Applied Thermal Engineering 100 (2016) 296–303

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

Experimental investigations of the performance of a standing wave thermoacoustic refrigerator based on multi-objective genetic algorithm optimized parameters

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HIGHLIGHTS

- Optimization of a standing wave thermoacoustic refrigerator using MOGA.
- Minimizing acoustic power and maximizing cooling power through MOGA to improve COP.
- L_{sn}, x_{sn}, and B are optimized simultaneously to obtain the optimum COP.
- Comparison of optimized results from MOGA with experimental work.

ARTICLE INFO

Article history:
Received 7 October 2015
Accepted 8 February 2016
Available online 23 February 2016

Keywords:
Thermoacoustic refrigerator
Multi-objective optimization
Stack
Genetic algorithm

ABSTRACT

This paper presents an experimental work based on the results from the Multi-objective Genetic Algorithm (MOGA) optimization scheme for a standing 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. Two different stacks were investigated: 1) spiral geometry build from Mylar, and 2) ready-made ceramic Celcor stack. Comparisons of the effects of the stack length, center position and plate spacing were completed between the optimized outcomes from MOGA and that measured experimentally. Results showed good agreement, confirming that values other than the optimized stack length of 4 cm, stack center position of 4 cm, and the stack separation gap of 0.36 mm did not give the desired high performance of the thermoacoustic refrigerator.

1. 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. Characterized by its low performance so far, optimization of the thermoacoustic system could make it more acceptable than it is now [1]. Thermoacoustic cooling technology is based on the interactions of acoustic waves with a solid boundary creating thermoacoustic cooling effects. At high pressure and with proper design, fabrication, and operation, the cooling effects that can be harnessed can be very significant. For commercial application, the first system was an ice cream cabinet using a standing wave, collaboration between The Penn State University, Unilever Engineering Excellent team, and Ben & Jerry's in 2004 with Sound Cool [2]. The Carnot coefficient of performance (COPc) of the device was 19% for a given operating condition. Although several prototypes and working systems have been developed prior to and after the Ben & Jerry chiller, COP of these thermoacoustic refrigeration systems is still relatively low [2–6]. Thus, design optimization of a thermoacoustic refrigeration system is important to make it attractive enough for consideration by the general market.

The most common approach utilized in the optimization of the thermoacoustic refrigerator is through discrete variation of each of the parameters of interest, experimentally or numerically, while holding other variables fixed. These optimization procedures are constrained by the range of values and parameters set obtaining only the local optimum/minimum. Furthermore, the use of DeltaE, a thermoacoustic software program [7], in past work to assist in the design of thermoacoustic devices needs tremendous amount of efforts [8]. The first attempt at a mathematical optimization based on a global search was probably completed by Andrew et al. in 2011 [9]. Their

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http://dx.doi.org/10.1016/j.applthermaleng.2016.02.028
1359-4311/

multi-objectives model showed possibilities of an optimized system based on personal design preference. Genetic algorithm (GA) is a relatively recent optimization scheme which has proven its capability in the optimization of some parameters of the standing wave thermoacoustic refrigerator [10–12]. This paper reports the results of experimental investigations based on the outcomes of the simultaneous optimization of the cooling power and acoustic power of the stack in a standing wave thermoacoustic refrigerator. Experiments were then conducted to compare the effects of the stack length, center position, material, and geometry, with that obtained from the optimized outcomes from GA. Comparison of the theoretical optimized performance from GA with that obtained experimentally has never been done before.

2. Design of the thermoacoustic refrigerator

The simplest thermoacoustic refrigerator consists of an acoustic driver, a porous media called the stack, hot and cold heat exchangers and the resonance tube where the acoustic waves interact with the stack to generate cooling at one of the stack. The performance of the thermoacoustic stack depends on three main stack design parameters: the center position measured from the acoustic driver, x_s; the length, L_s; and the cross-section, A. To simplify the analysis, dimensionless parameters have been used for the stack center position, x_{sn}, and stack length, L_{sn}. A good performance of the stack is defined by a high COP, a ratio of the cooling power to the acoustic power at the stack. The normalized cooling power, $Q_{cn} = Q_c/p_m aA$, and acoustic power, $W_n = W/p_m aA$, are given by Tijani et al. [5]:

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

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

where Λ is defined as:

$$\Lambda = 1 - \sqrt{\sigma}\delta_{kn} + 0.5\sigma\delta^2_{kn} \tag{3}$$

The stack length, L_s, and stack center position, x_s, are normalized by the wave number, k, which is given by $\lambda/2\pi$. The normalized temperature difference is given by the temperature difference across the stack length, ΔT_m, divided by the mean temperature, T_m. The thermal and viscous penetration depths are normalized by the half spacing of stack plate, y_0. The ratio between the amplitude of the dynamic pressure, p_0, and mean pressure, p_m, is called the drive ratio, D. The blockage ratio, B, which is also a dimensionless parameter, represents the porosity of the stack and l is the thickness of the stack plate. The dimensionless parameters used in this study are as listed in Table 1.

Generally, an increase in the cooling power increases the acoustic power required. Past optimization work is associated with the variation of certain parameters to achieve a high cooling power or a low acoustic power. The "optimized" parameters are identified from the combination of results completed over selected discrete values obtained. In this study, the optimization involves maximizing equation (1) and minimizing equation (2) simultaneously. Although these have been done before, they have not been compared against any experimental work [10–12]. The thermoacoustic cooler in this paper has been designed for a temperature difference, $\Delta T = 30$ K, across the stack using air as the working fluid at atmospheric pressure, 101 kPa. Even though the inert gas such as helium or a mixture of helium has been proven to give a better performance, air is often used when the focus of the study is not related to the working fluid [13]. The frequency selected is 400 Hz. The frequency selected is based on the manufacturability and the thermal penetration depth, δ_k, that is given by

$$\delta_k = \sqrt{\frac{2K}{\rho c_p \omega}} \tag{4}$$

where ω is the circular frequency. Increasing the frequency does increase the power density but at the same time, making the δ_k smaller and difficult to manufacture [5].

Table 1
Dimensionless parameters in the thermoacoustic refrigerator. Drive ratio.

Operating parameters	
Drive ratio: $D = p_0/p_m$	
Normalized cooling power: $Q_{cn} = Q_c/p_m aA$	
Normalized acoustic power: $Wn = W/p_m aA$	
Normalized temperature difference: $\Delta T_{mn} = \Delta T_m/T_m$	
Gas parameters	
Prandtl number: σ	
Normalized thermal penetration depth: $\delta_{kn} = \delta_k/y_0$	
Stack geometry parameters	
Normalized stack length: $L_{sn} = kL_s$	
Normalized stack position: $x_n = kx$	
Blockage ratio or porosity: $B = y_0/(y_0 + l)$	

2.1. Stack geometry

Stack geometry with a large surface area for the fluid–solid interactions is desired, constrained by the manufacturability of the component. Since cooling in a standing wave thermoacoustic refrigerator is achieved as the gas parcels oscillate next to the stack walls within a thermal boundary layer, the minimum separation between the walls would be twice the thermal boundary layer. In this study, the spiral and rectangular pore stack have been chosen.

2.2. Stack material

The stack material should have a low thermal conductivity, K, and a higher heat capacity than the working fluid. The rectangular pore stack selected here is ready-made from Celcor ceramic generally used in automotive industries. For the other geometry, it has been hand-made from the most common material used in thermoacoustic studies, Mylar. The related parameters of thermoacoustic refrigerator are listed in Table 2.

3. Stack design

Equations (1) and (2) are the objective functions; the former is to be maximized, while the latter is to be minimized, simultaneously, under optimized conditions of the stack length, L_{sn}, stack center position, x_{sn}, and plate spacing, $2y_0$. The standard optimization process associated with experimental and numerical work is associated with discrete variations of these three parameters. The outcomes would generally be a series of graphs whereby the best combinations that

Table 2
Operating parameters and air properties at T = 300 *K*.

Fluid properties	Operation parameters
Speed of sound in gas, a = 347.2 m/s	$p_m = 100$ kPa
Ratio of specific heat, $\gamma = 1.4$	$T_m = 300$ K
Gas Prandtl number, $\sigma = 0.7$	$\Delta T_m = 30$ K
Gas heat capacity, $c_p = 1005.5$ J/kg · K	$f = 400$ Hz
Gas thermal conductivity, $K = 26.3 \times 10^{-3}$ W/mK	