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A STUDY OF BOILING HEAT TRANSFER FROM A ROTATING HORIZONTAL CYLINDER

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Abstract—The influence of rotation on nucleate boiling heat transfer from a horizontal heated circular cylinder rotating about its own axis in Refrigerant-113 and in distilled water has been investigated. Both copper and brass cylinders, 1·125 in. (2·86 cm) dia. and 8 in. (20·32 cm) long, were tested at heat flux densities between 8000 and 26000 Btu/hft^2 (25200 and 82000 W/m^2) for a range of rotating Reynolds numbers from zero to $2{\cdot}6 \times 10^5$. It has been found that rotation has no significant effect on the heat-transfer coefficient for low to moderate speeds. In this region, for

$$0 \leqslant \left(Re = \frac{D^2\omega}{2\nu}\right) < 0{\cdot}35\left[Re_c = \left(\frac{Nu_{sb}}{0{\cdot}12(Pr)^{\frac{1}{3}}}\right)^{\frac{3}{2}}\right]$$

where Nu_{sb} is the Nusselt number based on the cylinder diameter for stationary boiling, existing correlations of boiling heat transfer for a stationary surface for a specific liquid–solid interface could be used to predict the heat transfer rate. For very high rotary speeds boiling ceased and forced convection became the only mode of heat transfer. In this region, for $Re > 1{\cdot}35\ Re_c$, the experimental data are correlated well by

$$Nu = 0{\cdot}12\left(\frac{D^2\omega}{2\nu}\right)^{\frac{2}{3}}(Pr)^{\frac{1}{3}}.$$

Between these limiting cases, a smooth and gradual transition region was observed in which the size and the frequency of the bubbles decrease with increasing speed until the bubbles completely disappear. The heat transfer results for both stationary surface boiling and forced convection were found to be in general agreement with those of other investigators. In addition, the values of "C_{sf}" and "r" in the Rohsenow equation for boiling from a stationary cylinder for the brass–R-113 combination were determined.

NOMENCLATURE

A, effective heater surface area [ft^2 (m^2)];
C, specific heat [Btu/lb$_m$°F (J/kg °C)];
C_{sf}, coefficient of equation (10);
D, diameter of cylinder [ft (m)];
D_b, bubble diameter [ft (m)];
Fr, Froude number, $\omega^2 D/2g$;
g, acceleration due to gravity [ft/h^2 (m/h^2)];
g_c, conversion factor $4{\cdot}17 \times 10^8$ [lb$_m$ft/h^2 lb$_f$];
Gr, Grashof number, $gD^3\rho^2\beta\Delta T/\mu^2$;
h, heat-transfer coefficient [Btu/hft^2 °F (W/m^2 °C)];
h_{fg}, latent heat [Btu/lb$_m$ (J/kg)];
k, thermal conductivity [Btu/hft °F (W/m °C)];
Nu, Nusselt number, hD/k;
Nu_{sb}, Nusselt number based on the cylinder diameter for stationary boiling;
Pr, Prandtl number $C_L\mu_L/k_L$;
Q, heat flux [Btu/h(W)];
r, exponent of equation (10);
Re, rotating Reynolds number, $\omega D^2/2\nu$;
Re_c, critical rotating Reynolds number

$$\left(\frac{Nu_{sb}}{0{\cdot}12\,(Pr)^{\frac{1}{3}}}\right)^{\frac{3}{2}};$$

s, exponent of equation (10);
T_s, saturation temperature [°F (°C)];

T_w, wall or surface temperature [°F (°C)];
ΔT_s, $= T_w - T_s$[°F(°C)];
We, Weber number, $\rho_L U^2 D_b/g_c$;
β, coefficient of thermal expansion [1/°F (1/°C)];
μ, dynamic viscosity [lb_m/fth (kg/mh)];
ν, kinematic viscosity [ft^2/h (m^2/h)];
ρ, density [lb_m/ft^3 (kg/m^3)];
σ, surface tension [lb_f/ft (J/m^2)];
ω, angular velocity [1/h].

Subscripts
L, liquid;
v, vapour.

INTRODUCTION

CONSIDERABLE research on boiling heat transfer from stationary heating surfaces has been carried out and the results made known to the public, but there appears to have been only one paper published recently on boiling heat transfer from a horizontal cylinder rotating about its own axis. The present paper, based on the work of Tang [1], is to investigate the effect of rotation of solid horizontal circular cylindrical heating surfaces made of copper and of brass about their axes in water and in Refrigerant-113 with nucleate boiling taking place at its surface.

SUMMARY OF PREVIOUS WORK

The present section briefly describes some of the previous investigations which are of importance to the subject in boiling and non-boiling convective heat transfer from a horizontal rotating surface.

Non-boiling

Studies of non-boiling convective heat transfer from a horizontal rotating cylinder have received considerable attention during the past decade. Among those were the significant quantitative studies of Anderson and Saunders [2], Etemad [3], Dropkin and Carmi [4], and Kays and Bjorklund [5] for air, Seban and Johnson [6] for oil and water, and that of Becker [7] for water.

Based upon their findings, the effect of rotation on non-boiling convective heat transfer from a rotating cylinder may be generalized into the following three regions:

(a) *Low Reynolds number region.* The Nusselt numbers depend almost entirely on the Grashof numbers. Therefore, existing free convection equations may be used to predict the heat transfer rate in this region.

(b) *Intermediate Reynolds number region.* Both natural convection and rotation influence the Nusselt numbers.

$$Nu = 0{\cdot}11[(0{\cdot}5\,Re^2 + Gr)\,Pr]^{0{\cdot}35} \tag{1}$$

and

$$Nu = 0{\cdot}095\,(0{\cdot}5\,Re^2 + Gr)^{0{\cdot}35}\ \text{(for air only)} \tag{2}$$

are the equations suggested by Etemad and by Dropkin and Carmi respectively to predict the Nusselt numbers in this region.

(c) *High Reynolds number region.* The Grashof number has a negligible effect and the rate of heat transfer is almost solely dominated by the rotating Reynolds numbers. The following are the equations proposed by different investigators to predict the heat transfer coefficients in this region:

Anderson and Saunders (for air only)

$$Nu = 0{\cdot}10\,(Re)^{\frac{2}{3}} \tag{3}$$

Etemad (for air only)

$$Nu = 0{\cdot}076\,(Re)^{0{\cdot}7} \tag{4}$$

Dropkin and Carmi (for air only)

$$Nu = 0{\cdot}073\,(Re)^{0{\cdot}7} \tag{5}$$

Becker

$$Nu = 0{\cdot}133\,(Re)^{\frac{2}{3}}\,(Pr)^{\frac{1}{3}}. \tag{6}$$

Equation (3) was later rearranged into the following form by Becker by introducing the Prandtl number in order to be applied to the general case of any liquid:

$$Nu = 0{\cdot}111\,(Re)^{\frac{2}{3}}\,(Pr)^{\frac{1}{3}} \tag{7}$$

Boiling

The only data available on boiling heat transfer from a rotating cylinder was published recently by Nicol and McLean [8, 9]. They investigated boiling heat transfer from a horizontal copper cylinder, 0·922 in. (2·34 cm) dia. and $10\frac{1}{2}$ in. (26·67 cm) long, rotating in still water at atmospheric pressure in a tank having a horizontal cross section 20 by 10 in. (50·8 by 25·4 cm) at heat flux densities between 8000 and 25000 Btu/hft^2 (25200 and 78900 W/m^2) with rotary speeds from zero up to 830 rpm. Each run was performed at a constant rotary speed while the power input to the test section was varied at increments of approximately 200 W. The surface finish of the cylinder was not specified. They found that an increase in the rate of heat transfer per unit temperature difference occurred for rotational speeds up to a critical value of 150 rpm. For rotational speeds greater than this value, the rate of heat transfer decreased.

Nicol and McLean correlated their results in terms of significant parameters and nondimensional groups influencing the boiling mechanism, which differed above and below the critical speed. Below the critical speed the effects of rotation were correlated by the inclusion of a Weber–Reynolds number term in the boiling equation of Lienhard [10] which took the form:

$$Q/A = 235(\Delta T)^{1\cdot 76}(k_L)(Pr)^{\frac{1}{3}} \times [1 + 178(We/Re)^{0\cdot 63}]. \tag{8}$$

Above the critical speed the same basic equation was modified to include a Froude number which gave the correlating equation:

$$Q/A = 520(\Delta T)^{1\cdot 76}(k_L)(Pr)^{\frac{1}{3}}(Fr)^{-0\cdot 186} \tag{9}$$

where k_L is the thermal conductivity of liquid. It should be noted that neither equation (8) nor equation (9) is dimensionless.

EXPERIMENTAL APPARATUS

The general arrangement of the experimental apparatus was as shown in Fig. 1.

The steel tank measured $11\frac{1}{2} \times 11\frac{1}{2} \times 18$ in. high (29·21 $\times$ 29·21 $\times$ 45·72 cm). Plexiglas windows were installed around the side walls of tank to permit visual observation and high-speed photography of the boiling phenomenon during the tests.

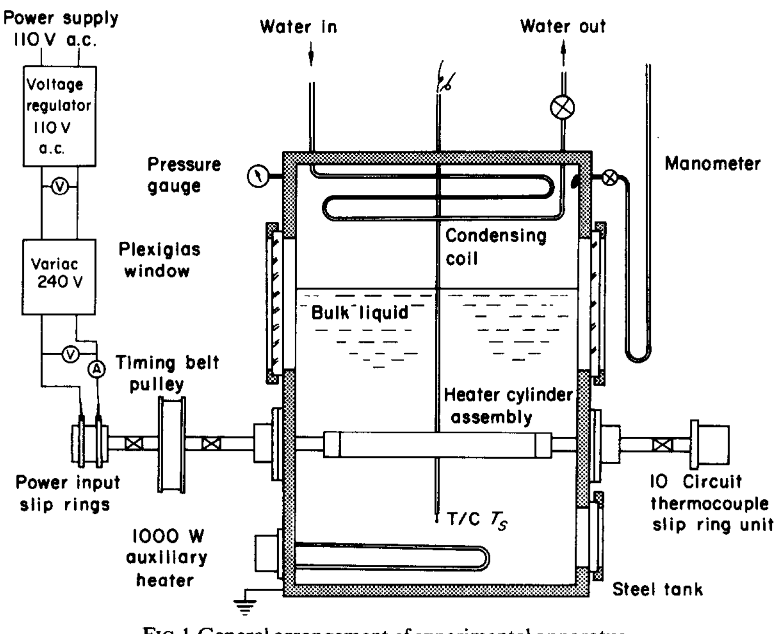


FIG. 1. General arrangement of experimental apparatus.

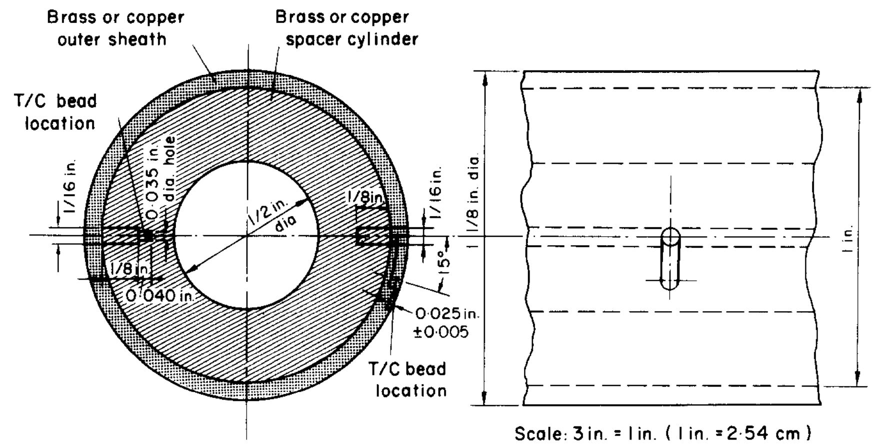


FIG. 2. Transverse cross section of heater cylinder assembly.

A liquid depth of approximately 11½ in. (29·21 cm) was maintained inside the tank during the tests. A thermocouple probe was used to measure the temperature of the bulk liquid. Two 1000 W auxiliary heaters, one at the front and the other at the rear of the tank, controlled by separate Variacs independently of the primary unit, were used to preheat the liquid to the saturation temperature. When boiling distilled water with a 500 W heat input it was necessary to use one of the heaters on a power input of approximately 200 W in order to maintain the bulk temperature of the boiling water at the saturation temperature.

The heater cylinder assembly, driven by a constant but adjustable speed motor, was 8 in. (20·32 cm) long with a diameter of 1·125 in. (2·86 cm). An electric cartridge heater, ½ in. (1·27 cm) dia. with a length of 8 in. (20·32 cm), was encased in the centre of the cylinder. Electrical power was supplied to this heater through three carbon brushes acting on each of two copper slip rings mounted on the hollow drive shaft. The temperature distribution within

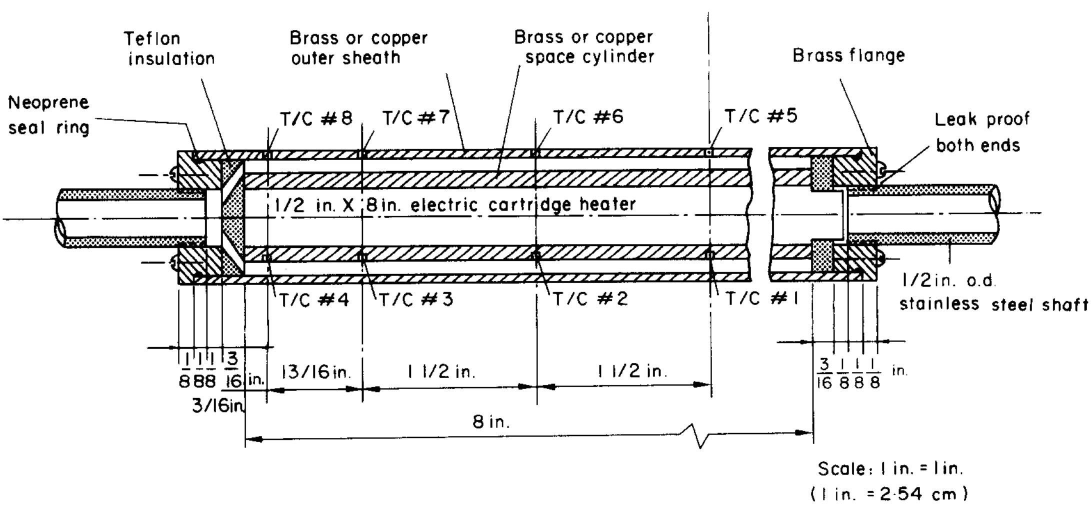


FIG. 3. Longitudinal cross section of heater cylinder assembly.

one half of the cylinder was measured using eight 30 gauge copper–constantan thermocouples embedded in the cylinder at two different radii 180° apart as shown in Figs. 2 and 3.

Two identical cylinders were tested which differed only in the material of the spacer cylinder and the outer sheath. For the copper cylinder, Alloys 110 and 122, manufactured by American Brass and Copper Company, were used to fabricate the spacer cylinder and the outer sheath respectively. For the brass cylinder, Alloys 360 and 260 were used respectively.

A condensing coil, made of $\frac{1}{2}$ in. copper tubing, was installed in the vapour-filled space above the liquid level inside the tank in order to maintain the pressure inside the tank equal to the atmospheric pressure during boiling.

EXPERIMENTAL PROCEDURE

Prior to each series of test runs, the heating surface was prepared by polishing it with No. 240 emery paper while rotating the cylinder. It was then washed with acetone.

The inside of the tank was thoroughly cleaned with acetone before being filled with the testing liquid to a depth approximately 6 in. above the test surface axis.

The power supply to the cylinder heater was adjusted to the desired level using a Variac. In addition the auxiliary heaters and condenser coil were used when necessary.

When steady state was reached, the values of all thermocouples, depth of liquid, power input of heater, manometer, barometer, rpm, room temperature, etc., were measured and recorded. In addition still and high-speed motion pictures were also taken during several test runs.

All experiments were conducted with atmospheric pressure at the vapour–liquid interface.

In order to observe clearly the effects of the rotary speed on the heat transfer coefficient, most of the test runs were conducted at a fixed heat input (500, 1000 or 1500 W) while the speed of the cylinder was varied incrementally from zero up to approximately 2000 rpm.

For each liquid–surface combination two specific series of tests were conducted and described as follows:

(a) *Stationary, constant heat flux, variable cylinder orientation tests*

The purpose of these tests was to establish the internal temperature distribution within the cylinder, to find the variation of heat transfer coefficients about the cylinder, and to compare the mean values with published data for a stationary surface.

(b) *Constant heat flux, variable speed tests*

The purpose of these tests was to study the effect of rotation on the heat transfer coefficients for a rotating surface.

Almost all test runs for each liquid–surface combination have been repeated in order to check the reproducibility of the experimental results. All instruments, thermocouples and equipment involved were carefully inspected, aligned and calibrated before and during each series of tests. Periodic checks indicated that the thermocouple readings were unaffected by the presence of electric equipment in the test area.

Longitudinal profilometer measurements of the roughness of the cylindrical heating surface were taken before and after a series of boiling runs. The roughness of the newly polished surface for both brass and copper cylinders was found to lie between 20 and 25 μin. (rms). No significant change in surface roughness for both cylinders was observed after the cylinder was used to boil either water or R-113.

DATA REDUCTION

The maximum possible end conduction loss from the heater cylinder assembly was estimated in order to determine whether their neglect would have any significant effect on the analysis of the surface temperature and heat flux calculations. This loss was determined by assuming that the heat was conducted through the stainless steel support shafts which were assumed to act as fins at each end and that the heater temperature was constant throughout. The *maximum* end loss through the hollow shafts at each end of the cylinder for various liquid–surface combinations was found to be less than 2·5 per cent of the total heat input to the cylinder.

Based upon the preceding calculations, it was concluded that the end conduction loss could be neglected for all surface temperature and heat density calculations. As a result, the heat flux density, Q, was evaluated directly from the electrical power input to the heater recorded by the voltmeter and ammeter by assuming that all the heat input to the test section was dissipated radially through the cylinder wall.

The temperature distribution near the cylinder surface along the longitudinal axis was found to be slightly erratic. These slight variations were probably caused by the thermal contact resistance generated at the interface between the spacer cylinder and the outer sheath [1]. Since fairly uniform temperature distribution throughout the centre part of the cylinder was observed, a $4\frac{1}{2}$ in. (11·43 cm) span along the centre section of the cylinder was selected and assumed to be the representative heat-transfer area for the calculation of the mean surface temperature.

DISCUSSION OF RESULTS

Figures 4 and 5 show plots of the heat flux density vs. mean temperature difference for different liquid–surface combination of the stationary surface. These figures also show the results of Corty and Foust [11], Blatt and Adt [12], Cryder and Gilliland [13], Cryder and Finalborgo [14], Linden and Montillon [15], and Nicol and McLean [8] for comparison. A better heat transfer performance was found for the copper surface compared with those of the previous investigators. For the water–brass combination, the curve for the present results agrees well with the data of Cryder and Gilliland but it is better than that of Cryder and Finalborgo. The disagreement is probably attribut-

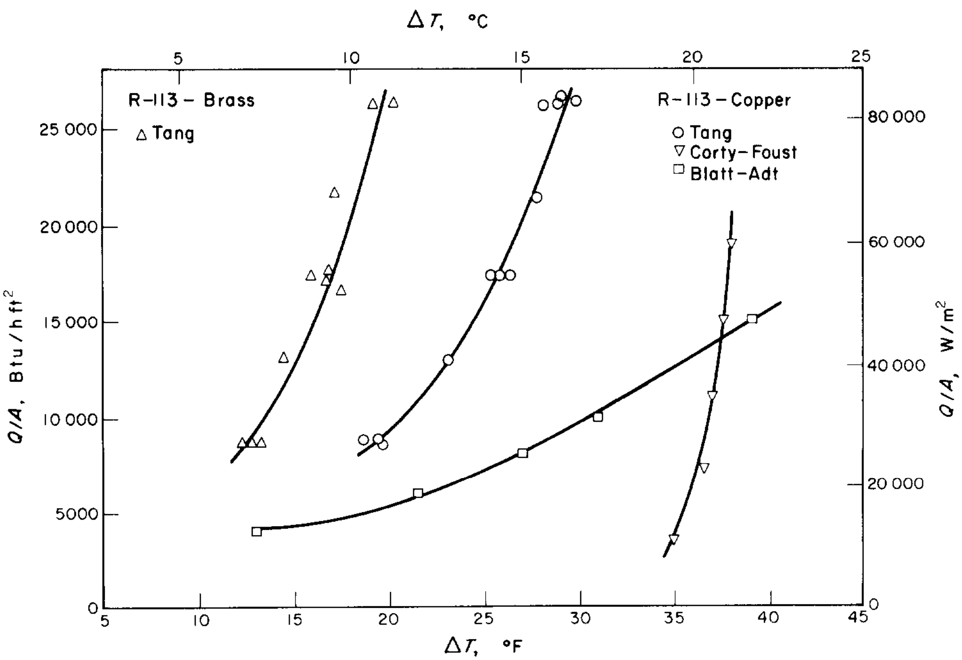


FIG. 4. Comparison of present results with previous investigation for a stationary surface (R-113).

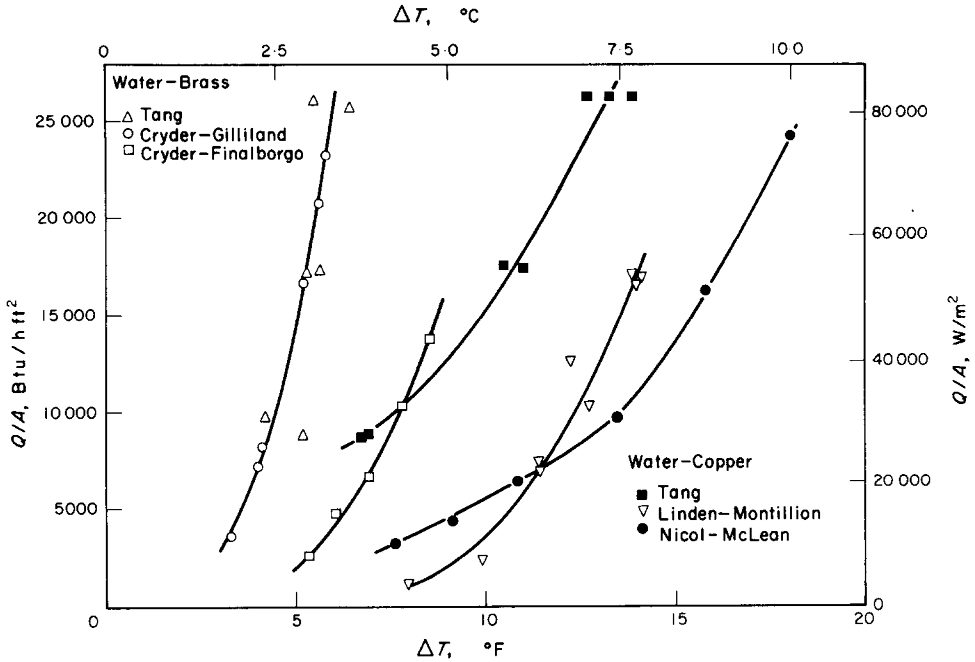


FIG. 5. Comparison of present results with previous investigation for a stationary surface (distilled water).

able to the difference in surface finish and to the geometric arrangement of the test section. The heat flux density is not only a function of the temperature but is also dependent upon the condition of the surface since the bubble population is greatly affected by this condition. By changing surface characteristics the shape of the nucleate boiling curve can be changed. Unfortunately, data for the R-113–brass combination are not available in the literature for comparison.

The experimental data of the stationary cylinder could be correlated satisfactorily by using the general form of the Rohsenow equation,

$$\frac{C_L(T_w - T_s)}{h_{fg}} = C_{sf}\left[\frac{Q/A}{\mu_L h_{fg}}\left(\frac{g_c\sigma}{g(\rho_L - \rho_v)}\right)^{\frac{1}{2}}\right]^r (Pr)^s \tag{10}$$

where $s = 1{\cdot}0$ for water and $1{\cdot}7$ for all other fluids. A comparison between the Rohsenow correlations for the present investigation and those for previous investigators using the same liquid–surface combination are shown in Figs. 6 and 7. Generally the correlation curves of the

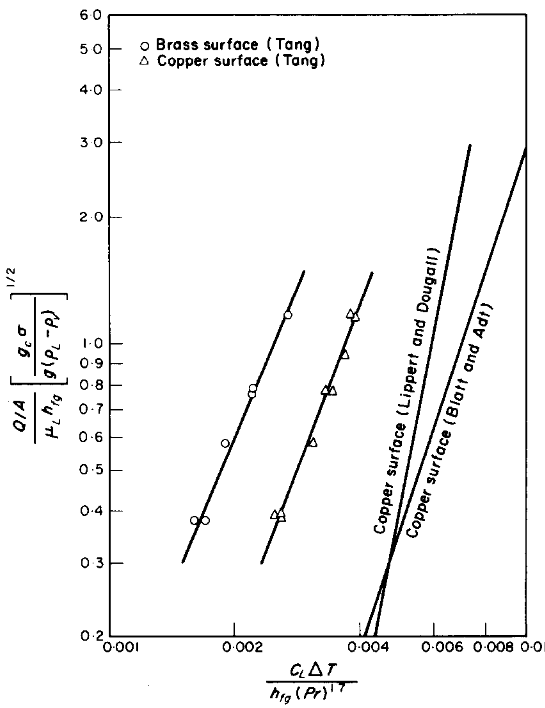


FIG. 6. Rohsenow correlations for boiling R-113.

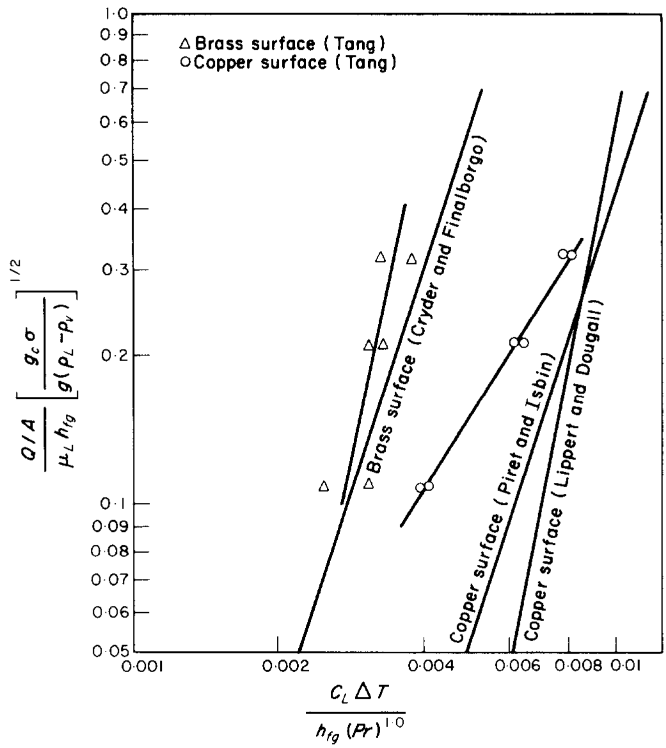


FIG. 7. Rohsenow correlations for boiling distilled water.

present study show a better heat transfer rate than those achieved by previous experimenters.

The calculated values of "C_{sf}" and "r" along with other values available from the literature are listed in Table 1. According to Vachon *et al.* [16], there is a definite dependency of "r" on surface preparation technique. This is the probable reason for the deviation of the measured values of "C_{sf}" and "r" among the different investigators and from the values suggested by Rohsenow.

McLean and Nicol [8, 9] found that a minimum ΔT occurred near a rotational speed of 150 rpm for a 0·922 in. (2·34 cm) dia. copper

Table 1. Values of C_{sf}, s, r *and correlation coefficient of Rohsenow equation equation* (10)

Surface fluid combination	s	r	C_{sf}	Coeff. of correlation	Investigators
Copper–water	1·0	0·642	0·017	0·996	Tang
Copper–water	1·0	0·33	0·013	—	Piret and Isbin [18]*
Copper–water	1·7	0·20	0·0112	—	Lippert and Dougall [17]
Copper–R-113	1·7	0·38	0·0037	0·993	Tang
Copper–R-113	1·7	0·200	0·0059	—	Lippert and Dougall [17]
Copper–R-113	1·7	0·33	0·007	—	Blatt and Adt [12]
Brass–water	1·0	0·220	0·0045	0·78	Tang
Brass–water	1·0	0·33	0·006	—	Cryder and Finalborgo [14]*
Brass–R-113	1·7	0·414	0·0025	0·990	Tang

* Values of C_{sf} and r were suggested by Rohsenow based on their experimental data.

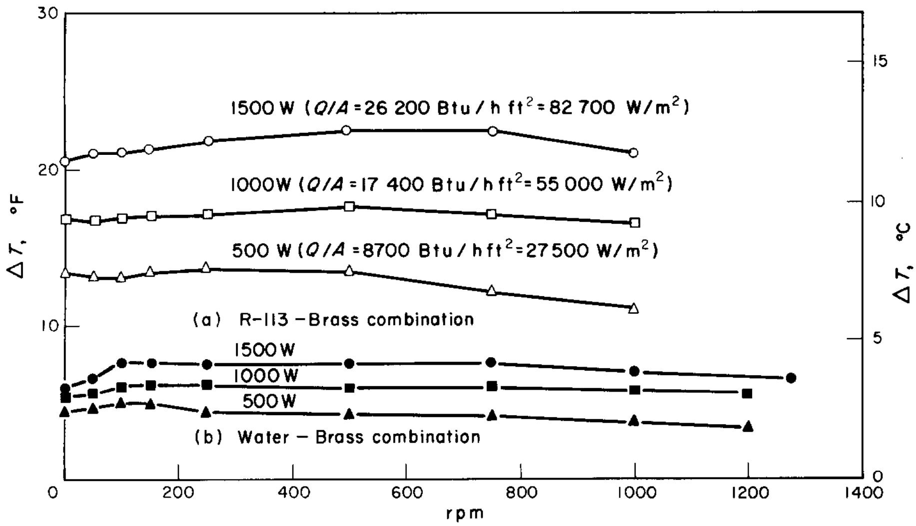


FIG. 8. Effect of speed on ΔT (brass surface).

cylinder in water. However, from Figs. 8 and 9 it may be observed that no significant change in ΔT occurs about this speed. Generally, these curves show a slight increase in ΔT as the speed increases up to a value about 500 rpm. Beyond this value a small but significant decrease of ΔT occurs with further increases in speed. The reason for the significant decrease in ΔT as the speed increases is probably due to the fact that at higher rotating speeds, where flow becomes turbulent, the boiling effects vanish and the heat transfer is mainly dominated by forced convection. Consequently, a lesser ΔT is required to remove the same amount of heat.

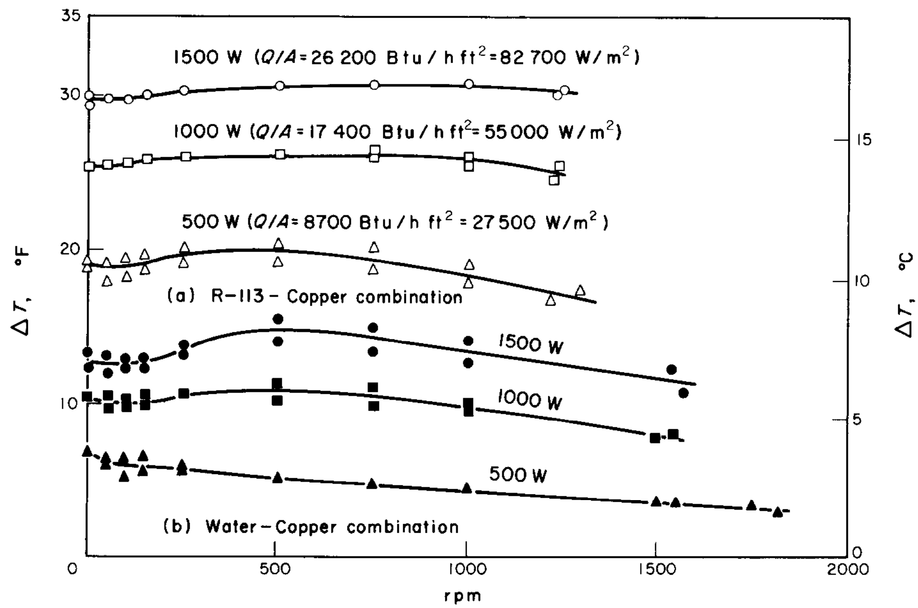


FIG. 9. Effect of speed on ΔT (copper surface).

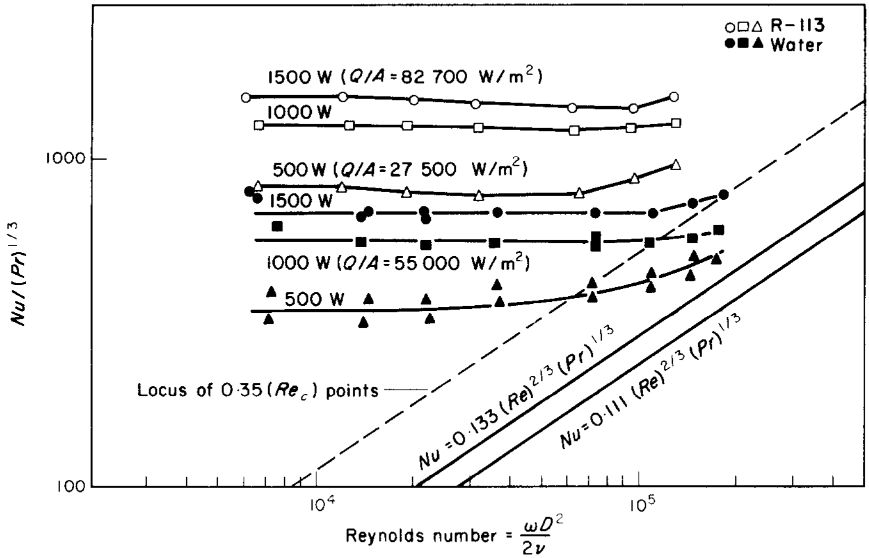


FIG. 10. Heat transfer from rotating horizontal cylinder (brass surface).

In order to facilitate a comparison of the present results with those for non-boiling forced convection from a rotating cylinder and with Nicol and McLean's results for boiling, a plot of $Nu/(Pr)^{\frac{1}{3}}$ vs. rotary Reynolds number $(\omega D^2/2\nu)$ was made. Figures 10 and 11 show these results along with the curves of Anderson and Saunders [2], and Becker [7]. The experimental data, based on the liquid properties evaluated at the saturation temperature, appear to be in reasonable agreement with the equation

$$Nu = 0{\cdot}12\,(Re)^{\frac{2}{3}}\,(Pr)^{\frac{1}{3}} \qquad (11)$$

in the non-boiling forced convection region. This equation is identical in form to the equations given by Anderson and Saunders and by

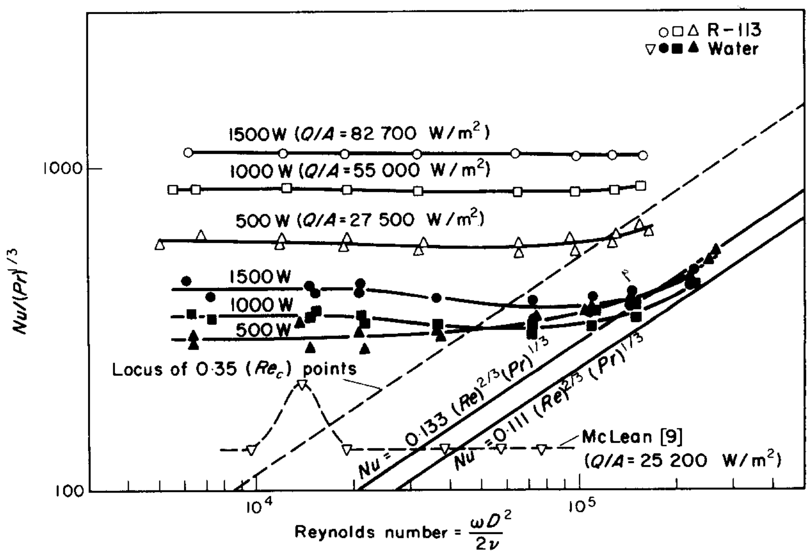
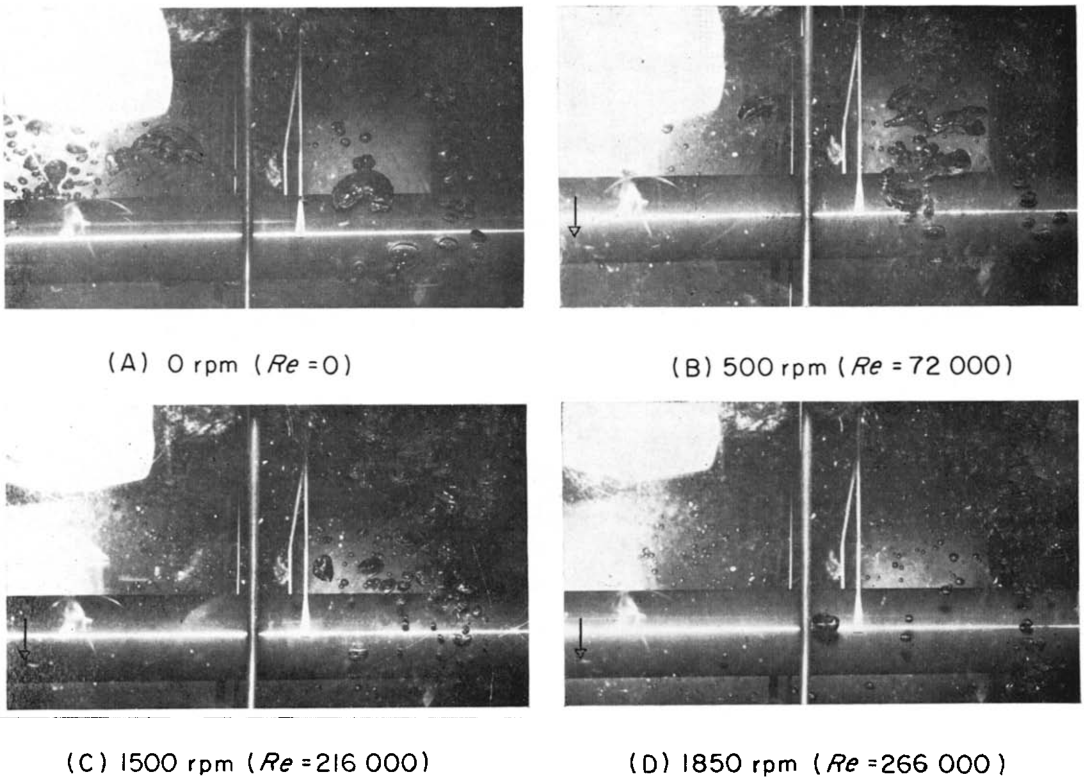
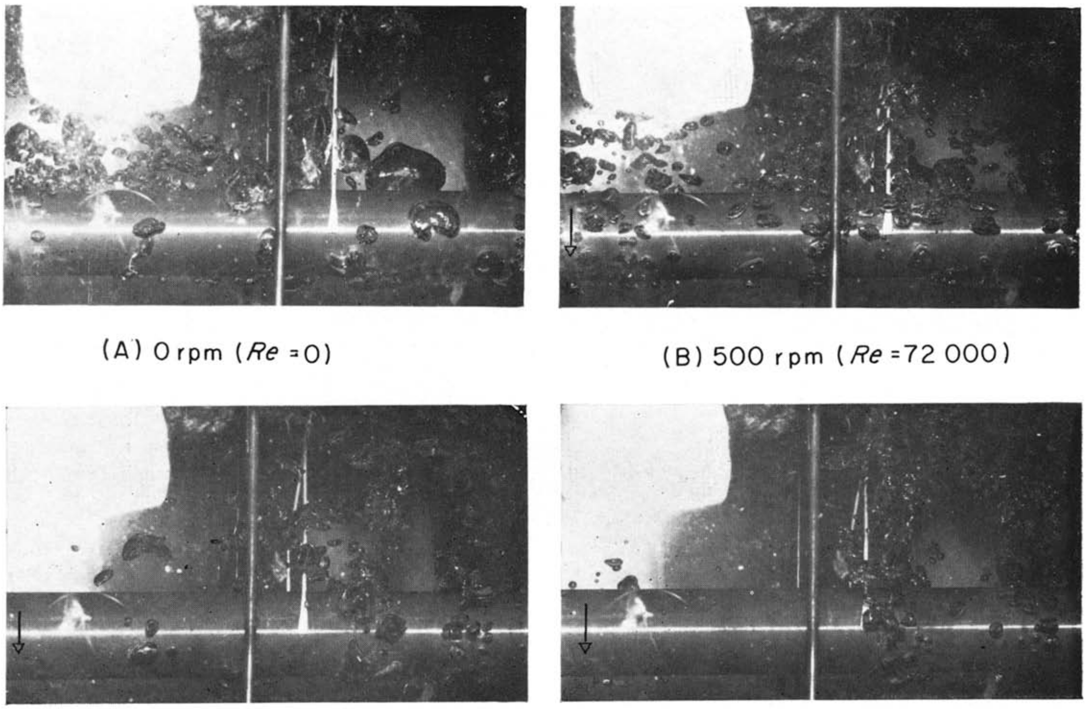


FIG. 11. Heat transfer from rotating horizontal cylinder (copper surface).



(Q/A = 27 500 W/m^2)

FIG. 12. Still pictures of boiling phenomena at various rotary speeds (water–copper combination; 500 Wheat input).



(A) 0 rpm (Re = 0)

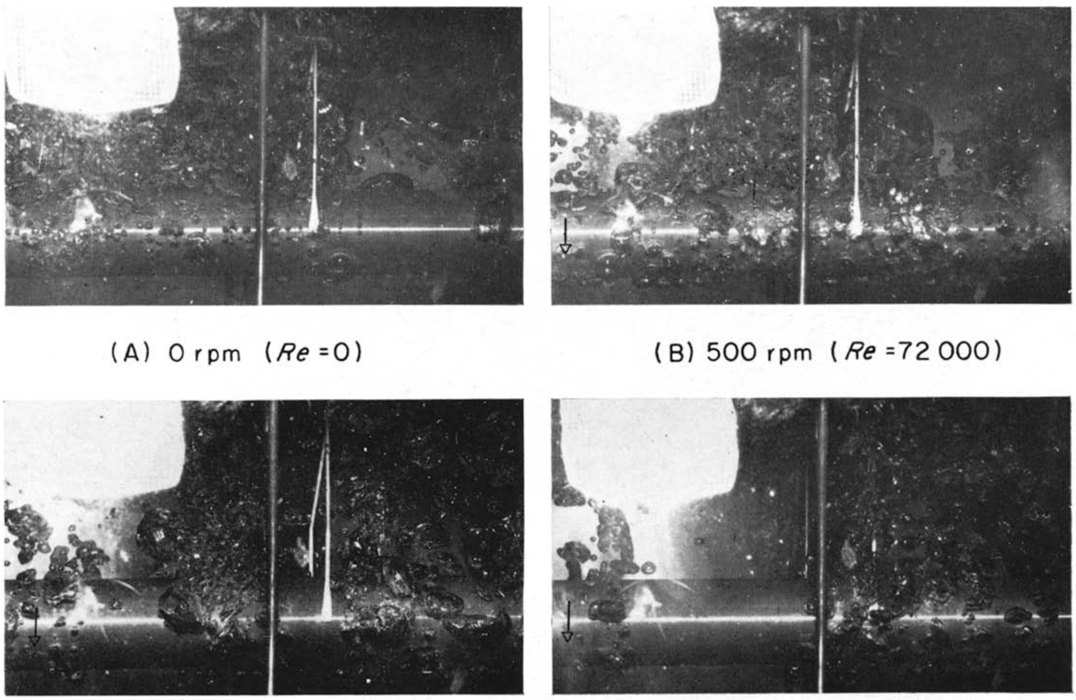
(B) 500 rpm (Re = 72 000)

(C) 1000 rpm (Re = 144 000)

(D) 1500 rpm (Re = 216 000)

(Q/A = 55 000 W/m^2)

FIG. 13. Still pictures of boiling phenomena at various rotary speeds (water–copper combination; 1000 W heat input).



(A) 0 rpm (Re = 0)

(B) 500 rpm (Re = 72 000)

(C) 1000 rpm (Re = 144 000)

(D) 1500 rpm (Re = 216 000)

(Q/A = 82 700 W/m^2)

FIG. 14. Still pictures of boiling phenomena at various rotary speeds (water–copper combination; 1500 W heat input).

Becker for properties evaluated at the mean film temperature. However, the constant, 0·12, in this equation is approximately equal to the average value of those used by Anderson and Saunders and by Becker.

A transition region between the boiling and non-boiling forced convection may be defined in terms of a critical rotating Reynolds number, Re_c, as that Reynolds number where the non-boiling forced convection correlation equation yields a value of Nu equal to that obtained for stationary boiling. Thus

$$Re_c = \left[\frac{Nu_{sb}}{0{\cdot}12(Pr)^{\frac{1}{3}}}\right]^{\frac{3}{2}} \qquad (12)$$

where Nu_{sb} is the Nusselt number based on the cylinder diameter for stationary boiling and where all physical properties are evaluated at the saturation temperature.

From Figs. 10 and 11, it is noted that the Nusselt number is almost independent of the Reynolds number up to approximately 35 per cent of the critical Reynolds number. Beyond this range the curves start to pitch upward and show a tendency to gradually approach the non-boiling convection curves as speed increases. The smooth transition is essentially complete when the Reynolds number is equal to approximately 135 per cent of the critical Reynolds number, the point when boiling has essentially ceased. Within this transition region, bubble size becomes more uniform and both bubble size and frequency reduce as rotary speed increases for the same heat flux. This phenomenon can be clearly observed from the still pictures, Figs. 12–14, by noting the decrease in the volume of the vapour bubbles near the cylinder as speed increases. This phenomenon was also observed in the high-speed motion pictures taken at various speeds and at various heat flux densities. Also, Fig. 12 (D) clearly indicates that the bubbles have almost completely disappeared at very high Reynolds number (Re = 266000) for the lowest heat flux density (500 W). Note that the majority of the small bubbles which appear behind the test cylinder are due to an auxiliary heater, not the test section.

Nicol and McLean's test results for water boiling about a similar 1 in. (2·54 cm) dia. copper cylinder with a heat flux of 8000 Btu/hft^2 (25 200 W/m^2) are also plotted in Fig. 11. Their results indicate a region of significantly improved heat transfer in the vicinity of 150 rpm. This phenomena was not observed during the present study at any heat flux with any liquid–surface combination. Generally, Nicol and McLean's data yield significantly lower heat transfer coefficients than those of the present investigation for a similar heat flux. Although the surface material, cylinder size, heat flux range and geometric arrangement of the heating surface of Nicol and McLean were similar to those of the present investigation, their instrumentation and test procedure were different. They measured the heater power input using a wattmeter with a current transformer and the thermocouple millivolt output using a potentiometer without an ice junction but compensated to ambient temperature. In addition, the heated cylinder of the present study had a smooth outside surface while that of Nicol and McLean had a discontinuous surface caused by the copper strips placed in the grooves covering the thermocouple lead wires. Unfortunately, Nicol and McLean did not state either the roughness of the heating surface or the technique used to prepare the surface. They claimed that the surface of the test surface was cleaned before each test run, however, they did not state whether the degrees of the cleanness of the heating surface for all test runs were identical. Nicol and McLean obtained their data by carrying out a series of constant-rotational-speed–variable-power-input tests whereas the present data were obtained using fixed-power-input–variable-speed tests. The advantage of the present test procedure is that the fixed-power-input–variable-speed tests enable each curve of ΔT vs. rpm to be obtained in a minimum time with minimum disturbance (aging, etc.) to the test surface. Since Nicol and

McLean performed their experiment at a constant speed by varying the power input to the test section, each of their ΔT–rpm curves (Fig. 11 of [9]) for a particular heat flux was not a continuous plot based on one series of test run but a cross plot based on the combination of all series of test runs. If the surface of one of the test runs had a significantly different degree of vection curves into a region where experimental data should not exist.

Due to structural considerations the speed of the rotating cylinder was limited to 2000 rpm. As a result only those curves (Fig. 11) for water–copper at various heat fluxes clearly show the merging of boiling curves into the non-boiling convection curves as the rotary speed increases.

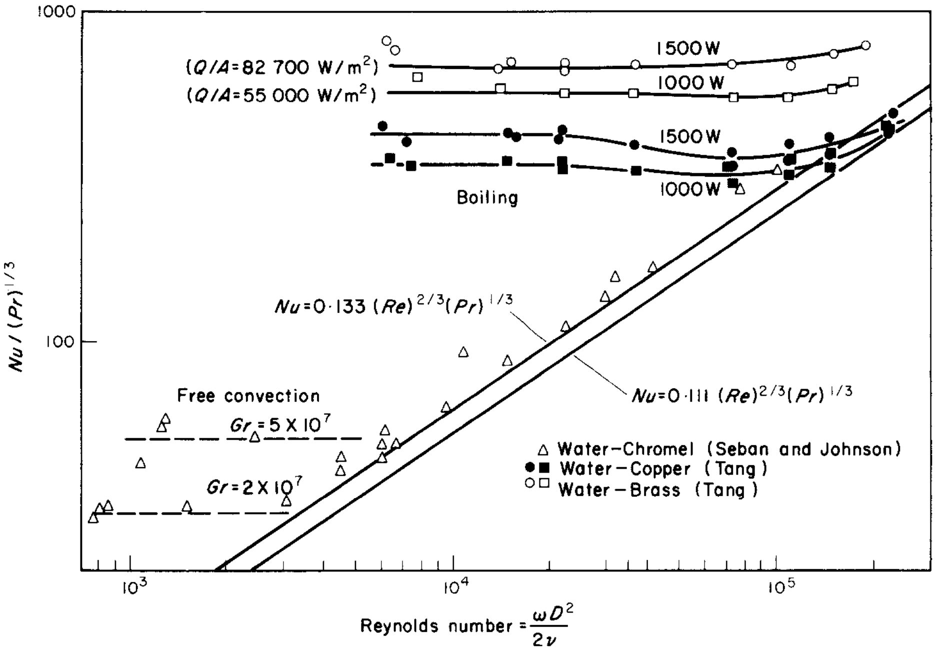


FIG. 15. Comparison between boiling and non-boiling heat transfer from a rotating horizontal cylinder.

cleanness from those of the rest then it may easily cause a dip or bulge on all of the ΔT–rpm curves. Nicol and McLean postulated that the significantly improved heat transfer in the vicinity of 150 rpm was due to differences in the behaviour and generation of bubbles at the seams of the copper strips covering the thermocouple lead wires.

In our opinion, Nicol and McLean's results, as shown in Fig. 11, are too low due to the fact that probably they did not properly estimate the true surface temperature of their cylinder. This opinion is further reinforced by the fact that their curve crosses over the non-boiling forced con-

To determine the effect of evaluating the liquid properties at various temperature, $Nu/(Pr)^{\frac{1}{3}}$ and Re were also calculated based on the mean film temperature in the non-boiling forced convection region. Compared to the plots for properties based on the saturation temperature, each point is shifted slightly upward and to the right. For example, for boiling R-113, the values of $Nu/(Pr)^{\frac{1}{3}}$ and Re based on properties evaluated at the film temperature are 5 and 9 per cent respectively greater than that evaluated at the saturation temperature for the maximum temperature difference of 15°F between the saturation and film temperatures at the highest (1500 W) heat input. For boiling water, these percentages are 1 and 4 per cent respectively arising from a temperature difference of 7°F. No significant change in the position of the resulting forced convection curves was observed.

A comparison between equations (6) and (7), data of Seban and Johnson [6] for non-boiling convection from a rotating cylinder, and the results of the present study for boiling heat transfer from a horizontal cylinder rotating in water is given in Fig. 15. For Reynolds numbers above about 3000, Seban and Johnson's data compare excellently with Becker's correlation. Below this value the rather high measured Nusselt numbers indicate the presence of free convectional effects. Figure 15 also provides evidence of similarity between the relationship of free convection and forced convection and the relationship of boiling and non-boiling forced convection on heat transfer from a rotating cylinder.

CONCLUSIONS

This experimental study on boiling heat transfer from stationary and rotating cylinders with heat fluxes in the range 8000–26000 Btu/hft^2 led to the following conclusions:

1. When a heated circular cylinder is rotated in a liquid at its saturation temperature, the thermal behaviour of the system falls into one of three distinct regions which may be defined in terms of the critical Reynolds number, Re_c, as follows:

(a) *Boiling heat transfer region* ($\mathrm{Re} < 0{\cdot}35\ \mathrm{Re}_c$)

In this region the heat transfer is almost independent of speed of rotation. Existing correlations, such as Rohsenow, Zuber and Forster, etc., for boiling from a stationary surface with the same kind of liquid–surface combination may be applied to predict the heat transfer characteristics.

No significant improvement in the heat transfer coefficient as a result of rotation, such as that observed by Nicol and McLean at a rotational Reynolds number of approximately 14500 for water boiling on a similar copper cylinder, was observed.

(b) *Transition region* ($0{\cdot}35\ \mathrm{Re}_c < \mathrm{Re} < 1{\cdot}35\ \mathrm{Re}_c$)

In this region the heat transfer is influenced both by the boiling process and the forced convection due to the cylinder rotation. All boiling curves smoothly and gradually merge into the forced convection heat-transfer curves as the speed increases while the vigor of the ebullition gradually reduces until the boiling process completely ceases.

(c) *Non-boiling convection region* ($\mathrm{Re} > 1{\cdot}35\ \mathrm{Re}_c$)

In this region boiling does not occur, all heat being removed from the surface by forced convection. The present experimental results in this region have been found to compare very well with the correlations of Anderson and Saunders and of Becker. Thus it is suggested that

$$Nu = 0{\cdot}12\,(Re)^{\frac{2}{3}}\,(Pr)^{\frac{1}{3}}$$

may be used to predict the heat transfer coefficient in this region.

It should be noted that these results may not be applicable for higher heat fluxes where some other boiling mechanism may dominate the heat transfer.

2. The values of C_{sf} and r in the Rohsenow equation for boiling from a stationary cylinder for the brass–R-113 combination were found to be 0·0025 and 0·414 respectively. This data were not previously published.

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UNE ÉTUDE DU TRANSFERT THERMIQUE LORS DE L'ÉBULLITION À PARTIR D'UN CYLINDRE HORIZONTAL TOURNANT

Résumé—On a étudié l'influence de la rotation sur le transfert thermique lors de l'ébullition nucléée à partir d'un cylindre horizontal circulaire chauffé tournant autour de son axe dans le réfrigérant 113 et dans de l'eau distillée. On a considéré des cylindres de cuivre et de laiton de diamètre 2,86 m et de longueur 20, 32 cm à des densités de flux variant entre 25 200 et 82 000 W/m² pour des nombres de Reynolds de rotation compris entre 0 et 2,6.10⁵. On a montré que la rotation n'a pas d'effet significatif sur le coefficient de transfert thermique pour des vitesses basses ou modérées. Dans cette région, pour

$$0 \leqslant \left(Re = \frac{D^2\omega}{2\nu}\right) < 0{\cdot}35\left[Re_c = \left(\frac{Nu_{s,b}}{0{\cdot}12\,(Pr)^{\frac{1}{3}}}\right)^{\frac{3}{2}}\right]$$

où $Nu_{s,b}$ est le nombre de Nusselt basé sur la diamètre du cylindre lors d'une ébullition stationnaire. On aurait pu à l'aide de correlations existantes sur le transfèrt de chaleur par ébullition pour une surface stationnaire avec un interface spécifique liquide–solide, prédire le flux thermique. Pour des vitesses de rotation très élevées, l'ébullition cesse et la convection forcée devient le seul mode de transfert thermique. Dans cette région pour $Re > 1{,}35\ Re_c$, les résultats expérimentaux sont unifiés par la relation:

$$Nu = 0{,}12\left(\frac{D^2\omega}{2\nu}\right)^{\frac{2}{3}}(Pr)^{\frac{1}{3}}$$

Entre ces cas limites on a observé une région de transition graduelle dans laquelle la taille et la fréquence des bulles décroissent avec l'augmentation de la vitesse jusqu'à la disparition totale des bulles. on a trouvé que les résultats de transfert thermique pour à la fois l'ébullition avec surface stationnaire et la convection forcée sont généralement en accord avec ceux des autres chercheurs. De plus, on a déterminé les valeurs de "C_{sf}" et de "r" dans l'équation de Rohsenow pour l'ébullition à partir d'un cylindre stationnaire pour laiton—R-113.

EINE UNTERSUCHUNG DES WÄRMEÜBERGANGS BEIM SIEDEN AN EINEM ROTIERENDEN HORIZONTALEN ZYLINDER

Zusammenfassung—Es wurde der Einfluss der Rotation untersucht auf den Wärmeübergang beim Blasensieden an einem horizontalen beheizten Kreiszylinder, der sich im Kältemittel-113 und in destilliertem

Wasser um seine Achse dreht. Versuche wurden gemacht mit Kupfer- und Messingzylindern von 2,86 cm Durchmesser und 20,32 cm Länge, bei Wärmestromdichten zwischen 2,52 und 8,2 W/cm² über einen Rotations-Reynolds-Zahlenbereich von 0–2,6 · 10^5. Es zeigte sich, dass die Rotation bei niedrigen und mässigen Geschwindigkeiten keinen spezifischen Einfluss auf die Wärmeübergangszahlen hat. In diesem Bereich für

$$0 \leqslant \left(Re = \frac{D^2\omega}{2\nu}\right) < 0{,}35\left[Re_c = \left(\frac{Nu_{s.b.}}{0{,}12(Pr)^{\frac{1}{3}}}\right)^{\frac{3}{2}}\right],$$

wobei $Nu_{s.b.}$ die auf den Zylinderdurchmesser bezogene Nusselt-Zahl für Sieden im Ruhezustand ist, konnten bestehende Beziehungen für den Wärmeübergang beim Sieden an einer feststehenden Oberfläche für eine bestimmte Flüssigkeits-Heizflächen-Paarung benutzt werden, um den Wärmefluss zu bestimmen. Bei sehr hohen Drehgeschwindigkeiten verschwand das Sieden, und allein die erzwungene Konvektion war für den Wärmeübergang massgebend. In diesem Bereich, für $Re > 1{,}35\ Re_c$, lassen sich die Versuchswerte gut korrelieren nach der Gleichung

$$Nu = 0{,}12\left(\frac{D^2\omega}{2\nu}\right)^{\frac{2}{3}}.(Pr)^{\frac{1}{3}}.$$

Zwischen diesen Grenzfällen konnte ein gleichmässig ansteigender Übergangsbereich beobachtet werden, in dem Blasengrösse und -frequenz mit wachsender Geschwindigkeit abnehmen, bis die Blasen völlig verschwinden. Die Ergebnisse im Wärmeübergang für Sieden an festen Flächen und für erzwungene Konvektion zeigten sich in allgemeiner Übereinstimmung mit denen von anderen Autoren. Ausserdem wurden noch die Wente für "C_{sf}" und "r" aus der Rohsenowbezüchung für Sieden an einem feststehenden Zylinder für die Messing-R-113-Paarung bestimmt.

ИССЛЕДОВАНИЕ ТЕПЛООБМЕНА ПРИ КИПЕНИИ ОТ ВРАЩАЮЩЕГОСЯ ГОРИЗОНТАЛЬНОГО ЦИЛИНДРА

Аннотация—Изучался теплообмен горизонтального нагретого кругового цилиндра, вращающегося вокруг своей оси в рефрижеранте и дистиллированной воде при пузырьковом (ядерном) кипении. Опыты проводились с медными и латунными цилиндрами диаметром 1,125 дюймов (2,86 см) и длиной 8 дюймов (20,32 см) при плотностях теплового потока от 25200 до 82000 вт/м² в диапазоне чисел Рейнольдса вращения от 0 до 2,6·10^5. Найдено, что вращение не оказывает существенного влияния на коэффициент теплообмена в диапазоне от низких до умеренных чисел оборотов. В этой области, т.е. при условии

$$0 \leqslant \left(Re = \frac{D^2\omega}{2\nu}\right) < 0{,}35\left[Re_c = \left(\frac{Nu_{s.b.}}{0{,}12(Pr)^{1/3}}\right)^{3/2}\right],$$

($Nu_{s.b.}$ — число Нуссельта, основанное на диаметре цилиндра применитель но к условиям стационарного кипения) можно пользоваться для количественных оценок величин тепловых потоков существующими корреляциями теплообмена при стационарном кипении на характерной поверхности раздела жидкость-твёрдое тело. При очень больших скоростях вращения кипение прекращается и вынужденная конвекция становится единственной формой теплообмена. В этой области при $Re > 1{,}35\ Re_c$ экспериментальные данные хорошо описываются соотношением

$$Nu = 0{,}12\left(\frac{D^2\omega}{2\nu}\right)^{2/3}(Pr)^{1/3}$$

Между этими предельными случаями наблюдалась гладкая и равномерная пе реходная область, в которой размер и частота пузырьков уменьшаются с увеличением скорости вплоть до полного исчезновения пузырьков. Результаты, полученные для стационарного поверхностного кипения и вынужденной конвекции, в общем, согласуются с данными других исследователей. Кроме того, определены величины параметров «C_{sf}» и «r» в уравнении Розенова для кипения на стационарном цилиндре пары латунь — R — 113.