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Semi-annual Technical Report No. 2
July 1963

PULSE TUBE REFRIGERATOR

by

William E. Gifford

Project No. 1620.1002

Sponsored by
Advanced Research Projects Agency
ARPA ORDER 268

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1. INTRODUCTION

This period has produced the very significant results of having a single stage pulse tube cool to 140°F below room temperature and a two-stage unit cool to 206°F below room temperature. The basic concepts of the pulse tube operation have been shown to be valid and studies of operational effects are continually improving its performance. The pulse tube is still in an early stage of development so it is expected that future designs will show a considerable additional improvement. This report contains three of these studies, a description of the pulse tubes that have been built and tested, and test results.

2. ANALYSIS

Of the three studies presented here only the first one has been applied to modifying the original pulse tube design. It produced a significant improvement and it is felt that the other two studies will also show significant advances.

A. Description of flow pattern in pulse tube

B. Flow timing

C. The heat exchange problem

A. Flow Pattern Distortion

In our initial analysis of the pulse tube we assumed that the velocity is uniform throughout the tube, this being the case that will give the best results. In the design of the experimental models we have tried in every way possible to achieve this.

This represents a very significant problem as there is a very large velocity head in the connecting tube which must be distributed evenly over the pulse tube end without a jet effect in one region. The problem is made more serious by the fact that this flow distribution must be achieved without the addition of any appreciable volume to the system. Experimentation was carried out with many possible flow distributors for the entrance end of the tube to assure constant velocity of the gas when it enters the open portion of the tube. A great many flow smoothers were contrived out of various sizes and meshes of wire screens, perforated plates, sintered metal disks, and many drilled holes in a disk. Many seemingly promising arrangements didn't work at all. Nearly the full force of a jet came through a surprising thickness of very fine screens. Various cored metal pieces gave very poor results. Very poor distribution can almost wipe out any pulse tube refrigeration effect.

A simple test rig was devised and built to study the performance of these various arrangements. It gave a chance to trace visually the flow pattern over any section of a tube with a flow distributor over one end admitting gas. Flow measurements were taken with a pitot tube.

Good results were finally achieved with a coarse grain sintered metal disk 1/4" thick. Finer grain sintered metal gave a good velocity pattern also, however, the disks had too high a pressure drop in the thicknesses which were tested. Thinner sections of the fine grain material might be better as the volume would be much smaller.

By means of the sintered disk we achieve quite even flow into one end of the pulse tube. The problem then is to keep this flow even until it enters the heat exchanger at the other end. There is, of course, a

definite force for destroying this even flow, namely, the viscous forces at the tube wall which create a retarding force that slows down the gas there. The flow pattern will tend to change from pattern (a) to pattern (b), as shown in Illustration 1, as it proceeds up the tube.

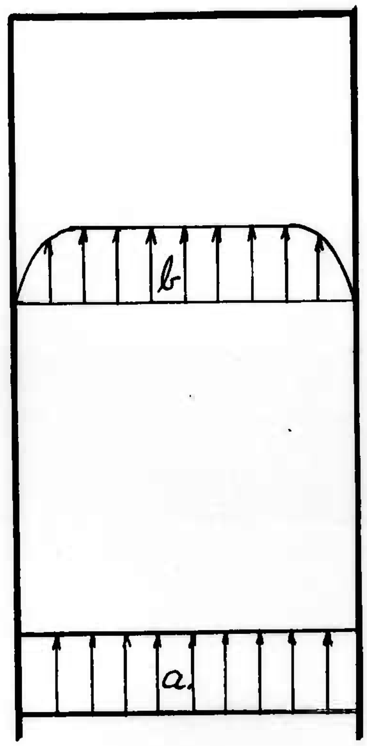


Illustration 1 Gas Velocity Profile in Pulse Tube

A natural first suggestion might be to install additional flow straighteners in the tube. However, this cannot be done as the flow straightener will necessarily have mass and a large area of fine

passages and will thus serve as a regenerator and be detrimental to our desired refrigerating effect.

Since the tube must be left open to achieve a good pulse tube refrigeration effect, the even velocity pattern deterioration must be tolerated and it becomes worthwhile to try to compute the degree of velocity effect to be expected and the effect it will have on the refrigeration.

Our particular problem for gas flow into a closed tube has not been solved in the literature. This is partly due to the fact that no one has ever been interested in this problem and partly because it is extremely difficult. The nearest solution in the literature is for laminar flow into an infinite tube at constant velocity, pressure, and density. This has been solved by Langhaar[1]. He has shown that the degree to which one proceeds from (a) to (b) of Illustration 1 is a function of

$$\frac{100\,\mu x}{\rho r_o^2 V_o}$$

where

x = Distance into tube

μ = Viscosity of the flowing fluid

V_o = Initial velocity

ρ = Gas density

For values of this parameter less than .1 the velocity is very slightly distorted. At a value of 5, however, it is almost completely converted to a laminar tube pattern. The value in the typical range

[1] Langhaar, H. L., Journal of Applied Mechanics, 9:2, A55-A58 (1942).

where one might expect to operate a pulse tube is about .1 to .3. Therefore, the effect should be there although not in its most developed state. It may be less than this predicts as the closed end puts added restraint on gas which should check the velocity to a uniform flow to a greater degree than that of a long open tube.

The next question then is if the velocity pattern is changed to that of (b) in Illustration 1 what will be the effect on the refrigeration? Also what, if anything, can be done to counteract any performance deterioration?

The effect of velocity variation will be that gas in the center of the tube will arrive at the heat exchanger sooner than that near the edges. Since the temperature of the gas will only depend on the time since entering, the temperature pattern of the gas in the tube will vary across the tube. Constant temperature curves will have a pattern much like the velocity curves. Therefore, the temperature of the gas entering the top heat exchanger will be different in the middle from that at the edges.

In Illustration 2 an attempt is made to show this effect by showing three patterns in a pulse tube; (a) for an ideal flow with no distortion (b) for the center section of increased velocity and (c) for the edge sections with decreased velocity. The refrigeration effect depends on cooling the gas in the top heat exchanger by the amount $T_3 - T_2$. For the center gas it will arrive sooner in the heat exchanger and therefore will not be quite so hot, thus reducing the net refriger-

ation. If the velocity distortion were sufficiently bad the gas might even arrive cooler than the heat exchanger and result in negative refrigeration.

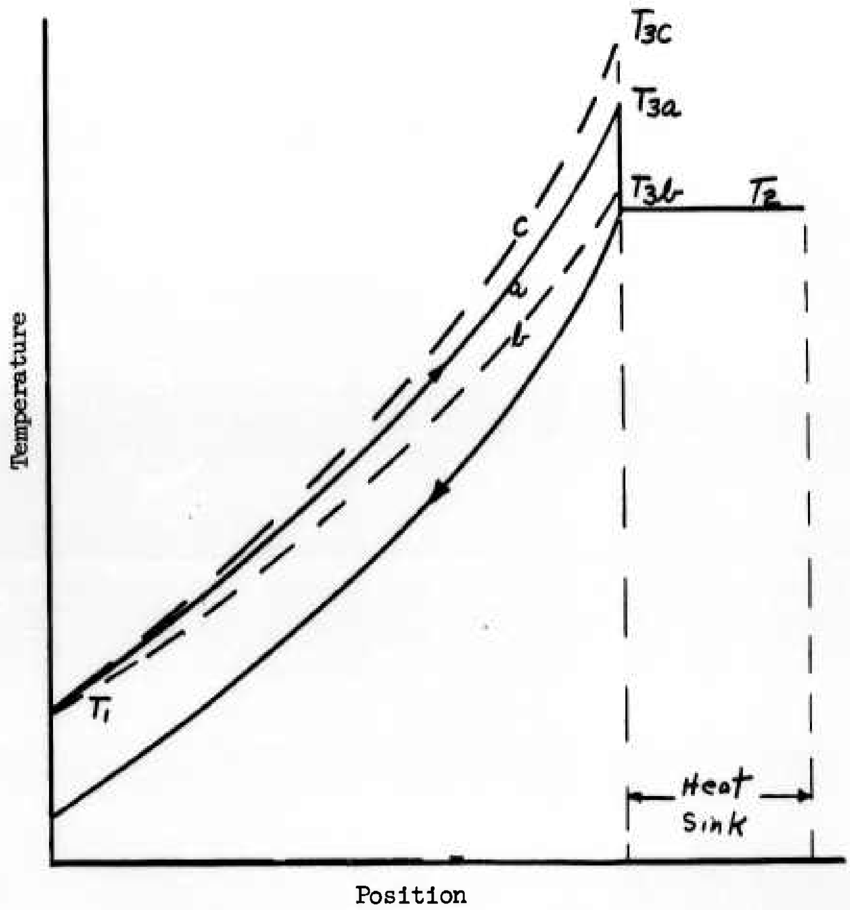


Illustration 2. Deviations in Gas Temperature Profile

The gas in the edge sections going slower gets hotter than for the ideal case. However, not as much gas arrives and therefore the refrigeration is also reduced here. If the velocity here is sufficiently slow it may not arrive in the heat exchanger at all and therefore no refrigeration will be achieved.

The total refrigerating effect will be the integrated effect of gas moving very slowly at the surface with a lot of heat, to that moving fastest in the middle with little heat. It would be an advantage to be able to know the velocity as a function of the radius and therefore make this calculation. However, the problem is a formidable one and has not been solved. A study of it has indicated that a rigorous solution would be a lengthy, difficult, and expensive program which is not warranted at this time when we are just demonstrating the principles and feasibility of the method.

A realization that this effect occurs, however, is very important. An attempt can be made to reduce the bad effects of this varying velocity by shaping the bottom of the top heat exchanger to fit the distorted temperature of the gas stream. This can be done by making the heat exchanger longer on the edges and shorter in the center as shown in Illustration 3. In this way the center gas goes farther and therefore heats more before it enters the heat exchanger and more of the hot edge gas is able to enter the heat exchanger.

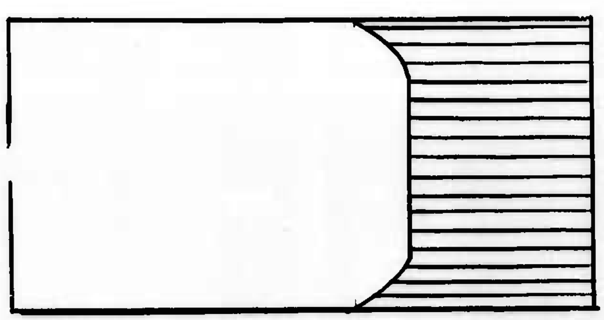


Illustration 3 Heat Sink Shaped to Match Gas Temperature Profile

A rough heat exchanger conforming to these patterns was made and gave very rewarding results. A markedly lower temperature was achieved. A heat exchanger of this type has been made standard for experimental models. It would be good to achieve an ideal profile; however, the ideal profile will be different for different speeds and temperatures. It is believed, therefore, that an experimental program testing different profiles will be less expensive and more rewarding than an extensive analytical one at this time.

B. Compression and Decompression Time

The pressure drop through the regenerator and heat station limits the speed at which the pulse tube may be operated, and since cooling capacity is proportional to speed--all other things being constant--it is necessary to understand the pressure drop problem.

Preliminary tests of the regenerators show that for acceptably low pressure drops the flow is laminar, so from the theory of laminar flow the following relations are introduced. For laminar flow the pressure drop, dp, per unit length, dx, of the regenerator is given by Poiseuille's law.

$$\frac{dp}{dx} = \frac{C_1 \mu v}{d^2} \tag{1}$$

where

C_1 = constant

μ = viscosity

v = velocity

d = effective diameter of flow passages

The velocity is a function of the mass flow rate, $\dot{m}$, the pressure, p, and the temperature, T,

$$v = \frac{\dot{m}\,R\,T}{A\,P} \tag{2}$$

R = gas constant

A = cross sectional area of regenerator

Substituting (2) into (1) and integrating gives

$$\frac{dp}{dx} = \frac{C_1}{d^2}\,\mu\,\frac{RT}{A}\,\left(\frac{\dot{m}}{p}\right) = C_2\,\left(\frac{\dot{m}}{p}\right) \tag{3}$$

$$\int_{P_1}^{P_2} pdp = \int_{0}^{L} C_2\,\dot{m}dx \tag{4}$$

$$P_2^2 - P_1^2 = 2\,C_2\,\dot{m}\,L \tag{5}$$

where L is the length of the regenerator. This yields an expression for the mass flow rate at a given instant,

$$\dot{m} = \frac{dm}{dt} = \frac{P_2^2 - P_1^2}{2\,C_2\,L} \tag{6}$$

For our case of pressurizing and depressurizing the volume of the pulse tube it is clear from this equation that the filling and exhausting rates and times will vary greatly with the two pressures as they change, and that filling will be a much faster procedure due to higher values of P^2 involved in the latter end of a filling procedure.

The total flow is, of course, the integral of the mass flow rate and is equal for both compression and expansion. If the pressures are known then the time of a complete cycle can be determined.

When the pulse tube is operated with a compressor, P_2 represents the discharge pressure and P_1 the pressure in the pulse tube during the compression cycle, while P_1 represents the pressure in the pulse tube and P_2 the compressor suction pressure during expansion. The notation to be used in determining the compression time, t_c, and the expansion time, t_E, for this case is shown in Illustration 4, a sketch of the cycle.

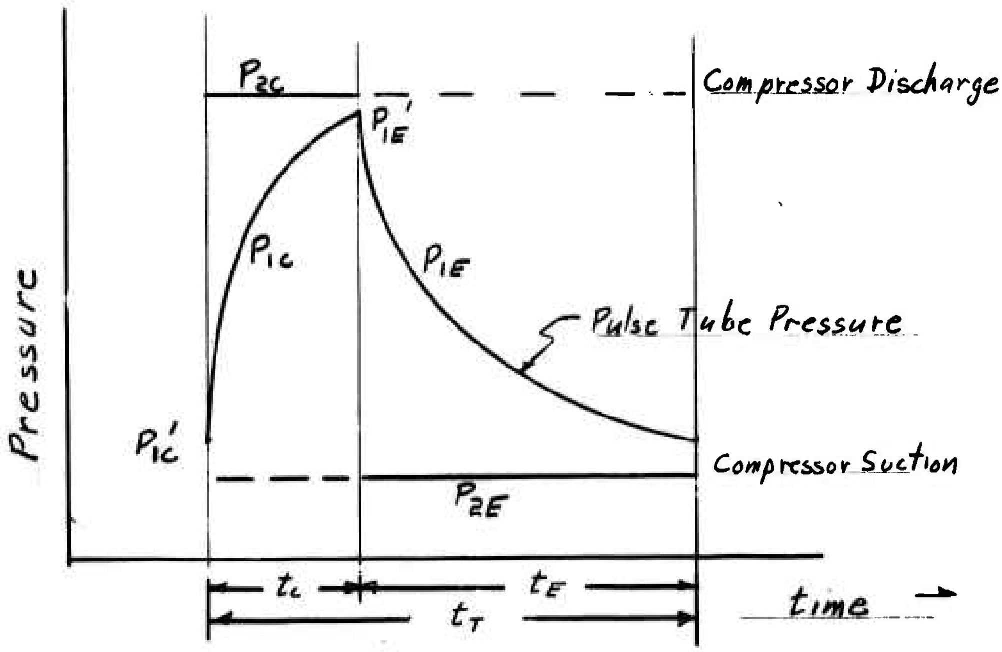


Illustration 4 Pressure Build Up and Decay in Pulse Tube

Since the gas flows into the pulse tube which has a volume, V, the change in pressure caused by the addition of gas, dm, is

$$dp = dm\ RT/V \tag{7}$$

When dm from (6) is substituted into (7) and the constant terms are collected, the resulting equation can be integrated to give the pressure build up and decay with time in the pulse tube,

$$\frac{dp_1}{dt} = C_3 \left(P_2^2 - p_1^2\right) \tag{8}$$

$$\int_{p_1'}^{p_1} \frac{dp}{p_2^2 - p_1^2} = \int_{t=o}^{t} C_3 \, dt \tag{9}$$

where the prime (') denotes the initial pressure.

For compression

$$C_3 t_c = \frac{1}{2P_{2C}} \tanh^{-1} \frac{P_{1c}}{P_{2c}} - \tanh^{-1} \frac{P'_{1c}}{P_{2c}} \tag{10}$$

For expansion

$$C_3 t_E = \frac{1}{2P_{2E}} \tanh^{-1} \frac{P_{2E}}{P_{1E}} - \tanh^{-1} \frac{P_{2E}}{P_{1E'}} \tag{11}$$

Curves for typical operating pressures of 20 atm. discharge and 1 atm suction pressures are plotted in Fig. 1. It is apparent from this curve that compression and expansion time curves are very different. The pulse tube works best for high pressure ratios. It therefore is of interest to determine for a given RPM the compression and expansion times for optimum pressure ratio. It will be obtained when the compression and expansion are terminated at the points where the slopes are proportional to the pressures, that is

$$\frac{P'_{1E}}{P'_{1c}} = \frac{d\ P_{1c}/dt}{d\ P_{1E}/dt} \tag{12}$$

From Eq. (8) this is expressed as

$$\frac{P'_{1E}}{P'_{1C}} = \frac{P_{2C}^2 - P_{1c}^2}{P_{1E}^2 - P_{2E}^2} \tag{13}$$

For different speeds the optimum pressure ratio and the ratio of compression to expansion time changes. These relationships are plotted in Fig. 2 and 3. From Fig. 3 it can be seen that expansion should be 5 to 10 times longer than compression. This, of course, assumes that the variation in filling and exhausting speeds will not affect regeneration performance or flow pattern in the tube. This is not true; however, for a first approximation this assumption is reasonable. Corrections will have to be made later.

Figure 4 in Report No. 1 shows that the percentage of gas that is effective in producing refrigeration decreases as the pressure ratio increases, and yet the total amount of cooling in a period of time increases with speed. The product of the percentage of cooling gas and the speed is, therefore, proportional to the total cooling. This relationship as a function of the speed is plotted in Fig. 4. If heat loss did not have to be considered, this curve would give the speed for maximum cooling.

As the speed is increased the total cooling increases, as one would expect, but the effective pressure ratio decreases. For this example the effective pressure ratio at maximum cooling is only about one-third the available pressure ratio but, of course, this would be an inefficient operating condition.

When regenerator losses, as described in the next section, are taken into account the speed for optimum cooling or efficiency will be much less than that indicated above.

C. The Heat Exchange Problem

In any low temperature refrigerator there is always a problem as to how efficient to make the heat exchange means. In this case our heat exchange means is, of course, a small thermal regenerator which can be made very efficient. However, the more efficient it is made the higher the pressure drop through it will be unless the total volume is greatly increased (another detrimental effect). Therefore, it is advisable to try to achieve about an optimum regenerator efficiency, not too high or low.

It is an advantage in general to refer to a regenerator inefficiency, I_e, for various reasons which will become obvious. It is the ratio of the heat that is not transferred in a regenerator or heat exchanger to that which could be transferred.

$$I_e = \frac{\dot{m}\, c_p\, \Delta T_e}{\dot{m}\, c_p\, (T_a - T_c)} \tag{14}$$

where

$\dot{m}$ = Mass flow rate

c_p = Heat capacity

ΔT_e = Temperature difference between entering and exhausting gas

T_a = Hot end of heat exchanger or regenerator

T_c = Cold end of the heat exchanger or regenerator

This of course reduces to

$$I_e = \frac{\Delta T_e}{T_a - T_c} \tag{15}$$

The parameters which determine this may be derived from a heat balance, equating the heat transferred in the regenerator to heat added to the flowing gas.

$$h\ A\ \frac{\Delta T_e}{2} = \dot{m}\ c_p\ (T_a - T_c)$$

$$I_e = \frac{\Delta T_e}{T_a - T_c} = \frac{2\dot{m}\ c_p}{hA} \tag{16}$$

From this it may quickly be seen that the I_e is inversely proportional to the product of the heat transfer coefficient and area.

One may also define for a refrigerator a heat exchange means efficiency, $E_{\Delta T}$, as the ratio of the total refrigeration production minus heat exchange means loss to the total refrigeration. The total refrigeration production, Q_r, is

$$Q_r = \dot{m}\ c_p\ \Delta T_r \tag{17}$$

where

ΔT_r = Average effective temperature drop due to refrigeration effect.

Therefore

$$E_{\Delta t} = \frac{\Delta T_r - \Delta T_e}{\Delta T_r} \tag{18}$$

In the pulse tube

$$\Delta T_r = .05 \text{ to } .10\ T_2$$

(assume .05)

also

$$\Delta T_e = I_e\ (T_a - T_c) \tag{19}$$

$$E_{\Delta t} = \frac{.05\ T_c - I_e\ (T_a - T_c)}{.05\ T_c}$$

and

$$T_2 = .6 \quad \text{to} \quad .8\ T_a$$

(assume .8)

$$E_{\Delta t} = \frac{.05 \times .8\ T_a - I_e \cdot 2\ T_a}{.04\ T_a}$$

$$E_{\Delta t} = 1 - 5\ I_e$$

Thus, for an inefficiency of 1 per cent the $E_{\Delta t}$ will be 95. For an I_e of 20 per cent the $E_{\Delta t}$ will be zero or all refrigeration achieved will be lost. The following table gives some values:

Table I

I_e	$E_{\Delta t}$
.01	.95
.02	.90
.03	.85
.04	.80
.05	.75
.06	.70

It would be desirable to have $E_{\Delta t}$ as large as possible. However, as can be seen from the table, even an I_e greater than about .06 would probably be intolerable and .01 would be quite good. We should, therefore, try to strive for about .01 or better if it can be conveniently made for a single stage unit.

When a second stage is added which cools the hot end of the first stage, the problem is exactly the same for the first low temperature

stage. Therefore, the regenerator inefficiency, I_e, in the first stage to be planned for would be roughly the same as for the single stage unit. The second stage regenerator, however, must handle the flow from both the tubes but will have only the cooling effect from the second stage tube. If tubes of equivalent size are assumed, the gas flow will be

$$m_2 = m_1 \left\{1 + \frac{T_{c2}}{T_{c1}}\right\} \tag{20}$$

where

m_1 = Mass of gas through list stage regenerator

m_2 = Mass of gas through 2nd stage regenerator

T_{c1} = Cold end temperature of 1st stage

T_{c2} = Cold end temperature of 2nd stage

Since the refrigerating effect which had previously been considered for a flow of m_1 must be averaged over m_2, the refrigeration effect of the second stage Δtr_2 now will be

$$\Delta tr_2 = \frac{\Delta_{tr}\ m_1}{m_1 \left\{1 + \frac{T_{c2}}{T_{c1}}\right\}} \tag{21}$$

$$\Delta tr_2 = \frac{\Delta tr + T_{c1}}{T_{c1} + T_{c2}} \tag{22}$$

Now assuming as before $\Delta_{tr} = .05$ and $T_{c1}/T_{c2} = .8$

$$\Delta tr_2 = \frac{.05\ T_a \times .8\ T_{c2}}{.8\ T_{c2} + T_{c2}}$$

$$= .05\ T_a \times .444$$

$$= .0222\ T_a$$

Table II then can be prepared again for the second stage regenerator of a two-stage system, giving values of $E_{\Delta t}$ and I_e.

Table II

I_e	$E_{\Delta t}$
.005	.934
.01	.888
.02	.776
.03	.664
.04	.552

It is interesting to note that here we need a smaller regenerator inefficiency to achieve equivalent $E_{\Delta t}$. The inefficiency .01 which gave an $E_{\Delta t}$ of .95 in the first stage gives only .888 in the second stage. It may be seen from this that the efficiencies of the higher temperature stages are going to need to be better than the lower temperature stages.

3. TEST UNITS

The construction of the first model pulse tube was described in the first report. Testing of this unit proved the design to be a bit less than ideal, but it had been constructed so that it could be modified. It was made so that the volume ratio could be adjusted; this feature introduced leakage problems and since the effect of varying the volume proved to follow the predicted results, the succeeding units were made with fixed volume ratios.

The most significant improvement in performance was achieved by smoothing the flow as it entered the pulse tube, the solution found most satisfactory being coarse sintered metal.

With the experience gained from the first pulse tube a simpler unit was built that proved to be even more easily modified. Since vacuum leaks had been found to be a major problem with the first unit the second one was built with a minimum of joints--only two--by adapting an encapsulating concept. This is sketched in Fig. 5, and a photograph of the unit appears in Fig. 6.

This unit worked as well as the first unit initially, but performance was later substantially improved by reshaping the heat sink. The study of the flow pattern in the tube indicated that the screen should be dished in and this was confirmed. Several different pieces of sintered metal were interchanged to determine their effect. The unit also lends itself well to changing the length of the regenerator for future studies in this area.

The success with the single stage encapsulated unit showed that this would be the best way to make a multi-stage test unit. A two-stage unit was built consisting of two single stage units end-to-end in a single shell. A sketch of this unit is shown in Fig. 7 and a photograph in Fig. 8. This unit, like its predecessor, can be easily modified.

A few comments about the construction of the pulse tube might be made. In general, it is fairly easy and straightforward to construct and assemble. Care should be taken in making the seals and brazing the joints so that there are no leaks at the heat sink or at the cold end

because they represent a loss in cooling effect. Also, in assembling the unit it is necessary to be sure that there are good thermal bonds at the points where heat is being transferred.

4. EXPERIMENTAL RESULTS

The experimental results have been quite good this period. The following important principles have been confirmed by the experimental results.

1.) A very substantial temperature difference may be maintained in a pulse tube. The best result to date is 140°F. A low temperature of -82°F was achieved with 58°F cooling water.

2.) Pulse tubes of this type may definitely be staged. A two-stage unit achieved -145° on the first stage and -39° on the second stage with 61°F cooling water.

The results that have been achieved by no means represent the optimum that could be achieved.

The -82°F represents a ratio of cold to hot temperature of .73. This was obtained for a tube in which V_T/V_1 was 3.7. The optimum value of T_1/T_2 for such a ratio would be .53. However, when you consider the necessary heat leaks by radiation and conduction, one could not expect to do better than about .64. When this is taken into consideration, our .73 seems quite good.

We are still at a stage where we see many things to do which should give improvements in performance. Some of these involve considerable additional study to determine just what to do. However, others involve purely the matter of making a more finished, smoother mechanical model.

The two-stage system gave values of T_1/T_2 of .748 between the first and second stages and .808 between the second stage and the cooling water. These are not really very good and will definitely be improved upon. The results, however, are good enough to confirm without any doubt that staging will work as predicted, and that very low temperatures are going to be possible with additional stages. The first stage here was only spanning a temperature interval of 100°F. However, it did make a base from which the second stage could achieve a further 106°F temperature drop.

The model with which staging was demonstrated was not in ideal shape. It is clear how it may be improved in many ways. The test was rushed through as it was felt it was important to establish the staging principle at an early point even if it had to be with a crude model. It should be possible to greatly improve these results.

The following tables summarize the best test results achieved to date.

Table III

Test Results of First Signle Stage Unit

	Air, $P_2/P_1 = 19$				Helium, $P_2/P_1 = 11$			
	3.8		3.1		2.4		3.8	
Sink Temp. T_a	51°F	511°R	52°F	512°R	49°F	509°R	47°F	507°F
ΔT, T_{c1}/T_a	95°F	.810	78°F	.848	53°F	.895	132°F	.741
1st Stage Temp, T_{c1}	-44°F	416°R	-26°F	434°R	-4°F	456°R	-85°F	507°R

Table IV

Test Results of Single Stage Encapsulated Unit V_T/V_1, 3.7

	Air, $P_2/P_1 = 19$		Helium, $P_2/P_1 = 11$	
Sink Temp. T_a	69°F	529°R	58°F	518°R
ΔT, T_{c1}/T_a	125°F	.764	140°F	.732
1st Stage Temp. T_{c1}	-56°F	404°R	-82°F	378°R

Table V

Test Results of Two Stage Encapsulated Unit $V_T/V_1 = 3.7$

	Air, $P_2/P_1 = 19$		Helium, $P_2/P_1 = 11$	
Sink Temp. T_a	61°F	521°R	61°F	521°R
ΔT T_{c2}/T_a	67°F	.873	100°F	.808
2nd State Temp. T_{c2}	-6°F	454°R	-39°F	421°R
ΔT, T_{c1}/T_{c2}	85°F	.813	106°F	.748
1st Stage Temp. T_{c1}	-91°F	369°R	-145°F	315°R
ΔT total, T_{c1}/T_a	152°F	.709	206°F	.605

5. CONCLUSIONS

1.) Pressurizing and depressurizing a constant volume system due to unsymmetrical transfer of heat and refrigeration may lead to the build-up of large temperature differences within the volume.

2.) A large smooth temperature gradient may be maintained in an open tube while it is filled and exhausted many times a minute.

3.) The unsymmetrical transfer of heat in pressurization and depressurization of a constant volume may be used in combination with heat exchangers and a regenerator to build a refrigerator which can achieve a temperature as low as 140°F below a toom temperature heat sink.

4.) Virtually any low temperature may be achieved by staging the refrigeration method of (3). This requires only the addition of more regenerators, heat exchangers, and open volumes.

5.) The refrigeration method of (3) and (4) makes most any low temperature available without any low temperature moving parts and without a non-ideal gas where the Joule-Thomson process is possible.

6.) Pressurization and depressurization of a constant volume system will lead to transfer of heat within the volume and possibly without the volume.

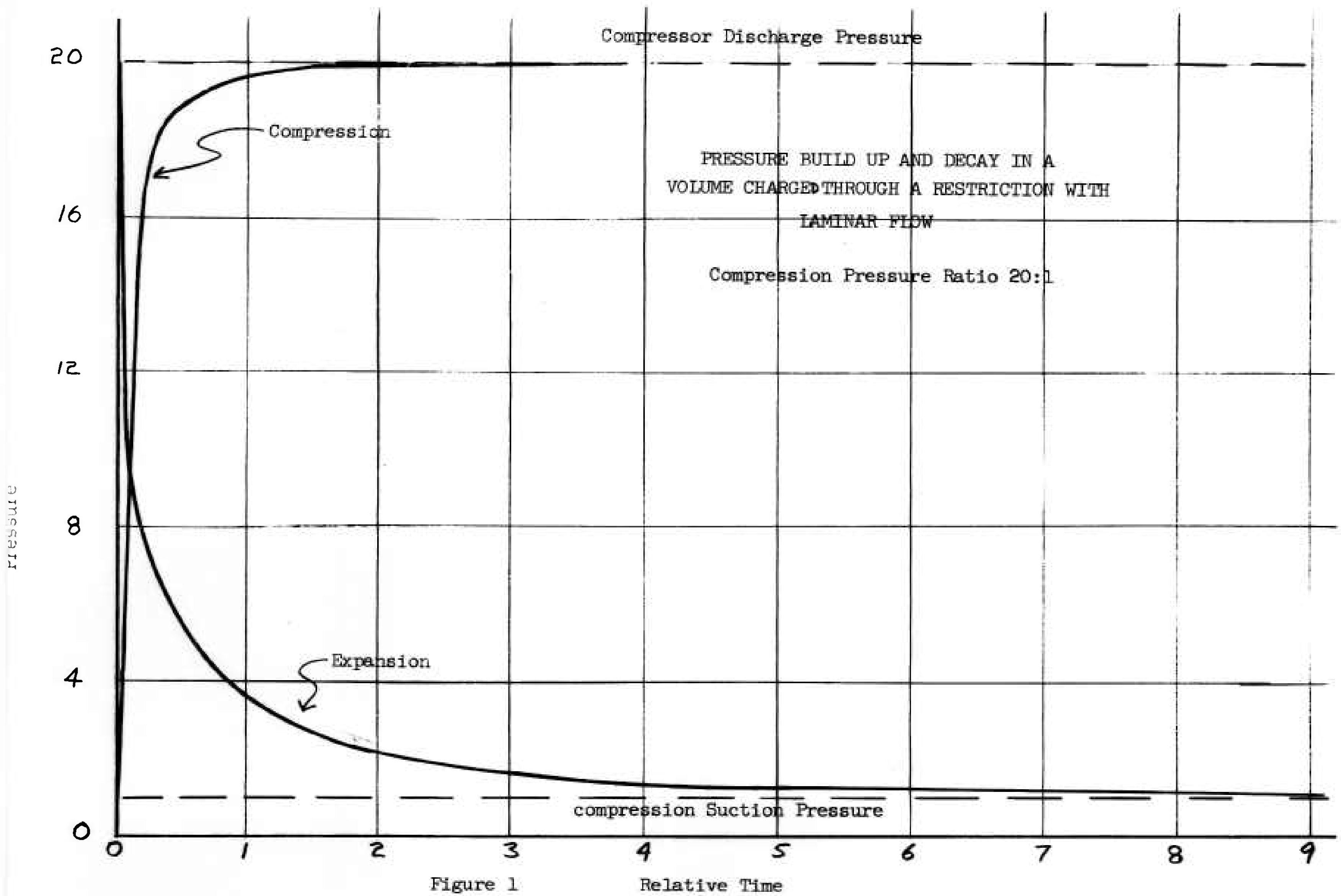

Compressor Discharge Pressure
Compression
PRESSURE BUILD UP AND DECAY IN A VOLUME CHARGED THROUGH A RESTRICTION WITH LAMINAR FLOW
Compression Pressure Ratio 20:1
Expansion
compression Suction Pressure
Pressure
20
16
12
8
4
0
0
1
2
3
4
5
6
7
8
9
Relative Time

Figure 1

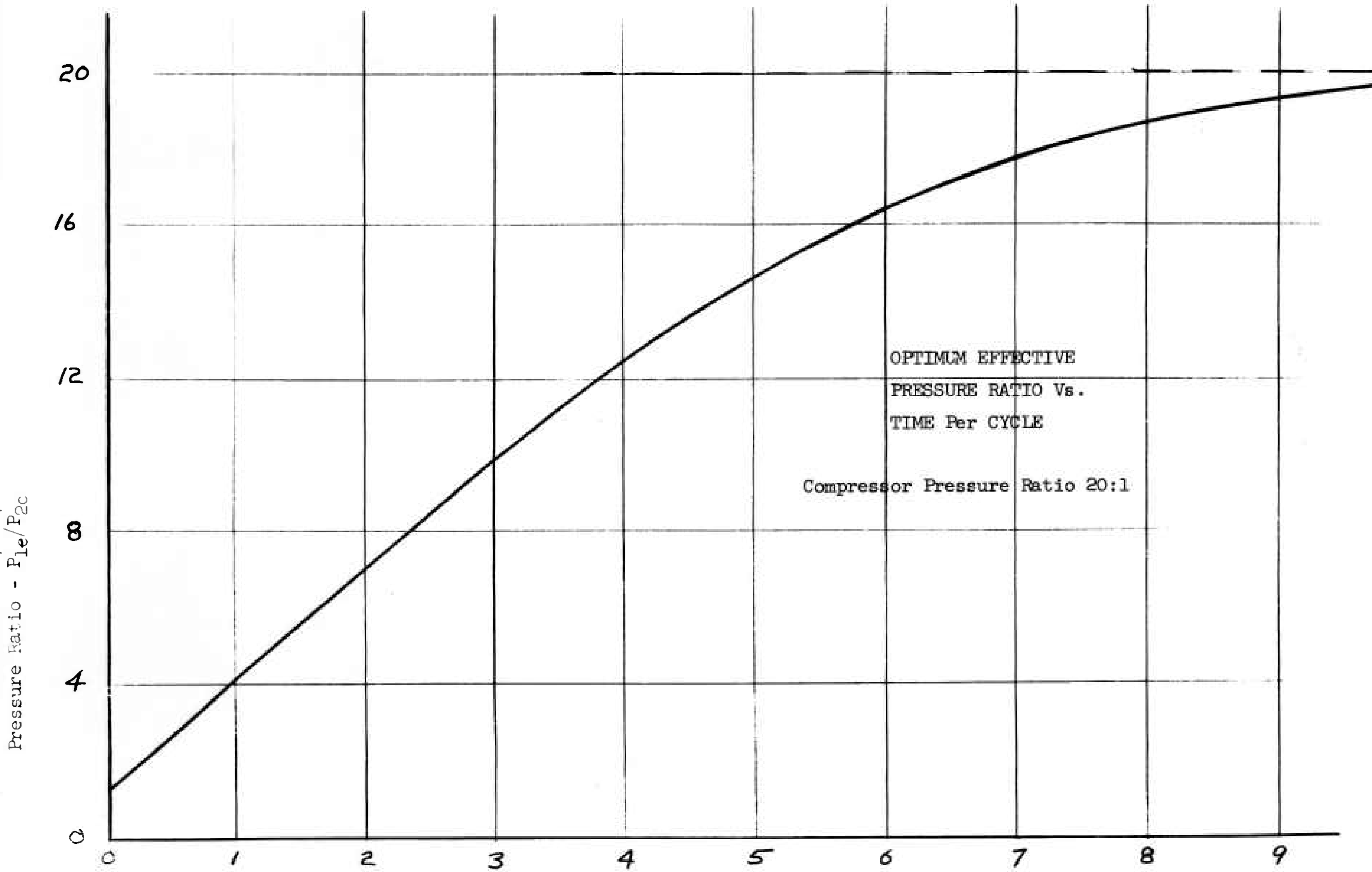
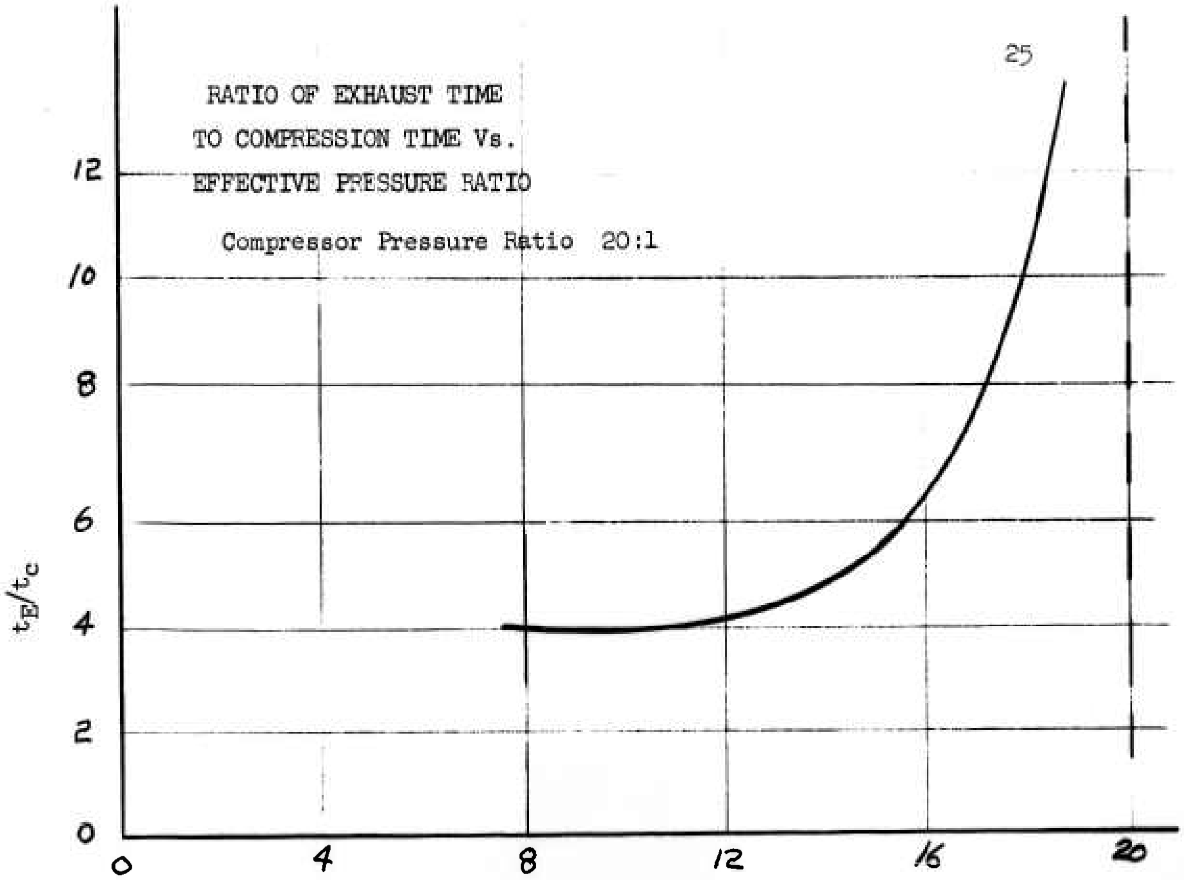
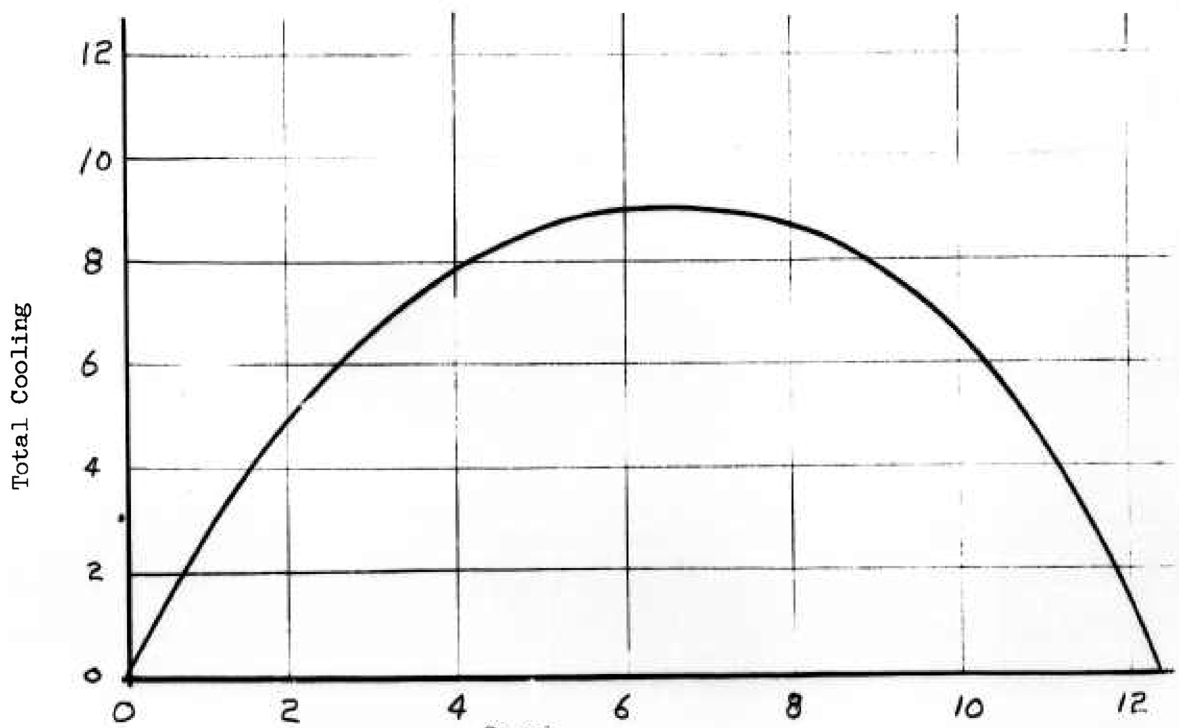


Figure 2 Time per Cycle



Effective Pressure Ratio, P_2/P_1

Figure 3



Speed

Figure 4

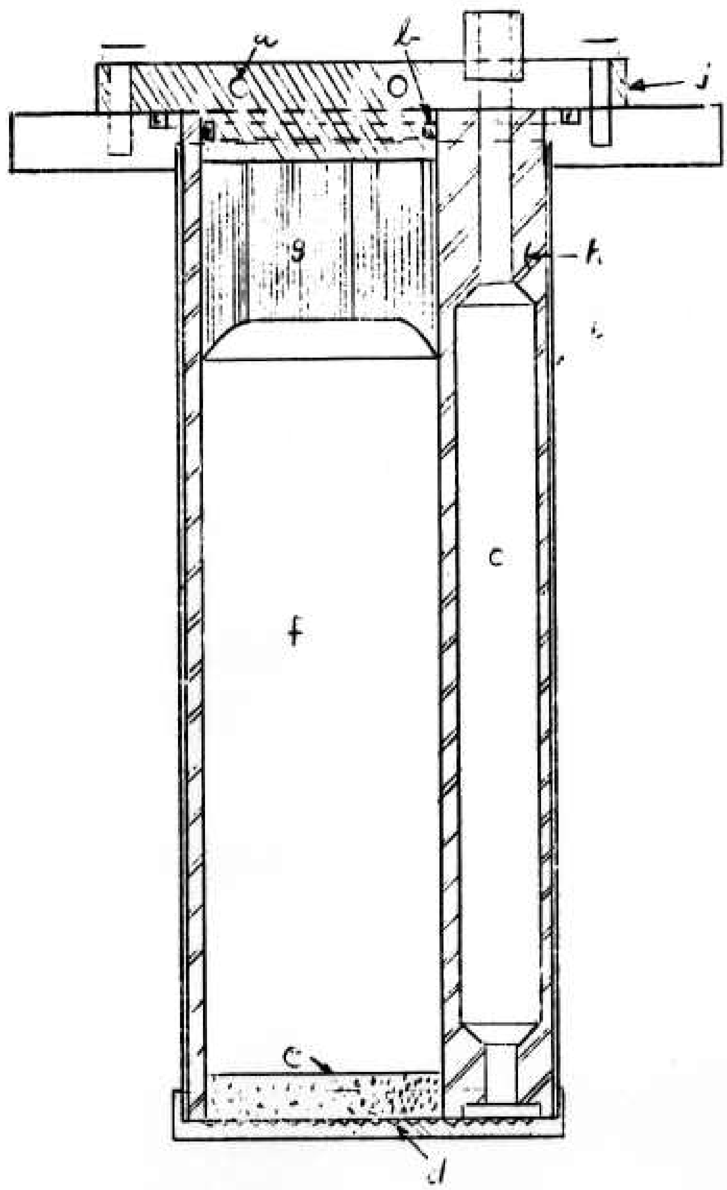


FIGURE 5. SINGLE STAGE ENCAPULATED PULSE TUBE.

A. WATER COOLING PART
B. "O" RING SEAL
C. REGENERATOR
D. HEAT SOURCE HEAT EXCHANGES
E. FLOW SMOOTHES
F. PULSE TUBE
G. HEAT SINK HEAT EXCHANGES
H. ENCAPSULATED PLASTIC
I. STAINLESS STEEL CASE
J. TOP CAP

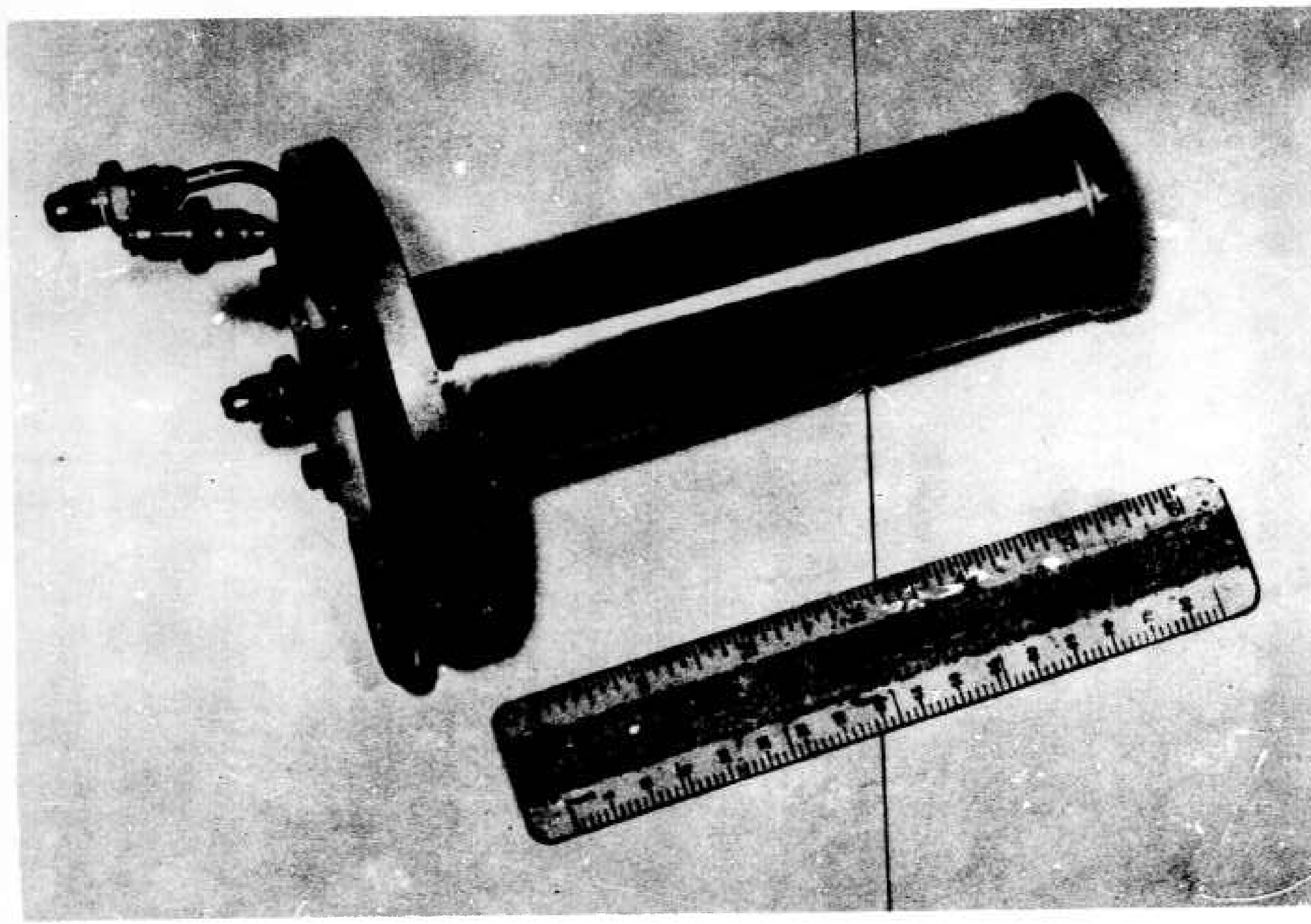


FIG. 6 SINGLE STAGE ENCAPSULATED PULSE TUBE

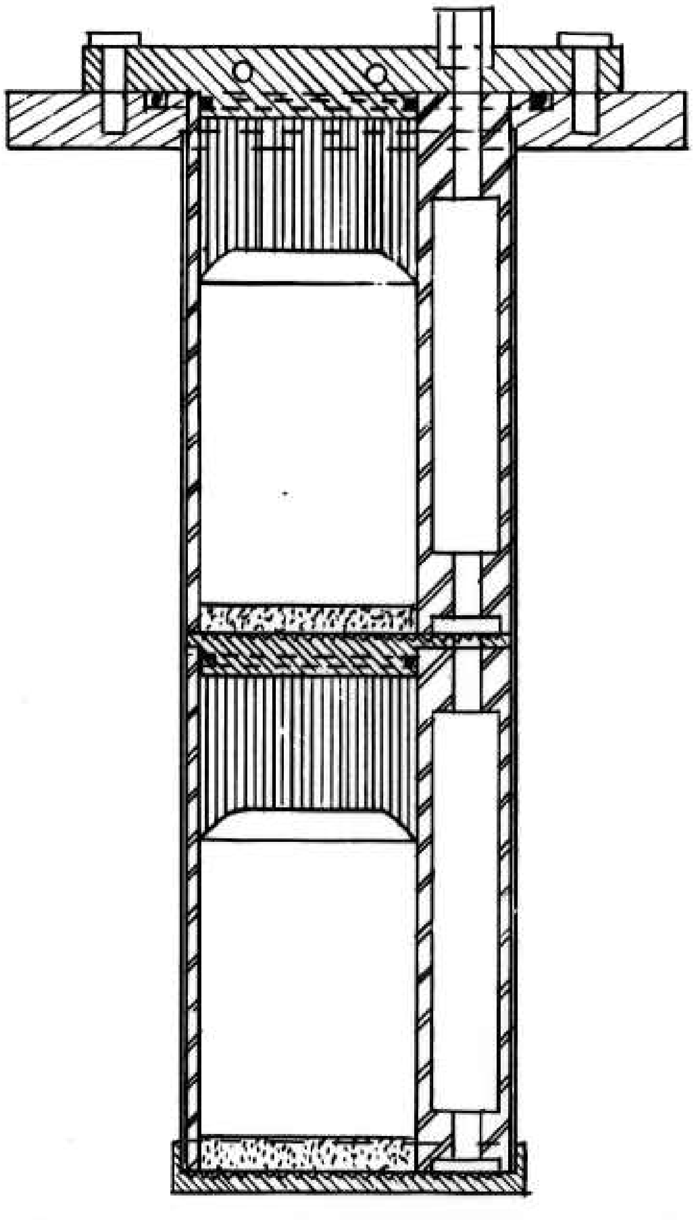


Fig. 7 Two Stage Encapsulated Pulse Tube

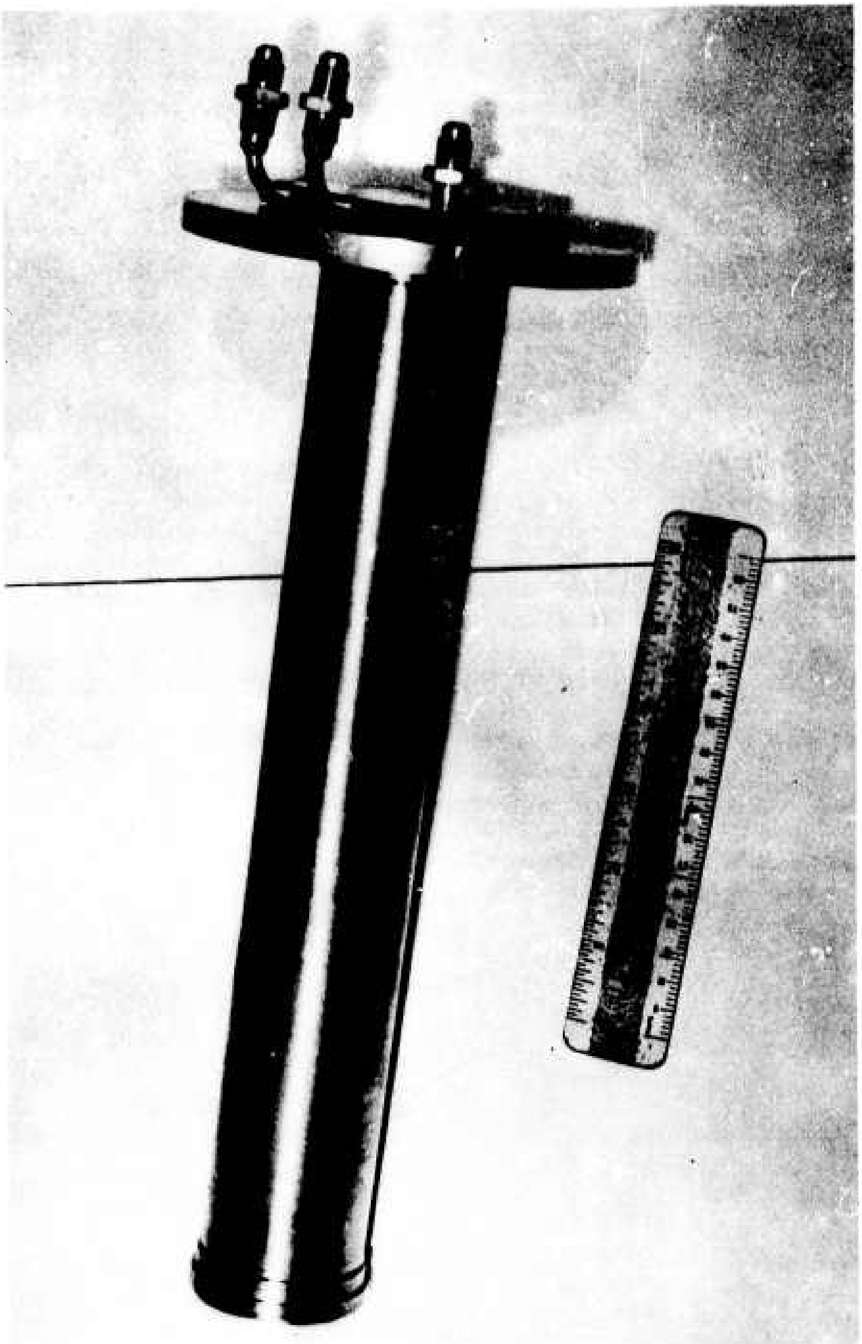


FIG. 8 TWO STAGE ENCAPSULATED PULSE TUBE

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