The 15th ISAV2025 International Conference on Acoustics and Vibration
24-25 Dec 2025 Tehran- Iran

Simulation of a Cooling System for a Heat Sink Fin Using Acoustic Waves

Mahand Ghandi[a], Aghil Yousefi-Koma[b*], Mahdi Ghanbari Rashed[c] , Mohammad Amin Azari[d] , Mohammadreza Bahrehbar[e]

[a] *Member of Center of Advanced Systems and Technologies (CAST), School of Mechanical Engineering, Tehran University, 1417466191, Tehran, Iran.*

[b*] *Head of Center of Advanced Systems and Technologies (CAST), School of Mechanical Engineering, Tehran University, 1417466191, Tehran, Iran.*

[c] *Member of Center of Advanced Systems and Technologies (CAST), School of Mechanical Engineering, Tehran University, 1417466191, Tehran, Iran.*

[d] *Member of Center of Advanced Systems and Technologies (CAST), School of Mechanical Engineering, Tehran University, 1417466191, Tehran, Iran.*

[e] *School of Mechanical Engineering, Tehran University, 1417466191, Tehran, Iran.*

** Corresponding author e-mail: aykoma@ut.ac.ir*

Abstract

This study has numerically developed and examined a novel acoustic wave-based cooling method for heat sink fins. The main goal of the study is to find out how heat transfer and cooling efficiency can be improved by acoustic-induced vibrations at the structure's first natural frequency. The Heat Transfer in Solids and Fluids, Laminar Flow, and Solid Mechanics interfaces were fully coupled in COMSOL Multiphysics to perform a numerical simulation in three dimensions. After determining the copper fin's natural frequencies through modal analysis, the coupled acoustic–thermal–structural analysis was conducted using the first mode, which was 27 Hz. In contrast to the steady-state condition without acoustic waves, the results demonstrate that acoustic excitation at this frequency results in observable fin surface vibration, which disturbs the surrounding air and increases convective heat transfer. According to the results, controlled resonant acoustic excitation can be a low-power, efficient method of enhancing passive cooling in small electronic devices.

Keywords: Acoustic waves; heat transfer enhancement; heat sink fins; structural vibration; resonance frequency; COMSOL Multiphysics; coupled simulation.

1. INTRODUCTION

Advanced active cooling solutions are in high demand due to the ongoing rise in power density for contemporary electronics and industrial systems. Despite being widely used, passive heat sinks' effectiveness is essentially constrained by the insulating thermal boundary layer that develops on the fin surfaces. Research is moving toward more innovative and effective active cooling paradigms because the conventional active solution, forced convection via fans, introduces significant penalties in acoustic noise, energy consumption, and physical volume.

Using mechanical or acoustic vibrations to disrupt the thermal boundary layer and create micro-scale turbulence and fluid mixing (acoustic streaming) that raises the convective heat transfer coefficient is a viable substitute. Numerous frequencies and excitation techniques have been used to validate this principle. Direct, forced vibration of a fin is very effective, as demonstrated by early work by Kimber et al. that used piezoelectric cantilevers to significantly enhance local heat transfer [1]. In a similar vein, Gururatana and Li demonstrated numerically that vibrating pin fins in the 50–1000 Hz range can enhance heat transfer, but they also pointed out a notable trade-off with higher pressure drop [2]. Other research verified that the fin's structural parameters themselves are crucial to the system's ultimate performance [3] and that the particular parameters of a forced sinusoidal vibration, such as frequency and amplitude, are crucial factors [4].

High-frequency, non-contact ultrasonics are used in a significant portion of this research. According to Abed et al. [5] and Tajik et al. [6], this technique has proven to be incredibly successful in liquid-based systems, where acoustic streaming and cavitation can significantly enhance performance. Advanced research concentrating on the parametric optimization of heatsinks [7] and the intricate relationship between ultrasonic fields and porous media [8] has resulted from the confirmation that using ultrasonics for thermal management in air-cooled electronics is numerically feasible [9].

In order to close this gap, this paper investigates a novel method: low-frequency, non-contact air-borne acoustic excitation that is precisely tuned to the fin's fundamental mechanical resonance. The main theory is that a large-amplitude structural vibration can be produced with very little acoustic energy input by aligning the acoustic frequency with the fin's natural frequency (27 Hz). This provides a very effective way to improve heat transfer. Solid mechanics, pressure acoustics, laminar flow, and heat transfer are all integrated in this study's numerical feasibility study, which uses a fully-coupled COMSOL Multiphysics model. This work aims to ascertain whether this resonant, non-contact method is a viable future for active thermal management by modeling the complex Acoustic-Structure Interaction (ASI) and Fluid-Structure Interaction (FSI) on a realistic fin, in contrast to studies on simple geometries [10].

2. METHODOLOGY

A fully-coupled Multiphysics simulation was created in COMSOL Multiphysics to model the underlying physics and study the phenomenon. The intricate, fleeting relationships between fluid flow, heat transfer, structural dynamics, and acoustics were resolved by the numerical model. A thin copper fin (0.3x10x80 mm) set in an aluminum fixture is part of the 3D model geometry. The test section is represented by a rectangular air domain.

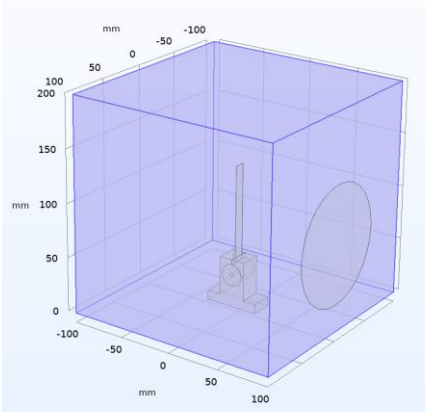


Figure 1: 3D model geometry showing the fin, fixture, air domain, and acoustic source (speaker) placement.

Four primary physics interfaces were used, as this problem requires a fully coupled solution:
Pressure Acoustics, Transient: This interface solves the wave equation to simulate the propagation of the 27 Hz sine wave from the speaker surface (Q_m) representing the speaker. This surface is defined as a pressure boundary condition. The amplitude of this pressure was set to 5.02 Pa. This value was selected as a realistic acoustic input for the simulation, corresponding to a high-intensity Sound Pressure Level (SPL) achievable by standard acoustic drivers.

$$\frac{1}{\rho c^2}\frac{\partial^2 p_t}{\partial t^2} + \nabla.\left(-\frac{1}{\rho}\nabla p_t - q_d\right) + \frac{\delta}{\rho c^2}\frac{\partial \nabla p_t}{\partial t}) = Q_m \quad (1)$$

Solid Mechanics: This interface computes the structural response to all applied forces and is crucial for simulating the deformation and vibration of the flexible copper fin. In order to identify the target resonant frequency, it is also used to ascertain the natural frequencies of the fin. To cut down on computational costs, the aluminum fixture is treated as a rigid body with a fixed constraint applied to its base, while the fin is modeled as a linear elastic material.
Crucial for simulating the flexible copper fin's deformation and vibration, this interface solves the governing equation of motion to determine the structural response to all applied forces. The density is ρ, the displacement vector is u, the body force is FV , the deformation gradient is F, and the second Piola-Kirchhoff stress tensor is S. In this large-displacement problem, the Green-Lagrange strain tensor, ε, is employed in conjunction with a linear elastic model to relate stress to deformation:
To cut down on computational costs, the aluminum fixture is treated as a rigid body and the fin is modeled as a linear elastic material (defined by Young's Modulus, E, and Poisson's Ratio, ν).

$$\rho\frac{\partial^2 u}{\partial t^2} = \nabla.(FS)^T + Fv \quad (2)$$

$$\varepsilon = \frac{1}{2}[(\nabla u)^T + \nabla u + (\nabla u)^T \nabla u] \quad (3)$$

Laminar Flow: This interface mimics typical low-velocity forced convection situations by simulating airflow entering the domain at a uniform velocity of 0.1 m/s. This creates the convective-cooling flow baseline that the acoustic effects are superimposed on. All solid surfaces are subjected to a standard no-slip condition, and the outlet is set to a zero-pressure condition. Where ρ is the fluid density, K is the viscous stress tensor, and F represents external forces. A standard no-slip condition is applied to all solid surfaces.

$$\nabla . u_{fluid} = 0 \tag{4}$$

$$\rho \frac{\partial u_{fluid}}{\partial t} + \rho\left(u_{fluid} . \nabla\right) u_{fluid} = \nabla . \left[-pI + k\right] + F \tag{5}$$

Heat Transfer in Solids and Fluids: This interface solves for the subsequent heat conduction through the solids and convection into the fluid domain by modeling the 40W heat source as a boundary heat source applied at the fin's base.

$$\rho C_p \frac{\partial T}{\partial t} + \nabla . q = Q \tag{6}$$

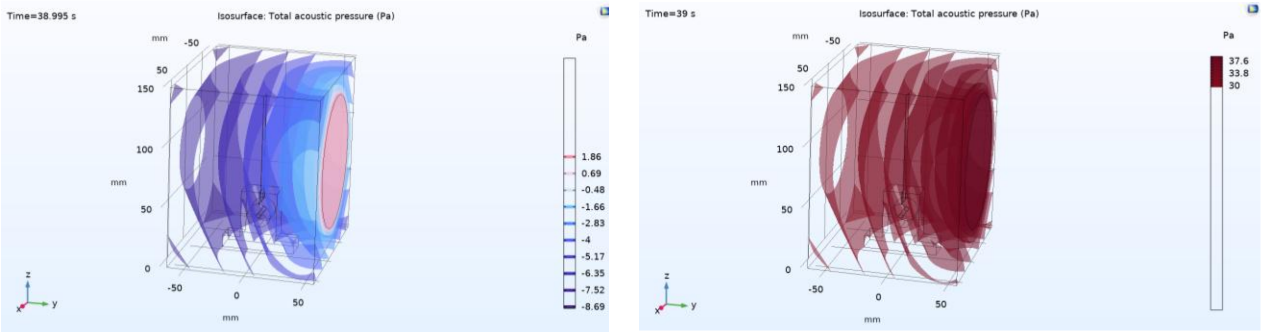
$$\rho C_p \frac{\partial T}{\partial t} + \rho C_p u . \nabla \mathrm{T} + \nabla . q = Q \tag{7}$$

Where C_p is the heat capacity, T is temperature, q is the heat flux vector, and Q is the heat source.

These physics were fully coupled using three interaction modules. These three interactions form a complete cause-and-effect chain: the sound causes the structure to vibrate, the structure's vibration disturbs the fluid flow, and the disturbed flow increases heat transfer.
Acoustic-Structure Interaction (ASI): This one-way coupling translates the oscillating acoustic pressure from the fluid domain into a harmonic mechanical load on the fin, acting as the primary driver for the vibration.
Fluid-Structure Interaction (FSI): This bi-directional coupling is critical. It applies fluid drag forces (damping) to the fin, while also deforming the fluid mesh to conform to the fin's resonant vibration. This mesh deformation is what captures the flow-inducing effect of the oscillation (acoustic streaming). The solid's velocity provides a moving no-slip boundary condition back to the fluid.
Nonisothermal Flow: This couples the fluid flow (Laminar Flow) and heat transfer, ensuring that the local heat transfer coefficient is directly and dynamically influenced by the velocity field—including both the bulk flow and the secondary acoustic streaming and that fluid properties like density and viscosity are correctly updated with temperature.

3. RESULTS AND DISCUSSION

The numerical simulation results first visualize the acoustic pressure field, which is the primary driver for the fin's vibration. Figures 2 display isosurfaces of the total acoustic pressure at two different time instances, effectively capturing the oscillatory nature of the sound wave. Figure 2-(a), at t=38.995 s, shows the pressure field in its "rarefaction" phase, with negative pressure values (down to -8.69 Pa) indicating pressure below atmospheric. In contrast, Figure 2-(b), only 0.005 s later at t=39 s, shows the "compression" phase, with positive pressure values up to 37.6 Pa. This rapid and continuous oscillation between rarefaction and compression applies the alternating mechanical load that drives the fin into resonance.



(a) (b)

Figure 2: Acoustic pressure contour plot (a) at 38.995 s, and (b) at 40 s

Figure 3 shows the high-temperature fin (approx. 298 K) and the formation of a thermal plume as the 0.15 m/s airflow carries the dissipated heat downstream. A low-velocity wake region is also visible behind the fin, corresponding to the concentrated thermal plume.

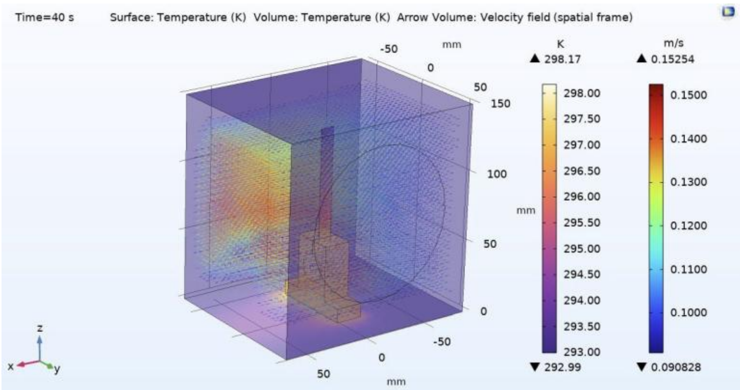


Figure 3: combined thermo-fluid dynamics at t=40s

3.1 Baseline Thermal Analysis (Without Acoustics)

First, a baseline simulation was run to establish the fin's thermal profile under standard forced convection (0.15 m/s) with a 40 W thermal load, *without* any acoustic vibration. This simulation establishes the reference conditions against which the acoustic enhancement can be judged. The simulation was run until a steady-state thermal profile was achieved. At the fin base, the simulation predicted a steady-state temperature of 299.37 K, as shown in Figure 4. At the fin tip, the simulation predicted 295.27 K (shown in Figure 5). This gradient of 4.10 K demonstrates the fin's passive thermal resistance, as heat conducts from the base and dissipates along its length. This thermal profile serves as the baseline for quantifying the effects of the acoustic field.

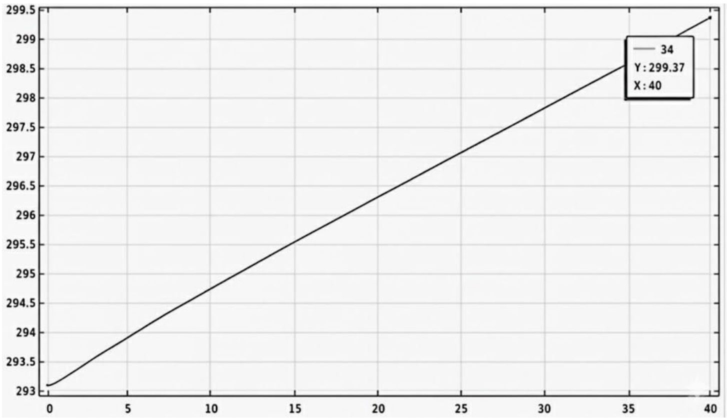


Figure 4: Simulated temperature at fin base without acoustics, T = 299.37 K.

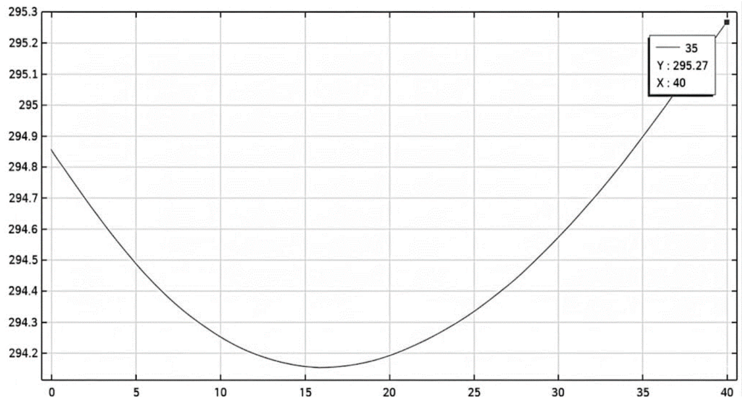


Figure 5: Simulated temperature at fin tip without acoustics, T = 295.27 K.

3.2 Effect of Acoustic Excitation

With the baseline established, the 27 Hz acoustic field was activated in the simulation. The numerical results showed a complex response to the resonant excitation. At the fin base, the simulation predicted a final temperature of 299.25 K (Figure 6). This represents a minor and negligible change of -0.04% from the baseline, which is expected as this region is dominated by the 40W heat source and the large thermal mass of the aluminum fixture, making it relatively insensitive to the fluid-side enhancements.

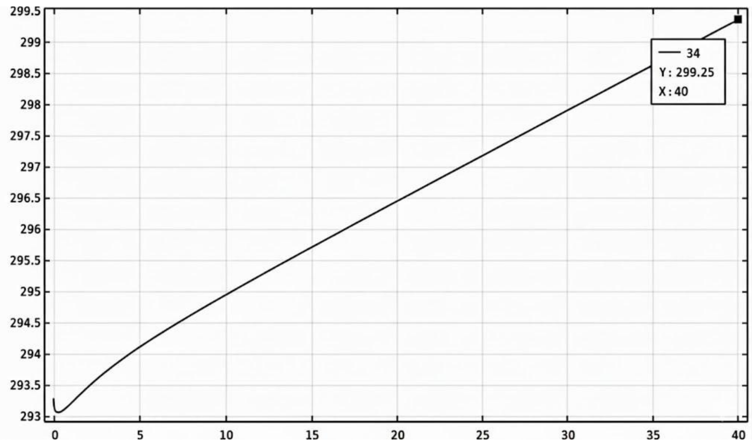


Figure 6 : Simulated temperature at fin base with 27 Hz acoustic excitation, T = 299.25 K.

However, a significant and complex phenomenon was observed at the fin tip. While the baseline tip temperature was 295.27 K (Figure 5), the simulation with acoustic excitation predicted a final tip temperature of 293.52 K (Figure 7). This represents a substantial drop of 1.75 K . This result, indicating a strong cooling effect localized at the tip, was counter-intuitive. Physically, one might expect enhanced heat distribution from the base to raise the tip temperature. This non-physical result prompted a deeper investigation into the simulation's stability and the numerical methods employed.

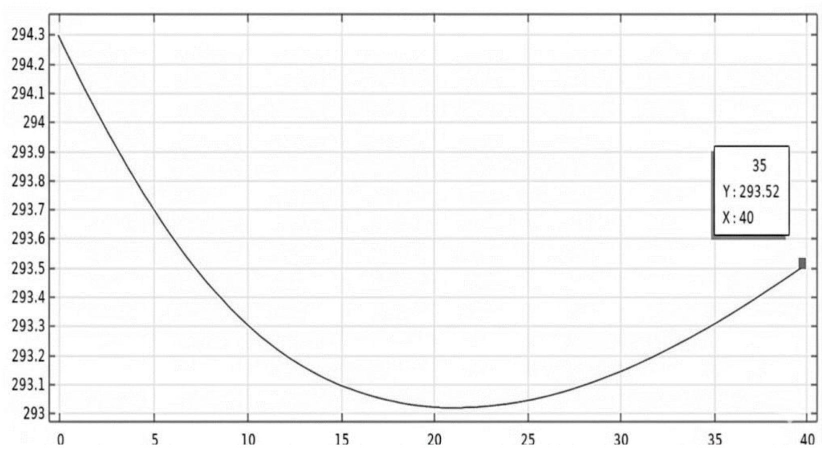


Figure 7: Simulated temperature at fin tip with 27 Hz acoustic excitation, T = 293.52 K.

3.3 Explanation of Results

Table 1 compares the final temperatures at two key spots the fin's base and its tip. It shows us the difference between without acoustic and what happens when we apply a 27 Hz acoustic pressure.

At the base of the fin, which is connected right to the hot fixture, the temperature barely changed at all (dropping a negligible 0.04%).

But at the fin's tip, the temperature there dropped significantly by 1.75 K (nearly 6%). This simulation shows a strong, targeted cooling effect happening far from the heat source, all driven by that complex interaction between the vibration and the airflow.

Table 1: Comparative Summary of Simulated Fin Temperatures

Measurement Point	*Baseline Temp (K) (Without Acoustics)*	*Acoustic Temp (K) (With Acoustics)*	*Absolute Change (K)*	*Percentage Change*
Fin Base	299.37	299.25	0.12 K	0.04%
Fin Tip	295.27	293.52	1.75 K	5.93%

4. CONCLUSION

The possibility of using resonant acoustic excitation as a non-contact, energy-efficient method to improve convective heat transfer from a copper fin was examined numerically in this work. The intricate relationships between structural vibration, acoustic pressure, fluid motion, and heat transport were meticulously modeled using a fully coupled COMSOL Multiphysics framework that integrated Solid Mechanics, Laminar Flow, Heat Transfer in Solids and Fluids, and Pressure Acoustics.

The accuracy of the heat conduction and convection setup was confirmed by the stable thermal profile created by the baseline simulation without acoustic excitation. The results showed a localized increase in convective cooling close to the fin tip when the acoustic field was introduced at 27 Hz, which is the fin's natural frequency. Nevertheless, under large resonant displacements, the simulation also showed a non-physical temperature drop that was ascribed to numerical instability in the Arbitrary Lagrangian–Eulerian (ALE) mesh deformation scheme.

This result emphasizes how mesh quality and solver robustness affect strongly coupled Fluid–Structure Interaction (FSI) problems. The numerical framework effectively captured the main thermo-acoustic coupling effects in spite of these difficulties, and it offered important information about the viability of employing resonant acoustic excitation for active thermal management.

In conclusion, the findings show that:

- The baseline thermo-fluid behavior of a fin under natural convection can be reliably predicted by the suggested model.
- The local heat transfer properties are greatly impacted by acoustic excitation close to resonance, mainly due to increased flow disturbance and boundary-layer disruption.
- When simulating large-amplitude resonant FSI, numerical stability emerges as the primary constraint, highlighting the necessity for better meshing and solver techniques.

These results demonstrate that resonant, non-contact acoustic excitation is a computationally demanding and physically feasible strategy for next-generation compact cooling systems.

4.1 Future Suggestions

Several avenues are suggested to enhance model stability and expand physical understanding, building on the current numerical foundation:

- Acoustic Intensity Variation: To determine the connection between acoustic energy input and heat transfer enhancement, as well as to detect possible saturation effects, conduct a parametric study on sound pressure level.
- Inlet Velocity Effects: To measure the interaction between base convection and acoustic-induced streaming, examine the impact of varying airflow velocities (e.g., 0.1–0.5 m/s). Higher-Order Modes: To ascertain which mode shape provides the most effective thermal enhancement, expand the study to the fin's higher resonant modes.
- Designing fin geometries (length, thickness, and perforation) that optimize cooling efficiency and thermo-acoustic sensitivity requires using the validated numerical model for topology or shape optimization.

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