THE MINIMAL NOISE CONCEPT AND ITS APPLICATION

T. SZENTMÁRTONY

Department of Fluid Mechanics,
Technical University, H-1521 Budapest

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Summary

One may define a certain amount of noise which is inherent in the fundamental work process and should be called "Minimum Noise". Single rotor axial flow fan's unavoidable noise source is the vortex shedding in the wake. Machines, using impact processes, to reshape workpieces, will show the so called acceleration noise of the hammer as unavoidable noise source.

Introduction

With the world wide introduction of limits on the noise in factories, entertaining centers and homes, purchasers of any noise producing machine demand less and less emission. It is frequently due to the customer's ignorance in acoustics that impossible requirements are set and ignorance on the manufacture's side to accept same. It happens very rarely that scientifically correct measurement justifies subjective judgement, hence commercial life relies on adjectives like: noiseless, quite etc. No wonder that not long ago a European country in which such adjectives meant, — due to legislation — a certain sound pressure level in well defined acoustical environment, checked the vacuum-cleaners on sale and found that 72% of them did not fulfil the relevant requirement. The blame is always put by both sides on the machine designer. However, unless acousticians provide a proof of feasibility of the required rate of noiselessness, all parties involved will continue to ignore limits or will not rack their brains to evade the use of the particular machine by changing technology, for instance.

The concept

Noise is not a veritable disaster and in most cases can be avoided to a certain limit by skilful design. It is therefore always important to know how much noise arises from design faults or material and economic limitations, not related directly to the work process.

On the other hand one may define a certain amount of noise which is inherent in the fundamental work process and should be therefore called: "Minimum Noise." The minimum noise is then the lowest level of radiated acoustic energy which is unavoidable if the requirements fixed for the designer are set. Once the minimum noise of a machine has been worked out, different designs can be evaluated on the basis of the actual noise emitted as compared to the calculated minimum one.

The next problem is the form in which the minimum noise should be rated. Some countries consider the peak acoustic pressure, hence the maximum level as the reference value, others insist on the daily dose of A-weighted noise energy. No other way seems to lead out of this dilemma as the way of considering always the relevant or legal limit as reference.

The minimum noise of axial flow fans

Axial flow fans used in airconditioning, ventilating or cooling systems are judged partly by their disturbing noise.

The usually continuous spectrum within certain cases dominating pure tone peaks are due to aerodynamically generated noise, radiated by several kinds of sources as follows:

Pre- and post-rotators serve to increase the static pressure rise of the axial flow fan. Let us assume that they are not required — as it is by many commercial low pressure fans — hence the so called potential and wake interaction noise does not enter into the picture. The single rotor may be still influenced by flow disturbances of struts supporting the center body of the machine. If, by a special construction struts also may be avoided (Fig. 1), only the turbulence remains as a sound source at the inlet.

Concerning the rotor, three sources should be examined. One of the three, the rotational noise — investigated by Gutin [1] — is considered as an impulse caused by blade thickness and the blade lift, assumed being constant to an observer moving with the blade. However, since axial flow fans working in airconditioning systems run usually with a tip speed of $u < 80$ m/s, far from the velocity of sound, the local hydraulic short-circuiting in the near field is strong, hence the radiation efficiency of this kind of sound source is very low. Calculations showed and experiments proved that its contribution to fan-noise can be neglected.

Oncoming turbulence may cause unsteady lift forces which in turn produce broad band noise. To avoid it, a bell-shaped inlet should stretch the

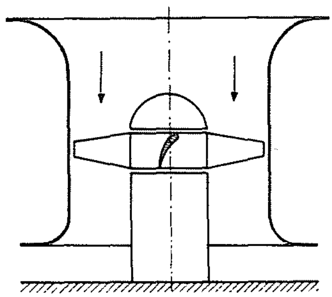

Fig. 1

turbulent eddies decreasing so the tangential intensity considerably and the axial one slightly. Even the thin boundary layer will not represent a dominant sound source.

Investigations have shown that the discrete frequency noise of a single rotor can be reduced to a level insignificant relative to the broadband vortex-shedding noise level from the rotor blades [2].

The so called vortex-shedding is due to the different flow velocity, leaving the trailing edge from the upper and lower side of the profile. The eddy wake caused by the velocity gradient generates in turn circulation, hence pressure fluctuation on the blade. The wake field, represented by an acoustic quadrupole field, hence triggers off a dipole source on the solid surface of the blade.

It may be noted here that the broad band vortex-shedding noise level can be predicted and has been proved by experiments as well [3].

Technically it is possible to design an axial flow fan which produces only a broad-band vortex-shedding noise which, as indicated, originates from the unavoidable wake caused by the velocity difference between the upper and lower surface of the blade. This difference however produces the lift and thus the pressure rise across the blade row which in turn is the goal for operating a fan. It may be certainly possible to remove the blade wakes by suction or blowing from the trailing edges, however the price and the reliability of such a construction is at present questionable.

We have reached the point where — excluding the possibility of wake manipulation — the source, from where the unavoidable noise will be radiated, has been found. This noise therefore can be considered for most of the commercial fans as minimum noise.

It was possible to build up a simple formula — based on fundamental aeroacoustic findings [3], [4], [5] — offering minimum sound power levels for the above mentioned "rotor only" axial flow fans [6], [7]

$$L_{W\,\min}=27+10\lg \dot{V}+23\lg \Delta p_t \quad [\mathrm{dB}]$$

where $\dot{V}$ [m^3/s] is the flow rate and Δp_t [Pa] the total head rise.

Concerning the emitted sound power level of an axial flow fan the widely used empirical formula is as follows:

$$L_W=40+10\lg \dot{V}+20\lg \Delta p_t \quad [\mathrm{dB}]$$

in which the constant may vary ± 4 dB according to experience. Taking the minimum $40-4=36$ compared to 27 seems to be quite a considerable difference. However, it should be noted that the multiplier of Δp_t is 23 instead of 20. This will compensate the 9 dB difference in the constants since e.g. taking $\Delta p_t=500$ Pa — a very common case — would immediately mean $3\lg 500=8.1$ dB.

Impact processes

Apart from aerodynamically generated machine noise, machines in the field of production are — generally speaking — reshaping the workpiece and radiating also considerable noise. The energy required for this process is usually built up over several hundreds of milliseconds and released for the process in microseconds.

The energy available is usually far more than the process needs, hence the excess is transformed into vibration. Vibrational energy finding a good radiator on any part of the machine will then be changed into radiative energy called noise.

Impact processes are numerous. Forging, stamping, riveting, sawing, planing, textile operations but also reciprocating engines, pneumatic drills or even the backlash in different mechanisms represent impact processes.

The noise generated arises from several sources. The hammer, anvil, punch, saw tooth etc. accelerates and decelerates on impact causing pressure perturbations, radiating pulse like sound, called acceleration noise. Immediately after the impact the workpiece changes shape as well, causing again a pressure perturbation, the so called billet noise. It is then followed by the vibrational noise from the workpiece due to the excess energy not absorbed by the material forming process. Some of this energy flows into the supporting structure and to the floor which then radiates sound, called ringing noise.

Having described the sound sources, let us see which of them can be — even if only theoretically — avoided. It seems that if there is no excess energy, no vibration will start, hence the so called ringing noise can be avoided. Practically it shows that not only the minimizing of the excess energy, but its absorbtion or prevention of its radiation is the main job of the designer.

There is however the acceleration noise which, if any solid body is accelerated or decelerated, will be more or less generated. It is a physical fact, hence it cannot be avoided. If so, we have reached the cause of the minimum noise, and it would be advantageous to know how strong does it depend on the various factors.

Longhorn [8] has calculated the work needed to accelerate slowly in an inviscid fluid a rigid sphere from rest to a velocity c_0 in time t_0. He found that over and above the expected 1/2 m c_0^2 where m is representing the mass of the sphere an additional 1/2 m*c_0^2 kinetic energy was required in which m* stands for the mass equal to one half that of the air displaced by the sphere. If no compressibility of air occurs this extra amount of energy may be recovered when a slow deceleration is secured. If however a fast acceleration occurs, so compressibility starts to play a role and an additional amount of energy is needed which must take the form of radiated sound. Thus the maximum of the radiated energy when halting a sphere in a very short t_0 time is $1/4\rho_0(V)c_0^2$, which is the available 1/2 m*v_0^2 extra energy travelling with the sphere. V stands for the volume of the sphere and ρ_0 represents the density of the air. Since usually at an impact one body is decelerated while the other is accelerated, it sounds reasonable to assume that the maximum value for the acoustically radiated energy must be $1/2\rho_0 Vc_0^2$. Taking it as a reference the acoustic efficiency

$$\eta_{\text{acc}} = \frac{E_{\text{acc}}}{\frac{1}{2}\rho_0 V c_0^2}$$

will tend towards one, if E_{acc} denotes the radiated energy. Note that it is very useful to know that since $\rho_{\text{steel}}/\rho_0 \approx 1.5 \cdot 10^{-4}$ the sound energy will never exceed $1.5 \cdot 10^{-4}$ times the kinetic energy of the impactor.

The general relation between peak sound pressure and impact duration (the actual contact time) t_0 was given by Koss and Alfredson [9] which smoothly closes to Longhorn's results when low frequency approximation is applied. Richards et al. [10] based on the above mentioned theories worked out practical expressions supported by experimental evidence. Proving that there is a good chance to replace the radius of the spheres by the actually accelerated body's $(\text{volume})^{1/3}$, the undimensional form of the $\hat{p}$ peak pressure at a distance r with "a" denoting the velocity of sound will be

$$\frac{\hat{p} \cdot r}{\rho_0 a c_0 (\text{vol})^{1/3}} = 1.5 \sqrt{\{a t_0/(\text{vol})^{1/3} + 4}$$

Since experimental points — containing colliding cylinders, cones etc. hence far from being spheres — lie under these values, a simple upper limit is indicated by

$$\frac{\hat{p} r}{\rho_0 a c_0 (\text{vol})^{1/3}} = 0.7 \qquad \text{for} \quad \frac{a t_0}{(\text{vol})^{1/3}} = \delta \leq 1$$

$$\text{and} \quad 0.7\delta^{-2} \qquad \text{for} \quad \delta > 1.0.$$

To change this result into a minimum peak sound pressure one has to rewrite the expression in the form

$$L = 10 \lg \frac{\hat{p}^2}{(2 \cdot 10^{-5})^2} \quad [\text{dB}]$$

But since the legal limits are marked in dBA the A-weighting has to be applied. Following Koss and Alfredson's suggestion the peak amplitude of the pulse may be expected at $f_A = 76.1/R$ for colliding metal spheres with radius R, which is the same as $f_A = 1/2t_0$. With this, the A-weighting correction read at f_A and marked by C_A [10]:

$$\text{for} \quad \delta \leq 1$$

$$L_{\min} = 117 - 20 \lg r + 20 \lg c_0 + 7 \lg(\text{mass}) - C_A \quad [\text{dBA}]$$

in which the volume in the previous expressions have been replaced by the mass in kg-s of the impactee or impactor. Hard blows ($\delta < 1$) may be calculated in a similar way by

$$L_{\min} = 117 - 20 \lg r + 20 \lg c_0 + 7 \lg(\text{mass}) - 40 \lg \delta - C_A \quad [\text{dBA}]$$

Following the train of thoughts one may soon reach conclusions. Considering the usual distance between the workpiece and worker at the moment of contact r, the impulse (mc_0) required by technology, and over and above forseeing die to die hard blows ($\delta = 1$), every pair of m, c_0 will represent an L_{min}, hence a peak sound pressure level which cannot be avoided and should be called therefore the minimum noise of the machine.

It may be useful to draw attention to the importance of c_0 in the expressions. At a required impulse (mc_0) large hammer mass (m) should be preferred to high impact speeds. Similar minimum levels may be obtained for the daily equivalent levels $L_{eq\,min}$ as well, if process repetition rates and the rate of hard and soft blows are known or estimated.

Should the acceleration noise be kept at a low level the dominating machine noise will be the ringing noise of the machine. The 1–2% of the kinetic energy not absorbed by the forming process but changed into flexural vibrations may find highly radiating machine parts causing sometime even 30 dBA higher noise than that of acceleration. There is no minimum noise level to be accepted in this field, since — even if only theoretically — excess kinetic energy should not exist or should be absorbed before being radiated.

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Prof. Dr. Tibor SZENTMÁRTONY H-1521 Budapest