Fan Noise 2007

COOLING FANS IN RAILWAY VEHICLES – APPLICATION OF NOISE CONTROL MEASURES TO A ROOF-MOUNTED ENGINE COOLER

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SUMMARY

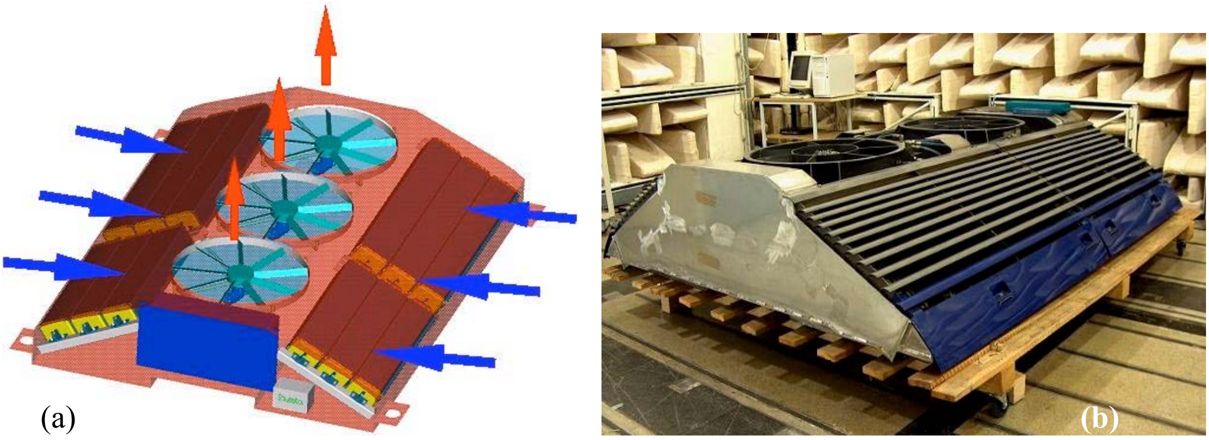
An experimental study has been undertaken on a railway vehicle cooling system in order to evaluate different noise control measures. The selected application is a large roof-mounted diesel engine cooling unit in the AGC (Autorail Grande Capacité) train. A strong presence of harmonic components from orders related to the 8-blade axial fan and the 7-piston hydraulic engine was found. In order to reach a substantial noise reduction, the broadband part of the noise needed to be brought down as well. A number of design modifications were proposed and tested in laboratory measurements: different ways to guide the air flow into the fans, reduced tip clearance, absorption, vibration isolation of the fan and of the hydraulic engine.

INTRODUCTION

The noise from rail and road traffic in urban areas is a major issue in many European cities. This topic is dealt with in the ongoing 6th framework EU-project "Silence" (ref TIP4-CT-2005-516288). In or around station areas cooling fans are typically the dominating noise sources from a train - both in terms of relatively high sound power levels and of high tonality. New legislation setting noise limits for stationary and accelerating trains puts further emphasis on increased noise reduction of cooling fans. The present study reports work carried out within the Silence project including a detailed examination of a typical railway vehicle cooling fan installation and an assessment of the noise reduction potential by different means. Two companion papers [1] and [2] describe the characterisation of the fan itself as an acoustic source and the fan noise modeling done within the project.

The selected application is a diesel engine cooling unit in the AGC (Autorail Grande Capacité) train. Even though this train is considered as a modern state-of-the-art product having very low noise emission, the engine cooler is identified as the dominating noise source when operating at its maximum capacity and the train is at rest or running at low speed. Because of geometrical, environmental and thermodynamic constraints the cooler is positioned on the roof. Generally for coolers, conventional noise reductions methods like encapsulation are not easily applicable because of the needed functionality (e.g. flow, thermodynamics).

The cooler under study has two axial fans operating under far from ideal flow conditions: therefore the aerodynamics needs to be improved for a reduction of flow induced noise. It is known that inflow disturbances, both steady distortions and turbulence, in particular can increase the sound generation from axial fans [3]. This and other flow related effects deviating from an ideal case or the case under which the fan has been tested, e.g., by the manufacturer, are called aerodynamic installations effects. To minimize such effects it is generally more advantageous to use an axial fan in a pushing configuration rather than a sucking. A cooling fan is required to deliver a certain volume flow and to do this it must overcome the pressure drop of the system. The pressure generated by a fan is via the fan scaling laws known to be proportional to the rpm squared. Therefore by designing a system with a smaller pressure drop one can reduce the fan rpm for a constant volume flow. This is important since the sound power from a fan can be expected to increase as the rpm raised to the power 5 [3]. Besides reducing the fan source strength by controlling the inflow and the fan rpm one must consider the acoustic response of the system. This is called the acoustic installation effect and is in particular of importance for low and intermediate frequencies or when there is a modal response in the system [6]. Regarding other works on noise control on cooling fan units a literature survey can be found in Ref. [4].



(a) (b)

*Figure 1: (**a**) Schematic roof-mounted engine cooling unit (note that the AGC train cooler has two fans instead of three). Direction of air flow indicated by the arrows. (**b**) Actual AGC train cooler.*

(a) (b)

*Figure 2: (**a**) Inlet side of fan, (**b**) Outlet side of fan.*
The hydraulic engine with seven pistons is visible inside the hub.

BASELINE TEST

An initial experimental laboratory study was carried out to establish the baseline situation for the cooling unit (i.e. the sound power spectra and directivity patterns for a complete range of operating speeds). The measurements were carried out in a semi-anechoic chamber at MWL, KTH (see Figure 1b). In the first measurements double peaks appeared in the sound power spectra, which implied that the two fans ran at slightly different speeds. This can occur with an hydraulic system arrangement in which the fans are connected in parallel circuits and the loads on the fans differ slightly. For the subsequent measurements one of the fans was disconnected to avoid this influence and to facilitate comparison with the measurements of the single fan in [1].

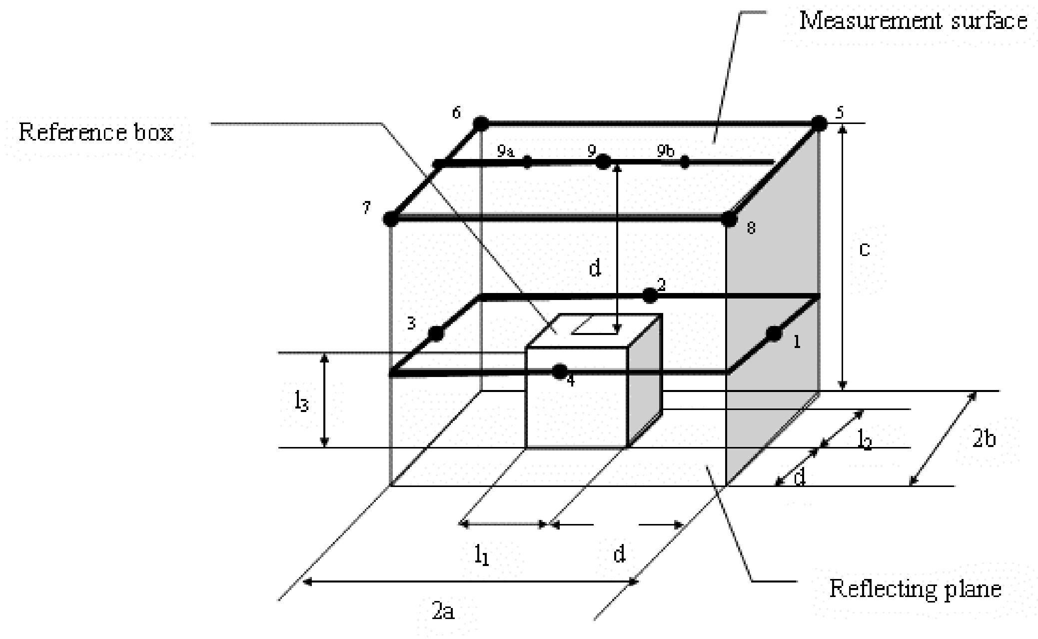


Figure 3: Microphone positions for sound power measurements in the semi-anechoic chamber at MWL, KTH.

The sound power measurement follows the ISO 3744 standard which is an engineering method to determine sound power levels using sound pressure data in an essentially free field over a reflecting plane. The microphone positions are shown in Figure 3. In order to avoid contamination from flow noise for microphone No 9, which was located directly in the outflow, it was replaced by two microphones (9a and 9b in Figure 3) on either side of it. Thus, in total 10 microphones were used. The calculation of sound power level L_w is given by the following equation

$$L_w = L_{pf} + 10\lg\left(\frac{S}{S_0}\right)$$

where L_{pf} is the energetic average of the sound pressure level at all microphones, S is the area of the measurement surface and $S_0 = 1\ \mathrm{m}^2$.

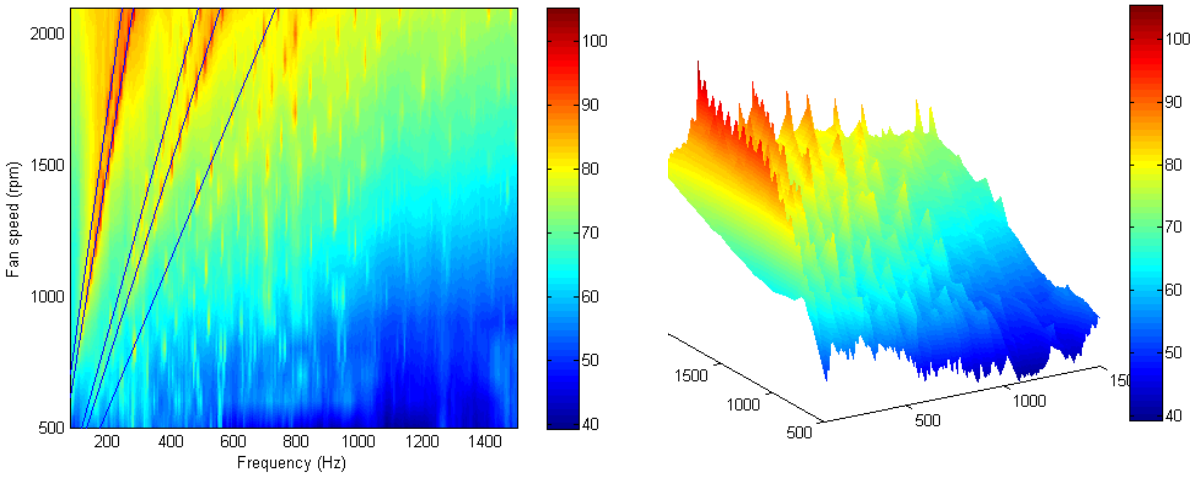


Figure 4: Waterfall and spectrogram plot of sound power level of cooling unit with one fan operating at speeds 500-2100 rpm. Five lowest fan and engine orders are indicated.

The waterfall plots concerning the sound power level generated from the cooling unit with one fan operating at different speeds are shown in Figure 4. The ridges in the waterfall plot are associated with specific fan and engine orders. The fan has 8 blades and the hydraulic engine has 7 pistons. In Figure 4 are inserted straight lines for the lowest three engine orders and the lowest two fan orders. It appears that the fan has most of its sound power concentrated at frequencies below 1 kHz.

As mentioned earlier the sound power from a fan is expected to vary as the rpm (N) raised to a power around 5. This implies a relationship of the form: $L_w=k+\alpha \lg N$ where L_w can be taken as the A-weighted sound power level, and α and k are coefficients to be fitted to the data. Figure 5 shows the relation between total A-weighted sound power and different fan speeds. The best fit for the fan tested here gave $\alpha=5.75$.

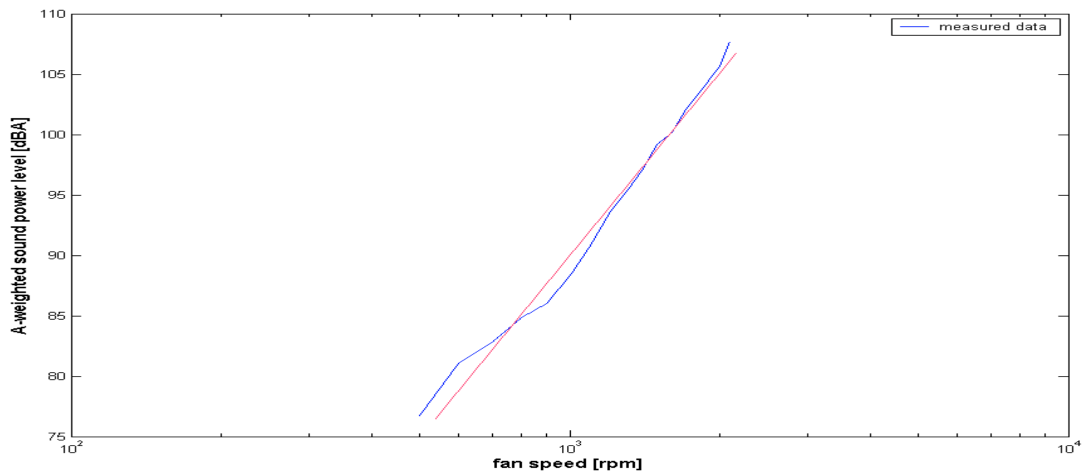


Figure 5: The relation between total A-weighted sound power level and the different fan speeds. The red curve (best fit) corresponds to $L_W \sim 5.75 \lg N$.

TRANSMISSION PATH ANALYSIS

To determine the acoustic response of the fan source and the effect of the enclosure for the cooling unit, i.e., the acoustic installation effects, frequency response functions (FRFs) can be used. Since the fan or its guide vanes generate sound via unsteady pressures (forces), the source character is of dipole type [3]. The noise radiated from the fan source is transferred to reach a receiver point in the far field. Here a small loudspeaker, see Figure 6, is used as a dipole source to obtain the acoustic FRF response of the actual fan source [6]. The goal of this FRF measurement is mainly to determine the FRF for a dipole source to pressure in the far field in order to see the acoustic loading created by the cooling unit. Therefore, the FRF in this measurement is defined as the relation between sound pressure $\hat{p}$ in the far field and dipole force $\hat{F}$ represented by a loudspeaker at a position as:

$$\hat{p} = H \cdot \hat{F}$$

where H is transfer function representing all transmission paths through the cooling unit. Since we are interested in the levels the square of this relationship expressed in dB, i.e., $10\log_{10}|H|^2$, is used below.

The frequency range within which the loudspeaker used can be accurately viewed as a dipole is from 250 Hz to 2500 Hz. It is also observed that the FRF shows no large differences between the two cases: with flow and without flow. This means that the measured FRF at the fan blade position at standstill will be close to the actual FRF when the fan runs.

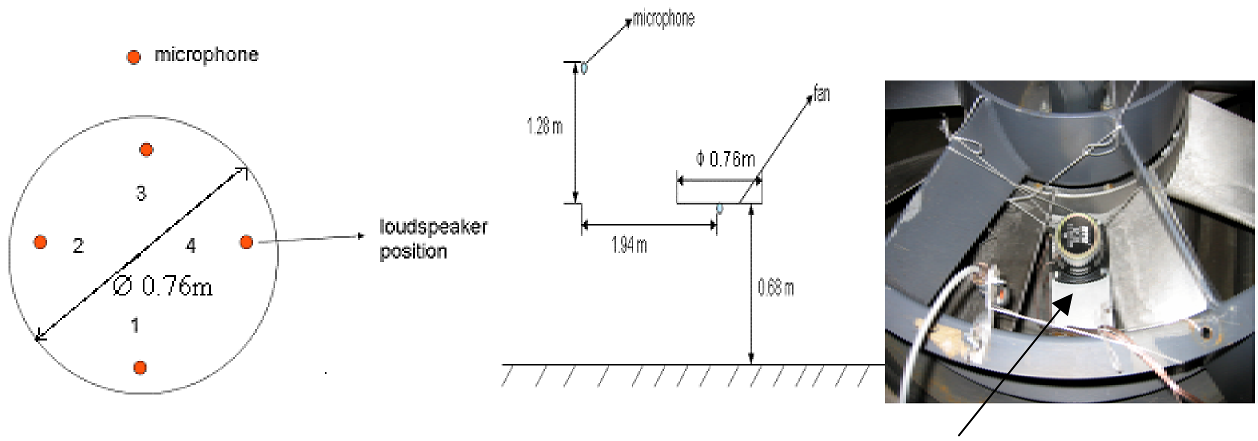


Figure 6: The schematic view of the FRF measurement setups with four loudspeaker positions at blade positions.

FRF measurements are performed in the semi-anechoic room with four positions of loudspeaker evenly distributed around the circumference and three directions (axial, radial and tangential) in order to more efficiently simulate a rotating dipole source [6]. A schematic view of the experimental setup can be seen in Figure 6. The microphone is kept at the same position for all measurements.

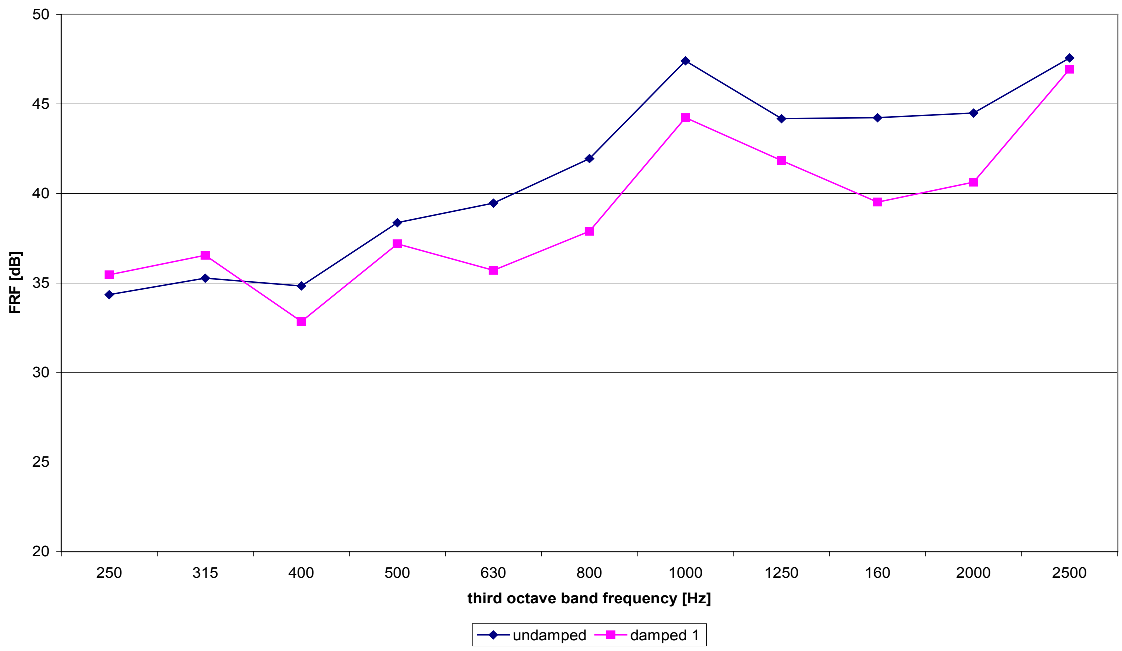


Figure 7: The comparison of FRFs between the undamped and damped enclosure.

To investigate the effects on the sound emission from adding damping treatments on the inside bottom of the cooling unit (see Figure 9a), transfer function measurements were performed to compare the difference between the two cases: with and without damping treatments. It is shown in Figure 7 that the damping treatment gives a reduction of the sound emission by 2 to 4 dB disregarding the region below 400 Hz where the wavelength is relatively large.

TESTS OF NOISE CONTROL TREATMENTS

An extensive test program was conducted on the cooling unit to evaluate different practical noise control measures. By examining the sound power spectrum of the baseline case it was found that a removal of all harmonic components from the fans and the hydraulic engine would give a reduction of 2-3 dB(A), which was considered insufficient. Therefore also the broadband noise needs to be reduced to give a substantial reduction of 5 dB(A) or higher. In total, nine cases were tested. The companion paper [1] describes modifications done on a single fan dismounted from the unit. Only two of these tests led to some reduction and these two are also applied here. .

The same setup and experimental procedure as for the baseline tests, described in one of the previous sections, was used. Due to the extensive number of cases only two fan speeds were considered: 950 rpm, which is the normal idling speed, and 1900 rpm, which is near the maximum. In order to reduce the otherwise very high volume flow one of the fans was deactivated and the opening was closed by a 50 mm thick sheet of absorption material to avoid short circuit of flow through it. Volume flow was measured with a hand-held scanning probe for the test cases where a changed pressure drop and thereby a changed volume flow could be expected. A wooden box (see Figure 8) was built and placed on the heat exchangers on one side to get a better controlled area to scan the flow. First a check was made of the symmetry between the two sides. For the remaining cases the flow was measured on one side only. No significant change in flow velocities was noted except from the case when the weather grills were removed. The measured flow velocities are listed in Table 2. The uncertainty in the measurements is believed to be of the order of ± 10%.

A short description of the cases tested follows in the numbered list below and some of them are also illustrated in Figure 9. The cases No 2, 5 and 8-10 relate to aerodynamic installation effects and No 6 and 7 to acoustic. No 4 it is related to reduction of structure borne sound and its radiation. Finally No. 3 is related to reduction of sound radiated from the hydraulic engine.

1. Baseline
2. "Velcro tape" also tested in [1]. A 30 mm wide Velcro tape is attached cylindrically along the inside surface of the stator in the same plane as the fan blades to decrease tip clearance.
3. "Hub covered" also tested in [1]. The outer hub is filled with absorption material up to the edge, and then a plywood plate, with the same diameter as the hub opening, is placed on top to reduce contribution from the hydraulic engine.
4. "Absorption outside" The space between two fan stators and between the stator and cooler end (on each side) is filled with absorption material.
5. "Grill removed". Weather grills are removed on both sides.
6. "Absorption inside + Cones". Bottom plate inside the cooling unit is covered by 60mm thick absorption material with 1mm thick perforate panel attaching on top. End wall inside the cooling unit on the operating fan side is covered by 90mm thick absorption material with corrugated (60mm thick) micro-perforated panel attached on it. Perforated cones filled with absorption material are mounted below each fan to guide the inflow.
7. "Absorption inside". As No 6 but without cones.
8. "Fan upside down". Fan is turned upside down to have an undisturbed inflow. The deactivated fan is closed by 50 mm thick absorption material to avoid short circuit of flow through the opening.
9. "Fan upside down + Cone". As No 8 but with the cone described under No 6 mounted below the fan.
10. "Fan upside down + Cone + Inflow disturbance". As No 9 but with a flow blocking strip mounted on the fan inlet in order to investigate the effect of inflow disturbances

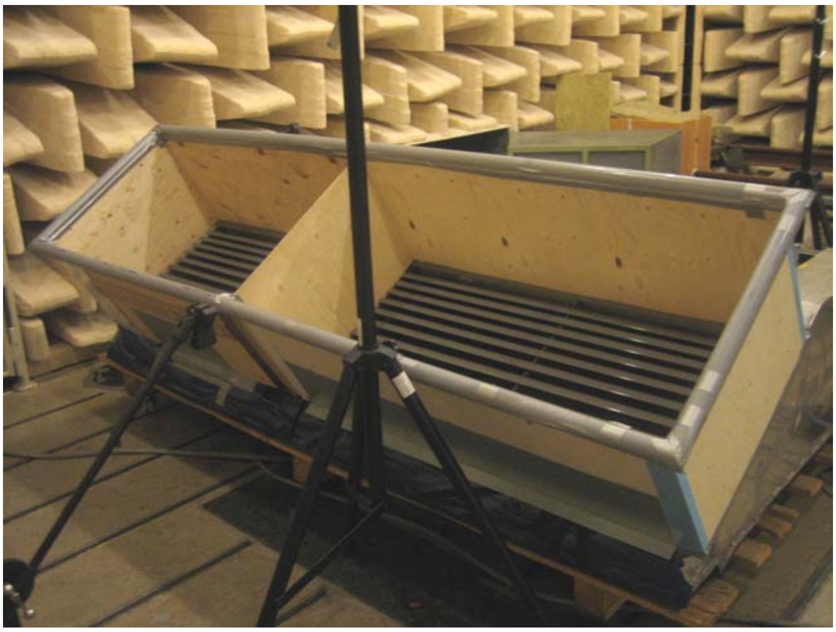


Figure 8: Box used for measurements of inlet/outlet flow velocity by hand held scanning hot wire probe.

(a) (b) (c) (d)

*Figure 9: (**a**) Inside of cooling unit with interior absorption and flow guide cones. (**b**) Upper side of cooling unit with one fan mounted in reversed mode. The other fan is deactivated and sealed off. (**c**) Velcro tape on stator circumference, (**d**) Shaded area shows where outside absorption was placed.*

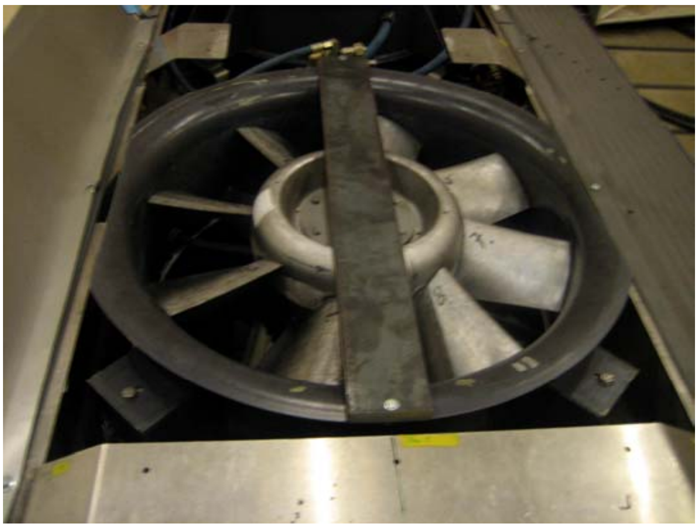


Figure 10: Strip (steel plate) placed on the fan in order to investigate the effect of inflow disturbances.

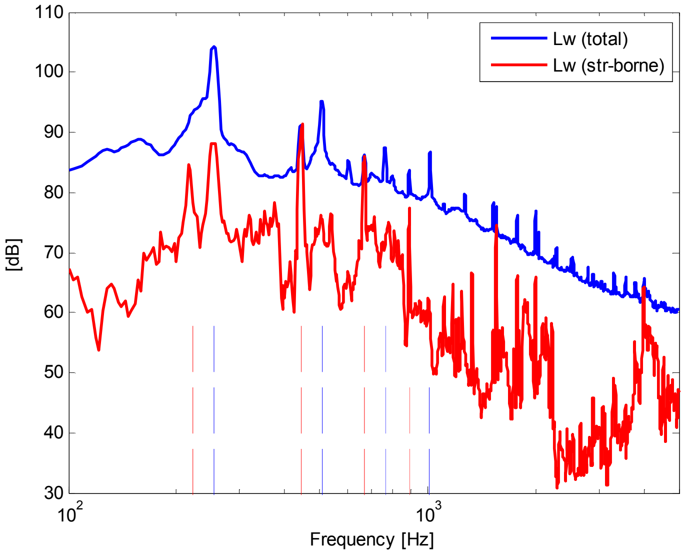


Figure 11: Separation of airborne and structure-borne contribution to sound power spectra.

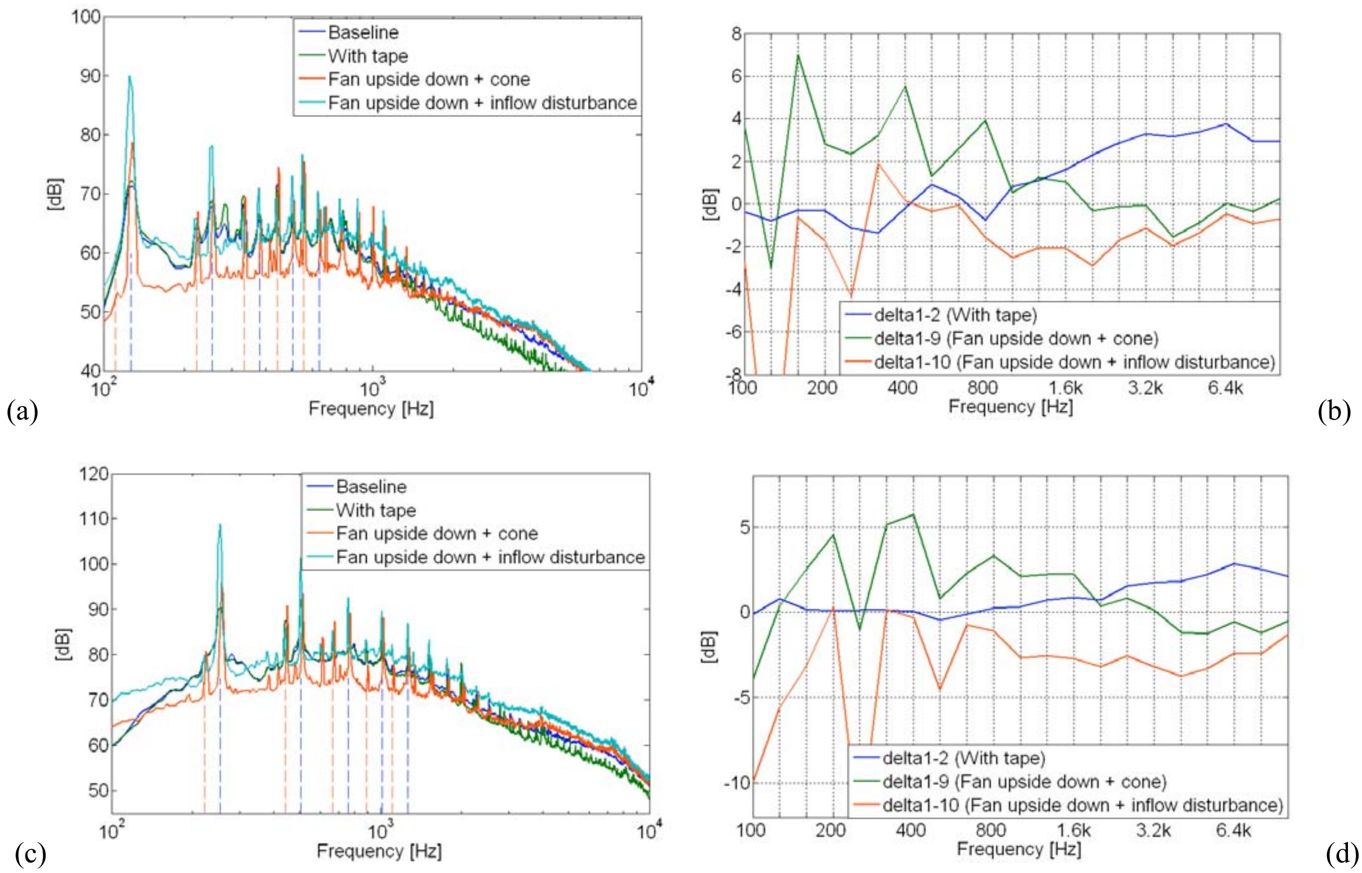


*Figure 12: **(a)** A-weighted sound power spectra with one fan running at 950 rpm for selected cases. Fan and engine orders indicated by red and blue vertical lines. **(b)** Sound power reduction in 1/3-octave frequency bands for the cases in (a) compared to the baseline case. **(c)** A-weighted sound power spectra with one fan running at 1900 rpm. **(d)**) Sound power reduction in 1/3-octave frequency bands for the cases in (c) compared to the baseline case.*

Table 1: Measured sound power level L_{wA}

	Short description	L_{wA}@950 rpm dB(A)	L_{wA}@1900 rpm dB(A)
1: Baseline	Original configuration	89.5	107.1
Case 2	Velcro tape	89.3	106.9
Case 3	Hub covered	89.1	107.1
Case 4	Absorption outside	88.6	106.7
Case 5	Grill removed	90.0	107.1
Case 6	Absorption inside + cones	89.3	106.3
Case 7	Absorption inside	88.8	107.0
Case 8	Fan upside down	87.9	106.0
Case 9	Fan upside down + cone	88.1	105.7
Case 10	Fan upside down + cone + inflow disturbance	95.5	113.3

Table 2: Measured volume flow Q

	Short description	Q@950 rpm (m^3/s)	Q@1900 rpm (m^3/s)
1:Baseline	Original config.	3.1	6.7
Case 5	Grill removed	3.5	7.1
Case 8	Fan upside down	4.1	7.6
Case 9	Fan upside down + cone	3.5	5.8
Case 10	Fan upside down + cone + inflow disturbance	3.2	6.6

Many of the cases tested had minor influence on the sound power. The most significant change came from the Velcro tape (for high frequencies) and the reversal of the flow direction (cases 8, 9) for mid frequencies. The inflow disturbance by the steel strip (case 10) gave a large increase of the blade passing frequency.

The identification of structure-borne noise generated by vibrations from the hydraulic engine was performed in two steps combining impact tests with running measurements. Transfer functions between hub vibration and acoustic pressure was measured and then signals from accelerometers at the hub with the fan running were used. The resulting contribution is plotted in Figure 11. It is seen that (as suspected) there a clear contribution to the peaks in the sound power spectrum related to orders of hydraulic engine but no contribution to the overall level.

MODIFIED FAN CONFIGURATIONS

The modifications to the cooling unit described in the previous section did not concern the design of the rotor or the stator of the fan itself. In the future steps of the project it is planned to include such tests. As an example, tests will be carried out with an increased number of stator vanes. According to the Tyler-Soffrin formula, the combination with eight rotor blades and seven guide vanes is far from ideal. It has been shown in calculations with the code described in [1] that a combination of 11 vanes and eight blades substantially reduces the amplitude of the dominating first two rotor harmonics compared to the existing configuration.

As a second case it is planned to replace the existing rotor with a rotor with curved blades. It is well-known that such rotors are less noisy than the standard rotors with straight blade edges [4,5]. The purpose of the tests will be to check if the installation effects are smaller for this fan than for the original axial fan.

A third case is to make a fundamental rebuild of the cooling unit by abandoning the axial fan concept totally and replace it by radial fans with vertical axes. This concept gives automatically a 90 degree shift in the flow direction and can therefore be better suited for the compact roof design required. A sketch of the principle is shown in Figure 13.

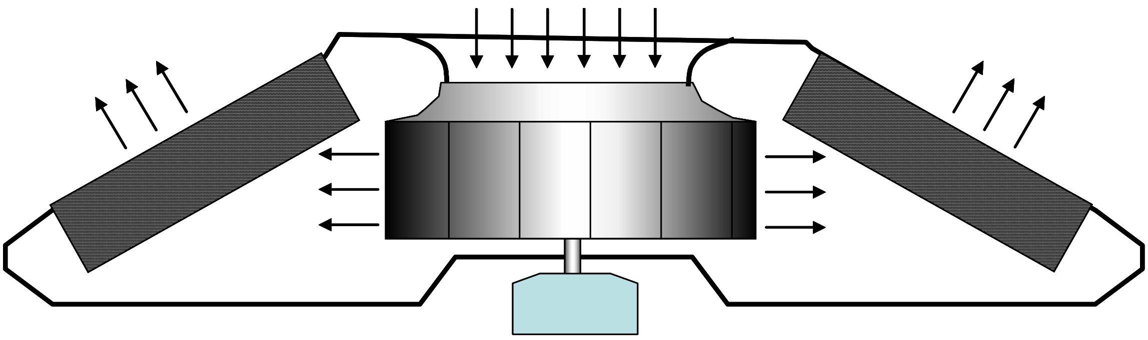


Figure 13: (a) Sketch of a proposed radial fan design concept.
Flow direction through fan and heat exchangers is indicated.

CONCLUSIONS

In addition to reducing the aerodynamically induced fan noise at a given flow speed it is, of course, of vital importance to also look into thermal management optimization. This means to bring down the cooling demand in general or to change the cooling cycles and make use of over-cooling when the train is running at high speeds, when other noise sources are dominating, in order to work up a thermal buffer that can be used when the train reaches the station areas so that the cooling can be reduced or even turned off. Another example of this type is to reduce the needed volume flow of cooling air by re-arranging the heat exchangers so that the high and low temperature circuits are placed in series instead of parallel. This complicates the construction for the heat exchangers somewhat but can lead to as much as 30% less cooling flow, which can directly be translated to a reduction in sound power by c:a 6 dB(A) assuming that the uniformity law holds. It can also be noted that arrangements where the fan units are pushing cold air through the heat-exchanger instead of sucking allows the fan to work with a smaller (cold) volume flow. This means that a lower fan rpm with less sound generation is possible for such configurations.

Lastly it is worth reminding that the findings reported here should be considered as intermediate project results and that final conclusions can not be drawn at the present stage.

ACKNOWLEDGEMENT

Support from Mr Kent Lindgren and Mr Danilo Prevelic at the Marcus Wallenberg Laboratory when carrying out the measurements is greatly appreciated. Mr Sönke Kraft at Bombardier Transportation assisted during some of the measurements and analyses. Funding was provided from the EU Commission under contract TIP4-CT-2005-516288.

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