

quick-frozen packaged products. This precludes containers from such materials as cork board, balsa wood, and the various proprietary insulating materials.

Corrugated fiber board has been found to be a very satisfactory material from which to manufacture insulated shipping containers, and considerable quantities are now being used for that purpose. It has approximately the same heat-insulating value as cork board or balsa wood, is light, strong, and inexpensive. Fiber board may be made with either straw or chip corrugations, and with kraft, jute, chip, or straw liners. It has approximately the same heat-insulating value no matter from which materials it is made.

Although the construction and thickness of the corrugated containers used by different producers of quick-frozen products vary greatly, an excellent container may be made by

placing four pads or liners within an ordinary slotted carton of the proper dimensions (Figure 4). The container should be as nearly cubical as possible, so as to expose the minimum surface to heat penetration; and, when possible, the corrugations in the liners should run horizontally so as to lessen convection currents. Obviously, the shipping container should be packed solidly full, because the heat leakage into the container is approximately in proportion to its surface area, whereas the amount of heat which can be absorbed by the product with a given temperature rise is in proportion to the volume of product (Figure 5).

Figure 6 indicates the effectiveness of corrugated fiber-board cases in protecting their contents from outside heat. It will be noted that the contents of the 50-pound container required 7 days to reach a temperature of 50° F., at which time the fish was still perfectly fresh.

The Condensation of Steam¹

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An apparatus has been built for studying the effect of temperature, concentration of small amounts of air, and temperature drop on the rate of condensation of steam on an isothermal condensing surface.

Within the experimental range and errors, the following empirical equations define the coefficient as determined by these individual independent variables:

$$f = a(\Delta T)^b$$

$$\frac{1}{f} = dC + e$$

$$f = Fg^T$$

where f is coefficient, ΔT is temperature drop, C is com-

position expressed as per cent air by volume, T is temperature in degrees Fahrenheit, and the other letters represent empirical constants. These individual equations have been combined with certain simplifying assumptions into a single equation:

$$\log f = \log \Delta T [1.213 - 0.00242T] + \left[\frac{\log \Delta T}{3.439} - 1 \right] \times [\log(C + 0.505) - 1.551 - 0.0097T]$$

Several theoretical reasons for the variations indicated by the equations have been pointed out in the light of the resistance concept.

The process of condensation of impure steam and that of gas absorption have been compared.

IT HAS been realized for almost a century that the rate of heat transfer between fluids and solids is dependent almost entirely on the physical properties of the fluids which cause them to form relatively stagnant films at contact surfaces. However, experimental work has not always been planned so that the mechanism of heat transfer through one film could be investigated independently, and in studies of the condensation of vapors there has usually been included another physical operation, the heating of water. In these reports references have been made to the amount of air present, but nowhere has its effect been closely scrutinized.

Reynolds^{1,*} in referring to air in condensing steam, wrote:

A priori it seemed probable that it might retard condensation very much; for when pure steam comes up to a cold surface and is condensed, it leaves an empty space which is immediately filled with fresh steam; so that the passage of steam up to the cold surface is unobstructed.*** If, however, the steam is mixed with air, then the steam will be condensed and the air be left between the fresh steam and the cold surface; so that after condensation has commenced that surface will be protected by a stratum of air, and fresh steam will have either to displace this or pass through it before it in turn can be condensed.

Later work^{2 to 18} has not changed the conclusions of Reynolds and other early experimenters, and many reports are to be regarded more as efficiency tests on specific commercial apparatus than as sources of data of general interest. Numerous studies have been made, but they have not satisfactorily determined the effect of non-condensable gas, temperature, and temperature drop from steam to tube (or amount of heat flowing) on the rate of condensation of steam. However, the literature shows that the coefficient varies between 1500 and 3500 (B. t. u. per square foot per hour per ° F.), and it is the purpose of the present work to show the quantitative effect of these conditions on this variation.

Apparatus

BOILER—The ends of a 38-inch section of standard 12-inch iron pipe were closed by welding on steel disks fitted with valves, packing boxes, etc., as shown in Figure 1. The inner surface was nickel-plated after the ends were machined for the fittings.

VAPORIZATION SPACE AND CONDENSER—If water is passed through a tube surrounded by condensing steam, the water is heated and hence the temperature of the tube is not constant. In order to avoid lengthwise variation in tube temperature, water was boiled inside the tube at a lower pressure than that of the steam outside, and the vapor formed was suitably condensed and the condensate returned. (To prevent ambiguity the outer 12-inch shell will be hereinafter

¹ Abstracted from a thesis submitted in partial fulfillment of the requirements for the degree of doctor of philosophy at the University of Michigan. Presented before the Division of Industrial and Engineering Chemistry at the 76th Meeting of the American Chemical Society, Swampscott, Mass., September 10 to 14, 1928.

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* Numbers in text refer to bibliography at end of article.

identified as the "boiler," and the tube system as the "vaporization space.")

The construction of this evaporator tube, its auxiliary condenser, and fittings is indicated in Figure 1. The tubular members were brazed together and the 3-inch o. d. by 47-inch evaporator tube was soldered to thin copper collars which absorbed expansion strains. These collars were soldered to the boiler ends. The tube was fitted with thermocouples previously attached by a nickel-plating process already described.¹⁹

COOLING-WATER SYSTEM—A constant-level tank (V, Figure 1) was installed three floors above. After passing the valve (M) and before entering the two $\frac{7}{8}$ -inch tubes, the water flowed by a thermocouple junction and a thermometer. After the two $\frac{7}{8}$ -inch tubes, the water flowed back through the jacket, then by the other junction of the thermocouple and to the orifice chamber.

An orifice chamber of 2.5 gallons capacity (Y, Figure 1, and Figure 2) was constructed as shown. Water rose in the open manometer attached to the can and indicated the hydrostatic pressure above the orifice plate. The orifice chamber was usually operated half filled with air, so that the compressibility of the gas would smooth any slight fluctuations of pressure. The calibration curve of the rate of flow against manometer readings was a straight line on logarithmic paper.

The three systems of water used in the apparatus may be

zation tube would boil toward one end, the boiler rested in its steel and asbestos setting at an angle of about 5 degrees with the horizontal (not shown in Figure 1). Two gas heaters were made by drilling small holes in two sections of pipe supplied with city gas and compressed air (D in Figure 1). The hot gases passed through a hood to the stack.

The two 2000-watt General Electric heaters (B, Figure 1) were installed as auxiliary and more readily adjustable sources of heat. Their small heat capacity and high rate of heat conduction to the water allowed an almost instantaneous change in the steam temperature to be produced by a small change in the current. The boiler temperature could be kept constant within 0.01°C . for days at a time by means of a thermostatic regulator described in another article.²⁰

AIR MANOMETER—A device for the analysis of the gas phase *in situ* was necessary. The partial pressure of air might be found by subtracting the vapor pressure of water from the total pressure, but the multiplicity of corrections and calibrations necessary to obtain this small difference accurately were undesirable. Other possible methods^{21,22} were also discarded and an apparatus was devised to indicate the partial pressure of the air directly on a manometer.²³

THERMOCOUPLE SYSTEM—The accuracy of the results depended largely on three measurements: (1) the temperature of the steam, (2) the temperature of the tube, and (3) the rise in temperature of the cooling water.

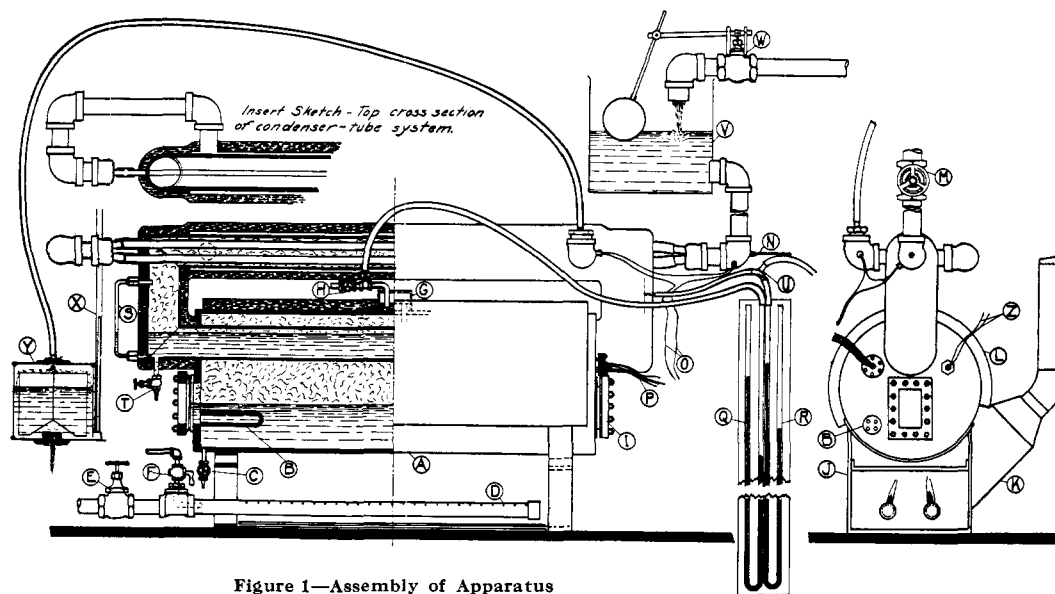


Figure 1—Assembly of Apparatus

A—Boiler body of 12-inch iron pipe
B—Electric heaters, 2 General Electric 2000-watt
C—Boiler drain valve
D—Gas heaters
E—Gas valve
F—Air valve
G—Water trap on manometer
H—Blow-off valve
J—Iron stand for boiler
K—Hood and flue for hot gases
L—Asbestos insulation
M—Valve regulating cooling water

N—Thermometer for entering cooling water
O—Water-difference thermocouple wires
P—Tube thermocouple wires
Q—Boiler manometer
R—Evaporative space manometer
S—Water gage in evaporative space
T—Evaporative space drain valve
U—Exhaust tube for evaporative space
V—Constant-level tank
W—Float valve on city water line
X—Water orifice manometer tube
Y—Orifice chamber
Z—Standard steam thermocouple wires

summarized: (1) the water boiling in the boiler, the steam from which condensed on the lower tube of the vaporization space in which (2) water was boiling at a lower temperature and pressure, the steam from which was condensed both on the inner wall of the upper 3-inch tube and the outer walls of the two $\frac{7}{8}$ -inch tubes through which (3) cooling water was circulated. The lower 3-inch tube was the one on which the heat transfer experiments were made, the upper tubes serving merely to discharge this latent heat to the cooling water, where it could be measured as sensible heat.

HEATING SYSTEM—In order that the water in the vapori-

The steam temperature and the rise in temperature of the cooling water were measured with thermocouples of copper-constantan. The steam thermocouple was an ordinary two-junction type, and the "water-difference" thermocouple was composed of two of these in series. These thermocouples were calibrated with an A. H. Thomas thermometer graduated to 0.1°C ., accurate to 0.02°C . and standardized by the Bureau of Standards. Their hot junctions were tied to the thermometer bulb and immersed in a liquid, the temperature of which could be varied. The cold junctions were immersed in melting ice in a Dewar flask. (The melting point of the

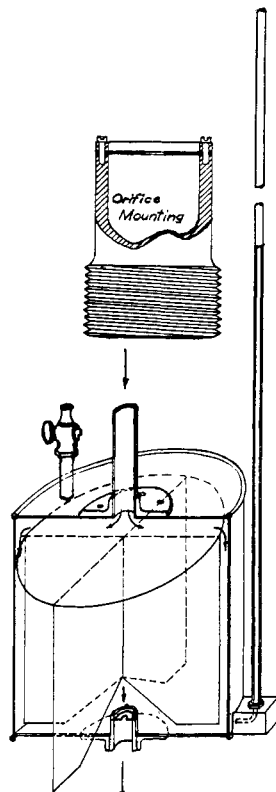


Figure 2—Orifice Chamber and Manometer

ice used in this and later work and that of ice frozen from conductivity water was compared with an accurate resistance thermometer and found to differ by less than 0.01°C . The free ends were connected by a selective switch to a special Type K potentiometer. The sensitivity of the galvanometer was such as to allow accurate readings to a microvolt. The standard cell was checked at intervals against an Eppley standard cell recently checked at the Bureau of Standards.

About fifty readings were taken in the desired range. The method adopted for using these and similar data was to plot them on sheets of millimeter graduated cross-section paper 50 cm. in width, as a series of nearly parallel lines, each of which represented 10° or 20°C . The thermometer used to measure the temperature of the entering cooling water as a zero-point compensator was graduated to half degrees, and after checking against the standard thermometer was installed next the thermocouple junction in the water line. The steam thermocouple was installed about 3 inches from the tube and was so shielded as to prevent loss of heat by radiation.

TUBE THERMOCOUPLES—The tube thermocouple system has been described.¹⁹ The three thermocouples not damaged in the installation checked closely and were connected in parallel. The boiler and tube were heated and maintained at a constant temperature by the electric heaters. This temperature was indicated by the standard steam thermocouple, and the e. m. f. of the three-tube thermocouples. About fifty sets of readings were taken between 70° and 120°C . On the resulting plots the farthest of the points was within 0.2°C . and the majority were within 0.1°C . of the curve which was drawn through them.

The temperature measurement desired in this work was that of the tube surface. In practice, the temperature as indicated by this calibration was the temperature of the nickel-copper interface and not that at the surface. The thickness of this nickel plate was known and the temperature gradient through it was found by Newton's law

$$q = kA \frac{\Delta T}{\Delta L}$$

where q is the rate of heat flow, k is thermal conductivity of nickel, A is the surface area, ΔT is temperature difference, and ΔL is thickness.

Note—This formula may be used instead of the logarithmic equation for the cylindrical flow of heat because the thickness of the plate is very small compared with the radius.

This correction was always evaluated and used in finding the temperature of the surface of the tube.

PROTECTION AGAINST HEAT LOSSES—The boiler above the setting was covered with a layer of magnesia insulation $1\frac{1}{4}$ inches thick. The vaporization space and its water jacket were similarly insulated, and both surfaces were given several coats of a varnish suspension of aluminum powder. Heat losses from the vaporization space were small, not only because the temperature difference between it and the room was small, but also because over half of the copper tubular rectangle outside of the boiler was water-jacketed, and the heat which otherwise would have been lost through the insulation was absorbed by the cooling water. In most cases the average temperature of the water was lower than room temperature, and the small amount of heat flowing into the cooling water from the outside would tend to balance the small amount flowing out from the rest of the surface.

Operation

CLASSIFICATION OF EXPERIMENTS—In a systematic study two of the three variables defining the coefficient must be kept constant while the effect of the third is studied. The temperature drop was altered more easily than either the temperature or composition of the steam, and it was changed during progressive experiments. When it also was fixed and a steady state was reached, the several readings were taken. Several of these runs taken at constant steam temperature and composition, but with different temperature drops, were called a "series." To find the effect of the temperature or composition it was necessary to hold one of these constant, with a different value of the other for each of several series. By suitable plotting and cross plotting, the coefficient could then be studied in its relation to any single one of the three variables.

REMOVAL OF AIR—The boiler was filled with distilled water which was boiled to expel dissolved gases. When about half had been vaporized, the exhaust valve was closed and the temperature maintained above 100°C . After a few hours the air manometer indicated the presence of a few hundredths per cent of a non-condensable gas, and it was impossible to prevent a slow evolution of this gas even after weeks of boiling.

Note—The absorption of air by water^{24,25} and the slowness of its evolution from a boiling liquid²⁶ have been mentioned by various writers. A liberation of absorbed gases from the surface film of water or, more prob-

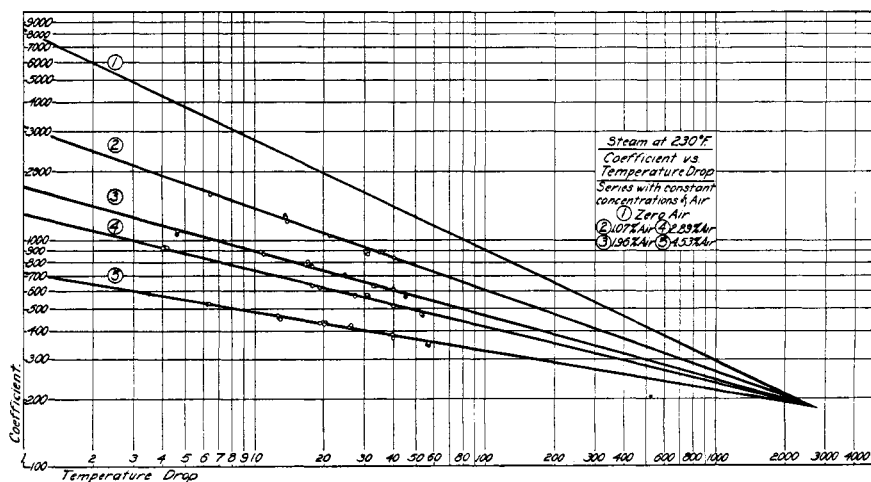


Figure 3

ably, the action of steam on iron incompletely covered by nickel to form hydrogen would account for that found in these experiments.

When gas-free steam was desired, a valve on the end (not shown in Figure 1) allowed a slight exhaust. This prevented the "air" from increasing to a measurable concentration during the time necessary to make a series. Air was ad-

mitted when desired by connecting this valve to a source of compressed air. The slight evolution of non-condensable gas was not consequential when a relatively large amount of air had been purposely introduced.

VAPORIZATION SPACE—The boiler conditions being fixed, the vaporization space was evacuated and the lower tube filled with water. Heat was supplied to this tube by the condensa-

minutes later established the value of the coefficient under the given conditions. The tables in this report include only such data which by duplication indicate the attainment of a steady state.

CHANGING THE TEMPERATURE DROP—Every series was made with the temperature drop as the single independent variable. The largest temperature drop of the series was that of the first run after which it was necessary to lessen it. Two methods were available—the first to reduce the flow of cooling water, and the second to diminish the coefficient of heat transfer from the low-pressure steam to the cooling water by introducing air into the vaporization space. This air would reduce the rate of heat transfer from the vaporization space to the condenser almost instantly, the temperature of the tube would immediately rise, and the heat supplied to the boiler would have to be diminished. A new temperature drop would be established and the second point on the curve of coefficient *versus* temperature drop could be obtained. This new temperature drop could not, however,

be arbitrarily set and exactly obtained, because it was not possible to know how much air to admit, but the amount (as shown by the manometer, *R* in Figure 1) could be roughly judged and the new temperature drop would be approximately that desired. A finer adjustment was sometimes made by changing the water velocity.

This work continued gave values of the coefficient as a function of the temperature drop. At the end of a series the condenser and vaporization space were drained of water, the air manometer was read, and the composition of the vapor calculated at thermal equilibrium.

DATA TAKEN—Probably twice as many runs as are here reported were made before the experimental difficulties were thought to be under control. These preliminary data have been disregarded, but all subsequent work is tabulated except

tion of steam on its outer surface and a part of the water was boiled off to remove the air present.

COOLING WATER SYSTEM—The water in the tank and lines was drained to waste until it came to a constant temperature. The flow was then diverted through the condenser tubes and adjusted to its proper value.

READINGS TAKEN—Heat was allowed to flow from the boiler to the vaporization space and thence into the cooling water until it was thought that a steady state had been reached. After a reasonable period, readings were made. These were always taken in the same order: (1) steam thermocouple, (2) tube thermocouples, (3) cooling-water differential thermocouple, and (4) water-orifice manometer. Throughout the series readings were also made on the entering cooling water thermometer and the air manometer. When there was no air in the boiler the air manometer was unchanged, but when air was present the readings during a run were so erratic as to be useless. The important readings of the air manometer were those taken before and after a series when there was no heat flowing through the tube wall. These were perfectly definite and their average was taken as indicative of the air present during the runs.

Note—Smith³ reported a similar variation of the amount of air and attributed the difference between that calculated at equilibrium and that calculated at a steady state of heat flow to large deviations from Dalton's law of partial pressures. Instead, there is probably a gradient of composition throughout the steam space of the boiler. Thus the total amount of air present could be found during a time when heat was flowing only by the integration of the composition at a large number of points. It must be noted in subsequent discussion that the average amount of air before the run is not the same as the average amount in the free space during a run since some will be held in a gas film around the tube. This work makes no distinction and its results are based on the total amount of air present.

After the first readings the apparatus was allowed to run 10 or 15 minutes before a second set. If these checked, a steady state had been reached and the coincidence of a third set 5

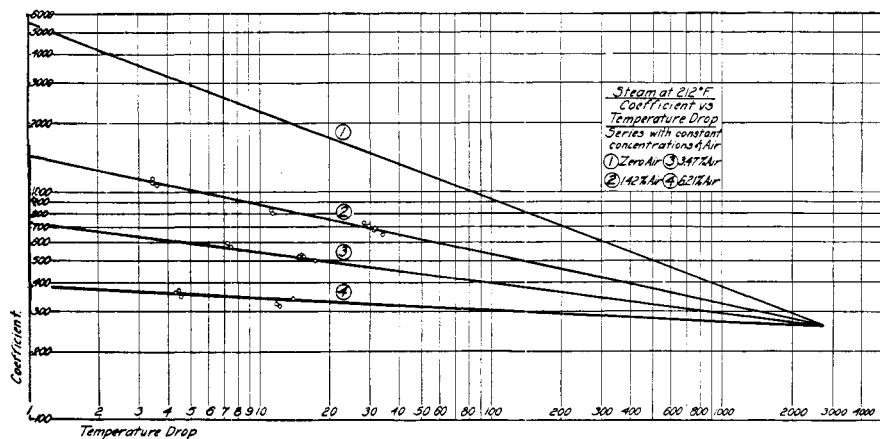


Figure 4

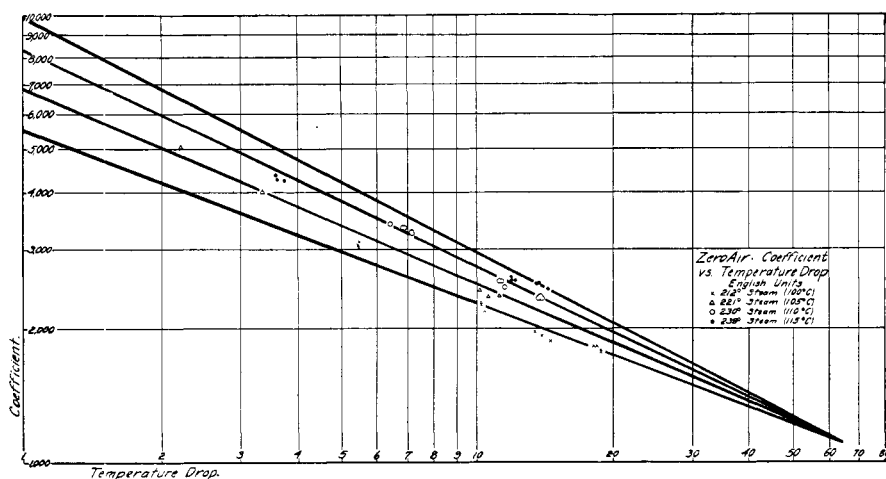


Figure 5

those individual readings taken before the attainment of a steady state. The three sets of series reported include (1) coefficient *versus* temperature drop at a very low or "zero" concentration of air and constant temperatures of 100°, 105°, 110°, and 115° C.; (2) coefficient *versus* temperature drop at a constant temperature of 100° C. and various constant percentages of air; and (3) the same for 110° C.

Sample Calculations

The results presented were calculated from the values read on the calibration curves with a 20-inch slide rule. The data taken were not all in one set of units, as is unfortunately the case rather often in engineering practice. The thermocouples were calibrated in Centigrade degrees and the water manometer in pounds of water per second. The potentiometer was read to the closest microvolt, and the manometer to the closest millimeter, although it was possible to estimate to one-tenth these values. The set of data shown in the tables for approximately 110° C. steam, 1.07 per cent air, and temperature drop of about 20° C. will have the calculations indicated.

The readings of the air manometer before and after the series and their arithmetic averages were respectively: 28.05, 27.65, and 27.85 cm. The zero point—i. e., the reading when there was no air in the boiler—was 29.10 cm. at 110° C. The difference between this and the average is 1.25 cm., which when corrected for the water above the mercury is $1.25 - 1.25/13.6 = 1.157$ cm. of mercury corresponding to the partial pressure of air in the boiler. Peabody's tables give the pressure of saturated steam at 110° C. as 107.45 cm. The total pressure in the boiler is the sum of the partial pressures, and the air present is thus

$$\frac{1.157 \times 100}{1.157 + 107.45} = 1.066 \text{ per cent}$$

Since the temperature of the steam was held within 0.1° C. by the thermostat, the average of the e. m. f.'s for a series was converted to degrees instead of using the temperature for each run individually. The values from the respective charts follow:

	° C.
Steam temperature	110.04
Interface temperature	89.93
ΔT	20.11
Rise in water temperature	22.50
Weight of cooling water, 0.507 lb. per second	

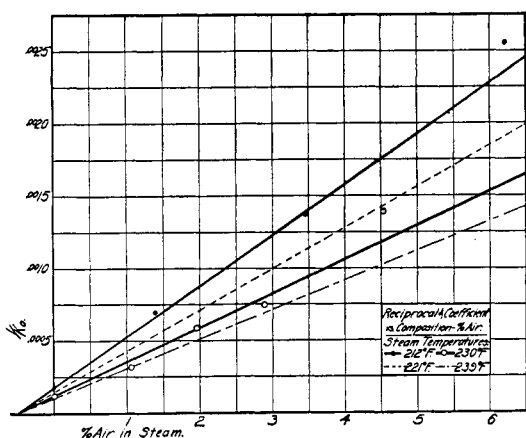


Figure 6

The term ΔT represents the difference between the temperature of the steam and that of the nickel-copper interface of the tube, and is corrected by subtracting the temperature drop through the nickel plate which is found, as mentioned above, from the relation $q = kA \Delta T / \Delta L$. The heating surface is 2.36 square feet (cylinder 35.75 × 3.025 inches) and the thermal conductivity of nickel is 0.1133 B. t. u. per square foot per second per ° F. per inch. q is the product of the weight of water per second flowing through the condenser by the increase in its temperature, or in English units

$$0.507 \times 22.50 \times 1.8 = 20.53 \text{ B. t. u. per second}$$

If the equation for heat flow is rearranged and the appropriate values substituted, the correction for the drop through the nickel is 0.43° C. and the corrected value of ΔT is $20.11 - 0.43 = 19.68$ ° C.

The steam film coefficient may not be calculated. It is equal to

$$f = \frac{q}{A \Delta T} = \frac{20.53 \times 3600}{2.36 \times 19.68 \times 1.8} = 883 \text{ B. t. u. per square foot per hour per } ^\circ \text{F.}$$

The relation

B. t. u. per hour per square foot per ° F. × 4.882 = kg. cal. per hour per square meter per ° C. and the reciprocal factor 0.2048 are often useful.

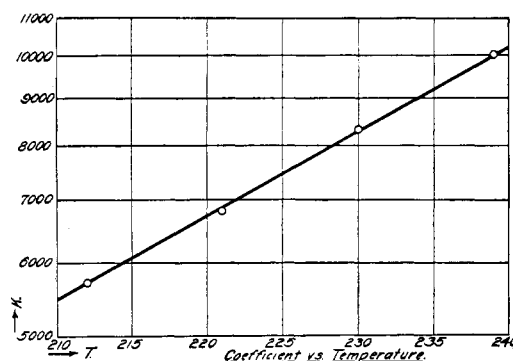


Figure 7

An Empirical Equation

Several empirical relationships between the coefficients obtained and the values of the independent variables were noticed: (1) With constant temperature and concentration the logarithm of the coefficient is a linear function of the logarithm of the temperature drop; (2) with constant temperature drop the reciprocal of the coefficient is a linear function of the composition expressed as per cent air by volume; (3) with constant temperature drop and concentration the logarithm of the coefficient is a linear function of the temperature.

The lines representing these data were drawn (Figures 3 to 7) without recourse to the method of least squares, but in relation to all of the data and not merely that of the individual series. These generalizations may, respectively, be expressed by the equations:

$$\log f = \log a + b \log \Delta T \quad (1)$$

$$\frac{1}{f} = sC + p \quad (2)$$

$$\log f = \log h + (\log F)T \quad (3)$$

where f is coefficient, ΔT is temperature drop, C is composition (per cent air by volume), T is temperature of steam in degrees Fahrenheit, and the other letters are constants to be evaluated from the geometric properties of the graphs. The logarithms in these and later equations are to the base 10.

Note—The points of Figure 6, used in (2) are obtained from values of the coefficient at 1° F. on the lines of Figures 3, 4, and 5. The points of Figure 7 used in (3) are obtained by reading values of coefficient at a temperature drop of 1° F. from Figure 5 where there is no air present in the steam. The third conclusion and Equation 3 is thus based only on experimental data for zero air. Values of the coefficient, when a measurable amount of air was present, were taken at only two temperatures of the steam and the third generalization may be definitely made only for the case where there is no air present in the steam. However, the final equation is applicable at these two temperatures to the data taken with air present.

Equation 2 and Figure 6 state that if the reciprocals of values read from the lines of Figure 3 or 4 are plotted against the per cent air, the resulting lines will be straight. If that is accepted, it may be shown that the lines of Figure 3 or 4 cannot converge to a common point although the intersections of any pair of lines may be infinitesimally close. Within the admittedly large experimental errors, this is of no consequence and since

the data indicate the possibility of such convergence and the mathematical handling is greatly simplified by its assumption, these lines are drawn as sheafs. This common intersection is, however, too far to the right to have any physical significance. The intersections have the same abscissas for both steam temperatures studied, but the ordinate of the higher steam temperature is the smaller. If the lines of Figure 5 representing different temperatures are extended through their pole, the intersections with the common ordinate of the poles of Figures 3 and 4 may be considered as poles of other sheafs. A plot of the ordinates of such intersections, constant A in Figure 8, is a straight line.

The equations may be combined if the geometric properties of the lines are considered, giving:

$$\log f = \frac{\log \Delta T [\log w + (\log u)T] + [\log (C + r) + \log(1/r) - \log h - T \log F]}{\log B} \times [\log \Delta T - \log B] \quad (4)$$

The constants in Equation 4 have been tabulated in English units.

CONSTANT	GEOMETRICAL SIGNIFICANCE	GRAPH	VALUE
$\log w$	Intercept	Constant A vs. temperature (Figure 8)	4.1737
$\log u$	Slope		-0.00832
r	Distance between intersection of $1/f$ vs. C lines and $1/f$ axis	Reciprocal of coefficient vs. composition (Figure 6)	0.505
$\log(1/r)$			0.2970
$\log h$	Intercept	Coefficient vs. temperature (Figure 7)	0.1848
$\log F$	Slope		0.00900
$\log B$	Logarithm of abscissa of intersection of lines of different compositions	Coefficient vs. temperature drop at constant T (Figures 3 and 4)	3.439

If these values are substituted in Equation 4 there follows:

$$\log f = \log \Delta T [1.213 - 0.00242 T] + \left[\frac{\log \Delta T}{3.439} - 1 \right] \times [\log (C + 0.505) - 1.551 - 0.009 T] \quad (5)$$

This relation expresses in B. t. u. per square foot per hour per ° F. the logarithm of the coefficient as a function of temperature, temperature drop, and composition of the steam. The constants are individually evaluated in the metric system and in kg. cal. per square meter per hour per ° C.

$$\log f = \log \Delta T [1.442 - 0.0047 T] + \left[\frac{\log \Delta T}{3.184} - 1 \right] \times [\log (C + 0.505) - 2.6568 - 0.0139 T] \quad (6)$$

Physical Discussion

Most problems dealing with potential gradients may be attacked by a method understood by workers in heat transfer and the analogous field of gas absorption to be based on the "resistance concept." Resistance is the reciprocal of conductivity, and the total resistance of several substances in series is the sum of the separate resistances, or in the transfer of heat from a condensing vapor to a solid:

$$1/f = 1/f_g + 1/f_c + 1/f_L$$

where f = coefficient from steam to tube

f_g = coefficient across gas film

f_c = coefficient at the condensation interface

f_L = coefficient across film of condensate

The mechanism of condensation may be studied by Langmuir's equation²⁷ for the condensation of metallic vapors. This may be combined with the Clausius-Clapeyron equation and the gas laws to give:

$$q = \sqrt{\frac{R}{2\pi M}} \times \frac{dP}{dT} \times (T_v^{3/2} - T_L^{3/2})$$

where q is the rate of heat transfer, T_v and T_L are absolute temperatures of the vapor and liquid sides of the interface, M is molecular weight, R is the gas constant, and dP/dT is the slope of the vapor-pressure curve which is assumed to be a straight line in this very narrow range. When the experimental values of q are substituted with the other quantities,

the temperature difference across the interface is found to be very small and the corresponding resistance negligible, which agrees with what Reynolds¹ said: "In fact there is no limit to the rate at which pure steam will condense but the power of the surface to carry off the heat." The total resistance may therefore be considered as due to the gas and condensate films, and the formula corresponding to Ohm's law might be written

$$\frac{q}{A} = \frac{\Delta T}{R}$$

where q/A is the rate of heat flow per unit of surface, ΔT is the temperature drop from steam to tube, and R is the sum of resistances of gas and liquid films. Few published experiments have been expressed in units of resistance.

The mathematical analysis of the mechanism of heat transfer through the liquid and vapor films is hindered by the large number of physical properties to be evaluated. Such things as the rate of mutual diffusion of steam and air; the viscosity, specific heat, thermal conductivity, and density of the air,

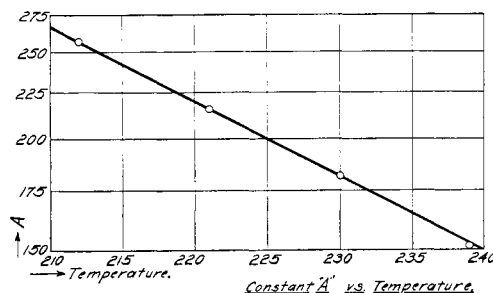


Figure 8

condensate, and steam; and the latent heat of the latter determine the amount of heat passed. These values all change with varying temperatures through the films, and the mean values can only be approximated. A knowledge of the thickness of the condensate film and of the air concentration and temperature at its surface would also be valuable. Because most of these data are not accurately known, a basic theory was not formulated for comparison with the experimental observations. However, the effect on the coefficient of a change of any one of the independent variables may be predicted and checked against the data.

The first relation which was considered was the linear function of the logarithms of the coefficient and the temperature drop. The graphs (Figures 3 and 4) start with a temperature drop of 1° F., but some data were taken with lower temperature drops. Although the errors of measuring these small temperature differences made the coefficients so unreliable that they are not plotted on the graphs, they indicate no change of the relationship.

Steam, in condensing, covers the tube with a film of water which is drained by gravity, but which increases in thickness as the rate of condensation increases. The resistance to heat transfer is increased and therefore the coefficient is decreased with increasing temperature drop. The condensate film is in viscous flow, and if the coefficients at zero air could be divided by the film thickness, the results would be approximately constant—i. e., the coefficient of thermal conductivity of the condensate. Since the coefficient is more easily determined than the film thickness, the latter may be calculated from the thermal conductivity of water which according to Jakob²⁸ varies as $k = 0.001325 (1 + 0.002984t)$, where k is the conductivity in calories per sq. cm. per second per ° C. per cm., and t is Centigrade degrees up to 80° C. Preliminary work in this laboratory has verified this equation up to 120° C.

The plot of the coefficients obtained with practically pure

steam (Figure 5) show comparatively large irregularities. The large amount of heat passed in these runs caused an uneven boiling in the vaporization space and a larger variation in tube temperature. Another reason was that a very slight amount of air makes a large difference in the coefficient when the steam is practically pure. The series at 212°, 221°, and 230° F. were made with a small amount of steam exhausting, to prevent a building up in the boiler of an appreciable concentration of non-condensable gas. The series for steam at 239° F. was obtained just after the steam had been freed from air by "boiling out," but with the exhaust valve closed during the series itself. The first points obtained on this last series are those with the highest temperature drops, and they are seen to be consistent with the lines of lower temperatures,

but those which were taken several hours later at lower temperature drops are considerably lower than expected. This shows that a slow evolution of non-condensable gas during a few hours had reduced the coefficient, and the line as drawn is based only on the points which were taken first, when the steam was substantially pure.

Some experimenters have indicated that a change of temperature of pure steam did not appreciably affect the coefficient. In the present work the coefficient is shown to increase with increasing temperature by the line in Figure 7 derived from the lines in Figure 5. This might be expected since the properties of water vary with temperature. At a given temperature drop the condensate drains faster and forms a thinner film when the viscosity is lowered by an increase of

Data

TUBE TEMP. ° C.	WATER DIFF. TEMP. ° C.	WATER Lb./sec.	ΔT ° C.	Δθ NICKEL ° C.	ΔT (COR.) ° C.	ΔT (COR.) ° F.	
AVERAGE TEMPERATURE OF STEAM 110.04° C.—1.066 PER CENT AIR							
89.93	22.50	0.507	20.11	0.43	19.68	35.42	883
89.93	21.86	0.521	20.11	0.43	19.68	35.42	882
92.46	33.70	0.295	17.58	0.37	17.21	30.98	881
92.46	33.43	0.295	17.58	0.37	17.21	30.98	873
92.95	17.98	0.542	17.09	0.36	16.73	30.12	888
92.85	18.00	0.540	17.19	0.36	16.83	30.30	881
98.01	14.82	0.542	12.03	0.30	11.73	21.12	1045
102.10	15.23	0.400	7.94	0.23	7.71	13.88	1205
102.20	15.69	0.400	7.84	0.23	7.61	13.70	1258
102.28	15.83	0.400	7.76	0.24	7.52	13.54	1284
102.17	16.58	0.402	7.87	0.25	7.62	13.72	1334
106.30	9.38	0.399	3.74	0.14	3.60	6.45	1585
AVERAGE TEMPERATURE OF STEAM 110.03° C.—1.96 PER CENT AIR							
84.35	17.34	0.538	25.69	0.35	25.34	45.28	561
84.35	17.61	0.540	25.69	0.36	25.33	45.28	572
84.57	17.29	0.541	25.47	0.35	25.12	45.20	568
91.32	25.68	0.297	18.72	0.29	18.43	33.17	631
91.55	25.56	0.297	18.49	0.28	18.21	32.78	636
91.31	25.73	0.297	18.73	0.29	18.44	33.22	632
96.00	21.34	0.296	14.04	0.24	13.80	24.85	698
96.15	21.22	0.296	13.89	0.24	13.65	24.59	702
96.05	21.19	0.297	13.99	0.24	13.75	24.75	697
100.65	16.07	0.295	9.39	0.18	9.21	16.58	784
100.45	16.84	0.295	9.59	0.19	9.40	16.91	806
100.13	16.97	0.293	9.91	0.19	9.72	17.51	780
99.98	16.97	0.295	10.06	0.19	9.87	17.78	774
103.85	11.68	0.294	6.19	0.13	6.06	10.91	864
104.00	11.75	0.294	6.04	0.13	5.91	10.64	892
107.40	6.29	0.293	2.64	0.07	2.57	4.62	1094
107.40	6.18	0.293	2.64	0.07	2.57	4.62	1075
AVERAGE TEMPERATURE OF STEAM 110.04° C.—2.89 PER CENT AIR							
80.10	19.31	0.470	29.94	0.34	29.60	53.30	468
80.10	19.28	0.470	29.94	0.34	29.60	53.30	467
80.45	19.24	0.470	29.59	0.34	29.25	52.65	472
92.48	13.67	0.470	17.56	0.24	17.32	31.18	566
92.62	13.72	0.470	17.42	0.24	17.18	30.94	572
94.82	11.92	0.471	15.22	0.21	15.01	27.06	571
94.72	11.92	0.471	15.32	0.21	15.11	27.20	570
94.72	11.99	0.471	15.32	0.21	15.11	27.20	570
99.98	13.73	0.298	10.06	0.15	9.91	17.83	630
99.28	14.55	0.295	10.76	0.16	10.60	19.08	617
107.92	4.63	0.294	2.12	0.05	2.07	3.73	1004
107.72	4.67	0.294	2.32	0.05	2.27	4.09	922
107.76	4.63	0.294	2.28	0.05	2.23	4.02	932
AVERAGE TEMPERATURE OF STEAM 110.04° C.—4.53 PER CENT AIR							
78.55	20.77	0.345	31.49	0.27	31.22	56.20	350
78.55	20.65	0.345	31.49	0.26	31.23	56.20	348
78.63	20.65	0.345	31.41	0.27	31.14	56.00	349
87.94	16.00	0.344	22.10	0.21	21.90	39.40	383
87.94	15.98	0.343	22.10	0.21	21.89	39.40	382
87.94	16.05	0.344	22.10	0.21	21.89	39.40	384
95.53	11.39	0.342	14.51	0.14	14.37	25.85	413
95.53	11.35	0.342	14.51	0.14	14.37	25.85	412
99.28	9.98	0.301	10.76	0.11	10.65	19.17	430
99.05	10.18	0.302	10.99	0.11	10.88	19.58	431
98.95	10.31	0.302	11.09	0.12	10.91	19.75	432
98.80	10.31	0.302	11.24	0.12	11.12	20.00	427
102.95	7.03	0.302	7.09	0.08	7.01	12.62	462
102.82	6.96	0.302	7.22	0.08	7.14	12.85	449
102.87	7.03	0.301	7.17	0.08	7.09	12.76	455
106.50	3.92	0.304	3.54	0.04	3.50	6.30	519
106.50	3.93	0.304	3.54	0.04	3.50	6.30	520
AVERAGE TEMPERATURE OF STEAM 110.04° C.—6.59 PER CENT AIR							
71.42	12.90	0.410	38.62	0.19	38.43	69.20	210
71.68	13.02	0.410	38.36	0.20	38.16	68.64	213
71.85	12.95	0.410	38.19	0.19	38.00	68.40	213
87.03	9.35	0.407	23.01	0.14	22.87	41.18	254
87.08	9.32	0.407	22.96	0.14	22.82	41.10	253
86.95	9.51	0.409	23.09	0.14	22.95	41.32	259
96.00	6.06	0.412	14.04	0.09	13.95	25.12	273
95.77	6.01	0.413	14.27	0.09	14.18	25.53	267
95.77	6.00	0.413	14.27	0.09	14.18	25.52	267
101.68	7.11	0.300	8.36	0.08	8.28	14.90	393
101.17	7.11	0.302	8.37	0.08	8.29	15.82	372
101.63	7.11	0.302	8.41	0.08	8.33	14.99	393
102.05	7.20	0.301	7.99	0.08	7.91	14.24	418
101.78	5.92	0.303	8.39	0.07	8.32	14.98	329
106.58	3.39	0.303	3.56	0.04	3.52	6.34	445
106.54	3.52	0.303	3.58	0.04	3.54	6.61	459
106.54	3.53	0.303	3.70	0.04	3.66	6.59	446
AVERAGE TEMPERATURE OF STEAM 99.97° C.—1.42 PER CENT AIR							
80.78	19.02	0.421	19.19	0.30	18.89	34.00	647
82.12	18.70	0.422	17.85	0.30	17.55	31.57	686
82.34	18.51	0.421	17.63	0.29	17.34	31.20	686
83.22	18.12	0.421	16.75	0.29	16.46	29.61	707
83.89	17.93	0.421	16.08	0.28	15.80	28.44	729
93.69	7.90	0.420	6.28	0.12	6.16	11.09	822
93.69	7.99	0.420	6.28	0.13	6.15	11.07	832
93.55	7.87	0.420	6.42	0.12	6.30	11.34	800
98.02	3.26	0.419	1.95	0.05	1.90	3.42	1097
98.02	3.39	0.419	1.95	0.05	1.90	3.42	1142
97.92	3.33	0.419	2.05	0.05	2.00	3.60	1085
AVERAGE TEMPERATURE OF STEAM 99.97° C.—3.47 PER CENT AIR							
90.14	10.04	0.317	9.83	0.12	9.61	17.48	500
91.31	9.24	0.317	8.66	0.11	8.55	15.39	523
91.48	9.16	0.317	8.49	0.11	8.38	15.08	529
91.58	8.98	0.317	8.39	0.11	8.28	14.90	525
95.77	4.92	0.317	4.20	0.06	4.14	7.44	575
95.90	4.95	0.317	4.07	0.06	4.01	7.21	598
AVERAGE TEMPERATURE OF STEAM 99.99° C.—5.21 PER CENT AIR							
92.18	5.71	0.302	7.81	0.06	7.75	13.95	340
93.18	4.64	0.302	6.81	0.05	6.75	12.15	317
93.41	4.58	0.302	6.58	0.05	6.53	11.75	323
97.50	1.98	0.301	2.49	0.02	2.47	4.44	368
97.55	1.93	0.301	2.44	0.02	2.42	4.35	363
97.45	1.90	0.301	2.54	0.02	2.52	4.53	346
AVERAGE TEMPERATURE OF STEAM 100.06° C.—11.44 PER CENT AIR							
74.32	10.85	0.303	25.74	0.12	25.62	46.09	196
74.32	10.85	0.303	25.74	0.12	25.62	46.09	196
74.08	11.04	0.304	25.98	0.13	25.85	46.49	198
84.57	8.23	0.303	15.49	0.09	15.40	27.70	247
84.12	8.57	0.303	15.94	0.10	15.84	28.50	250
84.12	8.50	0.301	15.94	0.10	15.84	28.50	247
84.29	8.33	0.303	15.77	0.09	15.68	28.20	246
AVERAGE TEMPERATURE OF STEAM 99.96° C.—ZERO AIR							
89.50	23.38	0.508	10.46	0.45	10.01	18.01	1810
89.10	24.11	0.508	10.86	0.46	10.40	18.72	1797
89.10	23.97	0.508	10.86	0.46	10.40	18.72	1787
89.32	23.80	0.508	10.64	0.45	10.19	18.34	1810
92.15	18.80	0.508	7.81	0.36	7.45	13.41	1957
91.63	19.24	0.507	8.33	0.37	7.96	14.33	1871
91.86	19.05	0.507	8.10	0.36	7.74	13.93	1904
93.88	16.36	0.507	6.08	0.31	5.77	10.39	1996
93.98	16.58	0.507	5.98	0.32	5.66	10.20	2268
93.98	16.71	0.506	5.98	0.32	5.66	10.19	2280
96.70	12.32	0.506	3.26	0.23	3.03	5.45	3138
96.70	12.06	0.506	3.26	0.23	3.03	5.45	3071
96.70	12.68	0.505	3.26	0.24	3.02	5.43	3236
AVERAGE TEMPERATURE OF STEAM 105.02° C.—ZERO AIR							
98.44	19.11	0.502	6.58	0.36	6.22	11.20	2363
98.81	18.38	0.493	6.21	0.34	5.87	10.57	2355
99.04	18.25	0.493	5.98	0.34	5.64	10.15	2435
102.96	10.04	0.491	2.06	0.18	1.88	3.38	4000
103.64	8.25	0.489	1.38	0.15	1.23	2.21	5005
AVERAGE TEMPERATURE OF STEAM 109.97° C.—ZERO AIR							
101.91	23.21	0.506	8.06	0.44	7.62	13.71	2352
101.96	22.89	0.507	8.01	0.44	7.57	13.62	2339
101.82	23.28	0.506	8.15	0.44	7.71	13.88	2332
103.36	20.51	0.506	6.61	0.39	6.22	11.20	2546
103.27	20.27	0.506	6.70	0.38	6.32	11.38	2477
103.32	20.70	0.506	6.65	0.39	6.26	11.27	2553
103.36	20.60	0.506	6.61	0.39	6.22	11.20	2557
106.08	16.07	0.506	3.89	0.31	3.58	6.44	3463
105.88	16.58	0.503	4.09	0.31	3.78	6.80	3380
105.67	16.84	0.503	4.30	0.32	3.98	7.16	3260
AVERAGE TEMPERATURE OF STEAM 115.04° C.—ZERO AIR							
108.00	37.35	0.302	7.04	0.42	6.62	11.91	2600
108.00	36.61	0.303	7.04	0.42	6.62	11.91	2559
107.93	36.52	0.306	7.11	0.42	6.69	12.05	2550
106.62	23.86	0.535	8.42	0.48	7.94	14.29	2454
106.91	23.68	0.535	8.13	0.47	7.66	13.79	2523
107.03	23.21						

temperature, and from Jakob's equation the thermal conductivity increases. Both tend to increase the coefficient at higher temperatures.

The enormous decrease of the rate at which nearly pure steam condenses when a slight amount of air is added shows that the non-condensable gas must be localized around the tube. The decrease of the coefficient by 50 per cent caused by adding 0.5 per cent air cannot be explained if the air is evenly distributed, since there would then be almost no change in the partial pressure of steam at the interface and condensation would take place unimpeded. There is, however, a backward diffusion of the air against the flow of steam and a steady state is reached when the rate of diffusion of the air away from the tube is balanced by the rate at which it is carried up to it. Obviously, the steady state is reached when the rate of diffusion of the steam through this air film is just equal to the rate at which condensation may take place as determined by the resistance of the condensate film. This rate of heat transfer through the condensate film probably depends only on the partial pressure of the steam at the interface and is independent of the presence or absence of air if the partial pressure of steam is unchanged.

The process of heat transfer from a vapor containing a non-condensable gas is seen from these considerations to be exactly analogous to the problem of gas absorption developed by Lewis and Whitman²⁹ and others. The air film is comparable to the stagnant gas film in the absorption of a gas from a mixture with one which is inert, and both are caused by molecular diffusion. In the case of a pure vapor passing over a liquid absorbent, Lewis and Whitman show that the resistance of the vapor film is negligible compared with the resistance of the liquid film. The absence of a film between pure vapor and condensate has been shown in another experiment in this laboratory, which will not be described here.

The heat flowing from steam to tube passes first through the gas film if air is present, then through the condensation interface, and finally through the film of water. The reciprocal of the coefficient represents the resistance and when plotted (Figure 6) it is seen to increase as a linear function of the per cent of air present. These lines do not intersect at the origin, and the resistance is not zero at zero air because of the resistance of the condensate film. The intersection of the lines on the graph does occur, however, at a point where the resistance to the flow of heat would be zero or the coefficient is infinity (since $1/\infty = 0$) and at this point the thickness of the water film approaches an infinitesimal.

From the fact that at zero air the only resistance is that of the condensate film, it is apparent that as the amount of air is increased the proportional effect of the condensate resistance to the total resistance becomes less and a given change of the thickness of the condensate film affects the coefficient to a smaller degree. At a given temperature of steam the thickness of the water film is largely dependent on the amount of heat flowing, which is in turn dependent on the temperature drop. The generalization which may be drawn from this is: Under constant temperature conditions, the rate of change of coefficient with respect to temperature drop varies inversely with concentration of air. Mathematically this may be formulated

$$\left[\frac{\delta f}{\delta(\Delta T)} \right]_T \propto \frac{1}{C}$$

The graphs in Figures 3 and 4 show the rate of change of the logarithm of the coefficient with the logarithm of the temperature drop (the slopes of the indicated lines) to decrease with increasing concentrations of air.

The curves of the reciprocal of the coefficient against composition (Figure 6) show that adding 0.5 per cent air decreases the coefficient to one-half its value at zero air, a second in-

crease of 0.5 per cent air decreases it to a third, a third increase of 0.5 per cent decreases it to a fourth, etc. In practice, air in a concentration of only a few per cent in the steam seriously reduces the capacity of steam-condensing equipment.

The points with the largest amounts of air in both the 212° and 230° F. series are seen to be off the line as determined by the other points in the graph in Figure 6. This is not believed to indicate the failure of the relation between the coefficient and the composition for steam with 6 or more per cent air, but was due instead to experimental difficulties. Until the runs with large concentrations of air were made, the reading made with the greatest ease and controlled with the greatest facility was that of the steam temperature. When, however, the air exceeded 4 per cent, this temperature varied considerably during readings, even though the pressure was kept constant by the relay circuit. Since it could not be accurately determined, the temperature drop and coefficient varied over an even greater range. The plotted values at these higher concentrations of air were therefore given less weight in drawing the representative lines.

Throughout the whole of this work no mention has been made of the mass velocity of the steam with regard to the tube. With flow of vapor past the heating surface, vapor velocity will affect the coefficient by changing the thicknesses of the gas and liquid films. It will probably change the gas film much more than the liquid film and thus in the case where pure vapors are being condensed, its effects will be comparatively small. From the hydromechanics of condensate drainage it is evident that velocity of steam will change the coefficient differently for every type, angle with horizontal, and length of tube surface.

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