CONCLUSIONS

The isothermal effect of pressure on the enthalpy of methane, a 5.1 mole % propane in methane mixture, and a 12.6 mole % propane in methane mixture was determined by means of isothermal flow calorimetry. The study covered the range 90 to 200°F. and 500 to 2,000 lb./sq. in. abs. The results are believed to be precise to within ± 2 B.t.u./lb.

Comparison of results with values from other experimental data showed good general agreement. Both the volumetric data of Sage and Lacey, and the Joule-Thomson data of Budenholzer, et al. yield enthalpies in substantial agreement with the present work. The Mollier charts from work at the University of Michigan are in poorer agreement with the data of this work, particularly at high pressures.

Comparisons of the experimental data with calculated enthalpies by using the Redlich-Kwong and Benedict-Webb-Rubin equations of state and Pitzer's corresponding states correlation showed close agreement. The equations of state and Pitzer's correlation predicted enthalpy values usually within 2 B.t.u./lb. of the experimental results, demonstrating the applicability of these predictive techniques in the range of conditions covered by this study.

ACKNOWLEDGMENT

This work was supported by Oklahoma State University and the Natural Gas Processors Association. D. D. Dillard was on a National Defense Education Act Fellowship. The analyses shown in Table 1 were provided by Phillips Petroleum Company, who prepared and donated the materials.

NOTATION

H = enthalpy, B.t.u./lb. mol.
h = enthalpy, B.t.u./lb.

Superscripts and Subscripts

o = calorimeter outlet pressure
p = calorimeter inlet pressure
T = run temperature
$*$ = ideal gas state

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Manuscript received October 26, 1966; revision received December 15, 1967; paper accepted January 10, 1968.

Condensation from Superheated Gas-Vapor Mixtures

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Superheated mixtures of vapor and noncondensable gas were cooled and condensed in downflow inside a single tube vertical condenser, 0.995 in. I.D. and 3.5 ft. long. The systems studied were air-water, carbon dioxide-water and helium-water. A bulk stream mixing probe was developed which enabled the measurement of point values of the stream temperature and partial pressure at various positions within the condenser. The Reynolds number range of the investigation was 5,700 to 34,000.

A theoretical solution of the problem, using Bras' design method, was solved rigorously on a digital computer. Results indicated excellent agreement between the theoretical and experimental cooling paths and the required transfer area. The inclusion of the Ackerman correction factor (for the sensible heat carried by the diffusing vapor) into the calculation gave significantly better agreement for all cases tested.

When a superheated vapor-gas mixture is contacted with a surface below its dew point, condensation will take place. Owing to the presence of the non-condensable component, the mechanism of the phenomena becomes one of simultaneous heat and mass transfer. Due to this parallel transfer, the changes in the mixture temperature

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and in the partial pressure of the condensable component is governed by the relative rates of heat and mass transfer.

Colburn and Hougen (*5*) in their pioneer investigation in 1934 presented a stepwise trial and error method for designing cooler condensers. Their basic equation was derived for a tubular water-cooled condenser and was based on a heat balance at the gas-liquid interface. The condensate loses heat through the tube wall to the cooling water and receives latent and sensible heat from the condensing vapor and the bulk mixture. The heat balance can be expressed by the following equation:

$$h_G(t_o - t_i) + k_G M_v \lambda_i (p_G - p_i) = U(t_i - t_l) \qquad (1)$$

A correction factor for the sensible heat carried by the diffusing vapor was suggested by both Ackerman (*1*) and Colburn and Drew (*4*). This factor is generally referred to as the *Ackerman correction.* By including the Ackerman correction in the Colburn and Hougen heat balance gives

$$h_G(t_o - t_i)\,\frac{\epsilon}{1 - e^{-\epsilon}} + k_G M_v \lambda_i (p_G - p_i) = U(t_i - t_l) \qquad (2)$$

The effect of the relative transfer rates was neglected in the Colburn and Hougen method since the mixture was assumed to enter the system saturated and to remain saturated. Colburn and Drew extended the analysis to handle superheated mixtures. They proposed a means of following the gas phase temperature partial pressure from point to point in the condenser by using the following relationship:

$$\frac{dp_G}{dt_G} = \frac{(p_G - p_i)}{(t_G - t_i)}\,\frac{k_G}{h_G}\,c M_m\,(\pi - p_G)\,\frac{e^{\epsilon} - 1}{\epsilon} \qquad (3)$$

The Ackerman correction has generally been neglected in most design considerations.

Equation (3) is valid only as long as the mixture remains superheated. Once the mixture reaches its dew point, heat released by bulk phase condensation should maintain the mixture in a saturated condition throughout the remainder of the condenser. However, some authors have suggested the possibility of supersaturation or fog occurring under certain conditions.

The equations of Colburn and Drew were incorporated into a design method by Bras (*2*) for sizing a cooler condenser operating on a superheated gas-vapor mixture. The stepwise method involved solving Equation (2) by trial and error at various points in the condenser for the interface temperature, t_i. The bulk temperature and partial pressure of the vapor are determined by a linear extrapolation over small temperature increments by using Equation (3). A numerical integration is then performed to evaluate the total transfer area by

$$A = \int \frac{dq}{U(t_i - t_l)} \qquad (4)$$

In the research program undertaken in this investigation the equipment and technique necessary to measure local conditions in a single tube condenser operating on a superheated gas-vapor mixture was designed and constructed. The experimental results obtained were then compared with a rigorous digital computer model of the physical process using Bras' design method. The effect of the Ackerman correction factor on the calculation was determined and all the experimental runs were simulated by the model. The systems studied were of widely varying characteristics air-water, helium-water, and carbon dioxide-water. Primary variables of interest were condenser inlet concentration and degree of superheat. The mixtures were cooled and condensed in down flow in a vertical single tube condenser 3.5 ft. long, with cooling water flowing countercurrent in the annulus.

EXPERIMENTAL EQUIPMENT AND PROCEDURE

A schematic flow diagram of the experimental apparatus is shown in Figure 1. The noncondensable gas was pumped by the blower (B) and metered by the rotameter (R). The gas was then preheated electrically by the heater (G) and mixed with steam from the steam lines in the chamber (M). The resulting gas-vapor mixture was heated to a controlled temperature by the electrically heated superheater (S). The line between the mixing chamber and the superheater was wrapped with resistance wire and insulated to prevent condensation from occurring on the tube walls of the section. The mixture then passed through a Swarthout separator (S1) which assured that the mixture was dry and thoroughly mixed. Mixing was important at this point since temperature control on the stream entering the test section was based on the measurement made in this section. The mixture was then cooled and condensed in the single tube vertical test condenser (C). Cooling water flowed in the annular space countercurrent to the gas flow. Condensate in the exit stream was separated in the condensate separator (S2) and collected in a reservoir (J) which was vented to the stream leaving the separator. The residual gas-vapor mixture was cooled to approximately the cooling water temperature in a total condenser (C1). Condensate in the stream leaving the total condenser was removed by separator (S3) and collected in reservoir (J). The noncondensable gas was then vented to the room for the air-water runs, and recycled back to the blower for the helium-water and carbon dioxide-water runs. Make-up gas for the recycle runs was supplied from a 4.5 cu.ft. gasometer. Pressure at the inlet of the blower was adjusted to slightly above atmospheric by adding weights to the top of the gas holder.

The rotameter temperature was measured by a mercury in glass thermometer (T). Copper-constantan thermocouples (TC) measured both the temperature of the mixture at the inlet to the test section, and that of the cooling water. The cooling water temperature rise through the test condenser was measured by a differential thermocouple. Concentration and temperature measurements within the condenser were made with a bulk stream mixing probe.

The test condenser was a double pipe exchanger with a 48 in. long inner section of 1⅛ in. O.D. (0.995 in. I.D.) type-K copper tubing inside a section of 2⅛ in. O.D. (1.981 in. I.D.) type-L copper tubing having an effective heat transfer length of 42 in. Seven copper-constantan thermocouples were imbedded in the wall of the inner tube; two were placed 1 in. from each end and the other five equally placed at 7 in. intervals. The ends of the inner tubing

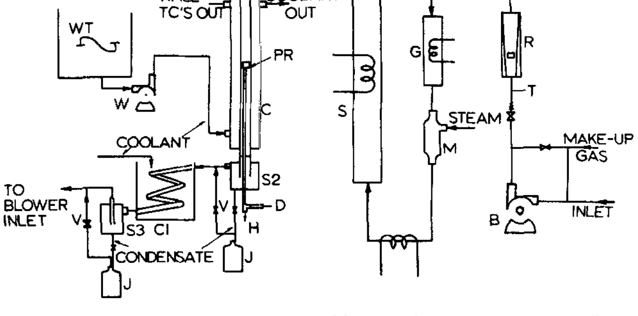
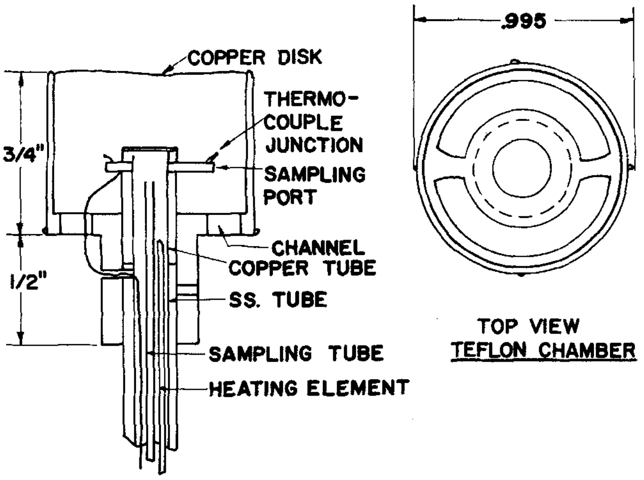


Fig. 1. Apparatus schematic: B = blower, T = thermometer, R = rotameter, P = pressure measuring tap, G = gas heater, M = mixing chamber, S = superheater, S1 = Swarthout separator, Tc = thermocouple, Pr = probe, C = test condenser, S2, S3 = condensate separator, H = probe heating element, D = probe vent for dew point measurement, J = condensate collection jar, C1 = total condenser. WT = cooling water tank, W = cooling water pump, V = vent line.



TOP VIEW TEFLON CHAMBER

CONSTRUCTED PROBE CUTAWAY

Fig. 2. Probe schematic.

were ground square and attached to the rest of the system by tight fitting rubber tubing. A 10 in. calming section was provided at the condenser inlet. The condenser is essentially the same as the one described by Votta and Walker (*10*).

Figure 2 is a schematic diagram of the probe used to measure the bulk mean concentration and temperature of the gas-vapor mixture at various positions within the condenser. It consisted of a small mixing chamber mounted on a 4.5 ft. length of stainless steel tubing (¼ in. O.D.). The tube left the system through a packing gland located in the bottom of the condensate separator and could be moved vertically to allow positioning of the chamber at any point in the condenser. A centering guide was attached about 2½ in. from the base of the chamber. Positioning was determined from a mounted yard stick.

The chamber was machined from a solid teflon cylinder and consisted of a thin walled cylinder ¾ in. long and 0.875 in. diameter on a small mounting adapter. Slots were cut in the base of the cylinder to enable the mixture to flow through. The clearance between the tube wall and the chamber was 0.060 in. which was sufficient to allow the condensate to flow past uninterrupted. The chamber was capped with a tight fitting copper disk having eight slots cut radially from the circumference to a point close to the center. The eight sectors were bent up so that the device resembled a propeller. Extending into the chamber from the bottom was a capped section of tight fitting ¼ in. O.D. copper tubing. The open end of this tube was attached flush up against the stainless steel mounting tube. Two concentration sampling ports made of 1/16 in. stainless steel hypodermic tubing were brazed into holes in the copper tubing 180° apart. The opening of one port was close to the copper tube and the other extended slightly further than midway between the chamber wall and the copper tube. The gas sample from both ports was mixed in the copper tube to insure a more representative sample in the event that mixing in the chamber was incomplete.

Temperature measurement in the chamber was made by two teflon coated thermocouple junctions, one draped over each of the sample ports. Agreement of these two temperature measurements indicated adequate mixing. The thermocouple leads left the system through the mounting tube.

Pressures were measured at the rotameter and at the condenser inlet with mercury manometers. The condenser pressure was taken as the inlet pressure since the pressure drop through the condenser was negligible.

The gas-vapor sample line was a section of 1/16 in. hypodermic tubing placed inside the mounting tube. A controlled flow of the gas mixture passed through the mounting tube outside of the sample line and thus served as an insulating layer to the sample. To completely eliminate any condensation in the sampling line a section of stainless steel sheathed chromel-alumel thermocouple wire, welded on one end and insulated from the sheath, was also run through the mounting tube and served as a resistance wire heating element capable of dissipating 150 w. All lines and the thermocouple wires left the system through a packed tee attached to the end of the stainless steel mounting tube.

Cooling water was pumped from an agitated 400 gal. tank by a centrifugal pump. Temperature control was maintained by a mercury-in-glass thermostat which actuated a solenoid valve that blew live steam into the agitated water. All runs were made with a constant inlet water temperature of 69.4°F., and control was maintained to within 0.2°F. The temperature of the gas-vapor mixture entering the test section was maintained to within ± 0.5°F. by a Brown Electronik on-off controller. Concentration measurements were made by a dew point technique using a modified dew point cup. The accuracy of the measurement was estimated to be 0.3°F.

Electronic potentiometers were used to make emf readings from the thermocouples with an accuracy of 1 uv (0.005°F.). Due to temperature fluctuations, no attempt was made in most cases to read temperatures closer than to 0.5°F.

In making a run, power to all the electrical heaters was first turned on, the temperature controller for the superheater was set, and the cooling water pump was started. The blower was turned on and the noncondensable flow rate was adjusted. Once the temperature of the gas leaving the preheater was sufficiently above the inlet dew point for the run, the steam was let into the system. When the test section inlet stream temperature reached the controller set point, adjustment was made to the heaters to minimize the control band. The system stabilization time was approximately two hours.

After stabilization, the probe measurements were made proceeding down the condenser from inlet to outlet. At each position the two probe thermocouples and the determined dew point were recorded. Wall temperatures and the cooling water rise were determined with the probe located at the condenser outlet. This eliminated probe interaction with the measurements.

The wall temperature readings were found to fluctuate from 0.5 to 1.0°F. in some cases, due to the high heat transfer rates. Fluctuations in the probe temperature measurements were a maximum of 0.5°F., with differences in readings between the two probe thermocouples being 0.5 to 0.75°F.

Before making any experimental runs, the probe was tested for adequate mixing, using air as a heat transfer medium in the test condenser.

Utilizing the experimentally determined radial temperature profile plots of Seban and Shimazaki (*7*) for air flowing at a Reynolds number of 10,400 and the universal velocity distribution, the following relationship between the bulk stream temperature and centerline temperature was derived:

$$t_m - t_w = 0.9529\ (t_c - t_w) \qquad (5)$$

A series of temperature measurements was made at the same condenser position using the bulk probe and then using a multiple junction probe for estimation of the centerline temperature with air flowing at a Reynolds number of 10,000. The section in which the measurements were made had essentially a constant wall temperature. The maximum difference between the bulk mean temperatures calculated from Equation (5) using the experimentally determined centerline temperature and the bulk probe measurements was 0.42°F. The maximum difference in the readings of the two mixing probe couples was 0.25°F.

The small difference between the two probe thermocouples indicated thorough mixing in the probe chamber. The close agreement on the mean temperatures determined by the two methods showed that the true mean was approached. Measurements made on the gas-vapor mixtures during the runs showed deviations of 0.5 to 0.75°F., which indicated that the conclusions about temperature mixing was applicable in these runs also. No attempt was made to verify the probe concentration measurements, but it was assumed that adequate temperature mixing was indicative of adequate concentration mixing. Since concentration samples were averaged from two positions in the probe chamber, it is felt that a reasonable accurate indication of bulk stream concentrations was obtained.

CALCULATION METHOD

The experimental temperature-concentrations and temperature-length paths of the gas-vapor mixture as it cooled in the condenser were compared with the calculated paths. The mathematical model utilized for this comparison was essentially the basic design method of Bras. All computation was programmed in the Fortran IV language and executed on an I.B.M. 1410 computer.

Initial studies indicated that the cooling water film coefficient showed a variation of from 150 to 1,000 Btu./hr.sq.ft. °F. in traveling through the test condenser. This was due to the abrupt entrance and exit conditions for the cooling water in the test condenser. The uncertainty that would have been present by basing the calculation on an average water film coefficient was removed by utilizing an experimentally determined wall temperature profile for each run. Equations (2) and (4) were, therefore, modified by expressing the overall driving force as $h_c\ (t_i - t_w)$ rather than $U\ (t_i - t_l)$.

The bulk phase transfer coefficients in Equation (2) were determined from the Colburn analogy (3) assuming the equality of J_H and J_M:

$$h_G = (J_H\, c\, G)/(N_{Pr})^{2/3} \qquad (6)$$

$$k_G = (J_M\, G)/M_m\, P_{GM}(N_{Sc})^{2/3} \qquad (7)$$

The mixture transport properties necessary in Equations (3), (6), and (7) were determined at each point in the condenser. The bulk stream thermal conductivity was calculated by an equation proposed by Lindsay and Bromley (6). The mixture viscosity was determined by Wilke's method (*11*). All other mixture properties were determined as a molar mean. Experimental values of the molecular diffusivity were corrected for pressure by an inverse proportionality and for temperature by a direct proportionality of the absolute temperature to the three-halfs power. The vapor pressure was determined from the Antoine form of the vapor pressure equation with the constants evaluated by a nonlinear least squared analysis of tabulated data.

The condensate film coefficient was calculated from the point value form of the Nusselt equation (7) with properties evaluated at the effective film temperature. No attempt was made to correct the condensate coefficient for shear due to the vapor flow rate. Generally the resistance of the condensate layer is small compared to the other resistances and the effect of the shear on the condensate coefficient should not greatly affect the result.

The detailed computation procedure and calculation flow diagram is given by Stern (9). The computation steps can be outlined as follows:

Starting at the inlet to the condenser where the condensable partial pressure, bulk temperature, and the wall temperature are known, the transport properties and transfer coefficients are determined. The interface conditions are then determined using Equation (2) and the vapor pressure relationship. The path derivative dp_G/dt_G, from Equation (3) is calculated, and by using this value and a linear extrapolation over a small temperature increment the bulk phase conditions are then determined at a point further along in the condenser. The following step-by-step procedure is then initiated.

1. The total heat transferred, condensate and total mixture flow rates are determined at the point from a heat and mass balance.
2. The transport properties and transfer coefficients are computed.
3. Equation (2) is solved by trial and error for the interface conditions using a trial value for the wall temperature at the point.
4. The total condenser area up to this position is established by Equation (4). The wall temperature is then established from the stored experimental wall temperature profile. If this value differs by more than 0.05°F. with the value used in step 3, the step is repeated with the new wall temperature.
5. The mixture temperature is checked to see if the condenser outlet has been reached. If it has the computation is stopped.
6. The path derivative is determined from Equation (3) and the bulk phase conditions are determined for a small increment in temperature down the condenser. The computation returns to step 1.

The integration in step 4 was carried out using the trapezoidal rule. The wall temperature profile was stored in the computer by a linearized segmenting technique from the smoothed experimental profile data. Pure component properties were fed into the computer as polynomials in temperature. The number of increments chosen for the calculation was based on an estimated error of 1% or less in the integration step.

The experimental data fed to the computer model were terminal temperatures, inlet dew point, condenser pressure, cooling water rate, wall temperature profile, and the noncondensable flow rate. The model then predicted the required condenser length and the path taken by the cooling mixture. These results were then compared with the actual length of 3.5 ft. and the experimentally measured cooling path. The calculation on each run was made in two phases; the first included the Ackerman correction factor, and the second neglected this factor.

RESULTS

A summary of the experimental results for the condenser terminal conditions is presented in Table 1. Also included are the calculated dew points of the gas-vapor mixture leaving the condenser. These were calculated using the Ackerman correction and may be compared with the measured outlet mixture dew points. For all the runs made, the average ratio of the calculated total condensate to the measured condensate was 0.95. The average ratio of the calculated total condenser area to the actual area was 0.93. The Ackerman correction was used in the calculation of the condensate and condenser areas above.

There was no evidence of any mist or fog formation in any of the tests. While most of the runs made were with highly superheated mixtures and fog formation would probably not be expected, a few runs were made in which the exit gas was saturated. Even for these no mist formation was noted.

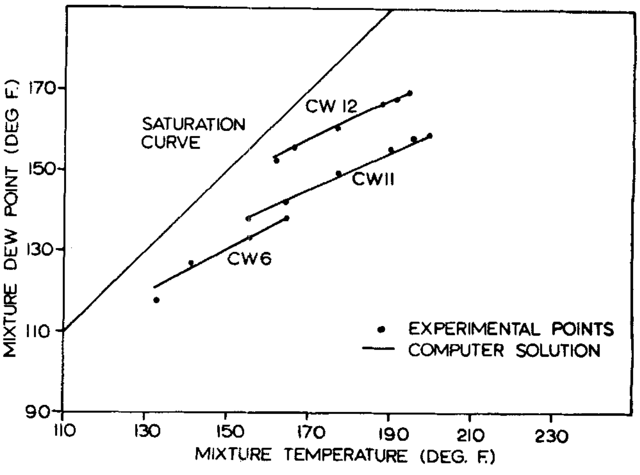


Fig. 3. Partial pressure-temperature path (carbon dioxide-water runs).

Table 1. Summary of Results

RUN NO.	GAS FLOW RATE	INLET PRESSURE	REYNOLDS NUMBER		MIXTURE TEMPERATURE (°F.)		MIXTURE DEW POINT (°F.)		
	lb./hr.	atm.	in	out	in	out	in	out	out(cal.)
AW1	36.62	1.088	13709	13171	214.4	150.3	146.2	123.8	124.1
AW2	35.15	1.094	14459	13192	216.0	157.4	158.0	136.8	135.5
AW3	32.16	1.101	17897	14206	216.1	175.7	179.8	159.5	160.2
AW4	36.54	1.088	13549	13077	230.9	160.5	147.6	125.2	126.0
AW5	36.33	1.091	14302	13406	180.6	135.1	147.6	126.0	125.9
AW6	36.36	1.088	14769	13593	155.5	126.1	147.4	125.2	126.1
AW7	37.93	1.119	13691	13525	121.6	101.2	116.0	99.0	101.2
AW8	37.33	1.049	13625	13389	131.6	105.4	120.6	101.2	104.0
AW9	38.15	1.091	13215	13318	159.0	118.3	116.8	99.5	100.6
AW10	36.99	1.052	13229	13109	159.8	119.4	124.5	104.8	106.5
AW11	36.37	1.019	13777	13243	163.2	124.6	135.8	116.9	116.5
AW12	32.53	1.093	17814	14196	195.2	161.2	177.2	157.5	156.9
AW13	36.09	1.083	16214	14188	195.1	151.5	163.4	140.2	141.9
AW14	23.14	0.979	9651	8777	181.9	135.7	150.4	128.0	128.0
CW1	83.72	1.367	33307	34025	170.9	138.8	131.5	114.0	117.0
CW2	70.63	1.273	32121	30591	195.0	162.1	169.2	152.5	151.9
CW3	72.66	1.272	29886	30000	185.2	146.3	147.4	128.0	129.1
CW4	73.24	1.264	29374	29877	182.2	145.0	138.2	119.0	122.0
CW5	73.84	1.268	29972	30390	172.9	137.0	137.8	118.5	120.3
CW6	73.84	1.272	30342	30582	164.3	134.7	137.8	116.8	122.0
CW7	48.62	1.133	19849	20050	190.3	145.0	142.8	119.8	123.4
CW11	48.02	1.167	20714	20236	199.9	155.6	158.5	137.5	138.5
CW12	48.00	1.164	21334	20612	180.9	146.1	158.8	134.3	139.8
CW13	50.33	1.155	20158	20616	189.9	140.9	137.9	114.8	117.6
CW14	50.90	1.155	30974	21189	170.5	130.2	138.0	114.2	117.5
HW1	8.87	1.122	7821	5720	200.2	143.4	152.9	136.3	133.5
HW2	8.56	1.136	10465	6784	202.3	153.9	166.0	150.0	145.5
HW3	8.69	1.122	8897	6084	183.8	139.1	158.0	138.5	137.7
HW4	12.53	1.256	9591	7593	191.3	143.1	150.5	133.6	133.9
HW5	12.71	1.243	8301	6977	189.6	138.5	141.8	125.7	126.5
HW6	12.70	1.243	8732	7215	165.9	130.0	142.8	126.2	128.0
HW7	12.69	1.239	8941	7354	155.3	129.2	143.2	128.2	129.2
HW8	12.74	1.233	7272	6339	159.6	119.6	130.9	115.5	116.4
HW9	12.77	1.233	7902	6543	140.1	118.8	134.3	116.4	118.8

AW = AIR WATER CW = CARBON DIOXIDE-WATER HW = HELIUM-WATER

DISCUSSION OF RESULTS

Figures 3, 4, and 5 show some sample cooling paths. The plots are presented in the form of the actual experimentally measured variables, dew point vs. temperature. The vapor concentration or partial pressure may readily be calculated from the dew point and total pressure. Plotting the data in this form has the added advantage that the saturation curve appears as a straight line and the degree of superheat is immediately determinable. The curves represent the computer model solution and the points represent the experimental measurements. The fit of the experimental points by the theoretical curve is excellent for the air-water system; maximum differences are of the order of 2°F. For the carbon-dioxide-water and helium-water systems differences are as high as 4°F., but the fit is still reasonable.

A series of three runs was made with overlapping conditions using the helium-water system to check the reproducibility of results. In other words, the same pressure-temperature path curve was followed for the three runs starting from different points along the path. The results of these runs are presented in Figure 4. Figure 5 illustrates the effect of the Ackerman correction factor on the cooling path prediction. Runs AW12 and AW3 have been included since they represent the highest concentration runs made and, therefore, would show the greatest effect of the interaction due to the high mass transfer rates. Neglecting the Ackerman correction, predicted outlet dew points were in error by as much as 4°F.

Figure 6 presents the effect of the Ackerman correction factor on design length reliability. Errors in the predicted design length by neglecting the factor were as high as 15%.

The excellent agreement between the calculated and experimental paths came as a surprise, as it was not expected. The agreement was good, not only for the entire

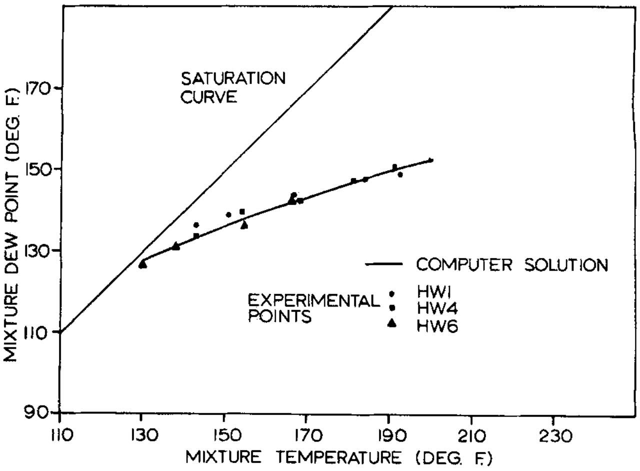


Fig. 4. Partial pressure-temperature path illustrating reproducibility of results.

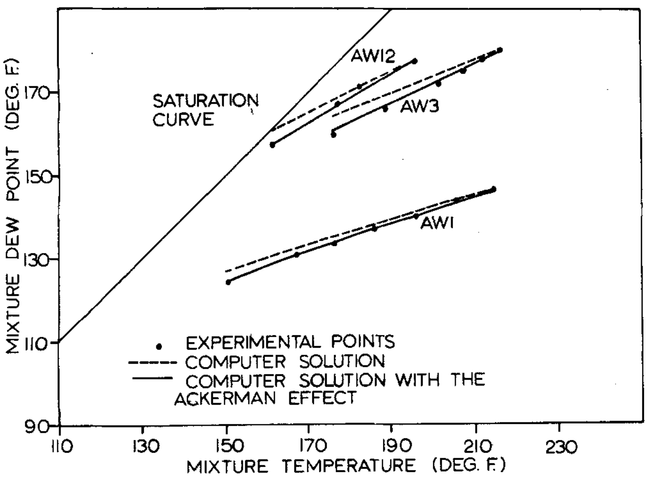


Fig. 5. Partial pressure-temperature path illustrating the Ackerman effect.

condenser but also for each point in the condenser. At the start of the investigation, it had been expected that the interaction of the high mass flux on the profiles would have invalidated the Colburn analogy. Apparently for the teste made, this interaction was not sufficient to appreciably affect the analogy. It would be interesting to carry out additional tests using higher mass flow rates and other systems in which water is not the vapor.

CONCLUSIONS

The results of the investigation indicate that Bras' method in conjunction with the Colburn analogy can be utilized for predicting the partial pressure-temperature path taken by a superheated gas-vapor mixture during cooling and condensing in a single tube condenser. Condenser design lengths were predicted within an accuracy of 15% for all runs made except two.

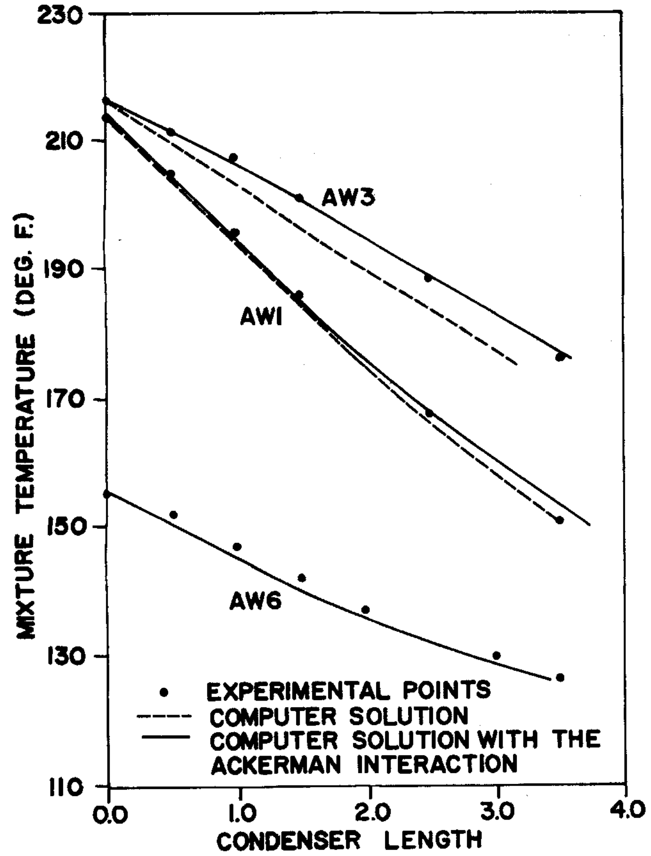


Fig. 6. Temperature-length path illustrating Ackerman effect.

Although in the past it has been considered that neglecting the Ackerman correction factor introduces negligible error into design calculation, it has been shown that the factor can make an appreciable difference for high mass transfer rates.

The feasibility of the use of a bulk stream mixing probe for determination of local conditions within a cooler condenser was demonstrated.

NOTATION

A = heat transfer surface area, sq.ft.
c = specific heat, B.t.u./lb. °F.
G = mass velocity, lb./hr.sq.ft.
h_c = condensate heat transfer coefficient, B.t.u./(hr.)(sq.ft.)(°F.)
h_G = gas film heat transfer coefficient, B.t.u./(hr.)(sq.ft.)(°F.)
J_H = dimensionless heat transfer factor
J_M = dimensionless mass transfer factor
k_G = gas film mass transfer coefficient, lb.moles/(hr.)(sq.ft.)(atm.)
M = molecular weight
N_{Sc} = Schmidt number
N_{Pr} = Prandtl number
p_G = partial pressure of condensable in bulk stream, atm.
p_{BM} = logarithmic mean partial pressure of nondiffusing gas, atm.
q = total heat transferred, B.t.u./hr.
t = temperature, °F.
U = overall heat transfer coefficient, B.t.u./(hr.)(sq. ft.)(°F.)
W_v = rate of condensation per unit area, lb./(hr.)(sq.ft.)
ϵ = ratio of sensible transfer by diffusing vapor to sensible transfer by bulk mixture, $W_v C_v / h_G$
λ = latent heat of vaporization, B.t.u./lb.
π = total pressure, atm.

Subscripts

c = center of stream
G = gas phase
i = interface
l = liquid
m = mean
v = condensable
w = wall

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Manuscript received July 18, 1967; revision received December 4, 1967; paper accepted December 4, 1967.