Effect of inclination on film boiling

Shih-Chang Lin

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EFFECT OF INCLINATION ON FILM BOILING

by

SHIH-CHANG LIN, 1942-

A

THESIS

submitted to the faculty of

UNIVERSITY OF MISSOURI - ROLLA

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Rolla, Missouri

1970

Approved by

Harry Shanes
(advisor)

Ralph E. Morrison

D. J. Dagano
ABSTRACT

The object of this investigation was to determine the effect of inclination on the heat transfer for film pool boiling. The test plate used as the heater was made of Inconel-600 and heated electrically. Liquid nitrogen at atmospheric pressure was used as the test fluid. The heater was oriented at various angles from the horizontal position with the heater surface facing both upward and downward and the heat transfer coefficient was determined for each angle over temperature differences ranging from 450°F to 750°F.

As the heater angle was increased from 0° (horizontal) to 90° (vertical), the heat transfer coefficient increased both for the heater surface facing upward and for the surface facing downward but the magnitude of the change was different.
ACKNOWLEDGEMENTS

The professional advice and leadership of my advisor, Dr. Harry J. Sauer, Jr., is gratefully acknowledged and sincerely appreciated.

The help of Richard D. Smith in the Mechanical Engineering Laboratory is appreciated.
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<td>Area</td>
<td>ft²</td>
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<tr>
<td>I</td>
<td>Current</td>
<td>amperes</td>
</tr>
<tr>
<td>K</td>
<td>Thermal conductivity</td>
<td>BTU/hr ft °F</td>
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<tr>
<td>Q</td>
<td>Heat flow rate</td>
<td>BTU/hr</td>
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<tr>
<td>ΔT</td>
<td>Temperature difference between heater and saturated liquid</td>
<td>°F</td>
</tr>
<tr>
<td>ΔTₜ</td>
<td>Temperature drop in transite backing</td>
<td>°F</td>
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<td>Voltage drop across test plate</td>
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<tr>
<td>g</td>
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</tr>
<tr>
<td>gₗ</td>
<td>Gravitational constant</td>
</tr>
<tr>
<td>h</td>
<td>Heat transfer coefficient</td>
</tr>
<tr>
<td>h'ₗ</td>
<td>Average enthalpy difference between vapor and liquid</td>
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<td>Viscosity</td>
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<td>( \rho )</td>
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<td>---</td>
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<tr>
<td>( l )</td>
<td>Liquid</td>
<td>---</td>
</tr>
<tr>
<td>( t )</td>
<td>Transite</td>
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<tr>
<td>( v )</td>
<td>Vapor</td>
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I. INTRODUCTION

Boiling heat transfer is a mode of heat transfer that occurs with a change in phase from liquid to vapor. Pool boiling is boiling on a heated surface in a pool of initially stagnant liquid.

Investigations of the mechanism of boiling have established the existence of four distinct regimes in which the boiling possesses different characteristics. These are termed the convective, nucleate, transition, and stable film boiling regimes. These regimes may be further subdivided as shown in Fig. 1.

In region I the heat transfer from the heater to the liquid takes place by conduction and single phase natural convection which maintains upward flow of superheated liquid and vapor is produced by evaporation at the free surface.

As the heater surface temperature is increased into region II, bubbles of vapor begin to form and condense before reaching the free surface while rising from active sites on the heating surface. This region is referred to as the individual bubble region which is in the nucleate region. As the heater surface temperature is increased into region III, nucleate boiling continues to occur with the bubbles rising to the free surface in continuous columns. This region is referred to as the
Figure 1. Typical boiling curve.
As the heater surface temperature increases, point A, the peak of curve is reached. This peak is called the burnout point, the critical excess temperature point. Beyond the peak of curve (region IV) an unstable film of vapor forms on the heater surface, and large bubbles are formed at the outer upper surface of the film. This vapor film is not stable, and collapses and reforms rapidly. The presence of this film provides additional resistance to heat transfer and reduces the heat-transfer rate.

As the temperature is increased into region V, point B is reached where the heat flux is a minimum in film boiling. This point is commonly called the Leidenfrost point. In region V a stable vapor film is formed on the heater surface which is blanketed with an insulating film of vapor and the heat-transfer rate is quite low.

By further increasing the heater surface temperature, the heat-transfer rate also is increased by thermal radiation from the heater surface and radiation becomes significant. However, too high a temperature would damage the heater. Hence, for practical purpose, the temperature is limited by the material properties.

This investigation was originated to provide the necessary data for evaluating the effect of surface orientation on boiling heat transfer in the stable film regime.
Nukiyama\textsuperscript{1} in 1934 found that at least two, and possibly three, distinct regions of boiling existed. Nukiyama submerged an electrically heated wire in a pool of saturated liquid water and measured the temperature of the wire as a function of the heat flux. The results of this experiment, summarized in Fig. 2, have great practical importance.

Nukiyama suggested that in addition to the two boiling regions represented by curves AB and CD, the boiling curve might be continuous between point B and D. If this was true, the curve connecting B and D would have the surprising characteristic that increasing the temperature difference would cause a decrease in the heat flux.

Farber and Scorah\textsuperscript{2} verified the above suggestion when they obtained the complete characteristic boiling curve as typified by Fig. 1. By carefully controlling their experiment they found it possible to obtain data in region IV, in spite of the fact that due to the negative slope, operation is inherently unstable in experiments where the heat flux is the controlled parameter. The general shape of the boiling curve is the same for all fluids at all pressures.

The first investigator to suggest a method of predicting heat transfer coefficients for film boiling was
Figure 2. Nukiyama's experimental results.
Bromley. He offered the following equation for the heat transfer coefficient in film pool boiling from a horizontal tube.

$$h = 0.62 \left( \frac{K_v h_f g \rho_v g (\rho_l - \rho_v)}{\mu_f \Delta T D} \right)^{1/4}$$ (1)

The coefficient 0.62 is empirical. The theoretical value is 0.512 for stagnant liquid around the vapor, and 0.724 for liquid moving with the same velocity as the vapor. The average, rounded-off, of these two numbers is 0.62.

Berenson has made many contributions to the area of boiling heat transfer. He concluded that the burnout heat flux and the film boiling curve are independent of surface material, cleanliness, and roughness provided that the roughness height is less than the film thickness. Berenson also concluded that transition boiling is a combination of unstable nucleate and unstable film boiling alternately at a given location on the heating surface. He derived the following analytical expression for the heat transfer coefficient in film pool boiling from a horizontal surface.

$$h = 0.425 \left( \frac{K_v h_f g \rho_v g (\rho_l - \rho_v)}{\mu_f \Delta T D} \right)^{1/4}$$ (2)
where \( E = \left| \frac{g_c \sigma}{g (\rho_1 - \rho_v)} \right|^{1/2} \)

To compare the above result with that of Bromley, the major difference is the substitution of \( E \) for the tube diameter \( D \). These are the geometrical scale factors for horizontal plates and tubes, respectively. The similarity between Eq. (1) and Eq. (2) provided added confidence in the validity of Eq. (2) since Bromley's equation has been thoroughly verified.

Chang\(^6\) was the first to point out that a standing wave existed over a plane surface in film boiling. In a subsequent paper, Chang\(^7\) used a wave approach and derived Eq. (3) for the heat transfer coefficient in film boiling from horizontal surfaces.

\[
h = \left| \frac{K_v^3 (\rho_1 - \rho_v) g}{8 \pi^2 \mu_v \alpha_c} \right|^{1/3}
\]

where \( \alpha_c = \frac{K_v \Delta T}{2 h_{fg} \rho_v} \)

\( \alpha_c \) is called an equivalent thermal diffusivity and may be thought of as the diffusivity of the vapor-liquid interface as a result of phase change.
Chang concluded that the effect of any variable might be calculated from its effect on the physical properties of the liquid and its vapor. An increase in pressure would increase the heat transfer coefficient.

Class, et. al.,\(^8\) have presented data for both film and nucleate boiling using electrically heated plates under a variety of conditions. These conditions include the angle of orientation of the surface, surface condition, and pressure. As was expected, at the higher pressures the boiling curves shifted to the left, causing higher heat fluxes at lower differential temperatures. That is, an increase in pressure will increase the heat transfer coefficient. The authors also pointed out that there was not much difference between the vertical, 45°, and the horizontal surfaces in the nucleate region. In the film region a shift to the right was always observed as the surface was rotated from the vertical to the horizontal, thus requiring a greater temperature difference for a given heat flux.

Hosler and Westwater\(^9\) have investigated film boiling from a flat plate with the objective of determining the actual validity of different theories. Film boiling was studied for water and Freon-11 at atmospheric pressure on a flat horizontal aluminum heating surface. High speed motion pictures were taken to support hydrodynamic calculations for both fluids. Hosler and Westwater conclude
that the method of Chang for predicting the film boiling curve is not reliable. Also, the method of Berenson for predicting the film boiling curve is good, but his predictions of temperature difference for the minimum flux are not reliable.

Brentari and Smith\textsuperscript{10} have made a significant contribution to the literature with their paper on correlation of pool boiling data for cryogenic fluids. The authors have a section devoted to the discussion of boiling variables. Nucleate boiling is generally regarded as insensitive to system geometry. As mentioned previously, the work of Class, et. al.,\textsuperscript{8} showed significant variation when changing from horizontal to vertical orientation with no other change in the system. The surface orientation with respect to an external force field (gravity) may have a major effect. For example, there exists marked reduction in film boiling fluxes for horizontal surfaces facing downward, where the influence of vapor removal is significant.

Kutateladze\textsuperscript{11} reports that electrically heated surfaces have slightly different heat transfer characteristics than those heated by vapor condensation, probably because condensation droplets cause surface temperature differences.

Flynn, et. al.,\textsuperscript{12} presented the complete curve representing boiling heat transfer in liquid nitrogen.
throughout the nucleate, transition, and film boiling region. The curve had been determined on a single surface. The authors state that in film boiling region, the problem of selecting the proper temperature for fluid properties becomes more acute due to the larger temperature gradients.

Ragsdell\textsuperscript{13} concluded that the heated surface material did not affect the stable film boiling region and presented the following modified Berenson equation to predict the film boiling heat transfer coefficient for horizontal heated surfaces.

$$h = 0.512 \left( \frac{K_{v} h_{fg} \rho_{v} g (\rho_{l} - \rho_{v})}{\mu_{f} \Delta T \sqrt{g_{c} - \sigma}} \right)^{1/4}$$

Price\textsuperscript{14} obtained experimental data for film pool boiling from flat plate heaters oriented at various angles from the horizontal position. The author concluded that as the angle of inclination was increased from 0° (horizontal) to 90° (vertical) a uniform increase in the heat transfer coefficient was observed. All experiments conducted with the heater surface facing upward.

Boiling heat transfer has been the subject of intensive studies for years. There are many other
researchers who have made significant contributions to this field.
III. EXPERIMENTAL INVESTIGATION

A. Method

The test plate was placed in a pool of liquid nitrogen and heated electrically. Thermocouples were welded to the unwetted side of the plate to measure the heater temperature and the same side was cemented to a transite block which served to insulate the back side of the heater. Power was supplied to the test plate from a single phase alternating current welder and was controlled by using the welder control and carbon-pile rheostats. An AC voltmeter was used to measure the voltage drop across the test plate. Current flow through the test plate was measured using an ammeter. The plate temperatures and the bulk temperature of the fluid were recorded on a multichannel recording potentiometer. A digital millivoltmeter was used for visual observation of the thermocouple outputs.

B. Equipment

The main components of the boiling apparatus are the console in which the instrumentation is mounted, the dewar containing the heater and test fluid, and the AC welder which was used as the power supply. Fig. 3 gives an overall view of the experimental set-up. A schematic of the apparatus is shown in Fig. 4. This schematic
Figure 3. Experimental set-up.
Figure 4. Schematic of experimental set-up.
serves as a listing of the equipment and shows the relationship of the components. More detailed descriptions of the test sections, power supply and control, and instrumentation are presented in the following sections.

C. **Test Sections**

The test sections used in this investigation were all made of Inconel-600, 0.005 inch thick and 1 inch in width. The effective length of the test plate was 3.5 inches when installed in position between the conduction bars. The test plate material was cut to a length of approximately four inches. This allowed one-half inch of extra material on each end of the test plate to connect the test plate to the conduction bars. A test plate assembly is shown in Fig. 5.

Each test plate was etched on one side using a solution of marble's reagent. This etching was done to roughen the surface of the test plate to improve the bond between the test plate and the epoxy.

After the test plates were etched, six 30 gauge chromel-alumel thermocouples were spotwelded to the etched surface by a solid state miniature spot welder. A piece of 7/16" thick transite insulation material grooved out to accommodate the thermocouples on the back side of the test plate was cemented to the test plate using an epoxy adhesive. Using this method, the
Figure 5. Heater assembly.
heater was constructed that would deliver heat to a boiling liquid from essentially one side. There was, of course, heat loss through the insulated side. To account for this, three thermocouples were cemented to the back side of the transite block and the backing temperature recorded. The reference junctions of all thermocouples were placed in a liquid nitrogen bath.

D. Power Supply and Control

The power was supplied to the heater through welding cables which carried the power to the conduction bars holding the heater. The power input to the test section was controlled using both the welder settings and two carbon-pile rheostats in series with the test plate.

E. Instrumentation

The voltage drop across the heater was measured with a Honeywell Model 333 Digital Multimeter capable of measuring voltages to four significant figures. The current through the heater was measured with G.E. Type P3 AC ammeter. The thermocouples associated with the heater plus the bulk temperature thermocouple were recorded on a Honeywell Electronik-16 potentiometric multi-channel recorder and observed on a Digitec Model 454 DC millivoltmeter.
F. Test Procedure

All instruments were turned on to allow a sufficient warm-up prior to each test. The heater was connected to the conduction bars and adjusted to the desired angle. The heater assembly was then placed in the dewar. Liquid nitrogen was poured slowly into the dewar and the thermocouple bath. The level of nitrogen was kept about four inches above the heater. The welder power was increased in small increments until the system went into stable film boiling. After equilibrium was reached, all readings were taken, and the power was increased to the next desired level. After the maximum desired temperature or maximum power level had been reached, the power was decreased in small increments, and a data set obtained at each stable point until the minimum point was reached. The welder was shut down and the heater was inspected for separation. If no separation occurred, the process was repeated.

The following items were taken at each data point: the six heater temperatures, the bulk temperature, the backing temperatures, and the current through and voltage drop across the heater.

G. Data Reduction

The average heater temperature, $T_H$, was determined by taking the average millivolt readings for each run and converting them into temperatures using the
National Bureau of Standards Circular No. 561. The millivolt readings for the backing temperatures were converted to temperatures in the same way as described above.

The power supplied to the heater was determined in the following manner:

\[
(\frac{Q}{A})_{\text{supplied}} = \frac{I \Delta V (3.4129)}{A}
\]

The heat loss through the backing was calculated using

\[
(\frac{Q}{A})_{\text{loss}} = \frac{K_t \Delta T_t}{\Delta X}
\]

where \(K_t = 0.48\) (BTU/hr ft °F). The net heat flux, which caused the boiling to occur, was obtained using

\[
(\frac{Q}{A})_{\text{net}} = (\frac{Q}{A})_{\text{supplied}} - (\frac{Q}{A})_{\text{loss}}
\]

The heat transfer coefficient was obtained using

\[
h = \frac{(\frac{Q}{A})_{\text{net}}}{\Delta T}
\]
IV. RESULTS

Tests were conducted with the heater surface oriented at 0°, 15°, 30°, and 45° from the horizontal with the heater surface facing downward and at 0°, 30°, and 60° from the horizontal with the heater surface facing upward, as well as with the heater surface at 90° (vertical).

From here on, "down" represents the heater surface facing downward, and "up" the heater surface facing upward. For example, "30° up" means a heater angle of 30° from the horizontal with the heater surface facing upward.

Figures 6, 7, 8, 9, 10, 11, 12, and 13 show the results of heat flux versus temperature difference for heater angles of 0° down, 15° down, 30° down, 45° down, 90°, 0° up, 30° up, and 60° up, respectively. The experimental data points are shown with the best polynomial fit curve drawn through them. The average and maximum deviations for each data set are shown on each figure.

Fig. 14 shows the results in terms of heat flux versus temperature difference for heater angles of 0°, 15°, 30°, and 45° with the heater surface facing downward and for 90°.
Figure 6. Heat flux for heater angle of 0° down.

Heater angle = 0° down

Best fit curve
Average deviation = 1.30%
Maximum deviation = 3.58%
Figure 7. Heat flux for heater angle of 15° down.
Heater angle = 30° down

Best fit curve

Average deviation = 0.60%
Maximum deviation = 1.28%

Figure 8. Heat flux for heater angle of 30° down.
Figure 9. Heat flux for heater angle of 45° down.
Heater angle = 90° (vertical)

Average deviation = 0.42%
Maximum deviation = 1.37%

Figure 10. Heat flux for the heater angle of 90° (vertical).
Figure 11. Heat flux for heater angle of 0° up.

- Best fit curve
- Average deviation = 0.60%
- Maximum deviation = 1.49%
Figure 12. Heat flux for heater angle of $30^\circ$ up.

Heater angle = $30^\circ$ up

Best fit curve
Average deviation = 0.65%
Maximum deviation = 1.37%
Heater angle = 60° up

Average deviation = 1.25%
Maximum deviation = -2.27%

Figure 13. Heat flux for heater angle of 60° up.
Figure 14. Heat Flux for heater surface facing downward.
Figure 15 shows the results in terms of heat flux versus temperature difference for heater angles of 0°, 30° and 60° with the heater surface facing upward and for 90°.

The curve indicates that as the heater angle is increased from the horizontal to the vertical position the heat flux necessary to maintain any given temperature difference is also increased for the heater surface facing either downward or upward.

Figures 16, 17, 18, 19, 20, 21, 22, and 23 are the results for heat transfer coefficient variation with temperature differences for heater angles of 0° down, 15° down, 30° down, 45° down, 90°, 0° up, 30° up, and 60° up, respectively. The experimental data points are shown with the best polynomial fit curve drawn through them.

Figure 24 shows the results of heat transfer coefficient versus temperature difference for heater angles of 0°, 15°, 30°, and 45° with the heater surface facing downward and for 90°. As the heater angle is increased from 0° to 45°, for any given temperature difference, the heat transfer coefficient is increased approximately 5 BTU/hr sq ft °F by changing the heater angle 15°. As the heater angle is increased from 45° to 90°, the heat transfer coefficient is increased approximately 6 BTU/hr sq ft °F only.
Figure 15. Heat flux for heater surface facing upward.
Figure 16. Heat transfer coefficient for heater angle of 0° down.
Figure 17. Heat transfer coefficient for heater angle of 15° down.

Heater angle = 15° down

Best fit curve
Average deviation = 0.61%
Maximum deviation = -1.67%

TEMPERATURE DIFFERENCE, °F

HEAT TRANSFER COEFFICIENT
BTU/HR SQ FT °F
Heater angle = 30° down

Average deviation = 0.59%
Maximum deviation = 1.27%

Figure 18. Heat Transfer coefficient for heater angle of 30° down.
Heater angle = 45° down

Best fit curve

Average deviation = 1.69%
Maximum deviation = 3.67%

Figure 19. Heat transfer coefficient for heater angle of 45° down.
Figure 20. Heat transfer coefficient for heater angle of 90° (vertical).

Heater angle = 90° (vertical)

Best fit curve

Average deviation = 0.42%

Maximum deviation = 1.31%
Heater angle = 0° up

Best fit curve

Average deviation = 0.60%
Maximum deviation = -1.51%

Figure 21. Heat transfer coefficient for heater angle of 0° up.
Heater angle = 30° up

Best fit curve

Average deviation = 0.63%
Maximum deviation = 1.11%

Figure 22. Heat transfer coefficient for heater angle of 30° up.
Heater angle = 60° up

Average deviation = 1.25%
Maximum deviation = -2.26%

Figure 23. Heat transfer coefficient for heater angle of 60° up.
Figure 24. Heat transfer coefficient for heater surface facing downward.
Figure 25 shows the results of heat transfer coefficient versus temperature difference for heater angles of 0°, 30° and 60° with the heater surface facing upward and for 90°. Figure 25 indicates that the heat transfer coefficient is increased uniformly with angle for an increase of approximately 4 BTU/hr sq ft °F for each increase of angle of 30°.

Figure 26 shows the results of heat transfer coefficient versus heater angle for the given temperature difference of 550°F. As the heater angle is increased the curve of the downward-facing heater surface is approaching the curve of the upward-facing heater surface and two curves approximately become one when the heater angle is over 45°.
Figure 25. Heat transfer coefficient for heater surface facing upward.
Figure 26. Heat transfer coefficient for temperature difference 550°F.
V. ESTIMATED ACCURACY

The average heater temperature used was the simple arithmetic average of the six thermocouples attached to the heater. Because of the relatively large temperature variation of as much as 50°F with position on the heater, \( \pm 5\% \) error might be introduced when calculating the temperature difference between the heater and fluid.

In calculating the heat loss through the backing material, the heater surface was assumed infinite with no edge effects taken into consideration. In order to determine the magnitude of this error, the temperature of the edge surface was assumed to be the same as the backing temperature. Actually, the former should be much higher than the latter. The heat transfer rate through the backing, considering two-dimensional flow, was obtained for several conditions using numerical methods. This analysis indicated that the error introduced by neglecting edge effects was less than 6%.

Additional error might have been introduced due to inaccuracies in the instrumentation for measuring current and voltage; however, it is considered small compared to those factors mentioned above.

Considering the sources of error mentioned above, the maximum predicted error in the heat transfer coefficient is 12%.
VI. CONCLUSIONS

1. For film boiling on a horizontal plate, the heat transfer coefficient of the heater surface facing upward is larger than that of the heater surface facing downward for heater angles less than 45°.

2. For the heater surface facing upward, the heat transfer coefficient appears to have approximately uniform increase as the heater angle is increased from 0° to 90°.

3. For the heater surface facing downward, the heat transfer coefficient increases very rapidly as the heater angle is increased from 0° to 45°.

4. The heat transfer coefficient of the heater surface facing downward is approximately the same as that of the heater surface facing upward if the heater angle is over 45°.
VII. BIBLIOGRAPHY


Shih-Chang Lin was born on June 26, 1942, in Taipei, Taiwan, China. He received a Bachelor of Science degree in Mechanical Engineering from National Taiwan University in June 1965.

He served in the Chinese Air Force from July 1965 to July 1966, then worked as a Mechanical Engineer for San Chin Machinery Co. from July 1966 to July 1967, and for Taiwan Electronic Co. from July 1967 to July 1968.

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