THE UNIVERSITY OF GLASGOW

CONVECTIVE HEAT TRANSFER IN CROSS-FLOW TUBE ARRANGEMENT.

A THESIS SUBMITTED FOR THE DEGREE OF Ph.D.

BY

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JANUARY 1950

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Part II - Experimental Investigation.

SYNOPSIS

An experimental study is presented of point - to-point variation in heat transfer around a tube placed across a hot air stream and water cooled internally. A single tube is first examined and then individual tubes in tube banks.

In the first part, methods and results of previous workers are reviewed; attention had been confined to variation at outer surface of single tube. The form of such variations are established. No comparable work has been carried out on the variation of heat transfer at inner surface, nor has there been any extension to individual tubes in banks.

The second part describes apparatus designed specifically for investigation. The outstanding feature is a fluted core which, placed in tube, allows measurement of actual heat transfer at inner surface for each 20^{o} angular interval round tube. Calibrations and preliminary tests are detailed.

In the third part experimental results are given and discussed. For single tube mean heat transfer coefficients are given, and also the variation in wall surface temperature around tube. The variations in heat transfer round inner surface are given and shown to diverge markedly from established variation for outer surface. Analysis, which allows for circumferential heat flow induced by wall temperature variation, gives correlation between inner and outer surface variations. The effects of varying conditions on relationship between inner and out~~w~~ variations in heat transfer coefficients are examined. Individual tubes in banks are similarly treated. Mean heat transfer coefficients for banks and for different rows in bank are derived.

Temperature and heat transfer variations explain increasing effectiveness of rear portion of tube from first to third row and emphasise exceptional conditions around second row tube. Results are summarised and conclusions given.

INTRODUCTION

The work described in this thesis forms the first part of a major investigation on natural circulation in boiler tube banks. To enable an assessment to be made of the variation in heat reception and therefore of natural circulation from row to row, detailed knowledge is required of the variation in heat transfer rate from row to row and also round any individual tube in a bank. A study of available data revealed that there were many gaps which could only be bridged by an experimental investigation using new techniques, in particular the measurement of local tube temperatures and heat reception rates.

In addition, attention had been drawn to the persistency of failures in the second row tubes of a water tube boiler, and it was decided to examine carefully the local variation in convective heating which must have given rise to these failures.

A hydrodynamic approach to the second row tube problem has already been made and, as a result, there is experimental evidence that the pressure gradients have their maximum values on the surface of the second row tubes. The absence of the knowledge necessary to translate this evidence into terms of heat transfer and temperature gradients have rendered it inconclusive.

The present work represents an approach to the problem from the heat transfer angle. It has been preceded by a survey of the published information relating to variations in temperature and in heat transfer coefficients round a tube in cross flow. It was found that the case of heat transfer from a gas stream to a cylinder in crossflow had been examined by several investigators. They had found that the heat transfer coefficient between the gas and the external surface varied locally round the cylinder and had established the form of that variation. Various experimental methods had been used but in no case were they carried far enough to allow of correlation between the variations in heat transfer rates at the inner and outer surfaces of a tube and the tube wall temperatures, nor of assessing the circumferential heat flow in the tube metal. The variations around a tube placed in a tube bank had received little attention.

It appeared necessary that further work should be undertaken with a view to filling gaps in existing information, to explaining certain anomalies, and to extending the fundamental knowledge to the detailed actions around the tubes in a tube bank.

The need for such knowledge is made more urgent by the ever increasing demands in heat transfer rates in boiler practice. A few years ago a heat intake of 120,000 B.ThU. per hour per square foot of tube surface was considered high for a fire-row tube in boiler practice, now 200,000 is accepted, and if modern advances in combustion technique are to be fully utilised much higher rates still must be faced. When it is remembered that these figures are mean rates for the tube then a detailed knowledge of the point-to-point variations around the tube assumes a new importance. Undoubtedly greater attention to temperature stresses will be demanded and these cannot be satisfactorily assessed without precise information on the temperature gradients in the tube wall. Thus any work which will advance existing knowledge on these important aspects is fully justified.

PART I - REVIEW

PRESENTATION AND CORRELATION OF HEAT TRANSFER RESULTS

The rate of heat transfer between two fluids separated by a solid wall is dependent upon the resistance to heat flow offered by the fluid on either side of the wall and the conductivity of the wall. For the case of a gas flowing past a bank of metal tubes through which a vapour or liquid is passing, the overall heat transfer coefficient is largely controlled by the resistance of the gas film. As this case is commonly met with in practice, many experiments have been carried out to obtain values of the heat transfer coefficient between gas and tube wall for various gas flow conditions.

The results of such tests are presented in many forms, and in the past empirical equations were often used; such equations had a very limited application.

The large number of independent variables affecting the convective heat transfer in any experiment have made the use of dimensionless groups universal in reporting results. In the case of forced convection, where natural convection effects are small, it can be shown by dimensional analysis that /

$$\frac{h_c d}{k} = f\left(\frac{\rho d v}{\mu}, \frac{C_p \mu}{k}\right)$$

$$\text{or Nu} = f\ (\text{Re, Pr})$$

where h_c = convective heat transfer coefficient gas to metal

d = Diameter of tube

k = Conductivity coefficient of the gas

ρ = density

v = velocity

μ = viscosity

C_p = Specific Heat

Nu = Nusselt Number = $\frac{h_c d}{k}$

Re = Reynolds " = $\frac{\rho d v}{\mu}$

Pr = Prandtl " = $\frac{C_p \mu}{k}$

That is, the Nusselt Number is a function of the Reynolds Number and the Prandtl Number, the two latter numbers being dependent respectively on the velocity distribution and the temperature distribution in the gas stream.

As Pr varies but slightly over a wide range of temperature for gases of the same atomic number, this equation may be reduced to

$$\text{Nu} = f\ (\text{Re})$$

which, when expressed as a power function, gives

$$\text{Nu} = a(\text{Re})^n$$

Thus the results of experiments using air or any other diatomic gas may be presented and correlated in this form.

The selection of a temperature at which to evaluate the physical properties of the gas in this equation has led to some difficulty in correlation. Some workers base their values of k, μ and ρ on the bulk temperature of the gas and others on the mean tube temperature. In this thesis, as in most recent works, k and μ are evaluated at a gas film temperature t_f, defined as

$$t_f = \tfrac{1}{2}(t + t_{s_m})$$

that is, the mean temperature between the gas stream and the tube; and ρ and V are combined in G, the mass flow of air per unit area based on the minimum flow area past the tube.

A further problem in correlation is the difference between the curves of different authorities for the physical properties used in evaluating results. Graphs of the variation of k and μ with temperature may differ by more than 5%. Throughout this paper, the values of the physical properties of air as given by Keenan and Kaye[1] are used.

SECTION A - SINGLE TUBE EXPERIMENTS

(1) Mean Convective Heat Transfer Coefficient

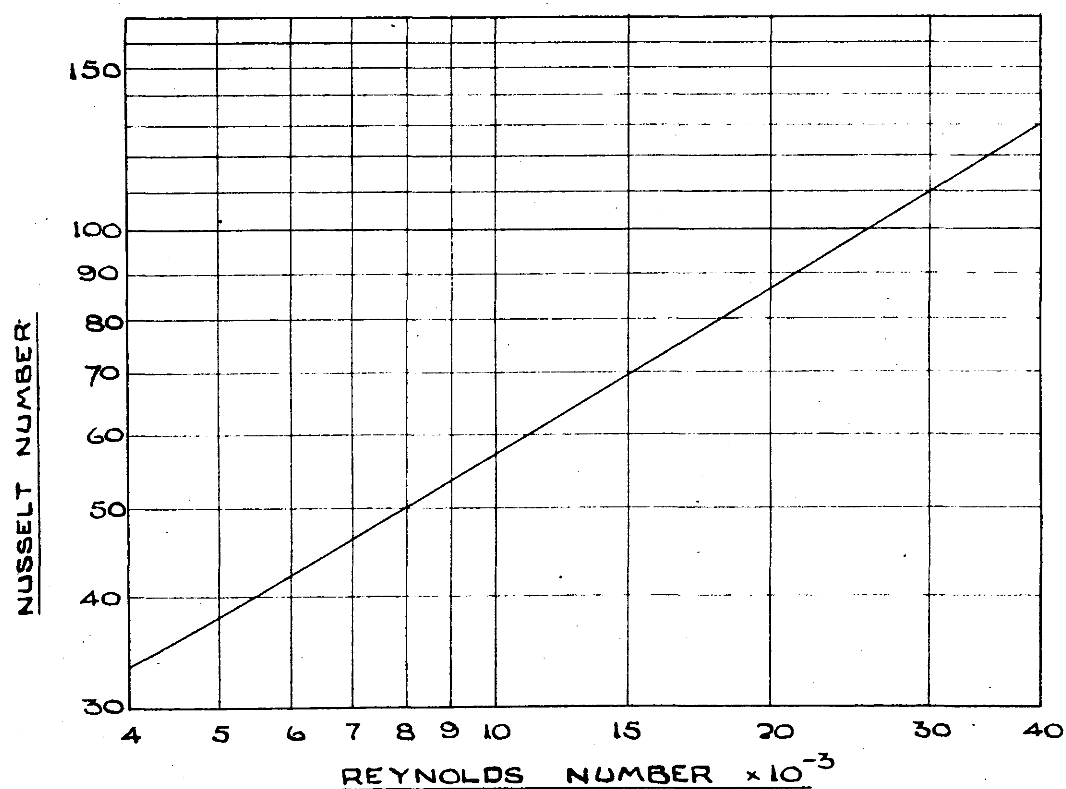
The majority of experimenters have used a single heated cylinder mounted across a duct through which a stream of air was passed. The heat transfer coefficient for any air flow condition was obtained by measuring the rate of heat loss from the tube surface, the mean temperature difference between the surface and the air, and the external surface area of the cylinder.

Various methods have been adopted to obtain the heat loss from the cylinder. Hughes[2] and others have taken the weight of steam condensed in the cylinder; Hilpert[3], Small[4] and Griffith and Awbery[5] have measured the electrical input to heaters mounted in the cylinder; while King[6] and Hilpert[3], for low values of Reynolds Number, have used an electrically heated wire as the 'cylinder'. Reiher[7], however, directly measured the heat received by water passing through a tube placed across a stream of hot air. The convective heat transfer coefficient was obtained after correcting for radiant heat transfer.

The results of many experiments have been correlated by Fishenden and Saunders[8], Schack[9] and more recently McAdams[10]. The recommended curve of McAdams is shown in Fig. 1 as a plot of Nu to a base of Re, the various experimental values agreeing within 20%. The results of

FIG. 1

NUSSELT NUMBER TO REYNOLDS NUMBER FOR SINGLE TUBE AS GIVEN BY McADAMS



the individual experiments, however, agree much more closely than this, the wide variation only being apparent when relating the results of different experimenters. This fairly wide variation in the general correlation cannot be completely accounted for by the possible differences in the values of the physical properties used in the evaluation of the results. It has been shown, 5, 7, that the degree of forced turbulence in the air stream flowing over a tube greatly influences the heat transfer. As this turbulence will vary in the different experiments, good agreement between the results cannot be expected, since the value of the Reynolds Number is not affected by the turbulence.

It may be seen that the slope of the curve in Fig. 1 is not quite constant. Hilpert[3] presented his results as a curve with a constant slope between values of Re 4 - 40, 40 - 4,000 and 4,000 - 40,000. There is a slight increase in the slope of the Nu - Re curve at each of these points.

In all tests the air flow has been at right angles to the tube. It has been shown[11] that the line of approach has little effect on the heat transfer unless the angle between the tube axis and the direction of flow is less than 60°.

(2) Temperature Variations round Tube:

In his classic paper, Reiher[7], has shown that for a tube receiving heat in a cross flow air stream there is a marked variation in surface temperature round the tube, with a maximum value at the upstream point and a minimum diametrically opposite, Fig. 2

Krujilin and Schwab[12], however, have shown that for a steam heated cylinder in a cold air stream, the temperature is greatest at the sides and least at the front and rear, Fig.3.

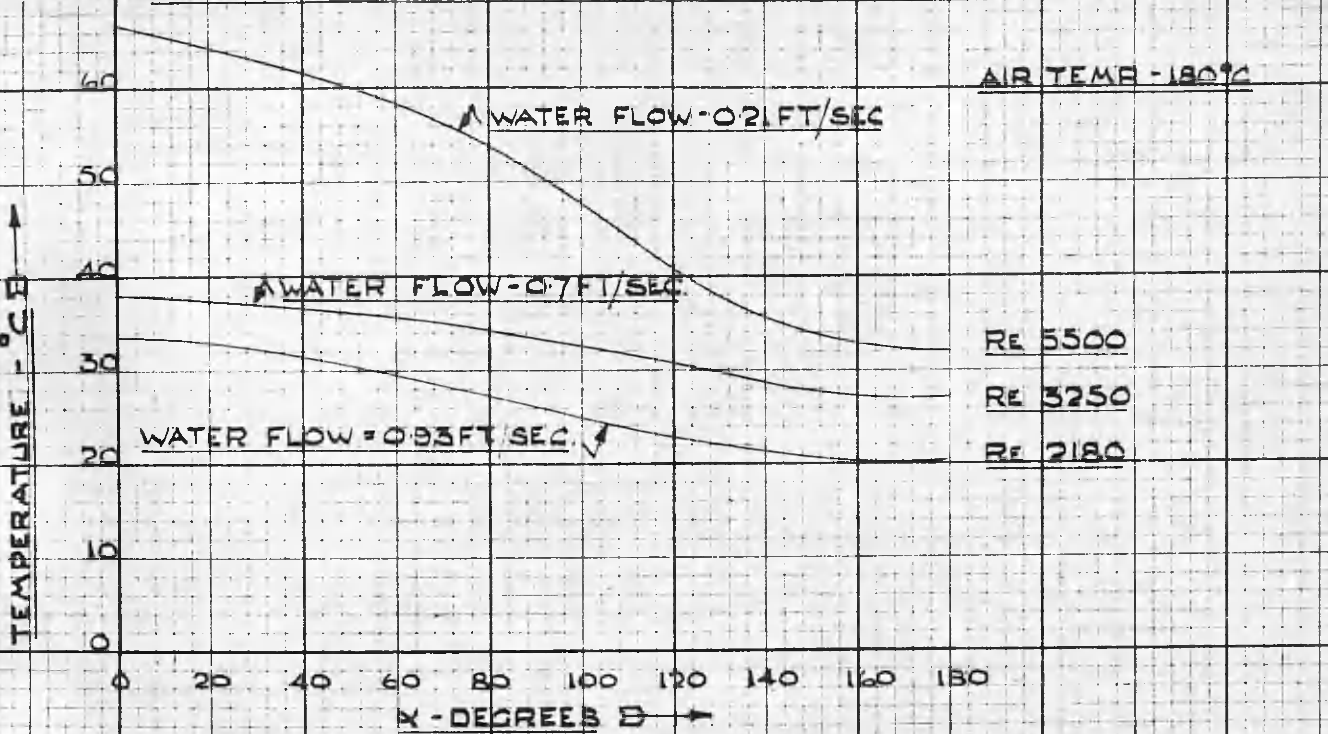
From such experimental evidence of temperature variations around a tube, it can be deduced that the rate of heat transfer taking place at different points around the tube cannot be constant. This may also be inferred from consideration of the air flow pattern around a tube, as the flow at the surface of the tube changes as it passes from the front to rear.

(3) Variation of Convection Heat Transfer Coefficient around Tube:

A number of experiments have been made to measure this variation in heat transfer around a cylinder and a brief summary of the experimental methods used will now be given.

FIG. 2

TEMP. PLOT BY REIHER FOR WATER COOLED TUBE



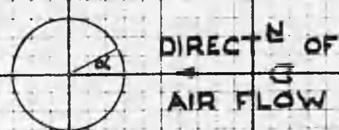
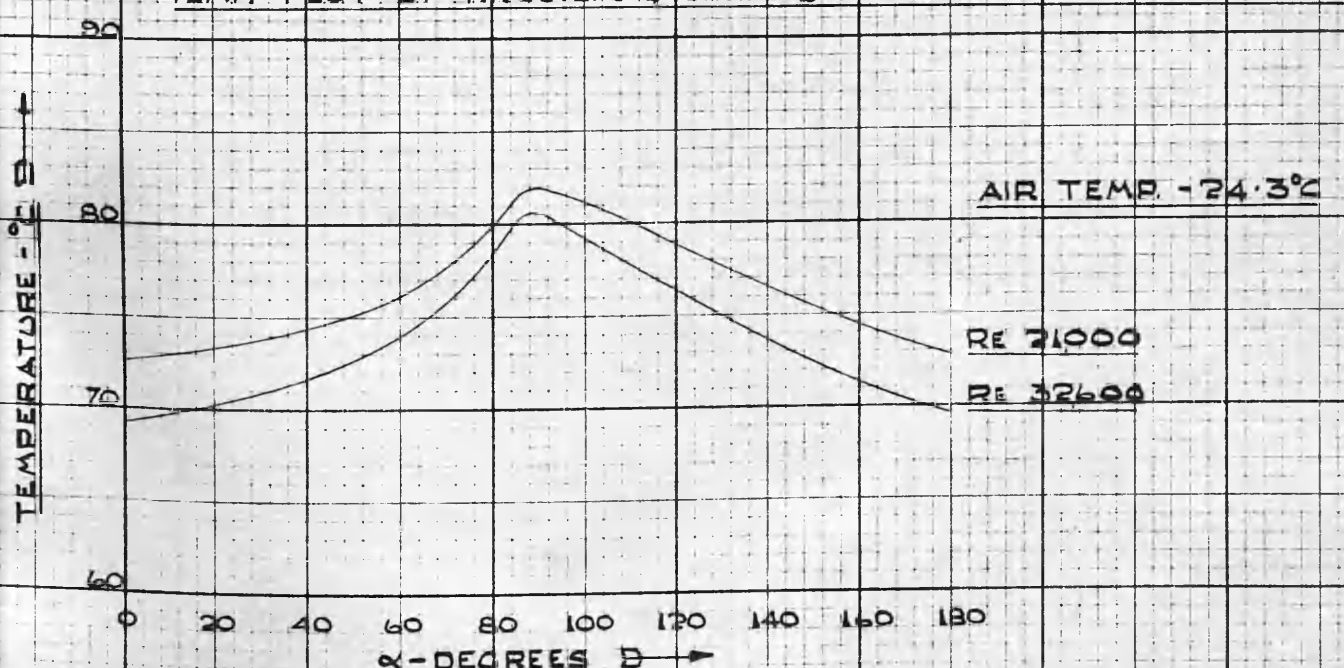


FIG. 3

TEMP. PLOT BY KRUJILIN & SCHWAB FOR STEAM HEATED CYLINDER



(a) Early Experiments: By measuring the heat loss from an electrically heated metal strip set into the wall of an ebonite cylinder, when the cylinder was placed in a cold air stream, Fage and Falkener[13], Small[14] and Kirpitchev[15] have shown that the rate of cooling of the strip varied as the cylinder was rotated through small angular displacements. These experiments all show a maximum value of heat transfer at about 40° from the upstream point and a minimum at the front and sides of the tube. The results of these tests cannot be applied generally to heated tubes, as the temperature conditions in the air stream around a locally heated cylinder will differ from those around a completely heated surface.

(b) Indirect Experiments: Circumferential variations in heat transfer have been deduced by Lohrisch[16], who related measurements of the rate of diffusion of a gas from the surface of a tube to values of heat transfer. Winding and Cheney[17] measured the change in dimensions of a cylinder of naphthalene placed in a wind tunnel, and by an analogy between mass transfer and heat transfer, converted their results to values of heat transfer.

Heat transfer values have also been presented by Klein[18], who used cylinders of ice placed in a hot air stream. The local variations in the rate of melting of the ice were used to calculate the variations in heat transfer coefficient for the surface of the cylinder.

The results of these three experimenters must be applied with caution, as the air flow pattern around the various cylinders used would be different from that around a heated or cooled tube, and thus it is to be expected that the heat transfer would also differ.

(c) <u>Using Heated Cylinders</u>: Small[4] carried out experiments using a heated metal cylinder in which an insulated thermopile was inserted in an axial slot. The tube was placed in an air stream and by rotating the tube, readings from the thermopile at different angular positions were obtained. By relating the mean value of these readings to the mean heat transfer to the cylinder, the variations in heat transfer around the cylinder were obtained.

Tests made by Paltz and Starr, reported by Drew and Ryan[19] give values of the heat transfer for axial strips around the inside of a 3.2 ins. outside diameter and $\frac{1}{8}$ in. thick brass tube for one value of Reynolds number. The results were obtained by measuring the condensation of steam in slots on the inside of the tube when cold air was blown past the outer surface of the tube. The assumption is made that the tube wall is at a uniform temperature corresponding to the saturation temperature of the steam, and the results are presented as heat transfer rates from the outer surface. This assumption can hardly be accepted in the light of the temperature distribution as given in Figs. 2 and 3.

Krujilin and Schwab[12] and Krujilin[20] deduce the heat transfer at the outer surface of an internally heated tube by calculating the conduction through the tube wall from experimentally obtained values of the surface temperature and an assumed constant internal surface temperature. The justification for taking the internal surface temperature as constant in this case is in doubt, as Hilpert[3] has measured small variations of internal surface temperature on a similarly heated tube.

A very complete set of results have been published by Schmidt and Wenner[21] who used a steam heated cylinder, in the side of which, in a small insulated slot, an electric element was inserted. The heat transfer at any point was obtained by measuring the electrical input to maintain the element at a constant temperature. A small error is to be expected in these results as they are based on readings of a tube at constant external temperature

(d) General: A selection of the results of these experimenters, covering the range of Reynolds Number being considered, are shown in Fig.4. In each case the variation in Nusselt Number round the half circumference of the tube

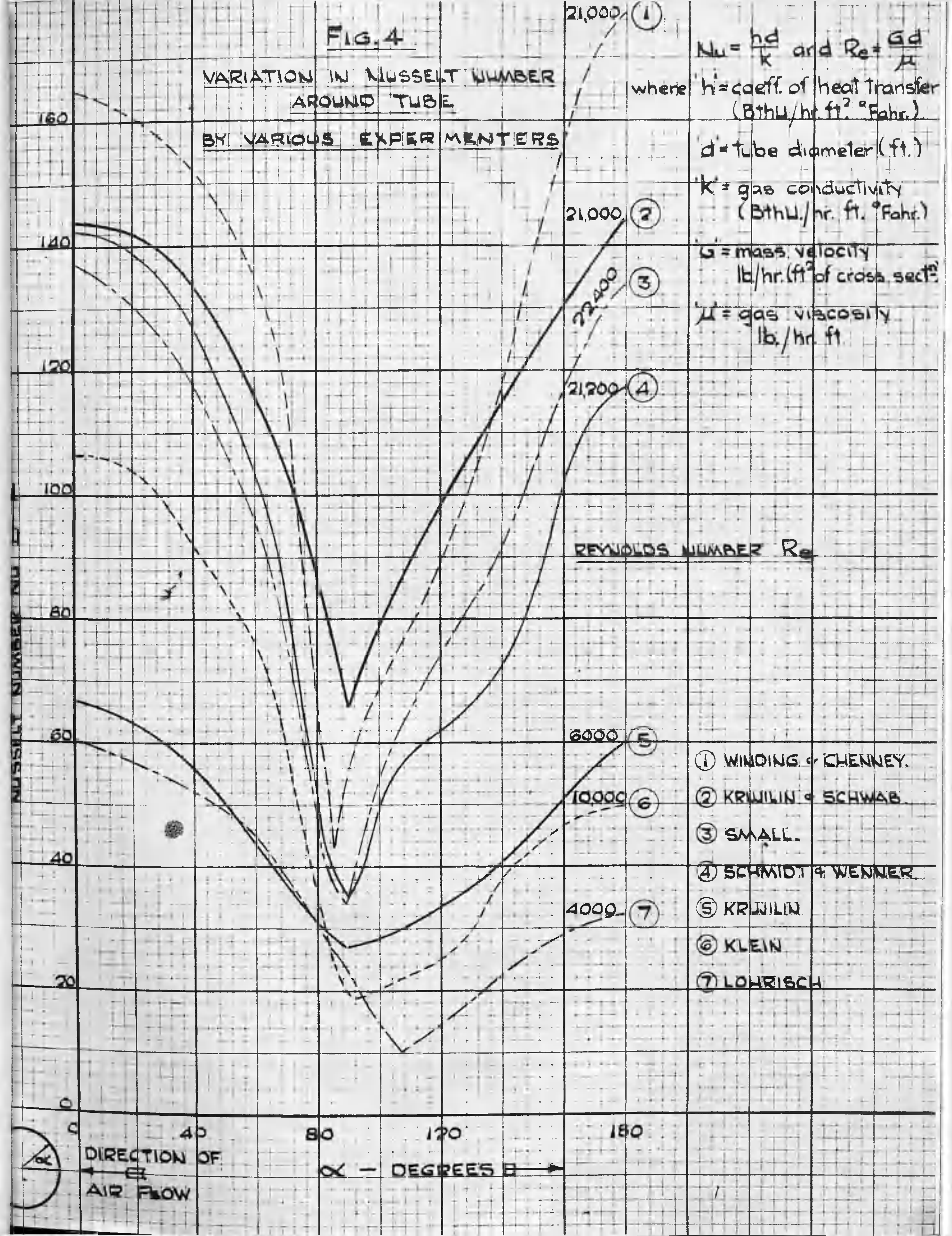


FIG. 4
VARIATION IN NUSSELT NUMBER AROUND TUBE
BY VARIOUS EXPERIMENTERS
Nu = hd/k and Re = Gd/μ
where h = coeff. of heat transfer (Bthu/hr. ft.² °Fahr.)
d = tube diameter (ft.)
k = gas conductivity (Bthu./hr. ft. °Fahr.)
G = mass velocity lb/hr.(ft² of cross sectn.)
μ = gas viscosity lb./hr. ft.
REYNOLDS NUMBER Re
21,000 (1)
21,000 (2)
22,400 (3)
21,200 (4)
6000 (5)
10,000 (6)
4000 (7)
(1) WINDING & CHENNEY.
(2) KRUJILIN & SCHWAB.
(3) SMALL.
(4) SCHMIDT & WENNER.
(5) KRUJILIN
(6) KLEIN
(7) LOHRISCH
NUSSELT NUMBER Nu
160
140
120
100
80
60
40
20
0
0
40
80
120
180
α
DIRECTION OF AIR FLOW
α — DEGREES

is shown to a base of angle measured from the front of the tube.

The general shape of these curves has been confirmed using optical methods by Joukovsky, Kirejew and Schamschew[22] and it can also be related to the characteristics of the gas flow pattern round the tube. The rate of heat transfer is dependent upon the air film thickness and the degree of turbulence at the tube surface and varies from a maximum at the front to a minimum where the flow lines break away from the surface. Thereafter there is an increase due to the vortices formed towards the rear of the tube.

It should be noted that the various results in Fig.4 show that the value of Nu at the rear of the tube, 180°, increases much more rapidly with increasing Re, than does the value at the front of the tube, 0°.

B - TUBE BANK EXPERIMENTS

1. Mean Heat Transfer coefficient:

The results of experimental investigations of the heat transfer between air and banks of tubes are, as in the case of single tubes, generally presented as a plot of Mean Nusselt Number to Reynolds Number. A comparison of the various results shows that the rate of heat transfer varies greatly with the tube arrangement and the number of rows of tubes present.

Extensive tests using banks of tubes ten rows deep have been carried out by Huge[23] and Pierson[24] to investigate the effect of tube arrangement on heat transfer. Their results have been correlated by Grimison[25] and fair agreement is found with the results of other experimenters. Kuznetzoff and Lokshin[26] have also carried out tests with varying tube spacing, their results are found to be somewhat lower than those given by Grimison, but show the same general trend.

The results of the many tube bank tests have been correlated by Lander[27] and Fishenden and Saunders[28]. To permit of easy application to design problem, factors that are dependent on the tube configuration and Re are given, to be applied in an equation of the form.

$$Nu = a(Re)^n \times \text{factor}.$$

In general it is found that the rate of heat transfer with "in-line" tube banks is lower than with staggered arrangements. In the case of the "in-line" tubes h_m is increased by placing the tubes closer together across the flow path, but only slight alteration in the value of h_m is noted when the spacing in the direction of flow is altered. For staggered banks, however, the heat transfer may be considerably increased by reducing the row to row distance, and only slightly varied by altering the distance between the tubes in the rows.

To assess the effect of the number of rows of tubes present in a tube bank, Reiher[7] carried out tests using banks containing different numbers of rows. From these tests a mean value of h_m for each bank was calculated and the value was found to increase with the number of rows. Similar effects are shown by Grimison[25].

Griffith and Awbery[5] and Winding[29] have shown that the mean heat transfer rate in a row increases from the first to the third row and then remains nearly constant for further rows. These experiments were made using a heated test tube placed in a bank of unheated tubes, the heat transfer coefficient being based on the approach air temperature.

As in the case of a single tube, it has been shown [30, 31], that the angle between the tube axis in a bank and the direction of air flow has little effect on heat transfer values.

2. Variation of Heat Transfer Coefficient around Tube.

Only one series of curves is available in the literature showing how the variation in heat transfer around a tube is affected by the position of the tube in a tube bank. These curves refer to a tube in the 5th row of a staggered tube bank and were obtained by Winding and Cheney[17] using their technique of measuring the change in dimensions of a naphthalene cylinder. Their curves are given in Fig.5 and show a very high rate of heat transfer at a point some 120° from the upstream point of the tube. It should be noted that a maximum value at this point has been shown to be present in single tube experiments when the value of Re is above 150,000 ([20, 21])

C - REVIEW SUMMARISED

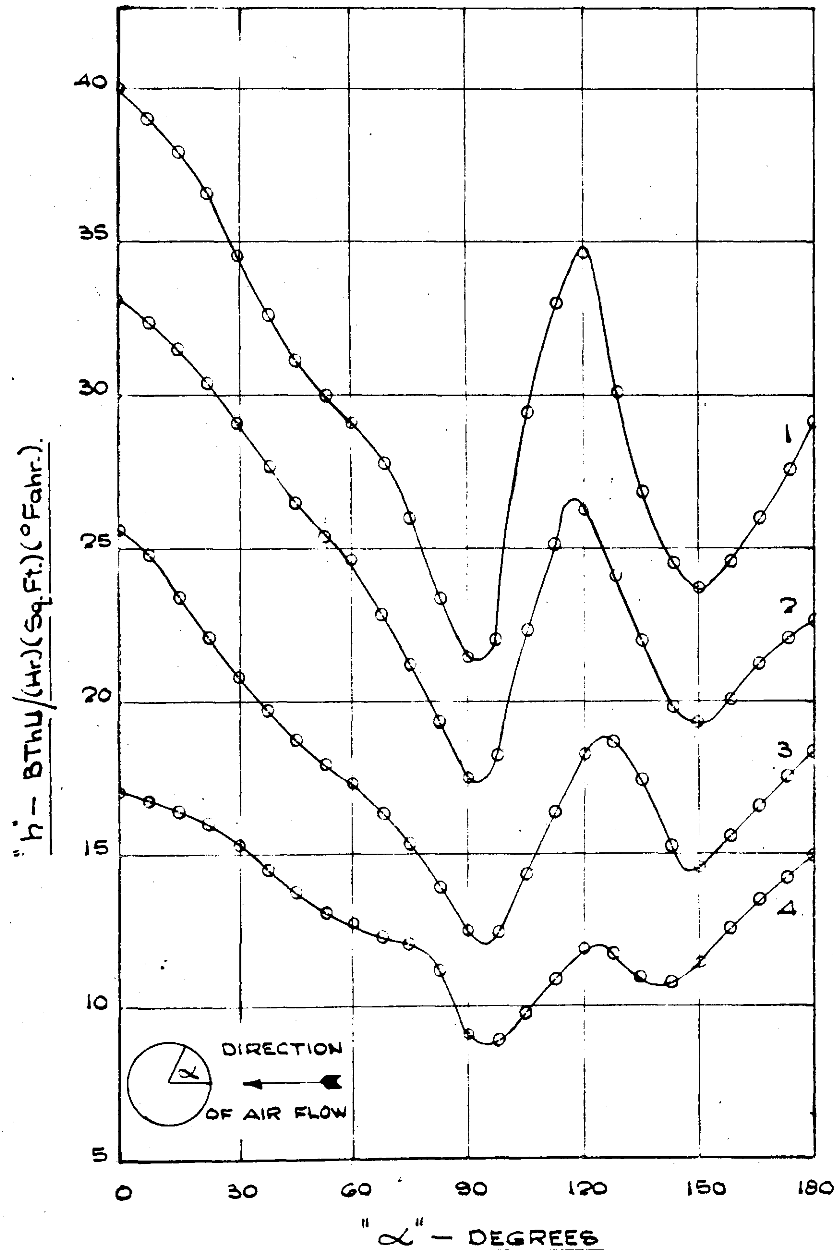
The review may be briefly summarised as follows:

(1) Experimental work on mean heat transfer rates to single tubes and also to tube banks has been fairly comprehensive and the results of the various workers properly correlated.

(2) The work on variation of heat transfer round the single tube has been very limited and has produced certain anomalies which cannot be satisfactorily explained in the light of existing knowledge. Attention, with one exception, has been confined to the outside surface of the tube, where it has been shown there is a marked variation in heat transfer rates. Where these experiments were carried out

FIG. 5

VARIATION OF LOCAL HEAT TRANSFER COEFFICIENT FOR ROW 5 OF A STAGGERED TUBE BANK.



CURVE Nº	REYNOLDS Nº
1	57,000
2	41,000
3	25,200
4	12,500

with tubes, it was generally assumed that the temperature round the inside of the tube was constant, thus implying a uniform rate of heat transfer at the inner wall surface; yet, in the one exception where heat transfer rates for the inside surface were measured, the results indicated a variation similar to that given by other workers for the outside.

(3) The variation of temperature around the tube wall has received very little attention although it has been clearly shown that a variation does occur. Such variation must induce circumferential heat flow and this, in turn, will affect the relationship between the variations in heat transfer coefficients at the outer tube surface and those at the inner surface. No consideration has so far been given to these aspects.

(4) Only one very limited and indirect attempt has been made to investigate variations round a tube placed in a bank.

A - APPARATUS

1. AIM AND SCOPE OF PRESENT INVESTIGATION

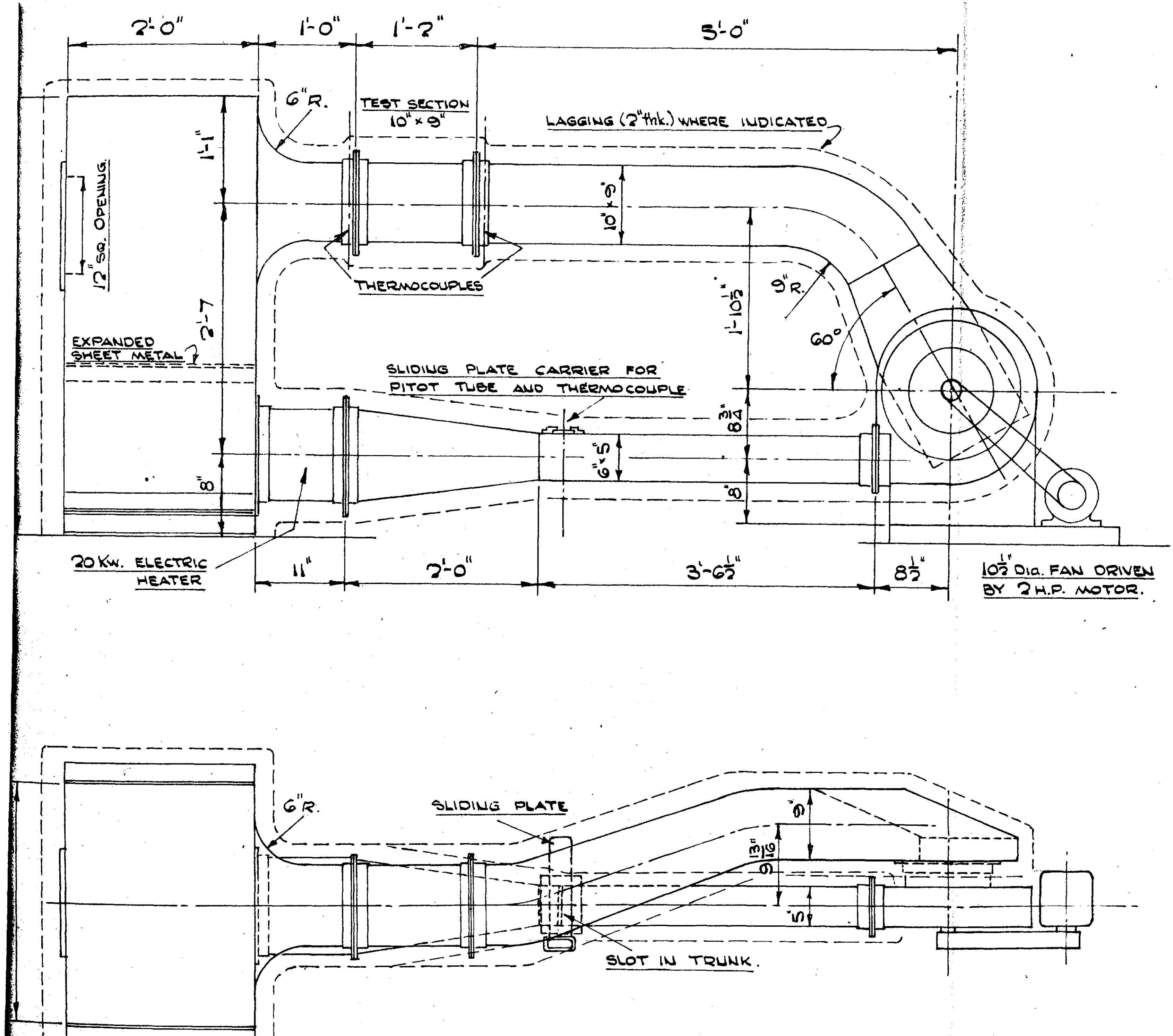
In the light of the Review, it was decided that the first step should be further experimental work on a single tube placed across a hot air stream to provide direct readings of tube temperature variations, and direct measurement of the variations in heat transfer rates round the inside surface of the tube. This information when correlated to existing knowledge of the variations in heat transfer rates at the outer tube surface, would give a much clearer picture of the local conditions around the wall of a tube and would probably give some knowledge of the radial and circumferential components of heat flow at any point. Such knowledge would represent a decided advance towards precise assessment of temperature stresses in the metal.

Having established the necessary technique, for the case of the single tube the work would then be extended to tube banks. In this way the influence of the tubes upon the gas flow pattern already seen in hydrodynamic tests could be examined in the light of the heat transfer results.

The apparatus used was designed specifically for this investigation. It comprised a recirculating wind tunnel with air heater and facilities for placing a single water-cooled tube or bank of tubes in the hot air stream. The measurement of tube wall temperatures and heat transfer rates round the tube was made possible by the design of a special test tube. This tube was used alone in the single tube tests, but during tests on tube banks it could be substituted for any one of the ordinary tubes forming the bank. The necessary instrumentation was provided.

2. Wind Tunnel:

To provide the hot air stream a totally enclosed recirculating wind tunnel was designed and constructed as in Figs. 6 to 8. Air passes from the variable speed fan along the lower duct, through a bank of controlled electric heating coils and into a large plenum chamber. Flow variations are damped out in this chamber, as the air flows at low speed up through a screen of expanded metal sheets. The air leaves at a uniform temperature and flows through a convergent passage to the test section, then returns to the fan inlet by way of the upper ducting. The design permits of air speeds up to 50 ft./sec. in the unrestricted test section and air temperatures from 200°F to 600°F.



SKETCH SHOWING ARRANGEMENT OF TUNNEL FOR HEAT TRANSFER RESEARCH.

FIG. 6.

FIG. 7.

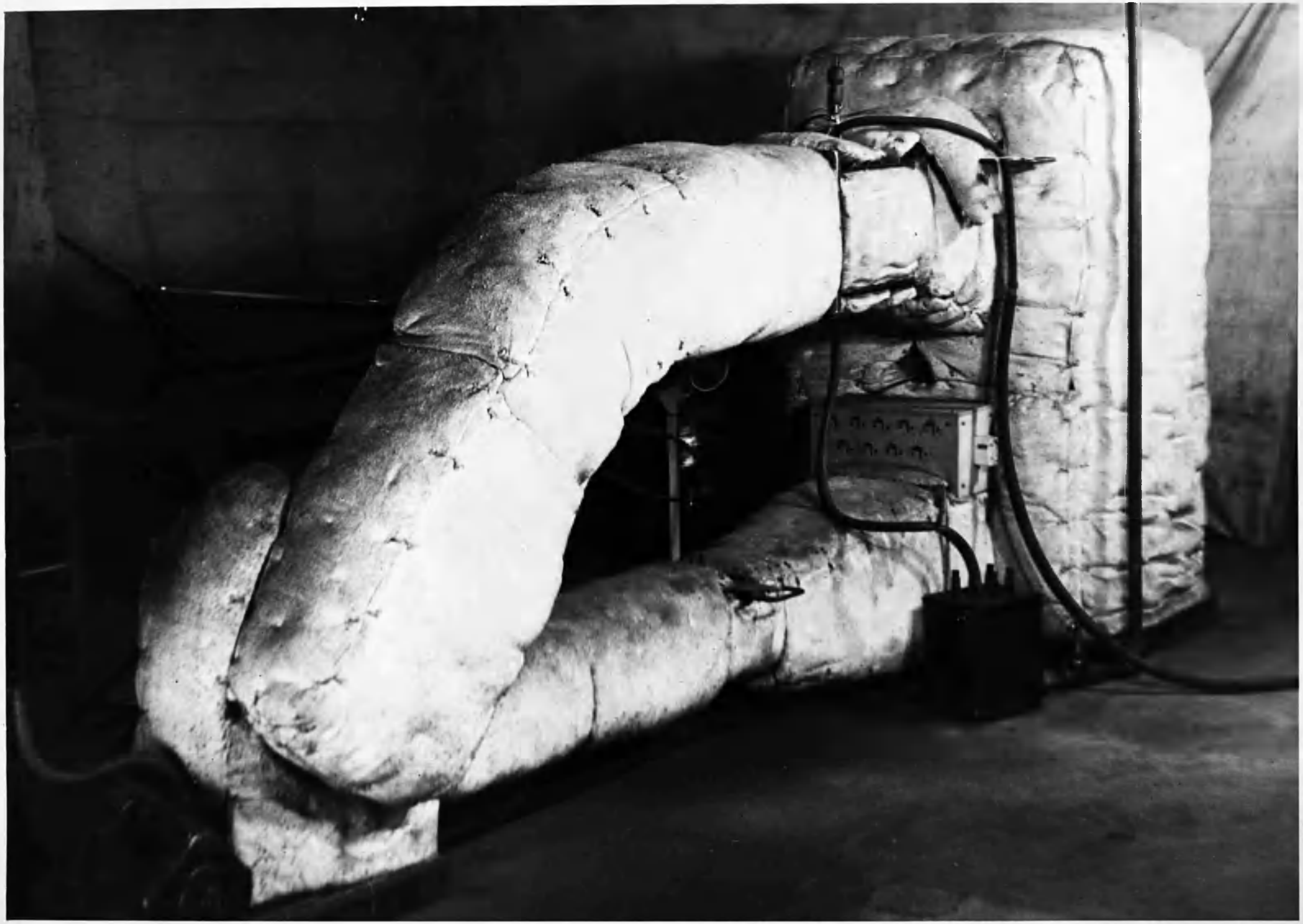


FIG. 8.

The mass flow of air in the circuit is calculated from readings of temperature and of velocity measured by a pitot-static tube mounted in the lower ducting some distance from the fan discharge. The temperatures of the air at this point and just before the test section are given by thermojunctions mounted in the air stream. Knowing these temperatures and the area to flow the velocity past a tube in the test section may be obtained.

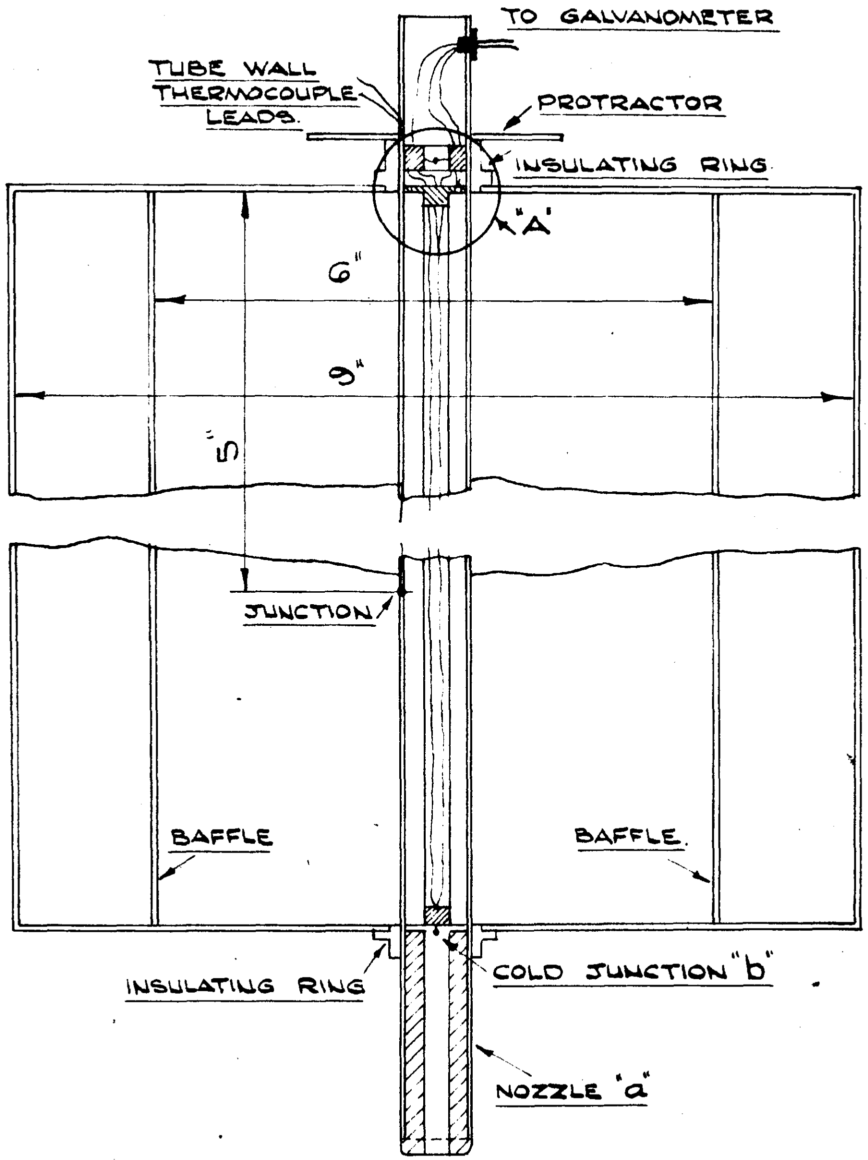
Preliminary tests indicated that at entry to the test section the temperatures and velocity distribution were uniform over the effective section of the duct.

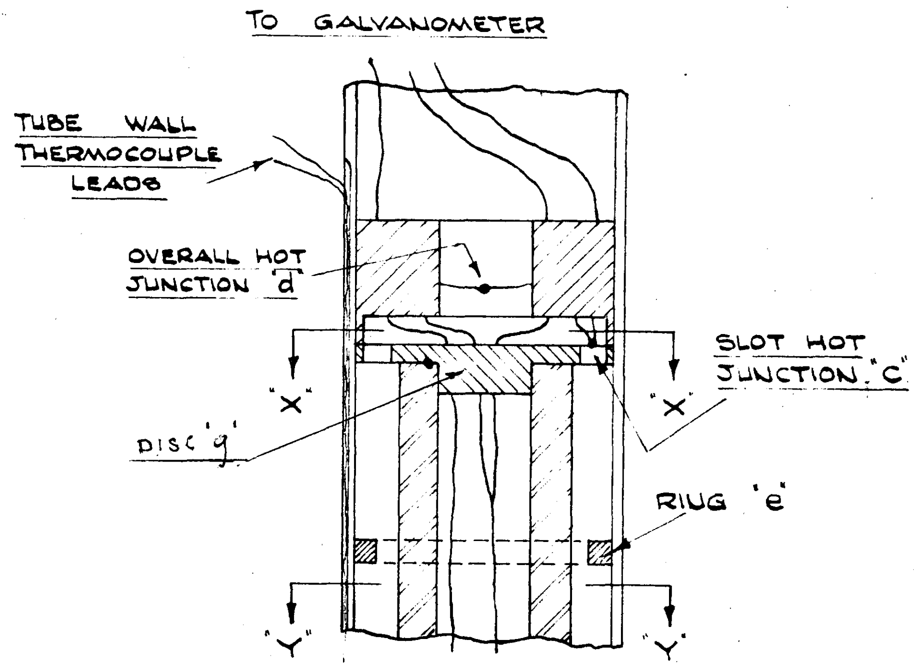
3. Tubes:

All tubes used were of solid drawn brass of nominal outside diameter $\frac{3}{4}$" and 10 s.w.g. thick. In the tests using banks of tubes, the tubes were spaced $1\frac{1}{2}$" centres across the duct, the rows being also at $1\frac{1}{2}$" centres. Thus it was possible to mount a bank of 6 rows of tubes in the test section.

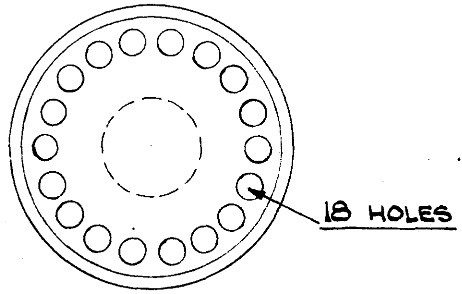
4. Test Tube:

A piece of the normal tubing was reamed out and a push fit slotted brass core was inserted, as in Fig.9. In the lower end of this tube was fitted a bakelite restrictor a, The cooling water flows through a and then equal fractions of the water pass along each of the 18 slots machined on the core.





ENLARGED DETAIL AT "A"



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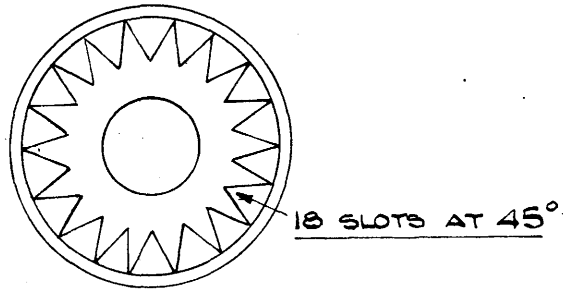

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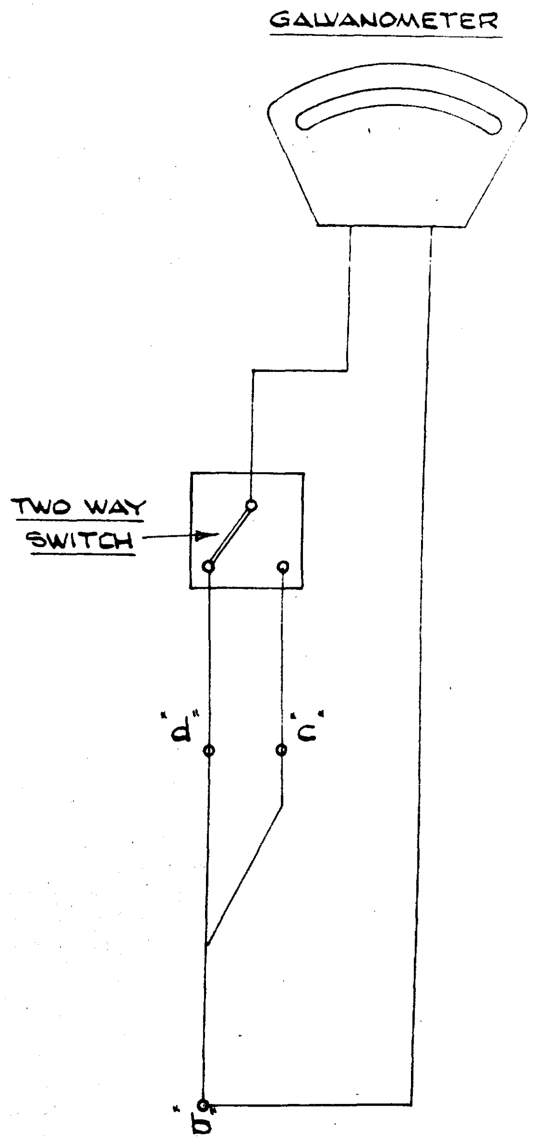


GALVANOMETER CIRCUIT.

ARRANGEMENT OF TEST TUBE

FIG. 9

To enable the tube temperature to be measured, a copper constantan thermocouple junction was soldered into a fine slot that had been milled axially on the outer surface of the tube at a point diametrically opposite the test slot. The insulated leads from this junction were led along the slot out of the tunnel and connected to a potentiometer and a cold junction maintained at the temperature of melting ice. The slot was carefully filled in and smoothed over using a red lead and glycerol cement to prevent any disturbance of the air flow pattern.

For the single tube tests, the tube was mounted in the centre of the test section, as in Fig. 9. By rotating the tube, readings of the temperature rise along the test slot and the tube wall temperature could be obtained for any angle around the tube.

B - EXPERIMENTAL TECHNIQUE

1. EXPERIMENTAL CALIBRATIONS

(a) Air Flow. A pitot-static tube of $^3/16$ ins. external diameter was constructed to the dimensions recommended by Ower[32]. This design is similar to the N.P.L. round nosed type with the right angle bend replaced by a radius which allows easier construction and easier entry into the ducting. This tube was calibrated against a standard N.P.L. sharp-nosed pitot-static tube by comparing the pressure head differences of the two tubes when they were placed in a uniform air stream. Tests were carried out in the main Wind Tunnel of the R.T.C. with velocities from 20 to 120 ft./sec., the velocity head readings being measured by an Askania Micromanometer. These readings were found to agree within 1% over the entire velocity range, and since the pitot-static factor of the N.P.L. tube may be taken as 1 in this range[32], the factor for the test tube was also taken as unity.

The inclined U-tube water manometer used with the pitot-static tube to give readings of pressure difference was calibrated by the method given by Ower[32] against Chattock and Askania Micromanometers. The factor by which the gauge readings must be multiplied to give readings in inches of water was found to be 0.0498. Over the complete range of the inclined gauge, this factor was constant within ½%.

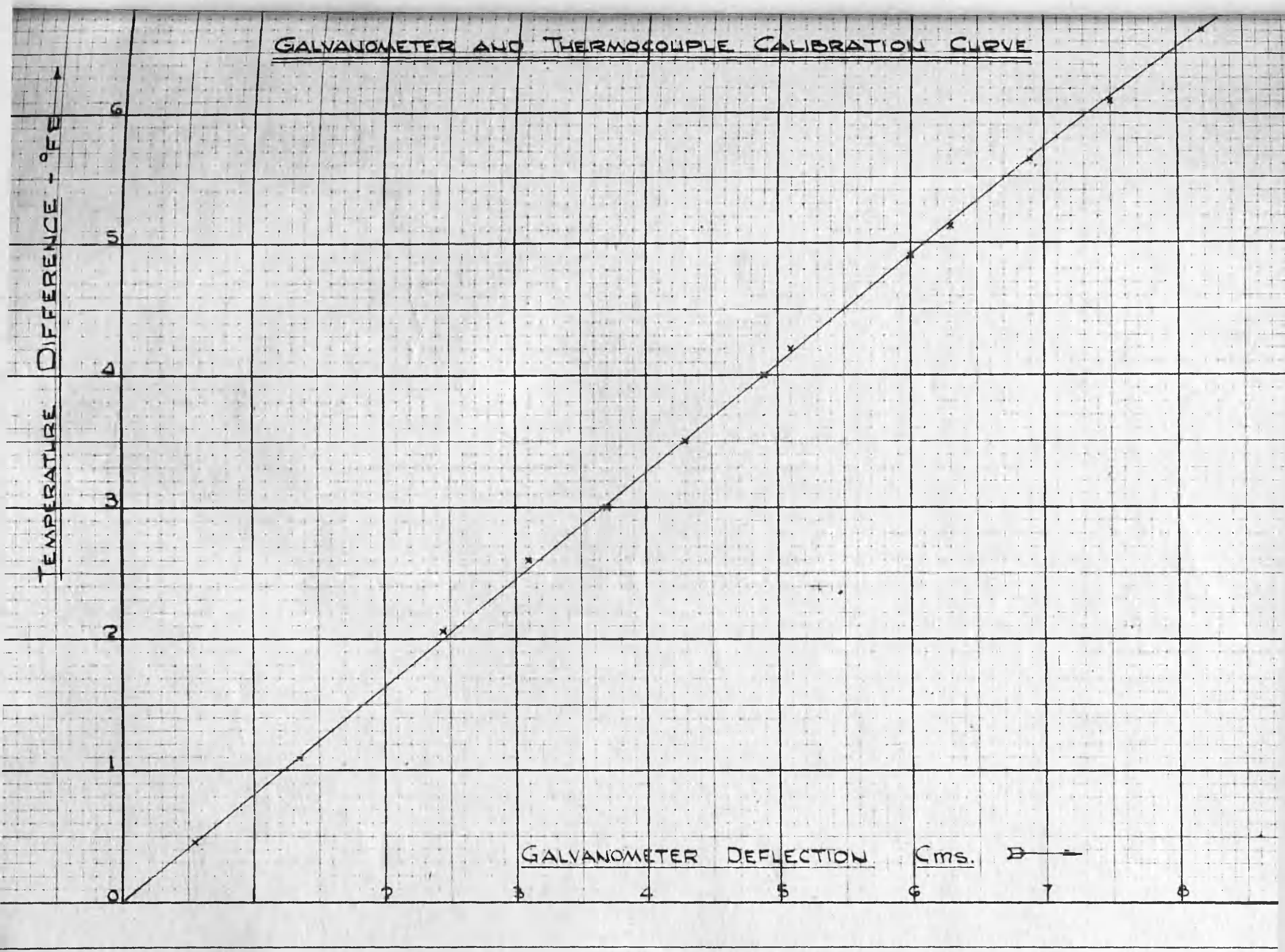
The velocity distribution across the duct at the pitot station was not uniform, but by obtaining readings at various points across the duct and integrating the velocity distribution curve, a mean speed was obtained. The ratio of this mean speed and the speed at a point on the centre line and 2 ins. from the bottom of the duct, was found within the range of test conditions to vary less than ½% from the average value of 0.897.

To obtain the true mean speed at the Pitot Station, these three calibration factors were combined to give the constant 0.0448. Thus the reading on the inclined gauge when connected to the pitot static tube at its fixed point in the ducting, could be converted to a true mean speed. Knowing the air temperature at this section, the actual mass flow of air could be obtained.

(b) Temperature Measurement: All temperature measurements were made using thermocouples. The thermocouple circuits, as in Fig. 9, were constructed using 28 s.w.g. copper and constantan wire. the length of wire in each circuit being exactly the same. Junction b was placed in a constant temperature water bath while junctions c and d were immersed in a controlled temperature bath. A Beckmann thermometer in each bath gave the temperature difference between the baths to within $^1/_{50}$th of a degree F. A check on this was also obtained using an eight-junction thermopile and potentiometer circuit.

The deflections on the galvanometer for various temperature differences are plotted in Fig.10. The mean curve shown was used throughout the tests to convert galvanometer readings to temperature difference for both circuits.

After many tests had been carried out using one test tube, the thermocouple circuit was removed from the test tube and again calibrated. This showed that the calibration had not altered.


GALVANOMETER AND THERMOCOUPLE CALIBRATION CURVE
TEMPERATURE DIFFERENCE - °F
GALVANOMETER DEFLECTION Cms.
0
1
2
3
4
5
6
7
8

Silk insulated 28 s.w.g. copper and constantan wires were used to make the thermocouples, for all other temperature readings. The e.m.f. - temperature relationship for six sample couples were obtained using a Cambridge Potentiometer with the junctions immersed in the following mediums of known temperature.

(i) Melting ice and steam in a hypsometer

(ii) Melting ice and subliming solid CO_2 using the method given by Scott[33].

(iii) Melting ice and solidifying lead.

The readings of e.m.f. and temperature from all the couples agreed within 1% of the e.m.f. temperature curve for copper-constantan thermocouples, as given by the American Institute of Physics[33], and this curve was subsequently used for all conversions.

The thermojunctions to measure the air temperature in the duct were mounted in the centre line of the duct, one lead being taken out through an insulated plug in each side of the ducting. No shields were required with these couples, as tests with shielded and unshielded couples showed that there was no measurable error in the air temperature read at 600°F. This was to be expected, as the heavy lagging of the ducting allowed the duct metal to attain a temperature very close to that of the air stream.

Account had, however, to be taken of the radiation between the ducting and the water cooled tube. To evaluate this radiation, it was necessary to know the temperature of the duct metal. This was done by soldering thermocouple junctions into small holes drilled on the outside of the duct. A very close approximation to the temperature of the radiating surface was thus obtained without disturbing the air stream.

Extensive investigations were made to evolve an exact method of measuring the tube wall surface temperature. Very large errors may be introduced by the conduction of heat along the wires leading to the thermojunction, but the method used in the present tests was found to give very satisfactory readings. A thermo-setting resin was at first used to fill in the slot on the tube surface, but equally satisfactory results were more easily obtained using a fine cement of glycerol and red lead.

2. Preliminary Tests:

Tests were carried out to find if any error in readings was caused by conduction effects in the brass core. A test tube was constructed with a core of bakelite instead of brass, but readings obtained from tests with this tube showed that there was no such error.

As the water temperature rises in passing through the test tube, there will be a corresponding rise in the temperature of the tube wall. Investigations were made to measure this axial variation in temperature. The water flow was regulated to the value used throughout the tests and the air flow regulated to 600°F. and the maximum speed at this temperature, thus giving the maximum water temperature rise. A tube twice the length of the normal test tube with a thermojunction mounted in the wall was used and readings of tube temperature along the upstream generator were obtained by moving the tube vertically through the duct. The variation in temperature was 6.2°F. As the variation was so small, the readings of tube temperature half way along the tube, as obtained in the main tests, are taken as the average tube temperature for any angle of tube.

The length of test tube in the air stream in all tests was 10 ins., but the distance between couples b and d, Fig. 9, is actually 10.25 ins. Tests made with a tube arranged with junctions b and d $\frac{3}{8}$" apart, showed that the heat transfer over a $\frac{3}{8}$" length at the tube ends was constant until b or d was $\frac{1}{4}$" outside the ducting. Also from tests to measure the tube temperature gradient at the tube ends, it was seen that the tube temperature dropped only 1°F. over the first $^{3}/16$" out from the air stream. It was therefore assumed that the actual water temperature rise for a 10 ins. length of tube was equal to the reading of temperature rise between b and d multiplied by

$\frac{10}{10.25}$. Further, no correction to the readings was necessary to allow for heat loss by endwise leakage.

The general reproducibility of results was tested by using two different tubes of similar dimensions. Under the same air flow conditions identical readings were obtained from the two tubes.

3. Experimental Procedure:

(a) Single Tube Tests: The test tube was mounted in the centre of the ducting, as shown in Fig.9. To give a higher air speed past the tube, the width of the test section was reduced to 6 ins. by carefully shaped baffles. The protractor fixed to the top of the tube showed the angular position of the test slot, this angle being varied by turning the tube. The water flow during any test was kept constant by means of a fixed level header tank and an open flow outlet.

The procedure at the start of each test was first to regulate the water flow rate to approximately 4.5lb. per minute and then to set the galvanometer to zero. The fan was started and the heaters switched on to bring the air temperature up to the required figure. The fan speed and heater input were then adjusted to give the necessary air flow conditions and the tube rotated so that the test slot faced directly upstream.

Sufficient time was allowed to enable conditions to stabilize and the following readings were taken:-

water flow rate
position of test slot
overall water temperature rise
water temperature rise along test slot
tube surface temperature
duct wall temperature at six points
air flow rate
air temperature before test section.

The tube was then rotated through 20° and a period of at least 5 minutes allowed before again taking the above readings. The procedure was repeated at each 20° interval up to 180°.

Thus for any air flow condition, it was possible to obtain the value of Re, the tube wall temperature variations, and the rate of heat flow to the tube and to the water in the test slot.

2. Tube Bank Tests:

The staggered banks of tubes used were arranged so that the area for air flow in each row was constant. Where necessary half tubes were attached to the duct sides. All the tubes, other than the half tubes, were water cooled.

Test readings as for the single tube experiments were obtained with the test tube substituted in turn for various tubes in the bank. In addition, the temperature of the air leaving the bank was measured when more than three rows of tubes were fitted. This was done by installing three thermocouples at different positions across the duct, the average of the readings of the three couples being taken as the mean air temperature.

PART III - EXPERIMENTAL RESULTS AND DISCUSSION

A - SINGLE TUBE RESULTS

The results of the tests on a Single Tube placed across a hot air stream are presented first as mean heat transfer coefficients from air stream to tube at different Reynolds Numbers. This allows comparison with the results of the many previous tests and indicates the general accuracy of the present tests. The variations in tube wall temperature are then considered and the detailed results of a single test are examined and explained. Finally, the complete results are presented and discussed.

1. Mean Convective Heat Transfer Coefficient between air and external Tube Surface:

The results obtained from the tests on the single tube are shown in Table 1(a), appendix 1. The range of Reynolds Number covered in these tests was 6,000 to 25,000 with air speeds of 40 to 100 ft./sec. and air temperatures from 200°F. to 600°F. Tests 1 to 26 were made using a tube of 0.747 in. outside diameter with wall thickness 0.059. in., while in tests 27 to 31 the tube used had been machined down to an external diameter of0.693 in., giving a wall thickness of 0.032 in. This table shows only the average temperature rise, θ_m, of the water through the 10" of tube in the air stream and t_{s_m}, the average tube temperature. The variations in these two quantities at points around the tube are considered later.

In Table 1(b), appendix 1, the results deduced from the data of Table 1(a) are given. Reynolds Number is calculated using G. the mass flow of air per unit of area based on the minimum flow area, d, the external tube diameter and μ_f the viscosity of air at the air film temperature t_{f_m}. From the temperature rise and the rate of flow of the water through the tube, the total heat transferred to the tube can be obtained. This will be the sum of the heat received by convection and by radiation. Since the duct temperature and the tube wall temperature are known, the radiant heat transfer can be calculated from

$$H_r = 0.173 \times 10^{-8} \times E\left(Td_m^4 - Ts_m^4\right)$$

where E is the emissivity of the brass tube, taken as 0.1. This assumes that the tube is small compared to the surrounding radiating surface. For convenience, a radiant heat transfer coefficient is used where

$$h_r = \frac{H_r}{\Delta t_m} \quad \text{and} \quad \Delta t_m$$

is the mean temperature difference between the gas stream and the tube wall. Thus the mean convective heat transfer coefficient h_{c_m} for the tube can be found from

$$h_{total} = h_{c_m} + h_r$$

It will be seen from Table 1(b) that the radiant heat transfer in these tests is very small, never exceeding

2% of the total heat transferred.

Values of Nusselt Number have been calculated for each test using the values of h_{c_m} so obtained, d the external tube diameter and k_f the conductivity of the air at temperature t_{f_m}.

The values of Nu and Re for each test are given in Table 1(b) and are shown plotted in log-log form in Fig.11. It will be noted that the points lie close to the straight line given by $Nu = 0.267(Re)^{0.595}$

For comparison, the results of various other experiments, together with the curve recommended by McAdams[10] are shown with the curve of Fig.11 in Fig.12. It will be seen that the present test results give higher values of Nu than the McAdams curve, but the points lie within the zone covered by the results of the many experiments. It should be noted that the results for all the curves in Fig.12 have been evaluated as described in this section.

Hilpert presents results for tests where the temperature of the wires used was varied from 200°F. to 1800°F. and correlates his results by considering Nu to be a function of Re and $\left(\frac{Tsm}{T}\right)^{\frac{1}{4}}$. The close agreement of the present tests covering the much smaller tube temperature range

FIG. 11

MEAN NUSSELT NUMBER TO REYNOLDS NUMBER FOR SINGLE TUBE TESTS

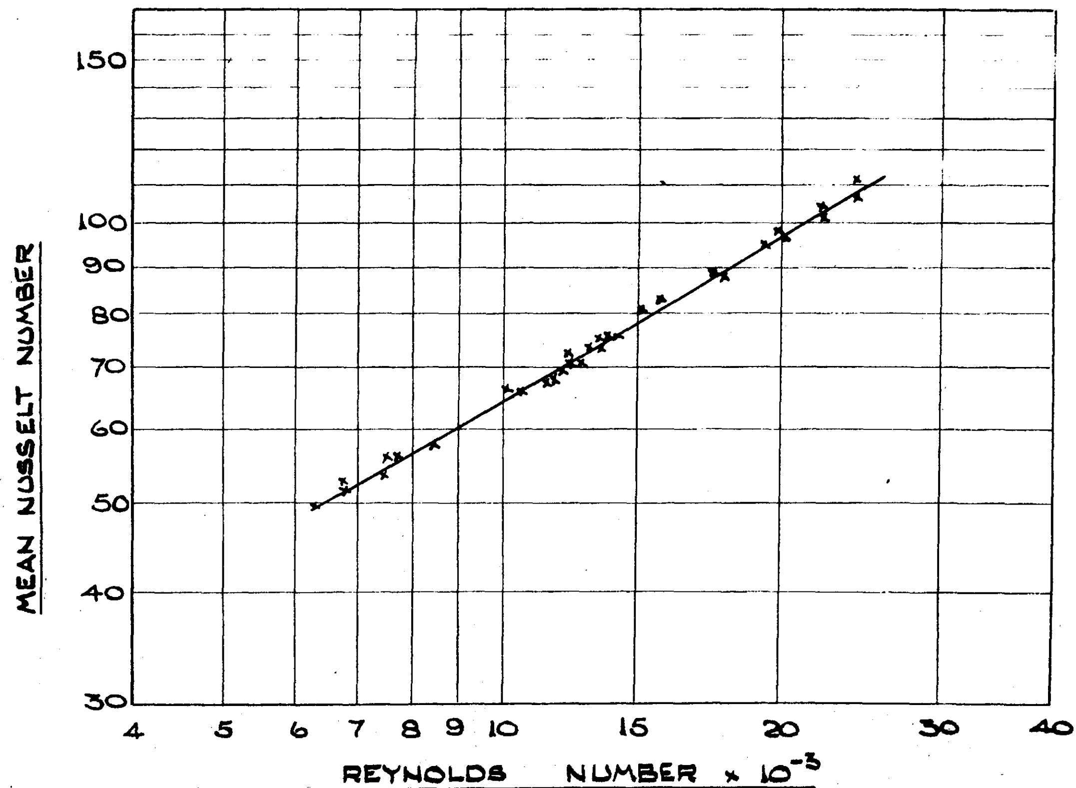
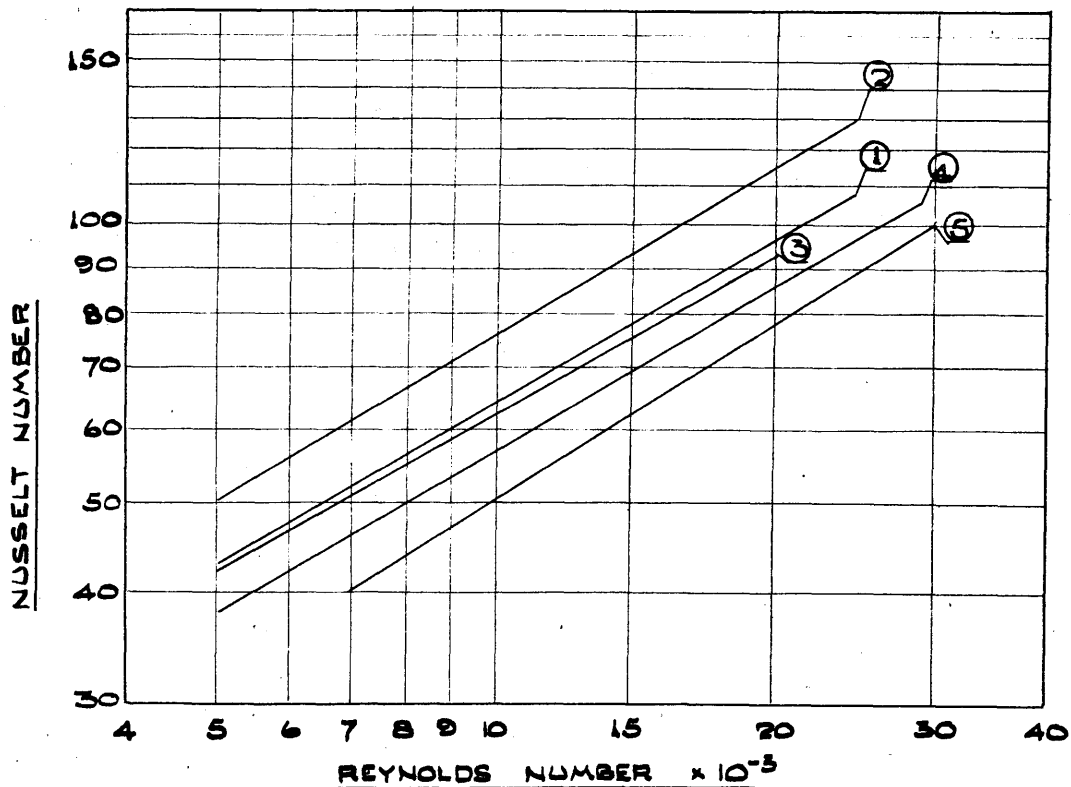


FIG. 12

NUSSELT NUMBER TO REYNOLDS NUMBER FOR SINGLE TUBE



① - PRESENT TESTS

② - GRIFFITH & AWBERY

③ - REIHER

④ - Mc ADAMS

⑤ - HILPERT

$200^{\circ}F.$ to $600^{\circ}F.$ when plotted in the form $Nu = f(Re)$ does not seem to justify this additional temperature factor.

2. Tube Wall Temperature Variation around Tube:

In Table 2, appendix 1, the experimental readings of tube temperature at different angles around one half of the tube are shown for the various tests of Table 1.

(a) Effect of Reynolds Number at Constant Air Temperature: In tests 7, 8 and 9, the approach air temperature was kept constant at $400^{\circ}F.$ and Re was varied by altering the air speed. The results of these tests are shown plotted in Fig.13.

As would be expected, the increase in heat transfer arising from the increased air speed, results in a corresponding general raising of tube temperature. It is interesting to note that the range of temperature at lower values of Re is greater than that at higher values. Also, at the lower values there is an almost constant fall in tube temperature from front to rear, and as Re is increased the point of minimum tube temperature moves from the rear towards the side of the tube.

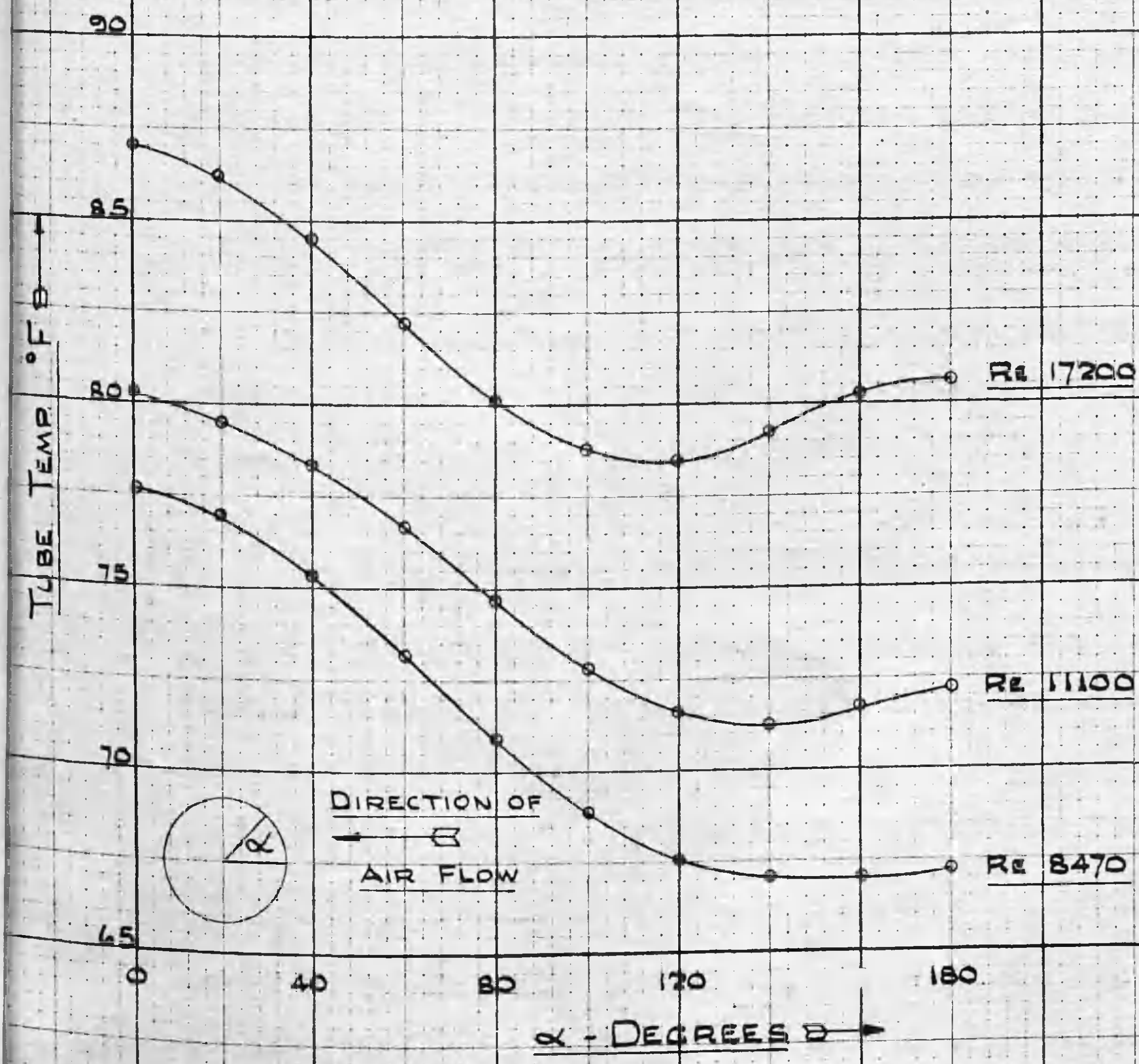
These curves seem to substantiate and extend to higher values of Re, the results of Reiher[7] shown in Fig.2, where for values of Re below 6000, the minimum value of tube temperature occurs at the rear of the tube. The form of the

FIG. 13

VARIATION IN WALL TEMP. WITH TUBE ANGLE

AIR. TEMP. 400°F

SINGLE TUBE



temperature-angle curve at values of Re above 10,000 agrees with that given by Krujilin and Schwab[12], Fig. 3, where the heat transfer was in the opposite direction, that is from tube to air stream.

(b) Effect of Air Temperature at Constant Reynolds Number
The temperature distribution for four tests where the Reynolds Number was approximately constant, are shown in Fig.14. Since the heat flow rate is increased by raising the air temperature, though the Nusselt Number remains constant, the overall temperature of the tube is also raised. It may also be seen that the range of circumferential temperature variation is increased by raising the temperature of the air stream.

It should be noted that the temperature variations around the tube are due to the variations in the heat transfer rates at the tube surface.

3. Variation of Convective Heat Transfer Coefficient around Tube:

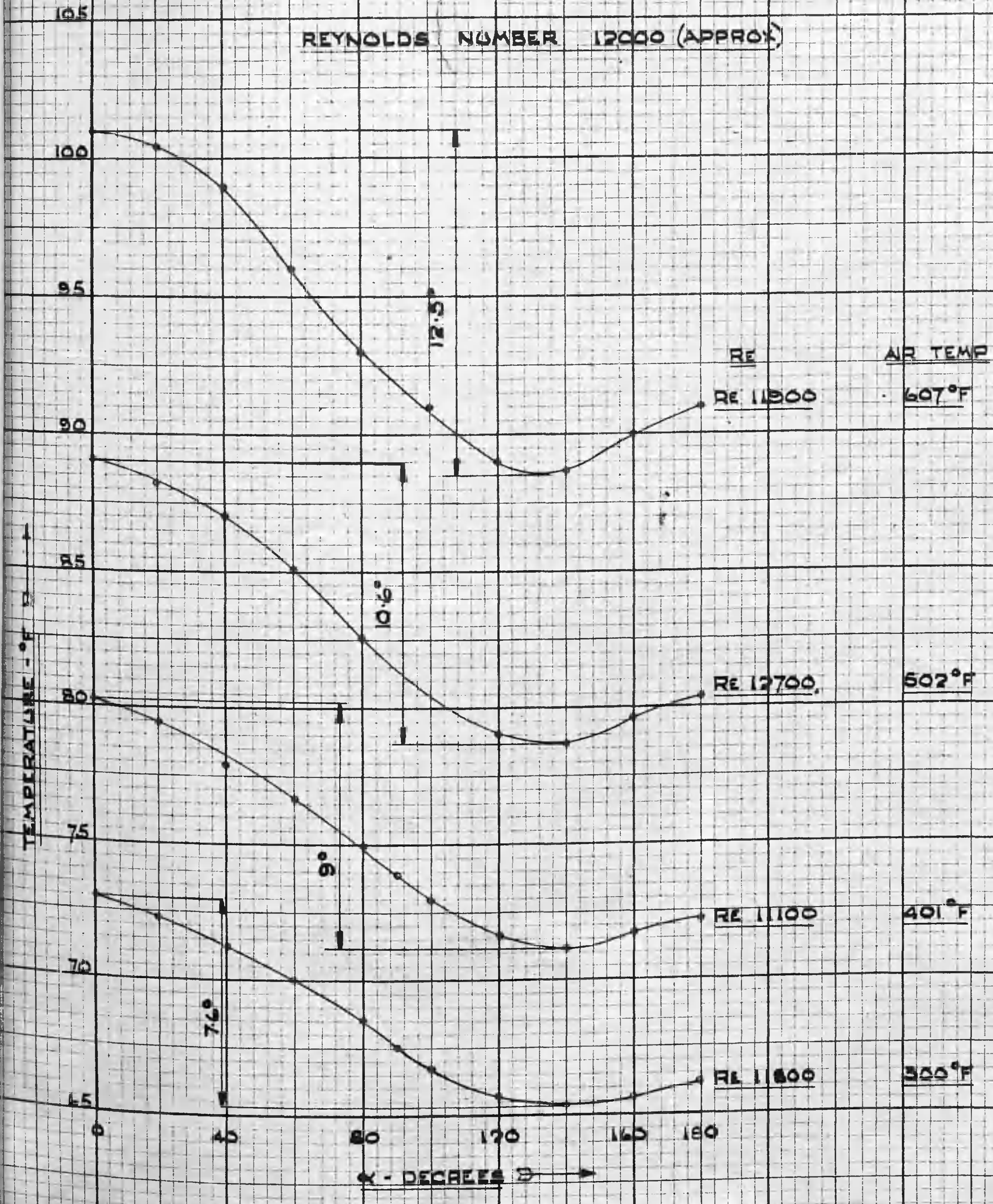
The values of the water temperature rise, θ, through the test slot at various angles to the air stream are given for the tests of Table 1 in Table 2, appendix 1. To simplify presentation, the results of a single test will first be considered.

In the test selected (test 3, tables 1 and 2) the air temperature was 250°F. and the air speed 87 ft./sec.,

Fig. 14

VARIATION IN WALL TEMP. WITH TUBE ANGLE – SINGLE TUBE

REYNOLDS NUMBER 12000 (APPROX)



giving a Reynolds Number of 22600. The experimental results are presented in Fig.15 in two curves A and B. These give respectively the angular variation in the rate of heat rejected from tube to water and the angular variation in the tube temperature. It is only necessary to show curves for 0° to 180°, as the distribution was found to be symmetrical about the meridian.

Curve C, Fig.15, is deduced from the results of Schmidt and Wenner and shows the angular variation in rate of heat reception at the outside surface of a tube, also for an Re of 22600.

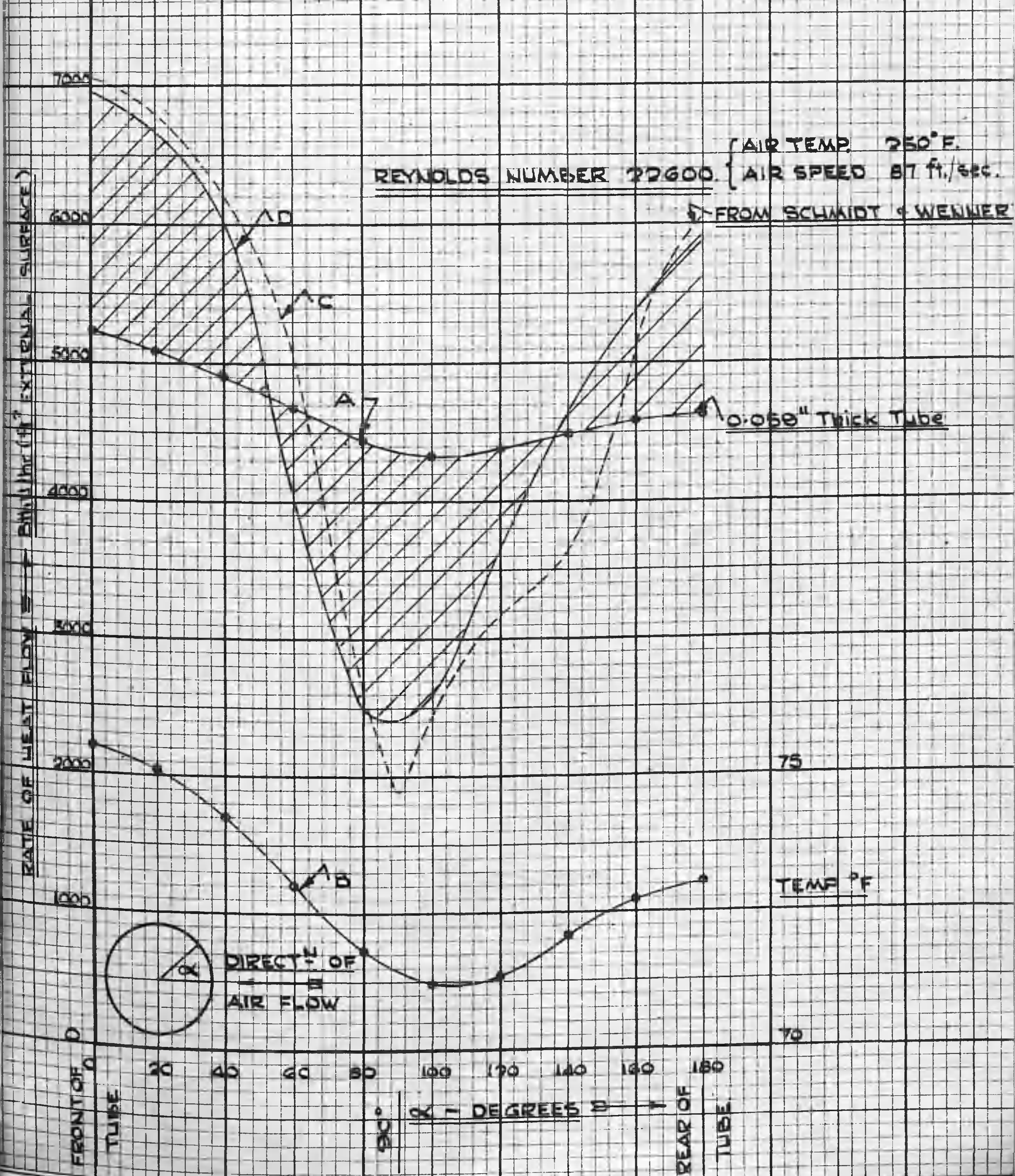
A comparison of the curves A and C reveals a marked difference between the variation in heat flow rate at the inner and outer surfaces. This difference is accounted for by the conduction of heat that takes place around the tube as a result of the circumferential variation in the tube wall temperature shown in curve B.

It will be seen from curve B that the coolest part of the tube wall occurs at an angle of about 110°, and, therefore, heat will tend to flow circumferentially in the tube wall towards that zone. It follows, for example, that at the front of the tube only a fraction of the heat which enters at any given point passes directly through to the water.

FIG. 15.

VARIATION OF TEMP. AND OF HEAT FLOW RATE AROUND TUBE

SINGLE TUBE



This circumferential heat flow in the tube wall at any section may be calculated as shown in the following section.

(a) Analysis of circumferential heat flow in tube wall:

For a small element of tube metal the rate of heat reception by the element is given by the standard equation

$$k\left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2}\right) = c\rho\frac{\partial t}{\partial S}$$

where x, y and z are the rectangular co-ordinates and S in this case denotes time.

This reduces for steady state conditions to

$$\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2} = 0$$

which expressed in terms of the polar co-ordinates r and α gives

$$\frac{\partial^2 t}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r} + \frac{\partial^2 t}{\partial \alpha^2} = 0$$

The radial and circumferential temperature distribution throughout the metal of the tube is required to obtain a complete solution of this equation. This is not possible in the case of a thin tube, because of the physical difficulty of inserting thermocouples at various radii in the wall of the tube. A good approximation, however, is possible by the following analysis:-

Consider a unit length of tube of internal radius r_i and external radius r_o which receives heat from a fluid flowing past its outer surface and rejects heat to a fluid flowing axially inside.

Assuming that no heat flow takes place in the direction of the tube axis, then for an element of tube subtended by angle $\delta\alpha$ Fig. 16, we have.

Where H_o = rate of heat flow per unit area at outer surface of tube element.

H_i = rate of heat flow per unit area at inner surface of tube element.

H = rate of heat flow by conduction per unit area around the tube.

Heat flow into element $= H_o r_o \delta\alpha + H(r_o - r_i)$ — — — — — 1

Heat flow out of element $= H_i r_i \delta\alpha + (H + \delta H)(r_o - r_i)$ — — — 2

For steady state equation 1 = equation 2.

$$\therefore \quad H_o = \frac{H_i r_i}{r_o} + \frac{\delta H (r_o - r_i)}{r_o\, \delta\alpha}$$

Now

$$H = -k \frac{dt}{r_m d\alpha}$$

where r_m = mean radius

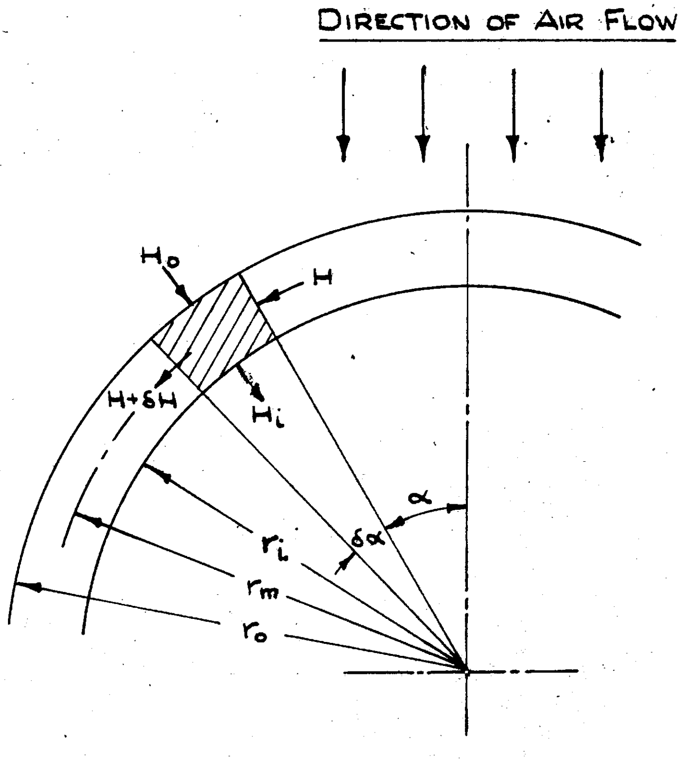
k = conductivity of metal

$\frac{dt}{d\alpha}$ = circumferential temperature gradient with respect to angle

and $\delta H = -k\,\delta\left(\frac{dt}{r_m d\alpha}\right)$

$$\therefore \quad \frac{\delta H}{\delta\alpha} = -k \cdot \frac{\delta}{\delta\alpha}\left(\frac{dt}{r_m d\alpha}\right) = -\frac{k}{r_m} \cdot \frac{d^2t}{d\alpha^2}$$

FIG. 16



DIRECTION OF AIR FLOW
H_o
H
$H+\delta H$
H_i
α
$\delta\alpha$
r_i
r_m
r_o

neglecting variations in circumferential temperature gradient across a given radius,

$$H_o = \frac{H_i r_i}{r_o} - \frac{k(r_o - r_i)}{r_o \cdot r_m} \cdot \frac{d^2t}{d\alpha^2} \qquad \text{----------- } 3$$

Substituting the dimensional values for the tube being tested and taking the conductivity of brass as 57 B.Th.U./hour. ft. ^{o}F. in equation 3 we have:-

$$H_o = 0{\cdot}832\, H_i - 314{\cdot}3\, \frac{d^2t}{d\alpha^2} \qquad \text{----------- } 4$$

The first term in this equation accounts for the difference in radial heat flow area between the inside and the outside of the tube, while the second term is the increment of circumferential heat flow at any given point round the tube.

(b) <u>Relationship between heat transfer rates at inner and outer surfaces</u>: Owing to the thinness of the tube it was possible only to measure the external tube temperature. From calculations based on the measured heat flow, it is obvious that the radial variation in tube temperature at a given point must be small. Hence it has been considered sufficiently accurate to take the circumferential variation in external tube temperature as being representative of the variation in mean tube temperature. By a method of finite differences [34], the second differential of this curve at any angle was obtained and by substituting the corresponding value of H_i, as given in Curve A, values of H_o for various angles have been calculated and are shown in curve D, Fig. 15.

The similarity between curves C and D shows clearly that the apparent disparity between curves A and C can be completely explained by consideration of circumferential heat flow effects in the tube wall. The small differences in curves C and D may be due to the fact that curve C was obtained from experiments where heat was being transferred from a cylinder to air, and the cylinder temperature was in effect constant. Also the readings of curve D are mean readings for 20° of surface, while in curve C the readings are the mean for 11° of tube surface. However, it must be noted that the double differentiation of the temperature curve must involve some error, even although extreme care was taken in measuring the tube temperature.

For comparison, in Fig.17 curves similar to these given in Fig.16 are shown for test 9. where Re was 8470, the results being obtained using the same test tube as was used in Test 3. It will be seen that the general form of the curves is somewhat altered by the different shape of the tube temperature curve B, but good agreement is obtained between the experimental curve of Schmidt and Wenner and the deduced curve D.

This Fig. 17 provides an explanation of the much discussed incompatability of Reiher's tube temperature curves, Fig. 2 and the heat transfer variation curves obtained by other workers, Fig. 4. It can be seen that

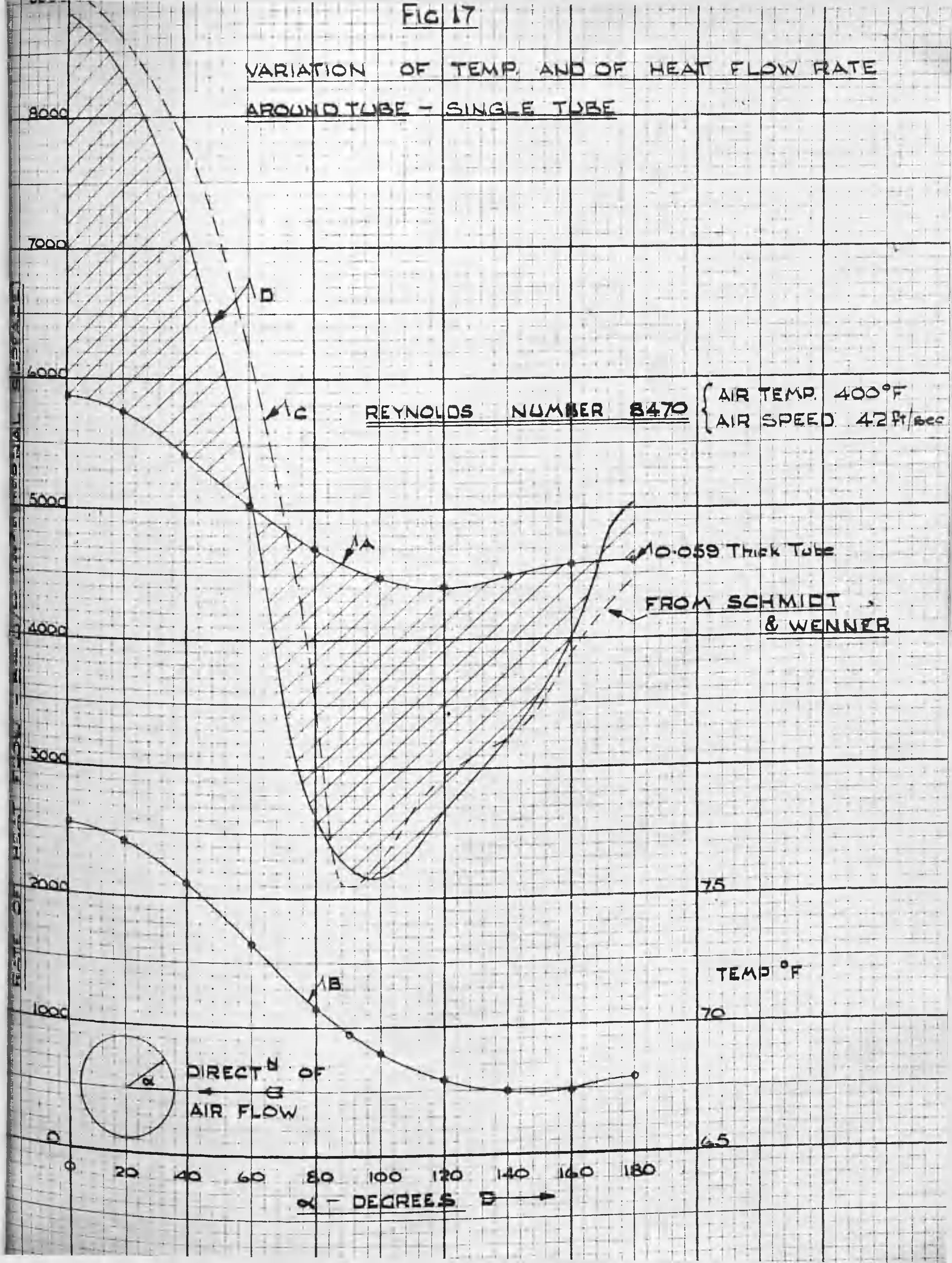


FIG 17
VARIATION OF TEMP. AND OF HEAT FLOW RATE
AROUND TUBE - SINGLE TUBE
8000
7000
6000
5000
4000
3000
2000
1000
0
D
C
A
B
REYNOLDS NUMBER 8470
AIR TEMP. 400°F
AIR SPEED 42 ft/sec
0.059 Thick Tube
FROM SCHMIDT & WENNER
75
70
65
TEMP °F
DIRECT^N OF AIR FLOW
α
0
20
40
60
80
100
120
140
160
180
α - DEGREES

maximum and minimum values of heat transfer do not correspond with maximum and minimum values of tube temperature, and that at low values of Reynolds Number, heat is being conducted from the front to the rear of the tube. It should be noted that for Re above 8,000, heat flows circumferentially from both the front and the rear of the tube to the sides.

Further interesting features emerge from a closer examination of the curves in Fig.15 and 17.

(1) Since the gross heat input to the external surface of the tube must equal the gross heat lost by the inner tube surface, it follows that the areas under curves D and A representing respectively, the above quantities must also be equal.

(2) At any angle the intercept between curves A and D shows the increment of circumferential heat flow at that section, this increment being positive above curve A and negative below curve A.

(3) Up to any value of α the algebraic integral of the area between the curves will give the actual rate of circumferential heat flow at the section at α. This nett area between 0° and 180° will be zero, and the nett area up to the point of minimum tube temperature will be zero.

(4) The points of intersection of curves A and D should correspond to maximum values of the circumferential temperature gradient, that is to the point of inflexion on the tube temperature curve.

All these conditions are approximately met in Fig. 15 and 17. Thus much information regarding the distribution of heat transfer at the outer surface of the tube may be obtained from examination of the curves showing the distribution of heat transfer at the inner surface and the tube temperature variations.

In the foregoing, the given test has been considered in detail for the purpose of demonstrating the nature of the actions which occur in transferring heat from an air stream to a colder fluid flowing in a tube. The extent of these variations under varying air flow conditions will now be considered.

(b) 'Apparent' and Real Nusselt Numbers; Previous experimenters have usually plotted results in the form of Nusselt Number to a base of angle as in Fig.4. To facilitate comparison with these, the results of the present tests, as given in Table 2, appendix 1, are plotted in Fig.18 using an "apparent" Nusselt Number.

$$\text{Apparent Nusselt number} = N' = \frac{h'd}{k}$$

where d = external tube diameter, ft.

k = conductivity of air for temperature t_m B.Th.U./hour.Ft.$^{\circ}$F.

$t_m = \frac{1}{2}(t + t_w)$

t = air stream temperature, $^{\circ}$F.

t_w = Tube wall temperature, $^{\circ}$F.

and $h' = \dfrac{Q_i}{a_o(t - t_w)}$ for an element of tube subtended by a small angle.

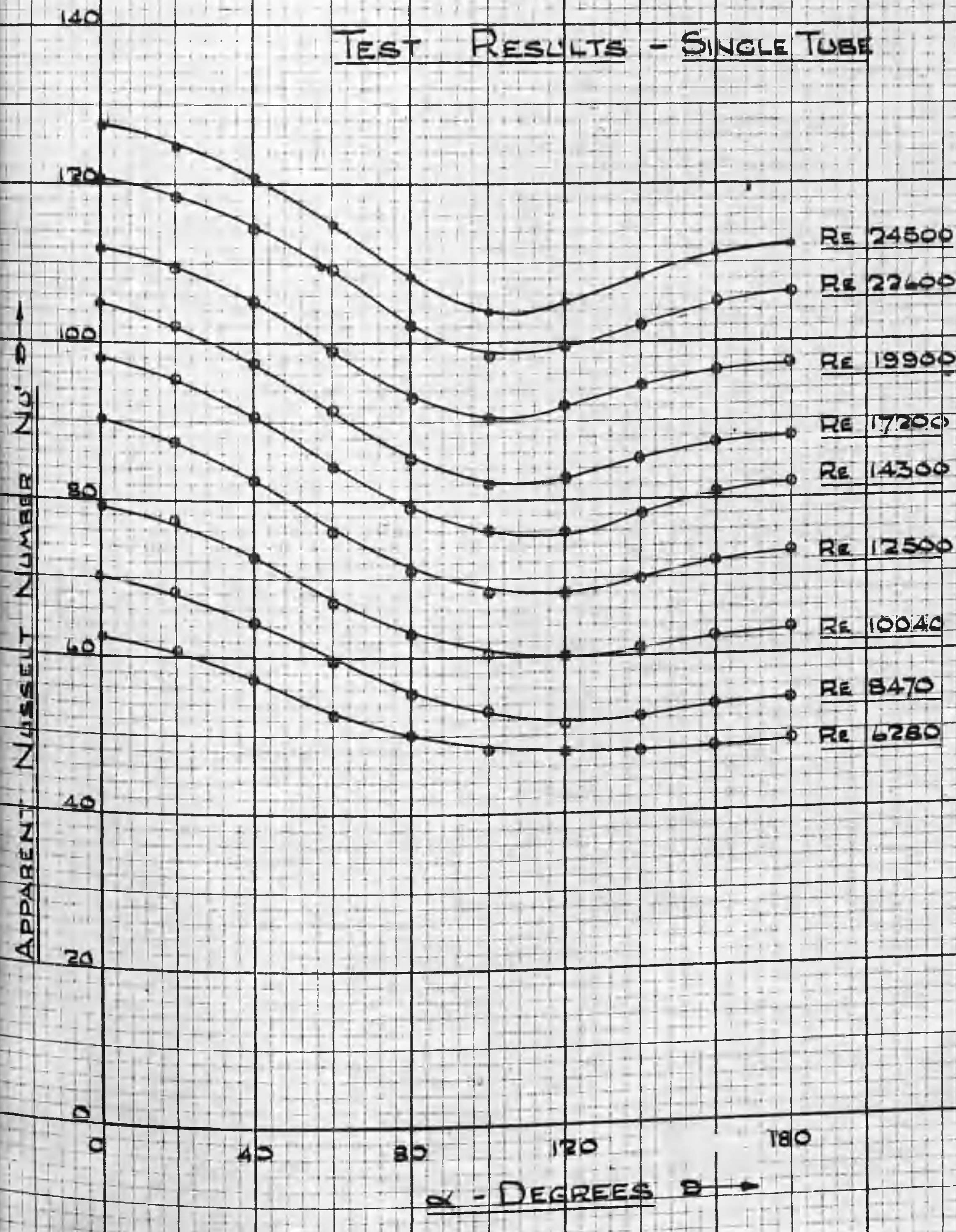
Q_i = heat rejected at inner surface of element

a_o = external surface area of element

Fig 18

VARIATION OF APPARENT NUSSELT NUMBER WITH ANGLE

TEST RESULTS - SINGLE TUBE



The real Nusselt Number for the external surface of a tube can be deduced from the "apparent" values given in Fig. 18 by applying equation 4. This has been done for many tests, the double differentiation being obtained by graphical means, or by a method of finite differences and the values so derived for four tests are shown in Fig.19. These curves, it will be seen, are in reasonable agreement with those deduced by other experimenters, as given in Fig.4.

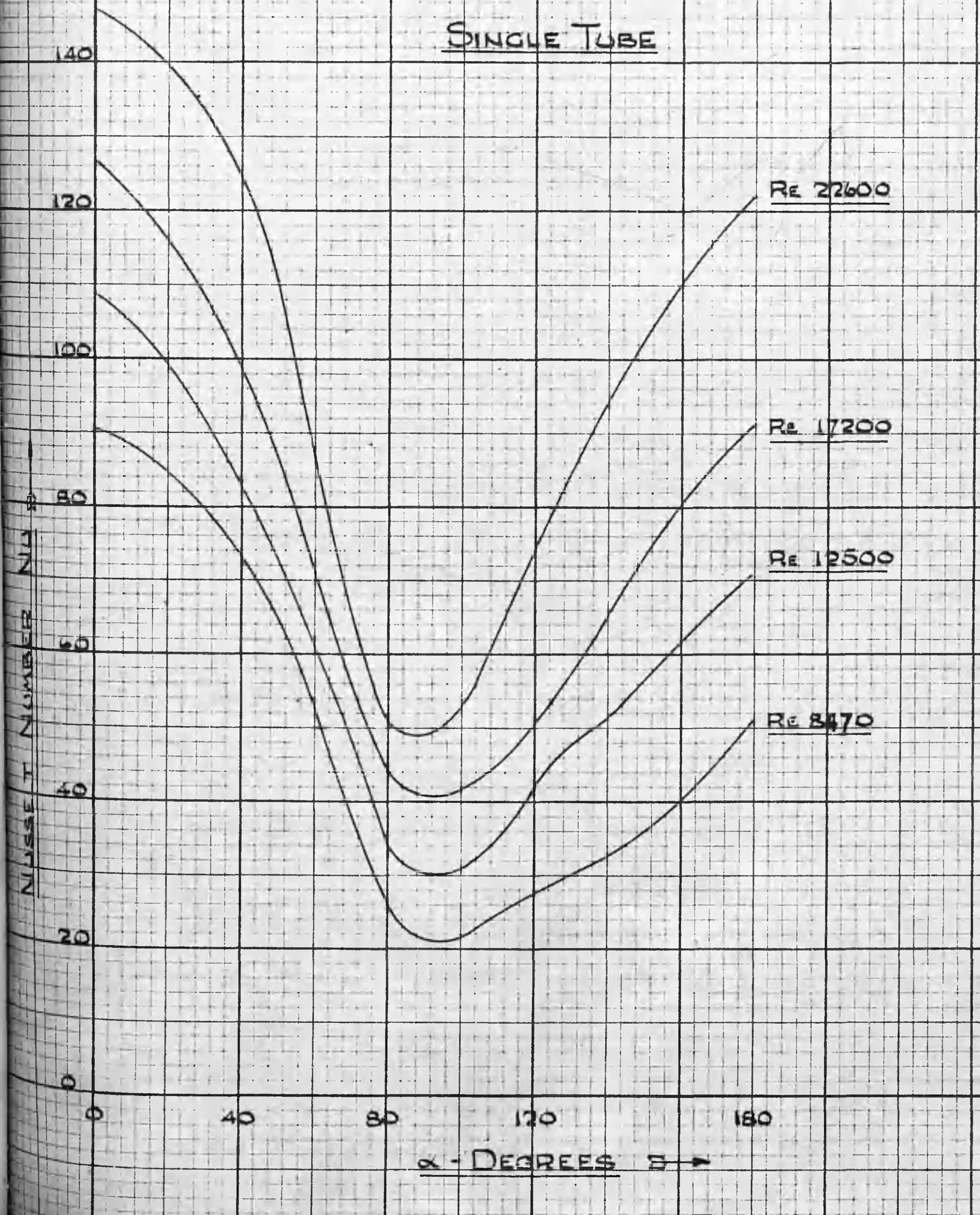
The mean ordinates of the curves of Fig.18 give the mean value of Nu for the tube at any Re. These mean values are found to agree with those given in Table 1, appendix 1, which were obtained by measuring the mean water temperature rise through the tube. Thus, the general accuracy of the readings is shown and justification provided for the assumption made that equal quantities of water pass through each slot in the test tube.

(c) <u>Effect of Reynolds Number on Heat Transfer at Front and Rear of Tube</u>: In the curves shown in Fig. 19 the relative increases in the Nusselt Number at front and rear of the tube for varying Reynolds Numbers can be seen. As Reynolds Number increases, the Nusselt Number increases more rapidly at the rear than at the front. This characteristic has already been noted by previous workers and is emphasised by the gradients of the graphs in Fig. 20, where the values of Nusselt Number at 0^{o} and at 180^{o} have been replotted to a

Fig 19

Variation of Nusselt Number with Angle

Single Tube



base of Reynolds Number. For comparison, the graph of Nusselt Number at 0^{o} as obtained theoretically by Squires[35] is also shown in Fig. 20 and is in remarkable agreement with the present work.

Similar features are evident to a lesser degree in the plots of apparent Nusselt Number for the inner tube surface as shown in Fig. 18, where, in addition, it will be noted that the minimum value of Nusselt Number occurs nearer the front of tube as the Reynolds Number increases. These two facts in conjunction indicate the very marked effect of Reynolds Number on the heat transfer distribution and the increasing effectiveness of the rear portion of the tube as the Reynolds Number rises.

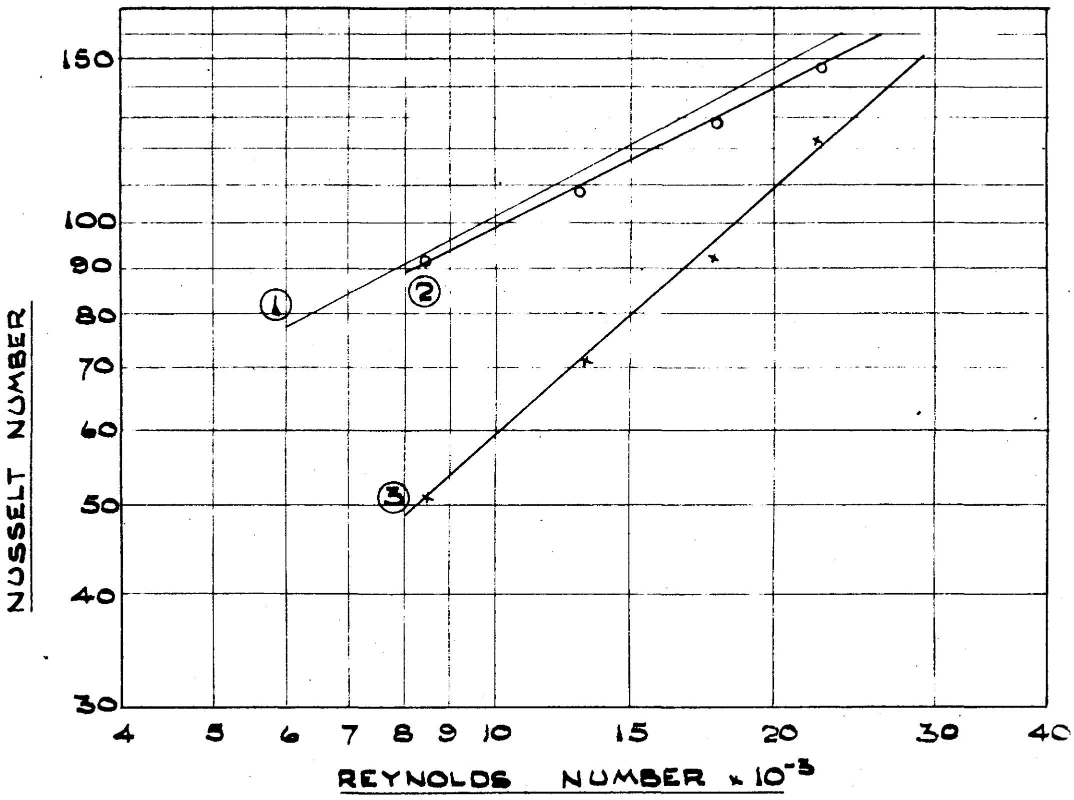
It should again be stressed that all these features are inter-related with the temperature distribution curves.

(c) Effect of Tube Thickness: It is obvious from equation 3, page 41 that the tube thickness has a marked effect on the relation between heat transfer rates at the inside and outside surfaces of a tube. As the tube thickness tends to zero the disparity between outside and inside rates tends to disappear.

To obtain evidence of this effect, tests 27 to 31 (Table 1) were carried out using a tube of wall thickness

FIG. 20

NUSSELT NUMBER AT 0° AND 180° TO REYNOLDS NUMBER FOR SINGLE TUBE



1 - SQUIRES - $Nu = 1.01\sqrt{Re}$

2 - 0°

3 - 180°

0.032 ins. as compared with the normal tube thickness of 0.059 ins. used in the other tests. The detailed results of these tests are given in Table 2, appendix 1, and are plotted as Nu^i to angle for various Re in Fig. 21. The variations in the rate of heat flow in tests 3 and 30, both at Re of 22600, are shown in Fig.22. The difference in the curves of heat transfer rate at the inside of the two tubes of different thickness is obvious, the thinner tube values of Nu^i being nearer those for the external tube surface.

Reference may be made here to the experiments of Paltz and Starr reported by Drew and Ryan[19], in which a thin tube 3.2 ins. outer diameter and 0.125 ins. thick was used. The results of the tests showed that the heat transfer distribution curve for the internal surface of the tube agreed closely with those for the external surface for similar values of Re, as obtained by other experimenters. It would appear now that this, so far from being general, is, in fact, due to the exceptional dimensions of the tube used.

This statement is substantiated by consideration of equation (3), page 41. The circumferential heat flow in any tube is dependent on the ratio $\frac{r_o - r_i}{r_o . r_m}$ where r_o and r_i are the tube radii at the outside and inside respectively and r_m is the mean radius. This factor has a value of 0.045 from the tube used by Platz and Starr as compared with a value of

Fig. 21

Variation of Apparent Nusselt Number with Angle.

Test Results using Thin Wall Tube

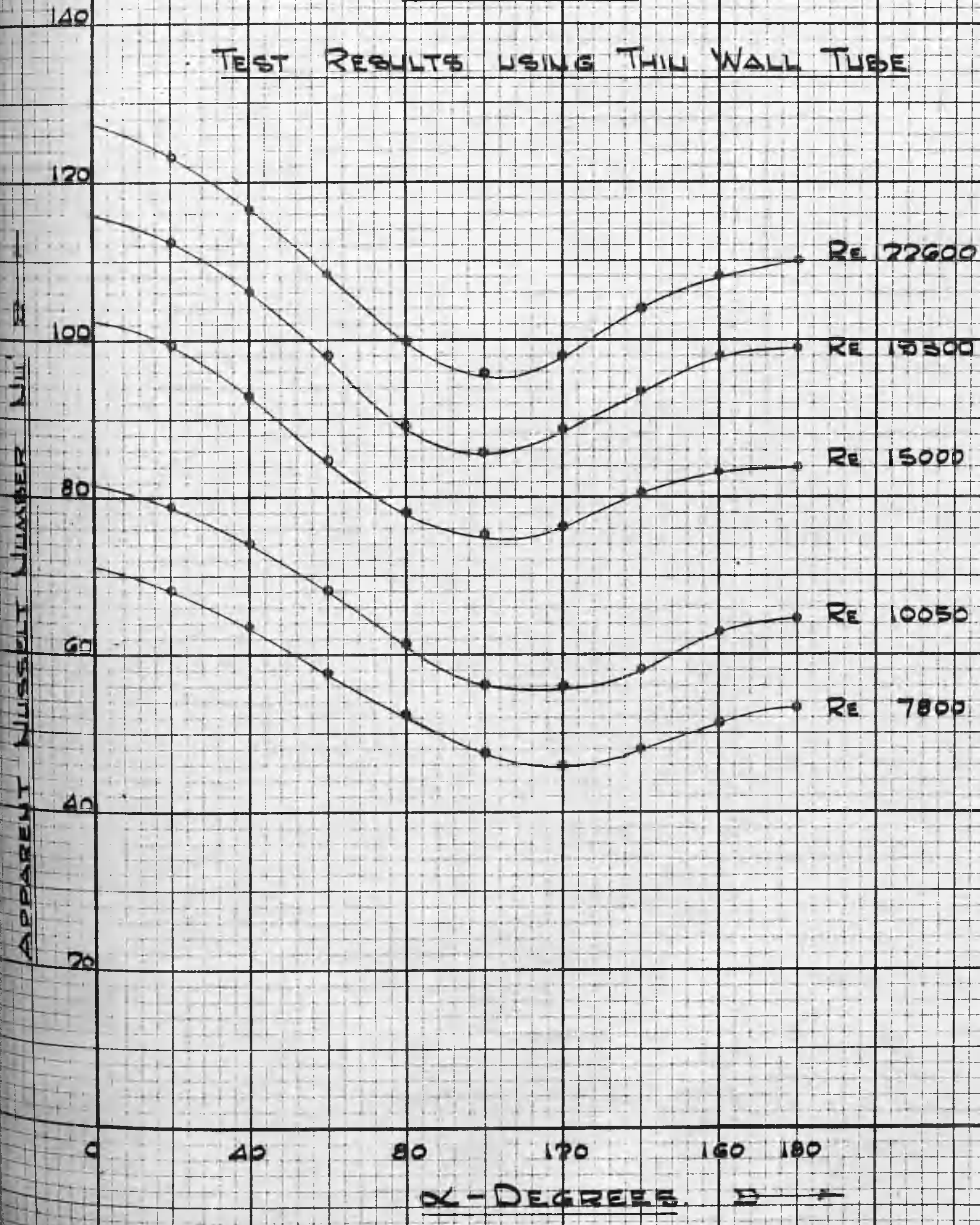
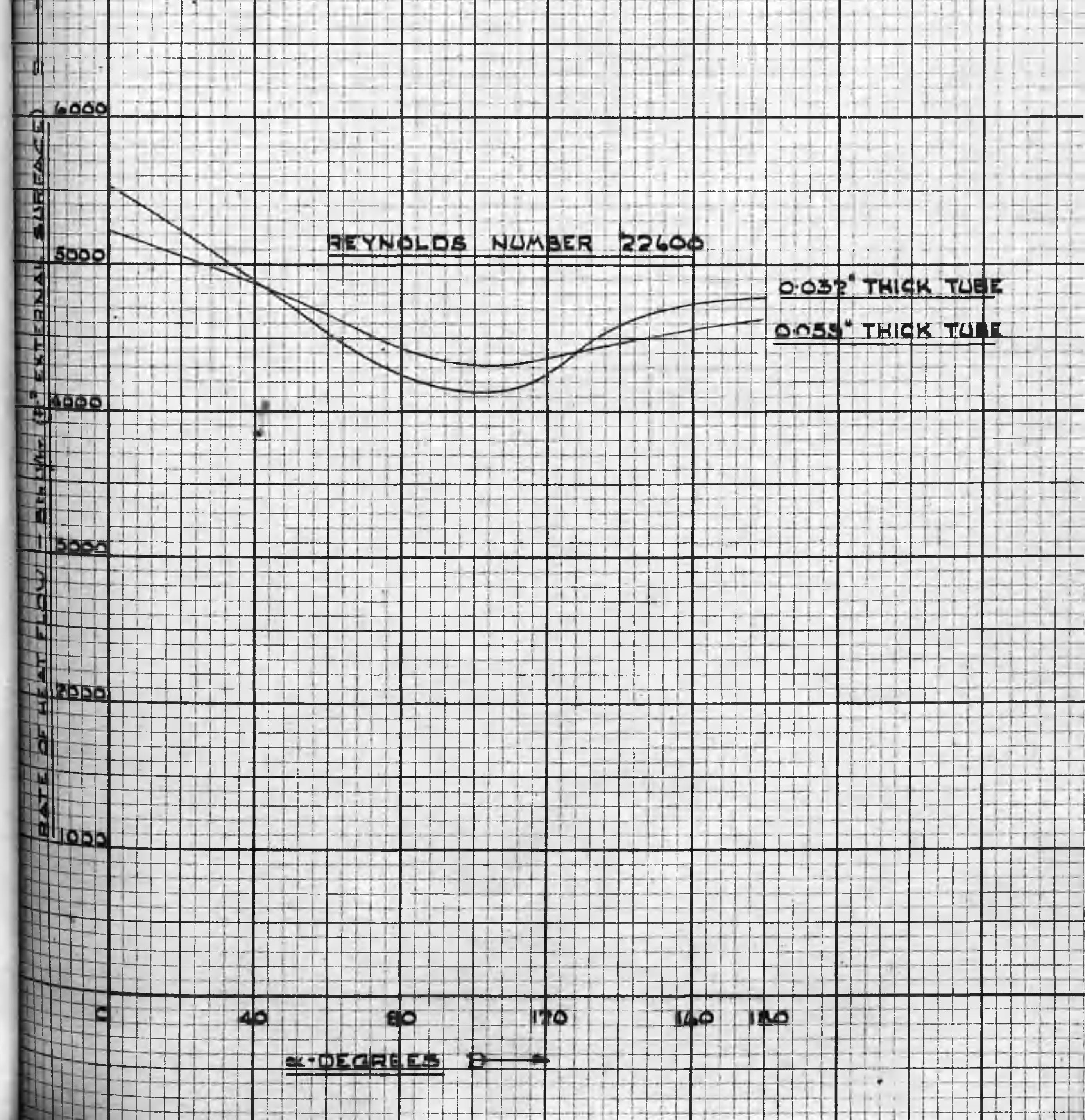


FIG. 22

EFFECT OF TUBE THICKNESS ON VARIATION OF HEAT FLOW RATE MEASURED AT INSIDE SURFACE, BASED ON OUTSIDE SURFACE AREA.



1.1 for the tube used in tests 1 to 26 in the present experiments. The change of circumferential heat flow at any point around the tube used by Paltz and Starr would thus be very much reduced.

It should be noted that the difference between the value of t_{s_m} in tests 3 and 30 is due to a difference of approximately 10^{o}F in the water temperature at entry to the tube.

B - TUBE BANK RESULTS

Tests were made using the various tube groupings as shown in Fig. 23. The tube pitch and spacing were kept constant throughout and the number of rows of tubes varied. In each series of tests, measurements of heat transfer and tube wall temperature were made, as previously described, at the positions marked ⊗ in Fig. 23. The central positions were chosen so that the air flow around the test tube would not be affected by the sides of the duct. The same test tube was used to take all readings by changing it from row to row as required.

In presenting the results of the tests, the variation in the average heat transfer coefficient between the air stream and the tube bank in the four arrangements will first be dealt with. This variation will then be examined by considering the mean coefficient for a tube in each row of tubes. Lastly an explanation of this row to row variation will be sought in the results of the tube wall temperature and heat transfer distribution around the test tube when placed in the various rows.

It should be noted that the results presented and the deductions drawn therefrom, strictly apply only to tubes spaced as in these tests, although an indication of conditions to be expected with other tube arrangements may be found.

Unless otherwise stated, the values of Nusselt Number and Reynolds Number are based on the measured temperature of the air just before the front row of tubes.

(1) Mean Heat Transfer Coefficient between air and external tube surface.

In Tables 3, 5, and 9, Appendix 1, are given the experimental and deduced results of the tests on the four different tube arrangements of Fig. 23. The results were obtained by similar methods to those described for the single tube except that no account has been taken of the small corrections for radiant heat transfer. For the tests on the six row arrangement, the temperature of the air leaving the bank is also given.

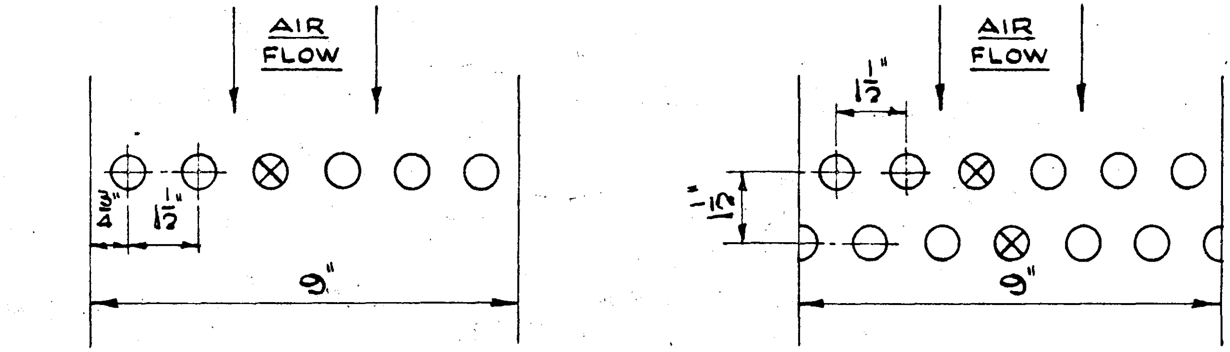
(a) Average Nusselt Number for Banks of Tubes: Fig. 24 shows the results plotted as the average Nu for the tube arrangement to a base of Re. Also shown to a base of Re is the average Nu for the six row bank evaluated using the mean air temperature through the bank.

From this figure it may be seen that, in the given range of Re, the mean Nu for the two row bank is similar to that for the single row of tubes, the value being some 4% less than that for a single tube, as shown in Fig.11. The average heat transfer to a three row bank, however, is some 8% higher and is again higher for the six row arrangement.

FIG. 23

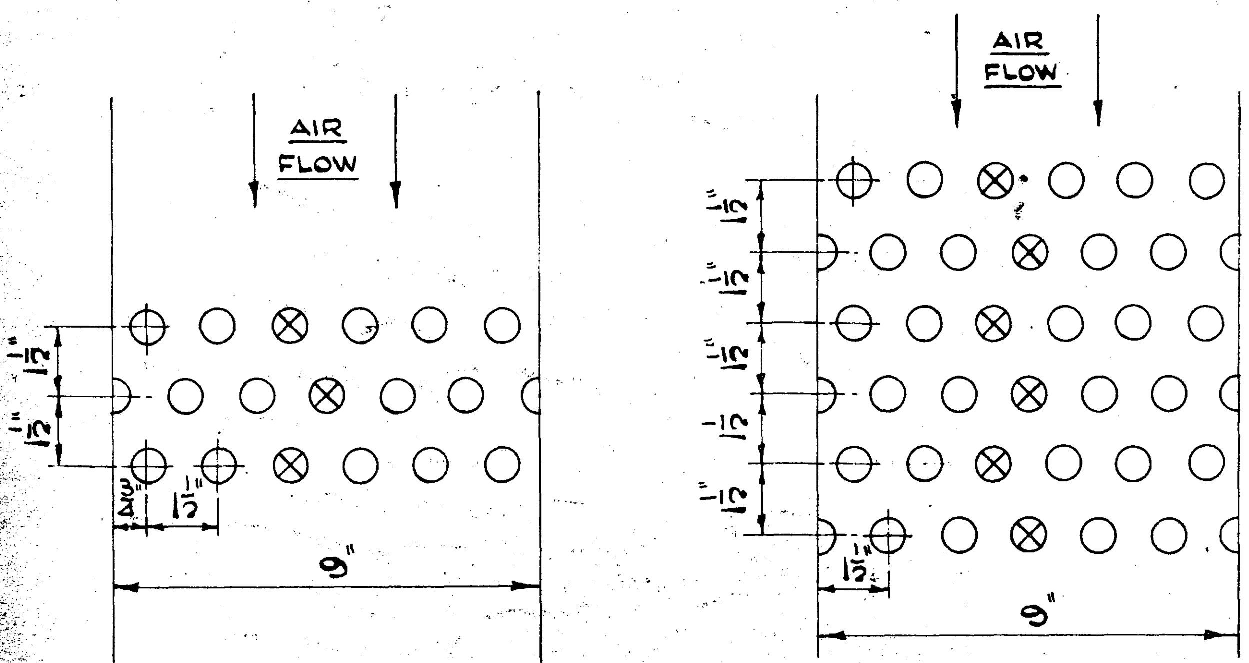
TUBE BANK ARRANGEMENTS

TUBE DIAMETERS IN ALL TESTS $\frac{3}{4}$".



(a) (b)

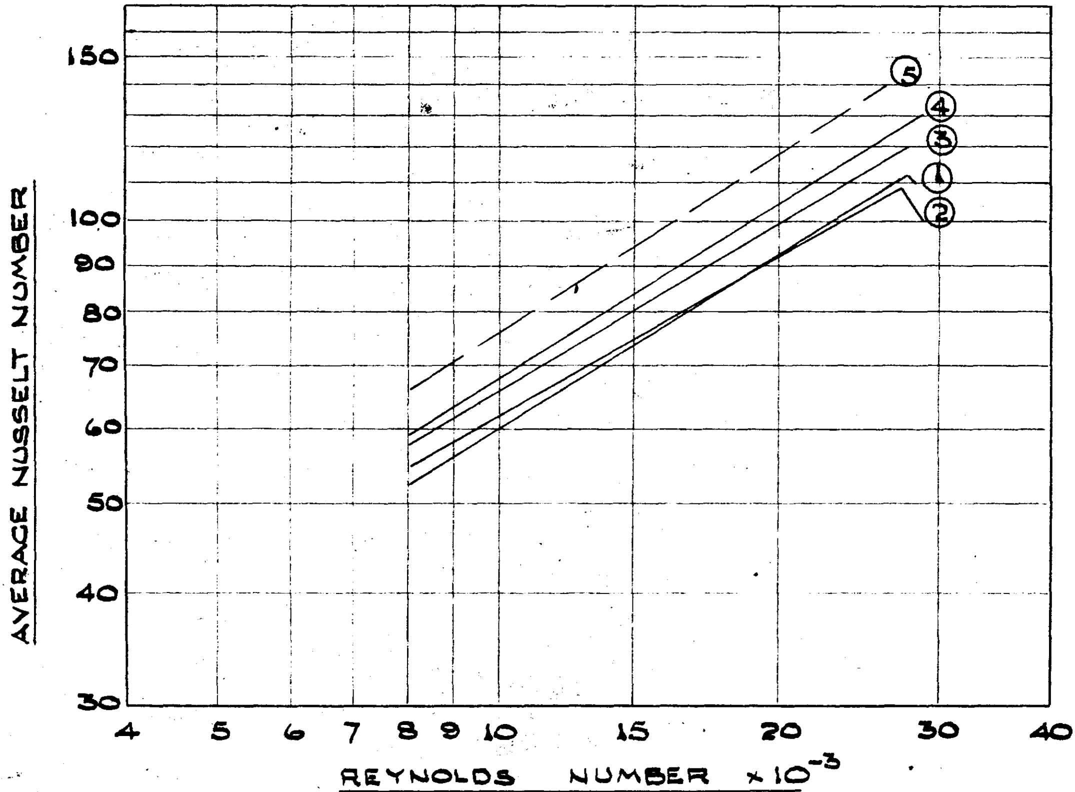
⊗ INDICATES TEST TUBE POSITION.



(c) (d)

FIG. 24

AVERAGE NUSSELT NUMBER FOR TUBE BANKS TO REYNOLDS NUMBER



(1) SINGLE ROW BANK
(2) TWO ROW BANK
(3) THREE ROW BANK
(4) SIX ROW BANK

} BASED ON AIR APPROACH TEMP.

(5) SIX ROW BANK - BASED ON MEAN AIR TEMP.

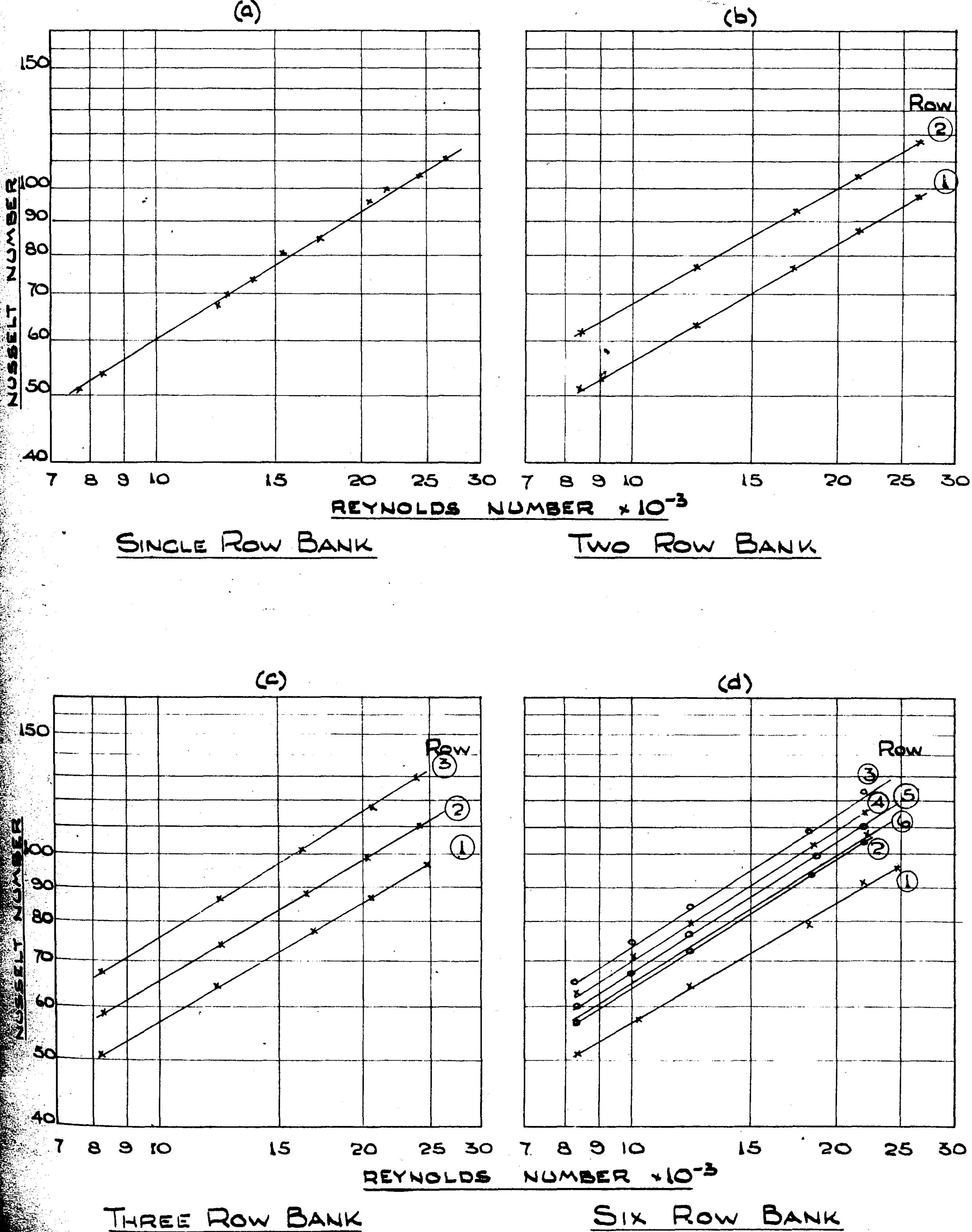
A comparison of the curves of Fig.24 with those obtained by previous experimenters using similar tube arrangements shows that the values of Nu obtained in the present tests are higher than those of Reiher[7] though lower by some 5% than those of Griffith and Awbery[5]. The most recent general results are those given by Fishenden and Saunders[28]. These are based on the experimental values of Huge[23] and Pierson[24] and are evaluated using the mean gas temperature in the bank. For a tube arrangement the same as that in the six row bank test, the values of Nu given correspond exactly with curve 5 in Fig 24 which is based on the results of the six row test using the mean air temperature in the bank.

(b) Row to Row Variation im Mean Nusselt Number in Banks of Tubes: In Fig.25 the test results are shown plotted in the form Nu_m to a base of Re for each row in the tube arrangements tested.

It may be seen from the curves of Fig.11 and Fig.25(a) that the heat transfer rate between hot air and a single row of tubes at a pitch of two diameters is less than that for a single tube at a corresponding Re. The placing of a second row of tubes behind this row of tubes causes a further reduction in the heat transfer rate as shown in Fig. 25 (b), but the placing of additional rows downstream does not further alter the heat transfer conditions in the first row.

FIGS. 25 (a), (b), (c), (d)

ROW TO ROW VARIATION OF NUSSELT NUMBER TO REYNOLDS NUMBER



The presence of the first row in front of the second row causes a greater heat transfer to take place in the second row than would be obtained in a single row. A similar increase is noted in the value of Nu_m for the third row tubes, though the value appears to decrease slightly in each succeeding row, the value in the sixth row being little above that for the second row. It should be noted, however, that the value of Nu_m for the third to sixth row tubes remains constant if based on the mean air temperature at the row, as deduced from the temperature drop through the bank. The lower value of average Nu obtained with banks of two or three rows is thus seen to be due to the low heat transfer rate in the first and second row tubes of a bank.

To compare the present test results with those of Griffith and Awbery[5] and Winding[29], who have obtained values of the row-to-row variation in Nu_m in the first three rows of a staggered tube bank, Fig.26 is given. All the results are based on the mean gas temperature at the row and relate to approximately similar values of Re. The tube spacings in the three tests vary widely, yet a very close agreement of results is obtained. The direction of heat transfer in the other experiments was opposite to that in the present tests, the heated tubes being cooled by an air stream.

FIG. 26

TABLE OF MEAN HEAT TRANSFER COEF. IN 1ST & 2ND ROW TUBES AS COMPARED WITH VALUES FOR 3RD ROW – STAGGERED TUBE BANKS.

SOURCE	RE	1ST ROW	2ND ROW	3RD ROW
WINDING (29)	20,200	64%	83%	100%
GRIFFITH & AWBERY (5)	22,000	61%	76%	100%
PRESENT TESTS	22000	66%	79%	100%

Experiments by Reiher[7] have shown that the heat transfer rate to a single tube may be increased by promoting turbulence in the approach air stream; and Griffith and Awbery[5] obtained equal heat transfer rates in each of the first three rows of a tube bank by making the air stream turbulent before it reached the bank. Thus the increased heat transfer in the second and succeeding rows can be attributed to the turbulence created in the air stream by the first and second row tubes.

The reasons underlying the row to row variation of Nu_m are further examined on page 55

It may be noted at this point, that the overall air temperature drop through the tube bank as measured for the six row tests is up to 5% greater than that calculated from the heat input to the tubes. This difference may be attributed to the difficulty of obtaining a true mean air temperature across the duct at the bank outlet and to heat lost to the ducting and half tubes in the test section.

2. Tube Wall Temperature Variation around Tubes

The curves of tube wall temperature to a base of angle for a tube in each row of the six row bank are shown for three values of Re in Fig. 27. The temperature variations in the cases of the other tube arrangements are given in tables 4, 6 and 8 in Appendix 1, while the complete values for the six row arrange-

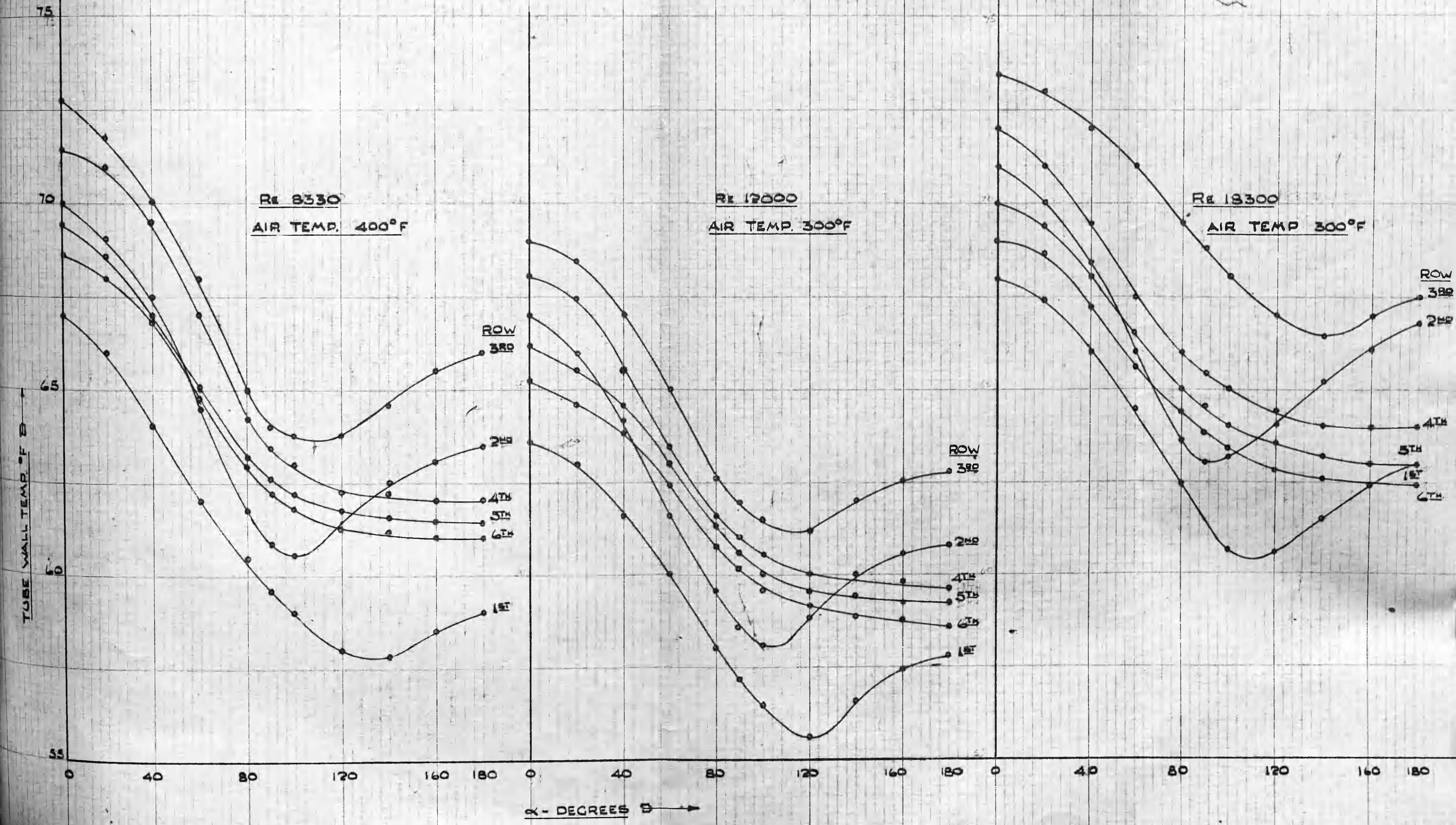
:ments are given in table 10 Appendix 1. The table values show that the temperature variations were the same in similar rows of all arrangements except in the case of the last row tubes, where the value of the tube temperature at the rear of the tube was slightly higher. Direct comparison of the various temperature readings is not permissible as the inlet water temperature was not constant.

The curves of Fig. 27 show the marked effect of tube position on tube wall temperature distribution. For each value of Re, the maximum temperature gradient round the tube in the second row was greater than that round the tube in any other row. From this it may be deduced that the maximum circumferential heat flow will be found in the second row tubes. This value occurs at approximately 60° from the upstream point.

The curves also show that the minimum temperature in the first and second row moves towards the sides of the tube as Re is increased, as was the case for the single tube experiments. For third row tubes, however, the minimum temperature moves nearer to the rear of the tube with increased values of Re. The tube temperature for fourth and succeeding rows falls to a minimum value at the rear of the tube for all values of Re.

FIG. 27

VARIATION OF TUBE WALL TEMP. FOR SIX ROW TUBE BANK.



(3) <u>Variation of Heat Transfer Coefficient around the Tubes</u>:

The water temperature rise, θ , through the test slot for the different angular positions are given in Tables, 4, 6, 8 and 10, Appendix 1 for the various tests of Tables 3, 5, 7 and 9. A selection of these results for each tube arrangement are plotted as "apparent" Nusselt Number to a base of tube angle for various Re in Figs. 28 to 31. The curves show the heat transferred at various points round the inside surface of the tube under various flow conditions.

(a) <u>Single Row of Tubes</u>: Comparison of the curves in Fig.28 and 25(a) shows that the heat transfer distribution around the inside of a tube in a single row differs considerably from that in a single tube. The value of Nu^1 at 0^o and 180^o is slightly less for the row tube due to the increased conduction round the tube caused by the greater temperature gradient. The minimum value of Nu^1 or true Nu at the sides is considerably reduced by the presence of the tubes along side.

The difference between the curves of Nu_m for the single tube and the single row of tubes (Fig.11 and 25(a)) at first cast doubt as to the use of the velocity based on the minimum flow area as a basis of comparison. From the curves of heat transfer distribution in the two cases it may be deduced that the flow pattern round a tube in a single row is different to that for a tube in open flow.

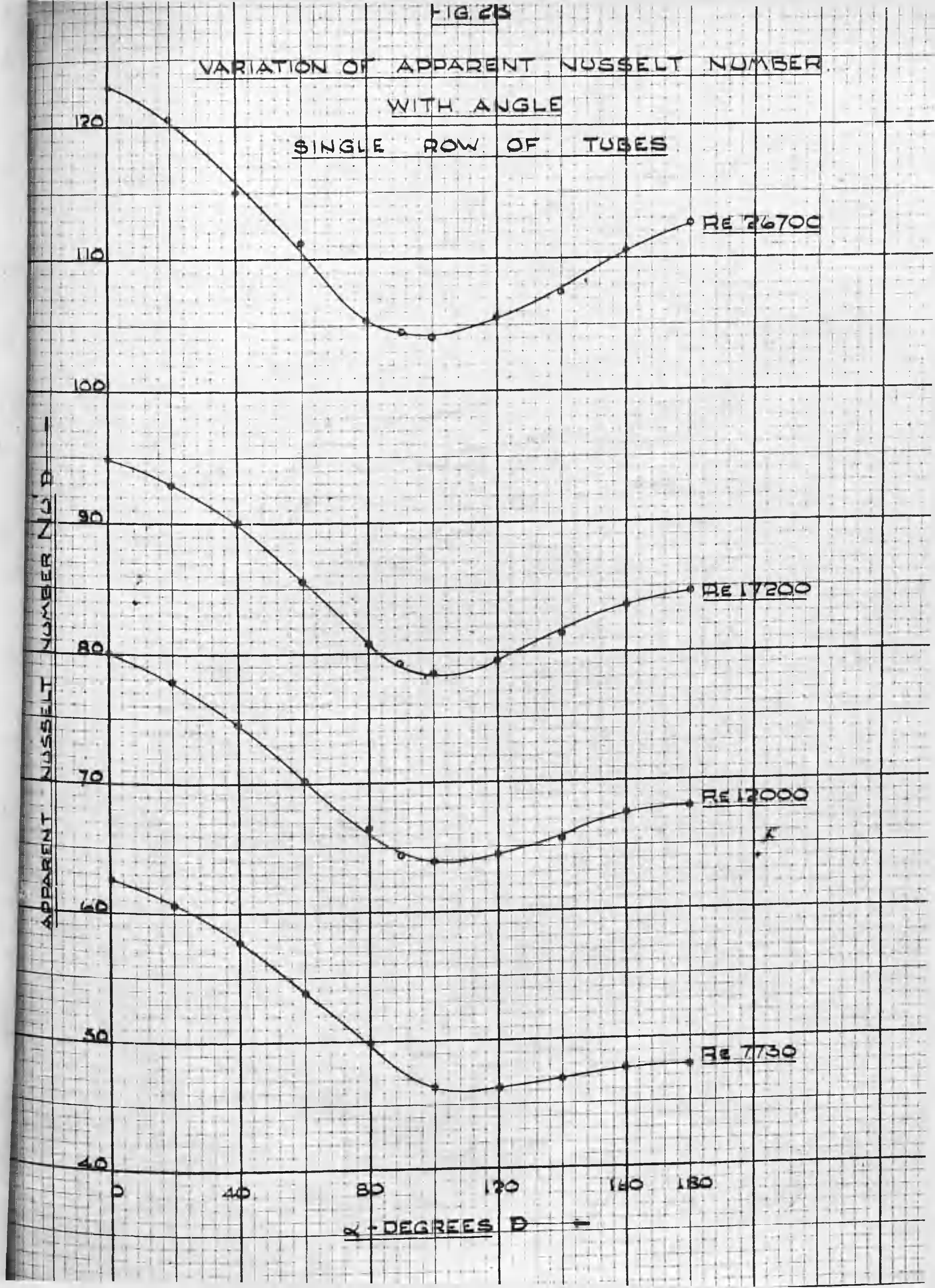


FIG. 28
VARIATION OF APPARENT NUSSELT NUMBER
WITH ANGLE
SINGLE ROW OF TUBES
Re 26700
Re 17200
Re 12000
Re 7730
120
110
100
90
80
70
60
50
40
0
40
80
120
160
180
APPARENT NUSSELT NUMBER Nu D
α - DEGREES

FIG. 29

VARIATION OF APPARENT NUSSELT NUMBER WITH ANGLE FOR TWO ROW TUBE BANK

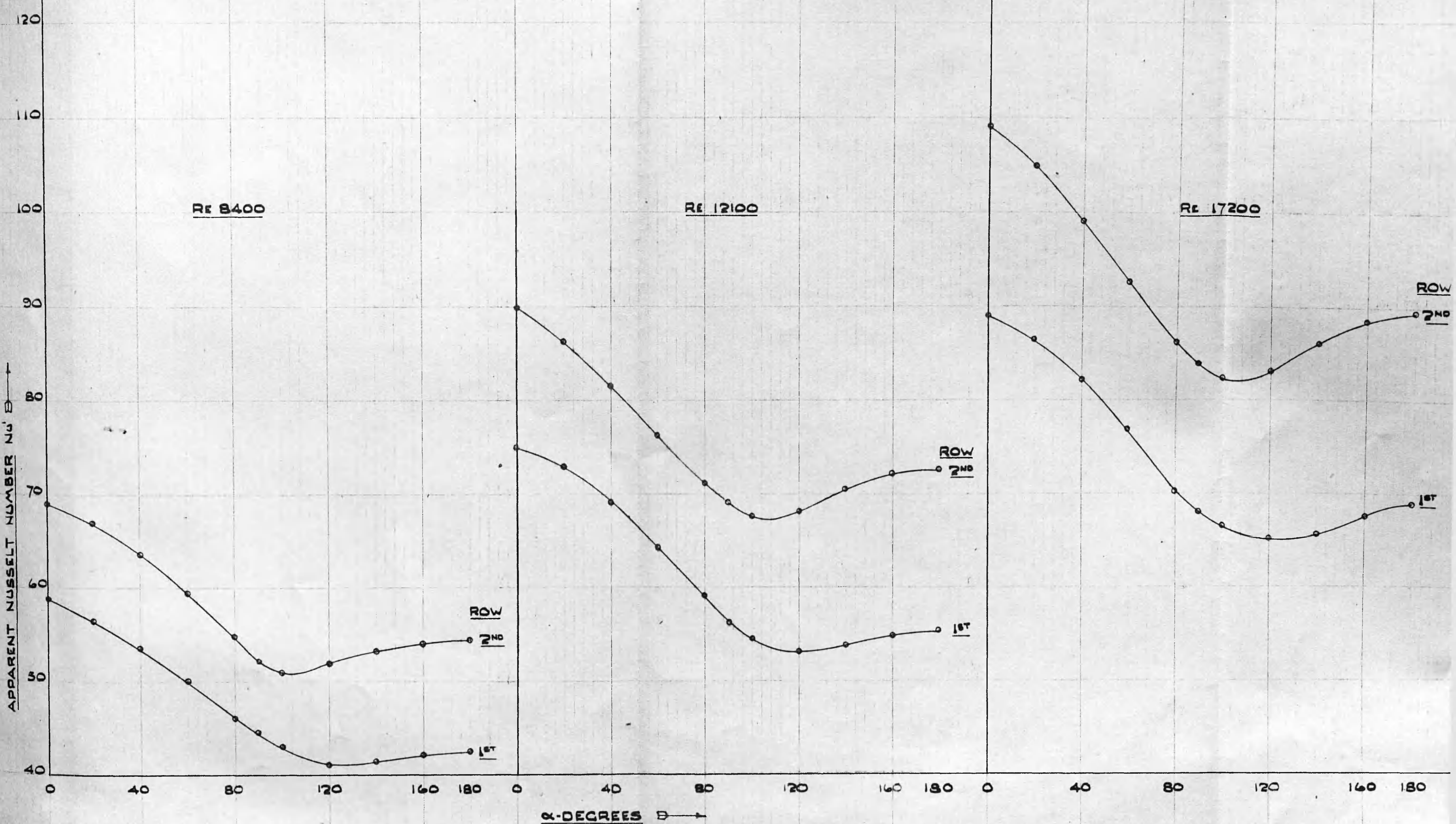


FIG: 30

VARIATION OF APPARENT NUSSELT NUMBER WITH ANGLE FOR THREE ROW TUBE BANK

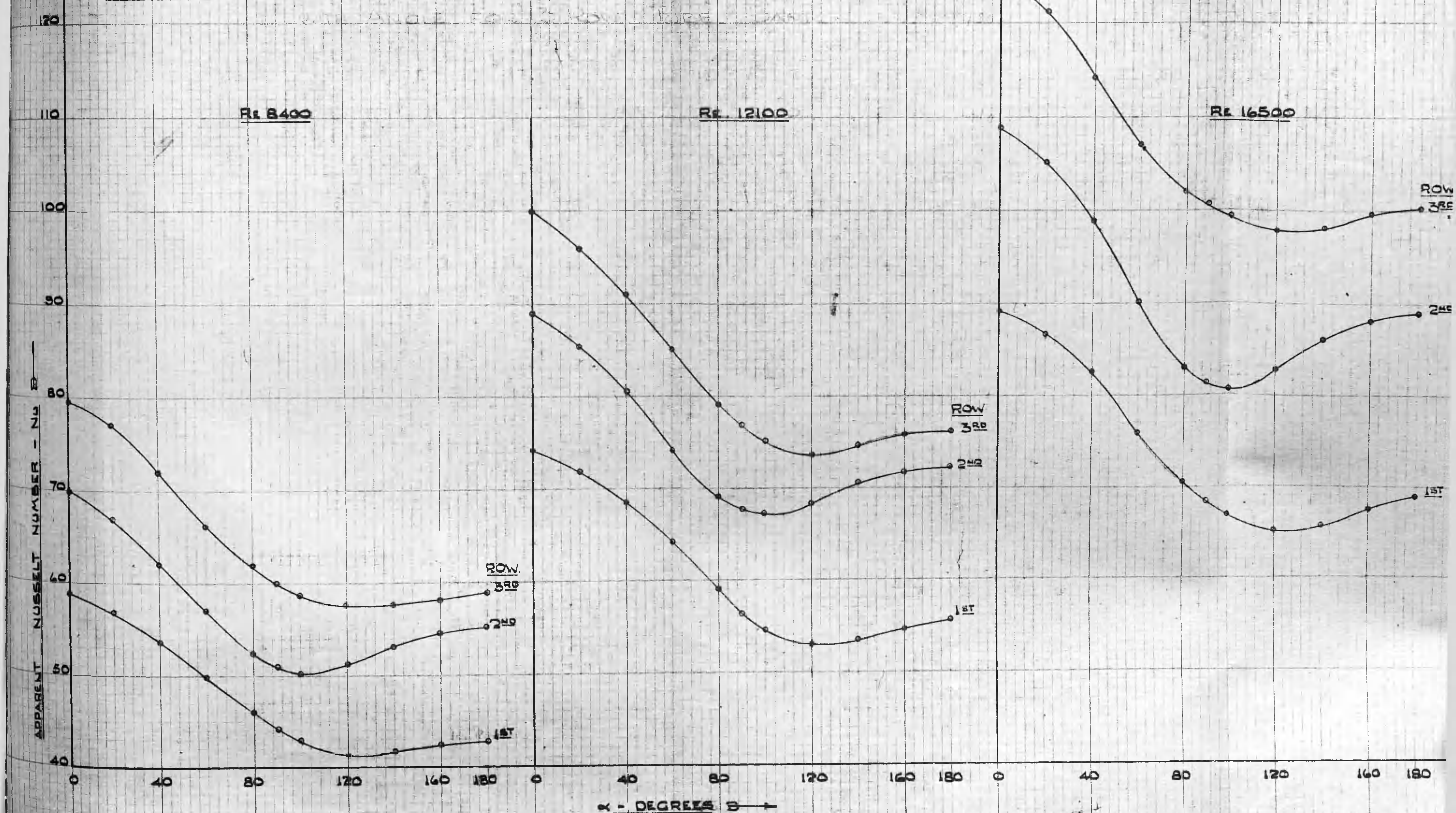
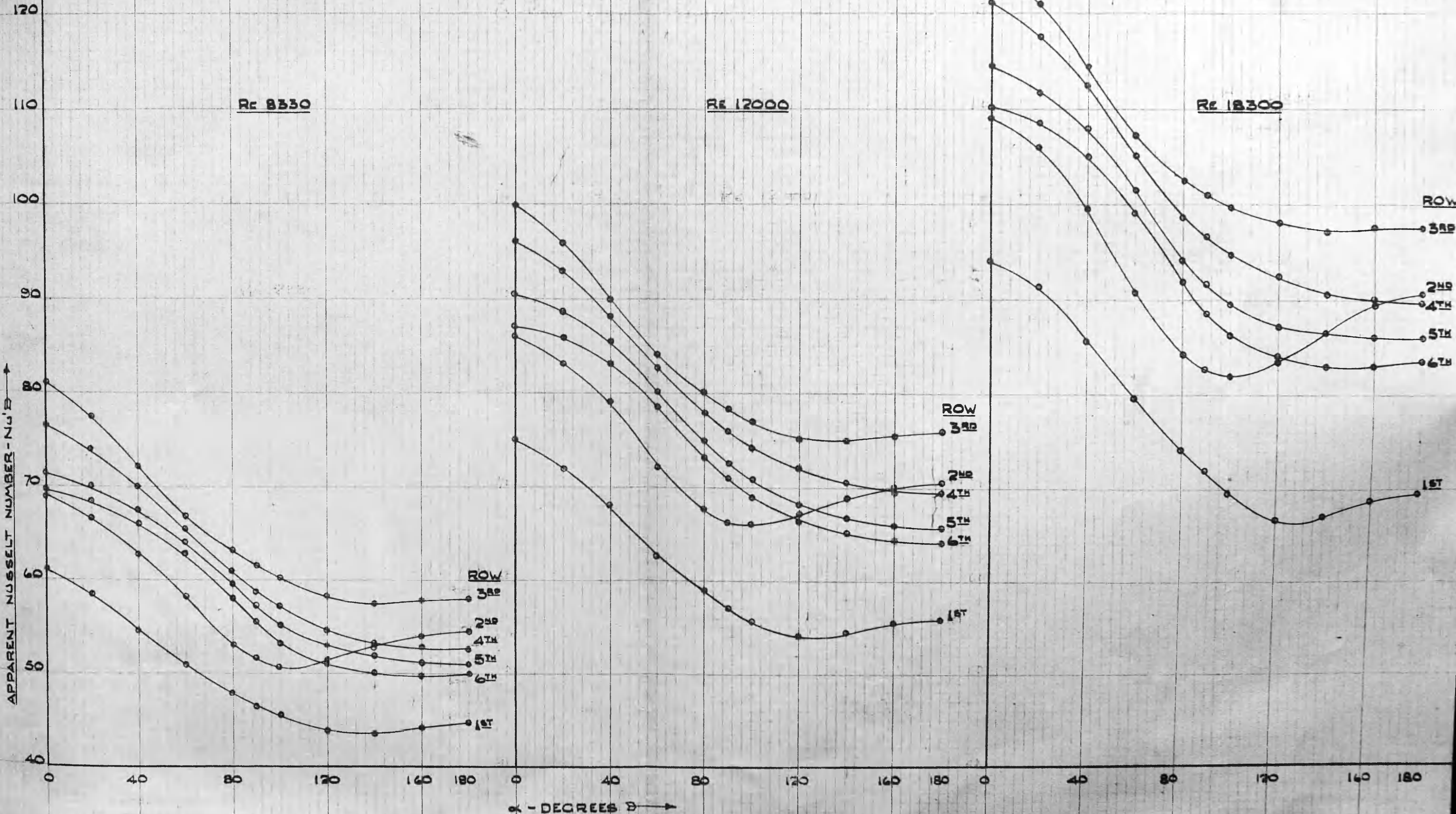


FIG. 31

VARIATION OF APPARENT NUSSELT NUMBER WITH ANGLE FOR SIX ROW TUBE BANK



(b) Two and Three Rows of Tubes: The decrease in mean heat transfer rate to first row tubes when two or more rows are present is due to the reduction in the heat transfer taking place at the sides and rear of the first row tubes. The point of minimum heat transfer is nearer the rear of the tube as also is the point of minimum temperature. Thus we see that the heat supplied to the front of the tube is the same as for a single tube, but due to the different air flow conditions less heat is given to the rear of the tube.

The higher mean value of Nu_m for second row tubes compared with the first row is fully explained by the transfer at the upstream point which falls to a minimum value nearer the front than in the case of a single row of tubes. There is also a pronounced increase towards the rear again. The mean heat transfer rate for the second row is only slightly reduced by the addition of the third row, Fig. 25(b) and (c), this small reduction being due to the point of minimum heat transfer shifting nearer to the front of the tube.

The rapid fall in Nu^1 from front to side of a second row tube and the rapidly changing tube temperature curve is evidence of the very high rate of heat transfer to the outer surface at the upstream point of this tube.

(c) Six Row Bank of Tubes: The points regarding circumferential heat transfer distribution in the previous section apply equally to the curves for the first three rows in the six row bank. In the fourth and succeeding rows the curves of Nu^1 to angle Fig. 31 show a similar form to the temperature curves for these rows, that is, a gradual fall from a maximum at 0^o to a minimum at 180^o.

In Fig. 32 are shown the approximate curves of Nu to tube angle, i.e., the true Nusselt Number for the outside of the tube to angle around the tube for a tube in each of the six rows, the Reynold's Number being 18300. These values are obtained by the application of equation 4, page 41. The values of $\frac{d^2t}{d\alpha 2}$ were obtained by graphical methods. A similar series of curves for Re equal 8330 are shown in Fig. 33.

From these curves it may be seen that the variation in heat transfer rate around the second row tube is greater than that around any other and the maximum value of heat input for any point in the bank is obtained at the front of the second row tube. The trend is for the variation in the heat transfer distribution to be smaller, the further the tube is from the front of the bank.

The heat transfer variations around a fifth row tube of a bank of staggered tubes as obtained by Winding and Cheny[17] are shown in Fig. 5. These curves indicate a rapid rise to a maximum value of heat transfer at about 120^o from the upstream point. From the results of the present tests, where the value of Nusselt Number is approximately constant over the rear part of

the/

FIG. 32

VARIATION OF NUSSELT NUMBER WITH ANGLE FOR SIX ROW BANK

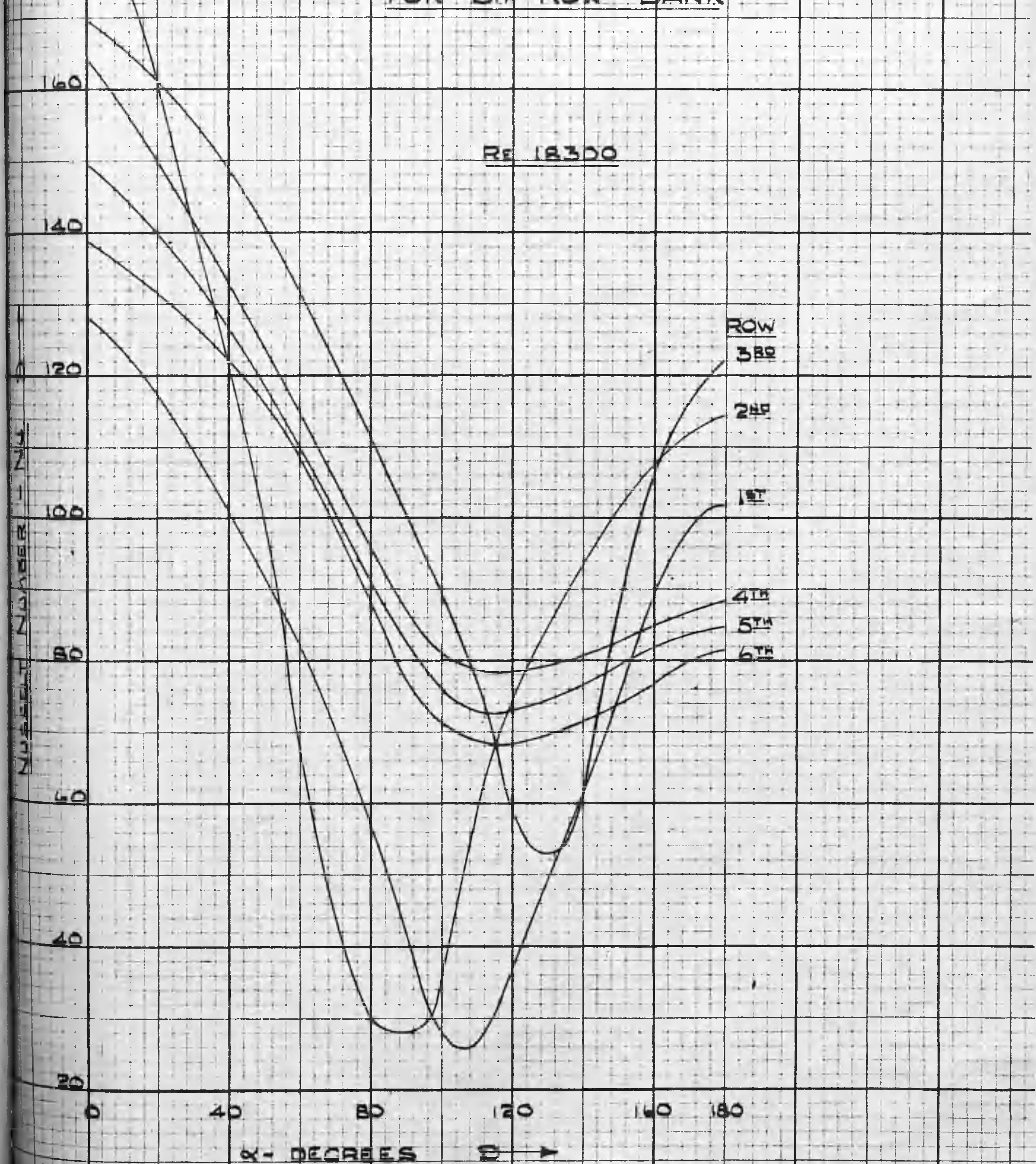
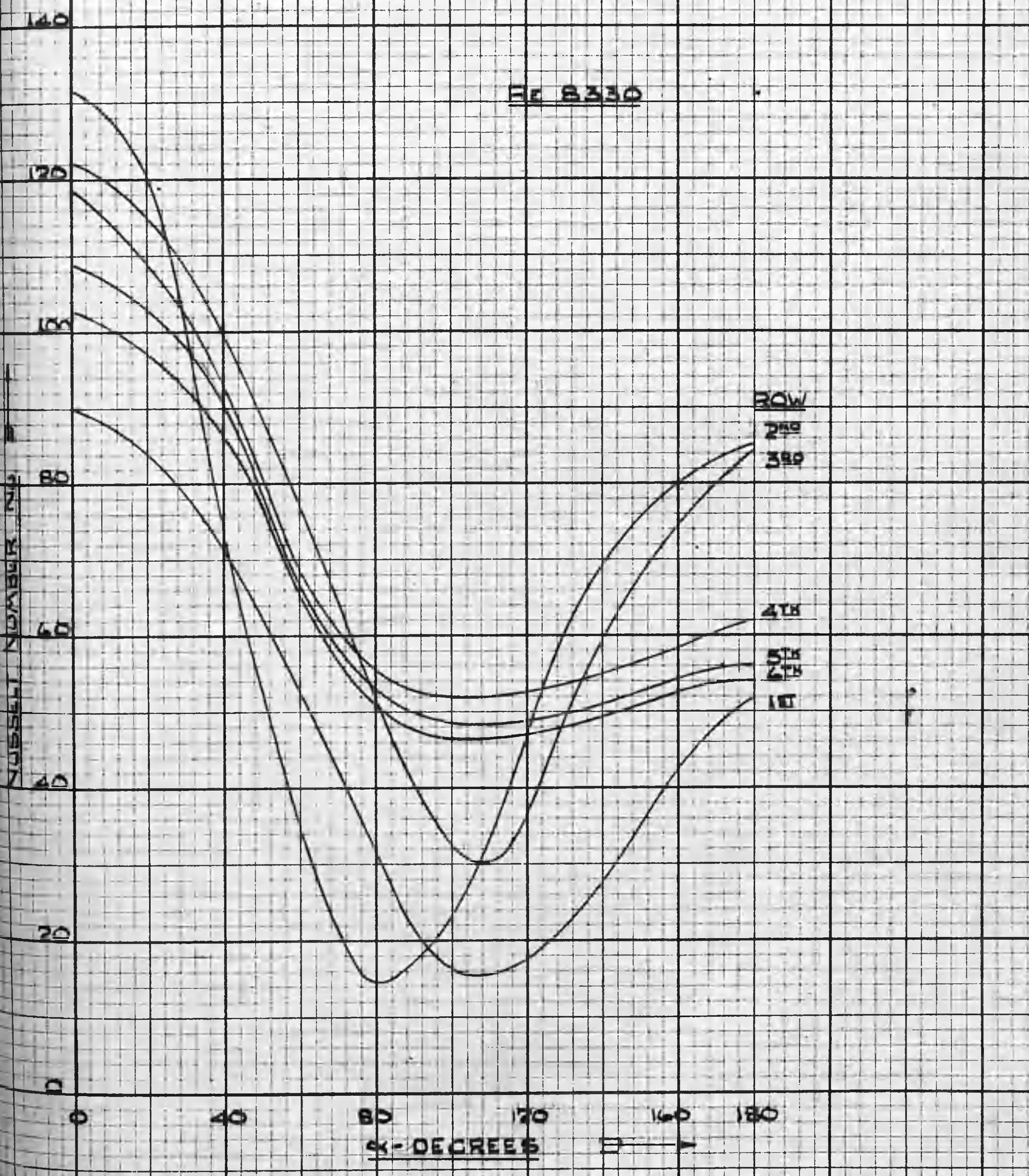


FIG 33

VARIATION OF NUSSELT NUMBER WITH ANGLE
FOR SIX ROW TUBE BANK



fifth row of tubes, there is no indication of this peculiar variation.

C - CONCLUSIONS.

The experimental techniques followed in this investigation have been specially devised to study heat transfer variations around a tube subjected to heating by a cross-flowing fluid, but the mean transfer coefficients, easily deduced from the results, agree well with the values established by previous investigators who were concerned only with mean values.

The correlation established between the inner and outer heat transfer variations and the temperature variation around a tube leads to a clearer impression of the lócal conditions around a tube in cross-flow and emphasises the magnitude of the variations to be expected.

With temperature variations around the tube known, the circumferential heat flow is approximately calculable. Probably in this work it has been somewhat overestimated because of the use of external wall temperatures but the order and significance of the effect is clear.

A noticeable feature of the results obtained is that although the general temperature level or mean temperature of the tube is affected by both the gas temperature and the Reynolds number yet the variations around the tube are dependant only on the Reynolds Number.

The influence of the latter on the variation of the heat transfer coefficients and also on the circumferential temperature and heat flow variations is graphically shown in Fig.34.

The rear portion of a tube in cross flow appears to be particularly responsive to the increased intensity of flow that is measured by higher Reynolds Numbers. The curve in fig. 20 shows very clearly the greater effectiveness of the rear with increasing Re.

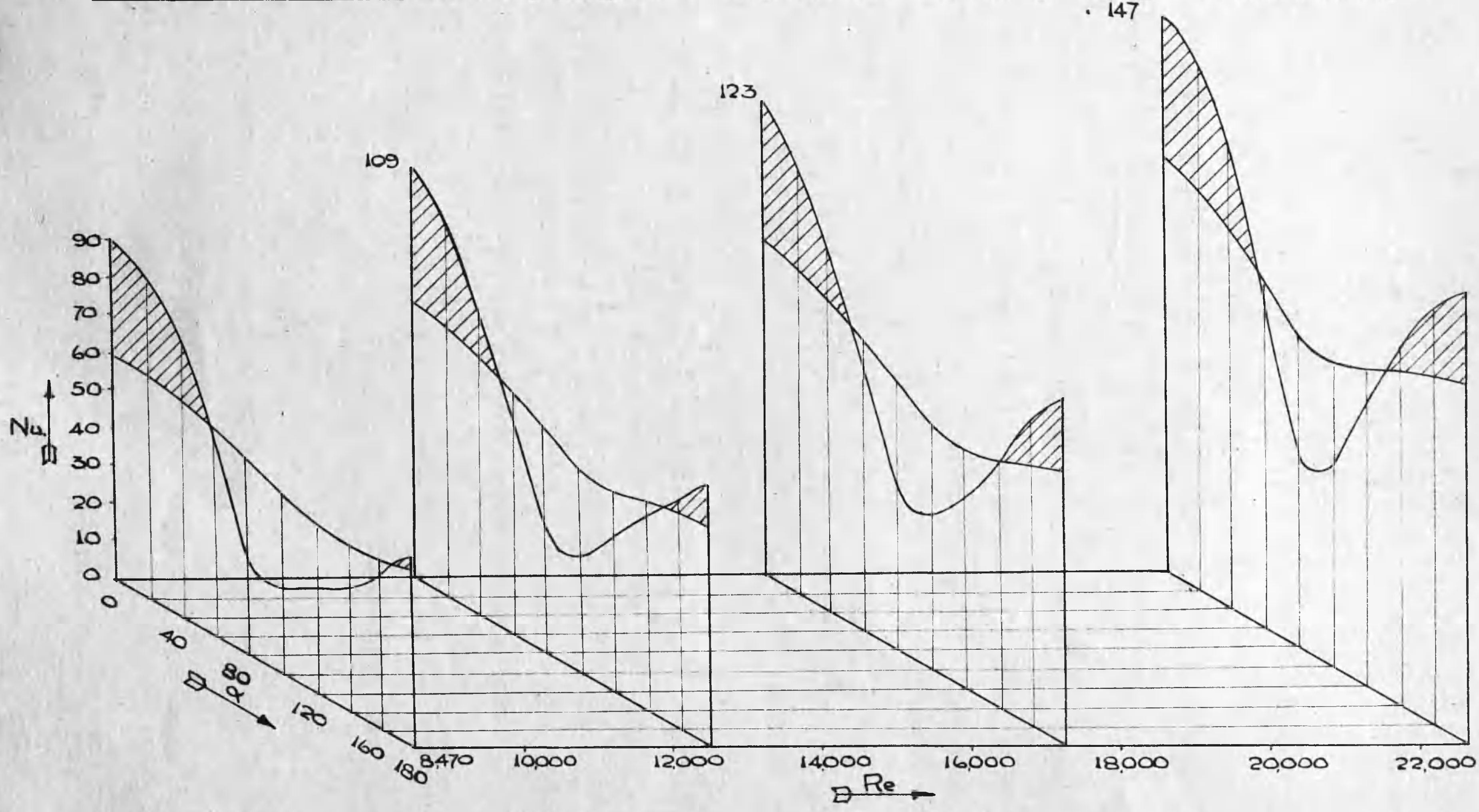
This fact explains the increase in heat transfer rates from the first to the third row in a tube bank. The increased turbulence at the rear of the third row brought about by the presence of the first two rows does in fact correspond to a Reynolds Number virtually higher than the actual.

It can also be deduced from fig. 20 that, for Reynolds Numbers above 40,000, the rear of the tube would become so effective that the maximum heat transfer rate would occur at the rear point instead of at the front point as with lower numbers.

It is seen that the second row boiler tube failures are to a fair extent explained by the uniqueness of the second row conditions established by the results of the investigation. Reference to figs. 27-32 shows that the second row tubes are

Fig. 34

EFFECT OF REYNOLDS NUMBER ON THE VARIATION OF NUSSELT NUMBER AT INSIDE AND OUTSIDE SURFACES – BASED ON EXTERNAL SURFACE AREA.



subjected to both the maximum rate of convective heating and the maximum circumferential temperature gradient. Hydrodynamic tests by previous workers have also established the unique nature of conditions around the second row.

The investigation has shown the close relationship existing between heat transfer characteristics and the features of the flow pattern. The following points serve to link the two types of investigation.

(a) The vortex sheet leaves the side of a first row tube at a point nearer to the rear of the tube than in the case of a single tube; the point of minimum heat transfer is also nearer the rear in a front row tube.

(b) In the second row, the flow breakaway point and the point of minimum heat transfer both take place nearer to the front of the tube than for a first row tube

(c) In third and succeeding rows there is again agreement in the position of points of flow breakaway and minimum heat transfer

(d) The area at the rear of a tube, often referred to in hydrodynamic tests as the "dead-water" area, is seen, in relation to heat transfer, to be a zone of very real activity

The implications of circumferential heat flow on tube temperature stresses are obvious. The results show that considerable temperature stress relief will be accorded the tube at the upstream point where normally due to both convection and radiation effects the local heat input will be a maximum. The temperature stress will also be a maximum there, but at the inside tube surface, and will obviously be decreased by any circumferential heat flow.

In view of the present demand for higher allowable tube stresses and greater heat inputs much more detailed work should be carried out on tube temperature condition, in particular on the shape of the tube isothermals. The existing apparatus, however, will have to be very considerably altered as the thin walled tube is quite unsuited for this type of investigation.

PART IV - APPENDIX

BIBLIOGRAPHY

1. KEENAN, J.H. and KAYE, J.: Thermodynamic Properties Air (1945)

2. HUGHES, J.A.; Philosophical Magazine (6) 31, (1916), p.118

3. HILPERT, R.; Forschung aug dem Gebiete (1933), p.215

4. SMALL, J.; Philosophical Magazine, Vol.19 (1935) p.251

5 GRIFFITH, E, and AWBERY, J.H.; Proceedings of Inst. of Mech. Eng. Vol. 125 (1933) p.319.

6. KING, L.V.; Transactions of Royal Society (London), A.214, (1914), p.373

7 REIHER, H.; Mitt. Forschungsarbeiten, (1925) p.269

8. FISHENDEN, M. and SAUNDERS, O.A.; The Calculation of Heat Transmission (1932)

9. SCHACK, A.; Industrial Heat Transfer (1933)

10. McADAMS, W.H.; Heat Transmission, 2nd Edition (1942)

11. SINELNIKOV, A.A. and CHASHCHIKHIN, A.S.; Zurnal Technicheskoi Fiziki, Vol.II No.9-10 (1932)

12. KRUJILIN, G. and SCHWAB, B.; Technical Physics, U.S.S.R. Vol. 2. (1935), p.312

13. FAGE, A. and FALKENER, V.M.; R. and M. No. 1408 April 1931

14. SMALL, J.; Engineering, Vol. 132 (1931) p.569

15. KIRPITCHEV, M.V.; Communication of Leningrade Physical-Technical Laboratories No. 2. (1926)

16. LOHRISCH, W.; Mitt. Forschungsarbeiten, (1929) p.322

17. WINDING, C.C.; and CHENEY, A.J.; Industrial and Engineering Chemistry (1948) p.1087

18. KLEIN, V.; Archiv fur Warmewirtschaft, (1934), p.150.

19. DREW, T.B. and RYAN, W.P.; American Institute of Chemical Engineers, (1931), p.118

20. KRUJILIN, G.; Technical Physics, U.S.S.R., Vol. 5., (1938) p.289.

21. SCHMIDT, E. and WENNER, K.; Forschung auf dem Gebiete, (1941) p.65.

22. JOUKOUSKY, V., KIREJEN, A., and SCHAMSCHEN, L.; Technical Physics U.S.S.R. Vol. 4, (1935) p.353

23. HUGE, E.C.; Transactions of A.S.M.E. Vol. 59 (1937) p.573

24. PIERSON, O.L.; Transactions of A.S.M.E. Vol. 59 (1937) p. 563.

25. GRIMISON, E.D.: Transactions of A.S.M.E. Vol. 59 (1937) p.583.

26. KUZNETZOFF, N.V. and LOKSHIN, V.A.; Teplo i Sila, No.10 (1937) p.19

27. LANDER, C.H.; Proceedings of Inst. of Mech. Eng. Vol. 148 (1942) p.81

28. FISHENDEN, M. and SAUNDERS, O.A.; Journal of Inst. of Fuels (1945) p. 62

29. WINDING C.C.; Industrial and Engineering Chemistry Vol.30. (1938) p. 942.

30. ORNATSKI, A.P.: Sovietskoye Kotlotorbostroyniye, No.2 (1940) p.48.

31. LOKSHIN, V.A.; Teplo i Sila, No. 8 (1940) p.20.

32. OWER, E.; The Measurement of Air Flow (1932)

33. American Institute of Physics, Temperature, its Measurement and Control in Science and Industry (1941)

34. TOFT, L. and McKAY, A.D.D.; Practical Mathematics (1942)

35. Modern Developments in Fluid Dynamics, Vol II. p.631

36. WALLIS AND WHITE, Engineering, Vol. 146 (1938) p.605.

NOTATION

c_p - Specific heat at constant pressure

d - diameter

E - emissivity

f - prefix designating function

G - mass flow per unit area of cross section

h - coefficient of heat transfer between fluid and surface

h_c - coefficient of convective heat transfer between fluid and surface

h_r - coefficient of radiant heat transfer between fluid and surface

H - heat transfer per unit area per unit time

k - thermal conductivity

L - length

Nu - Nusselt Number $= \frac{hd}{k} = \frac{Hd}{k\theta}$

Pr - Prandtl Number $= \frac{c_p \mu}{k}$

Q - Quantity of heat

r - radius

Re - Reynolds Number $= \frac{Gd}{\mu} = \frac{\rho d v}{\mu}$

s - time

t - temperature, bulk temperature of fluid

t_s - Tube surface temperature

t_f - film temperature $= \frac{1}{2}(t + t_{s_m})$

t_d - duct wall temperature

T - absolute temperature

V - velocity of fluid

α' - angle

θ - temperature rise

ρ - density of fluid

μ - absolute viscosity

Δt - temperature difference between fluid and surface

Suffixes other than given.

i - inside

m - true mean

o - outside

r - by radiation

Tests 1-26 - Tube O.Dia. = 0.747" Thickness = 0.059"

Tests 27-31 - Tube O.Dia. = 0.691" Thickness = 0.032"

Table Ia — Mean Heat Transfer to Single Tube

Test Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Pitot Static Head in H_2O	3.67	2.31	3.61	3.14	1.09	0.79	2.98	1.24	0.71	2.60	2.11	2.01	0.71	2.57	2.28	2.08	2.37	0.59	0.69	0.68	0.70	2.99	3.22	1.16	3.36	3.62	2.68	1.20	0.72	3.19	2.96
Air Temperature at Pitot Station °F	198	198	248	297	296	291	398	396	392	496	494	490	487	544	540	590	600	590	588	575	476	388	291	287	241	198	367	362	363	201	298
Air Temp. at Entry to Test Section °F	199	200	250	299	300	297	402	401	398	501	501	502	505	551	548	607	608	606	607	602	501	396	298	298	292	194	375	375	375	202	300
⌖ Average Tube Wall Temperature - t_{sm} °F	70	66	73	75	70	68	82	77	75	87	87	82	79	90	91	94	95	83	85	86	78	89	80	76	74	70	70	65	62	63	66
Average Duct Wall Temperature °F	193	190	235	286	283	272	375	372	371	480	472	470	465	502	500	554	555	547	546	545	485	375	285	280	234	190	345	341	340	192	287
Overall Water Temperature Rise - θ_m - °F	2.32	2.02	3.01	3.81	2.76	2.67	4.98	3.84	3.31	6.61	6.05	5.88	4.76	7.34	6.82	7.90	8.24	5.36	5.91	5.75	4.4	5.17	3.61	2.64	2.87	2.08	4.50	3.72	3.16	2.18	3.50
Total Water Flow lb/min.	4.45	4.5	4.59	4.5	4.47	4.5	4.72	4.7	4.7	4.5	4.58	4.75	4.49	4.4	4.5	4.5	4.5	4.5	4.5	4.46	4.53	4.47	4.52	4.51	4.52	4.5	4.52	4.5	4.5	4.52	4.6

Table Ib
Deduced Results

⌖ This quantity is based on detailed temperatures given in Table 2

Air Speed Past Tube ft/sec.	84	66	86.5	83	50	42	86	54.5	42	103	77	75.5	54	87	81.5	80.6	85.5	43	47	46	44.5	85	84.5	51	83	83	79	53.5	42	80	80
G - lb/hr ft² of minm flow area	18150	14500	17400	15800	9220	7900	14400	9200	7050	12600	11450	11200	6540	12200	11600	11000	11500	5800	6230	6250	6530	14300	16100	9660	17000	18200	13800	9250	7180	16900	15200
Mean Temperature Between Air and Tube Wall - t_{fm} - °F	134.5	132.5	161.5	187	185	183	242	239	237	294	294	292	292	320	319	350	352	345	346	344	288	243	189	187	158	132	222	220	219	132	183
Reynolds Number - Re	24500	19400	22600	18900	11800	10040	17200	11100	8470	14300	13000	12700	7560	13600	12900	11800	12500	6280	6760	6780	7450	17000	20200	12200	27100	24500	15000	10050	7800	22600	19300
Temperature Difference Air to Tube - Δt °F	129	134	177	224	230	229	320	324	323	414	414	420	426	461	457	513	513	523	522	516	423	307	218	222	168	124	306	310	313	138	234
Heat Transfer Coefficient - Bth U/hr ft² °F	29.4	24.9	28.7	28	19.7	18.6	27	20.4	17.6	26.3	24.5	24.4	18.4	25.4	25.7	25.4	25.6	17.0	18.6	18.0	17.3	27.6	27.4	19.6	28.3	27.6	26.6	21.6	18.1	28.6	28.8
Radiant Heat Transfer Coefficient - Bth U/hr ft² °F	0.15	0.15	0.15	0.17	0.17	0.16	0.21	0.21	0.21	0.29	0.28	0.27	0.26	0.28	0.29	0.33	0.32	0.31	0.31	0.31	0.29	0.17	0.18	0.17	0.16	0.15	0.20	0.20	0.20	0.15	0.17
Convective Heat Transfer Coefficient - Bth U/hr ft² °F	29.2	24.7	28.5	27.8	19.5	18.4	26.8	20.2	17.4	26.0	24.2	24.1	18.1	25.1	24.4	25.0	25.3	16.6	18.3	17.6	17.0	27.4	27.2	19.4	28.1	27.4	26.4	21.4	17.9	28.4	28.7
Mean Nusselt Number - Nu_m	110	93.7	104	98.4	69	65	88	66.4	57.5	80.2	74.7	74.6	56	75.1	72.8	71.6	73.2	49.1	53.2	51.3	53	89.5	96.2	70	103.6	106	82	66	56	105	94.

TABLE 2

Local variations around Single Tube of θ, the water temperature rise through the test slot, and t_s, the tube wall temperature.

Tube external diameter 0.747 ins.

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
1	θ °F.	2.62	2.57	2.49	2.39	2.24	2.17	2.14	2.19	2.24	2.29	2.31
	t_s °F.	72	71.6	71.2	70	69	68.6	68.6	69.2	69.6	70.4	71
2	θ °F.	2.43	2.38	2.26	2.14	2.06	1.99	2.00	2.04	2.05	2.07	2.11
	t_s °F.	68	67.4	66	65.4	64.6	64	63.6	64.2	65	65.6	66
3	θ °F.	3.42	3.31	3.20	3.03	2.90	2.85	2.82	2.86	2.93	2.99	3.00
	t_s °F.	75.8	75.4	74.2	73	71.8	71.4	71.2	71.2	72	72.8	73
4	θ °F.	4.25	4.20	4.04	3.80	3.62	3.54	3.49	3.56	3.66	3.70	3.71
	t_s °F.	77.6	77	76	75	74	73.5	73	73.5	74	74.5	75
5	θ °F.	3.32	3.25	3.04	2.85	2.66	2.64	2.58	2.56	2.64	2.74	2.76
	t_s °F.	73	72.4	71.2	70	68.6	67.4	66.8	65.6	65.4	65.6	66

TABLE 2 (Contd.)

Tube external diameter 0.747 ins.

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
6	θ °F.	3.01	2.93	2.77	2.62	2.49	2.46	2.41	2.40	2.44	2.47	2.48
	t_s °F.	72.6	72	70.6	69	67	66.4	66	65.2	65.4	66	66.4
7	θ °F.	5.80	5.67	5.44	5.13	4.84	4.73	4.71	4.71	4.84	4.96	5.00
	t_s °F.	87	86.4	84.6	82	80	79.4	78.6	78.4	79.2	80	80.4
8	θ °F.	4.70	4.62	4.32	4.03	3.73	3.71	3.66	3.63	3.72	3.82	3.86
	t_s °F.	80.2	79.4	78	76.6	75	73.8	73	71.6	71.2	71.6	72.2
9	θ °F.	3.93	3.81	3.62	3.39	3.12	3.06	3.04	2.93	2.98	3.04	3.07
	t_s °F.	77.8	77	75.4	73	70.6	69.6	68.8	67.8	67.4	67.4	67.6
10	θ °F.	7.63	7.54	7.32	7.00	6.54	6.30	6.12	6.18	6.35	6.62	6.69
	t_s °F.	94.6	93	91	88	84.6	83.6	83.4	84	85.6	86.6	87

TABLE 2 (Contd.)

Tube external diameter 0.747 ins.

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
11	θ °F.	7.45	7.30	6.92	6.41	5.96	5.79	5.56	5.57	5.91	6.12	6.22
	t_s °F.	93	92.4	91	89	86.2	85	84	83.6	84.8	85.2	87
12	θ °F.	7.02	6.81	6.42	5.85	5.50	5.44	5.32	5.33	5.47	5.52	5.63
	t_s °F.	89	88.2	87	85	82.4	81.4	80.2	79	78.6	79.4	80.4
16	θ °F.	9.06	8.73	8.14	7.62	7.10	6.95	6.83	6.77	6.94	7.21	7.30
	t_s °F.	101	100.6	99	96	93	91.6	91	89	88.5	90	91
17	θ °F.	9.45	9.15	8.65	8.10	7.52	7.46	7.33	7.34	7.54	7.76	7.90
	t_s °F.	103	102	99.2	95.4	91.4	90	89	88.6	89	89.4	89.6
18	θ °F.	6.70	6.53	6.06	5.66	5.43	5.32	5.21	5.16	5.17	5.24	5.28
	t_s °F.	90.6	90	88	86	83.4	81.6	80	78	76.8	76	75.6

TABLE 2 (Contd.)

Tube external diameter 0.747 ins.

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
19	θ °F.	6.90	6.57	6.27	5.82	5.61	5.49	5.44	5.32	5.34	5.36	5.41
	t_s °F.	92	91.6	90	88	84.4	83	81.6	79	78	77.4	77
20	θ °F.	6.81	6.49	6.31	5.86	5.65	5.55	5.45	5.33	5.35	5.37	5.42
	t_s °F.	93	92	91	88.6	85	83.4	82	79.4	79	78.6	78.4
21	θ °F.	5.35	5.22	4.95	4.56	4.32	4.20	4.12	4.08	4.09	4.13	4.16
	t_s °F.	85	84	82.8	81	78.2	76.2	75.5	74	73.2	72.6	72.4
24	θ °F.	3.40	3.33	3.16	2.92	2.76	2.70	2.61	2.61	2.70	2.76	2.81
	t_s °F.	74	73	72	71	69.8	69	68.2	67.2	67.2	67.6	68
26	θ °F.	2.64	2.57	2.51	2.39	2.25	2.18	2.15	2.19	2.22	2.27	2.29
	t_s °F.	72	71.6	71	70	69	68.6	68.6	69	69.4	70.2	71

TABLE 2 (Contd.)

Tube external diameter 0.691 ins.

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
27	θ °F.	5.68	5.45	5.12	4.80	4.48	4.38	4.38	4.43	4.60	4.74	4.78
	t_s °F.	75.6	75	73	70.2	67.2	66	64.6	66	67.6	69	70
28	θ °F.	4.59	4.43	4.15	3.80	3.48	3.42	3.36	3.30	3.41	3.64	3.69
	t_s °F.	70	69.6	68	66.2	64.6	63.6	63	61.4	62	63	63.6
29	θ °F.	4.15	3.96	3.68	3.25	3.06	2.88	2.76	2.55	2.76	2.96	3.06
	t_s °F.	67	66.4	65.4	63.2	62	61.2	60.2	59.2	59	60	60.4
30	θ °F.	2.75	2.68	2.48	2.34	2.22	2.14	2.12	2.21	2.29	2.36	2.49
	t_s °F.	65.4	65	64	63	61.6	61	61.4	62	63	63.6	64.4
31	θ °F.	4.45	4.29	4.08	3.72	3.46	3.42	3.40	3.49	3.65	3.72	3.76
	t_s °F.	69	68.4	67.4	66	64	63.6	63.4	64	64.6	66	66.6

TABLE 3

MEAN HEAT TRANSFER TO SINGLE ROW OF TUBES

TEST RESULTS	TEST NUMBER	1	2	3	4	5	6	7	8	9	10	11
PITOT STATIC HEAD	in. H_2O	2.18	0.63	2.07	0.64	1.33	2.82	1.74	2.55	3.23	3.15	1.13
AIR TEMPERATURE AT PITOT STATION	°F	391	567	567	474	485	288	390	288	200	240	390
AIR TEMPERATURE AT ENTRY TO TEST SECTION	°F	403	586	589	448	504	295	406	295	202	244	408
⊕ AVERAGE TUBE WALL TEMPERATURE t_{sm}	°F	86	83	96	80	86	80	84	80	73	75	80
OVERALL WATER TEMPERATURE RISE θ_m	°F	4.96	5.30	7.65	4.36	5.60	3.69	4.71	3.56	2.28	2.9	4.25
TOTAL WATER FLOW	lb./min.	4.51	4.5	4.51	4.5	4.49	4.52	4.58	4.5	4.5	4.5	4.52

⊕ THIS QUANTITY IS BASED ON DETAILED TEMPERATURES GIVEN IN TABLE 4

DEDUCED RESULTS

DEDUCED RESULTS	1	2	3	4	5	6	7	8	9	10	11
G - $lb/hr\ ft^2$ of min^m flow area	14400	7080	12860	7400	10600	17450	12850	16600	19900	19000	10400
MEAN TEMP. BETWEEN AIR AND TUBE WALL - t_{fm} °F	245	335	343	289	295	188	245	187	132	160	244
REYNOLDS NUMBER - Re	17200	7730	14000	8400	12000	21900	15300	21000	26700	24800	12300
TEMPERATURE DIFFERENCE AIR TO TUBE Δt - °F	317	503	493	418	418	215	322	215	129	165	328
HEAT TRANSFER COEFFICIENT h_m - $Btu/hr\,ft^2\,°F$	26.0	17.2	25.7	17.2	22.0	28.5	24.6	27.4	29.2	29.0	21.4
MEAN NUSSELT NUMBER Nu_m	85.2	51.3	74.6	53.6	67.8	100	80.4	96.2	111	106	70.3

TABLE 4

Local Variations around Tube in Single Row of Tubes of θ, the water temperature rise through test slot, and t_s, the tube wall temperature.

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	150°	180°
1	θ °F.	5.4	5.31	5.16	4.9	4.67	4.6	4.57	4.64	4.76	4.84	4.86
	t_s °F.	90	89.4	88.4	87.6	85.4	84.2	83	83	83.6	85	86
2	θ °F.	6.31	6.11	5.84	5.46	5.16	4.85	4.76	4.75	4.82	4.87	4.93
	t_s °F.	87	86	83	80.8	79	78	77	76.2	76.8	78	79.6
3	θ °F.	8.7	8.52	8.17	7.71	7.3	7.14	7.08	7.2	7.31	7.36	7.4
	t_s °F.	103	102	100	97.8	94.4	93.2	91	91	92	94.3	96
4	θ °F.	5.07	4.91	4.74	4.41	4.13	4.0	3.89	3.8	3.91	3.97	4.02
	t_s °F.	85.4	84	82.8	80.6	78	77	75.8	75.4	76.6	77	78
5	θ °F.	6.23	6.06	5.8	5.47	5.22	5.09	5.06	5.06	5.16	5.28	5.31
	t_s °F.	91	90	89	87	84.4	83.2	82	82.4	83.2	85	86

TABLE 4 (Contd.)

Test	Angle	0^o	20^o	40^o	60^o	80^o	90^o	100^o	120^o	140^o	160^o	180^o
6	θ ^oF.	3.93	3.84	3.71	3.54	3.39	3.34	3.35	3.42	3.51	3.58	3.61
	t_s ^oF.	83	82.4	80.6	79.4	79	78.2	77.4	77.8	78.6	79.4	80
7	θ ^oF.	5.15	5.02	4.88	4.56	4.34	4.23	4.19	4.23	4.34	4.42	4.44
	t_s ^oF.	89.6	89	87.4	85	82.6	81.4	80.6	80.8	81.8	83.2	84
8	θ ^oF.	3.87	3.8	3.7	3.59	3.41	3.34	3.29	3.32	3.34	3.39	3.41
	t_s ^oF.	82.6	82	81	80	78.6	78	77	77.4	78	78.8	79.4
9	θ ^oF.	2.46	2.42	2.31	2.24	2.14	2.13	2.13	2.17	2.20	2.24	2.26
	t_s ^oF.	74.4	74	73.4	72.6	71.8	71.4	71	71.4	72.6	73.2	74
10	θ ^oF.	3.08	3.04	2.93	2.8	2.69	2.65	2.65	2.72	2.8	2.86	2.87
	t_s ^oF.	78	77.6	76.6	75.4	74.4	74	73.6	74	74.6	75.4	76
11	θ ^oF.	4.60	4.52	4.36	4.13	3.95	3.89	3.82	3.9	3.94	4.00	4.02
	t_s ^oF.	83.6	83	82.2	81	79.6	79	78.2	78	78.6	79.2	79.8

Table 5 — Mean Heat Transfer to Two Rows of Tubes

Test Results	1st Row					2nd Row				
Test Number	1	2	3	4	5	6	7	8	9	10
Pitot Static Head in H_2O	2.01	1.35	0.62	2.99	2.66	2.07	1.33	0.63	2.06	2.65
Air Temperature at Pitot Station	385	477	457	197	287	383	471	457	196	286
Air Temperature at Entry to Test Section °F	403	504	497	202	294	403	304	497	200	294
⊕ Average Tube Wall Temperature t_{sm} °F	82	86	81	71	76	88	91	85	73	80
Overall Water Temp. Rise – θ_m °F	4.56	5.15	4.19	2.06	3.24	5.48	6.27	4.86	2.24	3.78
Total Water Flow lb/min.	4.49	4.5	4.5	4.5	4.51	4.5	4.47	4.5	4.55	4.5

⊕ This quantity is based on detailed temperatures given in Table 6

Deduced Results	1	2	3	4	5	6	7	8	9	10
G – lb/hr ft² of minm flow area	14400	10700	7400	19200	16900	14400	10700	7400	19100	16900
Mean Temp. between Air and Tube Wall – t_{fm} °F	242	295	289	137	185	245	298	286	136	187
Reynolds Number – Re	17200	12100	8400	25500	21400	17100	12100	8400	25500	21300
Temperature Difference Air to Tube – Δt °F	321	418	416	131	218	315	413	412	127	214
Heat Transfer Coefficient – h_m Btu/hr ft² °F	23.4	20.3	16.6	25.9	24.5	28.7	24.9	19.5	29.6	29.2
Mean Nusselt Number – Nu_m	76.8	62.5	51.6	98	86.7	94	76.6	60.5	112.5	103

TABLE 6

Local Variations around Tube in Two Row Bank of Tubes of water temperature rise through test slot θ, and tube wall temperature t_s.

1st Row

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
1	θ °F.	5.10	4.96	4.80	4.43	4.10	3.96	3.9	3.82	3.89	3.94	3.99
	t_s °F.	89	88.6	86	82.8	81	79.6	77	77	78	79.2	80.6
2	θ °F.	6.95	6.75	6.47	6.01	5.58	5.26	5.13	5.00	5.03	5.14	5.20
	t_s °F.	92.2	91.6	90	88	84.6	81.6	80.4	79	80	80.6	82.2
3	θ °F.	4.73	4.54	4.32	4.09	3.70	3.61	3.54	3.44	3.44	3.48	3.50
	t_s °F.	89	88.4	87	85.4	82	79.4	77.8	77	76	77	79
4	θ °F.	2.31	2.21	2.13	1.99	1.88	1.84	1.78	1.76	1.79	1.83	1.85
	t_s °F.	74.4	74	73.2	72	71	70.4	69.6	69	69	69.4	70
5	θ °F.	3.74	3.60	3.41	3.27	2.96	2.88	2.86	2.85	2.90	2.97	2.99
	t_s °F.	82	81.4	80	78	76.4	75	74	73	73.6	75	76

TABLE 6 (Contd.)

2nd Row.

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
6	θ °F.	6.15	5.87	5.70	5.46	5.13	4.90	4.76	4.79	4.96	5.09	5.14
	t_s °F.	93.4	92.8	89	86.6	85	84.4	84.6	85.6	87	89	90
7	θ °F.	7.18	6.90	6.54	6.07	5.72	5.52	5.47	5.42	5.65	5.78	5.83
	t_s °F.	98.4	98.4	95.2	92.6	89.4	86.6	87.6	88.6	90	92	93
8	θ °F.	5.40	5.21	4.95	4.53	4.23	4.13	4.06	4.08	4.19	4.26	4.28
	t_s °F.	92	91.6	88.6	84.4	82	81.6	81.6	82	83.4	85	86
9	θ °F.	2.69	2.51	2.48	2.33	2.21	2.19	2.18	2.24	2.34	2.40	2.42
	t_s °F.	76	75.6	74.8	73	71.6	71	70.6	71	71.6	72.6	73.4
10	θ °F.	4.46	4.29	4.14	3.96	3.75	3.65	3.61	3.59	3.60	3.71	3.76
	t_s °F.	86	85.4	83	80.2	78	77	77	78	80	86	81.4

Table 7 — Mean Heat Transfer to Three Rows of Tubes

Test Results	1st Row					2nd Row					3rd Row				
Test Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Pitot Static Head in H_2O	2.11	1.34	0.62	2.77	2.39	2.01	1.33	0.63	2.65	2.42	1.98	1.33	0.62	2.56	2.41
Air Temperature at Pitot Station °F	376	475	457	198	277	374	469	457	196	279	378	463	457	198	281
Air Temperature at Entry to Test Section °F	403	504	498	202	294	403	504	497	201	294	403	504	497	203	294
⊕ Average Tube Wall Temperature - t_{sm} °F	81	86	81	71	76	83	89	85	72	77	87	92	89	73	80
Overall Water Temperature Rise - θ_m °F	4.56	5.15	4.19	2.02	3.22	5.15	6.11	4.86	2.29	3.68	6.19	7.25	5.19	2.74	4.32
Total Water Flow lb/min.	4.5	4.5	4.5	4.51	4.5	4.51	4.5	4.5	4.5	4.51	4.49	4.5	4.5	4.5	4.52

⊕ This quantity is based on detailed temperatures given in Table 8

Deduced Results

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
G - lb/hr ft² of min^m flow area	14200	10700	7400	18400	16200	13860	10800	7400	18000	16200	13500	10800	7200	17800	16200
Mean Temp between Air and Tube Wall - t_{fm} - °F	242	295	289	136	185	243	297	286	136	186	245	296	293	137	187
Reynolds Number - Re	17000	12100	8400	24700	20400	16500	12200	8400	24100	20500	16300	12200	8200	23800	20400
Temperatur Difference Air to Tube - Δt - °F	322	418	417	131	218	320	414	412	129	217	316	412	408	130	214
Heat Transfer Coefficient - h_m - Btu/hr ft² °F	23.4	20.3	16.6	25.6	24.3	26.5	24.5	19.5	28.9	28.1	31.4	28.9	21.6	34.1	33.1
Mean Nusselt Number - Nu_m	76.8	62.5	51.6	96.7	86	87	74.7	60	110	99	102.3	88	67	129	117

TABLE 8

Local Variations around Tube in Three Row Bank of Tubes
of water temperature rise through test slot θ,
and tube wall temperature t_s.

1st Row

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
1	θ °F.	5.08	4.91	4.72	4.42	4.11	4.02	3.9	3.8	3.88	4.01	4.04
	t_s °F.	89	88.4	85.8	82	80	78.8	77	77	78	79.4	80.4
2	θ °F.	6.96	6.78	6.48	6.00	5.56	5.23	5.15	5.03	5.08	5.18	5.21
	t_s °F.	92	91.6	90	88	84.4	81	80.4	79	80	81	82
3	θ °F.	4.72	4.55	4.30	4.08	3.72	3.63	3.60	3.54	3.48	3.50	3.51
	t_s °F.	89	88.2	87	85	82.2	79.6	78	77.2	76.2	77	78
4	θ °F.	2.32	2.24	2.15	2.01	1.90	1.87	1.81	1.80	1.82	1.84	1.88
	t_s °F.	74.6	74	73	72	71.2	70.4	69.6	69	69	69.6	70.2
5	θ °F.	3.71	3.57	3.38	3.25	2.95	2.88	2.86	2.87	2.91	2.98	3.01
	t_s °F.	81	80.6	78.6	76.8	75	74	73	72.2	73	74	74.6

TABLE 8 (Contd.)

2nd Row

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
6	θ °F.	6.05	5.75	5.58	5.32	5.05	4.82	4.70	4.75	4.80	4.84	4.91
	t_s °F.	92.8	92	88	86	84.6	83.4	83.4	84.8	86	87	87.8
7	θ °F.	7.10	6.84	6.44	6.00	5.68	5.44	5.38	5.36	5.46	5.65	5.73
	t_s °F.	96.8	96	92.4	89	86	85	84.6	84.6	86	87.2	88
8	θ °F.	5.41	5.23	4.98	4.55	4.25	4.10	4.06	4.09	4.16	4.20	4.22
	t_s °F.	91	90.6	87.4	83	81	80.6	80.6	81	82.6	84	84.8
9	θ °F.	2.60	2.41	2.38	2.23	2.12	2.10	2.07	2.14	2.25	2.35	2.39
	t_s °F.	75.8	75.4	74.6	73.2	71.8	71.2	70.8	71.2	71.6	72.4	73
10	θ °F.	4.47	4.31	4.17	4.00	3.79	3.68	3.63	3.61	3.63	3.73	3.78
	t_s °F.	84	83.2	82	80	77.6	76.6	76.4	77	78	78.6	79.4

TABLE 8 (Contd.)

3rd Row

Test	Angle	0^o	20^o	40^o	60^o	80^o	90^o	100^o	120^o	140^o	160^o	180^o
11	θ ^oF.	7.20	7.11	6.60	6.26	6.07	6.00	5.93	5.84	5.85	5.89	5.92
	t_s ^oF.	94.6	94	92	89.6	87	86.4	86	85	86	87	87.6
12	θ ^oF.	8.12	7.82	7.38	6.85	6.51	6.30	6.12	6.09	6.13	6.15	6.20
	t_s ^oF.	98	97	95.2	93	90	89	87.4	86.2	87	87.6	88
13	θ ^oF.	6.15	5.92	5.54	5.07	4.75	4.61	4.49	4.40	4.43	4.46	4.50
	t_s ^oF.	94	93	91.2	88.2	86	85	84.8	85	85.4	86.6	87
14	θ ^oF.	3.16	3.05	2.93	2.75	2.67	2.65	2.63	2.60	2.60	2.63	2.64
	t_s ^oF.	77	76.6	75.4	74.2	73	72.2	71.8	71.2	71.2	72	73
15	θ ^oF.	5.24	5.07	4.92	4.60	4.47	4.39	4.35	4.31	4.33	4.35	4.37
	t_s ^oF.	86	85.6	84	82	80.2	79.8	79.6	79.4	79.6	80	80.4

TABLE 9 — MEAN HEAT TRANSFER TO SIX ROW BANK OF TUBES

TEST RESULTS	1ST ROW					2ND ROW					3RD ROW					4TH ROW					5TH ROW					6TH ROW				
TEST NUMBER	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
PITOT STATIC HEAD in H_2O	0·61	0·81	0·48	2·13	1·85	0·62	0·77	0·47	2·10	1·85	0·62	0·83	0·47	2·12	1·86	0·62	0·76	0·47	2·09	1·85	0·61	0·75	0·49	2·14	1·84	0·62	0·76	0·48	2·11	1·86
AIR TEMPERATURE AT PITOT STATION °F	294	256	330	187	266	293	257	329	184	264	293	256	328	187	263	294	257	331	186	264	294	258	331	184	264	295	257	331	184	264
AIR TEMPERATURE AT ENTRY TO TEST SECTION °F	350	294	403	200	294	350	294	403	200	214	350	294	403	200	294	350	294	403	300	294	350	294	403	200	294	350	294	403	200	294
AIR TEMPERATURE AT EXIT OF TEST SECTION °F	314	261	249	186	264	312	60	347	186	265	311	261	348	186	265	310	260	349	187	265	312	261	348	185	265	311	260	349	186	265
⊕ AVERAGE TUBE WALL TEMPERATURE t_{sm} °F	61	60	62	59	63	63	61	65	60	66	67	63	67	61	70	65	62	66	60	67	64	61	64	39	66	63	60	64	58	65
OVERALL WATER TEMPERATURE RISE θ_m °F	2·88	2·50	3·16	2·07	3·03	3·25	2·87	3·53	2·36	3·55	3·76	3·32	3·96	2·83	4·2	3·60	3·17	3·81	2·52	4·06	3·37	3·01	3·7	2·49	3·85	3·29	2·89	3·67	2·38	3·69
TOTAL WATER FLOW lb./min	4·5	4·5	4·5	4·5	4·52	4·5	4·5	4·51	4·5	4·5	4·5	4·5	4·5	4·5	4·5	4·5	4·5	4·5	4·5	4·5	4·51	4·5	4·5	4·5	4·5	4·5	4·5	4·51	4·51	4·5

⊕ THIS QUANTITY IS BASED ON DETAILED TEMPERATURES GIVEN IN TABLE 10

DEDUCED RESULTS

AIR SPEED PAST TUBE ft/sec.	8200	9670	7000	6400	14300	8160	9400	6930	16300	14400	8160	9670	6930	16300	14400	8760	9400	6920	16000	14400	8200	9300	7800	10,500	14300	8160	9400	7000	16200	14400
G - lb/hr ft² of min.^m flow area	205	177	233	129	179	206	178	234	131	180	208	179	235	130	182	207	178	235	130	180	207	177	234	130	180	206	177	233	129	179
REYNOLDS NUMBER - Re	10400	12200	8400	22100	18200	10000	12000	8330	22000	18300	10,000	12300	8330	22000	18300	10,000	12,000	8330	21700	18300	10400	11700	8650	22500	18200	10,000	12000	8400	22000	18300
TEMPERATURE DIFFERENCE AIR TO TUBE Δt °F	289	234	341	141	231	287	233	338	140	228	283	231	336	139	224	285	235	337	140	227	286	233	339	141	228	287	234	339	142	229
HEAT TRANSFER COEFFICIENT h_m Btu/hr ft² °F	16·8	18·3	15·4	24·1	21·7	18·7	20·4	17·4	27·6	25·7	21·6	23·7	19·4	33·6	30·9	20·8	27·5	18·7	29·7	29·7	19·6	21·3	18·2	28·6	27·8	19·0	20·5	17·6	27·8	26·7
MEAN NUSSELT NUMBER Nu_m	57·5	65	51·5	92	77	64	72	68	105	91·5	74	85	64·5	125	110	71	80	62	114	105	67	77	60·1	109	99	65	73	58·3	106	95

TABLE 10

Local variations around Tube in Six Row Bank of Tubes of water temperature rise through test slot θ, and tube wall temperature t_s.

1st Row

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
1	θ °F.	3.41	3.27	3.11	2.86	2.72	2.64	2.55	2.48	2.48	2.54	2.56
	t_s °F.	65	64.6	63.2	61.4	59.6	58.6	58	57	57.8	58.6	59
2	θ °F.	2.86	2.75	2.61	2.40	2.28	2.21	2.14	2.08	2.09	2.13	2.15
	t_s °F.	63.6	63	61.6	60	58	57.2	56.4	55.6	56.6	57.4	57.8
3	θ °F.	3.78	3.62	3.37	3.18	2.97	2.88	2.81	2.68	2.68	2.72	2.77
	t_s °F.	67	66	64	62	60.4	59.6	59	58	57.8	58.4	59
4	θ °F.	2.26	2.19	2.09	1.95	1.85	1.81	1.75	1.75	1.78	1.80	1.82
	t_s °F.	61.6	61	60	59	58.2	57.6	56.8	56.4	56.4	57	57.4
5	θ °F.	3.64	3.54	3.27	3.08	2.86	2.77	2.71	2.56	2.59	2.66	2.69
	t_s °F.	68	67.4	66	64.4	62.4	61.4	60.6	60.6	61.4	62.4	63

TABLE 10 (Contd.)

2nd Row

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
6	θ °F.	3.95	3.79	3.59	3.27	3.04	2.99	2.98	3.03	3.10	3.16	3.19
	t_s °F.	69	68.2	66.6	64.4	62.4	61.8	62	63.2	64.6	65.6	66
7	θ °F.	3.31	3.19	3.04	2.77	2.59	2.55	2.54	2.58	2.64	2.69	2.71
	t_s °F.	67	66	64.2	61.6	59.6	58.6	58	58.8	60	60.6	60.8
8	θ °F.	4.26	4.11	3.86	3.59	3.28	3.19	3.13	3.18	3.28	3.34	3.37
	t_s °F.	70	69	67.6	64.4	61.8	60.8	60.6	61.4	62.4	63	63.4
9	θ °F.	2.55	2.36	2.33	2.19	2.08	2.06	2.03	2.09	2.20	2.31	2.36
	t_s °F.	62.8	62.4	61.6	60.2	58.8	58.2	57.8	58.2	58.6	59.4	60
10	θ °F.	4.26	4.14	3.91	3.54	3.28	3.21	3.20	3.26	3.36	3.50	3.54
	t_s °F.	71	70	68.4	66	63.6	63	63.2	64	65.2	66	66.8

TABLE 10 (Contd.)

3rd Row

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
11	θ °F.	4.45	4.17	4.00	3.72	3.54	3.46	3.38	3.31	3.31	3.32.	3.33
	t_s °F.	72	71.2	70.2	68	65	64.2	64	64	65	65.8	66
12	θ °F.	3.83	3.59	3.46	3.23	3.08	3.02	2.96	2.89	2.88	2.90	2.92
	t_s °F.	69	68.4	67	65	62.6	62	61.6	61.2	62	62.6	62.8
13	θ °F.	5.01	4.78	4.45	4.11	3.90	3.80	3.71	3.58	3.56	3.58	3.58
	t_s °F.	73	72	70	68	65	64	63.8	63.8	64.6	65.6	66
14	θ °F.	3.20	3.03	2.83	2.66	2.62	2.6	2.59	2.55	2.51	2.52	2.54
	t_s °F.	64	63.6	62	61	60.4	60	59.6	59	59	59.6	60
15	θ °F.	4.73	4.56	4.34	4.03	3.90	3.85	3.80	3.78	3.76	3.76	3.76
	t_s °F.	73.4	73	72	71	69.4	68.8	68	67	66.4	67	67 .6

TABLE 10 (Contd.)

4th ROW

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
16	θ °F.	4.30	4.12	3.88	3.58	3.38	3.26	3.18	3.13	3.11	3.10	3.10
	t_s °F.	70.4	70	68.2	65.8	64.2	63	62.8	62.2	62.2	62	62
17	θ °F.	3.69	3.58	3.40	3.17	3.00	2.92	2.85	2.77	2.71	2.68	2.67
	t_s °F.	68	67.4	65.4	63.4	61.6	61	60.6	60	59.8	59.8	59.6
18	θ °F.	4.73	4.58	4.32	4.05	3.77	3.62	3.52	3.37	3.28	3.26	3.26
	t_s °F.	71.4	71	69.4	67	64.2	63.4	63	62.2	62.2	62	62
19	θ °F.	3.05	2.99	2.91	2.76	2.60	2.57	2.54	2.49	2.45	2.44	2.43
	t_s °F.	63.4	63	61.4	60.4	59.8	59.4	59	58.6	58.4	58.2	58
20	θ °F.	4.60	4.48	4.32	4.07	3.82	3.77	3.70	3.62	3.56	3.53	3.52
	t_s °F.	72	71	69.4	67.4	66	65.4	65	64.4	64	64	64

TABKE 10 (Contd.)

5th Row

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
21	θ °F.	4.11	3.94	3.71	3.42	3.20	3.07	2.98	2.97	2.97	2.95	2.95
	t_s °F.	69	68.6	67.2	64.8	63.2	62	61.6	61.0	61.0	60.8	60.8
22	θ °F.	3.47	3.40	3.28	3.08	2.88	2.79	2.73	2.61	2.57	2.54	2.52
	t_s °F.	66.2	65.6	64.6	63	61.4	60.6	60	59.4	59.4	59.2	59.2
23	θ °F.	4.42	4.32	4.17	3.96	3.71	3.56	3.40	3.28	3.20	3.16	3.16
	t_s °F.	69.4	68.6	67	65	63.2	62.6	62.2	61.8	61.6	61.4	51.4
24	θ °F.	2.94	2.88	2.80	2.65	2.59	2.43	2.38	2.34	2.33	2.33	
	t_s °F.	62.6	62.2	60.6	59.8	59	58.6	58.2	57.4	57.2	57	57
25	θ °F.	4.38	4.28	4.17	3.94	3.67	3.60	3.53	3.42	3.41	3.40	3.40
	t_s °F.	70	69.4	68	66.4	65	64.6	64	63.6	63.2	63	63

TABLE 10 (Contd.)

6th Row

Test	Angle	0°	20°	40°	60°	80°	90°	100°	120°	140°	160°	180°
26	θ °F.	3.95	3.80	3.55	3.18	3.05	2.91	2.78	2.77	2.76	2.75	2.75
	t_s °F.	67.4	67.0	66.0	64	63	61.6	61.0	60.4	60.4	60.6	60.8
27	θ °F.	3.35	3.30	3.19	3.00	2.81	2.73	2.65	2.56	2.50	2.48	2.46
	t_s °F.	65.2	64.6	63.8	62.4	60.8	60.2	59.6	59.2	58.8	58.8	58.6
28	θ °F.	4.30	4.24	4.08	3.90	3.62	3.43	3.31	3.15	3.09	3.07	3.09
	t_s °F.	68.6	68	67	64.6	63	62.2	61.8	61.2	61.2	61	61
29	θ °F.	2.71	2.67	2.58	2.43	2.27	2.21	2.14	2.07	2.02	2.00	2.00
	t_s °F.	61	60.4	59.4	58.4	57.6	57.4	57.2	57	56.8	56.8	56.6
30	θ °F.	4.31	4.25	4.12	3.88	3.59	3.49	3.37	3.29	3.25	3.25	3.27
	t_s °F.	69	68.4	67.2	65.6	64.4	63.8	63.4	62.8	62.6	62.4	62.4