Condensation of Vapor in the Presence of Noncondensing Gas

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When a pure vapor is passed over a cold surface, the vapor condenses at a rate which depends on the difference in temperature between the bulk of the vapor and the cold surface. The temperature at the interface between the vapor and liquid is assumed to be equal to that of the bulk of the vapor; thus the resistance to heat transfer between the vapor and the surface is due entirely to the layer of condensate separating them. The magnitude of this resistance can be estimated by methods such as the theoretical equations of Nusselt (*13*).

If a mixture of noncondensible gas and condensible vapor is passed over a surface colder than the dew point of the mixture, partial condensation will occur. In this case however it is necessary to consider an added resistance to the transfer of heat from the gas-vapor mixture to the tube wall. According to well-founded theories there exists between the bulk of the mixture and the condensate surface a resistance of such magnitude as to indicate that transfer of heat and mass through it occurs by very slow processes such as pure conduction and molecular diffusion. The magnitude of this resistance can be estimated from empirical correlations which provide a knowledge of the individual film coefficient of heat transfer and the individual film coefficient of mass transfer. An example of such a correlation is that of Colburn (*7*):

$$\left(\frac{h_G}{cG}\right)\left(\frac{c\mu}{k}\right)^{2/3} = 0.023\left(\frac{DG}{\mu}\right)^{-0.2} = \frac{k_G P_{nm} M}{G}\left(\frac{D_v \rho}{\mu}\right)^{-2/3} \qquad (1)$$

The method of using these ideas in the design of equipment was first proposed, by Colburn and Hougen (*10*), in 1934. The rate equation for transfer of sensible heat and latent heat may be written as

$$dq_T = Wc\,dt_L = h_{oi}(t_i - t_L)dA = h_G(t_v - t_i)dA + \lambda k_G(p_v - p_i)dA \qquad (2)$$

The heat transfer area required in a particular case is determined by graphical integration of the expression

$$A = Wc \int \frac{dt_L}{h_{oi}(t_i - t_L)} \qquad (3)$$

To carry out this graphical integration it is necessary to determine values of t_i corresponding to values of t_L:

$$h_{oi}(t_i - t_L) = h_G(t_v - t_i) + \lambda k_G(p_v - p_i) \qquad (4)$$

The points (p_v, t_v) and (p_i, t_i) are assumed to be on the vapor-pressure curve of the condensing material; that is, it is assumed that the bulk of the vapor is saturated and that equilibrium exists at the interface. For given conditions of the inlet vapor composition and rate, selection of a value of p_v at a particular point will permit calculation of the amount of condensation which has occurred to this point. The value of t_L can be evaluated from a heat balance. Values of h_{oi}, h_G, and k_G can be estimated from the flow rates and properties of the fluids. By using the vapor-pressure curve for the required second equation one can solve Equation (4) for p_i and t_i, usually by trial and error. However Bras (*4*) has recently proposed a method for carrying out this solution directly.

The method of Colburn and Hougen is in agreement with current methods of calculation in the fields of heat and mass transfer, but is very tedious to apply; therefore several attempts have been made to arrive at simpler methods of calculation (*1, 2, 3, 6, 12, 15*) based on various assumptions. Until quite recently, however, there were available in the literature very few experimental results which would serve as a basis for testing the method of Colburn and Hougen and the simplified methods which have been proposed, only a few sets of data being available (*5, 9, 16*).

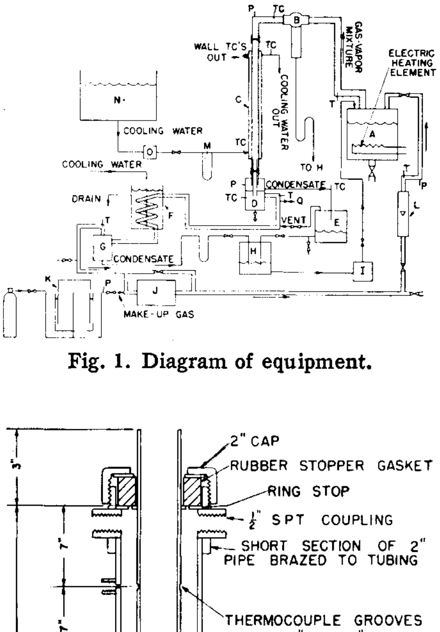

Fig. 1. Diagram of equipment.

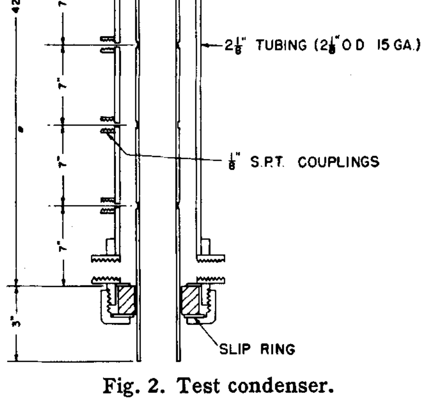

Fig. 2. Test condenser.

The research program described here was undertaken originally to provide a body of experimental data for such calculations. In the planning of the program it was considered essential that systems of widely varying characteristics be used: air-water, carbon dioxide–water, helium-water, and carbon dioxide–*n*-propyl alcohol. For each of these systems the effects of flow rate, temperature of the entering gas-vapor mixture, and cooling-water temperature were studied. Saturated mixtures of vapor and gas were cooled and condensed inside a single-tube vertical condenser, 1 in. I.D. and 42 in. long. All tests were made at atmospheric pressure with the vapor flowing downward inside the tube and countercurrent to the cooling water in the annular space. Data* are presented for a total of sixty-four experimental tests. The ranges of the variables covered in the investigation are given in Table 2. The interpretation of the data is presented in terms of a modified definition of a driving force involving the concept of availability of energy.

TABLE 2. RANGE OF VARIABLES INVESTIGATED

Systems	Water-air, water–carbon dioxide, water-helium, *n*-propyl alcohol–carbon dioxide
Flow rate	Total inlet mass flow rate of 0.198 to 1.78 lb./sec./sq. ft., or average Reynolds number of 1,585 to 15,000.
Inlet-gas temperature	118° to 185°F.
Cooling-water temperature	51° to 100°F.

EXPERIMENTAL EQUIPMENT

Figure 1 shows a flow diagram of the equipment used. The vapor was produced in an electrically heated boiler (*A*) and mixed in the boiler with a measured stream

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*Table 1 has been deposited as document 5766 with the American Documentation Institute, Photoduplication Service, Library of Congress, Washington 25, D. C., and may be obtained for $1.25 for photoprints or $1.25 for 35-mm. microfilm.

TABLE 3. TYPICAL WATER-AIR RUN
(Run A31)

Pressure in condenser	754.5 mm. Hg
Water flow rate	38.0 lb./min.
Water temperature in	69.6°F.
Water temperature out	70.6°F.
Water temperature rise (by differential thermocouple)	0.92°F.
Condensate flow rate	1.77 lb./hr.
Condensate temperature	99.6°F.
Temperature of gas-vapor mixture entering condenser	134.3°F.
Temperature of gas-vapor mixture leaving condenser	110.3°F.
Superheat of gas-vapor mixture leaving condenser	Negligible
Air flow rate	0.974 lb. mole/hr.
Wall temperatures	
7 in. from bottom of condenser	73.5°F.
14 in. from bottom of condenser	75.5
21 in. from bottom of condenser	77.0
28 in. from bottom of condenser	79.1

of noncondensing gas. The gas-vapor mixture then passed through a separator (*B*), where any condensate was removed before the stream entered the test condenser. The piping between the boiler and separator was left uninsulated, so that any superheat in the gas leaving the boiler would be removed before the mixture reached the separator. The mixture was cooled and condensed inside a single-tube vertical test condenser (*C*). The condensate was separated from the uncondensed gas-vapor mixture in the condensate separator (*D*) and collected in the condensate reservoir (*E*).

The uncondensed mixture was sent to the total condenser (*F*), where it was cooled somewhat below room temperature. Separation of this condensate from the noncondensing gas occurred in a separator (*G*) in the outlet of the total condenser. The combined condensate from condensers (*C*) and (*F*) was collected in a reservoir (*H*) and pumped back to the boiler by a small centrifugal pump (*I*). The noncondensable gas was recycled by a blower (*J*). A 4.5-cu. ft. gasometer (*K*) provided gas storage and pressure control.

The flow rate of gas to the boiler was measured by a rotameter (*L*); the cooling water flow was measured by a calibrated orifice (*M*) and checked by weighing for most runs. Temperature measurements were made by thermometers in the gas stream leaving the rotameter, in the gas-vapor mixture leaving the boiler, and in the exit streams from the separators (*D, G*). The temperatures of the gas-vapor mixtures entering and leaving the test condenser were measured by copper-constantan thermocouples (*TC*). Thermocouples were also installed in the inlet and exit cooling water and in the condensate leaving the test condenser. Pressure measurements (*P*) were made at the gasometer, at a point just beyond the rotameter, and at the inlet and outlet ends of the test condenser.

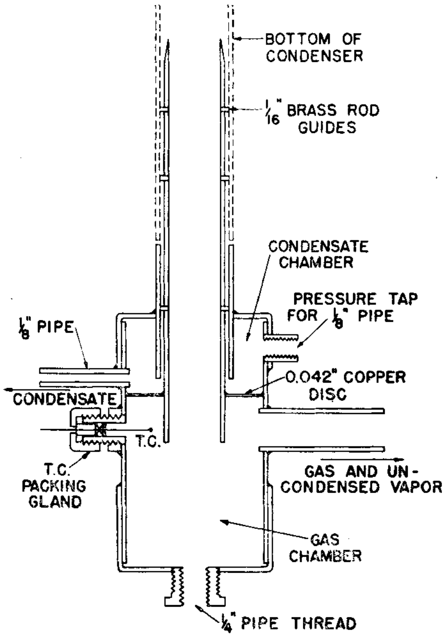

Fig. 3. Separator.

The piping from the boiler to the inlet separator was 1½-in. standard brass pipe, and from the inlet separator to the condensate separator the piping was 1⅛-in. O.D. copper tubing. All other gas-vapor lines in the recirculation system were constructed of ⅝-in. O.D. copper tubing. The test condenser and the separators (*B*) and (*D*) were thoroughly insulated.

Figure 2 shows a diagram of the test condenser consisting of a section of 1⅛-in. O.D. (0.995-in. I.D.) type-K copper tubing 48 in. long inside a section of 2⅛-in. O.D. (1.981-in. I.D.) type-L copper tubing. The actual heat transfer area was 0.912 sq. ft. (outside area of 1⅛-in. tube). Thermocouples were embedded in the wall of the inner tube at various locations in the condenser and also installed in the annular space midway between the two tubes in the cooling-water stream. (No use was made of the data from these cooling-water thermocouples, and the temperatures indicated by the wall thermocouples were used only to estimate the temperature of the condensate leaving the condenser.)

A tightly fitting rubber tube was used to join the butted ends of the test condenser and the interconnecting piping. A close fit between the ends of the copper tubing, resulting in minimum possible disturbance to the flow patterns, was effected by grinding. A length of 10 in., between the top of the test section and the adjacent elbow, was considered sufficient to ensure a fully developed flow pattern of the gas-vapor mixture entering the condenser.

The separator at the bottom of the condenser test section was constructed as shown in Figure 3. A ⅞-in. O.D. tube, centered by soldering $\frac{1}{16}$-in. brass-rod guides, was inserted into the condenser tube, its tapered end coinciding with the end of the heat transfer section. The uncondensed gas mixture passed through the ⅞-in. tube, into the gas chamber of the separator, and thence through the exit line. The condensate flowed in the annular space between the two tubes, into the condensate chamber, and then to the condensate receiver. (Another separator design was being used at the time the first twenty-two runs with the carbon dioxide–water system were being made. In that design the condensate and the exit gas-vapor mixture remained in contact for a relatively brief period after leaving the end of the condenser section. The heat exchange which occurred during this brief period was found to be appreciable. Although the magnitude of this exchange could be estimated from the data available, as was done in interpreting the data on the carbon dioxide–water runs, it was felt that a more satisfactory procedure would be to eliminate this heat exchange by the separator design already described.)

Cooling water maintained at a controlled uniform temperature was pumped from a 400-gal. mixing tank. A controller actuated a solenoid valve, which in turn regulated the flow of steam into the tank of well-stirred water. The temperature variation of the tank water was less than 0.3°F.

All thermocouples were checked against a Bureau of Standards calibrated thermometor in a water bath at room temperature and in a current of steam at atmospheric pressure. The steam temperature was calculated from the barometric pressure. In each case the cold junction was immersed in a water-ice bath. The thermocouples checked within 0.1°C. at both temperatures. The tube-wall thermocouples were calibrated in place against the water thermocouples. This was done first with cooling water flowing in the annular space and no flow in the condenser tube, that is, with no heat transfer, and then with steam flowing through both the annular space and the tube. The wall thermocouples checked within 0.05°C. of the water thermocouples.

The temperature rise of the cooling water in passing through the condenser was determined by a double differential thermocouple in the inlet and outlet water streams. This consisted of two thermocouples in each stream, all connected in series.

The electromotive force of the thermocouples was determined by a self-balancing electronic potentiometer with a sensitivity of 0.001 mv. (0.05°F.). Except for the differential thermocouple no attempt was made to read temperatures closer than 0.1°F. because of the occurrence of random variations in some of the temperatures. These were particularly noticeable in the case of high heat transfer rates, amounting to as much as 0.3° to 0.5°F. The temperature values recorded are averages.

EXPERIMENTAL PROCEDURES

The equipment was tested first by several runs involving simple heat transfer. Cooling water flowed in the annular space, and hot air or water through the inner tube. Heat transfer coefficients were determined from the results of these tests. The inside gas film coefficients were calculated for the air runs, and both the inside and outside coefficients for the water runs; this was possible because the tube-wall and cooling-water temperatures were measured at various points in the condenser.

The inside film coefficients of heat transfer for both air and water agreed within a few per cent with the values calculated by the Dittus-Boelter equation. The maximum deviation from the calculated values was

13.5%. However the average heat transfer coefficient for the cooling water in the annular space was 50 to 70% higher than the expected value, and the instantaneous coefficient seemed to vary considerably through the length of the condenser. The high values of the outside transfer coefficients can be accounted for by the relatively large effect of the entrance turbulence on the flow pattern in the short condenser used.

A detailed description of the operating procedure is not essential to this presentation. It may be noted that the data were taken over 30-min. intervals. Each run was continued until the data on successive 30-min. intervals indicated that steady state conditions prevailed.

When a run was begun with carbon dioxide or helium, some of the recirculating gas was vented to the air until tests of the gas indicated that it was essentially pure. In the helium runs the gas was analyzed for oxygen, and the quantity of air present was calculated from this result. To analyze the gas for the carbon dioxide runs a sample was diluted with an equal volume of air and then analyzed for carbon dioxide by absorption in potassium hydroxide solution. No difficulty was encountered in reaching an indicated purity of the recirculating gas above 99%.

For many of the tests the dew point of the gas-vapor mixture leaving the test condenser was measured. This was done by bringing a polished dew-point cup filled with hot water in contact with some of the gas leaving the condensate separator. As the water in the cup cooled, the temperature at which condensation first occurred on the polished surface was noted. For comparison with the dew point the temperature of the gas leaving the separator was also measured, since this was usually slightly different from that in the separator.

EXPERIMENTAL DATA

The results of the experimental tests are summarized in Table 1.* Data on a total of sixty-four tests are presented for the systems water-air (fourteen tests), water–carbon dioxide (twenty-five tests), water-helium (fourteen tests), and *n*-propyl alcohol–carbon dioxide (eleven tests). The results of a typical water-air run are given in Table 3. The variables studied were flow rate, vapor concentration, and cooling-water temperature. The gas-vapor mixture was in transition or turbulent flow for the majority of the tests but approached the viscous range in a number of cases, particularly in those tests in which helium was used as the carrier gas. All tests were made with the vapor mixture flowing downward inside the condenser tube and with cooling water flowing upward in the annular space. In all cases the pressure of the gas-vapor mixture was substantially atmospheric. The cooling-water rates were such that the Reynolds number for the annular space ranged from 3,400 to 5,300.

The first twenty-two tests on the water–carbon dioxide system were made by using a condensate separator in which there was appreciable heat transfer occurring between the gas-vapor mixture and the condensate beyond the bottom of the condenser. However, since the temperature of the condensate leaving the separator was measured, it was possible to calculate the magnitude of this effect and thus to estimate the condition of the gas-vapor mixture leaving the bottom of the condenser. This was done by making an enthalpy balance over the condensate separator and solving graphically for the temperature of the gas mixture leaving the condenser. For the purposes of this balance the temperature of the condensate leaving the condenser was assumed to be very close to the tube-wall temperature at this point, and the gas-vapor mixture was assumed to be saturated.

The flow rates of air and of carbon dioxide were determined by the rotameter and the manufacturer's calibration, corrected for the change in specific gravity. However in the case of the helium runs it was found that this procedure did not provide accurate values of the gas flow rates. The helium flow rates were therefore calculated from the quantity of condensate collected and the assumption of inlet and outlet saturation of the gas-vapor mixture. A subsequent partial calibration of the rotameter with helium at low flow rates, with a gasometer used as the primary standard, indicated that this procedure was quite satisfactory; it was further justified by the satisfactory nature of the material balances made on the other systems (by comparing the measured rate of condensate flow with the value calculated from data on the gas-vapor stream). The results of these material balances are shown in Table 1. The average ratio of the measured rate of condensate flow to the calculated rate was 1.01 for the air-water runs, 1.06 for the carbon dioxide–water runs, and 1.00 for the carbon dioxide–*n*-propyl alcohol runs. The higher value for the carbon dioxide–water system indicates that the mixture leaving the condenser was slightly superheated. Dewpoint measurements confirm this for three of the tests, but these data were not available for the first twenty-two runs of this series.

The consistency of the data was further checked by means of heat balances. The quantity of heat absorbed by the cooling water was calculated from the measured flow rates and the readings of a differential thermocouple in the inlet and outlet water lines. The heat given up by the gas-vapor stream was calculated from the known inlet and outlet conditions of the gas-vapor stream together with the measured flow rate of the condensate and an estimated value of the temperature of the condensate leaving the bottom of the condenser. (The data include a measured condensate temperature, but this temperature was taken at the outlet of the separator and was different from the temperature of the condensate leaving the bottom of the condenser.) In this calculation the heat content of the condensate is small compared with that of the exit gas-vapor mixture. It was assumed that the condensate left the condenser at the temperature of the tube wall. The error introduced by this assumption is not more than 1% in the calculated quantity of heat transferred from the gas-vapor mixture. The results of these heat balances are shown in Table 1. The average ratio of heat absorbed by water to heat given up by the gas-vapor mixture is 1.00 for the air-water runs, 1.12 for the carbon dioxide–water runs, 1.07 for the helium-water runs, and 1.29 for the carbon dioxide–propyl alcohol runs. When the difficulty of measuring the small change in temperature of the water stream is considered, these results are satisfactory for the first three systems. No explanation has been found for the larger discrepancy in the last system.

INTERPRETATION OF DATA

Interpretation of the body of data presented in this work might be approached in several ways. The method

TABLE 4. EQUIVALENT TEMPERATURE DRIVING FORCE

Saturated Gas-Vapor Mixtures
Reference temperature 492°R.
Total pressure 1 atm.

Values of Δ in °R. units

Temperature	Water-air	Water-carbon dioxide	Water-helium	*n*-Propyl alcohol-carbon dioxide	Benzene air
32	0	0	0	0	0
40	14.4	12.8	17.0	11.9	41.7
50	40.2	34.6	48.9	30.5	87.7
60	64.9	55.8	79.1	52.8	145
80	144	120	180	113	279
100	280	230	357	210	450
120	497	404	632	352	669
140	841	689	1,052	550	894
160	1,367	1,146	1,648	803	1,130
170					1,270
180	2,145	1,880	2,480	1,095	
200	3,310	3,120	3,495		

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*See footnote on page 413.

chosen here is based on the use of a driving force quite different from that ordinarily used. A method of correlating the data which employed a single driving force for the transfer of energy, similar to the one suggested by Colburn (*8*) and others, would be desirable. The success of the use of enthalpy as a potential for heat and mass transfer, even if limited to special cases, was encouraging; however enthalpy is not the correct driving force for mass transfer except for low concentrations of water vapor in air.

The resistance of a gas film to both heat and mass transfer is determined by the flow conditions and the physical properties of the system. The heat and the mass transfer coefficient are related, and most factors which affect one also affect the other. The question might then be asked, Is there a measure of driving force other than temperature, partial pressure, or enthalpy which could be used to measure the tendency for both heat and mass transfer? It seemed conceivable that such a driving force might exist, and the correlation of the data here is an attempt to find it.

Since any spontaneous process tends to result in a decrease in the availability of energy, the availability function was investigated as a possible driving force for both heat and mass transfer. It was assumed that the driving force was the difference in some function of the availability of energy between the main gas stream and the interface:

$$dq_T/dA = C[f(B) - f(B_i)] \tag{5}$$

The coefficient C would be expected to be a function of the physical properties of the stream, the geometry of the flow channel, and the mass flow rate.

The simplest function of availability is the availability itself. Interpretation of the data based on the availability of energy as a driving force was attempted first. In the case of air-water runs it was noted that the values of C calculated for runs in which dry air was used (heat transfer by convection only) were very nearly the same numerically as those calculated for runs in which condensation was occurring, provided comparisons were made at like conditions of flow rate and temperature level. To eliminate this effect of temperature level it was decided to try another function of the availability:

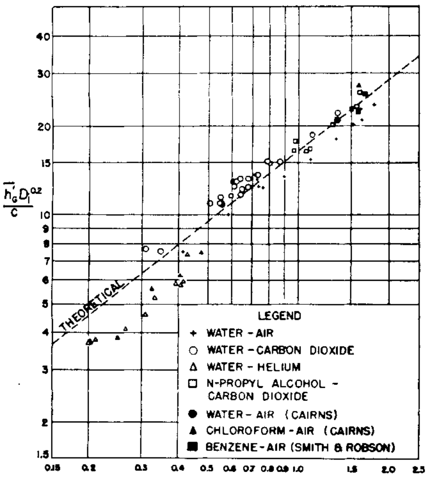

Fig. 4. Correlation of data.

Table 5. Comparison of Colburn and Hougen and Equivalent-Temperature Methods

Gas temperature, °F.	Colburn and Hougen t_i, °F.	Colburn and Hougen dq_T/dA $\frac{\text{B.t.u.}}{\text{(hr.)(sq. ft.)}}$	Colburn and Hougen ΔA, sq. ft.	This Work t_i, °F.	This Work dq_T/dA $\frac{\text{B.t.u.}}{\text{(hr.)(sq. ft.)}}$	This Work ΔA, sq. ft.
203	200.5	18,550	0	195.8	17,600	0
199.5	194.0	22,700	139.3	187.6	20,200	153.0
194	183.8	23,500	87.1	179.0	18,860	104.0
185	163.4	20,700	81.0	159.0	19,440	94.2
167	130.0	13,280	85.4	131.0	17,780	78.7
149	107.6	7,800	69.5	107.6	8,750	58.3
131	94.3	4,510	64.6	95.7	4,980	56.8
113	85.2	2,400	100.4	87.1	2,610	89.5
104	82.2	1,710	67.7	82.7	1,790	63.5
Total			695.0			698.0

$$\Delta = \frac{B_{nv}}{B_n}(T - T_0) \tag{6}$$

If this driving force is applied to a case in which sensible heat transfer alone is involved, it will be noted that $\Delta = T - T_0$. Thus the driving-force difference is

$$(\Delta - \Delta_i) = (T - T_0) - (T_i - T_0) = T - T_i \tag{7}$$

The rate of heat transfer for a noncondensing gas is then

$$dq/dA = C(\Delta - \Delta_i) = C(T - T_i) \tag{8}$$

Comparing this equation with the usual equation for transfer of sensible heat one will see that $C = h_G$ for the case of a noncondensing gas. For the case of transfer of both latent and sensible heat the coefficient C based on the equivalent temperature driving force will be denoted by h_G'.

For the case of saturated gas-vapor mixtures it has been shown (*17*) that

$$\Delta = (T - T_0) + \frac{RT_0\left[n_v \ln\frac{P}{P_0} - \ln\frac{\pi - P_0}{\pi - P}\right]}{nC_p\left[1 - \frac{T_0}{T - T_0}\ln\frac{T}{T_0}\right]} \tag{9}$$

For a superheated gas-vapor mixture it was assumed that

$$\Delta_{sup} = \Delta_{sat} + (T_{sup} - T_{sat}) \tag{10}$$

Values of Δ for mixtures of a particular gas and a particular vapor can be calculated from Equation (9), if data on vapor pressures and molal heat capacities are available. The results of such calculations (*17*) for several systems are given in Table 4.

To make use of this driving force it is necessary to know the interface temperature, which can be determined by noting first that

$$dq_T = h_G'(\Delta - \Delta_i)dA = h_{0i}(t_i - t_L)dA \tag{11}$$

$$\frac{h_{0i}}{h_G'} = \frac{\Delta - \Delta_i}{t_i - t_L} \tag{12}$$

Solution of this equation for t_i and Δ_i is by trial, the required second equation being given by the relation between Δ_i and t_i defined by Equation (9). This can also be done directly by plotting Equation (12) on the Δ vs. t diagram.

Values of the rate coefficient h_G' for the experimental runs were determined from the equation

$$dq_T = h_G'(\Delta - \Delta_i)dA \tag{13}$$

Several simplifications were possible in this work. The data indicated quite clearly that the major resistance to heat transfer was in the gas phase, and so Δ_i did not differ greatly from Δ_L in most cases. The variation in mass flow rate through the condenser was small enough that an average value of G could be used. Since it was assumed that in any one run h_G' was variable only to the degree that G and c vary, an average value of the heat transfer coefficient $\bar{h}_G'$ for each run was used. After Equation (13) had been integrated graphically a number of times, the results were compared with those obtained by using a logarithmic mean driving force and found to be nearly identical.

The calculated values* of $\bar{h}_G'D_i^{0.2}/c$ are presented in Figure 4. The experimental data of Cairns (*5*, *6*) and one run of Smith and Robson (*16*) were correlated by using the equivalent temperature driving force, and the results are shown on this plot. Also shown is the line representing the equation

$$h_GD_i^{0.2}/c = 16.6(G')^{0.8} \tag{14}$$

This equation is recommended (*14*) for estimating values of the gas-film heat

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*Sample calculation is given in reference (*17*).

transfer coefficient for gases in turbulent flow in pipes.

It is apparent from Figure 4 that the use of the driving force proposed in this work leads to values of a transfer coefficient which agree with an accepted empirical equation for correlating data on sensible heat transfer coefficients of gases. The agreement between experimental values and those calculated by Equation (14) is quite good (average deviation less than 10%) for the systems water-air, water–carbon dioxide, and *n*-propyl alcohol–carbon dioxide. The experimental points for the helium-water system lie appreciably below the empirical correlation (average deviation −22.5%). Two reasons can be given for the lower values in this case: The Reynolds numbers for the helium runs were low enough to indicate viscous flow in many cases, whereas Equation (14) is applicable in the turbulent flow region; and the experimental data indicate that the assumption of the equality of Δ_i and Δ_L is not as good in the case of the helium-water runs as it is in the other systems.

SOLUTION OF COLBURN AND HOUGEN'S EXAMPLE BY EQUIVALENT TEMPERATURE METHOD

The problem presented by Colburn and Hougen (*10*) in their classical design procedure was solved by the equivalent temperature method outlined here. This problem involves the condensation of water at 1 atm. from a mixture of nitrogen and water saturated at 95°C. The gaseous mixture is cooled to 40°C. with cooling water available at 25°C. and leaving the condenser at 60°C. Several calculated quantities used by Colburn and Hougen were also used in this solution: the quantity of heat transferred in the condenser at various points, the heat transfer coefficient for the combined resistances other than the gas film $h_{0i} = 310$ (assumed constant), and the value of the gas film coefficient at various points.

It was assumed that the gas-vapor mixture was saturated at all points in the condenser.

Interface temperatures and driving forces at the interface were calculated by using Equation (12). The area was calculated by using

$$A = \int_0^a \frac{dq_T}{h_G'(\Delta - \Delta_i)} \qquad (15)$$

Table 5 compares the results obtained by this method with those obtained by Colburn and Hougen.

The very close agreement between the total areas calculated by these two methods is a coincidence because of the particular temperature range covered. It should be noted that the Colburn and Hougen method predicts a somewhat more rapid rate of heat transfer at high vapor concentrations then does the equivalent temperature method, while the reverse is true at low concentrations; however agreement between the two methods is reasonably good for either condition over the range covered. The proposed equivalent-temperature procedure is easier to use and requires less time than the Colburn and Hougen method [even with the modifications proposed by Bras (*5*)], if the Δ vs. temperature data are available.

CONCLUSION

The results of this investigation indicate the possibility that the rate of gas-phase heat and mass transfer may be proportional to the heat transfer coefficient and a single driving force. The driving force suggested here is a function of the difference in availability of energy. While the proposed procedure has successfully correlated experimental data covering a wide range of conditions, it is believed that considerable additional testing of the method against other experimental data is needed. Reliable mass transfer data for gas mixtures containing a high concentration of vapor would be especially useful.

NOTATION

A = area of heat transfer surface, sq. ft.
B = availability of energy, $\Delta H - T_0 \Delta S$ (see *11*)
B_n = availability of energy of noncondensing gas having same physical properties as gas mixture from which heat is being transferred
B_{nv} = availability of energy of saturated gas-vapor mixture
C = general transfer coefficient
C_p = molal heat capacity at constant pressure $(C_{pn} + n_v C_{pv})/(1 + n_v)$
c = specific heat
D = diameter, ft.
D_i = diameter, in.
D_v = diffusion coefficient, sq. ft./hr.
$f(B)$ = function of availability of energy
G = mass velocity, lb./(hr.)(sq. ft.)
G' = mass velocity, lb./(sec.)(sq. ft.)
H = enthalpy, B.t.u./lb. mole
ΔH = enthalpy relative to T_0
h_G = gas-film heat transfer coefficient, B.t.u./(hr.)(sq. ft.)(°F.)
h_G' = instantaneous gas film heat transfer coefficient based on total heat transferred (sensible plus latent) and equivalent temperature driving force, B.t.u./(hr.)(sq. ft.)(°F.)
$\bar{h}_G'$ = average gas-film heat transfer coefficient based on total heat transferred (sensible plus latent) and equivalent temperature driving force, B.t.u./(hr.)(sq. ft.)(°F.)
h_{0i} = heat transfer coefficient for combined resistances other than gas film, based on inside surface, B.t.u./(hr.)(sq. ft.)(°F.)
k = thermal conductivity, B.t.u./(hr.)(sq. ft.)(°F./ft.)
k_G = gas film mass transfer coefficient lb. mole/(hr.)(sq. ft.)(atm.)
ln = natural logarithm
M = molecular weight
n = number of moles = $1 + n_v$
n_v = number of moles of vapor per mole of noncondensable gas
P = vapor pressure, atm.
p = partial pressure, atm.
P_{nm} = mean partial pressure of noncondensing gas, atm.
q_T = rate of flow of sensible heat plus latent heat, B.t.u./hr.
R = gas constant
S = entropy, B.t.u./(lb. mole)(°F.)
ΔS = entropy relative to T_0
T = absolute temperature, °R.
t = temperature, °F.
W = weight flow rate, lb./hr.

Greek Letters

Δ = finite difference when followed by a function
Δ = equivalent temperature driving force
μ = viscosity, lb./(hr.)(ft.)
π = total pressure, atm.
ρ = density, lb./cu. ft.
λ = latent heat, B.t.u./lb.-mole

Subscripts

i = interface
n = noncondensing gas
0 = datum state
L = liquid or cooling water
sat. = saturated
sup. = superheated
v = vapor

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