THE TECHNOLOGY OF SUPERFLUID HELIUM

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Abstract
The technical properties of helium II ("superfluid" helium) are presented from the user point of view. Its applications to the cooling of superconducting devices, particularly in accelerators, are discussed in terms of heat transfer capability and limitations in conductive and convective modes. Large-capacity refrigeration techniques below 2 K are reviewed, as concerns thermodynamic cycles as well as process machinery. Examples drawn from existing or planned projects illustrate the presentation.

1. INTRODUCTION
Once a curiosity of nature and still today an arduous research topic in condensed-matter physics, superfluid helium*) has also become a technical coolant for advanced superconducting devices, to the point that it is now implemented in industrial-size cryogenic systems, routinely operated with high reliability. Two classes of reasons call for the use of superfluid helium as a coolant for superconducting devices, namely the lower temperature of operation, and the enhanced heat transfer properties at the solid-liquid interface and in the bulk liquid.

The lower temperature of operation is exploited in high-field magnets [1, 2], to compensate for the monotonously decreasing shape of the superconducting transition frontier in the current density-versus-magnetic field plane, shown in Figure 1 for superconducting materials of technical interest. In this fashion, the current-carrying capacity of the industrial Nb-Ti superconducting alloys can be boosted at fields in excess of 8 T, thus opening the way for their use in high-field magnet systems for condensed-matter physics [3-5], nuclear magnetic resonance [6, 7], magnetic confinement fusion [8, 9] and circular particle accelerators and colliders [10-12]. In the case of high-frequency superconducting devices, such as acceleration cavities [13], the main drive for superfluid helium cooling is the exponential dependence of the BCS losses on the ratio of operating-to-critical temperature. Accelerators based on this technology, such as medium-energy, high-intensity machines [14, 15] and future high-energy lepton colliders [16-18] operate in the temperature range which minimizes capital costs and overall energy consumption. This issue is schematized in Figure 2.

The technical heat transfer characteristics of superfluid helium basically derive from peculiar transport properties [19, 20]. Its low bulk viscosity enables superfluid helium to permeate to the heart of magnet windings, while its very large specific heat (typically $10^5$ times that of the conductor per unit mass, $2 \times 10^3$ per unit volume), combined with excellent heat conductivity at moderate heat flux ($10^3$ times that of cryogenic-grade OFHC copper) can produce powerful stabilization against thermal disturbances. In order to fully exploit these properties, however, both in steady-state and transient regimes, e.g. for power heat transport over macroscopic distances as well as intimate stabilization of superconductors, an elaborate thermo-hydraulic design of the cooling circuits, conductor, insulation and coil assemblies is required. This often conflicts with other technical or economic requirements of the projects and acceptable trade-offs have to be found.

*) Strictly speaking, we are referring to the second liquid phase of helium, called He II, which exhibits the unusual bulk properties associated with superfluidity and is therefore also called "superfluid". This is not to be confused with the entropy-less component of the phenomenological two-fluid model accounting for the behaviour of He II, which some authors prefer to keep the qualitative "superfluid" for.
In the following, we will only address the specific issues of cryogenic technology pertaining to the use of superfluid helium as a technical coolant, namely different cooling methods as well as processes and machinery for sub-lambda temperature refrigeration [21]. Reference is made to companion lectures for cryogenic techniques which – however important in system design - are not superfluid-helium specific, such as thermal insulation and cryostat design [22, 23].

![Diagram of superconductors](image1)

**Fig. 1** Critical current density of technical superconductors.

![Diagram of optimal temperature](image2)

**Fig. 2** Optimal operating temperature of RF superconducting cavities.

2. DIFFERENT COOLING METHODS

2.1 Pressurized versus saturated superfluid helium

A look at the phase diagram of helium (Figure 3) clearly shows the working domains of saturated helium II, reached by gradually lowering the pressure down to below 5 kPa along the saturation line, and pressurized helium II, obtained by subcooling liquid at any pressure above saturation, and in particular at atmospheric pressure (about 100 kPa).

Although requiring one more level of heat transfer and additional process equipment - in particular a pressurized-to-saturated helium II heat exchanger - pressurized helium II cooling brings
several important technical advantages [24]. Avoiding low-pressure operation in large and complex cryogenic systems clearly limits the risk of air inleaks, and resulting contamination of the process helium. Moreover, in the case of electrical devices, the low dielectric strength exhibited by low-pressure helium vapour [25], in the vicinity of the minimum of the Paschen curve (Figure 4) [26], brings the additional risk of electrical breakdown at fairly low voltage. Operating in pressurized helium II avoids this kind of problem.

![Phase diagram of helium](image1.png)

**Fig. 3** Phase diagram of helium.

![Paschen curve for helium at 300 K](image2.png)

**Fig. 4** Paschen curve for helium at 300 K.

However, the most interesting and specific aspect of pressurized helium II in the operation of superconducting devices stems from its capacity for cryogenic stabilization. As a subcooled (monophase) liquid with high thermal conductivity, pressurized helium II can absorb in its bulk a deposition of heat, up to the temperature at which the lambda-line is crossed, and local boiling only then starts due to the low thermal conductivity of helium I. Quasi-saturated helium II, which is in fact slightly subcooled due to the hydrostatic head below the surface of the liquid bath, may only absorb...
heat deposition up to the point at which the saturation line is crossed, and change of phase occurs. The enthalpy difference from the working point to the transition line is usually much smaller in the latter case. The argument, developed in reference [27], typically yields an order of magnitude better performance in favour of pressurized helium II.

2.2 Conduction cooling

In the following we shall only consider conductive heat transport in helium II at heat fluxes of technical interest (typically above 1 kW.m$^{-2}$). For most practical geometries, this means working in the "turbulent" regime with full mutual friction between the components of the two-fluid model [28]. In this regime, helium II exhibits a large, finite and non-linear bulk heat conductivity, the value of which depends both on temperature and heat flux. While the general patterns of this behaviour can be predicted by the Gorter-Mellink [29] theory*), practical data useful for engineering design has been established in a number of experiments [30-35].

Consider conduction in one dimension, e.g. in a tubular conduit of length $L$, the ends of which are maintained at temperatures $T_C$ and $T_W$. The steady-state heat flux $q$ is given by:

$$q^n \cdot L = X(T_C) - X(T_W)$$

(1)

where the best experimental fit for $n$ is 3.4, and $X(T)$ is a tabulated function of temperature, physically analogous to a conductivity integral [30]. A plot of this function reveals that the apparent thermal conductivity of helium II goes through a maximum at around 1.9 K (Figure 5).

![Fig. 5 Thermal conductivity integral and apparent thermal conductivity of pressurized superfluid helium [22].](image)

As an example, the heat flux transported by conduction between 1.9 and 1.8 K in a 1-m long static column of helium II is about 1.2 W.cm$^{-2}$, i.e. three orders of magnitude higher than what would be conducted along a bar of OFHC copper of the same geometry! The non-linearity with respect to heat flux also results in a much weaker dependence of conduction upon length, or thermal gradient. Figure 6 shows the steady-state conduction $Q$ in superfluid helium between 1.9 and 1.8 K versus the static column length $L$ for different equivalent nominal diameters of the column. This abacus clearly shows that while the heat flux conducted in a solid is directly proportional to the thermal gradient applied, doubling the conduction length in a column of helium II only reduces the heat flux by some 20 %.

*) C.J. Gorter and J.H. Mellink introduced in 1949 the idea of an interaction producing mutual friction between the components of the two-fluid model, to account for the observed transport properties of helium II.
The variation of $X(T)$ also implies that, for each value of the cold boundary temperature $T_C$, there exists a maximum possible heat flux at which $T_w$ reaches the lambda point, and the helium column ceases to be superfluid. Values of this limiting heat flux, which also weakly depends on $L$, range from a fraction to a few units of W.cm$^{-2}$, for practical cases of interest. This clearly brings an intrinsic limitation in the applicability of helium II conduction for quasi-isothermal cooling of long strings of superconducting devices in an accelerator. Transporting tens of watts over tens of meter distances would then require several hundred mK temperature difference and a large cross-section of helium, which is both impractical and thermodynamically costly. For a more precise estimate, consider a uniformly heated tubular conduit of length $L$, operating between temperatures $T_C$ and $T_w$, and apply the helium II steady-state conduction equation to this fin-type geometry. After integration:

$$\dot{q}_{total}^n \cdot L = (n+1) \cdot [X(T_C) - X(T_w)]$$

where $\dot{q}_{total}$ is the total heat flux flowing through the section at temperature $T_C$, near the heat sink. Figure 7 shows the steady-state conduction $\dot{Q}_{tot}$ in superfluid helium of a cryomagnet string with linear heating $\xi$ between 1.9 K (temperature of the warmest magnet) and 1.8 K (temperature at the heat sink). As an example, cooling by conduction a 50-m long cryomagnet string, with a uniform linear thermal load of 1 W.m$^{-1}$, would require a helium II cross-section of 90 cm$^2$, i.e. a 10.7-cm diameter conduit. In view of such constraints, the conduction-cooling scheme originally considered for the LHC project [36] was later abandoned.

Conduction through static pressurized superfluid helium however remains the basic process for extraction and local transport of heat from the LHC magnet windings, across their polyimide-wrap electrical insulation. Although the polyimide tape, which constitutes the insulation of the superconducting cable, is wrapped in two layers with half overlap (Figure 8), in order to achieve sufficient mechanical toughness and dielectric strength, this still preserves sufficient percolation paths for helium II conduction to significantly improve the heat transfer, well above the solid conduction across the sole polyimide [37].
Fig. 7 Steady-state conduction cooling of cryomagnet string with linear applied heat load.

Fig. 8 Heat transfer across polyimide-wrap insulation of superconducting cable.

The high thermal conduction in helium II can also be exploited to ensure quasi-isothermality of helium enclosures of limited spatial extension, such as the helium bath of a superconducting magnet under test. Knowledge of temperature changes at any point in the bath permits to assess enthalpy changes of the system, and thus to perform calorimetric measurements. This technique proves very convenient for measuring minute heat inleaks [38] or substantial energy dissipation [39] such as produced by ramping losses or resistive transitions in superconducting magnets.

2.3 Forced-flow convection of pressurized superfluid helium

To overcome the limited conduction of helium II in long strings of cryogenic devices, the obvious issue is to create a forced circulation of the fluid in a cooling loop, thus relying on convective heat transfer. One can then benefit of an additional control parameter, the net velocity imparted to the bulk
fluid. In the following we shall only discuss convection in channel diameters of technical interest, i.e. typically greater than a few mm. The flow induced by a pressure gradient across an hydraulic impedance is then essentially determined by the viscosity of the bulk fluid. Assuming that internal convection between the components of the two-fluid model is independent of the net velocity, reduces the problem to the behaviour of a flowing monophase liquid with high, non-linear thermal conductivity. The steady-state convective heat transport \( \dot{Q} \) between two points 1 and 2 of the cooling loop is then given by the difference in enthalpy \( H \) of the fluid flowing with a mass flow-rate \( \dot{m} \):

\[
\dot{Q} = \dot{m} \cdot (H_2 - H_1)
\]

An estimate of the potential advantage of forced convection over conduction can be made, using the same geometry and temperature boundary conditions as described in paragraph 2.2 above. Consider helium II pressurized at 100 kPa, flowing in a heated pipe of length 1 m and cross section 1 cm\(^2\), and assume its temperature increases from 1.8 K at pipe inlet, to 1.9 K at outlet. It is easy to show that for flow velocities above 0.2 m.s\(^{-1}\), convective heat transport exceeds conduction.

The above calculation however neglects pressure drop along the flow. A look at the pressure-enthalpy diagram of helium (Figure 9) reveals a positive Joule-Thomson effect [40]: the enthalpy of the fluid increases both with increasing temperature and pressure, so that an isenthalpic expansion results in a temperature increase. For example, pressurized helium II flowing across a pressure gradient of 50 kPa will warm up from 1.8 K to 1.9 K, in absence of any applied heat load. The magnitude of this effect requires precise knowledge of the thermohydraulic behaviour of helium II, in order to validate its implementation in long cooling loops [41].

Following early work [42, 43], several experimental programs have investigated heated flow of pressurized helium II in pipes and piping components [44, 45], culminating with the 230-m long test loop in Grenoble [46, 47] which gave access to high Reynolds numbers and extended geometries characteristic of accelerator string cooling loops. In parallel to that work, mathematical models were developed for calculating combined conductive and convective heat transport processes in complex circuits [48, 49], and validated on experimental results. Pressure drop and heat transfer - both steady-state and transient - in flowing pressurized helium II may now be safely predicted for engineering purposes, using well-established laws and formulae.

![Fig. 9 Pressure-enthalpy diagram and forced-flow convection in superfluid helium.](image-url)
The implementation of forced-flow cooling requires cryogenic pumps operating with pressurized helium II. Although most of the experimental work has been performed using positive displacement, i.e. bellows- or piston-pumps originally developed for helium I [50], the thermomechanical effect, specific of the superfluid, may also be used for driving cooling loops by means of fountain-effect pumps [51-54]. In spite of their low thermodynamic efficiency [55], a drawback of limited relevance for using them as circulators which have to produce low pumping work, fountain-effect pumps are light, self-priming and have no moving parts, assets of long-term reliability e.g. for embarked applications in space [56]. At higher heat loads, they have been considered [57] and tested [58] for forced-flow cooling of superconducting magnets: the overall efficiency of the process may then be improved by configuring the cooling loop so as to make use of the heat load of the magnet proper to drive the thermomechanical effect in the pump [59].

2.4 Two-phase flow of saturated superfluid helium

The conductive and convective cooling systems described above both transport heat deposited or generated in the load, over some distance through pressurized helium II, up to a lumped pressurized-to-saturated helium II heat exchanger acting as quasi-isothermal heat sink. This is achieved at the cost of a non-negligible - and thermodynamically costly - temperature difference, thus requiring to operate the heat sink several hundred mK below the temperature of the load.

A more efficient alternative is to distribute the quasi-isothermal heat sink along the length of the accelerator string. In this fashion the conduction distance - and hence the temperature drop - in pressurized helium II is kept to a minimum, typically the transverse dimension of the device cryostat. This leads to the cooling scheme proposed for the LHC at CERN, schematized in Figure 10: the superconducting magnets operate in static baths of pressurized helium II at around atmospheric pressure, in which the heat load is transported by conduction to the quasi-isothermal linear heat sink constituted by a copper heat exchanger tube, threading its way along the magnet string, and in which flowing two-phase saturated helium II gradually absorbs the heat as it vaporizes [11].

Although potentially attractive in view of its efficiency in maintaining long strings of magnets at quasi-uniform temperature, this cooling scheme departs from the well-established wisdom of avoiding long-distance flow of two-phase fluids at saturation, particularly in horizontal or slightly inclined channels. Moreover, no experimental data was originally available on flowing saturated helium II, and very little for other cryogenic fluids in this configuration. Following first exploratory tests [60] which demonstrated the validity of the concept on a reduced geometry, a full-scale thermohydraulic loop [61] permitted to establish the stability of horizontal and downward-sloping helium II flows, to observe partial (but sufficient) wetting of the inner surface of the heat exchanger.
tube by the liquid phase, thanks to flow stratification, and to address process-control issues and
develop strategies for controlling uniformity of temperature at strongly varying applied heat loads, in
spite of the low velocity of the liquid phase. As long as complete dryout does not occur, an overall
thermal conductance of about 100 W.m$^{-1}$.K$^{-1}$ can be reproducibly observed across a DN40 heat
exchanger tube, made of industrial-grade deoxidized phosphorus copper.

Once the wetting of the inner surface of the tube is guaranteed, the heat transfer from the
pressurized to the saturated helium II is controlled by three thermal impedances in series: solid
conduction across the tube wall, and Kapitza resistance at the inner and outer interfaces between tube
wall and liquid (Figure 11). While the former can be adjusted, within technological limits, by
choosing tube material and wall thickness, the latter, which finds its origin in the refraction of
phonons at the liquid-solid interfaces and is thus strongly temperature-dependent, usually dominates
below 2 K [62].

![Fig. 11 Kapitza conductance at copper-helium II interface.](image)

The final validation of the two-phase helium II flow cooling scheme for LHC has been
performed successfully on a 100-m long test string, equipped with full-scale prototype cryomagnets,
operated and powered in nominal conditions [63, 64]. At varying heat loads exceeding 1 W.m$^{-1}$, all
magnets in the string were maintained in a narrow range of temperature, a few tens of mK above the
saturation temperature of the flowing helium II. Thermal buffering provided by the pressurized helium
II baths contributed to limit temperature excursions, at the cost of introducing strong non-linearities
and time delays in the system, which must be coped with by elaborate, robust process control [65, 66].
In complement of that applied work, more fundamental experimental studies have been conducted on
specially instrumented test loops at CEN-Grenoble, comprehensively equipped with diagnostics and a
transparent section for visual observation and interpretation of the flow patterns [67-69]. As long as
the vapor velocity remains sufficiently low to maintain stratified flow (up to a few m/s), engineering
design of such a cooling scheme rests on a few simple sizing rules [70]. At higher vapor velocity,
entrainment and atomization effects, still under investigation, complicate the flow pattern and impact
on the heat transfer [71].

This type of cooling scheme may also be used for extracting much higher linear heat loads,
typically about 10 W.m$^{-1}$, as present in the low-beta quadrupoles in the high-luminosity insertions of
the LHC [72, 73], at the expense of a larger-diameter heat exchanger tube to limit the saturated vapour velocity and thus preserve flow stratification.

3. REFRIGERATION CYCLES AND EQUIPMENT

The properties of helium at saturation (see Figure 3) impose to maintain an absolute pressure below 1.6 kPa on the heat sink of a 1.8 K cryogenic system. Bringing the saturated vapour up to atmospheric pressure thus requires compression with a pressure ratio exceeding 80, i.e. four times that of refrigeration cycles for "normal" helium at 4.5 K. Figure 12 shows the basic scheme for refrigeration below 2 K. A conventional refrigerator produces liquid helium at 4.5 K, later expanded down to 1.6 kPa in a Joule-Thomson expansion stage. The gaseous helium resulting from liquid vaporization is compressed above the atmospheric pressure and eventually recovered by the 4.5 K refrigerator. We will therefore start by presenting the Joule-Thomson expansion stage.

Three types of cycles, sketched in Figure 12, can be considered [74, 75] for producing refrigeration below 2 K:

- the “warm” compression cycle based on warm sub-atmospheric compressors,
- the “cold” compression cycle based on multistage cold compressors,
- the “mixed” compression cycle based on a combination of cold compressors in series with warm sub-atmospheric compressors.

We will then proceed to discuss thermodynamics and machinery for these three types of cycle.

![Generic process cycles for refrigeration below 2 K.](image)

**Fig. 12 Generic process cycles for refrigeration below 2 K.**

3.1 Joule-Thomson expansion stage

The efficiency of the Joule-Thomson expansion of liquid helium, say from 0.13 MPa and 4.5 K, down to 1.6 kPa and 1.8 K, can be notably improved if it is previously subcooled by the exiting very-low-pressure vapour (Figure 13). This is performed in a counter-flow heat exchanger, subcooling the incoming liquid down to 2.2 K by enthalpy exchange with the very-low-pressure saturated vapour. This heat exchanger has to produce limited pressure drop, particularly in the very-low-pressure stream. A maximum pressure drop of 100 Pa is generally acceptable, corresponding to a few per cent
of the absolute saturation pressure. The design of such heat exchangers for large flow-rate [76] is not straightforward, and their qualification impractical. As a consequence, the LHC cryogenic system features several hundred small-size (5 to 20 g/s) heat exchangers, distributed around the ring. This also avoids transporting subcooled helium over long distances, saving one header in the ring distribution line. Following prototyping, technical validation of different solutions [77, 78] and commercial selection, these heat exchangers are now series-produced by industry.

![Diagram of heat exchanger](image)

Fig. 13 Efficiency of Joule-Thomson expansion.

### 3.2 “Warm” compression cycle

For low-power refrigeration, e.g. in small laboratory cryostats, this is achieved by means of standard Roots or rotary-vane vacuum pumps, handling the very-low-pressure gaseous helium escaping from the bath after it has been warmed up to ambient temperature through a heat exchanger and/or an electrical heater. This technology may be pushed to higher flow-rates using liquid-ring pumps, adapted for processing helium by improving the tightness of their casing and operating them with the same oil as that of the main compressors of the 4.5 K cycle [79], or oil-lubricated screw compressors operating at low suction pressure. In any case, compression at ambient temperature is hampered by the low density of the gaseous helium, which results in large volume flow-rates and thus requires large machinery, as well as in costly, inefficient heat exchangers for recovering enthalpy of the very-low pressure stream.

All these compressors are positive-displacement machines having volumetric characteristics. Screw compressors are routinely used in helium refrigeration and their implementation in a 1.8 K cycle therefore follows from current practice. Special attention however has to be paid to the protection against air inleaks: in particular the motor shaft and its rotary sealing must be located on the discharge side to operate above atmospheric pressure.

A first limit to the use of subatmospheric screw compressors stems from volumetric flow requirements: the biggest available machines have a swept volume of about 4600 m³/h, so that higher flow-rates require parallel arrays. Moreover, the isothermal efficiency - defined as the ratio of isothermal compression work to the effective compression work of the machine - decreases markedly with the suction pressure as shown in Figure 15, thus precluding their use at very low pressure in efficient process cycles.
Fig. 14 Subatmospheric compressors.

(a) combination of Roots and rotary-vane vacuum pumps  
(b) compound screw.

![Subatmospheric compressors](image)

Fig. 15 Isothermal efficiency of warm sub-atmospheric compressors.

3.3 “Cold” compression cycle

The alternative process is to perform compression of the vapour at low temperature, i.e. at its highest density. The pumps and recovery heat exchangers get smaller in size and less expensive, but the work of compression is then injected in the cycle at low temperature, so that the inevitable irreversibilities have a higher thermodynamic weight. Moreover, the pumping machinery which handles cold helium must be non-lubricated and non-contaminating, which seriously limits the choice of technology. Hydrodynamic compressors, of the centrifugal or axial-centrifugal type, have been used in large-capacity systems [80]. Their pressure ratio limited to 2 to 3.5 per stage however imposes to arrange them in multistage configurations [81, 82], thus narrowing the operational range of the system, in particular for startup or off-design modes.

Depending on the operating temperature (2 K or 1.8 K), the “cold” compression cycle requires at least 4 or 5 stages in series in order to perform the overall pressure ratio of 45 to 80. The compressed helium is directly returned to the cold low-pressure (LP) stream of the 4.5 K refrigerator.

The main drawback of this cycle concerns turndown capability. The cold compressor set has to guarantee the same pressure ratio for any load. A typical operating field for hydrodynamic...
compressors (Figure 16) displays the pressure ratio as a function of the reduced flow $m^*$ and the reduced speed $N^*$. The working area is limited on the left side by the stall line, on the right by the choke line and on top by the maximum rotational speed of the drive. At constant pressure ratio, the compressor can handle a flow reduction of only about 20% before reaching the stall line. Below 80% of nominal, additional vapour generation by electrical heating must be used to compensate for the load reduction. Such a cycle is therefore not very compliant to turndown, and its operating cost is not optimised for part-load operation.

This led CERN to conduct, in view of the LHC project, a R&D programme on cold compressors, procuring from specialised industry three prototype hydrodynamic compressors of different designs [83-86] to investigate critical issues such as drive and bearing technology, impeller and diffuser hydrodynamics, mechanical and thermal design, as well as their impact on overall efficiency [87]. The choices eventually retained for the LHC series machines [88-90] are 3-phase electrical induction motor drives working at room temperature with rotational speed varying from 200 to 700 Hz, active magnetic bearings working at room temperature, axial-centrifugal (three-dimensional) impellers and fixed-vane diffusers (Figure 17 & 18).

\[
\begin{align*}
\dot{m}^* &= \frac{\dot{m}}{\dot{m}_0} \sqrt{\frac{T_{in}}{T_{in_0}}} \frac{P_{in}}{P_{in_0}} \\
N^* &= \frac{N}{N_0} \sqrt{\frac{T_{in_0}}{T_{in}}} \\
\text{with:} \quad m &\text{: mass-flow} \\
T_{in} &\text{: inlet temperature} \\
P_{in} &\text{: inlet pressure} \\
N &\text{: rotational speed} \\
N_0 &\text{: design condition}
\end{align*}
\]

Fig. 16 Typical operating field of hydrodynamic compressor.

Fig. 17 Axial-centrifugal cold compressor cartridges for the LHC.
Fig. 18 Isentropic efficiency and typical cross-section of cold compressors.

Thermodynamic efficiency of cold compressors, determined by hydrodynamic design as well as by limitation of heat inleaks along the drive shaft, has significantly improved (Figure 18). Here the relevant estimator is isentropic efficiency, defined as the ratio of compression work in the adiabatic, reversible case, to the real one. Recent machines can be expected to reach 75 % isentropic efficiency at their design point.

3.4 “Mixed” compression cycles

For large systems, oil liquid-ring pumps or lubricated screw compressors may be used in series with cold compressors, in “mixed” compression cycles. Cold compressors are well-suited for the lower stages, while the presence of volumetric machines in the upper stages permits independent adjustment of flow-rate or wheel inlet conditions, thus improving load adaptation [91, 92].

In “mixed” compression cycles, the number of cold compressor stages can be reduced to 3, depending on the swept volume and number of warm sub-atmospheric machines. The compressed helium can be returned to the 4.5 K refrigerator at different levels:

- at the warm medium-pressure (MP) side (connection #1 on Figure 12 d). This requires the use of screw compressors having a sufficient built-in pressure ratio. In this case, the enthalpy of the gas at the outlet of the cold compressors has to be recovered by the heat exchangers of the 4.5 K refrigerator. The main advantage of this solution is that the same oil-removal and final cleaning systems can be used for the warm sub-atmospheric compressors and for the booster stages of the 4.5 K refrigerators, thus minimising the investment cost of the system.

- at the warm low-pressure (LP) side (connection #2 on Figure 12 d). This solution is compatible with the use of either screw compressors or liquid ring pumps. The enthalpy of the cold gas at the outlet of the cold compressors also has to be recovered by the heat exchangers of the 4.5 K refrigerator. In this case, the warm sub-atmospheric stage requires its own oil-removal system.

- at the cold low-pressure side (connection #3 on Figure 12 d). This is required when the enthalpy of the cold gas at the outlet of the cold compressors cannot be recovered by the heat exchangers of the 4.5 K refrigerators (LHC case) [88]. In this case, the warm sub-atmospheric stage requires its own oil-removal and final cleaning system (coalescers and charcoal adsorbers), increasing the investment cost.
The main advantage of the “mixed” cycle resides in its turndown capability. With sub-atmospheric compressors having volumetric characteristics, the pressure at the outlet of the cold compressors decreases linearly with the flow-rate, i.e. if the temperature and rotational speed do not change, the reduced flow-rate $m^*$ stays constant, thus keeping the working point fixed in the operating field. Such a cycle can then handle a large dynamic range, e.g. 3 for the LHC, without any additional electrical heating. Moreover, the total pressure ratio of the cold compressor train is lowered and the speed of some machines can then be reduced, thus decreasing the total compression power and operating cost.

Another operational advantage concerns the possibility of maintaining the load in cold standby with the cold compressors freewheeling and all compression performed, though at much reduced flow, by the warm machines. This mode allows repair or exchange of a cold-compressor cartridge without helium emptying of the system. In addition, the load adaptation provided by the warm volumetric machines proves very useful during transient modes like cool-down and pump-down, in which the cold compressors operate far from their design conditions.

The only drawback of this cycle concerns the risk of air inleaks due to the presence of sub-atmospheric circuits in air. Helium guards are recommended to prevent pollution of the process helium [93].

### 3.5 Application range of low-pressure helium compression techniques

The practical ranges of application of the different techniques appear in Figure 19, setting a *de facto* limit for warm compression above $20'000 \text{ m}^3\text{h}^{-1}$, or typically 300 W at 1.8 K. The diagram also illustrates the large span of refrigeration power and diversity of projects using superfluid helium. Investment and operating costs of large superfluid helium refrigeration systems can be assessed from basic thermodynamics and practical scaling laws derived from recent experience [94], thus providing input for technical-economical optimisation of such systems.

![Fig. 19 Range of application of low-pressure helium compression techniques.](image)

### 4. CONCLUSION

Operating superconducting devices at 1.8 K, using superfluid helium as a technical coolant, has now become state-of-the-art. The specific aspects of superfluid helium technology addressed in this article can be combined with standard cryogenic practice to design, build and operate complete industrial-
type helium II systems. The particle accelerator projects in construction or in study, however represent major challenges and opportunities for further progress, in view of their large size, complexity and quest for reliability and efficiency.

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