Numerical investigation of thermoacoustic refrigerator at weak and large amplitudes considering cooling effect

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Abstract

In this paper, OpenFOAM package is used for the first time to simulate the thermoacoustic refrigerator. For simulating oscillating inlet pressure, we implemented cosine boundary condition into the OpenFOAM. The governing equations are the unsteady compressible Navier–Stokes equations and the equation of state. The computational domain consists of one plate of the stack, heat exchangers, and resonator. The main result of this paper includes the analysis of the position of the cold heat exchanger versus the displacement of the pressure node at large amplitude for successful operation of the refrigerator. In addition, the effect of the input power on the successful operation of the apparatus has been investigated. It is observed that for higher temperature difference between heat exchangers, the time of steady state solution is longer. We show that to analyze and optimize the thermoacoustic devices, both heat exchangers should be considered, coefficient of performance (COP) should be checked, and the successful operation of the refrigerator should be evaluated.

1. Introduction

A thermoacoustic refrigerator is a device that transfers heat from a low-temperature reservoir to a high-temperature reservoir by utilizing acoustic power. The standing wave thermoacoustic refrigerators consist mainly of four parts: acoustic driver, resonator, heat exchangers, and stack. The acoustic driver is attached to the resonator filled with a gas. In the resonator, the stack consisting of many parallel plates and two heat exchangers are installed as illustrated in Fig. 1. The acoustic driver sustains an acoustic wave in the gas at the fundamental resonance frequency of the resonator. The standing wave displaces the gas in the channels of the stack. The thermal interaction between the oscillating gas and the surface of the stack transfers heat from the cold side to the hot edge. The heat exchangers exchange heat between the apparatus and reservoirs.

Thermoacoustic devices use no moving parts, no exotic and poison materials; therefore, they seem to have the immediate potential to have comparably high reliability and low cost [1]. Cao et al. [2] simulated an oscillating gas near a 1D isothermal stack and computed energy flux density in thermoacoustic devices. Unusual vertical energy flux is found near the ends of the stack plate within an area whose length scale is proportional to the gas displacement amplitude. Worlikar and Knio [3] used an overall method consisting of a quasi-1D computation scheme for resonator and a multidimensional vorticity/stream-function potential formulation for the detailed simulation of flow around the stack. They demonstrated that the 1D code is capable of representing wave amplification through heat addition for weakly-nonlinear acoustic. Worlikar et al. [4] further extended their previous work by solving the energy equation in the fluid and the stack plates. They implemented fast Poisson solver for the velocity potential based on the domain decomposition/boundary Green’s function technique. They predicted the steady state temperature gradient across a two-dimensional couple and analyzed its dependence on the amplitude of the resonant wave. Ishikawa and Mee [5] used PHOENICS commercial code and solved 2D full Navier–Stokes equations and simulated flow near an isothermal zero thickness stack. They examined solver results in the form of energy vectors, particle paths, and overall entropy generation rates. It is observed that, the time-averaged heat transfer to and from the plates is concentrated at the edges of the plates. In constant Mach number, the width of the region where there is substantial heat transfer decreases as the plate spacing is reduced. Tasnim and Fraser [6] simulated a conjugate heat transfer in the thermoacoustic refrigerator. They solved unsteady compressible Navier–Stokes and energy equations with commercial code STAR-CD, and illustrated flow and thermal fields during a cycle. Ke et al. [7] used self-written program of the compressible SIMPLE algorithm and carried out
In this paper, two expressions are frequently employed: “weak amplitude” and “large amplitude”. For the conditions that inlet dynamic pressure increases and Mach number becomes greater than 0.1, the results show that nonlinear effects (e.g., pressure node displacement) influence on the operation of the apparatus and are not negligible [1]. Such conditions are defined as “large amplitude”. In contrast, “weak amplitude” is referred to conditions that inlet dynamic pressure cannot deliver enough heat to the ambient heat exchanger or cannot absorb heat from the cold heat exchanger to make COP > 0.

2. Governing equations and numerical method

The continuity, momentum and energy equations for compressible flow in a two-dimensional Cartesian coordinate system are as follows:

\[
\frac{\partial p}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0, \\
\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = -\frac{\partial p}{\partial x} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) + \mathbf{S}_{Mx}, \\
\frac{\partial (\rho v)}{\partial t} + \nabla \cdot (\rho v \mathbf{u}) = -\frac{\partial p}{\partial y} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) + \mathbf{S}_{My}, \\
\frac{\partial (\rho h_0)}{\partial t} + \nabla \cdot (\rho h_0 \mathbf{u}) = \frac{\partial p}{\partial t} + \nabla \cdot (k \nabla T) + \mathbf{S}_h, \tag{4}
\]

where \( h_0 = h + \frac{1}{2} (u^2 + v^2), h = c_p(T - T_m) \). The viscosity \( \mu \) and thermal conductivity of fluid \( k \) are temperature dependent. Working gas is air, assumed to be an ideal gas, therefore, the state equation is:

\[ p = \rho RT. \tag{5} \]

Energy equation in the solid domain of the parallel-plate stack and heat exchangers is given by:

\[ \frac{\partial (\rho c_p T)}{\partial t} = \nabla \cdot (k \nabla T). \tag{6} \]
Stack is made of glass with constant thermo-physical properties.

The above governing equations for laminar flow are solved with the chtMultiRegionFoam solver of the OpenFOAM V.2.1.1. These equations are discretized using the finite volume method. GAMMA scheme is used for discretization of the convective and diffusive terms. GAMMA is a second-order scheme and combination of the upwind differencing (UD) and central differencing (CD). In GAMMA scheme, the transition between UD and CD is smooth; this reduces the amount of switching in the differencing scheme and improves the convergence [10]. A first-order fully implicit scheme is employed for the discretization of the temporal term. The PISO algorithm is used in order to couple the momentum and the continuity equations [11].

Fig. 2 shows the computational domain, which consists of one plate of stack and heat exchangers. Stack is located between the plate of stack and heat exchangers. Stack is considered as conjugate heat transfer boundary condition.

Upper and bottom boundaries of the computational domain are set as symmetry boundary conditions as follows:

\[ \nu = 0, \quad \frac{\partial u}{\partial y} = 0, \quad \frac{\partial p}{\partial y} = 0, \quad \frac{\partial T}{\partial y} = 0. \]

Heat exchangers are considered as constant temperature, and the stack is considered as conjugate heat transfer boundary condition:

\[ T_{\text{solid}} = T_{\text{fluid}}, \quad k_{\text{solid}} \frac{\partial T_{\text{solid}}}{\partial n} = k_{\text{fluid}} \frac{\partial T_{\text{fluid}}}{\partial n}. \]

Velocity and pressure boundary conditions on stack and heat exchangers are as follows:

\[ \frac{\partial p}{\partial n} = 0, \quad u = 0, \quad v = 0. \]

The left boundary is located adjacent to the driver; therefore, we set the pressure equal to the inlet pressure there:

\[ p = p_m + p_A \cos(\omega t). \]  

(7)

Velocity at the left boundary is not zero; because this boundary is not on a solid surface, and the exact position of the velocity node is unknown. Therefore, we use zero gradient boundary condition there:

\[ \frac{\partial u}{\partial x} = 0, \quad \frac{\partial v}{\partial x} = 0, \quad \frac{\partial T}{\partial x} = 0. \]

The right wall has the following conditions:

\[ \frac{\partial p}{\partial x} = 0, \quad \frac{\partial T}{\partial x} = 0, \quad u = 0, \quad v = 0. \]

The initial values of pressure, temperature, and velocity are \( p_m \), \( T_m \), and zero, respectively. The initial temperature of the stack is crucial in determining the required time to reach a steady state solution. Average temperature of the hot and cold heat exchangers is a suitable initial value for the stack temperature, but it is better to set a linear function between the hot and cold heat exchangers temperature. It is because the temperature of the stack becomes approximately a linear function of the hot and cold heat exchangers temperature as the solution reaches to the steady state condition.

These boundary conditions represent a real thermoacoustic refrigerator behavior. An acoustic wave with known pressure amplitude is driven to the apparatus, at the end of the resonator it is reflected and contracted with another wave. As a result, a standing wave appears.

Energy flux density \( \dot{e}_r \) over a cycle in the fluid domain is computed as follows:

\[ \dot{e}_r = \rho v \left( \frac{1}{2} V^2 + h \right) - k \frac{\partial T}{\partial y}, \]  

(8)

where \( V = \sqrt{u^2 + v^2} \) and \( h = c_p(T - T_m) \). The dissipations are neglected in Eq. (8).

4. Grid independence and validation

For a grid independency test, nine grid sizes were employed. According to Fig. 3, the cooling load of the apparatus using a grid size of greater than 180 × 40 becomes independent of the grid scale.

![Fig. 2. Computational domain.](image1)

![Fig. 3. Grid independency.](image2)
temperature. Then, the gas parcel comes back over the cold heat exchanger and repeats this process.

Parcels in the mid-stack are the same as the middle members of a bucket brigade, passing heat along. At either end are parcels that oscillate between the stack and one of the heat exchangers. At the left end, such parcels absorb heat from the cold heat exchanger and release heat to the stack. At the right end, such parcels absorb heat from the stack and release heat to the ambient heat exchanger. In Overall, the net effect is to absorb heat from the cold heat exchanger and reject waste heat at the ambient heat exchanger.

Another interesting subject in oscillating flow is the variation of the velocity profile. Fig. 7 illustrates the velocity profile around the stack and during a short part of the cycle when the flow direction is reversed. In Fig. 7(a), gas flows to the right, then the velocity gradually reduces and finally it changes the direction and comes back. In Fig. 7(b) and (c), the velocity has positive and negative values, as there is a velocity phase-shift between fluid layers due to the distance from the wall. Further from the wall, the viscous stress is lower; therefore, the inertial force increases and the response of the flow to the pressure-change becomes slower. Consequently, flow near the wall responses more quickly to the pressure-change and moves in the reverse direction.

6. Results of the numerical simulation

Variations of the pressure and velocity during one-half of a cycle with a low-pressure amplitude of 1 kPa, are plotted in Figs. 8 and 9. In these plots, a cycle is divided into ten time step. As the inlet pressure amplitude is low, the pressure and velocity distributions in Figs. 8 and 9 are the same as the ideal standing wave. Within the stack, the cross section reduces, therefore, the velocity increases. At the end-wall, velocity is zero; therefore, the pressure antinode appears on this surface. Although the velocity boundary condition at inlet is zero gradient, but considering the length of the device that is equal to the half of the wavelength, the contraction of the waves creates a velocity node at the inlet. This observation shows that our boundary conditions are correct.

Fig. 10 is provided to show weak amplitude and its effect on the cooling load. Fig. 10 shows that, for \( p_a = 10 \) Pa, the apparatus cannot absorb heat from the cold heat exchange. Instead, it delivers heat to the cold heat exchanger (\( q_h < 0 \)). The reason of this phenomenon is due to the weakness of the pressure amplitude; therefore, temperature fluctuation on the cold heat exchanger cannot be lower than the heat exchanger temperature. As a result, the heat transfers from the gas to the cold heat exchanger. At the next stage, we increased the inlet dynamic pressure; in Fig. 10, it is clear that cooling effect of the apparatus become positive at more than 1 kPa, but it is not sufficient. Additionally the rejected heat to the ambient must be greater than the cooling load, because COP should be always positive for refrigerators.

\[
\text{COP} = \frac{q_c}{q_h - q_c},
\]

\( q_c < q_h \Rightarrow \text{COP} > 0 \)

At inlet dynamic pressure more than 1–5 kPa in Fig. 10, the \( q_c \) is positive but it is greater than \( q_h \), so the COP is negative; this shows that the apparatus does not work successfully. According to Fig. 10, for 10 kPa and greater than it, the \( q_h \) is greater than \( q_h \), therefore the COP, would be positive. The conclusion of this part is that, the inlet dynamic pressure should be set correctly according to the temperature of heat exchangers. Therefore, in the design of the thermoacoustic devices, it is necessary to consider the stack and cold heat exchangers to be sure that COP is positive. In many previous studies (e.g., Ref. [7]) one heat exchanger or both of them have been ignored.

**Fig. 4.** Independence of the time step.

**Fig. 5.** Energy flux density over the stack, \( p_a = 1 \) kPa, \( T_c = 297 \) K, \( T_s = 300 \) K, and \( y = 0.0006 \) m, comparison of the current work with that of Ref. [6].

**Fig. 6.** shows the temperature contour and the velocity profile in a thermoacoustic refrigerator. In Fig. 6(a), which corresponds to the beginning of the cycle, let’s consider a parcel of gas that is on the left side in the cold heat exchanger with zero velocity, which moves to the right. At this time, the pressure is minimum, and temperature arises more than the local temperature of the stack, so the parcel of gas releases the heat at a slightly higher temperature. Then, the gas parcel comes back over the cold heat exchanger and repeats this process.

Parcels in the mid-stack are the same as the middle members of a bucket brigade, passing heat along. At either end are parcels that oscillate between the stack and one of the heat exchangers. At the left end, such parcels absorb heat from the cold heat exchanger and release heat to the stack. At the right end, such parcels absorb heat from the stack and release heat to the ambient heat exchanger. In Overall, the net effect is to absorb heat from the cold heat exchanger and reject waste heat at the ambient heat exchanger.

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The cooling power of the cold heat exchanger and rejected heat of the ambient heat exchanger versus time, with $p_A = 10$ kPa, are shown in Fig. 11. Fig. 11 shows that the solution reaches the steady state after 10 s. Additionally, the rejected heat from the ambient heat exchanger is greater than the cooling load. This observation shows that the refrigerator works successfully.

We compare the energy flux density over the stack and heat exchangers with analytical results of Piccolo [12]. Figs. 12 and 13 at low inlet pressure and Figs. 14 and 15 at large amplitude have almost the same shapes, because two investigations have approximately the same operating conditions.

Inflection points, over the heat exchangers, in Figs. 12 and 14, vary with $p_A$ and are related to the gas parcel displacement. Gas parcel displacement is computed as follows [1]:

$$\delta = \frac{2u_A}{\omega}; \quad (10)$$

where $u_A$ is the velocity amplitude, which is related to the pressure amplitude, and $\omega$ is the angular frequency. Therefore, at constant frequency, the gas parcel displacement changes with $p_A$. For $f = 200$ Hz and $p_A = 10$ kPa, the velocity amplitude is 25 m/s according to Fig. 16, therefore the gas parcel displacement is obtained as:

$$\delta = \frac{2 \times 25}{2 \times \pi \times 200} = 0.04 \text{ m}.$$ 

For $p_A = 1$ kPa, the gas parcel displacement is lower than the length of heat exchangers. The gas parcel transfers heat from one side of the heat exchanger to the middle of the heat exchanger, at the next stage, it transfers heat to the upstream or stack. Therefore, the average of heat transfer over one cycle, at the middle of the heat exchanger is zero and at one side it will be positive and at other side it will be negative, as shown in Fig. 12 with low inflection at the middle of the heat exchangers. Whereas, for $p_A = 10$ kPa the gas parcel displacement is greater than the length of heat exchangers, therefore, the gas parcels over the cold heat exchanger only absorb heat and deliver it to the stack. Over the ambient heat exchanger, the gas only transfers heat from the stack to the heat exchanger. This phenomenon is shown in Fig. 14, where $q_c$ over the cold heat exchanger is always positive and $q_h$ over the ambient heat exchanger is always negative.

When the gas moves from the stack to cold heat exchanger, its pressure decreases gradually. At the left side of the cold heat exchanger, the pressure and temperature of the gas parcel are minimum, therefore, the temperature difference, and subsequently absorption of the heat from the left edge is maximum and its amount decreases at the right edge gradually. At the right side, $q_c$ increase, it is because of the gap between the stack and heat exchanger. The same phenomenon occurs for the ambient heat exchanger. Variation of velocity in one-half of the cycle is shown...
in Fig. 16. Concerning the driven pressure-amplitude, the graph is different at the inlet with Fig. 9 and the ideal standing wave.

The important point in Fig. 17 is that the pressure node is displaced. This note should be concerned in positioning of the plates. According to Fig. 8, that is at small amplitude as ideal standing wave, at first quarter of the cycle (0–\(\frac{\pi}{4}\) s) the pressure on the cold heat exchanger is lower than \(P_m\), therefore, the gas temperature becomes lower than the temperature of the cold heat exchanger, so the gas absorbs heat from it. This process repeats at last quarter of cycle. If the position of cold heat exchanger concerning ideal standing wave is set near the pressure node, then displacement of pressure node at large amplitude, reduces the cooling time of the cold heat exchanger to the lower than of the first and last quar-

![Fig. 7. Variation of the velocity profile during a short part of the cycle when the flow direction is reversed.](image)

![Fig. 8. Pressure distribution during half of a cycle, \(P_A = 1\) kPa, \(T_c = 297\) K, \(T_h = 300\) K, and \(y = 0.0001\) m.](image)

![Fig. 9. Velocity distribution during half of the cycle, \(P_A = 1\) kPa, \(T_c = 297\) K, \(T_h = 300\) K, and \(y = 0.0001\) m.](image)

![Fig. 10. Cooling load versus inlet dynamic pressure, \(T_i = 297\) K and \(T_h = 300\) K.](image)

![Fig. 11. Cooling load and rejected heat versus time, \(P_A = 10\) kPa, \(T_i = 297\) K, and \(T_h = 300\) K.](image)
ter time of the cycle, therefore, the gas cannot absorbs enough heat from the cold heat exchanger. Ke et al. [7] did not consider this point; they used large-scale plates and large inlet pressure ampli-
tude. In this case, it seems that there is a problem with the cold heat exchanger, as they omitted the cold heat exchanger and the temperature of the cold end of stack is considered for design and optimization.

Fig. 12. Energy flux density over stack and heat exchangers, $p_a = 1 \text{kPa}$, $T_r = 297 \text{ K}$, and $T_i = 300 \text{ K}$.

Fig. 13. Energy flux density over stack and heat exchangers based on the analytical solution, $p_a = 700 \text{ Pa}$, $T_r = 297 \text{ K}$, and $T_i = 300 \text{ K}$ [12].

Fig. 14. Energy flux density over stack and heat exchangers, $p_a = 10 \text{ kPa}$, $T_r = 297 \text{ K}$, and $T_i = 300 \text{ K}$.

Fig. 15. Energy flux density over stack and heat exchangers based on the analytical solution, $p_a = 7 \text{ kPa}$, $T_r = 297 \text{ K}$, and $T_i = 300 \text{ K}$ [12].

Fig. 16. Velocity distribution during half of a cycle, $p_a = 10 \text{ kPa}$, $T_r = 297 \text{ K}$, $T_i = 300 \text{ K}$, and $f = 200 \text{ Hz}$.

Fig. 17. Pressure distribution during half of a cycle, $p_a = 10 \text{ kPa}$, $T_r = 297 \text{ K}$, $T_i = 300 \text{ K}$, and $f = 200 \text{ Hz}$. 
Now, we do simulations with the condition discussed by Ke et al. [7]. They considered frequency equal to 100 Hz, so the length of the apparatus will be two times of the present work, and for pressure amplitude, they set the 15 kPa. Pressure and velocity distribution are shown in Figs. 18 and 19.

From the above figures, it is shown that the velocity and pressure node are displaced. In this model, the cooling load is 6 W and the rejected heat of the ambient heat exchanger is 58 W that is a large difference, therefore, the device does not work successfully. Concerning to the comparison of Figs. 8 and 18, it is clear that in Fig. 18, the pressure graph at time $2\tau/10$ s becomes near the $P_m$, therefore, the temperature different between the gas and the cold heat exchanger to be weak and cannot absorbs heat from it. In this situation, the cooling time of the cold heat exchanger is reduced to the lower than of the first and last quarter time of the cycle, therefore, the gas cannot absorb enough heat from the cold heat exchanger.

To explain the pressure displacement problem, we compare the COP of the simulation of Ke et al. [7] with Carnot COP.

The Carnot COP is:

$$\text{COP}_{\text{Carnot}} = \frac{T_c}{T_h - T_c} = \frac{297}{300 - 297} = 99.$$

Therefore, the performance of this case is 0.1% of Carnot COP. This performance for a refrigerator is very low.

In this paper, in the first case the frequency was set as 200 Hz, whereas in the second case of the Ke et al. [7] the frequency was set as 100 Hz. The frequency changes the length of the device because the length of the device is usually set equal to $\lambda/2$ or $\lambda/4$, where $\lambda = a/f$. To compare the pressure node displacement at $f = 100$ Hz and $f = 200$ Hz, we plot pressure distribution at $t = 8\tau/10$ s for $f = 100$ Hz and $f = 200$ Hz in Fig. 20. As this figure shows, the pressure node displacement for $f = 100$ Hz is more than $f = 200$ Hz.

Another point we considered is that the temperature difference of the heat exchanger affects the steady state solution. Fig. 21 shows the steady state solution versus different temperature of heat exchangers, $p_A = 10$ kPa and $f = 200$ Hz.
7. Conclusions

In this paper, the stack and heat exchangers of a thermoacoustic refrigerator at weak and large amplitude are simulated. The energy flux density over the stack and heat exchangers is compared with the analytical solution. This work has been performed with the cht-MultiRegionFoam solver in the open source CFD software OpenFOAM V.2.1.1, where we implemented cosine boundary condition to code. The results show that the input power and temperature of heat exchangers are affect on the steady state solution and successful operation of the refrigerator. As indicated, at weak pressure amplitude, the acoustic power cannot deliver waste heat to the ambient heat exchanger. At large amplitude, one should be careful about the position of the cold heat exchanger, as the displacement of the pressure node due to nonlinear effects reduces the cooling time of the cold heat exchanger, therefore, the parcel of gas cannot absorb enough heat from it. These subjects indicate that both heat exchangers must be considered in the simulation to check COP and successful operation of the apparatus.

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