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SUMMARY

Local heat-transfer data and static-pressure distributions for Freon-113 condensing inside a vertical tube are presented. The test condenser was a 0.293-inch-inside-diameter by 8-foot-long, water-cooled copper tube. Incomplete condensing occurred in the condenser with exit qualities ranging from 0.05 to 0.40. Local condensing heat-transfer coefficients varied from 3300 Btu per hour per square foot per $^{\circ}$F at the vapor inlet end to 200 Btu per hour per square foot per $^{\circ}$F at the discharge end. The local condensing heat-transfer coefficients for Freon-113 were satisfactorily correlated by using a Carpenter-Colburn type of relation for high-velocity condensing. Overall friction-pressure losses were computed and found to be a function of a group of variables used in single-phase pipe-friction problems.

INTRODUCTION

The research presented herein is a continuation of an experimental program initiated at the Lewis Research Center on inside-tube condensers. The program was designed to obtain local heat-transfer and static-pressure data for condensing with vapor velocities greater than 200 feet per second.

References 1 and 2 present the results of previous work in which steam was used as the test fluid. The results of the steam work showed that local condensing heat-transfer coefficients were proportional to local vapor flow rates. The vapor velocity effect corroborated the analytical work of references 3 to 5. In addition, overall friction-pressure losses were correlated with a group of variables that are commonly used in single-phase pipe-friction problems. The experimental results of references 1 and 2, however, left open the question of the influence of various fluid properties on condensing heat transfer. Therefore, the principal objective of the investigation described in this report was to study the fluid property effects.

Local heat-transfer coefficients and friction-pressure losses were obtained for Freon-113 (trichlorotrifluoroethane) condensing inside a tube over a range of operating conditions. Freon-113 was selected because its fluid properties differ enough from water, used in the previous tests, to permit comparisons of their effects. The fluid properties of particular interest were the liquid Prandtl number and the liquid-to-vapor density ratio. The Prandtl number of liquid Freon-113 is approximately $4\frac{1}{2}$ times that of water at saturation conditions and 1 atmosphere pressure. The liquid-to-vapor density ratio for Freon-113 is 195 at 1 atmosphere; the density ratio for water is 1600 at the same pressure. In addition to these considerations, the pressure, temperature, and flow ranges for Freon-113 were compatible with the experimental apparatus used to obtain the steam data of references 1 and 2.

The test condenser was a 0.293-inch-inside-diameter by 8-foot-long water-cooled copper tube. The condenser was mounted vertically with the vapor entering at the top and was cooled by water flowing countercurrently in an annulus around the tube.

The range of variables covered was as follows:

Test-fluid total flow rate, w, lb/hr	387 to 506
Test-fluid total mass velocity, G, lb/(hr)(ft^2)	613 000 to 1 080 000
Inlet-vapor pressure, P_{si}, psia	26.92 to 44.20
Inlet-vapor temperature, t_{vi}, ^{o}F	193 to 229
Inlet-vapor superheat, Δt_{sup}, ^{o}F	18 to 53
Coolant flow rate, w_k, lb/hr	434 to 1055
Coolant mass velocity, G_k, lb/(hr)(ft^2)	510 000 to 1 239 000
Coolant temperature, ^{o}F	
Inlet, t_{ki},	62 to 95
Exit, t_{ko}	97 to 133

APPARATUS AND PROCEDURE

Description of Facility

The condenser facility is shown in figure 1. The test-fluid side of the apparatus was a once-through system using Freon-113. Demineralized water was continuously circulated in the coolant loop. Building supply steam at 100 pounds per square inch gage was used as the heat source and cooling tower water as the final heat sink.

The equipment in the Freon-113 circuit consisted of a pot boiler, superheater, flow straightener, test section, condensate cooler, condensate flow measuring station, and receiver tank. The boiler was a 94-gallon tank with coiled tubes at the bottom of the tank, through which building supply steam flowed. A wire mesh screen and a baffle separator were located at the boiler exit to impede liquid droplet carryover. The superheater was a shell-and-tube heat exchanger with building supply steam on the shell side and Freon-113 vapor in the tubes. Wall heaters were installed around the vapor line between the superheater and the test-section inlet to reduce heat losses in this region. The wall heater consisted of 0.25-inch tubing spirally wrapped around the vapor line and soldered in place. Supply steam, flowing inside the 0.25-inch tubing, served as the heat source.

The single-tube condenser (fig. 2) was a shell and tube heat exchanger; the vapor condensed inside the inner tube and water flowed in the annulus between the inner and outer tubes. The test section was mounted vertically; the vapor entered at the top and the coolant flowed countercurrently in the annulus. The inner tube was a copper tube with an outside diameter of 0.541 inch and an inside diameter of 0.293 inch. The outer jacket was a copper tube with a 0.750-inch outside diameter and a 0.670-inch inside diameter. The space between the inner and outer tubes was 0.0645 inch. Spacer pins were placed in the annulus to maintain concentricity between the inner and outer tubes. The total length of the heat-exchange region was 8 feet. The inner diameter of the inlet-vapor line changed from 1.049 to 0.293 inch at a distance of 18.5 inches upstream of the test section. A bell-shaped fitting at this location accommodated the change in cross section. A stainless steel ring (inset in fig. 2) was placed between the inlet-vapor line and the beginning of the heat-exchange region of the test condenser to reduce axial heat conduction in the thick-wall tube. The lower end of the test section was equipped with a stainless steel bellows between the inner tube and the outer jacket to allow for thermal expansion. All vapor lines were insulated with molded magnesia, and the test-section shell was lagged with blanket insulation.

The condensate cooler was a shell-and-tube heat exchanger that condensed the excess vapor and/or subcooled the fluid flowing from the test section. Coolant-loop components included a variable-speed pump, a turbine-type flowmeter, a heat exchanger, and an expansion tank.

Instrumentation

The locations of the pressure and temperature measuring stations on the single-tube test condenser are shown in figure 3. Pressures were measured with U-tube manometers that used mercury as the manometer fluid and a hydrostatic liquid of known height, namely Freon-113, between the pressure tap and the manometer fluid. A horizontal run of bare metal tubing 12 inches long 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. Ambient pressure existed on the reference leg of the manometer. All temperatures were measured with iron-constantan thermocouples that were insulated with magnesium oxide and swaged inside a 1/16-inch-diameter sheath. The thermocouple wires were 0.005 inch in diameter. The coolant thermocouples were placed at the midpoint of the annular gap between the inner and outer tubes with the leads projecting radially outward. The inner-tube-wall thermocouple leads, as well as the static-pressure tubes, passed radially outward through the coolant stream, and bellows compensated for relative motion between the inner and outer tubes. The construction and installation of the wall thermocouples and the pressure taps are shown in the insets in figure 3.

Vapor temperature at the midpoint of the stream was measured in the 1-inch line at a position 2.16 feet upstream of the heat-exchange region of the test section. In addition, vapor temperatures were measured at three axial positions in the small-diameter tube (0.293-in. inner diam) between the transition section and the beginning of the test section. All temperatures were recorded on a self-balancing potentiometer.

Total condensate flow rate was measured volumetrically. The system consisted of a quick-shutoff valve located downstream of a 3-foot-long section of 2-inch tubing. The tube was equipped with a sight glass that allowed visual observation and timing of the liquid-level rise when the valve was closed. The temperature of the condensate at the flow-measuring station was recorded so that the flow rate could be evaluated in mass units. Coolant flow rate was measured with a turbine-type flowmeter.

Test Procedure

Prior to the acquisition of the experimental data, noncondensable gases were removed from the test fluid and the test facility. The air content of the Freon-113 as delivered was 20 parts per million on a weight basis. The air content of the liquid in the boiler during the tests remained at this value, as indicated by samples withdrawn from the boiler. The boiler was isolated from the remaining portion of the loop and was filled under vacuum with 80 gallons of the liquid. The test-section side of the U-tube manometer system was purged with Freon-113 to ensure an all-liquid hydrostatic head between the pressure tap and the manometer fluid. The manometer valves were closed, and the test loop was evacuated to approximately 50 microns of mercury. The vacuum pump was then turned off. The boiler was opened to the test loop, and 10 to 15 percent of the boiler inventory (8.0 to 12.0 gal) was boiled off. This procedure allowed any residual noncondensable gases to collect in the receiver tank, which was the low-pressure region of the system. The pressure in the receiver tank rose to the saturation pressure corresponding to the temperature of the liquid (4.5 to 5.5 psia) and remained constant during a run.

Early in the program, the vapor flow rate was found constant, regardless of the liquid level in the boiler. Therefore, obtaining a data point consisted in adjusting the flow rate of the test fluid (by adjusting steam pressure to the boiler) at a given coolant flow rate and monitoring the pressures, temperatures, and flow rates. All data were taken after the facility reached a steady-state condition where the vapor-inlet and coolant-exit temperatures had not changed for a period of at least 15 minutes. Approximately 10 minutes were required to record all data. The most time consuming were the pressure readings from the manometers. Measuring the condensate flow rate required less than 1 minute. System pressures did not change when the quick shutoff valve of the condensate-flow-rate measuring station was closed; therefore, it was assumed that the flow remained constant throughout the flow-measuring operation.

When the boiler inventory was reduced to approximately 20 gallons, the tests were terminated. The liquid in the receiver tank was transferred to the boiler, and the process of loop evacuation and boiloff was repeated. It was felt that this mode of operation always kept the noncondensable gases away from the heat-transfer test region.

METHOD OF DATA ANALYSIS

The experimental data obtained in the tests are presented in table I. The experimental measurements (pressures, temperatures, and flow rates) were used to calculate local heat flux, local condensing heat-transfer coefficients, mean condensing heat-transfer coefficients, and overall friction-pressure losses. These computed values are also given in table I.

The local condensing heat-transfer coefficient h_{cl} was calculated from the following relation (see appendix B):

$$h_{cl} = \frac{q_i}{t_{vs} - t_{iw}} \tag{B2}$$

(Symbols are defined in appendix A, and the methods used to obtain the heat flux and temperatures in equation (B2) are given in appendix B.)

The overall friction-pressure loss was obtained from the following relation, which defines the static-pressure change in the condenser:

$$\Delta P_s = \Delta P_f + \Delta P_m + \Delta P_g$$

The momentum-pressure change ΔP_m was obtained from

$$\Delta P_m = \frac{K_2}{g_c}\left(\frac{V_{vi}^2}{v_{vi}} - \frac{V_{vo}^2}{v_{vo}}\right)$$

The exit vapor velocity V_{vo} was calculated from the equation of continuity by assuming that vapor alone occupied the cross section of the tube.

Table I shows that, for all runs, the vapor 0.02 foot upstream of the condenser inlet was in a superheated state. In addition, the first wall thermocouple (0.14 ft downstream of the condenser inlet) measured a temperature higher than the vapor saturation temperature. To locate the position in the condenser where condensation began, it was assumed that the inner-wall temperature had to be lower than the local vapor saturation temperature. The difference between the inner-wall temperature and the measured wall temperature, however, was very small (1.3° F maximum). Therefore, the location where the measured-wall, axial-temperature profile intersected the axial saturated-vapor-temperature profile was taken as the start of the condensing portion of the test section.

Local vapor flow rates were computed from local heat balances over a short increment of length (0.25 ft), as outlined in detail in appendix B. No correction was needed for heat loss from the outer jacket of the test section in calculating the heat balances; the total heat loss was measured and found to be negligible compared with the total heat load of the condenser. The cooling of the liquid condensate and the vapor core was taken into consideration when the heat balance was calculated. With low-latent-heat fluids flowing under conditions of large pressure drops, the reduction in the enthalpies of the condensate and the vapor are significant portions of the total heat rejected. The assumption was made that, over a given increment of length, the change in the temperature of the condensate and the vapor was the same as the change in the vapor saturation temperature. For the temperature range of tests described herein, the latent heat of the Freon-113 was of the order of 60 Btu per pound.

A possible error in the measured wall temperature caused by the fin effect of the thermocouple sheath was investigated. A simplified analysis (appendix B) showed that the wall temperatures were only slightly affected by the presence of the fin; therefore, the wall temperatures reported herein are uncorrected.

RESULTS AND DISCUSSION

Axial-Pressure and Temperature Profiles

The experimental data obtained from the tests, including static pressure, wall temperatures, and coolant temperatures, are listed in table I. Figure 4 shows a typical example of the axial-pressure and temperature profiles for a single run. Figure 4(a) illustrates the manner in which the static pressure decreases with increasing length. The measured static-pressure change between the inlet and outlet of the test section varied from 10.18 to 33.10 pounds per square inch for flow rates varying from 287 to 506 pounds per hour. The starting point for condensing is shown in figure 4(b) by the intersection of the vapor-saturation-temperature and wall-temperature profiles.

Overall Friction-Pressure Loss

For the computation of overall friction-pressure losses, several simplifying assumptions were made. These assumptions and the details of the calculations are given in appendix B. Friction-pressure losses varied from 13.34 to 48.45 pounds per square inch for the range of variables investigated. The overall friction-pressure loss for each run is listed in table I.

The friction pressure loss data are presented in figure 5. Water data from reference 2 are included in the figure. Figure 5(a) shows the overall friction pressure loss as a function of the flow parameter $(v_{vi}G^2/2g_cK_1)(L/D_i)$, which is used in ordinary single-phase pipe friction problems. The mass velocity G is the test-fluid total flow rate divided by the inside cross-sectional area of the condenser. The vapor specific volume v_{vi} is the value at the inlet to the test condenser based on the static pressure and temperature measured 0.02 foot upstream of the condenser inlet. The combination of specific-volume and mass-velocity terms of the flow parameter is proportional to the kinetic energy of the vapor at the condenser inlet. The length-to-diameter ratio is a constant for the Freon-113 data reported herein. This ratio is included to compare the Freon-113 data with the steam data of reference 2, in which the condensing length was variable. Figure 5(a) indicates that the Freon-113 and water friction-pressure loss is directly proportional to the flow parameter.

The data of figure 5(a) were used to compute friction factors (ratios of overall friction pressure loss to the flow parameter) for each run. The results are shown in figure 5(b), where the friction factor is plotted as a function of the vapor inlet Reynolds number. The computed friction factors for the Freon-113 data fall in the transition zone between the curve for smooth pipes and the curve designating the wholly rough zone. The data from reference 2 (water) fall within ±40 percent of the curve for smooth pipes. The friction factors plotted in figure 5(b) show no apparent dependence on inlet vapor Reynolds number over the Reynolds number range investigated. Information on the relative roughness of the flow passage during two-phase operation is unavailable, and therefore a quantitative discussion of the effect of this parameter cannot be made.

Local Heat Transfer

The axial variation of the heat flux is shown in figure 6(a) for two typical data runs. Local heat fluxes at the vapor-inlet end of the test section for all runs were in the range 51 000 to 120 000 Btu per hour per square foot. At the discharge end of the condenser, the local heat fluxes ranged from 7500 to 20 000 Btu per hour per square foot.

Local condensing heat-transfer coefficients were computed from the heat flux and the temperature difference between the saturated vapor and the inner wall. Some typical results are shown in figure 6(b). Values of the coefficient at the vapor-inlet end of the condenser ranged from 2000 to 3300 Btu per hour per square foot per $^{\circ}F$. The local coefficient varied with length and generally decreased in magnitude. The coefficients at the discharge end ranged from approximately 200 to 1200 Btu per hour per square foot per $^{\circ}F$.

Several methods were tried in an attempt to obtain a general correlation for Freon-113 and water data of previous work (ref. 1 and 2). The experimental heat-transfer data are presented in terms of the classical Nusselt condensing parameters (ref. 6) in figure 7. The ordinate of the figure is a combination of the heat-transfer coefficient and a grouping of liquid-film-property values. The abscissa is the condensate Reynolds number as defined in appendix B. The range of data from references 1 and 2, in which water was the test fluid, is shown for comparison. The Freon-113 data, similar to the water data, show an increase in the condensing parameter with increase in total flow at a given condensate Reynolds number. This trend corroborates the results of analyses reported in references 3 to 5. The analyses showed that an increase in the condensing heat-transfer coefficient can be expected with an increase in vapor velocity.

The Freon-113 data yield higher values for the condensing parameter than water, mainly because the thermal conductivity of liquid Freon-113 is an order of magnitude less. The trend of the data support the theoretical work reported in references 7 and 8, in which the effect of the liquid Prandtl number was considered. The theoretical results showed that larger values of the condensing parameter would be obtained with fluids of higher Prandtl number. Figure 7 shows that the Nusselt parameter does not give a general correlation for high-velocity condensing.

Another method of correlation is shown in figure 8, in which the local condensing heat-transfer coefficients for Freon-113 and water are plotted as functions of the local vapor flow rate. The figure indicates that the local coefficient is strongly dependent on vapor flow rate but that other parameters are needed to obtain a general correlation for the three sets of data.

A third method of correlation based on the work of Carpenter and Colburn (ref. 3) was also tried. Carpenter and Colburn performed a basic study of high-velocity condensing by using boundary-layer principles. Their model consisted of a high-velocity vapor core with a thin liquid film on the tube wall. An expression was derived to predict the local condensing heat-transfer coefficient as a function of the local frictional drag of the vapor, the effect of gravity on the condensate, and the momentum change of the vapor that is condensed and brought essentially to rest. The friction of the vapor was expressed in terms of a two-phase friction factor taken from the work reported in reference 9. Carpenter and Colburn extended their work (ref. 3) to obtain a simplified expression for the mean condensing heat-transfer coefficient for the entire condenser. The simplification was to use a friction factor based on one-component flow (vapor alone) and to use an average value of vapor mass velocity. The details regarding the evaluation of the average vapor mass velocity are presented in appendix B.

In the analysis of the Freon-113 data and the water data from references 1 and 2, the local heat-transfer coefficients correlated with a relation similar to the Carpenter-Colburn simplified expression for the mean condensing heat-transfer coefficient. The results are shown in figure 9. The ranges of test conditions for the Freon-113 work and the work of references 1 and 2 are adequately represented by the data shown. The abscissa of figure 9 contains local values of liquid and vapor properties as well as the local value of the vapor mass velocity based on the inner diameter of the tube. The friction factor expression $(\sqrt{f/8})$ that was in the original Carpenter-Colburn relation is not included. Local friction factors require a knowledge of local changes in the momentum of both liquid and vapor. The local changes in the momentum of the liquid could not be accurately calculated from the data. The data of figure 9 show a fair correlation in the Carpenter-Colburn terms and indicate that the condensing heat-transfer coefficient is a function of both the vapor velocity and the liquid-film thermal resistance.

Mean Heat-Transfer Coefficients

The mean condensing heat-transfer coefficients were evaluated and plotted as a function of the mean Carpenter-Colburn parameter as shown in figure 10. The Freon-113 data as well as the water data from references 1 and 2 are shown. The average vapor mass velocity (defined in appendix B) is used in the Carpenter-Colburn relation in figure 10. The friction factor f was evaluated from the friction-pressure-drop data of figure 5 of this report. An average friction factor of 0.018 was used to compute the Carpenter-Colburn parameter. Figure 10 shows that the Carpenter-Colburn relation, as indicated by the curve, correlates the data satisfactorily over the range investigated. Most of the data fell within ±30 percent of the Carpenter-Colburn relation with a few individual values of the experimental coefficient being one-half those predicted from the relation at the high end of the range.

SUMMARY OF RESULTS

The results of the investigation for local heat transfer and pressure drop for condensation of Freon-113 in vertical downflow within a 0.293-inch-inside-diameter tube may be summarized as follows:

1. The local condensing heat-transfer coefficient varied with length down the tube; high values occurred at the vapor-inlet end and from there generally decreased in magnitude. The local condensing coefficient ranged from 3300 Btu per hour per square foot per $^{\circ}F$ at the vapor inlet to 200 Btu per hour per square foot per $^{\circ}F$ at the condenser exit.

2. The local condensing coefficient was shown to be proportional to the local vapor flow rate. In addition, the local coefficients for Freon-113 and water were correlated satisfactorily in terms similar to the simplified relation derived by Carpenter and Colburn, which includes vapor mass velocity, liquid-to-vapor density ratio, liquid specific heat, and liquid Prandtl number. The friction factor that was in the original Carpenter-Colburn relation was not included.

3. The mean Carpenter-Colburn parameter satisfactorily correlated the mean condensing heat-transfer coefficient for both Freon-113 and water.

4. The friction-pressure drop for the Freon-113 was found to be a function of a flow parameter used in single-phase pipe-friction problems.

Lewis Research Center,
National Aeronautics and Space Administration,
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APPENDIX A

SYMBOLS

c_{pk} specific heat of coolant, Btu/(lb)(^oF)

$c_{p,L}$ specific heat of test liquid, Btu/(lb)(^oF)

$c_{p,v}$ specific heat of vapor, Btu/(lb)(^oF)

D_i inner diameter, ft

D_t diameter at which tube wall thermocouple was placed, ft

f friction factor,

$$\Delta P_f \Bigg/ \frac{v_{vi} G^2}{2 g_c K_1} \frac{L}{D_i}$$

G test-fluid total mass velocity, lb/(hr)(ft^2)

G_k coolant mass velocity, lb/(hr)(ft^2)

G_m test-fluid mean mass velocity (defined by eq. (B5)), lb/(hr)(ft^2)

G_{vl} local vapor mass velocity (based on inner diameter of tube), lb/(hr)(ft^2)

G_1 vapor mass velocity at entrance to condenser

G_2 vapor mass velocity at exit of condenser

g_c conversion factor, 4.17×10^8 (lb mass)(ft)/(hr^2)(lb force)

h_{cl} local condensing heat-transfer coefficient, Btu/(hr)(ft^2)(^oF)

h_{cm} mean condensing heat-transfer coefficient, Btu/(hr)(ft^2)(^oF)

h_{fg} latent heat of vaporization, Btu/lb

K_1 conversion factor, 144 $in.^2/ft^2$

K_2 conversion factor, $9\times10^4(sec^2)(ft^2)/(hr^2)(in.^2)$

k_f thermal conductivity of condensate film evaluated at t_f, Btu/(hr)(ft)(^oF)

k_{mw} mean thermal conductivity of condenser tube wall, Btu/(hr)(ft)(^oF)

Δm_{Fi} rate of liquid formed from condensation in arbitrary increment of length upstream of n^{th} increment, lb/hr

Δm_{Fn} rate of liquid formed from condensation in n^{th} increment of length, lb/hr

ΔP_f overall friction-pressure loss between stations at -0.02 and 8.06 ft, psi

ΔP_g change in pressure due to change in elevation between stations at -0.02 and 8.06 ft, psi

ΔP_m overall momentum pressure change between stations at -0.02 and 8.06 ft, psi

P_s static pressure of test fluid, psia

ΔP_s	overall static pressure change between stations at -0.02 and 8.06 ft, psi
P_{si}	static pressure of test fluid at inlet (measured at -0.02 ft), psia
Pr	Prandtl number of liquid condensate evaluated at t_f
Q	total rate of heat flow, Btu/hr
Q_{cn}	rate of heat transfer from condensation in n^{th} increment of length, Btu/hr
$Q_{\ell n}$	rate of heat transfer from cooling of liquid condensate in n^{th} increment of length, Btu/hr
Q_n	total rate of heat transfer in n^{th} increment of length, Btu/hr
Q_{vn}	rate of heat transfer from cooling of vapor in n^{th} increment of length, Btu/hr
q_i	heat flux based on inner diameter, Btu/(hr)(ft^2)
t_f	mean temperature of condensate film, $t_{vs} - \frac{3}{4}(t_{vs} - t_{iw})$, ^{o}F
t_{iw}	inner-wall temperature, ^{o}F
t_k	coolant temperature, ^{o}F
t_{ki}	coolant inlet temperature, ^{o}F
t_{ko}	coolant exit temperature, ^{o}F
Δt_{sat}	change in saturation temperature, ^{o}F
Δt_{sup}	vapor superheat at condenser inlet (measured at -0.02 ft), ^{o}F
t_{vi}	vapor temperature at inlet (measured at -0.02 ft), ^{o}F
t_{vs}	vapor saturation temperature, ^{o}F
t_w	measured wall temperature, ^{o}F
V_{vi}	inlet-vapor velocity evaluated at -0.02 ft, ft/sec
V_{vo}	exit-vapor velocity evaluated at 8.06 ft, ft/sec
v_{vi}	inlet-vapor specific volume, cu ft/lb
v_{vo}	exit-vapor specific volume, cu ft/lb
w	test-fluid total flow rate, lb/hr
w_{cl}	local-condensate flow rate, lb/hr
w_{ct}	total-condensate flow rate, lb/hr
w_k	coolant flow rate, lb/hr
w_{vl}	local vapor flow rate, lb/hr
x_e	exit quality, $1 - w_{ct}/w$
Γ_{cl}	local rate of flow of condensate per unit periphery, $w_{cl}/\pi D_i$, 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/ft^3
ρ_v	vapor density, lb/ft^3

APPENDIX B

DATA REDUCTION AND COMPUTATIONS

Local Heat Flux

The following assumptions were made for the evaluation of the local heat flux:

(1) There was no heat loss from the coolant to ambient surroundings.

(2) Heat flowed radially only.

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

The local heat flux at the inner wall of the condenser was calculated from the following relation:

$$q_i = \frac{w_k c_{p,k}}{\pi D_i} \frac{dt_k}{dL} \tag{B1}$$

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 evaluated graphically by finding the tangent to the curve at a particular location.

Condensing Heat-Transfer Coefficient

The local condensing heat-transfer coefficient was evaluated from

$$h_{cl} = \frac{q_i}{t_{vs} - t_{iw}} \tag{B2}$$

where t_{vs} is the saturation temperature corresponding to the measured pressure. All Freon-113 fluid properties were taken from reference 10. The term t_{iw} in equation (B2) is the inner wall temperature and was calculated from the following relation from reference 11 (p. 13) for the radial flow of heat in a cylinder:

$$Q = \frac{2\pi L k_{mw}(t_{iw} - t_w)}{\ell n \dfrac{D_t}{D_i}} \tag{B3}$$

Equation (B3) may be solved for the inner wall temperature in terms of the local heat flux and measured wall temperature t_w. The physical junction of the wall thermocouple was placed 0.034 inch from the inner wall. The diameter ratio of equation (B3) is the ratio of the diameter at which the wall temperature was measured to the inner diameter of the tube. (See the section entitled "Wall Temperature Error Analysis", p. 17). The wall mean thermal conductivity was assumed constant over the temperature range encountered in the tests, and a value of 226 Btu per hour per foot per $^{\circ}F$ was used for oxygen-free copper (ref. 12). Equation (B3) reduces to the following expression after insertion of the constants:

$$t_{iw} = t_w + 1.128\times10^{-5} q_i \tag{B4}$$

The mean condensing heat-transfer coefficients were determined by finding the area under the curve of the local coefficients plotted as a function of length and by dividing by the length. The condensing length that was used for this calculation was that between the condensing starting point and the discharge end of the test section. The curve of the local coefficients as a function of length was extrapolated from the station at 6.98 feet to 8.00 feet.

The variables in the Carpenter-Colburn parameter were evaluated in the following manner. In reference 3, Carpenter and Colburn derived an expression for the mean mass velocity G_m by assuming that the condensing rate was uniform. The results showed that the proper average value should be

$$G_m = \left(\frac{G_1^2 + G_1 G_2 + G_2^2}{3}\right)^{1/2} \tag{B5}$$

This procedure was followed in the work reported herein. For convenience, the fluid properties were evaluated at the vapor saturation temperature and mean film temperatures that existed at one-half the condensing length.

Local Flow Rates

The local vapor flow rate was determined by subtracting the local condensate flow rate from the total measured flow rate. Local condensate flow was evaluated by heat balances over small increments of length (0.25 ft). The heat-balance calculation was performed in the following manner. The total rate of heat transfer in the n^{th} increment Q_n is given by

$$Q_n = Q_{cn} + Q_{\ell n} + Q_{vn} \tag{B6}$$

where Q_n is known from the local heat-flux measurements on the coolant side of the test section, and Q_{cn} is the heat released by the condensation process in the increment and is given by

$$Q_{cn} = \Delta m_{Fn} h_{fg} \tag{B7}$$

The notation $Q_{\ell n}$ represents the decrease in the enthalpy of the liquid in the increment. An average value of the liquid flow rate in the increment was used since the flow varies with length. The $Q_{\ell n}$ is represented by

$$Q_{\ell n} = \left(\frac{1}{2} \Delta m_{Fn} + \sum_{1}^{n-1} \Delta m_{Fi}\right) c_{p,L}\, \Delta t_{sat} \tag{B8}$$

where the summation sign represents the quantity of liquid that was formed by condensation upstream of the increment. It was assumed that the change in temperature of the liquid from the beginning of the increment to the end of the increment was the same as the change in the saturation temperature. This temperature is represented by Δt_{sat} in equation (B8). The Q_{vn} of equation (B6) represents the decrease in the enthalpy of the vapor in the increment and is given by

$$Q_{vn} = \left[w - \left(\frac{1}{2} \Delta m_{Fn} + \sum_{1}^{n-1} \Delta m_{Fi}\right)\right] c_{p,v}\, \Delta t_{sat} \tag{B9}$$

where w is the measured total flow rate. An average value was used for the vapor flow rate in the increment. It was assumed that the change in the vapor temperature in the increment was the same as the change in the saturation temperature. The fluid properties in equations (B7) and (B9) were evaluated at the saturation temperature at the beginning of the increment. The liquid specific heat $c_{p,L}$ was evaluated at the mean film temperature at the beginning of the increment.

Substituting equations (B7), (B8), and (B9) into equation (B6), rearranging, and using the product of the measured local heat flux and incremental area for Q_n result in the following expression for the rate of liquid formation in the increment:

$$\Delta m_{Fn} = \frac{1.92\times10^{-2}\, q_i - \Delta t_{sat}\left[\left(c_{p,L} - c_{p,v}\right)\sum_{1}^{n-1} \Delta m_{Fi} + c_{p,v} w\right]}{h_{fg} + \frac{1}{2} \Delta t_{sat}\left(c_{p,L} - c_{p,v}\right)} \tag{B10}$$

The local condensate flow rate at a particular location is the summation of the incremental condensate flow rates to that point.

The local condensate flow rate is used to define a local condensate Reynolds number by

$$\frac{4\,\Gamma_{cl}}{\mu_f}$$

where Γ_{cl} is the local condensate flow rate divided by the inner circumference of the condenser tube. Thus,

$$\Gamma_{cl} = \frac{w_{cl}}{\pi D_i}$$

All local liquid condensate properties were evaluated at a mean temperature defined by the following equation (ref. 11, p. 330):

$$t_f = t_{vs} - \frac{3}{4}(t_{vs} - t_{iw})$$

The total condensate flow rate at the exit end of the condenser was used to calculate the exit quality of the test fluid by the following relation:

$$x_e = 1 - \frac{w_{ct}}{w}$$

Pressure Drop

The static-pressure change in the condenser is given by the following relation:

$$\Delta P_s = \Delta P_f + \Delta P_m + \Delta P_g \qquad \text{(B11)}$$

The static-pressure change was obtained from experimental data. The momentum-pressure change was evaluated by assuming the following: (1) the specific volume of the vapor at the discharge end of the test section was the saturation volume corresponding to the pressure at this location; (2) the vapor at the discharge end of the test section occupied the entire cross-sectional area of the condenser tube so that the vapor velocity at the exit could be evaluated from the continuity principle; and (3) the acceleration pressure drop contributed by the formation of the liquid was neglected. The momentum-pressure change was calculated from the following:

$$\Delta P_m = \frac{K_2}{g_c}\left(\frac{V_{vi}^2}{v_{vi}} - \frac{V_{vo}^2}{v_{vo}}\right) \tag{B12}$$

Hydrostatic-pressure changes ΔP_g of equation (B11) were neglected.

Wall Temperature Error Analysis

Two sources of uncertainty are present in the condenser-tube wall temperatures. The first is the uncertainty in the temperature rise from the actual thermocouple junction to the physical thermocouple junction. It can be seen from the wall thermocouple detail of figure 3 that the thermocouple wires are embedded in a pool of silver solder for a distance of approximately 1/32 inch. The actual junction is at the end of the stainless steel sheath where electrical continuity between the two thermocouple wires first occurs. The physical junction is the position where the thermocouple wires are joined and welded together. It is difficult to correct for the temperature difference between the actual and physical junction since the exact value of thermal conductivity of the silver-solder fill material is unknown. The location of the physical junction was arbitrarily chosen as the position where the wall temperature was measured.

The second uncertainty is the thermal error of the condenser-tube wall thermocouple. The thermal error is defined as the difference in temperature of the tube wall with and without the thermocouple. Reference 13 presents an analytical solution for the temperature distribution in a flat plate with a heat source or sink when the plate is surrounded on either side by fluids of different temperatures. The thermocouple junction was considered as a heat sink as a result of the fin effect of the thermocouple leads. The analytical model closely approximates the case encountered in the experimental work presented herein. Thermal errors were computed by using the method outlined in reference 13. The results showed that the measured tube-wall temperatures were within 1.0° F of the tube-wall temperatures without the thermocouple. The dominant factors in the equations were (1) the high value of the condenser tube-wall thermal conductivity (226 (Btu)(ft)/(hr)(ft^2)($^{\circ}$F)), (2) the high values of the coolant heat-transfer coefficients (664 to 1685 Btu/(hr)(ft^2)($^{\circ}$F)), and (3) the high values of condensing heat-transfer coefficients (200 to 3300 Btu/(hr)(ft^2)($^{\circ}$F)).

REFERENCES

1. Goodykoontz, Jack H.; Dorsch, Robert G.: Local Heat-Transfer Coefficients for Condensation of Steam in Vertical Downflow Within a 5/8-Inch-Diameter Tube. NASA TN D-3326, 1966.
2. Goodykoontz, Jack H.; and Dorsch, Robert G.: Local Heat-Transfer Coefficients and Static Pressures for Condensation of High-Velocity Steam Within a Tube. NASA TN D-3953, 1967.
3. Carpenter, E. F.; and Colburn, A. P.: The Effect of Vapor Velocity on Condensation Inside Tubes. Proceeding of General Discussion on Heat Transfer. Inst. Mech. Eng. and ASME, 1951, pp. 20-26.
4. Rohsenow, W. M.; Webber, J. H.; and Ling, A. T.: Effect of Vapor Velocity on Laminar and Turbulent-Film Condensation. ASME Trans., vol. 78, no. 8, Nov. 1956, pp. 1637-1643.
5. Dukler, A. E.: Dynamics of Vertical Falling Film Systems. Chem. Eng. Progr., vol. 55, no. 10, Oct. 1959, pp. 62-67.
6. Nusselt, W. C.: Die Oberflächenkondensation des Wasserdampfes. Ver. Deut. Ing. Seit., vol. 60, 1916, pp. 541-546, 569-575.
7. Colburn, Allan P.: Calculation of Condensation With a Portion of Condensate Layer in Turbulent Motion. Ind. Eng. Chem., vol. 26, no. 4, Apr. 1934, 432-434.
8. Seban, R. A.: Remarks on Film Condensation with Turbulent Flow. ASME Trans., vol. 76, no. 2, Feb. 1954, pp. 299-303.
9. Bergelin, Olaf P.; Kegel, P. K.; Carpenter, F. G.; and Gasley, Carl Jr.: Co-Current Gas-Liquid Flow. Heat Transfer and Fluid Mechanics Institute, ASME, 1949, pp. 19-28.
10. Anon.: Thermodynamic Properties of FREON-113. Bul. No. T-113A, E. I. DuPont De Nemours Company.
11. McAdams, William H.: Heat Transmission. Third ed., McGraw-Hill Book Co., Inc., 1954.
12. Materials in Design Engineering - Materials Selector Issue, vol. 56, no. 5, Mid-Oct. 1962.
13. Boelter, L. M. K.; Romie, F. E.; Guilbert, A. G.; and Miller, M. A.: An Investigation of Aircraft Heaters. Part 28 - Equations for Steady-State Temperature Distribution Caused by Thermal Sources in Flat Plates Applied to Calculation of Thermocouple Errors, Heat-Meter Corrections, and Heat Transfer by Pin-Fin Plates. NACA TN 1452, 1948.

TABLE I. - EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, °F	Coolant temperature, t_k, °F	Static pressure, P_s, psia	Vapor saturation temperature, t_{vs}, °F	Local heat flux, q_i, Btu/(hr)(ft^2)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft^2)(°F)	Conditions
				Run 1			
0.14	175	---	29.68	159.0	60 500	----	Vapor inlet flow, w, 287 lb/hr; vapor temperature at -0.02 ft, 198° F; vapor pressure at -0.02 ft, 29.78 psia; vapor superheat at -0.02 ft, 37° F; condensing starting point, 0.28 ft; vapor inlet velocity, 204 ft/sec; vapor pressure at 8.06 ft, 7.85 psia; vapor quality at 8.06 ft, 0.24; mean condensing coefficient, 1378 Btu/(hr)(ft^2)(°F); coolant flow rate, w_k, 452 lb/hr; coolant inlet temperature, 70° F; coolant exit temperature, 109° F; overall friction pressure loss, 28.00 psi
.56	131	---	-----	-----	55 700	2371	
.98	---	100	26.66	151.8	51 100	----	
1.32	126	---	-----	-----	47 800	2118	
1.98	119	92	23.02	142.4	41 900	1828	
2.65	115	---	-----	-----	36 500	1687	
2.98	---	86	19.86	133.9	34 100	----	
3.65	106	---	-----	-----	29 500	1367	
3.98	105	81	17.06	125.6	27 400	1373	
4.98	---	76	14.68	117.7	21 600	----	
5.31	96	---	-----	-----	19 800	1061	
5.98	92	73	12.56	109.6	16 500	925	
6.98	80	71	-----	-----	11 800	569	
7.56	---	70	9.50	95.3	9 360	----	
				Run 2			
0.14	174	---	32.35	164.6	61 900	----	Vapor inlet flow, w, 324 lb/hr; vapor temperature at -0.02 ft, 193° F; vapor pressure at -0.02 ft, 32.61 psia; vapor superheat at -0.02 ft, 28° F; condensing starting point, 0.24 ft; vapor inlet velocity, 208 ft/sec; vapor pressure at 8.06 ft, 8.52 psia; vapor quality at 8.06 ft, 0.24; mean condensing coefficient, 1392 Btu/(hr)(ft^2)(°F); coolant flow rate, w_k, 452 lb/hr; coolant inlet temperature, 70° F; coolant exit temperature, 113° F; overall friction pressure loss, 30.79 psi
.56	136	---	-----	-----	57 100	2307	
.98	---	103	29.05	157.5	52 600	----	
1.32	131	---	-----	-----	49 400	2140	
1.98	123	95	25.03	147.6	43 500	1765	
2.65	119	---	-----	-----	38 100	1755	
2.98	---	88	21.58	138.6	35 700	----	
3.65	110	---	-----	-----	31 100	1405	
3.98	108	83	18.48	129.9	29 000	1357	
4.98	---	79	15.89	121.8	23 300	----	
5.31	99	---	-----	-----	21 500	1054	
5.98	94	75	13.65	113.9	18 200	925	
6.98	82	72	-----	------	13 700	594	
7.56	---	71	10.27	99.3	11 300	----	
				Run 3			
0.14	189	---	38.56	175.4	------	----	Vapor inlet flow, w, 349 lb/hr; vapor temperature at -0.02 ft, 211° F; vapor pressure at -0.02 ft, 38.85 psia; vapor superheat at -0.02 ft, 35° F; condensing starting point, 0.35 ft; vapor inlet velocity, 192 ft/sec; vapor pressure at 8.06 ft, 13.53 psia; vapor quality at 8.06 ft, 0.40; mean condensing coefficient, 1609 Btu/(hr)(ft^2)(°F); coolant flow rate, w_k, 434 lb/hr; coolant inlet temperature, 95° F; coolant exit temperature, 133° F; overall friction pressure loss, 33.32 psi
.56	153	---	-----	-----	48 200	2559	
.98	---	126	35.35	169.3	------	----	
1.32	146	---	-----	-----	42 600	2089	
1.98	141	119	31.40	162.7	38 200	1840	
2.65	137	---	-----	-----	34 100	1686	
2.98	---	113	27.84	154.6	------	----	
3.65	129	---	-----	-----	28 600	1470	
3.98	128	108	24.52	146.3	26 900	1519	
4.98	---	103	21.59	138.6	------	----	
5.31	119	---	-----	-----	20 700	1264	
5.98	115	100	18.80	130.8	17 800	1139	
6.98	104	97	-----	-----	13 900	1282	
7.56	---	95	14.36	116.6	------	----	

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, °F	Coolant temperature, t_k, °F	Static pressure, P_s, psia	Vapor saturation temperature, t_{vs}, °F	Local heat flux, q_l, Btu/(hr)(ft^2)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft^2)(°F)	Conditions
				Run 4			
0.14	191	---	41.90	181.1	------	----	Vapor inlet flow, w, 437 lb/hr; vapor temperature at -0.02 ft, 215° F; vapor pressure at -0.02 ft, 42.29 psia; vapor superheat at -0.02 ft, 33° F; condensing starting point, 0.24 ft; vapor inlet velocity, 220 ft/sec; vapor pressure at 8.06 ft, 13.41 psia; vapor quality at 8.06 ft, 0.18; mean condensing coefficient, 1662 Btu/(hr)(ft^2)(°F); coolant flow rate, w_k, 533 lb/hr; coolant inlet temperature, 73° F; coolant exit temperature, 123° F; overall friction pressure loss, 40.08 psi
.56	149	---	-----	-----	78 600	2900	
.98	---	113	37.41	173.4	------	----	
1.32	142	---	-----	-----	68 200	2494	
1.98	133	103	32.07	164.1	60 200	2028	
2.65	129	---	-----	-----	53 000	1947	
2.98	---	95	27.44	153.7	------	----	
3.65	119	---	-----	-----	43 600	1621	
3.98	117	89	23.40	143.4	40 800	1593	
4.98	---	84	19.90	133.9	------	----	
5.31	107	---	-----	-----	31 000	1345	
5.98	101	79	16.71	124.5	26 700	1183	
6.98	87	76	-----	-----	20 800	878	
7.56	---	74	12.55	109.6	------	----	
				Run 5			
0.14	192	---	41.20	179.9	------	----	Vapor inlet flow, w, 430 lb/hr; vapor temperature at -0.02 ft, 213° F; vapor pressure at -0.02 ft, 41.58 psia; vapor superheat at -0.02 ft, 33° F; condensing starting point, 0.29 ft; vapor inlet velocity, 222 ft/sec; vapor pressure at 8.06 ft, 13.42 psia; vapor quality at 8.06 ft, 0.26; mean condensing coefficient, 1683 Btu/(hr)(ft^2)(°F); coolant flow rate, w_k, 527 lb/hr; coolant inlet temperature, 73° F; coolant exit temperature, 122° F; overall friction pressure loss, 38.06 psi
.56	149	---	-----	-----	74 500	2822	
.98	---	112	36.76	172.2	------	----	
1.32	140	---	-----	-----	65 200	2360	
1.98	133	103	31.52	162.9	58 000	2030	
2.65	128	---	-----	-----	51 600	1908	
2.98	---	95	26.94	152.4	------	----	
3.65	118	---	-----	-----	43 100	1631	
3.98	117	89	22.97	142.3	40 600	1685	
4.98	---	83	19.55	133.0	------	----	
5.31	106	---	-----	-----	31 600	1363	
5.98	100	79	16.55	124.0	27 700	1167	
6.98	86	75	-----	-----	22 400	900	
7.56	---	73	12.50	109.4	-----	----	
				Run 6			
0.14	193	---	38.18	174.7	------	----	Vapor inlet flow, w, 388 lb/hr; vapor temperature at -0.02 ft, 213° F; vapor pressure at -0.02 ft, 37.41 psia; vapor superheat at -0.02 ft, 39° F; condensing starting point, 0.34 ft; vapor inlet velocity, 223 ft/sec; vapor pressure at 8.06 ft, 13.39 psia; vapor quality at 8.06 ft, 0.23; mean condensing coefficient, 1609 1609 Btu/(hr)(ft^2)(°F); coolant flow rate, w_k, 524 lb/hr; coolant inlet temperature, 73° F; coolant exit temperature, 119° F; overall friction pressure loss, 33.72 psi
.56	144	---	-----	-----	71 900	2783	
.98	---	109	33.98	167.4	------	----	
1.32	136	---	-----	-----	62 300	2330	
1.98	129	100	29.09	157.6	55 000	1986	
2.65	125	---	-----	-----	48 400	1941	
2.98	---	93	24.89	147.3	------	----	
3.65	115	---	-----	-----	39 800	1571	
3.98	113	87	21.18	137.5	37 200	1590	
4.98	---	82	18.06	128.6	------	----	
5.31	103	---	-----	-----	28 100	1235	
5.98	97	78	15.54	120.7	24 100	1048	
6.98	84	75	-----	-----	18 600	748	
7.56	---	73	11.81	106.4	------	----	

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, °F	Coolant temperature, t_k, °F	Static pressure, P_s, psia	Vapor saturation temperature, t_{vs}, °F	Local heat flux, q_i, Btu/(hr)(ft²)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft²)(°F)	Conditions
Run 7							
0.14	191	---	38.01	174.4	------	----	Vapor inlet flow, w, 370 lb/hr;
.56	146	---	-----	-----	66 200	2714	vapor temperature at -0.02 ft, 216° F;
.98	---	113	34.08	167.5	------	----	vapor pressure at -0.02 ft, 38.12 psia;
1.32	138	---	-----	-----	57 900	2309	vapor superheat at -0.02 ft, 41° F;
1.98	131	105	29.45	158.4	51 400	1965	condensing starting point, 0.28 ft;
2.65	127	---	-----	-----	45 500	1886	vapor inlet velocity, 208 ft/sec;
2.98	---	98	25.41	148.6	------	----	vapor pressure at 8.06 ft, 11.74 psia;
3.65	119	---	-----	-----	37 700	1646	vapor quality at 8.06 ft, 0.24; mean
3.98	116	92	21.86	139.3	35 400	1596	condensing coefficient, 1598 Btu/(hr)(ft²)(°F);
4.98	---	88	18.81	130.8	------	----	coolant flow rate, w_k, 517 lb/hr; coolant
5.31	106	---	-----	-----	27 000	1269	inlet temperature, 79° F; coolant exit temper-
5.98	102	84	16.30	123.2	23 300	1108	ature, 122° F; overall friction pressure loss,
6.98	90	81	-----	-----	18 200	759	34.58 psi
7.56	---	79	11.74	109.4	------	----	
Run 8							
0.14	175	---	33.86	167.2	------	----	Vapor inlet flow, w, 335 lb/hr;
.56	134	---	-----	-----	65 700	2301	vapor temperature at -0.02 ft, 203° F;
.98	---	103	30.10	160.0	------	----	vapor pressure at -0.02 ft, 34.44 psia;
1.32	127	---	-----	-----	56 900	1982	vapor superheat at -0.02 ft, 35° F;
1.98	120	96	25.82	149.6	50 200	1774	condensing starting point, 0.24 ft;
2.65	116	---	-----	-----	44 000	1682	vapor inlet velocity, 207 ft/sec;
2.98	---	90	22.21	140.2	------	----	vapor pressure at 8.06 ft, 12.68 psia;
3.65	108	---	-----	-----	35 800	1411	vapor quality at 8.06 ft, 0.11; mean
3.98	106	85	19.15	131.8	33 400	1349	condensing coefficient, 1326 Btu/(hr)(ft²)(°F);
4.98	---	81	16.73	124.5	------	----	coolant flow rate, w_k, 595 lb/hr; coolant
5.31	98	---	-----	-----	24 400	1019	inlet temperature, 74° F; coolant exit temper-
5.98	94	78	14.68	117.7	20 300	865	ature, 112° F; overall friction pressure loss,
6.98	85	76	-----	-----	14 700	528	30.26 psi
7.56	---	75	13.09	111.7	------	----	
Run 9							
0.14	189	---	33.34	166.4	------	----	Vapor inlet flow, w, 379 lb/hr;
.56	136	---	-----	-----	75 300	2826	vapor temperature at -0.02 ft, 215° F;
.98	---	101	30.46	160.7	------	----	vapor pressure at -0.02 ft, 33.60 psia;
1.32	127	---	-----	-----	64 900	2228	vapor superheat at -0.02 ft, 48° F;
1.98	121	94	25.86	149.7	57 000	2034	condensing starting point, 0.34 ft;
2.65	116	---	-----	-----	49 700	2068	vapor inlet velocity, 245 ft/sec;
2.98	---	87	20.84	136.5	------	----	vapor pressure at 8.06 ft, 13.12 psia;
3.65	107	---	-----	-----	40 200	1743	vapor quality at 8.06 ft, 0.20; mean
3.98	105	82	18.11	128.8	37 400	1651	condensing coefficient, 1629 Btu/(hr)(ft²)(°F);
4.98	---	78	15.19	119.5	------	----	coolant flow rate, w_k, 635 lb/hr; coolant
5.31	96	---	-----	-----	27 100	1340	inlet temperature, 72° F; coolant exit temper-
5.98	92	75	13.40	113.0	22 500	1120	ature, 111° F; overall friction pressure loss,
6.98	80	73	-----	-----	16 100	633	31.28 psi
7.56	---	72	10.74	101.6	------	----	

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, °F	Coolant temperature, t_k, °F	Static pressure, P_s, psia	Vapor saturation temperature, t_{vs}, °F	Local heat flux, q_i, Btu/(hr)(ft²)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft²)(°F)	Conditions
Run 10							
0.14	194	---	40.40	178.5	------	----	Vapor inlet flow, w, 461 lb/hr;
.56	142	---	-----	-----	84 500	2736	vapor temperature at -0.02 ft, 213° F;
.98	---	104	35.39	169.3	------	----	vapor pressure at -0.02 ft, 40.74 psia;
1.32	133	---	-----	-----	73 700	2304	vapor superheat at -0.02 ft, 34° F;
1.98	124	96	29.60	158.8	65 400	1967	condensing starting point, 0.27 ft;
2.65	120	---	-----	-----	57 900	1978	vapor inlet velocity, 244 ft/sec;
2.98	---	89	24.68	146.7	------	----	vapor pressure at 8.06 ft, 13.50 psia;
3.65	110	---	-----	-----	48 000	1703	vapor quality at 8.06 ft, 0.24; mean
3.98	108	84	20.48	135.6	45 000	1717	condensing coefficient, 1699 Btu/(hr)(ft²)(°F);
4.98	---	79	17.06	125.6	------	----	coolant flow rate, w_k, 691 lb/hr; coolant
5.31	98	---	-----	-----	34 400	1439	inlet temperature, 72° F; coolant exit temper-
5.98	94	76	14.55	117.2	29 700	1299	ature, 115° F; overall friction pressure
6.98	81	73	-----	-----	23 300	853	loss, 39.44 psi
7.56	---	71	11.42	104.7	------	----	
Run 11							
0.14	173	---	30.90	160.0	------	----	Vapor inlet flow, w, 310 lb/hr;
.56	124	---	-----	-----	59 500	1980	vapor temperature at -0.02 ft, 193° F;
.98	---	95	25.72	149.4	------	----	vapor pressure at -0.02 ft, 29.46 psia;
1.32	117	---	-----	-----	52 100	1840	vapor superheat at -0.02 ft, 35° F;
1.98	111	90	21.71	138.9	46 300	1661	condensing starting point, 0.29 ft;
2.65	106	---	-----	-----	40 900	1589	vapor inlet velocity, 222 ft/sec;
2.98	---	85	18.36	129.5	------	----	vapor pressure at 8.06 ft, 12.83 psia;
3.65	99	---	-----	-----	33 700	1352	vapor quality at 8.06 ft, 0.10; mean
3.98	98	81	15.77	121.4	31 400	1370	condensing coefficient, 1392 Btu/(hr)(ft²)(°F);
4.98	---	78	-----	-----	------	----	coolant flow rate, w_k, 697 lb/hr; coolant
5.31	90	---	-----	-----	23 300	628	inlet temperature, 72° F; coolant exit temper-
5.98	87	75	12.35	108.8	19 500	940	ature, 103° F; overall friction pressure
6.98	79	74	-----	-----	14 300	540	loss, 25.33 psi
7.56	---	73	11.31	104.2	------	----	
Run 12							
0.14	175	---	33.31	161.8	------	----	Vapor inlet flow, w, 346 lb/hr;
.56	128	---	-----	-----	64 400	2150	vapor temperature at -0.02 ft, 194° F;
.98	---	97	28.30	155.8	------	----	vapor pressure at -0.02 ft, 31.45 psia;
1.32	120	---	-----	-----	56 600	1891	vapor superheat at -0.02 ft, 31° F;
1.98	114	91	23.10	142.6	50 500	1794	condensing starting point, 0.25 ft;
2.65	110	---	-----	-----	44 800	1788	vapor inlet velocity, 232 ft/sec;
2.98	---	86	19.52	132.9	------	----	vapor pressure at 8.06 ft, 13.53 psia;
3.65	101	---	-----	-----	37 200	1483	vapor quality at 8.06 ft, 0.09; mean
3.98	100	82	16.67	124.4	34 900	1491	condensing coefficient, 1542 Btu/(hr)(ft²)(°F);
4.98	---	78	-----	-----	------	----	coolant flow rate, w_k, 704 lb/hr; coolant
5.31	93	---	-----	-----	26 500	755	inlet temperature, 72° F; coolant exit temper-
5.98	90	76	12.96	111.2	22 600	1093	ature, 105° F; overall friction pressure
6.98	80	73	-----	-----	17 300	610	loss, 28.02 psi
7.56	---	72	12.01	107.3	------	----	

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, °F	Coolant temperature, t_k, °F	Static pressure, P_s, psia	Vapor saturation temperature, t_{vs}, °F	Local heat flux, q_l, Btu/(hr)(ft²)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft²)(°F)	Conditions
				Run 13			
0.14	188	---	41.99	181.2	------	----	Vapor inlet flow, w, 449 lb/hr;
.56	149	---	-----	-----	80 100	2957	vapor temperature at -0.02 ft, 208° F;
.98	---	109	37.31	173.2	------	----	vapor pressure at -0.02 ft, 42.36 psia;
1.32	139	---	-----	-----	70 200	2388	vapor superheat at -0.02 ft, 26° F;
1.98	132	100	31.76	163.4	62 600	2085	condensing starting point, 0.24 ft;
2.65	126	---	-----	-----	55 800	1900	vapor inlet velocity, 223 ft/sec;
2.98	---	91	27.02	152.6	------	----	vapor pressure at 8.06 ft, 11.36 psia;
3.65	117	---	-----	-----	47 000	1692	vapor quality at 8.06 ft, 0.16; mean
3.98	115	85	22.91	142.1	44 300	1685	condensing coefficient, 1673 Btu/(hr)(ft²)(°F);
4.98	---	79	19.36	132.4	------	----	coolant flow rate, w_k, 540 lb/hr; coolant
5.31	105	---	-----	-----	35 100	1454	inlet temperature, 67° F; coolant exit temper-
5.98	97	74	16.47	123.7	31 000	1215	ature, 121° F; overall friction pressure
6.98	83	70	-----	115.1	25 700	808	loss, 42.70 psi
7.56	---	68	12.61	109.8	------	----	
				Run 14			
0.14	177	---	34.34	167.9	------	----	Vapor inlet flow, w, 368 lb/hr;
.56	141	---	-----	-----	65 100	2810	vapor temperature at -0.02 ft, 202° F;
.98	---	104	31.15	162.2	------	----	vapor pressure at -0.02 ft, 34.97 psia;
1.32	133	---	-----	-----	56 400	2289	vapor superheat at -0.02 ft, 33° F;
1.98	125	95	26.63	151.7	49 800	1921	condensing starting point, 0.11 ft;
2.65	121	---	-----	-----	43 800	1932	vapor inlet velocity, 223 ft/sec;
2.98	---	88	22.74	141.7	------	----	vapor pressure at 8.06 ft, 9.96 psia;
3.65	112	---	-----	-----	36 000	1559	vapor quality at 8.06 ft, 0.22; mean
3.98	110	82	19.34	132.3	33 700	1593	condensing coefficient, 1560 Btu/(hr)(ft²)(°F);
4.98	---	78	16.49	123.8	------	----	coolant flow rate, w_k, 495 lb/hr; coolant
5.31	100	---	-----	-----	25 400	1226	inlet temperature, 69° F; coolant exit temper-
5.98	94	74	14.21	116.0	21 700	1034	ature, 115° F; overall friction pressure
6.98	82	71	-----	-----	16 800	668	loss, 33.92 psi
7.56	---	70	10.95	102.5	------	----	
				Run 15			
0.14	181	---	40.17	178.1	------	----	Vapor inlet flow, w, 450 lb/hr;
.56	149	---	-----	-----	76 700	3124	vapor temperature at -0.02 ft, 199° F;
.98	---	108	36.47	171.7	------	----	vapor pressure at -0.02 ft, 41.58 psia;
1.32	140	---	-----	-----	66 900	2446	vapor superheat at -0.02 ft, 18° F;
1.98	131	98	31.24	162.4	59 500	1984	condensing starting point, 0.11 ft;
2.65	127	---	-----	-----	52 900	1941	vapor inlet velocity, 224 ft/sec;
2.98	---	90	26.52	151.4	------	----	vapor pressure at 8.06 ft, 10.61 psia;
3.65	116	---	-----	-----	44 300	1634	vapor quality at 8.06 ft, 0.22; mean
3.98	115	83	22.42	140.8	41 700	1668	condensing coefficient, 1674 Btu/(hr)(ft²)(°F);
4.98	---	77	18.86	131.0	------	----	coolant flow rate, w_k, 495 lb/hr; coolant
5.31	103	---	-----	-----	32 800	1346	inlet temperature, 65° F; coolant exit temper-
5.98	96	72	16.00	122.2	29 000	1158	ature, 120° F; overall friction pressure
6.98	82	68	-----	-----	23 900	791	loss, 41.77 psi
7.56	---	66	12.14	107.9	------	----	

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, $^{\circ}$F	Coolant temperature, t_k, $^{\circ}$F	Static Pressure, P_s, psia	Vapor saturation temperature, t_{vs}, $^{\circ}$F	Local heat flux, q_l, Btu/(hr)(ft^2)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft^2)($^{\circ}$F)	Conditions
				Run 16			
0.14	204	---	41.48	180.3	-------	----	Vapor inlet flow, w, 506 lb/hr;
.56	134	---	-----	-----	111 000	2799	vapor temperature at -0.02 ft, 229° F;
.98	---	95	35.28	169.2	-------	----	vapor pressure at -0.02 ft, 41.92 psia;
1.32	123	---	-----	-----	95 600	2388	vapor superheat at -0.02 ft, 48° F;
1.98	115	89	28.49	156.2	83 700	2108	condensing starting point, 0.30 ft;
2.65	109	---	-----	-----	72 800	2000	vapor inlet velocity, 264 ft/sec;
2.98	---	83	23.06	142.5	-------	----	vapor pressure at 8.06 ft, 8.82 psia;
3.65	101	---	-----	-----	58 500	1775	vapor quality at 8.06 ft, 0.16; mean
3.98	99	79	18.89	131.1	54 200	1735	condensing coefficient, 1566 Btu/(hr)(ft^2)($^{\circ}$F);
4.98	---	75	15.65	121.1	-------	----	coolant flow rate, w_k, 1055 lb/hr; coolant
5.31	---	---	-----	-----	-------	----	inlet temperature, 70° F; coolant exit temper-
5.98	85	73	13.39	112.9	31 400	1170	ature, 105° F; overall friction pressure
6.98	---	71	-----	-----	-------	----	loss, 48.50 psi
7.56	---	70	10.43	100.0	-------	----	
				Run 17			
0.14	193	---	35.89	170.7	-------	----	Vapor inlet flow, w, 403 lb/hr;
.56	134	---	-----	-----	83 400	2683	vapor temperature at -0.02 ft, 215° F;
.98	---	101	31.34	162.6	-------	----	vapor pressure at -0.02 ft, 35.80 psia;
1.32	126	---	-----	-----	72 700	2282	vapor superheat at -0.02 ft, 44° F;
1.98	118	94	26.16	150.5	64 400	2071	condensing starting point, 0.34 ft;
2.65	113	---	-----	-----	56 800	1973	vapor inlet velocity, 244 ft/sec;
2.98	---	89	21.86	139.3	-------	----	vapor pressure at 8.06 ft, 12.75 psia;
3.65	105	---	-----	-----	46 600	1705	vapor quality at 8.06 ft, 0.12; mean
3.98	103	84	18.43	129.7	43 600	1681	condensing coefficient, 1654 Btu/(hr)(ft^2)($^{\circ}$F);
4.98	---	80	15.68	121.1	-------	----	coolant flow rate, w_k, 815 lb/hr; coolant
5.31	95	---	-----	-----	32 300	1381	inlet temperature, 74° F; coolant exit temper-
5.98	91	78	13.70	114.1	27 300	1192	ature, 110° F; overall friction pressure
6.98	82	75	-----	-----	20 200	811	loss, 34.75 psi
7.56	---	74	11.39	104.6	-------	----	
				Run 18			
0.14	193	---	32.33	164.5	------	----	Vapor inlet flow, w, 367 lb/hr;
.56	131	---	-----	-----	76 500	2766	vapor temperature at -0.02 ft, 215° F;
.98	---	99	28.29	155.7	------	----	vapor pressure at -0.02 ft, 31.19 psia;
1.32	122	---	-----	-----	67 000	2359	vapor superheat at -0.02 ft, 53° F;
1.98	116	93	23.76	144.4	59 600	2176	condensing starting point, 0.38 ft;
2.65	111	---	-----	-----	52 700	2063	vapor inlet velocity, 257 ft/sec;
2.98	---	88	20.05	134.4	------	----	vapor pressure at 8.06 ft, 13.35 psia;
3.65	103	---	-----	-----	43 600	1758	vapor quality at 8.06 ft, 0.05; mean
3.98	102	84	17.11	125.7	40 800	1745	condensing coefficient, 1652 Btu/(hr)(ft^2)($^{\circ}$F);
4.98	---	80	14.99	118.8	------	----	coolant flow rate, w_k, 805 lb/hr; coolant
5.31	94	---	-----	-----	30 600	1333	inlet temperature, 73° F; coolant exit temper-
5.98	90	77	13.75	114.3	25 900	1090	ature, 107° F; overall friction pressure
6.98	82	75	-----	-----	19 500	680	loss, 29.78 psi
7.56	---	74	12.95	111.2	------	----	

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, °F	Coolant temperature, t_k, °F	Static pressure, P_s, psia	Vapor saturation temperature, t_{vs}, °F	Local heat flux, q_l, Btu/(hr)(ft^2)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft^2)(°F)	Conditions
				Run 19			
0.14	183	---	36.93	172.5	------	----	Vapor inlet flow, w, 422 lb/hr;
.56	135	---	-----	-----	80 100	2504	vapor temperature at -0.02 ft, 206° F;
.98	---	103	32.28	164.4	------	----	vapor pressure at -0.02 ft, 37.24 psia;
1.32	128	---	-----	-----	70 200	2202	vapor superheat at -0.02 ft, 33° F;
1.98	120	96	27.06	152.7	62 500	1955	condensing starting point, 0.22 ft;
2.65	115	---	-----	-----	55 400	1912	vapor inlet velocity, 240 ft/sec;
2.98	---	91	22.66	141.4	------	----	vapor pressure at 8.06 ft, 13.89 psia;
3.65	106	---	-----	-----	45 900	1646	vapor quality at 8.06 ft, 0.09; mean
3.98	105	86	19.20	132.0	43 000	1624	condensing coefficient, 1574 Btu/(hr)(ft^2)(°F);
4.98	---	82	16.48	123.8	------	----	coolant flow rate, w_k, 799 lb/hr; coolant
5.31	96	---	-----	-----	32 400	1306	inlet temperature, 75° F; coolant exit temper-
5.98	93	79	14.59	117.4	27 700	1166	ature, 112° F; overall friction pressure
6.98	84	77	-----	-----	21 100	738	loss, 33.35 psi
7.56	---	76	13.47	113.2	------	----	
				Run 20			
0.14	187	---	39.47	177.0	------	----	Vapor inlet flow, w, 457 lb/hr;
.56	140	---	-----	-----	90 000	2855	vapor temperature at -0.02 ft, 207° F;
.98	---	105	34.45	168.1	------	----	vapor pressure at -0.02 ft, 39.88 psia;
1.32	131	---	-----	-----	78 400	2458	vapor superheat at -0.02 ft, 29° F;
1.98	123	98	28.90	157.2	69 500	2110	condensing starting point, 0.21 ft;
2.65	118	---	-----	-----	61 200	2048	vapor inlet velocity, 241 ft/sec;
2.98	---	92	24.16	145.4	------	----	vapor pressure at 8.06 ft, 14.97 psia;
3.65	109	---	-----	-----	50 400	1761	vapor quality at 8.06 ft, 0.06; mean
3.98	107	87	20.41	135.4	47 100	1718	condensing coefficient, 1682 Btu/(hr)(ft^2)(°F);
4.98	---	83	17.48	126.9	------	----	coolant flow rate, w_k, 802 lb/hr; coolant
5.31	98	---	-----	-----	35 200	1357	inlet temperature, 75° F; coolant exit temper-
5.98	95	79	15.56	120.8	29 800	1176	ature, 115° F; overall friction pressure
6.98	84	77	-----	-----	22 500	724	loss, 38.91 psi
7.56	---	76	14.49	117.1	------	----	
				Run 21			
0.14	194	---	41.75	180.8	------	----	Vapor inlet flow, w, 476 lb/hr;
.56	145	---	-----	-----	87 100	2832	vapor temperature at -0.02 ft, 217° F;
.98	---	110	36.80	172.3	------	----	vapor pressure at -0.02 ft, 42.21 psia;
1.32	136	---	-----	-----	76 400	2415	vapor superheat at -0.02 ft, 36° F;
1.98	128	103	31.04	161.9	68 000	2067	condensing starting point, 0.26 ft;
2.65	123	---	-----	-----	60 300	2021	vapor inlet velocity, 243 ft/sec;
2.98	---	97	26.18	150.6	------	----	vapor pressure at 8.06 ft, 11.12 psia;
3.65	114	---	-----	-----	50 100	1691	vapor quality at 8.06 ft, 0.22; mean
3.98	113	91	22.68	141.5	47 000	1675	condensing coefficient, 1725 Btu/(hr)(ft^2)(°F);
4.98	---	87	18.89	131.1	------	----	coolant flow rate, w_k, 759 lb/hr; coolant
5.31	103	---	-----	-----	35 800	1446	inlet temperature, 79° F; coolant exit temper-
5.98	98	83	16.35	123.3	30 800	1275	ature, 120° F; overall friction pressure
6.98	88	81	-----	-----	23 900	921	loss, 43.59 psi
7.56	---	79	12.73	110.3	------	----	

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, ^oF	Coolant temperature, t_k, ^oF	Static pressure, P_s, psia	Vapor saturation temperature, t_{vs}, ^oF	Local heat flux, q_i, Btu/(hr)(ft^2)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft^2)(^oF)	Conditions
Run 22							
0.14	195	---	38.50	175.3	------	----	Vapor inlet flow, w, 417 lb/hr;
.56	140	---	-----	-----	78 500	2644	vapor temperature at -0.02 ft, 217^o F;
.98	---	107	33.88	167.2	------	----	vapor pressure at -0.02 ft, 38.79 psia;
1.32	131	---	-----	-----	68 400	2191	vapor superheat at -0.02 ft, 42^o F;
1.98	125	100	28.62	156.5	60 500	1975	condensing starting point, 0.30 ft;
2.65	120	---	-----	-----	53 300	1881	vapor inlet velocity, 233 ft/sec;
2.98	---	94	24.14	145.3	------	----	vapor pressure at 8.06 ft, 10.62 psia;
3.65	112	---	-----	-----	43 600	1649	vapor quality at 8.06 ft, 0.23; mean
3.98	109	90	20.53	135.7	40 700	1603	condensing coefficient, 1577 Btu/(hr)(ft^2)(^{o}F);
4.98	---	86	17.46	126.8	------	----	coolant flow rate, w_k, 739 lb/hr; coolant
5.31	101	---	-----	-----	30 000	1312	inlet temperature, 79^o F; coolant exit temper-
5.98	96	83	15.24	119.7	25 100	1082	ature, 116^o F; overall friction pressure
6.98	87	81	-----	-----	18 400	750	loss, 38.52 psi
7.56	---	80	11.99	107.2	------	----	
Run 23							
0.14	187	---	43.79	184.1	------	----	Vapor inlet flow, w, 470 lb/hr;
.56	150	---	-----	-----	82 700	2880	vapor temperature at -0.02 ft, 223^o F;
.98	---	113	38.70	175.6	------	----	vapor pressure at -0.02 ft, 44.20 psia;
1.32	141	---	-----	-----	72 500	2448	vapor superheat at -0.02 ft, 38^o F;
1.98	133	104	32.71	165.2	64 500	2073	condensing starting point, 0.17 ft;
2.65	129	---	-----	-----	57 300	2029	vapor inlet velocity, 231 ft/sec;
2.98	---	97	27.63	154.1	------	----	vapor pressure at 8.06 ft, no data ob-
3.65	118	---	-----	-----	47 900	1724	tained; vapor quality at 8.06 ft, 0.20;
3.98	116	91	23.45	143.5	45 100	1713	mean condensing coefficient, 1746 Btu/
4.98	---	85	19.87	133.9	------	----	(hr)(ft^2)(^{o}F); coolant flow rate, w_k,
5.31	106	---	-----	-----	35 000	1444	612 lb/hr; coolant inlet temperature, 76^o F;
5.98	101	81	16.59	124.1	30 500	1354	coolant exit temperature, 124^o F; overall
6.98	87	78	-----	-----	24 500	882	friction pressure loss, 43.60 psi
7.56	---	76	12.97	111.3	------	----	
Run 24							
0.14	198	---	39.90	177.7	------	----	Vapor inlet flow, w, 431 lb/hr;
.56	145	---	-----	-----	76 300	2820	vapor temperature at -0.02 ft, 222^o F;
.98	---	109	35.23	169.1	------	----	vapor pressure at -0.02 ft, 40.31 psia;
1.32	136	---	-----	-----	66 800	2357	vapor superheat at -0.02 ft, 44^o F;
1.98	129	101	29.75	159.1	59 600	2043	condenser starting point, 0.31 ft;
2.65	124	---	-----	-----	52 900	1964	vapor inlet velocity, 233 ft/sec;
2.98	---	94	25.22	148.1	------	----	vapor pressure at 8.06 ft, 10.26 psia;
3.65	115	---	-----	-----	44 200	1738	vapor quality at 8.06 ft, 0.25; mean
3.98	113	88	21.32	137.9	41 600	1699	condensing coefficient, 1706 Btu/(hr)(ft^2)(^{o}F);
4.98	---	84	18.02	128.5	------	----	coolant flow rate, w_k, 612 lb/hr; coolant
5.31	103	---	-----	-----	32 100	1474	inlet temperature, 75^o F; coolant exit temper-
5.98	97	80	15.44	120.3	28 000	1239	ature, 119^o F; overall friction pressure
6.98	85	77	-----	-----	22 300	854	loss, 40.25 psi
7.56	---	75	11.90	106.8	------	----	

TABLE I. - Continued. EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, °F	Coolant temperature, t_k, °F	Static pressure, P_s, psia	Vapor saturation temperature, t_{vs}, °F	Local heat flux, q_l, Btu/(hr)(ft^2)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft^2)(°F)	Conditions
Run 25							
0.14	183	---	32.22	164.3	------	----	Vapor inlet flow, w, 361 lb/hr;
.56	131	---	-----	-----	75 200	2669	vapor temperature at -0.02 ft, 204° F;
.98	---	95	28.21	155.5	------	----	vapor pressure at -0.02 ft, 32.63 psia;
1.32	122	---	-----	-----	64 500	2264	vapor superheat at -0.02 ft, 39° F;
1.98	115	87	23.65	144.1	56 400	1977	condenser starting point, 0.30 ft;
2.65	110	---	-----	-----	49 100	1885	vapor inlet velocity, 236 ft/sec;
2.98	---	81	19.79	133.6	------	----	vapor pressure at 8.06 ft, 7.86 psia;
3.65	101	---	-----	-----	39 400	1537	vapor quality at 8.06 ft, 0.19; mean
3.98	100	77	16.62	124.2	36 500	1532	condensing coefficient, 1518 Btu/(hr)(ft^2)(°F);
4.98	---	73	13.96	115.1	------	----	coolant flow rate, w_k, 652 lb/hr; coolant
5.31	90	---	-----	-----	26 200	1197	inlet temperature, 66° F; coolant exit temper-
5.98	85	70	12.14	107.9	21 600	972	ature, 104° F; overall friction pressure
6.98	75	68	-----	-----	15 200	638	loss, 34.37 psi
7.56	---	67	9.34	94.4	------	----	
Run 26							
0.14	180	---	26.60	151.6	------	----	Vapor inlet flow, w, 295 lb/hr;
.56	121	---	-----	-----	61 900	2439	vapor temperature at -0.02 ft, 203° F;
.98	---	89	23.35	143.3	------	----	vapor pressure at -0.02 ft, 26.92 psia;
1.32	113	---	-----	-----	53 400	2076	vapor superheat at -0.02 ft, 50° F;
1.98	106	83	19.63	133.2	46 700	1802	condenser starting point, 0.36 ft;
2.65	102	---	-----	-----	40 700	1702	vapor inlet velocity, 234 ft/sec;
2.98	---	78	16.53	123.9	------	----	vapor pressure at 8.06 ft, 6.82 psia;
3.65	94	---	-----	-----	32 600	1417	vapor quality at 8.06 ft, 0.17; mean
3.98	93	74	14.01	115.3	30 200	1398	condensing coefficient, 1397 Btu/(hr)(ft^2)(°F);
4.98	---	71	11.95	107.1	------	----	coolant flow rate, w_k, 655 lb/hr; coolant
5.31	85	---	-----	-----	21 300	1093	inlet temperature, 63° F; coolant exit temper-
5.98	81	69	10.48	100.3	17 300	919	ature, 97° F; overall friction pressure
6.98	73	67	-----	92.0	11 700	628	loss, 28.10 psi
7.56	---	66	8.06	87.1	------	----	
Run 27							
0.14	181	---	39.70	177.3	------	----	Vapor inlet flow, w, 470 lb/hr;
.56	138	---	-----	-----	87 700	2625	vapor temperature at -0.02 ft, 216° F;
.98	---	96	34.42	168.1	------	----	vapor pressure at -0.02 ft, 40.11 psia;
1.32	128	---	-----	-----	75 500	2165	vapor superheat at -0.02 ft, 38° F;
1.98	119	88	28.39	156.0	66 300	1866	condenser starting point, 0.16 ft;
2.65	114	---	-----	-----	58 100	1801	vapor inlet velocity, 254 ft/sec;
2.98	---	81	23.36	143.3	------	----	vapor pressure at 8.06 ft, 8.77 psia;
3.65	104	---	-----	-----	47 400	1542	vapor quality at 8.06 ft, 0.21; mean
3.98	102	75	19.35	132.4	44 200	1512	condensing coefficient, 1513 Btu/(hr)(ft^2)(°F);
4.98	---	71	15.92	122.0	------	----	coolant flow rate, w_k, 656 lb/hr; coolant
5.31	92	---	-----	-----	33 100	1237	inlet temperature, 62° F; coolant exit temper-
5.98	86	67	13.46	113.2	28 200	1056	ature, 107° F; overall friction pressure
6.98	73	64	-----	104.4	21 700	703	loss, 43.94 psi
7.56	---	63	10.31	99.5	------	----	

TABLE I. - Concluded. EXPERIMENTAL AND COMPUTED DATA

Location, ft	Measured wall temperature, t_w, °F	Coolant temperature, t_k, °F	Static pressure, P_s, psia	Vapor saturation temperature, t_{vs}, °F	Local heat flux, q_l, Btu/(hr)(ft^2)	Local condensing coefficient, h_{cl}, Btu/(hr)(ft^2)(°F)	Conditions
				Run 28			
0.14	190	---	36.53	171.8	------	----	Vapor inlet flow, w, 438 lb/hr;
.56	132	---	-----	-----	85 500	2529	vapor temperature at -0.02 ft, 217° F;
.98	---	93	31.51	162.9	------	----	vapor pressure at -0.02 ft, 36.97 psia;
1.32	123	---	-----	-----	73 200	2115	vapor superheat at -0.02 ft, 44° F;
1.98	114	85	25.83	149.7	63 800	1858	condensing starting point, 0.28 ft;
2.65	109	---	-----	-----	55 400	1769	vapor inlet velocity, 257 ft/sec;
2.98	---	78	21.15	137.4	------	----	vapor pressure at 8.06 ft, 8.03 psia;
3.65	99	---	-----	-----	44 500	1472	vapor quality at 8.06 ft, 0.26; mean
3.98	97	73	17.45	126.8	41 300	1450	condensing coefficient, 1454 Btu/(hr)(ft^2)(°F);
4.98	---	69	14.39	116.7	------	----	coolant flow rate, w_k, 701 lb/hr; coolant
5.31	87	---	-----	-----	29 800	1153	inlet temperature, 62° F; coolant exit temper-
5.98	83	66	12.30	108.5	24 700	974	ature, 103° F; overall friction pressure
6.98	71	64	-----	-----	17 700	641	loss, 39.64 psi
7.56	---	63	9.43	94.9	------	----	
				Run 29			
0.14	194	---	35.68	170.3	------	----	Vapor inlet flow, w, 417 lb/hr;
.56	134	---	-----	-----	81 800	2654	vapor temperature at -0.02 ft, 217° F;
.98	---	94	30.97	161.8	------	----	vapor pressure at -0.02 ft, 36.71 psia;
1.32	125	---	-----	-----	69 800	2226	vapor superheat at -0.02 ft, 45° F;
1.98	117	86	25.64	149.2	60 700	1927	condensing starting point, 0.34 ft;
2.65	111	---	-----	-----	52 600	1805	vapor inlet velocity, 245 ft/sec;
2.98	---	79	21.13	137.3	------	----	vapor pressure at 8.06 ft, 8.05 psia;
3.65	102	---	-----	-----	42 200	1509	vapor quality at 8.06 ft, 0.26; mean
3.98	100	73	17.49	126.9	39 100	1474	condensing coefficient, 1486 Btu/(hr)(ft^2)(°F);
4.98	---	69	14.51	117.1	------	----	coolant flow rate, w_k, 616 lb/hr; coolant
5.31	90	---	-----	-----	28 300	1156	inlet temperature, 62° F; coolant exit temper-
5.98	85	66	12.60	109.8	23 500	959	ature, 105° F; overall friction pressure
6.98	71	63	-----	-----	17 100	600	loss, 38.26 psi
7.56	---	62	9.50	95.3	------	----	

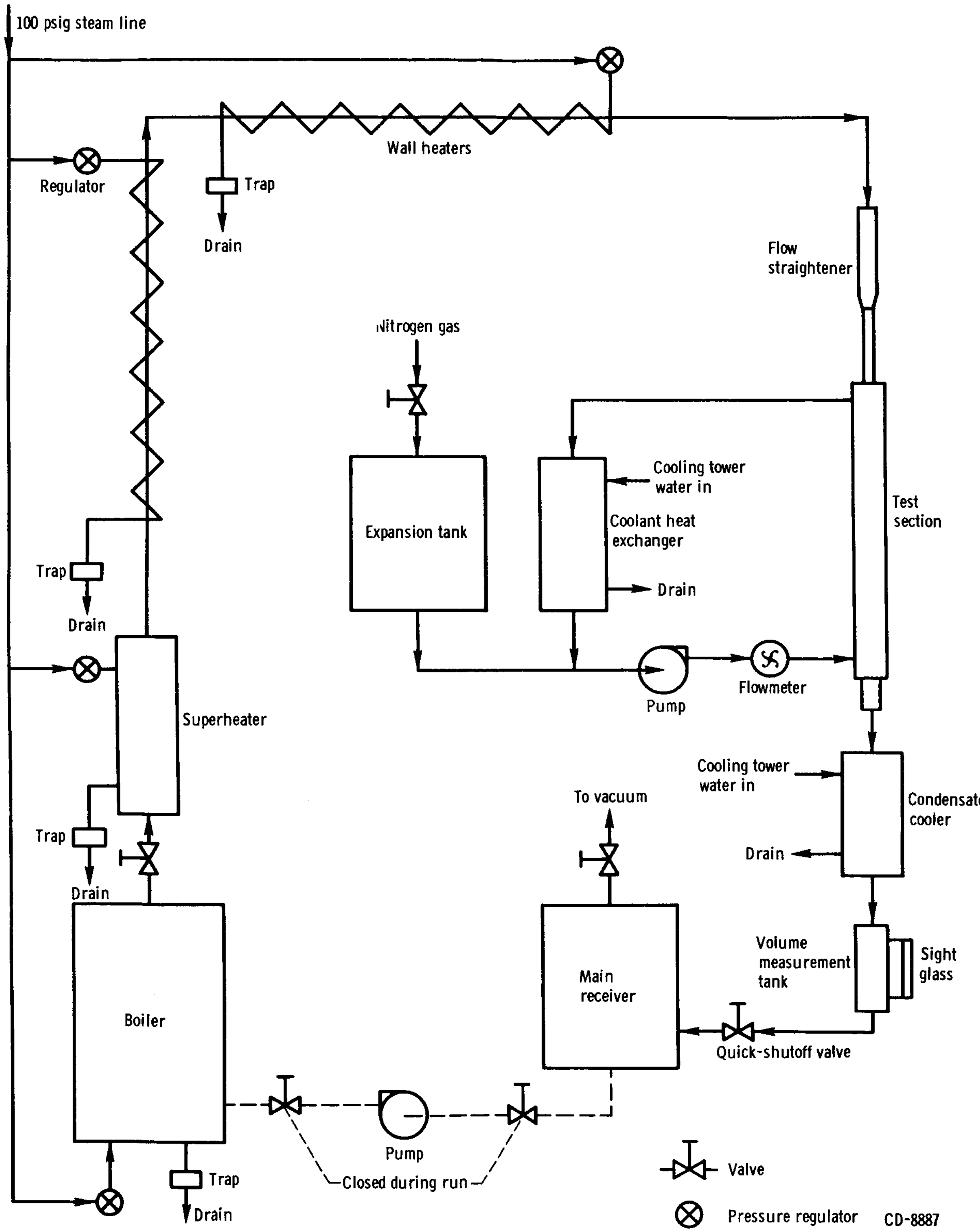


Figure 1. - Schematic drawing of single-tube condenser apparatus.

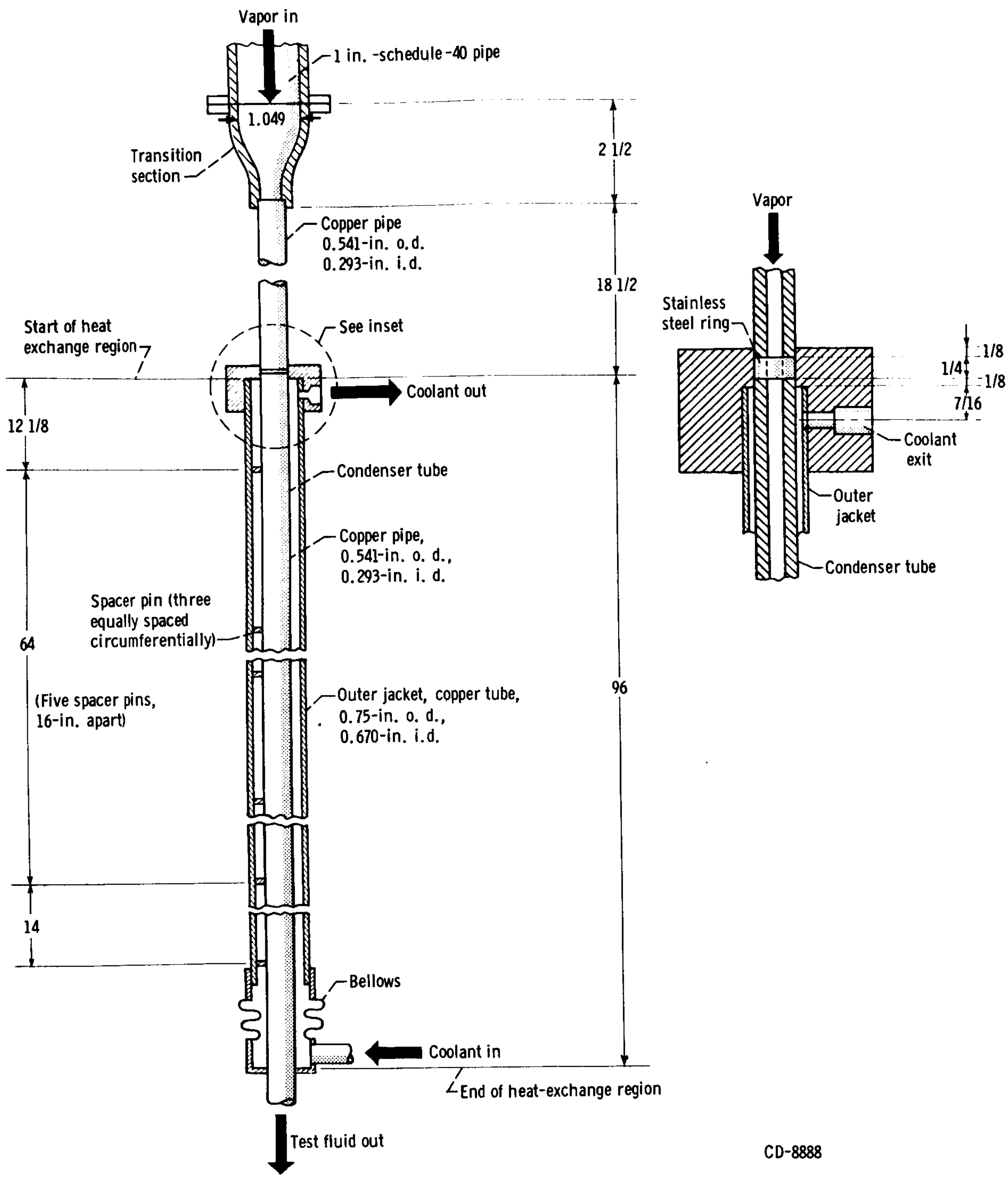


Figure 2. - Single-tube condenser test section. (Dimensions are in inches.)

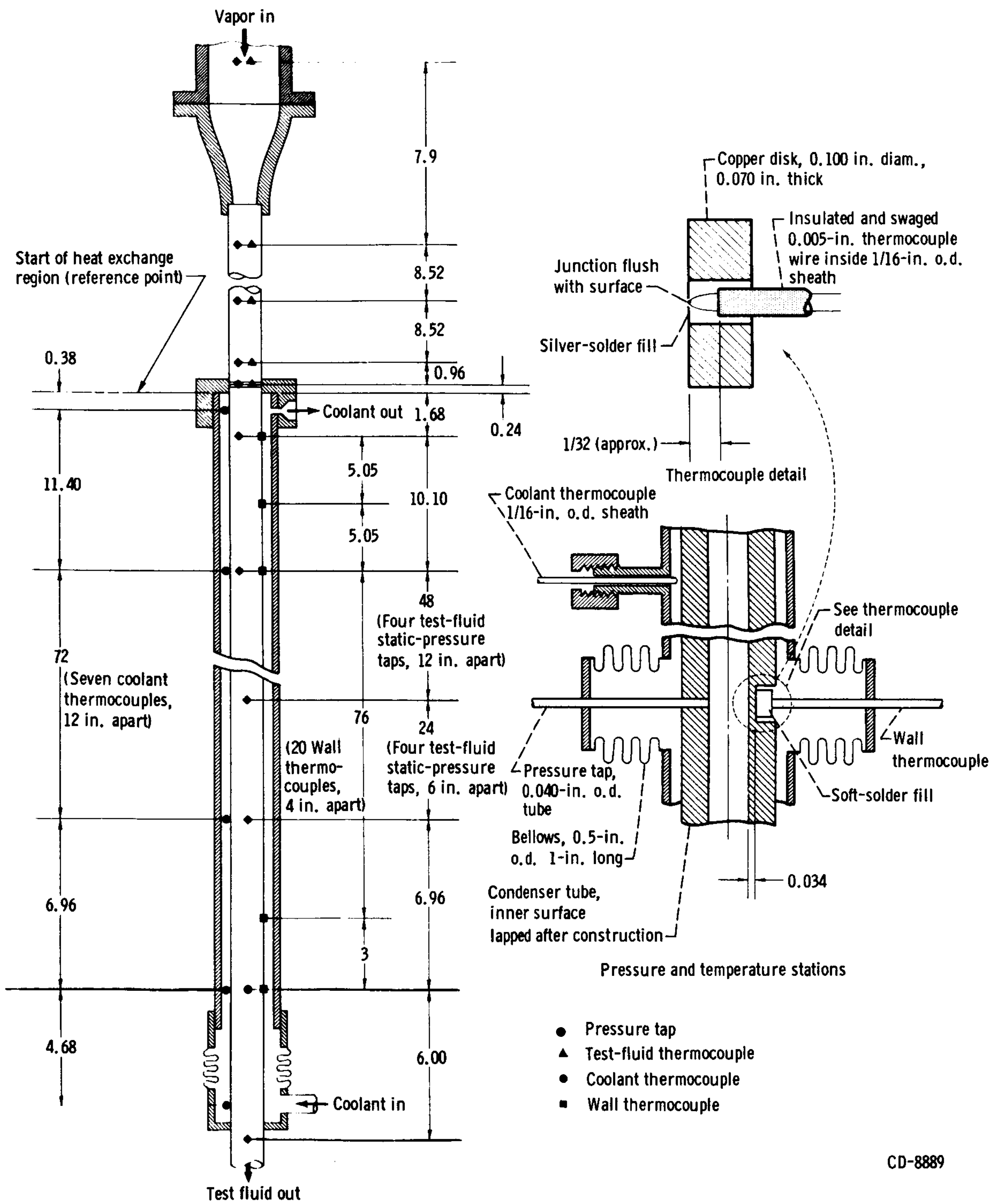
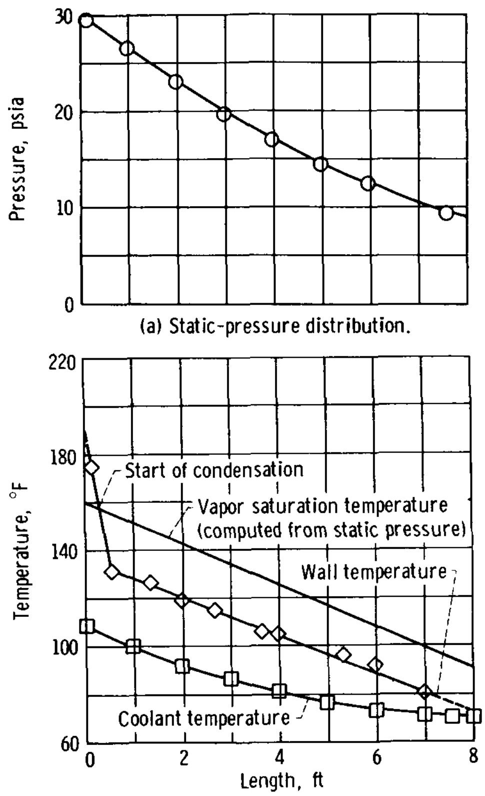


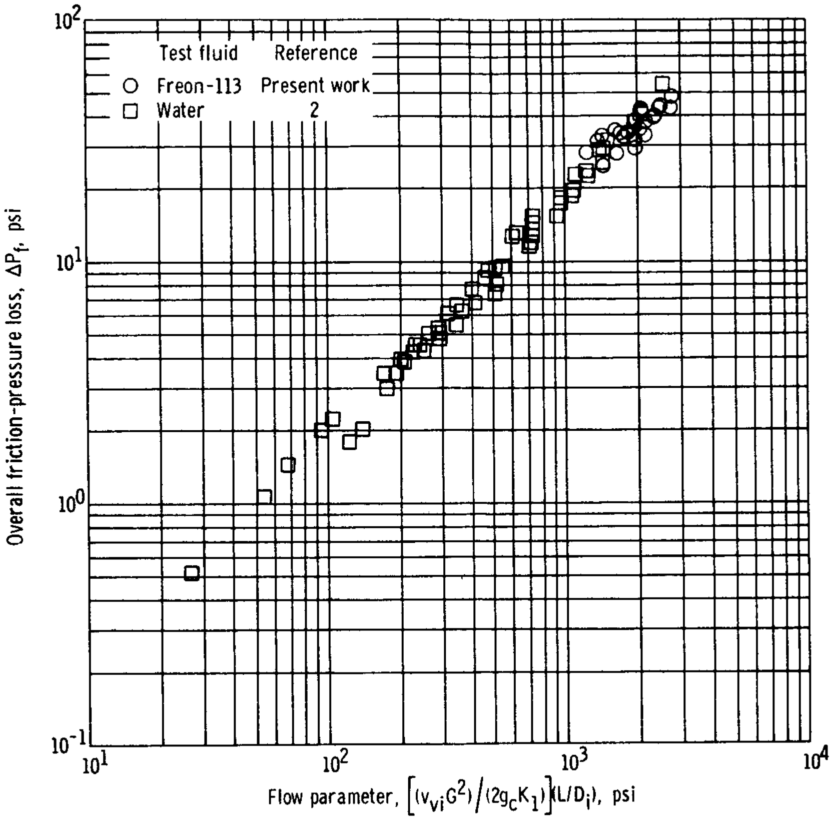
Figure 3. - Test-section instrumentation. (Dimensions are in inches.)



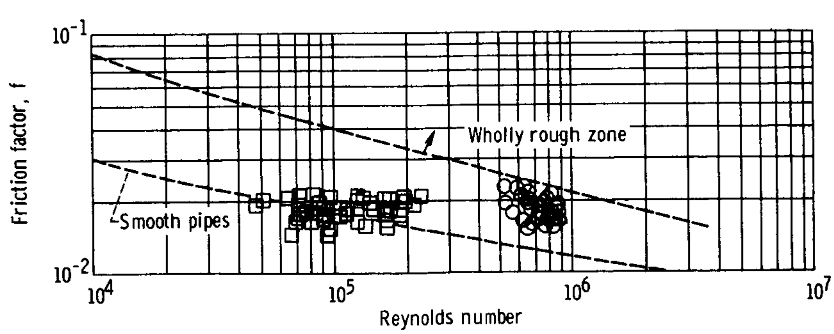
(a) Static-pressure distribution.

(b) Temperature distribution.

Figure 4. - Axial-pressure and temperature distributions for single-tube Freon-113 condenser (run 1). Freon-113 flow rate, 287 pounds per hour; inlet superheat, 37° F; exit quality, 0.24; coolant flow rate, 452 pounds per hour.

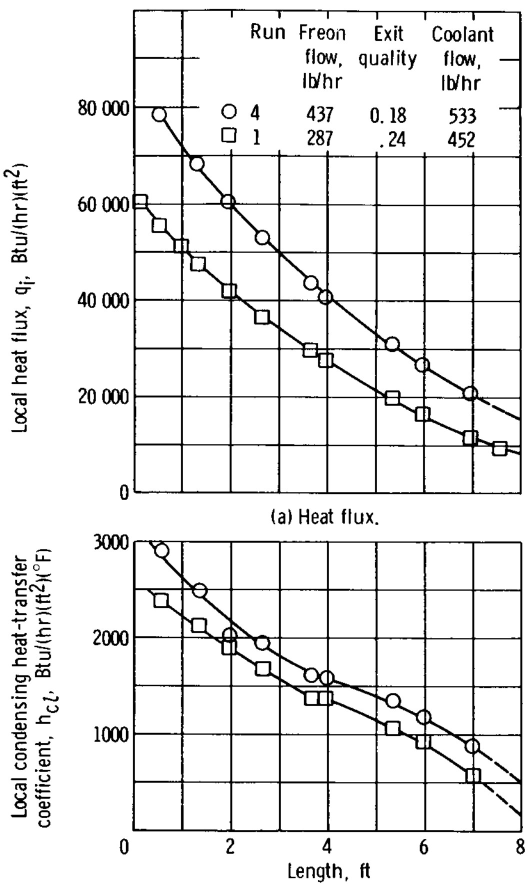


(a) Overall friction-pressure loss as function of flow parameter.



(b) Friction factor as function of inlet vapor Reynolds number.

Figure 5. - Friction-pressure loss data.



(b) Condensing heat-transfer coefficient.

Figure 6. - Local Heat flux and local condensing heat-transfer coefficient as functions of length.

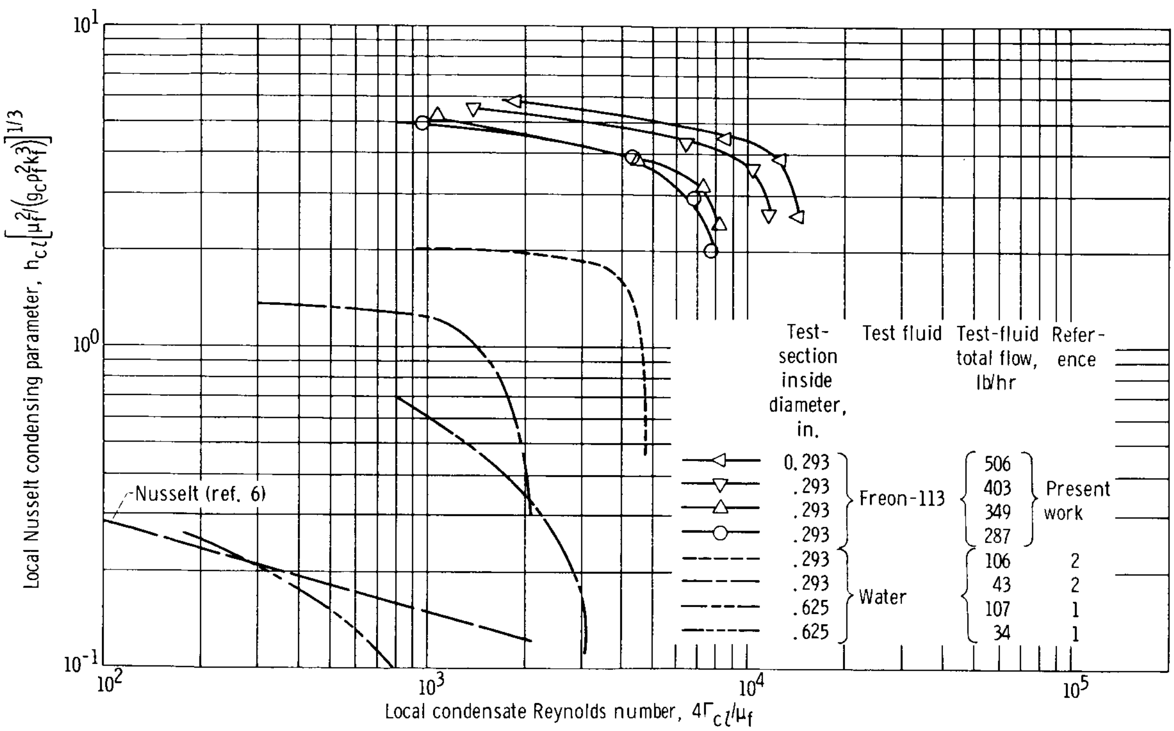


Figure 7. - Local Nusselt condensing parameter as function of local condensate Reynolds number.

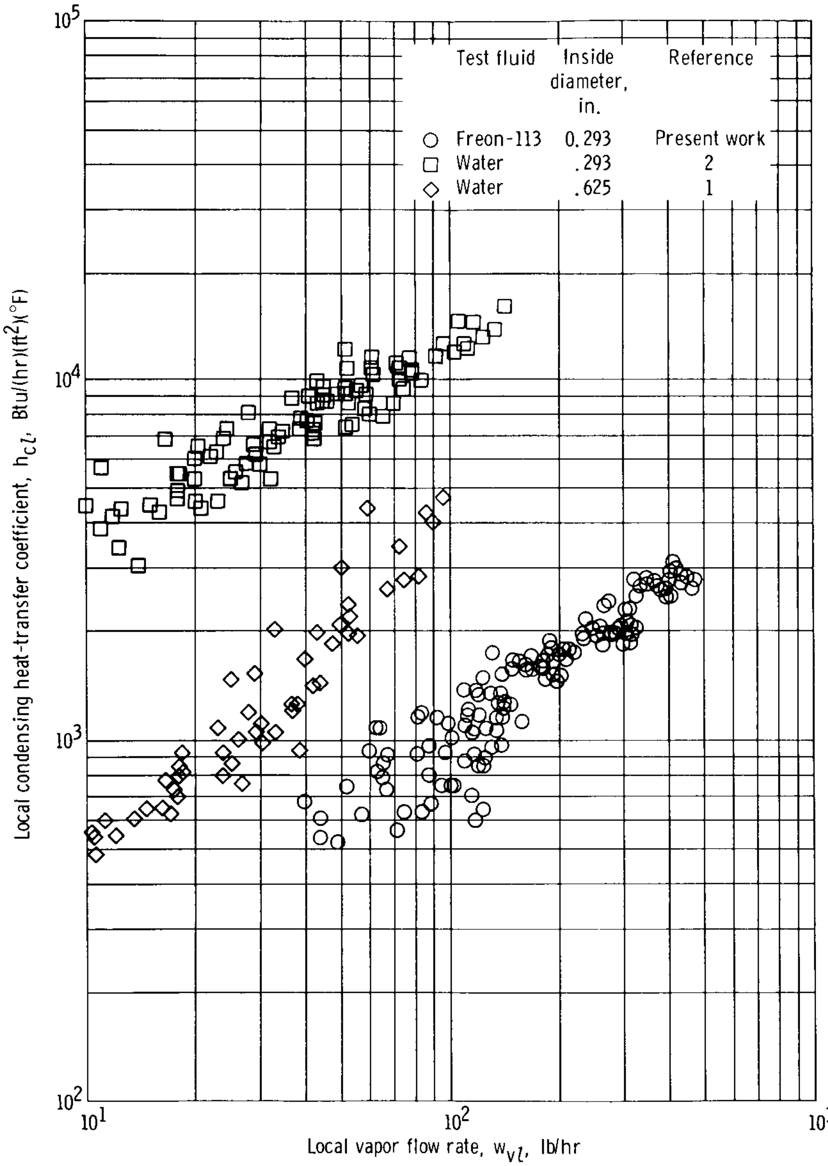


Figure 8. - Local condensing heat-transfer coefficient as function of local vapor flow rate.

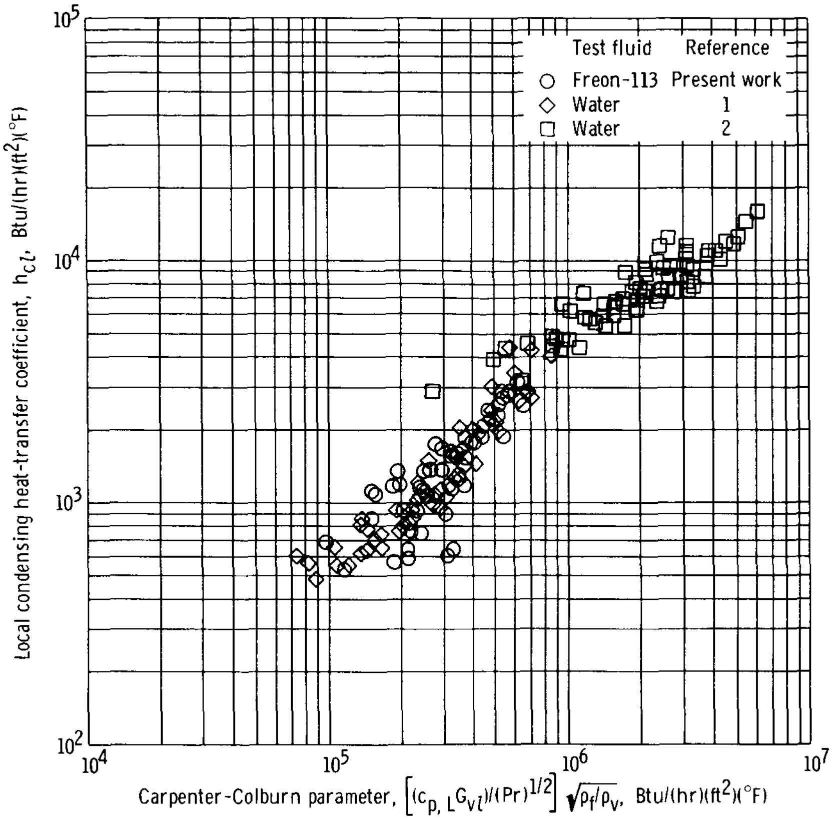


Figure 9. - Local condensing heat-transfer coefficient as function of Carpenter-Colburn parameter at local values.

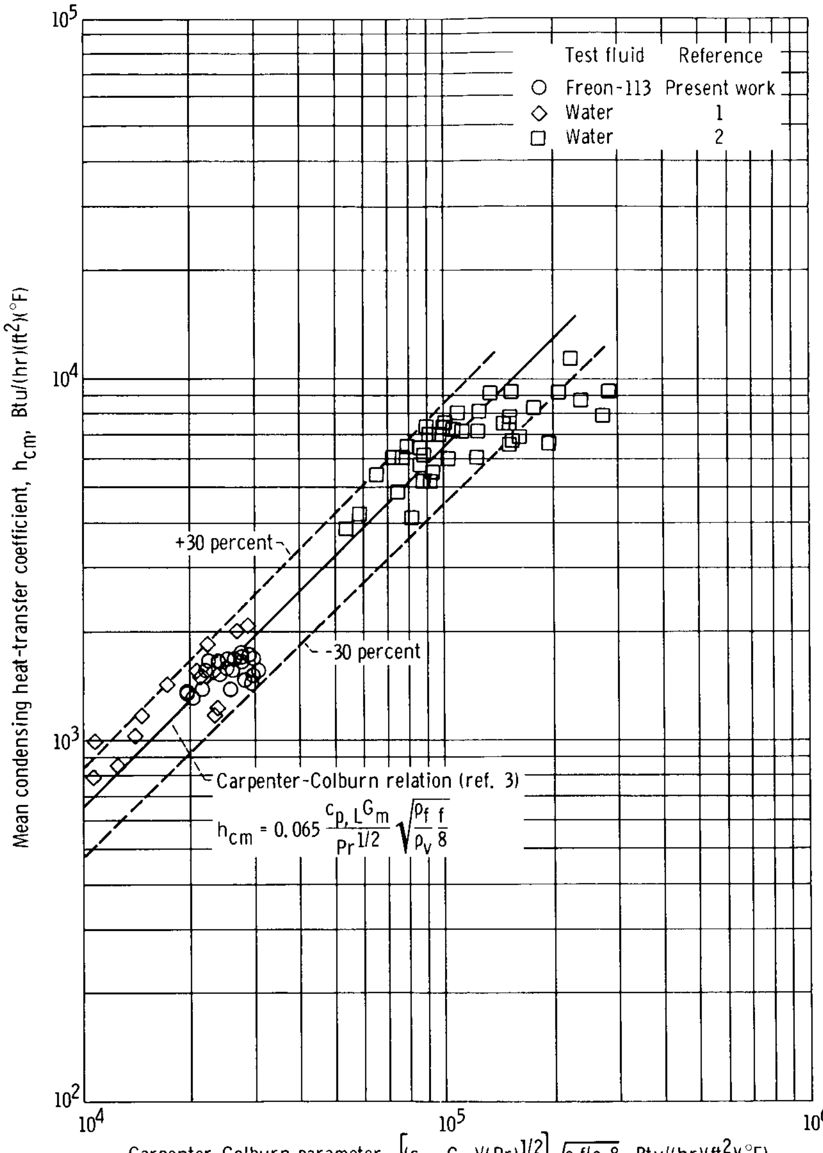


Figure 10. - Mean condensing heat-transfer coefficient as function of Carpenter-Colburn parameter at mean conditions.

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