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Acoustic field characteristics and performance analysis of a looped travelling-wave thermoacoustic refrigerator

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ABSTRACT

This paper focuses on a looped travelling-wave thermoacoustic refrigerator powered by thermal energy. Based on a simplified model for the regenerator, key issues for a highly efficient thermoacoustic conversion, including both thermal-to-acoustic and heat-pumping processes, are summarized. A looped travelling-wave thermoacoustic refrigerator with one engine stage and one refrigerator stage is proposed, with emphasis on high normalized acoustic impedance, sufficient volumetric velocity and appropriate phase relation close to travelling wave in the regenerators of both engine and refrigerator. Simulation results indicate that for the ambient temperature of 30 °C, the looped travelling-wave thermoacoustic refrigerator can be powered by the heat at 210–250 °C to achieve the refrigeration at −3 °C with the overall coefficient of performance above 0.4 and the relative Carnot coefficient of performance over 13%. The characteristics of the acoustic field inside the loop configuration are analyzed in detail to reveal the operation mechanism of the looped travelling-wave thermoacoustic refrigerator. Additional analyses are conducted on the impact of the cooling and the heating temperatures, which are of great concern to the refrigeration applications and the utilization of low-grade thermal energy.

1. Introduction

Thermoacoustic heat engine is a type of energy conversion machine, converting thermal energy into mechanical energy or consuming mechanical energy to pump heat, with the outstanding characteristics of high reliability and environmental friendliness [1]. As for the current challenges of energy and environment crisis, the thermoacoustic heat engine powered by low-grade thermal energy (waste heat, geothermal energy and solar thermal energy, etc.) has become a hot research area, facing but not limited to the applications of electricity generation, refrigeration, water pumping, circulation and pressurization [2–11].

The conception of travelling-wave thermoacoustic engine was proposed by Ceperley [12], trying to control the phases of velocity and pressure of oscillating gas parcels by travelling wave, thus to realize a reversible thermodynamic cycle similar to Stirling cycle with a high thermal efficiency. However, the low acoustic impedance in the regenerator caused severe viscous loss, leading to a poor thermal efficiency. Following Ceperley's idea, Yazaki et al. put two thermoacoustic cores (a stack sandwiched by two heat exchangers) into a closed loop (one of which acted as the thermoacoustic engine and the other one acted as the thermoacoustic refrigerator) [13], achieving a lowest cooling temperature of −27 °C with the heating power of 230 W. Sakamoto and Watanabe carried out a series of research on the similar configuration, and successfully obtained refrigeration effect under a heating temperature of 320 °C, with an overall coefficient of performance (defined as the ratio of the cooling capacity to the heating power) of 0.015 when the temperature drop was 16 °C [14].

The thermoacoustic Stirling heat engine, which could achieve travelling-wave acoustic field as well as high acoustic impedance in the regenerator by proper combination of the looped structure and the resonator [15], dramatically improved the efficiency of travelling-wave thermoacoustic engines. The invention of this novel thermoacoustic system and the underlining mechanism has also enlightened the research of thermoacoustically-driven thermoacoustic refrigerator. Ueda et al. added a regenerator in the loop of the thermoacoustic Stirling engine as a refrigerator stage, and its overall coefficient of performance was 0.052 at the cooling temperature of 0 °C [16]. Kang et al. proposed a system similar with Ueda's design and reached a cooling capacity of 40 W with an overall coefficient of performance of 0.13, when the cooling temperature was 0 °C [17]. Luo et al. proposed a different configuration by adding a secondary travelling-wave loop in a thermoacoustic Stirling engine,

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thus thermal energy was converted into acoustic power in one loop, and the acoustic power was then consumed in the other loop to pump heat, with an overall coefficient of performance of 0.11 at the cooling temperature of −22.1 °C [18].

The thermoacoustic refrigerators mentioned above, which are based on the thermoacoustic Stirling engine, are considered to have a promising prospect in the thermoacoustically-driven refrigeration with high efficiency. However, the required heat sources are generally with relatively high quality (the temperature is even as high as 600 °C) [17,18], which will severely limit its practical application scope.

In 2010, de Blok improved the looped travelling-wave thermoacoustic engine by inserting multiple thermoacoustic cores in the loop, and the required temperature of heat source has been significantly reduced. The multi-stage travelling-wave thermoacoustic engine, driven by the heat sources below 200 °C, was expected to realize a relative Carnot efficiency of 40% [3,4]. Another novel structure with three thermoacoustic engine stages and one thermoacoustic refrigerator stage equally spaced in a looped tube achieved an overall coefficient of performance of 0.073 when pumping heat from −40.5 °C to 24.2 °C, with the heating temperature of 211 °C [4]. Thus, the feasibility of thermoacoustic refrigerators driven by low-grade heat sources was verified by this novel technology based on the multi-stage travelling-wave thermoacoustic engine. Zhang et al. numerically analyzed the acoustic field in it and found that there exists significant difference in acoustic impedance of the four regenerators [5], indicating that the configuration is worthy of further optimization. Yu and Al-Kayiem conducted a numerical simulation to optimize this configuration by breaking the symmetric geometry so that the impedance distribution could be improved, and reached an overall coefficient of performance of 0.153 when pumping heat from 0 °C to 40 °C, with the heating temperature of 200 °C [6].

After a qualitative analysis from a simplified model for the regenerator of a travelling-wave thermoacoustic system, the present work proposed a looped travelling-wave thermoacoustic refrigerator with one thermoacoustic engine stage and one thermoacoustic refrigerator stage, connected by 2 resonators with lengths of approximately $3/4\lambda$ and $1/4\lambda$ (λ is the wavelength of the sound wave propagating inside the looped configuration), where the thermoacoustic cores are locally enlarged to reduce the viscous loss and the hydraulic radiuses of the regenerators is small to ensure good thermal contact between the gas parcels and the solid. These features help to make use of the merits of efficient thermoacoustic conversion in the travelling-wave acoustic field and to reduce the required temperature of driving heat source. A simulation based on thermoacoustic theory has then been conducted to analyze the performance of the looped system quantitatively. The emphasis is put on the analysis of the acoustic filed characteristics with respect to the thermal-to-acoustic conversion in the engine's regenerator and the heat-pumping effect in the refrigerator's regenerator. The thermodynamic performances of the system at various cooling and heating temperatures are also presented and discussed from the view of the variation in acoustic field.

2. Qualitative analysis based on a simplified model of the regenerator

For a travelling-wave thermoacoustic system, the regenerator is the place where the thermoacoustic conversion occurs. As one of the key components, it is significant to conduct a qualitative analysis on the performance of thermoacoustic conversion in the regenerator for a better conception design of a thermoacoustic system.

According to the thermoacoustic theory [19], the variation in the acoustic power through an element channel with the length of dx in one dimensional case can be depicted as

$$\frac{\mathrm{d}E_2}{\mathrm{d}x} = -\frac{r_v}{2}|U_1|^2 - \frac{1}{2r_\kappa}|p_1|^2 + \frac{1}{2}\mathrm{Re}[g\tilde{p}_1 U_1], \tag{1}$$

where E_2 is the acoustic power, r_v is the resistance caused by viscosity, r_κ is the effective resistance of thermal relaxation. U_1 and p_1 are the volumetric velocity and pressure, respectively. g is a sort of complex gain or attenuation coefficient for the volumetric velocity. Re represents the real part of a complex number. The tilde denotes the complex conjugation. As for Eq. (1), the first term and the second term on the right side are the acoustic power dissipations caused by the fluid viscosity and the thermal relaxation, respectively. The third term represents the acoustic power gain or attenuation, generated from the thermal-to-acoustic conversion or consumed for the heat-pumping effect, which are two opposite thermoacoustic conversion processes.

Generally, r_v, r_κ and g are functions of the geometry and dimensions of the channel, and g is even related to the temperature gradient along the channel [19]. For an ideal regenerator with perfect thermal contact between the working fluid and the solid channel boundary, as well as zero void volume, r_κ is infinite, and g can be simplified as [19,20]

$$g = \frac{1}{T_\mathrm{m}}\frac{\mathrm{d}T_\mathrm{m}}{\mathrm{d}x}, \tag{2}$$

where T_m is the mean temperature.

Consequently, Eq. (1) can be simplified into

$$\frac{\mathrm{d}E_2}{\mathrm{d}x} = -\frac{r_v}{2}|U_1|^2 + \frac{1}{2T_\mathrm{m}}\frac{dT_\mathrm{m}}{dx}|p_1||U_1|\cos\varphi, \tag{3}$$

where φ is the phase difference between p_1 and U_1. In addition, the expressions of r_v and U_1 are [19]

$$r_v = \frac{\omega\rho_\mathrm{m}}{A}\frac{\mathrm{Im}[-f_v]}{|1-f_v|^2}, \tag{4}$$

$$U_1 = v_1 A, \tag{5}$$

where ω, ρ_m, v_1 and A are the angular frequency, the mean density, the velocity, and the cross-sectional area of the channel, respectively. f_v is the spatially averaged viscous function [19]. Combining Eqs. (3)–(5), by eliminating r_v and U_1, yields

$$\frac{\mathrm{d}E_2}{\mathrm{d}x} = -\frac{\omega\rho_\mathrm{m}A}{2}\frac{\mathrm{Im}[-f_v]}{|1-f_v|^2}|v_1|^2 + \frac{1}{2T_\mathrm{m}}\frac{dT_\mathrm{m}}{dx}|p_1||U_1|\cos\varphi. \tag{6}$$

The first term on the right side in Eq. (6) indicates that decreasing the velocity amplitude $|v_1|$, instead of the volumetric velocity amplitude $|U_1|$, is the essential approach to reducing the viscous dissipation, although the group $A|v_1|^2$ can be expressed as $|U_1||v_1|$. According to the second term on the right side in Eq. (6), for a given mean temperature and temperature gradient, the higher values of $|p_1|$, $|U_1|$ and $\cos\varphi$ are all expected to enlarge the acoustic power generation and consumption for the engine's regenerator and the refrigerator's regenerator, respectively. The effects of the velocity amplitude $|v_1|$ and the volumetric velocity amplitude $|U_1|$ on the viscous dissipation and the acoustic power generation or consumption are different, which is the reason that we tried to distinguish the velocity amplitude $|v_1|$ from the volumetric velocity amplitude $|U_1|$. This differentiation, which usually does not attract enough attention, can inspire a better conception design of a thermoacoustic system for higher performance.

In a summary, as for the design of a travelling-wave thermoacoustic system, the following three points can be suggested for the regenerator: (1) high normalized acoustic impedance, i.e., $Z_\mathrm{n} \gg 1$ ($Z_\mathrm{n} = |p_1/v_1|/\rho_\mathrm{m}c$, where c is the sound speed), implying a relatively high pressure amplitude $|p_1|$ and a relatively low velocity amplitude $|v_1|$; (2) sufficient volumetric velocity amplitude $|U_1|$; (3) phase difference φ close to zero, leading to $\cos\varphi$ approaching to 1.