HEAT-TRANSFER COEFFICIENTS FOR FILM CONDENSATION OF STEAM ON AN INCLINED CYLINDER

by

TODD GARRETT

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Department of Mechanical Engineering

The University of British Columbia,
Vancouver 8, Canada.

Date September 1, 1960

ABSTRACT

This thesis discusses an investigation of film condensation of steam on the outside surface of an inclined cylinder. The objectives of the investigation were (1) to obtain heat-transfer coefficients for steam condensing on an inclined cylinder, (2) to determine the effect of the angle of inclination of the cylinder on the condensing coefficient, and (3) to verify values of the coefficient obtained from an accompanying theoretical development.

Experiments were conducted for steam condensing on a 12.45 in. length of the outside surface of a 3/4 in. diameter by 0.065 in. thick wall, water-cooled, copper tube.

Condensing coefficients were obtained for inclinations to the horizontal from 0° to 90° at 15° intervals. It was found that these coefficients decreased in value as the angle of inclination increased from 0° to 90°. At the same time good agreement was obtained between the measured and theoretical coefficients. No single equation could be found which would correlate the results for all the inclinations tested, but general equations were obtained which apply for each of two zones. One zone included inclinations from 0° to 30°, the other inclinations from 45° to 90°. The zone from 30° to 45° requires more investigation before it can be satisfactorily correlated.

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HEAT-TRANSFER COEFFICIENTS FOR FILM CONDENSATION OF STEAM ON AN INCLINED CYLINDER

HISTORICAL BACKGROUND

Heat-transfer coefficients for condensing vapours have been investigated by many experimenters over the last fifty years. Most of this work has been concentrated upon condensation on horizontal or vertical tubes.

In 1916, Nusselt derived the equation

$$h_T = 0.725(\rho^2 g \lambda K^3/\mu D \Delta t)^{1/4} \quad . \; . \; . \; . \; . \; . \; . \quad (1)^1$$

for the average heat-transfer coefficient for film condensation on a horizontal tube, and the corresponding equation

$$h_T = 0.943(\rho^2 g \lambda K^3/\mu L \Delta t)^{1/4} \quad . \; . \; . \; . \; . \; . \; . \quad (2)^2$$

for a vertical tube. His assumptions and his theory of film condensation will be presented later.

The results of Othmer (12) in 1929 for condensation on a horizontal tube are of interest in the present investigations. He determined heat-transfer coefficients for steam condensing on a 3 in. diameter horizontal tube, 3.9 ft. long. His values ranged from 1700 to 5000 Btu/hr ft^2 F for mean temperature differences between the steam and the tube surface of 19 to 2 F, respectively. The ratio of the actual to the theoretical coefficients ranged from 0.7 to 1.2.

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[1]Reference (11), equation (13-12), p. 338.

[2]Reference (11), equation (13-1g), p. 331.

Baker, Kazmark and Stroebe (2) investigated condensation on a 2 in. diameter vertical condenser tube 20 ft. long. Condensing coefficients from 700 to 1500 Btu/hr ft^2 F were obtained corresponding to mean temperature differences from 34 to 3 F. The average ratio of the actual coefficient to the theoretical was 1.2.

McAdams (10), on page 263, presents further information on the condensing coefficients of various vapours on horizontal and vertical tubes.

Tepe and Mueller (17) presented the first results for vapours condensing in an inclined pipe. Organic vapours were condensed inside a single water-cooled copper tube 7/8 in. diameter and 3 ft. long. They concluded that the condensing coefficients for benzene and methanol varied only slightly with the angle of inclination, but increased with increasing vapour velocities at all inclinations.

The next investigation of condensing coefficients on inclined cylinders was performed by Hassan and Jakob (9) who derived a theoretical relation. By condensing steam on the inside surface of a 1 1/2 in. diameter by 8 1/8 in. long copper tube, actual coefficients 28 to 100 per cent higher than the theoretical were obtained. It was found that the deviation from the theory increased with the inclination of the tube and the temperature difference between the tube surface and the steam. These deviations were attributed to ripples in the flowing condensate film.

The present investigation was begun to verify this theory

for vapours condensing outside an inclined tube, and to add to the amount of information available.

THEORY

As noted above, Hassan and Jakob developed the original theory for heat-transfer coefficients on inclined cylinders from Nusselt's basic concept of film condensation - the heat energy liberated by the condensing vapour is conducted through the film to the cooler surface. Their development depends upon the following ten assumptions, and an additional assumption 11, which is implicit in their work but was not stated as such.

1. The vapour is pure, dry, and saturated.
2. The condensate-film flow is laminar.
3. The vapour at the vapour-liquid interface is stagnant, and the shear stress is negligible at this interface.
4. The wall temperature is uniform.
5. The liquid-solid and liquid-vapour interfaces are smooth.
6. The curvature of the surface is large enough that the effect of capillary forces may be neglected.
7. The liquid temperature at the liquid-vapour interface is that of the saturated vapour.
8. The velocity distribution at any point on the cylinder surface is the same as that in a fully developed isothermal film flowing on a plane tangent to the surface at that point. Under such conditions the effect of acceleration in the film is neglected. Also, the effect of the curvature of the surface on the velocity distribution in the film is neglected.
9. The convective heat transfer along the condensate film is neglected. Thus considering the part of the condensate film in Figure 1, it is assumed that $q_3 = q_4$. Consequently, $q_1 = q_2$. Further, from the previous assumption, q_1 is equal to the latent heat liberated by the condensing vapour.
10. The physical properties of the condensate are constant.
11. The temperature distribution through the condensate film is linear.

Consider a droplet of condensate at any point P on the surface of an inclined cylinder. In accordance with the above assumptions equilibium is established between the force of gravity and the viscous force acting on the droplet. The droplet moves, momen-

tarily, along the line of intersection of the plane tangent to the cylinder at the point P and the vertical plane normal to the tangent plane.

In Figure 2 a cylinder of radius r is inclined at an angle α to the horizontal. Using coordinate axes as shown and letting a, b and c be unit vectors in the x, y and z-directions, respectively, then the unit normal vectors to the cylinder surface and the horizontal plane are, respectively,

$$n_c = b \sin \phi + c \cos \phi \quad \ldots\ldots\ldots \quad (3)$$

and

$$n_h = -a \sin \alpha + c \cos \alpha \quad \ldots\ldots\ldots \quad (4)$$

where ϕ is the angular position from top element of the tube.

Let θ be the angle between the momentary droplet path at the point P and the horizontal plane. This is the angle between the plane tangent to the cylinder at the point P and the horizontal plane and is, therefore, equal to the angle between the two unit normal vectors n_c and n_h. Hence,

$$\cos \theta = \cos \alpha \cos \phi \quad \ldots\ldots\ldots \quad (5)$$

In Figure 3 the angle ψ between the momentary droplet path at P and a cross section of the cylinder is the angle between the yz-plane and the line of intersection of the tangent plane at P and the plane of n_c and n_h. Now, the yz-plane and the plane of n_c and n_h are both normal to the tangent plane, thus, ψ is equal to the angle between the x-axis and the unit normal vector n_i to the plane of n_c and n_h. As n_i is parallel to the intersection of the

tangent and the horizontal planes, it is given by

$$n_i = (a \cos\alpha \sin\phi - b \sin\alpha \cos\phi + c \sin\alpha \sin\phi)/(\sin^2\alpha + \cos^2\alpha \sin^2\phi)^{1/2} \quad . \quad (6)$$

The angle ψ can then be obtained from

$$\cos\psi = \sin\phi\,(\tan^2\alpha + \sin^2\phi)^{1/2} \quad . \; . \; . \; . \; . \; . \quad (7)$$

Consider a small element on the surface of the cylinder as shown in Figure 4. Condensate of thickness Y enters the element with a mean velocity v_m making an angle ψ with the planes of constant x. Using assumptions 5, 7, 9, 10 and 11, the rate of heat transferred by conduction through the film equals the product of the latent heat of condensation of the vapour and the increase in mass of the element. Therefore,

$$(Kr/Y)\,\Delta t\; dx\; d\phi = \rho\lambda\left[\frac{\partial}{\partial\phi}(v_m\, Y \cos\psi\, dx)d\phi + \frac{\partial}{\partial x}(v_m\, Y\, r \sin\psi\, d\phi)dx\right] . \; . \; . \quad (8)$$

From assumption 8 Nusselt's expression for the mean velocity of an isothermal liquid film on an inclined plate can be used. Then

$$v_m = \rho g\, Y^2 \sin\theta / 3\mu \qquad (9)^1$$

Employing equations (5), (7) and (9), equation (8) reduces to

$$3\mu r\, K\Delta t/\rho^2 g\lambda = Y^4 \cos\alpha \cos\phi + 3Y^3 \cos\alpha \sin\phi \frac{\partial Y}{\partial\phi} + 3Y^3\, r \sin\alpha \frac{\partial Y}{\partial x} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (10)$$

Letting $$Z = \rho^2 g\lambda Y^4 \cos\alpha / 3\mu r\, K\Delta t \quad . \; . \; . \; . \; . \; . \; . \quad (11)$$

and letting the reduced distance X be defined as

$$X = x/r \tan\alpha \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (12)$$

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[1] Reference (3), equation (XVb-31), p. XV-8.

equation (10) simplifies to

$$(\partial Z/\partial X) + (\partial Z/\partial \phi) \sin\phi = (4/3)(1 - Z \cos\phi) \quad . . . (13)$$

Two boundary conditions apply. First, the condensate has zero thickness at the uppermost edge, and therefore

$$Z = 0 \text{ at } X = 0 \quad . . . . . (14)$$

The second boundary condition is based on the symmetry of the condensate film and its smoothness at $\phi = 0$ and π, therefore

$$\partial Z/\partial \phi = 0 \text{ at } \phi = 0 \text{ and } \pi \quad . . . . . (15)$$

Differential equation (13) was solved numerically by Hassan and Jakob. Their calculations were only carried out to $X = 3.2$, as the conditions at this cross section are, for practical purposes, very near to those at $X = \infty$.

With the values of Z obtained from equation (13), local values of the heat-transfer coefficient h can be calculated. Consider the heat transferred through a small element of the condensate film with surface area dA. Two laws apply:

Fourier's Law

$$q = K\, dA\, \Delta t/Y \quad . . . . . (16)^2$$

and Newton's Law

$$q = h\, dA\, \Delta t \quad . . . . . (17)^3$$

Equating equations (16) and (17) gives

$$h = K/Y \quad . . . . . (18)$$

---

[2]Reference (11) equation (2-1) p. 7.

[3]Reference (11) equation (1-4) p. 5.

If the value of Y from equation (11) is substituted in equation (18), then

$$h = (\rho^2 g \lambda K^3 \cos\alpha / 3\mu r \Delta t \; Z)^{1/4} \quad . \; . \; . \; . \; . \; (19)$$

which is the equation for the local values of the heat-transfer coefficient h.

The mean heat-transfer coefficient h_m for an infinitesimal ring of condensate on the cylinder surface a reduced distance X from the starting end, can be obtained from

$$h_m = \frac{1}{\pi}\int_0^{\pi} h(X, \phi)\, d\phi \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; (20)$$

Hassan and Jakob used Simpson's two-thirds rule to evaluate this integral. Figure 5 shows their results. The values of h_m form a smooth curve which varies from ∞ at X = 0 to a constant value of $0.8045(\rho^2 g \lambda K^3 \cos\alpha / 3\mu r \Delta t)^{1/4}$ at $X = \infty$.

For X = 3.2 the mean heat-transfer coefficient h_m is within 2 per cent of the value at $X = \infty$. As a result the part of the tube up to X = 3.2 may be considered as a starting length.

The mean heat-transfer coefficient h_T for the entire surface of an inclined cylinder is dependent upon the reduced length of the cylinder given by

$$X_L = L/r \tan\alpha \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; (21)$$

This mean heat-transfer coefficient can be derived from the expression

$$h_T = (1/X_L)\int_0^{X_L} h_m(X)\, dX \quad . \; . \; . \; . \; . \; . \; . \; . \; (22)$$

A numerical solution for equation (22) is given in Appendix I. The results are plotted in Figure 6. Again a smooth curve is

formed from $X_L = 0$ to ∞ with the same boundary values as for h_m. For a cylinder with $X_L = 50$ the mean heat-transfer coefficient is within 2 per cent of that for an infinitely long cylinder.

In Figure 7 the variation of the mean heat-transfer coefficient on the entire surface of geometrically similar cylinders versus the angle of inclination is presented. From this figure it is seen that for any size of cylinder there is an inclination for which the coefficient is a maximum. The dashed line represents the variation of this optimum inclination with the length to radius ratio of the cylinder. For the present apparatus the optimum inclination is 0°.

The results of equation (22) are used to calculate the mean heat-transfer coefficient for the present apparatus at inclinations below 90° to the horizontal.

Nusselt's equation

$$h_T = 0.943(\rho^2 g \lambda K^3/\mu L \Delta t)^{1/4} \quad . \; . \; . \; . \; . \; . \; . \; (23)$$ [4]

is used for the vertical position.

Values of the actual heat-transfer coefficient are calculated from the equation

$$q = h_A A \Delta t_m \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; (24)$$

The heat-transfer rate q is based on the heat gained by the water and is calculated from

$$q = w c_p \Delta t_w \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; (25)$$

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[4]Reference (11), equation (13-1g), p. 331.

APPARATUS

A photograph of the apparatus and instruments is shown in Figure 8. The general arrangement of the apparatus is shown in Figure 9. Steam from the U.B.C. power house is supplied to pressure regulator 5 at 150 psig. Moisture is drawn off from the separator 4 and discarded. The pressure regulator maintains a constant pressure in the apparatus. The steam enters the apparatus through two baffled manifolds 6. The condensate from the apparatus passes through the hotwell 7 and into the condensate collector 8. A constant head tank 9 maintains a steady water flow through rotameter 10 and condenser tube 3. The cooling water is discarded after it leaves the condenser tube.

At the start of a run all valves are opened allowing water and steam to enter the apparatus. The test is begun when steady-state conditions are reached.

Figure 10 shows the condenser tube and its steam chamber. The condenser tube 3 is a 3/4 in. O.D. by 0.065 in. wall copper tube 22 3/8 in. long. Teflon o-rings 11 hold the tube in place and seal the space between the tube and the top and bottom end plates 1 and 2, respectively. A 12 in. length of 6 in. I.D. pyrex pipe 12 completes the steam chamber. Rubber o-rings 13 seal the space between the end plates and the pyrex pipe. Four 1/4 in. diameter steel tie rods 14 inside of spacers 15 are used to hold the apparatus together. Steam enters the apparatus through the baffled manifolds 4. Thermocouples for measuring the steam tem-

perature enter through the attachments 16. The steam condensate leaves through the opening 17 in the bottom of end plate 2. The whole assembly, which is mounted on wooden supports as shown in Figure 8, can be rotated about the axis of end plate 2. The angle of inclination of the apparatus to the horizontal, α , can be varied from 0° to 90° at 5° intervals.

Several points had to be considered when the apparatus was designed, namely:

1. Contamination of the tube surface must be prevented.
2. The condensing surface must be visible.
3. Steam velocities in the chamber must be small.
4. End effects must be minimized.
5. The condenser tube must rotate inside the steam chamber.

The most important consideration in designing the apparatus was the need to maintain film condensation on the condenser tube. Drew, Nagle and Smith (5) investigated the conditions for drop-wise condensation and arrived at the conclusions that the type of condensation depends upon the purity of the steam, and the cleanliness and finish of the condensing surface. Film condensation will result if the steam and the surface are clean, regardless of surface finish. Drop-wise condensation will occur if the condensing surface is in some way contaminated, either from the steam or some other source. Substances such as oils, fatty acids, mercaptans and rubber will readily contaminate the condensing surface. Dropwise condensation will form more readily on smooth than rough

surfaces. Photographs of film and drop-wise condensation are shown in Figures 11 and 12, respectively.

Before assembling the present apparatus all the parts were thoroughly cleaned with ammonia soap to remove any oil or dirt. Wherever rubber had to be used, the passage that the steam had to follow to reach the rubber was made as small and constricted as possible.

Before choosing a condenser tube material, experiments were conducted to determine the type of condensation formed on various metal pipes by the steam available from the U.B.C. power house. In these experiments it was found that film condensation occurred on clean aluminum and stainless steel pipes, but not on clean copper or brass pipes. These same findings were obtained by Drew, Nagle and Smith when using steam from a power house. Based on these facts and the previous conclusions of Drew, Nagle and Smith, it is assumed that the steam available from the U.B.C. power house contains some contaminants.

These results also indicate that an aluminum or stainless steel pipe should be used as the condenser tube. Unfortunately, neither of these materials can be readily instrumented with thermocouples for measuring the surface temperature of the tube, since these thermocouples must be soldered in grooves in the tube wall. Consequently, it was necessary to find a method to obtain film condensation on a copper or brass tube.

A bimetallic pipe made from an instrumented copper tube,

metal sprayed with aluminum, gave the required film condensation. This tube maintained film condensation for periods up to forty-eight hours, but had two undesirable features. First, an oxide was formed at the interface between the copper and the aluminum, when the pipe was heated during the metal spraying process. Second, a heavy oxide formed on the aluminum surface during each experiment. As a result, two unknown oxide resistances entered into the values of the heat-transfer coefficient measured using this pipe.

The best method found for inducing and maintaining film condensation on a copper pipe was the use of a wetting agent.[1] Prior to each test the condenser tube was cleaned with steel wool and ammonia soap to remove any oxide formed, or contaminants deposited on the tube during or since the previous test. The condenser tube was then wiped with the wetting agent and assembled in the apparatus. With one application of wetting agent at the start of each test, film condensation was maintained for at least ten hours.

As it was necessary to view the type of condensation formed on the condenser tube, the steam chamber was made from a pyrex pipe. To reduce the steam velocities in the chamber the largest diameter pipe readily obtainable was used. Pipe instead of tubing was used, as fairly high pressures were expected (up to 20 psig).

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[1]The wetting agent used was Kodak Photo-Flo

The type of steam inlet greatly influenced the velocity in the chamber and also the symmetry of the circumferential surface temperature distributions of the condenser tube. Allowing the steam to issue into the chamber from the open end of a pipe produced unfavourable steam velocities and skewed surface temperature distributions. A single straight manifold made from a length of 1/4 in. diameter copper tubing with 11 equally spaced holes along its length was then used to direct the steam perpendicularly against the pyrex pipe wall. A second such manifold diametrically opposite the first gave a further improvement. Finally the baffles shown in Figure 13 were added. This was the best arrangement, but as can be seen in Figures 16 to 19 the circumferential temperature distributions are not perfectly symmetrical.

It was found that the position of the baffles, the number and placement of the holes, the steam pressure, and the angle B between the axis of the holes in the manifold and a diameter of the pyrex pipe all have a strong influence on the steam distribution in the apparatus and the temperature distribution over the pipe surface. The skewness of the temperature distributions increased with increasing angle B and increasing steam pressure.

Sealing the space between the end plates and the condenser tube poses the problems of (1) avoiding contamination of the tube surface, (2) providing for ready rotation of the tube, and (3) minimizing heat conduction between the end plates and the tube. Rubber o-rings and teflon o-rings were tried for this job. Rub-

ber was discarded as it contaminated the tube surface. Teflon did not contaminate the tube surface and was an excellent self lubricated bearing. Thus contamination of the tube surface was overcome and at the same time the tube could be readily rotated. Any heat conduction taking place between the end plates and the condenser tube had to pass through the teflon o-rings, which are good insulators having a thermal conductivity of 0.015 Btu/hr ft F. During the experiments it was found that little or no heat was conducted from the end plates to the condenser tube, but that heat was being conducted outside the apparatus along the copper pipe.

INSTRUMENTATION

The apparatus is equipped with the necessary instruments for measuring temperature, pressure and flow rate. The inlet and exit temperatures of the cooling water to the apparatus are measured with a bimetallic dial thermometer with 1 F subdivisions and a mercury in glass thermometer with 0.5 F subdivisions, respectively. These thermometers, which are calibrated to within ± 0.2 F, are placed in turbulent regions of the water stream in order to avoid incorrect temperature readings due to stratification of the water flow into different temperature zones. Four copper-constantan thermocouples connected in parallel and placed in random positions inside the steam chamber measure the steam temperature. These thermocouples, and the five thermocouples which measure the tube-wall temperature, are connected to a six point Bristol millivolt recorder. The recorder and the thermocouples were calibrated against certified thermometers to an accuracy of ± 0.5 F. The steam pressure is measured with a mercury manometer with 0.1 in. subdivisions. The water flow rate is measured with a rotameter calibrated to an accuracy of ± 0.02 lb/min.

Calculations based on the accuracy of the above calibrations and on experimental values give a maximum error due to calibrations of ± 5 per cent in the measured heat-transfer coefficient.

For heat balances only, the apparatus is insulated and the condensate level is maintained constant in the hot well. The con-

densate is weighed with a scale accurate to ± 0.5 oz. Its temperature is measured with an uncalibrated bimetallic dial thermometer with 2 F subdivisions. The resulting error in using an uncalibrated thermometer will be small compared with the 2 to 3 per cent agreement obtained in the heat balances.

Measuring the surface temperature of a condenser tube is a difficult task. The major difficulty is in obtaining the measurement without disturbing the actual surface temperature, the heat-transfer rate, or the surface conditions of the pipe. The measuring instrument generally changes the conditions at the point of measurement.

In most installations, thermocouples are embedded in grooves in the surface. Patton and Feagon (14) milled a longitudinal groove from the junction of the thermocouple to the outside of the apparatus along the surface of a condenser tube. The junction of a thermocouple was soldered in a chordal hole drilled from the milled slot. The remainder of the thermocouple leads were threaded through a small brass tube which was soldered in the milled groove.

Colburn and Hougen (4) used a 2 in. long narrow slot in an isothermal zone with a thermocouple soldered in the middle of the slot. The leads from the couple were laid in opposite directions along the slot and cemented in with glycerol-litharge which supposedly provides excellent thermal contact but poor electrical contact.

McAdams (11) has reviewed most of the literature on measurement of surface temperatures and suggests on page 199 the following methods of installing thermocouples in walls as satisfactory:

1. A groove is cut in that part of the outer surface later to be located in a substantially isothermal zone, the bare junction is placed in direct contact with the metal wall of the tube, the electrically insulated leads are installed in the grooves so that at least 1 in. of each lead is in the groove, and the grooves are filled with suitable material. If surface conditions are important, as in boiling or condensing, it is advisable to plate the assembly with a suitable coating of metal.

2. The junction is threaded through a chordal hole, each lead is submerged in a circumferential groove for at least 1 in., and the groove is filled as described above. This method does not disturb the surface of the metal near the junction. A modification of this method involves placing the junction in a hole drilled at an angle to the axis of the tube.

In the present apparatus the tube-surface temperature is measured with five thermocouples buried at known locations in the tube wall (see Figure 14). By rotating the tube five circumferential temperature distributions are obtained. With these temperature distributions the average tube-surface temperature was determined. Leeds and Northrop, 30 gauge, enamelled, copper-constantan thermocouple wire, insulated with Fibreglas, was used. The thermocouple leads were laid in milled grooves in the tube wall for a distance of 1 3/16 in. before entering the water stream through a hole at one end of the groove (see Figure 15). The leads followed the water stream until they were outside the steam chamber; then they were brought through the tube wall. A length of 14 gauge copper wire was placed on top of the thermocouple leads in the groove and soldered in place. The solder and copper wire were filed flush with the surface of the tube, and thus alteration of surface finish of the tube was minimized.

The thermocouple junction was formed by the solder which

flowed in between the thermocouple leads. This method of forming the junction was used in order to prevent the possibility of air pockets forming near the junction and thus affecting the temperature reading of the thermocouple. In previous installations by the author, junctions had been made by twisting and soldering the leads together before soldering them into the slot. With this method air pockets had formed in the space between the leads and the junction, as the junction and immediately surrounding insulation prevented the solder from flowing between the leads.

The law of intermediate metals for thermocouples states that, "the algebraic sum of the thermoelectromotive forces in a circuit composed of any number of dissimilar metals is zero, if all of the circuit is at a uniform temperature."[1] Therefore, the method used for making the thermocouple junctions should introduce negligible error provided the temperature difference between the point of contact of the copper and solder, and the point of contact of the constantan and solder is small (i.e. the slope of the temperature curve is small). From Figures 16 and 17 it is seen that the only locations where there are substantial changes of temperature are at $\phi = 135°$ and $\phi = 225°$.

At first it was suspected that soldering the thermocouple leads into the groove might destroy the Fibreglas insulation. This was not the case. In fact the insulation stood up very well and still retained an infinite electrical resistance.

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[1]Reference (1), p. 181.

The greatest error which might result from the present installation is the reduction in the thermocouple temperature due to heat conduction from the junction to the water stream. McAdams (11), on page 264, gives error curves for various types and gauges of bare thermocouple wires attached to the heat-transfer surface of a counter flow heat exchanger handling hot and cold air at 1000 F and 100 F, respectively. No literature could be found which dealt with the above error for insulated thermocouple leads.

To reduce the effect of heat conduction the leads were insulated and buried in grooves for 1 3/16 in. The thermocouple wire used was covered with two layers of Fibreglas insulation, one layer over each lead and then a common layer over both leads. The common layer was removed from the leads in the slot for reasons of space requirements, but the two layers of insulation were retained where the leads passed through the water stream.

McAdams' recommendation that the thermocouple leads be buried in an isothermal zone is impossible to obtain in the present apparatus as the isotherms on the pipe surface vary with the angle of inclination of the pipe and the rate of heat transfer. Consequently, straight longitudinal grooves, which are the easiest to make and instrument, were used. Also the leads were buried in the grooves so that they enter a higher temperature region before entering the cold water stream, thus tending to further reduce the conduction error.

Another problem encountered resulted from variations in the

heat-transfer rate as the condenser tube was rotated. The theory predicts that on a horizontal tube the greater percentage of the heat is transferred on the upper half of the tube. In a previous installation the thermocouples were placed in a line down the length of a tube, and significant variations in the heat-transfer rate were found as the tube was rotated. These variations are due to the resistance to heat transfer created by the solder and insulation in the thermocouple grooves. The heat-transfer rate will decrease as the thermocouples are moved from the bottom of the pipe to the top of the pipe. These variations will also decrease with increasing angle of inclination of the apparatus. For a vertical pipe no variation of the heat-transfer rate should result as the tube is rotated. These variations in the heat-transfer rate were reduced in the present apparatus by locating the thermocouples 90° apart circumferentially as shown in Figure 14.

As well as causing variations in the heat-transfer rate the thermocouple installations will also reduce the rate of heat-transfer. To keep this reduction in the heat-transfer rate to a minimum, the smallest grooves that could be readily instrumented were used. These grooves still satisfied McAdams' recommendation that at least 1 in. of each thermocouple lead be in the groove. In the present installation, a 2.8 per cent reduction in the heat-transfer rate will result if there is no heat transfer through the thermocouple grooves. Experiments were conducted to determine the exact reduction in the heat-transfer rate; the results will be presented later.

RESULTS

The Tube-Surface Temperature Distribution

Figure 16 shows the circumferential temperature distributions at five thermocouple locations along the condenser tube for a zero angle of inclination. Figures 17 and 18 give the corresponding distributions at 45° and 90° inclinations. All figures are for essentially the same mean temperature difference (about 37 F). The graphs for 0° and 45° are quite similar. Both exhibit a decrease in the local surface temperature as the angle of rotation, ϕ , from the top of the tube increases from 0° to 180°. This decrease is due to the increase in the thickness of the condensate film on the tube with an increase in the angle of rotation.

These figures also show that the symmetry which might be expected about the 0-180° axis is not attained exactly. In the vertical position the temperature of the tube surface is lower than for the previous inclinations. Some degree of angular symmetry of the temperatures is shown. This symmetry is upset, as shown in Figure 19, when the mean temperature difference is increased. The skewness is caused by the higher steam velocities which in turn are caused by the increased steam pressure and heat-transfer rate, which accompany an increase in the mean temperature difference. Note also that the tube-surface temperature is increased when the mean temperature difference is increased.

Figure 20 shows the average circumferential temperatures for the preceding results. Here again, the effects of varying the

angle of inclination and the mean temperature difference are evident.

The Heat-Transfer Coefficients

Actual and theoretical heat-transfer coefficients were calculated for each test run. A sample calculation of the coefficients is given in Appendix I, and a list of results is given in Table IV, Appendix I.

Figure 21 is a plot of the actual heat-transfer coefficient against the mean temperature difference between the steam and the tube surface, for various angles of inclination. At each inclination a straight line relation was obtained between the coefficient and the mean temperature difference using the method of averages described in Appendix I. The computed curves are shown as solid lines and the experimental points are grouped around them with a maximum scatter of ± 3.5 per cent. The equations for the various curves are listed in Table V, Appendix II.

Two solid lines are shown for the vertical tube. The lower line was obtained from the experimental data of the initial test runs. After the test program was completed for all the angles of inclination, the initial runs were repeated, five weeks later, to test for aging and other effects. The curves obtained for these latter runs averaged 2.7 per cent higher than the initial curves.

The negative slopes of the results agree with Figure 6, which shows that the heat-transfer coefficient is inversely proportional to the fourth root of the mean temperature difference.

This assumes that the other variables are constant. For the present experiments this is not true, but the changes in the mean temperature difference will have a much larger influence than those of the other variables.

The straight line relations between the heat-transfer coefficient and the mean temperature difference were used to develop general equations correlating the experimental results (see Appendix I for sample calculation). These equations are

$$h_A = (0.304 \alpha + 10.2)(28.9 - \Delta t_m) + 2000$$

for inclinations from 0° to 30° and for 45° to 90°

$$h_A = (0.286 \alpha - 26.3)(\Delta t_m - 70.3) + 1340$$

No general equation could be developed for the region from 30° to 45°, as sufficient data were not available. Straight line relations developed from these general equations are shown as dashed curves in Figure 21 and are listed in Table V, Appendix II. The maximum scatter of the experimental points about these curves is ± 5 per cent.

Ratios of the actual to theoretical coefficient for given inclinations are plotted in Figure 22 against the mean temperature difference. For inclinations from 0° to 60° the ratio is relatively constant between 0.93 and 0.98; for 75° the ratio averages 1.01 and for 90°, 1.09. As before, straight line equations for each inclination were calculated and are plotted as solid lines (see Table V, Appendix II). The maximum scatter of the experimental points about these lines is ± 3.5 per cent. The closing

results are 2.4 per cent higher than the initial results.

The results of Figure 22 for inclinations from 0° to 30° can be correlated by the equation

$$h_A/h_T = (0.177\alpha - 1.32)(32.2 - \Delta t_m)10^{-3} + 0.964$$

and for inclinations between 45° and 90° by

$$h_A/h_T = (0.128\alpha + 1.32)(\Delta t_m - 14.3)10^{-3} + 0.972$$

The equations of the dashed curves in Figure 22 were obtained from the above general equations and are listed in Table V, Appendix II. The experimental points are grouped around these lines with a scatter of ±7 per cent.

The usefulness of these results is limited by the fact that the surface temperature of the condenser tube must be known, before they can be applied to another geometrically similar tube. This temperature is generally not measured in industrial applications. Therefore, in order to make the results more useful, they have been replotted against the readily obtainable overall temperature difference.

In Figure 23 and 24, respectively, values of the actual coefficient and values of the ratio of the actual to theoretical coefficient are plotted against the overall temperature difference between the steam temperature and the arithmetic average of the inlet and exit water temperatures. The curves obtained using this new abscissa have the same general characteristics as those of Figures 21 and 22.

Straight line equations for the results of each inclination

are listed in Table VI, Appendix II. The following general equations correlating the results apply between 0° and 30°

$$h_A = (0.184\,\alpha + 5.94)(153.1 - \Delta T) + 1950$$

and

$$h_A/h_T = (0.104\,\alpha - 0.767)(153.0 - \Delta T)10^{-3} + 0.966$$

Between 45° and 90°

$$h_A = (0.169\,\alpha - 15.6)(\Delta T - 213.0) + 1330$$

and

$$h_A/h_T = (0.075\,\alpha - 4.47)(\Delta T - 117.0)10^{-3} + 0.982$$

The maximum scatter of the experimental points about the curves in Figures 23 and 24 is ±6 per cent.

Another comparison which can be made between the actual and theoretical coefficients is shown in Figure 25. Here, the dimensionless number $h_T(3\mu r\Delta t/\rho^2 g\lambda K^3)^{1/4}$ is plotted against the inclination. A theoretical curve and an average experimental curve are shown. Each exhibits a maximum at an inclination of 0° and an increasing negative slope with increasing inclination. From 0° to 74° the experimental curve is lower than the theoretical with a maximum difference of 5 per cent. Above 74° the experimental curve exceeds the theoretical with a maximum difference of 8 per cent at 90°.

The Heat-Transfer Rates

Figure 26 presents the heat-transfer rate data for test pipe VII, the pipe which was used to obtain all the preceding data. A question arose as to the validity of these results, since the

five tube-wall thermocouples were embedded in 2 in. long grooves in the tube surface and soldered in place. The transfer of heat through solder, thermocouples and insulation can not be the same as that through the undisturbed copper. Therefore, test pipe X was made. This pipe is identical to test pipe VII except that the tube-wall thermocouples are soldered in small radial holes in the tube wall instead of in grooves. These holes are in the same relative positions as those in test pipe VII. As the thermocouples in the wall of test pipe X will have negligible effect on the heat-transfer rate, it should be possible to determine the decrease in the rate of heat transfer through test pipe VII because of the solder, thermocouples and insulation.

Figure 27 presents the results obtained from test pipe X. As expected the heat-transfer rates for test pipe X are higher than those of test pipe VII. The minimum, average and maximum per cent increases in the heat-transfer rate are 2.6, 9.1 and 14.7, respectively.

In Appendix I an analysis based on the assumption of no heat transfer through the grooves in test pipe VII, predicts that the heat-transfer rate for test pipe X should be 2.8 per cent greater than that of test pipe VII. This is considerably less than the average experimental value. It is possible test pipe VII has a larger scale resistance on the inside surface than test pipe X, thus lowering the heat-transfer rate.

Straight line equations for the experimental results were

calculated using the method of averages. These equations are plotted in Figures 26 and 27 and are listed in Table VII, Appendix II. The maximum scatter of the experimental points about these straight lines is ± 7.0 per cent.

DISCUSSION OF RESULTS

In this section a general discussion of the results and the assumptions of the theory will be presented.

The Average Tube-Surface Temperature

A major item to be considered in the results is the average tube-surface temperature. This temperature was obtained with five thermocouples buried at known locations in the tube (see Figure 14). It will be noticed from Figure 14, that the thermocouples will only enable the calculation of the average surface temperature for a 9 in. length within the total 12.45 in. test length of the apparatus. Thus, the surface temperature distributions of the 2 in. length of tube between end plate 1 and thermocouple A and of the 1.5 in. length between thermocouple E and end plate 2 are unknown. Originally, in an attempt to eliminate end effects, it was planned to determine the heat-transfer coefficients for the above 9 in. length. To do this, it was necessary to measure the cooling water temperatures at thermocouples A and E. Thermocouples for measuring these temperatures were mounted in the water stream, but they gave unsatisfactory results. Instead, the inlet and exit water temperatures were used in the calculations. Consequently, the results include the end effects.

The influence of the end effects is shown in Figures 28 and 29. Here, curves of the average circumferential surface

temperatures for the horizontal and vertical positions are presented. The end effects, which are characterized by the sudden drops in the curves near the end plates, are the result of heat being conducted outside the apparatus along the condenser tube before entering the water stream. From Figures 28 and 29 it is possible to determine the change in the results due to end effects and due to using an average surface temperature for the 9 in. length. If there were no end effects, the temperature curves would extend to the end plates as shown by the dashed lines. The average surface temperature would then be larger than the true average for the entire tube; the mean temperature difference would decrease and also the heat-transfer rate, as can be seen from the graph of the heat-transfer rate versus the mean temperature difference in Figure 30. Calculations based on Figures 28, 29, and 30 indicate that the coefficient using the 9 in. average surface temperature is about 1.5 per cent lower than that obtained when the average surface temperature for no end effects is used. The coefficient for the true average temperature is about 3 per cent lower than that for no end effects (see Appendix I). Thus, the 9 in. average surface temperature is closer to the theoretical than is the true average surface temperature.

In Figure 28 the average circumferential temperature distribution for the horizontal position exhibits a dip in the region read by thermocouple D. This dip does not appear in Figure 29, but did appear in most of the previous results for test pipe

VII. For the initial runs, which were conducted at 90°, no dip was apparent, but as the experiments were continued at different inclinations a dip began to appear and to grow with time and the angle of inclination up to 75°. The closing results at 90° also exhibited a dip, but not as pronounced as at the other inclinations. This dip is also evident in Figures 16 and 17 for the circumferential temperature distributions. In these Figures the temperature curve of thermocouple E occasionally exceeds that of thermocouples C and D. In Figure 20, the curves of the average circumferential temperature for the results of Figures 16 and 17 clearly illustrate the dip. No definite explanation can be given by the author for the cause of this dip.

Two methods of averaging were utilized when calculating the average tube-surface temperature. An arithmetic average was used to obtain the average circumferential temperature at each thermocouple location. These averages were plotted against the thermocouple locations along the pipe as in Figure 20. Then the average tube-surface temperature was obtained by planimeter from the resulting graphs. An arithmetic average, which is much simpler to perform, was originally employed for the second average, but because of the previously mentioned dip in the temperature curves this method introduced a systematic error of 1 to 2 per cent in the actual heat-transfer coefficient. Negligible error resulted from using an arithmetic average for the circumferential temperature distributions.

Fluctuations of ± 0.3 F were frequently observed in the thermocouple readings. Maximum values of ± 1 F were reached at higher heat-transfer rates.

Since the junctions of the tube-wall thermocouples are 0.030 in. beneath the surface, the average tube-wall temperature was corrected to read the surface temperature. The applied correction was obtained from the following equation:

$$q = KA\,\Delta T/\Delta r$$

where ΔT is the correction

Δr = 0.030 in.

K is the thermal conductivity of the material above the junction. It is impossible to determine the exact thermal conductivity of the material above the junction, as this material is a combination of solder and copper. In the calculations the thermal conductivity of copper (220 Btu/hr ft F) was used. The true thermal conductivity will be less than this, thus the correction applied was low, making the mean temperature difference higher and the heat-transfer coefficient lower than the actual values.

The Heat-Transfer Coefficients

In general the heat-transfer coefficients are consistent, agree closely with the theory and are readily correlated. Reproducibility of the results was good. For a given inclination it was possible to reproduce a previous day's results to within ± 2.5 per cent. After five weeks of testing the maximum differ-

ence between the closing and the initial results was only 5 per cent.

The straight line equations developed from the experimental points and also from the general correlations satisfy the experimental results. However, the general equations for correlating the results give little information for inclinations between 30° and 45°. Figure 31 shows the graph of the intercepts versus the slopes of the straight line equations of the heat-transfer coefficient with respect to the mean temperature difference. For inclinations from 0° to 30° and from 45° to 90° the points form two straight lines which have an intersection between 15° and 30°. The plot of the slopes of these straight lines versus the pipe inclination, also in Figure 31, indicates an intersection at an inclination of 27°. A similar situation exists throughout the entire test program. Consequently, it has been necessary to divide the field into two test zones - low pipe inclinations up to about 30° and high inclinations above say 45°. General correlations have been obtained for each zone in addition to the individual relations for each pipe inclination. Further experimental points are required before the intervening region can be satisfactorily correlated.

The agreement between the actual and theoretical coefficients is very good, as was seen in Figures 22, 24, and 25. For inclinations up to 60° the ratio of the actual to the theoretical coefficient is relatively constant at approximately 0.95; for 75°

and 90° the average values are 1.01 and 1.09, respectively. It is suspected that the increase in the ratio with increase in the inclination above 60°, is due to an increase in the intensity of the ripples in the condensate-film flow. The behaviour of these ripples and their effect on the heat-transfer coefficient will be considered in the discussion of the theoretical assumptions.

Comparison of these experimental heat-transfer coefficients with those of other investigations is difficult, as the coefficient is dependent upon the length and diameter of the tube and the mean temperature difference. Data for conditions similar to those of the present experiments are non-existent. In a later section comparisons will be made between the ratios of actual to theoretical coefficients for the present experiments and those of other investigators.

A comparison of the effect of the angle of inclination on the heat-transfer coefficient can be made between the present results and those of Tepe and Mueller (17). They found that the angle of inclination had only a slight effect on the coefficient, whereas the present results indicate that the angle of inclination has a large effect.

The Theoretical Assumptions

The theoretical development for the heat-transfer coefficient for an inclined cylinder was based on several assumptions. In this section each assumption will be analyzed and its effect on the coefficient will be discussed.

Assumption 1. The vapour is pure, dry, and saturated. In the present work, it is probable that a small amount of air was present in the steam. Othmer (12) conducted experiments on the effect of the concentration of air on the measured heat-transfer coefficient and found that as the concentration increased, the measured coefficient decreased, as shown in Figure 32. Boelter and others (3), on page XV-2, explain the effect of the air as follows:

> When the mixture of a single condensable vapour in a non-condensable gas is in contact with a wall at a temperature below the dew point, some condensation occurs, and a film of liquid is formed on the cooling surface. A film of non-condensable gas and vapour collects next to the condensate layer, the concentration of vapour in the gas film being lower than in the main body of the mixture. Due to the difference in partial pressure of the vapour in the main body of the mixture and at the interface between the condensate layer and the gas film, the vapour diffuses from the main body through the gas film to liquify at the interface. The rate of condensation is thus governed by the laws of diffusion of vapour through a film of non-condensable gas.

Examination of the present experimental data indicated possible air concentrations of 1.35 per cent (see Appendix I for calculations). From Figure 32 the above air concentration would reduce the heat-transfer coefficient for a horizontal cylinder by about 45 per cent.

The presence of air in the steam would explain the low values of the ratio of actual to theoretical coefficient obtained in the experiments, compared with those of other investigations. For condensation of steam on horizontal cylinders, McAdams (10), on page 263, lists ratios of 0.7 to 2.2 for corresponding mean

temperature differences from 2 F to 66 F. In the present investigation, ratios of 0.96 to 0.98 were obtained for corresponding mean temperature differences of 27 F to 43 F. Also on page 263, McAdams lists, for condensation on vertical cylinders, average ratios of 1.5 to 1.7 for mean temperature differences from 5 F to 48 F. The present results were 1.05 to 1.12 for mean temperature differences of 33 F to 49 F.

Correcting the present results for the effect of the air will raise the ratios to average values of 1.94 and 2.18 for the horizontal and vertical positions, respectively. These values are not too reliable as the method of measuring the concentration of air in the steam was not very accurate.

One further observation can be made. As can be seen from Figure 32, a small change in the concentration of the air will cause a large change in the heat-transfer coefficient, but as reproducibility of the results was good the concentration of the air must have been constant.

Temperature and pressure measurements taken during the experiments indicate that for low heat-transfer rates the steam was possibly wet, and that for high heat-transfer rates the steam was superheated. Preliminary tests of the steam quality gave a minimum value of 97 per cent. It is shown in Appendix I that for this steam, the theoretical heat-transfer coefficient would be approximately 0.8 per cent high.

At this point it should be stated that the theoretical

derivation also assumes no subcooling of the condensate. Subcooling was obtained in all the experiments. It is shown in Appendix I that the amount of subcooling occurring during low heat-transfer rate runs would make the theoretical coefficient approximately 0.4 per cent low. Thus, the combined effect of neglecting the possibility of wet steam and of subcooling the condensate makes the theoretical coefficient approximately 0.4 per cent high for the low heat-transfer rate runs.

At the high heat-transfer rates, both superheating and subcooling were evident. For the worst case the theoretical coefficient is 0.6 per cent low, as can be seen from the calculations in Appendix I.

Assumption 2. The condensate-film flow is laminar. The validity of this assumption depends upon the Reynolds number of the flow. The maximum Reynolds number, 284, obtained during the experiments occurred at the bottom of a vertical tube. As the critical Reynolds number is about 2000 the condensate-film flow was in the laminar region, even though ripples were observed in the flow. Further discussion of these ripples will follow in the consideration of Assumption 5.

Assumption 3. The vapour at the vapour-liquid interface is stagnant and the shear stress is negligible at this interface. Even though a large diameter pyrex pipe was used as the steam chamber, the skewness of some of the surface temperature

distributions, especially at high heat-transfer rates, and the influence of the steam inlet on the same temperature distributions suggest that the vapour at the vapour-liquid interface was not stagnant.

McAdams (11), on page 336, states that for downward flow of condensing vapour at high velocity inside a tube, the measured heat-transfer coefficient may be as much as ten times the theoretical value. For upward flow of vapour in a vertical tube, the condensate film thickness is increased and thus the measured coefficient will be lower than the theoretical value. As neither the magnitude nor direction of the steam velocity is known in the present work, it is impossible to predict the effect of the same on the measured coefficient. However, every effort was made to minimize this effect by employing two baffled manifolds, as shown in Figure 13, to distribute the steam uniformly, and to reduce its velocity.

Assumption 4. The wall temperature is uniform. In practice the wall temperature varies both circumferentially and longitudinally. To obtain some knowledge of the effect of assuming an isothermal tube, consider another paper by Kamal-Eldin Hassan (8) in which theoretical results for non-isothermal surfaces are given.

For condensation on an inclined flat plate with temperature variation along its length only, the theory predicts that the heat-transfer coefficient is equal to that of a similar plate

with a uniform temperature equal to the integrated mean temperature of the non-isothermal plate.

Also, for an infinitely long inclined cylinder with only circumferential temperature variation, the heat-transfer coefficient is practically the same as that for a similar isothermal cylinder at the integrated mean temperature of the non-isothermal cylinder.

Qualitatively extending Hassan's results to the tube with both circumferential and longitudinal temperature variation, the heat-transfer coefficient for the non-isothermal tube should be substantially the same as that for an isothermal tube at the mean temperature of the non-isothermal tube.

Assumption 5. The liquid-solid and liquid-vapour interfaces are smooth. The first part of this assumption was attained by using a smooth condenser tube. The second part was not attained as ripples were apparent in the condensate-film flow.

The character of the liquid-vapour interface depended upon the inclination of the tube, the position on the tube surface and the heat-transfer rate. For a horizontal tube the condensate-film flow was smooth (see Figure 11). At an inclination of 15°, ripples were apparent in the condensate-film flow on the under surface of the tube, but not in the flow on the top surface. As the inclination of the tube was increased to 75° the same behaviour occurred, except that the ripples increased in intensity with the angle of inclination. At 75° ripples began

to appear in the condensate-film flow on the top surface of the tube. For a vertical tube, ripples were apparent in the entire film flow, as expected. At a given inclination, increasing the heat-transfer rate increased the intensity of the ripples.

Grimely (7) observed ripples in a liquid-film flow at a Reynolds number of 25, but says that the motion of the flow was quite regular in nature for Reynolds numbers up to 1000. On increasing the flow rate further a transition region appeared after which a more agitated flow suggesting turbulence occurred.

In the present experiments ripples were observed within a few inches of the top end of a vertical tube. Here, the Reynolds number would be close to the lowest Reynolds number at which Grimely first observed ripples to occur.

Further work on ripples in liquid-film flows is presented by Dukler and Bergelin (6) who developed new equations relating the liquid-film thickness in concurrent, gas-liquid flow to the flow rates, physical properties of the fluids and to the energy loss. These equations agree with Nusselt's film thickness equation for Reynolds numbers up to 1000, which is given by the new equations as the critical Reynolds number. Experimental results are shown which confirm the equations and verify the observations of other investigators that in spite of ripples in the condensate film, the Nusselt equation predicts accurate values of the mean film thickness in the viscous region.

As a result of the work of Grimely, Dukler and Bergelin,

it is quite certain that the condensate-film flow in the present experiments was laminar in spite of the ripples. Just the same the ripples will have an effect on the measured heat-transfer coefficient. The effect should be to increase the coefficient due to the increased surface area available for heat transfer.

Hassan and Jakob (9) cite calculations which predict that for a Reynolds number of 785 the heat-transfer coefficient of a rippling film will be nearly 50 per cent larger than that of an equivalent smooth film. In the present experiments a maximum Reynolds number of 284 was obtained, thus indicating that the effect should be considerably less than the above.

Experimental ratios of the actual to the theoretical coefficient are relatively constant for inclinations from 0° to 60°, tending to indicate that the effect of ripples was constant for these inclinations. Yet, there were no ripples in the condensate-film flow for the horizontal position, thus, it would appear that the ripples had negligible effect for inclinations up to 60°. From 60° to 90° the ratio gradually increased up to 11 per cent. For the vertical position, at which the ripples are the most intense, their presence increased the measured coefficient by no more than 11 per cent.

Assumption 6. The curvature of the surface is large enough that the effect of capillary forces may be neglected. This assumption need only be made for condensation on the inside surface of a tube and not for condensation on the outside surface,

as in the present investigation.

Assumption 7. The liquid temperature at the liquid-vapour interface is that of the saturated vapour. In the calculations it was assumed that the liquid at the liquid-vapour interface achieved the temperature of the vapour, whether saturated or superheated. With this assumption the heat-transfer coefficient between the steam and the condensate is included in that between the steam and the tube surface.

Assumptions 8 and 9. Acceleration and convection effects are negligible. Two papers by Sparrow and Gregg dealing with laminar film condensation on a horizontal cylinder (15) and on a vertical plane (16) are pertinent. Acceleration forces and energy convection terms are included in the developments, which are based on the techniques of boundary layer theory.

In reference (15) Sparrow and Gregg state that energy convection adds to the heat transfer by conduction, while inertia forces decrease the heat transfer. For high Prandtl number fluids, where the viscous forces override the inertia forces, the effects of energy convection win out and the heat transfer is higher than for the low Prandtl number fluids. In low Prandtl number fluids, energy convection is small and the effects of inertia forces win out. The effects of inertia forces are negligible for a fluid with a Prandtl number over 100.

In the present results, neglecting inertia forces and

energy convection terms for the vertical and horizontal cases reduces the theoretical coefficient by 1 and 2 per cent, respectively. It is reasonable to assume that the effect will be of the same order of magnitude for an inclined cylinder.

Assumption 10. The physical properties of the condensate are constant. This is a customary assumption to permit the use of arithmetic averaging. It is not unreasonable as the curves of the physical properties with respect to temperature are nearly linear for the temperature ranges obtained in the experiments.

Assumption 11. The temperature distribution through the condensate film is a straight line. This assumption has been checked against the work of Sparrow and Gregg (16). For the conditions of this test program their curves show that the temperature distribution through the condensate film is linear, when the pipe is vertical. They gave no data for the temperature distribution through the condensate layer on a horizontal cylinder.

In the foregoing analysis of the assumptions only two of them introduce the possibility of large discrepancies between the actual and theoretical heat-transfer coefficients. The discrepancies resulting from the remaining assumptions are within the $\pm$ 5 per cent accuracy with which the coefficients can be measured. The two doubtful assumptions assume air-free steam and no ripples in the flowing condensate film. The presence of

air in the steam will greatly reduce the heat-transfer coefficient, whereas ripples in the condensate-film flow will increase the coefficient. Although these effects are large, they are relatively uncertain in the present experiments, and as a result it is necessary to use care when applying these results.

CONCLUSIONS

The following conclusions apply only in the regions tested.

1. The actual heat-transfer coefficient and the ratio of the actual coefficient to the theoretical are dependent upon the tube inclination and the mean temperature difference between the steam and the tube surface. No single equation correlating the entire region of inclinations could be obtained, but general equations correlating these variables apply for each of two zones. For inclinations from 0° to 30°

$$h_A = (0.304\alpha + 10.2)(28.9 - \Delta t_m) + 2000$$

and

$$h_A/h_T = (0.177\alpha - 1.32)(32.2 - \Delta t_m)10^{-3} + 0.964$$

and for inclinations from 45° to 90°

$$h_A = (0.286\alpha - 26.3)(\Delta t_m - 70.3) + 1340$$

and

$$h_A/h_T = (0.128\alpha + 1.32)(\Delta t_m - 14.3)10^{-3} + 0.972$$

Further data is required before the region between 30° and 45° can be satisfactorily correlated. When values of the inclination were substituted into the general equations, the resulting linear relations were in agreement with the experimental values to within ± 7 per cent. As the experiments were conducted for mean temperature differences from 27 F to 49 F, the general equations can only be applied with certainty within this range.

More useful correlations can be developed using the overall temperature difference between the steam and the coolant

instead of the mean temperature difference between the steam and the tube surface. As before, general correlations employing this new variable apply in each of two zones. For inclinations from 0° to 30°

$$h_A = (0.184\alpha + 5.94)(153.1 - \Delta T) + 1950$$

and

$$h_A/h_T = (0.104\alpha - 0.767)(153.0 - \Delta T)10^{-3} + 0.966$$

and for inclinations from 45° to 90°

$$h_A = (0.169\alpha - 15.6)(\Delta T - 213.0) + 1330$$

and

$$h_A/h_T = (0.075\alpha - 4.47)(\Delta T - 117.0)10^{-3} + 0.982$$

Linear equations developed from these general equations were in agreement with the experimental values to within ± 6 per cent. These relations apply with certainty only within the region tested (i.e. for overall temperature differences between 142 F and 172 F).

2. As previously stated the tube inclinations, the mean temperature difference and the overall temperature difference all have a marked effect on the heat-transfer coefficient. Increasing these variables decreases the coefficient. For inclinations up to about 45° the effects of changing either temperature difference are greater than that of changing the inclination. Above 45° the reverse is true.

3. The slopes of the straight line relations for the heat-transfer coefficient with respect to the mean temperature differ-

ence, and with respect to the overall temperature difference, are all negative. The minimum and maximum negative slopes occur for inclinations of 90° and 30°, respectively. From 90° to 30° the slopes become increasingly negative. Below 30° the slopes become smaller again.

4. The angle of inclination, the mean temperature difference and the overall temperature difference all have a slight effect on the ratio of the actual to the theoretical coefficient. For inclinations from 0° to 60°, the average ratio of the actual to theoretical coefficient is relatively constant at 0.95. From 60° to 90° the average ratio increases from 0.97 at 60°, to 1.01 at 75° and then to 1.09 at 90°. As can be seen from the above, the agreement between the actual and theoretical coefficients is good.

5. Only two of the theoretical assumptions will greatly affect the agreement between the actual and theoretical coefficients. These are: that the steam is pure and that the liquid-vapour interface is smooth. The presence of air in the steam and, the presence of ripples in the condensate-film flow will produce a discrepancy between the observed and the theoretical values. As previously stated, the concentration of air present could lower the actual coefficient by as much as 45 per cent, while ripples in the condensate-film flow will increase the coefficient. The exact magnitude of the increase in the coefficient due to ripples is not known, but it should be less than 50 per cent for

the present experiments (9).

6. Film condensation of slightly contaminated steam on brass or copper pipes can be maintained for periods of at least ten hours if (a) the surface of the pipe is clean, and (b) the surface of the pipe is treated with an appropriate wetting agent. Film condensation of slightly contaminated steam will form on clean aluminum or stainless steel pipes without the use of a wetting agent.

7. The magnitude and direction of the steam velocity in the steam chamber has a large effect on the symmetry of the circumferential temperature distributions on the tube surface. Increasing the steam velocity increases the skewness of the temperature distributions. The magnitude and direction of the steam velocity is strongly dependent on the type of steam inlet and its location.

RECOMMENDATIONS

The present investigation has added a small amount of information to the data available on heat-transfer coefficients for condensing steam on inclined tubes. There is still much that can be done, and should be done, in order that more comprehensive experimental data may be available. Therefore, the following recommendations are presented in the hope that they may aid future investigations.

Film Condensation

Maintaining film condensation on the condenser tube is a fairly difficult task. As was previously stated, pure steam and a clean condenser surface are almost a must for film condensation. In the present experiments film condensation of slightly contaminated steam was promoted and maintained for periods of at least ten hours on a clean copper tube treated with a wetting agent. In future work a small boiler using distilled water is recommended. In this way pure steam would be available, so long as the apparatus itself did not contain any contaminants.

Tube-Wall Thermocouple Installation

The present tube-wall thermocouple installation is not the best available. At a late date in the experiments, knowledge was gained of an improved thermocouple installation using coaxial thermocouple wires [1]. Copper-constantan coaxial thermo-

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[1]Distributors, Philips of Canada Ltd.

couple wires consist of a constantan wire covered with a ceramic or inert metal oxide insulation, which is in turn covered with a copper sheath. Outside diameters down to 0.5 millimeters are available. These wires are very durable and can be readily formed.

The recommended method of installation is to solder the thermocouple wire in a groove extending from the thermocouple junction to the outside of the apparatus. For a small wire soldered flush with the surface, the difference between the surface temperature and that read by the thermocouple would possibly be small enough to neglect. Several of the problems in the present installation would be solved. There would be no possibility of air pockets near the thermocouple junction or of insulation breakdown of the thermocouple leads, heat conduction effects would be negligible and the resistance to heat-transfer through the thermocouple grooves would be small.

Steam Velocity

The steam velocity has a variable effect on the heat-transfer coefficient, depending upon its direction with respect to that of the condensate-film flow. In addition, the rate of heat transfer, and the sizes of the steam chamber and steam inlet all influence the steam velocity. Reducing the heat-transfer rate, increasing the size of the steam chamber and increasing the cross sectional area of the steam inlet all reduce the steam velocity. Therefore, in future work the instrumentation should be accurate

enough for measuring low heat-transfer rates and the steam chamber and inlet should be as large as possible.

End Effects

The influence of end effects, which was shown in a previous section, could be reduced with a longer tube. Another method would be to measure the heat-transfer coefficient over a length of the condenser tube that is not affected by end effects. Here, difficulties of instrumentation are encountered.

By using tubes of various lengths it might be possible to obtain a more accurate determination of the influence of end effects. For a very long tube these effects will be negligible, while for a short tube they will be significant. By comparing the ratios of the actual to theoretical coefficient for both tubes some knowledge of the influence of end effects might result. This method is dependent on maintaining similar conditions for both tubes, a requirement which might be quite difficult to provide.

Air

The presence of air in the steam, as per previous discussion, can greatly affect the measured heat-transfer coefficient. Therefore, it is imperative that steps be taken to obtain air-free steam or that some accurate means of measuring the air concentration be devised. In this respect Othmer (12) is very helpful. He used distilled water and a small boiler to obtain

air-free steam. In another paper (13) a description of his air manometer for measuring the concentration of the air is described.

Range of Values

In the present work the range of values of the heat-transfer coefficient was restricted by the limited range of mean temperature differences that could be attained. The major variable controlling the available mean temperature difference is the steam pressure. The range of steam pressures that could be obtained was from 0 to 6 psig. Pressures below atmospheric would have required more complicated apparatus than was available. The temperatures of steam above 6 psig. could not be measured accurately with the instruments available. In future experiments due consideration should be given to these limitations.

APPENDIX I

SAMPLE CALCULATIONS

Calculation of the Average Heat-Transfer Coefficient for the Entire Surface of an Inclined Cylinder.

This coefficient was calculated using numerical techniques on the values in Table I. In this table the values of the dimensionless mean circumferential heat-transfer coefficient for an infinitesimal ring of condensate are listed for corresponding reduced distances from the top end of an inclined cylinder. The reduced distance of any ring of condensate is given by the equation

$$X = x/r \tan \alpha$$

and the reduced length of the entire cylinder is given by

$$X_L = L/r \tan \alpha$$

For a cylinder with a reduced length X_L, the area under the curve of the dimensionless average circumferential coefficients (see Figure 5) was determined from the values listed in Table I by integrating between limits X = 0 and X = X .

Table I gives no data on values of X between 0 and 0.05 so this portion of the area is indeterminate from the table. Kamal-Eldin Hassan evaluated this area by the following method:

> At each cross section there is an angle $\phi = \phi'$ such that $h_m(X) = h(X,\phi)$. It was noted that $\phi' \rightarrow \pi/2$ as $X \rightarrow 0$. The change in ϕ' is slow, in fact, for $X = \infty$, $\phi' = 110$. Further, from the table[1], it may be noted that for small values of X,

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[1]Table 1 in Reference (9).

TABLE I

THE DIMENSIONLESS AVERAGE CIRCUMFERENTIAL HEAT-TRANSFER COEFFICIENT FOR DIFFERENT REDUCED DISTANCES FROM THE UPPERMOST END OF A CYLINDER

Coefficient $h_m(3\mu r \Delta t/\rho^2 g \lambda K^3 \cos\alpha)^{\frac{1}{4}}$	Reduced Distance $X = x/r \tan\alpha$
1.9694	0.05
1.6559	0.19
1.4965	0.15
1.3932	0.20
1.2603	0.30
1.1762	0.40
1.0668	0.60
1.0002	0.80
0.9533	1.00
0.9195	1.20
0.8953	1.40
0.8753	1.60
0.8597	1.80
0.8481	2.00
0.8390	2.20
0.8319	2.40
0.8262	2.60
0.8218	2.80
0.8182	3.00
0.8154	3.20
0.8045	∞

we have $\partial Z/\partial \phi \ll \partial Z/\partial X$. Thus for small values of X and for $\phi = \pi/2$, equation (15)[2] in the paper reduces to

$$dZ/dX = 4/3$$

Hence, using the boundary condition Z = 0 at X = 0, the value of Z at $\phi = \pi/2$, for very small values of X would be

$$Z = 4X/3$$

Consequently, for these conditions, the value of h_m can be obtained from

$$(3\mu r \Delta t/\rho^2 g \lambda K^3 \cos\alpha)^{1/4} h_m(X) = Z^{1/4} = (3/4X)^{1/4}$$

Thus the area up to $X_L = 0.05$ is equal to

$$\int_0^{0.05} (3/4X)^{1/4} dX = 0.1315$$

Areas starting at X = 0.05 were evaluated from the trapezoidal rule and Simpson's two thirds rule. Once the area under the curve is determined it is a simple matter to divide this area by the reduced length X to obtain the dimensionless number $h_T(3\mu r \Delta t/\rho^2 g \lambda K^3 \cos\alpha)^{1/4}$. By substituting appropriate values into this expression the integrated average heat-transfer coefficient, h_T, for the entire surface of an inclined tube can be calculated.

Calculation of the Actual Heat-Transfer Coefficient

The actual heat-transfer coefficient will be calculated for the data of Run 28 at an inclination of 15° (see Table II). The equation used in calculating the coefficient is

$$q = h_A A \Delta t_m \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (24)$$

where the heat-transfer rate q is given by

$$q = w \, c_p \Delta t_w \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (25)$$

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[2]Equation (13), p. 17 in the present work.

TABLE II

DATA FOR RUN 28 AT AN INCLINATION OF 15°

Angle Degrees	Inlet Water Temp. F	Exit Water Temp. F	Steam Pressure in. Hg		Steam Temp. Div.	Wall Thermocouple Readings Recorder Divisions A	B	C	D	E
135*	50.0	76.2	18.05	6.20	99.2	58.2	60.6	55.6	46.7	52.3
90	50.0	76.2	18.00	6.25	99.3	63.6	59.8	53.3	40.8	58.3
45	49.9	76.1	18.05	6.20	99.5	67.2	58.8	47.5	49.3	60.3
0	49.8	76.0	18.05	6.20	99.4	68.8	57.8	39.9	54.8	60.4
315	49.8	75.9	18.10	6.15	99.5	67.8	53.1	48.4	56.3	57.8
270	49.8	76.0	18.05	6.20	99.4	63.7	46.7	54.8	57.0	53.7
225	49.8	76.2	18.00	6.25	99.3	57.8	53.6	57.2	55.7	48.2
180	49.8	76.2	18.05	6.20	99.3	51.3	59.5	57.1	52.3	41.8
135**	49.8	76.2	18.05	6.20	99.4	58.3	60.5	55.9	46.7	52.4
Average	49.9	76.1	18.05	6.20	99.4	62.3	56.2	51.7	51.6	54.1

Water Flow Rate 10.0 lb/min.

Barometric Pressure 29.90 in. Hg

Cold Junction Temperature 136.3 F

* Initial test run

** Final test run

The water temperature rise Δt_w equals the temperature difference between the inlet and exit water temperatures t_i and t_e, respectively. Then

$$\Delta t_w = t_i - t_e \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (26)$$

Average exit water temperature reading = 76.1 F

Calibration correction = -0.1

Average exit water temperature = 76.0 F

Average inlet water temperature reading = 49.9 F

Calibration correction = 0.0

Average inlet water temperature = 49.9 F

Water temperature rise = 26.1 F

Water flow rate reading = 10.0 lb/min

Calibration correction = -0.11

Water flow rate = 9.89 lb/min

Specific heat of water at $(t_i + t_e)/2$ = 63 F is 1.0 Btu/lb F

Substituting in equation (25) gives

$$q = 9.89 \times 1.0 \times 26.1 \times 60 = 15450 \text{ Btu/hr}$$

Average steam thermocouple reading = 99.4 recorder divisions

Calibration correction = -1.2

Corrected reading = 98.2 recorder divisions

One recorder division = 0.025 m.v.

Therefore corrected reading = 2.455 m.v.

Cold junction temperature = 136.3F = 2.377 m.v.

Steam temperature t_s = 4.832 m.v. = 233.2 F

Finding the average tube-wall temperature requires the use

of two averaging techniques. First an arithmetic average of the readings of each thermocouple is performed. Then, this average reading is plotted against the thermocouple location. Then, a polar planimeter is used to determine the average tube-wall thermocouple reading from this graph. For this particular example the average is 54.4 recorder divisions.

Average tube-wall thermocouple reading		= 54.4 recorder div.
Calibration correction (see Figure 33)		= -1.2
Corrected average		= 53.2 recorder div.
		= 1.330 m.v.
Cold junction temperature	= 136.3 F	= 2.377 m.v.
Average tube-wall thermocouple temperature		= 3.707 m.v.
		= 189.8 F

As this temperature is measured at a point 0.030 in. beneath the tube surface, a further correction must be made in order to obtain the average tube surface temperature. This correction is obtained from the equation

$$q = KA \frac{\Delta T}{\Delta r}$$

where K is the thermal conductivity of copper = 220 Btu/hr ft F

A is the surface area of the tube = 0.204 ft^2

Δr is the depth of the thermocouple junction beneath the tube surface = 0.030 in.

and ΔT is the correction in degrees F.

In this particular case

$$\Delta T = 15450 \times 0.030/220 \times 0.204 \times 12 = 0.86 \text{ F}$$

Therefore the average tube surface temperature is

$$t_m = 189.8 + 0.9 = 190.7 \text{ F}$$

The mean temperature difference is then

$$\Delta t_m = t_s - t_m = 233.2 - 190.7 = 42.5 \text{ F}$$

Substituting in equation (24) gives

$$h_A = 15450/0.204 \times 42.5 = 1780 \text{ Btu/hr ft}^2 \text{ F}$$

The corrected data used in calculating the actual heat-transfer coefficients for all the test runs is listed in Table III. A list of results is given in Table IV.

Calculation of the Theoretical Heat-Transfer Coefficient

The theoretical heat-transfer coefficient was calculated from the general equation

$$h_T = X(L,r,\alpha)(\rho^2 \lambda g K^3 \cos\alpha / 3\mu r \Delta t_m)^{1/4} \quad . . . (27)$$

where X is obtained from Figure 6.

Calculations will again be made for the data in Table II, at an inclination of 15°. From Figure 6 the value of X is 0.809. Thus equation (27) becomes

$$h_T = 0.809(\rho^2 \lambda g\ K^3 \cos\alpha / 3\mu r \Delta t_m)^{1/4} \quad . . . (28)$$

The physical properties ρ, K and μ of the condensate are evaluated at the condensate film temperature t_f

$$t_f = (t_s + t_m)/2 = (233.2 + 190.7)/2 = 212.0 \text{ F}$$

Then $\rho = 59.8 \text{ lb/ft}^3$

$K = 0.395$ Btu/hr ft F

$\mu = 0.193 \times 10^{-3}$ lb/ft sec

The latent heat of condensation, λ, is evaluated at the steam

TABLE III

AVERAGE CORRECTED DATA

Run*	Inclination Deg.	Inlet Water Temp. F	Exit Water Temp. F	Water Flow Rate lb/min	Steam Press. psia	Steam Temp. F	Average Wall Thermocouple Readings** Recorder Divisions A	B	C	D	E	Mean Surface Temp. F	Cold Junction Temp. F
1	90	52.1	81.6	5.98	15.3	213.6	47.1	40.9	37.5	36.5	32.3	173.6	135.0
2	90	51.6	87.3	5.98	21.1	230.8	56.8	49.8	45.3	43.5	39.3	181.9	135.0
3	90	52.1	57.0	5.98	15.3	213.6	48.9	45.5	42.8	40.5	37.1	178.0	135.0
4	90	51.4	84.8	5.98	21.1	230.5	58.3	54.6	51.7	47.8	43.9	186.5	135.0
5	90	52.3	78.7	5.98	15.4	212.1	48.1	43.8	40.9	39.7	36.2	176.7	135.0
6	90	51.3	85.5	5.98	21.1	231.6	59.0	53.5	50.4	47.7	45.1	186.0	135.0
7	45	54.5	85.7	5.98	15.3	213.8	53.3	51.4	48.7	47.5	47.2	184.5	135.0
8	45	52.9	87.3	5.98	16.8	220.8	57.5	55.3	52.1	50.8	50.3	188.1	135.0
9	45	52.1	89.9	5.98	18.7	227.9	60.4	57.8	54.0	52.2	51.9	190.0	135.0
10	45	54.6	86.7	5.98	15.3	213.6	53.7	51.0	48.0	42.9	42.6	182.6	135.0
11	45	55.0	89.8	5.98	17.0	221.0	57.3	57.0	51.1	46.3	45.6	186.5	135.0
12	45	53.3	90.6	5.98	18.9	228.6	62.0	59.5	55.0	50.3	49.8	190.2	135.0
13	45	51.8	91.1	5.98	20.7	231.8	64.8	62.5	57.7	52.4	52.8	193.0	135.0
14	45	52.3	94.6	5.98	23.4	238.3	62.0	59.1	54.4	48.8	49.3	196.4	142.2
15	45	51.3	81.4	7.98	29.7	232.2	61.7	58.0	53.1	49.8	51.4	190.9	135.9
16	0	52.1	84.9	5.98	14.9	212.6	53.2	50.1	48.2	45.0	45.5	184.4	135.9
17	0	52.2	92.9	5.98	20.7	232.3	65.2	61.6	59.2	54.4	54.5	194.9	135.9

* Runs are arranged in chronological order

** Calibration corrections from Figure 33 were later applied to the averages of these readings.

TABLE III (continued)

Run	Inclination Deg.	Inlet Water Temp. F	Exit Water Temp. F	Water Flow Rate lb/min	Steam Press. psia	Steam Temp. F	Average Wall Thermocouple Readings Recorder Divisions A	B	C	D	E	Mean Surface Temp. F	Cold Junction Temp. F
18	0	52.2	87.8	5.98	17.2	220.0	58.1	54.6	51.9	45.5	47.3	187.4	135.9
19	0	52.4	90.2	5.98	18.8	226.0	62.0	58.3	55.5	49.4	50.4	191.0	135.9
20	0	51.2	84.0	7.98	21.2	233.6	64.2	59.6	56.4	51.3	52.6	192.7	135.9
21	0	51.7	84.4	5.98	15.7	214.7	54.7	51.5	49.4	45.0	46.2	185.5	136.0
22	0	52.0	89.5	4.97	15.2	213.1	54.7	52.2	49.9	45.9	46.9	185.5	136.1
23	0	51.0	78.5	9.89	21.1	233.9	62.7	56.9	52.5	52.7	52.9	191.4	136.1
24	15	52.1	84.3	5.98	15.1	212.6	53.2	51.2	48.1	44.9	46.1	184.8	136.1
25	15	52.4	92.0	5.98	21.3	233.2	67.0	63.1	59.0	54.7	56.4	196.1	136.5
26	15	50.2	84.1	5.98	15.7	217.1	56.2	53.2	50.4	47.1	49.2	187.0	136.2
27	15	50.6	87.8	5.98	18.8	227.1	62.4	58.9	55.6	51.9	53.4	192.2	136.3
28	15	49.9	76.0	9.89	20.5	233.2	62.3	56.2	51.7	51.6	54.1	190.7	136.3
29	15	50.1	81.0	7.98	21.0	233.6	64.0	59.2	55.7	53.1	55.4	193.0	136.2
30	15	50.3	86.0	5.98	17.2	221.1	58.9	55.8	52.6	49.6	51.2	189.3	136.1
31	15	50.0	90.2	5.98	20.3	232.3	66.3	62.0	58.6	54.9	55.4	195.0	136.1
32	30	50.2	81.6	5.98	15.0	212.7	53.5	51.7	49.0	45.7	48.2	185.2	136.0
33	30	50.7	88.7	5.98	20.7	232.3	65.8	63.1	59.1	55.2	57.9	195.7	136.0
34	30	50.5	75.6	9.89	20.3	231.4	61.6	56.7	51.7	51.8	54.2	190.4	136.0
35	30	49.8	80.2	7.98	20.6	232.3	63.2	59.1	55.1	53.2	56.4	192.8	136.0
36	30	50.6	85.6	5.98	17.4	220.9	58.5	56.1	52.9	49.8	52.8	189.5	136.0
37	30	49.9	86.2	5.98	18.1	225.5	60.9	58.4	55.0	51.9	55.0	191.7	136.0
38	45	50.7	83.4	5.98	16.1	217.1	55.5	53.0	49.3	46.9	49.5	186.4	136.0
39	45	51.5	88.7	5.98	20.9	231.6	64.4	61.3	57.0	53.5	56.9	194.0	136.1
40	45	49.7	80.4	5.98	15.0	212.7	52.0	49.3	46.6	43.9	46.7	183.3	136.1

TABLE III (continued)

Run	Inclination Deg.	Inlet Water Temp. F	Exit Water Temp. F	Water Flow Rate lb/min	Steam Press. psia	Steam Temp. F	Average Wall Thermocouple Readings Recorder Divisions A	B	C	D	E	Mean Surface Temp. F	Cold Junction Temp. F
41	60	49.8	88.3	5.98	20.3	231.5	61.6	57.7	52.8	44.9	46.8	188.6	136.1
42	60	49.9	81.6	5.98	14.9	211.7	30.1	46.2	43.2	36.8	38.5	178.9	136.2
43	60	49.9	85.8	5.98	18.1	224.2	57.8	53.9	49.4	42.3	44.5	185.5	136.2
44	60	50.7	87.4	4.97	15.4	214.2	53.7	50.4	47.2	40.3	43.0	182.5	136.0
45	60	50.4	84.1	5.98	16.9	219.8	56.0	53.1	48.5	42.7	47.3	185.0	136.0
46	60	50.1	80.0	7.98	21.0	232.3	61.8	57.2	50.9	46.5	50.6	188.7	136.0
47	60	49.7	84.5	5.98	17.9	223.3	57.8	55.0	49.7	44.2	47.5	186.4	136.0
48	60	49.2	73.9	9.89	20.3	231.4	58.9	53.8	46.4	44.5	48.4	185.8	136.0
49	60	50.0	81.2	5.98	15.1	212.6	51.7	48.3	44.3	39.9	42.9	180.7	136.0
50	75	49.5	81.7	5.98	17.5	220.7	55.5	51.8	46.4	41.3	44.7	183.4	136.0
51	75	50.0	86.2	5.98	21.0	231.8	62.5	58.8	52.4	46.7	50.8	189.6	136.0
52	75	50.1	79.6	5.98	15.0	212.2	50.5	46.2	41.9	38.0	40.9	178.9	136.0
53	75	50.8	80.8	5.98	15.8	215.7	52.5	48.6	43.4	38.9	41.9	180.6	136.0
54	75	51.2	84.7	4.97	15.1	212.5	52.1	48.5	43.9	38.7	41.8	180.4	136.0
55	75	49.5	73.2	9.89	21.0	233.1	59.3	53.2	45.6	43.7	47.5	184.9	136.0
56	75	50.1	83.4	5.98	18.8	224.3	58.0	54.3	48.0	42.9	47.1	185.4	136.0
57	75	50.1	86.0	5.98	20.8	231.2	62.4	58.5	51.9	46.2	51.2	189.2	136.0
58	75	50.0	78.2	7.98	20.7	231.4	60.0	55.3	48.3	44.4	49.3	186.8	136.0
59	90	51.7	82.3	5.98	17.8	222.3	56.8	50.2	47.0	41.9	42.0	183.2	136.0
60	90	51.4	78.1	5.98	14.8	211.3	49.3	43.4	40.5	36.2	35.6	176.6	136.0
61	90	51.3	84.9	5.98	21.2	231.6	63.0	55.6	51.9	45.4	46.4	187.9	136.0
62	90	51.3	77.5	7.98	20.6	230.2	62.5	51.6	46.9	43.2	42.8	184.6	136.0
63	90	51.0	82.4	4.97	14.9	211.5	50.1	45.0	42.1	37.3	37.9	178.2	136.0

TABLE IV

COMPARISON OF ACTUAL AND THEORETICAL
HEAT-TRANSFER COEFFICIENTS

Run	Inclination Degrees	Heat Transfer Rate Btu/hr	Mean Temperature Difference F	Overall Temperature Difference F	Actual Coefficient h_A Btu/hr ft^2 F	Theoretical Coefficient h_T Btu/hr ft^2 F	$\frac{h_A}{h_T}$
1	90	10600	40.0	146.7	1300	1215	1.071
2	90	12700	48.9	161.5	1280	1178	1.087
3	90	9650	35.6	148.0	1330	1260	1.056
4	90	11980	44.0	162.1	1335	1208	1.105
5	90	9720	35.4	146.2	1345	1263	1.066
6	90	12270	45.6	163.2	1315	1194	1.100
7	45	11190	29.3	143.7	1870	1930	0.969
8	45	12340	32.7	150.7	1850	1895	0.976
9	45	13350	37.9	156.9	1750	1850	0.945
10	45	11530	31.0	142.9	1825	1920	0.951
11	45	12490	34.5	148.6	1775	1870	0.949
12	45	13380	38.4	156.6	1705	1830	0.933
13	45	14100	38.8	160.3	1780	1830	0.972
14	45	15170	41.9	164.8	1775	1800	0.985
15	45	14440	41.7	165.8	1700	1795	0.948
16	0	11770	28.2	144.1	2045	2090	0.978
17	0	14600	37.4	159.7	1915	1970	0.972
18	0	12780	32.6	150.0	1920	2035	0.944
19	0	13570	35.0	154.7	1900	1980	0.959

TABLE IV (continued)

Run	Inclination Degrees	Heat Transfer Rate Btu/hr	Mean Temperature Difference F	Overall Temperature Difference F	Actual Coefficient h_A Btu/hr ft^2 F	Theoretical Coefficient h_T Btu/hr ft^2 F	$\frac{h_A}{h_T}$
20	0	15710	49.9	166.0	1880	1925	0.977
21	0	11740	29.2	146.6	1970	2070	0.952
22	0	11200	27.0	142.3	2035	2110	0.965
23	0	16300	42.5	169.1	1880	1915	0.982
24	15	11570	27.8	144.4	2040	2095	0.974
25	15	14200	37.1	161.0	1875	1975	0.949
26	15	12130	30.1	150.1	1975	2055	0.961
27	15	13350	34.9	157.9	1875	2005	0.935
28	15	15450	42.5	170.2	1780	1905	0.935
29	15	14800	40.6	168.0	1785	1930	0.925
30	15	12800	31.8	152.9	1970	2035	0.967
31	15	14400	37.3	162.2	1890	1970	0.960
32	30	11300	27.5	146.8	2010	2055	0.978
33	30	13650	36.6	162.6	1830	1935	0.945
34	30	14900	41.0	168.3	1780	1885	0.944
35	30	14580	39.5	167.3	1810	1900	0.953
36	30	12550	31.4	152.8	1960	2000	0.980
37	30	13050	33.8	157.5	1890	1975	0.964
38	45	11730	30.7	150.0	1870	1929	0.974
39	45	13350	37.6	161.5	1740	1850	0.943
40	45	11000	29.4	147.6	1830	1929	0.954
41	60	13800	42.9	162.4	1575	1660	0.949

TABLE IV (continued)

Run	Inclination Degrees	Heat Transfer Rate Btu/hr	Mean Temperature Difference F	Overall Temperature Difference F	Actual Coefficient h_A Btu/hr ft^2 F	Theoretical Coefficient h_T Btu/hr ft^2 F	$\frac{h_A}{h_T}$
42	60	11380	32.8	145.9	1700	1750	0.971
43	60	12900	38.7	156.3	1630	1695	0.962
44	60	10950	31.7	145.1	1690	1765	0.958
45	60	12100	34.8	152.6	1705	1735	0.983
46	60	14300	43.6	167.2	1605	1650	0.972
47	60	12500	36.9	156.2	1660	1710	0.971
48	60	14650	45.6	169.8	1575	1630	0.967
49	60	11200	31.9	147.0	1720	1770	0.973
50	75	11550	37.3	155.1	1515	1505	1.006
51	75	13000	42.2	163.7	1510	1475	1.024
52	75	10450	33.3	147.5	1540	1540	1.000
53	75	10770	35.1	149.9	1505	1505	1.000
54	75	10000	32.1	144.6	1530	1550	0.987
55	75	14100	48.2	171.7	1435	1425	1.005
56	75	11950	38.9	157.5	1505	1495	1.007
57	75	12900	42.0	163.1	1505	1470	1.024
58	75	13500	44.6	167.3	1480	1450	1.021
59	90	11000	39.1	155.3	1375	1240	1.109
60	90	9570	34.7	146.5	1350	1265	1.068
61	90	12050	43.7	163.5	1350	1215	1.111
62	90	12550	45.6	165.8	1350	1200	1.125
63	90	9410	33.3	144.8	1385	1280	1.082

pressure read by the mercury manometer. This pressure is actually the sum of the partial pressures of the steam and any air present in the steam. Reference to steam tables showed that any error resulting from using this pressure instead of the partial pressure of the steam is infinitesimal.

Steam pressure = 18.05 - 6.20 = 11.85 in. Hg gauge

Barometric pressure = 29.90

Steam pressure = 41.75 in. Hg absolute

Therefore $\lambda = 959$ Btu/lb

Substitution in equation (28) gives

$$h_T = 1910 \text{ Btu/hr ft}^2 \text{ F}$$

Calculation of the Straight Line equation for the Actual Heat-Transfer Coefficient with respect to the Mean Temperature Difference

These equations were calculated using the method of averages. In the following sample calculation the equation for an inclination of 15° will be developed using the appropriate values in Table IV. The desired form of the equation is

$$h_A = m\,\Delta t_m + b \quad \ldots\ldots\ldots\ldots \quad (29)$$

Using the method of averages the data taken at an inclination of 15° are arranged in the following form:

$$\begin{aligned} 2040 &= 27.8m + b \\ 1975 &= 30.1m + b \\ 1875 &= 34.9m + b \\ 1970 &= 31.8m + b \\ \hline 7860 &= 124.6m + 4b \end{aligned} \quad \ldots\ldots\ldots\ldots \quad (30)$$

$$\begin{array}{rcl} 1875 & = & 37.1m + b \\ 1780 & = & 42.5m + b \\ 1785 & = & 40.6m + b \\ \underline{1890} & = & \underline{37.3m + b} \\ 7330 & = & 157.5m + 4b \end{array} \quad \ldots\ldots\ldots\ldots (31)$$

Subtraction yields

$$530 = -32.9\ m$$

$$m = -16.1$$

Substituting m in equation (30) gives

$$7860 = -2010 + 4b$$

$$b = 2470$$

Substitution in equation (29) gives the straight line equation

$$h_A = -16.1\ \Delta t_m + 2470$$

<u>The Method Used to Calculate the General Equations Correlating the Results</u>

The straight line equations for the variation of the actual coefficient with respect to the mean temperature difference (see Table V, Appendix II) are of the form

$$h_A = m\ \Delta t_m + b \quad \ldots\ldots\ldots\ldots (32)$$

If the intercepts and slopes of these equations are plotted as in Figure 31, then two straight lines of the form

$$b = am + c \quad \ldots\ldots\ldots\ldots (33)$$

result for the intervals $0° \leq \alpha \leq 30°$ and $45° \leq \alpha \leq 90°$. The values of a and c can be determined by the method of averages, and thus b is known as a function of m.

In the next step the slopes of equation (32) are plotted against the angles of inclination as also shown in Figure 31.

Then two equations of the form

$$m = d\alpha + g \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (34)$$

can be obtained for the same intervals as before. Thus m is known as a function of .

Equations (33) and (34) are used to eliminate m and b in equation (32), yielding

$$h_A = (d\alpha + g)(a + \Delta t_m) + c \qquad (35)$$

where a, c, d and g are all known constants. Substituting the appropriate value for α in equation (35) yields a straight line equation for the variation of h_A with Δt_m at that particular pipe inclination.

<u>Calculation of the Reduction in Heat Transfer due to the Thermal Resistance of the Solder and Thermocouple Leads in the Tube Wall</u>

The thermal resistance of the solder and thermocouples in the tube wall is higher than that of copper. As a result the heat-transfer rate through the thermocouple grooves will be lower than that through the rest of the tube. In the following analysis it will be assumed that no heat is transferred through the grooves. This is the worst possible case. In actual practice there will be some heat transferred.

The following is the equation for one dimensional heat conduction through a cylindrical shell

$$q = KA_m \frac{\Delta \beta}{\Delta u}$$

where q = heat-transfer rate

K = thermal conductivity

A_m = average heat-transfer area

Δu = tube thickness

$\Delta \beta$ = temperature drop across tube thickness

For a tube with no thermocouples

$$A_m = 26.9 \text{ in.}^2$$

For the tube with thermocouples and assuming no heat transfer through the grooves

$$A_m = 26.9 - \text{surface area of grooves}$$
$$= 26.9 - 0.75 = 26.15 \text{ in.}^2$$

Assuming that K, $\Delta \beta$ and Δu are the same for both tubes then the per cent reduction in the heat-transfer rate is

$$0.75 \times 100/26.9 = 2.8 \text{ per cent}$$

<u>Calculation of the Heat-Transfer Coefficients for Different Average Tube-Surface Temperatures</u>

Figure 28 is a plot of the average circumferential temperature versus the distance along the tube in a horizontal position. The average surface temperature over any portion of the tube is found by integrating the curve over the appropriate length. Calculations were made for the following cases:

1. The average tube-surface temperature over the 9 in. length

$$t_{m1} = 187.5 \text{ F}$$

2. The average tube-surface temperature for no end effects

$$t_{m2} = 188.8 \text{ F}$$

3. The average tube-surface temperature for the entire tube

$$t_{m3} = 186.8 \text{ F}$$

Since the observed steam temperature was 214.2 F, the mean temperature differences for the above conditions are:

$$\Delta t_{m1} = 26.7 \text{ F}$$

$$\Delta t_{m2} = 25.4 \text{ F}$$

$$\Delta t_{m3} = 27.4 \text{ F}$$

Consider the temperature distributions in Figure 28 for cases 2 and 3. In case 3 the temperature difference between the steam and the tube surface is much larger near the end plates. Therefore more heat will be transferred in these regions and as a result the average heat-transfer rate will be larger. Heat-transfer rates obtained from Figure 30 using the above mean temperature differences are listed below:

$$q_1 = 11030 \text{ Btu/hr}$$

$$q_2 = 10640 \text{ Btu/hr}$$

$$q_3 = 11030 \text{ Btu/hr}$$

Note that $q_1 = q_3$. This is as expected since heat-transfer conditions are identical; they differ only in that Δt_m is calculated over a different length. However, $q_1 = q_2$ because in this case the heat-transfer conditions have changed.

Calculating the heat-transfer coefficients for the above conditions from the equation

$$q = h\ A\ \ \Delta t_m$$

gives

$$h_1 = 2020 \text{ Btu/hr ft}^2 \text{ F}$$

$$h_2 = 2050 \text{ Btu/hr ft}^2 \text{ F}$$

$$h_3 = 1970 \text{ Btu/hr ft}^2 \text{ F}$$

The per cent differences of h_1 and h_3 with respect to h_2 are -1.5 and -4.0, respectively.

Similarly, for a vertical tube (see Figure 29), the per cent differences for h_1 and h_3 with respect to h_2 and 0 and -2.2, respectively.

Calculation of the Effect of the Presence of Air in the Steam

The presence of air in the steam is detected by a discrepancy between the observed temperature and the saturation temperature corresponding to the observed pressure. The presence of air reduces the partial pressure of the water vapour and its corresponding saturation temperature. Therefore, the observed temperature is too low for saturation at the observed pressure, which is really the sum of the partial pressures of the steam and the air.

For a particular case the following data were obtained:

Observed steam temperature	= 211.7 F
Saturation pressure at this temperature	= 14.6 psia
Observed steam pressure	= 14.8 psia
Saturation temperature at this pressure	= 212.4 F

Then the partial pressure of the air is 14.8 - 14.6 = 0.2 psia or 0.2 x 100/14.8 = 1.35 per cent of the total pressure. From Othmer's graph in Figure 32, for a steam temperature of 212 F and the observed mean temperature difference of 34.7 F, the heat-transfer coefficients for 0 and 1.42 per cent air are approxi-

mately 1300 and 650 Btu/hr ft^2 F, respectively. Therefore, for a 1.35 per cent concentration of air the heat-transfer coefficient may be reduced by about 45 per cent.

Calculation of the Effect of Neglecting the Possibility of Moisture in the Steam

Tests have shown that the steam quality could be as low as 97 per cent. The effect of this will be found for the following data for condensation on a vertical tube. The equation for the heat-transfer coefficient is

$$h_T = 0.943(\rho^2 g \lambda K^3/\mu L \Delta t)^{1/4}$$

For saturated steam at 211.5 F, λ = 970.6 Btu/lb while for 97 per cent quality steam, λ = .97 x 970.6 = 947.0 Btu/lb. Thus by assuming saturated steam the theoretical heat-transfer coefficient will be

$$(970.6^{1/4} - 947.0^{1/4})100/947.0^{1/4} = 0.8 \text{ per cent high.}$$

Calculation of the Effect of Subcooling the Condensate

A calculation of the effect of subcooling of the condensate on the heat-transfer coefficient will be performed on a typical set of data for condensation on a vertical tube. The theoretical equation for the average coefficient for condensation on a vertical tube is

$$h_T = 0.943(\rho^2 g \lambda K^3/\mu L \Delta t)^{1/4}$$

This equation assumes that no subcooling of the condensate occurs. If subcooling occurs more heat will be given up by the steam than

that indicated by the latent heat of condensation, λ. For subcooling λ should be replaced with $\lambda + C$, where C is the heat given up by the subcooled condensate. The per cent error in the theoretical coefficient due to neglecting subcooling is then

$$[\lambda^{1/4} - (\lambda + C)^{1/4}]100 / (\lambda + C)^{1/4}$$

For saturated steam at 211.5 F, $= 970.6$ Btu/lb

At a mean surface temperature of 178.20 F

$$C = c_p \Delta t = 1.0(211.5 - 178.2)/2 = 16.6 \text{ Btu/lb}$$

Then $\lambda + C = 987.2$ Btu/lb

and the per cent error is

$$(970.6^{1/4} - 987.2^{1/4})100/987.2^{1/4} = -0.4 \text{ per cent}$$

Calculation of the Effect of Subcooling and Superheating

A calculation of the effect of subcooling of the condensate and superheating of the supply steam on the heat-transfer coefficient will be performed on a typical set of data for condensation on a horizontal tube. The theoretical equation for the average heat-transfer coefficient for condensation on a horizontal tube is

$$h_T = 0.725(\rho^2 g \lambda K^3 / \mu D \Delta t)^{1/4}$$

This equation assumes that the steam is saturated and that the condensate is not subcooled. If subcooling occurs and if the steam is superheated, more heat will be given up by the steam when it condenses than that indicated by the latent heat of condensation. Therefore, the latent heat of condensation, λ, in

the above equation should be replaced with $\lambda + H + C$, where H is the heat given up by the superheat in the steam and C is the heat given up by the subcooled condensate.

For the particular case in question:

Observed steam temperature	= 233.9 F
Observed steam pressure	= 21.1 psia
Latent heat of condensation corresponding to this pressure	= 958 Btu/lb
Saturation temperature at observed pressure	= 230.8 F
Observed mean tube-surface temperature	= 191.9 F

Then $H = c_p \Delta t = 0.5(233.9 - 230.8) = 1.6$ Btu/lb

$C = c_p \Delta t = 1.0(230.8 - 191.9)/2 = 19.5$ Btu/lb

Thus $\lambda + H + C$ = = 979 Btu/lb

and the per cent error is

$$(958^{1/4} - 979^{1/4})/979^{1/4} = -0.55$$

APPENDIX II

STRAIGHT LINE RELATIONS

The following tables present the straight line relations developed from the results. In the first row of each table the general form of the relation is given. The source of each relation is listed in the second row. The words "Original Data" mean that the relation was developed from the experimental points using the method of averages. The words "General Correlation" indicate that the relation was developed by substituting appropriate values of the inclination into the general equations correlating the equations under "Original Data."

The following abbreviations are used in the tables:

I.R. = initial test runs.

F.R. = final test runs after five weeks of testing.

Avg. = average of initial and final test runs.

TABLE V

RELATIONS FOR THE HEAT-TRANSFER COEFFICIENT AND THE MEAN TEMPERATURE DIFFERENCE BETWEEN THE STEAM AND THE TUBE SURFACE

Inclination	$h_A = m\,\Delta t_m + b$				$h_A/h_T = m\,\Delta t_m + b$			
	Original Data		General Correlation		Original Data		General Correlation	
	m	b	m	b	$m \times 10^3$	b	$m \times 10^3$	b
0	-10.2	2290	-10.2	2290	1.32	0.921	1.32	0.921
15	-16.1	2470	-14.8	2420	-2.07	1.024	1.34	1.007
30	-18.0	2510	-19.3	2560	-3.28	1.075	-3.99	1.092
45	-11.8	2210	-13.5	2280	-0.965	0.992	-1.85	0.999
60	-10.9	2060	-9.18	1983	-0.804	0.997	0.07	0.971
75	-4.14	1666	-4.91	1682	2.07	0.927	1.98	0.944
90 I.R.	-1.64	1386			3.82	0.921		
90 F.R.	-1.14	1407			3.86	0.944		
90 Avg.	-1.34	1396	-0.60	1379	3.84	0.932	3.91	0.916

TABLE VI

RELATIONS FOR THE HEAT-TRANSFER COEFFICIENT AND THE OVERALL TEMPERATURE DIFFERENCE

Inclination	$h_A = m\,\Delta T + b$				$h_A/h_T = m\,\Delta T + b$			
	Original Data		General Correlation		Original Data		General Correlation	
	m	b	m	b	$m \times 10^3$	b	$m \times 10^3$	b
0	-5.94	2860	-5.94	2860	0.767	0.848	0.767	0.847
15	-9.43	3390	-8.69	3280	-1.210	1.143	-0.795	1.088
30	-10.7	3580	-11.5	3700	-1.950	1.271	-2.360	1.294
45	-6.99	2870	-7.99	3030	-0.571	1.046	-1.100	1.111
60	-6.46	2660	-5.46	2490	-0.476	1.041	0.200	0.980
75	-2.34	1872	-2.93	1953	1.168	0.824	1.150	0.847
90 I.R.	-0.981	1469			2.29	0.726		
90 F.R.	-0.948	1510			2.28	0.745		
90 Avg.	-0.964	1490	-0.39	1411	2.28	0.735	2.28	0.715

TABLE VII

RELATIONS FOR THE HEAT-TRANSFER RATE AND THE OVERALL TEMPERATURE DIFFERENCE

Inclination	$q = m\Delta T + b$			
	Original Data for Test Pipe VII		Original Data for Test Pipe X	
	m	b	m	b
0	190.7	-15950	173.0	-12470
15	160.2	-11760	156.0	-10000
30	151.5	-10790	163.5	-11230
45	166.5	-12800	166.6	-11700
60	151.2	-10920	156.4	-10580
75	155.6	-12560	148.5	-10200
90	145.8	-11480	148.5	-10970

NOTE: Test pipe VII was used to obtain values of the heat-transfer coefficient. Test pipe X is similar to test pipe VII except that it has no tube-wall thermocouples.

APPENDIX III

NOMENCLATURE

A = area, ft^2

a,b,c = unit vectors in x, y, z-directions

c_p = specific heat, Btu/lb

D = diameter of tube, ft

g = acceleration of gravity, ft/hr^2

h = local heat-transfer coefficient, Btu/hr ft^2 F

h_A = actual heat-transfer coefficient for entire surface of a tube, Btu/hr ft^2 F

h_m = mean heat-transfer coefficient for a ring of infinitesimal width on tube surface, Btu/hr ft^2 F

h_T = theoretical heat-transfer coefficient for entire surface of a tube, Btu/hr ft^2 F

K = thermal conductivity, Btu/hr ft F

L = length of tube, ft

n_c = unit normal vector to tube surface

n_h = unit normal vector to horizontal plane

n_i = unit normal vector to plane of n_c and n_h

q = heat-transfer rate, Btu/hr

r = radius of tube, ft

t_e = exit water temperature, F

t_f = condensate film temperature, $(t_s + t_m)/2$, F

t_i = inlet water temperature, F

t_m = mean tube-surface temperature, F

t_s = steam temperature, F

Δt = temperature difference between the steam and an isothermal tube, F

Δt_m = temperature difference between the steam and the mean surface temperature of a non-isothermal tube, $t_s - t_m$, F

ΔT = temperature difference between the steam and the water, $t_s - (t_i + t_e)/2$, F

Δt_w = temperature rise of the water, $t_e - t_i$, F

v_m = mean velocity at any cross section of condensate film ft/hr

w = water flow rate, lb/hr

X = reduced distance from top end of tube, $x/r \tan\alpha$

X_L = reduced length of tube, $L/r \tan\alpha$

x,y,z = coordinate axes, shown in Figure 2

Y = thickness of condensate film, ft

Z = $g\rho^2\lambda\cos\alpha\, Y^4/3\mu K \quad r\Delta t$

α = angle of inclination of tube with horizontal

θ = angle between a plane tangent to a tube and the horizontal plane

λ = latent heat of condensation, Btu/lb

μ = Kinematic viscosity, lb/hr ft

ρ = density, lb/ft^3

ϕ = angular position from top element of tube

ψ = angle between a momentary droplet path and a cross section of the tube

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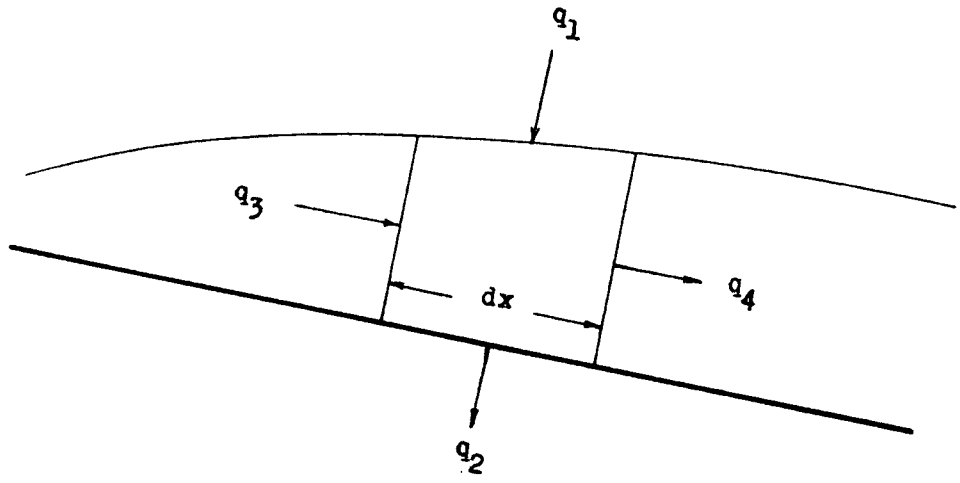


FIGURE 1

ELEMENT IN CONDENSATE FILM

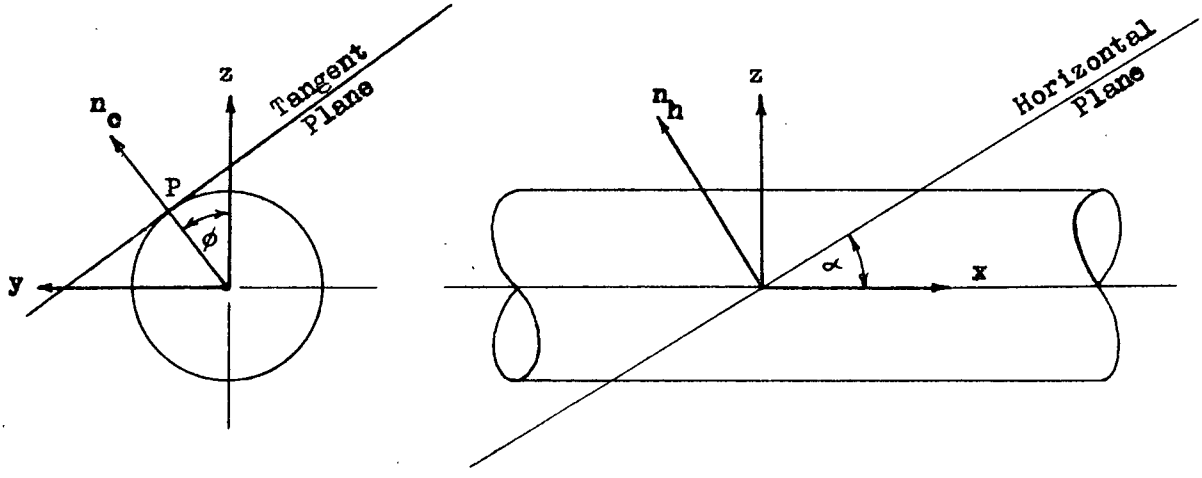


FIGURE 2

COORDINATE SYSTEM WITH DENOTATIONS

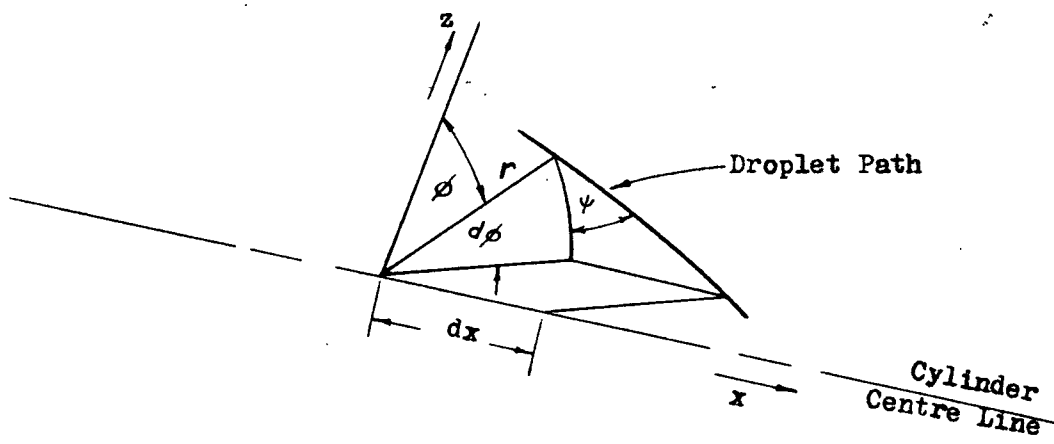


FIGURE 3

MOMENTARY DROPLET PATH

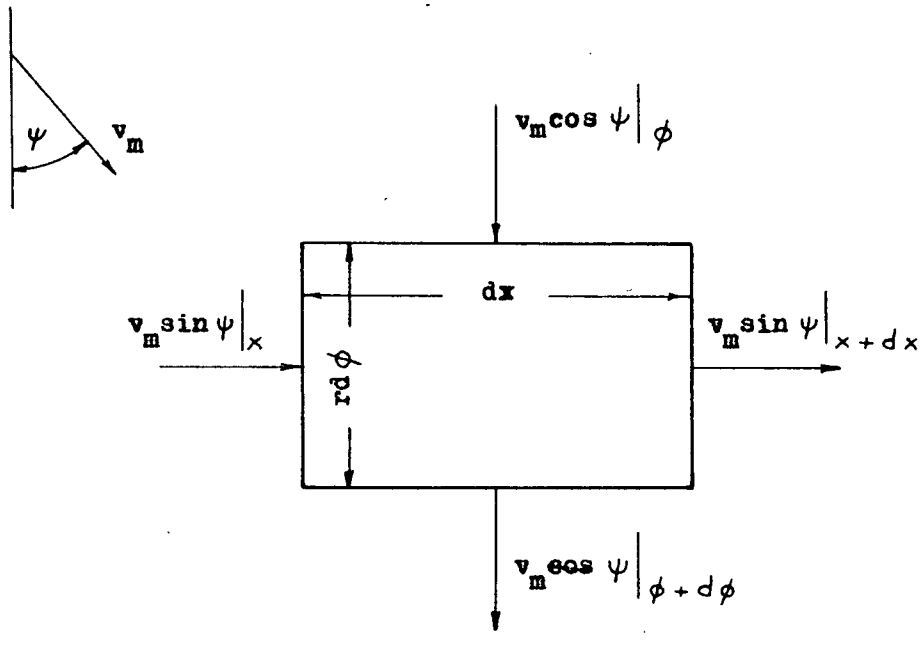


FIGURE 4

CYLINDER SURFACE ELEMENT

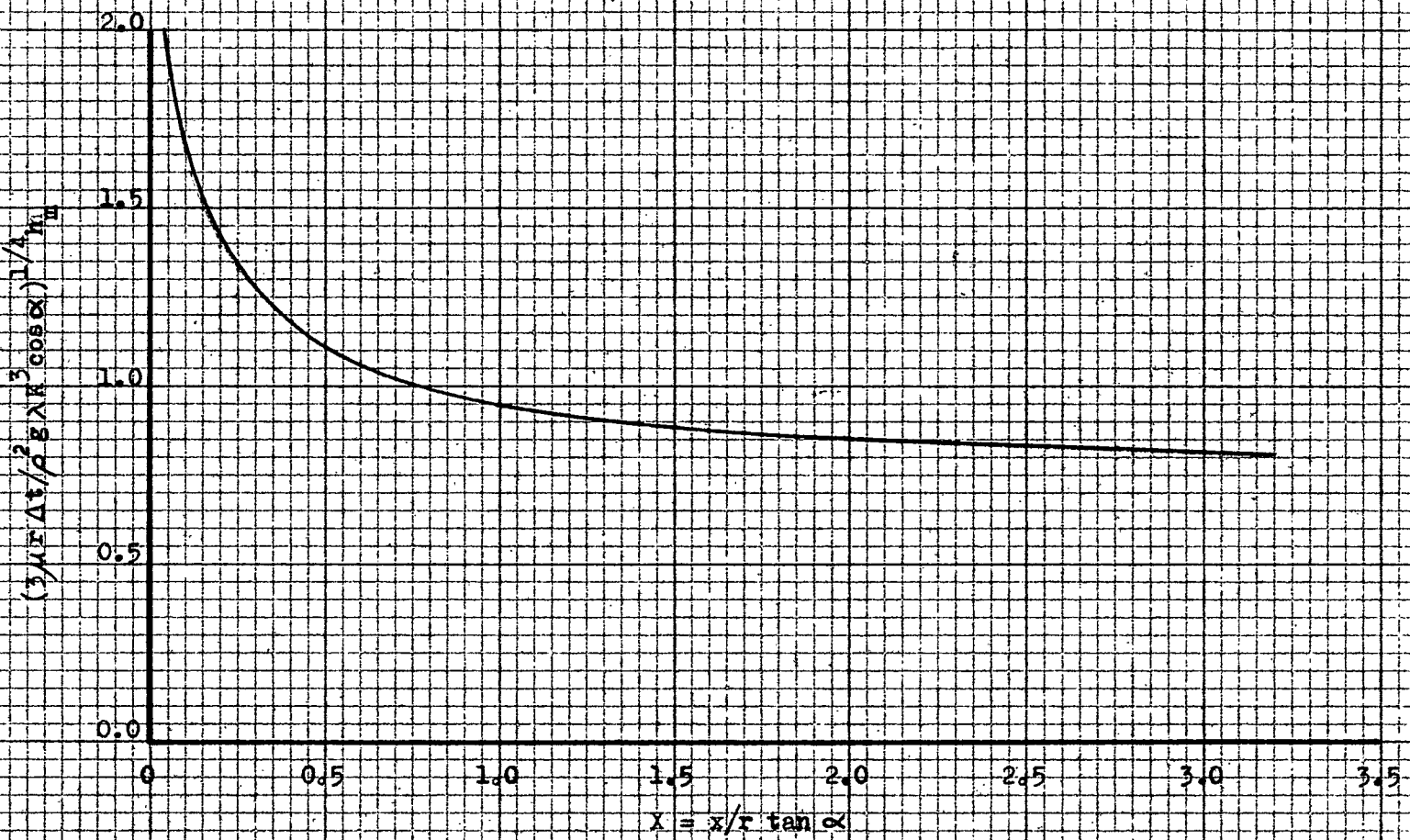


FIGURE 5

MEAN HEAT-TRANSFER COEFFICIENT AT DIFFERENT
REDUCED DISTANCES FROM UPPERMOST END
OF CYLINDER

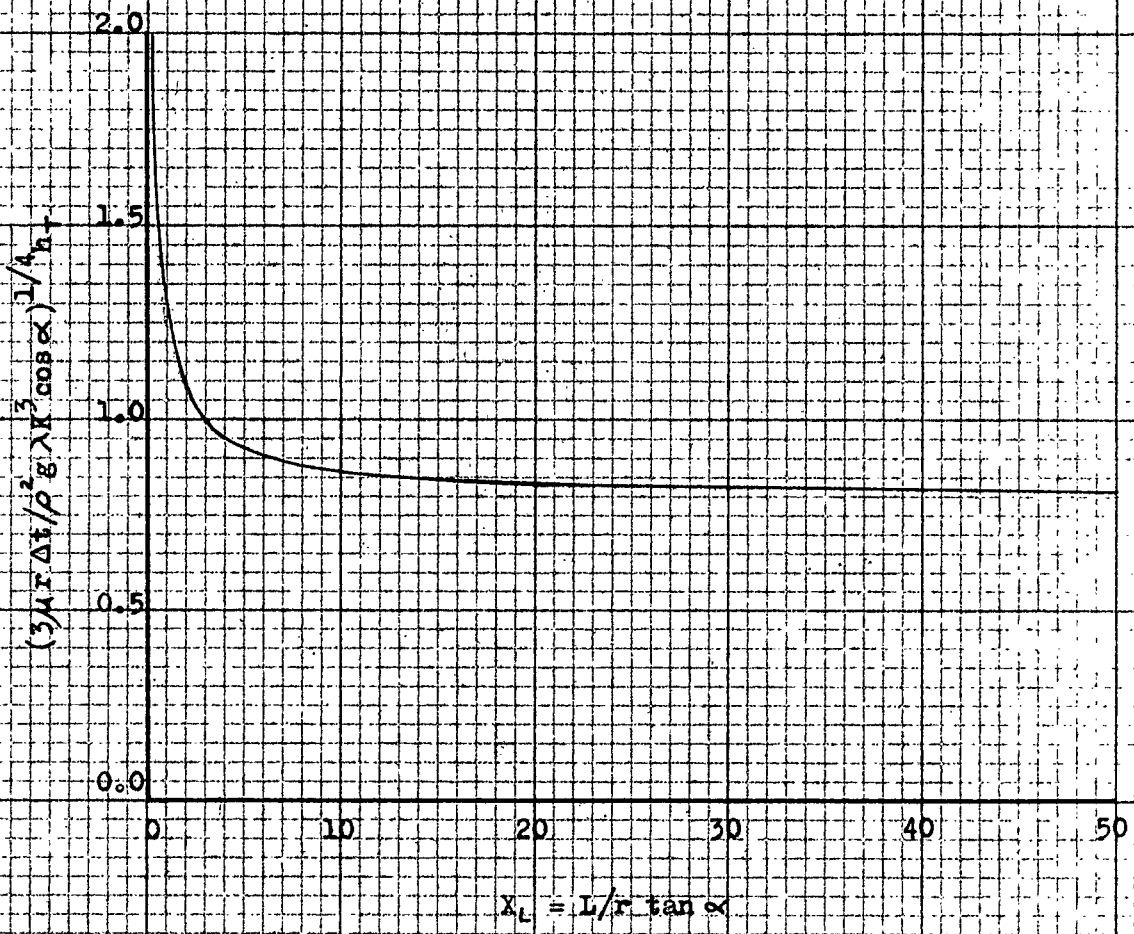


FIGURE 6

MEAN HEAT-TRANSFER COEFFICIENT
ON THE ENTIRE SURFACE OF AN
INCLINED CYLINDER

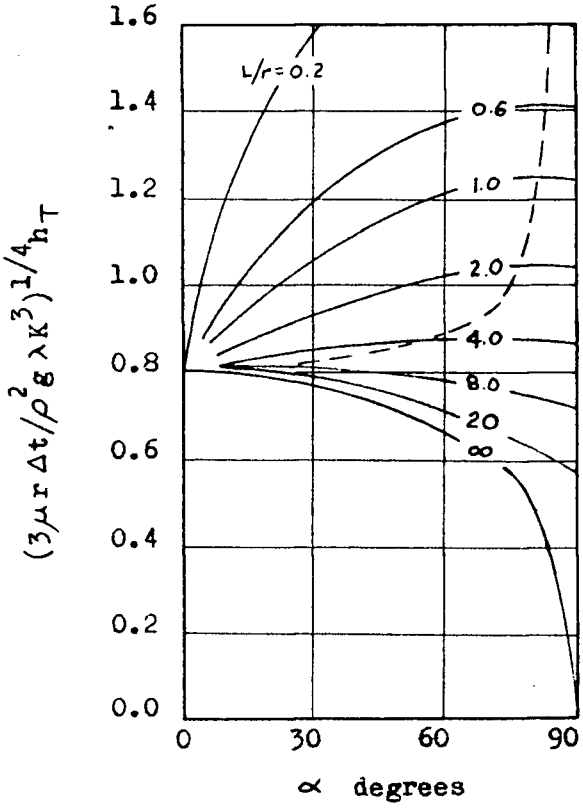


FIGURE 7

VARIATION OF MEAN HEAT-TRANSFER COEFFICIENT ON ENTIRE SURFACE OF GEOMETRICALLY SIMILAR CYLINDERS WITH THEIR INCLINATION [SELECTED DATA FROM REFERENCE (9)]

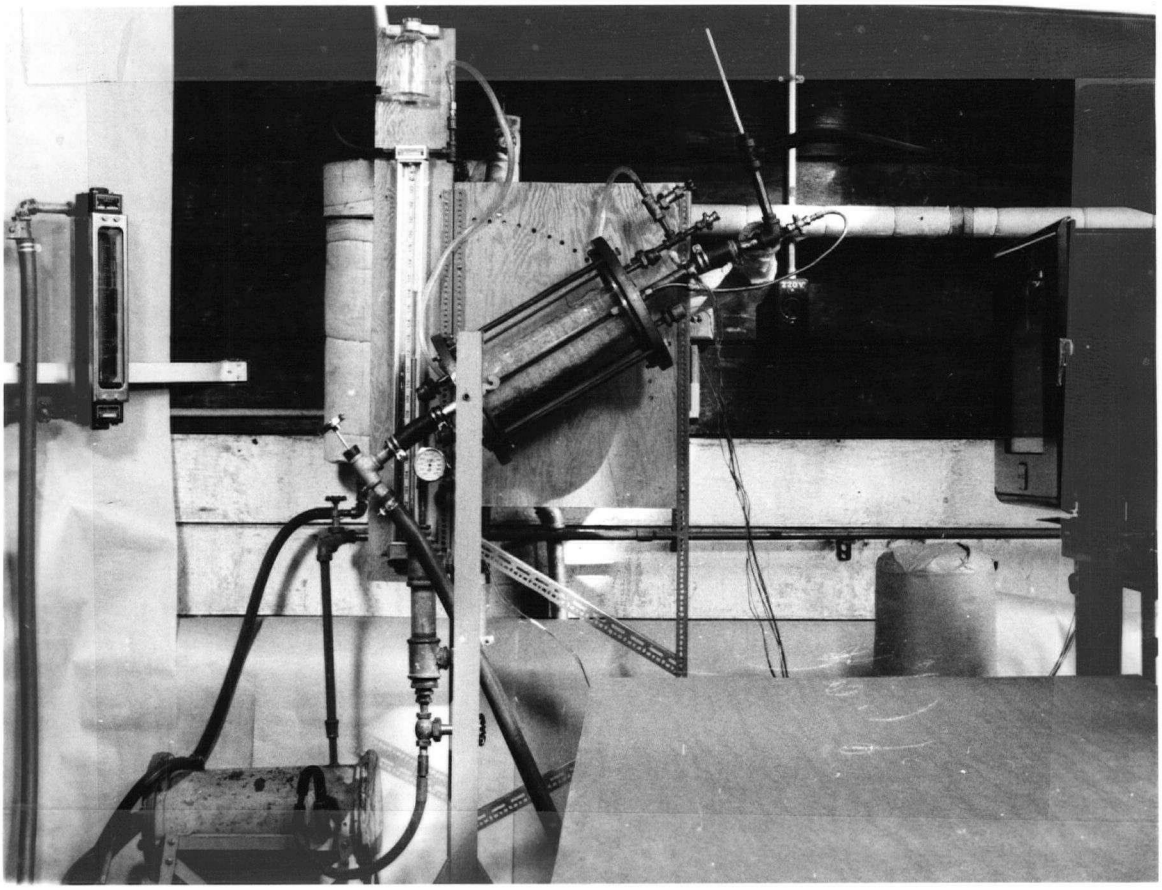


FIGURE 8

THE APPARATUS AND INSTRUMENTS

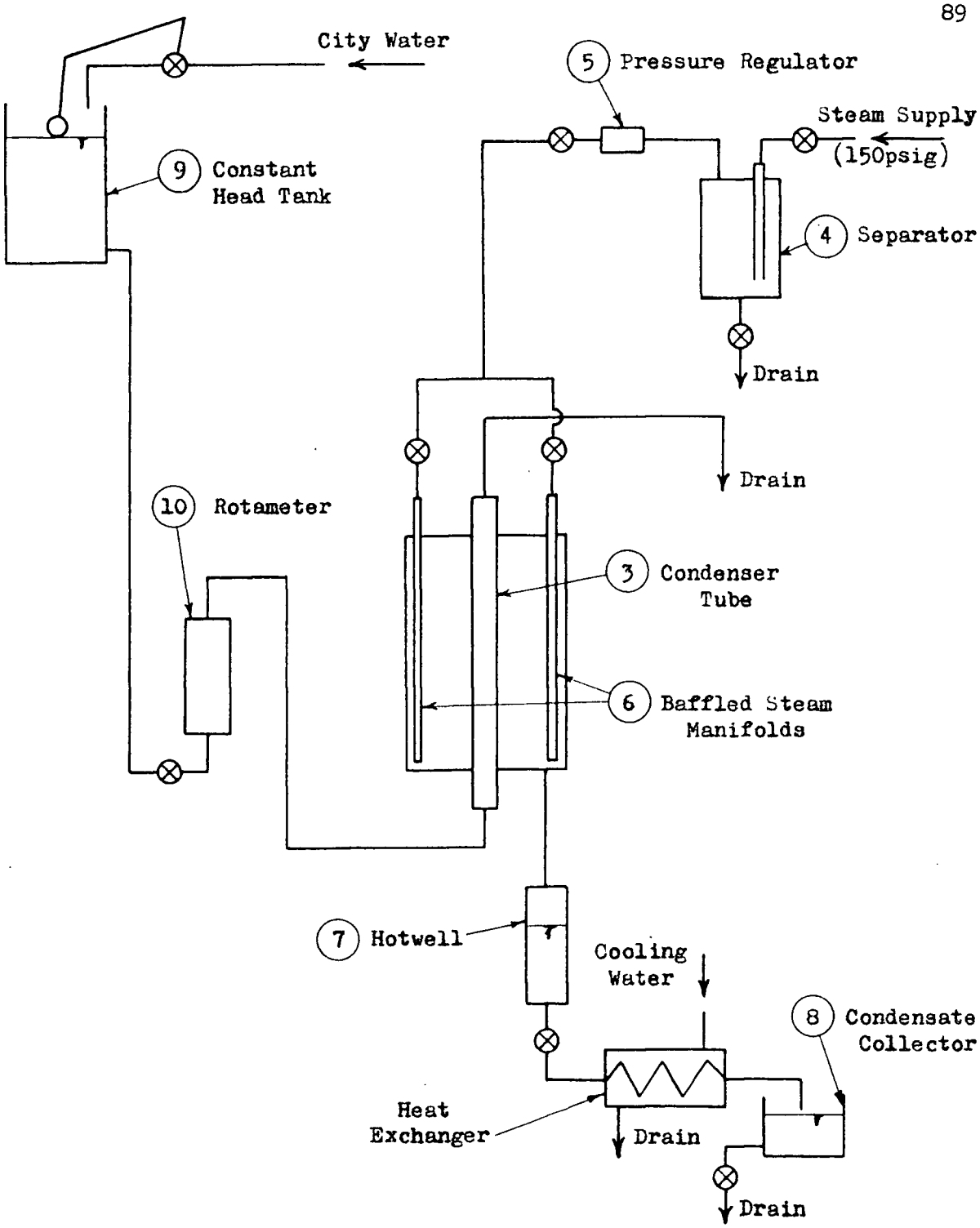


FIGURE 9

THE GENERAL ARRANGEMENT OF THE APPARATUS

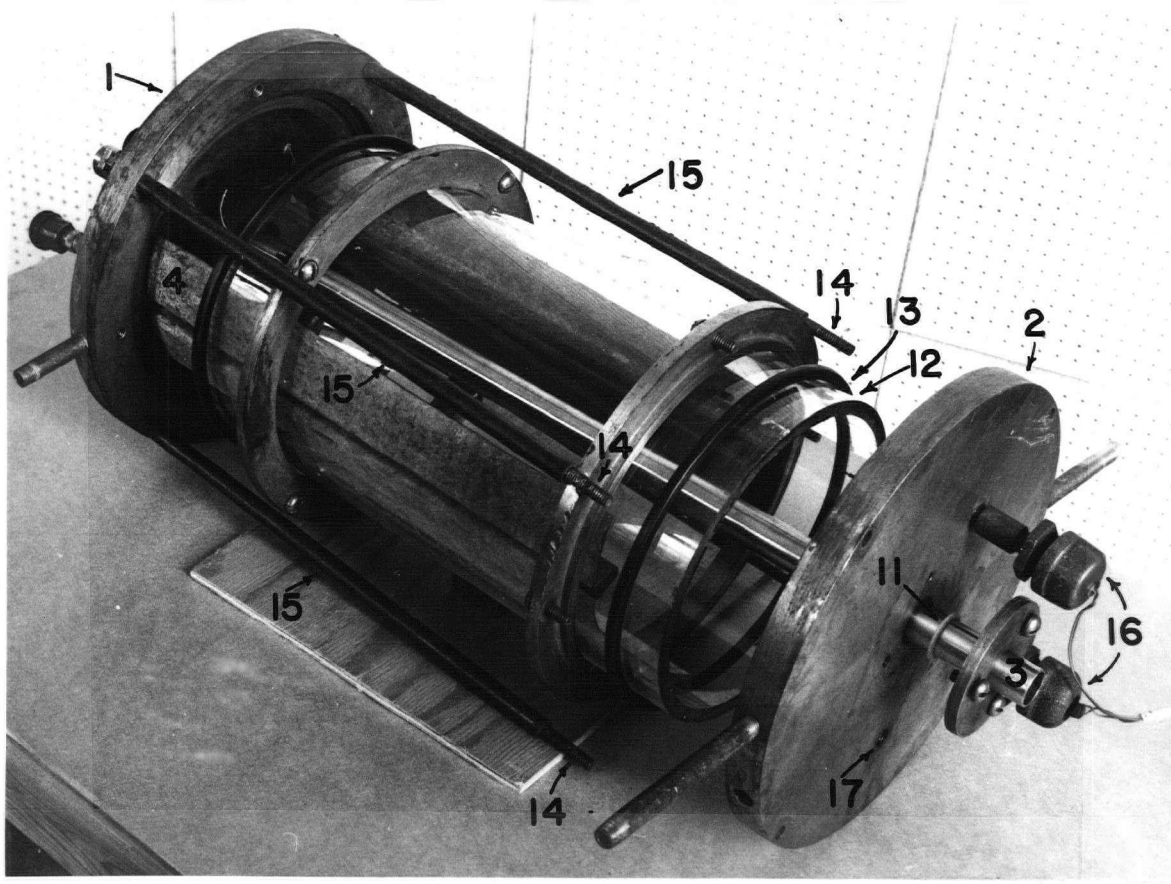


FIGURE 10

THE STEAM CHAMBER

KEY

1. Top End Plate
2. Bottom End Plate
3. Condenser Tube
4. Baffled Steam Manifold
11. Teflon O-Ring
12. Pyrex Pipe
13. Rubber O-Ring
14. Tie Rod
15. Spacer
16. Thermocouple Attachments
17. Condensate Outlet

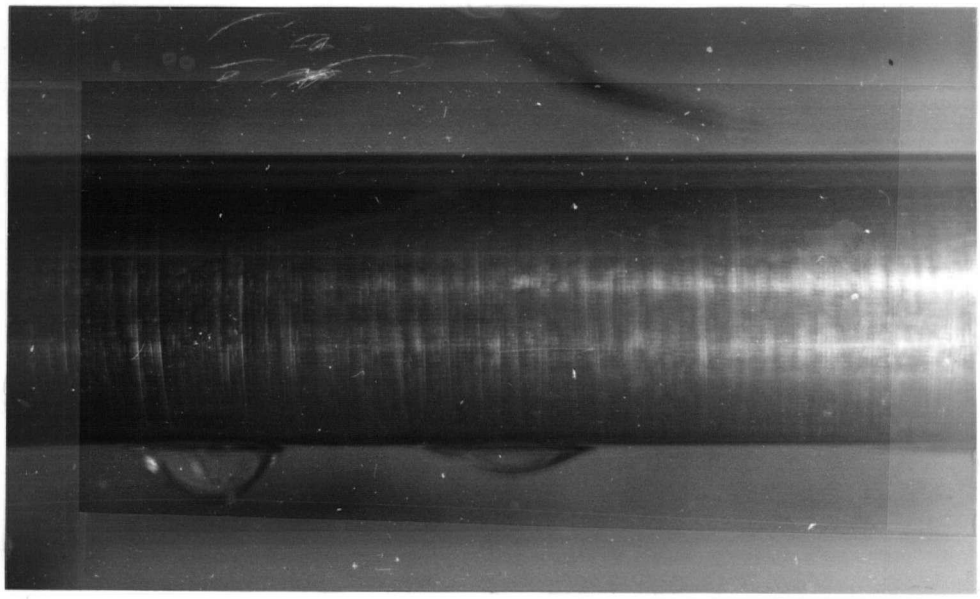


FIGURE 11

FILM CONDENSATION ON A
HORIZONTAL TUBE

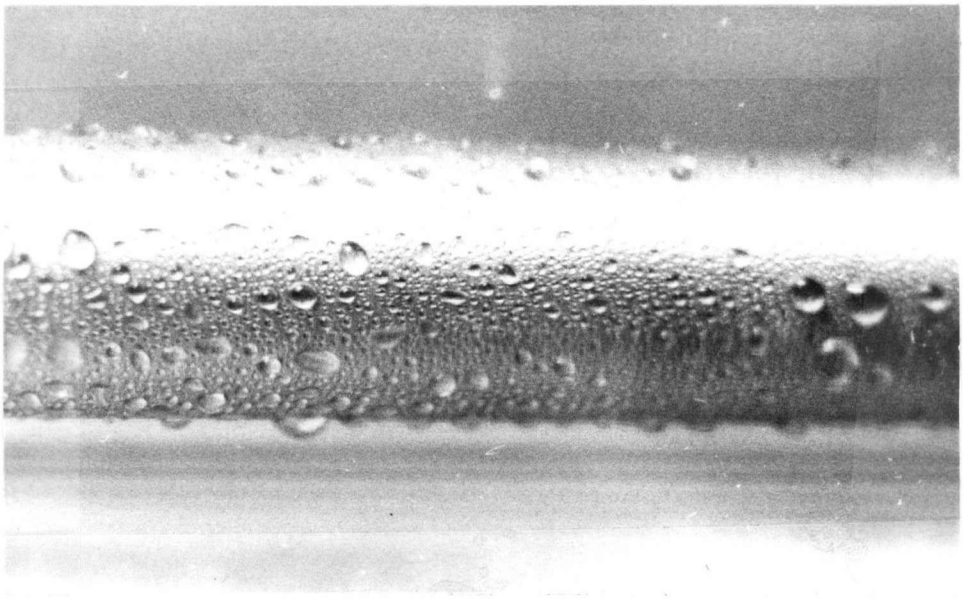


FIGURE 12

DROP-WISE CONDENSATION ON
A HORIZONTAL TUBE

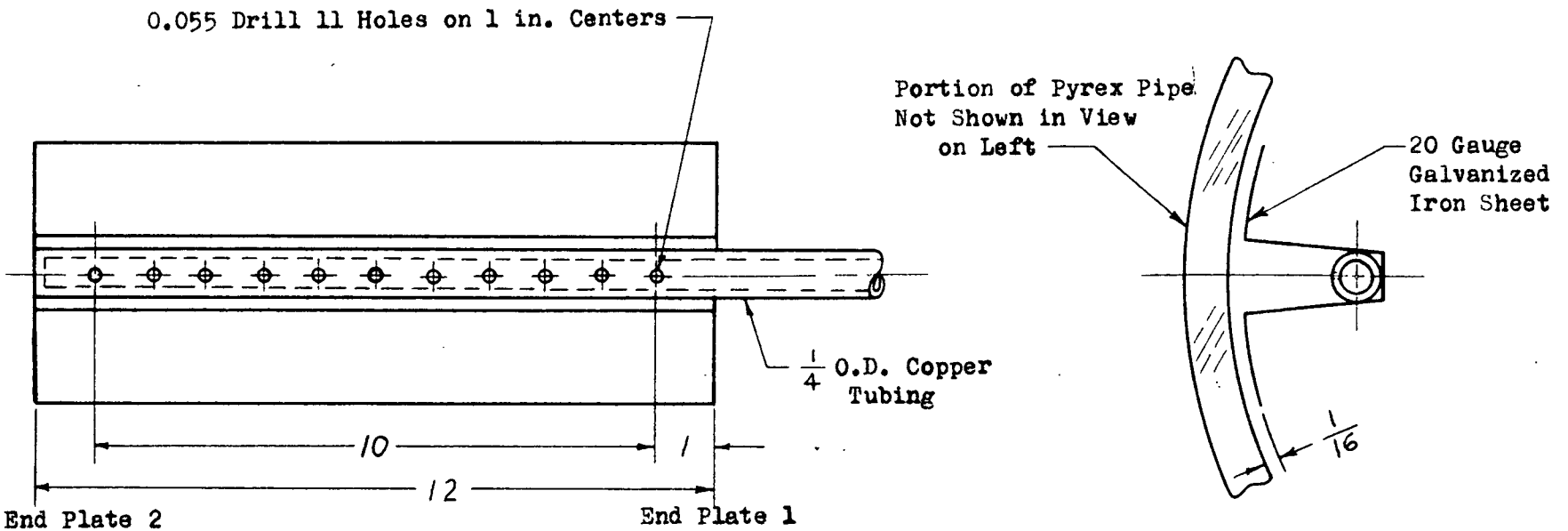


FIGURE 13

STEAM MANIFOLD AND BAFFLE

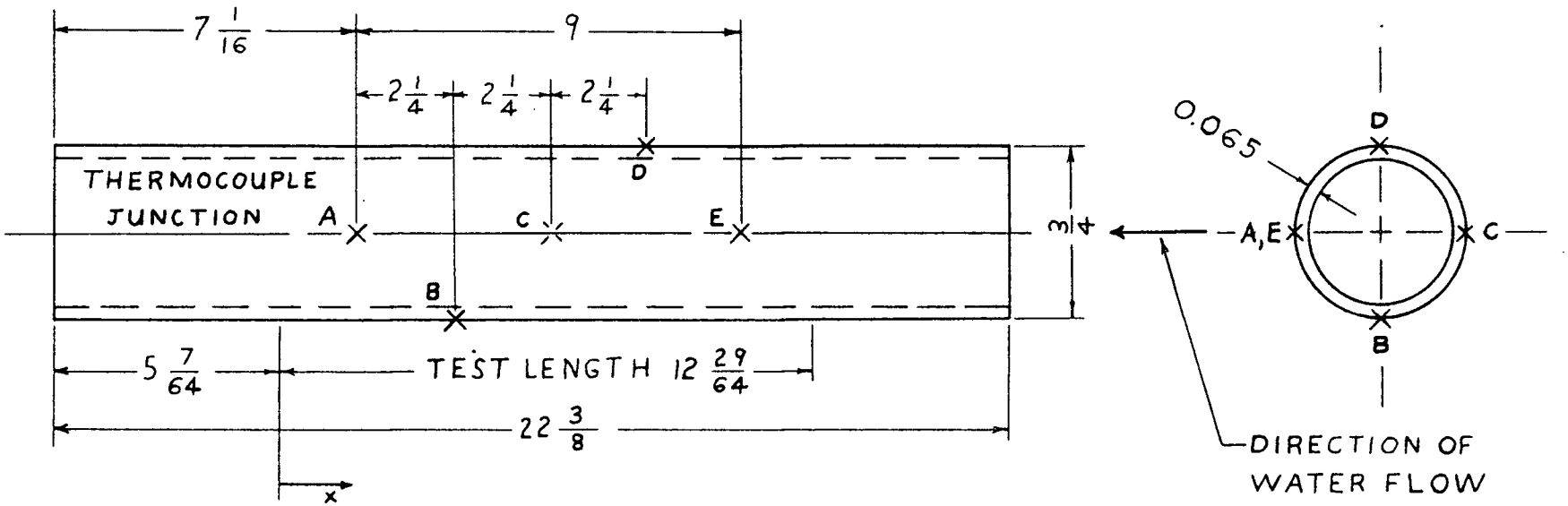
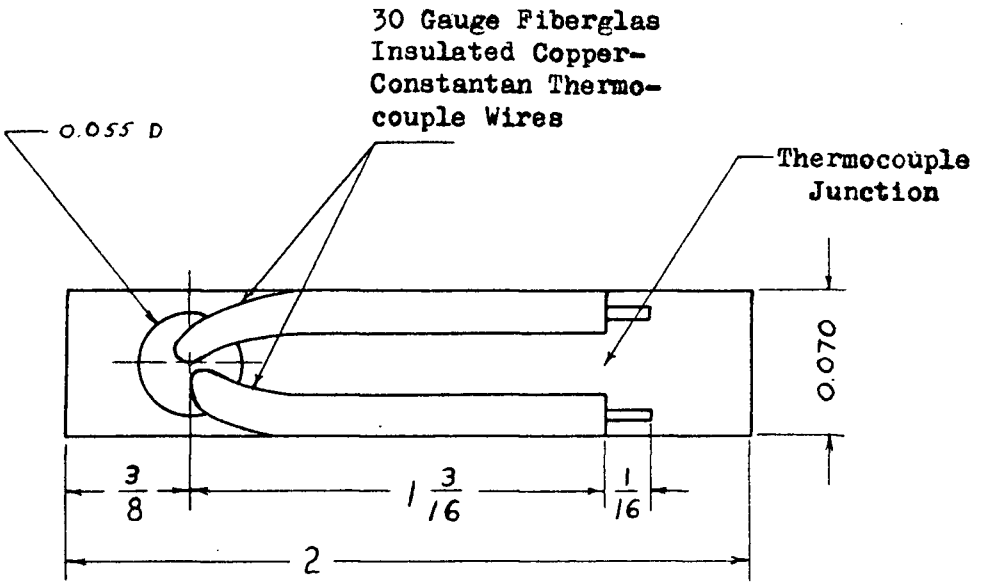


FIGURE 14

TUBE-WALL THERMOCOUPLE JUNCTION LOCATIONS



PLAN OF GROOVE

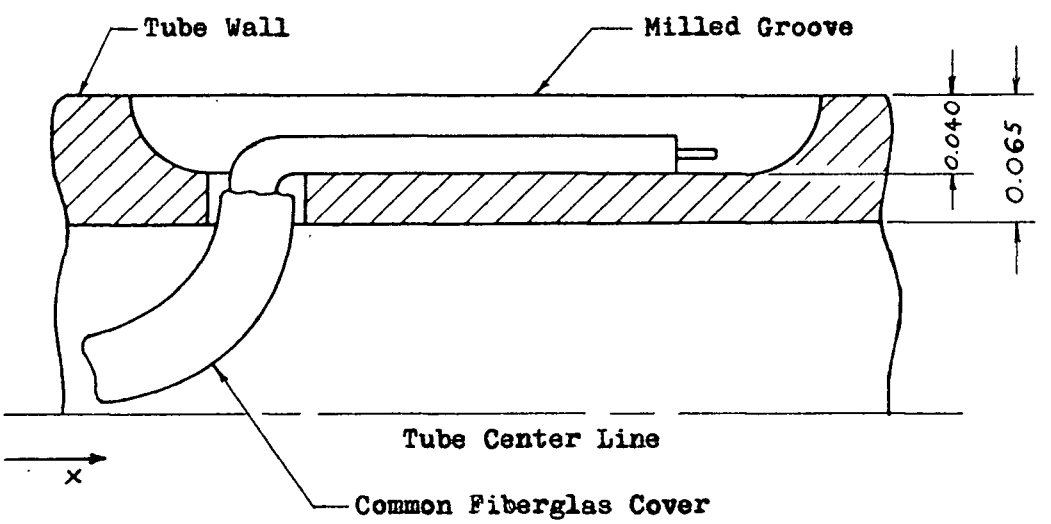


FIGURE 15

TUBE-WALL THERMOCOUPLE
INSTALLATION

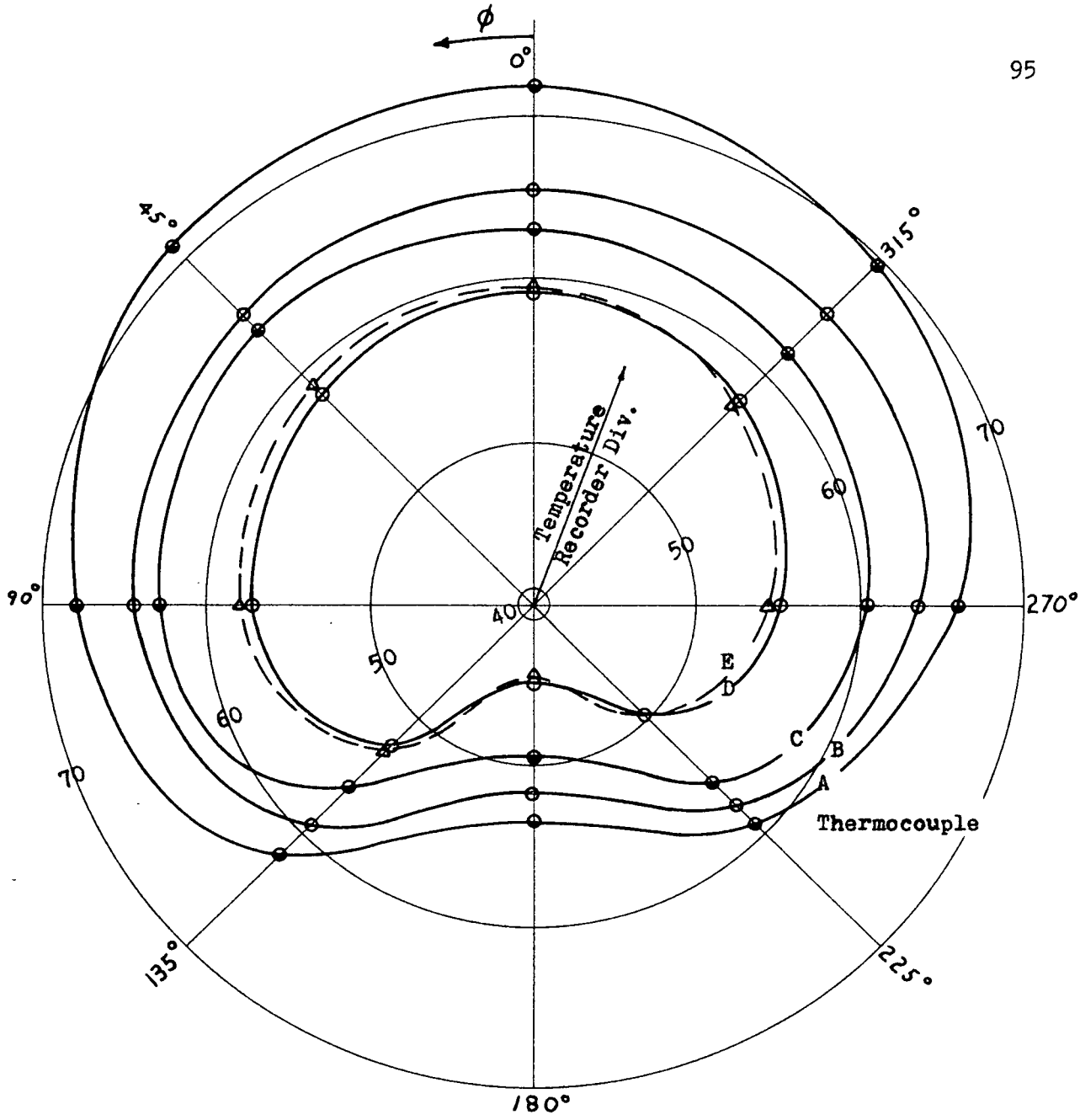


FIGURE 16

CIRCUMFERENTIAL TEMPERATURE DISTRIBUTIONS
ALONG A HORIZONTAL TUBE
(RUN 17, Δt_m = 37.4F)

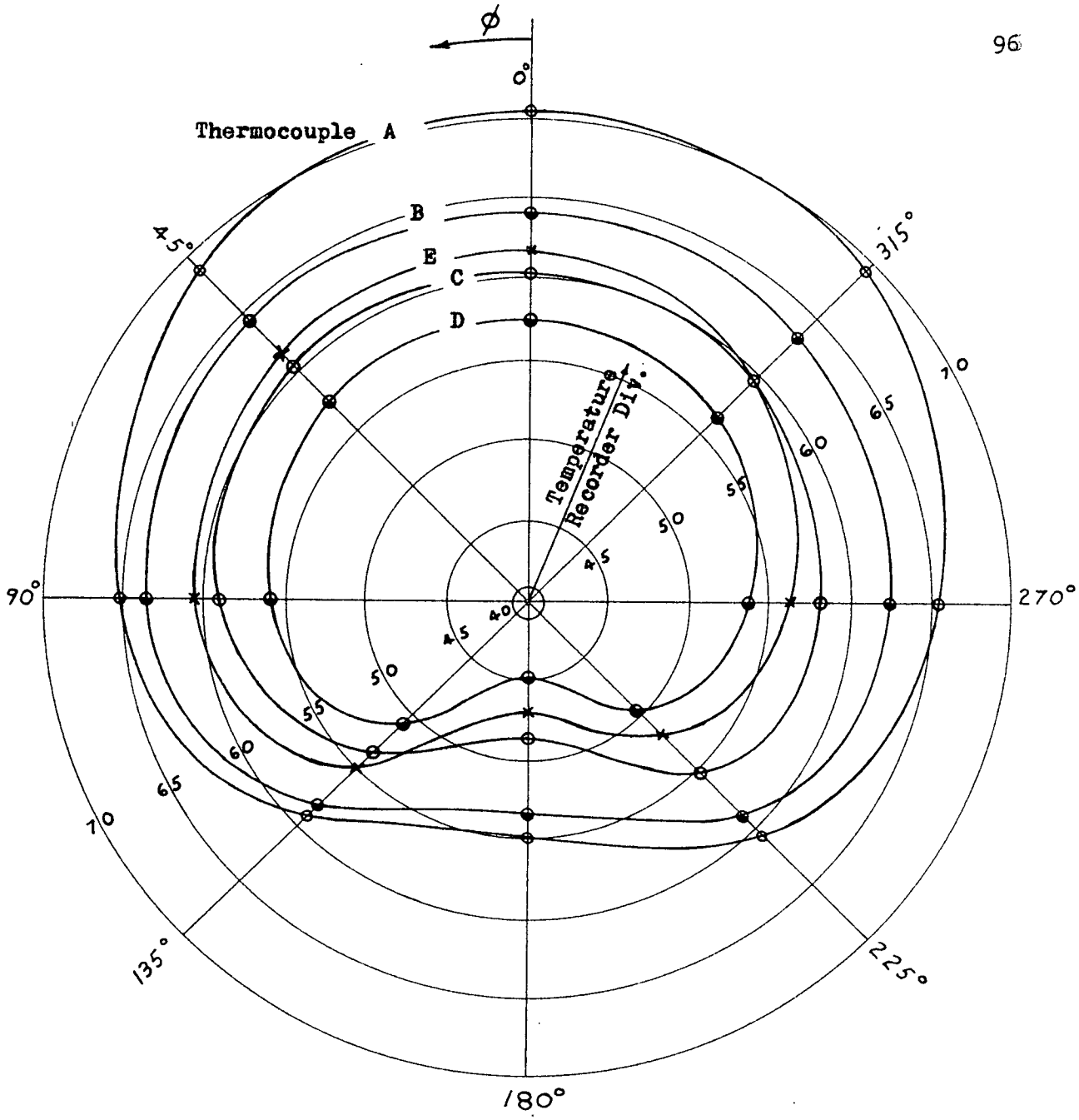


FIGURE 17

CIRCUMFERENTIAL TEMPERATURE DISTRIBUTIONS
ALONG A TUBE INCLINED AT 45°
(RUN 39, Δt_m = 37.6F)

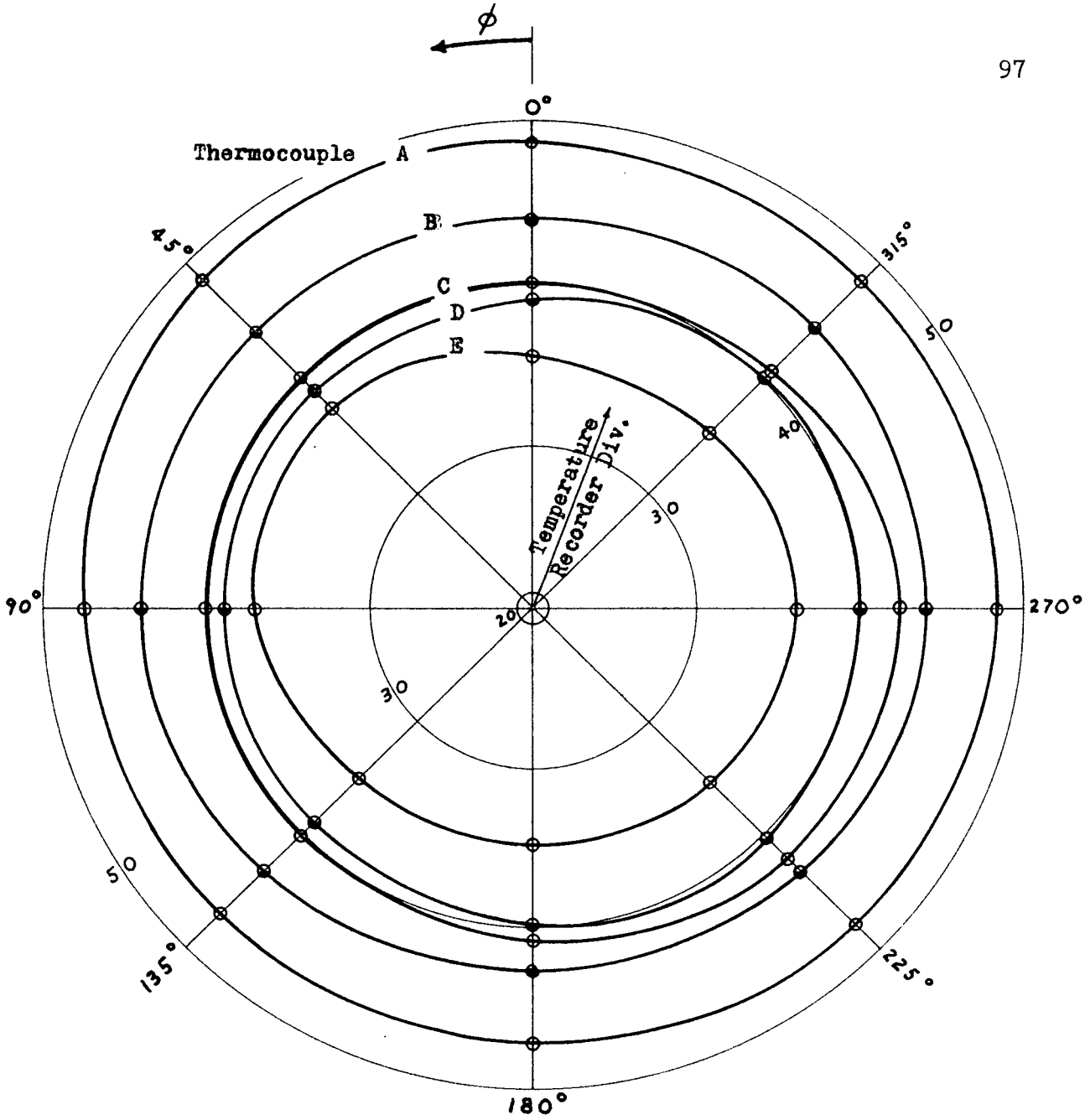


FIGURE 18

CIRCUMFERENTIAL TEMPERATURE DISTRIBUTIONS
ALONG A VERTICAL TUBE
(RUN 5, Δt_m = 35.4F)

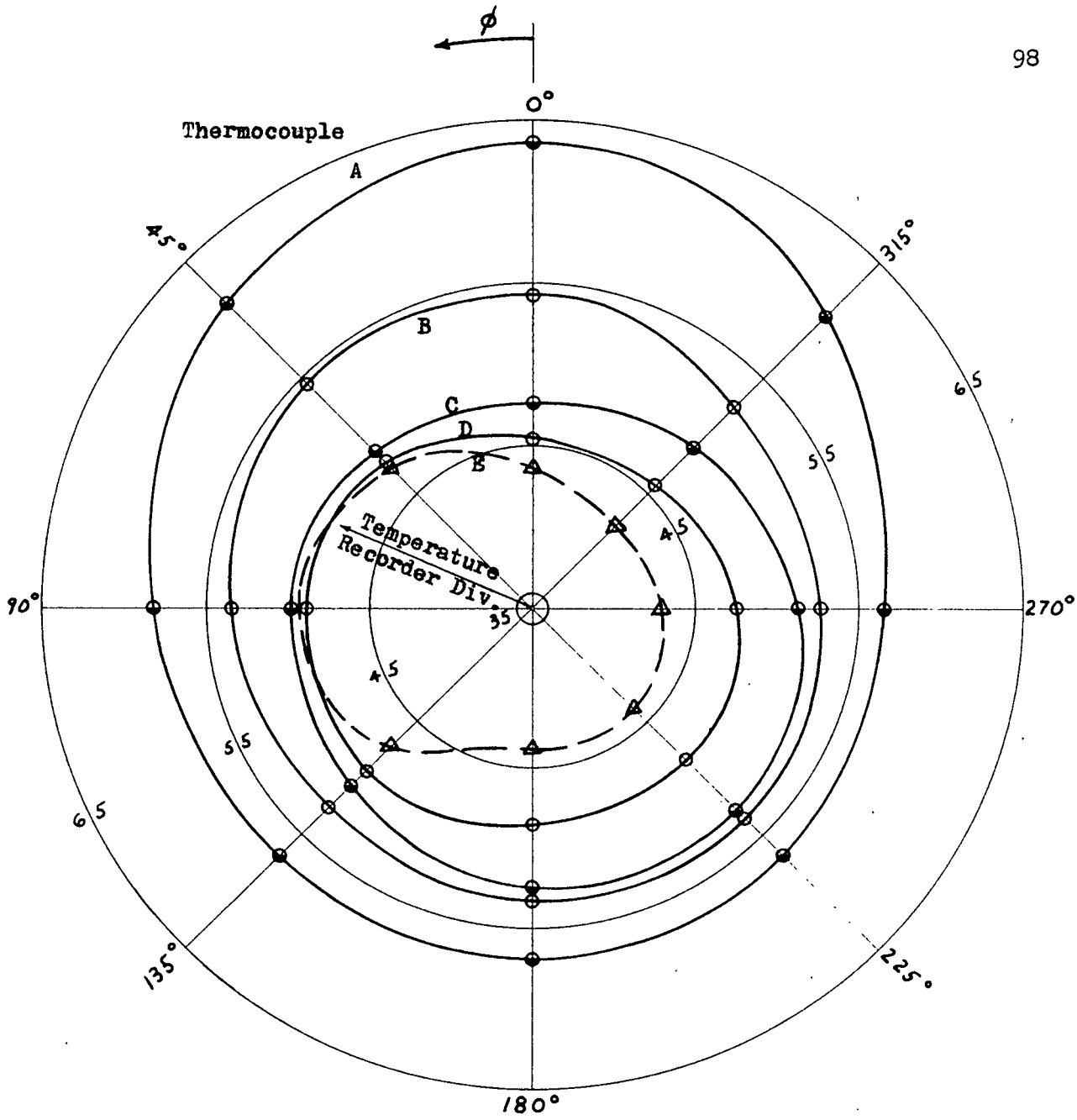


FIGURE 19

CIRCUMFERENTIAL TEMPERATURE DISTRIBUTIONS
ALONG A VERTICAL TUBE
(RUN 6, Δt_m = 45.6F)

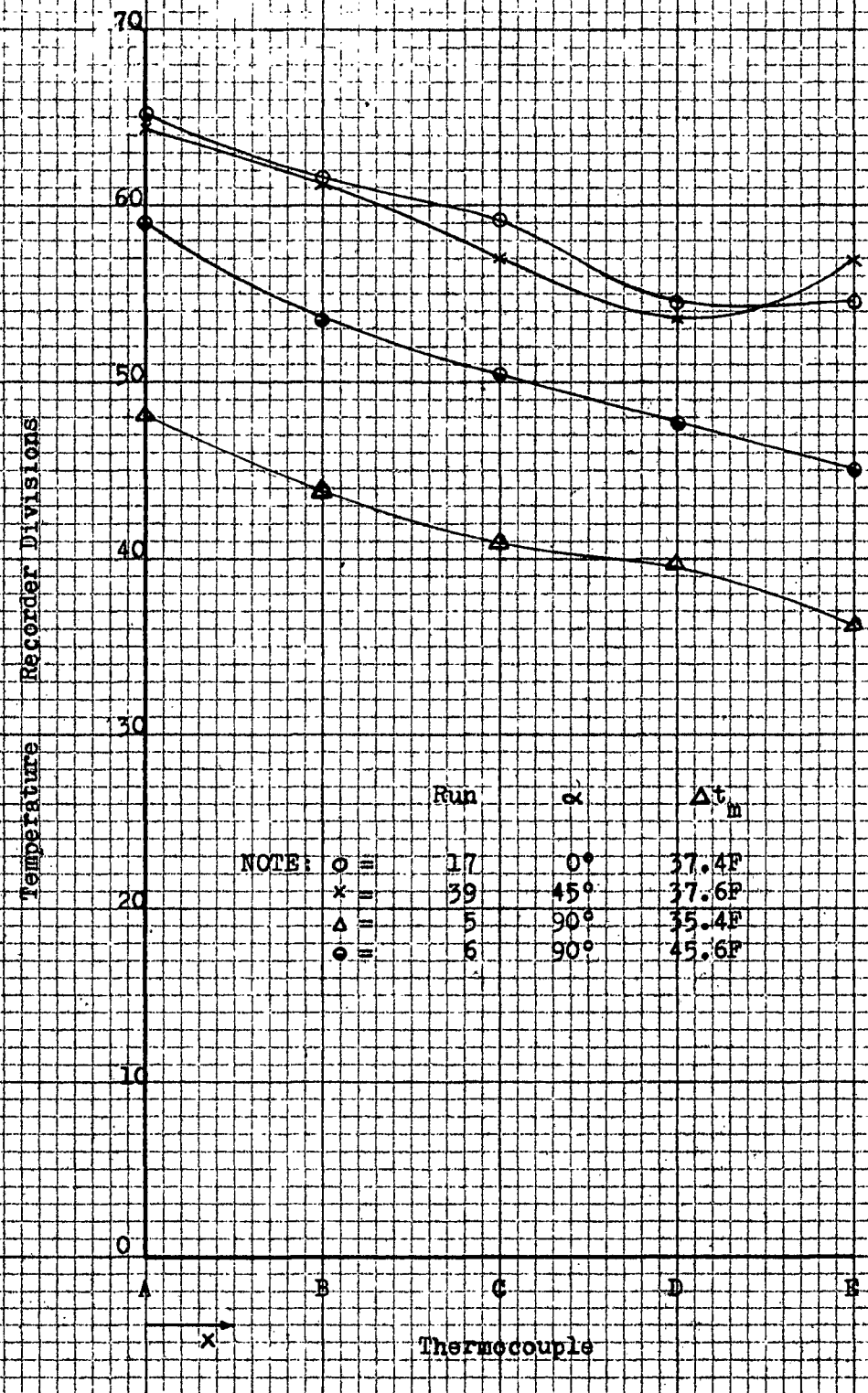


FIGURE 20

AVERAGE CIRCUMFERENTIAL TEMPERATURE
FOR EACH TUBE-WALL THERMOCOUPLE

FIGURE 21

VARIATION OF THE ACTUAL HEAT-TRANSFER COEFFICIENT WITH THE MEAN TEMPERATURE DIFFERENCE AND THE INCLINATION

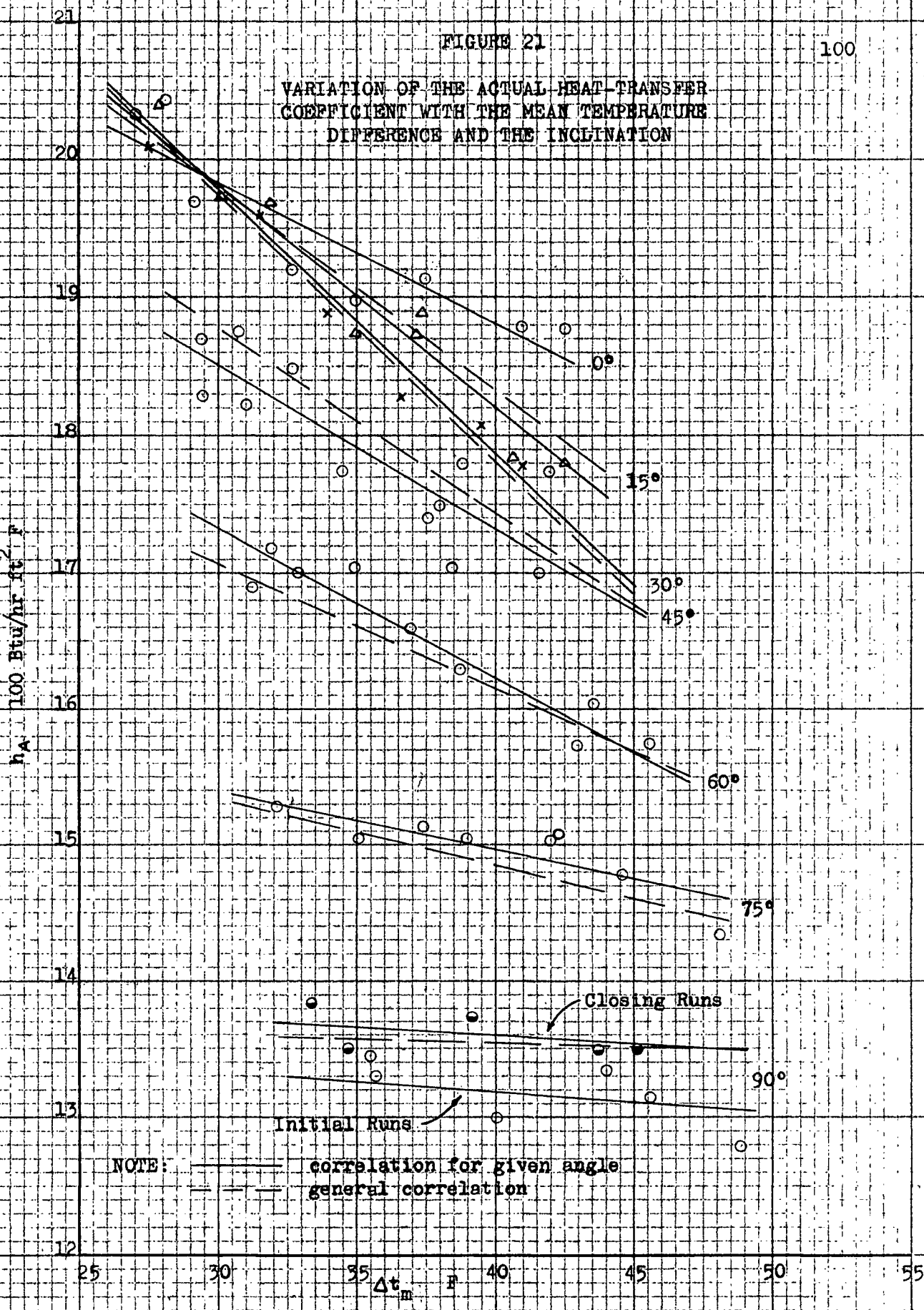


FIGURE 22

VARIATION OF THE RATIO OF THE ACTUAL COEFFICIENT TO THE THEORETICAL WITH THE MEAN TEMPERATURE DIFFERENCE AND THE INCLINATION

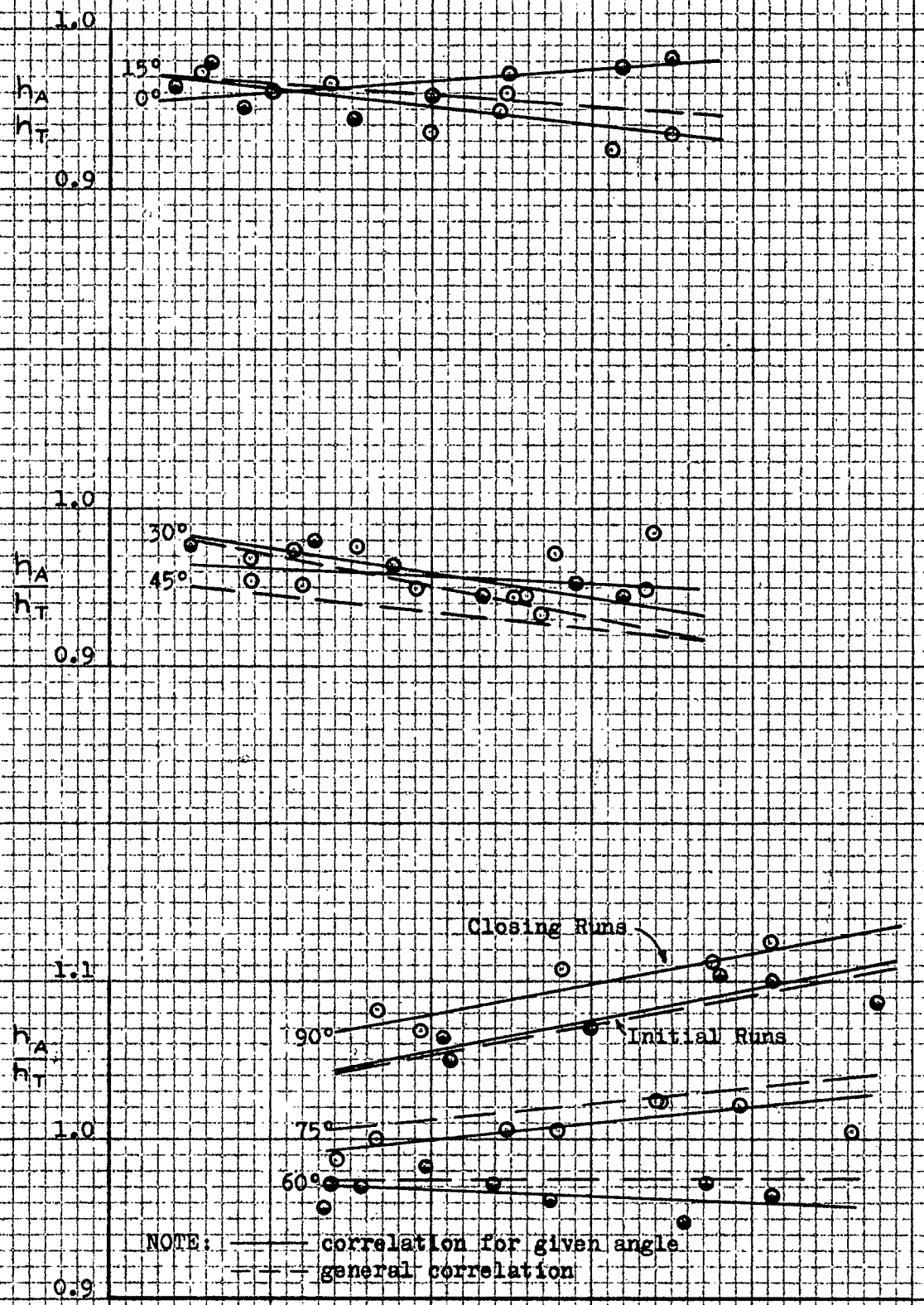


FIGURE 23

VARIATION OF THE ACTUAL HEAT-TRANSFER COEFFICIENT WITH THE OVERALL TEMPERATURE DIFFERENCE AND THE INCLINATION

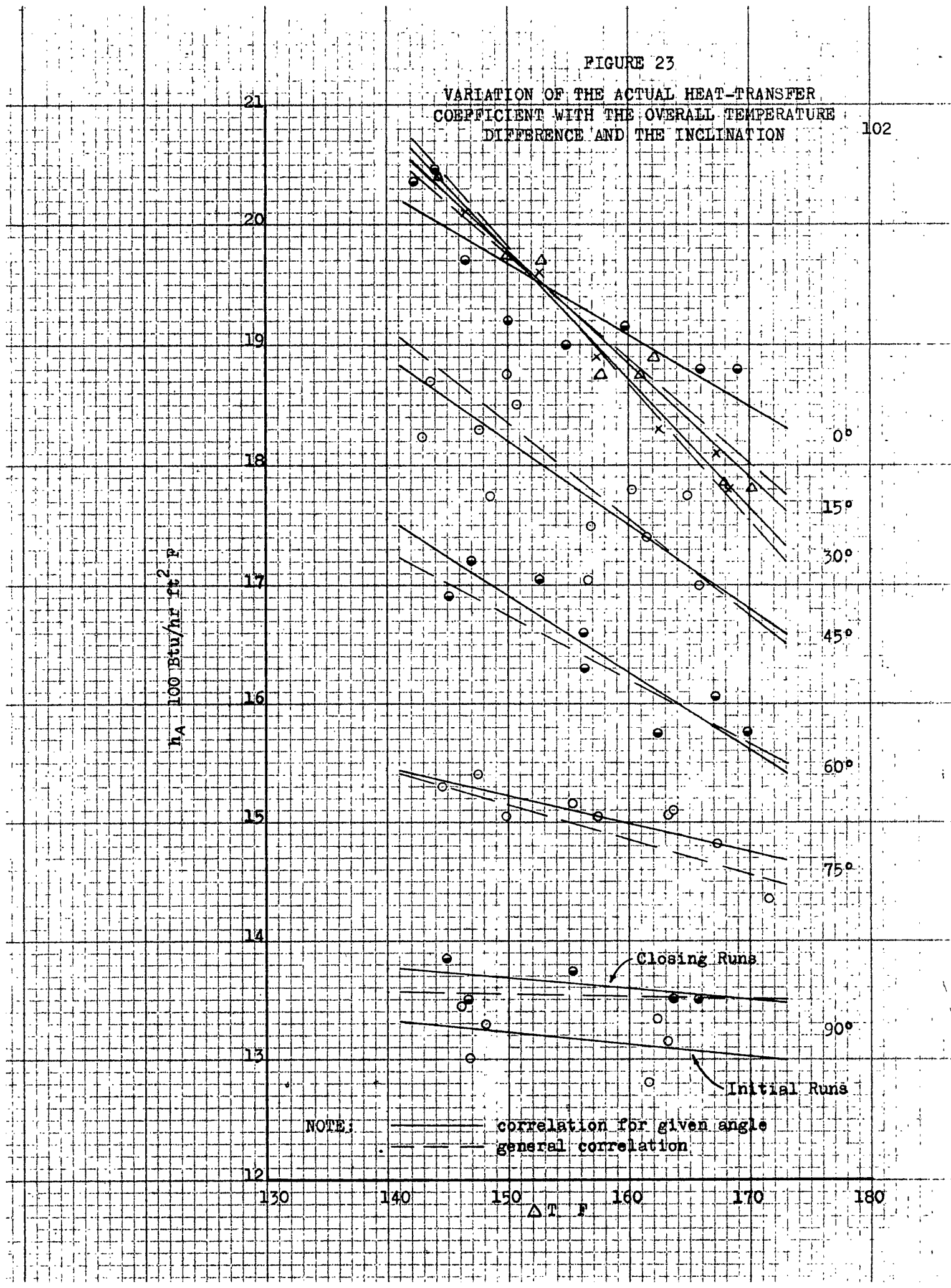
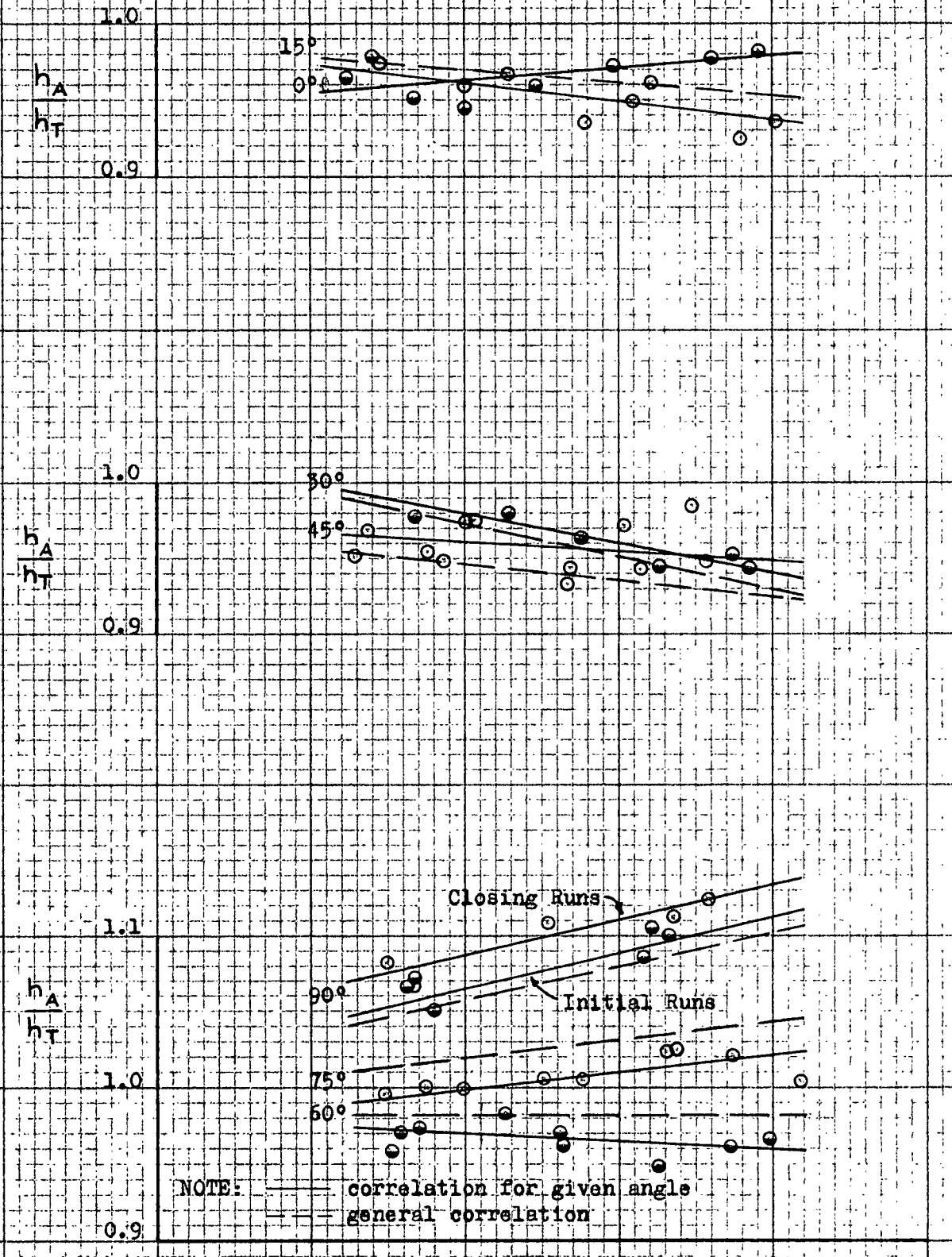


FIGURE 24

VARIATION OF THE RATIO OF THE ACTUAL COEFFICIENT TO THE THEORETICAL WITH THE OVERALL TEMPERATURE DIFFERENCE AND THE INCLINATION



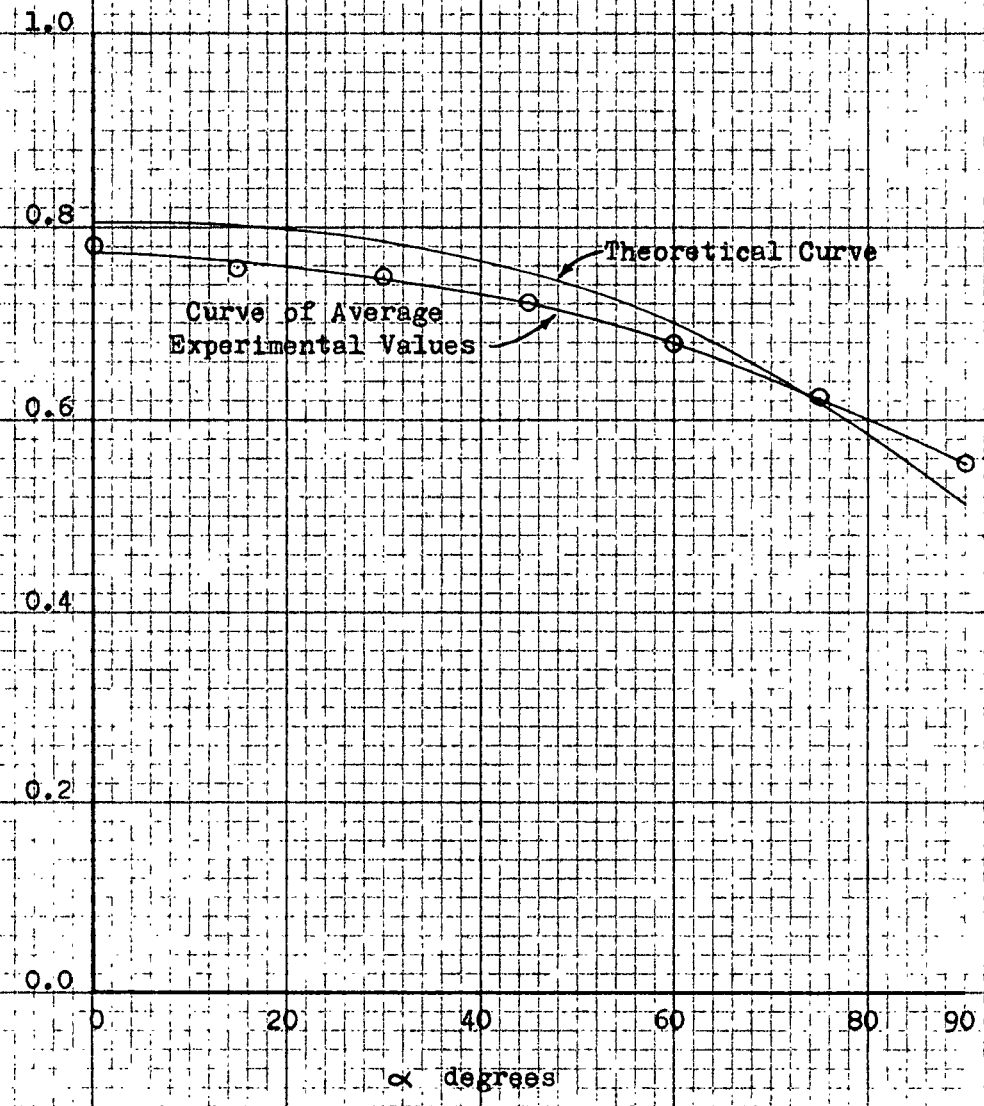


FIGURE 25

VARIATION OF THE MEAN HEAT-TRANSFER
COEFFICIENT WITH THE INCLINATION

FIGURE 26

VARIATION OF THE HEAT-TRANSFER RATE
WITH THE OVERALL TEMPERATURE DIFFERENCE
AND THE INCLINATION, FOR THE STANDARD TEST PIPE

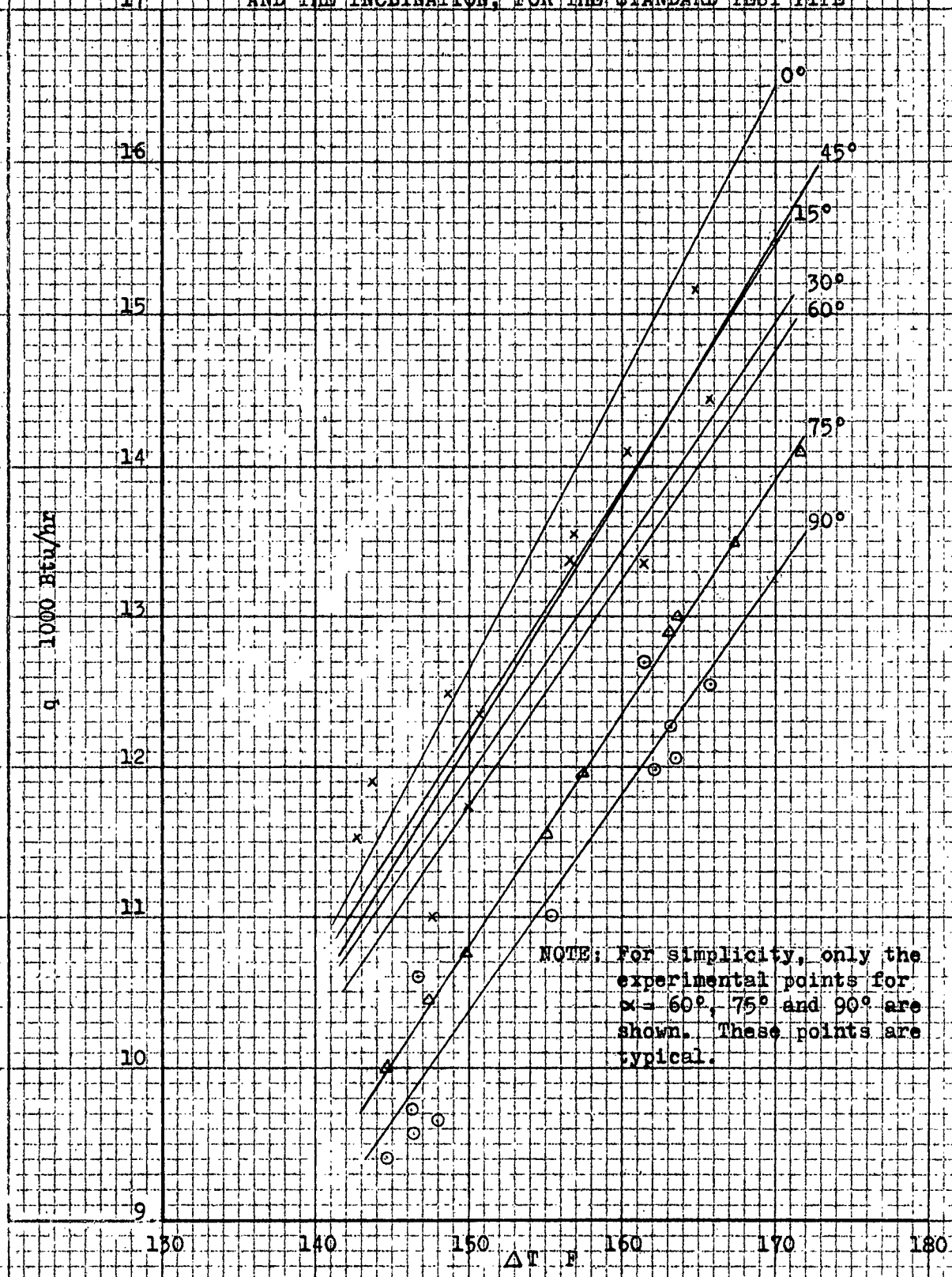
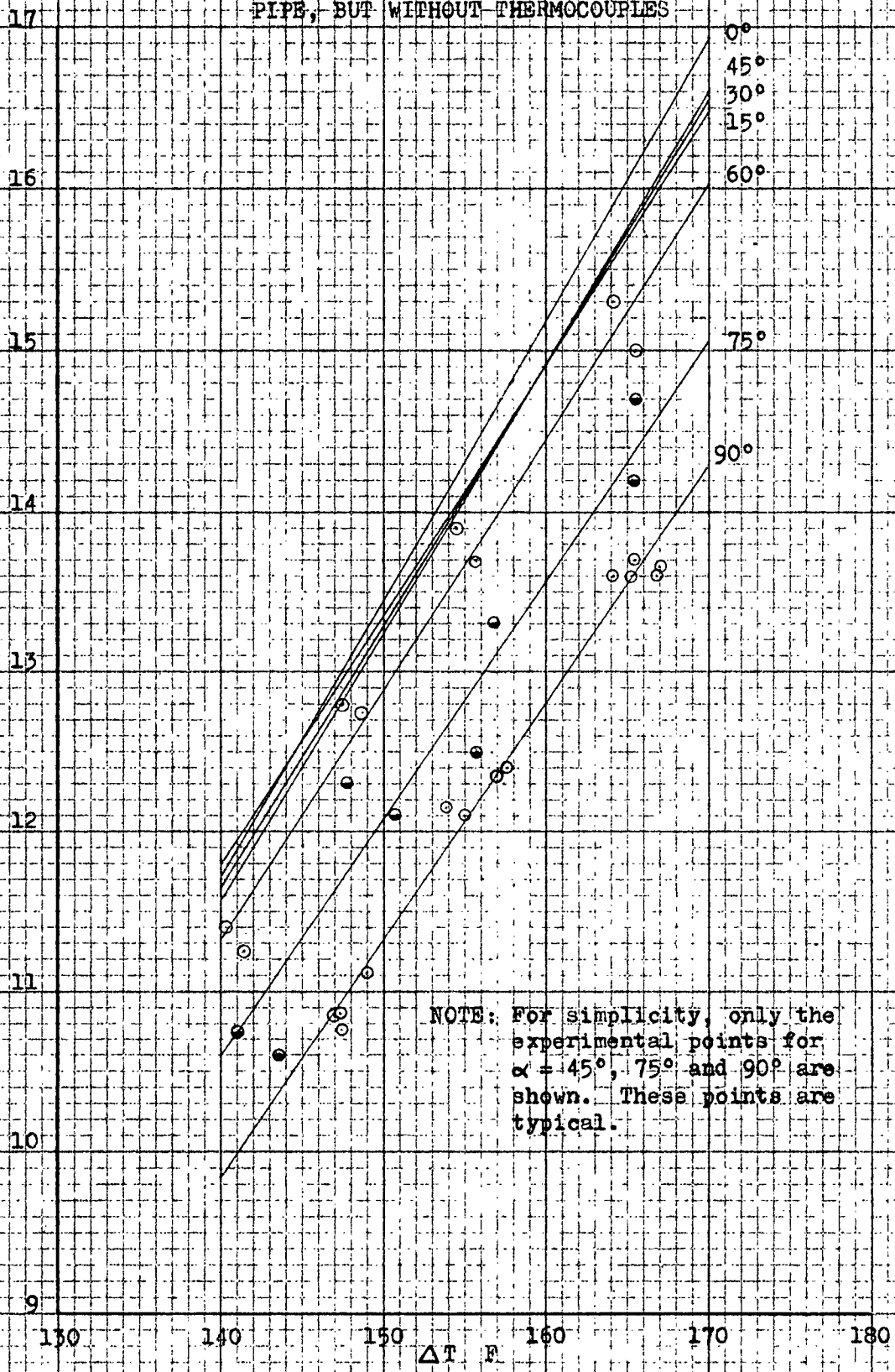


FIGURE 27

VARIATION OF THE HEAT-TRANSFER RATE WITH THE OVERALL TEMPERATURE DIFFERENCE AND THE INCLINATION, FOR A SIMILAR TEST PIPE, BUT WITHOUT THERMOCOUPLES



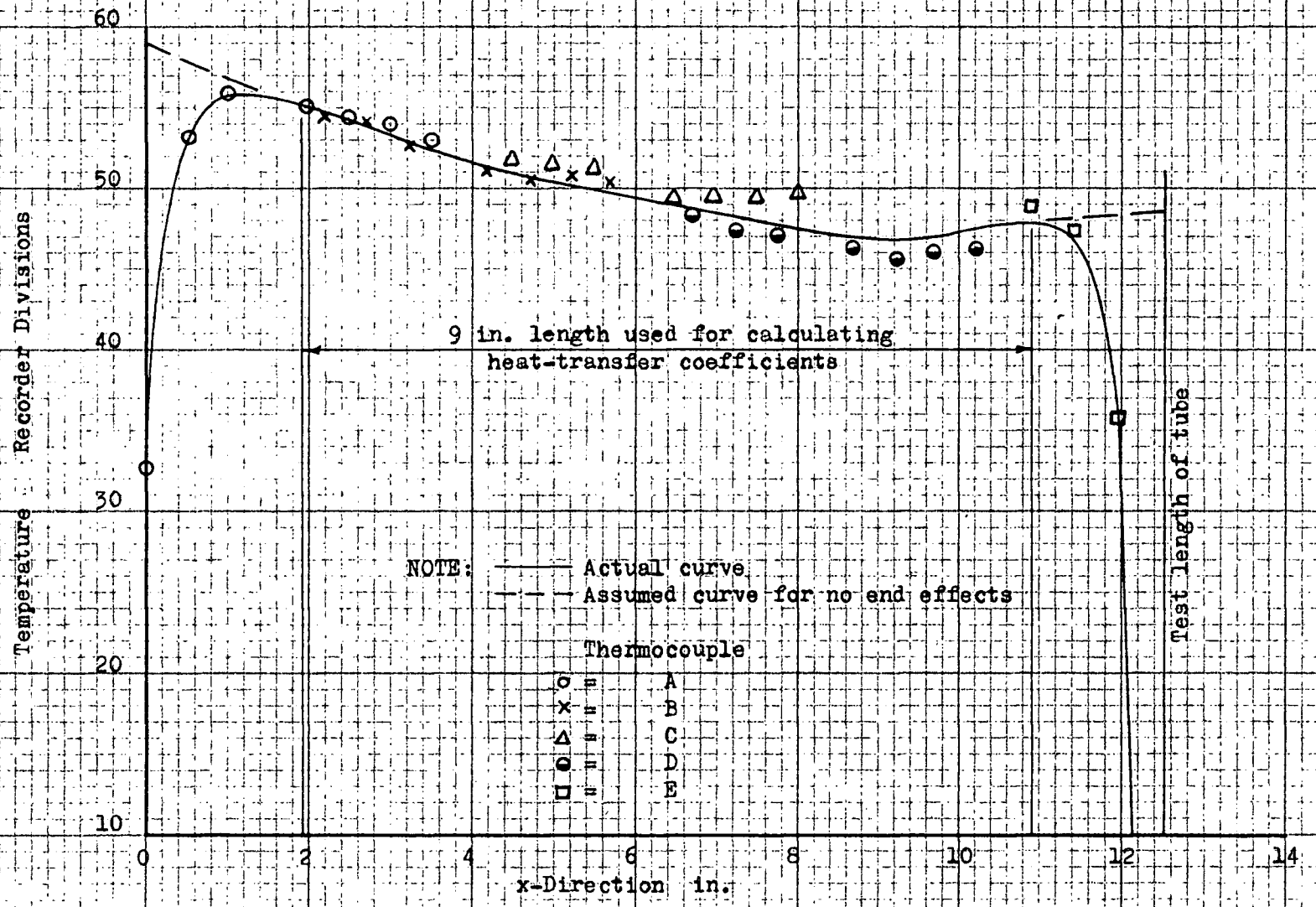


FIGURE 28

VARIATION OF THE AVERAGE CIRCUMFERENTIAL TEMPERATURE WITH THE DISTANCE ALONG A HORIZONTAL TUBE

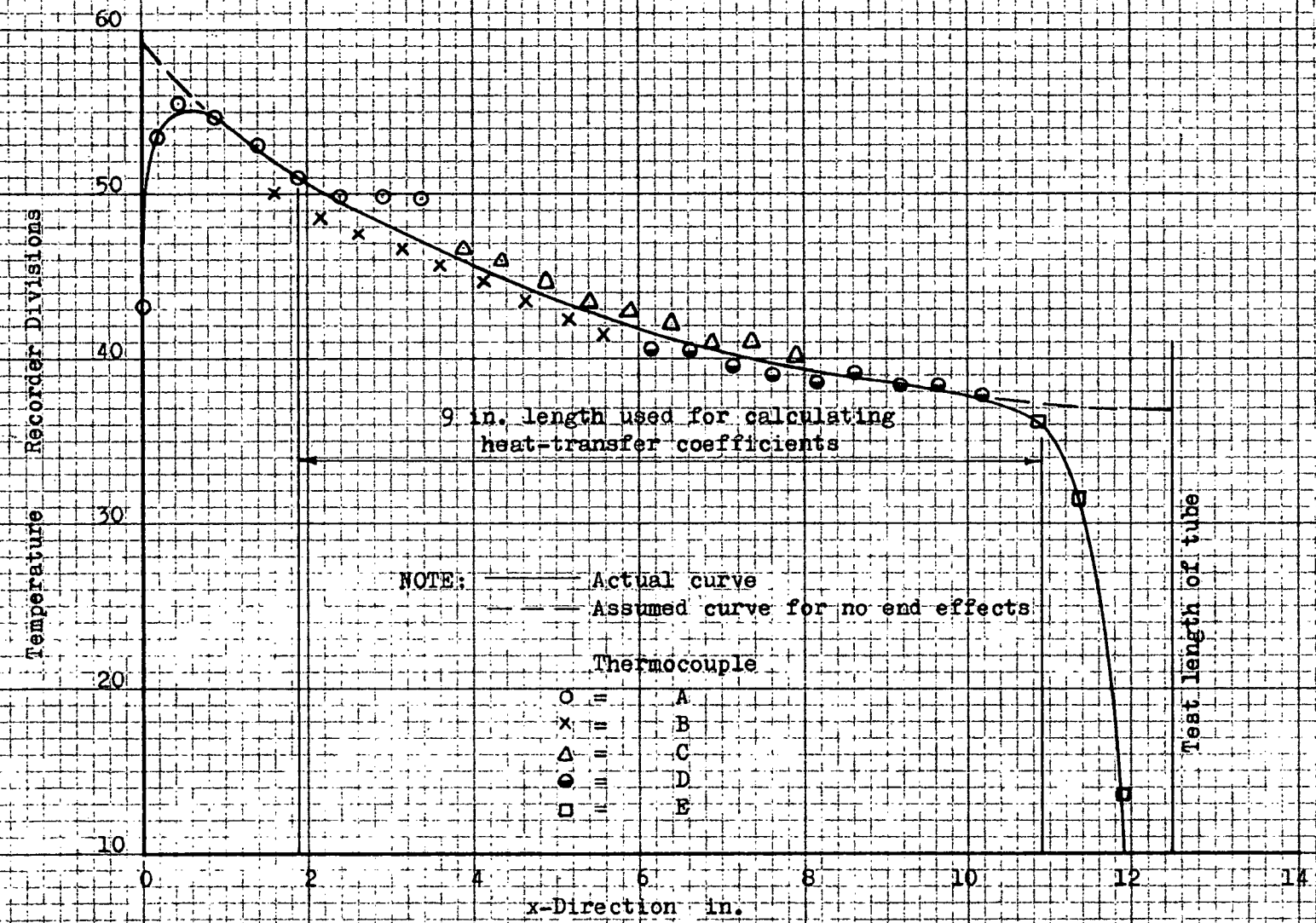
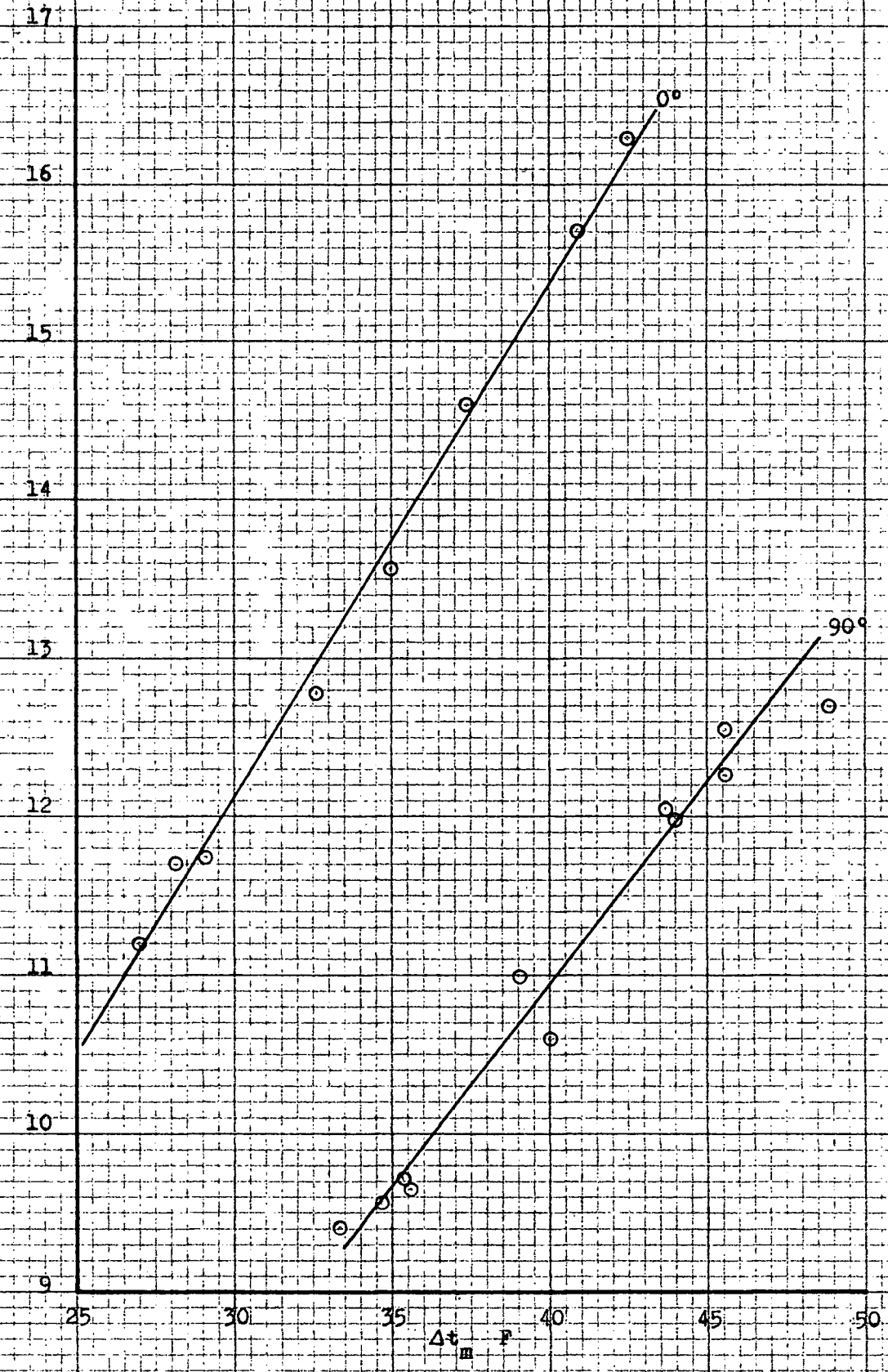


FIGURE 29

VARIATION OF THE AVERAGE CIRCUMFERENTIAL TEMPERATURE WITH THE DISTANCE ALONG A VERTICAL TUBE

FIGURE 30

VARIATION OF THE HEAT-TRANSFER RATE WITH THE MEAN TEMPERATURE DIFFERENCE



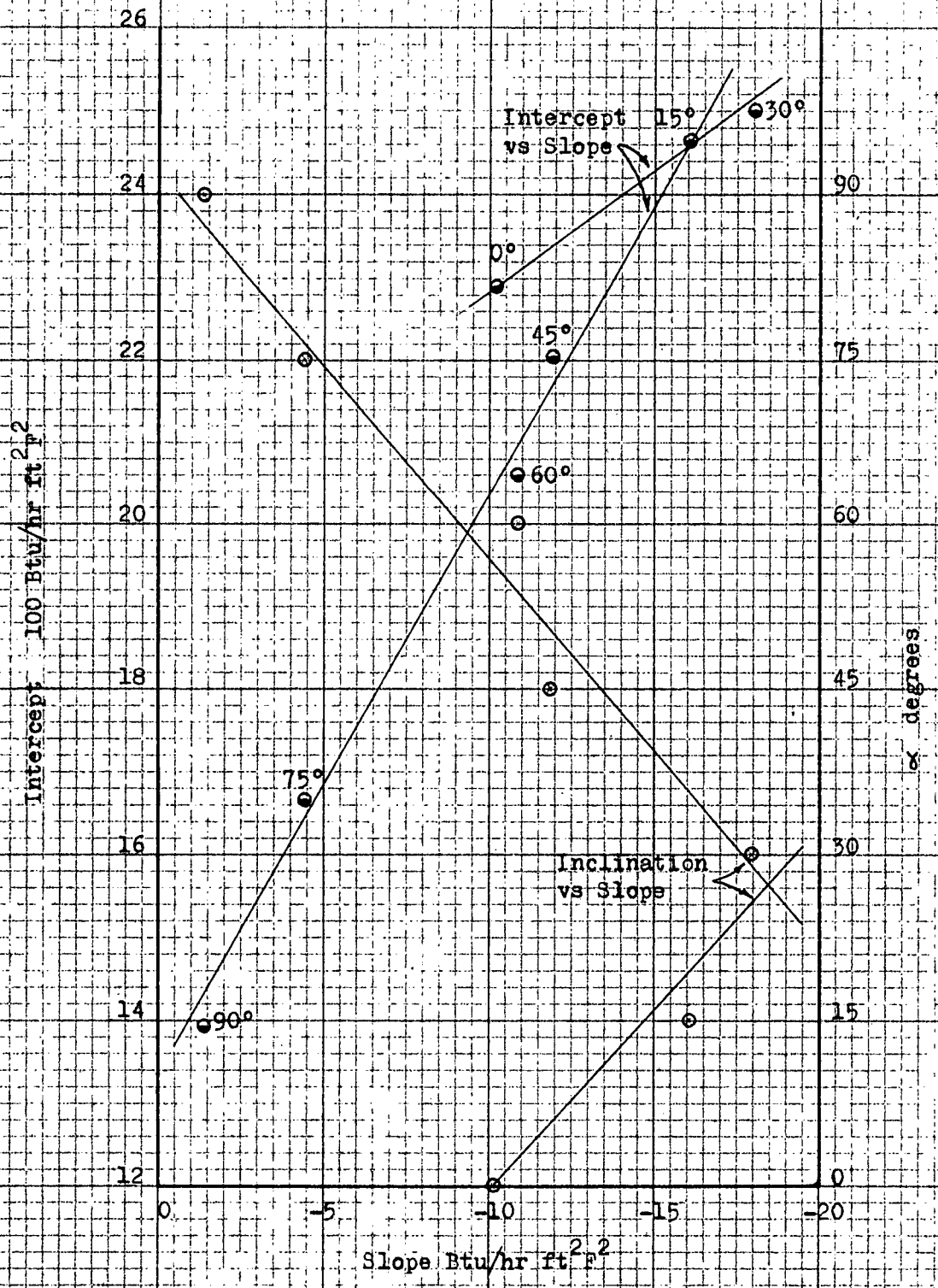


FIGURE 31

VARIATION OF INTERCEPT AND INCLINATION
WITH SLOPE FOR THE EQUATIONS OF
THE ACTUAL COEFFICIENT WITH RESPECT
TO THE MEAN TEMPERATURE DIFFERENCE

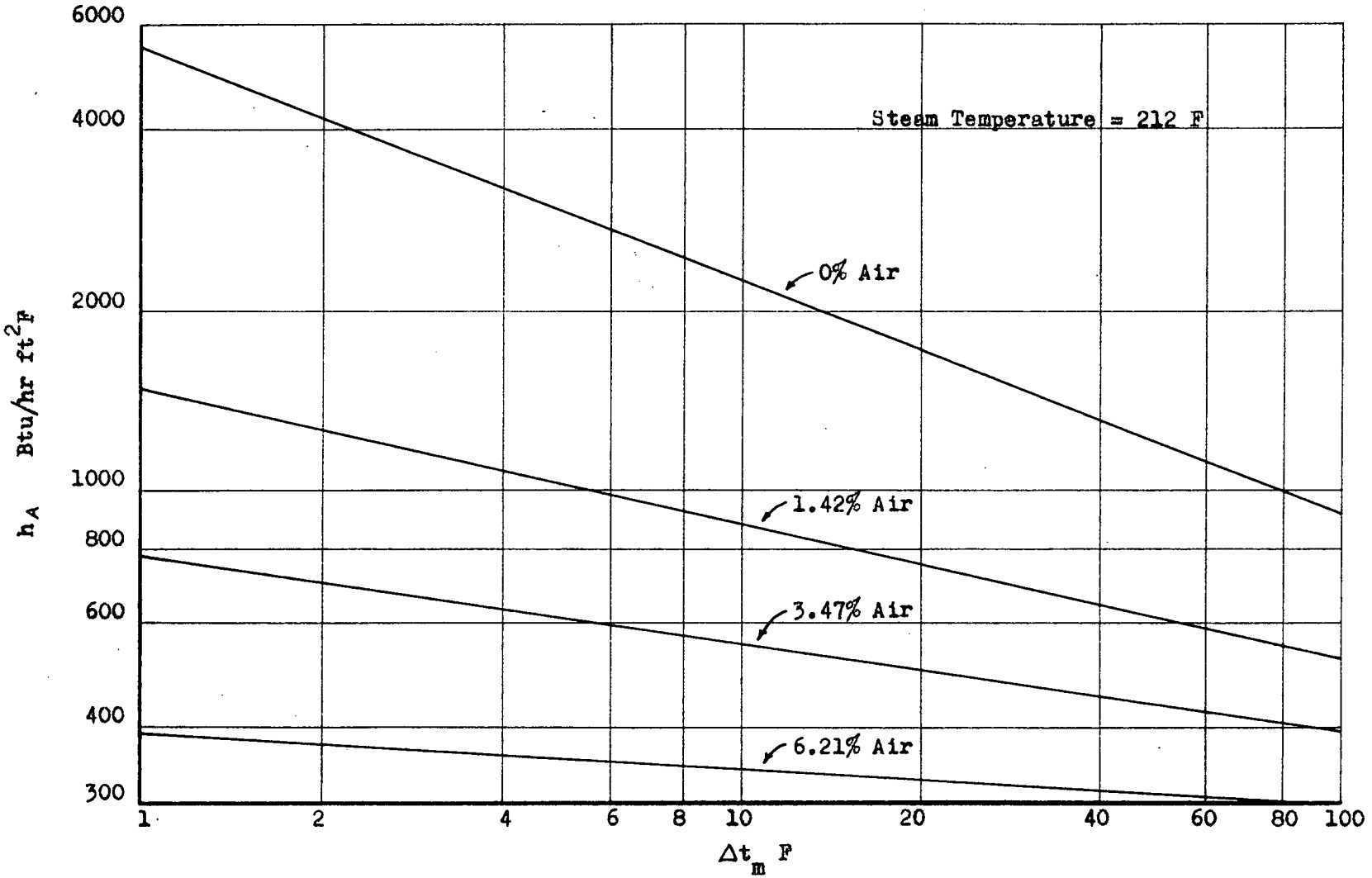


FIGURE 32

THE EFFECT OF THE CONCENTRATION OF AIR ON THE HEAT-TRANSFER COEFFICIENT FOR A HORIZONTAL TUBE [SELECTED DATA FROM REFERENCE (12)]

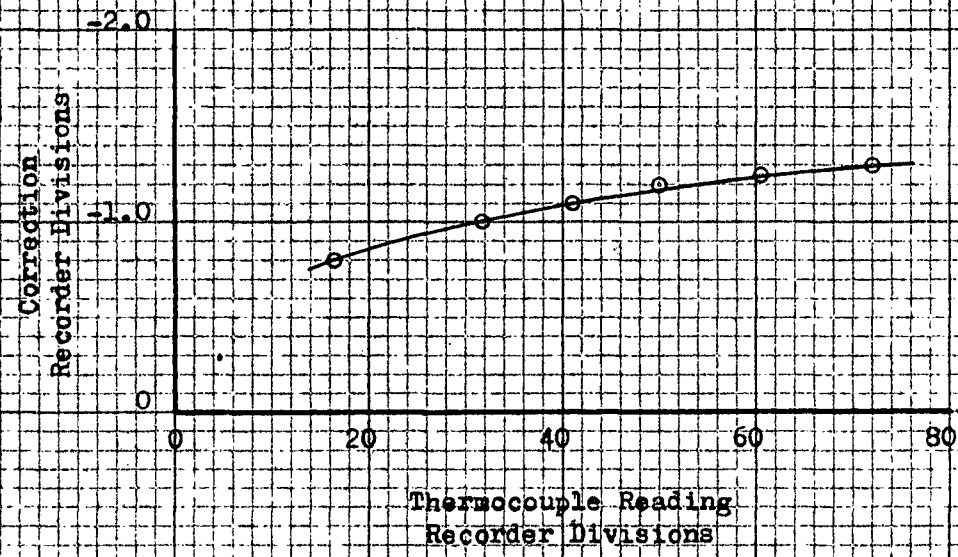


FIGURE 33

TUBE-WALL THERMOCOUPLE
CORRECTION