HEAT AND MASS TRANSFER
IN PACKED TOWERS

By

Jack B. Pohlenz

B.S.(Chem. Eng.) University of Colorado
1944

SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR THE DEGREE OF
DOCTOR OF SCIENCE

From the
Massachusetts Institute of Technology
1947

Signature of Author______________________

Department of Chemical Engineering, May 16, 1947

Signature of Professor in Charge of Research______________

Signature of Chairman of Departmental
Committee on Graduate Students______________

Signature of Head of Department______________

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May 16, 1947

Professor J. S. Newell
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Sir:

In accordance with the regulations of the Faculty, I hereby submit a thesis, entitled "Heat and Mass Transfer in Packed Towers", in partial fulfillment of the requirements for the degree of Doctor of Science in Chemical Engineering from the Massachusetts Institute of Technology.

Yours very truly,

Jack B. Pohlenz

ACKNOWLEDGMENT

The author wishes to express his deep appreciation to Professor William H. McAdams under whose supervision this thesis was performed. His active interest, encouragement, and many helpful suggestions were of inestimable value.

The author's thanks are also due the Research Corporation without whose financial aid it would not have been possible to carry out this research program.

The assistance of Messrs. R.C. St.John, W.C. Philoon, K.L. Hujsak, W.P. Herberg, and W.J. Rappaport in performing a large portion of the experimental work on the various phases of the project is appreciated.

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I. ABSTRACT

Introduction. - Many operations for which the chemical engineer is responsible involve the interphase transfer of heat, mass, or of both heat and mass. These processes are generally classified as "diffusional" and are exemplified by unit operations such as heat transfer, absorption, distillation, drying, and air-conditioning. It is convenient to further separate these diffusional processes into three broad classifications: (1) heat transfer without heat mass transfer; (2) mass transfer without heat transfer; and (3) combined heat and mass transfer. This thesis was confined to the first and third of these classifications wherein the interphase transfer of heat and mass occurred between a gas and a liquid in direct contact.

Objectives. - It was the purpose of this thesis to study the interphase transfer of heat between air and a non-volatile liquid (oil) and the simultaneous interphase transfer of heat and mass between air and a volatile liquid (water). Experimental data for the transfer of heat between a gas and a non-volatile liquid in direct contact are almost completely lacking in the literature. On the other hand, the vaporization of water into an air-stream has been the subject of numerous investigations but, due to experimental difficulties in obtaining adequate data for evaluating film coefficients, the theoretical equations dealing with the system have never been completely tested. In consequence, in the past, cooling-tower performance has been evaluated in terms of

two so-called over-all coefficients: Ua for heat transfer through both gas and liquid films, (despite the fact that the same quantity of heat does not pass through both films), and K'a for mass transfer through both films (although there is no mass-transfer resistance of water through water).

The objectives of this thesis were to evaluate the individual heat-transfer coefficients for the gas and liquid phases and the mass-transfer coefficient for the gas phase, and thus give the underlying theory a thorough test. Another objective was to determine whether or not the resistance of the liquid film was negligible compared with that of the gas film, as previously assumed in design of cooling towers.

Experimental. - Three experimental towers were designed and constructed, together with the necessary auxiliary equipment. An 8-inch packed tower was used in the air-oil study; a 4-inch packed tower and a 1-inch wetted-wall tower comprised the apparatus for the air-water investigation. The experimental work on the wetted-wall tower is still in progress, and the results of this work are not included in this thesis.

Procedure. - The theoretical equations pertaining to the air-water system involve true coefficients of heat transfer across the air and water films, and the true coefficient of mass transfer across the gas film. It is possible to measure directly the gas-film coefficient by employing the so-called "wet-bulb" technique, whereby water is fed to the adiabatically operated tower at the wet-bulb temperature of the inlet air and passes through the apparatus unchanged in temperature. Such an operation is termed an adiabatic humidifying run. Since the water temperature remains constant, its bulk temperature is also the temperature of the interface, and the gas-film driving force and coefficients can be determined. In view of the fact that the interchange of heat and mass is extremely rapid under these conditions, entering air temperatures from 600 to 700°F. were employed in order that a driving force of 5 to 20°F. could be obtained at the top of the tower packing, which consisted of 12 inches of 1-inch carbon Raschig rings. The apparatus was designed so that the heat-transfer coefficients for the gas film could be determined within ± 10%, and this precision required that heat balances over the tower close within 1% of the heat transferred across the gas-film.

RESULTS

With the use of resistance-wire heaters on the tower insulation, a negligible temperature driving force was maintained across the insulation and heat balances closed within the desired value.

By making a series of runs at constant air and water rates and with water temperatures of 135, 115 and 95°F., the gas-film temperature was varied between 110 and 210°F. From these runs the coefficient of heat transfer across the gas film was found to be proportional to $e^{0.0023t_f}$, where t_f is the air-film temperature in degrees Fahrenheit. In these runs the temperature driving force at the base of the tower between the hot air and the water ranged from 120°F. to more than 500°F., yet all data were correlated by the same general equation.

End-effects were evaluated by making adiabatic humidifying runs at constant air and water rates for packed heights of 12, 9 and 6 inches and extrapolating the data to infinite height. It was found that end-effects were independent of both air and water rates and were equivalent to 7.2 inches of one-inch rings. The coefficients of heat transfer across the gas film corrected for end-effects and reduced to a film temperature of 70°F. were found to be directly proportional to the product of the superficial mass velocity G of the air of 0.70 power and the 0.07 power of the superficial mass velocity L of the water over a gas-velocity range of 350-1000 lbs./(hr.)(sq.ft.) and a water-velocity range of 500-2500 lbs./(hr.)(sq.ft.).

Due to experimental difficulties, the humidity driving forces at the top of the packing, upon which the coefficient of mass transfer is based, could not be measured with the desired precision, and these coefficients are subject to a maximum error of ± 30%. However, the values of these coefficients were consistently smaller than would be predicted from the heat-transfer coefficients and the psychrometric ratio for the system, $ha_H/k'a_M s = 1$. This is thought to be due to the fact that the area a_H for heat transfer in a packed tower exceeds the area a_M for mass transfer, because of insufficiently wetted areas in the packing, causing $ha_{HT}/k'a_{MT}$ to exceed s. This explanation is substantiated by the fact that the mass-transfer coefficients $k'a_M$ more nearly approach the predicted values ha_H/s as the gas rate increases and water distribution improves. Furthermore, if the heat-transfer area exceeds that for mass transfer, the water temperature should exceed the adiabatic saturation temperature; this was found to be the case in all runs, the difference between the water and adiabatic saturation temperatures decreasing with increasing gas rate and consequently improvement in distribution of the water over the packing.

For water-cooling and dehumidification operations, it is possible to combine the basic equations in such a manner as to obtain a total heat or enthalpy potential as the driving force to allow for the transfer of sensible as well as latent heat. The use of this derived equation requires a knowledge of the gas-film coefficients ha and k'a as well as the coefficient of heat transfer across the water film. As mentioned earlier, it has been customary in the past to assume this latter coefficient infinite, or, in other words, to assume that the water film offers negligible resistance to the transfer of total heat. The coefficient of heat transfer for the liquid film varied as the 0.70 power of the gas rate and the 0.51 power of the liquid rate. From the results of water-cooling runs it was found that the water film offered 27 to 46% of the total resistance, depending upon the operating variables; the ratio of the resistances to total-heat transfer offered by the liquid film to that offered by the gas film is substantially independent of the air rate and is proportional to the 0.44 power of the water.

Mass-transfer coefficients, calculated from a series of oxygen-desorption runs made in the air-water tower, were found to check closely with those reported in the literature for a 20-inch tower containing the same packing.

The study of heat transfer between air and a non-volatile oil in an 8-inch tower includes both oil-cooling and oil-heating runs. It was necessary to electrically insulate this tower in much the same manner as was done for the air-water tower in order to obtain heat-balance closures within 1 to 2% of the heat given up by the hotter fluid. The packed heights equivalent to the end-effect were evaluated by varying the packed height and were found to be independent of the air and oil rates for the oil-heating runs, but were found to vary as the 2.1 power of the air rate for the oil-cooling runs.

The over-all coefficients of heat transfer for those runs in which heat was transferred from hot air to cold oil (oil-heating) varied as the 0.94 power of the air rate and the 0.25 power of the oil rate for gas rates of 250 to 800 lbs./(hr.)(sq.ft.) and oil rates of 500 to 2,000 lbs./(hr.)(sq.ft.). For the hot-oil (oil-cooling) runs the effect of the oil rate on the coefficients is approximately the same as for the hot-air runs, but the effect of air rate is much greater, the exponent varying between 2.2 and

2.8, depending on the oil rate. The coefficients of heat transfer are larger for the cold-oil runs than those for the hot-oil runs until high gas rates are reached where the coefficients for both are approximately equal. This discrepancy is believed due to the remarkable tendency for oil to channel at low gas rates and high oil temperatures. This fact was corroborated by visual observation of the oil leaving the packing. For the hot oil runs at low air rates wherein the oil at the base of the tower was at a high temperature and consequently had a low viscosity, the liquid was seen to leave the packing in the same number of separate streams and arranged in the same pattern as it was delivered to the packing by the multi-point distributor at the top of the tower. As the gas rate increased, the oil distribution improved since the channeling was no longer observed. This channeling effect was not observed for the cold-oil runs where the oil viscosity was higher. These observations were substantiated by the pressure drop data obtained for both types of runs. These clearly indicate that greater resistance to the air flow was offered by the wet packing in the oil-heating runs than was encountered in the oil-cooling runs at the same oil and air rates.

CONCLUSIONS

1. Adiabatic operation requires supply of external heat to the insulation, controlled to give a negligible radial gradient in temperature through the insulation. Heat balances closed to within 1 per cent for the adiabatic humid-

ifying runs, and to within 2 per cent for oil heating and cooling runs.

Adiabatic Humidifying Runs.

2. In adiabatic humidifying runs with water fed at the wet-bulb temperature of the entering air, no heat is transferred through the liquid film, and consequently the temperature of the liquid remains constant.

3. Wide variation in temperature driving force caused but minor changes in heat-transfer coefficient for the gas phase, to a degree identical with that previously established for absorption, and is due to variation in temperature t_f of the gas film:

$$ha \propto e^{0.0023 t_f}$$

where t_f is expressed in degrees Fahrenheit.

4. The heat-transfer coefficient for the gas phase varies as superficial mass velocity G of the gas to the 0.70 power, for all liquid rates employed, and varies as liquor velocity L to a small power increasing somewhat with increase in G, because of better distribution of liquid and consequent increase in wetted area a_H for heat transfer; it is adequate to employ a constant exponent of 0.07 on L.

5. End-effects, evaluated by varying packed depth, are independent of film temperature, gas velocity and liquid velocity. Visual observation through a window below the packing showed that the water mainly flowed transversely

to the periphery of the grid supporting the packing and thence down the walls of the base of the tower; since there was no so-called "spray section" at the base, and ~~a~~ multi-point distribution was used at the top to avoid the "coning" typical of a single central feed point, end-effects should have been substantially constant. The end-effects were found to be equivalent to an additional depth of 0.67 feet of packing.

7. The true coefficient for the gas phase with end effects eliminated, is

$$(ha)_{t_f} = 1.78\ e^{0.0023t_f}\ G^{0.7}\ L^{0.07}$$

The apparent coefficient (ha)', based on the actual depth z of packing, is given by

$$(ha)'_{t_f} = ha_{t_f}\ \left(\frac{z + 0.67}{z}\right)$$

8. In general the heat-transfer area a_H substantially exceeds that for mass transfer a_M, as shown by the fact that the temperature t_L of the water exceeds the temperature t_{AS} of adiabatic saturation. As gas velocity increases and water distribution improves, the wetted area more closely approaches the heat-transfer area, and in consequence $t_L - t_{AS}$ decreases.

9. Mass-transfer coefficients k'a for the gas phase, while inherently less precise than the values of ha, are affected by the same variables. If values of k'a are based on

measured temperatures of the water, they are less than those predicted by the relation

$$k'a_M = ha_H/s$$

but as G increases, ~~and~~ water is better distributed and the measured values of $k'a_M$ approach more closely to ha_H/s.

10. In design work, measured water temperatures are not available and in consequence the driving force for mass transfer would be based on the temperature of adiabatic saturation. On this basis the revised mass-transfer coefficients $(k'a)''$ agree closely with those predicted by

$$(k'a)'' = ha/s$$

which is recommended for design work.

11. Mass-transfer coefficients K_La for desorption of oxygen from water in the 4-inch tower closely checked those reported in the literature for a 20-inch tower, indicating that wall effect was unimportant.

Water-Cooling Runs

12. The coefficient h_La for the liquid film is given by the equation

$$h_La = 0.82\, G^{0.70} L^{0.51}$$

13. From this and the relation given for the gas film, the tie-line slope $(h_La/k'a)$ on the enthalpy diagram is independent of G and varies as the 0.44 power of L.

14. The slope of the tie-line is the ratio of the resistance of the gas-phase for enthalpy transfer to that of the liquid film for the same heat quantity. From the data it was shown that this resistance ratio ranged from 1.18 to 2.70 indicating that the liquid film offered 46 to 27 per cent of the combined resistance of both liquid and gas films. This is in sharp contrast with the usual assumption that the liquid-film resistance is negligible compared with the total resistance.

15. Now that film coefficients of heat transfer are available for the gas and liquid phases, and mass-transfer coefficients are available for the gas phase, it is no longer necessary to cling to the previously used so-called over-all coefficients of heat and mass transfer, which are fundamentally unsound.

Oil-Heating (Cold Oil) Runs

16. It was found that good distribution of oil was obtained, and consequently end-effects were independent of both L and G', as in the adiabatic humidifying runs with water and air, and the over-all coefficient Ua from air to oil increased more with increase in G' than with increase in L:

$$Ua = 0.083(G')^{0.94}L^{0.25}$$

Oil-Cooling (Hot Oil) Runs

17. The hot oil channeled badly and came out of the grid (supporting the packing) in the same number of streams as

fed at the top of the packing by the multi-point distributor. Because of the much poorer distribution of liquid than in the runs with cold oil, the coefficients were in general lower than with cold oil. As gas rate increased and distribution improved, both the wetted area and the over-all coefficient U increased, and consequently the product term Ua increased very rapidly, giving exponents on G' ranging from 2.2 to 2.8. End effects varied as $G'^{2.1}$. The combined effect of G' and L is given in graphical form, Figures 31 and 32 of this thesis.

II. INTRODUCTION

Many operations, for which the chemical engineer is responsible, involve the interphase transfer of heat, mass, or of both heat and mass. These processes are generally classified as "diffusional" and are exemplified by unit operations such as heat transmission, absorption, distillation, drying, and air conditioning. Since a sound approach to the solution of problems arising in these operations requires a firm underlying theory upon which experience and practice can be built, it is not surprising that considerable energy has been expended in the development of diffusional theory.

A great deal is known about "molecular" diffusion, which is the term applied to the mass transfer in gases and liquids when not agitated. Unfortunately, however, most engineering applications of diffusional processes employ conditions wherein the movement of mass and/or heat occurs in and between fluids which are in turbulent motion, and the molecular diffusion equations no longer control. The transfer of heat and/or mass by means of the mixing action due to the turbulence of the flowing fluid is called "eddy" diffusion and is considerably more rapid than molecular diffusion. However, since the mechanics of turbulent motion have not as yet been completely developed, a true understanding of the theory of interphase diffusion is not possible. As a result, most engineering research in the diffusional field has been along the line of determining a transfer coefficient which when multiplied by a suitable driving force will give the rate of interphase transfer per unit area.

This procedure has developed from the two-film theory as first introduced by Whitman (51) in 1923 to explain rate phenomena in heat and/or mass-transfer operations. This concept involves the resistance to diffusion of two fluid films, one on each side of the interface, and subsequent development of theory has centered around the two-film mechanism of interphase diffusion.

The gas phase is considered to consist of a true gas film flowing in streamline motion next to the interface and an eddy diffusion zone. The force tending to cause the movement of material against the resistance of a gas film is taken as the partial pressure gradient of the diffusing material across the true gas film. Similarly a concentration gradient causes the movement of mass through a true liquid film, and a temperature difference results in the transfer of heat. These true films are considered to be stagnant or in streamline motion so that the diffusion is truly molecular. However, the diffusion through the bulk of the fluid adjacent to each true film, which is in turbulent motion, occurs by eddy diffusion; and in the absence of adequate knowledge on the mechanism of turbulent motion it is customary to evaluate coefficients of diffusion over an entire phase or, indeed, more often from one phase to another. This has been necessary since experimental techniques have not as yet become sufficiently refined to permit measurements at the film boundaries.

Many of the diffusional operations used in chemical engineering are confined to the transfer of heat and/or mass between a liquid and a gas, and the following discussion, as well as the contents of this thesis, will be devoted exclusively to the liquid-gas systems as are normally encountered in chemical engineering work. In addition the discussion will also be concerned chiefly with the specific types of equipment used with this system, since the major percentage of research has been performed on apparatus under conditions which closely approximate those used in industrial equipment.

In order to examine the field of heat and mass transfer between liquids and gases more closely, it seems advisable to classify the general field into three sections:

(1) Heat transfer without mass transfer
(2) Mass transfer without heat transfer
(3) Combined heat and mass transfer.

Heat Transfer Without Mass Transfer

Typical examples of this first classification include the transfer of heat between a gas and a liquid in an ordinary shell-and-tube heat exchanger, or the transfer of heat between a gas and a non-volatile liquid in direct contact, such as occurs in a spray tower or a packed tower.

The last twenty or thirty years has seen great strides in the advancement of the science of heat transmission. A great mass of data on heat transfer have been organized, reduced to a common basis, and correlated in terms of physical properties and operational variables. Correlations based on dimensional analysis are available today (32) making it possible to take almost any combination of fluids, and by adding the film resistances of the two fluids to that of the tube wall, to obtain an over-all coefficient of heat transfer for a combination which has never been studied experimentally. The resulting coefficient can be used with over-all driving forces between the bulk temperatures of gas and liquid to predict with remarkable accuracy the operation to be expected with the chosen system. It has also been in the field of heat transmission that the greatest measure of success has been experienced in obtaining individual coefficients of heat transfer across gas films by means of a theoretical approach. Analogies between momentum and heat transfer as proposed by Reynolds (37), Prandtl (36), von Karman (48), and Martinelli (30), have resulted in equations which agree quite well with experimental data for gas films. However, the empirical correlations are still preferred today due to their ready availability and the ease with which problems can be handled mathematically by use of these correlations.

The rate of heat diffusion is directly proportional to the area available for heat transfer, so that it seems rather strange that so little work has been done toward the utilization of packed towers for heat transfer equipment. A tower packed with Raschig rings, Berl saddles, crushed rock, coke, or any of a large variety of materials, offers at low cost a very large contact area per unit volume. This type of apparatus also has several other distinct advantages. It offers a special advantage where corrosive materials are being handled since both the tower and the packing may be fabricated from corrosion resistant materials which are impracticable to use in a shell-and-tube exchanger because of low thermal conductivity. Furthermore, such a system offers a potentially better over-all coefficient of heat transfer since the thermal resistance of the tube wall and any scale deposits are eliminated. The limitations of direct-contact equipment for heat transfer are obvious. The method would be undesirable if one stream is easily contaminated by the other, or if the liquid is appreciably volatile.

In spite of these limitations such an apparatus could be advantageously used in many instances, the most common of which would be the heating or cooling of oils with high boiling points by use of flue gases, air, or other gases in direct contact with the oil. However, the design of such a system can not be accomplished by means of the general correlations of coefficients of heat transfer for gas and oil films, since the area of contact is unknown as is the actual velocity of the gas and liquid streams past the packing surfaces. As a result it seems that actual experimental data on the system will be necessary.

The quantitative treatment of such a system is quite simple, differing from the general treatment of a heat exchanger only in that the coefficient of heat transfer must be expressed in terms of a unit volume of the packing rather than the conventional unit area. Obviously such a coefficient will be a function of the size and shape of the packing. Consider an adiabatically operating packed tower with a non-volatile oil of specific heat c_L being fed into the top at a rate of L pounds per hour per square foot of gross cross section and a gas of specific heat c_G passing up through the tower at a rate of G' pounds per hour per square foot of gross cross section and countercurrently to the oil stream. In a differential height dz of the tower having gross cross-section S, the oil is at a bulk temperature T and the air at a bulk temperature of t. A heat balance between this section and the top of the tower gives

$$SL\, c_L(T_2 - T) = SG' c_G(t_2 - t) \qquad (1)$$

Subscript 2 refers to the top of the tower.
The rate of heat transfer within the section is

$$dq = +SL\, c_L\, dT = UaS(T - t)dz \qquad (2)$$

where U is the over-all coefficient of heat transfer per unit surface, and a is the interfacial area per unit of volume of packing. The differential of (1), substituted in (2), gives

$$\frac{dq}{S} = + L\, c_L\, dT = +G' c_G\, dt = Ua \quad (T - t)dz \tag{3}$$

Integrating (3) and solving for z,

$$z = \frac{L\, c_L}{Ua} \int \frac{dT}{T - t} = \frac{G' c_G}{Ua} \int \frac{dt}{T - t} \tag{4}$$

The integral term has been designated by Chilton and Colburn (9) as the number of transfer units, N_t and $\frac{z}{N_t}$ is termed the height of a transfer unit, H_t. From Eq. (3) it is seen that both T and t are linear in q, and therefore their difference is also linear in q. For these conditions Eq. (4) reduces to

$$z = \frac{Lc_L}{Ua} \frac{(T_2 - T_1)}{(\Delta t)_m} = \frac{G' c_G}{Ua} \frac{(t_2 - t_1)}{(\Delta t)_m}, \tag{5}$$

where $(\Delta t)_m$ equals

$$\frac{(T - t)_2 - (T - t)_1}{\ln \frac{(T-t)_2}{(T - t)_1}} \tag{6}$$

Solving for the volumetric over-all coefficient Ua:

$$Ua = \frac{Lc_L\, (T_2 - T_1)}{z\, (\Delta t)_m} = \frac{G' c_G (t_2 - t_1)}{z\, (\Delta t)_m} \tag{7}$$

All the quantities in the middle or last terms can be measured experimentally thus allowing the over-all coefficient to be computed. It is to be noted that the above analysis is based on adiabatic operation.

An unpublished thesis by G. A. Bennett at the Case School of Applied Science (4) provides the only known experimental treatment of the problem. Bennett passed hot oil countercurrently to air in a vertical eight-inch diameter tower, packed to a height of 10.25 feet with 1/2-inch ceramic Raschig rings. The oil was fed in a single central stream at the top. The tower was not insulated, and a considerable fraction of the heat removed from the oil was lost to the surroundings. In one case the oil lost 5600 Btu per hour and the air retained 2680 Btu per hour. Liquid rates ranged from 756 to 2010 pounds per hour per square foot of gross cross section, and gas rates ranged from 99 to 352 pounds per hour per square foot of gross cross section. Based on the heat absorbed and retained by the air, and on the logarithmic mean of the terminal differences in temperature from oil to air, the over-all coefficient of heat transfer, Ua, expressed in Btu per hour per cubic foot of tower packing per degree Fahrenheit is given by

$$Ua = 0.0011\, G'^{1.51}\, L^{0.159} \tag{8}$$

Now it must be pointed out that this over-all coefficient contains three quantities, the combined resistances of both air and oil films as well as the factor a. The effect of variables such as gas and liquor rates, viscosity, surface tension etc. on "a" are not known, so that it is impossible to quantitatively evaluate this correlation. However it is known from general correlations of gas and liquor-film coefficients that the gas film offers considerably more resistance to the flow of heat than does the liquid film. It is also a well established fact that the coefficient of heat transfer for gas films varies as the 0.6 to 0.8 power of the gas rate. In the light of these facts the high exponent (1.51) on the gas rate, G', is rather difficult to justify. H. C. Carlson (5) has reviewed Bennett's work and by basing the calculation of the over-all coefficient on the heat lost by the oil rather than the heat taken up by the air, Ua is substantially increased to U'a

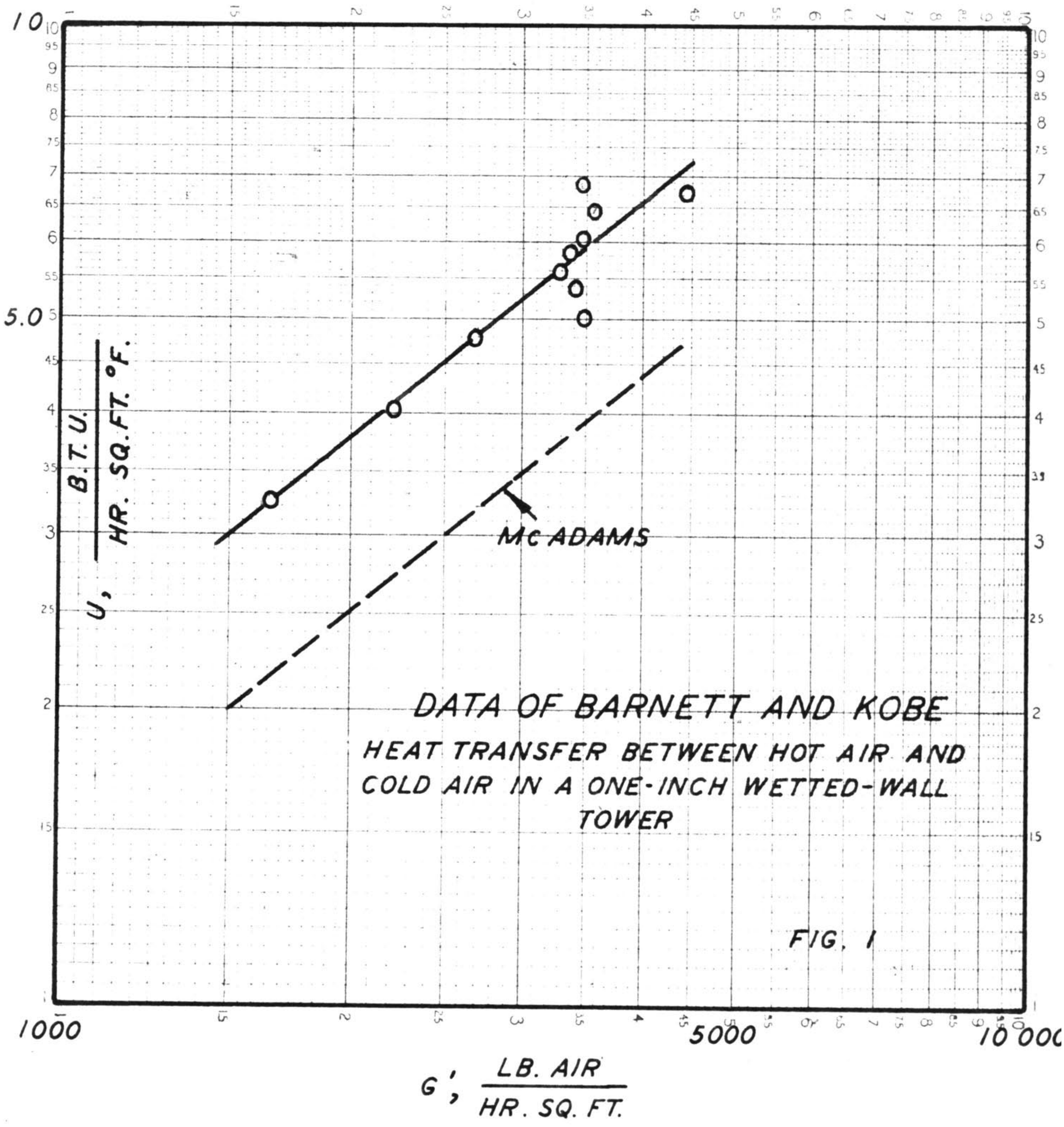
$$U'a = 1.15\, G'^{0.54}\, L^{0.067} \qquad (9)$$

Now the effect of gas rate is more normal. However, the entire picture leaves much to be desired as the large heat losses greatly lessen the value of the work.

The problem is somewhat clarified by Barnet and Kobe (3) who passed hot air countercurrently to oil in a vertical wetted-wall tower made from a four-foot length of standard one-inch seamless steel pipe. The oil was introduced into a cup-like reservoir at the top and flowed down the inner walls of the tower. The air was passed upward through a five-foot calming section before entering the tower in order that a normal turbulent flow pattern would be achieved before the gas came in contact with the liquid. The oil was removed through a slot in the bottom of the tower wall to prevent it from flowing into the air-calming section. Special precautions were taken to prevent the conduction of heat from the calming section into the tower which was well insulated. Oil flow rates varied from 64 to 212 pounds per hour per foot of inner circumference. Gas rates varied from

1650 to 7090 pounds per hour per square foot of gross cross section. Temperature variations in both entering streams were small. The over-all coefficients have been recalculated, taking as a basis the heat lost by the air and plotted versus the gas rate in Figure 1. Only the data from those runs with heat balances within 5% of the heat given up by the oil have been used. Since the apparatus was such that the interfacial area could be readily measured, the "a" of Eq. (7) can be computed, and the coefficient represents only the liquor and gas-film resistances. Wetted-wall tower data are of questionable value for packed tower work, but such equipment does indicate the extent to which gas and liquid velocities may be expected to affect the coefficient in a packed tower.

Although the data shown in Figure 1 scatter somewhat, a single straight line with a slope of 0.8 fits them reasonably well. It is also to be noted that liquor rate has apparently little or no effect on the coefficients since no definite trends can be determined by varying the liquor rate and holding the gas rate constant. This fact leads one to believe that the liquid film may offer but negligible resistance compared that of the gas film. Indeed the data indicate that the over-all resistance $1/U$ is even less than that expected for a single gas-film resistance $1/h$ as shown by the dashed line representing McAdams (32) general correlation for coefficients of heat transfer across single gas films on dry surfaces. The fact that the measured coefficients are approximately 50% higher than would be predicted if the oil



DATA OF BARNETT AND KOBE

HEAT TRANSFER BETWEEN HOT AIR AND COLD AIR IN A ONE-INCH WETTED-WALL TOWER

FIG. 1

film offered zero resistance is somewhat difficult to justify. One explanation that comes to mind is that the coefficient should be a function of the gas velocity relative to the liquid film whereas those plotted are velocities relative to the wall. However the gas rate is increased only about 10% when so calculated, causing but a small shifting of the data to the right. Gilliland (18) in his work on the vaporization of liquids in a wetted-wall tower, similar to the one later used by Barnet and Kobe, found that his data correlated better with actual rather than relative gas velocities. Gilliland advanced the explanation that the liquid film drags part of the adjacent gas film down to the bottom of the tower where it is mixed with incoming gas. This results in a decrease in the actual driving force at the bottom of the tower with the result that the calculated coefficient is too low. This tends to offset the increase in the coefficient due to the higher relative velocity. This explanation will not satisfactorily explain the discrepancy of Figure 1, since the coefficient seems already too high, even when the effects of relative gas velocity have been taken into account.

As a result of this survey of the available literature pertaining to the transfer of heat between a gas and a non-volatile liquid in direct contact, one of the aims of this theses is to measure such over-all coefficients of heat transfer and to correlate them with liquor and gas flow-rates.

Combined Heat and Mass Transfer

The large majority of direct-contact operations fall under this classification, and due to the fact that diffusion of both heat and matter are occurring simultaneously, the solution of problems of this nature become more complicated. The absorption of gases in liquids would involve mass transfer alone were it not for the fact that very often the absorption itself is exothermic. Even in those cases where the heat effect due to absorption is small, heat effects are encountered when the liquid stream vaporizes into the gas stream with the consequent cooling of the liquid film. Although considerable work has been done on the combined diffusion of heat and mass in systems involving an operation other than vaporization or condensation, these latter processes seem to present the more logical point of attack.

The rate of transfer of heat and mass through a gas or liquid phase is directly proportional to the film coefficient and to a driving force between the bulk of the stream and the interface causing the diffusion to occur. The driving force for the transfer of heat is, of course, a temperature difference, and that for mass transfer a difference in concentration or partial pressure. In dealing with vaporization and condensation it is generally convenient to use a humidity potential for mass transfer which is related to the partial pressure driving force (See APPENDIX).

The basic equations, introduced in the following discussion, were first proposed by Walker, Lewis, and McAdams (49) and by Robinson (38) in 1923. The equations are general in that they apply to any combination of gas and single pure liquid, as well as to either a packed or a wetted-wall tower.

Consider the flow of gas at a temperature, t_1, and a humidity, H_1, into the bottom of an adiabatic tower of infinite height. A volatile liquid is being introduced into the top of the tower at a rate exactly equal to the rate at which it is vaporizing into the gas stream. The volatile liquid is entering at a temperature, t_s, which is the temperature of the exit gas stream. An over-all enthalphy balance gives

$$i_{G1} + (H_s - H_1) \; i_{Ls} = i_{Gs} \tag{10}$$

where

i_{G1} = the enthalpy of the gas entering

i_{Ls} = the enthalpy of the liquid entering

i_{Gs} = the enthalpy of the gas leaving

and the subscript "s" refers to conditions at the top of the tower. If the assumption is made that the gas and the vapor from the liquid follow the perfect gas laws (See APPENDIX) it can be shown that

$$s_1 \, (t_1 - t_s) = (H_s - H_1)(r_s) \tag{10a}$$

where the humid heat s_1 is equal to $(c_G + c_V \; H_1)$; c_G is the specific heat of the gas, c_V is the specific heat of the vapor from the liquid, and r_s is the latent heat of vaporization at t_s. Since for a given total pressure H_s and r_s are fixed by t_s, it is seen that H_1 is a unique function of

t_1. A plot of this equation for various values of t_s with H and t as ordinate and abscissa, (the familiar humidity chart) gives a series of lines slightly concave upward which are known as adiabatic saturation lines. It can be seen from Eq. (10) that these lines cannot be lines of constant gas enthalpy (6) because of the middle term of Eq. 10 which represents the enthalpy of the liquid entering the tower. A tower of finite height would also operate along one of these adiabatic saturation lines; in this case, however, the gas leaving the top of the tower would be at some temperature, t_2, greater than t_s and some humidity H_2, lower than H_s. The equation would also hold for the case wherein the entering liquid rate exceeds the rate of vaporization, providing the exit liquid temperature was at t_s. It will be shown in a later section where this latter operation has special significance.

When a differential area of liquid surface is exposed to an unsaturated gas stream, the liquid surface will attain a dynamic equilibrium temperature known as the wet-bulb temperature t_w. At this temperature, the heat transferred from the surroundings will exactly supply the heat of vaporization r_w, of the liquid evaporating at t_w. A heat balance on the differential area gives

$$k' dA_w \, r_w \, (H_w - H) = h \, dA_H \, (t - t_w) \tag{11}$$

where k' is the coefficient of mass transfer across the gas film expressed in humidity units, h is the coefficient of heat transfer also across the gas film, dA_W is the differential area available for mass transfer, and dA_H is that for heat transfer. By making the logical assumption that $dA_W = dA_H$

$$\frac{r_W\ (H_W - H)}{t - t_W} = \frac{h}{k'} \qquad (11a)$$

This ratio of the coefficient of heat to mass transfer across the gas film has been determined experimentally by a number of observers (27), (29) for various liquid-gas systems by blowing gases at temperature t and humidity H past a thermometer bulb covered with a wick to which is supplied the liquid at the temperature t_W indicated by the thermometer. Knowing t_W, both H_W and r_W can be determined, and the ratio of coefficients is evaluated by means of Eq. (11a). It is often assumed that the ratio of h/k' is the same for the liquid surfaces in a tower fed with liquid at t_s as for a wet-bulb thermometer. This assumption is reasonable, but it has never been checked by experimental data.

The quantitative treatment of the general case of vaporization (or condensation) of a liquid into a gas stream when the two streams are flowing countercurrently in an adiabatic tower, yields five equations resulting from a material balance, a heat balance, and three rate equations. Of these rate equations, two refer to the transfer of mass and of heat across the gas film; the third refers to the transfer of heat across the liquid film. There can not, of course, be an equation for the rate of mass transfer across the liquid film since there can be no concentration gradient in a pure liquid. The water balance around the bottom of the tower gives

$$(L - L_1) = G(H - H_1) \tag{12}$$

where G is the pounds of bone dry air per hour per square foot of gross cross-section.

Over a section of the tower of height dz

$$dL = G\ dH \tag{12a}$$

An enthalpy balance around the bottom of the tower yields

$$G(i_{G1} - i_G) = L_1\ i_{L1} - L\ i_L \tag{13}$$

The enthalpy of the gas above a base temperature t_o is given by $i_G = c_G(t - t_o) + H\left[r_o + c_V\ (t - t_o)\right]$. Since the term $(c_G + c_VH)$ is the humid heat, s, Eq. (13) becomes

$$G(s_1(t_1 - t_o) + H_1\ r_o - s\ (t - t_o) - Hr_o) = L_1c_L(T_1 - t_o) - L\ c_L\ (T - t_o) \tag{13a}$$

Over the differential section dz

$$-Gsdt - G(t - t_o)ds - Gr_odH = -Lc_LdT - c_L(T - t_o)dL \tag{13b}$$

It is shown in the APPENDIX that the last term can usually be neglected since the liquor rate does not vary significantly. In addition the humid heat is essentially constant so that this expression simplifies to

$$- Gsdt - Gr_odH = - Lc_LdT \tag{13c}$$

The rate of transfer of sensible heat between the main body of the gas at t and the liquid-gas interface at t_i is

$$- Gsdt = ha\ dz\ (t - t_i) \tag{14}$$

The rate of diffusion of water vapor between the main body of the gas with humidity H and the interface with humidity H_i is

$$- GdH = k'a\ dz\ (H - H_i) \qquad (15)$$

The rate of heat transfer from the interface at t_i to the main body of the water at T is

$$- Lc_L\ dT = h_La\ dz\ (t_i - T) \qquad (16)$$

These five differential equations offer the general solution to the problem of direct contact between a pure volatile liquid and a gas in a steadily operated, adiabatic apparatus. Unfortunately a general solution has never been obtained. A knowledge of the necessary coefficients, combined with these five equations, is sufficient to solve such problems.

It is indeed fortunate that an important system, water and air, has physical characteristics such that the coefficients can be evaluated for this system, and thus problems dealing with this system can be handled quite rigorously.

W. K. Lewis (26) first derived the relationship between the coefficients of heat and mass transfer:

$$\frac{h}{k'} = s \qquad (17)$$

Although the derivation was later shown by Lewis to have been in error,(27) it is true that the ratio of coefficients has a constant numerical value equal approximately to the humid heat for a wet-bulb thermometer placed in an air stream and which is kept wet with water. After substituting Eq. (17) into Eq. (11a), it is seen that the wet-bulb equation:

$$\frac{H_W - H}{t - t_W} = \frac{s}{r_W} \qquad (17a)$$

and the equation for adiabatic saturation:

$$\frac{H_s - H}{t - t_s} = \frac{s_1}{r_s} \tag{17b}$$

are similar in form. Since in deriving the wet-bulb equation, equilibrium at the interface was assumed, H_w equals H_s, t_w equals t_s, and r_w equals r_s; neglecting any differences between s_1 and s, the two equations are identical. This means that for a given mixture of air and water, the wet-bulb temperature and the adiabatic saturation temperature are equal. Therefore it is possible to operate an adiabatic tower in such a fashion that, if the water fed to the tower is at the wet-bulb temperature of the entering air, it is also at its adiabatic saturation temperature, and the water would pass through the tower unchanged in temperature, provided interfacial areas for mass and heat transfer are equal.

Such an operation would afford an opportunity to obtain interfacial temperatures (and therefore interfacial humidities); since the water passes through the tower without temperature change there can be no temperature drop across the liquid film, and, consequently, the bulk temperature of the water is the same as that at the interface. Now with the interfacial temperature (and humidity) known and constant, it is possible to integrate Eqs. (14) and (15) directly and obtain both film coefficients, ha and ka. With the knowledge of these coefficients, and by making water-cooling runs, it is then possible to obtain the coefficient of heat transfer across the liquid film defined by Eq. (16).

The practical combination of the basic equation was first worked out by Merkel (31) who obtained an enthalpy or total heat potential as the driving force to allow for both sensible and latent heat transfer. Multiplying Eq. (15) by r_o, adding to Eq. (14), and assuming $h/k' = s$, the following relation is obtained:

$$-G(r_o dH + s dt) = k'a\, dz \quad (r_o H + st) - (r_o H_i + st_i) \qquad (18)$$

Since $r_o dH$ plus sdt equals di_G, Eq. (18) becomes

$$-G di_G = k'a dz\, (i_G - i_i) \qquad (18a)$$

where differences in s are overlooked.

Eq. (18a) can be rewritten

$$z = -\frac{G}{k'a} \int \frac{di_G}{i_G - i_i} \qquad (18b)$$

Since the enthalpy lost by the water must equal that gained by the air

$$c_L\, L dT = G di_G \qquad (19)$$

which in combination with Eqs. (16) and (18a) gives

$$\frac{i_i - i_G}{t_i - T} = -\frac{h_L a}{k'a} \qquad (20)$$

Dividing both sides of Eq. (20) by c_L, the heat capacity of water, (numerically equal to unity)

$$\frac{i_i - i_G}{i_i - i_L} = \frac{-h_L a}{k'a\, c_L} \qquad (20a)$$

It is seen that the ratio of coefficients is in reality the ratio of the resistances to total heat transfer of the gas film to the liquid film.

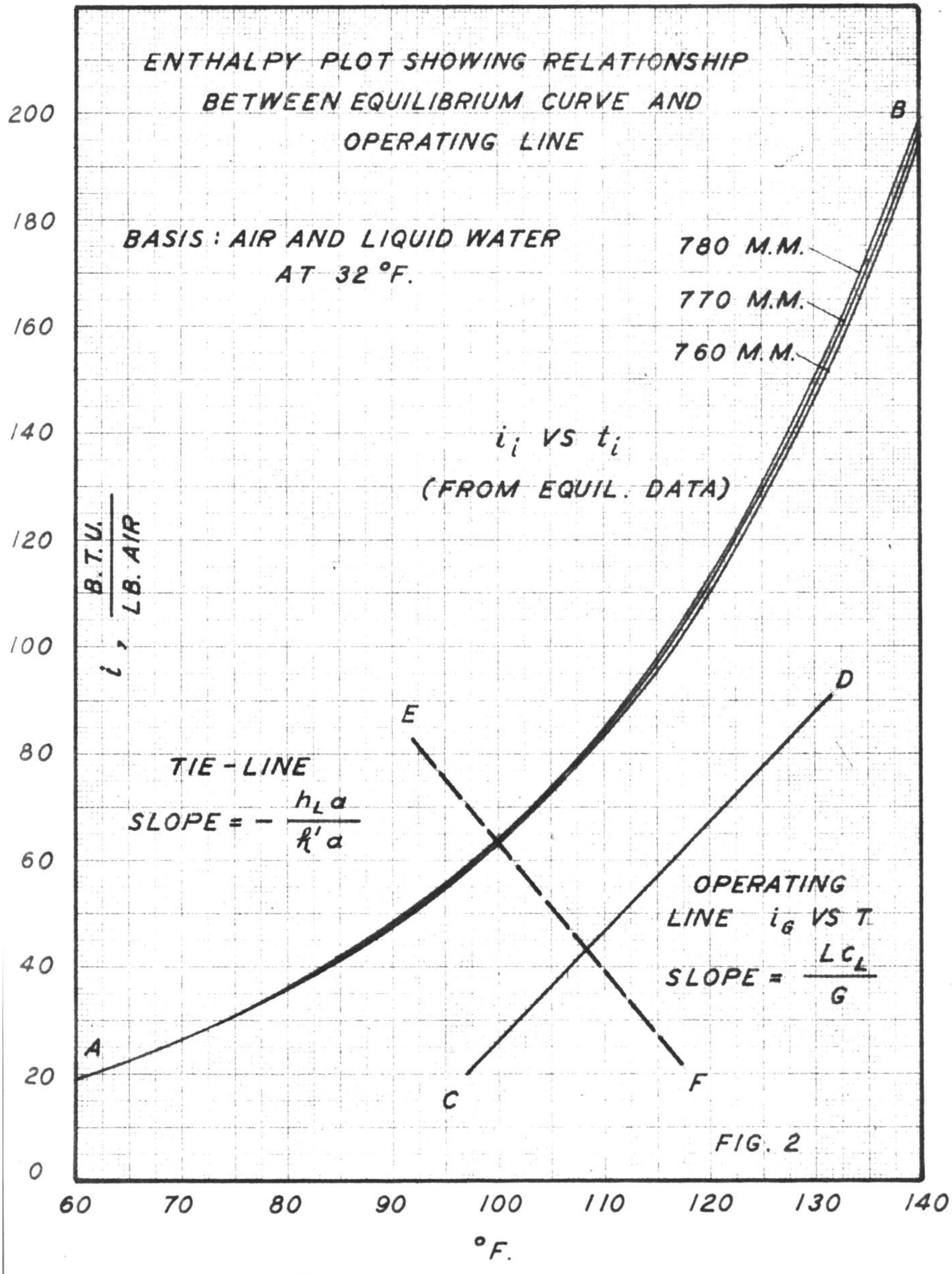

ENTHALPY PLOT SHOWING RELATIONSHIP BETWEEN EQUILIBRIUM CURVE AND OPERATING LINE

FIG. 2

The use of these equations can be best illustrated graphically on a plot of enthalpy versus temperature. The curve AB represents a plot of i_i versus t_i and is calculated from the equilibrium relations for water vapor and air at constant total pressure. The line CD represents the operating line as based on the heat balance Eq. (19) and is a plot of i_G versus T. The tie-line EF corresponds to Eq. (20). This tie-line relates the operating line and the equilibrium curve so that $(i_G - i_i)$ can be determined at any point in the tower, and, thus Eq. (18b) can be integrated graphically. The equations and Figure 2 furnish a method for the experimental determination of h_La. From an adiabatic humidifying run, wherein the water is fed at the wet-bulb temperature of the entering air, the coefficients of heat and mass transfer across the gas film can be obtained as outlined above. Then by operating the tower either as a dehumidifier (by introducing the water at a temperature below the wet-bulb temperature of the air) or as a water cooler (by introducing the water at a temperature above the wet-bulb temperature of the entering air), and employing the same values of L and G as before, the value of the integral in Eq. (18a) can be experimentally determined. By trial and error the slope of the tie-line $\frac{i_i - i_G}{t_i - T} \doteq - \frac{h_La}{k'a}$ is adjusted until the value of the integral for the cooling-tower run equals that for the adiabatic humidifying run. Since k'a is known, h_La can be then determined from this slope. Unfortunately, this analysis is applicable only to a liquid-gas system for which $h/k' = s$ (See APPENDIX).

Because the air-water system is of special importance due to the air conditioning field and because of the widespread use of water-cooling towers in industry, it is not surprising that many investigators have experimentally studied the system in wetted-wall, packed, and spray towers. However, it might be added that the procedure outlined has never been applied rigorously to obtain correlations for the film coefficients involved. This is mainly due to the experimental difficulties involved in the adiabatic humidifying runs.

In packed towers the large interfacial area, combined with the fact that heated air exchanges heat for mass very rapidly, causes the gas to be very near its saturation temperature as it leaves the tower. As a result the driving force at the top of the tower becomes too small to measure with the desired precision. One obvious solution is to decrease the height of packing, and this has often been done, but this magnifies the end-effects in the column. Another solution would be to preheat the air to a high temperature. However, this latter solution makes adiabatic tower operation more difficult. Since the theoretical equations are based on adiabatic operation, it is essential that the heat losses be kept low.

As a result of these difficulties it has not been uncommon for workers in the field to assume that the liquid film offers negligible resistance to heat transfer. It is possible to operate the apparatus as a water-cooling tower and thus obtain large driving forces at the top. Coefficients of mass transfer are calculated as a so-called over-all coefficient, K'a, with the use of Eq. (18b) and Figure 2 wherein the tie-lines are assumed vertical: $-\frac{h_La}{k'a} = \infty$, and t_i equals T. Likewise the coefficient of heat transfer is determined as an over-all coefficient, Ua, based upon the temperature difference between the bulk of the gas and the bulk of the liquid. However, such a procedure is open to question for several reasons. First, the concept of an over-all coefficient of heat transfer is misleading when dealing with vaporization or condensation processes since the sensible heat which passes through the gas film is never equal to the sensible heat passing through the liquid film. If, for example, a tower is functioning as a dehumidifier, the quantity of heat which is transferred from the interface to the bulk of the water is supplied by the sensible heat transferred across the gas film plus the latent heat which also crosses the gas film but as water vapor. Secondly, there is no resistance to mass transfer of water through the liquid film of water, hence an over-all mass transfer coefficient has no physical significance; this is in contrast to absorption, where resistances to the transfer of solute are offered by both gas and liquid films, and consequently the over-all mass transfer resistance is significant. Thirdly, there is no reason why $^{Ua}/K'a$ should equal s, so that the use of the enthalpy potential is hardly justified since its derivation is based on that assumption.

Whitman and Keats (50) made adiabatic humidifying runs in a coke-packed tower using the air-water system. This was the initial effort in the field and served to introduce the film concept of heat and mass transfer. Using a logarithmic mean temperature difference as the driving force, which was correct since the water temperature was presumably constant, they found a correlation between the coefficient of heat transfer across the gas film and the air rate. For dehumidifying runs the authors point out that the liquid temperature can not be used as a measure of heat transfer since the heat transfer across the liquid film is composed of both the heat of condensation and the sensible heat removed from the gas.

Johnstone and Singh (25) made numerous runs in a tower using various packings and the air-water system. Using one-inch Raschig rings, six inches deep, at a constant water rate of 1080 pounds per hour per square foot of gross cross section, they found that the H_t, height of a transfer unit, was approximately 0.4 foot for superficial gas velocities ranging from 625 to 2410 pounds per hour per square foot of gross cross section. They used no preheat on the inlet air and as a result the temperature driving force at the top of the tower was approximately 2°F. As a result their calculated values for these runs cannot be very precise.

Water temperatures were not reported, but presumably water was fed within 1°F. of the wet-bulb temperature of the exit air.

Sherwood and Holloway (41),(23) reported data on the vaporization of water in air in a 20-inch diameter tower packed with eight inches of 1.5-inch Raschig rings. The entering air was essentially at room temperature so that the packed height had to be short in order that a measurable driving force existed at the top of the tower. There was substantial vaporation in the spray section below the packing so that it was known that their coefficients were high.

London, Mason, and Boelter (28) report data on a forced-draft water-cooling tower packed with ovate slats. They used logarithmic-mean driving forces (based on terminal states) for the calculation of over-all heat and mass transfer coefficients. They acknowledge the fact that this procedure assumes zero resistance in the liquid film, and is, therefore, subject to criticism. They found that the so-called over-all coefficient of mass transfer varied with the 0.48 power of the gas rate.

M. Parekh (33) made a number of water-cooling runs in a 20-inch tower using 0.5, 1., and 1.5-inch Raschig rings and Berl saddles. These coefficients were calculated from over-all driving forces, and varied as the 0.43 to 0.64 power of the gas rate.

Gardner and Newton (20) attempted to follow the procedure indicated by the theoretical equations. Working with the 20-inch tower used by Parekh and packed with 20 inches of one-inch Raschig rings, they first made adiabatic humidifying runs and determined the coefficients of heat and mass transfer for the gas film. Using these data with data obtained from water-cooling runs in the same tower, they determined the only values of h_La found reported in the literature. The entering air was preheated to about 120°F., but they found this was not sufficient. Their results are few and somewhat questionable due to lack of sufficient preheat. At a constant air rate of 775 pounds per hour per square foot of gross cross section, the values of the liquid-film coefficient of heat transfer were found to be 2000, 2100, and 2700 Btu per hour per cubic foot per degree F. at water rates of 500, 1000, and 1500 pounds per hour per square foot respectively. The coefficient of heat transfer across the gas film for the same conditions was approximately 400 Btu per hour per cubic foot per degree F., corresponding to 0.5 feet of height per transfer unit.

Further consideration of these data on the air-water system will be deferred to a later section of this thesis.

From the previous discussion it is apparent that data concerning the rate coefficients for the air-water system are inadequate for use in solving problems pertaining to this system with any degree of confidence. It is desirable to know whether the equation $\frac{ha}{k'a} = s$ holds true for the air-water system in towers. The coefficient of heat transfer across the liquid film should be evaluated to determine whether or not the resistance of the liquid film is negligible.

Consequently it is one of the objects of this thesis to evaluate the several coefficients involved in the air-water system by following the pattern suggested by the theoretical equations themselves i.e., to determine coefficients of heat and mass transfer for air films by making adiabatic humidifying runs using highly preheated air in an electrically insulated packed tower fed with water at the wet-bulb temperature of the entering air, and using these data in conjunction with data from water-cooling runs, to determine coefficients of heat transfer across the water film.

It was pointed out at the beginning of the derivation of the equations presented in this section that the analysis was general in that it applied to any adiabatic contacting device which utilized the air-water system. An apparatus which often offers special advantages for a theoretical study is the wetted-wall tower because the interfacial area is subject to direct measurement. Furthermore the height of a transfer unit is large compared to a packed tower with the result that large driving forces are easily obtained at the exit-air end of the wetted-wall. Unfortunately it is not possible to use wetted-wall data in predicting packed tower performance, but nevertheless such data indicate the extent to which the operating variables may be expected to effect the coefficients in packed towers.

Due to the fact that the contact area can be determined in a wetted-wall tower, a considerable amount of basic mass-transfer theory has been tested in such equipment (8),(18),(4),(25). This theory has been based principally upon the assumption that mass is transferred in a manner analogous to the transfer of momentum, and consequently, the theory is evolving as our knowledge of the mechanics of turbulent flow increases. However, the discussion presented here will be confined to those wetted-wall tower data which are directly related to the theoretical treatment discussed above pertaining to the determination of the individual film coefficients for the air-water system.

Chambers (7) made adiabatic humidifying runs in a water-jacketed, glass, wetted-wall column 1.46 cm. inside diameter and 82.7 cm. in length. Inlet air was not preheated and the temperature driving force at the top of the tower ranged from 5 to 15°F. Inlet and exit air humidities were determined gravimetrically by means of a drying train. Air Reynolds numbers were varied from 1950 to 5180, but no water rates are reported. However, heat balances indicate that operation was far from adiabatic; heat losses varied from 27 to -88 per cent of the heat transferred across the gas film. Consequently the results are questionable.

Davis and Shipman (15) operated an insulated wetted-wall tower made from a 2.125 inch (inside) diameter brass pipe 139 inches long. Adiabatic humidifying runs were made over a gas Reynolds number range of 5,000 to 25,000. No air preheat was used; humidities were determined by means of wet and dry-bulb thermometers placed in the inlet and exit gas streams. Heat balances were made with the use of the reported data, and it was found that the column was gaining heat from the room in quantities which sometimes exceeded that being transferred from the air to the interface.

In view of this, it was decided that wetted-wall data available in the literature were inadequate, and that a thorough investigation paralleling that described above for a packed tower be made with a wetted-wall column.

Mass Transfer Without Heat Transfer

Although this thesis will not deal directly with this classification of diffusion, individual coefficients of mass transfer across a gas film will be determined in connection with the vaporization studies. As a result, the general status of the field will be briefly summarized in order to see whether or not the work as planned can contribute anything to this classification of diffusion.

As was pointed out in the section devoted to heat transfer, sufficient information is available so that it is often possible to add the film resistances to heat transfer and the resistance of a tube wall to obtain a reliable prediction of an over-all coefficient for a gas-liquid system that has never been experimentally studied. The two-film theory as applied to the isothermal absorption of a gas by a liquid infers that the individual film resistances should be additive in a manner analogous to that of heat transfer.

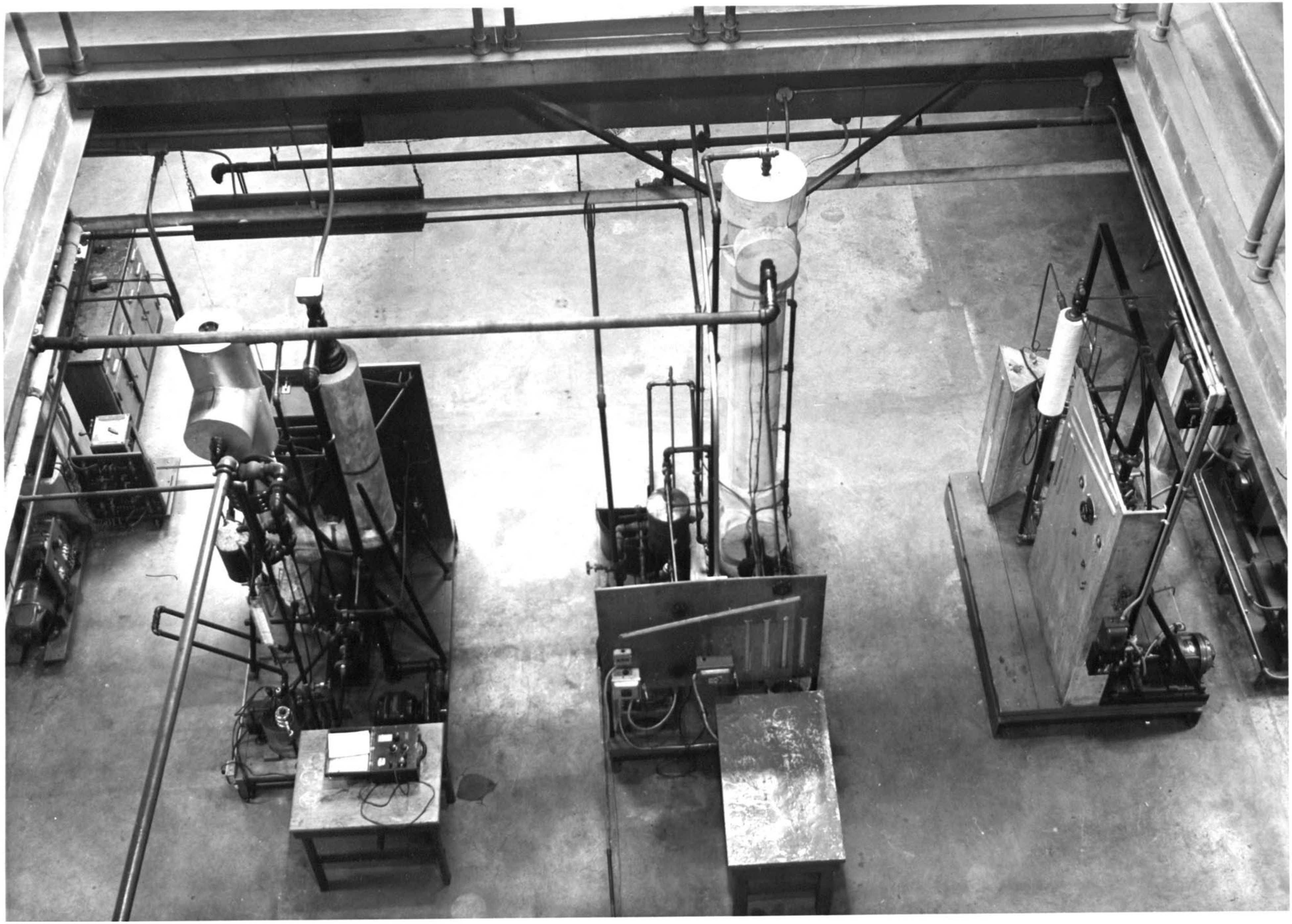
Sherwood and Holloway (43) desorbed very insoluble gases from water under conditions such that the resistance of the gas film was negligible and, as a result, obtained true coefficients of mass transfer across liquid films. By using a number of different systems they were able to correlate these coefficients in terms of operating variables and physical properties. No corresponding gas-film data are available to complete the picture, nor is it evident as to how they may be obtained (42). There is no gas-liquid system known wherein physical solubility of the gas in the liquid is ~~to~~ so high that the liquid film can be assumed to offer a negligible resistance. On the other hand the vaporization of water into air offers a method for the determination of true gas-film coefficients since there can be no concentration gradient in a pure liquid and consequently no resistance across the pure liquid film. It is, therefore, believed that true gas-film coefficients can be determined in this thesis which may be of value in the field of absorption. Whether or not simultaneous heat transfer will cause vaporation gas-film coefficients to differ from truly isothermal coefficients is not known.

III. PROCEDURE

General Program

Since the problems to be investigated involved the design and construction of three towers, each of which would require considerable time and planning due to its complicated form, it was decided to break the problem up into its major parts and to offer these as subjects of theses leading to the degree of Master of Science. Herberg and Rappaport (22) designed and built a 4-inch packed tower for the air-water study, but were unable to complete the construction and make any runs. The author redesigned and rebuilt this apparatus. R. C. St. John (45) as his Master's thesis cooperated in completing adiabatic humidifying runs and water-cooling runs on this equipment. Subsequently the author made additional runs to evaluate end-effects, and studied desorption of oxygen. W. C. Philoon, Jr. (35) and K. L. Hujsak (24) cooperated with the author in constructing and operating an 8-inch packed tower to study the air-oil system. The author designed and constructed the 1-inch wetted-wall tower, and T. R. Vick Roy is operating the apparatus; the latter thesis is still in progress.

These theses were directed by the author, and the data therefrom provided the basis for this work. The author's time was divided about equally between these theses and design and construction of the three towers. All three units were located in the same area so that a free exchange of information



existed between the workers. The units were all constructed as relatively permanent and general types of apparatus so that they may be used for additional experimental work of a nature differing from that of their initial use. In addition, each unit was mounted on a movable platform in order that each can be moved at will or stored until their use is again required at some future date. Such a procedure was possible due to the excellent facilities and the cooperation of the staff of the Chemical Engineering Shop and Stockroom.

It should be pointed out that this study was intended primarily to be one of a theoretical nature. The procurement of data which would be of value for industrial application was recognized as being desirable but was given secondary consideration.

Four-Inch Air-Water Tower

It was pointed out in the INTRODUCTION that adiabatic humidifying runs with the air-water system were difficult to accomplish with any degree of precision due to the rapid exchange of heat for mass between the hot air and the relatively cold water. This fact results in very small driving forces at the top of the tower. Methods of overcoming this difficulty were discussed, and it was shown that a high degree of preheat to the air offered the greatest promise as a solution. An error analysis (See APPENDIX) showed that, with a ten degree Fahrenheit driving force at the top of the tower, the heat transfer coefficients could be determined within

10 per cent of the true value. With a 12-inch packing of 0.5-inch ceramic Raschig rings, it was estimated that atmospheric air would have to be supplied to the tower at 700°F to provide this driving force at the top of the tower. Examination of Eq. (10a) shows that the ratio of the humidity driving force to the temperature driving force is equal to the ratio of the humid heat to the heat of vaporization, or approximately 0.24/1000. For a final temperature driving force of 10°F. this would mean that the final humidity driving force would be 0.0024 lbs. H_2O/lb. dry air. To obtain coefficients of mass transfer with the same accuracy as the coefficients of heat transfer would require that this humidity driving force be known to the nearest 0.0001 humidity unit. Preliminary tests for the determination of humidities by means of drying tubes filled with "Anhydrone", magnesium perchlorate, indicated that humidities could be determined with this precision.

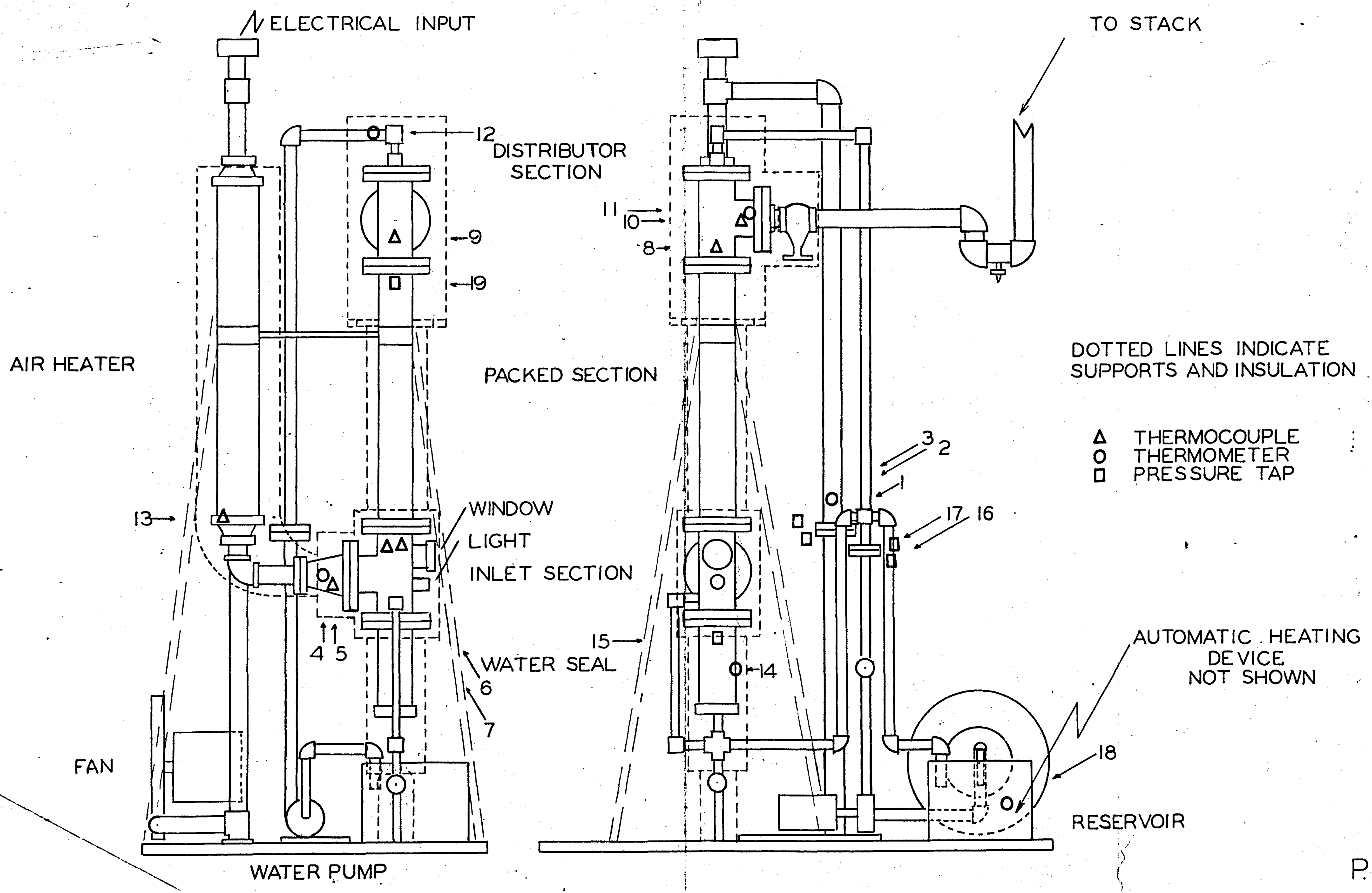
For these reasons it was decided to preheat the air to 700°F. by means of electrical heaters. In order to minimize the power consumption of these heaters it was decided to use a 4-inch tower and 0.5-inch Raschig rings as packing. This size tower would allow convenient handling of liquor and gas rates, and the pump and blower required for the two streams would be of such size as to be easily assembled on a portable platform. This size tower with 0.5 in. rings gave the ratio of tower diameter to ring size a value of eight, which is reported (<u>10</u>) as the minimum ratio to be used to subordinate wall effects. However, a few preliminary runs on the completed apparatus indicated that with 0.5-inch rings as packing, it would be necessary to use packed heights less than one-foot in order to obtain the desired driving forces at the top of the packing. For this reason it was decided to use one-inch carbon Raschig rings and one-foot of packing depth even though this arrangement might not give data representative of one-inch rings, because of possible wall effects. As will be shown later, it is believed that the tower diameter had a negligible effect on the values of the coefficients.

The experimental set-up is shown in Figure 3. Photographs of the completed apparatus follow Figure 3. The tower itself was assembled from standard 4-inch steel pipe and fittings. The section available for packing is 36 inches in height and was topped with a standard 4-inch tee to provide an exit for the air. The other opening of the tee was capped with a blank flange; through the center of this flange a 3/4-inch pipe delivered the water to a 4-point distributor placed directly over the top of the packing. At the bottom of the packed section, another standard 4-inch tee was assembled to provide an extrance for the hot air. Into the bottom opening of the tee was fitted a 4-inch nipple one foot long, capped at the bottom, to act as a sump and liquid seal. A sight glass was installed to indicate the liquid level. Water passed from this sump through a U-tube and vent arrangement, designed to maintain the liquid seal, into a glass reservoir equipped with electrical immersion heaters controlled with a thermostat. A stirrer provided sufficient agitation so that

FIG. 3

DIAGRAM OF APPARATUS

SCALE: 15:1



DOTTED LINES INDICATE SUPPORTS AND INSULATION

Δ THERMOCOUPLE
O THERMOMETER
□ PRESSURE TAP

P.F.H.

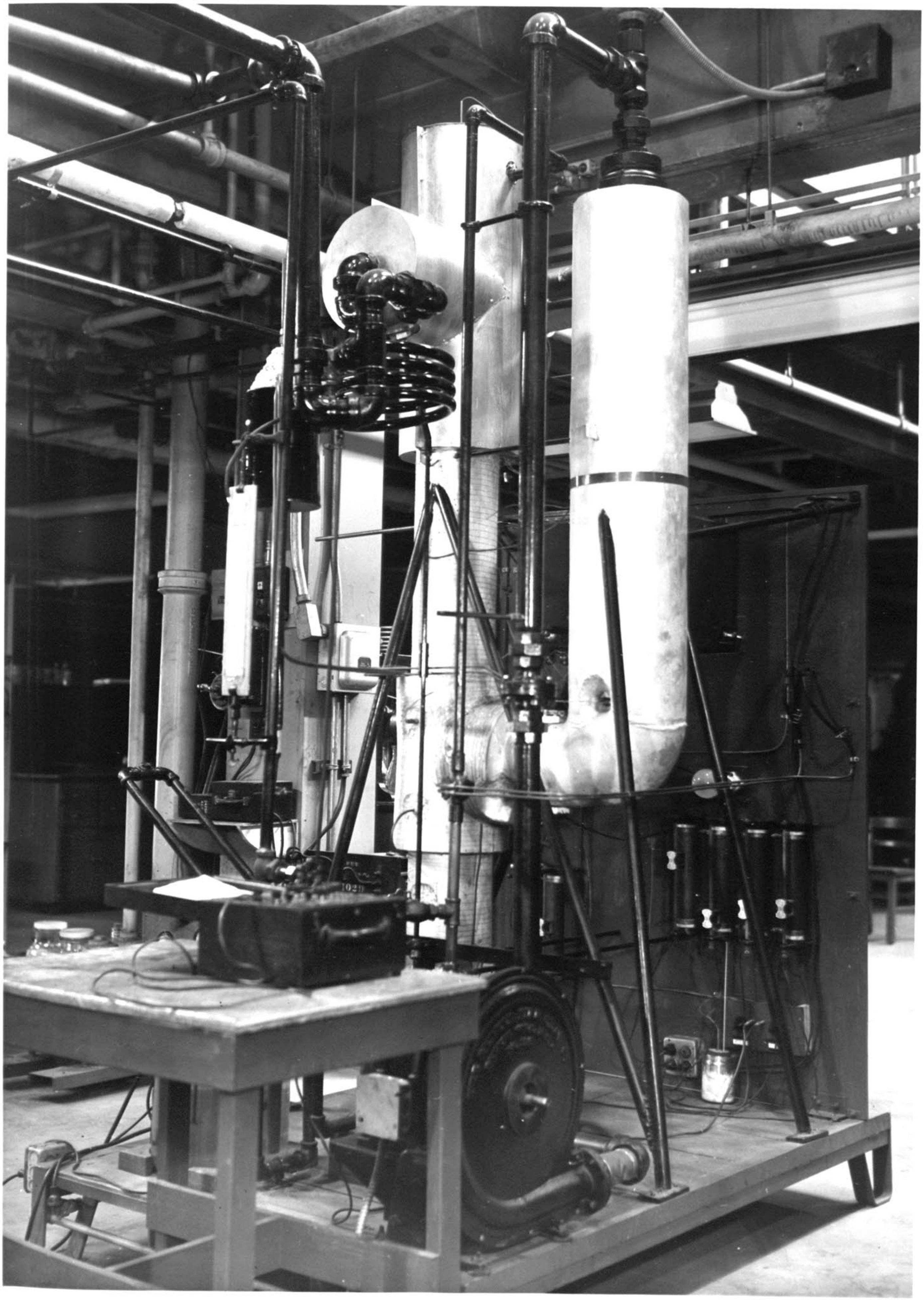
KEY TO FIGURE 3

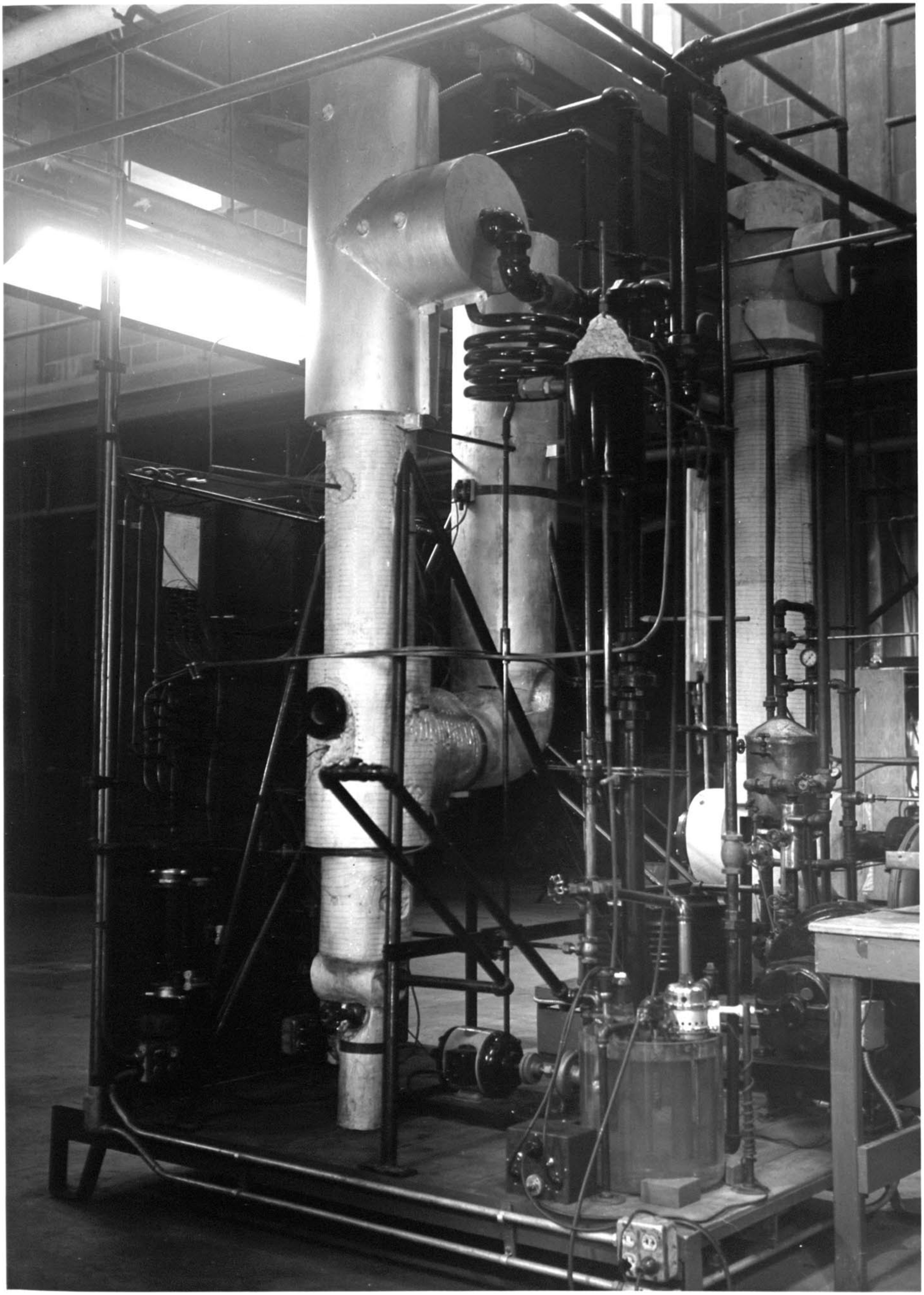
NO.	MEASUREMENT
1,2	Air orifice pressure tap.
3.	Thermometer - air orifice temperature.
4.	Shielded thermometer - inlet air temperature.
5.	High-velocity thermocouple - inlet air temperature.
6.	Thermocouple - temperature of air entering the packing.
7.	Thermocouple - temperature of water leaving the packing.
8.	Thermocouple and thermometer - temperature of air leaving the packing. (The position of 8 varied as the packed height such that the exit air temperature was measured four inches above the top of the packing.)
9.	Thermocouple - temperature of outside tower surface for regulating current input to the two electric strip heaters mounted inside the insulation and thus preventing condensation in the air leaving packed section.
10.	Thermocouple - temperature of air leaving the tower.
11.	Thermometer - temperature of air leaving the tower.
12.	Thermometer - temperature of water entering the tower. (Not shown is the thermocouple mounted in the distributor head which measured the temperature of water just as it flowed onto the packing.)
13.	Thermocouple - surface temperature of the electric air heaters.
14.	Thermometer - temperature of water in the sump.
15.	Pressure tap - static pressure at the base of the tower.

KEY TO FIGURE 3 (Cont.)

NO.	MEASUREMENT
16,17	Pressure taps - water orifice.
18.	Thermometer - temperature of water in the reservoir.
19.	Pressure tap - static pressure at the top of the tower.

The device used for measuring the exit humidity is not shown.





the water in the reservoir could be maintained at any predetermined temperature. From the reservoir, the water was forced by means of a centrifugal pump through a calibrated water orifice and into the distributor at top of the tower. The pump had sufficient capacity to force water into the tower at any desired rate up to 2500 pounds per hour per square foot of gross cross section. Two calibrated orifice plates were used to cover this range of liquor rates. Make-up water was added to the reservoir as needed.

The preheat to the air was provided by means of electrical strip heaters. These were assembled in a four-foot section of five-inch standard steel pipe. The air was supplied by a rotary blower at any desired rate up to 1000 pounds per hour per square foot of gross cross section. From the blower the air passed upward through straightening vanes and then through a sharp-edged orifice. Three sizes of orifices were calibrated to cover the desired range of gas velocities. The air then passed down through the preheater and into the bottom of the tower. From the top of the packing the air passed out of the tower, part of it going to a stack and the remainder passing through a device for measuring the humidity. This device is discussed below.

Temperature Measurements

Temperature measurements were made by means of calibrated thermometers and thermocouples. In many cases a thermometer and thermocouple were used at the same place in the apparatus to act as a check upon each other. A thermometer placed six inches downstream from the air orifice furnished the necessary air temperature for the orifice calibration. A thermocouple was welded directly to the surface of a strip heater at the outlet end of the preheater. It was necessary to operate these heaters at surface temperatures below 1200°F. to prevent burning out the heaters. The temperature of the air as it entered the tower was obtained by means of a high velocity thermocouple and a shielded thermometer. The temperature of the air leaving the packing was determined by means of a thermocouple and a thermometer located in the gas stream. Since this air was near its dew-point, heat was supplied to the top section of the tower, above the packing, by means of two strip heaters to prevent condensation. A thermocouple, located in the air stream leaving the tower, coupled with a resistance in series with the heaters, provided the control for these two strip heaters. This exit air temperature was maintained equal to or slightly higher than the temperature of the humid air leaving the packing.

In the water cycle, a thermocouple located in the distributor head gave the temperature of the water flowing onto the packing. Immediately below the perforated brass plate used for the packing support, was located a thermocouple to measure the temperature of the water. A glass window was built into the tee below the packed section and furnished visual assurance that this thermocouple was kept wet by the water leaving the packed section. Means was provided for relocating this couple from outside the tower and thus determine any temperature variation in the exit water. A thermometer was also placed in the sump at the bottom of the tower. In the early adiabatic humidifying runs it was found that the water temperature in the sump was always higher than the water leaving the packing. This was shown to be due to the fact that heat was being conducted along the wall of the hot air inlet pipe to the water flowing from the packed section into the sump. This conduction was preventing the closure of heat balances on the tower and was allowed for in subsequent runs (See APPENDIX).

Humidity Determinations

While designing the tower and laying out the experimental procedure, it was planned to measure the entrance and exit air humidities gravimetrically by passing metered quantities of the air-water mixtures thru a drying tube. The possibility of using wet and dry bulb thermometers for measurement of humidity was rejected due to the dependence of the wet and dry-bulb technique upon the assumption that $h/k' = s$. Sherwood (54) has pointed out that such a procedure prevents the independent evaluation of the two gas-film coefficients. However, it was found in early runs that the gravimetric method failed completely to give accurate results for the highly humid air leaving the tower. This air was usually at temperatures of 140-155°F., and its dew point was 135-140°F. As a result it was very difficult to prevent condensation in the sampling lines. When the sampling lines were electrically heated, water condensed in the stop-cocks of the weighing tube. Furthermore, the quantity of water to be removed per unit volume of air was so large that it was necessary to use two and three drying tubes in series to assure complete water removal from the air. Condensation occurred in the connections between the tubes, making it necessary to weigh the entire train as a unit. Since the weight and size of such a train prevented the use of any available analytical balance, the gravimetric technique was finally abandoned.

Numerous techniques for the precise determination of humidities were investigated and evaluated. However, no completely satisfactory method was found. The procedure finally used consisted of a modified dew-point determination. A metered portion of the exit air from the tower was passed through air-cooled coils where some condensation occurred. The condensate was removed from the saturated air stream in a cyclone separator and collected in a burette attached to the cone of the separator.

The pressure and temperature of the air in the cyclone were measured, and, since the air was assumed saturated at this point, its humidity was determined from water vapor-pressure data. This figure was then corrected for the condensate collected. Since no condensation occurred between the top of the packing and the cooling coils, this figure was taken to represent the humidity of the air leaving the packing.

During the initial runs, inlet air humidities were determined gravimetrically and by means of a sling psychrometer. Using experimental psychrometric data, as determined by the U. S. Bureau of Standards (53), the psychrometer readings were converted into humidity values which agreed very closely with those obtained by the gravimetric method. Consequently, wet and dry-bulb readings were employed to obtain the low humidities of the inlet air for the later runs.

Insulation

The entire apparatus was thoroughly insulated with magnesia and an error analysis assuming true adiabatic operation, indicated that heat balances should close within 1% of the sensible heat transferred across the gas film. Preliminary runs, however, indicated that the tower was losing considerable quantities of heat to the surroundings. To eliminate this loss, the surface of the insulation was heated by means of electrical heating coils (See APPENDIX) to maintain a zero radial temperature drop across the insulation. By this means the tower operated essentially adiabatically, and all data used in the air-water correlations were calculated from runs wherin the heat balances closed to within 1% of the sensible heat transfer.

The Effect of Temperature

The effect of temperature on the coefficients for the gas film was determined by adiabatic humidifying runs at different inlet air temperatures and corresponding wet-bulb temperatures, holding other variables unchanged. The gas-film coefficients were determined under temperature conditions quite different from those encountered in water-cooling runs, making a temperature correction necessary.

End-Effects

Although several plans were considered for the elimination of end-effects in the column, these were rejected, and the end-effects were determined directly by varying the packed height, holding all other variables constant. At the top of the column, end-effects refer to the fact that the liquid stream penetrates into the packing some distance before it becomes evenly distributed, leaving some dry areas unavailable for interphase diffusion. However, this so-called "coning" effect should be small with the multi-point distributor. At the bottom of the tower the water falls from the packing support into the sump forming a "spray section". Although these two effects tend to cancel one another, the spray-effect usually is of the greater importance. In order to obtain some direct measure of this spray-effect, the air temperature was measured just below the packing by means of a thermocouple placed under a small metal cone to keep it dry.

Effect of Tower Diameter to Ring Size

Sherwood and Holloway (43) desorbed oxygen from water by passing air countercurrently to an oxygen-enriched liquid stream in a 20-inch diameter tower packed with one-inch ceramic Raschig rings. End-effects were eliminated by sampling just above and below the packed section and the effect of temperature was evaluated for coefficients of mass transfer.

Similar runs were made in the air-water tower of this thesis, and the mass transfer coefficients compared with those of Sherwood and Holloway in an attempt to determine the effect of tower diameter.

Adiabatic Humidifying Operation

For the adiabatic humidifying runs, the operation of the equipment proceeded as follows: the water was started circulating by means of the centrifugal pump. As soon as the water sump had filled so as to seal the bottom of the tower, the air blower was started and the air preheater was turned on. The thermostat which controlled the temperature of the water in the reservoir was then adjusted so that the water entering the tower would be at the desired adiabatic saturation temperature. The water rate was next adjusted to the desired value by means of a needle valve on the discharge end of the pump. In order to bring the air up to temperature as rapidly as possible, the air rate was usually maintained at a maximum rate and the heaters loaded to capacity. After the air temperature had reached about 500°F., the air rates to the tower and through the cooling coils and cyclone separator were adjusted to the desired values, and the current input to the heaters reduced accordingly by means of an induction transformer. The heating coils on the tower insulation were then turned on, and a resistance in series with each coil was adjusted to give a zero temperature drop across the insulation. All thermocouple temperatures were recorded on a 16-point recording potentiometer which greatly facilitated the operation. Approximately five hours were required by the apparatus to reach steady state. When it was thought that steady-state operation had been achieved, the flow-rates of all streams, and all temperatures and pressures were recorded. From these data a heat balance was made. If the balance closed within 1% of the sensible heat transferred across the gas film, the run was considered complete, and conditions were changed for the next run. If the heat balance did not close properly the apparatus was allowed to run another thirty minutes and this procedure repeated. Once the apparatus had been brought up to temperature, subsequent runs required about three hours to reach steady-state conditions. The coefficients of heat and mass transfer across the gas film were then calculated directly from Eqs. (14) and (15). Sample calculations are given in the APPENDIX.

Water-Cooling Operation

The water-cooling runs were made in much the same fashion as were the adiabatic humidifying runs except that air at room temperature was supplied to the tower. As would be expected, a much shorter time was required for the apparatus to reach steady-state. Heat balances for these runs closed within 1% of the sensible heat transferred across the liquid film. The coefficients of heat transfer across the liquid film were determined by the procedure outlined in the INTRODUCTION.

End-Effects

The end-effect runs were a duplication of the adiabatic humidifying runs described above except for the fact that the packed height was varied. The effect of both gas and liquid rates on the end-effects was determined.

Oxygen Desorption

The very simple procedure developed by Holloway (23) and used by Vivian (47) was followed. The water fed to the tower was enriched with pure oxygen, analyzed for oxygen before and after passing through the packing, and the temperature and water rate recorded. In order to maintain the liquid film at a constant temperature, adiabatic humidifying conditions were employed. The equations used for the determination of coefficients of oxygen transfer are presented in the APPENDIX.

One-Inch Wetted-Wall Tower

At the beginning of this section it was stated that the general program of research included the construction and operation of a wetted-wall tower to be used in the study of the air-water system. Since, however, the experimental work of this phase of the program is still in progress, only the design and the construction of the apparatus can be considered here.

The schematic diagram of the apparatus is shown in Figure 3A; details of the tower construction are shown in Figure 3B. Photographs of the equipment follow Figure 3B.

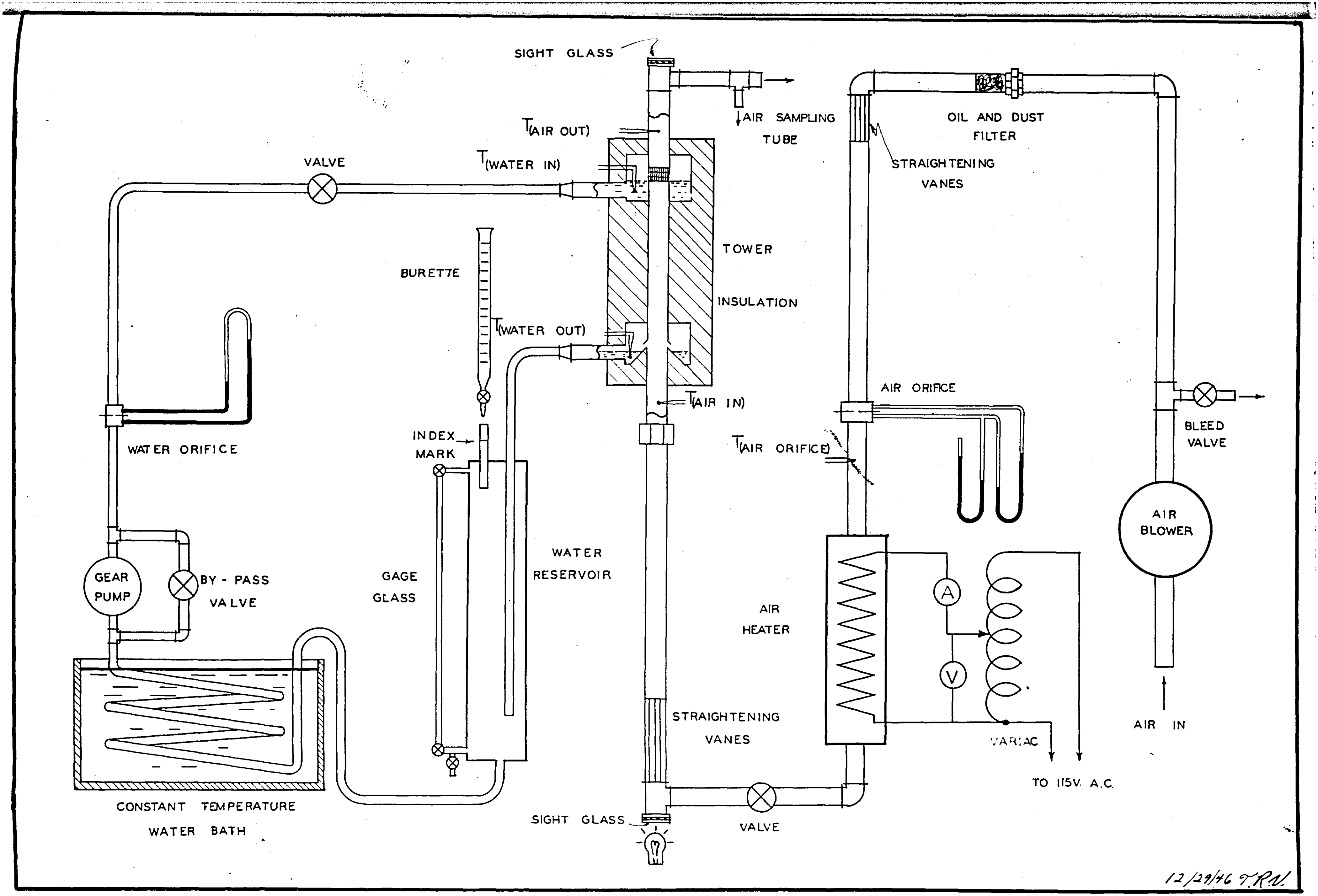


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FIGURE 3A SCHEMATIC DIAGRAM OF APPARATUS

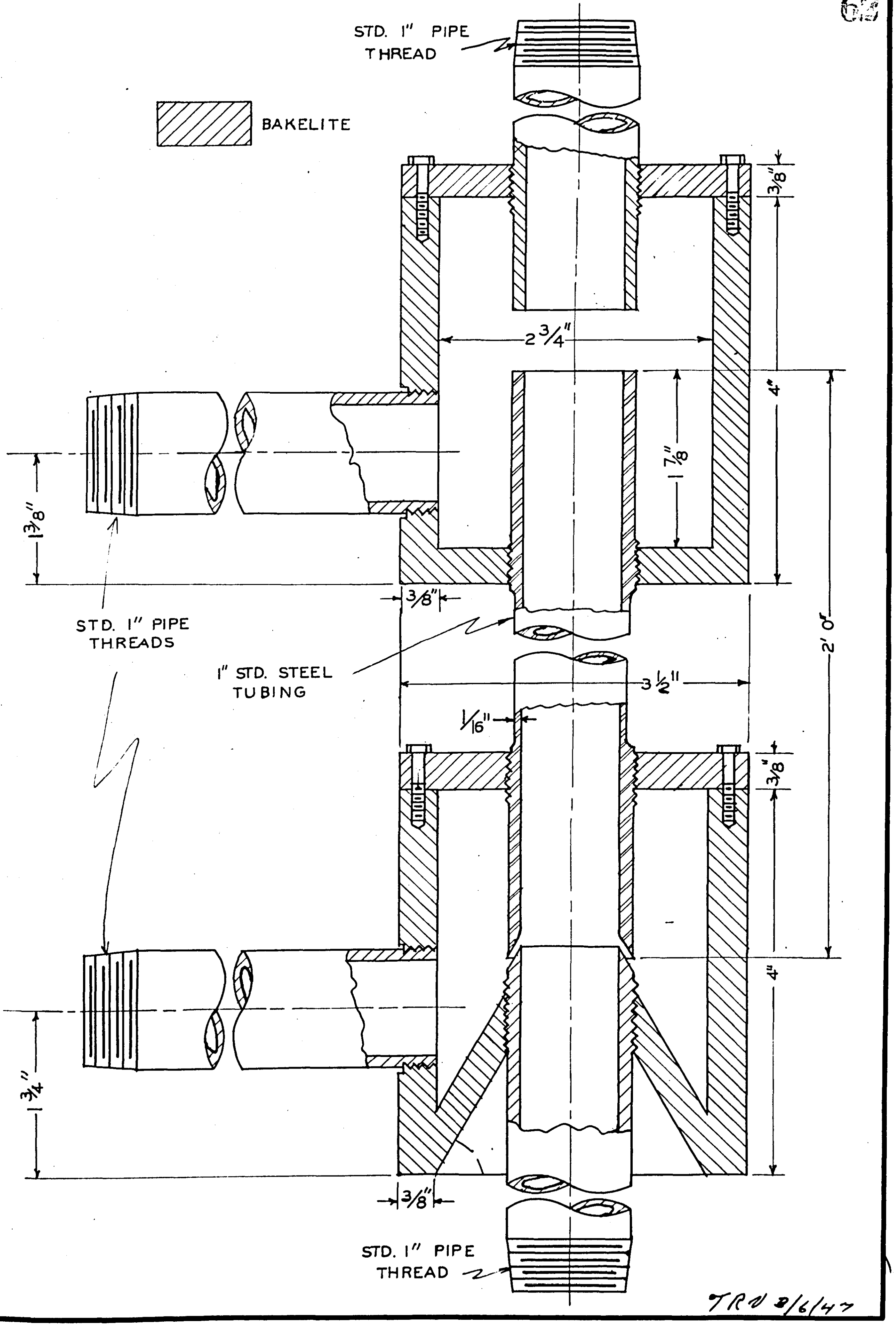
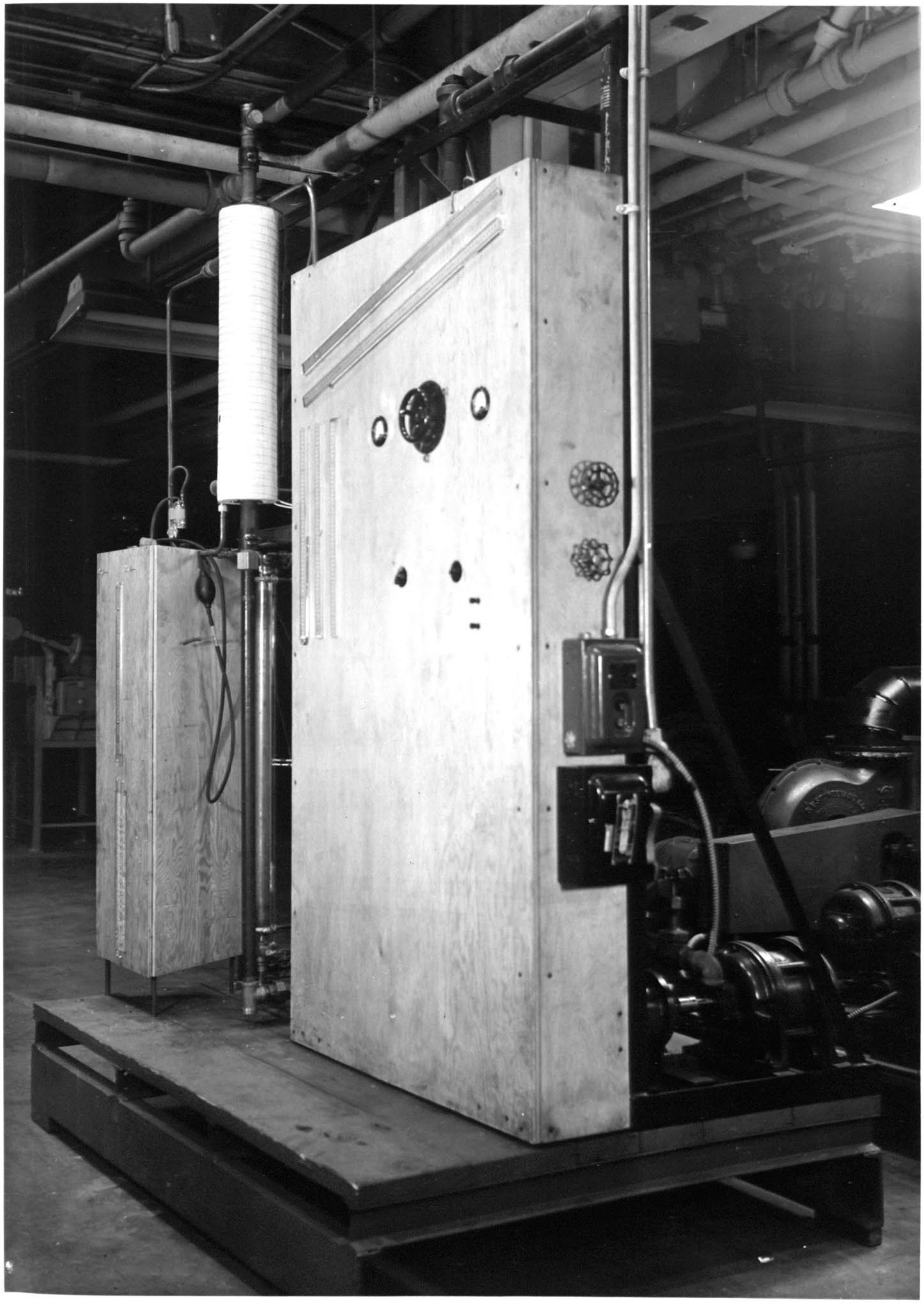
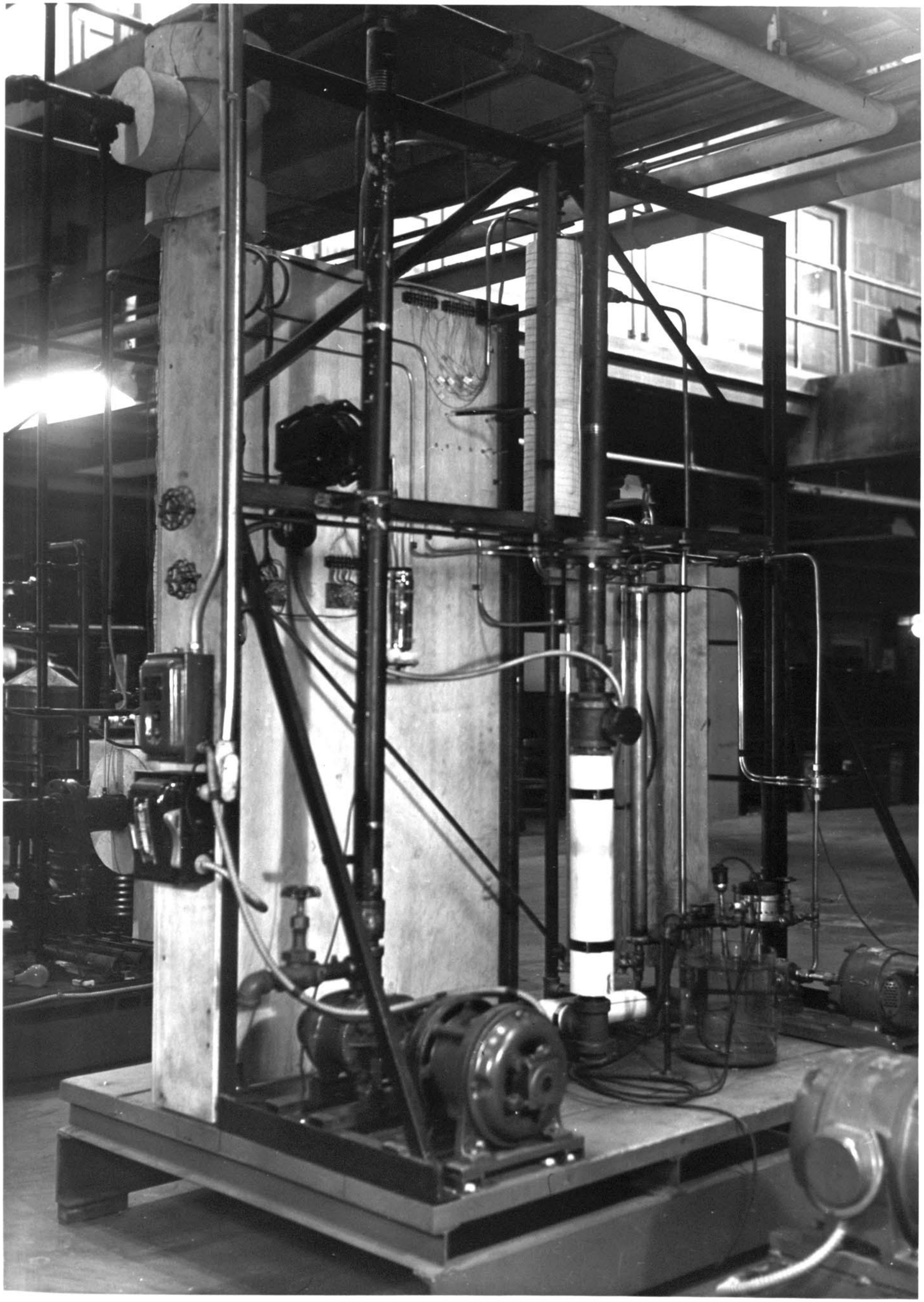


FIG. 3B DETAIL DRAWING OF WETTED-WALL TOWER





The general auxiliary equipment was much the same for this tower as for the 4-inch column in that it included an air blower, electrical air preheater, water pump, and constant temperature reservoir for recycling the water. Calibrated orifices were installed in both the air and water lines. Another water reservoir was employed in the water cycle in such a manner that the depletion of water in the system could be accurately measured. All water lines were of either copper or brass to eliminate rust formation.

The tower itself was originally constructed from a 2-foot length of standard 1-inch stainless steel pipe. However, it was found that only by means of frequent cleanings with a strong acid solution was it possible to maintain an even film of water on the inside wall, and the stainless steel column was discarded in favor of another made from mild steel which wetted much more easily. The inlet water flowed into a reservoir attached to the top of the tower which acted as an overflow pipe. The lower end of the column was turned to a sharp edge on the outside to carry the water away from the air stream and into another reservoir. The air line leading into this reservoir was a vertical 4-foot length of 1-inch pipe which functioned as a calming section. Similarly a short calming section was attached to the upper reservoir for the exit air. At the ends of the calming sections were fitted glass windows to allow visual inspection of the inside tower surface while a run was in progress.

There were two principal problems to be solved in the design of the wetted-wall column. Previous workers with such apparatus had been unable to achieve adiabatic operation, and therefore special precautions were taken in this respect. The tower was first well insulated with magnesia, and the outside surface was then provided with electrical wire heaters for maintaining a zero radial temperature gradient in the insulation. The use of electrically heated insulation prevented heat exchange between the tower and the air in the room, but would not eliminate any heat that would be conducted along the hot air inlet pipe into the water reservoir at the bottom of the tower. Consequently, the water reservoirs at the bottom and top of the tower along with the air and water inlet and outlet pipes were machined from a laminated bakelite plastic (Micarta) which has a thermal conductivity approximately half that of asbestos. In addition the water reservoir at the base of the tower was coned at the bottom to minimize the contact area between the hot air inlet pipe and the water (See Figure 3B). The surface of the air inlet pipe as well as the air-side of the cone surface were covered with polished aluminum foil to reduce heat transfer by radiation. With these precautions, trial runs indicated that it would be possible to obtain essentially adiabatic operation.

A method for the precise determination of inlet and exit air humidities presented the other major difficulty.

Experience gained from operation of the 4-inch packed tower indicated that a drying train offered a solution to this problem providing the dew-point of the exit air was below room temperature so that condensation would not occur in the sampling lines. Using the data of Davis and Shipman (15) preliminary calculations showed that with a tower two foot high, exit air dew points would be below 70°F for adiabatic humidifying runs with water temperatures of 90 to 95°F. Under these conditions it would be possible to determine heat-transfer coefficients within 2% of the true value and mass-transfer coefficients within 6%. The use of water temperatures of 90 to 95°F would require inlet air temperatures of 200 to 250°F. and so the air preheater was designed accordingly.

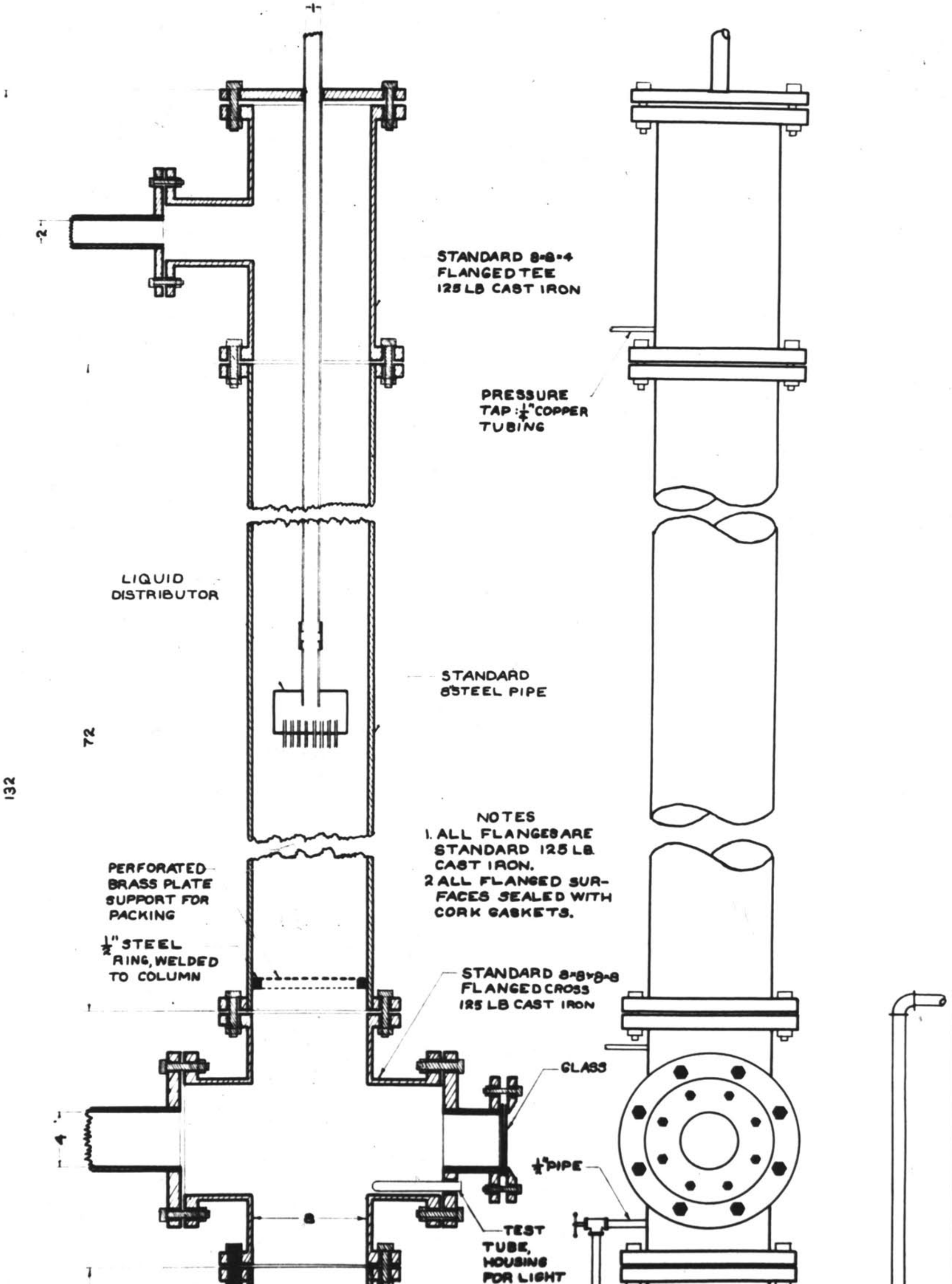
All temperature measurements were made by means of calibrated thermocouples and a precision potentiometer so that temperatures could be obtained to the nearest 0.1°F.

Eight-Inch Air-Oil Tower

The 8-inch air-oil tower was designed from the only available data, those of Bennett (4). Since Bennett's data were obtained with 0.5 inch Raschig rings whereas 1-inch ceramic Raschig rings were used as packing material in this tower, it was realized that the tower was somewhat over-designed.

The tower itself was made up from standard 8-inch steel pipe in a manner similar to that used for the 4-inch air-water tower described above. Details of the tower construction are shown in Figure 4 and in the photographs. The central length of the tower was six feet and eight inches long. The top was a standard tee, the bottom a standard cross. A cross was used at the bottom rather than a tee in order to easily provide an assembly for a window. At the bottom of the cross was fitted an eight-inch nipple, closed by a cap, to act as a sump. The oil was forced out of the sump by the pressure in the column through an adjustable, vented trombone arrangement. This provided a head of oil equal to the air pressure in the tower, thereby allowing a steady flow of oil, but preventing the escape of air. The trombone emptied the oil into a reservoir. From the reservoir the oil was forced by means of a centrifugal pump through a shell-and-tube heat exchanger, a calibrated orifice, and thence into the top of the tower where a 12-point distributor fed it onto the packing. Steam was supplied to the heat exchanger by means of a

FIGURE 4
DETAILS OF TOWER
(INSULATION NOT SHOWN)



NOTES
1. ALL FLANGES ARE STANDARD 125 LB CAST IRON.
2. ALL FLANGED SURFACES SEALED WITH CORK GASKETS.

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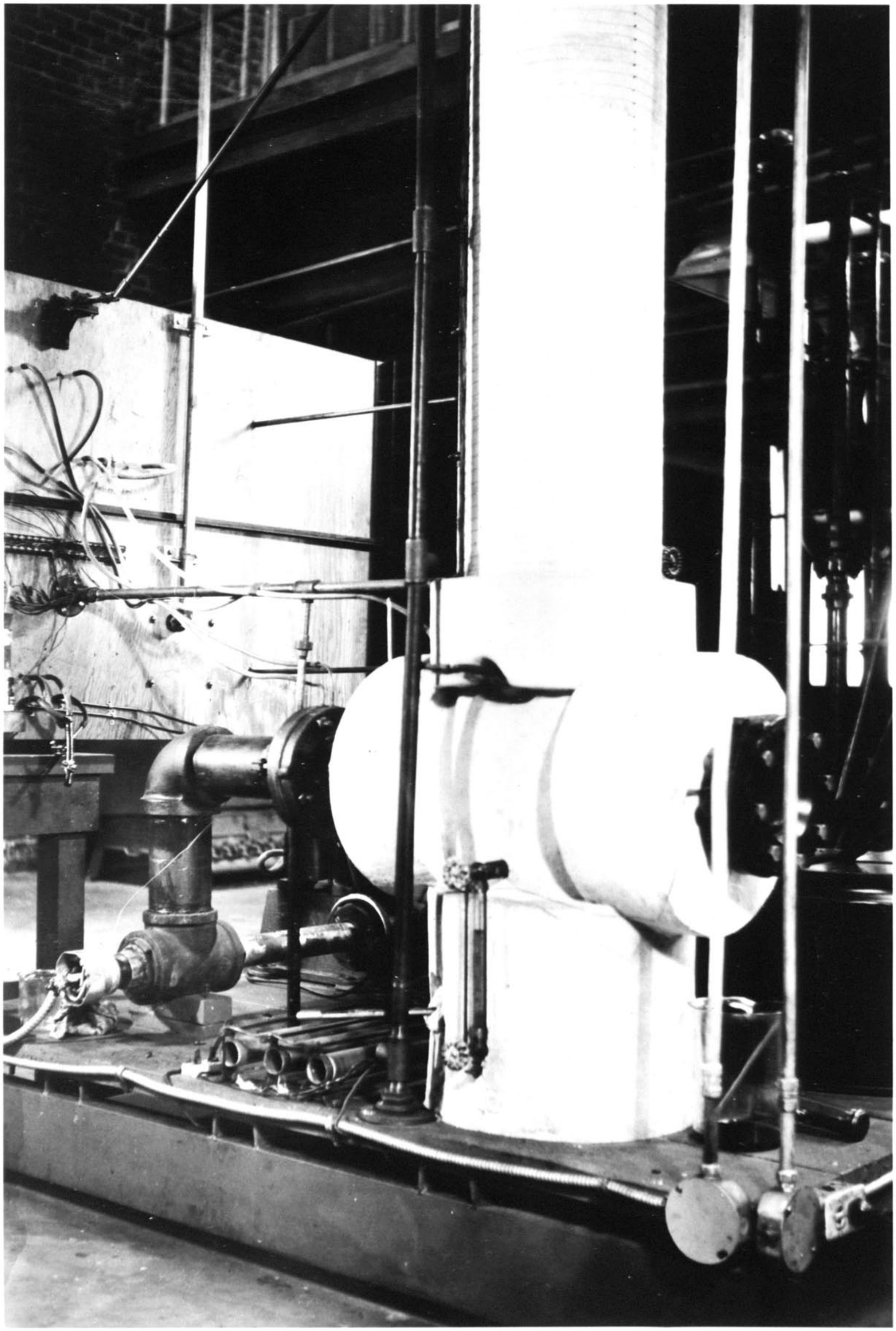
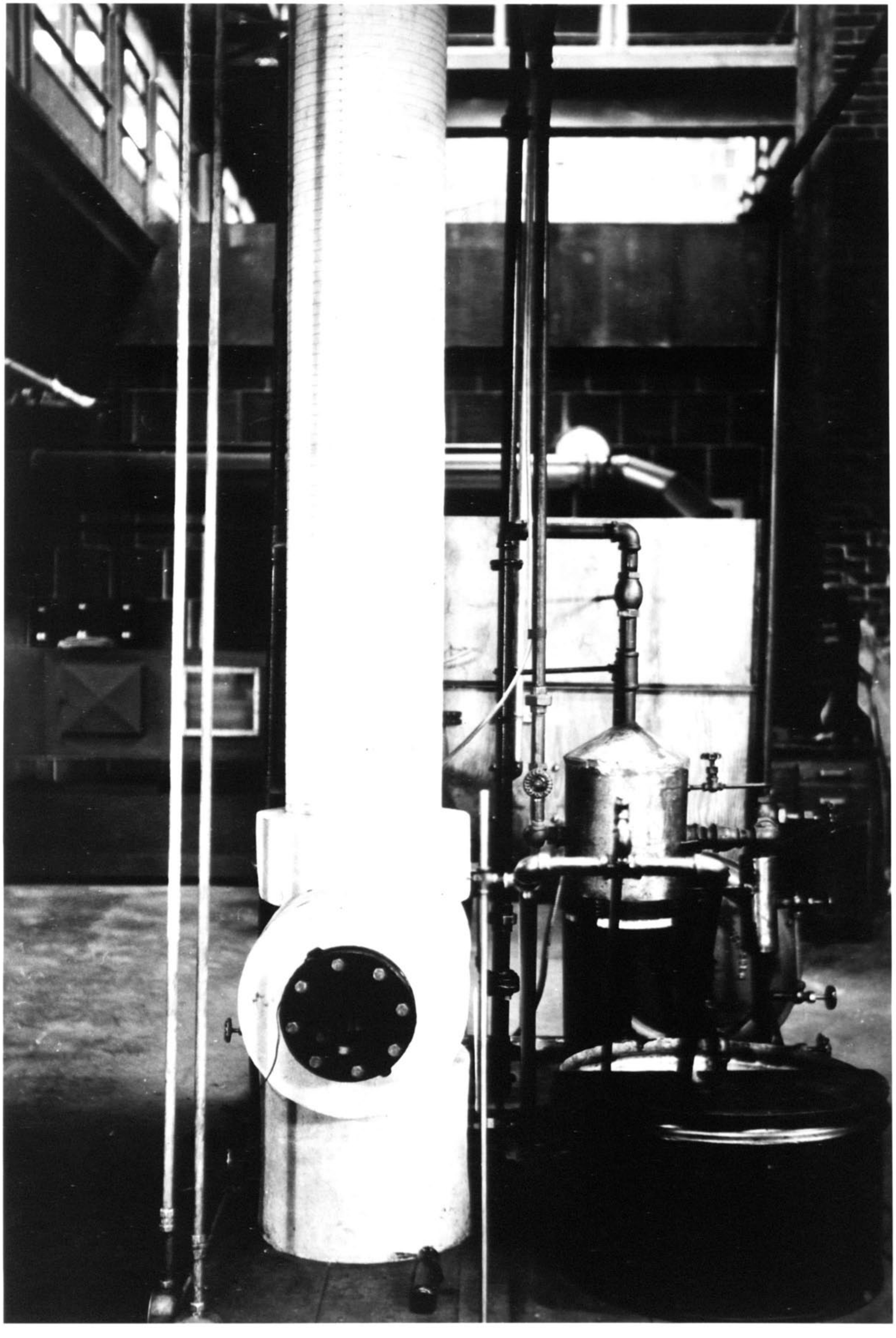




small, gas-fired boiler with automatic pressure controls. A valve arrangement permitted the flow of either cold tap water or steam to the shell of the heat exchanger to provide either cooling or heating.

The air was forced by means of a blower through an electrical air preheater which contained strip heaters similar to those employed in the air preheater on the air-water tower. From the preheaters, the air passed through a calibrated orifice and then into the bottom of the tower. The air from the tee at the top of the tower was vented to the roof.

Temperature Measurements

The original temperature measurements were made with thermometers placed in the entrance and exit oil and air streams. Most of these were subsequently replaced by the calibrated thermocouples. The final arrangement for temperature measurements was as follows: the entering air temperature was determined by means of a shielded thermometer located in the center of the air inlet duct at the base of the tower. Two thermocouples approximately three inches above the packing indicated exit air temperatures. The average of these was used as exit air temperature when any small difference was noted between the two readings. A thermocouple placed in the distributor head gave the oil inlet temperature. In the base of the tower, the exit oil temperature was measured by means of a thermocouple located in a funnel through which a large portion of the exit oil passed. This arrangement assured good contact between the thermocouple housing and the exit oil stream.

Insulation

In order to improve the apparatus over that used by Bennett, it was hoped that a thick magnesia insulation would be sufficient to reduce heat losses to a negligible value. It was found on preliminary runs, however, that this was not the case, and heat losses, though less than Bennett's, were still considerable. As a result the surface of the insulation was heated electrically to maintain a zero temperature drop across the insulation, as was done for the air-water tower. This procedure served to reduce the heat losses in all subsequent runs, so that substantially adiabatic operation was obtained.

End-Effects

End-effects were determined directly by varying the packed height. The effect of oil and air rates on the end-effects was also determined for both oil-cooling and oil-heating runs.

Operation

The operation of the air-oil tower was essentially the same as that for the air-water apparatus. The sump was first filled by starting the oil flow. The air blower was then put into operation and both rates adjusted by means of suitably located valves. The preheat for the oil stream or for the air stream was then adjusted depending on whether an oil heating or an oil cooling run was being made. The insulation heaters were turned on and their temperatures adjusted by means of suitable resistances in series with each heating element.

The apparatus required about three hours to come to steady state. When it was thought that steady state had been reached, all temperatures were recorded and a heat balance made. If the balance closed, the run was assumed complete and the variables changed. If the balance did not close the tower was allowed to operate for another 15-minute period and the procedure repeated. The over-all coefficients of heat transfer between the oil and air were calculated from Eq. (7). Heat balances were made by employing Eq. (1). Sample calculations are given in the APPENDIX.

IV. RESULTS

The results are presented in Figures 5 through 33 and can be classified into four sections.

(1) Figures 5 through 15 deal with the adiabatic humidifying runs on the air-water tower, showing the effects of temperature, packed height, water and air rates.

(2) Figures 16 through 19 give the results of the cooling-tower runs.

(3) Figures 20 through 25 deal with the hot air phase of the air-oil study in the 8-inch tower.

(4) Figures 25 through 33 refer to the hot oil data from the air-oil tower.

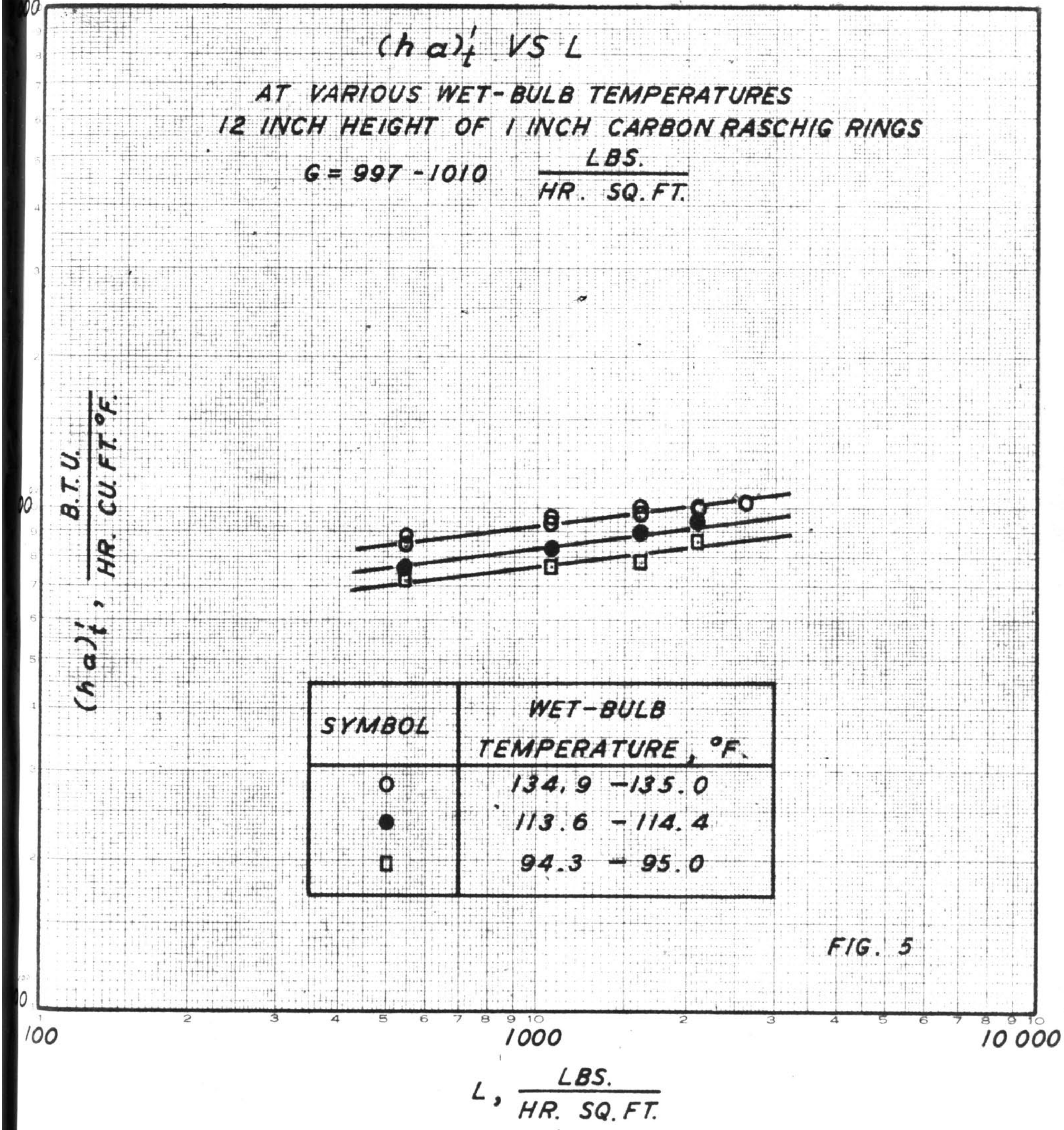
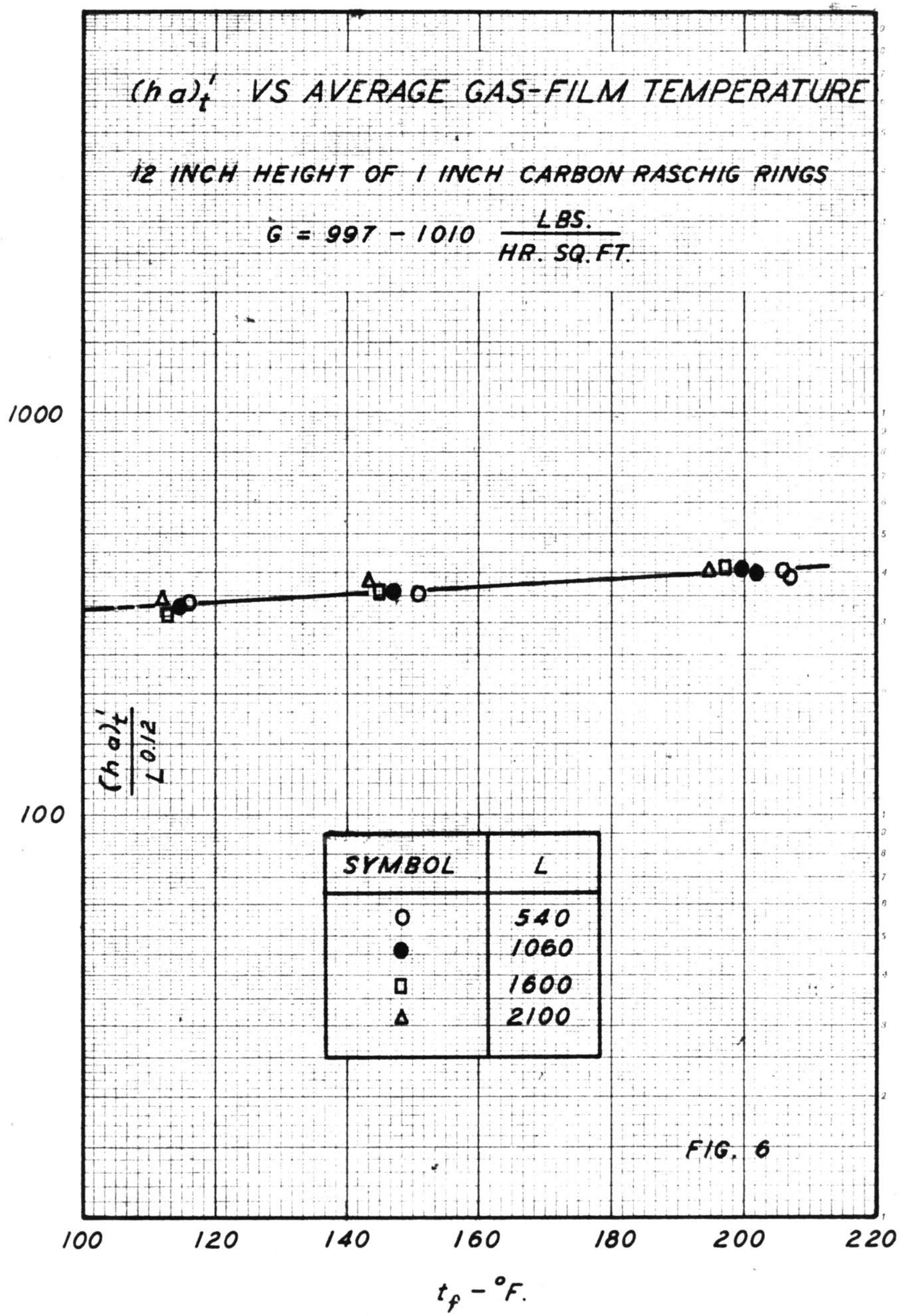

$(h\,a)'_t$ VS L
AT VARIOUS WET-BULB TEMPERATURES
12 INCH HEIGHT OF 1 INCH CARBON RASCHIG RINGS
G = 997 - 1010 $\frac{\text{LBS.}}{\text{HR. SQ. FT.}}$
$(h\,a)'_t$, $\frac{\text{B.T.U.}}{\text{HR. CU. FT. °F.}}$
SYMBOL
WET-BULB TEMPERATURE, °F
○ 134.9 - 135.0
● 113.6 - 114.4
□ 94.3 - 95.0
100
1000
10 000
L, $\frac{\text{LBS.}}{\text{HR. SQ. FT.}}$

FIG. 5



$(ha)'_t$ VS AVERAGE GAS-FILM TEMPERATURE

12 INCH HEIGHT OF 1 INCH CARBON RASCHIG RINGS

$G = 997 - 1010 \dfrac{\text{LBS.}}{\text{HR. SQ. FT.}}$

FIG. 6

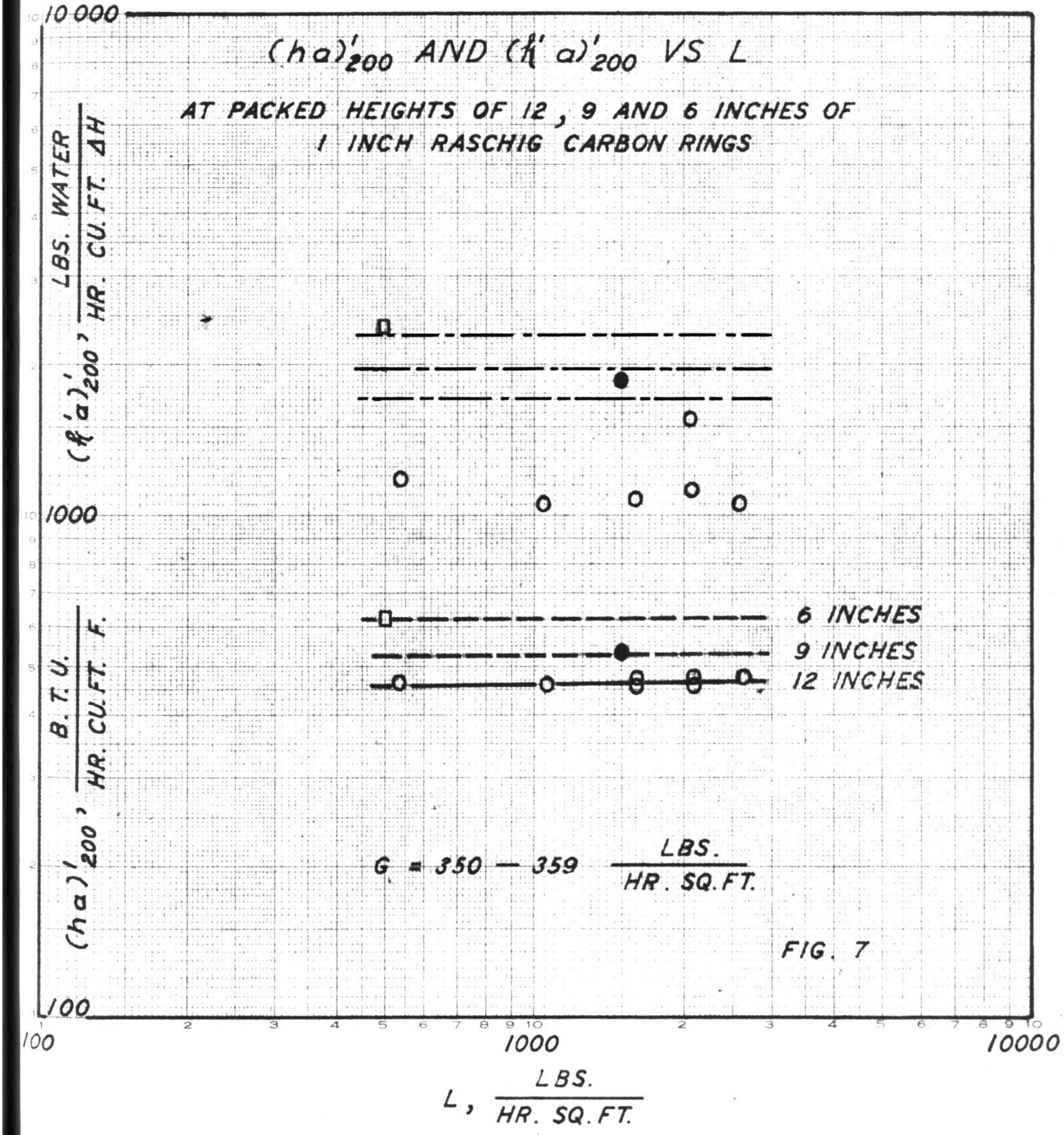
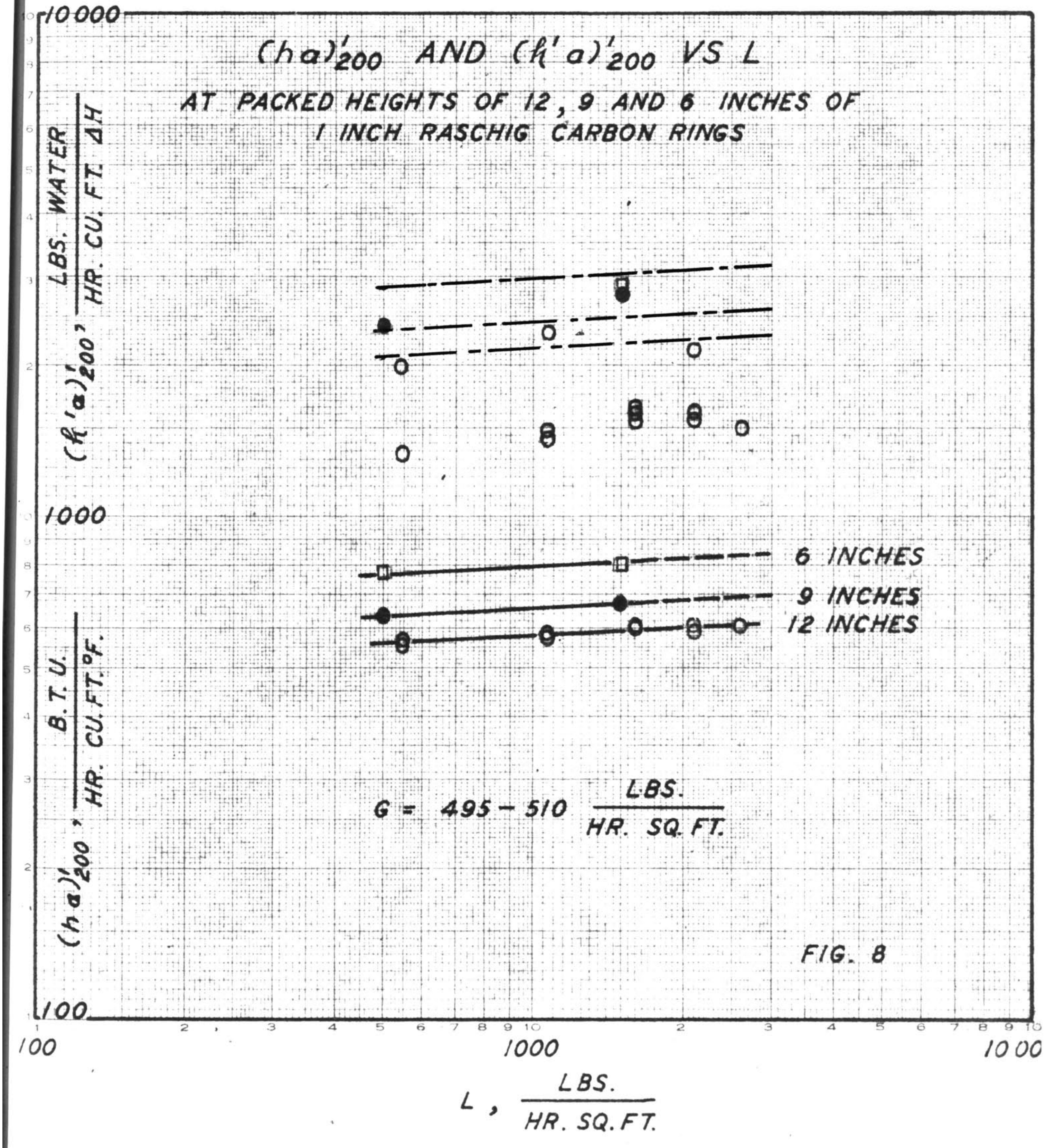

(ha)'200 AND (k'a)'200 VS L
AT PACKED HEIGHTS OF 12, 9 AND 6 INCHES OF
1 INCH RASCHIG CARBON RINGS
(k'a)'200, LBS. WATER / HR. CU.FT. ΔH
(ha)'200, B.T.U. / HR. CU.FT. F.
10 000
1000
100
6 INCHES
9 INCHES
12 INCHES
G = 350 — 359 LBS. / HR. SQ.FT.
100
1000
10000
L, LBS. / HR. SQ.FT.

FIG. 7



$(ha)'_{200}$ AND $(k'a)'_{200}$ VS L

AT PACKED HEIGHTS OF 12, 9 AND 6 INCHES OF 1 INCH RASCHIG CARBON RINGS

FIG. 8

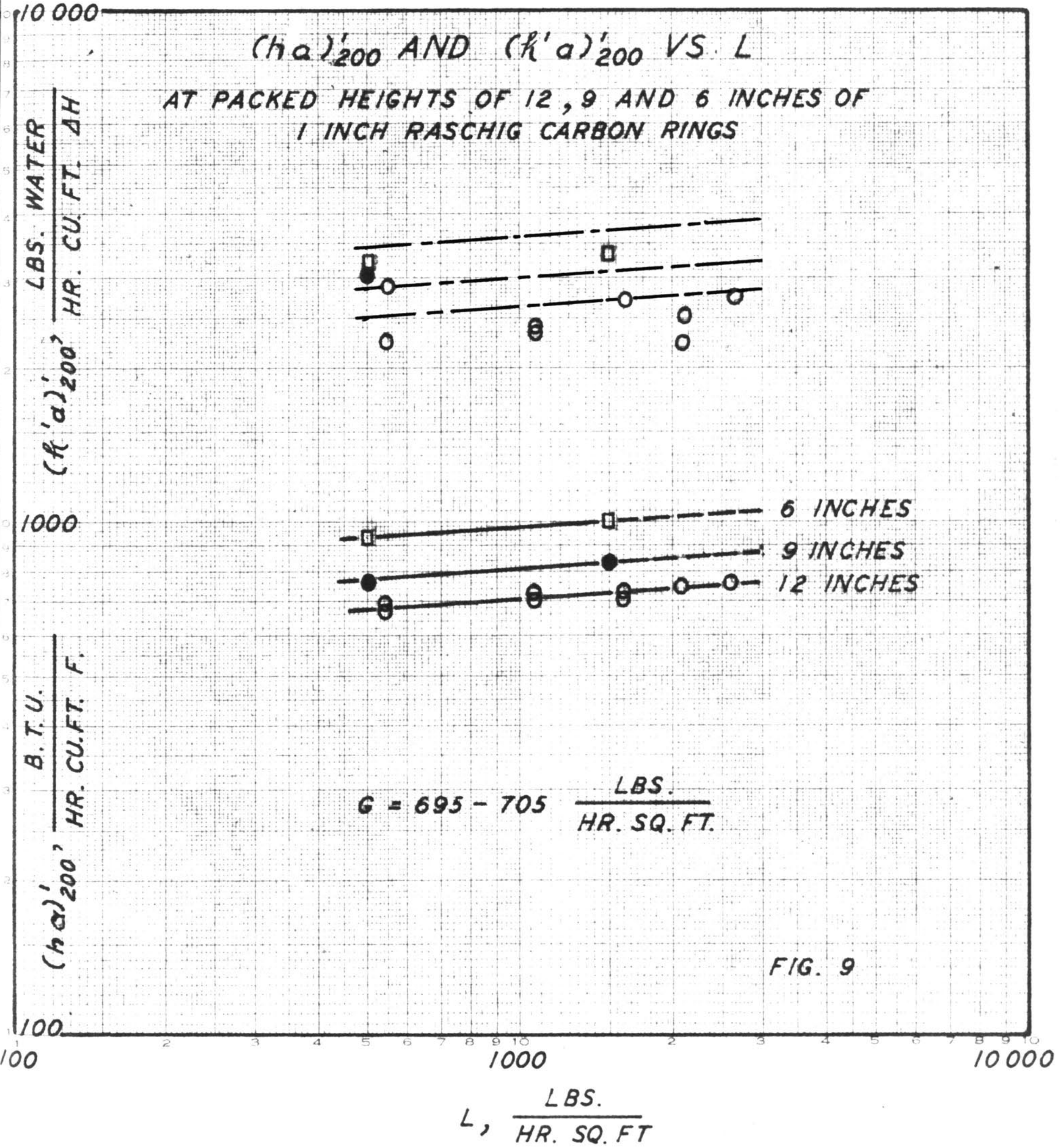

$(ha)'_{200}$ AND $(k'a)'_{200}$ VS. L
AT PACKED HEIGHTS OF 12, 9 AND 6 INCHES OF
1 INCH RASCHIG CARBON RINGS
$(k'a)'_{200}$, $\frac{LBS.\ WATER}{HR.\ CU.\ FT.\ \Delta H}$
$(ha)'_{200}$, $\frac{B.T.U.}{HR.\ CU.FT.\ F.}$
10 000
1000
100
6 INCHES
9 INCHES
12 INCHES
$G = 695 - 705\ \frac{LBS.}{HR.\ SQ.\ FT.}$
100
1000
10 000
L, $\frac{LBS.}{HR.\ SQ.\ FT}$

FIG. 9

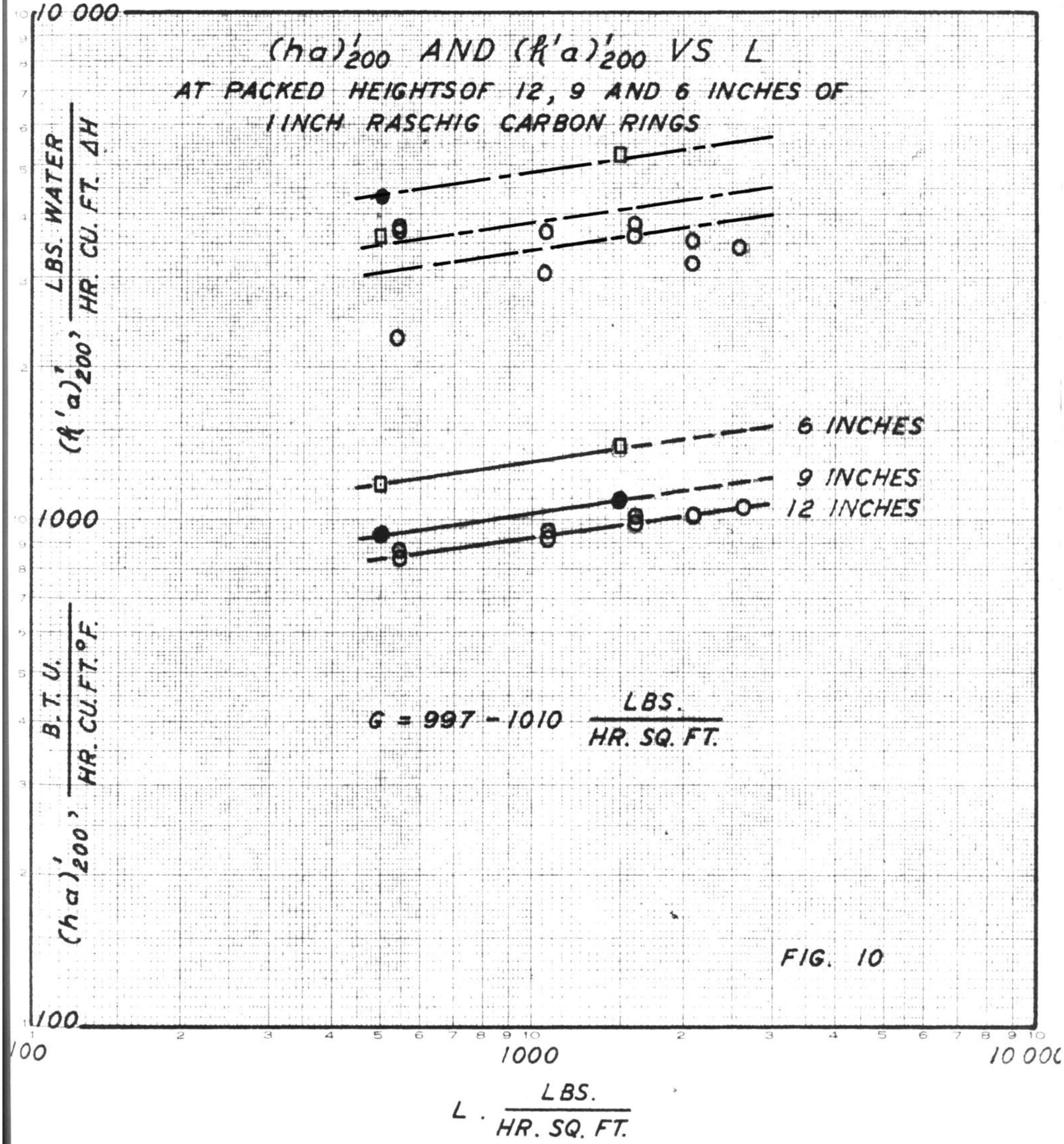

$(ha)'_{200}$ AND $(k'a)'_{200}$ VS L
AT PACKED HEIGHTS OF 12, 9 AND 6 INCHES OF
1 INCH RASCHIG CARBON RINGS
$(k'a)'_{200}$, $\frac{\text{LBS. WATER}}{\text{HR. CU. FT. } \Delta H}$
$(ha)'_{200}$, $\frac{\text{B.T.U.}}{\text{HR. CU. FT. °F.}}$
10 000
1000
100
6 INCHES
9 INCHES
12 INCHES
G = 997 – 1010 $\frac{\text{LBS.}}{\text{HR. SQ. FT.}}$
100
1000
10 000
L, $\frac{\text{LBS.}}{\text{HR. SQ. FT.}}$

FIG. 10

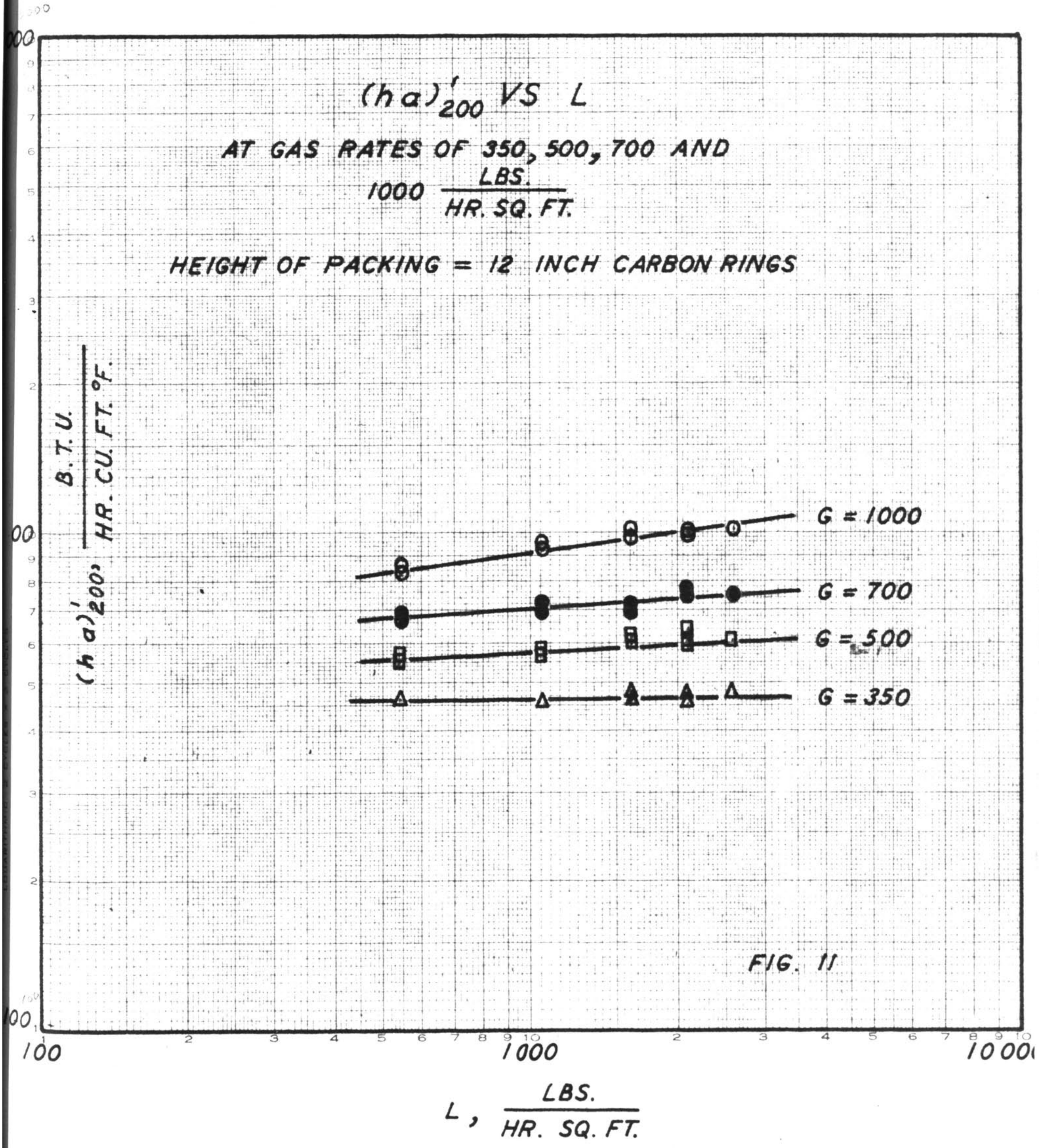
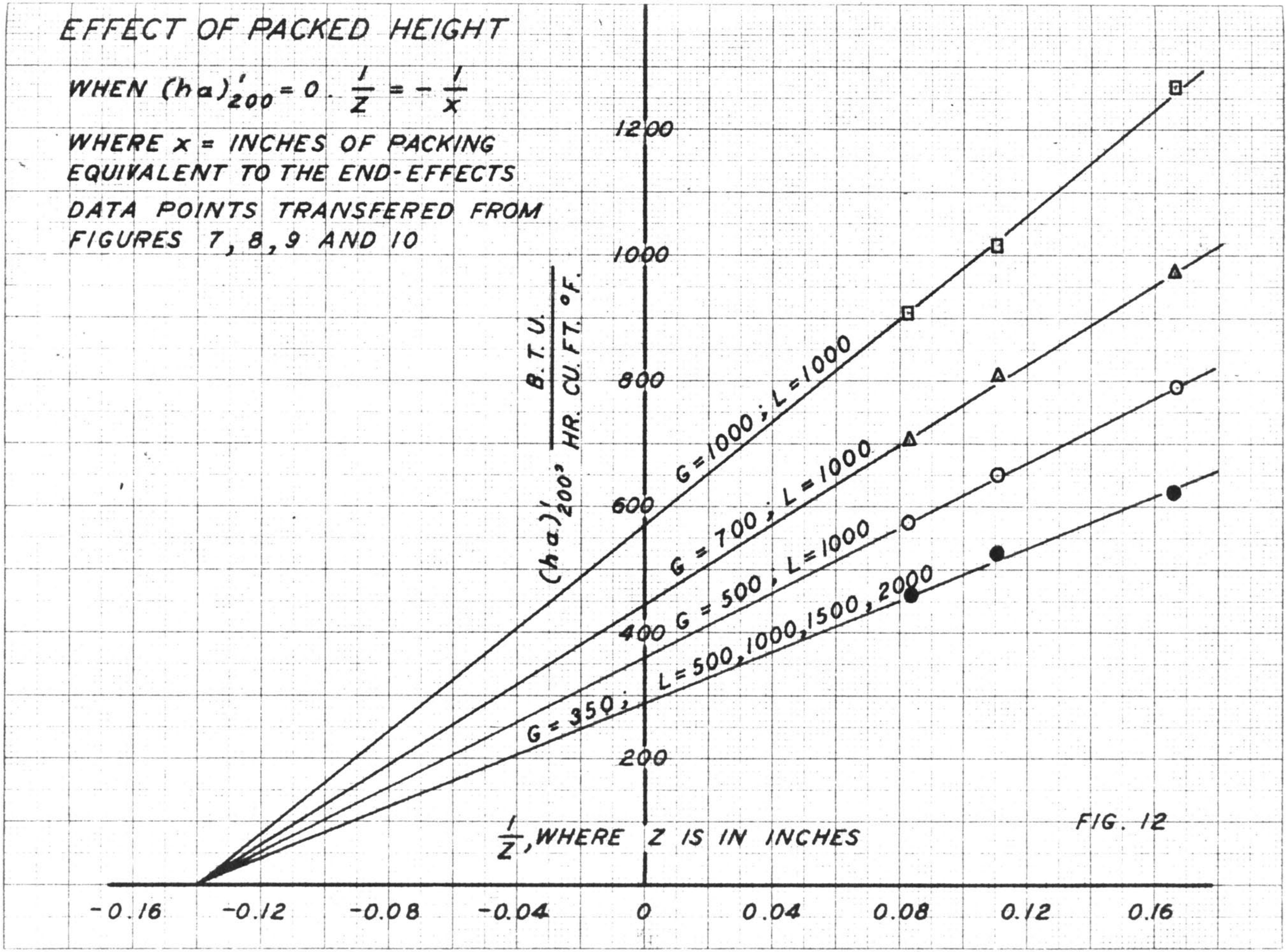

(h a)'200 VS L
AT GAS RATES OF 350, 500, 700 AND 1000 LBS. / HR. SQ. FT.
HEIGHT OF PACKING = 12 INCH CARBON RINGS
G = 1000
G = 700
G = 500
G = 350
(h a)'200, B.T.U. / HR. CU. FT. °F.
L, LBS. / HR. SQ. FT.
100
1000
10000

FIG. 11



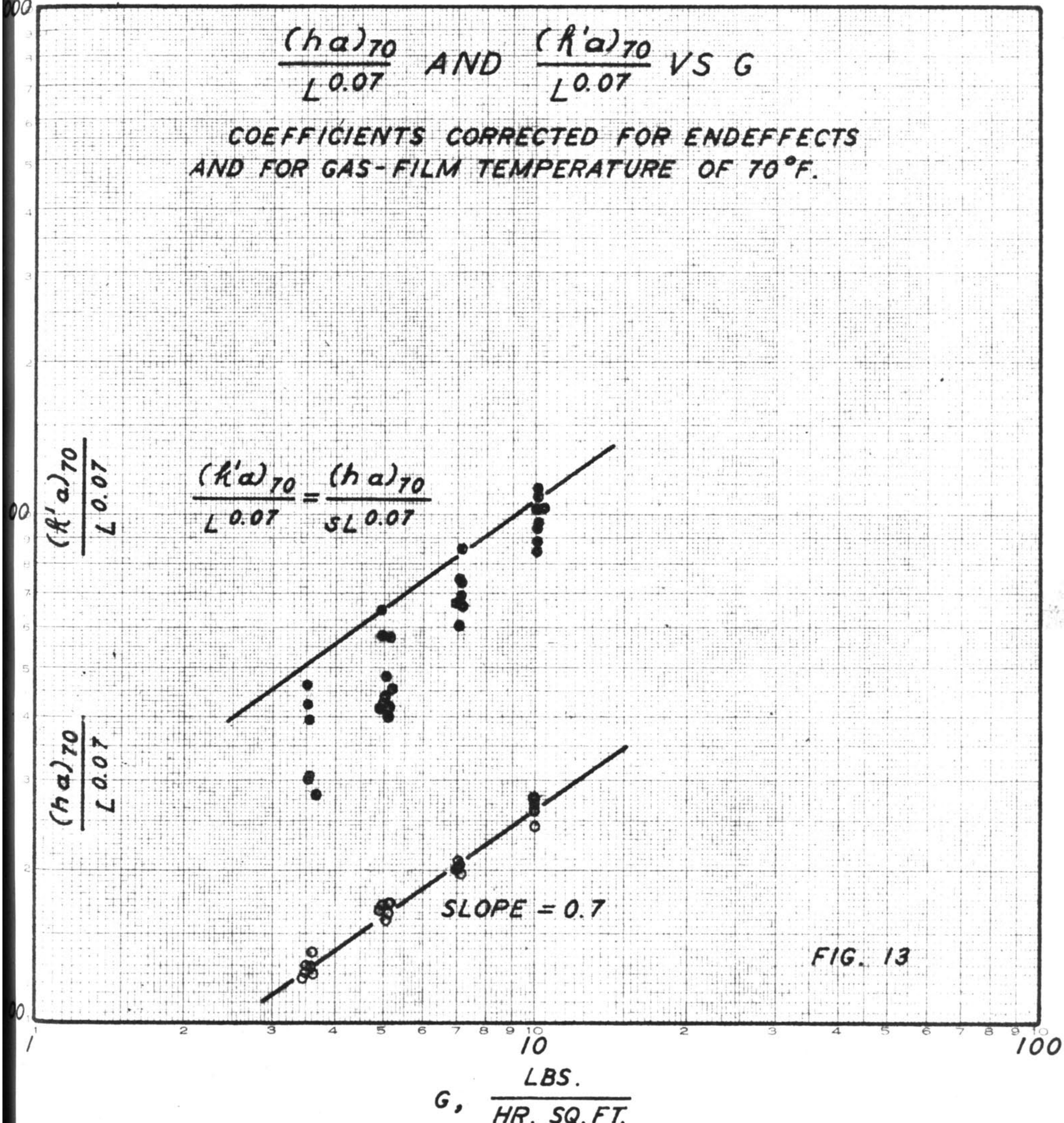
EFFECT OF PACKED HEIGHT

WHEN $(ha)'_{200} = 0$. $\frac{1}{Z} = -\frac{1}{x}$

WHERE x = INCHES OF PACKING EQUIVALENT TO THE END-EFFECTS

DATA POINTS TRANSFERED FROM FIGURES 7, 8, 9 AND 10

FIG. 12



$\frac{(ha)_{70}}{L^{0.07}}$ AND $\frac{(k'a)_{70}}{L^{0.07}}$ VS G
COEFFICIENTS CORRECTED FOR ENDEFFECTS AND FOR GAS-FILM TEMPERATURE OF 70°F.
$\frac{(k'a)_{70}}{L^{0.07}} = \frac{(ha)_{70}}{sL^{0.07}}$
$\frac{(k'a)_{70}}{L^{0.07}}$
$\frac{(ha)_{70}}{L^{0.07}}$
SLOPE = 0.7
1
10
100
G, $\frac{LBS.}{HR.\ SQ.FT.}$

FIG. 13

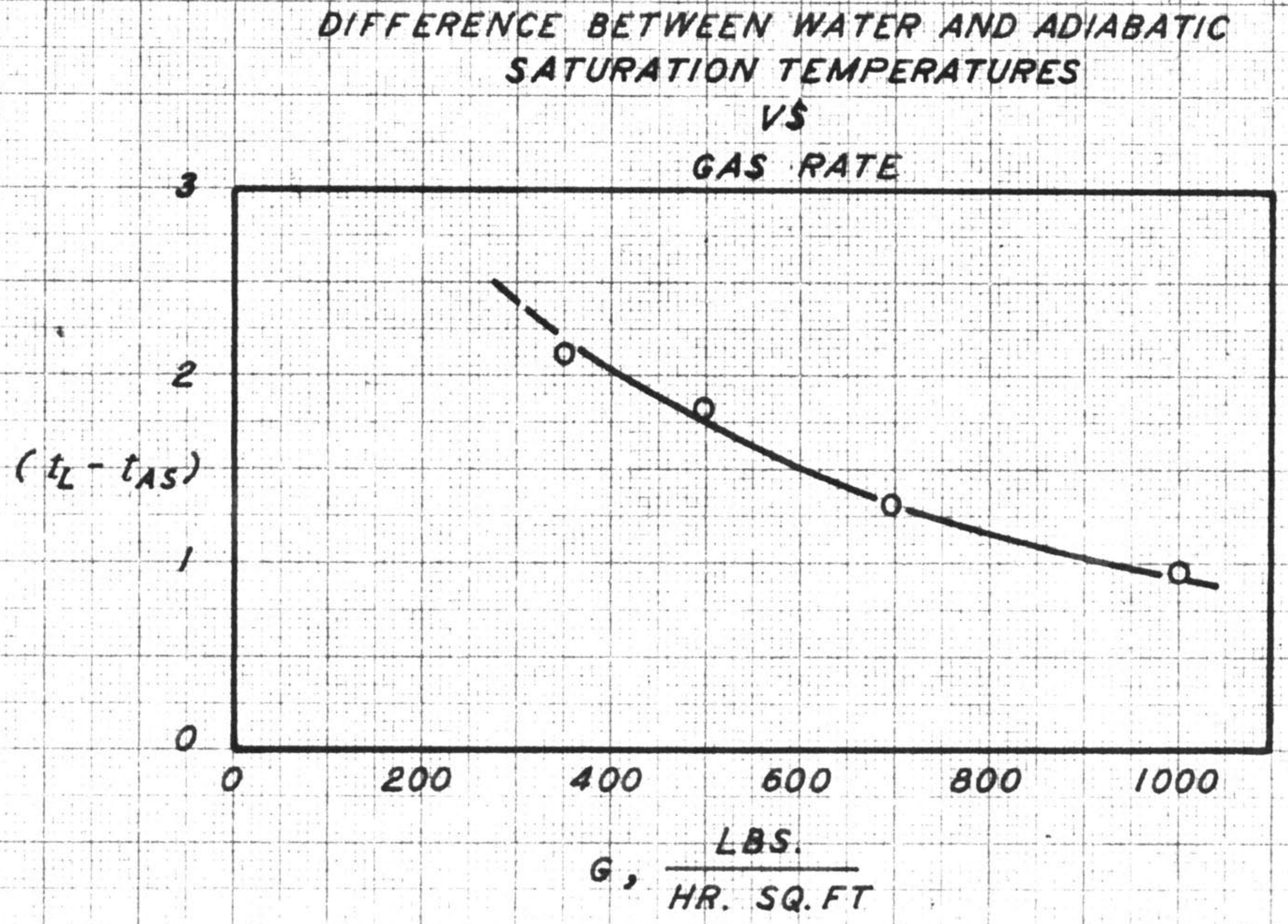

DIFFERENCE BETWEEN WATER AND ADIABATIC SATURATION TEMPERATURES VS GAS RATE

FIG. 14

REVISED COEFFICIENT $(k'a)''_{70}$ VS GAS RATE

BASED ON TEMPERATURE OF ADIABATIC SATURATION

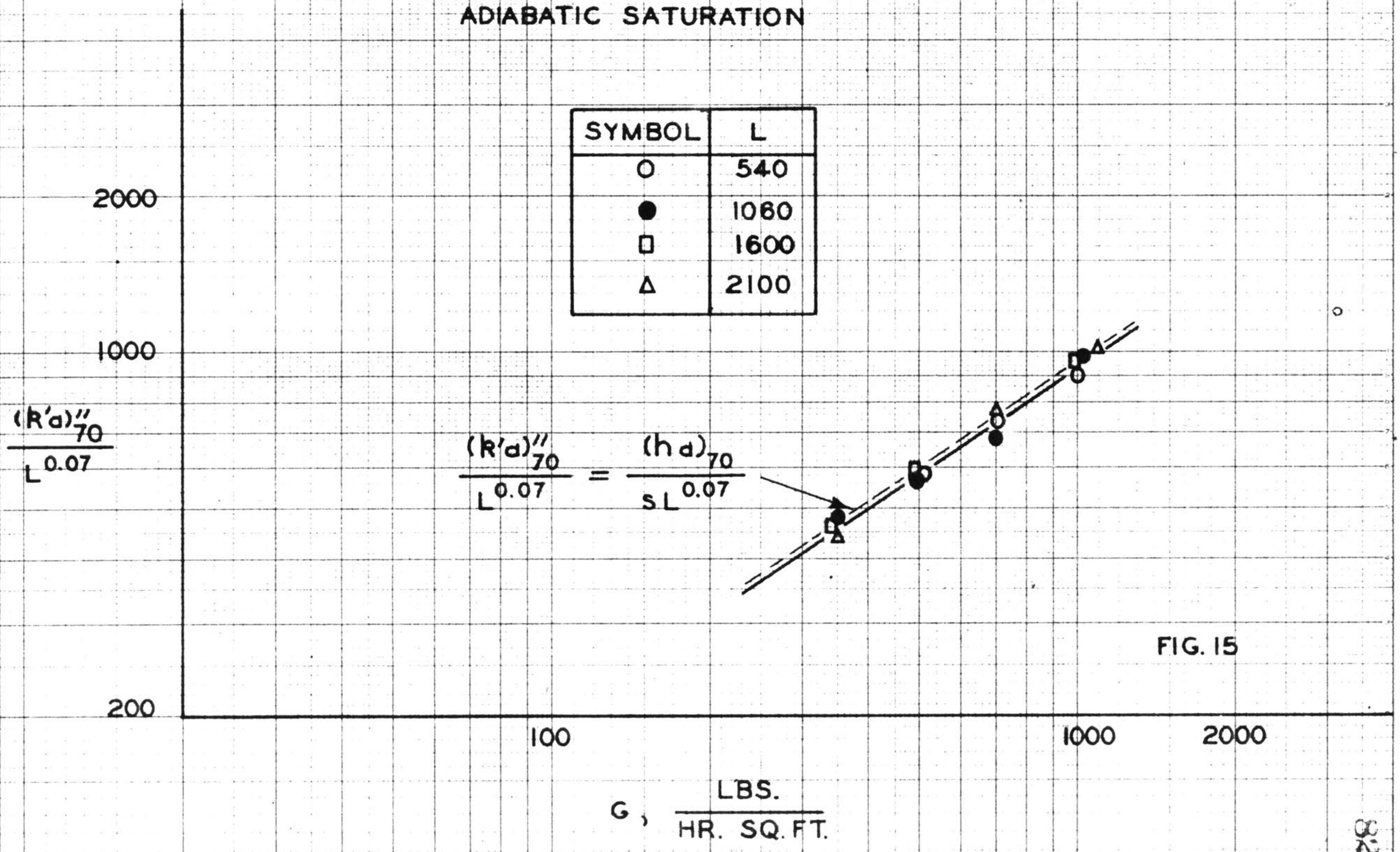


FIG. 15

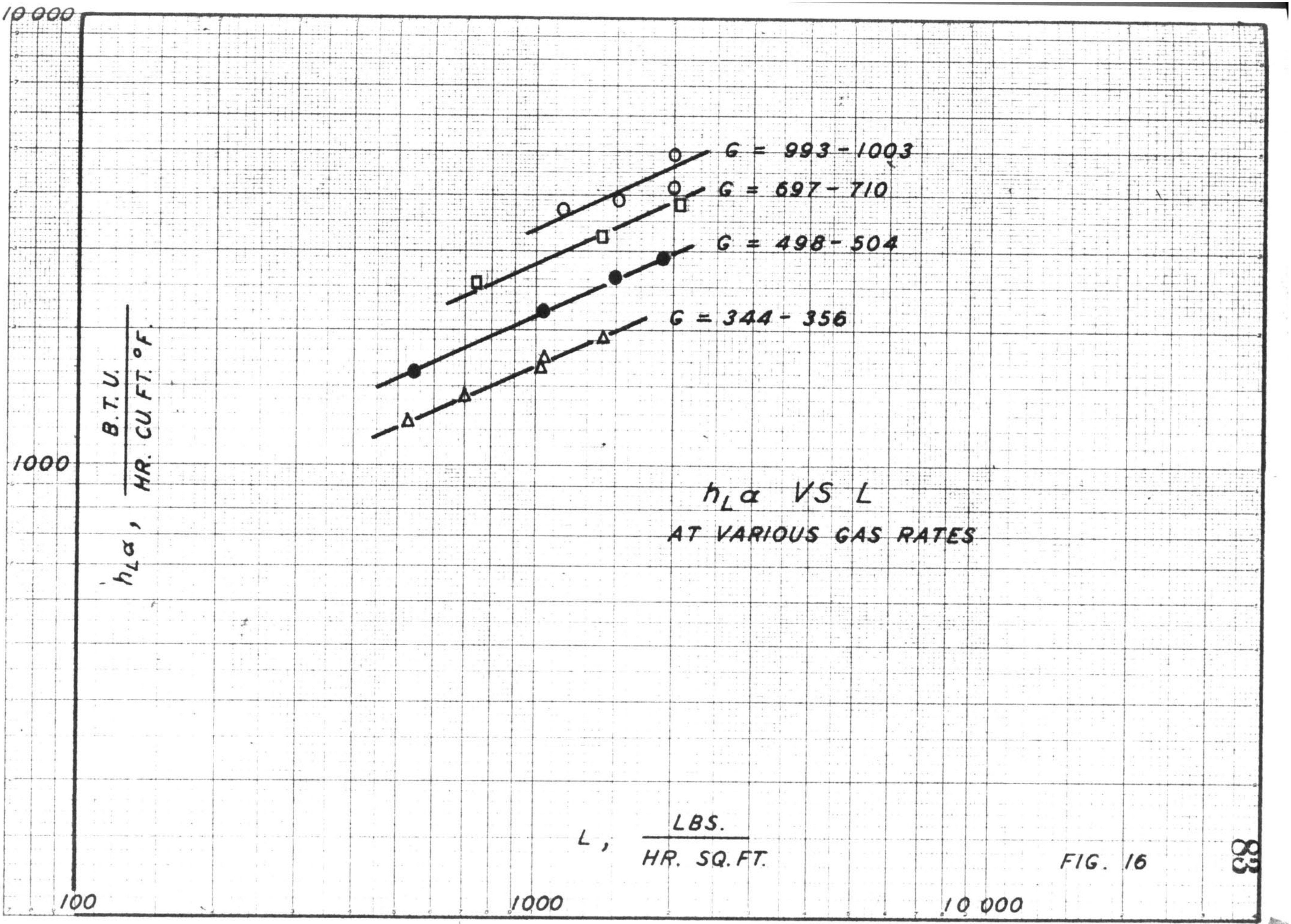

10 000
1000
100
1000
10 000
$h_L a$, B.T.U. / HR. CU. FT. °F.
L, LBS. / HR. SQ. FT.
G = 993 - 1003
G = 697 - 710
G = 498 - 504
G = 344 - 356
$h_L a$ VS L
AT VARIOUS GAS RATES

FIG. 16

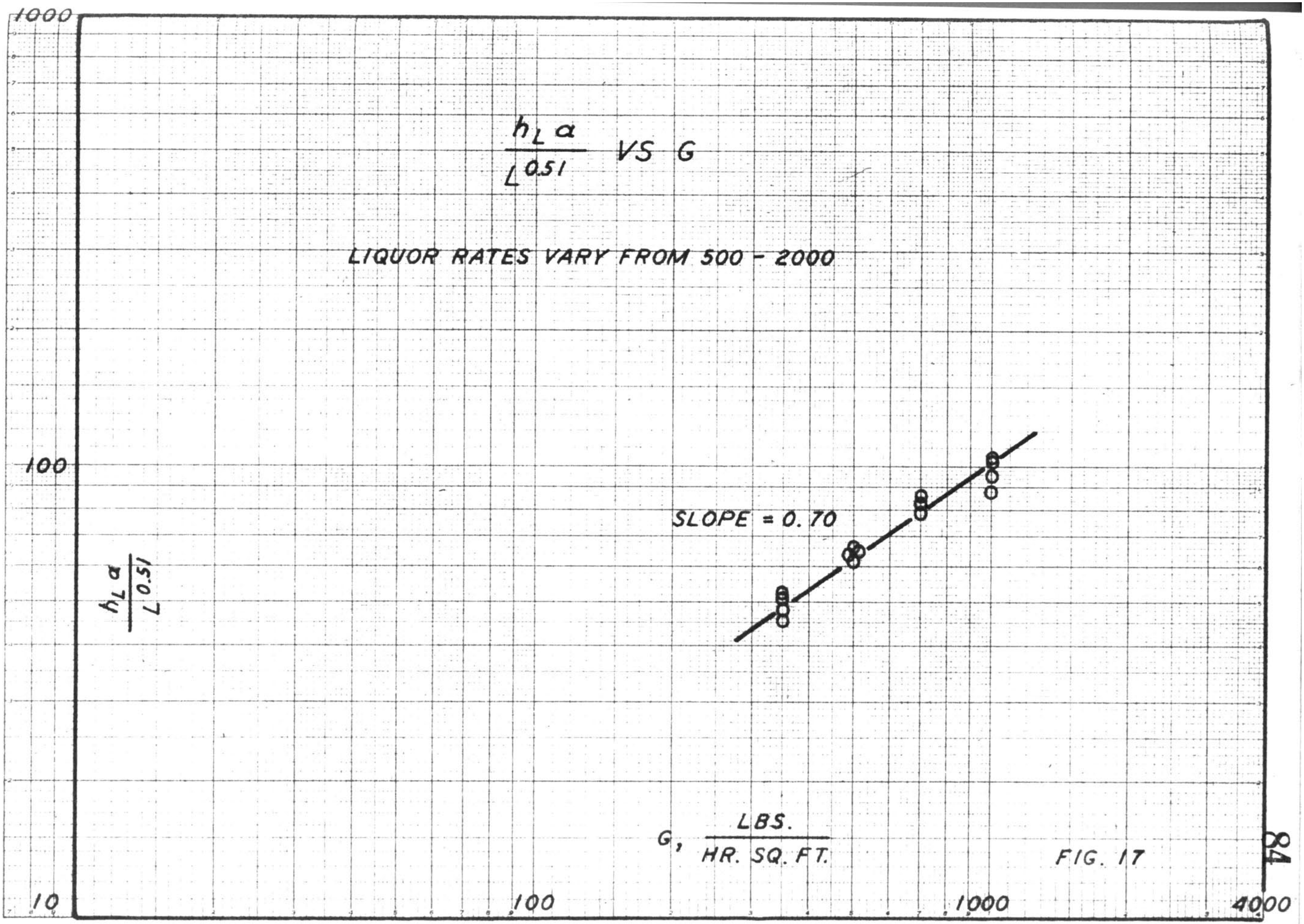

$\frac{h_L a}{L^{0.51}}$ VS G
LIQUOR RATES VARY FROM 500 - 2000
SLOPE = 0.70
$\frac{h_L a}{L^{0.51}}$
G, $\frac{\text{LBS.}}{\text{HR. SQ. FT.}}$
1000
100
10
10
100
1000
4000

FIG. 17

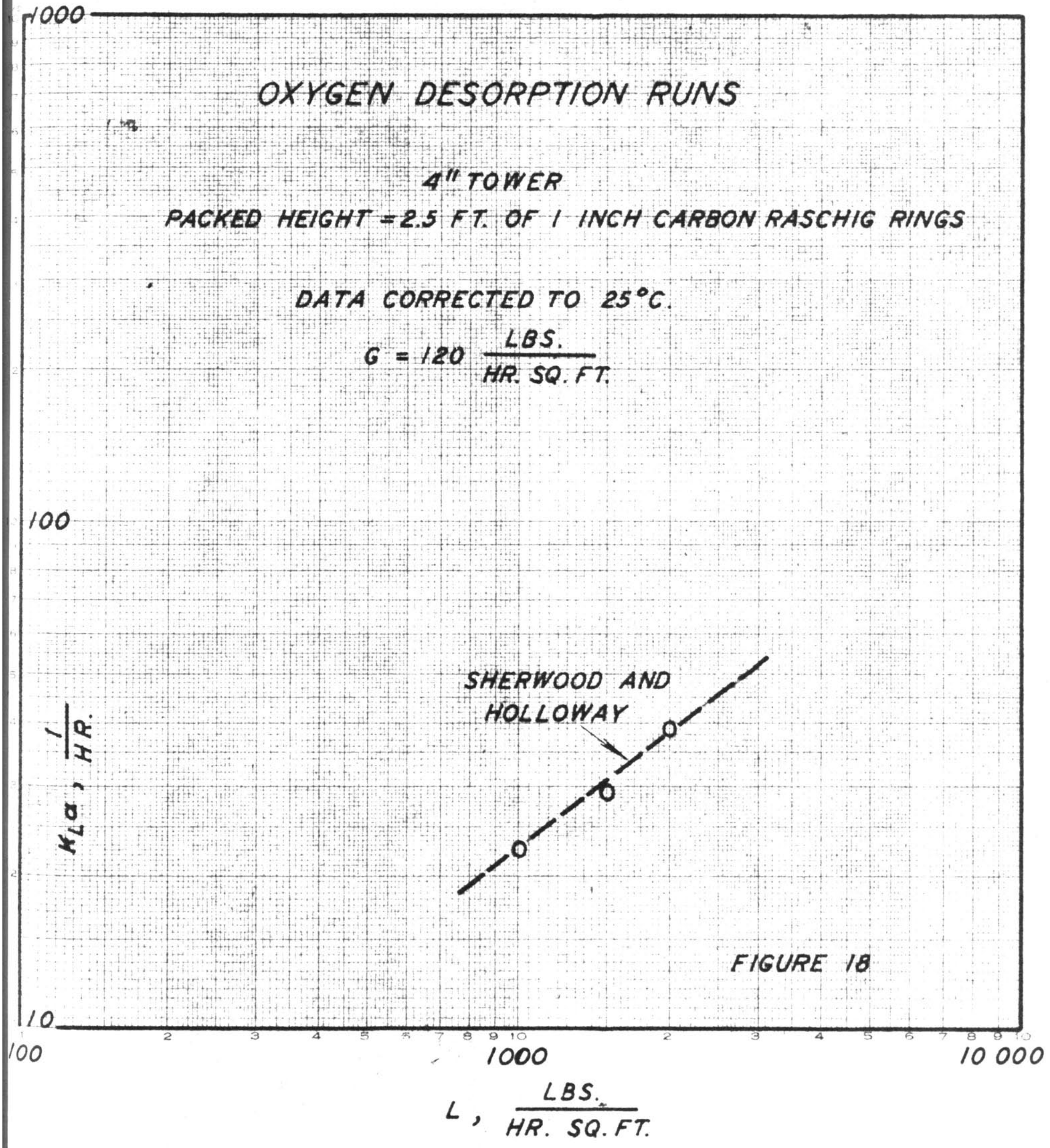

OXYGEN DESORPTION RUNS
4" TOWER
PACKED HEIGHT = 2.5 FT. OF 1 INCH CARBON RASCHIG RINGS
DATA CORRECTED TO 25°C.
G = 120 LBS. / HR. SQ. FT.
SHERWOOD AND HOLLOWAY
1000
100
10
100
1000
10 000
2
3
4
5
6
7
8
9
K_La, 1/HR.
L, LBS. / HR. SQ. FT.

FIGURE 18

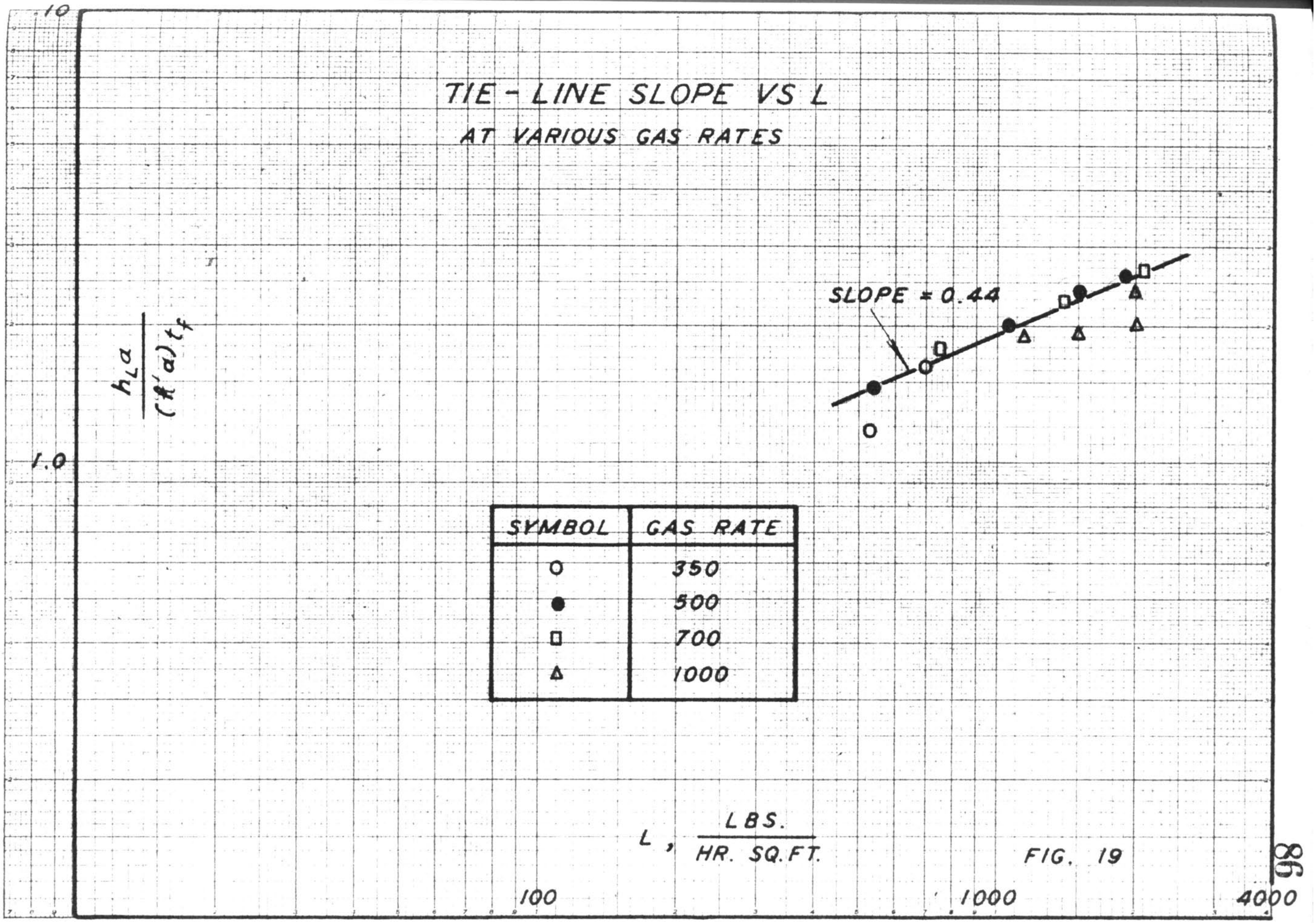
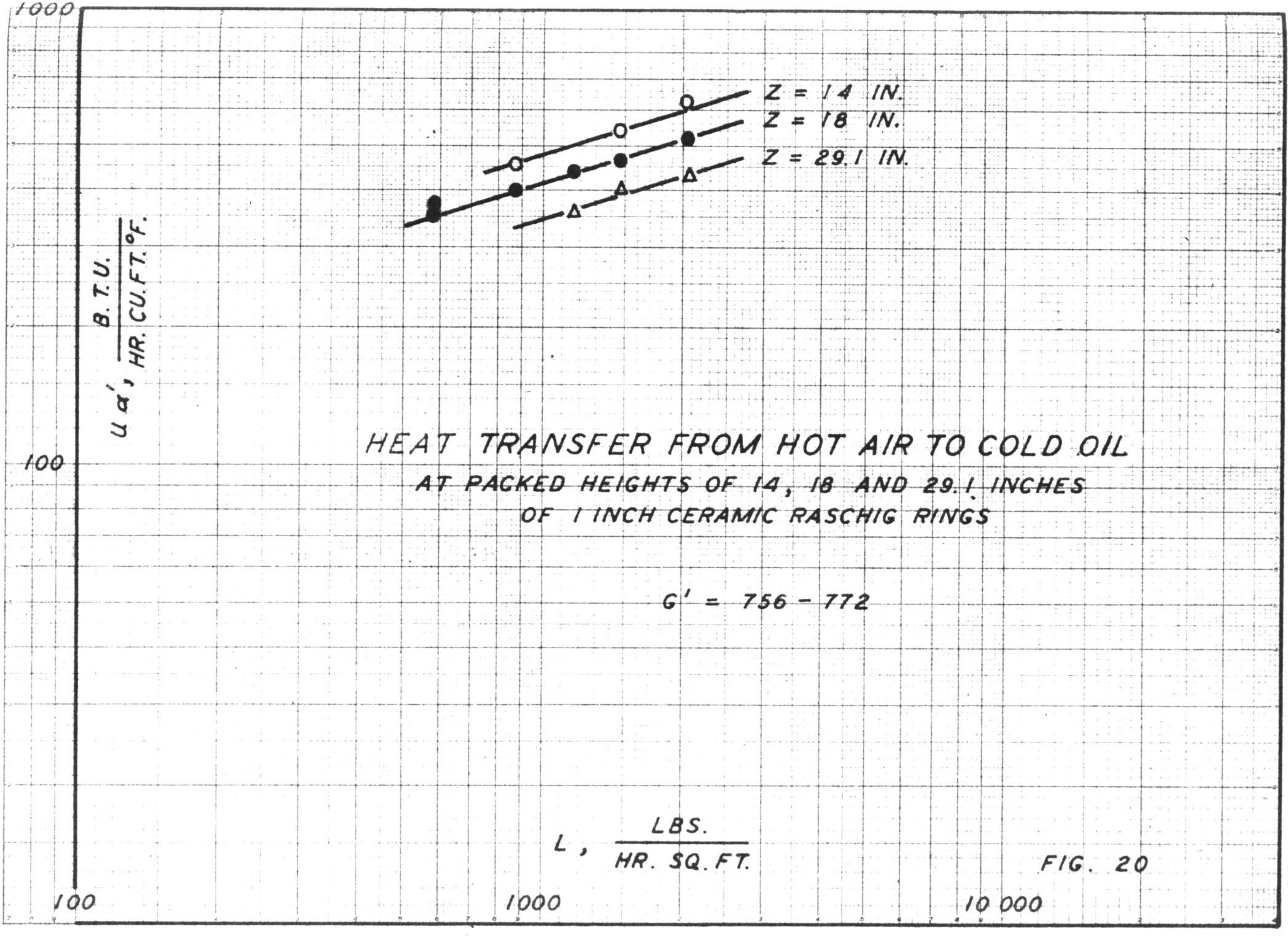
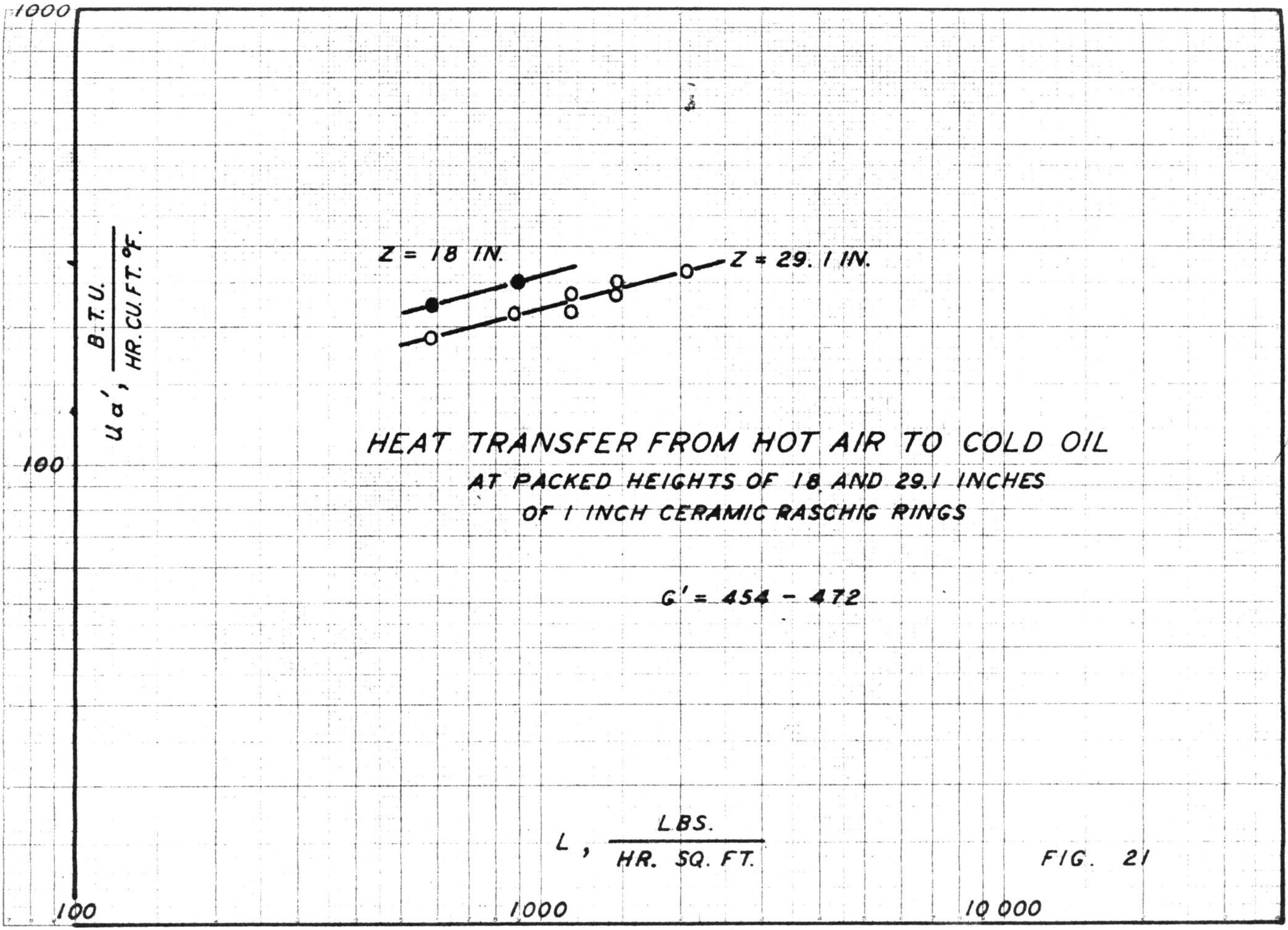

TIE - LINE SLOPE VS L
AT VARIOUS GAS RATES

FIG. 19



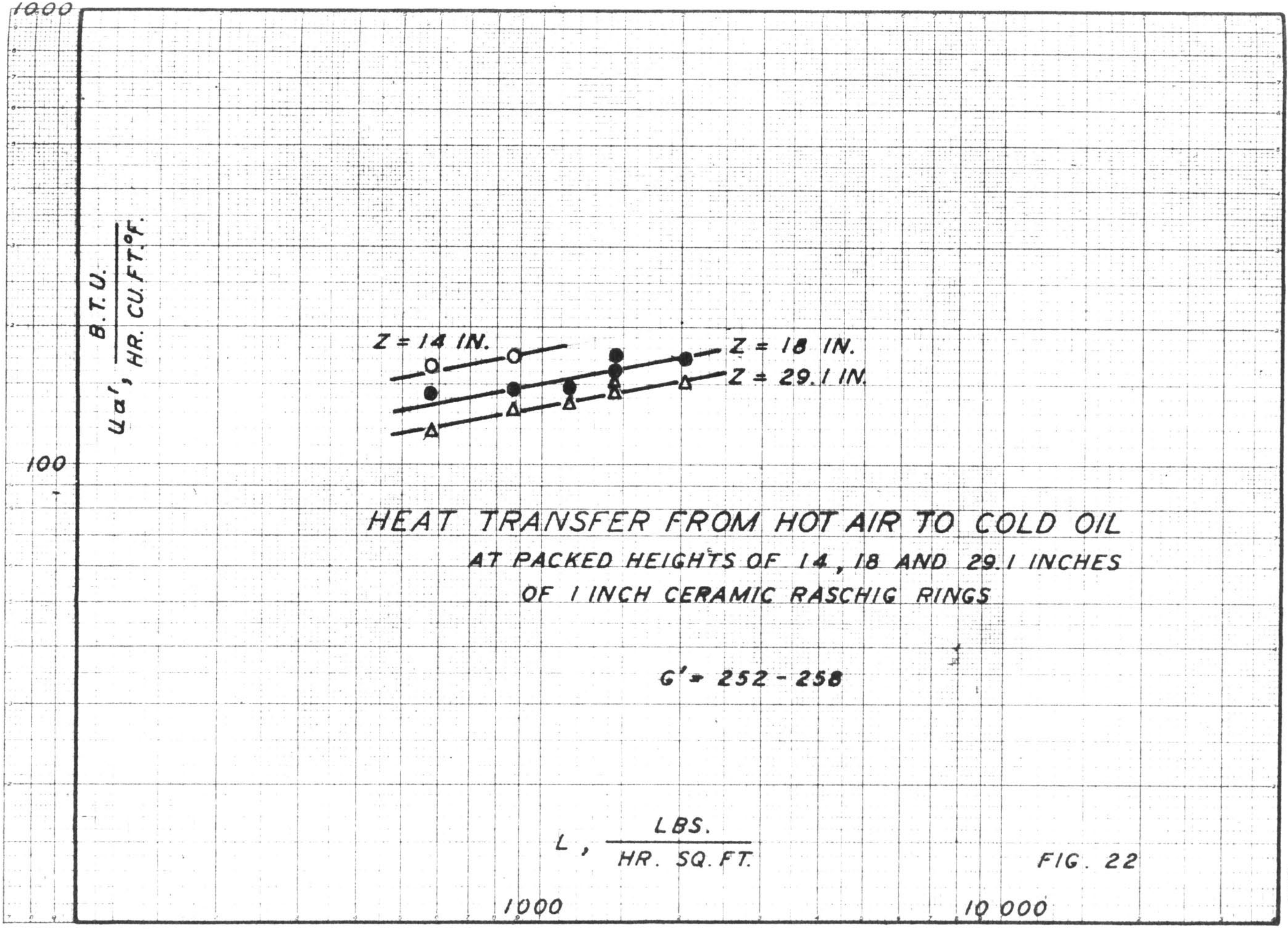
HEAT TRANSFER FROM HOT AIR TO COLD OIL
AT PACKED HEIGHTS OF 14, 18 AND 29.1 INCHES
OF 1 INCH CERAMIC RASCHIG RINGS

FIG. 20



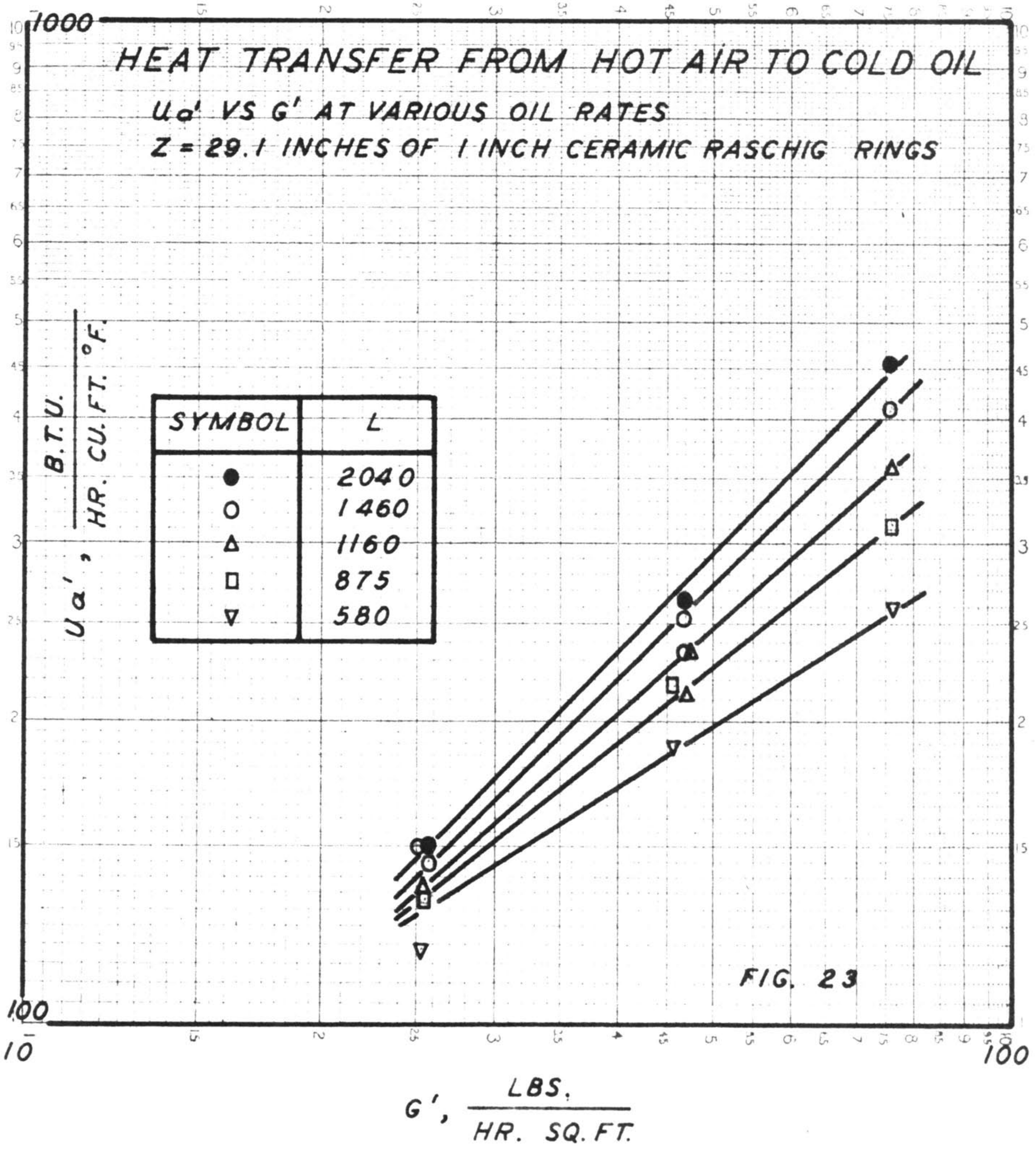
HEAT TRANSFER FROM HOT AIR TO COLD OIL
AT PACKED HEIGHTS OF 18 AND 29.1 INCHES
OF 1 INCH CERAMIC RASCHIG RINGS

FIG. 21



HEAT TRANSFER FROM HOT AIR TO COLD OIL
AT PACKED HEIGHTS OF 14, 18 AND 29.1 INCHES
OF 1 INCH CERAMIC RASCHIG RINGS

FIG. 22


HEAT TRANSFER FROM HOT AIR TO COLD OIL

Ua' VS G' AT VARIOUS OIL RATES

Z = 29.1 INCHES OF 1 INCH CERAMIC RASCHIG RINGS

FIG. 23

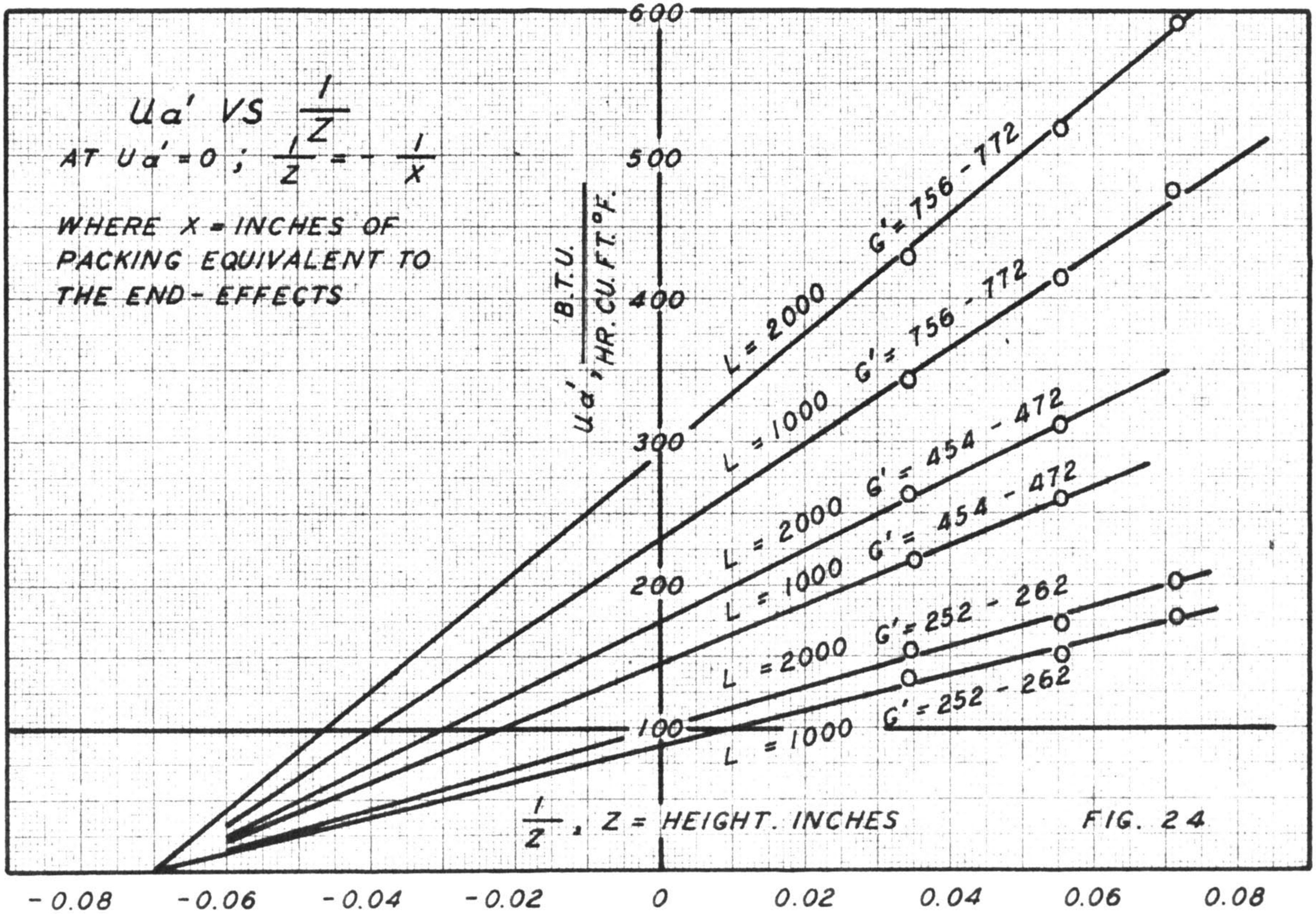
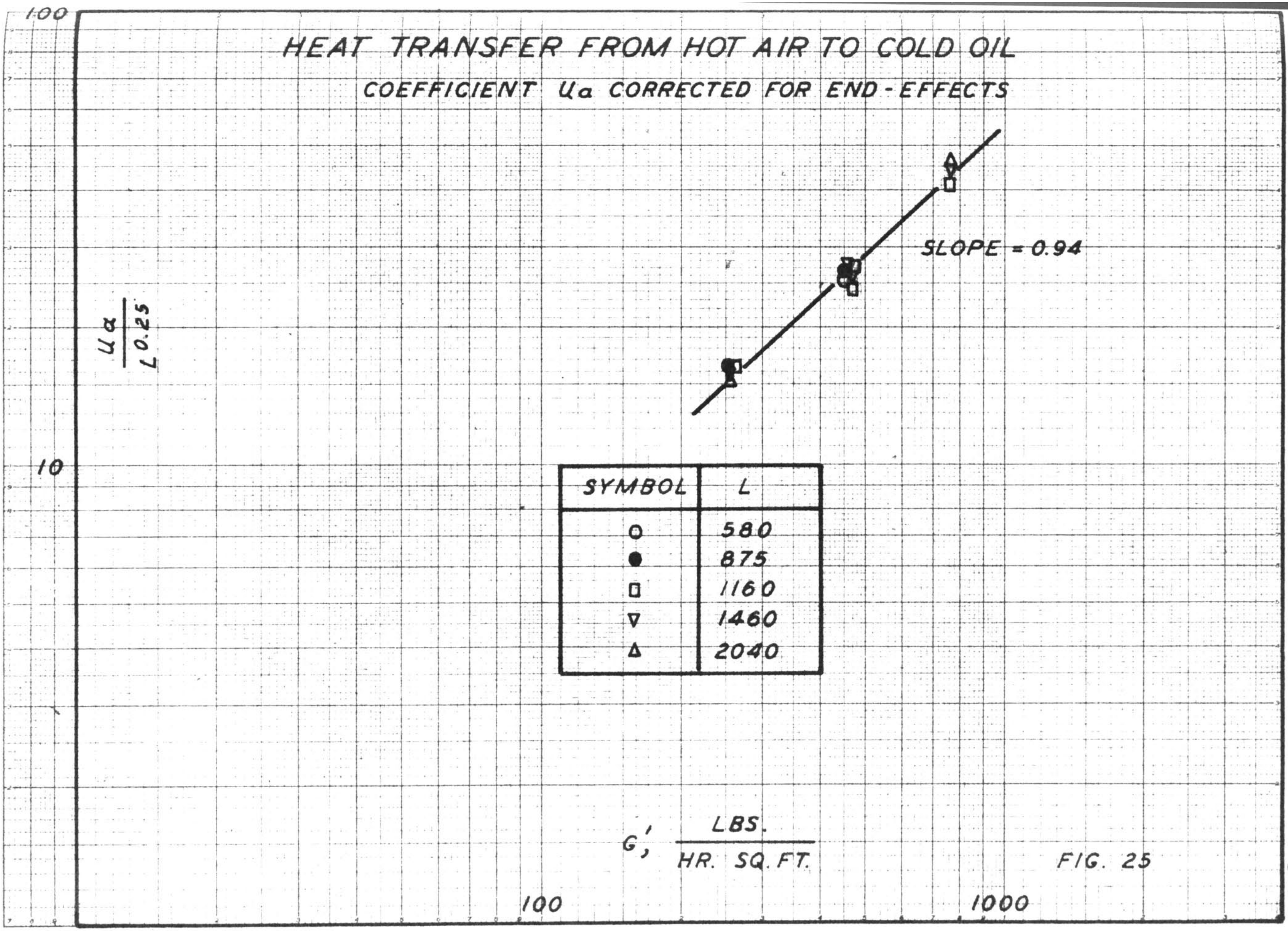

Ua' VS 1/Z
AT Ua' = 0 ; 1/Z = - 1/X
WHERE X = INCHES OF PACKING EQUIVALENT TO THE END - EFFECTS
Ua', B.T.U./HR. CU. FT. °F.
1/Z , Z = HEIGHT. INCHES
L = 2000 G' = 756 - 772
L = 1000 G' = 756 - 772
L = 2000 G' = 454 - 472
L = 1000 G' = 454 - 472
L = 2000 G' = 252 - 262
L = 1000 G' = 252 - 262
600
500
400
300
200
100
- 0.08
- 0.06
- 0.04
- 0.02
0
0.02
0.04
0.06
0.08

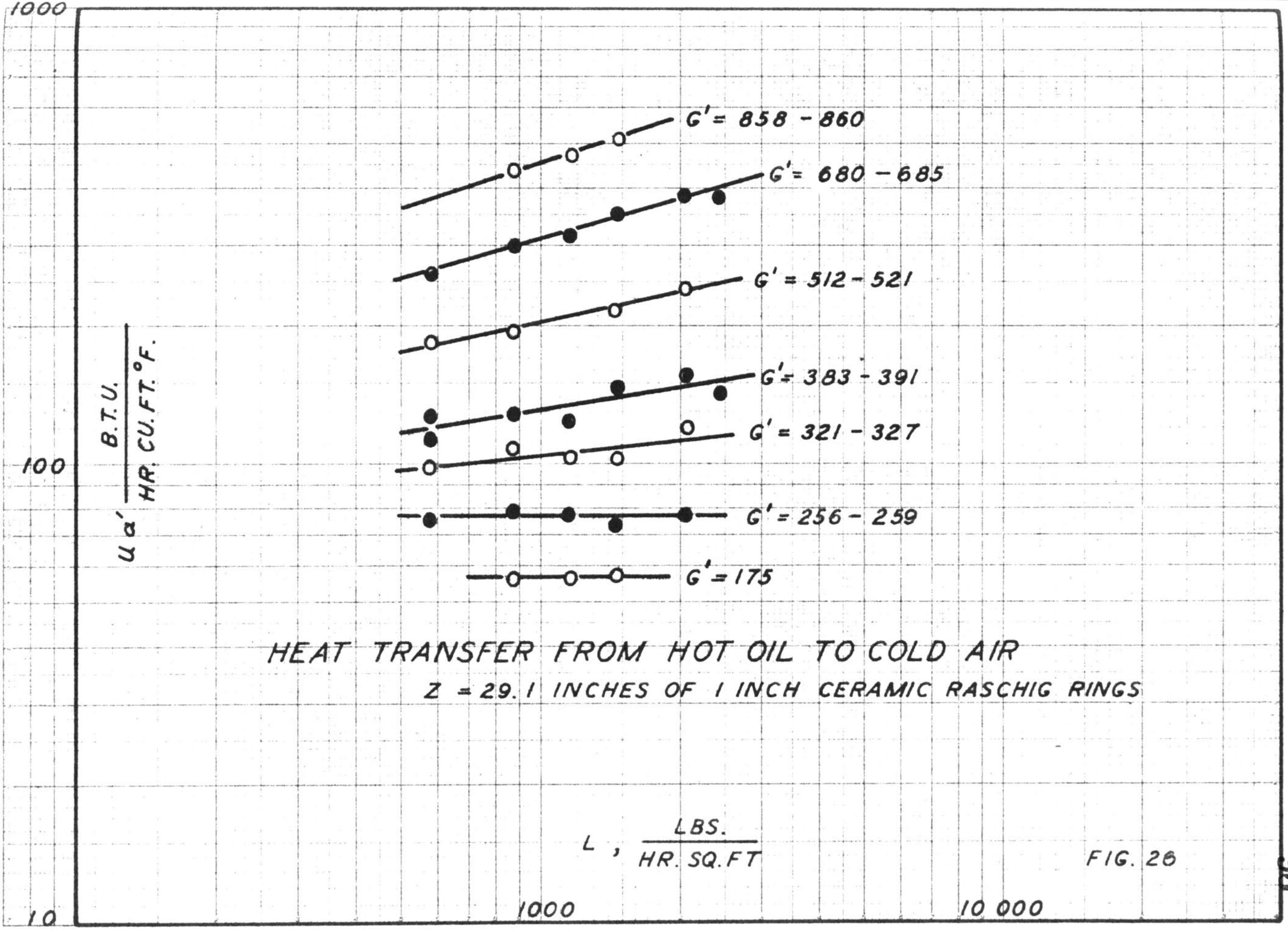
FIG. 24



HEAT TRANSFER FROM HOT AIR TO COLD OIL

COEFFICIENT Ua CORRECTED FOR END-EFFECTS

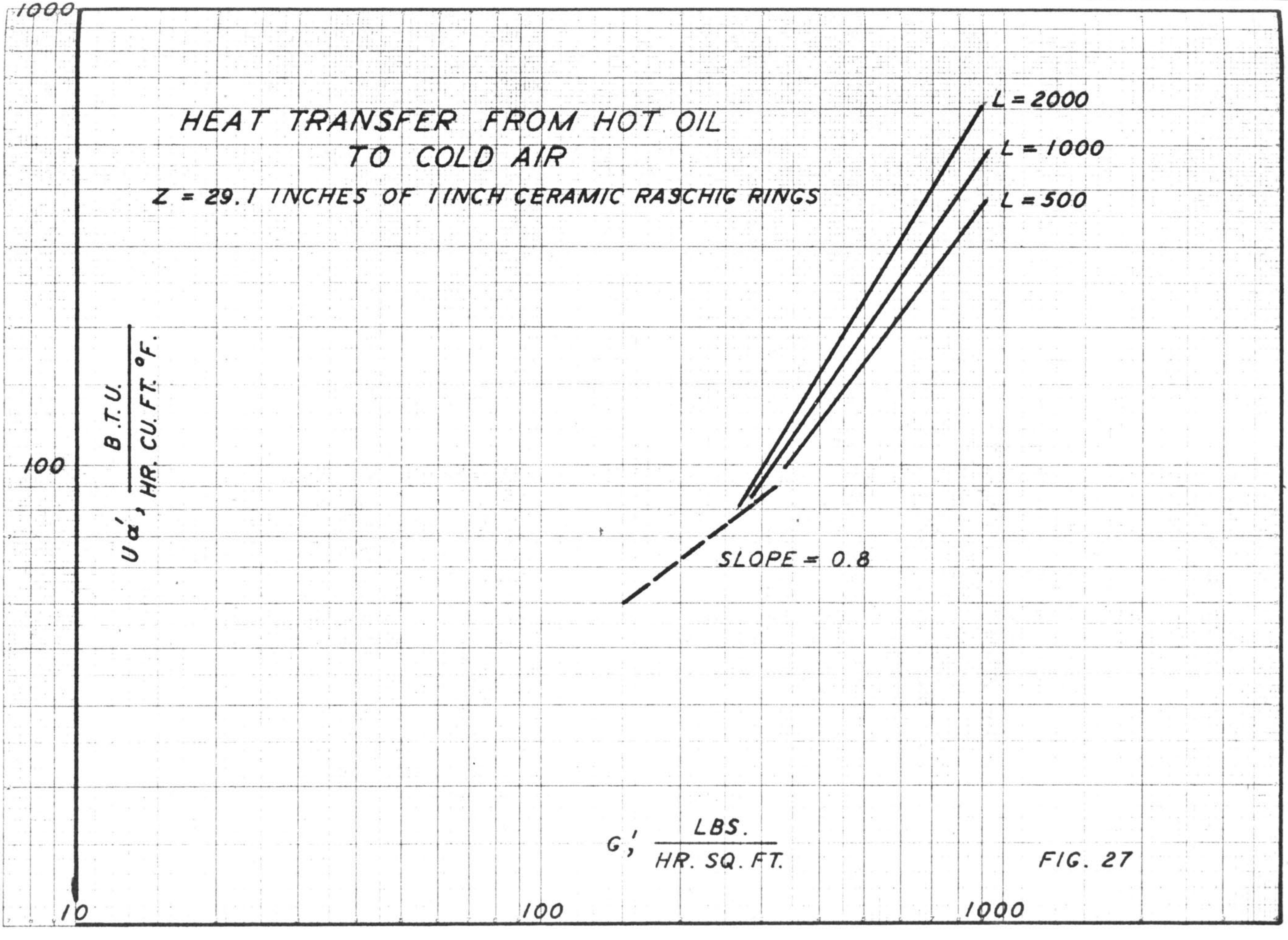
FIG. 25



HEAT TRANSFER FROM HOT OIL TO COLD AIR

Z = 29.1 INCHES OF 1 INCH CERAMIC RASCHIG RINGS

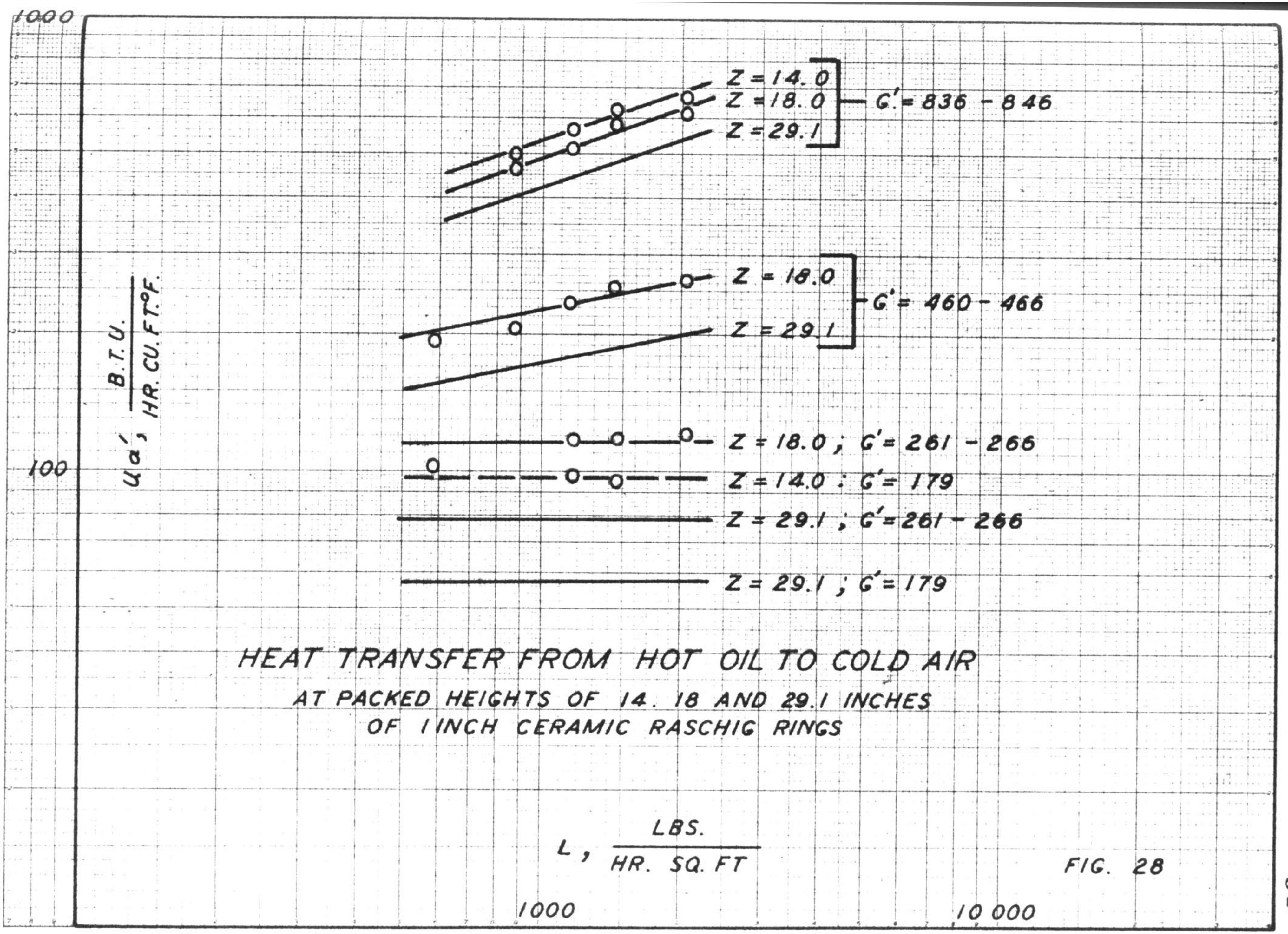
FIG. 26



HEAT TRANSFER FROM HOT OIL TO COLD AIR

Z = 29.1 INCHES OF 1 INCH CERAMIC RASCHIG RINGS

FIG. 27



HEAT TRANSFER FROM HOT OIL TO COLD AIR
AT PACKED HEIGHTS OF 14. 18 AND 29.1 INCHES
OF 1 INCH CERAMIC RASCHIG RINGS

FIG. 28

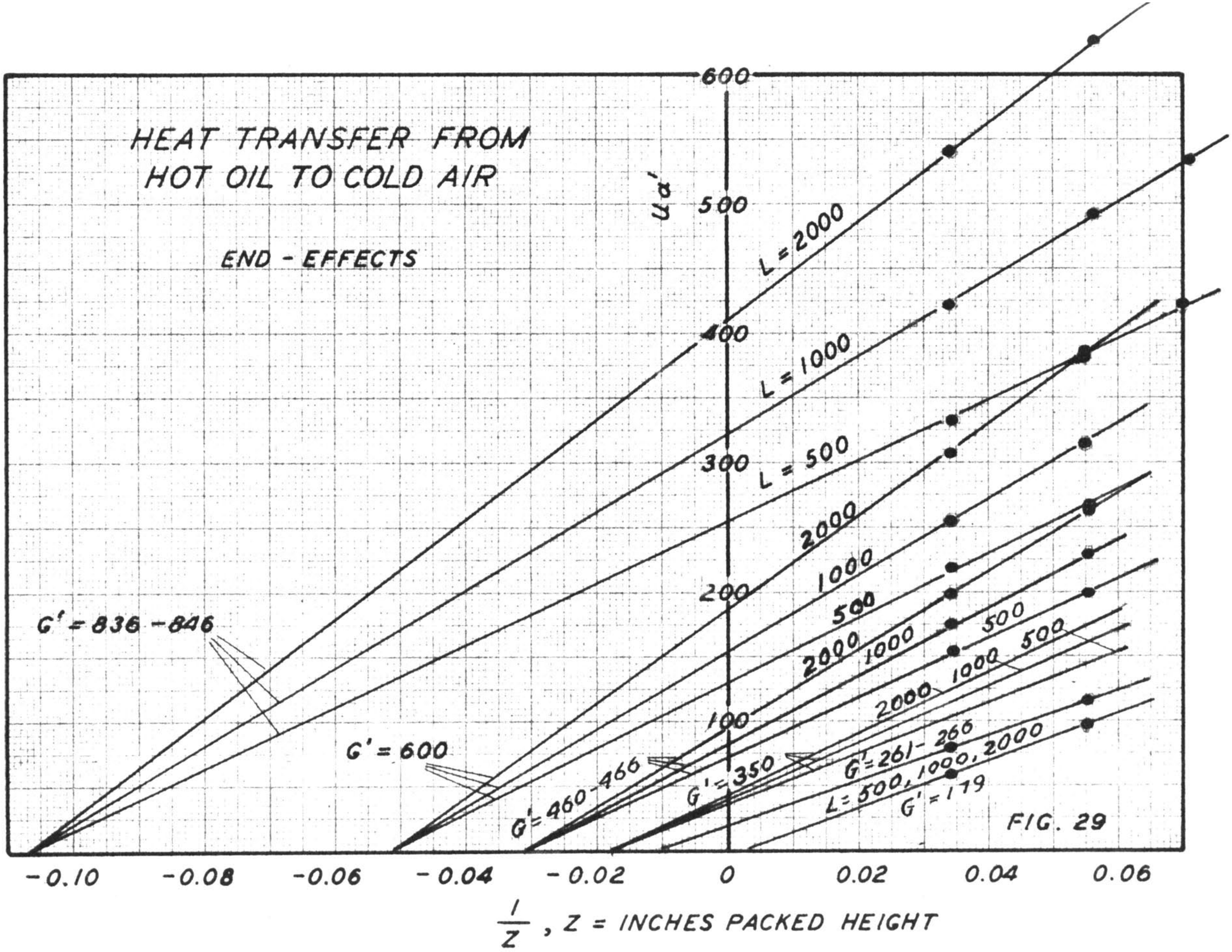
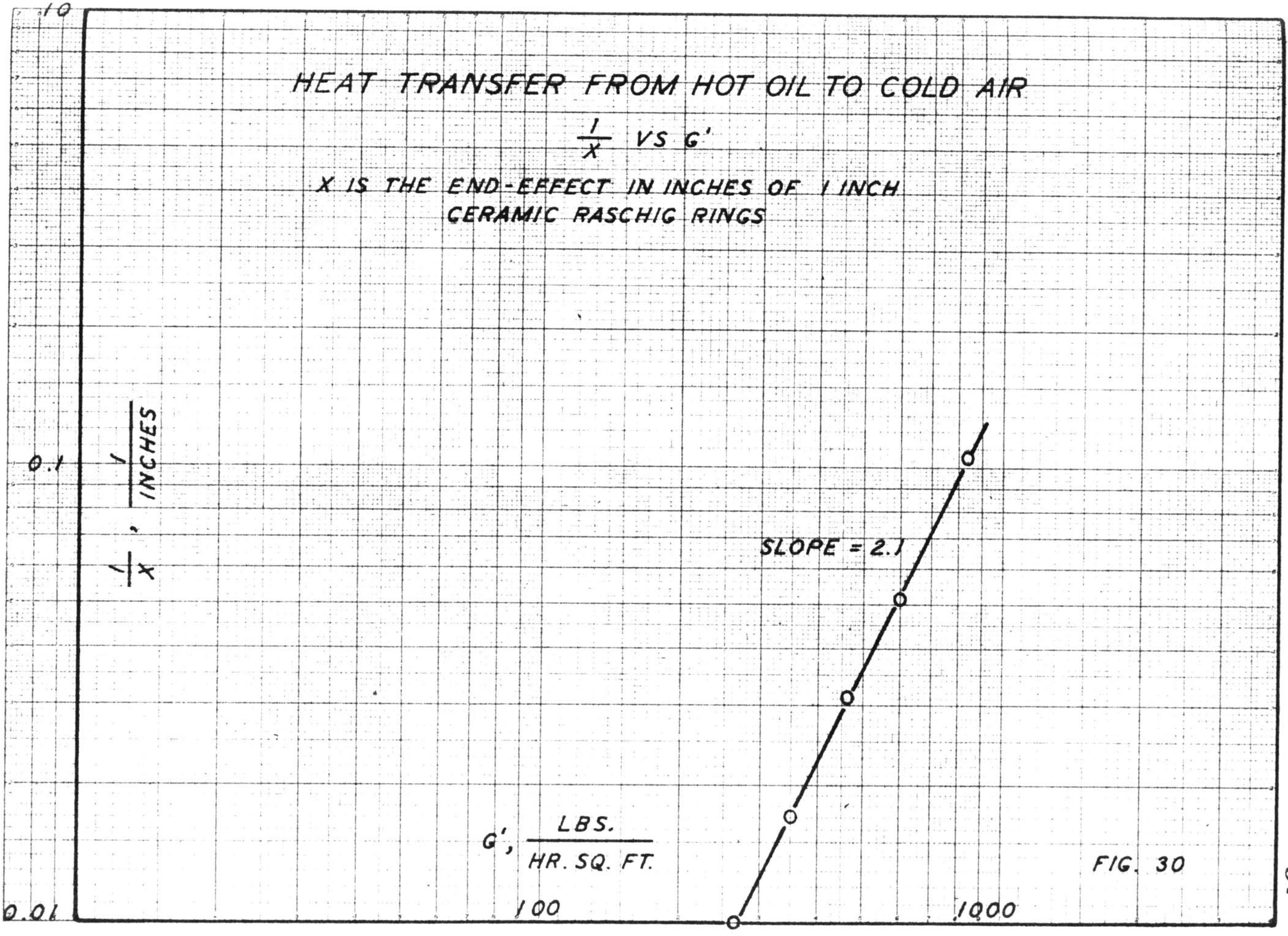

HEAT TRANSFER FROM
HOT OIL TO COLD AIR
END - EFFECTS
Ua'
600
500
400
300
200
100
L = 2000
L = 1000
L = 500
2000
1000
500
2000
1000
500
2000
1000
500
G' = 836 - 846
G' = 600
G' = 460 - 466
G' = 350
G' = 261 - 266
L = 500, 1000, 2000
G' = 179
-0.10
-0.08
-0.06
-0.04
-0.02
0
0.02
0.04
0.06
1/Z , Z = INCHES PACKED HEIGHT

FIG. 29

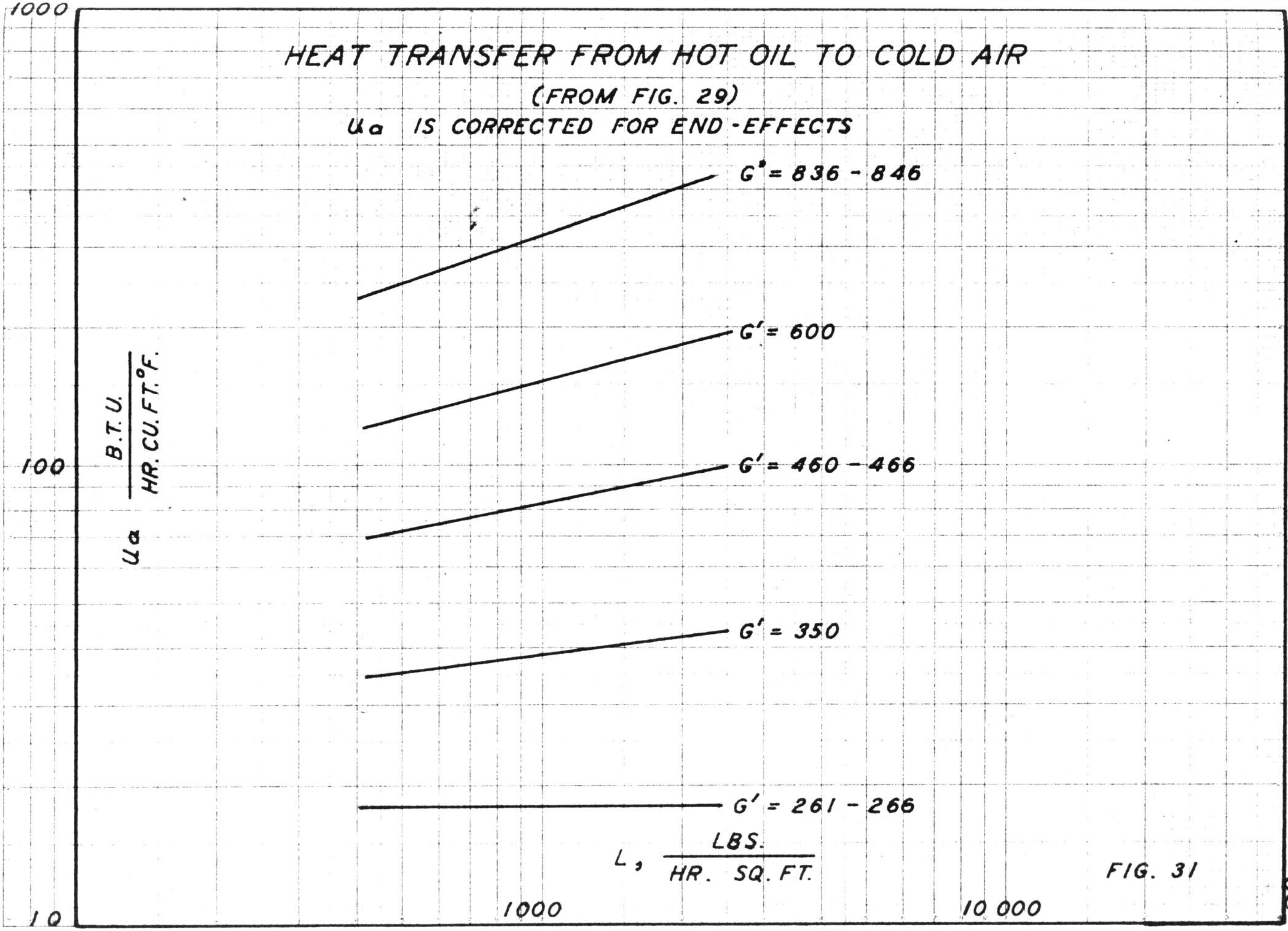


HEAT TRANSFER FROM HOT OIL TO COLD AIR

$\frac{1}{X}$ VS G'

X IS THE END-EFFECT IN INCHES OF 1 INCH CERAMIC RASCHIG RINGS

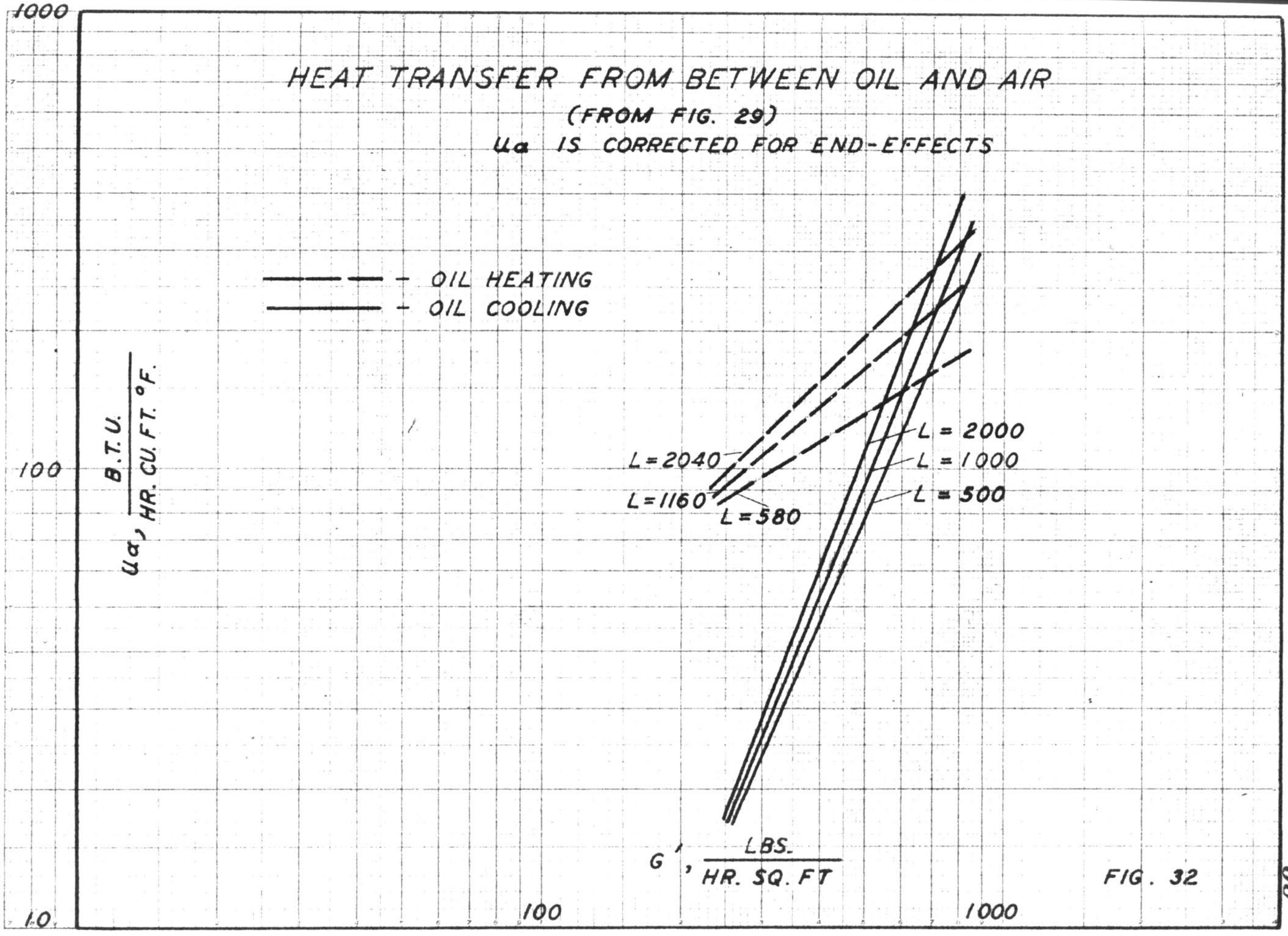
FIG. 30


HEAT TRANSFER FROM HOT OIL TO COLD AIR

(FROM FIG. 29)

Ua IS CORRECTED FOR END-EFFECTS

FIG. 31



HEAT TRANSFER FROM BETWEEN OIL AND AIR
(FROM FIG. 29)
Ua IS CORRECTED FOR END-EFFECTS

FIG. 32

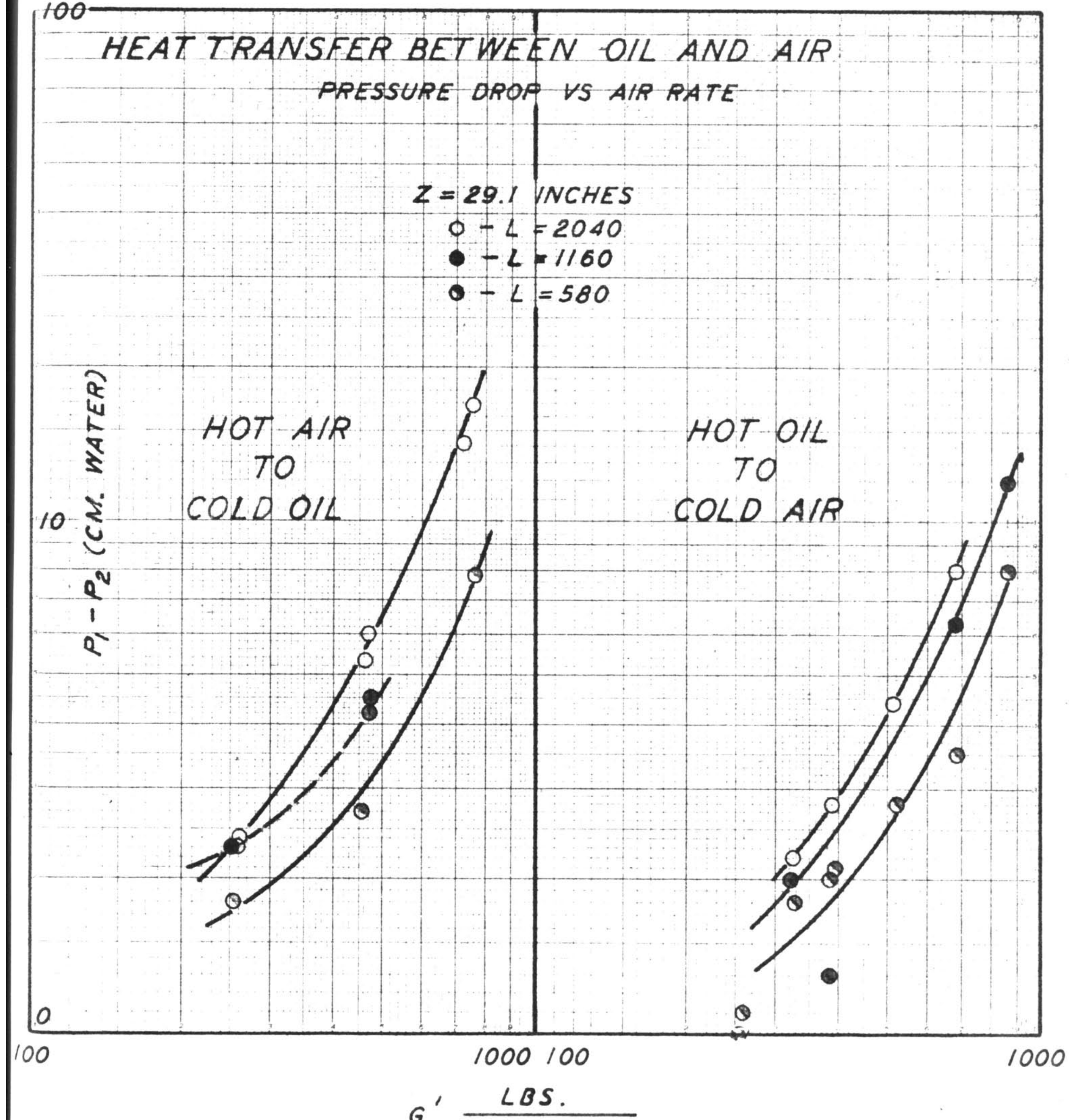

HEAT TRANSFER BETWEEN OIL AND AIR
PRESSURE DROP VS AIR RATE
Z = 29.1 INCHES
○ - L = 2040
● - L = 1160
◐ - L = 580
HOT AIR TO COLD OIL
HOT OIL TO COLD AIR
$P_1 - P_2$ (CM. WATER)
100
10
0
100
1000 100
1000
G', LBS. / HR. SQ. FT.

FIG. 33

V. DISCUSSION OF RESULTS

Air-Water Tower

An error analysis of the data for the adiabatic humidifying runs indicated that heat balances should close to within 1% of the heat transferred across the gas film, and this precision was obtained in substantially all of the runs. It was shown that the heat transfer coefficients (ha) should be reliable to within ten per cent. However, the coefficients of mass transfer are subject to deviations of ± 30 per cent due to errors in evaluating the humidity driving force at the top of the packing. An error of 0.0001 in the determination of the humidity of the air at the top of the tower was found to be sufficient to considerably alter the calculated coefficient of mass transfer but had a negligible effect on the closure of the heat balance. A small error in the determination of water temperature introduces further error in the humidity driving force. As a result the analysis of the adiabatic humidifying runs will deal first with the coefficients of heat transfer, and the relationship between the coefficients of heat and mass transfer will be discussed in a later section.

Several factors combined to somewhat limit the range of both water and air rates which were used in the air-water study. The lower limit on the water rate for one-inch rings was considered to be 500 lbs./(hr.)(sq.ft.); it is doubtful if a water rate less than 500 is adequate to wet the packing.

Loading and flooding data on one-inch rings (39) indicate that at gas rates of 1000 lbs./(hr.)(sq.ft.), loading begins when L reaches about 2500. As a result of these considerations it was decided to use a five-fold change in L, i.e. 500-2500, and investigate a gas rate range of 100-1000. In the course of the adiabatic humidifying runs it was found that gas rates G below 350 were impractical since temperature driving forces then became less than 4°F.

The first 26 runs were of the adiabatic humidifying type and were discarded because of poor heat balances. Several modifications in the apparatus were necessary before heat balance closures were obtained. Then 26 additional runs were made allowing the liquor rate to vary in increments of 500 lbs./(hr.)(sq.ft.) at gas rates of 350, 500, 700, and 1000 lbs./(hr.)(sq.ft.). At this point additional refinements were made in the apparatus in order to measure inlet and outlet water temperatures with greater precision. This was deemed necessary since the coefficient of mass transfer was not reproducible, and it was thought that such a procedure would give more accurate humidity driving forces. The complete ranges of water and gas rates were then repeated (runs 52-80).

Effect of Temperature

In each adiabatic humidifying run it was desired to use a chosen temperature of inlet water (95, 115, or 135°F.)

and to employ a temperature of the inlet air such that it would have a wet-bulb temperature equal to that of the water. In view of the various inlet humidities available (H from 0.006 to 0.016), the preheat temperature of the inlet air ranged from 217 to 675°F. and the corresponding temperature of the gas film, averaged throughout the tower, ranged from 110 to 210°F. Consequently it was necessary to first reduce the raw data to a standard gas-film temperature, arbitrarily selected as 200°F. Figure 5 shows the results of adiabatic humidifying runs made at a constant gas rate of 1000 pounds per hour per square foot of gross cross section wherein the coefficient of heat transfer is plotted on log-log paper versus the liquor rate for water temperatures of approximately 95, 115, and 135°F. These data are seen to vary as the 0.12 power of the liquor rate, and the coefficients decrease with decreasing temperature. In order to eliminate the effect of the liquor rate, $(ha)'_t \, / \, L^{0.12}$ is plotted versus the average film temperature on semi-logarithmic paper in Figure 6. The data fall onto a straight line which is represented by the equation

$$(ha)'_t = C \; e^{0.0023 t_f} \tag{21}$$

where C is a constant and t_f is the average gas-film temperature in degrees Fahrenheit. The data are reported in this fashion to conform with the practice established by previous investigators of reporting the effect of temperature on diffusion coefficients (23),(47). However it should be pointed out that over the range of gas-film temperatures investigated, the coefficient can also be correlated as a linear function of t_f; both functions give the same extrapolation to 70°F.

With a relationship known for the effect of temperature on the coefficient $(ha)'_t$, it was then possible to reduce the coefficients to a common gas-film temperature before determining the effect of other operating variables.

In the runs represented in Figures 5 and 6, temperature driving forces at the air inlet varied from a minimum of 120°F. to a maximum of almost 500°F. At the top of the tower the driving forces varied from 4 to 21°F. In the early stages of the investigation there was considerable doubt as to whether or not these very large driving forces at the bottom of the tower would exhibit some unknown effect upon the coefficients. Furthermore when such large driving forces prevail, water is being vaporized at the interface at an extremely rapid rate and might also affect the coefficients. However, as will be shown later, all data correlated satisfactorily, and so it is concluded that the coefficients are independent of the magnitude of the temperature potential.

Effect of Water Rate

In Figures 7, 8, 9, and 10, are plotted on log-log paper the coefficients $(ha)'_{200}$, which have been corrected to an average gas-film temperature of 200°F., versus the water rate at packed heights of six, nine, and twelve inches. Figure 11 shows $(ha)'_{200}$ plotted versus L at air rates of

350, 500, 700, and 1000 pounds per hour per square foot of gross cross section for a packed height of twelve inches. In this latter Figure it is seen that the coefficient varies with the liquor rate raised to an exponent which increases slowly from zero at a gas rate of 350 lbs./(hr.)(sq.ft.) to 0.15 at G of 1000 lbs./(hr.)(sq.ft.). With the existing apparatus it was not advisable to operate at gas rates below 350. At these low gas rates it was difficult to obtain accurate measurements of entering gas temperatures due to lack of good mixing of the air after it left the preheater. Furthermore, at gas rates below 350, the driving forces at the top of the tower become so small (less than 4°F.) that the precision is reduced. Consequently it was not possible to determine the exponent on the L term at low gas rates (below 350).

The increasing exponent is thought to be due to the effect of the gas rate on the distribution of the water stream. As the gas rate becomes greater the packing becomes more nearly wetted, increasing the interfacial area with a subsequent increase in the product $(ha)'_{200}$. This effect will be discussed further in connection with the air-oil tower where the same effect was noted.

Effect of Packed Height

In Figure 12 the coefficient $(ha)'_{200}$ is plotted on rectangular coordinates against the reciprocal of the packed height for various combinations of water and gas rates.

It is shown in the APPENDIX that when $(ha)'_{200}$ is zero, $-1/z = 1/x$ where x is defined as the height of the packing equivalent to the end-effects. The fact that the extensions of the lines, best representing the data, all intersect at a common point on the abscissa is quite remarkable. The negative reciprocal of the intercept, 7.2 inches, is seen to be independent of both water and gas rates (for clarity, other G and L combinations are not plotted) and represents a substantial correction to be applied to the coefficients when using small depths of packing. Since the inches of packed height equivalent to the end-effects is a constant, the interfacial area due to the combined end-effects is independent of L and G. These facts are apparently explained by visual observation of the "spray-section" by means of the window located below the packing. Throughout the course of the runs it was noted that the liquid leaving the packing did not fall into the sump as spray but rather the predominant portion flowed to the periphery of the packing support and thence down the wall to the sump. Such an action would have the tendency to give a substantially constant interfacial area below the packing and thus provide a constant value for the intercept $1/z$.

From Figure 11 it was pointed out that the coefficient varies with the water rate to a power which is also a function of the gas rate, the exponent ranging from zero at G of 350 to 0.15 at G of 1000. However, Figure 13 shows that little error is introduced by using a constant exponent of 0.07

on L. The coefficients used in this plot have been corrected for end-effects and reduced to 70°F. The data points on this plot represent about 55 runs made over the complete ranges of gas and water rates, and it is seen that numerous points fall one upon the other with very limited scattering. For this reason it is believed that the use of the 0.07 exponent on the water rate is to be preferred to a more complex function of the liquor rate which would allow for the variation in the exponent.

The plot of $(ha)_{70}/L^{0.07}$ versus G shows a slope of 0.70 which is of the magnitude expected from known correlations for heat transfer coefficients across gas films. The equation of the curve of Figure 13 is

$$(ha)_{70} = 2.1G^{0.70}L^{0.07} \tag{22}$$

In view of equation (21), $(ha)'_t$ proportional to $e^{0.0023t_f}$, the final correlation for the true coefficient for the packing itself (corrected for end-effects) therefore takes the form

$$(ha)_{t_f} = 1.78G^{0.70}L^{0.07}e^{0.0023t_f} \tag{23}$$

For the tower used in this thesis Eq. (23) becomes

$$(ha)'_{t_f} = 1.78G^{0.70}L^{0.07}e^{0.0023t_f}\left(\frac{z + 0.67}{z}\right) \tag{24}$$

where the end-effects are now included in the coefficient based on the measured packed height.

Mass Transfer Across the Gas Film

As was pointed out in the PROCEDURE the mass-transfer coefficients were not obtained with the same degree of accuracy as were the coefficients of heat transfer and, consequently, it was not possible to correlate them in an independent manner. For this reason it was assumed that the temperature and end-effect corrections as determined for the heat transfer coefficients were applicable to the coefficients for mass transfer. The following discussion is, therefore, subject to this assumption. In spite of this limitation, however, since the same corrections were applied to both heat and mass transfer coefficients, any comparison between the two is in effect a comparison of the original uncorrected coefficients.

In Figures 7, 8, 9, and 10, $(k'a)'_{200}$ is plotted versus the water rate for a packed heights of twelve, nine, and six inches and at constant gas rates of 350, 500, 700, and 1000 lbs./(hr.)(sq.ft.). The dot-and-dash lines on each plot represents the values of $(k'a)'_{200}$ predicted from the heat-transfer data assuming $ha_H/k'a_M = s$. Although the data scatter considerably, they are consistently low compared with the predicted dot-and-dash lines, indicating that the ratio $ha_H/k'a_M$ is considerably greater than the humid heat s. It should be noted that the ratio $ha_H/k'a_M$ is in reality the product of two ratios, that of the coefficients times the interfacial area per unit of tower volume available for heat transfer over that for mass transfer. In the event that during the operation of the tower all of the packing is not wetted, the ratio of the area for heat transfer will exceed that for mass transfer, tending to make the product ratio greater than s. If dry areas are present these are available for heat transfer but not for mass transfer. Such a condition should reveal its presence in two ways. First, the ratio of the two areas should approach unity as the water distribution improves at high gas rates; and, second the water temperature might well be higher than the adiabatic saturation temperature. If, for example, the inside surface of a ring is partially dry and the outside wet, heat is being transferred both by conduction through the ring wall to the water as well as through the gas-water interface.

As is shown in the plotted data, Figure 13, the deviations of the mass-transfer coefficient become less as the gas rate increases, indicating that the wetted area is approaching more closely the heat-transfer area. In every adiabatic humidifying run it was observed that the water temperature was higher than the adiabatic saturation temperature by one or two degrees indicating that dry sections did exist in the packing. Furthermore, Figure 14 shows that the average differences between the temperature of the water and that of adiabatic saturation decreases with increasing gas rate, as would be expected since the dry area decreases with better distribution. The adiabatic saturation temperature t_{AS} and the water temperature are shown for each run in the tabulated data (APPENDIX). Although $(t_L - t_{AS})$ showed no definite trend with liquor rate, it is noted that the difference becomes smaller as the gas rate increases. Since $(t_L - t_{AS})$ represents the difference between two quantities of approximately the same size, the precision is quite poor and for the purposes of Figure 14, the average of the differences at each set of gas rates is plotted versus the gas rate. Figure 13 also shows $(k'a)_{70} / L^{0.07}$ plotted versus G and further substantiates the above explanation by showing that the deviations from the predicted line, $(ha)_{70} / sL^{0.07}$, decrease with increasing air rate.

In the light of these facts it is not possible to state whether or not h/k's is unity in a packed tower. A wetted-wall tower, however, does offer a source of further information, since, in such an apparatus, not only is the area for heat transfer the same as that for mass transfer but this area can actually be measured. It is, therefore, recommended that the technique used in this thesis be applied to a wetted-wall tower, using sufficient air preheat to furnish adequate precision for the determination of these gas-film coefficients. Although it is recognized that data obtained in a wetted-wall tower can not be applied to a packed tower, nevertheless the problem is of sufficient theoretical interest to warrant the necessary research.

In the design of a tower for new conditions, the measured water temperature would not be available, and consequently it would be necessary to use the temperature of adiabatic saturation. Since the temperature of adiabatic saturation is somewhat less than that of the water, the humidity driving force is decreased, and the values of the revised coefficient called $(k'a)''_{70}$, are based on t_{AS} and corrected for end-effects.

The values of $(k'a)''_{70} / L^{0.07}$ are plotted in Figure 15 versus G, and are closely correlated by the equation

$$(k'a)''_{70} = 7.5G^{0.70}L^{0.07} \tag{25}$$

The dashed line is based on the assumption that

$$\frac{(ha)_{70}}{(k'a)''_{70}} = s \tag{26}$$

Water-Cooling Runs

The heat transfer coefficient (h_La) for several gas rates and corrected for end-effects is plotted in Figure 16 versus L on a log-log paper; it is seen that these coefficients vary as the 0.51 power of L. Figure 17 shows the same data plotted versus G with the effect of liquor rate removed by plotting $(h_La) / L^{0.51}$.

The final correlation is

$$(h_La) = 0.82G^{0.70}L^{0.51} \tag{27}$$

where (h_La) is the coefficient of heat transfer across the water film, corrected for end-effects;

$$(h_La) = (h_La)_{12''} \left(\frac{12}{12 + 7.2}\right) \tag{27a}$$

In the water cooling runs water was fed at a constant temperature of about 135°F. and the air temperature was so adjusted that the water cooled about 30°F., resulting in a substantially uniform average temperature of the liquid film.

The effect of temperature of the water film on h_La is recommended as a subject for future work.

The L/G ratio was varied from one to four for gas rates of 350 and 500, one to three for G of 700, and one to two for G of 1000. The variation in ratio L/G is limited first by the loading characteristics of the packing and second, by the positioning of the operating line (Figure 2) on the enthalpy plot in such a manner that there is no "pinching" of the equilibrium curve. When this pinching does occur at any place in the tower, driving forces become small, and lack of precision results.

The negative slope of the tie-line varied from 1.18 to 2.7 and does not correlate too closely with any of the operational variables. Since the tie-line slope is equal to $-h_La/(k'a)_t$, it is possible to combine Eqs. (23) and (27) and predict the ratio of the two coefficients. The average gas-film temperature for the water-cooling runs was approximately 115°F. and the average humid heat, 0.26 Btu/(lb.). Assuming $(k'a)_{115}$ equals $(ha)_{115}$ / s, Eq. (23) becomes

$$(k'a)_{115} = \frac{2.33}{0.26}\ G^{0.70}L^{0.07} \tag{27b}$$

and the tie-line slope is

$$-\frac{h_La}{(k'a)_{115}} = \frac{(0.82)(0.26)}{(2.33)}\ \frac{L^{0.51}G^{0.70}}{L^{0.07}G^{0.70}} = 0.091\ L^{0.44} \tag{27c}$$

Thus the log of the tie-line slope plotted versus the log of the water rate should be independent of the air rate and have a slope of 0.44. Such a plot is shown in Figure 19.

Although the data deviate somewhat from the solid line which represents Eq. (27c), it is believed that the h_La and k'a correlations are adequate for the solution of problems. It is recommended that dehumidification runs be made in order to obtain a more general correlation for h_La which will be adequate for a tower operating as a dehumidifier as well as a water cooler.

It was pointed out in the INTRODUCTION that due to lack of data on h_La, previous investigators had assumed the slope of the tie-lines on the enthalpy plot to be infinite i.e., negligible resistance was offered to the flow of heat by the liquid film. However, the slope of the tie-lines is also the ratio of the resistances offered by the air and the water films to total heat transfer, and from the reported slopes it is seen that the liquid film offers from 27 to 46 per cent of the total resistance. Consequently, it follows that the assumption mentioned above is subject to criticism. It is recommended that the coefficients as reported in this thesis be used to determine the tie-line slopes when dealing with water cooling towers packed with 1-inch carbon Raschig rings.

Oxygen Desorption

The coefficients of oxygen transfer across a water film are plotted versus L in Figure 18. Using the temperature correction as reported by Sherwood and Holloway for this system, the data are shown to agree very closely with results obtained by these workers on the same system in a 20-inch tower packed with one-inch carbon rings. Sherwood and Holloway sampled and analyzed the water stream at the top and the bottom of the packing and the same procedure was followed by the author. The good agreement between the data from the two towers indicates that a 4-inch tower gives representative values for one-inch rings insofar as liquid-film characteristics are concerned and that wall effects are negligible. Vivian (47), in his work on the absorption of chlorine in water in a 4-inch tower packed with one-inch rings, also made oxygen desorption runs and found that his data checked those of Sherwood and Holloway for the 20-inch tower very closely.

Air-Oil Tower

A series of runs were first made (35) using hot oil and air at room temperature at various gas and liquor rates. However, the data from these runs were not reliable due to poor heat balances in spite of excellent insulation. Since these heat losses were large (up to 46% of the heat transferred) it was possible to calculate two coefficients in much the same manner as Bennett (4) had done, one based on the heat taken up by the air and the other based on the heat lost by the oil. Ua', based on the air temperatures is

$$Ua' = 0.0032\ (G')^{1.57} L^{0.14} \qquad (28)$$

When the coefficient is recalculated and based on the heat given up by the oil, the exponent on the L term remained unchanged, but the exponent on G' is reduced to about 1.1.

It was realized that the problem could not be handled in a satisfactory manner until heat losses were considerably reduced, and consequently, the tower was electrically insulated. The data resulting from these early runs have not been plotted but are reproduced in the APPENDIX.

After installing the electrical insulation, the tower operated substantially adiabatically, and on subsequent runs, heat balances on the average closed within one to two per cent. The coefficients from these latter runs were based on the average of the heat loss by one stream and the heat gained by the other. In all cases this involved but a very small correction.

Hot-Air Runs

The data for the hot-air runs, in which heat was transferred from the hot air to the cold oil, are plotted in Figures 20, 21, and 22, as Ua' versus L for various gas rates and different packed heights. Ua' is the coefficient uncorrected for end-effects. In Figure 23, the coefficients are plotted versus G' for various oil rates at a packed height of 29.1 inches of one-inch ceramic rings.

From Figures 20, 21, and 22, it is noted that the slopes of the lines best fitting the data increase with increasing gas rate (0.19 to 0.31) in much the same manner as did the (ha)' coefficients in the air-water tower. As in the case of the air-water tower, it is believed that this changing slope is due to better distribution at higher gas rates with a subsequent increase in a.

Baker, Chilton and Vernon (2), working with the air-water system found that the principal effect of the gas rate was to increase distribution. Elgin and Weiss (17) also worked with the air-water system and found that, although the liquid hold-up in their tower was a function of liquor rate until high gas rates were reached, distribution was increased by high gas rates, especially near the loading point. The air-oil tower was found to load at gas rates above 800 as was evidenced by oil droplets being carried out of the tower. Schoenborn and Doherty (40) obtained pressure drop data across various types of commercial packings using water and oil as the liquids and air as the gas. They reported that oil was difficult to distribute, and the tendency to channel increased as the oil decreased in viscosity as was evidenced by a decrease in pressure drop.

Figure 23 shows Ua' plotted versus the gas rate at various oil rates. It should be pointed out that, at high gas rates, the exit oil from the tower was at a higher temperature when the oil rate was low, than it was for those runs in which the oil rate was high. This change in temperature of exit oil caused the viscosity of the oil to be high at the high oil rates and low at the low oil rates. It follows that distribution would be better at the high viscosities, i.e., high oil rates, causing a spread in the coefficients. As the gas rate decreases the same effect should occur but to a lesser extent causing the lines to converge with decreasing gas rates. In order to substantiate this explanation it is recommended that liquid hold-up data be obtained on both towers to determine the effect gas and liquid rates on distribution. Such an investigation should attempt to determine the effect of viscosity and other physical properties.

The effect of channeling was more pronounced for the hot-oil runs and will be considered in a later section.

In Figure 24 are plotted the coefficients (Ua') versus the reciprocal of the packed height. The data points were taken from the smoothed curves of Figures 20, 21, and 22. As was the case for the air-water adiabatic humidifying runs, it is seen that the lines intercept the abscissa at a common point, representing a constant end-effect equivalent to 1/0.07 or 14.3 inches of packing. This end-effect is independent of both oil and air rates, and the relation between Ua' and Ua, corrected for end-effects is

$$Ua = Ua' \left(\frac{z}{z + 14.3}\right) \qquad (28)$$

In spite of the fact that the coefficient varies as the 0.19 to 0.31 power of the oil rate (Figures 20, 21, 22), Figure 25 shows that an average exponent on L of 0.25 serves to correlate the data satisfactorily over the range studied. Here is plotted $Ua/L^{0.25}$ versus G', and inspection shows that the data fall closely about a line of 0.94 slope. Thus the final correlation for the hot air to cold oil runs becomes

$$Ua = 0.083\ (G')^{0.94} L^{0.25} \qquad (29)$$

A comparison of the $(ha)_t$ data from the adiabatic humidifying runs on the air-water with the over-all coefficient obtained on the oil-heating runs (See Tabulated Data, APPENDIX) indicates that the oil film does contribute to the over-all resistance to heat transfer, since at comparable air and liquid rates ha is greater than Ua. The relation between the over-all and the individual coefficients of heat transfer is

$$\frac{1}{Ua} = \frac{1}{ha} + \frac{1}{h_o a} \qquad (30)$$

where $h_o a$ is the coefficient of heat transfer across the oil film. Substituting ha values from the air-water tower into Eq. (30) presupposes that the a is the same for both towers, and in view of probable differences in the distribution of oil and water this procedure was considered unwise. However, assuming that ha is proportional to $G^{0.70}$ (Eq. 23), $1/Ua$ was plotted on log-log paper versus $1/(G')^{0.70}$ at constant liquor rates. The value of the intercept of a line through the data points is $1/h_o a$. A positive intercept was not obtained; for the majority of the liquor rates, the intercept was negative. This fact is explained by the distribution picture discussed above. As the gas rate increases($1/(G')^{0.70}$ gets small), the distribution of the oil is increased and a is increased causing $1/Ua$ to become too small and resulting in a small or negative intercept. The effective useof such a plot depends upon a constant value of a, and it has been pointed out that a is evidently also a function of the gas rate. As a result it was not possible to estimate the percentage of the total resistance to heat transfer which is offered by the oil film.

The data are not adequate to permit a determination of the effect of temperature. As was indicated above, temperature undoubtedly plays an important role with regard to oil viscosity which in turn affects the distribution. However, in addition to the effect of temperature on viscosity of the oil, temperature also effects the gas film coefficient and, no doubt, has also an effect upon the coefficient of heat transfer across the liquid film. As a result, the effect of temperature upon the over-all coefficient Ua becomes quite complicated. Air inlet temperatures in the hot-air runs varied from 200 to 250°F. and outlet temperatures from 65 to 90°F. Consequently, the temperature variation of the gas film, which undoubtedly offers that major percentage of resistance to heat transfer, was not large. In view of the fact that the effect of temperature on the gas-film coefficient was shown to be small in the air-water tower, temperature corrections on the coefficient (U) are probably insignificant. However, the viscosity of the oil changed several fold in these runs depending upon the L/G ratio, and its resulting effect on a is unknown. It is recommended that the effect of temperature on these coefficients be determined, preferably in the same apparatus used in this thesis.

Of the two phases of the air-oil study, the hot-air runs were the least satisfactory due to the difficulty of obtaining adequate driving forces at the top of the tower. Two factors contributed to this difficulty. First, the fact that the heat capacity of the oil (0.44) is approximately twice that of the air, somewhat limits the range of possible L/G ratios which will give satisfactory results. From heat balance considerations this ratio should be 1/2 or smaller. However, it was pointed out above that the tower showed loading characteristics at gas rates above 800 lbs./(hr.)(sq.ft.) and, consequently, the oil rate range was greatly restricted. Second, it was not possible to preheat the air to a temperature greater than 250-300°F. due to actual vaporization of the low-boiling components in the oil. Fortunately, these problems were not encountered in those runs in which heat was transferred from the hot oil to cold air.

Hot-Oil Runs

In Figures 26 and 27 are shown the over-all coefficients of heat transfer for the hot-oil runs at various air and oil rates. Inlet oil temperature varied from 190-205°F., and outlet oil temperatures from 118 to 205°F. depending on the L/G ratio. Air inlet temperatures were quite constant at 105-115°F. At low air rates (below 500) the tower showed distinct evidences of channeling. The oil was observed to leave the packing in 12 streams, symmetrically arranged in the same fashion that the 12-point oil distributor delivered oil at the top of the packing. As the gas rate increased, however, this tendency to channel decreased rapidly so that at gas rates of 800 lbs./(hr.)(sq.ft.) the oil was seen to leave the packing in a random manner, some dropping from the packing support directly into the sump and some running down the wall. Pressure drops across the packing are shown in Figure 33 for both the hot-oil and hot-air runs, and it is apparent that the pressure drop for the latter is larger than for the former until high gas rates are reached where they become nearly equal. This action is explained again by the fact that distribution is increased by high gas rates. At low gas rates the oil viscosity was several fold greater in the hot-air runs than for the hot-oil runs so that channeling occurred with a consequent decrease in a.

Figure 27 was constructed from the smoothed data of Figure 26 and shows that the coefficient Ua' varies as the 1.5 to 1.7 power of the gas rate, which checks reasonably well the values found by Bennett (4), 1.51, and Philoon (35), 1.57, when their coefficients were based on the heat taken up by the oil. Even though their heat balances were poor, the gas-film is undoubtedly the controlling resistance so that the coefficient should be based upon the heat actually crossing the gas film. The major portion of the heat loss would be expected to occur directly from the oil to the tower wall since the tower wall is presumably kept wet by the oil.

It was found that for the hot-oil runs the end-effect was a function of the gas rate in contrast to the adiabatic humidifying runs in the air-water tower and the hot-air runs in the air-oil tower where end-effects were independent of both liquid and gas rates. In view of the marked channeling which occurred in the hot-oil runs, it is not surprising that the end-effect corrections were more complicated and difficult to obtain. Unfortunately, this situation was not fully appreciated until after most of the work was completed. As a result the runs at various packed heights were not made over the full range of oil and gas rates so that some interpolation and extrapolation of the data was necessary in order to evaluate the end-effects.

Figure 28 shows actual coefficients obtained for 14 and 18 inches of packed heights plotted on log-log paper versus the liquor rates at the indicated gas rates. However, the runs at 29.1 inches of packed height were not made at the same air rates, and in view of the large effect of the gas rate on the over-all coefficients (Figure 27) the curves for the 29.1 inch packed height as shown in Figure 28 were taken from the smoothed data of Figure 27. From Figure 28 it was possible to construct Figure 29 which shows that the end-effects are independent of oil rate but a function of the air rate. At the low gas rates it is noted that the end-effect becomes very large (100 inches at G' = 261-266). At first glance this seems highly improbable but it must be emphasized that at these very low gas rates the oil was apparently running through the column in individual streams and that little or no distribution was being effected. With such a condition prevailing in the packing it is possible that the open section below the packing was offering more interfacial area for heat transfer than was the packing itself. As the gas rate increases the end-effect becomes smaller indicating better distribution in the packing.

The magnitude of the packed height equivalent to the end-effects was found to be a power function of the gas rate. In Figure 30, the log of the intercepts $1/z$ from Figure 29 was plotted versus the log of the gas rate and a straight line results, of slope 2.1.

The type of plot used to determine the end-effects (See APPENDIX) is such that at $1/z = o$, i.e., infinite packed height, Ua' is equal to the true coefficient, independent of end-effects. Thus for each combination of gas rate and oil rate, the true coefficient is indicated by the intercept on the vertical axis. This was the source of data for Figure 31 in which the corrected coefficient Ua is plotted on log-log paper versus the oil rate. Figure 32 was constructed from Figure 31, which shows that the coefficient varies as the 0-0.35 power of the oil rate. Due to this rapid change in slope it was found to be unsatisfactory to assign a single numerical exponent to L and obtain a general correlation for Ua. Similarly, Figure 32 shows the exponent on G' to vary from 2.2 to 2.8, and it is believed that a graphical representation of the results is to be preferred to an algebraic expression which would allow for these changing exponents.

Figure 32 also shows the results of the hot-air runs (dashed lines) also corrected for end-effects. It is interesting to note that the two types of runs give approximately the same results at air rates of about 700-800 lbs./(hr.)(sq.ft.). In this region the average oil temperature was the highest for the hot-air runs and lowest for the hot-oil runs so that oil viscosities were of the same magnitude causing approximately the same distribution. However, actual gas velocities in the two cases were somewhat different due to different ranges of air temperatures so that no definite conclusions can be drawn with regard to the effect of viscosity.

As for the case of the hot-air runs, time did not permit a study of the effect of temperature on the coefficients from the hot-oil runs. It is probable that the temperature has a much greater effect on distribution than on the actual coefficient. A logical attack on the problem would be to employ oils of differing viscosities in an attempt to separate these two effects.

VI. CONCLUSIONS

From the results of the air-water study it is concluded that:

1. Adiabatic operation requires external application of heat to the insulation, controlled to produce a zero radial temperature gradient in the insulation, and attainment of steady state; in all runs correlated, the heat balances close within a maximum deviation of 2.6 per cent and an average deviation of 0.6 per cent.

2. In counterflow adiabatic humidifying runs with water entering at the wet-bulb temperature of the inlet air, the water temperature is constant, independent of water rate, air rate, packed depth, and temperatures of entering air and water.

3. The effect of the air-film temperature, $t_f = (t_L + t)/_2$, is relatively small and is given by

$$ha \propto e^{0.0023t_f}$$

4. The gas-film coefficient for heat transfer is independent of the magnitude of the temperature driving force between the air and water; driving forces ranged from 4 to 545°F., yet all data correlate satisfactorily.

5. The effect of water rate on the heat transfer coefficient

for the gas film for packed depths ranging from 6 to 12 inches:

$$ha \Big/ e^{0.0023 t_f} \propto L^n$$

where n ranges from 0 to 0.15 as the air rate increases from 350 to 1000 lbs./(hr.)(sq.ft.), thereby increasing the wetted area; an average value of n of 0.07 is satisfactory.

6. The effect of packed depth z, ranging from 6 to 12 inches, is independent of air and water rates as shown by plotting ha corrected to a film temperature of 200°F. versus the reciprocal of the packed depth; the curves for various combinations of L and G converge at a common point at the reciprocal of 7.4 inches (0.617 feet).

7. The effect of air rate G for a given film temperature, liquor rate, and packed height is substantial

$$ha \propto G^{0.70}$$

8. The complete equation for the coefficient of the gas phase for the 4-inch tower used in this thesis and packed with one-inch carbon Raschig rings is

$$ha'_{t_f} = 1.78\, G^{0.70} L^{0.07} e^{0.0023 t_f} \left(\frac{z + 0.617}{z}\right)$$

This is shown by plotting

$$ha_{(z = \infty,\ t_f = 70^\circ F)} = (ha)'_{t_f} \left(\frac{e^{0.0023(70)}}{e^{0.0023 t_f}}\right)\left(\frac{z}{z + 0.617}\right) \Big/ L^{0.07}$$

versus $G^{0.70}$. As shown by Figure 13, this correlates the data within a maximum deviation of 8 per cent and an average deviation of 3 per cent.

9. Mass transfer coefficients $k'a$ for the gas phase are inherently less precise than those for ha. If based on measured temperatures of water and humidities of inlet and outlet air, the dimensionless ratio $ha_H/k'a_M s$ exceeds unity, as shown by Figure 13; a supporting argument is that a_H doubtless exceeds a_M, except possibly at the highest values of L and G. Furthermore, the water temperature exceeded the adiabatic saturation for all runs, the difference between the two temperatures decreasing with increasing gas rate as is shown by Figure 14.

However, based on adiabatic saturation temperatures and corresponding equilibrium humidity at the interface, the ratio $ha_H/(k')''a_M s$ is substantially unity, as shown by Figure 15. Since in design work the temperature of adiabatic saturation is used as the water temperature, and because of the utility of the assumption of unity for this ratio in the analysis of cooling-tower performance, it is concluded that the ratio should be taken as unity;

$$k'a = \frac{1.78}{s} G^{0.70} L^{0.07} (e^{0.0023 t_f}) \left(\frac{z + 0.617}{z}\right)$$

10. For cooling-tower runs, the liquid-film coefficient of

heat transfer depends on both G and L and is given by

$$h_La = 0.82G^{0.70}L^{0.51}$$

which covers values of G ranging from 350 to 1000, values of L ranging from 540 to 2100, and L/G values ranging from 1.5 to 4.0. Temperatures of inlet water were substantially constant at 135°F. and the water was cooled about 35°F. for all runs.

11. The slope of the tie-line in the cooling tower diagram, $-h_La/k'a$, which is the ratio of the resistance of the gas film to that of the liquid film for enthalpy transfer, ranged from 1.18 to 2.70. Hence it was found that the liquid film offered from 46 to 27 per cent of the total resistance to enthalpy transfer. This is in sharp contrast to the usual assumption that the resistance of the liquid film is negligible.

12. Oxygen desorption runs at room temperature gave mass-transfer coefficients, K_La, for the liquid film agreeing closely with those for the Holloway tower having five times the diameter, (20 versus 4 inches).

From the results of the air-oil study it is concluded that:

13. It was necessary to electrically insulate the tower in much the same manner as was done for the air-water tower in order to obtain operation approaching adiabatic conditions.

In all runs correlated, the heat balances close within an average deviation of 1.8 per cent of the heat transferred from the hotter fluid.

14. The effect of packed depth z, ranging from 14 to 29.1 inches, is independent of liquor and gas rates for the hot air-cold oil runs and is equal to 14.3 inches of packed height. However, for the hot oil to cold air runs the end effect, although independent of the oil rate, varies as the 2.1 power of the gas rate. This is explained by the fact that for the hot-oil runs the oil viscosity is low and channeling occurs to marked degree, especially at low gas rates, resulting in a decrease in "a". As the gas rate increases, the oil becomes more thoroughly distributed.

15. The effect of gas rate on the over-all coefficient of heat transfer between hot oil and cold air is

$$Ua \propto G^{0.94}$$

for gas rates ranging between 250-800 lbs.(hr.)(sq.ft.).

16. The coefficient for hot air-cold oil runs at various gas rates varies as the 0.19 to 0.31 power of the oil rate which ranged from 500-2000 lbs./(hr.)(sq.ft.). This diverging effect of the coefficients with oil rate is explained by the fact that contact area increases with increasing viscosity and gas rates. The use of an average exponent

of 0.25 on L is justified by the fact that the data are correlated within a maximum deviation of 10 per cent by the equation

$$Ua = 0.083(G')^{0.94}L^{0.25}$$

where the end-effects are eliminated from Ua.

17. The over-all coefficient of heat transfer between hot oil and cold air varied as the 0-0.39 power of the oil rate and the 2.2-2.8 power of the gas rate. The high exponent on the G' term is explained by the fact that channeling was predominant at low gas rates, and better distribution is obtained as the gas rate increases, causing a rapid increase in "a". This explanation is corroborated by the fact that at low gas rates (below $G' = 500$) the oil was observed to leave the packing in a number of streams equal to the number of delivery points in the top of the tower.

18. Since the exponents on the gas rate is a function of the oil rate and the exponent of the oil rate is a function of the air rate, the over-all coefficients of heat transfer from hot oil to cold air are more effectively presented in graphical form in Figures 31 and 32.

19. It is concluded that the coefficients of heat transfer for the hot-oil runs were considerably smaller than those from the cold-oil runs until high gas rates were reached where the oil was adequately distributed.

VII. RECOMMENDATIONS

It is recommended that:

1. Dehumidifying runs should be made using the air-water system to further test the applicability of the enthalpy driving force concept and to determine whether water-cooling and humidification operation give comparable results. For dehumidifying runs, the operating line would be located above the equilibrium curves.

2. The effect of temperature on h_La should be studied. In the runs made for this thesis, water was cooled from 135 to 100°F. so that the position of the operating line relative to the equilibrium curve (See Figure 2) was substantially unchanged. By shifting the operating line under the steep portion of the equilibrium curve, it would be possible to determine the effect of large enthalpy driving forces on h_La.

3. It is recommended that adiabatic humidifying runs be made in a wetted-wall tower. In such an apparatus the areas for heat and mass transfer are identical and measurable so that it would be possible to determine the psychrometric ratio h/k' for the air-water system. Since in a wetted-wall tower, the number of transfer units is small compared to a packed tower, by means of suitable design the mass-transfer coefficients and psychrometric ratio could be determined with greater accuracy than those obtained in this thesis.

4. It is recommended that the oil-air study be continued to study the effect of temperature and oil viscosity on the coefficients, using several oils of various viscosities at 100°F. Liquid hold-up data for oils of differing viscosities would be of value in determining the effect of viscosity on distribution characteristics of oils in packed towers, supplementing the conclusions based on pressure drop.

VIII. APPENDIX

A. SUPPLEMENTARY INTRODUCTION

Adiabatic Saturation Operation of Air-Water Tower

It was shown in the INTRODUCTION that an over-all enthalpy balance around a tower of infinite height, wherein the liquid stream is being fed at the temperature of the exit gas and at a rate equal to its rate of vaporization, resulted in Eq. (10)

$$i_{G1} + (H_s - H_1)i_{Ls} = i_{Gs} \tag{10}$$

where the subscript 1 refers to conditions at the base of the tower and s refers to conditions at the top. The gas enthalpy terms include the enthalpy of the dry gas at the temperature of the mixture and at the partial pressure of the dry gas in the mixture plus the enthalpy of the liquid vapor at its temperature and partial pressure.

Considering the air-water system specifically, the enthalpy of liquid water and its vapor at various temperatures and saturation pressures can be obtained from steam tables. If water vapor were a perfect gas, its enthalpy would depend only upon the temperature and would be independent of the pressure. Then values read from the saturated steam tables could be used without error at pressures other than saturation pressures. Actually the enthalpy of water vapor varies somewhat with pressure but for most psychrometric calculations the quantity of water vapor is small compared to the quantity of air and a slight error in the enthalpy of water vapor introduces only a negligible error in the enthalpy of air-water vapor mixtures. Thus for calculations from 32 to 212°F it is sufficiently satisfactory to take the enthalpy of water vapor directly from saturated steam tables, particularly in view of the fact that the saturation pressure of steam is low over this temperature range. Above 212°F the saturation pressure of steam increases rapidly. Therefore at temperatures above 212°F it is preferable to refer to superheated steam tables and to use the enthalpy of superheated steam at one atmosphere.

The enthalpy of dry air at any temperature t can be calculated from specific heat data which is, fortunately, practically independent of pressure since air behaves very nearly as a perfect gas at normal temperatures. Dry-air enthalpy values used in this thesis were determined from actual specific heat data obtained from spectroscopic analysis (53).

The enthalpies of air-water vapor mixtures were calculated from the equation

$$i_G = i_a + Hi_{wv} \tag{31}$$

where i_a is the enthalpy of the dry air and i_{wv} is the enthalpy of the water vapor as read from steam tables. This equation assumes that the enthalpies are additive, which is true only when the substances are ideal. Since neither air nor water is ideal, the equation is subject to a correction, the magnitude of which is unknown; however, at the low pressures involved in the work of this thesis, the deviation is probably small and Eq. (31) is sufficiently accurate.

The enthalpy of water vapor i_{wv} at any temperature t above a base temperature t_o is

$$i_{wv} = r_{to} + c_v\ (t - t_o)$$

where r_{to} is the latent heat of vaporization and c_v is the heat capacity of water vapor equal approximately to 0.45 Btu/(lb.)(°F.) Therefore,

$$i_G = c_G\ (t - t_o) + H\left[r_{to} + c_v(t - t_o)\right] \qquad (32)$$

where c_G is the specific heat of the gas, in this case air, equal approximately to 0.24 Btu/(lb.)(°F).

Defining the humid heat s as equal to $(c_G + c_vH)$ and substituting Eq. (32) in Eq. (10) results in Eq. (10a) as presented in the <u>INTRODUCTION</u>

$$s_1\ (t_1 - t_s) = (H_s - H_1)r_s \qquad (10a)$$

which represents the equation for the adiabatic saturation lines of the humidity chart.

Systems Other Than Air-Water

The basic equations which were used in the analysis of the air-water study are general in form in that they are independent of the liquid and gas. However, it is not possible to combine these equations to obtain an enthalpy or total heat driving force with any system other than one for which ha/k'a is equal to the humid heat. This can be seen by multiplying the relation for the rate of mass transfer, Eq. (15), and adding to the equation for the rate of heat transfer as was done for the air-water system in the <u>INTRODUCTION</u>. Assuming $ha/k'as = x'$ where x' has some numerical value other than unity

$$-G(r_o dH + s dt) = k'a\, dz \left[(r_o H + x's t) - (r_o H_i + x's t_i)\right] \qquad (33)$$

It is, of course, possible to define the functions in parentheses and employ this term as a psuedo-enthalpy for use in a plot similar to Figure 2, but the equation of the operating line

$$c_L\, LdT = Gdi_G \qquad (19)$$

still involves the enthalpy term as was used for the air-water system. It is not clear as to how the two "enthalpies" may be combined so as to furnish a method for the analysis of liquid-cooling or dehumidification operation.

Moreover, there is no evident method for the direct determination of gas-film coefficients. If the psychrometric ratio does not equal unity, it is not possible to make adiabatic humidifying runs in which the liquid passes through the tower unchanged in temperature. Consequently, the bulk temperature of the liquid cannot be the same as the temperature of the interface since there is a temperature difference across the liquid film. It appears as if a proper analysis of systems other than that for air-water must await a general solution of the basic equations.

Variation in the Air and Water Rates

Although the dry-gas rate G, which was used to correlate the air-water tower data, was by definition a constant, the actual gas rate through tower changed continuously due to the vaporization of water. However, the exit gas rate in terms of dry air plus water vapor never exceeded the entering wet-air rate by more than 10 per cent, and no attempt was made to allow for this effect. Furthermore, since it is customary to use a dry-air rate in calculations dealing with the air-water system, it seemed advisable to correlate the data in terms of the dry-gas rate.

For those adiabatic humidifying runs in which the air rate was large and the water rate small, the inlet water exceeded the outlet rate by as much as 20 per cent. However, in view of the small effect of the water rate on the coefficients of heat and mass transfer across the gas film, a 20 per cent change in L has a negligible effect upon the coefficients.

For the water cooling runs the water rate varied a maximum of 6 per cent of the inlet water rate and an average of 2 per cent. The variation of the gas rate for these runs was also considerably smaller, averaging about 3 per cent.

Determination of End-Effects

End-effects for both towers were evaluated in the same manner i.e., plotting measured coefficients versus the reciprocal of the packed height and extrapolating to infinite height. This procedure is based on the following analysis for the coefficient of heat transfer across the gas film as determined in the air-water tower.

$$(ha)'_{tf} = \frac{Gs}{z} \quad N_t \tag{34}$$

where z is the measured packed height and N_t is the number of transfer units. Eq. 34 can also be written

$$(ha)_{tf} = \frac{Gs}{(z+x)} \quad N_t \tag{35}$$

where $(ha)_{tf}$ is the true coefficient independent of end-effects, and x represents the height of packing equivalent to the end-effects. Dividing Eq. (34) by (35)

$$(ha)'_{tf} = (ha)_{tf} + \frac{x}{z} (ha)_{tf} \tag{36}$$

It is seen that if all variables which affect $(ha)_{tf}$ were held constant, a plot of $(ha)'_{tf}$ versus 1/z should be a straight line of slope $(x)(ha)_{tf}$ and should intercept the ordinate at $(ha)_{tf}$. Furthermore, when $(ha)'_{tf}$ is zero, $\frac{1}{z} = -\frac{1}{x}$.

VIII. APPENDIX

B. DETAILED PROCEDURE

Heat Balances on Air-Water Tower

The first ten runs which were made on the air-water tower had rather large heat losses in spite of the fact that the column was well insulated with a two-inch thick covering of magnesia. These losses ranged from 25 to 50 per cent of the heat transferred across the gas film. Consequently the tower was equipped with wire heaters wrapped around the column. The procedure for this installation is outlined below.

The problem was somewhat simplified by the fact that for the adiabatic humidifying runs, the water temperature remained constant throughout the tower so that the surface of the insulation covering those sections of the tower which contacted the water was essentially at a uniform temperature. However, the inlet air section of the tee at the base of the column would be expected to be at a temperature higher than the packed section as would also be that portion of the tower above the packing.

Holes one inch in diameter were drilled radially through the insulation to the external tower surface at six-inch intervals all the way up the column, from the sump to a point about six inches above the top of the packing. Another hole exposed the wall of the air inlet section of the tee. Through each hole a thermocouple was mounted directly to the external tower surface, and then each hole was plugged with rock wool. An adiabatic humidifying run was made, and the various tower-wall temperatures noted. These indicated that four different temperature zones existed on the surface of the column:

(1) the sump and bottom section of the tee (127-129°F)
(2) the top of the tee and the packed section (125-128°F)
(3) the unpacked tower section (126-130°F)
(4) the air inlet section of the tee (212°F)

Each of these four sections was then covered with nichrome (resistance) wire in such a manner that every square inch of insulation surface was in good thermal contact with one of the four heating elements. Next, thermocouples were mounted in the insulation surface so that there was a "surface" couple radially opposite each tower "wall" couple, and the four leadwires from each set of surface and wall couples were connected such that the wall couple acted as the hot junction and the surface couple became the cold junction. With this arrangement the voltage applied to the heating elements could be adjusted until each set of surface and wall thermocouples produced a zero deflection on a galvanometer, thus indicating that a zero temperature drop existed between tower wall and insulation surface.

Two runs were then made using heated air and no water in order to obtain a comparison of the heat losses with and without the use of the heating elements. After steady-state was obtained the following air temperatures were recorded at the indicated positions in the packing, (air rate = 69 lbs./hr.):

Position	Temperature	
	Heating Elements Off	Heating Elements On
Air inlet section of the tee	142	138
Bottom of the packing	136	137
Top of packing	131	137
Leaving the tower	126	137

Employing the heating elements to prevent heat losses, a series of adiabatic humidifying runs was started with the belief that heat balance closures could now be obtained. However, such was not the case. Now it was found that heat losses were negative, i.e., the tower was gaining heat from the surroundings, and the discrepancies amounted to 3 to 10 per cent of the heat transferred across the gas film. All thermocouples and thermometers were removed from the apparatus, tested, and found to be functioning properly. Finally, after many unsuccessful attempts to locate the source of trouble, thermocouples were placed at various positions in the tower packing to determine whether or not there was any detectable change in the temperature of the water stream between its inlet temperature at the distributor and the exit temperature in the sump. It was found that although the water passed through the packing at a constant temperature, there was a temperature difference between the water leaving the packing support and the water in the sump of 5°F., corresponding to a 425 Btu/(hr.) heat input over this section.

This temperature rise was attributed to the fact that heat was being conducted by the walls of the hot-air inlet pipe to the water flowing from the packing support into the sump (See Figure 3). This explanation was verified by calculating the heat which would be conducted due to the difference in temperature between the inlet air and the water leaving the packing support. The length of the conducting medium was taken as the measured distance between the thermocouple which recorded the inlet air temperature and the vertical section of the tower which was carrying the water to the sump. The area through which this heat was "finning" was determined from the dimensions of the tee. The total heat transferred was calculated to be 500 Btu/(hr.).

The facts indicated that the heat which was being conducted into the tower was picked up by the water after the water had left the packing and was thus being carried into the sump. Since it was believed that this "finned" heat was not affecting conditions in the packing itself, subsequent runs were based on an exit water temperature as measured at the packing support. After this modification, heat balances closed within the desired 1 per cent, and no further difficulty was experienced.

Orifice Calibrations

The 4-inch and 8-inch towers were each equipped with three orifice plates to cover the desired range of gas rates using water-filled inclined manometers, which were of such size as to measure heads up to 8 inches of water. These gas orifices were calibrated by means of the large gasometer (300 cu.ft.) in the Mechanical Engineering Laboratory at M.I.T. This device was used to obtain calibrations which were unusually accurate, the maximum deviation of data points from the correlations shown in Figures 35 and 37 being about one per cent. The liquid orifices were calibrated by collecting and weighing the efflux over a timed interval. These calibrations are shown in Figures 36 and 38.

Desorption of Oxygen

In order to obtain some knowledge of the importance of wall effects in the air-water tower, a series of oxygen desorption runs was made, and the coefficients of mass transfer were calculated for comparison with those obtained by previous investigators. Holloway (23) in his study of coefficients of mass transfer across liquid films worked out the technique for desorbing oxygen from a water stream by passing air countercurrently to the liquid in a 20-inch tower packed with one-inch rings. Vivian (47) using the same analytical technique as Holloway, checked Holloway's work in a 4-inch tower packed with one-inch ceramic rings.

It is realized that even though liquid-film coefficients from a 4-inch tower may check those obtained from a column five times larger in diameter, this does not preclude the possibility that wall effects are not negligible with respect to gas-film phenomena such as were studied principally in the 4-inch tower of this thesis. This is seen from the fact that for heat transfer a gas film is operative where no liquid film exists i.e., in dry sections of the packing. However, the fact that liquid-film coefficients from the two different columns did check does not eliminate the possibility that the gas-film coefficients obtained in the 4-inch tower are independent of tower diameter.

The procedure used was essentially that of Holloway. Pure oxygen was bubbled slowly at a constant rate into the water line leading to the constant temperature reservoir. Here the excess was allowed to escape into the room and the oxygen enriched water was pumped to the distributor above the packing. Samples were taken from the distributor and from a sampling cup placed directly beneath the packing. The exit water was not recycled but sent to the drain. In order to maintain the liquid (water) film at a constant temperature throughout the column, the water was introduced at the wet-bulb temperature of the entering air. It is worth noting that under these conditions both films were operating, each substantially independently of the other - oxygen was being transferred across the liquid film and gas film, with the gas-film offering a negligible resistance, and heat and water vapor being transferred in opposite directions across the gas film.

The procedure used for analyzing the water samples taken at both ends of the packing is presented in detail by both Holloway and Vivian and will not be described here. The equilibrium data used for oxygen dissolved in water was that of Winkler (55), which is reproduced in Figure (34). Samples of tap water which had been brought to equilibrium with the oxygen in the air, were analyzed for dissolved oxygen and found to check closely with Winkler's data.

The equation defining K_La is

$$(L/d)(c_t - c_b) = K_La(z)(c - c_e)_{ave.}$$

$$= K_La(z)\frac{(c_t - c_e) - (c_b - c_e)}{\ln\frac{(c_t - c_e)}{(c_b - c_e)}}$$

where d is the density of the liquid.

Since the partial pressure of oxygen in the gas is essentially constant throughout the tower $(c_e)_{bottom} = (c_e)_{top}$ and Eq. (34) becomes

$$K_La = \frac{L}{(d)(z)} \ln\frac{(c_t - c_e)}{(c_b - c_e)} \qquad (35)$$

Accuracy of Measurement of Gas-Film Coefficients

In order to illustrate the effect of various errors in measurements from which the gas-film coefficients were calculated, two adiabatic humidifying runs were selected (78 and 79) which show the extremes in the maximum error. In run 79 the air rate was low and the water rate high

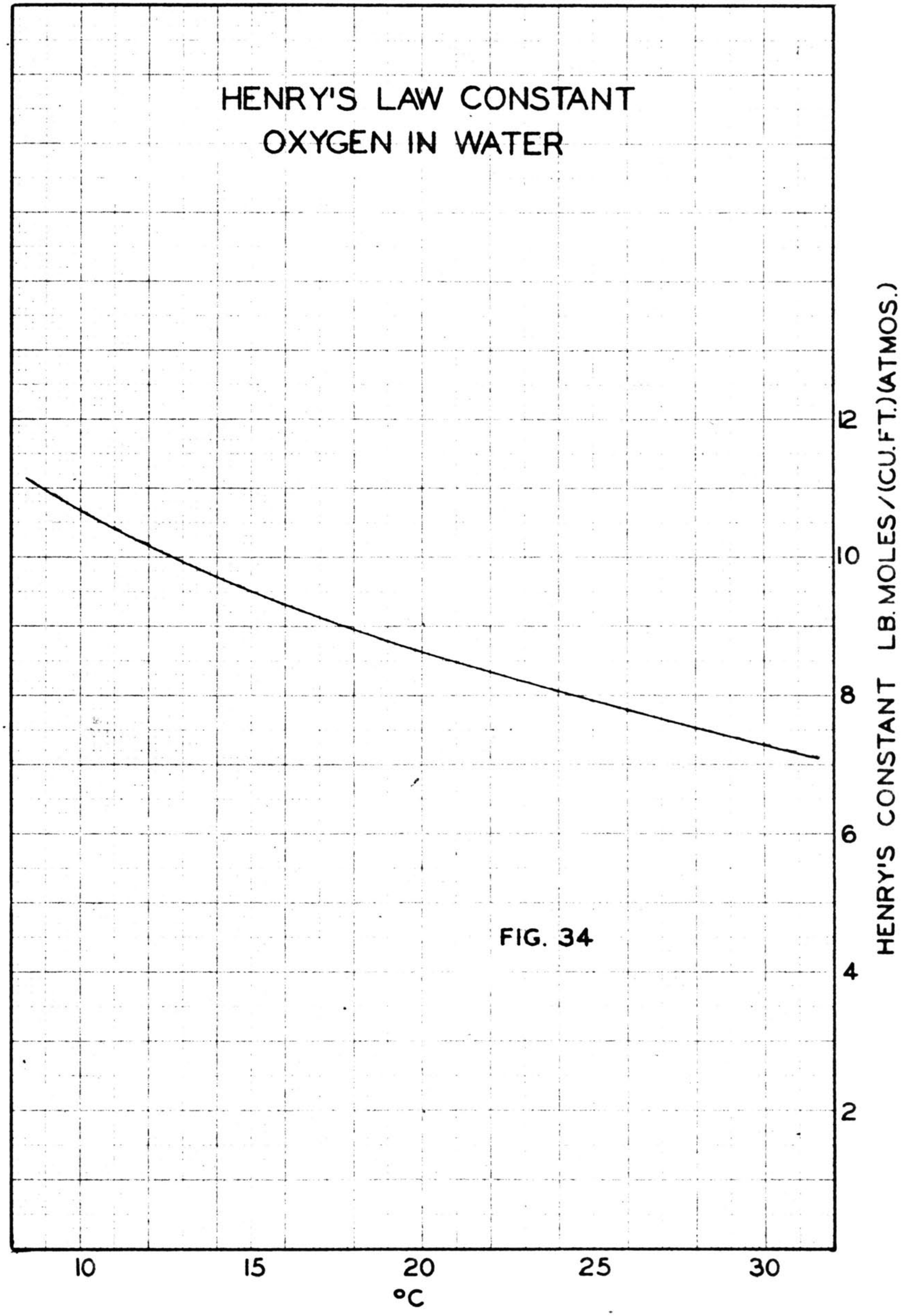

HENRY'S LAW CONSTANT
OXYGEN IN WATER
HENRY'S CONSTANT LB. MOLES / (CU. FT.) (ATMOS.)
12
10
8
6
4
2
10
15
20
25
30
°C

FIG. 34

OVER-ALL ACCURACY OF $(ha)'_t$ and $(k'a)'_t$

MEASUREMENT	RUN 79		RUN 78	
	Maximum Error	Value	Maximum Error	Value
G	2%	350±7	2%	1006±20
s	2%	0.270±0.005	2%	0.271±0.005
z	4%	1±0.04	4%	1±0.04
t_2	5°F	610±5	5°F	607±5
t_1	0.2°F	139.5±0.2	0.2°F	154.2±0.2
T	0.2°F	134.9±0.2	0.2°F	135.0±0.2
H_2	0.0010	0.1237±0.0010	0.0010	0.1248±0.0010
H_1	0.0005	0.0097±0.0005	0.0005	0.0122±0.0005
H_i	0.5%	0.1253±0.0007	0.5%	0.1274±0.0007
$N_t = \ln\left(\frac{t_2 - T}{t_1 - T}\right)$	2.4%	4.63±0.11	1.0%	3.2±0.03
$N_t = \ln\frac{(H_i - H_1)}{(H_i - H_2)}$	25%	4.28±1.08	18%	3.78±0.67
$(ha)'_t = \frac{Gs}{z} N_t$	10%	436±45	9%	869±78
$(k'a)'_t = \frac{G}{z} N_t$	31%	1496±460	24%	3790±910

resulting in small temperature and humidity forces at the top of the packing. Run 78 was made with a high gas and a low liquid rate which combined to give large exit driving forces. The estimated maximum errors (given in the following table) for the measured quantities are also indicated and are believed to be liberal. It is important to note that the height of packing measurement z accounts for 4% of the final maximum error determined for each coefficient. This fact was realized at the beginning of the runs and extreme care was used in measuring the packed height. A twelve inch length of standard 4-inch pipe, which was of the same diameter as the tower itself, was placed on a flat surface and the one-inch rings which were to be placed in the tower were dumped inside the section. The section was considered filled when no rings could be seen upon sighting across the rim. The rings were then removed and counted. This procedure was repeated several times and the average of the number of rings required each time to fill the pipe to a height of one foot was used as the tower packing for the subsequent runs. When the packed height was varied to determine end-effects, one fourth of the rings were removed for the runs at nine inches of packing and an additional one fourth for those runs at six inches of packing. As a result of this procedure, the data should be internally more consistent than the error analysis indicates.

Characteristics of the Oil Employed

The oil used was provided by the Standard Oil Development Company. It was a solvent-extracted paraffinic distillate stock, with the following characteristics:

A.P.I. Gravity		33.4
Flash Point		410°F.
Fire Point		460°F.
Viscosity	100°F.	148 SUS
	210°F.	43.7 SUS
Viscosity Index		113
Conradson Carbon Residue		0.009%
Cloud Point		+8°F.
Pour Point		+5°F.
Neutralization Number		0.04
Sulfur		0.09%

The heat capacity was experimentally determined as 0.44 Btu/(lb.)(°F).

VIII. APPENDIX

C. SUMMARIZED DATA

The summarized data are presented in the following tables. Those runs with poor heat balances have not been included.

Air-water tower adiabatic humidifying runs 121-127 were not used in the correlations because it is believed that the measurement of packed height z was in error.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	26	27	28	29	30
L, lbs./(hr.)(sq.ft.)	540	1060	1600	2100	2620
G, lbs./(hr.)(sq.ft.)	500	500	505	505	505
T_2, $^{\circ}$F, water at top	138	140	140	140	140
T_1, $^{\circ}$F, water at bottom	138	140	140	140	140
t_2, $^{\circ}$F, air at top	146.6	148.0	146.8	146.8	146.8
t_1, $^{\circ}$F, air at bottom	625	675	677	678	681
H_2, lbs. H_2O/(lb. air)	0.1278	0.1389	0.1414	0.1403	0.1398
H_1, lbs. H_2O/(lb. air)	0.0088	0.0088	0.0090	0.0090	0.0090
$(H_t)'_{HT}$, ft., (1)	0.248	0.242	0.229	0.229	0.229
$(H_t)'_{MT}$, ft., (1)	0.380	0.217	0.305	0.330	0.338
$(ha)'_t$, $\frac{B.t.u.}{Hr.-Cu.Ft.-^{\circ}F}$, (1)	546	543	604	602	603
$(k'a)'_t$, $\frac{Lbs.\ H_2O}{Hr.-Cu.Ft.-\Delta H}$, (1)	1316	2245	1656	1530	1492
s, B.t.u./(lb.)($^{\circ}$F)	0.271	0.270	0.274	0.273	0.273
Heat Loss, %	-0.6	-1.2	-0.9	0.3	0.9
P_2, mm. Hg., top	784.4	784.0	783.8	783.6	783.4
(P_1-P_2), cm. water	-	1.94	-	-	-
t_f, $^{\circ}$F	197.1	186.4	200.6	200.8	201.7
t_{AS}, $^{\circ}$F	135.2	137.6	138.1	137.9	137.7
$(ha)'_{200}$, (2)	550	560	604	601	601
$(k'a)'_{200}$, (2)	1320	1470	1650	1530	1490
$(ha)_{70}$, (3)	265	259	279	278	279

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects.

(3) Gas-film temperature = 70°F. Corrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	31	32	33	35	36
L, lbs./(hr.)(sq.ft.)	1060	1600	2120	540	540
G, lbs./(hr.)(sq.ft.)	495	495	500	986	1010
T_2, ^oF, water at top	138	138	138	135	135
T_1, ^oF, water at bottom	138	138	138	135	135
t_2, ^oF, air at top	141.5	142	143.5	155.0	156.5
t_1, ^oF, air at bottom	623	630	650	592	601
H_2, lbs. H_2O/(lb. air)	0.1302	0.1319	0.1331	0.1159	0.1168
H_1, lbs. H_2O/(lb. air)	0.0099	0.0099	0.0099	0.0112	0.0094
$(H_t)'_{HT}$, ft., (1)	0.203	0.227	0.222	0.320	0.325
$(H_t)'_{MT}$, ft., (1)	0.374	0.334	0.318	0.426	0.410
$(ha)'_t$, $\frac{B.t.u.}{Hr.-Cu.Ft.-^oF}$, (1)	660	593	610	830	834
$(k'a)'_t$, $\frac{Lbs.\ H_2O}{Hr.-Cu.Ft.-\Delta H}$, (1)	1323	1482	1571	2310	3740
s, B.t.u./(lb.)(^oF)	0.271	0.272	0.271	0.269	0.268
Heat Loss, %	-1.4	-2.6	1.0	1.8	0.7
P_2, mm. Hg., top	781.0	780.9	780.7	779.3	774.7
(P_1-P_2), cm. water	-	-	-	-	-
t_f, ^oF	186.8	188.8	193.9	204.8	207.2
t_{AS}, ^oF	-	135.8	136.1	132.9	133.3
$(ha)'_{200}$, (2)	680	609	618	821	820
$(k'a)'_{200}$, (2)	-	1520	1600	2280	3680
$(ha)_{70}$, (3)	315	282	286	380	380

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200^{o}F. Uncorrected for end-effects.

(3) Gas-film temperature = 70^{o}F. Corrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	37	38	39	40	42
L, lbs./(hr.)(sq.ft.)	1060	1600	2100	2620	540
G, lbs./(hr.)(sq.ft.)	1000	1004	1002	1000	695
T_2, °F, water at top	135	135	135	135	135
T_1, °F, water at bottom	135	135	135	135	135
t_2, °F, air at top	150	148.5	146.5	145.5	148.3
t_1, °F, air at bottom	609	610	614	614	600
H_2, lbs. H_2O/(lb. air)	0.1215	0.1245	0.1235	0.1229	0.1211
H_1, lbs. H_2O/(lb. air)	0.0100	0.0110	0.0101	0.0098	0.0099
$(H_t)'_{HT}$, ft., (1)	0.290	0.281	0.268	0.262	0.281
$(H_t)'_{MT}$, ft., (1)	0.329	0.266	0.288	0.298	0.309
$(ha)'_t$, $\frac{\text{B.t.u.}}{\text{Hr.-Cu.Ft.-}^\circ\text{F}}$, (1)	930	966	1009	1029	665
$(k'a)'_t$, $\frac{\text{Lbs. H}_2\text{O}}{\text{Hr.-Cu.Ft.-}\Delta\text{H}}$, (1)	3040	3770	3480	3360	2245
s, B.t.u./(lb.)(°F)	0.269	0.270	0.270	0.270	0.269
Heat Loss, %	0.3	-0.3	0.0	1.2	-0.6
P_2, mm. Hg., top	774.3	774.0	774.6	774.4	778.9
(P_1-P_2), cm. water	-	-	-	-	-
t_f, °F	201.7	199.7	197.7	196.4	198.5
t_{AS}, °F	133.9	134.5	134.2	133.9	133.7
$(ha)'_{200}$, (2)	926	966	1015	1037	668
$(k'a)'_{200}$, (2)	3030	3770	3500	3390	2250
$(ha)_{70}$, (3)	429	447	470	481	309

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects.

(3) Gas-film temperature = 70°F. Corrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	43	44	45	46	48
L, lbs./(hr.)(sq.ft.)	1060	1600	2100	2620	540
G, lbs./(hr.)(sq.ft.)	697	697	702	702	355
T_2, °F, water at top	135	135	135	135	135
T_1, °F, water at bottom	135	135	135	135	135
t_2, °F, air at top	147.2	146.5	146.1	144.6	139.2
t_1, °F, air at bottom	603	610	605	600	590
H_2, lbs. H_2O/(lb. air)	0.1219	0.1230	0.1228	0.1229	0.1198
H_1, lbs. H_2O/(lb. air)	0.0098	0.0100	0.0110	0.0125	0.0100
$(H_t)'_{HT}$, ft., (1)	0.274	0.269	0.257	0.258	0.214
$(H_t)'_{MT}$, ft., (1)	0.300	0.258	0.279	0.257	0.316
$(ha)'_t$, $\frac{\text{B.t.u.}}{\text{Hr.-Cu.Ft.-}^\circ\text{F}}$, (1)	685	697	738	736	446
$(k'a)'_t$, $\frac{\text{Lbs. } H_2O}{\text{Hr.-Cu.Ft.-}\Delta H}$, (1)	2320	2700	2520	2730	1123
s, B.t.u./(lb.)(°F)	0.270	0.270	0.270	0.270	0.269
Heat Loss, %	-0.4	0.2	0.7	1.4	0.5
P_2, mm. Hg., top	776.4	776.1	775.9	775.8	785.0
(P_1-P_2), cm. water	-	-	-	-	1.21
t_f, °F	194.4	197.4	196.3	193.8	183.1
t_{AS}, °F	133.8	134.1	134.0	133.9	132.8
$(ha)'_{200}$, (2)	694	703	744	748	463
$(k'a)'_{200}$, (2)	2350	2720	2540	2770	1170
$(ha)_{70}$, (3)	321	326	344	346	214

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects.

(3) Gas-film temperature = 70°F. Corrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	49	50	51	52	59
L, lbs./(hr.)(sq.ft.)	1060	1610	2100	2620	540
G, lbs./(hr.)(sq.ft.)	351	362	358	359	487
T_2, °F, water at top	135	135	135	135	133.7
T_1, °F, water at bottom	135	135	135	135	133.7
t_2, °F, air at top	139.4	139.2	139.1	139.0	142.8
t_1, °F, air at bottom	585	585	583	590	575
H_2, lbs. H_2O/(lb. air)	0.1182	0.1188	0.1198	0.1187	0.1203
H_1, lbs. H_2O/(lb. air)	0.0093	0.0098	0.0098	0.0096	0.0129
$(H_t)'_{HT}$, ft., (1)	0.216	0.214	0.213	0.213	0.258
$(H_t)'_{MT}$, ft., (1)	0.347	0.352	0.333	0.355	0.222
$(ha)'_t$, $\frac{\text{B.t.u.}}{\text{Hr.-Cu.Ft.-}^\circ\text{F}}$, (1)	437	455	452	454	510
$(k'a)'_t$, $\frac{\text{Lbs. } H_2O}{\text{Hr.-Cu.Ft.-}\Delta H}$, (1)	1011	1029	1074	1011	2190
s, B.t.u./(lb.)(°F)	0.269	0.269	0.269	0.269	0.270
Heat Loss, %	-0.1	0.2	-0.6	0.7	-1.1
P_2, mm. Hg., top	784.6	779.9	779.4	779.4	-
(P_1-P_2), cm. water	2.35	2.44	2.72	2.78	1.7
t_f, °F	183.1	183.1	182.2	182.7	189.3
t_{AS}, °F	132.4	132.6	132.8	132.5	-
$(ha)'_{200}$, (2)	454	473	471	472	523
$(k'a)'_{200}$, (2)	1050	1070	1120	1050	-
$(ha)_{70}$, (3)	210	219	218	218	242

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects.

(3) Gas-film temperature = 70°F. Corrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	60	61
L, lbs./(hr.)(sq.ft.)	1060	1600
G, lbs./(hr.)(sq.ft.)	487	501
T_2, $^{\circ}$F, water at top	133.9	134.0
T_1, $^{\circ}$F, water at bottom	133.9	134.0
t_2, $^{\circ}$F, air at top	141.0	141.8
t_1, $^{\circ}$F, air at bottom	575	568
H_2, lbs. H_2O/(lb. air)	0.1209	0.1193
H_1, lbs. H_2O/(lb. air)	0.0128	0.0129
$(H_t)'_{HT}$, ft., (1)	0.242	0.249
$(H_t)'_{MT}$, ft., (1)	0.217	0.300
$(ha)'_t$, $\frac{B.t.u.}{Hr.-Cu.Ft.-^{\circ}F}$, (1)	543	543
$(k'a)'_t$, $\frac{Lbs.\ H_2O}{Hr.-Cu.Ft.-\Delta H}$, (1)	2245	1670
s, B.t.u./(lb.)($^{\circ}$F)	0.270	0.270
Heat Loss, %	-1.2	-2.0
P_2, mm. Hg., top	775.2	774.0
(P_1-P_2), cm. water	1.94	2.32
t_f, $^{\circ}$F	186.4	187.0
t_{AS}, $^{\circ}$F	132.8	132.8
$(ha)'_{200}$, (2)	560	560
$(k'a)'_{200}$, (2)	2320	1720
$(ha)_{70}$, (3)	259	260

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects.

(3) Gas-film temperature = 70°F. Corrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	64a	66	67	68	70
L, lbs./(hr.)(sq.ft.)	2100	1600	540	1060	2100
G, lbs./(hr.)(sq.ft.)	497	500	503	500	705
T_2, °F, water at top	134.4	135.5	135.1	135.4	135.4
T_1, °F, water at bottom	134.4	135.5	135.1	135.4	135.4
t_2, °F, air at top	140.9	141.3	143.2	143.2	144.0
t_1, °F, air at bottom	578	578	610	595	597
H_2, lbs. H_2O/(lb. air)	0.1223	0.1229	0.1253	0.1224	0.1212
H_1, lbs. H_2O/(lb. air)	0.0158	0.0154	0.0110	0.0110	0.0106
$(H_t)'_{HT}$, ft., (1)	0.237	0.230	0.246	0.245	0.251
$(H_t)'_{MT}$, ft., (1)	0.241	0.324	0.260	0.363	0.318
$(ha)'_t$, $\frac{\text{B.t.u.}}{\text{Hr.-Cu.Ft.-°F}}$, (1)	568	589	555	552	757
$(k'a)'_t$, $\frac{\text{Lbs. H2O}}{\text{Hr.-Cu.Ft.-}\Delta H}$, (1)	2060	1543	1933	1378	2200
s, B.t.u./(lb.)(°F)	0.271	0.271	0.270	0.270	0.270
Heat Loss, %	0.5	0.3	0.1	-0.6	0.4
P_2, mm. Hg., top	772.1	772.4	778.1	777.8	782.1
(P_1-P_2), cm. water	2.68	2.50	1.6	2.10	3.46
t_f, °F	186.1	186.6	192.5	190.8	192.3
t_{AS}, °F	133.5	133.7	134.4	133.7	133.5
$(ha)'_{200}$, (2)	586	608	561	565	771
$(k'a)'_{200}$, (2)	2130	1600	1960	1410	2240
$(ha)_{70}$, (3)	272	282	260	262	357
$(k'a)''_{70}$, $\frac{\text{Lbs. H2O}}{\text{Hr.-Cu.Ft.-}\Delta H}$, (4)	980	982	909	936	1325
$(k'a)''/L^{0.07}$	575	589	587	575	778

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.
(2) Gas-film temperature = 200°F. Uncorrected for end-effects.
(3) Gas-film temperature = 70°F. Corrected for end-effects.
(4) Based on t_{AS} rather than T_1 and T_2. (See Sample Calculations in the APPENDIX).

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

$S = 0.0885$ sq.ft. $z = 12$ in. 1-inch Carbon Raschig Rings

Run No.	72	73	75	76	77
L, lbs./(hr.)(sq.ft.)	1060	540	2100	1600	1060
G, lbs./(hr.)(sq.ft.)	700	698	1016	1000	1006
T_2, °F, water at top	134.7	134.4	134.4	134.8	134.9
T_1, °F, water at bottom	134.7	134.4	134.4	134.8	134.9
t_2, °F, air at top	144.9	146.6	145.5	146.3	149.0
t_1, °F, air at bottom	595	595	598	606	605
H_2, lbs. H_2O/(lb. air)	0.1199	0.1204	0.1200	0.1230	0.1244
H_1, lbs. H_2O/(lb. air)	0.0106	0.0118	0.0116	0.0116	0.0125
$(H_t)'_{HT}$, ft., (1)	0.263	0.275	0.268	0.269	0.285
$(H_t)'_{MT}$, ft., (1)	0.297	0.243	0.327	0.284	0.272
$(ha)'_t$, $\frac{\text{B.t.u.}}{\text{Hr.-Cu.Ft.-}^\circ\text{F}}$, (1)	716	684	1023	1004	950
$(k'a)'_t$, $\frac{\text{Lbs. H2O}}{\text{Hr.-Cu.Ft.-}\Delta\text{H}}$, (1)	2360	2870	3110	3520	3680
s, B.t.u./(lb.)(°F)	0.269	0.270	0.270	0.270	0.271
Heat Loss, %	0.6	0.9	2.5	0.9	-0.1
P_2, mm. Hg., top	782.7	783.1	777.1	772.8	769.5
(P_1-P_2), cm. water	2.58	2.10	6.46	-	5.12
t_f, °F	193.8	196.1	194.9	197.7	199.9
t_{AS}, °F	-	133.5	133.3	134.0	134.5
$(ha)'_{200}$, (2)	726	690	1036	1015	950
$(k'a)'_{200}$, (2)	2390	2900	3150	3550	3680
$(ha)_{70}$, (3)	336	319	480	470	440
$(k'a)''_{70}$, $\frac{\text{Lbs. H2O}}{\text{Hr.-Cu.Ft.-}\Delta\text{H}}$, (4)	1107	1133	1759	1626	1561
$(k'a)''/L^{0.07}$	685	733	1030	976	966

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects.

(3) Gas-film temperature = 70°F. Corrected for end-effects.

(4) Based on t_{AS} rather than T_1 and T_2. (See Sample Calculations in the APPENDIX).

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft.   z = 12 in. 1-inch Carbon Raschig Rings

Run No.	78	79	80
L, lbs./(hr.)(sq.ft.)	540	2100	1610
G, lbs./(hr.)(sq.ft.)	1006	350	350
T_2, $^{\circ}$F, water at top	135.0	134.9	135.6
T_1, $^{\circ}$F, water at bottom	135.0	134.9	135.6
t_2, $^{\circ}$F, air at top	154.2	139.5	140.0
t_1, $^{\circ}$F, air at bottom	607	610	615
H_2, lbs. H_2O/(lb. air)	0.1248	0.1237	0.1246
H_1, lbs. H_2O/(lb. air)	0.0122	0.0097	0.0086
$(H_t)'_{HT}$, ft., (1)	0.312	0.216	0.213
$(H_t)'_{MT}$, ft., (1)	0.264	0.234	0.216
$(ha)'_t$, $\frac{\text{B.t.u.}}{\text{Hr.-Cu.Ft.-}^{\circ}\text{F}}$, (1)	869	436	442
$(k'a)'_t$, $\frac{\text{Lbs. } H_2O}{\text{Hr.-Cu.Ft.-}\Delta H}$, (1)	3790	1496	1620
s, B.t.u./(lb.)($^{\circ}$F)	0.271	0.270	0.269
Heat Loss, %	-1.1	0.6	0.1
P_2, mm. Hg., top	770.8	780.6	780.8
(P_1-P_2), cm. water	-	2.30	-
t_f, $^{\circ}$F	205.6	185.6	186.3
t_{AS}, $^{\circ}$F	135.0	134.0	134.0
$(ha)'_{200}$, (2)	859	451	456
$(k'a)'_{200}$, (2)	3750	1550	1670
$(ha)_{70}$, (3)	398	209	211
$(k'a)''_{70}$, $\frac{\text{Lbs. } H_2O}{\text{Hr.-Cu.Ft.-}\Delta H}$, (4)	1408	740	774
$(k'a)''/L^{0.07}$	910	435	463

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.
(2) Gas-film temperature = 200°F. Uncorrected for end-effects.
(3) Gas-film temperature = 70°F. Corrected for end-effects.
(4) Based on t_{AS} rather than T_1 and T_2. (See Sample Calculations in the APPENDIX).

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 Sq.Ft.   z = 12 in. 1-inch Carbon Raschig Rings

Effect of Gas-Film Temperature

Run No.	81	82	83	84
L, lbs./(hr.)(sq.ft.)	1600	2100	540	1060
G, lbs./(hr.)(sq.ft.)	997	997	1000	1000
T_2, °F, water at top	113.6	113.9	114.4	114.4
T_1, °F, water at bottom	113.6	113.9	114.4	114.4
t_2, °F, air at top	120.5	119.5	126.5	123.0
t_1, °F, air at bottom	336	338	336	334
H_2, lbs. H_2O/(lb. air)	0.0629	0.0640	0.0635	0.0641
H_1, lbs. H_2O/(lb. air)	0.0108	0.0114	0.0129	0.0135
$(H_t)'_{HT}$, ft., (1)	0.288	0.271	0.343	0.309
$(H_t)'_{MT}$, ft., (1)	0.281	0.269	0.338	0.315
$(ha)'_t$, $\frac{\text{B.t.u.}}{\text{Hr.-Cu.Ft.-}^{\circ}\text{F}}$, (1)	890	945	750	834
$(k'a)'_t$, $\frac{\text{Lbs. H}_2\text{O}}{\text{Hr.-Cu.Ft.-}\Delta\text{H}}$, (1)	3550	3710	2960	3170
s, B.t.u./(lb.)(°F)	0.257	0.257	0.257	0.257
Heat Loss, %	-0.5	-0.2	-0.4	0.1
P_2, mm. Hg., top	780.3	778.3	780.0	779.6
(P_1-P_2), cm. water	3.5	3.65	3.20	3.60
t_f, °F	144.6	143.6	150.5	147.0
$(ha)'_t/L^{0.12}$	367	378	354	361

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 Sq.Ft.   z = 12 in. 1-inch Carbon Raschig Rings

Effect of Gas-Film Temperature

Run No.	85	86	87	88
L, lbs./(hr.)(sq.ft.)	540	1060	1600	2100
G, lbs./(hr.)(sq.ft.)	1010	1000	1000	1000
T_2, °F, water at top	95.0	94.9	94.3	94.6
T_1, °F, water at bottom	95.0	94.9	94.3	94.6
t_2, °F, air at top	102.3	100.9	99.8	98.5
t_1, °F, air at bottom	222	222	217	217
H_2, lbs. H_2O/(lb. air)	0.0346	0.0345	0.0335	0.0344
H_1, lbs. H_2O/(lb. air)	0.0064	0.0063	0.0069	0.0066
$(H_t)'_{HT}$, ft., (1)	0.350	0.328	0.322	0.290
$(H_t)'_{MT}$, ft., (1)	0.288	0.288	0.320	0.281
$(ha)'_t$, $\frac{\text{B.t.u.}}{\text{Hr.-Cu.Ft.-°F}}$, (1)	719	760	773	859
$(k'a)'_t$, $\frac{\text{Lbs. } H_2O}{\text{Hr.-Cu.Ft.-}\Delta H}$, (1)	3510	3470	3125	3570
s, B.t.u./(lb.)(°F)	0.249	0.249	0.249	0.249
Heat Loss, %	-0.1	0.1	-0.2	0.1
P_2, mm. Hg., top	783.5	783.2	782.8	780.3
(P_1-P_2), cm. water	2.90	3.35	4.00	7.40
t_f, °F	116.0	114.7	113.1	111.8
$(ha)'_t/L^{0.12}$	339	329	319	343

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

AIR-WATER TOWER

WATER COOLING RUNS

S = 0.0885 Sq.Ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	91	93	95	96	97	98	99
L, lbs./(hr.)(sq.ft.)	1065	1500	2010	1510	1150	2080	1500
G, lbs./(hr.)(sq.ft.)	498	702	993	1000	1003	702	501
T_2, °F, water at top	134.8	136.1	136.8	136.7	136.7	136.7	136.1
T_1, °F, water at bottom	97.5	105.5	102.5	101.5	104	110	102.5
t_2, °F, air at top	120.5	120.0	118	118	122	121.5	122.5
t_1, °F, air at bottom	115.5	106.5	111.5	145	225	90	86
H_2, lbs. H_2O/(lb. air)	0.0697	0.0700	0.0714	0.0648	0.0662	0.0789	0.0780
H_1, lbs. H_2O/(lb. air)	0.0052	0.0083	0.0055	0.0058	0.0060	0.0070	0.0074
Heat in	245.5	249.6	237.2	191.8	173.4	331.6	333.1
Heat out	246.1	249.8	237.6	191.1	173.2	331.1	333.3
P_2, mm. Hg., top	774.8	779.2	778.7	777.4	777.4	773.3	777.9
(P_1-P_2), cm. water	2.9	3.2	4.55	4.05	3.8	4.75	2.6
t_f, °F	113.0	110.5	111.5	118.8	133.3	107.9	106
t_i, °F	107.5	108	-	-	-	110	107
h_La, B.t.u./(hr.)(cu.ft.)(°F)	2220	2645	4960	3930	3750	3840	2650
$-h_La/(k'a)t_f$	2.01	1.88	2.40	1.95	1.90	2.70	2.40

AIR-WATER TOWER

WATER COOLING RUNS

S = 0.0885 Sq.Ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	100	101	102	103	104	105	106
L, lbs./(hr.)(sq.ft.)	540	750	525	700	1050	1050	1050
G, lbs./(hr.)(sq.ft.)	503	697	350	346	344	346	504
T_2, ^oF, water at top	138.2	136.8	135.8	136.1	136.1	135.7	135.2
T_1, ^oF, water at bottom	104.5	103.5	102.5	99.0	100	103.5	97
t_2, ^oF, air at top	122	121.5	123.5	123	124	125	119
t_1, ^oF, air at bottom	216.5	203	205	111	82.5	81.5	98
H_2, lbs. H_2O/(lb. air)	0.0671	0.0669	0.0694	0.0704	0.0789	0.0827	0.0683
H_1, lbs. H_2O/(lb. air)	0.0088	0.0126	0.0055	0.0057	0.0058	0.0088	0.0030
Heat in	169.0	168.4	203.7	235.8	-	336.0	234.2
Heat out	169.2	168.3	203.4	235.0	-	336.3	234.0
P_2, mm. Hg., top	780.3	772.9	787.5	788.0	781.8	772.8	777.5
(P_1-P_2), cm. water	1.8	2.35	1.60	1.70	1.80	1.80	2.75
t_f, ^oF	132.2	130.1	130	110.5	-	106.4	107
t_i, ^oF	111.5	112.5	110	105	-	109.5	105.5
h_La, B.t.u./(hr.)(cu.ft.)(^oF)	1632	2530	1100	1419	1720	2310	2225
$-h_La/(k'a)t_f$	1.46	1.80	1.18	1.62	-	2.7	-

AIR-WATER TOWER

WATER COOLING RUNS

S = 0.0885 Sq.Ft. z = 12 in. 1-inch Carbon Raschig Rings

Run No.	107	108	109	110	111	112	115
L, lbs./(hr.)(sq.ft.)	2000	1900	1050	1500	1400	1400	1040
G, lbs./(hr.)(sq.ft.)	1000	502	356	500	354	710	348
T_2, ^oF, water at top	134.8	128.7	136.1	135.0	128.0	136.1	134.4
T_1, ^oF, water at bottom	100.5	103.5	102	103	101	105	103
t_2, ^oF, air at top	119	119.5	125.5	123.5	120	121	125
t_1, ^oF, air at bottom	98.5	82	81.5	84.5	80	114	86
H_2, lbs. H_2O/(lb. air)	0.0647	0.0722	0.0818	0.0779	0.0712	0.0696	0.0786
H_1, lbs. H_2O/(lb. air)	0.0036	0.0079	0.0063	0.0086	0.0036	0.0042	0.0048
Heat in	225.6	386.7	326.2	331.1	395.1	229.9	-
Heat out	224.9	387.8	325.6	334.5	395.4	232.4	-
P_2, mm. Hg., top	774.8	778.2	774.6	777.8	779.0	769.7	792.3
(P_1-P_2), cm. water	7.2	6.2	1.8	2.55	2.0	4.95	1.45
t_f, ^oF	107	104.5	107.4	-	104.7	112.3	-
t_i, ^oF	-	107.5	110	-	108	107.5	-
h_La, B.t.u./(hr.)(cu.ft.)(^oF)	4190	2920	2310	2660	1940	3270	1689
$-h_La/(k'a)t_f$	2.04	2.6	2.63	-	2.2	2.3	-

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 6 in. 1-inch Carbon Raschig Rings

Run No.	121	122	123	124	125
L, lbs./(hr.)(sq.ft.)	500	1000	1500	2000	500
G, lbs./(hr.)(sq.ft.)	996	1002	993	993	506
T_2, $^\circ F$, water at top	135.5	135.5	134.5	135.0	135.0
T_1, $^\circ F$, water at bottom	135.5	135.5	134.5	135.0	135.0
t_2, $^\circ F$, air at top	189.0	171.0	167	161.5	172
t_1, $^\circ F$, air at bottom	644	655	622	633	640
H_2, lbs. H_2O/(lb. air)	0.1129	0.1180	0.1213	0.1242	0.1177
H_1, lbs. H_2O/(lb. air)	0.0043	0.0037	0.0072	0.0072	0.0054
$(H_t)'_{HT}$, ft., (1)	0.223	0.199	0.185	0.171	0.191
$(H_t)'_{MT}$, ft., (1)	-	-	-	-	-
$(ha)'_t$, $\frac{B.t.u.}{Hr.-Cu.Ft.-^\circ F}$, (1)	1193	1350	1446	1570	708
$(k'a)'_t$, $\frac{Lbs.\ H_2O}{Hr.-Cu.Ft.-\Delta H}$, (1)	-	-	-	-	-
s, B.t.u./(lb.)($^\circ$F)	0.266	0.267	0.267	0.270	0.268
Heat Loss, %	0.0	0.2	-0.1	0.0	0.3
P_2, mm. Hg., top	-	-	-	-	-
(P_1-P_2), cm. water	2.8	4.25	6.95	7.10	1.45
t_f, $^\circ F$	-	-	-	-	-
$(ha)'_{200}$, (2)	-	-	-	-	-
$(k'a)'_{200}$, (2)	-	-	-	-	-

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 6 in. 1-inch Carbon Raschig Rings

Run No.	126	127	129	130	131
L, lbs./(hr.)(sq.ft.)	1000	1500	1500	500	500
G, lbs./(hr.)(sq.ft.)	506	503	985	1000	700
T_2, $^{\circ}F$, water at top	135.0	135.0	135.6	136.8	135.6
T_1, $^{\circ}F$, water at bottom	135.0	135.0	135.6	136.8	135.6
t_2, $^{\circ}F$, air at top	166	167	168.0	186.0	172.4
t_1, $^{\circ}F$, air at bottom	638	638	643	658	650
H_2, lbs. H_2O/(lb. air)	0.1168	0.1164	0.1179	0.1154	0.1182
H_1, lbs. H_2O/(lb. air)	0.0040	0.0040	0.0034	0.0036	0.0034
$(H_t)'_{HT}$, ft., (1)	0.179	0.182	0.182	0.212	0.189
$(H_t)'_{MT}$, ft., (1)	-	-	0.179	0.254	0.205
$(ha)'_t$, $\frac{B.t.u.}{Hr.-Cu.Ft.-{}^{\circ}F}$, (1)	760	740	1433	1270	989
$(k'a)'_t$, $\frac{Lbs.\ H_2O}{Hr.-Cu.Ft.-\Delta H}$, (1)	-	-	5500	3940	3410
s, B.t.u./(lb.)($^{\circ}F$)	0.267	0.267	0.267	0.267	0.267
Heat Loss, %	0.7	0.7	-0.3	0.9	-0.2
P_2, mm. Hg., top	-	-	771.7	775.1	772.4
(P_1-P_2), cm. water	1.95	2.50	6.95	3.9	2.4
t_f, $^{\circ}F$	-	-	221.8	236.8	226.1
$(ha)'_{200}$, (2)	-	-	1365	1168	931
$(k'a)'_{200}$, (2)	-	-	5240	3620	3210

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

$S = 0.0885$ sq.ft. $z = 6$ in. 1-inch Carbon Raschig Rings

Run No.	132	133	134	135
L, lbs./(hr.)(sq.ft.)	1500	1500	500	500
G, lbs./(hr.)(sq.ft.)	702	500	500	348
T_2, ^{o}F, water at top	136.0	136.5	136.8	135.4
T_1, ^{o}F, water at bottom	136.0	136.5	136.8	135.4
t_2, ^{o}F, air at top	168.6	161.0	163.1	153.0
t_1, ^{o}F, air at bottom	653	677	672	665
H_2, lbs. H_2O/(lb. air)	0.1210	0.1264	0.1241	0.1247
H_1, lbs. H_2O/(lb. air)	0.0034	0.0022	0.0022	0.0022
$(H_t)'_{HT}$, ft., (1)	0.180	0.161	0.167	0.148
$(H_t)'_{MT}$, ft., (1)	0.198	0.171	0.199	0.143
$(ha)'_t$, $\frac{B.t.u.}{Hr.-Cu.Ft.-^{o}F}$, (1)	1046	833	805	637
$(k'a)'_t$, $\frac{Lbs.\ H_2O}{Hr.-Cu.Ft.-\Delta H}$, (1)	3550	2920	2510	2440
s, B.t.u./(lb.)(^{o}F)	0.268	0.269	0.268	0.269
Heat Loss, %	-0.9	0.0	0.4	0.9
P_2, mm. Hg., top	772.1	768.6	769.8	771.5
(P_1-P_2), cm. water	2.6	2.2	1.90	1.85
t_f, ^{o}F	223.6	220.0	220.8	210.4
$(ha)'_{200}$, (2)	992	797	767	622
$(k'a)'_{200}$, (2)	3360	2790	2390	2380

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200^{o}F. Uncorrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 9 in. 1-inch Carbon Raschig Rings

Run No.	136	137	138	139
L, lbs./(hr.)(sq.ft.)	500	1500	1500	500
G, lbs./(hr.)(sq.ft.)	1015	1000	697	697
T_2, °F, water at top	136.0	135.3	135.2	134.4
T_1, °F, water at bottom	136.0	135.3	135.2	134.4
t_2, °F, air at top	170.4	159.6	152.0	156.0
t_1, °F, air at bottom	652	647	640	642
H_2, lbs. H_2O/(lb. air)	0.1161	0.1205	0.1207	0.1186
H_1, lbs. H_2O/(lb. air)	0.0018	0.0018	0.0018	0.0018
$(H_t)'_{HT}$, ft., (1)	0.278	0.242	0.221	0.241
$(H_t)'_{MT}$, ft., (1)	0.223	-	-	0.225
$(ha)'_t$, $\frac{\text{B.t.u.}}{\text{Hr.-Cu.Ft.-°F}}$, (1)	984	1104	845	772
$(k'a)'_t$, $\frac{\text{Lbs. } H_2O}{\text{Hr.-Cu.Ft.-}\Delta H}$, (1)	4560	-	-	3100
s, B.t.u./(lb.)(°F)	0.267	0.268	0.268	0.267
Heat Loss, %	0.3	-1.3	-0.9	-0.1
P_2, mm. Hg., top	779.9	-	-	778.3
(P_1-P_2), cm. water	2.20	3.90	2.80	2.40
t_f, °F	225.0	215.2	206.8	211.3
$(ha)'_{200}$, (2)	930	1069	834	754
$(k'a)'_{200}$, (2)	4310	-	-	3030

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects.

AIR-WATER TOWER

ADIABATIC HUMIDIFYING RUNS

S = 0.0885 sq.ft. z = 9 in. 1-inch Carbon Raschig Rings

Run No.	140	141	142
L, lbs./(hr.)(sq.ft.)	500	1500	1500
G, lbs./(hr.)(sq.ft.)	507	500	353
T_2, $^{\circ}$F, water at top	134.7	134.8	135.0
T_1, $^{\circ}$F, water at bottom	134.7	134.8	135.0
t_2, $^{\circ}$F, air at top	149.0	146.5	143.0
t_1, $^{\circ}$F, air at bottom	650	640	635
H_2, lbs. H_2O/(lb. air)	0.1247	0.1240	0.1239
H_1, lbs. H_2O/(lb. air)	0.0040	0.0040	0.0040
$(H_t)'_{HT}$, ft., (1)	0.215	0.202	0.182
$(H_t)'_{MT}$, ft., (1)	-	0.176	0.191
$(ha)'_t$, $\frac{B.t.u.}{Hr.-Cu.Ft.-^{\circ}F}$, (1)	634	666	520
$(k'a)'_t$, $\frac{Lbs.\ H_2O}{Hr.-Cu.Ft.-\Delta H}$, (1)	-	2840	1850
s, B.t.u./(lb.)($^{\circ}$F)	0.269	0.269	0.269
Heat Loss, %	0.1	-0.4	-0.6
P_2, mm. Hg., top	-	772.8	774.8
(P_1-P_2), cm. water	2.0	2.45	2.35
t_f, $^{\circ}$F	204.6	202.0	194.6
$(ha)'_{200}$, (2)	626	664	527
$(k'a)'_{200}$, (2)	-	2830	1870

Notes: (1) Gas-film temperature = t_f. Uncorrected for end-effects.

(2) Gas-film temperature = 200°F. Uncorrected for end-effects

AIR-WATER TOWER

OXYGEN DESORPTION RUNS

$S = 0.0885$ sq.ft. $z = 30$ in. 1-inch Carbon Raschig Rings

Run No.	144	145	146
L, lbs./(hr.)(sq.ft.)	1500	2000	1000
G, lbs./(hr.)(sq.ft.)	120	120	120
T_2, °F, water at top	55	56	65.5
T_1, °F, water at bottom	55	57	65.5
C_2, lb. moles/(cu.ft.) $\times 10^5$, top	5.04	6.07	3.88
C_1, lb. moles/(cu.ft.) $\times 10^5$, bottom	2.41	2.42	1.92
C_e, lb. moles/(cu.ft.) $\times 10^5$	2.1	2.05	1.82
$(K_La)_T$, lb. moles/(hr.)(cu.ft.)(lb. mole)/(cu.ft.) (1)	22.0	30.0	19.4
$(K_La)_{25}$, lb. moles/(hr.)(cu.ft.)(lb. mole)/(cu.ft.) (2)	29.1	38.7	22.4

Notes: (1) At water temperature, T

(2) K_La corrected to water temperature of 25°C using the relation $K_La \propto e^{0.023T}$

where $T =$ °C (23)

Heat Transfer from Hot Oil to Cold Air

$S = 0.347$ sq.ft. $z = 29.1$ in. Ceramic Raschig Rings

Run No.	1	2	3	4	5	6	7	8
L, lbs./(hr.)(sq.ft.)	875	875	580	580	875	580	875	580
G', lbs./(hr.)(sq.ft.)	1020	858	858	1018	683	683	521	521
T_2, °F, oil at top	196.3	200.0	194.0	194.3	201.0	196.6	199.3	196.6
T_1, °F, oil at bottom	131.3	144.0	126.0	118.0	159.4	145.4	170.6	158.4
t_2, °F, air at top	191.0	194.3	181.0	179.7	195.3	186.8	194.0	189.2
t_1, °F, air at bottom	103.3	104.1	102.8	101.0	106.4	107.0	107.4	108.0
P_1-P_2, cm. of water	-	8.0	8.0	12.3	5.2	3.5	-	2.8
$G'Sc_G(t_2-t_1)$, BTU./(hr.)	7510	6500	5640	6740	5100	4570	3800	3560
$LSc_L(T_2-T_1)$, BTU./(hr.)	8690	7480	6010	6750	5550	4540	3840	3390
Heat loss, %	13.6	13.1	6.1	0.1	8.82	-0.6	1.0	-3.2
Ua', BTU./(hr.)(cu.ft.)(°F) (1)	707	472	393	506	298	259	194	184
Ua, BTU./(hr.)(cu.ft.)(°F) (2)	-	-	-	-	-	-	-	-

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_2-t_1) + LSc_L(T_2-T_1)}{2}$

(2) True coefficients, corrected for end effects are shown in Figures 31, 32.

Heat Transfer from Hot Oil to Cold Air

S = 0.347 sq.ft. z = 29.1 in. Ceramic Raschig Rings

Run No.	9	10	11	12	13	14	15	16
L, lbs./(hr.)(sq.ft.)	875	580	580	875	1160	1460	2040	2040
G', lbs./(hr.)(sq.ft.)	383	383	259	258	256	256	256	175
T_2, °F, oil at top	201.5	199.0	198.6	202.6	206.0	205.2	208.0	209.3
T_1, °F, oil at bottom	182.5	169.3	179.0	189.2	196.0	197.2	202.0	205.3
t_2, °F, air at top	196.2	189.3	191.0	196.6	200.3	199.2	202.8	203.9
t_1, °F, air at bottom	111.5	110.2	109.6	113.0	116.4	117.1	118.9	116.4
P_1-P_2, cm. of water	-	2.0	1.1	1.2	1.0	1.1	-	0.9
$G'Sc_G(t_2-t_1)$, B.t.u./(hr.)	2720	2550	1770	1810	1800	1770	1805	1285
$LSc_L(T_2-T_1)$, B.t.u./(hr.)	2940	2630	1735	1790	1770	1780	1865	1245
Heat loss, %	7.5	3.0	-2.0	-1.1	-1.7	0.5	3.2	-3.2
Ua', B.t.u./(hr.)(cu.ft.)(°F) (1)	129	112	75	78	77	74	77	50
Ua, B.t.u./(hr.)(cu.ft.)(°F) (2)	-	-	-	-	-	-	-	-

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_2-t_1) + LSc_L(T_2-T_1)}{2}$

(2) True coefficients, corrected for end effects are shown in Figures 31, 32.

Heat Transfer from Hot Oil to Cold Air

S = 0.347 sq.ft. z = 29.1 in. Ceramic Raschig Rings

Run No.	17	18	19	20	21	22	23	24
L, lbs./(hr.)(sq.ft.)	1460	1160	875	1160	1160	1460	1160	1460
G', lbs./(hr.)(sq.ft.)	175	175	175	384	860	860	680	680
T_2, °F, oil at top	208.0	206.1	205.5	206.7	201.6	202.6	200.0	202.6
T_1, °F, oil at bottom	202.3	199.0	196.3	191.7	161.0	169.2	169.7	178.3
t_2, °F, air at top	204.0	201.8	201.1	201.6	199.2	201.2	197.0	201.0
t_1, °F, air at bottom	117.4	117.0	116.4	115.1	100.4	100.6	105.9	107.6
P_1-P_2, cm. of water	0.8	0.7	0.6	1.3	11.9	12.9	6.3	-
$G'Sc_G(t_2-t_1)$, B.t.u./(hr.)	1270	1248	1240	2790	7140	7260	5210	5340
$LSc_L(T_2-T_1)$, B.t.u./(hr.)	1270	1258	1230	2660	7190	7450	5360	5440
Heat loss, %	0.0	0.8	-0.8	-4.8	0.7	2.5	2.8	1.8
Ua', B.t.u./(hr.)(cu.ft.)(°F) (1)	57	56	56	123	471	511	315	351
Ua, B.t.u./(hr.)(cu.ft.)(°F) (2)	-	-	-	-	-	-	-	-

Notes: (1) Ua' based on measured packed height and $q_{ave} = \frac{G'Sc_G(t_2-t_1) + LSc_L(T_2-T_1)}{2}$

(2) True coefficients, corrected for end effects are shown in Figures 31, 32.

Heat Transfer from Hot Oil to Cold Air

S = 0.347 sq.ft. z = 29.1 in. Ceramic Raschig Rings

Run No.	25	26	27	28	30	31	32	33
L, lbs./(hr.)(sq.ft.)	2040	2040	1460	1460	2040	2040	1460	1160
G', lbs./(hr.)(sq.ft.)	680	512	512	389	388	324	322	320
T_2, °F, oil at top	203.0	204.6	204.1	203.0	205.3	206.0	201.6	201.0
T_1, °F, oil at bottom	186.8	192.3	186.4	189.6	196.0	198.0	191.0	188.6
t_2, °F, air at top	202.3	203.3	201.5	199.6	203.0	203.1	196.3	196.3
t_1, °F, air at bottom	111.0	111.4	110.9	110.3	114.5	111.4	110.4	115.4
P_1-P_2, cm. of water	8.0	4.4	4.0	2.6	2.8	2.2	2.1	2.0
$G'Sc_G(t_2-t_1)$, B.t.u./(hr.)	5220	3940	3900	2915	2910	2490	2320	2170
$LSc_L(T_2-T_1)$, B.t.u./(hr.)	5140	3860	3940	2980	2900	2490	2360	2175
Heat loss, %	-0.4	-2.0	1.0	2.2	-0.3	0.0	1.7	0.2
Ua', B.t.u./(hr.)(cu.ft.)(°F) (1)	385	241	216	145	155	119	101	102
Ua, B.t.u./(hr.)(cu.ft.)(°F) (2)	-	-	-	-	-	-	-	-

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_2-t_1) + LSc_L(T_2-T_1)}{2}$

(2) True coefficients, corrected for end effects are shown in Figures 31, 32.

Heat Transfer from Hot Oil to Cold Air

S = 0.347 sq.ft.   z = 29.1 in. Ceramic Raschig Rings

Run No.	34	35	36	37	38	39	40
L, lbs./(hr.)(sq.ft.)	875	580	580	875	875	2440	2440
G', lbs./(hr.)(sq.ft.)	321	327	391	388	860	685	385
T_2, °F, oil at top	197.6	192.0	190.3	198.3	196.3	203.0	206.3
T_1, °F, oil at bottom	181.1	168.6	163.3	178.6	148.7	188.3	199.0
t_2, °F, air at top	192.6	184.7	183.7	193.0	192.0	202.3	204.0
t_1, °F, air at bottom	113.2	108.0	109.4	111.7	103.5	105.9	116.0
P_1-P_2, cm. of water	1.8	1.8	2.1	2.1	-	9.7	3.0
$G'Sc_G(t_2-t_1)$, B.t.u./(hr.)	2140	2100	2430	2630	6380	5540	2840
$LSc_L(T_2-T_1)$, B.t.u./(hr.)	2205	2070	2390	2630	6370	5480	2720
Heat loss, %	3.0	-1.4	-1.7	0.0	-0.1	-1.1	-4.2
Ua', B.t.u./(hr.)(cu.ft.)(°F) (1)	106	97	127	128	435	384	143
Ua, B.t.u./(hr.)(cu.ft.)(°F) (2)	-	-	-	-	-	-	-

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_2-t_1) + LSc_L(T_2-T_1)}{2}$

(2) True coefficients, corrected for end effects are shown in Figures 31, 32.

Heat Transfer from Hot Oil to Cold Air

S = 0.347 sq.ft. z = 18.0 in. Ceramic Raschig Rings

Run No.	72	73	74	75	76	77	78
L, lbs./(hr.)(sq.ft.)	580	875	1160	1460	2040	2040	2040
G', lbs./(hr.)(sq.ft.)	841	841	841	841	841	261	465
T_2, °F, oil at top	193.7	197.6	202.6	202.0	203.0	206.6	205.3
T_1, °F, oil at bottom	135.0	148.4	164.0	164.3	179.0	199.6	193.0
t_2, °F, air at top	166.3	186.8	196.4	198.0	200.3	199.6	201.3
t_1, °F, air at bottom	94.1	94.6	95.3	95.3	95.7	104.0	100.4
P_1-P_2, cm. of water	7.1	8.0	9.3	10.2	12.2	-	3.3
$G'Sc_G(t_2-t_1)$, B.t.u./(hr.)	5100	6510	7140	7250	7400	2095	3900
$LSc_L(T_2-T_1)$, B.t.u./(hr.)	5200	6560	6840	7270	7460	2175	3820
Heat loss, %	1.9	-0.7	4.2	-0.3	0.8	3.8	0.5
Ua', B.t.u./(hr.)(cu.ft.)(°F) (1)	293	468	513	579	606	121	263
Ua, B.t.u./(hr.)(cu.ft.)(°F) (2)	-	-	-	-	-	-	-

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_2-t_1) + LSc_L(T_2-T_1)}{2}$

(2) True coefficients, corrected for end effects are shown in Figures 31, 32.

Heat Transfer from Hot Oil to Cold Air

S = 0.347 sq.ft. z = 18.0 in. Ceramic Raschig Rings

Run No.	79	80	81	82	83	84	85
L, lbs./(hr.)(sq.ft.)	1460	1160	1460	1160	580	875	580
G', lbs./(hr.)(sq.ft.)	466	466	266	266	266	460	460
T_2, °F, oil at top	204.6	203.3	205.6	205.3	196.0	201.6	196.6
T_1, °F, oil at bottom	186.1	181.7	195.6	193.0	195.7	175.0	163.0
t_2, °F, air at top	199.0	196.0	197.0	197.6	184.1	171.3	182.0
t_1, °F, air at bottom	97.0	96.9	101.9	102.1	101.2	98.4	98.5
P_1-P_2, cm. of water	3.0	2.6	1.8	1.6	1.2	2.5	2.2
$G'Sc_G(t_2-t_1)$, B.t.u./(hr.)	4000	3880	2145	2130	1850	3580	2960
$LSc_L(T_2-T_1)$, B.t.u./(hr.)	4120	3820	2230	2175	1820	3560	2975
Heat loss, %	3.0	1.6	4.4	2.1	1.6	0.6	0.5
Ua', B.t.u./(hr.)(cu.ft.)(°F) (1)	257	233	118	118	103	207	146
Ua, B.t.u./(hr.)(cu.ft.)(°F) (2)	-	-	-	-	-	-	-

Notes: (1) Ua' based on measured packed height and $q_{ave} = \frac{G'Sc_G(t_2-t_1) + LSc_L(T_2-T_1)}{2}$

(2) True coefficients, corrected for end effects are shown in Figures 31, 32.

Heat Transfer from Hot Oil to Cold Air

$S = 0.347$ sq.ft. $z = 14.0$ in. Ceramic Raschig Rings

Run No.	93	94	95	96	97	98
L, lbs./(hr.)(sq.ft.)	1460	1160	2040	875	1460	1160
G', lbs./(hr.)(sq.ft.)	846	846	840	836	179	179
T_2, °F, oil at top	202.5	202.3	203.3	199.6	206.6	206.3
T_1, °F, oil at bottom	170.6	114.0	181.0	155.3	200.3	198.6
t_2, °F, air at top	196.0	192.6	199.6	186.1	197.3	198.6
t_1, °F, air at bottom	96.8	96.8	99.8	100.4	106.0	107.5
P_1-P_2, cm. of water	8.8	7.9	12.5	7.3	1.1	1.0
$G'Sc_G(t_2-t_1)$, B.t.u./(hr.)	7010	6800	7030	6020	1370	1370
$LSc_L(T_2-T_1)$, B.t.u./(hr.)	7110	6790	6940	5940	1400	1370
Heat loss, %	1.4	-0.1	-1.3	-1.3	2.1	0.0
Ua', B.t.u./(hr.)(cu.ft.)(°F) (1)	629	566	686	500	95	98
Ua, B.t.u./(hr.)(cu.ft.)(°F) (2)	-	-	-	-	-	-

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_2-t_1) + LSc_L(T_2-T_1)}{2}$

(2) True coefficients, corrected for end effects are shown in Figures 31, 32.

Heat Transfer from Hot Air to Cold Oil

S = 0.347 sq.ft.   z = 29.1 in. Ceramic Raschig Rings

Run No.	41	42	43	44	45	46	47a	47b
L, lbs./(hr.)(sq.ft.)	1460	1160	1460	2040	1160	1460	2040	1160
G', lbs./(hr.)(sq.ft.)	466	470	470	470	756	756	756	472
T_2, °F, oil at top	67.7	66.6	68.0	69.4	71.1	73.6	69.6	68.7
T_1, °F, oil at bottom	94.7	99.6	94.6	87.8	116.0	109.0	95.7	105.4
t_2, °F, air at top	72.5	70.0	70.0	70.4	75.0	75.4	70.4	71.1
t_1, °F, air at bottom	220.0	218.5	219.1	218.5	201.8	203.0	200.6	236.0
P_1-P_2, cm. of water	4.8	4.2	4.4	6.0	-	12.9	16.8	4.5
$G'Sc_G(t_1-t_2)$, BTU./(hr.)	5790	5850	5880	5850	8020	8050	8210	6400
$LSc_L(T_1-T_2)$, BTU./(hr.)	6010	5840	5800	6000	7950	7900	8140	6490
Heat loss, %	-3.7	0.2	1.4	2.5	0.8	1.8	0.9	-1.4
Ua', BTU./(hr.)(cu.ft.)(°F) (1)	189	213	234	262	359	407	456	237
Ua, BTU./(hr.)(cu.ft.)(°F) (2)	127	143	157	176	241	273	306	159

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_1-t_2) + LSc_L(T_1-T_2)}{2}$

(2) Ua is the true coefficient, corrected for end effects.

Heat Transfer from Hot Air to Cold Oil

$S = 0.347$ sq.ft. $z = 29.1$ in. Ceramic Raschig Rings

Run No.	48	49	50	51	52	53
L, lbs./(hr.)(sq.ft.)	1460	2040	1160	1460	2040	1460
G', lbs./(hr.)(sq.ft.)	461	461	252	252	258	258
T_2, °F, oil at top	71.1	70.4	63.1	65.4	63.0	62.6
T_1, °F, oil at bottom	99.5	91.7	85.0	82.3	75.4	80.6
t_2, °F, air at top	72.5	71.4	64.5	66.1	63.7	63.7
t_1, °F, air at bottom	242.3	244.2	246.0	247.1	241.6	243.8
P_1-P_2, cm. of water	4.4	5.3	2.3	2.5	2.3	2.1
$G'Sc_G(t_1-t_2)$, BTU./(hr.)	6560	6675	3840	3830	3845	3900
$LSc_L(T_1-T_2)$, BTU./(hr.)	6350	6650	3870	3780	3850	4000
Heat loss, %	3.1	0.4	-0.8	1.3	-0.1	-2.5
Ua', BTU./(hr.)(cu.ft.)(°F) (1)	255	263	136	150	151	145
Ua, BTU./(hr.)(cu.ft.)(°F) (2)	171	177	91	101	101	97

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_1-t_2) + LSc_L(T_1-T_2)}{2}$

(2) Ua is the true coefficient, corrected for end effects.

Heat Transfer from Hot Air to Cold Oil

S = 0.347 sq.ft. z = 29.1 in. Ceramic Raschig Rings

Run No.	54	55	56	57	58	59
L, lbs./(hr.)(sq.ft.)	875	580	580	875	875	580
G', lbs./(hr.)(sq.ft.)	257	253	454	454	762	762
T_2, °F, oil at top	63.3	63.5	66.7	69.7	72.5	69.3
T_1, °F, oil at bottom	91.0	105.0	139.3	121.3	128.4	149.6
t_2, °F, air at top	65.1	67.4	77.5	74.3	80.0	87.8
t_1, °F, air at bottom	239.5	242.3	250.5	250.0	202.1	203.2
P_1-P_2, cm. of water	2.5	1.8	2.7	2.6	9.9	7.8
$G'Sc_G(t_1-t_2)$, BTU./(hr.)	3770	3710	6610	6710	7800	7380
$LSc_L(T_1-T_2)$, BTU./(hr.)	3700	3670	6940	6900	7460	7110
Heat loss, %	1.8	1.1	-2.6	-2.8	4.3	3.6
Ua', BTU./(hr.)(cu.ft.)(°F) (1)	133	118	189	217	313	260
Ua, BTU./(hr.)(cu.ft.)(°F) (2)	89	79	127	146	210	174

Notes: (1) Ua' based on measured packed height and $q_{ave} = \frac{G'Sc_G(t_1-t_2) + LSc_L(T_1-T_2)}{2}$

(2) Ua is the true coefficient, corrected for end effects.

Heat Transfer from Hot Air to Cold Oil

S = 0.347 sq.ft. z = 18.0 in. Ceramic Raschig Rings

Run No.	60	61	62	63	64	65
L, lbs./(hr.)(sq.ft.)	875	580	1160	1460	2040	2040
G', lbs./(hr.)(sq.ft.)	761	761	761	761	761	261
T_2, °F, oil at top	70.0	69.7	73.6	74.3	72.9	66.1
T_1, °F, oil at bottom	125.4	147.6	116.0	108.1	98.2	78.2
t_2, °F, air at top	83.0	94.0	82.0	80.0	76.1	70.0
t_1, °F, air at bottom	202.5	203.7	199.5	201.9	201.7	240.0
P_1-P_2, cm. of water	7.8	5.9	8.7	9.9	14.1	2.4
$G'Sc_G(t_1-t_2)$, BTU./(hr.)	7640	7000	7500	7780	8010	3750
$LSc_L(T_1-T_2)$, BTU./(hr.)	7400	6900	7490	7540	7810	3770
Heat loss, %	3.1	1.4	0.1	3.2	2.5	-0.5
Ua', BTU./(hr.)(cu.ft.)(°F) (1)	399	352	440	468	522	170
Ua, BTU./(hr.)(cu.ft.)(°F) (2)	222	196	245	261	290	95

Notes: (1) Ua' based on measured packed height and $q_{ave} = \frac{G'Sc_G(t_1-t_2) + LSc_L(T_1-T_2)}{2}$

(2) Ua is the true coefficient, corrected for end effects.

Heat Transfer from Hot Air to Cold Oil

$S = 0.347$ sq.ft. $z = 18.0$ in. Ceramic Raschig Rings

Run No.	66	67	68	69	70	71
L, lbs./(hr.)(sq.ft.)	1460	1160	875	580	580	875
G', lbs./(hr.)(sq.ft.)	262	261	261	260	467	467
T_2, °F, oil at top	63.5	62.6	62.3	63.5	66.4	66.7
T_1, °F, oil at bottom	79.6	83.0	90.3	107.1	124.7	108.4
t_2, °F, air at top	68.4	69.4	70.4	73.2	83.6	77.1
t_1, °F, air at bottom	237.0	241.3	243.0	245.4	220.1	221.9
P_1-P_2, cm. of water	2.8	2.6	1.8	1.6	2.3	2.8
$G'Sc_G(t_1-t_2)$, BTU./(hr.)	3700	3760	3780	3750	5380	5660
$LSc_L(T_1-T_2)$, BTU./(hr.)	3590	3610	3740	3860	5150	5560
Heat loss, %	2.8	3.9	1.1	-2.9	4.3	1.7
Ua', BTU./(hr.)(cu.ft.)(°F) (1)	160	147	145	144	222	250
Ua, BTU./(hr.)(cu.ft.)(°F) (2)	89	82	81	80	124	139

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_1-t_2) + LSc_L(T_1-T_2)}{2}$

(2) Ua is the true coefficient, corrected for end effects.

Heat Transfer from Hot Air to Cold Oil

S = 0.347 sq.ft. z = 14.0 in. Ceramic Raschig Rings

Run No.		86	87	88	89	90	91	92
L, lbs./(hr.)(sq.ft.)		875	580	875	580	1460	1460	2040
G', lbs./(hr.)(sq.ft.)		765	772	260	259	260	770	770
T_2, ^oF, oil at top		68.0	68.6	62.6	62.6	65.7	70.4	72.9
T_1, ^oF, oil at bottom		121.7	141.3	88.5	101.9	81.0	103.7	97.8
t_2, ^oF, air at top		84.3	99.2	72.2	76.0	74.0	78.1	77.5
t_1, ^oF, air at bottom		200.0	201.0	232.2	237.0	231.0	197.3	196.6
P_1-P_2, cm. of water		6.9	5.5	1.9	1.7	2.2	9.4	12.3
$G'Sc_G(t_1-t_2)$, BTU./(hr.)		7390	6580	3480	3490	3420	7690	7740
$LSc_L(T_1-T_2)$, BTU./(hr.)		7190	6450	3460	3480	3410	7450	7750
Heat loss, %		2.7	2.0	0.6	0.3	0.3	3.2	0.1
Ua', BTU./(hr.)(cu.ft.)(^oF)	(1)	456	370	173	164	173	544	624
Ua, BTU./(hr.)(cu.ft.)(^oF)	(2)	226	183	86	81	86	266	309

Notes: (1) Ua' based on measured packed height and $q_{ave} = \dfrac{G'Sc_G(t_1-t_2) + LSc_L(T_1-T_2)}{2}$

(2) Ua is the true coefficient, corrected for end effects.

Heat Transfer from Hot Oil(1) to Cold Air

S = 0.347 sq.ft. z = 29.1 in. Ceramic Raschig Rings

Run No.	1	2	3	4	5	6	7	8
L, lbs./(hr.)(sq.ft.)	650	970	285	2170	1360	1020	2170	1620
G', lbs./(hr.)(sq.ft.)	795	795	795	514	514	514	312	312
T_2, °F, oil at top	190.0	192.0	174.1	200.8	198.5	197.6	203.0	203.0
T_1, °F, oil at bottom	136.1	149.3	113.0	184.0	175.6	170.0	190.4	188.7
t_2, °F, air at top	175.9	184.4	140.2	189.1	189.5	188.1	181.4	189.5
t_1, °F, air at bottom	101.3	102.2	99.9	113.8	112.9	112.9	125.6	127.4
$G'Sc_G(t_2-t_1)$, Btu./(hr.)	5010	5510	2700	3280	3310	3240	1480	1640
$LSc_L(T_2-T_1)$, Btu./(hr.)	5380	6370	2680	5550	4770	4310	4180	3530
Heat Loss, %	7.3	13.5	0	41	31	25	65	54
Ua', Btu./(hr.)(cu.ft.)(°F) (2)	263	345	146	196	203	211	125	153
Ua', Btu./(hr.)(cu.ft.)(°F) (3)	244	298	146	116	141	159	44.4	71

Notes: (1) Data of Philoon (35). Not used because of large heat losses.

(2) Coefficient based on heat lost by oil.

(3) Coefficient based on heat gained by air.

Heat Transfer from Hot Oil(1) to Cold Air

S = 0.347 sq.ft.   z = 29.1 in. Ceramic Raschig Rings

Run No.	9	10	11	12	13	14	15	16
L, lbs./(hr.)(sq.ft.)	1210	810	500	1610	1610	2170	1210	1200
G', lbs./(hr.)(sq.ft.)	312	312	312	514	790	790	502	797
T_2, °F, oil at top	201.3	196.7	192.6	200.3	199.4	200.3	199.1	197.6
T_1, °F, oil at bottom	184.4	173.3	160.7	180.8	173.4	180.1	175.9	162.4
t_2, °F, air at top	189.1	182.5	177.7	182.0	194.5	195.8	188.4	192.1
t_1, °F, air at bottom	125.6	122.0	120.0	116.5	118.3	120.2	117.5	116.6
$G'Sc_G(t_2-t_1)$, Btu./(hr.)	1670	1600	2000	3210	5100	5040	3020	5090
$LSc_L(T_2-T_1)$, Btu./(hr.)	3100	2900	2450	4790	6390	6700	4300	6140
Heat Loss, %	45	45	18	33	20	25	30	17
Ua', Btu./(hr.)(cu.ft.)(°F) (2)	124	104	112	206	367	369	180	374
Ua', Btu./(hr.)(cu.ft.)(°F) (3)	67	58	92	138	292	278	127	310

Notes: (1) Data of Philoon (35). Not used because of large heat losses.

(2) Coefficient based on heat lost by oil.

(3) Coefficient based on heat gained by air.

Heat Transfer from Hot Oil(1) to Cold Air

S = 0.347 sq.ft. z = 29.1 in. Ceramic Raschig Rings

Run No.	17	18	19	20	21	22	23	24
L, lbs./(hr.)(sq.ft.)	1200	1200	1650	1650	1650	2160	2140	2160
G', lbs./(hr.)(sq.ft.)	329	1045	339	522	798	329	522	798
T_2, °F, oil at top	200.0	196.7	200.0	199.4	197.0	202.3	200.8	198.7
T_1, °F, oil at bottom	183.7	156.1	185.0	178.4	168.7	190.6	186.2	177.7
t_2, °F, air at top	191.5	191.3	186.5	191.5	191.5	190.7	184.0	182.6
t_1, °F, air at bottom	123.8	108.7	117.7	109.4	112.0	122.0	117.5	117.5
$G'Sc_G(t_2-t_1)$, Btu./(hr.)	1840	7300	2020	3620	5390	1915	3290	5100
$LSc_L(T_2-T_1)$, Btu./(hr.)	3190	7450	3780	5280	7090	3890	4810	6950
Heat Loss, %	42	2	46	31	24	49	31	27
Ua', Btu./(hr.)(cu.ft.)(°F) (2)	135	417	134	251	378	120	212	345
Ua', Btu./(hr.)(cu.ft.)(°F) (3)	78	408	72	172	287	59	145	253

Notes: (1) Data of Philoon (35). Not used because of large heat losses.

(2) Coefficient based on heat lost by oil.

(3) Coefficient based on heat gained by air.

Heat Transfer from Hot Oil(1) to Cold Air

S = 0.347 sq.ft. z = 29.1 in. Ceramic Raschig Rings

Run No.	25	26	27	28	29	30	31	32
L, lbs./(hr.)(sq.ft.)	2160	798	797	800	500	490	495	500
G', lbs./(hr.)(sq.ft.)	905	331	490	798	319	494	798	1175
T_2, °F, oil at top	198.5	195.6	194.7	193.5	194.0	193.3	192.2	189.6
T_1, °F, oil at bottom	174.6	169.3	161.3	145.2	158.4	146.8	125.2	115.4
t_2, °F, air at top	184.9	182.0	184.3	185.2	176.0	175.1	173.9	163.9
t_1, °F, air at bottom	107.0	117.3	111.2	108.5	115.3	110.3	106.7	103.1
$G'Sc_G(t_2-t_1)$, Btu./(hr.)	6730	1820	3050	5220	1640	2650	4670	6070
$LSc_L(T_2-T_1)$, Btu./(hr.)	7880	3230	4080	5920	2730	3520	5130	5710
Heat Loss, %	15	44	25	12	40	25	9	-
Ua', Btu./(hr.)(cu.ft.)(°F) (2)	426	134	190	368	112	158	329	374
Ua', Btu./(hr.)(cu.ft.)(°F) (3)	364	75	142	324	68	119	229	397

Notes: (1) Data of Philoon (35). Not used because of large heat losses.

(2) Coefficient based on heat lost by oil.

(3) Coefficient based on heat gained by air.

189 omitted in numbering

AIR ORIFICE CALIBRATIONS

4-INCH TOWER

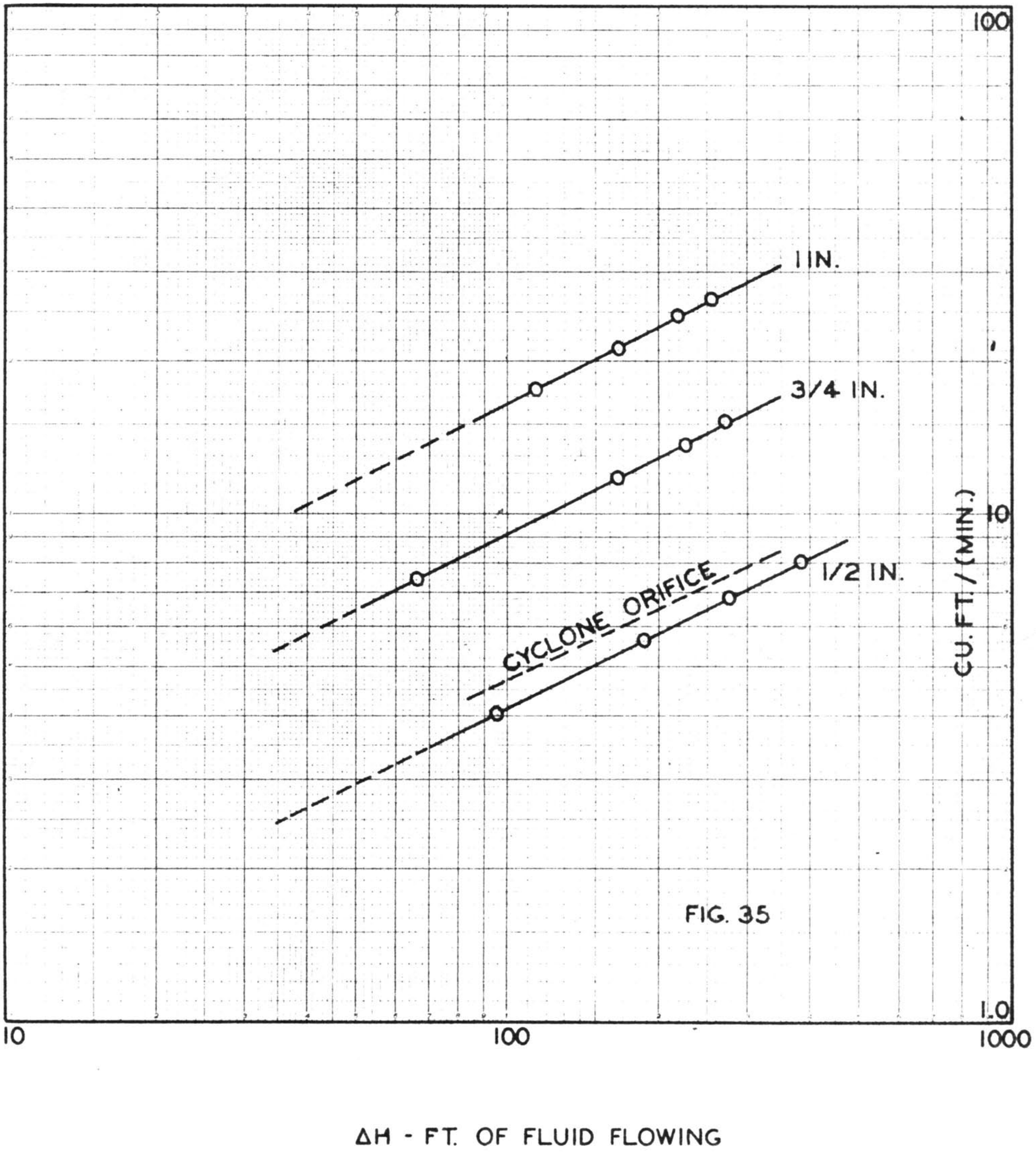


FIG. 35

WATER ORIFICE CALIBRATIONS
4-INCH TOWER

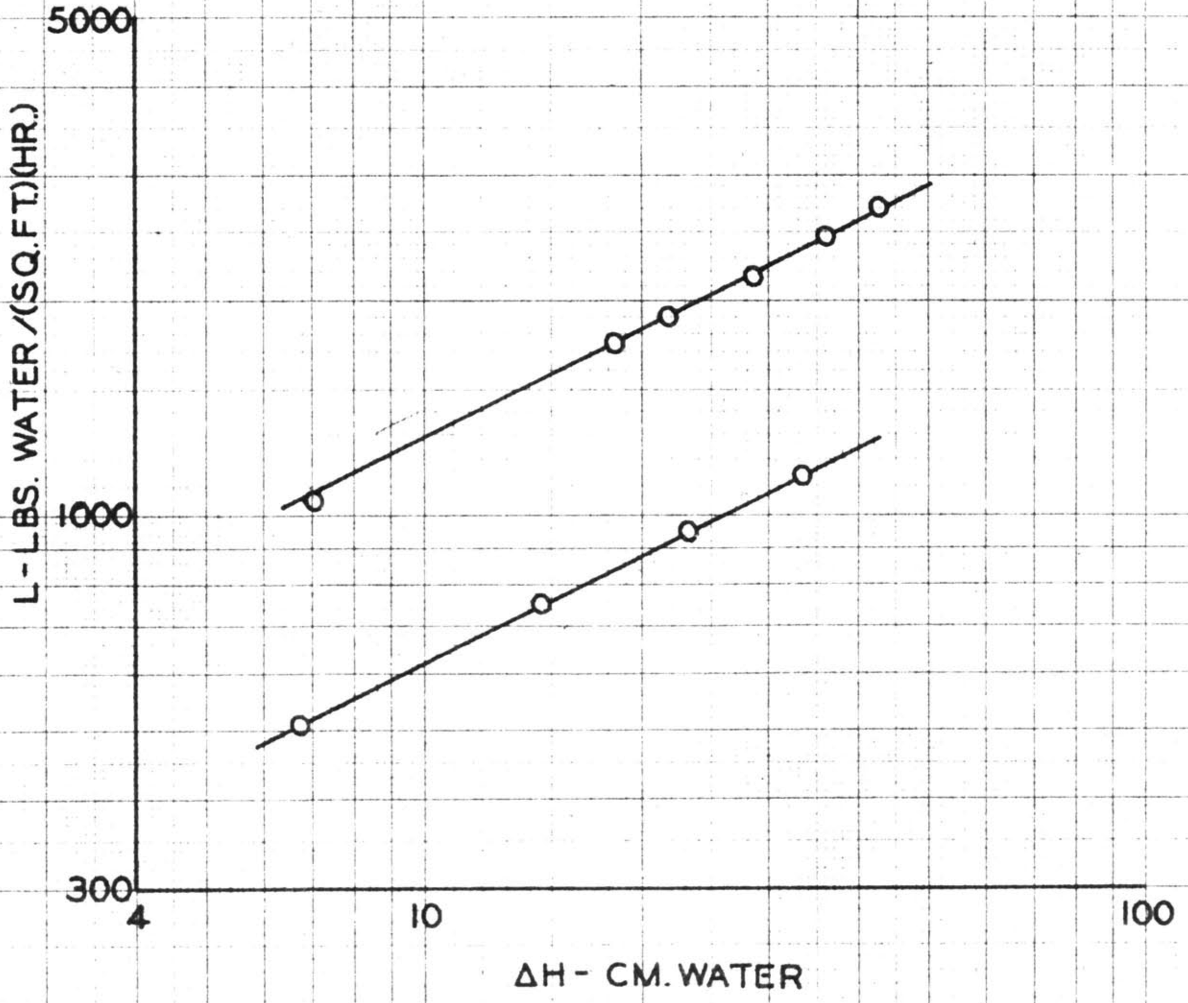


FIG. 36

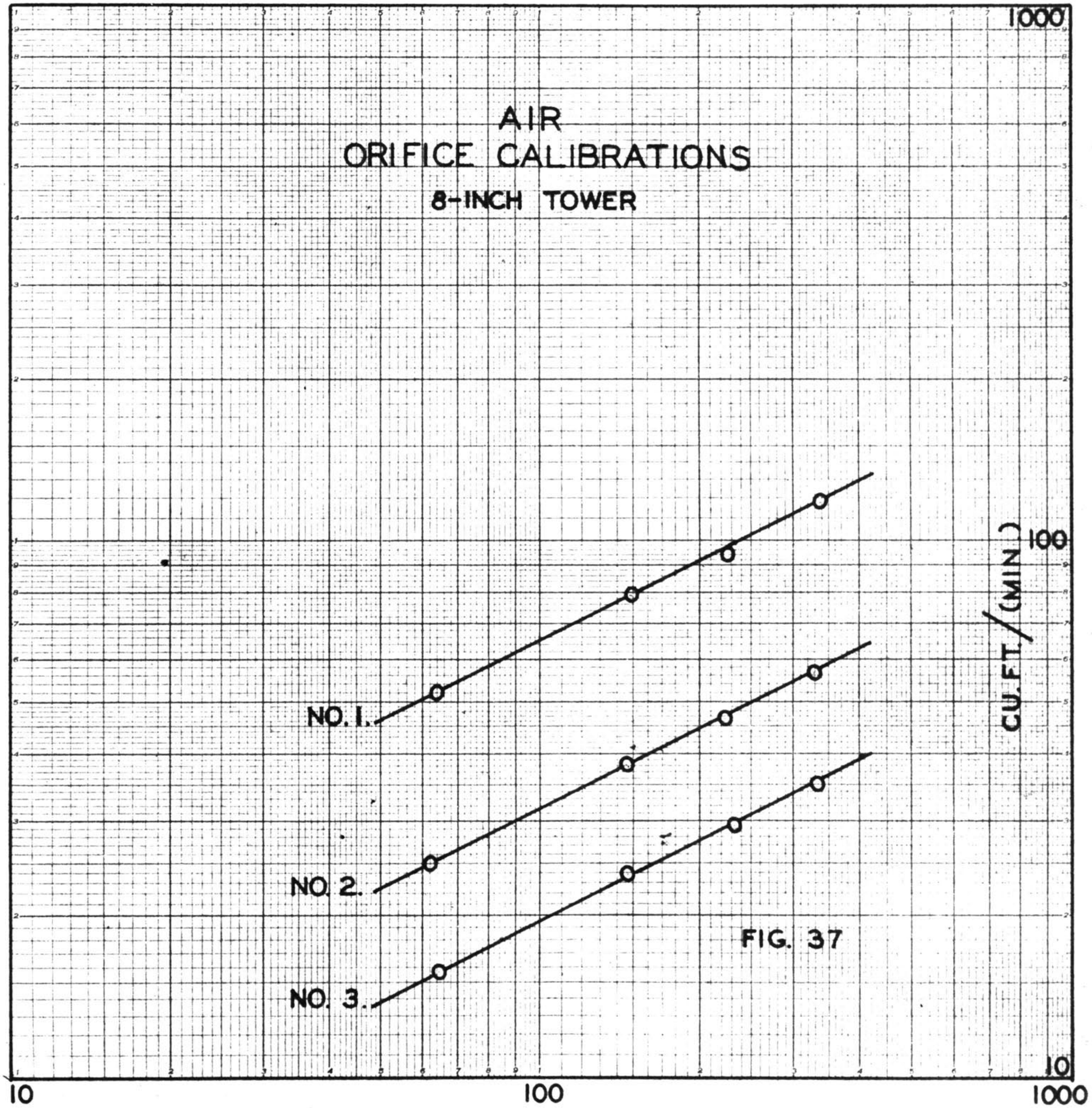

AIR
ORIFICE CALIBRATIONS
8-INCH TOWER
NO. 1.
NO. 2.
NO. 3.
1000
100
10
CU. FT. / (MIN.)
10
100
1000
ΔH - FT. OF FLUID FLOWING

FIG. 37

OIL ORIFICE CALIBRATIONS

8-INCH TOWER

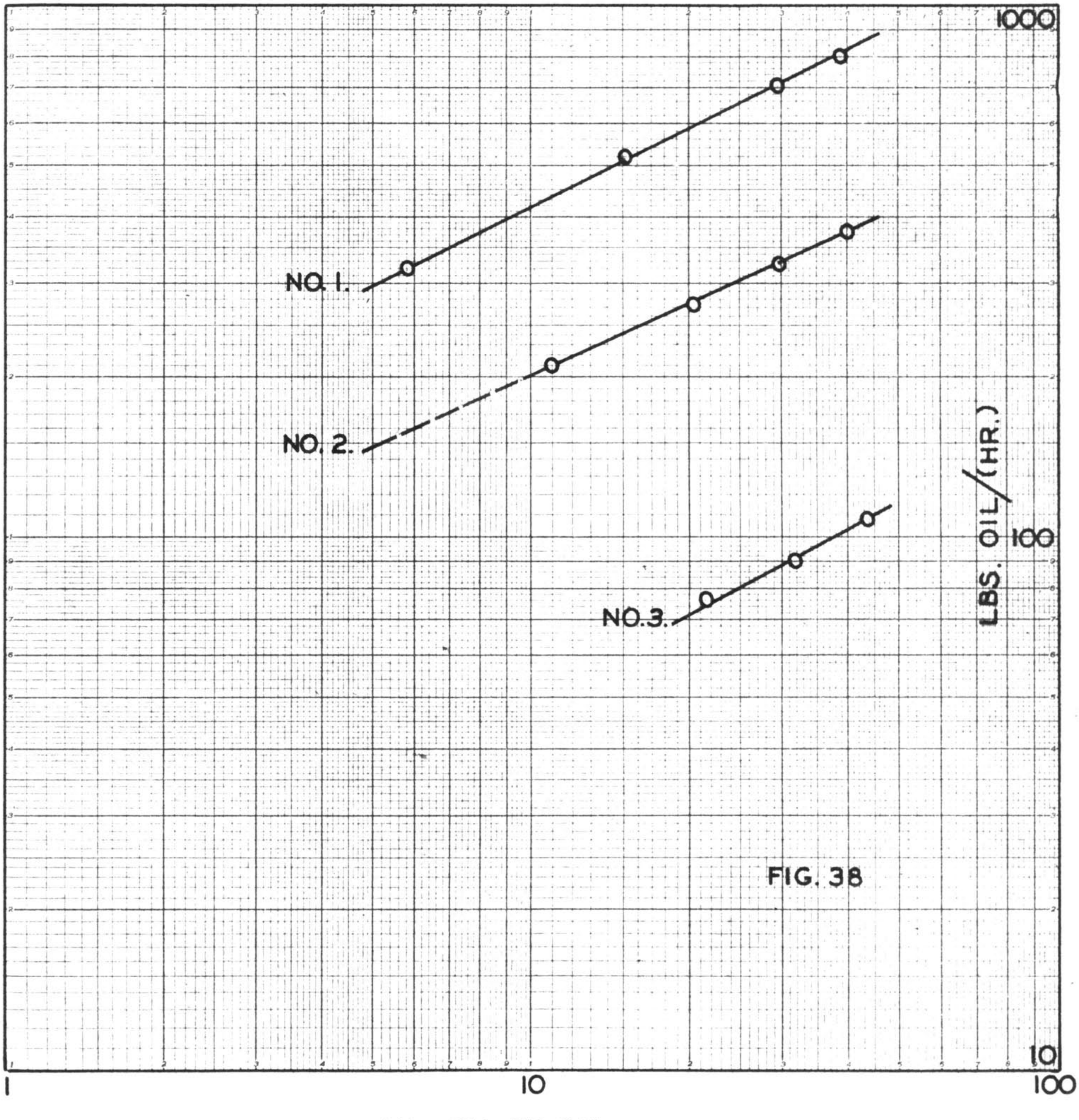


FIG. 38

VIII. APPENDIX

D. SUPPLEMENTARY DISCUSSION OF RESULTS

General Correlations for Oil-Heating and Oil-Cooling

Attempts were made to find a general correlation for the over-all coefficients of heat transfer obtained from both the oil-cooling and the oil-heating runs. It was felt that any such correlation would involve the viscosity of the oil since the oil viscosity (based on the average bulk temperature) was approximately fifteen fold greater for the oil-heating runs than for those in which the oil was cooled.

In the light of the fact that the individual coefficients of heat transfer, as obtained from adiabatic humidifying runs in the air-water tower, are of the same order of magnitude as the over-all coefficient of heat transfer from hot air to cold oil, it is concluded that the resistance to heat transfer offered by the oil film is small and that the gas film is controlling. Barnet and Kobe (3) heated oils with hot air in a wetted-wall tower and came to the same conclusion. Therefore, any temperature effect on the actual coefficient U must be restricted to the effect of temperature on the gas-film, and the results of the air-water study indicate this effect to be small. As a result, the a of Ua must be greatly influenced by oil temperature and oil

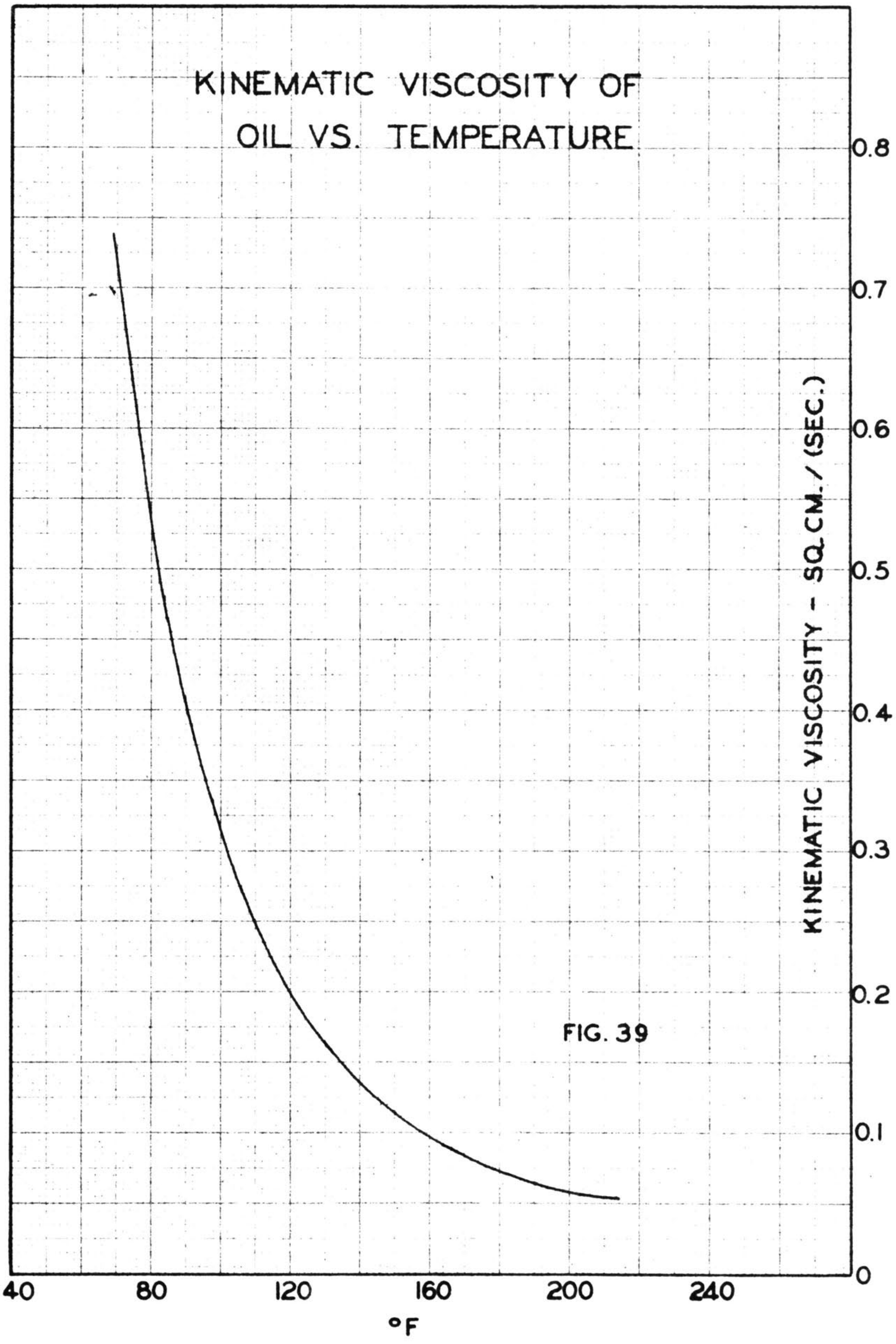

KINEMATIC VISCOSITY OF OIL VS. TEMPERATURE
KINEMATIC VISCOSITY - SQ.CM./(SEC.)
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
40
80
120
160
200
240
°F

FIG. 39

COEFFICIENT Ua VS. PRESSURE DROP

8 - INCH TOWER

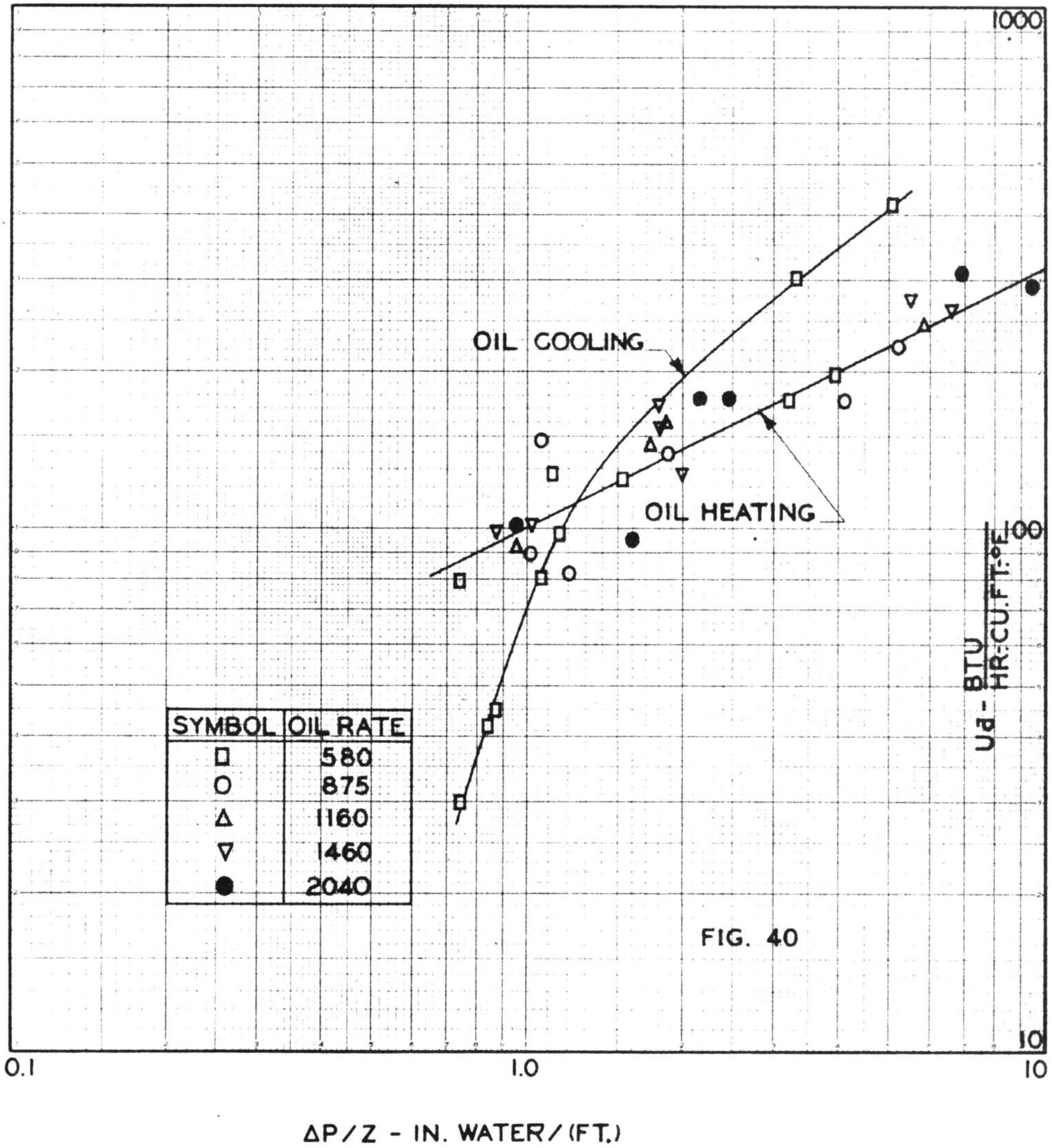
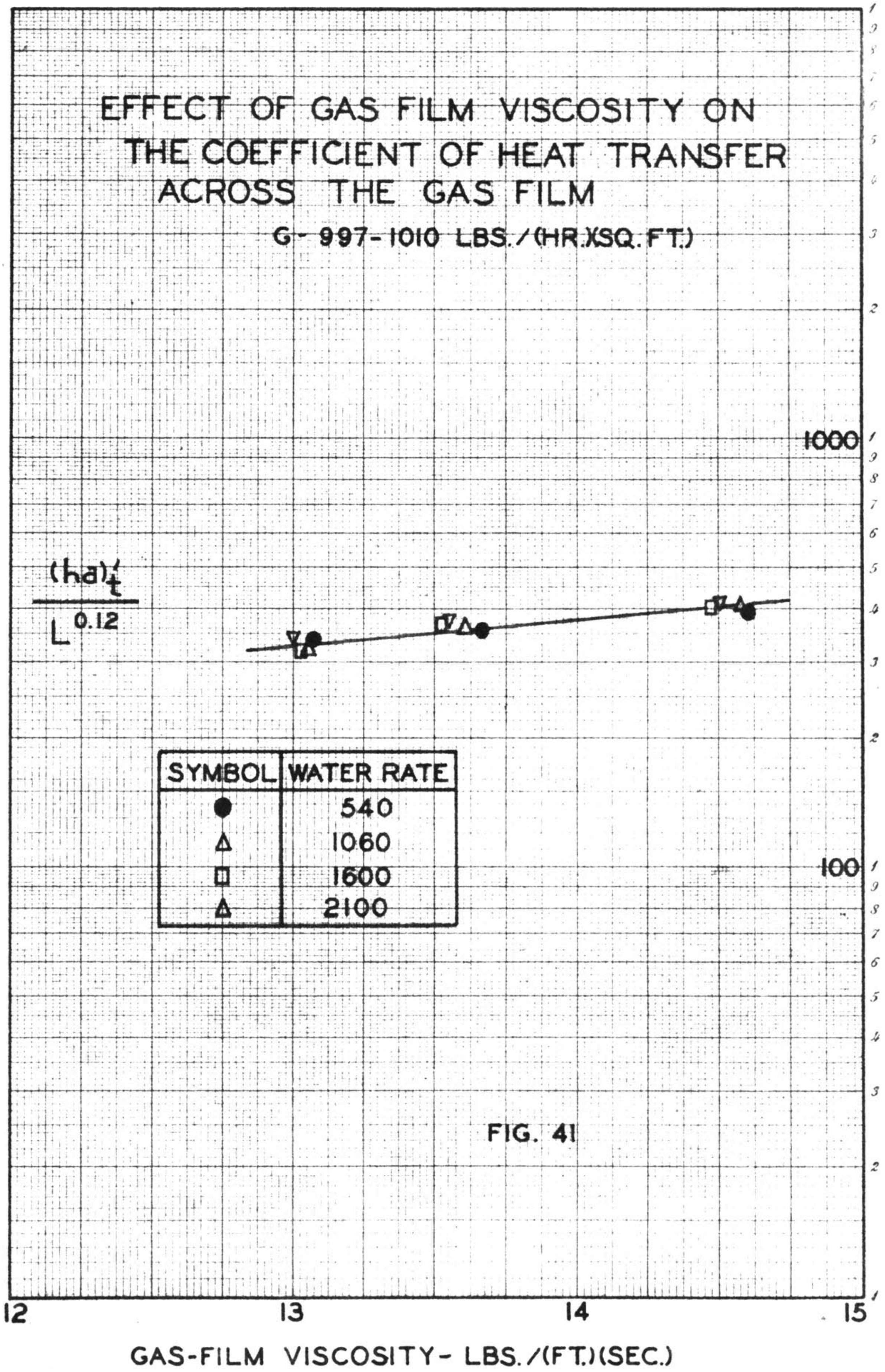


FIG. 40

viscosity as is evidenced by the large differences in coefficients obtained from the oil-heating and oil-cooling runs.

Special charts are available (1) which afford a relationship between oil viscosity and temperature providing the oil viscosity is known at two different temperatures. By means of these charts and experimental viscosity determinations at 100° F. and 210° F. it was possible to construct Figure 40 so that oil viscosities are available over the temperature range encountered in the experimental runs.

However, no satisfactory correlation was found by utilizing these oil viscosities. The distinct channeling which occurred in the hot-oil runs and which is the factor causing the large changes in the contact area per unit of packing volume seems to offer a more promising line of attack. Assuming that the pressure drop per unit of packed height is a measure of channeling and therefore of a, the coefficients for both oil-heating and oil-cooling runs are plotted in Figure 40 on log-log paper versus the pressure drop at various oil rates. Although the oil-heating data fall quite closely around a straight line 0.5 slope, the coefficients from the oil-cooling runs (at an oil rate of 580 lbs./(hr.)(sq.ft.) continue to show an unusual behavior.



EFFECT OF GAS FILM VISCOSITY ON THE COEFFICIENT OF HEAT TRANSFER ACROSS THE GAS FILM

G - 997-1010 LBS./(HR.XSQ.FT.)

FIG. 41

Effect of Temperature on ha.

The effect of temperature on the coefficient of heat transfer ha was presented as

$$ha \propto e^{0.0023t_f}$$

where t_f is the average air film temperature. The use of this form is purely conventional, and it was stated in the DISCUSSION OF RESULTS that the effect of temperature could be presented in other forms. From an examination of Figure 6, it is seen that the abscissa can also be considered as an expanded logarithmic scale of the absolute temperature. Consequently it is possible to express the effect of temperature as

$$ha \propto T^{m}$$

where T is the absolute temperature in degrees Rankine and m is equal to one third.

Perhaps a more fundamental approach is to present the temperature effect in terms of physical properties of the gas film. In Figure 41 the viscosity of the air film evaluated at the average film temperature is used as the abscissa on a semi-log plot and it is seen that the data points are well correlated by

$$ha = c' e^{0.15\mu_{t_f}}$$

E. SAMPLE CALCULATIONS

Calculation of ha and k'a from Air-Water Adiabatic Humidifying Runs

The following sample calculation is based on adiabatic humidifying run number 82. The recorded data with one foot of one-inch carbon Raschig rings as the packing were:

Manometer reading for large air orifice	5.34 cm. water
Manometer reading for air orifice static pressure	24.95 cm. water
Air temperature at air orifice	94°F
Manometer reading for orifice in cyclone separator	5.66 cm. water
Manometer reading for cyclone orifice static pressure (upstream tap)	15.25 cm. water
Air temperature of cyclone orifice (dew-point of exit air)	109.6°F
Pressure at top of the tower	19.05 cm. water
Pressure at bottom of the tower	22.70 cm. water
Pressure drop through tower	3.65 cm. water
Manometer reading for large water orifice	27.00 cm. water
Inlet humidity	0.0114 lbs. water per lb. of dry air
Inlet air temperature	338°F
Air temperature leaving packed section	119.5°F
Inlet and outlet water temperature	113.9°F
Barometer reading (corrected)	764.3 mm. Hg.

Pressure in cyclone separator	16.9 cm. water
Condensation rate from cyclone separator	1.27 cc per minute

(a) <u>Dry air rate through cyclone separator</u>.

Average orifice pressure,

$$764.3 + \left(\frac{15.25 - (0.5)(5.66)}{13.59}\right) 10, \text{ or } 773.5 \text{ mm. Hg.}$$

Molecular weight of moist air at orifice conditions is 28.0

Density of air at orifice conditions,

$$\frac{(28)\ (773.5)\ (492)}{(359)\ (760)\ (569.6)}, \text{ or } 0.0686 \text{ lbs./(cu.ft.)}$$

Head in feet of fluid flowing,

$$\frac{(5.66)\ (62.3)}{(12)(2.54)(0.0686)}, \text{ or } 169 \text{ ft.}$$

From the calibration plot at this head, the moist air rate is 6.0 cu.ft./(min.)

The flow rate of bone-dry air

$$(6.0)(60)(0.0686)\ \left(\frac{773.5-65.15}{773.5}\right)\ \left(\frac{29}{28}\right), \text{ or } 23.4 \text{ lbs./(hr.)}$$

where the vapor pressure of water at 109.6°F is 65.15 mm. Hg.

(b) <u>Humidity of air leaving packing</u>.

The humidity of the saturated air in the cyclone,

$$\frac{(65.15)(18)}{(776.8 - 65.15)(29)}, \text{ or } 0.0568 \text{ lbs.water/(lb. dry air)}$$

where 776.8 is the total pressure in the cyclone in mm. Hg.

The additional water condensed from the air and collected in the cyclone,

$$\frac{(1.27)(60)}{(454)(23.4)}\text{, or 0.0072 lbs. water/(lb. dry air)}$$

The exit tower humidity,

$$0.0568 + 0.0072 = 0.0640 \text{ lbs. water/(lb. dry air)}$$

(c) Air and water rates.

The calculation for the air rate is similar to that illustrated for the cyclone orifice.

The water rate is read directly from the water orifice calibration plot.

$G = 997$ lbs. dry air/(hr.)(sq.ft.)

$L = 2100$ lbs. water/(hr.)(sq.ft.)

(d) Heat balance.

A heat balance based on one pound of dry air showed that the run was nearly adiabatic. Enthalpy data are taken from tables compiled by Zimmerman and Lavine (53).

Base is 32°F and liquid water.

Enthalpy in (air at 338°F and water at 113.9°F),

$$73.89+(0.0114)(1210.8)+(0.064-0.0114)(114-32)= 92.0 \text{ Btu/(lb.}$$

Enthalpy out (air at 119.5°F)

$$21.03 + (0.0640)(1113) = 92.2 \text{ Btu/(lb.)}$$

Total heat transferred,

$$(0.24)(338 - 119.5), \text{ or } 52.4 \text{ Btu/(lb. dry air)}$$

Discrepancy in heat balance,

$$\frac{(92.2 - 92.0)\ 100}{52.4}, \text{ or } 0.4\% \text{ gain.}$$

(e) Transfer coefficients - gas film.

Since the heat balance closed within one per cent of the heat transferred across the gas film, the data were considered adequate for the calculation of the coefficients of heat and mass transfer across the gas film. From Eq. (14)

$$z = \frac{Gs}{ha_t'} \int_2^1 \frac{dt}{t - t_i} = H_t \ln \frac{(t - t_i)_1}{(t - t_i)_2}$$

$$H_t = \frac{1}{\ln \frac{(338 - 113.9)}{(119.5 - 113.9)}} = 0.271$$

$$(ha)_t' = \frac{Gs}{0.271} = \frac{(997)(0.257)}{(0.271)} = 945 \text{ Btu/(hr.)(cu.ft.)}(^\circ\text{F})$$

Similarly for mass transfer

$$z = \frac{G}{(k'a)_t'} \int_2^1 \frac{dH}{H - H_i} = H_t \ln \frac{(H - H_i)_1}{(H - H_i)_2}$$

The interfacial humidity is

$$H_i = \frac{19.15}{(778.3 - 19.15)} \left(\frac{18}{29}\right) = 0.0653 \text{ at gas outlet}$$

$$H_i = \frac{19.15}{(781.0 - 19.15)} \left(\frac{18}{29}\right) = 0.0651 \text{ at gas inlet}$$

$$H_t = \frac{1}{\ln \frac{(0.0651 - 0.0114)}{(0.0653 - 0.0640)}} = 0.269$$

$$(k'a)_t' = \frac{G}{0.269} = \frac{997}{0.269} = 3710 \text{ lbs./(hr.)(cu.ft.)(unit H)}$$

$$\left(\frac{ha}{k'a}\right)_t' = \frac{945}{3710} = 0.255 \text{ vs. } s = 0.257$$

Calculation of h_La from Water-Cooling Runs

The following illustrates the calculation of the liquid-film coefficient based on the data taken for run number 98.

Packed height corrected for end-effects (7.2 inches)		1.6 ft.
Gas rate		702 lbs./(hr.)(sq.ft.)
Water rate		2080 lbs./(hr.)(sq.ft.)
Air humidities	In	0.0070 lbs. water per lb. dry air
	Out	0.0789 lbs. water per lb. dry air
Humid heat		0.259 Btu/(lb. wet air)
Temperatures		
Inlet air		90°F
Exit air		121.5°F
Inlet water		136.7°F
Exit water		110°F
Interfacial (average) (See Fig. 2)		110°F
Film temperature (average)		108°F
Average pressure in the packing		775 mm. Hg.

Total enthalpy in	331.6 Btu/(lb. dry air)
Total enthalpy out	331.1 Btu/(lb. dry air)
Enthalpy of entering air	21.6 Btu/(lb. dry air)
Enthalpy of exit air	109.6 Btu/(lb. dry air)

(a) Determination of $k'a_{108^oF}$.

From adiabatic humidifying runs at L of 2080, G of 700, and $t_f = 193.8$, ha = 459 (corrected for end-effects). At a gas film temperature of 108°F,

$$ha_{108} = 459 \frac{e^{0.0023(108)}}{e^{0.0023(194)}} = 369 \text{ Btu/(hr.)(cu.ft.)(}^oF)$$

Assuming $\frac{ha}{k'a} = s$,

$$(k'a)_{108} = \frac{369}{0.259}, \text{ or } 1420 \text{ lbs./(hr.)(cu.ft.)(}^oF)$$

From Eq. (18b),

$$z = \frac{G}{k'a} \int_1^2 \frac{di_G}{i_i - i_G} = 1.6$$

$$N_t = \int_1^2 \frac{di_G}{i_i - i_G} = \frac{(1.6)(1420)}{(702)} = 3.24$$

The value of the integral must equal 3.24 and is evaluated graphically by plotting $1/i_i - i_G$ versus i_G and measuring the area under the curve (Figure 2). The values of $i_i - i_G$ are found by trial and error adjustment of the tie-line slope ($-h_La/k'a$) until the value of the integral is equal to 3.24. For this run, the tie-line slope is 2.7 and $h_La = (2.7)(1420) = 3840$ Btu/(hr.)(cu.ft.)(°F).

Calculation of K_La from Oxygen Desorption Runs

a. Equilibrium data

Equilibrium data are given by Winkler (55) in terms of Henry's Law constant H' and temperature T.

$$c_e = H'p$$

where c_e is the equilibrium concentration of oxygen in water and p is the corresponding partial pressure of oxygen in the gas phase. Preliminary tests with tap water in which air was bubbled through a sample of the water until equilibrium was reached checked these data.

Water temperature = 24.0°C.

Concentration of oxygen in the sample by analysis, 1.61×10^{-5} moles/(cu.ft.)

Calculating concentration from equilibrium data,

$$p = \frac{0.21\ (\text{Bar. Pressure - Vapor Pressure})}{\text{Barometric Pressure}}$$

$$= \frac{(0.21)(759.7 - 22.2)}{759.7}, \text{ or } 0.204 \text{ atmos.}$$

$$c_e = (8.00)(10^{-5})(0.204) = 1.63 \times 10^{-5} \text{ moles/(cu.ft.)}$$

Due to the good agreement with Winkler's data, these were used in the calculations.

b. Determination of the coefficient

The data from run number 144 will be used to illustrate the calculation

G = 120 lbs./(hr.)(sq.ft.)

L = 1500 lbs./(hr.)(sq.ft.)

z = 2.5 ft. of one-inch carbon Raschig rings.

Water temperature = 12.8°C.

For the oxygen analysis at the top and at the bottom of the packing, a variable amount of sample was titrated into 4.05 ml. of sodium thiosulphate. The normality of the thiosulfate was 0.0100.

Top

$$\frac{(4.05)(0.0100)(62.3)}{(4)(12.56)(1000)} = 5.04 \times 10^{-5} \text{ moles/(cu.ft.)}$$

Bottom

$$\frac{(4.05)(0.0100)(62.3)}{(4)(26.16)(1000)} = 2.41 \times 10^{-5} \text{ moles/(cu.ft.)}$$

Partial pressure of oxygen = 0.211 atmos.

$$c_e = (9.98 \times 10^{-5})(0.211) = 2.11 \times 10^{-5} \text{ moles/(cu.ft.)}$$

$$K_La = \frac{L}{z\ d} \ln \frac{(c_t - c_e)}{(c_b - c_e)}$$

$$= \frac{(1500)}{(2.5)(62.3)} \ln \frac{(5.04 - 2.11)}{(2.41 - 2.11)}$$

$$= 22.0 \text{ hr.}^{-1}$$

Using Holloway's (23) temperature correction

$$(K_La)_{25} = (K_La)_{12.8} \frac{e^{0.023(25)}}{e^{0.023(12.8)}}$$

$$= (22.0)(1.32) = 29.1 \text{ hr.}^{-1}$$

Calculation of Ua from Air-Oil Runs

The following calculation is based on run number 86 wherein heat was transferred from hot air to cold oil. The recorded data with 14 inches of one-inch ceramic Raschig rings were:

Manometer reading for air orifice No. 1	3.52 cm. water
Temperature of air in air orifice	200.0°F
Pressure in air orifice	790.1 mm. Hg.
Wet-bulb temperature of inlet air	50°F
Dry-bulb temperature of inlet air	68°F
Manometer reading for oil orifice No. 2	24.4 cm. oil
Temperatures:	
Exit oil	121.7°F
Inlet oil	68.0°F
Exit air	84.3°F
Inlet air	200.0°F

a. Flow Rates

The oil flow rate can be read directly from the calibration plot,

$$L = \frac{304}{S} = \frac{304}{0.347} = 875 \text{ lbs./(hr.)(sq.ft.)}$$

The air flow rate is calculated in the same manner as for air-water runs which is illustrated above,

$$G' = 765 \text{ lbs. wet air/(hr.)(sq.ft.)}$$

b. Heat Balance

Using the experimentally determined value of 0.44 Btu/(lb.) as the heat capacity of the oil, the heat lost by the air stream,

$$(0.347)(765)(0.44)(0.242)(200.0 - 84.3) = 7390 \text{ Btu/(hr.)}$$

The heat gained by the oil,

$$(0.347)(875)(0.44)(121.7 - 68.0) = 7190 \text{ Btu/(hr.)}$$

The heat loss,

$$\frac{(7390 - 7190)100}{7390} = 2.7\%$$

c. <u>Over-all coefficient Ua</u>

From Eq. (7)

$$Ua' = \frac{Lc_L\,(T_1 - T_2) + G'c_G\,(t_1 - t_2)}{(2)(z)(\ t_m)}$$

$$= \frac{(7290)}{(0.347)\left(\frac{14}{12}\right)\left(\frac{78.3 - 16.3}{\ln \frac{78.3}{16.3}}\right)} = 456 \text{ Btu/(hr.)(cu.ft.)}(^{\circ}\text{F})$$

where U'a is not corrected for end-effects.

Since the end-effect for the hot-air runs was found to be a constant (14.3 inches) independent of L and G', the corrected coefficient is obtained by multiplying Ua' by the ratio (14/28.3).

VIII. APPENDIX

F. TABLE OF NOMENCLATURE

a = wetted transfer surface per unit volume of tower packing, sq. ft./(cu.ft.)

c = specific heat, Btu/(lb.)(°F)

d = density, lbs./(cu.ft.)

G = superficial mass velocity of dry air, (lbs. bone-dry air)/(hr.)(sq.ft.)

G' = superficial mass velocity of non-dry air, (lbs. air)/(hr.)(sq.ft.)

H = absolute humidity of air (lbs. water vapor)/(lb. of bone-dry air)

H' = Henry's Law constant moles/(cu.ft.)(atmos.)

ha = individual coefficient of heat transfer independent of end effects, Btu/(hr.)(cu.ft.)(°F)

(ha)' = individual coefficient of heat transfer not corrected for end effects, Btu/(hr.)(cu.ft.)(°F)

H_t = height of a transfer unit, ft.

i = enthalpy, Btu/(lb.)

Ka = over-all coefficient of mass transfer, lbs./(hr.)(cu.ft.) (unit over-all driving potential.)

K_La = over-all coefficient of oxygen transfer, (lb. moles)/ (hr.)(cu.ft.)(lb. moles per cu.ft.)

ka = individual mass transfer coefficient, lb. moles/ (hr.)(cu.ft.)(atmos.). $k'a = ka\ p_{BM}\ M_B$.

k'a = individual coefficient of mass transfer across the gas film corrected for end-effects, lbs./(hr.)(cu.ft.)(unit humidity driving force).

$(k'a)'$ = individual coefficient of mass transfer across the gas film not corrected for end-effects, lbs./(hr.)(cu.ft.)(unit humidity driving force).

$(k'a)''$ = individual coefficient of mass transfer across the gas film, corrected for end-effects and based on temperature of adiabatic saturation, lbs./(hr.)(cu.ft.)(unit humidity driving force).

L = superficial mass velocity of the liquid, lbs./(hr.)(sq.ft.)

N_t = number of transfer units

p = partial pressure

q = quantity of heat, Btu.

R = resistance of phase to heat and/or mass transfer; reciprocal of the coefficient.

r = heat of vaporization, Btu/(lb.)

S = gross cross-sectional area, sq.ft.

s = humid heat, Btu/(lb.)

T = temperature of the liquid

t = temperature of the gas

Ua = over-all coefficient of heat transfer, corrected for end effects, Btu/(hr.)(cu.ft.)(°F)

$(Ua)'$ = over-all coefficient of heat transfer not corrected for end-effects, Btu/(hr.)(cu.ft.)(°F)

x = net end-effect equivalent to height, inches.

z = height of packed section, ft. or inches

Subscripts

B - refers to air

BM - refers to an average value of the inert gas

G - refers to the gas phase or gas film

H - refers to heat transfer

i - refers to conditions at the interface

L - refers to the liquid phase or liquid film

M - refers to mass transfer

o - refers to some base or standard

s - refers to saturation conditions

$t = t_f$ - refers to the temperature of the gas film

V - refers to vapor

wv - refers to water vapor

w - refers to wet bulb conditions

2 - refers to conditions at the top of the tower

1 - refers to conditions at the bottom of the tower.

VIII. APPENDIX

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