Cryogenic systems of superconducting accelerators

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The problems solved by cryogenic engineers when constructing superconducting accelerators are discussed. The main elements of cryogenic systems are considered, the loss sources in them are pointed out, and ways for improving them are indicated. The plans for the cryogenic systems of the largest superconducting accelerators are described.

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INTRODUCTION

It is well known that the energy of accelerated particles is directly proportional to the magnetic field in which the particles are accelerated and to the accelerator radius. Since the saturation of iron in electromagnets of normal type restricts the magnetic field induction to about 2 T, further increase in the particle energy is possible only by increasing the size of the accelerators. The accelerators most recently constructed in the world have already reached impressive sizes: The diameter of the accelerating ring at Serpukhov is 0.5 km, and at Batavia in the United States it is 2 km. At the same time, the energy requirements needed to supply the electromagnets approach the limits of the possible. Therefore, development of accelerator technology in the direction of superhigh energies depends directly on progress in technical superconductivity and cryogenics. The present plans for the largest accelerators in the world [the Energy Doubler and ISABELLE in the United States, the Accelerator and Storage Facility (UNK: Uskoritel'no-Nakopitel'nyi Kompleks) in the Soviet Union, and HERA in the German Federal Republic] are based on the use of superconducting magnetic systems.

The cryogenic systems of superconducting accelerators take up heat released in the accelerator coils when the magnetic field is changed and when scattered accelerated particles are stopped and also heat which flows in from the ambient medium through the thermal insulation, the supports, and the power leads. To maintain a constant cryogenic temperature[1)] of the magnets, it is necessary to do work. The minimal amount of work that must be done to take heat from an object that has constant temperature T can be calculated in accordance with the formula $e_q = q(T - T_0)/T$, where T_0 is the temperature of the ambient medium. The quantity e_q is called the energy of the heat flow q. The useful portion of the heat is characterized by the Carnot factor

$$\tau = e_q/q = (T - T_0)/T,$$

which shows how many watts must be expended to compensate 1 W at the cryogenic temperature (Fig. 1). As we see, the energy requirements increase rapidly with decreasing temperature: $|\tau| = 2.89$ at liquid-nitrogen temperature (77.4 °K), but the value is 13.7 at liquid-hydrogen temperature (20.4 °K), and 70.4 at liquid-helium temperature (4.2 °K). At very low temperatures, the energy requirements are approximately proportional to $1/T$, so that a decrease in the cryostating temperature from, for example, 6 to 3 °K (by just 3 °K) at least doubles the energy requirements. This circumstance must not be lost sight of when considering the requirements on the cryostating system, since otherwise the use of a superconducing system instead of an ordinary system may not make economic sense.

The work requirements given above are attainable only theoretically if all the processes are ideal, i.e., reversible. The actual requirements are significantly greater, and the efficiency

$$\eta = e_q/l,$$

of a cryogenic system is, as a rule, lower, the lower its refrigerating capacity (Fig. 2). In the above expression, l is the actual expenditure of work. At 4.2 °K, the minimal requirement is 70.4 W/W, but the real requirements lie in the range from 1000 W/W ($\eta \approx 7\%$) for small laboratory systems down to about 300 W/W ($\eta \approx 23\%$) for the currently constructed large systems. In the cryogenic systems of superconducting accelerators, the electrical power requirements may reach tens of megawatts,[1,2] and therefore the economic question acquires fundamental importance.

1. BASIC ELEMENTS OF THE CRYOGENIC SYSTEMS

The working temperature of the magnetic in the current projects for superconducting accelerators lie in the range of "helium" temperatures—usually 4.5 °K and below—and we shall therefore consider mainly

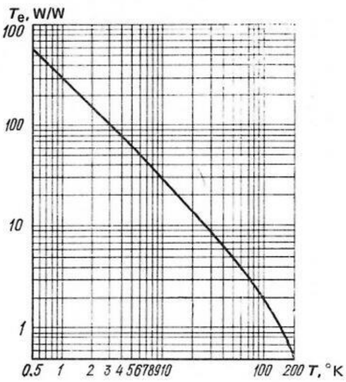

FIG. 1. Minimal specific work of cryostating as a function of the temperature.

[1)] Temperatures below 120 °K are called cryogenic.

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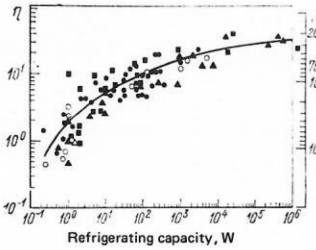

FIG. 2. Efficiency of cryogenic systems as a function of the refrigerating capacity.[1] The open circles are for planned refrigerators, the black circles (1.8–9 °K), black squares (30–90 °K), and black triangles (10–30 °K) are for existing refrigerators.

helium cryogenic systems.

In each such system, one can distinguish the following basic elements: the system for holding the refrigerant; the system for preparing the refrigerant, including its compression and decontamination from oil, moisture, and foreign gases; the refrigeration block, which produces the refrigerant at given parameters; and the cryostating system, which extracts head and keeps the magnets at the required temperature.

We shall consider only the elements that constitute the cryogenic facility proper, i.e., the elements that realize the cryogenic cycle and also the elements in the cryostating system. Typical schemes of cryogenic systems are shown in Fig. 3. There are three relatively independent steps: the compression (C) of the working medium, in which the refrigerant is compressed and simultaneously or subsequently cooled to the temperature of the ambient medium; precooling (PC), in which the temperature of the compressed refrigerant is reduced by means of certain cold sources; and the final cooling (FC), in which transformation of the exergy stored in the previous steps produces "cold" in the necessary amount and quality in the form of a flow of refrigerant with given prameters.

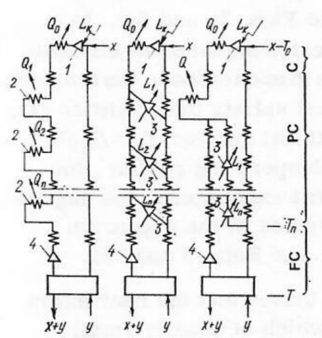

FIG. 3. Schemes of cryogenic systems: a) with external refrigeration sources, b) with gas-expansion machines connected in parallel, and c) in series. The compression stage is C, the precooling stage PC, and the final cooling stage FC; 1 is the compressor, 2 the external refrigeration sources, 3 the gas-expansion machines, 4 a gas–liquid expansion machine, x is the flow extracted from the cycle, and y is the refrigeration flow.

Compression of the working medium

The ideal here would be isothermal compression with the heat released by the gas continuously removed into the ambient medium. Then the amount of work that must be done would be minimal. However, in real compressors, the compression is closer to an adiabatic process than an isothermal one, since the surface of the region in which the compression takes place is usually too small to ensure sufficiently good heat extraction. The modern tendency to construct ever more powerful and rapidly operating machines has not only positive aspects—the reduction in the mass and size of the machines—but also negative features, since it is not easy to remove the heat from the gas during the compression process.

At the present time, reciprocating compressors are generally used in helium systems. Their isothermal efficiency is usually near 60%, the period of uninterrupted operation does not in practice exceed 500 h, and the total working life is 10 000 h. Reciprocating compressors are used in helium cryogenic systems with a refrigerating capacity from a few watts to 2–3 kW, which is generally adequate for the present-day requirements of science and technology. They are usually not required to ensure continuous uninterrupted operation, and therefore the relatively low reliability and short working life of reciprocating compressors are not an obstacle to their use.

For superconducting accelerators, much more powerful and reliable systems are needed. The duration of continuous uninterrupted operation of the compressors must be not less than 1500 h and preferably should be 3000–4000 h with a total working life of 60 000–90 000 h. The refigerating capacity required of the cryogenic system for the Accelerator and Storage Facility is estimated[2] at 40–50 kW, and the amount of circulating helium is approximately 100 000 nm^3/h. Because of the expense of helium, the sealing of the compressors and the system as a whole must be very good indeed. If the loss during an operating time of 5000 h in a year is only 0.001%, the cost of replenishing the helium is about 50 000 roubles. It is also important that the compressors should not contaminate the helium with air or other gases and should have a reliable system for decontamination from lubricating oil.

Turbine compressors would meet these requirements most fully. However, the specialists are pessimistic about the possibility of constructing an effective and reliable helium turbine compressor, since too many steps would be needed to compress helium to 2–3 MPa on account of its low molecular weight. In the Accelerator and Storage Facility it is proposed to use screw compressors, in which the gas is compressed in regions of variable volume formed by two synchronously rotating screws. But whereas the isothermal efficiency of screw compressors is practically the same as for piston compressors in the case of operation with air, it is much lower in the case of operation with helium because of the greater internal seepage of the compressed gas from the high-pressure to the low-pressure side made possible by the low viscosity of helium. In tests of a helium screw compressor, the isothermal efficiency did not exceed 50%.[3]

Thus, already in the first stage, i.e., in the compression of the refrigerant, 40–50% of the work is expended for nothing, and in the near future one cannot unfortunately expect any significant improvement in this respect.

Precooling

This part of the system is distinguished by the greatest variety in the methods used to achieve it. As cold sources one can use, for example, tanks with liquid refrigerants or gas refrigerators (see Fig. 3a). Another method is to cool one part of the compressed refrigerant by means of another part expanded in a cascade of gas-expansion machines connected in parallel (see Fig. 3b) or in series (see Fig. 3c). In helium systems, precooling combinations are not uncommon, the first source being a tank with liquid nitrogen, and the following sources being gas-expansion machines.

The aim of the precooling is to cool the refrigerant from the temperature T_0 of the ambient medium to the temperature T_n. In this connection, the main tasks in the development of the precooling technology are the choice of the number and type of cold sources, the determination of the optimal temperature levels at which they are activated, and, when gas-expansion machines are used, the choice of the optimal pressure ratios.

Analyzing the cycle of a helium liquifier with a cascade of gas-expansion machines connected in parallel, Kapitsa[4] showed that if the adiabatic efficiencies of the expansion machines are equal, the optimal choice is to use them in such a way that the temperatures after the machines form a geometrical series:

$$T_i = T_n \left(\sqrt[n]{T_0/T_n}\right)^{n-i}, \qquad (1)$$

where T_0 and T_n are the initial and final temperatures in the cooling process, and n is the number of expansion machines.

It is easy to show that this result can also be extended to the case of a cryogenic system with expansion machines in combined liquifaction–refrigeration (or purely refrigeration) regimes.[5,35] Moreover, it is also valid for precooling with any other cold sources, provided they have equal efficiencies.

The simplest cold sources are tanks with liquid refrigerants. However, they have a fundamental shortcoming—the heat is extracted from the refrigerated flow in the tank at a constant temperature, the boiling point of the refrigerant. This process is accompanied by unavoidable losses. The efficiency of the cooling process, which is equal to the ratio of the minimal work of cooling (the exergy of the flow) to the sum of the exergies of the heats taken up by the tanks,

$$\eta = [i - i_0 - T_0 (s - s_0)] / \sum_{1}^{n} q_i \tau_i, \qquad (2)$$

is far from 100% even if there are many cold sources (Fig. 4). In Eq. (2), i and i_0 are the enthalpies, and s

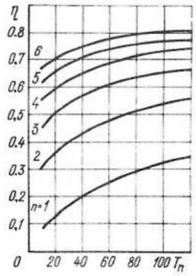

FIG. 4. Energetic efficieny of gas cooling by means of "cold" sources at constant temperature; T_n is the final temperature of the gas, and n is the number of cold sources.

and s_0 are the entropies of the gas in the initial and final states; T_0 is the temperature of the ambient medium; q_i is the heat extracted at the i-th temperature level, $\tau_i = (T_i - T_0)/T_i$ is the Carnot factor; and n is the number of cold sources. The low efficiency of the cooling process is a consequence of the fact that heat is extracted from the gas not continuously but in "portions," in tanks that have constant temperatures. Only if there were an infinitely large number of sources would the efficiency approach 100%. Figure 4 shows the efficiency in the case of ideal cold sources operating with reversible cycles. In real precooling systems, it is also necessary to take into account the losses in the refrigerating systems.

In addition, because of the restricted set of refrigerant boiling points, it is almost impossible to ensure that the temperatures of the cold sources form an optimal series. For example, for the precooling of a helium cryogenic system which has three cold sources—liquid nitrogen, nitrogen boiling in vacuum, and vacuum hydrogen—the real temperatures of the cold sources are approximately 78, 65, 15°K, while the optimal temperatures would be 110, 41, 15°K. Because of this alone the precooling efficiency will not be 0.5, as follows from Fig. 4 for $n = 3$ and $T_n = 15$°K, but about 30% lower. Much more effective precooling systems can be constructed by means of cascades of gas-expansion machines (see Figs. 3b and 3c). In a cycle with machines in parallel having the same efficiencies and pressure differences in each of them, the temperatures after the machines must satisfy the condition (1), and the flows form a geometrical series: $D_i = D_n q^{n-i}$, where D_n and q depend on the operating regime, the under-recuperation in the heat exchangers, the degree of expansion, and the efficiencies of the expansion machines (for more details, see Refs. 5 and 35).

If we ignore the hydraulic losses and the heat inflow from the ambient medium, which is usually small under cryogenic conditions, the remaining sources of loss in the cycle are the expansion machines, because the processes in them are not fully isentropic, and the heat exchangers, in which the heat exchange takes place at finite temperature differences. The first loss is technical and depends on the quality of the expansion machine, i.e., the extent to which the process of expansion of the gas approximates to an isentropic process; in principle, it can be reduced to an arbitrarily

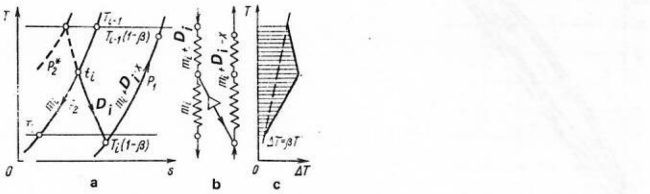

FIG. 5. Analysis of the loss and determination of the optimal pressure in a gas-expansion cycle: a) representation of part of the cycle in a $T-s$ diagram, b) scheme of part of a cycle, c) change in the temperature difference between the flows.

small amount by perfecting the expansion machine. But the losses in the heat exchangers are not merely technical. Even if we assume that we have at our disposal ideal heat exchangers with infinitely large heat-exchange surface for which the temperature difference at the boundaries of the section becomes equal to zero, there still remains a positive temperature difference at all remaining sections of the heat exchangers (Fig. 5) and there is still therefore an unavoidable loss of exergy associated with this temperature difference.

However, it is readily noted that if the pressure p_2 is chosen such that the expansion in a machine begins at the temperature T_{i-1}, the need for an upper (regenerative) heat exchanger disappears, and then, choosing the flows m_i, D_i, x appropriately, one can achieve an optimal linear distribution of the temperature difference between the flows in the heat exchanger (broken line in Fig. 5). In this case, the losses in the heat exchanger also become only technical and can in principle be eliminated, so that a precooling system constructed from a cascade of sections formed by ideal expansion machines and heat exchangers has maximal efficiency of 100%, in contrast to a precooling system with tanks. The pressure p_2^* is therefore optimal, enabling one under otherwise equal conditions (the same number of expansion machines and the same efficiencies of them, the same temperature range, and the same under-recuperation) to construct a cycle with the greatest efficiency.

It should be noted that the optimal pressure, or rather the optimal pressure ratio $(p_2/p_1)^*$, depends not only on the temperature range T_0/T_n of operation of the precooling stage, the number of expansion machines n, and their efficiency η (strongly) but also on the relative under-recuperation $\beta=\Delta T/T$ (weakly):

$$(p_2/p_1)^* = \left[\frac{1}{1-\beta}\sqrt[n]{T_0/T_n}\,\eta\Big/\left(1-\frac{1-\eta}{1-\beta}\sqrt[n]{T_0/T_n}\right)\right]^{\varkappa/(\varkappa-1)}, \quad (3)$$

where $\varkappa$ is the specific-heat ratio.

The optimal pressure ratio for precooling in a helium cryogenic system can be estimated approximately using Fig. 6. The graphs are plotted in accordance with Eq. (3) for $\eta=1$ and $\beta=0$. It can be seen that the precooling in helium cryogenic systems, which are characterized by $T_0/T_n=20-30$, cannot be optimized with respect to the pressure in the case of a single expansion machine, since excessively high pressures—thousands of atmospheres—would be needed. For p_2/p_1

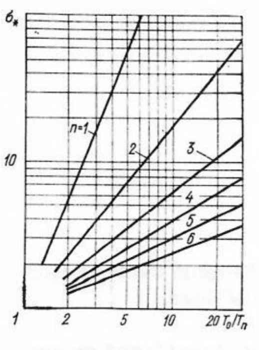

FIG. 6. Dependence of the optimal pressure ratio in the precooling stage of a helium cryogenic system on the range of working temperatures and on the number n of expansion machines.

10–20, values which are typical of helium systems, we obtain $n=3-4$. When allowance is made for the real efficiency of the expansion machines, the imperfect heat exchange, and additional losses, this number must be increased to 4–5. However, for precooling in helium systems one seldom uses more than two expansion machines, since it is assumed that an increase in the number of machines complicates the servicing and reduces the reliability of the system. If a preliminary cooling is introduced by means of liquid nitrogen, for example, then the expansion-machine part can be optimized with respect to the pressure by means of two expansion machines, but then the cycle becomes subject to the shortcoming described above of an arrangement with a cold source at constant temperature.

Optimization of the precooling with respect to the pressure or the number of expansion machines can have a significant effect, increasing the efficiency of this stage by 20–25%. Thus, for $T_0/T_n=25$, $\beta=\Delta T/T=0.05$, and $x=y$ the following results are obtained. Precooling with $p_2/p_1=20$ and three expansion machines with adiabatic efficiencies of 80% has an efficiency of 56%. For the optimal ratio $p_2/p_1=103.8$, the efficiency would be increased to 68%, i.e., by a factor 1.2. The same efficiency could be achieved with $p_2/p_1=23.9$ by increasing the number of expansion machines to four.

At the present time, helium cryogenic systems are usually equipped with turbine expansion machines. Since one can achieve only a relatively small pressure difference (usually $p_2/p_1<6$) in each of them, the expansion machines are connected in series (see Fig. 3c). For optimal pressure, when the intermediate regenerative heat exchangers become unnecessary, the expansion machines would have to be connected continuously. This means that the arrangement becomes equivalent to an arrangement with one expansion machine, which for large values of T_0/T_n is virtually impossible to optimize with respect to the pressure because of the need to use very high pressures. Therefore, if the other conditions are equal, such precooling will have a lower efficiency than one with expansion machines connected in parallel. For example, the precooling system of the industrial cryogenic system KGU-250, which includes a nitrogen stage and two expansion machines in series,

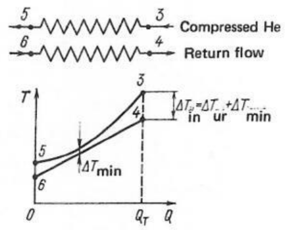

FIG. 7. The $Q-T$ diagram of heat exchange in the final cooling stage for variable specific heat of the flows.

has a rated efficiency in the regime considered above not exceeding 30%. Therefore, if we wish to mobilize all available resources for raising the efficiency of cryogenic systems, we must give preference to systems with expansion machines connected in parallel and choose the pressure or the number of expansion machines to be close to the optimal, for which reversibility of all the processes in the precooling stage becomes possible.

Final cooling

The final cooling stage can be considered independently only if the cryostating is achieved by immersing the object in liquid helium. In circulation systems, we shall show below that the final cooling must be optimized together with the cryostating with allowance for the parameters and characteristics of the object. The final cooling is done in the region of the critical point of the refrigerant, where its properties vary strongly with the temperature and the pressure. In particular, the specific heats of the forward and return flows in the heat exchanger are not the same and are not constant, and therefore even in the ideal case with an infinitely large heat-exchange surface it is impossible to obtain a zero temperature difference between the flows simultaneously in all sections of the heat exchanger (Fig. 7). In the given limiting case, only one quantity can be equal to zero—the minimal temperature difference $\Delta T_{\min}$. The under-recuperation ΔT_{ur} at the warm end of the heat exchanger corresponding to this limiting case depends on the temperatures of the flows, their pressures, and their relative qualities, and also on the type of expansion device. For the final cooling of a helium refrigerator with single throttling, this dependence is shown in Fig. 8. Efficient modern heat exchangers can be operated at temperature differences of order 0.1–0.3 °K, but this possiblity cannot be realized in the given case.

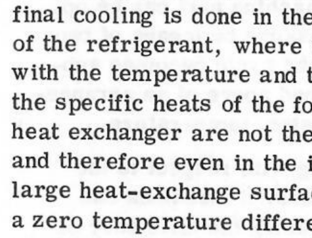

FIG. 8. Dependence of the minimal temperature difference at the warm end of the final cooling heat exchanger of a helium refrigerator with single throttling on the temperature of the forward flow at the entrance at different pressures.

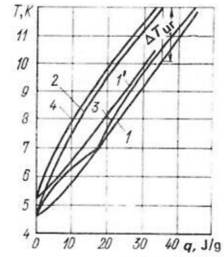

FIG. 9. The $Q-T$ diagram of heat-exchange processes in the final cooling stage of a helium refrigerator: 1) and 2) final cooling with single throttling; 1) and 3) final cooling with double throttling; 1) and 4) final cooling with vapor–liquid expansion machine; 1′) and 4) final cooling with vapor–liquid expansion machine and cryogenic forcing pump.

Several ways are known of improving the heat exchange and increasing the overall efficiency of the final cooling. The simplest of them is to introduce double throttling. In Fig. 9, we show the $Q-T$ diagrams of the heat-exchange processes in the case of single and double throttling. In the first case, the losses during heat exchange are large because of the unavoidable large temperature difference at the "warm" end of the heat exchanger. The efficiency of the corresponding stage does not exceed 26.2% (Table I). In the second case, the forward flow takes up almost all the cold of the return flow, and the efficiency is doubled.

If an ejector is used instead of a throttle valve, the cryostating level of the magnets can be reduced below 4.2 °K for the same expenditure (Fig. 10). In the evaporator 5, helium boils at reduced pressure. The useful thermal load Q_2 that the evaporator can take depends on the temperature; in the range from 4.2 to 3.5 °K it is 100% of the refrigerating capacity of the system. At lower temperatures, only part of the refrigerating capacity $Q = Q_1 + Q_2$ can be used in the evaporator. The other part Q_1 must be realized at a higher temperature level in the separator 3. At temperatures 1.6–1.8 °K, the useful refrigerating capacity in the evaporator is equal to zero.[6,7] An ejector can also be used to raise the pressure in the return flow of the refrigerator, which makes it possible to reduce by about 10% the amount of work needed to compress the helium in the compressor.

In principle, it was obvious that instead of a throttle valve one could use an expansion machine, which would raise the efficiency of the final cooling considerably. However, the development of a machine operating in

TABLE I. Exergetic efficiency of final cooling stages.

System	Entropy increment in heat exchanger	Entropy increment in expansion machine	$\Delta e_{\min}$	η_e
With single throttling	1.037	3.514	1849	0.262
With double throttling	0.058 + 0.347	2.14 + 0.798	2104	0.524
With expansion machine and η_{ad} = 0.8	1.168	0.738	1803	0.683
With expansion machine and η_{ad} = 0.8 with forcing pump and η_{ad} = 0.7	0.627	0.738 + 0.321	1734	0.708

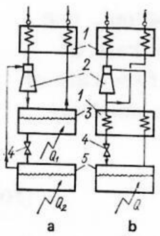

FIG. 10. Final cooling with ejectors; a) ejector used to lower the cryostating temperature, b) ejector used to compress the return flow; 1) heat exchanger, 2) ejector, 3) separator, 4) throttle valve, 5) evaporator.

the two-phase (vapor–liquid) region of states of the refrigerant encountered a kind of psychological inertia, namely, it was regarded as inadmissible to form liquid in the cylinder of a cryogenic reciprocating machine in view of known accidents with reciprocating machines of moderate cold due to hydraulic shock. A vapor–liquid reciprocating expansion machine for a hydrogen cryogenic system was developed for the first time in the Soviet Union at the Joint Institute for Nuclear Research in 1965.[8,9] In 1970, a vapor–liquid expansion machine for a helium system was constructed.[10] The capacity of the hydrogen liquifier was increased by 50–60%; in the helium system, an improvement of about 30% was achieved by using the expansion machine. The machines proved to be capable of stable and very efficient operation, the adiabatic efficiency exceeding 80%. The industrial helium cryogenic system KGU-400, which has a vapor–liquid expansion machine, is now produced in the Soviet Union.

In Ref. 11, it was recently proposed to introduce a cryogenic forcing pump into the final cooling to reduce the cryostating temperature level by creating a vacuum in the liquid-helium tank and (or) raising the pressure in the return flow (Fig. 11). For ideal expansion machine and ideal forcing pump, such an arrangement makes it possible to realize a Carnot cycle in the lower part of the final cooling. The use of the forcing pump makes it possible to reduce the unavoidable inherent losses in the heat exchanger when the temperature difference at the "cold" end of the heat exchanger is reduced. As a result, the cryostating level is reduced without additional expenditure of energy.

The efficiencies of the various types of final cooling are compared in Table I. The exergetic efficiency is

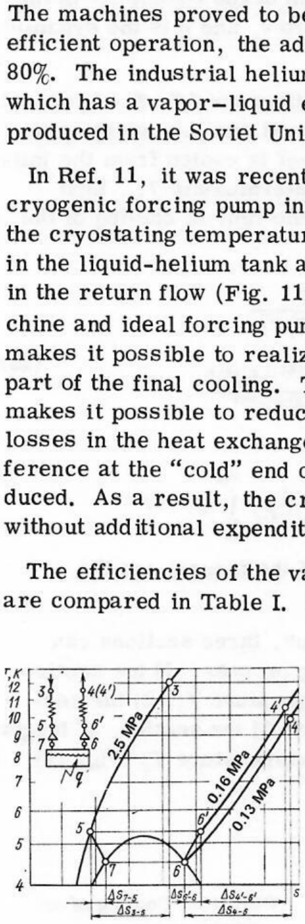

FIG. 11. Final cooling with vapor–liquid expansion machine and cryogenic forcing pump.

$$\eta_e = 1 - T_0 \Sigma \Delta s / \Delta e_{en},$$

where T_0 is the temperature of the ambient medium (300°K), Δs is the increment in the entropy, and $\Delta e_{en} = e_3 - e_1 = i_3 - i_1 - T_0(s_3 - s_1)$ is the difference between the exergies of the flows at the entrance to the final cooling stage. The calculation is made for the refrigerator regime ($G_3 = G_4$).

Cryostating system

The refrigerant obtained by the final cooling enters the cryostating system, where it takes heat from the magnets. Here, two limiting cryogenic regimes are possible:

1) the refrigerant is returned at low temperature to the cryogenic system for cooling of new portions of the refrigerant;

2) the refrigerant is extracted at low temperature in liquid or gaseous form from the system and is not returned to it.

If it is expedient to use only the heat of evaporation of the refrigerant in the cryostating system, the ratio of the refrigerating capacities of an ideal cryogenic system operating in these two regimes is

$$\chi = Q_R / Q_L = l_{\min} / r \mid \tau \mid, \tag{4}$$

where Q_R is the capacity in the first (refrigeration) regime, $Q_L = G_L r$ is the capacity in the liquifaction regime, G_L is the amount of refrigerant extracted in liquid form, $l_{\min} = i - i_0 - T_0(s - s_0)$ is the minimal work of liquifaction of the refrigerant, r is the heat of evaporation, and τ is the Carnot factor. The value of the coefficient is given in Table II for the most commonly used refrigerants. It can be seen that the refrigeration regime is much more advantageous, especially for helium systems.

For the cryostating of superconducting magnets of accelerators the cryogenic system is usually operated in a combined refrigeration–liquifaction regime, since besides the taking of the thermal load some of the cold helium is extracted in liquid or gaseous form through power leads to the temperature level of the ambient medium. For operation in such regimes, the characteristic of an ideal system, i.e., the connection between the refrigerating capacity Q and the amount G of extracted refrigerant, is expressed by linear equations:

$$Q = Q_R - \chi r G_L \tag{5}$$

in the case when liquid helium is extracted, and

TABLE II. Comparison of the refrigerating capacities of ideal cryogenic systems operating in the liquefaction and refrigeration regimes.

Refrigerant	Boiling point at atmospheric pressure, °K	Minimal work of liquefaction, J/g	Heat of evaporation at atmospheric pressure, J/g	$\lvert\tau\rvert$	χ
Nitrogen	77.35	771	199	2.59	1.34
Hydrogen (normal)	20.4	1200	450	13.7	1.96
Helium	4.2	6805	20.9	70.4	4.62

TABLE III. Performance of some industrial systems.

Designation	Q_R, W	G_L, liter/h	Q_L, W	χ
KhGU-150 (G-45) (Ref. 36)	150	45	31.5	4.76
KhGU-250 (Ref. 36):				
original variant	250	90	63	3.97
after optimization (Ref. 13)	340	104	72.8	4.67
GRO-500 (Ref. 14)	500	150	105	4.76
Linde (Ref. 15)	170	50	35	4.86
British Oxygen (Ref. 15)	175	50	35	5.00

$$Q = Q_R - (\chi - 1)\, rG_V \tag{6}$$

when gaseous helium is extracted.

The characteristics of real cryogenic systems are also usually nearly linear,[12,13] and the characteristic coefficient χ does not differ strongly from the theoretical value (Table III).

Three characteristic stages can be distinguished[16] in the bringing of the cryostating system into the working regime.

Cooling

We consider first the case when the magnet has a small extension and can be regarded as a concentrated mass having the same temperature T at all points. If we ignore the heat inflow from the ambient medium and the change in the mass of the refrigerant due to its accumulation in the cryostat of the magnet, then the following heat-balance equations will hold (Fig. 12):

$$Q = Gc_p(T_2 - T_1); \quad Q = \alpha F \overline{\Delta T},$$

where Q is the amount of heat extracted from the object per unit time, G is the flow rate of the refrigerant, c_p is its specific heat, α is the heat-transfer coefficient, F is the heat-transfer surface, and $\overline{\Delta T} = (T_2 - T_1)/[\ln(T - T_1)/(T - T_2)]$.

Hence, we find

$$Q = Gc_p(T - T_1)\,[1 - \exp(-\gamma)], \tag{7}$$

where $\gamma = \alpha F/(Gc_p)$.

We find the rate of cooling of the magnet from the equation

$$Q\,dt = -McdT, \tag{8}$$

where M is the massof the magnet, c is its specific heat, and t is the time. From (7) and (8), we obtain

$$dT/dt = \dot{T} = -(Gc_p/Mc)\,(T - T_1)\,[1 - \exp(-\gamma)]. \tag{9}$$

The construction of the magnet contains parts made from materials with very different thermophysical properties such as the thermal conductivity, the coefficient of linear expansion, and the specific heat. To avoid inadmissible deformations of the magnet, one usually specifies a maximally permissible rate of cooling: $\dot{T} = 5$–$10\,°$K/h. In this case, the flow rate of the refrigerant during the cooling process must not exceed

$$G \leqslant |\dot{T}|\, M\, \frac{c}{c_p}\, \frac{1}{(T - T_1)\,[1 - \exp(-\gamma)]}. \tag{10}$$

Usually, $\exp(-\gamma)$ is small compared with unity, and it can therefore be ignored.

In practice, it is convenient to carry out the cooling process maintaining a constant pressure of the refrigerant at the entrance to the cryostat of the magnet. Using the hydrodynamic dependences

$$\Delta p = \lambda \frac{G^2 L}{2\rho S^2 d}; \quad \lambda = \frac{0.3164}{\sqrt[4]{\mathrm{Re}}}; \quad \mathrm{Re} = \frac{Gd}{S\eta},$$

and bearing in mind that for helium the viscosity is $\eta = 5.023T^{0.647}$, we readily obtain the dependence of the helium flow rate on the temperature for this case:

$$G = G_w (T_w/T)^{0.664}. \tag{11}$$

Here, $\Delta p = p_{en} - p_{ex}$ is the difference between the pressures of the refrigerant as it enters and leaves the magnet, λ is the coefficient of friction, G is the flow rate of refrigerant at temperature T, G_w is the flow rate of the refrigerant at the end of the cooling process at working temperature T_w, ρ is the mean density of the refrigerant, L is the length of the channel, S is the area, d is the hydraulic diameter, and η is the dynamic viscosity of helium.

If a constant temperature difference $\delta T = T - T_1 = \mathrm{const}$ is maintained between the magnet and the refrigerant at the entrance when the magnet is cooled from the initial temperature T_{in} to some intermediate T_m, then from (9) and (11) we can find the time of cooling of the magnet [we ignore $\exp(-\gamma)$]:

$$t = -\int_{T_{in}}^{T_m} \frac{Mc\,dT}{Gc_p\delta T} = -\frac{M\bar{c}}{G_w \bar{c}_p \delta T T_w^{0,664}} \times \int_{T_{in}}^{T_m} T^{0,664}\, dT = 0.601\, \frac{M\bar{c}}{G_w \bar{c}_p}\, \frac{T_{in}^{1,664} - T_m^{1,664}}{\delta T T_p^{0,664}}. \tag{12}$$

Here,

$$\bar{c} = \frac{1}{T_{in} - T_m} \int_{T_{in}}^{T_m} c(T)\, dT; \quad \bar{c}_p = \frac{1}{T_{in} - T_m} \int_{T_{in}}^{T_m} c_p(T)\, dT$$

are the mean specific heats of the magnet and the refrigerant.

In extended magnetic systems, three sections can be distinguished in the cooling process: 1) the section, of length x, cooled to the temperature T_1; 2) the section, of length l, being cooled; 3) the section, of length $L - (x + l)$, having the initial temperature T_2 (Fig. 13).

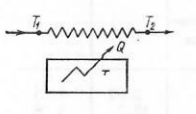

FIG. 12. Schematic representation of cooling of a concentrated mass with infinite thermal conductivity.

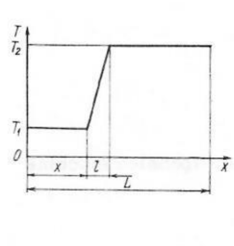

FIG. 13. Cooling of an extended mass.

Usually, $l \ll L$, where L is the length of the magnetic system. To determine the cooling time in this case, we use the method presented in Ref. 17, and we take $l = 0$.

In the absence of heat inflow from the ambient medium the heat-balance equation is

$$mc\,(T_2 - T_1)\,dx = Gc_p\,(T_2 - T_1)\,dt.$$

Here, $m = M/L$ is the mass per unit length. The cooling time is then

$$t = m\frac{\bar{c}}{\bar{c}_p}\int_0^L \frac{dx}{G}. \tag{13}$$

Suppose that during the cooling process the pressures at the entrance and exit to the system are kept constant, i.e.,

$$\Delta p = p_1 - p_2 = (p_1 - p_*) + (p_* - p_2) = \text{const}, \tag{14}$$

where p_* is the mean pressure over the section l. Let G_f be the refrigerant flow rate at the end of the cooling process, when the temperature of the complete magnetic system has become T_1. During this period,

$$\Delta p = p_1 - p_2 = \frac{G_f^2}{2\rho S^2}\lambda_1\frac{L}{d},$$

where $\rho = (p_1 + p_2)/2RT_1$. Hence, we obtain

$$p_1^2 - p_2^2 = \frac{G_f^2}{S^2 d}\lambda_1 L R T_1. \tag{15}$$

Similarly, for the current instant of time

$$p_1^2 - p_*^2 = \frac{G^2}{S^2}\lambda_1\frac{x}{d}RT_1;$$

$$p_*^2 - p_2^2 = \frac{G^2}{S^2}\lambda_2\frac{(L-x)}{d}RT_2;$$

$$p_1^2 - p_2^2 = \frac{G^2}{S^2 d}\left[\lambda_1 T_1 x + \lambda_2 T_2 (L-x)\right]R. \tag{16}$$

Using the hydrodynamic relations given above in (15) and (16), we obtain

$$G = G_f\left[\frac{L}{\xi L - x(\xi - 1)}\right]^{4/7}, \tag{17}$$

where

$$\xi = (T_2/T_1)\sqrt[4]{\eta_2/\eta_1}.$$

Substituting the expression for G in Eq. (13), and integrating, we find

$$t = \frac{m\bar{c}}{G\bar{c}_p L^{4/7}}\int_0^L \left[\xi L - x(\xi - 1)\right]^{4/7} J\,dx,$$

and finally

$$t = \frac{7}{11}\frac{M\bar{c}}{G_f\bar{c}_p(\xi - 1)}. \tag{18}$$

For helium

$$\xi = (T_2/T_1)^{1.162}.$$

Accumulation of liquid helium

We consider this process for the example of the operation of a cryogenic system with a magnet having cooled power leads. In this period, the cryogenic system compensates the heat inflow from the ambient medium and produces liquid helium, which must be accumulated in the cryostats of the magnets as rapidly as possible.

To determine the optimal regime, we must solve simultaneously the characteristic $Q = f(G)$ of the system

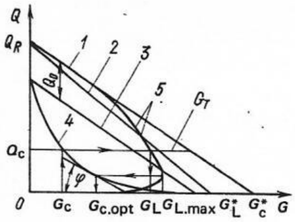

FIG. 14. Characteristic of refrigerator–magnet system during accumulation of liquid helium: 1) $Q = Q_R - \chi r G_L$, 2) $Q = Q_R - (\chi - 1) r G_c$, 3) $Q = Q_R - Q_0 - (\chi - 1) r G_c$, 4) $Q_c = f(G_c)_{I=0}$, 5) $G_L = f(G_c) = G_L^* - (Q_0 + Q_c)/\chi r - G_c(\chi - 1)/\chi$.

and the characteristic $Q_c = f(G_c)$ of the power lead; here, Q_c is the heat inflow through the power lead, and G_c is the flow rate of the gas cooling the power lead. We shall assume that the heat inflow Q_0 from the ambient medium at this period is constant. As a result of the solution, we find the dependence of the rate of accumulation of liquid helium on the amount of gas supplied to the power leads:

$$G_L = G_L^* - (Q_0 + Q_c)/(\chi r) - G_c(\chi - 1)/\chi. \tag{19}$$

Figure 14 shows the construction of the auxiliary curve 4, by means of which the required quantities can be determined. It can be seen that there is an optimal flow through the power leads for which the rate of accumulation of the liquid helium is maximal. From Eq. (19), we find

$$(\partial Q_c/\partial G_c)_{\text{opt}} = -r(\chi - 1).$$

We can find $G_{\text{c.opt}}$ by drawing the tangent to the curve $Q_c = f(G_c)$ at the angle $\varphi = \tan^{-1} r(\chi - 1)$ (see Fig. 14).

Cryostating

In this regime, the cryogenic system must compensate the heat inflow from the ambient medium and the heat release in the magnet due to the cyclic variation of the current that feeds the magnet. In addition, some of the cold helium is extracted through the power leads. The characteristic of the system in this regime is described by the equation

$$Q = Q_R - (\chi - 1)\,rG_c.$$

where G_c is the flow rate of the gas through the power leads.

We find the characteristic of the system by solving this equation simultaneously with the equation of the characteristic of the power lead for given current I:

$$Q_c = f(G_c)_{I = \text{const}},$$

and as a result we obtain the dependence between the admissible value of the dynamical heat release Q_d and the flow rate of the gas G_c that cools the power lead:

$$Q_d = Q_R - Q_0 - Q_c = -(\chi - 1)\,rG_c. \tag{20}$$

The construction of the characteristic of the system is shown in Fig. 15. For a certain flow rate of the gas that cools the power leads the admissible dynamical thermal load reaches a maximum. This regime is optimal for the given refrigerator–magnet system.

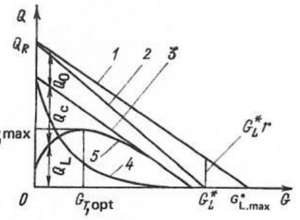

FIG. 15. Characteristic of refrigerator–magnet system during cryostating: 1) $Q = Q_R - \chi r G_L$, 2) $Q = Q_R - (\chi - 1) r G_c$, 3) $Q = Q_R - Q_0 - (\chi - 1) r G_c$, 4) $Q_c = f(G_c)_{I=I_1}$, 5) $Q_d = Q_R - Q_c - (\chi - 1) r G_c$.

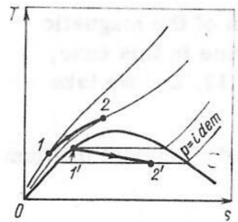

FIG. 16. Changes of state of the refrigerant when heat is taken from the cryostated object: 1)–2) circulation of single-phase flow, 1')–2') circulation of vapor–liquid flow.

2. CIRCULATION OF HELIUM IN SUPERCONDUCTING MAGNETIC SYSTEMS OF ACCELERATORS

The traditional way of cryostating superconducting magnets is by immersion in boiling helium. However, in extended systems a large pressure difference develops between the entrance and exit of the refrigerant, and this makes it impossible to maintain an equal level of liquid helium at different points of the cryostat. Therefore, this method can be used only for compact systems such as, for example, a superconducting cyclotron. In the overwhelming majority of the cases, a different method is used for the cryostating of superconducting magnets of accelerators; in it, heat is removed by circulating the refrigerant through channels placed within or around the coils. A further advantage of this method is that it reduces the amount of helium used to fill the system and, accordingly, decreases the danger of a rise in the helium pressure on transition of a magnet to the normal state.

The method gives rise to a number of problems associated with the construction of simple and reliable systems for circulating the helium, determining the optimal internal diameters and lengths of the channels for the refrigerant, and so forth. The starting regimes of the circulation systems require further study. Below, we consider some aspects of the use of the circulation method.

Classification and comparison of circulation systems

The most important criterion forming the basis for the classification of circulation systems in the phase state of the refrigerant. Bearing this in mind, we consider two forms of cryostating systems. In one form, supply of heat to the circulating helium does not cause a phase transition, while in the other the process takes place in the presence of a boiling vapor–liquid flow (Fig. 16). At present, both forms of system are realized in practice. However, attention has frequently been drawn to the danger that when a two-phase vapor–liquid mixture is used the flow regime could become unstable, the instability being characterized by a pulsating variation in the flow rate of the refrigerant, its pressure, and temperature or the blocking of some of the parallel channels by "vapor stoppers." On the other hand, there are various reasons that make the use of a two-phase flow preferable. The most important is that a lower cryostating temperature can be achieved. In addition, it should be noted that local releases of heat, which can occur, for example, when accelerated particles interact with elements of the magnetic system, are relatively less dangerous for a vapor–liquid flow, since the temperature does not rise until all the liquid has been evaporated. Because a comparatively large amount of heat can be taken up by a vapor–liquid flow, it is possible to pump less helium through the magnetic system. This reduces the energy output associated with circulating the refrigerant, and in some cases makes it possible to do without special circulation devices.

An arrangement in which this is achieved is shown in Fig. 17. The amount G_0 of circulating refrigerant is equal to the flow rate G_w in the forward flow in the final cooling stage of the refrigerator. In this case, the circulation factor $z = G_0/G_w$ is equal to unity. Such systems can be classified as *systems with single circulation*. If some additional means (reciprocating pump, jet apparatus, etc.) is used to circulate the refrigerant, one can obtain $z > 1$. Systems in which $z > 1$ can be called *systems with multiple circulation*. Examples of such systems are shown in Figs. 18–21.

Let us consider the reasons for using the latter systems.

It is well known that the refrigerating capacity of helium refrigerators per unit flow rate G_w (see Fig. 17) is $q = 10$–20 kJ/kg. The lower limit of q corresponds to refrigerators in which the process of cooling of the forward flow ends with throttling. Refrigerators in which the cooling process of the forward flow ends with an expansion machine have the largest values of q. In the case of single circulation of a single-phase flow, the refrigerant raises its temperature in passing through the channels of the cryostated object by ΔT

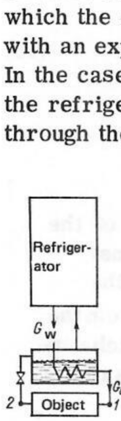

FIG. 17. Scheme with single circulation of refrigerant.

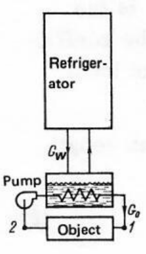

FIG. 18. Scheme with multiple circulation of a single-phase refrigerant by a mechanical pump.

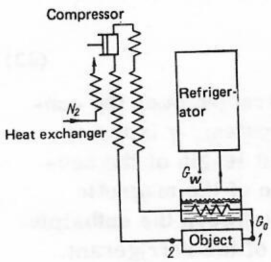

FIG. 19. Scheme with multiple circulation of a single-phase refrigerant by a compressor working at the temperature of the ambient medium.

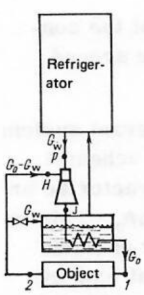

FIG. 20. Scheme with multiple circulation of a single-phase refrigerant with jet device.

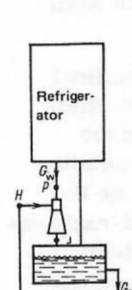

FIG. 21. Scheme with multiple circulation of vapor–liquid flow with jet device.

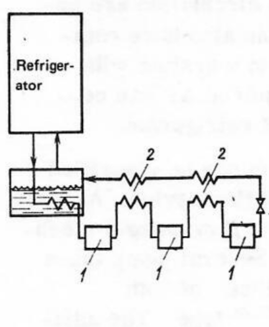

FIG. 22. Scheme with single circulation and multiple cooling of the refrigerant: 1) cryostated object, 2) coolers.

$=q/c_{1-2}$, where c_{1-2} is the mean specific heat in the process 1–2. Thus, for initial parameters $p_1 = 0.5$ MPa and $T_1 = 4.5$ °K, the increase in the temperature is $\Delta T = 1.7–2.5$ °K. This increase in the temperature of the refrigerant and, accordingly, the temperature of the superconductor gives rise to a significant reduction in the critical current density and an increased expenditure of the superconductor, and in the majority of cases is unacceptable. It is possible to reduce ΔT by introducing multiple circulation. In this case, $\Delta T = q/zc_{1-2}$. The value of ΔT for $z = 5–10$ is reduced to a few tenths of a degree.

Multiple circulation of a two-phase mixture can be used if it is necessary to reduce the vapor content of the refrigerant when it leaves the object. This may be necessary, for example, to achieve stable operation[18] or to improve the heat exchange by ensuring operation at vapor contents for which there is no "drying" of the liquid film on the channel walls.[19] In the case of single circulation, the change in the vapor content is $\Delta x = q/r = 0.5–1$ (r is the heat of evaporation of helium), while for $z > 1$ the vapor content is inversely proportional to the circulation factor: $\Delta x = q/rz$.

The required values of ΔT and Δx can also be obtained in the case of single circulation by introducing *multiple cooling or separation of the refrigerant*, as shown, for example, in Figs. 22 and 23. In these cases, means of circulation are not required but the construction of the cryostated object and the communications between it and the refrigerator are greatly complicated. If simple and reliable means of circulation are avail-

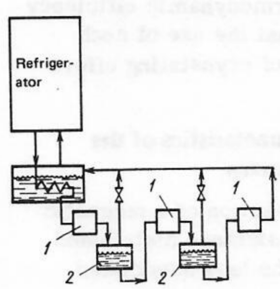

FIG. 23. Scheme with single circulation and multiple separation of the refrigerant: 1) cryostated object, 2) seperators.

able, arrangements with multiple circulation are undoubtedly more attractive. One can also have compromise solutions for which $z > 1$ in a system with multiple cooling, and then the required ΔT can be achieved with a smaller amount of refrigerant.

Systems with multiple circulation can be classified according to the type of the circulation device. A circulation scheme for a one-phase flow using a mechanical pump is shown in Fig. 18. Several pump types have been developed for liquid helium, of both reciprocating[20,21] and centrigugal[22,23] type. The adiabatic efficiency of such pumps is 20–75%, the reciprocating pumps having the higher efficiencies.

A compressor–heat-exchanger scheme was considered in Ref. 24 (see Fig. 19). In it, the circulation is realized by a compressor operating at the temperature of the ambient medium, and before the compressed gas is brought to the cryostated object it is cooled in a regenerative heat exchanger and cooler immersed in boiling helium. This arrangement has serious shortcomings compared with pump arrangements. The point is that in the case of multiple circulation the output of the circulation compressor must be several times greater than that of a refrigerator compressor. The working surface of the regenerative heat exchanger must also be several times greater than the surface of refrigerator heat exchangers. Such complications can hardly be accepted. Moreover, such a system uses significantly more energy than pump schemes.[24]

Multiple circulation of the refrigerant can be achieved by means of a *jet apparatus*.[25] A scheme with circulation of a single-phase refrigerant is shown in Fig. 20. Compressed gas in the amount G_w is accelerated after it leaves the refrigerator in the nozzle of a jet apparatus and in its further motion carries along with it an injected flow in the amount $G_0 - G_w$. The mixed flow G_0 reduces its temperature in a cooler immersed in boiling helium and is fed to the object. When the flow leaves the object, it is divided into two parts, one of which is directed to the jet apparatus, and the other, equal to G_w, is passed through a valve into a collector. A scheme with multiple circulation of a vapor–liquid flow is shown in Fig. 21.

In contrast to other circulation devices, the jet arrangement is exceptionally simple and can be prepared rapidly and with minimal expenditure. Such systems satisfy the most stringent reliability requirements. If it is borne in mind that their use instead of mechanical pumps hardly lowers the thermodynamic efficiency of the system,[26] it can be seen that the use of such schemes in circulation systems of cryostating offers wide possibilities.

Requirements on the hydraulic characteristics of the cryostat channels of a magnetic systems

The hydraulic resistance of a section of a magnetic system is made up of a linear resistance distributed over the complete length L and the loss associated with local resistances:

$$\Delta p = \lambda \frac{\rho w^2}{2} \frac{L}{d} + \Sigma\zeta \frac{\rho w^2}{2},$$

where ρ is the density of the refrigerant, w is the velocity, d is the hydraulic diameter, λ is the coefficient of friction, and ζ is the coefficient of the local resistances.

The hydraulic resistance of a section of unit length is

$$\Delta p_{\mathrm{un}} = a\rho w^2/2, \tag{21}$$

where

$$a = \lambda/d + \Sigma\zeta/L.$$

The flow velocity is related to the flow rate G and the cross-section area f by the continuity equation

$$G = \rho f w. \tag{22}$$

On the other hand, the flow rate can be determined from the relation

$$G = Q/\Delta i = qL/\Delta i, \tag{23}$$

where Q is the amount of heat extracted over the considered section of the magnetic system, q is the amount of heat extracted from unit length of the section, L is the length of the section of the magnetic system, and Δi is the difference between the enthalpies at the points of entrance and exit of the refrigerant.

Using (22) and (23), we obtain from (21)

$$\Delta p = \frac{a}{2} \frac{q^2 L^3}{f^2} \frac{1}{\rho \Delta i}. \tag{24}$$

The first part of the product,

$$k_m = \frac{a}{2} \frac{q^2 L^3}{f^2},$$

is determined by the size and construction of the considered element of the magnetic system; the second, by the parameters of the refrigerant.

In Ref. 26, in which a single-phase refrigerant system was analyzed in order to reduce variants of schemes to comparable forms, it was decided to characterize an object by the *reduced hydraulic resistance* Δp_0. This is defined as the hydraulic resistance to a flow of liquid helium heated in an object with given thermal load by 1°K and having temperature 5°K and pressure 0.5 MPa at the exit. It is readily seen that Δp_0 is proportional to k_m, and the coefficient of proportionality is determined by the parameters fixed in the definition of the reduced hydraulic resistance. In what follows, we shall use the results given in Ref. 26, but instead of characterizing the cryostated object by Δp_0 we shall use k_m.

Irreversibility losses also exist when all the final cooling elements are ideal, i.e., the adiabatic efficiencies of the machines are equal to unity and the working surfaces of the heat exchangers are infinite. Such losses are said to be inherent. They arise because of the unequal specific heats of the heat-exchanging elements, so that even if there is a zero temperature difference in one of the sections of the heat exchanger, this is not as a rule true in other sections.

The estimates made in Ref. 26 showed that the inherent losses in the final refrigeration can be very

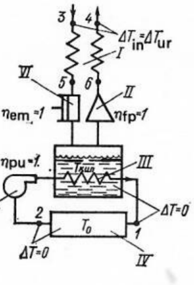

FIG. 24. Idealized final cooling stage: I is the heat exchanger, II the cryogenic forcing pump, III the cooler, IV the cryostated object, V the mechanical pump, and VI an expansion machine; η_{em}, η_{fp}, and η_{pu} are the adiabatic efficiencies of the expansion machine, forcing pump, and pump, respectively.

appreciable, so that it is expedient to identify these losses, which determine the limit to which the exergetic efficiency of final cooling can be raised by technical improvements in its elements. For this, we have considered a scheme in which there are only inherent losses, and the technical losses are due solely to the hydraulic resistance of the object. The highest efficiency is achieved in the scheme shown in Fig. 24. It is a combination of a Carnot cycle and regenerative heat exchange in a heat exchanger. The final cooling shown in Fig. 24 can be called idealized. It is characterized by the fact that the minimal temperature differences associated with the heat exchange in all elements of the stage are equal to zero, and the processes in all the machines are isentropic.

The efficiency η_{FC}^{id} of an idealized final cooling stage is shown in Fig. 25 as a function of k_m and the cryostating temperature T_0. The results of optimization are shown, i.e., the temperature T_b of boiling of the liquid in the collector for each pair of k_m and T_0 values is chosen to ensure the highest efficiency of the final cooling stage. It can be seen from the figure that even if all the elements of the arrangement are technically ideal the final cooling efficiency does not exceed 65–85% because of the inherent losses. Once k_m is raised above 10^{13}–10^{14} W^2/m^4, the value of η_{FC}^{id} falls sharply. Therefore, for objects with k_m above these values, no scheme can ensure a high thermodynamic efficiency of the cryostating system.

In contrast to idealized final cooling, in a real process (Fig. 26) the minimal temperature differences

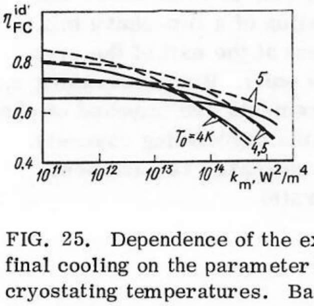

FIG. 25. Dependence of the exergetic efficiency of idealized final cooling on the parameter k_m of the object for different cryostating temperatures. Basic data: $p_6 = 0.13$ MPa; continuous curve, $T_3 = 15$ °K, $p_3 = 2.5$ MPa; broken curve, $T_3 = 10$ °K, $p_3 = 1.5$ MPa.

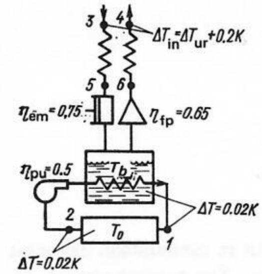

FIG. 26. Real final cooling stage with cryogenic forcing pump and mechanical pump; ΔT_{qr} is the temperature difference at the warm end of the heat exchanger corresponding to zero difference in one of the sections of the apparatus.

are nonzero, and the processes in the expansion machines, the cryogenic forcing pump, and the mechanical pump are not isentropic. Therefore, in all these elements there are additional losses associated with the technical imperfections of the machines and elements used in the process. Comparing the efficiencies of the idealized and real processes (Fig. 27), we can separate the technical and inherent losses. The technical losses are divided into two groups: $\Sigma d_{tech.1}$ and $\Sigma d_{tech.2}$. The former is due to technical imperfection of the machines and elements used in the final cooling, and the latter is due to the hydraulic resistance of the object. Accordingly, there are two ways of increasing the final cooling efficiency. One way, the tradiational one, is to improve the machines and the elements of the refrigerator; the other, which is peculiar to circulating systems, is to find a rational construction of the cryostated object itself.

It can be seen from the figure that a high thermodynamic efficiency of the final cooling cannot be achieved solely by decreasing $\Sigma d_{tech.1}$. This can only raise the efficiency to η_{FC}^{id}. The cryostated object must itself be correctly designed in order to reduce to a minimum $\Sigma d_{tech.2}$. Then η_{FC} approaches the limit governed exclusively by the inherent losses. The requirements on the object corresponding to these conditions can be formulated by means of k_m. For example, in the considered case it must not exceed 10^{13} W^2/m^4.

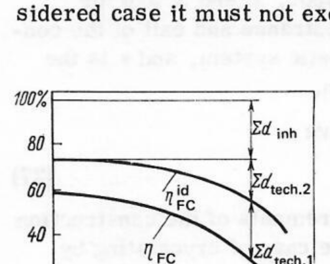

FIG. 27. Relationship between the technical and inherent losses (the schemes of Figs. 24 and 26); $\Sigma d_{tech.1}$ are the technical losses due to imperfection of the machines and the apparatus of the final cooling stage; $\Sigma d_{tech.2}$ are the technical losses due to the hydraulic resistance of the cryostated object; Σd_{inh} are the inherent losses. Basic data: $T_0 = 4.5$ °K, $p_3 = 2.5$ MPa, $T_3 = 15$ °K, $p_3 = 0.13$ MPa.

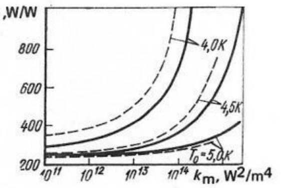

FIG. 28. Specific energy requirements in circulation cryostat systems with a cryogenic forcing pump. The continuous curves correspond to the scheme with a mechanical pump, the broken curves to schemes with a jet pump.

The specific energy expenditure in the circulating system of the cryostat can be calculated in accordance with

$$\varepsilon = |1 - T_{am}/T_0|/\eta_{PC}\,\eta_{FC},$$

where T_{am} is the temperature of the ambient medium.

The results of such calculations for parameters $T = 15\,°K$ and $p_3 = 2.5$ MPa at the entrance to the final cooling stage are given in Fig. 28. In accordance with Ref. 26, the precooling efficiency was taken equal to 0.42. In the same figure, we have plotted similar graphs for a scheme with a jet. It follows from the figure that for properly designed objects the specific energy losses in systems with jet pumps approach those in more complicated and less reliable systems with mechanical pumps. In the interval $T_0 = 4.5$–$5\,°K$, they are almost equal.

Since the density and velocity vary appreciably during the motion of a vapor–liquid flow, the expression (24) for a two-phase medium is valid only in differential form:

$$d(\Delta p) = a\frac{\rho w^2}{2}\,dL. \tag{25}$$

Assuming homogeneity of the moving medium after integration of (25), we can obtain

$$\Delta p = \frac{a}{2}\,\frac{q^2L^3}{f^2}\,\frac{v_l}{r^2}\,\frac{1+(1/2)(v_v/v_l-1)(x_2+x_1)}{(x_2-x_1)^2}, \tag{26}$$

where v_l and v_v are, respectively, the specific volumes of the liquid and the vapor, x_1 and x_2 are the mass vapor contents at the entrance and exit of the considered element of the magnetic system, and r is the heat of evaporation of helium.

For $x_1 = 0$ and $x_2 = 1$, we have

$$\Delta p = 0.5k_m(v_l + v_v)/r^2. \tag{27}$$

We now compare the requirements of the construction of the magnetic system in the case of cryostating by single- and two-phase flows.

Suppose the object is cryostated by the single-phase flow in accordance with the scheme in Fig. 26, and by the vapor–liquid flow in accordance with Fig. 29. In both cases, the cryostating temperature is $T_0 = 4.5\,°K$, and the energy expenditure per unit of employed "cold" is $\varepsilon = 280$ W/W. Under these conditions, it follows from Fig. 28 that in the case of a system with circulation of a single-phase flow it is necessary to ensure $k'_m = 9.4 \times 10^{12}$ W²/m⁴.

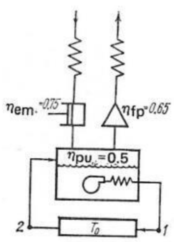

FIG. 29. Scheme for cryostating with vapor–liquid helium flow.

The left-hand branches of the graphs in Fig. 28 approach asymptotically the values of the specific energy requirements in the case of zero requirements for circulation. In these cases, the optimal circulation factor increases strongly, and the boiling point of the liquid in the collector approaches the temperature of the cryostated object: $T_b \to T_0$.

Suppose that in a system with two-phase helium the fraction of the energy requirement associated with circulation is negligibly small. Then for $T_b = 4\,°K$ the specific energy requirements are also 280 W (see Fig. 28). The hydraulic resistance is determined by the chosen boundary temperatures $T_1 = 4.5\,°K$ and $T_2 = 4.0\,°K$:

$$\Delta p = p_1 - p_2 = 0.130 - 0.081 = 0.049 \text{ MPa}.$$

Then in accordance with (27) the required value of k_m for the two-phase vapor–liquid flow is

$$k'_m = \Delta p r^2/[0.5\,(v_l + v_v)]$$
$$= 0.049\cdot 10^6\cdot(20.9\cdot 10^3)^2/[0.5\,(0.0082 + 0.0769)] = 503\cdot 10^{12}\ \text{W}^2/\text{m}^4.$$

The fraction of the energy requirement associated with circulation can be estimated as the ratio of the work of the pump to the refrigerating capacity: $v_l\Delta p/\eta_p r = 0.0082 \times 0.049 \times 10^6/(0.5 \times 20.9 \times 10^3) = 0.038$, i.e., as we assumed earlier, this can be ignored.

The ratio $k''_m/k'_m = 52.9$ shows that cryostating by a two-phase flow is more advantageous. In this case, it is possible to increase by $\sqrt{k''_m/k'_m} = 7.3$ the specific thermal load q or, for the same value of q, decrease the area f of the internal sections of the channels. Under otherwise equal conditions; transition from single- to two-phase cooling makes it possible to increase the length of the cooled section by a factor $\sqrt[3]{k''_m/k'_m} = 3.75$.

As we have already pointed out, in some cases one may require multiple circulation of a two-phase mixture in which the vapor content at the exit of the cryostated system does not reach unity. With decreasing x_2, increasingly stringent requirements are imposed on the hydraulic characteristics of the cryostating channels. In accordance with (26), the increased requirements can be characterized by the ratio

$$k_m(x_2)/k_m(1) = \psi(x_2)/\psi(1),$$

where

$$\psi(x_2) = \left[1 + \frac{1}{2}\left(\frac{v_v}{v_l} - 1\right)(x_2 + x_1)\right]\Big/(x_2 - x_1)^2.$$

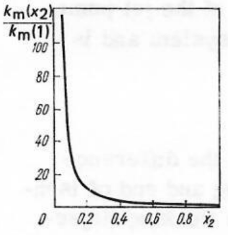

FIG. 30. Change in the parameter k_m when the fraction of vapor at the exit from the object is reduced.

The dependence $k_m(x_2)/k_m(1)$ for $x_1 = 0$ is plotted in Fig. 30. It can be seen that once $x_2 < 0.4$ the requirements on the hydraulic characteristics appreciably increase, though they become as high as for a single-phase flow $[k_m(x_2)/k_m(1) = 52.9]$ only when $x_2 < 0.1$.

Thus, the efficiency of a system that includes a refrigerator and an object cryostated by helium circulation is largely determined by the hydraulic characteristics of the object. Only if the parameter k_m is made to satisfy definite requirements can a high thermodynamic efficiency of the system be achieved. The transition from single-phase cooling systems to vapor–liquid systems makes it possible to reduce these requirements and, therefore, to increase the thermal load per unit length of the system and to decrease the internal cross section and increase the length of the channels carrying the refrigerant.

Calculation and testing of jet circulation systems

As we have already noted, it is for various reasons preferable to use jet arrangements to circulate the helium in cryostated objects rather than mechanical pumps. Therefore, we must discuss the principles of operation of such systems, the methods of calculation, and experimental testing.

A jet arrangement is shown in Fig. 31. Compressed gas in the amount G_1 is accelerated in a jet nozzle and in its subsequent motion carries along with it the injected flow G_2. In the mixing chamber, the flow velocities are equalized, and then in the diffuser there is a decrease in the velocity accompanied by an increase in the pressure of the mixed medium to the stagnation pressure.

If we ignore the interaction of the working flow and the injected flow as they approach the entrance section b of the cylindrical mixing chamber, i.e., we assume that the sections a and b coincide, for the interval between sections b and c we can write the momentum equation in the form[27]

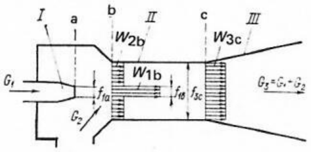

FIG. 31. Jet pump: I is the nozzle, II the mixing chamber, and III the diffuser.

$$\varphi_2 (G_1 w_{1b} + G_2 w_{2b}) - (G_1 + G_2) w_{3c} = (p_{3c} - p_{1b}) f_{1b} + (p_{3c} - p_{2b}) f_{2b}, \tag{28}$$

where φ_2 is the velocity coefficient that takes into account the loss during the flow in the mixing chamber, G is the mass flow rate, w is the velocity, p is the pressure, and f is the area of the flow section. In the system of indices, a number denotes the flow and a letter denotes the section of the jet system. The subscript 1 identifies the working flow, 2 the injected flow, and 3 the mixed flow.

Under the condition that sections a and b are coincident,

$$w_{1b} = w_{1a}; \quad p_{1b} = p_{1a}; \quad f_{1b} = f_{1a}.$$

If we assume that the injected and the mixed medium are incompressible, then, determining the velocities by means of Bernoulli's equation,

$$w_{2b} = \varphi_4 \sqrt{2 (p_2 - p_{2b})/\rho_2};$$
$$w_{3c} = (1/\varphi_3) \sqrt{2 (p_3 - p_{3c})/\rho_3},$$

using the continuity equations

$$G_1 = \rho_1 w_{1a} f_{1a}; \quad G_2 = \rho_2 w_{2b} f_{2b}; \quad G_1 + G_2 = \rho_3 w_{3c} f_{3c} \tag{29}$$

and remembering that $f_{2b} = f_{3c} - f_{1a}$, we can obtain from (28) an expression for the head $\Delta p = p_3 - p_2$ of the jet system, this being referred to as the *characteristic equation*[28]:

$$\Delta p = \frac{w_{1a}^2}{v_{1a}} \frac{f_{1a}}{f_{3c}} \Big[\frac{v_{1a}}{w_{1a}^2} (p_{1a} - p_2) + \varphi_2 + u^2 \frac{f_{1a}}{f_{2b}} \frac{v_2}{v_{1a}} \Big(\varphi_2 - \frac{0.5}{\varphi_4^2} \Big) - (1 + u)^2 \frac{f_{1a}}{f_{3c}} \frac{v_3}{v_{1a}} (1 - 0.5\varphi_3^2) \Big], \tag{30}$$

where φ_3 and φ_4 are the velocity coefficients that take into account the losses at the entrance section of the mixing chamber and in the diffuser, $u = G_2/G_1$ is the injection coefficient, and $v = 1/\rho$ is the specific volume.

In the regime without underexpansion, and also in all cases of subsonic exhausting from the nozzle, $p_{1a} = p_2$. Then the characteristic equation simplifies to

$$\Delta p = \frac{w_{1a}^2}{v_{1a}} \frac{f_{1a}}{f_{3c}} \Big[\varphi_2 + u^2 \frac{f_{1a}}{f_{2b}} \frac{v_2}{v_{1a}} \Big(\varphi_2 - \frac{0.5}{\varphi_4^2} \Big) (1 + u)^2 \frac{f_{1a}}{f_{3c}} \frac{v_3}{v_{1a}} (1 - 0.5\varphi_3^2) \Big) \Big]. \tag{31}$$

The parameters of the working flow at the nozzle that occur in Eqs. (30) and (31) can be determined from the continuity equation (29) and the relation for adiabatic flow with friction:

$$w_{1a} = \varphi_1 \sqrt{2 \Delta i_s},$$

where Δi_s is the difference between the enthalpies in the case of isentropic expansion in the nozzle, and φ_1 is the velocity coefficient of the nozzle.

The optimal ratio $\psi = f_{1a}/f_{3c}$ of the sections of the jet system, corresponding to maximal head, can be determined from the condition $d(\Delta p)/d\psi = 0$. After manipulation, we obtain

$$\psi_0 = \Big\{ \varphi_2 + B \Big[\frac{1}{(1 - \psi_0)^2} - 1 \Big] \Big\} \Big/ 2C, \tag{32}$$

where

$$B = u^2 \frac{v_2}{v_{1a}} \Big(\varphi_2 - \frac{0.5}{\varphi_4^2} \Big); \quad C = (1 + u)^2 \frac{v_3}{v_{1a}} (1 - 0.5\varphi_3^2).$$

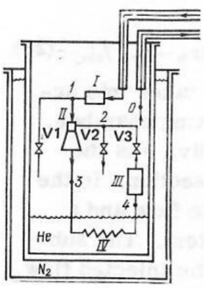

FIG. 32. Arrangement of experiment with circulation of liquid helium at supercritical pressure: I is an adsorber filter, II is the jet pump, III is an electric heater, IV is a cooling coil; 0,1,2,3,4 are the points at which the pressure is measured, and 1,2,3,4 are the points at which the temperature is measured.

The smallest of the three roots of (32) gives the required solution. This solution can be obtained iteratively by taking the zeroth approximation equal to zero. The admissibility of these relations was established experimentally at the Joint Institute for Nuclear Research.

A test facility[28] with circulation of liquid helium at supercritical pressure is shown schematically in Fig. 32. The injection coefficient is calculated in accordance with the formula

$$u=(i_1-i_3)/(i_3-i_2),$$

where i_1, i_2, i_3 are the flow enthalpies corresponding to the measured values of the temperature and pressure at the indicated points. The head Δp is measured by means of a differential manometer.

The experimental data are compared with the calculations made using the characteristic equation in Fig. 33. The calculations were made using the values of the velocity coefficients given in Ref. 27 as general recommendations: $\varphi_2=0.97$, $\varphi_3=0.90$, $\varphi_4=0.92$. The value of φ_1 determined during the experiments was 0.94. Figure 33 shows the adiabatic efficiency of the jet pump. It characterizes the efficiency of the system and is defined by

$$\eta=u\Delta i_{2-3s}/\Delta i_{1-3s},$$

where Δi_{1-3s} (respectively, Δi_{2-3s}) is the difference between the enthalpies at the beginning and end of isentropic expansion (compression) of the working (injected) flow to the pressure p_3. The largest values of the efficiency correspond to injection coefficients for which the calculated optimal ratio ψ_0 of the sections is equal to the ratio of the sections of the tested facility.

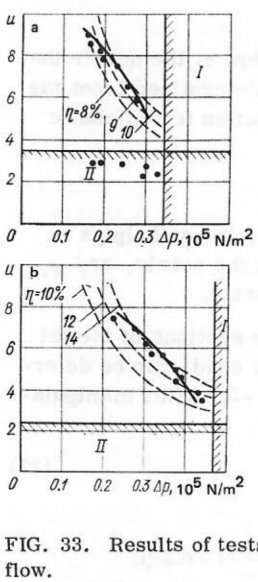

FIG. 33. Results of tests of jet pumps using a single-phase flow.

During the experiments, it was found that when the head was increased to values corresponding to region I in Fig. 33 flow separation occurs, this characterizing a sharp decrease in the injection coefficient. During subsequent smooth decrease in Δp, the injection coefficient hardly increases (see the point at the bottom in Fig. 33a). It was found that the position of the separation region I depends on the distance l between the sections a and b (see Fig. 31). Separation occurs when the injection coefficient has the value

$$u^*=0.31l/d_{1a}-1, \tag{33}$$

where d_{1a} is the exit diameter of the nozzle. The separation region, determined in accordance with Eq. (33), is region II in Fig. 33. It is necessary to choose l in such a way that in all regimes the injection coefficient is larger than u^*.

The arrangement of an experimental facility with circulation of a two-phase vapor–liquid flow of helium[29] is shown in Fig. 34. A measuring vessel, in which the level is measured by a superconducting level gauge, is placed in the cryostat. It communicates with the cryostat space at the top and the bottom: at the top directly, at the bottom through the valve V2. When V2 is in the open position, the levels of the liquid helium in the cryostat and the measuring vessel are equal. After V2 has been closed, the level H is lowered, and the flow rate of the injected helium can be determined from the rate of change of the level.

The working flow rate is found from the equation

$$G_1=Q/(i_0-i_1),$$

where Q is the power of the electric heater, and i_0 and

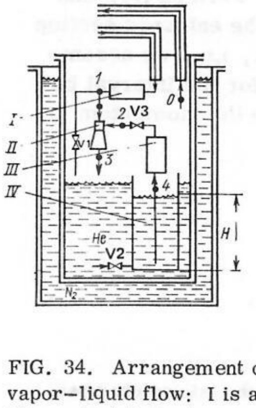

FIG. 34. Arrangement of experiment with circulation of vapor–liquid flow: I is an adsorber filter, II a jet pump, III an electric heater, and IV a measuring vessel.

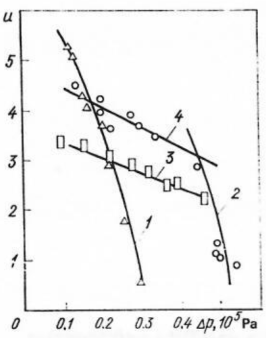

FIG. 35. Results of tests of jet pumps using vapor–liquid flow. Curves 1 and 2 are calculated on the basis of the characteristic equation; curves 3 and 4 are calculated using the relations for the limiting regimes.

i_1 are the enthalpies corresponding to the measured temperature and pressure at the points 0 and 1.

Comparison of the experimental data with the calculation based on the characteristic equation (curves 1 and 2in Fig. 35) shows that it gives a correct quantitative description in the case of circulation of a helium vapor–liquid mixture. Curves 3 and 4 correspond to the limiting regimes when the flow velocity in one of the sections of the system reaches the local velocity of sound. These curves were calculated using a thermodynamic equilibrium flow scheme for a two-phase flow.[29]

3. DESCRIPTION OF THE CRYOGENIC SYSTEMS OF THE LARGEST SUPERCONDUCTING ACCELERATORS

The superconducting accelerator at Fermilab

Three stages in the construction of the accelerator are foreseen.[30]

The first stage is the construction of the ring of the superconducting accelerator below the existing warm ring. The existing accelerator is a booster for the superconducting accelerator. The final energy of the accelerated protons is conserved. The field of the "warm" accelerator will be reduced, and the energy requirement accordingly. The expected economy is 5 million dollars per year. In this stage, the accelerator is called the Energy Saver.

The second stage is to raise the energy of the accelerated protons to 1 TeV with an intensity of 2.5 $\times 10^{13}$ proton/cycle. In this stage, the accelerator is called Tevatron I.

The third stage is to produce colliding $p\bar{p}$ beams with center-of-mass energy 2 TeV and luminosity 10^{30} $cm^{-2} \cdot sec^{-1}$; then the accelerator will be called Tevatron II. A dipole magnet of this accelerator is shown in section in Fig. 36.[31] The coils of the magnet are cooled at 4.6°K by liquid helium circulating in the channels of the coil 12 through the spaces 5 between the coil and the beam pipe 2 and between the bandage 11 and the wall of the cryostat. In the space 4, two-phase helium flows in the opposite direction, taking heat from

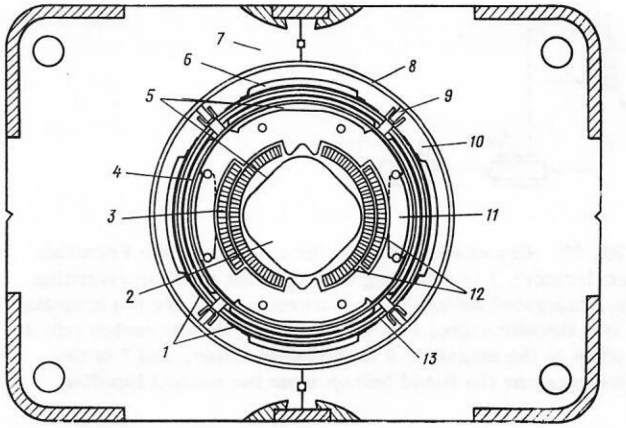

FIG. 36. Section of dipole magnet of the series E 22-14: 1 is the heat shield, 2 is the vacuum chamber for the particle beam (beam pipe), 3 is the superconducting coil, 4 is two-phase helium, 5 is liquid helium, 6 is liquid nitrogen, 7 is an iron magnetic shield, 8 is the outer vacuum casing, 9 is a roller support, 10 is the vacuum space with laminated insulation, 11 is the coil bandage, 12 is the electrical insulation with helium channels, and 13 is the mounting ring.

the liquid helium and absorbing the heat inflow from the ambient medium. Initially, it was assumed that the liquid helium passing through the coil would remove the heat released by it, ascend in the free space between the bandages, and exchange heat with the two-phase helium. Subsequently, it became necessary to fix the coil with greater care, which made it necessary to fix the ring of the bandage without axial gaps, and this made it difficult to circulate liquid helium in the tank of the two-phase helium. After this, the liquid helium was cooled unequally at different sections of the magnet and the mixing of it was begun after the passing of each magnet. The shielding system was also changed by replacing cold helium at 20°K by liquid nitrogen flowing in the space 6. During the testing of the magnets, it was found that rapid cooling causes a certain deformation and the nonuniformity of the field exceeds the admissible limit. For more rigid fixing of the magnetic part of the system, certain supports 9 were replaced by adjustable bolts with springs.[32]

It should be noted that in the Fermilab magnets a magnetic iron shield 7 at room temperature ("warm" iron) is used. A shortcoming of warm iron compared with cold is a smaller contribution to the magnetic field induction. The advantage is the decrease in the mass of the material that must be cooled to liquid-helium temperature. The beam pipe 2 has liquid-helium temperature. An advantage of such a solution is the simplification of the construction and the use of the inner surface of the beam-pipe wall as a condensation pump. There is a danger, not yet tested experimentally, that scattered particles could cause evaporation of the condensate from the surface of the beam pipe and spoil the vacuum within it.

The basic scheme of the cryostating of the magnets[31] is shown in Fig. 37. The cooling helium passes through

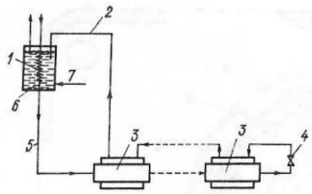

FIG. 37. Cryostating scheme for magnets of the Fermilab accelerator: 1 is a cooling coil, 2 is the pipe for returning the evaporated helium to the supercooler, 3 are the magnets, 4 is a throttle valve, 5 is the pipe carrying the cooled liquid helium to the magnets, 6 is the supercooler, and 7 is the pipe carrying the liquid helium from the central liquifier.

the cooling foil 1 of the supercooler 6. Single-phase liquid helium is passed through the coils of a number of magnets 3 arranged successively and through the throttle valve 4. The resulting two-phase helium streams in the opposite direction in the corresponding spaces of the magnets 3 and is returned to the supercooler through the tube 2. Liquid helium from the main liquifier is supplied through the tube 7.

The cryogenic system of the accelerator[31] is shown in Fig. 38. Around the almost 6-km ring of magnets 8 of the accelerator there are 24 satellite refrigerators 9, each of which supplies with liquid helium one arm containing 32 dipoles and eight quadrupoles. The liquid helium is supplied to the satellites from the central liquifier 1, which has an output of 4500 liter/h. Next to it, there is a central liquid-nitrogen recondenser 2, which supplies the precooling stage of the liquifier 1 at the temperature level 80°K and provides liquid nitrogen for the shields of the satellites and the magnets. Around the complete perimeter of the accelerator, there are five ring collectors of the cryogenic system: for liquid helium from the central helium liquifier to the satellites 7, for the compressed helium from the compressors of the satellites to the central helium liquifier 4, for returning the gaseous helium that cools the power leads to the gas system of the central helium liquifier 5, for the supply of liquid nitrogen to the shields 3, and for returning the evaporated nitrogen to the recondenser 6.

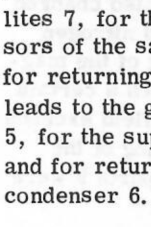

FIG. 38. Scheme of the accelerator cryogenic system: 1 is the central liquifier, 2 is the liquid-nitrogen recondenser, 3 is the collector for the supply of liquid nitrogen to the accelerator systems, 4 is the compressed-helium collector, 5 is the collector of low-pressure "warm" helium, 6 is the evaporated nitrogen collector, 7 is the liquid-helium collector, 8 are the magnets of the accelerator, and 9 is a satellite refrigerator.

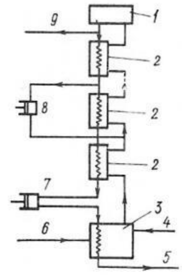

FIG. 39. Schematic arrangement of a satellite refrigerator: 1 is the compressor, 2 are heat exchangers, 3 is a Dewar vessel, 4 is the return of evaporated helium from the magnets, 5 is the cooled liquid helium supplied to the magnets, 6 is the pipe carrying the liquid helium from the central liquifier, 7 is a vapor–liquid expansion machine, 8 is a gas-expansion machine, and 9 is the pipe carrying the compressed helium to the central liquifier.

Each satellite (Fig. 39) has its own compressor 1 and is equipped with orindary 8 and vapor–liquid 7 expansion machines. Liquid helium is supplied to the satellite through the tube 6. An amount of helium equivalent to the liquid helium supplied to the satellite is returned to the central liquifier through the tube 9 by removing part of the compressed helium after the compressor 1 of the satellite.[31]

An alternative to combining the central liquifier and the satellites could be a scheme with several identical helium refrigerators distributed around the perimeter of the accelerator. Such a scheme is good from the point of view of emergencies. The breakdown of one of the expansion machines of a satellite can be compensated by increasing the amount of liquid helium that is taken off. The satellites are constructed in such a way that a vapor–liquid expansion machine can be replaced in 2 h. Breakdown of a satellite compressor can be compensated by taking compressed helium from the collector. Then the return gas flow from the satellite can be directed to the collector which takes gas from the power leads. If the central liquifier breaks down, the satellites provide 2/3 of the total refrigerating capacity of the helium system, which means that work can continue on the accelerator at a reduced rate. The supply of liquid helium to the power leads in this case is ensured by the reserve of liquid helium. If the nitrogen recondenser breaks down, operation can be maintained by filling the tank with liquid nitrogen brought in from outside. The liquid-helium ring collector is supplied from an intermediate tank of capacity 30–40 m^3 by means of a pump. The section of the collector and the performance of the pump make it possible to pass an excess amount of liquid helium through this line. The unused liquid helium is returned back to the tank. The collector is made of 24 sections. If one section breaks down, it can be disconnected. The ring is then no longer closed; it is possible to supply liquid helium to all 24 satellites, but it becomes impossible to return the excess liquid helium to the tank. Heat flow to the helium in the collector is reduced by shielding with liquid nitrogen.

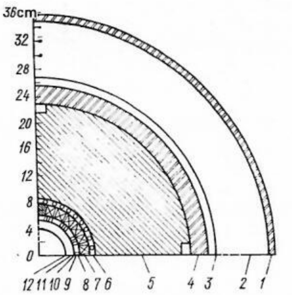

FIG. 40. Section of a quadrant of the superconducting dipole magnet of the ISABELLE accelerator: 1 is the vacuum casing, 2 is the vacuum space to take the superinsulation and helium pipes, 3 is a copper radiation screen, 4 is a stainless steel support pipe, 5 is an iron magnetic shield; 6 and 8 are plastic, 7 is the coil, 9 is a correcting coil, 10 is the inner coil of the cryostat, 11 is superinsulation, and 12 is the warm bore.

The superconducting accelerator ISABELLE at Brookhaven

A section through a quadrant of a superconducting dipole magnet is shown in Fig. 40.[30] Heat is removed by liquid helium, which passes through channels of the coil 7. The iron shield 5 has liquid-helium temperature, in contrast to the "warm" magnetic shield of the dipole magnets at Fermilab. The copper radiation shield 3 is cooled by helium at temperature 55°K.

The arrangement of the cryogenic system of the acceleator is shown in Fig. 41.[30] Liquid helium from the refrigerator is passed to the supercooler 4, is cooled there to 2.6°K at pressure 0.5 MPa, and is then passed to the ring supply collector 6. The low temperature in the supercooler 4 is achieved by evacuating helium by means of a cryogenic forcing pump.[33] From the collector 6, the helium is passed to the middle of each sextant 8, is heated to 3.8°K in the 45 successively connected magnets, is collected in the collector 5, and returned to the supercooler 4. The heat shields 9

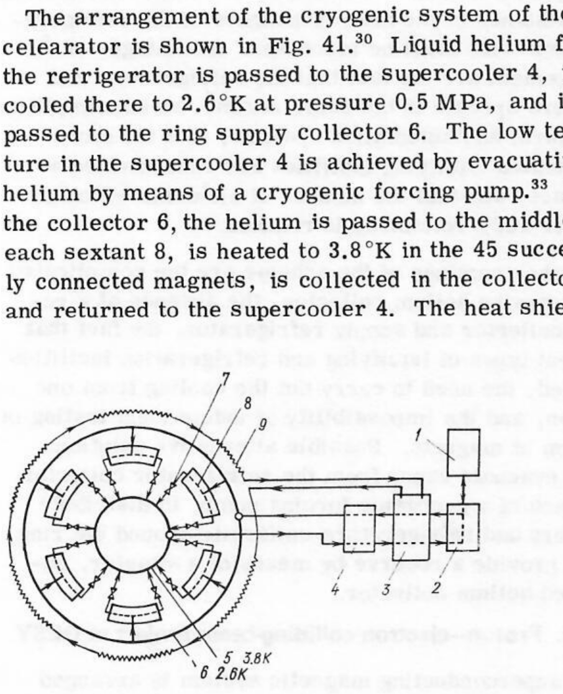

FIG. 41. Schematic arrangement of the cryogenic system for ISABELLE: 1 is the gas reservoir and purification plant, 2 is the compressor station, 3 is the main refrigerator, 4 is the supercooler with circulation system, 5 is the collector of the helium returned at 3.8°K, 6 is the collector of the cryostating liquid helium with temperature 2.6°K, 7 is a flow-regulation valve, 8 are the magnets of one sextant, and 9 is the heat shield.

of the magnets and the tubes are cooled at a temperature of about 55°K by helium taken from the refrigerator. The thermal load on one magnet is determined at 4.6 W, of which 2 W comes from the beam pipe. The insulation vacuum must be not worse than 1.3×10^{-4} GPa and the thermal load is 15.5 kW at 4°K and 36.8 kW at 55°K. The load at 4°K is made up of the heat release in the magnets (33%), the heat flow through the power leads (21%), and the heat flow to the outer helium system (34%). The refrigerator system does not contain a nitrogen refrigeration cycle. For all systems, one refrigerator with capacity 23.5 kW at 4°K and 55 kW at 55°K is foreseen. The scheme is designed for all possible variants of operation: cooling of the system, working regime, and the procedure for dealing with the emergency regime resulting from damage to the insulation vacuum or transition of a magnet to the normal state.

The cryogenic system of ISABELLE is not thermodynamically advantageous. The current density in a coil is determined by the temperature at the magnet exits: 3.8°K. The entrance temperature is 2.6°K, very low. The extra energy requirements in maintaining this are approximately proportional to the ratio of the absolute temperatures (3.8/2.6). It is not clear from the published material how high reliability of operation of the system is to be achieved—nothing is said about doubling the single installed liquifier.

Accelerator and storage facility at the Institute of High Energy Physics (USSR)

The Accelerator and Storage Facility[2] contains 2508 superconducting dipoles and quadrupoles placed around a ring approximately 20-km long. The magnets are cooled by liquid helium at 4.1°K. The total thermal load at this temperature is 48 kW. To shield the magnets and the collectors and to feed the helium liquifiers and refrigerators, liquid nitrogen with a total thermal load of 108 kW at 70–90°K is used. An arm of magnets is cooled as follows. At the entrance to a magnet, single-phase liquid helium is split into two streams (one passes through the channels of the superconducting coil and is slightly heated by the heat released in it; the other passes through a by pass channel between the outer surface of the bandage and the shell of the cryostat and is slightly cooled by heat exchange with boiling helium). After each magnet, the two streams are mixed and again split when the following magnet is reached. At the end of an arm, which is about 800-m long, the single-phase helium is passed through a valve, and the resulting two-phase helium streams in the opposite direction along a corresponding cavity in the magnets, taking heat from the supercooled helium flowing along the bypass channel.

Outer cryogenic system (Fig. 42)

Around the accelerator ring there are 12 stations with 24 refrigerators, each of which extracts heat from one arm. Each refrigerator 19 has an expansion machine 12, and, in addition, is supplied with liquid helium from a centrally positioned liquifying station in an amount constituting a few percent of the flow circulating

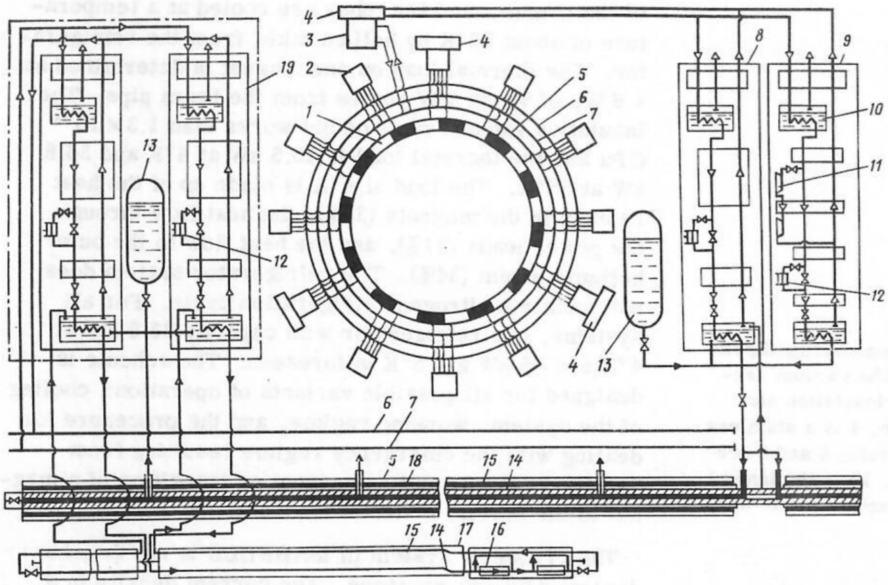

FIG. 42. Schematic arrangement of the helium cryogenic system of the Accelerator and Storage Complex: 1 is the accelerator ring, 2 is the cryostating station, 3 is the helium-liquification station, 4 is the compressor station, 5 is the cryogenic collector, 6 is the low-pressure collector, 7 is the high-pressure collector, 8 is the supply refrigerator, 9 is the helium liquifier, 10 is a nitrogen tank, 11 are turbine expansion machines, 12 is a reciprocating expansion machine, 13 is a helium cryogenic vessel, 14 is the cryostating liquid helium, 16 is a superconducting coil of a magnet, 17 is an arm of magnets, 18 is a nitrogen shield, and 19 is a refrigerator.

through the refrigerator. If the expansion machine 12 breaks down, the refrigerator can operate in the Joule-Thomson regime, using more liquid helium. Liquid nitrogen is also supplied to the refrigerators in order to decrease the size of the heat exchanger in the warm region due to the enhanced under-recuperation, which is compensated at the liquid-nitrogen temperature level. A cryogenic collector 5 is foreseen for supplying liquid helium to the refrigerators. To prevent the liquid helium in the collector from being heated, it is shielded by passing some of the liquid helium through a throttle valve and returning it in a counterstream as in the arm of magnets. The necessary head of the liquid helium in the collector is produced by means of a special refrigerator 8. The helium liquifier operates in accordance with the usual scheme with four cooling stages: in a nitrogen tank 10, and by expansion in two turbine expansion machines 11 and in a reciprocating expansion machine 12. The compressors are situated in four compressor stations, one of which serves the liquifiers and three the refrigerators. The high-pressure 7 and low-pressure 6 warm-helium collectors connect the compressor stations to the liquifiers and refrigerators. It is intended that liquid helium with temperature 4.1°K will be used to cool the magnets.

The decrease in the temperature of the cooling helium permits either an increase in the current density in the coils or operation with current density significantly below the critical density. To lower the helium temperature, it is necessary to lower the helium pressure in the refrigerator collector by lowering the pressure in the suction collector of the screw compressors to 0.06 MPa, which results in an increase in the number of compressors to maintain the necessary mass of the supplied helium. In addition, to reduce the hydraulic resistance of the heat exchangers of the refrigerator, their size must be increased. Advantages of the arrangement are the fact that the refrigerators and liquifiers operate in the same regime, which simplifies the control and automation systems, that the more complicated liquifying facilities are concentrated at one place, and that the number of elements which do not have 100% reliability is reduced.

The shortcomings of the scheme are the complicated and expensive helium collector, the absence of a reserve collector and supply refrigerator, the fact that different types of liquifying and refrigerating facilities are used, the need to carry out the cooling from one position, and the impossibility of independent testing of one arm of magnets. Possible alternative solutions are to evacuate vapor from the refrigerator collector by means of a cryogenic forcing pump, to distribute liquifiers and refrigerators uniformly around the ring, and to provide a reserve by means of a simpler, unshielded helium collector.

HERA: Proton–electron colliding-beam project at DESY

The superconducting magnetic system is arranged around a circle 6.5-km long. With magnetic fields of 4.75 T, the proton energy will be 820 GeV. Assuming reasonable synchrotron radiation losses, the maximal electron energy has been taken to be 30 GeV. The luminosity at maximal energy is 0.35×10^{32}, and the magnetic field of the accelerator is formed by 640 dipoles, 164 quadrupoles, and 80 special magnets for the beam-intersection positions. The dipoles have

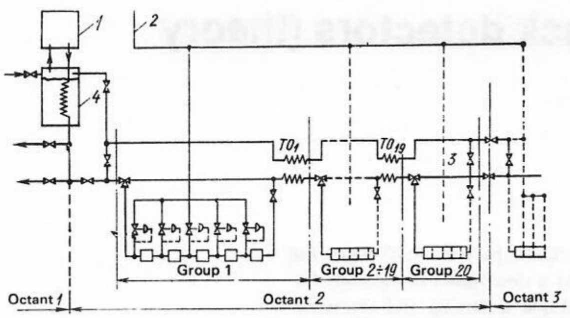

FIG. 43. Basic arrangement of the helium cryogenic system of the HERA accelerator: 1 is the refrigerator, 2 is the pipe for returning the warm helium from the power leads, 3 is a throttle valve, 4 is a collector, and TO_1–TO_{19} are heat exchangers.

a warm beam pipe, like the ISABELLE accelerator, and a warm iron magnetic shield, like the Fermilab accelerator. The cryostating will be done by pumping single-phase helium through the dipoles at 3.8°K by means of four refrigerators distributed uniformly around the ring next to the long straight sections. Figure 43 shows the arrangement of the cryogenic system of the accelerator.[30] Each refrigerator serves two arms of one octant in each arm. An octant consists of 22 groups of magnets, each group containing five magnets cooled in series. After each group of magnets, there are heat exchangers TO_1–TO_{19}, in which the cryostating liquid helium is again cooled by means of two-phase helium produced by passing the liquid helium at the end of the octant through the throttle valve 3. The evaporated helium is returned to the gas space of the collector 4 and is then sent to the refrigerator 1. The compressed helium from the refrigerator is cooled in the cooling coil of the collector. Neighboring octants are connected by valves, which ensures a standby if one of the refrigerators breaks down. Warm helium from the power leads is returned to the system through the tube 2.

In all the described cryogenic systems of the superconducting accelerators, heat is removed by single-phase liquid helium. In the accelerators at Fermilab and in the Accelerator and Storage Facility the low temperature of the cryostating liquid helium is maintained by heat exchange within the magnet between liquid helium and boiling two-phase helium. In ISABELLE, the liquid helium is split into parallel streams and intermediate cooling of it is not arranged. In HERA, special heat exchangers are foreseen for heat exchange between the liquid helium heated in the groups of magnets and two-phase helium. Here, we have not considered the cryogenic systems of superconducting cyclotrons,[34] since the well-known immersion method of cryostating is used for them.

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