SOUND GENERATION AND PROPAGATION IN A SYSTEM CONSISTING OF TWO PERIODIC ROWS OF CHANNELS

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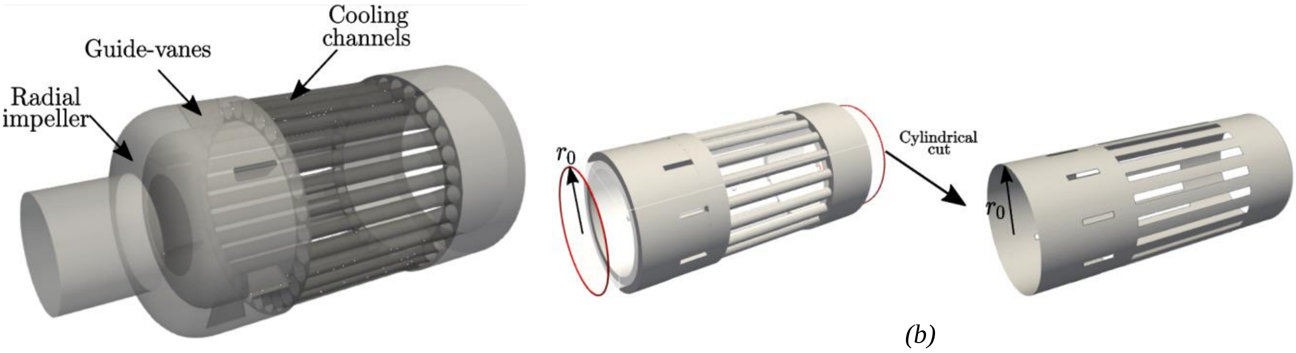
SUMMARY

The paper is dealing with the analytical modelling of sound generation and propagation in a system consisting of guide vanes and cooling channels, in order to improve the design of the cooling fan system integrated in traction motors. This system can be considered as two periodic rows of channels separated by a small distance. The first mechanism investigated in this study is the sound generated by the impingement of the wakes of an upstream impeller with circumferentially uneven blade-spacing on the guide vanes. The second one is the transmission of the acoustic waves generated by the latter through a row of thick-walled channels. The influence of various parameters on sound generation and propagation is discussed in this work.

INTRODUCTION

The major part of the noise radiated by traction motors used in railway applications at high rotation speeds is related to the ventilation system. The latter is used to maintain the machine components to allowable operating temperatures. A typical configuration of the ventilation system is shown in Figure 1a. It consists of a radial impeller, guide vanes and cooling channels. The impingement of the impeller wakes on the guide-vanes generates a noise at the blade passing frequency and its harmonics. The use of fans with uneven blade-spacing allows to reduce the annoyance related to this noise. But it induces new frequencies which are multiple of the rotational frequency. This noise is transmitted through all components of the ventilation system and it can be amplified by the presence of the cooling channels. This occurs when the rotational frequency harmonics coincide with the resonance frequencies of the cooling channels. It is therefore important to be able to predict these frequencies at the early design stage in order to avoid them.

Numerical simulations to predict the aerodynamic noise become more and more affordable in industrial context, especially with the growing use of the lattice Boltzmann method (LBM). Yet they can be expensive when repeated computations must be performed, typically to investigate variable configurations involving numerous parameters. The key issue is that all geometrical details, needed for the simulations, are often not available at the early design stage. In contrast, analytical approaches are well suited and attractive at that stage, because they rely on a simplified geometry and perform with very low computational times. The present methodology is to split the ventilation system into generic components addressed separately. The present work is focussed on the system consisting of guide vanes and cooling channels (see figure 1b). The model developed in this work is the combination of two sub-models. The first is devoted to the sound generated by the impingement of vortical waves on the guide vanes by using the models suggested in the literature [2,3,6]. The second one, is used for the transmission of the acoustic waves generated by the first model through a row of thick-walled channels. This model has been developed in a previous work [1] using the mode-matching method. In practical applications, the flow inside the ventilation system has a low Mach number (M<0.1), , and its effects on sound propagation are neglected in this study.



(a) *(b)*

Figure 1: (a) Cooling circuit of a traction motor. (b) Typical configuration of the ventilation ducts used in electrical machines.

The first part of this work is devoted to the mathematical formulation for the sound generation and propagation mechanisms. The results obtained are compared with the finite element method in order to validate the analytical model. The influence of physical parameters is discussed in the last part of this paper.

MATHEMATICAL FORMULATION

Two mechanisms are investigated in this paper. The first is the sound generation by the impingement of the impeller wakes on the guide vanes, and the second one is the sound propagation in a system consisting of guide-vanes and cooling channels. A cylindrical cut of the ventilation system at the radius r_0 is shown in figure 2a. The unwrapped representation of the later is shown in figure 2b. It consists of two periodic rows of channels separated by a distance d. The vanes and channels walls have respectively thicknesses b_1 and b_2 and are assumed to be perfectly rigid. The vanes and channels widths are given by: $a_1 = 2\pi r_0/V_1 - b_1$, $a_2 = 2\pi r_0/V_2 - b_2$, V_1 and V_2 being the numbers of vanes and channels respectively.

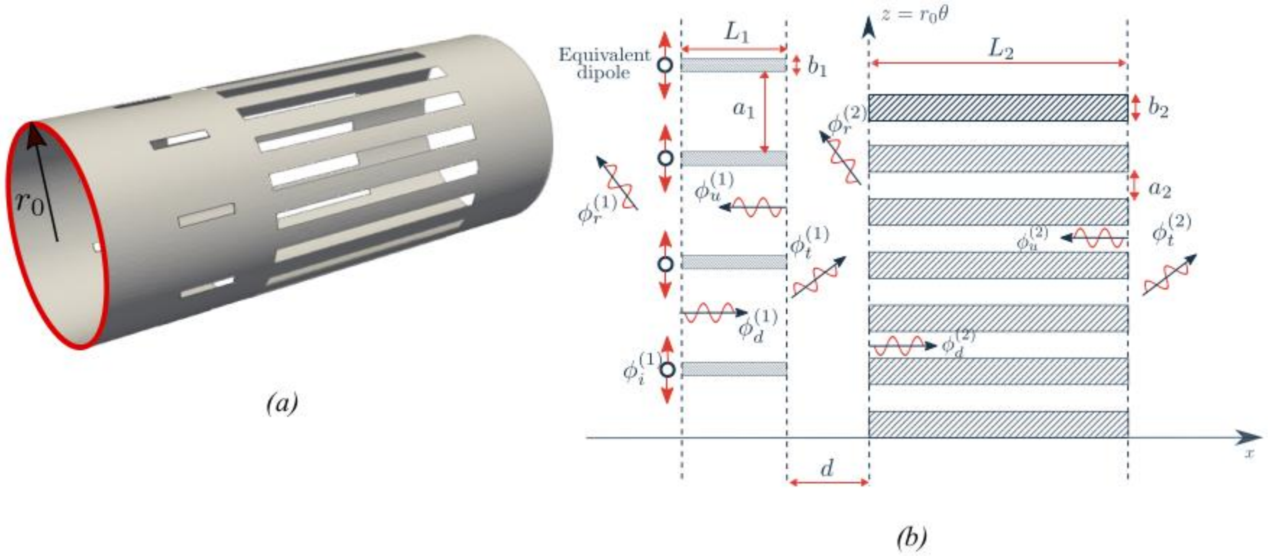


Figure 2: (a) Cylindrical cut of the ventilation system at a radius r_0 . (b) two-dimensional unwrapped representation. Propagation of waves generated by the diffraction of equivalent dipoles by the guide vanes.

Sound generation

The first part of this work is devoted to the sound generation mechanism. It is based on the work of Roger & Bouley [2,3,6]. The complete analytical modelling of the sound generation requires the description of the wake velocity deficit induced by the rotating blades of the impeller. In such a complex geometry, this parameter is not known and it needs to be extracted from an unsteady RANS simulation. Note here that the influence of this parameter is beyond the scope of this study. The aim is not to predict the absolute amplitude of the acoustic waves generated by the ventilation system, but to develop an analytical model to estimate relative variations, in order to optimise a cooling fan system. Furthermore, the model developed by Bouley *et al* [6] to predict the noise generated by the impingement of vortical waves on the guide vanes requires the definition of a mean flow. The hydrodynamic waves are convected by the flow. Roger *et al* [2,3] suggests an alternative approach to describe the response to a hydrodynamic disturbance as a diffraction problem of equivalent dipoles placed very close to the leading edge of each vane as shown in figure 2b. The use of this approach avoids the need for a mean-flow description, but it requires the description of the equivalent acoustic excitation ϕ_i. The latter is reproduced by the superposition of identical dipoles placed very close to the guide vanes. This excitation can be written as a sum of oblique plane waves having various propagation angles. The incident field ϕ_i can be written as:

$$\phi_i(x,z) = V_1 \sum_{s=-\infty}^{+\infty} A_s\, e^{i\alpha_s z} e^{ik_s(x+L_j)}, \alpha_s = \frac{n+sV_1}{r_0}, k_s = \sqrt{k_0^2-\alpha_s^2}, A_s = \frac{\alpha_s}{4\pi r_0 k_s} \qquad (1)$$

where $k_0 = n\,\Omega/c_0$ is the acoustic wavenumber, n the order of the rotational frequency harmonics and c_0 is the sound speed.

In practice the amplitude of the equivalent dipoles must be calibrated by the comparison with a reference solution [12,8]. It depends on the amplitude of the hydrodynamic waves $\phi_i \sim w_n$ .The scattering of this excitation can be achieved by using the mode-matching technique [2].

Blade modulation technique

The blade modulation technique consists in giving an irregular spacing of the impeller blades. This technique is very useful for reducing the tonal annoyance of a given fan by modifying the noise spectrum and distributing the sound energy to rotational frequency harmonics.

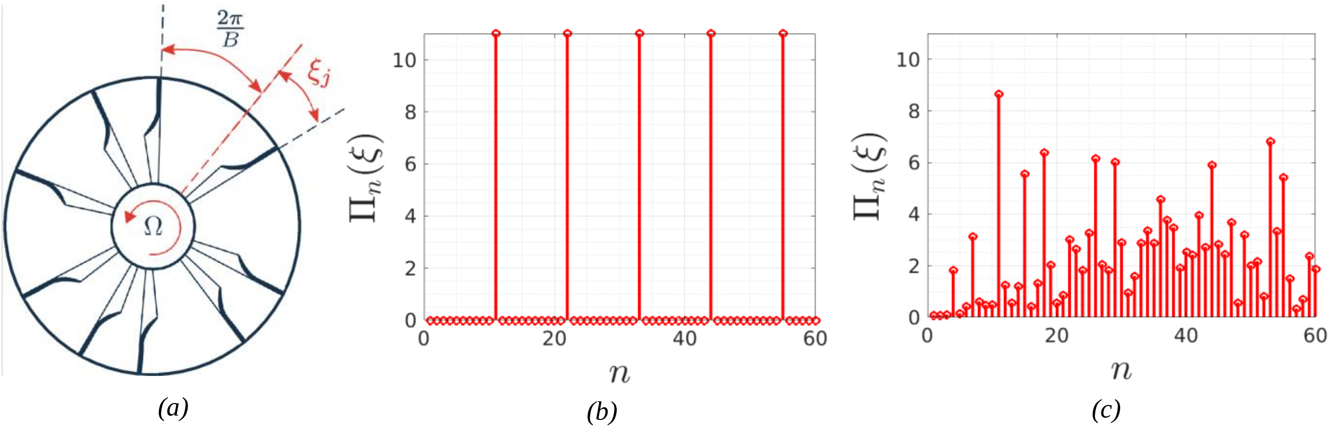


Figure 3: (a) Radial impeller. (b) Interference function for a fan with regular blade spacing $\xi = 0$. (c) Interference function for a fan with irregular blade spacing $\xi \neq 0$

The blade modulation effects have to be considered in a mathematical formulation of the hydrodynamic excitation. This was investigated, for instance, by Roger [7] who demonstrated that the mathematical expression of the hydrodynamic excitation w_n is written as:

$$w_n = \Omega \Pi_{-n}(\xi) F(n\Omega) \qquad (2)$$

where $F(n\Omega)$ is the Fourier transform of the velocity disturbance, Ω the rotational frequency and $\Pi_n(\xi)$ is an interference function, given by:

$$\Pi_n(\xi) = \sum_{j=0}^{B-1} e^{in\left(\frac{2\pi j}{B}+\xi_j\right)} \qquad (3)$$

The modulus of the interference function may have a value between 0 and the number of blades B. The influence of modulation angles ξ on the interference function Π_n is illustrated in Figure 3 for a 11-bladed impeller. It shows clearly that:

- For an equally spaced impeller $\xi = 0$ (Figure 3b), Π_n is equal to B if $n = gB$ and 0 if $n \neq gB$, with g is any integer.
- With unequal spacing of the fan blades $\xi \neq 0$ (Figure 3c) the interference is not completely constructive when $n = gB$ and it is not completely destructive when $n \neq gB$. Therefore, compared to a fan with regular blade spacing, a decrease of the peak levels at multiples of the blade-passing frequency is expected, with regeneration of harmonics of the rotational frequency.

In the present work, the interference function is used in order to select the most annoying harmonics to be studied.

Sound transmission

The acoustic waves generated by the diffraction of the equivalent dipoles propagate upstream and downstream the guide vanes. A part of the acoustic waves is transmitted through a row of cooling channels. These waves are partially reflected by the latter, generating an upstream field. Back-and-forth acoustic waves develop between the guide vanes and the cooling channels.

The sound propagation in this complex geometry is reproduced by the use of two transmission models which take into account the influence of the wall-thickness. The transmission model has been already presented in a previous work [1]. It is based on a two-dimensional mode-matching method [2, 4, 5, 6, 8, 9, 11, 12].

The main principles of the mathematical formulation of the mode-matching method are reminded here. In the first step, the geometry is divided into sub-domains in which the Helmholtz equation is separable. Then, the sound field is described as a sum of orthogonal modes in each sub-domain. For harmonic time dependence ($e^{-i\omega t}$), the acoustic fields produced by the diffraction are giving by:

$$\begin{pmatrix} \phi_t(x,z) \\ \phi_r(x,z) \end{pmatrix} = \sum_{s=-\infty}^{+\infty} \begin{pmatrix} T_s \\ R_s \end{pmatrix} e^{i\alpha_s z} \begin{pmatrix} e^{ik_s x} \\ e^{-ik_s x} \end{pmatrix}, \alpha_s = \frac{n+sV_1}{r_0}, k_s = \sqrt{k_0^2 - \alpha_s^2} \quad (5)$$

$$\begin{pmatrix} \phi_d^m(x,z) \\ \phi_u^m(x,z) \end{pmatrix} = \sum_{q=0}^{+\infty} \begin{pmatrix} D_q^m \\ U_q^m \end{pmatrix} cos\left(\alpha_q \left[z - m\frac{2\pi r_0}{V_1}\right]\right) \begin{pmatrix} e^{ik_q z} \\ e^{-ik_q z} \end{pmatrix}, \alpha_q = \frac{q\pi}{a_1}, k_q = \sqrt{k_0^2 - \alpha_q^2} \quad (6)$$

where ϕ_r and ϕ_t represent the reflected and transmitted fields in the unbounded domain, and ϕ_u^m and ϕ_d^m are the upstream and downstream acoustic fields in the m^{th} channel.

It may be noted from equations (5) and (6) that the acoustic potentials involve four unknowns R_s, T_s, U_q^m and D_q^m. They denote the modal amplitudes of the acoustic fields. They may be calculated from the matching equations, by imposing the continuity of the acoustic pressure and of the axial velocity on both interfaces of the channels:

- Channels inlet:

$$p_i + p_r = p_d + p_u$$
$$v_i^x + v_r^x = v_d^x + v_u^x$$

- Channels outlet:

$$p_d + p_u = p_t$$
$$v_d^x + v_u^x = v_t^x$$

The matching equations need to be projected on the two sets of eigenfunctions to get a system of linear equations that we can solve by a direct matrix inversion, and get the modal coefficients. This analytical model needs to be applied to the guide vanes and to the cooling channels. The output of the first subsystem is taken as the input of the next one. An iterative method is used in order to take into account the multiple reflections of the acoustic waves between the two subsystems.

NUMERICAL VALIDATION

In order to validate the analytical model, a comparison with the finite element method has been performed using the open source code FreeFem++ [13]. The latter is based on discretising the geometry of the ventilation system into small elements in which an approximate solution of the Helmholtz equation can be found.

Within the framework of linear acoustics, the validation needs to be performed for a single incident mode generated by the equivalent dipoles. The simulation domain must be divided into three different sub-domains:

1. The physical domain Ω_{phy}: represents the simulation domain.
2. A perfectly matched layer Ω_{pml}^o : placed downstream the cooling channels to attenuate the acoustic waves coming from the physical domain and avoid their reflection.
3. An active perfectly matched layer Ω_{pml}^i: in which the incident wave is imposed. This layer has the particularity to attenuate the acoustic waves in only one direction of propagation.

The simulation parameters are given in Table 1:

Table 1. Simulation parameters

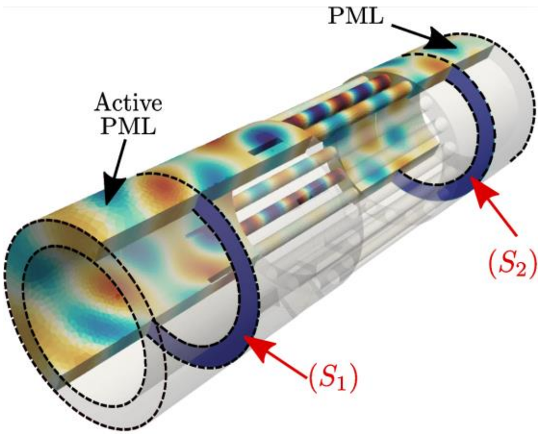
n	V_1	V_2	$d(m)$	$L_1(m)$	$L_2(m)$	$r_1(m)$	$r_2(m)$	N_E
3	9	18	0.1	0.1	0.4	0.13	0.17	$\approx 6 \times 10^5$

Figure 4a shows an instantaneous acoustic pressure field calculated by the finite element method. The physical domain is limited by the plane surfaces (S_1) and (S_2). The comparison between the mode-matching technique and the finite element method has been performed on the modal acoustic power. The associated acoustic powers $\mathcal{P}_r$ and $\mathcal{P}_t$ are evaluated by integrating the acoustic intensity over the areas of cross-sections (S_1) and (S_2) in figure 4a respectively:

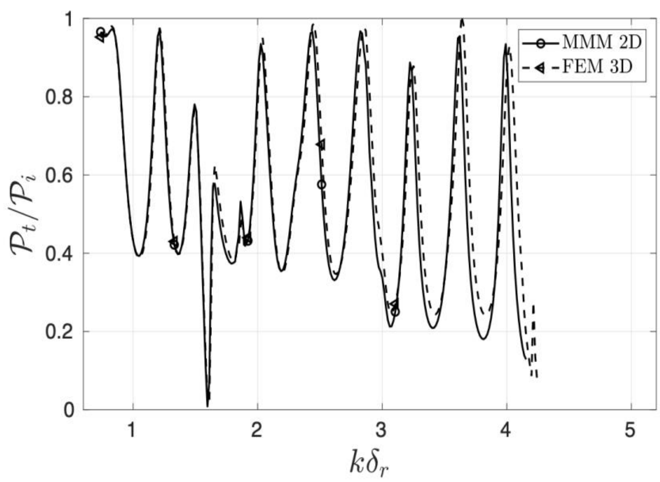
$$\mathcal{P}_r = \iint \frac{1}{2} \langle pv^* \rangle dxdy - \mathcal{P}_i \qquad (7)$$

$$\mathcal{P}_t = \iint \frac{1}{2} \langle pv^* \rangle dxdy \qquad (8)$$

Figure 4b shows the variations of the transmitted power obtained by the mode-matching method (MMM) and the finite element method (FEM) as functions of the Helmholtz number $k\delta r$, where $\delta r = r_2 - r_1$ is the thickness of the annular duct. Comparing the transmitted powers shows a good agreement of the results obtained by the two methods, up to relatively high frequencies. This confirms that the two-dimensional analytical model can be used with confidence for parametric studies.



(a)



(b)

Figure 4: (a) Acoustic pressure field obtained by the finite element method. (b) Transmitted acoustic powers calculated by the mode-matching method MMM (black line) and the finite element method FEM (dashed line)

Computing a complete spectrum with 300 frequencies on a personal computer with single core takes about 40 hours.

RESULTS

The analytical model is applied in this section to a ventilation system composed of 11-bladed impeller with circumferentially unsymmetrical blade-spacing, 8 guide vanes and 12 cooling channels. The test-case parameters are listed in Table 2.

Table 2. Test-case parameters

B	V_1	V_2	L_1 (m)	L_2 (m)	d (m)	r_0 (m)	Relative angles ξ (degree)
11	8	12	0.1	0.39	0.1	0.15	3°, -2°, 5°, -5°, 2°, 3°, -3°, 4°, -1°, -4°, 4°

The interference function for different values of n is shown in Figure 5a. The dominant harmonics are n=11, 15, 18, 26, 29. Each mode generates acoustic waves at the frequency $f = n\Omega/2\pi$. Figure 5b shows the variation of the transmitted powers as a function of rotational speed. It is found that all the dominant harmonics have no contribution in acoustic radiation when the fan speed is less than 1400rpm. Indeed, the acoustic waves generated by the ventilation system at this rotational speed are attenuated by the duct. These modes can be transmitted when the fan speed increases.

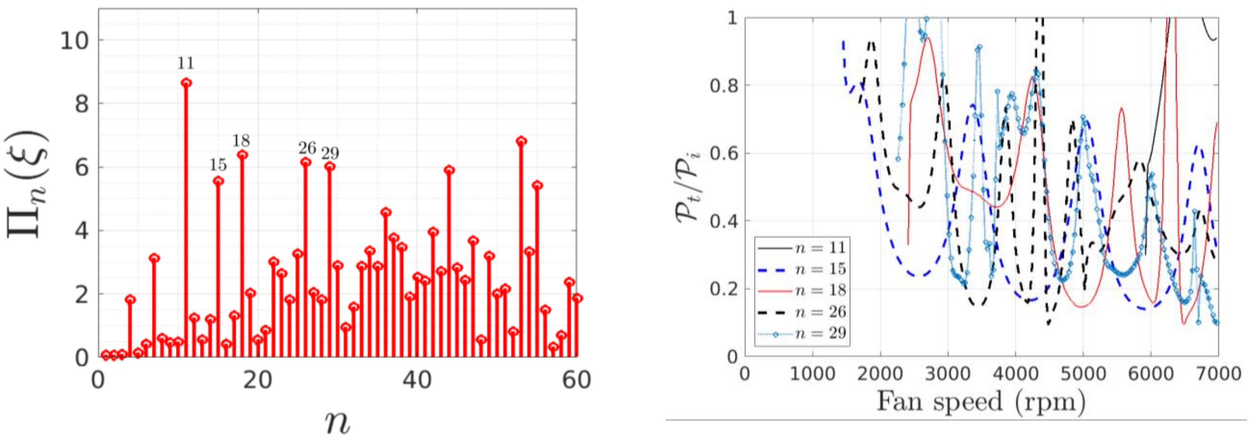
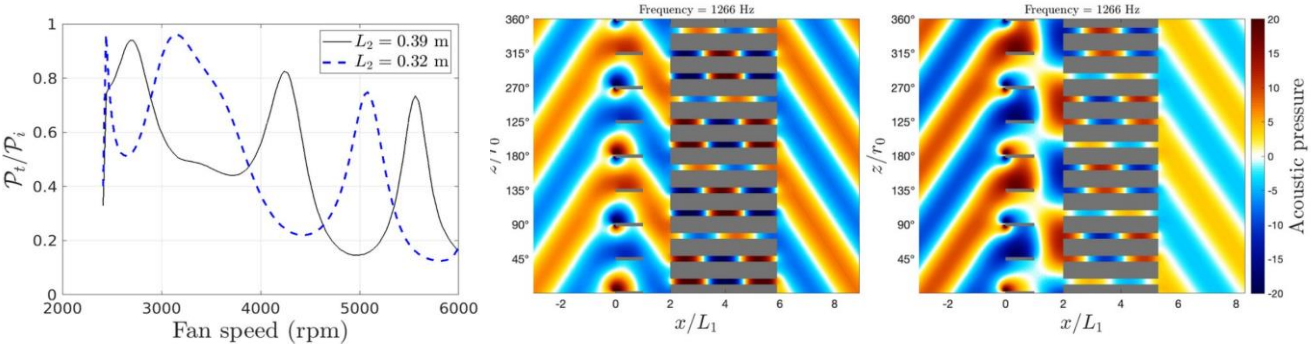


Figure 5: (a) Interference function. (b) Transmitted acoustic power.

Influence of channel length L_2

One of the most important aspects in this study is the influence of the length of the cooling channels on sound transmission. The transmitted acoustic powers are plotted as functions of fan speed in Figure 6a for n=18. Figures 6b and 6c show the instantaneous acoustic pressure fields for two different lengths of channels at 4200 rpm. The transmission peaks in Figure 6a are due to the acoustic resonances of the cooling channels. The latter are expected when the length of channels is a multiple of the half wave-length [10.1]. These frequencies are characterised by a strong amplification of the sound inside the cooling channels as shown in Figure 6b. It is therefore important to use channels having a length different from $L_2 = 0.39\, m$ at this rotational speed. The resonance frequencies of the channels must be different from the rotational frequency harmonics.

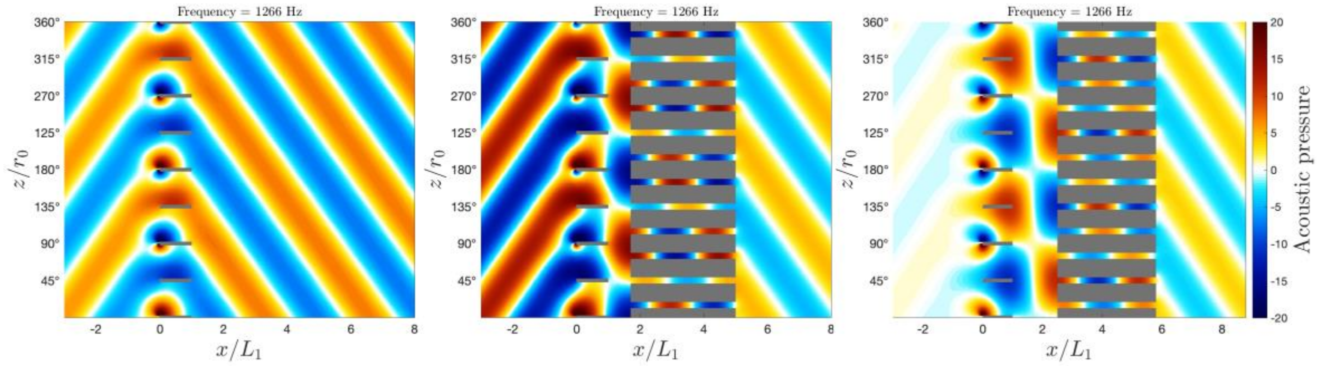


(a) (b) (c)

Figure 6: Influence of channel length. (a) Transmitted acoustic powers L_2 = 0.32 m (blue), L_2 = 0.39 m (black). Instantaneous acoustic fields, (b) L_2 = 0.39 m, (c) L_2 = 0.32 m. n = 18, B = 11, V_1 = 8, V_2 = 12, 4200 rpm

Effect of distance d

Figure 7 shows the acoustic pressure fields for 3 different configurations. The first configuration in figure 7a corresponds to the case without the cooling channels. The ventilation system emits upstream and downstream propagating waves. Figures 7b and 7c represent two configurations with different distances between the guide vanes and the cooling channels. Figure 7b shows that the acoustic field upstream of the guide vanes has been amplified compared with the first case (Figure 7a). This amplification is due to a constructive interference between the acoustic waves generated by the wake-interaction mechanism and those reflected by the cooling channels. Constructive interference occurs when two waves are in phase and propagate at the same frequency. The amplitude of the acoustic field is equal to the sum of the amplitudes of the two waves. In contrast, destructive interference occurs when the two waves are out of phase. This case is shown in figure 7c. When the distance between the two subsystems changes, the acoustic waves reflected by the cooling channels are phase-shifted. The low amplitude of the acoustic field upstream the guide vanes is due to the superposition of the maximum amplitude of the first wave with the low amplitude of the second one.



(a) (b) (c)

Figure 7: Instantaneous acoustic pressure fields. (a) Without channels, (b) d = 0.07 m, (c) d = 0.15 m, L_1 = 0.1 m, L_2 = 0.33 m, n = 18, Ω = 441.91 rad/s

It should be noted that the same interferences can be observed in the space between the guide vanes and the channels, because of the multiple reflections of the acoustic waves.

Effect of number of vanes

The influence of the number of vanes on sound generation is discussed in this section for n=18. Figure 8 shows the instantaneous acoustic pressure fields for three different numbers of vanes at the rotation speed 4200 rpm. In the first case (figure 8a), the configuration is composed of $V_1 = 5$, it generates two propagative modes, given by: $n_s = n + sV_1 = 18 + s5 = [3, -2]$. The acoustic field is dominated by the mode $n_s = 3$ and modulated by the mode $n_s = -2$. In the second case, for the configuration with $V_1 = 13$ vanes (Figure 8b), The modes generated by the system are cut-off, their amplitudes are attenuated exponentially from the interfaces. These modes do not contribute to the acoustic radiation. The last configuration with 16 vanes is shown in figure 8c, the ventilation system emits only one cut-on mode ($n_s = 2$).

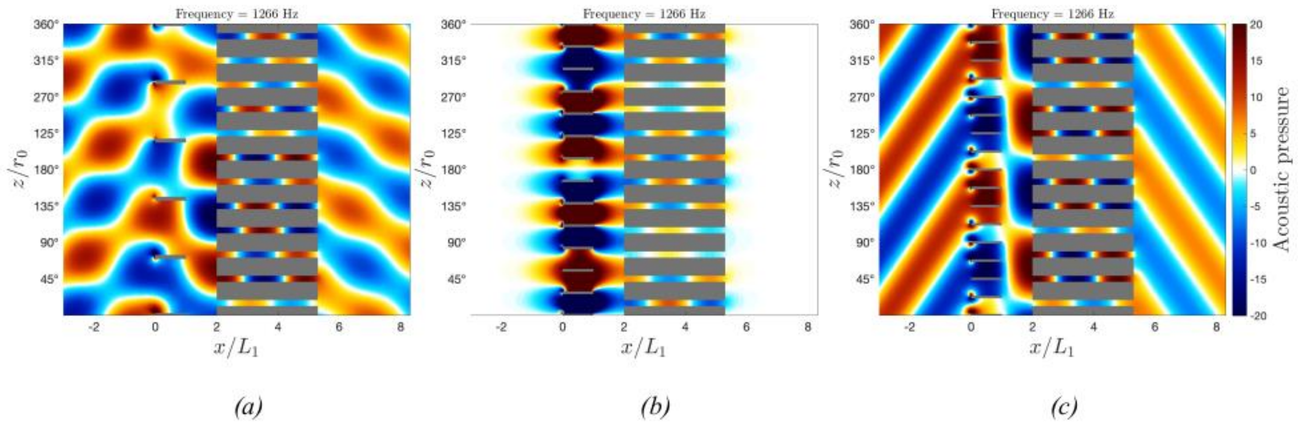


Figure 8: Instantaneous acoustic pressure fields for different numbers of vanes.
(a) $V_1 = 5$, (b) $V_1 = 13$, (c), $V_1 = 16$

Among these configurations, the second one ($V_1 = 13$) is quieter, because the acoustic waves generated at this frequency are cut-off.

CONCLUSION

In this paper, an analytical model for the sound generation and propagation in a system consisting of guide-vanes and cooling channels has been implemented. The model is aimed at improving the design of the cooling fan system integrated in the traction motors of trains to reduce the aerodynamic noise. The first part of the model addresses the sound generation by using the edge-dipole approach. Then, the sound propagation in this complex geometry was predicted by the use of two transmission models which take into account the influence of the wall thickness. The results were validated by a comparison with the finite element method, using an open source software "FreeFem++". A parametric study was carried out in order to show the influence of various parameters on sound generation and transmission. It has shown that the length of channels, the number of vanes and the distance between the guide-vanes and the cooling channels are very important parameters. The dimensions of the ventilation system must be adapted according to the fan speed and the blade spacing, in order to avoid the amplification of the most annoying tones produced by the cooling fan.

ACKNOWLEDGEMENT

This work was performed within the framework of the LABEX CeLyA (ANR-10-LABX-0060) of Université de Lyon, within the program "Investissements d'Avenir" (ANR-11-IDEX-0007) operated by the French National Research Agency (ANR).

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