Characteristics of a VM Type Thermal Compressor for Driving a Pulse Tube Cooler

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ABSTRACT

The Vuillemier (VM) type cryocooler is a classical kind of cryocooler. It utilizes the temperature difference, typically the temperature difference between liquid nitrogen and ambient, to generate the pressure wave to drive the cold head. The subsystem that generates the pressure wave could be viewed as a type of thermal compressor called the VM thermal compressor. From the thermoacoustic viewpoint, the output of the thermal compressor is strongly related to the characteristics of the load. The paper studies the characteristics of the thermal compressor with simulations based on thermoacoustic theory. Interesting features, such as the dependence of local acoustic characteristics on the load, and distribution of energy flow have been studied. The study sheds new light onto the working mechanism of the VM thermal compressor as well as setting the basis for optimizing a VM type thermal compressor to drive a pulse tube cooler at liquid helium temperature.

INTRODUCTION

Small scale 4 K cryocoolers have found important applications for cooling superconducting magnets in MRI instruments, and high sensitivity detectors or uses in low temperature physics. GM cryocoolers, and most recently the GM type pulse tube coolers, are the main technologies for this temperature region. With the advances in modern technologies, more requirements are put on the specific power, efficiency and ease of maintenance. For this reason, during the most recent decade, researchers have started to investigate the possibility of using the Stirling type pulse tube cooler to generate liquid helium temperature with a much smaller weight, a higher efficiency and low maintenance requirement. Using a four stage configuration, Lockheed Martin has successfully built a pulse tube cryocooler which can reach a temperature of 4.85 K with He-4 and 3 K with He-3 [1,2]. Qiu et al, with a thermally-coupled configuration, also reached a temperature of 4.26 K [3]. While the potential of the Stirling type configuration has been shown, the need for a high operating frequency (which is a key requirement of compressor resonance for high efficiency operation) and low-void-volume 4 K regenerators may lead to a poor efficiency, even when compared to GM technology which is normally 0.5-1% of Carnot efficiency. To look for more alternatives for 4 K region, the authors propose another configuration which is based on the VM thermal compressor concept.

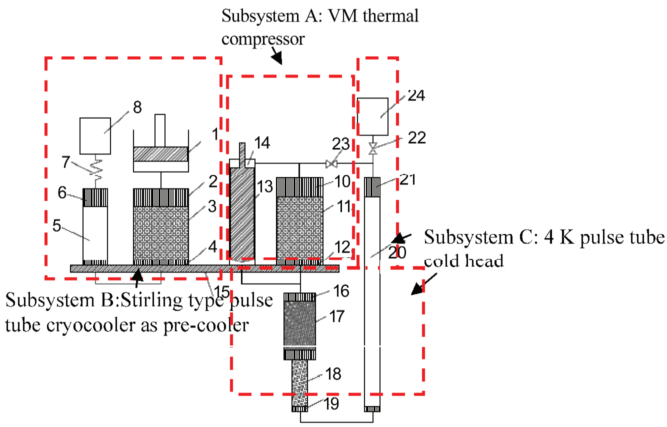


Figure 1. Illustration of the VM type pulse tube cooler for liquid helium temperature,
Subsystem B: 1.linear compressor, 2.Ambient HX, 3. Regenerator, 4.Cold HX, 5.Pulse tube, 6. Secondary ambient HX, 7. Inertance tube, 8. Reservoir;
Subsystem A:10. Ambient HX, 11.Regenerator, 12.Cold HX, 13.Displacer, 14. rod;
Subsystem C:17.2nd Regenerator,18.3rd regenerator, 19.4 K stage cold head, 20. 4 K Stage pulse tube, 21. 4 K stage secondary ambient HX, 22.23 and 24. 4 K stage phase shifters

The VM concept is a classical configuration [4]. Shown as Subsystem A in Fig.1, the so-called VM thermal compressor consists of a displacer, a regenerator, an ambient and cold heat exchanger. As the displacer piston displaces the gas between the ambient end and cold end, the pressure wave will be generated due to the change in gas temperature. In the 2000s, Matsubara et al first proposed using a modified VM thermal compressor with a thermal buffer tube to drive a pulse tube cooler and reach a temperature of 4 K with a GM cryocooler as the 20 K precooling stage [5,6]. The existence of the thermal buffer tube reduces the available pressure ratio and using liquid nitrogen is inconvenient. In this paper, the configuration is further evolved into the one shown in Fig.1. A classical VM compressor is used to drive the pulse tube cold head. A high efficiency Stirling type cryocooler is used to provide the required cooling power for the VM thermal compressor for an easy plug-and-go operation. Another advantage of this configuration is that the cold end temperature of the VM thermal compressor can be easily tuned for a high overall system efficiency.

This paper mainly deals with the thermoacoustic analysis of the thermal compressor subsystem. If not specially mentioned, words like regenerator or heat exchanger all refer to the component in this subsystem. Another paper in this conference [7] mainly deals with the CFD simulation of the lowest stage regenerator (i.e. component No.18 in Fig 1) which is most critical for the liquid helium operation. These studies shall all be grouped together for the design of the whole system. In the following, the 2nd section briefly introduces the thermoacoustic model. The 3rd section introduces the results. Finally, some conclusions are drawn.

NUMERIC MODEL AND GEOMETRICAL CONFIGURATION

Although theoretical analyses have long existed for the VM cryocooler system, most recent development of thermoacoustic theory has shed new insight into the working mechanisms of those cryocoolers and engines based on oscillatory flow [8]. For this reason, this paper chooses to use thermoacoustic theory to simulate the VM thermal compressor system. As will be shown later in the results section, the operation mode inside the regenerator changes from a standing wave without a load towards a travelling wave with a load, which is one of the interesting feature depicted through the analysis.

Table 1. Details of the VM Compressor

Displacer	Displacer dome diameter 6 cm, displacer rod diameter 1 cm, the upper and lower side of the dome is 2 cm long space for the displacer to move
Regenerator	I.d. 4 cm, length 15 cm, filled with 60# stainless steel mesh
Heat exchangers	Both are of length 2 cm, i.d. 4 cm, 50% porosity
Operating conditions	Charge pressure is 1.2 MPa, Operation frequency is 5 Hz for a relatively large specific power, ambient temperature is 300 K

The numeric model used here is based on a modular Matlab-based software package developed by the author. This model has been validated with the design of pulse tube coolers and free piston Stirling engines. For conciseness, the reader could refer to Ref. [9] for details. Simply speaking, the periodically-varying variables such as pressure, mass flow and temperature are taken as being sinusoidal. The control equations are simplified to retain the first order terms and re-written in the complex frequency domain. Then the equations are solved numerically to generate the distributions of variables which are further processed to show energy flow pictures.

In the model, gas flow is assumed to be laminar. Properties of the regenerator material and the gas are temperature-dependent. Thermal losses and clearance losses through the displacer are neglected. The average pressure is set at 1.2 MPa due to the non-ideal He-4 properties when the thermal compressor is used to drive the 4 K pulse tube cold head. The operating frequency is set to 5 Hz for a relatively high specific power. Since the driving power for the displacer is quite small, resonance or off-resonance operation of the driving motor only has a little effect on the overall system efficiency. This eases the design of the motor as compared with that of a linear resonant compressor in the Stirling type configuration [1,2] if we also want it to operate below 10 Hz.

RESULTS AND DISCUSSIONS

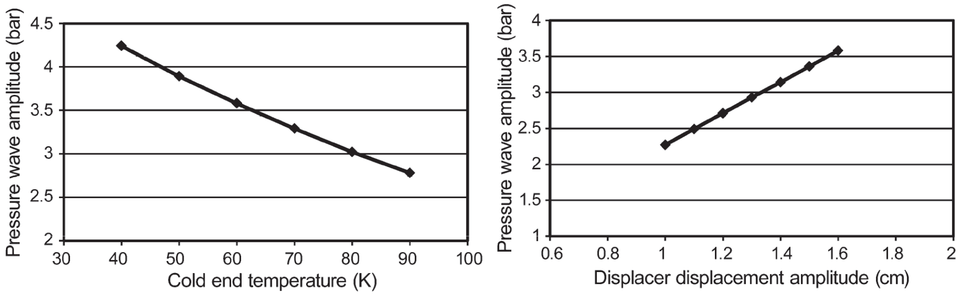
The analyses are grouped into two categories: with a load and without a load. The load is an RC load, i.e., resistance and capacitance, which typically represents the acoustic impedance at the warm end of the regenerator, i.e., at the entrance of component 17 in Fig.1. In this way, a quicker optimization of the system could be realized. In the following results, the x-axis starts from the top of component 10 in Fig.1 and heads downwards to the warm end.

Operation Characteristics Without Load

Without a load, the analysis of the VM thermal compressor is straightforward. Under this condition, only a small cooling power is required to compensate for the losses through the regenerator. Figures 2(a) and (b) show the dependence of the generated pressure ratio on the cold end temperature and on the displacement amplitude. It is easily understood that the pressure ratio increases with an increase in the temperature ratio between the ambient and the cold end, and also with an increase in the swept volume of the displacer. At the maximum pressure wave, the input work of the displacer is less than 10 watts to overcome the flow resistance through the regenerator and heat exchangers. Figure 3 further shows a typical phase difference between the pressure wave and the volume flow rate. The difference is very close to 90 degree, which means the regenerator operates in a standing wave mode.

Operation Characteristics With Load

As addressed in the beginning of the section, an RC load is used to represent the subsystem *C* depicted in Fig. 1. Here we only give a typical behavior of the VM thermal compressor with a limited range of load impedance. A more systematical study is currently underway.



(a) (b)

Figure 2. Depedence of pressure wave amplitude on (a) the cold end temperature, (b) the displacer displacement amplitude

Firstly, with a cold end temperature of 60 K, a displacement amplitude of 1.6 cm and a typical impedance of 9.5e8@(-50 degree), the thermal compressor can deliver up to 25.7 W acoustic power to the load. Meanwhile, 47.5 W cooling power is required from the Stirling type pulse tube cryocooler. The thermal efficiency in terms of power output divided by heat input at the ambient heat exchanger is 45.6%, which corresponds to a relative Carnot efficiency of about 57.0%. It should be noticed that part of the cooling power requirement is from the heat flow coming from lower stage. Fig.4 (a) and (b) show the acoustic power and enthalpy flow distribution inside the thermal compressor. For comparison, the phase difference between the pressure wave and the volume flow rate is also shown in Fig. 3, which clearly shows that the mode is shifting from nearly 90 degree towards a mode with more travelling wave component. Figure 5 fshows the power output and relative Carnot efficiency of the thermal compressor as the load changes the phase angle. It can be seen the output power and thermal efficiency changes dramatically with the phase angle, which means that we should carefully design the subsystem C shown in Fig.1 for a good overall system efficiency and power output.

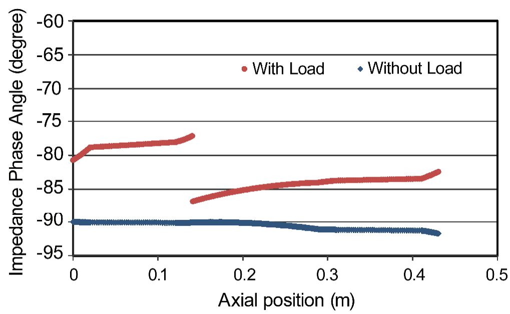
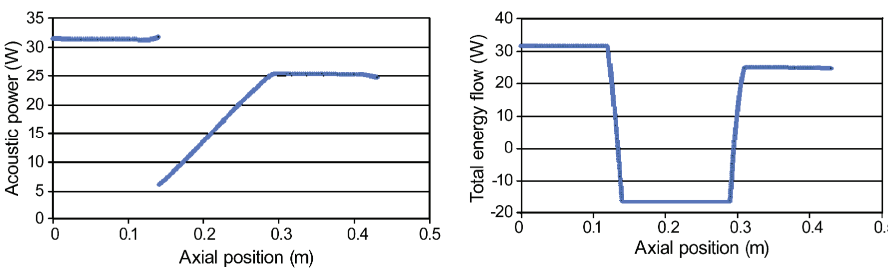


Figure 3. The distributions of phase angle across the thermal compressor



(a) (b)

Figure 4. Distribution of acoustic power and enthalpy flow inside the system

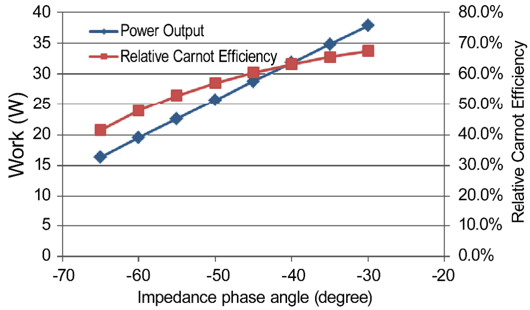


Figure 5. Power output and relative Carnot efficiency of the thermal compressor as the load

CONCLUSIONS

This paper presents a modified configuration of a 4 K pulse tube cryocooler driven by a thermal compressor. The infrastructure could be one of the candidates for future portable, dry and more efficient cryocooler working at liquid helium temperature. Simulation based on thermoacoustic theory has shown some behaviors of the thermal compressor system. One of the interesting features is that the regenerator works at standing wave mode without load but changes towards traveling wave mode when a typical load is connected. Meanwhile, the thermal compressor output and efficiency is strongly related to the load characteristics. More systematical study is currently underway for a better design of the whole system.

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REFERENCES

1. Olson J., Moore M., Champagne P., et al., “Development of a space-type 4-stage pulse tube cryocooler for very low temperature,” *Adv. in Cryogenic Engineering*, Vol. 51, Amer. Institute of Physics, Melville, NY (2006), pp.623-631.
2. Nast T., Olson J., Champagne P., et al. “*Development of a 4.5 K pulse tube cryocooler for superconducting electronics*”, *Adv. in Cryogenic Engineering*, Vol. 53, Amer. Institute of Physics, Melville, NY (2008), pp. 881-886.
3. Zhi X., Han L., Dietrich M., et al, “*A three-stage Stirling pulse tube cryocooler reached 4.26 K with He-4 working fluid*”, Cryogenics, Vol.58 (2013), pp.93-96.
4. G. Walker, *Cryocoolers*, Plenum Press, New York (1983).
5. Matsubara Y., Zhou S., “Feasibility study of applying thermal compressor to 4 K pulse tube cooler,” *Proc of the 18th ICEC* (2000) , p.535
6. Dai W., Matsubara Y. and Kobayashi H., “Experimental results on VM type pulse tube refrigerator,” *Cryogenics*, Vol.42, Issue: 6-7 (2002), pp.433-437.
7. Zhao, Y., Dai, W., Chen, Y., Wang, X., Luo, E., “Numerical Analysis of a 4 K Regenerator in a Pulse Tube Cryocooler,” *Cryocoolers 18*, ICC Press, Boulder, CO (2015), (this proceedings).
8. Swift, G.W., *Thermoacoustics: A Unifying Perspective for Some Engines and Refrigerators*, Acoustical Society of America Publications, Sewickley, PA (2002)

9. Dai W., Luo E., Zhang Y., et al, "Detailed study of a traveling wave thermoacoustic refrigerator driven by a traveling wave thermoacoustic engine," *Journal of Acoustical Society of America*, 119(5) (2006), pp.2686-2692.