating the slit width to the viscous boundary layer thickness, σ is the MPP porosity, M_g is the grazing flow Mach number, t_p is the MPP thickness and d_h is the slit width, α is 4 for sharp slit edges, $R_S = \sqrt{2\eta\rho_o\omega}/2$, the factor δ is the acoustic end correction for both side of the slit and put equal to $0.62d_h$ and $F_\delta = \left(1+\left(12.6\,M_g\right)^3\right)^{-1}$ is the flow effect on acoustic reactance. The shape of QWR and how it is connected is shown in Figure 12 and the working frequency is shown in Figure 13 (a). Also, the related absorption (normal incidence) coefficient (α) can be calculated as:

$$\alpha = \frac{4\,r_{MPP}}{\left(1+r_{MPP}\right)^2+\left(x_{MPP}-\cot(kh)\right)^2} \tag{6}$$

The absorption coefficient of the total designed unit is shown in Figure 13(b).

Figure 12. Semi – radial fan with, MPP shroud and MPP back plate fitted with QWRs. QWR dimensions; inner diameter 100 mm, height 80 mm, covered with MPP with, thickness 1 mm, slit width 0.2 mm and 1% porosity. QWR covered area is 40% of the total back plate area.



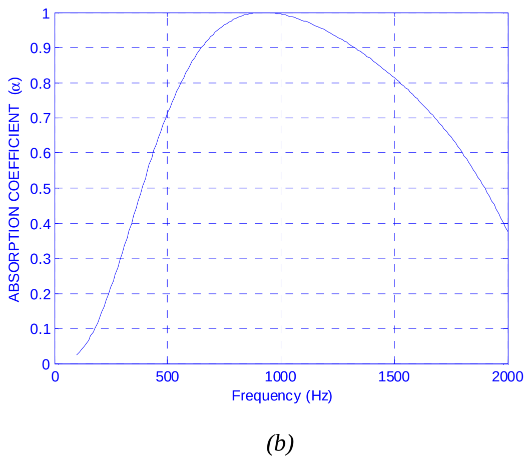


Figure 13. Acoustic performance of a tuned MPP damper based on the data in figure 12.
(a) Acoustic mobility magnitude of MPP and QWR, (b) Absorption coefficient of MPP and QWR at normal incidence.

Effect of tuned MPP damper on semi-radial fan radiated sound power

Based on the measured results presented in Figures 14, it can be concluded that the radiated sound power from the fan cooling unit can be reduced by 4 dB(A) in a given 1/3-octave, which can be increased to 6 dB(A) by combining with the MPP Shroud as shown in Figure 15(b). It can be noted that covering the whole back plate area would increase the damping and bandwidth. Also one can tune the resonators differently to (say) 2-3 consecutive 1/3 octaves to further increase the bandwidth. For the tested prototype the total sound power reduction amounts to 2 dB(A), see Figure 16.

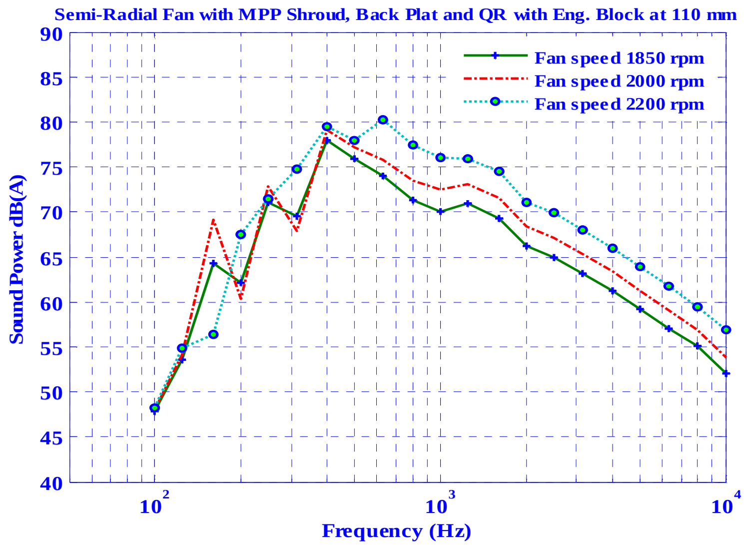


Figure 14. Sound power versus frequency for the semi-radial fan, with MPP shroud, MPP back plate quarter wave resonators (QWR) i.e. the tuned MPP damper.

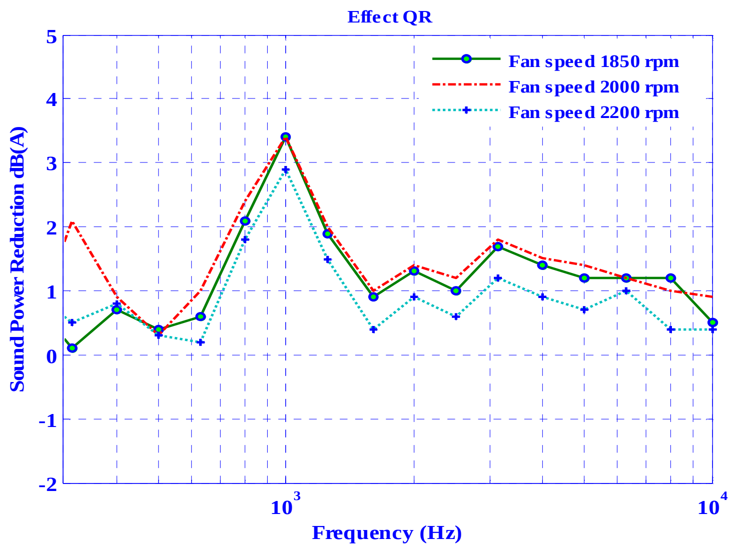


Figure 15(a). Reduction in sound power versus frequency for the semi-radial fan at different fan speeds, Tuned MPP damper only,

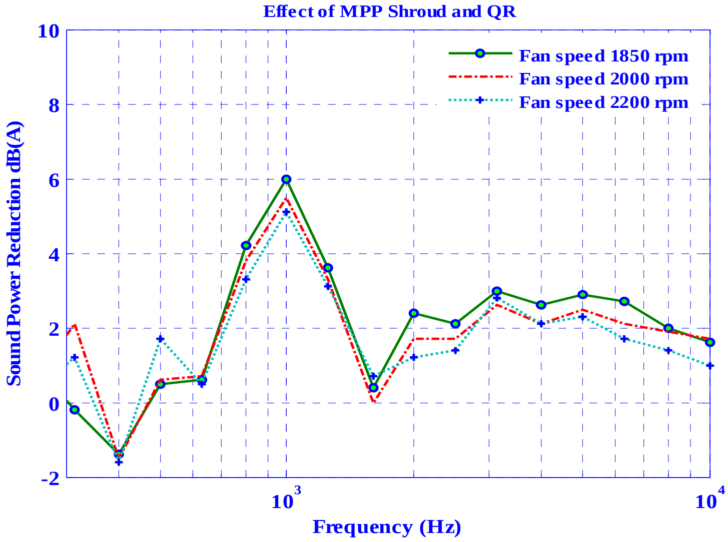


Figure 16.(b) Reduction in sound power versus frequency for the semi-radial fan at different fan speeds, MPP shroud and tuned MPP damper.

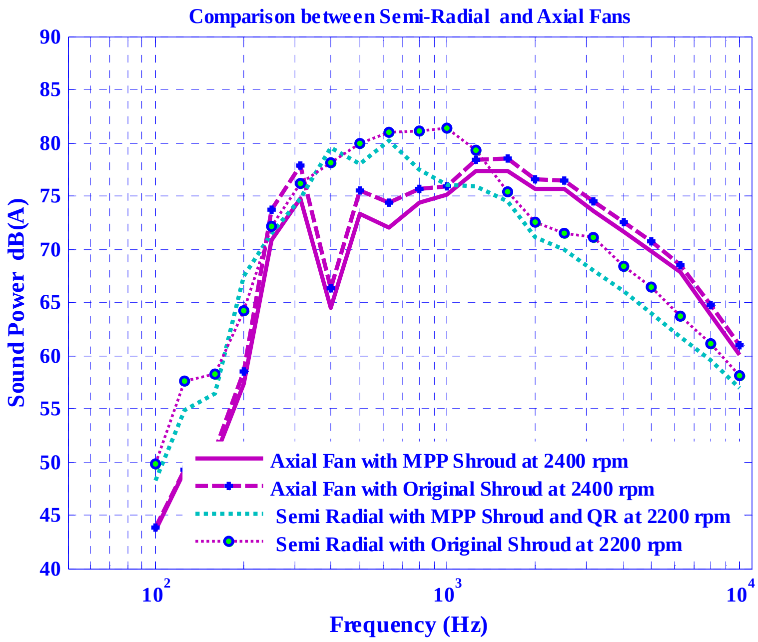


Figure 17. Comparison between semi-radial and axial fan at the desired operating conditions before and after adding modifications. Semi-radial fan with original shroud 89 dB(A), semi radial fan with MPP shroud and tuned MPP damper 87 dB(A), axial fan with original shroud 87.3 dB(A) and axial fan with MPP shroud 85.8 dB(A).

CONCLUSIONS

Based on the presented results, a micro-perforated (MPP) shroud can reduce the total sound power radiated from an automotive cooling fan unit with 1.5 to 4.5 dB(A), depending on covered area and fan speed, see Figures 6-9 and 11. Also the absorption on the back-side is significantly increased which can reduce the noise further. The concept of using a tuned MPP damper is also tested and is promising giving a reduction around 3-4 dB(A) at the operating points, see Figures 12-14. The MPP shroud plus the tuned MPP damper (=back plate and quarter wave resonators (QWR)), can reduce the total sound power around 6 dB (A), see Figure 15.

Acknowledgments

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