THE POTENTIAL OF AN AIR-OPERATED THERMOACOUSTIC COOLER AT LOW PRESSURE

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Abstract: Low cost thermoacoustic cooling system is still far beyond reach as it requires high-challenging pressurised gas which can only be obtained by using high cost fluids such as helium. This work investigates the potential enhancements of the thermoacoustic cooler using compressed air. The DeltaEC simulation is performed to study the coefficient of performance of the cooling system and the outcomes of this numerical simulation will be used to design a low cost thermoacoustic refrigeration system for developing countries. The study revealed the importance of the ambient duct length and the selection of the suitable frequency.

Keywords: Thermoacoustic, Compressed air, Refrigeration, Standing-wave, DeltaEC, COPR

1. Introduction

Standing-wave thermoacoustic refrigeration is a green technology which uses environmentally friendly working fluids to replace hazardous refrigerants. Besides that, another potential over the conventional cooling system is that it is simple in structure and has no moving parts. Moreover their miniaturisation is possible and would provide small-size refrigerators [1-2]. The standing-wave thermoacoustic cooler can potentially be applied in a wide range of cooling temperatures, from room temperatures to cryogenic temperatures. There are only limited research work done on room-temperature thermoacoustic coolers which use gases such as air to generate cooling [3]. Standing wave thermoacoustic refrigerator developed were mainly focused on the lowest achievable temperature for particular refrigeration applications including natural gas liquefiers or sensor cooling [4-5]. Most of the research work done up to date is making used of a relatively long stack compared with the resonator length and using helium as working gas and has a limited cooling load. As a result, some standing wave thermoacoustic refrigerators could reach cryogenic temperatures but with cooling capacities less than 10 W [6-9]. Presently thermoacoustic devices have low efficiency mainly due to the technical immaturity in designing the components of the devices. Hence, significant efforts are still needed to improve the thermoacoustic devices overall performance. The fundamental components of a thermoacoustic cooler include a stack, a resonance tube, two heat exchangers, and a loudspeaker which acts as a source to generate a standing acoustic wave. One of the noteworthy literatures, Wetzel and Herman showed a maximum value of 0.5 Carnot efficiency for the stack but the coefficient of performance for the commercial refrigerator can only reach up to 0.4 [10].

Efforts have been made to optimise the design of thermoacoustic coolers by improving the stack geometry. Several experimental works have shown that the linear thermoacoustic theory provides the optimum design of the thermoacoustic stack [11-12]. In order to generate an optimised temperature gradient, the stack is positioned between the pressure antinode and node. The stack materials should have low thermal conductivity, high heat capacity and an optimal value of the space between the stack layers. Tijani et al. investigated experimentally the optimisation of the stack spacing for maximum COP or for maximum cooling power [6-7]. It was observed that a stack spacing about three times larger than the thermal penetration depth is optimal for thermoacoustic cooling. Tasnim et al. conducted experiments on temperature fields at different locations on the stack plates and in the surrounding working fluid [13]. They found that axial heat transfer occurs in the stack extremities, as opposed to the hypothesis of a perfectly isolated stack used by Swift [14] in the linear thermoacoustic theory. The linear thermoacoustic theory prediction is based on the inviscid boundary-layer and short-stack approximation and neglecting the conduction of heat down the temperature gradient. Wetzel and Herman used a model based on the boundary layer approximation, and the short stack assumption to calculate the work flux and heat flux [10]. They optimised the system by adjusting nineteen design variables to achieve the best COP. Instead of using simplified work flux and heat flux equations, Minner et al. developed a design optimisation program that interacts with DeltaEC [15]. From a parametric study, they observed that the performance of the thermoacoustic refrigerator is sensitive to stack length, position, mean pressure and gas mixture, and less sensitive to the stack spacing.

The performance of the thermoacoustic cooler can also be affected by the design of the heat exchangers. Hence, two heat exchangers attached to both sides of the stack must also be optimised. Nsofor et al. studied the convection heat transfer coefficient on the outside surface of the heat exchanger in the thermoacoustic refrigeration system [16]. Results from the study showed that higher mean pressure will result in a greater heat transfer coefficient if the thermoacoustic cooler operates at the resonance frequency. Akhavanbazaz et al. conducted experiment on the impact of the heat exchange area in obstructing gas flow [17]. They found that heat exchanger with larger thermal contact area increase the heat exchange between the heat exchanger fluid and the stack, but it reduces the cooling power and increases the work input to the stack. Most of the literatures conceived to date only experimental work. In the present study, a numerical model of an air-operated thermoacoustic refrigerator for domestic cooling is designed with the aid of a linear one dimensional thermoacoustic software, DeltaEC developed by Ward et al. [18-19]. In order to predict the thermally induced acoustic waves accurately and also ensure the correct model used, the designed model was first validated against the work done by Tijani et al. [6-7]. The aim of this study is to perform parametric study on a household refrigeration system which operates in the temperature range between 273 and 305 K while having a high performance relative to Carnot value, COPR.

2. DeltaEC modelling

Design **E**nvironment for **L**ow-**A**mplitude thermoacoustic **E**nergy **C**onversion (DeltaEC) is used by researchers to evaluate the performance of the thermoacoustic devices [18-19]. The DeltaEC model of a thermoacoustic refrigerator is constructed by sixteen segments including the seven Reverse Polish Notations (RPN) segments which are used to perform the calculation of second law efficiency (COPR). A total number of four GUESSes and four TARGETs were chosen. The very first segment (zeroth segment) has been always a BEGIN segment. This segment contains global variables such as mean pressure, frequency, mean temperature and gas type. The calculations have been done using a constant temperature at the ambient heat exchanger T_h = 305 K and a regenerator length of 0.006 m. The BEGIN segment acts as a loudspeaker which generates sinusoidal standing wave in the refrigerator. A SURFACE segment comes next to account for oscillatory thermal losses at the first end of the resonator. The next segment is a lossy ambient temperature duct (DUCT) near the hot end of the stack. Heat exchangers (HX) are used to inject or remove heat. The first law of thermodynamics insists that this heat must equal to the difference between the upstream and the downstream total power ($\dot{H}_{tot}$) . Copper is used as the material for both heat exchanges. The material selected for the parallel-plate stack which forms the heart of the refrigerator is Mylar. Both ambient and cold heat exchangers also possess parallel-plate geometry. Stainless steel was used for the rest of the components in the refrigerator. A taper section is connecting the large diameter tube to the small diameter tube. A spherical bulb or compliance was used to terminate the resonator. Air was used as the working gas.

The GUESS vector, which has four components, shows what Deltaec will regard as solution variables: the mean temperature, the volume flow rate, the heat rejected at the ambient heat exchanger, AHX and the length of the small diameter tube. The initial guess of the mean temperature is taken as 300 K and the initial value of the volume velocity is taken as $1x10^{-4}ms^{-1}$. A heat rate value of -10 W was chosen to be the initial guess for the AHX and the initial length of the small diameter tube is taken as 0.19 m. Basically, DeltaEC integrates the wave equation from BEGIN to END. Refine the GUESS vector to find a solution to the acoustics problem that arrives at the HARDEnd with zero complex volume flow rate. The phase difference between the pressure and volume flow rate was also targeted to be equal zero to enforce the resonance effect. The calculations of COP, COP Carnot and COPR were obtained with the aid of Reverse Polish Notation (RPN).

3. Results and Discussion

All the results are produced by incrementing the independent variable with an appropriate step value and running the calculations repeatedly. Meanwhile, amendments are made on the GUESS values so that iterations will still converge (providing reasonable estimates of TARGETs values). Fig. 1 depicts the relationship between the temperature difference at the stack extremities and the operating frequency at different cooling loads and different mean pressure values. All the plots show a maximum temperature difference at approximately 150 Hz. It can be seen that increasing the pressure will have a significant effect on increasing the temperature difference in the cooler, this behaviour is more obvious and can be noticed at higher cooling loads. At a cooling load of 4 W, the increase of pressure from 7 to 10 bars resulting in an increase of the maximum temperature difference by almost double.

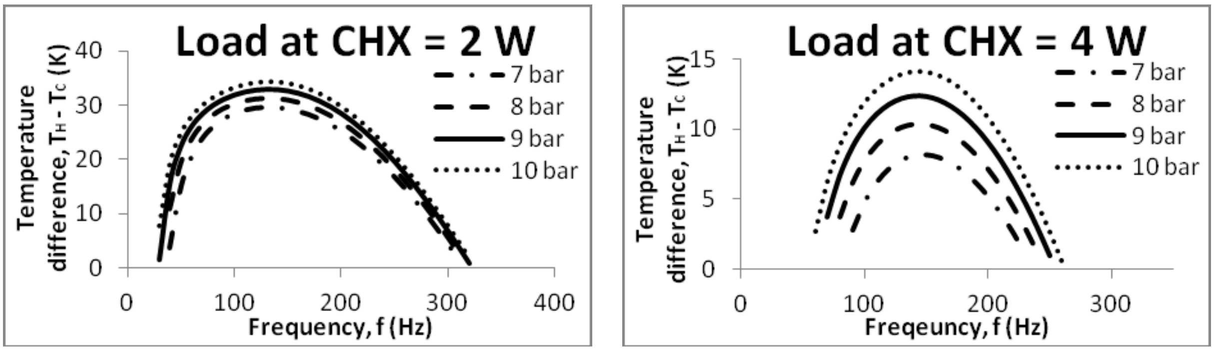

Figure 1: The results of temperature difference of the heat exchangers against the frequency of different heat load at the cold heat exchanger and different mean pressure when the drive ratio and the length of ambient duct are 2.5 % and 7.1 cm, respectively.

Furthermore, the coefficient of performance relative to Carnot's coefficient of performance is an important performance characterisation criterion for the thermoacoustic refrigerator. The response of COPR versus frequency for the different cooling loads and different mean pressure values is presented in fig 2. A nearly 30% of Carnot efficiency is achieved for a heat load at the cold heat exchanger of 0.5 to 1 W at the frequency around 45 Hz when mean pressure is between 7 to 10 bars. The mean pressure of 10 bar yields a higher COPR for all tested cooling loads compared to the other mean pressure values. For more reasonable load conditions between 3 to 4W the maximum COPR ranges between 7.5 to 15%. In general, the COPR decreases gradually from its peak value as both the frequency and the cooling load increase for all the examined mean pressure values.

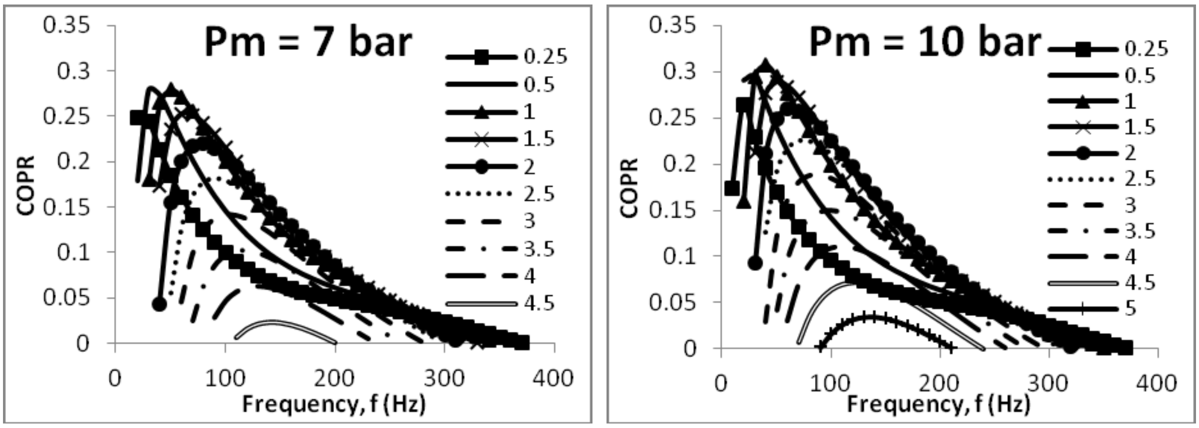

Figure 2: The graphs of COPR as a function of frequency for different mean pressure and different heat load at the cold heat exchanger when the drive ratio and the length of ambient duct are 2.5 % and 7.1 cm respectively.

In order to improve the thermoacoustic cooler footprint, thereby increase its potential application to be competed with the conventional refrigerator, the ambient duct is made smaller in size. Nevertheless, this will affect the performance of the overall system as the stack position from the loudspeaker also changes when the length of ambient duct is shortened. Fig. 3 shows the results of COPR against heat load at the cold heat exchanger for a constant frequency of 240 Hz. When a mean pressure of 10 bar is used, the best COPR is obtained when the ambient duct length is 2.1 cm. With the length of ambient duct increases, the COPR evolves in the same manner but with relatively low efficiency for each drive ratio value. It can be generally observed that the decreasing length of the ambient duct resulted in increasing the maximum COPR achieved at the same mean pressure. The drive ratio has no significant effect on the COPR or the location of its maxima at the same mean pressure.

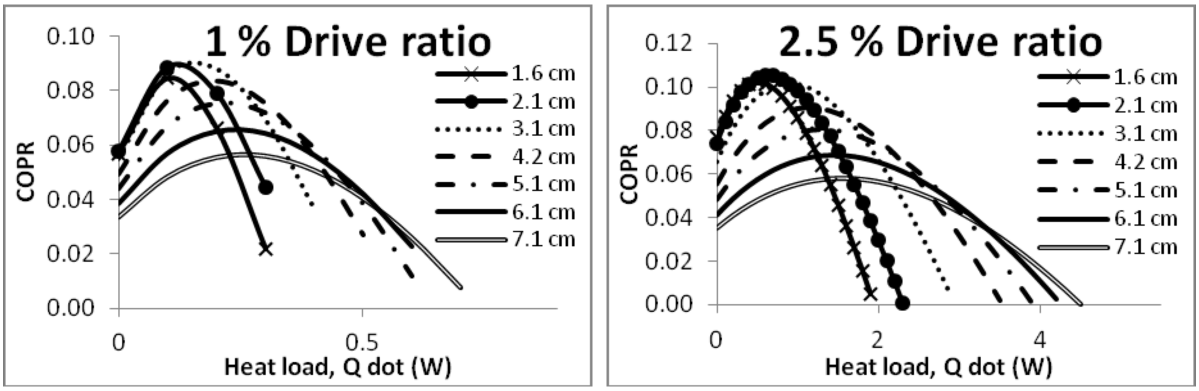

Figure 3: The results of COPR against heat load at the cold heat exchanger for different drive ratio and different the length of ambient duct when the mean pressure and operating frequency are 10 bar and 240 Hz respectively.

4. Conclusion

This work has successfully investigated the use of air as potential candidate for a low cost thermoacoustic cooler. In the proposed cooler, the pressure is not expected to be operating at more than 10 bars and the temperature difference is targeted to be ranging from 10-15 K below ambient conditions. The study revealed the importance of the ambient duct length as well as the selection of the suitable mean pressure and frequency. Air can be considered as an alternative fluid for a cooler at relatively low temperature difference requirements using renewable energy sources for the conditions of unavailability of conventional power sources.

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