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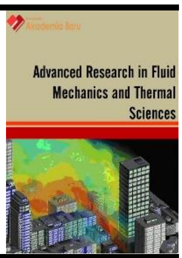

Thermoacoustic Energy Conversion Devices: Novel Insights

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ABSTRACT

Thermoacoustic engines and refrigerators have many advantages. They use environment-friendly working gases, their design is simple, and they can operate quietly. However, they have many design characteristics from geometric parameters and operating conditions. Besides this, they still have low efficiencies and performance. This paper summarizes important considerations of their design and presents the state-of-the-art developments in thermoacoustic energy conversion devices. This includes recent studies and designs of these devices towards more efficient thermoacoustic engines and refrigerators. New insights into the design of resonators, different power sources, different stack geometries and working mediums were considered. The challenges that face developments of thermoacoustic devices were also discussed. Far too little attention has been paid to looking at these devices comprehensively. In future research, the use of neural networks and metadata as optimisation methods could be a means of significantly increasing the performance of these devices. There is also abundant room for further progress in enhancing oscillatory heat transfer. This could be achieved by better design of the stacks and heat exchangers to suit the temperature variability inside the resonator. Moreover, further recommendations and studies were proposed for a better understanding of the interrelationship between the geometric parameters and operating conditions.

Keywords:
Thermoacoustics; engines; Refrigeration; Performance; Optimisation; Oscillatory heat transfer

1. Introduction
1.1 Background

There are many energy and environmental problems that face our world today. Performance optimization of energy-related systems becomes more important to meet energy needs with high efficiency. Besides that, problems of the diminution of the ozone layer and global warming produced by the harmful refrigerants in ordinary vapour compression cycles are facing the refrigeration and air conditioning area [1,2]. An emerging method for the clean technologies of refrigeration and engines is by implementing thermoacoustics. In recent years, there has been an increasing interest in

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thermoacoustics its significant advantages over other refrigeration and engine systems. These advantages include the use of environment-friendly working fluids, the continuous control of cooling capacity, the design simplicity, the ability to use waste energy and the quiet operation possibility [3,4]. It is now in the research and development stage, and it is predicted for observable spreading commercially [5].

1.2 Basic Operating Conditions and Geometric Parameters

Performance of thermoacoustic refrigerators and efficiency of thermoacoustic engines depend on some basic geometric parameters and operating conditions. Figure 1 shows the main thermoacoustic device components and dimensions for the heat exchanger, stack and resonator. The stack is described by its centre position (X_s), length (L_s) and porosity (B). The heat exchanger is described by its length (L_H for the hot heat exchanger and L_C for the cold heat exchanger), and its porosity and the resonator are described by the length (L_{res}) and the diameter (d). Cold and hot ducts are described by lengths $L_{C,\ duct}$, $L_{H,\ duct}$ respectively.

The main operating conditions are mean pressure (p_m), amplitude pressure (p_1) and frequency (f). The amplitude pressure is also called the oscillating pressure and it is the pressure above or below the mean pressure in the resonator. The drive ratio (D) is the ratio between amplitude pressure and the mean pressure.

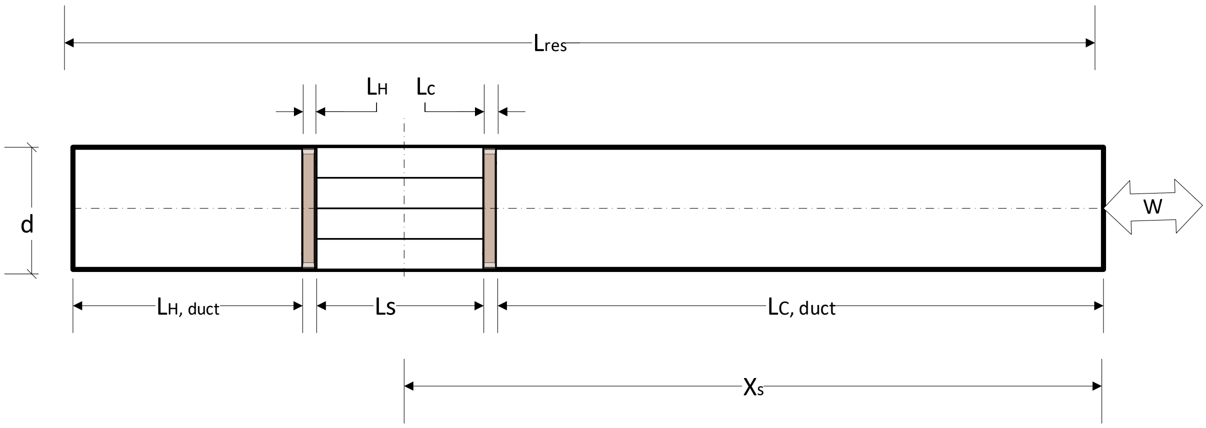

Fig. 1. The main thermoacoustic device geometric dimensions

A cooling effect (i.e. for thermoacoustic refrigerator) is obtained when work is consumed and there is an input power source such as loudspeakers and linear motors. Work is produced (i.e. for thermoacoustic engines) when there is a temperature difference between hot and cold heat exchangers. The position of hot and heat exchangers may be updated depending on design parameters

1.3 Working Principles

For sound waves, air particles oscillate around equilibrium positions, as shown in Figure 2. The disturbance flows to the tube end, but the particles propagate only in local places back and forth [2]. The parameter to describe this displacement is the gas amplitude displacement, $|X_1|$ and it is the gas parcel half excursion at one cycle. It is the ratio between the particle velocity amplitude $|u_1|$, and the wave angular frequency, ω as in Eq. (1). This is a consecutive action in a phenomenon called "Bucket Bridge". The gas parcel will do its cycle, and then it delivers the thermoacoustic effect to the next gas parcel. Figure 3 illustrates this phenomenon for three parcels A, B and C where the handling process is obvious [2].

$$|X_1| = \frac{|u_1|}{\omega} \tag{1}$$

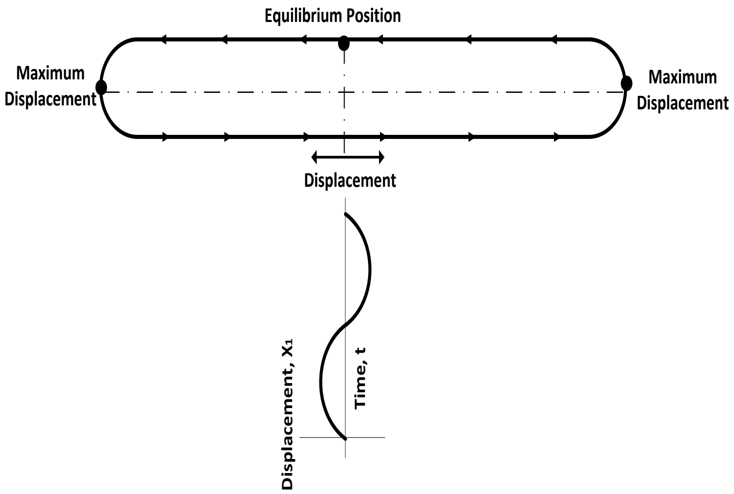

Fig. 2. An air particle vibrating about its equilibrium position

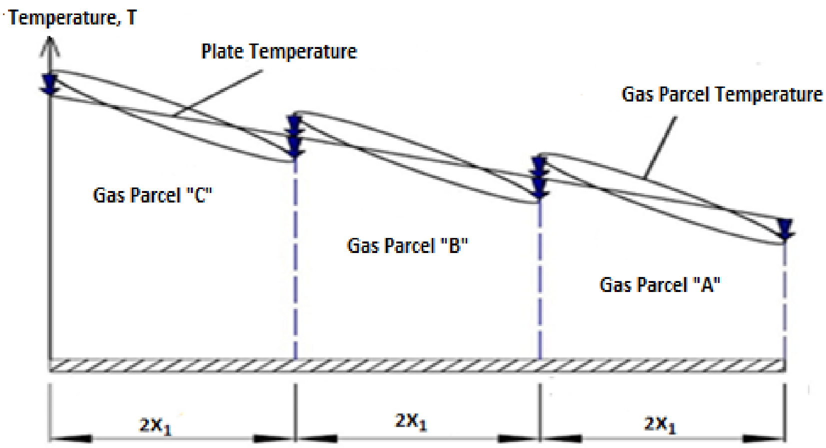

Fig. 3. Bucket bridge phenomenon [2]

1.4 Aims

This paper summarizes important considerations of the thermoacoustic energy conversion devices design and presents state-of-the-art developments towards more efficient thermoacoustic engines and refrigerators. New insights into the design of resonators, different sources of the input power, different stack geometries and working mediums were considered. The challenges that face the development of thermoacoustic devices were also discussed.

2. Thermoacoustic Device Components

2.1 Stack

In the stack, thermoacoustic effects take place, the acoustic work is consumed or produced, and the thermoacoustic cycle runs in the gas nearby its walls. The walls of the stack channel make a contact area between the working fluid and the solid surface which is vital to achieving the correct timing (i.e. time needed for the heat transfer). The stack geometries have a constant cross-section area across the flow direction of the stack. However, there should be a study to investigate the effect of different porosities in the same stack to suit the temperature variations [53]. The stack cross-section includes honeycombs, square tubes, pins arrays, parallel plates and spiral shapes as shown in Figure 4. The performance of the thermoacoustic stack is affected by many factors such as its material, its length, its position at the resonator, the pores shapes, and the porosity. Novel methods to optimise these parameters at the same time are discussed in section 4. The porosity (B) is known as the working fluid area to the total stack area [1]:

$$B = \frac{Ag}{A} \tag{2}$$

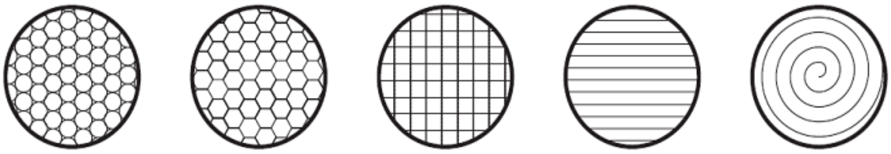

Fig. 4. Common geometries of the thermoacoustic stack

The stack porosity is dependent on the pores number, the surface and the pores thicknesses. The porosity of a parallel plate stack having a thickness of $2l$ and spacing between plates of $2y_0$ is:

$$B = \frac{y_0}{y_0 + l} \tag{3}$$

2.2 Working Gas

The working fluid is the sound wave propagation medium. Improving the characteristics of this medium affect performance and efficiency of thermoacoustic devices. Pure gases have Prandtl number 0.67 at the standard temperature and pressure. Lower Prandtl number values can be obtained for gases mixtures [1]. Figure 5 demonstrates how the working fluid affects the performance significantly [56].

The thermoacoustic stack values of thermal and viscous penetration depths provide how the working fluid parcels are reacting with the stack. The thermal penetration depth δ_k is the thickness of the working fluid layer where heat transfers at one-half cycle [1].

$$\delta_k = \sqrt{\frac{2k}{\omega \rho_m c_p}} \tag{4}$$

where k is the working fluid thermal conductivity, ω is the angular frequency, ρ_m is the working fluid mean density, and c_p is the working fluid isobaric heat capacity.

Viscous penetration depth δ_v is defined as the thickness of the working fluid layer where friction is effective at one-half cycle [1].

$$\delta_v = \sqrt{\frac{2\mu}{\omega \rho_m}} = \sqrt{\frac{2\nu}{\omega}} \tag{5}$$

where μ and ν are the working fluid dynamic and kinematic viscosities, respectively. The acoustic power dissipates at this layer thickness due to friction.

The viscous and thermal penetration depths are the important parameters to specify how the gas particles interact with the stack walls thermally. The designer should aim for higher thermal penetration depths for better heat transfer between the gas particles and the stack and lower viscous penetration depths so that the acoustic power is not dissipated. The Prandtl number is the ratio between the square of the thermal penetration depth to the square of the viscous penetration depth [1].

$$Pr = \frac{\mu c_p}{K} = \left(\frac{\delta_v}{\delta_k}\right)^2 \tag{6}$$

The acoustic power in the working fluid is the power that the acoustic propagating wave can produce, consume, or transmit. The acoustic power equals the pressure and particle velocity dot product [2].

$$\dot{W} = 0.5|p_1||u_1|\,cos(\Phi) \tag{7}$$

The acoustic power is the pressure and velocity dot product, so the phasing (Φ) between the phase angles (θ) of the pressure and the velocity is important. It can be also understood why increasing the input power increases the pressure and the velocity of the particle.

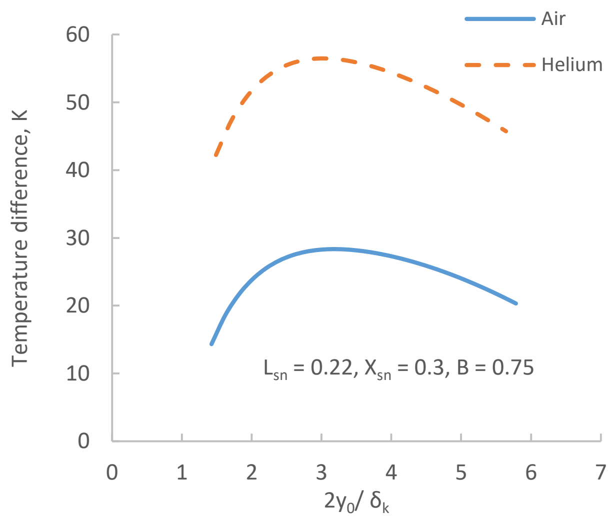

Fig. 5. Stack spacing to thermal penetration depth for Air and Helium effect on the temperature difference across the stack [56]

2.3 Frequency

The resonance frequency is important for high efficiency or performance of thermoacoustic devices. The frequency is affected by the overall resistance, inertance and compliance. So, the resonance frequency change may be explained that the stack interacts with the propagation of the sound wave and modifies the acoustic behaviour of the whole refrigerator. That is the main reason for the resonance frequency variation by changing the stack geometric parameters as shown in Figure 6. When looking at the inertance, impedance and compliance equations, it is found that these equations are strongly related to the gas density. From the universal gas law, gas density is a function of the temperature difference that varies with the stack geometric parameters. Increasing the temperature difference will increase the pressure and decrease the gas density, so the compliance, impedance and inertance of the system will decrease leading to an increase in the resonance frequency. Also, increasing the temperature difference across the stack affects the impedance which will be minimum, and the resistance will have the major role to impede the flow, as the inertance will be minimal due to the low gas density. Thus, the maximum energy flow can be obtained at the resonance frequency which will increase with the impedance decrease. Consequently, the resonance frequency is observed to increase as shown in Figure 6 for the obtained resonance frequency at different normalized stack positions through the resonator.

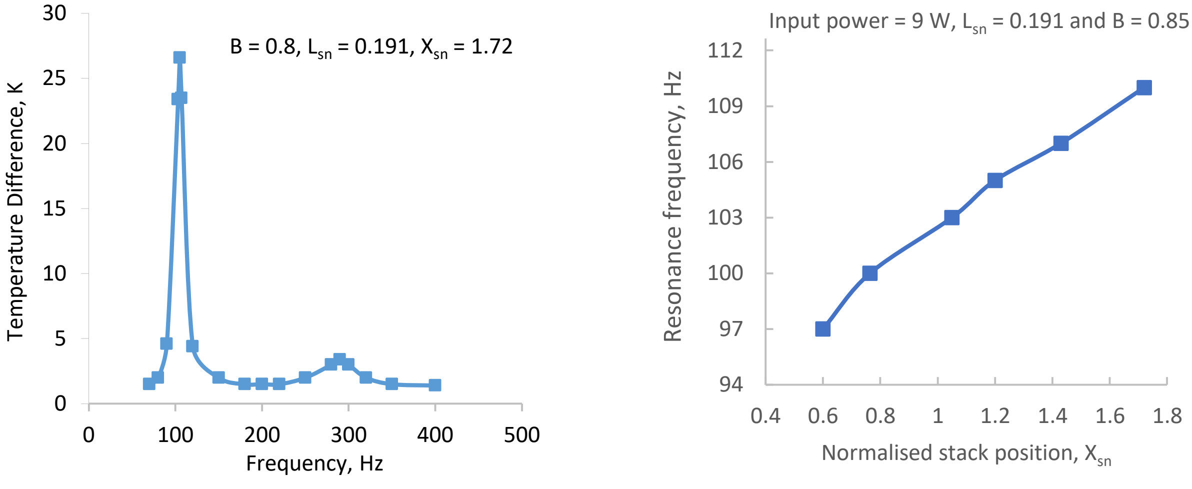

Fig. 6. (a) The frequency effect on the temperature difference across the stack (b) The change of the resonance frequency with the normalised stack positions through the resonator [53]

2.4 Amplitude Pressure

Figure 7 shows how amplitude pressure affects the temperature difference and performance in a thermoacoustic refrigerator [55]. The temperature difference begins with a low value at the first part of the curve shown in Figure 7(a) due to the weakness of pressure amplitude to make the change, then the increase of amplitude pressure increases the temperature difference until it reaches a maximum value that can be obtained by the increase of this value near a drive ratio of 3 %, after that the non-linear effects takes place. The amplitude pressure can make sensible changes to the working gas properties and lowers the thermal penetration depth. Increasing the pressure will increase the gas density and will change the gas properties, so it will decrease the gas thermal penetration depth (δ_k). The lower thermal penetration depth imposes a decrease in the heat transfer between the gas parcels and the stack plates as more gas parcels will be compressed and expanded without interacting with the plate and the temperature difference decreases again for the second part in the curve shown in Figure 7(a).

The consumed acoustic energy is proportional to the drive ratio, the input acoustic energy for a fixed mean pressure increases with the acoustic pressure increase which means lower Coefficient of performance. This is well demonstrated in Figure 7(b).

The maximum temperature difference occurs at a drive ratio equals 3%, but we are also concerned with Coefficient of performance. A drive ratio equals 2 % was chosen by Alamir and Elamir [55] to improve the Coefficient of performance and to account for the driver abilities to provide this drive ratio.

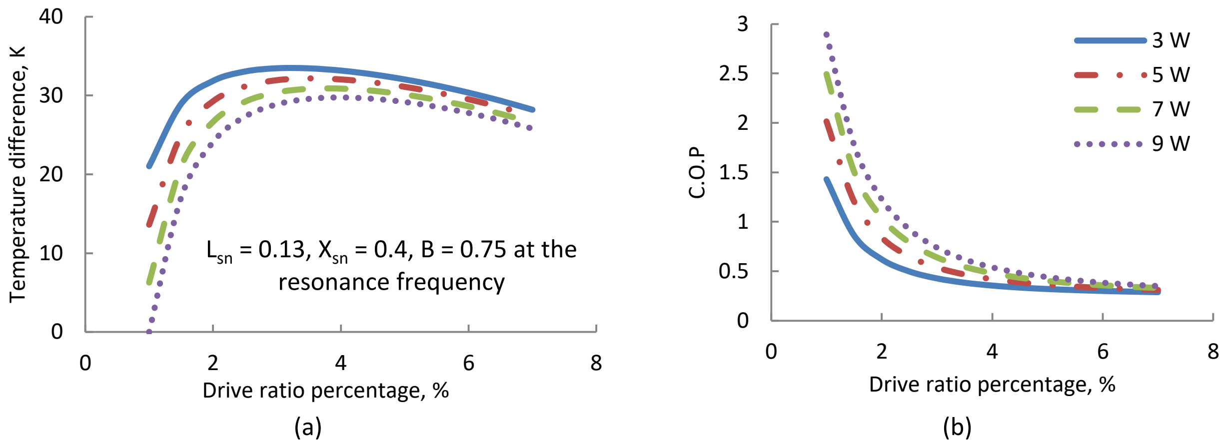

Fig. 7. The drive ratio effect on (a) Temperature difference (b) Coefficient of performance [55]

2.5 Heat Exchangers

Heat is transferred from or to the thermoacoustic refrigerators through two heat exchangers. The oscillating gas transfers the heat nearby the heat exchangers such that it forces the temperature gradient across the thermoacoustic stack permitting the occurrence of the thermoacoustic effect. The maximum displacement of the oscillating gas particles will occur in the temperature gradient direction. The fins of the heat exchanger that is longer than this displacement value do not exploit the additional length of the fin, so design restrictions of the two used heat exchangers are presented [1]. Many factors can affect heat transfer coefficients such as variability of the temperature difference, which vary considerably across the resonator as shown in Figure 8 [55].

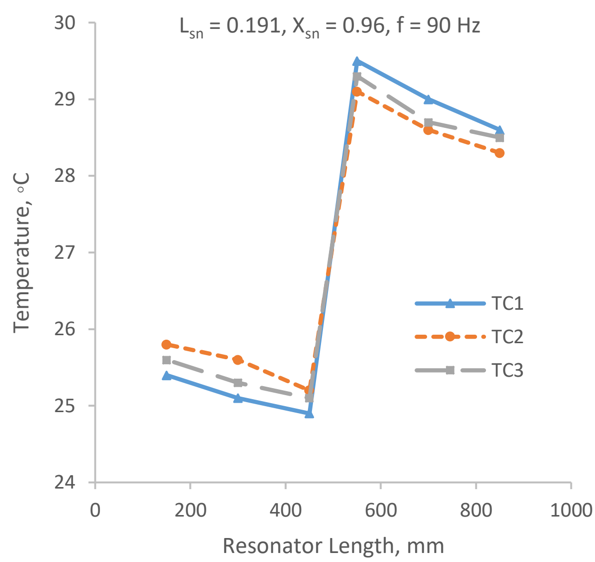

Fig. 8. The temperature distribution through the resonator at a frequency, f = 90 Hz [53]

2.6 The Driver

In thermoacoustic loudspeaker-based refrigerators, the system receives the acoustic energy by electro-dynamic drivers. The improvement of the efficiency of electro-acoustic conversion is important for these devices to improve the loudspeaker driven thermoacoustic refrigerators [1].

Thermoacoustic heat pumps can make use of waste heat as shown in Figure 9. A thermoacoustic engine can be built, then it feeds the acoustic wave by using the linear alternator to the refrigerator resonator. Adeff and Hofler [3] experimentally studied thermo-acoustically driven thermoacoustic refrigerator which made use of the solar thermal energy to drive a thermoacoustic engine. A cooling load of 2.5 W was produced with cold temperature of 5 °C and a temperature difference of 18 °C. Saechan [4] make use of the waste heat from cooking burner in his device. The maximum relative performance of 5.94 % was reached.

A heat engine to pump heat or produce spot cooling of specific microcircuit elements was tested experimentally by Symko *et al.,* [5]. Their system delivered 1 W cooling power and 10 °C temperature difference. A linear motor can be used directly as a driver for thermoacoustic refrigerator. For example, Trillium project by Keolian *et al.,* [7] used three linear motors.

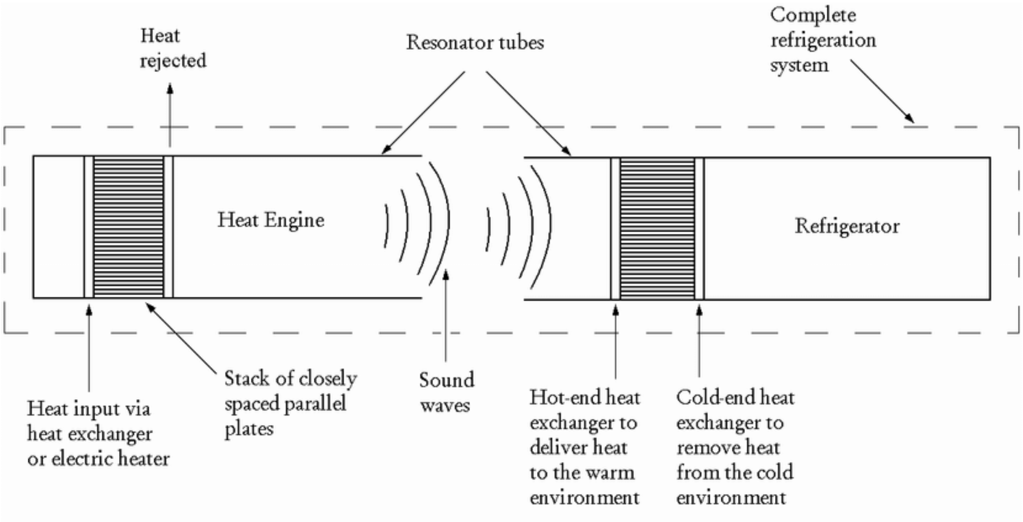

Fig. 9. A thermoacoustic refrigerator driven by a thermoacoustic engine [6]

3. Theoretical and Experimental Studies

Efforts for improving thermoacoustic systems theoretically or experimentally are significantly expanded. Many studies tried to generalize the design and optimization steps. Wetzel and Herman [17] developed a design algorithm for thermoacoustic heat pumps. The total consumed acoustic power was calculated as follows

$$\dot{W}_{tot} = \dot{W}_s + \dot{W}_{res} + \dot{W}_{hex} \tag{8}$$

The Los Alamos group, Ward *et al.,* [21] invented the DeltaEC program which is a computer program that can calculate the thermoacoustic device performance and helps for desired equipment design performance. DeltaEC is used in simulating thermoacoustic devices widely. DeltaEC was used as verification for the design and optimization algorithm. Srikitsuwan *et al.,* [19] presented an optimization design approach using genetic algorithms with a two-point boundary value problem. They demonstrated that this approach can maximize the performance of the thermoacoustic refrigerator. Arafa *et al.,* [22] investigated theoretically by using DeltaEC the operating conditions and stack geometry effect on the engine performance. The gas mean pressure was the main parameter that affects the onset temperature with 53.12 %.

Zoontjens *et al.,* [23] designed a thermoacoustic refrigerator driven by a loudspeaker and a temperature difference of 15 K was obtained. Prashantha *et al.,* [24] concluded that the 3 % drive ratio is better than operating at 2% drive ratio for helium gas and 10 W thermoacoustic refrigerator.

Sound wave interaction with the stack deepens the understanding of how to optimize the physical component of the refrigerator such as the stack and heat exchanger, especially at high-pressure amplitude. Namdar *et al.,* [25] used the OpenFOAM software to simulate the thermoacoustic refrigerator. The pressure node was observed to change due to non-linearity and the position of the cold heat exchanger was optimized. Wetzel and Herman [26] demonstrated the importance of the holographic interferometry and high-speed cinematography method for the development of the oscillatory heat transfer. Zhang *et al.,* [27] discussed the importance of the oscillatory flow field using particle image velocimetry (PIV) to determine the stack position and the resonator length which affect the working fluid velocity particles and thus the heat transfer process can be enhanced. Ke *et al.,* [28] simulated the best performance, which was at the dimensionless plate thickness ranged from 0.28 to 0.33 and the optimal heat exchanger length was close to the peak to peak amplitude of gas. Zink *et al.,* [29] made demonstration for thermally driven cooling using CFD analysis. Marx and Blanc-Benon [30] numerically simulated temperature distortion in the stack region and the optimal stack position was near the pressure antinode. Wu *et al.,* [31] presented a theoretical method for the optimum design of the stack in a standing wave thermoacoustic refrigerator. The optimal plate number increased with the increase of the working frequency and decreases with the increase of plates thickness. Lotton *et al.,* [32] provide an analytical approach to calculate the thermal quantities inside the stack which is overestimated in the linear theory of Thermoacoustics. Piccolo [33] numerically showed that the stack and heat exchanger act as the strong causes of irreversibility.

Many heat transfer studies of the oscillatory flow at the heat exchanger were presented. These studies results were correlated in terms of Prandtl number, Nusselt number and Reynolds number such as the work done by Kamsanam [34]. Tansim [35] derived temperature difference and acoustic work formulas and compared them with existing literature and a good agreement was observed. Abakr *et al.,* [36] experimentally found that the square wave has a better result on the temperature difference than the other acoustic waveforms. Campo *et al.,* [37] found that the gas mixture of helium-xenon gave the minimum Prandtl number of 0.12.

Ibrahim *et al.,* [38] demonstrated that the air in the enclosure controls the speaker's resonance frequency. Chinn [39] experimentally obtained resonance frequencies of a piezoelectric driven thermoacoustic refrigerator. Putra and Agustina [40] experimentally observed a temperature difference of 14.7 °C when testing different stack dimensions that varied with the temperature difference. Nsofor and Ali [41] recommended certain frequency and pressure for the system best performance. Setiawan *et al.,* [42] showed that the stack with thinner plate results in a larger temperature decrease. Sari *et al.,* [43] found that the farther stack from the sound source is the place giving the largest temperature difference. Nayak *et al.,* [44] found that the temperature difference increased with the increase of the input acoustic power. Alahmer *et al.,* [45] demonstrated that the difference between the theoretical and experimental results was due to losses which the theoretical model neglected. Akhavanbazaz *et al.,* [46] experimentally noticed the decrease of heat transfer with the increase of the gas blockage ratio. Allesina [47] presented a simple thermoacoustic refrigerator to give temperature difference of 24 °C was built and tested. Hariharan *et al.,* [48] studied twin thermoacoustic prime mover driven standing wave thermoacoustic refrigerator using helium at a pressure of 10 bar with two different mylar stack spacing. They demonstrated that their system raised the thermoacoustic refrigerator performance with the mylar stack with minimum spacing. Assawamartbunlue and Wantha [49] experimentally studied the resonance frequency change due to the mean pressure. The results indicate that the optimal operating frequency using helium is at 6 bar and 490 Hz which is 20% away from the frequency at 1 bar. Wantha and Assawamartbunlue [50] experimentally showed that the resonance frequency changed with the increase and the decrease of the back-volume size which changes the air stiffness in the back volume. Tartibu [51] experimentally found that the stack near the pressure antinode had the maximum performance.

4. Optimisation Algorithms

The stack length, position, and porosity affect the overall obtained cooling load and the required input acoustic power as shown in Eq. (9) and Eq. (10) for the normalized cooling power $\dot{Q}_{cn}$ and the normalized acoustic power $\dot{W}_n$ respectively. The normalized parameters in Eq. (9) and Eq. (10) are shown in Table 1. The coefficient of performance (C.O.P) is defined as the ratio of the obtained cooling power to the consumed power.

$$\dot{Q}_{cn} = -\frac{\delta_{kn} D^2 \sin(2X_{sn})}{8\gamma\Lambda(1+\sigma)}\left[\frac{\Delta T_{mn}\tan(X_{sn})}{L_{sn}B(\gamma-1)}\frac{1+\sqrt{\sigma}+\sigma}{1+\sqrt{\sigma}} - \left(1+\sqrt{\sigma}-\sqrt{\sigma}\delta_{kn}\right)\right] \tag{9}$$

$$\dot{W}_n = \frac{\delta_{kn} L_{sn} D^2}{4\gamma}(\gamma-1)B\cos^2(X_{sn})\left(\frac{\Delta T_{mn}\tan(X_{sn})}{\Lambda L_{sn}B(\gamma-1)(1+\sqrt{\sigma})} - 1\right) - \frac{\delta_{kn} L_{sn} D^2}{4\gamma}\frac{\sqrt{\sigma}\sin^2(X_{sn})}{\Lambda B} \tag{10}$$

where, $$\Lambda = 1 - \sqrt{\sigma}\,\delta_{kn} + \frac{1}{2}\sigma\,\delta^2{}_{kn} \tag{11}$$

Table 1
Normalized parameters of the thermoacoustic refrigerator

Normalized design requirements		Geometric parameters	
Normalized cooling load	$\dot{Q}_{cn} = \frac{\dot{Q}_c}{P_m a A}$	Normalized stack length	$L_{sn} = \frac{L_s}{\lambda/2\pi}$
Normalized acoustic power	$\dot{W}_n = \frac{\dot{W}}{P_m a A}$	Normalized stack position	$X_{sn} = \frac{X_s}{\lambda/2\pi}$
Drive ratio	$D = \frac{P_0}{P_m}$	Blockage ratio or porosity	$B = \frac{y_0}{y_0 + l}$
Normalized temperature difference	$\Delta T_{mn} = \frac{\Delta T_m}{T_m}$	Normalized thermal penetration depth	$\delta_{kn} = \frac{\delta_k}{2y_0}$

Recent studies focused on traditional optimisation algorithms to show the performance of thermoacoustic refrigerators. However, a comprehensive look at thermoacoustic devices is required. This should consider the devices as a whole piece. For example, some optimisation algorithms focused on geometric parameters and left important operating conditions such as resonance frequency. Zolpakar *et al.,* [20] presented experiments to validate the Multi-Objective Genetic Algorithm (MOGA) optimization approach results. Optimized dimensionless stack length of 0.29 and the optimized stack porosity of 0.72 for a thermoacoustic refrigerator were obtained. The resonance frequency was not optimized. Tijani [1] experimentally studied a thermoacoustic refrigerator driven by a loudspeaker. A simplified flow chart to design this refrigerators type was done based on the effect of dimensionless stack lengths and positions on the performance. Babaei and Siddiqui [18] investigated a general and effective optimization algorithm for thermoacoustic devices. The energy balance and entropy balance on the thermoacoustic device was the new algorithm basis to refine the optimization process.

However, problems are found with these optimisation algorithms. For example, they could disregard optimising important operating conditions such as the resonance frequency and not compromise temperature difference and performance [54,55]. Other problems include the theoretical background used as the basic linear theory equations developed shown in Eq. (9) and Eq. (10) and are usually used incorrectly. They can be different from the equations of real systems with different components. Other problems could be the incapability to track temperature variations [53].

Artificial neural networks (ANNs) are mathematical models that imitate the working principles of the human nervous system. There have been many applications of ANNs over the last thirty years in classifying, pattern recognition, regression and forecasting. They have been used in the environment, health and food sciences [57-61]. There is a new trend of using ANN for optimising thermal devices generally and thermoacoustic devices particularly. Using ANNs for thermoacoustic applications could help to model complex problems. For example, Abd Elaziz *et al.,* [62] showed that modelling oscillatory heat transfer using ANNs could give a high accuracy reaching to 0.98. Oscillatory heat transfer enhancement was also done by implementing an ANN approach by Rahman and Zhang [63]. The model can map the implicit relationship accurately. There are new studies that consider some parameters of thermoacoustic devices simultaneously for optimization [64-65]. Alamir [66] showed that the cooling temperature and performance (i.e. in case of thermoacoustic refrigerators) can also be predicted with high accuracy reaching 0.97. It was further suggested that this method can be used for modelling the power output and efficiency of thermoacoustic engines. The use of ANN algorithms can help develop these devices by considering the performance of the device in light of all its design parameters (i.e. operating conditions and geometric parameters) without the focus on a limited number of parameters.

5. Conclusions

The current paper presented some important insights into improving the performance of thermoacoustic devices. Design of thermoacoustic device components such as stack geometries, drivers, working fluids and heat exchanger can affect the performance and efficiency significantly. This review showed that each of these components has optimal geometric parameters and operating conditions. Given a large number of parameters needing optimization simultaneously, a wide range of proposed design optimization methodologies can help to improve the performance of these devices. Besides geometric parameters of thermoacoustic devices, there is a need to optimize operating conditions such as frequency, mean pressure and amplitude pressure of the driver. Implementation of neural networks in this area can be a particularly efficient solution rather than looking at some design parameters and neglecting others. Some recommendations were also provided for optimizing each parameter of the operating conditions and geometric variables such as the resonance frequency. For optimal thermoacoustic device performances and efficiencies, further progress is needed in designing novel geometries of the stack and the heat exchangers to take into account the temperature variability in thermoacoustic devices. This could be an important area towards enhancing oscillatory heat transfer.

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