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THE PASSIVE FLOW AND NOISE CONTROL ELEMENT OF THE ELECTRONIC COOLING SYSTEM

T. Avikainen

Nokia Research Center, Itämerenkatu 11-13, FIN-00180, Helsinki, Finland

Tel.: +358-9-43761 / Fax: +358-9-4376 7290 / Email: timo.avikainen@nokia.com

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ABSTRACT
Fans are commonly used for enhancing the cooling of electronics (improve heat transfer due to higher air velocities). Dynamic losses will increase flow vorticity and the turbulence level due to the complex flow fields of electronic devices. The flow turbulence is strengthened further at the fans that increase the pressure drop. The airflow turbulence functions as a local noise source. Near to the fans there would be a great need for flow controlling elements. This presentation describes a principle that is beneficial in terms of a more effective airflow and a sufficient thickness of noise absorption material to damp even low frequency components of fan noise.

1 - INTRODUCTION
The flow fields of electronic devices are typically complex, airflow channels are narrow and unsmooth, also there exist changes at the flow channel cross section and other flow obstructions (changes of flow velocity and maybe direction). Laminar and highly turbulent regions can appear at the same time, although the overall flow velocity is relatively low. The acoustic noise of the devices must fulfil the environmental requirements specified at the international standards.

The narrowband part of the fan noise occurs at the blade passing frequency (fan speed times number of blades) and at its harmonics, since the periodic nature of the blade movement. Turbulent inlet flow will strengthen at the blades in which the thickness of turbulent boundary layer grows. As a result the discrete peaks of sound associated with the blade passing frequency bandwidths are widened (low frequency velocity variations). Also, the stochastic velocity fluctuations generate random blade forces, and, hence broadband sound spectrum [1]. In addition it has been presented in aeroacoustic studies that the broadband noise is dependent on the interaction of turbulence with the trailing edge [2]. The turbulence dependent fan noise can be for example controlled by applying a serrated trailing edge at the fan blades [3].

According to equation 1, the influence of the pressure difference is higher than influence of the volume flow rate. The right fan selection is important in order to locate the operating point at the optimum region of the fan characteristic curve.

$$L_w\,(\mathrm{oct}) = C + 10\mathrm{lg}Q + 20\mathrm{lg}\Delta P + \Delta dB \ \ (\mathrm{dB}) \tag{1}$$

where C is characteristic depending on the fan type, Q is the volume flow rate, ΔP is the total pressure increment and ΔdB term can include parameters of spectral correction for frequency bands, deviations from the optimal operation point, fan efficiency, fan tip speed (m/s) and so on.

The pressure loss of the system can be defined according to the equation (2) in which the first term covers the friction losses and later the single airflow resistances, the multiplier is the dynamic pressure [4]. The pressure loss curve is typically defined by CFD (Computational Fluid Dynamics) and measurements.

$$\Delta p = \left(\frac{f}{d_h} \cdot L + \sum \xi\right) \cdot \frac{1}{2} \cdot \varphi \nu^2 \tag{2}$$

where f is dimensionless friction factor (for laminar flow is β/R_e in which β is constant depending on the shape of the cross area and R_e is Reynolds number), d_h is hydraulic diameter (mm), L is duct length (m), $\sum\xi$ is the sum of the single airflow resistances, φ is density (Kg/m^3), v is velocity (m/s).
Specially at the cases of axial fans any throttle of the inlet flow must be avoided, because of broadband noise increasing effect of stalling phenomena in which the angle of incidence increase too much at the fan blades. As a result the volume flow rate also decreases.

2 - SMOOTH DESIGN

The airflow eddies and turbulence of the cooling system is worth to be minimized and with that the pressure losses, especially at the inlet side of the fan by smooth design in terms of fluid dynamics. As a result of that acoustic energy is more concentrated at the discrete components. Often at the cases of complex airflow channels it is possible to use effective absorbing materials also at the low frequencies to smooth the discrete blade passing components. The noise reducing effect of absorption material is naturally clear at enough high frequencies, but a non-effective area can take place between the presented frequencies depending on the case. A thin resonator film/scrim at the surface of the absorption material can be used to control the best damping frequency area in relation to the sound source spectrum. Another way to influence the damping of noise would be to increase the sound reflections by using sudden changes at the cross section of the flow channel, but this increases the flow resistance.

3 - TESTCASE AND RESULTS

The testcase consists of the cover that prevents the sound waves moving directly from the fan to the outside surroundings (fig. 1a). The modified cavity (fig. 1b) was filled with relatively heavy polyether foam (70kg/m^3). The foam surface was coated by the scrim. In the middle of the bottom plate, foam was used at the design of passive flow control element.

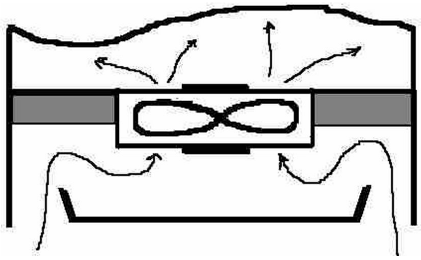

Figure 1(a): The original air cavity.

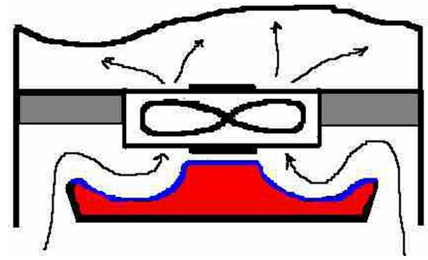

Figure 1(b): The modified air cavity.

The noise measurements consisted of the sound pressure measurements by one microphone at the distance of 1m from the system, the sound power of the total system was measured according to ISO 3744 standard. The fan operation point was controlled the same with the real testcase by using a pressure chamber. A-weighted sound pressure spectrums are presented in figure 2.
Comparative study was done by 2-dimensional axisymmetrical Finite Element (FE) models. Sound pressure spectrum of the fan (corresponding to the operation point) was used as noise source at the models. The boundary condition of the inlet hole of the system was taken into account by modelling part of the outer surroundings and by applying Sommerfeld's infinite boundary condition at the outest circle. Calculated sound pressure distributions are presented in fig. 3.
The total sound power reduction of the system was 1 - 4 dBA depending about the fan rotation speed in spite of relatively thin absorption element. In addition the volume flow rate increased about 10 - 15 %. By reducing the fan speed corresponding to the increased volume flow rate, about 2-3 dBA extra noise reduction can be achieved. According to the FEM-calculations the modified air cavity has its first resonance at clearly higher frequency (~1600 Hz) compared to the original air cavity (beginning from 1 kHz). This naturally guarantees more effective noise reduction by used relatively thin foam thickness. Roughly 1 dBA noise reduction can be achieved by coating simple 10mm foam at the fan cover.

4 - CONCLUSION

The decreased turbulence level and smaller pressure losses can be achieved by more ideal shape of the airflow channel. This also reduce the noise due to the decreased fan loading and turbulence noise by emphasizing the blade passing spectral components that are relatively easy to kill by even thin absorption material. The system space can be used more effectively in terms of thicker absorption material. Preform - molded absorbers are easy to mount at the right place.

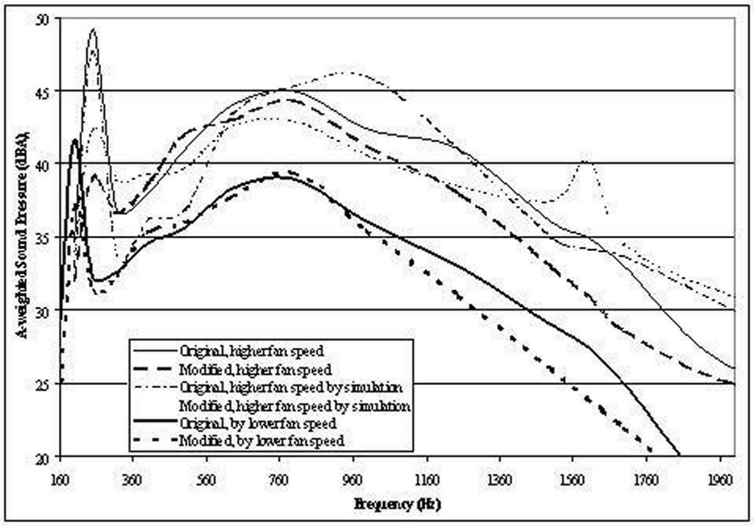

Figure 2: The measured A-weighted sound pressure spectrums.

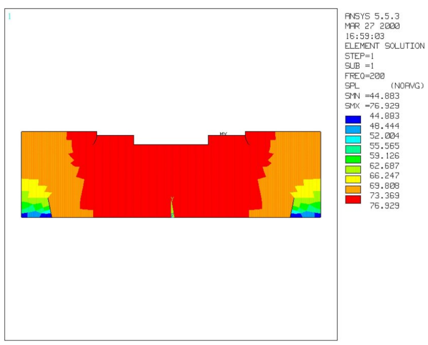

Figure 3(a): Original system.

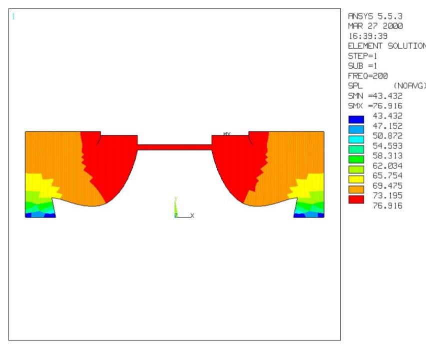

Figure 3(b): Modified system.

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