REGENERATION EXPERIMENTS BELOW 10K IN A REGENERATIVE-CYCLE CRYOCOOLER*

Ronald E. Sager**

AND

Douglas N. Paulson

S.H.E. Corporation, 4174 Sorrento Valley Blvd., San Diego CA 92121

ABSTRACT

At temperatures below 10K, regenerative cycle cryocoolers are limited by regeneration losses in the helium working fluid which result from the decreasing heat capacity of the regenerating material and the increasing density of helium. Our experiments are examining several approaches to improving the low-temperature regeneration in a four-stage regenerative cycle cooler constructed primarily of fiberglass materials. Using an interchangeable fourth stage, the experiments have included configurations with multiple regeneration passages, and a static helium volume for increased heat capacity. Experiments using helium-3 as the working fluid and a Malone stage are planned. Results indicate that, using these techniques, it should be possible to construct a regenerative cycle cooler which will operate below 6K.

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**Present address: Quantum Design, 11404 Sorrento Valley Road, San Diego, CA.

INTRODUCTION

The low temperature performance of regenerative cycle coolers is determined by fundamental physical properties of the solid regenerating material and the helium working fluid. Specifically, the decreasing heat capacity of solid regenerators precludes proper regeneration of the working fluid below about 10 K. In our experimental cooler, the thermal conductivity of the cylinder walls and displacer is of added importance since we use no regeneration matrix, but depend only on regeneration at the walls of a single regeneration gap. Our experiments, which are an extension of the work pioneered by Zimmerman and others [1-4] on plastic cryocoolers, are aimed at improving the regeneration properties of the coldest stage of a four-stage cooler, with the ultimate goal of obtaining temperatures below 5K.

THEORETICAL MODEL FOR REGENERATION LOSS

For estimating the effects of different fourth stage configurations on the low temperature performance of the cooler, we have developed a mathematical model for the regeneration loss in a cooler using single gap regeneration. The model assumes laminar flow along the gap, and that the fluid in direct contact with the walls is at the same temperature as the wall at that point. There will then be a temperature difference between the walls and the fluid in the middle of the gap due to the finite thermal conductivity of the fluid and its velocity along the gap.

Solving the heat flow equation for the temperature gradient in the fluid and the resulting thermal oscillation in the walls, we can compute the net enthalpy flow along the gap by averaging over one cycle of the cooler. For sinusoidal gas flow, the regeneration loss for a single stage of the cooler operating between temperature T_1 and T_2 will be given by:

$$\dot{Q}_R = \left(\frac{17}{280}\right)\left(\frac{d}{\pi DL}\right)(M\dot{n}_o)^2\left[\int_{T_1}^{T_2}\left(\frac{C_p^{\,2}}{K_f}\right)dT + \left(\frac{1.16\sqrt{\tau}}{d}\right)\int_{T_1}^{T_2}\frac{C_p^{\,2}}{\sqrt{K_w C_w}}\,dT\right] \quad (1)$$

Here d is width of regenerator gap, D is the displacer diameter, L is the length of the stage, M is the atomic weight of the working fluid (in grams/mole), $\dot{n}_o$ is the molar flow rate along the gap, τ is the period of the cooler cycle, C_w and K_w are respectively the heat capacity and thermal conductivity of the walls, and C_p and K_f are respectively the heat capacity at constant pressure (per gram) and thermal conductivity of the helium working fluid. The integrals account for the variation of C_w, K_w, C_p, and K_f over the temperature gradient, $\nabla_z T$ which is constant along the stage.

Since our experimental cooler uses a Gifford-McMahon cycle rather than a true sinusoidal Stirling cycle, the sinusoidal assumption is probably the most significant approximation in the calculation. However, the approximation should affect only the multiplicative constant in equation (1), not its functional form, and when applied to a nonsinusoidal gas flow, the essential error is an uncertainty in the value of the molar flow rate. To account for this problem in making comparisons with our experiments, we have used the molar flow rate as an adjustable parameter. We expect that this will not be necessary in future experiments with a true Stirling cooler since the molar flow rate should be more well defined.

The two terms in equation (1) arise from two different aspects of the regeneration process. The first term, which depends only on the heat capacity and thermal conductivity of the working fluid, arises from the temperature difference between the middle and the edges of the gap. Physically this term represents the regeneration loss incurred by imperfect regeneration of the fluid in the middle of the gap. The second term, which depends on the heat capacity and thermal conductivity of the walls, represents the ability of the walls to accept heat from the working fluid as it flows toward the cold end of the machine. At temperatures below about 10K the thermal conductivity and heat capacity of the walls become very small, so the walls can absorb less and less heat from the gas. The problem is made substantially worse at these temperatures by the rapidly increasing density (and increasing heat capacity per unit volume) of the helium.

It is interesting to evaluate the regeneration loss for a single stage constructed of nylon having a single regeneration gap and with its warm end operating at about 30K. The results are shown in Figure 1. The calculation assumed a gap width of .0025 cm, displacer diameter of .47 cm, length of 15 cm for the stage, and a maximum molar flow rate of .0043 moles/sec. This molar flow rate should be roughly characteristic of a cooler operating at one Hertz with a 0.7 cm stroke between pressures of 0.2 and 0.8 MPa.

Since the regeneration loss completely dominates other internal losses below about 15K, we expect the operating temperature of the cooler to be the point at which the internal regeneration loss just balances the total cooling power. The solid line in Figure 1 shows the estimated regeneration loss; the broken line gives the estimated cooling power assuming isothermal expansion. The peak in the cooling power at 5.2K, which reflects the behavior of the isobaric expansion coefficient for helium near its critical point, will probably be less dramatic in practice since the expansion will not be truly isothermal.

From this simple model, we would predict an operating temperature of about 10K for this stage. For comparison with experiment, the assumed operating parameters are those reported by Zimmerman for his four-stage machine which operated at about 8.5K. The discrepancy may arise from our assumption that the full pressure fluctuation is developed in the final stage of the cooler, which would assume a higher molar flow rate than actually exists in the cooler. In any event, the qualitative nature of the regeneration loss should accurately reflect the difficulty in achieving temperatures below about 8K. This seems to be consistent with Zimmerman's later results in which his five stage cooler reached about 7K, only a 1.5K improvement after adding an entire stage to his four stage machine.[3]

The dashed line in Figure 1 shows the contribution to the regeneration loss from the first term in equation (1). Physically this curve represents the regeneration loss to be expected if there were infinite heat capacity in the walls of the cylinder and displacer. While the contribution from this term is not insignificant, especially below about 6K, the comparison clearly demonstrates the requirement for first solving the heat capacity problem. The experiments we describe below specifically address the question of trying to improve the regeneration capacity of the cylinder and displacer walls at temperatures below 15K.

Although our primary interest is the thermodynamic behavior of the coldest stage, we have modeled the performance of the machine in terms of conduction, viscous, and shuttle heat losses, as well as radiation loading on the cooler. Our models for shuttle and conduction losses [2,5] have been verified in separate measurements [6]. Calculations for viscous and radiation losses are sufficiently straightforward that we do not expect any major errors in these estimates.

THE EXPERIMENTAL APPARATUS

To perform the experiments we have constructed a four stage cylinder and displacer, similar to Zimmerman's, having dimensions as given in Table I. The displacer is driven by a variable speed stepping motor through a scotch yoke, and a standard compressor of the type used for commercial cryocoolers provides the high and low pressure sources for the expansion/compression cycle. Electrically actuated pneumatic valves which control expansion and compression in the cooler are triggered from the displacer drive such that the phase of the displacer stroke and the duration of both expansion and compression can be adjusted.

While the eventual goal of our work is a more self-contained compact device, we felt that the Gifford-McMahon configuration provided added flexibility for the basic experiments with which to investigate in detail the effects of different parameters on the operation of the cooler. Another convenient aspect of the apparatus is the detachable fourth stage which allows quick modifications to the most interesting part of the cooler. This design, which uses an indium O-ring to make the seal, has functioned well, and we have had no particular problem repeatedly achieving a vacuum tight seal, at least for our pressures of up to about 1.0 MPa.

In the experiments, we are pursuing essentially three different ideas for improving the low temperature regeneration in the cooler. First, the heat capacity per unit volume of helium increases at low temperature rather than decreasing. This suggests the possibility of trying to use static helium fluid residing in the cylider walls of the last stage to provide the requisite heat capacity[7]. A possible configuration is shown in Figure 2a.

Thermal contact with the static helium must be provided by constructing a cylinder which has an extremely anisotropic thermal conductivity -- very high in the radial direction to allow heat transfer between the working fluid and the static helium, but very low parallel to the cylinder axis to prevent thermal conduction down the cooler. Ideally one might use a thin-walled laminated tube consisting of alternating rings of copper and stainless steel bonded together, but in practice the fabrication of such a structure which is helium leak tight has proven to be extremely difficult. Nonetheless, we have achieved a partial success, and have collected some data from a cooler using this type of assembly.

Our second technique is to use helium-3 as the working fluid in the cooler. Obviously for a Gifford-McMahon cycle, or any large cooler, this becomes prohibitively expensive. However, a small Stirling cooler could be charged with helium-3 for a few hundred dollars. If the cooler were constructed completely without dynamic seals, it should be able to operate for years before the helium charge would require replenishment.

The advantage of using helium-3 is that both its critical pressure and critical temperature are substantially lower than those of helium-4. At temperatures and pressures near their critical points, the density and other thermodynamic properties of the two isotopes will scale approximately as the ratio of their critical temperatures and pressures. Since the increasing density of the fluid contributes to the regeneration problem, we expect an improvement when using helium-3 since its density and heat capacity per unit volume will be somewhat less than those of helium-4 at the same temperature. Furthermore, since our type of cooler is designed to operate at low pressures, typically between 0.2 and 1 MPa, the lower critical pressure will be advantageous by allowing the cooler to operate completely above the critical pressure of the helium-3 and avoid the density and heat capacity problems encountered at the critical point of helium-4.

Finally, we are also investigating the use of a somewhat different type of refrigeration cycle for the final stage of the cooler. Allen, et al,[8] have observed that the counterflow heat exchange used in a Malone cycle[9] might help alleviate the problem of vanishing heat capacity in the regenerator. In the Malone cycle the helium is forced to flow through one channel as it moves toward the cold end of the machine, and through a second parellel channel as it returns to the warm end of the cooler. As shown in Figure 2b, heat exchange occurs across the wall separating the two channels, providing regeneration through a type of counterflow heat exchanger. In a cryocooler using a Malone stage, we are testing a tandem design in which the reciprocating flow of a staged Stirling cryocooler feeds the pulsating unidirectional flow of the Malone engine using a pressure seal and check valves. An essential quality of the Malone engine is the use of a pulsating, unidirectional fluid flow in which the regenerator behaves similarly to a counterflow heat exchanger so that, as constant pressure, the working fluid regenerates itself. Consequently, the thermal load on the Malone regenerator is greatly reduced over that in a Stirling machine. This is is particularly advantageous from the standpoint of the heat capacity problem.

There are two disadvantages to this approach, however. First, the reciprocating fluid flow in the upper three stages of the cooler must be converted to a pulsating unidirectional flow in the final stage. This will require check valves between the third and fourth stages to provide preferred paths for the incoming and outgoing fluid. Secondly, while the outgoing fluid stream can provide the heat capacity for regenerating the incoming fluid, the Malone regenerator requires a substantially larger regenerator dead volume, which introduces an additional heat capacity problem. Specifically, the larger dead volume means that a significant amount of helium remains in the fourth stage regenerator during the compression cycle. Hence, there must still be sufficient heat capacity available in the walls and displacer of the final stage to absorb the heat of compression of the fluid remaining in the regenerator. Because of this effect, a cooler using only a Malone-style regenerator in the fourth stage may not achieve a temperature significantly below one having only a loaded regenerator. However, we feel that there is considerable merit in the combination of a Malone stage operating inside a helium loaded regenerator.

EXPERIMENTAL RESULTS TO DATE

To provide a simple system for comparison with calculation, and to compare with Zimmerman's experiments, we first ran the cooler with no fourth stage at all. The operating temperatures of the three stages for this configuration are given in Table II. Since we imposed no external load on the machine (other than that produced by the radiation shields), each stage should operate at the temperature where the inherent thermal losses in the machine plus the radiation loading just equal the refrigeration produced by the stage.

To determine the refrigeration power which the cooler was producing, we measured the P-V function at the top of the cooler using an absolute pressure transducer with the volume axis simulated by a sinewave generator synchronized with the displacer. Over several measurements with different cooler configurations, we found that the P-V diagram thus generated gave a value for the total refrigeration very close to that predicted for a sinusoidal pressure fluctuation. Consequently, for later experiments we simply used that approximation.

For making comparisons between experiment and theory, we used the observed operating temperatures of the various cryocooler stages as data points, then computed the total thermal loading and cooling power expected at each stage of the cooler. Table II shows the comparison for the experiments with the three stage cooler. On the basis of our independent measurements of shuttle and conduction losses, we believe these estimates to be accurate to within a few percent. Futhermore, the estimates for the viscous losses are nearly negligible as are the radiation losses in stages two and three. Since the relative magnitudes of the other losses change dramatically over the length of the cooler, the overall agreement gave us reasonable confidence in our model.

In our first experiment with a four stage machine we used nested cylindrical sleeves as shown in Figure 2c for the final stage. The initial concept was that the additional regeneration passages would allow a greater volume of fiberglass in the cylinder walls to participate in the regeneration process. However, later investigation showed that the heat capacity of the fiberglass drops so quickly below 15K that even if the entire volume of the last stage participates in the regeneration, there is still far too little heat capacity to provide proper regeneration below about 10K.

Results from this configuration are given in Table III. The lowest operating temperature we achieved in this configuration was 7.6K, which compares favorably with the 8.5K obtained by Zimmerman in his four stage machine[1]. The improvement is probably due to a combination of the larger volume of fiberglass provided in the fourth stage and also perhaps to our use of a Gifford-McMahon cycle rather than the Stirling cycle. Although the actual operating parameters were somewhat different from those reported by Zimmerman, estimates for the regeneration loss and refrigeration power in our cooler are qualitatively the same as those in Figure 1. In particular, the regeneration effectiveness in our configuration also decreases precipitously below 10K, producing a virtual wall at about 8K.

As with the three stage experiments, the calculations for the estimated losses and the cooling power in the first three stages in the machine show reasonable agreement with the observed operating temperatures. While the estimates for the fourth stage are clearly incorrect the error is not surprising considering the complicated geometry of the cooler. The regeneration loss for the fourth stage was estimated by assuming that the fluid flow divided among the various regeneration passages in proportion to the flow impedances of each passage. The loss was then computed separately for each gap. This model is reasonable so long as the thermal penetration depth in the fiberglass is less than half of the radial distance between the gaps. In fact, for our geometry this approximation breaks down between 10 and 15K, so it is not surprising that the value in Table III underestimates the regeneration loss in the fourth stage.

In more recent experiments we have operated the cooler with the fourth stage constructed as in Figure 2a. Table IV gives the operating temperatures of the cooler with this fourth stage configuration using alternating rings of copper and stainless steel. As in the other experiments, the comparison between the observed operating temperatures and calculated losses in the machine is reasonable, with the exception of the third stage. We believe that the failure of this configuration to achieve a temperature of less than 7K resulted from a small leak between the static helium volume and working volume of the cooler. Our measurements of the leak rate suggested that the cyclical pressure fluctuations in the static volume would be a few percent of those in the working volume of the engine. Since even small pressure fluctuations in the relatively large static helium volume will involve a substantial heat of compression, we believe that such a leak could significantly degrade the cooler performance. Also viscous heating of liquid flowing through the leak will introduce an additional thermal load on the cold end of the machine. For the same reasons, the comparison with calculation is suspect.

An additional potential problem with our design is that the thermal conductivity of the static helium itself is rather poor in this temperature regime, so it becomes difficult to effectively access the inherent heat capacity of the static fluid. The spacing of the thermally conductive discs was based on the effective thermal penetration depth of the static fluid, so that ideally all of the static fluid could participate in the regeneration process. However, more effective thermal contact with the static fluid may be possible using a packed screen mesh or similar construction to provide the thermal contact between the static and working fluid [5].

Nonetheless, this four stage machine did achieve an ultimate temperature of 7.1K which is a significant improvement over the 7.6K performance of the machine with just a fiberglass regenerator. A quantitative feeling for the improvement can be obtained from noting in Figure 1 that a decrease in operating temperature from 7.6K to 7.1K represents a change from 370 mW to 530 mW in the regeneration loss in a fiberglass regenerator. We believe that the performance of a cooler with a helium-loaded final stage can be further improved with a completely sealed static volume, and we are currently developing more reliable techniques to fabricate this structure.

We are also presently preparing for experiments with a Malone-stage displacer and helium-3 as the working fluid. Experiments using both of these concepts should be completed within a few months. Nonetheless, the major problem is still to provide the proper regenerative heat capacity in the fourth stage, and we feel that the use of some type of static helum loading represents one of the best approaches to that problem.

COMMENTS AND CONCLUSIONS

We feel that our results to date provide encouragement for further work in trying to develop a small low-power cooler which can achieve a temperature of about 5K. We have, however, encountered several experimental problems which must also be addressed, and will probably require some clever engineering to completely solve. One of our most persistent problems has been the diffusion of helium through both the O-ring seals and the plastic materials of which the cooler is constructed. In our laboratory apparatus we have simply pumped on the vacuum container continuously to prevent thermal loading of the cooler via conduction through residual helium gas. Monitoring the vacuum can with a helium leak detector has provided quantitative information on the effectiveness of that approach. To guarantee the proper thermal isolation, the residual helium pressure will have to be kept below 10^{-5} Torr or so, which is a stringent long term requirement when using materials which allow helium diffusion. Zimmerman has begun to address the problem by incorporating metal-foil diffusion barriers in his more recent cooler,[10] but a complete solution will probably also require some innovative application of hermetic sealing techniques.

Our immediate problem has been that the possiblility of residual helium in the vacuum container has introduced a certain level of uncertainty in the interpretation of our data. While comparisons with calculation do not show any major thermal loading, the balance between internal losses and cooling power in the coldest part of the machine could be substantially altered by even a very small residual heat leak.

In addition to this uncertainty, we have encountered some problem of reproducibility in our experiments from run to run, most notably in reproducing the results of our initial experiments with the fiberglass regenerator. Part of the problem probably arises from an orientational effect in the displacer. Specifically, since the upper three stages of our machine use a rigid displacer, the interstage alignment is critical. In recent experiments we have seen a significant effect on the coldest stage by rotating the displacer to different azimuthal orientations. This result was completely unexpected since the displacer has centering stubs which should keep it properly aligned and it has not been observed to bind inside the cylinder even when cold. Two possible explanations are that the centering stubs have worn or that certain orientations produce enough frictional heating to adversely affect the coldest stage. On examination, we did find some wear on the centering stubs, but not enough to account for the effect. In future coolers it will probably be advisable to allow for some type of self-aligning mechanism between adjacent stages.

To summarize, we feel that our results to date provide hope for the eventual realization of a small low-power cooler which can operate below 6K. Eventually we hope to push that limit to below 5K. Although many practical problems still remain to be solved, once a prototype device has been demonstrated, its potential applications should generate a substantial interest in solving those remaining problems.

Table I. Physical dimensions of four stage experimental cooler. All values in centimeters.

Stage	1	2	3	4
Displacer Dia.	3.40	2.02	1.21	0.52
Cylinder O.D.	3.96	2.49	1.52	1.52
Displacer length	16.0	15.5	15.6	16.0
Regenerator gap	.0152	.0102	.0102	.002

Table II. Estimated losses and refrigeration in three stage cooler. All values are given in milliwatts. Stroke = .75 cm, Speed = 2.1 sec, P_{hi} = .78 MPa, and P_{lo} = .14 MPa.

Stage	1	2	3
Operating Temp (K)	164	53	12
Regeneration	170	185	168
Shuttle	178	46	5
Conduction	476	110	9
Viscous	33	14	0
Radiation	222	7	0
Total Losses	1079	362	182
Total Refrigeration	1064	372	208
Discrepancy	1.4%	2.7%	12.5%

Table III. Estimated losses and refrigeration in four stage cooler with multiple sleeve regenerator. All values are given in milliwatts. Stroke = 1.5 cm, Speed = 1.5 sec, P_{hi} = .79 MPa, and P_{lo} = .15 MPa.

Stage	1	2	3	4
Operating Temp (K)	164	68	30	7.6
Regeneration	578	464	286	45
Shuttle	760	179	25	6
Conduction	503	87	10	4
Viscous	97	23	4	2
Radiation	220	7	0	0
Total Losses	2158	760	325	56
Total Refrigeration	2210	774	354	152

Table IV. Estimated losses and refrigeration in four stage cooler with helium loaded refrigerator. Stroke = 1.0 cm, Speed = 1.75 sec, L_4 = 16 cm, P_{hi} = .75 MPa, and P_{lo} = .15 MPa.

Stage	1	2	3	4
Operating Temp (K)	164	76	26	7.1
Regeneration	330	306	358	83
Shuttle	341	90	16	1
Conduction	473	61	13	3
Viscous	76	42	4	9
Radiation	220	7	0	0
Total Losses	1440	506	391	96
Total Refrigeration	1528	534	245	121

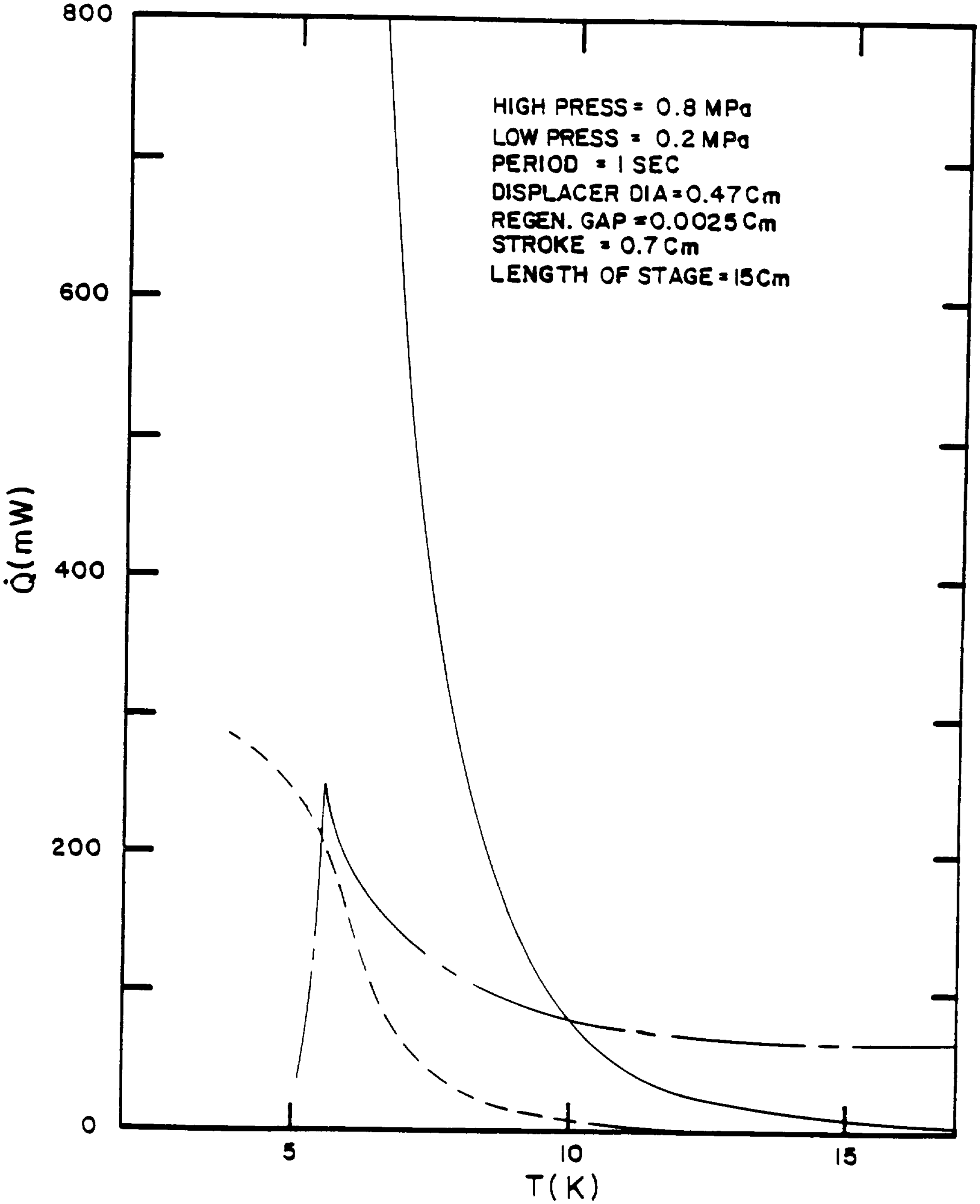

Figure 1. Estimates for regeneration loss and refrigeration for coldest stage of a Stirling-cycle cryocooler. The solid line shows the regeneration loss for operating parameters shown. The broken line (———) shows the expected refrigeration produced by the stage, and the dashed line (- - -) gives the contribution to the regeneration loss from the finite thermal conductivity of the fluid in the regenerator gap.

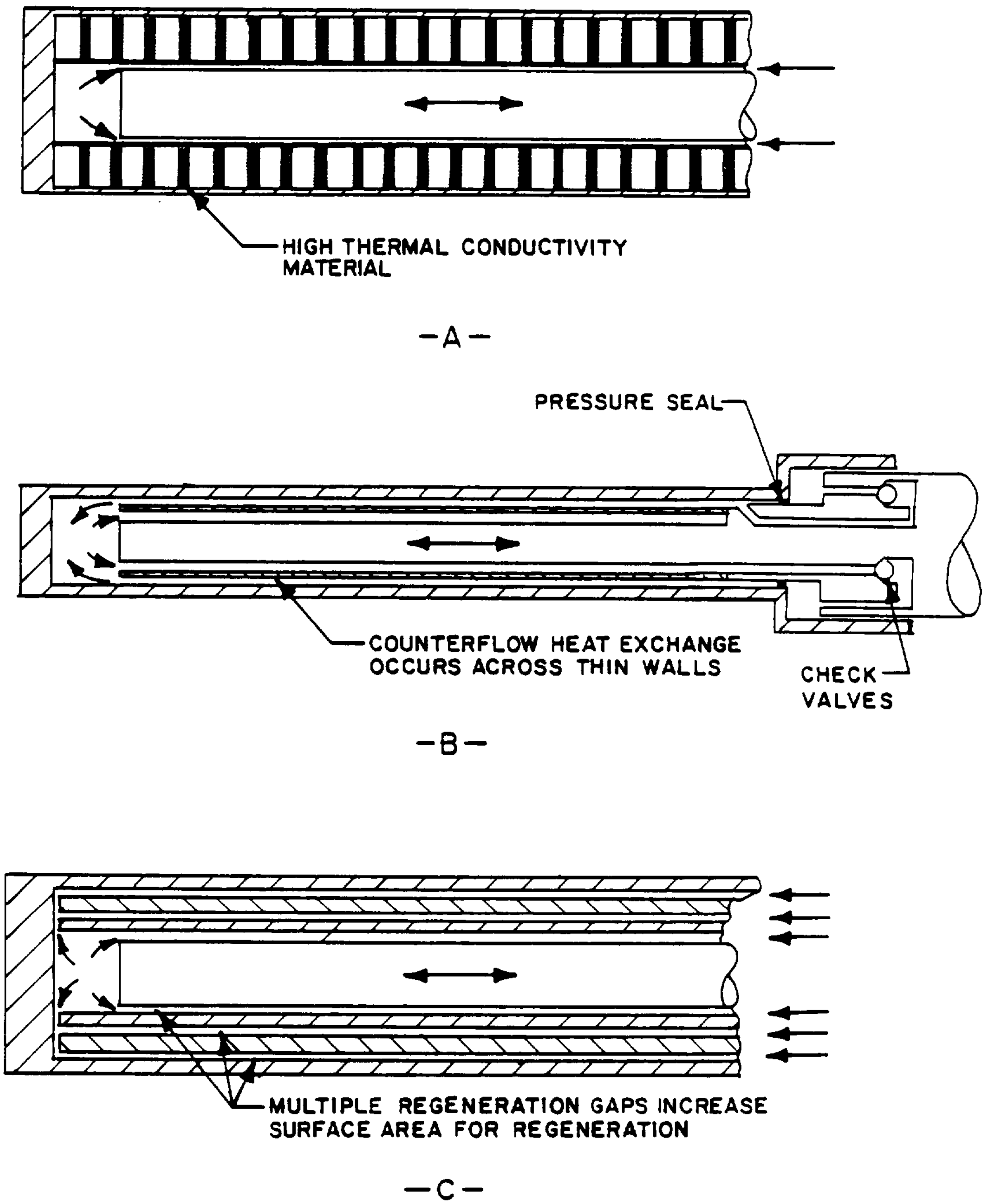

Figure 2. Part (a) shows scheme for loading the final stage of a Stirling cooler with static helium to provide additional heat capacity for regeneration. Part (b) shows counterflow heat exchange mechanism in a Malone cycle, and part (c) shows regeneration gap configuration in first experiments with a 4-stage Stirling cooler.

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