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Experimental Investigation on the Optimal Operating Frequency of a Thermoacoustic Refrigerator

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Abstract**—**This paper presents effects of the mean operating pressure on the optimal operating frequency based on temperature differences across stack ends in a thermoacoustic refrigerator. In addition to the length of the resonance tube, components of the thermoacoustic refrigerator have an influence on the operating frequency due to their acoustic properties, i.e., absorptivity, reflectivity and transmissivity. The interference of waves incurs and distorts the original frequency generated by the driver so that the optimal operating frequency differs from the designs. These acoustic properties are not parameters in the designs and be very complicated to infer their responses. A prototype thermoacoustic refrigerator is constructed and used to investigate its optimal operating frequency compared to the design at various operating pressures. Helium and air are used as working fluids during the experiments. The results indicate that the optimal operating frequency of the prototype thermoacoustic refrigerator using helium is at 6 bar and 490Hz or approximately 20% away from the design frequency. The optimal operating frequency at other mean pressures differs from the design in an unpredictable manner, however, the optimal operating frequency and pressure can be identified by testing.

Keywords**—**Acoustic properties, Carnot's efficiency, Interference of waves, Operating pressure, Optimal operating frequency, Stack performance, Standing Wave, Thermoacoustic refrigerator.

I. Introduction

THE thermoacoustic refrigerator is another cooling technology capable of converting acoustic energy into heat energy. It utilizes acoustic energy input to transfer heat from a low-temperature source to a high-temperature source. The standing-wave thermoacoustic refrigerator has few moving parts and operates using no environmentally harmful working fluids. These are the primary benefits of the thermoacoustic refrigerator. It can therefore be used in cooling electronic components, vehicles, space shuttles, and household refrigerators. Nevertheless, the applications of this technology are limited due to low performance of the device and the need for high amplitude of sound waves. The principles of thermoacoustic systems can be found in [1].

A standing-wave thermoacoustic refrigerator is comprised of four basic components: a stack, hot and cold heat exchangers at both ends of the stack so that heat can be removed from the thermoacoustic resonance tube, a resonance tube, and a loudspeaker or a driver to generate a standing wave within the resonance tube. Several experimental investigations have shown that the performance of the thermoacoustic system depends upon many factors, such as the stack, the drive ratio, the working fluid, and the resonance frequency [1], [2]. Wetzel [3] showed calculations of the coefficient of performance of the stack in a haft-wavelength of a standing-wave thermoacoustic refrigerator. They show 0.4-0.5 of Carnot's efficiency for the stack, while the coefficient of performance for the commercial refrigerator was in the range of 0.33-0.5. Nsofor et al. [4] studied the performance of a thermoacoustic refrigerator with some critical operating parameters. The results show that the frequency has influence performance of thermoacoustic refrigerator systems, but high pressure in the system does not necessarily result in a higher cooling load. Tijani [5] measured the performance of a thermoacoustic refrigerator with the mixing gas. The results show that optimal ratio of gases had affected with the performance of systems. Assawamartbunlue [6] found that the resonance frequency is a factor that influenced the temperature differences across the stack; however, they found that actual operating frequency of the thermoacoustic refrigerator was away from the design. Thus, the resonance tube must be prolonged to compensate for some acoustic effects to maintain the design frequency.

The objective of this paper is to investigate the effects of mean operating pressures on the optimal operating frequency of the thermoacoustic refrigerator. A prototype of the standing-wave thermoacoustic refrigerator was assembled for experimentations. To minimize the interactive effects of heat exchanger performance, the hot and cold heat exchangers were removed from the experiments.

II. Thermoacoustic Theory

The thermoacoustic effects occur in the stack through the interaction of sound waves and the stack plate. As a gas parcel moves toward the pressure antinode, the temperature of the gas is increased because of acoustic compression and the excess heat is transferred to the stack plate. When a gas parcel moves back to the pressure node and expands, the temperature of the gas parcel reduces lower than the stack plate. The gas parcel gains heat from the plate and starts the cycle again. Detailed explanation of the thermoacoustic effect can be obtained from other sources [1], [2].

To develop a standing-wave thermoacoustic refrigerator, the basic equation and theory for linear thermoacoustic systems is well covered by [1], [2]. The thermal and viscous penetration depths are given by:

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$$\delta_k = \sqrt{\frac{2K}{\rho C_p \omega}} \tag{1}$$

$$\delta_v = \sqrt{\frac{2\mu}{\rho\omega}} \tag{2}$$

where $\omega = 2\pi f_0$ is the angular frequency of the sound wave, f_0 is the design frequency, K is the thermal conductivity. μ, ρ and C_p are the viscosity, density, and isobaric specific heat of the gas, respectively. The normalized cooling power,Q_{cn}, acoustic power, and W_n are given by [2].

$$Q_{cn} = -\left[\frac{\delta_{kn} D^2 sin(2X_{sn})}{8\gamma(1+\sigma)\left(1-\sqrt{\sigma}\delta_{kn}+\frac{1}{2}\sigma\delta_{kn}\right)}\right] \times \left[\frac{\Delta T_{mn} tan(X_{sn})}{(\gamma-1)BL_{sn}} \times \frac{1+\sqrt{\sigma}+\sigma}{1+\sqrt{\sigma}} - \left(1+\sqrt{\sigma}-\sqrt{\sigma}\delta_{kn}\right)\right] \tag{3}$$

$$W_n = -\left[\frac{\delta_{kn} D^2 L_{sn} B(\gamma-1) cos^2(2X_{sn})}{4\gamma}\right] \times \left[\frac{\Delta T_{mn} tan(X_{sn})}{BL_{sn}(\gamma-1)(1+\sqrt{\sigma})\left(1-\sqrt{\sigma}\delta_{kn}+\frac{1}{2}\sigma\delta_{kn}^2\right)} - 1\right] - \left[\frac{\delta_{kn} D^2 L_{sn}}{4\gamma} \frac{\sqrt{\sigma} sin^2(X_{sn})}{B\left(1-\sqrt{\sigma}\delta_{kn}+\frac{1}{2}\sigma\delta_{kn}^2\right)}\right] \tag{4}$$

where $\delta_{kn} = \delta_k / y_o$ is the normalized thermal penetration depth, σ is gas Prandtl number, γ is ratio of specific heat, $B = y_o/(y_o + l)$ is the blockage ratio of the stack, y_o is half of the distance between the stack layers, l is half of the thickness of the stack layers, $\Delta T_{mn} = \Delta T_m / T_m$ is the normalized temperature difference, X_{sn} and L_{sn} are normalized stack position and length, respectively, and $D = P_o/P_m$ is the drive ratio. As shown in (1)-(4), the design of the thermoacoustic stack was affected by several parameters and is a function of the frequency, the blockage ratio, fluid properties, and some geometrical parameters of the stack.

III. Experimental Setup

The schematic of a standing-wave thermoacoustic refrigerator is shown in Fig. 1. Typical design parameters are listed in Table I. The stack has three main design parameters: the center position X_{sn}, the length L_{sn}, and the cross-section area A. The stack material should have good heat capacity and low thermal conductivity. In this experiment, the spiral stack is utilized and made of a 1.2x10-4 m thick Mylar sheet. The coefficient of the performance $COP_{st} = Q_{cn}/W_n$ of the stack is computed using (3) and (4). The prototype thermoacoustic refrigerator is designed at the cooling capacity of 120W. Based on the theoretical results and for practical reasons, the stack is located at $X_{sn} = 0.19$ and $L_{sn} = 0.15$ and the blockage ratio related to this design is $B = 0.744$. The diameter of the stack is 0.19 m. A resonance tube is made of steel, except the stack housing which is made of the plastic to reduce conduction heat losses. The inner diameter of resonance tube is 0.0869 m. To minimize heat losses, the resonance tube is covered by 12.7 mm thick insulations. An optimized quarter-wavelength, resonance tube, investigated by Hofler [7] is adopted to determine the length of the resonance tube at 400Hz.

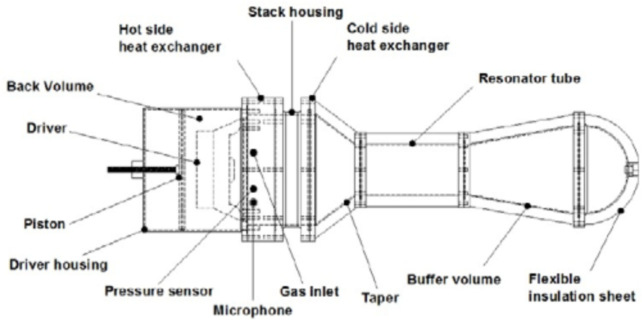

Fig. 1 Schematic of the standing-wave thermoacoustic refrigerator

TABLE I
Design and Operating Parameters of The Thermoacoustic Refrigerator

Property or parameter	Symbol	Value
Design frequency	f_0	400 Hz
Speed of sound in gas	a	1019.2 m/s
Ratio of specific heat	γ	1.66
Gas specific heat	C_p	5193.4 J/kg K
Gas thermal conductivity	k	0.15243 W/m K
Gas dynamic viscosity	μ	199.38 x 10^{-7} N s /m^2
Gas Prandtl number	σ	0.67
Mean temperature	T_m	300 K
Mean pressure	P_m	1-7 bars
Blockage ratio	B	0.744
Drive ratio	D	0.02
Normalized stack position	X_{sn}	0.19
Normalized stack length	L_{sn}	0.15

An eight-inch diameter loudspeaker with 6.3 ohm impedance is used as the acoustic driver that is further driven by a function generator and a power amplifier to obtain desired power and frequency. The driver is mounted in the housing at one end of the resonance tube. The driver housing is not insulated. The acoustic pressure and sound intensity level are measured using a condenser microphone with 5mV/Pa of sensitivity. The pressure of the gas within the resonance tube is measured using a pressure transmitter. Thermocouples are placed on each end of a stack to measure the temperature. The accuracy of the thermocouple is ±0.2oC. The OMB-DAQ BOARD-3000 via an OMB-PDQ30 and DAQVIEW software by Omega Engineering are used to acquire data.

IV. Results and Discussions

Fig. 2 shows the effect of temperature differences across the stack according to operating frequencies and mean pressures using helium as the working fluid. The pressure is varied from 1 to 7 bar. The results show that at a particular operating pressure, the temperature difference across the stack gradually increases and then decrease at some point where the temperature difference exists as the operating frequency increases. Fig. 3 shows that the optimal frequency and pressure is at 6 bars for the proposed thermoacoustic refrigerator. The temperature difference is increased approximately 62% compared to the one at 1 bar and the optimal frequency is approximately 20% away from the designed frequency. If the operating pressure is higher than 6 bar, the temperature difference across the stack is reduced because δ_k is inversely proportional to pressure according to (1). This means that the stack with small δ_k is required to operate at high pressure. Thus, for a particular stack, the optimal operating pressure and frequency exists.

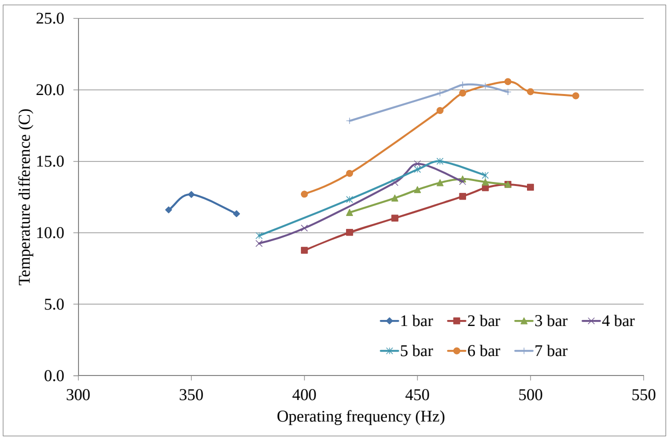

Fig. 2 Temperature difference across the stack versus mean pressure and frequency

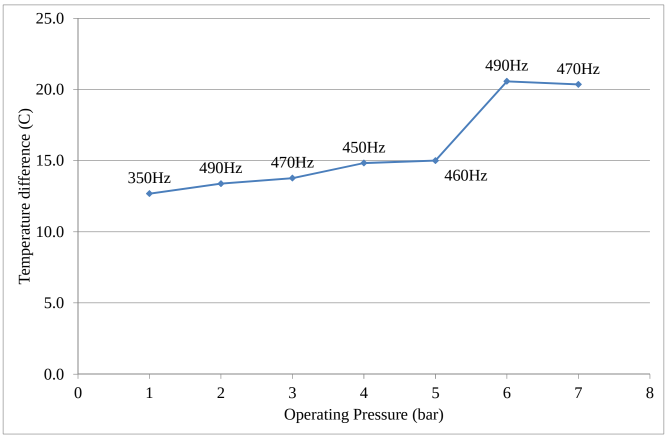

Fig. 3 Temperature difference across the stack at the optimal operating frequency

Fig. 4 compares the temperature differences between two working fluids. The position and length of the stack are similar, but the working fluids are altered from helium (Pr=0.67) to air (Pr=0.7). The temperature differences incurred by helium at 1 bar are close to ones incurred by air at atmospheric pressure. At higher operating pressure, the temperature differences using helium as working fluid is higher than ones using air as working fluid or approximately 58% improvement. The results are corresponding to (3) and (4) in the way that working fluids with low the Prandtl number is preferred for the thermoacoustic system. Most of noble gases has low the Prandtl number and thus is suitable for the thermoacoustic system as mentioned by [3].

The effects of power inputs are also studies in this paper as shown in Fig. 5. In this case, the operating frequency and pressure are maintained at 490Hz and 2 bar, respectively, and helium is used as working fluid. The electric power input to the loudspeaker is changed to study the responses of the temperature differences compared to the power input. As shown in the figures, the higher the power input, the larger the temperature difference across the stack. Based on thermoacoustic phenomenon, the electric power input is converted to acoustic energy that oscillates gas parcels and interaction with stack plates by which heat is transported from cold end to hot end of the stack. Fig. 5 points out that higher acoustic energy (i.e. input electric power) is able to migrate more heat from cold to hot end of the stack and eventually incur the maximum temperature difference across the stack. Unfortunately, not all of the acoustic energy can be achieved to pump heat through the stack due to the losses through the resonator surfaces and the heat exchangers [3]. Moreover, the efficiency to convert electric energy to acoustic energy of a commercial loudspeaker is only 3%. The rest is converted to heat that accumulates and rises the temperature of the loudspeaker that could damage the loudspeaker without proper cooling system.

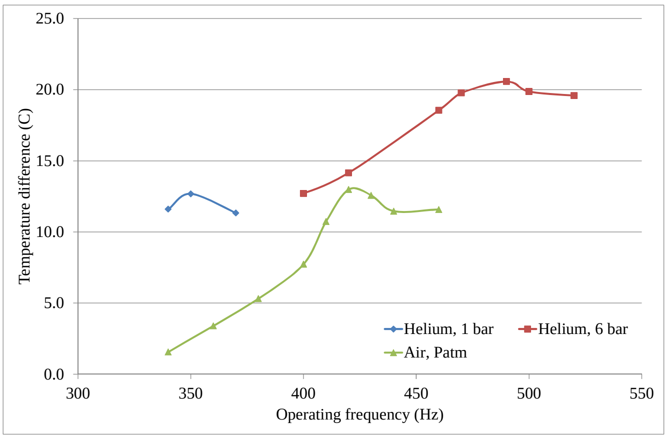

Fig. 4 The effect of the working fluid on the temperature difference across the stack

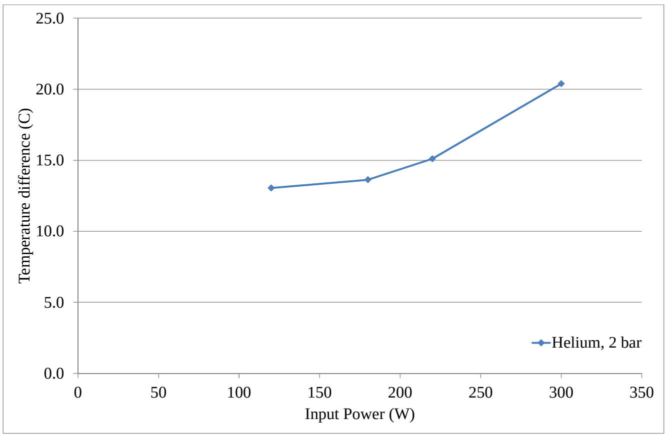

Fig. 5 The temperature difference across the stack versus input electric power for a constant frequency and pressure

V. CONCLUSION

The thermal response based on the temperature differences across the stack was studied. Experimental were performed on the prototype thermoacoustic refrigerator under various operating frequencies and mean pressures. The prototype is designed based on 120W cooling capacity. The results indicate that operating frequency and mean pressure are important and related to each other. Adjusting the operating frequency or mean pressure will not guarantee that the temperature differences across the stack is maximum. The 400Hz frequency and 7 bar mean pressure are used to design stack parameters, resonance tube length, and other parameters which are expected to obtain the maximum temperature differences across the stack at this design conditions. However, the results show that the optimal frequency and mean pressure are at 490Hz and 6 bars, respectively which is different from the design results. A problem is to create the stack according to the desired parameters. The prototype stack is by hand-made, so that the stack spacing is not equally throughout the stack. Also, parts that make up the thermoacoustic refrigerator have an influence on the operating frequency due to their acoustic properties, i.e., absorptivity, reflectivity and transmissivity. The interference of waves incurs and distorts the original frequency generated by the driver so that the optimal operating frequency differs from the designs. Unfortunately, the basic design equations used in the present does not apparently include these acoustic properties because of their complexity. So far, the optimal operating frequency and pressure can only be determined by testing.

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REFERENCES

[1] G.W. Swift, “Thermoacoustic engines,” *Journal of the Acoustical Society of America*, vol.84, pp.1145-1180, 1988.
[2] M.E.H. Tijani, J.C.H. Zeegers, and A.T.A.M. De Waele, “Design of thermoacoustic refrigerators,” *Cryogenics*, vol.42, pp.49-57, 2002.
[3] M. Wetzel, and C. Herman, “Design optimization of thermoacoustic refrigerators,” *International Journal of Refrigeration*, vol.20, pp.3-21, 1997.
[4] E.C. Nsofor and A. Ali, “Experimental study on the performance of the thermoacoustic refrigerating system,” *Applied Thermal Engineering*, vol.29, pp.2672-2679, 2009.
[5] M.E.H. Tijani, *Loudspeaker-driven thermo-acoustic refrigeration*, Mechanical Engineering, Eindhoven University of Technology, Eindhoven, 2001.
[6] K. Assawamartbunlue and C. Wantha, “The impact of the resonance tube on performance of a thermoacoustic stack”, *Frontiers in Heat and Mass Transfer*, vol. 2, pp.1-8, 2011.
[7] T.J. Hofler, *Thermoacoustic Refrigerator Design and Performance*, University of California at San Diego, 1986.