

CORRIGENDA.

Page 3. Insert between 3rd and 4th lines:

" θ_s Initial surface temperature of cooling water".

Page 5. 19th line - Delete "soakage" and insert "flow".

Page 30. 16th line should read: " $W_c H_s \pm W_w \theta'_m$ "

19th line should read; " $\left(\frac{\theta'_m}{\theta_s}\right)\left(\frac{W_w}{W_{so}}\right) = f_3\left(\frac{\alpha_w r}{l^2}\right)$ "

THE MECHANISM OF CONDENSATION IN A TURBULENT STEAM-AIR MIXTURE
IN A JET-TYPE CONDENSER.

by

John R. Botterill B.Sc.(Eng.). G.I.Mech.E.

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Imperial College of Science & Technology.

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NOTATION

- A Area of condenser cross-section; A_o , steam-air duct diameter at entry to condenser; A_w , cooling water orifice area, ft^2 .
- C_p Specific heat/unit mass at constant pressure, $\text{CHU/lb.}^\circ\text{C}$.
- D Diameter of duct; D_i , inside of cooling water jet, ft.
- E Eddy diffusivity; E_M , eddy mass diffusivity; E_H , eddy thermal diffusivity, ft^2/hr .
- f Function.
- G Mass velocity, $\text{lb/ft}^2\text{hr}$.
- g Gravitational acceleration, ft/sec^2 .
- H Enthalpy or heat content of steam-air mixture; H_s , steam, CHU/lb .
- h Metering head (Flowmeters), ins. water.
- I Diffusion rate of heat by eddy convection
Diffusion rate of heat by eddy conduction
- k Constant
- L Latent heat, CHU/lb .
- l Mixing length according to Prandtl's theory of turbulence.
- l Thickness of cooling water jet, ft.
- Δl Increase in jet thickness due to steam condensation, ft.
- M Molecular weight.

- p Pressure, ins. Hg.
- q Heat transfer rate/unit area, CHU/ft².hr.
- Re Reynolds number of steam-air mixture.
- r Turbulence factor = $\frac{\text{actual cooling water temperature rise}}{\text{theoretical cooling water temperature rise}}$
- S Surface area, ft².
- t Temperature of steam air mixture; t_w , water, °C.
- U Overall heat transfer coefficient in header section, CHU/ft².hr.°C.
- u" Instantaneous fluctuating velocity normal to duct wall, ft/sec.
- V Velocity of steam-air mixture; V_s , steam; V_w , water, ft/sec.
- v" Instantaneous fluctuating velocity in direction of flow, ft/sec.
- W Mass flow rate of the steam-air mixture; W_s , steam; W_a , air; W_c , condensate; W_w , cooling water, lb/hr.
- ω_s Steam mass diffusion rate/unit area, lb/ft².hr.
- x Position in a lamina of thickness $2l$, normal to surface, ft. Also mol fraction in viscosity equation, page. 63.
- y Radial distance, ft.
- z Distance along condenser from entry, ft.
- α_w Molecular thermal diffusivity of water, ft²/hr.

$$\gamma = \frac{E_M}{E_H}$$

θ Temperature at any point in cooling water jet assuming zero initial temperature, °C.

θ_m Theoretical mean temperature rise of cooling water, °C.

θ_{m1} Mean temperature rise if exposed to a constant surface temperature, θ_m .

θ_{m2} Mean temperature rise if exposed to a temperature $(\theta_s - \theta_m) \cdot e^{-k\tau}$.

θ_{surf} Surface temperature of cooling water assuming zero initial temperature.

Prime, ()', denotes actual temperature conditions of cooling water, °C.

λ Time with reference to equation (4), $0 < \lambda < \tau$

μ Viscosity of steam-air mixture; μ_s , steam component; μ_a , air component, lb/ft.hr.

ρ Density of steam-air mixture; ρ_s , steam component; ρ_a , air component, lb/ft.³

τ Time of exposure of cooling water to steam or steam-air mixture, hours.

Suffix "0" refers to condition at condenser inlet.

"h" refers to header section.

INTRODUCTION

The increased use of jet-type condensers in recent years has emphasized the lack of experimental data with which to make any pretence of design. It is well known that very high rates of heat transfer are possible compared with the more usual surface-type condenser and this makes for suitable applications where:-

- (i) the condenser space limitations are severe
- (ii) the condensing vapour is mixed with a
non-condensable gas
- (iii) there is a combination of (i) and (ii)

It was considered that a fundamental study of the mechanism of condensation of a vapour from a mixture with a non-condensable gas in a jet condenser was well merited but it was realised that all aspects of the problem could not be covered in the time available. The present research endeavours to provide data on the effect of inlet conditions on the condensation rate with parallel cooling water flow. The influence of water jet dispersion on the condensation rate and the general problem of pressure change across the condenser were not investigated as the apparatus could not accommodate the additional complications in duct construction for the former and extra condenser

length to make any useful readings on the latter. Steam and air were chosen as the vapour and non-condensable gas as these are most readily available and commonly encountered in practice.

Determination of the rate of steam condensation throughout the condenser, required a knowledge of the bulk temperature of the cooling water and an assessment of the diffusivity in the steam-air mixture required its temperature gradient. The fluid phase configuration in a normal jet condenser is a quasi-homogeneous water-steam-air mixture and will, of course, only yield a water temperature measurement of doubtful value. It was thus considered essential to depart from normal practice and separate the two fluid phases sufficiently to take temperature readings on each. The interpretation of the results is dealt with in the light of this expedient in the Discussion.

The theoretical analysis uses the transient heat soakage relation on the water phase but it is not self-sufficient as:

- a) turbulence substantially increases the heat transfer rate calculated in accordance with the molecular diffusivity and nominal surface area of the water jet.

- b) the true surface temperature decay rate of the water is not known.

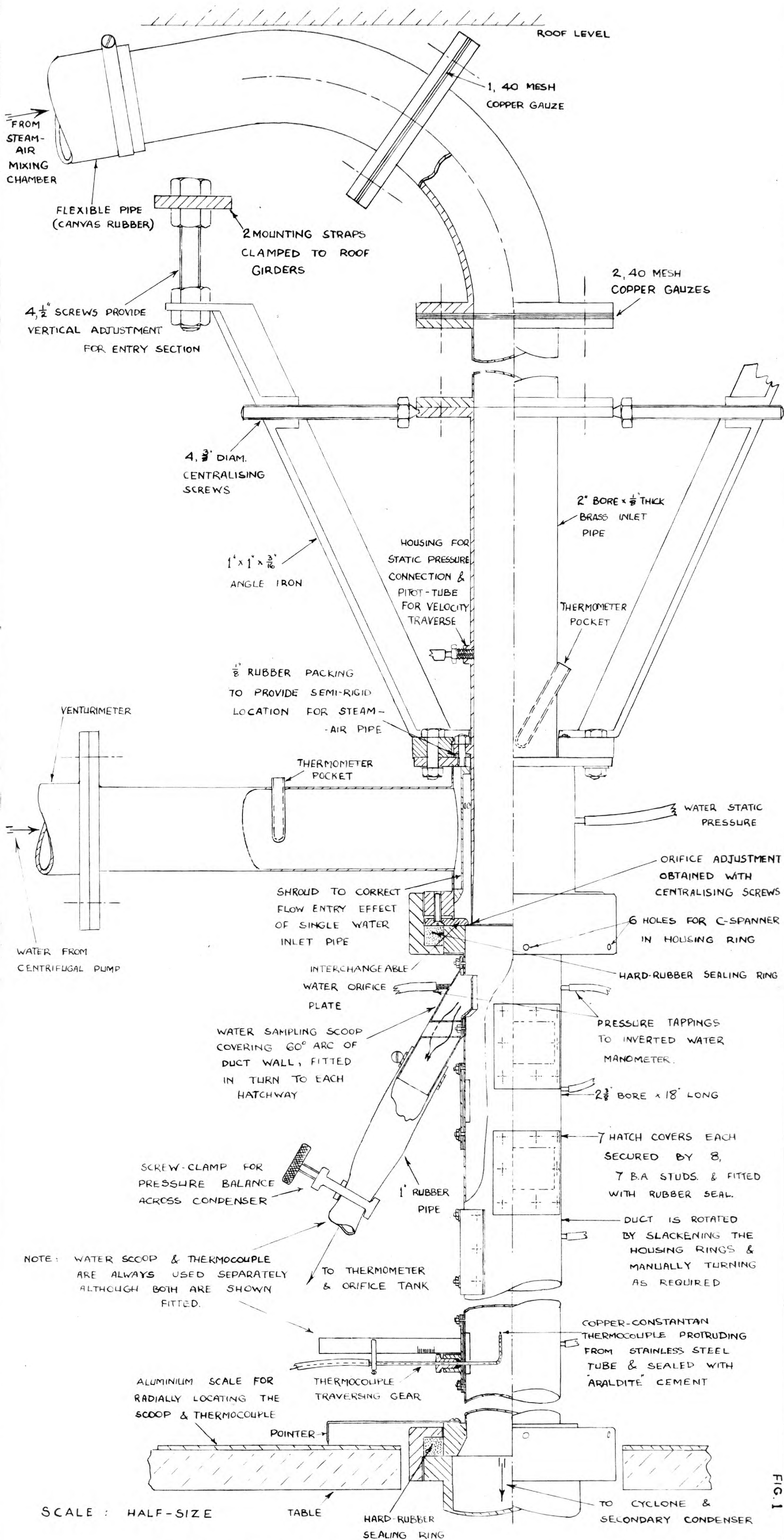
The effect of turbulence is allowed for empirically by using results for the condensation of steam without premixed air. A tentative assumption is used for the surface temperature of the cooling water and from this evolves a dimensionless parameter which permits correlation of results using steam-air mixtures.

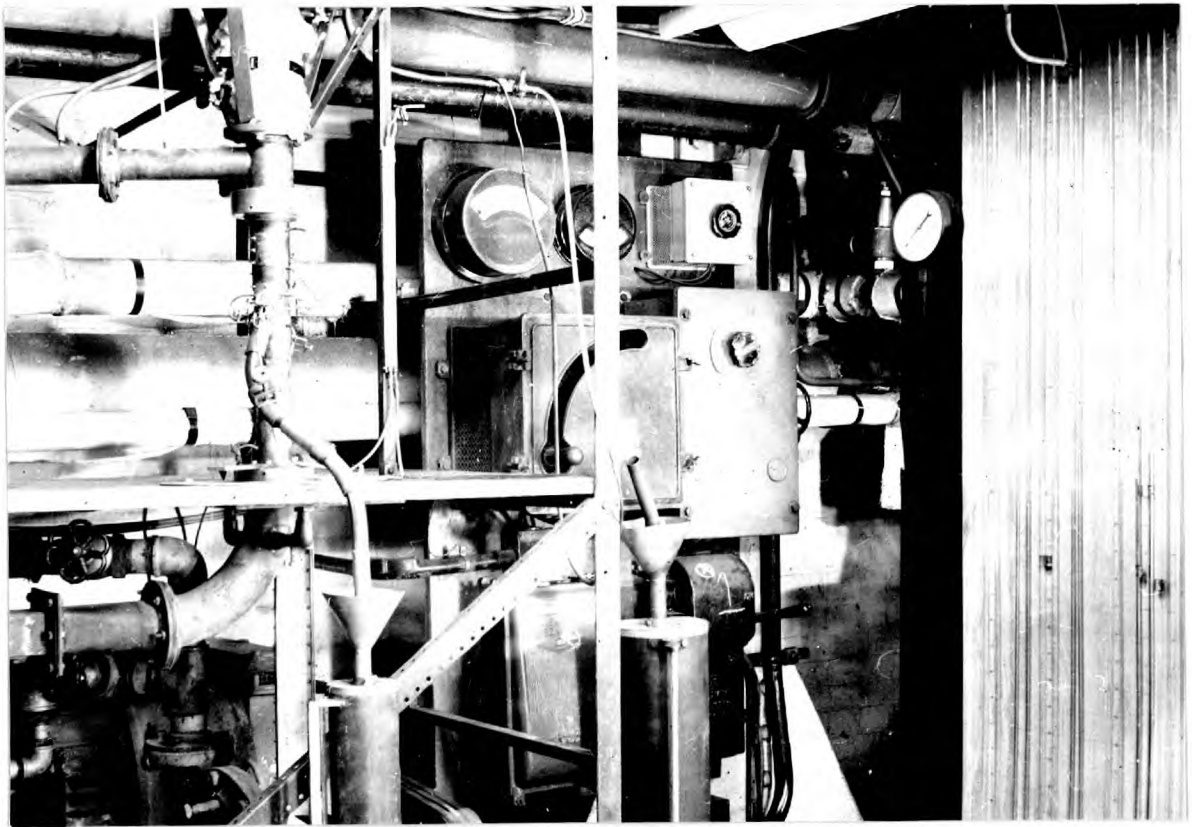
The steam condensation rate has been studied with and without premixed air at various inlet velocities and steam-air ratios and the influences of water inlet velocity and jet thickness have been determined. In each case the experimental data is presented on the same dimensionless coordinates. Values of the eddy diffusivity in the steam-air mixture have been calculated from the results and compared with existing data.

I. THE WORKING SECTION

The jet condenser body. This was essentially a vertical duct (fig. 1) down which flowed the condensing steam and air and the parallel-flowing cooling water. A vertical tube was used to eliminate gravitational cross-flow effects inherent in horizontal jet condensers and a circular cross-section offered the simplest solution to the practical side of the problem. The $2\frac{3}{8}$ " duct diameter was largely determined by the cooling-water sampling technique adopted and the necessity of ensuring that the inlet steam-air velocity was sufficiently high with the steam throughput available. The limited condenser length of 18" was imposed by the headroom which had to accommodate the inlet section, condenser and ancillary equipment, but in view of the very high rates of heat transfer, the limitation in length was not considered to be a disadvantage. Seven hatchways were cut down the length of the duct, each being supplied with a coverplate secured by 8, 7 B.A. studs and nuts, the inside surface of each cover being arranged to lie flush with the bore of the condenser.

The header section. Ideally in this type of investigation the cooling water should be brought into contact with the

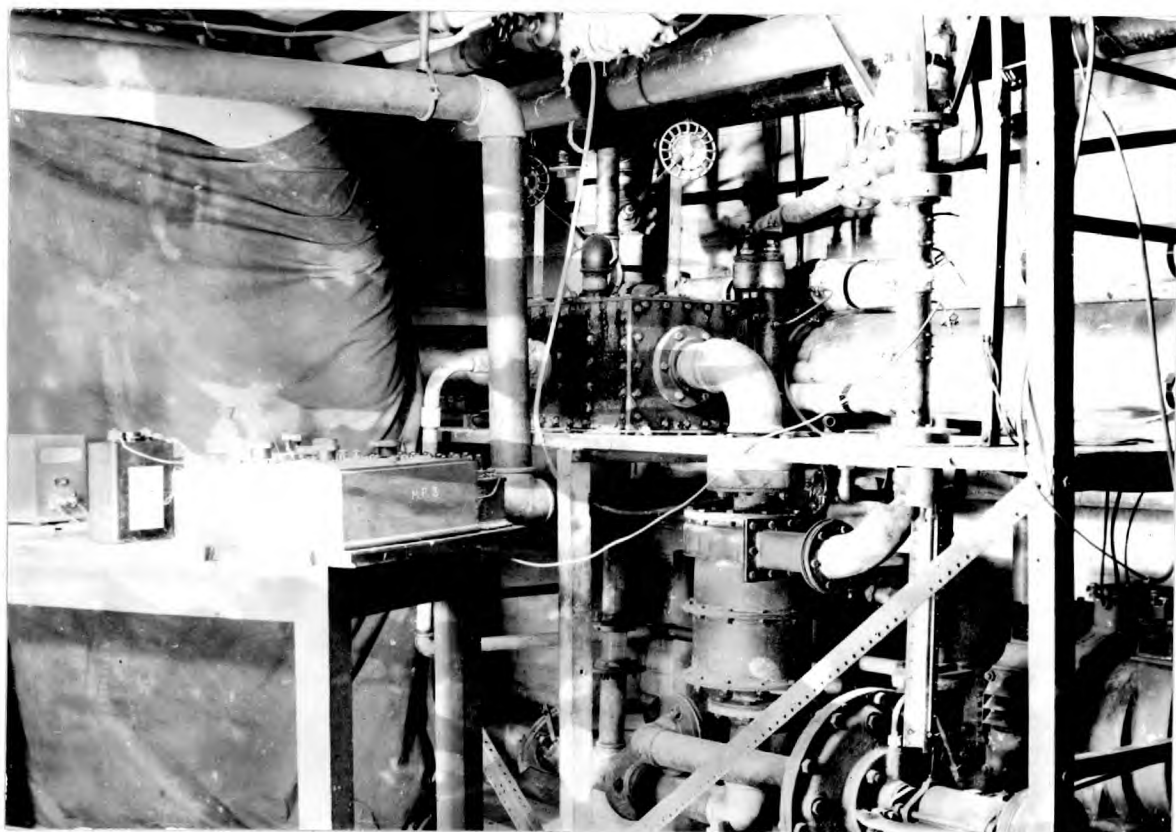




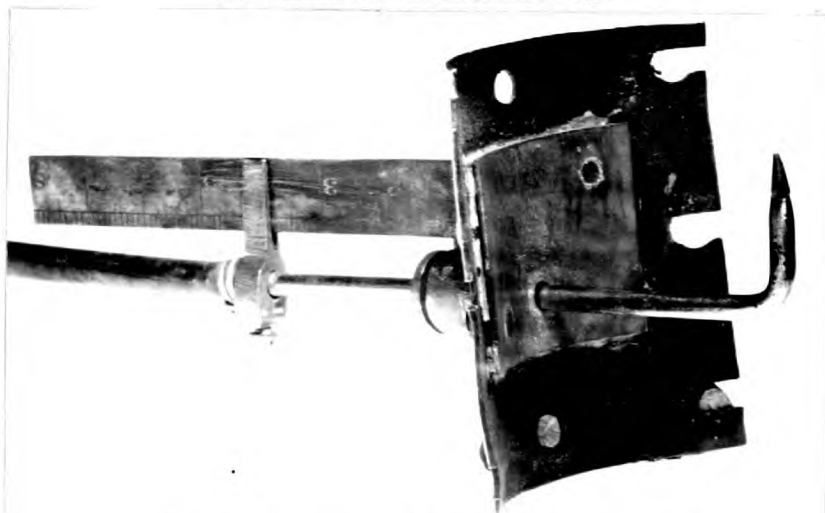
General View of Apparatus with scoop in No.4 Hatch



Cooling Water sampling scoop



General View of Apparatus with Thermocouple
probe in No. 4 Hatch



Traversing Thermocouple Probe

steam-air mixture without prior heat transfer having occurred between the steam and the water. At the time of building the apparatus this was considered impracticable and a short header annulus was incorporated as a means of supplying cooling water to the annular orifice (fig. 1); this header annulus was, in effect, a short length surface condenser. For the analysis of results using steam only, allowance has been made for the heat transfer occurring through the inner wall of the annulus (page 40). When condensing steam in the presence of air, the separate condensation occurring in the header was neglected due to the greatly decreased rates of heat transfer and lack of data on the effective surface temperature on the steam-air side. Any steam condensed in the header is considered to be of secondary order and is included in that measured at the first sampling position down the condenser.

The cooling water jet. Initially a few runs were made using a series of $\frac{1}{8}$ " holes as an annular array of narrow angle jets about the steam-air mixture, since this system approximated to that commonly in use. Nothing useful was forthcoming and it was realised that the steam-air mixture and the water had to be essentially separate to take

reliable temperature measurements. The cooling water jet configuration was changed to one using a continuous annular jet issuing from an adjustable orifice having a small clearance between its outer cheek and the duct wall (fig. 1). The orifice size was adjusted by means of interchangeable plates which provided the outer cheek of the orifice, the investigation being carried out with orifice widths of 0.025" and 0.050". Jets from orifices of less than 0.025" were found to be very susceptible to flow irregularities under continuous running conditions and jets from orifices greater than 0.050" experienced only comparatively small temperature rises in the condenser length available and were considered to be outside the range of practical interest.

Before the main runs could be attempted, some means of ensuring a circumferentially symmetrical water jet had to be developed; this proved to be much more difficult than was first envisaged. The circumferential flow distribution was checked by tapping off 1/6th of the water (60° lap) via a scoop (described in the next section) and measuring the flow in a suitable orifice tank. Constructional accuracy and rigidity of the header section was itself insufficient to ensure a symmetrical water jet

and a means of adjustment of the orifice had to be provided in the assembled apparatus. This was achieved by 4 centring-screws moving the inner orifice cheek (the steam-air inlet pipe) which was located elsewhere on a semi-rigid flange mounting (fig. 1). Correction of the orifice eccentricity did not eliminate the elliptical flow distribution of the jet caused by the asymmetric water inlet supply to the header. This fault was eventually corrected by incorporating a $2\frac{1}{2}$ " diameter shroud with 36 0.088" diameter holes in the header water annulus, as shown in the diagram.

Prior to every run the circumferential flow symmetry of the cooling water jet was adjusted to within $\pm 1\%$ of the mean flow rate, measured in No. 1 hatchway, i.e. 1.43" from the water orifice. This order of accuracy was essential to ensure that the jet was within $\pm 5\%$ error in circumferential flow distribution, at the lower end of the condenser. Reference to fig. 4 shows how rapidly a small initial error in uniformity deteriorates along the length of the condenser.

Cooling water sampling technique. To study the rate of steam condensation it was necessary to know the bulk temperature rise of the cooling water along the condenser.

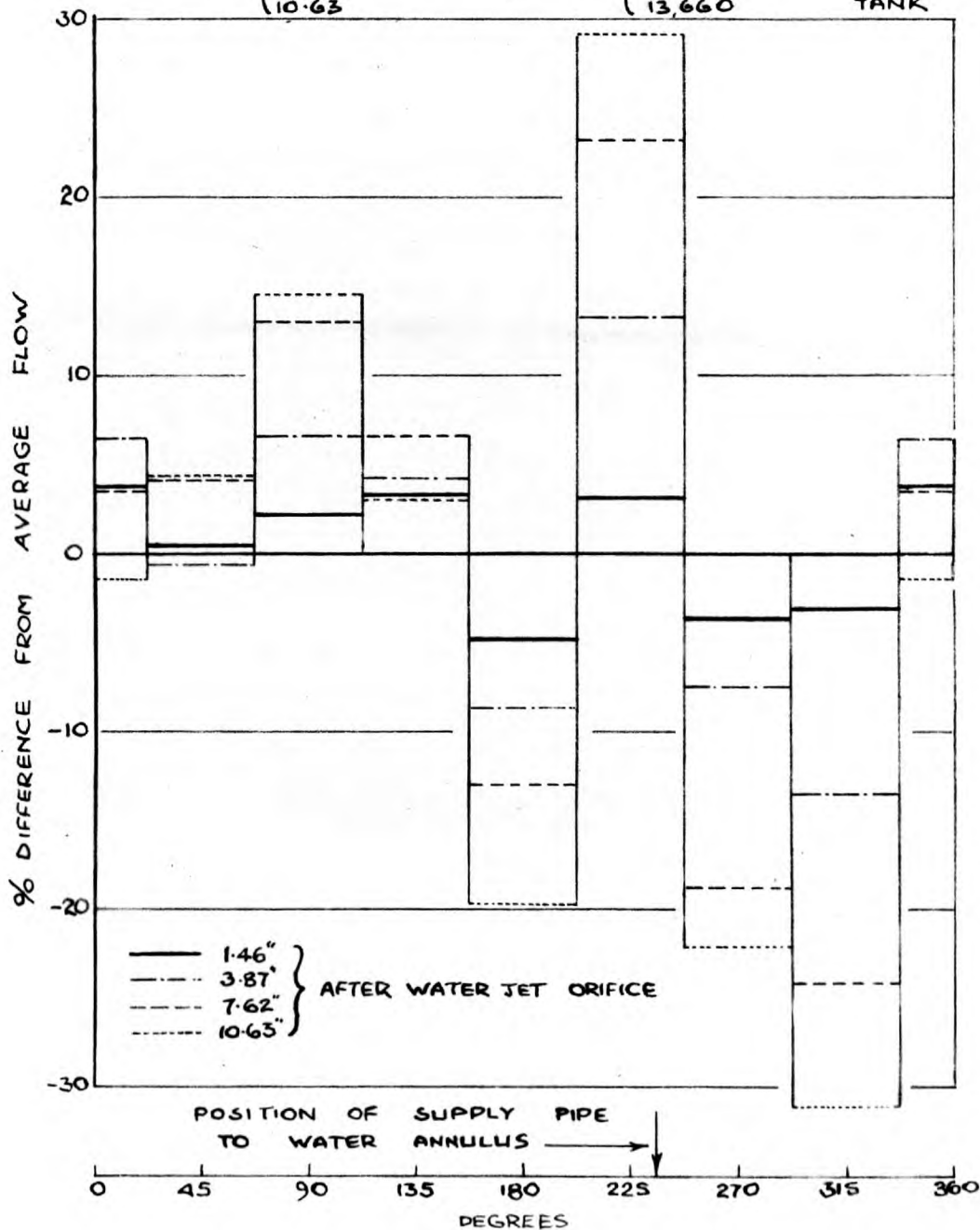
WATER FLOW DISTRIBUTION DOWN CONDENSER WITH A NON-UNIFORMITY OF FLOW AT THE WATER ORIFICE

[WATER ANNULUS FLOW CORRECTION SHROUD NOT FITTED]

AIR VELOCITY = 62 FT/SEC — NO STEAM PRESENT

WATER FLOW RATE THROUGH .050" ORIFICE = 14100 LB/HR — VENTURIMETER

WATER FLOW RATE	{	1.46" 3.87" 7.62" 10.63"	AFTER ORIFICE =	{	14,106 14,080 13,980 13,660	LB/HR	SCOOPE & ORIFICE TANK	% DIFFERENCE
								0
								-0.2
								-0.9
								-3.1



This was determined from temperature measurements with calibrated thermometers accurate to $1/10^{\circ}\text{C}$, at the condenser inlet and in water tapped from the main duct by a suitable scoop, the latter being fixed in turn to one of the six upper hatchways. Sampling from No. 7 hatchway was found to be impracticable due to the limited clearance above the rig table. The difficulty of changing the scoop from one hatchway to another with the condenser pressure above atmospheric and the complications arising in operating the scoop (described hereafter) at vacuum, limited this investigation to one at atmospheric pressure.

The construction of the scoop is shown in figs. 1 and 2, the optimum degree of penetration of the mouth being determined by trial and error. It was endeavoured to bleed off cooling water without extraneous steam-air mixture which would cause errors in the cooling water temperature measurement due to further heat transfer occurring before the cooling water contacted the thermometer. It follows that the scoop mouth had to penetrate into the duct just sufficiently to ensure collection of the cooling water to the exclusion of the steam and air. The unavoidable surface break-up of the cooling water, due to turbulence, made impossible an accurate assessment of the interface

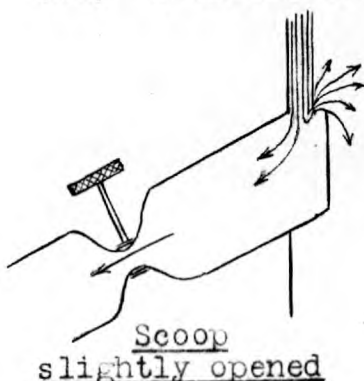
position but best results were obtained using a scoop having an outer lip at a radial position corresponding to the inner face of the cooling water orifice. During a test with this scoop, the cooling water collected 10.6" from the orifice was only 3% less than that passing through it as measured by the venturimeter, (fig. 4).

At the outset of the main runs it was noticed that the bulk temperature of the cooling water was very sensitive to the pressure difference between the scoop and the main duct and readings of temperature were therefore taken only when an accurate static pressure balance was achieved on an inverted water manometer. Although the manometer lines were purged with water before each reading the variations in cooling water temperature obtained at any hatchway at one condition, made essential further improvement in the experimental technique. The system of operation of the scoop at this time consisted of slowly opening the scoop pinch-cock at a uniform rate and noting the cooling water temperature when pressure balance was achieved. (Opening the scoop water line caused an increase in the water flow together with an increase in the average water temperature). It was observed that in the region of pressure balance a characteristic "flat"

was evident in the cooling water temperature rise and for reasons explained hereafter, the temperature corresponding to this "flat" was taken as the reading. The figure found by this technique was far more sensitive to pressure change than that obtained using the water manometer as an indication of pressure balance, the consistency of readings being high on rechecking. The following sequence of events is offered as an explanation for the changes in cooling water temperature in the scoop drain when the latter is opened steadily by unscrewing the pinch-cock on the scoop hose:-

(i) With a fully closed scoop the cooling water flow in the condenser is simply disturbed by the obstruction created by the scoop mouth.

(ii) With the scoop drain slightly opened, most of the

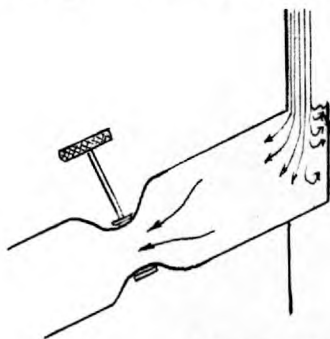


water entering the scoop is thrown out again into the main stream, that passing for temperature measurement and drain is from the region of the jet nearest to the condenser wall,

(iii) Further opening of the scoop will permit more cooling water to pass for temperature measurement,

the extra water coming from regions in the jet nearer the steam-air phase, i.e. warmer water.

(iv) The scoop will take away for temperature



measurement all the water flowing in that sector of the duct occupied by the scoop mouth.

Sequence (iv)

(v) Further opening will drop the level of the turbulent "back-flow" in the outer region of the scoop mouth and for a limited extent the temperature of the water alters little. When using steam-air mixtures, the very high air content about the scoop mouth further reduces any tendency for condensation inside the scoop.

(vi) The scoop appears to behave as an extractor, drawing steam and air noisily from the condenser, the condensing steam causing a temperature rise of $5 - 10^{\circ}\text{C}$ above that recorded in stages (iv) and (v).

At an axial distance down the condenser corresponding to the scoop mouth position in each hatchway the heat transfer rate from the steam-air mixture was calculated using the average of 6 bulk temperature readings on the

cooling water at 60° spacings around the jet. With circumferential non-uniformity of jet flow the heat transfer rate to the water should more correctly be assessed by summing the heat transfer rate at each of the 6 scoop positions. Fortunately it was found that the heat transfer rate calculated in accordance with the average bulk temperature was within 1% of that obtained by summing the heat transfer rates at each scoop position for a maximum circumferential flow error of 5% (a figure achieved). It was thus unnecessary to read the cooling water flow rate through the scoop to the orifice tank, at each scoop position. This would in fact be an extremely tedious and lengthy operation due to the settling time required in the tank.

Steam-air temperature traversing gear. A copper-constantan thermocouple, 29 S.W.G. wire (0.013 ins) was used in conjunction with a normal null-system potentiometer to read the steam-air temperature. The thermocouple was housed in a stainless steel tube and sealed with Araldite cement. The steel tubing was suitably cranked (fig. 1) to prevent cooling water from splashing onto the probe. The couple position in the duct was determined by the use of a steel scale attached to the mounting plate, the

latter being located in turn in different hatchways down the duct, fig. 3.

Flow visualisation. An interchangeable perspex condensing duct was used for observation of the flow pattern. Over the range of conditions covered when condensing steam in the presence of air, nothing worthy of note was discerned in the behaviour of the water phase. When condensing steam without premixed air, the cooling water jet was subject to instability at certain conditions. This is dealt with on page 38.

Main dimensions of jet condenser

Condenser body - brass tube. 2.375 ins. diameter x 0.033 ins wall thickness.

Steam-air inlet pipe at entrance to condenser, 2.145 ins. diameter.
(area, 3.62 sq.ins., 0.02515 sq.ft.)

Water orifice plates,

- a) 0.050 ins. width x 2.145 ins. inside diameter
(area, 0.345 sq. ins., 0.02397 sq.ft.)
- b) 0.025 ins. width x 2.145 ins. inside diameter
(area, 0.1705 sq.ins., 0.01184 sq.ft.)

Header section length, 3.75 ins.

Header cooling water annulus width, 0.35 ins.

Vertical location of cooling water sampling scoop and traversing thermocouple:-

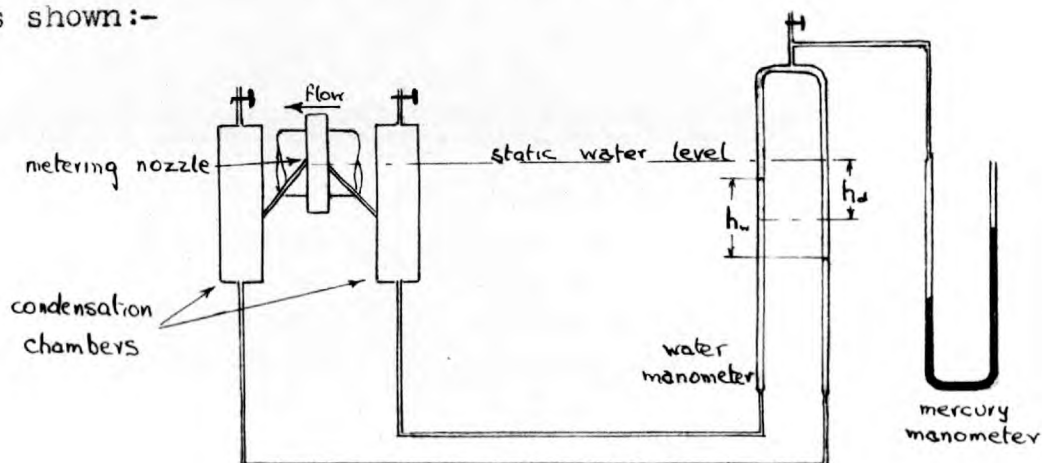
Hatchway Number	Distance from condenser entry (ins.)	
	0.050 ins. orifice plate	0.025 ins. orifice plate
1	1.425	1.460
2	2.400	2.435
3	3.865	3.900
4	5.620	5.655
5	7.620	7.655
6	10.630	10.665
7	13.620	13.655

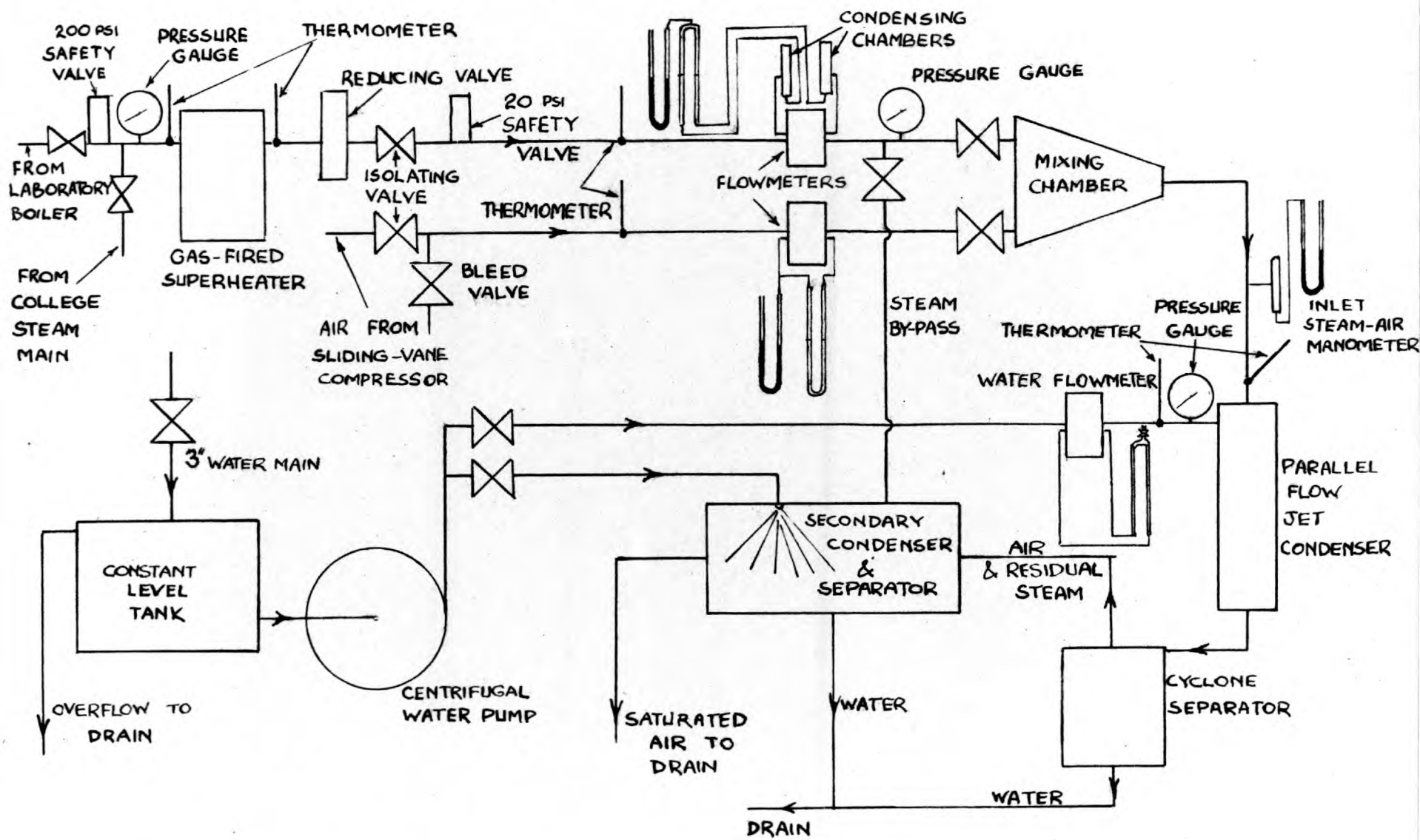
II. ANCILLARY EQUIPMENT

A diagrammatic sketch of the various services is shown in fig. 5. Descriptions of each are now given under their respective headings.

Steam, air and water metering. The flow meters were designed according to B.S.S. 1042 "Flow measurement" and fitted in 2" diameter pipes, each meter being provided with the stipulated lengths of straight piping on the upstream and downstream sections.

a) Steam flow The steam flow rate was measured by a nozzle type meter, 2" x 1.149". The pressure tappings were from Piezo rings to steam condensation chambers and then to an inverted water manometer, which measured the metering head. The metering pressure was tapped from the upper end of the water manometer as shown:-



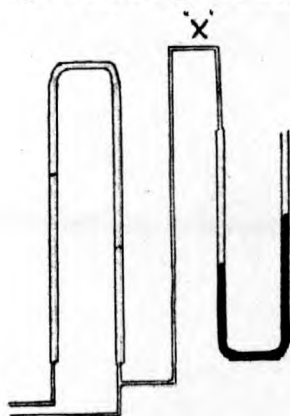


DIAGRAMMATIC ARRANGEMENT OF
COMPLETE APPARATUS

The true upstream pressure is given by:-

Upstream (ins.Hg.) = P_{observed} (ins.Hg.) + $\frac{hw.}{2 \times 13.6} - \frac{hd.}{13.6}$
 (allowing for whether the position of the mean level in the water manometer is above or below the static level.)

Initially, the mercury manometer was tapped directly to the upstream pressure line but the possibility of air bubbles in



the vicinity of "X", dictated the method shown over. The metering of the nozzle was checked by fitting it in the air supply line and comparing with a pitot traverse in another section of the line.

The percentage difference was as follows:-

Mean bulk velocity in 2" pipe (ft/sec)		% difference
By air meter	By pitot traverse	
172	171.1	+0.5
132.5	134.0	-1.1
93.7	92.3	+1.5
168	170	-1.2
122.5	122.5	0

} With no gauzes
 } in bend prior
 } to pitot tube
 }
 } With suitable
 } gauzes in bend
 } prior to pitot
 } tube to correct
 } velocity profile

The accuracy of the meter was thus established and a steam flow chart was calculated using the B.S.S. flow characteristics for steam nozzles.

b) Air flow To cover the wider range of flow rates required, compared with the steam, 2 interchangeable meters were used. The first was a facsimile of the steam nozzle, and for low air flow rates a sharp-edged orifice was made, 2" x 0.4485", corresponding to the highest area contraction ratio, 0.05 : 1, which B.S.S. permits. The nozzle flow meter was checked in the same manner as the steam meter and found to be within an average accuracy of $\frac{1}{2}\%$ over the same range of flow. The orifice plate could not be checked by this technique as the air velocities in the 2" pipe were too low for accurate pitot measurement, when within the range of the meter.

c) Water flow The suitably proportioned venturimeter was connected to an inverted water manometer fitted with an air valve and purging tap at its upper end. The flow curve was calculated from the B.S.S. data and the curve was verified by weighing tank and stop-watch.

System of steam, air and water supply. The general arrangement is shown in fig. 5.

a) Steam supply The high pressure boiler steam was reduced to about 6 - 8 psi gauge across the Forster-type reducing valve, further reduction being achieved after the flow-meter. Although the main reducing valve was automatically compensated for variations in upstream pressure, this was not very effective for fluctuations of the order of 2 - 3 psi and constant recourse had to be made to adjust the stop-valve on the boiler main. Steam flow rates of less than 250 lb/hr. were usually taken from the laboratory boiler but above this figure the College boilers had to be used. A steam by-pass was fitted across the jet condenser, to allow adjustments to the latter to be made without shutting down the steam; it also allowed the steam supply pipe to be warmed up while any necessary adjustments to the condenser were being made before running commenced and was a very useful expedient in saving time.

b) Air supply A sliding-vane compressor delivered air through a filter, passed a bleed valve to the metering pipe and mixing chamber. The air supply remained substantially constant after adjusting the bleed-valve

and allowing conditions to settle for 30 - 40 minutes and only minor adjustments of the bleed were required to correct the flow during the subsequent run.

c) Water supply The centrifugal pump, fed from a constant head tank, held the flow rate constant indefinitely, as the pump was for most of the time running nearly closed (maximum capacity - 42,000 lb/hr., maximum water flow rate used - 7,700 lb/hr.). A 120 mesh gauze was fitted in the water line to ensure that the condenser water jet orifice was not fouled.

The steam state at the condenser inlet was maintained at the saturated condition throughout, as the work was confined to condensation as distinct from desuperheating and condensation of steam. Provision had to be made for initially superheating the steam to allow for the heat lost when mixing with air at about 20°C. For ease of adjustment, heating of the steam is best carried out during the mixing process with air as this offers a quick response to control of the mixture temperature. However, the only available means of heating was a standard gas-fired steam superheater with $\frac{3}{4}$ " piping, by which the degree of high-pressure steam superheat was adjusted until the temperature of the low pressure steam-air mixture in

the condenser entry pipe, indicated about one degree of superheat. This system suffered from a slow response (about 30 minutes) in the steam-air mixture, to adjustment of the superheater gas supply.

Water separators and secondary condenser. It was appreciated that steam with premixed air in the quantities required, could not be completely condensed in the length of condenser possible on the available site and provision had to be made for the condensation of effluent steam from the working section. The uncondensed steam and air were first separated from the cooling water in a suitable cyclone and then passed to a secondary spray-condensing box using a $\frac{3}{8}$ " diameter spray nozzle at 50 psi water pressure. This condenser was able to cope with uncondensed steam over the conditions of the investigation. The saturated air from the secondary condenser passed through a side exit and the cooling water and condensate was run to drain together with the water from the cyclone separator.

III. MODE OF RIG OPERATION

Using steam without premixed air. The cooling water system was first started and the venturimeter manometer lines were thoroughly purged of air by opening the bleedcock at the upper end of the inverted water manometer when the water pressure in the venturimeter was about 3 - 5 psi. With the bleedcock closed, the manometer water levels were adjusted by introducing air into the upper end of the manometer through a valve adjoining the bleedcock. The circumferential flow distribution of the water jet was now checked and adjusted with the sampling scoop in No. 1 hatchway (page 9). Steam was introduced into the condenser after the flowmeter condensation chambers had been fully primed with water. When the steam supply line had been pressurised to 3 - 5 psi the inverted water manometer levels were adjusted in the same manner as those for the cooling water system. The superheat of the steam was then adjusted until it had a value of about 1°C in the condenser inlet pipe (page 22).

When conditions were steady a temperature traverse was made in two or three hatchways in the lower part of the condenser to determine the extent of the region of pure steam. This was necessary since the later stages

of steam condensation occurred in a mixture with air, the latter being present as the working section was run at atmospheric pressure and the absence of steam necessitated the presence of air. Bulk temperature readings were then taken on the cooling water from the scoop, by the procedure already described (page 10), in six circumferential positions at each hatchway in the zone of pure steam condensation. The time required to complete a run at one condition was made lengthy (6 - 9 hours) by the initial settling time and the 32 screwing operations required at each change of hatchway position for the thermocouple probe or scoop as 8 studs were necessary to ensure an adequately tight joint.

Using a mixture of steam and air. The operation of the cooling water and steam systems was the same as when condensing steam only. The most suitable air metering orifice was selected for the desired air content in the steam and the bleed valve between the compressor and the metering line was adjusted to give the correct air flow in the condenser. The superheat of the steam was then adjusted until it was about 1°C in the condenser inlet pipe.

As the steam-air phase had a finite velocity throughout the condensation process, there was no trouble due to the ingress of external air from the downstream end of the condenser, as occurred when condensing pure steam. Cooling water bulk temperature readings were taken at 6 different circumferential positions at each of 6 hatchways. Temperature traverses were made in the proximity of the water phase at 4 circumferential positions in each of 3 hatchways (Nos. 2, 4 and 6) as the time available for each day's running did not permit radial temperature traverses in every hatchway.

IV. THEORETICAL CONSIDERATIONS AND EXPERIMENTAL CORRELATION

Preliminary. The physical problem in this investigation concerns the diffusion of steam through air from a turbulent mixture stream to the surface of a thin sheet of water flowing in a direction parallel to the steam-air mixture. From the theoretical aspect this mechanism is best considered as a problem of heat transfer into a thin slab of infinite extent exposed on one side to a temperature which decreases as some function of time. No exact solution has been found for the relation between interfacial temperature and time of water exposure to the steam-air mixture. Thus it has not been possible to compare directly the actual heat transfer rates obtained by experiment with calculated figures based on the molecular diffusion of heat in the cooling water phase, as the true driving force in the water is not known.

As a first step in developing a satisfactory method of correlating the results it was necessary to determine the increase in heat transfer rate associated with turbulent flow over that expected under conditions of molecular diffusion of heat and use was made of the constant

surface temperature condensation obtained with pure steam. It will be shown later that a condensing steam-air mixture exhibits the same turbulence effect on the condensation rate as pure steam.

Condensation of steam onto a thin moving lamina of water. The annular water jet can with negligible error be considered as a thin flat moving lamina of infinite extent as the curvature is small (the minimum diameter/thickness ratio was 44 : 1). The heat transfer, associated with the condensation of steam within a given distance of the orifice, to the thin moving lamina or slab, is the same problem as the heat transfer to a static slab suddenly exposed to the steam saturation temperature* for a time τ , corresponding to the time of flow of the moving lamina. Neglecting conduction from the water surface remote from the steam, the slab may be considered to be of thickness $2l$ and exposed on both sides to a temperature, θ_s . If $-l < x < l$, the temperature at any point in the slab is given by:-

$$\theta = \theta_s - \frac{4\theta_s}{\pi} \sum_{n=0}^{\infty} \frac{(-1)^n}{2n+1} \cdot e^{-\frac{\alpha_w(2n+1)^2 \pi^2 \tau}{4l^2}} \cdot \cos\left(\frac{(2n+1)\pi x}{l}\right)$$

* ...ref.(2)
The surface coefficient of heat transfer is sufficiently high to neglect the temperature drop between the steam and the water face - ref. (1).

The bulk or mean temperature is

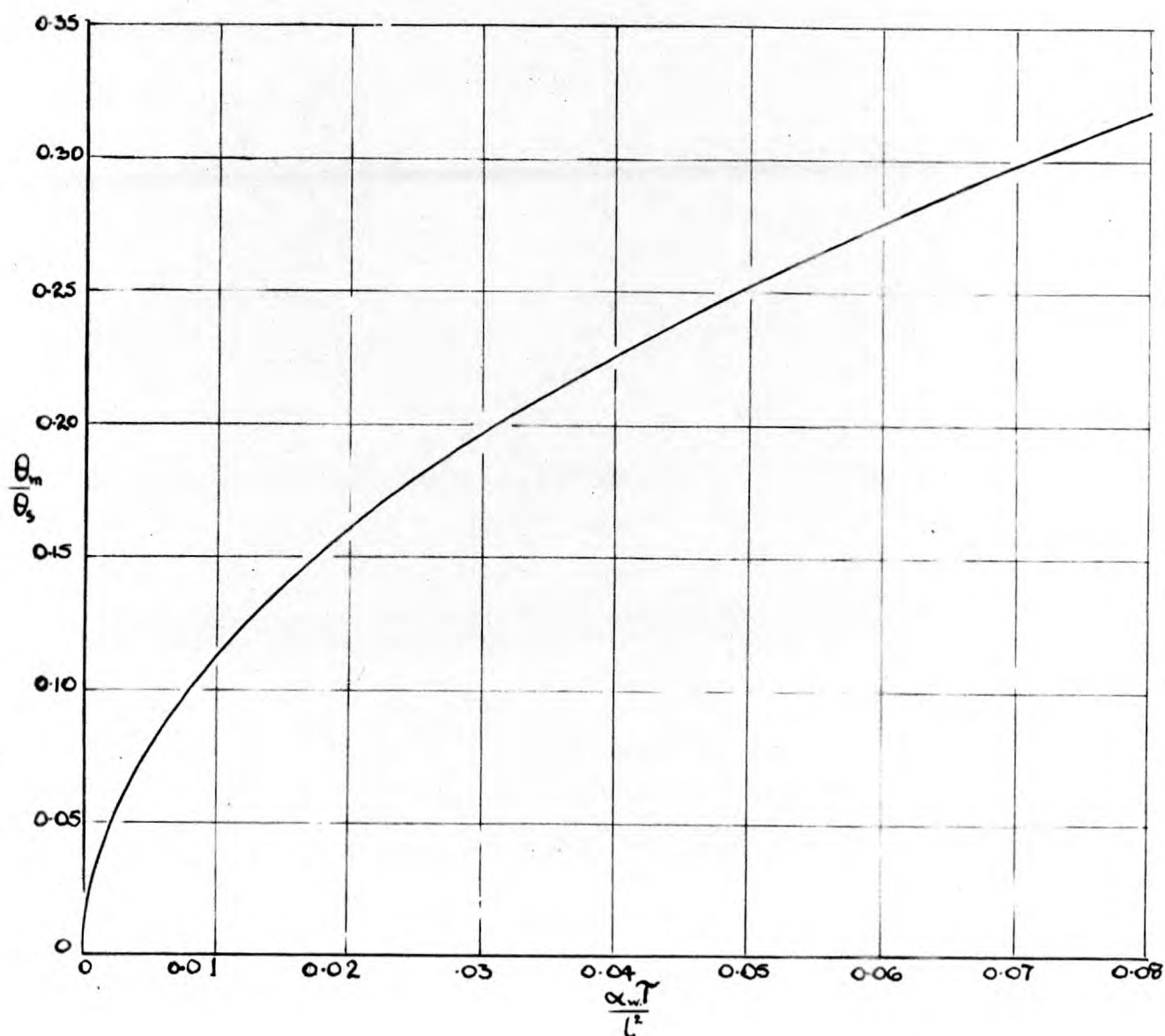
$$\theta_m = \frac{1}{2l} \int_{-l}^{+l} \theta \cdot dl = \theta_s - \frac{8\theta_s}{\pi^2} \sum_{n=0}^{\infty} \frac{1}{(2n+1)^2} \cdot e^{-\frac{\alpha_w (2n+1)^2 \pi^2 \tau}{4l^2}} \dots\dots(1)$$

For a slab of thickness l , exposed on one side to a temperature, θ_s , and insulated on the opposite side, the bulk temperature is given by the same equation. Fig. 6 shows a curve of $\frac{\theta_m}{\theta_s}$ vs. $\frac{\alpha_w \tau}{l^2}$.

Determination of turbulence factor when condensing

steam only. From fig. 6 the cooling water bulk temperature can be predicted providing τ and l are known. The time of exposure τ was calculated assuming the acceleration of the jet was only influenced by gravity. Drag on the jet due to wall friction and interfacial shear with the steam were not allowed for, the net effect of these two forces being incorporated in the turbulence factor mentioned hereafter. Using fig. 6 and actual temperature rises, a direct comparison can be made between the molecular and effective thermal diffusivities in the cooling water jet for different steam and water rates, or alternatively, the theoretical and actual temperature rises

MEAN TEMPERATURE, θ_m , OF A THIN LAMINA
WITH CONSTANT SURFACE TEMPERATURE θ_s
& INITIAL TEMPERATURE ZERO.



can be compared. An attempt was made to correlate the actual results with theory on a basis of diffusivity ratios but the scatter in the results was prohibitive.

From results obtained with varying steam inlet velocity at constant water rate (fig. 7) it was seen that the percentage of steam condensed was independent of the steam inlet velocity, being only a function of the condenser length (the percentage of steam condensed/unit volume was independent of the steam inlet velocity). More correctly the dimensionless time of cooling water exposure, $\frac{\alpha_w \tau}{l^2}$, instead of condenser length, is the basic parameter of transient heat flow and it follows that the percentage of steam condensed is a function of $\frac{\alpha_w \tau}{l^2}$.

$$\text{i.e. } \frac{W_c}{W_{so}} = f_1 \left(\frac{\alpha_w \tau}{l^2} \right)$$

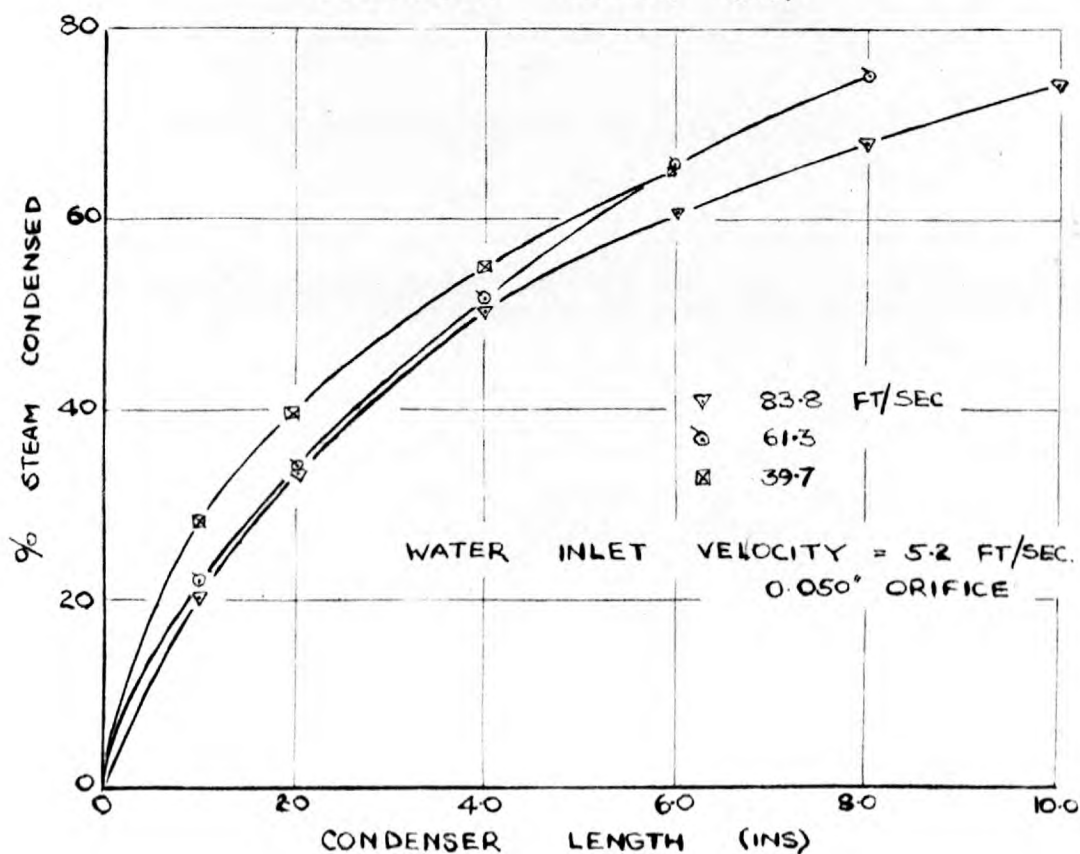
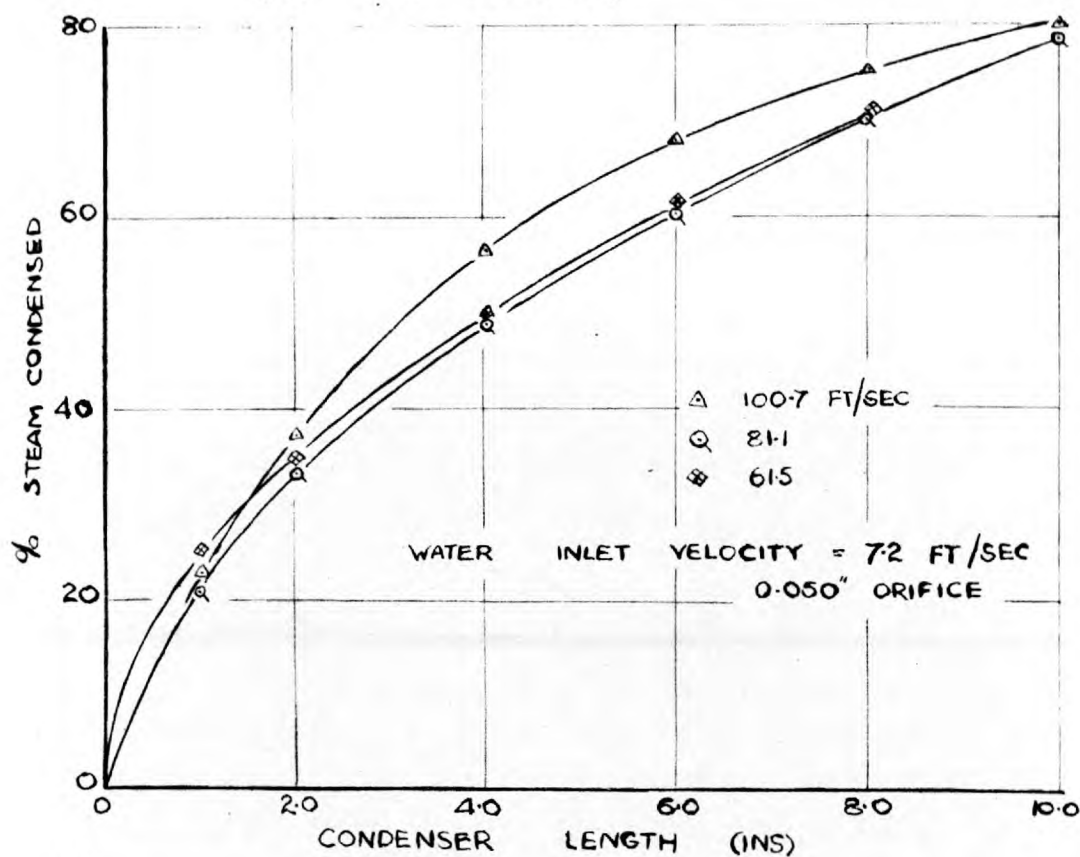
$$\text{Now } W_c H_s = W_w \theta'_m$$

$$\therefore \frac{W_w \theta'_m}{H_s W_{so}} = f_2 \left(\frac{\alpha_w \tau}{l^2} \right)$$

With constant pressure condensation, H_s is a function of θ_s

$$\left(\frac{\theta_m}{\theta_s} \right) \left(\frac{W_w}{W_{so}} \right) = f_3 \left(\frac{\alpha_w \tau}{l^2} \right)$$

EFFECT OF STEAM INLET VELOCITY
ON STEAM CONDENSED.
(NO AIR PRESENT)



Combining with equation (1)

$$\frac{\theta'_m}{\theta_m} = \frac{W_{so}}{W_w} \cdot f_4\left(\frac{\alpha_w \gamma}{l^2}\right)$$

The ratio $\frac{\theta'_m}{\theta_m}$ will be denoted by 'r'

$$\therefore \frac{r}{W_{so}} = \frac{1}{W_w} \cdot f_4\left(\frac{\alpha_w \gamma}{l^2}\right) \quad \dots\dots(2)$$

On plotting the results in the manner suggested by equation (2), figs. 8 and 9, it was found that the turbulence effect in the water was represented by an exponent of 0.17 and not unity as indicated by the equation. It will be noticed that the graphs are plotted using steam and water velocities and not flow rates as is also suggested by equation (2). This is explained as follows:- in the experimental work using steam without premixed air, only the 1 orifice, 0.050" width, was found capable of producing stable water jets as jets from smaller orifices invariably collapsed onto the steam core. The experimental cooling water rates used were thus always the same function of the water inlet velocity and with a fixed duct diameter the steam flow rate was, to the first order, a fixed function of the steam velocity. The turbulence effects of mass flow and velocity of the two phases could not, therefore, be separated in the results obtained. When analysing later

results using steam-air mixtures with different water orifices it was found that the cooling water turbulence correction factor was better expressed in terms of velocity and for this reason the steam and water rates in figs. 8 and 9 are presented in these units. The interpretation of figs. 9b and 9c is dealt with later (page 54).

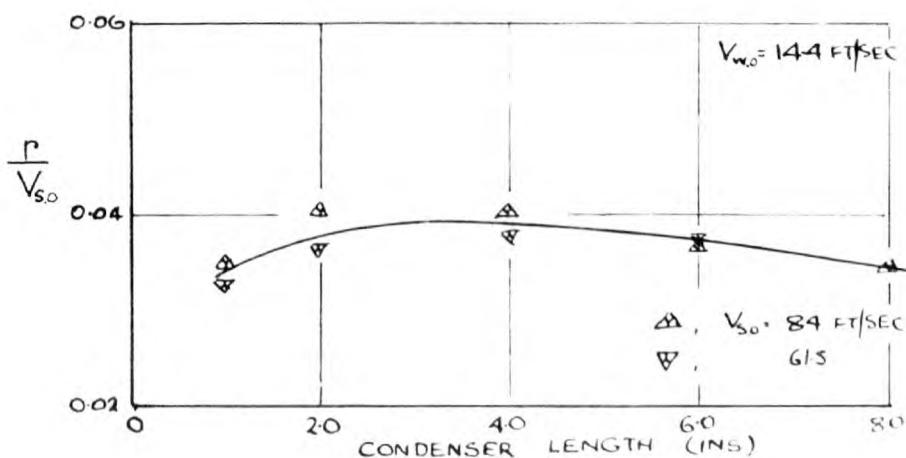
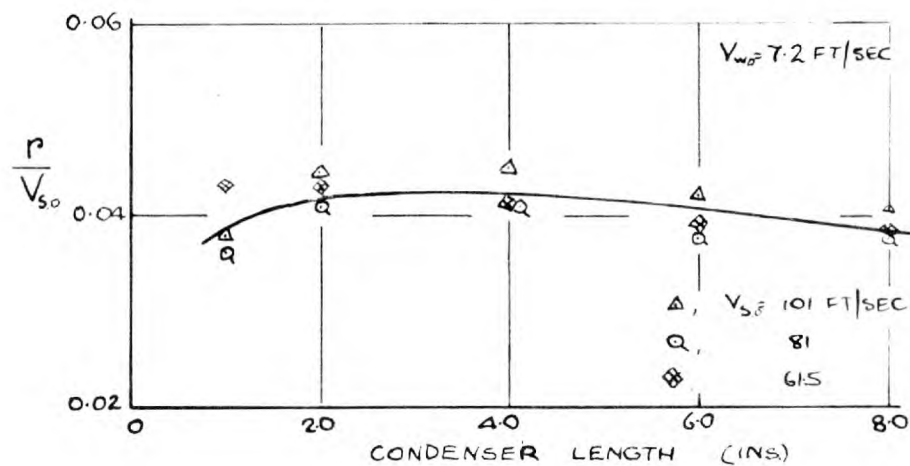
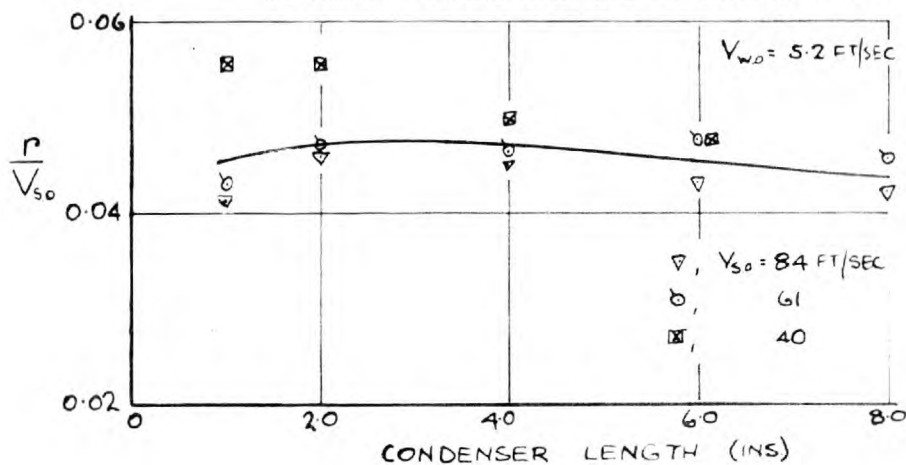
Variation of the water surface temperature when
condensing steam in the presence of air. The

condensation of steam when diffusing through air, very quickly lowers the water surface temperature due to the build-up of the air content in the steam-air mixture at the surface, the temperature of which for all practical purposes, corresponds to the partial pressure of the steam. The effect of the air is twofold; firstly, the creation of the diffusive resistance to the steam in the steam-air phase and secondly a decrease in the heat transfer driving force in the cooling water phase. Some knowledge of the behaviour of the water surface temperature was required in order to develop a means of comparison between experimental data and to determine the eddy diffusive resistance within the steam-air core. No direct theoretical analysis has been found possible

VARIATION OF $\frac{r}{V_{s0}}$ ALONG CONDENSER

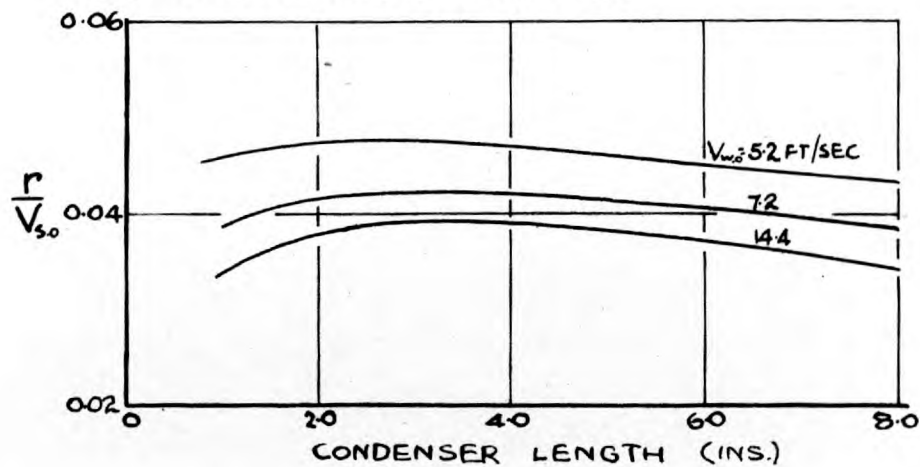
(NO AIR PRESENT)

$$r = \frac{\text{ACTUAL TEMPERATURE RISE OF COOLING WATER}}{\text{THEORETICAL TEMPERATURE RISE OF COOLING WATER}}$$

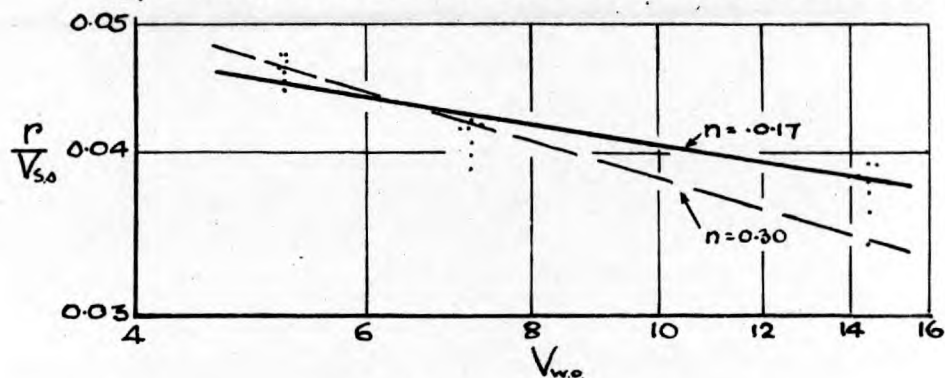
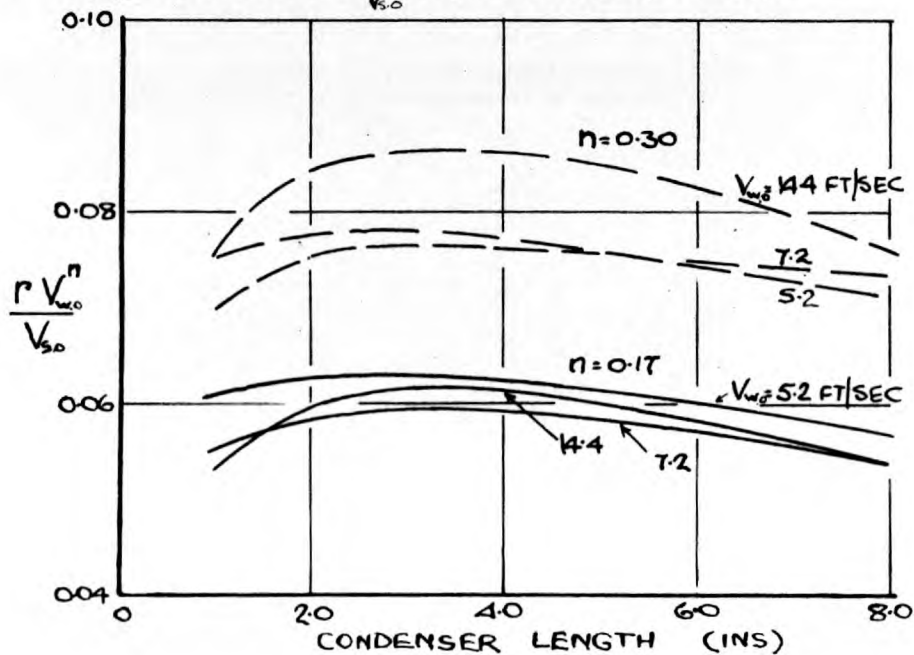


EFFECT OF WATER INLET VELOCITY, V_{w0} , ON r

(a) — AVERAGE CURVES FROM FIG. 8



(b) — LOG LOG CROSS-PLOT OF "a" AT 1' INTERVALS OF CONDENSER LENGTH

(c) — VARIATION OF $\frac{r V_{w0}^n}{V_{s0}}$ WITH CONDENSER LENGTH

for estimating the surface temperature and it is tentatively suggested that:-

$$\theta_{surf} = \theta_m + (\theta_s - \theta_m) e^{-k\tau} \quad \dots\dots\dots (3)$$

where k is some constant (Appendix 1)

This equation satisfied the two boundary conditions, viz:-

- (i) when $\tau = 0$ $\theta_{surf} = \theta_s$
- (ii) when $\tau = \infty$ $\theta_{surf} = \theta_m$

The mean temperature θ_m of a thin lamina of infinite extent with zero initial temperature and suddenly exposed to a medium of temperature $\theta_m + (\theta_s - \theta_m) e^{-k\tau}$ on one side, no heat flow occurring through the opposite side.

The lamina may be considered to be a thickness 2 and exposed to the medium on both faces.

The solution to the general case of a slab exposed to medium at temperature $f(\tau)$ is, ref (3):

$$\theta = 2 \sum_{n=0}^{\infty} e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4L^2}} \cdot \cos(2n+1) \frac{\pi x}{2L} \left\{ \frac{(2n+1) \pi \alpha_w (-1)^n}{2L} \int_0^{\tau} e^{-\frac{\alpha_w (2n+1)^2 \pi^2 \lambda}{4L^2}} \cdot f(\lambda) d\lambda \dots\dots\dots (4) \right.$$

$$\text{Now } f(\lambda) = \theta_m + (\theta_s - \theta_m) e^{-k\lambda}$$

Substituting in (4) and integrating:-

$$\begin{aligned} \theta = & \theta_m \left(1 - \frac{4}{\pi} \sum_{n=0}^{\infty} \frac{(-1)^n}{2n+1} \cdot e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4L^2}} \cdot \cos(2n+1) \frac{\pi x}{2L} \right. \\ & \left. + (\theta_s - \theta_m) 4\pi \sum_{n=0}^{\infty} \frac{(-1)^n (2n+1)}{\pi^2 (2n+1)^2 - 4kL^2} \left(e^{-k\tau} - e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4L^2}} \right) \cos(2n+1) \frac{\pi x}{2L} \dots\dots\dots (5) \right) \end{aligned}$$

Equation (5) can be conveniently written as:

$$\theta = \theta_1 + \theta_2$$

$$\text{And } \theta_m = \frac{1}{2L} \int_{-L}^{+L} \theta \cdot dx = \frac{1}{2L} \int_{-L}^{+L} \theta_1 \cdot dx + \frac{1}{2L} \int_{-L}^{+L} \theta_2 \cdot dx$$

$$\therefore \theta_m = \theta_{m1} + \theta_{m2} \quad \dots\dots\dots(6)$$

$$\text{where } \theta_{m1} = \frac{\theta_m}{2L} \int_{-L}^{+L} \left(1 - \frac{4}{\pi} \sum_{n=0}^{\infty} \frac{(-1)^n}{(2n+1)} \cdot e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4L^2}} \cdot \cos(2n+1) \frac{\pi x}{2L} \cdot dx\right.$$

$$\left. \therefore \theta_{m1} = \theta_m \left(1 - \frac{8}{\pi^2} \sum_{n=0}^{\infty} \frac{1}{(2n+1)^2} \cdot e^{-\frac{\alpha_w (2n+1)^2 \pi^2 \tau}{4L^2}}\right) \quad \dots\dots\dots(7)\right.$$

Equation (7) gives the mean temperature of the slab when exposed to a constant surface temperature, θ_m (cf. equation (1) and fig. 6)

$$\theta_{m2} = \frac{\theta_s - \theta_m}{2L} \int_{-L}^{+L} 4\pi \sum_{n=0}^{\infty} \frac{(-1)^n (2n+1)}{\pi^2 (2n+1)^2 - \frac{4kL^2}{\alpha_w}} \left(e^{-k\tau} - e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4L^2}}\right) \cos(2n+1) \frac{\pi x}{2L}$$

$$\therefore \theta_{m2} = 8(\theta_s - \theta_m) e^{-k\tau} \sum_{n=0}^{\infty} \frac{1}{(2n+1)^2 \pi^2 - \frac{4kL^2}{\alpha_w}}$$

$$- 8(\theta_s - \theta_m) \sum_{n=0}^{\infty} \frac{1}{(2n+1)^2 \pi^2 - \frac{4kL^2}{\alpha_w}} \cdot e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4L^2}}$$

$$\therefore \theta_{m2} = (\theta_s - \theta_m) \sqrt{\frac{\alpha_w}{kL^2}} \cdot \tan \sqrt{\frac{kL^2}{\alpha_w}} \cdot e^{-\left(\frac{kL^2}{\alpha_w}\right) \left(\frac{\alpha_w}{L^2}\right) \tau}$$

$$- 8(\theta_s - \theta_m) \sum_{n=0}^{\infty} \frac{1}{(2n+1)^2 \pi^2 - \frac{4kL^2}{\alpha_w}} \cdot e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4L^2}} \quad \dots\dots\dots(8)$$

Equation (8) gives the mean temperature of the slab, when exposed to a surface temperature $(\theta_s - \theta_m)e^{-k\tau}$.

The mean temperature of the slab when exposed to a surface temperature $\theta_m + (\theta_s - \theta_m)e^{-k\tau}$ is given by

$$\theta_m = \theta_{m1} + \theta_{m2} \quad \text{.....(6)}$$

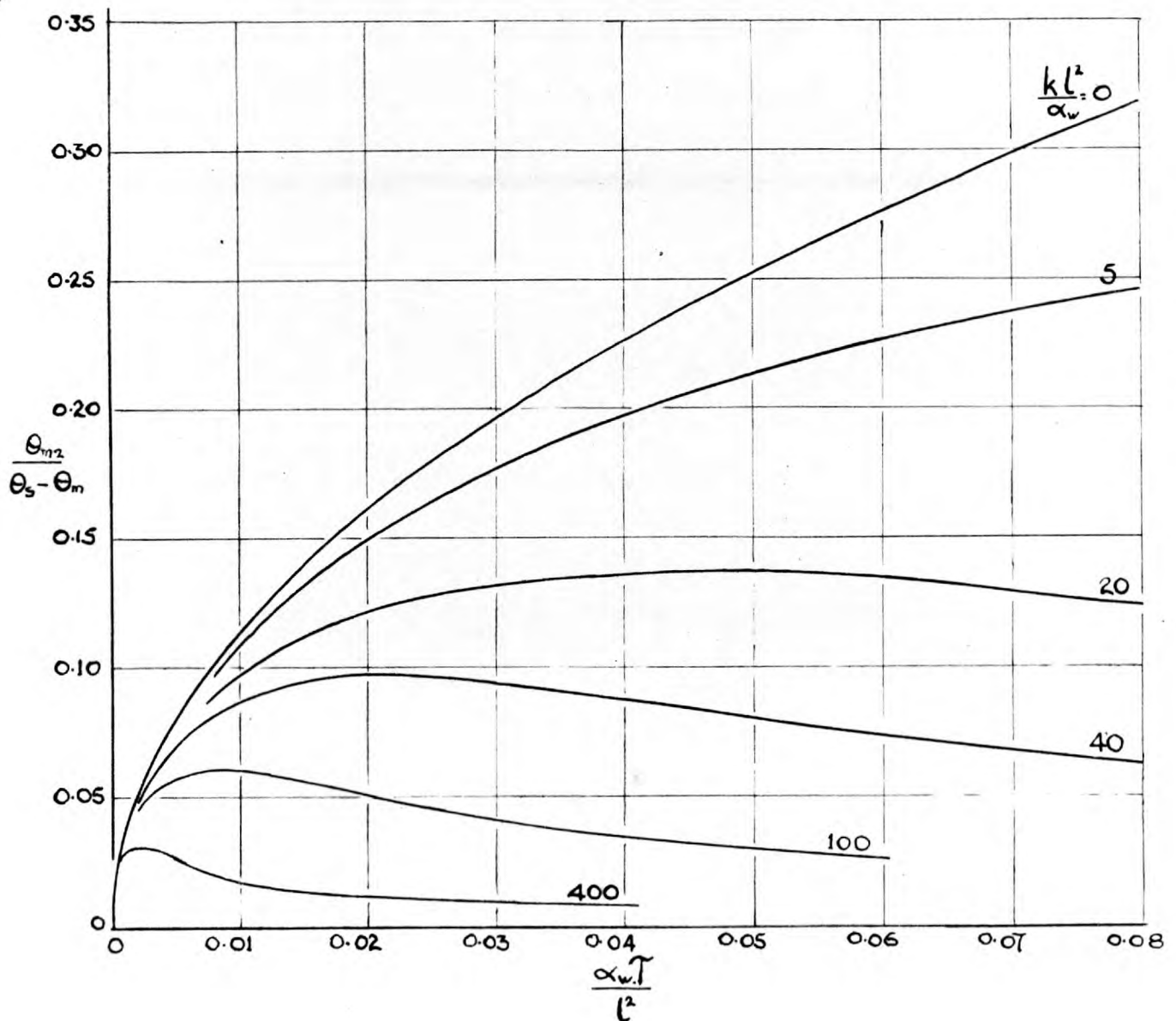
Curves of $\frac{\theta_{m2}}{\theta_s - \theta_m}$ vs $\frac{\alpha_w \tau}{L^2}$ have been computed from equation (8) for various values of $\frac{kl^2}{\alpha_w}$, (fig. 10). It will be noted that the curve, $\frac{kl^2}{\alpha_w} = 0$, is identical with fig. 6.

Justification for use of turbulence factor determined from the condensation of pure steam, for condensation from a steam-air mixture. Curves of percentage of steam condensed vs. condenser length at constant initial air content, are shown in fig. 11. With the exception of the run 22 (entry velocity of 42 ft/sec), these curves indicate the same independence of "gas" velocity as the results obtained using steam without premixed air, fig. 7. It is seen later (fig. 22) that the above mentioned run shows a substantially higher cooling water temperature rise than should be expected and this is accounted to erroneous temperature measurements on the cooling water. Neglecting this run, the curves in fig. 11 justified the use for steam-air mixtures of the turbulence factor "r", obtained by the

VARIATION OF $\frac{\theta_{m2}}{\theta_s - \theta_m}$ IN A THIN SLAB FOR
DIFFERENT VALUES OF $\frac{kl^2}{\alpha_w}$.

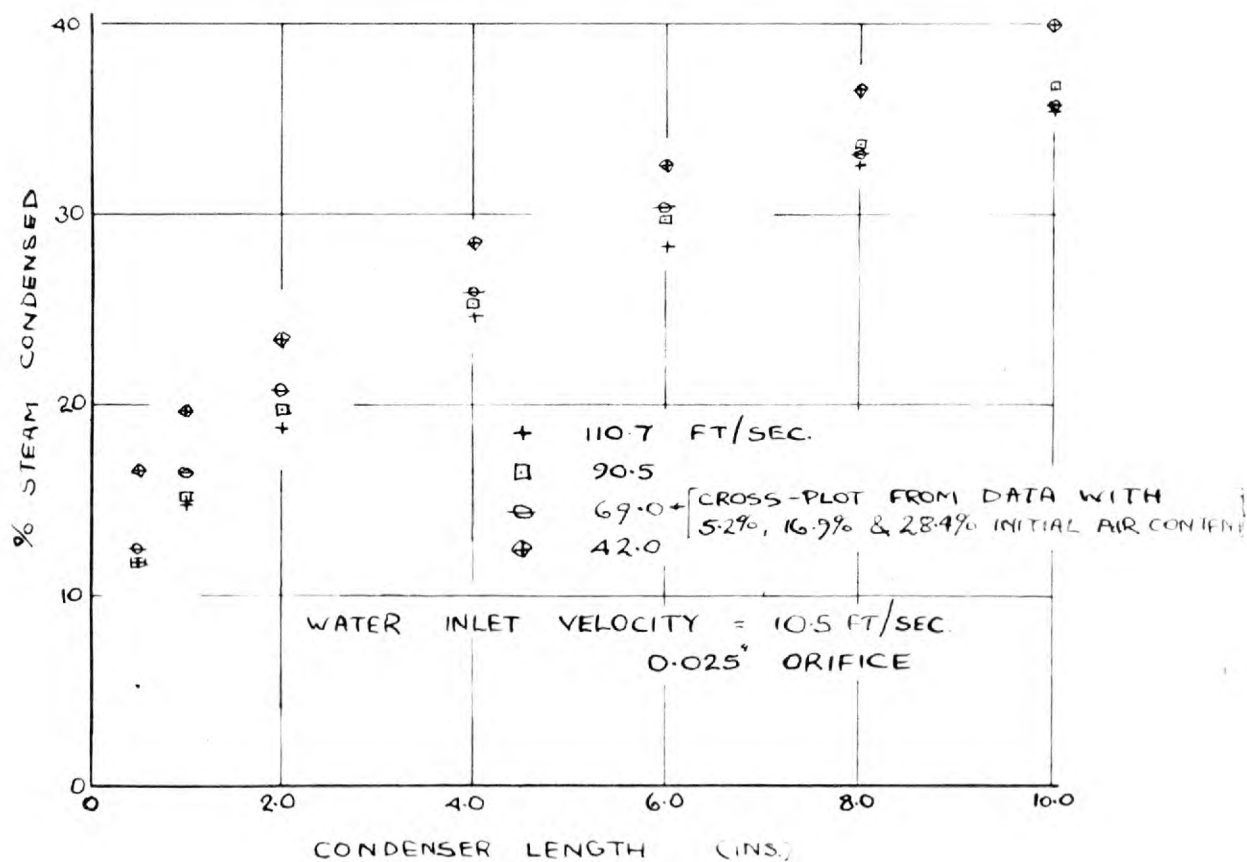
$$\theta_{surf} = \theta_m + (\theta_s - \theta_m) e^{-kT}$$

$$\theta_{m2} = \theta_m - \theta_{m1} \quad \text{WHERE} \quad \frac{\theta_{m1}}{\theta_m} = f\left(\frac{\alpha_w T}{l^2}\right), \text{ FIG 4.}$$



EFFECT OF STEAM-AIR INLET VELOCITY ON STEAM CONDENSATION

[10.4% INITIAL AIR CONTENT]



condensation of steam without premixed air, the inlet velocity of the "gas" now being that of the steam-air mixture. The effect of turbulence on steam-air condensation ratio is incorporated in the tentative theoretical relationship as follows:-

$$\theta'_m = r \theta_m \quad - \text{by definition.}$$

Combining with equation (8):

$$\theta'_{m2} = (r \theta_s - \theta'_m) \left[\sqrt{\frac{\alpha_w}{kl^2}} \cdot \tan \sqrt{\frac{kl^2}{\alpha_w}} \cdot e^{-\left(\frac{kl^2}{\alpha_w}\right) \left(\frac{x_w}{L}\right)} - 8 \sum_{n=0}^{\infty} \frac{1}{(2n+1)^2 \pi^2 - \frac{4kl^2}{\alpha_w}} \cdot e^{-\frac{\alpha_w \pi^2 (2n+1)^2}{4L^2} \tau} \right] \dots (9)$$

i.e. $\frac{\theta'_{m2}}{r \theta_s - \theta'_m}$ vs. $\frac{\alpha_w \tau}{L^2}$ for different values of $\frac{kl^2}{\alpha_w}$,

is the same relation as plotted in fig. 10. It will be seen later that these dimensionless parameters form a very convenient method of presenting the experimental data when condensing steam with premixed air, despite the fact that the actual results do not follow the trend of the theoretical curves, i.e. the cooling water surface temperature decay function does not truly represent the actual surface temperature variation.

V. EXPERIMENTAL RESULTS

The main experimental work was completed in two stages:-

(i) Condensation of steam without premixed air.

(ii) Condensation of steam with premixed air.

(i) Data at 8 conditions are presented for this part, these being the best results of 27 runs. These figures are indicative of the difficulty of obtaining reproducibility in the results during the earlier stages of the work; only when the final sampling technique had been developed was there any consistency in the bulk temperature readings of the cooling water. The conditions covered are summarised in the following table:-

Run No.	Symbol	Steam Flow (lb/hr) Allowing for Condensation in Header	Steam Inlet Velocity (lb/hr) Allowing for Condensation in Header	Cooling Water Flow Rate (lb/hr)	Cooling Water Entrance Velocity (ft/sec)	Initial Thickness of Cooling Water Jet (Ins.)
1	▽	287.2	84	2780	5.2	0.050
2	⊙	210.7	61	2780	5.2	0.050
3	⊠	135.7	40	2780	5.2	0.050
4	△	340.5	101	3840	7.2	0.050
5	⊙	278.5	81	3840	7.2	0.050
6	◇	211.5	61.5	3840	7.2	0.050
7	⊠	287	84	7700	14.4	0.050
8	▽	211	61.5	7700	14.4	0.050

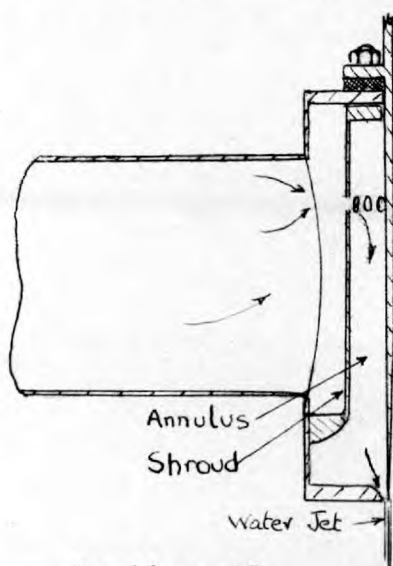
This data was solely obtained to estimate the effect of turbulence on the temperature rise in the cooling water. A wider range of results was desirable but stable conditions were not found possible at steam inlet velocities greater than about 100 ft/sec. and with a jet orifice width of less than 0.050". The symptoms of instability were violent manometer fluctuations on the pressure tappings, absence of cooling water in the scoop drain due to the collapse of the water jet, and the flooding of the main steam line by the "sucking back" of the cooling water.

(ii) The operating procedure in using the sampling scoop, had been finalised before runs were attempted with premixed air. This, together with the inherent stability of the steam-air mixture and cooling water, accounted for a greater degree of consistency in the results compared with runs using pure steam. 20 runs were completed at varying conditions with steam-air mixtures and of these 15 are presented in this thesis, the conditions being summarised as follows:-

Run No.	Symbol on Graphs	Initial % Air Content by Weight	Steam Flow (lb/hr)	Steam-Air Inlet Velocity (ft/sec)	Cooling Water Flow Rate (lb/hr)	Cooling Water Entrance Velocity (ft/sec)	Initial Thickness of Cooling Water Jet (ins)
9	◇	10.2	346	109.7	2780	5.2	0.050
10	⊙	10.2	353	111.0	5660	10.5	0.050
11	+	10.2	345	110.7	2780	10.5	0.025
12	×	10.3	344	108.0	5660	21.3	0.025
13	⊞	10.3	283	90.5	2780	10.5	0.025
14	⊗	16.9	213	69.4	2780	10.5	0.025
15	⊕	28.5	213	77.5	2780	10.5	0.025
16	Points not shown on fig. 23 for clarity	9.2	209	65.6	3840	14.5	0.025
17		19.9	212	71.6	3840	14.5	0.025
18		45.7	203	89.1	3840	14.5	0.025
19		60.3	206	114.1	3840	14.5	0.025
20		5.2	218	70.5	2780	10.5	0.025
21		5.2	218	70.3	5660	21.3	0.025
22		10.4	135	42.0	2780	10.5	0.025
23		26.7	133	47.2	2780	10.5	0.025

Runs 9 - 15 and 20 - 23 include temperature traverse data in the steam-air phase to enable calculation of the eddy diffusivity in the mixture.

Allowance for condensation in header section when using steam without premixed air. To estimate the heat transfer rate through the header section, the latter was treated as a simple annular chamber containing water flowing at a uniform rate throughout its length. The steam-side surface coefficient is sufficiently large to neglect during



Section of
header annulus

the early stages of condensation, ref (4), and the overall heat transfer coefficient is therefore given by:

$$U = \frac{1}{\frac{b_h}{k_b} + \frac{1}{h_h}}$$

where b_h is the mean thickness of the steam tube.

k_b is the thermal conductivity.

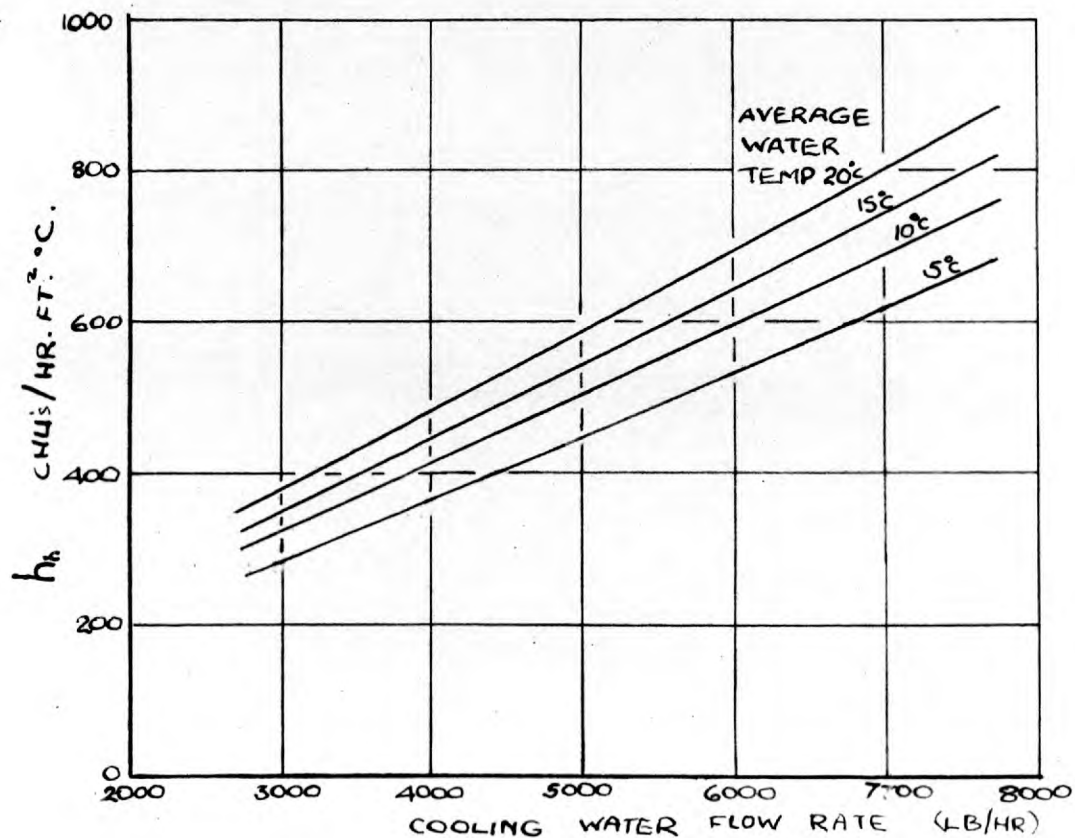
h_h is the annulus water film coefficient.

The annulus water film coefficient has been plotted (fig. 12) in terms of the cooling water flow rate and the mean water temperature from the data of Carpenter, Colburn & Schoenborn, ref. (5). From the diagram above, it is seen that the flow is not truly parallel over the whole extent of the annulus. It was, however, considered justifiable to

ANNULUS WATER FILM COEFFICIENT IN
HEADER SECTION OF CONDENSER

(CARPENTER, COLBURN & SCHOENBORN)

WIDTH OF WATER ORIFICE = 0.050"



treat it as such and estimate the heat transfer rate using a constant annulus water film coefficient calculated for the zone of parallel flow as the higher transfer coefficient associated with the increased water velocity at the surface in the approach to the main orifice, counters the lower coefficient in the region of water stagnation above the shroud orifices.

A balance between the heat transferred through the wall and the heat picked up by the cooling water gives:-

$$W_w \cdot \Delta\theta'_{mh} = U \cdot S_h \left[t_s - \left(t_w + \frac{\Delta\theta'_{mh}}{2} \right) \right]$$

$$\therefore \Delta\theta'_{mh} = \frac{U S_h (t_s - t_w)}{W_w + \frac{U S_h}{2}} \quad \dots\dots(10)$$

The steam condensed in the header section is given by:

$$W_{ch} = \frac{U \cdot S_h}{L} \left[t_s - \left(t_w + \frac{\Delta\theta'_{mh}}{2} \right) \right] \quad \dots\dots(11)$$

The measured cooling water inlet temperature and the steam flow rate to the header were corrected by equations (10) and (11) to give the true conditions at the entry to the jet condenser proper.

Analysis of results. The method of analysis of the experimental data is best illustrated by specimen calculations on one run from each of the two series of tests.

(1) Condensation of steam without premixed air. The following calculations are on data obtained in run 1, to determine the effect of turbulence on the theoretical temperature rise in the cooling water.

Experimental results

Steam flow rate by nozzle	=	295.5 lb/hr.
Condenser pressure	=	30.4 ins.Hg, 14.85 p.s.i.a.
Corresponding steam temperature	=	100.4°C
Cooling water flow rate by venturimeter	=	2780 lb/hr.
Cooling water inlet temperature to header	=	5.2°C
Cooling water orifice width	=	0.050 ins., 0.00417 ft.

A radial temperature traverse at station 6 (10.63 ins. from condenser inlet) indicated that the steam at this position was not contaminated with air from the downstream end of the condenser. Cooling water bulk temperature data taken to station 6, the lowest workable position for the scoop, was, therefore, relevant to the condensation of pure steam.

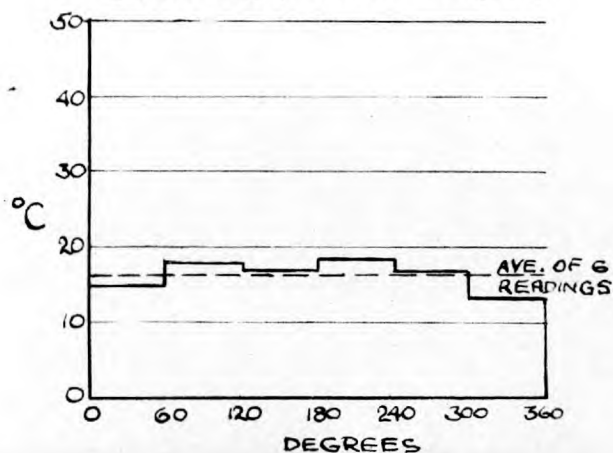
The bulk temperature rise of the cooling water
was :-

CIRCUMFERENTIAL TEMPERATURE RISE DISTRIBUTION OF COOLING WATER JET

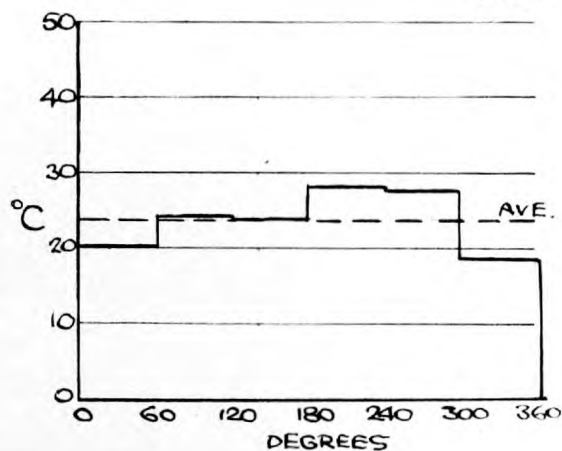
RUN 1.

(NO AIR PRESENT)

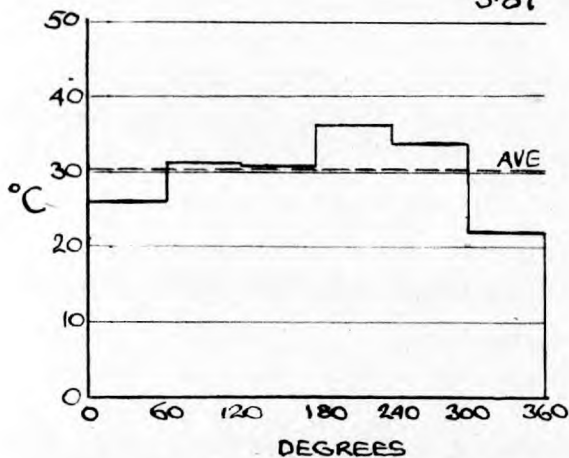
DISTANCE AFTER
CONDENSER INLET = 1.43"



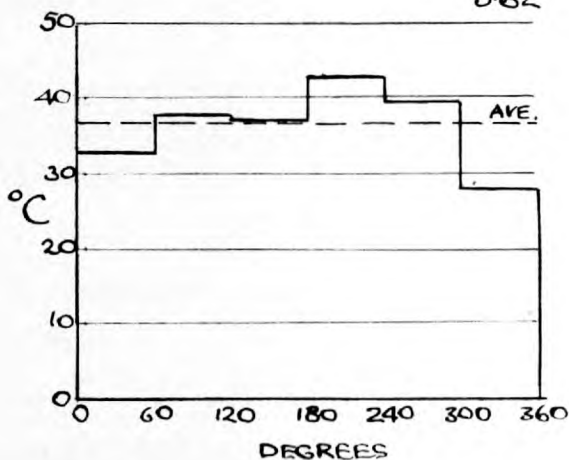
2.40"



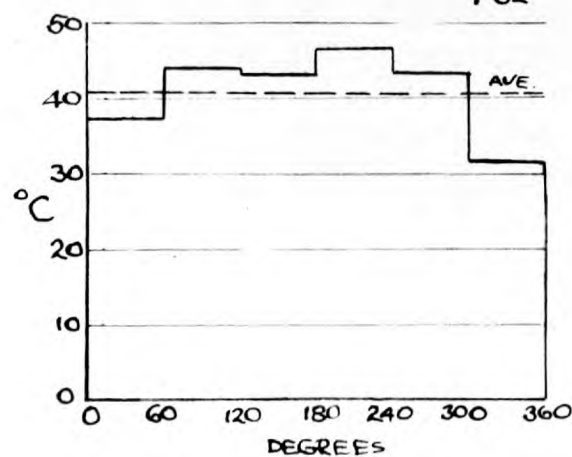
3.87"



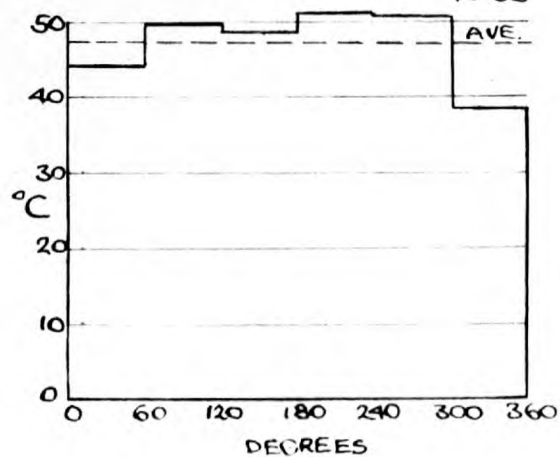
5.62"



7.62"



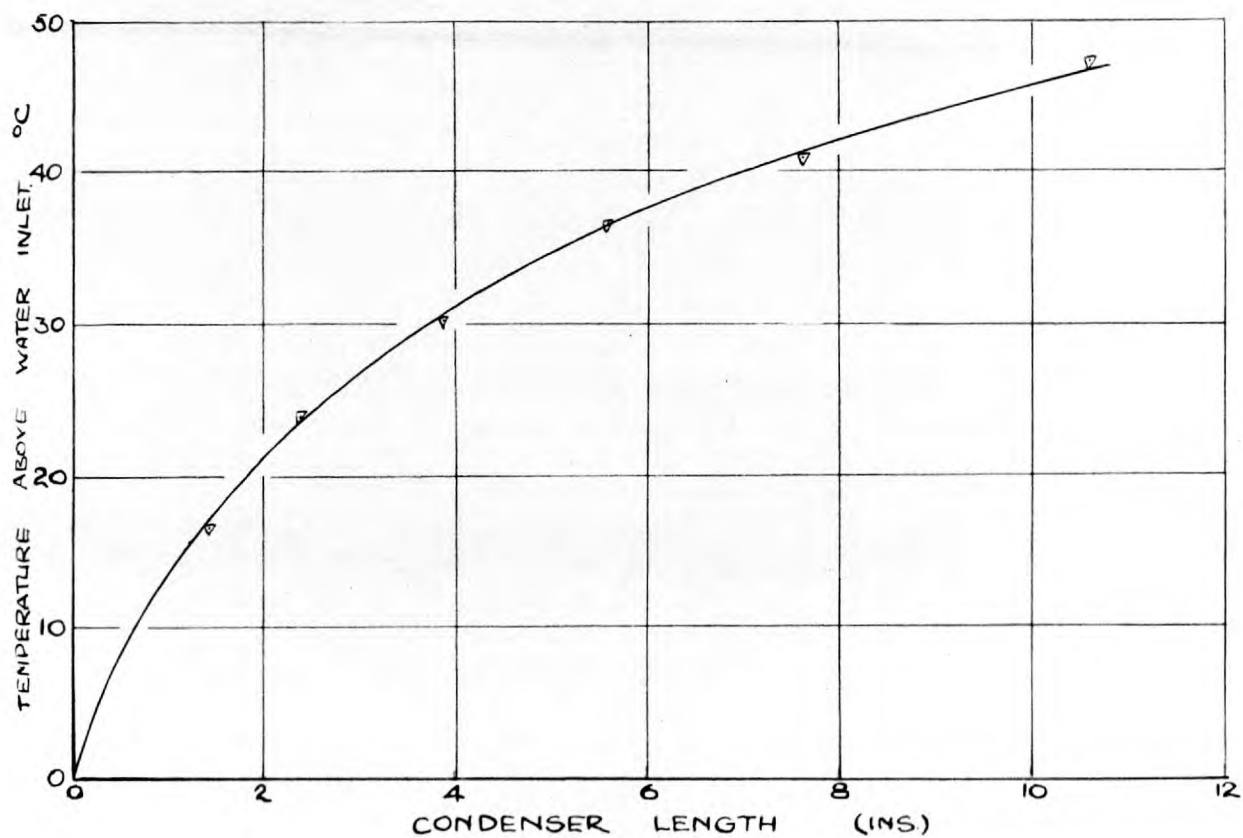
10.63"



BULK TEMPERATURE RISE OF
COOLING WATER JET

RUN. 1.

(NO AIR PRESENT)



Circumferential position of scoop - arbitrarily chosen.	Number of hatchway and distance from condenser entry					
	No. 1 1.43"	No. 2 2.40"	No. 3 3.87"	No. 4 5.62"	No. 5 7.62"	No. 6 10.63"
1	14.8	20.3	26.1	32.7	37.4	44.3
2	18.1	24.3	31.4	38.1	44.1	49.8
3	17.3	23.9	31.1	37.4	42.9	48.8
4	18.6	28.3	36.5	43.2	46.5	51.2
5	17.0	27.7	33.9	39.6	43.3	50.6
6	13.3	18.6	22.2	28.2	31.4	38.7
Average	16.52	23.85	30.20	36.53	40.93	47.22

Figs. 13 and 14 show the circumferential temperature distribution and the average bulk temperature rise down the condenser.

(i) Correction for steam condensation in the header section As a first approximation it was assumed that

$$t_{mh} = t_w, \quad \text{i.e.} \quad t_{mh} = 5.2^{\circ}\text{C.}$$

$$\text{From fig. 12, } h_h = 270 \text{ CHU/hr.ft}^2.^{\circ}\text{C.}$$

For the brass steam duct, $\frac{k_h}{b_h} = \frac{64}{0.10/12} = 7700 \text{ CHU./hr.ft.}^2 \text{ } ^\circ\text{C.}$
 where the mean thickness of the tubing is 0.10".

$$\therefore U = \frac{1}{\frac{1}{7700} + \frac{1}{270}} = 261 \text{ CHU./hr.ft.}^2 \text{ } ^\circ\text{C.}$$

The surface area, $S_h = \left(\frac{\pi \times 2.15 \times 3.75}{144} \right) = 0.1755 \text{ sq.ft.}$

based on the outside of the inner annulus tube.

From equation (10),

$$\Delta\theta'_{mh} = \frac{261 \times 0.1755 (100.4 - 5.2)}{2780 + \left(\frac{261 \times 0.1755}{2} \right)} = 1.55^\circ\text{C.}$$

By successive approximation, the annular water film coefficient; h_h , for a mean cooling water temperature of $(5.2 + \frac{1.55}{2})^\circ\text{C}$ is

$$h_h = 275 \text{ CHU/hr.ft.}^2 \text{ } ^\circ\text{C}$$

$$\therefore U = \frac{1}{\frac{1}{7700} + \frac{1}{275}} = 265 \text{ CHU/hr.ft.}^2 \text{ } ^\circ\text{C}$$

$$\therefore \Delta\theta'_{mh} = \frac{265 \times 0.1755 (100.4 - 5.2)}{2780 + \left(\frac{265 \times 0.1755}{2} \right)} = 1.58^\circ\text{C}$$

This is sufficiently accurate for the purpose

From equation (11)

$$W_{ch} = \frac{265 \times 0.1755}{539} \left[100.4 - \left(5.2 + \frac{1.6}{2} \right) \right]$$

$$W_{ch} = 8.32 \text{ lb/hr.}$$

∴ At the condenser entry proper,

$$\text{Cooling water temperature, } t_{w0} = (5.2 + 1.6) = 6.8^{\circ}\text{C}$$

$$\text{Steam flow rate, } W_{s0} = (295.5 - 8.3) = 287.2 \text{ lb/hr.}$$

$$\begin{aligned} \text{Steam velocity, } V_{s0} &= \left(\frac{W_{s0}}{3600} \times \frac{1}{A_o \rho_s} \right) = \left(\frac{287.2}{3600 \times 0.02515 \times 0.0378} \right) \\ &= 83.8 \text{ ft/sec.} \end{aligned}$$

(ii) Determination of turbulence factor From fig. 14, the actual temperature rise of the cooling water was read from the curve at 1", 2", 4", 6" & 8" from the condenser entry and the following calculation for conditions at a distance of 1", was repeated at each of the points in turn:-

$$\text{Actual cooling water temperature rise, } \theta_m' = 13.26^{\circ}\text{C}$$

$$\text{Actual cooling water temperature, } t_w = 20.06^{\circ}\text{C}$$

$$\text{Water orifice area, } = 0.345 \text{ sq.ins.}$$

$$\therefore A_w = \left(\frac{0.345}{144} \right) = 0.002396 \text{ sq.ft.}$$

$$\begin{aligned} \text{Water inlet velocity} &= \left(\frac{W_w}{3600} \times \frac{1}{A_w \rho_w} \right) \text{ ft/sec.} \\ &= 5.18 \text{ ft/sec.} \end{aligned}$$

By the laws of gravity, the time to fall 1" with an initial velocity of 5.18 ft/sec. = 0.0162 secs.

$$\text{Time of exposure, } \tau = \left(\frac{0.0162}{3600} \right) = 4.50 \times 10^{-6} \text{ hours.}$$

When calculating the dimensionless time factor $\frac{\alpha_w \tau}{l^2}$, it was appreciated that the theoretical relation between

$\frac{Q_w}{\theta_s}$ and $\frac{\alpha_w \tau}{L^2}$ (equation (1) and fig. 6), was obtained by integrating θ with respect to τ , α_w and L being considered constant throughout the period of heat transfer. Variations in α_w and L cannot therefore be accommodated with great accuracy. In this work, the changes in α_w

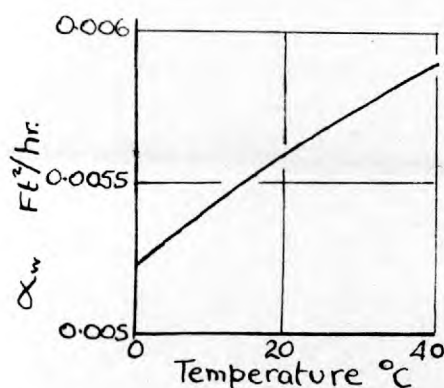


Fig. 15. Thermal diffusivity of water

are not critical (fig. 15) but the decrease in L amounts to 40% after a distance of 10" with a cooling water flow of 2780 lb/hr. To make some allowance for changes in α_w and L , these quantities were taken at their arithmetic mean values during the corresponding time of exposure, τ .

During condensation in the first inch of condenser length, the mean bulk cooling water temperature,

$$= 6.8 + \left(\frac{20.06 - 6.8}{2} \right) = 13.4^{\circ}\text{C}.$$

Hence the thermal diffusivity, $\alpha_w = 0.00549 \text{ ft}^2/\text{hr}$, (fig. 15)

The water jet thickness is modified by 2 factors during condensation;

- a) Thinning due to gravitational acceleration.
- b) Thickening due to steam condensation.

(The overall effect of turbulence on the jet, is found later)

a) Water jet velocity at 1" from the entry,

$$V_w = \sqrt{V_{w0}^2 + 2g \times \frac{1}{12}}$$

$$\therefore V_w = 5.67 \text{ ft/sec.}$$

$$\text{The corresponding jet thickness} = \left(0.050 \times \frac{5.2}{5.67} \right)$$

$$= 0.0456 \text{ ins.} = 0.00380 \text{ ft.}$$

b) The steam condensed to give a temperature rise, θ_m in the cooling water, can easily be shown to be:-

$$W_c = \frac{W_w \cdot \theta_m}{H_s - t_w}$$

$$\therefore W_c = \frac{2780 \times 13.26}{639.4 - 20.06} = 59.6 \text{ lb/hr.}$$

The increase in jet thickness due to steam condensation is

$$= \left(\frac{W_c}{3600} \times \frac{1}{V_w \cdot \rho_w \pi D_i} \right) \text{ ft.}$$

where D_i is the internal diameter of the water jet = $(D - 2l)$ ft.

$$D_i = \left(\frac{2.375 - 2 \times 0.0456}{12} \right) \text{ ft.}$$

$$\therefore D_i = 0.190 \text{ ft.}$$

$$\text{Hence } \Delta l = 7.75 \times 10^{-5} \text{ ft.}$$

$$= (0.00380 + 7.75 \times 10^{-5}) = 0.00388 \text{ ft.}$$

$$\therefore \text{Mean value of } l = \frac{(0.00417 + 0.00388)}{2} = 0.00402 \text{ ft.}$$

$$\text{From the above, } \frac{\alpha_w \gamma}{l^2} = \frac{0.00546 \times 4.5 \times 10^{-6}}{(0.00402)^2} = 0.00152$$

$$\therefore \frac{\theta_m}{\theta_s} = 0.041, \text{ (fig. 6).}$$

$$\therefore \theta_m = 0.041 \times (100.4 - 6.8) = 3.83^\circ \text{C}$$

$$\frac{\theta'_m}{\theta_m} = \frac{13.26}{3.83} = 3.46$$

$$\text{i.e. } r = 3.46$$

The correlation of r with steam and air velocity is dealt with elsewhere (page 30 et seq.).

(2) Condensation of steam with premixed air. The following calculations are on data obtained in run 10.

Experimental results

Steam flow rate by nozzle	= 353 lb/hr.
Air flow rate by nozzle	= 40.1 lb/hr.
Condenser pressure	= 30.3 ins.Hg, 14.8 p.s.i.a.
Steam-air temperature in condenser entry duct	= 98.8°C.
Cooling water flow rate by venturimeter	= 5660 lb/hr.
Cooling water inlet temperature to header	= 7.8°C.
Cooling water orifice width	= 0.050 ins.

The bulk temperature rise of the cooling water was :-

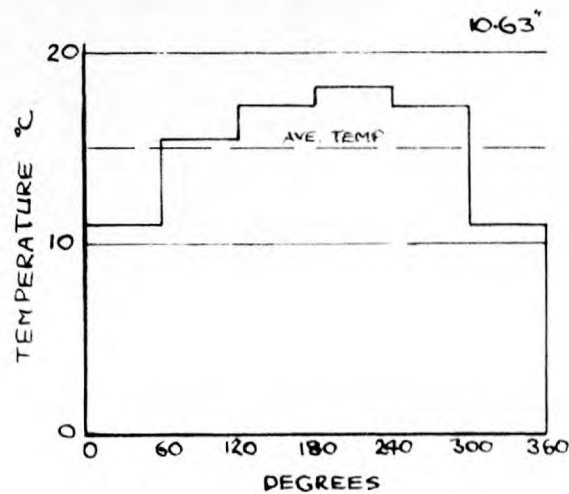
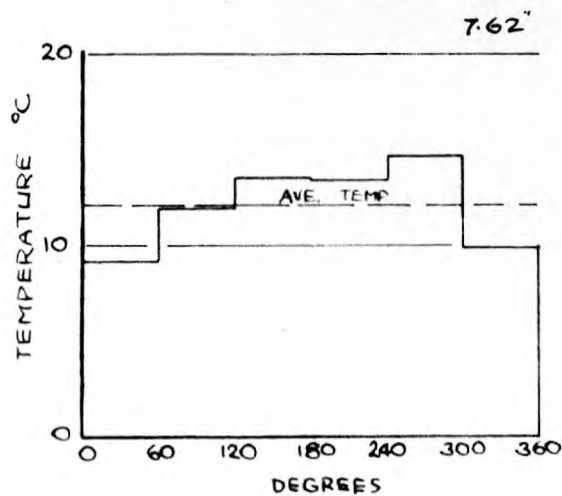
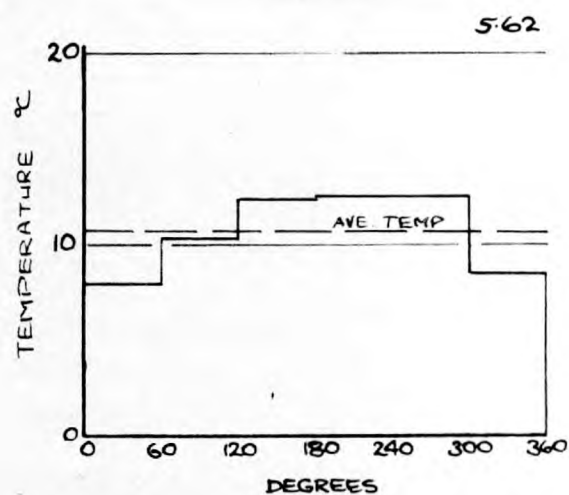
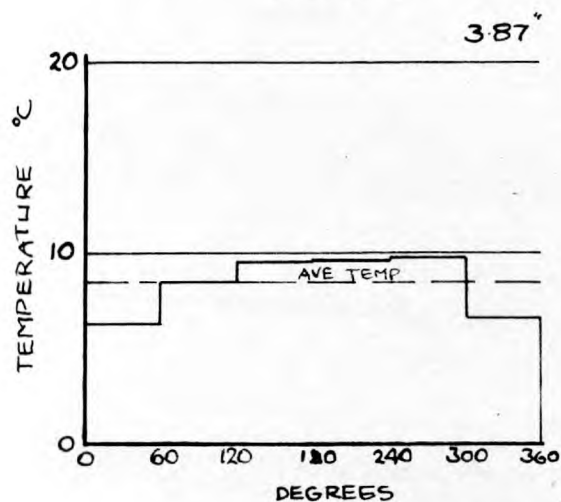
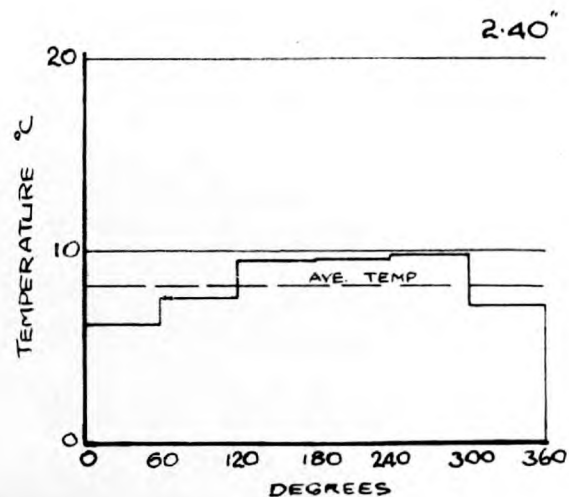
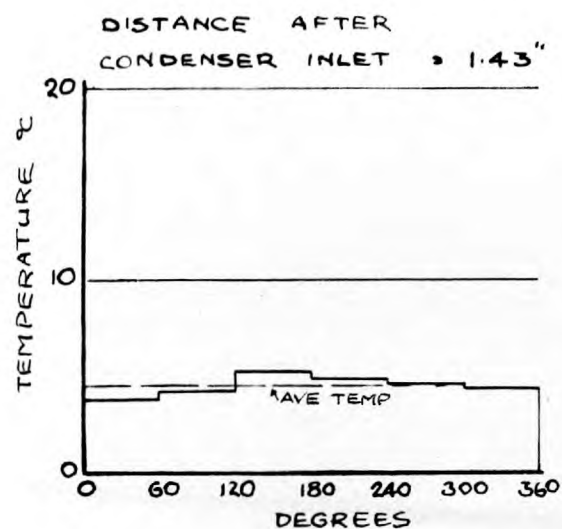
Circumferential position of scoop -arbitrarily chosen.	Number of hatchway and distance from condenser inlet.					
	No.1 1.43"	No.2 2.40"	No.3 3.87"	No.4 5.62"	No.5 7.62"	No.6 10.63"
1	3.7	6.1	6.3	7.9	9.1	11.0
2	4.1	7.5	8.5	10.3	11.9	15.4
3	5.2	9.5	9.5	12.3	13.5	17.2
4	4.9	9.6	9.6	12.5	13.4	18.2
5	4.5	9.7	9.7	12.5	14.7	17.2
6	4.3	7.1	6.6	8.5	9.9	11.0
Average	4.45	8.25	8.37	10.67	12.08	15.00

Figs. 16 and 17 show the circumferential temperature distribution and the average bulk temperature rise down the condenser.

Radial temperature traverse readings were taken through hatchways 2, 4 & 6:-

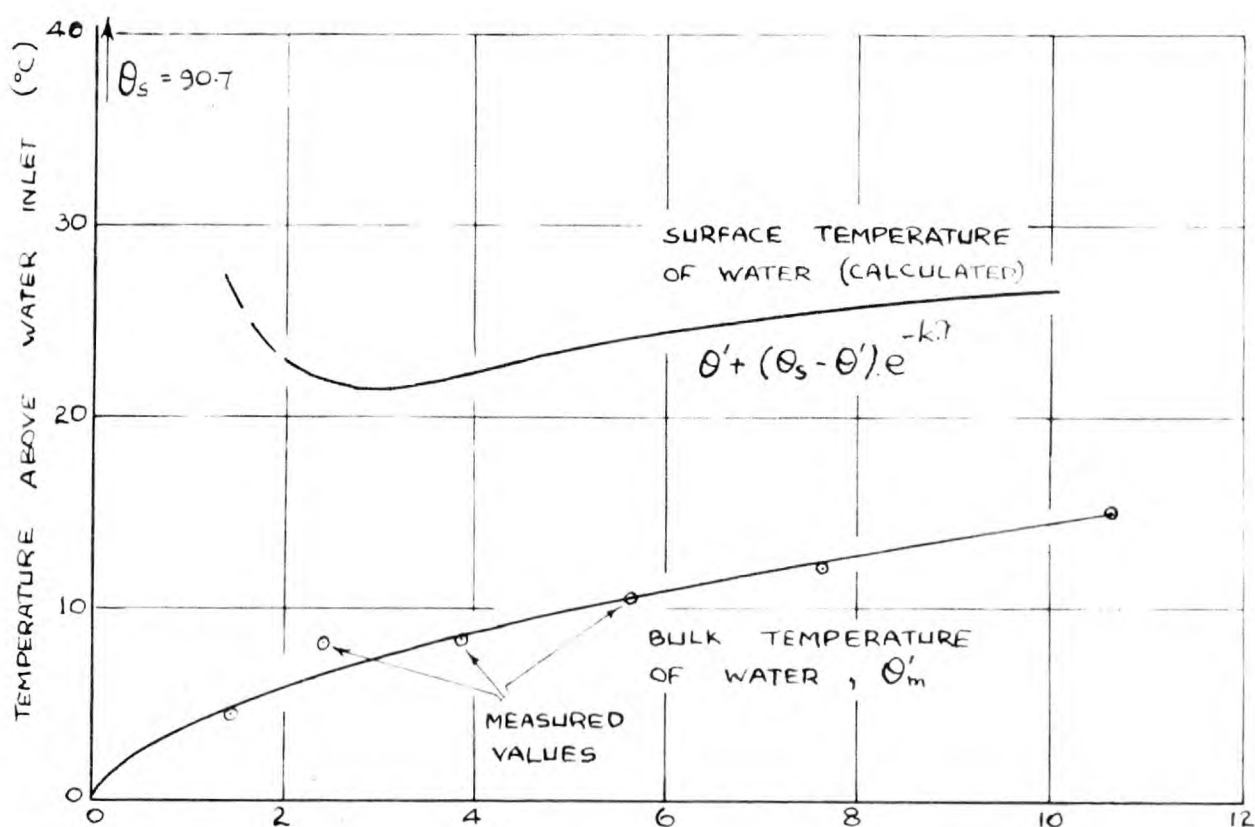
CIRCUMFERENTIAL TEMPERATURE RISE DISTRIBUTION
OF COOLING WATER JET

RUN 10.



BULK TEMPERATURE RISE & SURFACE
TEMPERATURE OF COOLING WATER JET

RUN 10
(10.2% INITIAL AIR CONTENT)



Station 2 - 2.40 ins. from condenser inlet

Circumferential position of probe	Distance from duct wall (ins.)				
	0.07	0.085	0.12	0.155	0.180
	Steam-air temperature °C.				
1	21.5	39.9	82.1	96.4	97.5
2	20.2	33.8	68.4	93.1	96.15
3	18.9	31.4	73.1	94.2	97.2
4	18.9	30.1	86.5	96.85	97.1

Radial traverses were taken at 4 circumferential positions in duct at 90° displacements

Station 4 - 5.62 ins from condenser inlet.

Circumferential position of probe	Distance from duct wall (ins.)				
	0.07	0.085	0.12	0.155	0.180
	Steam-air temperature °C.				
1	26.7	46.0	79.8	95.3	96.9
2	26.5	37.5	61.5	90.9	95.5
3	27.7	38.6	76.5	95.2	96.9
4	28.2	49.6	75.3	96.4	96.85

Station 6 - 10.63 ins. from condenser inlet.

Circumfer- ential position of probe	Distance from duct wall (ins)				
	0.07	0.085	0.12	0.155	0.180
	Steam-air temperature °C.				
1	27.7	38.2	74.8	93.2	95.93
2	27.7	32.8	60.3	89.85	95.2
3	24.0	33.6	73.3	94.15	96.3
4	20.7	29.7	66.1	92.0	95.75

Fig. 18 shows the steam-air temperature gradients at the above stations.

Inlet conditions. The accuracy of the steam and air metering was checked by comparing the observed steam-air inlet temperature with the saturation temperature of the steam corresponding to the steam-air weight ratio at the condenser pressure. From a graph of $\frac{W_a}{W_s} (= \frac{P_a}{P_s})$ vs. t drawn for mixture pressures of 29, 30 and 31 ins. H_g, fig. 19a, the saturation temperature is 98.5°C. This compares favourably with the observed temperature of 98.8°C in the entry duct. The saturation temperature, determined as above, was taken as the true inlet temperature of the steam-air mixture.

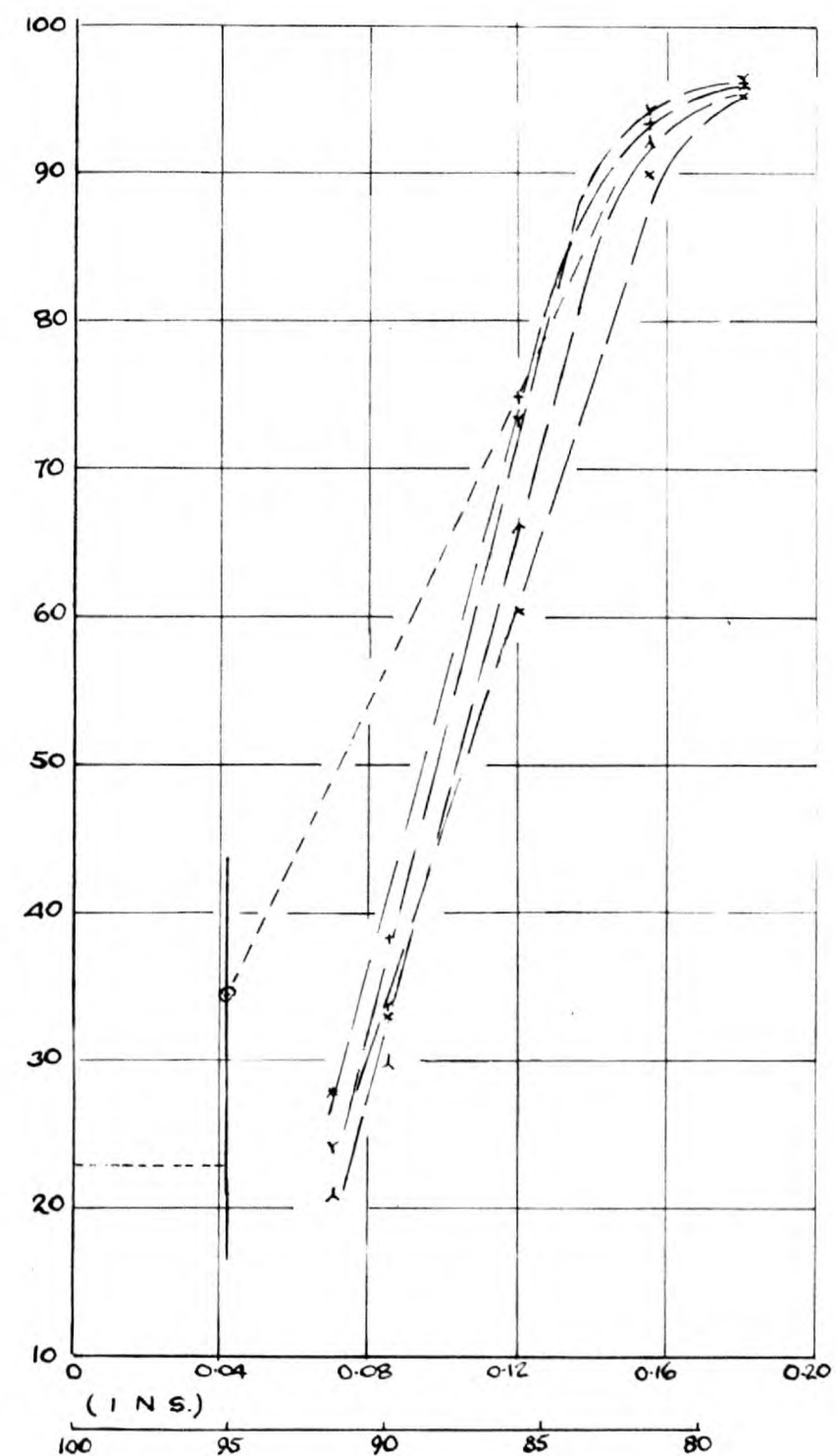
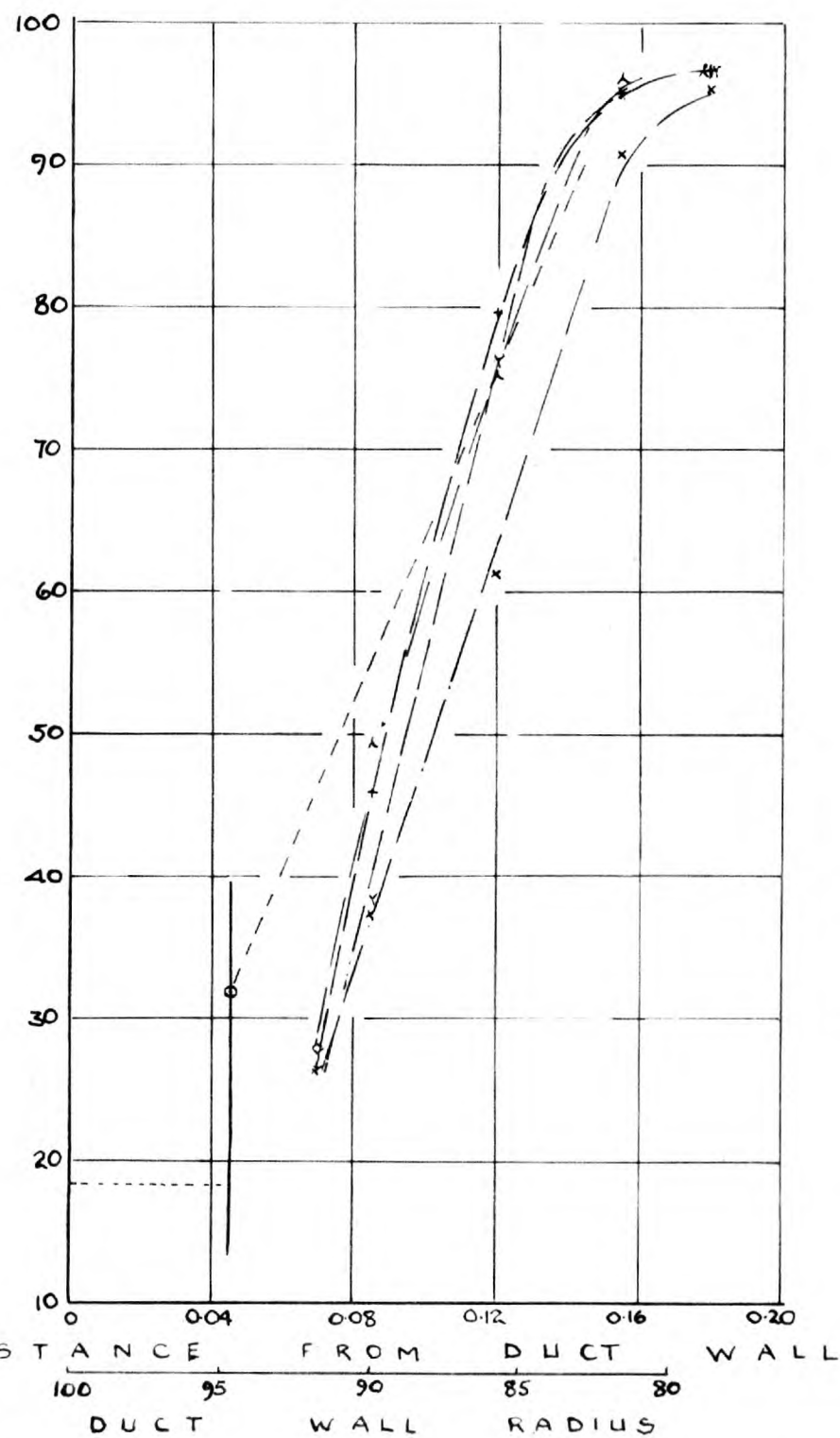
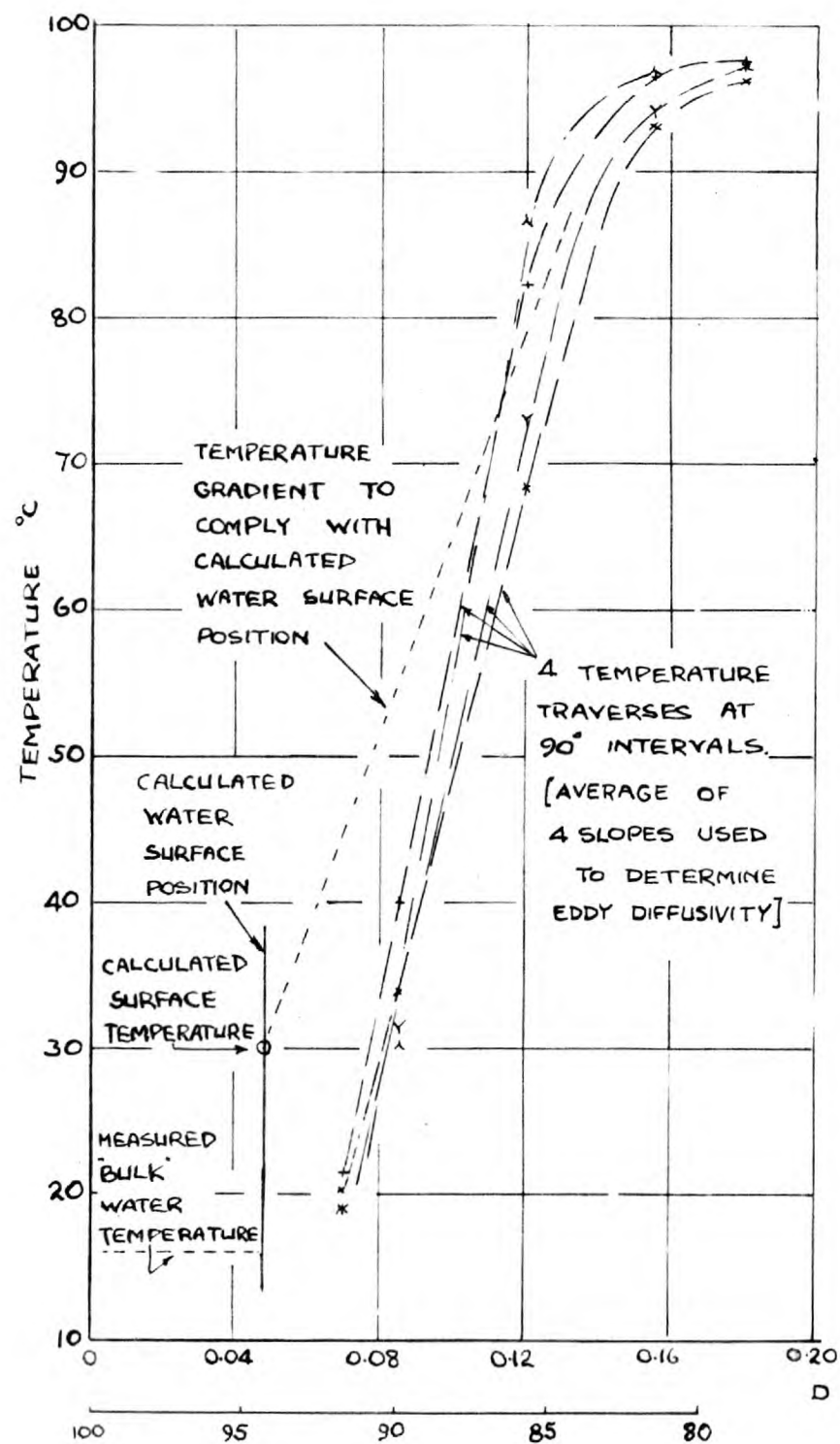
SPECIMEN RADIAL TEMPERATURE TRAVERSES IN STEAM-AIR MIXTURE

RUN 10. [STEAM-AIR INLET VELOCITY = 111 FT/SEC, AIR CONTENT, 10.2%]
 WATER INLET VELOCITY = 10.5 FT/SEC, .050" ORIFICE

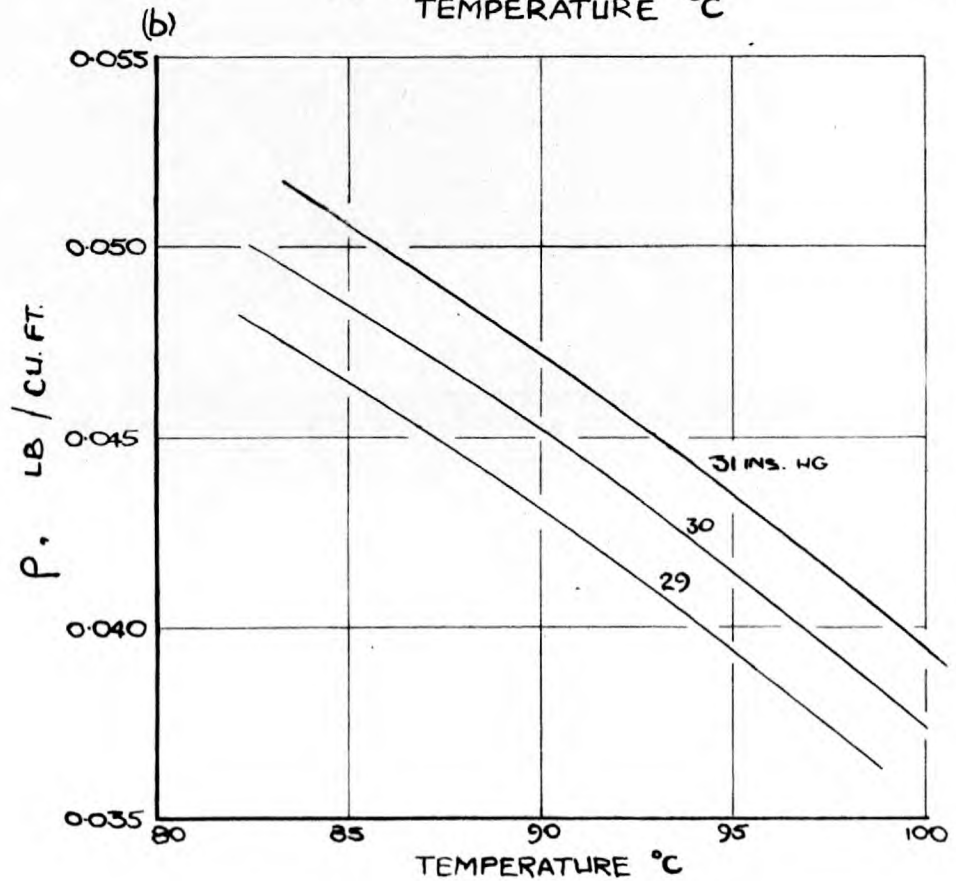
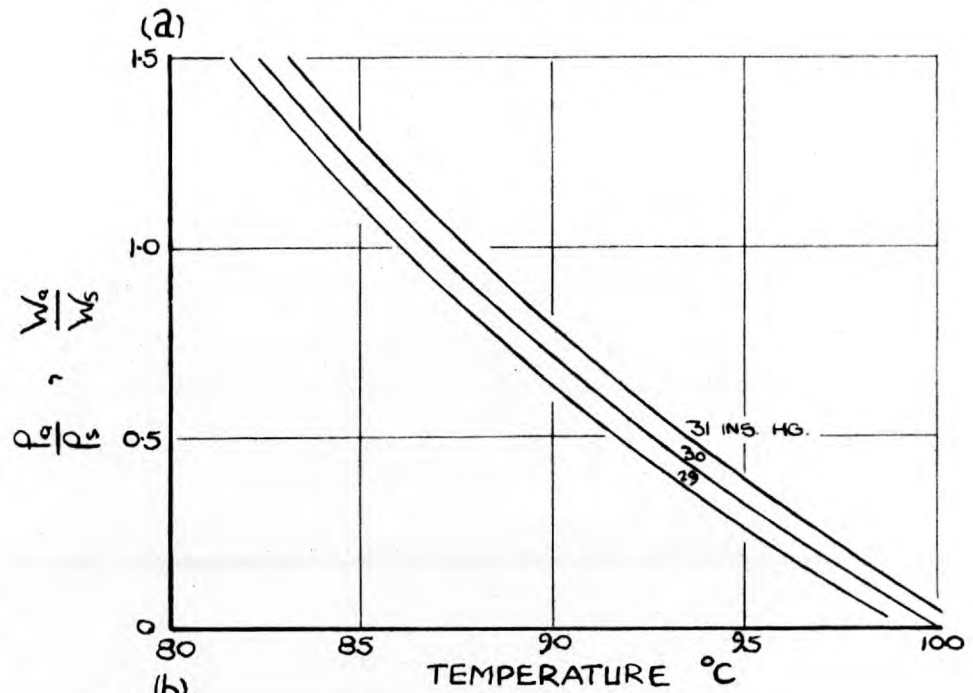
AXIAL DISTANCE AFTER
 CONDENSER INLET = 2.40"

5.62"

10.63"



VARIATION OF AIR/STEAM RATIO & MIXTURE
DENSITY WITH SATURATION TEMPERATURE
AT ATMOSPHERIC PRESSURE.



To facilitate analysis of the experimental data, a curve of ρ vs. t for constant mixture pressure, fig. 19b, was used to find the inlet steam-air mixture density.

From fig. 19b, at 98.5°C , $\rho = 0.0392 \text{ lb/cu.ft.}$

$$\text{The steam-air inlet velocity } V = \left(\frac{W}{3600} \times \frac{1}{A_m \rho} \right)$$

$$\therefore V = \left(\frac{(353 + 40.1)}{3600} \times \frac{1}{0.02515 \times 0.0392} \right) = 111.0 \text{ ft/sec.}$$

$$\text{Also, } \theta_s = (98.5 - 7.8) = 90.7^\circ\text{C.}$$

$$\text{The water inlet velocity} = \left(\frac{W_w}{3600} \times \frac{1}{A_w \rho_w} \right) = 10.5 \text{ ft/sec.}$$

For reasons mentioned on page 8, no allowance was made for the small percentage of steam condensed in the header from steam-air mixtures.

Determination of $\frac{\theta'_{m2}}{r\theta_s - \theta'_m}$. From fig. 17, the actual temperature rise of the cooling water was read from the curve at $\frac{1}{2}$, 1, 2, 4, 6, 8 & 10 ins. from the condenser entry and the following calculation for conditions at a distance of 1 in. was repeated at each of the points in turn.

$$\text{Actual cooling water temperature rise} = \theta'_m = 4.1^\circ\text{C.}$$

$$\text{Now } \theta'_{m2} = \theta'_m - \theta'_{m1} \quad \dots\dots(6), \text{ page 34}$$

$$\therefore \theta'_{m2} = \theta'_m \left(1 - \frac{\theta'_{m1}}{\theta'_m} \right) = \theta'_m \left(1 - \frac{\theta'_{m1}}{\theta'_m} \right)$$

Now $\frac{\alpha_w \cdot T}{l^2} = 0.000744$ - by the method of the previous section, page 46-7.

$$\therefore \frac{\theta_{m1}}{\theta_m} = 0.0325 \quad \dots\dots(\text{fig. 6})$$

$$\therefore \theta'_{m2} = 4.1 (1 - 0.0325) = 3.97^\circ\text{C}.$$

Referring to fig. 9c and ignoring at this stage the curves shown by broken lines, $\frac{rV_w^{0.17}}{V}$ was assumed constant valued at 0.06. (Full interpretation of figs. 9b and c is treated overleaf)

$$\therefore r = \frac{0.06 \times 111.0}{10.5} = 4.47$$

$$\therefore \frac{\theta'_{m2}}{r\theta_s - \theta'_m} = \frac{3.97}{(4.47 \times 90.7) - 4.1} = 0.01086$$

Presentation of results. The main endeavour of this work is to show quantitatively the effect on the steam condensation rate of the operating variables, viz. -

- a) Steam/air ratio and velocity.
- b) Cooling water velocity & jet thickness.

A means of reducing all these factors to two parameters was not found but it was hoped that the effect of steam and cooling water velocity and thickness would be allowed for in the parameters developed in Chapter IV.

Curves of $\frac{\theta'_{m1}}{r\theta_s - \theta'_m}$, vs. $\frac{\alpha_w \tau}{l^2}$ were therefore drawn for each of the runs 9 - 23. It should be noted that as θ'_m is small compared with $r\theta_s$ and θ'_{m1} is nearly equal to θ'_m (see page 53), $\frac{\theta'_{m1}}{r\theta_s - \theta'_m}$ can be interpreted as a measure of the temperature rise in the cooling water.

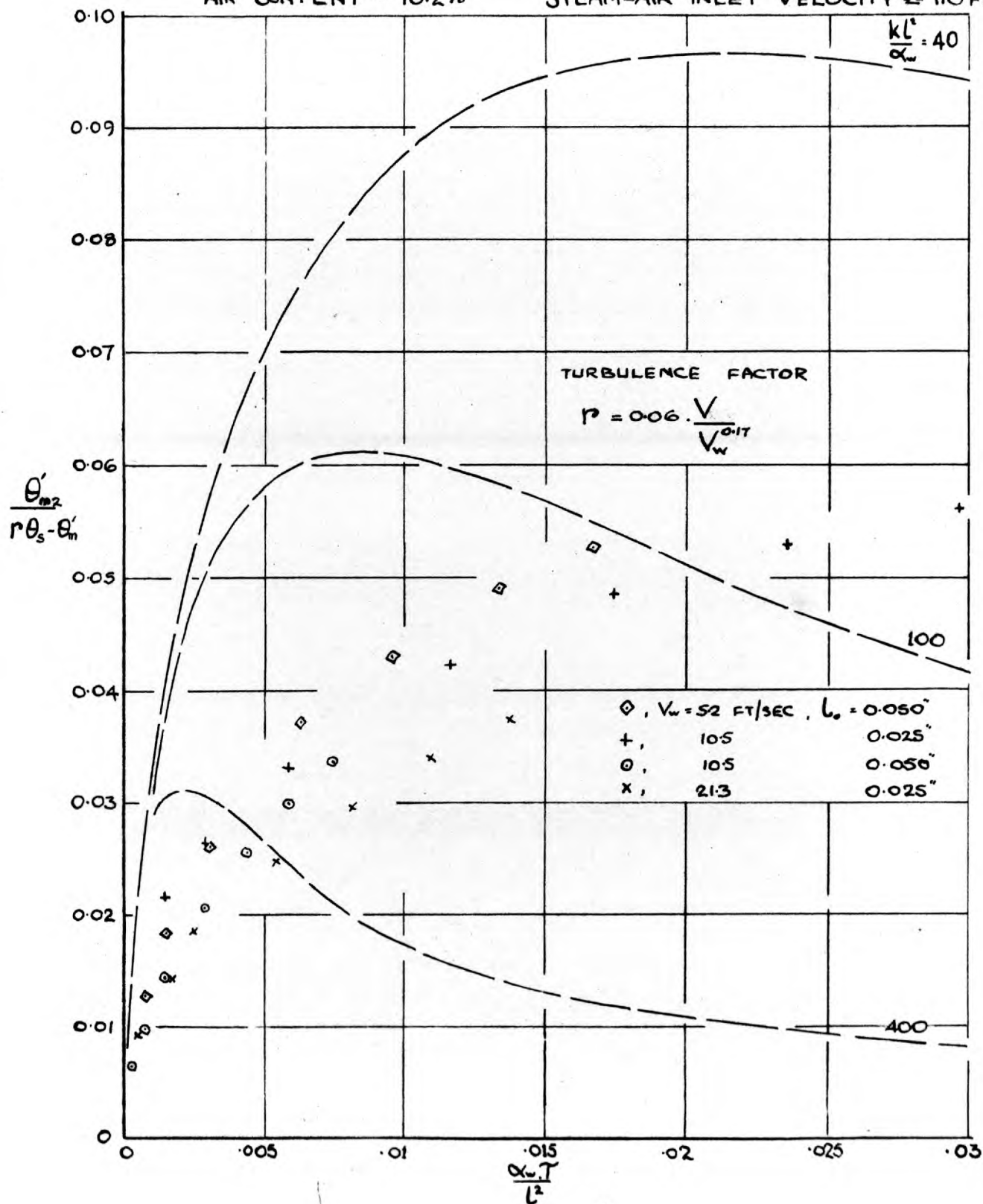
Effect of cooling water velocity and jet thickness Runs 9 - 12 were taken with constant initial steam/air ratio and steam-air velocity and varying cooling water velocity and jet thickness. The results are shown plotted in fig. 20 using a turbulence factor, $r = \frac{0.06V}{V_w^{0.17}}$, obtained from the results when condensing steam without premixed air, fig. 9c. The agreement is poor and inspection of the curves shows that the effect of the water velocity has been undercorrected, i.e. the exponent of 0.17 on V_w is too small. Other values were tried and a figure of 0.30 was found to provide the best agreement. Reference was then made to fig. 9b and a line of slope 0.30 was drawn on the graph. This slope was considered to be a plausible interpretation of the results from Series 1, the difficulty of obtaining consistent results in the first series (page 38) again being emphasized. Fig. 9c shows that an exponent of 0.30 lifts the curve, $V_w = 14.4$ ft/sec, (average of 2 runs) well above the

VARIATION OF $\frac{\theta'_{m2}}{r\theta'_s - \theta'_m}$ WITH $\frac{\alpha_w T}{L^2}$ FOR DIFFERENT ENTRY

CONDITIONS OF THE COOLING WATER i.e. DIFFERENT VALUES OF V_{w0} & L_0 .

AIR CONTENT = 10.2%

STEAM-AIR INLET VELOCITY ≈ 110 FT/SEC



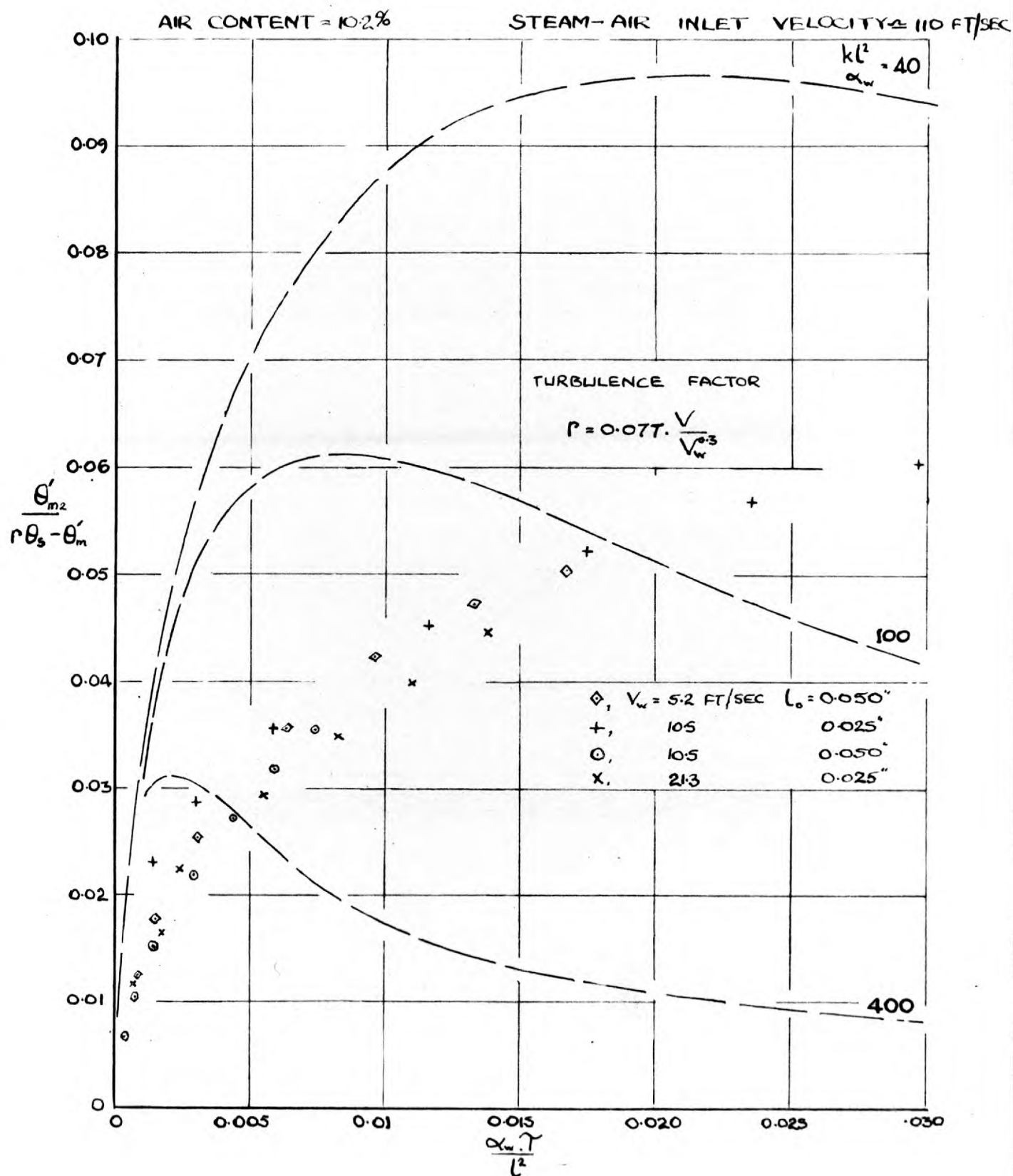
curves $V_w = 5.2$ and 7.2 ft/sec, (each being an average of 3 runs). Giving weight to the latter, the revised value of $\frac{V_w^{0.3}}{V}$ was taken as 0.077 and fig. 21 shows the results of runs 9 - 12 calculated using this figure.

The scatter is sufficiently small to conclude that variation in water velocity and jet thickness have been satisfactorily accommodated in the parameters $\frac{Q'_{m2}}{r\theta_s - Q'_m}$ and $\frac{\alpha_w \tau}{L^2}$.

Effect of steam-air entrance velocity Three curves are shown in fig. 22, with constant steam/air ratio and cooling water conditions and varying steam-air inlet velocity. The data from runs 14, 15 and 20, corresponding to 5.2, 16.9 and 28.5% air content, were cross-plotted to give the temperature rise in the cooling water expected with 10.2% air content. These figures were then reduced to the basic parameters and plotted with values from runs 11 and 13. The agreement between the curves substantiates the use of the steam-air velocity correction term in the turbulence factor.

Effect of initial steam/air ratio As the variation of cooling water temperature rise with steam-air entrance velocity and cooling water velocity and jet thickness,

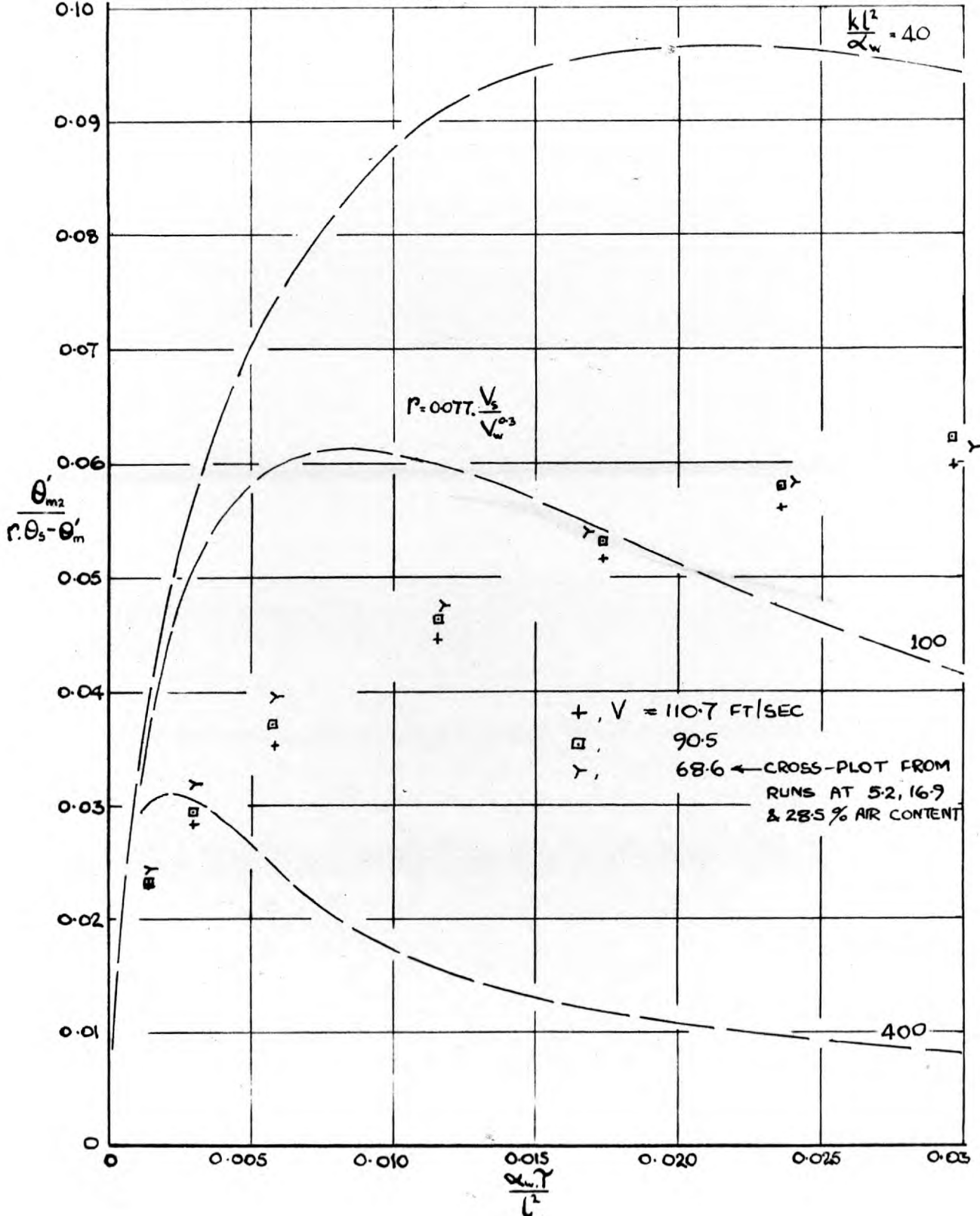
VARIATION OF $\frac{\theta'_{m2}}{r\theta_s - \theta'_m}$ WITH $\frac{\alpha_w \gamma}{l^2}$ FOR DIFFERENT ENTRY
CONDITIONS OF THE COOLING WATER



VARIATION OF $\frac{\theta'_{m2}}{r\theta'_s - \theta'_m}$ WITH $\frac{\alpha_w \gamma}{l^2}$ FOR DIFFERENT
STEAM-AIR INLET VELOCITIES

AIR CONTENT = 10.2%

$V_{w0} = 10.5$ FT/SEC. $L_0 = 0.025$ INS.

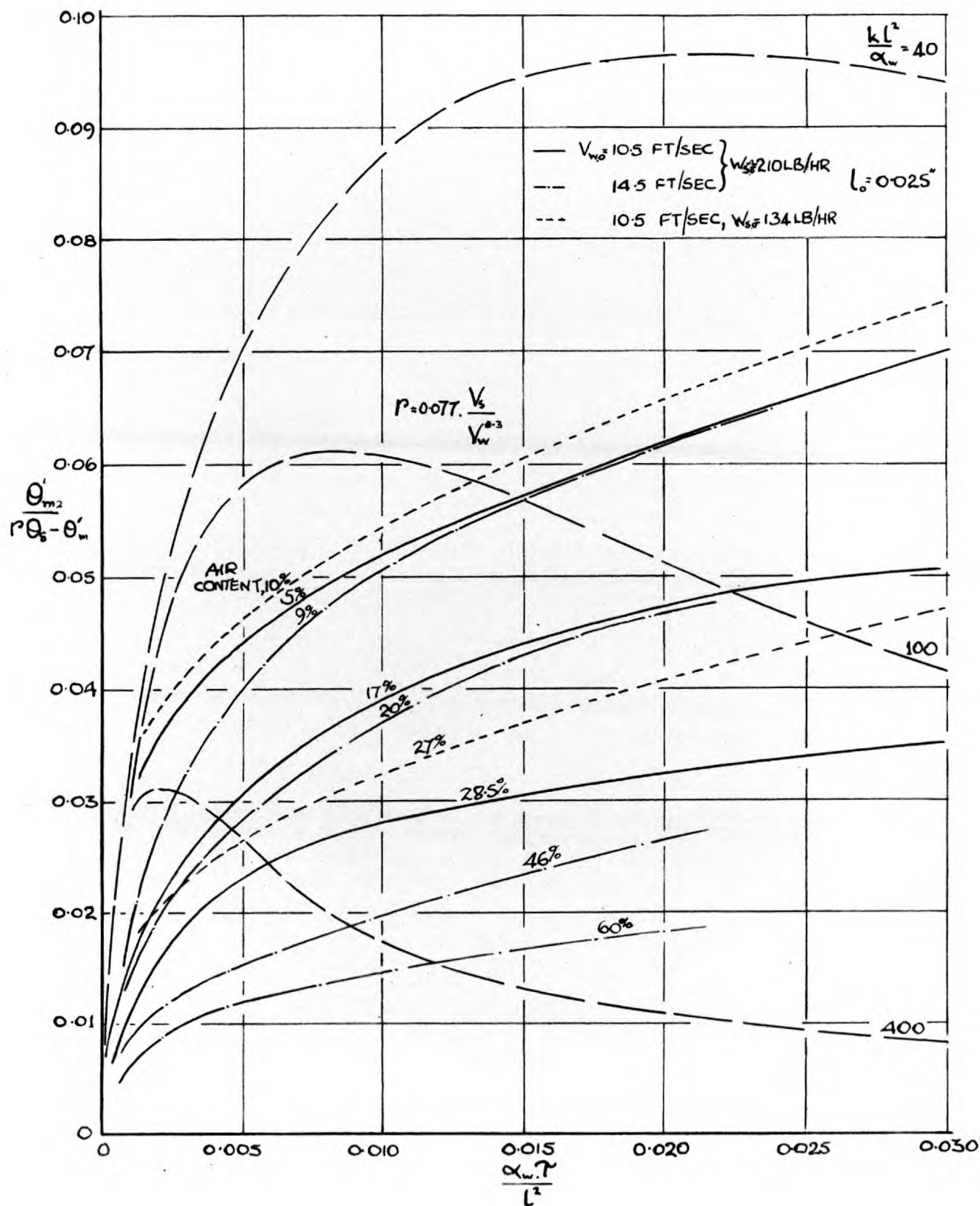


has been accommodated in the parameters $\frac{\theta_{m2}'}{r\theta_s - \theta_m}$ and $\frac{\alpha_w r}{l^2}$, the effect of varying the initial steam/air ratio can be expected to be a series of curves, each of fixed initial air content, on a graph with the above coordinates. The following runs are plotted in fig. 23 illustrating the quantitative effect of initial air content on the temperature rise in the cooling water.

Run No.	Initial % air content by weight	Steam flow (lb/hr)	Cooling water entrance velocity (ft/sec)	Initial thickness of cooling water jet (ins.)
20	5.2	218	10.5	0.025
14	16.9	213	10.5	0.025
15	28.5	213	10.5	0.025
16	9.2	209	14.5	0.025
17	19.9	212	14.5	0.025
18	45.7	203	14.5	0.025
19	60.3	206	14.5	0.025
22	10.4	135	10.5	0.025
23	26.7	133	10.5	0.025

Excepting run 22, these results are largely consistent with each other, on the effect of the initial air content in the steam on the temperature rise in the cooling water,

VARIATION OF $\frac{\theta'_{m2}}{p\theta_s - \theta'_m}$ WITH $\frac{\alpha_w T}{l^2}$ FOR DIFFERENT
INITIAL STEAM/AIR RATIOS



i.e. on the steam condensation rate.

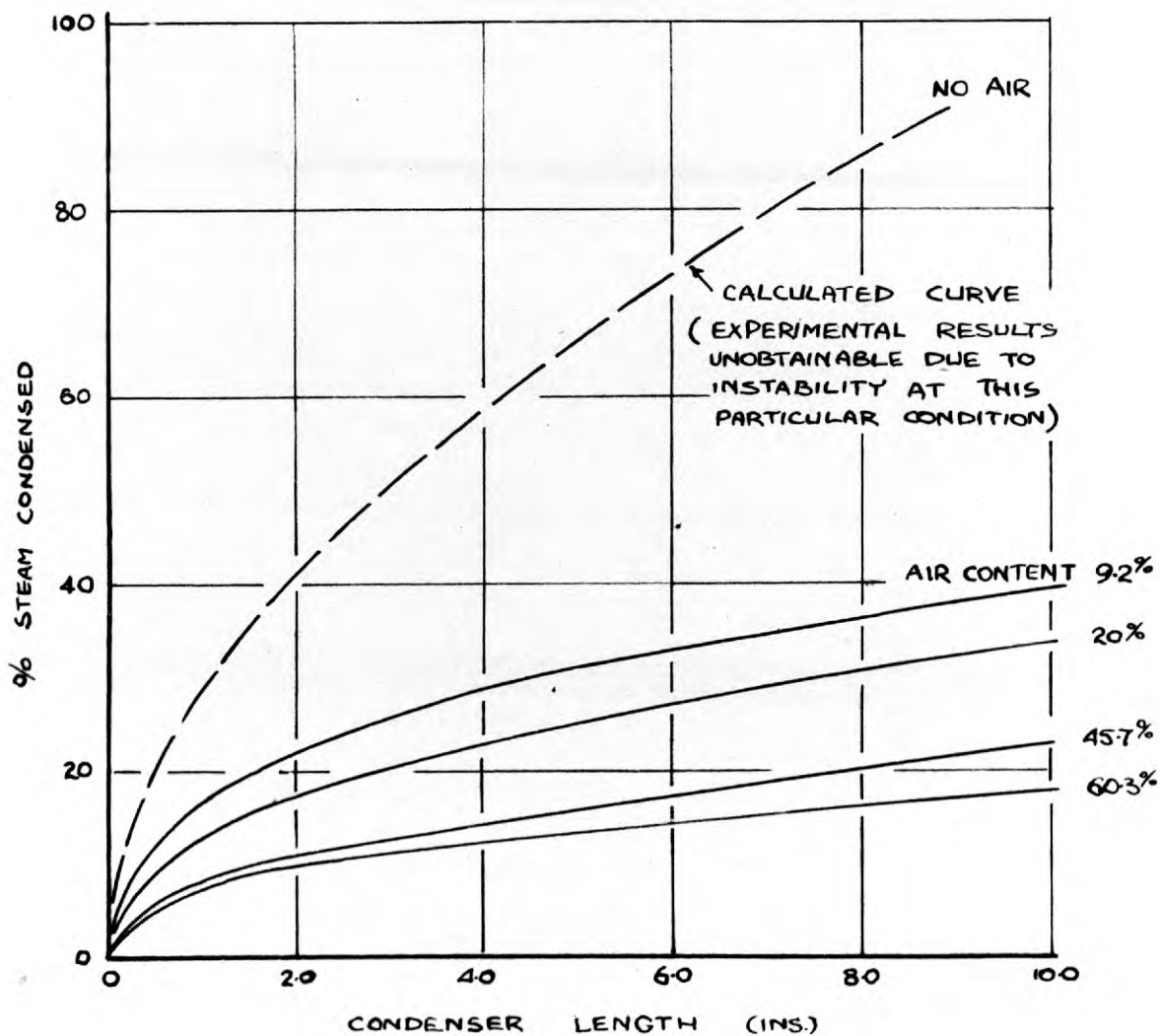
The overall effect of initial air content on the steam condensation rate is shown in fig. 24, for runs 16 - 19 where the steam condensed was calculated according to appendix 2. Unfortunately, instability of the cooling water jet prevented a run with no air present in the steam. For this case, using the average steam flow for the four runs, the turbulence factor, r , was calculated for the case of no air present and the temperature rise expected in the cooling water was then found for values of $\frac{\alpha_w T}{L}$, corresponding to the vertical distance down the condenser. The percentage steam condensation was readily estimated from the cooling water temperature rise, the curve being shown as a broken line. The curves together show clearly the marked reduction in the condensation rate with increasing air content in the steam, the major decrease occurring in this case with the first 9% of air. Runs 14, 15 and 20, fig. 23, indicate that the first 5% of air content in the steam is sufficient to provide the major part of the resistance to heat transfer in the steam-air phase.

EFFECT OF INITIAL AIR CONTENT
ON STEAM CONDENSATION

$$W_{s,g} = 210 \text{ LB/HR}$$

$$V_{w,0} = 14.5 \text{ FT/SEC.}$$

$$L_0 = 0.025''$$



VI EDDY DIFFUSIVITY IN THE STEAM-AIR MIXTURE

The steam condensation rate in the presence of air is primarily controlled by the rate of diffusion of the steam to the water surface and an investigation was made under different conditions of the variation in the turbulent coefficient of diffusion, better known as the eddy diffusivity.

The rate of eddy heat transfer through the steam-air mixture is: $-E_m \frac{d(pH)}{dy}$ CHU/ft².hr. ...Appendix 3.

The rate of mass diffusion from the turbulent steam-air mixture to the cooling water surface is:

$$\omega_s = -E_m \left(\frac{p}{p_a} \cdot \frac{dp}{dy} - \frac{p_s}{p_a} \cdot \frac{dp}{dy} \right) \text{ lb./ft}^2\text{.hr.} \dots \text{Appendix 4.}$$

The rate of heat transfer by convection in this diffusion of mass is: $\omega_s H_s$ CHU/ft².hr.

The total rate of heat transfer to the cooling water is therefore:

$$q = -E_m \frac{d(pH)}{dy} - E_m H_s \left(\frac{p}{p_a} \cdot \frac{dp}{dy} - \frac{p_s}{p_a} \cdot \frac{dp}{dy} \right) \text{ CHU/ft}^2\text{.hr.}$$

$$\therefore q = -E_m \frac{d(pH)}{dy} \left[1 + \gamma H_s \left(\frac{p}{p_a} \cdot \frac{dp}{d(pH)} - \frac{p_s}{p_a} \cdot \frac{dp}{d(pH)} \right) \right] \text{ CHU/ft}^2\text{.hr.}$$

where $\gamma = \frac{E_m}{E_h}$

Previous workers have found that the eddy diffusivities of mass and heat are not necessarily the same, but Perry

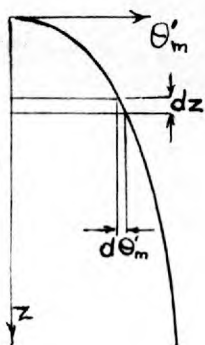
ref. 6 states that the use of $E_M = E_H$ does not involve differences of more than 30%. In this investigation E_M was assumed equal to E_H , i.e. $\gamma = 1$.

$$\therefore q = -E \frac{d(p_H)}{dy} (1 + I) \text{ CHU/ft}^2\text{hr} \quad \dots\dots(12)$$

$$\text{where } I = H_s \left(\frac{p}{p_a} \cdot \frac{dp_s}{d(p_H)} - \frac{p_s}{p_a} \cdot \frac{dp}{d(p_H)} \right)$$

I is a function of the temperature and pressure of the steam-air mixture and thus for constant pressure condensation, I only varies with temperature. Using the steam data of Callendar, ref. 7, curves of I vs. t were constructed for mixture pressures of 29, 30 and 31" Hg. The effect of pressure over this range was negligible and fig. 25 shows the variation of I with t for a nominal mixture pressure of 30" Hg.

Considering an elemental cooling water temperature rise $d\theta'_m$, the heat transferred is $(W_w + W_c) \cdot d\theta'_m$ CHU/hr.



The corresponding elemental surface area of the water is $\pi D_i \cdot dz$.

$$q = \frac{(W_w + W_c) \cdot d\theta'_m}{\pi D_i \cdot dz} \text{ CHU/ft}^2\text{hr}.$$

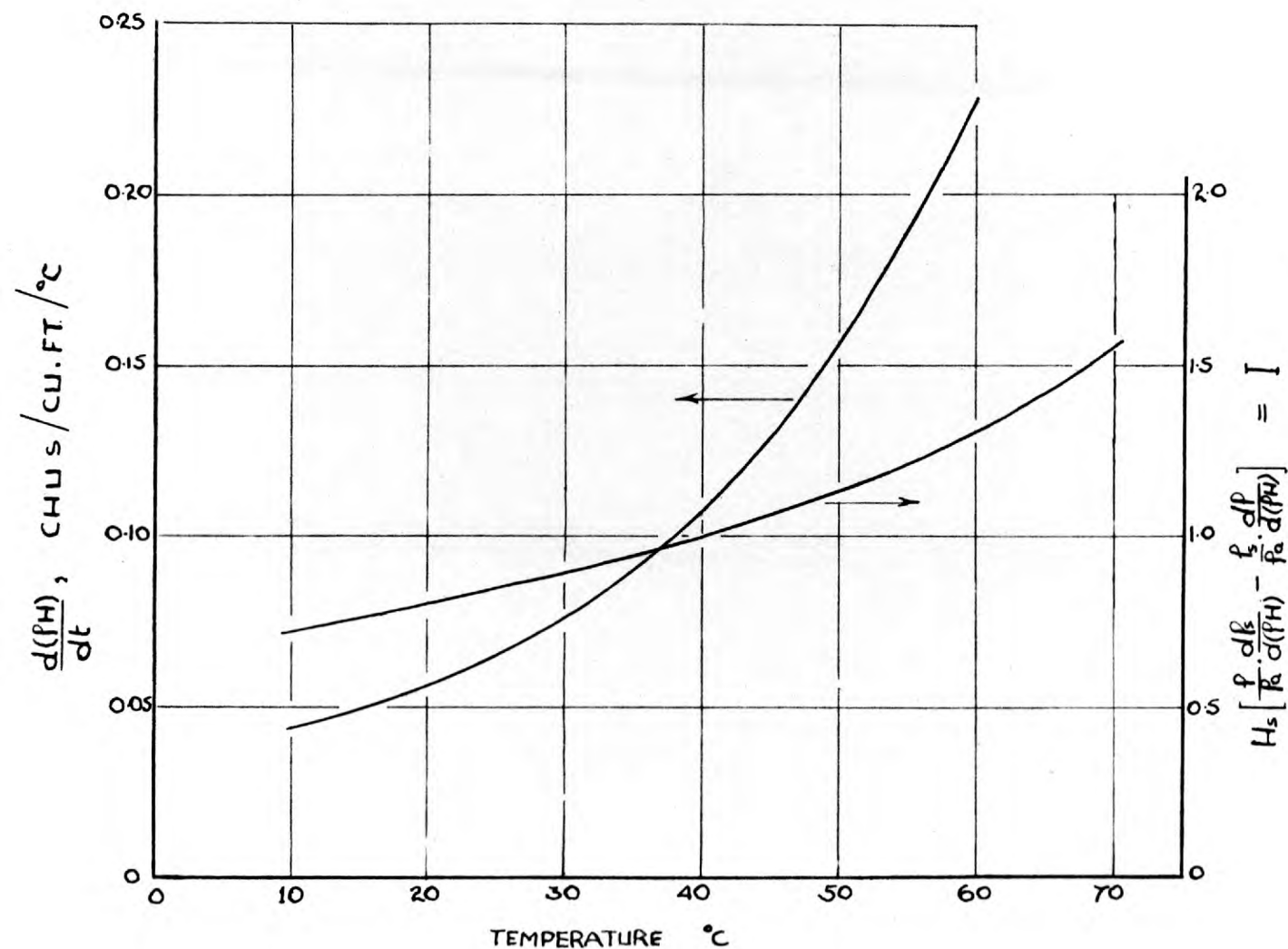
Combining with equation (12)

$$E = \frac{-(W_w + W_c) \cdot \frac{d\theta'_m}{dz}}{\pi D_i (1 + I) \cdot \frac{d(p_H)}{dy}} \text{ ft}^2\text{/hr}.$$

VARIATION OF $\frac{d(PH)}{dt}$ & I WITH STEAM-AIR
TEMPERATURE. TEMPERATURE.

(MIXTURE PRESSURE = 30 INS. HG.)

$$I = \frac{\text{DIFFUSION RATE OF HEAT BY EDDY CONVECTION}}{\text{DIFFUSION RATE OF HEAT BY EDDY CONDUCTION}}$$



$$\begin{aligned} \text{Now } \frac{d(\rho H)}{dy} &= \frac{d(\rho H)}{dt} \cdot \frac{dt}{dy} \\ E &= -\frac{(W_w + W_c) \cdot \frac{dQ'_m}{dz}}{\pi D_i \cdot (1+I) \cdot \frac{d(\rho H)}{dt} \cdot \frac{dt}{dy}} \quad ft^2/hr. \\ &\dots\dots(13) \end{aligned}$$

where $\frac{d(\rho H)}{dt}$ and I are at the surface temperature of the cooling water.

Surface temperature of the cooling water. It has been shown above that a knowledge of the water surface temperature is necessary before an estimate of the eddy diffusivity in the steam-air mixture may be calculated. The method now indicated is open to criticism but is used for lack of a more satisfactory solution.

A comparison between the shape of actual and theoretical curves of $\frac{\theta'_m}{r\theta_s - \theta'_m}$ and $\frac{\alpha_w \tau}{l^2}$, figs. 21 - 23, indicates an appreciable departure of the actual surface temperature of the cooling water from the theoretical assumption, $\theta_{surf} = \theta_m + (\theta_s - \theta_m) \cdot e^{-k\tau}$. This prevents an accurate assessment of the surface temperature.

Fortunately the degree of accuracy required in the surface temperature estimation is not high as the functions $\frac{d(\rho H)}{dt}$ and I_w ^{equation (13),} do not vary critically with temperature over the range of 15°C to 30°C, (fig. 25). This temperature range was found to cover values of the

surface temperature at the hatch positions of the thermocouple probe used in this work. For the present purpose, the surface temperature of the cooling water is taken as: $\theta_{\text{surf}} = \theta'_m + (\theta_s - \theta'_m)e^{-k\tau}$

where k is interpolated from the appropriate curve (figs. 21 - 23) at a value of $\frac{\alpha_w \tau}{L}$ corresponding to the relevant position along the condenser.

Reynolds number of the steam-air mixture in the condenser.

Sherwood & Woertz. ref. 8, found that the eddy diffusivity in a gas stream was a function of the Reynolds number and it was decided to plot the present diffusivity results on the same parameter. Evaluation of the Reynolds number necessitated a knowledge of the flow rate and viscosity of the steam-air mixture along the condenser. These were determined at the average steam-air condition at the point in question, the flow rate being readily found.

A search through the literature revealed that the most practicable method of determining the mixture viscosity was that developed by Wilke and Buddenberg, refs. 9 and 10. They developed an expression for the viscosity of a gaseous mixture from the physical properties and ratios of its components. Other workers have studied the problem but calculation by their techniques are

lengthy and require an extensive knowledge of the kinetic theory. The theory of Wilke and Buddenberg is not absolute but has been found to predict viscosities within 2½% of experimental values. Their expression for the steam-air mixture viscosity is:-

$$\mu = \frac{\mu_s}{1 + \frac{x_a}{x_s} \left[\frac{1 + \left(\frac{\mu_s \rho_a}{\mu_a \rho_s} \right)^{\frac{1}{2}} \left(\frac{M_s}{M_a} \right)^{\frac{1}{2}} \right]^2} + \frac{\mu_a}{1 + \frac{x_s}{x_a} \left[\frac{1 + \left(\frac{\mu_a \rho_s}{\mu_s \rho_a} \right)^{\frac{1}{2}} \left(\frac{M_a}{M_s} \right)^{\frac{1}{2}} \right]^2}$$

$$\frac{4}{\sqrt{2}} \left[1 + \frac{M_s}{M_a} \right]^{\frac{1}{2}} \quad \frac{4}{\sqrt{2}} \left[1 + \frac{M_a}{M_s} \right]^{\frac{1}{2}}$$

.....(14)

where ρ_s and ρ_a are the steam and air densities at the pressure of the mixture. To quote, "For ideal gases, the denominators are nearly independent of temperature and pressure. For components at temperatures below their boiling points, hypothetical vapour densities may be estimated on the basis of the gas laws". It was, therefore, necessary to select a gas constant for steam to cover its behaviour at partial pressures below atmospheric pressure. A figure of 15.25 ft.lb/lb.°C. for R was within an error of 1% from 0 to 80% air content and this was used to predict ρ_s . From equation (14), a curve of viscosity vs% air content in the steam-air mixture was drawn and

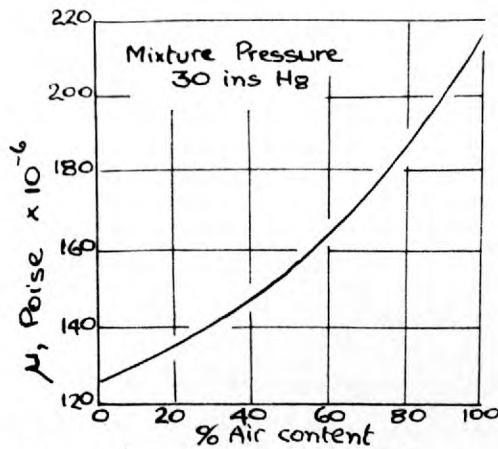


Fig. 26. Viscosity of the steam-air mixture.

is shown in fig. 26.

The characteristic diameter in the Reynolds number was taken as D_i i.e. the inside diameter of the water jet.

Specimen calculation for E. The following calculation is on data obtained in run 10, pages 48 - 50, and corresponds to the conditions in the condenser at station 2, i.e. 2.40" from the condenser entry. The calculating procedure was repeated at each of the other two thermocouple probe positions, station 4 and 6, used in this run.

$$\frac{d\theta'_m}{dz} = 1.58 \text{ degrees/inch, } 18.96 \text{ degrees/ft. (fig. 18)}$$

$$\text{Steam condensed, } W_c \approx \frac{W_w \theta'_m}{H_s - t_w} = \left(\frac{5660 \times 6.6}{638.8 - 14.4} \right) = 59.8 \text{ lb/hr.}$$

$$\text{Total water flow} = W_w + W_c = (5660 + 60) = 5720 \text{ lb/hr.}$$

Estimation of D_i :- Water inlet velocity $V_{w0} = 10.50 \text{ ft/sec.}$

$$V_w = \sqrt{10.50^2 + 2g \times \frac{2.40}{12}}$$

$$V_w = 11.12 \text{ ft/sec.}$$

$$\begin{aligned}\text{The corresponding jet thickness} &= \left(0.050 \times \frac{10.50}{11.12} \right) \\ &= 0.0472 \text{ ins.}, 0.00393 \text{ ft.}\end{aligned}$$

$$\text{Now } D_i = D - 2l$$

$$\therefore D_i = \left(\frac{2.375}{12} - 2 \times .0039 \right) = 0.190 \text{ ft.}$$

(the magnitude of l in comparison with D , justifies the neglect of change in thickness, Δl due to steam condensation on the jet).

Estimation of cooling water surface temperature:-

It is assumed that $\theta_{\text{surf}} = \theta'_m + (\theta_s - \theta'_m) \cdot e^{-k\tau}$

Now $\frac{\alpha_w \tau}{l^2} = 0.00176$, from a plot of $\frac{\alpha_w \tau}{l^2}$ vs. condenser length, using values of $\frac{\alpha_w \tau}{l^2}$ in fig 21.

$$\text{and } \frac{kl^2}{\alpha_w} = 950, \quad \text{from fig. 21 by cross-plotting.}$$

$$\therefore k\tau = 0.00176 \times 950 = 1.672$$

$$\therefore e^{-k\tau} = 0.188$$

$$\therefore \theta_{\text{surf}} = 6.6 + (90 - 6.6) \times 0.188 = 22.4^\circ \text{C.}$$

$$\text{Surface temperature} = (\theta_{\text{surf}} + t_{\text{wo}})^\circ \text{C} = (22.4 + 7.8) = 30.2^\circ \text{C}$$

$$\text{From fig. 25, } I = 0.89 \text{ and } \frac{d(IH)}{dt} = 0.0752$$

From equation (13), page 60,

$$E = \frac{-5720 \times 18.96}{\pi \times 0.190 \times 1.89 \times 0.0752 \left(\frac{dt}{dr} \right)} \text{ ft.}^2/\text{hr.}$$

$$\therefore E = - \left(\frac{1,278,000}{dt/dr} \right) \text{ ft.}^2/\text{hr.}$$

The actual temperature gradient in the steam-air mixture is:-

$$\frac{dt}{dy} = -1076 \text{ degrees/in.}, -12900 \text{ degrees/ft.}, (\text{fig. 18}).$$

$$\therefore E = \frac{1,278,000}{12900} = 99.0 \text{ ft.}^2/\text{hr.}$$

For reasons discussed elsewhere, page 71, a second value of E was determined using a temperature gradient complying with the calculated water surface position, whence

$$\frac{dt}{dy} = -683 \text{ degrees/in.}, -8196 \text{ degrees/ft.}, (\text{fig. 18})$$

$$\therefore E = \frac{1,278,000}{8196} = 156.0 \text{ ft.}^2/\text{hr.}$$

Reynolds number of the steam-air mixture:-

$$Re = \frac{Gd.}{\mu} = \frac{4W}{\pi D_i \mu}$$

$$\text{And } W = W_s + W_a = (353 - 60) + 40 = 333 \text{ lb/hr.}$$

$$D_i = 0.190 \text{ ft.} \quad \dots\dots(\text{page 64})$$

$$\text{Air content by weight} = \frac{40.1}{333} = 12.0\%$$

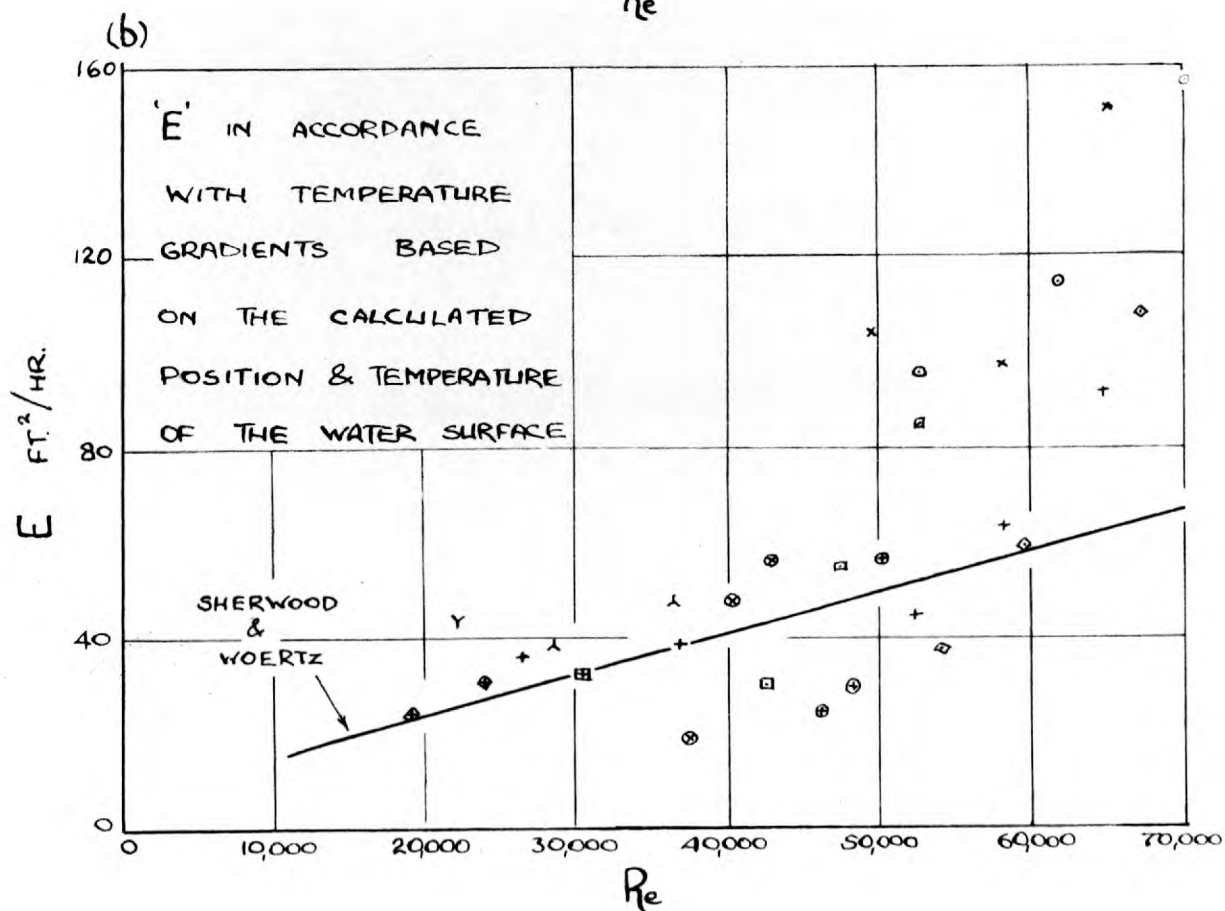
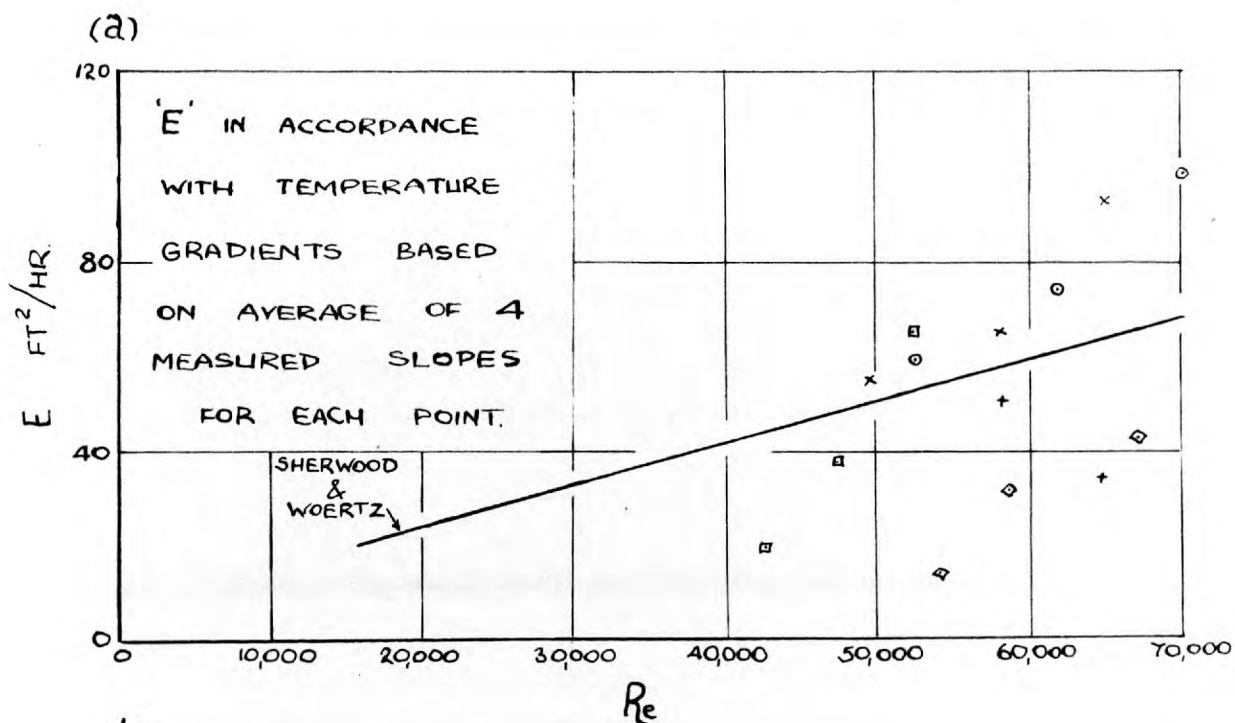
$$\therefore \mu = 131.6 \times 10^{-6} \text{ poise} \quad (\text{fig. 25})$$

$$\therefore \mu = (131.6 \times 10^{-6} \times 242) = 0.0319 \text{ lb/ft.Hr.}$$

$$\therefore Re = \left(\frac{4 \times 333}{\pi \times 0.190 \times 0.0319} \right) = 70.1000.$$

Presentation of the diffusivity data. Fig. 27a shows the variation of the eddy diffusivity E with the Reynolds number, where E is based on the measured temperature

VARIATION OF EDDY DIFFUSIVITY, E , WITH REYNOLDS
NUMBER Re FOR STEAM-AIR MIXTURES



gradient in the steam-air mixture. The number of points on the graph are limited as satisfactory readings at the lower end of the temperature curve were not obtained regularly, due in some part to elements of cooling water breaking away from the main jet and contacting the thermocouple. In every traverse, however, the upper end of the temperature curve was apparent and this was used in conjunction with the calculated surface temperature and interface position to determine a temperature gradient not influenced by the effect on the thermocouple of the "winnowing" of the cooling water. This alternative temperature gradient is shown in fig. 18 for run 10 and all values of E so calculated are plotted in fig. 27b.

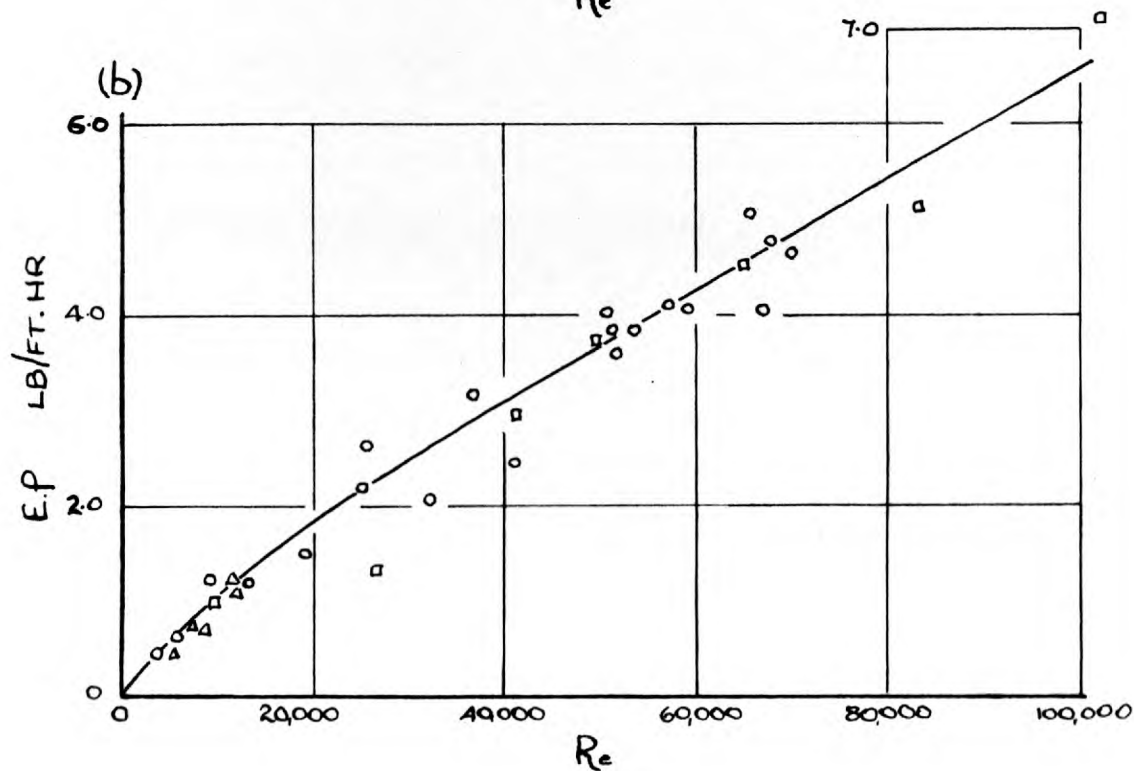
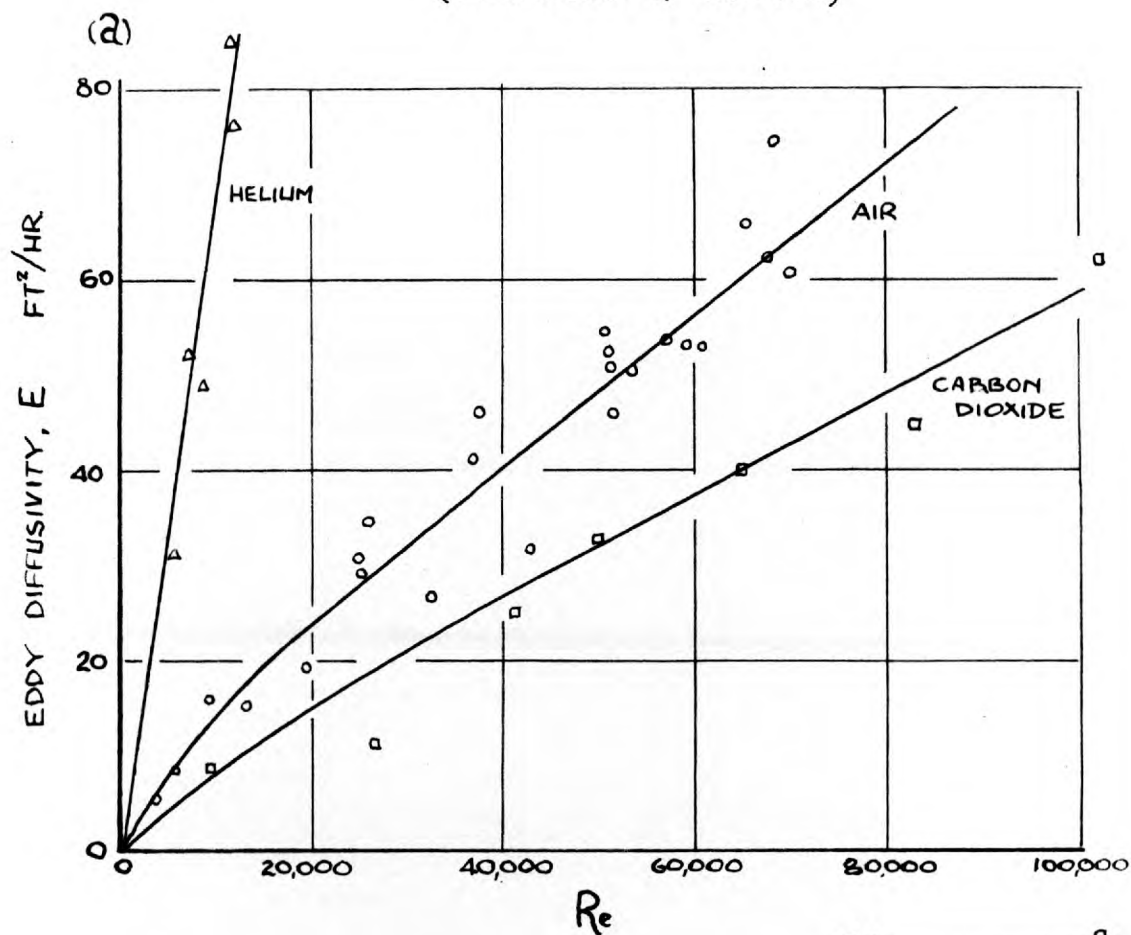
Comparison between the eddy diffusivity of this and other work. To the writer's knowledge the only other relevant data are those of Sherwood & Woertz, ref. 8. These workers used a vertical rectangular duct, 24" x 2.1" in cross-section, in which water and calcium chloride solution were allowed to flow down the opposite 24" walls. From pressure and hot wire anemometer measurements in the central gas stream and measurements of the mass transfer rate of the water, values of the eddy mass diffusivity and eddy viscosity were found, these being constant over any

cross-section and a function of the corresponding Reynolds number. The tests were run using air, carbon dioxide and helium in turn at room temperature. For comparison, the curve of Sherwood & Woertz for water diffusing through air is shown in figs. 27a and b. Bearing in mind the very different conditions in the two investigations the general agreement with Sherwood & Woertz is considered satisfactory.

The effect of noncondensable gases other than air, on the steam condensation rate. Figs. 28a and b show that the eddy diffusivity of steam mixed with a noncondensable gas, is inversely proportional to the steam-gas mixture density. From fig. 28b the eddy diffusivity of steam in a mixture with any noncondensable gas can easily be calculated.

However, the prediction of the heat transfer rate from the mixture to the cooling water is not easy and involves step-by-step integration of a tedious nature but a rough estimate of the effect of any particular noncondensable gas when compared with air, can be obtained. For example, the eddy diffusivity of steam in a mixture with carbon dioxide is about 66% of that in a mixture with air. It can therefore be assumed that the heat transfer rate from a carbon dioxide mixture will not be less than 66% of that from a steam-air mixture.

VARIATION OF E WITH Re FOR WATER VAPOUR
DIFFUSING THROUGH DIFFERENT GASES.
(SHERWOOD & WOERTZ)



VII DISCUSSION

Sources of error. The prime requirement for an investigation of this problem was a satisfactory means of measuring the rate of steam condensation. This necessitated the use of a device for measuring the bulk temperature of the complete cooling water jet over a number of cross-sections of the condenser as considerable variations can occur in bulk temperature "point" values at any fixed lengthwise position in the condenser (figs. 13 and 16). The arrangement adopted was not ideal but did permit temperature readings to be obtained over the whole jet. The development of the cooling water sampling system was accompanied by much trial and error as the author was not familiar with any previous work using a similar device. The range of conditions covered has not been as wide as was desired mainly because of limitations to the operating conditions under which the cooling water sampling device would function satisfactorily.

At a very early stage it was found that temperature readings on the cooling water could vary by 10°C or more by altering the back pressure on the scoop drain and thus the accuracy of the temperature measurements and, therefore, the whole results depended on the validity of the pressure

balancing technique. It was obvious that some means of registering pressure balance between the scoop and the duct was required with a greater sensitivity than that of a water manometer system. The latter experienced slow fluctuations of up to 4 or 5 ins. of water due to the splash and irregular flow pattern inside the scoop and this fluctuation trouble gave rise to latitudes of up to about 5°C when condensing steam with no air present and 2°C when using steam-air mixtures. The system finally adopted, whilst found somewhat fortuitously, did on examination seem plausible (pages 13 -14) and gave a degree of consistency not obtained previously. Any absolute errors of measurement that were incurred by the use of this technique will be consistent throughout and the relative effects of the variables investigated will not be altered.

Other sources of error can lie in the flowmeter and steam-air temperature readings. The flowmeters were regularly cleaned and before each run the steam and water manometer lines were well purged of any air present. A constant check on accuracy during the runs with steam-air mixtures was the comparison between the measured steam temperature and the saturation temperature corresponding

to the mixture ratio at the condenser pressure. With close agreement between these figures any error in flow rate must exist in both steam and air meters. Errors in the steam-air temperature readings will, of course, only affect the diffusivity data. It was feared that cooling water surface disintegration and splash onto the thermocouple probe would invalidate the readings by lowering the recorded temperature. Splash would be more prevalent nearer the water surface, i.e. at lower steam-air temperatures and would thus indicate a steeper temperature gradient than the true value. A comparison between measured temperature gradients and values in accordance with ^{the} λ calculated water surface temperature and position does bear this out (fig. 18) and more reliance is placed on eddy diffusivity figures complying with the latter gradient. It is considered that the errors involved in fixing the position of the lower end of the steam-air temperature curve by the calculation of cooling water surface temperature and position, will have a lower effect on the temperature gradient than that of splash on the thermocouple readings.

The cooling water surface temperature. Without knowledge of the exact mode of behaviour of the cooling water

surface temperature when condensing steam from a mixture with air, no exact theoretical analysis is possible. If the correct expression had been found for the surface temperature decay rate and the resultant equation for transient heat flow was capable of integration, the first series of tests using steam without premixed air would have been unnecessary for the analysis of the results using steam-air mixtures. As it was, with only a tentative assumption available for the surface temperature decay rate, the most that could be hoped for was a satisfactory means of correlating the experimental data. This required the use of a turbulence factor accommodating the effects of "gas" and water velocity on the cooling water temperature rise obtained from the results of steam condensation with no air present, as in this case the theoretical temperature rise can be predicted. It is thought that a more thorough study of the whole mechanism must involve a better understanding of the surface temperature behaviour of the cooling water although this is extremely complex.

An approximate calculation of the cooling water surface temperature (fig. 17) indicates an extremely quick decrease in the initial stages of condensation.

This emphasizes the need for a high degree of internal turbulence in the water jet and the necessity for a high cooling water surface area and a minimum jet thickness. The efficacy of the common practice of using circular cooling water orifices is questioned as these offer the smallest surface area/jet size ratio possible in the region where the greatest quantity of heat should be transferred. Thin slots would appear to offer the best means of introducing cooling water at the low initial velocities required.

Steam condensation in the presence of air. The effects of the initial steam-air ratio, steam-air velocity, cooling water jet thickness and velocity have been shown for what is otherwise a fixed condensing arrangement. The departure of the latter from normal practice was a necessary expedient but unfortunately it prevents any useful assessment of the heat transferred or steam condensed/unit volume as these figures bear little relation to the much higher rates obtainable with the usual multi-jet cooling water system dispersing throughout the steam-air mixture. "Seperation" of the steam-air and water phases does not, however, alter the general effect of steam/air ratio, steam-air velocity, cooling water velocity and

thickness on the condensation rate from that in a more intimate steam-air-water mixture.

The virtual independence of the percentage steam condensed/unit volume with initial "gas" velocity for a constant steam/air ratio is interesting but there is obviously some upper limit for which this is true. This limit is probably operative at a cooling water thermal ratio not reached in this work; the runs covered thermal ratios up to .48 and as far as is known this figure is not exceeded in practice. It would be unwise to assume the above identity holds for a steam-air velocity range much higher than that used (40 - 110 ft/sec.).

From a heat absorption standpoint, maximum utilisation of the cooling water is achieved with a minimum water entrance velocity although the net gain is small over the range investigated (fig. 9a). The pressure change across the condenser will suffer as a consequence of low water inlet velocities due to increased momentum drag but if water utilization is not important, its decrease with high water velocities may be outweighed by the improved pressure change across the condenser.

For the different steam-air mixtures used, the percentage of the steam condensed varied between 20 and

60% of the initial steam flow but this was sufficient to give a quantitative effect of the initial air content, fig. 23. The marked reduction in the heat transfer rate occurring with increasing air content, is shown to be progressive up to a maximum of 60% air, but the major influence is however exerted by the first few "percent" of air present. 5% was the lowest air content used, but much less than this will effectively blanket the cooling water surface.

Eddy diffusivity. The present results substantiate the data of Sherwood & Woertz on the effect of Reynolds number on the eddy diffusivity. This increase in diffusivity with Reynolds number accounts for the proportionality between steam condensation rate and steam-air entrance velocity. The correlation of E_p with Re , fig. 28b, shows the advantage of working the condenser at as low a pressure as possible. Such an arrangement would involve higher steam-air mixture velocities, which themselves are conducive to rapid heat transfer rates, but also the heat transfer rate is augmented by the increased eddy diffusivity associated with the lower mixture density.

The effect of other noncondensable gases compared with air, on the basic heat and mass diffusion rates is

easily found from fig. 28b but the overall effect on the steam condensation rate is very complicated. The percentage change in condensation rate will, however, be always less than the percentage eddy diffusivity change.

CONCLUSIONS

In a parallel-flow jet condenser, condensing steam from a turbulent mixture with air:-

1. The heat transfer rate has been presented in terms of the cooling water temperature rise on parameters developed from the transient heat flow equation for the cooling water, using a tentative assumption for the surface temperature decay relation.

Allowance has been made for the effect of steam-air mixture velocity, cooling water jet velocity and jet thickness on the condensation rate.

2. The steam condensed/unit volume is proportional to the steam-air inlet velocity over the range investigated, (40-110 ft./sec.), other factors being constant.

3. Water injected with a low initial velocity and in a fine state of dispersion is most efficiently utilized from a heat absorption stand-point.

4. Increased air content in the steam causes a progressive reduction in the heat transfer rate. The reduction is considerable with the first 5% of air, further increase in air content produces a less pronounced change.

5. Values of eddy diffusivity in the steam-air mixture confirm the work of Sherwood and Woertz, ref. 8, and examination of the latter results shows the influence of non-condensable gases other than air. It is also seen that the rate of steam condensation is improved by operation at low condenser pressures.

SUGGESTIONS FOR FURTHER WORK

This thesis is offered as some groundwork to the complex problem of heat transfer in a jet condenser when steam is mixed with air. Much more data than that given is required on the condensation process before an adequate understanding is reached for practical application.

The problem of pressure change through the condenser has not been touched and this can form the subject of a new line of research as it is necessary to link up the pressure change relations with the condensation rate data before any attempt at design can be made.

A theoretical analysis of the cooling water surface temperature relation would be useful, as it is considered that this provides the key to a better knowledge of the condensation mechanism discussed in this thesis.

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APPENDIX I

The surface temperature and mean temperature rise of the cooling water when condensing steam in the presence of air

The lack of any exact knowledge on the behaviour of the surface temperature necessitated the adoption of some relation which would:-

- a) appear a reasonable assumption.
- b) comply with the boundary conditions.
- c) be capable of integration in the general equation of transient heat flow.

An exponential decay function was considered to satisfy the above requirements and two alternatives were available.

$$(i) \theta_{surf} = \theta'_m + (\theta_s - \theta'_m).e^{-k\tau}$$

This satisfies the actual boundary conditions:-

$$\text{when } \tau = 0, \theta_{surf} = \theta_s$$

$$\tau = \infty, \theta_{surf} = \theta'_m$$

$$(ii) \theta_{surf} = \theta_m + (\theta_s - \theta_m).e^{-k\tau}$$

This satisfies the theoretical boundary conditions, the effect of turbulence being neglected:-

$$\text{when } \tau = 0, \theta_{surf} = \theta_s$$

$$\tau = \infty, \theta_{surf} = \theta_m$$

This first equation was thought to be the most applicable and was inserted in the general equation of transient heat

flow for a slab with zero initial temperature and with a surface temperature, $f(\gamma)$.

The lamina is considered to be of thickness, $2l$, and subject to the temperature, $f(\gamma)$ on both surfaces:-

Then

$$\theta = \frac{2}{l} \sum_{n=0}^{\infty} e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4l^2}} \cdot \cos(2n+1) \frac{\pi x}{2l} \left\{ \frac{(2n+1) \pi \alpha_w (-1)^n}{2l} \int_0^{\gamma} e^{-\frac{\alpha_w (2n+1)^2 \pi^2 \lambda}{4l^2}} \cdot f(\lambda) \cdot d\lambda \right. \dots\dots\dots (15)$$

$$\text{Now } f(\lambda) = \theta'_m + (\theta_s - \theta'_m) e^{-k\tau}$$

Substituting in (1) and integrating:-

$$\begin{aligned} \theta' = & \theta'_m \left(1 - \frac{4}{\pi} \sum_{n=0}^{\infty} \frac{(-1)^n}{2n+1} \cdot e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4l^2}} \cdot \cos(2n+1) \frac{\pi x}{2l} \right. \\ & + (\theta_s - \theta'_m) 4 \sum_{n=0}^{\infty} \frac{(-1)^n (2n+1)}{\pi^2 (2n+1)^2 - 4kl^2} \left(e^{-k\tau} - e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4l^2}} \right) \cos(2n+1) \frac{\pi x}{2l} \end{aligned} \dots\dots\dots (16)$$

Equation (2) can be conveniently written as

$$\theta' = \theta'_1 + \theta'_2$$

$$\text{And } \theta'_m = \frac{1}{2l} \int_{-l}^{+l} \theta' dx = \frac{1}{2l} \int_{-l}^{+l} \theta'_1 dx + \frac{1}{2l} \int_{-l}^{+l} \theta'_2 dx$$

$$\therefore \theta'_m = \theta'_{m1} + \theta'_{m2} \dots\dots\dots (17)$$

$$\text{And } \theta'_{m1} = \frac{\theta'_m}{2l} \int_{-l}^{+l} \left(1 - \frac{4}{\pi} \sum_{n=0}^{\infty} \frac{(-1)^n}{2n+1} \cdot e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4l^2}} \cdot \cos(2n+1) \frac{\pi x}{2l} \right) dx$$

$$\therefore \theta'_{m1} = \theta'_m \left(1 - \frac{2}{\pi} \sum_{n=0}^{\infty} \frac{1}{(2n+1)^2} \cdot e^{-\frac{\alpha_w (2n+1)^2 \pi^2 \tau}{4l^2}} \right) \dots\dots\dots (18)$$

Equation (18) gives the mean temperature of the slab when exposed to a constant surface temperature θ_m' . (cf. equation (1) and fig. 6)

$$\theta_{m_2}' = \frac{\theta_s - \theta_m'}{2L} \int_{-L}^{+L} 4\pi \sum_{n=0}^{\infty} \frac{(-1)^n (2n+1)}{\pi^2 (2n+1)^2 - \frac{4kl^2}{\alpha_w}} \left(e^{-k\tau} - e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4l^2}} \right) \cos(2n+1) \frac{\pi x}{2L} dx$$

$$\therefore \frac{\theta_{m_2}'}{\theta_s - \theta_m'} = \sqrt{\frac{\alpha_w}{kl^2}} \cdot \tan \sqrt{\frac{kl^2}{\alpha_w}} \cdot e^{-\left(\frac{kl^2}{\alpha_w}\right) \left(\frac{\alpha_w \tau}{l^2}\right)}$$

$$- 8 \sum_{n=0}^{\infty} \frac{1}{(2n+1)^2 \pi^2 - \frac{4kl^2}{\alpha_w}} \cdot e^{-\frac{\alpha_w \pi^2 (2n+1)^2 \tau}{4l^2}} \dots\dots\dots (19)$$

Equation (19) gives the mean temperature of the slab when exposed to a surface temperature $(\theta_s - \theta_m') \cdot e^{-k\tau}$

The graphical relation between $\frac{\theta_{m_2}'}{\theta_s - \theta_m'}$ and $\frac{\alpha_w \tau}{l^2}$ is identical with fig. 5. With the above coordinates it was hoped to accommodate the effect of turbulence on the temperature rise, but inspection of the parameter shows that it increases with turbulence and is thus unsuitable as a means of presenting the experimental results. Recourse was, therefore, made to the second alternative assumption for the cooling water surface temperature decay rate, $\theta_{surf} = \theta_m + (\theta_s - \theta_m) \cdot e^{-k\tau}$, (page 33).

APPENDIX II

Steam condensed from a steam-air mixture in terms of the temperature rise of the cooling water.

A heat balance between the cooling water and the steam-air mixture gives:-

$$W_w t_{wo} + W_{so} H_{so} + W_a C_p t_o = (W_w + W_c) t_w + W_s H_s + W_a C_p t$$

$$\therefore W_c = \frac{W_w(t_w - t_{wo}) - W_{so}(H_{so} - H_s) - W_a C_p(t_o - t)}{H_s - t_w}$$

assuming
specific
heat of
water
= 1.

Now $t_w - t_{wo} = \theta'_m$, by definition

$$\therefore W_c = \frac{W_w \cdot \theta'_m}{H_s - t_w} \left[1 - \frac{W_{so}}{W_w \cdot \theta'_m} \left\{ H_{so} - H_s + \frac{W_a}{W_{so}} \cdot C_p(t_o - t) \right\} \right] \text{ lb/hr.}$$

.....(20)

As t and H_s are not obtained experimentally, equation (20) is solved by successive approximation as follows:-

- (i) Assuming no air content in the steam, the condensation rate is calculated from the relation

$$W_c = \frac{W_w \theta'_m}{H_s - t_w}$$

- (ii) The residual steam flow, $W_s = W_{so} - W_c$ lb/hr. and the corresponding air/steam ratio, $\frac{W_a}{W_s}$, follows.

- (iii) The average steam-air mixture temperature, t , is found from fig. 19a. H_s is then determined.

- (iv) W_c is now calculated from equation (20).

(v) If W_c differs from that calculated in (i) by more than 1%, the procedure is repeated with the new value of W_c .

APPENDIX III

Eddy heat transfer in a fluid of varying density.

Reynolds, ref. 11, conceived that at any point in two-dimensional turbulent fluid system, the flow was composed of an average fluid velocity V on which were superimposed fluctuating velocities v'' and u'' . The diffusion is in the present case towards the wall and heat is transferred from a high to a low heat density level by the fluctuating velocity, u'' .

By Prandtl's mixing length theory, ref. 12.

$$u'' = v'' = l \frac{dv}{dy}$$

The heat density = heat/unit volume = ρH

$$\text{and } (\rho H)'' = l \frac{d(\rho H)}{dy}$$

The heat transferred, $q = -(\overline{\rho H u''})$, where $\overline{\quad}$ denotes the time average of the instantaneous fluctuations.

$$\therefore q = -l^2 \frac{dv}{dy} \frac{d(\rho H)}{dy}$$

But $E = l^2 \frac{dv}{dy}$ by definition.

$$\therefore q = -E \frac{d(\rho H)}{dy}$$

The eddy thermal diffusivity is usually denoted by E_H as investigations by different workers have shown that the diffusivities of heat mass and momentum are not necessarily the same.

$$\text{i.e. } q = -E_H \frac{d(\rho H)}{dy}$$

APPENDIX IV

Eddy mass transfer of steam through air to a receiving surface.

The basic equation of mass diffusion is

$$\omega = D \frac{dc}{dy} \quad \text{where } c \text{ is the concentration (wt/unit volume) of the diffusing component.}$$

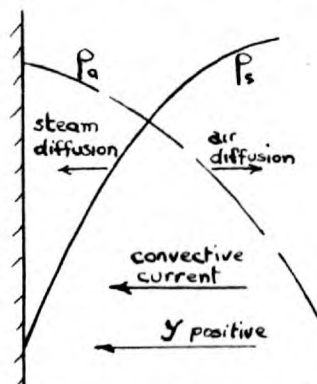
D is the diffusion coefficient.

This is known as Fick's law.

It can be shown in a similar manner to that of appendix III that the same formula is equally applicable to turbulent diffusion providing the coefficient is modified.

Thus $\omega = E_M \frac{dc}{dy}$ where E_M is the eddy mass diffusivity.

For the case of a vapour (steam) diffusing through a non-condensable gas (air), the density gradient of the "inert" component will cause it to diffuse in opposition to the vapour diffusion. As the air is not removed by



Steam and air density gradients

condensation and is in equilibrium in a direction normal to the surface of steam condensation, its diffusive flow must be balanced by an equal and opposite convective flow, velocity u . This convective

flow augments the vapour diffusive rate and the air and steam diffusion equations of mass rate are:

$$w_a = -E_m \frac{dp_a}{dy} + p_a \cdot u_i = 0 \quad \dots\dots(21)$$

$$w_s = -E_m \frac{dp_s}{dy} + p_s \cdot u_i \quad \dots\dots(22)$$

From (21), $u_i = \frac{E_m}{p_a} \cdot \frac{dp_a}{dy}$

$$\therefore w_s = -E_m \frac{dp_s}{dy} + E_m \frac{p_s}{p_a} \cdot \frac{dp_a}{dy}$$

$$\text{i.e. } w_s = -E_m \left[\frac{p}{p_a} \cdot \frac{dp_s}{dy} - \frac{p_s}{p_a} \cdot \frac{dp_a}{dy} \right]$$