DESIGN OF AN APPARATUS TO EXPERIMENTALLY VERIFY FILMWISE CONDENSATION THEORY ON SMALL DIAMETER HORIZONTAL TUBES

by

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NOMENCLATURE

A - surface area

c_p - constant pressure specific heat

D - diameter

g - acceleration due to gravity

g_c - dimensional constant $(32.2\ lb_m\ ft/lb_f\text{-}sec^2)$

h - heat transfer coefficient

h_{fg} - heat of vaporization

k - thermal conductivity

L - length

$\dot{m}$ - cooling water flow rate

Nu - Nusselt Number $(h\ D_o/k)$

Nu' - Nusselt Number for small diameter tubes $(h\ D_o/k)$

Pr - Prandtl Number $(\mu\ c/k)$

$\dot{q}$ - heat transfer rate

R - resistance to heat transfer

Re - internal Reynolds Number $(4\dot{m}/\pi D_i \mu)$

T - temperature

ΔT - temperature difference $(T_v - T_{w,o})$

ξ - acceleration parameter $\left(\frac{k_c \Delta T}{\mu_c h_{fg}}\right)$

μ - viscosity

ρ - density

σ - surface tension

Subscripts

B - bulk

c - pertaining to the condensate

i - pertaining to the inside of the tube

M - pertaining to the tube material

o - pertaining to the outside of the tube

t - pertaining to the tube

v - pertaining to the vapor

w - pertaining to the tube wall

1 - cooling water inlet

2 - cooling water exit

I. INTRODUCTION

Condensation is a common phenomenon which is vital to many industries. Tube type condensers are often used to change a vapor to its liquid phase so that it can be easily pumped back into the high pressure regions of a system.

Condensation occurs whenever a vapor is cooled below its saturation temperature. The energy released in the condensation process is transferred to the cooling substance. In the case of tube condensers, heat is transferred through the tube to the cooling fluid, which may be water.

There are large differences in the reported results of condensation heat transfer studies. These differences are probably the result of uncertainity about the experimental conditions. In many studies, pertinent information concerning the experiment was not recorded. The present project was undertaken to provide a means to obtain complete, accurate, and repeatable condensation heat transfer data.

There are three major condensation modes. If, upon condensing, the condensate collects in drops along the surface leaving areas of the surface unwetted, the mode is called dropwise. If a continuous layer of condensate is established over all of the surface, the mode is called filmwise. If there are areas of filmwise condensation separated by areas of dropwise condensation, the mode is mixed or streaky.

Dropwise condensation yields the highest heat transfer rates of the three modes and filmwise the lowest. Since in condenser design it is usually desired to transfer a specified amount of heat in the minimum

volume, dropwise is usually the preferred mode. However, the mixed mode, not the dropwise or filmwise modes, is the mode which is usually found [1]. As should be anticipated, a large range of heat transfer rates can be obtained depending upon the proportions of dropwise to filmwise areas. A conservative estimate of the heat transfer rate can be obtained by assuming completely filmwise condensation.

Analyses have been performed for filmwise condensation on the outside of horizontal tubes [2,3,4,5,6]. However, the effect of the assumptions made in these analyses on the accuracy of the results should be substantiated by experimental data. Complete, accurate, and consistent data for the heat transfer coefficient for filmwise condensation on small diameter horizontal tubes has not been reported. The objectives of this project were to design an apparatus which could be used to accurately determine this coefficient for the condensation of a dry, saturated, and stagnant vapor and to obtain preliminary data for pure steam.

The heat transfer coefficient is defined from Newton's "law of cooling" as

$$h = \frac{\dot{q}}{A(T_v - T_{w,o})} \quad .$$

All of the quantities necessary for the above calculation are easily determined using straight forward techniques except the surface temperature. There are a number of methods available for measuring surface temperature. The accuracy of the surface temperature measurement has a significant effect on the accuracy of the coefficient.

The techniques and concepts incorporated in the design of the apparatus to measure heat transfer coefficients are discussed in the following sections. A suggested procedure for the operation of the apparatus to obtain consistent results is also included.

II. LITERATURE REVIEW

Literature relating to the present study concerns four topics: (1) analyses of filmwise condensation on the outside of horizontal tubes, (2) the effect of surface tension forces, (3) the effects of noncondensible gases and contamination, and (4) experimental techniques and results.

The first reported analysis of filmwise condensation on horizontal tubes was developed by Nusselt in 1916. Nusselt's assumptions were [2,7]

1) the vapor velocity is negligible,
2) the vapor is pure, dry, and saturated,
3) the condensing and film surfaces are smooth,
4) the condensing surface temperature is uniform,
5) the condensate properties are constant and can be evaluated at the arithmetic mean of T_v and $T_{w,o}$,
6) the velocity profile is fully developed over the entire cylinder and the flow is laminar,
7) the only non-negligible forces are the viscous and gravitational forces, and
8) convection effects are negligible.

Assumptions one through three are under the control of the experimenter. These conditions can be approximated in an experimental facility. Assumption four can be approximated by using the average surface temperature. The fifth assumption affects the calculations and not

the design or operational procedures. The remaining assumptions are not under the control of the experimenter. Their validity should be verified by experimental data. Nusselt's analysis yielded

$$h = .725 \left[\frac{g\ \rho_c(\rho_c-\rho_v)k_c^{\ 3}h_{fg}}{D_o\mu_c\ \Delta T} \right]^{\frac{1}{4}}$$

Nusselt's analysis assumed that the only energy which was transmitted through the tube was the energy released by the change in phase. However, if the energy released by cooling the condensate from T_v to $T_{w,o}$ is of the same order of magnitude as the energy of vaporization, it should not be neglected. This condition may result from a large difference between the saturation temperature of the vapor and the wall surface temperature and/or a large specific heat. An analysis which included this effect was developed by Bromley [3]. His result was

$$h = .725 \frac{(1 + \frac{3}{8}\frac{\Delta Tc_p}{h_{fg}})}{(1 + \frac{7}{30}\frac{\Delta Tc_p}{h_{fg}})^{\frac{3}{4}}} \left[\frac{g\ \rho_c(\rho_c-\rho_v)h_{fg}k_c^{\ 3}}{D_o\mu_c\ \Delta T} \right]^{\frac{1}{4}} .$$

The boundary layer analysis developed by Sparrow and Gregg [4] included the effect of the inertia of the condensate film. It was assumed that the velocity gradient in the film was zero at the liquid-vapor boundary. For $Pr_c > 1$ (small velocity gradients in the film), the analysis predicts an increase in the heat transfer coefficient as the heat capacity of the film increases. For $Pr_c < 1$ (large velocity gradients in the film), a decrease in the heat transfer coefficient as the heat capacity of the film increases is predicted. For $Pr_c \approx 1$, the result differed from Nusselt's only by the constant, which was changed

to 0.733. It was noted that for small diameter tubes the boundary layer assumptions were inaccurate. A correction equation for small diameter tubes was included. The equation was

$$Nu' = \frac{2}{\ln(1 + \frac{2}{Nu})} .$$

Chen's [5] analysis included the effect of a nonzero velocity gradient at the liquid-vapor boundary. This difference from the previous boundary layer analysis is important only at $Pr_c < 1$. An additional decrease was predicted for this case. His analysis predicted a constant of 0.728.

One of the forces neglected in these analyses was the surface tension of the film. According to Markowitz, Mikic, and Bergles [8]

> "Surface tension could give rise to large pressure gradients in the liquid film due to the varying curvature of the condensate surface. Large pressure gradients necessarily lead to thin films, and coefficients of heat transfer several times larger than those observed in normal film condensation can be achieved."

This phenomenon may have affected the data reported by Ünsal [9]. Henderson and Marchello [10] tried to correlate deviations from Nusselt's result with the Ohnesorge Number, $\mu_c/[(\rho_c g_c D_o \sigma_c)^{1/2}]$, which is an indicator of the importance of viscous forces to surface tension forces used to correlate atomization data. However, a substantial portion of the data they used to obtain the correlation is questionable because the condensation mode was not observed. The results of the analysis reported by Buznik, Aleksandrov, and Smirnov [6], indicate that

> "under the conditions of laminar film flow in the condensation of steam on a smooth horizontal tube, the effect of surface tension forces on the heat transfer coefficient should not exceed 5 to 6 percent."

Results were reported for steam at atmospheric pressure for tubes with approximately the same outer diameters as those used in this study. It should be determined experimentally if the effect of surface tension is substantial or if it is small.

Two factors which affect condensation heat transfer rates are the condensation mode and the presence of noncondensible gases. According to Hampson [1]

> "Any contaminate in the vapor, including a noncondensible gas, will eventually affect the mode of condensation."

The drastic effect of small amounts of impurities was noted by Merte [11]. He reported that 3 to 50 parts per million of contaminant promoted dropwise condensation. In addition to its possible effect on the condensation mode, noncondensible gases also affect the value of the heat transfer coefficient. A noncondensible gas film forms around the tube and decreases the heat transfer coefficient [1]. Analytical results indicate that a 90 percent reduction in the heat transfer coefficient for the condensation of steam can be caused by a mass fraction of air of approximately 0.02 [12].

In spite of the quantity of data reported for filmwise condensation on horizontal tubes, complete and reliable data for small diameter tubes is not available. Some experiments [7,13,14,15,16] were performed without visual verification of the condensation mode. Other experiments

[17,18,19] were performed on large diameter tubes. An experiment which was applicable [20] reported results in terms of a "condensation coefficient" and did not include the data necessary to calculate the heat transfer coefficient.

Experimental data for the heat transfer coefficient during film-wise condensation on horizontal tubes, based on this review of the literature, is not complete. Because the results of all of the other analyses reduce to Nusselt's result (except for the constant) for $c_p \Delta T/h_{fg} = 0$ (approximately true for most common liquids), comparison of experimental data with his result is sufficient. Data should be obtained for tubes with diameters sufficiently large that the effect of surface tension forces is negligible as well as for small diameter tubes.

III. DESIGN PROCEDURE

The items which were considered to be essential in the design of the apparatus were

1) the ability to visually verify the condensation mode,
2) a method to accurately determine the heat transfer rate,
3) the complete removal of noncondensible gases from the system, and
4) a method to accurately determine the average tube surface temperature.

If these four criteria could be achieved, accurate and repeatable data would result.

It was decided that the easiest and most complete visual check of the condensation mode could be accomplished by using a glass condenser casing. However, the casing would have to be insulated and externally heated to prevent internal fogging.

The measurement of the heat flow rate was to be accomplished by measuring the flow rate of the cooling water and its inlet and exit bulk temperatures. The heat flow rate would then by determined by

$$\dot{q} = \dot{m}\, c_p (T_{B,2} - T_{B,1}) \ .$$

Care was taken to insure that only the tube transferred heat with the cooling water at the time measurements were taken. The cooling water was insulated from the rest of the apparatus except at the exit mixing

chamber. Under steady state operating conditions heat was transferred to or from the cooling water only through the tube. It was also necessary to insulate the tube from the apparatus to minimize heat conduction through the tube to the water from the apparatus.

The removal of noncondensible gases was to be accomplished as completely as possible before the condensation process was begun. After condensing had started the system was to be closed and operated at a slight positive pressure so that small leaks would result in the loss of vapor from the system and not in the introduction of air into the system.

A method for measuring the average surface temperature has already been mentioned as an area of concern. The local tube surface temperature depends upon circumferential as well as axial position. Studies have shown that the circumferential temperature variation does not significantly affect heat transfer results [21,22,23]. The average temperature of the tube surface was obtained by measuring a circumferential average temperature at evenly spaced intervals along the length.

Five surface temperature measurement techniques were considered. One technique was to embed thermocouples in the surface or in the tube from the inside [14,24]. This method disrupts the film in the vicinity of the thermocouple and gives a local tube temperature which may differ from the average surface temperature. Using the tube as a resistance thermometer [18,20] requires extremely accurate equipment and then yields surface temperature only after additional analysis. An indirect technique first proposed by Wilson [25] in 1915 has also been used

[9,17]. This technique, however, was developed using a questionable assumption and can yield inaccurate results. For a further discussion of Wilson's technique see Appendix I. Optical pyrometers can also be used to measure surface temperature. A pyrometer indicates temperature by measuring the electromagnetic radiation emitted by the surface. In condensation heat transfer experiments, the radiation emitted by the condensing surface has a long wave length and is absorbed by the condensate film. The film also emits radiation from its surface. The temperature indicated by a pyrometer, therefore, would be the surface temperature of the condensate film, not the temperature of the condensing surface. A fifth method for obtaining the surface temperature is to use an electroplated thermocouple as the surface [26]. Thin layers of two materials, electroplated around the circumference of the original tube, become the condensing surface. The only disturbance to the film is where the lead leaves the surface. The use of small lead wire minimizes the disturbance. Since the entire circumference of the tube is used as the thermocouple, an average temperature is obtained. The latter technique was selected in this experiment.

Using the techniques indicated in this section all of the necessary measurements could be accurately obtained, observation of the condensation mode would be maximized, and noncondensible gases would be removed from the condensing section.

IV. APPARATUS

The experimental facility consisted of five parts: (1) the condensing section, (2) the test specimens, (3) auxiliary equipment, (4) vapor generator, and (5) instrumentation. The condensing section enclosed and supported the test specimens. The condensing section was furnished with connections for the auxiliary equipment and instrumentation. The test specimens included the thermocouples used to measure the surface temperature. Auxiliary equipment included the noncondensible gas removal system, cooling water mixing chambers, and a part which passed the internal thermocouple leads to an exterior location which was called a feedthrough. The vapor generator and instrumentation were separate major pieces of equipment. A discussion of unforeseen difficulties encountered during the design of the apparatus is also included.

Condensing Section

The condensing section permitted complete visual inspection of the condensation mode. The section encased and supported the test specimens but did not transfer heat to them. Heat was exchanged only between the cooling water and the test specimen. An atmosphere of dry, saturated, stagnant vapor, free of noncondensible gases, existed within the section. The section was sealed from the atmosphere.

The condensing section is shown in Fig. 1. The casing was standard 6-inch ips by 18-inch long Pyrex tee sections. One or two sections were used depending upon the length of the test specimen. The sections were

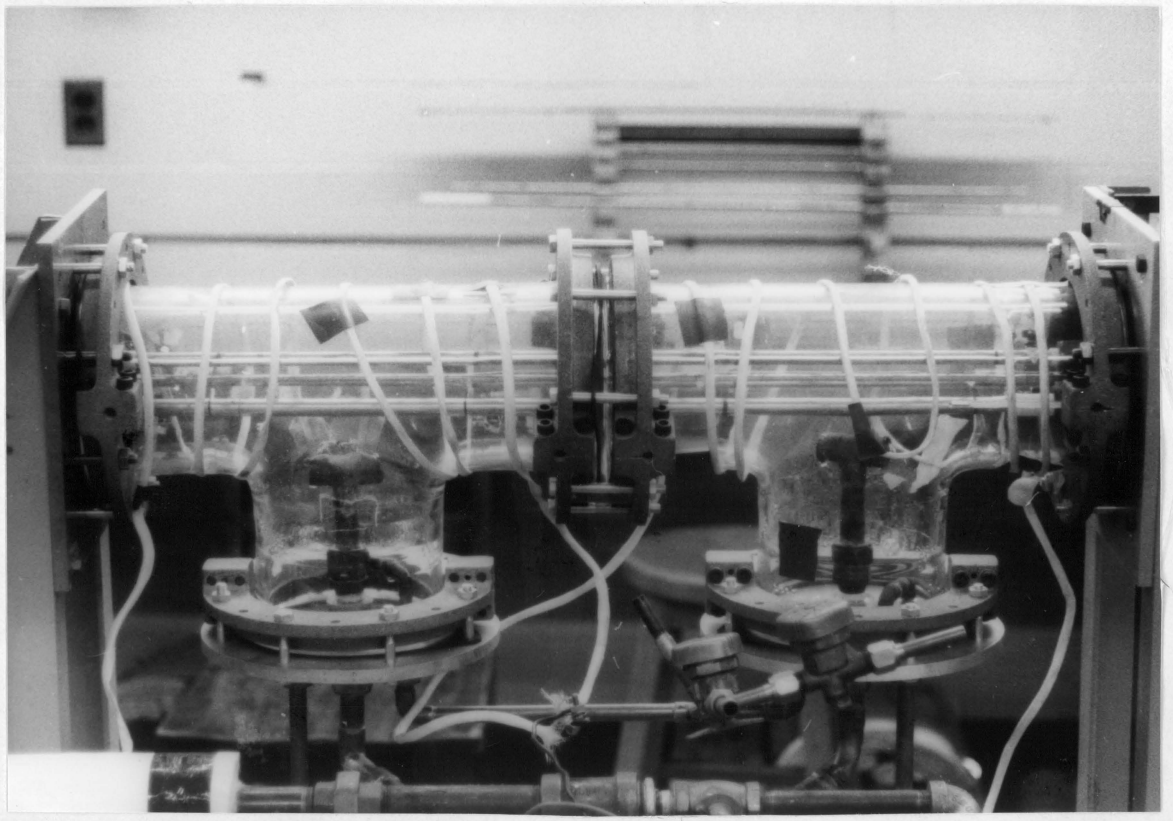

Fig. 1. Condensing Section

wrapped with insulated, electrically-heated nichrome wire which prevented internal fogging during operation of the experiment. Because of the relatively low temperatures at which the experiments were performed (220 F), the radiation emitted by the nichrome wire was long wave radiation which is not readily transmitted by glass. Radiation heat transfer, therefore, had an insignificant effect on the total heat transfer rate to the cooling water. The Pyrex sections were covered with 2 inches of fiber glass insulation during operation to minimize heat loss. Removable sections of insulation were cut to allow observation of the condensation mode.

The test specimens were supported as shown in Fig. 2. The tubes were insulated from the fittings by Teflon annuli and rubber gaskets. A table of annuli dimensions for each specimen is included in Appendix II. The inside of the tube was sealed from the condensing section at each end. The seals were effected by compressing the rubber gasket around the Teflon which compressed around the tube. At the cooling water inlet end, this seal also prevented leakage between the atmosphere and the condensing section. At the exit end the annulus was continued through another fitting which sealed the condensing section from the atmosphere. The annulus also insulated the cooling water from the fittings. The atmospheric seals also served as the tube supports.

Two sets of end flanges were prepared. Each set consisted of one 12-inch square by 1/2-inch thick brass plate and an identical aluminum plate. The brass plate was used so that soldering would be convenient if necessary. One set of end flanges was prepared to accommodate tubes 1/4-inch o.d. and smaller and the other set for tubes 1/4-inch o.d. to

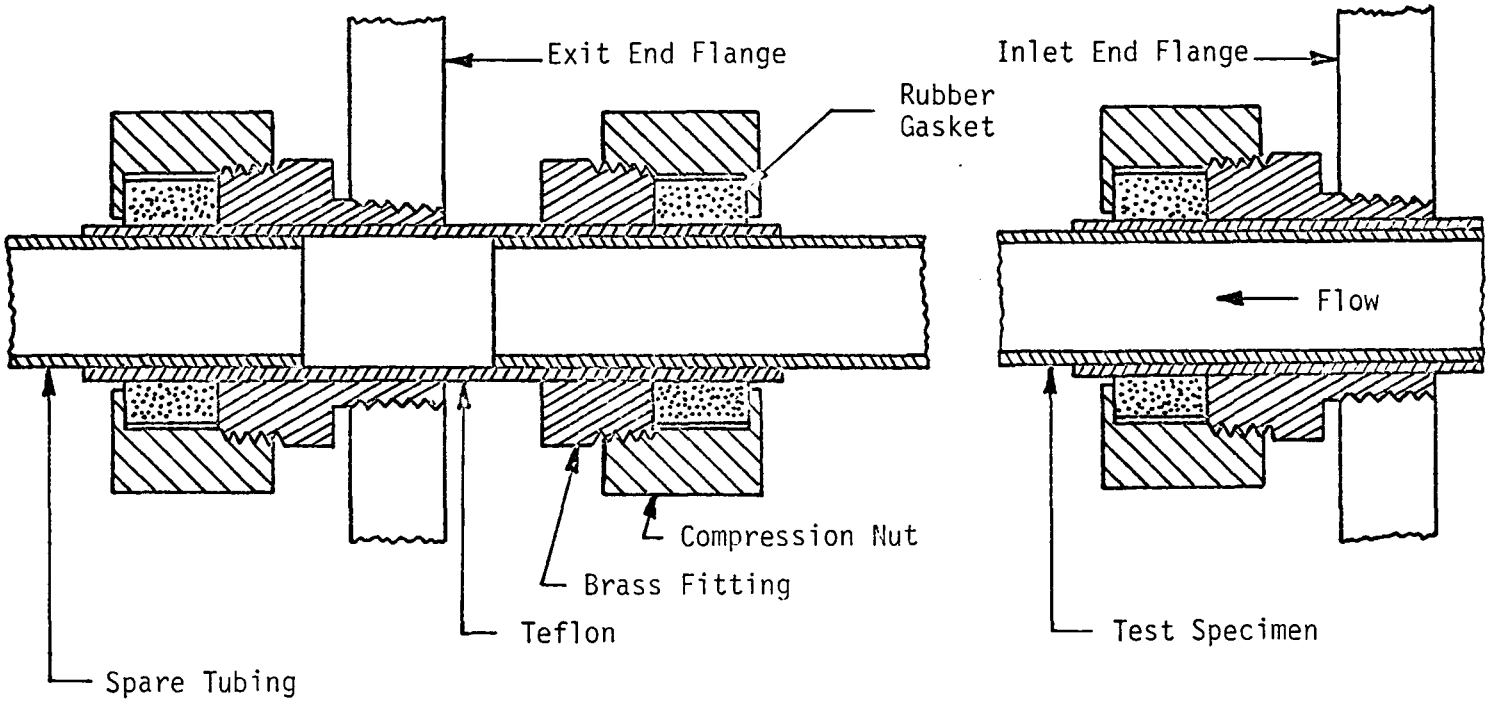

Fig. 2. Tube Insulation and Sealing Arrangement

7/8-inch o.d. Multiple installment positions were provided to allow installation of more than one tube at a time. The positions were arranged so that the condensate from one tube would not fall on another tube. One of each set of flanges was adapted for the installation of the thermocouple feedthrough. A short cylinder was cantilevered from each flange to hold a support tube for the surface temperature thermocouple leads.

A set of aluminum flanges was fabricated for the bottoms of the Pyrex tee sections. Connections for a vapor inlet line, a condensate removal line, and an evacuation line were provided in each flange. The vapor inlet line was as large as possible to minimize the vapor velocity within the condensing section. The vapor was diverted from direct impingement on the test specimens by a tee which also distributed the vapor along the section's length. The condensate removal line originated at the top surface of the flange. The condensate was returned by gravity to the vapor generator. The evacuation line originated just above the condensate level. A low elevation origin was chosen because the initial test was to be performed with steam with air being the noncondensible gas. Any air in the system would tend to collect at the bottom because of its higher density.

The tube surface thermocouple leads all left the condensing section at one end. The leads were supported in the condensing section by a 1/4-inch o.d. tube. The tube slipped over cantilevered supports in each end flange. Small wires were used to hold the leads in place.

The frame supported the Pyrex sections only by the end flanges. One end was fixed; the other end was movable to allow installation of

either one or two Pyrex sections. The frame also provided supports for the cooling water inlet and exit connections. Bolts were installed in the legs so that the condensing section could be leveled at its final location.

Test Specimens

The test specimens were commercially obtained tubes which were polished with steel wool prior to the electroplating of the thermocouples onto the surface. This polishing produced a smooth condensing surface. The tube surface temperature thermocouples had leads available for extension to the feedthrough. The specimens were designed to yield heat transfer rates which could be accurately measured.

Calculations were performed to determine minimum lengths for the specimens. Nusselt's solution was assumed to be valid for the calculation of the outside heat transfer coefficient for design purposes. The condensing vapor was assumed to be saturated steam at 212 F. An average cooling water bulk temperature of 70 F was assumed. A minimum change in cooling water bulk temperature of 10 F was desired. The internal Reynolds number was varied from 20,000 to 100,000. The cooling water flow rate, length of tubing, and the pressure drop across the test specimen were calculated.

Nine tubes were prepared for the experiment. Tube data are summarized in Table 1. Each tube was initially plated with a layer of nickel approximately 0.0005 in. thick. They were then plated with copper bands approximately 1/4-inch wide and 0.002 inch thick at

Outside Diameter (in.)	Inside Diameter (in.)	Length (in.)	Material	Number of Thermocouples
0.0625	0.03125	5	Brass	1
0.1250	0.09375	18	Brass	3
0.1875	0.15725	18	Brass	3
0.250	0.21875	18	Brass	3
0.250	0.21875	36	Brass	4
0.375	0.325	36	Copper	4
0.500	0.450	36	Copper	4
0.625	0.569	36	Copper	4
0.875	0.811	36	Copper	4

Table 1. Test Specimen Data

evenly spaced intervals along the tube. The number of copper bands depended upon the length of the tube.

A test specimen with the leads attached is shown in Fig. 3. Leads were attached to the copper bands by wrapping a 0.003 copper wire around the tube before the copper plating was initiated and allowing the plating process to effect the connection. Nickel leads, 0.005 inch diameter, were attached to the 1/4-inch o.d. by 36 inch and 3/8-inch o.d. tubes by solder. All of the other nickel leads were attached by plating a very thin copper base onto the nickel, wrapping a 0.005 inch diameter nickel lead around the tube, and then plating a layer of nickel 0.0007 inch thick over it. This was necessary because nickel will not plate over a nickel base. All leads which were broken after the original attachment were reattached by soldering.

Auxiliary Equipment

Equipment was necessary to remove noncondensible gases from the condensing section, to mix the cooling water so that bulk temperatures could be obtained, and to extend the surface thermocouple leads to the exterior of the condensing section.

An aspirator was used to remove the noncondensible gases from the system. It had a venturi type constriction to produce the suction and it used city water. The aspirator was connected directly to the condensing section. The maximum vacuum obtained using the aspirator was approximately 22 inches of mercury.

The inlet cooling water was assumed to be well mixed so no mixing chamber was provided. The temperature of the inlet water was measured

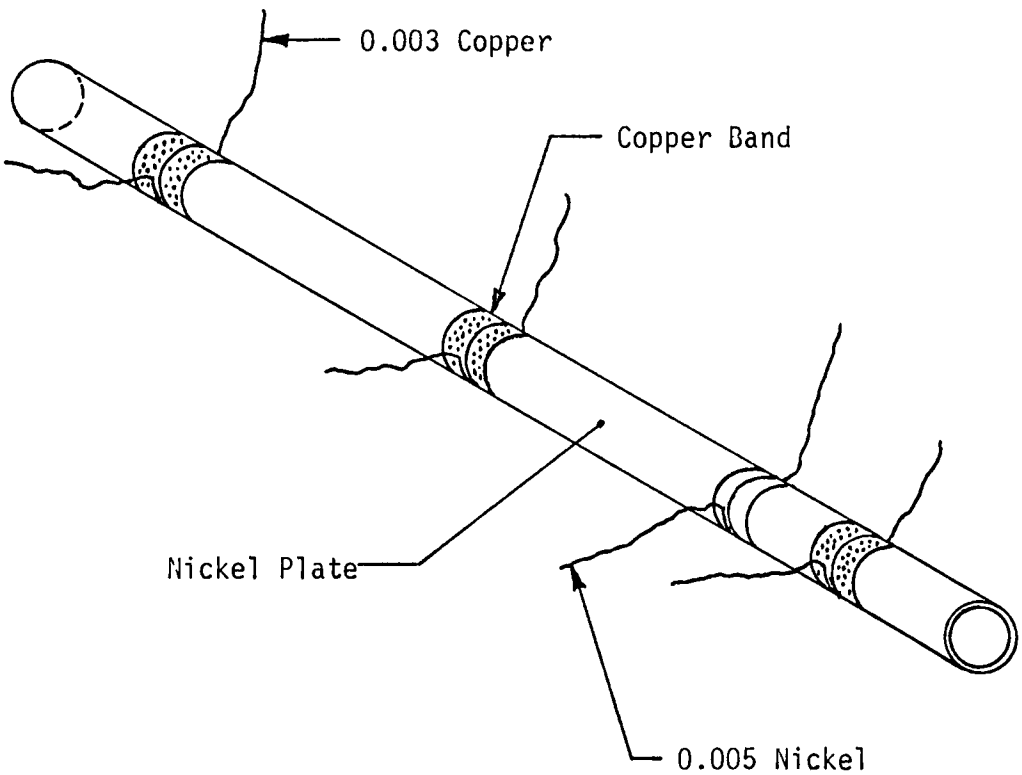

Fig. 3. Surface Thermocouples

just prior to entering the condensing tube. The outlet water temperature was measured after the water had passed through a mixing chamber. The chamber was made in accordance with the American Society of Heating, Refrigeration, and Air Conditioning Engineers Standards [27] and is shown in Fig. 4. The line connecting the tube outlet with the mixing chamber and the mixing chamber were both insulated to minimize heat loss prior to the temperature measurement.

The thermocouple feedthrough is shown in Fig. 5. It was located in the cooling water inlet end flange. The feedthrough wires were coated with silicone rubber, inserted through the aluminum piece, and then surrounded with another silicone rubber to prevent electrical short circuits.

Vapor Generator

The vapor generator supplied the pure, dry, saturated vapor to the condensing section. It was completely sealed from the atmosphere.

The generator is shown in Fig. 6. It was a four coil, single pass heat exchanger which used building steam as the heating source. The outer casing was an 18-inch section of standard 6-inch ips Pyrex pipe. The Pyrex allowed constant monitoring of the fluid level. The coils were made of 3/8-inch o.d. soft copper tubing. Building steam was passed through the coils and heated the surrounding fluid. The tubes were soldered in place. All of the internal parts were made from copper, aluminum, brass, or neoprene rubber. Two large vapor outlets were provided at the top of the generator to allow the vapor to flow with minimal pressure differential between the generator and the

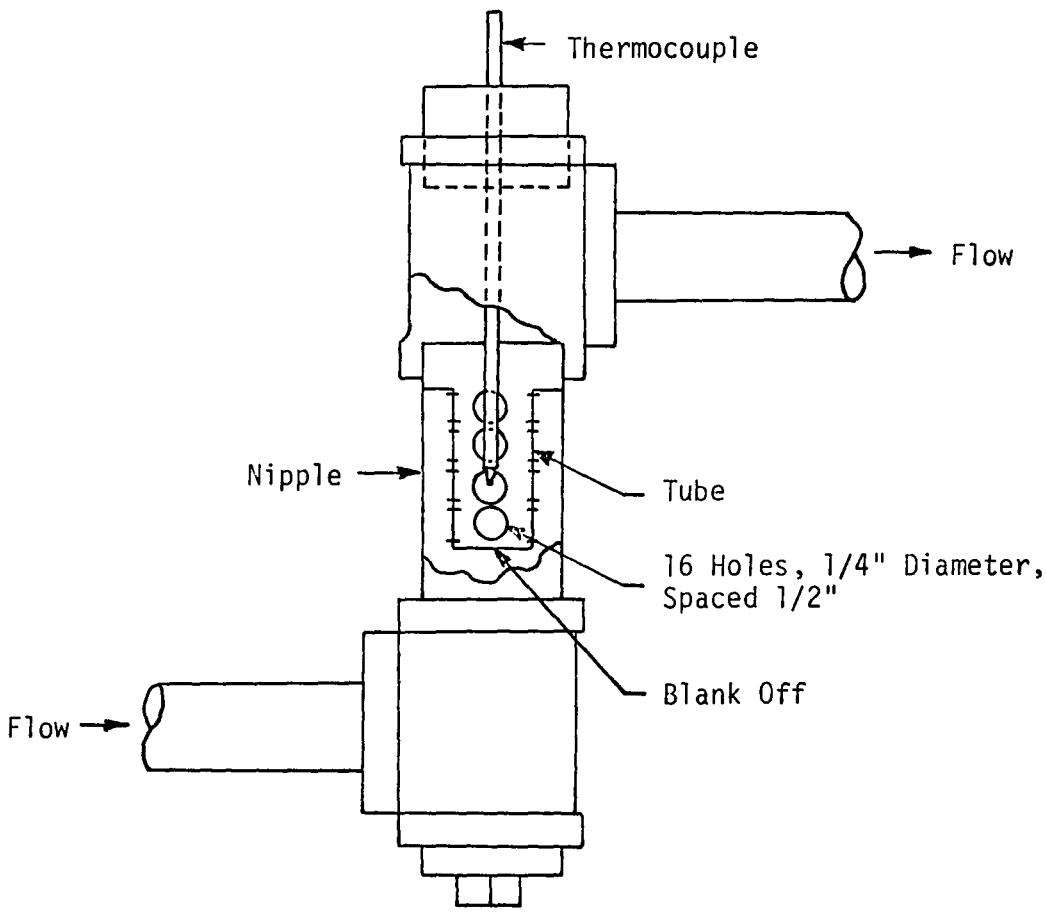

Fig. 4. Mixing Chamber

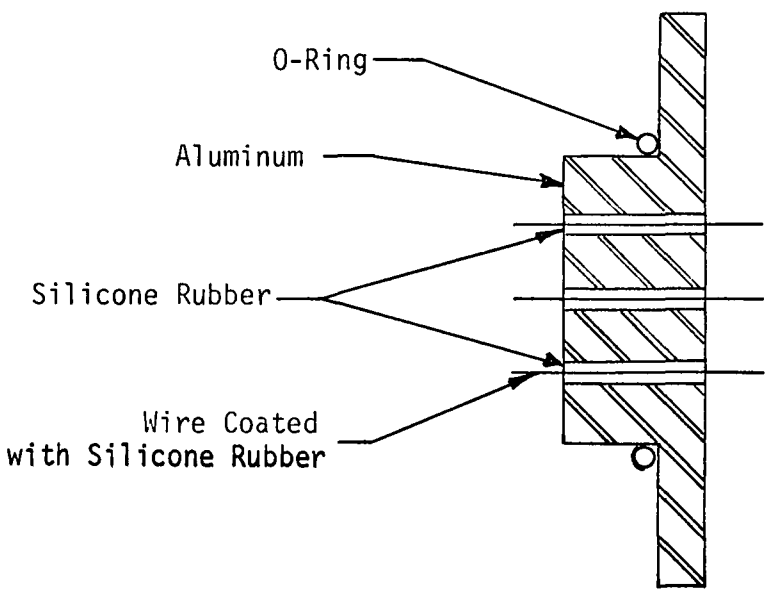

Fig. 5. Surface Thermocouple Lead Feedthrough

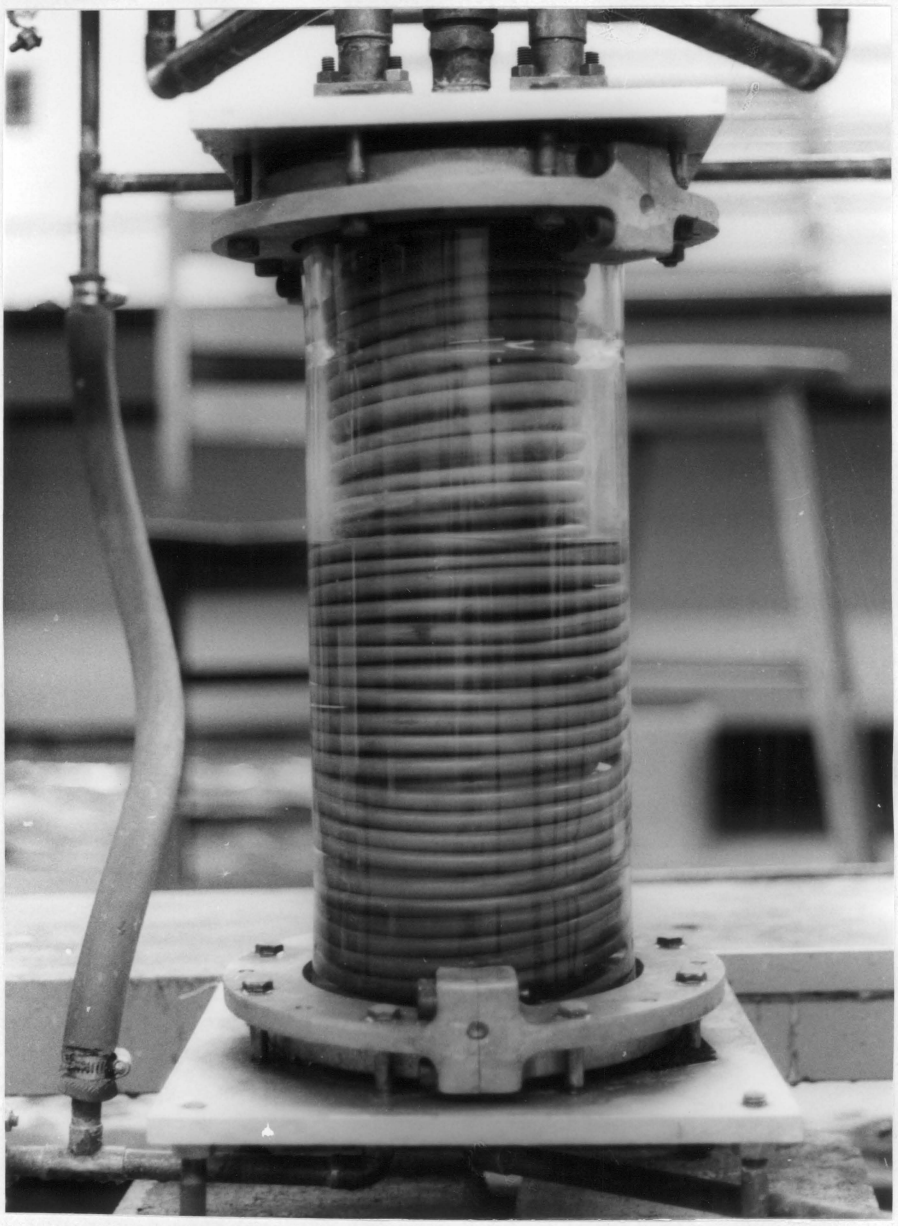

Fig. 6. Vapor Generator

condensing section. A common fill, drain, and condensate return line connected through the base. All of the inlets and outlets were capped when not in use. Neoprene rubber and rubber O-rings were used for all gaskets.

The building steam was throttled to between one and nine psig and desuperheated before entering the generator. Therefore, since the condensing substance was water in the initial testing, the pressure could not exceed the pressure in the building steam supply line. The Pyrex case had a pressure rating of 20 psig.

Instrumentation

Instrumentation were required to measure cooling water flow rate, the vapor temperature, the cooling water temperatures at the inlet and outlet, and the tube surface temperature. A U-tube manometer was used to monitor the condensing section pressure.

The cooling water flow rate was measured by full-view glass tube rotameters. Three rotameters having a combined range of 0.04 gpm to 21 gpm were installed next to the apparatus. Calibration data for the two largest rotameters are provided in Appendix III. The smallest rotameter was not calibrated because the fluctuations in city water pressure caused unsteady flow rates. The calibration data were obtained by measuring the time it took to collect a predetermined weight of water.

Vapor temperature was measured by a mercury-in-glass thermometer. The cooling water inlet and exit temperatures were measured with copper-constantan thermocouples. The thermocouples were calibrated

against a mercury-in-glass thermometer to within $\pm$ 0.1 C over a range of 23.6 C to 53.7 C. Calibration data are included in Appendix III. The reference junctions were in an ice bath for calibration and operation.

Tube surface temperatures were measured by the electroplated nickel-copper thermocouples on the tube. These thermocouples were calibrated against a mercury-in-glass thermometer with a range of 0 to 230 F divided into 2 F graduations. The scale could be read to within 0.5 F. The reference junctions for the calibrations were kept at 32 F. Calibration data are included in Appendix III. The reference junctions during operation were in condensing section vapor.

Two recording potentiometers were used to record thermocouple output. A Honeywell Electronic 15 Strip Chart Multipoint Recorder recorded the output of each thermocouple several times during each recorder cycle. The potentiometer had a range of 0 - 5 millivolts with the smallest graduation of the paper being 0.02 mv. Readings could be made to within 0.005 mv. A Honeywell continuous recording potentiometer was used to record the output of one of the tube thermocouples continuously. The range was -0.25 to +2.5 mv with the smallest division being 0.025 mv. Readings could be made to within 0.005 mv. The potentiometers were calibrated before each run against a Honeywell precision potentiometer.

A mercury filled U-tube manometer was installed to indicate condensing section pressure. It was useful during leak testing of the apparatus to indicate the strength of the vacuum and also indicated when and to what extent positive pressure was attained.

Discussion of Design Changes

Two major changes in the apparatus were made because of unforeseen problems. The vapor generator used in the experiment was a replacement for the original generator. The aspirator replaced a vacuum pump during the late stages of the project.

The original vapor generator was a commercially available electric steam generator. The outlet and drain were made from iron pipe which rusted when water was used. The rust contaminated the water making the purity of the vapor doubtful. This prompted the design and fabrication of a generator with no internal ferrous parts.

A vacuum pump was first used to remove noncondensible gases from the system. However, when difficulty in obtaining filmwise condensation was incurred, the possibility that vacuum pump oil vapor might have migrated from the pump back into the condensing section and caused the dropwise condensation mode prompted the abandonment of the pump. It was replaced with the aspirator which used tap water as the working substance.

All of the other parts of the system functioned as expected and presented no major difficulty. The test specimens were visible for their entire length. The surface thermocouple leads were easily broken but were not hard to replace. The thermocouples produced voltage on the order of 1 mv per 75 F.

V. OPERATING PROCEDURE

The test specimen was first polished with fine steel wool to remove obvious dirt and impurities. It was then dipped in phosphoric acid, rinsed with distilled water, and installed in its supports. The thermocouple leads were connected to the feedthrough and solder joints cleaned with methyl ethyl ketone. The tube was washed with a consumer detergent product, coated with phosphoric acid, and rinsed with distilled water.

The outer casing was assembled and put into place. Connections were made to the vapor generator and to the evacuation system. The system was then leak tested by filling the system with distilled water and removing the remaining air. Leaks were located by determining the origin of air bubbles and were fixed as required. The condensing section was then drained.

The aspirator was started. While the air was being removed, the cooling water inlet and exit connections were made. After a steady vacuum reading was obtained, the vapor generator was activated and allowed to pressurize the system to a slight positive pressure. The aspirator was then disconnected and the vapor was allowed to escape to the atmosphere. This venting was continued for approximately 30 minutes to further remove remaining air.

External heating of the condensing section Pyrex case was started. A variable voltage transformer was used to adjust the power input. A maximum input of approximately 3,300 BTU/hr (18 volts applied across 3 ohms) was required to clear enough of the Pyrex to permit observation

of the entire tube.

The vent was closed and the cooling water was started. The condensing rate was varied by adjusting the cooling water flow rate. The system was allowed to stabilize and readings were recorded. At each flow rate five parameters were measured: (1) flow rate (gpm), (2) steam temperature (F), (3) inlet thermocouple voltage (mv), (4) outlet thermocouple voltage (mv), and (5) tube thermocouple voltages (mv). The tube thermocouple voltages were actually multiple readings but were averaged after completion of the run. Observation of the tube was accomplished by removing two small sections of insulation on opposite sides of the condensing section. Sufficient light entered the section through one of the holes to permit observation of the tube through the other.

VI. RESULTS

Complete filmwise condensation was not attained on any clean tube. Consequently, heat transfer measurements were suspended and further efforts were directed toward obtaining filmwise condensation on selected tubes. The tubes used in this phase of the experiment were 1/4-inch o.d. by 36-inch long, 3/8-inch, 1/2-inch, and 7/8-inch o.d.

Filmwise condensation occurred, however, on surfaces which had become tarnished during the cleaning and initial leak testing of the apparatus. The system had been cleaned by filling it with weak solutions of phosphoric acid and acetic acid and tap water. The treatment resulted in part of the nickel surface becoming blackened. Filmwise condensation was observed on this surface for approximately two hours after which time it changed to the streaky mode. After the transition began, filmwise condensation could not be reestablished.

Some preliminary data were taken to evaluate the surface temperature measurement technique. It was noted that the surface temperature fluctuated approximately $\pm$ 10 F with time. The fluctuations probably resulted from variations in the film thickness which were caused by droplet formation and movement along the bottom of the tube. The width of the thermocouple was small enough that a drop could cover the entire junction. Increasing the thermocouple width would decrease the percentage of the thermocouple which would be affected. Steadier readings should result from a larger thermocouple. The thermocouple size could be increased by first plating the entire tube with copper and then covering the copper with large nickel bands leaving only

small areas of copper exposed. The procedure would yield a bright smooth surface which would not tarnish when exposed to the atmosphere. The preliminary data is not reported because all of the surface thermocouple leads were not intact at the time and the results were not repeatable. Also, at this time the single channel continuous recording potentiometer was not part of the instrumentation.

The inability to obtain filmwise condensation was probably due to small amounts of contaminants introduced into the system during the assembly process. This was unavoidable because soldering the thermocouple leads to the feedthrough required the use of flux which may not have been completely removed during the cleaning process. Contaminants from the atmosphere were probably also present since the installation and assembly time for one long tube was approximately two hours. The technique of freezing a film onto the tube before beginning the condensing process [23,26] was not tried in this experiment.

An evaluation of the noncondensible gas removal system was not made because a technique for detecting the presence of the gases was not available.

VII. CONCLUSIONS AND RECOMMENDATIONS

The overall performance of the apparatus was generally satisfactory with the exception of attaining filmwise condensation. The tube was visible for its entire length. A practical method of insulating the tube and cooling water from the apparatus while effecting a positive seal against the influx of air was found. The instrumentation permitted accurate determination of the necessary quantities. The fluctuations of the surface thermocouple outputs was an unexpected occurrence. This could be eliminated by increasing the thermocouple size as noted in the previous section.

A procedure for insuring consistent filmwise condensation needs to be established. The following procedures are recommended:

1. Clean the apparatus with a degreasing solution after assembly. Flush the system with distilled water before starting the experiment.
2. Treat the surface with an agent which promotes filmwise condensation. The agent should be adsorbed by the surface and not by the condensing vapor. It would have to withstand a temperature equal to the saturation temperature of the vapor before the start of the condensation process.
3. Freeze a film of condensed vapor onto the surface before starting the condensing process.

LITERATURE CITED

1. Hampson, H., "Heat Transfer During Condensation of Steam," Engineering, Vol. 172, 1951, pp. 221-223.

2. Boelter, L. M. K., V. H. Cherry, H. A. Johnson, and R. C. Martinelli, Heat Transfer Notes, McGraw-Hill Company, Inc., New York, 1965, pp. 563-567.

3. Bromley, L. A., "Effect of Heat Capacity of Condensate," Industrial and Engineering Chemistry, Vol. 44, No. 12, pp. 2966-2969.

4. Sparrow, E. M., and J. L. Gregg, "Laminar Condensation Heat Transfer on a Horizontal Cylinder," Journal of Heat Transfer, Vol. 81, No. 4, 1959, pp. 291-295.

5. Chen, M. M., "An Analytical Study of Laminar Film Condensation: Part 2 - Single and Multiple Horizontal Tubes," Journal of Heat Transfer, Vol. 83, No. 1, 1961, pp. 55-60.

6. Buznik, V. M., V. A. Aleksandrov, and G. F. Smirnov, "The Effect of Surface Tension Forces on the Heat Transfer Coefficient in the Condensation of Vapours on Inclined Tubes," International Chemical Engineering, Vol. 9, No. 2, April 1969, pp. 216-219.

7. Hassan, K. E., and M. Jakob, "Laminar Film Condensation of Pure Saturated Vapors on Inclined Circular Cylinders," Transactions of the ASME, Vol. 80, 1958, pp. 887-894.

8. Markowitz, A., B. B. Mikic, and A. E. Bergles, "Condensation on a Downward-facing Horizontal Rippled Surface," ASME paper 72-HT-33.

9. Ünsal, M., "Heat Transfer Measurements in Film Condensation of Steam on Small Diameter Horizontal Tubes," M.S. Thesis, Virginia Polytechnic Institute and State University, 1972.

10. Henderson, C. L., and J. M. Marchello, "Role of Surface Tension and Tube Diameter in Film Condensation on Horizontal Tubes," AIChE Journal, Vol. 13, No. 3, 1967, pp. 613-614.

11. Merte, H., Jr., "Condensation Heat Transfer," Advances in Heat Transfer, Vol. 9, 1973, pp. 181-272.

12. Minkowycz, W. J., and E. M. Sparrow, "Condensation Heat Transfer in the Presence of Noncondensibles, Interfacial Resistance, Superheating, Variable Properties, and Diffusion," International Journal of Heat and Mass Transfer, Vol. 9, No. 10, 1966, pp. 1125-1144.

13. Hawakawa, T., "Condensation of Steam in the Presence of Air," Heat Transfer - Japanese Research, Vol. 2, No. 9, 1973, pp. 41-45.

14. Kirkbride, C. G., "Heat Transmission by Condensing Pure and Mixed Substances on Horizontal Tubes," Industrial and Engineering Chemistry, Vol. 25, 1933, pp. 1324-1330.

15. McAdams, W. H., and T. H. Frost, "Heat Transfer," Industrial and Engineering Chemistry, Vol. 14, 1922, pp. 13-18.

16. Wallace, J. L., and A. W. Davidson, "Condensation of Mixed Vapors," Industrial and Engineering Chemistry, Vol. 30, 1938, pp. 948-953.

17. Hampson, H., "The Condensation of Steam on a Tube with Filmwise or Dropwise Condensation and in the Presence of a Non-condensible Gas," International Heat Transfer Conference, 1961, pp. 310-318.

18. Selin, G., "Heat Transfer by Condensing Pure Vapours Outside Inclined Tubes," International Heat Transfer Conference, 1961, pp. 279-289.

19. Henderson, C. L., "Heat Transfer During Vapor Condensation in the Presence of Noncondensible Gas," Ph.D. Dissertation, University of Maryland, 1967.

20. Magal, B. S., "Film Condensation of Saturated Steam on a Horizontal Tube," Indian Journal of Technology, Vol. 10, No. 10, 1972, pp. 370-376.

21. Baker, E. M., and A. C. Mueller, "Condensation of Vapors on a Horizontal Tube," AIChE Journal, Vol. 33, 1937, pp. 531-538.

22. Bromley, L. A., and R. S. Brodkey, "Effect of Temperature Variation Around a Horizontal Condenser Tube," Industrial and Engineering Chemistry, Vol. 44, 1952, pp. 2962-2966.

23. Peck, R. E., and W. A. Reddie, "Heat Transfer Coefficient for Vapours Condensing on Horizontal Tubes," Industrial and Engineering Chemistry, Vol. 43, 1951, pp. 2926-2931.

24. Rausher, J. W., A. F. Mills, and V. E. Denny, "Experimental Study of Film Condensation From Steam-Air Mixtures Flowing Downward Over a Horizontal Tube," Journal of Heat Transfer, Vol. 96, Series C, No. 1, pp. 83-88.

25. Wilson, E. E., "A Basis for Rational Design of Heat Transfer Apparatus," Transactions of the ASME, Vol. 37, 1915, pp. 47-82.

26. Hyman, S. C., and C. F. Bonilla, "Heat Transfer by Natural Convection From Horizontal Cylinders to Liquid Metals; Final Report for July 1, 1949 to June 30, 1950," U. S. AEC, NYO-560, June 1950, pp. 22-37.

27. American Society of Heating, Refrigeration, and Air Conditioning Engineers Standards 41-66, Part 1, 1967, p. 13.

28. Mills, A. F., and R. A. Seban, "The Condensation Coefficient of Water," International Journal of Heat and Mass Transfer, Vol. 10, No. 12, 1967, pp. 1815-1817.

29. Kays, W. M., Convective Heat and Mass Transfer, McGraw-Hill, Inc., New York, 1966, pp. 173-196.

Appendix I

WILSON'S METHOD FOR DETERMINING HEAT TRANSFER COEFFICIENTS

A graphical technique to determine the relationship between the internal heat transfer coefficient and the cooling water velocity was originally proposed by Wilson [25]. In his analysis, however, Wilson introduced a questionable assumption which affects the accuracy of the results. Ünsal [9] extended the technique (using the same assumption) to obtain the outside heat transfer coefficient. His results are, therefore, subject to the same limitation. Wilson's original procedure and Ünsal's extension are both described below.

Heat transfer data for a tube was collected and grouped according to the average bulk temperature of the cooling water. The Reynolds Number of the cooling water flow was sufficiently high to insure turbulent flow. The overall resistance to heat transfer, $R_{overall}$, was calculated and plotted as a function of $(1/V)^N$. Because ρ and μ depend only on the average bulk temperature, the grouping of the data made this, effectively, a graph of $R_{overall}$ as a function of $(1/Re)^N$. A linear relationship between $R_{overall}$ and $(1/Re)^N$ was desired. The value of N was determined by trial and error to be 0.82.

$R_{overall}$ is the sum of three components.

$$R_{overall} = R_i + R_t + R_o$$

Wilson assumed that R_o and R_t were independent of the cooling water velocity. This is the questionable assumption and will be discussed later. It was further assumed that

$$R_i = 0 \quad \text{at} \quad (1/V)^{.82} = 0 \ .$$

Therefore, a line parallel to $R_{overall}$ passing through the origin of the axes was, using these assumptions, a graph of R_i as a function of $(1/V)^{.82}$. The internal heat transfer coefficient can be determined by

$$h_i = \frac{1}{R_i A_i}$$

Ünsal's extension of Wilson's method was to extrapolate the $R_{overall}$ line to $(1/V)^{.82} = 0$. Since

$$R_i = 0 \quad \text{at} \quad (1/V)^{.82} = 0$$

$$R_o + R_t = R_{overall} \quad \text{at} \quad (1/V)^{.82} = 0$$

But if R_o and R_t are independent of $(1/V)^{.82}$ as assumed

$$R_o + R_t = \text{constant} = Z$$

The value of Z is equal to the value of the intercept. It is known that

$$R_t = \frac{\ln \frac{D_o}{D_i}}{2\pi k_m L}$$

which is independent of the cooling water velocity. Therefore, since

$$R_o = Z - R_t ,$$

R_o can be determined. The outside heat transfer coefficient can be calculated by

$$h_o = \frac{1}{A_o R_o} .$$

If Nusselt's solution for the value of the outside heat transfer coefficient is assumed to be valid, it can be shown that R_o is not a constant but depends on the internal Reynolds Number. This dependence exists because h_o depends on the surface temperature which, in turn, is dependent on the internal Reynolds Number.

Calculations were performed to determine the extent of this dependence. The internal heat transfer coefficient was calculated by the following equation [29].

$$h_i = 0.0155\ Pr^{0.5} Re^{0.83} (1 + \frac{6D_i}{L})\ (\frac{\mu_{w,i}}{\mu_B})^{0.14}$$

Nusselt's equation was used to calculate h_o. An iterative technique was used to determine the wall temperatures. The calculations were performed for the following set of conditions:

D_o = 0.250 in.

D_i = 0.21875 in.

L = 33 in.

k_m = 92 B/hr-ft-deg F

T_{B1} = 75 deg F

The vapor was assumed to be saturated steam at atmospheric pressure. The exit bulk temperature was allowed to vary with the Reynolds Number and was found by iteration.

Graphs of R_o, R_i, and $R_{overall}$ as functions of $(1/Re)^{.83}$ are included in Fig. 7. R_o is not a constant but increases by 60 percent over the entire range of Reynolds Numbers. $R_{overall}$ and R_i are not

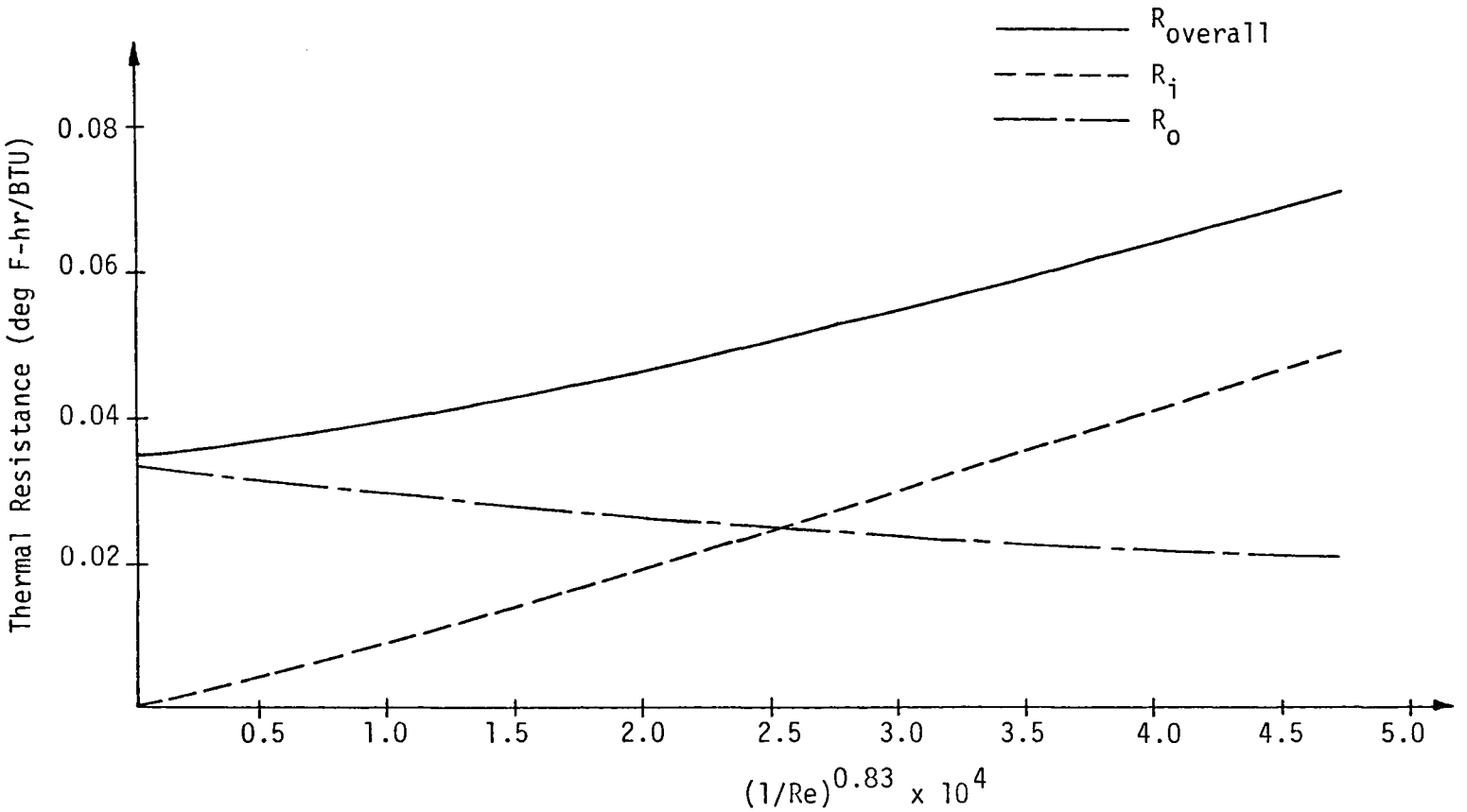

Fig. 7. Resistance to Heat Transfer vs. Internal Reynolds Number

linear functions of $(1/Re)^{.83}$. The exponent can be adjusted to yield a straight line for either curve but no exponent can make both curves become linear. Thus, the data obtained from a Wilson's plot do not correlate with the results obtained by applying other theories.

The computer program that was used to compute these results is listed on the following pages. The subroutines STEAMT and STMTHP were packaged subroutines which calculated steam properties and steam thermal properties, respectively, and are not listed.

```
      DIMENSION SXP(500),SYP(500),SYP2(500)
      REAL K,LENGTH,NU,MDOT
      HC(A,B,C,D,E,F,G,H) = 0.725*((32.2*A*(A-B)*(C**3)
     2*D/E/F/(G-H)*12.*3600.)**0.25)
      PI = 3.14159
      XMAX = 0.0
      YMAX = 0.0
      MDOT = 1.0
      N1 = 1
C READ IN TB1 = INLET BULK TEMPERATURE (DEG F)
C          TSTEAM = STEAM TEMPERATURE (DEG F)
C          OUTER AND INNER DIAMETERS (INCHES)
C          LENGTH (INCHES)
C         K = MATERIAL THERMAL CONDUCTIVITY (B/HR/FT/F)
      READ,TB1,TSTEAM,DIAOUT,DIAIN,LENGTH,K
C RTUBE = DEG F-HR/BTU
      RTUBE = ALOG(DIAOUT/DIAIN)/2./PI/LENGTH/K*12.
      CALL STEAMT (TSTEAM,PSAT,ROWF,ROWG,4)
      CALL STEAMT (ROWF,TSTEAM,HF,PRESS,1)
      CALL STEAMT (ROWG,TSTEAM,HG,PRESS,1)
      HFG = HG - HF
      WRITE (6,50) DIAOUT,TB1,TSTEAM
   50 FORMAT (5X,17HOUTER DIAMETER =,F7.4,/
     2         5X,6HTBU = ,F7.2,/
     3         5X,9HTSTEAM = ,F7.2,/)
      DO 108 I = 1,500
C  MDOT IS INCREMENTED BY 2 % EACH ITERATION
   34 MDOT = MDOT*1.02
      TB2 = TB1 + 1.
C BEGIN ITERATION OF TB2
  107 TBULK = (TB1 + TB2)/2.
      CALL STMTHP (TBULK,20.,VISCI,1)
```

```
      RE = MDOT * 4./VISCI/PI/DIAIN*12./60.
      TWI = TBULK + 5.
      TWO = TWI + 2.
      CALL STMTHP (TBULK,20.,PR,5)
      NU = 0.0155*(PR**0.5)*(RE**0.83)*(1.+6.*DIAIN/LENGTH)
      CALL STMTHP (TBULK,20.,CONDI,3)
      HI1 = NU*CONDI/DIAIN*12.
C BEGIN ITERATION FOR INSIDE AND OUTSIDE WALL TEMPERATURES
  103 TFILM =(TSTEAM + TWO) /2.
      CALL STEAMT (TFILM,PRESS,DENF,DEFG,4)
      CALL STMTHP (TFILM,15.,CONDO,3)
      CALL STMTHP (TFILM,15.,VISCO,1)
      CALL STMTHP (TWI,15.,VISCW,1)
C HI  = H INSIDE (BTU/HR/SQ FT/F)
      HI = HI1*((VISCW/VISCI)**0.14)
C RIN = INTERNAL RESISTANCE TO HEAT TRANSFER (DEG F-HR/BTU)
      RIN = 1./PI/DIAIN/LENGTH/HI*144.
C HO FROM NUSSELT EQUATION
C HO = OUTSIDE HEAT TRANSFER COEFFICIENT (BTU/HR/FT**2/F)
      HO = HC (DENF,ROWG,CONDO,HFG,VISCO,DIAOUT,TSTEAM,TWO)
C RO = OUTSIDE RESISTANCE TO HEAT TRANSFER (DEG F-HR/BTU)
      RO = 1./DIAOUT/PI/LENGTH/HO*144.
      RTOTAL = RIN + RTUBE + RO
      TWI1 = RIN/RTOTAL*(TSTEAM-TBULK) + TBULK
      TWO1 = TSTEAM - RO/RTOTAL*(TSTEAM-TBULK)
      IF (ABS(TWI1-TWI)-0.10) 101,102,102
  102 TWI = TWI1
      TWO = TWO1
      GO TO 103
C STOP ITERATION FOR INSIDE WALL TEMPERATURE
  101 IF (ABS(TWO1-TWO)-0.10) 104,102,102
C STOP ITERATION FOR OUTSIDE WALL TEMPERATURE
```

```
  104 TB21 = TSTEAM - (TSTEAM - TB1)/(EXP
     2(1./RTOTAL/MDOT/60.))
      IF (ABS(TB21-TB2)-0.010) 105,106,106
  106 TB2=TB21
      GO TO 107
C END OF ITERATION FOR TB2
C
C CHANGE STATEMENT 105 TO CHANGE EXPONENT FOR PLOTTING
C
  105 REIN = RE**0.83
      RREIN = 1./REIN
      UA = 1./RTOTAL
      IF (I - 1) 108,25,24
   25 IF (RE - 5000.) 30,30,31
   31 IF (RE - 15000.) 24,33,33
   30 MDOT = MDOT * 2.
      GO TO 34
   33 MDOT = MDOT / 2.
      GO TO 34
   24 IF (RE - 10000.) 109,2,2
    2 IF (RE-10.**7) 14,14,26
   14 IF (XMAX) 3,3,9
    3 CALL PLOT (0.,6.,3)
      CALL PLOT (0.,2.,2)
      CALL PLOT (7.,2.,2)
      CALL PLOT (7.,6.,2)
      CALL PLOT (0.,6.,2)
      CALL PLOT (0.,2.,-3)
      WRITE (6,23)
   23 FORMAT (//6X,3HTB2,6X,7HR TOTAL,7X,4HR IN,7X,5HR OUT,
     29X,3HTWI,9X,3HTWO,9X,2HRE,4X,9H(1/RE)**N,8X,4HMDOT,
     34X,9HITERATION)
```

```
      DO 5 J = 10,150,5
      IF (REIN-J*100. ) 6,6,5
    6 XMAX = 1./((J-5)*100.)
      GO TO 4
    5 CONTINUE
    4 DO 7 J = 1,100,5
      IF (RTOTAL - (J-1)*0.001) 8,8,7
    8 YMAX = (J-1)*0.001
      GO TO 10
    7 CONTINUE
   10 XPLOT = RREIN/XMAX*7.
      YPLOT = RTOTAL/YMAX*4.
      DO 11 I1=1,9
      CI = FLOAT(I1)
      XTIC = CI*7./10.
      CALL PLOT (XTIC,4.05,3)
      CALL PLOT (XTIC,3.95,2)
      CALL PLOT (XTIC,0.05,3)
      CALL PLOT (XTIC,-.05,2)
   11 CONTINUE
      DO 12 I1=1,4
      CI = FLOAT(I1)
      YTIC = CI*4./5.
      CALL PLOT (-0.05,YTIC,3)
      CALL PLOT (+0.05,YTIC,2)
      CALL PLOT (6.95, YTIC,3)
      CALL PLOT (7.05, YTIC,2)
   12 CONTINUE
      XPLOT = RREIN/XMAX*7.
      YPLOT = RTOTAL/YMAX*4.
      CALL PLOT (XPLOT,YPLOT,3)
      YPLOT2 = RIN*4./YMAX
```

```
    9 XPLOT = RREIN/XMAX*7.
      YPLOT = RTOTAL/YMAX*4.
      WRITE (6,18)TB2,RTOTAL,RIN,RO,TWI,TWO,RE,RREIN,MDOT,I
   18 FORMAT (3X,F7.2,3F12.6,2F12.2,F12.1,F11.8,F12.3,I9)
      CALL PLOT (XPLOT,YPLOT,2)
      SXP(I) = XPLOT
      SYP(I) = RIN/YMAX*4.
      SYP2(I) = RO/YMAX*4.
      GO TO 108
  109 N1 = I + 1
  108 CONTINUE
   26 N2 = I-1
      XPLOT = 0.0
C
C PLOT RI AS A DOTTED LINE
C
      DO 208 I = N1,N2
   99 IF (XPLOT) 98,98,97
   98 XPLOT = SXP (I)
      YPLOT = SYP(I)
      CALL PLOT (XPLOT,YPLOT,3)
      X1 = XPLOT
   97 XPLOT = SXP (I)
      YPLOT = SYP(I)
      IF ((X1 - XPLOT) - 0.25) 19,19,21
   19 CALL PLOT (XPLOT ,YPLOT ,2)
      GO TO 208
   21 IF ((X1 - XPLOT ) - 0.3) 20,22,22
   20 CALL PLOT (XPLOT,YPLOT,3)
      GO TO 208
   22 X1 = XPLOT
      CALL PLOT (XPLOT,YPLOT,3)
```

```
  208 CONTINUE
C END OF RI PLOT
      CALL PLOT(SXP(N1),SYP2(N1),3)
C
C PLOTS RO AS A SOLID LINE
C
      DO 300 I = N1,N2
      CALL PLOT(SXP(I),SYP2(I),2)
  300 CONTINUE
C END OF RO PLOT
   15 WRITE (6,13) XMAX,YMAX
   13 FORMAT (5X,8HX MAX = ,F12.7,/5X,8HY MAX = ,F12.7,/)
      CALL PLOT (0.,0.,-4)
      STOP
      END
```

Appendix II

TUBE INSULATION DATA

Tube o.d. (in.)	Teflon o.d. (in.)	Fitting o.d. (in.)
0.0625	0.1225	1/8
0.1250	0.1850	3/16
0.1875	0.2475	1/4
0.250	0.3100	5/16
0.375	0.500	1/2
0.500	0.675	5/8
0.675	0.8125	7/8
*0.875	0.810	7/8

* The 0.875 tube diameter was reduced to 0.750 before passing through a fitting.

Table 2. Tube Insulation and Fitting Data

Appendix III

CALIBRATION OF EQUIPMENT

EMF (mv)	Temperature (deg F)
0.928	74.5
1.415	95.5
1.518	100.2
1.872	115.2
2.196	128.7

Table 3. Cooling Water Thermocouple Calibration Data

EMF (mv)	Temperature (deg F)
2.405	224.5
2.200	210
1.780	180
1.610	167
1.080	128
0.930	116
0.040	71

Table 4. Surface Thermocouple Calibration Data

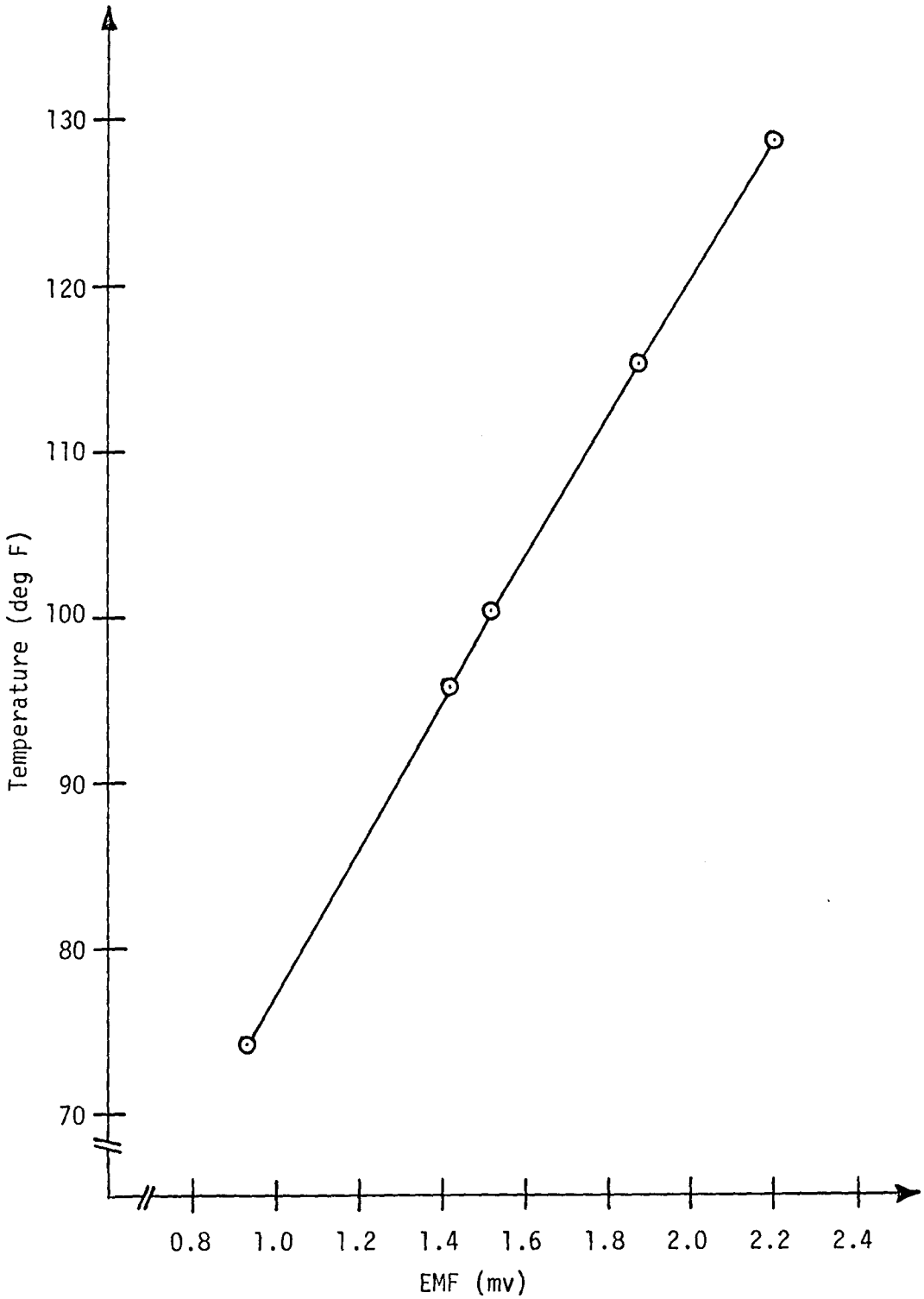

Fig. 8. Cooling Water Thermocouple Calibration Curve

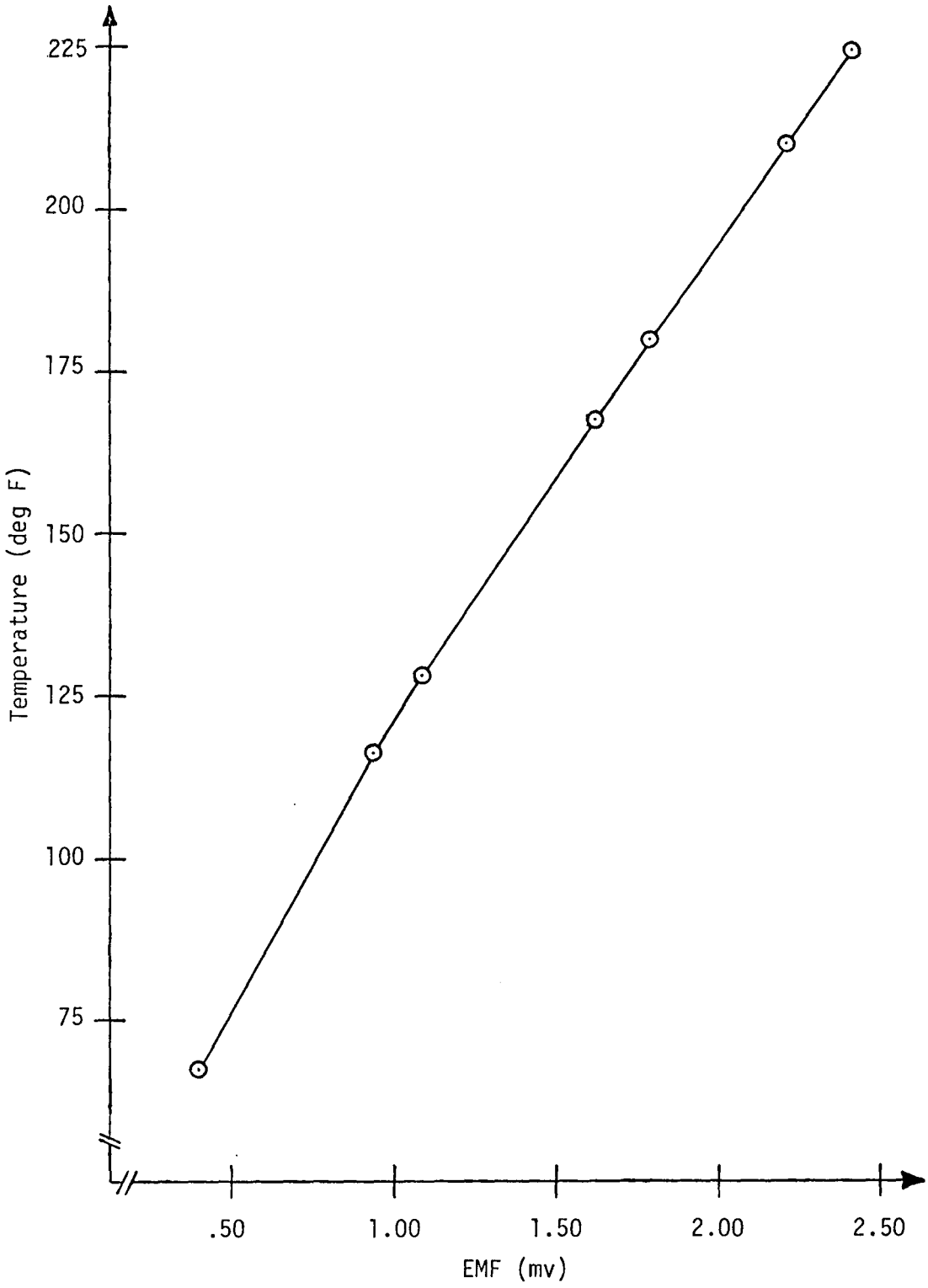

Fig. 9. Surface Thermocouple Calibration Curve

Reading (gpm)	Flow Rate (lb_m/min)
14.00	116.9
12.10	101.9
8.95	75.2
6.55	55.2
3.10	27.2

Table 5. Large Rotameter Calibration Data

Reading (gpm)	Flow Rate (lb_m/min)
2.50	20.6
2.00	16.4
1.50	12.3
0.90	7.3

Table 6. Intermediate Rotameter Calibration Data

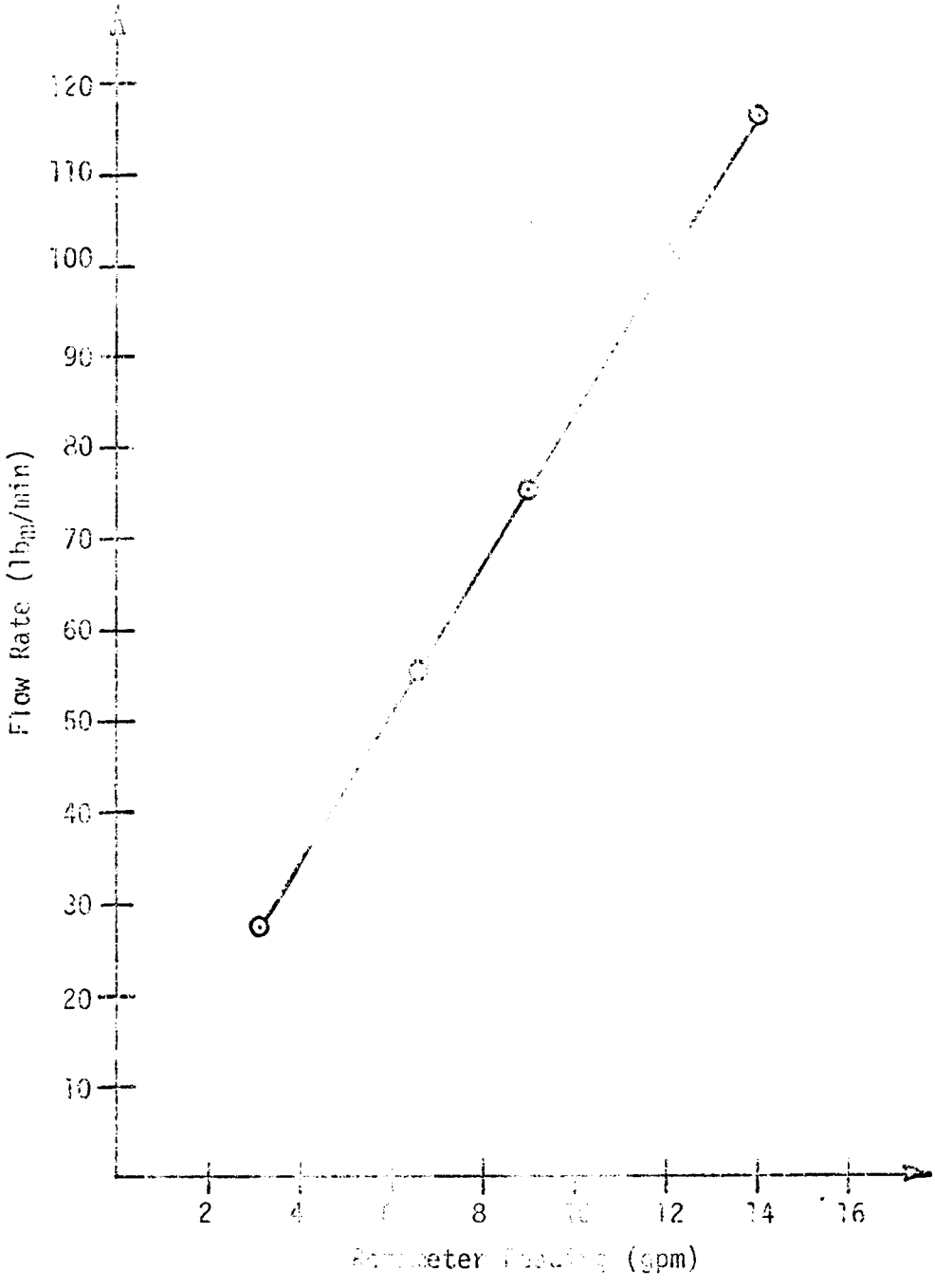

Fig. 10. Large Rotameter Calibration Curve

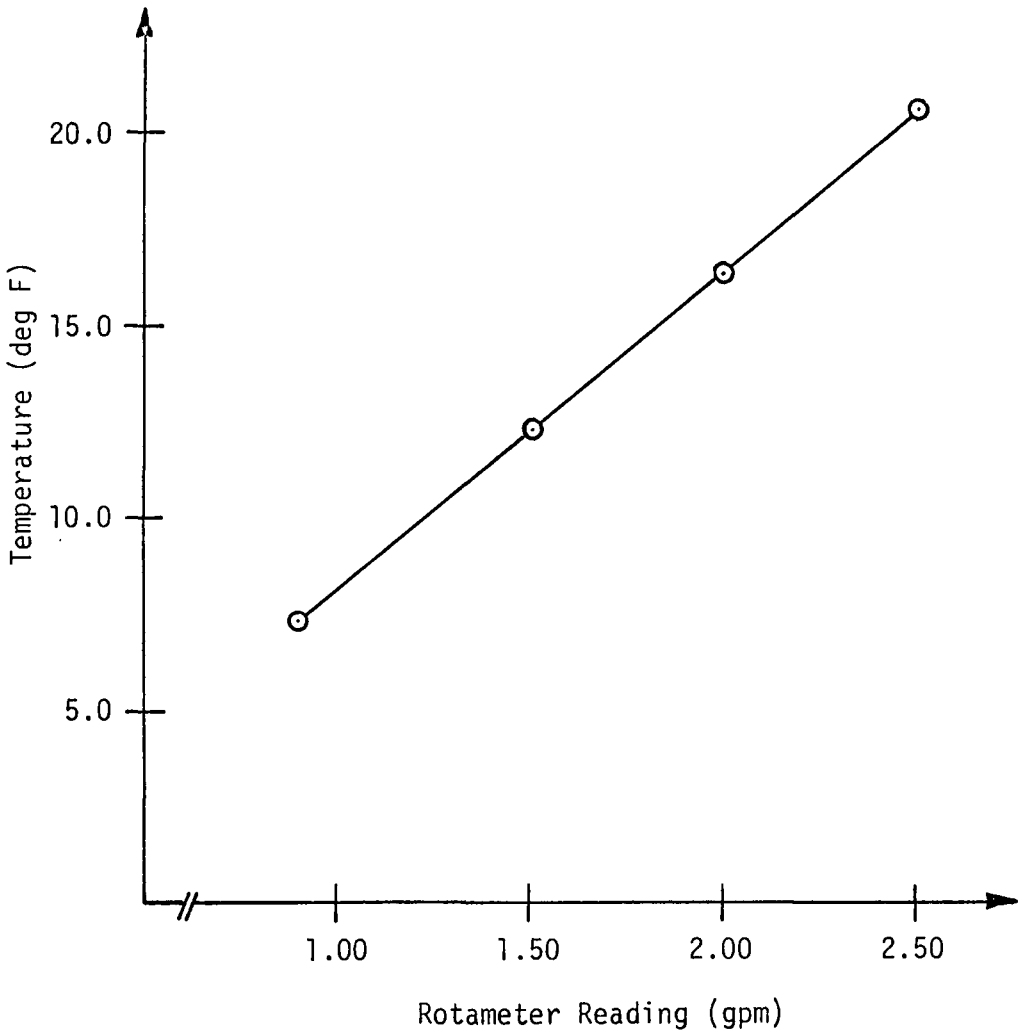

Fig. 11. Intermediate Rotameter Calibration Curve

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DESIGN OF AN APPARATUS TO EXPERIMENTALLY
VERIFY FILMWISE CONDENSATION THEORY
ON SMALL DIAMETER HORIZONTAL TUBES

by

Mark R. Schoonover

(ABSTRACT)

An apparatus to experimentally verify the accuracy of filmwise condensation heat transfer theories for small diameter horizontal tubes was developed. The apparatus was designed so that the condensation pattern was visible at all locations, all of the parameters necessary to calculate the average heat transfer coefficient were accurately determinable, and the removal of noncondensible gases was effected before the start of the condensation process. A method was devised to insulate the test specimen and cooling water from the rest of the apparatus to obtain an accurate measure of the heat transfer rate. The surface temperature of the tubes was directly measured by copper-nickel thermocouples which were electroplated onto the tube.

Initial testing of the apparatus was performed using steam as the condensing vapor. Difficulty in obtaining and maintaining filmwise condensation was caused by small amounts of impurities which were introduced during the assembly of the apparatus. The surface thermocouples produced output of approximately 1 mv per 75 F and were responsive to transient temperature fluctuations.