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PULSE TUBE REFRIGERATION PROGRESS

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INTRODUCTION

It is not generally realized that delivery of constant temperature gas into a closed volume, thus increasing the pressure, will result in very large temperature gradients in the volume if the turbulence is low. By suitable arrangement of a thermal regenerator, heat exchangers, and turbulence eliminators in this volume, it is possible to preserve these temperature gradients in an essentially static state even though there is a rapid flow of gas in and out of the volume, pressure variation is great, and the pressure oscillation from a maximum to a minimum occurs many times per minute. These temperature gradients are maintained by the pulsating gas. Therefore, if the hot end of the gradient is cooled to room temperature, the cold end will descend to a lower temperature.

Pulse Tube refrigeration [1] is a method which uses this principle to achieve low-temperature refrigeration in simple compact tubes. Refrigeration across temperature intervals as large as 209°F with a single-stage system has been achieved. The method may also be conveniently multistaged with virtually no additional complications so that any low temperature may be achieved. Temperatures as low as 85.5°K have been achieved with staging in essentially preliminary work.

The processes in the Tube are asymmetrical in nature so that heat is delivered from the cold to the hot end. However, they are also of a fairly reversible nature, so that if the pressure increase and decrease are also accomplished reversibly, the refrigeration may be achieved quite efficiently.

REFRIGERATION MECHANISM

The series of operations by which refrigeration is achieved is shown in Fig. 1. Beneath the schematic diagram of the Pulse Tube (shown without the compressor), a plot is given of the gas temperature of an element of gas as the pressure is increasing (solid line) and decreasing (dashed line). During the pressure build-up phase the valve admits high-pressure gas through the regenerator where it is cooled to the cold end temperature T_c. As this gas enters the Tube through the flow smoothing heat exchanger No. 1, it compresses the gas that is already in the Tube, thus acting as a gas piston. This gas piston effect does two things: first, it compresses the gas in the Tube nearly adiabatically so that the temperature of this gas increases, and secondly, it pushes this gas to the far end of the Tube.

If a mass of gas M_c with specific heat C_p is compressed to a temperature T_m just before it enters heat exchanger No. 2 at temperature T_h, then when it enters this heat exchanger it will give up an amount of heat equal to the refrigeration effect Q_r. During the exhaust phase of the cycle, gas leaves heat exchanger No. 2 at temperature T_h and expands to an exhaust temperature T_e which is less than T_c, thus making the refrigeration Q_r available at T_c to cool the load and to recool the regenerator for the next cycle as it flows out:

$$Q_r = M_c C_p (T_c - T_e) \tag{1}$$

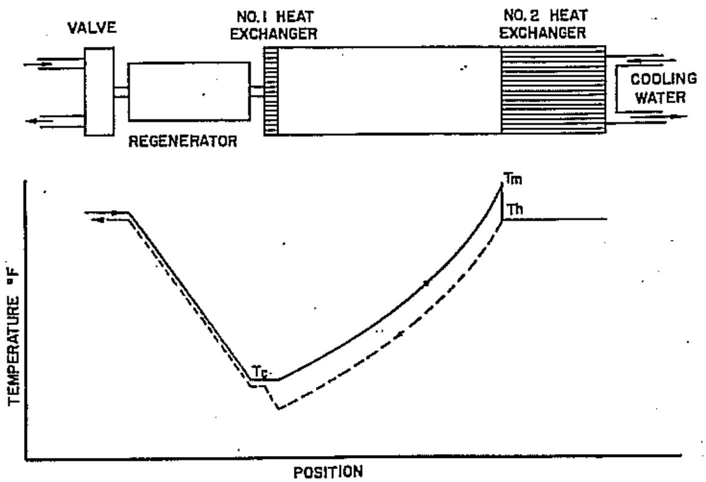

Fig. 1. Pulse Tube with filling and exhausting temperature pattern.

The equation for the temperature pattern of the gas as it flows out of heat exchanger No. 2, which was derived in a previous paper [1], assumed that the gas flowed out as it pushed against a flat piston. The relation for the temperature at any point x in the Tube, T_x, in terms of the volume of heat exchanger No. 2, V_h, and the total volume from the top of the Tube to point x, V_x, for a gas whose ratio of specific heat is γ, is given by

$$\frac{T_h}{T_x} = \left[\frac{(V_x/V_h) + \gamma - 1}{\gamma}\right]^{\gamma-1} \tag{2}$$

Figure 2 is a plot of this temperature pattern for air, $\gamma = 1.4$, and helium, $\gamma = 1.66$. This relation is used to obtain the temperature of the exhaust gas T_e and also shows that the cold end temperature depends on the geometry of the Tube. As can readily be seen, very substantial temperature differences are predicted.

In addition to the mass of gas M_c which accomplishes the refrigeration, there is gas present in the Tube at the beginning of the cycle which follows the dashed temperature pattern (Fig. 1) as it flows back and forth, and an additional mass of gas which fills the Tube but never reaches heat exchanger No. 2. It follows the solid-line temperature pattern as it flows back and forth.

Originally, it was believed that these latter two gas quantities contributed nothing to the refrigeration effect. Practical aspects of the performance of a Pulse Tube refrigerator, however, have shown that these gas quantities, depending on the design and the conditions of operation, can contribute more to the refrigeration effect and efficiency of operation than the gas previously described which makes the complete passage between the two heat exchangers.

The contribution of these gas quantities to the refrigeration effect is due to heat transfer between the gas and the walls of the Tube in the open volume. It is interesting to note that this heat transfer increases the refrigeration. The gas from heat exchanger No. 2 spends nearly all of its time in the open volume of the Tube adjacent to a hotter wall, which heats the gas so that when it returns to heat exchanger No. 2 it carries heat from the walls of the open volume of the Tube. Similarly, gas from heat exchanger No. 1 spends nearly all of its time adjacent to a Tube wall colder than the gas which thus cools it. When it returns to heat exchanger No. 1 it is colder than when it left, and therefore cools No. 1.

By this mechanism alone, it is possible to have a Pulse Tube operate to relatively low temperatures with such a low pressure ratio that no gas travels the complete distance between the two heat exchangers. It has been found possible to obtain a temperature of $-133°F$ in a single-stage Tube 6 in. long, 0.75 in. diameter, with pressures of 300 and 155 psia, and only 42 cycles/min.

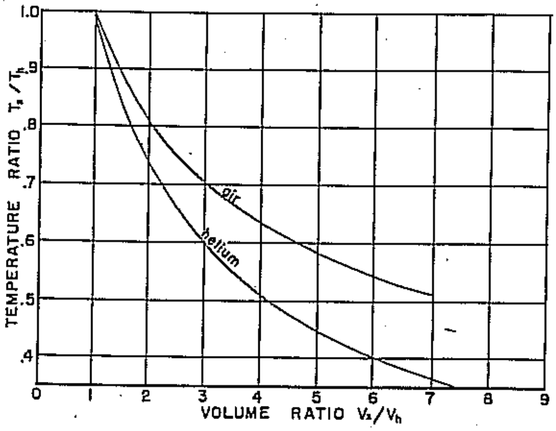

Fig. 2. Ideal temperature ratio for Pulse Tube as a function of its volume ratio.

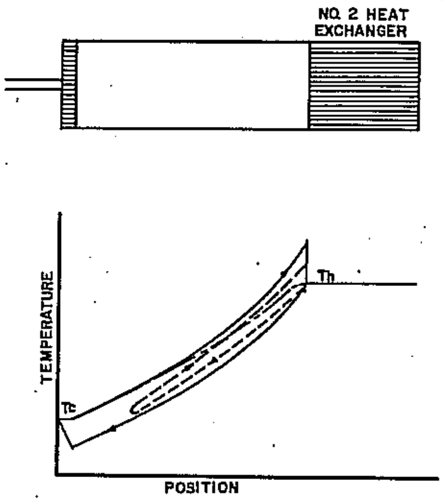

Fig. 3. Temperature pattern of gas exchanging heat with the walls: ———— gas temperature without heat transfer; ------------ temperature of gas from heat exchanger No. 2 with heat transfer; —·—·— wall temperature.

The details of just what occurs in this heat transfer effect may be understood by referring to Fig. 3. This figure shows the temperature patterns in the Tube for the gas from each of the end heat exchangers (assuming there is no heat transfer to the walls), the approximate wall temperature, and the gas temperature path for the gas from heat exchanger No. 2 with the effect of heat transfer. Note that the gas returns to heat exchanger No. 2 hotter than it leaves and thus carries heat into it from a lower temperature. A similar procedure is carried out with the gas from heat exchanger No. 1, but it is not shown here because the temperature differences are smaller and hard to see clearly.

This description of the refrigeration mechanism assumes that the gas flows down the Tube smoothly and maintains a plane front as if it were separated by a series of pistons. It is possible to build heat exchangers that give smooth flow up and down the Tube. However, due to viscous effects along the wall, the flow does not maintain a plane front, and the velocity profile of the gas becomes more nearly parabolic. A photograph of the smoke flow pattern of gas into a Pulse Tube is shown in Fig. 4. Both the smooth flow of the gas and the effect of a nearly parabolic velocity profile are observed.

The viscosity of the gas modifies the description of the ideal refrigeration mechanism in the following manner: First, the outline of the smoke pattern represents gas that entered at the same time and has thus been compressed an equal amount. This is, therefore, a constant temperature surface. Gas in the center of the Tube reaches heat exchanger No. 2

without being compressed as much as if the flow were plane, and a similar effect occurs when the flow expands, so that the temperature difference is less than that predicted by (2). The gas at increasing distances from the Tube axis is compressed more before it reaches heat sink No. 2 and thus is at a higher temperature than the gas in the middle. In Fig. 5, these temperatures are compared with the temperature pattern of (2). The net effect is a temperature difference less than that predicted by (2).

The second effect of viscosity is to modify the pattern of heat exchange with the walls because the gas near the wall does not move back and forth very far. This heat pumping effect in the boundary layer, which gives large temperature differences with low pressure ratios, is currently under investigation.

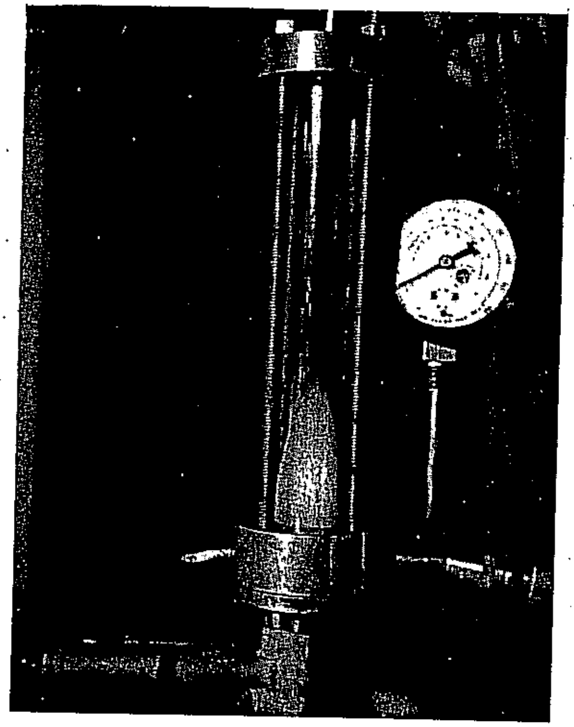

Fig. 4. Smoke flow pattern of gas into a Pulse Tube.

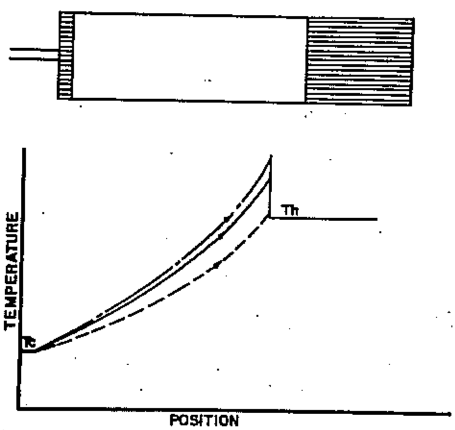

Fig. 5. Distortion of temperature pattern due to effect of viscosity: ———— gas temperature from equation (2) + ΔT; ------------ gas temperature near wall; ———— gas temperature in center of Tube.

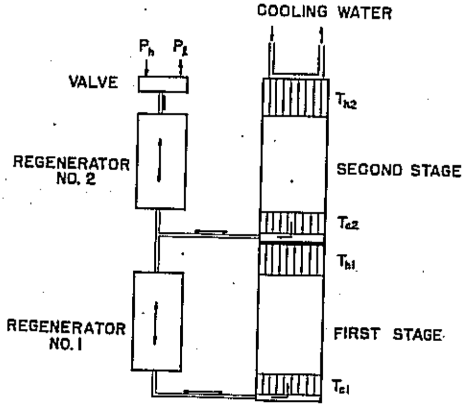

Fig. 6. Two-stage Pulse Tube schematic.

STAGING

The Pulse Tube refrigeration effect may be staged conveniently so that much lower temperatures can be achieved, i.e., by using the entrance end cooling to cool the hot end of another Pulse Tube. Such a refrigerator has been built and operated at a temperature 288°F below the cooling water temperature. A system for a two-stage unit is shown in Fig. 6. In such a system, one adds no moving part, only closed conduits. There is still only one valve at room temperature.

The heat removed from the cold end heat exchanger of the first stage is transferred to the cold end heat exchanger of the second stage and then to the heat sink. Many stages may be incorporated together very simply and conveniently so that virtually any temperature within the limits of regenerator design may be achieved without any low-temperature moving part. A three-stage system has been operated to 356°F below the cooling water temperature.

REVERSIBLE PULSE TUBE

The method using valves for pressure increases and pressure decreases is not very efficient because the free expansion through the inlet and exhaust valves is an irreversible process. Losses would be large for high compression ratios which were originally anticipated, e.g., 10 or 20:1. However, the successful operation of a single-stage unit at −133° from 50°F, with a pressure ratio of 2:1, demonstrates that these losses can be greatly reduced.

Another approach that would eliminate this source of inefficiency is to raise and lower the pressure reversibly by means of a piston. Such a system, which would eliminate the valve, is shown schematically in Fig. 7. Power required to build up the pressure would be taken from a flywheel, and all but the power lost in small irreversible effects in the Pulse Tube and Carnot work would be returned to the flywheel during expansion. Also shown in Fig. 7 is the pressure–volume history of the Pulse Tube refrigerator.

If everything is considered to operate ideally with no friction, it is possible for the piston to do work only if the pressure on its return stroke is smaller than on its forward stroke. Thus, the dotted line may be followed on the P–V diagram on the return stroke rather than retracing the solid compression line. This procedure will be possible if some of the working volume is at a lower temperature on the return stroke than it was on the forward stroke. The only volume which can achieve this in ideal operation is the open volume in the system when it is correctly operating for Pulse Tube refrigeration.

On the forward stroke the temperature is set everywhere by the bottom heat exchanger which gives a higher temperature throughout the volume and delivers some hot gas to the top heat exchanger, where it is cooled. On the return stroke, the temperature is set everywhere by the top heat exchanger and it is colder than it was on the forward stroke, so that some gas is delivered to the bottom heat exchanger colder than the bottom heat exchanger,

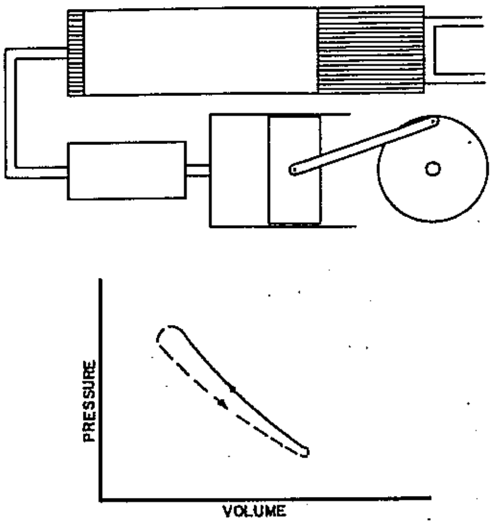

Fig. 7. Reversible Pulse Tube refrigerator.

giving the refrigeration effect. If the temperature pattern were the same in the open volume on both strokes, no heat would be transported from the cold to the hot end and no net work could be put into the piston. Thus, such a system is potentially very efficient.

EFFICIENCY

The efficiency of a Pulse Tube refrigerator is a complex combination of many interrelated effects which encompass thermodynamics, fluid dynamics, and heat transfer. It is not at this time possible to give a complete detailed analysis of how these effects contribute to the inefficiency of the system; however, a list can be made of the important sources of inefficiency along with short discussions of them. The sources of irreversibility (and therefore inefficiency) are the following:

1. Flow smoother heat-exchanger pressure drop.
2. Regenerator pressure drop.
3. Regenerator thermal inefficiency.
4. Heat conduction in gas.
5. Viscous dissipation.
6. Turbulent mixing.
7. Heat transfer across finite temperature differences.

The first three of these are concerned with problems of optimization of heat exchange and pressure drop in highly efficient small-volume heat exchangers and thermal regenerators. Based on present experimental work, these losses will not be large for refrigerators achieving temperatures of 50°K and higher. Lower temperatures may require a better understanding of the regenerator if losses are not to become very high.

The next three losses listed—heat conduction in the gas, viscous dissipation, and turbulent mixing—are all very small and will never be a source of significant efficiency. Surprisingly, turbulent mixing can be small; interpretation of test results and flow pattern studies with smoke indicate this to be true for the Pulse Tubes that have been built.

The last loss, which cannot be avoided, is the most significant. The following is a rough description and estimate of this effect. Refrigeration is achieved in a Pulse Tube by batches of gas which are transported back and forth in the Tube due to compression and expansion from one end. The small batches of gas pick up heat at the lower end of the Tube and deliver it at the top end. This process occurs when the gas makes the complete passage between the heat exchangers, as well as when it is involved with only one of the heat exchangers. This absorbing and delivering of heat involves heat transfer through finite temperature differences, ΔT. The work involved, W, to transfer a quantity of heat, Q, in an otherwise reversible cycle which, however, did involve irreversible ΔT's for heat transfer would be

$$(W/Q)_{\Delta T} = \frac{(T_h + \Delta T) - (T_c - \Delta T)}{T_c - \Delta T} \tag{3}$$

An estimate of this irreversible ΔT effect alone on refrigerator performance can be computed by the ratio of $(W/Q)_i$ for the ideal Carnot cycle to that of (3). This would be

$$\frac{(W/Q)_i}{(W/Q)_{\Delta T}} = \frac{(T_h - T_c)(T_c - \Delta T)}{T_c(T_h - T_c + 2\Delta T)} \tag{4}$$

These ΔT's will vary over a considerable range. Our lack of a complete analysis at this time makes it impossible to compute the exact integrated effect. However, from a little judgment based on experimental results and preliminary analysis, it is possible to estimate the average ΔT to be in the vicinity of 20°R. Thus, with $T_h = 520$°R and $T_c = 340$°R, the efficiency of the Pulse Tube would only be reduced by 23 % by this effect alone. This effect should not result in Pulse Tubes of extremely low efficiencies.

CONSTRUCTION OF MODELS

One of the most important and desirable features of Pulse Tube refrigerators is their simple construction. There are few parts, none of which presents any difficulty in construction. Due to this simplicity, refrigerators can be built in many different configurations, thus making it possible to construct units that lend themselves readily to testing. The first models had the open Tube and regenerator hole drilled in a solid piece of plastic which, in turn, was encased in a thin-walled stainless steel pressure shell. Heat exchangers were lightly press-fitted into the plastic at either end and could be readily changed. This design lent itself well to the testing of various flow smoothers, regenerator packing, etc., at the expense of some additional heat leak.

More recent units have been constructed in single-stage modules with the regenerator and open Tubes consisting of separate thin-walled stainless steel Tubes in a brazed assembly. These sections cannot be modified readily, but they have the advantages of a low conduction heat loss, and are lightweight, providing fast cooldown. Another advantage of this arrangement is that the system can be tested as a single-stage unit, and then two or more can be soft-soldered together to form a multistage unit.

Figure 8 shows the cross section of a typical module drawn approximately to scale. For the $\frac{3}{4}$-in. unit, the four basic components have the following dimensions:

1. Regenerator: $\frac{3}{8}$ in. diameter × $2\frac{1}{4}$ in. long packed with 200 mesh bronze screens.
2. No. 1 heat-exchanger flow smoother: The copper bottom cap has flow distribution channels machined into it, then $\frac{1}{16}$ in. of medium coarse sintering metal is sintered and bonded to the copper.
3. Open Tube: $\frac{3}{4}$-in.-diameter × 7-in.-long × 0.020-in.-thick stainless steel tube.
4. No. 2 heat-exchanger flow smoother: The top cap, like the bottom cap, is copper with $\frac{1}{16}$-in.-sintered metal bonded to it. The gas flows through the sintered metal into a free volume which is 22.5 % of the total volume of the Tube (volume ratio of 4.5:1).

TEST RESULTS

Tests of the new modular units have shown a substantial improvement over the original encapsulated units. Also, some tests have had startling results, which led to the realization of the inadequacy of the original description of the Pulse Tube effect in terms of plane flow down the Tube with a mass transfer of heat into the top heat sink.

The first of these unusual tests was one conducted to find the minimum pressure ratio needed to have the gas make the full passage from the top heat exchanger to the bottom and

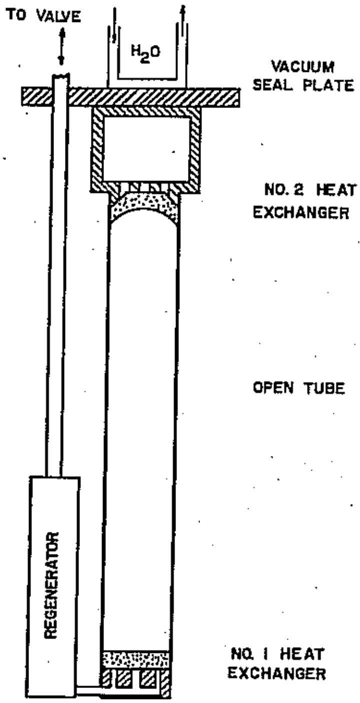

Fig. 8. Single-stage Pulse Tube model.

thus produce refrigeration. It was thought that until this minimum pressure ratio was reached, which was 7:1 for this unit, that the unit would not begin to cool. The test was started at a 4:1 compression ratio with air and an hour later the unit was at −56°F, 98°F colder than the cooling water. Pressures were 0 and 45 psig. This led to the conclusion that there is a heat transfer effect between the walls of the Tube and the gas which gives a heat pumping refrigeration effect.

This conclusion was later confirmed by a test on the ¾-in. modular unit with helium at a pressure ratio of 1.94:1. The gas leaving heat exchanger No. 2 flows only part way down the Tube, expanding from 50°F to a minimum of −67°F, and yet the temperature at heat exchanger No. 1 was −133°F.

The second test that produced interesting results was the measurement of the cooling capacity of the ¾-in. Pulse Tube at various temperatures. From (1) it is seen that for a given M_c the cooling capacity Q increases linearly with the temperature of the bottom end heat exchanger. When cooling capacity is plotted against cold end temperature, the slope of the line should be equal to M_cC_p and the temperature with no load should be given by (2). Figure 9 shows the comparison of test results on the ¾-in. modular unit with results anticipated on the basis of the plane flow in the Tube with no heat transfer to the walls. The test results show that more gas is effective in producing refrigeration than is predicted by the plane flow theory. This can be explained in part by the heat exchange with the walls and by the effect that viscosity has in rounding out the profile of the gas as it flows up the Tube. Not all the gas that was initially in the Tube is pushed into the top heat exchanger; thus, there is room for more cooling gas. This effect also occurs when the gas flows back out and causes the gas in the center of the Tube to arrive at the bottom heat exchanger quite a bit warmer than predicted by plane flow theory, and would account in part for the difference between the expected no load temperature and the measured temperature.

The temperature gradient in the 6½-in. pulsating gas column operating between the two heat exchangers under no heat load ranged from 47° to −159°F, a difference of 206°F, with

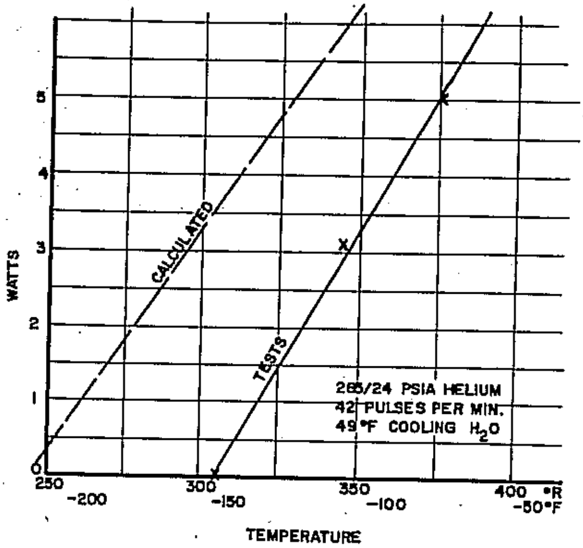

Fig. 9. Capacity test results on $\frac{3}{4}$-in.-diameter Pulse Tube.

Table I. Test Results of $\frac{3}{4}$-in.-Diameter Single-Stage Pulse Tube—Helium (No Load)

Pressure, psia				Temperature, °F		
P_D	P_S	P_D/P_S	rpm	T_h	T_c	$T_h - T_c$
285	28	10.2	42	49	−155	204
215	54	4.0	42	49	−143	192
300	155	1.94	42	50	−133	183
335	217	1.55	42	50	−89	139
345	45	7.7	30	47	−139	186
355	48	7.4	76	47	−159	206
325	45	7.2	104	47	−155	202

Table II. Test Results of Two-Stage Pulse Tube

	Helium	Air
Heat sink temperature	43°F	45°F
Temperature difference	127°F	114°F
Absolute temperature ratio	0.749	0.775
Interstage temperature	−84°F	−69°F
Temperature difference	154°F	115°F
Absolute temperature ratio	0.589	0.706
Heat source temperature	−238°F	−184°F
Total temperature difference	281°F	229°F
Total absolute temperature ratio	0.441	0.548

helium at a pressure ratio of 7.4:1. When run on air at a pressure ratio of 18:1, it cooled to −118°F, a difference of 160°F. Table I gives a summary of results at other conditions.

A two-stage unit was constructed by combining a 1-in.-diameter module for the upper stage with the ¾-in.-diameter module. The 1-in.-diameter module had been tested separately with temperatures almost identical to those of the ¾-in. unit. This combination cooled to −236°F (123°K) with helium and to −186°F with air.

A three-stage unit was made by adding a ⅜-in.-diameter module to the above two-stage unit. With it, a temperature of 85.5°K was achieved with helium and 119°K with air. Cryogenic temperatures are clearly obtainable by this method.

CONCLUSIONS

Pulse Tube refrigeration is much more than an interesting curiosity. It is a refrigeration method which makes it possible to get virtually any temperature with no low-temperature moving parts and almost any gas pressures in small, simple devices with small gas flows. It will undoubtedly find applications.

It is difficult at this time to predict what overall efficiencies will be obtainable. However, experimental work now indicates that some sources of inefficiency which had originally been considered quite serious are actually negligible. Also, the contributing effect of heat transfer and possibility of operation at low pressure ratios make reasonably good efficiencies a probability.

NOTATION

C_p = specific heat at constant pressure
M = mass of gas
Q = quantity of heat
T = temperature
V = volume
W = work
γ = ratio of specific heats
ΔT = temperature difference

Subscripts
c = cold
h = hot
i = ideal
m = maximum
r = refrigeration
x = arbitrary position

ACKNOWLEDGMENT

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REFERENCE

1. W. E. Gifford and R. C. Longsworth, "Pulse-Tube Refrigeration," ASME paper No. 63-WA-290 presented at Winter Annual Meeting of the American Society of Mechanical Engineers, Philadelphia, Pennsylvania (November 17–22, 1963).

DISCUSSION

Question by H. J. Smith, Ferranti-Electronics: Is the heat evolved at the warm end equal to the refrigeration produced?

Answer by Author: Yes.

Question by J. P. Roos, RCA Service Corp.: Is there a limitation on the frequency with which the gas can be pulsed efficiently?

Answer by Author: Yes. The refrigeration involves heat transfer in the gas after a temperature gradient has been set up by the pressure variation. Time must be allowed for this heat transfer. Pulse rate should be relatively slow for good efficiency.

Question by C. A. Stochl, U.S. Army Electronics Laboratories: The statement was made that "any temperature can be achieved with this device." Wouldn't the liquefaction aspects tend to limit this achievable temperature?

Answer by Author: Yes. Any temperature above the liquefaction point could be achieved. Thus with helium gas and suitable low-temperature regenerators, a temperature of 6°K could be achieved.

Question by M. P. Hnilicka, National Research Corporation: What is the nature of gas boundary motion (piston or laminar)?

Answer by Author: The flow in the boundary layer is essentially laminar.