EFFECT OF ORIENTATION ON HEAT AND MASS TRANSFER IN STACKED BEDS OF SPHERES

by

LESLIE ROBERT GALLOWAY

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ABSTRACT

Heat, mass and momentum transfer rates have been measured in two stacked beds of porous spheres having equal fractional void volume but different orientation with respect to the direction of fluid flow. An air-water system was studied under essentially adiabatic conditions over a Reynolds number range 100-1200. Orientation had negligible effect on heat and mass transfer rates though considerable effect on friction factor.

An explanation for this behaviour is presented in terms of a difference in the degree of turbulent wake formation for the two assemblages, similar to that observed in comparable banks of closely packed staggered and in-line heat exchanger tubes.

The experimental results contradict simple analogies between momentum, heat and mass transfer which show a direct proportionality between total friction factor and heat and mass transfer factor.

Measured friction factors were about 50% in excess of those obtained by Martin for similar assemblages of smooth metal spheres. This is explained by the higher surface roughness of the refractory-like spheres used in the present investigation.

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NOMENCLATURE

a	=	Surface area of the solids per unit volume of bed, sq.ft./cu.ft.
A_s	=	Area available to heat or mass transfer, sq.ft.
A	=	Constant dimensionless
B	=	Constant, dimensionless
b	=	Constant, dimensionless
c	=	Constant, dimensionless
C_1	=	Concentration of diffusing component in inlet stream, lb. moles/cu.ft.
C_2	=	Concentration of diffusing component in exit stream, lb. moles/cu.ft.
C^*	=	Equilibrium concentration of diffusing component in stream, lb. moles/cu.ft.
C_p	=	Specific heat at constant pressure, B.t.u./(lb.)(°F.)
D	=	Inside pipe diameter, ft.
D_p	=	Effective particle diameter, ft.
D_v	=	Diffusion coefficient of gas in the film, sq.ft./hr.
f	=	Friction factor $= \dfrac{g_c \Delta P D_p \rho}{2G^2 L}$ in packed bed, dimensionless
f_k	=	Modified friction factor $= \dfrac{2f}{g_c} \cdot \dfrac{\epsilon^3}{1-\epsilon}$, dimensionless
G	=	Mass velocity based on empty column, lb./(hr.)(sq.ft.)
g_c	=	Gravitational constant, (lb.-force)(ft.)/(lb.-mass)(sec.^{2})
h	=	Heat transfer coefficient for gas, B.t.u./(hr.)(sq.ft.)(°F.)
j_d	=	Mass transfer factor $= \dfrac{k_g p_{gf} M_m}{G}\left(\dfrac{\mu}{\rho D_v}\right)_f^{2/3}$, dimensionless
j_H	=	Heat transfer factor $= \dfrac{h}{C_p G}\left(\dfrac{C_p \mu}{k}\right)_f^{2/3}$, dimensionless
k'	=	Mass transfer coefficient, lb./(hr.)(ft.^{2})(humidity difference)

k	=	Thermal conductivity of fluid, B.t.u./(hr.)(sq.ft.)(°F./ft.)
k_g	=	Mass transfer coefficient of gas film, (lb. moles)/(hr.)(sq.ft.)(atm.)
K	=	Orifice flow coefficient, dimensionless
L	=	Height of column, ft.
M_m	=	Average molecular weight of fluid, lb./lb. mole
Nu	=	Nussult number for heat transfer = hD_p/k, dimensionless
Nu'	=	Mass transfer number analagous to Nu for heat transfer = $k_g p_{gf} M_m D_p / \rho D_v$ for packed beds, dimensionless
q	=	Rate of heat transfer, B.t.u./hr.
p_{gf}	=	Arithmetic mean partial pressure of the non-transferred gases in the gas film, in.Hg = $\left[(P_1 - p_1) + (P_2 - p_2)\right]/2$
p_{w_1}	=	Partial pressure of water vapor at temperature t_{w_1}, in. Hg
p_{w_2}	=	Partial pressure of water vapor at temperature t_{w_2}, in. Hg
p_1	=	Partial pressure of water vapor in entrance air, in. Hg
p_2	=	Partial pressure of water vapor in exit air, in. Hg
$\Delta p_{l.m.}$	=	Log mean partial pressure of the transferring gas in the gas film = $\dfrac{(p_{w_1} - p_1) - (p_{w_2} - p_2)}{\ln \dfrac{p_{w_1} - p_1}{p_{w_2} - p_2}}$, atm.
P	=	Total pressure, atm.
ΔP	=	Pressure drop, lb-force/sq.ft.
$-dP$	=	Decrease in pressure, lb.-force/sq.ft.
P_1	=	Absolute pressure at inlet of bed, in. Hg
P_2	=	Absolute pressure at outlet of bed, in. Hg
Pr	=	Prandtl number = $C_p \mu / k$, dimensionless
Re	=	Reynolds number = $D_p V_o \rho / \mu$ in packed bed, dimensionless
Sc	=	Schmidt number = $\mu / \rho D_v$, dimensionless

t_1	=	Temperature of inlet air, °F.
t_2	=	Temperature of outlet air, °F.
t_{w_1}	=	Wet bulb temperature of inlet air, °F.
t_{w_2}	=	Wet bulb temperature of exit air, °F.
$\Delta t_{l.m.}$	=	Log mean temperature difference = $\dfrac{(t_{w_1} - t_1) - (t_{w_2} - t_2)}{\ln \dfrac{t_{w_1} - t_1}{t_{w_2} - t_2}}$, °F.
$-(\Delta t)$	=	Decrease in temperature of the transfer medium, °f.
V_o	=	Superficial velocity based on empty column, ft./sec.
V	=	Volume of packing, cu.ft.
w	=	Rate of mass transfer, lb. moles diffusing component/hr.

Greek Symbols

ϵ	=	Fractional void volume in packed bed, dimensionless
ϕ	=	Function in momentum transfer equations, dimensionless
ϕ'	=	Function in mass transfer equations, dimensionless
ϕ''	=	Function in heat transfer equations, dimensionless
ρ	=	Density of fluid, lb./cu.ft.
μ	=	Viscosity of fluid, lb./(ft.)(sec.)

INTRODUCTION

In the past decade the field of heat, mass and momentum transfer in systems where assemblages of particles are contacted by a fluid stream has received considerable attention. New industrial applications, the demand for reliable design equations and the desire to understand more fully the basic mechanisms of these transfer processes have been reasons for this increased activity.

Fixed beds, that is beds in which the fluid moves past a stationary assemblage of particles, were probably the first venture into this very broad field. For example, blast furnace operation and filtration have been used for centuries. Moving beds, fluidized beds and a late innovation - spouted beds - are developments of more recent years.

The subject of heat, mass and momentum transfer in fixed beds has been investigated extensively (27). Many empirical correlations relating various modified Reynolds numbers with friction factor, mass transfer factor and heat transfer factor have been presented. The bulk of this work, however, has been made with random packed beds.

The fact that published data on pressure drop through packed beds have not correlated too well led Martin et al (43) to investigate the effects of orientation of packing on pressure drop. They found that considerable effect due to orientation did exist. Little or no attention, however, has been paid to heat and mass transfer rates in orientated beds.

The object, then, of this investigation has been to measure heat and mass transfer rates in two spedific packings used by Martin. These packings, although having the same voidage and even the same basic arrangement when viewed in isolation, differed in orientation with respect to the direction of flow and yielded considerably different friction factors, all other variables being equal.

LITERATURE REVIEW

The transfer processes that occur when a fluid flows through a fixed assemblage of solid particles have received the attention of a multitude of investigators in the past. Their objectives were to obtain equations that could be used for design purposes and to increase the knowledge of the basic mechanisms involved in these transfer processes. It was apparent that the complexity of these mechanisms did not lend themselves to immediate theoretical treatment. Hence, the treatment of this subject by most workers has been on an empirical basis.

1. METHODS OF CORRELATING DATA

a. Pressure Drop

When fluid flows in a circular duct the pressure drop due to friction is found, by application of dimensional analysis, to be expressed by the following equation

$$-dP = \frac{\rho V_0^2 dL}{g_c D} \phi\left(\frac{D V_0 \rho}{\mu}\right) \qquad (1)$$

This expression can be integrated across the length of the duct if the velocity, density, and diameter of the duct are assumed to remain constant to give

$$\Delta P = \frac{\rho V_0^2 L}{g_c D} \phi\left(\frac{D V_0 \rho}{\mu}\right) \qquad (2)$$

This can be written as

$$\Delta P = 2f\left(\frac{\rho V_0^2 L}{g_c D}\right) \qquad (3)$$

where

$$2f = \phi \left(\frac{DV_o \rho}{\mu} \right) \tag{4}$$

Equation 3 is the so-called Fanning equation. By rearrangement of equation 3 it is found that

$$f = \frac{\Delta P g_c D}{2 \rho V_o^2 L} \tag{5}$$

By experimentation it is possible to establish the relation that exists between friction factor and Reynolds number.

A modified version of equation 5 has been used to calculate friction factor in packed beds:

$$f = \frac{\Delta P g_c D_p}{2 \rho V_o^2 L} \tag{5a}$$

This value of friction factor is obtained as a function of a modified Reynolds number, $D_p V_o \rho / \mu$. Several authors have proposed other modifications of the Fanning equation to take account of variables in the packing such as voids, roughness and shape of particles. These shall be considered under separate headings.

b. Mass Transfer and Heat Transfer

When a concentration gradient of a component exists within a phase, there is a potential available tending to transfer the component in the direction of decreasing concentration. The rate at which this component is transferred is directly proportional to the concentration gradient and the area available for transfer. Thus,

$$w \propto A_s(-\Delta C) \tag{6}$$

or

$$w = k A_s(-\Delta C) \tag{7}$$

Under steady state conditions, for mass transfer in the gas phase, equation 7 becomes (24)

$$w = k_g A_s \Delta p_{l.m.} \tag{8}$$

An analagous situation exists for heat transfer in as much as the rate of heat transfer is also directly proportional to the driving force, in this case temperature gradient, and the area available for heat transfer. That is,

$$q \propto A_s(-\Delta t) \tag{9}$$

or

$$q = hA_s(-\Delta t) \tag{10}$$

Under steady state conditions, equation 10 becomes (24)

$$q = hA_s \Delta t_{l.m.} \tag{11}$$

Three methods are available for expressing transfer rates. These are the transfer coefficients, k_g for mass transfer and h for heat transfer; the transfer factors j_d for mass transfer and j_H for heat transfer; and the height of a transfer unit, $(H.T.U.)_H$ for heat transfer and $(H.T.U.)_d$ for mass transfer. The transfer coefficients have the advantage of simplicity but have the disadvantage of not being dimensionless and not relating the properties of the system. Chilton and Colburn overcame this problem by developing the transfer factors (10) and the height of the transfer unit (11).

The development of the transfer factors came about in the following manner. If dimensional analysis is applied to the correlation of mass transfer coefficients in wetted-wall columns and to the correlation of heat transfer coefficients in circular ducts for turbulent flow, the following equations are obtained: for mass transfer,

$$\frac{k_g p_{gf} M_m D}{\rho D_v} = \phi' \left[\left(\frac{DV_o \rho}{\mu} \right), \left(\frac{\mu}{\rho D_v} \right) \right] \qquad (12)$$

and for heat transfer,

$$\frac{h\, D}{k} = \phi'' \left[\left(\frac{DV_o \rho}{\mu} \right), \left(\frac{C_p \mu}{k} \right) \right] \qquad (13)$$

For empirical correlation purposes it is usually assumed that equations 12 and 13 may be simplified respectively to

$$\frac{k_g p_{gf} M_m D}{\rho D_v} = A \left(\frac{DV_o \rho}{\mu} \right)^b \left(\frac{\mu}{\rho D_v} \right)^c \qquad (14)$$

and

$$\frac{h\, D}{k} = B \left(\frac{DV_o \rho}{\mu} \right)^y \left(\frac{C_p \mu}{k} \right)^z \qquad (15)$$

By rearranging the terms in equations 14 and 15, they become respectively

$$\frac{k_g p_{gf} M_m}{G} = A \left(\frac{DV_o \rho}{\mu} \right)^{b-1} \left(\frac{\mu}{\rho D_v} \right)^{c-1} \qquad (16)$$

and

$$\frac{h}{C_p G} = B \left(\frac{DV_o \rho}{\mu} \right)^{y-1} \left(\frac{C_p \mu}{k} \right)^{z-1} \qquad (17)$$

Chilton and Colburn (10) have defined the transfer factors as

$$j_d = \frac{k_g p_{gf} M_m}{G} \left(\frac{\mu}{\rho D_v} \right)_f^{2/3} \qquad (18)$$

and

$$j_H = \frac{h}{C_p G} \left(\frac{C_p \mu}{k} \right)_f^{2/3} \qquad (19)$$

If $z = C = 1/3$, as has been demonstrated experimentally (55), then equations 16 and 18, and equations 17 and 19 can be combined respectively to give

$$j_d = A(Re)^{b-1} \qquad (20)$$

and

$$j_H = B(Re)^{y-1} \qquad (21)$$

These correlations have been extended to heat and mass transfer in packed beds by making the necessary modifications to the dimensionless groups. These modifications are attempts to adequately describe the flow of fluid past the solid particles and include substitution of D_p for D and, in some cases, the introduction of a voidage term and a particle shape factor.

A more general expression for equations 20 and 21 applied to packed beds would be, respectively,

$$j_d = \phi'(Re) \qquad (22)$$

and

$$j_H = \phi''(Re) \qquad (23)$$

since it is found that the constants A, B, b and z when the fluid is turbulent are different in value from those when the fluid is laminar.

2. THE EFFECTS OF ORIENTATION

Of the multitude of works published in heat transfer (8, 21, 24, 42, 49, 52, 60), mass transfer (12, 13, 17, 22, 24, 26, 27, 29, 46, 51, 52, 56, 57, 61) and momentum transfer (4, 7, 8, 14, 16, 24, 35, 38, 39, 47) in packed beds, comparatively no attention has been paid to possible effects of orientation.

Martin, McCabe and Monrad (43) made perhaps the only formal investigation on the effects of orientation of packing on transfer rates. Their work was confined only to friction factor measurements. They found that packings of equal voidage but different orientation produced, at equal Reynolds numbers, widely differing friction factors. Orientation effects in heat and mass transfer have received even less attention. Taecker and Hougen (57) mention, in passing, that no significant differences in j_H were obtained in comparing random with staggered arrangements of packings (saddles and rings).

3. THE EFFECTS OF SURFACE ROUGHNESS

Surface roughness effects on pressure drop through packed beds have been studied by Leva et al (40). They report an increase in friction factor as surface roughness is increased when testing aloxite granules, clay Raschig rings, alundum cylinders and clay balls in tubes in turbulent flow. Campbell and Huntington (7a) report similar results. Brownell and Katz (5) found that comparison of data on lead spheres and on celite spheres indicated that the celite spheres exhibited a greater resistance to flow than did the lead spheres under similar conditions. This difference they attribute to roughness.

No studies on the effects of particle surface roughness on heat and mass transfer between fluids and packed beds have been found reported.

4. THE EFFECTS OF VOIDS

The effects of void volume on pressure drop have been investigated by many workers (3, 6, 7, 14, 16, 19, 20, 25, 33, 34, 40, 41, 44, 58). The importance of including a void volume term in correlating friction factor measurements is well known, but how this should be done has become a point of controversy (14).

The effects of voids on heat and mass transfer have not received the same amount of consideration. Several authors (15, 17, 22, 23, 29, 31) use the void fraction term in their correlations of mass transfer with Reynolds number. In some cases, it is used in an attempt to define a

Reynolds number of the fluid moving past the solid particles (15, 22). Others introduce the term in order to correlate fixed beds with fluidized beds (17, 31), while still others have used it to relate published data (23,29) for different packings. Gamson (23), when he plotted reported mass transfer data for spherical particles (24, 27, 46) as j_d versus a modified Reynolds number, $6G/a\mu$, found that a series of curves resulted with the void volume of the system as parameter. He was able to consolidate all these reported data for spherical particles into a single generalized correlation by plotting $j_d/(1-\epsilon)^{0.2}$ versus $6G/a\mu$ $(= D_pG/\mu(1-\epsilon))$. Data reported by Hobson and Thodus (27) and McCune and Wilhelm (46) were not in as good agreement in the transition region ($10 < 6G/a\mu < 100$). This lack of agreement was attributed by Gamson to the indefinite flow pattern of this region. Gamson et al (24) in their investigation found that while pressure drop was a function of the voidage, mass and heat transfer factors were not affected at all.

5. THE ANALOGY BETWEEN HEAT AND MASS TRANSFER AND MOMENTUM TRANSFER

Considerable theoretical and empirical work has been done to establish an analogy between heat, mass and momentum transfer in circular conduits (32). Several authors (15, 31, 50) have attempted to extend this analogy to packed beds.

Ranz (50) considers that transfer rates in packed beds of spheres occur as a summation of the transfer rates about the consituent spheres in isolation, the effective velocity past the spheres being taken as the superficial velocity divided by the minimum fractional free area of the packing. He is thus able to correlate turbulent heat, mass and momentum transfer data in randomly packed beds with those for an isolated sphere. His derivation leads to the result that two packed beds of spheres with the same voids, but so aligned as to offer quite different minimum fractional free area to fluid flow, would not only show markedly different fluid friction characteristics, but also correspondingly different heat and mass transfer rates.

Ergun (15) has proposed for packed beds an equation which he found correlated fluid friction data quite well. The equation presented is

$$f_k = 150 \mu \frac{(1-\epsilon)}{D_p G} + 1.75 \tag{24}$$

The analogy for mass transfer claimed here is that

$$f_k = \frac{D_p}{L} \frac{\epsilon}{1-\epsilon} \frac{\mu}{\rho D_v} \frac{C_2 - C_1}{C^* - C_2} \tag{25}$$

for complete longitudinal mixing of the fluid in the bed and

$$f_k = \frac{D_p}{L} \frac{\epsilon}{1-\epsilon} \frac{\mu}{\rho D_v} \ln \frac{C^* - C_1}{C^* - C_2} \tag{26}$$

for the case of no longitudinal mixing. Some degree of correlation was obtained between mass transfer and fluid friction for liquid systems on assuming no fluid mixing. However, little success was obtained with gaseous systems for which perfect mixing was assumed. Ergun claims that this was due to the deficiency and uncertainty of published gas stream data but he offers no direct experimental evidence for his mixing assumptions. No attempt was made to correlate heat transfer data.

Ju Chin Chu et al (31) have investigated mass and momentum transfer in fixed and fluidized beds and have proposed a modification to the Chilton and Colburn analogy (10) which may be written

$$\frac{(f/2)\ (\epsilon^3/1-\epsilon)}{k_g p_{gf} M_m/G} = 5(Sc)^{2/3} \tag{27}$$

or

$$j_d = (f/10)\ (\epsilon^3/1-\epsilon) \tag{28}$$

Fair agreement with experimental data for randomly packed and fluidized beds is obtained over a Reynolds number range of 1 - 10,000. Here again the results indicate, as in equation 28, a direct dependence of j_d on f, regardless of what factors (e.g. orientation) bring about the variation of f at a given Reynolds number and packing voids.

APPARATUS

The rates of heat, mass and momentum transfer were made using an air-water system. Air was passed through a bed of porous spheres (to be described later) which had been previously soaked in water. This method corresponds to that used by Gamson et al (24), Taecker and Hougen (57), Wilke and Hougen (61) and Hobson and Thodos (27).

The apparatus is illustrated schematically in Figure 1. Air, which was obtained from the building supply, was conveyed to the packed bed through 2-inch commercial steel pipe. Air flow rates were measured with a standard orifice using flange pressure taps. The pressure drop through the orifice was measured with a 60-inch vertical water manometer. Calibrated thermometers reading to the nearest 0.1°F were positioned at the inlet and outlet of the column housing the packing. A series of sampling lines shown schematically in Figure 2 were used to enable humidity determinations to be taken of both inlet and outlet air streams throughout the run. Humidity was measured with a Foxboro "Dewcel" Dew Point Recorder. Pressure drop measurements through the packing were made with a Hays Corporation Draft Gauge reading to the nearest 0.005 inches of water.

A more detailed description of the apparatus will now follow.

1. AIR SUPPLY

The air, which was used at room temperature for all runs, was obtained from the building supply. It has a maximum rate of 127 lb./hr. which corresponds to a Reynolds number of approximately 1200 through the packing. A centrifugal air blower driven by a 2 H.P. motor and delivering air at a maximum flow rate of 50,000 cu. ft./hr. at a pressure of 12 inches of water was also installed in the system in order to obtain higher Reynolds numbers; however, it was not used.

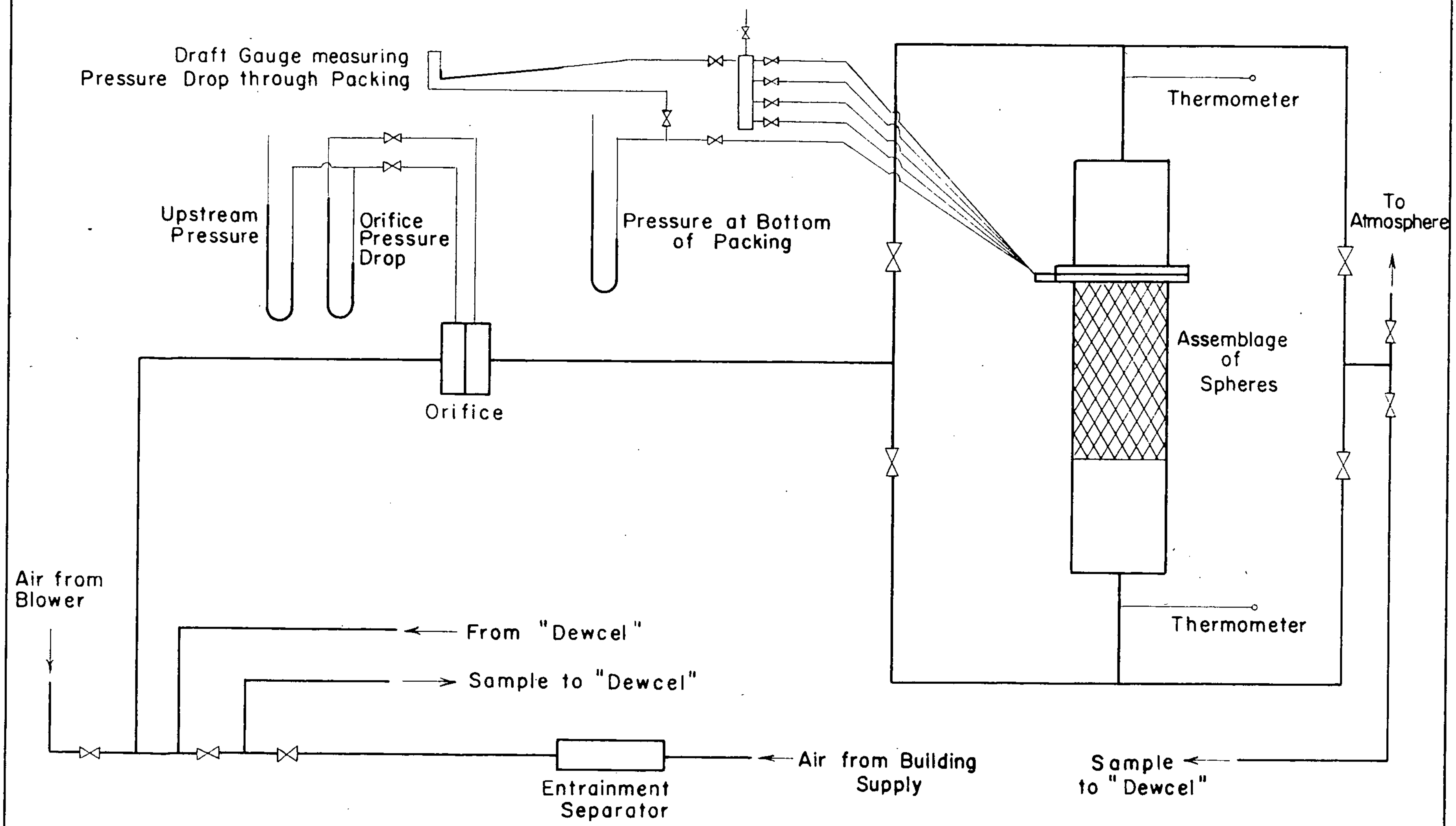


Figure 1. Schematic Diagram of Apparatus

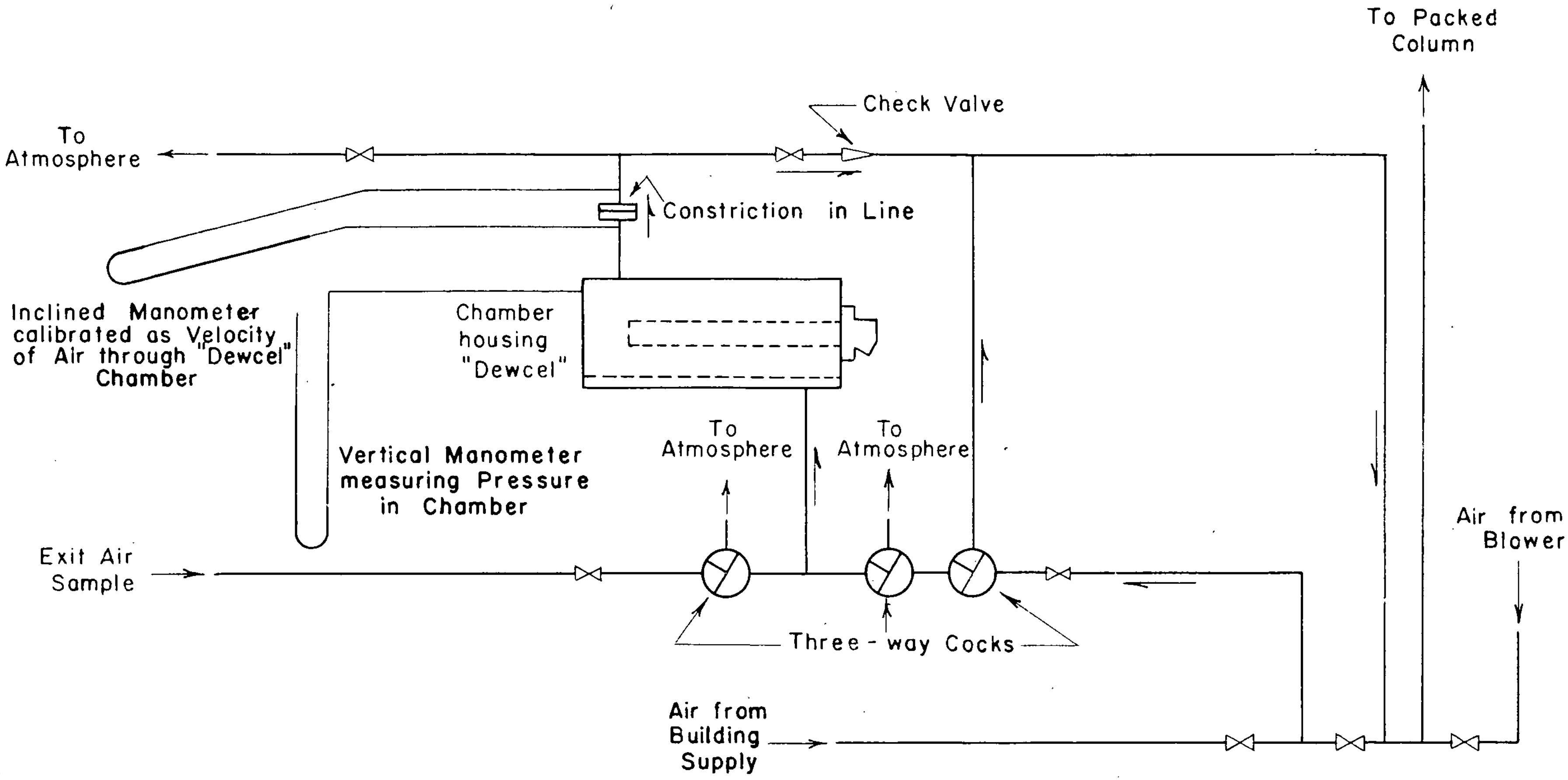


Figure 2. Schematic Diagram of Air Sampling Lines

2. ENTRAINMENT SEPARATOR

An entrainment separator was installed in the lines coming from the building air supply to remove entrained water. It consisted of a closed cylinder 2 3/4 inches in diameter and 9½ inches long, fitted with standard 3/4-inch pipe couplings at both ends. Two baffles were placed perpendicular to the air flow and 4 inches from either end of the cylinder. These baffles were circular and of the same diameter as the inside of the cylinder. Holes, 3/8-inch in diameter, were drilled in the baffles in such an arrangement that the air which passed through the holes of the first baffle would impinge upon the second baffle. 1½-inch lengths of 3/8-inch brass tubing were pressed into the holes in order to prevent the separated water from being picked up again by the air stream. Drains were installed slightly upstream from each baffle.

3. ORIFICE

Air was metered through standard orifices constructed according to the specifications given in the A.S.M.E. Report on fluid meters (2). Pressure drops were measured with flange taps made according to the recommendations in the report. Three orifice plates were machined having openings of ¼, ½ and 3/4-inches, thereby allowing flow rates to be measured over a wide range. Values of flow coefficient K were taken from this report and plotted as a function of the Reynolds number through the orifice with the ratio of the diameter of the orifice to the diameter of the pipe as parameter. This plot may be found in the appendix. The ½-inch orifice was calibrated using a 900 cu. ft. per hr. capacity diaphragm-type gas meter calibrated to an accuracy of 2%. The calibration of the orifice showed an average deviation in K from those given in the report of only 1%. It was therefore considered unnecessary to calibrate the other two orifices.

4. HUMIDITY DETERMINATION

The determination of moisture content by measuring the dew point is considered by Ewell (18) as the most accurate absolute method. Wet bulb measurements require elaborate set-ups (54) while gravimetric methods have been found inaccurate for highly humid air (45). Consequently, a Foxboro "Dewcel", which measures dew point automatically to the nearest 0.5°F, was considered best for this investigation. Moisture determination by the "Dewcel" is based on the fact that for every water vapor pressure in contact with a saturated salt solution, there is an equilibrium temperature at which this solution neither absorbs nor gives up moisture to the surrounding atmosphere. The "Dewcel" is a thin-walled metal socket covered with a woven glass tape impregnated with lithium chloride, and wound with a pair of silver wires connected to a 25-volt alternating current power supply. The lithium chloride, being hygroscopic, absorbs moisture and becomes a solution. The conductivity of the salt is increased, allowing a larger current to flow through the silver wires with the result that the temperature of the "Dewcel" rises, the solution dries up and the amount of current passing through the wires is reduced. The "Dewcel" then cools, absorbs more moisture and the cycle is repeated until equilibrium is attained. A liquid expansion thermometer indicates the temperature of the "Dewcel" and is recorded on a chart calibrated in dew point temperature.

An attempt to calibrate the instrument with a gravimetric determination resulted in the "Dewcel" reading consistently higher humidities than the gravimetric method. This result would be expected if the absorbing material (in this case magnesium perchlorate) did not remove all the moisture. A further check was made using wet and dry bulb thermometers. In this case the "Dewcel" indicated a lower humidity. Since it is probable that the wet bulb thermometer was reading too high and therefore indicating too high a moisture content, and since the gravimetric and wet and dry bulb determinations bracketed the "Dewcel" determination, it was believed that the "Dewcel" was reading accurately. A further calibration was made by checking the temperature indicating element of the "Dewcel" against a calibrated thermometer. This resulted in an average deviation of 0.38% in the humidity corresponding to the temperature of the "Dewcel" element from the humidity corresponding to the temperature indicated by the calibrated thermometer.

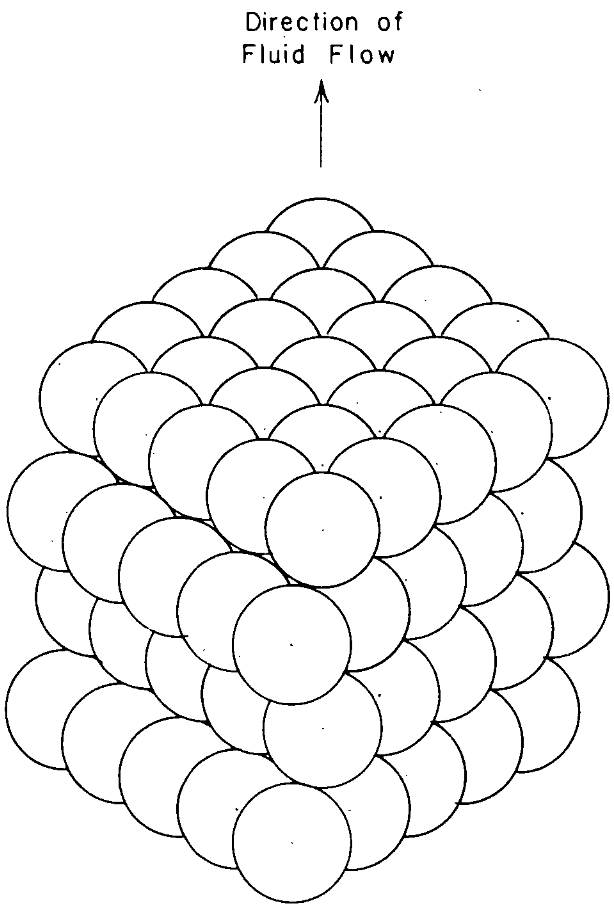
5. THERMOMETERS

The thermometers were calibrated against a Leeds & Northrup Co. platinum resistance thermometer bearing a National Bureau of Standards certificate dated August 14, 1939. Calibration curves are included in the appendix.

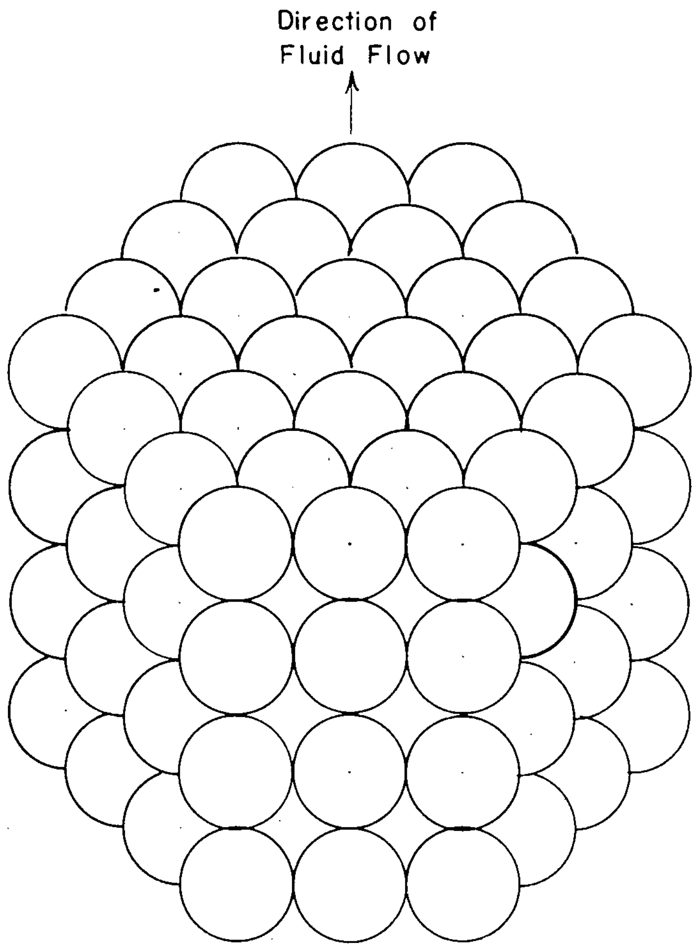
6. PACKING

Perhaps the major portion of this investigation was spent in formulating a suitable packing material, finding a method of molding the packing and performing the manufacturing operation.

The objective of this investigation was to compare two packings used by Martin et al (43) having the same voidage but showing widely different friction factors. Such packings are those designated by Martin as Orthorhombic No. 2 Clear Passage and Orthorhombic No. 4. Figure 3 shows these packings in isometric view. It will be noted that the basic arrangement of the spheres is the same in both packings when the packings are viewed in isolation; however, when viewed along the major axis of flow the orientations are quite different. This investigation was, therefore, a study of



ORTHORHOMBIC NO. 2



ORTHORHOMBIC NO. 4

Figure 3. Isometric Views of the two Orientations of Packing Used

were required for the two packings. Each sphere was measured to the nearest .001-inch across three diameters with a micrometer and an average diameter determined. The average diameter was 0.673-inch with a standard deviation of 0.004-inch. In order to pin the spheres together, it was necessary to drill six holes in each sphere in appropriate locations. The spheres were pinned together with 0.022-inch diameter stainless steel fishing wire, the wire being secured in each hole with Araldite AN-104 cement.

The characteristics of each of the two packings are given in Table I. In determining surface area, the correction for the transfer area lost by drilling six holes in each of the spheres was calculated to be only 1.08% and was considered negligible.

Wall porosity was eliminated by using fractional spheres at the walls as was done by Martin et al (43). It was therefore necessary to construct two columns in which to housethe packings: a square column for Orthorhombic No. 2 and a hexagonal column for Orthorhombic No. 4.

The bundles of spheres were enclosed on all sides except the top and bottom by 1/16-inch brass plate glued to the faces of the fractional spheres with Araldite AN-104. This was done mainly to afford protection to the somewhat delicate packing and had the additional advantage of avoiding the use of a supporting grid, thereby eliminating a source of entrance effects.

In order to measure entrance and exit effects in the packing as well as the total pressure drop through the packing, pressure taps were located in one of the brass sides at five different locations: at the bottom of the packing, between the 2nd and 3rd layers of spheres, between the 4th and 5th layers, between the 6th and 7th layers, and at the top of the 8-layer packing. This allowed pressure differentials between the bottom and any of

TABLE I

CHARACTERISTICS OF PACKINGS

Orientation	Shape of Container	Cross-Sectional Dimensions Inches	Cross-Sectional Area ft^2	No. of Spheres	Height of Bed Inches	Smallest Fraction Free Area	Surface Area	Void Volume
Orthorhombic No. 2	Square	$4\frac{11}{16} \times 4\frac{11}{16}$	0.1526	392	4.660	0.219	3.8690	0.3954
Orthorhombic No. 4	Regular Hexagon	$2\frac{11}{16}$ on all sides	0.1303	384	5.381	0.093	3.7900	0.3954

the other four positions to be measured.

The two columns used to house the assemblages of spheres were made from $\frac{1}{4}$- and 1/8-inch aluminum plate. The inside cross-sectional dimensions of these columns were slightly larger than the outside dimensions of the corresponding packing. This afforded a snug fit when the assemblage of spheres with brass side plates was placed in the column. In order to maintain a constant cross-sectional area throughout the entire length of the column, the column was lined with brass plate above and below the packing. The columns were made in two longitudinal sections, bolted together with a flange. The bottom section housed the packing assembly, the top of which was flush with the top of this section. Pressure lines from the taps in the side of the assemblage were brought through the column at the flange. This was done by running the lines from the taps to a brass plate at the top of the packing assembly. This plate, which was placed perpendicular to the direction of flow and parallel to the flange, was attached to the top of the wall containing the pressure taps. It contained five 1/8-inch diameter channels, one for each of the pressure lines. The plate was of sufficient length to project through the aluminum column past the periphery of the flanges. Compression fittings were screwed into the projecting end of the plate, to allow connection of pressure leads to the draft gauge.

The columns were insulated with approximately 2 inches thick glass wool.

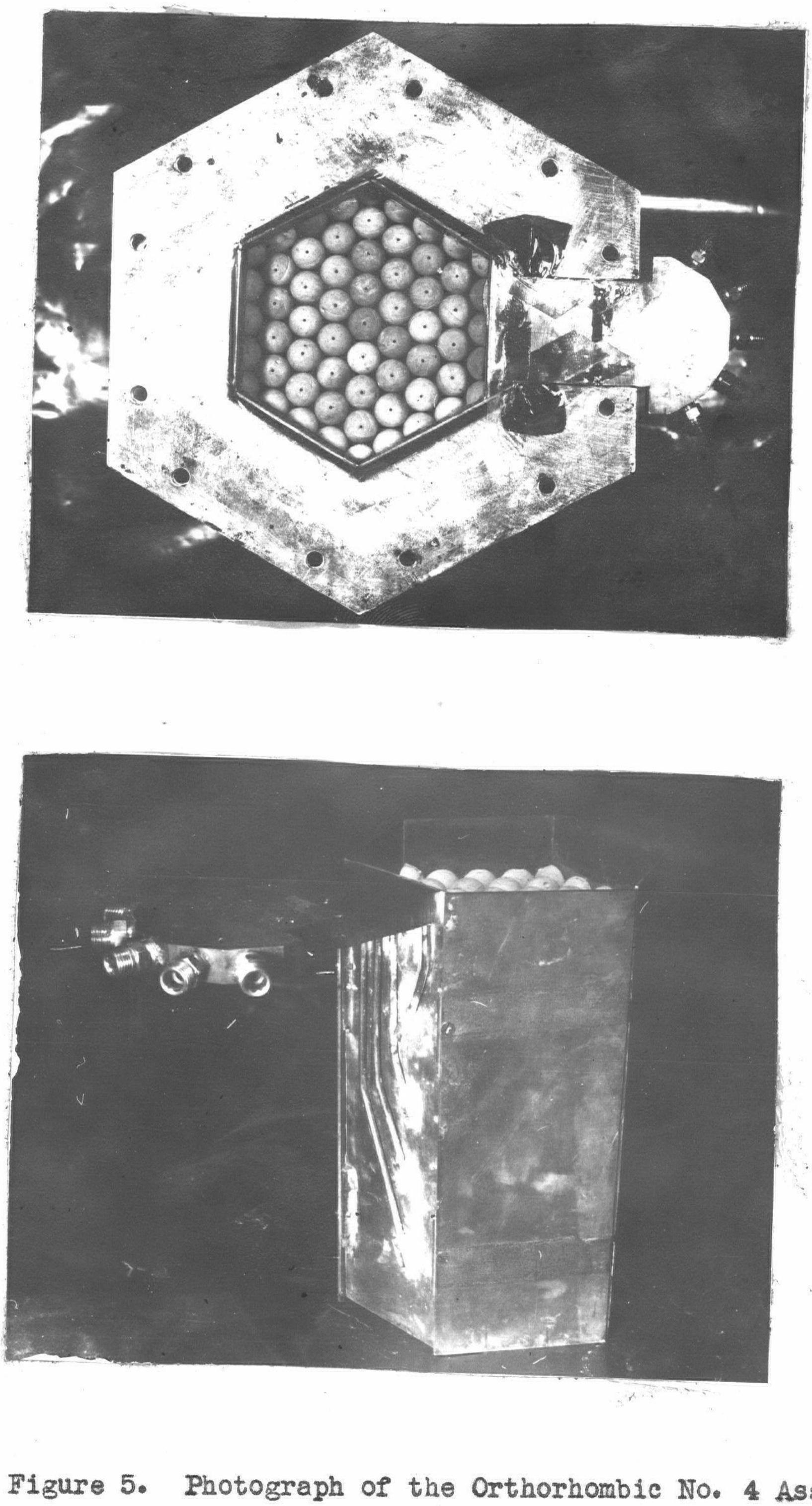


Figure 5. Photograph of the Orthorhombic No. 4 Assemblage

EXPERIMENTAL PROCEDURE AND RESULTS

1. OPERATING PROCEDURE

Each packing was soaked in tap water for a period of not less than three hours. The temperature of the water was controlled by placing the container holding the packing and water in a constant temperature bath. The temperature was held as close as possible ($\pm$ 3.0°F.) to the wet bulb temperature of the air entering the packing during the experimental run.

The packing, when removed from the water, was shaken vigorously to remove excess water, and then immediately placed in the bottom section of the column. In order to prevent air by-passing the packing by flowing in the small space between the outside wall of the packing and the inside wall of the column, this space was sealed off at the top of the column with scotch tape. A gasket of latex dental dam was used around the brass plate housing the pressure lines to prevent air leaking to the atmosphere. The entire operation of preparing the column for a run required about 15 minutes.

Once the column was secured in place, pressure lines attached, thermometers installed and insulation applied, the run was begun. The air rate was adjusted to the desired setting and the time clock started. Readings of inlet air temperature and humidity, orifice pressure drop, upstream pressure, pressure at the bottom of the packing, pressure drop through the packing, and pressure in the "Dewcel" sampling chamber were taken either every 15 minutes or every 30 minutes depending upon the rate of flow of air.

2. CALCULATING PROCEDURE

Orifice pressure drop, orifice upstream pressure, pressure at the bottom of the packing, and pressure drops through the packing were averaged from the data taken over the entire length of the run, thus eliminating the effect of small cyclical flow fluctuations caused by the on-off building compression.

Inlet and exit temperatures and humidities used in the calculations were taken at the point when the column was believed to have reached steady state. Some difficulty was experienced in deciding when this situation occurred for the lower flow rates. For runs of high flow rate the column reached steady state, as indicated both by a constant exit temperature and constant exit humidity, in approximately 15 minutes. However, at low flow rates, the time required to bring the temperature of the column and its large volume of insulation to a steady state condition was much longer, resulting in a slowly but detectably falling outlet air and packing temperature. The corresponding effect on outlet air humidity was even smaller. The procedure followed in this case was to use the data taken when the exit humidity had reached a constant value even though the exit temperature may not have become perfectly constant. Waiting for the exit temperature to become absolutely constant was not feasible in runs using low flow rates because there existed the danger of reaching the falling rate period of drying before complete steady state was attained.

Flow rates were calculated according to the method and equations set forth in the A.S.M.E. Report on flow meters (2). Appropriate temperature and pressure corrections were applied to convert from orifice to column conditions.

Moisture content of the air was determined from the dew point reading according to the method described by the "Dewcel" operating manual supplied by the Foxboro Company. This included a correction for deviation of the "Dewcel" chamber pressure from 760mm. of mercury.

The rates of liquid evaporation were calculated from the change in humidity of the air stream and the flow rate of air.

The mass transfer coefficient, k_g, was calculated according to equation 8 and the mass transfer factor, j_d, according to equation 18. The Schmidt number, which is temperature dependent but practically pressure independent, was plotted as a function of temperature (see appendix) and the value used in equation 18 was that corresponding to the average temperature in the column. In calculating k_g from equation 8, the log mean partial pressure difference of the transferring gas, $\Delta p_{l.m.}$, was evaluated by assuming that the surface temperature of the packing was equal to the wet bulb temperature of the air. Partial pressure of water vapor at the surface temperatures and at the dew point temperatures of the air were taken from the Foxboro operating manual for the "Dewcel". These values were identical with the values listed in Table I, page 762 of Perry (48).

The evaluation of the heat transfer coefficient was made according to equation 11. The log mean temperature difference was calculated from the assumed surface temperature and the measured air temperatures. The heat transfer factor, j_H, was evaluated according to equation 19. The Prandtl number was assumed to be constant over the small range of temperatures used in this investigation. It was given a value of 0.8280 at 70°F, which was calculated from a value of C_p = 0.2401 B.t.u./(lb.)(°F) as listed on page 79 of the International Critical Tables (30); k = 0.01284 B.t.u./(sq. ft.)(hr.)(°F/ft.) as listed on page 213 of the International Critical Tables (30); and $\mu = 1.23 \times 10^{-5}$ (lb.)/(ft.)(sec.) taken from Figure 2 of Gamson et al (24). The last mentioned plot is reproduced in the appendix and was used for determining all values of viscosity.

Friction factor was calculated according to equation 5a. Pressure drops between the top of the second layer and the top of the sixth layer were used for the calculations. Pressure drop data were plotted against the superficial velocity on a log-log plot, with the number of layers of spheres encompassed as a parameter. This resulted in four straight, parallel lines (see appendix). Calculation of the average incremental pressure drop per layer of packing from these lines showed that entrance and exit effects, if present at all, were very small. However, to ensure that such effects were not included in the calculated friction factors, the pressure drop across the four middle layers were used in calculating them. This is essentially the method employed by Martin et al (43).

The mass transfer factor, j_d, the heat transfer factor, j_H, and the friction factor, f, were plotted on log-log paper against the Reynolds number based on particle diameter, defined by

$$Re = \frac{D_p V_o \rho}{\mu} \tag{29}$$

Empirical equations giving j_H and j_d as exponential functions of Reynolds number were determined by the method of least squares.

3. RESULTS

Figures 6, 7 and 8 represent graphically all the results obtained from the main experimental portion of this work. The two assemblages showed entirely different fluid friction characteristics, but similar rates of mass and heat transfer.

The data for the mass transfer factor of both assemblages were correlated by the empirical equation

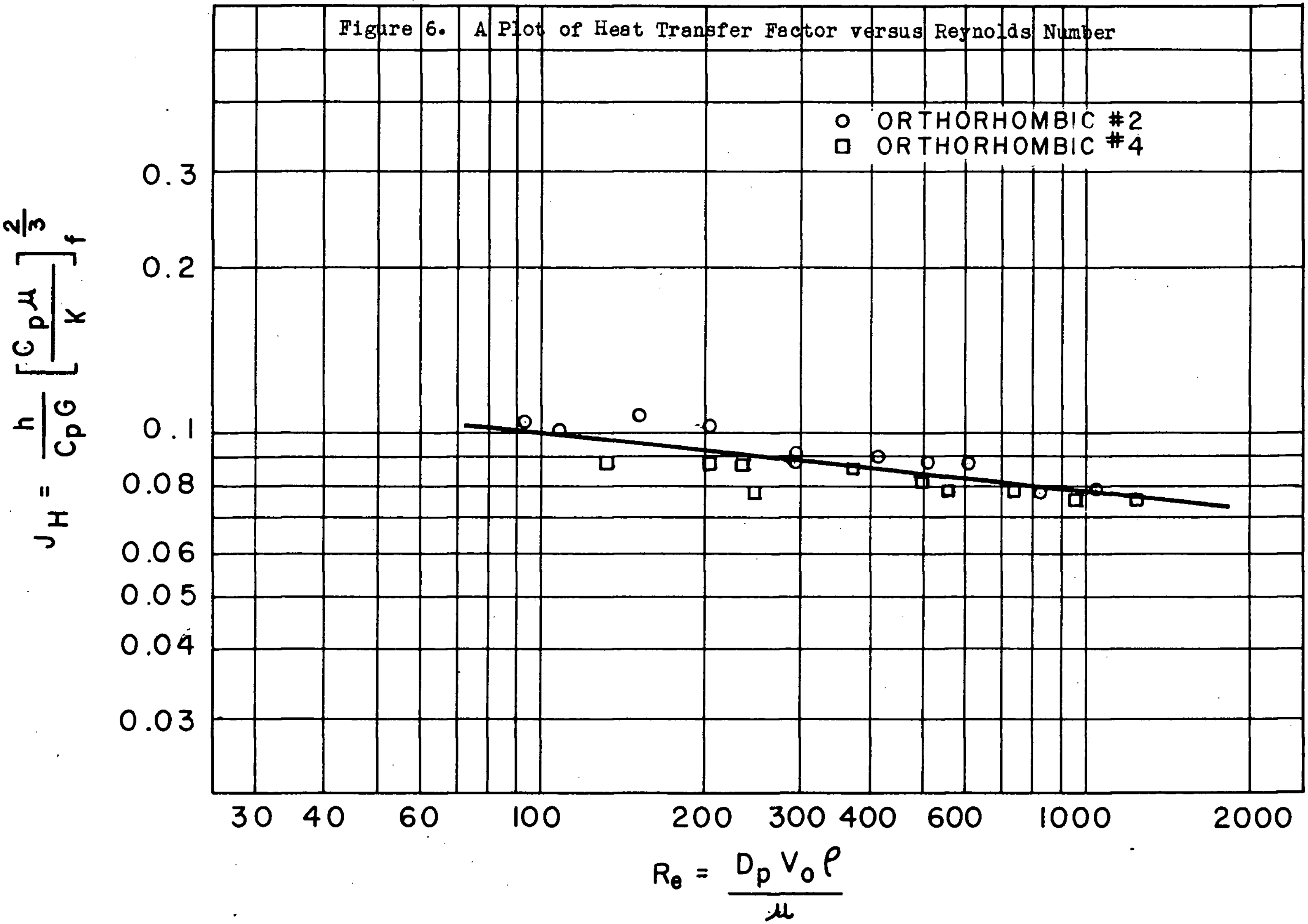


Figure 6. A Plot of Heat Transfer Factor versus Reynolds Number

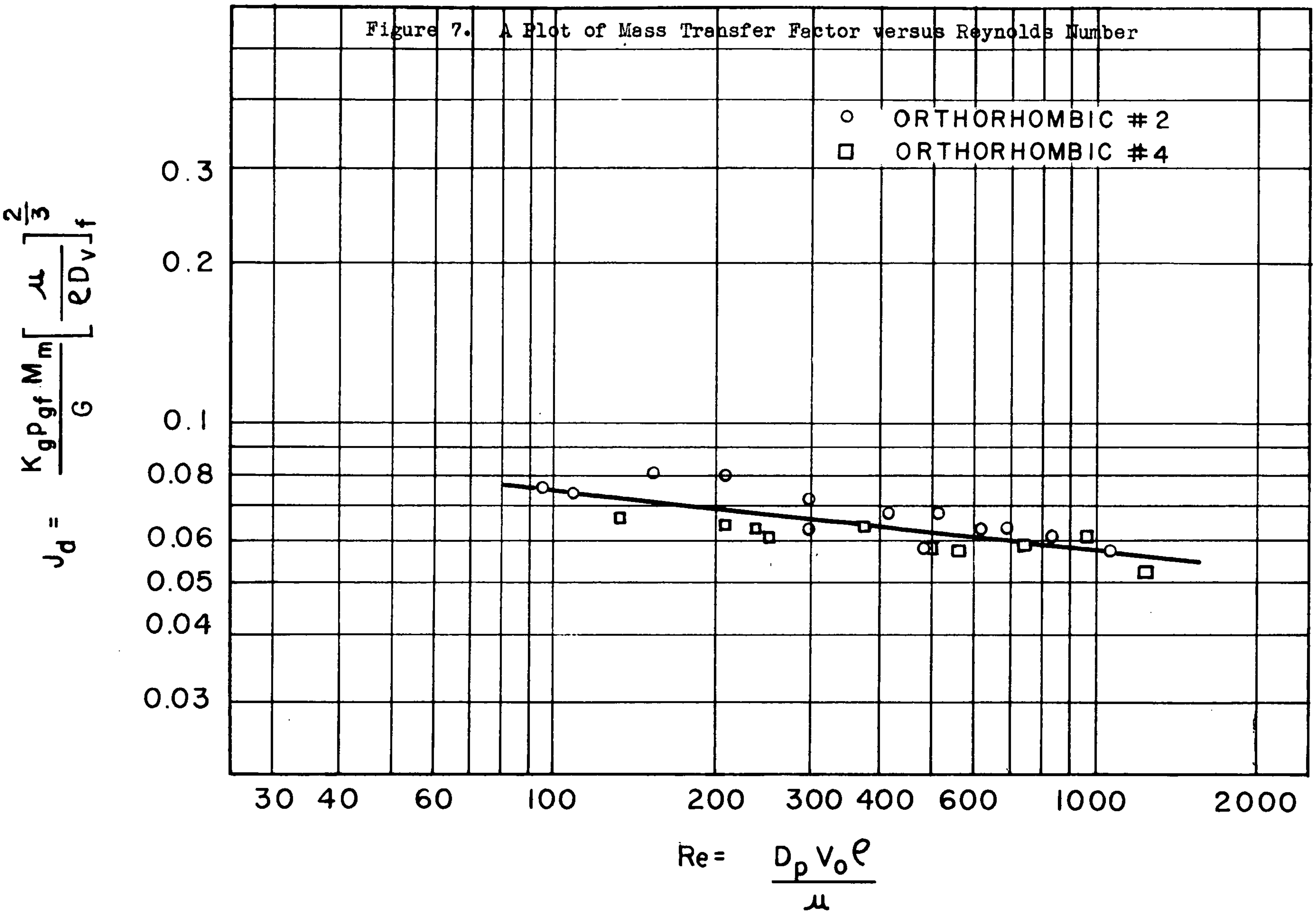


Figure 7. A Plot of Mass Transfer Factor versus Reynolds Number
ORTHORHOMBIC #2
ORTHORHOMBIC #4
$J_d = \frac{K_g P_{gf} M_m}{G} \left[\frac{\mu}{\rho D_v}\right]_f^{\frac{2}{3}}$
0.3
0.2
0.1
0.08
0.06
0.05
0.04
0.03
30
40
60
100
200
300
400
600
1000
2000
$Re = \frac{D_p V_o \rho}{\mu}$

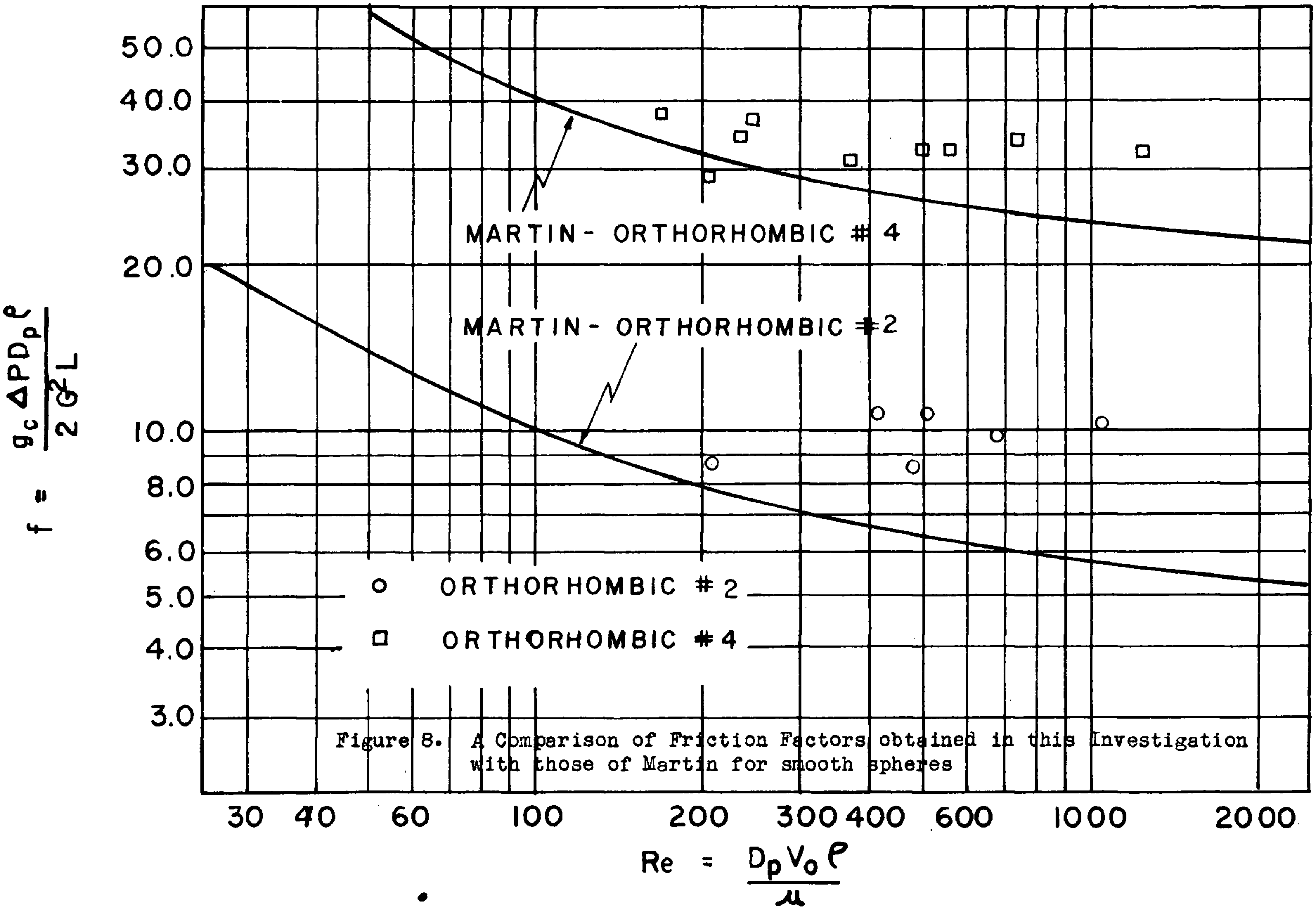


Figure 8. A Comparison of Friction Factors obtained in this Investigation with those of Martin for smooth spheres

$$j_d = 0.1261\ (Re)^{-0.1107} \tag{30}$$

with an average deviation of $\pm$ 6.05%, while the combined data for heat transfer were correlated by

$$j_H = 0.1669\ (Re)^{-0.1123} \tag{31}$$

with an average deviation of $\pm$ 4.78%.

Table II lists the observed values of j_H, j_d and f and the corresponding Reynolds numbers. The average ratio of heat transfer factor to mass transfer factor, j_H/j_d, was 1.310.

The original and calculated data are included in the appendix.

TABLE II

EXPERIMENTAL VALUES OF j_H, j_d, AND f

Run No.	Orientation	Re	j_d	j_H	f
2-1	Orthorhombic #2	607.6	0.06440	0.08727	
2-2		294.2	0.06408	0.08745	
2-3		824.6	0.06194	0.07767	
2-4		295.2	0.07290	0.09085	
2-5		480.3	0.05918	0.08036	8.555
2-6		679.0	0.06397	0.08059	9.749
2-7		414.5	0.06919	0.08993	10.74
2-8		207.6	0.08093	0.1026	8.658
2-9		151.6	0.08253	0.1083	
2-10		108.1	0.07521	0.1008	
2-11		94.56	0.07670	0.1053	
2-12		509.4	0.06865	0.08765	10.71
2-13		1049.4	0.05877	0.07856	10.28
4-1	Orthorhombic #4	741.0	0.06010	0.07822	33.93
4-2		956.2	0.06253	0.07467	
4-3		368.8	0.06458	0.08566	31.23
4-4		495.9	0.05893	0.08063	32.51
4-5		247.1	0.06198	0.07729	37.03
4-6		205.1	0.06520	0.08742	29.43
4-7		233.2	0.06441	0.08673	34.66
4-8		132.1	0.06669	0.08722	
4-9		557.2	0.05878	0.07857	32.67
4-10		1234.9	0.05344	0.07544	32.29

DISCUSSION

1. ASSUMPTION OF WET BULB TEMPERATURE AT THE SURFACE OF THE PACKING

The assumption that the surface of the packing is at the wet bulb temperature has become a very controversial issue. This assumption was first employed by Gamson et al (24), and later by Wilke and Hougen (61) and Taecker and Hougen (57). In their first paper Gamson et al (24) made no checks on the actual surface temperature, but in view of their excellent correlation ($\pm 3\frac{1}{2}\%$) they felt that this assumption was valid. Moreover, it was stated by T.H. Chilton during the discussion of this paper (24) that D.M. Hurt had made an attempt to determine " ... the temperature of the wetted solids during evaporation and as close as the experimental data could be obtained the check with the temperature of adiabatic saturation, or the wet bulb temperature, was as good as the agreement is between these two temperatures." Wilke and Hougen (61) found that after many trials surface temperatures could not be measured with any degree of accuracy by attaching thermocouples to the surface. Taecker and Hougen (57) report no attempts to measure surface temperature. Hobson and Thodus (27) doubted the accuracy of this assumption at low Reynolds numbers. In order to overcome this assumption they embedded thermocouples in the surface of the packing and have reported differences between wet bulb temperatures and measured surface temperatures as high as 5.5°F. In two out of the five runs made they report measured surface temperatures to be less than wet bulb temperatures. No attempt was made to make their process adiabatic, however, and exit wet bulb temperatures calculated from their data are consistently higher than the measured inlet wet bulb temperatures, the difference ranging from 1.7°F to 6.1°F. No mention is made of the temperature at which their packings were soaked.

An attempt was made in the present investigation to measure surface temperatures and to compare the measured surface temperature with the adiabatic saturation temperature of the air. Measurements were made in a random packed glass column, 3 inches in diameter, containing approximately 100 porous spheres similar to the spheres used in the orinetated packing. Height of the bed was approximately $5\frac{1}{2}$ inches. Surface temperatures were measured with thermocouples calibrated with a Leeds and Northrup platinum resistance thermometer certified by the National Bureau of Standards. Each thermocouple was placed in a groove inscribed in the surface of the sphere. Four such spheres were fitted with thermocouples and distributed at random throughout the bed in positions approximately 1, 2, 3, and 4 inches above the inlet of the bed. Reynolds numbers for each of these runs were in excess of 1200. The first few runs showed a decrease in measured surface temperature from inlet to exit. Adiabatic saturation temperature of the air was found to be less than the measured surface temperatures, although near the exit of the packing the difference was only 1.5°F. It was thought that contacting the thermocouples may have been causing some error in the measurement. Therefore, the thermocouples were shielded from direct contact with the air by placing a small strip of plastic adhesive tape over them. Runs with these shielded thermocouples showed marked reductions in the measured surface temperatures. Those measured 1 inch from the inlet differed from the adiabatic saturation temperature by as much as 4.5°F., while the surface temperatures measured 1 inch from the exit were only 0.4°F. above the adiabatic saturation temperature. In all runs the adiabatic saturation temperature of the inlet and exit air differed by only 0.2°F. In no case was the measured surface temperature less than the adiabatic saturation temperature.

It was believed that the air, which was higher in temperature than the surface of the spheres, was still affecting the temperature indicated by the thermocouples, causing them to read higher than the actual surface temperature. The plastic strip did prevent the thermocouples from being in direct contact with the air; however, quite conceivably, the plastic strip could be heated by the air to some degree, and since it was indirect contact with the thermocouple a higher temperature would be indicated. As the air proceeds through the packing, it is cooled. Hence the tendency of the air to cause the thermocouples to read higher than the actual surface temperature is reduced. This is indicated by the reduction in measured surface temperature proceeding from the inlet to the outlet of the packing.

The conclusions deduced from this preliminary investigation were that reliable surface temperature measurements could not be obtained by attaching thermocouples to the surface, and that the assumption of either wet bulb or adiabatic saturation temperature at the surface of the sphere was more accurate than direct measurement.

This argument would hold for the turbulent region of flow but extending it to the laminar and transition region without further investigation may be open to criticism. The data of Hobson and Thodus (27) would indicate that it could not be extended to the laminar region. However, the reliability of their measurements is open to question, especially in two cases where they report surface temperatures lower than the wet bulb temperature, despite the fact that the surroundings were at a higher temperature than the packing. It is hard to conceive that such a situation would occur at steady state, even in the unpredictable laminar and transitional zones.

In making runs with the orientated packings it was at first planned to run adiabatically. This was achieved with runs of high Reynolds number; however, with the lower flow rates the danger of entering the falling rate period of drying before adiabatic conditions were established became apparent Consequently, the wet bulb temperature, although only slightly different in value from the adiabatic saturation temperature, was considered to be a more reliable assumption of the surface temperature. A psychrometric chart was constructed using equation 47, page 812 of Perry (48) with a value of $h_c/k' = 0.26$ as reported in Table VII, page 100 of Sherwood and Pigford (55). This value does not include radiation effects, which were absent in the present set-up. Wet bulb temperatures were read from the chart, which is included in the appendix, to the nearest 0.1°F. using the measured values of dry bulb temperature and humidity. A similar chart was made for adiabatic saturation curves using equation 46, page 811 of Perry (48). Increases in wet bulb temperature from inlet to exit air streams were found to be never greater than 3.0°F and in most cases less than 0.5°F. Increases in adiabatic saturation temperatures were generally higher, though these never deviated by more than 1.5°F from the corresponding wet bulb temperatures.

Calculations of j_d for all runs were made using both wet bulb temperature and adiabatic saturation temperature as the assumed surface temperature. No noticeable change occurred in the spread of results; however, the assumption of wet bulb temperature at the surface yielded approximately 3% lower values of j_d. No noticeable difference in the values of j_H occurred.

2. EFFECTS OF ORIENTATION

Figures 6, 7 and 8 illustrate rather clearly that in the Reynolds number range covered, orientation has negligible effect on heat and mass transfer, whereas it has considerable effect on friction factor.

An explanation for the above results may be presented in view of work done with the flow of fluids past immersed bodies and past banks of heat exchanger tubes (32). The resistance to the movement of a solid in a fluid (or conversely, a fluid moving past a stationary solid) is known as drag.

This drag may be brought about by the shear stresses exerted in the boundary layer of the fluid next to the solid surface, in which case it is referred to as surface drag or skin friction.

In the case of fluid flow across circular cylinders, the pressure gradient in the fluid varies from negative to positive. This variation in pressure gradient causes the phenomenon of flow known as "separation" of the boundary layer. Separation of the boundary layer occurs at the point on the cylinder surface where the pressure gradient is zero. This can be visualized if a circular cylinder, placed at right angles to the fluid flow, is considered. As the fluid in the main stream flows past the cylinder, it is accelerated as a result of moving around the cylinder. This acceleration, which is an increase in kinetic energy, is accompanied by a decrease in pressure making the pressure gradient negative. However, as the fluid in the main stream goes past the cylinder, the expanding cross section of flow requires a deceleration of the fluid and a corresponding increase in pressure, making the pressure gradient positive. The boundary layer is thus flowing against an adverse pressure gradient as it moves around the cylinder. This results in a marked change in the velocity profile in the boundary layer. In order to maintain flow in the direction of this adverse pressure gradient, the boundary layer separates from the solid surface and continues in space. Beyond the point of separation of the boundary layer from the surface of the cylinder the fluid is flowing in a direction opposite to that in the main stream. Thus, the area behind the cylinder is an area of disturbed flow characterized by eddies. This area of disturbance beyond the cylinder is known as the turbulent wake.

If separation of the boundary layer accurs, causing a turbulent wake behind the solid body, a loss of energy in addition to that lost owing to surface drag also occurs. This loss of energy due to the turbulent wake is known as form drag and is a function both of the form or shape of the body past which the fluid is flowing, and of the Reynolds number.

An increase in turbulence which does not affect the laminar sub-layer results only in an increase in energy loss and does not appreciably increase the heat transfer (32). A turbulent wake behind an immersed body aids only slightly in transferring heat to the body but contributes to a considerable extent to the drag of the body (32).

Wallis (59), as reported by Knudsen and Katz (32), has studied visually the flow of fluids perpendicular to tube banks. The tube banks investigated were four different in-line or rectangular arrangements and four different staggered or triangular arrangements. The in-line arrangements compare, to some extent, with a cross-sectional view, taken parallel to the fluid flow direction, of the Orthorhombic No. 4 orientation used in this investigation while the staggered arrangement is similar to Orthorhombic No. 2 packing, taken in the same cross-section. Photographs of the pattern of fluid flow are shown. For the tubes in the in-line arrangement, it appears that the turbulent wake continues to the next tube in line and only a very thin boundary layer forms on that tube. For the closely packed staggered arrangement, the turbulent wake behind each tube is considerably reduced. The tubes are so placed that they are not in the turbulent wake of the tubes immediately upstream. This results in a considerable reduction of the size of the turbulent wake, and thus there should be a considerable reduction in energy dissipation (32).

It would seem, then, that here is a plausible explanation for the results obtained in this investigation. If the fluid flows in the packed beds according to the patterns witnessed by Wallis, then the spheres in the Orthorhombic No. 4 packing would have a greater turbulent wake on their downstream side than the spheres in the Orthorhombic No. 2. This would explain the fact that the Orthorhombic No 4 arrangement displays a considerably greater pressure drop than Orthorhombic No. 2. The reason that heat and mass transfer factors are not affected could be explained by the statement of Knudsen and Katz that this turbulent wake behind an immersed body aids only slightly in transferring heat from the body.

3. ANALOGIES BETWEEN HEAT, MASS AND MOMENTUM TRANSFER IN PACKED BEDS

The results of this investigation would appear to contradict any notion that a simple universal analogy exists between heat and mass transfer and momentum transfer. Because orientation does affect friction factor but not heat and mass transfer, in the turbulent region at least, some method must be introduced to take account of orientation.

Two statistically random packed beds would show no difference in orientation. It is doubtful, however, whether beds as they are packed in practice achieve such statistical randomness. This probably explains the fact that even the best correlations for fluid friction in "randomly" packed beds, though they employ elaborate functions to account for voids, still yield some spread in the data points (40). Attempts to express heat and mass transfer as a simple function of friction factor, without reference to orientation, are therefore, at best, approximate only. Furthermore, such attempts are strictly empirical and limited to particular cases, unless based on skin friction alone rather than on total drag. By subtracting form drag from total drag in the case of flow around a cylinder, Sherwood (53) extimated $f/2$ based on skin friction alone for flow normal to an isolated cylinder, and showed that it was very close to both j_H and j_d for this case. Unfortunately, the proportions of skin friction and form drag for other cases such as packed beds are not known at present.

4. SURFACE ROUGHNESS

In figure 8 the data for friction factor obtained in this investigation are compared with the friction factor curves for the same orientations obtained by Martin et al (43). In both cases the results are higher than those reported by Martin. This may be expected when it is considered that the spheres used by Martin were smooth steel ball bearings, while the packing material used here was an assemblage of rough refractory spheres. That is, the difference, it is believed, can be attributed to surface roughness, an effect recorded by other investigators (5, 7a, 40). Leva (40), for instance reports that, in turbulent flow, clay and alundum particles packed to the same voids as glass spheres, show a 50% increase in pressure drop, while rougher particles show an even greater increase. As clay and alundum are large constituents of the spheres used here, the results obtained are in accord with Leva's findings.

5. RELIABILITY OF THE DATA

It is difficult to make an overall quantitative estimate of the reliability of the data due to uncertainties arising out of the assumption of the surface temperature. However, it is possible to investigate the probable errors in isolated data.

The values of the heat transfer factor are believed to be more accurate than the mass transfer factor. If equation 8 is considered, the mass transfer coefficient is seen to be a function of the log mean partial pressure difference, $\Delta p_{l.m.}$, which is defined by

$$\Delta p_{l.m.} = \frac{(p_{w_1} - p_1) - (p_{w_2} - p_2)}{\ln \dfrac{p_{w_1} - p_1}{p_{w_2} - p_2}} \qquad (32)$$

The driving force at the top of the column ($p_{w_2} - p_2$), is generally quite small so that small errors in the values of p_{w_2} and p_2 result in large errors in the value of $\Delta p_{l.m.}$. This is also true for the log mean temperature difference, $\Delta t_{l.m.}$, which is used in the calculation of heat transfer coefficient. However, the errors in the measurement of individual temperatures are approximately 0.3% compared to approximately 1.6% for partial pressure terms. Calculations have shown that an approximate error of 0.3% in measuring temperatures could result in an approximate error of 2.5% in the log mean temperature difference, while 1.6% error in partial pressure terms could result in a 7.0% error in the log mean partial pressure difference.

Pressure drop data at Reynolds numbers below 150 for the Orthorhombic No. 4 arrangement and below 250 for the Orthorhombic No. 2 arrangement are not reliable due to the very small pressure drop. In this region the pressure drops were of the order of 0.010 to 0.020-inch of water, while readings could be estimated only to the nearest 0.005-inch of water. However, in the higher Reynolds number range, the results should be quite reliable.

6. COMPARISON WITH PUBLISHED RANDOM PACKING DATA

The values of j_H and j_d obtained in this investigation agree quite well with the results on random packing obtained by other workers (23, 24, 26, 27, 52) at a Reynolds number of 1000. However, the slope of the straight line through the points is found to be less than that reported by several investigators (23, 24, 27, 52, 57, 61). This discrepancy is, however, no greater than the discrepancies existing within the previously reported data (15). No reason can be put forth as to why this slope should be less than the slope reported by Gamson et al (24), who used the same system and who made the same assumption regarding surface temperature.

That a discrepancy exists in absolute values of j_H and j_d between those reported and those obtained here is, however, not important for the present purpose, which was not to measure absolute values of j_H and j_d, but rather to compare the results obtained from two different orientations, both measured on the same basis.

The ratio of j_H to j_d obtained here is slightly higher than that reported by Gamson et al (24) but agrees quite well with the value of 1.37 obtained by Scatterfield and Resnick (52).

PROPOSALS FOR FURTHER STUDY

1. Heat and mass transfer measurements on the two orthothombic assemblages should be extended into the laminar region in order to establish the effect of orientation where molecular transfer of heat and mass dominates completely over eddy transfer.

In order to reduce the time required for the column to reach equilibrium at these low flow rates, the inlet air should be heated to a point where its adiabatic saturation temperature is close to the room temperature.

To eliminate the possibility of entering the falling rate period of drying during the experimental run, studies should be made on each packing to determine the length of the constant drying rate period as a function of the Reynolds number through the packing. The length of time for each experimental run could then be safely determined in advance.

It would be necessary to know how the surface temperature of a material during the constant rate period of drying behaves at low flow rates of air. A number of ways of arranging thermocouples on or under the surface should be tried in order to determine some method of obtaining reliable surface temperature measurements.

2. A formal investigation of the effect of fractional void volume on heat and mass transfer rates can be made using the present apparatus. It would, however, require the construction of two or three additional packing assemblages of different voidage--for instance, a simple cubic which represents the loosest arrangement of spheres and a face-centred cubic which represents the tightest arrangement of spheres. Only one orientation per arrangement would have to be constructed, as the present investigation has already shown

that no appreciable orientation effect on heat and mass transfer exists in turbulent flow, while Martin's (43) fluid friction data points to no orientation effects for a given arrangement in laminar flow except for the two assemblages studied here.

Orderly arrangements of uniform spheres display a voidage range of 26% to 47.6%, while the spread between random dense and random loose beds of spheres is less than half this range (43). The advantage of studying fractional void volume in orderly arrangements is thus apparent.

SUMMARY

1. Experimental measurements have been made of the rates of heat, mass and momentum transfer in two packed beds having the same voidage, and the same arrangement when viewed in isolation, but different orientation with respect to the direction of fluid flow. The results indicate that over the range of Reynolds numbers covered orientation, while having considerable effect on pressure drop, has little or no measurable effect on the rates of heat and mass transfer.

2. The packing arrangements have been compared with in-line and staggered arrangements of heat exchanger tube banks. The observations made on these tubes have been used in an attempt to explain the results obtained in this investigation.

3. It is suggested that no simple analogy between momentum transfer and mass and heat transfer exists in packed beds. Neglecting the effects of orientation in deriving these analogies is believed to be erroneous in principle and, therefore, they can be regarded only as empirical approximations.

4. The empirical equations

$$J_d = 0.1261(Re)^{-0.1107}$$

and

$$J_H = 0.1669\ (Re)^{-0.1123}$$

have been used to relate the experimentally obtained values of J_H and J_d with Reynolds number over a Re-range of 100 to 1200. Average deviation in the mass transfer factor was $\pm$ 6.05% while that of heat transfer factor was $\pm$ 4.78%.

5. Friction factors were found to be higher than those reported for smooth spheres. This was attributed to surface roughness.

6. A number of attempts have been made to measure surface temperatures of the packing during the constant rate period of drying. The conclusions reached were that surface temperatures were difficult to measure reliably by attaching thermocouples to the surface, and that the assumption of wet bulb temperature was more accurate than direct measurement, at least in turbulent flow.

7. Proposals for further study have been presented and include the extension of the measurements of heat and mass transfer rates into the laminar region, an investigation to determine more reliable methods of measuring surface temperature and the initiation of a project to determine the effects of voids on heat and mass transfer rates.

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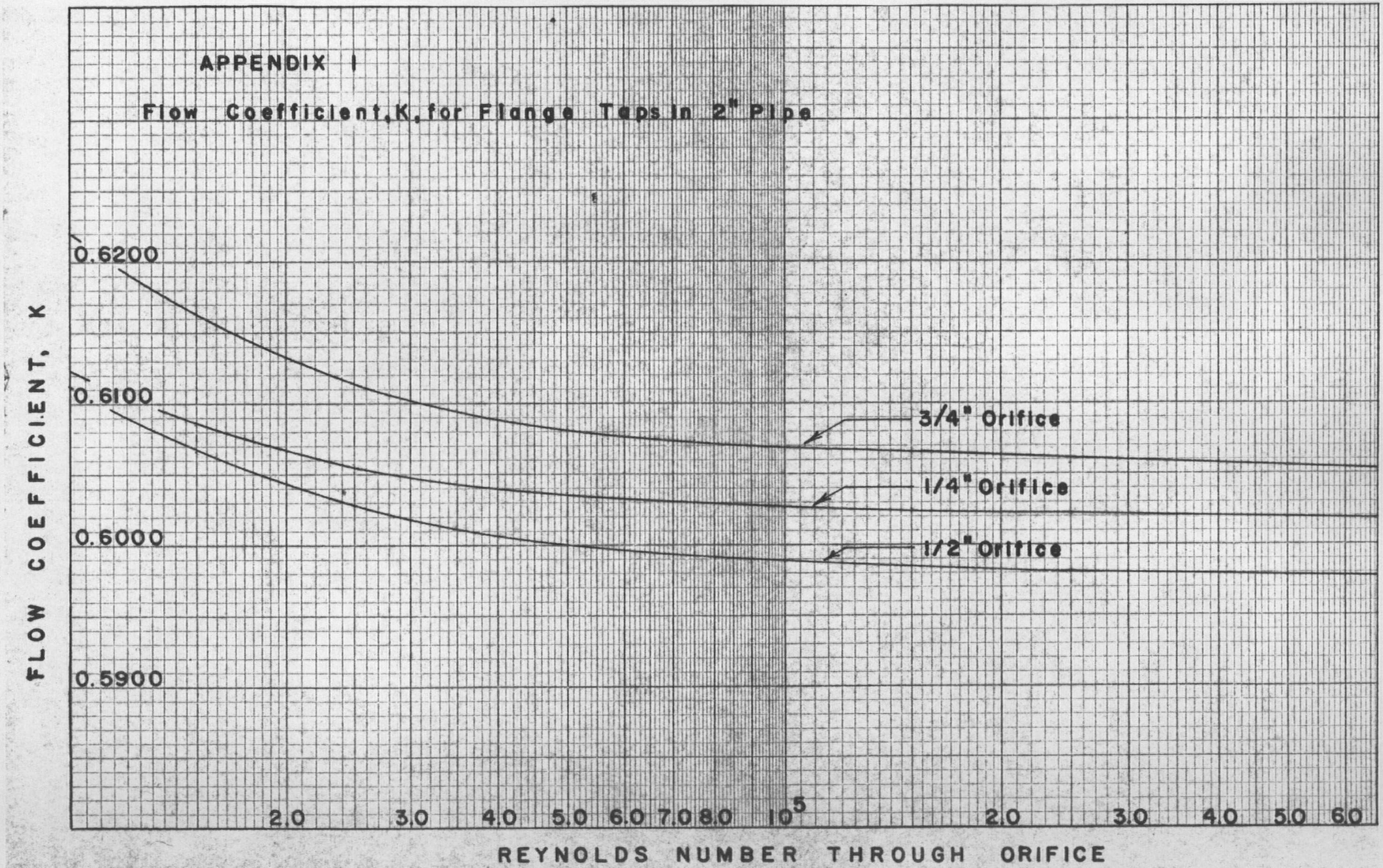
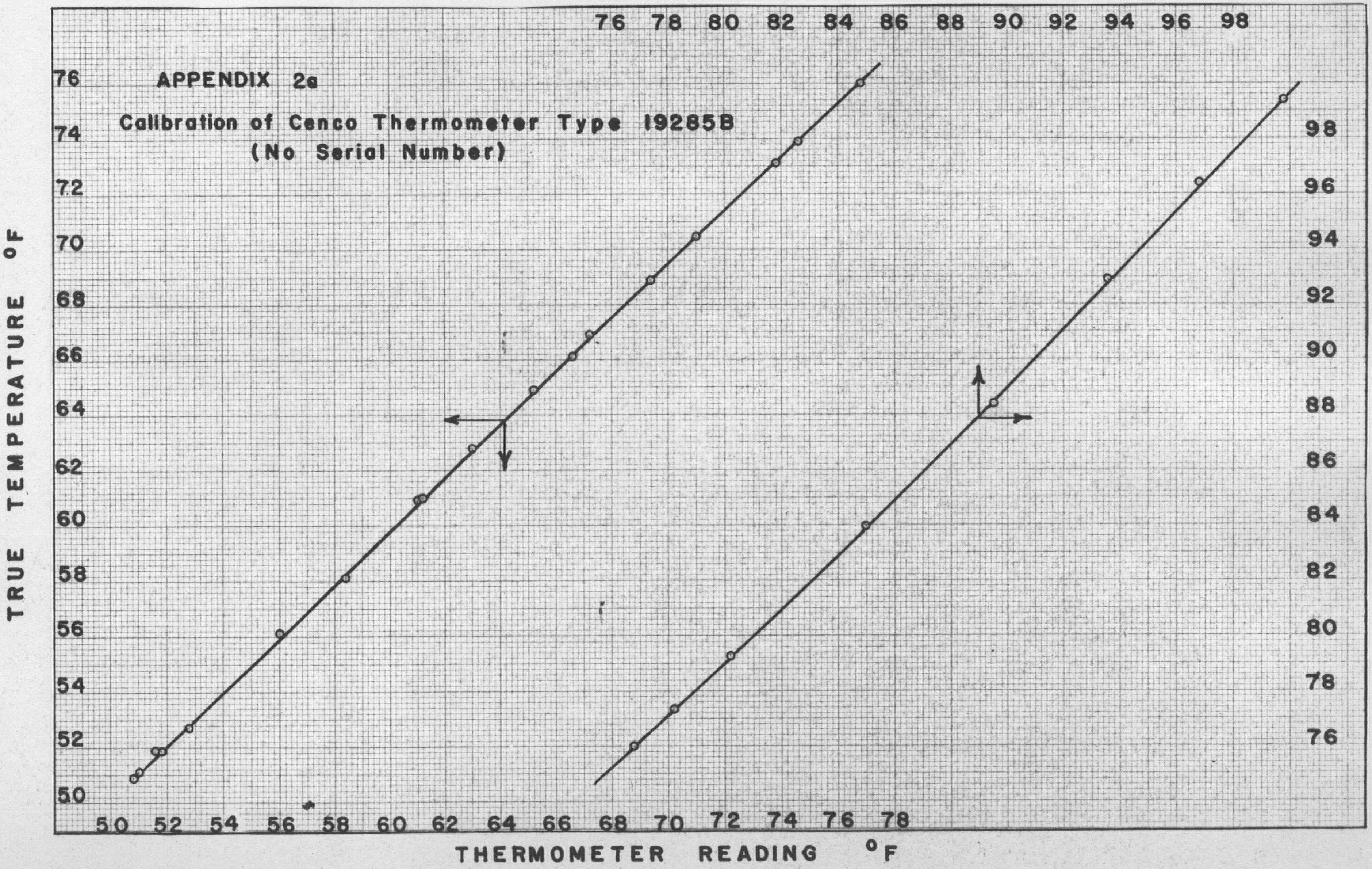
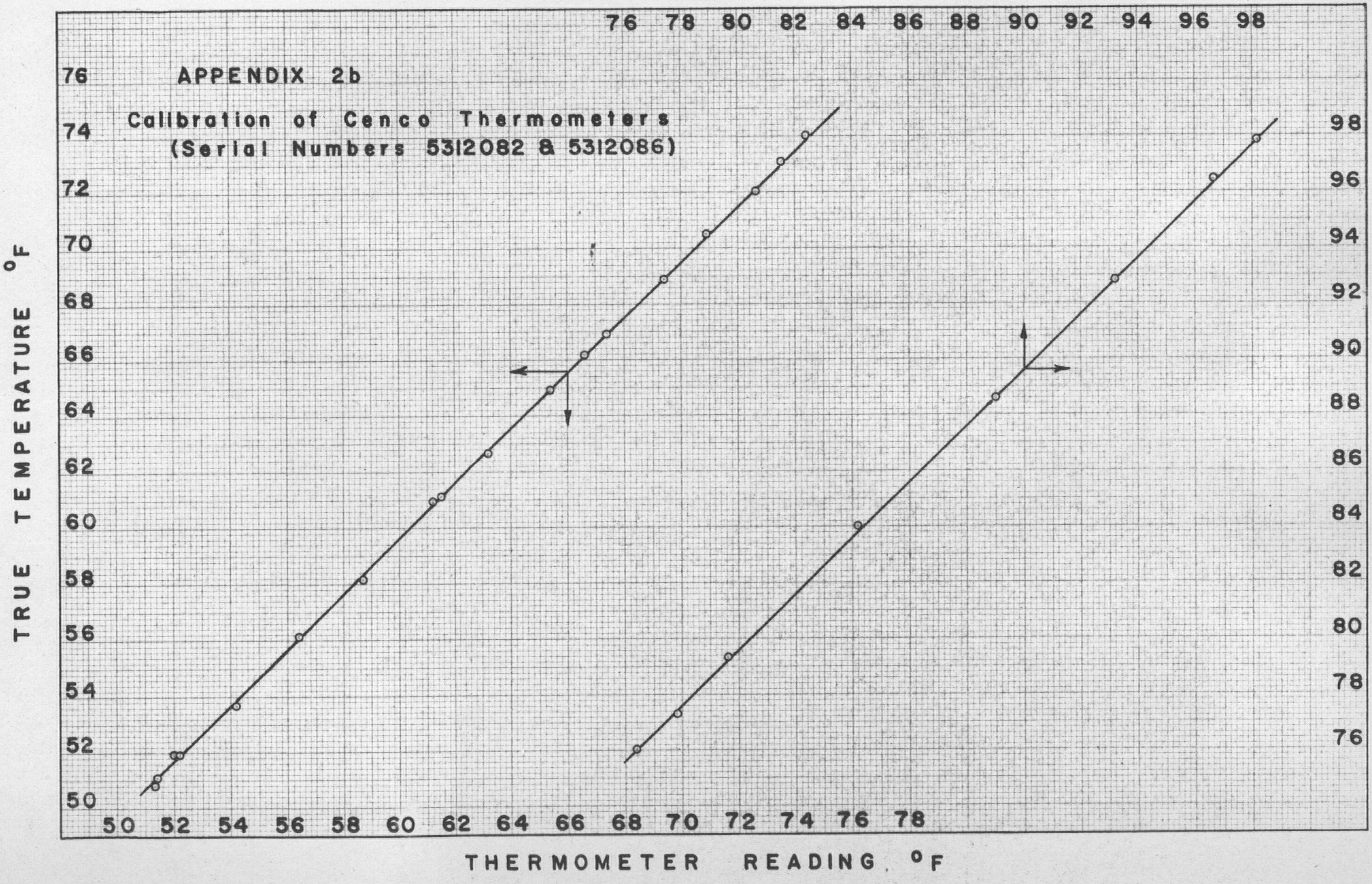
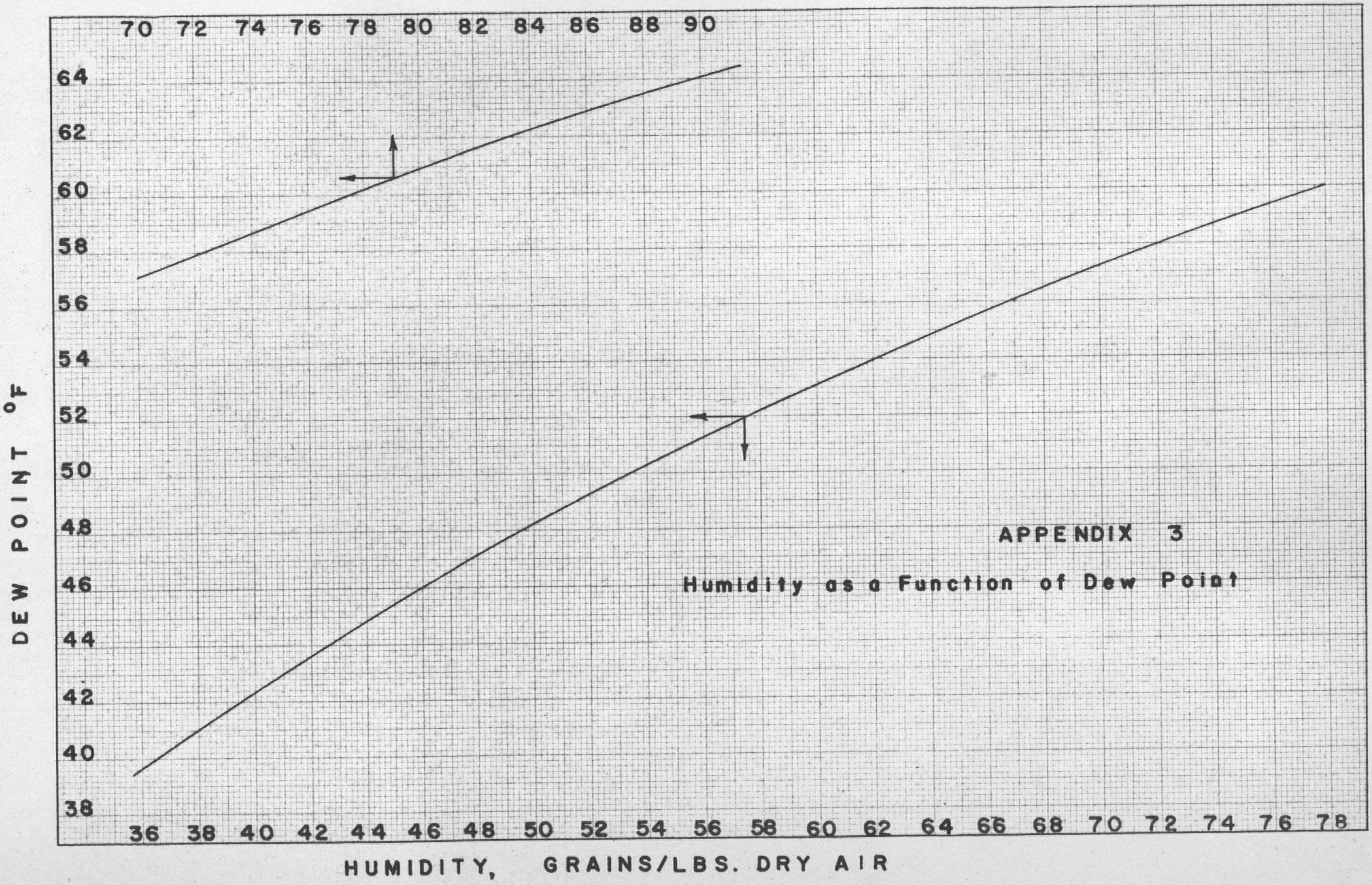
APPENDIX


APPENDIX I
Flow Coefficient, K, for Flange Taps in 2" Pipe
FLOW COEFFICIENT, K
0.6200
0.6100
0.6000
0.5900
3/4" Orifice
1/4" Orifice
1/2" Orifice
2.0
3.0
4.0
5.0
6.0
7.0
8.0
10^5
2.0
3.0
4.0
5.0
6.0
REYNOLDS NUMBER THROUGH ORIFICE

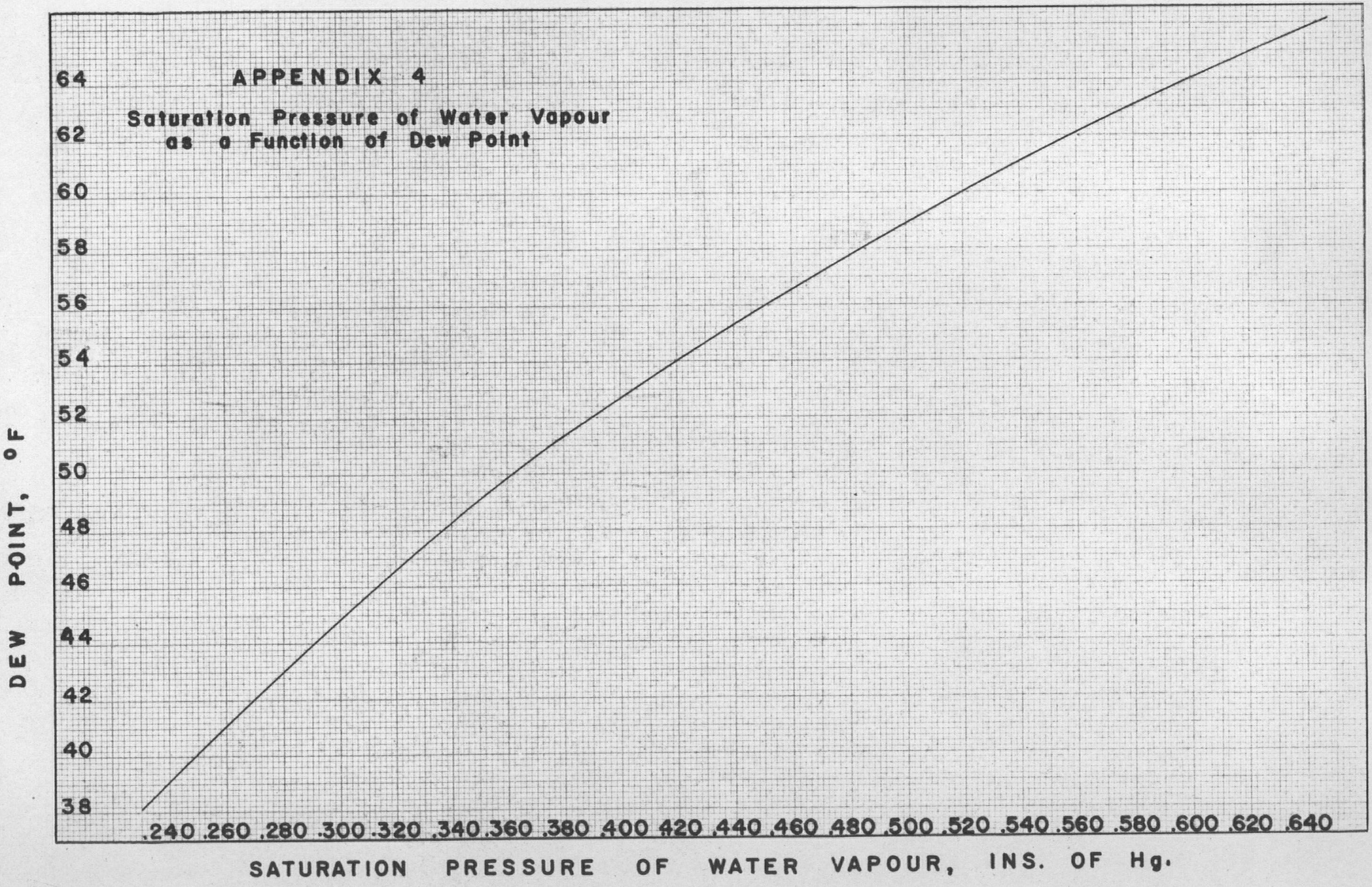

APPENDIX 2a
Calibration of Cenco Thermometer Type 19285B
(No Serial Number)
TRUE TEMPERATURE °F
THERMOMETER READING °F
76 78 80 82 84 86 88 90 92 94 96 98
76 74 72 70 68 66 64 62 60 58 56 54 52 50
98 96 94 92 90 88 86 84 82 80 78 76
50 52 54 56 58 60 62 64 66 68 70 72 74 76 78

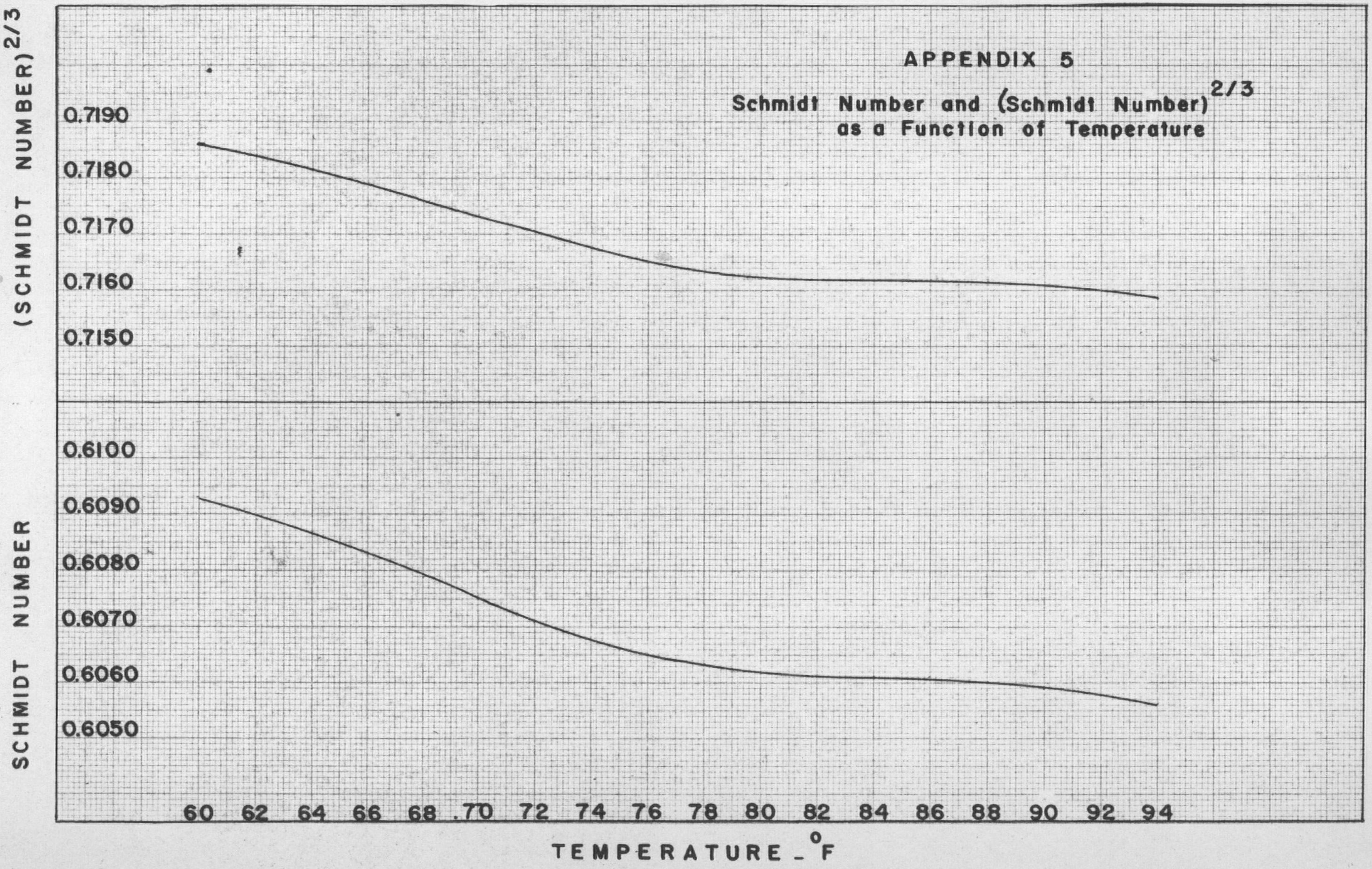

APPENDIX 2b
Calibration of Cenco Thermometers
(Serial Numbers 5312082 & 5312086)
TRUE TEMPERATURE °F
THERMOMETER READING °F
50 52 54 56 58 60 62 64 66 68 70 72 74 76 78
76 78 80 82 84 86 88 90 92 94 96 98
50 52 54 56 58 60 62 64 66 68 70 72 74 76
76 78 80 82 84 86 88 90 92 94 96 98

APPENDIX 3



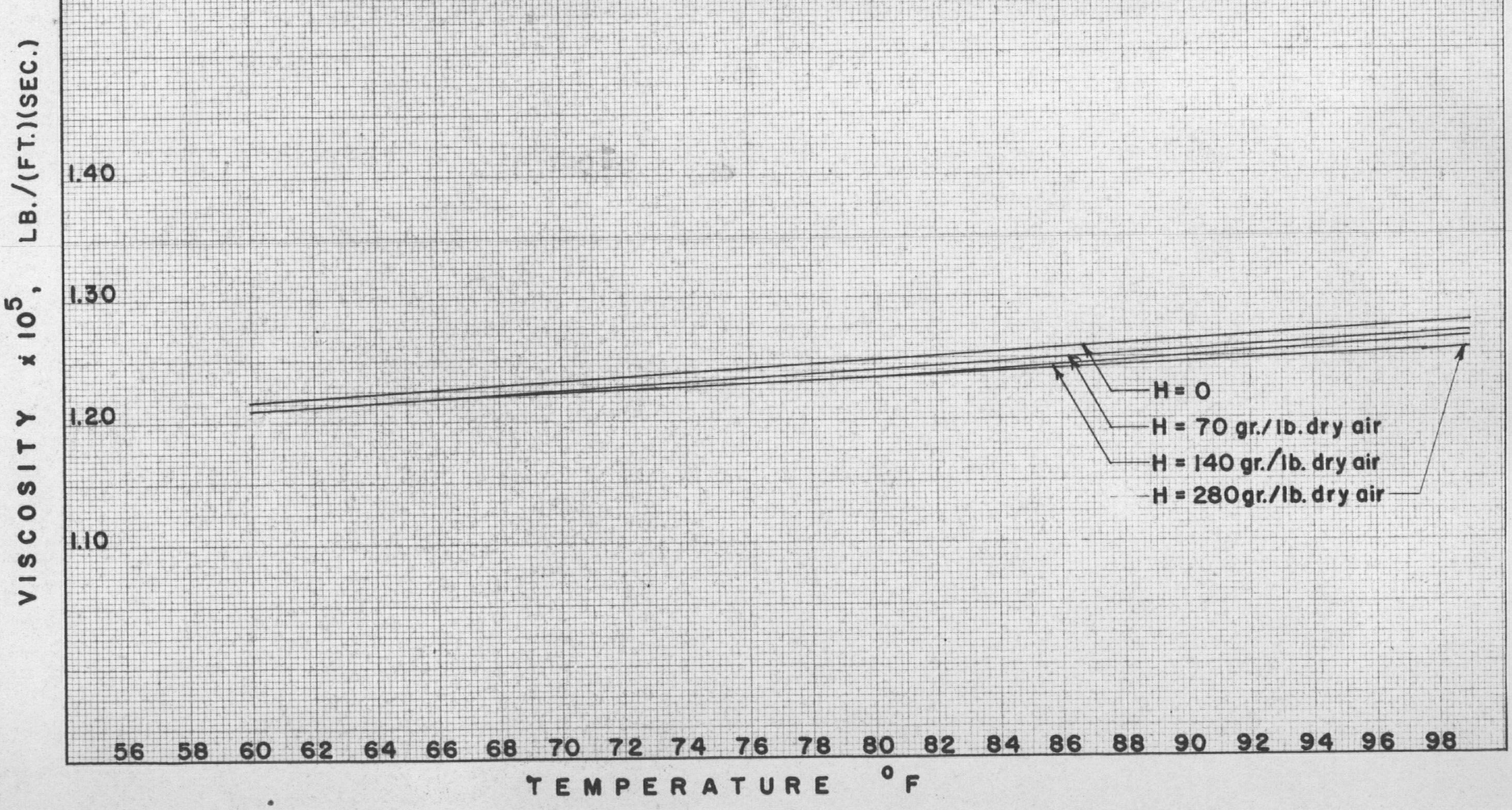
Humidity as a Function of Dew Point

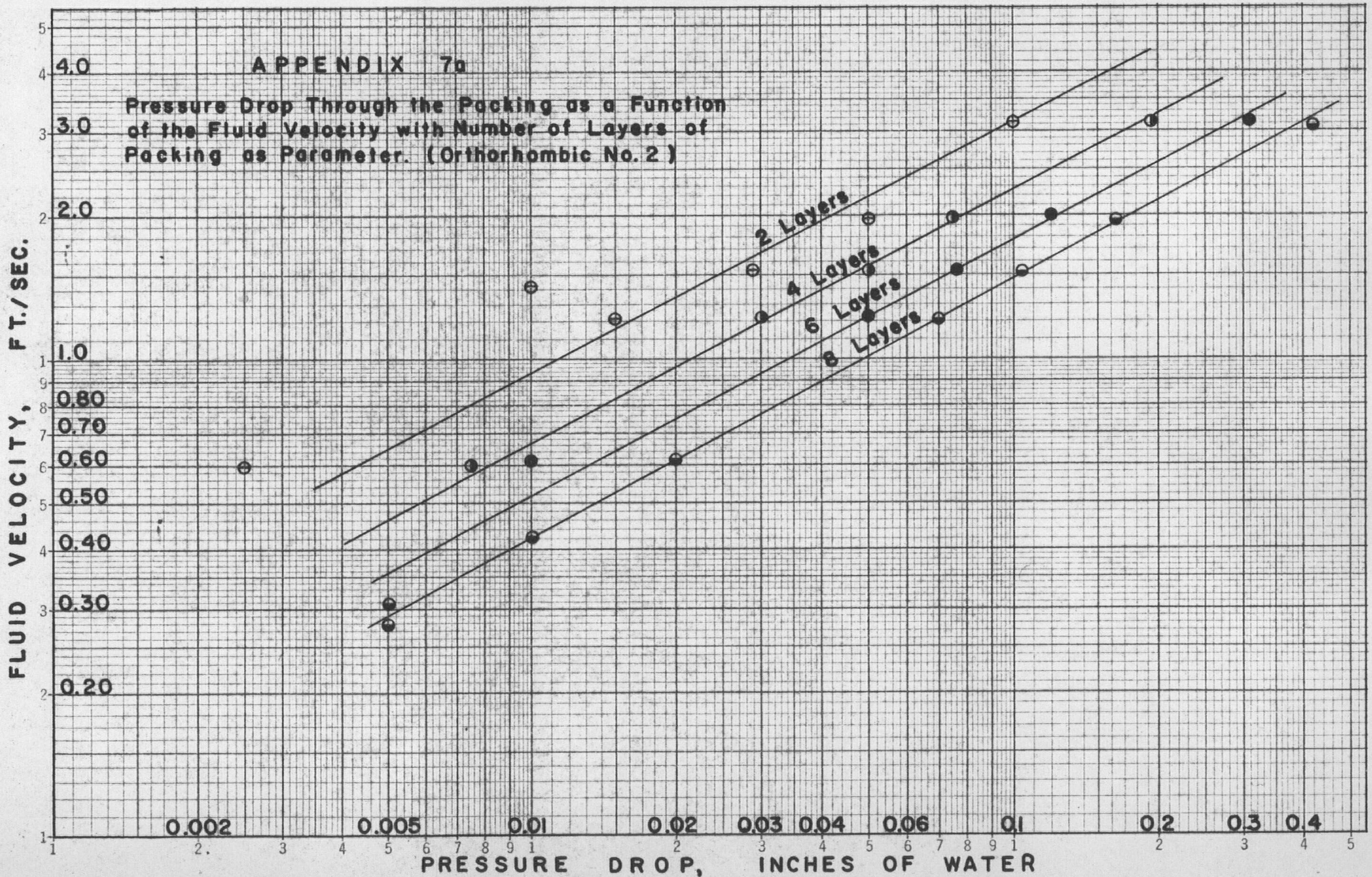

APPENDIX 4
Saturation Pressure of Water Vapour
as a Function of Dew Point
DEW POINT, °F
64
62
60
58
56
54
52
50
48
46
44
42
40
38
.240 .260 .280 .300 .320 .340 .360 .380 .400 .420 .440 .460 .480 .500 .520 .540 .560 .580 .600 .620 .640
SATURATION PRESSURE OF WATER VAPOUR, INS. OF Hg.

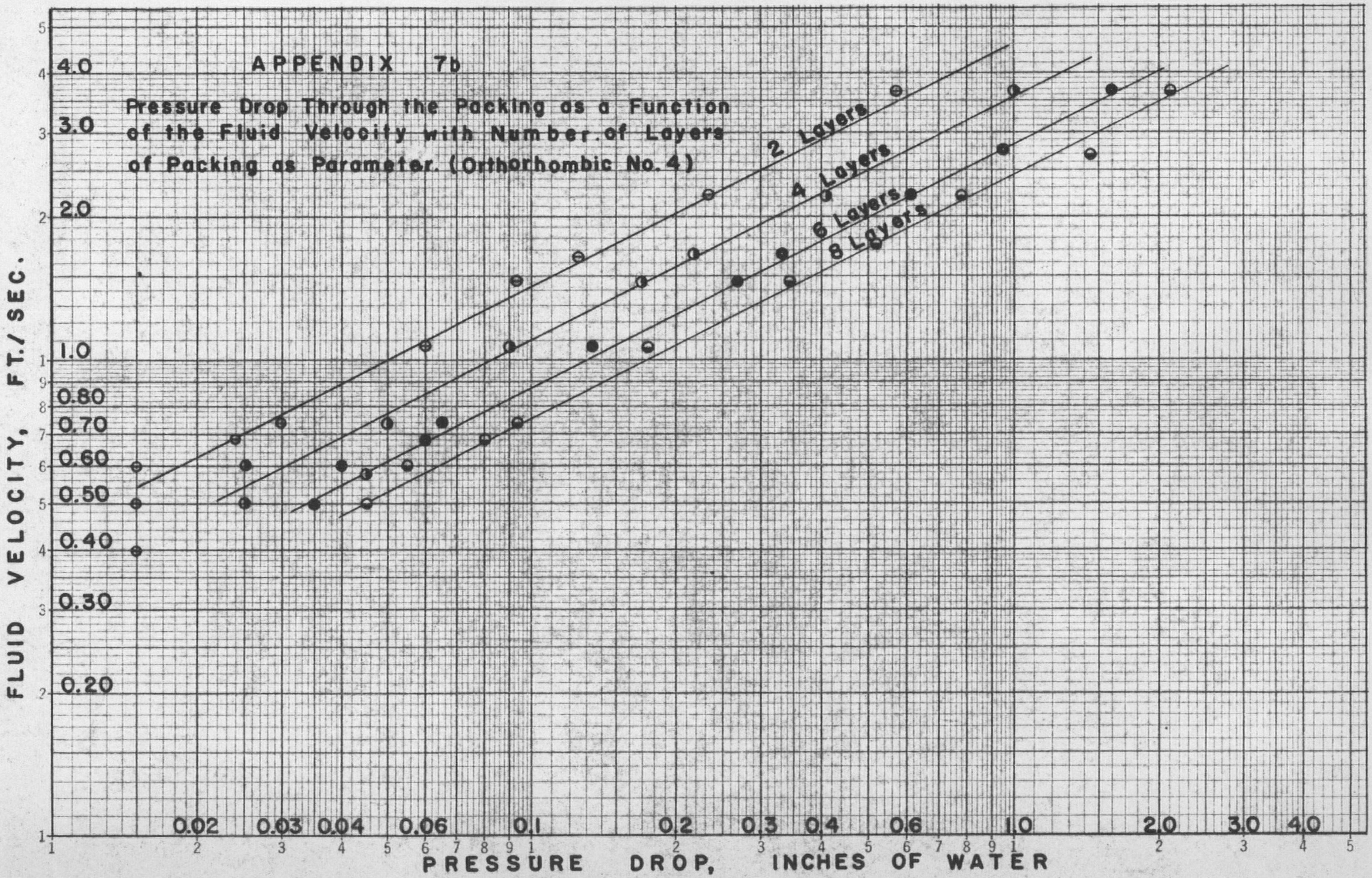

APPENDIX 5
Schmidt Number and (Schmidt Number)$^{2/3}$ as a Function of Temperature
(SCHMIDT NUMBER)$^{2/3}$
0.7190
0.7180
0.7170
0.7160
0.7150
SCHMIDT NUMBER
0.6100
0.6090
0.6080
0.6070
0.6060
0.6050
60 62 64 66 68 70 72 74 76 78 80 82 84 86 88 90 92 94
TEMPERATURE – °F

APPENDIX 6

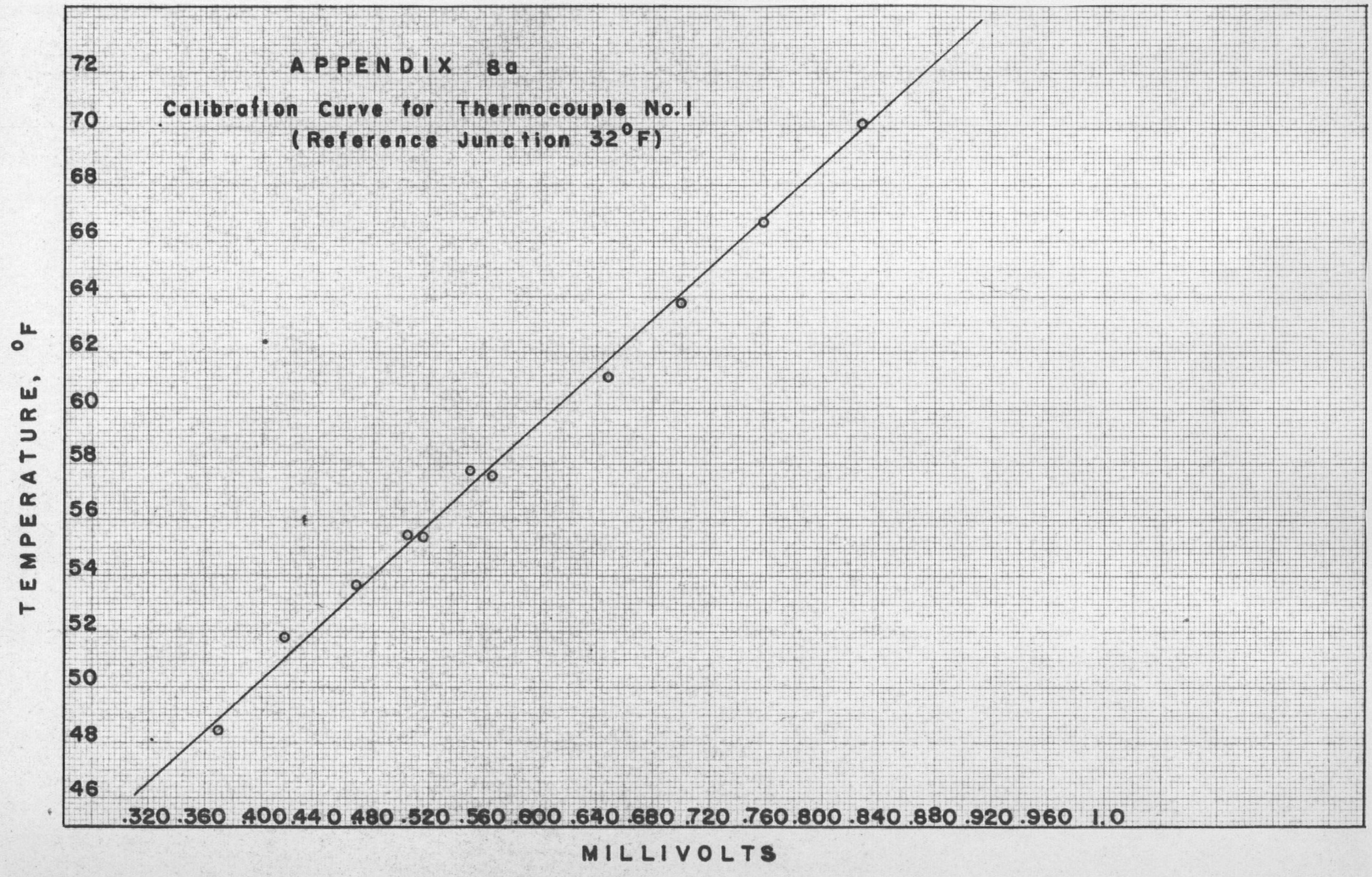
Viscosity of Air as a Function of Humidity and Temperature
(Taken from Gamson, Thodos and Hougen (24))

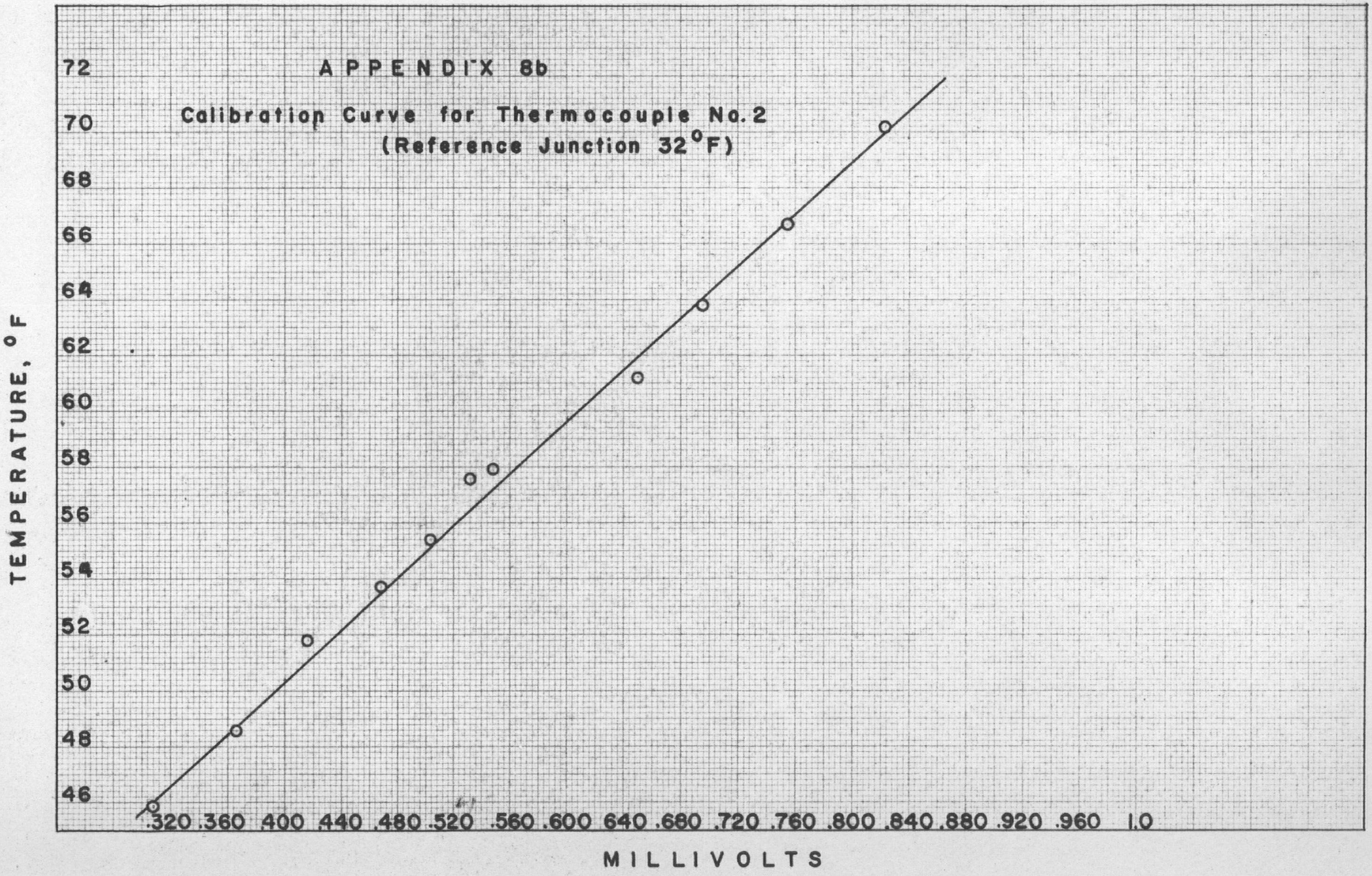


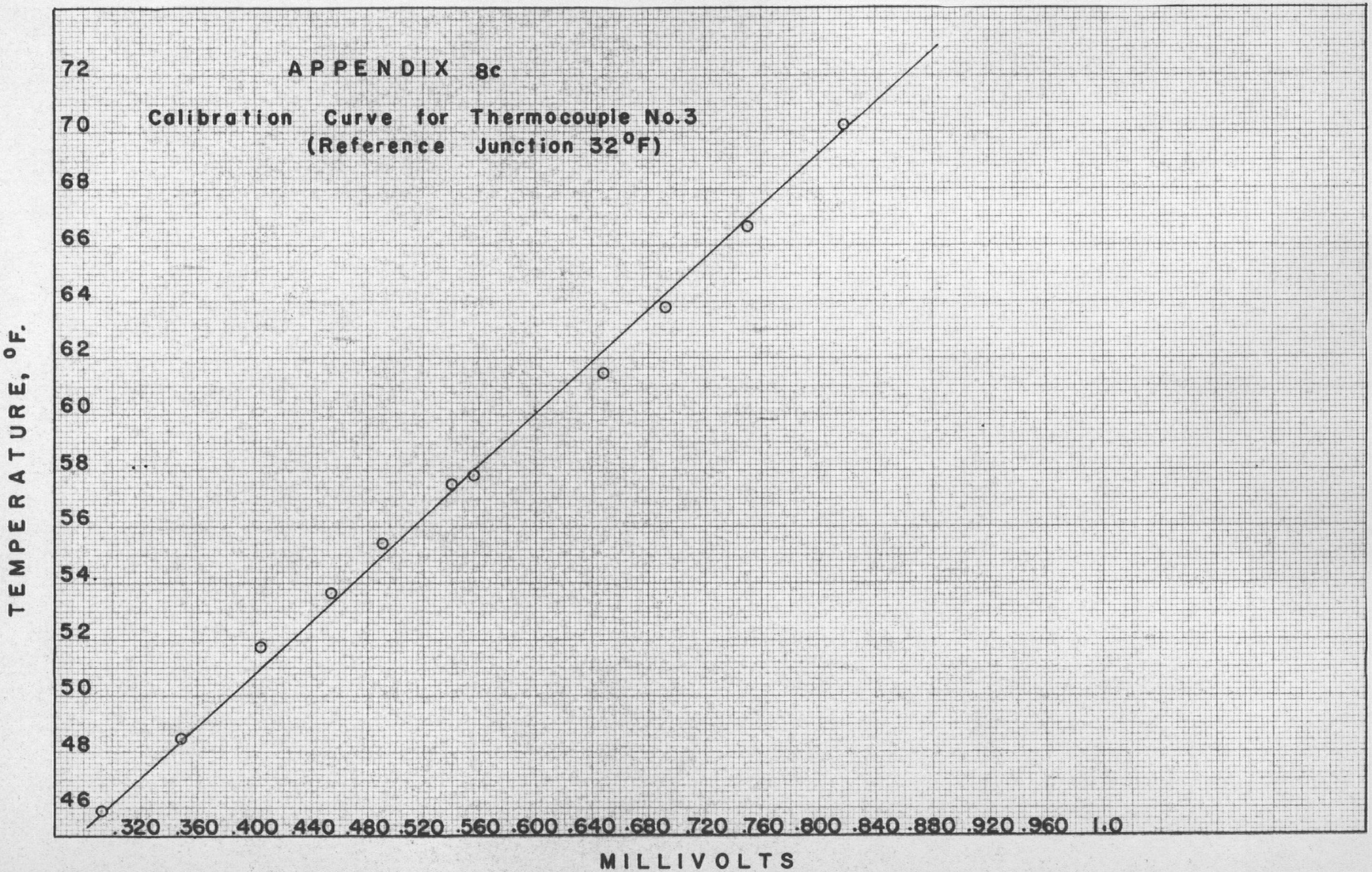

APPENDIX 7a
Pressure Drop Through the Packing as a Function of the Fluid Velocity with Number of Layers of Packing as Parameter. (Orthorhombic No. 2)
2 Layers
4 Layers
6 Layers
8 Layers
FLUID VELOCITY, FT./SEC.
4.0
3.0
2.0
1.0
0.80
0.70
0.60
0.50
0.40
0.30
0.20
PRESSURE DROP, INCHES OF WATER
0.002
0.005
0.01
0.02
0.03
0.04
0.06
0.1
0.2
0.3
0.4



APPENDIX 7b
Pressure Drop Through the Packing as a Function of the Fluid Velocity with Number of Layers of Packing as Parameter. (Orthorhombic No. 4)
2 Layers
4 Layers
6 Layers
8 Layers
FLUID VELOCITY, FT./SEC.
0.20
0.30
0.40
0.50
0.60
0.70
0.80
1.0
2.0
3.0
4.0
PRESSURE DROP, INCHES OF WATER
0.02
0.03
0.04
0.06
0.1
0.2
0.3
0.4
0.6
1.0
2.0
3.0
4.0

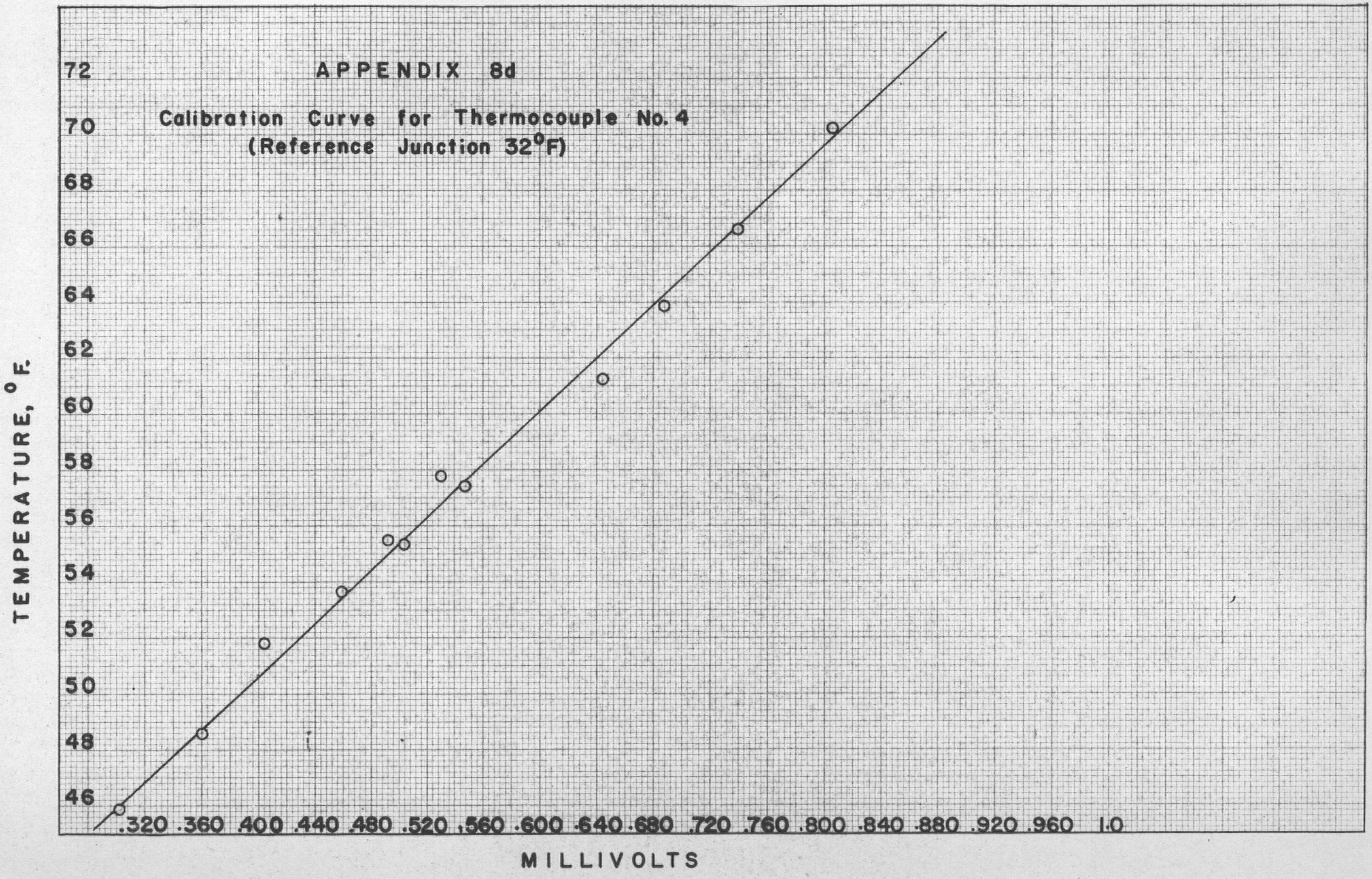

APPENDIX 8a
Calibration Curve for Thermocouple No. 1
(Reference Junction 32°F)
TEMPERATURE, °F
72
70
68
66
64
62
60
58
56
54
52
50
48
46
.320 .360 .400 .440 .480 .520 .560 .600 .640 .680 .720 .760 .800 .840 .880 .920 .960 1.0
MILLIVOLTS

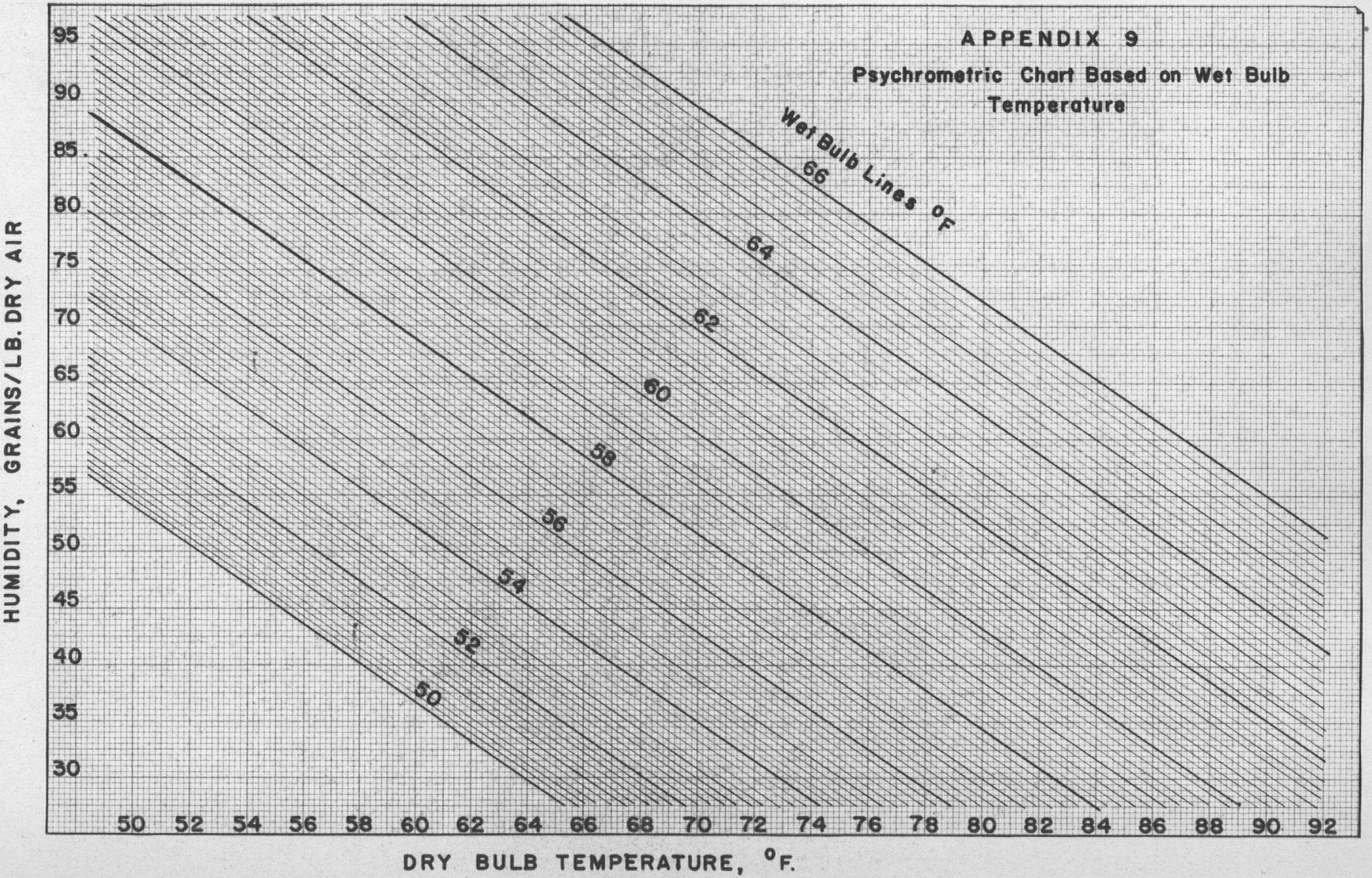

APPENDIX 8b
Calibration Curve for Thermocouple No. 2
(Reference Junction 32°F)
TEMPERATURE, °F
72
70
68
66
64
62
60
58
56
54
52
50
48
46
.320 .360 .400 .440 .480 .520 .560 .600 .640 .680 .720 .760 .800 .840 .880 .920 .960 1.0
MILLIVOLTS


APPENDIX 8c
Calibration Curve for Thermocouple No.3
(Reference Junction 32°F)
72
70
68
66
64
62
60
58
56
54
52
50
48
46
TEMPERATURE, °F.
.320 .360 .400 .440 .480 .520 .560 .600 .640 .680 .720 .760 .800 .840 .880 .920 .960 1.0
MILLIVOLTS

APPENDIX 8d

Calibration Curve for Thermocouple No. 4
(Reference Junction 32°F)





APPENDIX 9
Psychrometric Chart Based on Wet Bulb Temperature
Wet Bulb Lines °F
66
64
62
60
58
56
54
52
50
95
90
85
80
75
70
65
60
55
50
45
40
35
30
HUMIDITY, GRAINS/LB. DRY AIR
50 52 54 56 58 60 62 64 66 68 70 72 74 76 78 80 82 84 86 88 90 92
DRY BULB TEMPERATURE, °F.

APPENDIX 10. ORIGINAL DATA AND CALCULATED DATA

Run No.	Orientation	P across orifice in. H_2O	Absolute upstream pressure in. Hg	Temperature of inlet air °F.	Density of dry air at orifice lb./cu.Ft.	Dew Point of inlet air °F.	Correction to density for moisture	Density of moist air at orifice lb./cu.ft.
2-1	Orthorhombic #2	24.04	31.50	72.2	0.07854	39.0	0.9971	0.07831
2-2		5.78	30.23	84.0	0.07373	39.4	0.9970	0.07351
2-3		45.78	33.04	84.9	0.08047	42.4	0.9969	0.08022
2-4		5.89	30.50	83.7	0.07444	38.1	0.9970	0.07422
2-5		15.90	31.14	88.3	0.07537	41.9	0.9968	0.07513
2-6		30.61	32.16	80.2	0.07901	41.7	0.9969	0.07877
2-7		11.74	30.95	85.3	0.07532	40.8	0.9968	0.07508
2-8		44.42	33.04	79.9	0.08121	41.2	0.9970	0.08097
2-9		24.42	31.66	84.1	0.07722	39.5	0.9972	0.07700
2-10		12.27	30.72	78.5	0.07572	38.4	0.9971	0.07549
2-11		9.57	30.53	83.6	0.07453	38.5	0.9971	0.07431
2-12		17.52	31.25	88.3	0.07564	44.0	0.9964	0.07537
2-13		14.67	30.77	85.7	0.07483	46.3	0.9961	0.07454
4-1	Orthorhombic #4	27.30	32.01	88.6	0.07744	43.3	0.9966	0.07718
4-2		43.68	33.27	81.7	0.08151	42.8	0.9970	0.08127
4-3		6.03	30.37	78.4	0.08203	38.6	0.9970	0.08178
4-4		12.26	30.95	85.8	0.07525	41.7	0.0068	0.07501
4-5		47.64	33.30	91.3	0.08016	42.6	0.9970	0.07992
4-6		32.12	32.39	82.8	0.07919	40.5	0.9969	0.07894
4-7		40.57	32.86	78.5	0.08098	40.7	0.9970	0.08074
4-8		7.44	30.51	81.6	0.07476	39.6	0.9971	0.07454
4-9		15.43	31.17	85.2	0.07587	43.4	0.9965	0.07560
4-10		14.66	30.86	83.3	0.07538	47.1	0.9961	0.07509

APPENDIX 10. ORIGINAL DATA AND CALCULATED DATA (CON'T)

Orifice diameter	Expansion factor	Viscosity of air through orifice x 10^5	Discharge coeff.	Flow Rate	Reynolds number through orifice	Temperature of exit air	Average temperature of air in column	Average Absolute pressure in column
inch		lb./(ft.)(sec)		lb./sec.	x 10^{-4}	°F.	°F	in. Hg
0.500	0.9835	1.232	.6002	.02017	5.00	56.2	64.2	29.85
0.500	0.9959	1.250	.6030	.00990	2.42	63.3	73.6	29.81
0.500	0.9701	1.252	.5995	.02777	6.78	63.0	73.9	28.88
0.500	0.9959	1.250	.6030	.00990	2.42	61.7	72.7	30.08
0.500	0.9890	1.256	.6010	.01619	3.94	64.4	76.4	30.05
0.500	0.9794	1.243	.5999	.02274	5.59	60.5	70.4	30.05
0.500	0.9918	1.253	.6015	.01396	3.40	62.7	74.0	30.14
0.250	0.9710	1.247	.6047	.006942	3.40	59.9	69.9	29.83
0.250	0.9834	1.250	.6058	.005094	2.49	64.3	74.2	29.90
0.250	0.9914	1.241	.6075	.003620	1.78	62.5	70.5	29.82
0.250	0.9932	1.250	.6084	.003178	1.55	65.2	74.4	29.84
0.500	0.9979	1.257	.6008	.01717	4.17	64.3	76.3	30.04
0.750	0.9896	1.253	.6078	.03534	5.75	65.4	75.6	29.81
0.500	0.9816	1.257	.6002	.02132	5.18	63.9	76.2	30.05
0.500	0.9717	1.246	.5995	.02735	6.71	60.3	71.0	30.21
0.500	0.9957	1.241	.6027	.01051	2.59	58.5	68.5	29.93
0.500	0.9922	1.253	.6014	.01426	3.48	62.4	74.1	30.09
0.250	0.9691	1.261	.6046	.007128	3.45	64.9	78.1	29.91
0.250	0.9786	1.248	.6052	.005881	2.88	62.6	72.7	30.04
0.250	0.9734	1.241	.6048	.006647	3.27	59.3	68.9	29.92
0.250	0.9947	1.246	.6092	.002815	1.38	64.5	73.1	29.96
0.500	0.9893	1.253	.6010	.01600	3.90	63.5	74.4	30.09
0.750	0.0896	1.248	.6078	.03540	5.78	63.1	73.2	29.88

APPENDIX 10. ORIGINAL DATA AND CALCULATED DATA (CON'T)

Density of Air in Column	Superficial air velocity (based on empty column)	Viscosity of air in column x 10^5	Modified Reynolds number	Corrected Humidity inlet air	Dew Point exit air	Corrected Humidity exit air	Relative Humidity exit air	Adiabatic Sat'n Temp. inlet air
lb./cu.ft.	ft./sec.	lb./ft.(sec.)		gr./lb. dry air	°F.	gr./lb. dry air	%	°F.
0.07556	1.749	1.219	607.6	33.2	53.0	60.4	88.8	54.0
0.07414	0.8750	1.236	294.2	35.0	58.7	74.5	84.7	59.1
0.07426	2.4506	1.237	824.6	35.8	57.9	72.5	83.5	59.6
0.07493	0.8658	1.232	295.2	33.0	58.0	72.1	87.6	58.5
0.07434	1.4272	1.238	480.3	37.3	59.5	76.7	85.1	61.2
0.07519	1.9819	1.230	679.0	35.7	56.3	68.2	85.9	57.8
0.07490	1.2214	1.237	414.5	35.2	58.8	74.6	86.8	59.6
0.07468	0.6091	1.228	207.6	34.9	57.2	70.6	91.0	57.6
0.07428	0.4494	1.234	151.6	33.4	60.6	80.3	87.8	58.8
0.07460	0.3176	1.228	108.1	32.7	58.3	73.5	85.5	56.5
0.07410	0.2811	1.235	94.56	33.4	60.8	80.5	85.8	58.6
0.07434	1.5135	1.238	509.4	40.4	60.6	79.6	87.8	61.9
0.07386	3.1355	1.237	1049.4	44.6	60.6	80.0	84.6	61.9
0.07437	2.1999	1.238	741.0	38.3	60.2	78.6	87.7	61.5
0.07550	2.7799	1.230	956.2	36.2	56.9	69.6	88.4	58.6
0.07515	1.0732	1.225	368.8	33.7	55.9	67.2	91.1	56.7
0.07477	1.4636	1.237	495.9	37.6	59.1	75.5	88.6	60.4
0.07377	0.7415	1.241	247.1	35.8	60.7	80.1	86.5	61.9
0.07484	0.6030	1.233	205.1	34.1	59.0	75.2	87.6	58.5
0.07507	0.6795	1.226	233.2	33.7	56.3	68.3	90.0	56.7
0.07457	0.3897	1.233	132.1	34.9	60.1	78.2	85.6	58.2
0.07453	1.6474	1.235	557.2	39.5	59.8	77.3	87.5	60.6
0.07437	3.6528	1.233	1234.9	45.7	59.7	77.6	88.4	61.3

APPENDIX 10. ORIGINAL DATA AND CALCULATED DATA (CON'T)

Adiabatic Sat'n Temp. exit air	Wet Bulb Temp. of inlet air	Wet Bulb Temp. of exit air	Average Wet Bulb Temp.	Partial Press. of Water Vap. at t_{w1}	Partial Press. of Water Vap. in inlet air	Partial Pressure of Water Vap. at t_{w2}	Partial Press. of Water Vap. in exit air	Log Mean Partial Pressure Difference
°F.	°F.	°F.	°F.	in. Hg	in. Hg	in. Hg	in. Hg	in. Hg
54.4	54.6	54.5	54.6	0.430	0.239	0.428	0.404	0.08060
60.4	59.7	60.6	60.2	0.516	0.243	0.534	0.498	0.1171
60.0	60.2	60.0	60.1	0.526	0.272	0.522	0.485	0.1128
59.4	59.2	59.4	59.3	0.506	0.231	0.510	0.486	0.1030
61.5	61.8	61.5	61.7	0.557	0.267	0.553	0.512	0.1274
58.0	58.4	58.1	58.3	0.492	0.265	0.487	0.457	0.09745
60.4	60.2	60.4	60.3	0.526	0.256	0.530	0.500	0.1094
58.4	58.1	58.4	58.3	0.487	0.260	0.492	0.473	0.08394
62.2	59.4	62.1	60.7	0.510	0.244	0.564	0.534	0.1083
60.0	57.0	60.0	58.5	0.469	0.233	0.522	0.489	0.1033
62.6	59.2	62.6	60.9	0.506	0.234	0.574	0.538	0.1168
62.0	62.5	62.0	62.3	0.572	0.289	0.561	0.534	0.1091
62.6	62.4	62.5	62.5	0.570	0.315	0.572	0.534	0.1141
61.7	62.1	61.7	62.4	0.566	0.282	0.555	0.526	0.1119
58.3	59.1	58.3	58.7	0.585	0.276	0.490	0.467	0.08974
57.0	57.2	57.0	57.1	0.473	0.236	0.469	0.450	0.08648
60.5	61.0	60.5	61.3	0.542	0.265	0.532	0.505	0.1075
62.4	62.6	62.3	62.5	0.574	0.274	0.568	0.537	0.1187
60.5	59.1	60.5	59.8	0.505	0.254	0.534	0.503	0.1053
57.6	57.3	57.7	57.5	0.474	0.255	0.482	0.457	0.08949
61.8	58.8	61.8	60.3	0.498	0.245	0.557	0.524	0.1081
61.3	61.2	61.3	61.3	0.546	0.283	0.548	0.518	0.1074
61.2	61.8	61.2	61.5	0.557	0.325	0.546	0.516	0.09886

APPENDIX 10. ORIGINAL DATA AND CALCULATED DATA (CON'T)

Change in Humidity lb.water/ lb.dry air	Rate of Liquid Transfer lb.mole/ hr.	Mass Transfer Coeff. lb.mole/ (hr.)(atm) (sq. ft.)	Pressure at Bottom of Column in. Hg	Press. Drop through Packing in. Hg	Mean Part. Press. of non-trans. component atm.	Mass Velocity lb.mass/ (hr)(ft^2)	(Schmidt No.) 2/3	j_d	Log mean Temp. Diff. °F.
0.00388	0.01557	1.4939	0.19	0.02	0.9866	475.83	0.7182	0.06440	6.8104
0.00564	0.01110	0.7330	0.15	0.04	0.9840	233.55	0.7167	0.06408	9.8416
0.00524	0.02893	1.9834	0.27	0.05	0.9860	655.12	0.7167	0.06194	10.3048
0.00559	0.01100	0.8259	0.31	0.03	0.9933	233.55	0.7169	0.07290	9.3944
0.005628	0.01811	1.0993	0.4412	0.0061	0.9913	381.94	0.7165	0.05918	10.6788
0.004642	0.02099	1.6657	0.4493	0.0121	0.9923	536.46	0.7172	0.06397	8.8023
0.005628	0.01562	1.1041	0.4941	0.0050	0.9947	329.33	0.7167	0.06919	9.5506
0.005100	0.007039	0.6485	0.2610	0.0013	0.9843	163.77	0.7173	0.08093	7.5932
0.006700	0.006787	0.4846	0.2478	0.0008	0.9863	120.17	0.7168	0.08253	9.4466
0.005828	0.004190	0.3137	0.2390	0.0005	0.9846	85.28	0.7173	0.07521	8.8398
0.006728	0.004252	0.2815	0.1625	0.0004	0.9846	74.97	0.7167	0.07670	9.7470
0.005600	0.01910	1.3539	0.3257	0.0077	0.9903	405.06	0.7165	0.06865	9.7319
0.005057	0.03549	2.4054	0.2103	0.0308	0.9820	833.71	0.7166	0.05877	9.8010
0.005757	0.02439	1.7210	0.4926	0.0568	0.9916	589.04	0.7166	0.06010	9.7749
0.004771	0.02594	2.2819	0.6566	0.1066	0.9973	755.64	0.7172	0.06253	8.5050
0.004786	0.01000	0.9129	0.3221	0.0130	0.9890	290.38	0.7175	0.06458	7.4465
0.005414	0.01534	1.1265	0.4875	0.0254	0.9933	393.98	0.7167	0.05893	8.9239
0.006329	0.008968	0.5967	0.2507	0.0069	0.9863	196.94	0.7164	0.06198	10.8807
0.005871	0.006866	0.5147	0.3963	0.0040	0.9916	162.48	0.7170	0.06520	8.9225
0.004942	0.006532	0.5762	0.3162	0.0059	0.9883	183.65	0.7175	0.06441	7.5936
0.006185	0.003461	0.2528	0.1934	0.0012	0.9886	77.77	0.7169	0.06669	9.4316
0.005400	0.01717	1.2621	0.3897	0.0309	0.9923	442.06	0.7167	0.05878	9.1332
0.004557	0.03203	2.5578	0.4044	0.1544	0.9846	978.05	0.7169	0.05344	8.0875

APPENDIX 10. ORIGINAL DATA AND CALCULATED DATA (CON'T)

Heat of Evap'n at Average Surface Temp. B.t.u./lb.	Heat Transferred B.t.u./hr	Heat Transfer Coeff. B.t.u./(hr)(ft^2)(°F.)	j_H	Press. Drop across 2 Layers in. H_2O	Press. Drop across 4 Layers in. H_2O	Press. Drop across 6 Layers in. H_2O	Press. Drop across 8 Layers in. H_2O	Press. Drop between 2 and 6 Layers in. H_2O	f
1062.2	297.95	11.3074	0.08727						
1058.8	211.76	5.5612	0.08745						
1059.9	552.42	13.8555	0.07767						
1059.5	209.99	5.7773	0.09085						
1058.2	345.29	8.3571	0.08036	0.010	0.044	0.054	0.080	0.054	8.555
1060.1	400.93	11.7724	0.08059	0.0150	0.074	0.120	0.165	0.120	9.749
1058.9	297.97	8.0637	0.08993	0.0150	0.035	0.050	0.070	0.050	10.735
1060.1	134.42	4.5754	0.1026	0.0025	0.0075	0.010	0.020	0.010	8.658
1058.7	129.48	3.5426	0.1083	0	0	0	0.010		
1060.0	80.02	2.3396	0.1008	0	0	0	0.005		
1058.6	81.09	2.1503	0.1053	0	0	0	0.005		
1057.8	363.99	9.6660	0.08765	0.029	0.047	0.076	0.105	0.076	10.707
1057.7	676.29	17.8343	0.07856	0.110	0.210	0.311	0.419	0.311	10.278
1057.7	464.75	12.5448	0.07822	0.232	0.378	0.610	0.784	0.610	33.93
1059.8	495.24	15.3638	0.07467	0.950			1.45		
1060.7	191.14	6.7726	0.08566	0.050	0.085	0.135	0.175	0.135	31.23
1058.4	292.54	8.6494	0.08063	0.093	0.167	0.260	0.345	0.260	32.51
1057.7	170.92	4.1447	0.07729	0.030	0.045	0.075	0.094	0.075	37.03
1057.2	130.78	3.8673	0.08742	0.015	0.025	0.040	0.055	0.040	29.43
1060.5	124.82	4.3370	0.08673	0.025	0.035	0.060	0.080	0.060	34.66
1058.9	66.02	1.8469	0.08722	0	0	0	0.015		
1058.4	327.36	9.4571	0.07857	0.127	0.203	0.330	0.420	0.330	32.67
1058.3	615.82	20.0908	0.07544	0.570	1.03	1.60	2.10	1.60	32.29