THE COMPONENT HEAT EXCHANGES

IN THE

EVAPORATIVE CONDENSER.

BY

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INTRODUCTORY.

An outstanding feature of steam power engineering at the present day is the high order of performance obtainable from the modern surface condenser. In this unit are embodied the fruits of years of intense research and investigation, and, to such a pitch has the technique of design been brought, that further improvement seems improbable. In striking contrast to this state of affairs, is the position with regard to the evaporative condenser, which may be said to be still in the most primitive stage of development, technically and practically. This latter aspect is seen in the average colliery evaporative condenser, which both in bulk and in area covered, is out of all proportion to its usefulness, while the shortcomings on the technical side may be gauged from the fact that design is still largely specula: :tive. It is difficult to understand why such an unprogressive policy should have been pursued, and the weakness of the system has been fully exposed in some recent attempts to design portable evaporative condensers for railway work, which have resulted only in costly failures.

That this type of condenser is worthy of more consideration may be appreciated from a comparison with the standard surface condenser, cooling tower system. The latter scheme comprises two separate and costly units, and the cooling tower must of necessity, be capable of dealing with the same amount of heat as the condenser. The water evaporated in the process may equal 85 per cent of the steam condensed. On the other hand, the evapora: :tive condenser combines the condensing and water cooling processes in one plant, the loss of water by evaporation is no greater, and a much smaller rate of water circulation is required.

Such advantages as these cannot be overlooked, particularly for portable units, but they can only be turned to practical use when the principles of the heat transmission processes involved are fully understood, and a rational basis of design established.

When the question of the application of condensers to locomotives was first raised, research on the evaporative condenser was instituted at the Royal Technical College, and the results of the earlier work have already been embodied in an important paper by D.S. Anderson, Ph.D. Much information of a very valuable nature was obtained from these initial investi: :gations, and the complex character of the problem was fully demonstrated. Several of the issues raised, however, were of a distinctly controversial nature, but the limited scope of the original apparatus, which was designed to deal only with atmos: :pheric steam, prohibited further examination of these points.

In order to carry development a stage further, therefore, the investigation described in this paper was undertaken, and a practical type of condenser was evolved, incorporating most of the features likely to be met with under actual operating condi: :tions. The steam side was arranged for working under varying vacuum, the cooling water was circulated continuously, and the evaporation found directly from the decrease in weight of the water in the system. It has thus been possible to carry out tests under widely varying conditions on both steam and water sides, and, in consequence, the general design basis developed may be confidently applied to practical cases.

A new line of attack on the problem has been developed. The component heat transfer processes have been separately analysed, and it is clearly demonstrated that the generally accepted laws of heat transmission and evaporation, are still applicable to the evaporative condenser.

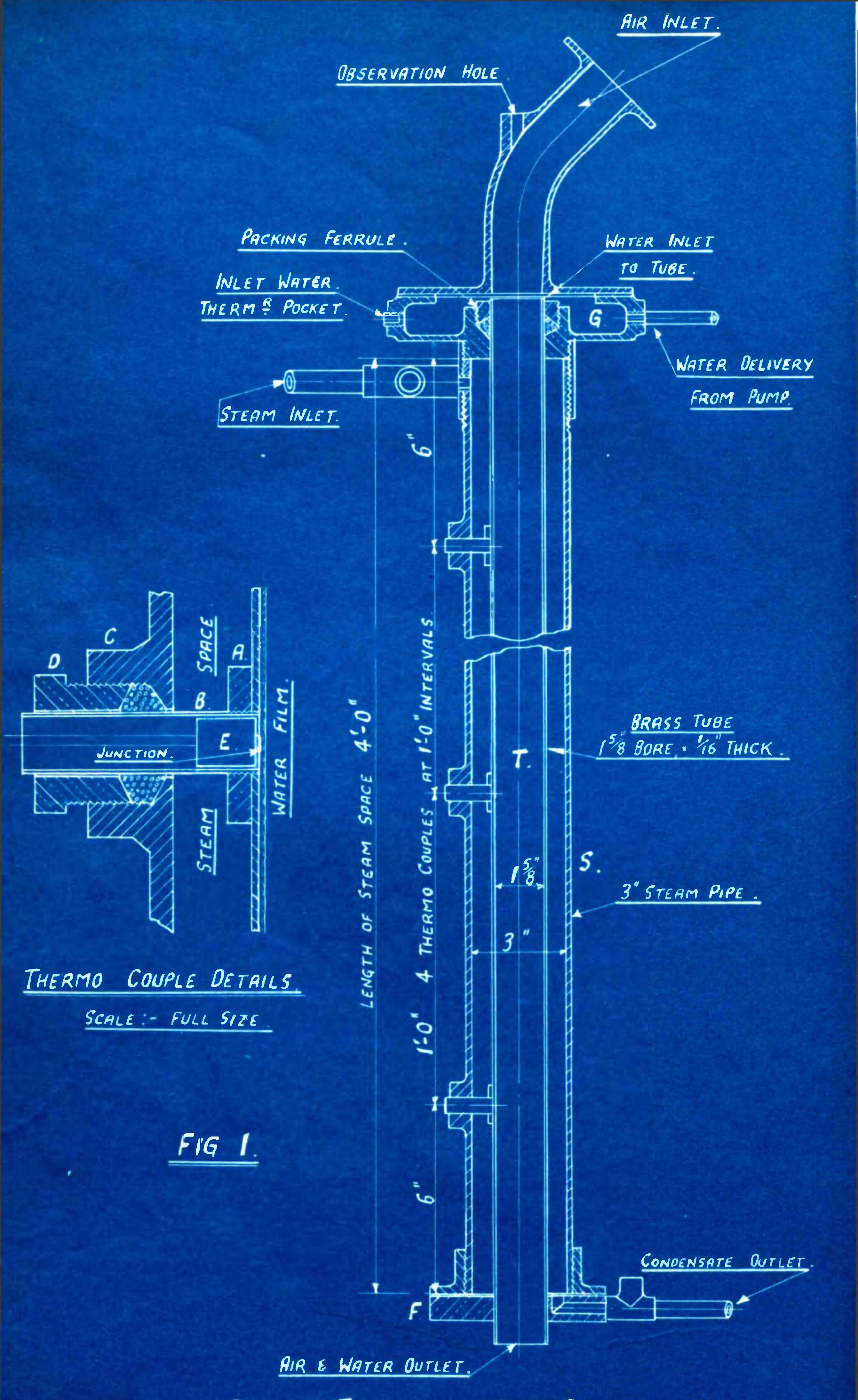


FIG 1.

SINGLE TUBE EVAPORATIVE CONDENSER.

SCALE 3" = 1 FOOT.

MEAN COOLING SURFACE, (STEAM - WATER) - 1·77 SQ. FT.

DESCRIPTION of PLANT.

The principal features of the apparatus are shown in Figs. 1 - 3. Fig. 1 gives the condenser unit in detail, while the general arrangement of the complete plant is illustrated in the other figures. As will be seen from Fig. 1, the condenser comprises a single brass tube T_T $1\frac{5}{8}$" bore, encased in a length of steam piping S, 3" internal diam. x 4 feet long. The cooling water flows down over the inside face of the brass tube, and is swept by the air stream, which also flows downward through the tube T. In the annular space between S and T the steam is condensed.

Referring in detail to the figure, at the bottom end of the condenser, the tube T is solidly brazed to the heavy flange F, which in turn is bolted to the bottom flange of the outer pipe S. At the top end, pipe S is closed by a brass casting G, through which tube T passes, being ferrule packed as in standard surface condenser practice. This casting G carries a water thermometer pocket and the water inlet pipe, and also acts as a header for admitting the water to tube T. The inlet air branch pipe is also bolted to G, in such a manner that a space of $\frac{1}{16}$ inch is left clear above the top edge of tube T. This slit serves to spread the water in an even film over the tube surface. The inlet air branch carries a $\frac{5}{8}$" dia. hole which serves the dual purpose of a thermometer pocket, and window for inspecting the water film. The steam enters at the top end of the condenser as indicated, and the condensate is pumped off through a passage in the flange F.

The water temperature at four points along the tube is recorded by thermo-couples, and the details of the necessary fixing are also given in Fig. 1. A metal pad A, sweated to the brass tube T, carries the small brass tube B, $\frac{1}{2}$" external diameter.

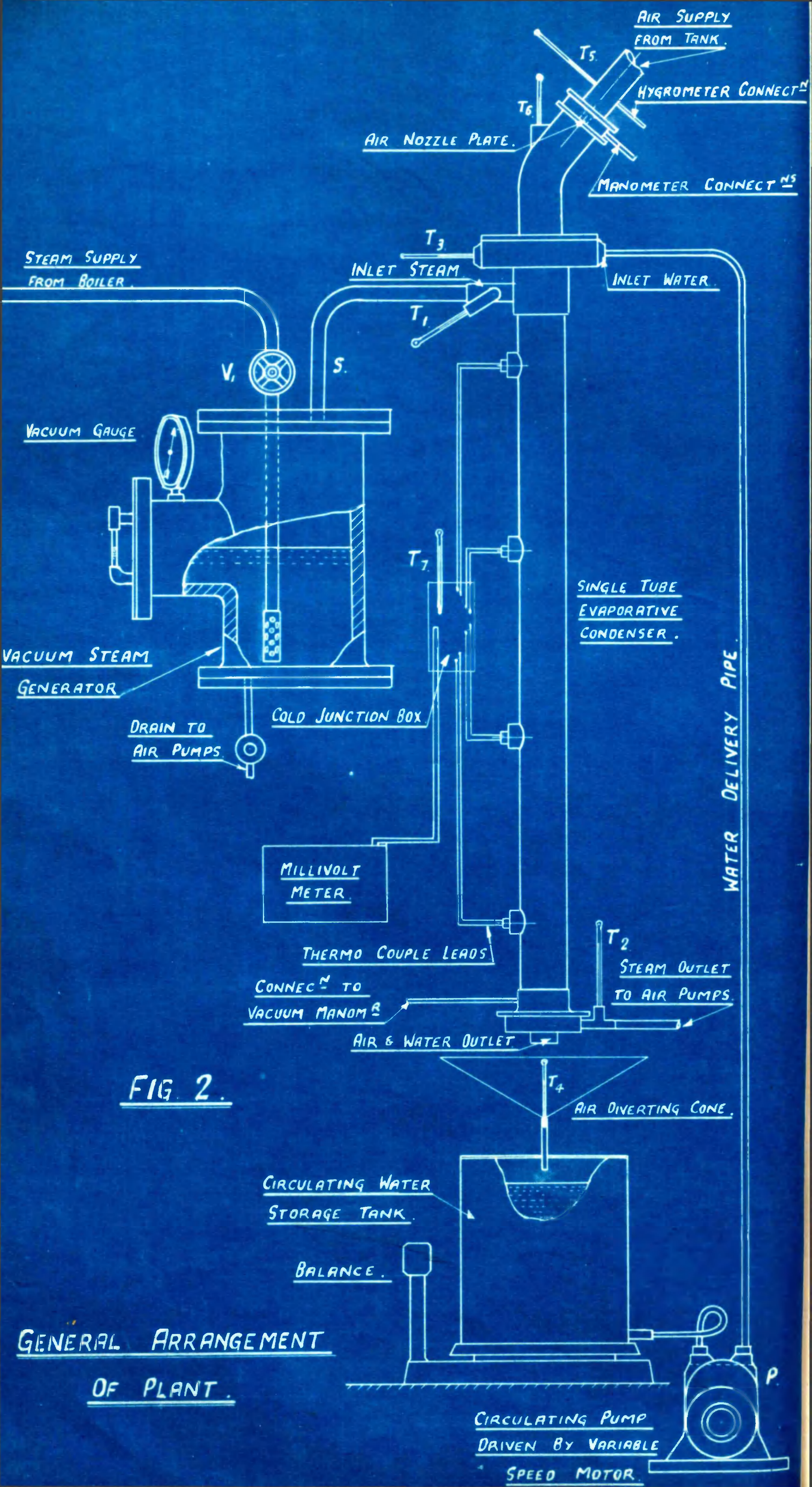


FIG. 2.

GENERAL ARRANGEMENT OF PLANT.

A steel boss, C, is also welded to the external steam tube, and in this box is incorporated a stuffing box. Through this passes B, and leakage of either air or steam is prevented by means of asbestos packing and ferrule D. A cylinder of cork, E, insulates the thermo-couple wires from the metal, and is adjusted until the actual thermo-junction is just protruding into the water film, as indicated. The couples used were of copper-constantan.

The general scheme of testing is best described by considering separately the circuit of each fluid involved.

Steam Circuit: Steam throttled directly from a boiler into a vacuum of say 24 inches, builds up an extra-ordinary superheat, and is therefore useless for vacuum tests purporting to represent practical condenser conditions. This difficulty is overcome in the present plant by passing the boiler steam into a "vacuum steam generator", Fig. 2. This vessel is maintained about half full of water, which is heated by the condensation of the steam led into it. From the space above the water a pipe S leads directly into the condenser, hence the water in the generator boils at a temperature slightly above the saturation temperature at the condenser vacuum. Thus, by regulating the flow of the inlet boiler steam, a steady supply of low temperature steam is fed to the steam space of the condenser. The condensate outlet from the condenser is connected to the Edwards air pumps of a surface condenser installed in the laboratory.

Control of the steam supply is effected by valve V_1. During a given test, the condenser vacuum, recorded on a mercury manometer, was kept constant, and V_1 regulated so that the outlet steam thermometer T_2, recorded the corresponding saturation temperature, or as near to this as the air content of the steam permitted. Great care was taken to prevent air leakage into the steam system, but the supply steam itself contained a fair amount of air, as the boiler feed circuit is open. As a result there was a steam temperature gradient along the tube in the high vacuum tests, and

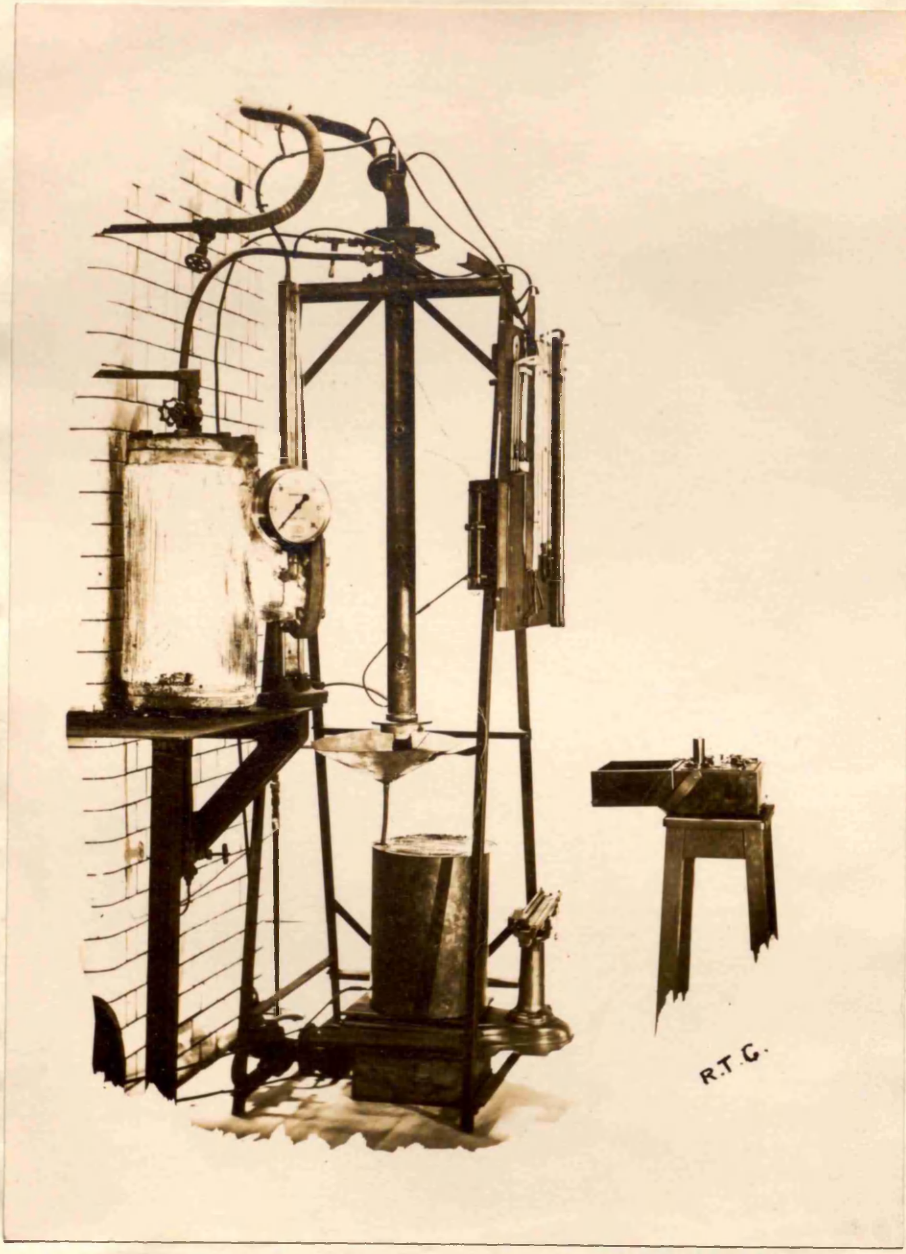


FIG. 3

GENERAL VIEW OF PLANT.

in such cases the mean steam temperature has been taken as the average of the saturation and outlet temperatures.

Air Circuit: As a rule fans supply sufficient air for evapora:tive condensers, but in the present case it was more convenient to take the air from a compressor storage tank. In the supply pipe a measuring nozzle plate is fitted as indicated in Fig. 2. This plate is of the standard type for pipe flow work, the manometer connections being taken from either side of the nozzle out through the body of the plate. The manometers record the pressure drop across the nozzle in ins. water, and the air pressure on the bottom side of the plate. In order to minimise the disturbance set up in the air stream, a relatively large sized nozzle is used. The humidity of the supply air is found by tapping off a stream of air before the nozzle and passing it over a wet and dry bulb hygrometer. The temperature of the air at inlet to the condenser tube is taken at T_6. The air stream passes through the condenser, collecting the vapour evolved from the water, and at the outlet end is diverted through roughly 180° and thus prevented from carrying off solid water in suspension.

Water Circuit: The water circulation is maintained by a gear pump, P, Fig. 2, which is driven by a small shunt motor, fitted with a rheostat in the armature circuit. Alteration of the rate of circulation is effected by varying the motor speed. From the pump a long copper pipe, $\frac{3}{16}$" bore, leads to the header at the top of the condenser, whence the water passes to the inner tube, as previously described. Although little difficulty was experienced in maintaining a water film with the vacuum tests, in order to ensure regularity of results, the inner tube surface was faced with a fine silk widely meshed net. At the foot of the tube the unevaporated water is collected in the metal cone, where its temperature is taken at T_4, and is passed back to the storage tank as shown. In addition the temperature of the water leaving the tube was taken. In earlier tests a calibrated weir was fitted below the cone to measure the rate of water circulation, but it was found to be more accurate to measure the water flow directly, before and after each test. The storage tank is placed on a weighing machine, and a flexible tube leads from the foot of the tank to the suction of pump P. With a constant rate of water circulation, and no evaporation taking place, the balance reading remains constant. With evaporation, however, the air carries off a percentage of the water as vapour, and the loss due to this is recorded by a decreased balance reading.

The leads from the four water thermo-couples are taken to a common switch box, where the cold junction temperature is read on T_7. Leads are also taken from the box to a calibrated Cambridge millivolt-meter, which by means of a four way switch, may be put in circuit with each of the thermo-couples separately.

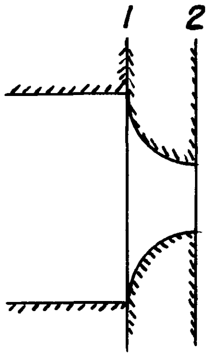
Before concluding this section, it may be stated that previous to starting the tests, the balance, gauges and thermome::ters were calibrated. No test was started until all conditions had been steady for at least 15 minutes. The actual length of a test depended on the rate of evaporation, the average time being about 25 minutes.

The principal observed readings and derived results are given in Tables 1-3, while the main features of the calculations are dealt with in the following section.

TEST MEASUREMENTS AND REDUCTION OF DATA.

(a) Calculation of Air Flow. A pure-convergent nozzle, $\frac{7}{16}$" diameter, was used, and the pressure drop across it was varied from 1 to 23 inches of water. The quantity

of air passing was calculated on the following basis:



Let w = specific weight of air, lb./ft^3, taken as constant over the given pressure range.

p_1, p_2 = air pressure lb/ft^2 at sections (1) and (2).

v_1, v_2 = air speed ft/sec. " " and (2).

a_1, a_2 = cross section ft^2 " " and (2)

Then, by Bernoulli's equation,

$$\therefore \quad v_2 = \sqrt{\frac{2g.(p_1 - p_2)}{w.\left\{1 - \left(\frac{a_2}{a_1}\right)^2\right\}}} \text{ -----------------(1)}$$

And lb. air passing per sec. = $Cd.\ a_2.\ w.\ v_2$--(2)

Where Cd, the coefficient of discharge was taken as 0.97.

Also, if H_1 = pressure at (1), inches Hg.

and T_1 = tempr. " " ^oF abs.

and h = pressr. drop across nozzle, ins. water

Then $w = \frac{p_1}{53.3\ T} = 1.33 \frac{H_1}{T}$

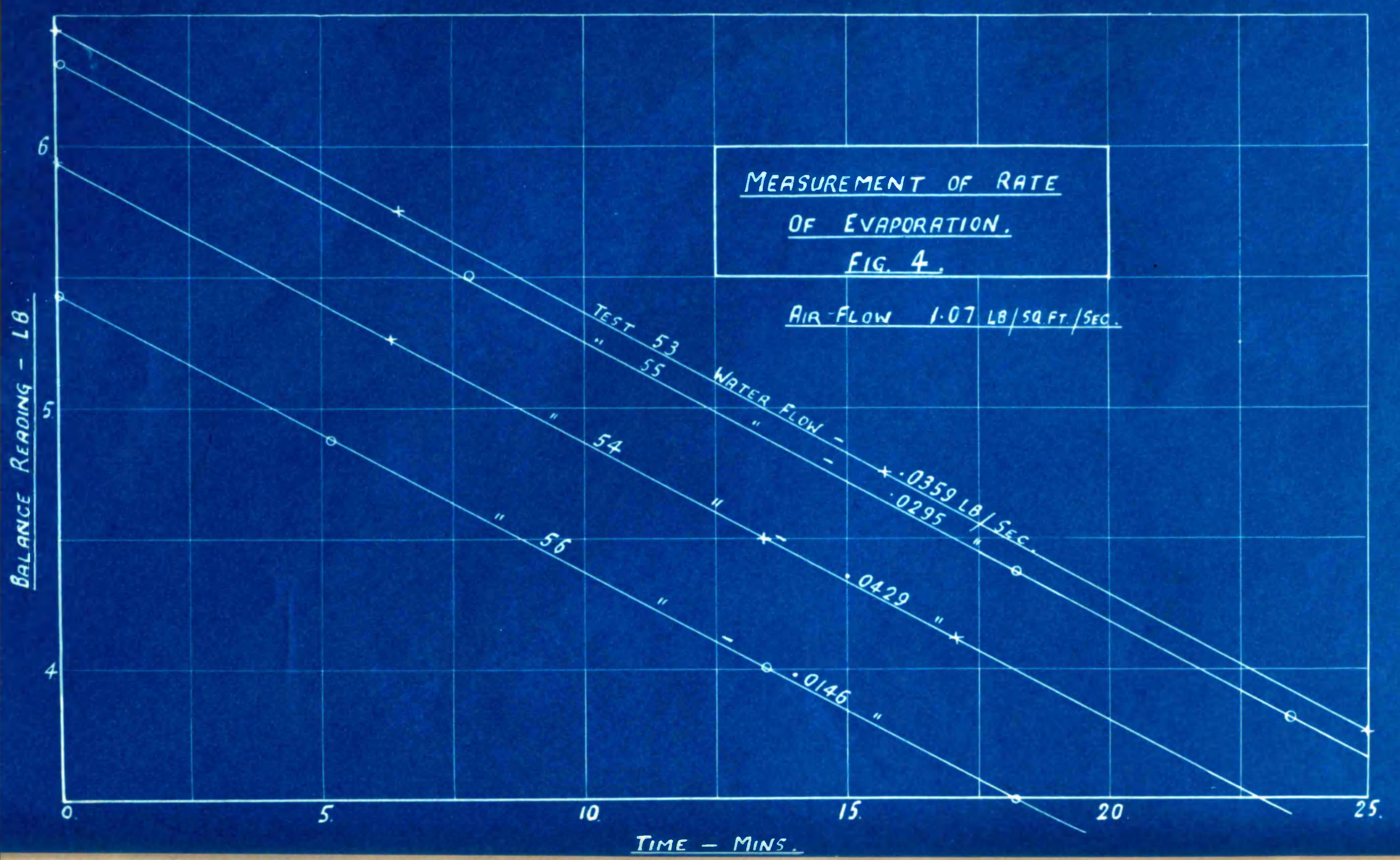
and $(p_1 - p_2) = 5.2\ h.$

Combining (1) and (2) and making the above substitutions gives finally for the air flow through a $\frac{7}{16}$" nozzle,

$$W \text{ lb/sec.} = .0216 \sqrt{\frac{H_1 h}{T_1}}$$

The smallest air speed used corresponded to a flow rate in the condenser tube of 0.36 lb/sec/sq. ft., while the critical velocity for air in the tube, with air at 180^{o}F, is in the region of 0.22 lb/sec./sq. ft.

(b) Measurement of Evaporation. This was estimated by means of a stop watch and an accurate balance. The sensitivity of the balance was increased by artificially raising the centre of gravity



MEASUREMENT OF RATE
OF EVAPORATION.
FIG. 4.
AIR-FLOW 1·07 LB/SQ.FT./SEC.
TEST 53 WATER FLOW – ·0359 LB/SEC.
" 55 " – ·0295 "
" 54 " – ·0429 "
" 56 " – ·0146 "
BALANCE READING – LB.
TIME – MINS.
0.
5.
10.
15.
20.
25.
4
5
6

of the beam, so that there was a definite movement of the beam between its stops while passing through the point of balance. At the beginning of a test, the rider on the beam was set at a noted weight, and when, due to the loss of water from evaporation, the beam swung through the balance point, the watch was started. This was repeated for suitable weight decrements, the time being noted in each intermediate case. The results were then plotted as indicated in Fig. 4, and the gradient of the straight line graph gave directly the rate of evaporation during the test. The results obtained in this way were remarkably consistent, and for any given test the plotted points always lay on or very close to the mean line. The outstanding advantage of the method, however, is that it acts as a check on the steadiness of conditions during a test, since varying conditions would be indicated by uneven plots. The only possible sources of error which might occur are leakage in the water system, and the carrying off by the air stream of solid unevaporated particles of water. Freedom from the former trouble was ensured by very careful packing of all the joints concerned; and lengthy tests with an air vacuum in the steam space, and no air flow on the water side, failed to reveal any measurable leakage even with the smallest amount of water circulating. Regarding the latter trouble mentioned above, it was found that there was a maximum air speed below which no visible water globules were carried off, and the highest air speed used, about 25 feet per sec, was well below this maximum.

A certain amount of water is evaporated between the condenser outlet and the entry to the storage tank, and it is necessary to correct the total evaporation for this, to find the actual evaporation in the tube. This correction is simple, as the heat lost by the circulating water between the above two points is known. This heat is a direct measure of the vaporisation which has taken place, and is converted to equivalent evaporation by dividing by the heat required per lb vapour formed. All evaporation rates are based on a water air surface of 1.90 sq. ft.

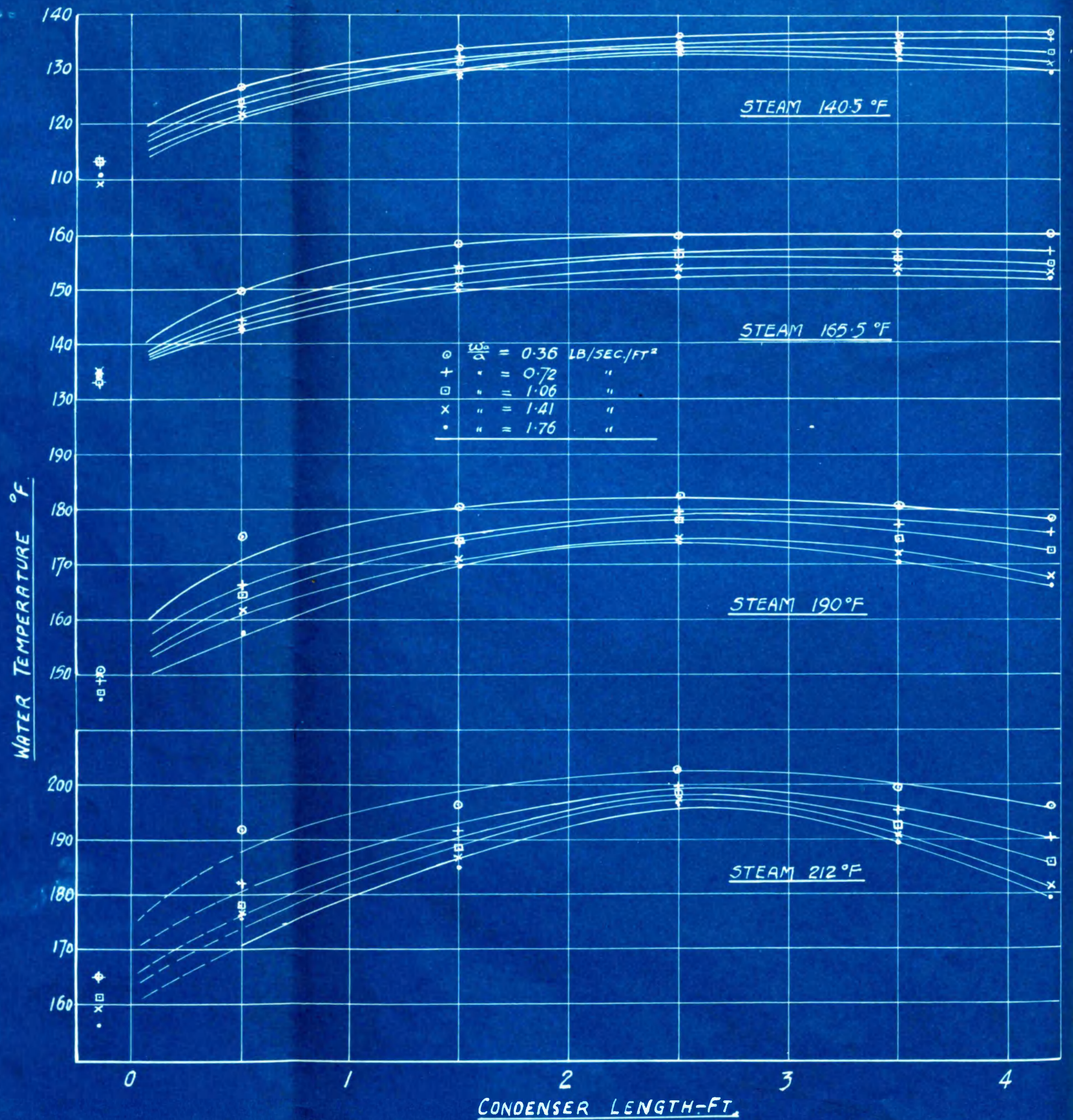


FIG 5.

WATER TEMPRS FROM THERMO COUPLES

The series represented in Fig. 4 comprised tests with varying water flow, and a constant air speed. The method of plotting indicates very clearly how little the evaporation rate is affected by alteration in the water supply, since the gradient is practically equal in each case.

(c) Measurement of Water Temperature by Thermo-Couples.

Before being put in position, the couples were calibrated in water, against a standard mercury thermometer, for temperatures from 60^{o}F to 212^{o}F. From this a graph was plotted of temperature difference (hot junction - cold junction) to a base of millivolts, on which a range of 150^{o}F corresponded to 3.8 millivolts. The scale on the millivoltmeter is subdivided into 0.01 m.v. During a test, the cold junction temperatures and the millivoltmeter reading for each couple, were taken at intervals, and the mean temperature readings plotted to a base of condenser length, as in Fig. 5. In addition there have been added to the figure, for each test, the inlet and outlet water temperatures, as recorded on the respective mercury thermometers. A curve was constructed through each set of points, and from this the mean water temperature over the tube was found. The figures show the effect of increased air speed on the water temperature; for a given air speed there was little variation in the water temperature with different water flows.

With a few exceptions the results may be considered very regular, in view of the comparatively delicate nature of the observation. The only irregularities encountered were with the top thermo-couple in the atmospheric steam tests, which appeared to read high. It was pbserved, however, that during these tests the inlet steam current impinging on the tube above the couple created a local turbulent effect, resembling ebullition; this disturbance which was entirely absent with the lower steam temperature tests was damped out very quickly. The only direct application of the mean water temperature is in the calculation of "h_1", the steam to water heat transmission coefficient, and in view of the above uncertainty, the "h_1" values with atmospheric steam have not been considered.

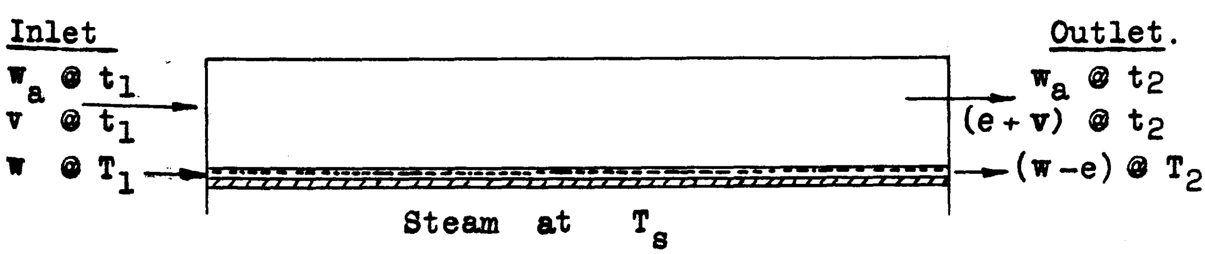
The falling off in water temperature in the latter end of the condenser may be taken as evidence of the increasing capacity of the air stream to absorb vapour, and therefore heat, from the water. This effect would not proceed indefinitely, however, as after a certain length equilibrium would finally be established when the air became saturated at the water temperature.

(d) The Heat Balance.

Throughout the remainder of the paper the following symbols are used.

T_s	-	steam temperature, $^\circ$F.
T	-	water " , $^\circ$F.
t	-	air " , $^\circ$F.
w	-	water flow, lb/sec.
w_a	-	air " lb/sec.
v	-	water vapour initially associated with air lb/sec.
s_a	-	specific heat of dry air.
s_v	-	" " water vapour.
p_a	-	partial air pressure, ins. Hg.
p_v	-	" vapour " ins. Hg.
H_t	-	total heat of water vapour at $t\,^\circ$F B.Th.U /lb.
h_T	-	liquid heat of water at $T\,^\circ$F, B.Th.U./lb.
e	-	water evaporated in condenser - lb/sec.

The total heat transmitted through the tube surface was calculated on the following basis. The inlet and outlet air conditions are conveniently represented diagrammatically as shown.



Then, Heat to dry air B.Th.U/sec. $= w_a.s_a.(t_2 - t_1)$ -------(1)

" " initial vapour " $= v.(H_{t2} - H_{t1})$ ----------(2)

" " evolved vapour, " $= e.(H_{t2} - h_{T1})$-----------(3)

" " surplus water, " $= (w - e).(T_2 - T_1)$-------(4)

The first three terms may be combined to give

Total Heat carried off by the air, B.Th.U./sec.

$$= w_a s_a.(t_2 - 32) + (v + e).H_{t_2} - w_a s_a(t_1 - 32) - vH_{t_1} - eh_{T_1}$$

$$= w_a \left\{ \left[s_a (t_2 - 32) + \frac{(v + e)}{w_a} \cdot H_{t_2} \right] - \left[s_a(t_1 - 32) + \frac{v}{w_a} \cdot H_{t_1} \right] \right\} - eh_{T_1}$$

$= w_a$ [Total heat of moist air at t_2, (vapr. content $\frac{v + e}{w_a}$ lb/lb air)

– " " " " at t_1, (" " $\frac{v}{w_a}$ lb/lb air)]

– eh_{T_1} ------------------(5)

Thus, Total heat transmitted through tube, B.Th.U/sec.

$$= \underline{\underline{(4) + (5)}}$$

Expression (5) is a very suitable form for calculation purposes as standard graphs of "Vapour Content", and "Air Total Heat" per lb. air may be readily plotted to a base of air temperature. These graphs vary a little with atmospheric pressure. The total heat of the air entering the condenser is known from its observed temperature and humidity. The vapour content of the air at outlet is also known from its initial humidity and the measured evaporation; and taking the air as saturated, the corresponding air total heat and mean temperature at outlet are got from the above mentioned standard charts.

In connection with the expression for the heat necessary for evolution of the vapour, an error frequently made is to assume that the heat required per lb. vapour is simply the Latent Heat at the water temperature, i.e.

Heat of Vaporization $= eL_T$ B.Th.U./sec. (a)

Comparing (a) with (3) above, it is evident that (a) only applies where the final temperature of the vapour is equal to the water temperature.

THE HEAT TRANSFER PROCESSES.

In the type of investigation at present under discussion, the ultimate aim of research may be regarded as the establishment of a working theory or hypothesis, which covers all experimental results and is readily applicable to practical use. In the development of such a scheme there are three distinct stages. First, a previous analysis of the variables involved and their incorporation in a simple rational theory, based on accepted fundamental physical laws; secondly, the application of this theory to given experimental results; and lastly, if possible, a comparison of the final values obtained with those of a related nature for already established theories. Such a line of develop:ment has been followed in the investigation described in this paper and the present section is devoted to a brief consideration of the principles governing the various heat transfer actions in the evaporative condenser.

There are four component processes involved. These are, first the heat exchange between the steam and cooling surface, and secondly, the metal to water interchange on the other side of the tube. At the water-air interface, two simultaneous actions are involved, the convective heating, by the water, of the air, and its associated vapour, together with the process involving the conversion of a portion of the cooling water into vapour at the air temperature.

<u>Steam-Metal-Water Heat Transfer</u>. In ordinary surface condenser investigations the laws governing the rate of heat transfer are stipulated from Reynolds' work, and, for the purposes of design, the experimental researches are concerned only with the determina:tion of a single curve, giving the variation of the heat trans:mission coefficient, from steam to water, on a base of cooling water speed. Strictly speaking, two curves are necessary, one covering steam to metal, and the other giving the transfer rate from metal to water, but it has been found that one overall curve meets all design needs, provided the steam side conditions are similar in the practical and experimental plants.

Passing to the evaporative condenser, so far as the metal ?, is aware, the cooling process is similar to that in a surface condenser. In the latter case, the water is cooled, e.g., by evaporation in cooling towers, between exit and entrance, while in the evaporative condenser the cooling of the water by the air takes place continuously along the condenser length. Otherwise, the metal-water heat interchange is identical in both cases, being proportional to the diffusion at the metal surface, and also to the temperature difference between metal and water. Consequently, an overall heat transmission coeffi: :cient may be also applied to evaporative condenser design.

For this new coefficient Reynold's equation must still hold, $H = (A + B\rho f(v))(T_s - T)$, but the addition of the evaporation effect modifies the usual application of the equation in two respects. The factor "A" is included to cover "the internal diffusion of the fluid when at rest", and in ordinary convective problems, this factor is often negligible. Consider, however, the case of evaporation taking place from a horizontal water surface. It then becomes possible for an appreciable heat flow to be established between the heating surface and the water, without the water being given a <u>positive</u> velocity. This is due to the flow of heat from the water surface into the air causing convection currents to be set up within the water. Thus, at zero water velocity,"h" still has an appreciable value. If the water is given a definite velocity past the surface, there will be a corresponding increase in the rate of heat transfer, as in pure convective heating. Thus, water velocity is still the correct basis of comparison for the heat transmission coefficient in evaporative condensers. An absolutely rigid estimate of this velocity is impossible as the nature of the flow in such steam-heated, air-cooled films is very indeterminate. A measure of the velocity, however, sufficiently accurate for a practical basis of comparison may be obtained if laminar flow is assumed in the film. As shown later, a mean water velocity may then be calculated from the standard hydro-dynamical relationships for fluid flow in open channels.

The effect of evaporation also causes a modification in the temperature difference factor. In the surface condenser the temperature difference between steam and water follows a hyperbolic law through the condenser, and the mean is calcul: :able on a log. formula, covering only inlet and outlet tempera: :ture difference. It may safely be said, however, that no general formula, involving only inlet and outlet water temperatures can possibly be used for the mean temperature difference in the evaporative condenser; some of the possible temperature variations are well illustrated in Fig. 5. The method employed in the present tests, of directly measuring the temperature difference between steam and water represents the only accurate means of determining "h_1". Fortunately the application to practice is not so difficult, as the water then circulates continuously without cooling, and suffers little temperature change through the condenser.

<u>The Heat Transfer at the Water Air Interface.</u> The process of the <u>convective heating of the air</u> and its entrained vapour is a simple heat transmission problem, analagous to the metal-air interchange in tubular air heaters. As such, it is reducible to a similar simple basis involving only a heat transmission coefficient, air speed and temperature difference. In addition, however, to the fluid diffusion set up by the relative motion between the air and the water surface, there is an additional amount of diffusion caused by the evaporation taking place at the surface.

i.e. $$H = \left(A + f(e) + B\rho f.(v_a)\right)\ (T - t).$$

The evaporation factor, f(e) is in turn a function of the air speed v_a, hence v_a may be conveniently taken as a basis of comparison, for the heat transmission coefficient, incorporating both diffusion effects. For a given air speed, the evaporation effect will be reflected in a higher value of "h", than for the usual metal air case without evaporation.

The process of evaporation of a part of the cooling water represents the outstanding feature of the condenser. The subject of evaporation from water surfaces into air has received a great deal of attention from physicists, and has long been recognised by them as a process of diffusion. Water vapour is regarded as a gaseous collection of water molecules. In intimate contact with any water surface, provided the water is not boiling, there is always a layer of air constantly being charged with molecules from the water and, therefore, saturated with vapour at a pressure corresponding to the water temperature. So long as the vapour pressure in this layer is greater than that in the air beyond, there is a steady diffusion outwards of the higher pressure vapour. The rate of diffusion, and therefore of evaporation, is thus dependent, other factors remaining constant, on the difference between the saturation pressure at the water temperature, and the mean vapour pressure in the main body of the air. In addition, in accordance with the fundamental laws of fluid diffusion, the rate of evaporation is also proportional to the eddy diffusion set up by the passage of a positive air current over the water surface. The process is very similar to the ordinary case of heat transfer by diffusion and may be represented by the same type of equation.

Thus, if E = rate of evaporation lb/ft^2/sec.

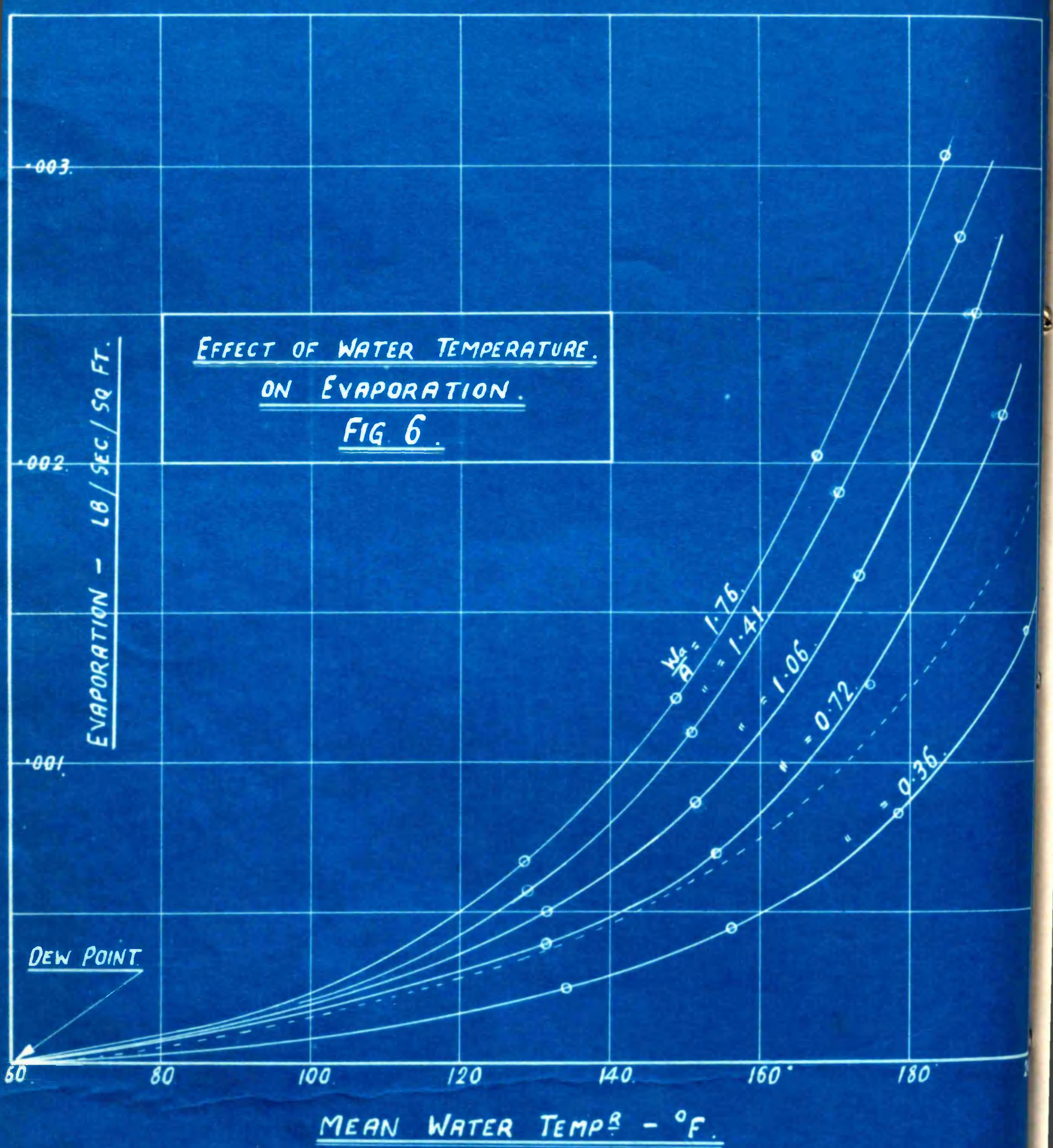
V_a = velocity of air relative to water surface ft/sec.

p_T = saturation pressure at water tempr.- ins.Hg

p_t = partial vapr. pressure in air - ins.Hg

Then $$E = \left(A + Bf(\rho V_a)\right).\ (p_T - p_t) \quad \text{----------(a)}$$

The factor, $\frac{E}{p_T - p_t}$, is the equivalent of "h", and may be termed the "Coefficient of Diffusion". As seen



EFFECT OF WATER TEMPERATURE. ON EVAPORATION.
FIG 6.

from eqn. (a), the proper basis of comparison for this coefficient is rate of air flow, as with the convective heating coefficient.

Since evaporation is purely a surface effect, the above laws should hold for the water film surfaces in the evaporative condenser, in the same way as for the surface of large water bodies. This is fully confirmed in the results of the present tests. For a given air speed, the sole factor controlling evaporation is the vapour pressure difference between the water and the air; neither film thickness nor water velocity have any discernible effect. The mode of evaporation is well illustrated by Fig. 6 in which Evaporation Rates have been plotted to a base of mean water temperature. The given curves are exactly similar in trend to those obtained by passing air over still water maintained at a constant temperature; such a curve, to a separate scale, is shown dotted. For the test series, the average dew point of the supply air was 60°F, and this, of course, marks the lowest water temperature at which evaporation is possible. From that point, the evaporation curves rise, with an increasing gradient, and finally tend to a very high rate of evaporation for boiling water.

Certain workers[x] in this field have encountered rather anomalous evaporation results in their work, such as higher rates of evaporation with low water temperatures than with water films much higher in temperature. These phenomena, however, have since been found to be due to ebullition caused by entrained air being deposited from solution in the water. In the present series of experiments, and in practice, where the same water is circulated continuously, the small air content of the water has negligible effect, and hence ebullition effects are absent.

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[x] e.g. Anderson, Journal, Royal Technical College, 1925.

Evaporation is also a heat transmission process, and the heat loss experienced by the water is a measure of the energy required to sustain the outward flow of water molecules. When the air above the water becomes saturated, the evolution of molecules from the water surface still continues, but these, instead of dispersing into the air, now return to the water at the same rate as they are evolved, and no further heat is extracted from the water. Since it is desirable to reduce all heat transfer processes to a temperature basis, an attempt was made to express evaporation as a function of temperature difference between water and air. It was found, however, that this could not be effected on a basis, sufficiently accurate and simple for practical application. In converting $(p_T - p_t)$ to a function of $(T - t)$, the saturation pressure p_T bears the following relation to T,

$$\log p_T = A - \frac{B}{T + 460} - C.\log.(T + 460)$$

If the air is saturated at t_1, p_t may also be expressed as above, but for unsaturated air,

$$p_t \overset{?}{=} \frac{P_A}{\frac{K}{w} + 1}$$

where P_A = atmospheric pressure and w = vapour content of air.

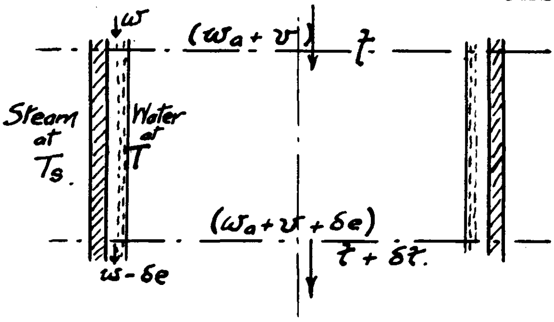
Either of these quantities may vary considerably and independently without alteration in t. K is a constant.

In the following treatment, therefore, the rate of heat transfer by evaporation has been expressed as a compound factor involving the product of heat quantities and pressure difference.

THE HEAT TRANSFER COEFFICIENTS.

From the considerations of the foregoing section it follows that a comprehensive working theory of Evaporative Condenser Design will require to incorporate at least three separate component theories, in order to take full account of all the possible variables. This combination is most suitably effected by the use of separate coefficients for each process concerned, as in the following development:

Consider a section of the condenser tube.



Over the element of water surface, δA, δe lb. water are evaporated.

As in the typical practical case, sensible heating of the water may be neglected, and the water temperature taken as constant at T.

Then if h_1 = heat transmission coefficient, steam to water, B.Th.U/Sec.×Sq.ft.× °F.

Heat transmitted from Steam to Water = $h_1(T_s - T).\delta A$ ----(1)

Also

Sensible Heat transmitted from Water to Air $= (w_a s_a + v.s_v)\delta t$------------(2)

and Evaporative " " $= \delta e(H_{t+\delta t} - h_T)$ -------(3)

And, since Heat from Steam = total heat to air.

$$h_1.(T_s - T).\delta A = (w_a s_a + v s_v).\delta t + \delta e.(H_{t+\delta t} - h_T) \text{ --(4)}$$

For design purposes the ideal arrangement of the right hand of (4), would be to cover both the contained terms under a single overall coefficient. As already noted, however the evaporation term does not lend itself to reduction to a temperature basis, hence the use of two coefficients is necessary for the total heat transmitted to the air.

Thus, if h_2 = heat transfer coefficient, for sensible heating of air by water, B.Th.U./sec.×sq. ft×°F.

Equation (2) reduces to

$$(w_a s_a + v.s_v).\delta t = h_2.(T - t).\delta A \text{ -----------(5)}$$

Also, if a new coefficient be taken for the mass diffusion of the vapour, the evaporation over the section may be expressed as

$$\delta e = C.(p_T - p_t).\delta A \text{ ---------------------(6)}$$

where C = vapour diffusion coefficient, lb/sec/sq.ft/in.Hg. vapour pressure diff.

and $(p_T - p_t)$ = vapour pressure difference.

between water and air.

The corresponding evaporative heat transfer in equ. (3) is thus expressed as

$$\delta e(H_{t+\delta t} - h_T) = C.(p_T - p_t).(H_{t+\delta t} - h_T).\delta A \text{-----}(7)$$

Eqn. (4) page 16 may then be re-written as

$$h_1.(T_s - T).\delta A = \left(h_2.(T - t) + C.(p_T - p_t).(H_{t+\delta t} - h_T)\right).\delta A, \text{----------}(8)$$

And, integrating over the total condenser area,

$$\underline{\underline{h_1.(T_s - T).A = \left[h_2(T - t_m) + C.p_m.(H_{t_2} - h_T)\right].A \text{------}(9)}}$$

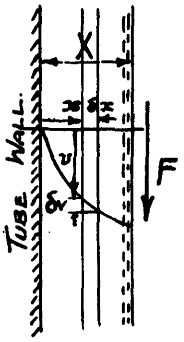
where t_m = mean air tempr. ^oF.

p_m = " vapour pressr. diff. between water and air.

t_2 = air temperature at outlet ^oF.

Eqn. (9) represents the fundamental equation for heat transmission in an evaporative condenser, and provides a complete and general basis for design, demanding only the provision of experimental values for the coefficients concerned. The primary object of the present investigation was to provide the necessary values for these coefficients, and the following sections deal with the methods of derivation and the final results obtained.

<u>Calculation of Water Velocity.</u> As a preliminary to plotting the coefficient curves, a method of calculating the mean and surface water velocities had to be derived, and the following basis was adopted.



For viscous flow on a vertical wall, the velocity distributn. in the film will be as indicated.

Let v = velocity flow, ft./sec., at a depth x ft. from the wall.

X = total depth of film - ft.

F = viscous drag of the air stream lb/ft^2.

w = specific weight of water - lb/ft^3

u = viscosity " " $lb.sec/ft^2$

W = total water flow, lb/ft. width of wall.

In the case of an elemental layer, δx thick and of unit area in a plane parallel to the wall, for unaccelerated motion,

Wt. of water in element = supporting viscous forces

$$\therefore w.\delta x = u.\frac{dv}{dx} - u\frac{d}{dx}(v + \delta v)$$

0

$$\therefore w = -u.\frac{d^2v}{dx^2}$$

$$\therefore -u.\frac{dv}{dx} = wx + A$$

But $(wX + F) = u\cdot\left(\frac{dv}{dx}\right)_{(x=0)}$

$$\therefore A = -(wX + F)$$

$$\therefore -u.\frac{dv}{dx} = wx - (wX + F)$$

$$\therefore uv = -\frac{wx^2}{2} + (wX + F)x + B$$

When $x = 0$, $v = 0$ $\therefore$ $B = 0$

$$\therefore uv = -\frac{w}{2}x^2 + (wX + F).x \text{ --------------(1)}$$

$\therefore$ From (1)

$$\text{Surface Velocity, } V_s = \frac{1}{u}\left(-\frac{wX^2}{2} + wX^2 + FX\right)$$

$$= \frac{1}{u}\left(\frac{w}{2}X^2 + FX\right) \text{--------(2)}$$

and Mean Velocity,

$$V_m = \frac{1}{X}\cdot\int_0^X (v.dx.)$$

$$= \frac{1}{u}\cdot\left[\frac{1}{3}\cdot wX^2 + \frac{1}{2}FX\right] \text{-----(3)}$$

Also, from the general equation for mass flow

$$X = \frac{W}{w.v_m} \text{ --------------(4)}$$

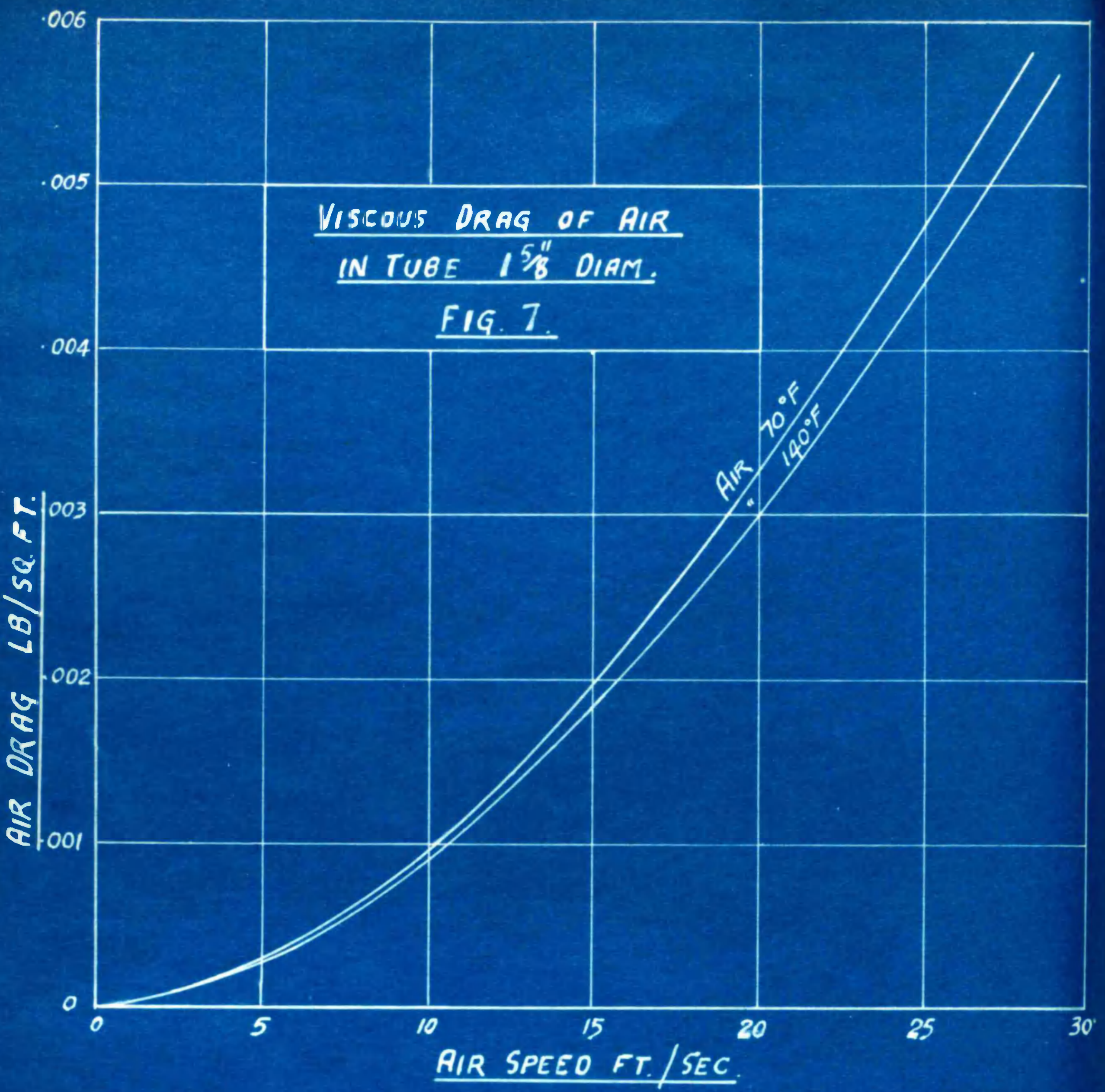
Substitution of eqn. (4) in equn. (3) gives

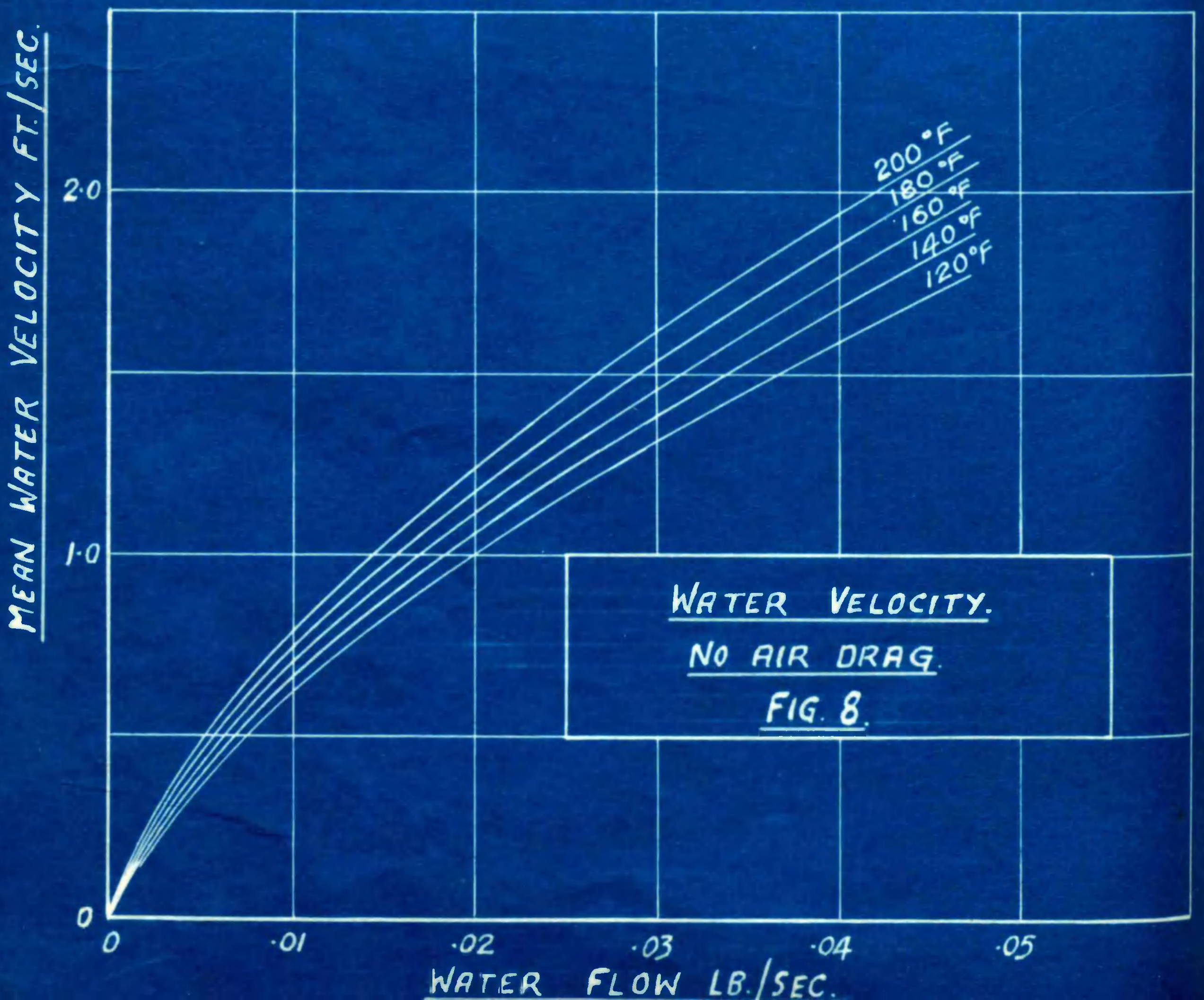
$$V_m = \frac{1}{u}\cdot\left[\frac{w}{3}\cdot\frac{W^2}{w^2 v_m^2} + \frac{1}{2}F.\frac{W}{w.v_m}\right]$$

$$\text{or } wu = \frac{W}{V_m^2}\cdot\left[\frac{W}{3v_m} + \frac{1}{2}\cdot F\right] \text{--------------(5)}$$

For a given test, the only unknown in eqn. (5) is V_m.

The right hand side may be plotted to a base of V_m, for selected values of V_m, ? and the interception with the horizon :tal through the ordinate at w.u, gives the required value of V_m.


VISCOUS DRAG OF AIR
IN TUBE 1 5/8" DIAM.
FIG. 7.
AIR DRAG LB/SQ. FT.
·006
·005
·004
·003
·002
·001
0
0
5
10
15
20
25
30
AIR SPEED FT./SEC.
AIR 70°F
" 140°F


WATER VELOCITY.
NO AIR DRAG.
FIG. 8.
MEAN WATER VELOCITY FT./SEC.
2·0
1·0
0
0
·01
·02
·03
·04
·05
WATER FLOW LB./SEC.
200°F
180°F
160°F
140°F
120°F

Where the viscous air drag is negligible, eqn. (5) reduces to

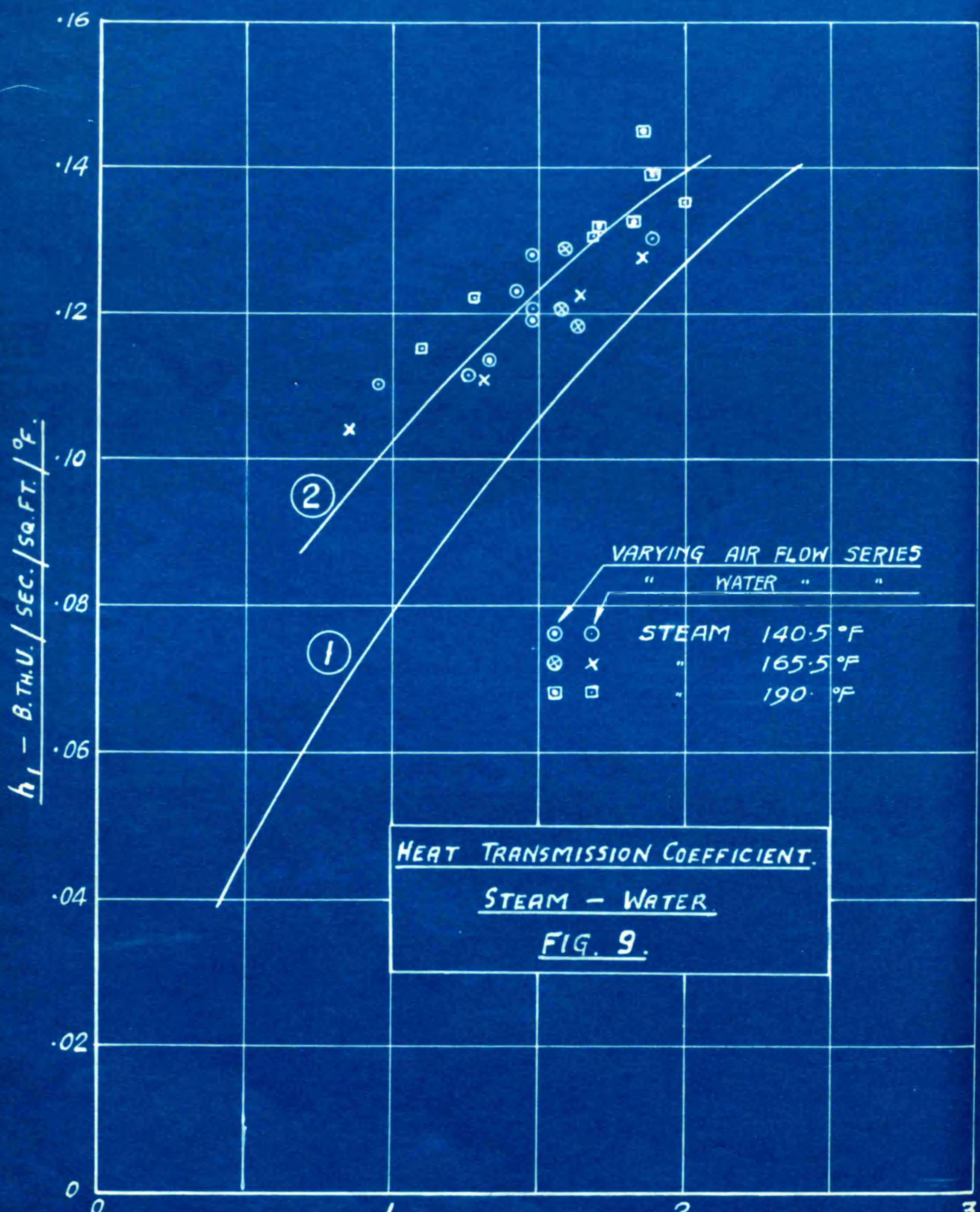
$$V_m = \left(\frac{W^2}{3\,u\,w}\right)^{\frac{1}{3}} \text{-----------------------(6)}$$

and eqn.(2) reduces to

$$V_s = 1.5\left(\frac{W^2}{3uw}\right)^{\frac{1}{3}} \text{-------------------(7)}$$

The effect of the air drag was examined, using the National Physical Laboratory curve for the skin friction of fluids in smooth tubes. Fig. 7 shows the derived curves of F, the drag in lb. per sq. ft, for air flowing in an $1\frac{5}{8}$" tube, at temperatures of 70°F and 140°F. These curves were applied directly to the present tests, there being assumed to be little difference between the resistance offered by the smooth water surface and that offered by a metal tube. For inter: :mediate air temperatures, interpolation was applied. In estimating F from Fig. 7, the relative velocity between the air and the water surface is the correct value of V to take. For this purpose, the ~~water~~ surface velocity may be taken from eqn. (7) with sufficient accuracy. In Fig. 8, eqn. (6) is plotted for various water temperatures. For convenience the base was taken as the total water flow on a surface, equal in width to the perimeter of an $1\frac{5}{8}$" tube, but by suitable modification of the base scale, the curves can be used for any width of surface. The equations for a plane vertical surface have been taken as applicable to the tube surface, as the dimensions of the film are small compared to the radius of curvature of the tube.

For the smallest air flow, the water velocity was obtained directly from Fig. 8, the air drag being negligible. The maximum increase in water velocity, at the highest air speed, was 6% over the velocity given by Fig. 8. For a given air flow, the percentage increase in speed is greater with the thinner films.



HEAT TRANSMISSION COEFFICIENT.
STEAM – WATER.
FIG. 9.

The Steam to Water Coefficient "h_1". The test series for the investigation of h_1, comprised sets of tests at four steam temperatures. For a given steam temperature, tests were carried out with four rates of water supply and a constant air speed.

For each test the total heat transmitted through the tube, and the mean water temperature, were found as described in Section 2. Denoting these respective quantities by H and T_m, then

$$h_1 = \frac{H \text{ B.Th.U/sec.}}{1.77 \text{ ft.}^2 \times (T_s - T_m)\,^\circ\text{F.}}$$

Fig. 9 gives h_1 for the vacuum tests, plotted to a base of mean water velocity, the velocity being calculated as described in the preceding sub-section. The curve (2) represents a mean line through the test points. Any irregular::ities in the points are entirely due to the small temperature differences which exist in the evaporative condenser; as a result, moderate experimental errors,in temperature measure::ment may result in appreciable percentage errors in h_1. Several broad conclusions, however, may be drawn from the figure, to which has been added a mean curve (1), for a small surface condenser employing an annular water flow space. The given curve is for steam condensing at a constant temperature of 212°F.

Referring to the evaporative condenser results, it may be stated that neither steam temperature nor air speed appear to have any very definite effect on h_1, for the tests given. With the atmospheric steam series, however, there was a tendency for the h_1 values to be higher.

The effect of the water velocity presents some interesting features. The maximum velocity of 2 feet per sec. represents a generous water supply to the condenser, such as would not likely be exceeded in practice. Thus, if an extrapolation is ventured, it is seen that in an average condenser, of the total value of h_1, only 50-60% will be due to velocity, and the remainder to the "still-water" effect. This serves to explain the apparent anomaly that higher rates of heat transmission are obtained with the laminar flow of the evaporative condenser than with the forced convection of the surface condenser, as represen:ted by curve (1). The fact, that an increase in the heat transfer rates is obtained with higher water speeds, has an important practical application. The prevalent idea regarding evaporative condensers appears to be that for efficient working and high evaporation, the water film must be kept as thin as possible. The results of these tests, however, would indicate that a more copious water supply would be detrimental neither to heat transfer nor evaporation, and, at the same time, the danger of dry patches developing on the tubes would be considerably minimised.

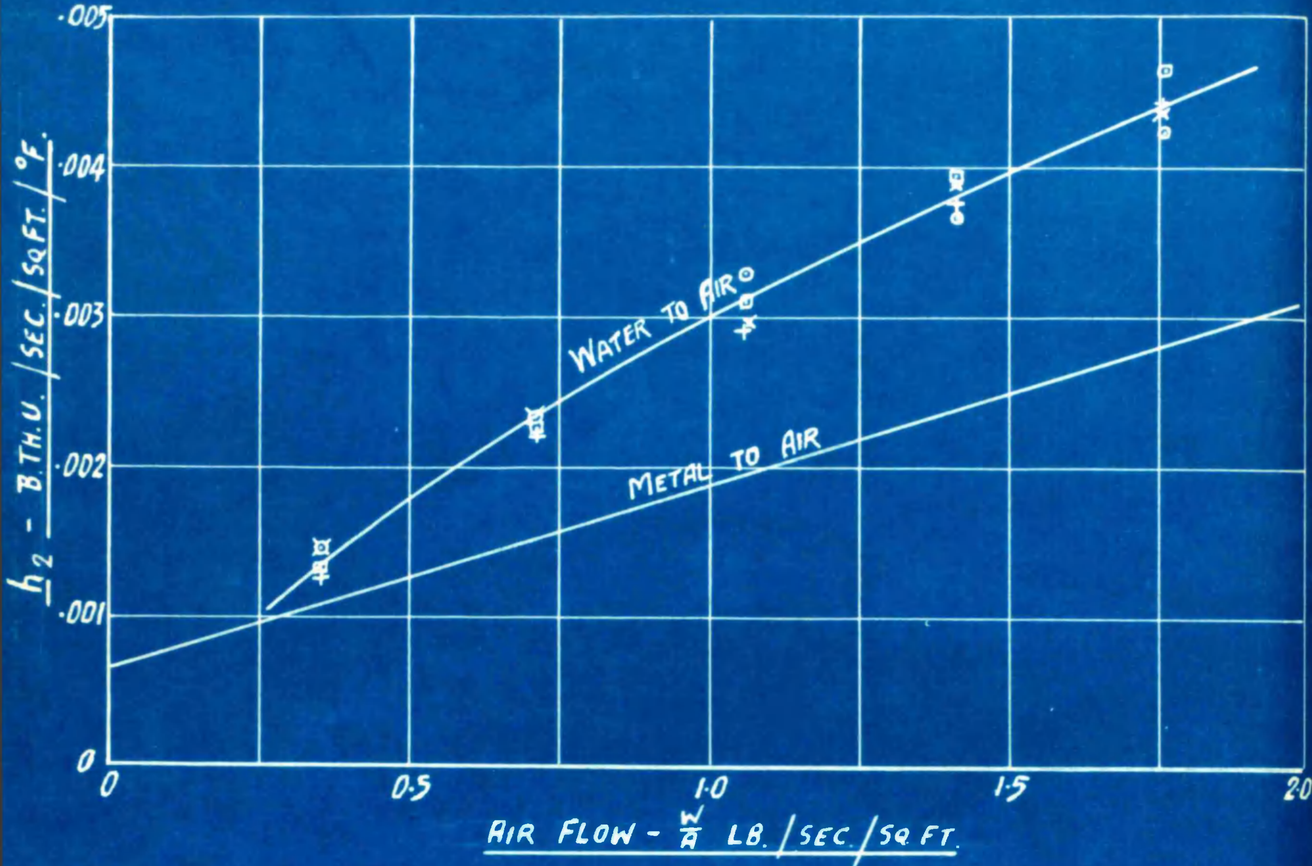


FIG. 10.

HEAT TRANSMISSION COEFFICIENT FOR CONVECTIVE HEATING OF AIR.

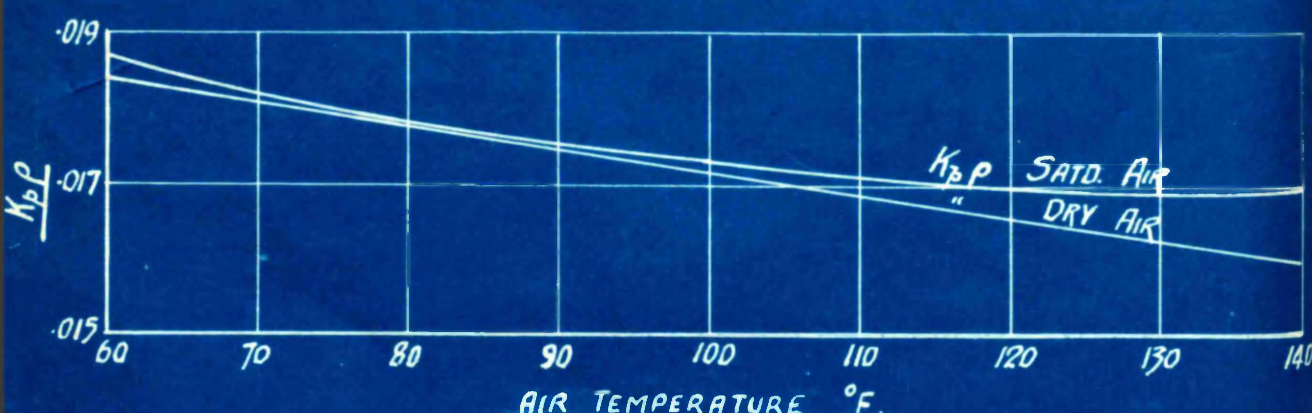


FIG. 11.

COMPARISON OF THERMAL CAPACITIES OF DRY & SATURATED AIR.

The Convective Heating Coefficient, Water to Air, "h"$_2$. The test series analysed for h_2, and also for the diffusion coefficient, comprised twenty tests, in all of which the water supply was kept approximately constant. Tests were run at four steam temprs. at each of which five air speeds were used.

In deriving h_2, the standard method for convective coefficients was employed, i.e.

Heat given to air = Wa. Sa. $(t_2 - t_1)$ B.Th.U./sec.

and Mean tempr. difference between air and water $\Big\}\ t_m = \dfrac{(T_1 - t_1) - (T_2 - t_2)}{\log e \dfrac{T_1 - t_1}{T_2 - t_2}}$ ^oF

Then, $$h_2 = \frac{Wa. \times .24 \times (t_2 - t_1)}{t_m \times 1.90 \text{ sq. ft.}} \text{ - B.Th.U./sec/ft}^2/^oF.$$

The water area used is the total tube surface exposed to the air flow. For T_2 the end Values of the curves in Fig. 5 were not taken for those tests in which the temperature curve fell towards the outlet end. In such cases, in order to apply the log. mean difference expression the water temperature was assumed to rise continuously while still maintaining the same mean temperature. The end values then obtained were taken for T_2, as given in Table 3. This ensures that the h_2 values do not err on the high side.

The calculated values of h_2 for all the tests are given in Fig. 10, and the results are exactly in accordance with theoretical considerations. The lower curve of "h" from metal to air, is taken from a paper by [x]Sneeden, and the similarity between the processes represented by the two curves is at once evident. For a given air speed the difference in the "h" values is a measure of the increased diffusion caused by the evaporation at the water-air interface.

The curve, for the present tests, given in Fig. 10 takes no account of the sensible heating of the vapour in the air, but its use can easily be extended to air saturated to any degree thus -

For a given air speed,

x Journal, Royal Technical College, 1926.

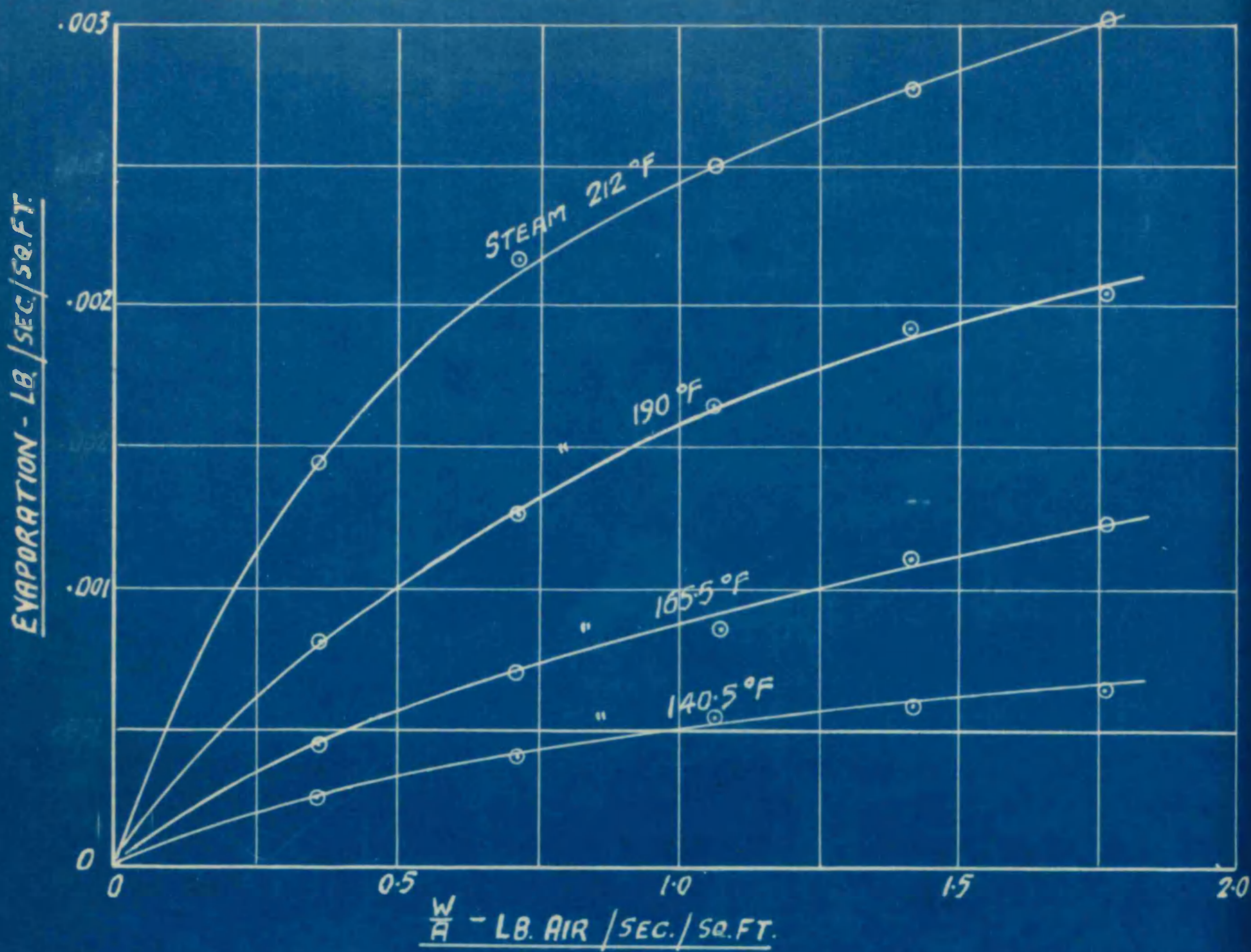


FIG. 12

EFFECT OF STEAM TEMP^R. ON EVAPORATION.

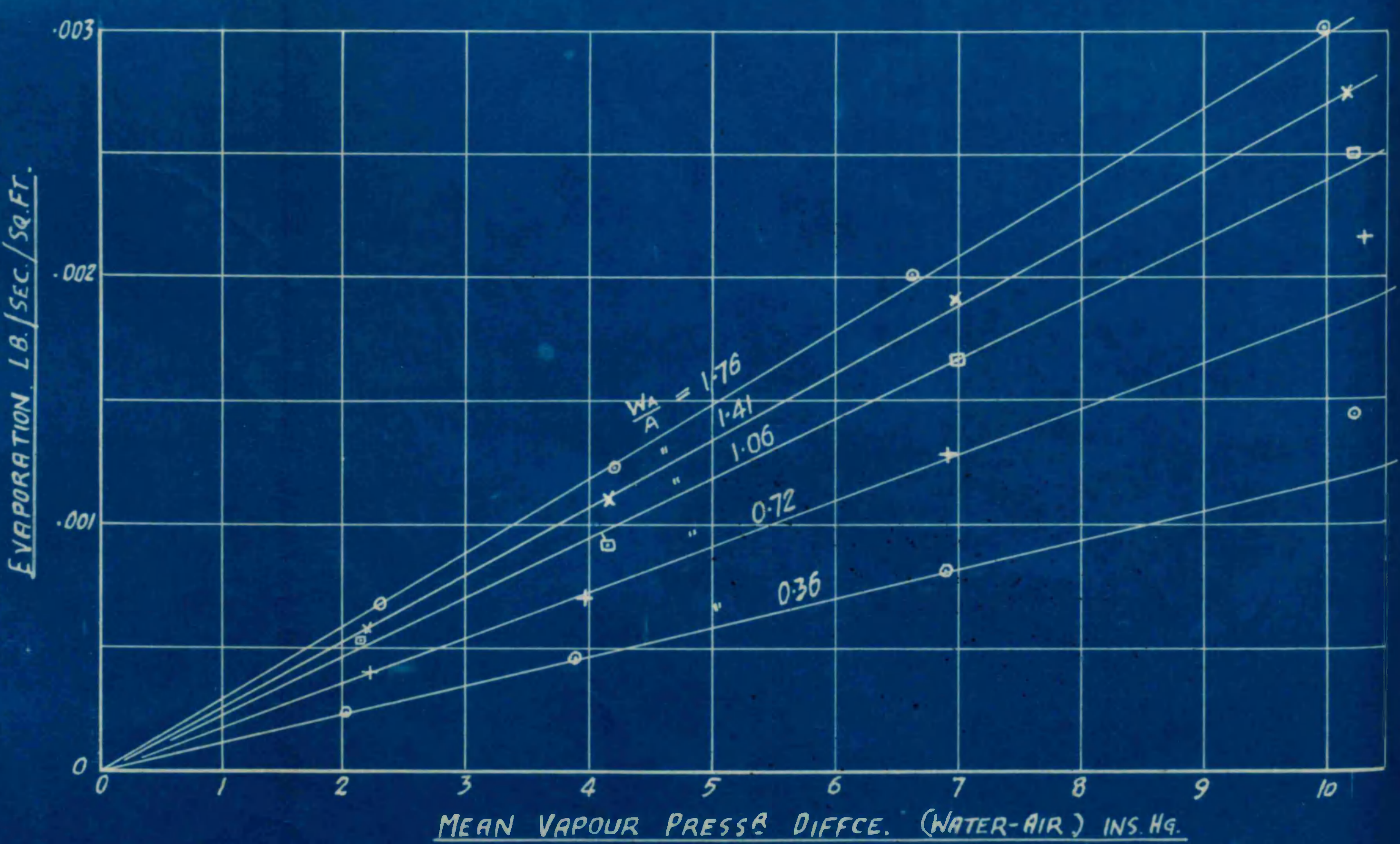


FIG. 13

VARIATION OF EVAPORATION WITH VAPOUR PRESS^R. DIFFERENCE.

$$\frac{h_2 \text{ for moist air}}{h_2 \text{ for dry air}} = \frac{K_p \times \rho \text{ for moist air.}}{K_p \times \rho \text{ for dry air.}}$$

where K_p = specific heat @ constant pressure.

and ρ = density.

<u>For dry air</u>, $K_p = .24$

and $\rho = \frac{144 \times pa}{53.3(t_a + 460)}$ lb/ft^3.

<u>For 1 lb of air and vapour mixture,</u> containing Wa. lb. air and W_v. lb. vapour.

$K_p = Wa. \times .24 + W_v. \times .46$ (.46 = mean spec. heat of water vapour.)

And if p_a = partial air pressure, lb/in^2,

ρ for the mixture $= \frac{144\ p\ a}{53.3\ (t - 460)} \times \frac{1}{W_a}$ lb/ft.^{3}

On this basis, the thermal capacities of dry and <u>fully</u> saturated air are compared in Fig. 11. It will be seen that up to about 110°F, there is little difference between the two gases as heat transmitting agents. This is an interesting point, in view of the fact that in at least one type of practical air condenser, the trouble has been taken of saturating the inlet air, possibly in the hope that a nearer approach to a water-cooled effect would result.

<u>The Vapour Diffusion Coefficient, C.</u> The successive stages in the reduction of the evaporation results to a fundamental basis are represented in Figs. 12-14. Figure 12. illustrates the general nature of the results obtained, and brings out well the comparatively low evaporation rates which may be expected at ordinary vacuum steam temperatures. For each test the mean vapour pressure difference between the water and the air was calculated; in no case was there much difference between the arithmetic and logarithmic mean of the inlet and outlet vapour pressure differences. The evaporation rates of Fig. 12 were then plotted to a base of mean vapour pressure difference, as in Fig. 13. Allowing for experimental aberrations, it is seen that the linear relationship between evaporation and vapour pressure difference is followed fairly closely. The probable explanation of the irregularities in the high temperature tests

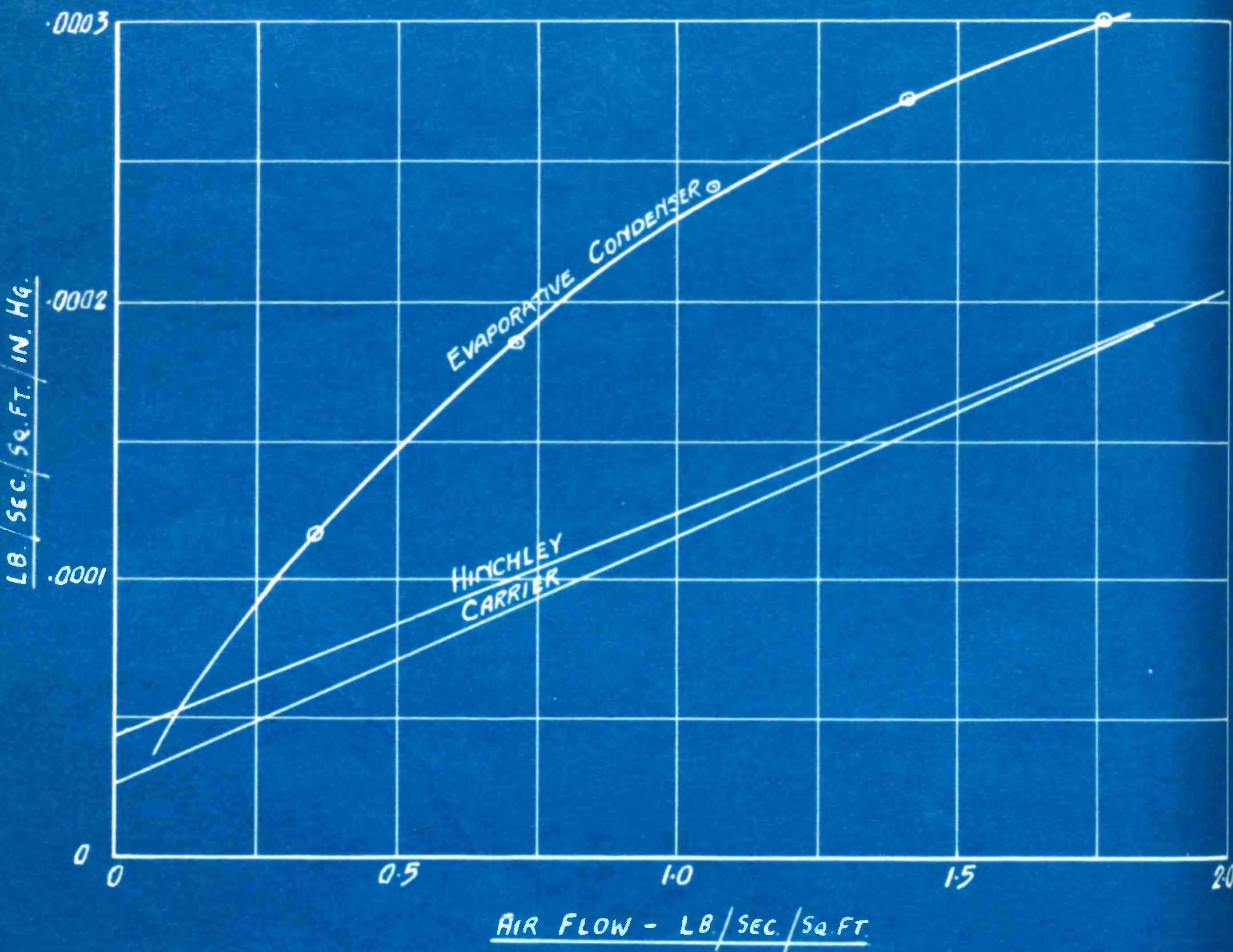


FIG. 14.

COEFFICIENT OF DIFFUSION.

is that the mean pressure diff. based on inlet water temperature is too low, since in these tests there was a sudden rise in water temperature near the inlet. Since the application of the derived coefficients, however, is to vacuum conditions, the mean lines have been kept below these atmospheric steam tests as shown.

The gradients of the straight lines in Fig. 13 give the mean diffusion coefficient for the respective air speeds, and the resultant coefficients are shown plotted to a base of air flow, in Fig. 14. The general nature of the curve is similar to the usual heat diffusion coefficient curves. As has already been mentioned, the subject of evaporation from still water surfaces has received much attention from other workers, and it is interesting to compare some of the figures obtained with the results from the evaporative condenser. In chemical engineering circles, the work of Carrier[1] and Hinchley[2] is outstanding. Each has carried out research of an eminently practical nature, and has developed a working formula connecting evaporation, air speed, and pressure difference. Their respective formulae, when converted to lb. ft. sec. units, and expressed in a coefficient form, give for C.

$C = .000026\,(1 + .26v)$ lb/sec/sq.ft./in.Hg. - Carrier.
$C = .000045\,(1 + .133v)$ do. do. - Hinchley.

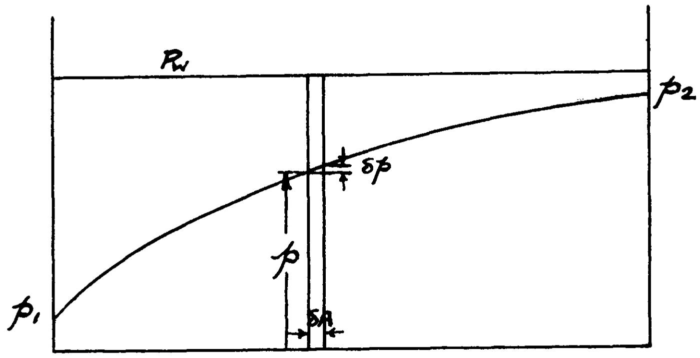
and the linear law is stated to be an approximation. The intercept on the zero air flow line covers the "still-air" condition, where, although there is no positive air current above the water, evaporation may still take place so long as a vapour pressure difference exists. This condition is usually maintained by the convection air currents set up above the water, which result in the continuous induction of fresh air to the water surfaces. The process is rather indeterminate and the "still-air" coefficient equally so.

The experimental conditions under which these results were obtained were vastly different from those existing in the

1 - Journal Industrial and Engineering Chemistry, May, 1921.
2 - Trans. Inst. Chemical Engineers, 1924.

experimental evaporative condenser used for the present tests. e.g. In Hinchley's work, the evaporation took place from a flat pan into a draught channel 1'6" x 9" high. In view of this fact, the comparative agreement of the results is interesting, when allowance is made for the increased diffusion caused by the motion of the water surface in the condenser. If the coefficient of diffusion, for a given air speed, can be shown to be very little affected by the dimensions of the air passage, such an experimental curve as given in Fig. 14, will have very great practical value. Its use need not be confined to evaporative condensers, but could be extended e.g. to the design of cooling towers; it would there be used in combination with the convective heating coefficient of Fig. 10.

Logarithmic Mean Pressure Difference. For the practical case of evaporation from water at a constant temperature into an air stream, the use of the arithmetic mean vapour pressure difference is not accurate and the logarithmic mean should be used as indicated in the following development.



If P_w = vapour pressure at water tempr. - ins. Hg.

p = " " in air "

the vapour pressure diagram for the condenser is as indicated.

In a mixture of air and water vapour,

$$\frac{\text{Weight of vapour}}{\text{" " air}} = \frac{\text{Spec. Vol. of air at partial air press. } p_a}{\text{" " " vapour " " } p_v}$$

$$= \frac{53.3 \times T \times p_v}{144 \times p_a \times .5948 \times T}$$

$$= \underline{\underline{.622 \cdot \frac{p_v}{p_a}}}$$

Hence, considering a section of the condenser surface, δA,

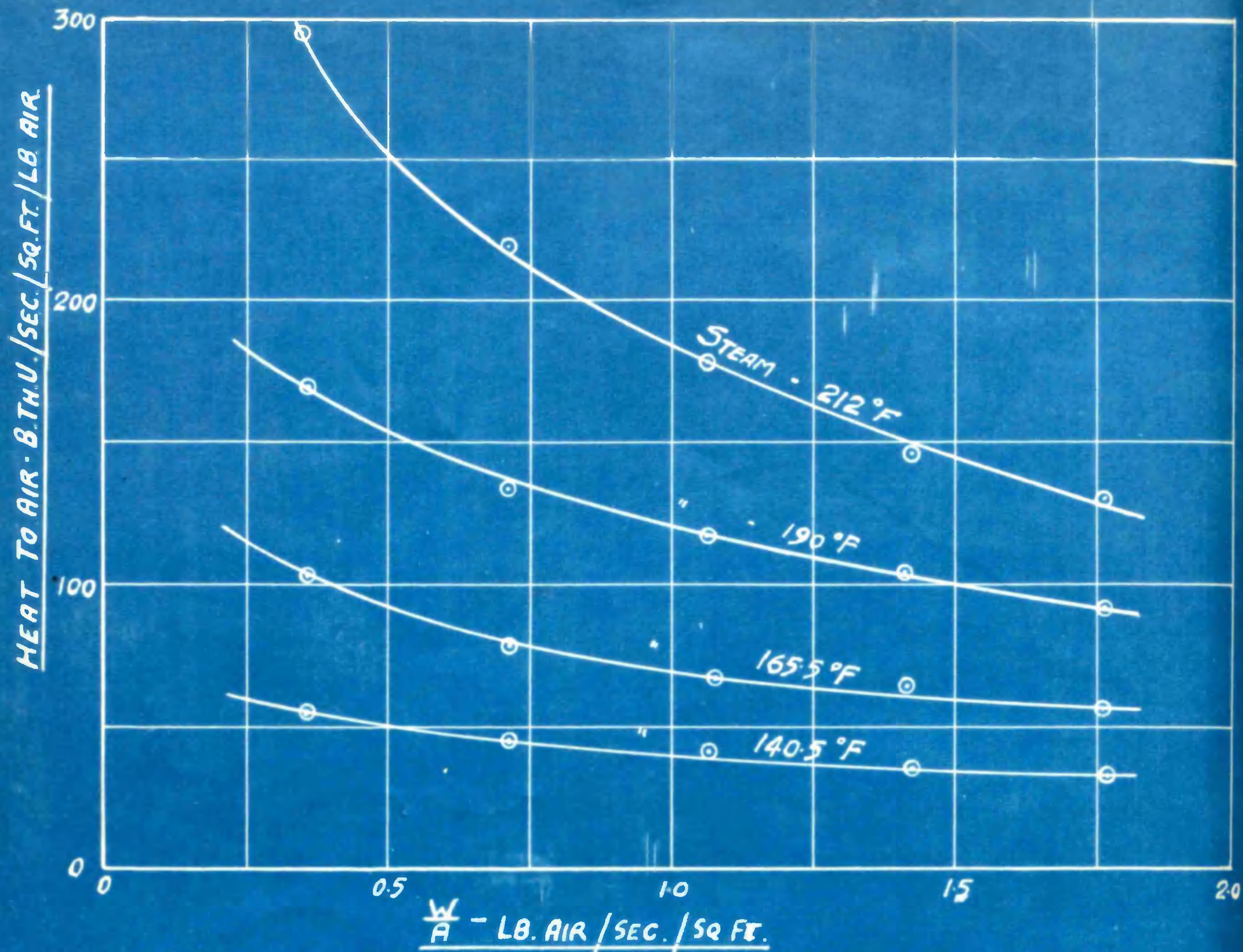


FIG. 15.

TOTAL HEAT TAKEN UP PER. LB. AIR.

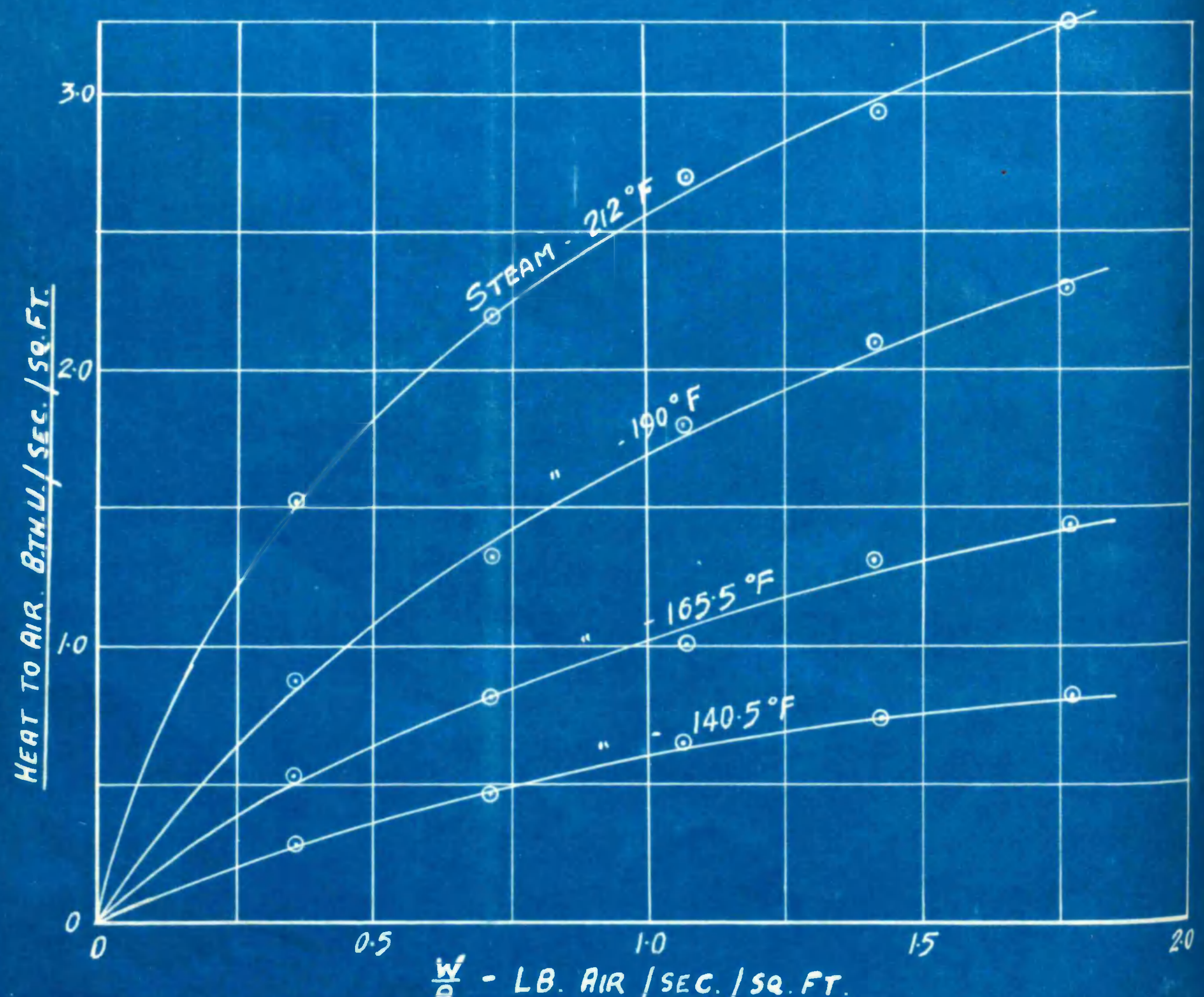


FIG. 16.

TOTAL HEAT TAKEN UP BY AIR PER SEC.

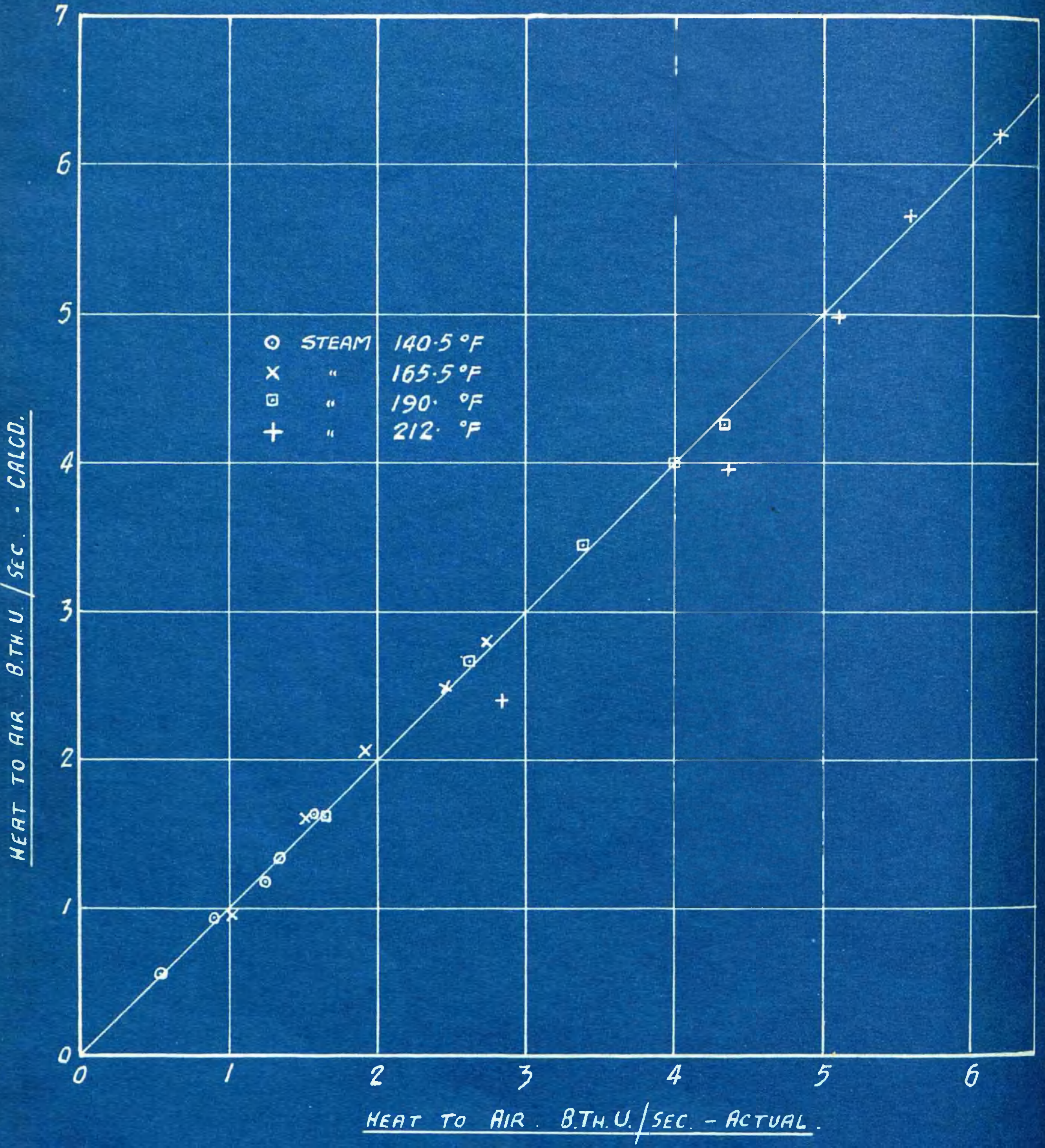


FIG. 17.

TOTAL HEAT TAKEN UP BY AIR

COMPARISON OF ACTUAL & CALCULATED VALUES.

$$\text{Evaporation over } \delta A,\ \delta e = C(P_w - p).\delta A \quad - \quad (1)$$

$$\text{Also} \qquad \delta e = .622 \frac{\delta p}{p_a} \quad - \quad (2)$$

In the average practical case, where the partial vapour pressures involved are of a low order, the absolute value of p_a varies very little through the condenser, and may be taken as constant.

$\therefore$ Equating (1) and (2)

$$.622 \cdot \frac{\delta p}{p_a} = C(P_w - p).\delta A$$

$$\therefore \int_{p_1}^{p_2} \frac{\delta p}{(P_w - p)} = \int_0^A \frac{C.pa}{.622} \delta A.$$

$$\therefore \log_e \cdot \frac{P_w - p_1}{P_w - p_2} = \frac{C.Pa}{.622} \cdot A.$$

And, for whole condenser,

$$C.\ pm.\ A = \frac{.622}{pa} \cdot (p_2 - p_1)$$

$$\therefore\ p_m = \frac{.622}{C.A.pa}(p_2 - p_1)$$

$$= \frac{p_2 - p_1}{\log_e \cdot \dfrac{P_w - p_1}{P_w - p_2}}$$

The remaining figures, 15-17, have been prepared as a matter of practical interest. Fig. 15 illustrates the fact that although increased air speed means an increased rate of heat transmission, it also results in a decreased usefulness, per lb. air circulated. Fig. 16 shows the overall effect of increased air speed on the heat carried off by the air, the plotted points being taken from the test results. Also, for each of these tests, the water and air conditions being known, the total heat transferred to the air was calculated by the use of the mean coefficient curves in Figs. 10 and 14. These Calculated and Actual Values were then plotted against each other, in Fig. 17; if complete agreement existed between the two sets of values, all points would fall on the 45° line shown. The object of plotting this figure is to demonstrate that the actual test results in general have not suffered undue distortion in being analysed to give the overall coefficients. The only tests which show any marked deviation from the mean line are the two atmospheric steam tests which gave the irregularities in Fig. 13.

PROCEDURE IN DESIGN.

For a given case of condenser design, the steam temperature and the required total heat transmission per second are known. In the case of the evaporative condenser, there are 3 expressions for this total heat, as derived previously in the paper.

$$\text{Total Heat/sec} = W_a \cdot \left[\left(S_a + \frac{v}{W_a} \cdot S_v\right) . (t_2 - t_1) + \frac{e}{w_a} (H_{t_2} - h_T) \right] \quad (1)$$

$$= h_1 . (T_s - T) . A \quad (2)$$

$$= h_2 . (T - t_m) . A + C . (p_T - p_m) . (H_{t_2} - h_T) A \quad (3)$$

A = total cooling surface - sq. ft.

p_m = mean vapour pressure in air - ins. Hg.

T = the water temperature, taken as constant.

As in surface condensers, certain preliminary stipulations are necessary. The following points would require to be previously decided.

Water Speed, and therefore, h_1.

Air Speed, and therefore, h_2 and C.

The average atmospheric temperature and humidity would fix the inlet air conditions.

In surface condenser work, the outlet water temperature is kept at a given amount, e.g. 8-16°F, below the steam temperature, and a similar stipulation requires to be made with regard to the outlet air condition in the evaporative condenser. As the question is mainly one of evaporation, a convenient way of stating this limiting condition is to fix the vapour pressure in the outlet air at a given amount below the absolute steam pressure. This is not an irrational basis, as, if the condenser were sufficiently long, the water would rise to the steam temperature with the air saturated, also at that temperature. In the highest vacuum tests of the present tests, the average vapour pressure difference between the steam and outlet air was 3 ins. Hg.

Having fixed the vapour pressure in the outlet air, as a safe approximation, the outlet air temperature is taken as the saturation temperature at this pressure. If the air is not completely saturated it simply means that the outlet temperature will be a few degrees higher.

A tentative estimate may then be made of the mean air temperature (t_m) and the mean vapour pressure (p_m) in the condenser.

Equating (2) and (3) above, then gives

$$h_1.(T_s - T) = h_2.(T - t_m) + C.(p_T - p_m).(H_{t_2} - h_T) \quad - - -(4)$$

The only unknowns in equation (4) are T, p_T and h_T.

Over short temperature ranges p_T can be approximately expressed in terms of T, e.g. from $T = 100^oF$ to $T = 110^oF$, the saturation pressure at any intermediate T is given by

$$p_T = 1.94 - .068(T - 100) \text{ ins. Hg.}$$

And $h_T = (T - 32)$ B.Th.U./lb.

Hence eqn. (4) may be solved for the required T.

The total condenser area, A, is then got from either eqn. (2) or (3). The total air flow, Wa, required is got from eqn. (1). Wa and $(\frac{Wa}{a})$ determine the air flow area, and if a specified size of tube is used, the total number of tubes also becomes fixed. The total perimeter of the air-flow cross section being then known, the total cooling surface divided by this perimeter fixes the tube length.

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TABLE I. VARYING WATER FLOW SERIES.

TEST Nº	94.	95.	92.	93.	56.	55.	53.	54.	64.	63.	59.	62.	79.	80.	77.	78.
STEAM. ABSOL. PRESS$^{\text{E}}$ INS. HG.	6.0				11.0				19.0				ATMOS.			
" SATURAT$^{\text{N}}$ TEMP$^{\text{E}}$ °F.	140.5	140.5	140.5	140.5	165.5	165.5	165.5	165.5	190	190	190	190	212.2	212.2	212.2	212.2
" OUTLET " °F.	136.5	136.4	136.8	136.6	164.0	163.3	163.7	163.5	189.9	189.9	189.9	189.9	212	212	212	212
" MEAN " °F.	138.5	138.5	138.7	138.6	164.8	164.4	164.6	164.5	190	190	190	190	212.2	212.2	212.2	212.2
WATER. INLET " °F.	110	109	109	112	132	134	133	134	145.5	146.5	150	152	155.5	158.5	165	163
" TEMP$^{\text{E}}$ @ T_4 °F.	125.5	127	129	129.5	147.5	149.5	152	152	164	166	171.5	171.5	186.5	188	189.5	191.5
" MEAN TEMP$^{\text{E}}$ FROM-TH. COUPLES °F.	129.9	129.1	129.1	129	152.4	151.9	152.1	152	175.3	175.4	175.1	175	197.6	197	197.1	196
MEAN TEMP$^{\text{E}}$ DIFF. STEAM-WATER °F.	8.6	9.4	7.6	9.6	12.4	12.5	12.5	12.5	14.7	14.6	14.9	15	14.6	15.2	15.1	16.2
WATER LEAVING CONDENSER LB/SEC	.016	.0245	.0315	.046	.0137	.025	.035	.042	.017	.021	.0325	.0425	.020	.0285	.0345	.0455
TOTAL EVAPORATION LB./SEC.	.00117	.00114	.00114	.00115	.00178	.00179	.00177	.00177	.00244	.00252	.0025	.00251	.0030	.0031	.0030	.00307
AIR. FLOW - LB./SEC.	.0204				.0154				.0103				.0051.			
" SUPPLY TEMP$^{\text{E}}$ °F.	70	70	70	70	69.5	69.5	69	69.5	70.5	71	70	70.5	72	71	71	72
INLET VAPOUR CONTENT LB/LB. AIR.	.0091	.0091	.0091	.0091	.009	.009	.009	.009	.0086	.0087	.0088	.0084	.0085	.0087	.0083	.0085
FINAL " " LB/LB AIR	.0665	.065	.065	.0655	.127	.125	.124	.124	.246	.253	.252	.252	.598	.616	.596	.611
FINAL TOTAL HEAT OF AIR BTHU./LB.	93.2	91.4	91.4	92	166.5	164	164	164	306	314	313	313	716	734	713	730
INITIAL " " " " " "	18.8	18.8	18.8	18.8	18.6	16.6	18.6	18.6	18.4	18.7	18.6	18.2	18.8	18.7	18.2	18.8
SENSIBLE HEAT OF EVAPORATION " "	4.5	4.3	4.3	4.5	11.8	11.8	11.6	11.7	26.9	27.9	28.6	29.2	72.7	77	78.2	78.9
HEAT TO AIR BTHU. / LB.	69.9	68.3	68.3	68.7	136.1	133.6	133.8	133.7	260.7	267.4	265.8	266.6	624	638	617	632
" " " BTHU. / SEC.	1.42	1.40	1.40	1.40	2.06	2.06	2.06	2.06	2.68	2.75	2.74	2.75	3.18	3.24	3.15	3.22
HEAT TO SURPLUS WATER BTHU/SEC.	.25	.44	.63	.81	.21	.39	.67	.76	.31	.41	.70	.83	.62	.84	.85	1.3
TOTAL HEAT FROM STEAM BTHU/SEC.	1.67	1.84	2.03	2.21	2.27	2.45	2.73	2.82	2.99	3.16	3.44	3.58	3.80	4.08	4.00	4.52
h_1 STEAM-WATER BTHU/FT^2/SEC °F.	.110	.111	.12	.13	.104	.111	.123	.128	.115	.122	.131	.135	.149	.154	.152	.16
MEAN WATER FLOW - LB/SEC.	.0166	.0251	.0321	.0466	.0146	.0259	.0359	.0429	.0182	.0223	.0338	.0438	.0215	.0301	.0360	.0471.
MEAN WATER VELOCITY FT./SEC.	.95	1.26	1.48	1.89	.84	1.32	1.65	1.86	1.10	1.27	1.68	2.00	1.26	1.60	1.80	2.15

TABLE 2. VARYING AIR FLOW SERIES.

TEST Nº	43.	44.	90	91	51.	52.	49.	50.	60.	61.	57	58.	73.	74.	75.	76.
STEAM. ABSOL. PRESSᴿ INS HG.	6·0				11·0				19·0				ATMOS.			
" SATURATᴺ TEMPᴿ °F.	140·5.	140·5.	140·5	140·5	165·5	165·5	165·5	165·5	190	190	190	190	212·2	212·2	212·2	212·2.
" OUTLET. " °F.	138·5	138·5.	137·8	137·6	164·1	164·1	163·3	163·4	189·7.	189·9	189·7	189·9.	212	212	212	212
" MEAN " °F.	139·5	139·5	139·2	139·1.	164·8	164·8	164·4.	164·5	189·9.	190	189·9.	190	212·2	212·2	212·2	212·2.
WATER. INLET " °F.	113.	113.	113	111	133·5	133	135	135	151	150	150	144	163	161	159	156.
" TEMPᴿ @ T4 °F.	137.	134·5	132	125·5	159	154	148	146	175·5	170	163·5	159·5	181·5	179·5	172·5	170.
" MEAN TEMPᴿ FROM-TH. COUPLES °F.	133·5	131·8	130·6	129	156·5	153·6	150·5	149	179·3	173	170	168·3	191·9.	188·3.	187.	184·8
MEAN TEMPᴿ DIFF. STEAM-WATER °F.	6·0	7·7.	8·6	10·1	8·3	11·2	13·9	15·5	10·6	17·0	19·9	21·7	20·1	23·7	25·0	27·2.
WATER LEAVING CONDENSER LB/SEC	·0335	·0325	·030	·026	·0335	·0345	·0310	·0315	·0385	·034	·036	·031	·034	·035	·0322	·030.
TOTAL EVAPORATION LB/SEC.	·000485	·00077	·00106	·00134	·00091	·00145	·00219.	·00241	·00164	·00317	·00374	·00404	·00444	·00507	·00546	·00596
AIR. FLOW - LB./SEC.	·00514	·0104	·0152.	·0255	·0515	·01035	·0202	·0253	·00516	·0151	·0202	·0253.	·0104	·0152	·0203	·0254
" SUPPLY TEMPᴿ °F.	66	67	70	69	68·5	70	68·5	68	69	69·5	70·5	70·5	70·5	70·5	71	71
INLET VAPOUR CONTENT LB/LB. AIR	·0075	·008	·009	·009	·0085	·0085	·0085	·0084	·0082	·0083	·0091	·0091	·0091	·0088	·0092	·0086
FINAL " " LB/LB. AIR.	·102	·082	·080	·0618	·186	·149	·117	·104	·326	·218	·194	·169.	·435	·342	·278	·244
FINAL TOTAL HEAT OF AIR BTHU/LB.	130·5	112	110	87·3	237	193	155	140	401	274	246	217	525	420	343	305.
INITIAL " " " " " "	15·9	16·4	18·9	18·9	17·8	18·2	17·8	17·6	17·2	18	19	19	19	18·6	19·4	18·7
SENSIBLE HEAT OF EVAPORATION. " "	7·7	6·0	5·8	4·1	18·0	13·9	11·2	9·8	37·8	24·8	21·8	17·9	55·8	43	34·2	31·6
HEAT TO AIR BTHU. / LB.	106·9	89·6	85·3	64·3	201·2	156·9	126	112·6	346	231·2	205·2	180	450	358	289	255
" " " BTH.U. / SEC.	·55	·93	1·30	1·64	1·04	1·62	2·54	2·85	1·78	3·49	4·14	4·55	4·68	5·44	5·87	6·48
HEAT TO SURPLUS WATER BTHU/SEC.	·81	·70	·57	·38	·85	·72	·40	·34	·94	·68	·49	·48	·63	·65	·44	·42
TOTAL HEAT FROM STEAM BTHU/SEC	1·36	1·63	1·87	2·02	1·89	2·34	2·94	3·19	2·72	4·17	4·63	5·03	5·31	6·09	6·31	6·90
h_1 STEAM-WATER BTHU/FT²/SEC. °F.	·128	·119	·123	·113	·129	·118	·120	·118	·145	·139	·132	·131	·149	·146	·143	·144
MEAN WATER FLOW - LB./SEC.	·0338	·0329	·0305	·0267	·034	·0352	·0321	·0327	·0393	·0356	·0379	·033	·0362	·0375	·0352	·033
MEAN WATER VELOCITY FT./SEC.	1·48	1·48	1·42	1·32	1·58	1·63	1·57	1·62	1·85	1·88	1·83	1·7	1·84	1·89	1·83.	1·76

TABLE 3. CALCULATION OF "h_2 & C.

TEST No.	43	44	90	92	91	51	52	53	49	50	60	59	61	57	58	77	73	74	75	76
WATER TEMPᴿ INLET TO TUBE °F	·113	·113	·113	109	111	133·5	133	133	135	135	151	150	150	150	144	165	163	161	159	156
AIR " " " " °F	72	70	70	69	68	75	73	71	69	69	78	75	72	71	70	85	68	68	71	69
WATER TEMPᴿ OUTLET FROM TUBE °F	138	136·5	135·5	134·5	133	160	157	155	153·5	151·5	182·5	179	177	175	174	201	198	195	193	192
AIR " " " °F	125·5	119·5	118·5	112	110	145	137	133·5	131	127	161	154	150	146·5	142	176	168·5	163	157	153
MEAN TEMPᴿ DIFFERENCE °F.	24	28	28	30·4	32	31·9	36·6	40	40·5	42·6	42	45·7	48	49	49·5	47·2	55	57·2	58	60
SENSIBLE HEAT TO AIR B.TH.U/SEC/SQ.FT.	·0347	·065	·093	·111	·135	·0465	·0837	·118	·158	·186	·054	·102	·149	·193	·23	·0587	·121	·167	·217	·26
h_2 - B.TH.U/SEC./SQ.FT./°F.	·00145	·00232	·00332	·00365	·00421	·00146	·00229	·00295	·0039	·00436	·00129	·00224	·0031	·00394	·00465	·00124	·0022	·00292	·00374	·00433
WATER VAPOUR PRESSᴿ INLET "HG.	2·85	2·85	2·85	2·55	2·70	5·0	4·9	4·9	5·18	5·18	7·8	7·6	7·6	7·6	6·5	10·8	10·3	9·9	9·4	8·8
AIR " " " "HG.	·35	·36	·43	·43	·44	·40	·40	·43	·40	·40	·38	·42	·39	43	43	·39	·43	·43	·43	·43
WATER " " OUTLET "HG.	5·6	5·38	5·25	5·10	4·9	9·7	9·0	8·60	8·25	7·8	16·25	15·0	14·25	13·6	13·3	24·0	22·6	21·2	20·3	19·9
AIR " " " "HG.	4·0	3·4	3·3	2·78	2·6	6·7	5·45	4·7	4·65	4·18	9·9	8·4	7·6	6·9	6·2	14·0	11·75	10·3	9·0	8·2
MEAN VAPOUR PRESSᴿ DIFFᴿ "HG.	2·02	2·22	2·17	2·22	2·28	3·88	3·98	4·18	4·16	4·20	6·89	6·89	6·93	6·94	6·59	10·2	10·36	10·19	10·14	9·95
EVAPORATION - LB/SEC./SQ.FT.	·000245	·000397	·000542	·000574	·000664	·000463	·00070	·000852	·00111	·00122	·000805	·00126	·00163	·00191	·00202	·00145	·00217	·0025	·00276	·00303