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Heat Transfer and Flow of Helium in Channels--Practical Limits for Applications in Superconductivity

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ABSTRACT

Heat transfer and fluid mechanics of helium flowing in channels are reviewed. Emphasis is placed on observed or anticipated limits of operation which might be expected to apply in applications of superconductivity. Topics included are: the high-heat-flux degradation of heat transfer and possible effects of buoyancy forces in supercritical helium; transition to film boiling in subcritical helium; limiting heat currents in helium II; and the possibility of oscillations in forced flow helium systems.

Key Words: Buoyancy; channels; film boiling; forced convection; helium I; helium II; mixed convection; nucleate boiling; subcritical; supercritical; transition; turbulent flow.

Introduction

The development of large superconducting devices is intimately related to the fluid mechanics and heat transfer characteristics of cryogenic helium. In the earliest successfully developed magnets for bubble chambers and accelerator beam transport and focussing, the main function of the helium was to cool the conductor matrix down, to stabilize it against flux jumps, and to provide a heat sink for the relatively low losses which occur in charging. The success of this phase of development of superconducting technology is attested to by the existence of several such devices with 1000 hours or more of routine operation behind them [1,2]. The wide range of applications under consideration for the future, however, demands much more of the helium as a heat transfer medium, and will exercise the ingenuity of designers to the full. The simple expedient of immersing a device in a bath of liquid helium at a temperature close to 4 K will not suffice or may simply be impractical.

Our research philosophy at the National Bureau of Standards has been to explore as far as possible all modes of application of helium as a heat transfer medium in order to preserve as many options as possible for the designer. As a consequence we are interested in all phases of helium and we are particularly interested in exploring the possible boundaries of operation imposed by the thermodynamic and transport properties of helium.

In this paper, after a brief discussion of the relevant properties of helium, we consider some important characteristics of helium flowing in channels, since we anticipate that this mode of cooling in some form or other will be preferred over natural convection in future large devices. First we discuss heat transport to helium I above the critical pressure, then heat transport to helium I below the critical pressure. We then discuss some possibilities for cooling to lower temperatures by means of helium II. In the final section we consider the problem of flow stability and oscillations in channels cooled by forced flow of helium.

Heat transfer under steady state conditions is the traditional approach to technology of cooling superconducting devices and is the subject of most of the following material. But we should like to take this opportunity to urge that future research concentrate more on transient conditions; it appears that performance limits may often only occur during such unsteady circumstances, as for example in electrical power network faults or in necessary pulsed operation. No work to date has been done under flow conditions, but measurements in static liquid helium have already shown that under pulsed heating orders of magnitude increase in heat flux may be sustained for a given wall temperature rise [3,4,5].

Thermodynamic and Transport Properties of Helium

With reference to the phase diagram, figure 1, the first point to make is that, with critical temperature and pressure 5.20 K and 0.2275 MPa (2.245 atm) respectively, we can hardly avoid the near-critical region in cooling superconductors. At pressures greater than critical, the fluid undergoes a smooth, albeit steep, transition from liquid-like density to gas-like density as the temperature increases and crosses the so-called transposed critical, or pseudocritical, line. This is the dashed line in the figure and is the locus of maxima in the specific heat at constant pressure. At sub-critical pressures, we have to contend with the vapor-liquid phase boundary and the associated boiling and two-phase phenomena.

This proximity to the critical point, either at super-or sub-critical pressures is no disadvantage from a heat transfer point of view, but the processes involved in the transport of heat are complex and are perhaps best described by reference to experimental results. As we shall see, there are boundaries to the regimes of good heat transfer and these must be established experimentally.

In the supercritical pressure region we have to admit to some uncertainties in transport properties which cloud the interpretation of experimental results somewhat. First of all, for the viscosity [6] of dense helium between 4 and 20 K the experimental uncertainty is rather larger than for most fluids (possibly as much as 8%). Secondly, until quite recently, no experimental results were known to us on the thermal conductivity in this range. McCarty [6] has published interim estimates made by Roder based on published data above 20 K and below 4 K [7,8],

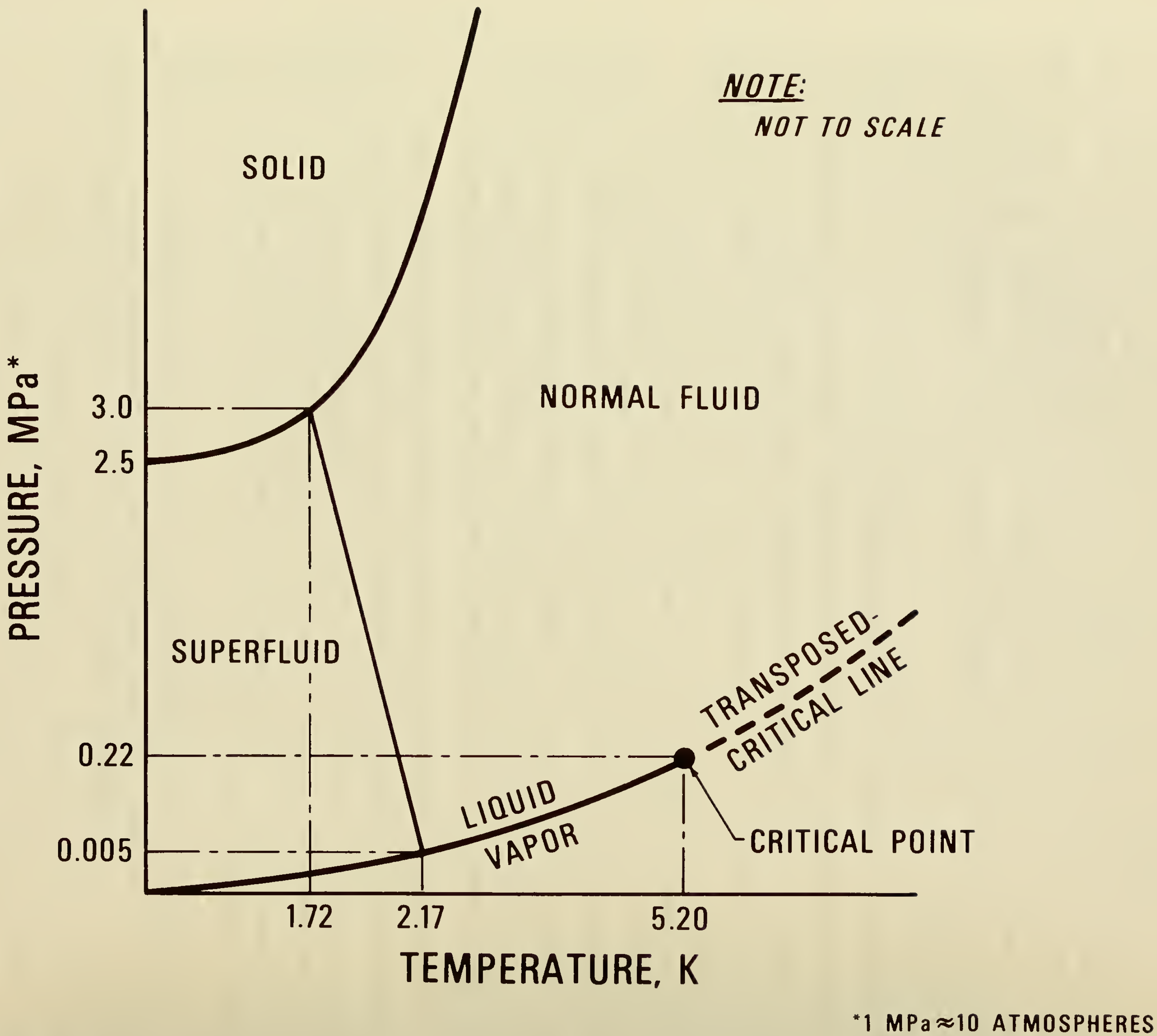

Fig. 1. Phase diagram of Helium-4.

and on the thermal conductivity of hydrogen. Recently, we have become aware of new measurements, yet to be published, by Kellner [9]. We have the provisional reassuring news that the new measurements are reasonably close to Roder's estimates. Nevertheless, when interpreting heat transfer results it is important to quote the source of data used for properties and to bear in mind these uncertainties. In all of our work on helium I discussed below, we have used the data on properties published by McCarty.

Below the lambda temperature, which decreases smoothly from 2.177 K at 0.0053 MPa (0.0497 atm) to 1.763 K at 3.013 MPa (29.74 atm), as is well known, we encounter the helium II liquid phase (superfluid). The implications of the unique superfluid properties are discussed further below; but briefly, it is no longer possible to assign a unique value of the transport properties to any particular thermodynamic state [10]. Here again, these properties are at best very favorable from a heat transfer viewpoint, but there are boundaries to the existence of such favorable properties and these must be established.

Heat Transport to Helium I Above the Critical Pressure

Heat transport to fluids near the critical point, and in particular at pressures above critical, has been a subject of investigation for many years. Indeed, a review paper was published in 1971 by Hall [11] containing 57 references to previous work. However, the majority of experimental papers referred to studies with water and carbon dioxide. None of the papers quoted dealt with helium although a few dealt with hydrogen and oxygen. In fact the first useful data on helium were not available in the open literature until 1971.

The advantages of supercritical helium as a heat transfer medium are that, with high enough flow rates, heat transfer coefficients comparable to subcritical nucleate boiling can be realized ($\sim$1.0 W/cm^2·K) without the attendant possibility of "burn out" which, as will be seen below, can lead to a large temperature rise sufficient to exceed the critical temperatures of all known superconductors. Furthermore, the fluid has almost as high a heat capacity as subcritical, i.e., liquid helium, to cope with transient heat release in the superconductor. However, one thing that is clear from studies with other fluids is that a considerable range of wall temperatures can sometimes be obtained under seemingly identical conditions. Most disturbing of all, the phenomenon of temperature "spikes" along a vertical heated channel is well known. In carbon dioxide and water these spikes may be of the order of 100 K. This somewhat unpredictable nature of heat transfer at supercritical pressures should make us cautious and, one would hope, would lead to some very careful experimental work with helium. It is probably fair to say, however, that great strides have been made towards understanding temperature spikes and so-called deterioration in heat transfer during the last few years, but unanimity has not yet been reached and some unexplained phenomena remain [12].

It is convenient to divide the rest of this discussion into two parts. In part a. we discuss heat transfer which is dominated by pressure drop forces, i.e., pure forced convection; this appears to include all of the available experimental data. In part b., we discuss the possible effects of buoyancy and acceleration, i.e., mixed convection.

a. Pure forced convection.

If the Reynolds number of the flow is high enough, flow should be dominated by pressure drop forces and results should be independent of channel orientation. Under these conditions a variety of behavior can be expected. First, with relatively low heat flux, the wall and bulk fluid densities are almost equal and the heat transfer coefficient is predictable by the usual empirical relationships among Nusselt, Reynolds and Prandtl numbers for turbulent flow, e.g.,

$$Nu = C_1 \, Re^{C_2} \, Pr^{C_3} \qquad (1)$$

where C_1, C_2 and C_3 are constants. The heat transfer coefficient in this case is a constant independent of the heat flux for a given thermodynamic state and we have a linear heat transfer process. The experimental results of Johannes [13] appear to be of this kind.

In order for this to apply, the thermodynamic state of the helium must be away from the region of sharp property variation, i.e., temperature well above or below the transposed critical, or the heat flux must be small.

Whereas the results of Johannes all pertain to the situation where both bulk fluid and wall temperatures remained below the transposed critical (i.e., $T_B < T_W < T_{TC}$) those of Giarratano et al. [14], covered a wide range of thermodynamic states and all situations were encountered, i.e., $T_B < T_W < T_{TC}$, $T_B < T_{TC} < T_W$, and $T_{TC} < T_B < T_W$. In this case the equation

$$Nu = 0.0259\ Re_B^{0.8}\ Pr_B^{0.4} \left(\frac{T_W}{T_B}\right)^{-0.716} \qquad (2)$$

fitted the data with a standard deviation of 8.5%. For a given thermodynamic state we now have a slightly non-linear heat transfer process, but we note that for low heat flux and temperature rise equation (2) reduces to equation (1). We note also that in the experiments of Giarratano et al. the density ratio ρ_B/ρ_W did not exceed about 2.5. Equation (1) might be considered the limiting form of the heat transfer coefficient for near homogeneous conditions.

Equation (2), in this limit, predicts a heat transfer coefficient proportional to the quantity $k^{0.6}\ (C_p/\mu)^{0.4}$ for given channel diameter and flowrate. This quantity is dominated by the behavior of C_p, and reaches a maximum at the transposed critical temperature when plotted against temperature at constant pressure. Thus, if we take a channel with constant flow rate and inlet conditions below T_{TC}, the heat transfer coefficient at any point along the channel should rise with heat input as the fluid temperature approaches T_{TC}. This can be clearly seen in the data of Johannes. For high heat flux, however, the wall temperature may be expected at some point to exceed the transposed critical and we could have a wide variation of properties across the channel. In the experiments reported by Ogata and Sato [15] density ratios of about 5 were achieved at 3 atm pressure and in this case heat transfer coefficients a good deal less than those given by equation (2) were obtained at the highest heat fluxes.

In order to investigate this situation in more detail we performed experiments on an improved apparatus at 2.5 atm [16]. This is only 10% above the critical pressure and we were able to obtain density ratios ρ_B/ρ_W up to 10, when in all cases we had $T_B < T_{TC}$ and both $T_W < T_{TC}$ and $T_W > T_{TC}$. In figure 2 we show a typical example of heat transfer coefficient profiles for a given flow rate and inlet temperature. The data are for a 2.13 mm inside diameter tube.

The most significant feature is the variation in heat transfer coefficient with change in heat flux. In all cases studied the pattern was the same: the whole profile rises at first with increase in heat flux and then falls below its original low heat flux level, dropping proportionately more toward the outlet. If we take a fixed position in the tube and for a given flow rate and inlet temperature we plot the local heat transfer coefficient against heat flux the result is typified by the two examples given in figure 3. We have also plotted as dashed lines the limiting values given by equation (1) using the constants of equation (2) for comparison. Thus, a maximum in heat transfer coefficient is reached long before any maximum that would be predicted by equation (1). Since T_B was always less than T_{TC} of course the result calculated from equation (1) never in fact reaches a maximum for these data.

In figure 4 all of the results for the thermometers at respectively 5, 7 and 10 cm from inlet are plotted as the ratio of experimental heat transfer coefficient to the limiting value by equation (1) versus the ratio ρ_B/ρ_W. This plot clearly reveals the range of behavior we have been discussing, from the homogeneous or limiting low heat flux behavior of equation (1) to departures as much as 90% as the stream becomes less and less homogeneous.

The central question, of course, is whether the departure from homogeneous condition is predictable. The problem is that in the presence of such inhomogeneities any application of the well-known semiempirical theories of turbulent transport involve hypotheses; the prescriptions for

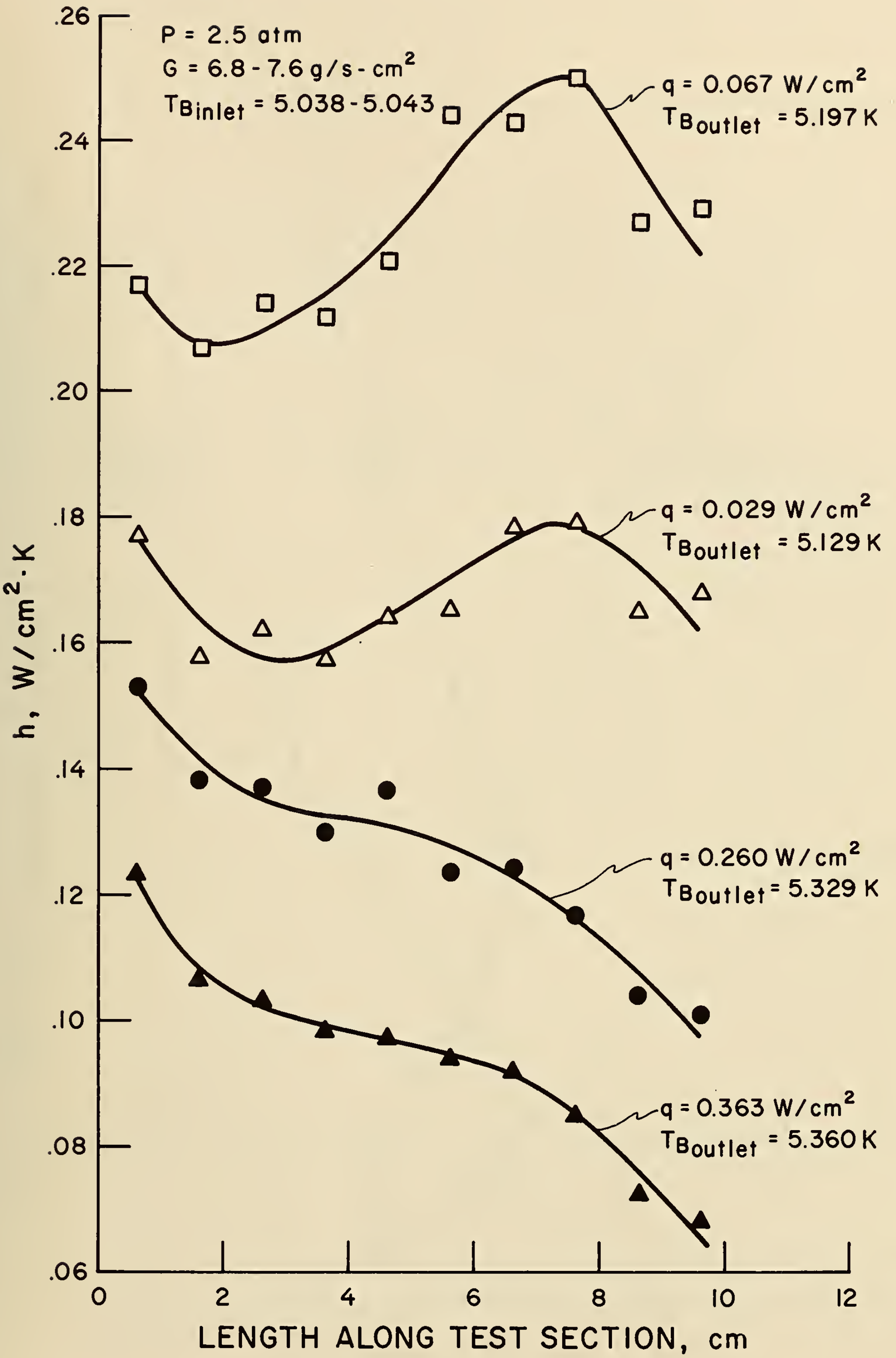

Fig. 2. Heat transfer coefficients for supercritical helium [16] in a vertical tube 2.13 mm diameter, downward flow.

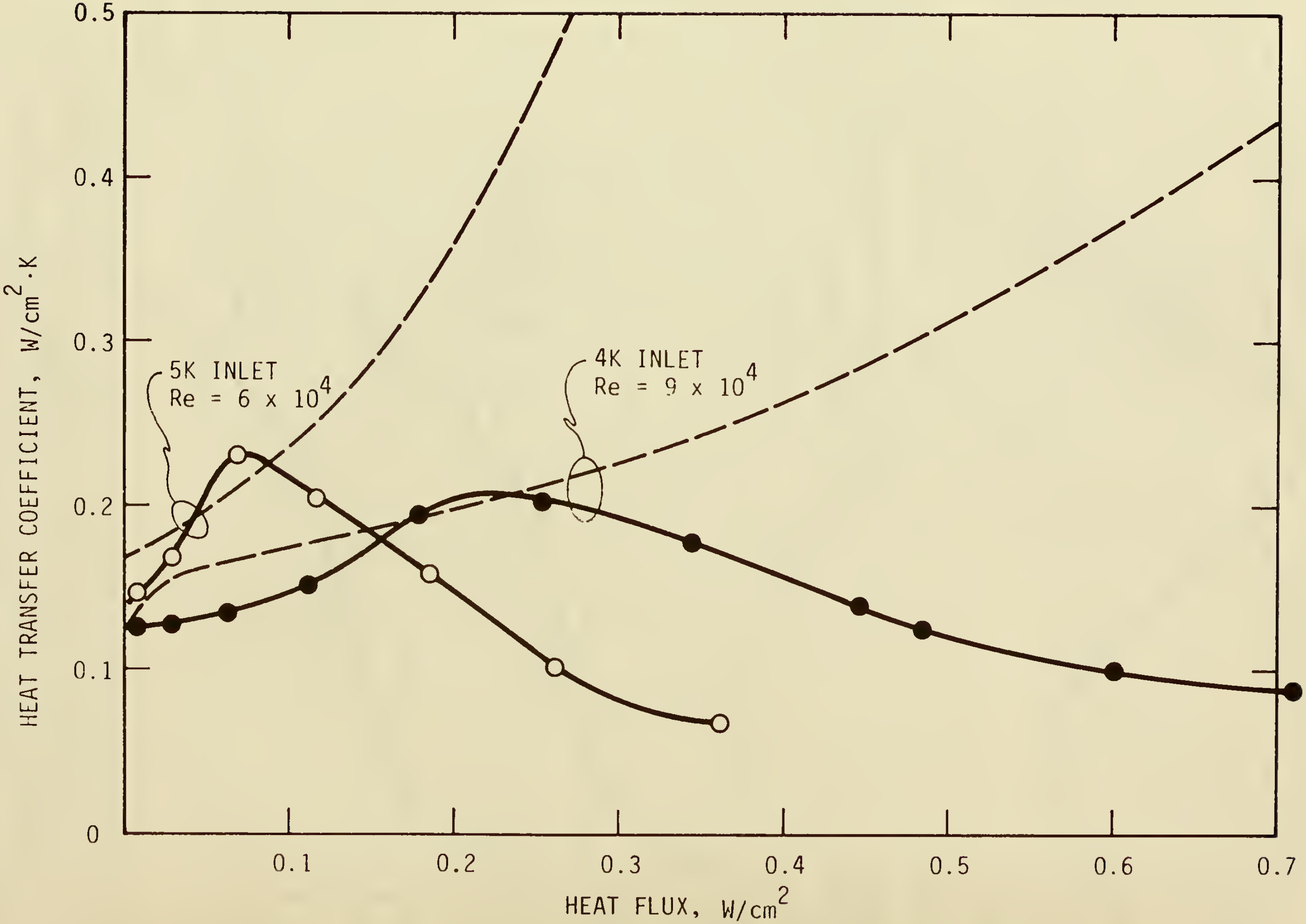

Fig. 3. Heat transfer coefficient vs. heat flux at z/d = 45 for fixed inlet conditions.

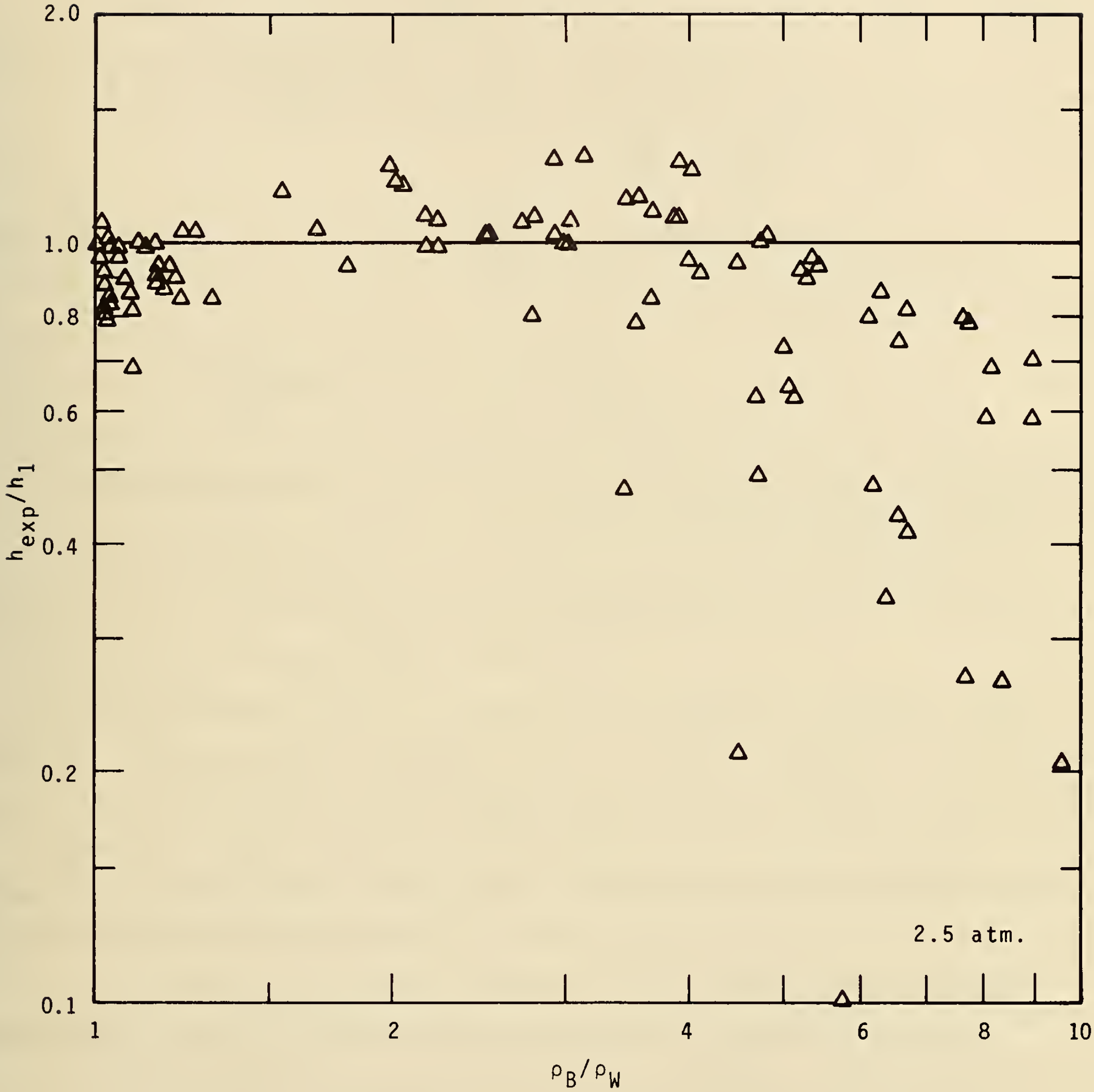

Fig. 4. Ratio of experimental heat transfer coefficient to theoretical by equation (1).

calculating turbulent diffusivities all are taken from isothermal turbulence studies. However, as we shall show below some understanding of the gross effects may be gained by a proper appreciation of the role of fluid properties.

To achieve this understanding, let us go back to the simplest of theories: namely, the old mixing length theory of Prandtl applied to tube flow. Neglecting the convective terms, the shear stress and heat flux may be related to local velocity and temperature gradients in fully developed steady turbulent flow by

$$\tau = \rho(\varepsilon_M + \nu) \frac{\partial u}{\partial y} \tag{3}$$

and

$$q = \rho C_p(\varepsilon_H + \alpha) \frac{\partial T}{\partial y} = \rho(\varepsilon_H + \alpha) \frac{\partial H}{\partial y} . \tag{4}$$

Equation (3) defines an eddy diffusivity for momentum transport which, by the mixing length theory, is given by

$$\varepsilon_M = \ell^2 \left|\frac{\partial u}{\partial y}\right| . \tag{5}$$

Under classical isothermal conditions, ℓ, in the near proximity of the wall, is a universal function of the distance y from the wall. Various semiempirical expressions have been proposed for ℓ [17], but we shall simply write $\ell = \ell(y)$. Also, for fully developed turbulence, when only pressure drop and friction forces are present, τ and q are linear functions of radius decreasing to zero at the channel center. In the region close to the wall, we may approximate τ and q by their values τ_W and q_W at the wall. In any case, the variation of τ and q across the channel will not be dependent on fluid properties. Finally, we shall assume, as is customary and has been born out by experiment, that ε_H is approximately equal to ε_M.

Under the above assumptions and approximations equations (3) and (4) may be integrated using (5) to give

$$u^+ = \frac{u}{u^*} = \int_0^{y^+} \frac{2\, dy^+}{\left(\frac{\mu}{\mu_W} + \sqrt{\left(\frac{\mu}{\mu_W}\right)^2 + 4\ell^{+2}\left(\frac{\rho}{\rho_W}\right)^2}\right)} \tag{6}$$

$$(H_W-H)^+ = \frac{\rho_W(H_W-H)u^*}{q_W} = \int_0^{y^+} \frac{2\, dy^+}{\left(\frac{2}{Pr} - 1\right)\frac{\mu}{\mu_W} + \sqrt{\left(\frac{\mu}{\mu_W}\right)^2 + 4\ell^{+2}\left(\frac{\rho}{\rho_W}\right)^2}}\ , \tag{7}$$

where we have used the friction velocity $u^* = (\tau_W/\rho_W)^{1/2}$ to define the dimensionless velocity u^+ and the dimensionless distance from the wall, $y^+ = yu^*/\nu_W$. Note that for $Pr \approx 1$ we have similarity of enthalpy and velocity profiles rather than temperature and velocity.

In equations (6) and (7) the standard dimensionless profiles are obtained under homogeneous conditions i.e., $\rho = \rho_W$, $\nu = \nu_W$ etc. In general, however, these profiles could not be evaluated except by numerical means making use of the equation of state at each step and using an empirical expression for ℓ^+. We would also need to determine the friction velocity $u^* = (\tau_W/\rho_W)^{1/2}$ by integral mass conservation:

$$\dot{m} = \int_0^R 2\pi r \rho u dr\ . \tag{8}$$

On the other hand, we can readily find an approximate solution enabling us to pick out the major property effect.

This can be done by noting first of all with reference to figure 5 that by a suitable choice of reference enthalpy the specific volume $1/\rho$ is proportional to the enthalpy when $T_W > T_{TC}$, at least in some region close to the wall. Then $\rho/\rho_W = v_W/v = H_W/H = 1 + (H_W-H)/(H-H_0)$. We notice also that, in this region, the viscosity is only a weak function of enthalpy. Thus, since the integrands appearing in (6) and (7) are strongly weighted towards small y^+, we may set $\mu/\mu_W = 1$ and $\rho/\rho_W = 1$. The integrals in (6) and (7) are now the same as the constant property integrals to first order.

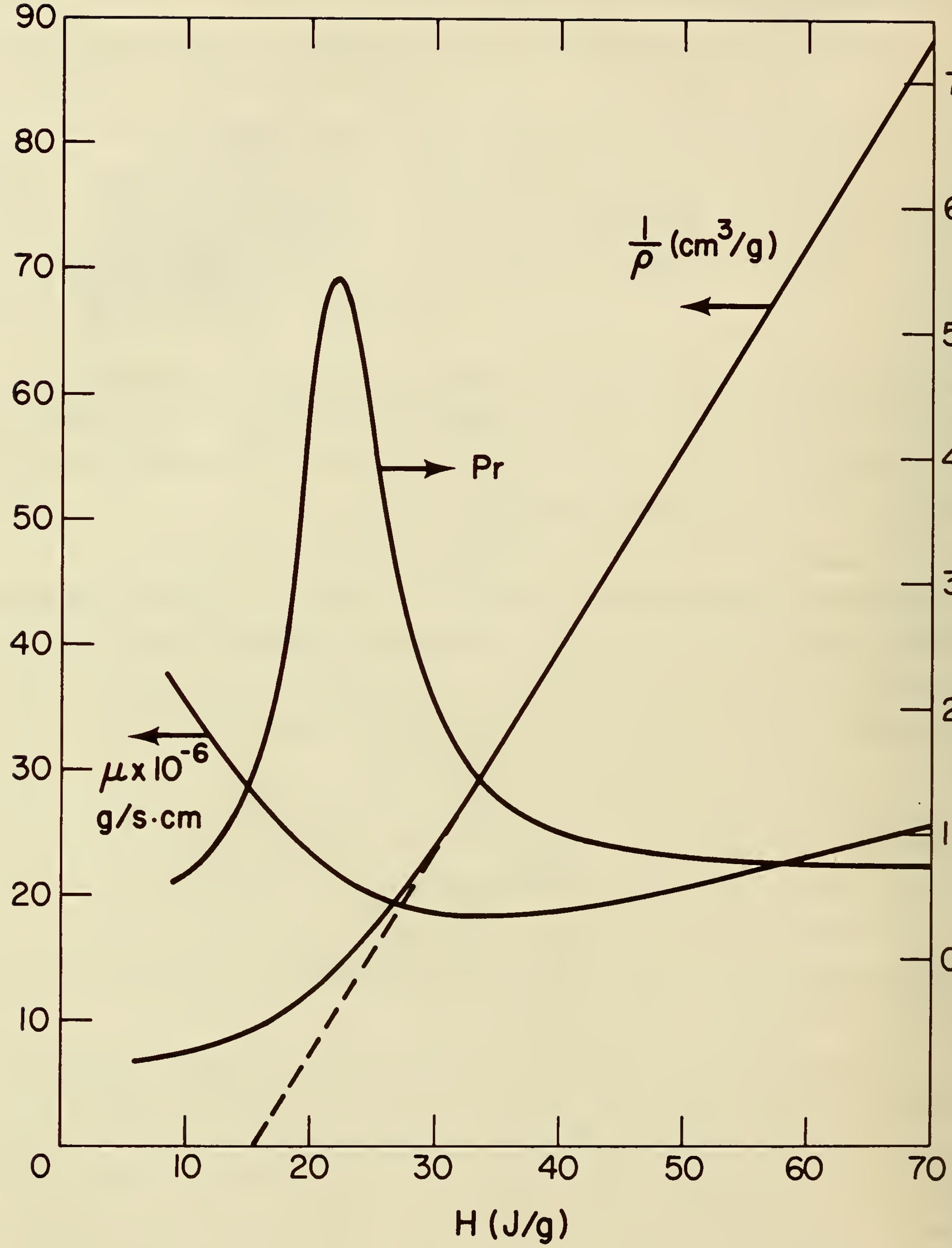

Fig. 5. Prandtl number, viscosity and specific volume vs. enthalpy at 2.5 atm pressure.

The constant property equivalents of (6) and (7) for $T_B \approx T_W < T_{TC}$ are

$$u_1^+ = \frac{u_1}{u_1^*} = \int_o^{y^+} \frac{2\,dy^+}{1 + \sqrt{1 + 4\ell^{+2}}} \tag{9}$$

$$(H_W-H)_1^+ = \rho_B Cp_B \frac{(T_W-T)_1}{q_{W_1}} u_1^* = \int_o^{y^+} \frac{2\,dy^+}{(\frac{2}{Pr_B} -1)+\sqrt{1+4\ell^{+2}}} \tag{10}$$

Using (8) it is readily shown that $u_1^* \approx u^*$, again to first order, and hence it follows that

$$\frac{h}{h_1} \approx \frac{\rho_W \langle C_p \rangle}{\rho_B\ Cp_B} \frac{\int_o^{R_1^+} \frac{2\,dy^+}{(\frac{2}{Pr_B} -1)+\sqrt{1+4\ell^{+2}}}}{\int_o^{R^+} \frac{2\,dy^+}{(\frac{2}{Pr_W} - 1)+\sqrt{1+4\ell^{+2}}}} \tag{11}$$

where $\langle Cp \rangle = (H_W-H)/(T_W-T)$. Finally, it is well known that the ratio of the integrals appearing in (11) is $(Pr_B/Pr_W)^{0.5}$ for $1 < Pr < 20$. Then the ratio of experimental heat transfer coefficient to that calculated by (1) should be

$$\frac{h}{h_1} = \frac{\rho_W}{\rho_B} \frac{\langle Cp \rangle}{Cp_B} \left(\frac{Pr_B}{Pr_W}\right)^{0.5} \tag{12}$$

Summarizing equation (12), there is first a correction ρ_W/ρ_B due to a thickening of the viscous sublayer resulting from the decreased wall density. Secondly, there is a correction $\langle Cp \rangle/Cp_B$ because of the similarity of enthalpy (rather than temperature) profiles, and finally there is the well-known Prandtl number effect.

Equation (12) gives an explanation for the departure of the experimental heat transfer coefficient from that given by equation (1) for the situation $T_B < T_{TC} < T_W$. In this case ρ_W is always less than ρ_B and at high enough wall temperatures $\langle Cp \rangle$ is less than Cp_B. The Prandtl number at the wall is again usually less than Pr_B but the dependence in (12) is weak. On the other hand, when T_W is in the vicinity of T_{TC} it is possible for $\langle Cp \rangle$ to be greater than Cp_B and the ratio given by (12) to be greater than 1.0 as is observed. But in fact (12) predicts too low a heat transfer coefficient by as much as 50%. Shiralkar and Griffith [18] found similar discrepancies even with accurate numerical evaluation of the integrals in (6) (7) and (8) in their comparisons with data for water and carbon dioxide. We should recall that we have assumed the mixing length to be of the same form as for the isothermal case in the absence of any real knowledge of its behavior in the presence of strong property gradients. Considering all the approximations involved it might still be useful to determine empirically the exponents a, b and c in a modified form of equation (12), i.e.

$$\frac{h}{h_1} = \left(\frac{\rho_W}{\rho_B}\right)^a \left(\frac{\langle Cp \rangle}{Cp_B}\right)^b \left(\frac{Pr_B}{Pr_W}\right)^c \tag{12a}$$

As a final point, Shiralkar and Griffith have observed that the deteriorated heat transfer is strongly dependent on upstream conditions. They feel that the thickened viscous sublayer is easily perturbed. Perhaps quantitative prediction is too much to expect without much greater control of experiment.

Based on our experimental results, a fairly safe conclusion for supercritical helium seems to be that equation (1) should not be used if ρ_B/ρ_W is greater than 2.5.

b. Mixed convection

i. Vertical channels

In steady flow of a fluid in a vertical channel there is a balance between the forces acting on the fluid. These are inertial, body forces, pressure gradient forces and shear stresses. In a. above, only the latter two were considered and it is under these conditions that a linear shear stress distribution exists, falling from a wall value to zero at channel center. When a third force is present the shear stress distribution can be entirely different. Thus, when the gravitational force or buoyancy force is considered, it is possible for this force acting on a thin warmer layer adjacent to the inside wall to be balanced entirely by the shear stress at the wall. This can only occur in upward flow with heating or downward flow with cooling. Then, there can be no frictional pressure drop force acting. As a consequence, the core of the flow -- practically at constant temperature -- cannot experience any shear stress, which is then zero all the way to the channel center. In downward flow with heating it is in principle possible for the buoyancy force on the warm layer to be entirely balanced by the pressure drop force. This would result in zero shear stress at the wall, but not in the core.

The importance of shear stress in turbulent flow is that the working of the shear stresses against the mean flow in the proximity of the wall is what generates the turbulence. The turbulent intensity is a result of a balance between generation, convection, diffusion and dissipation. Then the reduction of the shear stress can cause the intensity to decay, with a deterioration in heat transfer coefficient as a consequence.

It is now generally thought [11,19] that the wall temperature spikes referred to above, when observed only in upward flow, are due to this anomalous shear stress distribution caused by the buoyancy forces. A similar phenomenon is observed in accelerated flows where relaminarization can take place. Temperature spikes are not generally observed in downward flow although a broader wall temperature maximum may occur in any orientation due to the transverse property variation discussed in section a. above.

On the basis of these considerations criteria have been proposed to signify conditions under which drastically reduced shear stress can be expected. Thus, Hall [11] gives the criterion

$$Re_B^{2.7} \approx 8.3 \times 10^3 \ Gr \tag{13}$$

where the Grashof number Gr is defined as $Gr = (\rho_B - \rho_W) g (4b)^3 / \rho_B \nu_B^2$, b being the channel half width. Tanaka et al. [19] gave a similar criterion based on a slightly different model

$$Re_f^{21/8} \approx 1.55 \times 10^3 Gr \tag{14}$$

where Gr can be $Gr_g = (\rho_B - \rho_f) g d^3 / \rho_f \nu_f^2$ for the effect of buoyancy or $Gr_a = u_B \ (du_B/dz)(\rho_B/\rho_f) d^3/\nu_f^2$ for the effect of acceleration. In Tanaka's expressions the subscript f indicates that a property is to be evaluated at $T_f = (T_W + T_B)/2$. These criteria effect a division of all heat transfer experiments into two classes: on the one hand pure forced convection, and on the other pure free convection with mixed convection in some ill defined region of the line. In figure 6 we have drawn rough regions encompassing different experimenters results for heat transfer to supercritical helium together with a line representing equation (13). We see that all belong to the forced convection class. The nearest to being mixed convection are the experiments of Giarratano and Jones at 2.5 atm and it is noteworthy that the shaded region closest to the line is where all data from those experiments lie in which experimental heat transfer coefficients are less than 70% of the values given by equation (1). But we have seen above in this case that another explanation exists for such departures; therefore we hesitate to ascribe these departures to buoyancy effects. The two effects will be difficult to separate without simultaneous radial velocity profiles or variation of the channel orientation or flow direction.

It is also interesting to note that, for a given channel at a fixed operating pressure and gravitational acceleration, Grashof numbers obtainable are practically limited by the density differences achievable in the region of the transposed critical temperature. So we may ask the question: under what conditions could we get into a mixed convection region? Assuming a fixed upper limit to density differences, we can either reduce the Reynolds number by reducing the flow rate, or we can increase the Grashof number by increasing the acceleration (e.g., by rotation) or the channel dimension at constant Reynolds number (corresponding reduction in flow rate). In view of practical applications in rotating machinery, or large magnets with large diameter channels, we strongly urge that future experiments cover these situations and that both upward and downward flow be investigated. In all cases, in order to observe temperature spikes a number of thermometers should be used to measure a complete axial temperature profile since a single isolated thermometer could miss such a phenomenon. Pressure drop measurements should also accompany these heat transfer measurements in order to correlate possible observations of temperature anomalies with shear stress in the fluid. The usefulness of radial velocity profiles has already been mentioned.

ii. Horizontal channels

Study of horizontal channels under mixed convection conditions does not appear to be as well advanced as for vertical channels and no studies have been reported with helium. While one expects no such dramatic effect as temperature spikes, caution should nevertheless be exercised because a circumferential variation in wall temperature can result from a buoyancy-induced secondary flow.

A criterion similar to those presented for vertical channels has been developed by Petukhov et al. [21] for a 1% effect of buoyancy on the heat transfer coefficient. With the Grashof number defined as $Gr = \beta_B q_W g d^4/k_B \nu_B^2$ the criterion is given as

$$Gr = 3.0 \times 10^{-5}\ Re_B^{2.75}\ Pr^{0.5}\ [1+2.4\ Re_B^{-1/8}(Pr^{2/3}-1)] \qquad (15)$$

and is plotted in figure 6 with Pr as a parameter. Interestingly, the criterion is very similar to those given for vertical tubes for strong buoyancy effects, but note that, as defined, the Grashof number here may be two orders of magnitude larger.

For typical proposed operating conditions of superconducting power transmission lines operating points fall below the dashed lines representing equation (15). Such conditions may be difficult to simulate in the

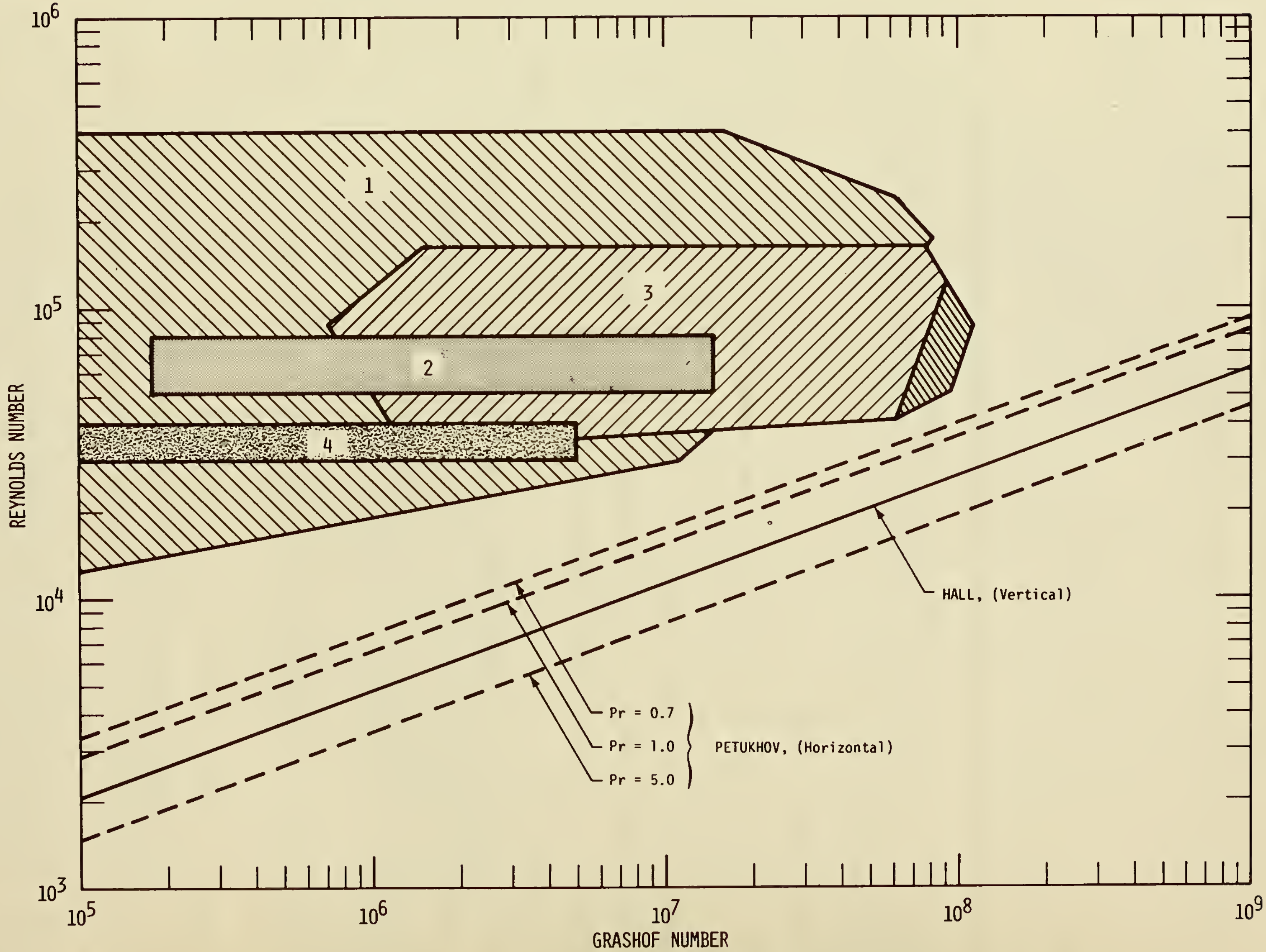

Fig. 6. Convective heat transfer regimes: 1, data of ref. [14]; 2, data of ref. [15]; 3, data of ref. [16]; 4, data of ref. [20].

laboratory, mainly on account of the large disparity between laboratory scale measurements, where diameters are measured in millimeters, and full scale equipment where the diameters may be a factor of 10 larger and Grashof numbers 10^4 larger. Tests should perhaps be performed on large scale prototypes.

In any case, experiments should be performed stressing the measurement of circumferential temperature distributions in the free convection dominated region.

<u>Heat Transfer to Helium I Below the Critical Pressure</u>

The highest heat transfer coefficients for helium flowing in channels are obtained below the critical pressure [22]. This occurs when nucleate boiling takes place at the wall, or in other words, we are in a wetted-wall regime. Indeed, at 2 atm we have observed heat transfer coefficients as high as $\sim 10\ W/cm^2 \cdot K$ under forced flow conditions, although it must be admitted that this is a very approximate figure since the temperature rise was often of the same magnitude as the total estimated systematic error (0.02 K). One very significant fact which emerges from our observations and others is that, in the nucleate boiling regime, the heat transfer coefficient, or the relation between the heat flux q and the temperature rise, $T_W - T_B$, is almost independent of the mass flow rate and vapor mass fraction of helium. It is essentially the same as for unconfined surfaces, or "pool boiling". The relationship is well represented by the well-known semiempirical equation of Kutateladze [23]

$$\frac{h}{k_L}\left(\frac{\sigma}{g\rho_L}\right)^{1/2} = 3.25 \times 10^{-4}\left[\frac{q\ Cp_L\rho_L}{\lambda\rho_V\ k_L}\left(\frac{\sigma}{g\rho_L}\right)^{1/2}\right]^{0.6}$$

$$\left[g\left(\frac{\rho L}{\mu_L}\right)^2\left(\frac{\sigma}{g\rho_L}\right)^{3/2}\right]^{0.125}\left[\frac{P}{(\sigma g\ \rho_L)^{1/2}}\right]^{0.7} \qquad (16)$$

which gives $q \propto \Delta T^{2.5}$ for a given pressure, all properties being evaluated on the saturation line. This relationship was derived by a dimensional analysis of the governing equations and a fairly detailed empirical knowledge of boiling phenomena.

Far more important from a practical standpoint, however, is the transition from this very attractive heat transfer regime to the film boiling or dry-wall regime; for here large temperature excursions take place. In figure 7 we show temperature profiles along our 2 mm dia. vertical test section with downward flow at 1.1 atm pressure. The temperature excursions are obviously quite unacceptable for any superconductivity application. In figure 8 the situation for 2 atm pressure, but with all other conditions essentially unchanged, is seen to be quite different; for now the excursion may be acceptable for some high field superconductors. It is clear that a thorough knowledge of this transition and the conditions under which it takes place are very important, and this was the thrust of our study reported earlier. A similar study was carried out by Ogata and Sato [24] and we find good agreement with their results.

An interesting and useful point of agreement between these two studies is that the transition at any given pressure is predictable within the scatter of the data by a unique relationship between the critical heat flux q_c and the local critical quality x_c. This seems to account implicitly for the separate effects of mass flow rate and position along the channel at which transition occurs. We found that our data on transition supplemented by the low mass flux (natural convection) data of Johannes and Mollard [25] could be represented by the simple relationship

$$Ku = 0.031 + 0.078\,(1-x_c)^{3.92} \tag{17}$$

where Ku is the dimensionless critical heat flux parameter $q_c/(\lambda\rho_V^{1/2}\,[\sigma g(\rho_L-\rho_V)]^{1/4})$ of Kutateladze which has been so successful in correlating transition data for pool boiling. This correlation is shown in figure 9.

We were particularly pleased that this relationship predicted some data of Keilin et al. [26] to within 5%, because this data was for a transition under somewhat different conditions than our own. These authors observed the transition at 278 diameters downstream and local quality of about 0.4 compared to our conditions of less than 48 diameters and quality less than 0.2, for similar pressure and mass flux.

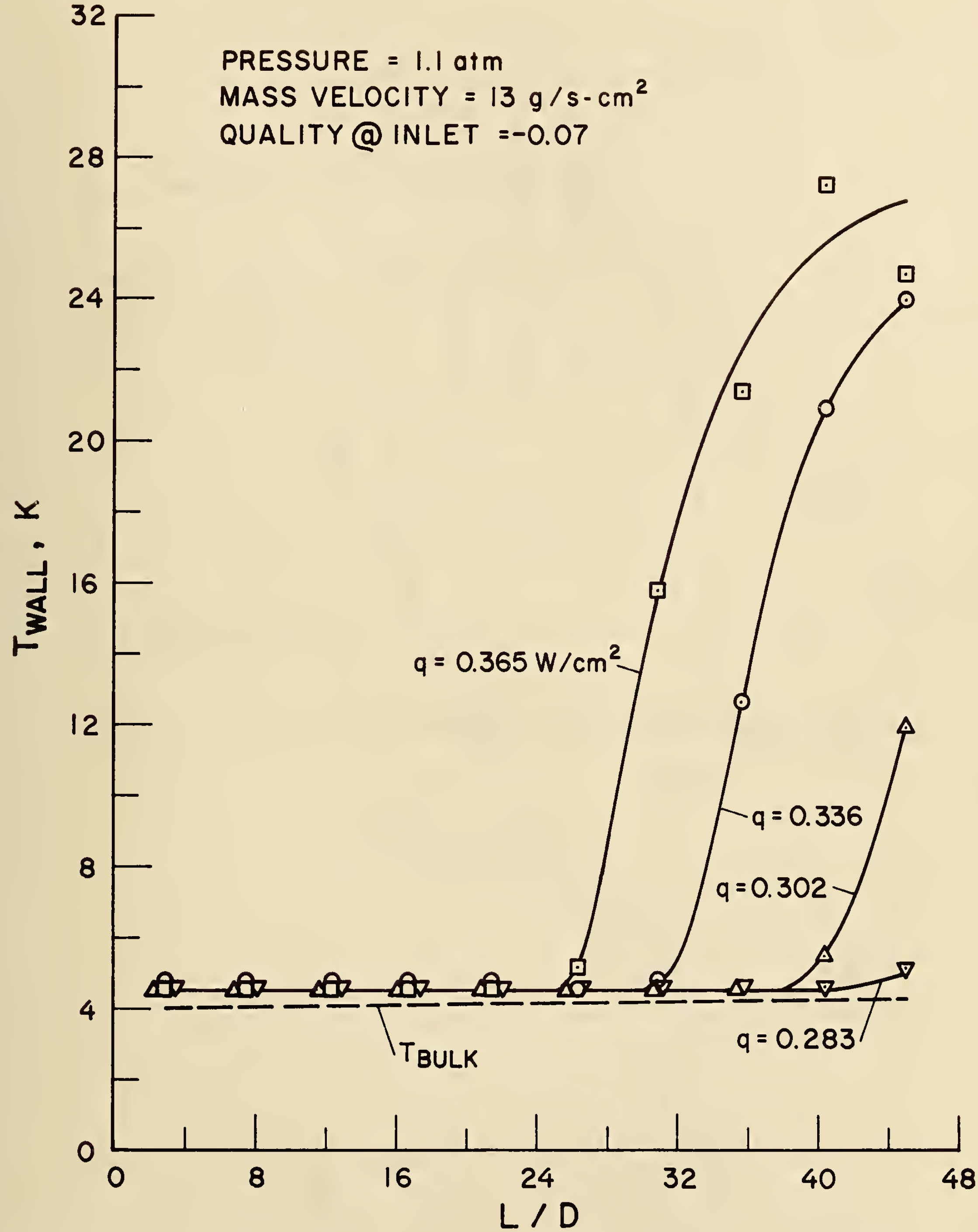

Fig. 7. Wall temperature profiles for sub-critical helium showing transitions to film boiling; 1.1 atm pressure.

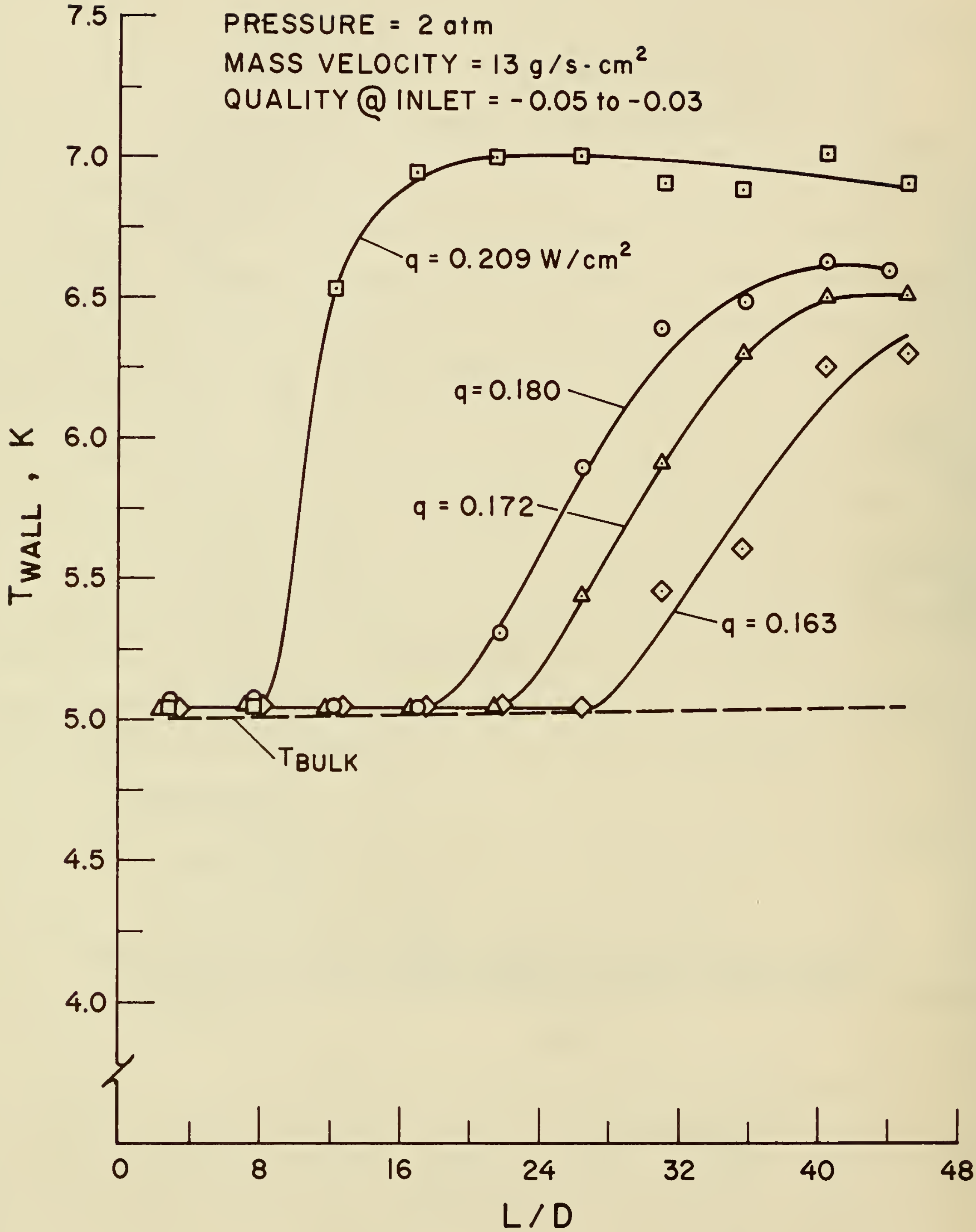

Fig. 8. Wall temperature profiles for subcritical helium showing transitions to film boiling; 2 atm pressure.

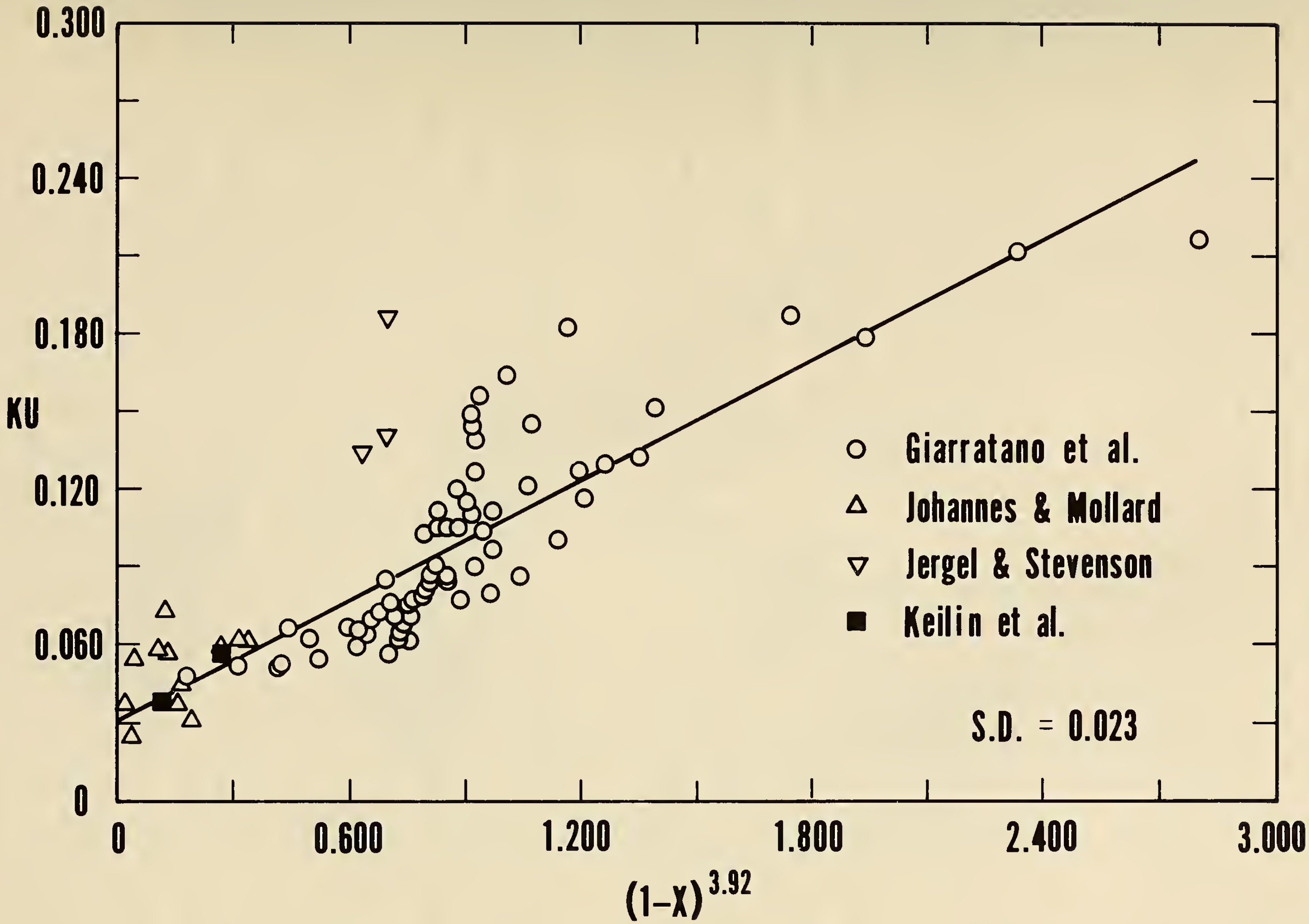

Fig. 9. Correlations for transition to film boiling in forced convection flow in channels.

In a more recent study, Keilin et al. [27] found that their data on transition to film boiling at pressures between 1.1 and 1.5 atm correlated well on a plot of Ku vs. Fr where Fr is a Froude number, $u_L/\sqrt{g\ell}$. The characteristic length ℓ was taken as the characteristic bubble diameter $(\sigma/(g(\rho_L-\rho_V)))^{1/2}$. (Note that this is the square root of the usual definition for Froude number). All data on transition were within ±10% of a line given by

$$Ku = 0.031\ Fr^{0.53} \qquad (18)$$

While these authors do not feel that quality has any explicit influence on the value of the critical heat flux in their experiments, it is not clear that their results are inconsistent with our results or with other authors' results. The difficulty in making precise statements of comparison is that conditions tend to be concentrated in different areas in different experimental arrangements. For instance, while Keilin et al. observed the transition to film boiling at qualities from 0.33 to 0.6, many of our transitions occurred under subcooled boiling conditions (negative thermodynamic quality) and our range was from -0.24 to +0.38. We should like to recommend a review of critical heat flux correlations for helium under flow conditions in channels with all the data in hand.

In the final figure in this section, figure 10, we have summarized our subcritical heat transfer results on a q vs. ΔT plot with shaded areas representing the entire data as labeled. On the same plot for comparison we have shown supercritical heat transfer at 3 atm calculated from equation (2) for a bulk fluid temperature of 4.2 K. This plot shows clearly the unquestioned superiority of the nucleate boiling regime at 2 atm, although extrapolation to even higher subcritical pressures is dangerous since the transition heat flux q_c should go to zero at the critical pressure. This is inherent in the parameter Ku. If a few degrees ΔT may be permitted there is little to choose between film boiling at a subcritical 2 atm pressure and the supercritical 3 atm pressure, but it is clear that film boiling at 1 atm should be avoided. Again, referring to the previous section, ΔT greater than ∿ 2 K is not properly represented by equation (2) for here the density ratio ρ_L/ρ_V is beyond the limit for which the equation was developed.

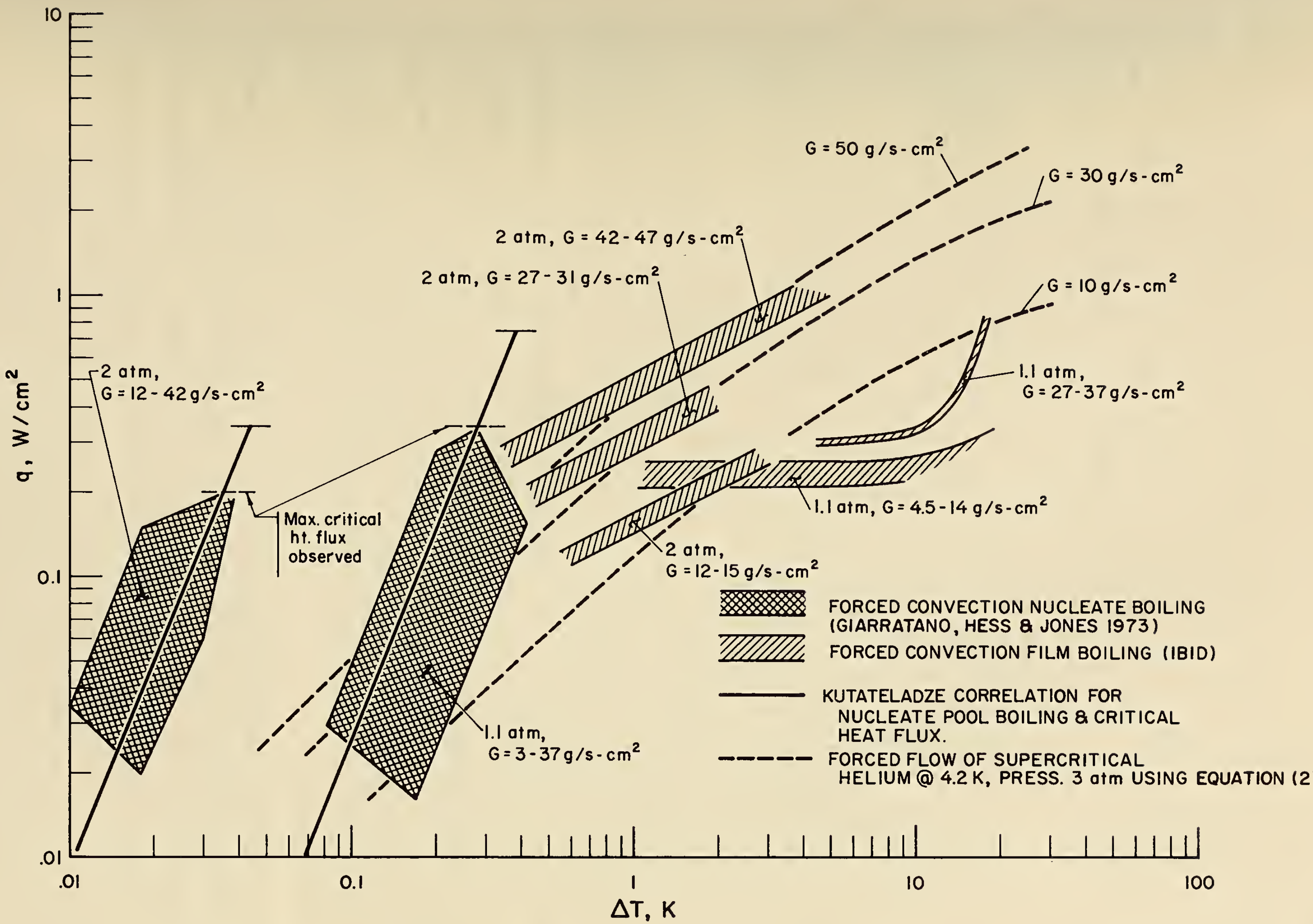

Fig. 10. Summary of subcritical heat transfer data from reference [22] and comparison with supercritical heat transfer by equation (2) for 3 atm.

Heat Transfer to Helium II

It appears likely at this time that some applications of superconductivity will require operation at temperatures below the lambda point, 2.17 K, of helium. The study of heat transfer in the He II phase (the superfluid phase of liquid helium) remains largely separate from studies of heat transfer in all other liquids. The main phenomena of superfluidity are so different, and the vocabulary that describes them so specialized, that communication to persons outside the field can be difficult. Before discussing superfluidity itself, however, we should first mention briefly one special problem in heat transfer that probably has nothing to do with superfludity but is ordinarily only detectable and important at He II temperatures, and that is the "thermal boundary resistance" ("Kapitza resistance") between dissimilar materials. It is characterized by a temperature jump at the interface between two materials which is proportional to the heat flux passing through that interface.

Most studies have been done on He II - metal interfaces, where the effect is largest, and where low bulk thermal conductivities do not obscure it. Its strong temperature dependence ($\propto T^{-3}$), and the low thermal conductivity of He I, ordinarily make it unimportant for $T > 2.2$ K. In He II its magnitude can become a limiting factor in heat transfer. Perhaps the bigger problem is that its value is difficult to predict and control. For "clean" Cu-He interfaces, reported values at $T = 1.9$ K range from 1.3 to 8.3 $cm^2 \cdot K/W$ [28]. Even surfaces prepared by the same procedure may differ by 50%. This variation is known to be caused by the great sensitivity of the effect to the exact condition of the surface, but the detailed effects of, e.g., dislocations, is still controversial.

One of the most characteristic features of He II work concerns conduction of heat through the liquid itself; the unique behavior can be grasped once the main features of the "two-fluid" model are understood. It supposes we can view He II as consisting of a homogeneous mixture of two fluids, called the "superfluid" and the "normal fluid". The relative concentrations are determined only by the temperature (and weakly, the pressure), but the velocities of the two fluids may differ, thus nearly doubling the number of variables as compared to ordinary hydrodynamics.

In most respects, the normal fluid fraction behaves like an ordinary fluid. On the other hand, the superfluid fraction has the unique properties of carrying no entropy, and, in certain circumstances, of obeying an Euler type equation of motion (perfect fluid equation). Both fluids have the unique property of being accelerated (strongly) by a temperature difference as well as a pressure difference, the superfluid fraction being driven toward the higher temperature and the normal fluid fraction being driven away.

These properties allow heat to be transfered within the stagnant liquid by a unique type of "internal convection" or "counterflow". The superfluid fraction flows toward the heat source, absorbs heat by its conversion to normal fluid, which then flows back toward the heat sink. At low heat currents (low relative velocities) the two fluids do not interact, so that the two fluids can flow through each other unhindered, and effective thermal conductivities as measured along the axis of a tube can easily be many orders of magnitude larger than in good metals. Unfortunately this excellent state of affairs has a limit, because the two fluids begin to interact, at some "critical axial heat flux" (typically .01 - .1 W/cm^2) which is dependent on the geometry, by a process which is called "mutual friction". The temperature gradient then starts to increase strongly, approximately as the third power of the heat flux. In most practical situations, i.e., diameters $\gtrsim$ .05 cm, T $\gtrsim$ 1.5 K, q > .01 W/cm^2, mutual friction will be dominant; even so, the effective thermal conductivity is still orders of magnitude superior to any metal at these temperatures, as can be seen in figure 11. For quantitative discussions of these topics, the reader can consult existing reviews [10] and monographs [29,30]. We note that natural convection, and nucleate boiling are never observed.

For technological purposes, e.g., cryogenic stabilization of superconducting magnets, it is the peak, or limiting, heat flux (also called the "critical heat flux") that can be accommodated without a large temperature increase, that is of most interest. Most of the available information suggests that in He II this limit corresponds to the transition to film boiling: the wall is no longer wetted and its temperature jumps dramatically. The simplest assumption is that this occurs when the local temperature (determined by the heat current distribution and mutual friction) becomes somewhat

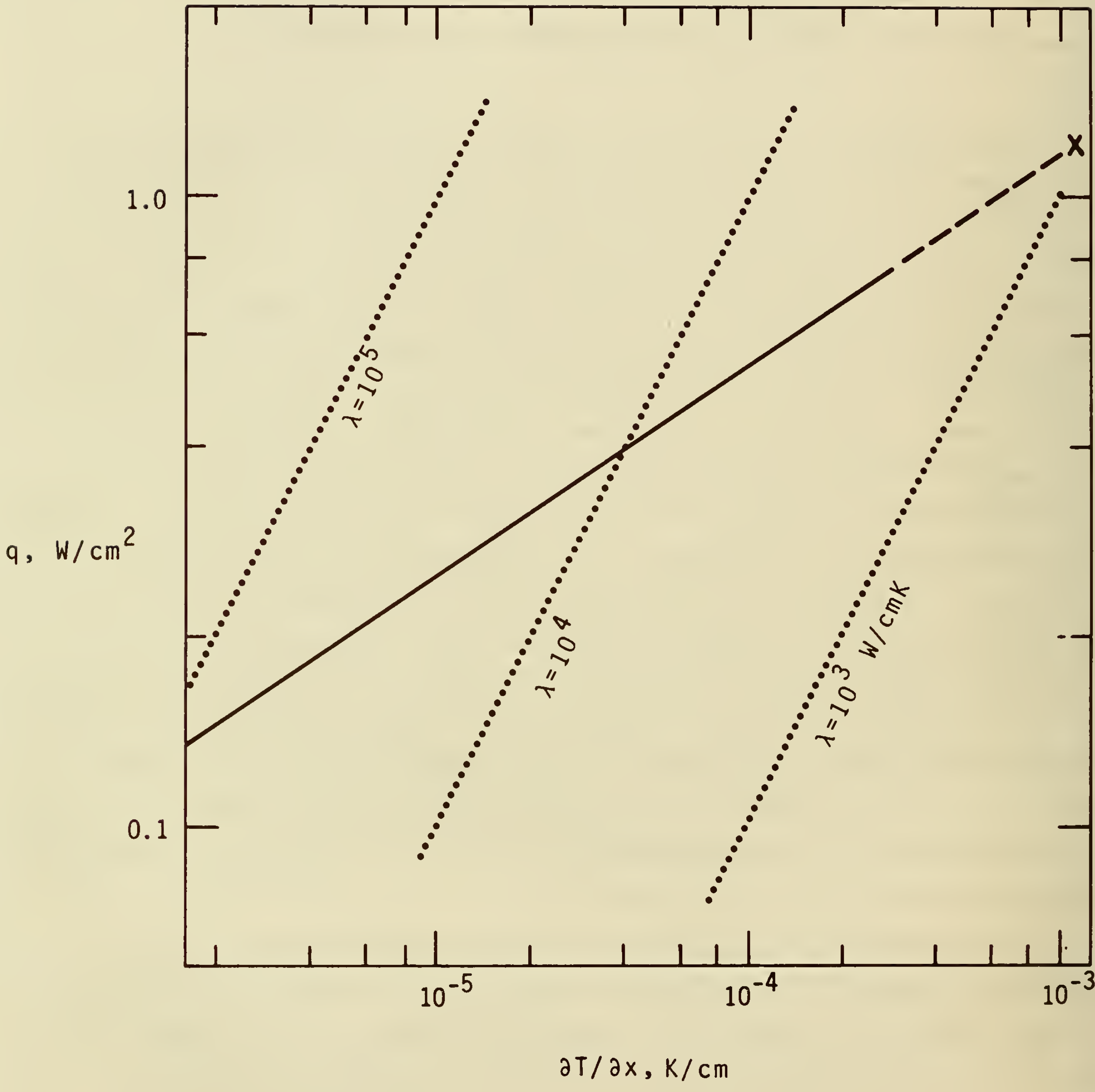

Fig. 11. The axial heat flow (solid line) along a tube filled with He II at T = 1.9 K, as a function of the temperature gradient, where "mutual friction" is dominant. The dashed portion of the curve indicates the approximate range of the limiting heat flux. The dotted curves are lines of equal effective thermal conductivity.

greater than the saturation temperature appropriate to the local pressure, just as in helium I. In long heated tubes (perhaps length/diameter > 10) without net axial mass flow, this limiting heat flux turns out to be roughly independent (± 30%) of length, width, and heat flux distribution at the liquid boundary, and to have a value of about 1 watt per cm^2 <u>of tube cross-section</u>. It should be pointed out that substantial pressurization (subcooling) increases the limiting heat flux only slightly [31].

One outstanding problem is that of limiting heat fluxes in more open geometries, such as flat plates or cylinders far from a wall, where the heat flux need be large only quite near the heated surface. Experiments have indicated considerably larger peak fluxes, 3-6 W/cm^2 for centimeter-sized surfaces, and a marked depth dependence (sub-cooling dependence), in qualitative agreement with the model outlined above [32,33]. We need more experiments where wall temperatures and liquid temperatures near the wall are measured, and where careful attention is paid to nearby surfaces that can channel or obstruct the heat currents. The influence of the surface condition, and the degree of super-heating are also of interest.

Whether the limiting heat flux in long channels can be increased is an important and open question. The most straight-forward, and possibly only way, i.e., by use of forced convection of the He II, raises a number of questions.

We can formulate the model situation as follows: net mass flow of He II through a heated tube is produced by a pressure drop along it, and sensible heat is absorbed by the temperature rise of the fluid. If such a situation can be produced, it will undoubtedly increase the limiting flux: a temperature rise from 1.8 to 2.0 K and a velocity of 100 cm/s corresponds to a heat flux of about 10 watts per cm^2 of tube cross-section.

The implicit assumption, which we take for granted with an ordinary fluid, is that mass flow is driven by the pressure gradient <u>only</u>. In He II we know that in at least certain special situations, e.g., the "fountain effect", a large mass flow is driven by a temperature gradient only. We can not predict with any certainty how the temperature gradient will change the pressure gradients needed to produce a given net mass flow. The difficulty is that there are large gaps in our knowledge of two-fluid hydrodynamics. For example, the conventional formulations of the theory of "mutual friction", and almost all of the experiments, consider only the case of zero net mass flow. It would not be surprising if, at the least, some of the empirically determined quantities of that theory are a function of the net mass flow. What we need to know is how both the heat flow rate and the mass flow rate along the tube depend on both the temperature gradient and the pressure gradient, and also how they depend on tube diameter and temperature. Despite these uncertainties, our best guess is that in millimeter-and larger-sized geometries and at large mass velocities the usual benefits of forced convection will be obtained.

A final question concerns pumping the superfluid fraction. A truly ideal fluid, e.g., could not be set into rotation, hence could not be centrifugally pumped. In fact, the ideality of the superfluid fraction usually breaks down for large velocities, so we expect conventional pumps to work in some fashion [34]. However, there have been no in depth investigations of pump characteristics in helium II. One speculative possibility that has occurred to us is to close off one end of a tube with a "super-leak", the other end being open, net mass flow then being driven through the tube by the "fountain effect". Such a system was demonstrated long ago with a point heat source near the superleak [35]. Our guess is that such a "fountain pump" will also work well with a distributed heat source, and prove attractive from an engineering standpoint, because of the absence of moving parts.

For a long time it has been known that He II is a superior heat transfer medium for small heat fluxes. We expect that further research, if care is taken in understanding the unique properties of He II, will show it to be a superior medium at high heat fluxes and to have a place in the technology of superconductivity.

Oscillations in Forced Flow Helium Cooling Systems

To date no systematic experimental study of oscillations in forced flow helium systems has been reported. Rather, this has been a phenomenon frequently referred to, but more as a nuisance in the pursuit of other goals, notably the heat transfer studies reported above, than as a subject for study.

We can identify three distinct modes of oscillation in forced flow systems from a large body of literature on other cryogenic and higher temperature fluids. These are

a) Density wave oscillations [36,37]
b) Acoustic oscillations [38,39]
c) Pressure drop or Helmholtz oscillations [40].

In a) the mechanism is that of an enthalpy and density disturbance travelling at the stream velocity. It can be shown that such a disturbance becomes amplified due to thermal expansion as it travels down the channel. The resulting pressure drop disturbance is propagated back at sonic velocity, but arrives at the inlet delayed in time by the residence time of the fluid in the heated channel. Depending on the nature of the source flow impedance this pressure drop disturbance can again induce a flow and hence density and enthalpy disturbance at inlet. If it is larger than the initiating disturbance a self-sustained oscillation is set up with period approximately equal to the residence time of fluid in the channel.

The oscillations of type b) are of the "organ pipe" kind with a standing wave pattern set up between flow discontinuities (e.g., a valve). It appears necessary for such oscillations to be self-sustained that heat addition be concentrated within a part of the standing wave where velocity fluctuations are all in the same sense and such as to cause a fluctuating heat addition in phase with the pressure.

Type c) appears to occur when a reservoir of compressible fluid feeds a heated channel which exhibits negative differential flow resistance, a phenomenon characteristic of two phase and supercritical fluids. In this case, the channel pressure drop is in phase with the reservoir pressure when the reservoir-channel system oscillates in the Helmholtz (mass/spring) mode, which is thereby sustained.

Provided the time constant for radial thermal diffusion is small compared to the period of oscillation, these can all be studied by the application of classical linear stability theory to the one dimensional conservation equations of mass, energy and momentum describing flow in the heated channel. But of equal importance to the dynamics of the channel itself are the external flow impedances or acoustic impedances at inlet and outlet of the channel.

We have made calculations for type a) for supercritical helium systems. We find that under certain circumstances (e.g., low pressures and proximity to the transposed critical temperature) these oscillations are a real possibility in supercritical helium. An example of computed frequency response for a heated channel at 2.5 atm taken from [37] is given in figure 12. The "open loop gain" and "phase angle" refer to a velocity perturbation at the entrance of the heated channel. This case is that of a simple uniformly heated circular channel fed from a constant pressure reservoir via an inlet flow restriction and discharging via a second flow restriction to a second constant pressure reservoir. The flow restrictions define the flow impedances external to the channel. With the open loop gain greater than unity at a frequency where the phase angle is $-\pi$, this system must be unstable. Two-phase helium systems should be even more susceptible to this type of instability. However, this system can be stabilized by increasing the upstream flow restriction at the cost of increased system pressure drop.

Stability computations for acoustic oscillations, type b), are considerably more difficult and those that have been reported [39] have not permitted generalization of the results. We therefore still lack criteria that can be applied to helium systems.

Arp [41] has studied the possibility of negative differential flow resistance in helium cooled channels and finds that this is only a possibility at inlet conditions of below about 3.5 K and below about 3 atm for short channels with high heat flux. Thus, oscillations of type c) may be of very limited possibility in applications of superconductivity, but we are investigating other possibilities for sustaining this mode of oscillation.

In all cases it is necessary to understand that, for a given mode, stability is not determined by helium properties and the parameters of the heated channel alone. Of equal importance are the flow impedances at either end of the channel and the external coupling, if any, between hydraulic and thermal disturbances. These determine the manner in which propagated disturbances are reflected or transformed.

A thorough understanding of the mechanisms of oscillations and their stability criteria will enable successful design ensuring stable performance. This should not be left to chance. Experimental verification of the theoretical methods at our disposal is urgently needed.

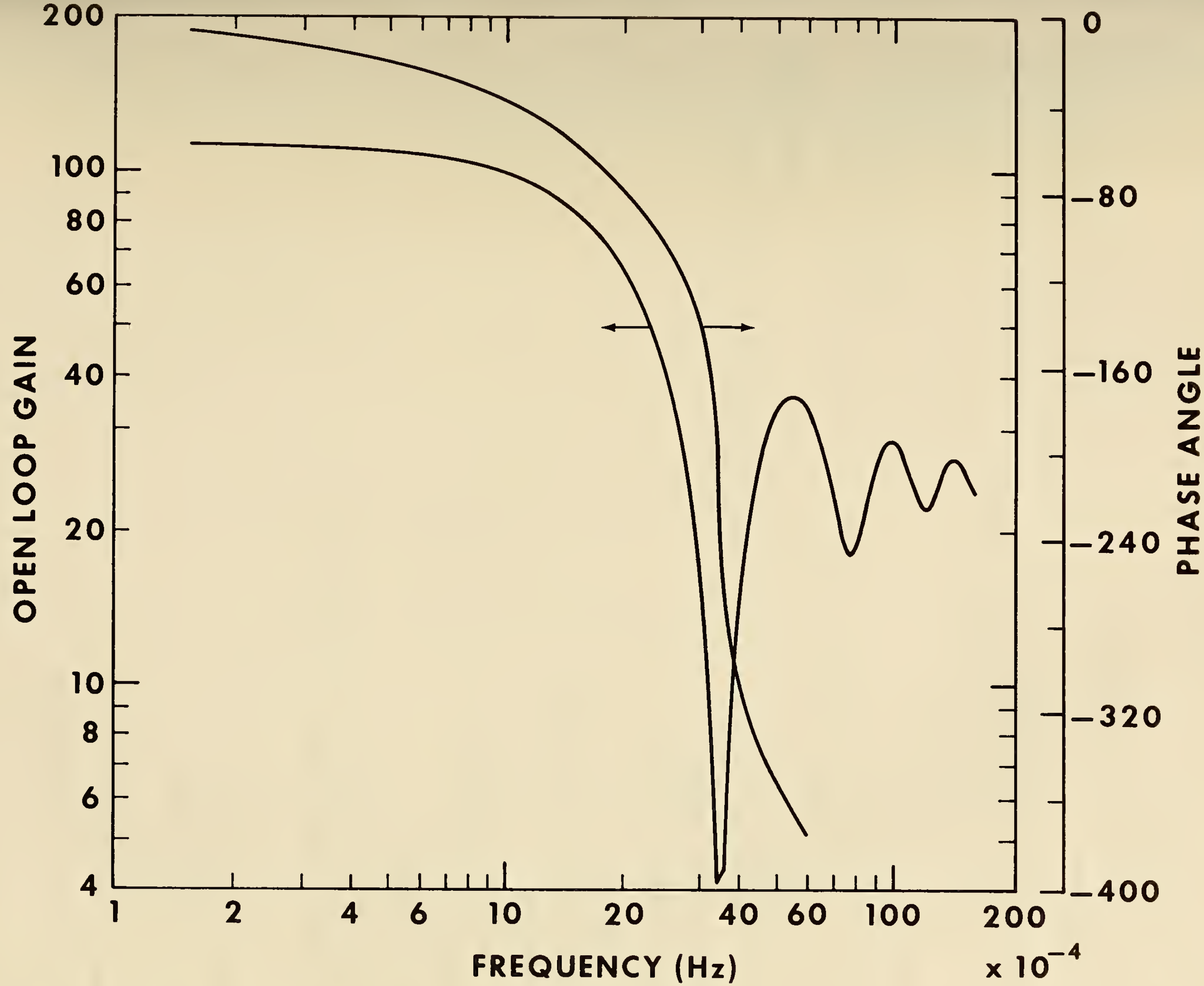

Fig. 12. Computed frequency response for a channel 240 m in length, 0.5 cm diameter with 1.0 g/sec helium entering at 4.5 K and 2.5 atm. Heat load 0.157 W/m. Inlet and exit restriction equivalent to 0.25 cm diameter orifices.

Acknowledgments

The many contributions of our colleagues Patricia J. Giarratano and Vincent D. Arp are gratefully acknowledged.

Nomenclature

b	channel half width
C_p	constant pressure heat capacity
d	tube diameter
g	gravitational acceleration
Gr	Grashof number, variously defined in text
h	heat transfer coefficient
H	enthalpy
k	thermal conductivity
Ku	Kutateladze parameter, defined in text
ℓ	mixing length
$\dot{m}$	mass rate of flow
Nu	Nusselt number, hd/k
Pr	Prandtl number, $\mu C_p/k$
q	heat flux
Re	Reynolds number, $\rho \langle u \rangle d/\mu$
T	temperature
u	axial velocity component
u*	friction velocity
v	specific volume
x	thermodynamic quality
y	coordinate perpendicular to channel wall

Greek

α	thermal diffusivity
β	thermal expansivity
ε	eddy diffusivity
ν	kinematic viscosity
λ	latent heat of vaporization
ρ	density
σ	surface tension
τ	shear stress
μ	viscosity

Subscripts

B	bulk fluid
C	critical, referring to boiling transition
f	film
H	heat
L	liquid
M	momentum
TC	transposed critical
V	vapor
W	wall
0	reference value
1	constant property case

Superscripts

+	dimensionless parameters defined in text

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15. SUPPLEMENTARY NOTES

16. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here.)*

Heat transfer and fluid mechanics of helium flowing in channels are reviewed. Emphasis is placed on observed or anticipated limits of operation which might be expected to apply in applications of superconductivity. Topics included are: the high-heat-flux degradation of heat transfer and possible effects of buoyancy forces in supercritical helium; transition to film boiling in subcritical helium; limiting heat currents in helium II; and the possibility of oscillations in forced flow helium systems.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)* Buoyancy; channels; film boiling; forced convection; helium I; helium II; mixed convection; nucleate boiling; subcritical; supercritical; transition; turbulent flow.

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