The influence of periodic structures on sound propagation through heat exchangers

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Introduction

In a large number of industrial applications, heat exchangers are operated on the suction side of axial fans. This arrangement serves to dissipate or supply heat from these systems. Since these devices are often used in the direct vicinity of people, they must also comply with strict regulations regarding their acoustic noise emission. This is to prevent noise pollution caused by these devices and to avoid a negative impact on people's lives. In this context, the axial fan is the dominant sound source, and the sound radiation from it is usually amplified by the heat exchanger. The cause of this interaction lies in the inhomogeneous and turbulent flow field induced on the suction side of the axial fan by the heat exchanger [1]. However, the heat exchanger also influences the sound field. Due to the periodic arrangement of the heat exchanger tubes, amplifications and attenuations of specific frequencies can occur within the heat exchanger as they propagate through the heat exchanger. The influence on the sound field is mainly caused by reflections at the coolant tubes [2]. Figure 1 shows a schematic drawing of the flow field (blue) and the sound field (orange) through a heat exchanger axial fan module.

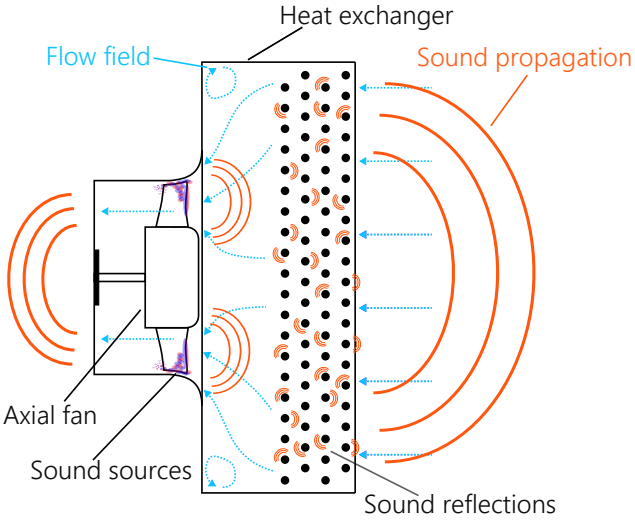

Abbildung 1: Sketch of the flow field and sound field of an axial fan heat exchanger module.

The aim of this investigation is to determine the influence of the heat exchanger on the sound field. This influence will be analyzed separately from the flow field. The sound propagation through the heat exchanger will be considered simulatively and analytically on the one hand and validated with experimental investigations on the other.

This article is structured as follows: First, a short introduction to the theory of periodic structures is given along with the simulation setup. Subsequently, the experimental setup is described. This is followed by the results of the sound propagation through the heat exchanger and a summary.

Theory & Simulation

The main objective is showing that the heat exchanger tubes can be described in terms of acoustics of periodic media [7, 6, 8]. Since the geometry is the key factor influencing the system (without the fluid flow – see [2] for further commentary), it is possible to neglect the thermo-viscous losses and let the system be governed by Helmholtz equation $\nabla^2 p + k^2 p = 0$, where p and k denote the acoustic pressure and wavenumber respectively. In the first approximation, one can predict the characteristic frequency of the bandgap by Bragg's diffraction law (see e.g., [5]). For our case of hexagonal lattice in the above shown orientation it gives the so-called Bragg frequency:

$$f_b = \frac{c_0}{\sqrt{3}a} \,, \tag{1}$$

where c_0 is the speed of sound and a the lattice constant of the periodic structures. In order to enhance this rather crude approximation, it is possible to use the (Floquet) periodic boundary conditions employed on a single unit cell to capture the effects of periodicity (see [6, 7]). Subsequently, one can predict the band structure of the heat exchanger tubes from the dispersion diagram (see Fig. 3 left, the bandgaps are marked in blue) obtained by solving the eigenvalue problem depicted in Fig. 2. To illustrate the sound propagation even closely, a 2D simulation of the sound propagation through the heat exchanger under more realistic geometry was conducted as well (see Fig. 3 center). A plane wave enters the system from the direction where the axial fan would be placed. All simulations were conducted in Comsol Multiphysics 5.5 with the Pressure acoustics, Frequency domain interface. In all cases the computational mesh had at least 12 second-order elements per shortest wavelength investigated.

Experimental Setup

The investigations are carried out in a standardized axial fan test rig. This has a low-reflection chamber on the suction side of the axial fan in which the sound propagation can be investigated. In this experimental environment, investigations of heat exchanger axial fan modules

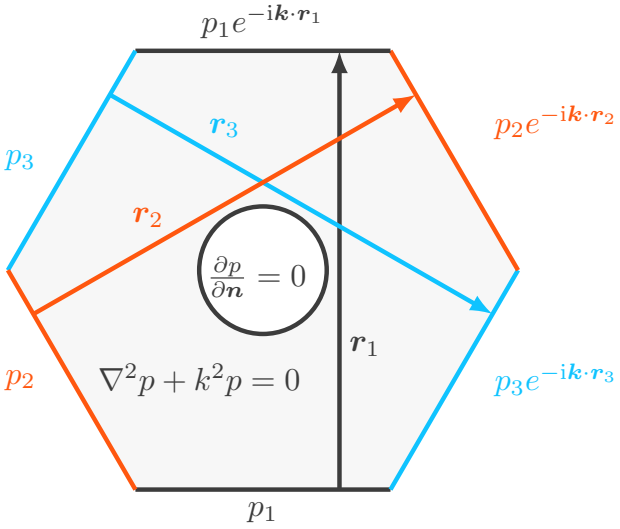

Abbildung 2: Schematic drawing of the setup employed for calculation of the dispersion diagram. The interior of the unit cell is governed by the Helmholtz equation, the heat exchanger tube is assumed perfectly sound hard and the Floquet symmetric conditions are supplemented to the respective boundaries.

are normally carried out, which is the reason why this test setup offers the best possibilities to combine the results found with those from previous work [3]. Since the sound field is to be investigated without the influence of the flow, the fan, which is the actual sound source, is replaced by a loudspeaker. The loudspeaker serves as the sound source in these investigations. The heat exchanger is mounted on the wall of the anechoic chamber. Within the low-reflection room, microphones according to DIN EN ISO 3745 [4] are arranged to determine the sound power. A white noise is used as excitation signal. The sound power of an empty heat exchanger housing is determined and subsequently that of a heat exchanger. Using equation (2), the insertion loss between the empty housing and the heat exchanger can be formed. Thus, the influence of the cooling fins and the cooling tubes of the heat exchanger on the sound propagation can be determined. Negative values of the insertion loss represent a sound reduction.

$$IL = L_{w,housing} - L_{w,hex} \tag{2}$$

The dimensions of the heat exchanger housing are 800 mm x 800 mm. The coolant tube radius of the heat exchanger used is r = 6 mm. The lattice constant of the periodic structure formed by the coolant tubes of the heat exchanger used is a = 40 mm.

Results

In the process of these investigations, the influence of heat exchangers on the sound field of axial fans was analyzed. Figure 3 shows the dispersion diagram obtained from the simulation of a single unit cell. The indicated directions of the reduced wave vector refer to the irreducible Brillouin zone, which is shown in the dispersion diagram.

At frequencies around 5000 Hz a pseudo-band gap opens up, which is shown in blue in figure 3 (left). This indicates that in this frequency range sound waves are attenuated when they travel through the periodic structures (coolant tubes). In addition, a complete band gap opens up within the investigated sonic crystal, which is located at 9000 Hz. The simulations of the dispersion diagram, were supplemented with propagation calculations in the frequency domain through the heat exchanger. A plane wave with specific frequencies is used as excitation. In the middle of figure 3 the sound pressure in the simulation area is shown for four specific frequencies. It shows that for the frequencies of the pseudo-bandgap and for the complete bandgap only low sound pressures are present on the right side of the heat exchanger. This confirms the sound reflecting properties of the periodic structures from the dispersion diagram. For two further frequencies a focusing of the sound pressure on the center axis can be observed. If the results from the dispersion diagram and from the propagation calculation are compared with the experimentally determined insertion loss, good agreements are found. Where the pseudo-bandgap and the complete bandgap are located, sound reductions of more than 5 dB are found. In addition, it is noticeable that the insertion loss decreases further at high frequencies. This can be attributed to the thermoviscous losses due to the high area of the cooling fins. This effect was not taken into account in the simulations due to complexity. The range of the pseudo-bandgap can additionally be determined via the analytical Bragg frequency. It can be seen that this lies in the middle of the pseudo-band gap with a value of f_b = 4951 Hz. This shows that the acoustic properties of heat exchangers can already be determined by simple analytical approaches as well as by simplified simulations on unit cells. It could be proven that the periodic structures inside the heat exchanger (the coolant tubes) behave like a sonic crystal and can contribute to sound reduction. With the help of this knowledge of these properties, heat exchangers can be modified in the future in such a way that they better attenuate disturbing frequencies of the axial fans.

Conclusion

It was demonstrated that the periodicity of heat exchanger tubes causes an effective sound propagation similar to sonic crystals. The presented experimental results match the theoretical predictions. These findings are useful for deeper understanding of the sound spectra measured in the vicinity of heat exchangers as well as for possible future optimization. It turned out that both the analytically determined Bragg frequency and the relatively simple simulation of the dispersion diagram from a unit cell offer the possibilities to analyze the heat exchangers with respect to their acoustic properties already before manufacturing. In addition, based on this knowledge, the heat exchanger can be adapted early in the development process so that unwanted tonal components can be reduced.

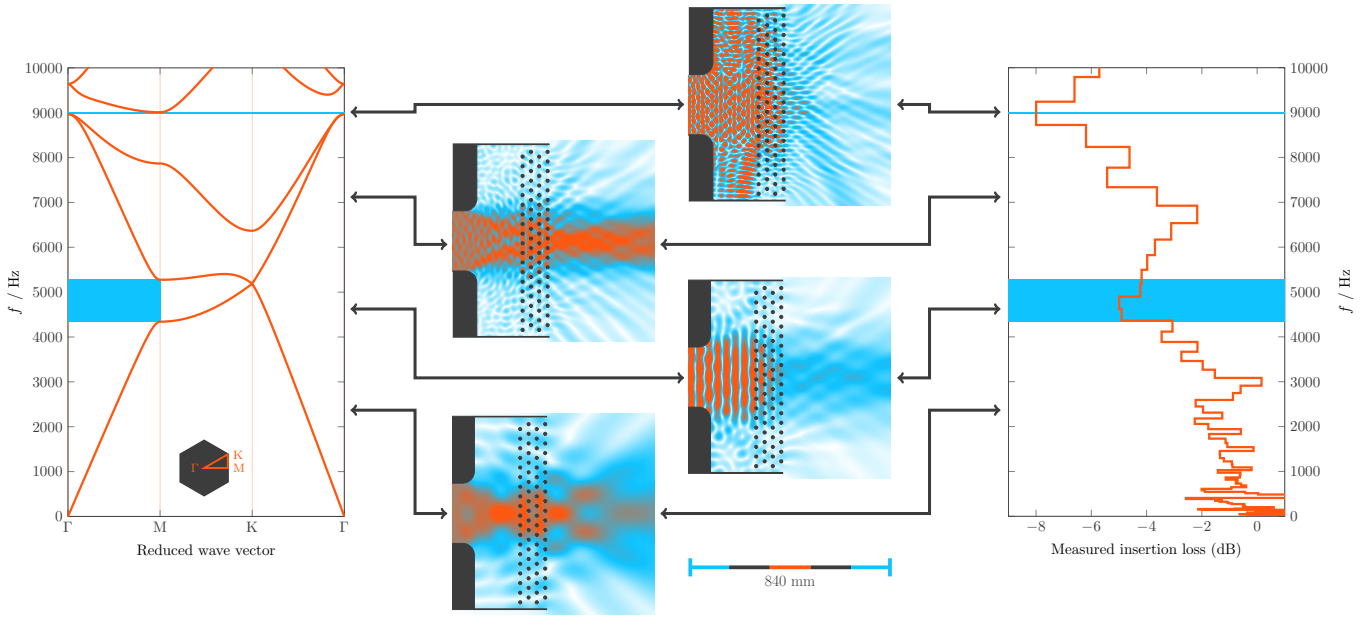

Abbildung 3: Dispersion diagram of a single unit cell (left), sound propagation through a heat exchanger, the depicted quantity is the magnitude of the complex pressure amplitude (center) and experimentally determined insertion loss (right).

Literatur

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