A Miniature Coaxial Pulse Tube Cooler Operating above 100 Hz

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ABSTRACT

Increasing the drive frequency can yield a more compact pulse tube cooler, which is very attractive for many applications. We have studied high frequency (>100 Hz) pulse tube coolers for years, and in our previous work, the numerical and experimental results show that the same cooler can achieve better performance at both 100 Hz and 300 Hz when different inertance tube phase shifters are used. This article focuses on the influence of frequency on system performance. The analytical results indicate that performance decreases as the frequency increases from 100 to 300 Hz. A miniature coaxial pulse tube cooler has been designed and fabricated, and the influence of various parameters such as frequency and pressure ratio have been investigated experimentally and analytically. Typically, this cooler can produce 2.8 W of cooling at 77 K with an acoustic power of 40 W and reaches a no-load temperature of 49.5 K when operating at 105 Hz. After the frequency is increased to 280 Hz, the no-load temperature raises to 63.3 K and a maximum cooling power of 1.0 W at 80 K is achieved.

INTRODUCTION

Pulse tube coolers have the advantage of low cost, long life, and low mechanical vibration, which gives them potential application in many fields. But compared with Stirling coolers, their relatively larger size and lower power density can limit their application in some fields in which size and weight are critical. To address this issue, an increase in drive frequency can effectively lead to a decrease in the size and mass of the compressor for the same output acoustic power, and thus yield a more compact system. Other researchers that have focused on increasing system frequency in recent years include Radebaugh et al., Vanapalli et al., and Dai et al.[1-3]

Our laboratory has conducted research on high frequency pulse tube coolers (>100 Hz) for years. A miniature in-line pulse tube cooler was designed and manufactured to operate around 300 Hz; it was driven by a standing-wave thermoacoustic compressor. In 2009, it achieved a no-load temperature of 60 K and a cooling power of 1.04 W at 80 K.[4] When the same PTC was driven by a linear compressor working at 100 Hz, it achieved better performance with a cooling power at 80 K of 2.6 W.[5]

This article introduces a miniature coaxial pulse tube cooler that has similar structural parameters to the one previously referenced.[4,5] Operation at both 100 Hz and 280 Hz is investigated. A linear compressor is used as a substitute for the thermoacoustic compressor when working at 280 Hz.

In the following sections, we first present numerical simulations that were carried out to investigate the effect of frequency on performance. Then, the experimental systems are introduced. Thirdly, the main experimental results are presented. This includes the noted influence of various parameters such as frequency and pressure ratio that were investigated both experimentally and analytically. Lastly, conclusions are drawn.

THEORETICAL ANALYSIS

Thermoacoustic theory[6,7] is a powerful tool for regenerative coolers. A numerical model[8] based on thermoacoustic theory was used to optimize the parameters. This cooler was originally designed to operate with a frequency in the 300 Hz range. A large number of calculations were carried out to optimize different components such as the regenerator, pulse tube and inertance tube. Since high frequency operation decreases the thermal penetration depth, the required hydraulic diameter in the regenerator must be reduced as the frequency is increased. Previous calculations indicated that stainless steel screens with a mesh number of 600 could provide a suitable hydraulic diameter for 100 Hz operation. However, the maximum mesh number available on the market is 600, and this is insufficient for meeting the requirements for operation at 300 Hz. An interesting thing is that the optimized size of the regenerator remains unchanged as the frequency increases from 100 to 300 Hz if the regenerator material is kept constant.

The influence of frequency was investigated in detail in the current studies. For these, the pressure ratio was fixed at 1.22, and the mean pressure was 3.55 MPa. Figure 1 shows the optimized impedance at the hot end of the pulse tube at different frequencies as required to achieve a good relationship between pressure and volumetric flow rate. As shown in Figure 1, the phase angle of the impedance becomes larger as the frequency increases, and the amplitude becomes smaller.

As the frequency increases, the impact of void volume in the regenerator gets larger. After the impedance at the hot end of the pulse tube has been optimized, the phase difference between the pressure and flow rate is almost zero. As shown in Figure 2, the phase difference becomes larger at the hot end of the pulse tube, which creates a greater loss in the regenerator.
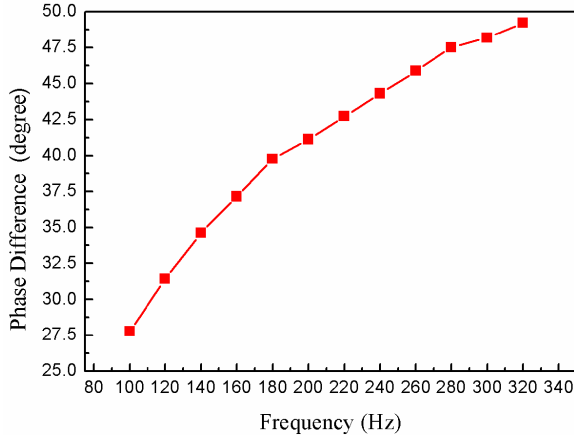
Figure 3 shows the input acoustic power and cooling power at 77 K for different drive frequencies; a cooling power of 1 W at 77 K requires an acoustic power of 13.6 W at 100 Hz. The performance becomes worse as the frequency increases, and a cooling power of 1 W at 77 K requires 22 W at 300 Hz.

EXPERIMENT SYSTEM AND MEASUREMENT

The experimental system was composed of linear moving magnet compressor, coaxial pulse tube cooler, and connecting tube. Figure 4 shows a schematic of the experiment system. The ambi-
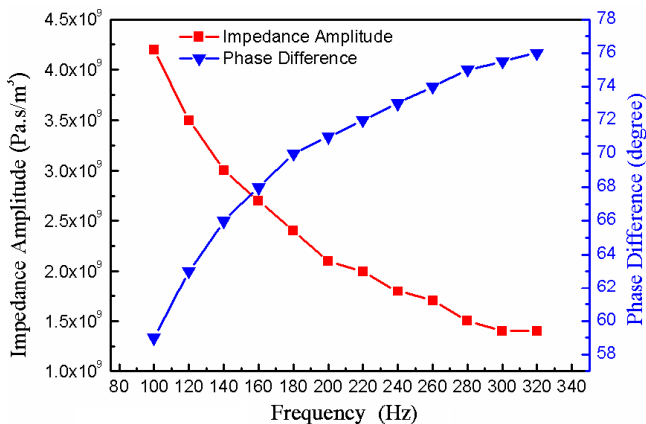

Figure 1. Acoustic impedance amplitude and phase difference versus frequency, with a pressure ratio of 1.22 and a mean pressure of 3.55 MPa.

Figure 2. Phase difference between dynamic pressure and volumetric flow rate versus frequency, with a pressure ratio of 1.22 and a mean pressure of 3.55 MPa.
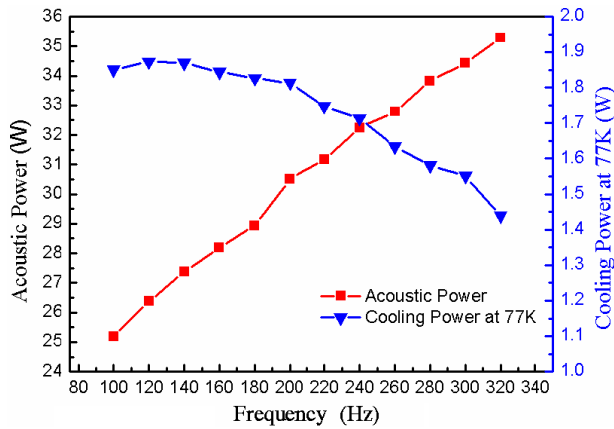
Figure 3. Input acoustic power and cooling power at 77 K versus frequency, with a pressure ratio of 1.22 and a mean pressure of 3.55 MPa.
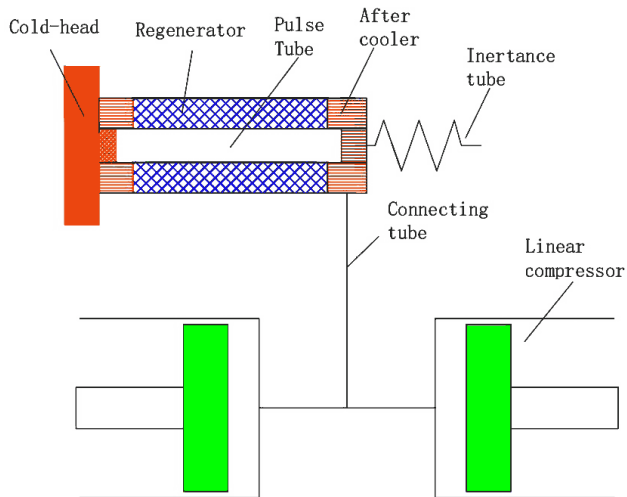
Figure 4. A schematic drawing of Stirling-type coaxial pulse tube cooler driven by linear compressor.

ent heat exchangers were kept at room temperature using chilled water, with the temperature maintained around 20°C. The temperature of the cold-head was measured by a calibrated PT100 thermistor, while the cooling power of the PTC was measured using a resistor heater with external DC power supply. The instantaneous pressure and mean pressure at both the front and backside of the compressor were also measured. A dual-piston moving magnet linear compressor was used to reduce the vibration.

Because the study mainly focused on the performance of the pulse tube, the characteristics were mostly investigated in terms of the pressure ratio at the inlet of the PTC. With the assumption of adiabatic compression in the backside space, the piston velocity was computed using the measured dynamic pressure in the backside space. The acoustic power (PV power) was then approximated using the dynamic pressure in the compression space and the piston velocity.

EXPERIMENT RESULTS AND DISCUSSION

Performance of PTC at 100 Hz

The characteristics of the pulse tube cooler operating at both 100 Hz and 280 Hz were studied. When operating at 100 Hz, the cooler was driven by a linear compressor that was originally designed for a 10 W pulse tube cooler requiring a drive power of up to 200 We. When connected to the miniature coaxial pulse tube cooler in this study, the compressor was not working at resonance, but it could still supply a pressure ratio of 1.3. In order to evaluate the performance of the cooler, the pressure and estimated acoustic power were recorded as described in last section. A copper inertance tube with a diameter of 1.5 mm and a length of 1.4 m, plus a 200 cc reservoir, were used as the phase shifter.

Figure 5 shows a typical cooldown curve for the coldhead. The mean pressure was 3.55 MPa. It takes about 5 minutes to cool down to 77 K. The lowest no-load temperature achieved was 49.5 K. The driven pressure ratio reached 1.23, and the measured compressor output acoustic power was about 39 W.

Figure 6 shows the no-load coldhead temperature for different pressure ratios at the inlet of the PTC with the mean pressure of 3.55 MPa and frequency of 100 Hz. The dotted line with triangle points represents the simulation results, while the solid line with square points represents the experimental results. Although both curves have the same trends, the simulation results are about 10 K lower than the experimental results. The cooler consumed about 12 W acoustic power to maintain a no-load temperature of 75 K.

The cooling power at 77 K for different pressure ratios is presented in Figure 7. Again, the mean pressure is 3.55 MPa. A maximum cooling power of 2.68 W at 77 K was acquired and the corresponding estimated acoustic power was 44.5 W. Referenced to the acoustic power, the COP is 0.06, and the relative Carnot efficiency is 17%, approximately. The relative Carnot efficiency reaches a maximum value of 18% with a pressure ratio of 1.236. As the pressure ratio increases, turbulence
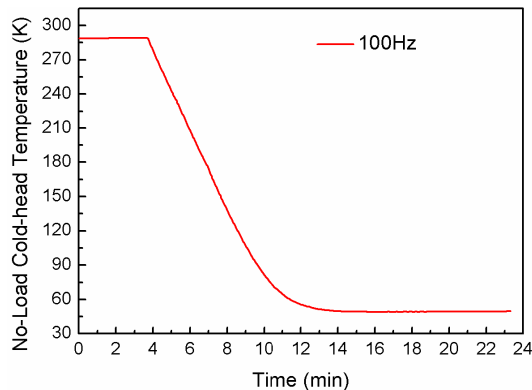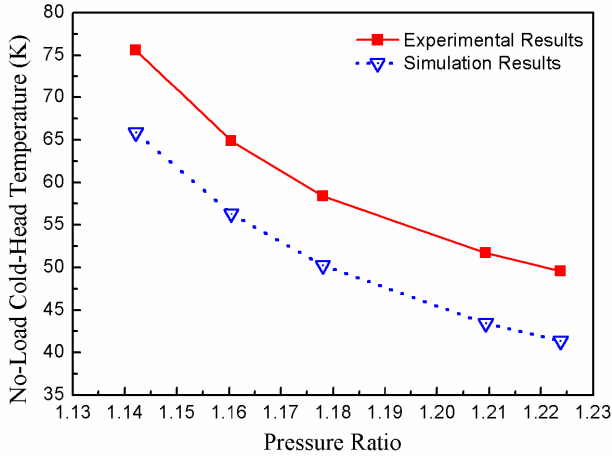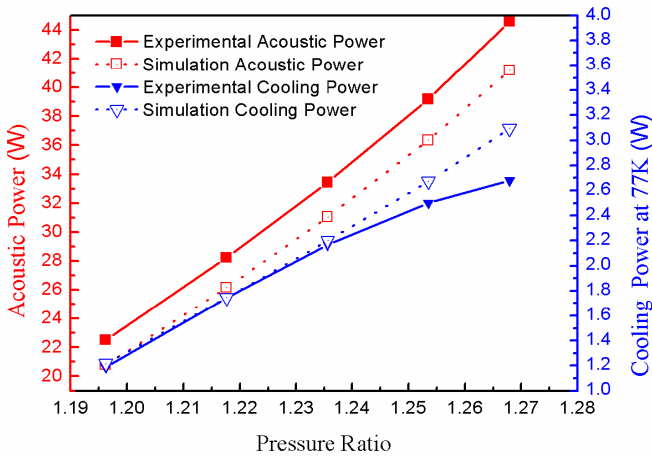

Figure 5. Cool-down curve of the cold-head, with frequency of 100 Hz.

Figure 6. No-load cold-head temperature versus pressure ratio at the inlet of PTC, with a frequency of 100 Hz and a mean pressure of 3.55 MPa
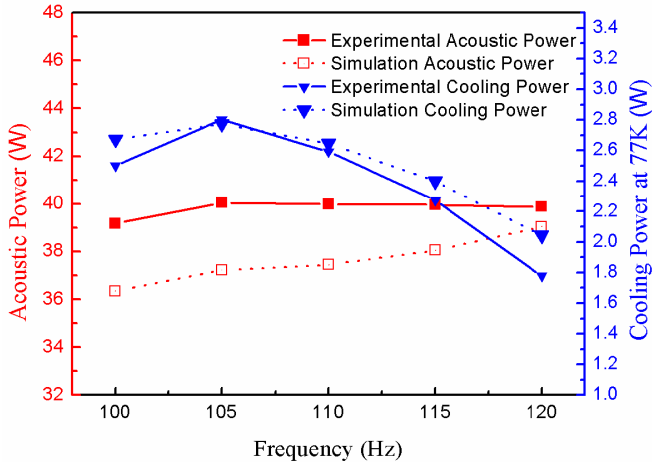

Figure 7. Compressor output acoustic power and cooling power at 77 K versus pressure ratio at the inlet of PTC, with a frequency of 100 Hz and a mean pressure of 3.55 MPa

within the inertance tube causes a decrease of phase angle. Enhancement of jet effects within the pulse tube is likely to also cause additional losses; these cannot be simulated in our computational model. As a result, the experimental results and simulation results are very close at lower pressure ratios, but the difference becomes larger as the pressure ratio increases.

The main problem for this system is the terrible performance of the compressor. The efficiency is only 25%. The compressor losses include three parts: 1) Joule heat losses generated by the coil; 2) iron losses produced by eddy currents; and 3) mechanical losses due to friction between the piston and cylinder. Firstly, the compressor is not working in a resonant state; this leads to a larger Joule heat loss. Secondly, an optimum impedance match between the cooler and compressor could not be realized. And lastly, the compressor itself has a large coil resistance and mechanical damping coefficient, which causes larger losses.

As shown in the Figure 8, the influence of frequency was investigated next. Pressure ratio and mean pressure were used in the numerical model to obtain simulated results for comparison. The compressor output acoustic power was maintained around 40 W. The acoustic power in the simulation increased slowly as the frequency increased. Both the experimental and simulation cooling power reached a maximum at 105 Hz. A maximum cooling power of 2.8 W at 77 K was achieved

Figure 8. Compressor output acoustic power and cooling power at 77 K versus frequency, with a mean pressure of 3.55 MPa.

with an acoustic power of 40 W. At the frequency of 105 Hz, the inertance tube can supply a good phase relationship between pressure and volumetric flow rate. As the frequency increases further, the compressor gradually approaches its resonant frequency, which causes its drive current to decrease to 7 Arms from 10 Arms. So the compressor efficiency increases to 33%.

As mentioned above, although the total system efficiency is very low, the performance of the PTC meets its design goal. As a next step, the compressor parameters will be optimized. Then the compressor will be manufactured by Lihan Technologies, with a predicted efficiency above 60%.

Performance of PTC at 280 Hz

When operating at 280 Hz, two single-piston moving-magnet compressors were connected together to generate the pressure wave. Different from the 100 Hz compressor, the compressor front-side volume was adjusted to 11.6 cc to achieve a resonant state around 280 Hz. The maximum pressure ratio achievable was about 1.23, with a mean pressure of 4.1 MPa. A combination of two copper inertance tubes was used as the phase shifter.

Figure 9 shows the cooldown curve for the cold-head when operating at 280 Hz. Compared to the performance at 100 Hz, the cooldown time increased to 12 minutes, and the lowest no-load temperature reached 63.3 K with a pressure ratio of 1.228. The estimated compressor output acoustic power was 41.5 W.
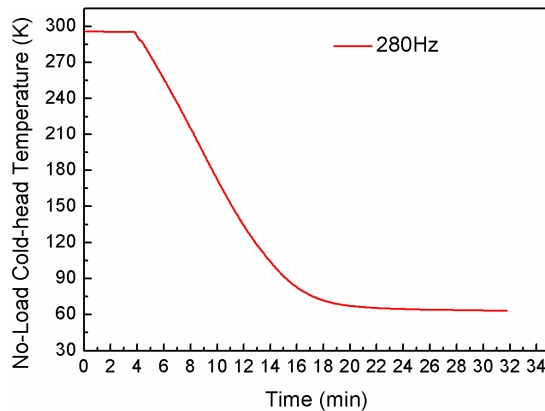
Figure 9. Cold-down curve with a frequency of 280 Hz and a mean pressure of 4.1 MPa.
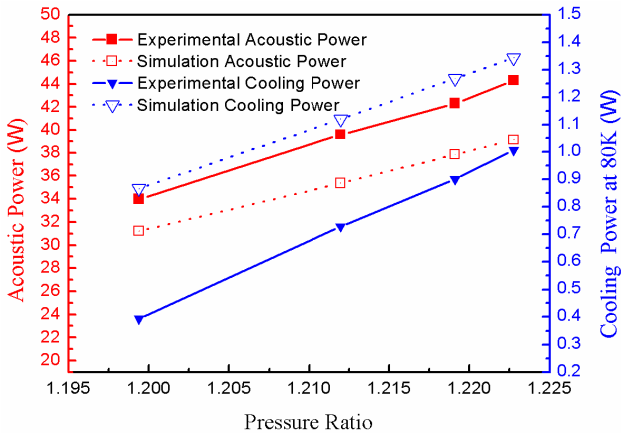
Figure 10. Compressor output acoustic power and cooling power at 80 K versus pressure ratio at the inlet of PTC, with a frequency of 280 Hz and a mean pressure of 4.1 MPa.

At so high an operating frequency, the characteristics are very sensitive to operating and structure parameters. The influence of void volume is very obvious, most of the volumetric flow rate swept by the piston enters the compressor front side, so the phase difference between the dynamic pressure on the front versus back side of the piston is about 90 degrees. Consequently, the phase measurement error will significantly affect the calculated values of acoustic power, so the measurement results for acoustic power are only for reference.

Cooling power at 80 K and 280 Hz is presented in Figure 10 as a function of pressure ratio. The mean pressure is 4.1 MPa. A maximum cooling power of 1.0 W at 80 K was achieved with a pressure ratio of 1.223 and an estimated acoustic power of 43 W. This is compared to a cooling power of 2.68 W at 100 Hz. First of all, 600# stainless steel mesh cannot meet the need for thermal penetration depth at 280 Hz, as mentioned earlier. In addition, some losses get larger at higher frequency operation, especially the losses associated with void volume. Meanwhile, unlike our results at 100 Hz, the experimental results and simulation results have a larger difference at 280 Hz.

A key issue is that our compressor can only achieve an efficiency of 30%. The piston diameter needs to be further optimized, and the coil resistance and mechanical damping coefficient need to be controlled to achieve lower losses. The compressor performance around 300 Hz should also be studied in the next work.

CONCLUSIONS

This paper describes the performance of a miniature coaxial PTC with an operating frequency above 100 Hz. The PTC was optimized using a theoretical model based on thermoacoustic theory. The simulation results indicate that, because of the absence of a suitable regenerator material with a very low void volume, the performance gets worse as the frequency increases, even though the best phase shifter impedance is used at the hot end of the pulse tube. To validate the results experimentally, a miniature coaxial pulse tube cooler was designed and fabricated. The influence of various parameters such as frequency and inertance tube were then investigated. In the experiments when operating at 100 Hz, the cooler was able to achieve a cooling power of 2.8 W at 77 K with an acoustic power of 40 W and reached a no-load temperature of 49.3 K. After the frequency was increased to 280 Hz, the no-load temperature increased to 63.3 K, and a maximum cooling power of 1.0 W at 80 K was achieved.

ACKNOWLEDGMENT

REFERENCES

1. R. Radebaugh and A. O'Gallagher, "Regenerator operation at very high frequencies for microcryo-coolers," *Advances in Cryogenic Engineering* 51B, AIP, New York (2006), pp.1919-1928.

2. S.Vanapalli, M.Lewis, Z. Gan, R. Radebaugh , "120 Hz pulse tube cryocooler for fast cooldown to 50 K," *Appl. Phys*. *Letters* 90 (2007), 072504.

3. Wei Dai, Guoyao Yu, Shanglong Zhu, Ercang Luo., "300 Hz thermoacoustically driven pulse tube cooler for temperature below 100 K," *Applied physics letters*, 90 (2007), 024104.

4. Xiaotao Wang, Guoyao Yu, Wei Dai, Ercang Luo, Yuan Zhou, "Influence of acoustic pressure ampli-fier tube on a 300 Hz thermoacoustically-driven pulse tube cooler," 2010.10 *Journal of Applied Phys-ics*, 2010.

5. Xiaotao Wang, Wei Dai, Ercang Luo et al, "Characterization of a 100 Hz miniature pulse tube cooler driven by a linear compressor," *Advances in Cryogenic Engineering*, 54 AIP, New York (2010), pp. 1077-1084.

6. G. Swift, "*Thermoacoustics: A unifying perspective for some engines and refrigerator,*" Sewickley, 2002, pp. 99-102 and 136-139, respectively

7. Xiao, J.H., *Thermoacoustic effects and thermoacoustic theory for regenerative cryocoolers (heat en-gines)*, Ph.D Dissertation, Institute of Physics, Chinese Academy of Sciences (1990) (in Chinese)

8. Dai, W., Luo, E. C., Zhang, Y., et al., "Detailed Study of a Traveling Wave Thermoacoustic Refrigera-tor Driven by a Traveling Wave Thermoacoustic Engine," *J. Acoust. Soc. Am.*, 119 (5), pp. 2686-2692.