Study of a Thermoacoustic Refrigeration System with a Spiral Rubber Stack

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Abstract. Thermoacoustic refrigeration technology converting sound energy into thermal energy offers environmental benefits including zero risks of ozone depletion or global warming potential due to the absence of hazardous refrigerants in the process. This study aims to investigate the effects of a spiral rubber stack on the cooling performance of a thermoacoustic refrigeration system made from an acrylic resonator tube. Air is used as a working gas. The results show that as time progresses, the temperature on the hot side of the stack increases, while that on the cold side of the stack reduces, leading to an increase in temperature difference across the stack. The findings of this study can potentially further expand knowledge and scientific experimental data in the field of thermoacoustic refrigeration.

1. Introduction
In the early 20th century, the CO₂ refrigerant was widely used in refrigeration and air-conditioning industry due to its higher safety compared to other substances such as ammonia and sulphur dioxide. However, given the drawbacks of CO₂ in high-pressure and high-temperature applications, the CO₂ refrigerant was replaced by non-toxic, non-flammable and non-corrosive chlorofluorocarbons in 1930s and 1940s [1]. Approximately 30-40 years later, it was found that chlorofluorocarbons significantly caused ozone depletion and also greatly contributed to global warming issues [2,3]. Hydrochlorofluorocarbons and hydrofluorocarbons were later introduced and replaced chlorofluorocarbons. However, hydrochlorofluorocarbons consisting of chlorine molecules has ozone depletion potential and both refrigerants also contribute to global warming potential [4]. As a result, a number of cooling technologies including adsorption refrigeration, absorption refrigeration, thermoelectric refrigeration and thermoacoustic refrigeration have been introduced and developed [5-8]. Among others, a thermoacoustic refrigeration system offers several benefits including low cost and an absence of refrigerants. A thermoacoustic refrigeration system consists of a loudspeaker, a resonator filled with a gas, a stack, and two heat exchangers, one of which is placed on the cold side and the other one on the hot side of the stack. The working principle of an thermoacoustic refrigeration system is similar to a heat pump system where work input is required. The detailed working principle of a thermoacoustic refrigeration is well explained by Swift [9] and Wheatly et al. [10]. In short, the system converts the acoustic energy to cooling power by oscillating gas particles inside the resonator using an acoustic standing wave. Gas particles oscillate, expand, and compress, creating temperature differences as heat is transferred from one end of the resonator to the other. The stack is installed in the path of this wave to store heat, and heat exchangers are used to extract the thermal energy for further use [11-16].
A number of experimental studies have been conducted in order to shed light on the effects of designed parameters on the cooling performance of the thermoacoustic refrigeration system [11-16]. Tasnim et al. [11] conducted an experiment on a Corning Celcor ceramic stack in a Pyrex resonator tube filled with atmospheric air. The stack length was 4 cm. A frequency of 350 Hz was used. When the stack was placed 5 cm away from a resonator end, the hot side temperature of the stack increased from 27 °C to 45 °C, while the cold side temperature of the stack decreased from 23 °C to 13.8 °C. Tijani et al. [12] experimentally studied a POM-Ertacetal resonator in a 10-bar pressurized condition. A parallel circular stack made from mylar sheets and fishing lines with the plate spacing of 0.3 mm was used. Helium as a working gas was supplied to the resonator. Given a very high average pressure applied, the very low temperature of -65 °C was obtained. Hariharan et al. [13] used parallel and spiral stacks made from mylar and photographic film with two gap spaces of 0.4 mm and 0.8 mm in their investigation. Helium was used as a working fluid, and the acoustic frequency of 460 Hz was applied. They reported that the photographic film stacks for both gap spaces achieved higher temperatures at the hot side of the stacks when compared with mylar stacks because the photographic film has greater thermal conductivity and specific heat capacity. The findings also showed that both the photographic film stack and the mylar stack at the gap space of 0.4 mm gave higher temperature differences across the stacks than those at the gap space of 0.8 mm. The system with the mylar stack with the gap space of 0.4 mm showed the greatest temperature difference of 16 °C. Wantha and Assawamarbunluea [14] studied the effects of the resonator on the thermoacoustic refrigeration performance. A circular spiral mylar stack was placed inside an acrylic resonator filled with air. The results showed that the length of the resonator affected the temperature difference across the stack. Allesina [15] conducted an experiment of thermoacoustic refrigeration using a rolled film stack and a parallel plate stack in a circular shape. The resonator material was PVC with the fixed length of 225 mm, and the working fluid was air. At the working frequency of 383 Hz and the resonator diameter of 36 mm, it was reported that the temperature difference across the rolled film stack was higher than that across the parallel plate stack by 35%. However, when the resonator diameter reduced to 22 mm, the temperature difference significantly increased. Putra and Agustina [16] experimentally studied the effects of voltage input of a loudspeaker and the stack plate thickness on the temperature difference across the stack. The parallel circular acrylic stacks with the thicknesses of 0.15 mm, 0.5 mm, and 1 mm were used. The experiment was conducted in an acrylic resonator. The findings showed that at 9 voltage peak-to-peak and 0.5 mm thickness, the maximum temperature difference of 14.8 °C was achieved.

Although the temperature difference across the stack has been investigated in many aspects with a number of stack and resonator materials and with different working gases, there is still a lack of experimental data and understanding of a thermoacoustic refrigeration system consisting of a rubber stack. Consequently, this study aims to investigate the temperature difference across a rubber spiral stack in an acrylic resonator tube. Air is utilized as a working gas. It could not be emphasized enough that this is the first study in which the rubber stack is experimentally investigated. Designed parameters of the thermoacoustic refrigeration system will be explained in Section 2. An experimental setup will be described in Section 3. In Section 4, results of the experiment will be reported and discussed. A brief conclusion will be drawn in Section 5.

2. Designed parameters

A thermoacoustic refrigeration system mainly consists of an acoustic resonator with the length of \( L \) and the inner diameter of \( D_i \), an acoustic driver or a loudspeaker with a frequency of \( f \), and a stack with the stack length of \( L_s \). The acoustic driver is installed on one end of the resonator, while the other end of the resonator is kept closed. The stack is placed inside the resonator with the distance between the mid length of the stack and the acoustic driver of \( X_s \). \( \lambda \) representing wavelength of the standing wave is the ratio of the speed of sound \( (a) \) to the frequency of the acoustic driver as shown in equation (1).

\[
\lambda = \frac{a}{f}
\]
In the current study, an acoustic driver frequency of 345 Hz is selected. The sound speed in air is approximately 345 m/s. Thus, the wavelength from equation (1) is equal to 1000 mm. Based on previous studies [17-19], the length of the resonator is set at half wavelength. As a result, the resonator length in this study is set at 500 mm. A normalized stack length ($L_{sn}$) and a normalized stack position ($X_{sn}$) are also given in equations (2) and (3), respectively.

$$L_{sn} = \frac{2\pi f}{a} L_s$$  \hspace{1cm} (2)  

$$X_{sn} = \frac{2\pi f}{a} X_s$$  \hspace{1cm} (3)  

The length of the stack is set at 40 mm. According to equation (2), the normalized stack length is 0.25. Given that there is reportedly a relationship between the normalized stack length and the normalized stack position which affects the performance of the refrigeration system [20], the normalized stack position of 0.22 is chosen for this study. From equation (3), the corresponding stack position is 35 mm away from the acoustic driver.

Apart from the length and the position of the stack, the gap space between stack layers also plays a major role and affects the performance of the system. The gap space links to two characteristic length scales namely, thermal penetration depth ($\delta_k$) and viscous penetration depth ($\delta_v$). The thermal penetration depth represents the distance between the stack layers which allows heat diffusion through the gas, while the viscous penetration depth relates to momentum diffusion [19,20]. As shown in equations (4) and (5), thermal penetration depths and viscous penetration depths depend on gas properties including thermal conductivity of a gas ($k$), dynamic viscosity of a gas ($\mu$), density of a gas ($\rho$), constant specific heat of a gas ($c_p$), and angular frequency of the wave ($\omega = 2\pi f$). Based on the previous studies [19,20], the gap space between the stack layers should range from $2\delta_k$ to $4\delta_k$. Moreover, the ratio between the thermal penetration depth and the viscous penetration depth is equal to square root of Prandtl number ($\sigma$) as shown in equation (6).

$$\delta_k = \frac{2k}{\rho c_p \omega}$$  \hspace{1cm} (4)  

$$\delta_v = \frac{2\mu}{\rho \omega}$$  \hspace{1cm} (5)  

$$\sigma = \left(\frac{\delta_k}{\delta_v}\right)^2$$  \hspace{1cm} (6)  

Atmospheric air at standard temperature and pressure is utilized as the working gas ($k = 0.025$ W/m-K, $\rho = 1.3765$ kg/m$^3$, $c_p = 1.005$ kJ/kg-K, $\mu = 1.849 x 10^{-5}$ kg/m-s). The thermal penetration depth is 0.129 mm, leading to the allowable range of the gap space between 0.258 mm and 0.516 mm ($2\delta_k$ to $4\delta_k$). In the current experiment, the gap space of 0.45 mm is selected. Also, the viscous penetration depth is 0.1113 mm, corresponding to the Prandtl number of 0.745.

3. Experimental setup

In this study, a circular acrylic resonator with the inner diameter of 140 mm was fabricated along with two acrylic adjustable shafts. One adjustable shaft was attached with an acoustic driver and the other one was used as a solid wall at the other end. Based on the designed parameters, the resonator length was adjusted to 500 mm. A spiral stack was fabricated by wrapping a rubber sheet around a small PVC rod as shown in figure 1. The gap space of 0.45 mm between the rubber layers was made by inserting fishing lines with a uniform diameter of 0.45 mm between each rubber layer. The stack size is equal to the inner diameter of the resonator. The stack was placed inside the resonator with the distance between the mid length of the stack and the acoustic driver of 35 mm. Two thermocouples were inserted on each
side of the stack and also connected to a datalogger on the other end as illustrated in figure 2. A function generator and an amplifier were used to deliver sound input at the preset frequency of 345 Hz. It should be noted that the experiment was conducted at the standard temperature and pressure, and the resonator was not covered with any insulation sheets.

![Figure 1. A spiral rubber stack.](image)

**Figure 1.** A spiral rubber stack.

![Figure 2. A schematic diagram of the experimental setup.](image)

**Figure 2.** A schematic diagram of the experimental setup.

### 4. Results and discussions

As aforementioned, two thermocouples were inserted to the resonator on each side of the stack. The measured temperatures shown in figure 3 were determined from the average value of two thermocouples on each side of the stack. The experiment lasted 60 minutes. The results showed that as time progressed, the temperature on the hot side of the stack continuously increased, and the temperature on the cold side of the stack gradually decreased. At the 60th minute, the temperature of the hot side increased by 3.45 °C, while the temperature of the cold side decreased by 2.40 °C. Therefore, the temperature difference across the stack was obtained as shown in figure 4. After one hour of applied sound input, the maximum temperature difference of 8.82 °C was acquired.
Figure 3. Measured temperatures at the hot side and the cold side of the spiral rubber stack as functions of time.

There are several reasons to explain why the temperature drop at the cold side of the stack was smaller than those reported in previous studies. Firstly, the resonator size affects the cooling performance. Based on the study from Allesina [15], it was reported that when the resonator size decreased, the temperature on the cold side of the stack significantly dropped, and the temperature of the hot side greatly increased, resulting in greater temperature difference. Secondly, the length of the resonator impacts on the thermoacoustic refrigeration system. Similar to certain previous studies [17-19], the current study utilized the half wavelength method on the resonator tube. However, although both half wavelength and quarter wavelength resonators can be used in thermoacoustic refrigeration systems, it has been reported that the quarter wavelength resonator provided better performance because losses on the resonator walls were significantly reduced [8]. Thirdly, based on the performance curve showing the relationship between the normalized stack length and normalized stack position [20], the stack length and the stack position which are inter-dependent affect the temperature difference across the stack. Lastly, a working fluid in the thermoacoustic refrigeration system plays a significant role because cooling power depends on sound velocity in a working fluid. Although air is widely used in many studies due to its availability, helium offers greater cooling performance due to its properties [8]. The sound velocity in helium is higher than that in air. In addition, thermal conductivity of helium is also greater than that of air, resulting...
in heat transfer enhancement and an increase in thermal penetration depth. In short, a quarter wavelength resonator with smaller tube size, properly designed stack length and stack position, and helium as a working gas can potentially enhance the cooling performance of the spiral-rubber-stack thermoacoustic refrigeration system.

5. Conclusions
In this study, a thermoacoustic refrigeration system with an air-filled acrylic resonator and a spiral rubber stack was designed, built, and experimentally tested. The resonator length and the stack position were determined based on the selected values of the acoustic driver’s frequency and the stack length. The stack was fabricated using the rubber sheet, the PVC rod and the fishing lines. The findings show an increase in the temperature on the hot side of the stack and a decrease in the temperature on the cold side of the stack, leading to greater temperature difference across the stack as time progresses.

6. References
[1] Kim MH, Pettersen J and Bullard CW 2004 Fundamental process and system design issues in CO2 vapor compression systems Progress in Energy and Combustion Science 30 119–174
[2] Glynn S 2002 Constructing a selection environment: competing expectations for CFC alternatives Research Policy 31 935-946
[3] Bolaji BO and Huan Z 2013 Ozone depletion and global warming: Case for the use of natural refrigerant–a review Renewable and Sustainable Energy Reviews 18 49-54
[4] Zink F, Vipperman JS and Schaefer LA 2010 Environmental motivation to switch to thermoacoustic refrigeration Applied Thermal Engineering 30 119-126
[5] Srikhirin P, Aphornratana S and Chungpaibulpatana S 2001 A review of absorption refrigeration technologies Renewable and sustainable energy reviews 5 343-372
[6] Anyanwu EE 2003 Review of solid adsorption solar refrigerator I: an overview of the refrigeration cycle Energy Conversion and Management 44 301–312
[7] Rawat MK, Chattopadhyay H and Neogi S 2013 A review on developments of thermoelectric refrigeration and air conditioning systems: a novel potential green refrigeration and air conditioning technology International Journal of Emerging Technology and Advanced Engineering 3 362-367
[8] Zolpakar NA, Mohd-Ghazali N and El-Fawal MH 2016 Performance analysis of the standing wave thermoacoustic refrigerator: A review Renewable and sustainable energy reviews 54 626-634
[9] Swift GW 1988 Thermoacoustic engines The Journal of the Acoustical Society of America 84 1145-1180
[10] Wheatley J, Hofler T, Swift GW and Migliori A 1985 Understanding some simple phenomena in thermoacoustics with applications to acoustical heat engines American journal of physics 53 147-162
[11] Tasnim S, Mahmud S and Fraser R 2011 Measurements of thermal field at the stack extremities of a standing wave thermoacoustic heat pump Frontiers in Heat and Mass Transfer 2
[12] Tijani MEH, Zeegers JCH and De Waele ATAM 2002 Construction and performance of a thermoacoustic refrigerator Cryogenics 42 59-66
[13] Harirharan NM, Sivashanmugam P and Kasthurirengan S 2013 Experimental investigation of a thermoacoustic refrigerator driven by a standing wave twin thermoacoustic prime mover. International journal of refrigeration 36 2420-2425
[14] Wantha C and Assawarnartbunluea K 2012 The impact of the resonance tube on performance of a thermoacoustic stack Frontiers in Heat and Mass Transfer 2
[15] Allesina G 2014 An experimental analysis of a stand-alone standing-wave thermoacoustic refrigerator International Journal of Energy and Environmental Engineering 5 1-9
[16] Putra N and Agustina D 2013 Influence of stack plate thickness and voltage input on the performance of loudspeaker-driven thermoacoustic refrigerator Journal of Physics: Conference Series 423 012050
[17] Lotton P, Blanc-Benon P, Bruneau M, Gusev V, Duffourd S, Mironov M and Poignand G 2009 Transient temperature profile inside thermoacoustic refrigerators International Journal of Heat and Mass Transfer 52 4986-4996

[18] Poignand G, Lihoreau B, Lotton P, Gaviot E, Bruneau M and Gusev V 2007 Optimal acoustic fields in compact thermoacoustic refrigerators Applied Acoustics 68 642-659.

[19] Akhavanbazaz M, Siddiqui MK and Bhat RB 2007 The impact of gas blockage on the performance of a thermoacoustic refrigerator Experimental thermal and fluid science 32 231-239

[20] Tijani MEH, Zeegers JCH and De Waele ATAM 2002 Design of thermoacoustic refrigerators Cryogenics 42 49-57.