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Influence of the connecting tube at the cold end in a U-shaped pulse tube cryocooler

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ABSTRACT

In some special applications, the pulse tube cryocooler must be designed as U-shape; however, the connecting tube at the cold end will influence the cooling performance. Although lots of U-shape pulse tubes have been developed, the mechanism of the influence of the connecting tube on the performance has not been well demonstrated. Based on thermoacoustic theory, this paper discusses the influence of the length and diameter of the connecting tube, transition structure, flow straightener, impedance of the inertance tube, etc. on the cooling performance. Primary experiments were carried out in two in-line shape pulse tube cryocoolers to verify the analysis. The two cryocoolers shared the same regenerator, heat exchangers, inertance tube and straightener, and the pulse tube, so the influence of these components could be eliminated. With the same electric power, the pulse tube cryocooler without connecting parts obtained 31 W cooling power at 77 K; meanwhile, the other pulse tube cryocooler with the connecting parts only obtained 27 W, so the connecting tube induced more than a 12.9% decrease on the cooling performance, which agrees with the calculation quite well.

1. Introduction

Pulse tube cryocoolers (PTCs) can be divided according to their shape [1]. If the regenerator and the pulse tube are in line, it is called an in-line pulse tube cryocooler. Its disadvantage is that the cold region is in the middle of the cryocooler. By bending the cryocooler at the cold end of the regenerator and the pulse tube, a U-shaped cryocooler is produced. It is much more convenient since it can produce cooling at the end of the cryocooler. For better compactness, the regenerator can be constructed as an annular shaped space surrounding the pulse tube, which is called a coaxial shape. The in-line shape pulse tube cryocooler can obtain the best performance and is often employed in the laboratory. The coaxial shape is very popular in practice for its compactness although the thermal contact between the pulse tube and the regenerator may decrease the performance [2]. The U-shape is a tradeoff between the convenience and the performance. It is the most common shape in GM type pulse tube cryocoolers [3].

In the in-line PTC, the pulse tube, the cold heat exchanger and the regenerator are connected directly. In the U-shaped PTC, there are connecting tube, flow straightener and transition sections at the cold end. These parts may influence the cooling performance of the cryocooler. The experiment by Charles et al. shows that the efficiency of a U-shaped PTC was about 30% lower than that of an in-line one [4]. Meanwhile, in Hiratsuka's experiment, it was about 9% lower [5]. Radebaugh did some qualitative explanation for the influence of the connecting part at the cold end in a U-shape pulse tube cryocooler, "It requires void space at the cold end to reverse the flow direction and introduce turbulence into the pulse tube" [1]. But no one has yet systematically and quantitatively demonstrated how these connecting parts influence the performance and why the influence was different in the various experiments. This is the goal of the present study.

In this paper, the linear thermoacoustic theory will be employed to analyze the influence of the connecting parts. Using a numerical approach, we will demonstrate how the connecting tube, straightener, flow transition sections and impedance of the inertance tube influence the cooling performance. Because the cooling performance of the U-shaped pulse tube cryocooler is easily influenced by the flow ununiformity, a systematic experimental verification is not easy to be precisely realized. Here only a primary experiment will be carried out. At last, some conclusions will be drawn.

2. Theoretical model

Linear thermoacoustic theory gives the following equations to describe the flow inside the PTCs [6,7].

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$$\frac{dp}{dx} = -[\xi_\mu r_\mu + i\omega\xi_l l]U \tag{1}$$

$$\frac{dU}{dx} = -\left\{i\omega\xi_c c + \frac{1}{\xi_\kappa r_\kappa}\right\}p + gU \tag{2}$$

$$\frac{dH}{dx} = Q, \quad H = c_1 + c_2\frac{dT}{dx}, \tag{3}$$

where p, U, H, Q, and dT/dx are the pressure, volume flow rate, total energy flow, heat absorbed from the surroundings and mean temperature gradient, respectively. r_μ, l, c, r_κ and g are the viscous resistance, inertance, compliance, thermal-relaxation resistance and complex gain/attenuation constant, respectively. Because the velocity in the connecting tube may be fairly high and turbulent flow may occur, four correction factors based on the assumptions of Iguchi et al., ξ_μ, ξ_l, ξ_c, and ξ_κ, are introduced in the first two equations [8,9]. It assumes that oscillatory-flow losses can be calculated by using the Moody friction factor (valid for steady flow) at each instant of time during the oscillatory flow. This assumption was well approved in Ref. [10]. After appropriate discretization [11], the three equations are used to calculate the acoustic and temperature distributions in tubes, heat exchangers and regenerators. What must be mentioned is that this model is one dimensional. The flow inhomogeneity cannot be taken into consideration.

When high-Reynolds-number flow makes an abrupt transition from a pipe of small cross-sectional area to a larger area, the transition is accompanied by work dissipation. This phenomenon is often referred to as "minor loss". For oscillating flow, the time-averaged power dissipation can be expressed as [12]

$$E = \frac{\rho U^3}{3\pi A^2}(K_{\exp} + K_{con}) \tag{4}$$

where ρ, A, and K are the density, cross sectional area and minor loss coefficient. When the flow expands into a much larger cross-sectional area, $K = K_{\exp} \approx 1$. For the opposite direction, contracting into the smaller pipe, $K = K_{con}$ is strongly dependent on geometrical details. If the edge of the entrance is sharp, $K_{con} = 0.5$. In this paper, the two values are used to evaluate the power dissipation at the two ends of the connecting tube. The minor loss will result the same decrease in cooling power.

The direction change of the flow will also induce minor loss [13]. For a typical connecting tube with diameter of 5 mm and length of 30 mm, if it is bent into a half circle as shown by the solid line in Fig. 1(a), the minor loss will only be 20% of that induced by the abrupt transitions at the two ends of the connecting tube. If the tube is longer, the loss will even be less. But, if the connecting tube it is bent into the shape as shown by the dashed line in Fig. 1(a), the minor loss could be 1.3 times of that. So, the minor loss induced by the direction change varies much with the shape of the connecting tube and it is fixed as 20% of that the loss induced by the abrupt transitions in the following calculation.

Fig. 1 shows the three configurations that we will study in this paper. The calculation will focus on the cold finger. The linear compressor will be simplified as a pressure ratio of 1.28 since it is just a pressure wave generator. The inertance tube and the reservoir will be simplified as an acoustic impedance. In the calculation, the phase angle and modulus of the impedance are optimized to make the efficiency of the cryocooler maximum. As we know, in pulse tube cryocoolers with high pressure ratio and small cooling power, the optimum impedance cannot be realized by an inertance tube [14]. There is still no model to precisely predict the impedance that an inertance tube provides because of the turbulent flow. So it is very difficult of decide what impedance should be employed in the calculation for the inertance tube. However, if the inertance tube is replaced by an expansion piston, any impedance can be realized at the end of the secondary water cooler. In order to make the study more universal, whether the inertance tube can realize the optimized impedance is not taken into consideration.

In a real PTC, the connecting tube cannot be directly connected with the cold heat exchanger or the pulse tube because the sudden change of the cross section area may induce jet-driven steaming. Conical transition sections are often used to disperse the jet [15]. Here, the diameter of its one end equals to that of the connecting tube, the other end is the same as the diameter of the regenerator or the pulse tube. The connecting tube, transition section and cold heat exchanger are often machined in one copper block in practice, so it is reasonable to assume their temperatures are the same in the calculation. Their temperatures are set as 77 K. The main parameters of the pulse tube cryocoolers are presented in Table 1.

3. Numerical results

3.1. Influence of the location of the connecting tube

In order to know the influence of the connecting tube, the performance of the pulse tube cryocooler with the connecting tube removed should first be known (Of cause, the transition sections and straightener are also removed. In fact, it is the same as a general in-line pulse tube cryocooler). Table 2 presents the result. It can be seen that the relative Carnot efficiency could be as high as 35.3%.

In Fig. 1(a), the connecting tube is installed between the cold heat exchanger and the pulse tube, meanwhile, in Fig. 1(b), it is installed between the cold heat exchanger and regenerator. It is found that the performance of these two configurations is quite different. With Fig. 1(b), the efficiency is much lower. If a coordinate originating from the main water cooler (MWC) and passing through the regenerator (RG), the connecting tube (CT), the pulse tube (PT) and the secondary water cooler (SWC) is set, the gas temperature distribution in the cold end of the pulse tube cryocooler can be presented in Fig. 2. It can be seen, although the gas temperatures in the cold heat exchanger are also almost the same, but the temperature in the cold end of the regenerator in Fig. 1(b) is much lower. The cooling power is not extracted from the coldest part in Fig. 1(b), so the efficiency is lower.

3.2. Influence of the flow straightener

Although the conical transition section is used to disperse the jetting flow, it is not enough to make the flow laminar in the pulse tube because it needs a very long distance. Some screens are necessary to be fixed between the transition section and pulse tube to further straighten the flow. Fig. 3 shows the influence of the length of the flow straightener (FS) on the relative Carnot efficiency. Obviously, the straightener has negative influence on the efficiency. This may also be explained by the fact that the lowest temperature is not located in the cold heat exchanger when the straightener is used as shown in Fig. 4. With higher mesh screens (300 mesh stainless steel screens), the heat exchange between the working gas and the straightener is better, and the straightener acts as a more ideal regenerator and produces a greater negative temperature gradient in it. So, the efficiency is deteriorated more compared with 80 mesh stainless steel screens. If there are screens made of an imagined material whose heat conductivity, density and heat capacity are all 0.1 times of those of stainless steel, the straightener will influence the efficiency much less. As shown in Fig. 4, with the imagined material, the straightener does not show any regenerative effect and the positive temperature gradient in the straightener is very similar to that in a pulse tube.