ON THE DEVELOPMENT OF CO-AXIAL MINIATURE PULSE TUBE COOLERS FOR SPACE APPLICATIONS

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ABSTRACT

Cryocoolers for cooling infrared sensors in space applications require high reliability, long lifetime, low power and minimum weight. In this paper we report work on a miniature pulse tube cooler specifically designed for such applications.

A series of engineering model co-axial miniature pulse tube coolers with a flexure bearing linear compressor of 1cc swept volume have been designed and fabricated in our laboratory. A theoretical model is established based on the analyses of thermodynamic and hydrodynamic behaviors of oscillatory flows in regenerator, for performance prediction, optimization and as a rough guide in the early stages of system design. An experimental apparatus, including a hot wire anemometer, has been set up to study the flow resistance of regenerators under oscillatory flow conditions. The co-axial, multi-bypass, and symmetric nozzle structure has been used in the coolers. We will present here the performance of two sizes of coolers with 9mm and 8mm diameter of cold fingers. The 9mm cooler currently provides 500mW net cooling power at 80K with input power of 47W, and the 8mm cooler, provides 450mW at 80K with 51W input power with a 65% efficient compressor. The cold fingers of our co-axial pulse tube coolers have the similar size of miniature Stirling coolers and are the only one that could meet the geometry specifications of the Standard Advance Dewar Assembly (SADA) for thermal imaging systems in most military applications.

INTRODUCTION

The growing need for cryogenic cooling of space-borne infrared devices has brought about a strong requirement for high reliability, long lifetime, low power and minimum weight in coolers. Miniature coolers with small cooling power at 65-80K are thus strongly required. Stirling coolers are compact and reliable machines and have been used for commercial application in the last few decades. However, they have a moving piston at the

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cold head, thereby the need of multiple sliding seals, which causes mechanical vibrations, electromagnetic interference (EMI), and possible limitation of lifetime. Although many technical methods have been used to damp the vibrations, they did not completely eliminate the vibrations thereby unreliability and high-cost of sliding seal, remains as a severe problem for many applications [1, 2].

The elimination of moving parts in the cold finger of pulse tube coolers gives them many potential advantages over Stirling coolers in many applications. Their development and improvement in recent years enables pulse tube coolers to be an ideal alternative of Stirling coolers in terms of design fabrication, efficiency, reliability, and lifetime for space applications. An extensive review of the development of pulse tube coolers as an efficient and reliable cryocooler can be seen in Reference [3].

There are three different geometries, in-line type, U type and co-axial (concentric) type, which have been widely developed for the system design of pulse tube coolers. Obviously, the in-line arrangement is the most efficient. Chan et al. [4] in TRW developed a in-line type miniature pulse tube cooler that achieved a cooling power of 0.53W at 80K with input power of 17.8W to a 1cc swept volume linear drive compressor. The cooler efficiency was about 8% of Carnot. Later, they improved the cold head and demonstrated that the new cold head has 3.7 times the cooling power at 65K of the previous design [5] with efficiencies as high as 24% of Carnot with their in-line miniature pulse tube coolers [6]. However, the cold head located at the middle region of the two warm ends is the main disadvantage for connecting to the cooled devices. The most compact and convenient for practical application, just like the geometry of Stirling cooler, is the co-axial type pulse tube cooler. It can replace Stirling cooler without any change to the Dewar or the connection to the cooled sensors. However, there are several potential design problems, like mismatch of temperature profiles between the regenerator and the pulse tube, void space at the cold end, and the reversal of the gas flow direction in the cold end space, that retard the overall performance of the co-axial pulse tube cooler. The problem has been minimized by various techniques. Recently, a coaxial pulse tube cooler developed at NIST used as an oxygen liquefier for NASA demonstrated 19W cooling power at 90K with 222W input PV power for an efficiency of 17% of Carnot [7]. Unfortunately, for the 0.15W to 1W at 80K low power miniature co-axial pulse tube coolers, the efficiency is still less than that of the Stirling coolers and other types of pulse tube coolers.

In order to improve the performance of co-axial pulse tube coolers to be used to cool space-borne infrared sensors, work on both the linear compressor and cold finger have been carried out in Cryogenic Laboratory, Chinese Academy of Sciences (CAS) since 1992. A computational model was established based on the analyses of thermodynamic and hydrodynamic behaviors of gas flows in oscillating flow regenerator, for performance predictions, optimization and as a rough guide in the early stages of cooler system design. An experimental apparatus, including a hot wire anemometer, has been set up to study the flow resistance of regenerators under oscillatory flow conditions. In recent year a series of engineering models of miniature pulse tube coolers incorporating flexure bearing linear compressor of 1cc swept volume have designed, fabricated, and tested in our laboratory. The co-axial geometry, multi-bypass version, and symmetric nozzle structure have been used in the pulse tube coolers. We have reported results on a 0.25W/85K class miniature pulse tube cooler, which produced 0.28W at 85K with the electrical input power of 30W from the compressor, whose efficiency was estimated only about 60% [8].

We will present in this paper the overall performance of two sizes of coolers with 9mm and 8mm outer diameter of the cold finger, respectively. The 9mm cooler currently provides 500mW net cooling power at 80K with input power of 47W; the 8mm cooler have 450mW at 80K with input power of 51W with a 65% efficient compressor.

THERMODYNAMIC ANALYSIS

The coefficient of performance (COP) of pulse tube coolers is defined as the ratio of the average cooling power $\dot{Q}_L$, obtained at the low temperature T_L, to the input power $\dot{W}_0$ from the compressor at high temperature T_H

$$COP = \frac{<\dot{Q}_L>}{<\dot{W}_0>} \tag{1}$$

where <> symbol indicates a time-averaged quantity. The power at the hot end of the regenerator is simply the PV power of the compressor.

The COP of a practical pulse tube cooler is mainly determined by irreversible processes in the regenerator, in the heat exchangers, in the pulse tube, in the orifices, etc. In the ideal pulse tube cooler that all the main components are assumed to be perfect and there is no flow resistance, no net heat flux, and no irreversible dissipations in them. The only loss is the irreversible expansion through the orifice. For an ideal gas, the COP of the ideal pulse tube cooler is given by the well-known result [9]

$$COP_{ideal} = \frac{T_L}{T_H} \tag{2}$$

It is well known that the COP of the Carnot cycle is

$$COP_{Carnot} = \frac{T_L}{T_H - T_L} \tag{3}$$

In an ideal case that isothermal compression and expansion processes are reversible and the regenerator and the heat exchangers are perfect, the COP of the ideal Stirling cooler is the same as the COP of Carnot. It is clear that $COPT_{ideal} < COP_{Carnot}$. This is due to the dissipation in the orifice compared to that of Stirling coolers. In order to achieve the same cooling power, additional compressor work is needed for an ideal pulse tube cooler due to the irreversible dissipation in the orifice than for an ideal Stirling cooler.

SYMMETRIC NOZZLE

Generally a pulse tube cooler requires a phase shifter, located at the hot end of the pulse tube, to get an optimal phase shift between mass flow and pressure wave at the warm end of pulse tubes to increase cooler performance. The orifice and double-inlet are two of the most well known configurations to cause a beneficial phase shift.

Adjustable needle valves are commonly used as the orifice and double-inlet valve as the phase shifter. It has been found that the oscillatory flow through the valves may cause a change in the open setting of the valves, so that the cold end temperature oscillated and the performance of the pulse tube cooler is not stable. In addition, the valves may cause irreversible dissipation in the orifice. From the discussion in last section it is of importance to minimize the dissipation through the valves at the warm end of the pulse tube. With a warm expander [10], it is possible to recover work from the pulse tube, but it introduces an additional moving part in the cooler system. Swift et al. [11] recently described a system of inertance and compliance elements to recover the acoustic power at the warm end of the pulse tube. However, the inertance tube is too long to be used in a strictly limited space and it is of little use in miniature pulse tube coolers.

It has been found in our Lab. that replacing the needle valves for the orifice and especially for the double-inlet with the symmetric nozzle results in the improvement of the performance under the same operating conditions [12]. Experimental data of a 1cc linear compressor driven pulse tube cooler showed that the low temperature reduced 3~8K, just replacing the needle valves in the orifice and double-inlet with the symmetric nozzle [8]. Figure 1 shows the schematic diagram of a typical symmetric nozzle of length L, of throat diameter d and of entrance diameter D. The critical parameters for a symmetric nozzle used in a pulse tube cooler is of the throat diameter d, ranging from 0.2~0.5 mm.

The improvement in performance by using a symmetric nozzle possibly comes from: (1) it may give a desirable phase shift between the mass flow and the pressure in the pulse tube; (2) it may cancel the intrinsic tendency for DC gas flow; (3) it may recover the work and feed it back into the pulse tube to help the compression and expansion processes since there exists the energy transformation for the gas pressure and velocity as the gas moves forth and back through a symmetric nozzle.

MULTI-BYPASS VERSION

According to our experiments [13-15] and computational calculations [16,17], the multi-bypass version is an effective way to improve the performance of pulse tube coolers. In the multi-bypass configuration, a bypass tube with flow impedance is added to a certain position between the regenerator and pulse tube, and to allow one or more streams of mass flow from the middle of the regenerator in or out the pulse tube directly due to the pressure drop between the two. The mass flow and the effect of the multi-bypass on the cooler performance depend on the position and the flow resistance of each bypass tube.

The advantages of the multi-bypass method are as follows. Firstly, it can regulate the phase shift between the pressure and mass flow in the pulse tube [13,15]. Secondly, it can improve the regenerator efficiency since part of gas flows into the pulse tube through the bypass tube, thereby reducing the quantity of gas flow through the regenerator. Thirdly, it can change the gas temperature distributions along the pulse tube and reduce the longitudinal heat flux between the cold and hot ends of the pulse tube. And last, it can reduce the effect caused by DC gas flow by allowing part of the DC flow from the warm end to flow to the midpoint of the regenerator through the by-pass tube instead flowing directly to the cold end [15,17].

In the co-axial pulse tube cooler, the regenerator and the pulse tube are assembled co-axially. It is easy to use the multi-bypass version in the co-axial arrangement by drilling a suitable hole in the common concentric tube.

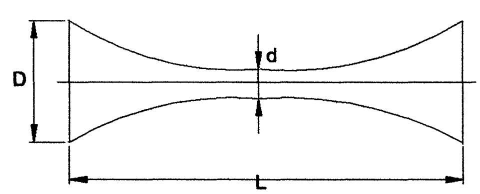

FIGURE 1. Schematic diagram of a typical symmetric nozzle

FLOW RESISTANCE OF OSCILLATING FLOW REGENERTATOR

The ability to predict pressure drops and phase shifts of the regenerator is crucially important in optimum design for the pulse tube cooler, especially in the high frequency miniature coolers. There are important differences between the steady and the oscillating flow. It is apparent that the correlations based on the steady flow would not be able to predicate accurately the pressure drop in the oscillating flow regenerator. Many researchers [18-21] found that higher friction factor in oscillating flow regenerators than that in steady flow at the same Reynolds number. The oscillating behavior for regenerators demonstrates not only pressure drops but also phase shifts.

Since 1996, our laboratory has been working on the experimental measurements of the flow resistance characteristics of the regenerator under the oscillating flow conditions. A test apparatus, including a hot-wire anemometer, was designed and set up. The pressure drops and phase shift of the gas through the test section (made of stacked screens) of the regenerator subjected to an oscillatory flow generated by a self-made compressor have been investigated. In all tests, we studied experimentally the effects of the orifice setting, the system average pressure and the mesh size on the flow resistance of the regenerator.

The maximum and cycle-averaged friction factors were correlated in terms of Reynolds numbers and dimensionless distance based on the cross-sectional mean velocity and frequency. The dimensionless phase shift between the gas velocity and pressure wave was also correlated. It showed that the product of the friction factor and the dimensionless distance increased as the Reynolds number decreased. The dimensionless phase angle tended to be higher at lower Reynolds numbers.

We compared the pressure drops from experimental data for the oscillating flow and the steady flow and found that the cycle-averaged pressure drop of the oscillating flow is 2 to 3 times higher than that of the steady flow calculated by the correlations given by Kays and London [22] at the same Reynolds numbers based on the cross-sectional mean velocity. The test conditions are very close to the operating condition of the practical high frequency pulse tube cooler, resulting the experimental data is useful for our performance prediction and practical design. Measurements on the flow resistance of the regenerator at different operating temperature are being carried out.

PERFORMANCE OF THE COOLERS

Based on the above technology developments and using self-made pressure oscillators a series of engineering model co-axial miniature pulse tube coolers, specifically designed to be used to cool space-borne infrared sensors, have been fabricated and tested in our laboratory. The pressure oscillator is a single piston flexure bearing linear compressor and has a maximum swept volume of 1.06cc. The compressor uses an AC power source with a voltage range from 12 to 16V and a current range from 2.8 to 3.5A, resulting in large joule heating as high as 25W in the coil. The compressor is not so efficient and is estimated only about 65% efficient compared with 85% efficient compressors typically used for a Stirling cooler in most space applications. The pulse tube cooler is of a co-axial structure with double-inlet and multi-bypass, as illustrated in Figure 2. The cooler looks like a split Stirling cooler. Two cold fingers, one with a 9mm outer diameter and the other with 8mm outer diameters with total length of 65mm, are fabricated and tested. In order to reduce the conduction losses, both the two pulse tubes are made of Teflon with a wall thickness of 0.5mm, an inner diameter of 3.5mm and a length of 58mm. The regenerators for the two cold fingers are of 8mm and 9 mm in outer diameter, respectively, 50mm in length, and made of titanium alloy filled with 400 mesh stainless steel screen. The flow channel of the pulse tube and regenerator is connected by a cold tip heat exchanger. Symmetric nozzles for the orifice and the double-inlet are welded inside a gas buffer with a volume of 40cm^3. The hot end of the cold finger is bolted on a hot end flange and is connected to the compressor by a copper coil as long as 0.5m in order to arrange conveniently the cold finger fit into envelopes (Standard Dewar) to cool infrared sensors. This design is very compact and compatible for space application.

Two small piezoresistive sensors were placed at the hot ends of the pulse tube and the regenerator to measure the pressure oscillations. The temperatures of the cold tip and the hot end of the cold finger were measured with thermocouples. A resistance heater was provided at the cold tip for cooling power measurement at a particular temperature. The current and the applied voltage were measured to determine the cooling power.

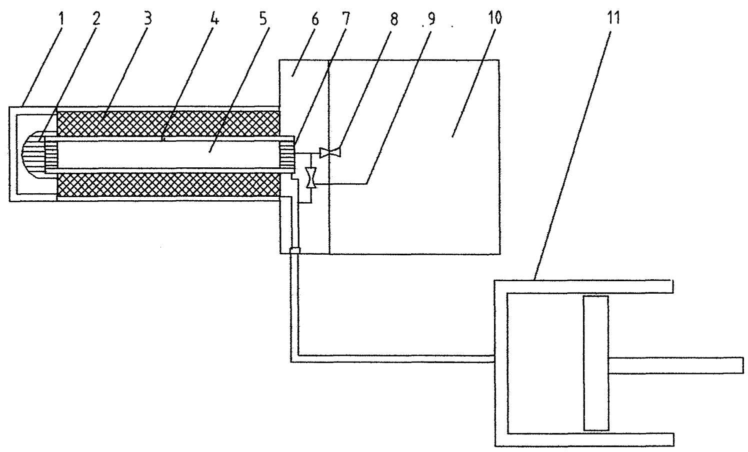

FIGURE 2. Schematic diagram of the co-axial miniature pulse tube cooler
1. cold tip; 2. flow straighter; 3. regenerator; 4. multi-bypass; 5. pulse tube; 6. hot end flange; 7. flow straighter; 8.orifice; 9. double-inlet; 10. gas buffer; 11. compressor

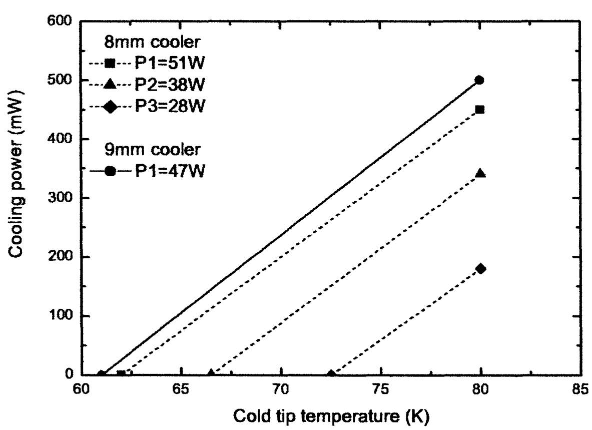

FIGURE 3. Cooling power as a function of the cold tip temperature with different input power

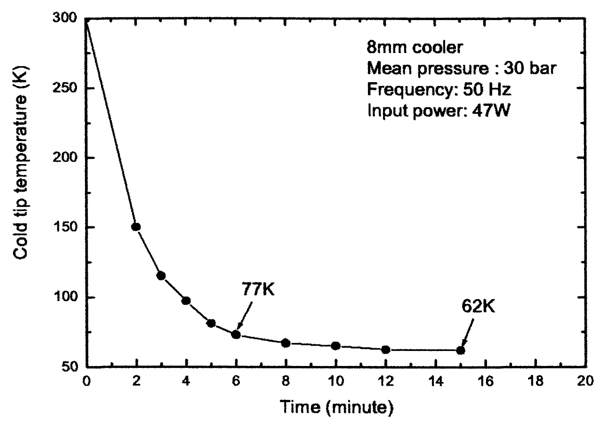

FIGURE 4. Typical cooling-down rate of the co-axial miniature pulse tube coolers

Test results at the input mean pressure of 30bar and at operating frequency of 50Hz are given in Figures 3 and 4. Figure 3 shows the cooling power as function of the cold tip temperatures with different electronic input power to the compressor. The 9mm-cooler currently provides 500mW net cooling power at 80K with input power of 47W, and 300mW with input power of 40W. The 8mm-cooler provides 450mW at 80K with input power of 51W. The cooling-down process of the 8mm pulse tube coolers is depicted in Figure 4. It takes about 6 min to reach 77K and 12 min to the lowest temperature of 61K. The cooling-down time of 9mm pulse tube cooler is almost the same as that of 8mm cooler.

It should be noted that the self-made compressor used for our co-axial pulse tube coolers is not so efficient and it is estimated to convert electrical input power to PV power only at about 65% efficiency. As we know, a well-designed linear compressor operating at the resonant frequency of the oscillatory piston can convert electrical power to PV power at an efficiency of about 85%. Provided an 85% efficient compressor is used, the 9mm cooler would produce 0.5W at 80K with 17W input PV power. In other words, the cooler would have an efficiency of 8% of Carnot compared to an efficiency of 14% of Carnot based on our theoretical prediction. For comparison, one of the first miniature pulse tube coolers, described by Chan et al. [4] for space applications, was the 0.5W/80K in-line miniature pulse tube cooler. It achieved a cooling power of 0.53W at 80K with input power of 17.8W and was about 8% of Carnot.

The cold fingers of the 8mm and 9mm outer diameter co-axial pulse tube coolers have the similar size of miniature Stirling coolers required for the geometry specifications of the Standard Advance Dewar Assembly (SADA) for thermal imaging systems in most military applications. The co-axial pulse tube cooler is the only one that can be located at the warm end and fitted inside the SADA sleeve.

Lifetime of 5 years is the basic requirement for most space applications. To our knowledge, continuous operation of such coolers has shown that a cumulative contamination problem can arise and therefore major risks including temperature instability and thermal efficient degradation are involved in long-term operation. We have designed and built a life-test experimental facilities, which incorporate a number of specific measurements such as the temperature stability and efficiency degradation as function of time, environment temperature, humidity, the start/stop cycles over mission life, the thermal-vacuum heat-sink conditions, etc. Life test is currently underway in our lab.

CONCLUSION REMARKS

A series of engineering models of miniature pulse tube coolers incorporating a flexure bearing linear compressor have been designed, fabricated, and tested in our lab. The co-axial geometry, multi-bypass version, and symmetric nozzle structure have been used in the coolers in order to demonstrate practical use. We presented here the performance of two sizes of coolers with 9mm and 8mm outer diameter of the cold fingers, respectively. The 9mm cooler currently provides 500mW net cooling power at 80K with input power of 47W and the 8mm cooler provides 450mW at 80K with input power of 51W with a 65% efficient compressor. The further development of such miniature coolers has benefited much from the development of reliable, high efficient linear compressors.

The cold fingers of our 8mm and 9mm outer-diameter coaxial pulse tube cooler are the only one that meet the geometry specifications of the Standard Advance Dewar Assembly (SADA), which were developed many years ago when the JT and Stirling coolers were the only options for cooling infrared sensors in most military tactical applications. Compared with Stirling coolers, our co-axial miniature pulse tube coolers are still less efficient, but have advantages of easy fabrication, less vibration, high reliability and possible long-lifetime. Improvements are being carried out to further increase the cooling power and the efficiencies of compressors.

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