DESIGN OPTIMIZATION OF PARALLEL PLATE HEAT EXCHANGERS IN THERMOACOUSTIC DEVICES

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1. INTRODUCTION

Cooling by sound is a new environmentally friendly technology developed rapidly during the past three decades. The refrigeration system based on this technology is called thermoacoustic refrigeration. Due to inherent simplicity, reliability and no hazardous materials, thermoacoustic refrigeration systems have a strong potential to replace conventional refrigeration systems. However, at present, these devices have lower efficiency mainly attributed to the poor understanding of the fundamental processes and technical immaturity to design different components of these devices. Thus, significant efforts are needed to improve the fundamental understanding of the thermoacoustic process, and the role/impact of the main components of these devices on the overall performance. This will lead to the development of efficient devices that will benefit the society from environmental as well as economical perspectives.

One of the main components of a thermoacoustic device is the heat exchanger. Heat exchangers are used to exchange heat with the warm and cold environments external to the device. However, the impact of heat exchanger design on the thermoacoustic process and the overall performance of the device are not well understood. In the present study, parallel plate heat exchangers of a thermoacoustic refrigerator with an acoustic driver and a heat-driven thermoacoustic refrigerator are optimized theoretically by using the second law of thermodynamics and the computer code DeltaE which solves the one dimensional wave equation in a geometry defined by the user. DeltaE is a great tool to simulate and predict the behavior of thermoacoustic devices in the linear range of the acoustic wave [1]. The manufacturing issues are also considered for the optimized heat exchangers to facilitate easy fabrication and thus, practical usage.

2. METHODOLOGY

To have minimum working gas flow disturbance in the vicinity of stack edges and heat exchangers, some studies have recommended that the heat exchangers should have the same porosity as that of the stack [2]. However, to keep the same porosity as the stack, the plate spacing in parallel plate heat exchangers needs to be reduced which results in increasing the entropy generation due to viscous effects quadratically [3]. Thus, it could be inferred that keeping the same porosity for the heat exchangers as that of the stack is not an optimized choice. A better design choice for parallel plate heat exchangers is investigated in the present study by applying the analysis developed by Ishikawa and Hobson [4] which is based on the second law of thermodynamics for the time-averaged entropy generation in parallel plate heat exchangers of thermoacoustic devices. The effects of plate thickness, spacing and the porosity of parallel plate heat exchangers on the performance of the two thermoacoustic devices are investigated using DeltaE. Swift [5] recommended that the optimum length of a heat exchanger is equal to the peak to peak gas displacement at the heat exchanger location. In this study, the length of the cold heat exchangers and the refrigerator ambient heat exchangers are assumed equal to the gas peak to peak displacement at the cold heat exchanger locations and the length of the hot heat exchanger and the engine ambient heat exchanger are assumed equal to the gas peak to peak displacement at the engine ambient heat exchanger location.

3. RESULTS

A thermoacoustic refrigerator is designed with the air as the working gas. The mean pressure is considered to be atmospheric with the resonance frequency of 140 Hz. The sketch of the designed device is shown in Figure 1. The overall length and cross sectional area of the device are 1.22 m and 0.0132 m^2, respectively. This device could provide 3 watt of cooling power at 9°C.

Fig. 1. Sketch of the thermoacoustic refrigerator with air, all dimensions are in millimeters

In the first simulation by DeltaE, the blockage ratio of the heat exchangers is assumed equal to that of the stack (BR=0.8). Through the second law analysis and by assuming the plate thickness of the heat exchangers equal to their length (i.e. 3.6 mm), the porosity of the heat exchangers is optimized. The optimized blockage ratio for the cold heat exchanger and ambient heat exchanger are estimated 0.56 (plate spacing 4.6 mm) and 0.345 (plate spacing 1.9 mm), respectively. The heat exchangers with modified blockage ratio are used to simulate the same apparatus by DeltaE. The results of the two simulations are summarized in table 1. The results show that with the optimized heat exchangers whose porosity is less than that of the stack, the overall performance of the thermoacoustic refrigerator is increased by 16%.

Cases	BR same as stack	BR optimized based on second law
Required acoustic power(w)	2.28	1.97
Dissipated acoustic power in the stack(w)	0.88	0.88
Dissipated acoustic power in the CHX(w)	0.17	0.01
Dissipated acoustic power in the AHX(w)	0.16	0.02
Ratio of the dissipated acoustic power in HXs to required acoustic power	14.4%	1.52%
Overall coefficient of performance	1.31	1.52

Table 1: simulation results for the thermoacoustic refrigerator

A heat driven thermoacoustic refrigerator is also designed with helium as the working gas at the mean pressure of 7 atm. The resonance frequency is 400 Hz. The sketch of the designed device is shown in Figure 2. The overall length and cross sectional area of the device are 1.25 m and 0.012 m^2, respectively. This device could provide 30 watt of cooling power at 2˚C.

Fig. 2. Sketch of the heat driven thermoacoustic refrigerator with helium, all dimensions are in millimeters

The same procedure is used to optimize the heat exchangers of the second device. That is, the device is simulated first by assuming the blockage ratio of all heat exchangers to be the same as the stack i.e. 0.8. Then the blockage ratio of the heat exchangers is changed to optimize the design based on the second law analysis. The simulation results for both cases are presented in Table 2. Similar to the first refrigerator, the results show that for the heat-driven thermoacoustic refrigerator, with the optimized heat exchangers whose porosity is less than that of the stack, the overall performance of the thermoacoustic refrigerator is increased by 35%. The modified blockage ratio of hot heat exchanger, engine ambient heat exchanger, cold heat exchanger and refrigerator ambient heat exchanger are 0.13, 0.19, 0.32 and 0.20, respectively.

Cases	BR same as stack	BR optimized based on second law
Required heat energy (w)	215	159
Produced acoustic power in engine stack (w)	35.1	25.2
Dissipated acoustic power in the CHX/Ref. AHX (w)	2.49/2.66	0.23/0.67
Dissipated acoustic power in the HHX/Eng. AHX(w)	5.15/3.28	1.82/0.57
Ratio of the dissipated acoustic power in HXs to produced acoustic power	38.7%	13%
Overall efficiency	14%	18.9%

Table 2: simulation results for the heat driven thermoacoustic refrigerator

4. DISCUSSION

The present study shows that parallel plate heat exchangers with the same porosity as that of the stack are not optimized due to higher acoustic power dissipation. Furthermore, they are not practical from manufacturing aspect. It is also been shown that the porosity of the optimized parallel plate heat exchangers is lower from that of the stack which also reduces the manufacturing challenges.

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