Acoustic and thermal optimisation of systems including a fan and a heat exchanger

Yvon Goth,
Philippe Thoquenne,
CETIM,
Noise and Vibration Department,
52 avenue Felix-Louat,
BP 80067,
60300 Senlis CEDEX,
France

This study concern axial fans for the engine cooling systems of fixed or slowly moving devices, as earth moving equipment. Both acoustic and aerodynamic fan performances have been measured on a test rig, varying the fan blade geometry, the gap, and the distance between the fan and the heat exchangers. A specific installation effect parameter has been introduced to take into account the interaction between fan and heat exchanger, in order to be able to compute accurately the acoustical and aerodynamic performances of fan-exchanger systems.

The European legislation will constrain in a short term earth moving equipment manufacturers to decrease strongly the noise level of their machines, and measurement on existing systems have shown that the main contribution to the global noise level of the system is that of the cooling fan. On the other hand, the new low-emission engines tend to have a heat to power ratio that is higher than for older ones. Manufacturers will then face the challenge to reduce fan noise while improving the cooling efficiency of the engine cooling system.
In that context, the performances of some types of axial fan have been tested, in order to evaluate what could be the best compromise between noise and air flow for an axial fan associated to a heat exchanger.

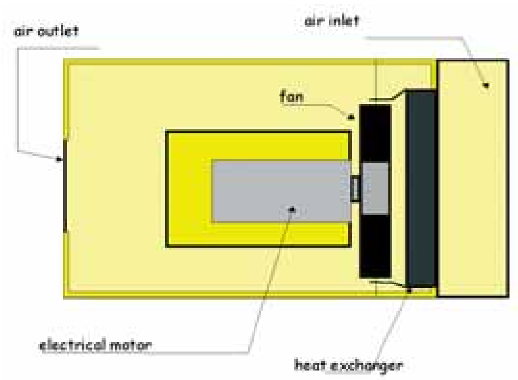

Fig. 1 : Test rig

Comparative test of fan

Test Loop

The test loop is given by a box-like structure, representative of the geometry of a real equipment.[fig 1]
From air inlet to outlet, the configuration is composed of :

- an air inlet, with adjustable opening areas,
- a heat exchanger,
- a fan
- an electrical motor, in a close box that represent the engine,
- an air outlet, with a circular hole of adjustable diameter.

Overall dimensions are 1m x1m x 2 m.

Noise measurement

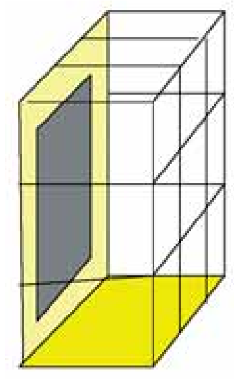

Fig. 2 : Integration grid for noise measurement

Noise measurement are made in the front part of the test box, by intensity method. Total power level is obtained by integration of 18 points over the following grid:[fig 2]

Air flow measurement

Air flow is calculated by integration of the air speed field in the front of the air exchanger. The air speed is measured with a small anemometer (1 cm diameter) in a 8 x 8 point grid.
Because of the difficulty to obtain accurate absolute measurement in that way, air flow must be considered only as a comparative data: different fans have been tested in the same condition and only their relative performances can be compared.
The total air flow is made to vary by changing the diameter of the outlet circular hole. Three diameters are used : 420 mm, 550 mm and 610 mm.

Fan

Three types of fan have been tested [fig 3] :

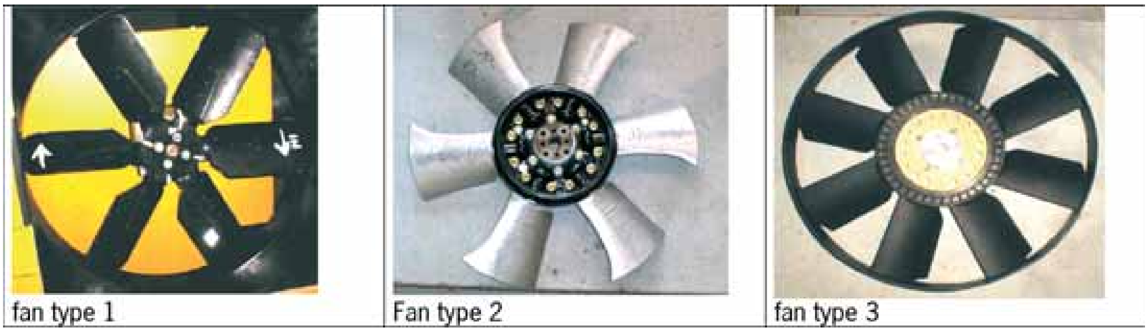

Fig. 3 : The three fan tested

All three fans have a diameter of 600 mm.
Fan 1 is a standard 6 blades fan of circular shape.
Fan 2 is a 6 airfoil shape blades with a possibility of angle adjustment. Test have also been conducted with 3 blades.
Fan 3 is a 8 blades fan with ring.

Tests results

About 100 test have been conducted in various configurations, with speed varying form 1500 to 2300 rpm.

The major observations are:
1 – When expressed in dB(A) scale, the fan tonal noise has roughly no influence.[fig 4]

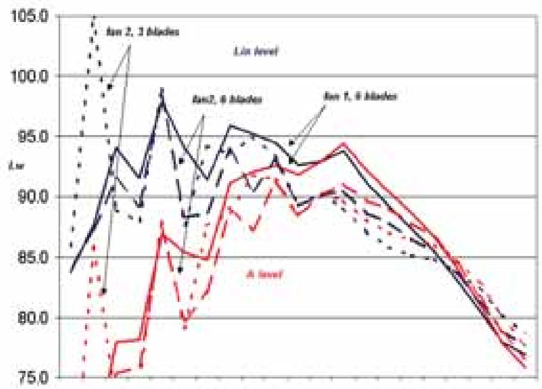

Fig. 4 : Comparative spectra in linear and A scale level

2 – Reducing the gap at the blade tip increases the air flow, as well as the noise level.[fig 5]

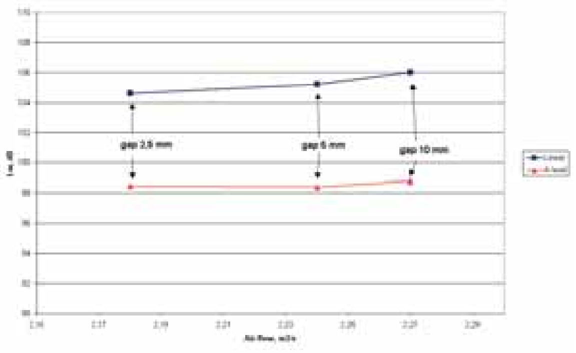

Fig. 5 : influence of gap on the air flow- noise curve. Fan 1

3 – Figures 6 represent the results of the measurements made with the three fans in different configurations. (varying speed, gap, and, for fan 2, blade number and blade pitch angle) For these points, the area of the air outlet is large (680 mm hole diameter), and the resulting global pressure loss in the system are small. In figure 6, dotted lines are for fan 2 with small blade pitch angles (12° to 18°).
If figure 7, the same results are presented for fans 1 and 3 with larger pressure loss in the system (outlet diameter 420 mm).
One can note that:

- there is an increase in noise level with small angles;
- the blade shape has not a strong effect on the airflow-noise curve;
- the fan with the ring generate increased airflow with lower noise, and the difference is greater in air circuit with high pressure losses.

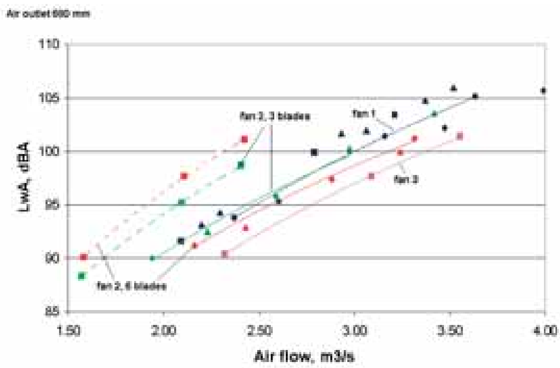

Fig. 6 : Relative performances of fans, low pressure loss

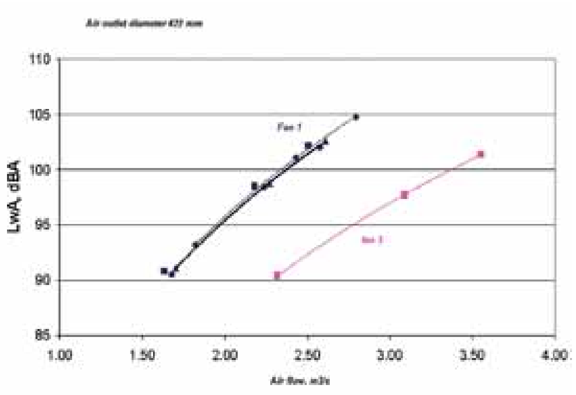

Fig. 7 : Relative performances, high pressure losses

Numerical models

Use of a correction parameter for air flow

A numerical model of the test rig has been created. In a first step, air flow is computed on the basis of the fan pressure–air flow curve and the pressure losses characteristics of the heat exchanger and the air inlet and outlet. With that model, the air flow is overestimated by 25%.

In fact, as far as the fan is close to the heat exchanger, air flow is not constant on the exchanger, and pressure losses varies along the distance from the centre. If the total air flow is calculated by taking in account the maximum pressure loss in the heat exchanger instead of the average, a better estimation of the flow is obtained:

$$\Delta P_{\max} = \Delta P_{avg} r^n$$

$$r = \frac{U_{\max}}{U_{avg}}$$

where U_{avg} is the average air speed and U_{max} the maximal air speed on the exchanger, and D_{Pavg} the pressure losses at the average speed, and n defines the dependency between air speed and pressure losses. For normal exchangers, it is close to 1.75.

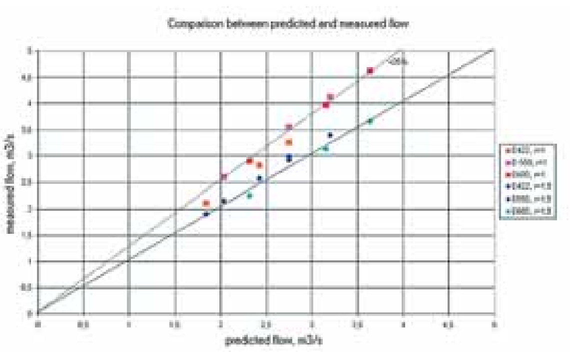

Fig. 8 : Effect of corrective factor on predicted air flow

The experimental value r=1.5 allows to fit well with the experimental data [fig 8]. Of course, the amount of data used to test this correction parameter is small, and it has to be checked whether it can be useful in others configurations.

Thermal and acoustic simulation of fan cooled systems in enclosures.

A thermal model of fan cooled heat exchangers systems in enclosures has been developed, linked to a software that allows to calculate the acoustic behaviour of enclosures, 'CETIM-CAPOT' [1][2]. In that model, the fan noise source model is based on the ASHRAE correlation, of the form [fig 9]:

$$Lw(f) = K_w(f) + 10\log Qv + 20\log \Delta P_T$$

where Kw(f) : specific power level by octave band, given for a fan type,

Qv : air flow, m³/s

ΔP_T : fan pressure, kPa

With a correction added at tonal frequency [3].

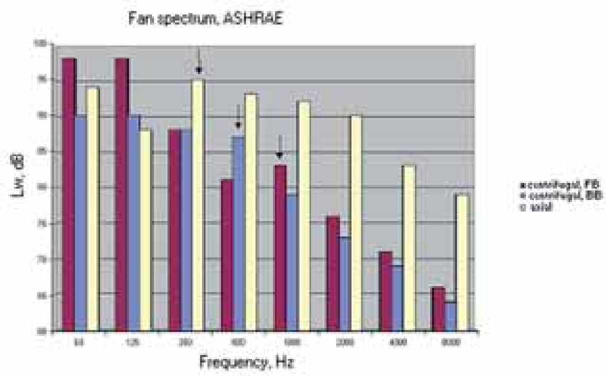

Fig. 9 : Typical specific power Kw(f)

When coupling thermal, aerodynamic and noise simulation, it is possible to find the optimal configuration (giving the minimal noise level for a given air flow).

For example, in some configurations one can obtained lower noise by having larger air inlet and outlet opening, because the lower pressures losses allows to get the same air flow with a lower fan speed.

In the following illustrative example, in order to get better cooling of a heat source under a housing, one wants to improve the air flowing over a heat exchanger in the wall of the housing, with the minimum over noise. The housing is 1m x 1m x 1m and the source is an engine with a total sound power level of 106.5 dB(A). The fan associated with the heat exchanger has a total sound power level of 100.4 dB(A) at 2000 rpm.

There are two options:

- increase the fan speed, with the same air outlet area
- increase air outlet area, with constant fan speed.

For the same increase in air flow, these two options have a different impact on the total sound power generated over the housing, as showed in the following table:

It is mandatory to introduce a correction parameter in order to properly compute the flow when the fan is close to the heat exchanger.

Configuration	fan speed	air outlet area	air flow	fan power level	**total power level**
base	2000 rpm	800 x 200 mm	2.31 kg/s	100.4 dB(A)	**100.8 dB(A)**
increase fan speed	2300 rpm	800 x 200 mm	2.74 kg/s	103.4 dB(A)	**102.6 dB(A)**
increase air outlet area	2000 rpm	800 x 420 mm	2.73 kg/s	100.4 dB(A)	**101.5 dB(A)**

Conclusions

In order to optimise the configuration of axial fans–heat exchangers cooling systems in practical cases, an experimental and simulation approach has given the following conclusions:
When associated with an heat exchanger, very different fan blades shapes and different configurations can have similar results when regarding the air flow-noise curves. In general, the noise level increases in parallel with air flow.
Fan with rings seems to increase both air flow and acoustic performances.

Finally, it is necessary to take into account both aerodynamic and acoustic behaviour of fan-heat exchangers systems in enclosures to be able to define the best compromise between noise level and cooling performances.

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