THE EFFECT OF PURE FORCED CONVECTION ON THE BOILING HEAT TRANSFER BETWEEN A TWO-DIMENSIONAL SUBCOOLED WATER JET AND A HEATED SURFACE

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This paper is concerned with heat transfer from a heated surface with uniform heat flux to a two-dimensional water jet. It describes quantitatively the results on the effect of pure forced convection on heat transfer with boiling in a flow of subcooled water. Experiments were carried out in two flow fields of an impinging water jet, at the stagnation point and in the parallel flow zone. Approximation of the superposition, that is, the total heat flux at forced convective boiling consisting of pool boiling heat flux and pure convective heat flux, held fairly good up to the burn-out heat flux region.

Introduction

This paper is concerned with heat transfer between a two-dimensional water jet and a heated surface with uniform heat flux. An impinging water jet onto the heated surface is used as a method of increasing heat transfer rates at low temperature differences. Heat transfer associated with nonboiling and boiling in an impinging water jet system is a problem of great current interest in many technological applications. Most recent work in this field has been stimulated by the application of gaseous impingement cooling to high-temperature gas turbine engines; consequently, a large number of papers have appeared about impinging gas jets, but there seems to be scant heat transfer information about a two-dimensional water jet.

It is the purpose of this paper to present data on the effects of the impinging velocity on the rate of heat transfer in the nonboiling and the boiling process. Experimental data were obtained for natural and forced convective regions, pool and forced convective nucleate boiling regions at high subcooling.

1. Experimental Apparatus and Procedure

Figure 1 shows the experimental apparatus for the impinging water jet. Tap water was pumped up to a height of about 15 m and poured into the storage tank. The rectangular nozzle of 10 mm × 30 mm and the parallel partition plates attached to the end of a feed pipe were designed to maintain the two-dimensional character of the jet. The water jet was impinged downward normally on a heated surface. The jet velocity was changed by the regulative ball valve in the range of up to 16 m/sec. The distance between nozzle and heated surface was kept at 15 mm. The jet velocity was calculated from the measured values of the static pressure at the stagnation point. The jet water temperature measured at the outlet of the nozzle was kept at about 15°C.

Figure 2 shows details of the test section, heat transfer surface and thermocouple location. The electrodes were brazed with a platinum foil of thickness 0.1 mm. This platinum foil was heated directly by alternating current. The rectangular central part of this foil was 4 mm × 8 mm and was used as a heated surface. The a.c. power input was controlled with a 10 KVA adjustable transformer and supplied to the electrodes through another transformer of low voltage and high current. The temperature was measured by alumel-chromel thermocouples of 50 μm dia. attached to the back side of the foil by spot welding. The platinum wires of 50 μm dia. used as voltage taps were attached in the same way as the thermocouples. The thermocouples and voltage taps were electrically insulated by a Teflon coating. The back side of the platinum foil was coated with Teflon to prevent entry of the jet water into the back side and heat loss from the same side. The back side was stiffened with epoxy resin to prevent swelling of the platinum foil induced by heat expansion. The surface of Teflon coating was pretreated to stick firmly to the epoxy.
Fig. 1 Experimental apparatus

Fig. 2 Details of test section

resin. Five or seven alumel-chromel thermocouples attached to the back side of the foil in the direction of flow were used to measure the temperature distribution of the heated surface. This distribution, measured at right angles to the direction of flow was uniform. The voltage was measured by a highly sensitive a. c. mV meter. The electric current was measured through the current transformer. The temperature of the heated surface $T_w$ was determined by

$$T_w = T_0 - q \cdot \delta / (2 \cdot \lambda)$$  \hspace{1cm} (1)

where $T_0$ was the back-side temperature of the foil, $\delta$ was the foil thickness, $\lambda$ was the thermal conductivity and $q$ was the heat flux. $T_0$ was a time-average value measured by the galvanograph recorder. Figure 3 shows the static pressure distribution and the velocity distribution at the outer edge of the boundary layer. The temperature and the voltage were measured at the two points $X_s$ and $X_u$ shown in Fig. 3. The point $X_s$ was located at the stagnation point, $X_u$ at a distance of 25 mm from the stagnation point. The test piece used for the measurement of the temperature and the voltage at the stagnation point $X_s$ was moved from $X_s$ to $X_u$ and was used to measure at the point $X_u$. Static pressure distributions of the impinging water jet were measured with seven static holes of 0.5 mm dia. installed with separation distance of about 7 mm, flush with the plate surface. The measured static pressure distribution was used to compute the velocity distribution at the outer edge of the boundary layer.

Experimental apparatus for natural convection and pool boiling is shown in Fig. 4. The test liquid, about ten liters of water, was placed into the pool vessel of 215 mm ID. Two 40 mm-dia. plane glass windows were located at the bottom of the vessel. Pressure in the vessel was provided by nitrogen gas fed from a cylinder. The test piece used for pool boiling was identical to that shown in Fig. 2. The liquid temperature distribution was measured by five copper-constantan thermocouples which were flush and arranged with a separation of 10 mm.

Before each experimental run the heat transfer surface was cleaned with a 5/0 emery paper and acetone. After a steady-state condition was reached, as indicated by temperature measurements, the experimental data were taken.

2. Experimental Results and Discussion

2.1 Temperature distribution of the heated surface

Figure 5 shows the distribution of the back-side temperature $T_0$ at the measured location $X_s$. $T_0$ is plotted against $l/L$ with a parameter of the heat flux $q$. Here $l$ is the distance from the center of the heated surface in the flowing direction, $L$ is the half of the
heated length. It was considered roughly that the temperature distribution was uniform in the nonboiling region under 50°C and in the nucleate boiling region above 120°C, but in the range of $T_0$, 50°C to 120°C it was quadratic curve. Calculations of the heat conduction based on this temperature distribution showed that the heat losses through both cross-sections of heat transfer surface were negligible compared with the rate of heat transfer. Figure 6 shows the temperature distribution at the measured location $X_0$. Figure 6 is similar in temperature profiles to Fig. 5. In the nonboiling region, the maximum point of temperature moved to a point at a little distance downstream from the center of the heated surface. As boiling started, the maximum point of temperature moved to a point of the upper stream at a little distance from the center. In high heat flux nucleate boiling the temperature rose again as the heat flux increased and the temperature distribution became roughly uniform over the whole heated surface. Such a variation of temperature is closely related to the population density of active sites on the heated surface.

2.2 Heat transfer of forced convection at the stagnation point

In Fig. 7, $q$ vs. $(T_w-T_i)$ at the stagnation point of the impinging jet was plotted with a parameter of the jet velocity. The data can be plotted roughly on the lines with a slope of unity and the heat flux increases proportionally to the square root of the jet velocity in the nonboiling region. When $(T_w-T_i)$ is increased sufficiently to cause boiling to begin, the curve bends sharply upward. In the nonboiling region, the results for each velocity are expressed with those expected from conventional equations for forced convection. In several books, the heat transfer coefficient at the stagnation point of an impinging jet was derived as a basis of the so-called wedgeflow solutions of the laminar boundary layer equations obtained for two-dimensional, incompressible flow over wedges. In the neighborhood of the stagnation point, the heat transfer coefficient can be expressed approximately by

$$\alpha_{st} = 0.57 \Pr^{1/3} \cdot \lambda_i \cdot \sqrt{(a \cdot u_0)}(B \cdot \nu_l)$$

where $a$ is the dimensionless velocity gradient and $B$ is the width of the jet nozzle. The heat transfer coefficient measured in the present experiments showed a value about 1.8 times $\alpha_{st}$, and the following experimental equation was obtained.

$$\alpha_{eo} = 1.03 \Pr^{1/3} \cdot \lambda_i \cdot \sqrt{(a \cdot u_0)}(B \cdot \nu_l)$$

The heat flux $q_{eo}$ can be expressed by

$$q_{eo} = \alpha_{eo}(T_w-T_i) = \alpha_{eo}(\Delta T_{sat} + \Delta T_{sub})$$

The solid line shown in Fig. 7 is expressed by Eq. (4). All the experimental data in the nonboiling region seem to satisfy Eq. (4) sufficiently. The slope of the solid lines shown in Fig. 7 wasn’t unity, but increased
slightly as the temperature of the heated surface rose because of the variation of properties with temperature. The effect of the variation of properties with temperature has been correlated with the above equation by evaluating the properties at the “film” temperature \((T_{w} + T_{i})/2\). The increase of heat transfer rates at the stagnation point over the theoretical predictions may be caused by the turbulence of the jet. In this investigation the intensity of the turbulence was neither controlled nor measured. However, the turbulence generated by the jet itself plays an important role in determining the heat transfer characteristics of impinging jets, and further study will be necessary to completely determine the effect of this turbulence. The boiling region of Fig. 7 is magnified in Fig. 12 by plotting the data with \((T_{w} - T_{sat})\) instead of \((T_{w} - T_{i})\). The effect of the jet velocity is clearly revealed. In Fig. 8 the heat transfer coefficient defined by Eq. (4) is plotted against the heat flux up to the high heat flux nucleate boiling region. In the natural convective region, the experimental data are very scattered. The scattering is due to the temperature variation caused by the generation and departure of gas bubbles having a diameter of about 0.5 mm. This test liquid containing gas releases the gas at the hot surface. The resulting gas bubbles agitate the liquid in the same way as vapor bubbles. In the natural convective region, the temperature of the heated surface continued to decrease while the gas bubbles grew on the surface, then increased rapidly at the moment of departure of the gas bubbles. In the case of forced convection, on the contrary, the temperature of the surface continued to increase while the gas bubbles grew on the surface and decreased rapidly at the moment of departure. There is not so much scattering in forced convection owing to the effect of blowing off the gas bubbles.

2.3 Natural convection

The data on heat transfer by natural convection from a horizontal plate already were shown in Fig. 8. Figure 9 shows a logarithmic graph of the Nusselt number plotted as ordinate vs. the product of the Grashof and Prandtl numbers. A geometrical factor in both the Nu and Gr was taken as the smaller length of a horizontal rectangular surface. The physical properties were evaluated at a film temperature. It can be considered that there is no effect of system pressure within the limits of this experiment on heat transfer coefficients for natural convection. In several books\(^7\), the data on heat transfer by natural convection from horizontal plates facing upward in the laminar range have been correlated by the following equation.

\[
Nu = 0.54 \left( Pr \cdot Gr \right)^{1/4} \tag{5}
\]

Our experimental result agrees well with Eq. (5).

2.4 Heat flux for incipient boiling

We would like to confine ourselves here to the conditions of incipient boiling with forced convection at high subcooling. This problem has been investigated by many researchers\(^2\rightarrow\)\(^8\)\(^\rightarrow\)\(^14\), and it has been reported that the heat flux of initial boiling varied with fluid temperature, liquid velocity, system pressure and so on. The relation \(\alpha\) and \(q\) in the boiling region shown in Fig. 8 was expressed with a fairly good approximation by the following equation.

\[
\alpha = 0.039 q^{0.90} \tag{6}
\]

If the condition which simultaneously satisfied Eq. (3), Eq. (4), and Eq. (6) was obtained, the heat flux of initial boiling and \(\Delta T_{sat}\) were determined. The heat flux of initial boiling was expressed approximately within a range of \(\Delta T_{sub} 85^\circ C\) to \(110^\circ C\) and at atmospheric pressure by

\[
q_i = 1.20 \times 10^6 u_0^{0.68} \tag{7}
\]

where \(q_i\) is in kcal/m\(^2\)-hr, \(u_0\) in m/sec. Equation (7)
Fig. 10 Heat flux for incipient boiling against
jet velocity

Fig. 11 Correlation of local heat transfer

is shown in Fig. 10 and for comparison the equation of Sato et al.\textsuperscript{14)} is also shown.

2.5 The local heat transfer

Figure 3 suggests that the flow field can be divided into three flow fields; the first is a stagnation flow zone where \( U_{\text{max}} = cX \), the second an impingement flow zone where \( 0.5 < U_{\text{max}}/u_0 < 1.0 \) and the last is a uniform parallel flow zone where \( U_{\text{max}}/u_0 = 1.0 \). The correlation of the local heat transfer is shown in Fig. 11. An obstacle to determination of the local distance \( X \) in the \( Re_x \) is the fact that no actual leading edge of the flat plate exists in the uniform parallel flow zone. However, the local distance \( X \) in the \( Re_x \) was measured as the distance from the stagnation point, according to the thinking described by McMurray et al.\textsuperscript{9)}. The \( Re_x \) must be based on the local velocity just outside the boundary layer. It is necessary to consider the effect of an unheated starting length on heat transfer since in this experiment there exists an unheated starting length \( X_0 \) with the wall temperature maintained at the liquid temperature. The experimental data shown in Fig. 11 were expressed by the following equation:

\[
Nu_x = 0.014 \ Pr^{1/3} \ Re_x^{0.8} \left[ 1 - \left( \frac{X_0}{X} \right)^{0.4} \right]^{-1/3}
\]  

Equation (9) is for the laminar boundary layer and Eq. (10) for the turbulent. Their experiment was carried out with water velocities from 9.1 to 18.9 m/sec heat fluxes from \( 3.3 \times 10^5 \) to \( 6.5 \times 10^5 \) kcal/m\(^2\)-hr. It is found that Eq. (8) agrees well with Eq. (10) because in the right-hand term of Eq. (8), the value 0.014\([1 - (X_0/X)^{0.4}]^{-1/3}\) is 0.033. It can be considered that there is no effect of an unheated starting length on heat transfer in the range very near the heated starting point.

2.6 Surface boiling of forced convection

The data for surface boiling of the impinging water jet are plotted in Fig. 12 vs. \( DT_{\text{sat}} \). Although the study of surface boiling with forced convection has been reported by many researchers, only a few experimental reports of surface boiling studied at high heat flux above \( 10^6 \) kcal/m\(^2\)-hr can be found. Gunther\textsuperscript{5)}, employing an electrically heated steel foil in a rectangular water channel, reported a maximum heat flux \( 3.1 \times 10^6 \) kcal/m\(^2\)-hr with water velocities from 1.5 to 12 m/sec, pressures from 1 to 11.1 atm and \( DT_{\text{sat}} \) from 12.2 to 157°C. Rohsenow and Clark\textsuperscript{13}) reported a maximum heat flux of about \( 8.2 \times 10^6 \) kcal/m\(^2\)-hr for forced convective boiling of water in an electrically heated nickel tube (4.57 mm dia.) with water velocities from 3.0 to 9.0 m/sec, pressure from 70 to 140 ata, and Rohsenow\textsuperscript{12}) showed for nucleate boiling with convection that the convective effect was
superimposed on the vapor bubble motion effect:

\[ q_t = q_{so} + q_{boli} \]  

(11)

Here \( q_t \) is total heat flux for nucleate boiling with convection, \( q_{so} \) is the heat flux associated with either forced or natural convection in the absence of boiling and \( q_{boli} \) is the boiling heat flux associated with bubble motion alone in the absence of convection. In all the above-mentioned reports many problems remain unsolved, whether the boiling curves separated correspondingly to the velocities approach an asymptote at high heat flux or not, and whether the asymptote is considered to be an extension of the saturated pool boiling curve or not. The solid lines in Fig. 12 show the data at the stagnation point with jet velocities from 1.1 to 15.3 m/sec and the broken lines show the data obtained in a uniform parallel flow zone. All the boiling curves have the shape of S curves at low \( \Delta T_{sat} \). Boiling curves which exhibit a definite decrease in \( \Delta T_{sat} \) as the heat flux rises have been termed S-shaped boiling curves. The phenomenon of S-shaped boiling curves in this experiment can be explained by a way of thinking reported by Aluf Orell11. In Fig. 12, the phenomenon of S-shaped boiling curves occurs more remarkably in the case of uniform parallel flow showed by the broken lines. The boiling curves showed by the broken lines and the solid lines approach an extension of the subcooled pool boiling curve expressed by a dot-dash line and appear on it as \( \Delta T_{sat} \) rises. To make clear the above-mentioned facts, the data obtained by another experimental apparatus10 with the boiling surface made of a copper block are plotted in Fig. 12. The dot-dash-line shown in Fig 12 is expressed by the following experimental equation.

\[ q_{pool} = 68 \Delta T_{sat}^{1.5} \]  

(12)

where \( q_{pool} \) is in kcal/m²·hr and \( \Delta T_{sat} \) in °C. The experimental data plotted in Fig. 12 show that the burn-out heat flux obtained at a stagnation point is higher than that obtained in a uniform parallel flow zone. This tendency seems to be proved by the fact that the effect of blowing off the vapor bubble by the impinging jet at the stagnation point is stronger than in a uniform parallel flow zone. Figure 13 shows the data of Fig. 12 plotted with \( (q_t - q_{so}) \) and \( \Delta T_{sat} \). Here \( q_t \) was measured heat flux and \( q_{so} \) was calculated by Eq. (3) and Eq. (8) according to the definite Eq. (4) of heat transfer coefficient. The solid line shown in Fig. 13 represents Eq. (12) and a broken line represents the nucleate boiling curve obtained by Katto and Kunihiro6 and a dot-dash-line represents Nishikawa-Yamagata’s11 correlation equation for nucleate boiling heat transfer in pool boiling. The approximation of the superposition evaluated by Eq. (11) holds fairly good though there is some scattering. It seems to be an important fact that the heat flux \( (q_t - q_{so}) \) associated with bubble motion alone extends to \( 3 \times 10^7 \) kcal/m²·hr and appears on a linear extension of the pool boiling curve. The cause or mechanism by which such a high heat flux was obtained may be explained by the fact that the transport of latent heat is reinforced by the impinging water jet, that is, the subcooled water is supplied continually on the heated surface and coalescent bubbles are blown off immediately by the impinging jet. Since there exists a critical value of heat flux according to velocity and subcooling, the boiling curve cannot help departing from the nucleate boiling curve above the burn-out heat flux.

**Conclusion**

1) Heat transfer coefficient measured at the stagnation point was about 1.8 times the theoretical value for the laminar boundary layer.

2) For the turbulent boundary layer, there is no effect of an unheated starting length on heat transfer in the range very near the heated starting point.

3) Approximation of the superposition, that is, the total heat flux at forced convective boiling, consisting of pool boiling heat flux and pure convective heat flux, held fairly good up to the burn-out heat flux region.

4) The heat flux associated with bubble motion alone extended to \( 3 \times 10^7 \) kcal/m²·hr and appeared on a linear extension of the pool boiling curve.

**Nomenclature**

- \( a \) = dimensionless velocity gradient
- \( B \) = jet nozzle width [m]
- \( c \) = velocity gradient [1/sec]
- \( Gr \) = Grashof number [-]
- \( L \) = half of the heated length [m]
\( l \) = distance from center of the heated surface [m]

\( Nu \) = Nusselt number [—]

\( P \) = static pressure [kgf/cm²]

\( P_s \) = system pressure [kgf/cm²]

\( P_w \) = atmospheric pressure [kgf/cm²]

\( Pr \) = Prandtl number [—]

\( q \) = heat flux [kcal/m²·hr]

\( Re \) = Reynolds number [—]

\( T_f \) = film temperature [°C]

\( T_0 \) = back-side temperature of the heated plate [°C]

\( T_	ext{sat} \) = saturated temperature [°C]

\( T_	ext{w} \) = heated surface temperature [°C]

\( dT_	ext{sat} \) = superheat temperature [°C]

\( dT_	ext{sub} \) = subcooling [°C]

\( U_{\text{max}} \) = velocity outer edge of the boundary layer [m/sec]

\( u_0 \) = jet nozzle outlet velocity [m/sec]

\( X \) = local distance from the stagnation point [m]

\( X_0 \) = unheated starting length [m]

\( \alpha \) = heat transfer coefficient [kcal/m²·hr·°C]

\( \delta \) = thickness of the foil [m]

\( \lambda \) = thermal conductivity [kcal/m·hr·°C]

\( \nu \) = kinematic viscosity [m²/sec]

\( th \) = theory

\( u \) = location of 25 mm from the stagnation point

\( x \) = local

\(<\text{Subscripts}>\)

boil = boiling

co = convection

i = incipient boiling

l = liquid

pool = pool boiling

s = stagnation point

t = total

Literature Cited


