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SUMMARY

Local heat-transfer data were obtained for steam condensing in vertical downflow inside a tube. A 5/8-inch-inside-diameter by 8-foot-long stainless-steel water-cooled tube was used as the test condenser. The coolant flowed counter currently in the surrounding annulus. Complete condensing occurred in the test section. The downstream vapor-liquid interface was maintained inside the tube in all runs by throttling the condensate flow at the exit. Axial variations of the local condensing heat-transfer coefficient are presented. High values (some in excess of 6000 Btu/(hr)(sq ft)(^{O}F)) occurred at the vapor inlet decreasing with distance down the tube to values below 300 Btu per hour per square foot per ^{O}F at the downstream end of the condensing section. The local condensing heat-transfer coefficients were strongly dependent on the local vapor flow rate. The mean condensing heat-transfer coefficients for the entire condensing region varied from 2090 to 790 Btu per hour per square foot per ^{O}F and showed an approximate linear relation with the total mass velocity of the test fluid. Axial temperature distributions for the test fluid, condenser tube wall, and coolant are also presented. The measured axial temperature profiles of the vapor agreed closely with the local saturation-temperature profiles obtained from measured static pressures when the inlet vapor was near saturation conditions. However, temperature measurements made with the inlet vapor in a superheated state showed that the core of the vapor could remain superheated the entire length of the two-phase region, although condensation was occurring at the wall.

INTRODUCTION

Inside-tube-type vapor condensers are being considered for use in Rankine cycle spacecraft power-generation systems. The alkali-metal working-fluid vapors would be condensed within small diameter tubes, and the condensate would flow toward the discharge end of the condenser under the influence of shear and inertia forces.

In this type of condenser, the local value of the condensing heat-transfer coefficient varies with distance down the tube. In cases where the coolant and effective tube-wall coefficients are of the same order of magnitude as the condensing coefficient, the local heat flux will also vary with length down the tube. Since the rate of heat transfer is closely coupled to the hydrodynamic characteristics of the unit, an improved knowledge of the condensing heat-transfer coefficient is needed to predict the performance of an inside-tube condenser and to optimize the design.

Studies of inside-tube condensing have been conducted by a number of investigators (e.g., refs. 1 to 6). However, much of the data reported in these studies is for overall or mean values of the heat-transfer coefficient. Furthermore, where local values are reported, generally only overall test-section pressure drops are given so that it is difficult to determine the local vapor pressures. Because of the close interdependence between local fluid-flow conditions and local heat-transfer coefficients, it is important to make both local temperature and local pressure measurements. This becomes particularly important as the mass flow rate and, therefore, the local vapor velocities are increased. Hilding and Coogan were among the first to recognize this and a very limited amount of such data is presented in reference 1.

A program was therefore initiated at the Lewis Research Center to obtain additional data on inside-tube condensing. Steam was selected as the test fluid. Low-pressure (near 1 atm) steam, in addition to being a wetting fluid like the alkali metals, has a liquid to vapor density ratio very similar to the alkali metals at the temperatures contemplated for space powerplant operation. It is therefore likely to have similar hydrodynamic two-phase flow patterns. The insight obtained and the analytical models formulated from the steam data should be useful as a guide in analyzing the relatively small amount of high-cost alkali-metal-condensing data as it becomes available.

The principal objective of the investigation described in this report was to obtain local heat-transfer data for steam condensing inside a tube. In particular, local values of condensing heat-transfer coefficients and their variation with distance down the tube were desired. Special emphasis was placed on obtaining data while the downstream liquid-vapor interface was within the tube since this mode of operation would more nearly simulate an actual space-flight system.

The test condenser was a 5/8-inch-inside-diameter by 8-foot-long stainless-steel tube mounted vertically with vapor entering at the top and flowing downward. The condenser was cooled by water flowing upward (counter currently) in an annulus around the tube. The range of variables employed was as follows:

Test fluid flow rate, w_{tt}, lb/hr	34 to 117
Inlet vapor pressure, psia	16.2 to 55.8
Inlet vapor velocity, V_{vi}, ft/sec	65 to 232
Total condensing length, L_c, ft	5.9 to 7.7
Coolant flow rate, w_k, lb/hr	1085 to 7680
Coolant temperature, t_k, ^{o}F:	
Inlet	87 to 104
Exit	96 to 140.5

(All symbols are defined in appendix A.) Vapor inlet qualities were nominally 100 percent. A few special runs were made with the inlet vapor having superheat up to 17^{o} F.

APPARATUS AND PROCEDURE

Description of Rig

The test facility (fig. 1) consisted of two closed loops that use demineralized water as the working fluid in both loops. The equipment in the vapor loop consisted of a pot-type boiler, superheater, test section, flow-rate-measuring station, receiver, and boiler feed pump. The boiler was equipped with a wire mesh screen and baffle-type separator to remove liquid droplets. The coolant-loop components included a variable-speed pump, flowmeter, heat exchanger, and expansion tank. Building supply steam at 100 pounds per square inch gage was used as the heat source, and cooling-tower water was used as the final heat sink.

The single-tube test condenser was a coaxial-type shell and tube heat exchanger with vapor condensing inside the inner tube and cooled by water flowing in the annulus between the inner and outer tubes (fig. 2). The test section was mounted in the vertical position with vapor entering at the top and coolant flowing counter currently in the annulus. The inner tube was stainless steel with a 3/4-inch outside diameter, a 5/8-inch inside diameter, and a total length of 8 feet. The outer jacket was a 2-inch schedule 40 pipe that gave an annular gap width of 0.66 inch. A low-velocity plenum chamber and an adiabatic region were located upstream of the condensing section. The plenum chamber consisted of a cylinder $2\frac{3}{4}$ inches (i.d.) by 3 inches long with the entrance to the condenser tube being essentially sharp edge. A distance of $5\frac{11}{16}$ inches between the plenum chamber and condensing section comprised the adiabatic region and consisted of a dead-air space between the condenser tube and coolant exit chamber. The dead-air space was constructed by placing a $1\frac{1}{4}$-(o.d.) by 0.065-inch-wall tube around the main condenser tube and sealing both ends. The vapor plenum chamber was insulated from the coolant exit chamber by an additional dead-air space (fig. 2). The downstream end of the test section also had an insulating air space between the condenser tube and the coolant inlet chamber.

Mixing rings were located in the annulus between the condenser tube and outer jacket so that the measured temperature of the coolant would approximate the bulk temperature. The first ring was positioned 8 inches from the start of the condensing section and thereafter at 1-foot intervals and mounted to four 1/4-inch rods that ran the entire length of the test section. The test section (shown disassembled in fig. 3) and all vapor lines were lagged with blanket-type insulation to minimize heat losses. A throttle valve was installed downstream of the test section to assist in controlling the vapor condensing length by throttling the outflow of the condensate.

The receiver was a 40-gallon tank, partly full during a run, to act as a supply reservoir for the boiler feed pump. The boiler feed pump was automatically controlled to maintain a constant liquid level height in the evaporator for assurance of steady-state conditions.

Instrumentation

The test section was provided with instrumentation to measure vapor inlet temperature, vapor and condensate temperature inside the test section, condenser tube wall temperatures, coolant temperatures, vapor inlet pressure, static pressure inside the test section at different axial positions, and condensate and coolant flow rates. Figure 4 and tables I and II give the axial locations of the test-section thermocouples and test-fluid static-pressure taps.

Temperatures were measured with sheathed iron-constantan thermocouples. The vapor inlet temperature was measured in the low-velocity plenum chamber upstream of the condensing section. Vapor and condensate temperatures inside the test section were measured by inserting a temperature probe from the exit end of the condenser. The probe consisted of a 0.075-inch (o. d.) tube with the thermocouple wires inside the tube. The thermocouple junction was held in place at the tip of the probe by a bead of soft solder that also closed off the end of the tube. The probe could be moved to any desired axial position and was kept centered inside the condenser tube by three equally spaced vanes mounted 1 inch from the end.

Condenser-tube-wall temperatures were measured at 4-inch intervals in one longitudinal plane. The first thermocouple was positioned 2 inches from the start of the condensing section. The thermocouple wires were swaged inside a 0.040-inch (o. d.) stainless-steel tube insulated with magnesium oxide. The sheath material of the wall thermocouple was stripped back 1/8 inch from the measuring end, thus exposing the thermocouple junction. The bare junction and approximately 1 inch of the sheathed leads were placed in a 0.040-inch-deep longitudinal groove in the outer wall of the condenser tube. The assembly was then soldered into place, and the surface was ground smooth.

All wall thermocouple wires left the annular passage at the coolant exit plenum chamber and were tied to the 1/4-inch rods in the annular passage for support. Figure 5 illustrates the wall thermocouple installation.

Coolant temperatures were measured at equal intervals in one longitudinal plane in the annulus of the test section. The axial locations of the thermocouples are listed in table I. The coolant thermocouples, starting at 0. 51 foot, were positioned approximately 2 inches downstream of the mixing rings so that the temperature of the fluid was measured in a well-mixed region. The radial location of each coolant thermocouple was at the midpoint of the annular gap between the condenser tube and outer jacket. In addition, two temperature measurements were made at the coolant inlet plenum chamber and two were made at the coolant exit plenum chamber.

All pressures inside the test section were measured with mercury filled U-tube manometers with a hydrostatic water leg on the test-section side and atmospheric pressure on the reference side. A horizontal run of bare metal tubing of at least 6 inches was installed between the pressure tap in the test section and the vertical line to the manometer for assurance of an all-liquid hydrostatic head. Vapor inlet pressure was measured in the low-velocity plenum chamber upstream of the condensing section with a mercury filled U-tube manometer. Static-pressure taps were placed at the beginning, at the midpoint, and at the end of the condensing section.

Condensate flow rate was measured by using a modified weigh-tank technique that consisted of measuring the time required to fill a known volume. The temperature of the condensate was monitored so that the flow rate in mass units could be evaluated. The flow-rate-measuring station was located between the condensate throttle valve and main receiver tank (fig. 1) where the line sizes were large enough and the flow rate small enough that the liquid did not completely fill the cross section of the lines while flowing. The measuring station consisted of a quick shut-off valve positioned downstream of a section of 2-inch (o. d.) by 3-foot-long glass tube. This arrangement allowed visual observation and timing of the rise of the liquid interface when the quick shut-off valve was closed. It was assumed that the condensate flow rate did not change when the quick shut-off valve was closed since both the pressure drop across the condensate throttle valve and the pressure in the test section remained constant during the volumetric measurement. The coolant flow rate was measured continuously by a commercial turbine-type flowmeter.

Test Procedure

Before a data run was started, noncondensables were eliminated from the system as follows: The evaporator was filled with demineralized water and isolated from the remaining portion of the system that was evacuated to 28 or 29 inches of mercury through the vacuum line on the receiver. Ten to 15 percent (6 to 9 gal) of the original evaporator inventory was then boiled off to atmosphere to rid the water of dissolved and entrained gas. The air content of the water in the evaporator was reduced to 2 or 3 parts per million, on a mass basis, as determined by a gas analyzer. The evaporator was then opened to the system, and with the condensate throttle valve fully open, vapor flowed through the system to the receiver tank and out the vacuum line to purge any residual air pockets. The vacuum line was then closed and the receiver was allowed to fill partly with liquid. The pressure in the receiver rose to the saturation pressure corresponding to the temperature of the liquid. The receiver pressure ranged from 2 to 6 pounds per square inch absolute for the conditions of the tests. The test-section side of the manometer system was purged with water for assurance of an all-liquid hydrostatic head between the pressure tap and manometer fluid.

Establishment of test conditions was initiated when the receiver was approximately half full of liquid (20 gal). A coolant flow rate and evaporator pressure were set and the outflow of the condensate was throttled until the liquid-vapor interface was in the test section. The coolant inlet temperature was dependent on the temperature of the cooling-tower water, which was rather constant and served as the final heat sink.

The data were taken only after the system came to a steady-state operating condition; steady-state being defined as a constant axial position of the liquid-vapor interface, which was monitored with the temperature probe. The condensing lengths were arbitrarily chosen, since attempting to adjust to a given condensing length was very time consuming. The temperature of the vapor region of the test section, as well as the condensate subcooling portion, was measured with the movable axial probe. The presence of the probe did not appreciably affect the pressures and/or flow rate inside the test section since the condenser tube cross-sectional area was reduced by only 1.2 percent. The adiabatic temperature rise of the probe was neglected since it was estimated to be only 2.3° F at the highest inlet velocity obtained. Samples of liquid were withdrawn from the evaporator periodically to check the air content of the water. The air content never exceeded 3 parts per million.

The range of test variables for the investigation were as follows:

Test-fluid total flow rate, w_{tt}, lb/hr	34 to 117
Test-fluid total mass velocity, lb/(hr)(sq ft).	15 940 to 54 900
Coolant flow rate, w_k, lb/hr	1085 to 7680
Coolant mass velocity, lb/(hr)(sq ft)	57 600 to 408 000
Inlet vapor pressure, psia	16.2 to 55.8
Total condensing length, L_c, ft	5.9 to 7.7

DATA ANALYSIS

The data obtained in the tests are tabulated in tables I and II. The values of length are the distances from the start of the condensing portion of the test section with the positive direction measured downstream. In order to calculate the local condensing heat-transfer coefficients presented in table I, the local vapor saturation temperature, local inner-wall temperature, and local heat flux must be known.

The local vapor temperatures were evaluated by assuming thermodynamic equilibrium between the two phases and taking the saturated-vapor temperature corresponding to the local static pressure. The static pressures were determined from the manometer readings taking into account all hydrostatic corrections in the manometer system. The pressures downstream of the vapor-liquid interface were corrected for the liquid leg between the interface and the station at which the pressure was measured. Fluid properties were taken from reference 7.

The local inner-wall temperatures were computed from equation (B3) of appendix B by adding the temperature drop through the wall to a measured temperature of the outer wall of the condenser tube. The wall temperature drop was calculated by using the experimental value of the local heat flux and the tube-wall thermal conductivity. The measured outer-wall temperatures are tabulated in table II and show considerable scatter attributable mainly to the mixing rings. Therefore, a curve was faired through the wall-temperature data points, and local values of outer-wall temperatures were read from the curve at the axial stations of interest. The local faired values of outer-wall temperatures are included in table I. The inner-wall temperatures are not explicitly shown in table I but may be calculated by taking the difference between the local vapor-saturation temperature and temperature drop across the condensate film, Δt_f.

The local heat flux was computed by the method shown in appendix B. The heat flux is a function of the slope of the coolant-axial-temperature profile. The determination of the slope is a critical portion of the analysis of the data. The percent error that can be encountered is inversely proportional to the magnitude of the gradient. In the tests described herein, the slope of the coolant-temperature profile, especially in the vicinity of the vapor-liquid interface, was small, and consequently errors in heat-flux determination may be present. Errors in determining heat flux would be reflected in erroneous values of condensing heat-transfer coefficients. No estimate is given for the magnitude of the errors since no independent method for determining local heat flux was available. All runs, except run 10, gave conservative (low) values for the local heat fluxes and, consequently, local condensing heat-transfer coefficients. This was determined by plotting the values of local heat flux against length and graphically integrating the area under the curve to find the heat-rejection rate in the condensing portion of the test section. The heat-rejection rates thus computed ranged from 3 to 18 percent lower than that evaluated by taking the product of the measured condensate flow rate, inlet vapor quality, and latent heat of vaporization. Run 10 gave an integrated heat-rejection rate that was 5 percent higher.

For evaluation of overall heat-balance errors on the test condenser, the heat rejected by the test fluid was computed by taking into account vapor inlet quality, and also the subcooling of the condensate. To determine the overall temperature rise of the coolant, the average of the two thermocouple readings at the coolant inlet and also at the coolant exit were used. The heat gained by the coolant was then used as the reference to compute the percent errors presented herein. The total heat rejected from the test-fluid side ranged from 11 percent greater to 3.4 percent less than that gained by the coolant. The heat-balance errors are attributed partly to end losses.

The amount of superheat at the inlet vapor plenum chamber was determined by comparing the measured vapor temperature at this point with the saturation temperature corresponding to the measured pressure. The data that indicated superheat at the test-section inlet were used to develop an empirical correlation that gave the heat losses from the vapor line upstream of the test section as a function of test-fluid total flow rate and the average temperature between the evaporator and test section. Since no pressure drop was measured between the evaporator and vapor inlet plenum chamber, the drop in temperature was caused solely by heat losses. This correlation was used to determine the heat loss for the runs with no indicated superheat and to compute the vapor quality at the inlet plenum chamber. Qualities thus computed were never less than 0.94. It was assumed that the quality of the vapor leaving the evaporator was 100 percent.

Condensing lengths were determined from the test-fluid temperature profiles. The point where the slope of the test-fluid temperature profile changed abruptly was taken as the position of the downstream vapor-liquid interface.

RESULTS AND DISCUSSION

Temperature Profiles

Examples of axial temperature profiles are shown in figure 6. Plotted against length are measured vapor and condensate temperature, vapor saturation temperature, computed inner-wall temperature, measured outer-wall temperature, and coolant temperature. Because of the low vapor flow rate, no static pressure drop was encountered in the condensing portion of the test section, and consequently there was no change in saturation temperature. Since pressure drop from the vapor plenum to the condensing section was less than 0.5 pound per square inch, and if it is assumed that there were no heat losses in the zone between the plenum chamber and the condenser, the vapor state at the beginning of the condensing section should be the same as that at the plenum chamber. Figures 6(a) and (b) show the close agreement between measured vapor temperature and the saturation temperature obtained from the measured static pressure. This is typical of all runs that indicated near-saturation conditions at the test section inlet. Figure 6(c) compares measured vapor temperature with saturation temperature for a run that had 17.0° F superheat at the vapor inlet plenum chamber. The data indicate that the core of the test fluid is superheated the entire length of the condensing section, and yet condensation is occurring at the tube wall as indicated by the gradient of the coolant-temperature profile and the presence of the liquid-vapor interface in the test section.

In general, the vapor temperature upstream of the interface was quite constant with time, although run 14 gave a variation of $\pm 3^{\circ}$ F at several axial positions. The temperature of the condensate in the region of the interface was not constant with time. As much as $\pm 20^{\circ}$ F variation was measured in run 7. Presumably this was caused by alternate slugs of vapor and liquid coming into contact with the thermocouple probe or by axial fluctuations of the vapor-liquid interface. The condensate temperature downstream of the interface falls off quite rapidly with distance in the subcooling portion of the condenser. The condensate temperature fluctuations die out within a 6-inch distance downstream of the interface.

The slope of the coolant-temperature profile in the subcooling portion of the condenser and in the region of the interface was very small. This is a result of the low rate of heat transfer in this area caused by a low value of the overall heat-transfer coefficient and a decrease in the overall temperature difference.

Local Heat Transfer

Examples of the axial heat-flux distributions obtained in the tests are shown in figure 7. The local fluxes presented are based on the inner surface area of the test condenser. The heat-flux-distribution curves indicate a trend toward steeper slopes as the coolant flow rate is increased. Figure 7(a) shows that increasing the coolant flow rate at constant test-fluid vapor pressure and condensing length required an increased flow rate into the condenser. In general, high values of local heat flux existed at the vapor inlet end of the condenser and decreased in magnitude with increasing length. A uniform heat-flux distribution was approached as the coolant flow rate was reduced to the lower portion of the range (approx 1000 lb/hr). The variation in heat flux with length is caused partly by the variation in condensing heat-transfer coefficient.

Figure 8 shows typical plots of local condensing heat-transfer coefficients against length. The coefficients start at high values near the vapor-inlet end of the condenser and decrease in magnitude as the end of the condensing section is approached. At the vapor-inlet end of the condenser, the local condensing coefficient ranged from 15 850 (run 8) to 1460 Btu per hour per square foot per $^{\circ}F$ (run 5). At the downstream end of the condensing section, the local coefficients were generally below 300 Btu per hour per square foot per $^{\circ}F$.

The local condensing heat-transfer data are shown on the traditional Nusselt plot in figure 9. The theoretical Nusselt curve is given for reference. The ordinate of the figure is a function of the local condensing heat-transfer coefficient and a grouping of liquid condensate property values. The abscissa is the local condensate Reynolds number (the detailed relations are given in appendix B). Each set of symbols represents data from one run and therefore gives the variation in the condensing parameter with length down the condenser. The higher values of the condensing parameter occur at the vapor-inlet end of the condenser. The decrease in the condensing parameter is caused primarily by the decrease in the heat-transfer coefficient since the liquid property group, being a function of temperature, does not change appreciably with length.

The theoretical curve of Nusselt (ref. 8) shown in figure 9 was developed without considering the effect of interfacial shear between the vapor and liquid condensate. Later studies (refs. 9 and 10) have indicated that the presence of interfacial shear would increase the local heat-transfer coefficient. The data of figure 9 show a strong trend of increasing heat-transfer coefficient with increasing values of test-fluid total flow rate for a given value of condensate Reynolds number. The higher values of total flow rate give higher values of vapor flow rate at the same condensate Reynolds number. The dependence of condensing heat transfer on vapor flow rate is shown more directly in figure 10, where the local condensing heat-transfer coefficient is plotted as a function of the local vapor flow rate. The data fall within a relatively narrow band and the heat-transfer coefficient shows a nearly linear dependence on vapor flow rate. Figure 10 demonstrates the strong influence of the convective process on condensing heat transfer. These results are similar to those obtained by Akers and Rosson (ref. 4) for methanol and Freon-12 condensing in a horizontal tube.

The local condensing, coolant, wall, and overall heat-transfer coefficients are compared in figure 11 for two runs. Figure 11(a) shows data from run 2, where the coolant flow rate is at the high end of the range and indicates considerable variation in the coolant-side heat-transfer coefficient. Figure 11(b), on the other hand, shows data from run 5, where the coolant flow rate was at the low end of the flow range and indicates much less variation in coolant-side heat-transfer coefficient. The local coolant coefficients were higher than predicted by the equations for fully developed flow partly because of the mixing rings in the flow channel. The local coolant-side heat-transfer coefficients, as determined from the data, ranged from 2500 to 400 Btu per hour per square foot per $^{\circ}F$ at the vapor-inlet end of the test section and decreased in magnitude with increasing length. The effective wall-heat-transfer coefficient was constant at a value of 1650 Btu per hour per square foot per $^{\circ}F$, since it was assumed that the thermal conductivity of the wall material did not change over the temperature range encountered. The local overall heat-transfer coefficient, based on the outer diameter of the condenser tube, ranged from 910 to 257 Btu per hour per square foot per $^{\circ}F$ at the vapor-inlet end of the condenser and decreased with length.

Mean Heat-Transfer Coefficients

The mean condensing heat-transfer coefficients presented in the conventional form of the condensing parameter plotted against condensate Reynolds number are shown in figure 12. The property values were evaluated at the mean film temperature at the downstream vapor-liquid interface. The condensate Reynolds number is also evaluated at the end of the condensing length. The theoretical curves of Nusselt (ref. 8) and Colburn (ref. 11) (zero vapor shear) are shown for reference. Also shown in the figure are total condensing data of reference 2. As shown by figure 12, the experimental condensing parameter based on the mean condensing coefficient for flowing vapors is higher than would be predicted from the zero-vapor-shear curves.

The mean condensing heat-transfer coefficients from the data of the present work and the data of reference 2 are compared in figure 13 on the basis of total mass velocity. The general trend is shown to be an approximate direct proportionality between the mean condensing heat-transfer coefficient and the total mass velocity, with the data from this work falling slightly lower than that of reference 2.

SUMMARY OF RESULTS

The results of the investigation for local heat transfer for condensation of steam in vertical downflow within a 5/8-inch inside-diameter tube may be summarized as follows:

1. The local condensing heat-transfer coefficient varied with distance down the tube; high values occurred at the vapor-inlet end of the condenser and progressively decreased in magnitude with increasing distance as the quantity of condensate increased. The local condensing coefficient ranged from values as high as 15 850 Btu per hour per square foot per $^{\circ}F$ at the vapor-inlet end to values below 300 Btu per hour per square foot per $^{\circ}F$ near the downstream end of the condensing section.

2. The local condensing heat-transfer coefficients increased markedly with increasing values of local vapor flow rate.

3. The mean condensing heat-transfer coefficient showed an approximate linear relation with the total mass velocity of the test fluid. The mean coefficient varied from 2090

to 790 Btu per hour per square foot per $^{\circ}F$ over a test-fluid total mass velocity range from 54 900 to 15 940 pounds per hour per square foot.

4. Measured temperature profiles of the vapor agreed closely with the local saturation temperatures obtained from the static pressures when the vapor was near saturation conditions at the condenser inlet. Temperature measurements made with the inlet vapor in a superheated state showed that the core of the vapor could remain superheated the entire length of the two-phase region although condensation was occurring at the wall.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November, 30, 1965.

APPENDIX A

SYMBOLS

c_{pk}	specific heat of coolant, Btu/(lb)(^{o}F)
D	diameter, ft
D_i	inside diameter, ft
D_o	outside diameter, ft
G_k	coolant mass velocity, lb/(hr)(sq ft)
G_{tt}	test-fluid total mass velocity, lb/(hr)(sq ft)
g_c	conversion factor, 4.17×10^{8} (lb mass)(ft)/(hr^2)(lb force)
$h_{c\ell}$	local condensing heat-transfer coefficient, Btu/(hr)(sq ft)(^{o}F)
h_{cm}	mean condensing heat-transfer coefficient, Btu/(hr)(sq ft)(^{o}F)
h_k	coolant-side heat-transfer coefficient, Btu/(hr)(sq ft)(^{o}F)
h_w	effective tube-wall heat-transfer coefficient, Btu/(hr)(sq ft)(^{o}F)
k_f	thermal conductivity of condensate film evaluated at t_f, (Btu)(ft)/(hr)(sq ft)(^{o}F)
k_{mw}	mean coefficient of thermal conductivity of tube wall, (Btu)(ft)/(hr)(sq ft)(^{o}F)
L	length, ft
L_c	total condensing length, ft
Pr	Prandtl number ($c_p\mu/k$) of liquid condensate
P_s	test-fluid static pressure, psia
Q	rate of heat flow, Btu/hr
q	heat flux, Btu/(hr)(sq ft)
q_i	heat flux based on inside diameter, Btu/(hr)(sq ft)
t_f	mean temperature of condensate film, $t_{vs} - 3/4\,\Delta t_f$, ^{o}F
Δt_f	temperature difference between saturated vapor and inner wall, $t_{vs} - t_{iw}$, ^{o}F
t_{iw}	inner-wall temperature, ^{o}F
t_k	coolant temperature, ^{o}F
t_{ow}	outer-wall temperature, ^{o}F
t_p	test-fluid temperature as measured by probe, ^{o}F
ΔT_{sup}	amount of superheat at vapor inlet plenum chamber, ^{o}F
t_{vp}	vapor temperature at vapor inlet plenum chamber, ^{o}F
t_{vs}	saturated vapor temperature, ^{o}F
t_w	faired value of wall temperature from experimental data, ^{o}F
V_{vi}	vapor velocity at start of condensing portion of test condenser, ft/sec
$w_{c\ell}$	local condensate flow rate, lb/hr
w_{ct}	total condensate flow rate, lb/hr

w_k coolant flow rate, lb/hr

w_{tt} test-fluid total flow rate, lb/hr

$w_{v\ell}$ local vapor flow rate, lb/hr

x vapor quality at vapor inlet plenum chamber

$\Gamma_{c\ell}$ local rate of flow of condensate per unit periphery $w_{c\ell}/\pi D_i$, lb/(hr)(ft)

Γ_{ct} total rate of flow of condensate per unit periphery $w_{ct}/\pi D_i$, lb/(hr)(ft)

μ_f absolute viscosity of condensate film evaluated at t_f, lb/(ft)(hr)

ρ_f density of condensate film evaluated at t_f, lb/cu ft

APPENDIX B

DATA REDUCTION AND COMPUTATIONS

For evaluation of the local heat flux, the following assumptions were made:

(1) No heat loss from the coolant to ambient surroundings

(2) Radial flow of heat only

(3) The measured change in the temperature of the coolant was the change in the bulk temperature of the coolant

The local heat flux was obtained by the relation

$$q = \frac{w_k c_{pk}\, dt_k}{\pi D\, dL} \tag{B1}$$

where the numerator represents the increase in the enthalpy of the coolant and the denominator the area normal to the flow of heat. The slope of the coolant-temperature profile, dt_k/dL, was determined graphically from a plot of coolant temperature as a function of length. The flux at the inner wall was obtained by using the inner diameter of the condenser tube in equation (B1).

For determination of the local condensing heat-transfer coefficient the difference between the vapor temperature and inner-wall temperature was needed. Therefore, the inner-wall temperature was calculated from the following relation from reference 12 (p. 13) for the radial flow of heat in a cylinder

$$Q = \frac{2\pi L k_{mw}(t_{iw} - t_{ow})}{\ln \dfrac{D_o}{D_i}} \tag{B2}$$

which may be solved for inner-wall temperature in terms of the measured wall temperatures and the local heat flux

$$t_{iw} = t_{ow} + \frac{q_i D_i \ln \dfrac{D_o}{D_i}}{2k_{mw}} \tag{B3}$$

The thermal conductivity of the condenser tube wall k_{mw} was assumed constant over the temperature range encountered in the tests, and a value of 9.4 Btu per hour per foot per $^{\circ}F$ for stainless-steel 304 was used. The diameter ratio in equation (B3) is the ratio of the diameter of the actual location of the temperature being measured to the inner diameter of the tube. Since the wall thermocouple was placed in a 0.040-inch-deep groove milled into the outer surface of the condenser tube, the diameter at which the measured wall temperature was located was taken as the diameter to the midpoint of the groove. Equation (B3) reduces to the following relation after substituting the known values of the diameters, thermal conductivity, and faired value of wall temperature t_w:

$$t_{iw} = t_w + 3.52\times10^{-4}\, q_i$$

The local condensing heat-transfer coefficient was evaluated by

$$h_{c\ell} = \frac{q_i}{t_{vs} - t_{iw}}$$

where t_{vs} is the saturation temperature of the vapor corresponding to the pressure. The local values of condensing coefficient were plotted as a function of length and a mean coefficient was obtained by integrating to obtain the area under the curve and dividing by the condensing length.

For the purpose of comparing the data obtained herein with the laminar theory and the work of other investigators, a mean condensate film temperature is needed for the evaluation of film properties to compute a condensing parameter and a condensate Reynolds number. The mean film temperature is obtained from

$$t_f = t_{vs} - \frac{3}{4}\Delta t_f$$

as suggested by reference 12 (p. 330). The local condensing parameter is defined as

$$h_{c\ell}\left(\frac{\mu_f^2}{g_c \rho_f^2 k_f^3}\right)^{1/3}$$

where $h_{c\ell}$ is the local condensing heat-transfer coefficient and the quantities inside the parenthesis are the local values of the film properties. The local condensate Reynolds number is defined as

$$\frac{4\Gamma_{c\ell}}{\mu_f}$$

where $\Gamma_{c\ell}$ is the local condensate flow rate divided by the inner circumference of the condenser tube; thus,

$$\Gamma_{c\ell} = \frac{w_{c\ell}}{\pi D_i}$$

Local quantities of condensing parameter and condensate Reynolds number are shown in figure 9. Figure 12 is a correlation of condensing parameter as a function of the condensate Reynolds number based on the mean condensing heat-transfer coefficient and total quantity of condensate formed in the test section. The property values of the liquid are evaluated at the mean film temperature at the end of the condensing section.

The vapor velocity included in table I was computed from the continuity equation at zero length with the assumption that the state of the vapor did not change from the vapor inlet plenum chamber. The coolant mass velocity in table I was based on the annulus cross-sectional area minus the area of the four 1/4-inch support rods.

The coolant-side heat-transfer coefficient was calculated from the local heat flux and temperature difference between the outer surface of the condenser tube and coolant. The local heat flux used was based on the outer surface of the tube. The temperature drop from the location of the wall thermocouple to the outer diameter of the condenser tube was accounted for in evaluating the temperature of the outer surface of the tube.

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TABLE I. - EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, ^oF	Faired value of wall temperature, t_w, ^oF	Test-fluid temperature as measured by probe t_p, ^oF	Test-fluid static pressure, P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, ^oF	Temperature difference between saturated vapor and inner wall, Δt_f, ^oF	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(^oF)
				Run 1				
-0.60	-----	---	---	16.2	------	---	----	----
-.23	105.0	106	---	----	------	---	----	----
.05	103.5	155	---	----	102 000	216	25.1	4060
.14	-----	---	---	15.8	------	216	----	----
.51	102.0	155	---	----	86 000	216	30.7	2800
1.51	100.0	154	---	----	60 500	216	40.7	1487
2.51	98.5	154	---	----	51 500	216	43.9	1174
3.51	97.0	153	---	----	51 500	216	44.9	1149
3.95	-----	---	230	----	------	---	----	----
4.00	-----	---	---	15.9	------	---	----	----
4.51	95.0	151	---	----	47 500	216	48.3	983
5.51	94.5	147	---	----	35 500	216	57.2	585
6.09	-----	---	226	----	------	---	----	----
6.51	93.5	138	---	----	13 500	216	73.3	184
6.60	-----	---	225	----	------	---	----	----
7.05	-----	---	220	----	------	---	----	----
7.16	-----	---	171	----	------	---	----	----
7.48	-----	---	152	----	------	---	----	----
7.51	93.5	118	---	----	------	---	----	----
7.55	-----	---	145	----	------	---	----	----
7.85	-----	---	---	15.5	------	---	----	----
7.97	93.5	105	---	----	------	---	----	----
8.28	93.5	---	---	----	------	---	----	----

Conditions:

Test fluid total flow rate, w_{tt}, 69 lb/hr

Test fluid total mass velocity, G_{tt}, 32 350 lb/(hr)(sq ft)

Coolant flow rate, w_k, 5600 lb/hr

Coolant mass velocity, G_k, 298 000 lb/(hr)(sq ft)

Total condensing length, L_c, 7.00 ft

Vapor temperature at vapor inlet plenum chamber, 234^o F

Vapor velocity at start of condensing portion of test condenser, V_{vi}, 232 ft/sec

Mean condensing heat-transfer coefficient, h_{cm}, 1235 Btu/(hr)(sq ft)(^{o}F)

Amount of superheat at vapor inlet plenum chamber, ΔT_{sup}, 17.0^o F

Heat balance error, 9.2 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure, P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
				Run 2				
-0.60	-----	---	---	24.2	-------	---	----	----
-.23	107.5	108	---	----	-------	---	----	----
.05	105.5	163	---	----	118 500	238	33.3	3560
.14	-----	---	---	24.0	-------	238	----	----
.51	104.0	163	---	----	99 000	238	40.2	2460
1.51	101.5	163	---	----	72 500	238	49.5	1465
2.51	99.5	163	---	----	68 000	238	51.1	1330
3.51	97.5	161	---	----	61 000	238	55.5	1100
4.00	-----	---	---	24.0	-------	238	----	----
4.51	96.5	158	---	----	51 000	---	62.1	822
5.51	94.5	153	---	----	39 000	238	71.3	547
6.51	93.0	145	---	----	22 000	238	85.3	258
7.13	-----	---	241	----	-------	---	----	----
7.35	-----	---	185	----	-------	---	----	----
7.51	93.0	130	---	----	-------	---	----	----
7.70	-----	---	165	----	-------	---	----	----
7.85	-----	---	---	23.8	-------	---	----	----
7.97	93.0	110	---	----	-------	---	----	----
8.18	-----	---	160	----	-------	---	----	----
8.28	93.0	---	---	----	-------	---	----	----

Conditions:

Test fluid total flow rate, w_{tt}, 85.5 lb/hr
Test fluid total mass velocity, G_{tt}, 40 000 lb/(hr)(sq ft)
Coolant flow rate, w_k, 5550 lb/hr
Coolant mass velocity, G_k, 295 000 lb/(hr)(sq ft)
Total condensing length, L_c, 7.1 ft
Vapor temperature at vapor inlet plenum chamber, 245° F
Vapor velocity at start of condensing portion of test condenser, V_{vi}, 189 ft/sec
Mean condensing heat-transfer coefficient, h_{cm}, 1172 Btu/(hr)(sq ft)(°F)
Amount of superheat at vapor inlet plenum chamber, ΔT_{sup}, 6.6° F
Heat balance error, 8.0 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental: Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure, P_s, psia	Computed: Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
Run 3								
-0.60	-----	---	---	37.3	-------	---	-----	----
-.23	120.5	119	---	----	-------	---	-----	----
.05	120.0	199	---	----	119 000	263	22.1	5380
.14	-----	---	---	37.1	-------	263	-----	----
.51	118.0	199	---	----	114 000	263	23.9	4770
1.51	115.0	198	---	----	101 000	263	29.5	3430
2.51	112.5	195	---	----	87 000	263	37.4	2330
3.51	110.0	191	---	----	74 700	263	45.7	1635
4.00	-----	---	---	37.0	-------	---	-----	----
4.51	108.5	183	---	----	67 000	263	56.4	1188
5.51	106.0	173	---	----	63 000	263	67.8	929
6.51	104.0	163	---	----	40 500	263	85.7	473
7.51	103.5	153	---	----	5 500	263	108.1	51
7.60	-----	---	265	----	-------	---	-----	----
7.75	-----	---	208	----	-------	---	-----	----
7.85	-----	---	---	37.0	-------	---	-----	----
7.97	103.5	127	---	----	-------	---	-----	----
8.28	103.5	---	---	----	-------	---	-----	----

Conditions:

Test fluid total flow rate, w_{tt}, 107 lb/hr
Test fluid total mass velocity, G_{tt}, 50 100 lb/(hr)(sq ft)
Coolant flow rate, w_k, 5600 lb/hr
Coolant mass velocity, G_k, 298 000 lb/(hr)(sq ft)
Total condensing length, L_c, 7.7 ft
Vapor temperature at vapor inlet plenum chamber, 260° F
Vapor velocity at start of condensing portion of test condenser, V_{vi}, 157 ft/sec
Mean condensing heat-transfer coefficient, h_{cm}, 2020 Btu/(hr)(sq ft)(°F)
Quality at inlet plenum chamber, x, 0.99
Heat balance error, 9.0 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, ^{o}F	Faired value of wall temperature, t_w, ^{o}F	Test-fluid temperature, as measured by probe, t_p, ^{o}F	Test-fluid static pressure P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, ^{o}F	Temperature difference between saturated vapor and inner wall, Δt_f, ^{o}F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(^{o}F)
Run 4								
-0.60	-----	---	---	36.6	-------	---	-----	----
-.23	117.0	115	---	----	-------	---	-----	----
.05	116.0	192	---	----	132 000	262	22.6	5830
.14	-----	---	---	36.4	-------	262	-----	----
.51	114.5	192	---	----	130 000	262	24.2	5370
1.51	112.0	190	---	----	120 000	262	29.8	4030
2.51	110.0	186	---	----	103 000	262	39.7	2590
3.51	107.5	180	---	----	85 000	262	52.1	1630
4.00	-----	---	---	36.3	-------	262	-----	----
4.51	106.0	172	---	----	69 000	262	65.7	1050
5.51	104.5	163	---	----	45 000	262	83.2	540
6.51	103.5	152	---	----	25 000	262	101.2	250
7.51	103.5	145	---	----	2 000	262	116.3	17
7.70	-----	---	261	----	-------	---	-----	----
7.80	-----	---	213	----	-------	---	-----	----
7.85	-----	---	---	36.1	-------	---	-----	----
7.97	103.5	127	---	----	-------	---	-----	----
8.28	103.5	---	---	----	-------	---	-----	----

Conditions:
- Test fluid total flow rate, w_{tt}, 117 lb/hr
- Test fluid total mass velocity, G_{tt}, 54 900 lb/(hr)(sq ft)
- Coolant flow rate, w_k, 7680 lb/hr
- Coolant mass velocity, G_k, 408 000 lb/(hr)(sq ft)
- Total condensing length, L_c, 7.7 ft
- Vapor temperature at vapor inlet plenum chamber, 260^{o} F
- Vapor velocity at start of condensing portion of test condenser, V_{vi}, 175 ft/sec
- Mean condensing heat-transfer coefficient, h_{cm}, 2090 Btu/(hr)(sq ft)(^{o}F)
- Quality at inlet plenum chamber, x, 0.99
- Heat balance error, 6.5 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure, P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
Run 5								
-0.60	-----	---	---	39.4	------	---	----	----
-.23	140.5	142	---	----	------	---	----	----
.05	136.5	225	---	----	40 000	266	27.4	1460
.14	-----	---	---	39.5	------	266	----	----
.51	133.5	225	---	----	39 000	266	27.3	1430
1.51	128.0	223	---	----	37 500	266	29.8	1260
2.51	122.0	219	---	----	37 500	266	33.8	1110
3.51	117.0	213	---	----	36 800	266	40.1	920
4.00	-----	---	---	39.6	------	266	----	----
4.51	112.5	206	---	----	34 700	266	47.8	726
5.51	107.5	198	---	----	31 000	266	57.1	543
6.51	102.5	189	---	----	25 000	266	68.2	367
7.51	99.5	175	---	----	8 500	266	88.0	97
7.60	-----	---	260	----	------	---	----	----
7.75	-----	---	222	----	------	---	----	----
7.85	-----	---	212	39.6	------	---	----	----
7.90	-----	---	209	----	------	---	----	----
7.97	99.5	135	---	----	------	---	----	----
8.15	-----	---	206	----	------	---	----	----
8.28	99.5	---	---	----	------	---	----	----

Conditions:

Test fluid total flow rate, w_{tt}, 49 lb/hr

Test fluid total mass velocity, G_{tt}, 22 950 lb/(hr)(sq ft)

Coolant flow rate, w_k, 1109 lb/hr

Coolant mass velocity, G_k, 58 900 lb/(hr)(sq ft)

Total condensing length, L_c, 7.6 ft

Vapor temperature at vapor inlet plenum chamber, 265° F

Vapor velocity at start of condensing portion of test condenser, V_{vi}, 65.3 ft/sec

Mean condensing heat-transfer coefficient, h_{cm}, 847 Btu/(hr)(sq ft)(°F)

Quality at inlet plenum chamber, x, 0.96

Heat balance error, 1.3 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental: Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure, P_s, psia	Computed: Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
				Run 6				
-0.60	-----	---	---	44.6	------	---	----	----
-.23	138.5	140	---	----	------	---	----	----
.05	136.5	224	---	----	72 500	274	23.8	3050
.14	-----	---	---	44.5	------	274	----	----
.51	133.0	224	---	----	63 500	274	27.7	2290
1.51	127.0	223	---	----	49 000	274	33.8	1450
2.51	122.0	220	---	----	42 500	274	39.0	1090
3.51	117.5	214	---	----	40 000	274	45.9	872
4.00	-----	---	---	44.5	------	274	----	----
4.51	113.5	207	---	----	38 000	274	53.6	709
5.51	109.5	196	---	----	36 000	274	65.3	551
6.51	105.0	184	---	----	31 000	274	79.1	392
7.51	102.5	182	---	----	10 000	274	88.5	113
7.65	-----	---	255	----	------	---	----	----
7.75	-----	---	235	----	------	---	----	----
7.85	-----	---	222	44.4	------	---	----	----
7.95	-----	---	217	----	------	---	----	----
7.97	102.5	170	---	----	------	---	----	----
8.05	-----	---	212	----	------	---	----	----
8.28	102.5	---	---	----	------	---	----	----

Conditions:

Test fluid total flow rate, w_{tt}, 59 lb/hr
Test fluid total mass velocity, G_{tt}, 27 700 lb/(hr)(sq ft)
Coolant flow rate, w_k, 1479 lb/hr
Coolant mass velocity, G_k, 78 600 lb/(hr)(sq ft)
Total condensing length, L_c, 7.7 ft
Vapor temperature at vapor inlet plenum chamber, 273° F
Vapor velocity at start of condensing portion of test condenser, V_{vi}, 70.2 ft/sec
Mean condensing heat-transfer coefficient, h_{cm}, 1025 Btu/(hr)(sq ft)(°F)
Quality at inlet plenum chamber, x, 0.96
Heat balance error, 2.9 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure, P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
Run 7								
-0.60	-----	---	---	38.7	------	---	----	----
-.23	125.0	126	---	----	------	---	----	----
.05	123.5	202	---	----	89 600	265	30.9	2900
.14	-----	---	---	38.6	------	265	----	----
.51	121.5	202	---	----	88 000	265	32.0	2750
1.51	118.0	201	---	----	84 000	265	34.4	2440
2.51	114.5	198	---	----	76 500	265	40.1	1910
3.51	112.0	194	---	----	67 500	265	47.2	1430
4.00	-----	---	---	38.5	------	265	----	----
4.51	109.5	189	---	----	56 000	265	56.3	995
5.51	107.5	182	---	----	47 000	265	66.5	707
6.51	105.5	173	---	----	40 000	265	77.9	514
7.00	-----	166	---	----	33 000	265	87.4	378
7.25	-----	161	---	----	20 000	265	97.0	206
7.38	-----	---	240	----	------	---	----	----
7.51	104.0	155	---	----	------	---	----	----
7.60	-----	---	206	----	------	---	----	----
7.85	-----	---	196	38.2	------	---	----	----
7.97	104.0	134	---	----	------	---	----	----
8.05	-----	---	191	----	------	---	----	----
8.28	104.0	---	---	----	------	---	----	----

Conditions:

Test fluid total flow rate, w_{tt}, 84 lb/hr

Test fluid total mass velocity, G_{tt}, 39 400 lb/(hr)(sq ft)

Coolant flow rate, w_k, 3930 lb/hr

Coolant mass velocity, G_k, 209 000 lb/(hr)(sq ft)

Total condensing length, L_c, 7.4 ft

Vapor temperature at vapor inlet plenum chamber, 264° F

Vapor velocity at start of condensing portion of test condenser, V_{vi}, 115 ft/sec

Mean condensing heat-transfer coefficient, h_{cm}, 1490 Btu/(hr)(sq ft)(°F)

Quality at inlet plenum chamber, x, 0.97

Heat balance error, 2.2 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure, P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
Run 8								
-0.60	-----	---	---	42.3	-------	---	-----	------
-.23	105.5	112	---	----	-------	---	-----	------
.05	104.5	195	---	----	181 000	270	11.4	15 850
.14	-----	---	---	41.9	-------	270	-----	------
.51	102.5	195	---	----	145 500	270	23.8	6 110
1.51	99.0	194	---	----	95 000	270	42.6	2 230
2.51	98.0	192	---	----	63 500	270	55.6	1 142
3.51	97.0	187	---	----	53 000	270	64.3	824
4.00	-----	---	---	42.1	-------	270	-----	------
4.51	95.5	178	---	----	51 500	270	73.9	698
5.51	93.5	165	---	----	45 000	270	89.2	505
6.51	93.0	150	---	----	22 000	270	112.3	196
6.75	-----	146	---	----	12 500	270	119.6	104
7.00	-----	142	---	----	2 500	270	127.1	------
7.22	-----	---	272	----	-------	---	-----	------
7.40	-----	---	226	----	-------	---	-----	------
7.51	93.0	132	---	----	-------	---	-----	------
7.55	-----	---	206	----	-------	---	-----	------
7.85	-----	---	197	41.5	-------	---	-----	------
7.97	93.0	112	---	----	-------	---	-----	------
8.15	-----	---	190	----	-------	---	-----	------
8.28	93.0	---	---	----	-------	---	-----	------

Conditions:

Test fluid total flow rate, w_{tt}, 94 lb/hr

Test fluid total mass velocity, G_{tt}, 44 100 lb/(hr)(sq ft)

Coolant flow rate, w_k, 6540 lb/hr

Coolant mass velocity, G_k, 348 000 lb/(hr)(sq ft)

Total condensing length, L_c, 7.2 ft

Vapor temperature at vapor inlet plenum chamber, 273° F

Vapor velocity at start of condensing portion of test condenser, V_{vi}, 123 ft/sec

Mean condensing heat-transfer coefficient, h_{cm}, 1850 Btu/(hr)(sq ft)(°F)

Amount of superheat at vapor inlet plenum chamber, ΔT_{sup}, 1.9° F

Heat balance error, -0.9 percent

I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
				Run 9				
-0.60	-----	-----	---	20.2	------	---	----	----
-.23	124.0	125.0	---	----	------	---	----	----
.05	120.0	192.0	---	----	40 500	228	22.3	1818
.14	-----	-----	---	20.2	------	228	----	----
.51	117.0	192.0	---	----	36 000	228	23.3	1546
1.51	113.0	191.0	---	----	29 000	228	26.8	1083
2.51	108.0	188.0	---	----	25 400	228	31.1	816
3.51	104.5	183.0	---	----	23 600	228	36.7	644
4.00	-----	-----	---	20.2	------	228	----	----
4.51	101.5	179.0	---	----	22 500	228	41.1	547
5.51	98.0	169.0	---	----	21 300	228	51.5	414
6.51	95.0	149.0	---	----	16 000	228	73.4	218
6.80	-----	-----	213	----	------	---	----	----
6.90	-----	-----	179	----	------	---	----	----
7.05	-----	-----	159	----	------	---	----	----
7.28	-----	-----	139	----	------	---	----	----
7.51	94.0	118.0	---	----	------	---	----	----
7.60	-----	-----	129	----	------	---	----	----
7.85	-----	-----	---	19.3	------	---	----	----
7.97	94.0	100.0	---	----	------	---	----	----
8.28	94.0	-----	---	----	------	---	----	----

Conditions:

Test fluid total flow rate, w_{tt}, 34 lb/hr
Test fluid total mass velocity, G_{tt}, 15 940 lb/(hr)(sq ft)
Coolant flow rate, w_k, 1093 lb/hr
Coolant mass velocity, G_k, 58 300 lb/(hr)(sq ft)
Total condensing length, L_c, 6.8 ft
Vapor temperature at vapor inlet plenum chamber, 227° F
Vapor velocity at start of condensing portion of test condenser, V_{vi}, 84 ft/sec
Mean condensing heat-transfer coefficient, h_{cm}, 790 Btu/(hr)(sq ft)(°F)
Quality at inlet plenum chamber, x, 0.94
Heat balance error, 4.2 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure, P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
Run 10								
-0.60	-----	---	---	55.8	-------	---	-----	----
-.23	114.5	115	---	----	-------	---	-----	----
.05	113.5	205	---	----	150 500	288	30.2	4990
.14	-----	---	---	55.7	-------	288	-----	----
.51	111.5	204	---	----	143 500	288	33.5	4280
1.51	108.5	202	---	----	119 000	288	44.1	2700
2.51	106.0	198	---	----	80 000	288	61.8	1295
3.51	104.5	193	---	----	62 000	288	73.2	846
4.00	-----	---	---	55.6	-------	288	-----	----
4.51	103.0	184	---	----	59 000	288	83.2	710
5.51	101.5	172	---	----	59 000	288	95.2	620
6.51	100.0	157	---	----	47 000	288	114.5	410
6.75	-----	151	---	----	38 000	288	115.6	329
6.96	-----	---	292	----	-------	---	-----	----
6.98	-----	---	290	----	-------	---	-----	----
7.02	-----	---	283	----	-------	---	-----	----
7.11	-----	---	243	----	-------	---	-----	----
7.29	-----	---	232	----	-------	---	-----	----
7.44	-----	---	226	----	-------	---	-----	----
7.51	99.5	122	---	----	-------	---	-----	----
7.65	-----	---	222	----	-------	---	-----	----
7.85	-----	---	---	55.5	-------	---	-----	----
7.97	99.5	104	---	----	-------	---	-----	----
7.98	-----	---	218	----	-------	---	-----	----
8.28	-----	---	---	----	-------	---	-----	----

Conditions:

Test fluid total flow rate, w_{tt}, 97 lb/hr

Test fluid total mass velocity, G_{tt}, 45 500 lb/(hr)(sq ft)

Coolant flow rate, w_k, 6650 lb/hr

Coolant mass velocity, G_k, 353 500 lb/(hr)(sq ft)

Total condensing length, L_c, 7.0 ft

Vapor temperature at vapor inlet plenum chamber, 290° F

Vapor velocity at start of condensing portion of test condenser, V_{vi}, 97.5 ft/sec

Mean condensing heat-transfer coefficient, h_{cm}, 1555 Btu/(hr)(sq ft)(°F)

Amount of superheat at vapor inlet plenum chamber, ΔT_{sup}, 1.7° F

Heat balance error, -3.4 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe t_p, °F	Test-fluid static pressure, P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
Run 11								
-0.60	----	---	---	16.9	-------	---	----	----
-.23	96.0	96	---	----	-------	---	----	----
.05	95.5	158	---	----	100 000	218	24.8	4030
.14	----	---	---	16.5	-------	218	----	----
.51	94.5	158	---	----	95 500	218	26.4	3620
1.10	----	---	220	----	-------	---	----	----
1.51	92.5	158	---	----	83 000	218	30.8	2695
2.51	91.0	156	---	----	68 000	218	38.1	1785
3.51	89.5	153	---	----	53 000	218	46.3	1146
3.60	----	---	220	----	-------	---	----	----
4.00	----	---	---	16.7	-------	218	----	----
4.51	88.5	148	---	----	41 000	218	55.6	737
5.40	----	---	219	----	-------	---	----	----
5.51	88.0	136	---	----	31 500	218	70.9	445
6.51	87.0	120	219	----	13 500	218	93.2	145
6.65	----	---	218	----	-------	---	----	----
6.75	----	---	173	----	-------	---	----	----
7.00	----	---	158	----	-------	---	----	----
7.30	----	---	141	----	-------	---	----	----
7.45	----	---	131	----	-------	---	----	----
7.51	87.0	100	---	----	-------	---	----	----
7.75	----	---	124	----	-------	---	----	----
7.85	----	---	---	15.5	-------	---	----	----
7.95	----	---	120	----	-------	---	----	----
7.97	87.0	90	---	----	-------	---	----	----
8.28	87.0	---	---	----	-------	---	----	----

Conditions:

Test fluid total flow rate, w_{tt}, 72 lb/hr

Test fluid total mass velocity, G_{tt}, 33 750 lb/(hr)(sq ft)

Coolant flow rate, w_k, 7450 lb/hr

Coolant mass velocity, G_k, 396 000 lb/(hr)(sq ft)

Total condensing length, L_c, 6.7 ft

Vapor temperature at vapor inlet plenum chamber, 221° F

Vapor velocity at start of condensing portion of test condenser, V_{vi}, 226 ft/sec

Mean condensing heat-transfer coefficient, h_{cm}, 1572 Btu/(hr)(sq ft)(°F)

Amount of superheat at vapor inlet plenum chamber, ΔT_{sup}, 1.6° F

Heat balance error, 11.0 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental				Computed			
	Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure, P_s, psia	Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
Run 12								
-0.60	-----	---	---	30.7	------	---	----	----
-.23	129.0	131	---	----	------	---	----	----
.05	127.0	209	---	----	53 500	252	23.8	2250
.14	-----	---	---	30.8	------	252	----	----
.51	122.0	209	---	----	51 000	252	25.1	2030
1.51	116.5	208	---	----	43 000	252	28.9	1489
2.51	111.0	204	---	----	33 500	252	36.2	925
3.42	-----	---	253	----	------	---	----	----
3.51	105.5	198	---	----	27 000	252	44.5	607
4.00	-----	---	---	30.9	------	252	----	----
4.02	-----	---	253	----	------	---	----	----
4.51	102.5	192	---	----	24 000	252	51.6	465
4.91	-----	---	253	----	------	---	----	----
5.51	98.5	184	---	----	19 500	252	61.1	319
5.75	-----	---	253	----	------	---	----	----
6.00	-----	177	---	----	16 500	252	69.2	238
6.10	-----	---	253	----	------	---	----	----
6.15	-----	---	216	----	------	---	----	----
6.26	-----	---	203	----	------	---	----	----
6.51	95.5	165	188	----	------	---	----	----
6.95	-----	---	169	----	------	---	----	----
7.51	95.5	109	153	----	------	---	----	----
7.85	-----	---	---	29.8	------	---	----	----
7.97	95.5	99	143	----	------	---	----	----
8.28	95.5	---	---	----	------	---	----	----

Conditions:

Test fluid total flow rate, w_{tt}, 40 lb/hr

Test fluid total mass velocity, G_{tt}, 18 750 lb/(hr)(sq ft)

Coolant flow rate, w_k, 1085 lb/hr

Coolant mass velocity, G_k, 57 600 lb/(hr)(sq ft)

Total condensing length, L_c, 6.1 ft

Vapor temperature at vapor inlet plenum chamber, 251° F

Vapor velocity at start of condensing portion of test condenser, V_{vi}, 66 ft/sec

Mean condensing heat-transfer coefficient, h_{cm}, 994 Btu/(hr)(sq ft)(°F)

Quality at inlet plenum chamber, x, 0.94

Heat balance error, 3.8 percent

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental: Coolant temperature, t_k, °F	Faired value of wall temperature, t_w, °F	Test-fluid temperature, as measured by probe, t_p, °F	Test-fluid static pressure, P_s, psia	Computed: Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, °F	Temperature difference between saturated vapor and inner wall, Δt_f, °F	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(°F)
Run 13								
-0.60	-----	---	---	35.4	------	---	----	----
-.23	118.0	120	---	----	------	---	----	----
.05	117.5	201	---	----	97 000	260	24.5	3960
.14	-----	---	---	35.5	------	260	----	----
.51	113.5	200	---	----	85 000	260	30.1	2720
1.51	110.0	198	---	----	55 000	260	42.6	1290
2.51	107.5	194	---	----	46 000	260	49.8	924
3.51	105.0	188	---	----	43 000	260	56.9	756
3.79	-----	---	262	----	43 000	---	----	----
4.00	-----	---	---	35.5	------	260	----	----
4.51	102.0	178	262	----	41 000	260	67.6	606
5.15	-----	---	262	----	------	---	----	----
5.51	101.0	167	---	----	35 000	260	80.7	434
5.58	-----	---	262	----	------	---	----	----
6.00	-----	160	---	----	19 000	260	93.3	204
6.10	-----	---	262	----	------	---	----	----
6.15	-----	---	255	----	------	---	----	----
6.41	-----	---	196	----	------	---	----	----
6.51	99.5	150	---	----	------	---	----	----
6.75	-----	---	182	----	------	---	----	----
7.00	-----	---	177	----	------	---	----	----
7.51	99.5	108	---	----	------	---	----	----
7.55	-----	---	163	----	------	---	----	----
7.85	-----	---	---	34.6	------	---	----	----
7.90	-----	---	156	----	------	---	----	----
7.97	99.5	104	---	----	------	---	----	----
8.28	99.5	---	---	----	------	---	----	----

Conditions:
- Test fluid total flow rate, w_{tt}, 57 lb/hr
- Test fluid total mass velocity, G_{tt}, 26 700 lb/(hr)(sq ft)
- Coolant flow rate, w_k, 2982 lb/hr
- Coolant mass velocity, G_k, 158 000 lb/(hr)(sq ft)
- Total condensing length, L_c, 6.1 ft
- Vapor temperature at vapor inlet plenum chamber, 259° F
- Vapor velocity at start of condensing portion of test condenser, V_{vi}, 87 ft/sec
- Mean condensing heat-transfer coefficient, h_{cm}, 1170 Btu/(hr)(sq ft) (°F)
- Quality at inlet plenum chamber, x, 0.99
- Heat balance error, 2.9 percent

TABLE I. - Concluded. EXPERIMENTAL AND COMPUTED DATA

Length, L, ft	Experimental: Coolant temperature, t_k, ^oF	Faired value of wall temperature, t_w, ^oF	Test-fluid temperature, as measured by probe, t_p, ^oF	Test-fluid static pressure, P_s, psia	Computed: Heat flux based on inside diameter, q_i, Btu/(hr)(sq ft)	Saturated-vapor temperature, t_{vs}, ^oF	Temperature difference between saturated vapor and inner wall, Δt_f, ^oF	Local condensing heat-transfer coefficient, $h_{c\ell}$, Btu/(hr)(sq ft)(^oF)
Run 14								
-0.60	-----	---	---	35.3	-------	---	----	----
-.23	114.0	114	---	----	-------	---	----	----
.05	113.0	190	---	----	138 500	260	21.3	6500
.14	-----	---	---	35.3	-------	260	----	----
.51	110.0	189	---	----	110 000	260	32.3	3410
1.51	108.0	187	---	----	72 000	260	47.7	1510
2.51	105.5	183	---	----	60 000	260	55.9	1075
3.51	103.5	179	---	----	54 000	260	62.0	872
3.55	-----	---	263	----	-------	---	----	----
3.85	-----	---	262	----	-------	---	----	----
4.00	-----	---	---	35.3	-------	260	----	----
4.45	-----	---	264	----	-------	---	----	----
4.51	102.0	173	---	----	47 000	260	70.5	667
4.90	-----	---	262	----	-------	---	----	----
5.40	-----	---	264	----	-------	---	----	----
5.51	100.5	158	---	----	37 000	260	89.0	416
5.75	-----	151	263	----	27 000	260	99.5	271
5.86	-----	---	258	----	-------	---	----	----
6.25	-----	---	192	----	-------	---	----	----
6.51	100.0	127	---	----	-------	---	----	----
6.55	-----	---	182	----	-------	---	----	----
6.90	-----	---	175	----	-------	---	----	----
7.35	-----	---	168	----	-------	---	----	----
7.51	100.0	105	---	----	-------	---	----	----
7.85	-----	---	---	34.2	-------	---	----	----
7.97	100.0	102	---	----	-------	---	----	----
8.28	100.0	---	---	----	-------	---	----	----

Condition:

Test fluid total flow rate, w_{tt}, 68 lb/hr
Test fluid total mass velocity, G_{tt}, 31 850 lb/(hr)(sq ft)
Coolant flow rate, w_k, 4815 lb/hr
Coolant mass velocity, G_k, 256 000 lb/(hr)(sq ft)
Total condensing length, L_c, 5.9 ft
Vapor temperature at vapor inlet plenum chamber, 262^o F
Vapor velocity at start of condensing portion of test condenser, V_{vi}, 105 ft/sec
Mean condensing heat-transfer coefficient, h_{cm}, 1438 Btu/(hr)(sq ft)(^oF)
Amount of superheat at vapor inlet plenum chamber, ΔT_{sup}, 1.7^o F
Heat balance error, 0 percent

TABLE II. - MEASURED CONDENSER TUBE-WALL TEMPERATURES

Length, L, ft	Run													
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
	Temperature, ^{o}F													
0.17	162	164	201	195	228	229	207	201	195	209	164	214	206	199
.50	150	159	188	179	221	216	195	184	188	189	151	204	191	183
.83	162	164	201	196	229	229	207	199	196	207	160	214	203	194
1.17	150	163	193	186	225	221	200	190	191	197	152	208	194	183
1.83	159	161	201	196	223	223	206	196	192	205	158	206	197	191
2.16	150	165	199	193	221	219	203	195	188	202	156	201	190	186
2.50	155	162	186	176	212	215	193	182	183	187	146	199	182	174
2.83	150	167	209	204	221	224	212	204	189	211	161	204	197	191
3.16	161	160	197	192	216	216	202	193	184	194	150	197	186	182
3.49	148	156	176	169	206	200	184	170	171	175	139	191	170	164
3.82	154	160	196	186	215	216	198	188	185	194	151	201	188	181
4.15	150	160	185	178	210	207	188	179	179	188	146	194	178	173
4.49	151	160	191	176	211	214	199	187	181	190	149	197	189	182
4.82	146	160	190	183	208	208	194	185	177	191	150	194	182	180
5.15	149	154	170	165	200	199	176	162	167	168	135	184	166	164
5.48	149	155	178	164	201	199	184	169	168	173	140	186	173	170
5.82	141	148	160	156	194	188	169	153	160	158	128	181	162	153
6.15	140	150	163	155	193	187	169	155	159	158	128	179	164	141
6.48	139	146	161	152	186	178	169	154	157	165	132	162	154	129
7.15	131	141	159	150	191	189	167	153	135	152	112	114	111	105
7.48	117	127	155	148	185	183	158	134	117	115	101	109	109	103
7.82	110	118	146	142	171	179	143	137	105	111	97	106	107	102

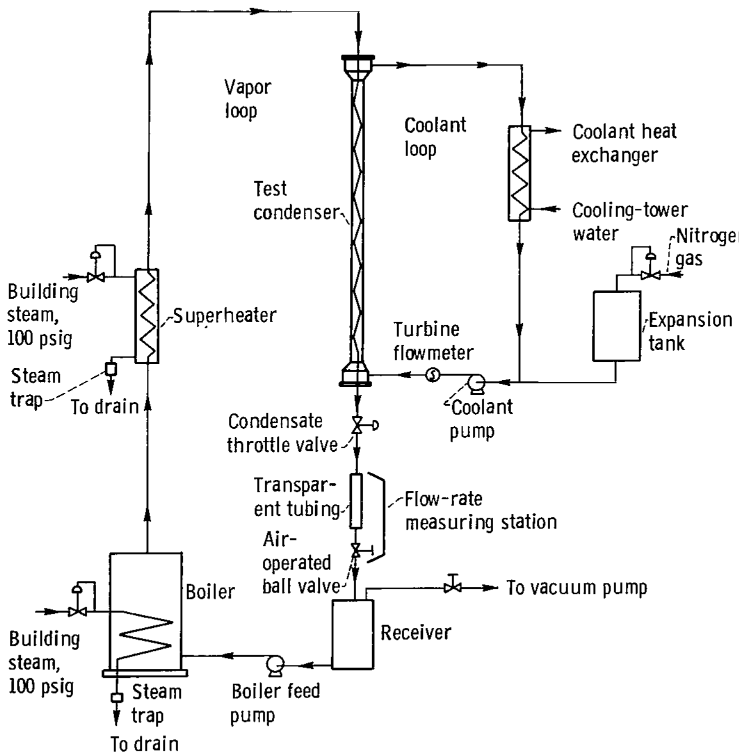


Figure 1. - Schematic drawing of single-tube steam-condensing apparatus.

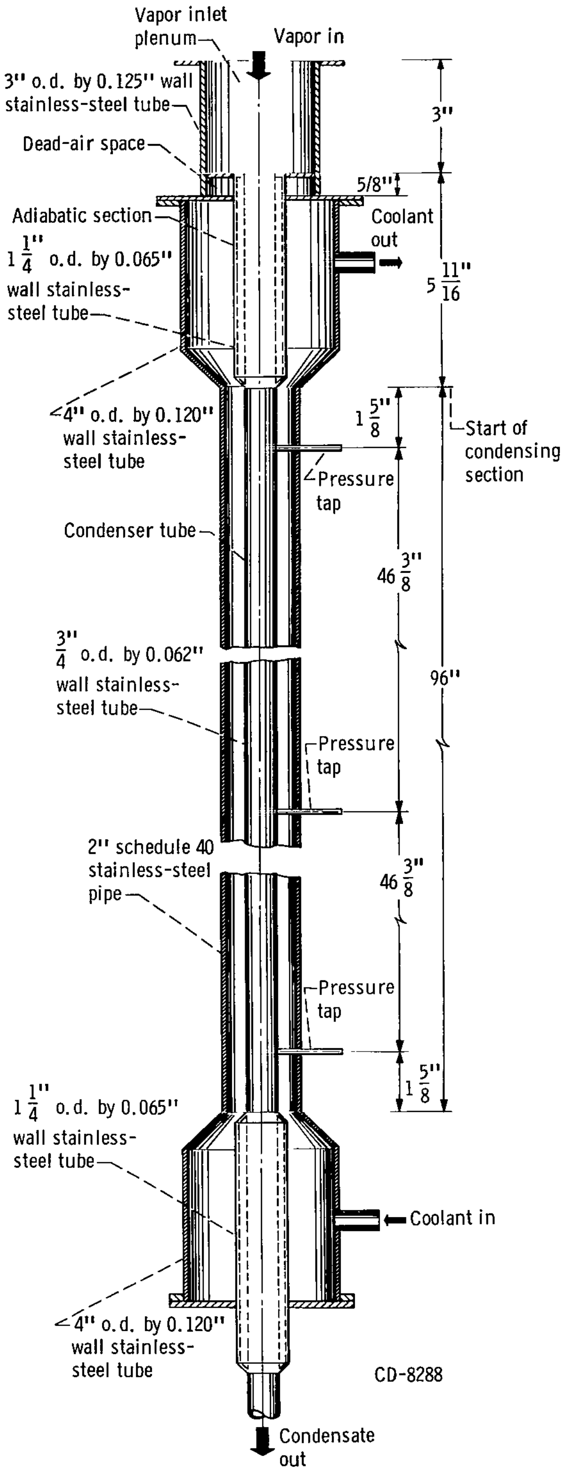


Figure 2. - Schematic drawing of single-tube-condenser test section.

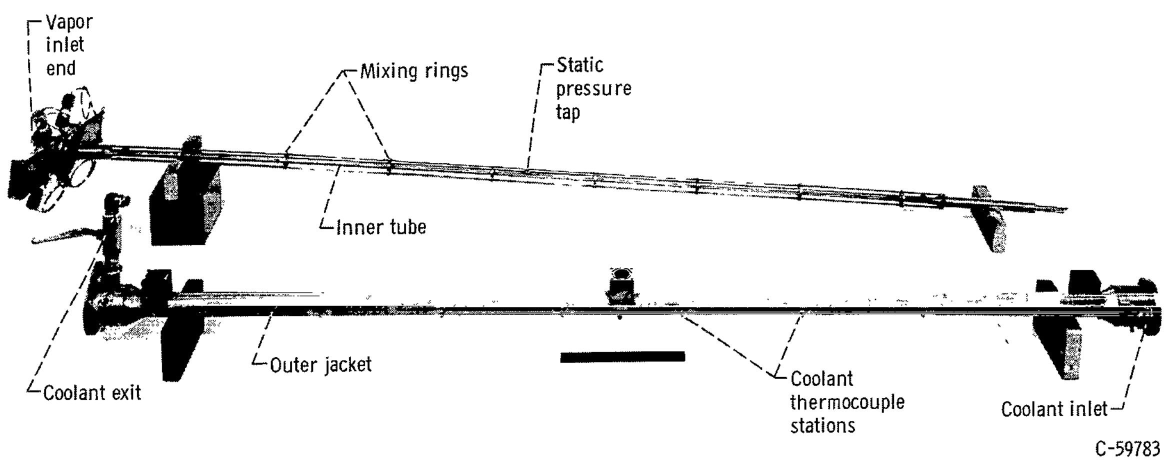


Figure 3. - View of inner tube and outer jacket of test condenser.

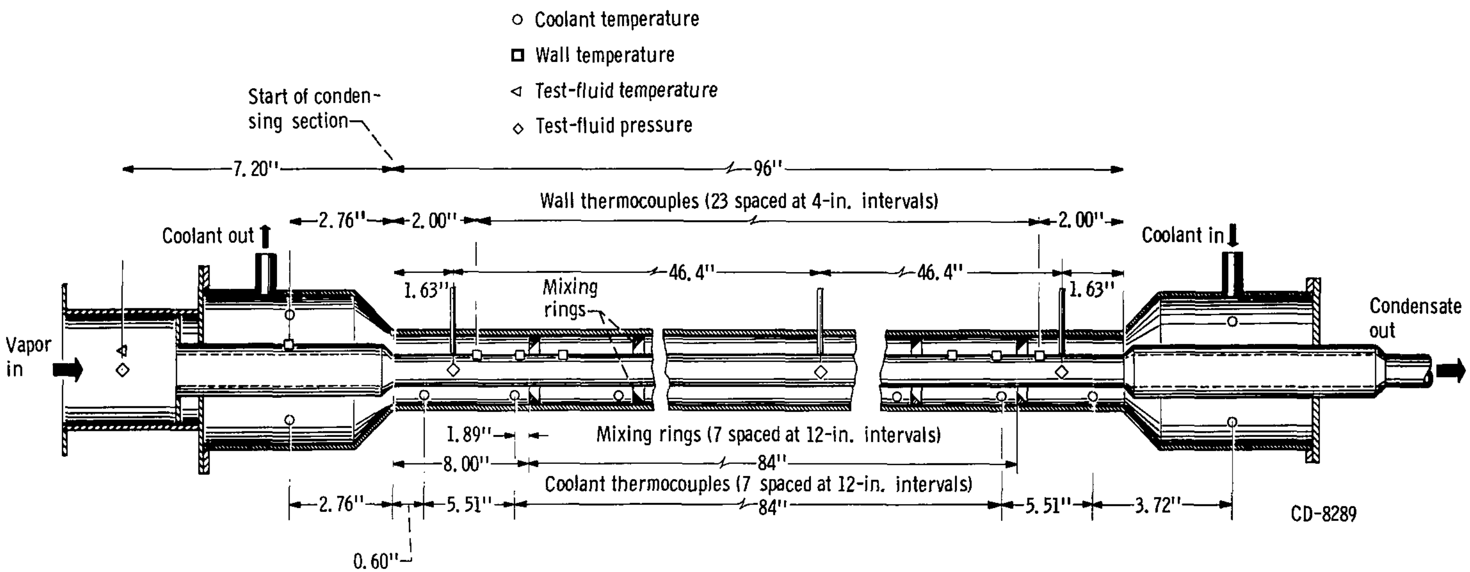


Figure 4. - Test-section instrumentation.

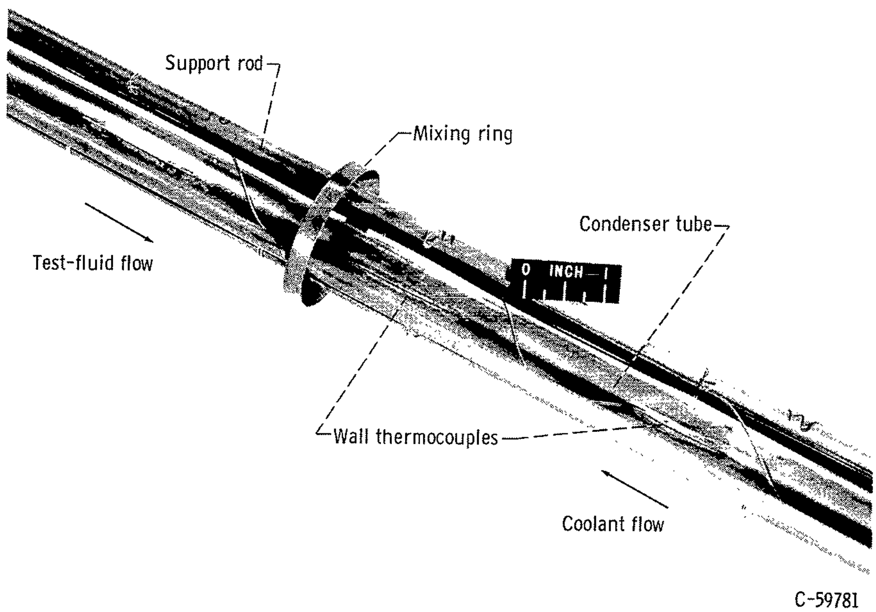
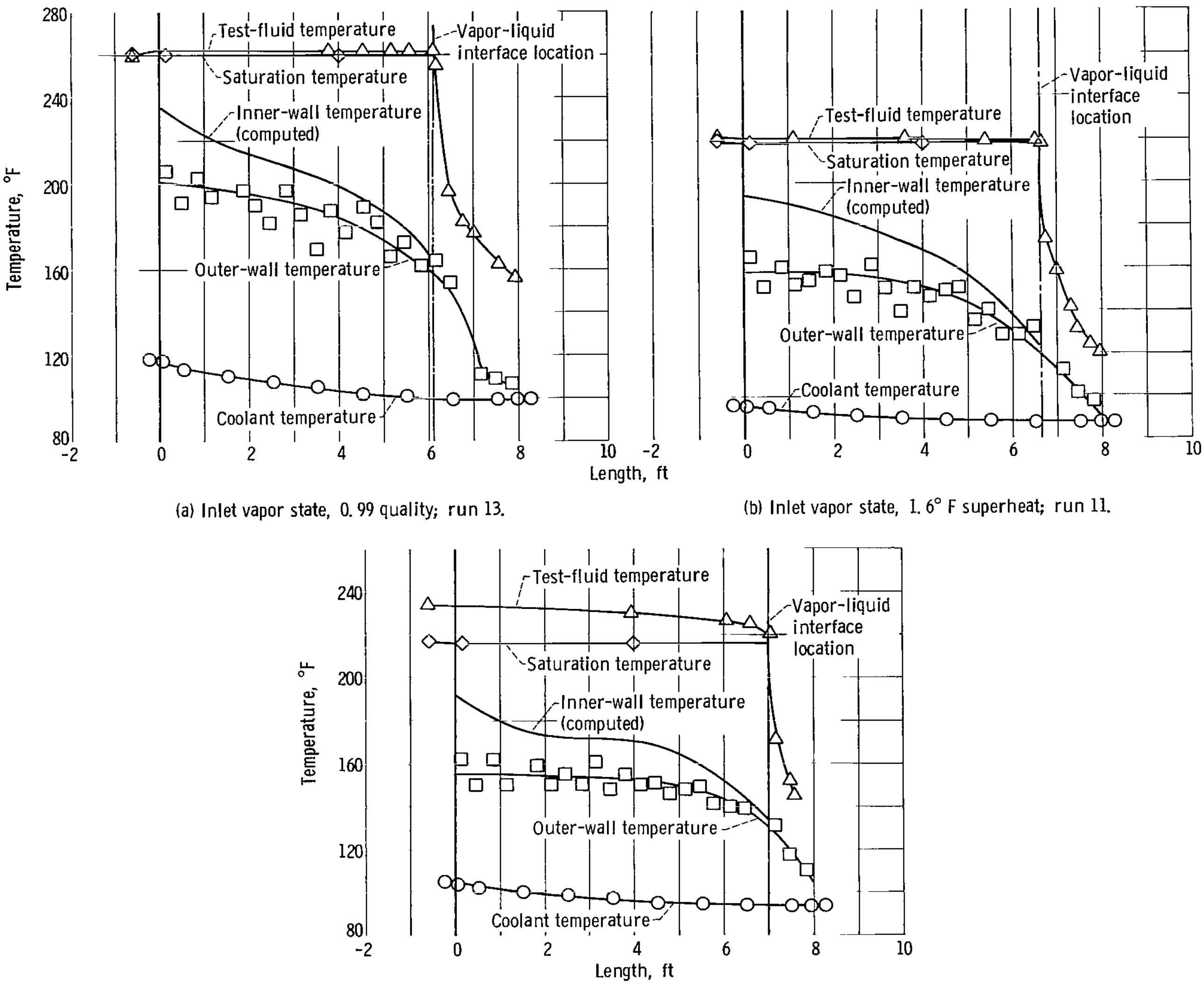


Figure 5. - Condenser tube wall thermocouple installation.



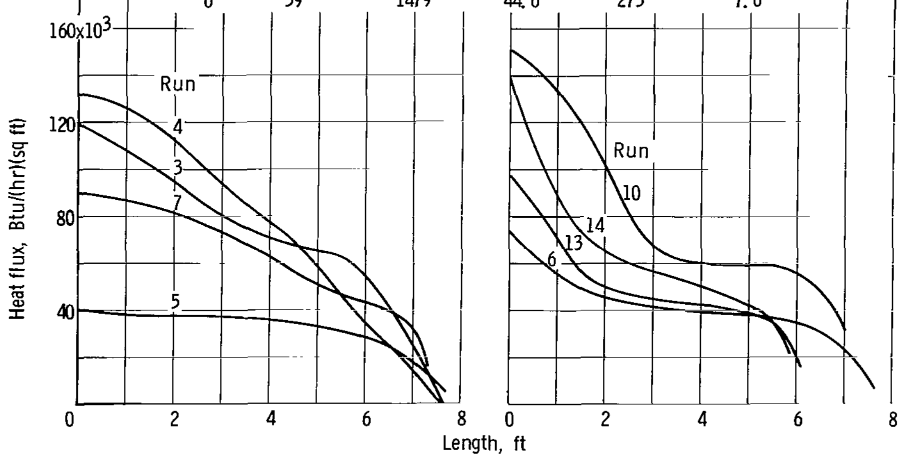
(a) Inlet vapor state, 0.99 quality; run 13.

(b) Inlet vapor state, 1.6° F superheat; run 11.

(c) Inlet vapor state, 17° F superheat; run 1.

Figure 6. - Typical axial temperature profiles obtained in single-tube steam-condensing tests.

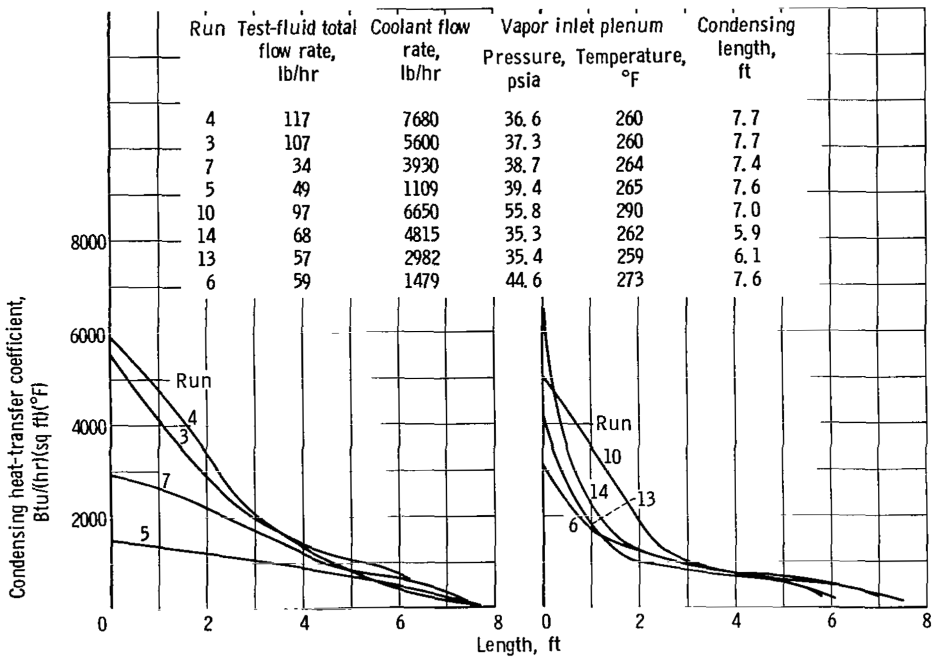
Run	Test-fluid total flow rate, lb/hr	Coolant flow rate, lb/hr	Vapor inlet plenum Pressure, psia	Vapor inlet plenum Temperature, °F	Condensing length, ft
4	117	7680	36.6	260	7.7
3	107	5600	37.3	260	7.7
7	84	3930	38.7	264	7.4
5	49	1109	39.4	265	7.6
10	97	6650	55.8	290	7.0
14	68	4815	35.3	262	5.9
13	57	2982	35.4	259	6.1
6	59	1479	44.6	273	7.6



(a) Runs 3, 4, 5, and 7.

(b) Runs 6, 10, 13, and 14.

Figure 7. - Variation of local heat flux with length for single-tube steam-condensing tests.



Run	Test-fluid total flow rate, lb/hr	Coolant flow rate, lb/hr	Vapor inlet plenum Pressure, psia	Vapor inlet plenum Temperature, °F	Condensing length, ft
4	117	7680	36.6	260	7.7
3	107	5600	37.3	260	7.7
7	34	3930	38.7	264	7.4
5	49	1109	39.4	265	7.6
10	97	6650	55.8	290	7.0
14	68	4815	35.3	262	5.9
13	57	2982	35.4	259	6.1
6	59	1479	44.6	273	7.6

(a) Runs 3, 4, 5, and 7.

(b) Runs 6, 10, 13, and 14.

Figure 8. - Variation of condensing heat-transfer coefficient with length for single-tube steam-condensing tests.

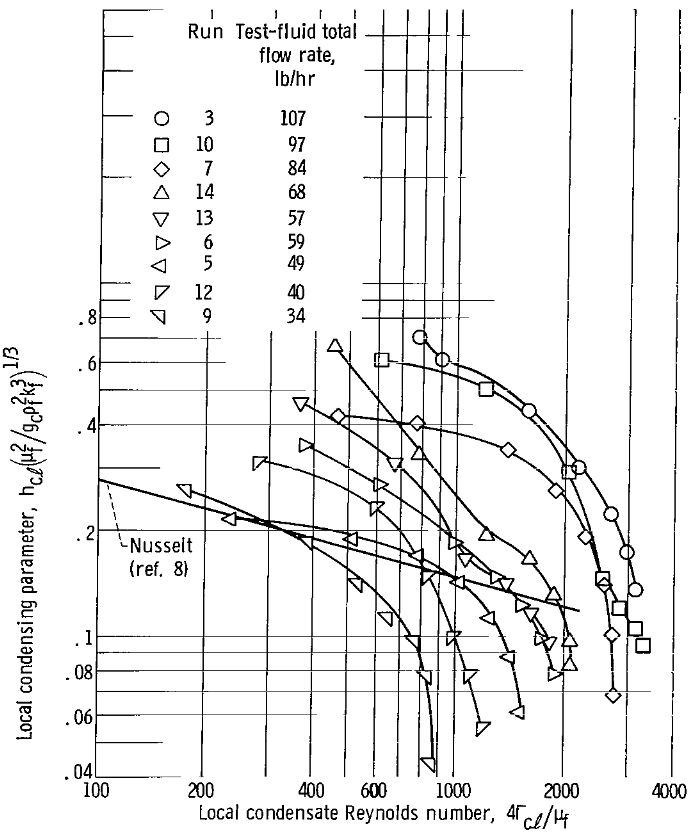


Figure 9. - Local condensing heat-transfer parameter as function of local condensate Reynolds number.

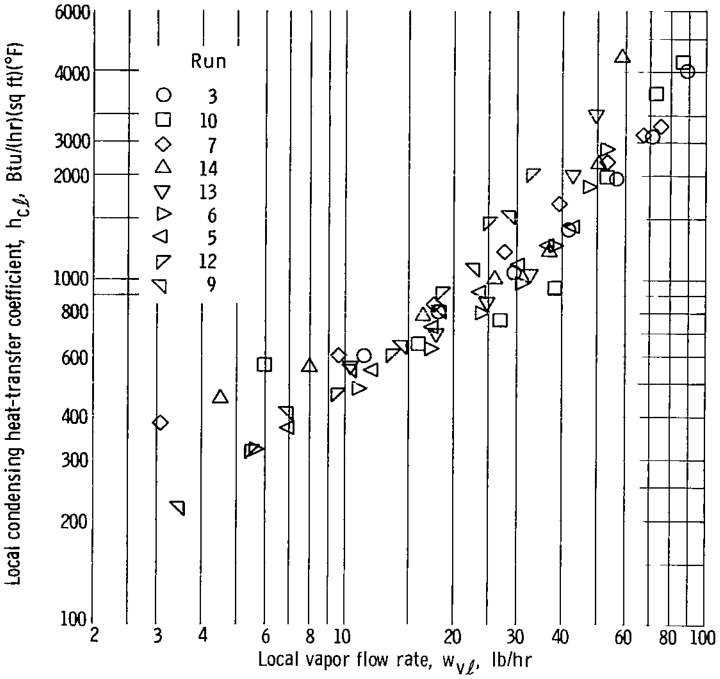
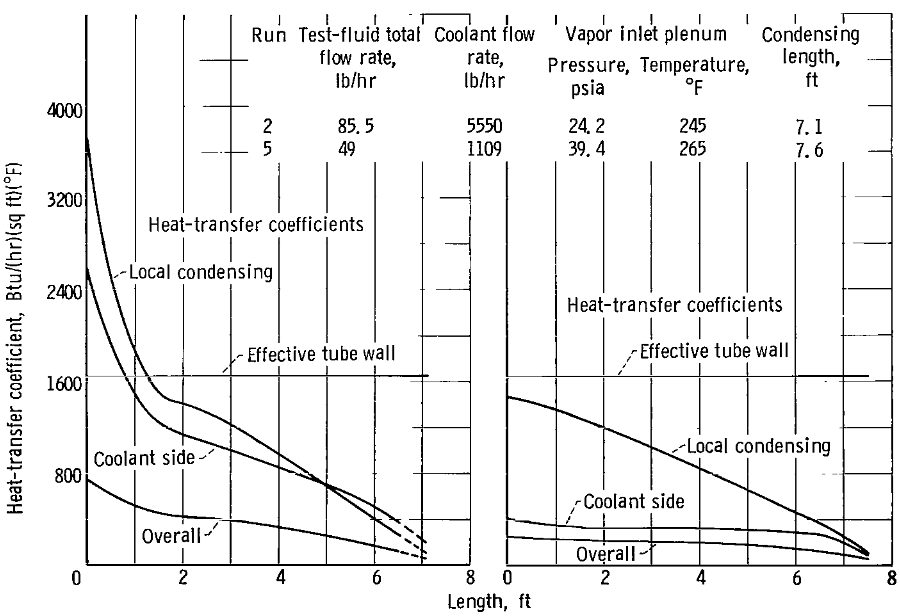


Figure 10. - Local condensing heat-transfer coefficient as function of vapor flow rate.



(a) Run 2.

(b) Run 5.

Figure 11. - Variation of individual and overall heat-transfer coefficients with length.

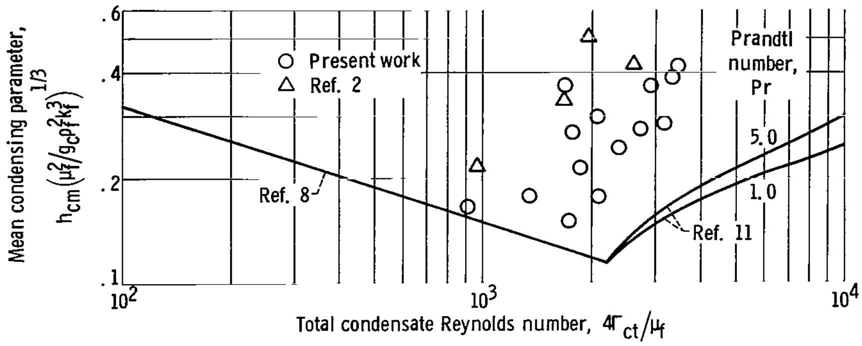


Figure 12. - Mean condensing heat-transfer parameter as function of total condensate Reynolds number.

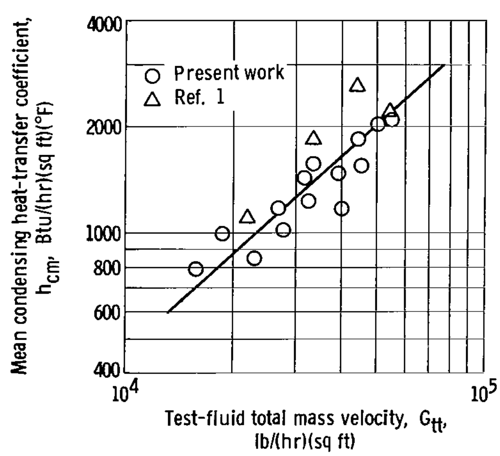


Figure 13. - Mean condensing heat-transfer coefficient as function of test-fluid total mass velocity.

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