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THERMAL RESISTANCES OF FOULING DEPOSITS ON
MARINE CONDENSER TUBES AND THEIR EFFECTS
ON STEAM POWER PLANT PERFORMANCE

BY

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ABSTRACT

The thermal resistances of the fouling deposits on 72 used condenser tubes have been measured in a low pressure steam condenser under conditions comparable with those in the condensers from which the tubes were obtained.

Thirty of the tubes were removed in batches of six at intervals over a period of seven years from the condensers of the S.S. Pyrrhus, a cargo boat, and most of the remainder in batches of three tubes each from a wide variety of vessels ranging from the S.S. Queen Mary to a cross-channel car ferry, the S.S. Lord Warden.

The experimental method used was the so called Wilson Line Plot which was apparently used for the first time in a manner consistent with the assumptions on which it is based.

The average increase in resistances in the tubes from the S.S. Pyrrhus was such as to decrease the heat transfer coefficients at normal operating conditions by 16 per cent during the first ten months of actual use and by 35 per cent after seventy months. The majority of the results from the other tubes tested fell between the results from the S.S. Pyrrhus and a rate which gave a reduction of 33 per cent after thirty months use.

Measurements of the distribution of the deposits on

a number of tubes showed that they were sensibly uniform.

The effect of cleaning tubes by brushing and with acid was investigated and it was found that brushing reduced the resistance of the internal deposits by between 40 and 80 per cent and that acid cleaning could remove the deposits completely.

Finally, an estimate was made of the effects of the fouling deposits on the specific steam consumption of a typical marine power plant and it was found that for an initial condenser pressure of $28\frac{1}{2}$ in Hg vac it would increase by 5 per cent during between forty and one hundred months use, and, for an initial pressure of 27 in Hg vac, during between twenty and sixty months.

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CHAPTER 1

INTRODUCTION

The growing recognition of the importance of the subject of Heat Transfer has, during the last thirty years, resulted in the accumulation of a large body of experimental data reinforced and correlated by theoretical analysis. This data covers a wide range of heat exchanger configurations and of fluids and is continuing to accumulate at such a rate that each new attempt to produce an up-to-date summary is out of date almost before publication.

It would appear that, with the aid of all this information, it should now be possible to calculate with considerable precision, the dimensions of a heat exchanger, using fluids of known properties, to perform a given duty.

There is, however, one aspect of this design problem which appears to have received little attention. The majority of the fluids used in commercial heat exchangers contain substances in solution or suspension which can become attached to the heat transfer surfaces. This effect is usually referred to as fouling or scaling. The layer of deposited material on the heat transfer surfaces introduces an unknown quantity into the calculation of heat exchanger dimensions which largely nullifies the precision which the available data appears to make possible.

The effect of these deposits on the performance of heat exchangers is well known and most reports on experimental

work give details of the measures taken to ensure that the heat transfer surfaces of the apparatus used were kept clean. In some cases the change in the performance of the apparatus due to such deposits have been reported and shown to be substantial.

Text books on Heat Transfer which do not ignore this problem suggest that a fixed allowance or fouling factor should be used in heat transfer calculations. This takes the form of a thermal resistance to be added to those calculated from data obtained with clean apparatus, its magnitude depending usually on the fluids concerned. The impression given is that this method will allow for the effects of the deposits on the heat transfer surfaces once and for all.

However, it is probable that in most cases the thickness of the deposits and therefore their thermal resistance will continue to grow for as long as the heat exchanger is in use. The use of a fixed allowance for fouling in such cases means that initially the exchanger will give a better performance than is required. The performance will then deteriorate until the thermal resistance of the deposits is equal to the allowance, the exchanger will then be giving the required performance. Unless the deposits are removed the performance of the heat exchanger will thereafter fall short of that required.

Heat exchangers are usually used in conjunction with other apparatus which has to be shut down at intervals for overhaul or maintenance. It would be logical to use a fouling factor which ensured that the heat exchanger gave the required

performance for the same intervals of time.

The fouling factor used in the design of a heat exchanger should therefore depend on two considerations. Firstly, the rate at which the thermal resistance due to the deposits increases and secondly, the period for which the exchanger will be required to operate between shut downs for cleaning.

The second piece of information will usually be relatively easy to obtain. The first will only be available if the deterioration in performance with time of a similar heat exchanger has been measured. If such measurements have been made they have not been widely reported.

This is understandable. Such tests are most likely to have been made by the manufacturers of heat exchangers to obtain data for future designs. As the allowance made for fouling influences the size and thus the cost of a heat exchanger the manufacturer is unlikely to let his competitors have the benefit of his own experience. So, although it is highly probable that there is a considerable amount of data in existence, it is unlikely that it will ever be made generally available.

That this is an undesirable state of affairs was recognised by the American Society of Mechanical Engineers who, in 1949, arranged a symposium on the "Cleaning and Fouling of Heat Exchangers"²⁷ under the chairmanship of Professor O.P. Bergelin of the University of Delaware.

After the discussion that followed the reading of the various papers presented, Professor Bergelin summed up as follows: "Information on fouling is not easy to obtain nor is it ordinarily of a nature to permit generalization or justify publication. In the past (i.e. prior to the symposium) it has been generally accepted that surface fouling takes place under such widely different conditions and involves so many variables that each case should be treated specifically". He also pointed out that industrial concerns must possess considerable data but that this was difficult to obtain, also that the data that was available was too fragmentary to merit publication.

This thesis gives an account of an attempt to treat one case specifically and so far as the author is aware is the first that has been published since the symposium.

The case investigated was that of the marine steam condenser and the work was carried out on behalf of and with the assistance of the British Shipbuilding Research Association.

The modern marine steam condenser has been developed from experience gained during over one hundred years of operation, with the result that there is now little to choose between the designs of different manufacturers. This, together with the fact that there can be little variation in the properties of the two fluids concerned, i.e. low pressure steam and sea water, reduces the number of variables involved. The case investigated was therefore a relatively simple one.

It must also be admitted that those concerned with the day to day working of marine condensers do not regard fouling as a problem. This investigation in fact showed that, although cleaning schedules are laid down for marine condensers, they appear to operate satisfactorily for long periods without cleaning. This could mean that the design fouling allowances used are too large and the condenser heat transfer surface areas larger than necessary.

CHAPTER 2

PREVIOUS WORK

Information on the fouling rates to be expected in marine condensers is even more sparse than that for other common types of heat exchanger. This is in part due to "condenseritis", i. e. condenser tube failure due to corrosion, having been the main problem in the operation of marine condensers until the advent of aluminium-brass and cupro-nickel tubes. Until condenser tubes made from these alloys were used condenser tube life was short, sometimes less than twelve months and the fouling deposits accumulated in this time were insufficient to cause any appreciable deterioration in condenser performance. When this corrosion problem was overcome it was the fouling problems in other types of marine heat exchangers, i. e. fuel oil heaters, sea water evaporators etc., that engaged the attention of marine engineers.

The subject does occasionally receive passing mention. Sim,²² who is still regarded as a leading authority on marine condenser design, gives, in his textbook, the results of tests made to compare heat transfer coefficients obtained with a condenser tube taken from a passenger liner, after three voyages to Calcutta and back, with those with a new tube. These results showed that, with a cooling water velocity of 6.5 ft/s, the overall heat transfer coefficients

were 300 and 850 Btu/ft²h deg F respectively. This is a greater increase in thermal resistance than any found during this investigation. In the absence of detailed information as to how this data was obtained it must be treated with suspicion. A possible explanation of Sim's result is that at the time he made his measurements it was the practice to warm up the ship's turbines with a condenser pressure of about 5 in Hg vac. At the higher condenser temperature resulting, i.e. greater than 110°F, calcium carbonate could have been precipitated from the cooling water to form a hard adherent deposit. The fact that this practice is no longer used may explain the differences between Sim's results and those obtained during this investigation from condenser tubes from modern vessels.

Some of the papers given at the symposium referred to in the previous chapter mentioned the fouling of marine condensers but only fouling due to unusual circumstances, such as the incubation of marine animals in the warmed cooling water and fouling due to fuel oil from sunken ships entering the condensers with the cooling water. Fouling due to the steady growth of deposits during normal operation were not mentioned.

The composition of the deposits, however, were reported. Those on the outer surface of the tubes, i.e. the steam side, being made up of the chemicals used to condition the boiler feed water together with substances derived from

decomposed lubricating oil. On the inner tube surface, i. e. the cooling water side, the deposits were a mixture of corrosion products, remains of marine animals and plants, and of the salts and other substances dissolved in and in suspension in sea water. The rate at which these deposits accumulate was said to depend on many variables including tube diameter, material and surface condition and water velocity and temperature but no information was given to support these statements or to indicate the relative importance of the different variables.

It was also stated that the fouling allowance usually made when designing marine condensers resulted in the heat transfer area being about 18 per cent greater than the use of clean tube data would predict and that this allowance appeared to be generous.

Some idea of the magnitude of the fouling allowances which are used in condenser design can be obtained from the Standards of the (American) Tubular Exchanger Manufacturers Association (T.E.M.A.)²³ These standards list the fouling factors that members of the association should use when designing heat exchangers using the more common liquids and gases. For each fluid, factors are given for two ranges of temperature and of velocity. It is claimed that these factors, stated to have been obtained from "field trials", lead to the design of heat exchangers which will "maintain satisfactory

performance in normal operation with reasonable service time between cleaning." In view of these qualifications it is probable that industrial designers use data obtained from their own experience.

An analytical approach to the problem of fouling has been made by McCabe and Robinson¹⁵ when investigating fouling in petroleum refinery evaporators, whose useful period of service between cleaning is measured in days or hours. Assuming that the thickness and therefore the thermal resistance of the deposits at any time to be proportional to the heat transferred up to that time, and the overall temperature difference to be constant they showed that:-

$$R^2 = B.t + R_c^2$$

where R is the overall thermal resistance after time t, R_c the overall thermal resistance of the exchanger when clean and B a constant depending on the properties of the substance deposited.

The assumptions made by McCabe and Robinson are not applicable to the marine condenser though the same general approach appears to be valid. In a marine condenser the cooling water flow rate is approximately constant while the overall temperature difference and to a lesser extent, the heat transfer rate increase, as the thickness of the fouling deposits increase. If it can be assumed that the thickness and the thermal resistance of the deposits are proportional

to the amount of cooling water that has passed through the condenser, then the variation of the overall thermal resistance with time should be linear, i.e. $R = B't + R_c$ where B' is a constant, depending on the mass of material deposited per unit mass of cooling water. If this varies with cooling water temperature then the seasonal and geographical variations in sea water temperature makes simple analysis impossible.

Considerable though fragmentary data exists on the fouling of land based steam condensers both in this country and in the United States. The pollution of inland waters causes very rapid fouling on the water side of such condensers, principally due to the growth of algae and slimes but also to corrosion products and water borne debris. Methods developed to remove these deposits without stopping the plant range from periodic chlorination of the cooling water to kill living deposits¹¹ and zero buoyancy spheres which, carried through the tubes by the cooling water, remove soft deposits, to the provision of cooling water circulation systems which permit the direction of cooling water flow to be reversed.⁵ The fouling problems of land based condensers will only apply in the marine field when ships are required to use their main engines for considerable periods in inland waters. As it is usual to use tugs in these circumstances and to fill the ships' condensers with clean water while in port, the experience gained from land service condensers is not applicable to marine use.

Of the many reports from experimental workers on the fouling of experimental apparatus that of Eagle and Ferguson⁸ is typical. Working with an electrically heated condenser tube and fresh water at temperatures and velocities similar to those in marine condensers, they reported a decrease in heat transfer coefficient of 15.5 per cent after 50 hours at a water velocity of 7 ft/s. This again is a greater rate of increase in thermal resistance than has been measured in this investigation and could have been due to the experimental method used. The condenser tube was used as a resistance heater, so the potential difference between the tube surface and the water may have caused electrostatic precipitation. The deposit in this case appeared to consist of iron oxide.

Thus, apart from the T.E.M.A. fouling factors, this investigation was made in the absence of any general background of information and previous investigations.

CHAPTER 3

EXPERIMENTAL METHOD

The main object of the investigation was to determine the relationship between the thermal resistance of the deposits on marine condenser tubes and the time for which the tubes had been in service.

It was decided to obtain the required data by measuring the resistance to heat transfer through used condenser tubes, whose history was known, under conditions similar to those existing in modern marine steam condensers. The resistance of the deposits could then be obtained by comparison with the results of similar measurements made using a new tube of the same dimensions and material.

The selection of sample tubes for testing was based on the following considerations; to obtain a picture of the pattern of the deposit resistances within a condenser, about 50 tubes would be taken from a condenser with about three years service; to cover as wide a range of operating conditions and time as possible, three tube samples would be taken from as many vessels as practicable; to examine the effect of the progressive accumulation of deposits, six tube samples would be taken, at approximately yearly intervals, from one particular vessel. Owing to the time taken to develop the experimental apparatus and a reliable experimental

procedure the first item in the programme had to be abandoned.

Sample tubes were provided by ship-owning members of the British Shipbuilding Association. Details of the ships from which they were taken are given in Table 1, and a copy of the questionnaire, sent to each shipowner supplying tubes, in Appendix 3. The ship selected to provide the series of samples was the S.S. Pyrrhus. The factors considered when selecting sample tubes are discussed in Chapter 10.

FOR TABLE 1 SEE OVER.....

TABLE 1

Details of Ships Supplying Sample Condenser Tubes and their Condensers
 (All tubes tested were $\frac{3}{4}$ -in. o.d. 18 gauge)

Name and code	Normal route	Vacuum Hg. in.	Water velocity ft/s	Tube material	Cleaning Schedule
Pyrrhus (Py)	U.K.-Far East	$28\frac{1}{2}$	6.5	Aluminium brass	Brushing every four months
Dorset (Do)		$28\frac{1}{2}$	6.75		
Papanui (Pa)		$28\frac{1}{2}$ - $27\frac{3}{4}$	6.5		
Isle of Guernsey (I.G.)	Southampton- Guernsey	29-29	6.5	Bemal (70/30), brass (+Pb+Fe)	Wire brush yearly
Lord Walden (L.W.)	Cross-channel	$28\frac{1}{2}$	6.0	Aluminium brass	
Queen Mary (Q.M.)	U.K.-U.S.A.	29	8.3 (max)	Cupro-nickel (Cu 66.82 percent, Fe 0.88 percent, Mn 0.92 percent, Ni remainder)	Not cleaned since installation

TABLE 1 (cont.)

Name and code	Normal route	Vacuum Hg. in	Water velocity ft/s	Tube material	Cleaning Schedule
Velutina (Ve)	U.K. -Middle East (oil-tanker)	28½	6.75	Aluminium brass	Wire brush yearly
Clan Maclachlan (C. M.)	U.K. -India- S.Africa	28-29	5.74		Wire brush twice yearly
Hadra (Ha)	World-wide (oil-tanker)	28½	6.67		Rods and brushes once since installation
Hindsia (Hi)	World-wide (oil-tanker)	28½	6.67		
Clan MacLennan (C. M.L)	U.K. -India- S.Africa	28-29	5.74		Wire brush once or twice yearly
Stratheden (St)	U.K. -Australia	28-29	6.0	Cupro-nickel (Cu 70 percent, Ni 30 percent)	Wire brush three times per year

To measure the thermal resistances of the sample tubes it was decided to use the experimental method usually known as the "Wilson Line Plot".

Although this method has been used by many investigators during the last forty years, chiefly to measure heat transfer coefficients for condensing vapours, it does not appear from the literature that the assumptions involved have ever been examined.

The justification of this method and, in the interests of historical justice, an account of its origin are given in Appendix 1.

Briefly the method depends on the fact that the resistance to heat transfer from a vapour condensing on the outer surface of a tube to a liquid flowing through the tube is the sum of several separate resistances in series.

In the case of steam condensing on a clean tube through which water is flowing there are three resistances the sum of which is equal to the overall thermal resistance, which is the reciprocal of the overall heat transfer coefficient, i.e.

$$R = \frac{1}{h_{sw}} = R_s + R_t + R_w$$

If the steam contains air, or any other non-condensable gases, then there is a fourth resistance, R_a ,

$$\text{and } R = R_s + R_a + R_t + R_w$$

If the thermal resistance is measured for a constant

value of R_s and R_a , and for a range of values of R_w then, provided that R_w can be expressed as a linear function of the experimental variables the relationship between R and this function will be of the form $y = mx + c$ where c is the intercept of the straight line drawn through the experimental points with the overall resistance axis and m the slope of the line. The intercept will be equal to $R_s + R_a + R_t$ and as R_t can be calculated from the dimensions of the tube and the thermal conductivity of the tube material, $R_s + R_a$ can be obtained by difference.

If a similar test, at the same value of $R_s + R_a$, is made with a fouled tube of the same dimensions and material then the intercept on the resistance axis will be greater than that for a clean tube by an amount equal to the thermal resistance of the deposits, which can then be obtained by difference. Provided that the thickness of the deposits is small, relative to the tube diameters, then the lines obtained by plotting the experimental results from different tubes should be parallel. (see Appendix 1).

This method may seem unduly elaborate in view of the information required. However, the published results of even the most carefully conducted heat transfer experiments of this type show considerable scatter of the experimental points, particularly when one of the fluids concerned is condensing. It appeared unlikely therefore that any single point experiments,

say at a fixed value of R_s and at a common water velocity, could be made with sufficient accuracy for the differences between the results obtained to be considered reliable.

In the event the scatter of the plotted results was small and usually there was little difficulty in drawing the best straight line through them. When the scatter was such as to make it difficult to draw the best straight line by eye the slope and intercept were calculated using the "least mean square" method.

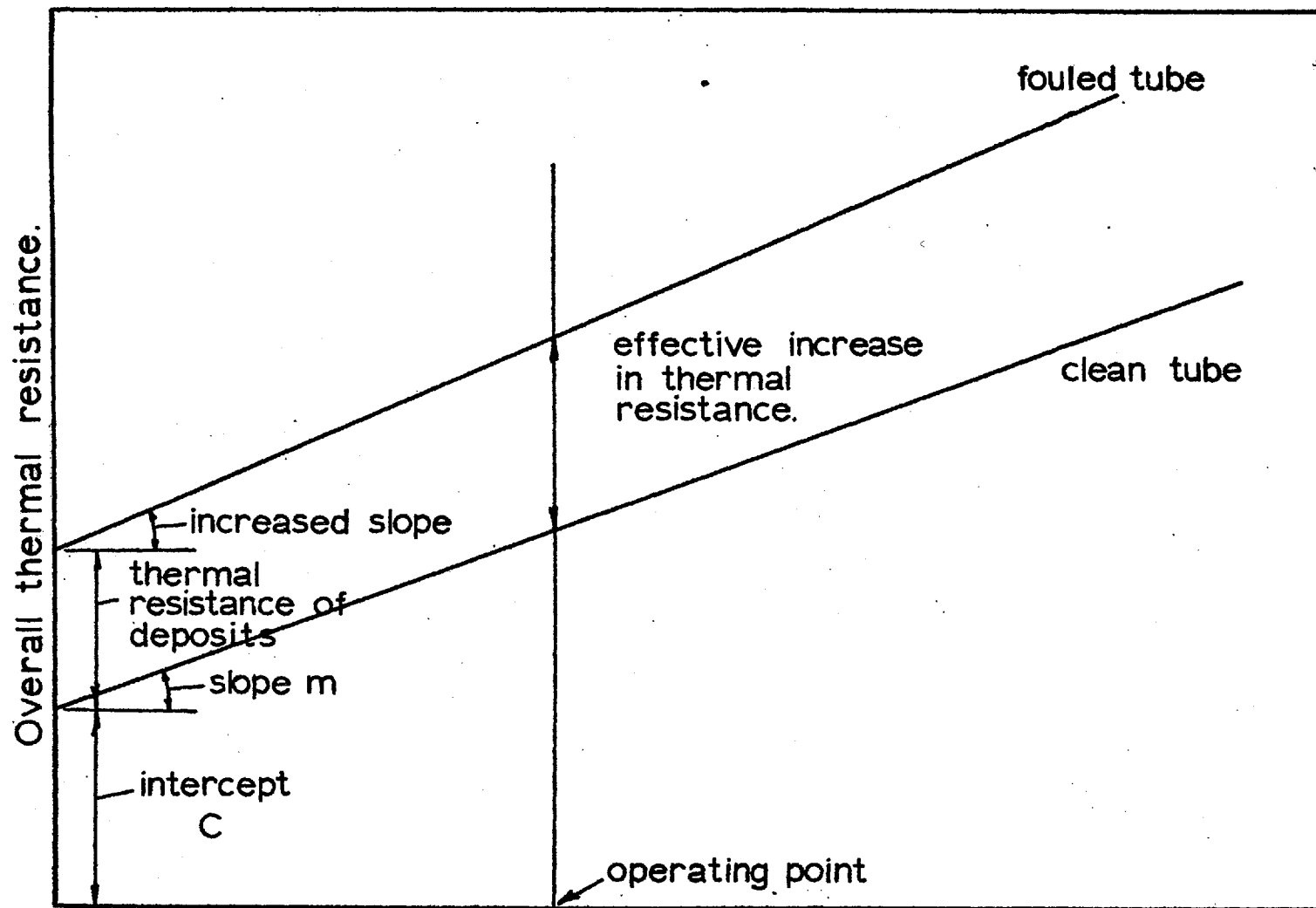
The steam side resistance, $R_s + R_a$, was kept constant by maintaining a constant condenser temperature and condensation rate, by ensuring that the steam was not superheated, and that its air content was constant. The standard condenser pressure and condensation rate chosen were approximately 28.5 in Hg vac, corresponding to a condenser temperature of 91.5°F, and 10 lb/ft²h, these values being typical of the normal operating conditions in the condensers from which the sample tubes were taken.

The variation in the water side resistance, R_w , was obtained by varying the cooling water velocity and temperature, the combinations used being those required to maintain the standard condenser conditions. This involved using cooling water velocities in excess of those normally used in condensers and thus the possibility of dislodging some of the deposits on the inner surface of the tube. This effect was only noticed in the case

of one of the tubes tested, in which the internal deposits had previously been disturbed by brushing, see Chapter 12.

As has already been stated the Wilson Line Plots for clean and fouled tubes should be parallel. It was found, however, that they were not, the slope of the lines through the experimental points increasing with fouling as in Fig.1. This meant that the increase in thermal resistance given by the difference between the intercepts on the resistance axis was less than the difference in thermal resistance at normal condenser water side conditions. The effective increases in resistances reported are therefore those measured at a cooling water velocity of 6.5 ft/s and a mean cooling water temperature of 70°F. These conditions were selected as being typical of those found in modern condensers.

This particular combination of cooling water velocity and temperature is referred to as the Operating Point and as the differences in slope were not large it does not require close definition.



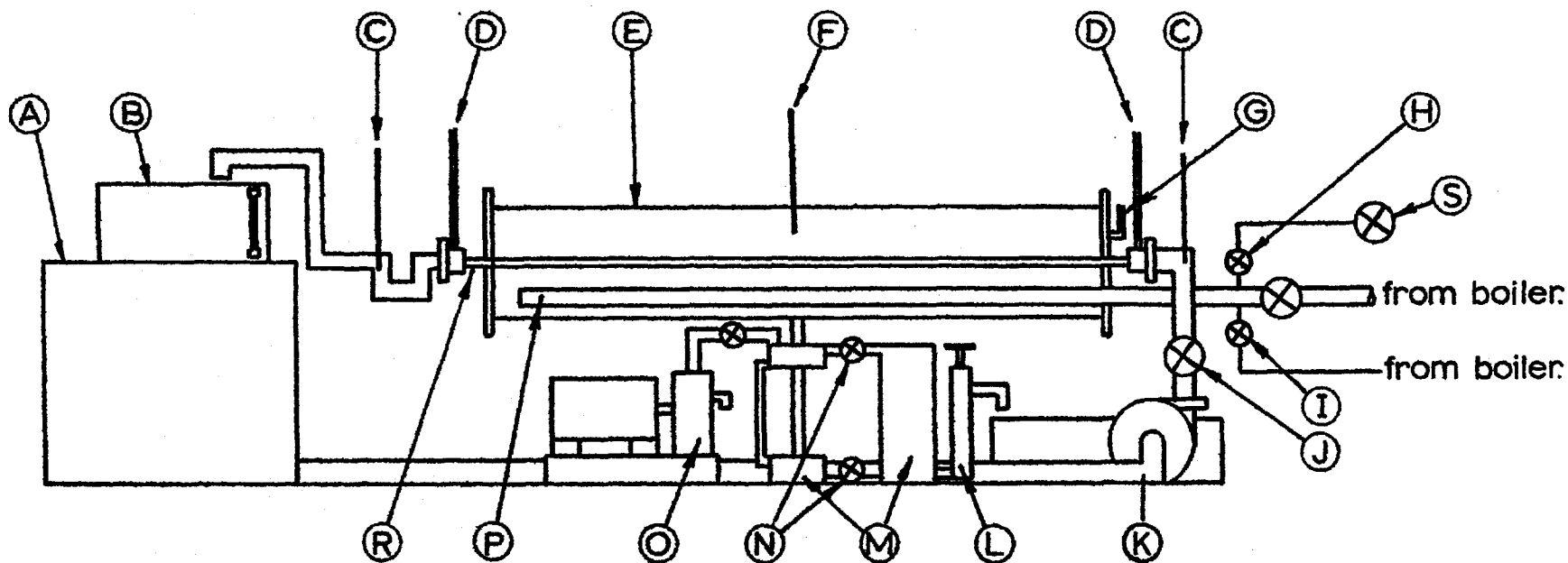
Function of experimental variables.

Fig.1. WILSON LINE PLOT.

CHAPTER 4

APPARATUS

The apparatus, in its final form is depicted in Fig.2, (for photographs of the apparatus see Figs. 3, 4 and 5.) It consisted of a horizontal cylindrical condenser chamber, 6 ft long and 9 inches in diameter, in which the condenser tube to be tested was mounted axially. The condenser chamber was made up of three sections, each 2 ft long, so that tests could be made on 2 ft, 4 ft or 6 ft lengths of condenser tube. However all measurements were made using the maximum length. The tubes passed through gland nuts in the tube plates closing the ends of the condenser chamber and projected at both ends to allow the cooling water connections to be attached. When using all three condenser sections the length of condenser tube required was 7 ft. Steam was raised in a vertical gas fired fire-tube boiler with a nominal rating of 50 lb of steam per hour from and at 212°F. The boiler delivery pressure was maintained approximately constant by an on/off valve in the gas supply pipe controlled by the boiler pressure. This device could, initially, be set to give mean delivery pressures within a range between 20 and 200 lbf/in² gauge. However, as described in the following section, the control was modified to extend the lower end of the range to pressures below atmospheric pressure. The boiler was connected, via a throttle valve, to a perforated distribution pipe, also made up of 2 ft sections,



Key

A cooling water reservoir tank.
 B cooling water measuring tank.
 C cooling water thermometer.
 D connections for water manometer.
 E condenser.
 F condenser thermometer.
 G connection for condenser manometer.
 H air bleed valve.
 I desuperheater valve.

J cooling water throttle valve.
 K cooling water circulating pump.
 L condensate pump
 M condensate chambers.
 N condensate chamber isolating valves.
 O vacuum pump.
 P steam distribution pipe.
 R condenser tube.
 S air valve.

FIG. 2. DIAGRAM OF APPARATUS.

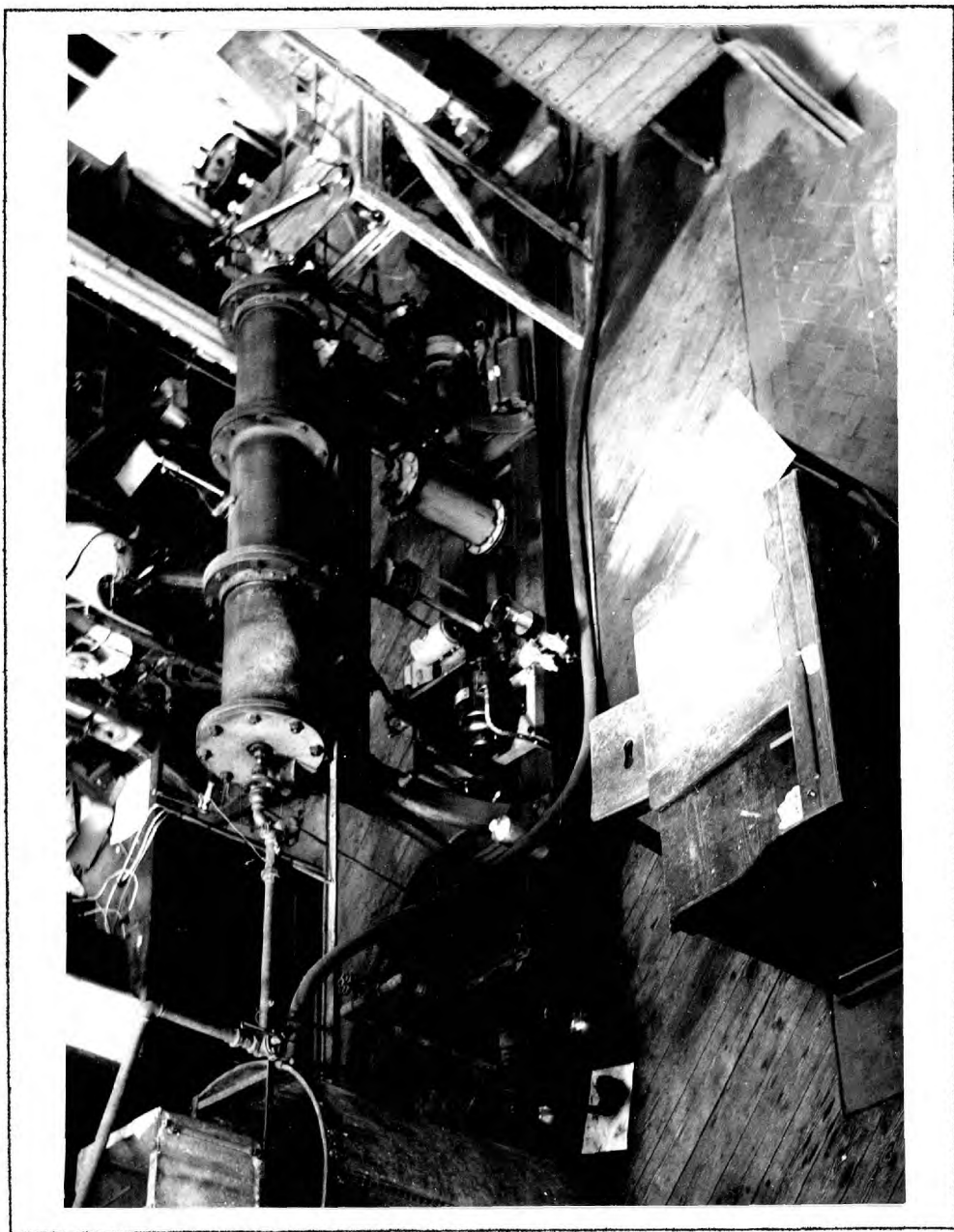


FIG. 3. PHOTOGRAPH OF APPARATUS.

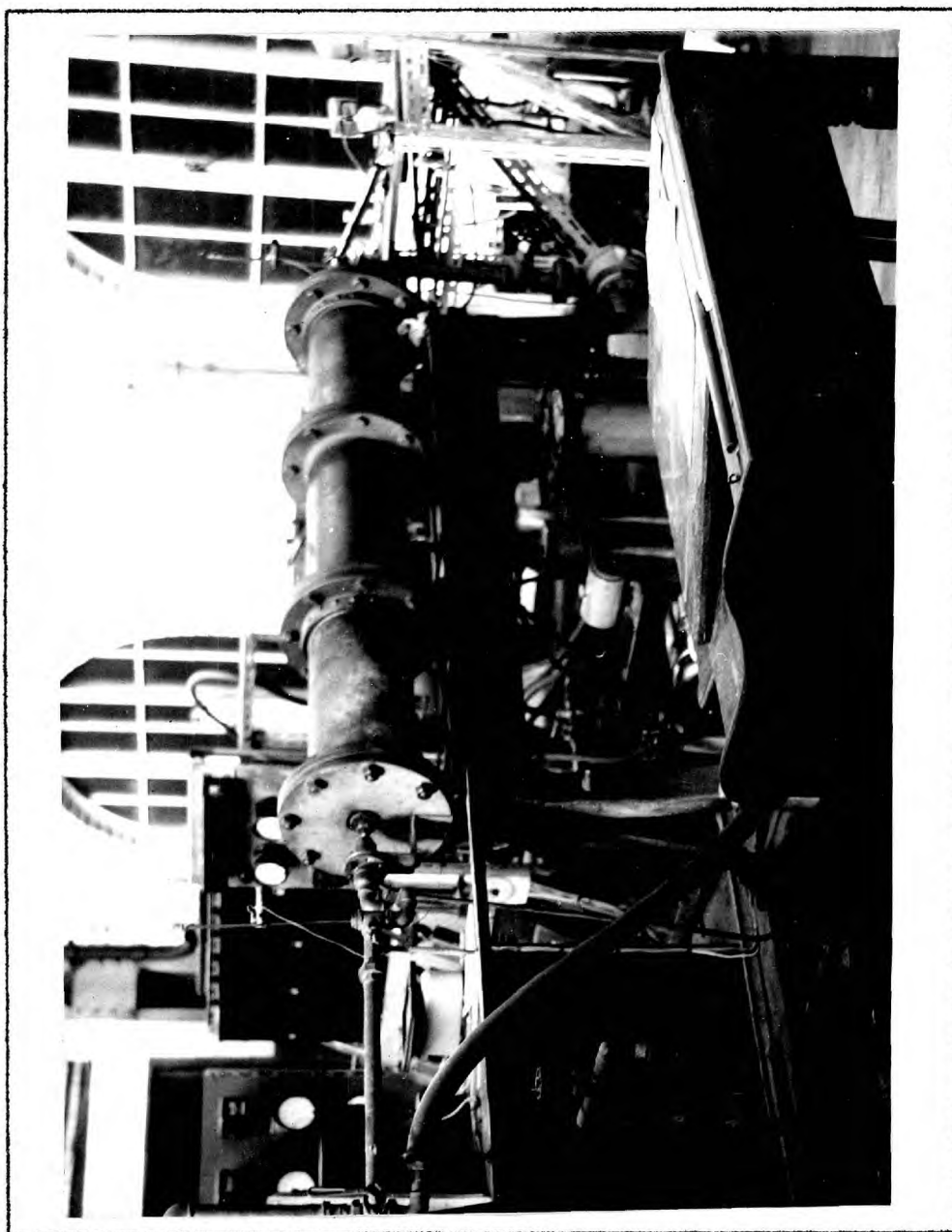


FIG. 4. PHOTOGRAPH OF APPARATUS.

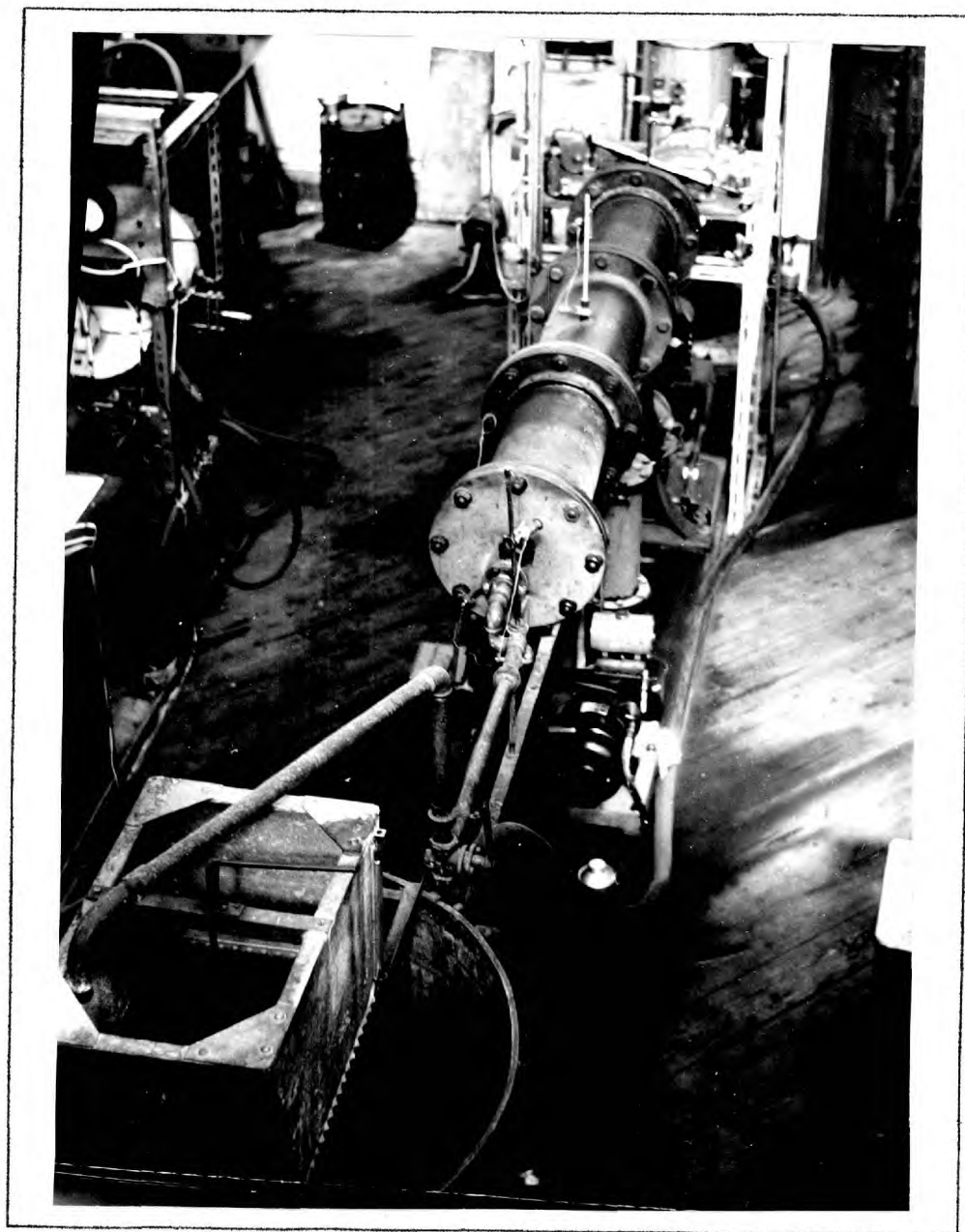


FIG. 5. PHOTOGRAPH OF APPARATUS.

running the length of the condenser chamber. This pipe was in the same horizontal plane as the condenser tube and the perforations were arranged so that the steam was not discharged directly on to the condenser tube surface. Two auxilliary valves were fitted to the condenser side of the body of the steam throttle valve. One, a fine control air bleed valve, controlled the quantity of air entering the condenser with the steam. The other controlled the flow of water, from the bottom of the boiler, used to desuperheat the steam entering the condenser. Condensate drained from the condenser through a pipe, attached to the underside of the centre section, into a double compartment condensate chamber.

The two compartments were mounted side by side and were connected by two pipes, one connecting the tops and the other the bottoms of the two compartments.

There was a valve in each connecting pipe and also a non-return valve in the lower pipe. The first compartment, i.e. that into which the condensate drained initially, was of known cross section area and fitted with a sight glass so that the condensate flow rate could be measured. The second compartment was fitted with a valve opening to atmosphere.

The condensate was removed from the second section of the condensate tank by a hand pump and discharged into a settling tank which was also the supply tank for the boiler feed pump. The settling tank was connected to the mains

water supply, so that condensate losses could be replaced, and to a drain so that the rust and debris from the condenser that settled out could be removed.

Cooling water was circulated through the condenser tube by an electrically driven centrifugal pump capable of delivering 20,000 lb/h against a head of 20 ft. The cooling water flow rate was varied by varying the pump speed by adjusting a variable resistance in the driving motor field circuit. Fine control was obtained by a throttle valve in the pump delivery pipe. The pump was supplied with water from a 170 gallon reservoir tank to which the water was returned after passing through the condenser tube. Provision was also made for discharging the water from the tube to waste and for supplying the pump and the reservoir tank with water from the mains.

A calibrated cooling water flow rate measuring tank, fitted with a sight glass and a quick release valve, was placed between the discharge from the condenser tube and the reservoir tank.

The condenser tube was connected into the cooling water circuit by the entry and exit sections shown in Fig.6. As these sections contain the pressure tapings, used to measure the pressure drop over the condenser tube, their interior dimensions are similar to those specified for measuring orifices in B.S. 1048.

The condenser tubes were cut to the required length and their ends squared off in a special jig. The outer diameter

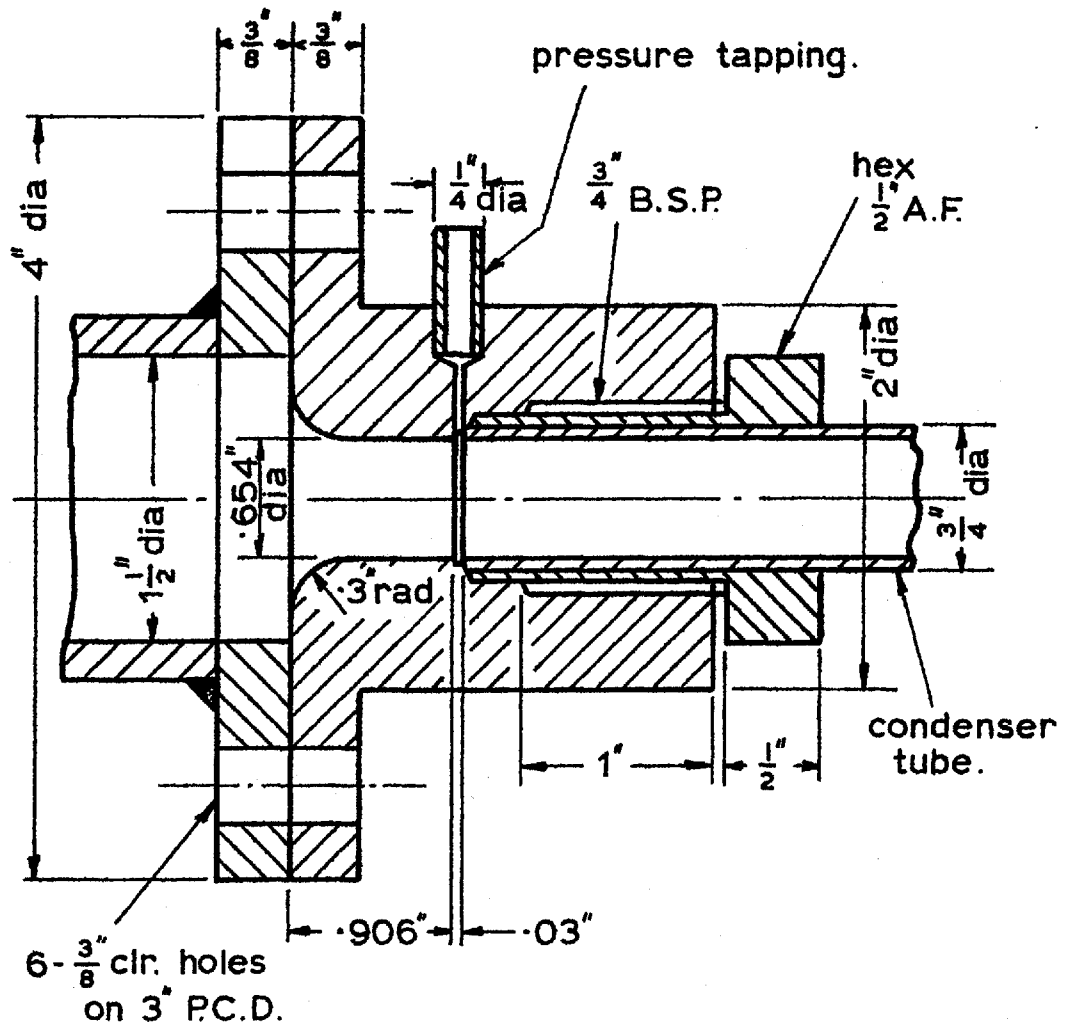


FIG. 6. ENTRY AND EXIT SECTIONS
FOR $\frac{3}{4}$ " O.D. 18 GAUGE.
CONDENSER TUBES.

at each end was scraped down to obtain a push fit in the entry and exit sections. The interior deposits, and if necessary the inner surface of the tube, were then scraped away locally to bring the inner surfaces of the tube and the entry and exit sections into line.

The pressure tappings in the entry and exit section were connected to the lower ends of an inverted vertical U tube water manometer. The upper end of the U tube was filled with air to the pressure required to make use of the full height of both limbs of the manometer. Air was removed from the condenser by an eccentric-vane type vacuum pump, swept volume 4 ft³/min, driven by a constant speed electric motor. The pump, which was connected to the top of the first section of the condensate tank, was drip fed with a special oil which was discharged from the pump as an oil-water emulsion. To recover the oil the emulsion was heated to a temperature of 230°F until boiling ceased and then filtered and allowed to stand for twelve hours to separate out the remaining water.

The condenser temperature was measured by a mercury-in-glass thermometer placed in a mercury filled pocket which projected vertically into the middle section of the condenser chamber. The inner end of the pocket was 0.5 in from the surface of the condenser tube.

The condenser pressure was measured by an inclined mercury manometer the upper end of which was

connected, via a safety reservoir, to a second vacuum pump and the lower end to a pressure tapping in one of the condenser chamber tube plates. The reference pressure, i.e. that in the safety reservoir between the inclined tube and the second vacuum pump, was measured by a McLeod type pressure gauge. The connecting tubes and valves for this manometer were so arranged that the connecting tubes could be cleared of condensate and the mercury prevented from entering the condenser and the vacuum pump. A vertical mercury manometer, connected to the same pressure tapping, was used to measure the condenser pressure when it exceeded the range of the inclined manometer i.e. 2.5 in Hg abs.

The cooling water temperatures were measured with mercury-in-glass thermometers. The inlet thermometer was placed in the right angled bend where the 2 in diameter delivery pipe from the pump was connected to the inlet section to the condenser tube. The outlet thermometer was placed in the 1 in diameter discharge pipe which was connected to the exit section via four right angled bends. It was assumed that due to the turbulence caused by those bends that the temperatures measured would be the mean mixing temperatures.

Illuminated magnifying readers were used to read all thermometers. The final form of the apparatus was the result of modifications made necessary by failure first to maintain constant steam side conditions and then to obtain

repeatable results.

As a considerable proportion of the time and effort devoted to this project was absorbed in overcoming these initial difficulties, an account of the various problems encountered and their solutions is given in the next chapter.

CHAPTER 5

MODIFICATION OF APPARATUS

This account of the development of the apparatus is not in chronological order. The attempts to overcome the various shortcomings of the original design were made simultaneously as was the development of the experimental procedure. For the sake of clarity each problem is discussed separately.

Condensate extraction

In the original design condensate was to be removed from the condenser by a pump trap operated by steam from the boiler. Flow through the trap was controlled by a float which, when raised by the condensate, closed the condensate inlet valve and opened a steam inlet valve. Steam from the boiler then forced the condensate out of the trap through a non-return valve. The return of the float to the bottom of the trap closed the steam valve and re-opened the condensate valve. This method had to be abandoned because, when the condensate valve re-opened, the steam in the trap entered the condenser. This steam together with that formed by evaporation of condensate by the heated walls of the trap,

changed the condenser steam side conditions which the experimental method required to be kept constant.

The effect of using compressed air to operate the trap was investigated but the air entering the condenser from the trap when the condensate valve re-opened also caused too great a change in the steam side conditions.

Eventually, as none of the electrically driven condensate extraction pumps available appeared to be suitable, a hand pump was obtained. This pump, though generally satisfactory, had one major failing. Due to air leakage back into the pump through the pump outlet valve the pump periodically "deprimed" and would not "reprime" at the standard condenser pressure.

This difficulty was overcome by using two separate condensate chambers in series, the first connected to the condenser and the hand pump connected to the second. When the pump deprimed the two isolating valves between the condensate chambers were closed and air admitted to the second chamber until the pressure was high enough for the pump to prime. When the isolating valves were re-opened the air that had been admitted flowed into the condenser and was removed by the vacuum pump. The effects of the introduction of this air on the experimental procedure are explained in another chapter. Even in its final form this method of removing the condensate was not entirely satisfactory. Depriming of

the hand pump during a test sometimes resulted in the first condensate chamber filling before the pump could be reprimed and in water entering the vacuum pump. This increased the clearance between the eccentric rotor and the pump body and reduced the volumetric efficiency of the pump and therefore the equilibrium air partial pressure at the standard steam temperature and condensation rate. The pump had then to be dismantled and re-assembled with the correct clearance.

Measurement of Cooling Water Flow Rate

In the original design the cooling water flow rate was measured by discharging the water into a tank mounted on a calibrated weighing machine. This tank was emptied after a measurement had been made through a connection to the inlet side of the cooling water circulating pump.

The cooling water was diverted into the weighing tank by a two-way valve in the discharge pipe and it was found that the operation of this valve changed the water flow rate, and thus the experimental conditions, appreciably. Attempts to prevent this change by equalising the hydraulic resistances of the two flow paths were only partially successful. It appeared that the momentary stopping of the flow during the movement of the valve disturbed the flow through the pump to such an extent that it took some time to return to its previous operating point.

The original method of emptying the weighing tank also altered the cooling water flow rate and so, instead, an auxiliary water pump was used to pump the water from the weighing tank to the reservoir tank.

However, owing to the failure to prevent the change in the flow rate caused by the operation of the two-way valve, it was decided to measure the flow rate volumetrically. This was done by placing a calibrated tank, fitted with a sight glass, above the reservoir tank, and discharging the cooling water from the condenser tube into it. The water then flowed into the reservoir tank through a quick opening valve in the bottom of the measuring tank. To measure the cooling water flow rate this valve was closed and the time taken for the water level to rise between two markers on the sight glass measured. The valve was then re-opened, the tank emptying in less than five seconds at the maximum cooling water flow rate. The fall in the water level in the reservoir tank during these measurements did not affect the pump delivery. This method of flow measurement ignored the change in the density of the water with temperature. The range of outlet water temperatures was between about 60° and 85° F for which the decrease in density is 0.3 per cent and this change was partly compensated for by the expansion of the measuring tank, so the error involved was negligible.

Measurement of Condenser Pressure and Air Leakage

It was realized from the beginning that the heat transfer coefficients from the condensing steam would be considerably influenced by the presence of non-condensable gases, i.e. air, and that to maintain constant steam side conditions the partial pressure of the air would have to be kept constant or, alternatively, the effect of changes in partial pressure measured.

In the event it was found that the control and measurement of the air leakage into the condenser was the major experimental difficulty encountered and that the quantities of air required to change the experimental conditions appreciably were very small indeed.

It was intended, initially, to determine the partial pressure of the air from measurements of the condenser temperature and pressure, the latter being measured by the vertical mercury manometer. It soon became evident that the difference between the manometer reading and the local barometric pressure could not be measured with sufficient accuracy, so the inclined mercury manometer was constructed. This manometer, which could be read to the nearest 0.005 in Hg over a range of from 0 to 2.5 in Hg abs, was also found to be too insensitive.

An attempt was then made to construct a vertical manometer, using vacuum pump oil as the manometric fluid,

to measure the difference between the condenser and auxiliary vacuum pump pressures. This manometer appeared to be capable of providing the required sensitivity but it was abandoned as being too complicated. The main operating difficulties were due to the need to prevent the oil entering the condenser and to keep the connecting pipes free of condensate; to the sensitivity of the manometer to changes in oil temperature; to the "bubble pump" effect caused by the release of the air and water vapour in the oil and the difficulty in measuring the liquid column to the required accuracy. The photographs of the apparatus, Figs. 3, 4 and 5, were taken while work on this manometer was in progress.

More elaborate methods of determining the quantity of air in the condenser, such as condensing samples of steam and measuring the air separated out, were considered but also rejected as being too complicated.

No further attempts were made to measure the air partial pressure directly. Instead, the air leakage rate into the condenser was used as a measure of the air partial pressure during standard tests. This leakage rate was measured over the period that the apparatus was shut down between tests.

The assumption was that if the air leakage rate into the condenser was constant then the vacuum pump would establish

the same equilibrium air partial pressure at the standard conditions. The tests made to check this assumption are described in Chapter 9.

For convenience the leakage rate was expressed as the rate of increase in the air partial pressure, measured when the apparatus was shut down, corrected to 65°F. This rate was calculated by subtracting the saturation pressure, at the condenser temperature, from the condenser pressure, measured with the inclined mercury manometer, dividing by the time that the apparatus had been shut down and then correcting to 65°F by multiplying by the ratio of the absolute temperatures. The volume of the condenser and associated pipe work etc., involved in this measurement was calculated to be 3.22 ft^3 thus an increase in the air partial pressure of 1 in Hg/h at 65°F corresponds to an air leakage rate of $8.2 \times 10^{-3} \text{ lb/h}$ and the standard air leakage rate, 0.0225 in Hg/h, to $1.85 \times 10^{-4} \text{ lb/h}$. This gives, for the standard condensation rate, a ratio of steam to air flow rates of about 6.5×10^4 which is lower than those reported to have a significant effect on condensation heat transfer coefficients. This ratio, and the corresponding thermal resistance, have no general significance because, for the type of condenser used in these experiments, the effect of air leakage depends primarily on the characteristic of the air extraction pump, which determines the equilibrium air partial pressure.

It is worth pointing out that, because of this characteristic of vacuum pumps, for very small leakage rates the time taken to reduce the air partial to the equilibrium value can become excessive. From the practical point of view it is an advantage to have appreciable air leakage into the condenser provided that it can be measured and controlled.

During the development period the air leakage rate into the condenser was very high but by treating all welds, joints etc., with jointing compound and repacking all valve glands, it was eventually reduced to 0.0225 in Hg/h and it remained at this value for the first six months of routine testing. This leakage rate was therefore adopted as the standard. Then, following the replacement of a brass valve that had failed, due to attack by mercury distilled from the manometers, an overhaul of all joints and packings reduced the rate to less than 0.005 in Hg/h.

Because of the change in the equilibrium air partial pressure, it was then found that the results of tests made prior to the overhaul could not be repeated, the measured heat transfer coefficients being higher, and the time taken by the apparatus to attain equilibrium conditions was considerably longer. (see Chapter 7 for a description of the experimental procedure).

To re-establish the original experimental conditions air was bled into the condenser so as to restore the initial

leakage rate. The first attempt to do this, using lengths of fine hypodermic tube, failed because the smallest diameter tube readily available gave leakage rates higher than required. The required rates were eventually obtained by using an Edwards variable flow air bleed valve which was connected to the condenser side of the steam throttle valve. The air thus mixed with the steam as it entered the condenser. The maximum air flow rate through this valve gave an air partial pressure of about 2 in Hg at the standard condenser temperature and condensation rate. When first fitted the air bleed valve was used to maintain the total air leakage rate at the original value i.e. 0.0225 in Hg/h. As it required ten hours or more to measure the leakage rate, a series of tests were made to determine the relationship between air leakage rate and the resultant change in thermal resistance. Details of these tests are also given in Chapter 9 and the results are shown in Fig. 13.

Thereafter the air bleed valve was only adjusted to maintain the leakage at approximately 0.0225 in Hg/h and the measured thermal resistances were corrected for differences from this rate using the data from Fig. 13.

The assumption that a constant air leakage rate would give a constant air partial pressure assumed that the vacuum pump performance also remain constant. To make sure that

this was so, the clearance between the rotor and the pump body was checked periodically. Using the air leakage rate as a measure of air partial pressure also required that the air content of the steam from the boiler had to be maintained constant or preferably reduced to zero.

Water usually contains appreciable amounts of air in solution and during the first experiments it was noticed that, when fresh water was pumped into the boiler by the feed pump, the measured heat transfer coefficients decreased abruptly. Thereafter the boiler was filled with water at the beginning of a test and the feed pump switched off. The test period was then limited by the capacity of the boiler.

To reduce the air content of the steam the boiler was brought up to its operating pressure under vacuum i.e. with the steam throttle valve fully open and the condenser vacuum pump running. The normal boiler operating pressure was maintained at 4.0 lbf/in^2 gauge to ensure that there was no air leakage into it. Another reason for using a low boiler pressure is discussed in the next section. The tests described in Chapter 9 also checked the air content of the steam from the boiler and showed that it was virtually air free.

Prevention of Super-heating

Owing to the difficulty experienced in measuring the condenser pressure, the condenser temperature and the

condensation rate were used as measures of the standard test conditions. This assumed that the steam in the condenser was dry-saturated or wet.

As the water level in the boiler fell during a test the heat transfer surface area in the steam space above the water increased. This decreased the steam dryness fraction and after a time the steam, after throttling down to the condenser pressure, became super-heated. Heat transfer coefficients for the condensation of super-heated steam are lower than those for wet steam but provided the amount of super-heat is small, the coefficients for super-heated and wet steam are found to be the same if the saturation temperature is used in calculating the super-heated coefficient. Because the condenser pressure could not be measured with sufficient accuracy to obtain the saturation temperature, the experimental method could not be used when condensing super-heated steam. The first attempt to prevent super-heating was by lowering the boiler pressure to the lowest value that the automatic control would give i.e. 25 lbf/in^2 gauge. This delayed the onset of super-heating but did not prevent it. The boiler pressure control was then modified by preloading the control diaphragm with a mercury column. This extended the range of the control down to less than 5 lbf/in^2 abs but even at this pressure the steam eventually super-heated.

The difficulty was eventually overcome by injecting

water into the steam as it entered the condenser, through a needle valve at the same point as the air bleed. The water used was initially taken from the condensate settling tank, the difference between atmospheric and condenser pressure providing the required flow. This water was found to contain appreciable quantities of air so, to provide air free water, the supply to the needle valve was taken from the bottom of the boiler through a lagged pipe. The water thus entered the valve approximately saturated at boiler pressure. The "flashing" of the steam formed as this water entered the condenser would form a spray of water droplets and hence provide the conditions required to desuper-heat the steam. The amount of desuper-heating water required was small, so this additional demand on the boiler did not materially reduce the experimental time available. As the desuper-heating water flow rate appeared to introduce another experimental variable tests were made to see if the heat transfer coefficients were affected by it. It was found that there was no difference in the coefficients measured over the whole range of desuper-heating water flow rates, i.e. from zero to the rate which provided sufficient "flash" steam to give the standard steam condensation rate.

This was probably because the method used to distribute the steam in the condenser separated out the water droplets before the steam reached the condenser tube surface.

CHAPTER 6

CALIBRATION OF INSTRUMENTS

Cooling Water Thermometers

Cooling water temperatures were measured with specially made mercury-in-glass thermometers. These thermometers were constructed to measure temperatures between 40°F and 100°F. The scales, 11 in long, were sub-divided to 0.1 deg F and were read to the nearest 0.01 deg F using illuminated magnifying viewers. The depth of immersion was $1\frac{1}{2}$ in.

The outlet cooling water temperature thermometer was first compared with the inlet temperature thermometer in a variable temperature water bath. This bath, see Fig. 7, was specially constructed for this purpose and was designed to permit the water circulation velocity past the thermometer bulbs and the rate of water temperature rise to be varied.

Prior to a calibration the bath was filled with water at a temperature less than 40°F and the power input to the heater set to give a temperature rise of 0.23 deg F/min, the maximum rate at which the cooling water temperature increased during experiments when using the standard experimental procedure. The thermometer readers were so positioned that the two thermometers could be read

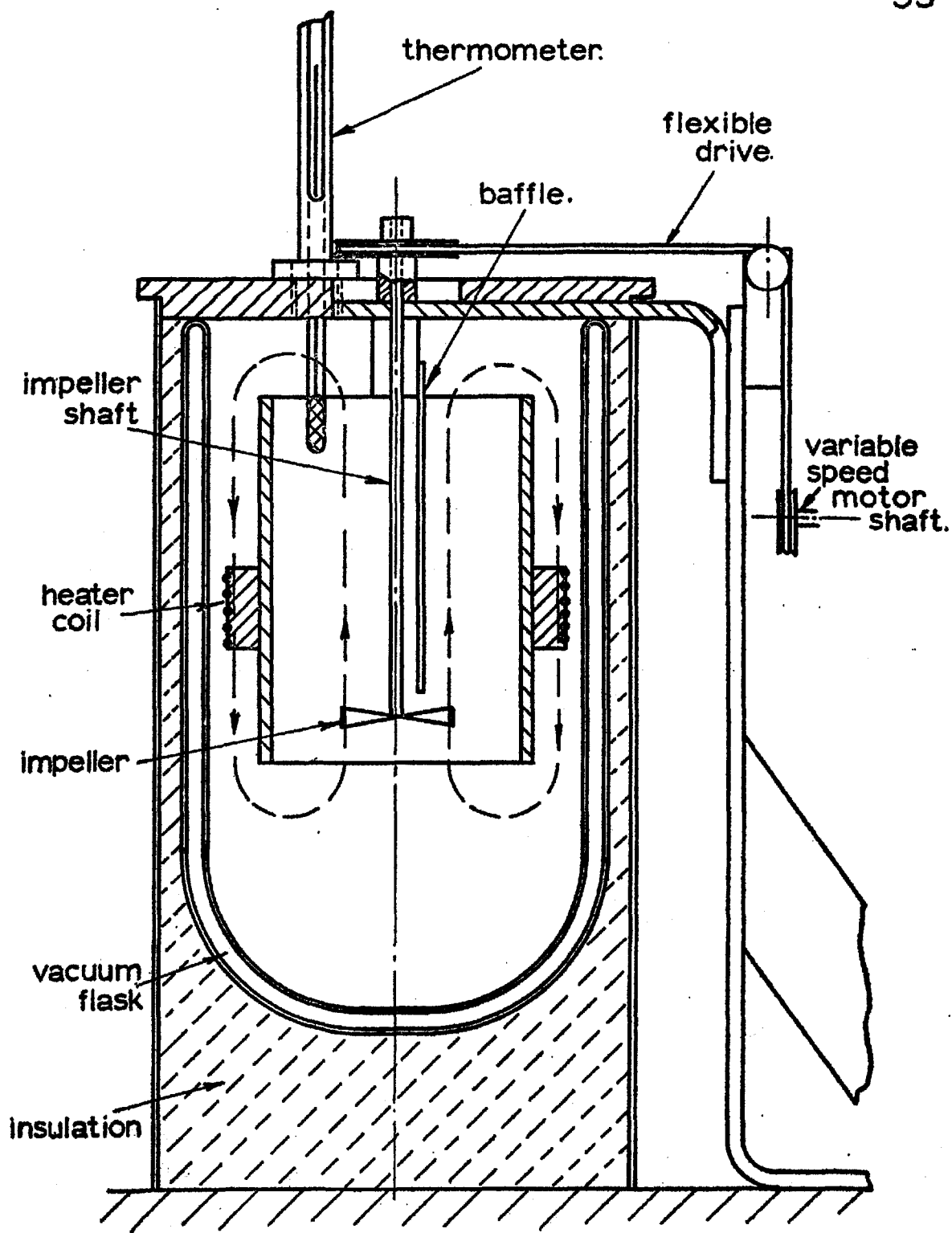


FIG. 7. VARIABLE TEMPERATURE WATER BATH.

virtually simultaneously. Both thermometers were read at intervals of about 0.4 deg F over their working range.

Initially it was found that the relative calibration varied from day to day, differences being at first as great as 0.2 deg F. However, after the thermometers had been taken through their working range about fifty times over a period of a fortnight, a repeatable calibration was obtained. When this was checked three years later, the maximum change was less than 0.02 deg F.

The effect of varying the rate of temperature rise on the calibration was investigated over the range 0.1 deg F/min (that due to the work transfer by the water circulating impeller) to 1 deg F/min and also of the effect of the water circulation rate. It was found that provided the protective skirts fitted by the manufacturer around the thermometer bulbs were removed, neither of these variables affected the calibration.

When in place in the apparatus the thermometer bulbs projected into the cooling water delivery and discharge pipes and were thus subjected to the water pressures. It was noticed during the preliminary experiments that these pressures could produce changes in thermometer readings greater than 0.10 deg F. A correction curve for this effect, Fig.8, was obtained by observing the instantaneous decreases in the

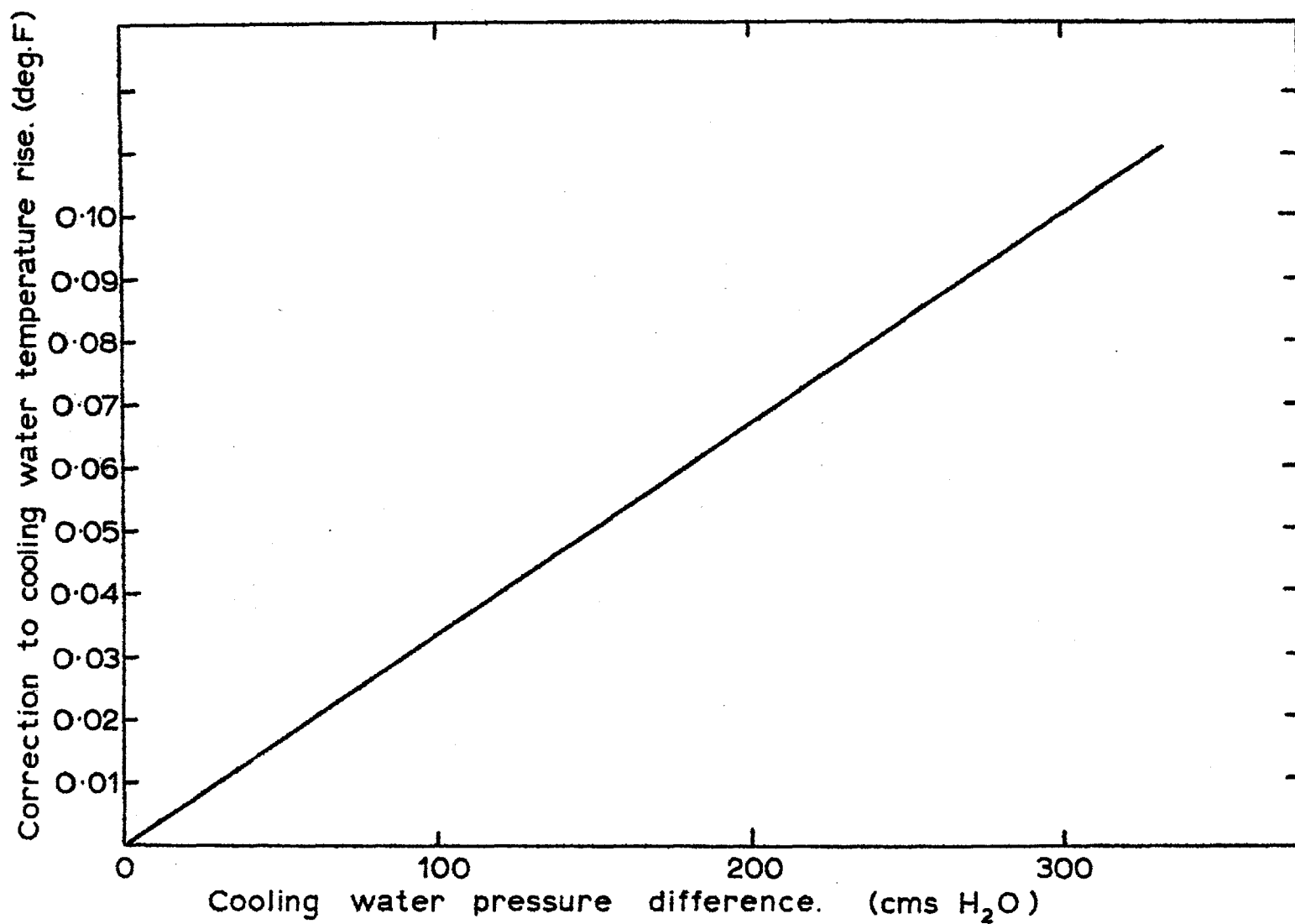


FIG. 8. TEMPERATURE CORRECTION CURVE FOR COOLING WATER PRESSURE DIFFERENCE.

readings of the thermometers when the water pump motor was switched off. The differences between these decreases were measured over the full range of flow rates, and therefore of pressure differences. The difference between the water pressures at the thermometer bulbs was assumed to be proportional to the pressure drop over the condenser tube. This assumption is not strictly correct, but in view of the magnitude of the corrections concerned, the error involved is insignificant.

The inlet thermometer was then calibrated against a thermometer for which an N. P. L. calibration certificate had been obtained. From the two calibrations the table of corrections, Table 2, was compiled.

The thermometer used to measure the condenser temperature was also calibrated against the N. P. L. certificated thermometer.

Cooling Water Measuring Tank.

The relationship between the depth of water in the tank, as measured against a centimeter scale behind the sight tube, and the mass of water in the tank was obtained by placing the tank on a calibrated weighing machine and adding quantities of water, at a temperature of about 70°F, sufficient to increase the depth by about 1 cm.

TABLE 2COOLING WATER THERMOMETER CORRECTIONS

<u>Thermometer Reading</u> °F	<u>Corrections</u>		<u>Thermometer Reading</u> °F	<u>Corrections</u>	
	<u>Inlet</u> deg F	<u>Outlet</u> deg F		<u>Inlet</u> deg F	<u>Outlet</u> deg F
46	-0.16	-0.18	66	+0.02	-0.02
47	-0.18	-0.20	67	+0.02	-0.03
48	-0.20	-0.22	68	+0.02	-0.03
49	-0.21	-0.23	69	+0.03	-0.02
50	-0.22	-0.24	70	+0.04	0
51	-0.22	-0.24	71	+0.04	+0.02
52	-0.22	-0.24	72	+0.05	+0.04
53	-0.21	-0.23	73	+0.07	+0.03
54	-0.20	-0.22	74	+0.07	+0.01
55	-0.19	-0.21	75	+0.06	-0.02
56	-0.18	-0.20	76	+0.05	-0.06
57	-0.15	-0.18	77	+0.04	-0.06
58	-0.12	-0.16	78	+0.02	-0.09
59	-0.10	-0.14	79	-0.01	-0.12
60	-0.06	-0.10	80	-0.06	-0.13
61	-0.02	-0.06	81	-0.08	-0.14
62	-0.02	-0.04	82	-0.09	-0.14
63	-0.01	-0.03	83	-0.10	-0.14
64	+0.01	-0.03	84	-0.09	-0.13
65	+0.02	-0.03	85	-0.09	-0.13

This data was used to plot curves, Fig. 9, of cooling water mass flow rate ($\dot{Q}$ lb/h) against the time taken for the depth of water in the tank to increase by 9, 18 and 27 cm (τ).

Inclined Mercury Manometer

This was calibrated against the vertical water U tube manometer used to measure the pressure drop over the condenser tube. From the calibration a scale card was constructed so that the condenser pressure could be read directly in inches Hg abs.

The first condensate chamber was also calibrated so that the condensation rate could be measured directly, but as explained in Chapter 7, this measurement was not made.

The stop watch, used to time the rise in water level in the measuring tank, was checked against TIM.

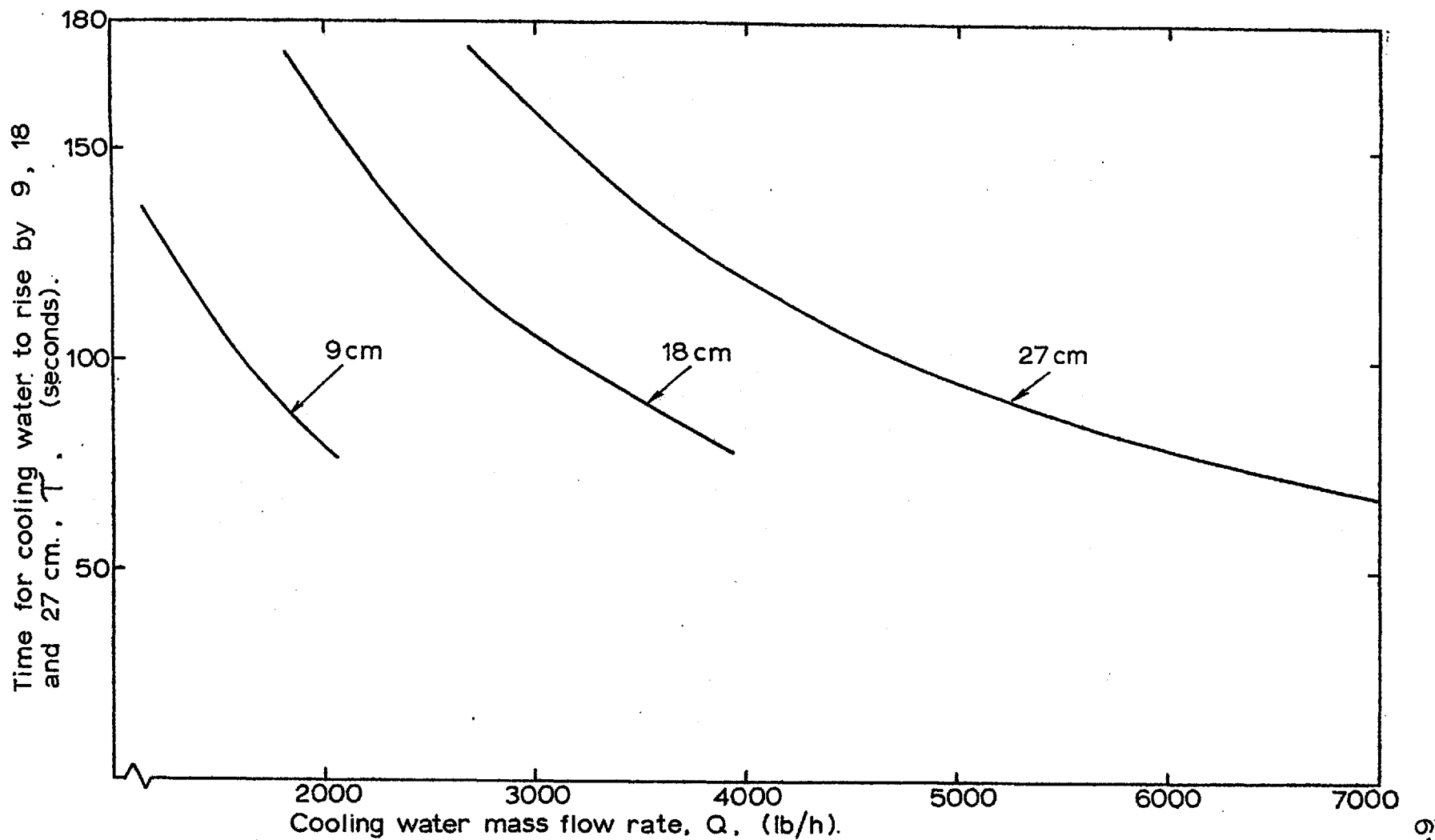


FIG. 9. CALIBRATION CURVES FOR COOLING WATER MEASURING TANK.

CHAPTER 7

EXPERIMENTAL PROCEDURE

The section of condenser tube to be tested was cut to size and fitted into the condenser on the day before a test. The cooling water connections were made and the tube left filled with water. The boiler was filled from the condensate tank up to the level of the steam delivery pipe and the vacuum pump run, with the steam delivery valve open, until the pressure in the condenser and the boiler, as measured by the sloping mercury manometer, was equal to the saturation pressure corresponding to the condenser temperature. The steam delivery valve and vacuum pump isolating valve were then closed, the time recorded and the vacuum pump switched off.

On the following day the condenser pressure and temperature were again measured, the partial pressure of the air in the condenser determined and the air leakage rate calculated. If this was within the range for which corrections could be made the test was started. If not, the condenser was filled with air to a pressure of about 20 lbf/in² gauge and the leak, or leaks, detected by painting likely points, i.e. glands, joints, etc. with a strong solution of detergent. Major leaks revealed themselves at once but those usually responsible for a change in the leakage rate were indicated by the accumulation,

over a period of an hour or more, of a distinctive layer of very white foam. The leaks detected were closed by repacking glands or painting the areas concerned with a shellac jointing compound with the condenser under vacuum. The leakage rate was then measured again. As this required a period of ten hours or so, the test was delayed by a day.

When the air leakage rate was acceptable the condensate extraction pump was primed, air being admitted to the second condensate chamber if necessary, the vacuum pump started and the steam valve opened. The boiler gas burner was ignited and cooling water from the mains passed through the condenser tube at a low velocity, about 2 ft/s, and discharged to waste. At the same time the reservoir tank was filled from the mains. If the mains water temperature was too high crushed ice was added to the water in the reservoir tank. As the boiler pressure increased so did the condenser temperature and as the latter approached the standard value, i.e. about 91.5°F, the steam valve was closed to the setting which gave the standard steam flow rate, and the desuperheater water valve opened. The standard condenser temperature was maintained thereafter by adjusting the cooling water flow rate. During this warming up period the cooling water circulating pump was started and run at minimum speed and the mains supply to the reservoir tank turned off when

the tank was filled to the standard level.

When the boiler pressure reached the standard value, 4.0 lbf/in² gauge, the cooling water velocity was adjusted to give the standard condenser temperature and then the cooling water flow rate and temperature rise were measured to check the steam condensation rate. The steam valve setting and water velocity were adjusted if necessary. The required experimental conditions having been obtained a small quantity of air, sufficient to raise the condenser temperature by about 5 deg F, was admitted as a check on the functioning of the vacuum pump. If the condensate pump "deprimed" during the warming up period this air was admitted first to the second condensate chamber and the pump "reprimed". When the vacuum pump had removed this air i.e. the condenser temperature had returned to its initial value the mains water was turned off and the water from the reservoir tank circulated through the tube, the water velocity being adjusted to give a condenser temperature of about 90.5°F. The main control of the water velocity was by the variable resistance in the pump motor field circuit. The throttle valve in the pump delivery pipe was only used for fine control as it was found that, for large pressure drops over this valve, air in solution in the cooling water was released and carried through the condenser tube. This two phase flow had of course to be avoided.

As the temperature of the recirculated cooling water rose so did the condenser temperature. When the latter reached about 91.0°F the valve in the cooling water measuring tank was closed and the time taken for the water level to rise through one of the standard heights measured. At the same time the cooling water and condenser temperatures were recorded together with the readings of the water manometer measuring the pressure drop over the tube.

The sequence in which the temperatures were read was as follows; cooling water inlet, condenser, cooling water outlet. The cooling water inlet and condenser temperatures were then checked. The mean cooling water inlet temperature being used to calculate the increase in water temperature. As the closing of the measuring tank valve reduced the rate of cooling water temperature rise, the differences between the two values of the inlet temperature were small. If the condenser temperature had changed by more than 0.05°F , the readings were discarded.

Because the boiler pressure was controlled by an on-off valve in the gas supply, there was a fall of about 3 lbf/in^2 in the boiler pressure during the period the gas supply was shut off. To avoid the effect of this variation the temperature reading sequence was always started at the moment the gas supply was shut off i.e. when the boiler pressure was a maximum.

When a set of readings had been taken the measuring tank was emptied and the cooling water flow rate increased to lower the condenser temperature to 90.5°F again. During the interval between readings the product of the cooling water flow rate and temperature rise (corrected for cooling water pressure differences and calibration errors) and the overall heat transfer coefficient were calculated. The constancy of the former provided a check against gross error and on the steam condensation rate.

Similar sets of readings were taken at increasing water velocities and temperatures until the maximum water velocity that the pump could provide i.e. about 12 ft/s, was reached. The reservoir tank was then emptied, via a two way valve in the condenser tube outlet pipe, and refilled from the mains. The standard condenser conditions were maintained by passing mains water through the tube and to waste, as during the warming up period.

When the tank had been refilled the first three or four sets of readings were repeated to ensure that there had been no change in the experimental conditions during the test.

A typical set of experimental readings are given in Appendix 2, together with an example of the subsequent calculations.

If the second set of readings agreed with the first,

the apparatus was shut down until the following day when the air leakage rate was again calculated. If it was unchanged, another condenser tube was installed. When there was an appreciable change in the leakage rate the test was repeated. If the difference between the thermal resistances obtained from the two tests was equal to the difference between the corrections for the two air leakage rates, then the results were accepted. When the results of the two tests could not be reconciled the air leakage rate was measured again and a third test made. On only two occasions was a third test required and on both of these the second and third tests agreed.

For the first thirty tubes tested a second test was made on the day after the first but as the modifications to the apparatus had put the programme a long way behind schedule, and because there was close agreement between the two tests, the second test was discontinued unless there was a change in the air leakage rate.

It had been intended initially to use the calibrated condensate chamber to relate the heat transfer to the cooling water to the change in enthalpy of the steam condensed, chiefly because most experimentalists measuring condensation heat transfer coefficients had done so. But as both the dryness fraction of the steam and the heat transfer from the condenser to the surroundings were unknowns it was decided that no useful information would be obtained from such an analysis.

CHAPTER 8.
PRELIMINARY TESTS

These tests were made to obtain the information required to answer the following questions: (1) Could the required thermal resistance be measured using fresh cooling water or would a supply of sea water have to be obtained?; (2) How were sample tubes to be stored while awaiting test?; (3) With what standard was the thermal resistance of a used tube to be compared?

(1) The deposits on the inner surfaces of marine condenser tubes are composed partially of the substances in solution and suspension in sea water. To use fresh water as the cooling medium would be to run the risk of dissolving part of the internal deposits and thus to reduce their thermal resistance. But to arrange a regular supply of sea water in Central London involved considerable administrative and experimental difficulties.

(2) Sample tubes could only be withdrawn from ships condensers when the ships concerned returned to ports in the United Kingdom. When sample tubes from different vessels arrived at the laboratory together several months might elapse before the last tube could be tested. The choice of storage methods was between dry storage and wet, i. e. filling

the tubes with sea water and plugging, with the possibility that both would change the thermal resistance.

(3) The effect of manufacturing methods and storage prior to installation on the thermal resistance of new heat exchanger tubes was mentioned by several of the contributors to the symposium mentioned in Chapter 1, so the resistance of a new condenser tube selected at random could not be used as a standard of comparison.

With the assistance of the Royal Navy 250 gallons of sea water were obtained and a series of standard tests using first sea and then fresh water were made on a new condenser tube. At this stage it was assumed that sea water would have to be used and the original object of these tests was to determine whether the differences between the properties of fresh and sea water would be significant. From the data available it appeared that the water side resistance for sea water could be up to about 4 per cent higher than for fresh water, the difference in the overall resistances would of course be smaller.

The results of two tests using sea water and one for fresh water are given in Fig. 10. In the calculations the differences between the properties of sea and fresh water were ignored, and from these results it was concluded that the differences in properties was not significant.

At this point, it was decided to make further tests

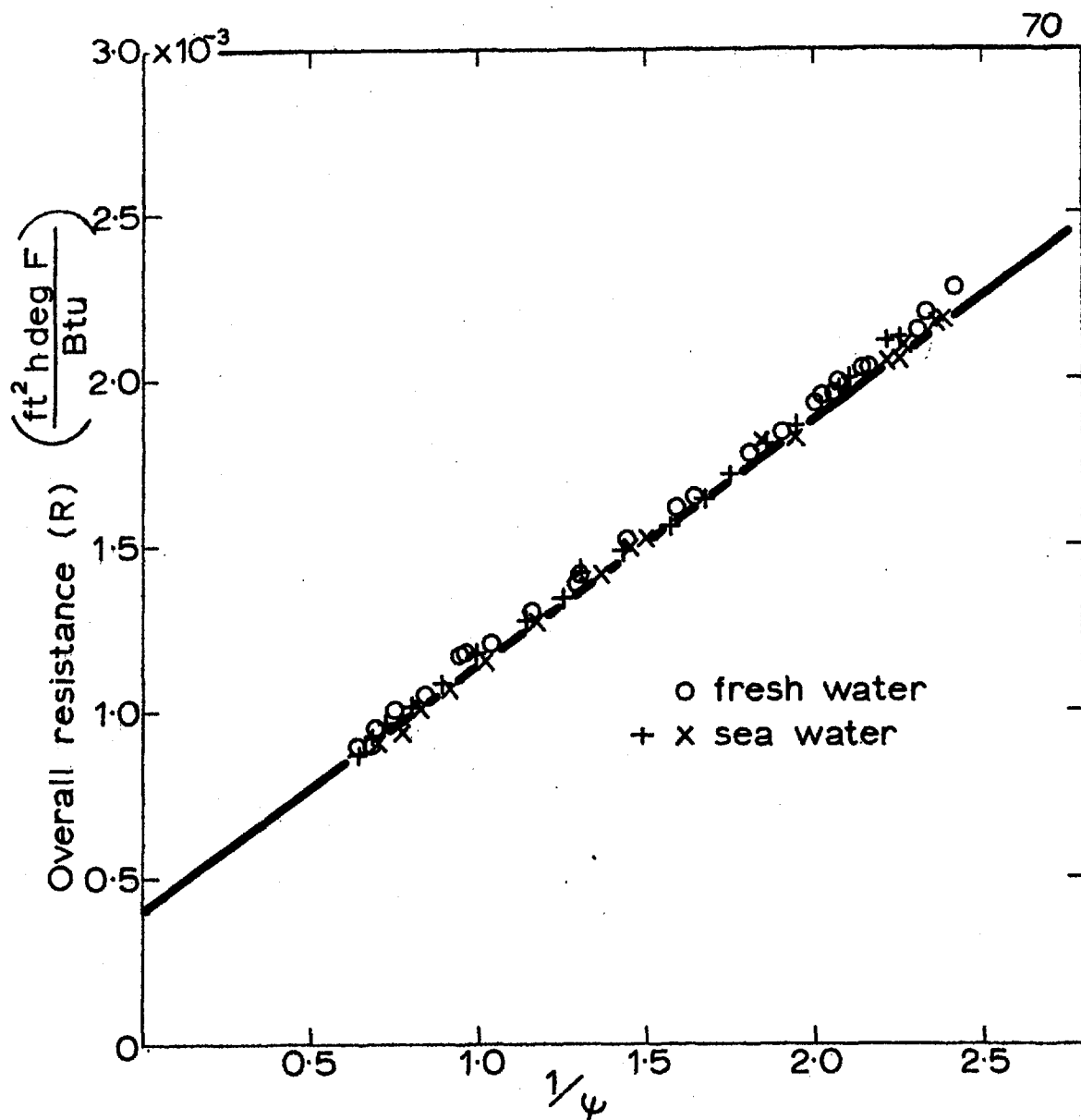


FIG. 10. WILSON LINE PLOT. FRESH WATER-SEA WATER COMPARISON TESTS. (Py. NEW TUBE)

to see if it was in fact necessary to use sea water.

For these tests three tubes, filled with sea water and plugged, were delivered to the laboratory four hours after having been withdrawn from the condenser of the S.S. "Brittany". Two of these tubes (tubes I and II) were tested first using sea water and then fresh water, the first test on tube I being completed three hours after delivery. The third tube (tube III) was tested with sea water only and then refilled with sea water and replugged.

Tubes I and II were stored dry for about six weeks and were then retested using fresh water. Tube III was also retested with fresh water after the same interval.

TABLE 3

Tube I

Date		$\frac{Rop \times 10^3}{ft^2 h \text{ deg F}}$ Btu	$\frac{hop}{ft^2 h \text{ deg F}}$ Btu
22.1.55	Sea water	1.78	562
24.1.55		1.75	572
25.1.55		1.75	572
26.1.55	Fresh water	1.73	578
27.1.55		1.75	572
3.3.55		1.71	584

TABLE 3 cont.

Date		$R_{op} \times 10^3$ $\frac{ft^2 h \text{ deg F}}{Btu}$	$\frac{hop}{Btu}$ $\frac{ft^2 h \text{ deg F}}{Btu}$
	Tube II		
28.1.55	Sea water	1.69	592
31.1.55		1.64	610
3.2.55	Fresh water	1.60	626
4.2.55		1.64	610
4.4.55		1.59	629
6.4.55		1.58	633
24.4.55		1.57	637
	Tube III		
7.2.55	Sea water	1.90	526
8.2.55		1.87	535
10.2.55		1.91	524
29.4.55	Fresh water	2.01	496
4.5.55		1.90	527

The results of these tests are given in Table 3 from which it can be seen that the thermal resistances at the operating point, R_{op} , were not changed appreciably by the change from sea to fresh water or, in the case of tubes I and II by drying. The small decreases in resistance could have been due to the removal of non-adherent deposits by the high water velocities involved in the experimental method or by

the removal of part of the external deposit when removing and reinserting the tubes in the experimental condenser. Further evidence was provided by tests on a tube from the S.S. "Pyrrus", which showed no change in thermal resistance over a period of 12 months during which the tube was stored dry.

When tube III was unplugged for retesting the sea water filling was found to be discoloured. As the thermal resistance had also increased, wet storage with sea water was ruled out. However as the tests on tubes I and II showed that drying appeared to have no effect, wet storage was not investigated further.

Chemical analysis of the internal deposits from the condensers of H. M. S. "Cockade"² (see Appendix 5) showed that they contained only about 6.5 per cent of water soluble matter. This provided further justification for the use of fresh water. As the samples of deposits for the above analysis had been obtained by tapping tubes with a hammer, condenser tubes were also handled carefully to avoid similarly detaching the deposits.

As a result of these tests all other tubes were stored dry and tested using fresh water.

The selection of a suitable standard "new tube" resistance was investigated by making standard tests with a number of new and acid cleaned tubes. Details of the acid

cleaning process are given in Chapter 12.

TABLE 4.

Results of Tests on New and Acid-Cleaned Tubes

Results of Tests on New and Acid-Cleaned Tubes				
Tube No.	Condition	Material	Intercept c. $\left\{ \frac{\text{ft}^2 \text{ h deg F}}{\text{Btu}} \times 10^4 \right\}$	
			Measured	Calculated
Do	New tube	Alumin. brass	3.55	3.8
Py			4.0	
I. G.		Bemal	3.7	
			3.85	
			3.75	
Py 6C(A)	Acid - Cleaned	Alumin. brass	2.95	$\left\{ \begin{array}{l} 6.0, k=14.5 \frac{\text{Btu}}{\text{ft h deg F}} \\ 5.5, k=17.0 \frac{\text{Btu}}{\text{ft h deg F}} \end{array} \right.$
Py 6C(F)			3.3	
Q. M. A(i)		Cupro- nickel	6.1	
Q. M. B (i)			5.9	
			6.3	
L. W. C.		Alumin. brass	3.1	3.8

The sums of the steam side and tube wall resistances for these tubes, obtained from Wilson Line Plots, are given in Table 4. Those for acid cleaned tubes are lower than for new tubes but this may have been due in part to the acid

cleaning having produced a non-wetting surface which, giving drop-wise condensation, would lower the steam side resistance. This would give too low a resistance to use as the standard.

As a check on the results obtained from these tests the sum of the steam side and tube wall resistances was calculated.

The resistance due to the condensate film was obtained in the first instance from the Nusselt equation i.e.

$$\frac{1}{R_s} = h_s = 0.725 \left(\frac{k^3 L p^2 g}{D_o \Delta T} \right)^{0.25}$$

By trial and error, h_s for the standard condenser conditions was found to be 2,600 Btu/ft²h deg F, i.e.

$$R_s = 3.8 \times 10^{-4} \text{ ft}^2 \text{ h deg F/Btu.}$$

Condensation coefficients calculated from the Nusselt equation are frequently lower than those obtained in practice. Fishenden and Saunders⁹ recommend increasing the Nusselt coefficient by 20 per cent whilst McAdams¹⁶ quotes measured coefficients which are up to 2.2 times greater than the Nusselt values.

An attempt was made by Peck and Redie¹⁹ to account for these differences. They tried to ammend the Nusselt formula to allow for the variation in temperature difference

around the circumference of the tube and for the acceleration of the condensate film. Their analysis showed that the variation in temperature difference was insignificant and they failed to ammend the Nusselt equation to allow for the film acceleration. Their analysis did show, however, that the ratio of the measured to the Nusselt coefficient should be a function of the dimensionless group $\frac{L \mu}{k \Delta T}$. The properties being evaluated as for the Nusselt equation i.e. at the mean film temperature.

To test their conclusion they plotted the ratio of measured to Nusselt coefficients against this group for a large selection of published data. From their correlation they obtained the following empirical relationship for the ratio of measured to Nusselt coefficient $\frac{hs}{hs (Nu)}$

$$\frac{hs}{hs (Nu)} = 0.0206 \left(\frac{L \mu}{k \Delta T} \right)^{0.5} + 0.79$$

The agreement between the published data and this relationship, see Fig. 11, is good for high values of $\frac{L \mu}{k \Delta T}$ though data in this region is sparse. For the standard condenser conditions $\frac{L \mu}{k \Delta T} \sim 36$ so the actual condensation coefficient should be about 1.55 x the Nusselt coefficient.

Accepting this ratio, gives a condensate film resistance of $2.5 \times 10^{-4} \text{ ft}^2 \text{ h deg F/Btu}$.

The increase in the steam side resistance due to the

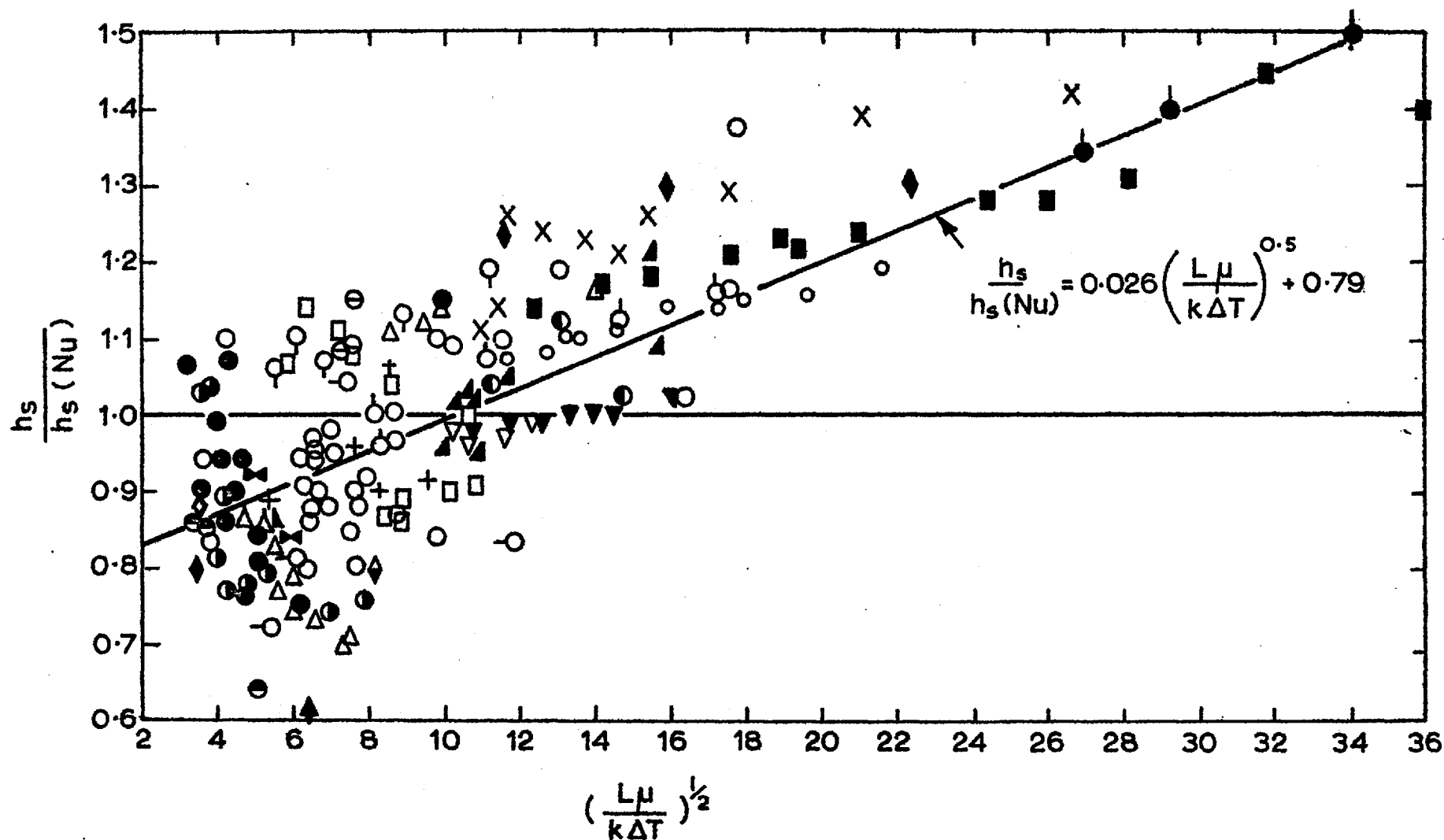


FIG. 11. PECK AND REDDIE CORRELATION OF MEASURED TO NUSSELT CONDENSATION COEFFICIENTS.

presence of air in the condenser was found to be about $0.6 \times 10^{-4} \text{ ft}^2 \text{ h deg F/Btu}$. The method used to estimate this resistance is given in Chapter 9.

The resistance of the tube wall was calculated for the various metals concerned, from $R_t = \frac{D_o(D_o - D_i)}{k (D_o + D_i)}$

The sums of these resistances, i.e. the estimated values of the Wilson Line Plot intercepts, c , is also given in Table 4.

As these estimated resistances were lower than those measured for new tubes but higher than those for acid clean tubes, it was decided to use them as the standard with which to compare the thermal resistances of used tubes.

The agreement between the measured and calculated intercepts for the cupro-nickel tubes depends on the value of the thermal conductivity used. The value given by the makers of the tubes is $17.0 \text{ Btu/ft h deg F}$, but the effective conductivity¹² appears to be about $14.5 \text{ Btu/ft h deg F}$. This difference is due to the oxide film which forms on cupro-nickel surfaces and gives it its anti-corrosion properties.

When the tubes from the S.S. "Dorset" and S.S. "Papanui" were received it was reported that they were made from aluminium bronze, which has a lower conductivity than aluminium brass. The estimated values of c were thus higher than those measured. Samples of both tubes were

analysed by the Yorkshire Copper Company who reported that they were in fact made of aluminium brass.

Although the values of c for the various new and cleaned tubes tested differed the slopes, m , of the best straight lines through the experimental points were equal.

The linear relationship between the water side resistance and the experimental variables was obtained from the Sieder-Tate²¹ formula i.e. $Nu = K.Re^{0.8} Pr^{1/3} (\mu / \mu_s)^{0.14}$, and, as shown in Appendix 1, the slope, m , is a function of the tube diameters and the Sieder-Tate constant K . The value of m given by these tests gives a value for K of 0.027 which is that suggested by McAdams.¹⁶ Other workers have obtained values of K ranging from 0.018 to 0.031 and in these tests the value of K was found to be lower for used tubes, falling to a minimum of about 0.024. This point is discussed further in Chapter 14.

The agreement between the measured and estimated values of both the slope and intercepts obtained from the Wilson Line Plots for these new and cleaned tubes was taken to confirm the accuracy of the instrumentation and the suitability of the experimental method.

CHAPTER 9

AIR LEAKAGE TESTS

As has been explained in Chapter 5, attempts to measure the condenser pressure with sufficient accuracy to enable the air partial pressure to be deduced were unsuccessful so the air leakage rate was used instead.

The air partial pressure during a standard test would be that at which the air flow rate into the condenser was equal to the vacuum pump extraction rate. So, if the flow rate into the condenser could be kept constant, then, provided the vacuum pump performance did not change, the air partial pressure would be constant. Air could enter the condenser with the steam from the boiler and through glands, joints etc., so the problem was to ensure that the sum of these flows did not change. The method of providing air free steam by generating it under vacuum and of supplementing the varying gland leakage rate by an air bleed valve have already been described. The tests now described were made to determine, so far as practicable, whether or not these precautions did result in a constant air partial pressure during standard test conditions and also to provide the data required to correct resistances measured when the leakage rate differed from the standard.

The test procedure was as follows.

First the air leakage rate into the condenser was reduced, by repacking glands and remaking joints, to 0.003 in Hg/h, on the assumption that the effect of the change in the temperature of the apparatus on the leakage rate between shut down and standard test conditions would be relatively greater when the leakage rate was small.

Then, using a clean tube whose thermal resistance had been measured at standard conditions, cooling water from the mains, at approximately constant temperature, was pumped through the tube at the rate required to maintain the standard condenser temperature and condensation rate. The air bleed valve had been previously set to give a known total air leakage rate but, while the standard condenser conditions were being established the air valve was closed. When the vacuum pump had reduced the air partial pressure to its equilibrium value, i.e. the overall thermal resistance at the experimental point used had remained constant for about 15 minutes, the vacuum pump valve was shut, the air valve opened and the stop watch started. Thereafter the air partial pressure increased, at a known rate if the assumptions being tested were correct, and the condenser temperature increased to maintain the constant heat transfer rate against the increasing thermal resistance.

Periodic readings of condenser temperature were taken at the constant cooling water temperatures and flow rate. The steam flow rate remained constant, the pressure ratio over the throttle valve being at all times less than the critical.

Three such tests were made at different shut down leakage rates and the results plotted as increase in overall thermal resistance against air partial pressure calculated from the shut down leakage rates, see Fig.12.

At the higher air partial pressures it was possible to measure the condenser pressure with sufficient accuracy to deduce the air partial pressure from the measured condenser temperature and pressure. These agreed with those calculated from the shut down leakage rate.

Because the experimental points from the three tests lie on a single continuous curve it can be assumed that the steam was virtually air free and that the shut down leakage rate was equal to that at standard test conditions.

It is possible that the characteristics of the uncontrolled leaks could have changed with time i.e. given a different leakage rate during tests to that measured during shut down.

To check this a clean tube was retested about once every three weeks and, by continuous maintenance, the uncontrolled leakage rate was kept below the standard.

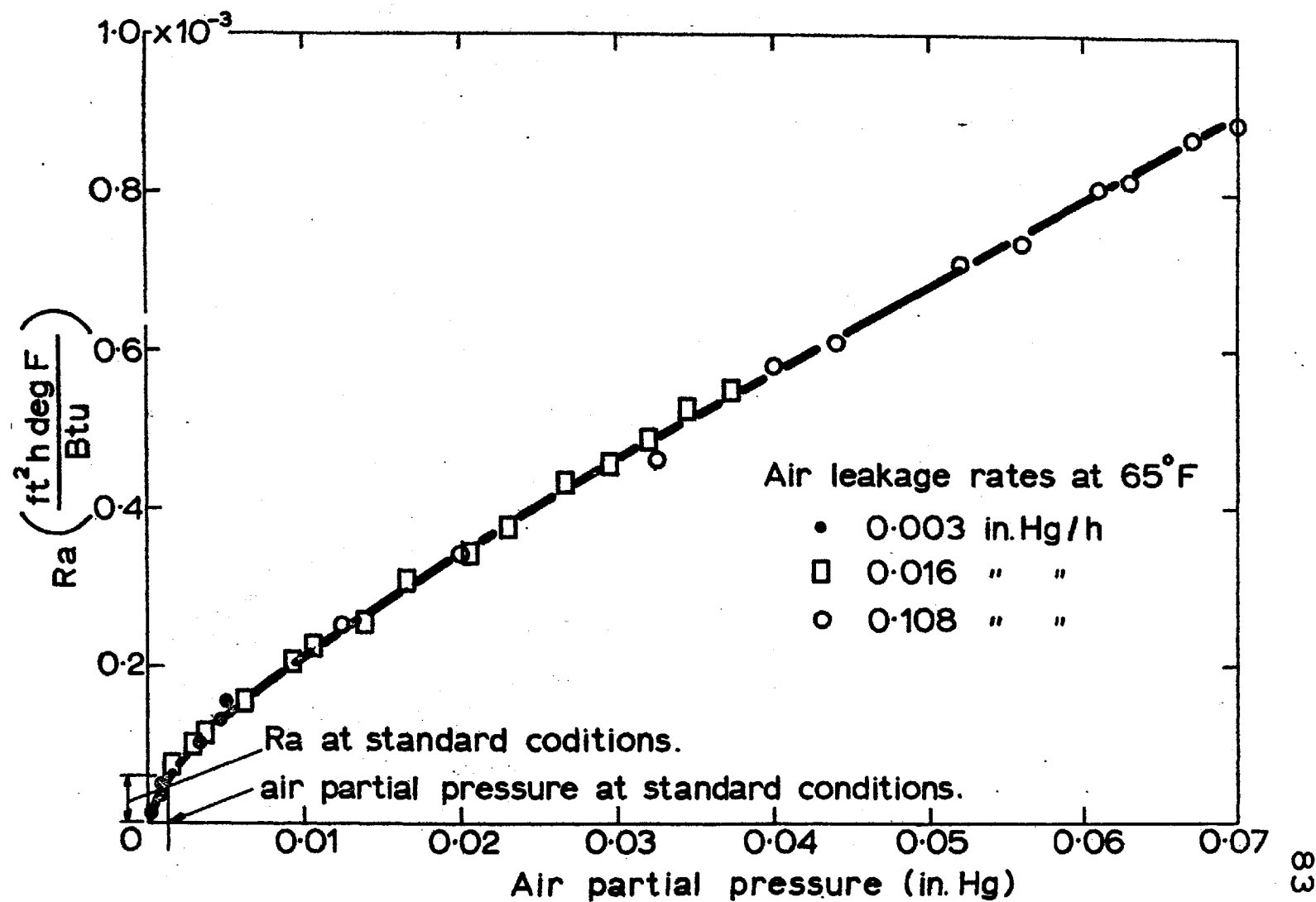


FIG. 12. THERMAL RESISTANCE OF AIR (R_a) v AIR PARTIAL PRESSURE.

A further series of tests were made, again using a clean tube, to obtain a relationship between total air leakage rate and increase in overall thermal resistance at standard condenser temperature and condensation rate. This was done by making standard tests at different settings of the air bleed valve, the corresponding total leakage rate being measured before the test and checked afterwards. The results of these tests are given in Fig. 13, which also shows how the thermal resistance due to the air, at standard conditions, mentioned in Chapter 8, was deduced.

This curve was used to correct resistances measured at leakage rates different from the standard and to check the differences between tests on the same tube made at different leakage rates.

From Figs. 13 and 14 it appears that the air partial pressure at standard conditions is of the order of 0.0015 in Hg.

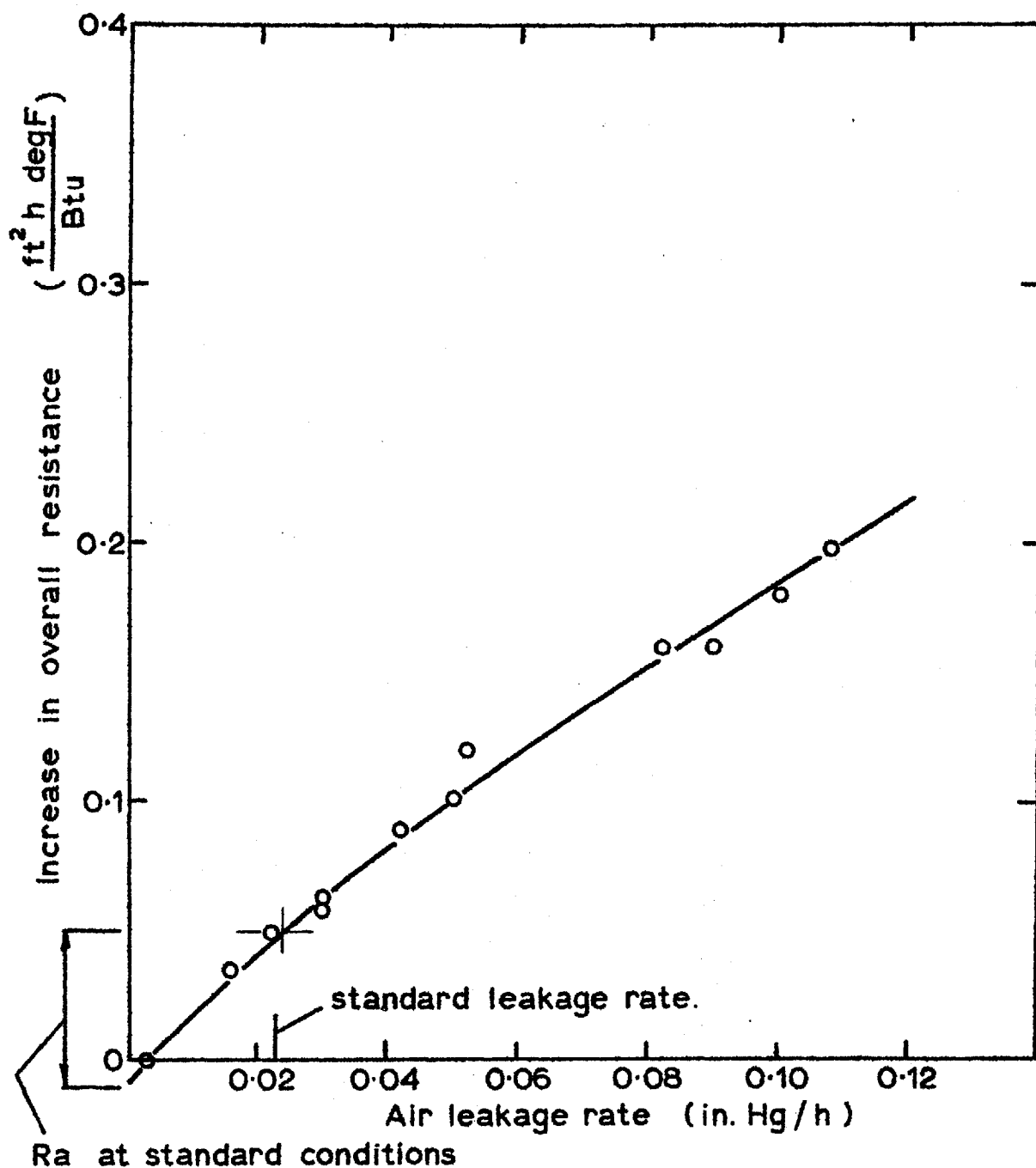


FIG. 13. THERMAL RESISTANCE OF AIR v AIR LEAKAGE RATE.

CHAPTER 10

SELECTION OF CONDENSER TUBES FOR TESTING

There have been several investigations to measure the variation in heat transfer rates and coefficients from place to place inside large steam condensers. The usual method used was to measure the cooling water flow rate and temperatures through certain selected tubes by means of thermocouple and pitot-static probes passing through the walls of the water boxes into the inlet and outlet ends of the tubes. Though it appeared in some cases that the probes used must have disturbed the flow through the selected tubes the differences between the heat transfer rates measured from tube to tube probably gave a reliable indication of the differences during normal operation. Russian investigations have shown variations in heat transfer coefficient from 720 to 80 Btu/ft²h deg F while tests on Naval type condensers by Pamatrade³ gave ratios of local to mean heat transfer rate from 1.24 to 0.7. Other investigations gave similar results. The maximum variation above the mean being generally less than the minimum.

The general pattern of variation was of decreasing heat transfer rate from the steam inlet to the interior of

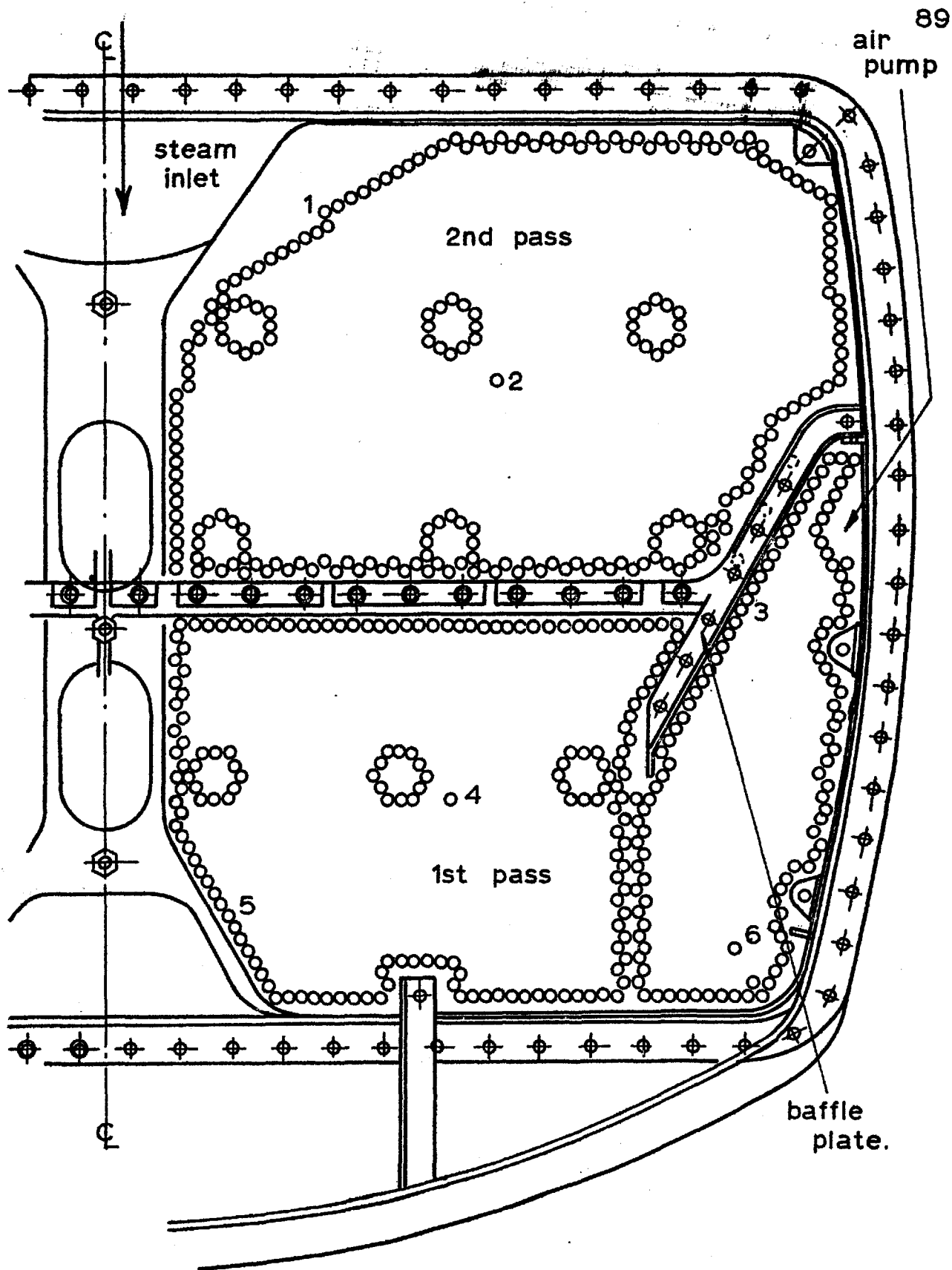
the condenser and from the surface to the interior of the various tube bundles.

These variations with tube position are due, on the steam side, to differences in steam velocity and air concentration. The steam velocity over the tubes near the steam inlet are of the order of 200 ft/s and it is in this region that the highest heat transfer rates and coefficients occur. As the steam penetrates further into the condenser, the velocity decreases, due to condensation and increase in flow area, air partial pressure increases and the heat transfer rates and coefficients decrease. The lowest occur in the air cooler i.e. that portion of the condenser remote from the steam inlet where the air concentration is highest. The connection to the air extraction pump is made at this point. The art of designing steam condensers consists mainly of so arranging the tubes that all the steam flow paths lead eventually to the air extraction point i.e. there are no static accumulations of air in other parts of the condenser.

On the water side, variation in heat transfer rates from tube to tube can be caused by differences in water velocity. In one case reported¹⁰ it was found that the water velocity through some tubes was practically zero whilst in others it was even reversed. The highest water velocity measured was over twice the mean.

This wide variation in heat transfer from tube to tube means that some tubes in a condenser are virtually inoperative and so any increase in their thermal resistance will have little or no effect on the performance on the condenser as a whole. A similar increase in the resistance of tubes where the heat transfer rate is high would have a very considerable effect on the overall performance.

All the tubes tested came from so called "regenerative type" condensers. Fig. 14 gives the general arrangement of the tubes in one half of such a condenser, the other half being symmetrically similar about the condenser's vertical centre line, including the baffle plate shrouding the tubes near the air extraction point. The purpose of this baffle is to extend the steam flow path to, and to increase the air partial pressure at, the inlet to the air extractor. The passage between the two main banks of tubes is intended to ensure that the steam flows into the tube banks over the greater proportion of the outer surface, particularly from below. The "Regenerative Effect" is due to the steam, flowing down the central passage, coming into contact with the surface of the condensate at the bottom of the condenser. This, together with the heat transfer to the condensate, from the steam, as it falls to the bottom of the condenser reduces the under cooling of the condensate i.e. the difference between the saturation



temperature corresponding to the condenser pressure and the condensate exit temperature. In modern condensers this is as low as 5 deg F.

It was originally intended to base the selection of tubes for test on the results of tests on a large number of tubes taken at the same time from a single condenser. Owing to the time taken to develop the apparatus and the experimental method, this part of the programme was abandoned. Tubes were selected on the basis of the variations in performance found in the investigations previously discussed.

The selection of tubes from the S.S. "Pyrrhus" was based on the following considerations. Positions 1 and 2 as representing the highest and the average performance of tubes in the second pass, positions 5 and 4 similarly representing the first pass and position 3, in the air cooler, and 6, an area where a static accumulation of air is possible, representing the regions of low heat transfer rates. The selection of the tubes taken from the S.S. "Isle of Guernsey", Fig. 15, was based on similar considerations.

The three tube batches obtained from the other vessels were drawn from the centre and bottom of the first pass and from the centre of the second pass. These positions were modified as required to allow the tubes to be withdrawn through the inspection doors in the water boxes.

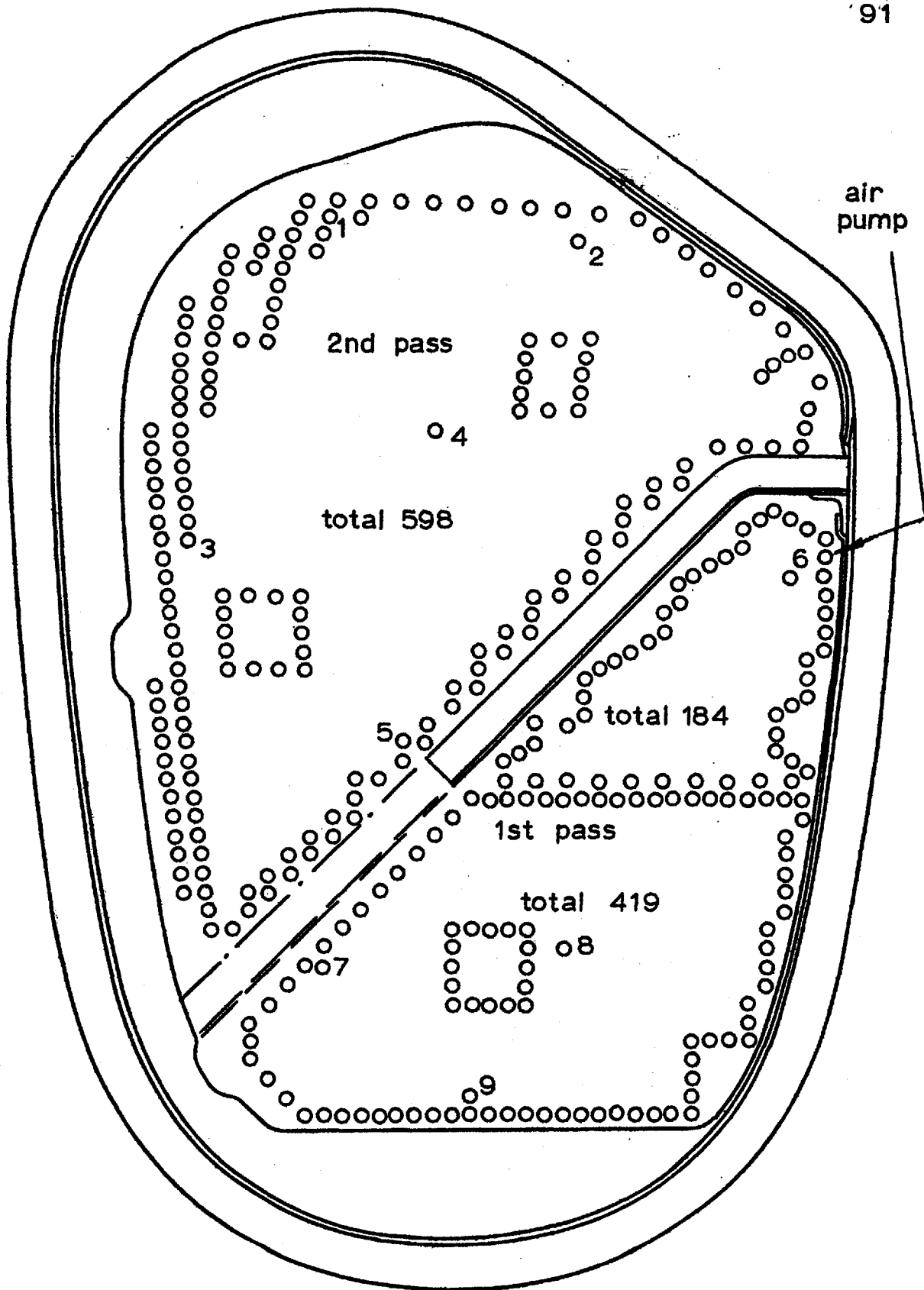


FIG. 15. POSITIONS IN CONDENSER OF S.S. ISLE OF GUERNSEY FROM WHICH SAMPLE TUBES WERE TAKEN.

A questionnaire, see Appendix 3, was sent to each of the companies supplying tubes. In addition they were asked to label the inlet end of each tube to indicate its position in the condenser. These instructions were not always followed and some of the answers to the questions in the questionnaire were subsequently found to be incorrect.

CHAPTER 11

IDENTIFICATION OF TUBES

Tubes are identified first by the initials, or the first two letters of the name of the vessel from which they were obtained. The key to this code together with details of the normal routes on which the vessels were used and the nominal working conditions for their condensers, are given in Table 1.

A number following the identifying letters corresponds to the numbered positions in the condenser from which the tube was removed (see Fig.14 and 15). A new condenser tube received with a batch of used tubes is identified merely by the letters of the vessel concerned.

When more than one batch of tubes was taken from a vessel the position number for the second batch is followed by the letter A and the third batch by B etc.

In the case of the S.S. "Pyrrhus" the fourth batch of tubes was taken half from the forward and half from the after side of the condenser. These positions are identified by (F) and (A) respectively, following the batch letter.

The tubes obtained from the S.S. "Queen Mary" were sufficiently long for two test lengths to be obtained from each tube. When this was done the two sections are numbered (i) and (ii).

When three tube samples only were obtained they were withdrawn as far as possible from the middle and bottom of the first pass and from the middle of the second pass. These general positions are identified by the letters A, B and C respectively. When the positions in the condenser from which a three tube sample was obtained was not known the letters X, Y and Z are used.

CHAPTER 12

TUBE CLEANING TESTS

A series of tests was made to measure the effect of the two most common methods used to clean condenser tubes, i.e. by brushing and by washing with acid.

Other cleaning methods that have been used to clean marine condensers include firing plugs of various types through the tubes with compressed air or high pressure water. Experience has shown that such methods tend to remove only the outer layer of the deposit while at the same time compacting and consolidating that remaining. Such deposits had eventually to be removed by sand blasting or by passing a power driven rotary cutter through the tube. The labour and time required by all such methods, and by brush cleaning, is such that it appears probable that acid cleaning will be the only method used in future.

For the brush cleaning tests a selection of so-called condenser tube cleaning brushes was obtained from a firm of ship repairers. It was stated that brushes of this type were used to clean the condensers of ships as part of the routine overhauls. However, it was stated that acid cleaning was also used.

The brushes had brass, steel or bronze bristles, were 4.5 in long and about 0.65 inches in diameter, i.e. less than the nominal internal diameter of the tubes that they were supposed to clean. Contact between the tube wall and bristle depended on the brush being slightly bent.

Visual inspection of the interiors of several tubes after the passage of these brushes showed that the removal of the deposits was very patchy except when the tube internal diameter was less than the nominal. In one case it was possible to drop a brush through a tube and only scratch the surface of the deposits. Where the deposits had been removed the underlying metal surface was seen to be scratched.

Recent American experience in the cleaning of power station condensers⁴ has shown the superiority of nylon bristle brushes. Tests have shown that these brushes remove more of the deposit than do wire brushes and that they do not scratch the metal surface. Such damage has been held responsible for the onset of corrosion and also for an increase in the rate at which subsequent deposits accumulate. It was decided therefore to base the tube cleaning experiments on such a brush and not to use the wire brushes. The nylon brush used, 6 in long and 1 inch in diameter, was also obtained from the ship repairing firm.

The acid cleaning compounds used were those marketed as Clensol P. and Clensol L.C.P. They are both based on hydrochloric acid and contain additives which retard the acid attack on the metal once the deposit has been dissolved. Both compounds were used but no difference in their cleaning properties was noticed.

An attempt was made to clean a tube by circulating the cleaning solution through the apparatus, but, due to the acid reaction with the galvanizing on the water pipes, the cooling water pump filled with gas. The mist caused by the reaction with the galvanizing on the reservoir tank and the effervescence of the solution leaving the tube also presented a health hazard. The tubes therefore were cleaned internally by plugging one end and filling with cleaning compound. The filling was changed each time the reaction ceased and when no reaction followed a refilling, the tube was assumed to be internally clean. Visual examination of the interior surface was difficult due to the reflection of light from the exposed bright metal surface, but all deposits appeared to have been removed. For external cleaning the tube was immersed in a bath of solution until the whole of the exterior surface had a bright metallic appearance. After cleaning, some tubes appeared to have a non-wetting surface.

The testing sequence was as follows. Standard tests to measure the thermal resistance were made, first with the tube in the as received condition, then after brushing through once in each direction with the nylon brush, again after internal acid cleaning and finally after internal and external acid cleaning. This sequence showed the effect of brushing and gave, by difference, the individual resistances of the internal and external deposits.

The successive reductions in fouling resistance for four tubes tested in this way are given in Table 5. The percentage reduction in resistance is based on the resistance of the uncleaned tube over that of the calculated resistance of a new tube. For tubes C.M.Z. and L.W.Z. acid cleaning did not reduce the resistance to the calculated new tube value probably because the inner tube surface had been roughened. In the case of tubes Py 6C(A) and (F) the resistance was reduced to below the calculated new tube value, the sum of the successive reductions therefore total more than 100 per cent.

The effect of brushing was found to vary considerably, the thermal resistance being reduced by between 2.65 and $5.5 \times 10^{-4} \text{ ft}^2 \text{ h deg F/Btu}$. This is equivalent to the increase in resistance during a tube life of between 20 and 65 months and thus gives some guidance as to suitable time

TABLE 5
TUBE-CLEANING TESTS

Tube No.	Tube condition	$R_{op} \times 10^4$ $\frac{ft^2 h \text{ deg } F}{Btu}$	h_{op} $\frac{Btu}{ft^2 h \text{ deg } F}$	f at $Re=3.3 \times 10^4$	ΔR_{op} $\times 10^4$	Percentage re- duction in fouling resistance
C. M. Z	283 months service	19.85	505	0.0098	-	-
	Brushed through	14.4	696	0.0068	5.5	71
	Acid-cleaned internally	12.85	780	0.0064	1.5	19
	Acid-cleaned internally and externally	12.45	805	0.0065	0.4	5
L. W. Z	54 months service	18.0	555	0.00835	-	-
	Brushed through	15.35	650	0.00725	2.65	43
	Acid-cleaned internally	13.15	760	0.0070	2.2	36
	Acid-cleaned internally and externally	12.25	815	0.0070	0.9	15
Py 6C(A)	81 months service	17.6	570	0.0072	-	-
	Brushed through	14.15	705	0.00705	3.35	60
	Acid-cleaned internally	12.3	815	0.0068	1.85	33
	Acid-cleaned internally and externally	11.5	870	0.0068	0.8	14
Py 6C(F)	81 months service	18.4	545	0.0071	-	-
	Brushed through	13.3	750	0.0068	5.1	80
	Acid-cleaned internally	12.4	807	0.0067	0.9	14
	Acid-cleaned internally and externally	11.55	865	0.0067	0.85	13

intervals for this method of cleaning. The resistance of the external deposits appears to be small enough to make it unnecessary to undertake in situ external cleaning.

When retesting tube C.M.Z. after brushing it was found that the fouling resistance decreased as the water velocity increased during the experiment and that it was not reduced to the given value until the cooling water had been passed through the tube at about 12 ft/s for 10 minutes. Therefore water flushing, or lancing, may be necessary to remove the deposits loosened by brushing. As this effect was not noticed during the tests on the other tubes, the lower water velocities used at the beginning of the experiment must have been sufficient in these cases to remove the debris. Tube C.M.Z. was much older than the other three tubes so it is possible that the deposits become more difficult to remove with time. Fig. 16 shows the Wilson Line Plots obtained from the successive tests on tube C.M.Z. and the reduction in fouling resistance as the increasing water velocity removed the deposits loosened by brushing. Fig. 17 shows the effect of brushing and acid cleaning on the friction factor.

The results of these tests are discussed further in the next chapter.

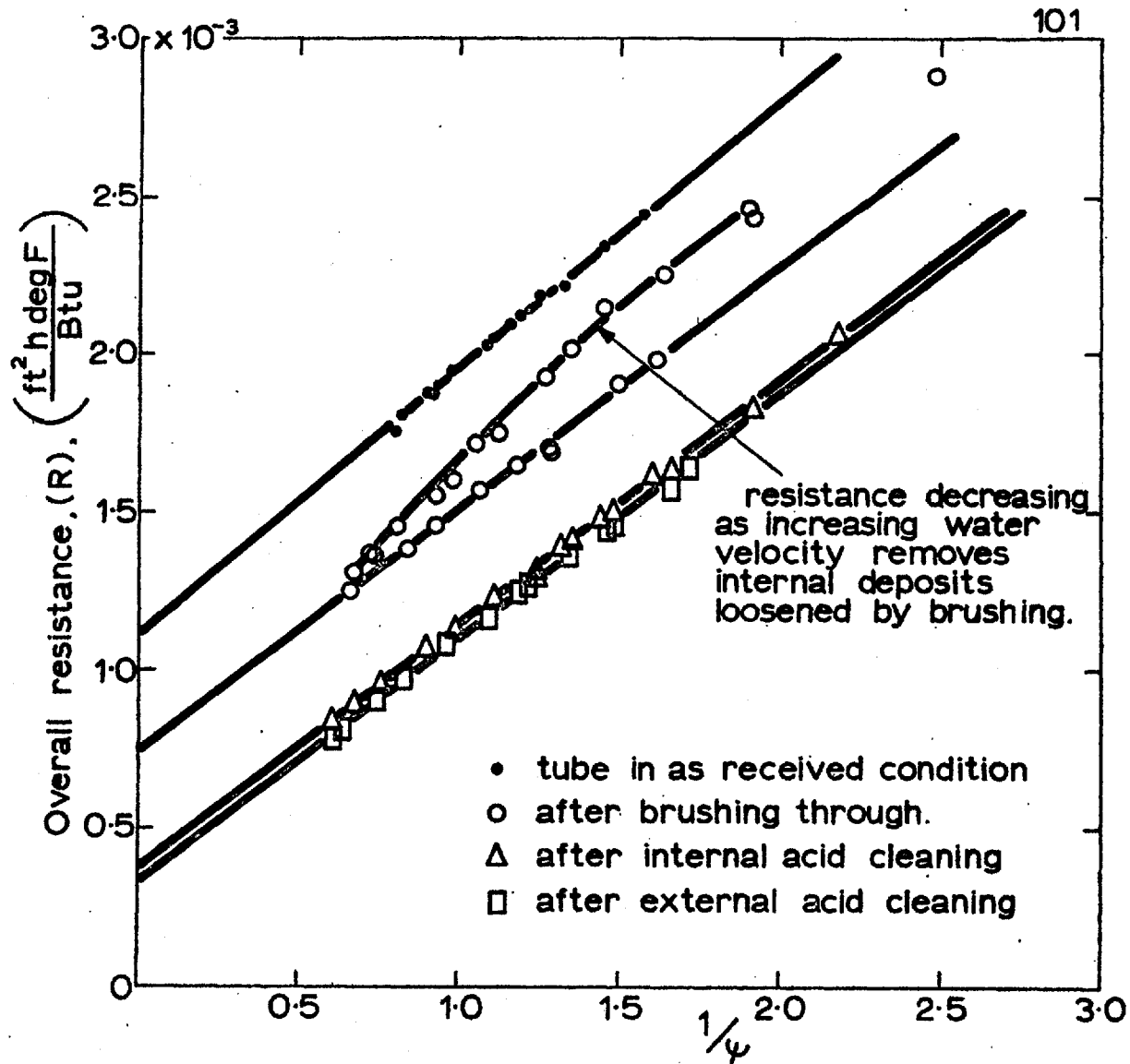


FIG. 16. WILSON LINE PLOT. TUBE CLEANING TESTS. TUBE C.M.Z.

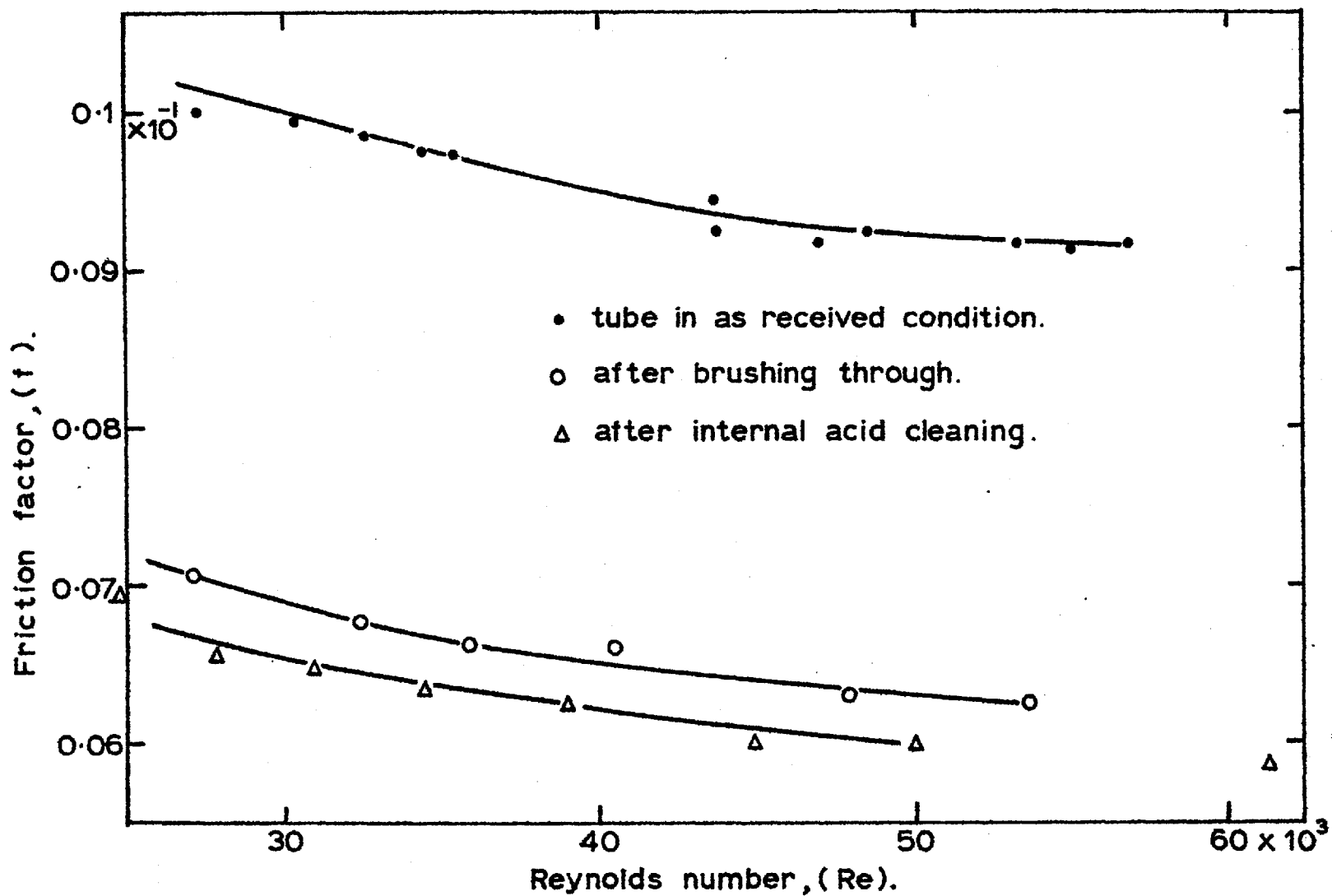


FIG.17. TUBE CLEANING TESTS. TUBE C.M.Z.
 FRICTION FACTOR v REYNOLDS NUMBER.

TABLE 6
TEST RESULTS

Tube No.	Tube life (months)	$Rop \times 10^4$ $\frac{ft^2 h \text{ deg } F}{Btu}$	hop $\frac{Btu}{ft^2 h \text{ deg } F}$	f at $R=3.3 \times 10^4$	$Rf \times 10^4$	Remarks
Py 1	8.4	14.35	697	0.00685	2.3	Calculated Rop of a new tube $= 12.05 \times 10^{-4}$ $\frac{ft^2 h \text{ deg } F}{Btu}$ hop (new tube) $= 830 \frac{Btu}{ft^2 h \text{ deg } F}$
2		14.9	670	0.0073	2.85	
3		13.6	735	0.0071	1.55	
4		14.1	710	0.0071	2.05	
5		14.0	715	0.0071	1.95	
6		14.2	705	0.0071	2.15	
Average		14.2	705	0.0071	2.15	
Py 1A	28	16.55	605	0.0068	4.5	
2A		16.5	605	0.0070	4.45	
3A		15.1	660	0.00695	3.05	
4A		15.95	625	0.0066	3.9	
5A		16.0	625	0.0069	3.95	
6A		14.95	670	0.0068	2.9	
Average		15.85	630	0.00685	3.8	
Py 1B	36	17.6	570	0.0069	5.55	
2B		17.8	560	0.0075	5.75	
3B		16.9	590	0.0072	4.85	
4B		16.25	615	0.0068	4.2	
5B		15.85	630	0.00695	3.8	
6B		15.4	650	0.00695	3.35	
Average		16.65	600	0.00705	4.6	

TABLE 6 cont.

Tube No.	Tube life (months)	Rep x 10 ⁴ $\frac{\text{ft}^2 \text{h deg F}}{\text{Btu}}$	hop $\frac{\text{Btu}}{\text{ft}^2 \text{h deg F}}$	f at R=3.3 x 10 ⁴	Rfx10 ⁴	Remarks
Py 2C(A)	57	17.0	587	0.00715	4.95	Calculated Rop of a new tube $= 12.05 \times 10^{-4} \frac{\text{ft}^2 \text{h deg F}}{\text{Btu}}$ hop (new tube) $= 830 \frac{\text{Btu}}{\text{ft}^2 \text{h deg F}}$
2C(F)		19.0	525	0.00715	6.95	
4C(A)		17.35	575	0.00705	5.3	
4C(F)		18.05	555	0.0069	6.0	
6C(A)		17.6	568	0.0072	5.55	
6C(F)		18.4	545	0.0071	6.35	
Average		17.9	560	0.0071	5.85	
I. G. 1	19.5	15.1	662	0.00645	3.05	Calculated Rop of a new tube $= 12.05 \times 10^{-4} \frac{\text{ft}^2 \text{h deg F}}{\text{Btu}}$ hop (new tube) $= 830 \frac{\text{Btu}}{\text{ft}^2 \text{h deg F}}$
2		15.45	648	0.0067	3.4	
3		16.15	620	0.00675	4.1	
4		16.6	602	0.00695	4.55	
5		15.7	637	0.00685	3.65	
6		14.2	705	0.0068	2.15	
7		14.65	682	0.00705	2.6	
8		14.65	682	0.0068	2.6	
9		17.95	557	0.00815	5.9	
Average		15.6	640	0.00695	3.55	
I. G. 4A	36	16.95	590	0.0072	4.9	
8A		18.25	550	0.00805	6.2	
9A		19.10	525	0.00855	7.05	
Average		18.1	555	0.00790	6.05	

TABLE 6 cont.

Tube No.	Tube life (months)	Rop $\times 10^4$ $\frac{\text{ft}^2 \text{ h deg F}}{\text{Btu}}$	hop $\frac{\text{Btu}}{\text{ft}^2 \text{ h deg F}}$	f at $R=3.3 \times 10^4$	Rf $\times 10^4$	Remarks
L.W.A.	2	12.35	810	0.0070	0.3	Calculated Rop and hop for new tube as for Isle of Guernsey
B.		12.8	780	0.0069	0.75	
C.		13.05	765	0.0075	1.0	
Average		12.75	785	0.0070	0.7	
L.W.X	16	16.1	620	0.0072	4.05	
Y		14.65	684	0.0075	2.6	
Z		18.0	555	0.00835	5.95	
Average		16.25	615	0.0077	4.2	
Q.M.A. (i)	16	16.95	590	0.0070	2.7	Calculated Rop of a new tube $=14.25 \times 10^{-4}$ $\frac{\text{ft}^2 \text{ h deg F}}{\text{Btu}}$ hop (new tube) $=700 \frac{\text{Btu}}{\text{ft}^2 \text{ h deg F}}$
A. (ii)		17.25	580	0.00695	3.0	
B. (i)		17.5	570	0.0070	3.25	
B. (ii)		17.5	570	0.0068	3.25	
C.		16.9	590	0.0070	2.65	
Average		17.2	580	0.00695	2.95	
Do A	13	16.25	615	0.0069	4.2	Calculated Rop of a new tube $=12.05 \times 10^{-4}$ $\frac{\text{ft}^2 \text{ h deg F}}{\text{Btu}}$ hop (new tube) $=830 \frac{\text{Btu}}{\text{ft}^2 \text{ h deg F}}$
B		16.00	625	0.00715	3.95	
C		15.35	650	0.0071	3.3	
Average		15.85	630	0.00705	3.8	

TABLE 6 cont.

Tube No.	Tube life (months)	$Rop \times 10^4$ $\frac{ft^2 h \text{ deg F}}{Btu}$	hop $\frac{Btu}{ft^2 h \text{ deg F}}$	f at $R=3.3 \times 10^4$	$Rf \times 10^4$	Remarks
Pa X	36	17.95	557	0.0069	5.9	Calculated Rop of a new tube $= 12.05 \times 10^{-4}$ $\frac{ft^2 h \text{ deg F}}{Btu}$ $= 830 \frac{Btu}{ft^2 h \text{ deg F}}$
Y		17.6	570	0.0075	5.55	
Z		16.7	600	0.0070	4.65	
Average		17.4	575	0.0071	5.35	
Ve A	73	14.15	705	0.00815	2.1	
B		14.6	685	0.00825	2.55	
C		16.85	595	0.0079	4.8	
Average		15.2	657	0.0081	3.15	
C.M.X	167	18.35	545	0.0075	6.3	
Y		20.55	486	0.0076	8.5	
Z		19.85	505	0.0098	7.8	
Average		19.6	510	0.0083	7.55	
St A	33	19.75	506	0.0075	5.5	Calculated Rop of a new tube $= 14.25 \times 10^{-4}$ $\frac{ft^2 h \text{ deg F}}{Btu}$ hop (new tube) $= 700 \frac{Btu}{ft^2 h \text{ deg F}}$
B		19.75	506	0.0076	5.5	
C		20.6	486	0.0079	6.35	
Average		20.05	499	0.00765	5.8	

TABLE 6 cont.

107.

Tube No.	Tube life (months)	$Rop \times 10^4$ $\frac{ft^2 h \text{ deg } F}{Btu}$	hop $\frac{Btu}{ft^2 h \text{ deg } F}$	f at $R=3.3 \times 10^4$	$Rf \times 10^4$	Remarks
Ha A	25	18.0	556	0.0075	5.95	Calculated Rop of a new tube = $12.05 \frac{ft^2 h \text{ deg } F}{Btu}$ hop (new tube) = $830 \frac{Btu}{ft^2 h \text{ deg } F}$.
B		17.25	580	0.0071	5.2	
C		19.5	513	0.0088	7.45	
Average		18.25	546	0.0078	6.2	
Hi A	16	17.5	570	0.00725	5.45	
B		15.85	630	0.00725	3.8	
C		17.85	560	0.0074	5.8	
Average		17.05	586	0.0073	5.0	
Py 1D	70	17.85	560	0.0075	5.8	(Excluding tubes 3D and 6D)
2D		18.2	550	0.00805	6.15	
3D		16.0	625	0.0077	3.95	
4D		19.35	517	0.0075	7.3	
5D		18.4	543	0.0073	6.35	
6D		15.5	645	0.00765	3.45	
Average		17.55 (18.45)	570 (540)	0.0076 (0.0076)	5.5 (6.4)	
CM _L B	83	20.95	477	0.0077	8.9	
CM _L C		18.75	533	0.00715	6.7	
Average		19.85	505	0.00745	7.8	

CHAPTER 13

DISCUSSION OF RESULTS

THERMAL RESISTANCE OF DEPOSITS

The measured increases in the thermal resistance at the operating point (fouling resistances) of all the tubes tested are given in Table 6 together with the nominal friction factors, f , at $Re = 3.3 \times 10^4$, the overall heat transfer coefficients at the operating point, h_{op} , and those, calculated for new tubes, used as standards of comparison.

The average increase in resistance at the operating point for each batch of tubes is plotted against tube life in Fig.18 and the corresponding overall heat transfer coefficient, expressed as a percentage of that calculated for a new tube, against tube life in Fig.19. Tube life is the estimated time that the tube had actually been in use and was obtained by multiplying the age of the tube by a factor of 0.9 in the case of oil tankers and passenger liners, 0.7 for cargo vessels and 0.3 for the cross channel boats. These factors were estimated from the itineraries of the vessel provided by their owners. The average increase in resistance is the arithmetic mean of the increases of all the tubes in a batch except for batch D from the S.S. "Pyrrhus". The resistance of two of the tubes,

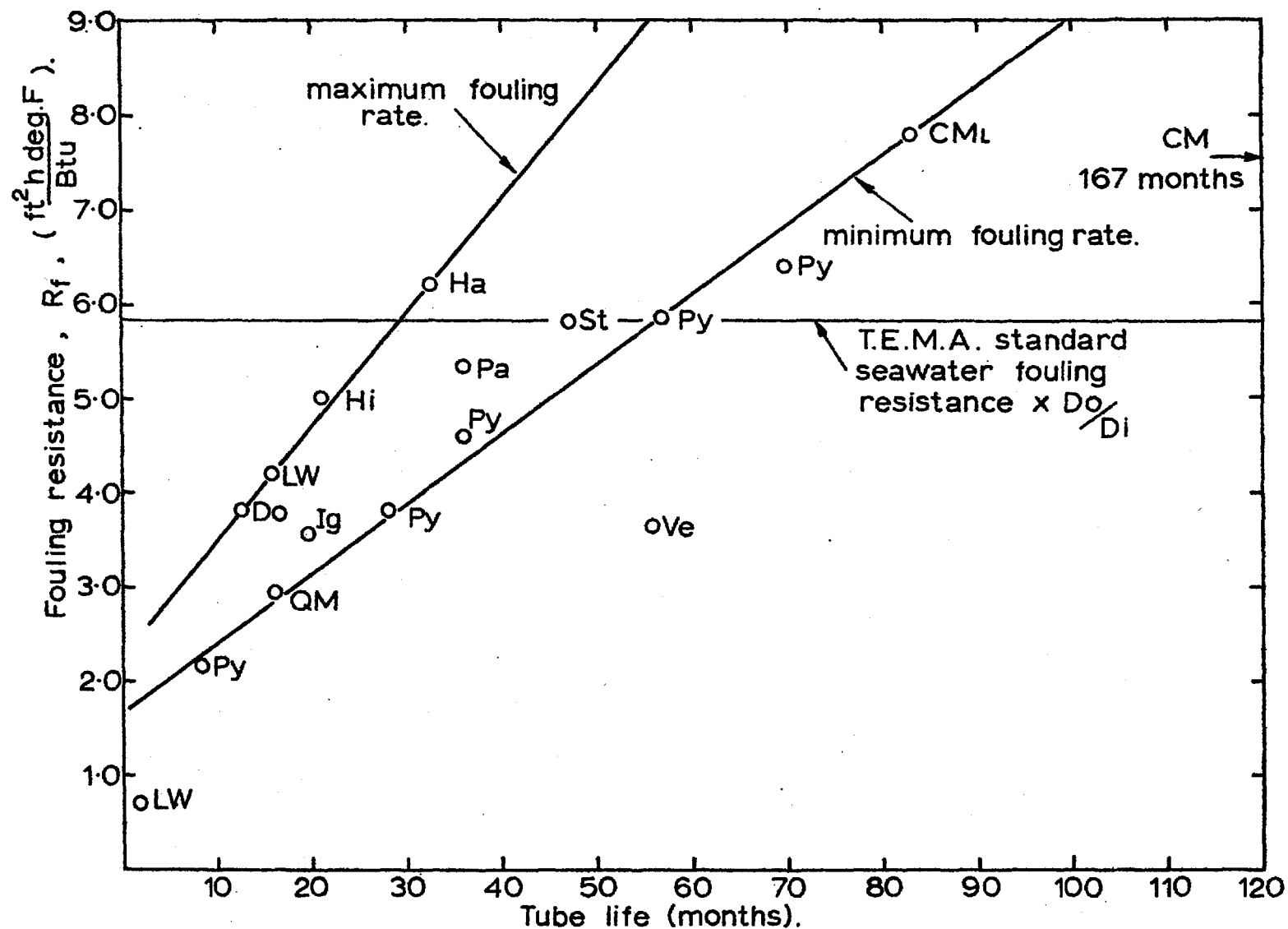


FIG.18. FOULING RESISTANCE AT THE OPERATING POINT v TUBE LIFE.

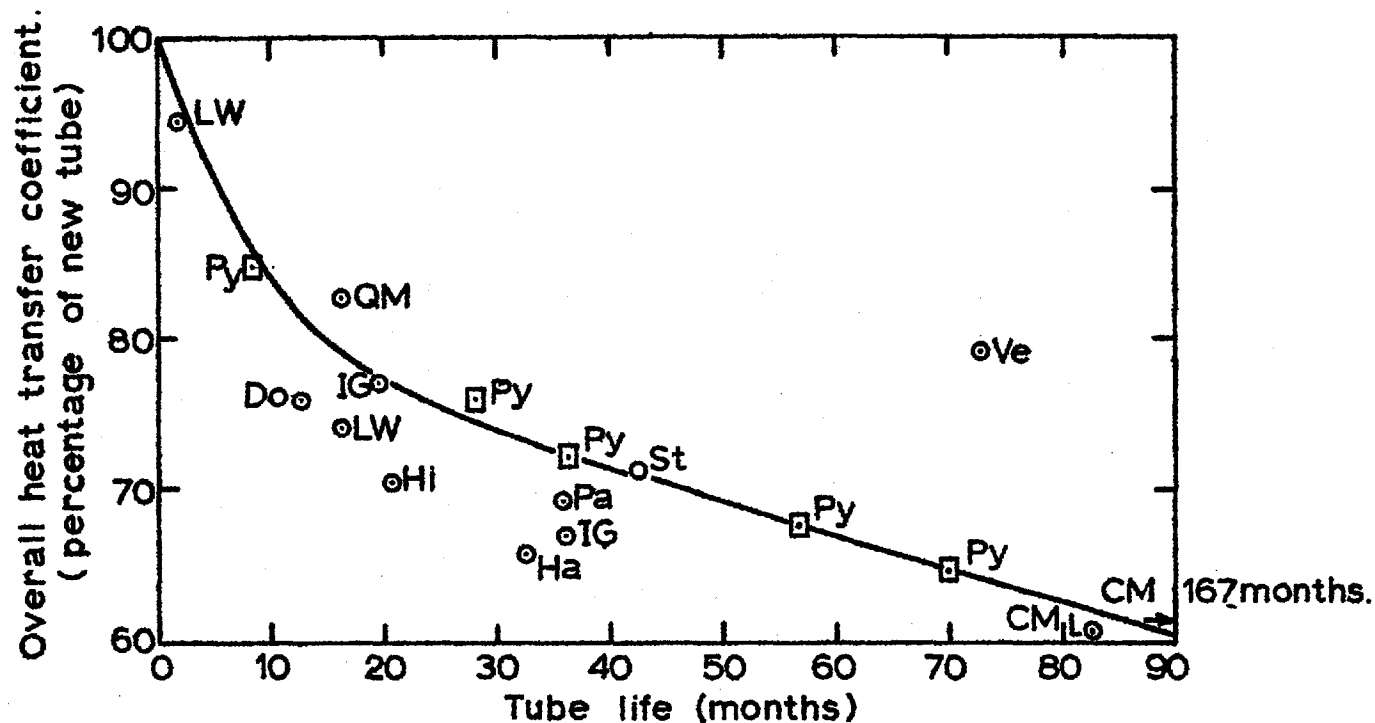


FIG. 19. OVERALL HEAT-TRANSFER COEFFICIENT (PERCENTAGE OF NEW TUBE AT OPERATING POINT) v TUBE LIFE.

those from positions 3 and 6 were so low that it appeared probable that tubes that had replaced earlier samples, had been selected in error. As sample tubes were withdrawn through the inspection doors in the water boxes the number of tubes easily accessible was small and the chance of selecting a tube from the same position on two occasions high.

The curves in Fig. 18 are drawn to estimate the highest and lowest fouling rates to be expected. The intersections of these curves with the T.E.M.A. seawater fouling allowance line shows the allowance to allow for the increase in resistance during between 30 and 55 months use and that, during this period, the overall heat transfer coefficient falls to about two thirds of its new tube value.

The fact that the tubes from the S.S. "Pyrrhus" show a progressive increase in resistance with tube life and that the other results with two exceptions conform fairly well with this trend makes it extremely doubtful that the stated cleaning schedules were in fact complied with. The tube cleaning tests showed that brushing decreased the resistance by between 5.1 and 2.65×10^{-4} ft²h deg F/Btu while the difference between the maximum and minimum increases in resistance was only about 2.0×10^{-4} ft² h deg F/Btu after a life of three years. It is unlikely that this could be accounted for by the differences between the effect of the wire

cleaning brushes and the nylon brush used in the tube cleaning experiments.

Brushing through with the standard wire brushes would produce a considerable decrease in resistance. If this had been done up to four times a year, as stated, the resistances should have varied between a pre-cleaning maximum and a minimum, depending on the diameters of the brushes i.e. the amount of deposit left after brushing.

Conversations with ships engineers tended to confirm that condensers were cleaned only when the fall in vacuum became appreciable. As will be shown in Chapter 14 the decrease in vacuum caused by fouling in modern condensers is not likely to be large and therefore, to the engine room staff there seems no need to disturb a condenser that appears to be operating satisfactorily, even if the calendar demands it.

The curves of Figs. 18 and 19 may therefore be assumed to show the change in the resistance of condenser tubes without periodic cleaning.

The two batches of tubes that did not conform to the general trend were those from the S.S. "Velutina" and the S.S. "Clan MacLachlan". The "Velutina" tubes were nearly seven years old when they were withdrawn, so it is probable that the vessel had undergone a major refit and

overhaul during this time. It is the practice of at least some of the firms specializing in this work to remove the condenser tubes during a major overhaul and clean them in an acid bath. An alternative explanation is, of course, that the tubes had been cleaned on one or more occasions by wire brushing or possibly prior to despatch to the laboratory for testing.

The S.S. "Clan MacLachlan" had not been under the owner's control during the war and they had no record of the overhauls that it must have had during that period. The increase in the resistance of both these sets of tubes probably occurred over a shorter period than the given age of the tubes.

In plotting the point for the S.S. "Hadra" the claim that these tubes had been cleaned 18 months before withdrawal was, in view of the general uncertainty as to the accuracy of the reported cleaning programmes, ignored. Despite this the rate of increase of resistance was among the highest measured.

A practical interpretation of these results would be that during the first ten months of a tube's life the thermal resistance increases by between 2.4×10^{-4} and 3.5×10^{-4} $\text{ft}^2 \text{ h deg F/Btu}$ and thereafter increases at a rate of $0.074 \times 10^{-4} \text{ ft}^2 \text{ h deg F/Btu}$ per month for the lower initial increase and $0.12 \times 10^{-4} \text{ ft}^2 \text{ h F/Btu}$ per month of tube life for the higher. The higher rate is only justified by the experimental data up to a total life of 3 years and the lower up to 5 or possibly 7 years. There appears to be no correlation between

the type, duty or route of a vessel and the rate of increase of resistance.

The initial increase in resistance depends to some extent on the standard of comparison used. If a less optimistic standard than the calculated new tube performance used here is adopted, then the initial increase will be less. If the highest new tube resistance were used, i. e. the Py tube in Table 4, the initial increase would be 0.2×10^{-4} ft² h deg F/Btu less. The higher initial increases in resistance may be due to the condensers being filled with polluted water while the ships were in the fitting-out basin and during the preliminary steaming trials. It is also possible that the deposits accumulate more rapidly on a clean metal surface than they do when a continuous layer of deposits has been formed.

It is unlikely that further laboratory tests of this type could provide the designer with closer estimates of the increase in thermal resistance of condenser tubes due to fouling.

More precise information could be obtained by fitting a new ship's condenser with instruments to measure the performance of as many individual tubes as practicable. A continuous record of the deterioration of the selected tubes and of the condenser as a whole would then provide more useful design data. In particular such a record would show whether the

deposits accumulated at a uniform rate or only under certain conditions, such as in inland waters, in passages through the Red Sea etc. With technically competent engine room staff always on duty such data could be obtained relatively easily.

DISTRIBUTION OF FOULING

Some measurements were made to determine the distribution of the internal deposits along the length of condenser tubes. Sim²² states in his book that, in a new tube, the deposits first start to build up at the outlet end and that it took some time for the layer to become continuous up to the inlet. Had this been so then the fouling resistance would have varied from a maximum at the outlet to a minimum at inlet. It would have been difficult from the measurement of the resistance of only part of a tube to decide what would be the increase in the resistance of the tube as a whole. However the examination of the condenser tubes from the H.M.S. Cockade² seemed to show that the deposits to which Sims referred were those laid down during the turbine warming period as already discussed. In the Cockade investigation, pieces of the internal deposits were removed by tapping the tube with a hammer and measuring their thickness with a microscope. The variation in thickness was only investigated over a 6 in length at inlet and outlet and "at the centre of the tube". The average thickness measured for

three cupro-nickel tubes, 5 years old, was 0.0046 in at inlet and 0.0048 in at outlet and 0.0029 in at the centre. The difference between the least and greatest thickness was about 0.002 in at inlet and outlet and 0.0012 in at the centre. The chemical analysis of the deposits removed during these tests is given in Appendix 5. An attempt was made to use this method to find the deposit thickness on some of the tubes tested. Tapping the tube with a hammer did dislodge some of the deposits, but in all cases when the tube was cut in half, longitudinally, for visual examination, it was found that part of the deposit remained attached to the tube wall. Some of the pieces of material detached also separated into two layers.

In an attempt to measure the scale thickness more accurately, sections of condenser tube were filled with a cold setting resin and, after cutting and polishing, examined under a microscope. It was found that the cutting and polishing processes formed a burr on the inner edge of the tube. This made it impossible to measure the deposit thickness with any accuracy. Acid etching to remove the metal burr was tried but the acid rapidly dissolved the deposit and then attacked the tube wall below the burr. This method was not developed further because a determination of the actual thickness of the deposits was not required. Such measurements that were made, together with visual examination, showed that the

deposit thickness varied in a random fashion along the length of a tube and around the circumference.

The distribution of the deposits, by weight, along a tube was measured by cutting sections, approximately 6 in long, out of some used tubes at intervals of 12 in. The sections were measured and weighed and then cleaned, first externally and then internally, in a bath containing the same cleaning solution as was used in the tube cleaning tests. The results of two such tests are given in Table 7. Each of the cleaning stages was continued until the weight of each of the sections became sensibly constant. The maximum and minimum tube wall thickness and the external diameter were also measured. The external diameter was found to be constant but the tube wall thickness and internal diameter varied by an amount within the limits given in the manufacturers specification. The inlet and outlet 6 in of the tubes were excluded from these tests because they contained traces of anti-fouling paint, which had probably been used to protect the sea water side of the tube plates, and possibly of the lubricant used when rolling the tubes into the plate. The presence of such substances in the Cockade tubes would explain the thicker deposits found at the ends of the tubes.

These weighing tests show that the distribution of the deposits, by weight, along the tubes was sensibly uniform

TABLE 7
DISTRIBUTION OF DEPOSITS. WEIGHING TESTS

S S Dorset 12 months service					S S Papanui. 51 months service					
Section 6 in.	Weight of deposit gm		Tube-wall thickness, in.		External diameter, in.	Weight of depositions, gm		Tube-wall thickness, in.		External diameter, in.
	Internal	External	Maximum	Minimum		Internal	External	Maximum	Minimum	
1	0.43	0.17	0.053	0.046	0.748	0.57	0.15	0.049	0.046	0.746
2	0.36	0.11				0.63	0.16	0.043		
3	0.42	0.14	0.052	0.045		0.57	0.15			
4	0.46	0.12				0.58	0.15			
5	0.50	0.13		0.59		0.20	0.048	0.045		
6	0.48	0.15		0.61		0.20				
7	0.48	0.17		0.044		0.59	0.19	0.47	0.046	
8	0.47	0.11	0.60			0.17				
9	0.44	0.13	0.051	0.58		0.16				

Provided that the deposits were homogeneous then the ir thermal resistance would also be constant along a tube.

As a further check the resistances of two sections of two tubes from the S.S. Queen Mary were measured. These tubes had only been in use for 16 months so if Sims observations were correct the outlet sections should have had a higher resistance than the inlet sections. The results of one of these tests are given in Fig. 20 which shows that the resistances of the two sections were equal. The resistances of the two sections of the other tube, Q.M.A. differed by about $0.3 \times 10^{-4} \text{ ft}^2 \text{ hr deg F/Btu}$ which is within the limits of experimental error. The tubes from the S.S. Queen Mary were the only ones supplied that were long enough to provide two test lengths.

Although there appeared to be no variation in resistance along a given tube there was considerable variation from tube to tube. This variation for the tubes from the Pyrrhus and Isle of Guernsey is illustrated in Fig. 21 in which the resistances of the individual tubes have been plotted against tube life. In general the increase in resistance is greater for tubes from the second pass of the condenser, though for batch D from the Pyrrhus this order is reversed. This difference is not confirmed by all the results from the three tube samples from other vessels. The first pass tubes from the

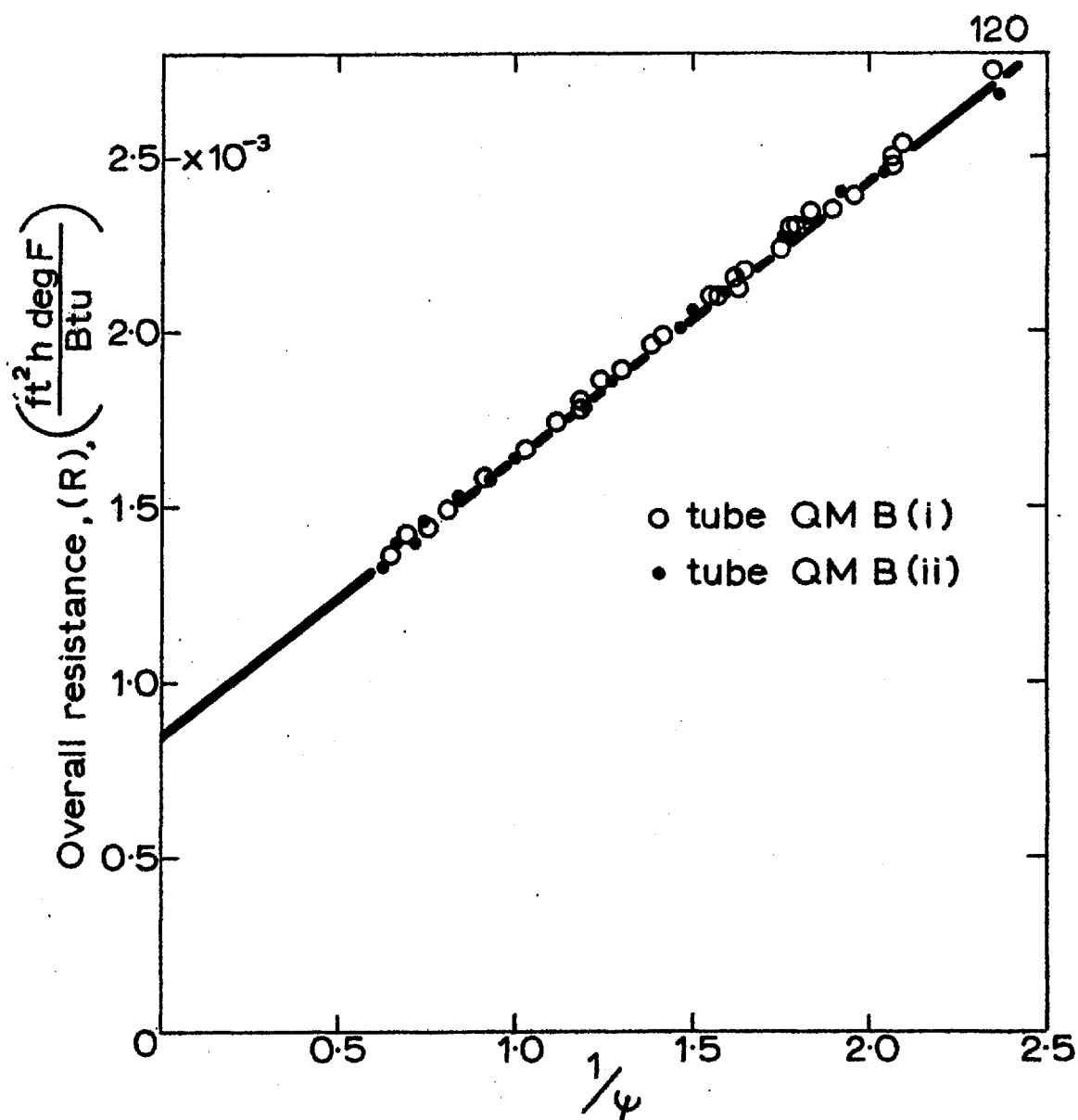


FIG. 20. WILSON LINE PLOT. DISTRIBUTION OF DEPOSITS. COMPARISON OF THE RESISTANCES OF TWO SECTIONS OF A TUBE FROM THE S.S. QUEEN MARY.

Stratheden, Hadra, Hindsia and Velutina had higher resistances than those from the second passes.

From this data it is only possible to make tentative suggestions as to the variables which may affect the rate of increase of resistance and to their relative importance.

Assuming that the proportion of depositable material per unit mass of cooling water is the same for each tube, and that the important variables are the water temperature and velocity, then a possible explanation of the different fouling rates between tubes is as follows. The difference between the two passes is due to the higher water temperature in the second pass, while the differences between tubes in the same pass is due to differences in water velocities. The analysis of the increase in thermal resistance in evaporators by McCabe referred to in Chapter 2 was based on the assumption that the only important variable was the total heat transferred through the tube surface up to the time considered. In the case of the water side of a condenser it is improbable that this variable is important except in so far as a high heat transfer rate means a higher water temperature. If the deposits accumulate more rapidly at higher water temperatures then for tubes where the rate of accumulation is high the water temperatures will decrease more rapidly than in tubes where the rate is lower and differences between the resistances of

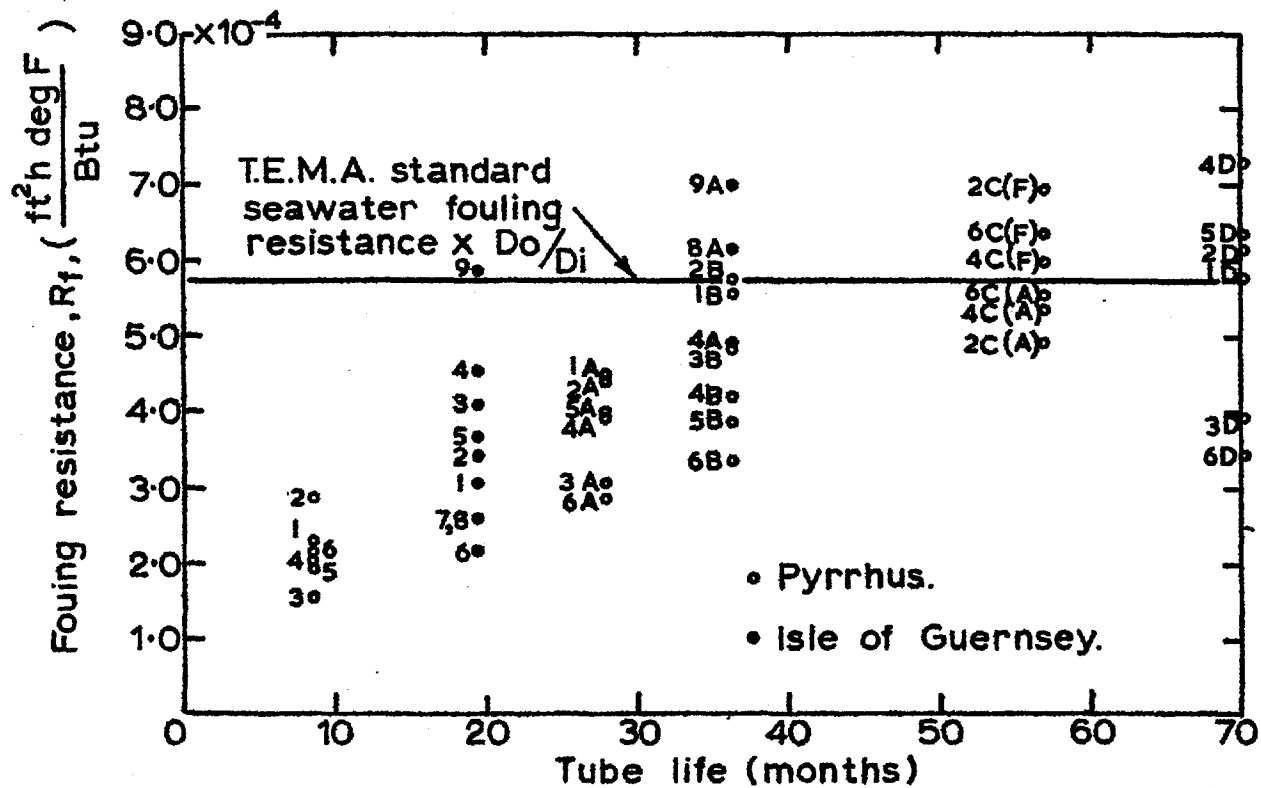


FIG. 21. FOULING RESISTANCES OF INDIVIDUAL TUBES FROM S.S. PYRRHUS AND S.S. ISLE OF GUERNSEY.

tubes in different parts of the condenser should decrease with time. The Pyrrhus and Isle of Guernsey results show this tendency.

The results that do not conform to these assumptions i.e. the high first pass resistances already mentioned, and those of the position 9 tubes from the Isle of Guernsey, could have been due to mud and silt etc. entering the condenser and, due to sedimentation in the inlet water box, being deposited preferentially in the lower tubes of the first pass. This possibility of sudden random increases in the resistance of all, or some of, the tubes makes all attempts to explain these results hypothetical.

A clearer picture could only be obtained from a programme of continuous observation as suggested previously.

FRICION FACTORS

The Fanning friction factor, at $Re = 3.3 \times 10^4$, was calculated for all the tubes tested from the measured cooling water mass flow rates and pressure drops. In the calculation the nominal internal diameter of the tube was used. In Figs. 22 and 23 the average friction factors for each batch of tubes are plotted against tube life and against average fouling resistance. These show in general the expected increase in friction factors with tube life and the

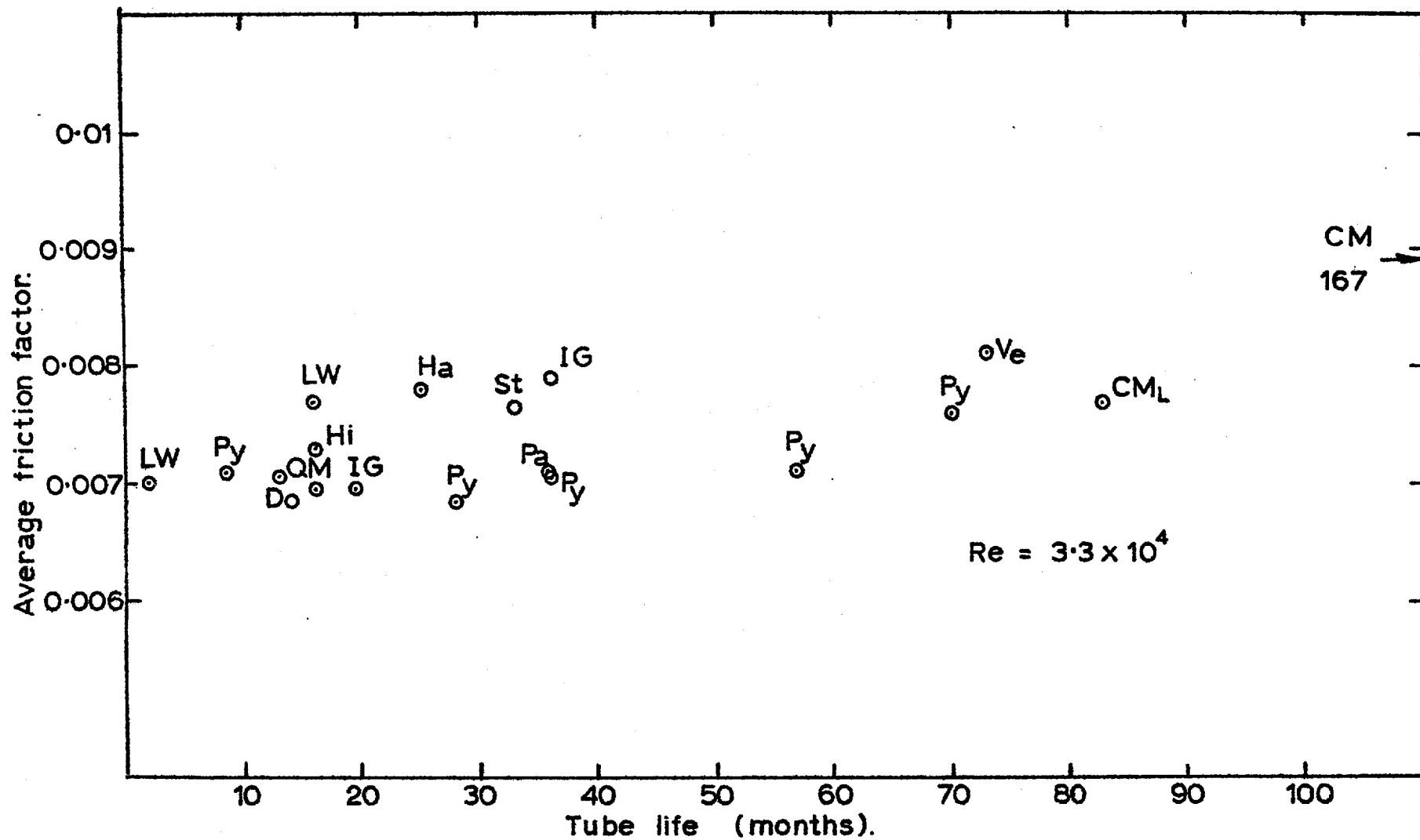


FIG. 22. AVERAGE FRICTION FACTOR v TUBE LIFE.

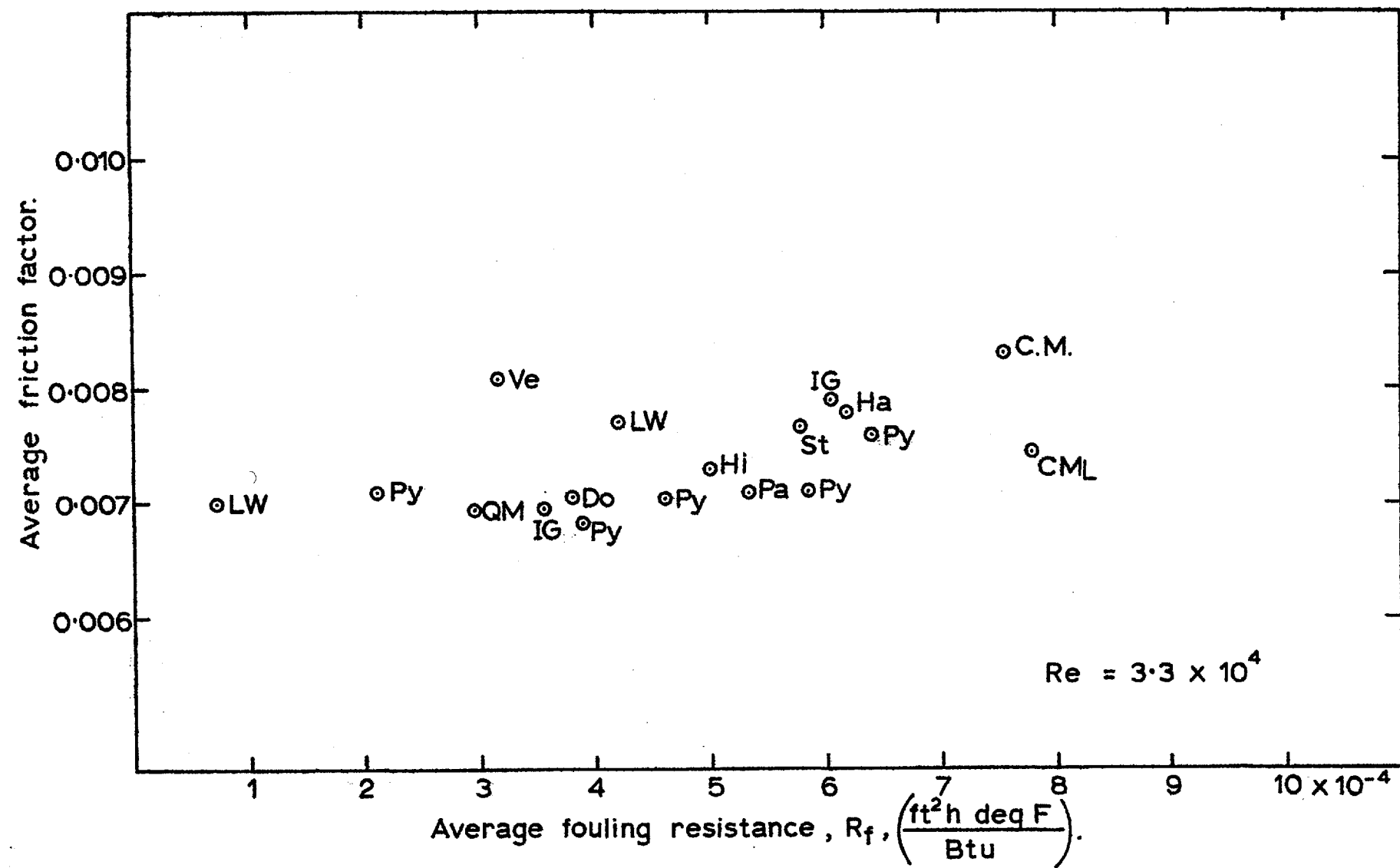


FIG. 23. AVERAGE FRICTION FACTOR v AVERAGE FOULING RESISTANCE.

association of high friction factor with high fouling resistance, but the correlation is very poor and the scatter of the experimental points too great to permit any general relationship between friction factor and tube life or fouling resistance to be deduced. Plots of individual tube friction factors against the same two variables were so scattered that they were not worth reproducing.

As has already been described the inlet and outlet connections to the section of tube under test incorporated the pressure tapings, Fig. 6. The tube was lined up with these connections by scraping the inner surface of the tube. Owing to the confined space it was not possible to determine whether or not this was done accurately. When the tube internal diameter was greater than nominal, and when the inner and outer tube surfaces were not concentric, lining up was of course impossible. Such inaccuracies are only a partial explanation of the scatter of the experimental points. The chief cause was the differences in the internal diameters of the tubes from the nominal, these differences being due to the internal deposits and the manufacturing tolerances. The thickness of the deposits was of the order of 0.005 in and assuming the manufacturing tolerances to be within those specified by B.S. 1464 of 1948 and B.S. 378 of 1953, variations in the internal diameter could be from 0.008 in greater to 0.013 in less than the nominal.

A difference of 0.01 in between the actual and nominal internal diameters will make a difference of 8 per cent in the friction factors, and a difference of 0.02 in 18 per cent.

The friction factors obtained are thus only of interest in that they give an indication as to the magnitude of the factor which should be used, in conjunction with the nominal internal tube diameter, when calculating the pressure required to give the required water flow rate through a dirty condenser.

CHAPTER 14

COMPARISON OF THE MEASURED WATERSIDE HEAT TRANSFER COEFFICIENTS WITH PREVIOUS WORK

The linearity of the Wilson Line Plots shows that the Sieder-Tate formula accounts successfully for the variation of the water-side heat transfer coefficients with water velocity and temperature for values of Re greater than 1.7×10^4 and, as has been pointed out in Chapter 8, the slope of these plots, for new and acid cleaned tubes, gives a value of the multiplying constant, K , of 0.027 which is the value recommended by McAdams.¹⁶

The increase in the slope of plots obtained from tests on dirty tubes was, however, unexpected. As is shown in Appendix 1 the slope of the plot, m , is equal to $\frac{D_o}{K} \left(\frac{\pi D_i}{4000} \right)^{0.8}$. Internal deposits will decrease D_i so the value of m should be less for fouled tubes than for clean ones.

To make sure that this change in slope was real tests were made to make certain that it was not due to thermometer calibration errors. The main difference between the water-side conditions during tests on clean and on fouled tubes was that, for the same cooling water velocity, the cooling water temperature was lower for a fouled tube than for a clean one,

so tests were made on a clean tube using the standard heat transfer rate but a lower condenser pressure, (approximately 29 in Hg Vac) chosen so that the water temperatures were similar to those for standard tests on tubes for which the value of m was a maximum. The slope of the plot from a non-standard test was found to be the same as that from the standard tests, see Fig. 24. The difference between the intercepts is due to differences in the air leakage rate, to tube Py 6C(A) having been acid cleaned and to the different condenser pressures. The slope of the plot from the standard test of tube C.M.Y., in which the cooling water temperature velocity relationship was similar to that for the test at 29 in Hg vac, was about 9 per cent greater, i.e. the value of K , based on the nominal internal diameter was 0.0245.

Interchanging the inlet and outlet cooling water thermometers also failed to change the slope of the plot. The conclusion must be that the internal deposits, in some way, decrease the heat transfer coefficient between the wetted surface and the water. The chief difference between the inner surface of a clean tube and the wetted surface of the internal deposits on a dirty tube is that the latter surface is rougher. It is usually assumed that, for forced convection heat transfer, an increase in surface roughness increases the heat transfer coefficient but there is some evidence to show that this is not always the case.

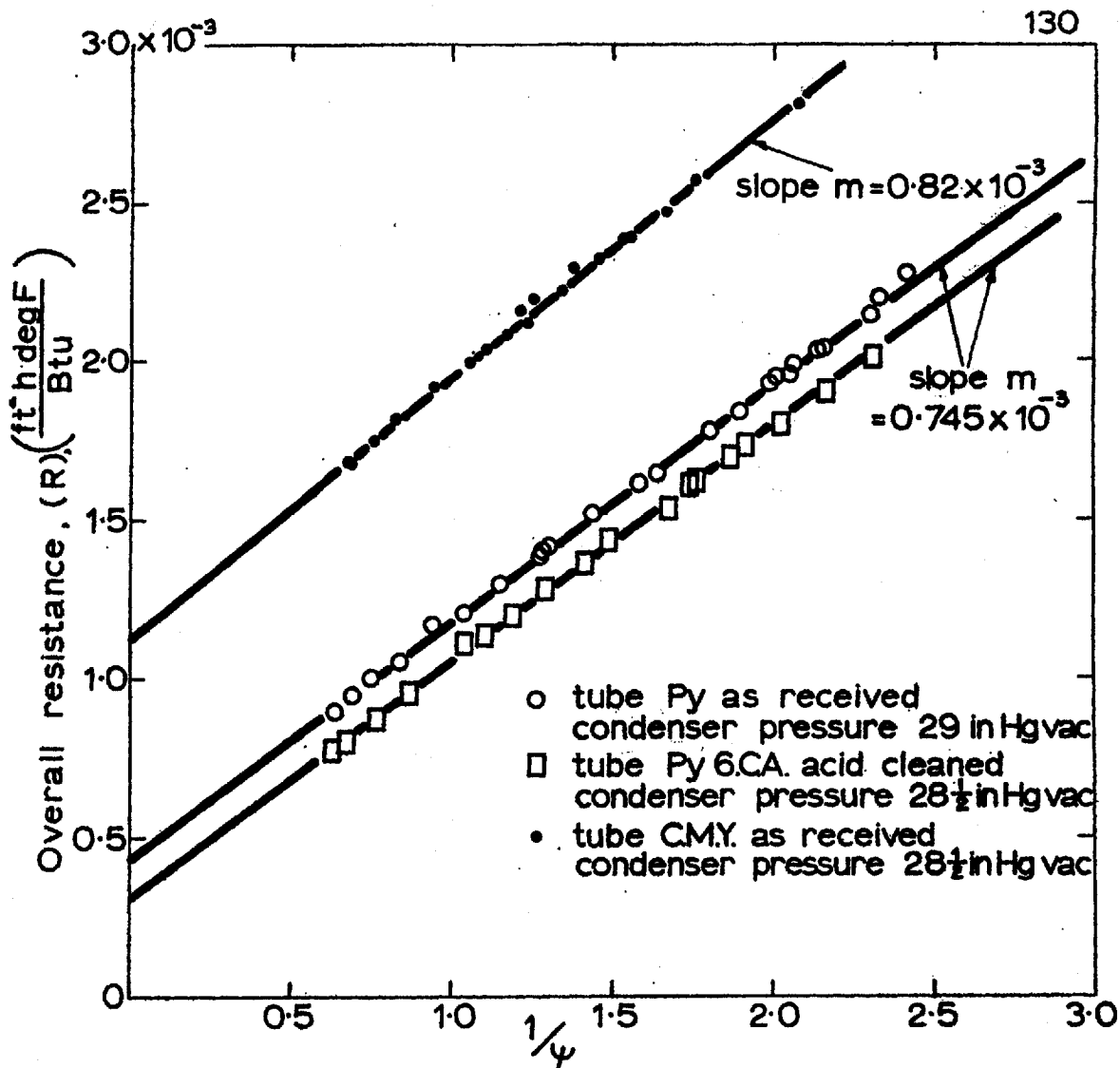


FIG. 24. WILSON LINE PLOT. CHANGE IN SLOPE WITH FOULING.

In the report on the condenser tubes from H. M. S. Cockade mention was made of the effects of corrosion revealed when the tubes had been acid cleaned. This had produced roughening of the inner surface of the tubes in the form of shallow pits. In the opinion of the authors this roughening would have an adverse effect on the heat transfer coefficient. Unfortunately, the report did not make it clear whether this opinion was based on experience or intuition.

Experimental evidence of a decrease in heat transfer coefficient with small scale roughness was obtained by Cope⁷ who investigated the effect of surface roughness on heat transfer coefficients for water heated and cooled in flow through tubes. In one series of experiments the inner surface of a 1 in internal diameter tube was roughened by cutting multi-start left and right hand Whitworth threads, 0.010 in deep, so as to remove 10 per cent of the original smooth surface. In a second experiment a $\frac{3}{4}$ in internal diameter tube was roughened by knurling the inner surface so as to produce a regular pattern of contiguous pyramids about 0.008 in high. The friction factor for the 1 in tube was approximately equal to that of a similar smooth tube but for the $\frac{3}{4}$ in tube the friction factor at $Re = 3.3 \times 10^4$ was about 0.009 and at higher values of Re became constant at about 0.012, i.e. the tube was aerodynamically rough. Although

friction factors of the same order at $Re = 3.3 \times 10^4$ were calculated for some of the older tubes tested this was due more to the decrease in internal diameter than to the increase in roughness and in all cases the friction factor decreased with increasing Reynolds number. Copes $\frac{3}{4}$ in tube was therefore rougher than any of the condenser tubes tested.

The Sieder-Tate formula was used to recalculate Copes data for rough and smooth tubes and the results, plotted in Fig. 25, show that the roughened tubes gave lower coefficients. Most of Copes experimental points are in the transition region so it is not possible to estimate the effect of the roughness on the value of K.

Cope also made measurements with two other internally knurled tubes with pyramid heights of 0.025 and 0.048 in. Both these gave higher coefficients than smooth tubes. So it appears that with increasing roughness heat transfer coefficients first decrease and only increase to become greater than the smooth tube values when the roughness exceeds some particular value. As the values of K calculated from Copes⁶ smooth tube readings, for Re greater than 17×10^3 , were lower than those measured in this investigation, results obtained by other workers, i. e. Eagle and Ferguson⁸, Sherwood and Petrie²⁰, Jurgensen and Montillon¹⁴ and El Arabi,¹ were also recalculated to find out the values of K obtained. These values of K are summarised in Table 8 which shows that, despite the care taken by the various

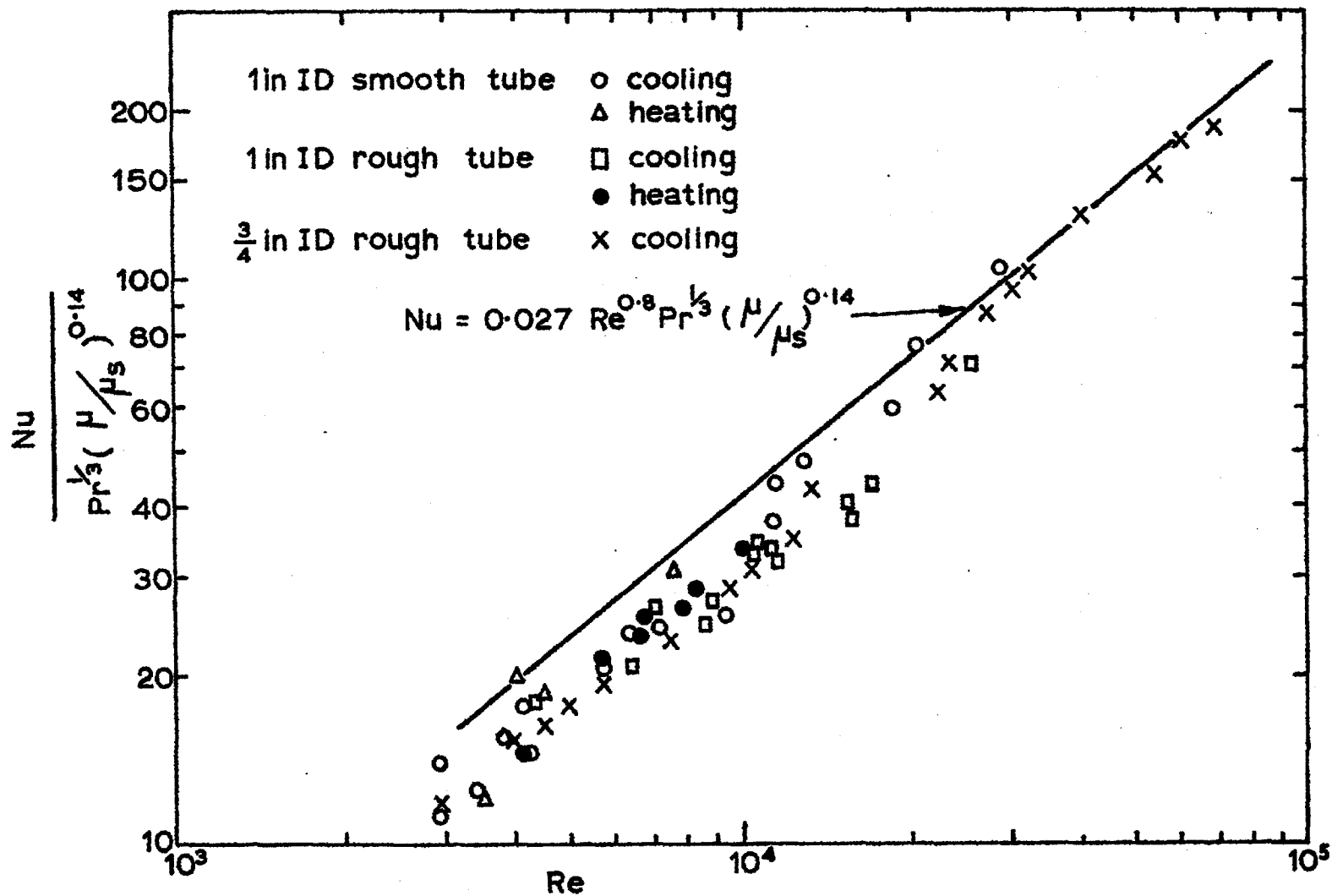


FIG. 25. DATA OF COPE FOR ROUGH AND SMOOTH TUBES.

TABLE 8
Values of K obtained by other experimentalists

<u>Author</u>	<u>Brief details of Apparatus</u>	$\text{Range of } K = \frac{\text{Nu}}{\text{Re}^{0.8} \text{Pr}^{\frac{1}{3}} (\mu/\mu_s)^{0.14}}$ (for Re greater than 17×10^3)	<u>Remarks</u>
Eagle and Ferguson	Electrically heated $\frac{3}{4}$ in 18G condenser tube.	0.028-0.031	K constant for experiments with constant water velocity and heat transfer rate.
Sherwood and Petrie	4 ft copper tube, 0.675 in O.D. 0.494 in I.D. heated by steam at atmospheric pressure	0.019 - 0.027	
Jurgensen and Montillon	5 ft copper tube, 1.315 in O.D., 1.062 in I.D. heated by steam at atmospheric pressure	0.029 - 0.038	
Cope	6 ft brass tube, 1.0 in I.D., heated and cooled by water	0.022 - 0.028	K increasing with increasing Re from 0.022 at Re = 17×10^3 to 0.028 at Re = 37×10^3
Al Arabi	2.5 ft copper tube, 1.0 in O.D., 0.75 in I.D. heated electrically	0.0190 - 0.0195	

experimentalists, there were considerable differences between the results they obtained under apparently similar experimental conditions.

The reports from which these data were obtained are not sufficiently detailed to make it possible to explain these differences, but there is one possibly significant variable that does not appear to have been considered, i.e. the amount of air dissolved in the cooling water used.

The effect of dissolved air has been reported²⁵ for heat transfer from mercury. Early measurements of forced convection heat transfer coefficients for mercury failed to give reproduceable results. Maximum values were fairly well defined but much lower values were frequently obtained. At first it was thought that this depended on whether or not the mercury wetted the heat transfer surface but eventually it was discovered that the variation was due to air, dissolved in the mercury, separating out at the surface and forming a film between the surface and the flowing metal. It was found that mercury can absorb sufficient air to reduce its specific gravity by 12 per cent. Experiments with mercury, containing air, flowing through a transparent tube showed that surface roughness promoted the release of the air which could be seen to form a fan shaped cavity, between the tube wall and liquid metal, downstream of a roughness projection. It is possible that the same mechanism can operate when water is heated in a roughened tube.

It is reasonable to assume, therefore, that the deposits on the inner surface of condenser tubes decrease the heat transfer coefficients at the wetted surface and therefore the increase in the thermal resistance of tubes with time is greater than that of the thermal resistance of the deposits.

CHAPTER 15

THE EFFECT OF CONDENSER TUBE FOULING ON MARINE POWER PLANT PERFORMANCE.

Owing to the number of variables involved it is not easy to estimate the change in the performance of a ship's steam power plant which will result from the progressive accumulation of deposits on the condenser tubes.

When the overall thermal resistance of a condenser, operating with constant cooling water temperatures and velocity and constant condensation and heat transfer rates, increases the condenser temperature will increase. The increase will be such that the overall temperature difference will be that required to maintain the heat transfer rate against the higher resistance. The increase in condenser temperature will be accompanied by an increase in condenser pressure, i.e. the corresponding saturation pressure, or in conventional terms, a fall in condenser vacuum. Condenser vacuum is traditionally used as a measure of condenser performance though at the operating pressures in modern condensers, i.e. 29 - 28½ in Hg vac, the rate of change of saturation pressure with temperature is so small that measurements of the condenser temperature would provide a more sensitive and reliable indication of changes in condenser conditions.

The change in condenser pressure that would result from an increase in thermal resistance depends on the initial condenser operating conditions and performance. If the initial resistance is high, i.e. the overall heat transfer coefficient is low, then the increase will have little effect. If the initial resistance is low then the same increase will produce a greater change. The corresponding changes in condenser pressure will depend on the initial pressure, being smaller when the initial pressure is low, i.e. for high vacuum condensers.

The following example illustrates these points. If the resistance of a condenser, operating with constant cooling water inlet and outlet temperatures of 60°F and 70°F and an initial vacuum of 29 in Hg vac, increases by 0.6×10^{-3} ft² h deg F/Btu the vacuum will fall by 0.12 in to 28.88 in Hg vac if the initial heat transfer coefficient is 400 Btu/ft² h deg F and by 0.26 in to 28.74 in Hg vac if the initial coefficient is 800 Btu/ft² h deg F.

For cooling water temperatures of 60°F and 100°F and an initial vacuum of 27 in Hg vac the same increase in resistance will, for initial coefficients of 400 and 800 Btu/ft² h deg F, cause the vacuum to fall by 0.60 and 1.34 in to 26.40 and 25.66 in Hg vac respectively. The increase in thermal resistance used in this example is that

due to deposits accumulated during a tube life of about between 30 and 60 months. The cooling water outlet temperatures used in these examples are those found by experience to correspond to the chosen initial condenser pressures.

The examples based on an initial coefficient of $400 \text{ Btu/ft}^2 \text{ h deg F}$ are unlikely to predict, accurately, the change in conditions in a condenser in which the average overall coefficient is of this value. A low coefficient would probably be due to poor steam and cooling water distribution and a wide range of local coefficients within the condenser. In such a condenser the bulk of the heat transfer would take place through tubes where the local coefficients were higher than the mean and the change in condenser vacuum would probably be of the same order as that calculated for the higher initial coefficient. As shown in Chapter 13 tubes in those parts of a condenser where heat transfer rates are high may accumulate deposits at a higher rate than the average, so again the actual decrease in condenser vacuum would be greater than estimates based on average overall performance.

The economically important effect of condenser tube fouling is that the resulting increase in condenser pressure increases the specific steam consumption of the ships

power plant which, being approximately proportional to the fuel consumption, increases the running costs. Assuming that a plant is required to maintain a constant power output, as is, approximately, the duty of a ship's power plant, then an increase in condenser pressure will result in an increase in the steam flow rate as well as an increase in the enthalpy of the steam entering the condenser. To estimate the resulting increase in specific steam consumption some assumptions must be made regarding the operating conditions and the performance of the steam plant.

To provide a basis for further calculations it is assumed that the initial steam conditions for the S.S. Pyrrhus, i.e. a pressure of 525 lbf/in^2 gauge and a temperature of 525°F , are representative. It is further assumed that the isentropic efficiency of the expansion is 80 per cent and that the relationship between enthalpy and entropy during the expansion is linear, i.e. the line of steam condition on the enthalpy-entropy diagram is straight. Measurements of steam condition during flow through multi-stage turbines show that the slope of the line of condition in fact decreases towards the end of the expansion and, as modern marine steam power plants use regenerative feed water heating, only about 80 per cent of the initial steam flow enters the condenser, so this latter assumption will tend to over-estimate the increase

in specific steam consumption with increase in condenser pressure.

The resulting relationship between specific steam consumption and condenser vacuum thus obtained, for initial condenser pressures of 28.5 and 27 in Hg vac, are given in Fig. 26, together with the corresponding relationship given by Stodola and Loewenstein²⁴ for "high pressure" turbines, (i.e. initial steam pressures greater than 200 lbf/in² gauge).

To estimate the increase in specific steam consumption with tube life, due to condenser tube fouling, the following sets of initial condenser conditions were assumed:

- (1) Vacuum 28.5 in Hg vac, cooling water inlet and outlet temperatures 60 °F and 78°F.
- (2) Vacuum 27 in Hg vac, cooling water inlet and outlet temperatures 60 °F and 100°F.

The cooling water mass flow rate was assumed to be constant. This assumption is discussed later.

For each set of conditions calculations were made for initial overall heat transfer coefficients of 800 and 600 Btu/ft²h deg F. Details of tests on new condensers are seldom published but a private sight of the acceptance test data for some new land based condensers for the Central Electricity Generating Board showed that they achieved overall coefficients between 640 and 700 Btu/ft² h deg F.

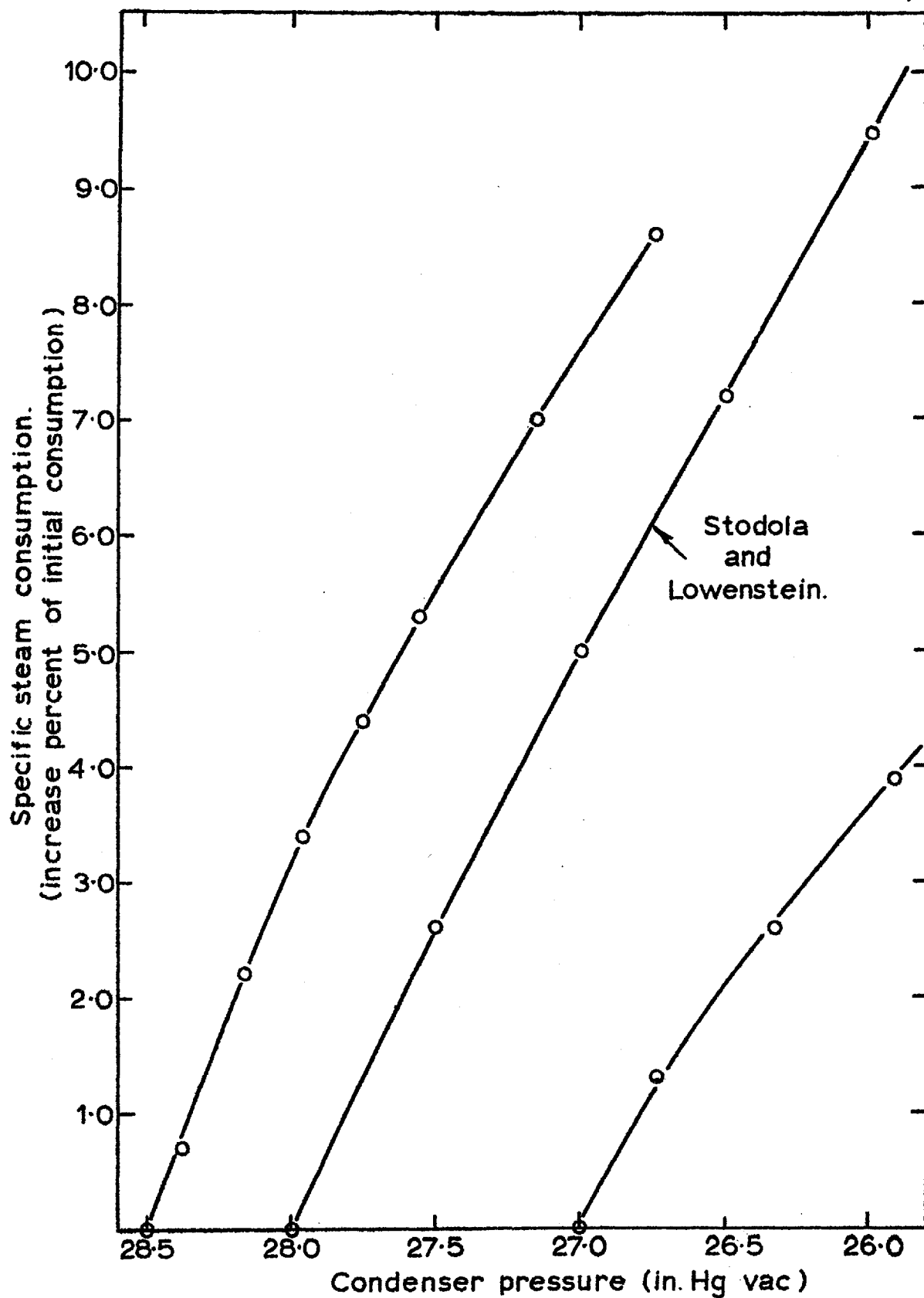


FIG. 26. SPECIFIC STEAM CONSUMPTION v CONDENSER PRESSURE.

The calculation procedure was as follows:-

- Stage 1. From the relationship $\dot{q}'' = h_{sw} \ln(T_c - T_w)$
where $\dot{q}''$ = heat transfer rate, Btu/ft²h,
 $\ln(T_c - T_w)$ = logarithmic mean temperature
difference and h_{sw} = overall heat transfer
coefficient, calculate $\dot{q}''$ for an assumed set of
initial condenser conditions.
- Stage 2. For an assumed higher condenser pressure
obtain from Fig. 27 the increase in heat transfer
rate due to the increase in specific steam con-
sumption and of the enthalpy of the steam entering
the condenser. Hence calculate the new values
of $\dot{q}''$ and cooling water outlet temperature.
- Stage 3. For the new cooling water outlet temperature
and condenser temperature (corresponding to
the assumed higher condenser pressure) determine
the new value of h_{sw} . From the initial and new
values of h_{sw} determine the increase in overall
thermal resistance.
- Stage 4. From Fig. 18 obtain the maximum and minimum
tube lives required to give this increase in
resistance.

The curves of Fig. 27 were obtained assuming that
the condensate temperature remained constant despite the

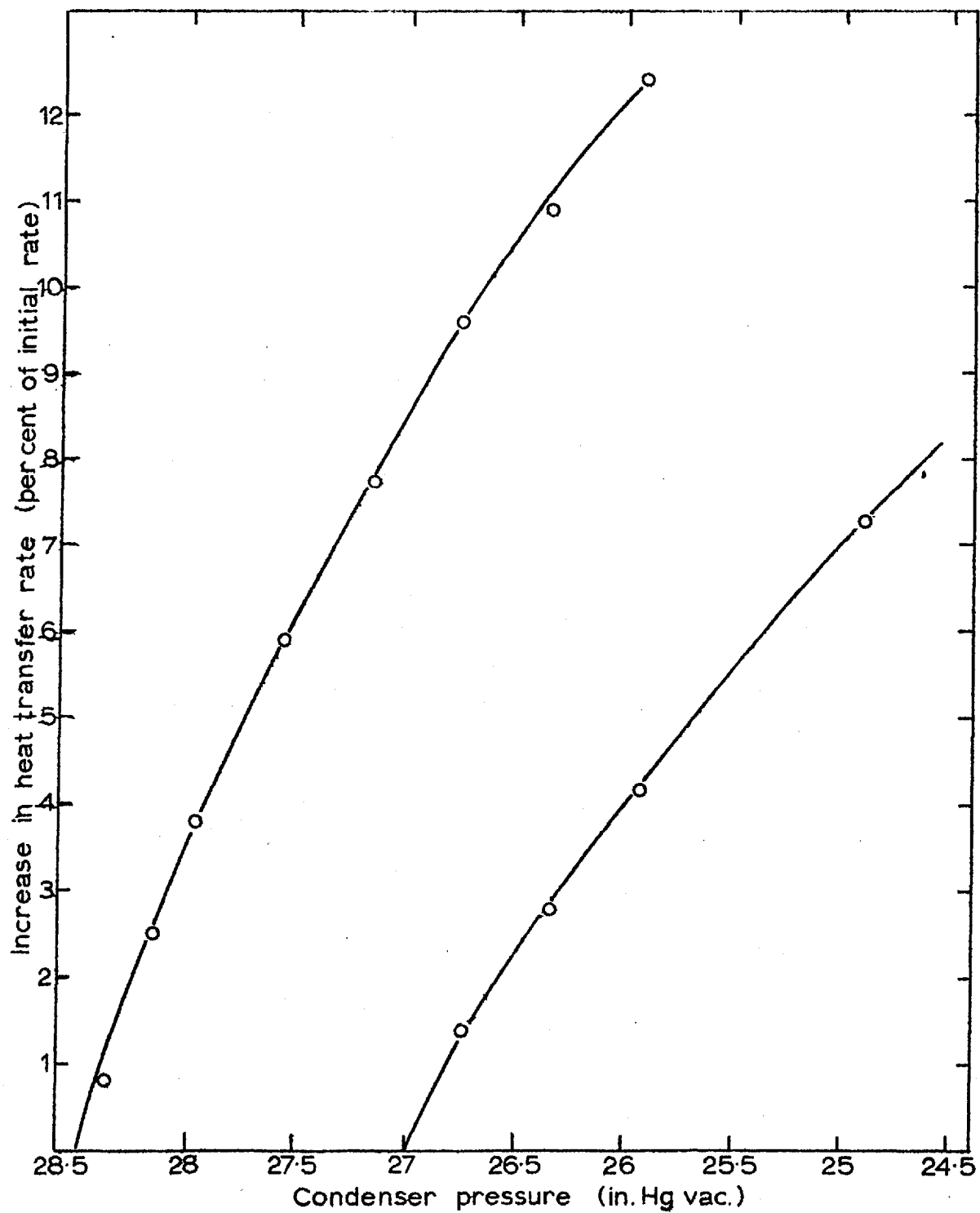


FIG. 27. HEAT TRANSFER RATE v CONDENSER PRESSURE.

increase in condenser temperature. As the condensate temperature depends chiefly on the cooling water temperatures this is a reasonable assumption.

The relationships thus obtained between condenser pressure and tube life are given in Figs. 28 to 31, and between specific steam consumption and tube life in Figs. 32 to 35.

An example of the calculation described above is given in Appendix 4.

An important variable that cannot be allowed for in these calculations is the cooling water velocity. It appears to be the practice to run the steam turbines, driving the centrifugal cooling water pumps, with steam inlet valves fully open. These turbines are usually of the single stage impulse type and their performance will not vary appreciably with condenser pressure. However, as the hydraulic resistance of the condenser tubes increases, due to the accumulation of deposits, the performance of the turbine-pump unit will change. To determine the change in cooling water velocity would require not only more reliable friction factor-tube life data than has been obtained in this investigation but also the water delivery pressure-flow rate characteristics for the turbine-pump unit. The usual form of these characteristics is such that an increase in delivery pressure

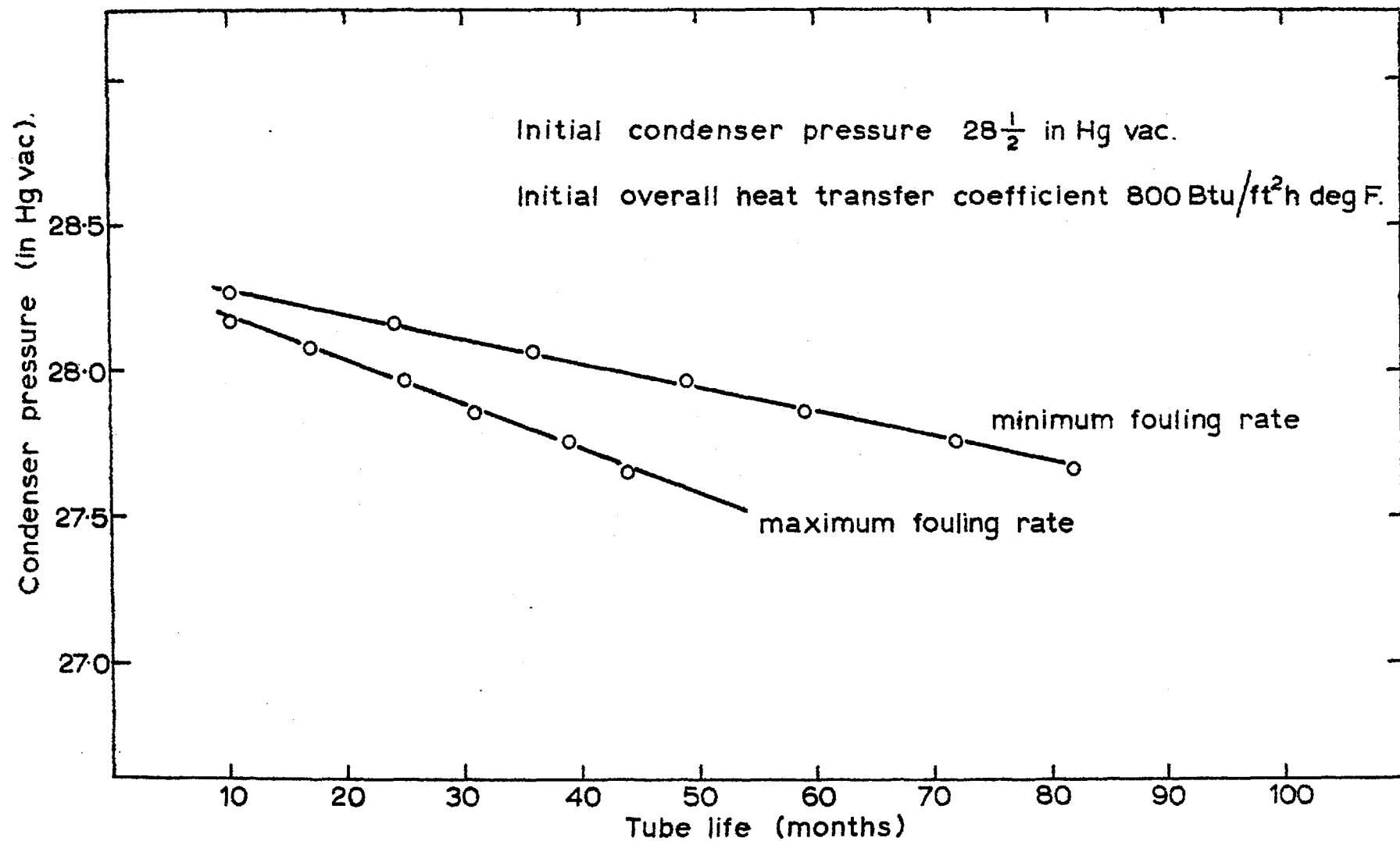


FIG. 28. CONDENSER PRESSURE v TUBE LIFE.

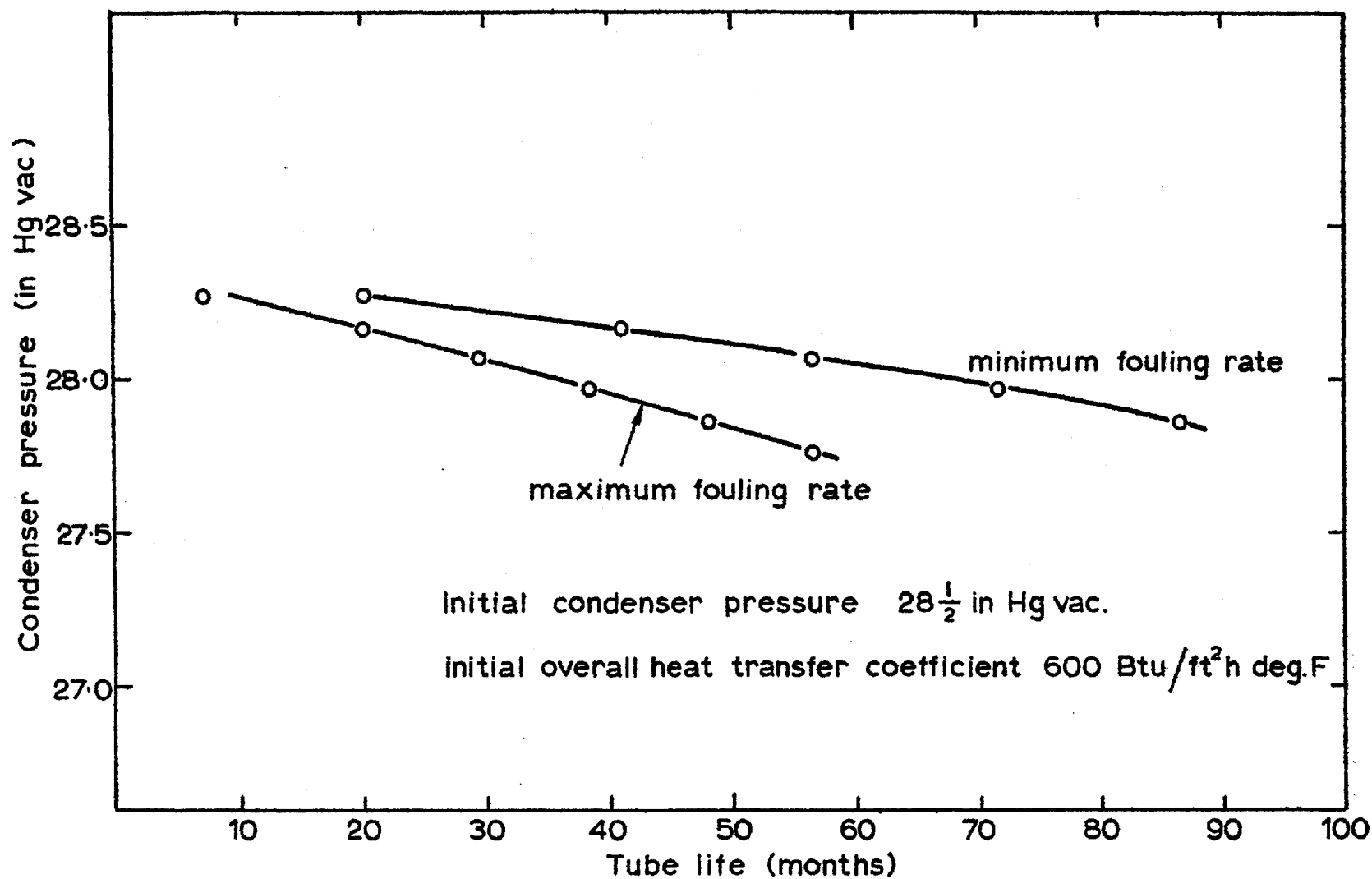


FIG. 29. CONDENSER PRESSURE v TUBE LIFE.

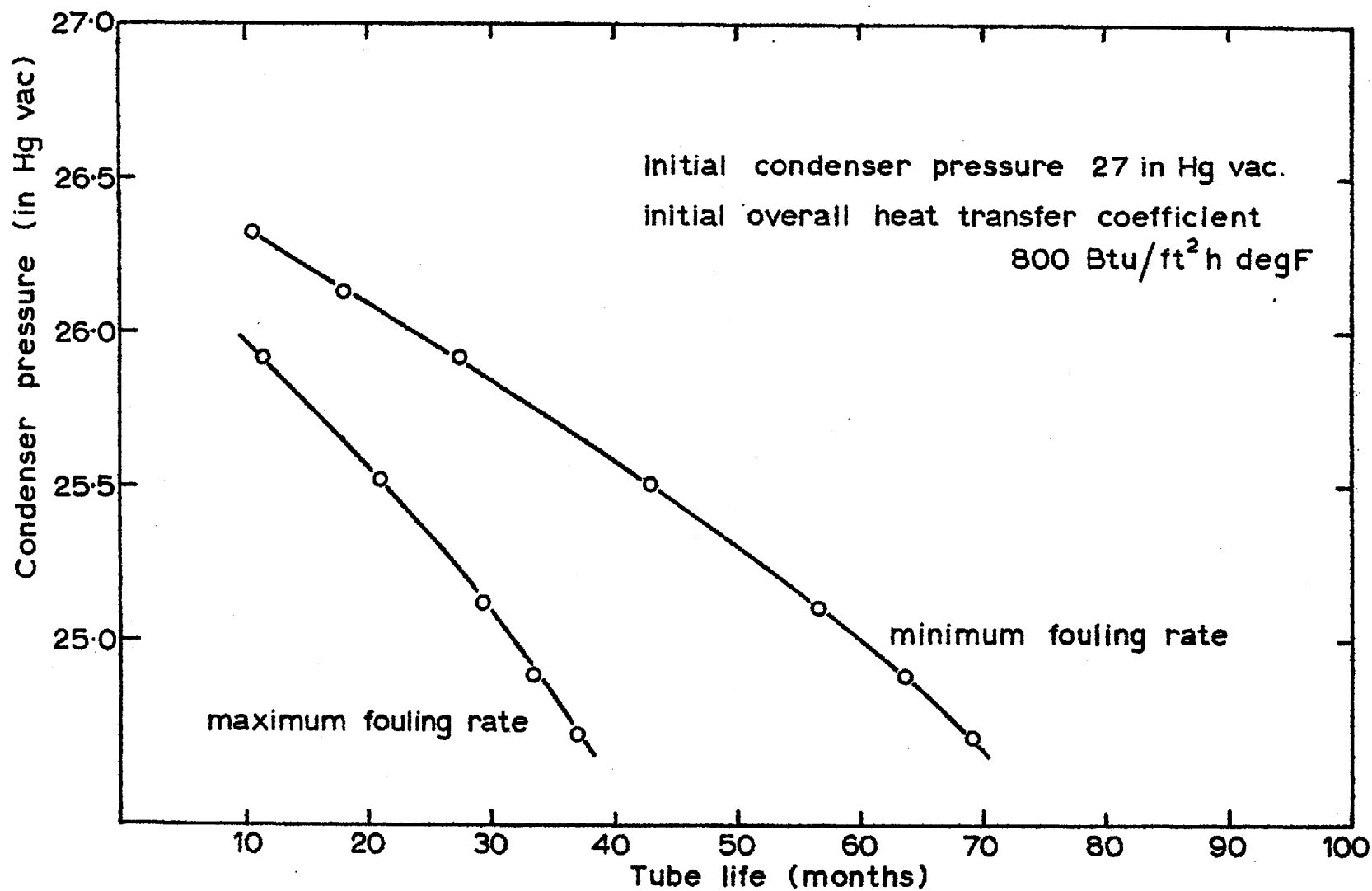


FIG. 30. CONDENSER PRESSURE v TUBE LIFE.

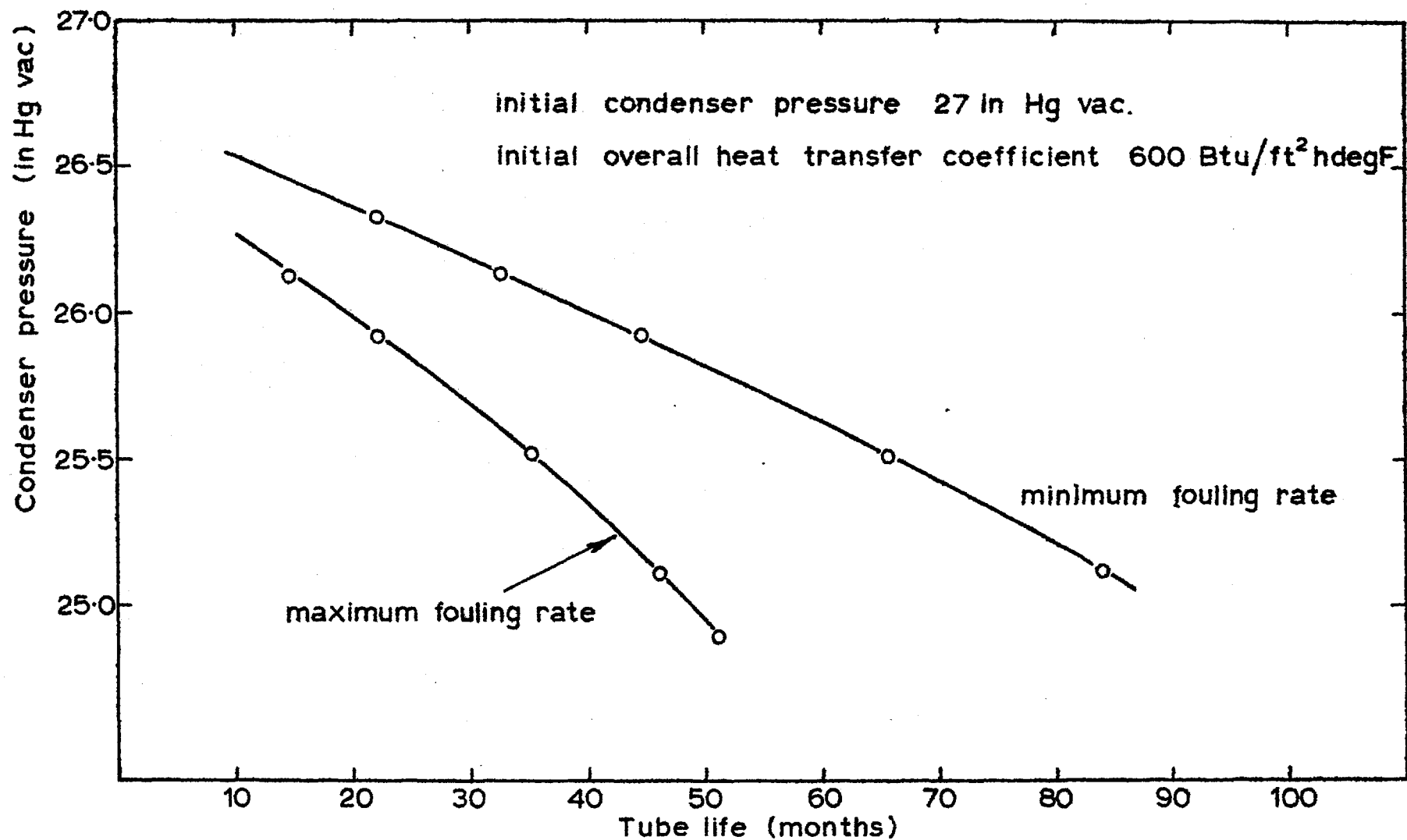


FIG. 31. CONDENSER PRESSURE v TUBE LIFE.

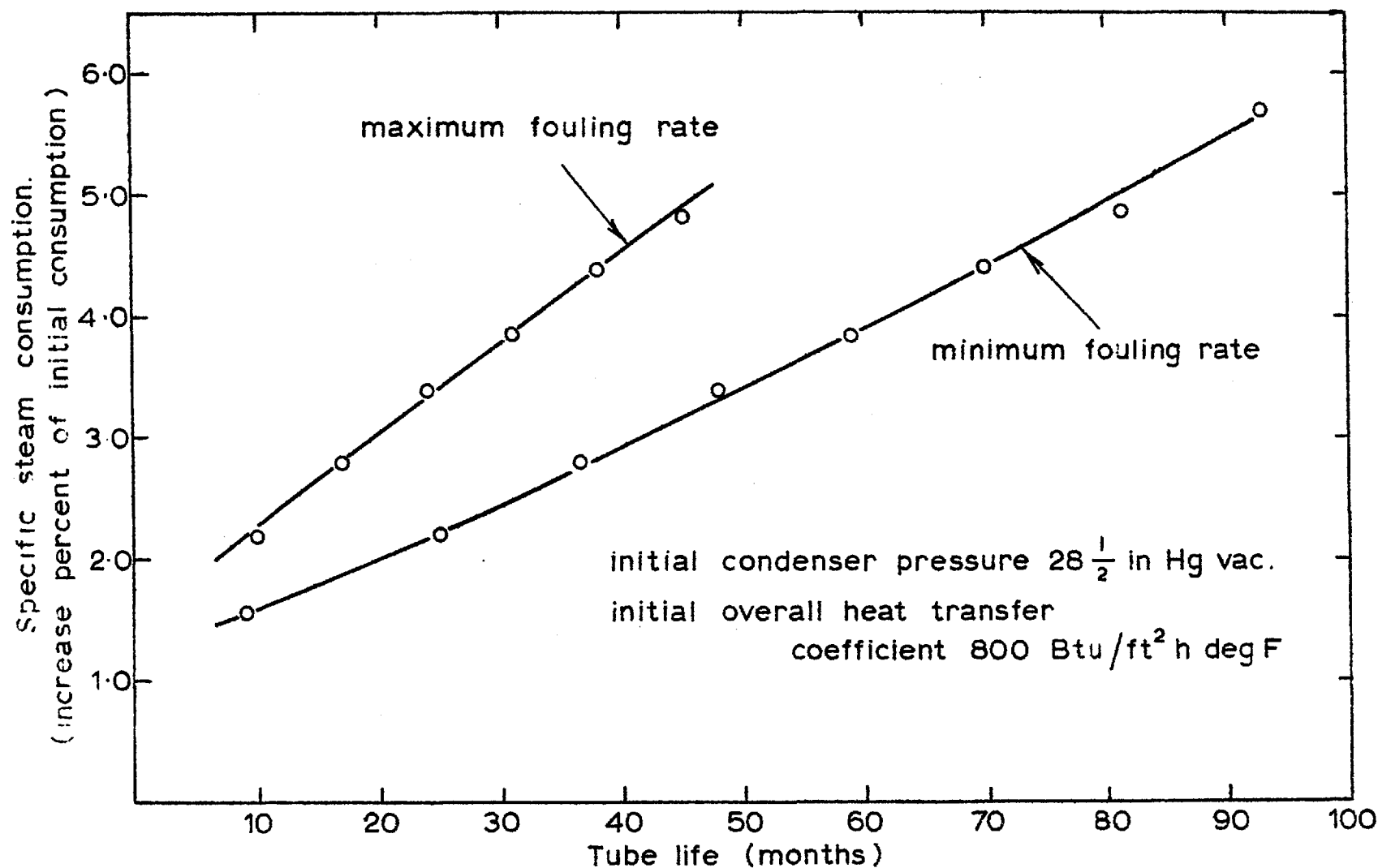


FIG. 32. SPECIFIC STEAM CONSUMPTION v TUBE LIFE.

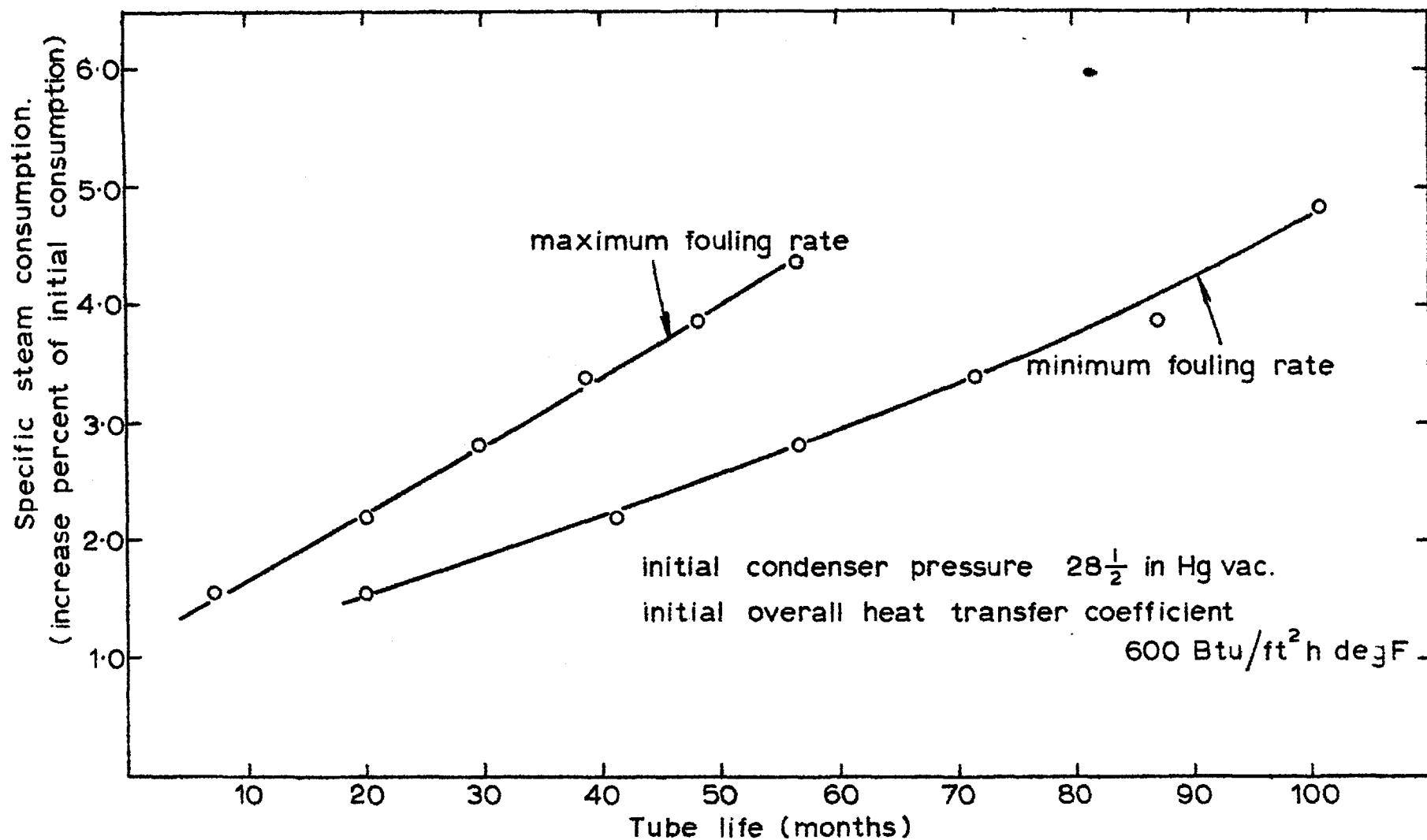


FIG. 33. SPECIFIC STEAM CONSUMPTION v TUBE LIFE.

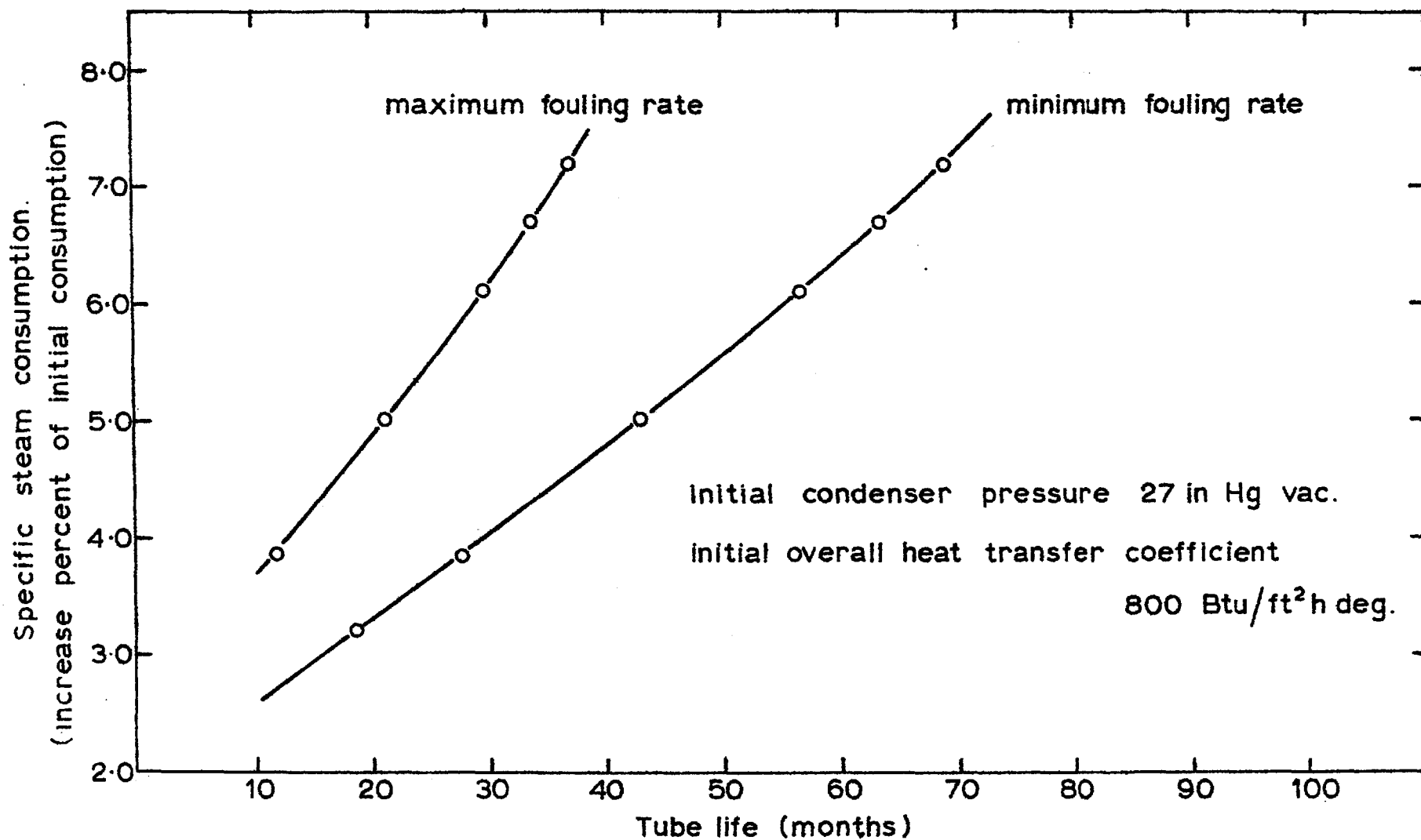


FIG. 34. SPECIFIC STEAM CONSUMPTION v TUBE LIFE.

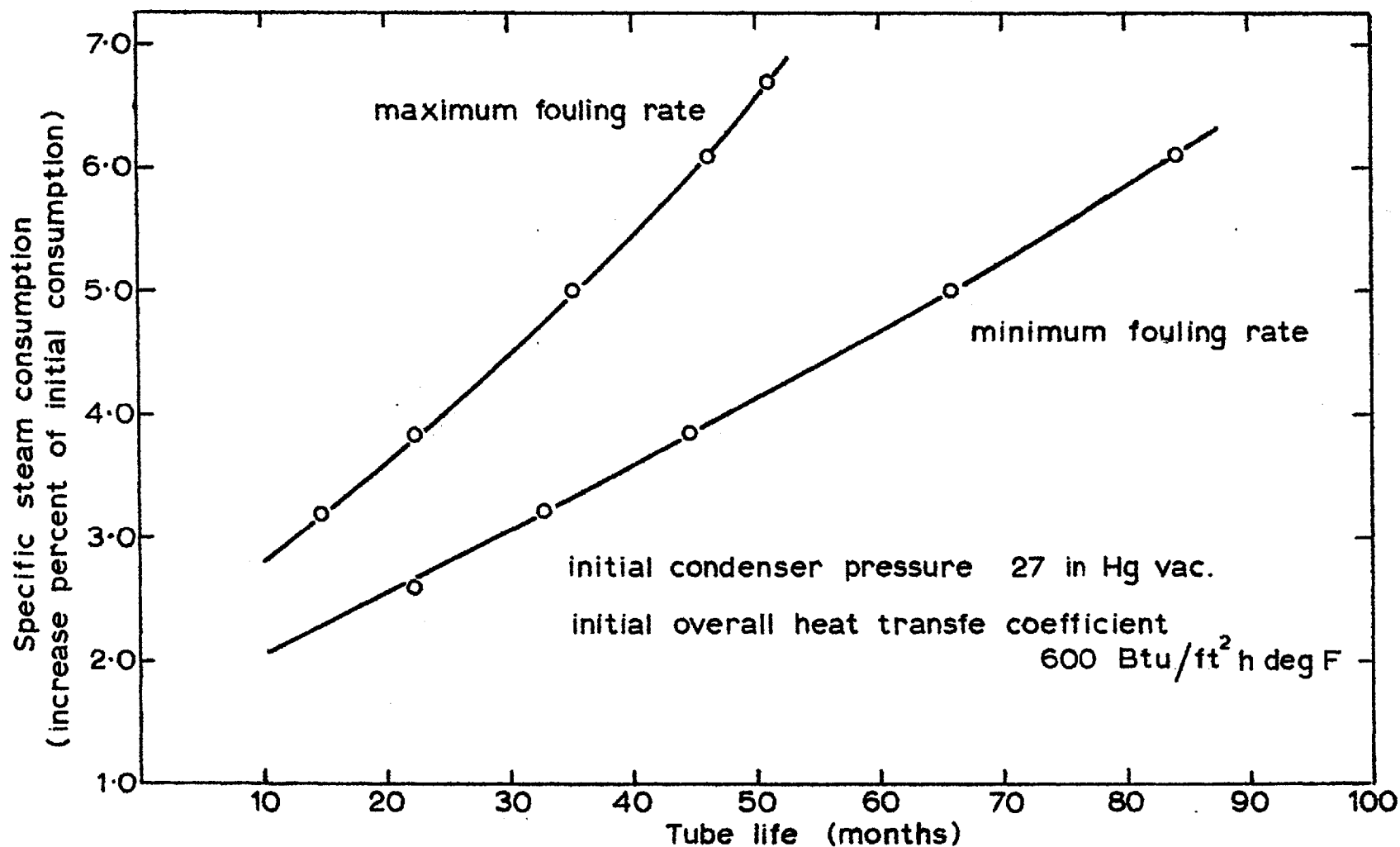


FIG. 35. SPECIFIC STEAM CONSUMPTION v TUBE LIFE.

is accompanied by a decrease in the flow rate. A decrease in water velocity will further increase the thermal resistance of the condenser and therefore the condenser pressure and specific steam consumption.

The increase in thermal resistance due to the increase in the condensation rate will be small and will be partially compensated for by the increase in the condensate film temperature.

The assumption that a ship's power plant is required to maintain a constant power output is questionable. It is more probable that a constant ship's speed is required. If this is so then the progressive foul^{ing} of the ship's hull will require an increase in power and of steam flow rate which would mask the change in condenser performance due to condenser tube fouling. Only a continuous measurement of the flow through selected tubes in the condenser, as already suggested, could, together with an experimentally determined specific steam consumption - condenser pressure relationship for the turbines, enable the effect of the tube fouling alone to be determined.

However, if it can be assumed that the over-estimation of the increase in specific steam consumption with increase in condenser pressure is compensated for by

the neglect of the decrease in the cooling water flow rate due to tube fouling then Figs. 32, 33, 34 and 35 will if anything underestimate the effect of condenser tube deposits on the specific steam consumption of marine steam power plants.

APPENDIX ITHE WILSON LINE PLOT

Most heat transfer text books, particularly those published in the U.S.A., use the above title to describe an experimental technique that can be used when it is only possible to measure the sum of several thermal resistances in series and it is required to obtain the value of one of them. The case usually treated is of a vapour condensing on the outer surface of a tube through which a fluid is flowing. In this case there are usually three resistances to be considered, i.e. that due to the tube wall, and those at the inner and outer tube surfaces. It is relatively easy to measure experimentally the overall resistance but difficult to measure that at the condensation surface alone.

The so-called Wilson Line Plot method for obtaining this resistance from measurements of the overall resistance is as follows. The overall resistance is measured for a series of fluid velocities and the results plotted in the form overall resistance against the reciprocal of the fluid velocity to the power 0.8. The best straight line is then drawn through the experimental points and extrapolated to cut the overall resistance axis. The intercept on this axis is taken to give the sum of the thermal resistances of the tube wall and at the condensation surface. The resistance

of the tube wall can be calculated and that at the condensation surface is then obtained by difference.

The reference given when mentioning this method is to a paper written by E.E.Wilson²⁶ and published in the A.S.M.E. Journal in September 1915. All that Wilson did, however, was to reproduce, without specific acknowledgement, material from a series of unsigned articles,¹⁷ entitled "The Theory of the Surface Condenser", which had appeared in "Engineering" commencing in January 1914. These had been written by the late H.M.Martin, who was at the time a member of the staff of "Engineering".

Martin, starting from Reynolds' analogy between heat and momentum transfer, had come to the conclusion that the resistance to heat transfer between the cooling water and the condenser tube depended on the tube diameter and the kinematic viscosity of the water, as well as the water velocity. To test his theory he plotted the experimental results of Orrook¹⁸ and others in the form of overall resistance against the reciprocal of the water velocity "corrected" to 60°F. This correction was made by multiplying the measured velocity by the ratio of the kinematic viscosity of water at 60°F to that at the actual mean water temperature. He then drew a best straight line through the points obtained and pointed out the significance of the intercept on the resistance axis.

Unfortunately, Martin used an empirical formula to calculate the variation of the kinematic viscosity of water with temperature and, owing to a misprint in the text he consulted, his values were too high. He corrected this mistake in a later article but not before he had become involved in an acrimonious correspondence with T.E.Stanton, who had pointed out the error.

In his final article, Martin replotted overall resistance against the reciprocal of the corrected water velocity to the power of 0.82, which by trial and error he found best to correlate the data available. The correlation was not particularly good, but in view of the conditions in which Orrock obtained his data, this is not surprising.

The intercept on the overall resistance axis gave a steam side resistance in agreement with estimates then available. At that time Nusselt's analysis of condensation heat transfer had not been published and Martin made the then common error of assuming the resistance at the condensation surface to be constant. However, he pointed out, in explaining the scatter of the experimental points, that air in the steam and deposits on the surfaces would increase the overall thermal resistance.

Martin's final conclusion was that the heat transfer coefficient between the cooling water and the inner surface of the tube varied as $Di \left(\frac{\rho V}{\mu} \right)^{0.82}$ which was a

considerable step towards the more recent correlations of the form $Nu = a$ function of Re and Pr .

Wilson's contribution was merely to reproduce Martin's final plot without the accompanying argument, which was a pity. If the many experimentalists who have used the Wilson Line Plot had started from Martin's original articles, they would have used it more intelligently and possibly the now familiar correlating formula for forced convection heat transfer data would have been developed earlier.

The justification and development of the Wilson Line Plot in the light of more recent knowledge is, for the case of steam condensing on the outer surface of a clean condenser tube through which water is flowing, as follows:-

The heat transfer rate per foot length of tube, $\dot{q}/ft$, can be expressed in four ways:

For the overall heat transfer from steam to cooling water

$$\dot{q}/ft = h_{sw}(T_c - T_w)\pi D_o \quad (1)$$

For the heat transfer from the steam to the outer surface of the tube

$$\dot{q}/ft = h_s (T_c - T_{so})\pi D_o \quad (2)$$

For the heat transfer through the tube wall by conduction, assuming that the thickness of the tube wall is small compared with the tube diameters

$$\dot{q}/ft = \frac{2k\pi}{(D_o - D_i)} (T_{so} - T_{si}) \frac{(D_o + D_i)}{2} \quad (3)$$

Where k is the effective thermal conductivity of the tube material.

For the heat transfer from the inner surface of the tube to the water

$$\dot{q}/ft = hw (T_{si} - T_w)\pi Di \quad (4)$$

By adding equations (2), (3) and (4)

$$(T_c - T_w) = \dot{q}/ft \left[\frac{1}{hs\pi Do} + \frac{(Do - Di)}{\pi k(Do + Di)} + \frac{1}{hw\pi Di} \right]$$

From equation (1)

$$(T_c - T_w) = \frac{\dot{q}/ft}{hsw\pi Do}$$

$$\therefore \frac{1}{hsw\pi Do} = \frac{1}{hs\pi Do} + \frac{(Do - Di)}{\pi k(Do + Di)} + \frac{1}{hw\pi Di}$$

$$\therefore R = \frac{1}{hsw} = \frac{1}{hs} + \frac{Do(Do - Di)}{k(Do + Di)} + \frac{Do}{hwDi}$$

i.e. the overall thermal resistance, R , is equal to the sum of the thermal resistance at the inner and outer surfaces of the tube and of the tube wall, related to the external surface area of the tube.

$$\text{i.e. } R = R_s + R_t + R_w$$

Additional terms, representing the thermal resistance of surface deposits, can be included as can a term for the resistance due to the presence of air in the condensing steam.

The Wilson Line Plot assumes that this relationship between R and the separate resistances is of the form

$y = mx + c$ where m and c are constants

i.e. the overall resistance is a linear function of the variables controlling the individual resistances.

The resistance of the tube wall, R_t , is a constant, depending only on the tube diameters and the effective thermal conductivity of the tube material and may be calculated for a given tube. The effective conductivity of the tube material may differ from the actual conductivity if the metal is one which acquires an oxide film under normal conditions. Cupro-Nickel is such a metal, its effective thermal conductivity being lower than the actual, see Chapter 8.

The resistance at the inner surface of the tube, R_w , depends on the heat transfer coefficient, h_w , for the forced convection heat transfer to the water flowing through the tube. This coefficient can be related to the water temperature and velocity by the Sieder-Tate²¹ equation, i.e.

$$Nu = \frac{h_w D_i}{k} = K \cdot Re^{0.8} \cdot Pr^{1/3} \left(\frac{\mu}{\mu_s} \right)^{0.14}$$

Although the term (μ/μ_s) was introduced to correlate the results of experiments made with high viscosity oils in laminar flow, Sieder and Tate found that it also improved the correlation of data from experiments made with liquids in turbulent flow. This formula was used in preference to others because the experimental method made

the mean value of T_{si} , and thus μ_s , constant for a given test. This temperature could be deduced by trial and error and hence (μ/μ_s) could be evaluated.

Strictly speaking terms involving similar ratios of thermal conductivity and density should be included but as the change of these properties with temperature is small for most fluids for the usual range of temperatures, they are omitted.

$\therefore$ The waterside resistance, R_w , related to the external surface area, is given by

$$R_w = \frac{1}{h_w} \cdot \frac{D_o}{D_i} = \frac{1}{K} \frac{D_o}{D_i} \cdot \frac{D_i}{k} \cdot \frac{1}{Re}^{0.8} \cdot \frac{1}{Pr}^{1/3} \cdot (\mu/\mu_s)^{0.14}$$

Assuming the density of the water to be constant

$Re = \left(\frac{4Q}{\mu \pi D_i} \right)$ where Q is the cooling water mass flow rate.

$$\begin{aligned} \therefore R_w &= \frac{D_o}{K} \left(\frac{\mu \pi D_i}{4Q} \right)^{0.8} \cdot \frac{1}{k Pr}^{1/3} \cdot (\mu/\mu_s)^{0.14} \\ &= \frac{D_o}{K} \left(\frac{\pi D_i}{4Q} \right)^{0.8} \cdot \frac{\mu^{0.66}}{k Pr^{1/3}} \cdot \mu_s^{0.14} \end{aligned}$$

$\frac{\mu^{0.66}}{k Pr^{1/3}}$ is a function of the mean cooling water temperature, T_{wm} , and is designated $f(T_{wm})$.

For the experimental method used T_{si} , and there-

fore μ_s , is constant for a given tube and can be found by trial and error.

For convenience Q was measured in units of 1,000 lb/hour, thus:

$$R_w = \frac{D_o}{K} \cdot \left(\frac{\pi D_i}{4000}\right)^{0.8} \cdot f(T_{wm}) \cdot \mu_s^{0.14} \left(\frac{1000}{Q}\right)^{0.8}$$

$$\text{Designating } \left(\frac{Q}{1000}\right)^{0.8} \cdot \frac{1}{\mu_s^{0.14}} \cdot \frac{1}{f(T_{wm})}$$

as ψ

$$\therefore R_w = \frac{D_o}{K} \left(\frac{\pi D_i}{4000}\right)^{0.8} \cdot 1/\psi$$

i.e. R_w is a linear function of $1/\psi$ for that range of Reynolds numbers for which the Sieder-Tate equation holds.

The steam side resistance can also vary but if it is maintained constant then the conditions required for the use of the Wilson Line Plot will be obtained i.e. R will be a linear function of $1/\psi$ and, comparing $R = R_s + R_t + R_w$ with $y = mx + c$, if R is plotted against $1/\psi$ the slope, m , of the straight line through the experimental points will be equal to $\frac{D_o}{K} \left(\frac{\pi D_i}{4000}\right)^{0.8}$ and the constant c , which will be given by the intercept by the line through the experimental points on the R (i.e. x) axis, equal to $R_s + R_t$.

For all the tubes tested the nominal values of D_o and D_i were 0.0625 ft. and 0.0545 ft. respectively;

$$\text{therefore } m = \frac{2.00(3) \times 10^{-5}}{K} = \Delta R / \Delta 1/\psi$$

Maintaining a constant steam side resistance meant maintaining constant the condenser temperature, the condensation rate, the quantity of air entering the condenser and also ensuring that the steam was not superheated. Measurements of overall heat transfer coefficients at those combinations of cooling water flow rate and temperature required to maintain the constant steam side resistance provided the required data.

A typical example of a Wilson Line Plot is given in Fig.36 which is a plot of the results of the test on tube Py 1 given in Appendix 2. Fig.37 shows the plot of a test to determine the lower limit of linearity of the Sieder-Tate formula and shows the formula to hold for Reynolds numbers greater than about 1.7×10^4 .

Many attempts have been made to use this experimental method to measure condensation heat transfer coefficients. The apparatus usually consisted of a vessel, in which the vapour was formed by boiling the liquid phase at the bottom, traversed by a water tube. The data was obtained by measuring the overall coefficient at different water velocities. The usual assumptions made were that the vapour side resistance was constant, despite the fact that

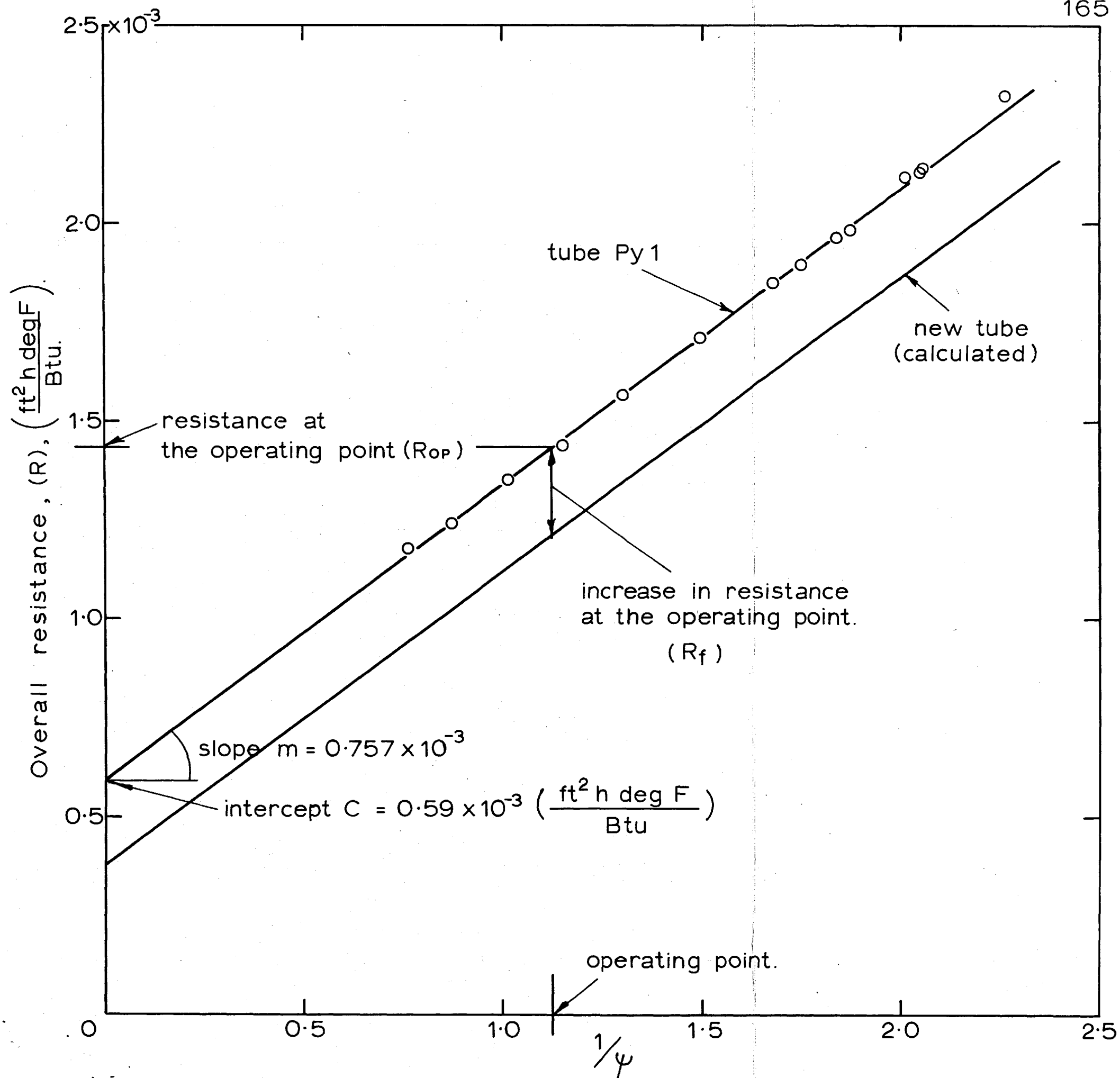


FIG. 36.

WILSON LINE PLOT. TUBE Py 1.

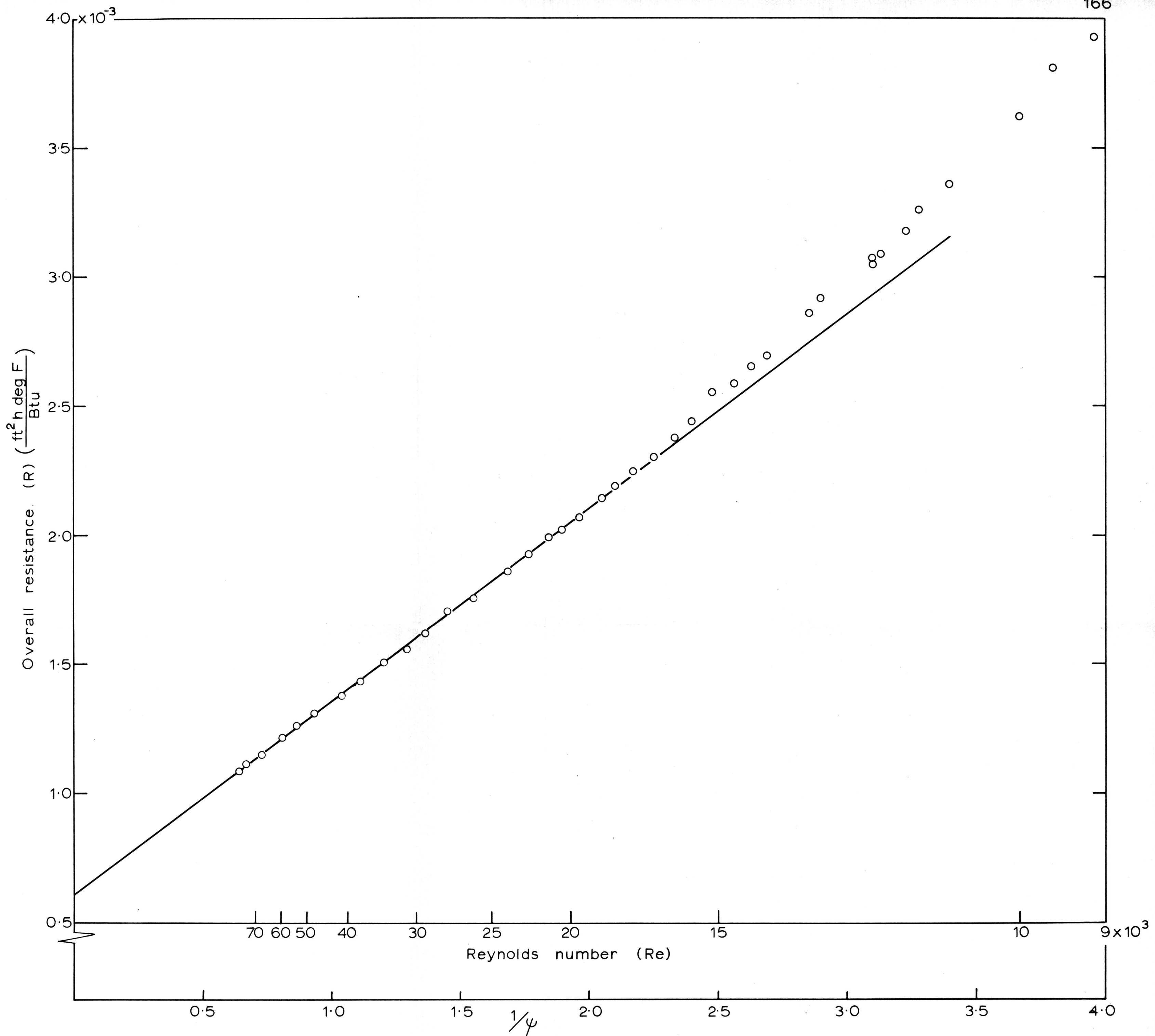


FIG. 37. WILSON LINE PLOT. DETERMINATION OF LIMIT OF LINEARITY OF SIEDER-TATE FORMULAR.

this resistance must increase with the increased condensation rate resulting from the increased water velocity, and that the water side resistance depends only on the water velocity. Intercepts obtained by drawing a line through as few as three non-linear points obtained by plotting R against $V^{-0.8}$ were used to determine the required coefficients. The relationship between the slope of the line through the experimental points and the tube dimensions does not appear to have been noticed by anyone.

The only set of experiments found for which full rigour is claimed are those of Ju Chin Chu and others¹³. In their case the variation in the properties of water were allowed for by using the McAdams formula i.e. $h_w = \text{a constant} \times (1 + 0.011T_{wm})V^{0.8}$. They also realised that the vapour side coefficient could vary. This they claimed was kept constant by maintaining a constant heat transfer rate by varying the vapour pressure, and hence the temperature difference. This ignores the effect of the condensate film temperature and the variation in the condensation enthalpy change.

The method used to maintain a constant heat flux involved operating at pressures below atmospheric, yet no mention was made of the possibility of air leakage affecting the results. Further, a secondary condenser was used so that there was a finite vapour velocity over the

heat transfer surface.

The weakest point experimentally was that they made their measurements over a 12.47 in length of 0.37/0.295 in tube.

The sum of these imperfections is shown by the difference in the slopes of some of their straight line plots Figs. 38 and 39, the significance of which was not appreciated.

No account has been found of an attempt to use the Wilson Line Plot in a manner consistent with the assumptions upon which it is based.

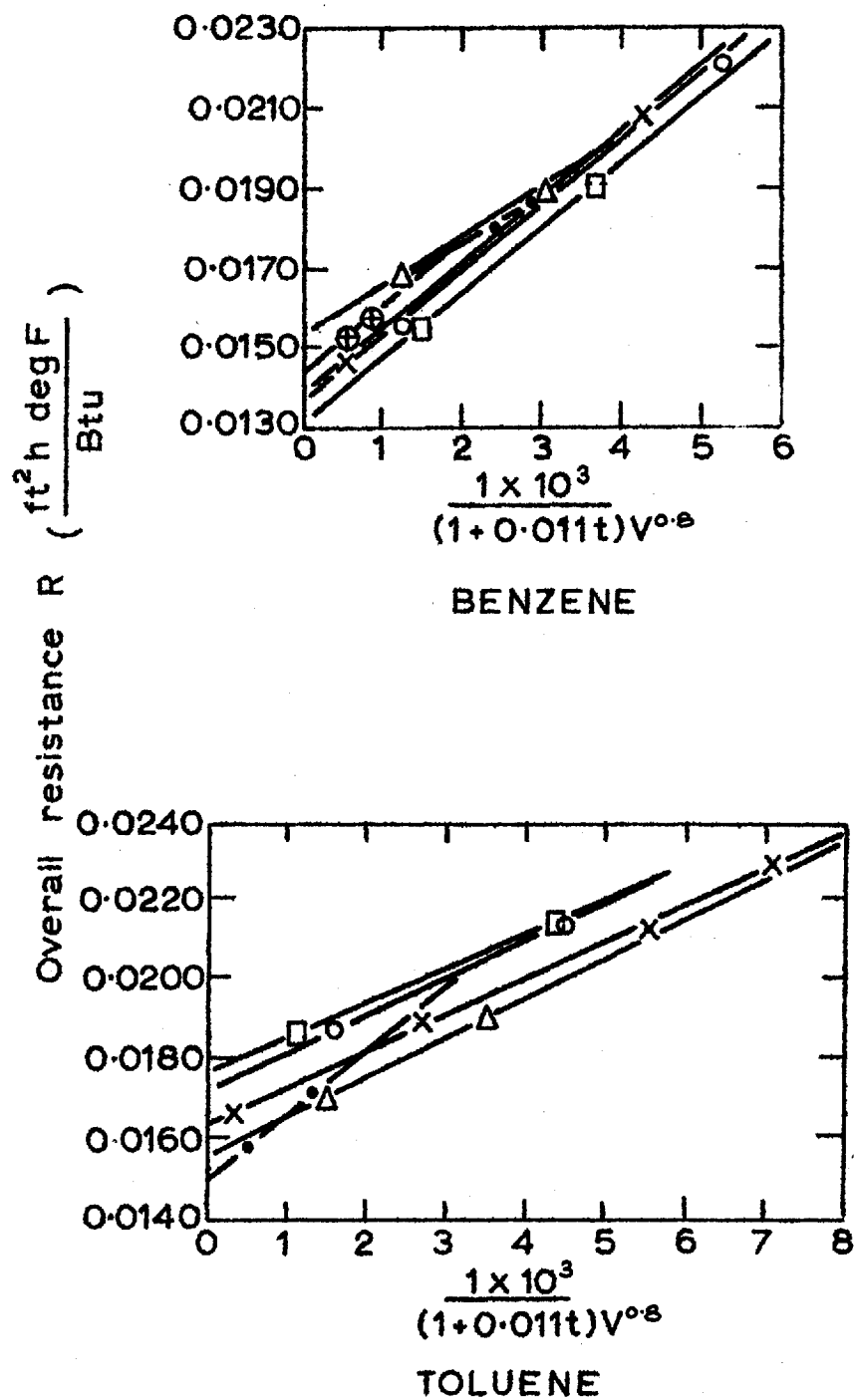


FIG. 38. WILSON LINE PLOTS OF JU CHIN CHU AND OTHERS.

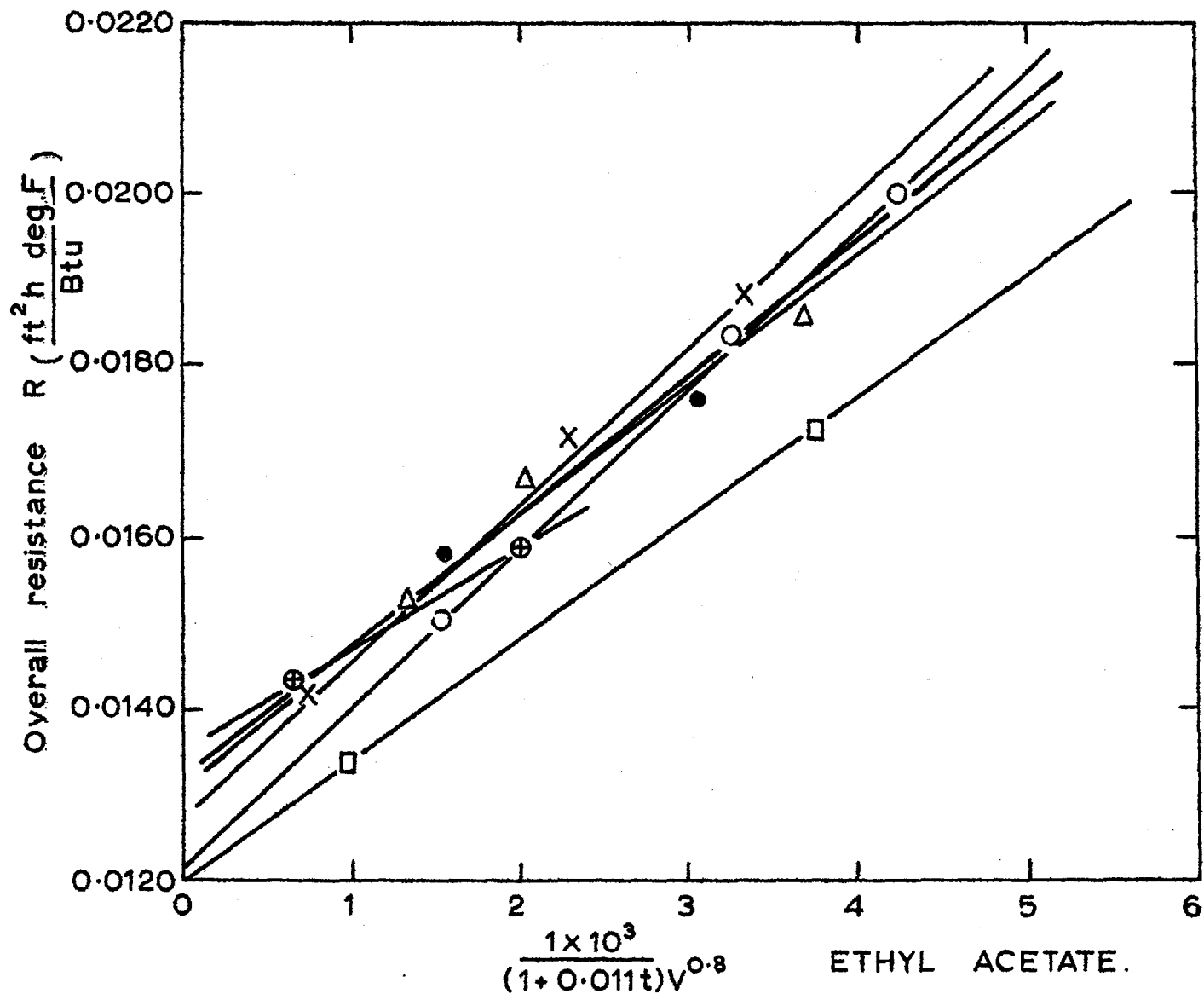


FIG. 39. WILSON LINE PLOTS OF JU CHIN CHU AND OTHERS.

APPENDIX 2.

SPECIMEN SET OF EXPERIMENTAL READINGS

AND CALCULATION PROCEDURE

DATA SHEET 1

[illegible]

DATA SHEET 2

No.	Twm	$R \times 10^3$	$(Q \times 10^{-3})^{0.8}$	$f(Twm)$	$1/\psi$	$Re \times 10^{-4}$	$f \times 10^2$
1	64.0	2.38	1.295	2.78	2.385		
2	65.9	2.425	1.265	2.75	2.395		
3	66.9	2.32	1.33	2.73	2.26		
4	68.3	2.135	1.45	2.71	2.055		
5	69.8	1.985	1.58	2.69	1.875		
6	71.2	1.895	1.68	2.67	1.75		
7	72.0	1.845	1.74	2.66	1.68		
8	73.2	1.71	1.95	2.64	1.49	2.74	0.740
9	74.5	1.565	2.21	2.62	1.305	3.21	0.690
10	75.8	1.435	2.50	2.61	1.15	3.74	0.66
11	77.0	1.355	2.81	2.59	1.015	4.34	0.651
12	78.1	1.24	3.27	2.58	0.87	5.21	0.649
13	78.8	1.18	3.71	2.57	0.76		
14	66.5	2.46	1.235	2.74	2.445		
15	68.3	2.13	1.455	2.71	2.05		
16	69.2	2.115	1.475	2.70	2.01		
17	70.1	1.965	1.61	2.69	1.84		

CALCULATION PROCEDURE

The following calculation is for experimental point No.12.
For the plot comparing this test with that calculated for a clean tube, see Fig. 36.

Cooling water temperatures:	$T_{w1} = 76.62^{\circ}\text{F}$	$T_{w2} = 79.56^{\circ}\text{F}$
Calibration corrections	+0.05	-0.12
(from Table 2)	76.67 $^{\circ}\text{F}$	79.44 $^{\circ}\text{F}$
Cooling water temperature rise	ΔT_w	= 2.77 $^{\circ}\text{F}$
Pressure correction		+ 0.04 $^{\circ}\text{F}$
$\therefore \Delta T_w$ corrected		= 2.81 $^{\circ}\text{F}$
Condenser temperature	T_c (corrected)	= 91.1 $^{\circ}\text{F}$
Time taken for depth of water in measuring tank to increase by 27 cm	τ	= 108 s
Cooling water flow rate (Fig. 9.)		4,390 lb/h
Log mean temperature difference, $\ln (T_c - T_w)$		13.0 $^{\circ}\text{F}$
(obtained from a Nomogram)		
Overall heat transfer coefficient, h_{sw}	$= \frac{Q \times \Delta T_w}{A \times \ln (T_c - T_w)}$	
	$= \frac{4,390 \times 2.81}{1,177 \times 13.0} = 806 \text{ Btu/ft}^2\text{h deg F}$	
Overall resistance, R	= $1.24 \times 10^{-3} \text{ ft}^2\text{h deg F/Btu}$	
Mean cooling water temperature	$T_{wm} = 78.1^{\circ}\text{F}$	

$$f(T_{wm}) \text{ (Fig. 40)} = 2.58$$

$$\mu_s^{0.14} \text{ (see later for explanation)} = 1.10$$

$$(Q/1000)^{0.8} = 4.39^{0.8} = 3.27$$

$$\therefore 1/\psi = \frac{T_{wm} \times \mu_s^{0.14}}{(Q/1000)^{0.8}} = \frac{2.58 \times 1.10}{3.27} = 0.87$$

The mean inner surface temperature of the tube, T_{si} , was found by assuming $\mu_s^{0.14}$ to be unity and calculating and plotting sufficient points to estimate the intercept on the overall resistance axis. Multiplying this resistance by the heat transfer rate per unit external surface area gave $(T_c - T_{si})$ and hence T_{si} and μ_s .

For the test from which this example is taken the estimated intercept was $0.6 \times 10^{-3} \text{ ft}^2 \text{ h deg F/Btu}$

$$\therefore (T_c - T_{si}) = \frac{0.6 \times 10^{-3} \times 12,300}{1.177} = 6.3 \text{ deg F}$$

$$\therefore T_{si} = 91.2 - 6.3 = 84.9 \text{ deg F}$$

$$\therefore \mu_s = 1.98 \text{ lb/ft h and } \mu_s^{0.14} = 1.10$$

The slope, m , of the best straight line through the experimental points was found using "the least mean square" method, to be 757×10^{-4} giving a value of K of 0.0264. By calculation the intercept on the overall resistance axis was 0.59 and the overall resistance at the operating point $(1/\psi = 1.125) = 1.435 \times 10^{-3} \text{ ft}^2 \text{ h deg F/Btu}$.

The Fanning Friction Factor, f , and Reynolds

number, Re , were calculated for the experimental points near the operating point and from a plot of f against Re the value of f at $Re = 3.3 \times 10^4$ obtained.

To calculate the Reynold's number the viscosity at Tsi was used after experiments had shown that this gave the same $Re - f$ relationship as for isothermal flow at Tsi.

Data Sheet 1 was completed in the laboratory as the test proceeded. The arithmetic involved was not checked unless the plot showed a large deviation from the best straight line through the bulk of the experimental points. Scatter of the experimental points was thus also due to small arithmetical inaccuracies as well as variations in the condenser conditions, errors in temperature measurements, etc.

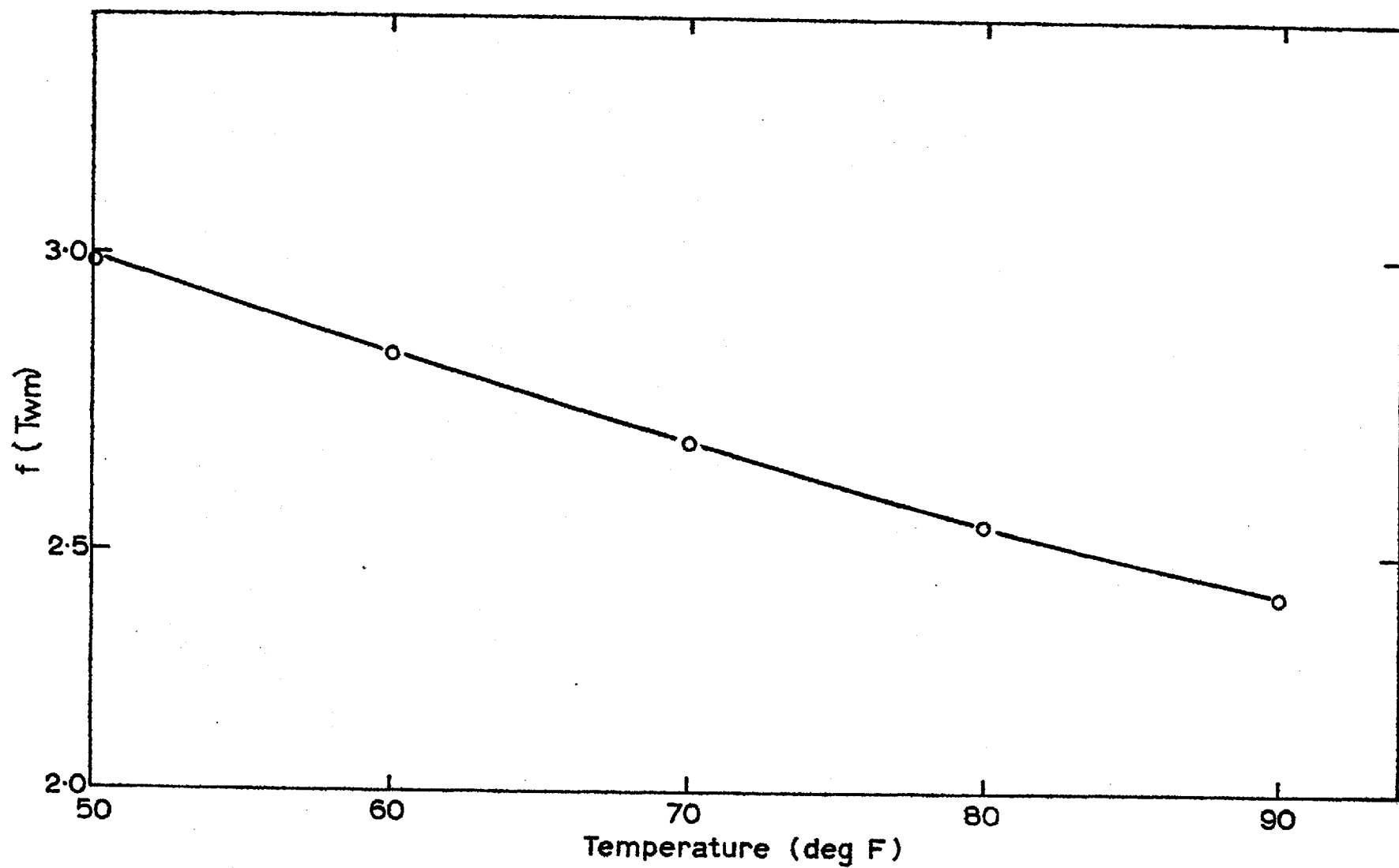


FIG. 40.

$f(T_{wm})$ v TEMPERATURE.

APPENDIX 3.

The following questionnaire was sent to all the shipping firms supplying used condenser tubes.

B. S. R. A. Research Item M. 8.

<u>Used Condenser Tubes</u>	<u>Information Required</u>
1. Condenser Tube	(a) Tube size and gauge (b) Tube material (c) Length of service - in months if possible.
2. Details of Condenser	(a) Type (b) Working Pressure (c) Nominal water velocity (d) Position of tube in condenser (e) Details of any cooling water treatment, and of any corrosion and scale inhibiting devices. (f) Normal route of vessel and proportion of time condenser is shut down.
3. Cleaning	(a) Number of times tube has been cleaned or the normal condenser maintenance programme. (b) Method of cleaning (1) Internal. (2) External.

It is realized that all this information may not always be available, but until it is known which of the variables are important, the more information that can be obtained the more valuable any specimen will be.

Item (2d) will most easily be given by a sketch showing the position of the tube in relation to the cooling water inlet and outlet, steam inlet, condenser drain and air extraction pump.

Where more than one tube is available from any one source, some system of tube numbering will have to be used to connect individual tubes with the relevant information.

An $8\frac{1}{2}$ ft length of tube is to be preferred as this will allow for an effective 6 ft test length. If shorter lengths only are available the test length will have to be reduced to 4 ft.

For purposes of comparison, it will be desirable to have a new tube of the same type as any used tubes supplied.

If a relatively short test length is cut from a long tube Item (2d) should indicate which portion of the tube is provided for test.

As the frequency with which the ships engines are stopped and started may affect the rate of condenser tube fouling, a brief resume of engine operating procedure from start to finish of a voyage would be useful. If the ships engines are started, or 'warmed up' on low vacuum this information could be of considerable importance.

APPENDIX 4EFFECT OF CONDENSER TUBE DEPOSITS
ON MARINE POWER PLANT PERFORMANCE

Specimen of calculation used to obtain the data
for Figs. 28 to 35.

Initial condenser conditions

Condenser pressure, P_c ,	28.5 in Hg. Vac
Overall heat transfer coefficient, h_{sw}	800 Btu/ft ² h deg F
∴ Overall thermal resistance, $R = 1.25 \times 10^{-3}$	ft ² h deg F/Btu
Cooling water temperatures. Inlet, T_{w1}	60°F
Outlet, T_{w2}	80°F
Condenser temperature, T_c	92°F
(Saturation temperature at P_c)	
∴ Inlet temperature difference, $\theta_1 =$	32 deg F
∴ Outlet temperature difference, $\theta_2 =$	12 deg F
∴ Log mean temperature difference, $\ln(T_c - T_w) =$	20.4 deg F
∴ Heat transfer rate $q'' = 800 \times 20.4 =$	16,320 Btu/ft ² h

Assume a new condenser pressure of 27.97 in Hg vac.

When the condenser vacuum has fallen to 27.97 in Hg ^v Vac
the heat transfer rate will have increased 3.8 per cent (from
Fig. 27), and the condenser temperature to 101.7 °F. (the

increase in the heat transfer rate is due to an increase of 3.4 per cent in the specific steam consumption and of 0.4 per cent to the increase in the enthalpy of the steam entering the condenser).

$$\therefore \dot{q}'' = 16,320 \times 1.038 = 16,950 \text{ Btu/ft}^2\text{h}$$

$$\text{and } T_{w2} = 60 + 20 \times 1.038 = 80.7^\circ\text{F}$$

$$\therefore \theta_1 = 101.7 - 60 = 41.7 \text{ deg F}$$

$$\text{and } \theta_2 = 41.7 - 20.7 = 21.0 \text{ deg F}$$

$$\therefore \ln (T_c - T_w) = 30.2 \text{ deg F}$$

$$\therefore \text{Overall resistance} = 30.2/16,950 = 1.78 \times 10^{-3} \text{ ft}^2 \text{ h deg F / Btu}$$

$$\therefore \text{Increase in resistance} = 1.78 - 1.25 = 0.53 \times 10^{-3} \text{ ft}^2\text{h deg F/Btu}$$

This is the increase in resistance due to tube deposits during a tube life of between 25 and 49 months (from Fig. 18),

Therefore, depending on the fouling rate, after a life of between 25 and 49 months the condenser vacuum will have fallen from 28.5 to 27.97 in Hg Vac and the specific steam consumption of the power plant will have increased by 3.4 per cent. (From Fig. 26).

APPENDIX 5ANALYSIS OF DEPOSITS FROM INTERNAL SURFACES
OF CONDENSER TUBES FROM H. M. S. COCKADE

Silicon as SiO_2	6.8 per cent
Copper as CuO	17.2 " "
Iron as Fe_2O_3	6.8 " "
Nickel as NiO	14.0 " "
Magnesium as MgO	1.0 " "
Chlorides as NaCl	2.3 " "
Nitrates }	a trace
Phosphates }	
Water soluble matter	6.5 per cent
Calcium as CaCO_3	Remainder

The tubes from which this sample was obtained were made of cupro-nickel and had been "in service" for five years.

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SYMBOLS

A	Area	ft ²
B, B'	Constants	
Do	External Diameter	ft
Di	Internal Diameter	ft
K	Constant in the Sieder-Tate formula	
L	Evaporation enthalpy change	Btu/lb
Pc	Condenser pressure	in Hg
Pw ₁	Cooling water inlet pressure	cm H ₂ O
Pw ₂	Cooling water outlet pressure	cm H ₂ O
Q	Cooling water mass flow rate	lb/h
R	Overall thermal resistance	ft ² h deg F/Btu
Rop	Overall thermal resistance at the operating point	ft ² h deg F/Btu
Rc	Overall thermal resistance of a clean heat exchanger	
Ra	Thermal resistance due to air in condensing steam	)
Rf	Thermal resistance due to fouling deposits at the operating point	) (ft ² h deg F/Btu per unit external surface area)
Rs	Thermal resistance at the condensation surface	)
Rt	Thermal resistance of the tube wall	)
Rw	Thermal resistance at the internal wetted surface	)
T	Temperature	°F
Tc	Condenser temperature	°F
Tsi	Internal wetted surface temperature	°F
Tso	External tube surface temperature	°F

T_w	Cooling water temperature	$^{\circ}\text{F}$
T_{wm}	Arithmetic mean cooling water temperature	$^{\circ}\text{F}$
$\ln(T_c - T_w)$	Logarithmic mean temperature difference	deg F
T_{w1}	Cooling water inlet temperature	$^{\circ}\text{F}$
T_{w2}	Cooling water outlet temperature	$^{\circ}\text{F}$
V	Velocity	ft/s
c	Intercept of the Wilson Line Plot on the overall resistance axis	$\text{ft}^2 \text{ h deg F/Btu}$
f	Fanning friction factor	
h_{sw}	Overall heat transfer coefficient	$\text{Btu/ft}^2 \text{ h deg F}$
h_{op}	Overall heat transfer coefficient at the operating point	$\text{Btu/ft}^2 \text{ h deg F}$
h_s	Heat transfer coefficient at outer surface	$\text{Btu/ft}^2 \text{ h deg F}$
h_w	Heat transfer coefficient at inner surface	$\text{Btu/ft}^2 \text{ h deg F}$
k	Thermal conductivity	Btu/ft h deg F
m	Slope of Wilson Line Plot	
$\dot{q}$	Heat transfer rate	Btu/h
$\dot{q}''$	Heat transfer rate per unit area	$\text{Btu/ft}^2 \text{ h}$
t	Time	h

τ	Time taken for water level to rise between standard depths in measuring tank	s
μ	Viscosity	lb/ft h
μ_s	Viscosity at internal wetted surface temperature	lb/ft h
ρ	Density	lb/ft ³
ψ	Defined on page 163	
Nu	Nusselt Number	
Re	Reynolds Number	
Pr	Prandtl Number	

Other symbols as per B.S. 1991.