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Effect of inclination on film boiling

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EFFECT OF INCLINATION ON FILM BOILING

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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 co-efficient increased both for the heater surface facing upward and for the surface facing downward but the magnitude of the change was different.

ACKNOWLEDGEMENTS

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LIST OF ILLUSTRATIONS

NOMENCLATURE

SYMBOL	SIGNIFICANCE	UNITS
CAPITAL LETTERS		
A	Area	ft^2
I	Current	amperes
K	Thermal conductivity	BTU/hr ft °F
Q	Heat flow rate	BTU/hr
ΔT	Temperature difference between heater and saturated liquid	°F
ΔT_t	Temperature drop in transite backing	°F
ΔV	Voltage drop across test plate	volts
ΔX	Thickness of backing material	ft
LOWER CASE LETTERS		
g	Acceleration due to gravity	ft/sec^2
g_c	Gravitational constant	$lb_m\ ft/lb_f\ sec^2$
h	Heat transfer coefficient	$BTU/hr\ ft^2\ °F$
h'_{fg}	Average enthalpy difference between vapor and liquid	BTU/lb_m
h_{fg}	Latent heat of vaporization	BTU/lb_m

SYMBOL	SIGNIFICANCE	UNITS
GREEK LETTERS		
α_c	Equivalent thermal diffusivity	ft^2/hr
μ	Viscosity	$lb_f\ hr/ft^2$
ρ	Density	lb_m/ft^3
σ	Surface tension	lb_f/ft
SUBSCRIPTS		
f	Film	---
l	Liquid	---
t	Transite	---
v	Vapor	---

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 super-heated 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
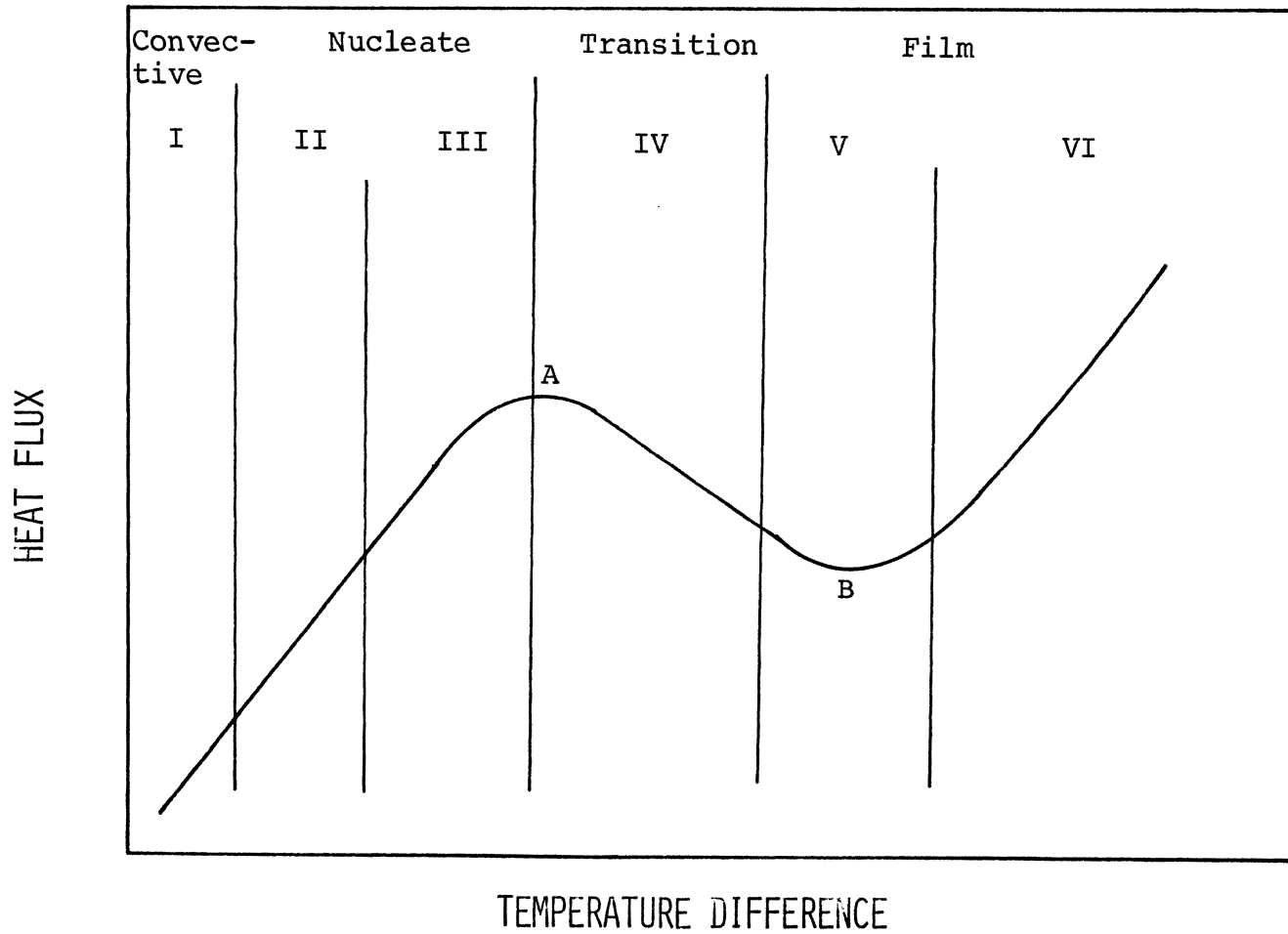
Figure 1. Typical boiling curve.

continuous column region.

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.

II. LITERATURE REVIEW

Nukiyama[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[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.

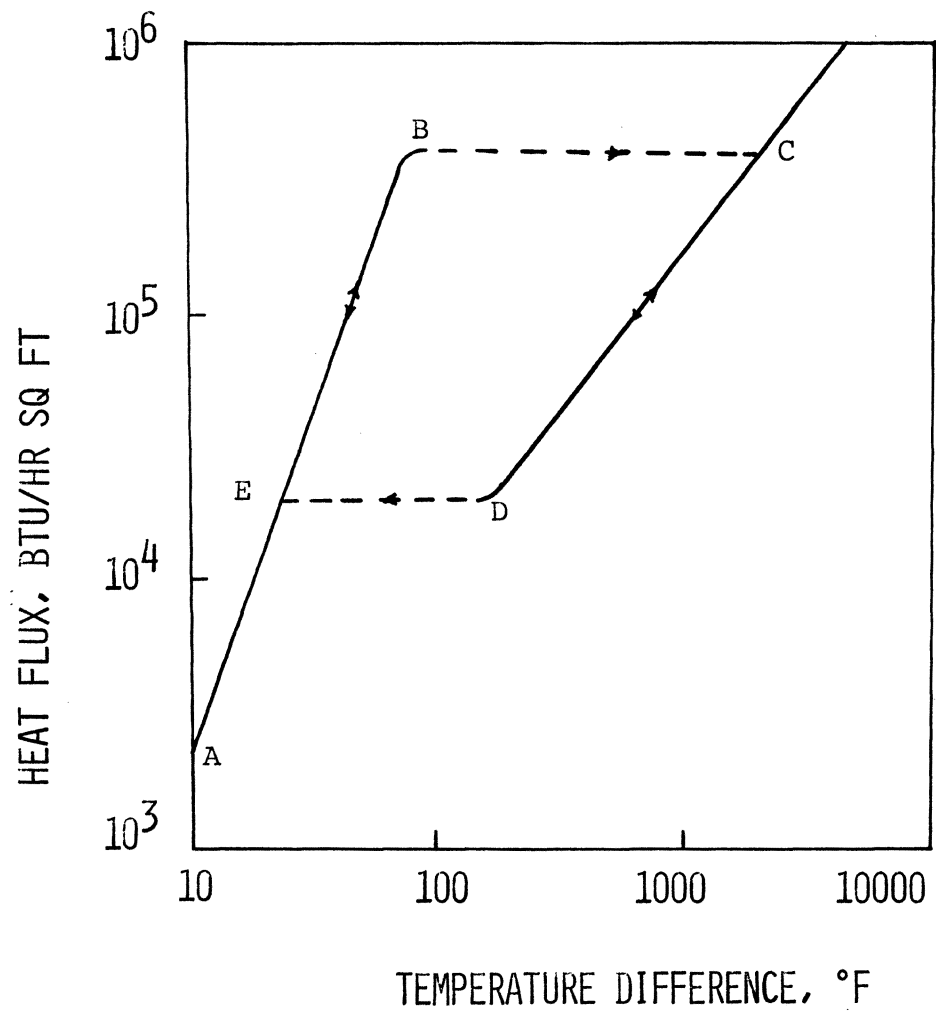The first investigator to suggest a method of predicting heat transfer coefficients for film boiling was

Figure 2. Nukiyama's experimental results.

Bromley.[3] He offered the following equation for the heat transfer coefficient in film pool boiling from a horizontal tube.

$$h = .62 \left| \frac{K_{vf}^3 \; h_{fg}' \; \rho_{vf} \; g \; (\rho_1 - \rho_v)}{\mu_f \; \Delta T \; D} \right|^{\frac{1}{4}} \tag{1}$$

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[4,5] 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.

$$h = .425 \left| \frac{K_{vf}^3 \; h_{fg}' \; \rho_{vf} \; g \; (\rho_1 - \rho_v)}{\mu_f \; \Delta T \; E} \right|^{\frac{1}{4}} \tag{2}$$

where $E = \left| \dfrac{g_c\, \sigma}{g\,(\rho_1 - \rho_v)} \right|^{\frac{1}{2}}$

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.

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

where $\alpha_c = \dfrac{K_v\,\Delta T}{2\,h_{fg}\,\rho_v}$

α_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[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.,[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[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.,[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 sur-
face. 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[13] concluded that the heated surface mater-
ial 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.

$$h = .512 \left| \frac{K_{vf} \, h'_{fg} \, \rho_{vf} \, g \, (\rho_1 - \rho_v)}{\mu_f \, \Delta T \, \sqrt{\dfrac{g_c \, \sigma}{g \, (\rho_1 - \rho_v)}}} \right|^{\frac{1}{4}}$$

Price[14] obtained experimental data for film pool
boiling from flat plate heaters oriented at various
angles from the horizontal position. The author con-
cluded 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 in-
tensive 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 nitro-
gen 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 con-
trolled 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
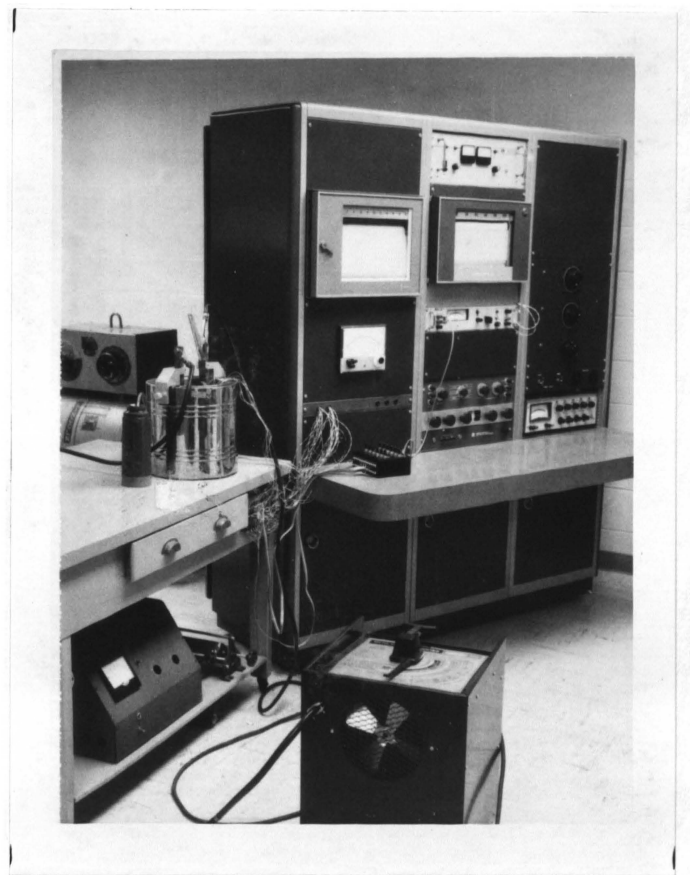of the apparatus is shown in Fig. 4. This schematic
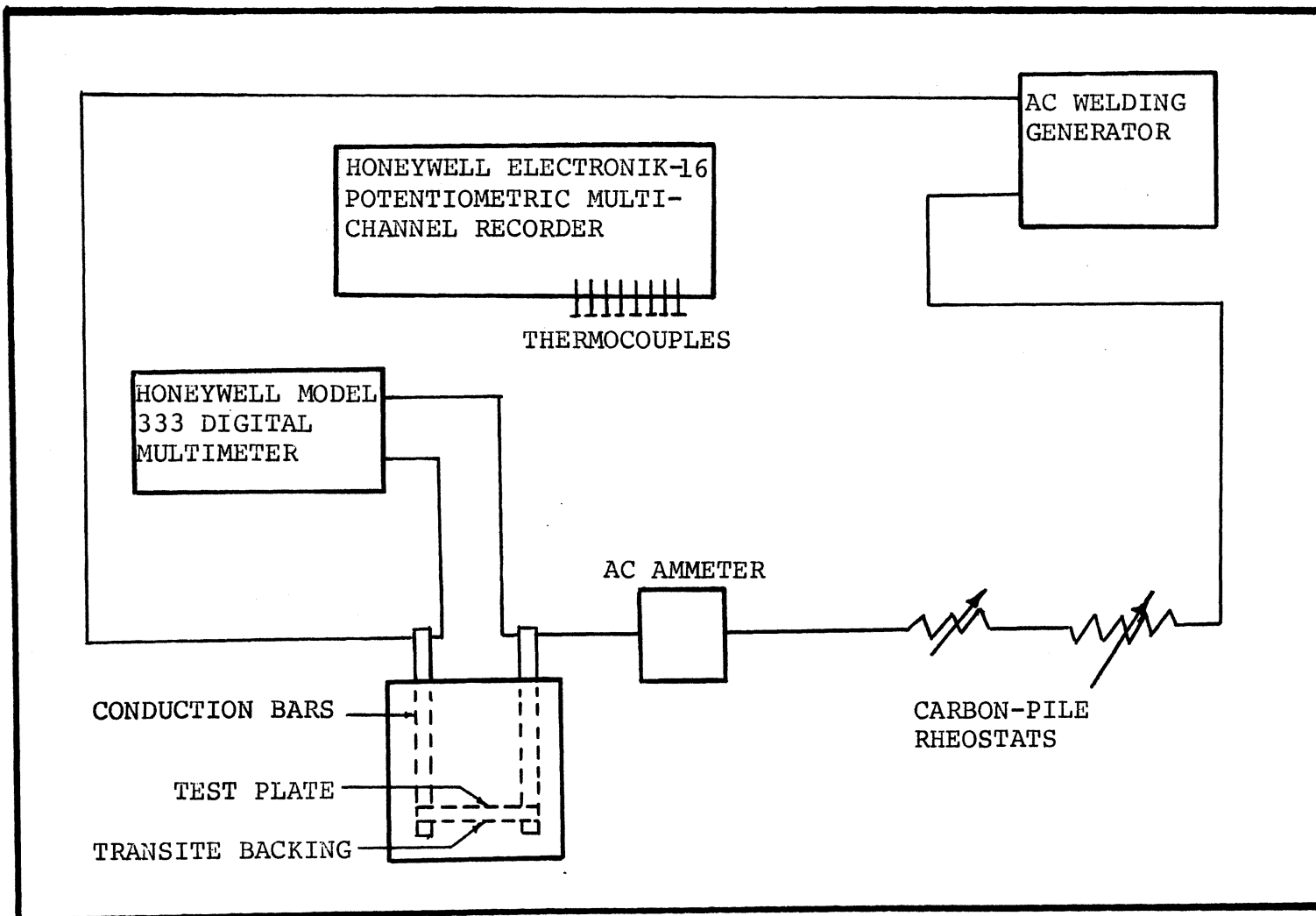
Figure 3.   Experimental set-up.

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
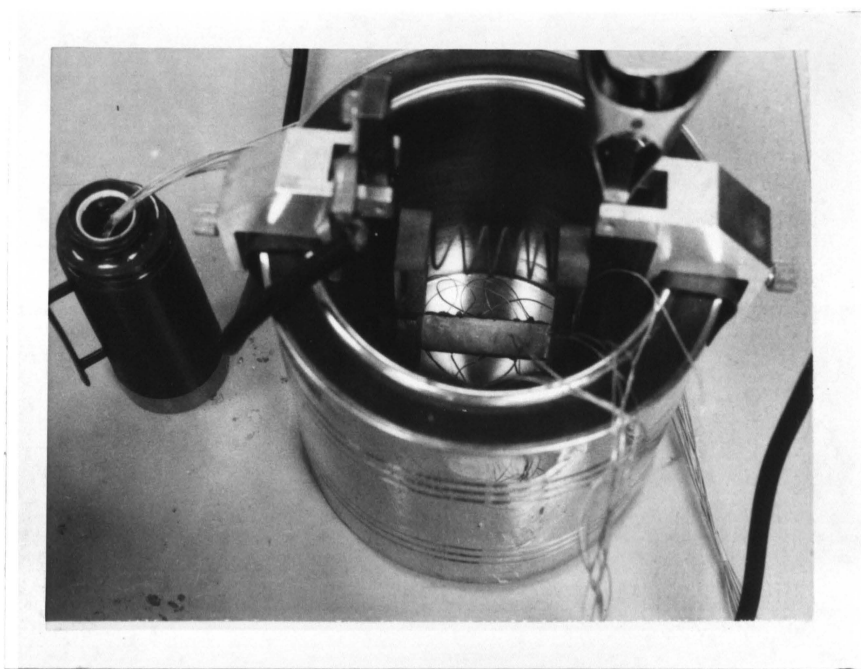
Figure 5.   Heater assembly.

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 tempera-
ture recorded. The reference junctions of all thermo-
couples 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 re-
corded 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:

$$(Q/A)_{supplied} = \frac{I \; \Delta V \; (3.4129)}{A}$$

The heat loss through the backing was calculated using

$$(Q/A)_{loss} = \frac{K_t \; \Delta T_t}{\Delta X}$$

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

$$(Q/A)_{net} = (Q/A)_{supplied} - (Q/A)_{loss}$$

The heat transfer coefficient was obtained using

$$h = \frac{(Q/A)_{net}}{\Delta T}$$

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°.
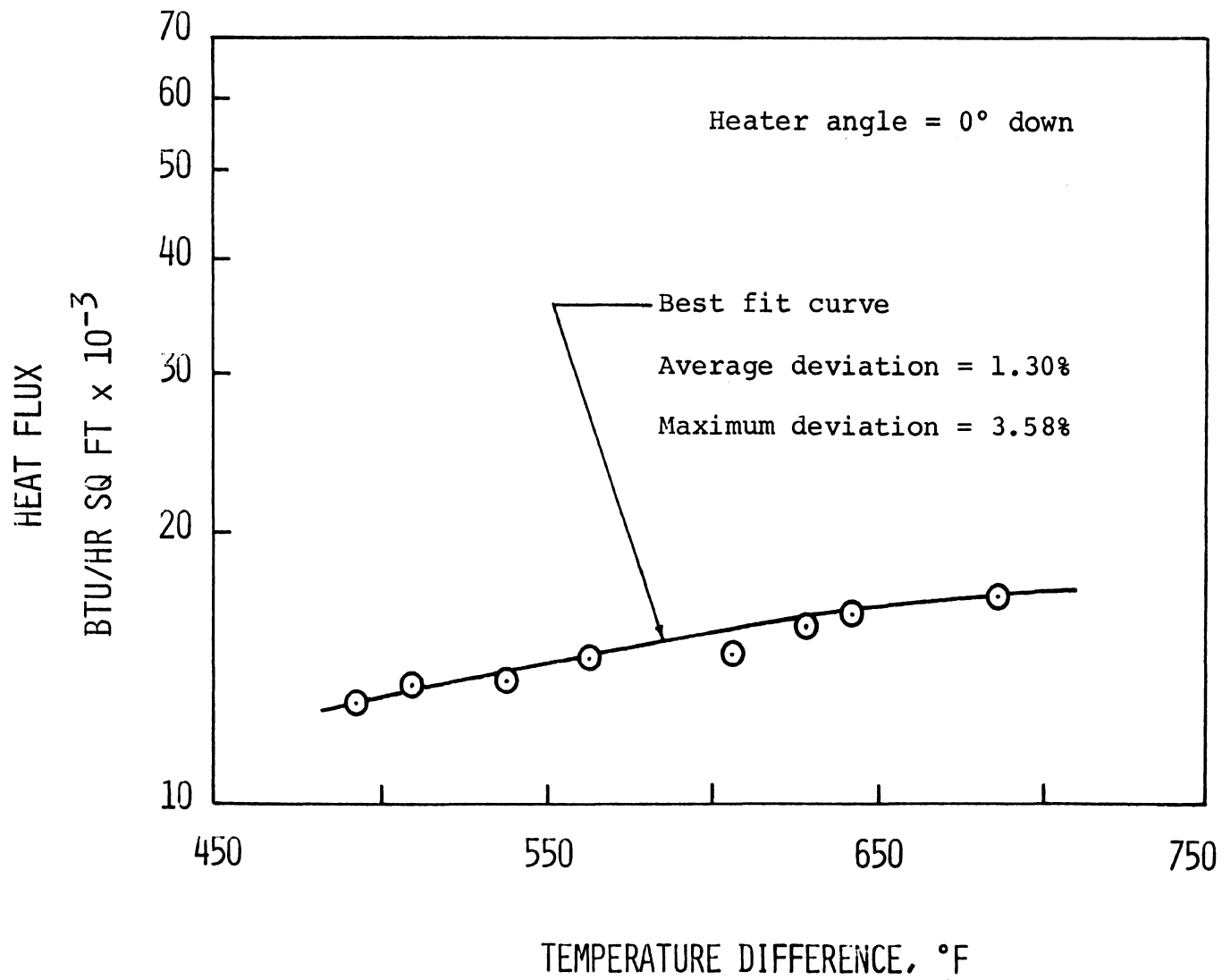
Figure 6. Heat flux for heater angle of 0° down.
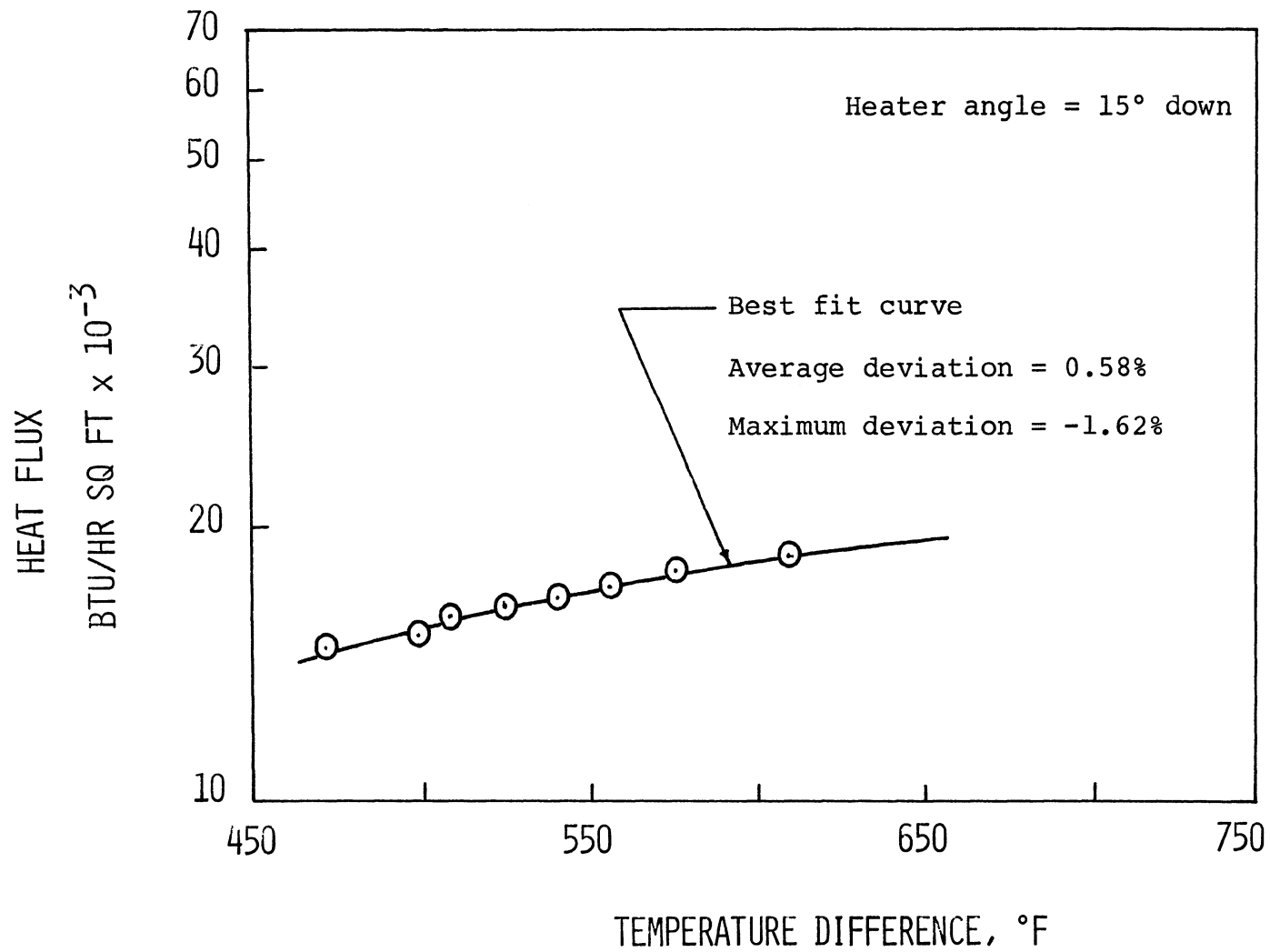
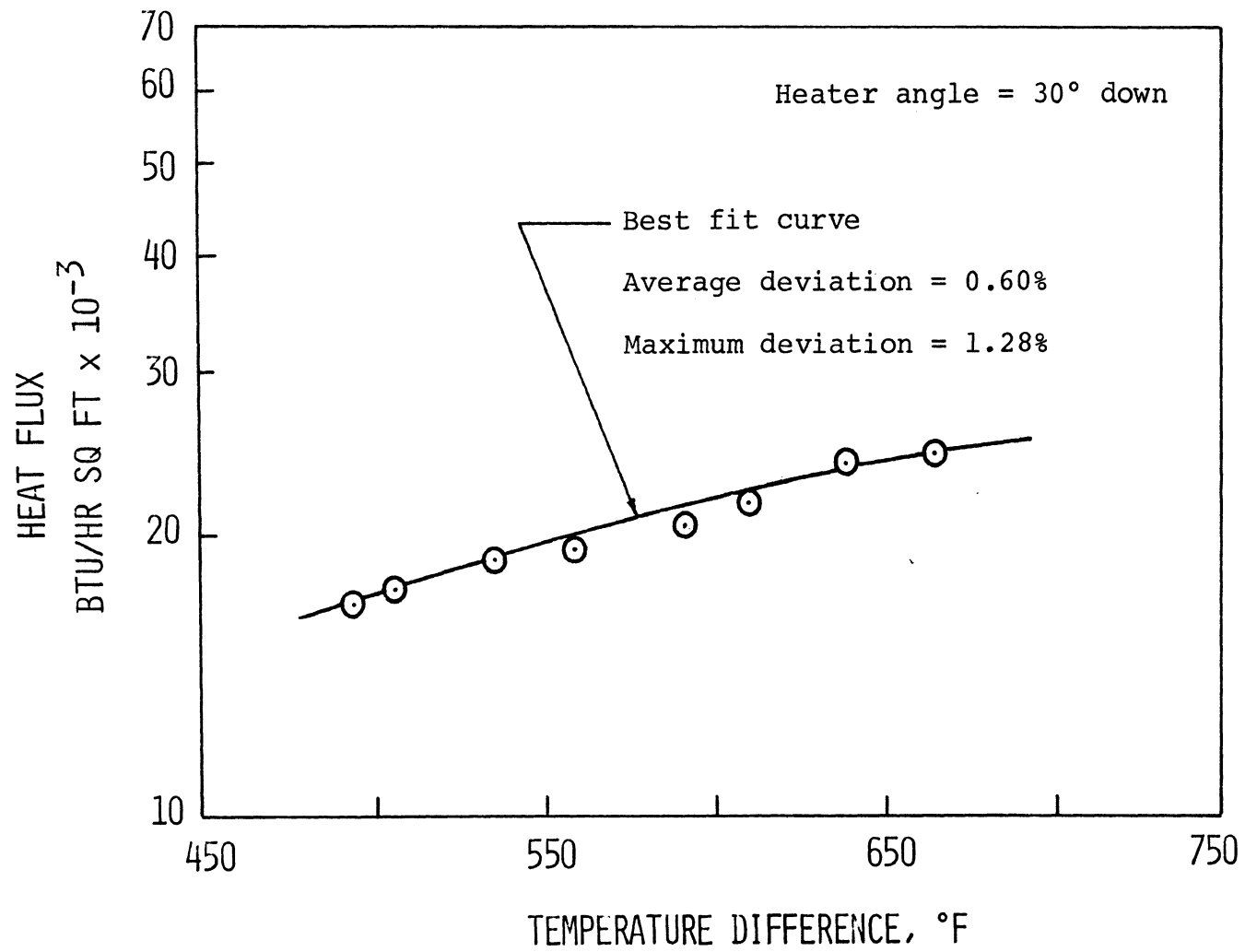Figure 7. Heat flux for heater angle of 15° down.

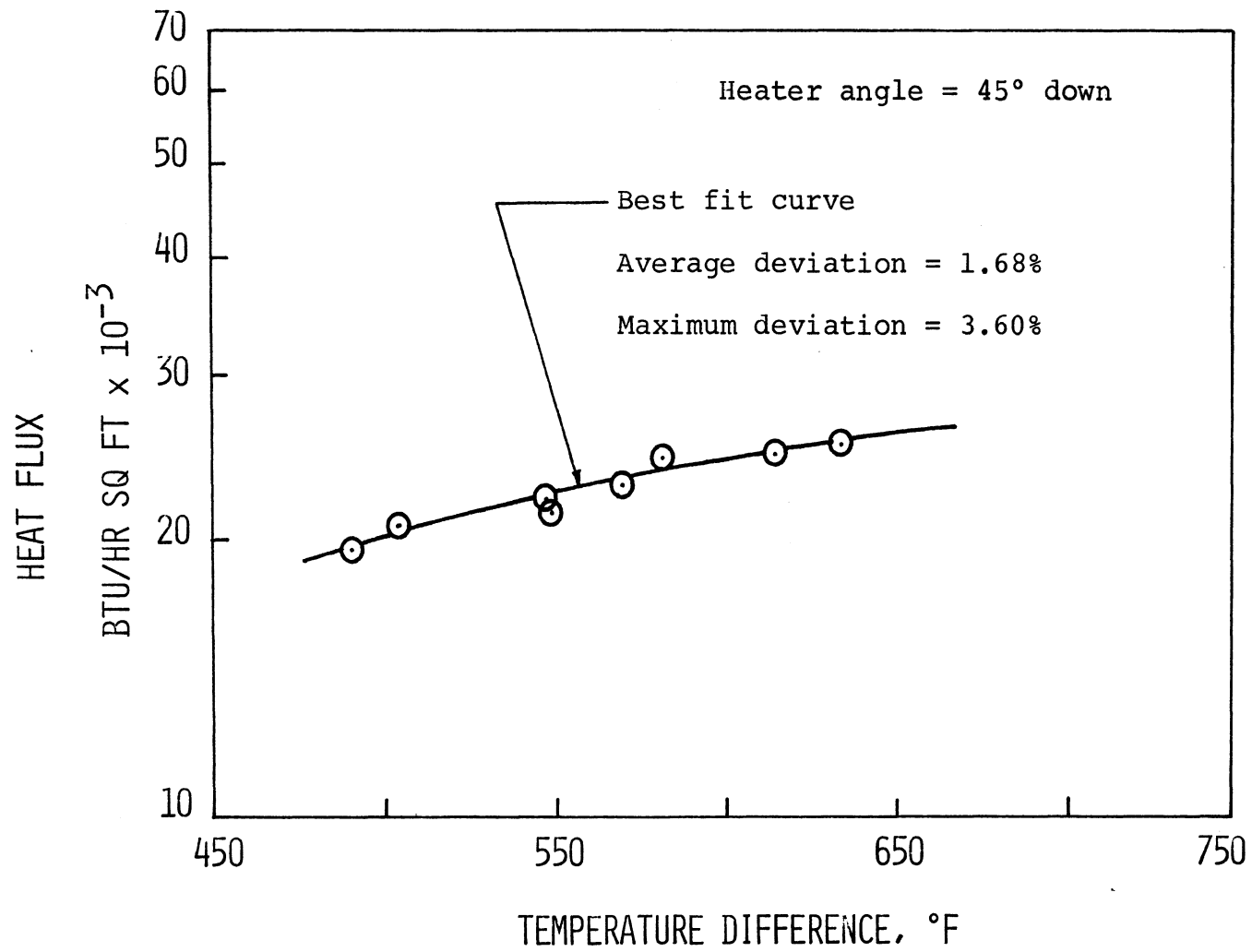Figure 8. Heat flux for heater angle of 30° down.
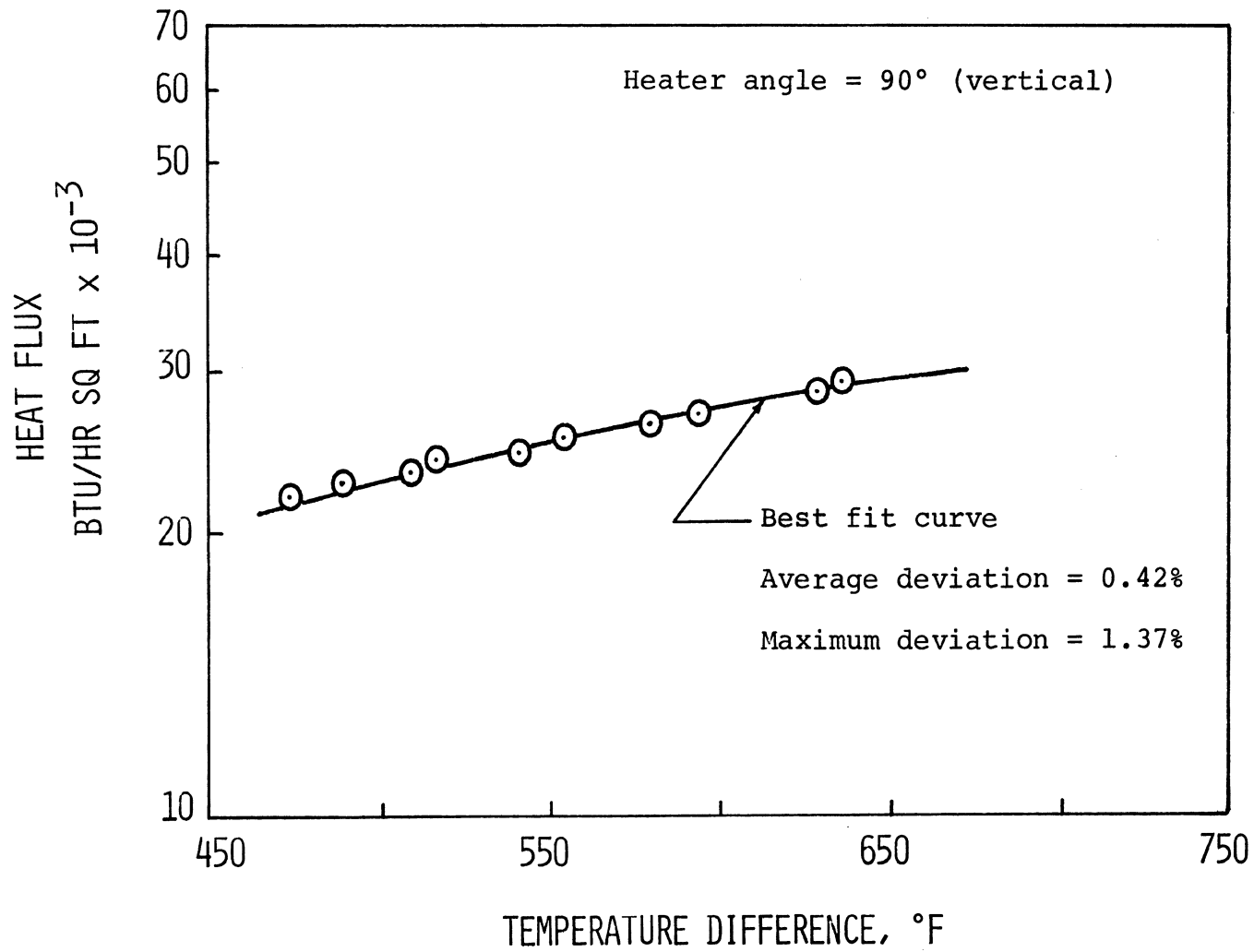
Figure 9. Heat flux for heater angle of 45° down.

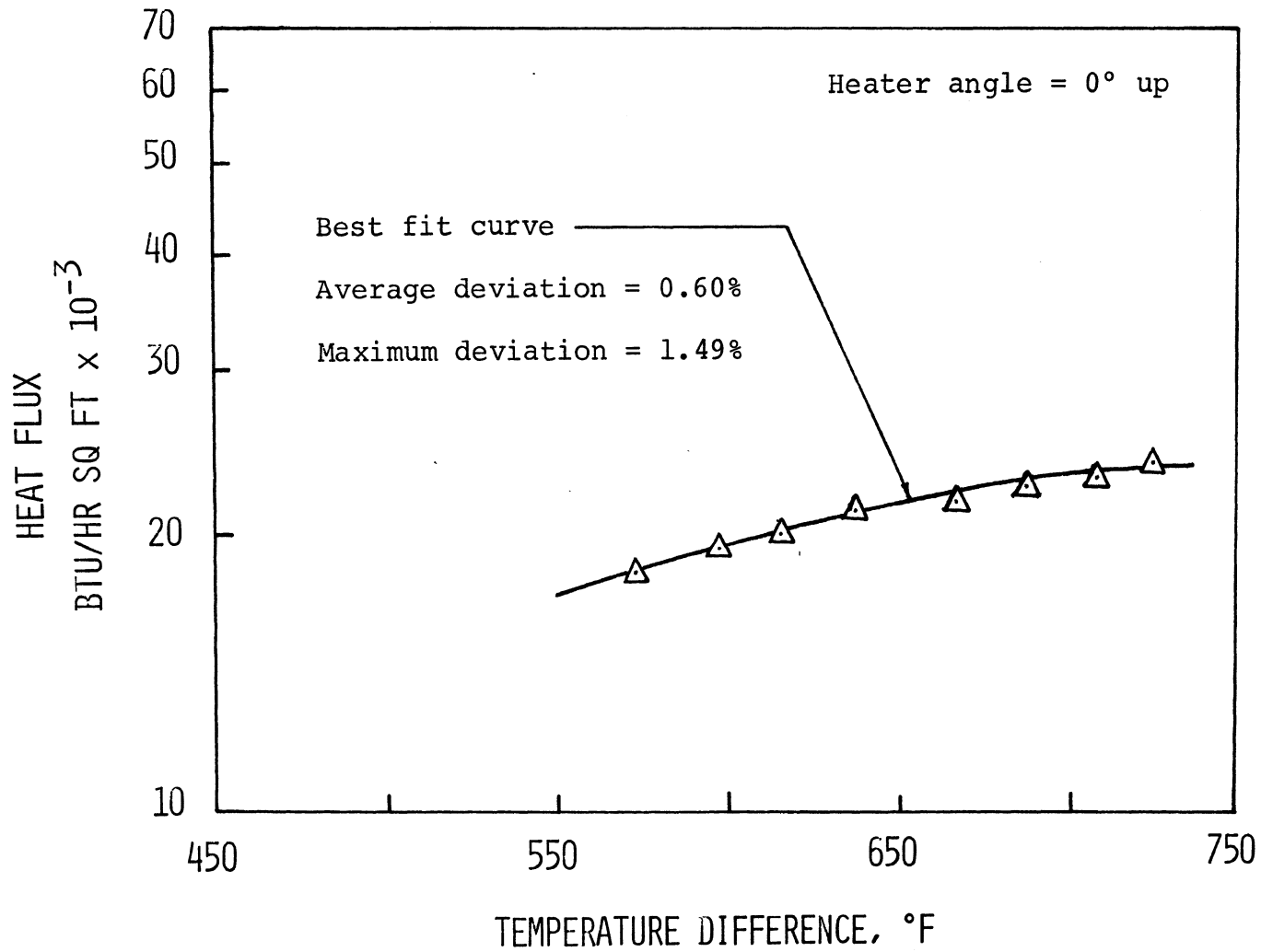Figure 10. Heat flux for the heater angle of 90° (vertical).
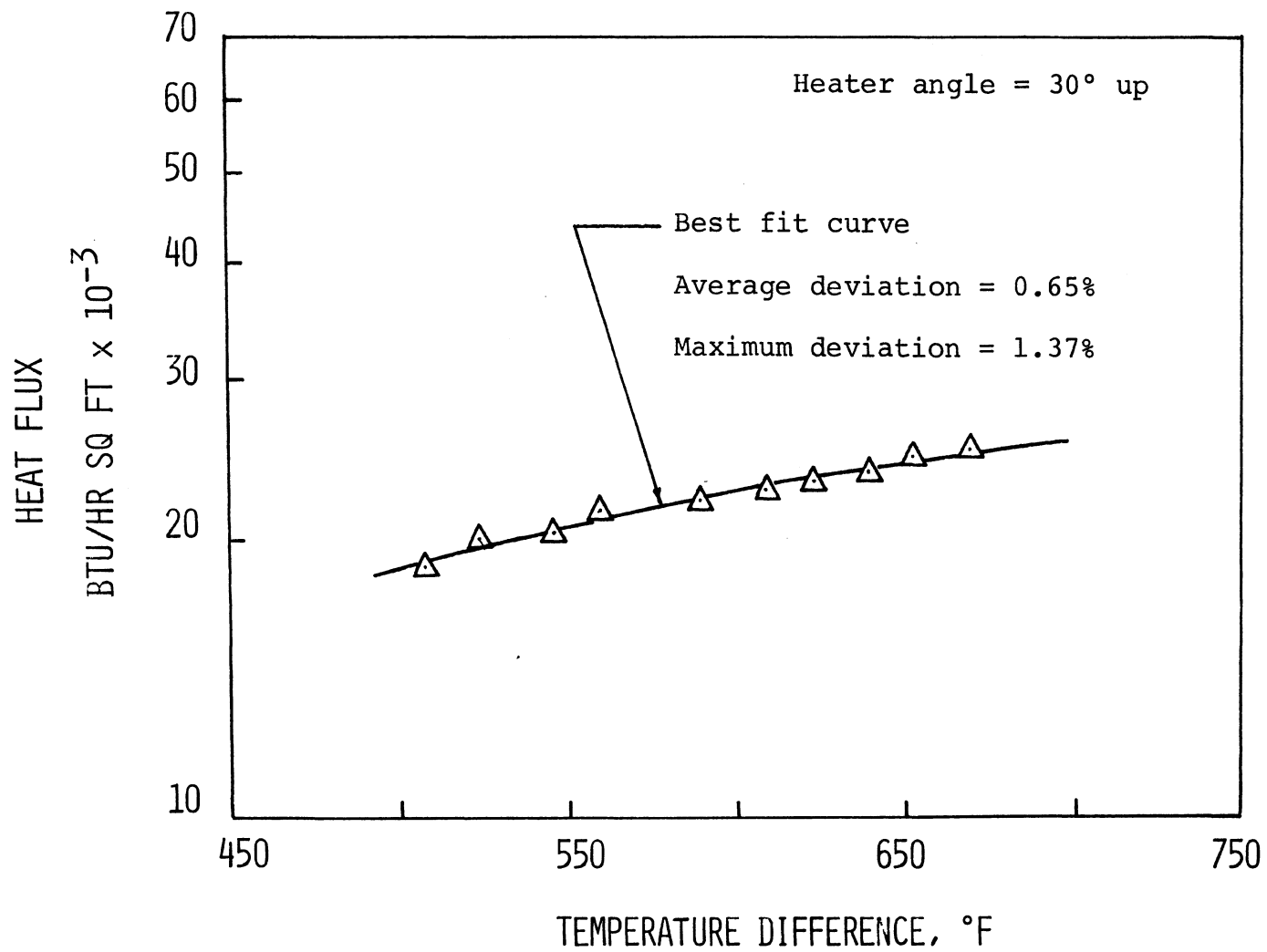
Figure 11. Heat flux for heater angle of 0° up.

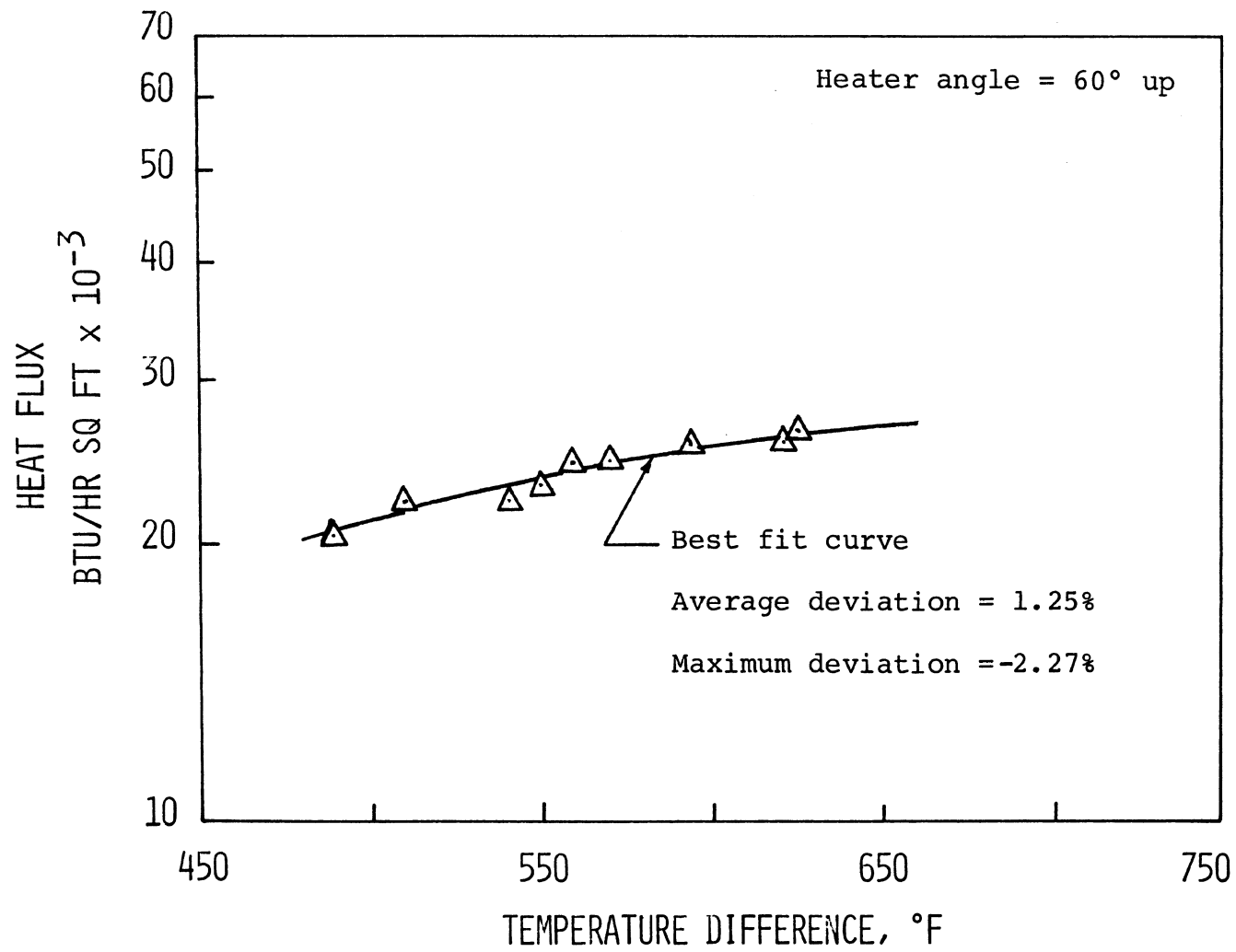Figure 12. Heat flux for heater angle of 30° up.
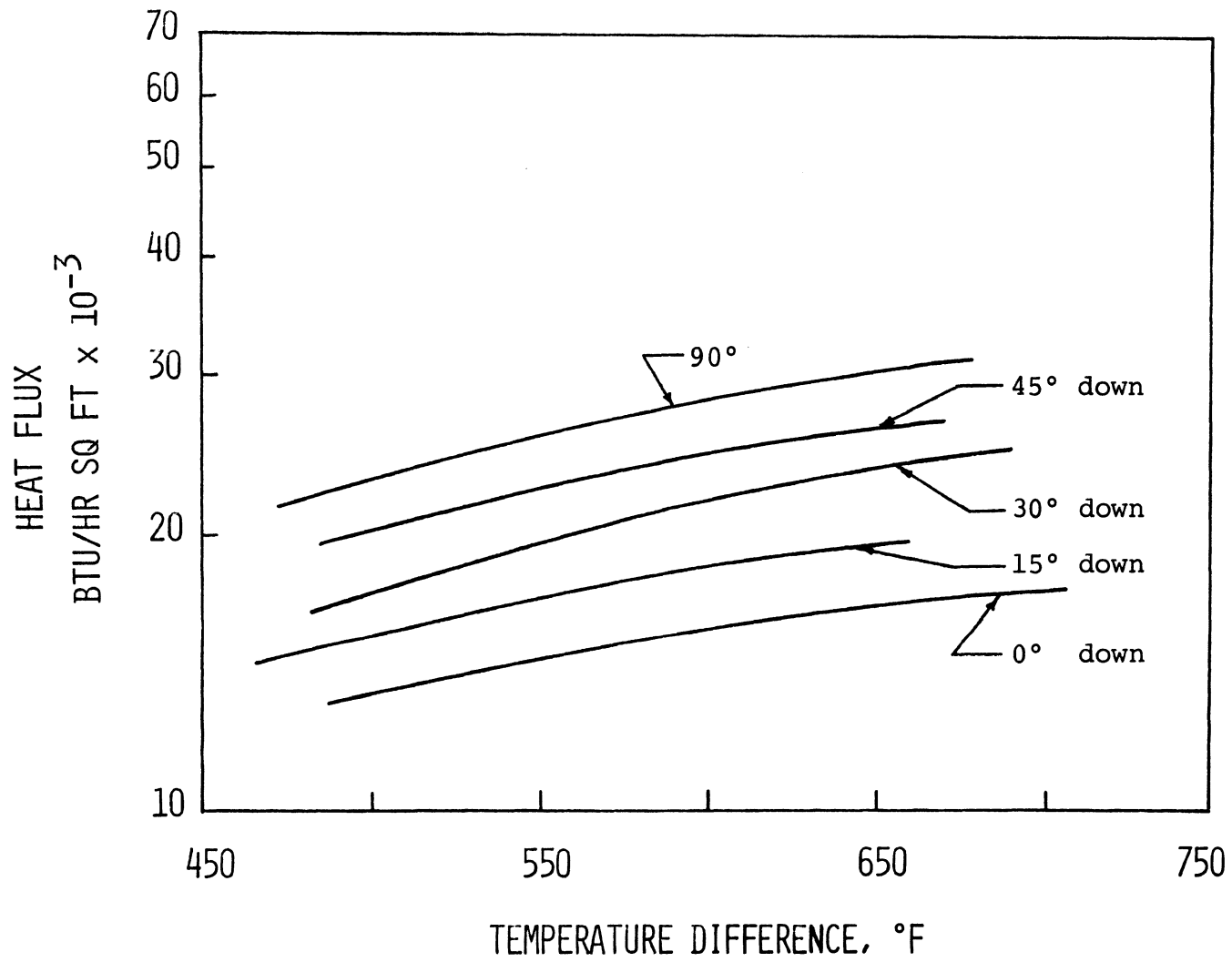
Figure 13. Heat flux for heater angle of 60° up.

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 co-efficient 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.
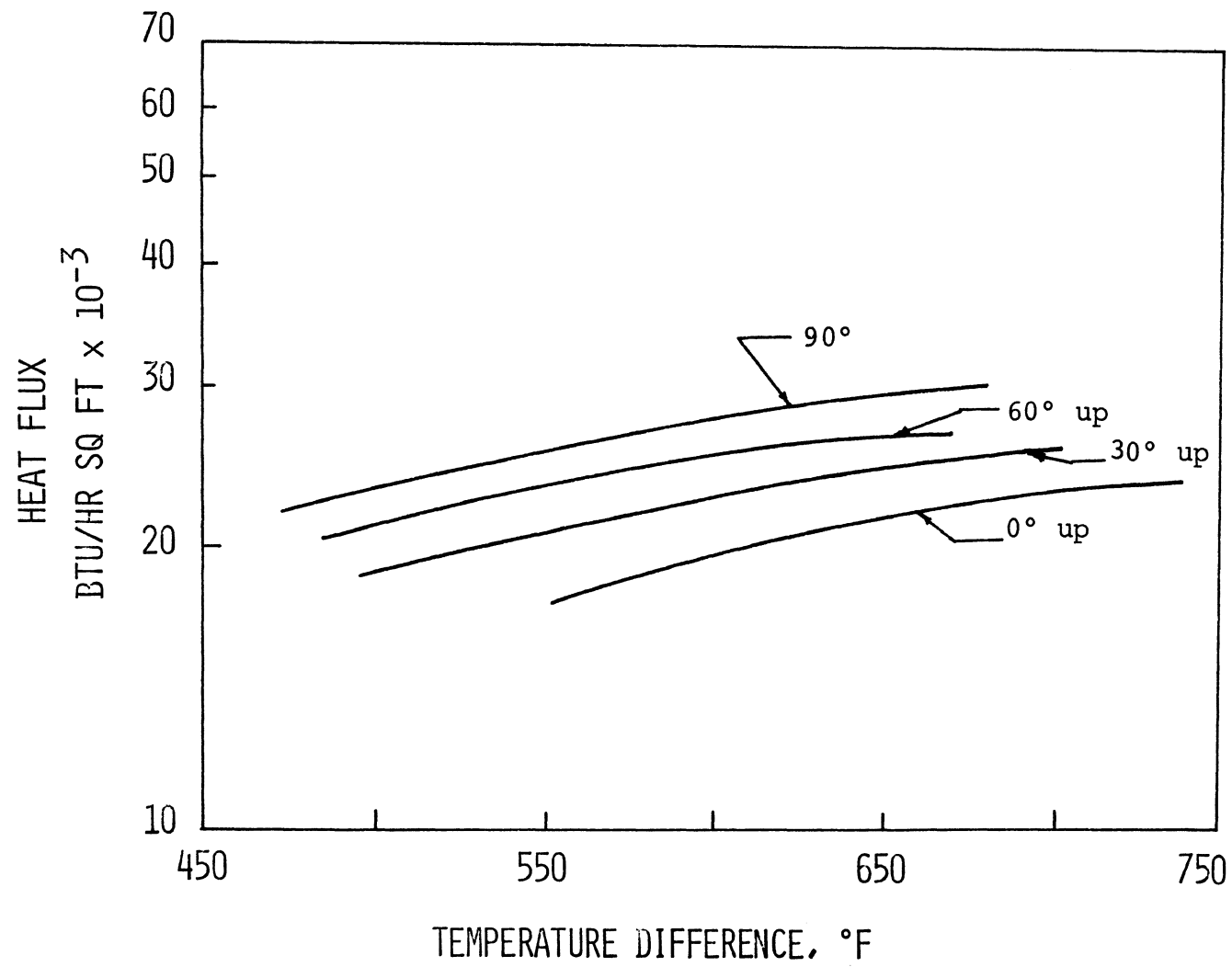
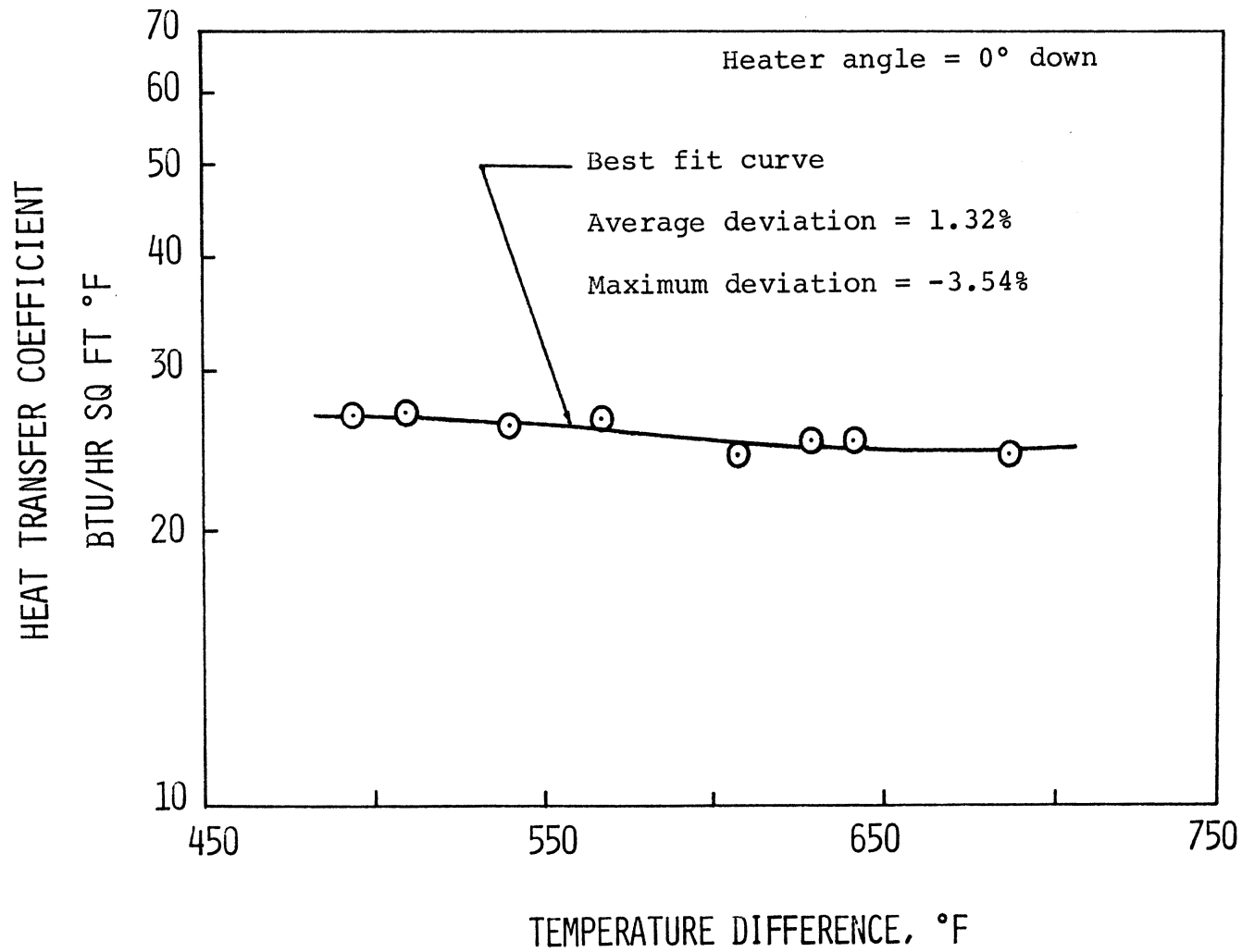Figure 15. Heat flux for heater surface facing upward.

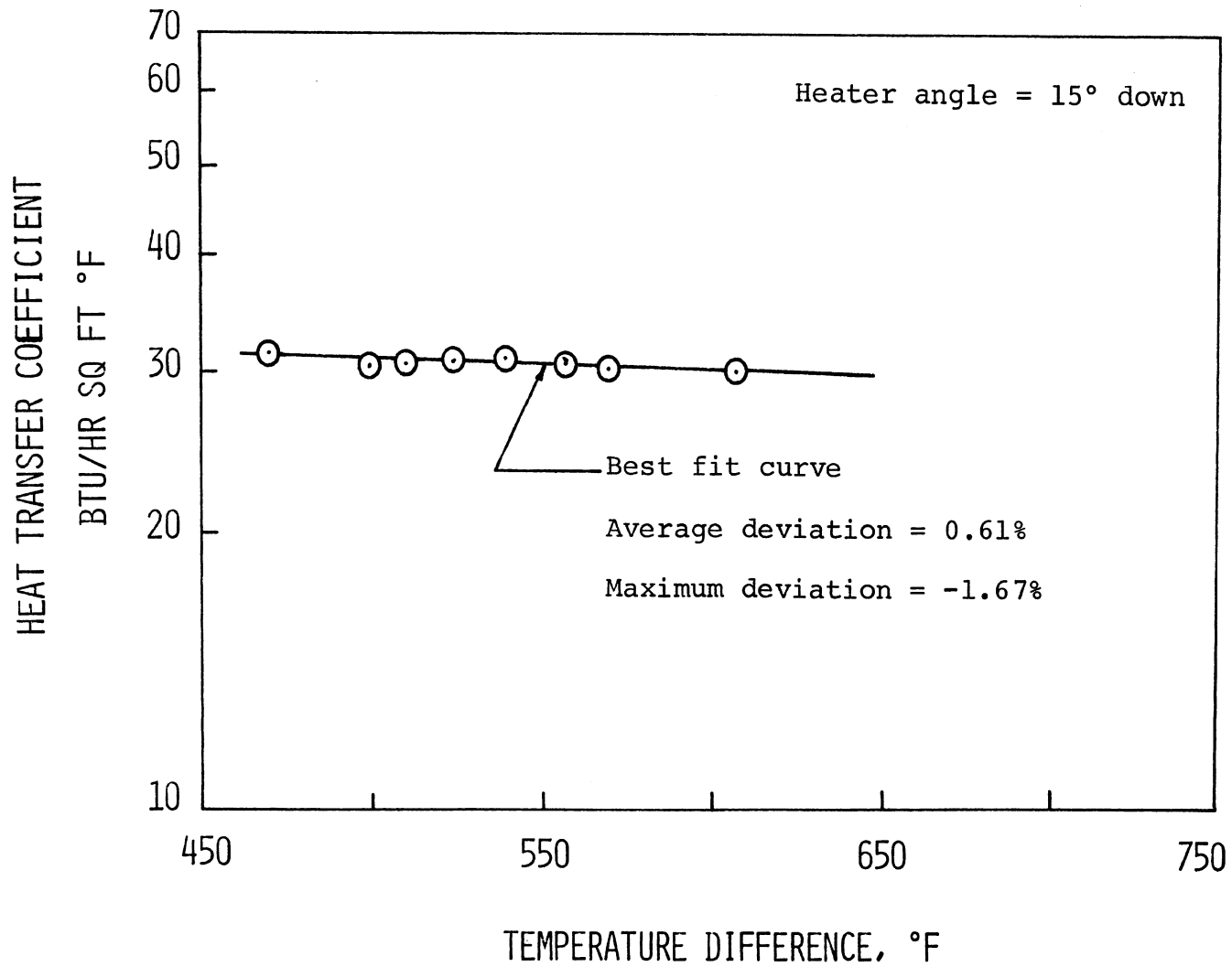Figure 16. Heat transfer coefficient for heater angle of 0° down.

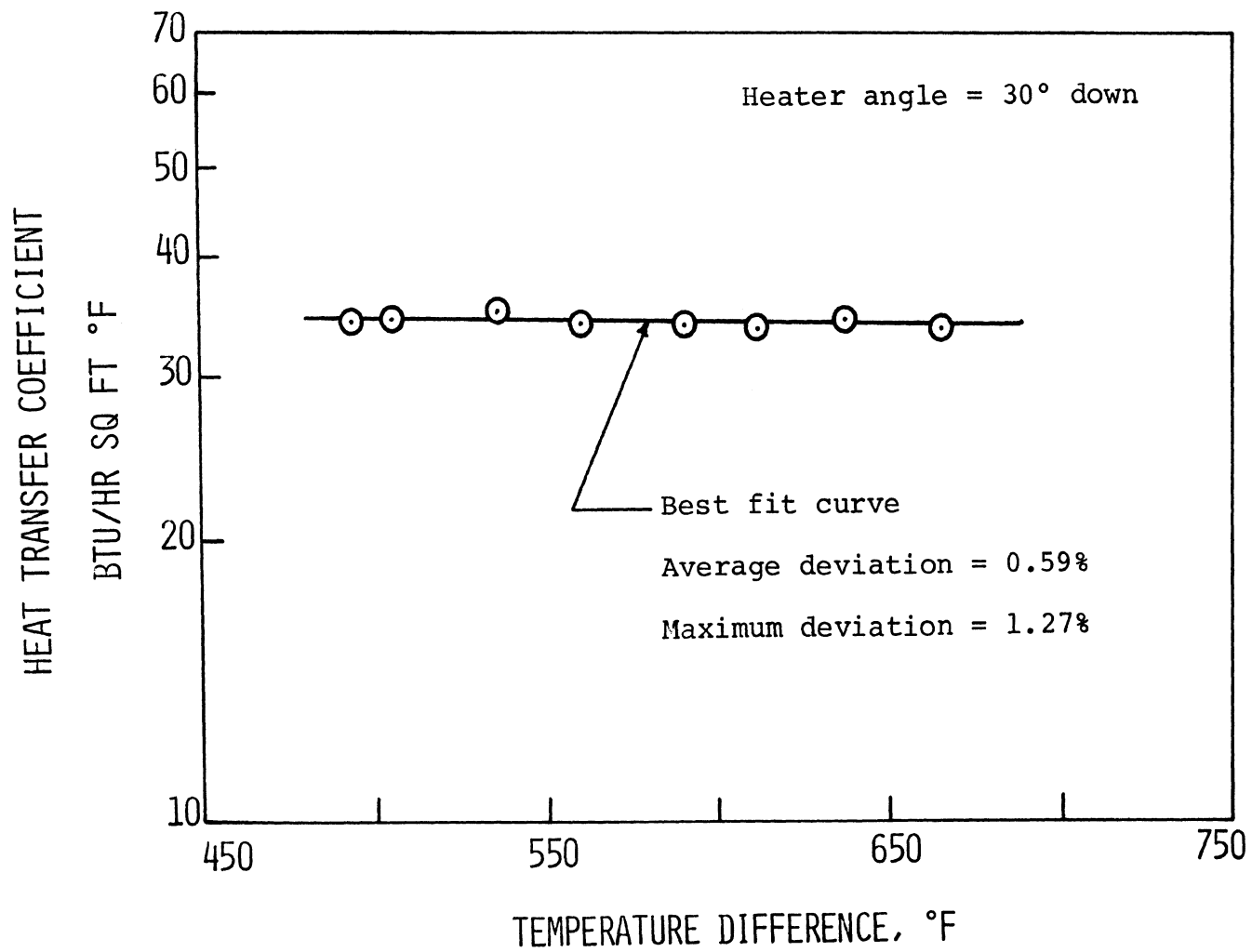Figure 17. Heat transfer coefficient for heater angle of 15° down.

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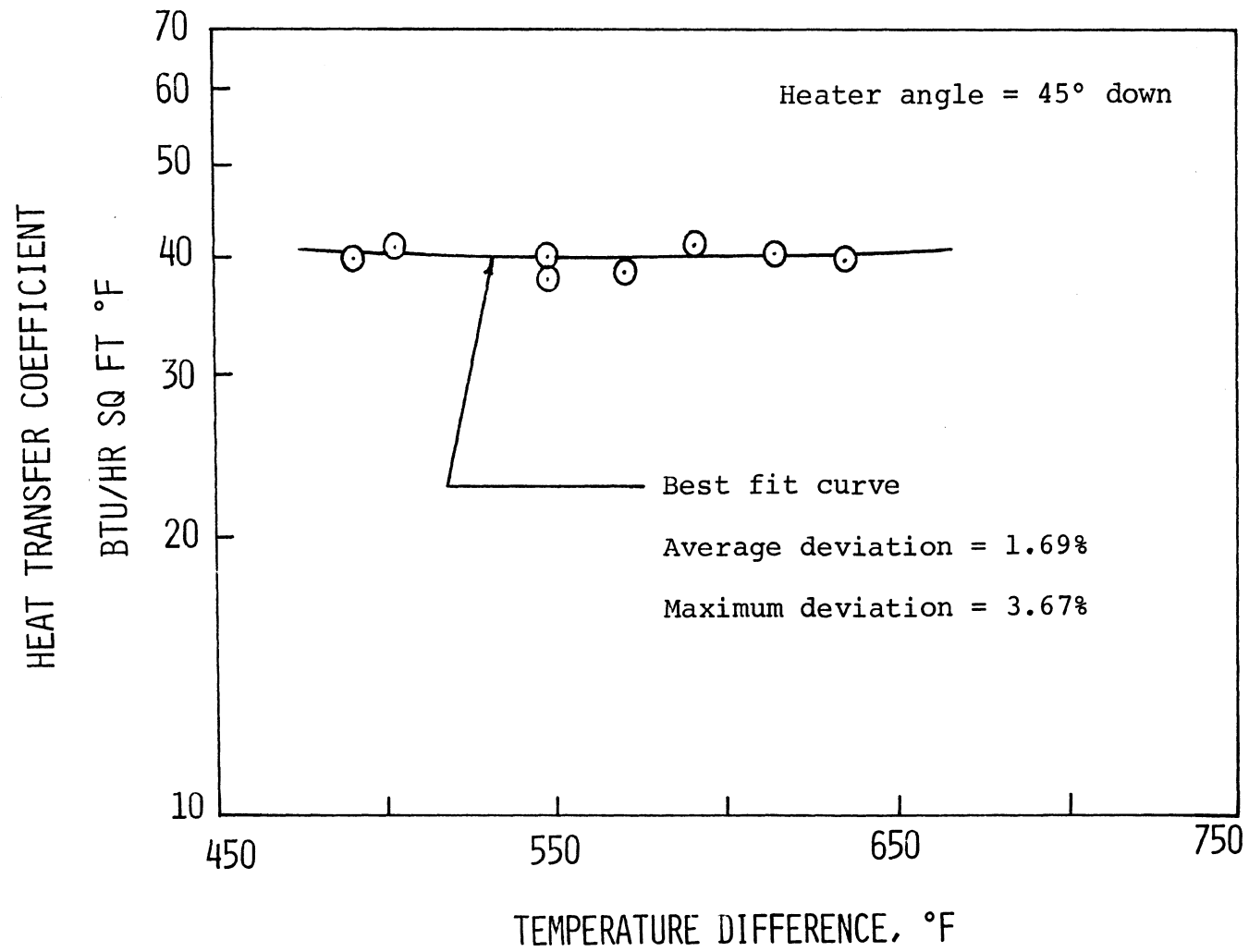down.

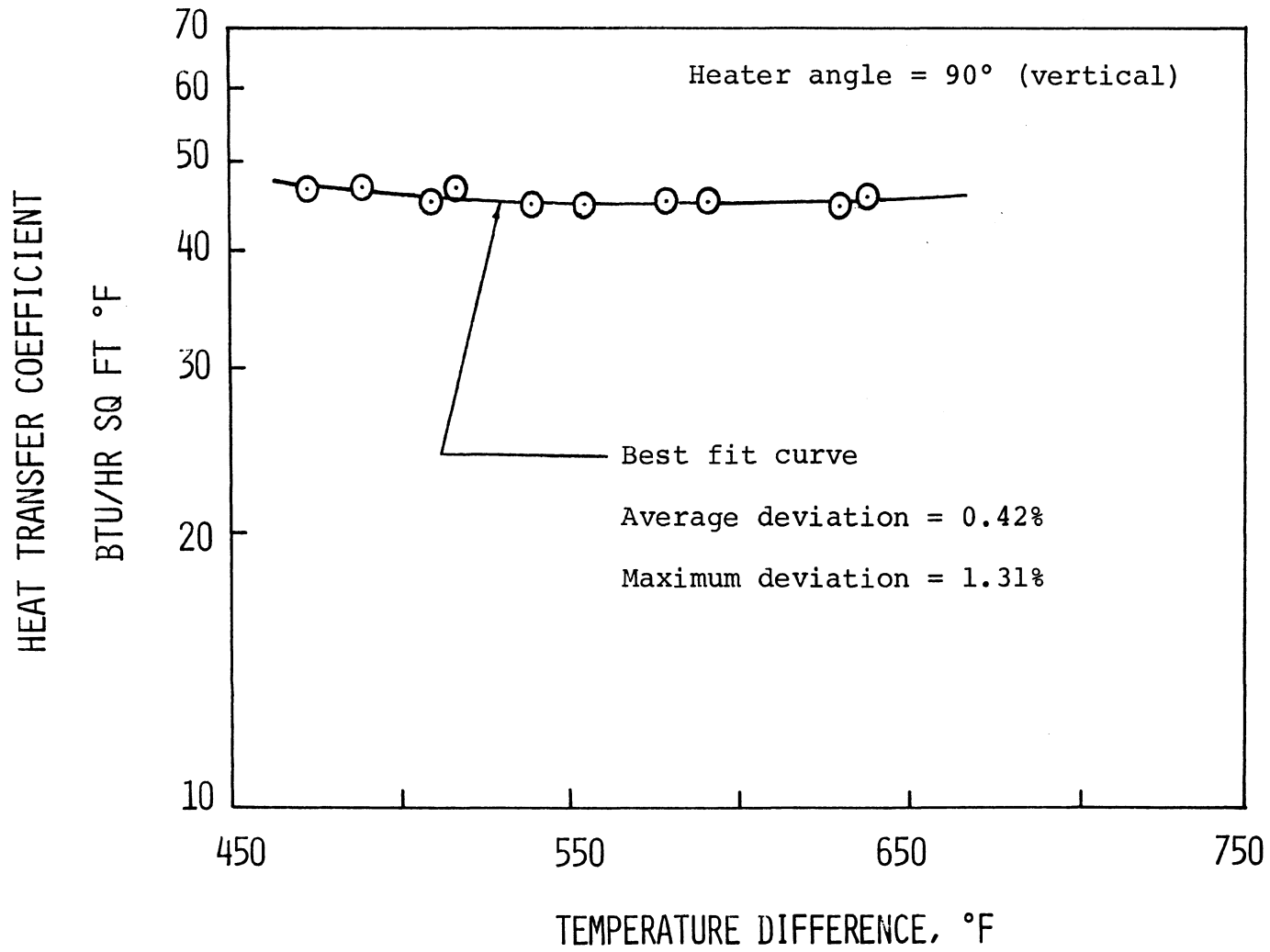Figure 19. Heat transfer coefficient for heater angle of 45° down.

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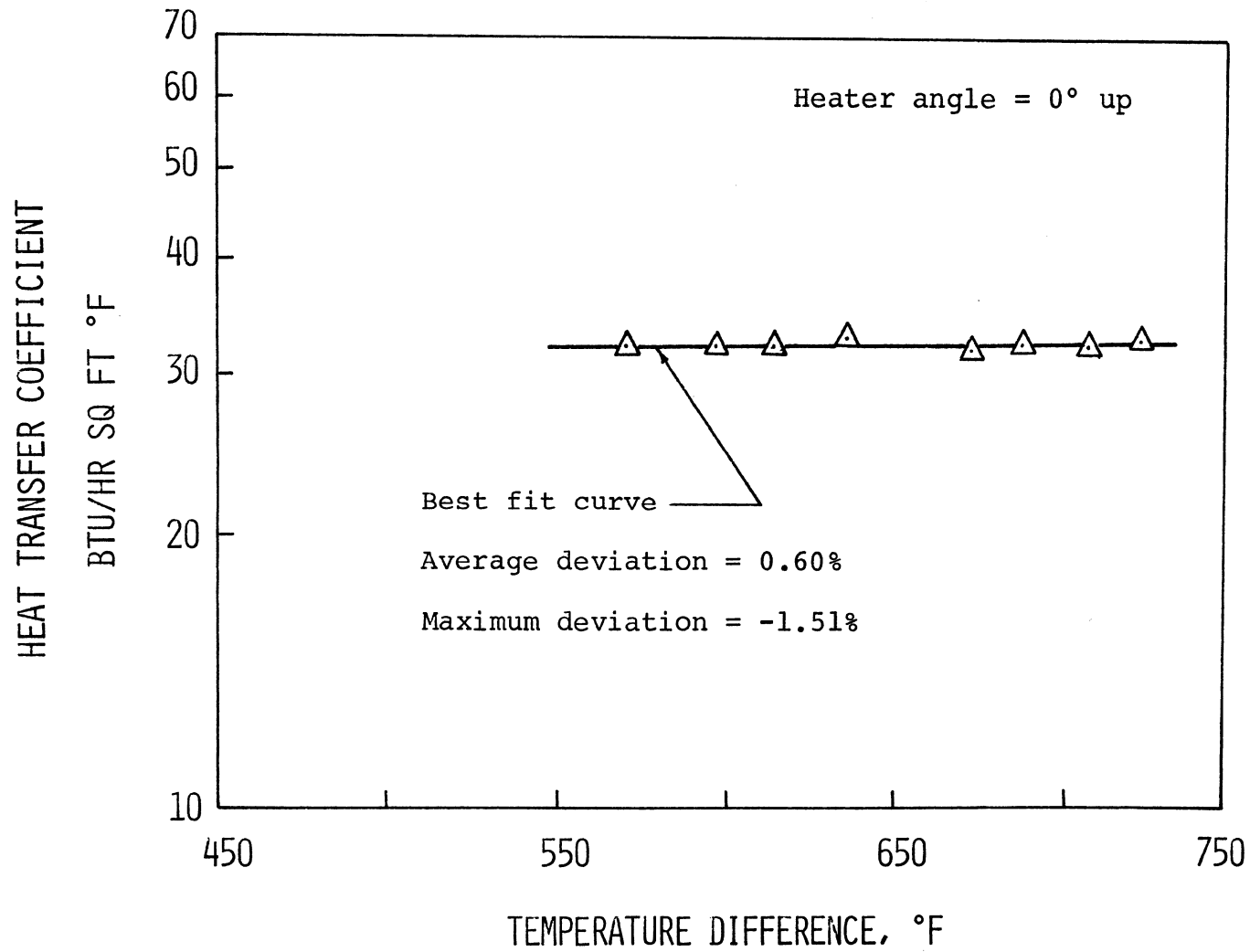(vertical).

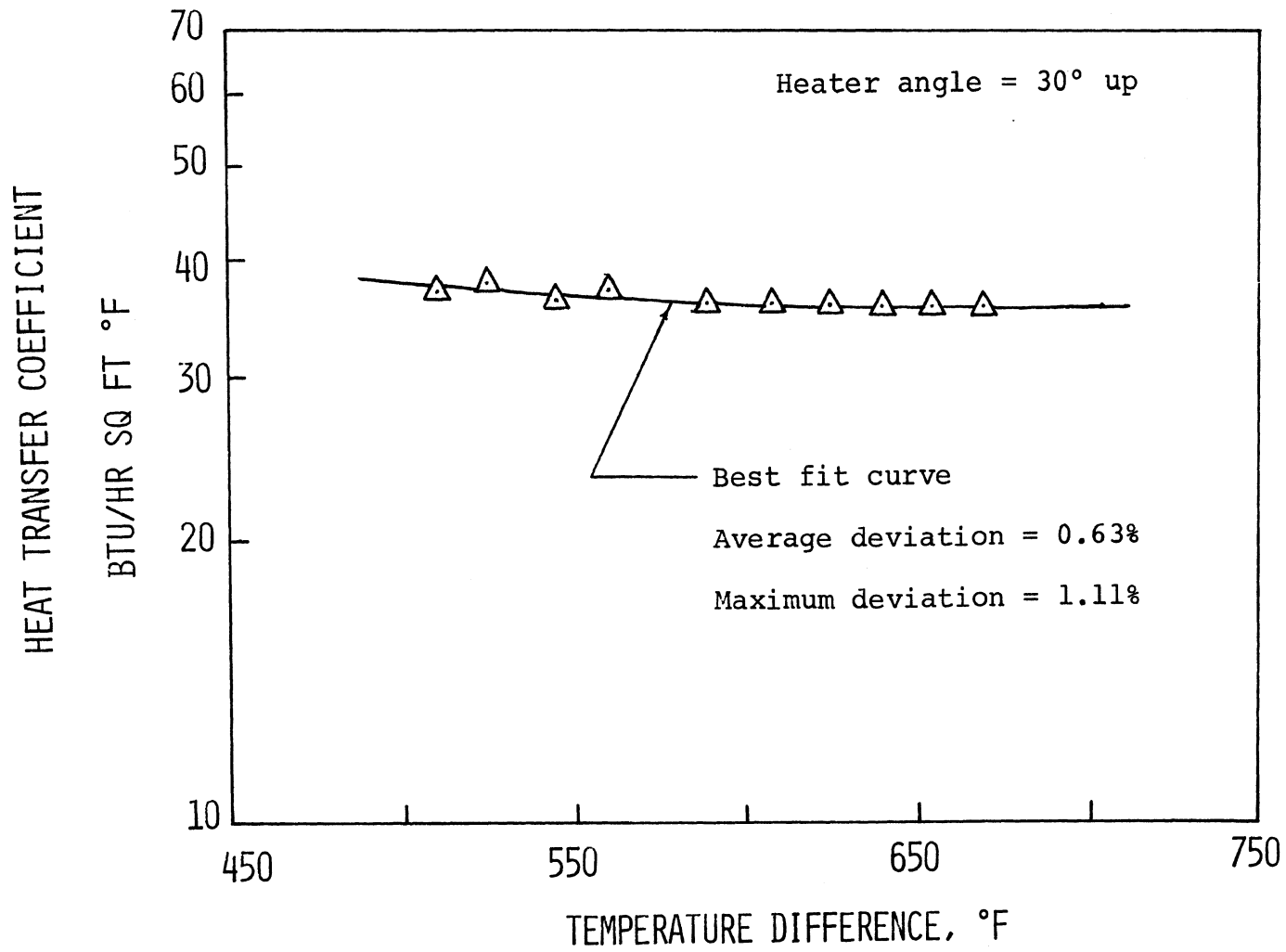Figure 21. Heat transfer coefficient for heater angle of 0° up.

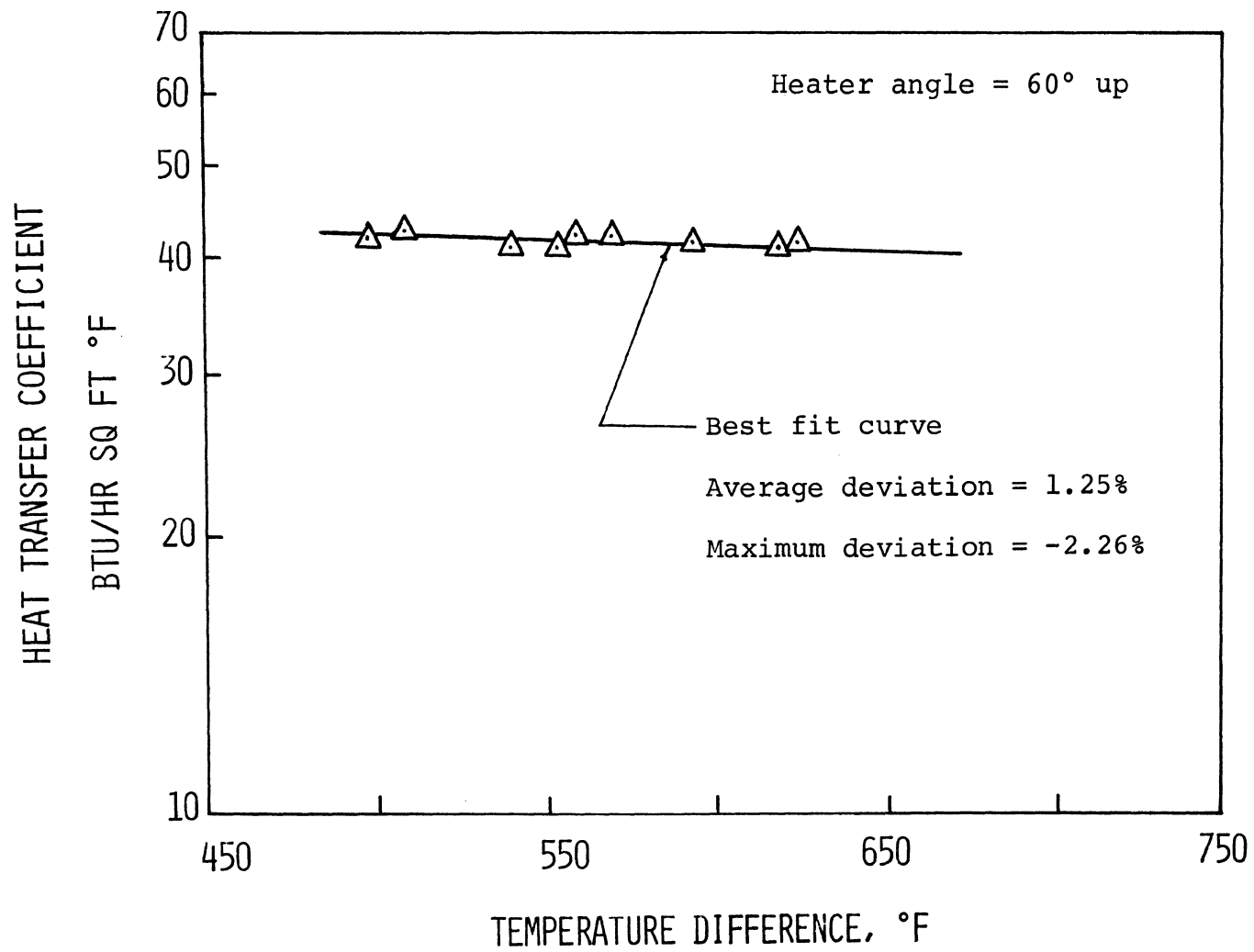Figure 22. Heat transfer coefficient for heater angle of 30° up.

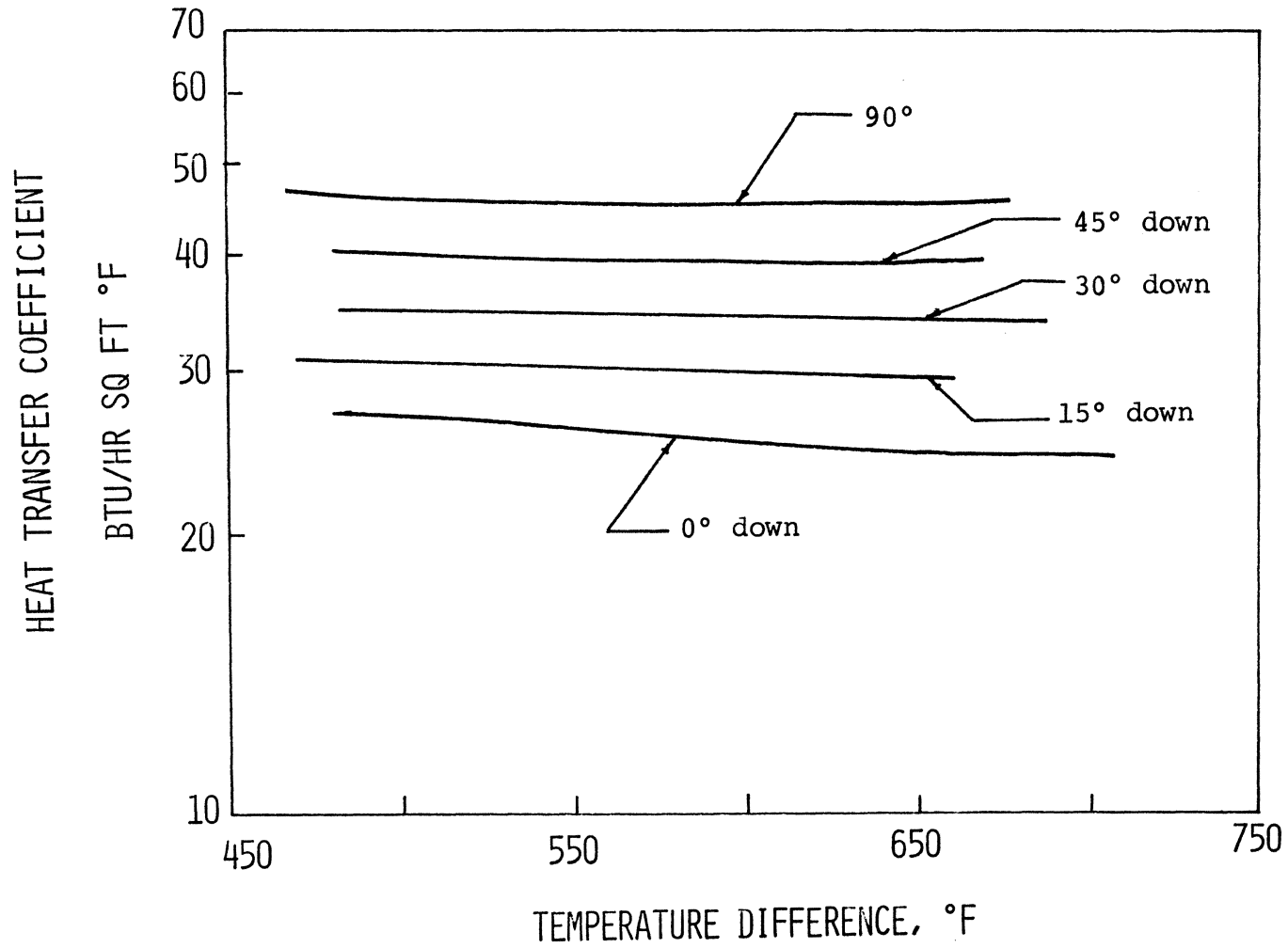Figure 23. Heat transfer coefficient for heater angle of 60° up.

Figure 24. Heat transfer coefficient for heater surface facing down-ward.

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°.
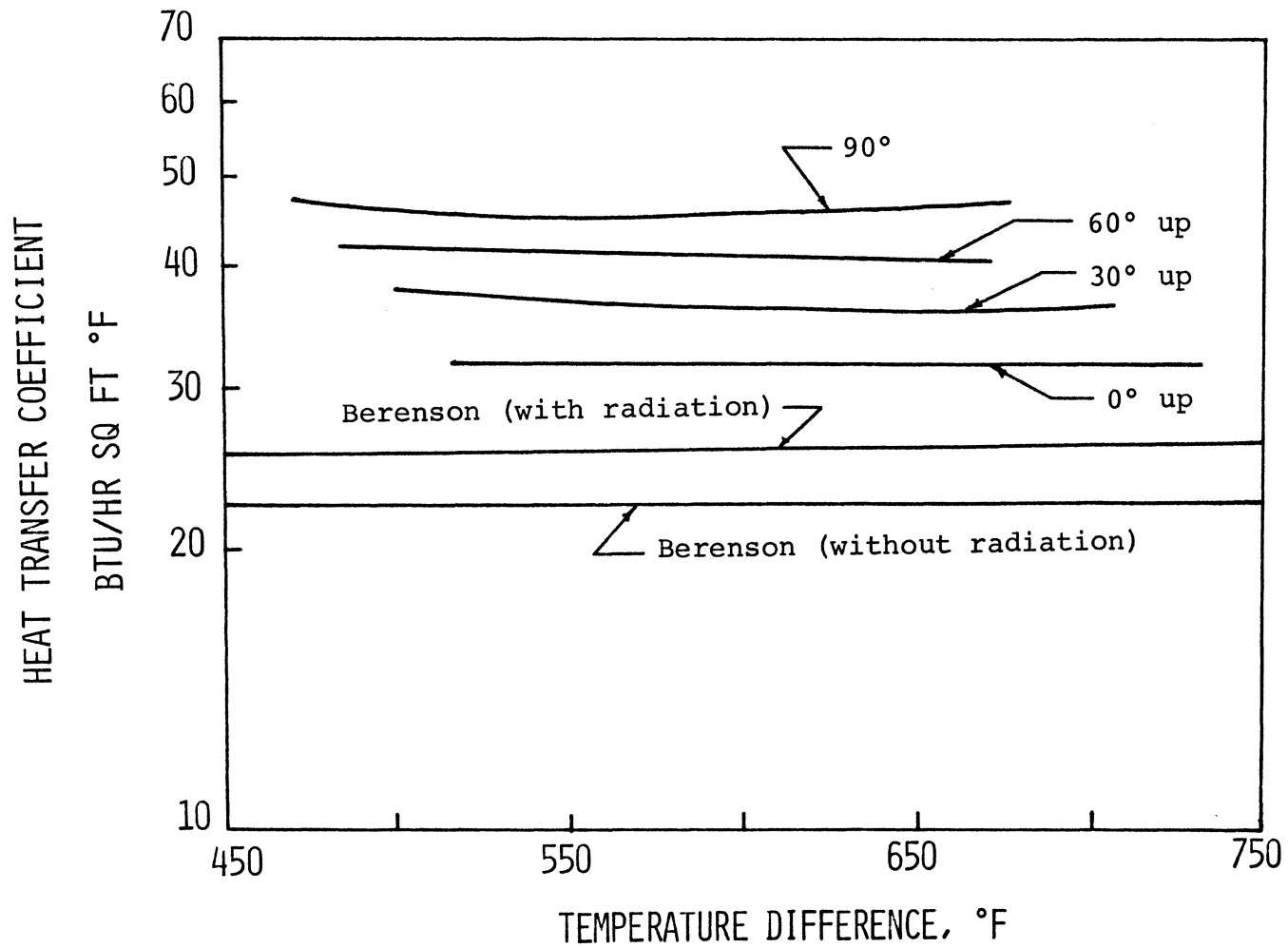
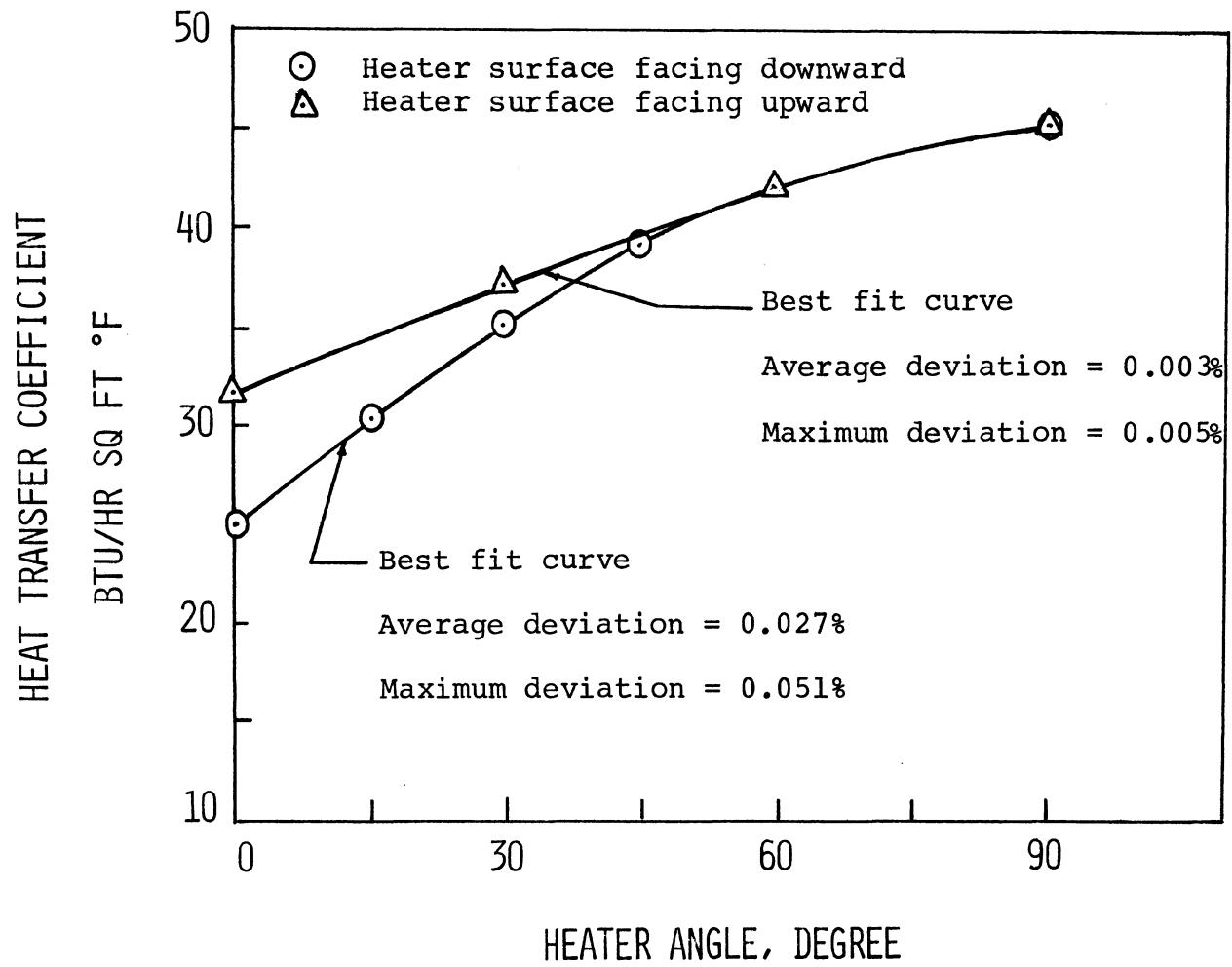Figure 25. Heat transfer coefficient for heater surface facing upward.

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, ±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°.

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