Experimental Investigations on the Performance of Eco- Friendly Thermoacoustic Refrigerator

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Abstract

Development of refrigerators based on Thermoacoustic technology is a novel solution to the present day need of cooling without causing environmental hazards. With added advantages of absence of moving parts and circulating refrigerants, these devices can attain very low temperatures maintaining a compact size. This paper presents an experimental work based on the results obtained from the sound wave thermoacoustic refrigerator. In this study, the performance of the thermoacoustic refrigerator is based on the temperature difference measured at the end of extremities of the stack. The optimized variables are the stack length, stack center position and the plate spacing represented by the blockage ratio.

Keywords: Stack, Thermoacoustic, Working Gas

I. INTRODUCTION

Environmental degradation caused by hazardous refrigerants from the refrigeration industry has driven extensive research into alternative clean technologies as well as more efficient cooling systems. The absence of refrigerants and a compressor in a thermoacoustic refrigerator deems it attractive to be explored further in our search for a more benign cooling technology. During the last two decades thermoacoustic refrigeration is explored as a new cooling technology. Thermoacoustic refrigerators are systems which use sound waves to produce cooling power. They consist of a loudspeaker, attached to one end of an acoustic resonator, which is closed at the other end [1]. The loudspeaker sustains an acoustic wave in the resonator. A stack and two heat exchangers are appropriately installed in the resonator. The thermal interaction between sound and the surface of the stack generates a heat-pumping process. The way these systems work is extensively explained in figure 1. Characterized by its low performance so far, optimization of the thermoacoustic system could make it more acceptable in several applications; the thermoacoustic refrigerator is a well-known device.

Sound waves are use in thermoacoustic refrigeration system to obtained cooling effect. The medium in which they move undergoes vibrations, thus experiences compression and rarefaction. This is associated with change in temperature and pressure. When the gas carrying a wave is brought in contact with a solid surface, it absorbs the heat as the gas gets compressed. Since the specific heat capacity of solids is generally quite greater than that of fluids, the solid absorbs heat without much change in its temperature. Similarly, it rejects heat to the gas molecules nearby during expansion, thus maintaining stable temperature. It is obvious that large amplitudes of pressure lead to an increase in heat transfer since it generates larger temperature difference between the gas and the solid. If continuous constant frequency vibration (by means of sound waves) is supplied to the gas in a tube with one end (reflector end) close, the reflected and the originally supplied sound waves will interfere with each other. The length of the tube is selected so as to obtain resonance. The pressure amplitude thus obtained is high and can be used to produce temperature difference across the tube.

II. LITERATURE REVIEW

The field of thermoacoustics has attracted researchers from 1816 when Pierre-Simon Laplace took into account the local change in temperature and pressure due to the compression and rarefaction that takes place in the medium during sound propagation [2].

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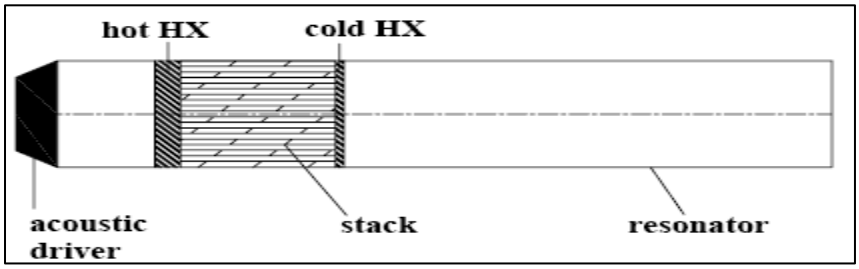

Fig. 1: Schematic of thermoacoustic refrigerator

19th century glass blowers used to hear pure tones emitted by their heated vessels. The reverse phenomenon was recognized after more than a century and it was realized that sound could conversely produce a temperature difference [3]. Notable advances were done after the work of P. Rijke, who employed a glass tube open at both closings and a wire mesh to locally heat the air. Tone close to the fundamental tone of the glass tube was obtained [4]. In a similar experiment, Lord Rayleigh used two gauze layers, the extra one to retain heat for a longer time. This reportedly produced strong oscillations which shook the room [5]. N. Rott made a considerable contribution by modelling of thermodynamic phenomenon through his linear theory [6]. He laid the mathematical foundation for thermoacoustics [7]. The field was further galvanized by notable contributions from G. Swift, who associated it with the broader field of thermodynamics [8].

The basic equations and concepts for the thermoacoustic theory and the early history of thermoacoustics are documented by Swift [1,8]. Jebali et al. [9] studied the performance of a thermoacoustic refrigerator subjected to variable loading. The hot heat exchanger of the system was maintained at ambient temperature while the temperature of the cold heat exchanger was varied to achieve a number of temperature differences along the stack. The cooling load was measured for these temperature differences while varying the driving frequency. Calculations of the contribution of the progressive and stationary waves and the losses on the thermoacoustic heat flow were performed. Results from the study showed that the cooling power is dependent on the working frequency and on the temperature difference between the stack ends. The maximum refrigeration power occurred around the resonance frequency.

Bai et al. [10] investigated the effect of the working fluid, resonator length, charging pressure and heating temperature on the performance of the prime mover. The study did not investigate the effects of variable loads on the system's performance. Zhou [11] performed experiments on a thermoacoustic prime mover with stacks made of copper wire mesh and studied the influence of gas properties, frequency, mean pressure, mesh size and stack length on overall performance of the prime mover. Wetzel and Herman [12] studied design optimization of thermoacoustic refrigerators using the short stack boundary layer approximation. By visualizing and quantifying the temperature fields in the neighborhood of a stack plate, they [13, 14] investigated the thermal interaction between a heated solid plate and the working fluid and used a combination of holographic interferometry and high-speed cinematography to study the thermoacoustic effects on the plate. The authors summarized the design steps and suggested related further areas of research. Worlikar et al. [15–17] performed numerical studies on thermally stratified flow in the neighborhood of an idealized thermoacoustic stack. They also visualized the energy flux density around the heat exchangers and examined the implications on the heat exchanger design. Following the studies on working gases for thermoacoustic systems, Belcher et al. [18] reported on the best working gases for thermoacoustic refrigeration. This present study is on the thermal response of the thermoacoustic refrigerating system when subjected to various operating conditions. This is necessary for design, understanding and improvement of the system. The experimental setup used consists mainly of the thermoacoustic refrigeration system. Helium was selected as the working fluid used in the system for this study. Results from the experimental study are based on the systems' response to various working pressures, frequencies, and refrigerant loading.

III. EXPERIMENTAL SETUP

For the design of the resonator tube, an optimized λ/4 length parameter introduced by Hofler [19] was adopted but a cone shaped buffer was substituted for the sphere in order to produce an open-end condition [20]. The resonator consists of the large diameter portion and a small diameter portion. The ratio of the small diameter tube portion to the large diameter tube portion was 0.54 for minimum energy losses. The stack used in the resonator is the parallel plate type made from a thermoplastic material. The oscillating gas flows in the spaces between the plates. In the design of the stack, consideration was given to the fact that the stack plate material should have a low thermal conductivity in order to reduce axial heat conduction. Some of the recommendations by Swift [1] and Wetzel and Herman [12] were used for this design. The auxiliary components of the system include the audio frequency generator, power amplifier, pressure transducers and gauges, thermocouples, an oscilloscope. Table 1 shows some parameters related to the design and operation of this thermoacoustic refrigerating system. The thermo physical properties of the helium, the working gas given in the table are with respect to standard temperature and pressure conditions.

Table – 1
Some design and operating parameters of the system

Property or parameter	*Symbol*	*Value*
Speed of sound in gas	*a*	*1013 m/s*

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Gas specific heat	c_p	*5193 J/kg K*
Gas thermal diffusivity	α	*13.2 x10^{-5} m^2/s*
Gas thermal conductivity	k	*0.155 W/m K*
Gas dynamic viscosity	μ	*197 x 10^{-7} N s/m^2*
Gas Prandtl number	σ	*0.663*
Gas density	ρ	*0.8845 kg/m^3*
Drive ratio	D	*0.02*
Ratio of specific heats	γ	*1.67*
Mean pressure	*Pm*	*3–6 bars*
Normalized stack position	x_n	*0.276*
Normalized stack length	L_{sn}	*0.262*
Blockage ratio	B	*0.5*

The desired cooling power, temperature difference between the hot heat exchanger and the cold heat exchanger and the operating frequency of the system were some of the items selected for the design of the system. Thermal penetration depth δk was calculated using the equation

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

and the viscous penetration depth dv was calculated using the equation

$$\delta_v = \sqrt{\frac{2\mu}{\rho_m \omega}}.$$

Figure 2 illustrate the arrangement of the experimental setup. A thermo acoustic refrigerator consists of an acoustic driver/speaker with a housing attached to a gas filled resonator tube in which a stack and two heat exchangers are placed. The system is constructed in such a way that all components are independent so that specific parts can be exchanged when the design changes or for other reasons. An electrical resistance heater arrangement was located at the cold side of the resonator to supply the variable load for the refrigerating system. An audio generator with frequency range from 10 Hz to 1 MHz was used to produce the sound that was transferred to the amplifier. The amplified sound is transferred to the acoustic driver which powers the thermoacoustic system. Fluid inside the resonator interacts with the stack plates which are aligned in the direction of vibration of the sound waves. For temperature measurement thermocouples was installed, one thermocouple was installed near the electric heater, one was installed at the surface of the acoustic driver, two were each installed at the inlet and outlet of the heat exchanger and one was installed at the middle of the resonator tube. Similarly for measurement of pressure, mass flow rates the bourdon tube pressure gauge, pressure transducer, oscilloscope, flow meter, was utilized. The range of the pressure transducers was from 0 to 100 psig. This pressure transducer was selected because of its small size and high sensitivity. An oscilloscope capable of measuring up to a sampling rate of 200 Msa/s and 100 MHz was used to analyze the pressure waves from the pressure transducer. The flow meter used for measuring the flow rate of the cooling water flowing in the system has the measuring range of between 4 and 40 GPH.

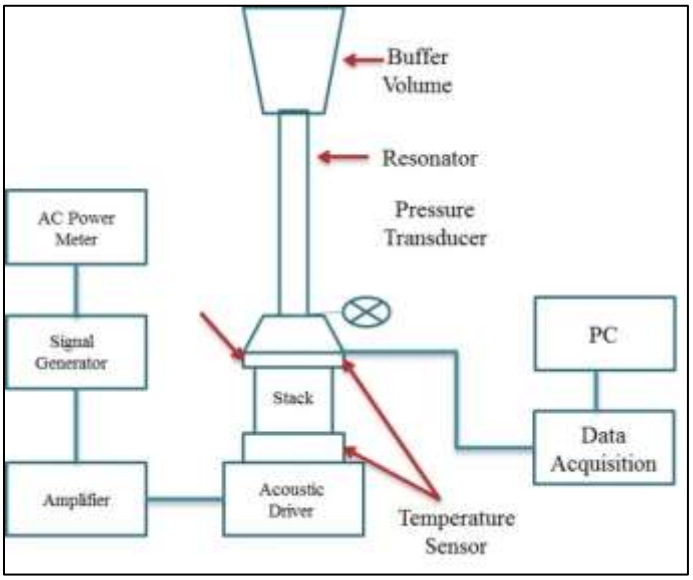

Fig. 2: Arrangement of experimental setup

Experiments were conducted to investigate the thermal response of the system under various operating conditions. The mean pressure was set initially at the lowest pressure of 3 bars to begin the experiments. The desired frequency was selected and then

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increased slowly from the minimum to the maximum value for the course of each of the experiments. The cooling load which was controlled using resistance heating in place of the cold side heat exchanger was initially set at a constant load. For each experiment, the data readings were taken from the initial time until conditions became stable. The frequency was then set at the next level and the experiment was repeated. After running the experiments for the desired frequency range, the pressure was adjusted to 4 bars and the experiment was repeated for the same set of frequencies and cooling load. Further experiments were performed for pressure values of 5 bars and 6 bars. The experiments were also repeated in the same way for various values of the cooling load from the selected lowest cooling load to the highest. The results show the ranges of the operational parameters.

IV. RESULT AND DISCUSSION

The graph of temperature (at the hot end of the stack) with time for various frequencies at constant mean pressure and cooling load is shown in Fig. 3. It shows how the frequency affects the temperature in the system. For each frequency, the temperature increased at the beginning of the experiment and then stabilized after a time. The experiment was performed for various constant pressure and cooling load values and it was found that the stabilization time increased as the pressure was increased. This could be because the thermoacoustic process increased with mean pressure in the system and therefore it took more time for the temperature to stabilize. Fig. 4 shows the history of the temperature at the hot end of the stack when the cooling load was varied at constant pressure and frequency. It was also seen that the temperature increased at the beginning and stabilized with time. For the experiments, the maximum temperature occurred at the maximum power tested and the lowest temperature occurred at the lowest power confirming that the maximum temperature increases with the cooling load. This is good for the system since higher temperatures are likely to result in higher temperature differences between the two

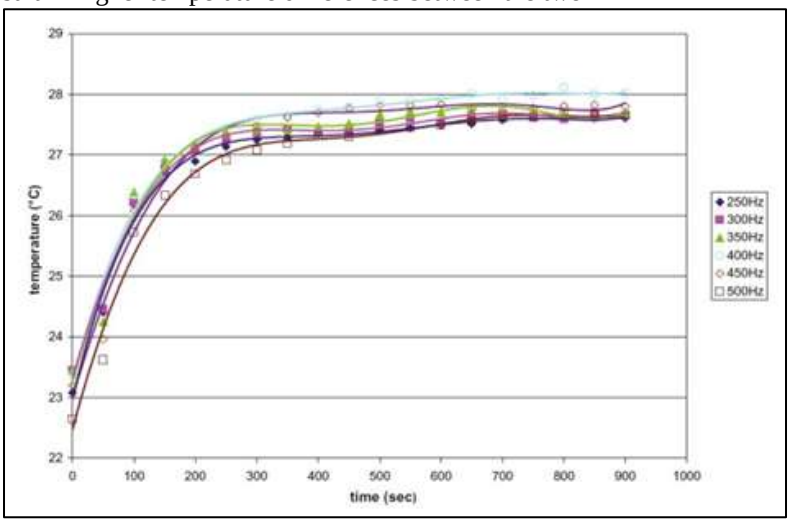

Fig. 3: Temperature time history (at the hot end of stack) for constant cooling load and mean pressure for various frequencies.

ends of the stack and therefore better performance by the system. Also, the stabilization time increased as the cooling load increased. This could be because the working temperature in the system increased as the cooling load increased. As the cooling load increased, the experiment showed that more time is needed to reach equilibrium in the system.

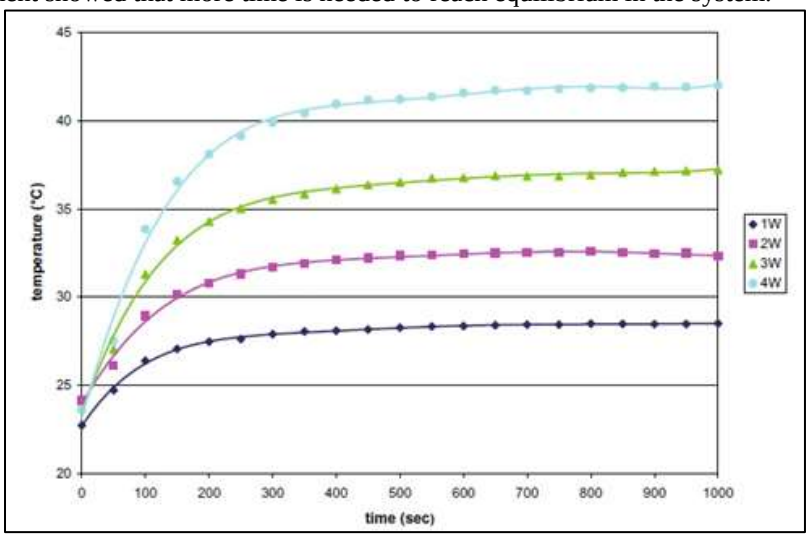

Fig. 4: Stack hot end temperature time history for constant mean pressure and frequency.

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The temperature variations along with different frequencies are shown in fig. 5, the various frequencies are tested under experimentation to obtained the better performance, frequency started from 200 Hz to 500 Hz were utilized and it is observed that the highest temperature variation was obtained on 384 Hz frequency. This could be because the thermoacoustic process increased with the frequency in the system and therefore it shows better performance in particular range, but after that the turbulence was increased which disturbed the thermoacoustic process and that's why the at higher frequency temperature variation was not obtained.

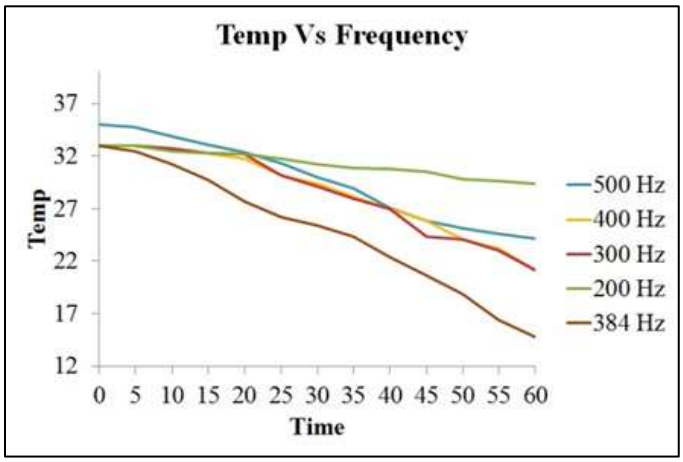

Fig. 5: Temperature variations for different frequencies.

V. CONCLUSION

As experimentations was conducted to check the performance of thermoacoustic refrigeration system when subjected to various operating conditions. The cooling load was controlled using resistance heating in place of the cold side heat exchanger. Results from this study show that the cooling load increases with the temperature difference between the two ends of the stack. It was also found that high pressure alone in the system does not necessarily result in a higher cooling temperature difference and thus a higher cooling load. For the thermoacoustic refrigerating system, there exists for a given frequency, an optimum pressure that results in the maximum temperature difference which in turn results in the possible maximum cooling load. . The most favorable temperature difference between hot end and cold end of the stack is obtained at frequency 384 Hz.

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