THE INFLUENCE OF THE UNSYMMETRIC ALTERNATOR FAN
BLADE-SPACING TO THE TOTAL SOUND PRESSURE LEVEL AND SPECTRA
OF AERODYNAMIC NOISE

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1. INTRODUCTION

Demand for smaller and more efficient alternators forced designers to place
two radial cooling fans inside the casing which improved cooling. Due to this
change the aerodynamic noise that is dominant at higher rotating speed of
rotor increased.

In our work the sources of the aerodynamic noise of alternator were
identified and the influence of irregular blade-spacing on total sound
pressure level (SPL) and spectra was theoretically and experimentally
analysed. According to the theoretical results various fans with irregular
blade-spacing were manufactured and measurements of total SPL and
spectra of radiated noise were performed.

2. THEORETICAL APPROACH

The forces exerted by the air flow on the fan blades and casing are the
primary cause of the noise radiated from fans with low blade-tip mach
number [5]. Forces with periodic time history lead to discrete sound
components and random forces are responsible for the broadband noise.

Assuming blades are thin, expression (1) describes sound pressure
radiated by a point blade force $T_j$ acting at the aerodynamic center of blade
(Fig. 1) [2,3].

\begin{equation}
\end{equation}

Fig. 1: Radial fan blade model
\[ 4\pi(p - p_0) = -\frac{\partial}{\partial x_j} \left[ \frac{T_j}{r|1 - M_j|} \right] \]  

(1)

Defining the fluctuating force \( T_j \) by a Fourier series and adopting the theory describing the discrete frequency sound radiated by axial fans to the radial fans, expression (2) for calculating the complex magnitude of sound pressure of the \( n \)-th harmonic radiated by a single blade was obtained \([4]\).

\[ c_n = -\frac{\text{imag}}{4\pi r_0^2} e^{\frac{n(\pi + \frac{\pi}{6})}{n^2}} e^{-i\omega t} \]

\[ \sum_{n=-\infty}^{\infty} T_j (i)^{-\lambda} \left[ i \cos \beta_s (J_{n-\lambda+1}(nM) - nM) \right] + 2 \sin \beta_s \frac{n-\lambda}{nM} J_{\lambda-\lambda}(nM) \]

(2)

If variations of fan blade spacing angles \( \phi \) are small, blade size and shape changes can be neglected. Therefore all blades radiate much the same but phase shifted sound fields. Expression (3) describes the interference of sound fields radiated from blades \([1]\) where \( B \) stands for the number of blades.

\[ F(\phi_n) = 1 + e^{i\phi} + e^{i(\phi + \phi_1)} + \ldots + e^{i(\phi + \phi_1 + \ldots + \phi_k)} \]

(3)

Discrete frequency sound pressure radiated from radial fan is given by the equation (4) from which the expression for sound pressure level can be derived.

\[ p(x, t) = \sum_{n=-\infty}^{\infty} c_n(x, t) F(\phi_n) e^{-i\omega t} \]

(4)

3. EXPERIMENTS

The measurements of total SPL and spectra of noise radiated from various fans were performed in an anechoic chamber with following dimensions: \( W \) 1.70 m, \( L \) 2.60 m and \( H \) 2.65 m. Alternator was belt driven by the electromotor and the microphone was mounted in the plane of fan, 0.50 m away from the axis of rotation.

4. COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS

Radial cooling fans are made of one millimeter steel-sheet. Fans that are currently being manufactured have 12 equally spaced blades and are named FAN-R (Fig 4a). In order to avoid unbalance first studies of the influence of irregular blade spacing on total SPL and noise spectra were performed on symmetrical fans - FAN-1 (Fig. 5a).

Definition of blade force with periodic time history

Since fan is mounted inside the casing the flow over the blades is unsteady causing fluctuating blade forces. Presumed periodic blade force time history is shown in figure 2.
According to the calculated results we found that sound pressure magnitude depends primarily on the amplitude of the blade force fluctuation \( \frac{1}{2}(T_B - T_a) \) and that the relative relations of calculated sound pressure for various fans are equal in wide area of changing the amplitude. Therefore the desired fan blade spacing can be obtained without knowing actual blade force magnitude [4].

**Total sound pressure level**

Measured values (Fig. 3a) and relative comparison of calculated values (Fig 3b) for various fans shows that total sound pressure level doesn’t depend on fan blade spacing what matches with results found in literature [5].

**Noise power spectrum**

In power spectrum of fan with equally spaced blades FAN-R there is strongly represented sound pressure at 12th harmonic (Fig. 4b and c) that correspond to the blade passing frequency. Using fans with irregular blade spacing FAN-1 radiated sound energy is spread over several harmonics (Fig. 5).

Due to the prospective results in reducing the siren effect the fan with unsymmetrical blade spacing FAN-O whose sound power is spread over
many harmonics was designed (Fig. 6). Unbalance of fan was reduced with minor decrease of lengths of some blades.

![Diagram of fan](image1)

![Graphs of spectra](image2)

**Fig. 5:** a) drawing, b) measured and c) calculated spectra for FAN-1 at 12000 min⁻¹. Amplitudes not to scale.

**Fig. 6:** a) drawing and b) calculated spectra for FAN-0 at 12000 min⁻¹

### 5. CONCLUSIONS

Linear acoustics formulas for calculation of radial rotating blade noise were derived on the basis of the acoustical analogy. Theoretical and experimental results confirm that fluctuating forces exerted by the unsteady flow on the fan blades and casing are the primary cause of the noise radiated from radial fan. By irregular fan blade-spacing the blade passage sound is spread over a wider frequency band. This is desirable from a subjective point of view but the total sound power level remains unchanged. Unbalance can be simply reduced by changing blade lengths.

**References:**


