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TRANSIENT BOILING HEAT TRANSFER TO WATER*

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Abstract—Experimental transient boiling heat transfer in water for exponential heat inputs to thin metallic ribbons of $0.015 \times 10^6 e^{t/t_0}$ Btu/ft²h are compared with steady state predictions. The nominal values of controlled variables included velocity: $\leqslant 1$, 14 ft/s; pressure; 14·7, 500, 1000, 2000 psia; subcooling: $\leqslant 10$, 42, 112°F; exponential period: 5, 15, 50 ms.

NOMENCLATURE

a, thermal diffusivity (0·09290 m²/h) [ft²/h];
c, specific heat (1 kcal/kg°C) [Btu/lb°F];
H, ribbon thermal capacitance 0·5 ($\rho c\delta$), (4·882 kcal/m²°C) [Btu/ft²°F];
h, heat transfer coefficient (4·882 kcal/m²°C) [Btu/ft²/h°F];
h_{fg}, enthalpy of vaporization ($1{\cdot}8^{-1}$ kcal/kg) [Btu/lb];
k, thermal conductivity (1·488 kcal/mh°C) [Btu/fth°F];
L, ribbon length (0·3048 m) [ft];
p, pressure, psia (0·07031 kg/cm²);
Pr, Prandtl modulus;
q, heat flux (2·712 kcal/m²h, $31{\cdot}54 \times 10^{-6}$ W/cm²) [Btu/ft²h];
S, relative thermal capacitance, $H^2/(k\rho c)_f t_0$, [dimensionless];
T, temperature ($1{\cdot}8^{-1}$°C) [°F];
T_R, relative temperature rise, $q_0\,[t_0/(k\rho c)_f]^{\frac{1}{2}}$($1{\cdot}8^{-1}$°C) [°F];
t, time [h, ms];
t_0, exponential period [ms];
u, velocity (0·3048 m/s) [ft/s];
v, void per unit area [mm];
δ, ribbon thickness, mils (mils × 0·0254 mm) [ft];
μ, viscosity (0·4134 centipose) [lb/hft];
ν, kinematic viscosity (0·09290 m²/h) [ft²/h];
ρ, density, (16·02 kg/m³) [lb/ft³];
σ, surface tension ($14{\cdot}59 \times 10^3$ dynes/cm) [lb/ft].

Subscripts

d, designates variables at incipient boiling, i.e. time of first bubble;
f, liquid;
g, vapor;
m, mean;
o, initial, datum or period;
po, power off;
P, pool chamber;
ss, steady state;
V, visual chamber;
x, x-ray chamber.

* A "Tabular Summary of the Test Conditions" (Tables A-1–A-7) are on deposit with the National Auxiliary Publications Service of the American Society for Information Science, NAPS-00985.

INTRODUCTION

THE INTENT here is to present a summary of the heat transfer results from an extensive experimental investigation [1] for the transient response in water of metallic ribbons which, to simulate nuclear reactor excursions, were subject to exponential Joule heat inputs of $q_0\,e^{t/t_0}$. Details of the experimental systems and data reduction procedures are given in [1] and will not be repeated here; however, Table 3 presents a summary of the test dates and major system modifications.

All tests were made on 3 in. length ribbons, primarily with 4 mil Deltamax (50 per cent nickel, 50 per cent iron), although 1 mil Platinum was often used in atmospheric pool boiling chamber tests. These metals having been selected as suitable for resistance thermometry.

Mounted to span the center of 0·270 × 0·81 in. rectangular visual and x-ray flow channel chambers, results for ribbon widths of $\frac{1}{10}$, $\frac{1}{8}$ and $\frac{1}{4}$ in. show no measurable differences.

The heat flux input q and the ribbon temperature rise ΔT were obtained from oscilloscope displays of voltage and current while the surface to fluid or net transient heat flux was calculated from the energy balance relation:

$$q_{\text{net}} = q_0\, e^{t/t_0} - H\frac{\partial T}{\partial t},$$

solved graphically for times greater than one half period.

Void production was obtained from high-speed motion picture film exposed through the glass walls of pool and visual (flow) chambers and alternatively by x-ray absorption through a Beryllium tube wall x-ray chamber. The fields of view were about 1 in. so the void for only one third of the ribbon could be obtained. For this reason the void measurement location was initially varied between the first in. (upstream) and the third in. (downstream) positions along the length of the three in. ribbon.

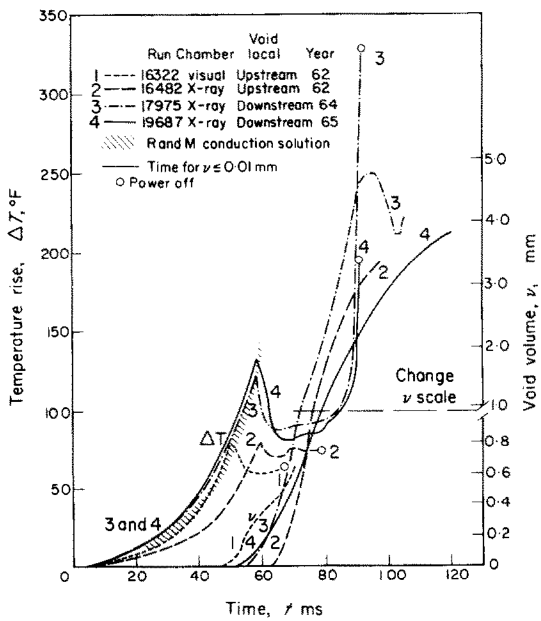

FIG. 1. Measured transient temperature rise and void production for 4 runs on 4 mil Deltamax (500 psia, $u \leqslant 1$ ft/s, $\Delta T_{\text{sub}} = 42$°F, $t_0 =$ ms).

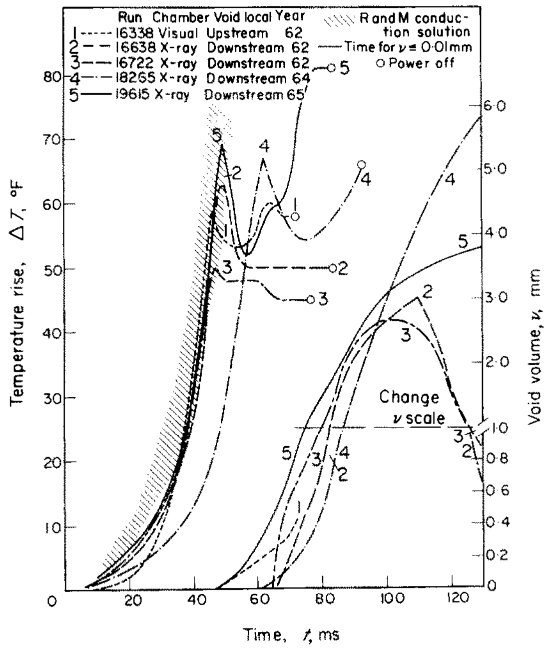

FIG. 2. Measured transient temperature rise and void production for 5 runs on 4 mil Deltamax (1000 psia, $u \leqslant 1$ ft/s, $\Delta T_{\text{sub}} = 42$°F, $t_0 = 15$ ms).

Figures 1 and 2 illustrate typical measured results. The nominal test conditions of low velocity ($u \leqslant 1$ ft/s), 42°F subcooling, 15 ms period and the two system pressures of 500 and 1000 psi selected for these figures represent the maximum number of replicate tests. The run number establishes the test sequence wherein the last two digits reveal the number of the particular test on a ribbon number given by the preceding digit, i.e. Run 16322 was the twenty-second test made on ribbon number 163.

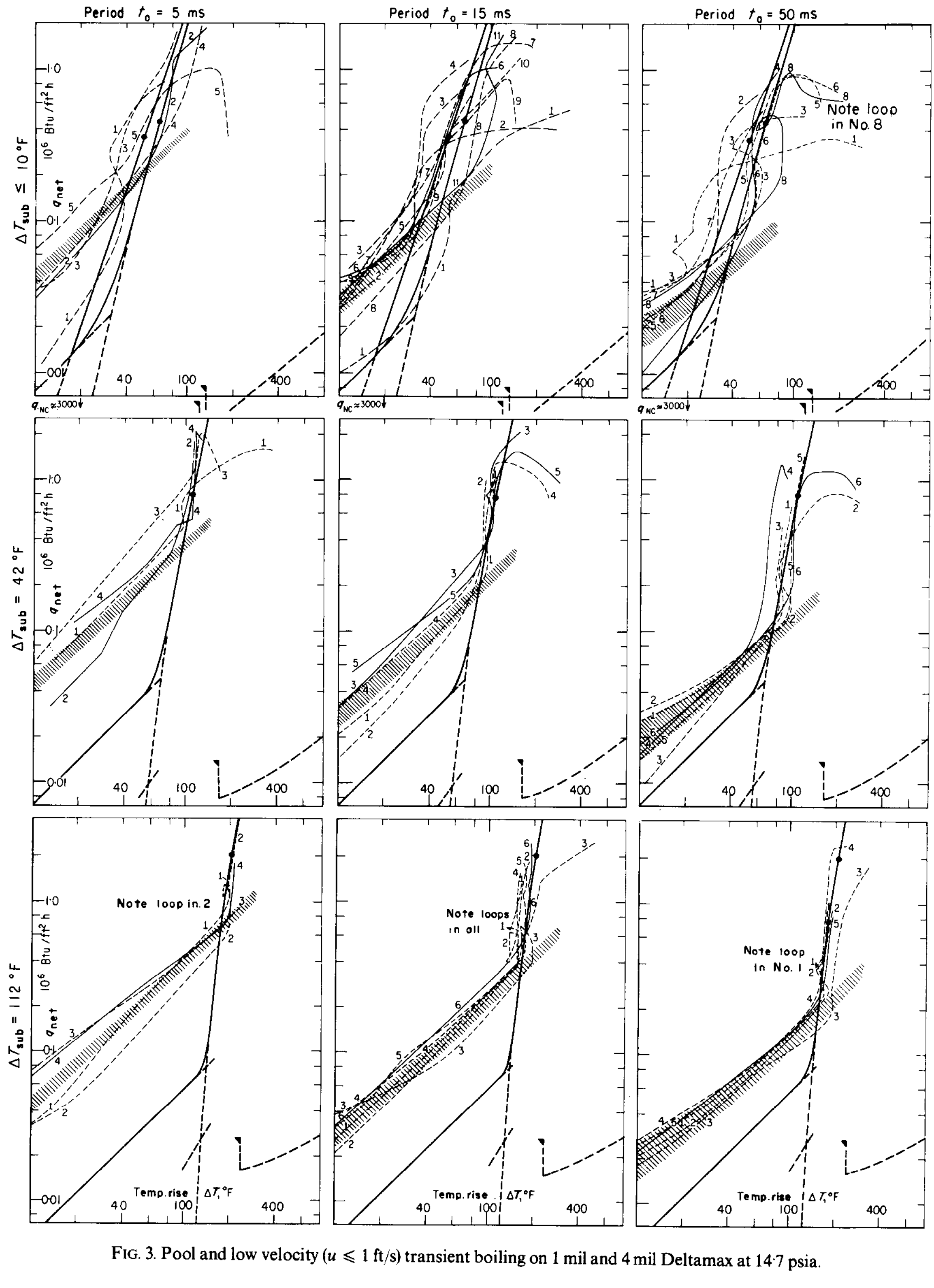

FIG. 3. Pool and low velocity ($u \leqslant 1$ ft/s) transient boiling on 1 mil and 4 mil Deltamax at 14·7 psia.

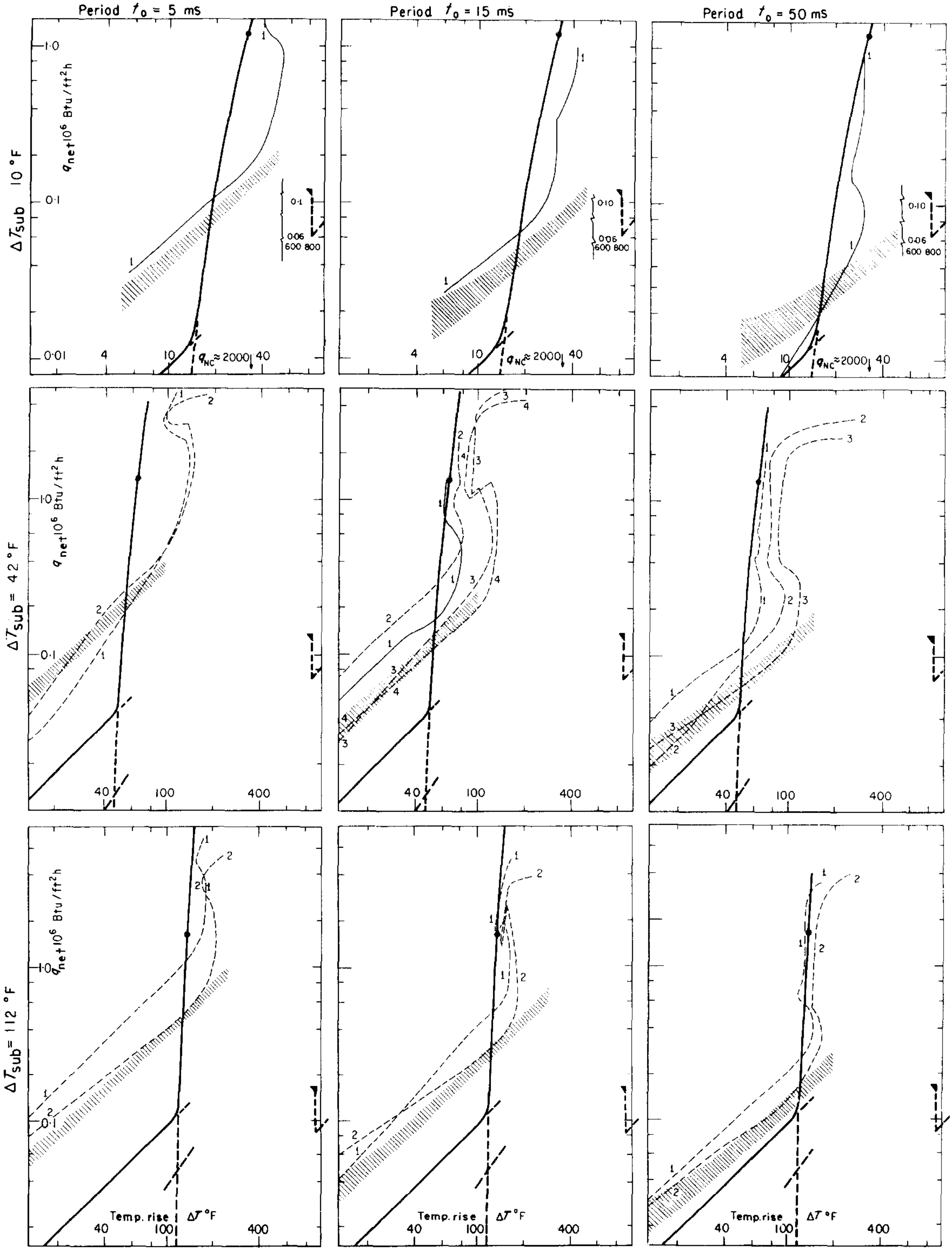

FIG. 4. Pool and low velocity ($u \leqslant 1$ ft/s) transient boiling on 4 mil Deltamax at 500 psia.

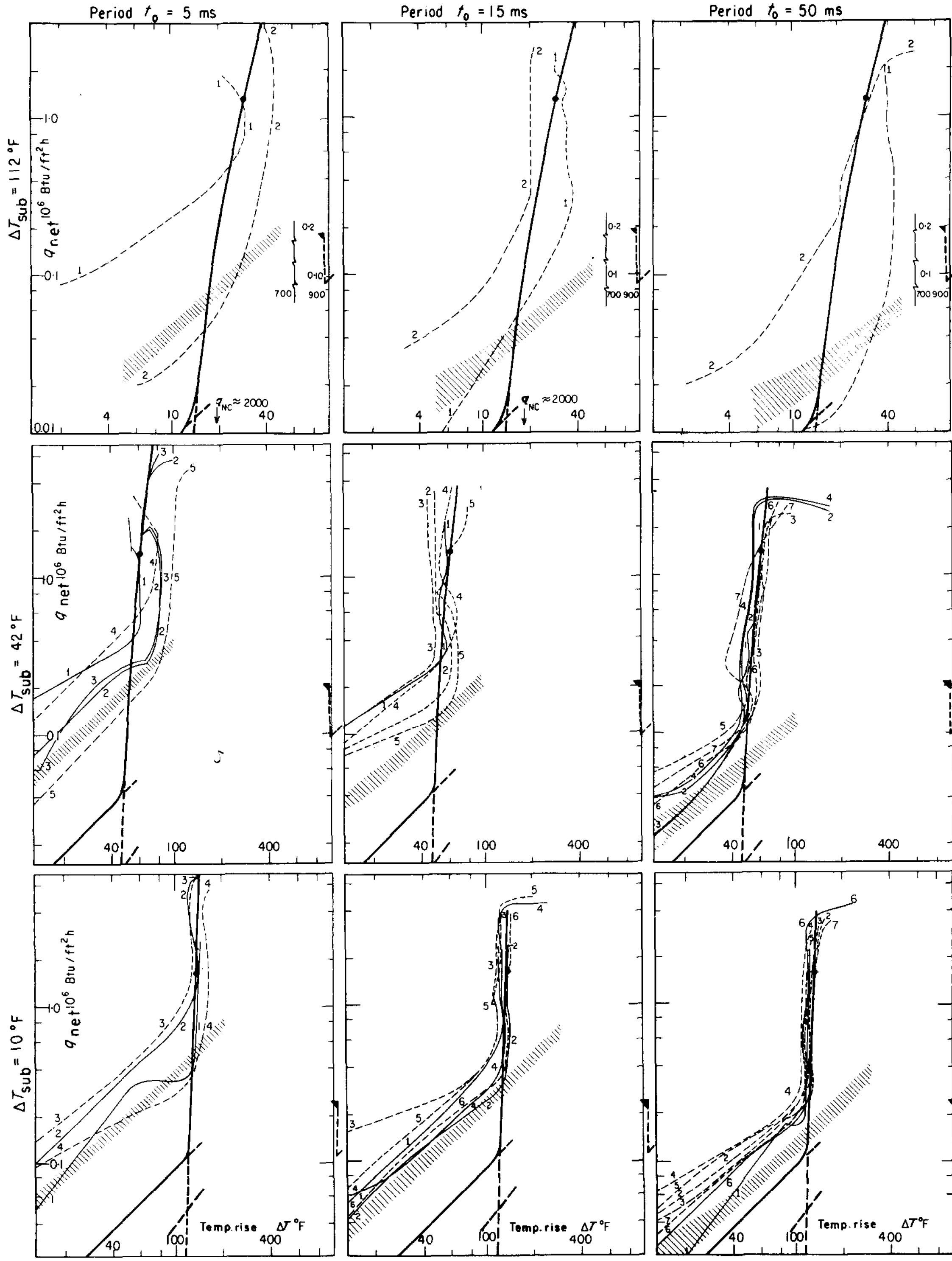

FIG. 5. Pool and low velocity ($u \leqslant 1$ ft/s) transient boiling on 4 mil Deltamax at 1000 psia.

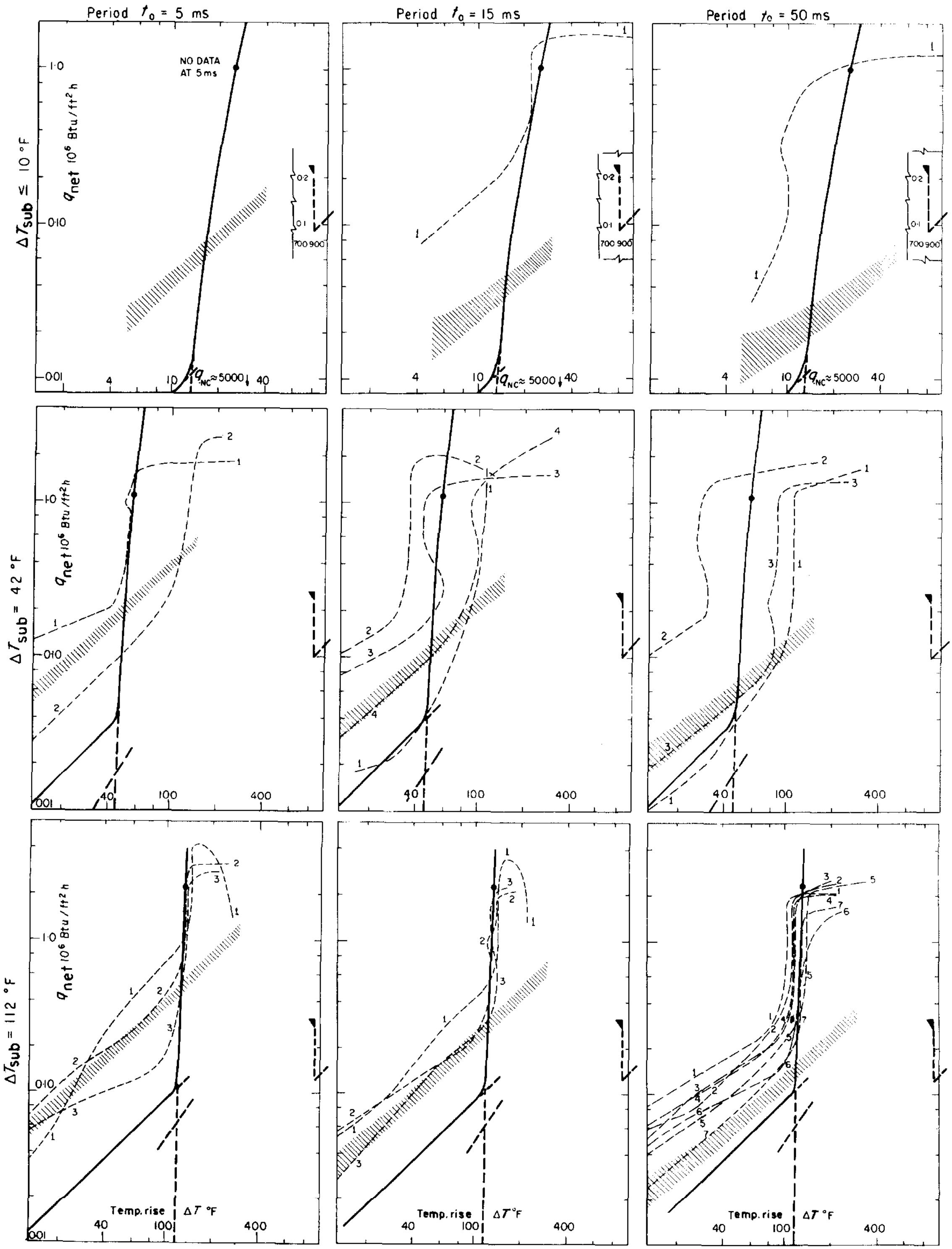

FIG. 6. Pool and low velocity ($u \leqslant 1$ ft/s) transient boiling on 4 mil Deltamax at 2000 psia.

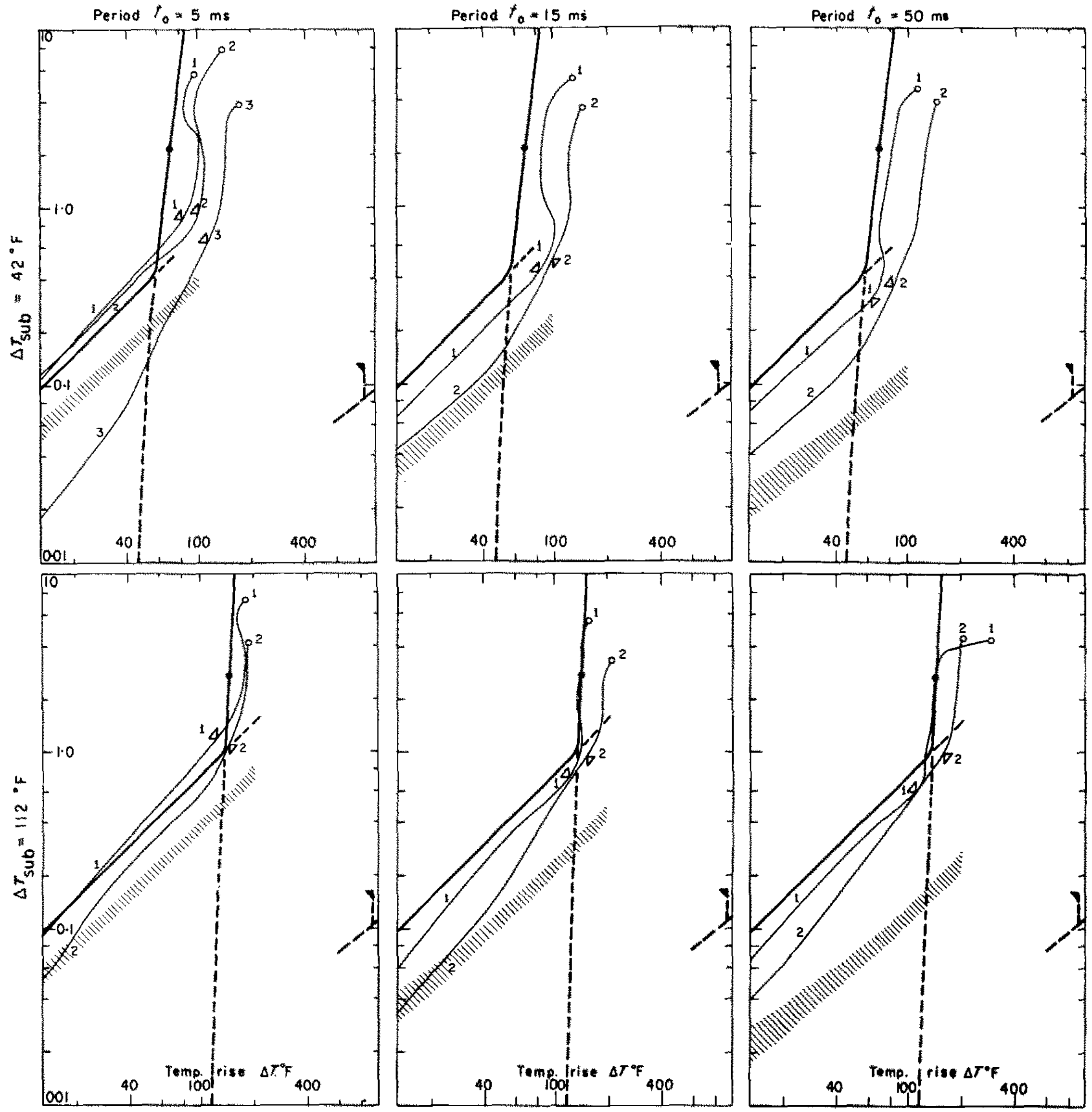

FIG. 7. High velocity transient boiling on 4 mil Deltamax, downstream at 14 ft/s and 500 psia.

It is believed that Figs. 1 and 2 warrant reasonable confidence in the data since the agreement revealed includes data taken over a period of five years during which time several major modifications were made in the test systems, the instrumentation and data processing techniques (see Table 3).

The shaded area shown on these figures is the Rosenthal and Miller ([2], see Discussion on Figs. 3–9) analytical conduction solution for the nonboiling transient temperature rise in which variations between the several runs of $14{\cdot}1 < t_0$ ms $< 16{\cdot}0$ and $0{\cdot}011 < q_0$ 10^6 Btu/ft^{2}h $< 0{\cdot}018$ is accounted for. Studies of individual runs comparing the Rosenthal and Miller solution revealed excellent agreement for both the pool and the flow chambers for velocities from 0 to 1 ft/s. In this regard the 1962 Run 16482 in Fig. 1 and the 1964 Run 18265 in Fig. 2 are suspect, but whatever the cause, the improvements made between 1962 and 1964 contradicts the possibility of instrumentation being at fault; further, error analysis reveals temperature measurement accuracies of 5–10°F which account in part for these deviations.

Once nucleate boiling begins the cooling rate increases and the rate of temperature rise is markedly reduced in spite of the exponential heat input. Of course, if film boiling starts the temperature rise will again be rapid and may proceed to burn out. In both Figs. 1 and 2, however, there are definitive peaks in ΔT vs. t curves which are accentuated on the temperature scale in Fig. 2. This so-called overshoot phenomenon was dramatically demonstrated in motion picture films at low (atmospheric) pressure wherein the early void is clearly seen to condense in place. For instance, in one case for a 16 ms period run: 2·3 ms after the time a first bubble was observed, a void of 0·09 mm had been produced; 0·8 ms later the void was 0·004 mm which was followed in 23 ms at power off by a void of 0·11 mm.

Except for low pressures and small void production (less than 1 mm) the heat transfer effects of the overshoot phenomenon is not believed to be severe as evidenced in part by the early monotonic increase for all the void curves in Figs. 1 and 2. For this reason transient incipient boiling is interpreted throughout as beginning at the time of first bubble t_b which is obtained from the films for the visual chamber and from the limiting initial response ($v = 0{\cdot}01$ mm) of adsorption for the x-ray chamber.

While this paper does not treat the void production it should be noted that the void increase after power off is substantial. Also that for curves 3 in Figs. 1 and 5 in Fig. 2 where the late temperature rise to power off is large there is an indication that the void production rate is noticeably decreased suggesting the possibility of film boiling. Using the time at which ΔT exhibits a rapid rate of rise as a conservative estimate of a possible start of film boiling (90 ms in Fig. 1, 70 and 80 ms in Fig. 2) an independent study reveals that with few exceptions (Fig. 2 is one) film boiling did not occur at void values of less than 1 mm which appears as a sufficient criterion to assure that all observed overshoot phenomena were confined to the nucleate boiling régime. This tentative conclusion must, however, be qualified since Hall and Harrison [15] report severe fluctuations in net heat transfer for periods of less than 5 ms which appear to be accompanied by film boiling.

METHOD OF PRESENTATION OF RESULTS

To affect an inclusive summary to bracket the heat transfer results in [1], all data from 1959 through 1965 for the velocity extremes of $u \leqslant 1$ ft/s and $u = 14$ ft/s are presented excluding the intermediate subcooling of 82°F and system pressures of 40 and 125 psia. That is, for each of these two velocities there is included the following pattern of nominal values for the test parameters, all of which are for a nominal q_0 of 0·015 million Btu/ft^{2}h.

Pressure, p, psia		14·7, 500, 1000, 2000
Period, t_0, ms		5, 15 and 50
Subcooling, ΔT_{sub}, °F	$\leqslant$	10, 42 and 112.

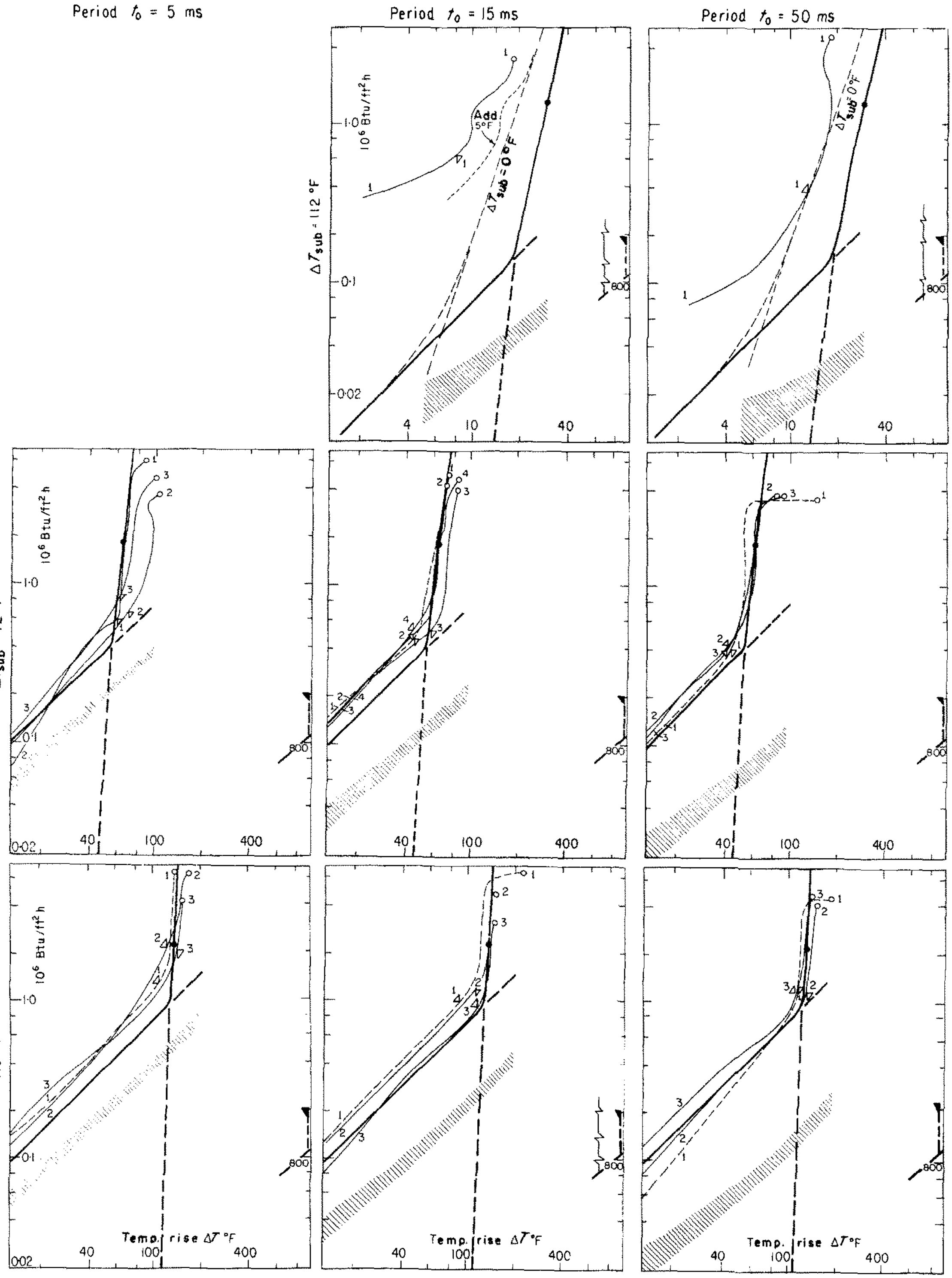

FIG. 8. High velocity transient boiling on 4 mil Deltamax, downstream at 14 ft/s and 1000 psia.

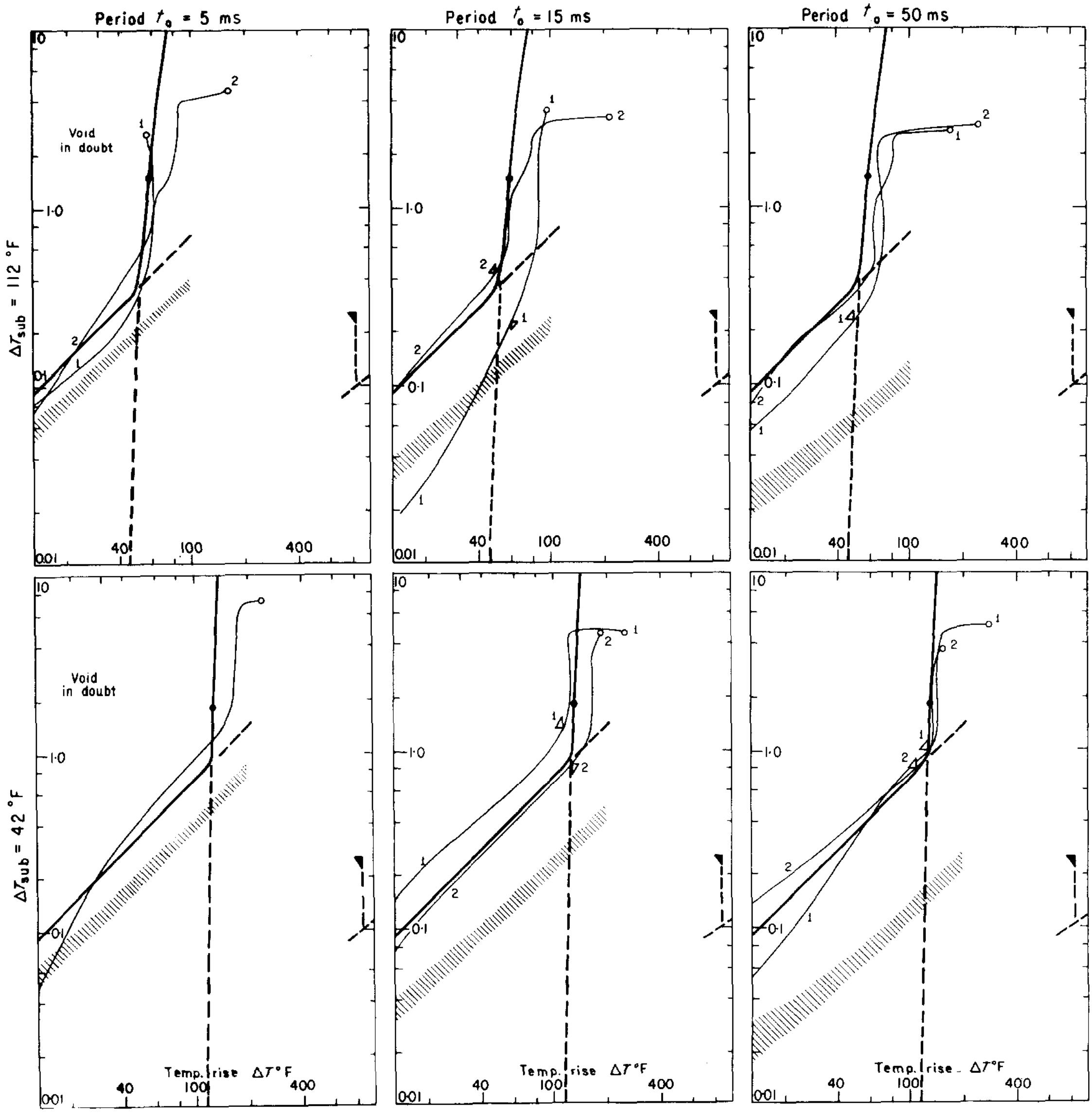

FIG. 9. High velocity transient boiling on 4 mil Deltamax, downstream at 14 ft/s and 2000 psia.

Data at 14·7 psia could not be obtained at high velocity because the minimum test section pressure required for circulation at 14 ft/s was about 30 psia.

Plotting the transient net heat flux q_{net} vs. the ribbon surface excess temperature,

$$\Delta T = (T_w - T_0) = \Delta T_{sat} + \Delta T_{sub},$$

eliminates time and facilitates direct comparison with steady state heat transfer. This is the presentation method here in Figs. 3–6, for pool boiling (14·7 psia) and low velocity ($u \leqslant 1$ ft/s), and in Figs. 7–9 for the high velocity ($u \approx 14$ ft/s) results.

DISCUSSION OF FIGS. 3–9

Throughout the curves of net heat flux vs. the excess surface temperature were determined as follows.

Non-boiling region

(a) Steady state natural convection is based on the Eckert [3] turbulent boundary layer correlation equation:

$$\frac{hL}{k} = 0{\cdot}024\left[\left(\frac{\beta g}{\nu^2}\right)L^3\,\Delta T\frac{(Pr)^{1\cdot17}}{1 + 0{\cdot}494\,(Pr)^{\frac{2}{3}}}\right]^{0\cdot4}$$

in which the characteristic length was taken at 3 in. consistent with the ribbon orientation and length for the visual and x-ray chambers.

(b) Steady-state forced convection is based on the well known turbulent flow flat plate correlation

$$\frac{hL}{k} = 0{\cdot}037\left(\frac{u_\infty L}{\nu}\right)^{0\cdot8}(Pr)^{\frac{1}{3}}$$

where u_∞ was taken as $u/0{\cdot}8$. Use of this correlation was established by steady state tests ([1], USAEC Report SAN 1003) wherein Reynolds to the 0·8 power was shown to apply for measured velocities from 1 to 14 ft/s. Agreement for the visual chamber was satisfactory while the results for the x-ray chamber were 15 per cent high.

(c) Analytic transient heat transfer (shaded area) is based on the Rosenthal and Miller conduction solution [2]:

$$\frac{\Delta T}{T_R} = \frac{e^{t/t_0}}{1 - S}\left\{\mathrm{erf}\sqrt{\left(\frac{t}{t_0}\right)} - (\sqrt{S}) + (\sqrt{S})\exp\left[\left(\frac{1}{S} - 1\right)\frac{t}{t_0}\right]\mathrm{erfc}\sqrt{\left(\frac{1}{S}\frac{t}{t_0}\right)}\right\}$$

$$q_{net} = \frac{q_0 e^{t/t_0}}{1 - S}\left\{1 - (\sqrt{S})\,\mathrm{erf}\sqrt{\left(\frac{t}{t_0}\right)} - \exp\left[\left(\frac{1}{S} - 1\right)\frac{t}{t_0}\right]\mathrm{erfc}\sqrt{\left(\frac{1}{S}\frac{t}{t_0}\right)}\right\}.$$

(d) At high velocity (Figs. 7–9) the transient convection is approximated by a quasi-steady state model in which a constant heat transfer coefficient [from (b) above] is applied to the lumped capacity energy equation as suggested in [4] for conditions which apply here. It should be noted that for this model, while ΔT is a function of time given by the energy balance equation

$$\Delta T = \frac{q_0 t_0}{H + h_m t_0}\left[\exp(t/t_0) - \exp(-h_m t/H)\right],$$

the net heat flux vs. ΔT, i.e. $q_{net} = h_m \Delta T$, is independent of time. For this reason the calculated incipient boiling values $(\Delta T_b)_{ss}$ and $(q_b)_{ss}$ are independent of the period. The R and M solution (shaded areas) are repeated in Figs. 7–9 for comparison.

In the nonboiling region Figs. 3–6 for pool and low velocity reveal generally fair agreement with the Rosenthal and Miller transient conduction solution as required. There are, however, a substantial number of tests for which the deviations apparently exceed the error analysis limits. The fault can be shown to be due to excessive computational errors at early times. That is, at early time when $q_0\,e^{t/t_0}$ is of the same order as the negative ribbon heat capacitance term $H(\partial T/\partial t)$ the calculated net heat flux is thus sensitive to evaluation of the temperature time derivative which is determined by a graphical technique using faired plots of the T vs. t data. For example, at 1000 psia, 42°F sub-cooling and 15 ms period, comparison of Figs. 2 and 5 shows that while temperature vs. time data for curves 1, 2, 3 and 5 are in good agreement with Rosenthal and Miller, all five curves for q_{net} vs. ΔT are uniformly high compared with Rosenthal and Miller (see Fig. 5).

Study of all data presented in these figures shows no evidence favorable to any one of the three systems or the year of test. Further it is observed in Table 1 that at the so-called time of first bubble t_b, i.e. at incipient boiling, agreement in ΔT_b between measured and R and M values is predominantly satisfactory which supports the argument above as to the cause of deviations at early times. Consequently, for pool and low velocity nonboiling heat transfer the validity of the Rosenthal and Miller conduction solution appears to be well established by

Table 1. Surface superheat temperatures $(\Delta T_{sat})_b$ *for incipient boiling*

p (psia)	14·7			500			1000			2000		
				Fabic transient model [1] SAN 1008								
ΔT_{sub} °F 10		10–20			10–23			10–20			At 2000 psia, extrapola-	
42		27–50			28–47			24–31			tion of this model indi-	
112		36–60			37–54			27–30			cates all values near zero	
				Hsu ss model [5]								
ΔT_{sub} °F 10		28			11			10			10	
42		95			49			46			44	
112		180			120			118			115	
				Bergles and Rohsenow ss model [6]								
ΔT_{sub} °F 10		15			12			12			11	
42		50			47			45			44	
112		123			119			117			115	
t_0 ms	5	15	50	5	15	50	5	15	50	5	15	50
				Predicted by quassi-ss nucleate boiling criteria								
ΔT_{sub} °F 10	46	37	29	8·5	6·5	4·5	6·7	4·7	3·2	4·8	2·8	2
42	54	44	37	13	11	8	11	9	7	9	7	5
112	48	41	36	17	14	11	15	12	9	14	11	6
				Mean of measured values. A single value signifies measured and R and M values agree within ±5°F. Otherwise the upper value is that measured and the lower that for R and M								
⩽ 10°F	34	27	29	22	18	14	30	16	28	—	6	−2
								17			54	28
42°F	46	35	37	61	41	39	37	11	7	10	32	40
								31	30	31	37	25
112°F	64	46	34	43	29	52	26	6	4	18	5	5
								41	44	44	27	63
				High velocity ($u = 14$ ft/s) for incipient boiling on 4 mil Deltamax								
				Quasi-ss, constant h_m, Bergles and Rohsenow [6]								
10					—			2–16			—	
42					16			10			6	
112					21			17			10	
				Mean of measured values								
10								8	8			
42				58	54	32	22	6	2	—	13	7
112				22	25	23	15	4	13	—	8	8

Pool and low velocity ($u \leqslant 1$ ft/s) boiling on 1 mil Pt and 4 mil Deltamax ribbons these results. For high velocity nonboiling there is considerable support for the constant heat transfer coefficient prediction, especially at 1000 psia. Noting that throughout Figs. 7–9 the highest numbered curves are the most recent (1965) data the very low values at 500 psia cannot be explained since this condition does not persist at the two higher pressures. It should be emphasized here that while reasonable predictions of q_{net} vs. ΔT appear for constant h_m the deviations in predictions of ΔT vs. t using this model are poor (see [4]).

[6] which are observed to be in substantial mutual agreement. The fourth method of prediction is based on equating the Rosenthal and Miller q_{net} and quasi-steady state nucleate boiling heat transfer. It is noted, also, that only the fourth technique is a function of the exponential period.

Review of the measured results in Table 1 reveals the stochastic response that is also commonly observed for steady state incipient boiling. There does appear to be some tendency for this incipient transient superheat $(\Delta T_{sat})_b$ to decrease with both increasing period and pressure as implied by the steady state criteria. At high velocity the lower values for higher subcooling is contrary to expectations, in any event, these results must be viewed with much caution.

Table 2. Transient crises heat flux, q_{max} — net 10^6 Btu/ft^{2}h

	Pool and low velocity				High velocity (u = 14 ft/s)			
	Exponential period t_0(ms)			Steady	Exponential period t_0(ms)			Steady
ΔT_{sub} °F	5	15	50	state	5	15	50	state
				p = 14·7 psia				
⩽ 10	1·6	1·2	0·76	0·40	—	—	—	—
42	2·2	1·6	1·1	0·78	—	—	—	—
112	2·2	2·1	2·2	1·50	—	—	—	—
				p = 500 psia				
⩽ 10	—	—	—	1·21	—	—	—	—
42	4·8	5·1	2·9	1·34	5·7	4·5	4·7	2·14
112	6·4	4·4	3·3	1·63	5·7	4·4	4·4	2·70
				p = 1000 psia				
⩽ 10	—	—	2·5	1·31	—	—	—	1·34
42	5·8	—	2·8	1·41	4·7	4·7	3·8	1·84
112	6·5	4·7	3·9	1·65	6·3	6·6	4·7	2·28
				p = 2000 psia				
⩽ 10	—	1·6	1·2	1·02	—	—	—	—
42	2·2	2·8	1·5	1·10	3·5	3·3	2·7	1·48
112	3·4	2·6	2·1	1·29	7·1	4·5	5·0	1·83

Incipient boiling

The surface superheat for incipient boiling $(\Delta T_{sat})_b$ is presented in Table 1 which includes four prediction techniques for low velocity and one for high velocity. The first is from Fabic's transient model, the details for which are treated in [1] (SAN 1008), that also includes an extensive survey of other proposed models. Second and third are from the steady state models of Hsu [5] and Bergles and Rohsenow

Nucleate boiling region

Throughout in Figs. 3–9 there is shown Rohsenow's [7] steady state correlation for nucleate boiling and that of Bergles and Rohsenow for forced convection boiling [6], taken in the following forms.

$$q_{NB} = (0{\cdot}013)^3 \frac{\mu_f c_f^3}{h_{fg}^2 Pr_f^n}\left(\frac{\rho_f - \rho_g}{\sigma}\right)^2 (\Delta T_{sat})^3$$

where $n = 5{\cdot}1$ for $T_{sat} < 600°F$ and $3{\cdot}0$ for $T_{sat} > 600°F$ and

$$q_{FNB} = q_{FC}\sqrt{\left\{1 + \left[\frac{q_{NB}}{q_{FC}}\left(1 - \frac{q_{1B}}{q_{NB}}\right)\right]^2\right\}}$$

where

$q_{FC} = h_m \Delta T$ with h_m evaluated as described above,

$q_{1B} = q_{NB}$ at $(\Delta T_{sat})_i$ determined by the incipient boiling equation:

$$15{\cdot}6(\text{psia})^{1{\cdot}156}(\Delta T_{sat})_i^{2{\cdot}30/(\text{psia})^{0{\cdot}0234}} = h_m[(\Delta T_{sat})_i + \Delta T_{sub}].$$

nucleate boiling curve is the pool boiling critical heat flux according to the Zuber *et al.* [10] correlations:

$$q_{max\,sat} = 0{\cdot}131\, h_{fg}\rho_g\left[\frac{\sigma(\rho_f - \rho_g)}{\rho_g{}^2}\right]^{\frac{1}{4}}$$

and

$$q_{max\,sub} = q_{max\,sat}[1 + \phi_{Z.\,T.\,W}(T_{sat})\Delta T_{sub}]$$

where

$$\phi_{Z.\,T.\,W} = 5{\cdot}3\frac{\sqrt{[(k\rho c)_f]}}{h_{fg}\rho_g}\left[\frac{g}{\sigma^3}\rho_g^2(\rho_f - \rho_g)\right]^{\frac{1}{8}}.$$

For high velocity the steady state crises values are based on the Jens and Lottes [9] pipe flow correlation:

$$q_{JL} = C(G/10^6)^m(\Delta T_{sub})^{0{\cdot}22},\ 10^6\ \text{Btu/ft}^2\text{h}$$

Table 3. Schedule of test dates and system modifications

Ribbon numbers	Year of test	Chamber	Modifications
USAEC Report [1] SAN 1001, Jan. 1961			
32, 128	1959	pool	Initial pool boiling system
USAEC Report SAN 1002, Nov. 1961			
130–144	1961	visual	Ribbon temperature calibration duplicated *in situ*
146		x-ray	
USAEC Report SAN 1007, Mar. 1963			
153	1962	pool	Added provisions to increase power off temperatures and current limit to 630 A. Programmed correction for non-linearity in resistivity coefficients
147–49, 51, 55, 56, 58, 59, 63		visual	
150, 52, 54, 57, 64, 67		x-ray	
USAEC Report SAN 1913, May 1966			
181	1964	visual	System rebuilt and relocated. Paralleled second oscilloscope for early time data
179, 182, 183		x-ray	
194–198	1965	x-ray	Eliminated second oscilloscope programed corrections for nonlinearities in the scope and improved resistivity coefficients. Measured transient pressures

This procedure was followed throughout and thus is presumed to supercede Rohsenow's original forced convection boiling proposal [8].

Also shown on each low velocity steady state

where

psia	500	1000	2000
C	0·817	0·626	0·445
m	0·16	0·28	0·50

for: u from 5 to 30 fps, p from 500 to 2000 psia and ΔT_{sub} from 3 to 160°F.

To complete orientation with steady state heat transfer there is included the incipient natural convection film boiling state (q_{min}, ΔT_{min}) as estimated by Berenson [11], the film boiling analyses for natural convection by Sparrow and Cess [12] and for forced convection by Cess and Sparrow [13]. These estimates included an arbitrary subcooling correction for q_{min}, ΔT_{min} based on [10] and for film boiling a radiation heat transfer correction following Bromley's technique as suggested by Sparrow [14].

The results in Figs. 3–9 demonstrate substantial agreement with the nucleate boiling predictions of [6] and [7]; however, Hall and Harrison [15] have demonstrated that for exponential periods of less than 5 ms and particularly for periods of less than 1 ms, nucleation is soon followed by: film boiling, severe fluctuations in temperature and net heat flux, and probably severe limitations in void production.

As observed by others [15, 16] the present data also exhibit maximum nucleate boiling heat fluxes which exceed the steady state crisis values by factors as large as 4. Tachibana *et al.* [16] present ramp input ($q = 0.37 \times 10^6\ t/t_0$ Btu/ft^{2}h) data for saturated transient pool boiling at atmospheric pressure which clearly show that q_{net} vs. ΔT plots can be used for this purpose and in fact their plot of such crisis values vs. ramp period reveals a progressive increase from the steady state value at a ramp period of 100 ms to six times that value at a ramp period of 1 ms. Unfortunately, there are too few runs for which the flattening of the q_{net} vs. ΔT curves is sufficiently evident to establish firm values for net q_{max} and thus for transient incipient film boiling. A study of such crisis values as were obtained revealed a tendency for increases with decreased period, increased subcooling and increased pressure (see Table 2).

CONCLUSION

The results demonstrate that transient nucleate boiling heat rates for exponential heat inputs, excluding periods of less than 5 ms, are reasonably well represented by steady state nucleate boiling correlation techniques including incipience at a transient non-boiling heat flux equal to that for steady state nucleate boiling and crisis values that exceed steady state predictions.

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REFERENCES

1. H. A. Johnson, V. E. Schrock *et al.*, Reactor heat transients project, University of California, Berkeley, USAEC, TID 4500 Reports: SAN 1001 (Jan. 1961) to SAN 1013 (May 1966).
2. M. W. Rosenthal and R. L. Miller, An experimental study of transient boiling, ORNL-2294, Oak Ridge National Laboratory, Oak Ridge, Tennessee (1957); also H. A. Johnson and P. L. Chambré, Transient heat transfer for steady slug flow over a flat plate with uniform exponential heat generation, *L. M. K. Boelter Anniversary Volume*. McGraw-Hill, New York (1964).
3. E. R. G. Eckert and R. M. Drake, *Heat and Mass Transfer*, p. 324. McGraw-Hill, New York (1959).
4. M. Soliman and H. A. Johnson, Transient heat transfer for forced convection flow over a flat plate of appreciable thermal capacity and containing an exponential time-dependent heat source, *Int. J. Heat Mass Transfer* **2**, 27 (1968).
5. Y. Y. Hsu, On the size range of active nucleation cavities on a heating surface, *J. Heat Transfer* **84**, 207 (1962).
6. A. E. Bergles and W. M. Rohsenow, The determination of forced convection surface-boiling heat transfer, *J. Heat Transfer* **86**, 365 (1964).
7. W. M. Rohsenow, A method of correlating heat transfer data for surface boiling liquids, *Trans. ASME* **74**, 969 (1952). Also, W. M. Rohsenow, Lecture series on boiling and two-phase flow for heat transfer engineers, Univ. of Calif. Extension, Berkeley (May 1965).
8. W. M. Rohsenow, Heat Transfer, A Symposium 1952, Eng. Res. Inst., Univ. of Mich., Ann Arbor (1952).
9. W. H. Jens, Boiling heat transfer; What is known about it? *Mech. Engng* **76**, 981 (1954).

10. N. Zuber, M. Tribus and J. W. Westwater, The hydrodynamic crisis in pool boiling of saturated and subcooled liquids, International Heat Transfer Conference. Boulder. Colorado. ASME, Part II (1961).
11. P. J. Berenson, Film boiling heat transfer from a horizontal surface, *J. Heat Transfer* **3** 351 (1961).
12. E. M. Sparrow and R. D. Cess, The effect of subcooled liquid on laminar film boiling. *J. Heat Transfer* **84**. 149 (1962).
13. R. D. Cess and E. M. Sparrow, Film boiling in a and concentration profiles through the boundary layer. For forced-convection boundary layer flow, *J. Heat Transfer* **83**. 370 (1961).
14. E. M. Sparrow, The effect of radiation on film boiling heat transfer, *Int. J. Heat Mass Transfer* **7**, 220 (1964).
15. W. B. Hall and W. C. Harrison, Transient boiling of water at atmospheric pressure, International Heat Transfer Conference. Inst. Mech. Eng.. Chicago. Illinois, (1966).
16. F. Tachibana, M. Akiyama and H. Kawamura, Heat transfer and critical heat flux in transient boiling, *J. Nucl. Sci. Technol.* **5**, 3 (March 1968).

TRANSFERT THERMIQUE DANS L'EAU EN ÉBULLITION TRANSITOIRE

Résumé—Les résultats expérimentaux de transfert thermique dans l'eau en ébullition transitoire pour des rubans métalliques minces alimentés thermiquement suivant la loi exponentielle $4{,}7 . 10^4\ e^{t/t_0}$ W/m² sont comparés aux prédictions de régime stationnaire. Les valeurs nominales des variables contrôlées correspondent à des vitesses $\leqslant 0{,}3$ ou 4,2 m/S, des pressions $1{,}01 . 10^5$; $3{,}44 . 10^6$; $6{,}89 . 10^6$; $1{,}38 . 10^7$ N/m², des sous-refroidissements 5,56; 23,35; 62,2°C et des périodes exponentielles 5; 15; 50 ms.

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Zusammenfassung—Die experimentellen Ergebnisse des instationären Wärmeüberganges beim Sieden an dünnen Metallbändern in Wasser mit einer exponentiellen Wärmezufuhr von $4{,}731 . e^{t/t_0}$ W/cm² werden mit den stationären Voraussagen verglichen.

Die Werte der untersuchten Variablen betragen für Geschwindigkeit: $\leqslant$ 0,3, 4,3 m/sek; Druck: 1, 34,5, 69, 138 bar; Unterkühlung: $\leqslant -$ 12, 5.5, 45°C; exponentielle Periode: 5, 15, 50 Millisekunden.

НЕСТАЦИОНАРНЫЙ ПЕРЕНОС ТЕПЛА ПРИ КИПЕНИИ ВОДЫ

Аннотация—Экспериментальные данные по нестационарному теплопереносу при кипении воды в случае экспоненциального подвода тепла к тонким металлическим лентам, равного $0.015 \times 10^6\ e^{t/t_0}$ *Btu/ft² h*, сравниваются с расчетами для стационарного теплообмена. Номинальные значения контролируемых переменных включали скорость $\leqslant$ 1,14 фут/сек, давление 14,7; 500; 1000; 2000 psia, недогрев 10,42, 112°*F*; экспоненциальный период 5, 15, 50 миллисекунд.