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A COMPARISON OF THREE TYPES OF PULSE TUBE REFRIGERATORS: NEW METHODS FOR REACHING 60K*

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ABSTRACT

Pulse tube or thermoacoustic refrigerators require only one moving part--an oscillating piston or diaphragm at room temperature. Refrigeration occurs within a tube connected to the pressure wave generator when the thermal relaxation time between gas and tube is comparable to a half period. Three types have been discussed in the literature recently by Gifford, by Mikulin, and by Wheatley. A record low temperature of 60 K was achieved in our work using a single stage pulse tube similar to that of Mikulin. Previously 105 K was the lowest temperature achieved. Because of only one moving part, all three types have the potential for long life, but their efficiency and intrinsic limitations have never been investigated. This paper compares the three types with each other and with common refrigerators such as Joule-Thomson and Stirling refrigerators. An apparatus is described which can measure the intrinsic behavior of the different types from temperatures of about 30 K to 300 K. Overall cycle efficiency as well as sources of loss such as conduction and regenerator ineffectiveness are discussed and the advantages of various phase shifting techniques to increase refrigeration capacity are compared.

INTRODUCTION

High reliability in small cryocoolers has been a problem which has been studied for many years. One approach to increased reliability is the elimination of some of the moving parts in a mechanical refrigerator. Stirling refrigerators have only two moving parts--the compressor piston and the displacer. In 1963 Gifford and Longsworth[1] discovered a refrigeration technique which eliminates the displacer from the Stirling refrigerator. They called this new technique pulse tube refrigeration. Under Gifford's direction,[2-5] the pulse tube refrigerator was advanced to the point where temperatures achieved were 124 K in one stage and 79 K in two stages.[4] A single stage unit operating from 65 K achieved 30 K.[4]

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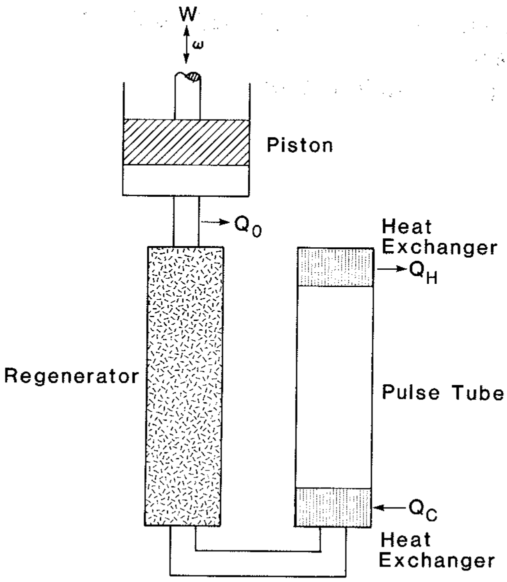

Fig. 1. Schematic of basic pulse tube refrigerator.

A schematic of the basic pulse tube refrigerator is shown in Fig. 1. The principle of operation, as given by Gifford and coworkers[1-5] and by Lechner and Ackermann[6], is qualitatively simple. The pulse tube is closed at the top end where a good heat transfer surface exists between the working gas (helium is best) and the surroundings in order to dissipate heat Q_H. The bottom end is open but also has a good heat transfer surface to absorb refrigeration power Q_C. The open end is connected to a pressure-wave generator via a regenerator. During the compression part of the cycle any element of gas in the pulse tube moves toward the closed end and at the same time experiences a temperature rise due to the adiabatic compression. At that time the pressure is at its highest value P_h. During the plateau in the pressure wave, the gas is cooled somewhat by heat transfer to the tube walls. In the expansion part of the cycle the same element of gas moves toward the open end of the pulse tube and experiences a cooling due to the adiabatic expansion. During the plateau at P_ℓ the gas is warmed through heat transfer from the tube walls. The net result of cycling the pressure in this manner is a shuttle heat-transfer process in which each element of gas transfers heat toward the closed end of the pulse tube. Note that the heat pumping mechanism described here requires that the thermal contact between the gas and the tube be imperfect, so the compression and expansion processes are somewhere between isothermal and adiabatic. The best intermediate heat transfer generally occurs when the thermal relaxation time τ_t between the gas and the tube walls is $\omega\tau_t \approx 1$, where ω is the frequency of operation in rad/s.

The work of Gifford and coworkers in the mid 1960's on the basic pulse tube led to some semi-empirical expressions for the performance. Overall efficiencies were estimated to be about comparable to a Joule-Thomson refrigerator. The low temperature of 124 K using a pressure ratio of 4.25/1 was certainly interesting, but most practical applications begin at about 80 K or below. No further work on pulse tube refrigeration occurred until the last few years when some new concepts were introduced. The similarities and differences between the various types are discussed in the next section.

THREE PULSE TUBE TYPES

Basic Pulse Tube

The basic pulse tube refrigerator as developed by Gifford and coworkers[1-5] has already been discussed and is shown in Fig. 1. The early work on these refrigerators was done using a valved compressor and a rotary valve to switch the pulse tube between the high and low pressure side of the compressor. Such a technique lowers the overall efficiency since no work is recovered in the expansion process even though the actual refrigeration process is improved because of the short times for compression and expansion and long times for heat transfer at P_h and P_ℓ. The lowest temperature of 124 K was achieved in 1967 by Longsworth[4] on a pulse tube 19 mm diameter by 318 mm long with the top 31.8 mm made of copper instead of stainless steel to provide an isothermal section. The high and low pressures were 2.38 MPa and 0.56 MPa, respectively, for a pressure ratio, P_r, of 4.25 at a frequency of 0.67 Hz. A gross heat pumping rate of 5 W was achieved at the low temperature limit of 124 K, which was totally absorbed by loss terms such as conduction and regenerator ineffectiveness.

Also in 1967 Gifford and Kyanka[5] used a valveless compressor, as shown schematically in Fig. 1, to recover work in the expansion process and increase overall efficiency. The pressure wave in that case is more sinusoidal than a square-wave. The pressure ratio used in this work was 4.2/1 and a low temperature limit of 165 K was achieved for one stage as depicted on the temperature scale of Fig. 2. The pulse tube consisted of a bundle of 4 stainless steel tubes each 6.35 mm diameter by 102 mm long with the top 10% as the hot isothermal section. The pressures were P_h = 2.17 MPa and P_ℓ = 0.52 MPa at a frequency of 6.2 Hz. They estimated that the work input was about 10% of an equivalent system with a valved compressor.

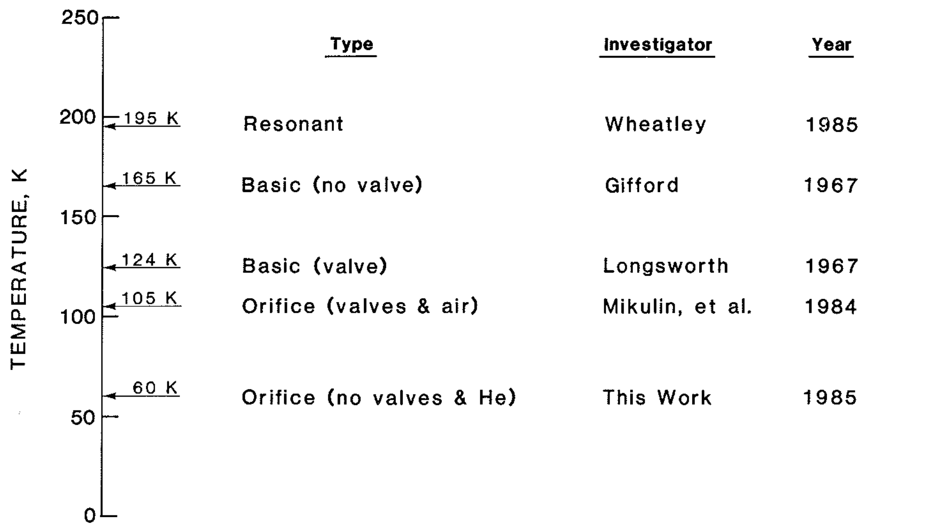

Fig. 2. Temperature limits reached with various pulse tube refrigerators.

Resonant Pulse Tubes

In 1975 Merkli and Thomann[7] showed that in a gas-filled tube, closed at one end and driven at resonance with an oscillating piston at the other end, cooling of the tube walls occurs at positions near the center of the tube (pressure node) and heating occurs at the ends (pressure antinode). Since no internal structure was used in the tube, the high frequency of 122 Hz resulted in nearly adiabatic conditions ($\omega\tau_t >> 1$) so little refrigeration was produced. More recently Wheatley[8,9] has improved the resonant pulse tube or thermoacoustic refrigerator by adding internal structure at the appropriate position along the tube to get $\omega\tau_t \approx 1$ (see Fig. 3). The internal structure of the 39 mm diameter tube consisted of cloth-epoxy plates 67 µm thick and spaced by 380 µm. A frequency of 516 Hz was used with a mean pressure P_m of 1 MPa and a dynamic pressure amplitude of 0.034 P_m. The lowest temperature achieved with this type of device has been 195 K as depicted in Fig. 2.[9] Note that this device has no regenerator and the thermodynamically active region in the gaps between plates is located differently from that of the basic pulse tube. The reason for this difference will be explained later but the resonant pulse tube can use the same geometry, including regenerator, as in the basic pulse tube.[10]

Orifice Pulse Tube

In 1984 Mikulin et al[11] inserted an orifice at the top of the pulse tube to allow some gas to pass through to a large reservoir volume. Figure 4 shows our version of this modification although the original work of Mikulin et al[11] placed the orifice just below the isothermal section instead of above it as in our work. They applied the pressure

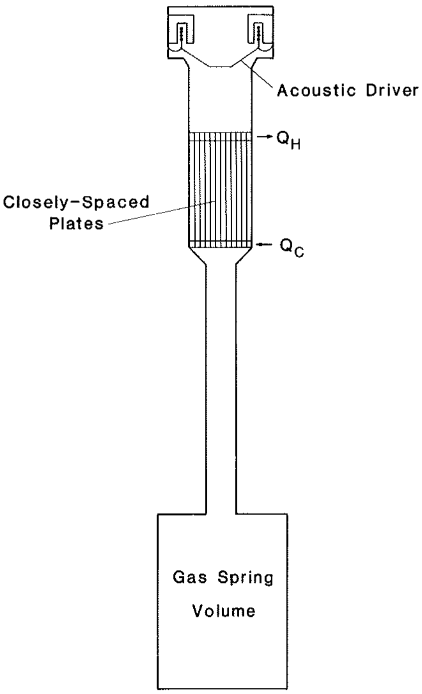

Fig. 3. Schematic of the resonant pulse tube (thermoacoustic) refrigerator.

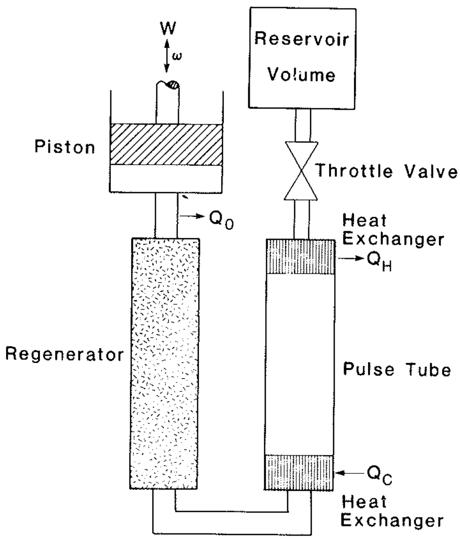

Fig. 4. Schematic of the orifice pulse tube refrigerator.

wave by the use of valves and used air as the working gas even though the Joule-Thomson effect is not used in the orifice. They were able to achieve a low temperature of 105 K using a pulse tube 10 mm in diameter by 450 mm long with pressures of $P_h = 0.4$ MPa and $P_\ell \approx 0.2$ MPa at a frequency of 15 Hz. The net refrigeration capacity at about 120 K was 10 W. It is not clear from their work what fraction of the gas passes through the orifice.

Experiments in our laboratory were done using the modified geometry of Fig. 4. The orifice was a needle valve which could easily be adjusted to give the minimum temperature. The pulse tube was a thin wall stainless steel tube 12.7 mm O.D. by 237 mm long. A 22 mm long copper section was soldered to the top end and packed with 80 mesh copper screen to enhance heat transfer to the water-cooling coil wrapped on the outside of the copper section. The reservoir volume, large when compared with the pulse tube, was 1 L. The regenerator was a stainless steel tube 19 mm diameter by 127 mm long filled with 1050 discs of 150 mesh phosphor-bronze screen. A valveless compressor was used to provide the pressure wave to the system using helium gas.

Optimum orifice adjustment gave a low temperature of 60 K with P_h = 1.24 MPa and P_ℓ = 0.71 MPa (P_h/P_ℓ = 1.75) and a frequency of 9 Hz. The orifice adjustment which gave 60 K was such that the pressure variation in the reservoir volume was about 0.9% of the average pressure. It was then calculated that mass flowing in and out of the reservoir volume was about half the total mass flow at the cold end of the pulse tube. Gross heat pumping values from the cold to hot ends were about 18 W at 60 K with those conditions. Net refrigeration at higher temperatures was not measured. Shortening either the regenerator or the pulse tube led to higher minimum temperatures. With our test setup we were unable to completely optimize the frequency and amplitude of the pressure wave. Evidence indicated that a lower temperature would have been achieved at higher frequency and perhaps also a greater compression ratio, P_h/P_ℓ.

Multiple Stage Pulse Tubes

It is easy to add any number of stages, still using one pressure wave generator, for the basic and orifice type of pulse tubes. The hot end of the second-stage pulse tube is attached to the cold end of the first stage. A second regenerator is added below the first with some gas passing on to the second stage. (It is not entirely clear how a multiple stage resonant pulse tube would be best configured.) The only experiments done to date on multiple stages have been the work of Longsworth. He reached 79 K with two stages of a valved basic pulse tube[4] and 32 K with 4 stages[12] using $P_h \approx 2.0$ MPa and $P_\ell \approx 0.68$ MPa. It is interesting to speculate on the low temperature limit with about four stages of the orifice pulse tubes. It may be possible to reach 10 K if sufficiently good regenerators can be made.

ANALYSIS

A unified explanation of pulse tubes as well as the Stirling refrigerator based on enthalpy flow is given here to provide a basis for comparing different types of pulse tubes with each other and with other refrigerators. For steady-flow recuperative systems, such as the Joule-Thomson refrigerator, the enthalpy flow analysis is simple and has always been used. The enthalpy flow in regenerative systems, such as the Stirling and pulse tube systems, is more complex because of the oscillating flows.

The enthalpy flow for an ideal gas in any region of an open system is

$$\dot{H} = \dot{m}h = \dot{m}C_pT, \tag{1}$$

where $\dot{m}$ is the mass flow rate through the region, h is the specific enthalpy, C_p is the specific heat at constant pressure, and T is the temperature. For an oscillating flow the time average of the enthalpy flow over one cycle of period τ is

$$\langle\dot{H}\rangle = (C_p/\tau) \int_o^\tau \dot{m}T\ dt. \tag{2}$$

According to the first law of thermodynamics, the average heat flow rate $\langle\dot{Q}\rangle$ into a region is related to the change in enthalpy flow through that region and the average work rate $\langle\dot{W}\rangle$ produced in the region by

$$\langle\dot{Q}\rangle = \Delta\langle\dot{H}\rangle + \langle\dot{W}\rangle. \tag{3}$$

For a pulse tube, equation (2) cannot be used directly since T must be determined from the pressure P by the equation of state for ideal gas,

$$T = P/R\rho. \tag{4}$$

Thus an alternative and often more useful equation for the enthalpy flow becomes

$$\langle\dot{H}\rangle = (C_pA_g/R\tau) \int_o^\tau u\ P\ dt, \tag{5}$$

where A_g is the gas cross-sectional area perpendicular to the flow direction and u is the gas velocity. Since both equation (2) and (5) are cyclic integrals, only the dynamic temperature T_d and pressure P_d need be used in those equations.

Consider a sinusoidal pressure wave,

$$P_d = P_{do} \sin \omega t, \tag{6}$$

in a tube closed at one end, which leads to a flow velocity, mass flow rate and dynamic temperature of

$$u = u_o \sin (\omega t - \phi), \tag{7}$$

$$\dot{m} = \dot{m}_o \sin (\omega t - \theta), \tag{8}$$

$$T_d = T_{do} \sin (\omega t - \alpha), \tag{9}$$

where ϕ, θ, and α are the phase angles by which each of these waves leads the pressure wave. The assumption of sinusoidal behavior implies that (P_d/P) and (T_d/T) are both small compared to unity. The integral in equations (2) and (5) is of the form

$$I = (1/\tau) \int_o^\tau (\sin \omega t) \sin (\omega t - \delta)\ dt, \tag{10}$$

which becomes

$$I = \tfrac{1}{2} \cos \delta. \tag{11}$$

Thus, $\langle\dot{H}\rangle$ is a maximum when u and P_d, or $\dot{m}$ and T_d, are in phase and is zero when the phase angle between them is $\pi/2$.

We now consider three ranges of heat transfer between the gas and the walls. For adiabatic conditions ($\omega\tau_t >> 1$) $\phi = \pi/2$, $\theta = \pi/2$, $\alpha = o$, and from equation (5) $\langle\dot{H}\rangle = 0$. For isothermal conditions ($\omega\tau_t << 1$) $T_d = 0$, and from equation (2) $\langle\dot{H}\rangle = 0$. For intermediate conditions ($\omega\tau_t \approx 1$), ϕ and θ are less than $\pi/2$ and α becomes finite. The phase angle between $\dot{m}$ and T_d is then less than $\pi/2$, so by equation (2) $\langle\dot{H}\rangle$ is finite.

The enthalpy flow as a function of position is shown in Fig. 5 for both the basic and resonant pulse tubes. A positive $\langle\dot{H}\rangle$ means the enthalpy flow is to the right. The enthalpy flow at the left (cold) end of the pulse tube is initially greater then that on the right (hot) end because $\dot{m}$ is greater there; T_d is the same on both ends. The abrupt changes in $\langle\dot{H}\rangle$ at the pulse tube boundaries give rise to heat flow terms according to equation (3), since no work crosses the boundary at those points. If the net heat input at the left end is less than $\dot{Q}_c$, that end of the pulse tube cools until equilibrium is reached. At that time $\langle\dot{H}\rangle$ will become a constant for all x within the $\omega\tau_t \approx 1$ region and heat flows occur only at the boundaries. A first law analysis shows that $\dot{Q}_c = \dot{Q}_H$ at equilibrium, which means that a measurement of $\dot{Q}_H$ gives the gross refrigeration power of the pulse tube.

At higher frequencies, where resonance develops within the closed tube, nodes and antinodes occur in the standing waves of pressure and velocity. The dot-dash sine wave in Fig. 5(b) shows the extreme position

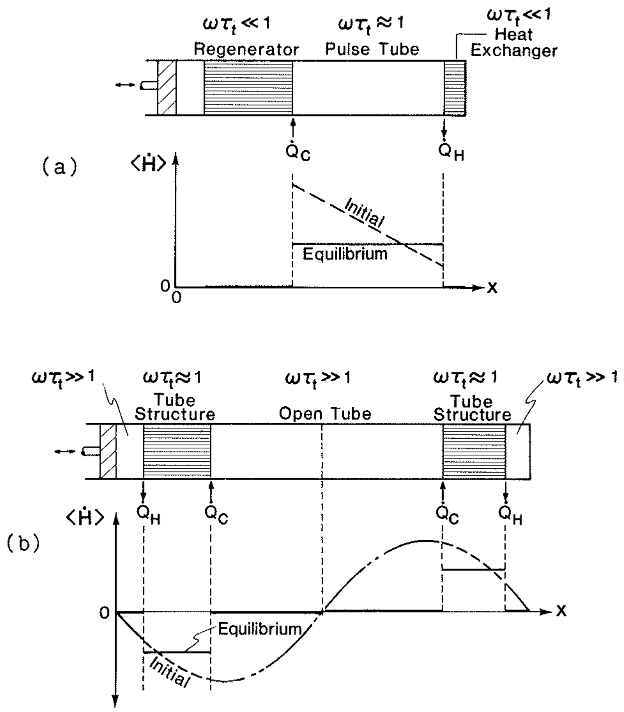

Fig. 5. (a) Enthalpy flow rate at various positions along a basic pulse tube refrigerator. (b) Enthalpy flow rate at various positions in a resonant pulse tube refrigerator.

for the velocity wave at resonance. The pressure wave is shifted by $\pi/2$ in the x direction as well as in time, but only for the case where no heat transfer occurs ($\omega\tau_t >> 1$). Because $\phi = \pi/2$, $\langle\dot{H}\rangle = 0$ by equation (5) in those regions of the tube that are not filled with any structure. In the regions where plates are inserted in the tube the condition $\omega\tau_t \approx 1$ exists and the resulting phase shifts away from $\pi/2$ lead to a finite $\langle\dot{H}\rangle$. The dot-dash line shows how $\langle\dot{H}\rangle$ would behave initially if the entire tube were filled with plates. Cooling occurs where $d\langle\dot{H}\rangle/dx$ is positive and heating occurs where it is negative as found by Merkli and Thomann[7]. By inserting the plates and obtaining $\omega\tau_t \approx 1$, Wheatley produced the abrupt changes in $\langle\dot{H}\rangle$ at the boundaries. At equilibrium the enthalpy flow is constant through the plates. The only heat exchange with the environment occurs at the discontinuites in $\langle\dot{H}\rangle$. Wheatley et al. [8,9] placed the plates close to the piston as shown in Fig. 5(b). The hot end of the plates is toward the piston; hence, there is no need for a separate regenerator.

Both the basic and resonant pulse tubes have relied on heat transfer to the walls to bring about a phase shift between u and P_d away from $\pi/2$. In the pulse tube described by Mikulin et al[11] the orifice at the warm end is the phase shifting mechanism between u and P_d in analogy to an RC electrical system. In our version of the orifice pulse tube, as shown in Fig. 4, phase shifts due to heat transfer with the walls can also occur, but with the frequency of 9 Hz in a 12.7 mm diameter tube it is quite small. It is unclear at present whether the addition of internal structure to maintain $\omega\tau_t \approx 1$ through the pulse tube would improve the performance of the orifice pulse tube.

The enthalpy flow analysis is easily extended to the Stirling refrigerator. Because $\omega\tau_t << 1$ in the regenerator section, $\langle\dot{H}\rangle = 0$. The sections on each end of the regenerator can have any value of $\omega\tau_t$ in practice. If $\omega\tau_t >> 1$ (adiabatic), the movement of the pistons causes a phase shift between u and P_d away from $\pi/2$ to a phase angle of about zero. The enthalpy flow according to equations (5) and (11) is then a maximum and heat flows occur at the discontinuities in $\langle\dot{H}\rangle$ at each end of the regenerator. The term $\langle\dot{H}\rangle$ drops to zero at the boundary with the piston which means the First Law in equation (3) is satisfied by balancing the $\Delta\langle\dot{H}\rangle$ with the work flow $\langle\dot{W}\rangle$. If $\omega\tau_t << 1$ (isothermal) in the sections between the regenerator and the pistons, the enthalpy flow is zero. The energy balance at the piston boundary is satisfied by having $\langle\dot{Q}\rangle = \langle\dot{W}\rangle$ at that boundary. The similarity between the Stirling refrigerator and the pulse tube refrigerators is now evident when the cold piston or displacer is viewed simply as a means of shifting the phase between the velocity and pressure waves, just as intermediate heat transfer or flow through an orifice causes such a phase shift in the pulse tubes. However, the $\pi/2$ phase shift cannot be reached in the pulse tubes except in some asymptotic fashion where $T_d \to 0$ or $\dot{m} \to \infty$.

TEST APPARATUS

Previous measurements of pulse tube behavior have been made in systems where the regenerator loss is mixed with the refrigeration effect. Also no measurements of efficiency have been made. An apparatus as shown in Fig. 6 has been built at the National Bureau of Standards for measurements of the intrinsic efficiency and refrigeration capacity per unit mass flow rate. The technique used to separate the regenerator loss from the intrinsic refrigeration uses dual isothermal heat exchangers. The isothermalizers have excellent heat transfer between the working gas and the body of the isothermalizer and a high heat capacity to eliminate temperature oscillations at the working frequency. The two

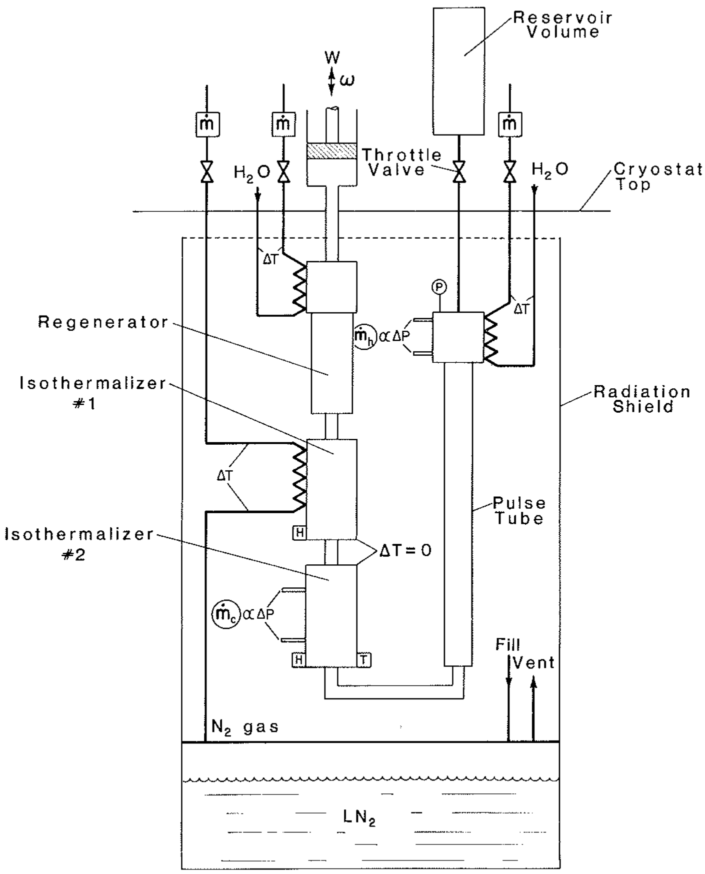

Fig. 6. Schematic of the NBS test apparatus for measuring the intrinsic behavior of various pulse tube refrigerators.

isothermalizers are thermally isolated from each other but are maintained at the same temperature by electronic temperature control that uses a ten junction thermopile to sense the temperature difference. The heat absorbed from the gas by isothermalizer #1 is the regenerator loss term and the heat added to the gas from isothermalizer #2 is the intrinsic refrigeration of the pulse tube minus the conduction loss through the pulse tube. Heat is removed from isothermalizer #1 by the boil-off gas from a liquid nitrogen bath.

The mass flow rate $\dot{m}_c$ at the cold end of the pulse tube is determined from a measurement of the pressure drop across the laminar flow channel in the isothermalizer. Calibration was done during steady flow measurements. Presently mass flow $\dot{m}_h$ through the orifice is determined from an observation of the pressure wave in the reservoir volume assuming adiabatic conditions. Plans include measuring this flow from a ΔP measurement across a laminar flow heat exchanger at the top of the pulse tube.

The thermodynamic efficiency η of the various types of pulse tube refrigerators is used for cycle comparisons. This efficiency considers only the basic refrigeration element and neglects regenerator and compression losses. The efficiency relative to Carnot is expressed as

$$\eta = \dot{Q}\,(T_o - T)/\dot{W}T, \qquad (12)$$

where T_o is ambient temperature. The term $\dot{W}$ is the thermodynamic work input given by

$$\dot{W} = \oint PdV = (1/\tau)\int_o^\tau P\dot{V}dt \qquad (13)$$

where $\dot{V}$ is the volume variation rate of an ideal compressor with no dead volume between it and the cold end of the pulse tube. The value $\dot{V}$ gives the correct $\dot{m}_c$ and P and is derived from simultaneous measurements of the $\dot{m}_c$ and P curves.

DISCUSSION

The excellent performance measured here for the orifice pulse tube indicates a definite advantage over the basic and resonant pulse tubes, although the addition of a large orifice or constriction to the resonant pulse tube could improve its performance. With our 12.7 mm O.D. by 237 mm long orifice pulse tube a temperature of 60 K and a gross heat pumping rate of 18 W was achieved using a pressure ratio of about 1.75.

An ideal Stirling refrigerator has a specific refrigeration capacity of $\dot{Q}$ / $\dot{m}$ T = 1.4 J/(g•K) for a pressure ratio of 2 while a Joule-Thomson expansion of helium from 2 MPa to 0.1 MPa gives $\dot{Q}$ / $\dot{m}$ T = 2.0 J/(g•K). Calculations for $\dot{Q}$ / $\dot{m}$ T in pulse tubes give values around 0.2 J/(g•K). In actual Stirling refrigerators the $\dot{Q}$ / $\dot{m}$ T values may be reduced by a factor of two since isothermal expansion is not achieved. The low $\dot{Q}$ / $\dot{m}$ T for the pulse tubes indicates that its regenerator must be better than those in a Stirling refrigerator in order to handle the greater mass flow. If temperatures below about 20 K are desired, the disadvantage of low $\dot{Q}$ / $\dot{m}$ T is offset by not having frictional heating from moving parts at low temperature. Also conduction loss in the displacer is eliminated in the pulse tube.

The pressure ratio in resonant pulse tubes is only about 1.05 to 1.10 which means an even lower $\dot{Q}$ / $\dot{m}$ T. With such a low pressure ratio, pressure drops in the channels become very significant. The low $\dot{Q}$ / $\dot{m}$ T in all pulse tubes also means that the P versus V diagram for the compressor has a relatively small opening. Consequently, it is important to efficiently recover the expansion work if high overall cycle efficiency is to be achieved. The linear resonant compressors now being used on some split-Stirling cryocoolers appear to offer the most efficient means of supplying the pressure wave to the pulse tubes.

Measurements are yet to be made on the intrinsic pulse tube efficiency, but we would expect efficiencies as high as the approximately 20% of Carnot achieved by the expansion process in Joule Thomson refrigerators.

CONCLUSIONS

The orifice pulse tubes can now reach 60 K in a single stage and they offer a viable alternative to Stirling and Joule-Thomson refrigerators for situations where high reliability is needed. Of the three types of pulse tube refrigerators, the orifice type is the most promising. Advantages over other cycles are:

1. Only one moving part, and that is at room temperature.

2. Uses modest pressure and pressure ratios.

3. Possible to achieve temperature ratios of greater than 4 in one stage.

4. Works on ideal gas, which implies one fluid for all temperatures.

5. Large orifice will not collect impurities at high temperature part of cycle.

6. Good intrinsic efficiency.

7. Several stages can be operated from the same pressure wave generator.

The one disadvantage is the low refrigeration rate per unit mass flow, which means better regenerators are required. Further studies of intrinsic efficiencies, refrigeration capacities, regenerator performance, and compressor designs are needed to better understand and improve the overall system performance.

ACKNOWLEDGMENT

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