Performance of a Stirling-Type Pulse Tube Cooler for High Efficiency Operation at 100Hz

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ABSTRACT

High efficiency pulse tube coolers driven by linear compressors usually operate between 30 Hz and 60 Hz. This article presents the performance of a high efficiency in-line PTC with an operating frequency of 100 Hz. The theoretical model of a PTC is based on thermoacoustic theory. Analytical results indicate that the PTC operating at 100 Hz could obtain a high efficiency with appropriate parameters. To couple the PTC, the parameters of linear compressor are optimized using compressor governing equations. Experimental results show that a no-load temperature of 31.8 K and a cooling power of 12.4 W at 77 K are obtained with an input electric power of 185.2 W.

INTRODUCTION

In comparison to Stirling or Gifford-McMahon (G-M) coolers, the pulse tube cooler (PTC) eliminates moving parts in the cold region, and brings the advantages of low cost, long life and low mechanical vibration. The Stirling pulse tube coolers driven by linear compressors usually operate between 30 Hz and 60 Hz and have been developed for decades.[1] The coolers operating at these frequencies have already reached a high level of efficiency.[2, 3, 4] Higher frequency operation will lead to a reduction in the size and mass of the compressor with the same acoustic power and, therein, provide a more compact system. In 2007, NIST setup an in-line Stirling PTC operating at 120 Hz, which provide a cooling power of 3.44 W at 80 K[5]. In 2008, NGST reported their coaxial pulse tube cooler operating at about 100 Hz and a cooling power of 1.3 W at 80 K was achieved with an input electric power of 49 W.[6]

The research of 100 Hz PTC at our laboratory has been carried out for years. By using the CFIC 2S102W compressor in our lab, a miniature pulse tube cooler was designed in 2008.[7] In the early experiments, a cooling power of 0.8 W at 80 K was achieved. Through a series of improvements, the cooling power at 80 K was raised to 2.6 W in 2009. Because this compressor is originally designed at 40-60 Hz, the compressor can only reach 44% of its design power even though the resonance frequency has been adjusted to 100 Hz. In 2009, a bigger PTC designed to provide 10 W cooling powers at 77 K is designed and fabricated. To couple with this cooler, a moving magnet linear compressor was designed and manufactured by Lihan Technology Inc. The cooler design and experimental system are introduced in the next sections. The main experimental results and further plan are also presented.

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PULSE TUBE COOLER DESIGN

The principal design goal of this cooler is to provide 10 W of cooling power at 77 K within 120 W of acoustic power. A numerical model[8] based on thermoacoustic theory[9, 10] is used for the optimization. A large number of calculations have been carried out to optimize the different components such as the regenerator, the pulse tube and the inertance tube. The key parameters of these components are chosen appropriately. The modeling was successful because the simulation results are very close to the experimental results.

Figure 1 shows the PTC cooling power at 77 K with different levels of input acoustic power. The regenerator is the key component of PTC and regenerator material will directly affect the cooler performance. Since the high frequency operation decreases the thermal penetration depth, the required hydraulic diameter in the regenerator is much smaller than that under conditions of 30 Hz to 60 Hz. The influence of the different regenerator hydraulic diameters is investigated, as shown in Figure 1. The performance is degraded when the hydraulic diameter is too large, but the difference is not great when the value of hydraulic diameter ranges from 26 to 34 µm. The cooler can provide 12.3 W of cooling powers with 125 W of input acoustic power. Too small a value for the hydralic diameter is not suitable because the pressure drop become bigger as the hydraulic diameter decreases.

The analytical results also indicate that an appropriate phase at the hot end of the pulse tube can be easily obtained at a high frequency with an inertance tube plus and a reservoir as the phase shifter.

The other goal of the simulation is to compute the acoustic impedance (Z) of the PTC, which is used to optimize the compressor parameters using the compressor governing equations. The compressor is a single-piston configuration. A spring-mass oscillator with a resonance frequency of 100 Hz is installed at the back of the compressor to reduce the vibration.

EXPERIMENT SYSTEM AND MEASUREMENT

Figure 2 shows the schematic of the experiment system with a in-line PTC configuration chosen. The input electric power is calculated by measuring the compressor's driving voltage and current. With the assumption of an adiabatic compression in the backside space of the compressor, the piston velocity is computed through the dynamic pressure measured in the backside space. The dynamic pressure and the mean pressure in the compression space and reservoir are also measured. The acoustic power (PV power) can be calculated approximately using the dynamic pressure in the compression space and piston velocity.

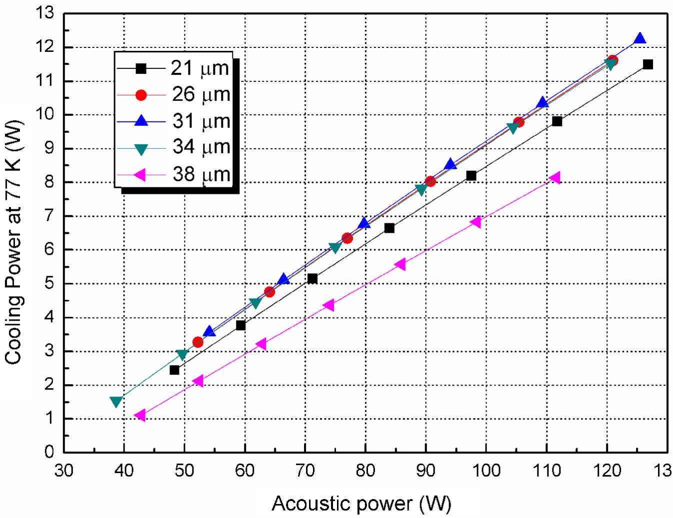

Figure 1. Cooling power at 77 K Vs Input acoustic power, different regenerator hydraulic diameter values are used in simulation.

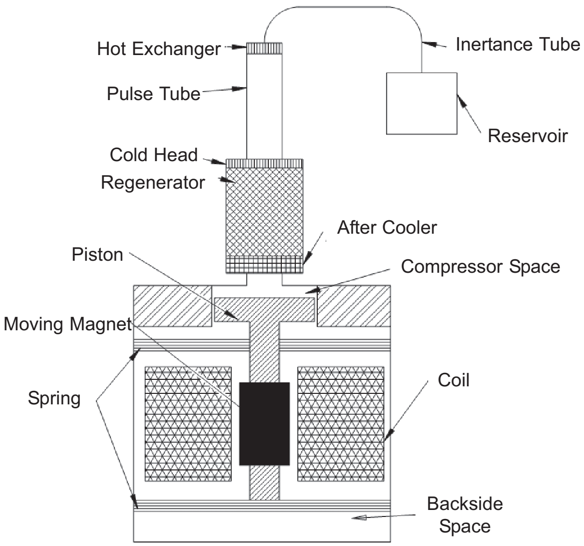

Figure 2. A schematic drawing of Stirling-type pulse-tube cooler driven by linear compressor

The ambient heat exchangers are kept at room temperature with chilling water at 288 K . The temperature of the cold-head was measured by a calibrated PT100 sensor. The cooling power of the PTC is measured via a resistor with the external DC power supply.

EXPERIMENT RESULTS AND DISCUSSION

Figure 3 shows the typical cool down curve of the cold-head with a driving voltage of 114 Vrms. The cold-head is made from copper with a mass of 53 g. The cool-down time to 77 K is 4.76 minutes from 297 K at a mean pressure of 3.55 MPa and it takes 4.33 minutes from 291 K to 77 K at 3.0 MPa. The input electric power becomes larger as the cold-head temperature decreases. The input electric power is about 183.2 W when cold-head reaches at the lowest temperature. The lowest cold-head

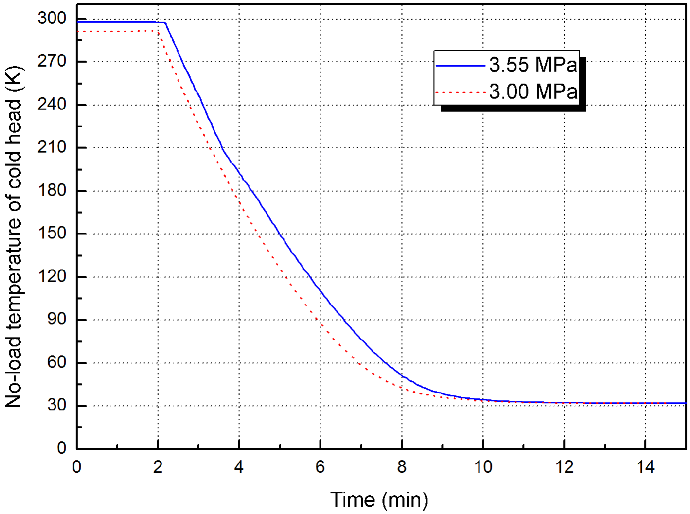

Figure 3. Cooldown curve of the cold head, with mean pressure 3.55 MPa and 3.0 MPa, respectively

temperature of 31.8 K at 3.55 MPa is acquired and the difference of the lowest temperatures between two mean pressures is very slight.

Figure 4 shows no load cold-head temperature at different input electric power at the mean pressure of 3.55 MPa. The cold-head temperature can be maintained at 52 K with 42 W input electric power and the acoustic power is about 28 W. This figure also shows the compressor efficiency. It can be seen that compressor efficiency decreases as the input electric power increases. With 183.2 W input power, the approximately calculated acoustic power using the dynamic pressures at both sides of the piston is 111.4 W with an compressor efficiency of 61 %, which is a little lower than our design goal.

The cooling power at 77 K with different input electric power is presented in Figure 5. The mean pressure usedi s 3.55 MPa. A cooling power of 12.4 W at 77 K is required with an input electric power of 185.2 W. The corresponding calculated acoustic power is 113.6 W. The whole

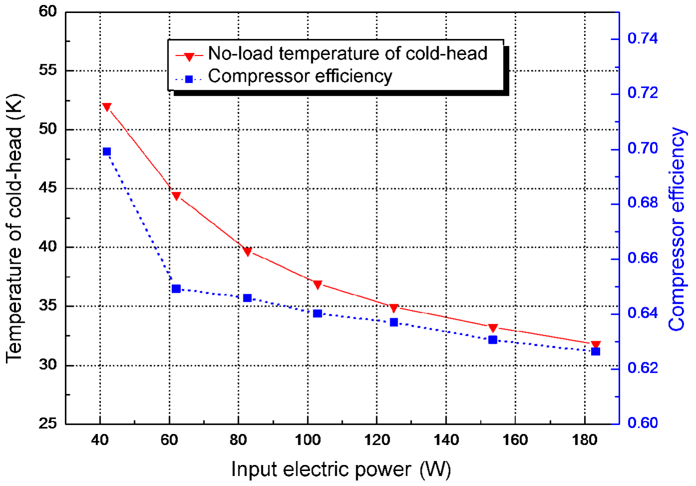

Figure 4. No load temperature of the cold head & compressor efficiency vs input electric power.

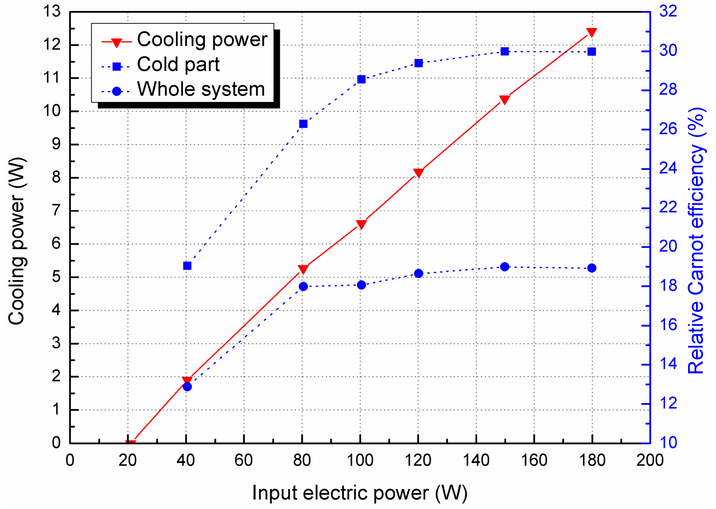

Figure 5. Cooling power at 77 K and relative Carnot efficiency vs input electric power. The whole system means the relative Carnot efficiency using input electric power, while the cold part means the relative Carnot efficiency using estimated acoustic power

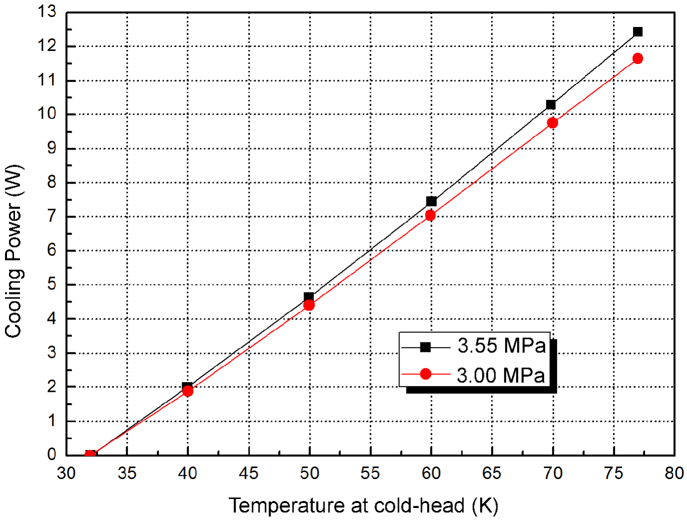

Figure 6. Cooling power at 77 K vs cold-head temperature.

system-relative Carnot efficiency is also shown in Figure 5 and it reaches 18.4 % at 185.2 W. In order to evaluate the performance of the cooler, the relative Carnot efficiency of the cold-head is also calculated using an approximation of the acoustic power and is presented in Figure 5. As this figure shows, the maximum Carnot efficiency of cold-head can reach approximately 30 %. Figure 6 shows the cooling power at different cold-head temperatures when the input electric power is maintained around 185 W. The cooling powers are a little lower at a lower mean pressure, although the lowest temperatures are almost the same.

Compared with the design goal, the compressor efficiency is not satisfied. In the next step, the design parameters of compressor will be optimized. The single piston configuration will be replaced by a dual-piston configuration because of lower vibration and easiness to adjust frequency. An air-cooled co-axial PTC will also be developed.

CONCLUSIONS

In this paper, the performance of a high efficiency in-line PTC with an operating frequency of 100 Hz is presented. The PTC is designed using a theoretical model based on thermoacoustic theory. In the experiments, a no-load temperature of 31.8 K and a cooling power of 12.4 W at 77 K is obtained with 185.2 W of input electric power. The corresponding relative Carnot efficiency of the whole system including compressor reaches 18.4 %. The PTC cool-down time is 4.76 minute from 297 K to 77 K. The simulation results agree well with the experimental results.

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