ON THE TRANSMISSION OF HEAT BETWEEN FLUIDS ACROSS AN INTERVENING METAL WALL, WITH SPECIAL REFERENCE TO AIR AND SUPERHEATED STEAM.

A THESIS

BY

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INTRODUCTION.

There are very few branches of modern engineering practice in which the subject of Heat Transmission in its widest sense does not enter under one aspect or another: from the simple process of proportioning a domestic hearth for a room of given size, to the more complicated design of a modern steam plant for a large power station.

It is also one of the subjects which lends itself to fruitful research, albeit the final results achieved seem very meagre when compared to the amount of preparation necessary to ensure a satisfactory degree of accuracy of these results.

It was on account of the possibilities of this subject that, as far back as 1920, Dr. A.L. Mellanby suggested that it should be taken up, and directed attention to the literature which should form the basis of any studies; Osborne Reynold's paper ("On the Extent and Action of the Heating Surface for Steam Boilers") and Nicolson's classical contribution to the Proceedings of the Institution of Engineers and Shipbuilders in Scotland (Boiler Economics and the use of high gas speeds) immediately decided the choice of that form of Heat Transmission which deals with the transfer of heat from one fluid to another through a metal wall, as a subject of research.

Various pieces of apparatus were placed at my disposal in the Laboratory of the Mechanical Engineering Department at the Royal Technical College, and I take this opportunity to express my thanks and my indebtedness to Dr. Mellanby, for his courtesy and helpful advice.

The first objective, after a study of the subject covering 3 or 4 years had been made, was to find out in what manner a new line of research could be initiated and developed. Numerous experiments were met with, carried out by renowned experimenters, dealing almost exclusively with hot air and water, dry or wet steam and water, but very few indeed concerning a fluid and metal surface. These numerous experiments are all then of what might be termed an overall character: for example, there are several experiments on the transmission of heat from Air (or hot flue gases) to water, of which the results are expressed as a plot of "heat transmitted per sq. foot of heating surface per hour/per °F. from Air (or hot flue gases) to water" on a base of air (or hot flue gases) speed, and this simply because it was experimentally proved that, within the limits of these experiments, there seemed to be very little change in the quantity of heat which was transmitted by altering the water speed while maintaining a constant flow of air (or hot flue gases), whereas the effect of varying the gas speed was more noticeable.

Until recent years, such plants as air preheaters were hardly deemed one of the necessary adjuncts of medium and large boiler plants, and even superheaters were probably designed by empirical rules, since as recently as 1928, the author came across a design of superheaters by Specialists in that class of work who used the relation heat transmission rate = constant x (temp. difference)^2. With the advent of such plants as air preheaters, superheaters, interheaters or resuperheaters and desuperheaters, together with the enormous proportions which these units may assume in modern plants, the necessity for accurate determination of the heating or cooling surfaces is of first importance and involves a more intimate knowledge of what exactly are the possible values of the heat transmission coefficients concerned.

It became quite clear that, since the speed of the hot gases (air, flue gases or superheated steam) had such an important bearing upon the magnitude of the heat transmission/

transmission coefficient when dealing with one of these gases and water, a similar and equally important effect would be produced if the water were replaced by any one of these gases as a cooling agent. Exactly in the same way, when hot oil is cooled by water, the speeds of both liquids could be expected to influence the overall rate of heat transmission to a more or less equal extent. An overall rate of heat transmission, in cases such as these, could not very well be expressed solely as a function of the speed of one of the fluids. The conclusion was that the overall method of research should be abandoned, at least temporarily, and something else substituted instead.

While Reynold's law, that the heat transmitted by each square foot of metal surface per unit time (H), is usually expressed in the form $H = (A + B\rho v)(T - \theta)$, it is often more convenient to transform it into $h = A + B\frac{w}{a}$. "h" is the heat transmitted between fluid and metal per square foot of surface per unit time; per °F difference of temperature.

With the second as the unit of time, and the British Thermal Unit as the unit of heat, this quantity will be called the "<u>Rate of heat transmission</u>" throughout this thesis, and various subscripts will be affixed to this symbol to enable it to be retained in cases where more than one heat transmission rate is concerned.

A & B are the usual Reynold's constants,
w is the rate of fluid flow in lb/sec.,
a is the cross sectional area of flow in sq.feet.
$\frac{w}{a}$ ($= \rho v$) will be called the "Mass-Flow Rate" throughout this thesis.

The equation, when written in this last form, has the advantage of containing only one variable $\frac{w}{a}$, and is to be preferred from this point of view.

Let any hot fluid flow on one side of a metal tube at a known mass flow rate $\frac{w_1}{a_1}$, to which corresponds the coefficient h_1, and let any cold fluid flow on the other side, mass flow rate $\frac{w_2}{a_2}$, coefficient h_2.

At any section of a thin tube, let T, θ, t, be the/

the temperatures of the hot fluid, the mean metal and the cold fluid respectively.

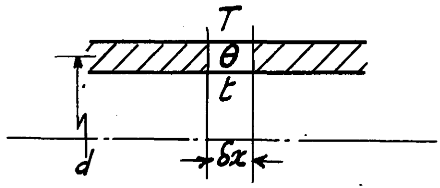


FIG. 1.

Over an infinitesimal length of tube δx, the temperatures will be sensibly constant, and if the mean metal surface be considered as the heat transmitting surface,

$$h_1 . \pi . d. \delta x. (T - \theta) = h_2 . \pi . d. \delta x. (\theta - t).$$

Again, if h_o = the overall coefficient, each of the above

$$= h_o . \pi . d. \delta x. (T - t).$$

Hence:- $$1 + \frac{T - \theta}{\theta - t} = 1 + \frac{h_2}{h_1}$$

$$\frac{T - t}{\theta - t} = \frac{h_1 + h_2}{h_1}$$

But, $$\frac{T - t}{\theta - t} = \frac{h_2}{h_o}$$

$$\therefore \frac{1}{h_o} = \frac{h_1 + h_2}{h_1 h_2}$$

or $$\frac{1}{h_o} = \frac{1}{h_1} + \frac{1}{h_2}$$

Thus the above consideration leads to the conclusion that the overall rate is dependent upon each of two individual rates which in turn are functions of the individual fluid speeds. HENCE, IN AN ABSOLUTE SENSE, NO CASE OF OVERALL HEAT TRANSMISSION RATE CAN BE REGARDED AS BEING DEPENDENT ON ONE OF THE FLUID FLOWS ALONE. This probably accounts for the fact that the results of different investigators show some lack of agreement: While each experimenter recognised the apparent ineffectiveness of the motion of the one specific fluid throughout the range of his own particular field of research, yet, the student who wishes to compare these several results must take account of the degree of ineffectiveness of that particular fluid.

As an example of the above, the surface condensation of steam by a current of cold water presents the strongest evidence. In no other instance of heat transmission work is there such wide disagreement between the results of different investigators. It is surprising to note how wide apart are/

are the plotted points showing overall h values on a base of $\frac{w}{a}$ for water.

Tables A and B have been compiled from W.G. Webster's data. (Proceedings of the Institution of Engineers and Shipbuilders in Scotland - 1913-14).

TABLE A. STEAM PRESS: 18·5 lb/in² abs.

1	No. of Experiment	34	35	36	37	81	46	60
2	Water speed ft/sec.	2.46	4.66	9.83	16.1	12.5	7.32	1.23
3	$\frac{w}{a}$ water.	153	291	615	1005	781	458	76.8
4	Heat/sec/sq.ft, water. H_w. B.Th.U.	32.7	43.2	55.8	69.7	69.0	54.3	23.9
5	Mean metal temp., water side θ_w °F	190	178	153	135.5	159.6	170	188
6	Mean water temp. t_w °F	73.4	64.1	57.6	54.3	56.8	59.1	84.4
7	$h_w = H_w/(\theta_w - t_w)$.	0.281	0.380	0.585	0.810	0.673	0.490	0.231
8	$\frac{1}{h_w}$, corrected to mean surface area.	4.90	3.52	2.35	1.70	2.05	2.81	5.96
9	Heat/sec/sq.ft. steam. H_s. B.Th.U.	19.8	26.2	23.8	42.3	41.8	32.8	14.5
10	$T_s - \theta_s$ °F	31.4	41.0	65.0	81.5	58.3	48.6	33.0
11	Steam speed ft/sec. Mean	36	32	34	30.5	68	54	22
12	$h_s = H_s/(T_s - \theta_s)$	0.63	.64	.52	.52	.715	.675	.440
13	$\frac{1}{h_s}$ corrected to mean surface area.	1.25	1.23	1.51	1.51	1.10	1.165	1.79
14	$\frac{1}{h_s} + \frac{1}{h_w} = \frac{1}{h_o}$	6.15	4.75	3.86	3.21	3.15	3.975	7.75
15	h_o	0.163	0.211	0.259	0.312	0.318	0.252	0.129
16	T_s °F	225.6	223.7	223.9	223.9	228.6	225.5	223.5
17	Overall mean temp. diff. t_m °F	152.2	159.6	166.3	169.6	171.8	166.4	139.1
18	Heat/sec/sq.ft. mean. H	26.3	34.7	44.8	56.0	54.9	43.6	19.2
19	$h'_o = H/t_m$	0.173	0.218	0.269	0.331	0.319	0.262	0.138

TABLE B. STEAM PRESS: 90 lb/in^2 abs.

1	No. of Experiment	71	75	98	99	100	101	102
2	Water speed ft/sec.	2.7	6.36	2.76	2.76	6.2	6.55	11.6
3	$\frac{w}{a}$ (water)	169	398	173	173	338	410	725
4	Heat/sec/sq.ft. water. H_w. B.Th.U.	60.5	77.3	62.5	67.5	74.5	87.7	112.5
5	Mean metal temp. water side (θ_w) °F	242.5	214.2	243.4	248	217.8	231.7	210.5
6	Mean water temp. (t_w) °F	90.7	70.6	95.3	98.9	73.2	76.2	68.2
7	$h_w = H_w/(\theta_w - t_w)$	0.398	0.538	0.422	0.453	0.516	0.565	0.790
8	$\frac{1}{h_w}$ corrected to mean surface area.	3.45	2.56	3.26	3.04	2.68	2.44	1.74
9	Heat/sec/sq.ft. steam. H_s. B.Th.U.	36.7	46.8	37.8	40.8	45.2	53.2	68.2
10	$T_s - \theta_s$ °F	72.3	98.5	71.5	67.3	97.0	81.3	100.7
11	Steam speed ft/sec. (mean)	5.2	11.3	10.3	18.0	8.5	27.2	32.0
12	$h_s = H_s/(T_s - \theta_s)$	0.508	0.475	0.529	0.608	0.466	0.655	0.678
13	$\frac{1}{h_s}$ corrected to mean surface area.	1.42	1.65	1.48	1.29	1.69	1.51	1.16
14	$\frac{1}{h_s} + \frac{1}{h_w} = \frac{1}{h_o}$	4.87	4.21	4.74	4.33	4.37	3.95	2.90
15	h_o	0.205	0.237	0.211	0.231	0.229	0.253	0.345
16	T_s °F	320.5	320	320.7	321.4	321.5	321.1	322.1
17	Overall mean temp. diff. t_m °F	229.8	249.4	225.4	222.5	248.3	244.9	253.9
18	Heat/sec/sq.ft. mean.H.	44.0	56.2	45.5	49.1	54.2	63.8	81.8
19	$h_o' = H/t_m$	0.192	0.221	0.202	0.221	0.219	0.261	0.322

(NOTE:- Since average metal temperatures only are given by Webster, the various mean temperature differences have been taken as arithmetical means, and heat transmission rates have been based upon these).

Line 14 shows the overall rate as calculated from the individual rates, after correction had been made for the mean surface area. Line 19 has been directly calculated from overall data, and the agreement with line 14 (15?) is sufficient to demonstrate the relation

$$\frac{1}{h_o} = \frac{1}{h_1} + \frac{1}{h_2}$$

Now it is very obvious that all the figures from line 12 which express the effect of the steam speed on the overall rate of heat/

heat transmission, on the steam side, are comparable to, or of the same order as the figures in line 7, which express the effect of the water speed on the water side. Indeed, under test 102, the one is 0.678 and the other 0.79, a ratio of 1 to 1.16. Under those circumstances, agreement cannot be expected amongst overall rates when plotted on a base of water speed.

It is only in a case such as air and water where the order of the individual rates is as 0.005 for air to 0.20 for water, that any such overall plot can have any significance and be of any use.

Again the full importance of the relation $\frac{1}{h_o} = \frac{1}{h_1} + \frac{1}{h_2}$ will be realised, when it comes to a question of obtaining values for h for a fluid which has not yet been experimented with.

For example: suppose the investigator desires to obtain h for a metal and superheated steam. Choosing as a heating or cooling fluid (as the case may be), air whose values of h to metal we shall say have been previously ascertained, he will obtain the following overall data:-

(1) air flow, steam flow;
(2) air temperatures and steam temperatures;

From these, the overall rate h_o is first calculated, and from the formulae $\frac{1}{h_o} = \frac{1}{h_{air}} + \frac{1}{h_{steam}}$

the value of h_{steam} can be found for various steam flows, while maintaining a steady air flow. But of course, this last condition is not sine qua non, since it is supposed that h_{air} is known as a function of the air $\frac{"w"}{a}$.

Hence the objects of experimenters should be directed towards establishing rates of heat transmission between 'metal' and various fluids. These would allow of the building up of a chart somewhat as per figure (2), over.

And it is towards this end that efforts have been directed in the experimental work which follows.

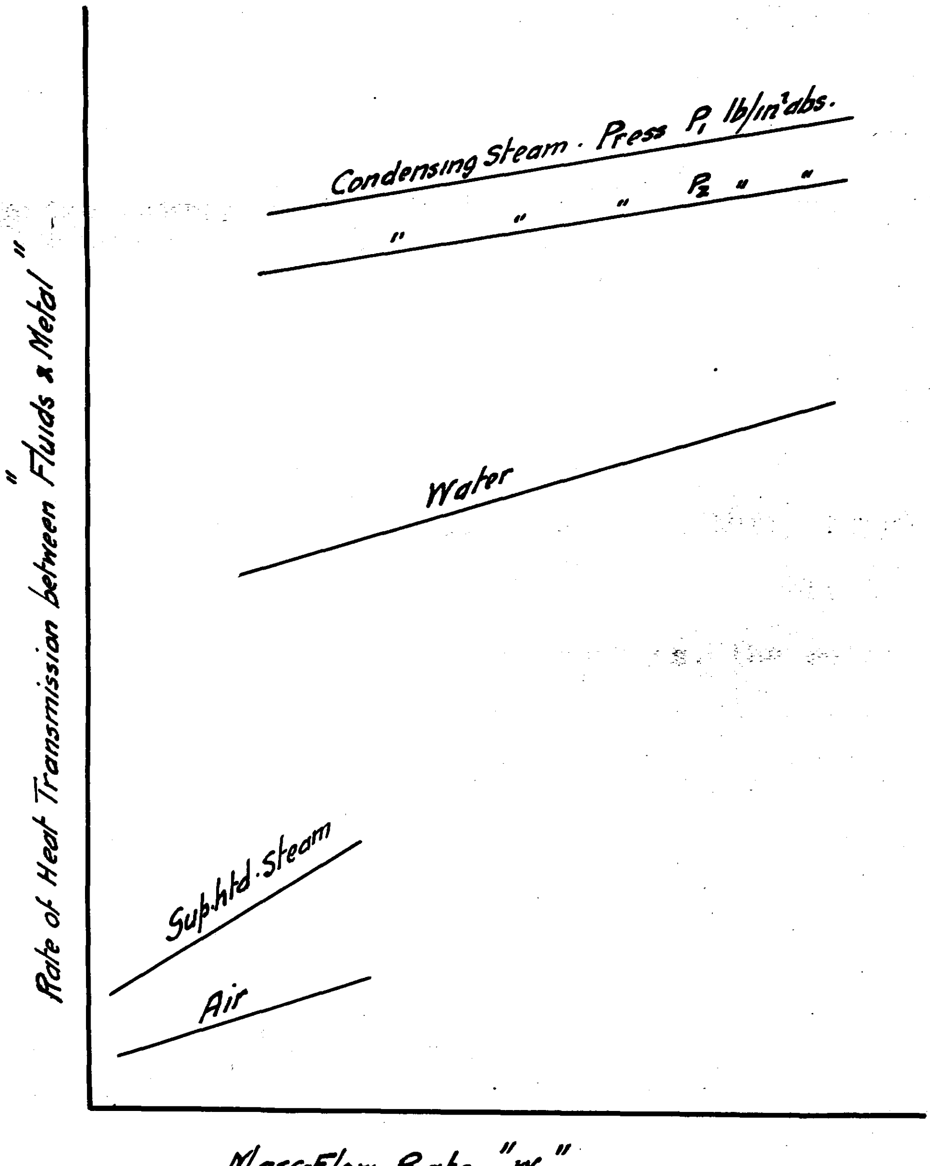


FIG. 2. Not to Scale

Hence, considering a small element of the tube δx, heat passing from hot gas to metal wall per unit time $= h_1(T-\theta).\pi d_1.\delta x$, heat passing from metal to cold gas wall per unit time $= h_2(\theta - t).\pi d_2.\delta x$, where d_1 and d_2 are the inside and outside diameters of the tube.

Again, heat lost by hot gas and that gained by cold gas per unit time, in passing over the element δx, are $-w_1 S_1.\delta T$ and $-w_2 S_2.\delta t$ respectively. S_1 and S_2 are the specific heats of the hot and cold gas respectively.

These four expressions being equal to one another, it follows that $-w_1 S_1 \delta T = h_1(T - \theta).\pi d_1.\delta x$

Again, $\theta = \theta_1 - m_1 x$

$$\therefore -\frac{\delta T}{\delta x} = h_1 \frac{(T - \overline{\theta_1 - m_1 x})}{w_1 S_1} . \pi d_1$$

In the limit $-\frac{dT}{dx} = \frac{h_1}{w_1 S_1}\left\{T - (\theta_1 - m_1 x)\right\}. \pi d_1$

$$\text{or} \quad \frac{dT}{dx} + \frac{h_1 \pi d_1}{w_1 S_1} T = \frac{h_1 \pi d_1}{w_1 S_1}(\theta_1 - m_1 x)$$

Let $\frac{h_1 \pi d_1}{w_1 S_1} = c_1$

then $\frac{dT}{dx} + c_1 T = c_1(\theta_1 - m_1 x)$;

multiply both sides by $e^{c_1 x}$

$$\frac{dT}{dx} . e^{c_1 x} + c_1 T e^{c_1 x} = c_1(\theta_1 - m_1 x).e^{c_1 x}$$

$$\text{or } dT.e^{c_1 x} + c_1.T.e^{c_1 x}.dx = c_1(\theta_1 - m_1 x)\, e^{c_1 x}.dx$$

Integrating both sides

$$Te^{c_1 x} = \int c_1(\theta_1 - m_1 x)e^{c_1 x} dx + C,$$

$$= c_1\theta_1 . \frac{e^{c_1 x}}{c_1} - c_1 m_1 \int x e^{c_1 x}\, dx + C.$$

To find $\int x e^{c_1 x}\, dx$,

$$\text{we have } \int x e^{c_1 x}\, dx = \frac{1}{c_1}.xe^{c_1 x} - \frac{1}{c_1}\int e^{c_1 x}\, dx$$

$$= \frac{1}{c_1} xe^{c_1 x} - \frac{1}{c_1^2}.e^{c_1 x}$$

$$\text{hence} \quad T e^{c_1 x} = \theta_1 e^{c_1 x} - m_1.e^{c_1 x}\left(x - \frac{1}{c_1}\right) + C$$

$$(T - \theta_1 + m_1 x)\, e^{c_1 x} = \frac{m_1}{c_1}.e^{c_1 x} + C$$

when $x = 0$, $e^{c_1 x} = 1$, $T = T_1$

$$\therefore \quad T_1 - \theta_1 - \frac{m_1}{c_1} = C$$

When $x = l$, $T = T_2$

$$\therefore (T_2 - \theta_1 + m_1 l)\, e^{c_1 l} = \frac{m_1}{c_1} . e^{c_1 l} + (T_1 - \theta_1 - \frac{m}{c_1})$$

$$\therefore (T_2 - \theta_2 - \frac{m_1}{c_1}) e^{c_1 l} = (T_1 - \theta_1 - \frac{m_1}{c_1}) - m_1 l\, e^{c_1 l} + \theta_1 e^{c_1 l} - \theta_2 e^{c_1 l}.$$

and since $m_1 l = (\theta_1 - \theta_2)$

$$\therefore \quad e^{c_1 l} = \frac{T_1 - \theta_1 - \frac{m_1}{c_1}}{T_2 - \theta_2 - \frac{m_1}{c_1}},$$

or $$c_1 l = \log_e \frac{T_1 - \theta_1 - \frac{m_1}{c_1}}{T_2 - \theta_2 - \frac{m_1}{c_1}},$$

or $$c_1 l = \log_e \frac{1 - \frac{c_1}{m_1}(T_1 - \theta_1)}{1 - \frac{c_1}{m_1}(T_2 - \theta_2)}.$$

Since all the values in this equation are determined experimentally, c_1 can be calculated most readily by the method of trial and error.

Also $h_1 = \frac{c_1 S_1 w_1}{\pi d_1}$, hence h_1 can be calculated.

The mean temperature difference tm_1 between the gas and the metal can be calculated thus

$$tm_1 = \frac{w_1 S_1 (T_1 - T_2)}{\pi d_1 . h_1}.$$

$\therefore$ the mean gas temperature is $= \frac{\theta_1 + \theta_2}{2} + tm_1$ (approx)

Proceeding along the same lines, and equating $-w_2 S_2 dt$ to $h_2 (\theta - t)\pi d_2 . \delta x$, it can be shown that a consideration of the conditions on the cold side of the tube would lead to similar equations, viz. :-

$$c_2 l = \log_e \frac{1 - \frac{c_2}{m_2}(\theta_2 - t_2)}{1 - \frac{c_2}{m_2}(\theta_1 - t_1)},$$ (m_2 is here equal to m_1)

$$h_2 = \frac{c_2 S_2 w_2}{\pi d_2},$$

$$tm_2 = \frac{w_2 S_2 (t_1 - t_2)}{\pi d_2\, h_2}.$$

It is now possible, therefore, to obtain the value of the rate of heat transmission from gas to gas, namely $h_o = \frac{1}{\frac{1}{h_1} + \frac{1}{h_2}}$

It will be observed that, provided values of h_1 and h_2 are available for different "mass flow," a value for h_o can always be calculated if necessary. Further, were the graphs of h_1 against

against $\frac{w_1}{a_1}$ and h_2 against $\frac{w_2}{a_2}$ to fall near enough to one another for a single curve to be taken in practice to represent both graphs, then h will be a maximum (for the highest value of h_2 permissible) when $h_1 = h_2$—i.e., when $\frac{w_1}{a_1} = \frac{w_2}{a_2}$; under these conditions it will be found that the performance of the heater will be improved. The performance on any preheater can be defined as the ratio:-

$$\frac{\text{actual rise in temperature of cold air}}{(\text{temp. of flue gas entering heater}) - (\text{temp. of cold air entering heater})}$$

Now, in one particular case where $\frac{w_1}{a_1} = 5{\cdot}14$, $\frac{w_2}{a_2} = 0{\cdot}58$, $h_1 = 0{\cdot}00858$, $h_2 = 0{\cdot}00130$, $T_1 = 430^\circ F.$, $T_2 = 319^\circ F.$, $t_1 = 75^\circ F.$, $t_2 = 185^\circ F.$, $w_1 = w_2 = 0{\cdot}00212$ lb./sec., A = heating surface = 2·016 sq.feet.

$\therefore$ coefficient of performance $= \frac{110}{355} = 0{\cdot}31$.

Suppose the outer pass where the cold air is flowing were redesigned so as to make its cross-section a_2 equal to a_1, so that $\frac{w_1}{a_1} = \frac{w_2}{a_2}$. (In practice, since w_1 and w_2 are very nearly equal to one another, $\frac{w_1}{a_1}$ will be very nearly equal to $\frac{w_2}{a_2}$ as a_2 is designed equal to a_1.)

Then, on the assumption of the single graph already mentioned, h_2 will be equal to 0·00858, and therefore h_o will be equal to 0·00429.

Let x denote the altered temperature of the hot gas leaving the preheater, and y that of the cold gas leaving the preheater, °F, the flow being counter-current.

Then $T_1 - x = y - t_1$

$$x + y = T_1 + t_1 = 505 \quad . \; . \; . \; . \; . \; . \quad (a)$$

Again, t_m, the mean temperature difference from hot to cold gas, =

$$T_1 - y = x - t_1$$

$$\therefore\; h_o t_m \cdot A = wS_1(T_1 - x) = wS_1(y - t_1)$$

$$0{\cdot}00429(T_1 - y) \times 2{\cdot}016 = 0.0212 \times 0.24 \times (430 - x)$$

$$\text{or } 8{\cdot}66y - 5{\cdot}1x = 1540. \quad . \; . \; . \; . \; . \quad (b)$$

Solving for x and y from (a) and (b) we have $x = 206^\circ F.$, $y = 299^\circ F.$

Hence the coefficient of performance $= \frac{224}{355} = 0{\cdot}63$.

This shows that, by careful designing and judicious choice of $\frac{w}{a}$, it is possible to construct a tubular heater with a performance which will compare very favourably with that of the regenerative types on the market, which claim a coefficient of/

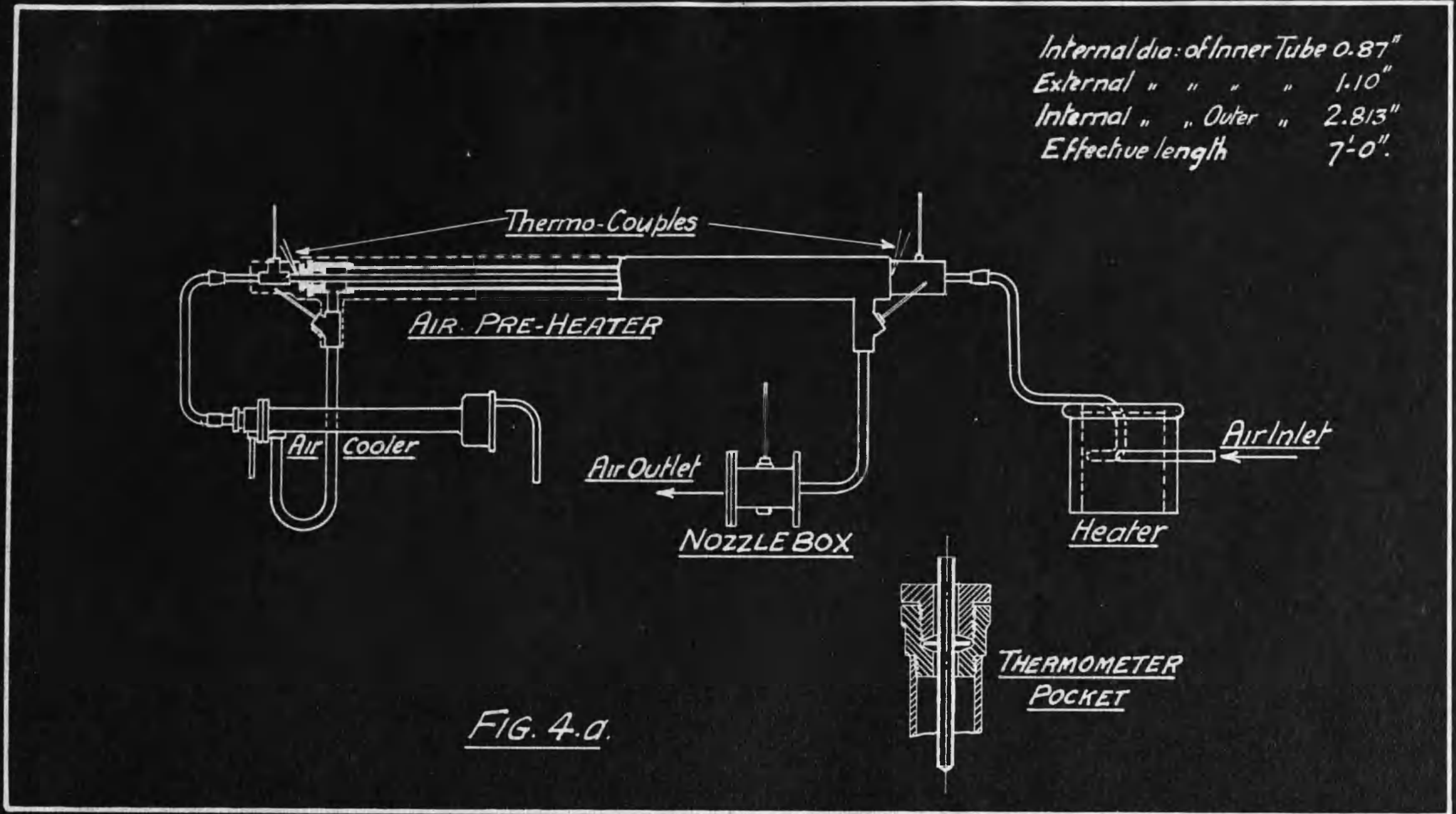

Internal dia: of Inner Tube 0.87"
External " " " " 1.10"
Internal " " Outer " 2.813"
Effective length 7'-0".
Thermo-Couples
AIR. PRE-HEATER
Air Cooler
Air Outlet
NOZZLE BOX
Heater
Air Inlet
THERMOMETER
POCKET

FIG. 4.a.

of 70 per cent. or more, and are usually more cumbersome.

The apparatus required to obtain the necessary data for the determination of the coefficients h_1 and h_2 previously mentioned, was constructed in the Laboratory of the Mechanical Engineering Department of the Royal Technical College. A large number of experiments were carried out, the first set of trials took place about four months after the erection of the plant and the second set a year later, thus the heat-transmitting surfaces must have become coated with rust and dust. Each test was conducted with great care, while very few of these lasted less than one hour. The results of these trials may, therefore, be looked upon as representing with fair accuracy what might be realized in actual practice with a preheater of a similar type, i.e., of tubular form. It must be pointed out at this stage that the material available did not permit of the two areas of flow a_1 and a_2 being anywhere near to one another, but they were made in the ratio 0.00413 to 0.0365 or 1 to 8.84; this will explain the low efficiency of 0.31 which has been previously mentioned.

To facilitate the description of the plant, shown in Fig.4a the latter has been divided into four sections or elements, which in their relative positions are in the following order:- (1) the air heater,* (2) the air preheater, (3) the air cooler, and (4) the nozzle-box.*

The air heater consists of a coiled pipe placed inside a cylindrical steel shell lagged with asbestos. One end of the coiled pipe is connected by a valve to the pipe from the receiver of an air compressor, and the other end is connected to the inner pipe of the preheater by a pair of flanges separated by means of a thick piece of asbestos sheeting.

The coil was heated by means of an ordinary gas ring.

The air preheater consists of two concentric steel pipes. The air enters from the heater, passes through the inner tube and leaves at the other end into the cooler, re-enters from the cooler into the annular space of the preheater and is discharged into the atmosphere through the nozzle-box. The joint between/

* For details see Appendix II

between the inner pipe and the cooler is also of the flanged and insulated type. This type of joint was found necessary, in order to reduce to a minimum the conduction losses which otherwise caused differences of 10 per cent. and over in the heat balances. The figures employed in the plot shown in Fig. 5 are derived from those tests where a difference of less than 10 per cent. obtained in the heat balance, which may be accounted for by radiation and conduction losses.

The air temperature at entrance to the preheater and at the nozzle-box was measured by mercury thermometers inserted directly into the air current, whilst at the other three points indicated mercury thermometers were placed in thin steel pockets insulated from the steel containers. The temperature of the metal of the inner tube was measured by means of thermocouples, of copper and Constantan wires, placed at the two ends of the pipe, but quite close to the outer pipe.

The cooler is simply a small high-speed water-cooled condenser which permits the cooled air being maintained at a fairly constant temperature before re-entering the preheater.

The nozzle-box is a cylindrical casting about 6" diam. X 10" long, fitted with a thermometer at the top, a pressure tube at the bottom, and a nozzle-plate at the end, while fine wire gauze discs are placed at intervals inside to break up the flow of the incoming air, and allow it to come to rest before being discharged through a smooth convergent nozzle into the atmosphere. A nozzle, $\frac{1}{4}$" in diameter, was used, the air temperature in the box was measured by a mercury thermometer with the bulb directly exposed to the air, and the air pressure registered by a column of mercury.

The method adopted in conducting the experiments was to heat the air before its entrance to the preheater and to maintain it at a temperature as constant as possible once the air used was flowing at a steady rate. Next, the air leaving the preheater was cooled to a predetermined temperature before its re-entrance to the preheater and maintained at this temperature throughout all the tests.

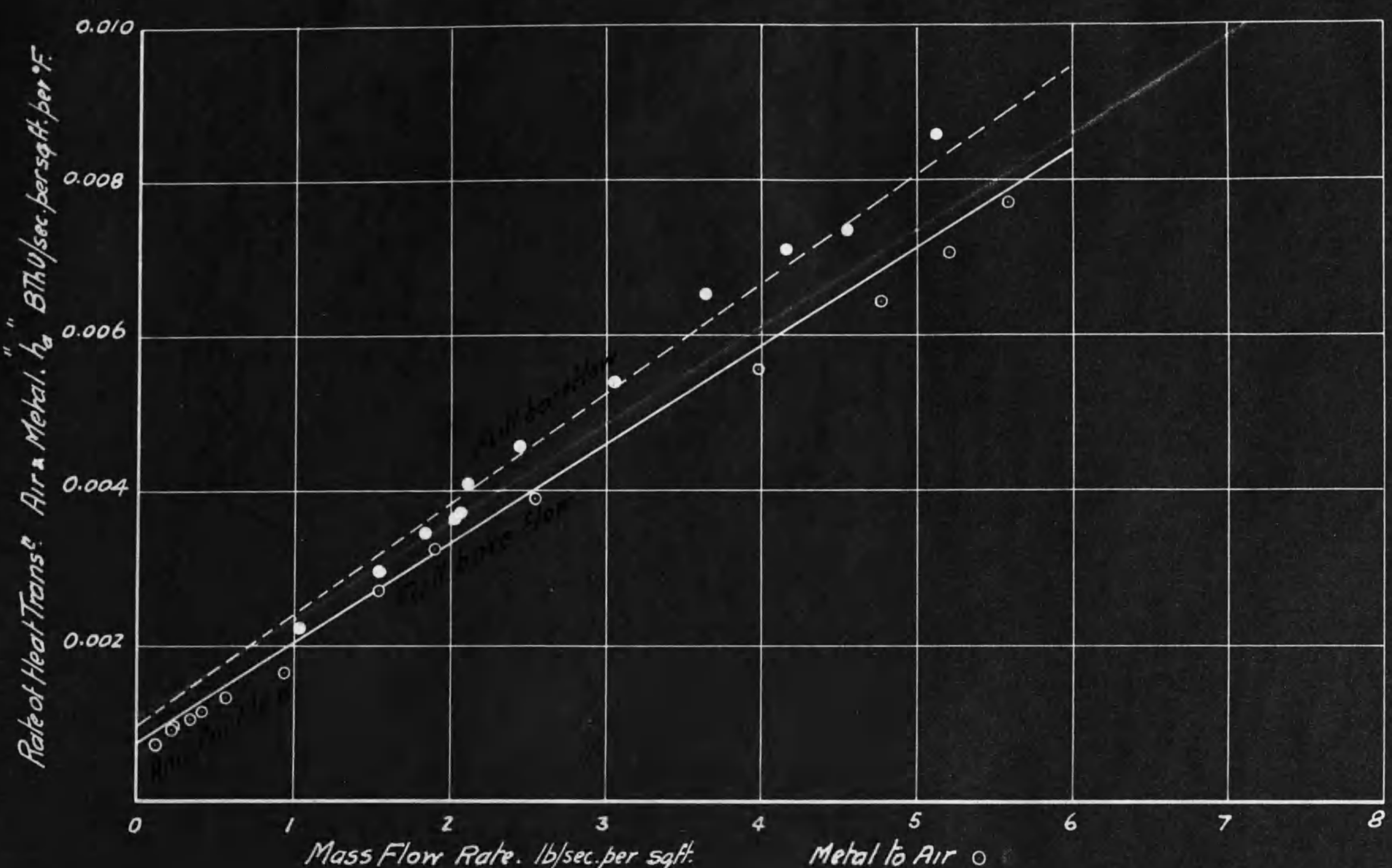

Rate of Heat Trans.^e Air ⇄ Metal. h_a BTh.U/sec. per sq.ft. per °F.
0.010
0.008
0.006
0.004
0.002
0
1
2
3
4
5
6
7
8
Mass Flow Rate. lb/sec. per sq.ft.
Metal to Air ○
Air to Metal ●

FIG. 5

The air quantities were varied from test to test, but the above temperature conditions were maintained throughout the one set of experiments. Several sets were carried out with different initial hot-air temperatures. The values of h_1 and h_2 were deduced as has been indicated previously, and these have been plotted on a base of massflow in Fig. 5.

Another series of tests has been carried out with the cold air flowing in the inner tube and the hot air in the annular space, but only the cold side has been dealt with and therefore only h_2 appears on the graph. The main readings for these tests are shown in Table C in the Appendix.

It will be noticed that the rate of heat transmission from metal to air does not seem to be affected to any considerable extent, whether the flow is annular or full bore, as indicated by the full line; whereas there is an appreciable difference between h_1 and h_2 especially at the higher velocities, as shown by the relative positions of the broken and full lines; neither does a variation of 100°F. in the initial temperature of the hot air seem to affect these coefficients to any great extent.

In conclusion, taking into account all the possible factors, both known and unknown, which may in practice affect the results which any plant is likely to yield, the author is of opinion that the full-line graph given in Fig. 16 may be safely used by designers in the manner set forth elsewhere in this Thesis.

It has been pointed out in the Introduction, that (for steam and water), plotting overall rates of heat transmission on a base of massflow for one of the fluids does not always yield satisfactory results. Yet for practical purposes this is a most desirable procedure and should be done wherever possible. It is thus relevant at this point to find out the approximate speed at which the water side ceases to affect the overall rate of heat transmission when "air to water" are the fluids concerned.

To that end Table D has been compiled.

Column (1) indicates air mass-flow rates $\frac{w}{a}$;

Column (2) indicates h_a (air to metal) from the preceding experiments.

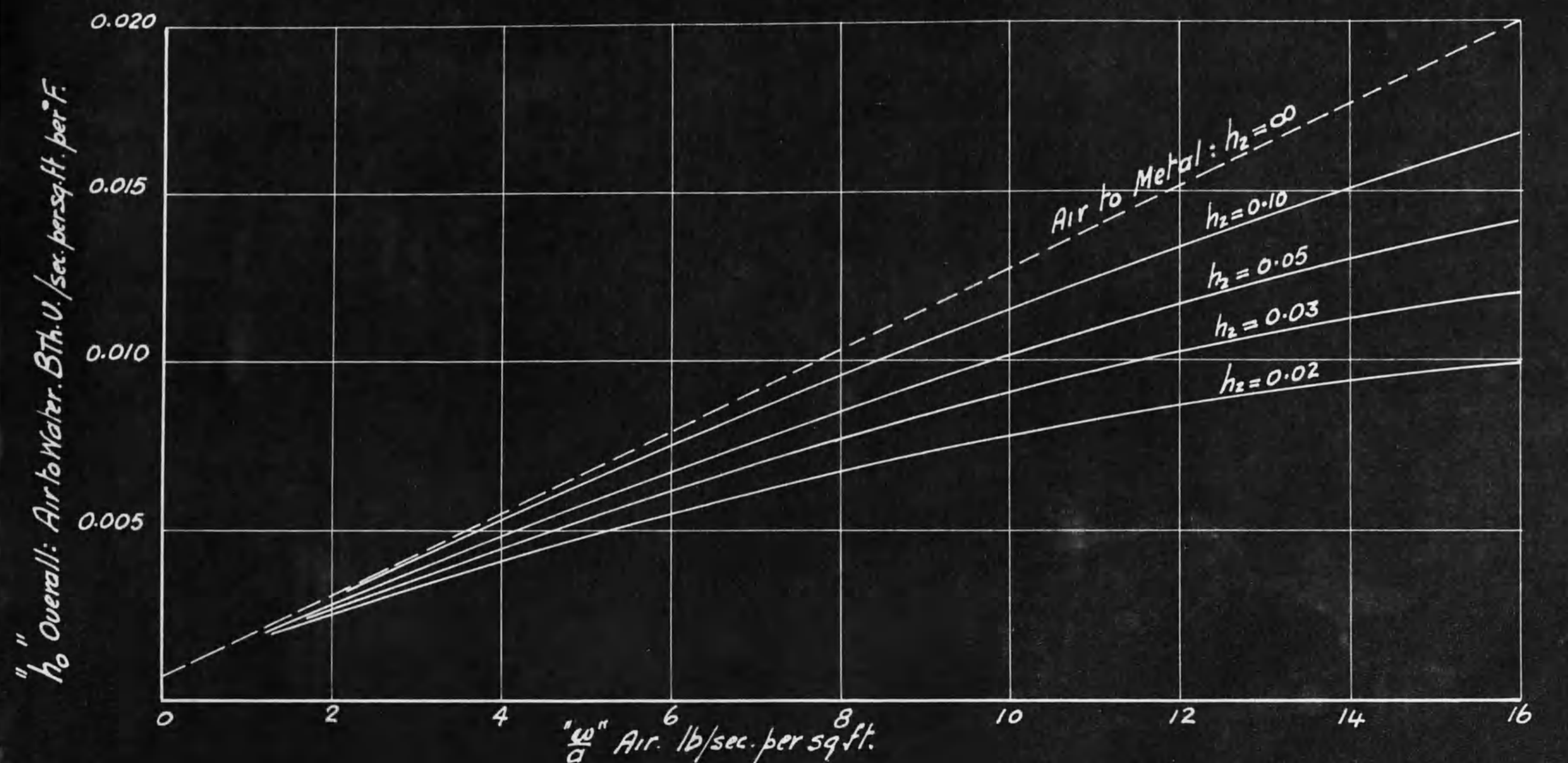

"h_o" Overall: Air to Water. BTh.U./sec. per sq.ft. per °F.
0.020
0.015
0.010
0.005
0
2
4
6
8
10
12
14
16
"$\frac{w}{a}$" Air. lb/sec. per sq ft.
Air to Metal : $h_2 = \infty$
$h_2 = 0{\cdot}10$
$h_2 = 0{\cdot}05$
$h_2 = 0{\cdot}03$
$h_2 = 0{\cdot}02$

FIG. 6.

Columns (3) to (7) indicate h_o overall, for h_w (water to metal) as derived from Jordan's curves, these embracing water mass-flow rates 0 to 20 or speeds 0·0 to 0·32 ft/sec.

TABLE D.

$\frac{w}{a}$ Air.	h_a	h_o				
		hw = 0.02	hw = 0.03	hw = 0.04	hw = 0.05	hw = 0.10.
1	0.0019	.00174	.0018	.0018	.018	
2	0.0031	.00268	.0028	.00297	.00292	
4	0.0055	.0043	.0047	.0048	.0050	
6	0.0079	.0057	.0063	.0066	.0068	.0074
8	0.0103	.0068	.0077	.0082	.0084	.0094
10	0.0127	.0078	.0089	.0096	.0101	.0114
12	0.0151	.0086	.0101	.0110	.0116	.0132
14	0.0175	.0094	.0111	.0120	.0130	.0150
16	0.020	.010	.0120	.0133	.014	.017
Column 1	2	3	4	5	6	7

In fig (6) the resulting overall rates have been plotted on a base of air speeds, and it is sufficiently evident that for water speeds below about 0·16 ft/sec ($\frac{w}{a}$ = 10), it is not advisable to discard the effect of the water speed.

R. Royds has carried out a considerable number of experiments on overall heat transmission from air to water, and in his Book:- "Heat Transmission by Radiation, Conduction and Convection" he states, on pages 178 to 179:- "It is probable that in most of these experiments, (air to water), the velocity of the water was below the critical value." An inspection of fig. 79 page 178, (loco cit.), will show that the limits chosen for figure (6) cover most of Royds' experiments.

Now, using the overall h_o values of Royds with h_w values for water at the speeds used by Royds and Campbell in their experiments, described in a paper to the Institution of Engineers and Shipbuilders in Scotland ("The possibilities of flue gas economisers on board ship"), values of h_a for air to metal have been derived and these are shown in fig 7, with Jordan's figures (from Air to brass) and those of the author.

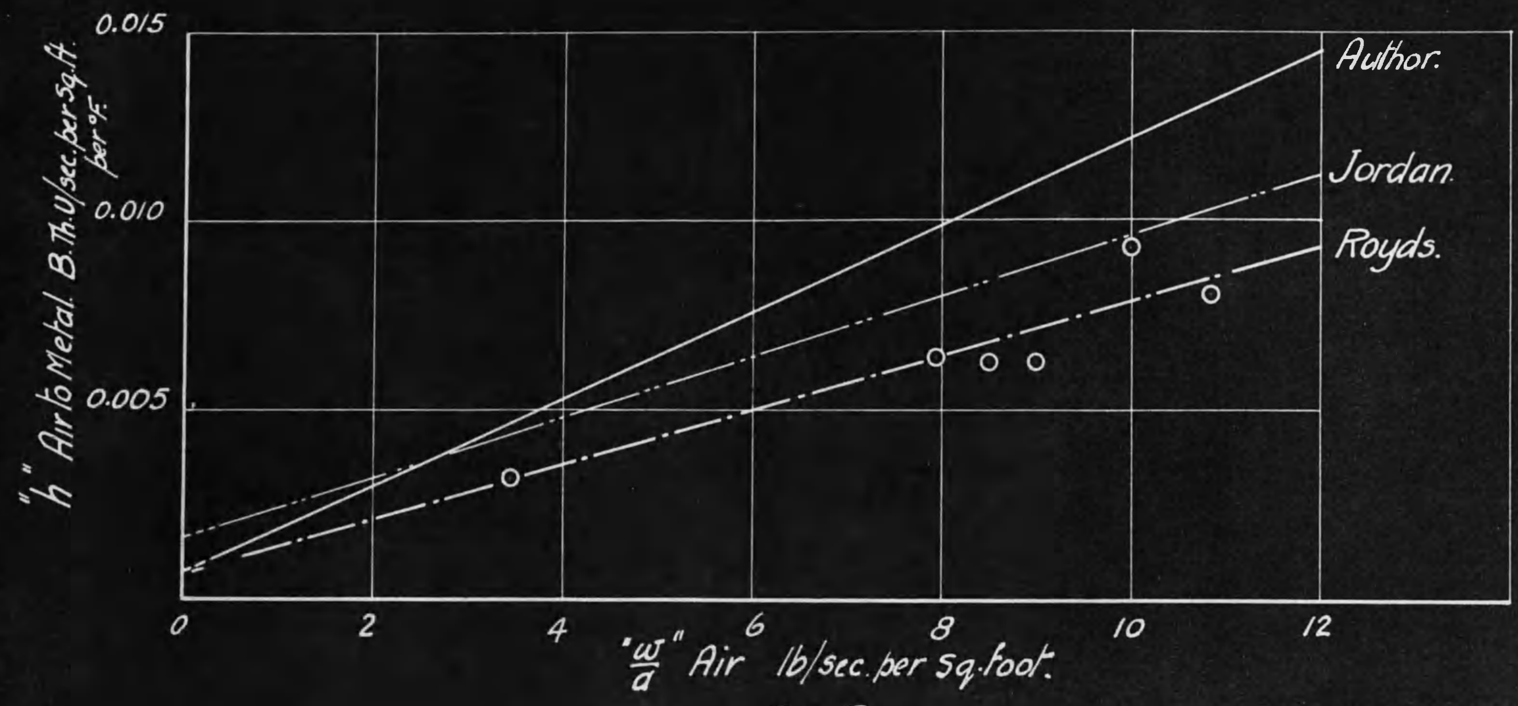

"h" Air to Metal. B.Th.U/sec. per Sq. ft. per °F.
0.015
0.010
0.005
0
2
4
6
8
10
12
Author.
Jordan.
Royds.
"w/a" Air lb/sec. per Sq. foot.

FIG. 7.

APPLICATION TO INDUSTRIAL APPARATUS INVOLVING THE TRANSMISSION OF HEAT FROM AIR TO METAL.

The most important of these may be classed as follows:-

1. (a) Boilers, (b) Economisers, (c) Superheaters and Resuperheaters.
2. Exhaust gas boilers.
3. Air preheaters for (a) boiler plants, (b) muffles.
4. Air coolers for air compressors.

1. (a) and (b). These have been dealt with by Nicolson and others.

   (c) These will be considered in the second part of this Thesis, in connection with superheated steam.

2. Exhaust gas boilers or water heaters, and 4, air coolers for air compressors, of the tubular type.

Only those having the hot air or gases inside the tubes and the water on the outside will be considered. The tubes used may be of the following types:-

Plain parallel tubes, tapered tubes, Serve tubes either parallel or tapered, parallel or tapered tubes fitted with Retarders.

First it is necessary to ascertain how the overall rate of heat transmission varies with the air speed for the plain parallel tubes, Serve tubes, and tubes fitted with retarders. It is assumed of course that the water speed will be above the limit mentioned previously.

Fortunately, curves are available to meet these cases, and they indicate that the overall rate, h_o, can be expressed as a function of $\frac{W}{a}$, such as: $h_o = k(\frac{W}{a})^n$.

Taking the most general case of the tapered tube of which the parallel is only a special case, the following development will be found useful.

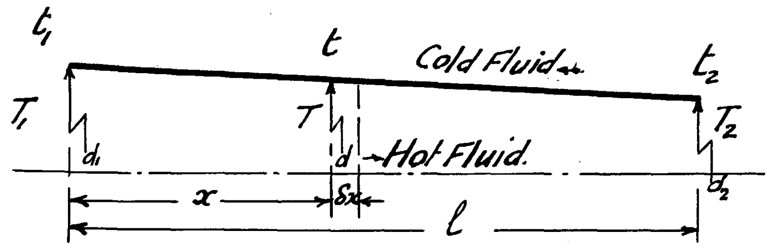


FIG. 8

Consider a short length δx of the tube after a section distant x from the origin taken at entrance of the hot fluid to the tube, i.e., at the larger end, where T, θ, t, are the temperatures of the hot gas, metal and water respectively. °F.

From the geometry of the tube we have $\frac{d_1 - d}{x} = \frac{d_1 - d_2}{l}$

$\frac{d_1 - d_2}{l}$ is the rate of taper of the tube, let this be denoted by m, then $d = d_1 - mx$.

Let w_1 = the rate of gas flow in lb/sec.

S_1 = specific heat at C.p. for the gas, in B.Th.U., (lb., °F).

let δT = fall in temperature of gas while flowing over the elementary surface area $\pi d.\delta x$.sq.ft.

Since the cross sectional area varies continuously along the length of the tube, let h be the value of the rate of heat transmission at the considered section, so that

$$h.\ \pi d.\delta x.(T - t) = -w_1 S_1.\delta T; \quad \ldots \quad (1)$$

$$\text{again } w_1 S_1 (T_1 - T) = w_2(t_1 - t), \quad \ldots \quad (2)$$

$$\therefore -1 + \frac{t_1 - t}{T_1 - T} = \frac{w_1 S_1}{w_2} - 1,$$

$$\frac{-T_1 + T + t_1 - t}{T_1 - T} = \frac{w_1 S_1 - w_2}{w_2} = r$$

$$(T - t) - (T_1 - t_1) = \frac{(w_1 S_1 - w_2)}{w_2}(T_1 - T)$$

$$\therefore (T - t) = r.(T_1 - T) + (T_1 - t_1)$$

$$\therefore h.\ \pi d.\ \delta x \ .\ \frac{1}{w_1 S_1} = \frac{-\delta T}{r(T_1 - T) + (T_1 - t_1)} \quad \ldots \quad (3)$$

$$\text{or } k\left(\frac{w_1}{a}\right)^n .\pi.(d_1 - mx).\ \delta x \ .\ \frac{1}{w_1 S_1} = \frac{-\delta T}{r(T_1 - T) + (T_1 - t_1)}$$

or in the limit

$$\frac{k}{S_1}\cdot\left(\frac{w_1}{a_1}\right)^{n-1}.\ \pi\frac{(d_1 - mx)}{a^n}\ .\ a_1^{\ n-1}.\ dx = -\frac{dT}{r(T_1 - T) + (T_1 - t_1)}$$

$$\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}.\ \frac{\pi(d_1 - mx)}{\left(\frac{\pi}{4}\right)^n.(d_1 - mx)^{2n}}\ .\ \left(\frac{\pi}{4} d_1^{\ 2}\right)^{n-1} dx = -\frac{dT}{r(T_1 - T) + (T_1 - t_1)}$$

$$\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}.\ \frac{4}{d_1}.\ \left(1 - \frac{mx}{d_1}\right)^{1-2n}.\ dx = \frac{-dT}{r(T_1 - T) + (T_1 - t_1)} \quad \ldots (4)$$

Integrating, we have:-

$$\therefore -\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}\ \frac{4}{d_1}.\ \left(1 - \frac{mx}{d_1}\right)^{2-2n}\ .\ \frac{d_1}{m(2 - 2n)} = \frac{1}{r}\log_e\{r(T_1 - T) + (T_1 - t_1)\} + C \quad \ldots (5)$$

$x = 0 \quad T = T_1,$

$$\therefore -\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}.\ \frac{4}{m\ (2-2n)} = \frac{1}{r}\log_e (T_1 - t_1) + C,$$

$x = l \quad T = T_2$

$$\therefore\ -\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}\frac{4}{m(2-2n)}\left(1-\frac{ml}{d_1}\right)^{2-2n} = \frac{1}{r}\log_e\left\{r(T_1 - T_2)+(T_1 - t_1)\right\} + C,$$

$$\therefore\ \frac{k}{S_1}\cdot\left(\frac{w_1}{a_1}\right)^{n-1}\frac{4}{m(2-2n)}\left\{1-\left(1-\frac{ml}{d_1}\right)^{2-2n}\right\} = \frac{1}{r}\log_e \frac{r(T_1 - T_2)+(T_1 - t_1)}{(T_1 - t_1)}$$

expanding $(1 - \frac{ml}{d_1})^{2-2n}$, to the 2nd power,

$$\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}\frac{4}{m(2-2n)}\left\{(2-2n)\frac{ml}{d_1} - \frac{(2-2n)(1-2n)}{2}\cdot\frac{m^2l^2}{d_1^2}\right\} =$$
$$\frac{1}{r}\log_e \frac{r(T_1 - T_2)+(T_1 - t_1)}{(T_1 - t_1)}.$$

$$\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}\frac{4l}{d_1}\left\{1 + \frac{2n-1}{2}\cdot\frac{d_1-d_2}{d_1}\right\} = \frac{1}{r}\log_e\left\{1+\frac{r(T_1 - T_2)}{(T_1 - t_1)}\right\}. \quad . \quad . \quad .(6)$$

This equation is a perfectly general one, and the particular case of any of the tubes enumerated above can be easily derived; but it is perhaps advisable to simplify this further by expressing r as a function of w, and temperatures of hot and cold fluids.

Thus:-

$$r = \frac{w_1S_1 - w_2}{w_2} = \frac{w_1S_1}{w_2} - 1$$

now $w_2(t_1 - t_2) = w_1S_1(T_1 - T_2)$

$$r = \frac{(t_1 - t_2) - (T_1 - T_2)}{T_1 - T_2}$$

and the right hand side of equation (6) reduces to

$$\frac{T_1 - T_2}{(T_2 - t_2)-(T_1 - t_1)}\cdot\log_e\left\{1 + \frac{(t_1 - t_2)-(T_1 - T_2)}{(T_1 - t_1)}\right\};$$

or $$\frac{T_1 - T_2}{(T_1 - t_1)-(T_2 - t_2)}\ \log_e \frac{T_1 - t_1}{T_2 - t_2}.$$

The equation is then in its final form.

$$\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}\cdot\frac{4l}{d_1}\left(1+\frac{2n-1}{2}\cdot\frac{d_1-d_2}{d_1}\right) = \frac{T_1 - T_2}{(T_1 - t_1)-(T_2 - t_2)}\ \log_e \frac{T_1 - t_1}{T_2 - t_2} \quad . \quad . \quad (7)$$

Should it be a question of an exhaust gas boiler it is possible to use heat transmission rates of air to water since the molecular weight and specific heats of air and average flue gases are about equal; this determines the value of the index n.

Again in that case the temperature of the water side may be taken as constant, when $t_1 = t_2$ and the equation further simplifies to:

$$\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}\cdot\frac{4l}{d_1}\left\{1+\frac{2n-1}{2}\frac{(d_1-d_2)}{d_1}\right\} = \log_e \frac{T_1 - t_1}{T_2 - t_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7a)$$

If parallel tubes are to be used, $d_1 = d_2$

and $$\frac{k}{S_1}\left(\frac{w_1}{a_1}\right)^{n-1}\frac{4l}{d_1} = \log_e\ \frac{T_1 - t_1}{T_2 - t_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7b)$$

The factor $1+\frac{2n-1}{2}(\frac{d_1-d_2}{d_1})$ expresses the effect of the tapering. Unless n is less than $\frac{1}{2}$ this function is greater than 1. As a rule n is greater than $\frac{1}{2}$, therefore the tapering has the effect of reducing the length of tube required for a given initial tube diameter and a given value of "$\frac{w_1}{a_1}$".

Whether or not this reduction is worth very much may be matter for controversy, this, however, does not invalidate in any way the value of equation (7) as a perfectly general solution for the transmission of heat from air or flue gases to water.

Of course the equation alone does not suffice to determine all the necessary sizes for such plants, but it must be used along with the equations:-

$\frac{w}{a}$ - arbitrary value, or value determined from consideration of fan power or friction loss, which, if tubes of suitable diameter be chosen, will determine the number of tubes.

This part of the problem is found in most text books and is therefore left out of the question here.

....

3. Airpreheaters of the tubular type.

The use of a curve for overall heat transmission rates is out of the question.

The available data is the known relation $h = A + B\frac{w}{a}$ for air to metal or metal to air , which is not affected by the direction of flow.

Consider the case of plain parallel tubes, with the same notation as before, and with the pitch of the tubes denoted by p, we have first, for equal mass-flow rates at any section $\frac{w_1}{a_1} = \frac{w_2}{a_2}$

hence $a_2 = a_1(\frac{w_2}{w_1}) = a_1(1 - \frac{1}{w_1})$

since $w_1$lb. flue gases = 1 lb. fuel + $w_2$lb. cold air,

$\therefore\ 0{\cdot}866p^2 - \frac{\pi}{4}d^2 = \frac{\pi}{4}d^2\ (1 - \frac{1}{w_1})$

$$\therefore\ 0{\cdot}866p^2 = \frac{\pi}{4}d^2(2 - \frac{1}{w_1}) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (8)$$

Also the heat lost by the hot fluid flowing over the element of/

of surface $\delta x \cdot \pi d = h.\pi d.\delta x(T-\theta) = -w_1 S_1 \delta T$

and the heat gained by the cold fluid

$$= h.\pi d.\delta x(\theta - t) = -w_2 S_2 \delta t.$$

$$\therefore \frac{T-\theta}{\theta - t} = 1$$

$$\therefore \frac{T-t}{T-\theta} = 2$$

But, $\dfrac{h.\pi d.\delta x}{2 w_1 S_1} = -\dfrac{\delta T}{T-t}.$

Also, $w_1 S_1 (T_1 - T) = w_2 S_2 (t_1 - t).$

$$-1 + \frac{w_1 S_1}{w_2 S_2} = \frac{t_1 - t}{T_1 - T} - 1 = r$$

$$r = \frac{(T - t) - (T_1 - t_1)}{T_1 - T}$$

$$T - t = r(T_1 - T) + (T_1 - t_1)$$

$$\therefore \frac{h\pi d}{2 w_1 S_1} dx = -\frac{dT}{r(T_1 - T) + (T_1 - t_1)}$$

On Integration,
this reduces to:-

$$\frac{h\pi d}{2 w_1 S_1} \cdot l = \frac{1}{r} \log_e \frac{r(T_1 - T_2) + (T_1 - t_1)}{(T_1 - t_1)}$$

or, since $\dfrac{\pi d^2}{4} = a$,

$$\frac{2h.}{\left(\frac{w_1}{a}\right) S_1} \cdot \frac{l}{d} = \frac{T_1 - T_2}{(T_1 - t_1) - (T_2 - t_2)} \log_e \frac{T_1 - t_1}{T_2 - t_2}$$

substituting for $h = (A + B\frac{w_1}{a})$

$$\frac{2}{S_1}\left(\frac{A}{\frac{w_1}{a}} + B\right) \cdot \frac{l}{d} = \frac{T_1 - T_2}{(T_1 - t_1) - (T_2 - t_2)} \log_e \frac{T_1 - t_1}{T_2 - t_2} \quad . \quad . \quad . \quad (9)$$

This is the fundamental equation for the ratio $\frac{l}{d}$.

. . . .

There are two functions in the above equation (9) which have a very great bearing upon the surface area required and upon the ratio $\frac{l}{d}$; these are T_2, the outlet temperature of the hot gases and $\frac{w_1}{a}$, the mass-flow rate. It is now proposed to indicate the effect of these two factors.

With the same notation as before

$$S_1(1 + w_2)(T_1 - T_2) = S_2 w_2 (t_1 - t_2)$$

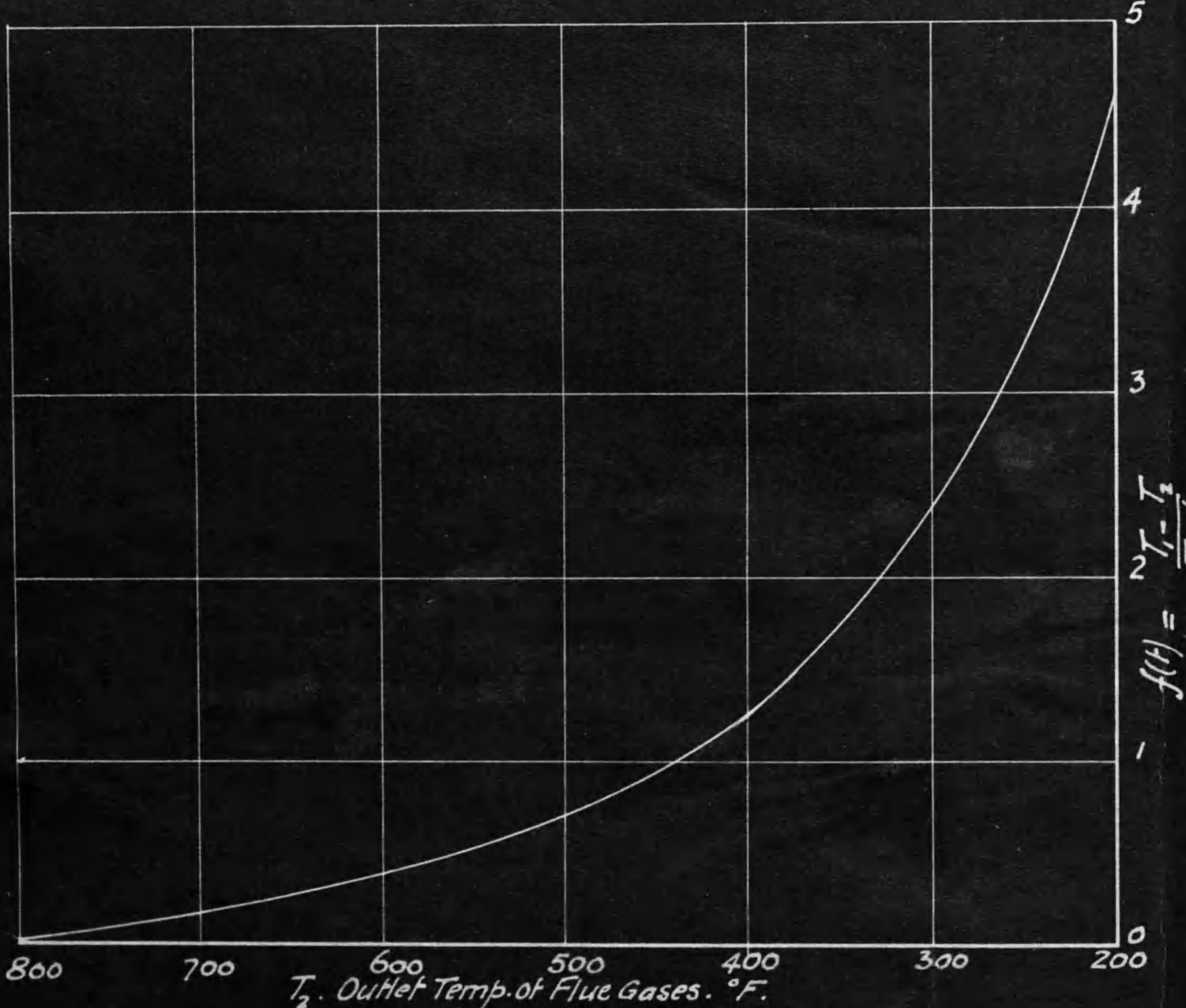
and $S_1 w_1$ is nearly equal to $S_2 w_2$

$$\therefore \quad T_1 - T_2 = t_1 - t_2$$

$$\text{or } T_1 - t_1 = T_2 - T_2 \text{ (nearly)}$$

$$\therefore \quad \text{Mean Temp. difference } t_m = T_1 - t_1 = T_2 - t_2.$$

again/



5
4
3
2
1
0
$f(t) = \frac{T_1 - T_2}{T_2 - t_2}$
800
700
600
500
400
300
200
T_2. Outlet Temp. of Flue Gases. °F.

FIG 9.

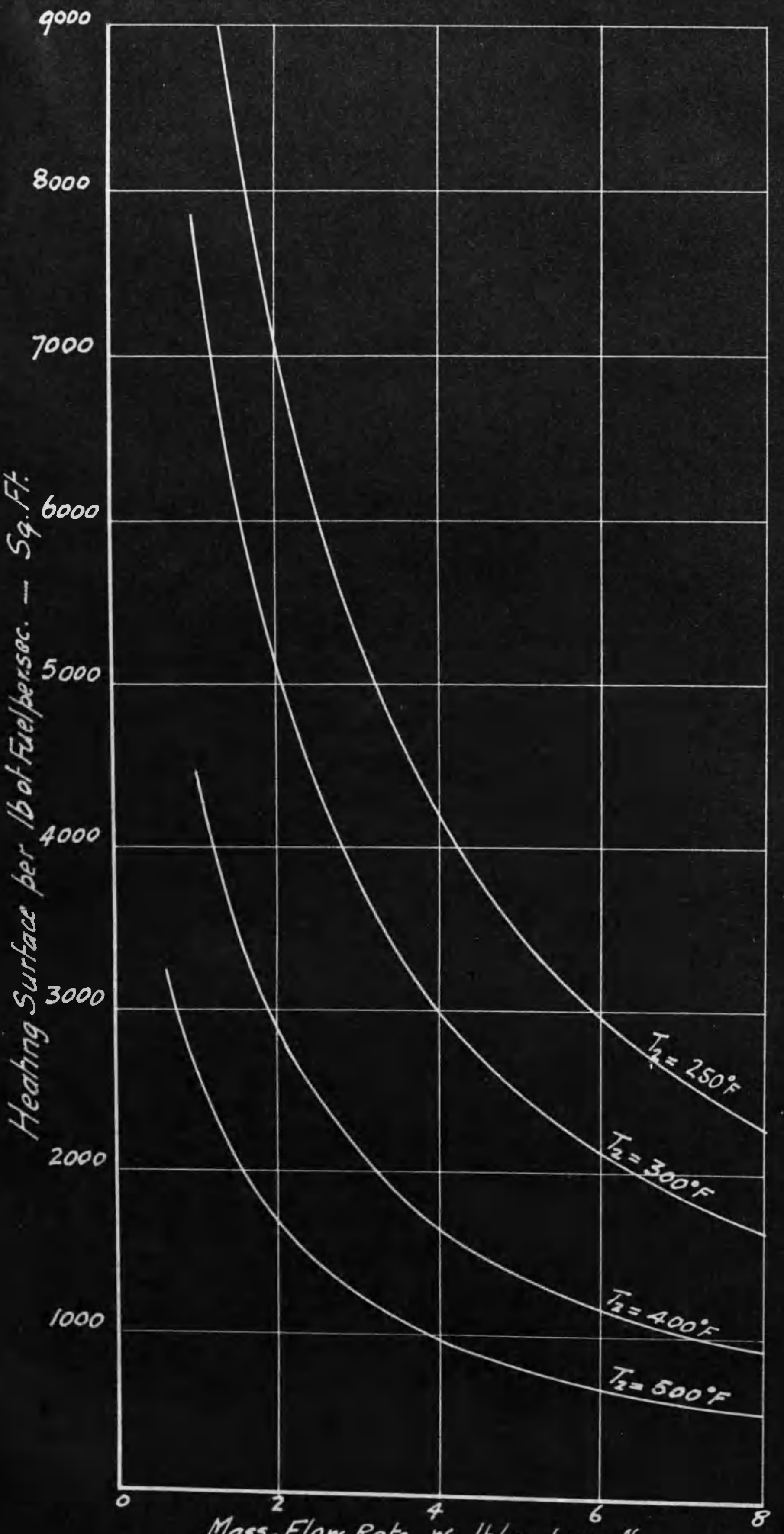

9000
8000
7000
6000
5000
4000
3000
2000
1000
0
2
4
6
8
Heating Surface per lb of Fuel per sec. — Sq. Ft.
Mass-Flow Rate $\frac{W_2}{a_2}$ lb/sec-per sq.ft.
T_2 = 250°F
T_2 = 300°F
T_2 = 400°F
T_2 = 500°F

FIG 10(a).

again:

$$S_1(1 + w_2)(T_1 - T_2) = h_o \cdot (T_2 - t_2)M$$

where h_o = overall rate of heat transmission = $\frac{1}{2}(A + B\frac{w_2}{a_2})$

where M = heating surface in square feet;

$$\therefore \quad M = S_1(1 + w_2) \cdot \frac{1}{\frac{1}{2}(A + B\frac{w_2}{a_2})} \cdot \frac{(T_1 - T_2)}{(T_2 - t_2)}$$

Assuming an average value of 15 lb. of gases per lb. of fuel, and 0·26 for the specific heat, and a fuel comsumption rate of 1 lb/sec., the heating surface M in square feet is given by:-

$$M = \frac{8 \cdot 36}{A + B\frac{w_2}{a_2}} \cdot f(t) \quad \ldots \ldots \ldots \ldots \quad (9)$$

$$\text{where } f(t) = \frac{T_1 - T_2}{T_2 - t_2} \quad \ldots \ldots \ldots \ldots \quad (10)$$

T_1 is usually about 800°F, and t_2 about 70°F. with these two arbitrary values inserted in (10)

$$f(t) = \frac{800 - T_2}{T_2 - 70}.$$

this function has been plotted in fig (9) on a base of T_2. (Outlet temperature of the flue gases).

By taking a series of values for $\frac{w_2}{a_2}$ (from 1 to 8), and reading off fig 16 the corresponding values of $(A + B\frac{w_2}{a_2})$, the heating surface M can be calculated for any particular outlet temperature of flue gases.

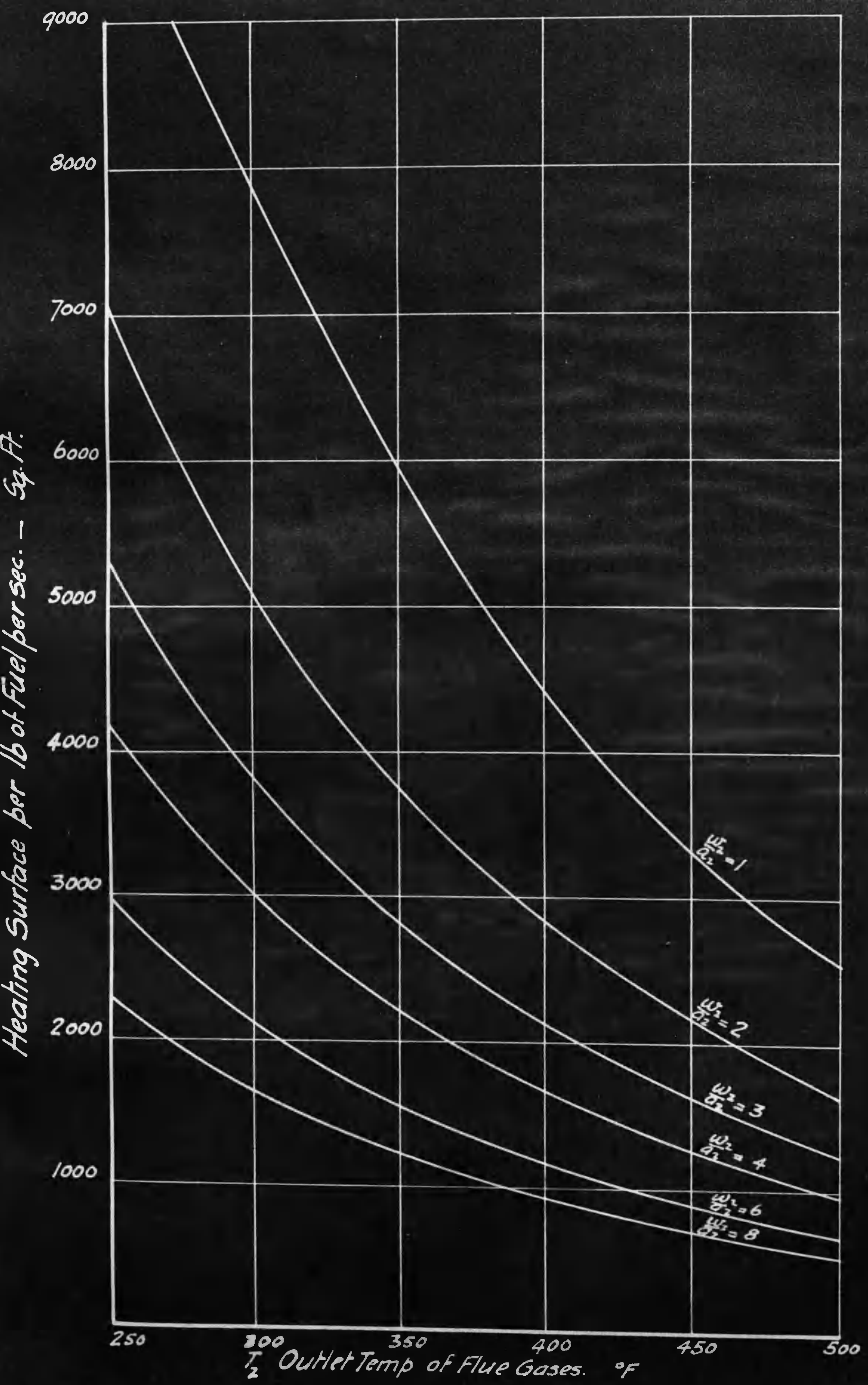
Thus with $\frac{w_2}{a_2} = 4$, $(A + B\frac{w_2}{a_2}) = 0.0061$

for $T_2 = 400$°F, $f(t) = 1{\cdot}212$

$\therefore \quad M = \frac{8.36}{.0061} \times 1{\cdot}212 = \underline{\underline{1660}}$ sq.ft. for each lb. of fuel burnt/sec.

Fig 10(a) shows the effect of the mass-flow rate on the heating surface required for various outlet temperatures of flue gases, ranging from 250°F to 500°F, and fig 10(b) shows more clearly the effect of lowering the outlet temperature of the flue gases.

The importance of the two factors: mass-flow rate and outlet temperature of the flue gases is well illustrated by the curves in figures 10(a) and 10(b).



9000
8000
7000
6000
5000
4000
3000
2000
1000
Heating Surface per lb of Fuel per sec. – Sq. Ft.
250
300
350
400
450
500
T_2 Outlet Temp of Flue Gases. °F
$\frac{\omega_2}{\sigma_2} = 1$
$\frac{\omega_2}{\sigma_2} = 2$
$\frac{\omega_2}{\sigma_2} = 3$
$\frac{\omega_2}{\sigma_2} = 4$
$\frac{\omega_2}{\sigma_2} = 6$
$\frac{\omega_2}{\sigma_2} = 8$

FIG 10(b).

PREHEATING OF AIR OF COMBUSTION
PROBABLE FURNACE TEMPERATURES

In the case of a boiler not fitted with any sort of economizer, the flue gases on leaving the boiler are discharged into the atmosphere, carrying with them and dissipating a quantity of heat, which may be as much as 35 to 40 per cent. of the heat generated by the combustion of the fuel. By introducing, therefore, either a feed water heater or an air heater in the path of the flue gases after their exit from the boiler, a certain proportion of this heat may be recovered, and experience now shows that it is preferable, from several points of view, to make use of an air heater for that purpose.

Let w = the weight in lb. of air used per lb. of fuel, c_p = the specific heat at constant pressure of the flue gases. Let also T_1 = the temperature of the gases leaving the boiler with no preheater and T_a = the temperature of the atmospheric air, then the heat lost will be $(w + 1)c_p(T_1 - T_a)$.

If a preheater is now fitted to the boiler and T_1' = the new temperature of the gases leaving the boiler and entering the preheater and T_2 = the temperature of gases leaving the preheater, the heat lost under these conditions will be $(w + 1)c_p(T_2 - T_a)$.

Therefore, the heat saved = $(w + 1)c_p(T_1 - T_2)$ per lb. of fuel, and not $(w + 1)c_p(T_1' - T_2)$, as might at first sight be supposed.

The immediate effect of the introduction of hot air into the furnace is an increase in the temperature of the furnace. This increase, however, is by no means equal to the rise in the temperature of the air supply, but is a function of it and of the original temperature of the furnace, as the following will show:-

Let θ_1 = temperature of the fuel before combustion(°F.abs.),
S_1 = specific heat of the fuel.(BThU.,lb,°F.)
C = calorific value per lb. of the fuel.(BThU)
θ = temperature of the products after combustion(°F.abs.),
c_p = specific heat of the products at constant pressure.
w = weight of air used per lb. of fuel.
T_a = temperature of the atmosphere (°F.abs.),
S_a - specific heat of air at constant pressure (BThU. lb.°F.)

Total heat supplied to the furnace per lb. of fuel before combustion $= S_1\theta_1 + wS_aT_a + C$ . . . . (1)

Total/

Total heat generated during combustion = heat radiated by the burning fuel + heat in the products of combustion

$$= R + (w + 1)c_p.\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$\therefore S_1\theta_1 + wS_aT_a + C = R + (w + 1)c_p.\theta \quad . \quad . \quad . \quad . \quad (3)$$

If the air is now preheated by an amount $x^oF.$, θ will now be increased by $y^oF.$, say, and the radiation from the bed will be increased from R to R + r.

Hence $S_1\theta_1 + wS_a(T_a + x) + C = R + r + (w + 1)c_p(\theta + y)$.(4)
subtracting (3) from (4),

$$wS_ax = r + (w + 1)c_p.y \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

r is evidently here a function of the furnace temperature, and therefore also a function of y.

Let it be assumed that all the heat radiated by the burning bed of fuel is finally received by the boiler; and, further, that this radiation takes place according to the fourth power law of Stefan and Boltzmann, so that if T is the furnace temperature and T_o that of the boiler, in $^oF.$ abs.,

$$R = k(T^4 - T_o^4) \text{ for the first set of conditions } ..(6)$$

$$R + r = k(\overline{T + y}^4 - T_o^4) \text{ for the second set of conditions } ...(7)$$

Subtracting (6) from (7),

$$r = k(4T^3y + 6T^2y^2 + 4Ty^3) \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

or $r = kT^4 \left(\frac{4y}{T} + \frac{6y^2}{T^2} + \frac{4y^3}{T^3}\right)$

Neglecting the 3rd term $r = kT^4 \left(\frac{4y}{T} + \frac{6y^2}{T^2}\right)$ . . . . . (9)

Substituting in (5) we have

$$wS_ax = kT^4 \left(\frac{4y}{T} + \frac{6y^2}{T^2}\right) + (w + 1)c_py \quad . \quad . \quad . \quad (10)$$

This is a series of parabolas, the vertices of which are all very remote from the origin, and the portion of the curve which is of interest is almost straight—i.e., neglecting the y^2 term

$$\left(\frac{4kT^3}{wS_a} + \frac{w + 1}{w} \cdot \frac{c_p}{S_a}\right) y = x \quad . \quad . \quad . \quad . \quad . \quad (11)$$

M. Rozak, (mémoires de la Société des Ingénieurs Civils de France,1923), from a series of experiments on a boiler plant, has been able to plot the curves represented by equation (10) above, and from these results a suitable value of k for equation (11) is $0{\cdot}135 \times 10^{-9}$.

Again, since c_p and S_a are very nearly equal to one another we can write $\frac{w + 1}{w} \cdot \frac{c_p}{S_a} = 1$ and equation (11) reduces to

$$x = \left(0{\cdot}225 \times 10^{-8} \frac{T^3}{w} + 1\right) y \quad . \quad . \quad . \quad (12)$$

This equation gives the increase in furnace temperature to be expected from a definite amount of preheating, and shows that this increase is smaller the higher the original furnace temperature.

Thus, if the air is preheated by 400°F., the increase in the furnace temperature when T = 2000°F. abs. is 182°F., and when T = 2500°F. abs. it is 120°F., with an air to fuel ratio of 15·0, in both cases.

The increased furnace temperature causes an increased rate of heat transmission per square foot of heating surface, and also results in the temperature of the gases leaving the boiler proper being higher than would be the case without the interposition of the preheater. At the same time the temperature of the gases leaving the preheater is less than that of the gases leaving the same boiler without a preheater.

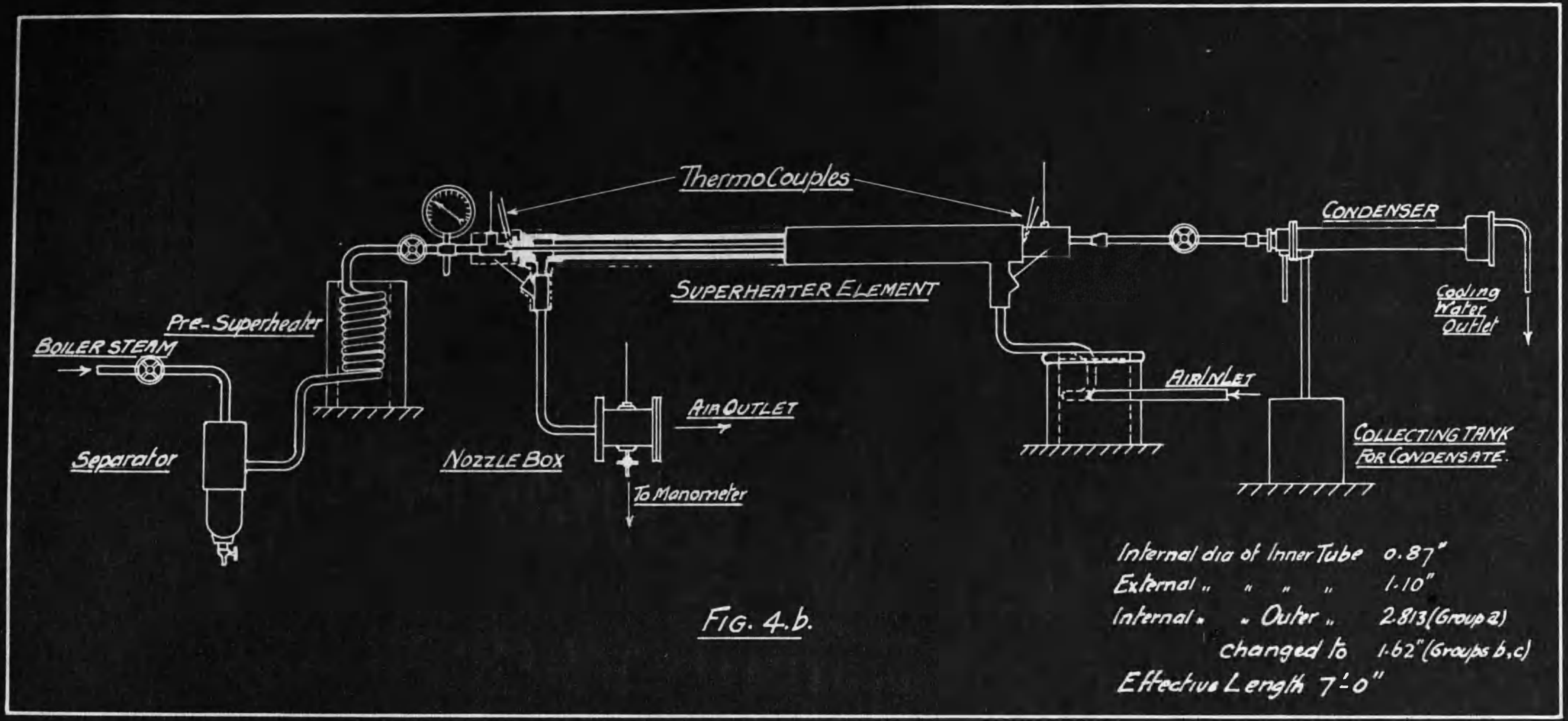

ThermoCouples
CONDENSER
Cooling Water Outlet
SUPERHEATER ELEMENT
Pre-Superheater
BOILER STEAM
AIR INLET
AIR OUTLET
Separator
NOZZLE BOX
To Manometer
COLLECTING TANK FOR CONDENSATE.
Internal dia of Inner Tube 0.87"
External " " " " 1.10"
Internal " " Outer " 2.813 (Group a)
changed to 1.62" (Groups b, c)
Effective Length 7'-0"

FIG. 4.b.

HEAT TRANSMISSION BETWEEN METAL AND SUPERHEATED STEAM.

It has been demonstrated that the relation $\frac{1}{h_0} = \frac{1}{h_1} + \frac{1}{h_2}$ is an important one when investigating problems in heat transmission; it has also been shown how this equation suggests as line of research the determination of values of h for any fluid which may receive the attention of the experimenter.

A choice is made of a suitable first fluid, of which the variations of h_1 with $\frac{w_1}{a_1}$ have been already established (such as air); the second fluid can be either heated or cooled by the first, and overall values of h_0 are obtained experimentally. It is then a simple matter to deduce h_2 and plot these values on a base of $\frac{w_2}{a_2}$.

In this section of the work air was the first fluid, and superheated steam the second.

In order to render the results more conclusive, the experiments were grouped as follows:-

(a) Experiments giving directly values of h_2.

(b) Experiments giving directly values of h_2, h_1 and h_0.

(c) Experiments giving indirectly values of h_2, from those of h_0 and h_1.

These experiments were all carried out on the same apparatus, with some modifications for groups b and c. The apparatus Fig. 4b was an adaptation of that used in the experiments with air, described in the first part of this thesis. Saturated steam from the laboratory distributor was led into a separator which was allowed to drain continuously with a small excess steam flowing, then through a gas-heated superheater coil where it could be further dried and superheated if necessary. The dried steam now entered the superheater element proper through a regulator valve. This element consisted of one straight length of steel tubing. The superheated steam left the element through a second regulator/

regulator valve and was finally condensed in a small high speed surface condenser which was capable of dealing with the largest quantity of steam used while securing ample cooling of the condensate.

The condensate was collected and weighed.

The circuit for the hot air was sensibly the same as in the air to air experiments. The hot air, after flowing through the annular space surrounding the inner superheater element was discharged through a "Nozzle-Box". The pressure and temperature of the air in the nozzle-box were taken to enable the air flow to be calculated.

Thermocouples placed at the ends of the superheater element gave the metal temperatures required for the experiments of groups (a) and (b).

The whole apparatus was carefully lagged to reduce radiation losses.

The chief precaution observed in these experiments was to increase the duration of each test, in order to ensure a reasonable quantity of steam to be collected.

From the outset it was recognised that difficulties would be encountered from radiation effects, and the first experiments confirmed this expectation. The hot air temperatures were too high and a reasonable heat balance could not be struck. Calculations indicated however that the values of h_2, from superheated steam to metal, were about double those of h_1(air-metal) for equal mass-flow rates ($\frac{W}{a}$). This was of course quite regular, from Reynolds' Law, as the specific heat of steam at constant pressure is approximately twice that of air.

It was therefore decided to continue with the experiments and deal with the steam side only. Table E, in the appendix, shows four tests which come under group (a). The discrepancy in the heat balance is indicated.

Table F, also in the appendix, shows the average readings for the remainder of the tests carried out under group (a). In series B and C, the pressure of the steam was that of the/

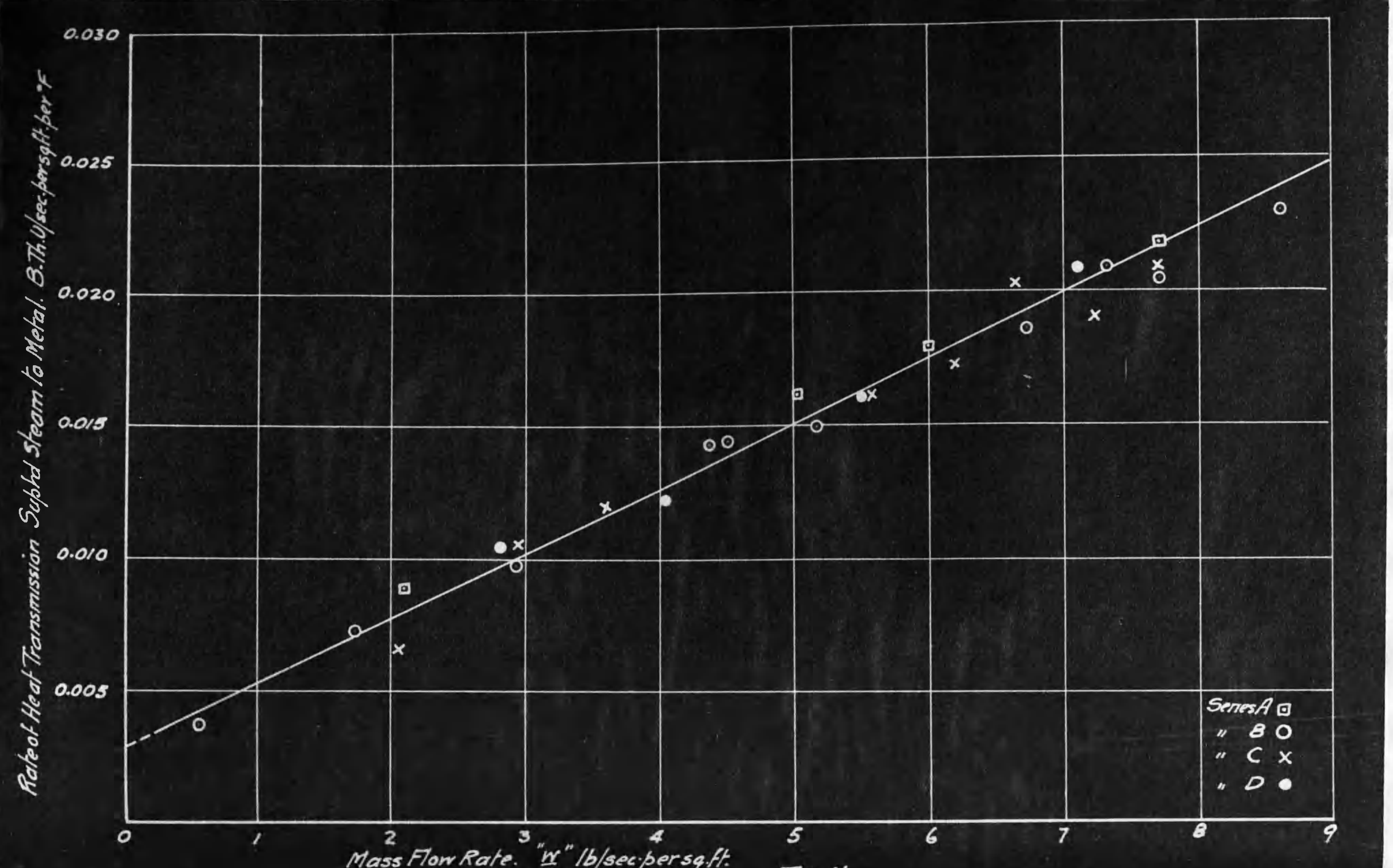

Rate of Heat Transmission Suphd Steam to Metal. B.Th.U/sec.per sq.ft.per °F
0.030
0.025
0.020
0.015
0.010
0.005
0
1
2
3
4
5
6
7
8
9
Mass Flow Rate. "W/a" lb/sec.per sq.ft.
Series A ⊡
" B O
" C X
" D ●

FIG 11

the atmosphere while the initial temperature varied from an average of 238°F in series B to 245°F. in series C. In the next series D, the steam pressure was increased to 24 lb/sq.in.abs., and the initial temperature was also increased to 250°F. The rates of heat transmission from metal to superheated steam, as deduced from these experiments, have been plotted on a base of mass-flow rate and fig. 11 is the result.

The plots average up to a straight line, in agreement with the fundamental law of Osborne Reynolds': $h = A + B\frac{W}{a}$.

It was now essential to confirm the figures obtained from these experiments by more tests with closer heat balances. Accordingly, the tests in group b were carried out, with a radical modification in the apparatus: the outer pipe was removed, and a smaller one, 1·62" internal diameter, was substituted, giving a 79% reduction in the annular area, with a possibility of higher air speeds. In view of the high air temperatures registered in the nozzle-box, an air cooler was interposed between the air discharge from the superheater element and the nozzle-box.

As there seems to be no reason to suppose that the rate of heat transmission from fluid to metal should be different from the rate of heat transmission from metal to that fluid, the steam supply was highly superheated before entering the superheater element, while cold air was admitted to the outer annular space: thus converting the superheater into a desuperheater or air heater. This change successfully fulfilled what had been anticipated: a reasonable heat balance was obtained throughout the tests.

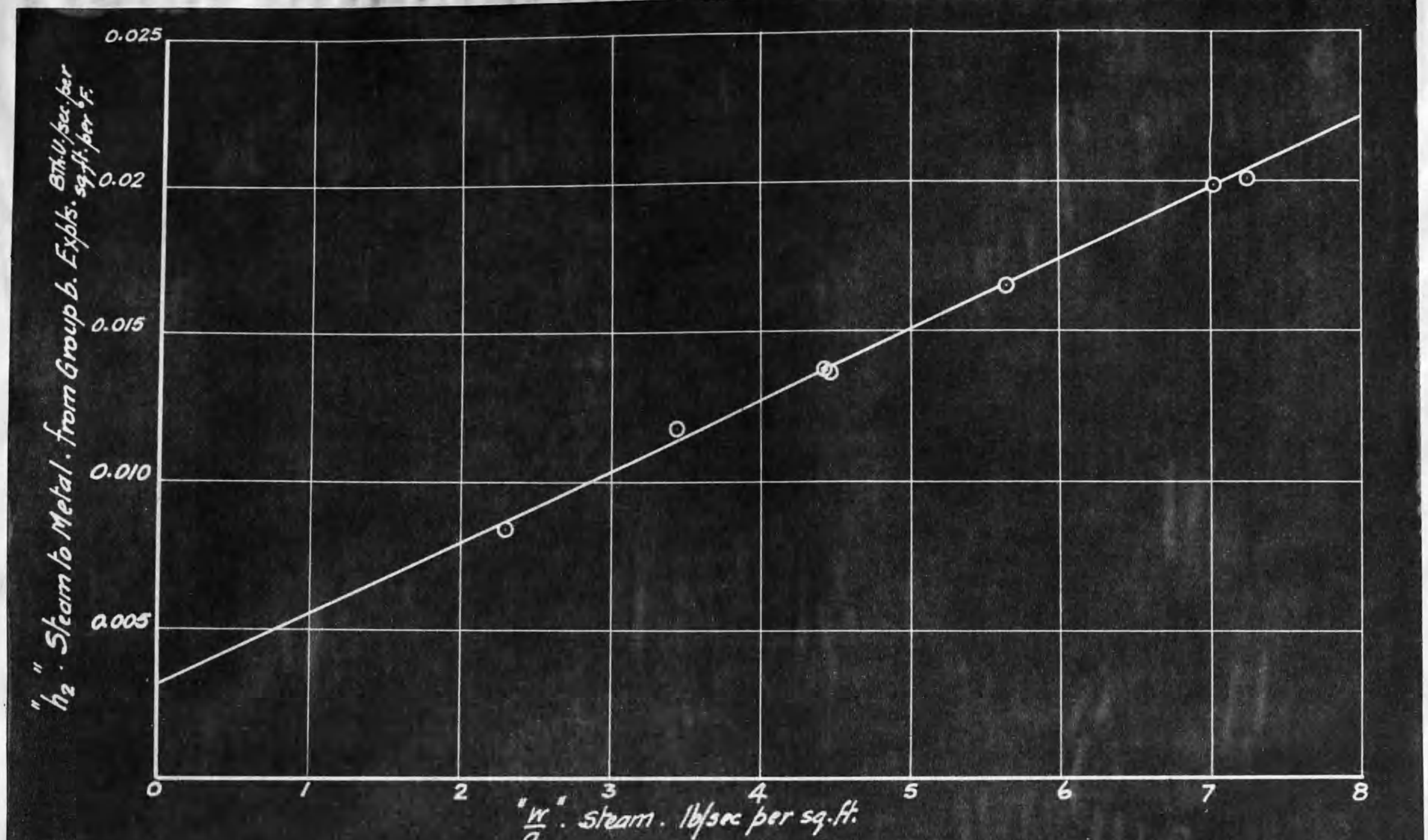
The average readings for this group are given in Table G, in the appendix..

The rate of heat transmission from steam to metal was calculated, as in group (a), directly from the equations :-

$$c_2 \ell = \log_e \frac{T_1 - \theta_1 - \frac{m}{c_2}}{T_2 - \theta_2 - \frac{m}{c_2}} .$$

$$\text{and } h_2 = \frac{c_2 S_2 w_2}{\pi d_2} .$$

as derived for the air to air experiments.



0.025
0.02
0.015
0.010
0.005
"h2" : Steam to Metal . from Group b. Expts. B.Th.U./sec. per sq.ft. per °F.
0
1
2
3
4
5
6
7
8
"w/a" : steam . lb/sec per sq.ft.

FIG 12.

These rates of heat transmission (denoted by h_2) are plotted in fig.12 on a base $\frac{"w"}{a}$. These points confirm the average line obtained from the experiments in group (a), and show that there is no substantial difference between the rates of heat transmission from fluid to metal and from metal to fluid.

The readings taken in this series enabled the coefficients to be determined for the air side. These were calculated and plotted in fig.13. The points are shown relative to the average air-metal line obtained previously. While they are on the whole below the mean line, they show good agreement with the metal to air rates of fig.5.

Lastly, it was desired to test the relation $\frac{1}{h_0} = \frac{1}{h_1} + \frac{1}{h_2}$. Hence, h_0 was calculated, first from the individual rates (after correcting h_1 and h_2 to a mean surface area), and second in the usual way, i.e., heat transmitted per square foot per second, divided by the overall mean temperature difference. The two values are plotted, one against the other, in fig.14.

A third group of 4 tests was undertaken, in which the metal temperatures were excluded in order that the rate of heat transmission from steam to metal might be calculated from the overall rate and from the air rate as taken from the mean line in fig.13.

The readings are recorded in table H, in the appendix.

The results of this group are plotted in fig.13 relative to the mean steam to metal line previously derived from the experiments in groups (a) and (b). Agreement between the two methods, i.e., the direct and the indirect, is now well established.

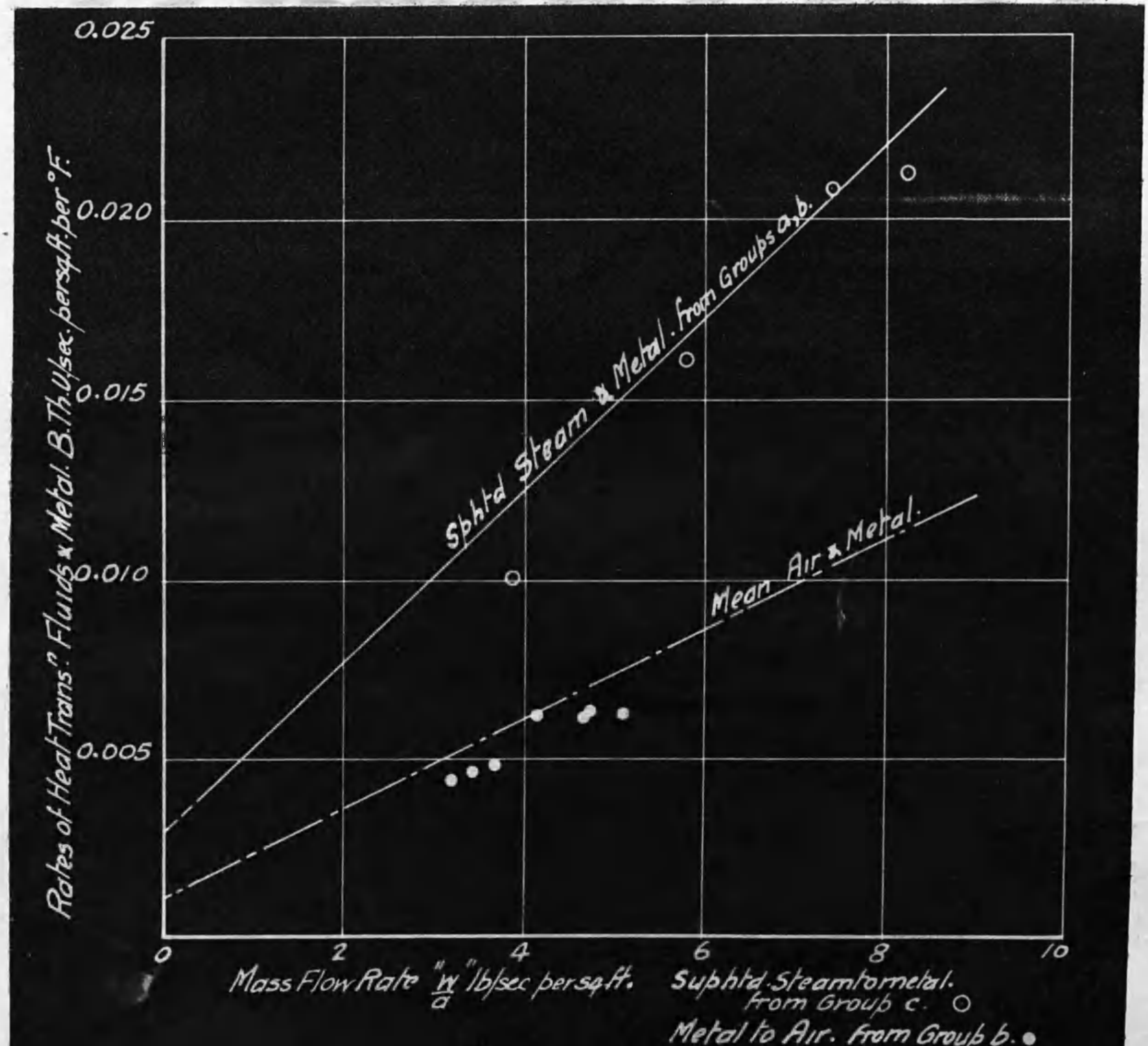

Rates of Heat Trans.ⁿ Fluids & Metal. B.Th.U/sec. per sq.ft. per °F.
0.025
0.020
0.015
0.010
0.005
0
2
4
6
8
10
Sphtd Steam & Metal. from Groups a, b.
Mean Air & Metal.
Mass Flow Rate "w/a" lb/sec per sq.ft.
Suphtd. Steam to metal. from Group c.
Metal to Air. from Group b.

FIG. 13.

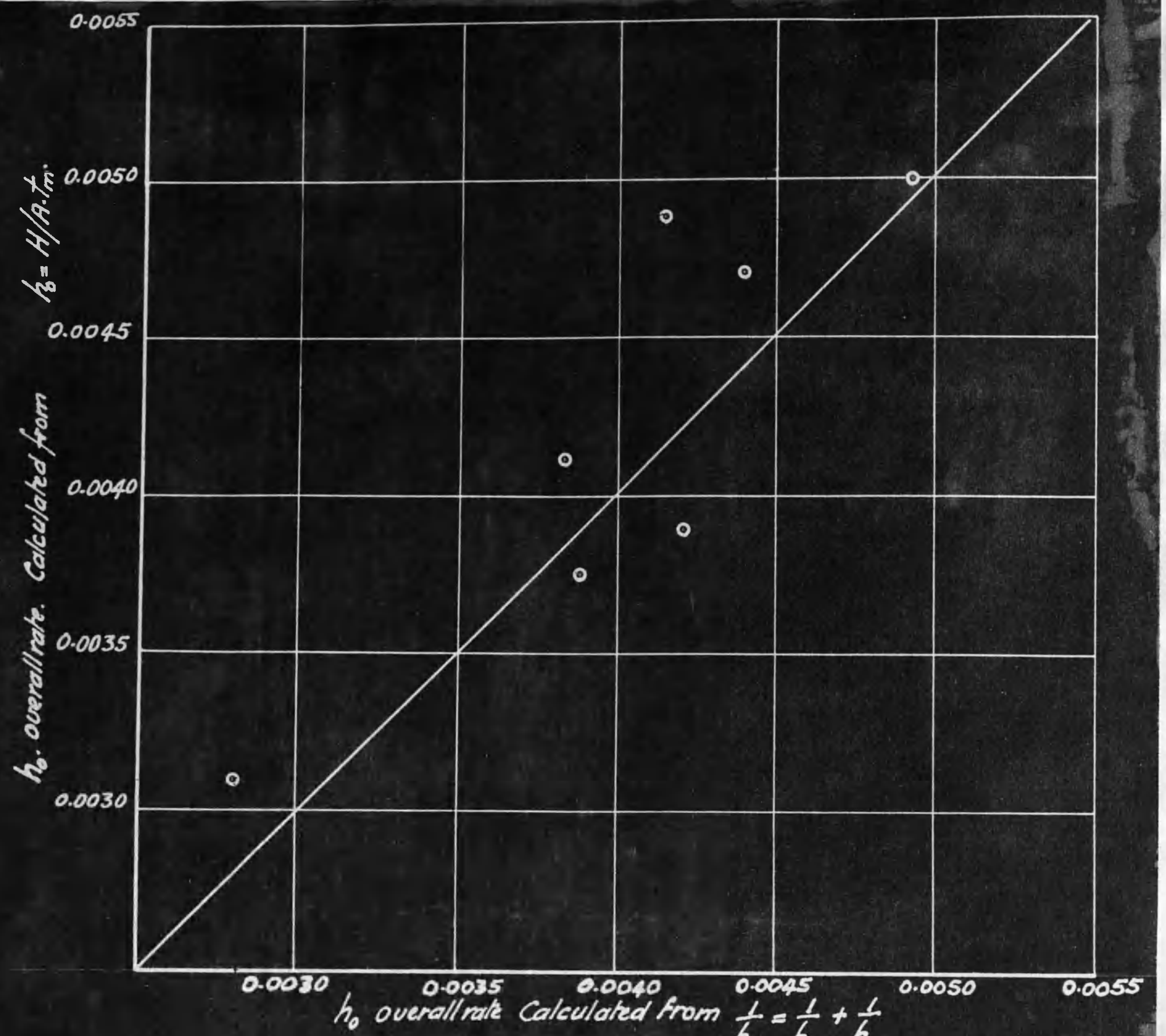

0.0055
0.0050
0.0045
0.0040
0.0035
0.0030
h_0. overall rate. Calculated from $h_0 = H/A \cdot t_m$.
0.0030
0.0035
0.0040
0.0045
0.0050
0.0055
h_0 overall rate Calculated from $\frac{1}{h_0} = \frac{1}{h_2} + \frac{1}{h_1}$

FIG 14

APPLICATION TO INDUSTRIAL APPARATUS INVOLVING THE TRANSMISSION OF HEAT FROM SUPERHEATED STEAM TO METAL.

DESUPERHEATERS OR LAST-STAGE FEED WATER HEATERS UTILIZING SUPERHEATED STEAM.

In small marine turbine plants using steam with a fairly high superheat and fitted with auxiliaries driven by reciprocating engines, the steam supply to these auxiliary engines has to be "desuperheated", as these are not usually suitably designed to utilize superheated steam. In such cases either sea water or the feed water itself may be used as cooling agent. In either case the speed of the cooling liquid will certainly not be the factor controlling the overall rate of heat transmission.

A comparison between the highest value of h_s, for steam, which is likely to be adopted, 0·01, ($\frac{W}{a} = 3$) and the lowest probable value of h_w, for water, 0·2, ($\frac{W}{a} = 80$), will show that the overall rate is likely to be small, 0·00955, and therefore practically dependent on the rate of flow of the superheated steam. This low value of the overall rate of heat transmission requires to be compensated by relatively large cooling surfaces.

For example: to desuperheat 1 lb. of steam per second, at 250 lb/sq.in.abs., 120°F superheat (specific heat 0·586), would require $\frac{0{\cdot}586 \times 120}{0{\cdot}00955} = 7360$ square feet of surface, with 1°F. mean temperature difference. With the saturation temperature of the steam at 401·2°F. and an average water temperature of 80°F, the mean temperature difference would be approximately 380°F and hence the surface required would be about 19·3 sq.feet per lb. of steam.

On the other hand if the cooling agent was the feed water itself, the mean temperature difference would be smaller and the cooling surface would have to be extended in proportion.

Supposing that the feed water has already been heated to a temperature of 250°F, and that it is desired to heat it up further to the saturation temperature of the steam in the/

the boiler, and that this is done by superheated steam extracted from the main steam pipe, then the surface area required would depend upon whether the latent heat of the steam is available or not.

Case (a): The latent heat is available.

Quantity of steam required/lb. feed $= \frac{h_1 - h_2}{H - h_1}$ lb. $= 0\cdot175$ lb.

where H = total heat/lb. of superheated steam at the conditions already specified: 250 lb/sq.in.abs. 120°F = 1280 B.Th.U.

h_1 = liquid heat/lb. water at saturation temp. = 376·3 B.Th.U.

h_2 = liquid heat/lb. water at 250°F = 218·5 B.Th.U.

If the heater be regarded as made up of two heaters in series, the first becomes a desuperheater, and the second a surface condenser working at 250 lb/sq.in.abs.

The value to be assigned to h_o for the desuperheater part will be that corresponding to a mass-flow ($\frac{W}{a}$) of 1·5, giving an overall rate 0·006 B.Th.U/sec. per sq.ft. per °F.

The author had an opportunity to examine the performance of a triple expansion pumping engine, in which reheating of the cylinder steam was accomplished by superheated steam extracted from the pipe supplying the engine. The quantity of superheated steam actually used on test and the dimensions of the reheaters allowed the probable mass-flow rate for the heating steam to be calculated: This figure was less than 1. It is therefore admissible that 1·5 is about as high a mass-flow as would be allowed in this instance.

Superheat lost by steam = 0·586 X 120 X 0·175 = 12·3 B.Th.U/lb. feed.

Latent heat lost by steam = 833 X 0·175 = 145·5 B.Th.U/lb.feed.

∴ temperature rise of feed due to condensation of steam = 139·8°F nearly.

& temperature rise due to loss of superheat of steam = 11·4°F nearly.

∴ on the steam side of the desuperheating surface the mean steam temperature will be about $\frac{401 + 521}{2} = 461$°F

on/

on the water side the mean water temperature will be about $250 + 140 + 5 = 395^{\circ}F$: Thus giving an average temperature difference of $461 - 395 = 66^{\circ}F$.

$\therefore$ Cooling surface required for desuperheating

$$= \frac{12.3}{\cdot006 \times 66} = 31.1 \text{ sq.feet for a flow rate of 1 lb. of feed/sec.}$$

In the condenser part, an average value for the rate of heat transmission from steam to metal is about 1·0 (Webster), and about 0.4 for water at a speed of about 2.5 feet/sec.

$\therefore$ Overall rate of heat transmission

$$= \frac{1}{\frac{1}{0.4} + \frac{1}{1}} = 0.286 \text{ B.Th.U./sec. per sq.ft. per } ^{\circ}F.$$

The steam temperature on the steam side is $401.2^{\circ}F$; the average water temperature on the water side is about $250 + \frac{140}{2} = 320^{\circ}F$.

$\therefore$ Mean temperature difference $= 401.2 - 320 = 81{\cdot}2$

$\therefore$ Cooling surface required for condensation

$$= \frac{145{\cdot}5}{0.286 \times 81.2} = 6{\cdot}28 \text{ sq.feet for a flow rate of 1 lb. of feed/sec.}$$

$\therefore$ Total cooling surface required for a flow of 1 lb. of feed/sec. = 37·38 sq.feet, of which about 83% is necessary to cause the steam to lose its superheat, before it will condense on the remaining 17% of the cooling surface.

Case (b): The latent heat of the steam is not available.

This reduces of course to the desuperheater problem, with a mean temperature difference of about $136^{\circ}F.$, $\left(\frac{401 + 521}{2}\right) - \left(\frac{250 + 401}{2}\right)$.

Even allowing for a higher mass-flow rate for the superheated steam and a correspondingly increased rate of heat transmission of 0·01 B.Th.U/sec. per sq.ft. per $^{\circ}F.$, the cooling surface necessary works out at $\frac{157{\cdot}8}{\cdot01 \times 136} = 116$ sq. feet for a flow rate of 1 lb. of feed/sec.

These calculations have been somewhat elaborated in order to show the difficulties with which the designer has to contend, since both these types of feed water heaters have been proposed for modern steam plants.

STEAM SUPERHEATERS.

Probably the most important application of heat transmission factors for superheated steam is in the design of superheaters and reheaters, which are identical so far as their duties are concerned. The possible differences in these two are to be found in the arrangement of the tubes and in the path given to the heating gases.

On one side of these tubes, the hot gas flows at such rates which are capable of influencing the overall rate of heat transmission to a large extent, on the other side, there flows a colder gas at rates which are almost equally capable of influencing the overall rate of heat transmission.

Indeed, considering that 1 lb. of coal requires an average of about 15 lb. of air for its combustion, and generates about 7 lb. of steam, the relative weight of gases to superheated steam is about 2 to 1. Again, from Fig.13, it appears that the rate of heat transmission from metal to superheated steam is approximately twice that from air to metal for equal mass flow rates. It follows therefore that with equal sectional areas of flow for the gases and for the superheated steam, the individual rates of heat transmission will be about equal and the average temperature of the metal will be midway between that of the gases and superheated steam.

For instance, if the average temperature of the gases is 900°F., that of the steam 450°F., the mean metal temperature will be about 675°F.

Should a higher steam temperature be desired than is implied by the 450°F. above, then the steam speed will have to be increased in order to reduce the temperature difference on the steam side, and retain the temperature of the metal within the safe limits required by the elastic properties of the material. For example Dr. A.L. Mellanby and Dr. W. Kerr, in a paper, "The use and economy of high pressure steam plants", suggest as a limiting steam temperature about 850°F. at a pressure of 1000 lb/sq.in.abs. (saturation temperature 547°F.)

This/

This would mean an average steam temperature of about 690°F., hence the steam speed would have to be rather high in order to bring down the mean metal temperature, and this speed would have to be maintained especially at the lighter loads when the steam flow is likely to decrease. Such high speeds may have serious effects on the pressure drops through the tubes of the superheater.

This safeguarding of the metal from overheating at all loads has been achieved in the Löffler high pressure boiler in a very positive manner, by what is known as the "Steam Pumping Process". In this process, steam is produced in an evaporator or vaporizer to start with, this steam is then highly superheated in special superheater tubes. Part of the superheated steam is led to the engines and part of it is returned to the vaporizer to evaporate the requisite quantity of steam for the cycle of operations. A pump is used between the vaporizer and the superheater coils to maintain a positive circulation. When starting up, the superheating is carefully controlled by reduced firing, but once started, the control of the superheating is effected by varying the speed of the pump so as to maintain a very high mass-flow rate in the coils at all loads.

The normal working pressure of this boiler is about 1400 lb/sq.in.abs., and the mean working steam temperature 880°F.

The average speed of the steam in the superheater coils is about 65 ft/sec.

Under these conditions hard open-hearth steel tubes were found to be quite satisfactory.

A superheater design for a locomotive boiler will now be considered.

As a general proposition a 4-flow element type will be assumed, usually known as a "Schmidt Superheater", and fitted in the top three or four rows of smoke tubes of a locomotive boiler.

Let A and a be the cross sectional areas of flow in one large flue tube containing the superheating elements and in one/

one of the smaller flue tubes respectively, sq.ft;

Let P and p be the corresponding perimeters, ft;

P_W and P_S the perimeters of the metal in contact with water and steam respectively in the large tubes, such that $P = P_W + P_S$, ft;

W and w the weightsof gas flow per tube per hour, for large and small tubes respectively, lb;

N the number of large flue tubes and n the number of smallflue tubes;

G the total weight of flue gases, lb/hour;

S the lb. of steam evaporated per lb. of coal.

In view of the fact that the members constituting the one superheater element are usually of unequal lengths and do not extend for the whole length of the flue tube, the following will be assumed to meet the case with sufficient accuracy, for practical design.

(a) the length of the element is the mean of the individual lengths;

(b) the temperature drop from furnace end to smoke box end is the same for large and small tubes;

(c) the pressure drop from furnace to smoke box is the same for large and small tubes;

(d) the heat absorbed by the water surrounding the tube is proportional to the metal surfaces in contact with the water.

Assumption (c) leads to the key equation

$$\left(\frac{W}{A}\right)^2 \cdot \frac{P}{A} = \left(\frac{w}{a}\right)^2 \cdot \frac{p}{a}$$

$$\therefore \quad \frac{W}{w} = \sqrt{\frac{p}{P} \cdot \left(\frac{A}{a}\right)^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Again $NW + nw = G$, $\therefore$ from (1) $NW = \dfrac{G}{1 + \frac{n}{N}\sqrt{\frac{P}{p} \cdot \left(\frac{a}{A}\right)^3}}$

$$= \frac{G}{1 + \frac{n}{N} \cdot \left(\frac{P}{p}\right)^{\frac{1}{2}} \cdot \left(\frac{a}{A}\right)^{\frac{3}{2}}}$$

$$W = \frac{G}{N + n\left(\frac{P}{p}\right)^{\frac{1}{2}} \cdot \left(\frac{a}{A}\right)^{\frac{3}{2}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

If H and h are heat lost by the flue gases per hour in large and small tubes respectively, B.Th.U;

$$\frac{H}{h} = \frac{NW}{nw} = \frac{N}{n} \cdot \left(\frac{p}{P}\right)^{\frac{1}{2}} \cdot \left(\frac{A}{a}\right)^{\frac{3}{2}};$$

if H_w = heat to water from large tubes, per hour, B.Th.U;

$$\frac{H_w}{h} = \frac{NP_w}{np} \text{ from assumption (d)}$$

hence $$\frac{H}{H_w} = \left(\frac{p}{P}\right)^{\frac{1}{2}} \cdot \left(\frac{A}{a}\right)^{\frac{3}{2}} \cdot \frac{p}{P_w}$$

$$\frac{H}{H_w} = \left(\frac{p}{P_w}\right)^{\frac{3}{2}} \left(\frac{A}{a}\right)^{\frac{3}{2}} \cdot \left(\frac{P_w}{P}\right)^{\frac{1}{2}} = X \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Now, if H_s = superheat taken in per hour by steam

$$H_s = H - H_w$$

$$\therefore \frac{H_s}{H} = 1 - \frac{H_w}{H} = 1 - \frac{1}{X}$$

$$\text{or } H_s = H\left(1 - \frac{1}{X}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Let ${}_gc_p$ = mean specific heat of the flue gases, at approximately constant pressure;

T_1 and T_2 the temperatures of the flue gases at firebox end and smoke box end respectively;

then $H = NW \cdot {}_gc_p(T_1 - T_2)$.

$$\therefore \quad (T_1 - T_2) = \frac{H_s}{{}_gc_p \cdot \left(1 - \frac{1}{X}\right) \cdot W.N} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

It must be borne in mind that the steam as supplied by the boiler to the superheating elements is by no means dry, and of a dryness fraction which may not easily be determined; but an average figure might be assumed as consistent as possible with the usual working conditions of the boiler. The quantity H_s, as used above, is meant therefore to cover the addition of the latent heat necessary to dry the steam prior to its being superheated, over and above the superheat proper.

Let Q = total coal consumption in lb/hour

then steam/hour = QS lb.

Let "$\frac{w_s}{a_s}$" for the steam side = k lb/sec per sq.ft. of cross sectional area, then $$N = \frac{QS}{3600\ k.a_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where a_s = cross sectional area of Superheater Element.

It remains to develop an additional equation to determine the length of each element.

The difficulty here lies in the arrangement of each limb of the superheater element. While the flue gases maintain a simple uniform path, the steam flow is partly in the same direction as the flue gases and partly in the opposite direction. An exact solution for the mean temperature/

temperature difference would therefore be an elaboration not quite justified by the approximations stated at the beginning of the problem.

It is probably quite sound to take the average temperature of the steam between the two limits of outlet and inlet temperatures and subtract this from the mean temperature of the flue gases. Calling this mean temperature difference T_m, the overall rate of heat transmission h_o B.Th.U/sec., per sq.ft., per °F., the heating surface $P_s \cdot \ell'$, where ℓ' = mean length of each member of one element, then

$$h_o \cdot T_m \cdot N \cdot P_s \cdot \ell \cdot = \frac{H_s}{3600} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Recapitulating:-

From (1) $\frac{W}{w} = (\frac{p}{P})^{\frac{1}{2}} \cdot (\frac{A}{a})^{\frac{3}{2}}$

From (3) $X = (\frac{p}{P_W})^{\frac{3}{2}} \cdot (\frac{A}{a})^{\frac{3}{2}} \cdot (\frac{P_W}{P})^{\frac{1}{2}}$

From (5) $W.N.(1 - \frac{1}{X}) = \frac{H_s}{g c_p (T_1 - T_2)}$

From (6) $N = \frac{Q.S}{3600\ k.a_s}$

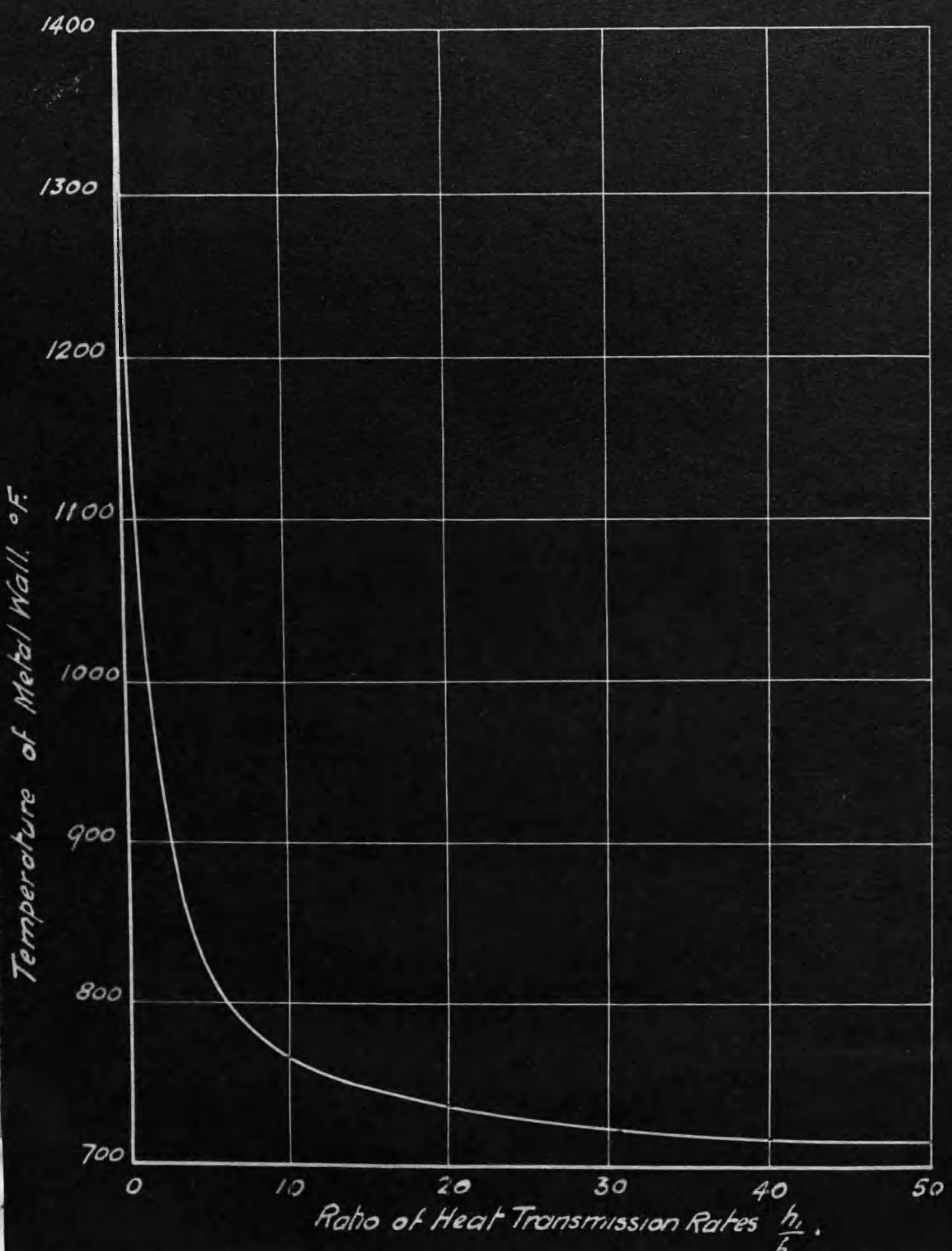
From (7) $\ell = \frac{H_s}{3600 . h_o . T_m . N . P_s}$

and lastly (8) $NW + nw = G$.

These six equations provide a complete solution to the problem.

Probably the most important of these equations is (6), where the number of elements and k values must be such as to ensure the metal temperature remaining within safe limits at all loads. The fluctuation of the temperature of the metal must be kept small, in order to prevent undue fatigue from the repeated and unequal contraction and expansion. It has been shown in the introduction that if T = temperature of hot fluid, θ = temperature of metal wall and t = temperature of cold fluid at any section, then $\frac{T - \theta}{\theta - t} = \frac{h_1}{h_2}$, where h_1 and h_2 are the rates of heat transmission from metal to cold fluid (superheated steam), and from hot fluid (flue gases) to metal respectively.

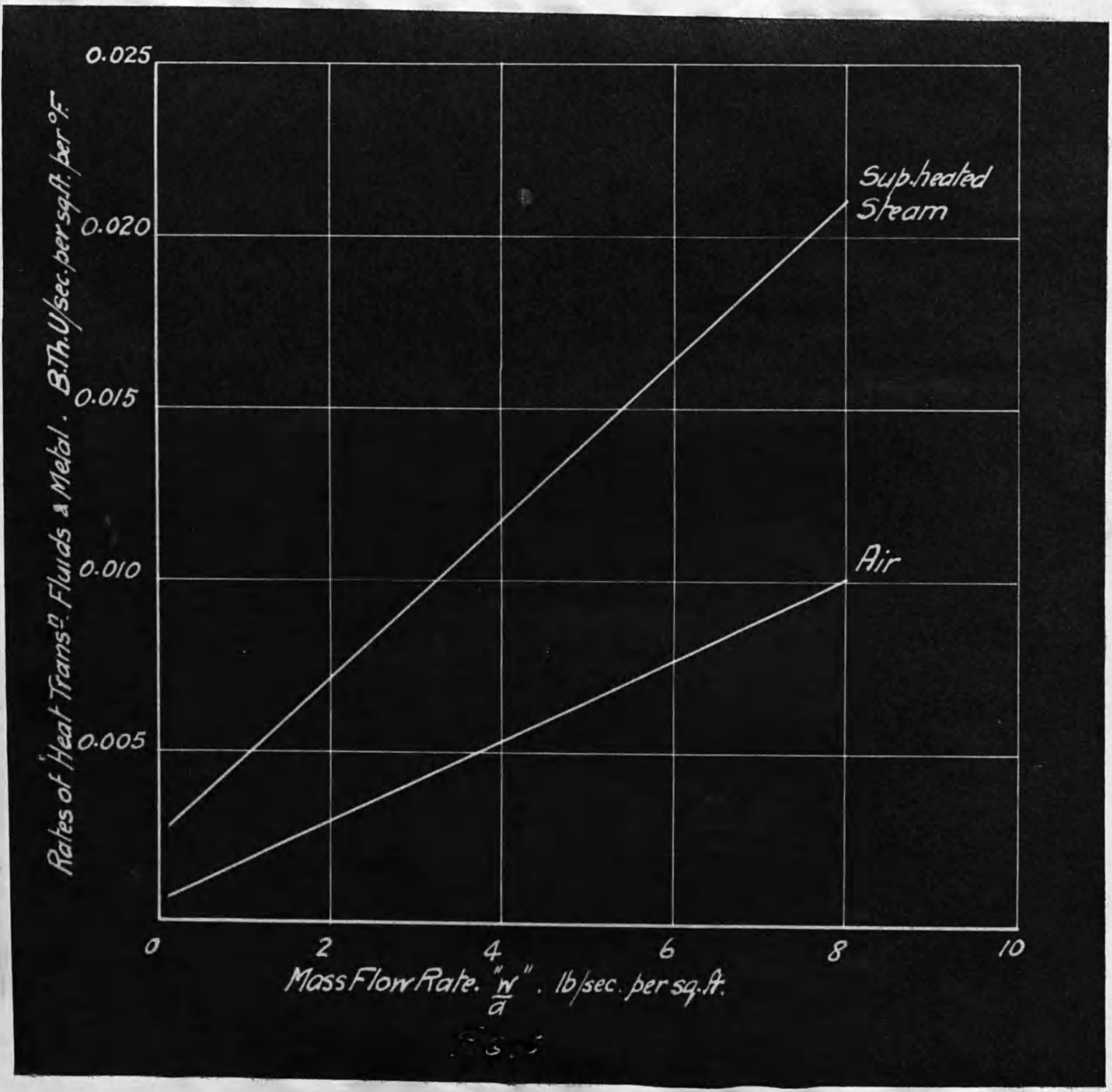
Taking/



1400
1300
1200
1100
1000
900
800
700
Temperature of Metal Wall. °F.
0
10
20
30
40
50
Ratio of Heat Transmission Rates $\frac{h_1}{h_2}$.

FIG 15

Taking T = 1400°F, t = 700°F, θ has been plotted against $\frac{h_1}{h_2}$ in fig. 15, it will be seen that ratios of $\frac{h_1}{h_2}$ varying from 10 to 50 must be provided for in the design in order to limit the range of fluctuation of the metal temperature.



0.025
0.020
0.015
0.010
0.005
Rates of 'Heat Transn. Fluids & Metal. B.Th.U/sec. per sq.ft. per °F.
Sup.heated Steam
Air
0
2
4
6
8
10
Mass Flow Rate. "w/a". lb/sec. per sq.ft.

FIG. 16.

CONCLUSION.

The main object of this thesis is the determination of the rate of heat transmission between air and metal, and superheated steam and metal. This object has been achieved and fig.16 shows the results, the outstanding feature of which is the confirmation of the Reynold's law.

The relative positions of the lines in fig.16 also show that the rates of heat transmission between fluids and metal surfaces are proportional to the specific heats of these fluids, provided there is no change of state (such as from liquid to vapour), and the velocities of the fluids are higher than the critical.

Fig.16 has been drawn after due allowance had been made for the lowest points in the preceding plots, and forms what might be termed a working chart for design purposes.

The full application of these important coefficients has also been indicated in connection with the design of industrial plants.

The question of pressure drops in all the practical applications considered has been purposely omitted as being a separate entity from the purely heat transmission side.

A P P E N D I X . 1.

Tables C, E, F, G, H, showing the principal readings and results for all the tests which have been plotted on the graphs contained in this thesis.

Table C . Air-Metal-Air.

Hot Air inside Inner Tube. Area of Flow for Hot Air = 0.00413 sq.ft; for Cold Air 0.0365 sq.ft

Number of Test	2	3	6	10	14	23	24	25	26	27	28	29
Hot Air Temp. at Inlet °F	566	566	572	549	440	430	471	491	502	512	512	510
" " " " Outlet °F	346	344	339	315	243	320	343	350	348	342	326	315
Cold Air. Temp at Oulet °F	277	278	286	285	255	185	202	215	223	232	241	247
" " " " Inlet °F	75	75	75	73	72	75	75	74	75	75	75	74
Air Flow "w" lb/sec.	0.00855	0.00845	0.00767	0.00643	0.00434	0.02123	0.01888	0.01726	0.01513	0.01275	0.0101	0.00877
Metal Temp at Hot Air Inlet °F	524	524	530	503	402	403	449	469	473	476	472	467
" " " " " Outlet °F	255	253	247	228	178	273	282	286	283	272	256	245
Mass Flow Rate. "$\frac{w}{a}$" Cold Air	0.2345	0.2315	0.2105	0.1762	0.1190	0.580	0.517	0.473	0.415	0.350	0.277	0.241
Rate of Heat Trans^n. Metal to Air B.Th.U/sec. per sq.ft. per °F	0.000928	0.000922	0.000894	0.000825	0.00071	0.00130	0.00122	0.00122	0.00114	0.00106	0.000943	0.00089
Mass Flow Rate. "$\frac{w}{a}$". Hot Air.	2.07	2.06	1.86	1.56	1.05	5.14	4.57	4.18	3.66	3.09	2.45	2.12
Rate of Heat Trans^n. Air to Metal B.Th.U/sec. per sq.ft. per °F	0.00369	0.00364	0.00343	0.00299	0.00221	0.00858	0.00737	0.00711	0.00657	0.00542	0.00458	0.00407

Cold Air Inside. Area of Flow for Cold Air 0.00413 sq.ft.

Number of Test	35	36	37	38	39	40	41	42
Hot Air Temp at Inlet °F	449	452	451	451	442	454	445	443
" " " " Outlet °F	311	311	311	305	279	257	242	213
Cold Air Temp at Outlet °F	165	169	172	179	194	202	206	213
" " " " Inlet	64	64	65	67	68	70	70	72
Air Flow "w" lb/sec.	0.0231	0.0215	0.0197	0.0165	0.0106	0.00789	0.00641	0.00403
Metal Temp at Hot Air Inlet °F	221	227	232	240	256	257	262	273
" " " " " Outlet °F	87	89	92	93	98	99	102	109
Mass Flow Rate "$\frac{w}{a}$" Cold Air	5.6	5.21	4.78	3.99	2.56	1.91	1.55	0.976
Rate of Heat Trans^n. Metal to Air B.Th.U/sec. per sq.ft. per °F	0.0077	0.00683	0.00640	0.00555	0.00386	0.00322	0.00269	0.00162

Table E Experiments Group 2.

Series A

Number of Experiment	1	2	3	4
Air Temp at Inlet °F	509	420	504	506
" " " Outlet °F	440	365	419	415
Air Flow lb/sec	0.0516	0.065	0.0519	0.0521
Heat lost by Air BThU/sec	0.855	0.856	1.060	1.14
Steam Press	Atmospheric			
Steam Temp at Inlet °F	240	240	240	239
" " " Outlet °F	349	285	303	293
Steam Flow lb/sec. W_s	0.00862	0.02075	0.0248	0.032
Steam Mass Flow Rate "W_s/a_s"	2.08	5.02	6.00	7.74
Metal Temp at Air Inlet °F	385	310	340	327
" " " Air Outlet °F	266	243	248	246
Heat gained by Steam BThU/sec	0.471	0.467	0.781	0.864
"h_s" Rate of Heat Trans^n Metal to Spd. Steam BThU/sec. per sq. ft. per °F.	0.00885	0.0161	0.0178	0.0217

Table F Experiments. Group a.

Series:	B ⊙										C ×								D •			
Number of Expt.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
Air Temp at Inlet °F	404	404	414	411	407	409	406	413	418	413	414	409	409	405	410	407	407	408	500	501	494	492
" " " Outlet °F	372	362	357	356	355	351	350	355	356	353	369	361	360	351	354	347	348	351	429	424	409	406
Steam Press.	← Atmospheric →										← Atmospheric →								← 24.0 lb/in² abs. →			
Steam Temp at Inlet °F	239	237	230	241	236	241	239	240	241	239	246	245	246	244	245	245	247	244	250	250	249	250
" " " Outlet °F	312	302	293	290	283	282	275	275	277	271	303	300	297	282	281	281	278	276	336	319	307	299
Metal Temp at Air Inlet °F	326	324	320	315	308	307	298	297	301	293	330	324	322	305	304	302	300	298	371	357	340	328
" " " Air Outlet °F	258	252	248	248	242	246	244	244	247	242	261	255	265	249	260	248	250	247	269	263	260	255
Steam Flow W_s lb/sec.	0.00236	0.0071	0.0121	0.0181	0.0186	0.0214	0.0278	0.0304	0.0322	0.0356	0.00845	0.0122	0.0149	0.0230	0.0257	0.0275	0.030	0.032	0.01162	0.0167	0.0228	0.0295
Steam Mass Flow Rate "w_s/a_s"	0.57	1.71	2.92	4.36	4.50	5.17	6.72	7.35	7.76	8.63	2.04	2.95	3.60	5.56	6.20	6.66	7.25	7.74	2.81	4.04	5.52	7.14
"h_s" Rate of Heat Trans[n] Metal to Spd. Steam, B.Th.U/sec. per sq.ft. per °F.	0.00362	0.0072	0.0097	0.0143	0.0143	0.01445	0.0185	0.0208	0.0204	0.0229	0.0065	0.0105	0.0119	0.0160	0.0172	0.0203	0.0190	0.0208	0.0104	0.0121	0.0160	0.0208

Table G Experiments. Group b.

Number of Experiment	1	2	3	4	5	6	7
Steam Temp at Inlet °F.	449	535	462	601	570	508	433
" " " Outlet °F.	345	357	316	306	309	308	317
Steam Flow lb/sec W_s	0.0292	0.0190	0.0234	0.0095	0.01463	0.01835	0.030
Steam Mass Flow Rate "W_s/a_s"	7.03	4.47	5.66	3.68	3.45	4.45	7.26
Air Temp at Inlet °F	63.0	64.5	62.0	64.0	66.5	67.0	68.0
" " " Outlet °F.	316	335	285	270	269	266	276
Air Flow lb/sec W_a	0.0246	0.0266	0.0321	0.0283	0.0392	0.0365	0.036
Air Mass Flow Rate "W_a/a_a"	3.20	3.45	4.17	3.68	5.10	4.74	4.68
Metal Temp at Steam Inlet °F	395	440	390	401	391	381	401
" " " " Outlet °F	300	290	254	252	240	245	261
"h_s" Spd. Steam to Metal. BThU/sec. per sq.ft. per °F.	0.0198	0.0136	0.0165	0.00838	0.0117	0.0137	0.0200
"h_a". Metal to Air. BTh.U./sec per sq.ft. per °F.	0.00447	0.00465	0.00620	0.00480	0.00623	0.00638	0.00612
"h_o" overall from $\frac{1}{h_o} = \frac{1}{h_s} + \frac{1}{h_a}$.	0.00389	0.00374	0.00471	0.00310	0.00411	0.00488	0.00494
t_m. overall Mean Temp diff. °F	197.5	243	214	277	274	241	200
"h_o" from overall observations.	0.00422	0.00388	0.00441	0.00280	0.00383	0.00415	0.00500

Table H. Experiments. Group C.

Number of Experiment	1	2	3	4.
Steam Press.	← Atmospheric →			
Steam Temp. at Inlet °F. T_1	443	447	427	530
" " at Outlet °F. T_2	340	307	305	311
Steam Flow. W_s lb/sec.	0.0304	0.0240	0.0339	0.0160
Steam Mass Flow. Rate $''\omega_s/a_s''$.	7.35	5.80	8.20	3.88
Air Temp. at Inlet °F. t_2	64	66	67	67
" " at Outlet °F. t_1	295	270	255	274
Air Flow. W_a lb/sec.	0.0272	0.0331	0.0432	0.0338
Air Mass Flow Rate $''W_a/a_a''$.	3.53	4.30	5.60	4.40
Rate of Heat Trans.^{n} Metal to Air. h_a BThU/sec. per sqft. per °F.	0.0053	0.0065	0.0077	0.0063
$1/h_a$.	188	154	130	159
Mean temp diff. °F. $t_m = \dfrac{(T_1-t_1)-(T_2-t_2)}{\log_e \frac{T_1-t_1}{T_2-t_2}}$	205	202	203	250
Rate of Heat Trans.^{n} Overall h_o. BThU/sec. per sqft per °F	0.00424	0.00463	0.00567	0.00389
$1/h_o$	236	216	177	257
$1/h_s$.	48	62	47	98
h_s Rate of Heat Trans.^{n} Spd. Steam to Metal. BThU/sec. per sqft per °F.	0.0208	0.0161	0.0213	0.0102

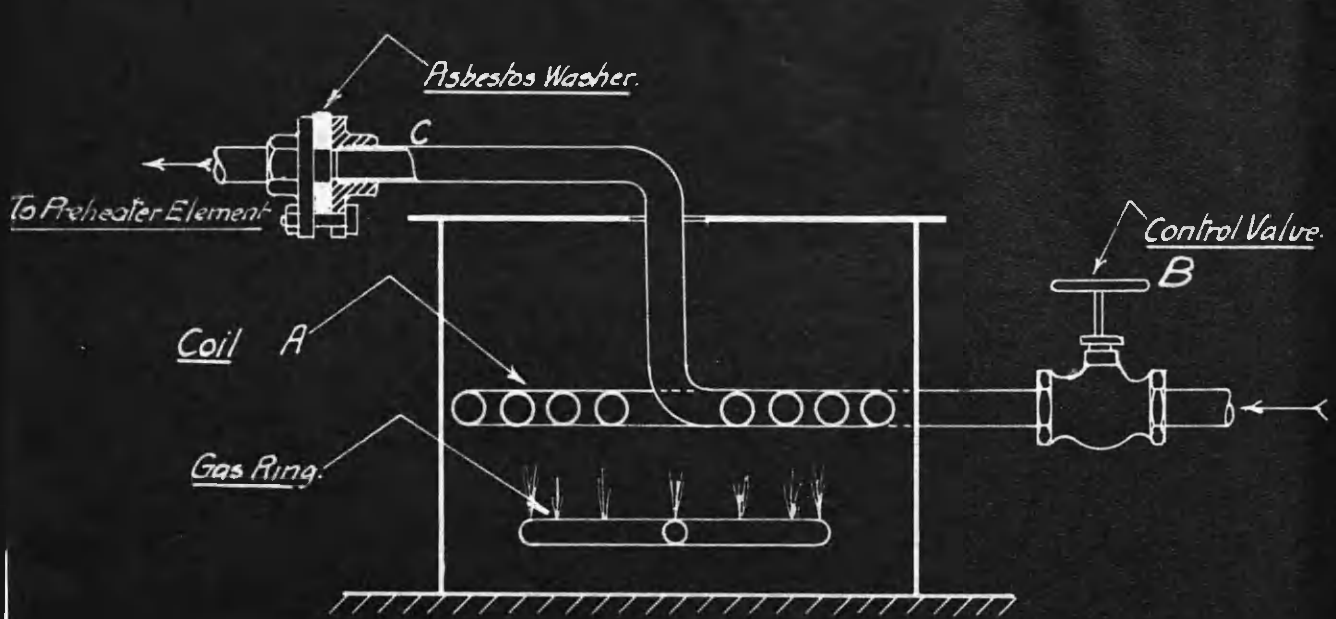


FIG 17.

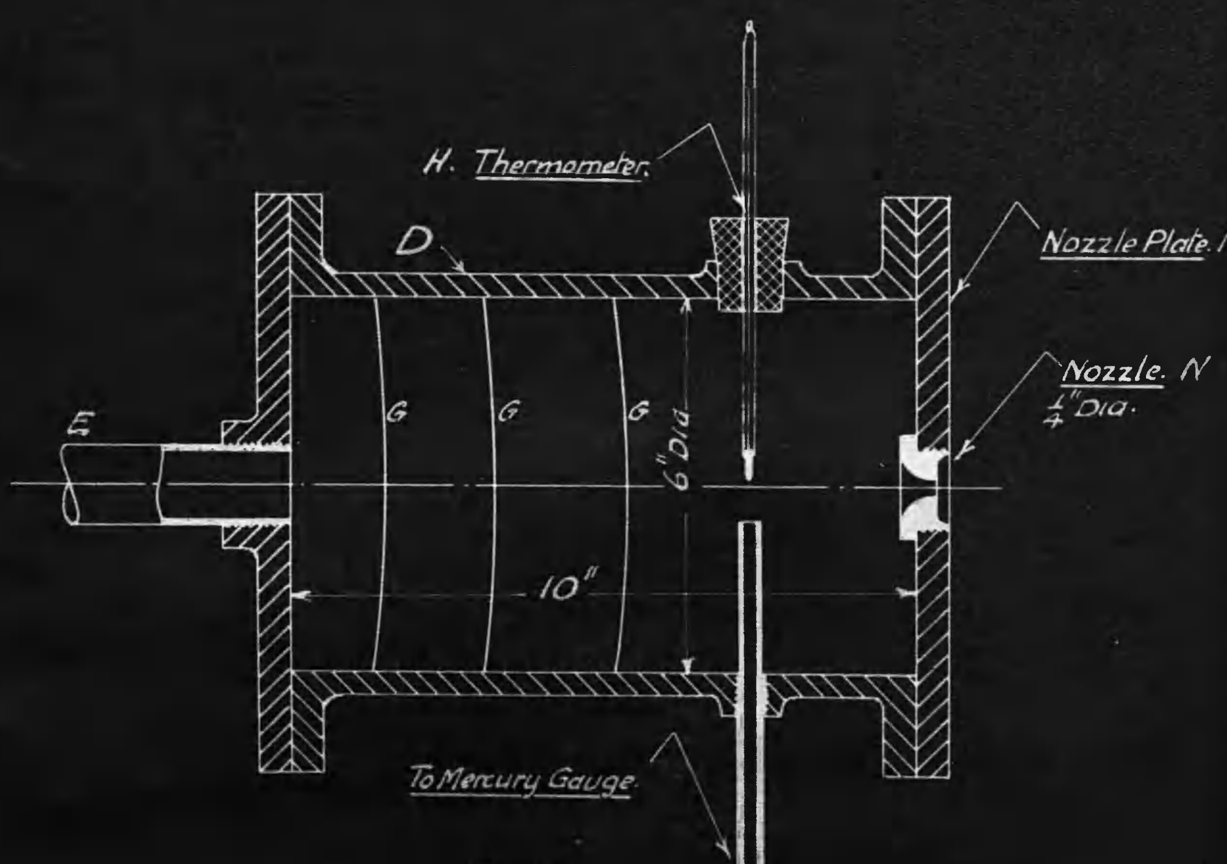
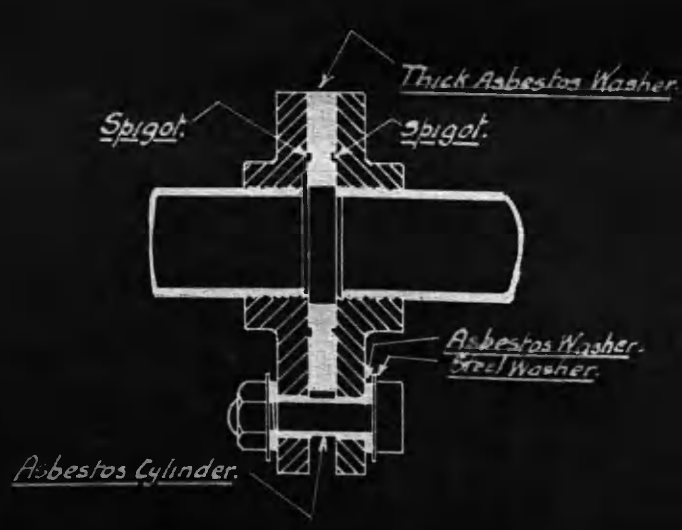


FIG 18.



DETAILS OF INSULATED JOINT.

APPENDIX II.

Particulars of air heater and nozzle box.

The air heater, fig.17, consists of a steel pipe 1" internal diameter, coiled in the form of a spiral of four complete turns. One end of this coil A, is connected to the pipe from the receiver of an air compressor, by a valve B, the other end C is connected to the inner pipe of the preheater by a pair of flanges, separated by means of a piece of thick asbestos sheeting. The shanks of the connecting bolts are sheathed in asbestos, while thick asbestos washers are placed under the head and nut of each bolt. These refinements were made in order to reduce to a minimum any longitudinal conduction of heat along the length of the preheater element.

The nozzle box, Fig.18, is a flanged cylindrical casting D, 6" dia. x 10" long. It is connected at one end to the air discharge pipe E, from the cooler, and at the other end it is fitted with a cover F, into which the nozzle N is screwed. Fine wire gauge discs G, placed inside the cylinder, steady the air flow. The temperature of the air in front of the nozzle is measured by the mercury thermometer H, while tube K, screwed into the nozzle box is connected to the mercury manometer used to measure the pressure of the air.

The nozzle N is of the convergent type, $\frac{1}{4}$" dia. at exit, with well rounded entrance and $\frac{1}{32}$" of parallelism at outlet.

The discharge coefficient for the nozzle was not assumed, but was obtained directly from experiments, as follows.

In the Laboratory where these researches were conducted, is a "Nozzle Testing" apparatus, designed about 5 years ago by Professor W. Kerr, Ph.D., A.R.T.C. By this apparatus, it is possible to supply steam, saturated or superheated, at definite pressures to a nozzle screwed into a removable plate. The discharge from the nozzle is condensed in a surface condenser, while the discharge pressure can be regulated by a valve placed on the discharge pipe to the condenser. The $\frac{1}{4}$" dia. nozzle, proposed to measure the air discharge was placed in the "nozzle testing" apparatus; steam, first at 20 lb/in^{2}abs., second at 25 lb/in^{2}abs. with relatively/

relatively high superheat, was passed through the nozzle; the discharge from the condenser was weighed; and both supply and exhaust pressures were noted over considerable periods. The theoretical discharge was calculated by means of Callendar's steam tables, and this divided into the actual discharge gave the required coefficient. It is this coefficient which was used in calculating the actual air discharge from the air preheater, by means of the usual formula for the flow of gases through nozzles.