Int. J. Heat Mass Transfer. Vol. 20, pp. 31–40. Pergamon Press 1977. Printed in Great Britain

POOL HEAT TRANSFER TO LIQUID AND SUPERCRITICAL HELIUM IN HIGH CENTRIFUGAL ACCELERATION FIELDS

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(Received 29 December 1975 and in revised form 5 April 1976)

Abstract—In relation to the problem of cooling the rotating field windings of a superconducting alternator, a preliminary experimental study of pool heat transfer to helium under high speed rotation is reported. Heat transfer from a flat copper surface has been measured for a range of heat flux 0.2–650 mW cm^{-2} and in local centrifugal acceleration fields of 180–1720 g perpendicular to the surface. Analysis of the data indicates that the mode of heat transfer by free convection at both subcritical and supercritical pressures follows the simple correlation $Nu = 0.29\,Ra^{0.29}$. At subcritical pressures, nucleate boiling is an additional process at sufficiently high heat fluxes.

NOMENCLATURE

a, acceleration field [m·s^{-2}];
g, earth's gravitational acceleration field [m·s^{-2}];
G, mass flux [g·cm^{-2} s^{-1}];
h, heat-transfer coefficient [W·m^{-2} K^{-1}];
h_{fg}, liquid–vapour latent heat [J·kg^{-1}];
k, thermal conductivity [W·m^{-1} K^{-1}];
L, duct length [m];
p, fluid pressure [Pa];
q, heat flux [W·m^{-2}];
r, duct radius [m];
R, electrical resistance of thermometers [Ω];
T, temperature [K];
w, flow velocity magnitude in duct [m·s^{-1}];
z, longitudinal co-ordinate of the duct.

Dimensionless groups

Pr, Prandtl number, ν/α;
Re, Reynolds number, wr/ν;
Gr, Grashof number based upon duct radius, $\dfrac{a\beta r^3 \Delta T}{\nu^2}$;
Ra, Rayleigh number, $Gr \times Pr$;
Nu, Nusselt number, $\dfrac{h}{k} r$;
a/g, ratio of centrifugal to gravitational acceleration.

Greek symbols

α, thermal diffusivity [m^2 s^{-1}];
β, volume expansivity [K^{-1}];
Δ, increment;
ν, kinematic viscosity [m^2 s^{-1}];
ρ, mass density [kg·m^{-3}];
σ, surface tension [N·m^{-1}];
Ω, rotational angular frequency [rad·s^{-1}].

Subscripts

A, ambient values;
B, bulk fluid properties;
b, bubble properties;
l, liquid properties;
NC, natural convection;
sat, saturation properties;
T, total;
v, vapour properties;
w, wall properties.

Arbitrary constants

Where necessary arbitrary constants are defined in the text.

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INTRODUCTION

HELIUM, in a liquid, vapour, or supercritical state, will be required as a coolant in the rotating frame for the rotating field windings of proposed superconducting alternators [1–3]. Heat transfer to helium at the rotor periphery at temperatures between 4 and 10 K under centrifugal acceleration fields up to 5000 g will be required either under conditions of forced flow in a system of rotating ducts or to a rotating pool of fluid.

The preliminary experiments described in this paper are concerned with free convection and boiling heat transfer to liquid and supercritical helium contained in a single duct which rotates about an axis perpendicular to its length and connects with a reservoir of unheated fluid. This rotating "open thermosyphon" system has been investigated as a possible primary means of removing heat from the field windings at the periphery of a rotor and rejecting it to a heat sink on or near the rotor axis.

The rotating open thermosyphon was proposed by Holzwarth [4] and Schmidt [5] as a means of cooling the blades of high temperature gas turbines. A comprehensive review of the rotating open thermosyphon system has been published by Bayley and Martin [6] and some of their conclusions are relevant to the results for liquid helium described in this paper.

For a free convection mode of heat transfer in the open thermosyphon, experimental data have generally been used to obtain correlations of the form

$$Nu = Nu(Pr, Gr, r/L) \qquad (1)$$

or

$$Nu = Nu(Ra) \qquad (2)$$

where $Ra = Gr \times Pr$ and r/L is the radius to length ratio for the thermosyphon. These correlations have been applied under the condition of uniform heat flux at the boundary wall over the whole length of the thermosyphon. In studies of the rotating thermosyphon, the limited amount of experimental data [7, 8] again refers to heat transfer via distributed wall heating.

This paper is concerned with heating at the closed end only. Furthermore, in the case of helium, the changes in fluid property, such as density and thermal conductivity, are considerable along the length of the duct. At the same time, the viscosity of helium is very low, so that very high Grashof and Rayleigh numbers apply in free convection processes.

As far as can be ascertained, no experimental data exist for rotating thermosyphons heated at the closed end only. Some of the data obtained for turbine blade cooling have been described by Jakob [9]. He also discussed the effect of mixing between heated and unheated fluid streams upon the free convection correlation.

While closed end heating in a stationary vertical open thermosyphon results in strong mixing at sufficiently high Rayleigh numbers [10], later work by Martin [11] with a tilted open thermosyphon at Rayleigh numbers up to 10^8, suggested that Coriolis acceleration may act so as to demarcate heated and unheated fluid streams in a rotating system. He observed that, as the Rayleigh number was increased, the onset of a fully mixed turbulent flow regime with a vertical open thermosyphon was delayed by tilting from the vertical. The turbulent heated and unheated fluids separated, with the heated fluid being confined to the "leading" edge of the duct.

The helium filled rotating thermosyphon can therefore be expected to exhibit a behaviour which may not correlate very closely with data pertaining to other fluids—only qualitative comparisons can be drawn from this preliminary study. Another interesting feature is that at sufficiently high heat fluxes and for helium at subcritical pressures, subcooled nucleate pool boiling from the closed end of the rotating open thermosyphon becomes an additional mode of heat transfer. The improvement in heat transfer after the onset of boiling appears to be independent of the free convection mechanism. The data for the boiling region may be correlated in terms of forced convection associated with bubble agitation. As expected above a particular speed of rotation, the additional heat transfer due to boiling disappears when the local fluid pressure at the closed end exceeds the critical value.

2. APPARATUS

2.1. *The rotating cryostat*

The heat-transfer experiments were performed using a rotating helium cryostat, a section drawing of which is shown in Fig. 1. The helium space comprises a cylindrical liquid reservoir on the axis of rotation to which are attached two radial ducts. These ducts are

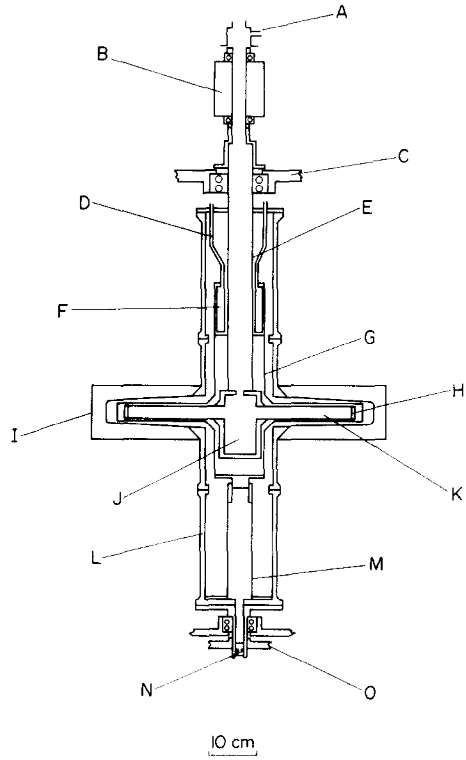

FIG. 1. Section drawing of the rotating cryostat. (A) rotating gas seal; (B) 8-channel slip ring; (C) Bearing support; (D) Liquid N_2 fill tubes; (E) Liquid helium fill-tube; (F) Liquid N_2 reservoir; (G) Radiation shield; (H) Heat-transfer "cell"; (I) Drag reduction shield (enclosing radial arms); (J) Helium reservoir; (K) Radial helium ducts; (L) Vacuum container; (M) Support tube for radiation shield; (N) Vacuum seal-off valve; (O) Drive pulley.

made of thin-wall (0.35 mm) stainless steel, have closed outer ends, and open into the reservoir. A heat-transfer "cell", Fig. 2, containing a heater and thermometers, is located at the outer closed end of one of the ducts. The helium space, including the fill-tube, forms an inverted "T"-shape which is enclosed by a liquid nitrogen cooled thermal radiation shield. The rotating cryostat is mounted in a frame which can gimball so that the axis of rotation of the cryostat can be made either vertical or horizontal. For static heat-transfer measurements, the axis is horizontal and the heat-transfer surface is arranged to face vertically upwards in the "lower" radial arm. During rotation liquid helium fills the radial ducts to a maximum distance of 233 mm from the rotational axis. Since the rotational frequency of the cryostat is variable up to 44 Hz a maximum acceleration field of $1.69 \times 10^4\,\mathrm{m\,s^{-2}}$ (1720 g) is produced at the experimental heat-transfer surface.

The helium cryostat has been designed to have a low liquid boil-off rate under rotation by the use of a

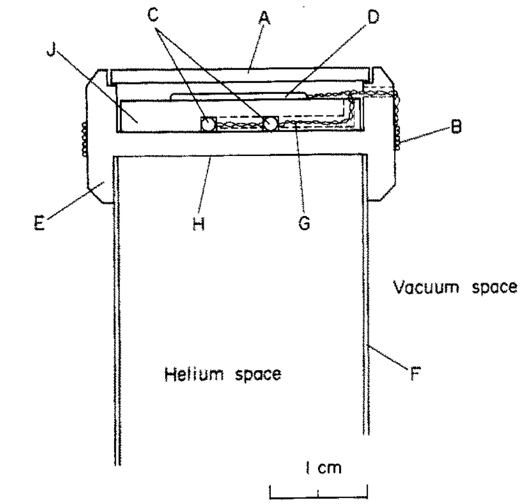

FIG. 2. The heat-transfer "cell". (A) Copper end-plate; (B) Thermal anchorage of thermometer leads; (C) Carbon resistors (CRT's); (D) Constantan heater; (E) Copper end-cap; (F) Stainless steel duct, 0.35 mm wall thickness; (G) Channel for CRT leads; (H) Heat-transfer surface; (J) Copper mounting block for CRT's.

convection baffle system in the fill-tube [2]. This system allows filling with liquid nitrogen (radiation shield) and liquid helium (experimental reservoir) while the cryostat is stationary and gives approximately 2 h of running at full speed in the absence of heater power. The cryostat was designed to have a critical speed in excess of 80 Hz and no vibration problems were experienced. Static balancing of the rotating parts was done with extreme care and no further dynamic balancing was needed.

2.2. *The heat-transfer cell*

A cross section of the heat-transfer "cell" is shown in Fig. 2. The temperature of the OFHC copper heat-transfer surface was taken as the average over a 3 mm thickness perpendicular to the direction of heat flow. The consequent uncertainty in the true surface temperature was estimated to range from 0.02 mK at the lowest level of heat flux (0.2 mW cm^{-2}) up to 50 mK at the maximum level of heat flux (650 mW cm^{-2}). The error in ΔT, the temperature excess of the surface over the unheated fluid, was estimated to be less than 1% over the whole range of heat flux.

2.3. *Experimental details*

The average temperature of the 3 mm thickness of copper was measured by either one of two calibrated Allen–Bradley 220 Ω 1/8 W carbon resistors. Using a stabilised 1 μA DC supply, the resistance of the sensors was determined using 4 lead potentiometry, electrical connections to the rotor being via an 8-channel precision slip ring assembly. The overall resolution of the thermometry system was ± 2 mK at 5 K. The rotating cryostat was not suitable for carrying out a full calibration of the thermometers and so prior to each experimental run a previously determined mean calibration temperature-resistance curve of the form [13]

$$1/T = a/\ln R + b + c \ln R \qquad (3)$$

where a, b and c are constants, was adjusted to pass through a single fixed point (the boiling point of helium under the ambient pressure). The resultant uncertainty in absolute values of temperature was estimated to be ± 4 mK. In measuring ΔT however, the same thermometer registers the fluid temperature in the absence of heater power and then the surface temperature for a given heat flux. Used differentially in this manner the thermometer was capable of measuring ΔT to an estimated accuracy of ± 2 mK. Three full temperature-resistance calibrations of the carbon resistors on completion of the experiments revealed no systematic shifts in sensor characteristics as a result of high speed rotation.

The heater power was variable and stability at each setting was better than 0.1% over the duration of a measurement. Heat fluxes at the heat-transfer surface were subject to a correction to allow for thermal leakage by conduction down the wall of the stainless steel duct. At the lowest limit of heat flux employed (0.2 mW cm^{-2}) these corrections were calculated, according to a heat leakage model discussed in (14), to be as high as 10% of the total. They diminished to less than 5% at heat fluxes greater than 20 mW cm^{-2}. The heat leakage correction was probably the largest single source of error in determining the heat-transfer coefficients.

The following experimental cases have been investigated, all involving pool heat transfer from a flat, copper surface:

(1) Heat transfer to saturated liquid helium under conditions of standard gravity with the heat transfer surface facing upwards and perpendicular to the gravitational acceleration field (i.e. no rotation, and rotation axis horizontal).

(2) Heat transfer to sub-cooled liquid helium (i.e. pressures below critical) under acceleration fields between 184 and 1000 g perpendicular to the surface.

(3) Heat transfer to helium at supercritical pressures under acceleration fields up to 1720 g perpendicular to the surface.

3. RESULTS AND DISCUSSION

3.1. *The question of isentropic, unheated flow to the periphery*

Before presenting and discussing the heat-transfer results, it is important to justify the assumption made in the analysis that the fluid reaching the heat-transfer surface has only suffered isentropic compression and has not become mixed with heated fluid leaving the surface. In fact there are several pieces of evidence from these experiments which support this assumption. In one experiment, the fluid temperature at the heat-transfer surface was monitored as a function of rotational angular frequency Ω for zero heater power. The resulting temperatures are shown in Fig. 3 as a function of the corresponding local fluid pressures at the heat-transfer surface calculated from the expression

$$p = p_A + \Omega^2 \int_{z_0}^{z_1} \rho z \, dz \qquad (4)$$

where p_A is the known ambient pressure acting on the

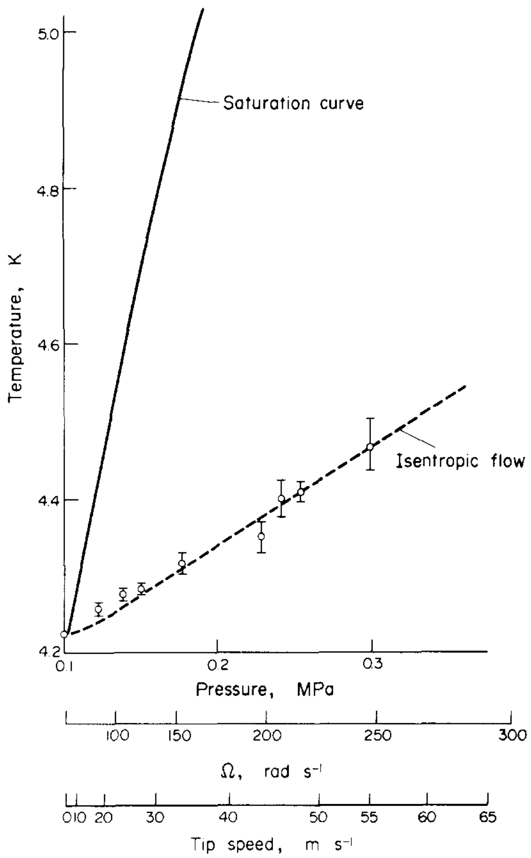

FIG. 3. Measured fluid temperature at heat-transfer surface for zero heater power as function of calculated fluid pressure (M Pa), rotational speed (rad s^{-1}) and tip speed (m s^{-1}), using $\Delta P = \frac{1}{2}\Omega^2 R^2 \bar{\rho}$ with $\bar{\rho} = 127\,\mathrm{kg\,m^{-3}}$, $R = 0.223\,\mathrm{m}$. The saturation T.P. curve is also shown to indicate subcooling generated by centrifugal compression.

liquid free surface, and z_0 and z_1, are respectively the distances of the free surface and the heat-transfer surface from the rotational axis. The saturation vapour pressure curve for helium is also shown for comparison to indicate the subcooling due to the centrifugal compression. During an experimental run, z_0 can be determined from the initial volume of liquid and a knowledge of the helium space geometry.

Within experimental error, the measured temperatures are consistent with the helium having undergone an isentropic, centrifugal compression. Moreover, observation of the approach to equilibrium of these temperatures suggests that there is a steady, recirculating flow of helium between reservoir and heat-transfer cell. If the radial column of helium were stagnant, any rise in fluid temperature at the periphery as the speed of rotation was increased would be expected to decay (i.e. disappear) with a time constant characteristic of thermal conduction in the duct wall and fluid. The dominant (shorter) time constant would be that due to wall conduction and would be of the order of 10^2 s. In fact, no such transient behaviour of fluid temperature was observed. Following each increase in speed of rotation, a new equilibrium temperature was reached within a few seconds. The conclusion drawn is that in the case of no power dissipation in the heater the appropriate reference temperature for the bulk fluid adjacent to the heat-transfer surface is that due to a steady, isentropic flow from the liquid reservoir.

Clearly, this assumption would be invalid if, during heat-transfer measurements, there was mixing of the heated and unheated fluid. The extent to which mixing occurs might be expected to become apparent as a progressive deviation of the data, as the heat flux is increased to high values, from any free convection correlation fitted to the low heat flux data. As can be seen from the results described below, no such systematic effect is apparent and the assumption that unheated fluid reaches the closed end of the thermosyphon under all operating conditions appears to be valid. Clearly, further experiments using another thermometer mounted in the fluid adjacent to the heat-transfer cell are required to confirm this assumption. Practical difficulties prevented this technique being used in the current work.

3.2. *Static experiments*

The experiments involving static (non-rotating) pool heat transfer to the saturated liquid were intended to test both the functioning of the heat-transfer cell and to characterise the copper used for the nucleate boiling experiments under rotation. The OFHC copper surface was maintained grease and water free throughout the experimental work but received no special treatment beyond that due to machining. Some measure of the RMS surface roughness is attainable via the work of Cummings and Smith [15] who correlated surface finish with the value of γ in the relation for q given by

$$q = C(\Delta T)^{\gamma} \tag{5}$$

where C and γ are constants.

The results of the static (non-rotating) experiments are summarised in Fig. 4 in the form of heat flux q vs ΔT, the temperature excess of the heat-transfer surface over the saturation value. The slope of the best fit straight line through the data points is 1.9, indicating an approximate RMS surface roughness of 3×10^{-3} mm. Further pool boiling experiments using a heat-transfer surface with a measured roughness of 3.8×10^{-3} mm CLA (Centre Line Average) have closely reproduced the results of the above study and in turn are in reasonable agreement with the conclusions of Cummings and Smith. Some of these latter data points are included in Fig. 4 for comparison.

The saturated liquid temperature was measured using the thermometers located in the heat-transfer cell with no heater power. The cell was sufficiently well shielded thermally to produce negligible error in the liquid temperature as measured by this method.

The Kutateladze correlation [16] has been found by Brentari and Smith [17], amongst others, to describe the nucleate boiling process in saturated helium quite successfully and the results of the present study are also in reasonable agreement with it. The data of Fig. 4 are representative of results collected over a period of 6 months during which time the heat-transfer behaviour of the surface exhibited good reproducibility with no

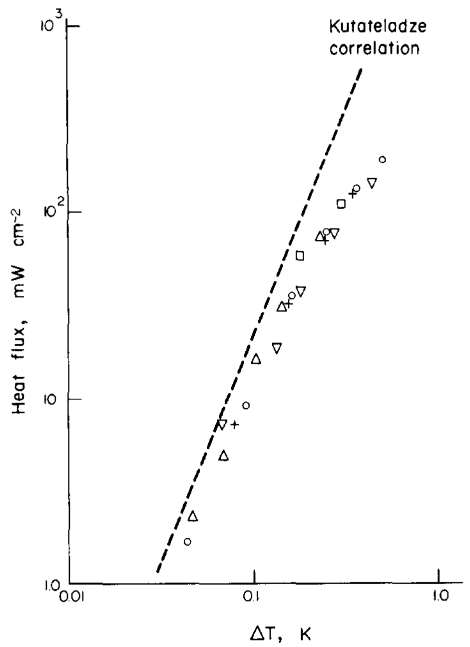

FIG. 4. Static pool boiling in saturated helium at 0.1 MPa + results for $3.8 \cdot 10^{-3}$ mm CLA (measured) copper surface. All other symbols are for different runs on copper surface used in rotating experiments.

sign of any hysteresis effects. This static pool boiling curve is useful as a reference in assessing the combined effect of enhanced acceleration field and increasing sub-cooling upon the nucleate boiling performance of the particular surface used. Clearly it does not automatically follow that other surfaces behaving according to Fig. 4 will be similarly affected by rotation.

3.3. *Heat transfer at low rotational speeds below the critical pressure*

Since the number of controllable variables in the experiments was limited to a single thermosyphon and heat-transfer surface geometry, the following analysis of the data seeks mainly to apply existing forms of dimensionless correlation to the heat-transfer results.

The experimental data for heat transfer can be presented in the basic form

$$q = f(\Delta T) \tag{6}$$

from which a heat-transfer coefficient h can be determined. Data at four different speeds of rotation have been plotted in this form in Fig. 5. At sufficiently low speeds of rotation, the pressure in the helium adjacent to the heat-transfer surface is sub-critical, and sub-cooled nucleate boiling can occur. Referring to Fig. 5, two distinct modes of heat transfer are discernible. A free convection process at low heat flux is seen to give way to a nucleate boiling regime at higher heat fluxes. The point P corresponds to heat-transfer surface temperatures which are 0.1 K above local saturation values for the helium and appears to mark the transition to nucleate boiling with reasonable accuracy for all the speeds of rotation. As the speed of rotation is increased the consequent increase in level of sub-cooling plus the greater contribution of free convection to heat transfer tends to suppress the onset of nucleate boiling, shifting P to higher values of ΔT. In the boiling region above P, the value of the exponent "n" in the curve

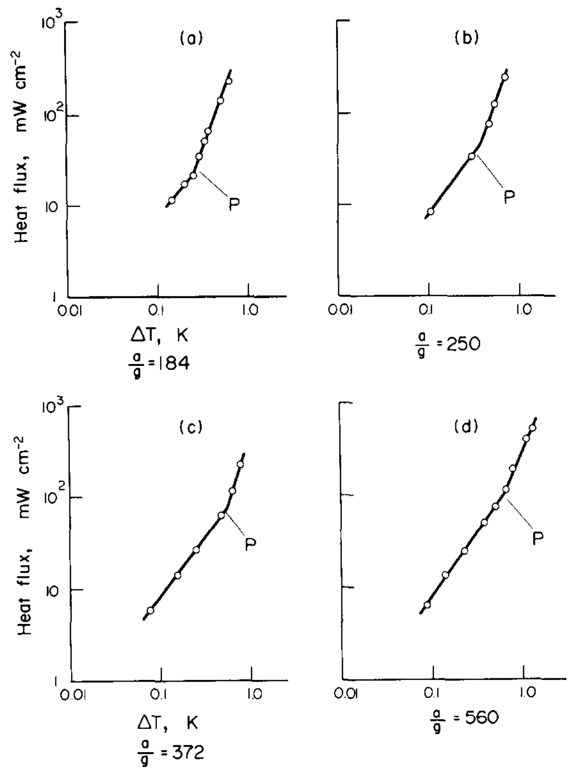

FIG. 5. Heat flux vs ΔT at low rotational speeds with sub-critical pressures, showing transition to nucleate boiling. (a) $a/g = 184$, $\Omega = 90$ rad s^{-1}, $P = 0.124$ MPa, $T_s = 4.243$ K. (b) $a/g = 250$, $\Omega = 108$ rad s^{-1}, $P = 0.133$ MPa, $T_s = 4.254$ K. (c) $a/g = 372$, $\Omega = 128$ rad s^{-1}, $P = 0.149$ MPa, $T_s = 4.274$ K. (d) $a/g = 560$, $\Omega = 157$ rad s^{-1}, $P = 0.173$ MPa, $T_s = 4.306$ K.

$$q = \text{const.}\,\Delta T^n \tag{7}$$

is approximately constant and independent of acceleration field, as found by Fainzilberg and Usenko [18].

When the helium data were expressed as

$$h = \text{const.}\,q^m \tag{8}$$

the value of the index "m" is given by

$$m = \frac{n-1}{n}. \tag{9}$$

The boiling heat-transfer results, over a range of acceleration field 180–560 g, give a value of 0.56 for m with an 11% standard deviation. In contrast Fainzilberg and Usenko find $m = 0.8$ for Freon 11 and 12. The nature of the experiments so far with helium precludes an explanation for this difference.

The effect of an increasing speed of rotation in reducing the contribution of boiling to the overall rate of heat transfer may be correlated in terms of forced convection associated with bubble agitation.

Using a forced convection correlation of the form

$$Nu_b = Nu_b(Re_b, Pr_l, a/g) \tag{10}$$

Körner [19] was able to fit his data on the nucleate boiling of water for varying levels of subcooling 0–40 K and acceleration field 1–1000 g to the expression:

$$Nu_b = 300(Re_b)^{0.667}(Pr_l)^{-0.7}(a/g)^{-0.5} \quad (11)$$

where

$$Nu_b = \frac{q_T - q_{NC}}{(T_w - T_B)k_l}\sqrt{\left[\frac{2\sigma}{g(\rho_l - \rho_v)}\left(\frac{g}{a}\right)\right]}$$

$$Re_b = \frac{q_T - q_{NC}}{\rho_l\, h_{fg}\, v_l}\left(\frac{T_w - T_{sat}}{T_w - T_B}\right)\sqrt{\left[\frac{2\sigma}{g(\rho_l - \rho_v)}\left(\frac{g}{a}\right)\right]}.$$

The correlation obtained by Judd and Merte [20] from their data on Freon 113 (for $1 < a/g < 100$), was similar but with a weaker dependence of Nu_b upon (a/g) as follows:

$$Nu_b = 150(Re_b)^{0.667}(Pr_l)^{-0.7}(a/g)^{-0.17}. \quad (12)$$

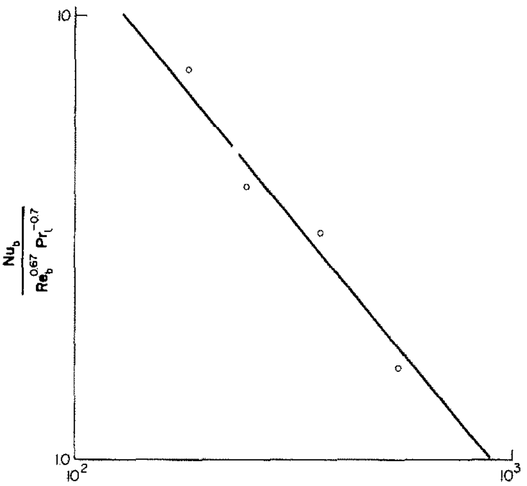

FIG. 6. Application of Körner-type correlation to the nucleate boiling results.

The same functional dependence of Nu_b upon Re_b and Pr_l has been assumed in this work and the result is shown in Fig. 6. The boiling data have been found to fit the following expression for bubble Nusselt number:

$$Nu_b = 2390(Re_b)^{0.67}(Pr_l)^{-0.7}(a/g)^{-1.1}. \quad (13)$$

The introduction of the dimensionless subcooling parameter

$$\frac{T_w - T_{sat}}{T_w - T_B}$$

into the expression for bubble Reynolds number appears to be somewhat arbitrary and may be the reason for the disagreement between the above correlations and the present results. The effect of an increasing (a/g) is seen to be a rapid reduction in the contribution to overall Nusselt number of the nucleate boiling process. This could be an outcome of the rapid reduction, as a/g increases, in the departure size of vapour bubbles, proportional to

$$\sqrt{\left[\frac{2\sigma}{g(\rho_l - \rho_v)}\frac{g}{a}\right]}.$$

The independent influence of sub-cooling and acceleration field was not easily measured in the apparatus used and so the reduction in Nu_b with increasing (a/g) expressed above has only empirical significance.

3.4. *Heat transfer at high rotational speeds above the critical pressure*

At higher rotational speeds, when the local pressure at the closed end of the duct exceeds the critical pressure of 0.224 MPa for helium, nucleate boiling effects disappear, as might be expected, and heat transfer is by free convection in a single phase fluid. This behaviour is clearly shown in Fig. 7 at two speeds,

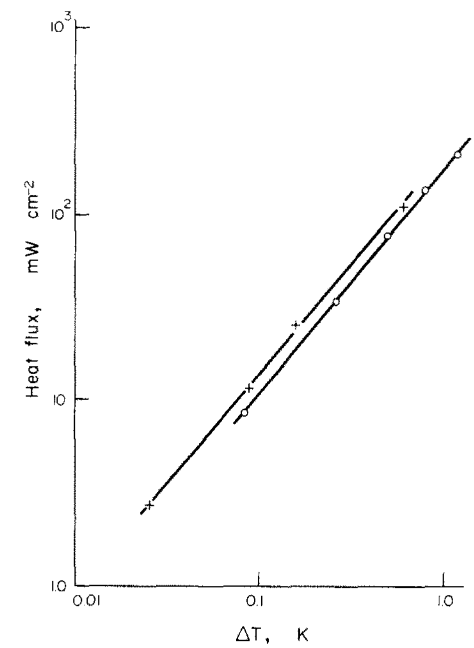

FIG. 7. Heat flux vs ΔT at high rotational speeds with supercritical pressures, (+) $a/g = 1120$, $\Omega = 220\,\text{rad s}^{-1}$, $P = 0.250$ MPa, $T_s = 4.409$ K; (○) $a/g = 1720$, $\Omega = 275$ rad s^{-1}, $P = 0.334$ MPa, $T_s = 4.516$ K.

where the log–log plot of q against ΔT remains linear up to the highest heat fluxes. This linear behaviour also supports the assumption that unheated fluid reaches the closed end of the duct. Although the correlation described by equation (2) has been found by previous workers to fit the free convection data for thermosyphons with distributed wall heat flux, it seems a reasonable assumption that a similar type of correlation should also fit when the heating is at the closed end only.

With this assumption in mind, all of the heat-transfer data in Figs. 5–7 have been recalculated in terms of a free convection correlation given by

$$Nu = \text{constant}\ (Ra)^m \quad (14)$$

where m is a constant and the Rayleigh number is based on duct radius in order to compare with the previous work. The results are displayed in Fig. 8.

Fluid properties calculated at the temperature of the unheated fluid give the best correlation of the data. Nusselt and Rayleigh numbers are based upon the duct radius of 12.5 mm. Equation (14) fits the data quite well except for the disconnected curves A, B, C and D which represent the added influence of nucleate boiling in improving the overall Nusselt number. The best fit for all the free convection data ($180 < a/g < 1720$) is

$$Nu = 0.29(Ra)^{0.29} \quad (15)$$

with a calculated RMS standard deviation in Nu of 14%.

This is a remarkably simple result bearing in mind the number of fluid property changes and the wide range of acceleration fields involved. The magnitude of the exponent of Ra suggests the existence of a thermal boundary-layer regime of flow [10], wherein a stream of heated fluid moves essentially independently of the unheated fluid. The results of Fig. 8, based on the assumption of no mixing, suggest that a well-defined flow loop within the single duct is the mechanism of heat transport to the liquid reservoir. More precise interpretation of these preliminary results to infer the likely flow pattern is not possible. It does appear, however, that unheated fluid penetrates to the closed end of the duct, over the whole range of heat flux employed in the experiments. Finally, it should be noted that the effect of mixing in progressively reducing the value of "m" in equation (14) has been discussed by Jakob [9] and is also demonstrated by the results of Ellerbrock [7].

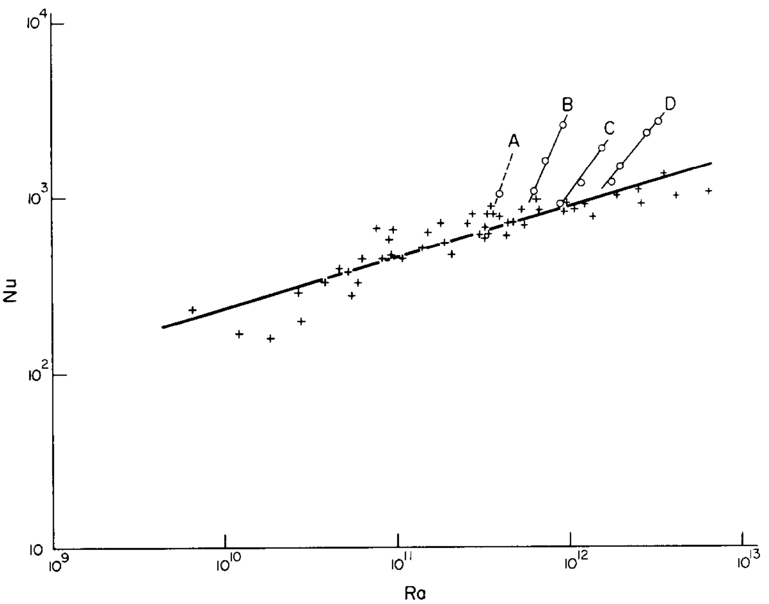

FIG. 8. Free convection correlation of all data. Nucleate boiling data shown by curves A, B, C, D; respectively a/g = 184, 250, 372, 560.

4. THE INFLUENCE OF FREE CONVECTION IN A FORCED FLOW SYSTEM

Free convection heat transfer is expected to be the dominant mechanism for any helium cooling system in rotors of 1 m dia and rotating with a frequency of 50 Hz. This can be demonstrated by an order of magnitude comparison of the expected forced convection Nusselt numbers, for the likely mass flow rates of coolant needed, with the free convection Nusselt numbers measured in this study. Typical mass flow rates in 2-cm dia ducts for a 1300 MW machine are in a range 0.5–2 g.s^{-1} producing Reynolds numbers of between 3000 and 12000. Although for flow in a duct, rotating about an axis parallel to but displaced from its own longitudinal axis, a rotationally-induced enhancement of heat transfer has been predicted [21, 22] and measured [23, 24], an estimated Nusselt number may be based on results obtained for supercritical helium in a stationary duct. From their results, both Giarratano *et al.* [25] and Brassington and Cairns [26] produced correlations using modifications of the Dittus–Boelter equation for turbulent forced convection heat transfer, a typical correlation for supercritical helium being of the form

$$Nu = 0.0259(Re)^{0.8}(Pr)^{0.4}(T_w/T_E)^{-0.716}. \quad (16)$$

Using the range of Reynolds number cited for helium flow in the rotor of a superconducting alternator and using the above correlation, the forced convection Nusselt number should lie in a range 10–100. Extrapolation of the free convection correlation obtained in this study to typical values of Rayleigh number in a 1.0 m dia rotating helium system yields free convection Nusselt numbers in the range 10^3–10^5. It follows therefore that local free convection heat transfer may well be the dominant mechanism in rotating systems with comparatively low mass flow rates of helium.

The need to assume that helium flows isentropically to the closed end of the thermosyphon during the heat-transfer process is avoided by expressing the data in the form of an effective thermal conductance for the column of circulating helium. Since the free surface of the reservoir acts as an evaporative heat sink at a known temperature of approximately 4.22 K, an effective thermal conductance can be determined from the measured heat flux and temperature at the outer end of the helium column. The effective thermal conduc-

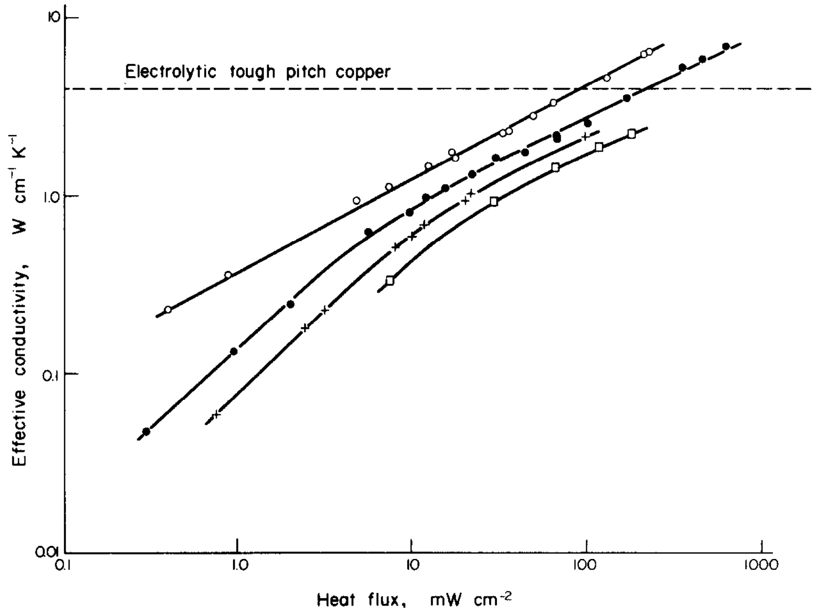

FIG. 9. Effective thermal conductivity of the helium column. ○ $-a/g = 184$, $p = 0.124$ MPa; ● $-a/g = 560$, $p = 0.173$ MPa; + $-a/g = 1120$, $p = 0.250$ MPa; □ $-a/g = 1720$, $p = 0.334$ MPa. Values of acceleration field and fluid pressure are those at the heat-transfer surface.

tivity is expressed in Fig. 9 as a function of heat flux for various rotational frequencies. The use of the helium-filled open thermosyphon as the primary heat-transfer device in a rotor appears therefore to be quite attractive, especially at higher heat fluxes. At a given heat flux the effective thermal conductivity decreases as the rotational frequency is increased. This effect is simply due to the larger isentropic temperature rise experienced by the helium as it is compressed to the

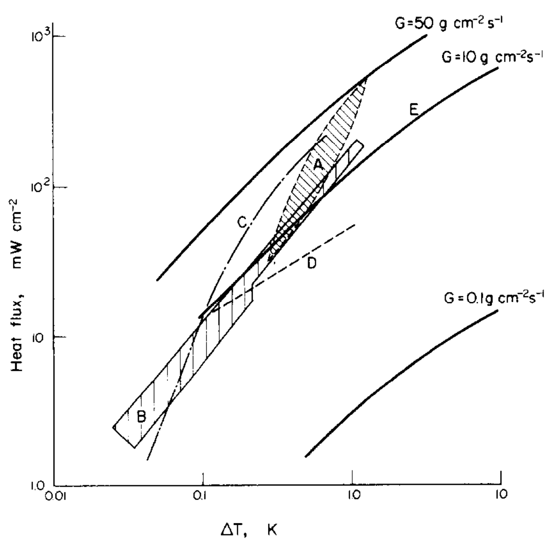

FIG. 10. Heat flux vs ΔT—summary of the data and comparison with other modes of heat transfer to liquid and supercritical helium. (A) Nucleate boiling under rotation; (B) Free convection under rotation; (C) Saturation pool boiling at $a/g = 1$; (D) Free convection to supercritical helium at 5.4 K, 0.24 MPa and $a/g = 1$; (E) forced convection to supercritical helium at mass flows of 0.1, 10 and 50 g cm^{-2} s^{-1} respectively.

higher pressures produced by increased speeds of rotation.

A further consideration in the design of refrigeration for the field windings of a superconducting alternator is that the cooling system should be able to cope with increased transient heat loads arising from electrical fault conditions. In this respect the rapid improvement in effective thermal conductivity as the heat flux increases is a positive advantage of the open thermosyphon configuration.

Finally, all the data are displayed in block form on a heat flux vs ΔT plot in Fig. 10 together with a summary of previous heat-transfer data on liquid and supercritical helium.

5. SUMMARY AND CONCLUSIONS

The experimental limitations of this study have made accurate conclusions difficult regarding the boiling and free convection heat-transfer processes in helium under high speed rotation.

Instead, a comparison of the results with existing data correlations allows the following points to be made.

1. For the range of centrifugal acceleration field imposed, a transition from free convection to nucleate boiling heat transfer is well defined and corresponds to a heat-transfer surface temperature 0.1 K above the local saturation value.
2. The free convection data follows a relationship

$$Nu = 0.29 Ra^{0.29}$$

even for the highest heat flux used, supporting the assumption that the reservoir helium flows isentropically to the closed end of the thermosyphon.
3. An increase in rotational speed, which affects the

levels of subcooling and acceleration field simultaneously, produces a marked reduction in the influence of bubble agitation compared to free convection for the overall heat transfer process.

4. Owing to the interdependence of sub-cooling and acceleration field, their relative importance in suppressing the effect of bubble agitation is not established.

5. As a result of the extremely low kinematic viscosity of liquid helium (of the order $3 \times 10^{-4}\,\mathrm{cm^2\,s^{-1}}$) and the large centrifugal acceleration (up to $5000\,g$), very high Rayleigh numbers apply in the full-scale rotor transfer process. Consequently, extrapolating the results which have been obtained for helium to higher Rayleigh numbers, it is concluded that free convection heat transfer will be the dominant mode in a full-scale rotor cooling system.

6. Finally, some advantages of using the rotating open thermosyphon to remove heat from the periphery of the rotor have been pointed out.

Acknowledgements—GKT acknowledges the support of Science Research Council for a research studentship and subsequently a personal research fellowship.

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TRANSFERT THERMIQUE EN RESERVOIR A L'HELIUM LIQUIDE ET SUPERCRITIQUE DANS UN CHAMP INTENSE D'ACCELERATION CENTRIFUGE

Résumé—En relation avec le problème du refroidissement des enroulements à champ tournant d'un alternateur à supraconducteurs, on présente une étude expérimentale préliminaire du transfert de chaleur à l'hélium au repos, entrainé par une vitesse de rotation importante. Le transfert de chaleur sur une surface plane de cuivre a été mesuré dans un domaine de flux thermiques allant de 0,2 à $650\,\mathrm{mW\,cm^{-2}}$ et dans des champs d'accélération centrifuge locale de 180 à 1720 g perpendiculaires à la surface. L'analyse des données montre que le régime de transfert thermique par convection naturelle aux pression sub-critiques et supercritiques suit la loi simple $Nu = 0{,}29Ra^{0,29}$. Aux pressions subcritiques, et pour des flux thermiques élevés, s'ajoute le processus d'ébullition nucléée.

DER WÄRMEÜBERGANG AN FLÜSSIGES UND ÜBERKRITISCHES HELIUM IN BEHÄLTERN UNTER DEM EINFLUSS GROSSER ZENTRIFUGALER BESCHLEUNIGUNGSFELDER

Zusammenfassung—Im Hinblick auf das Problem der Kühlung rotierender Feldwicklungen supraleitender Wechselstromgeneratoren wird über eine vorläufige experimentelle Untersuchung über den Wärmeübergang an Helium in Behältern unter dem Einfluß hoher Rotationsgeschwindigkeiten berichtet. An einer ebenen Kupferoberfläche wurde der Wärmeübergang bei Wärmestromdichten von 0,2 bis 650 mW cm^{-2} und bei lokalen, senkrecht zur Oberfläche gerichteten Beschleunigungsfeldern von 180 bis 1720 **g** gemessen. Die Auswertung der Daten zeigt, daß der Wärmeübergang bei freier Konvektion sowohl bei unter- wie bei überkritischen Drücken der einfachen Beziehung $Nu = 0{,}29 \cdot Ra^{0{,}29}$ gehorcht. Bei genügend hohen Wärmestromdichten tritt bei unterkritischen Drücken das Blasensieden als zusätzlicher Vorgang auf.

ТЕПЛОПЕРЕНОС В БОЛЬШОМ ОБЪЕМЕ К ЖИДКОМУ И СВЕРХКРИТИЧЕСКОМУ ГЕЛИЮ ПРИ УСКОРЕНИИ В ПОЛЕ СИЛЬНЫХ ЦЕНТРОБЕЖНЫХ СИЛ

Аннотация — В связи с проблемой охлаждения вращающейся обмотки возбуждения сверхпроводящего генератора приведены результаты предварительного экспериментального исследования теплопереноса в большом объеме к гелию при высокой скорости вращения. Проведены измерения теплопереноса от плоской медной поверхности для теплового потока в диапазоне 0,2–650 мватт см$^{-2}$ при наличии ускорения в поле центробежных сил в 180–1720 g, направленных перпендикулярно поверхности. Анализ данных показывает, что теплоперенос с помощью конвекции как при критическом, так и при сверхкритическом давлении подчиняется простому соотношению $Nu = 0{,}29\ Ra^{0{,}29}$. При докритическом давлении и достаточно высоких тепловых потоках, кроме того, имеет место пузырьковое кипение.