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SUPERFLUID HELIUM AS A TECHNICAL COOLANT

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Abstract

The characteristics of superfluid helium as a technical coolant, which derive from its specific transport properties, are presented with particular reference to the working area in the phase diagram (saturated or pressurised helium II). We then review the principles and scaling laws of heat transport by equivalent conduction and by forced convection in pressurised helium II, thus revealing intrinsic limitations as well as technological shortcomings of these cooling methods. Once properly implemented, two-phase flow of saturated helium II presents overwhelming advantages over the previous solutions, which dictated its choice for cooling below 1.9 K the long strings of superconducting magnets in the Large Hadron Collider (LHC), a 26.7 km circumference particle collider now under construction at CERN, the European Laboratory for Particle Physics near Geneva (Switzerland). We report on recent results from the ongoing research and development programme conducted on thermohydraulics of two-phase saturated helium II flows, and on the validation of design choices for the LHC cooling system.

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The characteristics of superfluid helium as a technical coolant, which derive from its specific transport properties, are presented with particular reference to the working area in the phase diagram (saturated or pressurised helium II). We then review the principles and scaling laws of heat transport by equivalent conduction and by forced convection in pressurised helium II, thus revealing intrinsic limitations as well as technological shortcomings of these cooling methods. Once properly implemented, two-phase flow of saturated helium II presents overwhelming advantages over the previous solutions, which dictated its choice for cooling below 1.9 K the long strings of superconducting magnets in the Large Hadron Collider (LHC), a 26.7 km circumference particle collider now under construction at CERN, the European Laboratory for Particle Physics near Geneva (Switzerland). We report on recent results from the ongoing research and development programme conducted on thermohydraulics of two-phase saturated helium II flows, and on the validation of design choices for the LHC cooling system.

1. INTRODUCTION

Sixty years after its discovery [1], helium II ("superfluid" helium 4) is still an advanced topic in condensed-matter physics: the archetype of a quantum fluid, it constitutes a simple physical system which permits clean investigation of such basic processes as phase transitions, evaporation/condensation kinetics, and turbulence [2]. Over the last two decades, however, superfluid helium has also become an efficient technical coolant for high-field superconducting devices, in laboratory experiments as well as in large research projects of industrial size. The rationale for using superfluid helium as a cooling fluid rests on two of its basic properties, namely the lower temperature of operation and the enhanced heat transfer at the solid-liquid interface, as well as in the bulk liquid.

The lower temperature of operation is exploited in high-field magnets [3, 4], 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 [5-8], magnetic confinement fusion [9-10] and more recently circular particle accelerators and colliders [11-13]. The latter are of large geographic extent and therefore demand efficient production and long-distance transport of high refrigeration capacity at low temperature, thus requiring to exploit the specific thermophysical properties of the fluid. In the case of high-frequency superconducting devices, such as acceleration cavities [14], 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 [15, 16] and future high-energy lepton colliders [17-19] operate in the temperature range which minimizes capital costs and overall energy consumption. This issue is schematised in Figure 2.
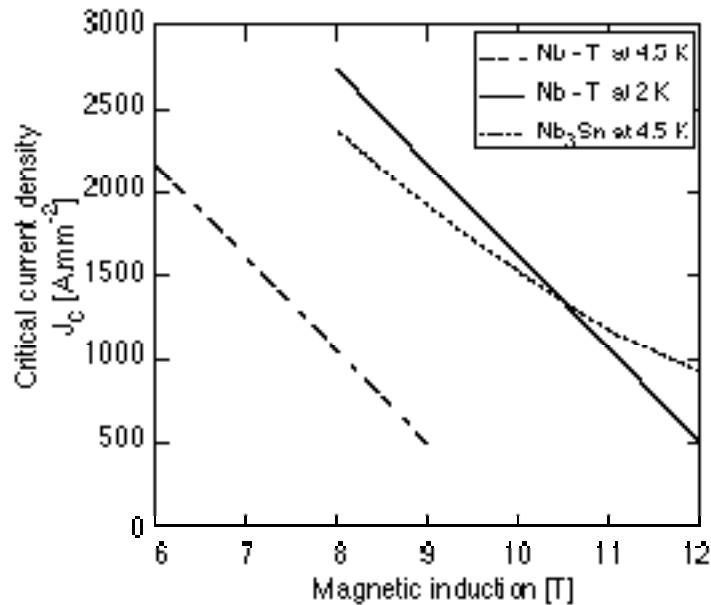
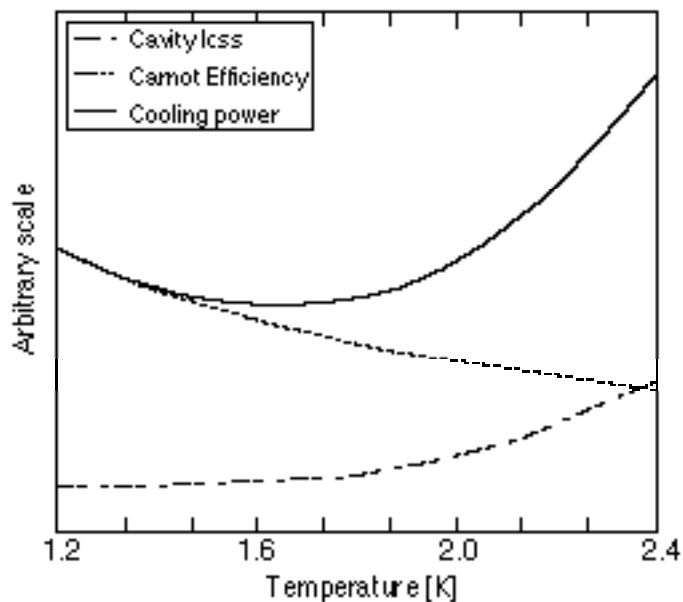Figure 1. Critical current density of technical superconductors

Figure 2. Optimal operating temperature of superconducting RF cavities

The technical heat transfer characteristics of superfluid helium basically derive from its peculiar transport properties, such as high heat capacity, low viscosity and good effective thermal conductivity [20]. In order to be fully exploited, however, both in steady-state and transient regimes, e.g. for large-capacity heat transport over macroscopic distances as well as intimate stabilisation of superconductors, they require elaborate thermohydraulic design of the cooling circuits, conductor, insulation and coil assemblies. This often conflicts with other technical or economic requirements of the projects, and acceptable trade-offs have to be found.

2. PRESSURISED 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 pressurised helium II, obtained by subcooling liquid at any pressure above saturation, and in particular at atmospheric pressure (100 kPa).
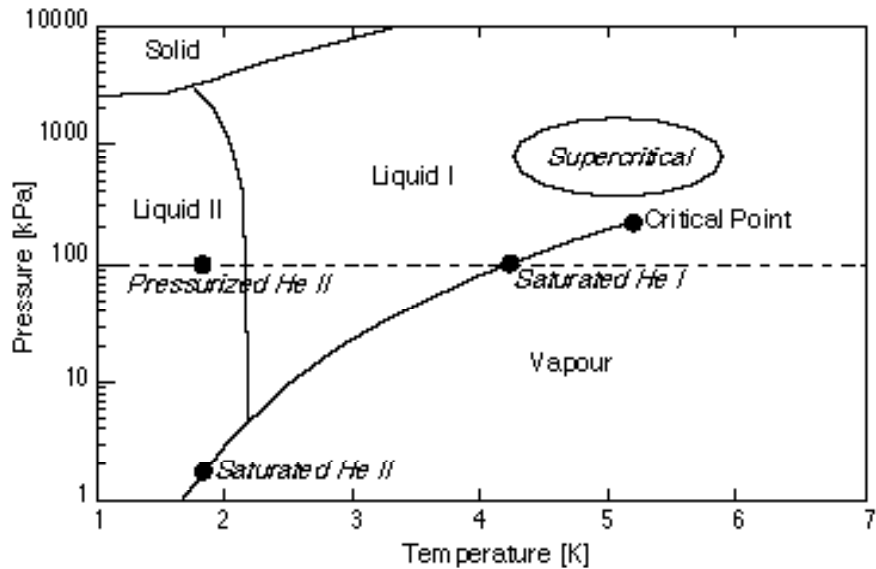
Figure 3. Phase diagram of helium

Although requiring one more level of heat transfer and additional process equipment - in particular a pressurised-to-saturated helium II heat exchanger - implementation of pressurised helium II for cooling devices brings several important technical advantages [21]. 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 [22], in the vicinity of the minimum of the Paschen curve [23], brings the additional risk of electrical breakdown at fairly low voltage. Operating in pressurised helium II avoids this kind of problem.

However, the most interesting and specific aspect of pressurised helium II in the operation of superconducting devices, stems from its capacity for cryogenic stabilisation. As a subcooled (monophase) liquid with high thermal conductivity, pressurised 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 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 [24], typically yields an order of magnitude better performance in favour of pressurised helium II.

3. 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 [25]. 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 theory [26], practical data useful for engineering design has been established in a number of experiments [27-32].

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:

$$q'^n L = X(T_c) - X(T_w) \tag{1}$$

3

where the best experimental fit for *n* is 3.4, and *X(T)* is a tabulated function of temperature, physically analogue to a conductivity integral [27]. A plot of this function in Figure 4 reveals that the apparent thermal conductivity of helium II goes through a maximum at around 1.9 K.
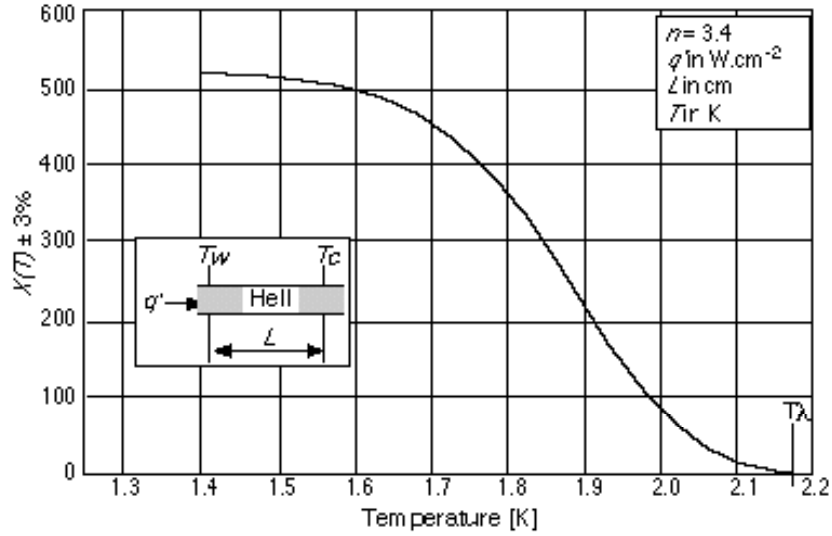

Figure 4. Thermal conductivity integral of pressurised superfluid helium [27]

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. 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 %.

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. 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:

$$q'_{total}{}^n L = (n+1) [X(T_c) - X(T_w)] \qquad (2)$$

where q'_{total} is the total heat flux flowing through the section at temperature T_c, near the heat sink. As an example, cooling a 50-m long cryomagnet string, with a uniform linear thermal load of 1 W.m^{-1}, by conduction between 1.9 K (temperature of the warmest magnet) and 1.8 K (temperature at the heat sink), 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 proposed for the LHC project [33] was later abandoned. It is however currently used for cryogenic testing of single prototype LHC magnets [34]. Inside the magnet windings, helium II conduction is also exploited locally to extract heat across the several layers of glass-fibre and polyimide tape tightly wrapped around the superconducting cables, which constitute their electrical insulation. Even in such a confined geometry, it has been shown [35] that the presence of minute conduction paths in superfluid helium strongly contributes to bypass the thermal impedance of the electrical insulation system, and thus helps stabilise the superconducting cable against thermal perturbations.

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 then permits to assess enthalpy changes of the system, and thus to perform calorimetric measurements. This technique proves very convenient for measuring minute heat inleaks [36], as well as energy dissipation [37] produced by ramping losses or resistive transitions in superconducting magnets.

4. FORCED CONVECTION OF PRESSURISED 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 1 cm. 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 heat transport Q' between two points 1 and 2 of the cooling loop is then given by the difference in enthalpy of the fluid flowing with a mass flow-rate m':

$$Q' = m' (H_2 - H_1) \tag{3}$$

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 3 above. Consider helium II pressurised 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 5) reveals a positive Joule-Thomson effect [38]: the enthalpy of the fluid increases both with increasing temperature and pressure, so that an isenthalpic expansion results in a temperature increase. For example, pressurised 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 [39].
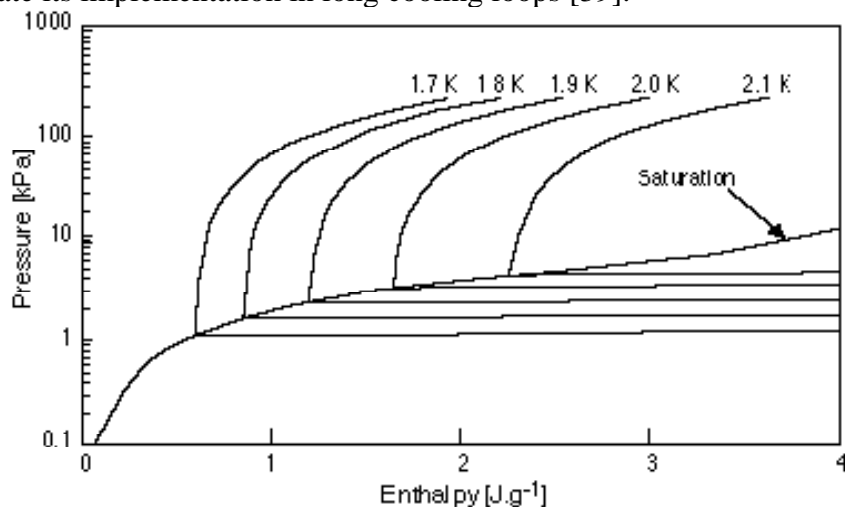

Figure 5. Pressure-enthalpy diagram of superfluid helium

Following early exploratory work [40, 41], several experimental programs have investigated heated flow of pressurised helium II in pipes and piping components [42, 43], culminating with the

230-m long test loop at CEA-Grenoble [44, 45] 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 [46, 47], and validated on experimental results. Pressure drop and heat transfer - both steady-state and transient - in flowing pressurised helium II may now be safely predicted for engineering purposes, using well-established laws and formulae.

The implementation of forced-flow cooling requires cryogenic pumps operating with pressurised 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 [48], the thermomechanical effect, specific of the superfluid state, may also be used for driving cooling loops by means of fountain-effect pumps [49-52]. In spite of their low thermodynamic efficiency [53], 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 [54]. At higher heat loads, they have been considered [55] and tested [56] 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 [57]. However, no large cryogenic system today relies on the use of superfluid helium pumps, which constitute a key technology - still under development and not yet mature - for the implementation of forced-flow convection cooling using pressurised helium II.

5. 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 pressurised helium II, up to a lumped pressurised-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 string of devices to be cooled. In this fashion the conduction distance - and hence the temperature drop - in pressurised 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, and schematised in Figure 6: the superconducting magnets operate in static baths of pressurised 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 vaporises [12].
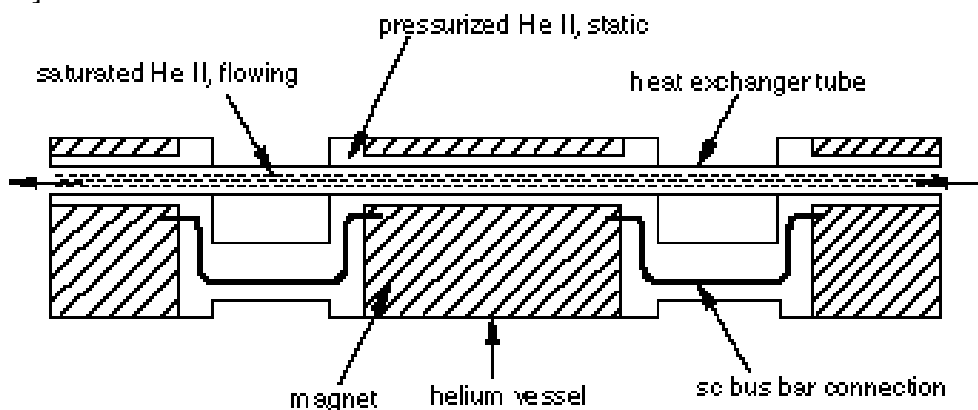

Figure 6. Principle of the LHC superfluid helium cooling scheme

Although potentially attractive in view of its efficiency for 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, at the time we proposed its use, no experimental data was available for two-phase flows of saturated helium II, and very little for other cryogenic fluids in this configuration. Following first exploratory tests [58] which demonstrated the validity of the concept on a reduced geometry, a full-scale thermohydraulic loop [59] 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 ND40 heat exchanger tube, made of industrial-grade deoxidised phosphorus copper.

To complete in-depth understanding of thermohydraulic processes occurring in co-current flow of two-phase saturated helium II in horizontal or slightly inclined tubes, a dedicated test set-up was built and operated at CEA-Grenoble (France). The 90-m long, 1.4 % slope, ND40 heated tube was comprehensively equipped with sensors and diagnostics, as well as with two Pyrex glass sections for observation of the flow patterns by means of CCD video cameras operating at cryogenic temperature [60]. Tests performed at vapour velocities of up to 10 m.s^{-1} showed that the flow remains stratified, and the pressure drop in good agreement with the predictions of a separated-flow model. Heat transfer measurements also show good agreement with the wetting of the tube wall predicted by the model, up to a vapour velocity of about 4 m.s^{-1}. At higher velocity, the measured heat transfer appears to improve, due to liquid droplet entrainment in the vapour stream [61]. For most practical purposes, and provided the vapour velocity remains below a few m.s^{-1}, the stratified flow model applies conservatively.

Once the wetting of the inner surface of the tube is guaranteed, the heat transfer from the pressurised 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. 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, largely dominates below 2 K [62]. The use of high-purity, cryogenic-grade copper such as OFHC is in general not required, and pressurised-to-saturated helium II heat exchangers can be made of DLP grade - "plumber's" copper.

The final validation of the two-phase helium II flow cooling scheme for LHC has been performed successfully on a 50-m long test string, equipped with full-scale prototype cryomagnets (Figure 7), operated and powered in quasi-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 pressurised helium II baths contributed to limit temperature excursions (Figure 8), 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].
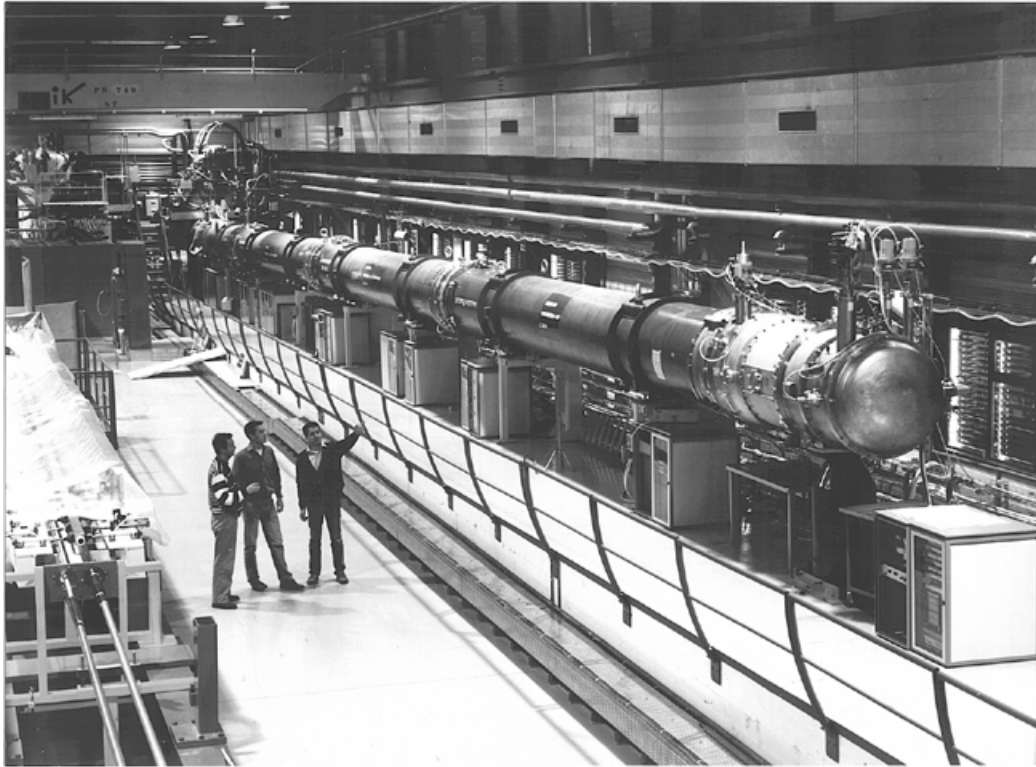
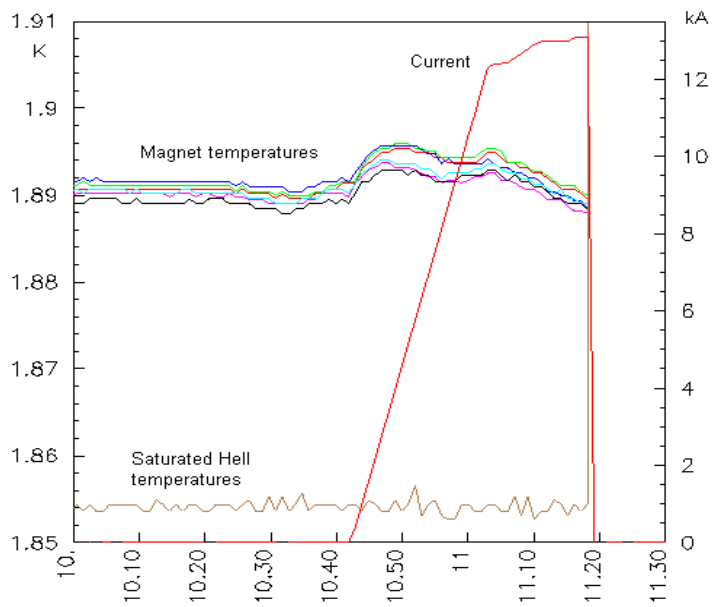Figure 7. General view of the LHC test string

Figure 8. Response of the LHC test string cooling system to a current ramp
(horizontal scale: time in hours, minutes)

7. SUPERFLUID HELIUM CRYOSTATS

The high thermodynamic cost of low-temperature refrigeration may only be rendered acceptable provided the heat loads at the 1.8 K level are tightly budgeted and contained. Setting aside the dynamic heat loads, produced by Joule heating, hysteresis and eddy-current losses, or interactions with the circulating particle beams, the base load in the cryogenic system of a large accelerator is the cryostat heat inleak, falling onto several kilometres (or tens of kilometres) of cold mass. It is therefore very important, in large projects, to design, construct and assemble reproducibly device cryostats with low residual heat inleak.

The structure of accelerator cryostats for devices operating in superfluid helium [66-70] (Figure 9) reflects these preoccupations. The device cold mass is supported and precisely positioned from the ambient-temperature vacuum vessel by post-type supports, made of non-metallic composites, with several levels of heat interception at intermediate temperatures [71]. In this fashion the residual heat inleak to the 1.8 K level can be kept more than two orders of magnitude smaller than the heat drawn by conduction along the posts from the ambient-temperature environment. The thermal performance achieved in practice, however, critically depends on the quality of the heat intercepts, i.e. on the ability to realise good solid-to-solid thermal contacts under vacuum.

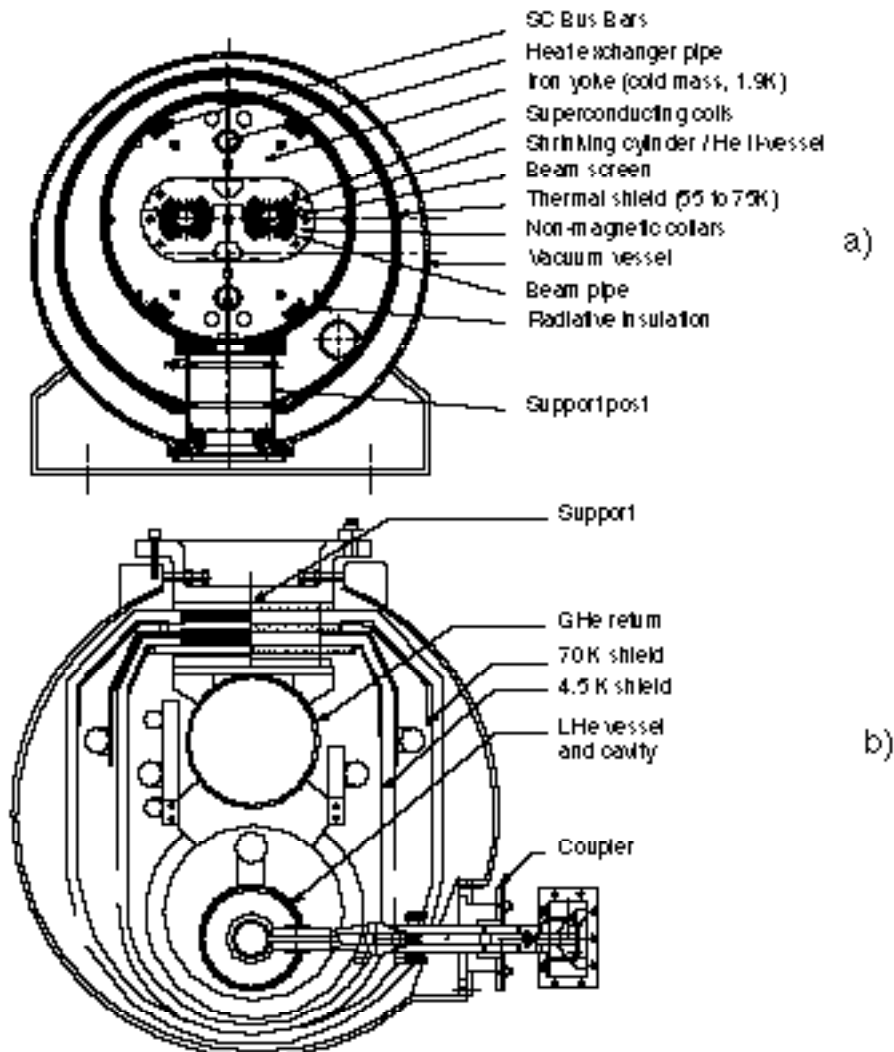Figure 9. Transverse cross-sections of superfluid helium cryostats for accelerator devices:
(a) LHC, (b) TESLA

Radiation and residual gas conduction in the evacuated insulation space are limited by one or two nested actively cooled shields, wrapped with multilayer reflective insulation. Although single low-emissivity surfaces are in principle sufficient to reduce the radiative heat inleak below 80 K,

the use of multilayer systems appears essential to provide robustness of performance against degradation of the insulation vacuum [72], a mishap bound to occur locally around the multi-kilometre circumference of a large accelerator. Thermalized multilayer insulation systems incorporating an actively cooled screen may also provide a means of further improving both nominal performance and robustness against vacuum degradation [73]. With proper industrial-scale implementation of such techniques, the distributed heat flux reaching the cold mass may then be kept well below 0.1 W.m^{-2} [74].

The experimental validation of cryostat performance requires precise calorimetric measurements, which may only be performed on well-instrumented, dedicated thermal models [75]. It is however satisfying to observe that industrially-produced cryostats reproducibly perform in accordance with budgeted heat loads [76]. It must also be noted that the leaktightness requirements of superfluid helium enclosures can be met by the standard joining and testing technology employed for helium I cryostats.

8. CONCLUSION

Operating large superconducting devices below 2 K, using superfluid helium as a technical coolant, has become state-of-the-art. Basic principles of heat transfer and dimensioning rules for helium II cooling systems are now well established, on the basis of recent research and development work. New projects presently in construction or in study, nevertheless represent major challenges in view of their large size, refrigeration capacity and quest for reliability and efficiency.

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