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Compact counter-flow cooling system with subcooled gravity-fed circulating liquid nitrogen

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ABSTRACT

A liquid nitrogen (LN2) is usually used to keep the high-temperature superconducting (HTS) cable low temperature. A pump is utilized to circulate LN2 inside the cryopipes. In order to minimize heat leakage, a thermal siphon circulation scheme can be realized instead. Here, we discuss the effectiveness of thermal siphon with counter-flow circulation loop composed of cryogen flow channel and inner cable channel. The main feature of the system is the existence of essential parasitic heat exchange between upwards and downwards flows. Feasibility of the proposed scheme for cable up to 500 m in length has been investigated numerically. Calculated profiles of temperature and pressure show small differences of T and p in the inner and the outer flows at the same elevation, which allows not worrying about mechanical stability of the cable. In the case under consideration the thermal insulating properties of a conventional electrical insulating material (polypropylene laminated paper, PPLP) appear to be sufficient. Two interesting effects were disclosed due to analysis of subcooling of LN2. In case of highly inclined siphon subcooling causes significant increase of temperature maximum that can breakup of superconductivity. In case of slightly inclined siphon high heat flux from outer flow to inner flow causes condensation of nitrogen gas in outer channel. It leads to circulation loss. Results of numerical analyses indicate that counter-flow thermosiphon cooling system is a promising way to increase performance of short-length power transmission (PT) lines, but conventional subcooling technique should be applied carefully.

1. Introduction

A number of projects of HTS PT lines have been successfully implemented in recent years. This advance makes it possible to begin to construct of industrial hundreds-meter class lines right now. However, standard technology is still far from perfect despite the significant achievements in enhancing the operational properties of HTS tapes and cables. The main difficulty is the lack of efficiency of the HTS cable cooling system. Typically, LN2 is used as a refrigerant, and cryopumps are utilized to circulate LN2. However, existing apparatus are not optimized for this goal. They show up an additional large heat load on the system, a large hydraulic resistance and require power supply. Moreover, existing models of cryopumps are very expensive. Therefore, our group develops several approaches to improve cooling systems. In particular, the experimental facility in Chubu University uses smooth cryopipes separated with short bellows segments to reduce the pressure drop and pump power, instead of standard entirely corrugated cryopipes. An active thermal shutters based on the Peltier effect are used to reduce the penetration of heat through the current leads inside the cryostat [1]. We believe that the technological breakthrough in the field of the HTS system optimization may be achieved by implementation of the natural circulation of LN2 and, as a result, a complete rejection of the cryopump installation.

A preliminary set of specifications for exploring the possibility of cooling HTS cable using thermosiphon effect was done several years ago by Yamaguchi [2]. Thermal siphon is a simple device wherein working fluid moves under the action of gravitational forces due to thermally generated density difference in communicating pipes. A variety of machines based on thermosiphon principle are widely used in industry due to their low cost, relative simplicity and reliability. However, the operational capability of the thermal siphon may be limited, depending upon the relative magnitude of driving forces with respect to hydraulic resistance inside the circulation loop [3,4]. A conceptual design of the HTS cable cooling system is shown in Fig. 1. It is a canonical U-shaped thermal siphon inclined depending on the cable run. The system is composed of LN2 reservoir placed at the top of the circuit, and two cryopipes, connected at its bottom. In the case of open siphon

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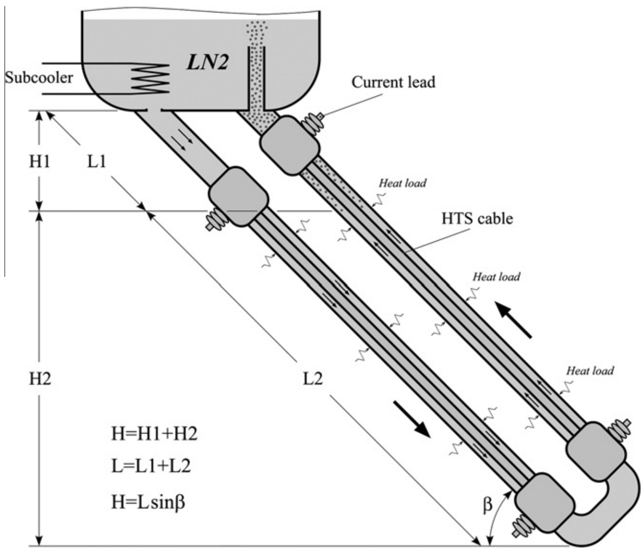

Fig. 1. Normal configuration of the thermosiphon HTS cable cooling system.

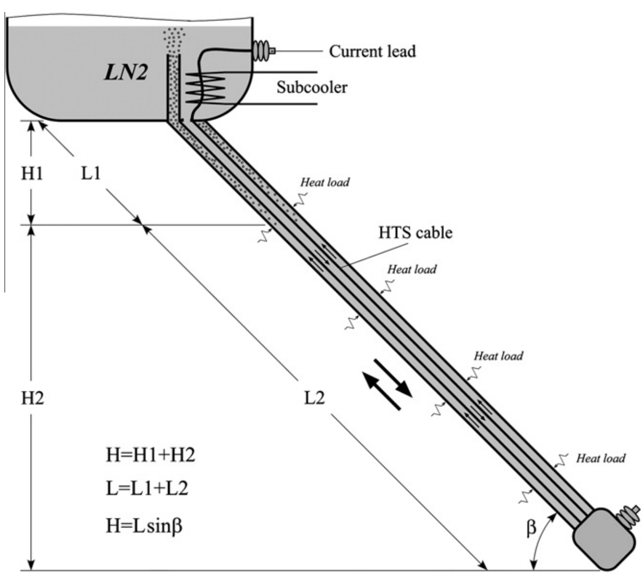

Fig. 2. Compact configuration of the thermosiphon HTS cable cooling system.

the reservoir has to contain the coolant at the atmospheric pressure $p_0 = 0.1013$ MPa, and the boiling temperature at the inlet is $T_0 = 77.35$ K. It is required that the vaporization happens only in the second channel, significantly reducing the average density of the two-phase fluid. This can be easily reached by maintaining a sufficiently low heat load to prevent temperature rise from exceeding the boiling temperature as a function of local pressure. The pressure at the bottom point of the thermal siphon is defined by the hydraulic head above it, and the pressure drop in the first line. There is no boiling here, and the flow goes up the hill absorbing the incoming heat due to the thermal capacity of the coolant at a rate defined by its mass flow. At some point the temperature and the pressure of LN2 reach the boiling point. That is where the two-phase flow begins. Above this point the temperature and the pressure correspond to the saturation curve. The flow returns LN2 into the atmospheric pressure reservoir at the top of the loop. Nitrogen vapor is vented to the atmosphere, while liquid is returned to the inlet of the cooling circuit. The analysis of HTS PT line with the length of up to 100 km, and the up/down-hill elevations varying by up to 2 km, which is able to be built in a mountainous terrain confirmed the feasibility of the similar arrangement [5]. Recently, we proposed a one more version of the design [6]. Referenced paper was addressed to the effectiveness of open thermal siphon with coolant flows down through the inner channel of the superconducting cable, instead of through a separate supply pipeline as in the conventional system (Figs. 2 and 3). The main features of this system are as follows: the hydraulic resistance in the narrow inner channel is high, the difference of hydraulic cross-sections of inner and outer channels is significant, and parasitic heat exchange between upwards and downwards flows due to a relatively high thermal conductivity of electrical insulating material is essential. Our analysis confirmed that the siphon can operate even without subcooling; it is enough to compensate the loss of LN2 instead of evaporated one [6]. It is also pertinent to note that the similar vertical long thermal siphons are applied to extract of geothermal energy [7]. The evaluation of heat exchange in HTS cable was done, for example, in [8,9].

2. Model

Present paper is devoted to the efficiency of subcooling using to improve the performance of the open thermal siphon cooling system shown in Fig. 2. In order to properly simulate the circulation process of LN2 geometrical parameters of the cable and the cryopipe were selected in accordance with geometry of 20 m experimental facility in Chubu University [1,6]. Namely, I.D. of the inner cable channel was 14 mm, O.D. of the cable was 40 mm, and I.D. of the inner cryopipe was 80 mm as shown in Fig. 3. Heat conductivity of LN2-impregnated PPLP was prescribed to be 0.2 W/m K, while reported values vary over the range 0.05–0.2 W/m K [10]. Anyway, overall heat transfer coefficient k can be controlled by varying thickness of insulation layer. It was assumed that the distributed heat load is 1 W/m, and the loss of heat through the end current lead is 50 W. The steady-state design equations were obtained from Bernoulli's formula

$$p + \rho g h + \rho \frac{v^2}{2} = const \tag{1}$$

and heat balance conditions. Equations for the upper part (L1) are as follows:

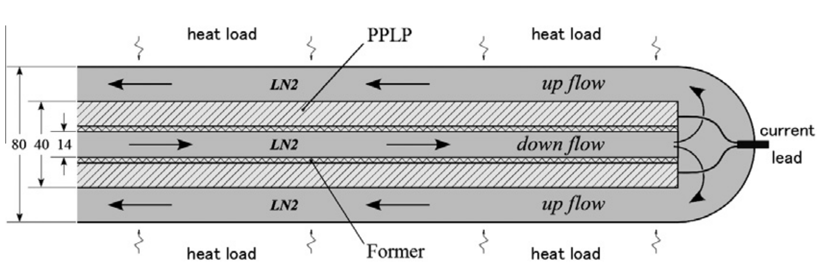

Fig. 3. Simplified model of HTS cable with counter-flow cooling. The geometry was prescribed to be the same as of the test facility of Chubu University [1].

$$\begin{cases} \dot{m}C_{pl}\frac{dT_1}{dx} = k(T_2 - T_1) \\ \dot{m}[(1-\chi)C_{pl} + \chi C_{pg}]\frac{dT_2}{dx} + r\dot{m}\frac{d\chi}{dx} = k(T_2 - T_1) - q \\ \frac{dp_1}{dx} = \rho_l g \sin\beta - \frac{2f_1 G_1^2}{\rho_l D_{h1}} \\ \frac{dp_2}{dx} = \left(\frac{\chi}{\rho_g} + \frac{1-\chi}{\rho_l}\right)^{-1} g\sin\beta + \frac{2f_2 G_2^2}{\rho_l D_{h2}} \frac{1+(\rho_l/\rho_g - 1)\chi}{[1+(\mu_l/\mu_g - 1)\chi]^{0.25}} \\ T_2 = T_{sat}(p_2) \end{cases} \quad (2)$$

and for the lower part (L2) are as follows:

$$\begin{cases} \dot{m}C_{pl}\frac{dT_1}{dx} = k(T_2 - T_1) \\ \dot{m}C_{pl}\frac{dT_2}{dx} = k(T_2 - T_1) - q \\ \frac{dp_1}{dx} = \rho_l g \sin\beta - \frac{2f_1 G_1^2}{\rho_l D_{h1}} \\ \frac{dp_2}{dx} = \rho_l g \sin\beta + \frac{2f_2 G_2^2}{\rho_l D_{h2}} \end{cases} \quad (3)$$

where C_p is the specific heat, J/kg K; D_h is the hydraulic diameter, m; f is the friction factor; g is the gravitational acceleration, m/s^2; G is the specific mass flow, kg/m^2 s; k is the overall heat transfer coefficient, W/m K; $\dot{m}$ is the mass flow, kg/s; p is the pressure, Pa; q is the heat load, W/m; r is the specific heat of vaporization, J/kg; Re is the Reynolds number; $\sin\beta$ is the sine of inclination; T is the temperature, K; $T_{sat}(p)$ is the saturation curve, K; μ is the dynamic viscosity, Pa s; ρ is the density, kg/m^3; and χ is the vapor quality. Subscripts l and g denote liquid and gas, respectively; subscripts 1 and 2 denote inner and outer channels, respectively. The temperature and pressure dependences of the physical properties of the nitrogen are taken into account. In case of cylindrical geometry the overall heat transfer coefficient k is given by

$$k = 2\pi\left(\frac{1}{h_I r_I} + \sum_i \frac{\ln(r_i + 1/r_i)}{\lambda_i} + \frac{1}{h_{II} r_{II}}\right)^{-1} \quad (4)$$

where h is the surface heat transfer coefficient, W/m^2 K; r is the radius, m; λ is the heat conductivity of cable material, W/m K. Subscripts I and II denote internal (channel) and external surfaces of the cable, respectively; subscript i denotes ith layer of cable material. A contribution of layers with high thermal conductivities λ_i (HTS tapes, copper former, etc.) can be neglected. Therefore, second term can be written in short as $\ln(r_2/r_1)/\lambda$, where alone PPLP layer is taken into account. Terms $1/h_I r_I$ and $1/h_{II} r_{II}$ are also negligible compared to $\ln(r_2/r_1)/\lambda$. Final expression for k becomes:

$$k = 2\pi\frac{\lambda}{\ln(r_2/r_1)} \quad (5)$$

where λ is the heat conductivity of PPLP, W/m K; r_1 and r_2 are inner and outer radii of equivalent PPLP layer, respectively. It was assumed that r_1 = 24 mm and r_2 = 40 mm. In case of one-phase turbulent flow friction factor is given by

$$f = 0.079Re^{-0.25} \quad (6)$$

where Re is the Reynolds number. Frictional pressure gradient for two-phase flow is determined by multiplying f of the liquid by a so called two-phase multiplication factor. A typical correlation for the two-phase multiplication factor is as follows:

$$\Phi_{lo}^2 = \frac{1 + (\rho_l/\rho_g - 1)\chi}{[1 + (\mu_l/\mu_g - 1)\chi]^{0.25}} \quad (7)$$

Correlations described by Eqs. (6) and (7) are already entered into Eqs. (2) and (3) in explicit form. It should be noted that acceleration pressure drop was omitted from consideration because of estimated value was several Pa per 500 m cable. To solve the systems of design equations a numerical fourth-order Runge–Kutta method was used.

3. Analysis and results

Analysis revealed that refrigerant flow is turbulent over the entire range of lengths (100–500 m) and angles (0–90°). One-phase flow through the wide outer channel is characterized by Reynolds numbers within the range of 4000–5000, and through the inner narrow channel by Re = 32,000–69,000. An area with two-phase flow near the outlet of outer channel is characterized by Re = 5000–11,000. This is because vapor quality (χ) does not exceed 3.8%. Mass flow ($\dot{m}$) is relatively small being within the range between 0.056 and 0.087 kg/s. Calculated profiles of temperature and pressure revealed small differences of T and p in the inner and the outer flows at the same elevation, which allows not worrying about mechanical stability of the cable (Fig. 4). Subcooling stimulates parasitic heat exchange between upwards and downwards flows. Results of the series of calculations allowed us to identify two technologically important cases.

I. Vertical (and highly inclined) siphons can be used to supply electrical equipment in high-rise buildings. In this case, a large pressure gradient leads to the fact that even at high heat loads LN2 boiling begins only near the outlet and heat transfer through electrical insulation has no time to significantly affect on the distribution of χ. Subcooling leads to an almost linear decrease in mass flow and vapor quality at outlet against ΔT. At the same time, subcooling causes an increase in the average temperature gradient along the cable, and an increase in T_{max}. As shown in Fig. 5 for the

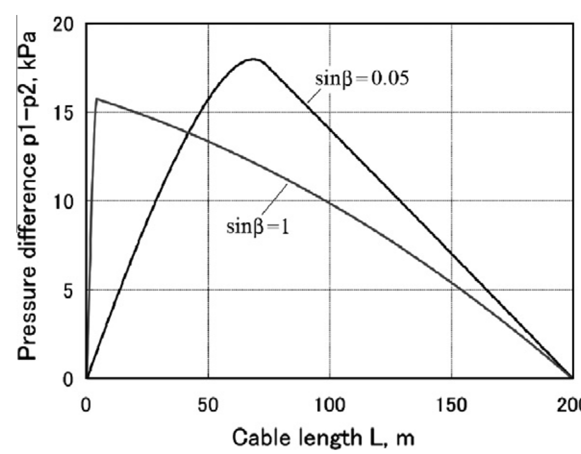

Fig. 4. Pressure difference between inner and outer LN2 channels at the same elevation for 200 m siphon.

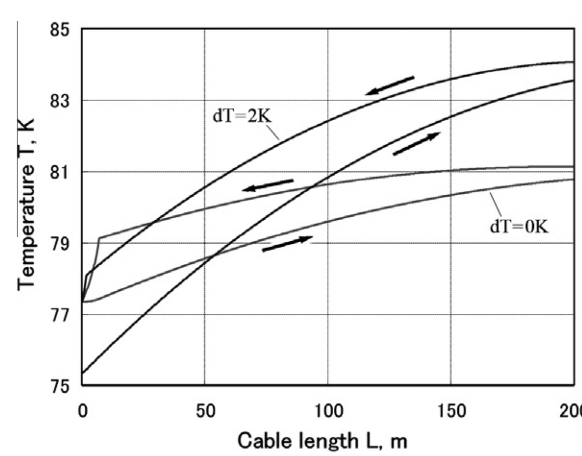

Fig. 5. Temperature profiles for 200 m vertical ($\sin\beta$ = 1) siphon with and without subcooling.

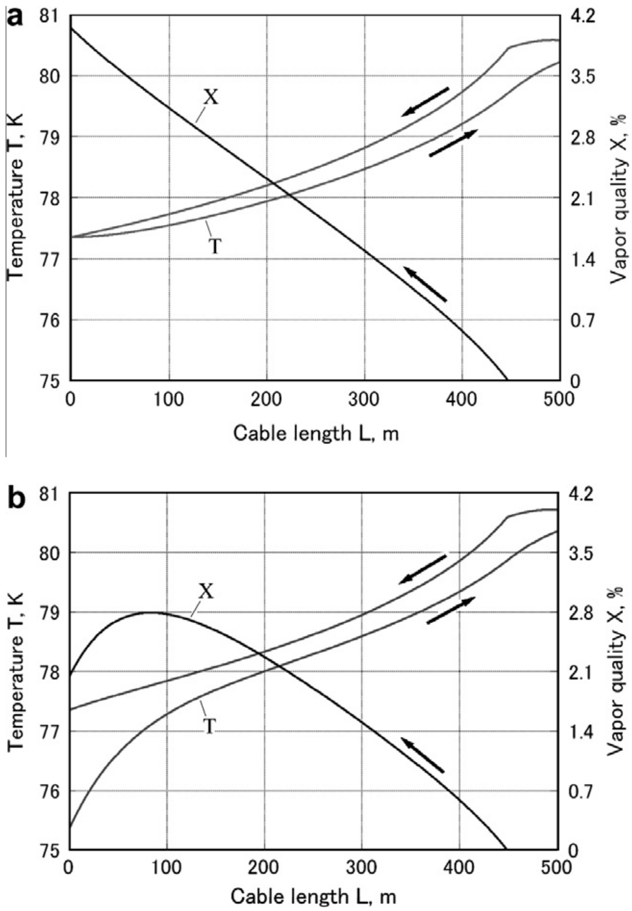

Fig. 6. Temperature profile (loop curve) and vapor quality for slightly (sin β = 0.04) inclined 500 m siphon: (a) without subcooling and (b) with 2 K subcooling.

200 m vertical siphon, subcooling by 2 K raises the temperature maximum by 3 K. Unlike the conventional system with a separate supply channel wherein the maximum temperature is achieved at the initial boiling point, in the case under consideration T_{max} can be at some point of the upward flow. The position of this point depends on length and slope of the cable. That can lead to overheating of the superconductor and requires obligatory hardware control of T in the areas of the cryostat, far from the outlet.

II. Slightly inclined siphon can be useful for in-plant and Internet Data Center applications [11]. Most of the outer channel is filled with liquid–gas mixture which is at saturation because of the small $\dot{m}$ due to the small I.D. of the inner channel and a small driving force (proportional to sin β), so the boiling begins near the far end of cryopipe (and in some cases just from the terminal current lead). Subcooling virtually does not affect on T_{max} and the distribution of T changes significantly only in the initial part, where large difference $T_2 - T_1$ between the channels exists. Undesirable effect manifests itself in this place. Large outflow of heat (defined by $T_2 - T_1$) from the outer two-phase fluid becomes so high that the reverse process, i.e. the partial condensation of gaseous nitrogen begins. This leads to an increase of the average density of the medium in the upper part of the channel and, therefore to loss of circulation (Fig. 6). In principle, the effect can be somewhat weakened by increasing the thermal resistance of insulator, but it is difficult and expensive to be implemented. Therefore, although subcooling reduces the consumption of LN2 being lost in the form of gas, it can critically disrupt the functionality of the system, if incoming LN2 will subcooled by more than 1–2 K.

4. Conclusions

Results of the numerical analyses indicate that the cooling system with counter-flow thermosiphon is a promising way for increasing performance of short-length HTS PT lines. Subcooling by 1–2 K can be used to reduce the consumption of LN2, but it requires obligatory hardware control of T. It seems the technologically advanced one-phase thermal siphon have to be analyzed from the same point of view in future.

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