Study of Nitrogen and Neon Pool Boiling on a Short Vertical Pipe

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An experimental study in the behavior of nitrogen and neon pool boiling was performed. The test section consisted of a vertical pipe 3 in. O.D. × 7½ in. long. The variables studied are: heat input, submergence, and surface finish. Several copper heaters were tested, in addition to nickel and cadmium plated heaters. The results were correlated for nitrogen by an equation which gave an average deviation of ± 16% for all the heaters except the cadmium plated heater. The paper describes the experimental equipment and technique, summarizes the data, and compares them to literature values and equations.

The operation of cryogenic magnets requires continuous removal of heat, usually by boiling refrigerant, at temperatures below the superconductivity critical point. At 30°K. the only fluid refrigerants are helium, hydrogen, and neon. Hydrogen and neon are more suitable from the heat transfer point of view, but safety considerations preclude the use of liquid hydrogen until more operating experience with cryomagnets is obtained. Other refrigerants are available for operation at higher cryogenic temperatures. Some cryogenically cooled magnets have been operated at 78°K. by using liquid nitrogen as a coolant.

Comprehensive surveys of heat transfer to boiling cryogenic liquids have been presented by several authors (*1, 2, 3, 4*). A recent paper published by Bewilogua and Mahn (*5*) presents experimental data for boiling neon. Furthermore, general descriptions of the various types of boiling, which include cryogenic boiling, are given in text books such as McAdams "Heat Transmission" and in many technical papers (*6, 7, 8, 9*).

In general, information on boiling heat transfer is scarce in the case of cryogenic fluids. Since the mechanical design of magnets includes narrow annuli and confined spaces between conductors, it is necessary to develop the required design information concerning the cooling characteristics under such conditions.

This research study was undertaken to investigate the various factors affecting the boiling performance of nitrogen and neon. It is entirely limited to the nucleate boiling region.

EXPERIMENTAL EQUIPMENT

The apparatus consisted of a double glass dewar holding the liquid being boiled, with boiling taking place on the walls of a cylinder immersed, at least partly, in the liquid. An auxiliary refrigeration system was used to recondense neon vapors evolved at the boiler.

Test Section

Heating of the boiling surface was accomplished through use of a boiler-condenser scheme. The boiler condenser consisted of a small electric heating element which boiled liquid nitrogen or liquid neon inside of a pressure vessel having a nominal 3 in. O.D. The outer surface of the pressure vessel formed the heating face. A shield inside the pressure vessel around the heater formed a vapor-lift pump that pumped liquid nitrogen or liquid neon so that the inner wall of the pressure vessel was continuously washed by the percolating fluid and was therefore maintained at a constant pressure. Under test conditions, the pressure-vessel heater was submerged in either liquid nitrogen or liquid neon. Resistance heating by means of the partially immersed electric element resulted in simultaneous vapor generation and condensation inside the 3-in. vessel and boiling on its outer wall. At a given electric power input, the heater pressure would rise until the temperature driving force resulting from the difference between the saturation temperature of the boiling-condensing liquid inside the heater at the heater pressure, and the saturation temperature of the boiling liquid outside the heater was sufficient to dissipate the heat input. The thermosyphon pumping assured the existence of a uniform temperature potential through the vessel wall, thus resulting in the uniform heat generation simulation desired.

In mechanical design of the condenser-boiler heating element attempts were made to meet the American Society of Mechanical Engineers Boiler and Pressure Vessel Code (*10*), to minimize the resistance to heat transfer of

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the vessel walls, and also to limit the effects of pressure and temperature changes on the heater diameter. For these reasons the heater wall was made of OFHC (oxygen-free high-conductivity) copper pipe 3/8 in. thick. This material, of certified American Society of Testing Materials Specification 13-75 (*11*), has a thermal conductivity at liquid neon temperatures of 695 B.t.u./hr.-ft.-°F. (*12*). The estimated maximum temperature difference occurring across this wall at the higher heat fluxes was 0.3°F.

In order to minimize thermal end effects, end closures were specified as 1/4 to 5/16 in. nominal thickness stainless steel plates. The plates were soldered into appropriately machined recesses at the respective ends of the OFHC copper pipe length. One plate had a centrally located acess hole, while the other closure was solid. Each end plate was fitted with a short length of stainless steel rod attached to it, thus making the pressure vessel much like a rolling pin in appearance. One of the rods was bored out to provide access to the pressure vessel cavity through the end plate.

The heating element with a nominal heating capacity of 550 w. at 120 v. A.C. when operated in liquid nitrogen or liquid neon had a 1/2 in. O.D. and a 2 1/8 in. length.

The element was mounted inside a 5/8 in. O.D. thin wall stainless steel tube. The heater element-tube assembly was inserted into the pressure vessel cavity through the access tube and constituted the thermosyphon pump.

Figure 1 shows schematically the assembled test section. The complete flow sheet for the experimental test section is shown on Figure 2.

Instrumentation

The observations and measurements deemed necessary to study the pool boiling behavior of the liquids investigated were as follows.

1. Saturation pressure of fluid in heater cavity (pressure vessel cavity). Saturation pressure, along with the vapor pressure data for nitrogen and neon, was used as one measure of the temperature of the internal surface of the boiler-condenser heater wall. Saturation pressure in the pressure vessel cavity was measured on a bank of manometers and a precision pressure gauge. The various manometers and pressure gauge were specified to allow accurate pressure readout from 0.02 to 600 lb./sq. in gauge. The pressure measurement bank tapped into the gas fill line of the heater and allowed continuous monitoring of the cavity saturation pressure. An auxiliary piping circuit employing the precision high pressure gauge was specified to permit metering the gas introduced into the vessel cavity.

2. Temperature of heater surface (pressure vessel wall) and of liquid bath into which test section is immersed. Space limitations and practicality of instrumentation dictated the use of copper-constantan thermocouples for temperature measurement. It was also decided that the measurement of the boiling liquid temperature was of secondary importance, since boiling occurred at atmospheric pressure. The primary measurement was therefore a direct measurement of the temperature difference between the wall and boiling liquid. Copper-constantan thermocouples mounted in the wall of the pressure vessel and referenced to a copper-constantan junction submerged in the liquid nitrogen or neon bath yielded the desired electromotive force output corresponding to the temperature difference between the pressure vessel wall and the liquid bath. The dewar containing the liquid being boiled was 12 in. in diameter and contained a pool of liquid approximately 24 in. deep. The reference thermocouple was located about 6 in. below the liquid surface. In preliminary tests temperature traverses were made

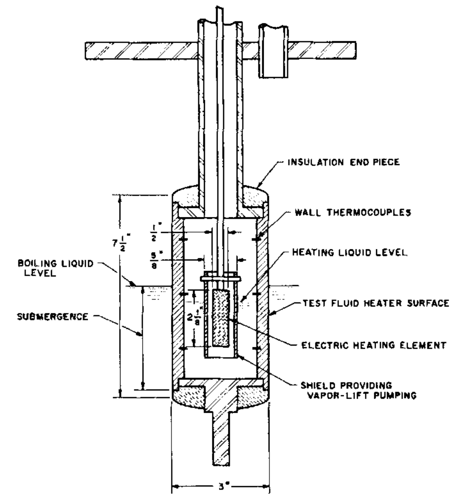

Fig. 1. Sketch of assembled test section.

throughout the pool of liquid in the dewar. It was found that this liquid was essentially isothermal. The small signal obtained from the thermocouples was amplified to provide a sensitivity for temperature measurement of ± 0.1°F. Thus the temperature measurement instrumentation consisted of a copper-constantan differential thermocouple circuit, a preamplifier unit, a multipoint recorder, 0 to 1 mv. full scale, and a precision potentiometer for suppression voltage.

3. Rate of heat input to the heater which is also heat transferred to the boiling liquid. The power input to the electric heating element was measured by means of a multiple range precision watt meter. The assumption was made that all power indicated by the watt meter was transferred through the walls of the pressure vessel. This implies that a negligible amount of heat was lost through the vessel end plates and in I^2R heating of the heater element lead-in wires.

4. Rate of gas generation or gas flow rate out of the boiling liquid. Vapor generation rate, or vapor flow rate, was measured in two independent ways. First, the generated vapor was directed to a rotameter through an access tube for direct measurement. The rotameter was located between the dewar and the boil off recovery gas bag. Secondly, the watt meter employed in measurement of power input also served as a monitor of the rate of vapor generation. That is, on the assumption that all heat input

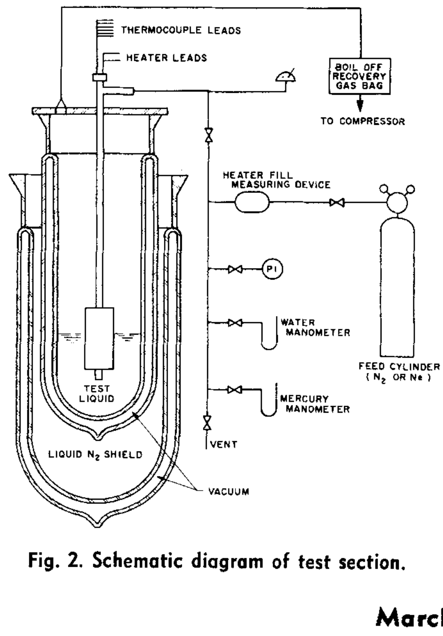

Fig. 2. Schematic diagram of test section.

TABLE 1. CHARACTERISTIC DIMENSIONS OF TEST SECTIONS

Heater no.	Heater O.D., D_1, in.	Heat transfer area, sq. ft./in.	Surface finish, root mean square, μ-in.
12b	2.9975	0.06540	9-11
6	2.9695	0.06478	16-20
9	2.9675	0.06474	6-8 (Ni)
10	2.9645	0.06468	8-11 (Cd)
7b	2.9030	0.06333	11-13
8	2.8495	0.06217	13-15

to the test section was consumed in vaporization, the vapor generation rate would be directly proportional to the power consumption, the constant of proportionality being the latent heat of the liquid.

Trial operations of the test section yielded a very good agreement (better than ± 3%) between vapor flow rate measurements by the two independent methods. It was then decided that the watt meter measurement of vapor flow would suffice since the flow meter method introduced a slight but undesirable back pressure on the boiling fluid.

5. Heated surface wetted by the liquid-gas mixture. The glass external parts of the test apparatus allowed direct visual observation of the wetted area of heated surface. A graduated scale proximate to the visible area of the heated surface assisted in this measurement. The boiling liquid surface was always plainly visible. Wetted surface included that surface covered by the swell resulting from boiling. It did not include the end-plate areas. Preliminary estimates of the error resulting from end-plate heat transfer showed this to be less than 2% of the total heat transfer.

6. Surface roughness and finish. The surface finish of each heater was carefully checked, and its roughness was measured with a profilometer (*13*).

Experimental Procedure and Data Collection

The test section was assembled and introduced into the glass dewars. The heater cavity was evacuated, and the whole section was then cooled down with liquid nitrogen. The shield section of the dewars assembly was also filled with liquid nitrogen. The 500 cc. heater cavity was filled with about 250 cc. of liquid nitrogen. This was accomplished by introducing enough gaseous nitrogen in the heater and condensing it. After the heater was filled with the desired amount of liquid nitrogen, the system was assumed to have reached equilibrium when the heater pressure was less than 3 to 4 in. of water for over 15 min., with zero heat input to the test section.

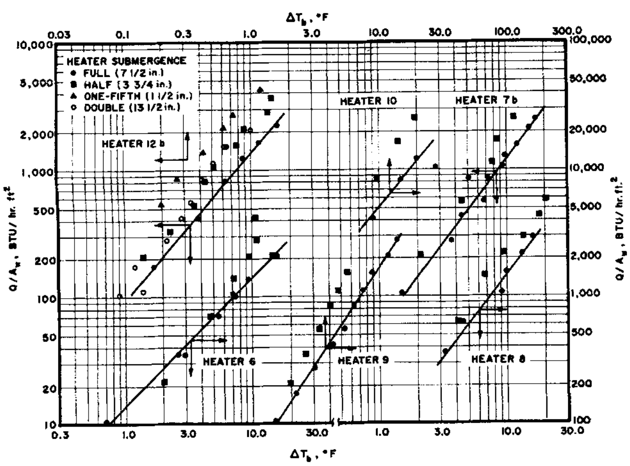

Fig. 3. Heat flux for liquid nitrogen pool boiling.

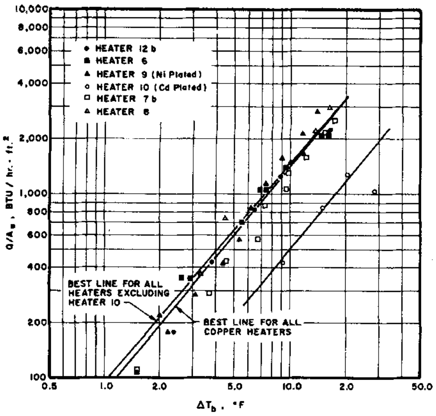

Fig. 4. Heat flux in nitrogen pool boiling, comparison between all heaters at 7½ in. submergence.

A complete series of runs for pool boiling consisted of varying the electrical power input to the heating element from 0 to about 400 w. at at least two different submergences, submergence being defined as the depth of the lower edge of the heating surface below the surface of the boiling liquid, as shown in Figure 1, and measured with a visible graduated scale. A 7½ in. submergence corresponds to 100% submergence since the various heaters tested had a 7½ in. effective heating length. For each run readings were taken of heater pressure, wattage, heater wall temperature measurements, and submergence. Six heaters were tested under pool boiling conditions: four copper heaters, Nos. 12b, 6, 7, and 8; and heaters Nos. 9 and 10 which were nickel and camium plated respectively. Dimensions and surface finish of the heaters are summarized in Table 1.

The experimental procedure used for the neon boiling study was the same as the one used for nitrogen boiling, but the heater cavity was filled with neon. Unfortunately, in the case of neon, only the overall temperature driving force was obtained because of damage to the wall thermocouples.

The physical properties of the two cryogenic fluids used for calculation purposes were obtained from standard sources (*14, 15, 16, 17, 18*). The experimental data are summarized in Figures 3, 4, and 5.

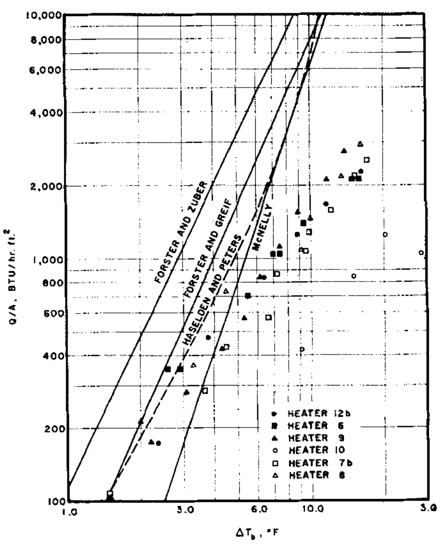

Fig. 5. Nitrogen heat flux data (total submergence) compared with literature equations and data.

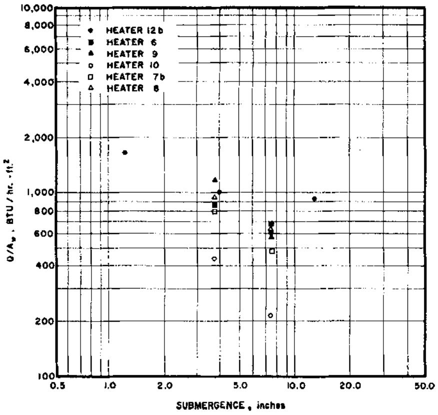

Fig. 6. Heat flux vs. submergence at ΔT_b = 5°F. nitrogen pool boiling.

As can be seen from Figure 4, a generalized equation relating heat flux to boiling temperature driving force for the case of full submergence can be obtained for all nitrogen runs except for the cadmium plated heater, this generalized equation being

$$Q/A_w = 87.2\ (\Delta T_b)^{1.20}$$

while the equation for the cadmium heater is

$$Q/A_w = 30.0\ (\Delta T_b)^{1.20}$$

Examination of Table 1 shows a variation of 9 to 20 μ in. root mean square for the surface finish of the four copper heaters, with the nickel plated and cadmium plated heater having a surface finish of 6 to 8 and 8 to 11 μ in. root mean square respectively. However, the test results, Figure 4, indicate that surface finish had no effect on the boiling heat flux. The smoothest heater (nickel plated heater, 6 to 8 μ in. root mean square) had essentially the same heat flux characteristic as the coarsest copper one (16 to 20 μ in. root mean square). The only noticeable difference was observed with the cadmium plated heater having a surface finish of 8 to 11 μ in. root mean square, intermediate between the nickel plated heater and the copper ones. There was no apparent explanation for this behavior.

Figure 5 compares the heat flux data obtained here for boiling nitrogen with the six heaters at 7½ in. submergence with three empirical equations for nucleate boiling and with one other set of nitrogen boiling data. The equations of Forster and Greif (*19*) and of Forster and Zuber (*20*) predict heat flux values higher than the experimental data at all but the lowest ΔT's. The Forster and Greif equation predicts a Q/A varying with the 2.1 power of ΔT_b in compar son to the 1.2 exponent found here. The McNelly correlation (*21*), which crosses the path of the experimental data at a ΔT_b of about 4°F., predicts a variation of Q/A dependent on the 2.9 power of ΔT_b. In making these calculations ΔT was taken as $(T_o - T_s)$ and also equal to ΔT_b. The data of Haselden and Peters (*22*) is shown here as a line representing the average of their data on boiling nitrogen outside copper cylinders of ½ × ⅜ in. horizontal, ⅜ × 3 in. vertical, and ⅜ × 3 in. horizontal. Their data agrees with that obtained here at $\Delta T_b < 3$°F. but at higher ΔT_b's predicts higher heat fluxes than those obtained here.

The effect of submergence on heat flux is illustrated in Figure 6, which is a plot of heat flux vs. submergence at 5.0°F. boiling temperature driving force. The following equation can be derived from the data:

$$\frac{(Q/A_w)_h}{(Q/A_w)_{7\,1/2}} = 4.0\ h^{-0.7} \text{ for } h < 7\tfrac{1}{2} \text{ in.}$$

and

$$\frac{(Q/A_w)_h}{(Q/A_w)_{7\,1/2}} = 0.2\ h^{0.7} \text{ for } h > 7\tfrac{1}{2} \text{ in.}$$

Variation in submergence probably causes several opposing effects on boiling. The results, Figure 6, indicate that heat flux decreases with increasing submergence, reaches a minimum at about 100% submergence, and then increases with increasing submergence. Some of the factors that may explain this behavior are surface blanketing with vapor, turbulence, number of bubbling sites, and bubble size. Estimates of the effect of heat transfer from the exposed heating surface to the vapor phase show that this transfer probably contributes less than 5% to the flux. It seems that the blanketing effect is the controlling one for the case of partial and full submergence. The bubbles being formed in the lower part of the heater blanket the upper surface with vapor and reduce its effectiveness. However when submergence increases beyond the 100% point, the other effects are offsetting the detrimental effect of blanketing, and heat flux increases with increasing submergence. It is believed that further study of this problem is necessary to develop a proper understanding of the several effects involved and to establish their magnitude.

Results from neon pool boiling experiments are shown in Figure 7. For neon boiling, an equation based on overall temperature driving force (ΔT_t = temperature difference between vapor inside heater and surrounding boiling liquid) was obtained, namely $Q/A_w = 135\ (\Delta T_t)^{1.11}$. The equation for nitrogen also based on overall temperature driving force is $Q/A_w = 66.0\ (\Delta T_t)^{1.11}$. Therefore, the heat flux for neon is about twice the heat flux for nitrogen at the same temperature driving force.

It is also interesting to compare the equations of Forster and Greif, Forster and Zuber, and McNelly (Table 2) with the neon and nitrogen data. The Forster and Zuber equation predicts a neon heat flux 70% as great as that for nitrogen at ΔT = 5°F. The Forster and Greif equation predicts a neon heat flux six times that for nitrogen at this same ΔT, whereas the McNelly equation gives a factor of 10. Experimentally, the authors found the heat flux for neon boiling to be twice that for nitrogen boiling at ΔT_t = 5°F.

It is interesting to note that the results of Bewilogua and Mahn (5) can be represented by the equation $Q/A_w = 225\ \Delta T_b^{1.40}$ (Figure 7).

TABLE 2. SOME EMPIRICAL EQUATIONS FOR PREDICTING NUCLEATE BOILING HEAT FLUXES

a. Forster and Zuber:

$$\frac{Q'}{A} = 1.5 \times 10^{-3} \frac{(T_o - T_s) k_L \lambda \rho_V \Delta p^{0.75}}{\Delta T\, C_L\, \rho_L^{1.25} \sqrt{2\pi a_L \sigma}} \left\{ \frac{\rho_L}{\mu_L} \left[\frac{(T_o - T_s) C_L \rho_L \sqrt{\pi a_L}}{\rho_L \rho_V} \right]^2 \right\}^{0.62} N_{Pr}^{0.35}$$

b. Forster and Greif:

$$\frac{Q'}{A} = 1.2 \times 10^{-3} \left(\frac{A_L\, C_L\, \rho_L\, T_s}{j\, \lambda\, \rho_V \sqrt{\sigma}} \right) \left(\frac{C_L\, T_s \sqrt{a_L}}{j\, \lambda^2\, \rho_V^2} \right)^{1/4} \left(\frac{\rho_L}{\mu_L} \right)^{5/8} N_{Pr}^{1/3}\, \Delta p^2$$

c. McNelly:

$$\frac{Q'}{A} = 0.0082 \left(\frac{C_L}{\lambda} \right)^{2.22} \frac{k_L \rho_L}{\sigma} \left(\frac{\rho_L}{\rho_V} - 1 \right)^{1.06} \Delta T^{3.22}$$

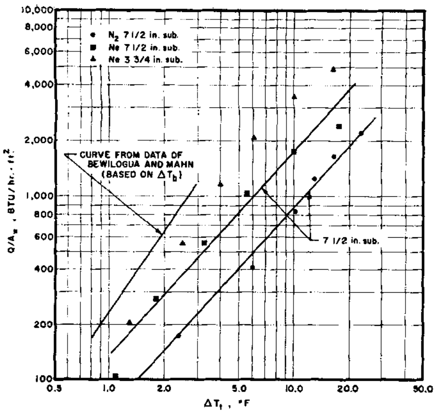

Fig. 7. Comparison between neon and nitrogen pool boiling heat flux vs. overall temperature driving force, heater 12b.

This indicates that the heat flux obtained with neon is about two and one half times larger than the one for nitrogen. This is in close agreement with the present results which show that on the basis of an overall temperature driving force, the neon heat flux is also about twice the one for nitrogen. It is also noted that Bewilogua's results show that the heat flux is proportional to the temperature driving force to the 1.4 power which is reasonably close to the 1.20 power obtained from this study.

Exponents for the temperature driving force calculated from the data presented by Richards et al. (Figures 2 and 3, reference *3*) for liquid helium and nitrogen are 0.98 and 0.94 respectively. All of these exponents are substantially lower than the value of three to four usually indicated in the literature (*6*, *7*).

CONCLUSION

An apparatus was developed permitting a study of nitrogen and neon boiling on a short, vertical, 3-in. diameter pipe. A generalized equation for the nitrogen boiling heat flux was derived based on boiling temperature driving force, and two equations based on overall temperature driving force were obtained, one for nitrogen and one for neon. Heater surface roughness, though varied, was not found to have a significant effect on the boiling coefficients. However, heater submergence appeared to have an appreciable effect on boiling heat flux.

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NOTATION

A_L = wetted effective heat transfer area, sq. cm.
A_w = wetted effective heat transfer area, sq. ft.
a_L = thermal diffusivity of liquid, sq. cm./sec.
C_L = specific heat of liquid, joules/g. °K.
D_1 = outside diameter of heater chamber, in.
h = submergence of heater surface, in.
j = mechanical equivalent of heat, erg./joule
k_L = thermal conductivity of liquid, w./cm. °K.
Δp = pressure difference corresponding to the temperature difference ΔT, dynes/sq. cm.
N_{Pr} = Prandtl number of liquid, $C_L \mu_L / k_L$
Q_1 = heat transfer rate, B.t.u./hr. sq.ft.
Q = heat transfer rate, w.
T_o = temperature of liquid, °R. (except in Forster and Zuber equation)
T_s = saturation temperature of boiling liquid, °R. (except in Forster and Zuber equation)
ΔT = temperature difference, °R., $\Delta T = T_o - T_s$ (except in Forster and Zuber equation)
ΔT_b = boiling heat transfer driving force, temperature difference between boiling liquid and heater wall, °R.
ΔT_t = overall boiling heat transfer driving force, temperature difference between saturation temperature of fluid inside heater and boiling liquid temperature, °R.
λ = latent heat of vaporization, joules/g.
μ_L = absolute viscosity of liquid, poise
ρ_L, ρ_V = density of liquid and vapor, g./cc.
σ = surface tension, dynes/cm.

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