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A LITERATURE SURVEY ON THE APPLICABILITY OF STEADY-STATE
HEAT-TRANSFER COEFFICIENTS TO TRANSIENTS IN POWER OR IN
FLOW WITH SINGLE-PHASE FLOW AND WITH TWO-PHASE FLOW

by

R. H. Norris

Report No. 57GL346

November, 1957

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TECHNICAL INFORMATION SERIES

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TITLE / Literature Survey on the Applicability of Steady-State Heat-Transfer Coefficients to Transients in Power or in Flow with Single-Phase Flow and with Two-Phase Flow		
ABSTRACT The known "unclassified" literature was reviewed and found to provide quantitative information on this subject for only certain simple cases, as indicated below under "Conclusions." The scope and limitations of our available information are briefly discussed, and a bibliography presented.		
G.E. CLASS III GOV. CLASS. NONE	REPRODUCIBLE COPY FILED AT LIBRARY OF GENERAL ENGINEERING LABORATORY SCHENECTADY, NEW YORK	NO. PAGES 15
CONCLUSIONS Quasi-steady-state heat transfer coefficients seem to be applicable for time intervals Δt (after a step change) greater than the following values: A. For forced-convection flow along flat plates: 1. With step change of velocity, with laminar layer: $\Delta t = 3.4 x/V$ 2. With step change of temperature: Lam. $\Delta t = 6 (c_{pu}/k)^{1/3} x/V$; Turb. $\Delta t = 10 c_{ppk}/(h)^2$ 2. For "ramp" change of temp., see eq. 2a. B. For forced convection in a duct, step change in temp. $\Delta t = 3 D/V$ C. For free convection on vertical surface, with step change of temperature: $\Delta t = 5.24 (0.95 + c_{pu}/k)^{1/2} (g\beta\Delta t)^{1/2} x^{1/2}$ D. For boiling with step change in surface heat source, at 1 atmosphere: $\Delta t = 10$ milliseconds (roughly); for heat sources in the water itself, Δt is much greater.		

INFORMATION PREPARED FOR Knolls Atomic Power Laboratory

TESTS MADE BY No new tests.

AUTHOR R. H. Norris *R. H. Norris*

COUNTERSIGNED R. H. Norris *R. H. Norris*

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Introduction

This report presents a survey of all the literature known to the author (or located after a reasonable search effort) on the subject indicated by the title. It was prepared at the request of the Knolls Atomic Power Laboratory. This survey consists of four parts and two appendices, namely:

- I. Response of Single-Phase-Fluid Forced-Convection Heat Transfer to a Rapid Change of Velocity p. 2
- II. Response of Single-Phase-Fluid Forced-Convection Heat Transfer to a Rapid Change of Temperature of the Surface or of the Incoming Fluid p. 4
- III. Response of Single-Phase-Fluid Free-Convection Heat Transfer to a Rapid Change of the Surface Temperature or of the Heat Flux Through the Surface p. 7
- IV. Response Time for Boiling Heat Transfer to a Sudden Increase in Heat Flux, or in Temperature of the Wall p. 8
- Appendix A. Summary of Range and Nature of Published Test Results on the Response to a Sudden Rapid Rise in Heat Input Which Brings a Surface Immersed in Water to a Temperature Above the Boiling Point p. 9
- Appendix B. Symbols - Nomenclature p. 12
- References p. 13

1. Response of Single-Phase-Fluid Forced-Convection Heat Transfer to a Rapid Change of Velocity

For turbulent flow, the writer has so far located no literature giving practically useful data directly on this specific topic. It is common practice, in analysis and test of skin temperatures of missiles accelerating at several "g", to assume "quasi-steady-state" conditions for evaluating convective heat transfer even though the velocity is changing at a moderate rate. By "quasi-steady-state" is meant that the heat transfer is evaluated from the steady-state correlations applied to the bulk velocity existing at each instant considered.

It seems reasonable to assume the validity of quasi-steady-state conditions when the rate of change of velocity, V , is slow enough so that the time, $\Delta\tau$, required for a significant change (say 10%) in velocity, V , corresponds to a displacement ($\Delta x = V\Delta\tau$) of the fluid which is at least several times larger than the characteristic dimension, L , of the surface considered (e.g. diameter of a duct, or length of flat plate parallel to the flow).

This is equivalent to a time interval:

$$\Delta\tau = N \frac{L}{V} \quad (1)$$

where N is several times unity.

The one case where analysis exists for transient flow conditions is the case of laminar flow over a flat plate parallel to the flow. Topper (18) has derived the criterion

$$\Delta\tau = 3.4 \frac{x}{V} \quad (2)$$

for the time interval, after a sudden ("step function") velocity change, required to make the boundary layer thickness at distance x , from the leading edge, "substantially reach its final value." But Topper does not seem to specify "substantially" in quantitative terms. Note that eq. (2) is of the form proposed in eq. (1).

Moore (14) proposes a slightly different form of criterion for laminar flow with a velocity changing at the rate $dV/d\tau$. He suggests that if the quantity

$$\left(\frac{dV}{d\tau}\right) \delta^2 / (\nu V)$$

is "small" then quasi-steady-state flow conditions may be assumed to exist. Here " δ " is the thickness of the boundary layer. This proposal of Moore can be shown to reduce to a relation of the same form as eq. (1) or eq. (2), with the constant N in eq. (1) remaining unknown, however, since Moore does not define his term "small" quantitatively.

One type of changing velocity deserves special comment, namely a decelerating flow. If this deceleration is rapid enough to require a positive gradient of static pressure (increasing pressure in the streamwise direction), this gradient will tend to cause separation of the boundary layer (as happens in steady flow in an expanding duct), as remarked by Schlichting (16). This separation will affect the convective heat transfer; but no method for predicting this effect is known to this writer.

Another feature of decelerating flow which deserves comment is the fact that with such flow it seems conservative to assume quasi-steady-state values for the heat transfer coefficient, i.e., such coefficients will be smaller than actually occur. The reason is given by Larkin, (29), who notes that with such flow the velocity-boundary-layer thickness increase tends to lag, slightly, the decrease in the bulk flow.

II. Response of Single-Phase-Fluid Forced Convection Heat Transfer to Rapid Temperature Changes of the Surface or of the Incoming Fluid

Assumption of quasi-steady-state conditions seems reasonable for evaluating local heat transfer coefficients, with time-varying surface temperature provided the rate of change of temperature is not too rapid. The criterion by which to judge what rate of change is "too rapid" will of course depend on both the shape of the surface, and on whether the boundary layer, or velocity profile, is that of laminar flow or of turbulent flow. These two types of flow are considered in turn below.

The assumption of quasi-steady-state heat transfer coefficients is conservative in any case where the surface temperature, or the heat flow from surface to fluid, is increasing with time; i.e. the assumption will yield calculated surface temperature higher than actual. The reasoning supporting this statement has been presented by Larkin (28).

A. Laminar Flow

For surface temperature, t_w , changing at a finite rate, a theoretical expression for the deviation of the heat-transfer coefficient from its quasi-steady-state value has been derived by Sparrow and Gregg (22), for one simple configuration, namely a hot flat plate of uniform temperature parallel to a flowing fluid. Their result is:

$$\frac{h_{\text{inst}}}{h_{\text{quasi-steady}}} = 1 + 2.40 \frac{dt_w/d\tau}{\Delta t} \left(\frac{x}{V}\right) - 0.80 \frac{d^2 t_w/d\tau^2}{\Delta t} \left(\frac{x}{V}\right)^2 + \dots \quad (3a)$$

For a sudden "step-function" jump in surface temperature of such a plate, Topper (18) has derived a theoretical expression for the time interval, $\Delta\tau$, immediately subsequent to this jump, beyond which quasi-steady-state heat transfer coefficients are "well-approximated." His result is

$$\Delta\tau = 6 \left(\frac{c_p \mu}{k}\right)^{1/3} \left(\frac{x}{V}\right) \quad (3b)$$

This expression is similar in form and in order of magnitude, to that in eq. (2), except that the cube root of Prandtl number has been included as a factor.

No test data appear to be available by which to check eq. (3a) or (3b).

B. Turbulent Flow

For turbulent flow, the only criteria available for the limitations to the assumption of quasi-steady-state for evaluation of heat transfer coefficients seems to be the "order of magnitude" estimates of Larkin (28). For a turbulent boundary layer, with an effective laminar sub-layer of thickness δ_L ,

(defined by: $\delta_L = k/h$) he derives the estimate:

$$\Delta\tau = 10 \frac{c_p \rho \delta_L}{h} = 10 \frac{c_p \rho k}{h^2} \quad (3c)$$

For a duct, by an alternate method of reasoning, Larkin (28) derives the criterion:

$$\Delta\tau = 3 \frac{D}{V} \quad (3d)$$

which does not differ greatly from eq. (1), eq. (2), and eq. (3b). For the one example considered by Larkin, the results by eq. (3c) and by eq. (3d) differ by only 10%.

No test data for checking Larkin's criteria seem to be available. In spite of the lack of criteria for the limitations to the assumption of quasi-steady-state heat transfer (Larkin's paper is only recent, with very limited distribution), it has been common practice to use this assumption in analyses of transient temperature changes for turbulent flow in fluid-to-fluid heat exchangers (11, 12, 13), for coolant of nuclear reactors (10 and 24), for skins of wind-tunnel models tested with short duration "flows" (19), and for flow in aircraft ducts (30).

The assumption of quasi-steady-state heat transfer has also been used in two types of non-steady-state tests to measure heat transfer coefficient, both by the transient cooling rate of a hot surface suddenly immersed in a cool flow (20), and by observing the effect of sinusoidally varying the inlet fluid temperature to a matrix of heat exchanger elements (21); in the latter case, the reduction in amplitude of the outlet temperature of the fluid, relative to that of the inlet temperature is used as a measure of the heat transfer coefficient.

But the results of the reported tests, using these non-steady-state methods are unfortunately not in a range of conditions which would be suitable for yielding adequate evidence on the question of the time required for the heat-transfer coefficient itself to reach a quasi-steady-state condition. In the reported use of the method of sudden immersion (20), the time constant of the changing temperature of the surface was relatively very long (20 to 90 seconds). In the reported use of the method of sinusoidal variation of fluid temperature (21), the period of the fluctuations, although less, was not less than 2 seconds, still too long to be adequate for the purpose here considered (judging from the results explained below). For two of the three matrix samples tested, the heat transfer coefficient obtained by the sinusoidal temperature method agreed reasonably well with the correlations obtained previously by steady-state methods. Although the results for a third matrix showed a 20% discrepancy, no correlation with the temperature fluctuation frequency was established.

Some theoretical consideration has been given by Topper (26) to the limitation to increase in frequency of the sinusoidal temperature of the fluid, but it is limited to the effect of lengthwise diffusion of heat flow, and does not consider the effect on the transverse profiles of temperature and of velocity (which determine the convective heat transfer coefficient). Topper's paper (26) considers the laminar as well as the turbulent types of flow (in a duct).

C. Effect of Surface Temperature Non-Uniformity

One consideration in the assumption of quasi-steady-state heat transfer particularly deserves mention, namely that streamwise non-uniformities of surface temperature may occur, and may vary with time. If these non-uniformities are large, they can appreciably affect the local heat transfer coefficient. This effect can be taken account of as explained, for example, in ref. 15a, 15b, 15c, and 29.

If the surface temperature changes too rapidly to justify the assumption of quasi-steady-state conditions, but the flow remains reasonably constant, then the temperature distribution spacewise and time-wise for turbulent flow in a long straight duct could be obtained by computation, using the eddy conductivity distribution transversely (which is known for steady flow). A procedure for a closely related problem is explained in ref. (17).

III. Response of Single-Phase-Fluid Free-Convection Heat Transfer to Rapid Changes of the Surface Temperature, or of Heat Flux, Through the Surface

For this case, theoretically derived results have only recently become available (22 and 23). They give the time, τ_c , required to reach approximately steady-state conditions of heat transfer for a vertical plate in the laminar-flow range of free convection.

For a uniform surface temperature, after a sudden step-function rise above ambient temperature, Siegel's results (22) are available by two different alternative approximations. They are, in dimensionless form (for symbols list, see Appendix B):

$$\left(\frac{k}{c_p \rho}\right) \frac{\tau_c}{L^2} = 5.24 (0.95 + c_p \mu/k)^{1/2} \left(\frac{g \beta \Delta t L^3}{\nu^2}\right)^{-1/2} \left(\frac{c_p \mu}{k}\right)^{-1} \left(\frac{X}{L}\right)^{1/2} \quad (4)$$

and

$$\left(\frac{k}{c_p \rho}\right) \frac{\tau_c}{L^2} = 7.10 (0.377 + c_p \mu/k)^{1/2} \left(\frac{g \beta \Delta t L^3}{\nu^2}\right)^{-1/2} \left(\frac{c_p \mu}{k}\right)^{-1} \left(\frac{X}{L}\right)^{1/2} \quad (5)$$

Siegel (22) suggests using the arithmetic average of these two results.

In dimensional form, eq. (4) reduces to a simpler form, namely:

$$\tau_c = 5.24 (0.95 + c_p \mu/k)^{1/2} (g \beta \Delta t)^{-1/2} X^{1/2} \quad (4a)$$

For the example of air at 70°F and one atmosphere pressure, with $\Delta t = 200^\circ\text{F}$, and $X = 1$ foot, this yields

$$\tau_c = 1.9 \text{ seconds} \quad (6)$$

Topper (23) obtains a result equivalent to eq. (4) except that the numerical factor 5.24 is replaced by 1.4. Siegel's analysis seems more refined and detailed than Topper's so his results are preferred by this writer. The difference seems to be mainly in the difference in the value specified for the boundary-layer thickness, considerably smaller in Topper's analysis.

For a heat flux suddenly raised from zero to q'' , the approximate relation derived and recommended by Siegel (22) is:

$$\frac{k}{c_p \rho} \frac{\tau_c}{L^2} = 4.78 (0.80 + c_p \mu/k)^{2/5} (g \beta q'' L^4 / \nu^2 k)^{-2/5} (c_p \mu/k)^{-4/5} \left(\frac{X}{L}\right)^{2/5} \quad (7)$$

Topper (23) does not consider this case.

No test data appear to be available for comparison with these theoretical results.

IV. Response Time for Boiling Heat Transfer with Sudden Increase in Heat Input, or in Temperature at the Wall

There is indeed a slight time delay in the response of the heat transfer coefficient to a very rapid or sudden rise in temperature of the surface of a solid heat source submerged in liquid water. But this delay is of the order of only 10 milliseconds, or less, for subcooled water at a pressure of one atmosphere (absolute). This conclusion is supported by test evidence, from several sources, as reviewed in Appendix A. For saturated water, at atmospheric pressure, the bubble growth rate indicates that good heat transfer is well established in 10 to 15 milliseconds (see Appendix A, item 5).

For surface heat sources submerged in water at much higher pressure, no data on the amount of time delay are known to this writer.

For volume-heated water (heat generation in the water itself, instead of in the wall), test data have been reported by Greenfield et.al. (25). In this case, time delays of from 80 to 400 milliseconds were observed in the start of density decrease after a sudden step in heat input to initially saturated water; the pressures ranged from one to nine atmospheres. The longer time delays in this case seem due to the need for nuclei for formation of bubbles at low superheat.

APPENDIX A

SUMMARY OF RANGE AND NATURE OF PUBLISHED TEST RESULTS ON THE RESPONSE TO A SUDDEN RAPID RISE IN HEAT INPUT WHICH BRINGS A SURFACE IMMERSED IN WATER TO A TEMPERATURE ABOVE THE BOILING POINT

All the test results considered in this appendix apply to a pressure of approximately one atmosphere (absolute).

1. Exponentially rising Heat Input to a Thin Ribbon

Rosenthal and Miller (1) have presented extensive data for a rapidly increasing heat input to a vertical flat ribbon. The heat input was caused to rise exponentially with time. The rate of rise was expressed as the "exponential period," i.e., the time interval required for the heat input, q to multiply by the factor "e" (=2.73). Tests were made for various values of the following:

- (a) "Exponential period," from 5 to 70 milliseconds
- (b) bulk temperature of water, 90 to 160°F
- (c) materials: aluminum and platinum
- (d) dissolved oxygen = 0.2 to 5 parts per million

The measured surface temperature was found to behave as follows:

- (a) a rise, with increasing time, to a sharp peak, somewhat above saturation temperature;
- (b) a drop back to a lower, slowly increasing temperature rise (above saturation) representative of nucleate boiling;
- (c) a second sudden, very rapid rise, as film boiling began to occur, due to the further increase in q .

The "delay time" is defined as the time interval between the time when the surface passes the boiling point (212°F) and the time when the surface temperature reaches its first peak.

This "delay time" never exceeded 9 milliseconds over the whole range of the tests reported. It was in most cases no more than 5 milliseconds.

2. Uniform Heat Input Pulse of 30 Milliseconds to a Thin Ribbon

Cole (2) has observed the temperature of a thin horizontal nickel ribbon, in water, during power pulses of various intensities, ranging from 0.08×10^6 to 2.8×10^6 Btu/(hr ft²). The sub cooling ranged from 9 to 14.0°F below saturation.

For inputs in the range which produced only the nucleate type of boiling, the surface temperature reached its steady-state value in from 4 to 16 milliseconds.

For inputs large enough to produce film boiling, the surface temperature rose nearly linearly, except for, in some cases, a lower rate of rise while passing through the nucleate boiling temperature range.

Free convection of the liquid water was said to have negligible effect during the 30 millisecond heating interval considered.

3. A.C. (120 cycle) Heat Input to Water-Filled Thin Vertical Tube

Lottes (5) reports the temperature response of, and the time required to start "blowing" water up, a vertical 0.31" I.D. copper tube, for various watts input at 120 cycles per second (from 60 cycle current and voltage in phase with each other). The initial subcooling ranged from 12 to 130°F below saturation.

For inputs which produce nucleate boiling, the surface temperature shows a 120-cycle ripple, thereby showing that it is responsive to changes with a period of 1/120 or 8 milliseconds. Furthermore, the water rises more than 0.08 inch in the tube within a very few milliseconds of the time when the wall temperature starts to level off, due presumably to start of nucleate boiling. This boiling, in turn, occurs within 2 or 3 milliseconds of the time the wall temperature passes the saturation value (212°F).

For very high inputs which produce film boiling, (935°F after one pulse) the wall temperature rise is almost the same as for an empty tube. Hence, the vapor film formation must occur within a small fraction of the 8.3 millisecond duration of the input pulse.

4. Bubble Growth and Collapse for Boiling of Sub-Cooled Water

Ellion (2) observed bubble growth and collapse in sub-cooled water. Examples of his results are presented by Tribus (3). They show that, at 50% of "peak heat flux," and a subcooling of about 80°F, the bubble revealed its maximum radius about .02 inch, in about 0.5 milliseconds from the time it became visible (at a radius of about .005 inch). The time required to reach this initially observed radius was, of course, not known; extrapolation of the radius versus time curve to zero radius is merely an assumption, but the analysis of Forster and Zuber (8) indicates that there is no decrease of practical significance in the growth rate for radii below the observable range. Extrapolation to zero radius, is therefore reasonable and indicates a growth from 0 to .005 inch in about .05 millisecond.

The bubble collapsed completely about 0.5 milliseconds after reaching its maximum radius.

The total bubble lifetime is, therefore, about one millisecond.

Similar data for forced convection of sub-cooled water are presented by Gunther. (4)

The bubble life duration increases with decrease of subcooling. For example, McLean et al (9) report, for a .015" diameter wire, bubble life-times of 2 milliseconds for 16 to 30°F of subcooling, and of 10 milliseconds for 70°F subcooling. Saturated water (no subcooling) is considered below.

5. Bubble Growth for Boiling of Water at Saturated Temperature

Griffith (7) includes, in his report on bubble growth rates, only one Figure (Fig. 3) showing bubble radius as an explicit function of time (for a wall temperature of 218°F and atmospheric pressure).

A growth to a radius of .04 inch in 15 milliseconds is shown; no data for times beyond this were reported, but the figure shows good agreement between calculated and observed growth rate, so the calculation method could presumably be applied to evaluate growth for a longer period if desired.

Forster and Zuber (8) have also derived an equation for bubble growth which checks well with the test data they considered. This equation is reproduced in Fig. 8 of Tribus' summary (3). It has been established, furthermore, by Forster and Zuber (8) that the hydrodynamic effects (including surface tension) on bubble growth are confined to the first 0.1 millisecond of the bubble life, so that time for a bubble to reach observable size is not of serious concern in considering time lag of effect of changes in heating rate. (see first paragraph of item 4 above).

Symbols - Nomenclature

c_p = specific heat at constant pressure

D = diameter of a duct

g = acceleration of gravity

k = thermal conductivity of the fluid

L = characteristic length, total height of the plate in eq. (4), (5), and (7)

N = a constant

q'' = heat flux per unit of surface area

Δt = temperature rise of the surface above ambient, or above bulk fluid

t_w = temperature of surface (wall)

V = velocity

x = distance from leading edge, in eq. (2)

X = local distance vertically above lower edge of vertical plate

β = volumetric coefficient of expansion of the fluid

δ_L = thickness of laminar sub layer of a turbulent boundary layer

μ = viscosity

ν = kinematic viscosity, $= \mu/\rho$

ρ = density of the fluid

τ = time, after sudden change

$\left. \begin{matrix} \Delta t \\ \tau_o \end{matrix} \right\}$ = time to reach steady state condition, approximately

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