Greek Symbols

α = fitting ratio b_1/b_2 or m_1/m_2
δ = deviation
θ = coverage of the solid surface by adsorbates
μ = chemical potential
ξ = empirical function of θ defined by Equation (7) or (23) for binary mixtures
Ξ = same as ξ but as function of (N^*/A)
π = spreading pressure of the adsorbed phase
ϕ = empirical function of θ defined by Equation (8) or (24) for pure component
Φ = same as ϕ but as function of (N^*/A)
ψ = empirical function of θ defined by Equation (9) or (25)
Ψ = same as ψ but as function of (N^*/A)

Superscripts

o = pure component adsorption; standard state condition
$'$ = derivative of the function with respect to parameter within parentheses

Subscripts

i = immobile model
j, k = species
m = mobile model
1, 2 = methane, ethane

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Heat Transfer Coefficients and Circulation Rates for Thermosiphon Reboilers

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Heat transfer coefficients and circulating data were obtained for a single tube thermosiphon reboiler operated under vacuum and at atmospheric pressure. The results are compared with existing correlations. An analysis is presented for local heat transfer coefficients in the single phase and two-phase regions.

Normal operation of vertical thermosiphon reboilers at atmospheric and higher pressure results in a two-phase region for more than one-half of the tube length with the remainder of the tube length required for essentially single-phase liquid heating. Under vacuum conditions, the liquid heating section may represent as much as 90% of the tube length. The relatively short two-phase region at subatmospheric conditions would be expected to result in a lower circulation rate than is obtained with the longer two-phase region at atmospheric and higher pressure conditions. Experimental heat transfer and circulating rate data at subatmospheric conditions are limited with the result that design methods must be extrapolated for these conditions.

This paper reports an experimental study of a single tube thermosiphon reboiler operated under vacuum and at atmospheric pressure. Heat transfer coefficients and circulating rate data were obtained and are analyzed to provide confidence in design methods for vacuum conditions.

EXPERIMENTAL EQUIPMENT AND PROCEDURE

Figure 1 shows a sketch of the equipment. The tube was 8 ft. long ¾ in. schedule 80 carbon steel pipe. Electrical heating was used to provide a known flux at point locations on the pipe. Four heaters were used to provide a maximum flux of 9,000 B.t.u./hr. sq. ft. The heaters were made of Nichrome wire of 0.41 ohms/ft. resistance and were wound with ceramic beads around the wire. Each wire was 26.33 ft. long. Total power output at 117 v. was 5,700 w. Powerstats were used to control the heat to the pipe.

Iron-constantan thermocouples were attached to the outside pipe wall at the ends of the 8 ft. section and at 1 ft. intervals along the pipe. Holes 0.05 in. deep were drilled in the pipe wall, and the thermocouple junctions were pressed into these holes. A traveling thermocouple passed downward through the pipe to obtain process temperatures corresponding to the 1 ft. intervals.

The orifice was 0.500 in. diam. in a run of pipe the same as the heated pipe to minimize the pressure drop. Water and ethylene glycol were used to calibrate the orifice. The orifice differential pressure was obtained with a manometer.

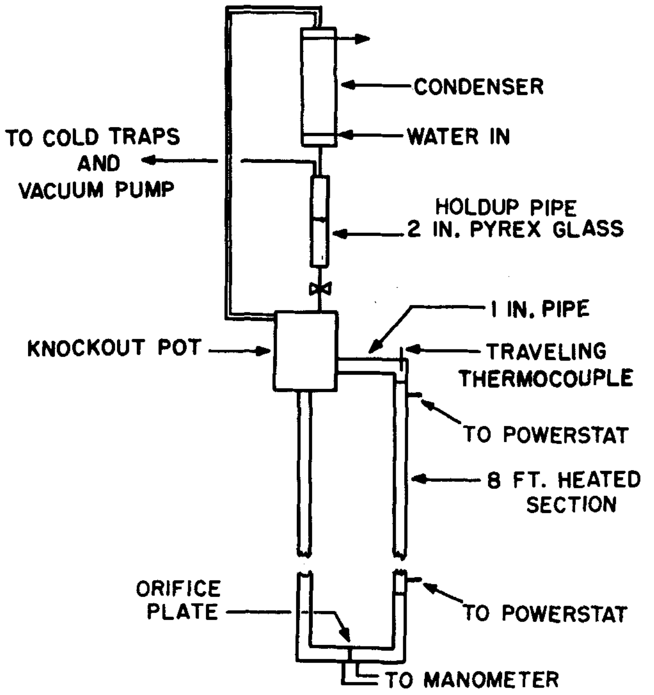

Fig. 1. Diagram of equipment.

The manometer seal fluid was the test liquid in the reboiler and the manometer fluid was water, salt saturated water, or mercury.

The system was charged with 1½ gal. of the liquid to be tested and the liquid was used as the seal fluid for the manometer. Heat to the pipe was set by a voltmeter reading on the powerstats. Level was held approximately constant in the holdup pipe and data were taken when the wall thermocouples showed that steady state temperatures had been reached. The wall temperatures and the corresponding fluid temperatures were taken starting at the bottom of the pipe. The orifice pressure drop was then read with the traveling thermocouple out of the heated section. Observed pressure drop was the same with the thermocouple in the pipe section.

The system was cleaned by boiling methanol and then draining the methanol. Vacuum was pulled on the unit to remove the residual methanol.

Table 1 summarizes the systems and operating conditions.

Table 1. Systems and Operating Conditions

System	Range of top tube pressure conditions—mm. mercury Abs.	Range of heat fluxes B.t.u./hr. sq. ft.
Dowtherm A	24 to 125	2,000 to 9,000
ethylene glycol	13 to 85	2,000 to 9,000
1-dodecanol	39 to 113	2,000 to 9,000
nitrobenzene	28 to 95	2,000 to 6,000
1-dodecene	up to 160	6,000 to 9,000
toluene	atmospheric	1,000 to 9,000
carbon tetrachloride	atmospheric	1,000 to 9,000
heptane	atmospheric	1,000 to 9,000
n-butanol	atmospheric	1,000 to 9,000
methanol	atmospheric	1,000 to 9,000
ethylene dibromide	63—atmospheric	1,000 to 9,000
isopropanol	165—atmospheric	1,000 to 9,000

COMPARISON WITH EXISTING METHODS

Figure 2 shows the experimental data with the correlation suggested by Hughmark (5) for the heat transfer coefficient. Figure 3 compares experimental circulating velocities with those calculated from a computer program based upon gas-liquid two phase flow correlations (7).

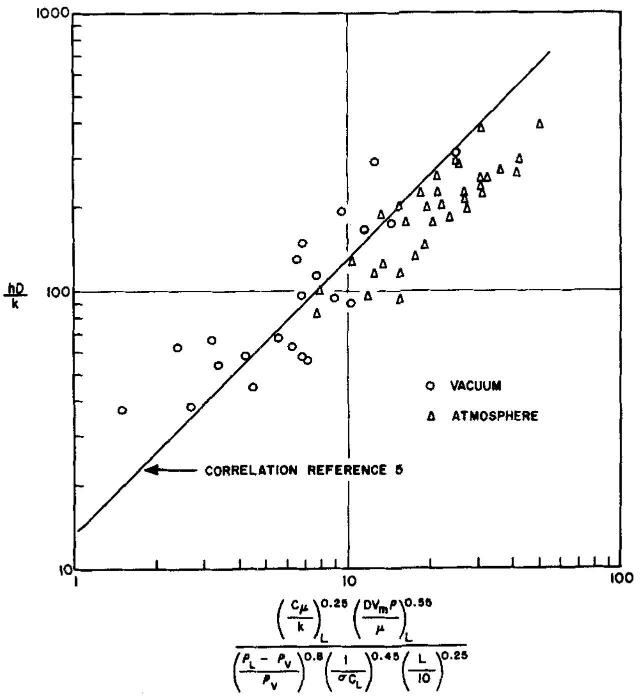

Fig. 2. Heat transfer correlation.

Agreement is good with consideration of the assumptions that are inherent in such a program:

1. Holdup and pressure drop correlations obtained from adiabatic gas-liquid data are applicable to diabatic vapor-liquid conditions.
2. Thermal equilibrium exists throughout the system.
3. The sensible heating zone density is that of the liquid phase.

LOCAL COEFFICIENTS

A single coefficient for the process side of a thermosiphon reboiler represents the integration of a range of local coefficients at points along the tube. Figure 4 shows the data for ethylene glycol in which the coefficient varies with position along the tube. This difference arises from the phase differences which occur. Liquid enters the bottom of the tube and flows primarily as a single phase until

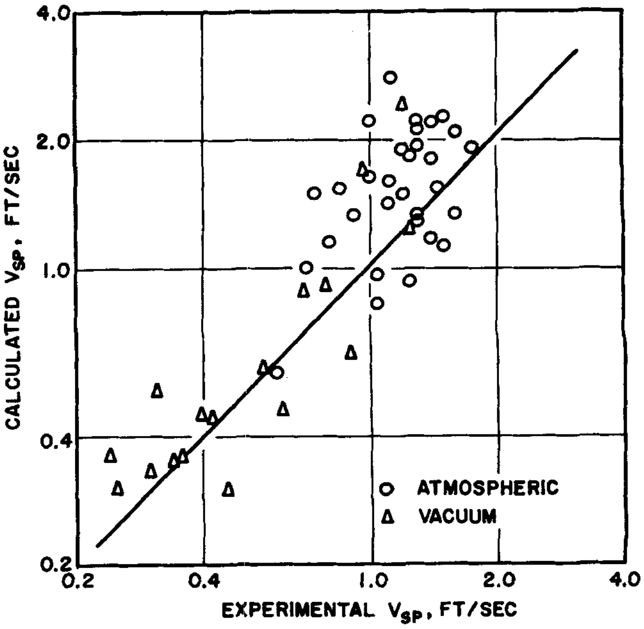

Fig. 3. Comparison of calculated and experimental circulating velocities.

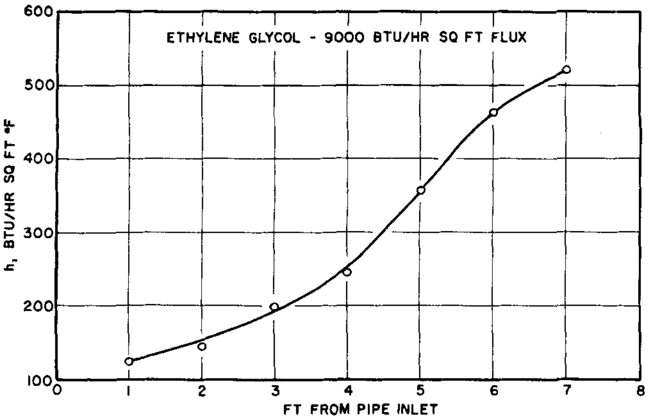

Fig. 4. Local coefficients.

the temperature reaches the boiling point for the corresponding pressure. Vapor then exists as a second phase and the quantity of vapor increases to the tube outlet. The single phase region represents heat transfer by convection and, if the wall temperature is high enough, nucleate boiling at the wall with collapse of the vapor in the subcooled liquid. The two-phase region coefficients are much higher than those for the single phase because the presence of the vapor increases the liquid velocity in contact with the tube wall. Thus, a more fundamental approach to the heat transfer coefficient is to use a series of local coefficients for the length of the tube.

SINGLE PHASE CONVECTION

Nucleate boiling occurs in the single phase region in the lower part of the tube if the tube wall temperature is sufficiently greater than the saturation temperature of the liquid. If the wall temperature is less than the minimum temperature for nucleate boiling, heating will occur only from convection. This is forced convection heating because the average liquid velocities from natural circulation are in the range of 0.3 to 2 ft./sec. Many of the experimental runs were made at low heat fluxes to evaluate single phase non-boiling coefficients. These data would be expected to fit the modified Dittus Boelter equation:

$$\frac{hD}{k} = 0.023\,(N_{Re})^{0.8}\,(N_{Pr})^{0.4}\,(\mu/\mu_w)^{0.14} \tag{1}$$

Average absolute deviation between experimental and calculated coefficients was 13.5% for the data with Reynolds numbers greater than 10,000 where the equation is valid.

NUCLEATE BOILING

Heat transfer with both forced convection and nucleate boiling can be represented by superposition of the two fluxes (*12*):

$$(q/A)_{\text{total}} = (q/A)_{\text{forced convection}} + (q/A)_{\text{nucleate pool boiling}}$$

or in terms of the coefficients and temperature differences:

$$h_{\text{total}}\,(T_W - T_B) = h_C(T_W - T_B) + h_N(T_W - T_S) \tag{2}$$

The coefficient, h_C, is obtained from Equation (1) and the nucleate pool boiling coefficient, h_N, from the equation developed by Rohsenow (*11*):

$$\frac{C_L(T_W - T_S)}{\lambda} = C_{Sf}\left(\frac{q/A}{\mu_L\lambda}\sqrt{\frac{g_c\sigma'}{g(\rho_L-\rho_V)}}\right)^{0.33}\left(\frac{C_L\mu_L}{k_L}\right)^{1.7} \tag{3}$$

The heat flux calculated from Equation (3) corresponds to the final term in Equation (2).

The experimental data were treated to derive C_{Sf} from the runs in which nucleate boiling occurred. Runs with $T_W - T_S$ greater than 15°F. showed nucleate boiling. Table 2 compares the values of C_{Sf} obtained from other forced convection data. The value of C_{Sf} represents both metal surface properties and liquid system properties. The values obtained in this experimental work do not appear unrealistic.

TABLE 2. VALUES FOR C_{Sf} IN FORCED CONVECTION BOILING

Reference	Heating surface	Fluid	C_{Sf}
Rohsenow—Clark (*12*)	Nickel (0.18 in. diam.)	water	0.006
Piret—Isbin (*10*)	Copper (1.068 in. diam.)	isopropanol	0.0022
Piret—Isbin	Copper (1.068 in. diam.)	*n*-butanol	0.003
Piret—Isbin	Copper (1.068 in. diam.)	water	0.013
Piret—Isbin	Copper (1.068 in. diam.)	carbon tetrachloride	0.013
This work	Steel (0.74 in. diam.)	isopropanol	0.004
This work	Steel (0.74 in. diam.)	*n*-butanol	0.007
This work	Steel (0.74 in. diam.)	carbon tetrachloride	0.011

TWO-PHASE REGION

The two-phase region is composed of the bubble, slug, and annular flow regimes. Heat transfer is by forced convection with nucleation suppressed. A forced circulation model that has been used (*1, 3*) is

$$\frac{h_{TP}}{h_L} = \left[\frac{(N_{ReL})_{TP}}{(N_{ReL})_{SP}}\right]^{0.8} = \left[\frac{(V_L)_{TP}}{(V_L)_{SP}}\right]^{0.8} \tag{4}$$

With the definitions

$$(V_L)_{TP} = \frac{(V_L)_{SP}}{R_L} \quad \text{and} \quad (V_L)_{SP} = \frac{(1-x)W_L}{A\rho_L}$$

knowledge of R_L will provide for estimation of the two-phase coefficient. Here the classical approach is to use the Lockhart-Martinelli (*9*) relationship that holdup is a function of the term

$$\frac{1}{X_{tt}} = \left(\frac{W_V}{W_L}\right)^{0.9}\left(\frac{\rho_L}{\rho_V}\right)^{0.5}\left(\frac{\rho_V}{\rho_L}\right)^{0.1} \tag{5}$$

and plot h_{TP}/h_{SP} vs. $1/X_{tt}$. Figure 5 shows such a plot for the experimental two-phase data. The results are not surprising because vertical-upward gas-liquid flow data show significant deviation from the $1/X_{tt}$ plot (*4*). Hughmark has proposed correlations for prediction of holdup for the annular (*6*) and slug flow (*9*) regimes of gas-

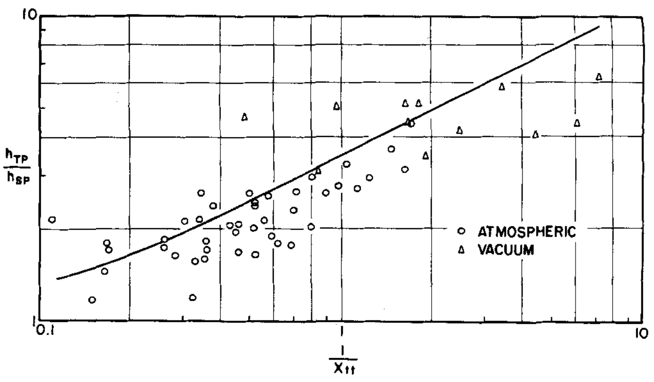

Fig. 5. Comparison with Denger-Addoms correlation.

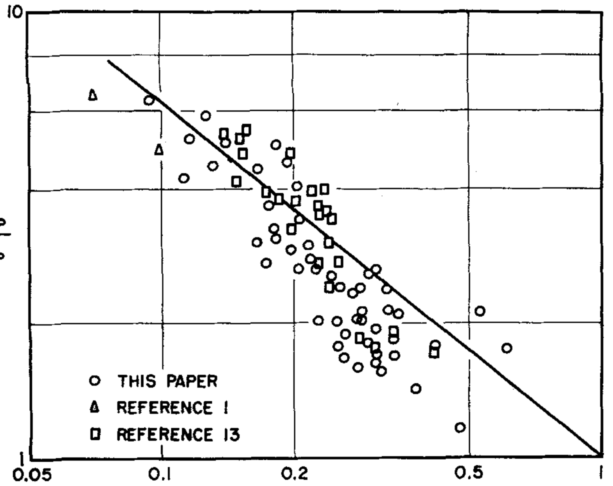

Fig. 6. Two phase heat transfer as a function of holdup.

liquid flow. Figure 6 shows a plot of the experimental two-phase data vs. R_L. A line is shown for

$$h_{TP}/h_{SP} = (1/R_L)^{0.8} \qquad (6)$$

The fit is better than that shown by Figure 5. Figure 6 also shows the data of Talty (*13*) and Dengler and Addoms (*1*).

The annular flow heat transfer data for air-water of Groothuis and Hendel (*2*) shows good agreement with Equation (6).

GAS ADDITION

Vacuum thermosiphons have the disadvantage in that the long sensible heating zone results in a rather high temperature rise in the circulating liquid. One way of reducing this temperature rise is to add a small amount of inert gas to the bottom of the heater. A connection was added below the heater pipe for such a test. A pressure reducer and rotameter were used for the nitrogen addition. Table 3 shows the results for runs with 1-dodecene at a 6,000 B.t.u./hr.sq.ft. flux with and without nitrogen addition.

TABLE 3.

Nitrogen added	Maximum fluid temp. °F	Overall coefficient B.t.u./hr. sq. ft. °F.	Circulating Velocity ft./sec.
0	292	275	0.62
105 SCFM	275	260	1.1
186 SCFM	280	290	1.15

Thus, in this case, the gas addition significantly reduces the liquid temperature and increases the circulating velocity but does not affect the heat transfer coefficient.

SUMMARY

Experimental data for a single tube thermosiphon reboiler show that there is nothing unusual about vacuum operating conditions. The circulating rates and the heat transfer coefficients can be estimated by published design methods.

Local coefficients for the sensible heating zone are predicted by convective coefficients if the wall temperature is of the order of 15°F. or less than the saturation temperature. At higher wall temperatures, nucleation occurs in addition to the convection heating. Local coefficients for the two-phase region appears to be consistent with other published data.

NOTATION

A = heat transfer area, sq.ft.
C = specific heat, B.t.u./(lb.) (°F.)
C_{sf} = coefficient of Equation (3)
D = tube I.D., ft.
g = acceleration of gravity, ft./(sec.)^2
g_c = 32.17 (ft.) (lb. mass)/(lb. force) (sec.)^2
h = film coefficient, B.t.u./(hr.) (sq.ft.) (°F.)
k = thermal conductivity, B.t.u./(hr.) (sq.ft.) (°F.)/ft.
L = tube length, ft.
N_{Pr} = Prandtl number, $(C\mu/k)$
q = heat transferred, B.t.u./hr.
N_{Re} = Reynolds number, $(DV\rho/\mu)$
R_L = liquid volume fraction in two phase flow
T = temperature, °F.
U = over-all heat transfer coefficient, B.t.u./(hr.) (sq. ft.) (°F.)
V = linear velocity, ft./sec.
V_m = log mean liquid-vapor velocity in pipe, $= (V_2 - V_1)/\ln V_2/V_1$
W = mass rate of flow, lb. mass/sec.
X_{tt} = Lockhart-Martinelli parameter, Equation (5)
x = vapor weight fraction

Greek Letters

σ = surface tension, dynes/cm.
σ' = surface tension, lb. force/ft.
λ = latent heat, B.t.u./lb.
ρ = density, lb. mass/cu.ft.
μ = viscosity, lb. mass/(ft.) (sec.)

Subscripts

B = bulk
C = convection
L = liquid
N = nucleate
S = saturation
SP = single phase
TP = two phase
V = vapor
W = wall
1 = point of inlet to section
2 = point of outlet of section

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