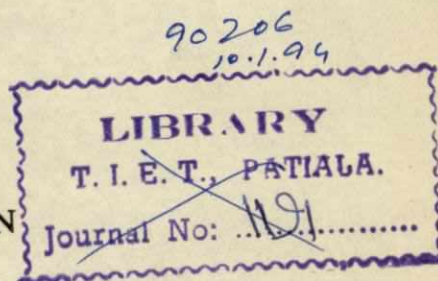


Evaluation of Overall Heat Transfer Coefficient for Feedwater Heaters of A Modern Steam Power Station

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ABSTRACT

In this work, regenerative feedwater heating cycle of one of the units of an existing 4 x 110 MW, Coal-Fired Steam Power Station has been analysed with a view to evaluating analytically overall heat transfer coefficient for various low pressure and high pressure feedwater heaters under full load rated steady-state conditions.

Break up of total thermal resistance to heat transfer into various resistances on steam side, water side and through tube wall has also been carried out and commented upon.

It has been concluded that the overall heat transfer coefficient is different for different feedwater heaters, being somewhat lower for high pressure heaters. It has further been inferred that in case of feedwater heaters, unlike surface condensers, water side resistance may not be the controlling resistance; rather steam side resistance and sometimes even the tube wall resistance may be major/controlling resistance to heat transfer.

The work is based primarily on various theoretical formulations available for convective/condensing heat transfer phenomenon and the results agree well with the figures assumed by the manufacturer.



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NOMENCLATURE

<u>Symbol</u>	<u>Explanation</u>	<u>Units</u>
A, A_i, A_o	Heat Transfer Area, Water Side, Steam Side	m^2
$A_c, (A_o)_c$	Steam Side Heat Transfer Area of Condensing Section	m^2
$A_d, (A_o)_d$	Steam Side Heat Transfer Area of Desuperheating Section	m^2
A_f	Feedwater Flow Area	m^2
a_o, a_i	Constants of Wilson Plot	-
$^{\circ}C$	Unit of Temperature (Celsius Scale)	$^{\circ}C$
C_{p_w}, C_v	Specific Heat of Water/Vapour at Constant Pressure	Kcal/Kg $^{\circ}C$
cm	Unit of Length (Centimetre)	cm
D	Mean Diameter of Helical/Spiral Coil	m
D_e	Equivalent or Hydraulic Diameter	m
D_i, ID	Inside Diameter of Heater Tubes	m
D_o, OD	Outside Diameter of Heater Tubes	m
F	LMTD Correction Factor	-
ΔT	Wall Temperature Variation Correction Factor	-

<u>Symbol</u>	<u>Explanation</u>	<u>Units</u>
f_1	Ratio of Uniform Heat Flux/ Uniform Wall Temperature Nusselt Number	-
f_2, f_3	Flow Development/Tube Curvature Correction Factor	-
GTD	Greater Temperature Difference	$^{\circ}\text{C}$
g	Acceleration Due to Gravity	m/s^2
HP, HPH	High Pressure, High Pressure Heater	-
h	Specific Enthalpy	Kcal/Kg
$h_{fg}, h_{fg}^i, h_{fg}^u$	Latent Heat of Vapourization of Water	Kcal/Kg
h_1, h_{11}	Water Side Heat Transfer Coeff., by check calculation	Kcal/hr m^2C
h_m, h_m^i, h_m^u	Mean Condensing Heat Transfer Coefficient	Kcal/hr m^2C , Kcal/ $\text{s m}^2\text{C}$
h_o, h_{oDe}	Steam Side Heat Transfer Coeff., based on Equivalent Diameter	Kcal/hr m^2C
hr	Unit of Time (hour)	hr
h_s, h_{sat}, h_{sup}	Specific Enthalpy of Steam/ Saturated/Superheated Steam	Kcal/Kg
h_{s1}, h_{s2}	Specific Enthalpy of Steam at Inlet/Exit Conditions	Kcal/Kg
h_{w1}, h_{w2}	Specific Enthalpy of Feedwater at Inlet/Exit Conditions	Kcal/Kg

<u>Symbol</u>	<u>Explanation</u>	<u>Units</u>
h_{wc}, h_{wdr}	Specific Enthalpy of Condensate/ Drip or Drain	Kcal/Kg
h_{wc1}	Specific Enthalpy of Feedwater at Inlet to Zone I of Condens- ing Section	Kcal/Kg
h_{wd1}	Specific Enthalpy of Feedwater at Inlet to Zone II of Condens- ing Section	Kcal/Kg
h_{wc2}, h_{wd2}	Specific Enthalpy of Feedwater at Exit from Condensing/Desuper- heating Section	Kcal/Kg
Kcal	Unit of Heat (Kilocalorie)	Kcal
Kg, Kgf	Unit of Mass, Force (Kilogram, Kilogram force)	Kg, Kgf
k_f, k_f'	Thermal Conductivity of Fluid	Kcal/s m °C Kcal/hr m °C
k'_{wb}	Thermal Conductivity of Feed- water at t_{bw}	Kcal/hr m °C
k'_{wl}	Thermal Conductivity of Tube Wall at Mean Wall Temperature	Kcal/hr m °C
KWh	Unit of Energy (Kilowatt Hour)	KWh
L	Tube Length	m
LP, LPH	Low Pressure, Low Pressure Heater	-
LTD	Lesser Temperature Difference	°C

<u>Symbol</u>	<u>Explanation</u>	<u>Units</u>
LMTD, LMTD'	Logarithmic Mean Temperature Difference	$^{\circ}\text{C}$
(LMTD) _c	LMTD of Condensing/Desuperheating Section	$^{\circ}\text{C}$
(LMTD) _d		
(LMTD) _{cI}	LMTD of Zone I/Zone II of Condensing Section	$^{\circ}\text{C}$
(LMTD) _{cII}		
ln	Natural Logarithm	-
MW	Unit of Power (Megawatt)	MW
m	Unit of Length (Metre)	m
$\dot{m}^s, \dot{m}_s^s, \dot{m}_w^s$	Mass Flow Rate, of Steam, of Water, of Drip/Drain	Kg/hr
$\dot{m}_{wdr}^s$		
$\dot{m}_{wc}^s, \dot{m}_{wd}^s$	Mass Flow Rate of Feedwater in Condensing/Desuperheating Section	Kg/hr
NA	Not Applicable	-
Nu, Nu_{∞}	Nusselt Number, in fully Developed Region	-
Nu_x	Local Nusselt Number in the Entry Region	-
Nu_H, Nu_H^s	Uniform Heat Flux Nusselt Number	-
Nu_T, Nu_T^s	Uniform Wall Temperature Nusselt Number	-
Nu_{DeH}	Nu_H Based on Equivalent Diameter	-
n	Number of Tubes	-

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<u>Symbol</u>	<u>Explanation</u>	<u>Unit</u>
P, P_s, P_w	Absolute Pressure, of Steam, of Feedwater	Kgf/cm ²
P_i	Tube Pitch or Spacing	m
Pr, Pr_f	Prandtl Number, at Film Temperature	-
$Q, (Q)_c, (Q)_d$	Heat Transfer Rate, of Condensing/Desuperheating Section	Kcal/hr
$(Q)_{cI}, (Q)_{cII}$	Heat Transfer Rate of Zone I/II of Condensing Section	Kcal/hr
$ Q _{m, wdr} = 0$	Heat Transfer Rate excluding Drip Enthalpy	Kcal/hr
R	Heat Capacity Ratio of Tube Fluid/Shell Fluid	-
Re, Re_D	Reynolds Number, Based on Diameter	-
Re_{De}, Re_f	Reynolds Number based on Equivalent Diameter, on Film Temperature	-
Re_h	Reynolds Number for Condensate Flow Outside Horizontal Tubes	-
R_o	Overall Thermal Resistance based on Outside Area	hr m ² °C/Kcal
R_o'	Characteristic Gas Constant	m Kgf/Kg °C
r_d, r_d'	Total Fouling Resistance of Both Surfaces of Tubes based on Steam Side Area with/without Drip Enthalpy	hr m ² °C/Kcal

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<u>Symbol</u>	<u>Explanation</u>	<u>Units</u>
$(r_d)_c + (r_d)_d$	Condensing/Desuperheating Section Total Fouling Resistance based on Steam Side Area	hr m ² °C/Kcal
$r_i + r_{io}$	Water Side Thermal Resistance, based on Steam Side Area	hr m ² °C/Kcal
$r_{id} + r_{ido}$	Water Side Fouling Resistance, based on Steam Side Area	hr m ² °C/Kcal
r_o	Steam Side Thermal Resistance	hr m ² °C/Kcal
r_{od}	Steam Side Dirt Coefficient	hr m ² °C/Kcal
r_{wl_o}	Tube Wall Thermal Resistance based on Steam Side Area	hr m ² °C/Kcal
S	Thermal Effectiveness	-
s	Unit of Time (second)	s
T, t	Shell/Tube Fluid Temperature (for Equation 3 only)	°C
TTD	Terminal Temperature Difference	°C
$t_{bs} + t_{bw}$	Bulk - Mean Temperature of Steam/Feedwater	°C
$t_f + t_{ref}$	Film/Reference Temperature	°C
$t_s + t_{sat}$	Heating Steam/Saturated Steam Temperature	°C
$t_{s_1} + t_{sup}$	Inlet/Superheated Steam Tempera- ture	°C
t_{s_2}	Steam Temperature at Desuper- heating Section Exit	°C

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<u>Symbol</u>	<u>Explanation</u>	<u>Units</u>
t_{w1}, t_{w2}	Feedwater Inlet/Exit Temperature	$^{\circ}\text{C}$
t_{wc1}	Feedwater Temperature at Inlet to Zone I of Condensing Section	$^{\circ}\text{C}$
t_{wd1}	Feedwater Temperature at Inlet to Zone II of Condensing Section	$^{\circ}\text{C}$
t_{wc2}, t_{wd2}	Feedwater Temperature at Exit from Condensing/Desuperheating Section	$^{\circ}\text{C}$
t_{wl1}, t_{wl0}	Average Tube Wall Temperature on Water/Steam Side	$^{\circ}\text{C}$
Δt	Temperature Difference between Shell/Tube Fluid	$^{\circ}\text{C}$
$\Delta t_i, \Delta t_o$	Water/Steam Side Average Temperature Drop	$^{\circ}\text{C}$
Δt_m	Mean Temperature Difference	$^{\circ}\text{C}$
Δt_{wl}	Temperature Drop in Tube Wall	$^{\circ}\text{C}$
U, U_o	Overall Heat Transfer Coefficient, based on Steam Side Area	$\text{Kcal/hr m}^2 \text{ }^{\circ}\text{C}$
U_{cl}, U_{clo}	Clean Overall Heat Transfer Coefficient, based on Steam Side Area	$\text{Kcal/hr m}^2 \text{ }^{\circ}\text{C}$
U_D, U_{Do}	Design or Dirty Overall Heat Transfer Coefficient, based on Steam Side Area	$\text{Kcal/hr m}^2 \text{ }^{\circ}\text{C}$
U'_{Do}	U_{Do} excluding Drip Enthalpy	$\text{Kcal/hr m}^2 \text{ }^{\circ}\text{C}$

<u>Symbol</u>	<u>Explanation</u>	<u>Units</u>
$(U_{Clo})_c$, $(U_{Clo})_d$	U_{Clo} for Condensing/Desuperheating Section	Kcal/hr m^2 $^{\circ}C$
$(U_{Do})_c$, $(U_{Do})_d$	U_{Do} for Condensing/Desuperheating Section	Kcal/hr m^2 $^{\circ}C$
U_{Supp}	Value of U assumed by Supplier of Heaters	Kcal/hr m^2 $^{\circ}C$
V_w	Volumetric Flow Rate of Water	m^3/s
v_w	Specific Volume of Feedwater	m^3/Kg

GREEK LETTERS

ρ_v, ρ_f	Density of Steam/Condensate at t_{ref}	Kg/m^3
μ, μ_f	Dynamic Viscosity, at t_f	Kg/ms
μ_b, μ_{wl}	Dynamic Viscosity of Water at t_{bw}/t_{wl_1}	Kg/ms
ν_b, ν_f	Fluid Kinematic Viscosity at Bulk-Mean/Film Temperature	m^2/s
T	Condensate Mass Flow Rate per Unit Dimension	Kg/ms
T_h	T per Unit Length (for Horizontal Tubes)	Kg/ms
T_v	T at the Lowest Point per Unit Perimeter (for Vertical Tubes)	Kg/ms



CHAPTER 1

INTRODUCTION

1.1 REGENERATIVE FEEDWATER HEATING (1)

There is hardly any Steam Power Station today that does not make use of Regenerative Feedwater Heating in which boiler feedwater is heated with steam extracted after it has partially expanded in the turbine. It is because of the well recognized principle that Regenerative Cycle has a higher thermal efficiency than that of the basic Rankine Cycle and the improvement increases as the number of feed heating stages increases. Feedwater heating is carried out by means of equipment known as Feedwater Heaters. FIGURE 1 (A, B & C) shows that each additional feedwater heater adds a progressively small increment of efficiency and theoretically, an infinite number of heaters will raise the efficiency of the Regenerative Rankine Cycle to that of Carnot Cycle. It is observed that as the number of heaters tends to infinity, temperature difference between the heating steam and the incoming feedwater tends to zero so that water heating process approaches 'reversibility'. However, because of inherent irreversibility of superheating process, the ideal Carnot Cycle efficiency is not achieved. Nevertheless, the infinite heater cycle is of academic interest because it demonstrates the basic principle of regenerative feedwater heating process.

For five to seven stage feedwater heating cycle, a decrease in turbine heat input per unit electrical energy output of the order of nine to fourteen percent has been reported (1). These figures, though very impressive as it appears to be,

cannot be taken in isolation as adoption of the system of regenerative feedwater heating necessitates increased capital investment and running costs so that a balance has to be struck between the incremental increase in efficiency and the additional costs involved in achieving the same. Moreover, flue gases leaving the boiler economiser tend to have a higher temperature, thus inevitably reducing the boiler efficiency to an extent which could largely counterbalance the increase in turbine thermal efficiency. It is because of this reason that Air Preheater are also incorporated in modern boiler design and this again calls for increased capital investment and running costs.

Faras and Ozisk (2) point out that regenerative feedwater heating serves also to reduce the main condenser size because the condensers for large steam turbines become so large that installation problems become awkward indeed. Thus having about a third of steam flow bled to feedwater heaters is a real help in this respect. The cost of low pressure end of the turbine is also reduced (1).

1.2 OPTIMISATION OF FEEDWATER HEATERS

In the recent past, due to increased cost of both power plant equipment and of energy resources, particularly of fossil fuels, considerable interest has been shown by research workers to the application of various optimisation techniques to the design of feedwater heaters. Salisbury (3) has described optimisation of heater design conditions of each individual heater of a six-heater nuclear power plant and reported a reduction in turbine heat rate of about 6.8 Kcal/Kwh and a net

increase in power output of about 2.5 MW in the 880 MW nuclear power plant as a result of application of the optimisation techniques to the heater design. He has assumed the same average value of overall condensing heat transfer coefficient for each heater ($3175 \text{ Kcal/hr m}^2 \text{ } ^\circ\text{C}$) and is of the opinion that this coefficient will be different for different heaters which will affect the results.

Gopinathan (4) has demonstrated the technique of 'dynamic programming' to be quite useful in determining the optimal distribution of feedwater heater surface areas in steam power plants. He has also assumed the same average value of overall heat transfer coefficient for all the seven heaters in the feedwater heater chain which is $1800 \text{ Kcal/hr m}^2 \text{ } ^\circ\text{C}$ and $720 \text{ Kcal/hr m}^2 \text{ } ^\circ\text{C}$ for the condensing and desuperheating sections respectively. He has clearly stated that this is one of the limitations of his work and suggested that better results would be available by using the actual values of overall heat transfer coefficients for various heaters.

1.3 OBJECTIVES AND SCOPE OF THE PRESENT WORK

The widely varying/contradicting values of overall heat transfer coefficient assumed by Salisbury (3) and Gopinathan (4) in their respective analysis have led the author to analyse the feedwater heater chain of an existing modern steam power station with a view to determining, analytically, the average value of overall heat transfer coefficient for each individual heater.

The plant under study is the 4 x 110 MW Guru Nanak Dev

Thermal Plant, Bhatinda (Punjab) owned and operated by Punjab State Electricity Board. All the four units are of M/s Bharat Heavy Electricals Limited make and are identical in design. The units are coal based with oil firing at part - load operation and/or for flame stabilization. Each unit is a single - reheat set with the principal steam parameters as 139 Kg/cm^2 (abs.)/ 535°C at turbine stop-valve, 36.4 Kg/cm^2 / 360°C at reheater input and 33.8 Kg/cm^2 / 535°C at reheater outlet. Rated steam output is 375 tonne/hr at the superheater outlet and the boiler drum conditions are 160 Kg/cm^2 / 345°C . Feedwater is heated to about 232°C , before its entry to the economiser, by eight non-regulated extractions from the steam turbine. In all, about 96.2 tonne/hr i.e., about 25 percent of total steam generation is used as bled steam for feedwater heating. This figure excludes about 8 tonne/hr of turbine gland steam which is also used for feedwater heating. There are five LP heaters, one deaerator and two HP heaters in the feed-heat cycle. Whereas extractions for the first three LP heaters are taken from the LP cylinder of the turbine, the next four extractions are taken from the medium pressure cylinder or casing of the turbine while extraction for the last heating stage is taken from the 'cold reheat' line i.e., from the steam main connecting exhaust from the HP casing of the turbine to the reheater inlet.

The present work is restricted to evaluating the average value of overall heat transfer coefficient for the various feedwater heaters, excluding the deaerator, at rated full load of 110 MW and under design conditions with all the heaters

'charged' i.e., in circuit. No calculation have been presented for their behaviour at part load, although a few remarks have been included on the basis of a separate exercise carried out by the author. Breakup of various resistances to heat transfer from steam to water has also been carried out and commented upon. Values of individual film coefficients have also been presented and compared with those available in relevant literature. The author was, however, unable to lay hands on consistent field data in the absence of which the actual values of overall heat transfer coefficient could not be compared with those determined analytically.

The work is based primarily on theoretical formulations and various assumptions made have been brought out in relevant sections.



CHAPTER 2

LITERATURE REVIEW

2.1 FEEDWATER HEATERS

2.1.1 TYPES

There are two basic types of feedwater heaters in service namely, the 'Direct Contact Type' and the 'Surface Type' (1, 2, 3). This classification is well known and needs no elaboration. A typical example of the direct contact type heater is a Deaerator and of the surface type heater is the common LP or HP heater. Use of segmental baffle plates or tube-support plates in surface type heaters is popular as this arrangement increases turbulence, allows thermal expansion of tubes and helps in directing the flow of steam through the tube-nest (1). A special advantage of using baffle plates for vertical units has been reported (1) in that these can be arranged also to strip the steam condensate at intervals as it runs down the outside of long vertical tubes. The formation of thick film and an increase in resistance to heat transfer is thereby prevented.

Singh (6) has described 'Coil Type' design of HP feedwater heaters. This design completely eliminates dish type headers and tube nests. In this design, spirals of tubes are welded to long cylindrical headers and in each plane, there are four coils. The coils in each column are connected to an inlet header and an outlet header, arranged adjacent to each other.

In a feedwater heater chain, a particular heater is referred to as 'Low Pressure' (LP) Heater or 'High Pressure' (HP) Heater depending upon whether it is located, in the heater chain, before the boiler feed pump or after it (1). After the LP Heaters, a deaerator is arranged in an elevated position on the suction side of the boiler feed pump so as to ensure a positive suction head under all conditions of service (1).

Bled steam condensing outside the tube surfaces of feedwater heaters, popularly known as drip or drain, is usually, though not always, cascaded to the next lower heater in the chain (1). References (1,5) describe other methods also.

2.1.2 TUBE MATERIAL

The material for LP heaters is usually non-ferrous because of obvious benefit of better thermal conductivity and also because of the fact that appreciable quantity of air is present in LP heaters operating under vacuum so that rusting could occur if ferrous material were used (1). Admiralty metal (70% Copper, 4% arsenic, 1.25% tin and remaining zinc), Cupro - Nickel Alloys and Aluminium Brass are the commonly used materials for LP heater tubes (1).

In early sixties, an industry-wise trend towards carbon steel feedwater heaters began (7). Experiments by Morris and Phillips (2) demonstrated the acceptability

of carbon steel for feedwater heater services, particularly for the HP heaters. Experiments at the Research Laboratories of M/s Hitachi Ltd., Japan (8) also demonstrated that the difference in overall heat transfer coefficient for the steel tubed heaters and for the non-ferrous heaters was not much. Decidedly, carbon steel is the material most commonly used for high pressure/temperature heater tubes as it facilitates the use of welding for tube joints and also simplifies the chemical control of feedwater in plants in which the major surfaces are ferrous (1,9). Faras and Ozisk (2) report that in cases where acid-cleaning of feedwater heater tubes may be necessary, stainless steel tubes are used.

2.1.3 PERFORMANCE CHARACTERISTICS

It has been established (1) that optimum efficiency is achieved in all heaters with non-reheat cycle and in the heaters below the reheat point on reheat sets, when equal enthalpy rise takes place over each heater. Fenton (5) has written that the last heater in a reheat set should have an enthalpy rise of about 1.8 times the rise of other heaters in order to obtain the best heater spacing. In actual practice, however, the distribution of enthalpy rise between various heaters and the number of feed heating stages is a compromise between the theoretical or economic optimum and the practical optimum (1).

Thermal performance of feedwater heaters is usually

gauged by the so called 'TTD' (terminal temperature difference) which is the difference in temperature between the water leaving the heater and saturation temperature corresponding to the heater shell steam pressure (1.5). Lower the TTD, better it is, although it requires much more additional surface area to lower TTD and the lower limit of TTD is governed by the necessary temperature gradient required for heat flow by conduction from the condensate film formed on the outside surface of tubes (1).

As live steam temperatures have gone up and the steam is reheated, bled steam temperatures in HP heaters are quite high (1). Fenton (5) has written that this superheat of steam cannot be used in an ordinary heater because the temperature in the heater shell is determined by the pressure prevailing in the shell where condensation of steam is taking place. So, in HP heaters it is customary to include a 'desuperheating' section in the main shell where superheated steam is first brought down to near its saturation temperature (1). It complicates the heater design and construction but reduces the capital cost, as compared with providing a separate desuperheater and hence finds favour (1). Fenton (5) has listed another advantage of desuperheaters/ desuperheating sections that these help in reducing TTD (or 'approach') even down to zero or in some cases, even a 'cross-over' i.e., heating a part of the feedwater to or beyond the saturation temperature is possible (1.10).

It is customary (1,10) to allow the steam to leave the desuperheating section with a residual superheat of about 25°C or so. According to Horlock (11), superheated steam under these conditions can be considered as an isentropically perfect gas i.e., C_p & $R'_0 = p.v/T$ are very closely functions of entropy alone. Thus the mechanism of heat transfer in this section is entirely by convection and the greater part of resistance to heat transfer is in the steam film formed on the tubes (1). In some HP heater designs, heater drain-cooling sections are also provided so as to help in recovering a part of the heat lost in HP heater drains (1,5).

Many factors are listed in references (2,12) regarding choice of fluid on the shellside and on the tube-side of a shell and tube type of heat exchanger. For feedwater heaters, however, it is usual to make feedwater pass through tubes and bled steam flows in the shell around the tubes (1).

2.2 HEAT TRANSFER CONSIDERATIONS

2.2.1 LOGARITHMIC MEAN TEMPERATURE DIFFERENCE (LMTD)

Surface type heat exchangers are designed by the usual equation (12)

$$Q = U.A.(\Delta t_m) \quad \dots \quad (1)$$

where Q is the heat transfer rate,

U is the relevant average overall heat transfer coefficient over the entire tube length,

A is commonly specified as the outside surface

area of tubes and

Δt_m is the mean temperature difference, usually known as 'Logarithmic Mean Temperature Difference' (LMTD) defined as (12)

$$LMTD = \frac{1}{L} \int_0^L \Delta t \cdot dx = \frac{GTD - LTD}{\ln(GTD/LTD)} \dots (2)$$

where L is the effective length of heat exchanger path,
 dx is the elemental length,
 Δt is the elemental change in temperature and
 GTD & LTD are respectively the greater and lesser temperature difference between the two fluids, reckoned at either end of the heat exchanger flow path.

The concept of LMTD is based on a number of simplifying assumptions. One of the most important assumptions is that there are no partial phase changes (vapourisation or condensation) taking place in the system. The derivation is applicable for sensible heat changes only or when vapourisation or condensation is isothermal over the whole length of the path (12, 13).

Colburn (14) studied the problem of varying ' U ' over the flow path and concluded that whenever ' U ' varies considerably over the flow path, LMTD is not the true temperature difference. For linear variation of ' U ', he proposed a mathematical model for determining the average value of $U \cdot (\Delta t)$. Kern (13) has introduced the concept of 'Caloric Temperature' based on Colburn's work but recommended the use of average bulk or arithmetic mean

temperature of a fluid for cases when the temperature rise or fall of the fluid is small.

When the design of a heat exchanger results in extreme lengths, it is usual to employ multipass units (12). In a single-phase section, increasing the number of passes on the liquid side makes that section of the exchanger more like a counterflow heat exchanger than a cross-flow heat exchanger (13). Mott (15) has stated that as the number of passes is increased, the liquid velocity increases which increases Reynolds Number resulting in increased heat transfer coefficient.

The problem of evaluating exact LMTD for multipass exchangers has been studied by a number of research workers. Nagle (16) obtained a correction factor 'F' for upto 1 - 6 exchangers by graphical integration and commented that the 'F' factors for 1 - 2 exchangers applied with negligible error to 1 - 4 and 1 - 6 exchangers. Recent work by Kraus and Kern (17) has, however, revealed the variation to be upto 6.8 percent in extreme cases. Underwood (18) and Bowman (19) rederived/extended the results given by Nagle (16). Fischer (20) also worked on multipass exchangers and gave a set of general rules. Bowman, Mueller and Nagle, in their comprehensive paper (21), presented graphs of 'F' factors for shells with one to six passes and number of continuous tube passes respectively double the number of shell passes. Kern (13) has suggested the use of following equation for LMTD correction factor 'F' for 1 - 2 exchanger:

$$F = \frac{\sqrt{R^2 + 1} \ln \left(\frac{1-S}{1-RS} \right)}{(R-1) \ln \left[\frac{2 - S(R+1 - \sqrt{R^2 + 1})}{2 - S(R+1 + \sqrt{R^2 + 1})} \right]} \dots\dots (3)$$

where R = Heat capacity ratio of tube fluid to shell

$$\text{fluid} = \frac{T_1 - T_2}{t_2 - t_1},$$

$$S = \text{Thermal effectiveness} = \frac{t_2 - t_1}{T_1 - t_1},$$

T & t are the shell and tube fluid temperatures respectively,

Suffix 1 and 2 respectively denote inlet and outlet conditions.

The correction factor ' F ' has been graphically presented with R & S as parameters in a number of references (12,13). Kern (13) has, however, recommended that when the value of R & S is close to the vertical portion of a curve, it is difficult to read the figure and ' F ' should be computed from equation (3) directly. This factor ' F ' when multiplied by the counterflow LMTD gives the true mean temperature difference for 1 - 2 exchangers and the same value can approximately be used for 1 - 4 exchangers (13).

When one of the fluids changes phase throughout the length of exchanger path, then $F = 1$ since LMTD will be the same for counterflow and parallel flow conditions (12,13). Many references (2,12,22) contain correc-

tion factors for various crossflow conditions usually encountered.

Ten Broeck (23), Granet (24) and Kays & London (25) have presented graphs of the dimensionless groups of S , R and NTU (Number of Heat Transfer Units).

Kern (13) has discussed the case of partial phase change taking place in a heat exchanger such as a desuperheater or subcooling condenser and has suggested splitting up of the problem into two zones viz. one desuperheating/subcooling zone and the other as condensing zone, determining LMTD & U_{Cl} separately for each zone and then combining the two as per following equation :

$$\text{LMTD (Weighted)} = \frac{Q_{\text{total}}}{\sum \left(\frac{Q}{\text{LMTD}} \right) \text{ each zone}} \dots (4)$$

where Q_{total} is the heat transfer rate for the entire heat exchanger,

Q & LMTD are respectively the heat transfer rate and counterflow LMTD for each zone.

Knowing LMTD(Weighted), design U or U_D can be determined from equation (1).

The equation

$$U_{Cl}(\text{Weighted}) = \frac{\sum (U_{Cl} \cdot A) \text{ each zone}}{A_{\text{total}}} \dots (5)$$

can be used to determine weighted U_{Cl} , knowing U_{Cl} & A for each zone and A_{total} , the total heat transfer area

for the exchanger. Gupta and Prakash (26) have also illustrated use of this concept.

2.2.2 OVERALL HEAT TRANSFER COEFFICIENT

The most common equation for overall heat transfer coefficient, as presented by TEMA (22) and reference (12) is of the form

$$\frac{1}{U_o} = \frac{1}{h_o} + r_{o,d} + \frac{D_o \ln(D_o/D_i)}{2k'_{wl}} + r_{i,d_o} + \frac{1}{h_i} \left(\frac{D_o}{D_i} \right) \dots (6)$$

where the various terms have their usual meaning.

Equation (6) may also be written with usual notations, in the form

$$R_o = r_o + r_{o,d} + r_{wl_o} + r_{i,d_o} + r_{i_o} \dots (6A)$$

where

$$r_{wl_o} = \frac{D_o \ln(D_o/D_i)}{2k'_{wl}}$$

The first and last terms of equation (6) have received maximum attention from various research workers and for second and fourth terms, only empirical relations are available (12,22).

2.2.3 SCALE OR FOULING RESISTANCE

Fouling - the buildup of unwanted deposits such as scales, sludges and slimes on plant surfaces is a widely encountered problem in industry (27). Brune and Moskiment (28) and Benenson, Velikovich and Rezhikova (29) have highlighted that fouling often causes a marked re-

duction in heat transfer capability of plant and forces plant designers to specify additional heat transfer surface area to overcome this problem and this costs. One estimate (27) puts the additional cost equivalent to about 1500 to 3600 crores of Rupees per annum to the Western World alone.

Faras and Ozisk (2) and Kern (13) have suggested a reduction factor of 0.85 to be applied to U_{C1} to obtain U_D in case of steam to water heat exchangers with no contamination on steam or water side. A term 'Cleanliness Factor' to represent dirt or fouling factor has also been introduced by Kern (13).

TEMA (22) have proposed following values for fouling resistance (as converted into MKS units) in their Standards :

Nature of Fluid	Fouling resistance r_{o_d} or r_{i_d} (hr m ² °C/Kcal)
(a) Steam (non oil bearing)	1.024254×10^{-4}
(b) Exhaust Steam (oil bearing)	2.048509×10^{-4}
(c) Treated boiler feed- water upto 52°C with heating medium tempera- ture upto 116°C and water velocity over 0.9144 m/s	1.024254×10^{-4}
(d) Treated boiler feedwater over 52°C with heating	2.048509×10^{-4}

medium temperature between
116°C and water velocity
over 0.9144 m/s.

Although this is the only empirical data readily available, it has received severe criticism (12) such as ignoring the tube material, lack of test data and lack of time interval for buildup of specified coefficient and non-linearity of buildup with time.

Kern (13) has made frequent use of the equation

$$\text{Dirt/fouling resistance/coefficient} = \frac{U_{C10} - U_{D0}}{U_{C10} \cdot U_{D0}} \dots (7)$$

where the symbols have their usual meaning.

2.2.4 SURFACE BOUNDARY CONDITIONS

Two different concepts have been used by various research workers while investigating forced convective heat transfer phenomenon (2,12,30,31). These are:-

(i) Uniform Heat Flux Concept which is useful when the temperature difference responsible for heat transfer process is constant over the length of heat exchanger such as pure counterflow in which temperature rise in one fluid equals the temperature drop in the other fluid or electrically heated surfaces and nuclear reactors (2).

(ii) Uniform Wall Temperature Concept which is useful when the wall temperature of heat transfer matrix, rather than the heatflux, is substantially

uniform throughout the unit such as boilers or condensers having tube wall temperature approximately equal to the saturation temperature of the fluid handled (2).

It has been recorded (12) that uniform heat flux Nusselt Number ' Nu_H ' is always higher than the uniform wall temperature Nusselt Number ' Nu_T ', and that results for turbulent flow conditions are invariably derived for uniform heat flux concept and then converted into ^{uniform} wall temperature concept. London (30) stated that for laminar flow conditions, the convective heat transfer coefficient for uniform heat flux concept is about 9 to 28 percent higher than that for uniform wall temperature concept.

Kays (31) has presented curves, due to the work of Sleicher and Tribus, correlating Nu_H & Nu_T versus Reynold Number for fully developed turbulent flow conditions inside a circular tube. These are available in FIG. 3 in which ' $f_i = Nu_H / Nu_T$ '. It is observed that for $Pr > 1$, this factor is quite small.

2.2.3 FLOW DEVELOPMENT AND ENTRY LENGTH

Most of the standard results for forced convective heat transfer are available for fully developed thermal/hydrodynamic boundary layers (2,12,31). Faras & Ozisk (2) have opined that because of relatively thin boundary layer near the entrance region, heat transfer coefficient is very high in this region and have defined 'entry length' as that region of flow path beyond which the flow pattern becomes

fully developed. Characteristically, thermal entry length in case of turbulent flow is much shorter than that for laminar flow (2, 12). FIGURE 4 (A & B) shows the effect of entry length on Nusselt Number for turbulent flows (12).

Faras and Ozisk (2) presented a correction factor curve (reproduced in FIGURE 5) to account for flow development. The curve is based on the work of Deissler (32) and the fully developed Nusselt Number when multiplied by the factor ' f_2 ' gives an average value of Nusselt Number over the flow path. Harnett's work (33) for $Re = 17,000$ to $48,000$ showed that thermal entry length varies from 10 to 15 tube diameters and that for $Pr > 1$, Pr has no effect on entry length.

2.2.6 TURBULENT FLOW INSIDE SMOOTH CIRCULAR TUBES

Heat Transfer Coefficient

Many algebraic equations are used extensively to represent turbulent convective heat transfer data for flow inside smooth circular tubes. Mc Adams (34) pointed out that for the same power loss or pressure drop, greater heat transfer is obtained from a smooth rather than a rough pipe and gave the following equation, known as 'Dittus - Boelter Equation':-

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad \dots (8)$$

This equation agrees very well in the range $Re = 10,000$ to $1,20,000$, $Pr = 0.7$ to 120 & $L/D > 60$ (12, 34). For

liquids, all fluid properties are evaluated at their 'bulk-mean' or 'cup-mixing' temperature which is the arithmetic mean temperature of the fluid at its entry to and exit from the tube (2,12,34). For gases, however, Faras and Ozisk (2) have cited the work of NASA Lewis Laboratory over large temperature difference range which demonstrated that for both air and helium, the data are correlated nicely by evaluating the physical properties of the fluids at the 'film temperature' i.e., the arithmetic mean temperature between the tube wall and the bulk free stream (35,36).

Another very popular equation, proposed by Colburn (37), is

$$Nu = 0.023 Re^{0.8} Pr^{1/3} \quad \dots (9)$$

This equation also agrees very well in the range $Re = 10,000$ to $1,20,000$, $Pr = 0.7$ to 120 & $L/D > 60$ when fluid properties, excluding C_p , are evaluated at the film temperature (34).

For air, however, Mc Adams (34) proposed the coefficient 0.023 of equation (8) and (9) to be replaced by 0.021.

Kays (34) gave the following equation in the range $Pr = 1$ to 20 for both uniform heat flux and uniform wall temperature concepts

$$Nu = 0.0155 Re^{0.83} Pr^{0.5} \quad \dots (10)$$

Sieder and Tate (38), after much experimentation, proposed that in order to account for viscosity variation

of the fluid from tube wall to the middle of fluid stream, the Nusselt Number given by equation (9) should be multiplied by the term $(\frac{\mu_b}{\mu_{wl}})^{0.14}$ and that all properties, except μ_{wl} , of the fluid be evaluated at its bulk-mean temperature (38). According to Faras and Ozisk (2), equation (8) does not require any viscosity correction for gases since their properties are evaluated at film temperature and in case of liquids, the exponent of Pr should be changed from 0.4 (for heating of liquids) to 0.3 (for cooling of liquids). Kays and London (25), however, proposed the viscosity correction factor as $(\frac{\mu_b}{\mu_{wl}})^{0.09}$ and $(\frac{\mu_b}{\mu_{wl}})^{0.12}$ for heating and cooling of liquids respectively, in the turbulent range with $Pr = 1$. Recent work by Petukhov (39) proposed the exponent for $(\frac{\mu_b}{\mu_{wl}})$ to be 0.11 for heating and 0.25 for cooling. Subsequently, Hanna and Sandall (40) have compared Petukhov's results with those of Sieder and Tate (38) and concluded that Sieder and Tate correction factor gives results higher than those of Petukhov (39).

Longitudinal Conduction

In equations (8) to (10), it has all through been assumed that no heat flows by conduction through the fluid mass because of the temperature gradient available from tube inlet to tube outlet. Kays (31) has upheld this assumption, both for laminar and turbulent flows, provided the product $Re.Pr$ (Peclet Number) is greater than 100. According to him, at values appreciably lower than this, longitudinal conduction can be expected to become important and will have the effect of lowering Nusselt Number.

Effect of Curvature

Mc Adams (34), while reporting the investigations of Jeschke, has suggested that for turbulent flow of air in helically coiled tubes, convective heat transfer coefficient, as obtained for straight tubes, should be multiplied by a factor $(1 + 3.5 \frac{D_1}{D})$ and that in the absence of data on specific liquids, the same factor may be applied to them as well.

Rogers and Mayhew (41) investigated heat transfer for turbulent flow of water in steam heated coiled pipes and gave the following equation for average peripheral value of Nusselt Number

$$Nu = 0.021 Re^{0.85} . Pr^{0.4} \left(\frac{D_1}{D} \right)^{0.1} \quad \dots (12)$$

Kern (13) has reported that Richter obtained overall coefficients for double pipe helical water to water exchanger which are about 20 percent greater than those computed for straight pipes using similar flow velocities. He reasoned that because of increased turbulence caused by tube curvature, tube side coefficients for a coiled tube can be expected to be greater for a given weight flow than for a straight tube.

Kubair and Kuloor (42), in their work with Re upto 5000, have reported that spiral coils, under certain conditions, give performance better than that of helical coils and that in all cases, heat transfer coefficients are certainly higher for helical/spiral coils as compared

with those of straight tubes.

2.2.7 TURBULENT FLOW OUTSIDE CIRCULAR TUBES

Gupta and Prakash (26) recommended that for flow parallel to but outside tubes, the values of convective heat transfer coefficient may be determined by using the equations for flow inside the tubes provided 'hydraulic' or 'equivalent' diameter is used rather than outside and the fluid properties are evaluated at the film temperature.

Faras and Ozisk (2) gave following relations for computing the equivalent diameter for cases where wetted area of the exchanger shell is small as compared with that of tube bundle,

$$D_e = D_o \left[1.271(P_1/D_o)^2 - 1 \right] \text{ for square pitch ... (13)}$$

and

$$D_e = D_o \left[1.102(P_1/D_o)^2 - 1 \right] \text{ for equilateral triangular pitch ... (14)}$$

Kays (31) cited Diessler and Taylor's work (43) for $Re = 1,00,000$ to conclude that the concept of equivalent diameter is not necessarily a good approximation in case of flow parallel to but outside the tubes. Their results for staggered square pitch configuration can be approximated by the curve of FIG. 6. In the figure, Nu_{D_e} is the Nusselt Number for fully developed conditions inside a circular tube having the same equivalent diameter. Kays (31) further commented that the curve is in fair agreement with the available experimental data, although his own experiments suggested a value of $Nu/Nu_{D_e} = 1.20$

for $Pi/D_o = 1.15$ to 1.25 . For $Pi/D_o = 1.2$, the curve, however, yields $Nu/Nu_{De} = 1.05$ so that for Pi/D_o between 1.2 to 1.3 , an average value of 1.10 can be taken for Nu/Nu_{De} .

Shell side heat transfer has been investigated by Short (44) and Gardner & Siller (45) and the effect of crossflow on shell side has been dealt with by Bell, Tinker and Devore (12) in their respective work. Their work, however, appears to be applicable only to those cases in which no phase change takes place and the flow is predominantly a crossflow one.

Kern (13) has quoted Donohue to record that baffle pitch, and not the usual 25% cut of the baffle, determines the effective velocity of the shell fluid. He gave the following equation for shell side or bundle crossflow area from which velocity of the shell side fluid can be determined,

$$\text{Shell Side Area} = \frac{\text{Shell ID} \times \text{Mean Clearance} \times \text{Baffle Pitch}}{\text{Mean Tube Pitch}} \quad \dots (15)$$

where Mean Clearance = Mean Tube Pitch - Tube Outside Dia.

2.2.8 CONDENSING HEAT TRANSFER

Nusselt's Theory of Film Condensation

Mc Adams (34), while reporting the basic theory developed by Nusselt, has given the following equation for average value of condensing heat transfer coefficient outside a VERTICAL TUBE,

$$h_m' \left[\frac{\mu_f^2}{k_f^3 \rho_f (\rho_f - \rho_v) g} \right]^{1/3} = 1.47 \left(\frac{4T_v}{\mu_f} \right)^{-1/3} \dots (16)$$

Equation (16) can also be written in the forms

$$(12, 34) \quad h_m' = 0.943 \left[\frac{g \rho_f (\rho_f - \rho_v) k_f^3 h_{fg}}{L \mu_f (t_{sat} - t_{wl_o})} \right]^{1/4} \dots (17)$$

These equations hold good for fully developed laminar flow conditions with the assumption that no non-condensable is there in the vapour, the liquid-vapour interface is smooth and is at saturation temperature, the wall temperature is constant and the temperature profile in the liquid (condensate) film is linear (34). The test for flow to be laminar has been suggested by Faras and Ozisk (2) as $4T_v / \mu_f \leq 2000$ while Mc Adams (34) placed the limit between 1800 to 2100.

To allow for the fact that condensate is cooled from the saturation temperature to the film temperature, Brombley (34) proposed that ' h_{fg} ' of equation (17) should be multiplied by the term

$$\left[1 + 0.4 C_p \frac{t_{sat} - t_{wl_o}}{h_{fg}} \right]$$

especially at high pressures when C_p of the liquid film increases and h_{fg} decreases. Rohsenow (12) also proposed a similar correction and replaced h_{fg} by

$$h_{fg}' = h_{fg} + \frac{3}{8} C_p (t_{sat} - t_{wl_o})$$

He re-evaluated the problem of film condensation to account for non-linear temperature distribution and proposed a modified equation wherein he replaced h_{fg} of equation (17) by the term

$$h_{fg} + 0.68 C_p (t_{sat} - t_{wl_o})$$

Sparrow et al (46) proposed that the effect of variation of viscosity with temperature may be accounted for by using equation (17), as corrected by Rohsenow (12) with k_f , ρ_f , ρ_v and C_p evaluated at reference temperature given by

$$t_{ref} = t_{wl_o} + 0.31 (t_{sat} - t_{wl_o})$$

Faras and Ozisk (2), Rohsenow (12) and Mc Adams (34) have commented that experimental data generally agree with the equations suggested above or are slightly higher. Rohsenow (12) quoted Kapitza to record that in case of water, waves or ripples begin to form at $4\Gamma / \mu \geq 33$ and that these waves or ripples reduce the effective film thickness so that the net effect is to increase the measured h'_m above that predicted by theory by as much as 20 percent. Faras and Ozisk (2) also recommended that h'_m , as evaluated from the above equation should be increased by 20 percent to arrive at the value of condensing heat transfer coefficient.

Mc Adams (34) has presented the following equation, due to Nusselt, for condensation taking place outside

a single HORIZONTAL TUBE for the laminar region of the condensate flow

$$h_m' \left[\frac{\mu_f^2}{k_f^3 \rho_f (\rho_f - \rho_v) g} \right]^{1/3} = 1.2 \left(\frac{2T_h}{\mu_f} \right)^{-1/3}$$

$$= 1.51 \left(\frac{4T_h}{\mu_f} \right)^{-1/3} \quad \dots (18)$$

Rohsenow (12) has presented the above equation in another form

$$h_m' = 0.728 \left[\frac{g \rho_f (\rho_f - \rho_v) k_f^3 h_{fg}'}{D_o h (t_{sat} - t_{wl_o})} \right]^{1/4} \quad \dots (19)$$

In these equations, the term h_{fg}' is the same as discussed in the case of condensation taking place outside vertical tubes and T_h designates condensate flow rate, per unit tube length, at its lowest point. These equations are valid for

$$\frac{2T_h}{\mu_f} \leq 1800 \text{ to } 2100 \quad (12,34)$$

For condensation taking place outside a vertical bank of n horizontal tubes, Nusselt (12,34) assumed all of the condensate dropping from any one tube to fall on the next lower tube in the bank and modified equation (19) as

$$h_m' = 0.728 \left[\frac{g \rho_f (\rho_f - \rho_v) k_f^3 h_{fg}'}{n D_o h (t_{sat} - t_{wl_o})} \right]^{1/4} \quad \dots (20)$$

i.e., the average h_m' for n tubes equals h_m' for a single tube divided by $n^{1/4}$. Mc Adams (34) has opined

that equation (18) is equally valid for tube banks because the effect of n is automatically allowed for in $\bar{T}_h$ based on condensate on the lowest tube in the tier.

Chen (47) suggested that since the condensate film was subcooled, it is possible that additional condensation occurs on the liquid layer between tubes. Assuming all of this subcooling to be removed, he proposed equation (20) to be multiplied by a factor

$$\left[1 + 0.2 \frac{C_p(t_{\text{sat}} - t_{wlo})}{h_{fg}} \cdot (n - 1) \right]$$

Rohsenow (12) commented that this is a good approximation provided

$$\frac{C_p (t_{\text{sat}} - t_{wlo})}{h_{fg}} < 2$$

Mc Adams (34) commented that for steam, average ratio of measured to predicted coefficients was 1.23. Rohsenow (12) and Faras & Ozisk (2) also commented that most experimental data for pure vapours agree with or are slightly higher than equations (18) to (20) and they attributed it to external vibrations causing ripples in the liquid surface, splashing off tubes in a bank or uneven run off due to bowing or inclination of tubes.

Mc Adams (34) has also highlighted the fact that h_m^* for horizontal position has a value larger than that for vertical position.

Kirkbride (48) and Badger (49) proposed the following dimensionless equation for condensation taking place outside vertical tubes in the TURBULENT REGION i.e., for $4T_v/\mu_f$ exceeding 1800

$$h_m^* \left[\frac{\mu_f^2}{k_f^3 \rho_f (\rho_f - \rho_v) g} \right]^{1/3} = 0.0077 \left(\frac{T_v}{\mu_f} \right)^{0.4} \dots (21)$$

Mc Adams (34) proposed that in the unlikely event of $2T_h/\mu_f$ exceeding 2100, equation (21) may be used for horizontal tubes/tube bank as well, replacing the right hand side term $4T_v/\mu_f$ by $2T_h/\mu_f$ and further commented that this equation is in fair agreement with experimental values.

Factors Affecting Film Condensation Phenomenon

For condensation of pure SUPERHEATED vapour, Rohsenow (12) suggested, on the basis of experimental data, the use of

$$h_{fg}'' = h_{fg}' + C_v (t_{\text{sup}} - t_{\text{sat}})$$

inplace of latent heat in various equation and stated that the heat transfer is still governed by conduction $(t_{\text{sat}} - t_{w|_0})$ through the liquid film. Thus

$$\frac{h_m^* (\text{superheated})}{h_m^* (\text{saturated})} = \left[\frac{h_{fg}' + C_v (t_{\text{sup}} - t_{\text{sat}})}{h_{fg}'} \right]^{1/4} \dots (22)$$

Kern (13) and Mc Adams (34) stated that except in

case of desuperheater, it is customary to disregard the temperature range of superheating/desuperheating and consider all the heat to be delivered at the saturation temperature corresponding to operating pressures. Mc Adams (34) has recorded that even with as much as 100°C of superheat, h'_m is only about 3 percent higher than for the saturated steam at the same pressure.

Assuming linear VARIATION OF WALL TEMPERATURE difference ($t_{\text{sat}} - t_{w,0}$) from top to bottom of a vertical surface, Mc Adams (34) suggested a correction factor F_t to be multiplied with h'_m obtained from equations (16) and (18) so as to get the corrected value of h'_m .

Ratio of top to bottom temperature : 0.5 1.0 2.0 5.0
difference

' F_t ' : 0.96 1.0 1.06 1.15

Rohsenow (12) reported that presence of ELECTRO-STATIC FIELD aids in increasing condensing heat transfer coefficient inside a vertical tube.

McAdams (34) recorded that when the VELOCITY of uncondensed VAPOUR is substantial as compared with the velocity of the condensate at the vapour-condensate interface, the vapour velocity influences the velocity and thickness of condensate film so that with downward flow of condensing vapour at high velocity, the heat transfer coefficient is found to increase. Furman and Hampson (50) also reported similar findings for cross-flow of steam at high velocities.

Reynolds (51) explained the effect of PRESENCE OF AIR in condensing vapour in the following words:

"If the steam is mixed with air, then the steam will be condensed, air will be left between the fresh steam and the cold surface so that after condensation has commenced, that surface will be protected by a stratum of air and fresh steam will have either to displace this or pass through it before it, in turn, can be condensed".

Rohsenow (12) and others (1) have explained that the total pressure of vapour and non-condensables approaching tube wall surface remains constant but near the condensation surface (wall) where concentration of vapour and, therefore, its partial pressure is lower than that in the main body of the mixture so that the concentration of non-condensable gas and its partial pressure is correspondingly higher. Due to difference in partial pressures, the vapour diffuses from the main body of the mixture through the gas or air film to liquefy at the condensate film. The rate of condensation is thus governed by diffusion or mass transfer. The interface saturation temperature is significantly below the temperature of the main vapour gas mixture. The combined mass and heat transfer process requires a lengthy trial and error stepwise calculation to predict the performance after accounting for this effect. The diffusion phenomenon is shown in FIGURE 7(A).

Langen's results, as reported by Rohsenow (12), revealed that a significant decrease in heat transfer coefficients results from the presence of very small amount of non-condensable gas or air in the vapour. This

is shown in FIGURE 7(B). Othmer (51) also investigated the effect of presence of air in condensing vapour and his results are shown in FIGURE 7(C).

Rohsenow (12) stated that it is because of this reason that in most of the condensing equipment such as power plant steam condensers and feedwater heaters, a provision is made to bleed off or vent non-condensable gases that leak into the system. Fenton (5) has highlighted that exact location of vents is most important in feedwater heaters and that in many cases, a marked improvement has been obtained by merely repositioning of the vents.

2.3 GRAPHICAL METHODS

2.3.1 WILSON PLOT

Equation (6-A) may be rewritten in the form

$$\frac{1}{U_{C10}} = r_o + r_{wl_o} + r_{i_o} \quad \text{for a clean tube and}$$

$$\frac{1}{U_{D0}} = r_o + r_{wl_o} + r_{i_o} + r_d \quad \text{for a dirty tube}$$

where r_d is the sum of fouling resistances on inside and outside surfaces of the tube.

Wilson (52) showed that the sum of r_o , r_{wl_o} and r_d for a series of tests is approximately constant say, a_o , and that r_{i_o} is the controlling resistance i.e., the water side resistance is the major resistance from condensing steam to water, except where very high

water velocities are used. He further stated that the film coefficient for water flowing inside tubes is proportional to the 0.8th power of velocity through the tubes so that r_{i0} can be replaced by $1/a_1 v^{0.8}$ and the equation given above may be written as

$$\frac{1}{U} = a_0 + \frac{1}{a_1 v^{0.8}}$$

He pointed out that a plot of $\frac{1}{U}$ versus $\frac{1}{v^{0.8}}$ on rectangular co-ordinates should yield a straight line, permitting the evaluation of both the constants a_0 and a_1 . Mc Adams (34) has commented that ' a_0 ' is actually the intercept for an infinite water velocity and ' a_1 ' is the slope of the line and gives the tube side heat transfer coefficient for a unit velocity based on outside area of the tube. Heat transfer coefficient based on inside area can then be worked out. He pointed out that by comparing the value of ' a_0 ' for old and new tubes, the dirt, fouling or scale resistance can be worked out and the tube wall resistance can be worked out by the usual steady-state conduction equation. When wall and dirt resistances are subtracted from ' a_0 ', the true value of condensing coefficient can be obtained by the equation

$$r_0 = \frac{1}{h_0} = a_0 - r_{wl_0} - r_d \quad \dots (23)$$

Thus, with the aid of graphical method of Wilson, the observed overall thermal resistance has been subdivided into component resistances i.e., outside resis-

tance r_o , inside resistance r_{io} , wall resistance r_{wl_o} and fouling or dirt resistance r_d . Experimental data of Orrick (53), as presented by Mc Adams (34) using the graphical method of Wilson (52) is shown in FIGURE 8. The upper and lower curves are for old/new condenser tubes respectively.

Mc Adams (34) has commented that when water velocity becomes too low, the assumption of 0.8 exponent will be incorrect; hence this relation breaks down at low water velocities.

2.3.2 MISCELLANEOUS PLOTS

Van Hangel (54) in his tests for brass tube surface condensers having heat flux in the range 7,300 to 20,000 Kcal/hr m² and $R_o = 10,000$ to 60,000 determined that at the upper range of heat flux, tube side resistance was nearly 70 percent of the total thermal resistance. He further commented that LMTD concept was true only for a single spot in a condenser and that it did not apply for the overall heat transfer coefficient because of the wide variation of temperature throughout the condensers both for steam and for cooling water.

Gilmour (55) in his series of three papers has presented nomograms for quick calculations of tube side, shell side, tube wall and scale resistances for cases where no phase change, for cases with parallel flow in shells and for cases involving change in phase such as heaters and condensers. He has also divided the tempe-

perature drop from hot fluid to cold fluid into various temperature drops on shell side, tube side, through tube wall and through scale. His nomograms are based on a number of simplifying assumptions and yield a fair idea about the various coefficients.



CHAPTER 3

FORMULATION OF THE PROBLEM

3.1 PROBLEM

Feedwater heating cycle of an existing 110 MW reheat unit, broad outlines of which have been given in Section 1.3, has been analysed. The problem considered here is to determine, analytically, the overall heat transfer coefficient based on outside area of the heating surface for all the seven feedwater heaters at the rated full load of 110 MW with all the seven heaters in operation. Breakup of total thermal resistance into various components has also been carried out. In the analysis, basic performance parameters as per supplier's data have been made use of.

3.2 DETAILS OF FEEDWATER HEATERS (56,57)

A simplified flow diagram of the feedwater heating cycle is shown in FIGURE 9. All the LP & HP Heaters are of vertical design and their details are given in the subsequent paragraphs.

LPH I & II

The first two LP heaters do not have any separate shell of their own, rather the tube nest is built up in the LP casing of the turbine itself. Separate water-box and condensate collection chamber are, however, provided. Each heater is, in turn, built up in two identical halves.

The tubes are bent into 'U' shape and their both ends are expanded into a steel tube plate. Typical arrangement of tube bundle layout for these heaters is shown in FIGURE 10.

LPH III, IV & V

Their design is almost identical but for the shell whose thickness is different for LPH V because of higher operating pressure in the shell. The heating area consists of tubes bent into 'U' shape and their both ends are expanded into a steel tube plate. Typical arrangement of tube-packing for these heaters is shown in FIGURE 11.

Feedwater entry to these heaters is from one side near the top portion of the heater and the exit is also from the same level but at a staggered location. Heating steam enters from one side near top of the shell, at a location which is at right angles to the water entry/exit, and the condensate leaves through a drain at the lowest point on the heater shell. Drip/drain from a higher stage heater cascades into the next lower stage heater on the same side of the shell as water entry/exit side at a location nearly 1250 mm above bottom flange of the shell.

All the LP heaters (I to V) consist entirely of condensing section alone, there being no desuperheating or under-cooling section.

HPH I & II

These two heaters are also of identical design but for the shell thickness which is higher for HPH II.

Arrangement of tubes in these heaters is much more complex than the one for LP heaters. Each heater consists of two sections, viz, the 'desuperheating section' and the 'condensing section'. Whereas the desuperheating section forms the central core of the heater and consists of vertical tubes bent into double 'U' i.e., 'UU' shape, the condensing section is built around the desuperheating section in the shape of an annulus packed with horizontal tubes bent into double spiral coil shape.

Feedwater enters/leaves the water box at top of the heaters and is divided into two paths - one fraction (about 10%) flows through the tubes of the desuperheating section and the remaining fraction flows into two inlet headers to which the horizontal bank of tubes, bent into double spiral coils, are connected and leaves through two outlet headers. The tubes are arranged in six horizontal tube banks placed vertically one above the other and each bank consists of ten layers of horizontal tubes arranged vertically above each other. In each horizontal plane, there are four coils and the coils in each column are independently connected to an inlet header and an outlet header. The arrangement of tubes and tube details are shown in FIGURES 12 & 13 respectively.

It will be seen that the entire mass of steam to be condensed comes in contact with four tubes at a time in a given horizontal plane.

Extraction steam for heating the feedwater first enters the central desuperheating section or core from bottom of the heaters, gets desuperheated, escapes into the main shell near its top and then flows down in the annulus shaped condensing section where it gets condensed on the outside of spiral tubes. Steam thus condensed leaves the heater near its bottom.

Disposal of the Condensate Drain/Drip

The condensate of heating steam from LPH II cascades into LPH I and the entire drip from LPH I is led to the suction of condensate pump. The condensate of heating steam from LPH V cascades into LPH IV, that from LPH IV cascades into LPH III and the combined drip of LPH III is led into the outlet of a flash tank/expander (not included in the analysis here) wherefrom the condensate is repumped, with the help of a booster pump, into the main condensate or feed piping just before LPH III.

The condensate of heating steam of HPH II cascades into HPH I at a point about 1800 mm above bottom flange of the heater and the total drip of HPH I is led to the deaerator or at low loads, it goes directly to LPH V or even to the flash tank/expander when LPH V is not in the circuit.

Arrangement for by-passing one or more of the heat-

ers also exist, although it has not been included in various flow diagrams.

FIGURES 14 & 15 show detailed flow diagrams for LPH I to V and HPH I & II respectively.

Miscellaneous Heat Exchangers

In addition to the feedwater heaters described above, a 'Deaerator' and two more small heat exchangers namely, 'Chimney Steam Condenser' and 'Gland Steam Condenser' are also provided in the main feed-line. Whereas former of the two condenses approximately 900 Kg/hr of chimney steam from the turbine stuffing boxes, the latter condenses steam escaping from labyrinth seals of the turbine. Flash tank/expander and steam-jet air ejector also act as small heat exchangers. However, separate identity of all those heat exchangers has been avoided in this work.

3.3 AVAILABLE DATA

TABLES I & II give the available data for various heaters as follows :-

TABLE I : Salient features of various heaters such as their physical dimensions and material of tubes.

TABLE II : Supplier's performance parameters for the regenerative feedwater heating cycle.



TABLE I

SALIENT FEATURES OF VARIOUS FEEDWATER HEATERS

Sr. No.	Item	LPH I	LPHII	LPHIII	LPHIV	LPH V	HPH I*		HPH II*	
							(a)	(b)	(a)	(b)
1.	Shell ID (mm)	NA	NA	1084	1084	1080	1472	375 x 375	1450	375x 375
2.	Overall Ht. (mm)	5094	5094	5500	5500	5500	7600	-	7600	-
3.	Shell Thickness (mm)	--	--	8	8	10	14	-	25	-
4.	Tube OD/ID (mm)	15/13	15/13	15/13	15/13	15/13	25/15	25/15	25/15	25/15
5.	Tube Length (mm)	7000	7000	7475	7475	7475	13800	18520	13800	18520
6.	No. of Tubes	242	242	710	710	710	240	33	240	33
7.	Tube Count	484	484	1420	1420	1420	4	132	4	132
8.	Steam Side Area (m ²)	80	80	250	250	250	260	48	260	48
9.	Water flow area (cm ²)	160.606	160.606	471.199	471.199	471.199	424.115	58.316	424.115	58.316
10.	Tube Matl.	AM*	AM*	AM*	AM*	AM*	CS*	CS*	CS*	CS*

*(a) ... Condensing Section, (b) Desuperheating Section.

*AM ... Admiralty Metal, *CS ... Carbon Steel, AST 35/DIN 17135

- Notes: 1. No. of tubeside passes = 4 for LPH I to V and HPH I(b), II(b).
 2. Sr. No. 6 & 8 for LPH I & II pertain to only one half heater.
 3. LPH III to V have 4 trans. baffles at an average spacing=600 mm.
 4. Water flow area (S.No.9) = $\pi / 4 D_o^2 \times$ Tube Count per pass except in case of HPH I(a) and HPH II(a).

TABLE II

SUPPLIER'S PERFORMANCE PARAMETERS AT 110 MW LOAD

Sr. Para- No. meter	LPH I	LPH II	LPH III	LPH IV	LPH V	HPH I*		HPH II*	
						(a)	(b)	(a)	(b)
1. P_s (Kgf/cm ²)	0.1964	0.4140	0.9672	2.2376	5.450	17.175	17.175	32.590	32.590
2. P_w^{***} (Kgf/cm ²)	21	21	21	21	21	173	173	173	173
3. m_s' (kg/hr) (for each half)	1981	3912	7774	12756	16092	25000	25000	29100	29100
4. m_w' (Kg/hr) (for each half)	134496	134496	305614	305614	305614	325183	36131	325183	36131
5. h_{s_1} or h_{sup} (Kcal/Kg)	608.70	630.80	660.80	694.70	751.00	711.49	808.30	687.88	748.70
6. h_{sat} (Kcal/Kg) (approximate)	608.70	630.8	638.47	647.40	657.95	667.54	667.54	669.50	669.50
7. t_{s_1} or t_{sup} (°C)	59.27	78.00	145.00	224.78	338.89	276.40	460.10	261.30	364.60
8. t_{sat} (°C)	59.27	76.22	98.15	123.18	154.35	203.90	203.90	237.40	237.40
9. t_{w_1} (°C)	48.85	57.35	80.82	97.67	122.78	160.30	160.30	197.16	197.16
10. $U_{supplier}$ Kcal/hrm ² °C	3085	-	3637	-	-	2780	422	2912	419

* (a) Condensing Section, (b) Desuperheating Section.

** Neglecting pressure drop in the desuperheating section.

*** Neglecting pressure drop in heaters.

**** Mixed with gland steam.

CHAPTER 4

THEORETICAL ANALYSIS AND CALCULATIONS

4.1 DESIGN OVERALL HEAT TRANSFER COEFFICIENT ' U_{Do} '

In order to work out the design figure of U_{Do} , the basic equation (1) can be used. If we know ' Q ' and ' Δt_m ', ' A_o ' being already known, U_{Do} can be worked out.

Calculation of ' Q '

Specific enthalpy drop of steam can be worked out by assuming that there is no undercooling of the condensate so that specific enthalpy of condensate leaving a particular heater corresponds to the saturation temperature of water. Multiplying the mass flow rate of steam by the specific enthalpy drop of steam, total rate of enthalpy drop of steam can be worked out. To this must be added the time rate of enthalpy drop of drain/drip cascading from a higher pressure heater into the next lower pressure heater. The resulting sum gives the total rate of heat transfer ' Q ' for a particular heater.

Dividing ' Q ' by the mass flow rate of water, specific enthalpy rise of feedwater in a particular heater can be worked out so that final temperature of feedwater leaving the heater can be determined.

Calculation of ' Δt_m ' or 'LMTD'

Having known all the four terminal temperatures, it

is now possible to calculate LMTD.

For LPH I & II, the quality of steam is almost that of saturation state and thus steam temperature can be taken to be constant throughout the heaters and LMTD can accordingly be worked out with the help of equation (2). Since no partial phase change takes place ' F ' = 1. Since the two halves of each of these heaters are identical, calculations need only to be made for either half of the heater.

For LPH III, IV & V, the inlet steam is superheated and for the purpose of evaluating LMTD, desuperheating and condensing process LMTDs are separately worked out. The condensing zone may be designated as Zone I and the desuperheating zone as Zone II. Whereas $F = 1$ for Zone I, its value for Zone II may be determined by using equation (3) by assuming the flow to be a counterflow one. Percentage of enthalpy drop taking place in the two zones can also be worked out. Knowing $Q/LMTD$ for each zone, weighted LMTD can be determined for the heater as a whole, by using equation (4).

FIGURE 16. shows various temperature profiles for LMTD determination of LPH I to V.

Having known Q , A_o and LMTD, design value of overall heat transfer coefficient based on outside area U_{Do} can be determined. For sake of comparative study, U_{Do} and U_{Do}^* have been worked out by both including and excluding

the effect of enthalpy of drip cascading from a higher pressure heater into a lower pressure heater.

In HP heaters, there are 33 tubes in the desuperheating section out of a total of $33 + 240 = 273$ tubes i.e., water flow area for desuperheating section is about 12 percent of the total flow area. In order to account for different weight densities of water in the two sections and also the contribution of inlet/outlet headers in the condensing section, it may be assumed that about 10 percent of the total mass flow rate of water goes to the desuperheating section and the remaining to the condensing section.

For the desuperheating sections of HPH I & II, counterflow LMTD can be determined and the correction factor 'F' can be applied as per equation (3).

For the condensing sections of HPH I & II, the flow may be assumed to be a parallel flow one and LMTD can be determined in a similar manner as for LPH III to V. Because of crossflow conditions existing in the condensing section, the LMTD determined as above may be suitably increased.

FIGURE 17 shows various temperature profiles for LMTD determination of HPH I & II.

Weighted LMTD for entire HPH I & II and also U_{Do} and U'_{Do} can now be determined. Detailed calculations for working out U_{Do} & U'_{Do} for various heaters have been presented in TABLES III to V.

TABLE III

LMTD AND U_{Do} CALCULATION SHEET FOR LPHI & LPHII

(Calculations presented are for one of the two identical sections)

Sr. No.	Quantity/Parameter	Units	LPHI	LPHII
1.	P_s	Kgf/cm ²	0.1964	0.4140
2.	t_{s1}	°C	59.27	73.00
3.	t_{sat}	°C	59.27	76.22
4.	h_{s1}	Kcal/Kg	608.70	630.80
5.	m'_s	Kg/hr	1981.00	3912.00
6.	h_{wdr}	Kcal/Kg	76.22	NA
7.	m'_{wdr}	Kg/hr	3912.00	NIL
8.	h_{wc}	Kcal/Kg	59.27	76.22
9.	$(h_{s1} - h_{wc})$	Kcal/Kg	549.43	554.58
10.	$(h_{wdr} - h_{wc})$	Kcal/Kg	16.95	NA
11.	$Q = m'_s(h_{s1} - h_{wc}) + m'_{wdr}(h_{wdr} - h_{wc})$	Kcal/hr	1154729.00	2169517.00
12.	t_{w1}	°C	48.85	57.35
13.	h_{w1}	Kcal/Kg	49.22	57.78
14.	m'_w	Kg/hr	134496.00	134496.00
15.	$h_{w2} - h_{w1} = \frac{Q}{m'_w}$	Kcal/Kg	8.56	16.13
16.	$h_{w2} = h_{w1} + (15)$	Kcal/Kg	57.78	73.91
17.	t_{w2}	°C	57.35	73.51
18.	$GTD = t_{sat} - t_{w1}$	°C	10.42	18.87
19.	$LTD = t_{sat} - t_{w2}$	°C	1.92	2.71
20.	GTD/LTD	-	5.43	6.96

Sr. No.	Quantity/Parameter	Units	LPHI	LPHII
21.	$\ln(\text{GTD}/\text{LTD})$	-	1.69	1.94
22.	$\text{GTD} - \text{LTD}$	$^{\circ}\text{C}$	8.50	16.16
23.	$\text{LMTD}' = \frac{\text{GTD} - \text{LTD}}{\ln(\text{GTD}/\text{LTD})}$	$^{\circ}\text{C}$	5.03	8.33
24.	F	-	1.00	1.00
25.	$\text{LMTD} = \text{LMTD}' \times F$	$^{\circ}\text{C}$	5.03	8.33 Say, 8.55*
26.	A_o	m^2	80.00	80.00
27.	$U_{Do} = \frac{Q'}{A_o \cdot \text{LMTD}}$	$\text{Kcal/hr.m}^2 \text{ } ^{\circ}\text{C}$	2872.00	NA
28.	$U_{Do}' = \frac{ Q \text{m}^2 \text{wd}x = Q}{A_o \cdot \text{LMTD}}$	$\text{Kcal/hr m}^2 \text{ } ^{\circ}\text{C}$	2705.00	3171.00

Ref. Figs 16(a) and 16(b)

*To account for a slight superheat.

TABLE IV

LMTD AND U_{Do} CALCULATION SHEET FOR LPH III, IV & V

Sr. No.	Quantity/Parameter	Units	LPH III	LPH IV	LPH V
1.	$P_s \times 10^2$	Kgf/cm ²	96.72	223.76	545.00
2.	t_{s1}	°C	145.00	224.78	338.89
3.	t_{sat}	°C	98.15	123.18	154.35
4.	h_{s1}	Kcal/Kg	660.80	694.70	751.00
5.	m'_s	Kg/hr	7774.00	12756.00	16092.00
6.	h_{wdr}	Kcal/Kg	123.60	155.50	NA
7.	m'_{wdr}	Kg/hr	28848.00	16092.00	NIL
8.	h_{wc}	Kcal/Kg	98.23	123.60	155.50
9.	$(h_{s1} - h_{wc})$	Kcal/Kg	562.57	571.10	595.50
10.	$(h_{wdr} - h_{wc})$	Kcal/Kg	25.37	31.90	NA
11.	$Q = m'_s(h_{s1} - h_{wc})$ $+ m'_{wdr}(h_{wdr} - h_{wc})$ Kcal/hr		5105293.00	7798287.00	9532786.00
12.	t_{w1}	°C	80.82	97.67	122.78
13.	h_{w1}	Kcal/Kg	81.36	98.06	123.58
14.	m'_w	Kg/hr	305614.00	305614.00	305614.00
15.	$h_{w2} - h_{w1} = \frac{Q}{m'_w}$	Kcal/Kg	16.70	25.52	31.24
16.	$h_{w2} = h_{w1} + (15)$	Kcal/Kg	98.06	123.58	154.82
17.	t_{w2}	°C	97.67	122.78	153.53
18.	h_{sat}	Kcal/Kg	638.47	647.40	657.95

Sr. No.	Quantity/Parameter	Units	LPH III	LPH IV	LPH V
19.	$h_{s1} - h_{sat}$	Kcal/Kg	22.33	47.30	93.05
20.	$\frac{h_{w2} - h_{wd1}}{(h_{s1} - h_{sat})m'_s} \cdot m'_w$	Kcal/Kg	0.57	1.97	4.90
21.	Cp_w (mean) x 10	Kcal/Kg°C	10.06	10.14	10.29
22.	$\frac{t_{w2} - t_{wd1}}{h_{w2} - h_{wd1}} \cdot Cp_w$	°C	0.56	1.95	4.76
23.	t_{wd1}	°C	97.11	120.83	148.77
24.	$\frac{h_{s1} - h_{sat}}{h_{s1} - h_{wc}} \times 100$	-	3.97	8.28	15.63

ZONE I Ref. FIG.16(c), 16(e) & 16(g)

25.	$GTD = t_{sat} - t_{w1}$	°C	17.33	25.51	31.57
26.	$LTD = t_{sat} - t_{wd1}$	°C	1.04	2.35	5.58
27.	$GTD - LTD$	°C	16.29	23.16	25.99
28.	$GTD/LTD, \times 10^2$	-	1666.35	1085.53	565.77
29.	$\ln (GTD/LTD), \times 10^2$	-	281.32	238.47	173.30
30.	$LMTD' = \frac{GTD - LTD}{\ln(GTD/LTD)}$	°C	5.79	9.71	15.00
31.	F	-	1.00	1.00	1.00
32.	$LMTD_I = LMTD' \times F$	°C	5.79	9.71	15.00

ZONE II

33.	$GTD = t_{s1} - t_{w2}$	°C	47.33	102.00	185.36
34.	$LTD = t_{sat} - t_{wd1}$	°C	1.04	2.35	5.58

Sr. No.	Quantity/Parameter	Units	LPH III	LPH IV	LPH V
35.	GTD - LTD	$^{\circ}\text{C}$	46.29	99.65	179.78
36.	GTD/LTD, $\times 10^2$	-	4550.96	4340.43	3321.66
37.	$\ln(\text{GTD/LTD}) \times 10^2$	-	381.79	377.06	350.31
38.	$\text{LMTD}' = \frac{\text{GTD} - \text{LTD}}{\ln(\text{GTD/LTD})}$	$^{\circ}\text{C}$	12.12	26.43	51.32
39.	$R = \frac{t_{s1} - t_{sat}}{t_{w2} - t_{wd1}}$	$\times 10^4$	[116.93	187.59	250.37]
40.	$S = \frac{t_{w2} - t_{wd1}}{t_{s1} - t_{wd1}}$	$\times 10^4$			
41.	$F = \frac{\sqrt{R^2 + 1} \ln\left(\frac{1-S}{1-RS}\right)}{(R-1) \ln\left(\frac{2-S(R+1-\sqrt{R^2+1})}{2-S(R+1+\sqrt{R^2+1})}\right)}$	-	0.94	0.89	0.88
42.	$\text{LMTD}_{II} = \text{LMTD}' \times F$	$^{\circ}\text{C}$	11.35	23.51	45.23
43.	$\text{LMTD} = \frac{Q}{\left(\frac{Q}{\text{LMTD}}\right)_I + \left(\frac{Q}{\text{LMTD}}\right)_{II}}$	$^{\circ}\text{C}$	5.91	10.21	16.75
44.	A_o	m^2	250.00	250.00	250.00
45.	$U_{Do} = \frac{Q}{A_o \cdot \text{LMTD}}$	Kcal/hr m^2 $^{\circ}\text{C}$	3455.00	3056.00	NA
46.	$U_{Do}' = \frac{ Q _{m'_{wd1}=0}}{A_o \cdot \text{LMTD}}$	Kcal/hr m^2 $^{\circ}\text{C}$	2960.00	2855.00	2289.00

*Taking $Q = 1$, $(Q)_I = 1 - \frac{(24)}{100}$ and $(Q)_{II} = \frac{(24)}{100}$.

TABLE VLMTD AND U_{DO} CALCULATION SHEET FOR HPH I & II

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
1.	$P_s \times 10$	Kgf/cm ²	171.75	325.90
2.	t_{s1}	°C	460.10	364.60
3.	t_{s2}	°C	276.40	261.30
4.	t_{sat}	°C	203.90	237.40
5.	$t_{s2} - t_{sat}$	°C	72.50	23.90
6.	h_{s1}	Kcal/Kg	808.30	748.70
7.	h_{s2}	Kcal/Kg	711.49	687.88
8.	h_{sat}	Kcal/Kg	667.54	669.50
9.	h_{wc}	Kcal/Kg	207.70	244.80
10.	h_{wdr}	Kcal/Kg	244.80	NA
11.	m'_s	Kg/hr	25000.00	29100.00
12.	m'_{wdr}	Kg/hr	29100.00	NIL
13.	$h_{s1} - h_{wc}$	Kcal/Kg	600.60	503.90
14.	$h_{wdr} - h_{wc}$	Kcal/Kg	37.10	NA
15.	$Q = m'_s (h_{s1} - h_{wc})$ $+ m'_{wdr} (h_{wdr} - h_{wc}) \times 10^2$ Kcal/hr		160946.17	146634.90
16.	t_{w1}	°C	160.30	197.16
17.	h_{w1}	Kcal/Kg	163.88	202.31
18.	m'_w	Kg/hr	361314.00	361314.00
19.	$h_{w2} - h_{w1} = \frac{Q}{m'_w}$	Kcal/Kg	*	40.58

Sr. No.	Quantity/Parameter	Unit	HPH I	HPH II
20.	$h_{w_2} = h_{w_1} + (19)$	Kcal/Kg	202.31*	242.89
21.	t_{w_2}	$^{\circ}\text{C}$	197.16*	235.02

(* Adopted from Supplier's data)

(A) DESUPERHEATING SECTION

22.	$h_{s_1} = h_{s_2}$	Kcal/Kg	96.81	60.82
23.	$(Q)_d = m'_s (h_{s_1} - h_{s_2})$	Kcal/hr	2420250 .00	1769862.00
24.	m'_{wd}	Kg/hr	36131.00	36131.00
25.	$h_{wd2} - h_{w_1} = \frac{(Q)_d}{m'_{wd}}$	Kcal/Kg	66.99	48.98
26.	$h_{wd2} = h_{w_1} + (25)$	Kcal/Kg	230.87	251.29
27.	t_{wd2}	$^{\circ}\text{C}$	224.04	242.69

Ref. FIG. 17(a) & 17(d)

28.	$\text{GTD} = t_{s_1} - t_{wd2}$	$^{\circ}\text{C}$	236.06	121.91
29.	$\text{LTD} = t_{s_2} - t_{w_1}$	$^{\circ}\text{C}$	116.10	64.14
30.	$\text{GTD} - \text{LTD}$	$^{\circ}\text{C}$	119.96	57.77
31.	GTD/LTD	-	2.03	1.90
32.	$\ln (\text{GTD}/\text{LTD})$	-	0.71	0.64
33.	$\text{LMTD}' = \frac{\text{GTD} - \text{LTD}}{\ln(\text{GTD}/\text{LTD})}$	$^{\circ}\text{C}$	169.04	89.95
34.	$R = \frac{t_{s_1} - t_{s_2}}{t_{wd2} - t_{w_1}}$	-	2.88	2.27
35.	$S = \frac{t_{wd2} - t_{w_1}}{t_{s_1} - t_{w_1}} \times 10^2$	-	21.26	27.19

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
36.	$F = \frac{\sqrt{R^2+1} \ln \left(\frac{1-S}{1-RS} \right)}{(R-1) \ln \frac{2-S(R+1-\sqrt{R^2+1})}{2-S(R+1+\sqrt{R^2+1})}}$	-	0.93	0.89
37.	$(LMTD)_d = LMTD' \times F$	$^{\circ}C$	156.44	80.20
38.	$(A_o)_d$	m^2	48.00	48.00
39.	$(U_{Do})_d = \frac{(Q)_d}{(A_o)_d (LMTD)_d}$	Kcal/hr m^2 $^{\circ}C$	322.00	460.00
(B) <u>CONDENSING SECTION</u>				
40.	m'_{wc}	Kg/hr	325183.00	325183.00
41.	$h_{s2} - h_{wc}$	Kcal/Kg	503.79	443.08
42.	$(Q)_c = Q - (Q)_d$	Kcal/hr	13674367.00	12893628.00
43.	$h_{wc2} - h_{w1} = \frac{(Q)_c}{m'_{wc}}$	Kcal/Kg	42.05	39.65
44.	$h_{wc2} = h_{w1} + \frac{(Q)_c}{m'_{wc}}$	Kcal/Kg	205.93	241.96
45.	t_{wc2}	$^{\circ}C$	200.71	234.23
46.	$h_{s2} - h_{sat}$	Kcal/Kg	43.95	18.38
47.	$h_{wc1} - h_{w1} = \frac{(h_{s2} - h_{sat}) m'_s}{m'_{wc}}$	Kcal/Kg	3.38	1.64
48.	Cp_w (mean)	Kcal/Kg $^{\circ}C$	1.02	1.05
49.	$t_{wc1} - t_{w1} = \frac{(h_{wc1} - h_{w1})}{Cp_w}$	$^{\circ}C$	3.30	1.56
50.	t_{wc1}	$^{\circ}C$	163.60	198.72

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
<u>ZONE I Ref. FIG. 17(c) & 17(f)</u>				
51.	$GTD = t_{sat} - t_{wcI}$	$^{\circ}C$	40.30	38.68
52.	$LTD = t_{sat} - t_{wc2}$	$^{\circ}C$	3.19	3.17
53.	$GTD - LTD$	$^{\circ}C$	37.11	35.51
54.	GTD/LTD	-	12.63	12.20
55.	$\ln (GTD/LTD)$	-	2.54	2.50
56.	$LMTD' = \frac{GTD-LTD}{\ln(GTD/LTD)}$	$^{\circ}C$	14.63	14.20
57.	F	-	1.00	1.00
58.	$(LMTD)_{cI} = LMTD' \times F$	$^{\circ}C$	14.63	14.20
<u>ZONE II</u>				
59.	$GTD = t_{s2} - t_{w1}$	$^{\circ}C$	116.10	64.14
60.	$LTD = t_{sat} - t_{wcI}$	$^{\circ}C$	40.30	38.68
61.	$GTD - LTD$	$^{\circ}C$	75.80	25.46
62.	GTD/LTD	-	2.88	1.66
63.	$\ln (GTD/LTD)$	-	1.06	0.51
64.	$LMTD' = \frac{(GTD-LTD)}{\ln(GTD/LTD)}$	$^{\circ}C$	71.64	50.34
65.	F (assumed value)	-	1.00	1.00
66.	$(LMTD)_{cII} = LMTD' \times F$	$^{\circ}C$	71.64	50.34
67.	$\frac{(h_{s2} - h_{sat})m_s'}{(Q_c)} \times 100$	-	8.04	4.15
68.	$(LMTD)_c' = \frac{(Q)_c}{\left(\frac{Q}{LMTD_{cI}}\right) + \left(\frac{Q}{LMTD_{cII}}\right)} \times ^{\circ}C$	$^{\circ}C$	15.63	14.63

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
69.	$(LMTD)_c$ **	$^{\circ}C$	16.50	15.50
70.	$(A_o)_c$	m^2	260.00	260.00
71.	$(U_{Do})_c = \frac{(Q)_c}{(A_o)_c \cdot (LMTD)_c}$	Kcal/hr m^2 $^{\circ}C$	3188.00	3199.00
72.	$\frac{(Q)_c}{(LMTD)_c}$	Kcal/hr $^{\circ}C$	828749.51	831846.96
73.	$\frac{(Q)_d}{(LMTD)_d}$	Kcal/hr $^{\circ}C$	15470.79	22068.10
74.	$LMTD = \frac{(Q)_c + (Q)_d}{\left(\frac{Q}{LMTD}\right)_c + \left(\frac{Q}{LMTD}\right)_d}$	$^{\circ}C$	19.06	17.17
75.	$A_o = A_c + A_d$	m^2	308.00	308.00
76.	$U_{Do} = \frac{Q}{A_o \cdot LMTD}$	Kcal/hr m^2 $^{\circ}C$	2741.00	NA
77.	$U'_{Do} = \frac{ Q _{m_{wdr}=0}}{A_o \cdot LMTD}$	Kcal/hr m^2 $^{\circ}C$	2557.00	2772.00

* Taking $(Q)_c = 1$, $(Q)_{cI} = 1 - \frac{(67)}{100}$, $(Q)_{cII} = \frac{(67)}{100}$.

** Modified to account for crossflow etc.

4.2 CLEAN OVERALL HEAT TRANSFER COEFFICIENT U_{C10}

It can be determined by using equation (6-A) with the terms $r_{o,d}$ and $r_{i,d}$ missing and then taking reciprocal of R_o . The terms r_o , $r_{i,o}$ and $r_{w1,o}$ can be determined by finding out h_o and h_i from the theoretical formulations and by taking thermal conductivity of the tube material at the mean tube wall temperature.

The main difficulty in applying theoretical formulations arises from the fact that tube wall temperature must be known before any formulae can be made use of. Although this temperature varies in different sections of the tube, an estimate of the mean value of tube surface temperature can be had by trial and error method by keeping in view the fact that heat flux is constant from steam condensing outside the tube surface to heat conduction through tube wall and from the tube inside surface to the water flowing in the tubes, the heat flux being worked out on a particular area (say, outside area) basis. Thus,

$$A_o h_o (\Delta t_o) = \frac{A_o - A_i}{\ln(A_o/A_i)} \cdot \frac{k'_{w1} (\Delta t_{w1})}{(D_o - D_i)/2} = A_i h_i (\Delta t_i)$$

Considering per unit tube length,

$$\pi D_o h_o (\Delta t_o) = \frac{\pi (D_o - D_i)}{\ln(D_o/D_i)} \cdot \frac{k'_{w1} (\Delta t_{w1})}{(D_o - D_i)/2} = \pi D_i h_i (\Delta t_i)$$

whence

$$h_o (\Delta t_o) = \frac{k'_{w1} \Delta t_{w1}}{(D_o/2) \ln(D_o/D_i)} = \left(\frac{D_i}{D_o}\right) \cdot h_i (\Delta t_i) \dots (24)$$

Substituting the values of D_o and D_i in equation (24)

and simplifying we get,

$$\begin{aligned}\text{For LPH I to V, } h_o(\Delta t_o) &= 931.744 k'_{wl}(\Delta t_{wl}) \\ &= 0.8666667 h_i(\Delta t_i) \dots (24-A)\end{aligned}$$

$$\begin{aligned}\text{\& For HPH I \& II, } h_o(\Delta t_o) &= 156.609 k'_{wl}(\Delta t_{wl}) \\ &= 0.6 h_i(\Delta t_i) \dots (24-B)\end{aligned}$$

Individual temperature drops Δt_i , Δt_{wl} & Δt_o are not known. However, ' Δt ', which is the difference between temperature of steam and bulk temperature of water, is known. To start with, certain values of h_i , h_o and k'_{wl} may be assumed and Δt_o can be worked out as under :

$$\Delta t = \Delta t_i + \Delta t_{wl} + \Delta t_o$$

$$\begin{aligned}\text{For LPH I to V, } \Delta t_i &= \Delta t_o \left(\frac{h_o}{h_i} \right) \left(\frac{15}{13} \right) \\ \text{and } \Delta t_{wl} &= \frac{h_o(\Delta t_o)}{931.744 k'_{wl}}\end{aligned} \quad \left| \begin{array}{l} \text{From equation} \\ (24-A) \end{array} \right.$$

$$\text{so that } \Delta t = (\Delta t_o) \left[1 + h_o \left\{ \frac{15}{13h_i} + \frac{1}{931.744 k'_{wl}} \right\} \right]$$

whence

$$\Delta t_o = \frac{\Delta t}{1 + h_o \left[\frac{15}{13h_i} + \frac{1}{931.744 k'_{wl}} \right]} \dots (24-C)$$

Similarly, for HPH I & II,

$$\Delta t_o = \frac{\Delta t}{1 + h_o \left[\frac{25}{15h_i} + \frac{1}{156.609 k'_{wl}} \right]} \dots\dots (24-D)$$

Having known Δt_o , the values of Δt_i and Δt_{wl} can be determined with the help of equations (24-A) and (24-B). Knowing Δt_o , Δt_i and Δt_{wl} , h_o and h_i can be worked out as under:

STEAM SIDE FILM COEFFICIENT ' h_o '

For LPH I to V, which are entirely condensing heaters, the condensate flow is found to be in the laminar region so that equation (16) can be used. The term T_v is worked out from the relation:

$$T_v = \frac{\text{Steam Condensation Rate}}{\pi D_o \times \text{Tube Count}} \dots\dots (25)$$

The value of h'_m determined from equation (16) is increased by 20 percent to account for ripple effect etc. (Section 2.2.8).

In the condensing sections of HPH I & II, the tubes are arranged in horizontal double spiral coils and the condensate is found to be in the turbulent region so that use of equation (21) with the term $4T_v$ replaced by $2T_h$ is appropriate, where

$$T_h = \frac{\text{Steam Condensation Rate}}{\text{Effective Tube length in the Lowest Tier}}$$

$$= \frac{m_s' / 3600}{\frac{13.800}{2} \times 4}$$

$$\text{i.e., } T_h = \frac{m_s'}{99360} \quad \dots (26)$$

$$\text{so that } \frac{2T_h}{\mu_f} = \frac{m_s'}{49680 \mu_f} \quad \dots (26-A)$$

The value of h_m so obtained can be increased by at least 5 percent to account for the effect of ripple formation and tube bowing etc. (Section 2.2.8).

In the desuperheating sections of HPM I & II, superheated steam flows outside the tubes and it is essentially the case of gas flow parallel to outside of vertical tubes. The concept of hydraulic diameter (Section 2.2.7) can be used here.

$$D_e = \frac{4 \text{ times the flow area}}{\text{Wetted perimeter}}$$

$$= \frac{4 \left[(0.375 \times 0.375) - 132 \cdot \frac{\pi}{4} (0.025)^2 \right]}{(4 \times 0.375) + 132 \pi (0.025)}$$

(Ref. FIG. 12 & 13)

$$= 0.02556 \text{ m}$$

$$\text{i.e., } D_e = 25.56 \text{ mm}$$

Mean pitch P_1 of the tube packing is the average of 375/12 and 375/11 mm i.e., about 32.6 mm so that,

$$P_1/D_o = \frac{32.6}{25.00} = 1.304 \quad \text{Say, } 1.30$$

For this Pi/D_o ratio, a correction factor of about 1.1 can be applied to the value of h_o determined on the basis of equivalent diameter (Section 2.2.7).

Reynolds Number for steam flow in the desuperheating section of HPH I & II may be determined from the following relation :

$$\begin{aligned}
 Re_{D_o} &= \frac{V D_o}{\nu_f} \\
 &= \frac{4 \cdot m_s}{h_{f, \text{Wetted Perimeter}}} \\
 &= \frac{4 \cdot m_s}{3600 \cdot h_f \cdot 132 \cdot \pi \cdot D_o} \quad (D_o = 0.02556 \text{ m}) \\
 \text{i.e., } Re_{D_o} &\simeq 1.0481 \times 10^{-4} \frac{m_s}{h_f} \quad \dots\dots (27)
 \end{aligned}$$

The coefficient ' h_o ' for the desuperheating sections of HPH I & II can be evaluated by using equation (8) with various fluid properties evaluated at film temperature.

WATER SIDE COEFFICIENT ' h_1 '

For LPH I to V, water side heat transfer coefficient h_1 may be determined by using equation (9) with the viscosity correction applied as proposed by Sieder and Tate (38) . Since $Re.Pr \gg 100$ for all the heaters, longitudinal conduction effect can be ignored. Whereas correction factor f_1 to convert Nu_H to Nu_T may be had from FIGURE 3, the correction factor f_2 for entry

length can be read off from FIGURE 5.

For desuperheating sections of HPH I & II, the calculation procedure is similar to that for LPH I to V with the difference that no conversion of Nu_H to Nu_T is required as the tube wall temperature cannot be assumed to be uniform so that the factor f_1 is not required to be applied.

For condensing sections of HPH I & II, an additional correction factor ' f_3 ' for tube curvature needs to be applied as detailed below :

Total length of double spiral coil

$$\text{having six turns} = 13800 \text{ mm}$$

$$\text{So, average periphery of each turn} = \frac{13800}{2 \times 6} \text{ mm}$$

$$= 1150 \text{ mm}$$

$$\& \text{ mean diameter of the coil} = \frac{1150}{\pi}$$

$$\approx 366 \text{ mm}$$

$$\text{So that mean inside dia of the coil} = 366 - 2 \times 5$$

$$\text{i.e., } D = 356 \text{ mm}$$

$$\text{Also, } D_1 = 15 \text{ mm}$$

$$\therefore \text{ Correction factor } f_3 = 1 + 3.5 \frac{D_1}{D}$$

$$= 1 + \frac{3.5 \times 15}{356}$$

$$\text{i.e., } f_3 = 1.1475$$

Check calculations for the effect of curvature may also be made by using equation (12).

CHECK

Having determined h_i and h_o as detailed above, Δt_o can be recalculated with the help of equations (24-D) or (24-C), as the case be, and compared with the value determined earlier. This procedure is to be repeated till successive trials yield consistent (almost identical) results.

From the final values of h_i and h_o so determined and taking k'_{wl} of the tube material at average wall temperature, U_{C1o} can be determined. Weighted clean overall heat transfer coefficient for HPH I & II can also be determined by using equation (5).

SOURCE OF VARIOUS PHYSICAL PROPERTIES

Whereas thermal conductivity of tube material k'_{wl} has been taken from TEMA Standards (22), various physical properties of water and steam, including transport properties, have been taken from reference (58). Since all these properties were available only in FPS units and no published work in MKS units was readily available, these properties, in the limited range of interest, have been converted into MKS units and have been presented in the shape of various curves in the APPENDIX (FIGURES A-1 to A-13).

4.3 DIRT/FOULING/SCALING COEFFICIENT/RESISTANCE

Since U_{Do} , U'_{Do} and U_{C1o} are known (Sections

4.1 and 4.2), the dirt coefficient provided for/available in the design of various feedwater heaters can be worked out by using equation (7).

Final trial calculations for U_{Cl_0} and dirt Coefficient have been presented in TABLES VI to VIII.

TABLE VI

U_{C10} CALCULATION SHEET FOR LPH I & II

(Calculations presented are for one of the two identical sections)

Sr. No.	Quantity/Parameter	Units	LPH I	LPH II
1.	$t_{sat} = t_s$	$^{\circ}\text{C}$	59.27	76.22
2.	$t_{bw} = \frac{t_{w1} + t_{w2}}{2}$	$^{\circ}\text{C}$	53.10	65.43
3.	$\Delta t = t_{sat} - t_{bw}$	$^{\circ}\text{C}$	6.17	10.79
4.	Δt_o	$^{\circ}\text{C}$	3.54	6.75
5.	$t_{wl_o} = t_{sat} - \Delta t_o$	$^{\circ}\text{C}$	55.73	69.47
6.	$t_f = t_{wl_o} + 0.5(\Delta t_o)$	$^{\circ}\text{C}$	57.50	72.85
7.	t_{ref} (for k_f^i, ρ & C_p) $= t_{wl_o} + 0.31 (\Delta t_o)$	$^{\circ}\text{C}$	56.83	71.56
(a) <u>STEAM SIDE COEFFICIENT</u>				
8.	$T_v = \frac{m_s^i}{3600 \cdot \pi \cdot D_o \cdot \text{Tube Count}}$			
	, $\times 10^4$	Kg/ms	241.27	476.44
9.	μ_f , $\times 10^4$	Kg/ms	4.79	3.82
10.	μ_f^2 , $\times 10^{10}$	$\text{Kg}^2/\text{m}^2\text{s}^2$	2294.41	1459.24
11.	$Re_f = \frac{4T}{\mu_f}$	-	201.47 < 1800	498.89 < 1800
			$\therefore$ Laminar	$\therefore$ Laminar
12.	k_f^i , $\times 10^2$	Kcal/hr m°C	55.85	56.95
13.	$k_f = k_f^i / 3600$, $\times 10^7$	Kcal/sm $^{\circ}\text{C}$	1551.38	1581.94
14.	k_f^3 , $\times 10^{15}$	$\text{Kcal}^3/\text{sm}^3\text{C}^3$	3733.89	3958.89

Sr. No.	Quantity/Parameter	Units	LPH I	LPH II
15.	ρ_f	Kg/m ³	985.20	976.45
16.	ρ_v	Kg/m ³	0.13	0.26
17.	$g \cdot \rho_f (\rho_f - \rho_v)$	$\times 10^{-2}$ -	95205.16	93508.99
18.	$h'_m = 1.47 \left[\frac{g \cdot \rho_f (\rho_f - \rho_v) k_f^3}{Re_f \cdot \mu_f^2} \right]^{1/3}$	Kcal/sm ² °C $\times 10^2$	134.68	117.34
19.	$h_m = 3600 h'_m$	Kcal/hrm ² °C	4848.33	4224.17
20.	$h_o = 1.2 h_m$	Kcal/hrm ² °C	5818.00	5069.00
(b) <u>CONDUCTION THROUGH TUBE WALL</u>				
21.	k'_{wl}	Kcal/hr m°C	99.50	101.10
22.	$\Delta t_{wl} = \frac{h_o (\Delta t_o)}{931.744 k'_{wl}}$	°C	0.22	0.36
(c) <u>WATER SIDE COEFFICIENT</u>				
23.	$t_{wl_1} = t_{wl_o} - \Delta t_{wl}$	°C	55.51	69.11
24.	t_{bw}	°C	53.10	65.43
25.	$\Delta t_1 = t_{wl_1} - t_{bw}$	°C	2.41	3.68
26.	v_w	$\times 10^4$ m ³ /Kg	10.12	10.20
27.	m'_w	Kg/hr	134496.00	134496.00
28.	$V_w = \frac{v_w \times m'_w}{3600}$	$\times 10^6$ m ³ /s	37808.32	38107.20
29.	A_f	$\times 10^5$ m ²	1606.06	1606.06
30.	$V = V_w / A_f$	$\times 10^2$ m/s	235.41	237.27

Sr. No.	Quantity/Parameter	Units	LPH I	LPH II
31.	ν_b , $\times 10^7 \text{ m}^2/\text{s}$		5.21	4.36
32.	$Re_D = VD_1 / \nu_b$	-	58740.00	70746.00
33.	$Re_D^{.8}$	-	6533.52	7581.61
34.	P_z	-	3.38	2.68
35.	$Re_D \cdot P_z$	-	$\gg 100$	$\gg 100$
So, longitudinal conduction effects are negligible.				
36.	$P_z^{1/3}$, $\times 10^3$	-	1500.74	1389.03
37.	μ_b , $\times 10^4 \text{ Kg/ms}$		5.15	4.27
38.	μ_{wl} , $\times 10^4 \text{ Kg/m s}$		4.94	4.03
39.	$(\mu_b / \mu_{wl})^{.14}$, $\times 10^4$	-	10058.45	10083.07
40.	$Nu_H = .023 Re_D^{.8} P_z^{1/3} (\frac{\mu_b}{\mu_{wl}})^{.14}$, $\times 10^3$	-	226835.88	244226.97
41.	$f_1 = Nu_H / Nu_T$, $\times 10$	-	10.06	10.05
42.	Nu_T , $\times 10^3$	-	225482.98	243011.91
43.	L/D_1	-	538.50	538.50
44.	f_2 , $\times 10^2$	-	101.25	101.25
45.	$Nu_T = Nu_T \times f_2$	-	228.30	246.05
46.	k'_{wb} , $\times 10 \text{ Kcal/hr m}^2\text{C}$		5.57	5.67

Sr. No.	Quantity/Parameter	Units	LPH I	LPH II
47.	$h_1 = \frac{Nu_T \cdot k'_{wb}}{D_1}$	Kcal/hr m ² C	9782.00	10732.00
(d)	<u>OVERALL HEAT TRANSFER COEFFICIENT U_{C10}</u>			
48.	$r_o = 1/h_o$, $\times 10^7$	hr m ² C/Kcal	1718.80	1972.77
49.	$r_{i0} = D_o/D_1 h_1$, $\times 10^7$	hr m ² C/Kcal	1179.56	1075.14
50.	$r_{wl0} = \frac{D_o \ln(D_o/D_1)}{2 k'_{wl}}$, $\times 10^7$	hr m ² C/Kcal	107.86	106.16
51.	$R_o = r_o + r_{i0} + r_{wl0}$, $\times 10^7$	hr m ² C/Kcal	3006.22	3154.07
52.	$U_{C10} = 1/R_o$	Kcal/hr m ² C	3326.00	3171.00
53.	$r_d = \frac{U_{C10} - U_{Do}}{U_{C10} \cdot U_{Do}}$, $\times 10^8$	hr m ² C/Kcal	4752.79	NA
54.	$r'_d = \frac{U_{C10} - U'_{Do}}{U_{C10} \cdot U'_{Do}}$, $\times 10^8$	hr m ² C/Kcal	6902.43	*

* $U'_{Do} = U_{C10} = 3171$ Kcal/hr m²C, so no dirt coefficient appears to be available.

TABLE VII

U_{C10} CALCULATION SHEET FOR LPH III, IV & V

Sr. No.	Quantity/Parameter	Units	LPH III	LPH IV	LPH V
1.	t_s	$^{\circ}\text{C}$	145.00	224.78	338.89
2.	t_{sat}	$^{\circ}\text{C}$	98.15	123.18	154.35
3.	$t_{\text{bw}} = .5(t_{w1} + t_{w2})$	$^{\circ}\text{C}$	89.25	110.23	133.16
4.	$\Delta t = t_{\text{sat}} - t_{\text{bw}}$	$^{\circ}\text{C}$	8.90	12.95	16.19
5.	Δt_o	$^{\circ}\text{C}$	5.00	7.85	10.33
6.	$t_{w1_o} = t_{\text{sat}} - \Delta t_o$	$^{\circ}\text{C}$	93.15	115.33	144.02
7.	$t_f = t_{w1_o} + 0.5(\Delta t_o)$	$^{\circ}\text{C}$	95.65	119.25	149.19
8.	t_{ref} (for k_f^1 , ρ & C_p) $= t_{w1_o} + 0.31(\Delta t_o)$	$^{\circ}\text{C}$	94.70	117.76	147.22
(a) <u>STEAM SIDE COEFFICIENT</u>					
9.	$\dot{m}_s = \frac{\text{m}_s^1}{3600 \pi D_o \cdot \text{Tube Count}}$ $\times 10^4$	Kg/m s	322.71	529.52	668.00
10.	μ_f , $\times 10^5$	Kg/m s	29.30	23.15	18.05
11.	μ_f^2 , $\times 10^{10}$	$(\text{Kg/m s})^2$	858.49	535.92	325.80
12.	$Re_f = 4\dot{m}_f / \mu_f$	-	440.56 < 1800 $\therefore$ Laminar	914.94 < 1800 $\therefore$ Laminar	1480.34 < 1800 $\therefore$ Laminar
13.	k_f^1 , $\times 10^2$	$\text{Kcal/hr m}^{\circ}\text{C}$	58.30	59.10	59.09
14.	$k_f = k_f^1 / 3600$, $\times 10^7$	$\text{Kcal/ s m}^{\circ}\text{C}$	1619.40	1641.66	1641.38

Sr. No.	Quantity/Parameter	Units	LPH III	LPH IV	LPH V
15.	k_f^3 , $\times 10^{15}$	(Kcal/sm ² C) ³	4247.15	4424.40	4422.16
16.	ρ_f	Kg/m ³	961.65	944.38	918.96
17.	ρ_v	Kg/m ³	0.57	1.25	2.83
18.	$g \rho_f (\rho_f - \rho_v)$, $\times 10^3$	-	9066.62	8737.50	8258.91
19.	$h_m' = 1.47 \left[\frac{g \rho_f (\rho_f - \rho_v) k_f^3}{Re_f \mu_f^2} \right]^{1/3}$ $\times 10^4$	Kcal/s m ² C	14789.36	13581.03	13398.15
20.	$h_m = 3600 h_m'$	Kcal/hr m ² C	5324.17	4889.17	4823.33
21.	$h_o = 1.2 h_m$	Kcal/hr m ² C	6389.00	5867.00	5788.00
(b) <u>CONDUCTION THROUGH TUBE WALL</u>					
22.	k_{wl}'	Kcal/hr m °C	104.20	107.00	110.50
23.	$\Delta t_{wl} = \frac{h_o (\Delta t_o)}{931.744 k_{wl}'}$	°C	0.33	0.47	0.58
(c) <u>WATER SIDE COEFFICIENT</u>					
24.	$t_{wl_i} = t_{wl_o} - \Delta t_{wl}$	°C	92.82	114.86	143.44
25.	t_{bw}	°C	89.25	110.23	138.16
26.	$\Delta t_i = t_{wl_i} - t_{bw}$	°C	3.57	4.63	5.28
27.	v_w , $\times 10^4$	m ³ /Kg	10.34	10.52	10.77
28.	m_w' , $\times 10^{-2}$	Kg/hr	3056.14	3056.14	3056.14
29.	$v_w = \frac{v_w \times m_w'}{3600}$, $\times 10^5$	m ³ /s	8777.93	8926.48	9142.95

Sr. No.	Quantity/Parameter	Units	LPH III	LPH IV	LPH V
30.	A_f , $\times 10^5$	m^2	4711.99	4711.99	4711.99
31.	$V=V_w/A_f$, $\times 10^2$	m/s	186.29	189.44	194.04
32.	ν_b , $\times 10^7$	m^2/s	3.24	2.63	2.12
33.	$Re_D = \frac{V.D_1}{\nu_b}$, $\times 10^{-2}$	-	747.46	936.40	1189.84
34.	$Re_D^{.8}$	-	7922.65	9487.88	11491.87
35.	Pr , $\times 10$	-	19.25	15.50	12.25
36.	$Re_D.Pr$	-	$\gg 100$	$\gg 100$	$\gg 100$

So, longitudinal conduction effects are negligible.

37.	$Pr^{1/3}$, $\times 10^3$	-	1243.97	1157.30	1069.99
38.	μ_b , $\times 10^5$	Kg/m s	31.40	25.20	19.68
39.	μ_{wl} , $\times 10^5$	Kg/m s	30.20	24.13	18.90
40.	$(\frac{\mu_b}{\mu_{wl}})^{.14}$, $\times 10^3$	-	1005.47	1006.09	1005.68
41.	$Nu_H = .023 Re_D^{.8}$				
	$Pr^{1/3} (\frac{\mu_b}{\mu_{wl}})^{.14}$, $\times 10^3$	-	227917.47	254884.91	284417.46
42.	$f_1 = Nu_H / Nu_T$				
	$\times 10$	-	10.05	10.06	10.04

Sr. No.	Quantity/Parameter	Units	LPH III	LPH IV	LPH V
43.	Nu_T	-	226.78	252.57	283.28
44.	L/D_1	-	575.00	575.00	575.00
45.	f_2 , $\times 10^2$	-	101.19	101.19	101.19
46.	$Nu_T = Nu_T^* \cdot f_2$	-	229.48	255.58	286.66
47.	k_{wb}^* , $\times 10^2$	Kcal/hr $m^{\circ}C$	58.05	59.10	59.23
48.	$h_1 = \frac{Nu_T \cdot k_{wb}^*}{D_1}$	Kcal/hr $m^{2\circ}C$	10247.00	11619.00	13060.00
(d) <u>OVERALL HEAT TRANSFER COEFFICIENT U_{Clo}</u>					
49.	$r_o = 1/h_o$, $\times 10^7$ hr $m^{2\circ}C/Kcal$		1565.19	1704.44	1727.71
50.	$r_{io} = \frac{D_o^2}{D_i \cdot h_i}$, $\times 10^7$ hr $m^{2\circ}C/Kcal$		1126.03	993.07	883.47
51.	$r_{wl_o} = \frac{D_o \ln(D_o/D_i)}{2k_{wl}^*}$, $\times 10^7$ hr $m^{2\circ}C/Kcal$		102.30	100.30	97.13
52.	$R_o = r_o + r_{io} + r_{wl_o}$, $\times 10^7$ hr $m^{2\circ}C/Kcal$		2793.52	2797.81	2708.31
53.	$U_{Clo} = 1/R_o$	Kcal/hr $m^{2\circ}C$	3580.00	3574.00	3692.00
54.	$r_d = \frac{U_{Clo} - U_{Do}}{U_{Clo} \cdot U_{Do}}$, $\times 10^8$ hr $m^{2\circ}C/Kcal$		1010.59	4742.65	NA
55.	$r_d = \frac{U_{Clo} - U_{Do}}{U_{Clo} \cdot U_{Do}}$, $\times 10^8$	hr $m^{2\circ}C/Kcal$	5850.80	7046.41	16601.60

TABLE VIII

U_{Cle} CALCULATION SHEET FOR HPH I & II

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
(A) <u>DESUPERHEATING SECTION</u>				
1.	$t_{bs} = .5(t_{s_1} + t_{s_2})$	$^{\circ}\text{C}$	368.25	312.95
2.	$t_{bw} = .5(t_{w_1} + t_{w_2})$	$^{\circ}\text{C}$	192.17	227.00 (modified value)
3.	$\Delta t = t_{bs} - t_{bw}$	$^{\circ}\text{C}$	176.08	85.95
4.	Δt_o	$^{\circ}\text{C}$	155.93	74.00
5.	$t_{wl_o} = t_{bs} - \Delta t_o$	$^{\circ}\text{C}$	212.32	238.95
6.	$t_f = .5(t_{bs} + t_{wl_o})$	$^{\circ}\text{C}$	290.29	275.95
(a) <u>STEAM SIDE COEFFICIENT</u>				
7.	m'_s	Kg/hr	25000.00	29100.00
8.	μ_f	$\times 10^6 \text{ Kg/m s}$	19.93	19.05
9.	$Re_{De} = 1.0481 \times 10^4 \frac{m^3}{\mu_f}$	-	131473.00	160104.00
10.	$Re_{De}^{.8}$	-	12447.13	14572.11
11.	Pr_f	$\times 10$ -	10.49	11.78
12.	$Pr_f^{.4}$	$\times 10^4$ -	10193.19	10677.22

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
13.	$Nu_{DeH} = .023 Re_{De}^{.8} Pr_f^{.4} \times 10^2$	-	29181.48	35785.60
14.	$k_f \times 10^2$	Kcal/hr m°C	3.81	3.95
15.	$h_{oDe} = \frac{Nu_{DeH} \cdot k_f}{De}$	Kcal/hr m ² °C	434.91	552.94
16.	$h_o = 1.1 h_{oDe}$	Kcal/hr m ² °C	478.41 SAY, 478.00	603.23 SAY, 608.00

(b) CONDUCTION THROUGH TUBE WALL

17.	k'_{wl}	Kcal/hr m°C	41.60	40.95
18.	$\Delta t_{wl} = \frac{h_o (\Delta t_o)}{156.609 k'_{wl}}$	°C	11.44	7.02

(c) WATER SIDE COEFFICIENT

19.	$t_{wl1} = t_{wl0} - \Delta t_{wl}$	°C	200.88	231.93
20.	$t_{bw} = .5 (t_{w1} + t_{wd2})$	°C	192.17	227.00
21.	$\Delta t_1 = t_{wl1} - t_{bw}$	°C	8.71	4.93
22.	$v_w \times 10^4$	m ³ /Kg	11.34	11.90
23.	m'_w	Kg/hr	36131.00	36131.00
24.	$V_w = \frac{v_w \times m'_w}{3600} \times 10^6$	m ³ /s	11138.12	11943.30
25.	$A_f \times 10^6$	m ²	5831.60	5831.60

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
26.	$V = V_w / A_f$, $\times 10^2$	m/s	195.17	204.80
27.	ν_b , $\times 10^7$	m ² /s	1.61	1.43
28.	$Re_D = V.D_1 / \nu_b$	-	181831.00	215130.00
29.	$Re_D^{.8}$	-	16133.70	18456.99
30.	Pr, $\times 10$	-	9.15	8.35
31.	$Re_D \cdot Pr$	-	$\gg 100$	$\gg 100$
So, longitudinal conduction effects are negligible.				
32.	$Pr^{1/3}$, $\times 10^3$	-	970.82	941.66
33.	μ_b , $\times 10^6$	Kg/m s	142.00	120.00
34.	μ_{wl} , $\times 10^6$	Kg/m s	135.12	118.00
35.	$(\mu_b / \mu_{wl})^{.14}$, $\times 10^4$	-	10069.77	10023.56
36.	$Nu_H = .023 Re_D^{.8} Pr^{1/3} (\frac{\mu_b}{\mu_{wl}})^{.14}$	-	362.76	400.69
37.	L/D_1	-	1234.70	1234.70
38.	f_2 , $\times 10$	-	10.06	10.06
39.	$Nu = Nu_H \cdot f_2$	-	364.94	403.09
40.	k'_{wb} , $\times 10^2$	Kcal/hr m°C	58.63	56.56

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
41.	$h_i = \frac{Nu \cdot k_{wb}}{D_i}$	Kcal/hr m ² °C	14264.00	15200.00

(d) OVERALL HEAT TRANSFER COEFFICIENT

42.	$r_o = 1/h_o$, x 10 ⁷	hr m ² °C/Kcal	20920.50	16447.37
43.	$r_{io} = D_o/D_i \cdot h_i$, x 10 ⁷	hr m ² °C/Kcal	1168.44	1096.49
44.	$r_{wl_o} = \frac{D_o \ln(D_o/D_i)}{2k_{wl}}$, x 10 ⁷	hr m ² °C/Kcal	1534.93	1559.29
45.	$R_o = r_o + r_{io} + r_{wl_o}$, x 10 ⁷	hr m ² °C/Kcal	23623.87	19103.15
46.	$(U_{Clo})_d = 1/R_o$	Kcal/hr m ² °C	423.00	523.00
47.	$(r_d)_d = \frac{(U_{Clo})_d - (U_{Do})_d}{(U_{Clo})_d + (U_{Do})_d}$, x 10 ⁷	hr m ² °C/Kcal	7462.41	2618.67

(B) CONDENSING SECTION

48.	t_{s_2}	°C	276.40	261.30
49.	t_{sat}	°C	203.90	237.40
50.	$t_{bw} = .5(t_{w_1} + t_{w_{c2}})$	°C	180.51	215.70
51.	$\Delta t = t_{sat} - t_{bw}$	°C	23.39	21.70
52.	Δt_o	°C	7.22	6.02
53.	$t_{wl_o} = t_{sat} - \Delta t_o$	°C	196.68	231.38

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
54.	$t_f = .5(t_{sat} + t_{wl_o})$	$^{\circ}\text{C}$	200.29	234.39
55.	$t_{ref}(\text{for } k_f^i, C_p \text{ \& } \rho)$ $= t_{wl_o} + .31 (\Delta t_o)$	$^{\circ}\text{C}$	198.92	233.25

(a) STEAM SIDE COEFFICIENT

56.	m'_s	Kg/hr	25000.00	29100.00
57.	$\Gamma_h = m'_s/99360$	$, \times 10^4 \text{ Kg/m s}$	2516.10	2928.74
58.	μ_f	$, \times 10^5 \text{ Kg/m s}$	13.23	11.32
59.	μ_f^2	$, \times 10^{10} (\text{Kg/m s})^2$	175.03	128.14
60.	$Re_h = 2\Gamma_h / \mu_f$	-	3803.63 > 2100 So, turbulent	5174.46 > 2100 So, turbulent
61.	$.0077 \left(\frac{2\Gamma_h}{\mu_f} \right)^{.4}$	$, \times 10^5 -$	20824.28	23552.47
62.	k_f^i	$, \times 10^2 \text{ Kcal/hr m } ^{\circ}\text{C}$	57.25	54.80
63.	$k_f = k_f^i/3600$	$, \times 10^6 \text{ Kcal/s m } ^{\circ}\text{C}$	159.03	152.22
64.	k_f^3	$, \times 10^{15} (\text{Kcal/s m } ^{\circ}\text{C})^3$	4021.78	3527.23
65.	ρ_f	Kg/m^3	859.09	822.29
66.	ρ_v	Kg/m^3	8.61	16.20

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
67.	$g \cdot \rho_f (\rho_f - \rho_v)$, $\times 10^{-2}$ -		71675.67	63024.58
68.	$\left[\frac{\mu_f^2}{k_f^3 \cdot g \cdot \rho_f (\rho_f - \rho_v)} \right]^{1/3}$, $\times 10^5$	s m ²⁰ C/Kcal	8467.91	8236.21
69.	$h_m^n = (61/68)$, $\times 10^2$	Kcal/s m ²⁰ C	245.92	285.96
70.	$h_m^i = 1.05 h_m^n$, $\times 10^2$	Kcal/s m ²⁰ C	258.22	300.25
71.	$h_o = 3600 h_m^i$	Kcal/hr m ²⁰ C	9296.00	10809.00

(b) CONDUCTION THROUGH TUBE WALL

72.	k_{wl}^i	Kcal/hr m ⁰ C	42.00	41.08
73.	$\Delta t_{wl} = \frac{h_o (\Delta t_o)}{156.609 k_{wl}^i}$	⁰ C	10.21	10.11

(c) WATER SIDE COEFFICIENT

74.	$t_{wl_1} = t_{wl_o} - \Delta t_{wl}$	⁰ C	186.47	221.27
75.	t_{bw} (From Sr. No.50)	⁰ C	180.51	215.70
76.	$\Delta t_1 = t_{wl_1} - t_{bw}$	⁰ C	5.96	5.57
77.	v_w , $\times 10^4$	m ³ /Kg	11.18	11.68
78.	m_w^i , $\times 10^{-2}$	Kg/hr	3251.83	3251.83
79.	$V_w = v_w \cdot m_w^i / 3600$, $\times 10^4$	m ³ /s	1009.87	1055.04

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
80.	A_f	$\times 10^5 \text{ m}^2$	4241.15	4241.15
81.	$V = V_w/A_f$	$\times 10^2 \text{ m/s}$	238.09	248.74
82.	ν_b	$\times 10^8 \text{ m}^2/\text{s}$	16.94	14.77
83.	$Re_D = VD_1/\nu_b$	$\times 10^{-2} -$	2108.24	2526.12
84.	Re_D	-	18160.84	20987.62
85.	Pr	$\times 10 -$	9.53	8.54
86.	$Re_D \cdot Pr$	-	$\gg 100$	$\gg 100$

So, longitudinal conduction effects are negligible.

87.	$\frac{\gamma^3}{Pr}$	$\times 10^4 -$	9840.81	9487.52
88.	μ_b	$\times 10^5 \text{ Kg/m s}$	15.15	12.65
89.	μ_{wl}	$\times 10^5 \text{ Kg/m s}$	14.62	12.35
90.	$(\mu_b/\mu_{wl})^{.14}$	$\times 10^2 -$	100.50	100.34
91.	$Nu_H = .023 Re_D^{.8} Pr^{.43} \left(\frac{\mu_b}{\mu_{wl}} \right)^{.14}$	$\times 10^2 -$	41310.45	43951.84
92.	f_1	$\times 10 -$	10.02	10.02
93.	$Nu_T = Nu_H/f_1$	$\times 10^2 -$	41228.00	45860.12

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
94.	L/D_1	-	920.00	920.00
95.	f_2 , x 10	-	10.01	10.01
96.	$Nu_T = Nu_T' \times f_2$, x 10 ²	-	41269.23	45905.98
97.	f_3 , x 10 ²	-	114.75	114.75
98.	$Nu_T = Nu_T'' \times f_3$, x 10 ²	-	47356.44	52677.11
99.	k_{wb}' , x 10 ²	Kcal/hr m ^o C	59.21	57.39
100.	$h_i = \frac{Nu_T \times k_{wb}'}{D_1}$	Kcal/hr m ^{2o} C	18693.00	20155.00

CHECK BY EQN. (12)

101.	$h_{iI} = .021 . Re_D^{.85} . Pr^{.4} . \left(\frac{D_i}{D}\right)^{.1} \frac{f_2}{f_1} \frac{k_{wb}'}{D_1}$	Kcal/hr m ^{2o} C	19839.00	21462.00
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(d) OVERALL HEAT TRANSFER COEFFICIENT

102.	$r_o = \frac{1}{h_o}$, x 10 ⁸	hr m ^{2o} C/Kcal	10757.31	9251.54
103.	$r_{io} = D_o/D_1 \cdot h_i$, x 10 ⁸	hr m ^{2o} C/Kcal	8915.99	8269.24
104.	$r_{wl_o} = \frac{D_o \ln(D_o/D_1)}{2 k_{wl}'} , x 10^8$	hr m ^{2o} C/Kcal	15210.30	15543.62

Sr. No.	Quantity/Parameter	Units	HPH I	HPH II
105.	$R_o = r_o + r_{io} + r_{wi_o} \times 10^8$	hr m ² C/Kcal	34883.60	33064.40
106.	$(U_{Clo})_c = 1/R_o$	Kcal/hr m ² C	2867.00	3024.00
107.	$(r_d)_c = \frac{(U_{Clo})_c - (U_{Do})_c}{(U_{Clo})_c \cdot (U_{Do})_c}$	hr m ² C/Kcal	*	*

(C) ENTIRE HEATER

108.	$U_{Clo} = \frac{(A)_c (U_{Clo})_c + (A)_d (U_{Clo})_d}{(A)_c + (A)_d}$	Kcal/hr m ² C	2486.00	2634.00
109.	$r_d = \frac{U_{Clo} - U_{Do}}{U_{Clo} \cdot U_{Do}}$	hr m ² C/Kcal	*	NA
110.	$r_d^* = \frac{U_{Clo} - U_{Do}^*}{U_{Clo} \cdot U_{Do}^*}$	hr m ² C/Kcal	*	*

* These figures work out to be negative.

CHAPTER 5

DISCUSSION OF RESULTS

5.1 OVERALL HEAT TRANSFER COEFFICIENT

TABLE IX contains a summary of overall heat coefficient values for various feedwater heaters at rated full load of 110 MW.

TABLE IX

OVERALL HEAT TRANSFER COEFFICIENT FOR VARIOUS HEATERS.

		<u>Kcal/hr m²⁰C</u>										
		<u>(Based on Steam Side Area)</u>										
Sr. No.	Heat Transfer Coeff.	LPH I	LPH II	LPH III	LPH IV	LPH V	HPH I *			HPH II*		
							(a)	(b)	(c)	(a)	(b)	(c)
1.	U _{C10}	3326	3171	3580	3574	3692	2867	423	2486	3024	523	2634
2.	U _{D0}	2872	NA	3455	3056	NA	3188	322	2741	NA	NA	NA
3.	U _{D0}	2705	3171	2960	2855	2289	--	--	2557	3199	460	2772
4.	U _{Supp}	3085	--	3637	--	--	2780	422	--	2912	419	--

*(a) Condensing Section, (b) Desuperheating Section,
(c) Entire Heater.

A perusal of the above TABLE reveals that 'clean' overall heat transfer coefficient 'U_{C10}' varies from 3171 Kcal/hr m²⁰C to 3692 Kcal/hr m²⁰C for various LP heaters and that the figures are generally higher for the heater higher in the chain. These figures are in fair agreement with the normal range of values for condensing heaters as per various references quoted below:

<u>Sr. No.</u>	<u>Source</u>	<u>Ref.</u>	<u>Range of U', Kcal/hr m² C.</u>	<u>Remarks</u>
1.	C.E.G.B.	(1)	2440 - 4880	-
2.	Faras & Ozisk	(2)	2196 - 3710	For Surface Condensers.
3.	Salisbury	(3)	3175	For Nuclear Plant Feed- water Heaters.
4.	Kern	(13)	996 - 3400	Including Dirt Factor.
5.	Mc Adams	(34)	1900 - 2500	For Large
6.	Furman & Hampson	(50)	1950 - 5860	Surface
7.	Van Hengel	(54)	4880 - 5425	Condensers.

The overall heat transfer coefficient is somewhat higher for the heater placed higher in the chain because of relatively increased turbulence on water side and higher condensate velocity on the steam side as we go up in the chain.

For HP heaters, however, U_{C10} values are relatively lower than those for LP heaters. This may be attributed to the following two factors:

- (i) Presence of desuperheating sections in these heaters.
- (ii) Poor thermal conductivity of carbon steel as compared with that of admiralty metal and thicker tubes.

It is observed that U_{C10} for the desuperheating

section is only about 15 to 17 percent of its corresponding value for the condensing section. The value of U_{Do} for the desuperheating section is, however, only about 10 to 14 percent of the figure for condensing section. These figures are relatively lower probably because the dirt coefficient may not be significantly different for the two sections. These figures are in fair agreement with Reference (1) where a figure of about 10 percent onwards has been indicated. The desuperheating sections are, however, necessary/customary in HP heaters (discussed in Section 2.1.3) although these inevitably lead to reduced overall heat transfer coefficient because of the poor thermal conductivity of steam as compared with that of condensate film (water) formed in the condensing process.

As expected, dirty or design values of overall heat transfer coefficient U_{Do} & U'_{Do} are lower than the clean coefficient U_{C10} with the exception of LPH V and condensing sections of HPH I & II. Whereas the deviation for LPH V may be due to the uncertainty involved in ascertaining the correct saturation temperature of bled steam mixed with gland steam; that for condensing section of HPH I & II may be attributed to the uncertainty involved in finding out LMTD because of the fact that flow pattern in this section is a complex combination of parallel, counter and crossflow.

It is observed that the various values of overall

heat transfer coefficient are in fair agreement with U_{Supp} i.e., the figures assumed by the manufacturer/supplier of the feedwater heaters.

In a separate exercise (not reported here), it was revealed that the value of overall heat transfer coefficient at part loads is lower than that at full load. This is but expected because of reduced fluid velocities and hence reduced turbulence.

5.2 STEAM/WATER SIDE HEAT TRANSFER COEFFICIENT

A perusal of TABLE VI & VII reveals that steam side condensing heat transfer coefficient h_o is almost equal for LPH I to V (except LPH II) and is of the order of 6,000 Kcal/hr m^2 °C. For HPH I & II (TABLE VIII), it varies in the range of 9,000 to 11,000 Kcal/hr m^2 °C. These figures are in fair agreement with References (13,34).

Water side convective heat transfer coefficient h_i ranges from about 10,000 to 13,000 Kcal/hr m^2 °C for various LP heaters; 14,000 to 15,000 Kcal/hr m^2 °C for desuperheating sections of HPH I & II and 18,000 to 20,000 Kcal/hr m^2 °C for condensing sections of HPH I & II. Relatively higher values for HP heaters are attributable to higher Reynolds Number. The figures for condensing section of HPH I & II are higher than those for desuperheating section because the former consists of tubes bent in the shape of double spiral coils which helps in increasing the heat transfer effects (discussed in Section 2.2.6). It is also noted

that check calculations by using equation (12) yield values of water side coefficient which are about 6 to 6.5 percent higher than those obtained by Jeschke's formulations (34) for helical coiled tubes. The values of h_1 obtained in various calculations are well within the range presented by Faras and Ozisk (2).

3.3 DISTRIBUTION OF THERMAL RESISTANCE

TABLE X shows percentage contribution of water side, steam side and tube wall resistance to the total thermal resistance to heat transfer, for clean tubes, for various heaters.

TABLE X
DISTRIBUTION OF THERMAL RESISTANCE

Sr. No.	Percent Thermal Resistance.	LPH I	LPH II	LPH III	LPH IV	LPH V	HPH I*		HPH II*	
							(a)	(b)	(a)	(b)
1.	r_o	57.17	62.55	56.03	60.92	63.79	30.84	88.56	27.98	86.10
2.	r_{i0}	39.24	34.09	40.31	35.49	32.62	25.56	4.95	25.01	5.74
3.	r_{wl0}	3.59	3.36	3.66	3.59	3.59	43.60	6.49	47.01	8.16

*(a) Condensing Section, (b) Desuperheating Section.

A perusal of TABLE X reveals that steam side thermal resistance r_o is about 1.4 to 2 times the water side thermal resistance r_{i0} for LPH I to V and is about 1.1 to 1.2 times r_{i0} for HPH I(a) and HPH II(a). This goes against the basic ideology of Wilson Plot (discussed in Section 2.3.1) wherein

Wilson (52) showed that when steam condenses outside brass tubes carrying water at moderate velocities, water side thermal resistance r_{10} is the controlling (major) resistance to heat transfer. Van Hengel (54) also showed that water side resistance was about 70 percent of the total thermal resistance in the range of $Re = 10,000$ to $60,000$.

It may, however, be noted that these investigations appear to have been made mainly for surface condensers using brass tubes wherein:

(i) Contribution of tube wall thermal resistance is negligible because of better thermal conductivity of brass and thinner tubes in view of low operating pressures.

(ii) Water velocity and temperature are low thus giving lower Reynolds Number.

(iii) Heat flux is relatively low i.e., of the order of 7,500 to 16,000 (20,000 in extreme cases) Kcal/hr m^2 of steam side area.

These features are conspicuously absent in case of feedwater heaters wherein:

(i) Contribution of tube wall to thermal resistance may or may not be negligible. Whereas it is only about 3.6 percent for LPH I to V and about 6 to 8 percent for HPH I(b) and HPH II(b), it is as high as 43 to 47 percent for HPH I(a) and HPH II(a) i.e., for condensing sections. It is thus seen that it is the tube wall that offers maximum resistance to heat transfer for condensing sections of HP

heaters. Such a high contribution of tube wall to total thermal resistance may be attributed to relatively thicker tubes (25/15 mm OD/ID) and a relatively low thermal conductivity of carbon steel (about 40 percent of that for admiralty metal). Percent contribution of tube wall for desuperheating sections of HP heaters is relatively lower primarily because of the fact that absolute value of r_o itself is quite large for these sections.

(ii) Water velocity and temperature encountered are high thus giving higher Reynolds Number on water side. A perusal of TABLE VI to VIII reveals that water side $Re = 58,740$ to $2,52,612$. Such high values of Re inevitably increase h_i thus lowering r_{i0} . Coiled tubes for HP heaters also aid in lowering r_{i0} .

(iii) Heat flux is relatively high. In the case of feedwater heaters analysed here, design heat flux value ranges between $13,600$ to $52,600$ Kcal/hr m^2 of steam side area.

It may thus be seen that in case of LP heaters, steam side resistance is the major resistance and in case of HP heaters, tube wall resistance is the highest. Thus the ideology of Wilson Plot does not seem to be applicable to feedwater heaters in the sense that water side resistance is not the controlling/major resistance. However, the sum of steam side and tube wall resistances may not vary much over a range of tests for the same heater and in that restricted sense, Wilson Plot technique may be applicable.

Since no consistent actual field data were available, the graphical technique could not be applied.

5.4 MEAN TEMPERATURE DIFFERENCE

TABLE XI shows LMTD as well as Δt for various heaters.

TABLE XI

MEAN TEMPERATURE DIFFERENCE, °C

	LPH I	LPH II	LPH III	LPH IV	LPH V	HPH I*			HPH II*		
						(a)	(b)	(c)	(a)	(b)	(c)
LMTD	5.03	8.55	5.91	10.21	16.75	16.50	156.44	19.06	15.50	80.20	17.17
Δt	6.17	10.79	8.90	12.95	16.19	23.39	176.08	-	21.70	85.95	-

*(a) Condensing Section, (b) Desuperheating Section
(c) Entire Heater.

It may be recalled here that whereas an attempt has been made to account for the degree of superheat while working out LMTD, Δt merely represents the difference between saturation steam temperature and bulk-mean temperature of feedwater for various heaters except desuperheating sections of HP heaters.

A perusal of TABLE XI reveals that in general, LMTD values are lower than Δt values. This difference is maximum in case of desuperheating section of HPH I where, incidentally, the degree of superheat is also maximum (about

256°C).

An examination of the product of the terms h_o & Δt_o for various heaters reveals that this value is substantially higher (by about 20 to 56 percent) than the design value of heat flux including the effect of drip enthalpy. It shows that:

(i) The actual temperature drop may be lower than Δt_o .

(ii) The design heat flux values are lower so as to account for fouling of tube surfaces etc.

While working out U_{C10} , Δt has been split up into Δt_o , Δt_i and Δt_{wl} . If LMTD were taken as Δt , different set of results would have been obtained. However, the concept of LMTD itself is subject to a number of simplifying assumptions not many of which are realised in practice, in the strictest sense. For instance, counter flow pattern has been assumed for LPH III to V and desuperheating sections of HP heaters and parallel flow pattern for condensing sections of HP heaters. As a matter of fact, however, the flow pattern is a complex combination of parallel, counter and crossflow conditions. As discussed in Section 2.3.2, Van Hengel's (54) comments bear out this view that LMTD concept is true for a single spot in a heat exchanger and is not applicable to the equipment as a whole.

5.5 FOULING OR DIRT COEFFICIENT

TABLE XII shows various approximate total values of fouling or dirt coefficient for both surfaces of heater tubes, expressed on the basis of steam side area.

TABLE XII

FOULING COEFFICIENT $\times 10^5$, hr m² °C/Kcal

Sr. No.	Dirt Coeff:	LPH					HPH I*			HPH II*		
		I	II	III	IV	V	(a)	(b)	(c)	(a)	(b)	(c)
1.	r_d	4.75	NA	1.01	4.74	NA	-ve	74.62	-ve	NA	NA	NA
2.	r_d'	6.90	Nil	5.85	7.05	16.60	-	-	-ve	-ve	26.19	-ve
3.	r_d (TEMA)	22.06	33.88	33.88	33.88	44.12	44.38	44.38	-	44.38	44.38	-

*(a) Condensing Section, (b) Desuperheating Section,
(c) Entire Heater.

A perusal of TABLE XII brings out the following:

(i) For HP heaters, fouling coefficients work out to be negative which means that U_{Do} or U'_{Do} values are greater than U_{Clo} values.

(ii) TEMA (22) dirt or fouling coefficients are, in general, much higher than those obtained by using equation (7).

It may be recalled here that U_{Do} or U'_{Do} values have been worked out by dividing total heat transferred 'Q' by the product of heat transfer area and LMTD. A slight variation in LMTD values can change the dirt coefficients from negative to

positive and this variation, as brought out in Section 5.4, is not ruled out.

The dirt coefficients given by TEMA (22) have been very severely criticized by Mueller in Reference (12), on the following lines:

- (i) The list is based on experience of members but it is doubtful that much test data were available as a basis.
- (ii) The tube material is ignored as a factor affecting fouling.
- (iii) Variation in design/operating parameters may effect fouling.
- (iv) The coefficient is meaningless unless time interval for building up of a specified coefficient is given as well as information concerning its linearity with time.

It may, however, not be in order to dismiss outright TEMA (22) recommendations which are based on the experience of members. These recommendations are widely used and the calculation procedure for U_{C10} needs to be investigated.

Various factors which may affect U_{C10} values are listed below:

- (i) Drip/Drain Enthalpy: About 5.74, 14.34, 6.59 and 6.08 percent of the total heat transferred for LPH, III, IV, V and HPH I respectively is contributed by heater drip cascading from a higher stage heater to a lower stage heater.

Its effect has been taken care of only partly on tube side coefficient in the shape of increased water temperature/velocity and this effect is absent on steam side where the effect of only steam flow/condensation has been considered.

Contribution of chimney and gland steam condensers has also been ignored. Further, no undercooling of the condensate has been considered.

(ii) Degree of Superheat : Whereas inlet steam to LPH I has a dryness fraction of about 0.975, it has a degree of superheat in the range of about 2°C to 256°C for LPH II to HPH II. The percentage enthalpy drop due to desuperheating process ranges from about 4 to 15.6. While working out h_o , however, steam has been assumed to be in saturation state (except in case of desuperheating sections) and also entire mass of steam is assumed to have been completely condensed on the tubes. This may, however, not be wholly true in practice.

(iii) Tube Wall Temperature Variation has been ignored for sake of simplicity, although its effect needs to be considered (Ref. Section 2.2.8).

(iv) Viscosity Correction as proposed by Sieder and Tate (38) has been used in the calculations, although different correction factors have been proposed by various research workers (Ref. Section 2.2.6).

(v) Effect of Vapour Velocity & Electrostatic Field on film condensation (Ref. Section 2.2.8) is to increase

heat transfer. It may be noted that feedwater heaters are installed not far away from turbo-alternators and the influence of electrostatic/magnetic field cannot be ruled out.

(vi) Radiation Effects have been ignored, although these may not be insignificant in case of HP heaters.

(vii) Reynolds Number : Strictly speaking, equation (9), as corrected by Sieder & Tate (38) applies very well only in the range $Re = 10,000$ to $1,20,000$ and $Pr = 0.7$ to 120 . Even equation (8) is valid in the range $Re = 5,000$ to $3,00,000$ within 20 percent (34). Thus h_1 , as worked out by equation (9), may not represent the true value of water side coefficient for HP heaters where Re ranges from about $1,81,831$ to $2,52,612$.

(viii) Pressure Drop : Steam/water side pressures in various heaters have been assumed to be constant for sake of simplicity. In practice, however, pressure drop does take place in various flow paths.

(ix) Film Condensation has been assumed to take place, although a slight contribution of dropwise condensation, thus improving heat transfer, cannot be ruled out.

(x) Non-Condensables : Whereas most of the factors listed above may tend to increase U_{C10} , the presence of even small quantities of non-condensables in steam such as air tend to reduce the steam side heat transfer coefficient drastically.

The cumulative effect of various factors (i) to (x) brought out above may be to increase the value of U_{C10} a little bit so that the value of fouling coefficient may not be much in disagreement with those of TEMA (22).

A comparison of U_{D0} and U_{C10} values (TABLE IX) reveals that with the exception of LPH II and condensing sections of HPH I & II, the ratio U_{D0} / U_{C10} ranges between 0.620 to 0.965, giving an arithmetic mean value of about 0.82 which is not much different from the suggestion of taking $U_{D0} / U_{C10} = 0.85$ made by Faras & Ozisk(2) and Kern (13), as discussed in Section 2.2.3.



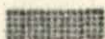
CHAPTER 6CONCLUSIONS

From the foregoing discussion of results, following inferences may be drawn:

1. The values of overall heat transfer coefficient for various feedwater heaters assumed by the manufacturer are in fair agreement with the calculated figures.
2. The figure of overall heat transfer coefficient is different for different feedwater heaters, being somewhat lower for HP heaters wherein the presence of a desuperheating section tends to lower this coefficient.
3. The actual value of overall heat transfer coefficient for a given heater varies with load and many other operating parameters, including the accumulation of dirt/sludge/scale and presence of air/non-condensable gases in steam.
4. The ideology of Wilson Plot that water side resistance is the controlling resistance, does not appear to be applicable to feedwater heaters wherein steam side resistance/tube wall resistance for LP and HP heaters respectively turns out to be the major resistance to the heat transfer phenomenon.
5. Evaluation of mean temperature difference between the shell side fluid and tube side fluid can, at best, be an approximation only.
6. Theoretical formulations yield a fair estimate of overall heat transfer coefficient for various heaters, although the analysis is restricted in the sense that a number of simplify-

ing assumptions are made.

7. In order to obtain more reliable/realistic results of optimal distribution of heat transfer areas amongst various feedwater heaters, actual values of overall heat transfer coefficient for various heaters should be used.



CHAPTER 7

SUGGESTIONS FOR FURTHER WORK

The present work is limited to the evaluation of overall heat transfer coefficient of various low pressure and high pressure feedwater heaters under full load rated conditions with all the heaters "charged" i.e., in circuit and various parameters in equilibrium state i.e., steady state conditions exist.

In practice, however, these are certain occasions when one or more of the heaters in the chain have to be by-passed for the purpose of maintenance or for certain operational requirements. Further, it is very rare that power plant equipment operates under full load rated condition. More often than not, it is either operated at part load or at over load and also at varying load. Under these conditions, mass and heat transfer pattern is likely to change which needs to be examined in a separate study.

In order that results of theoretical formulations may be of real practical utility, it is imperative that these results be compared with those actually obtaining in the power plant. Unfortunately, the author was not in a position to lay hands on consistent field data for operation of the various feedwater heaters at full load rated conditions. A separate study can be made to determine the heat transfer behaviour of various heaters under full load

operating conditions by taking actual measurement of various parameters such as flow rates, temperatures and pressures at various stages.

The work can further be extended by determining optimal distribution of total heat transfer area amongst various feedwater areas by making use of overall heat transfer coefficient values computed for each feedwater heater.



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CARNOT CYCLE SUPERIMPOSED
ON RANKINE CYCLE FOR
SATURATED STEAM

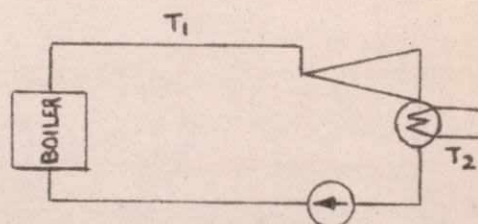
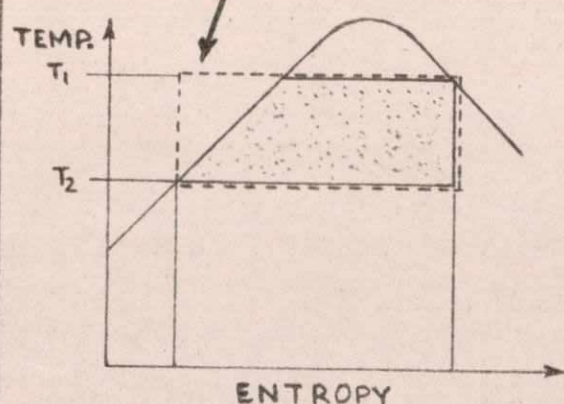


FIG. 1(A) WATER HEATING IN BOILER ONLY

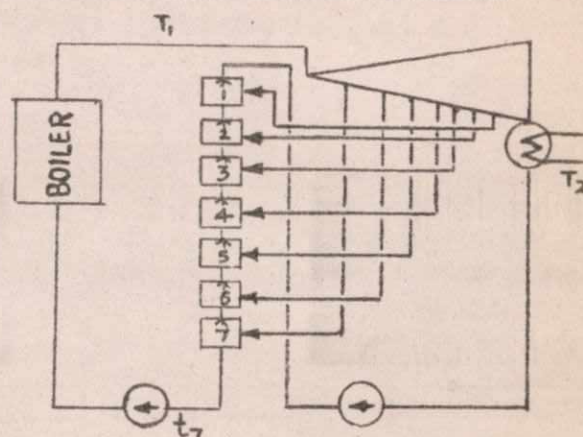
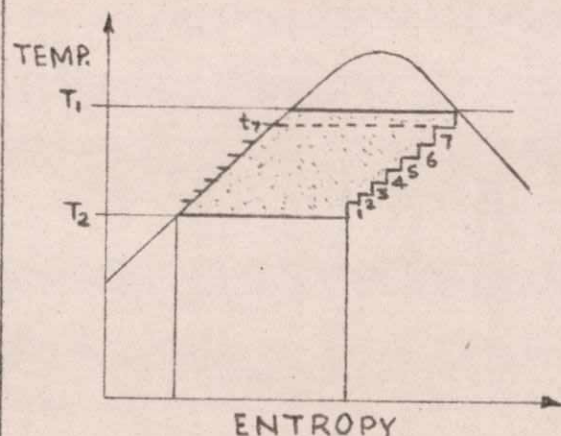


FIG. 1(B) SEVEN-STAGE FEED HEATING SYSTEM

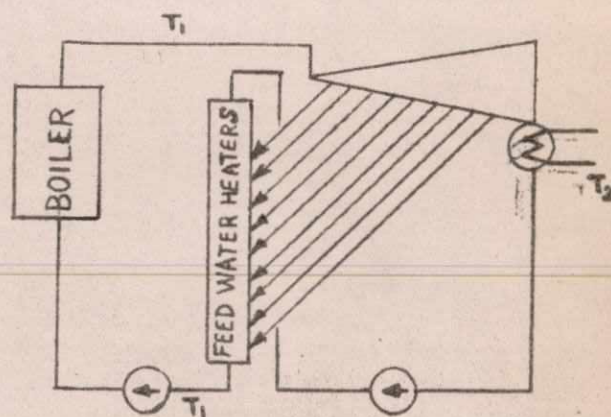
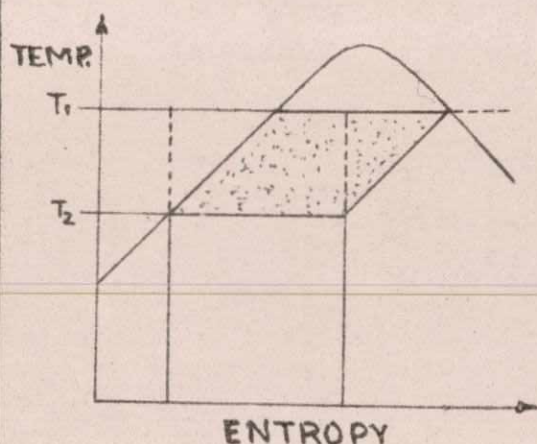


FIG. 1(C) FEED HEATING SYSTEM WITH AN
INFINITE NUMBER OF HEATERS

FIGS. 1(A;B,C) REGENERATIVE FEED HEATING(1)

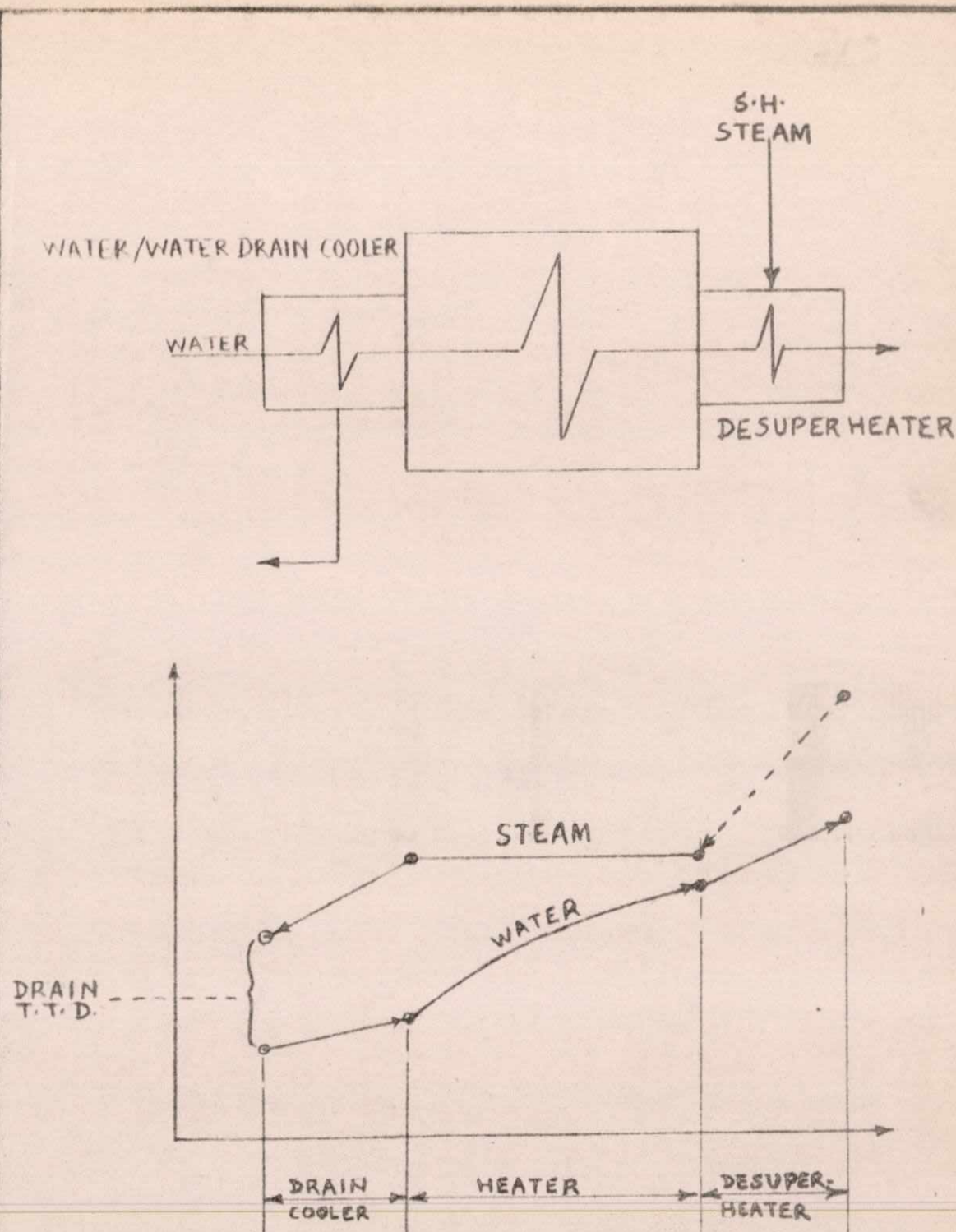


FIG.2. TEMPERATURE RISE OF FEED WATER IN A SURFACE TYPE HEATER (1)

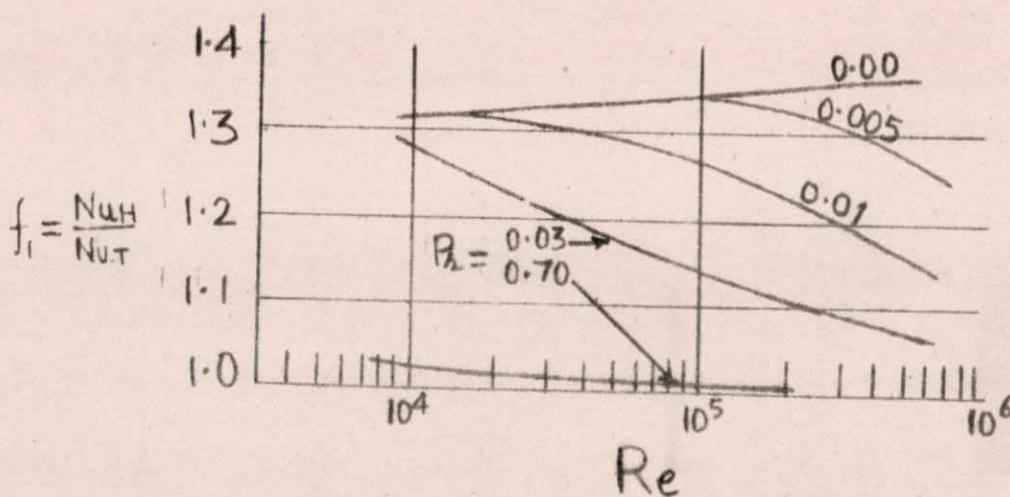


FIG. 3. RATIO OF NUSSULT NUMBER FOR
UNIFORM HEAT FLUX TO UNIFORM WALL
TEMPERATURE CONCEPT
FULLY DEVELOPED CONDITIONS IN A CIRCULAR TUBE (3)

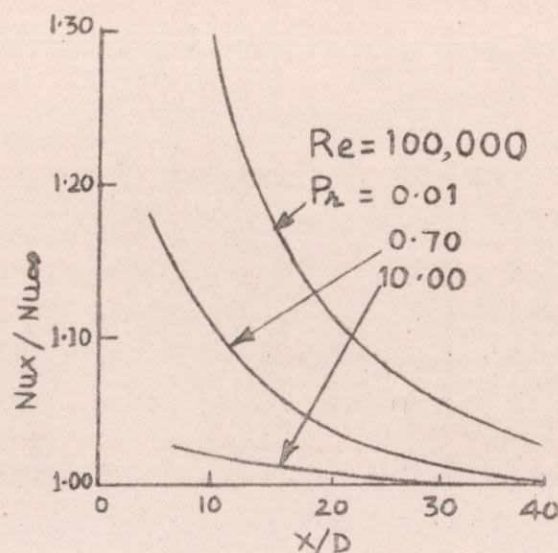


FIG.4(A) NUSSELT NUMBER IN THERMAL ENTRY LENGTH OF A CIRCULAR TUBE, CONST. HEAT RATE (12)

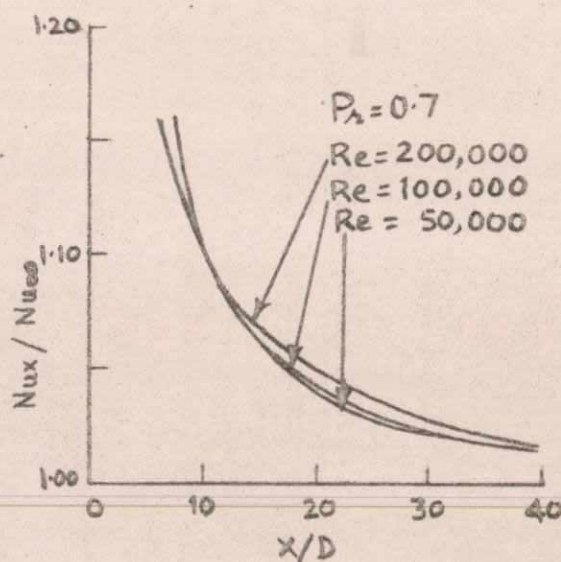


FIG.4(B) NUSSELT NUMBER IN THERMAL ENTRY LENGTH OF A CIRCULAR TUBE, CONST. HEAT RATE (12)

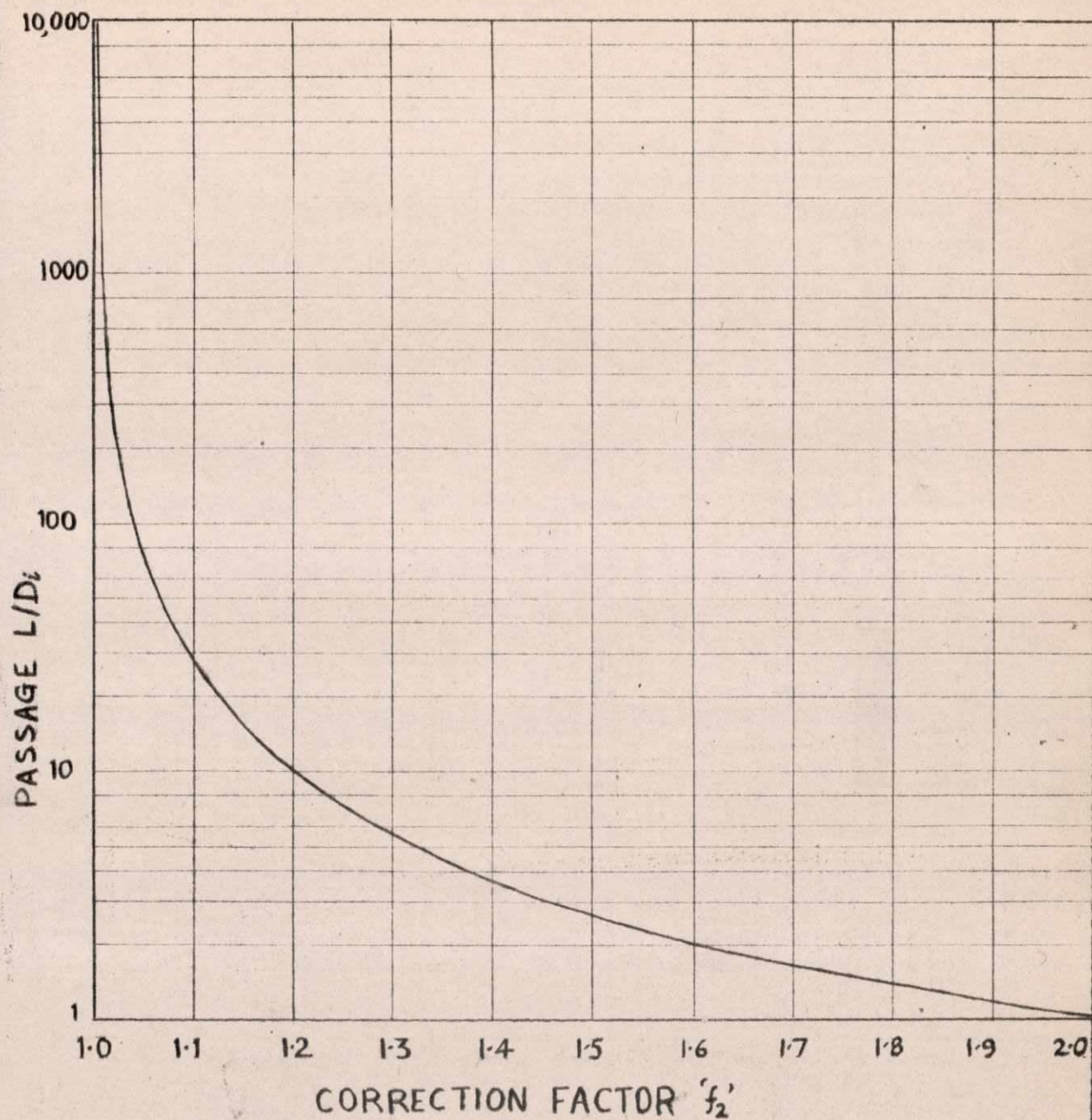


FIG.5. CORRECTION FACTOR TO OBTAIN AVERAGE HEAT TRANSFER COEFFICIENT IN THE TURBULENT FLOW ENTRANCE REGION(2)

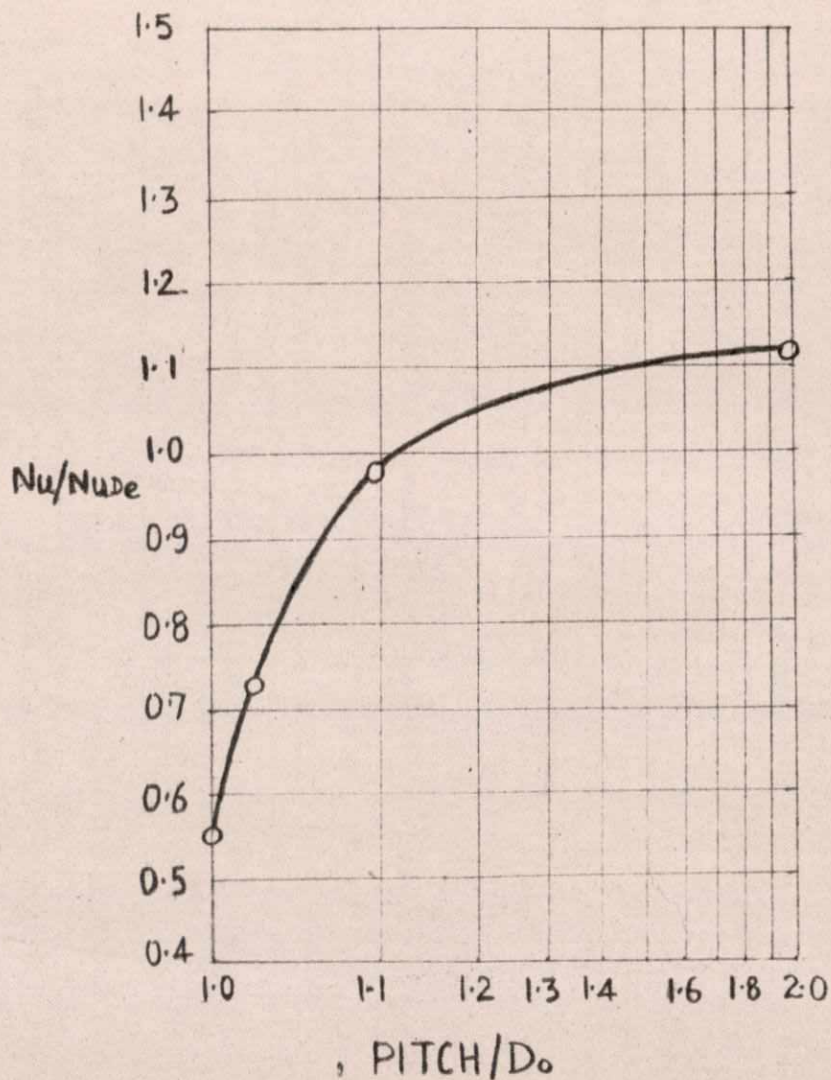


FIG. 6. , RATIO OF NUSSELT NUMBER
& NUSSELT NUMBER BASED ON HYDRAULIC
DIAMETER , $Re = 10^5$, Re INFLUENCE IS SMALL

(31)

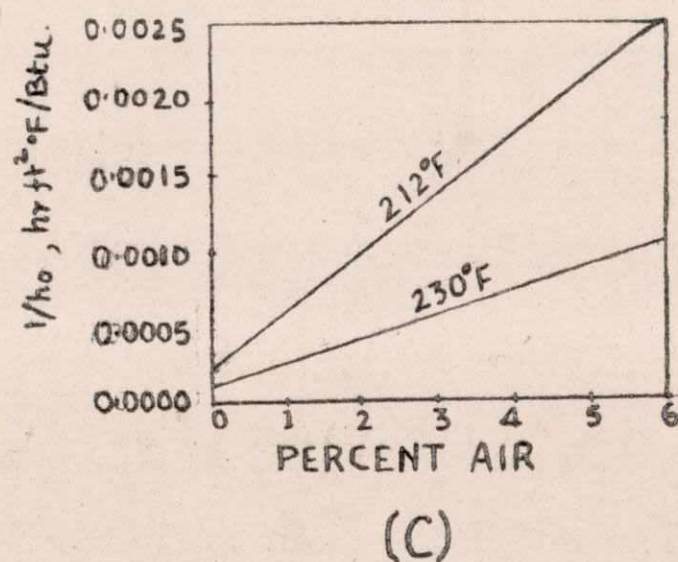
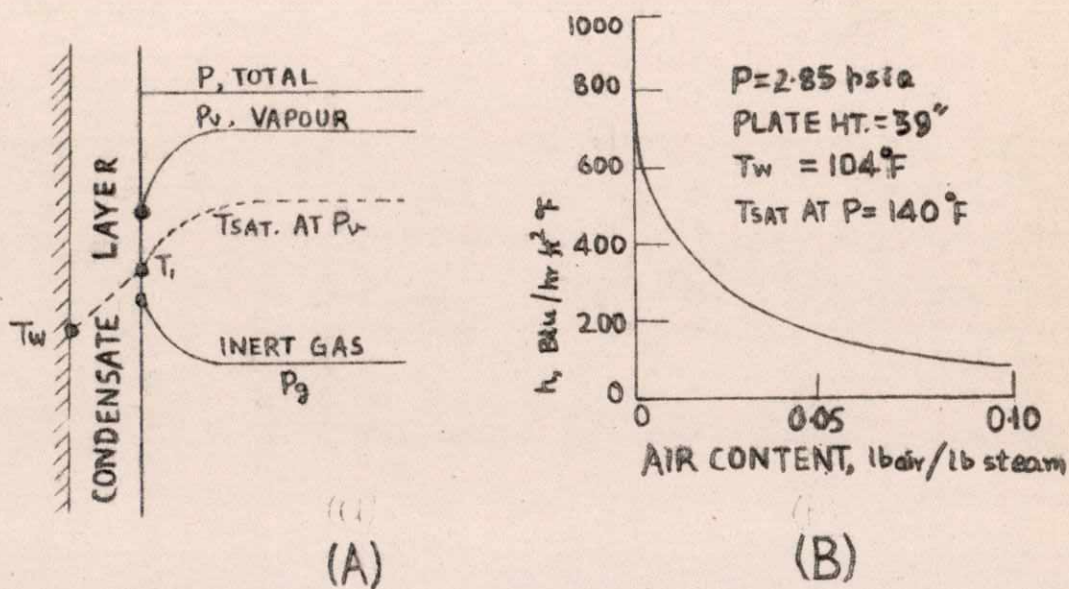


FIG. 7(A,B,C) EFFECT OF NONCONDENSABLE GAS ON CONDENSING VAPOUR HEAT TRANSFER COEFFICIENT

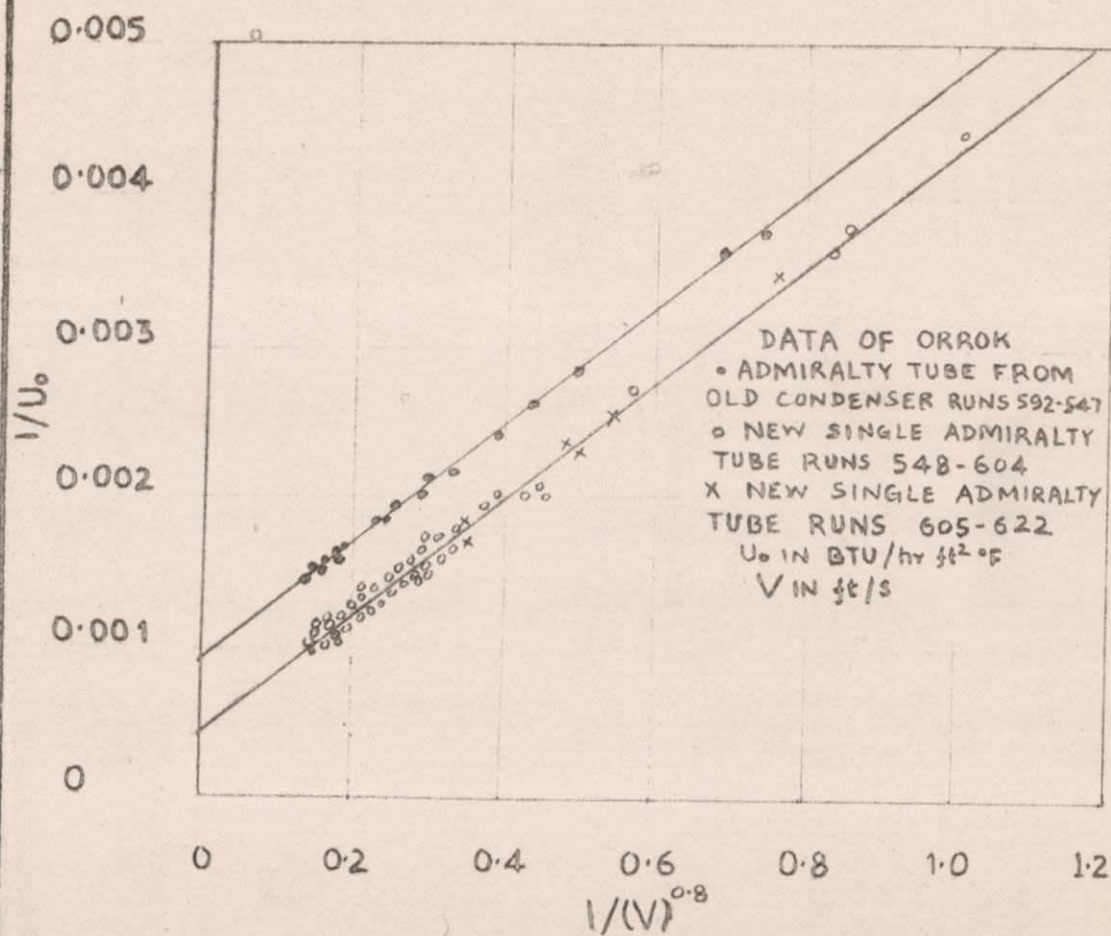


FIG. 8. WILSON PLOT(34)

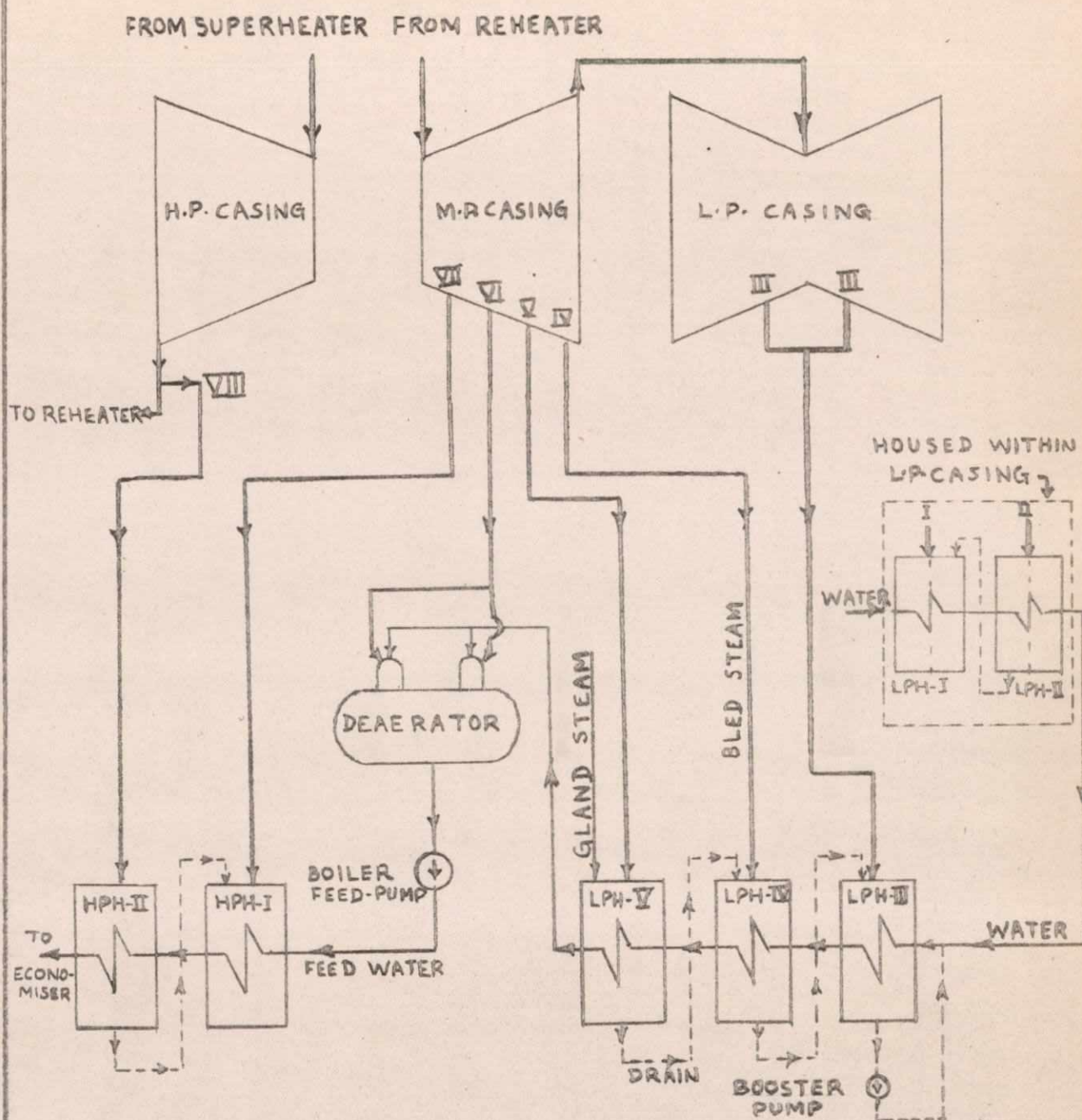
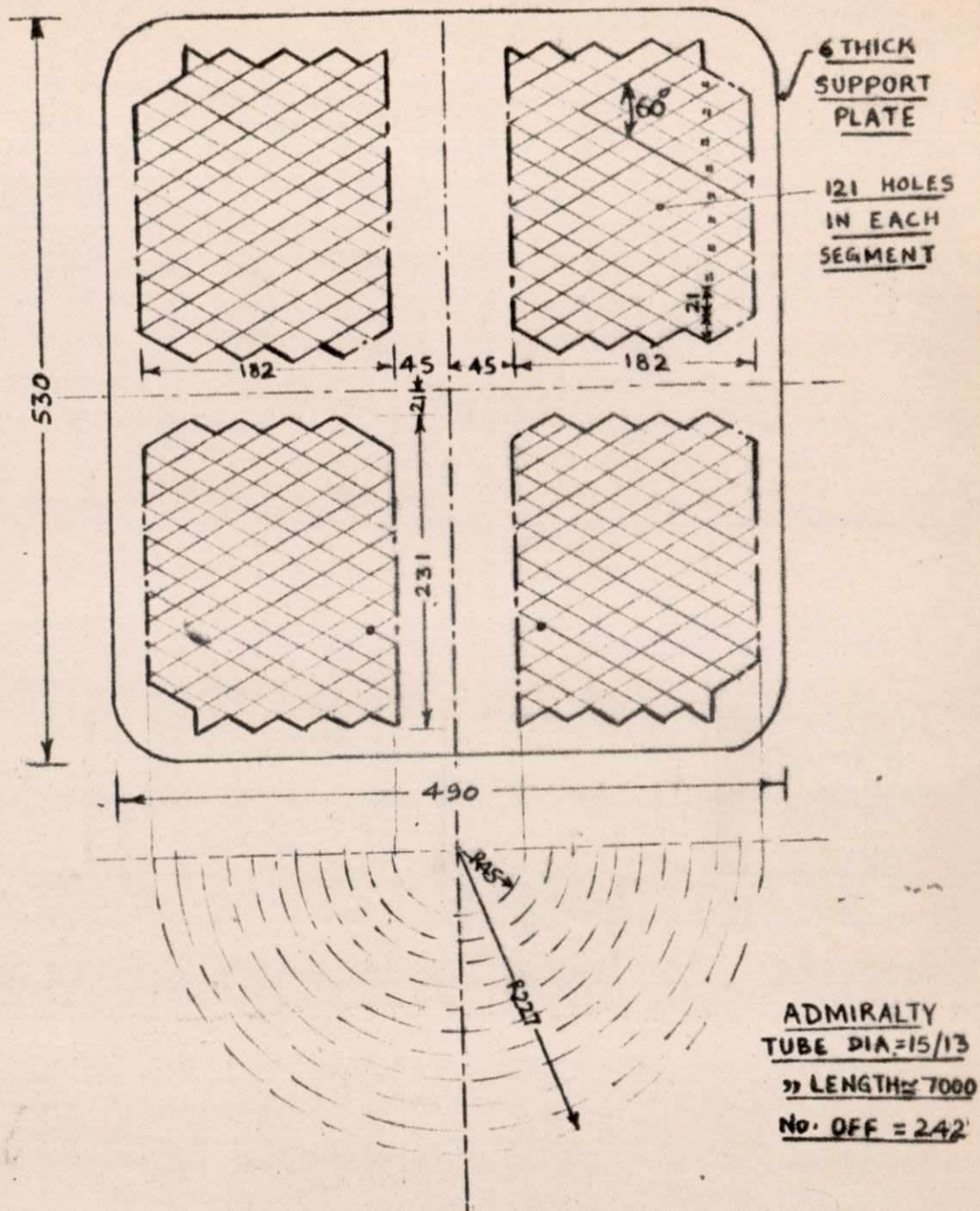
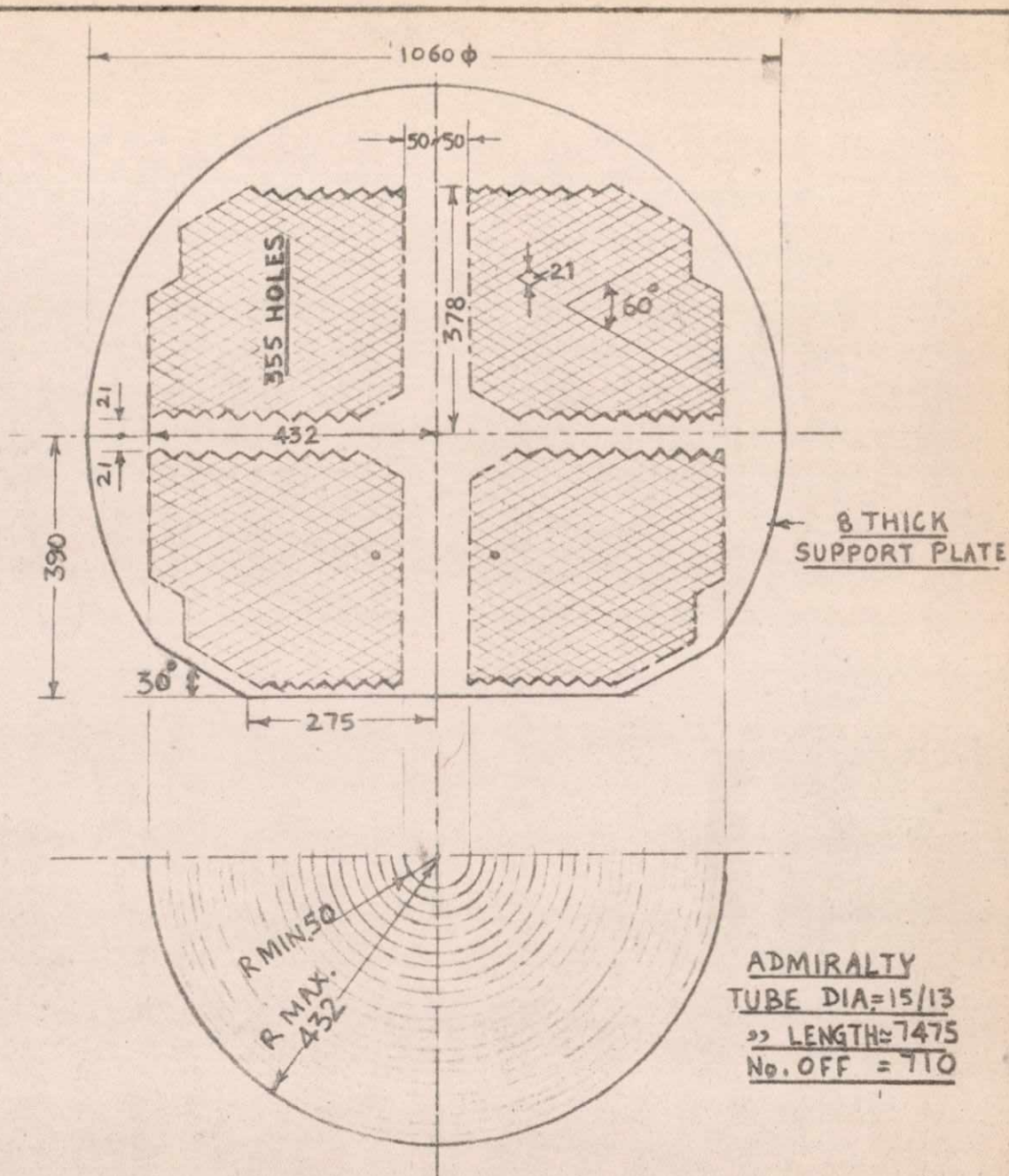


FIGURE 9
SIMPLIFIED FLOW DIAGRAM OF FEED
WATER HEATER CHAIN (110 MW UNIT)



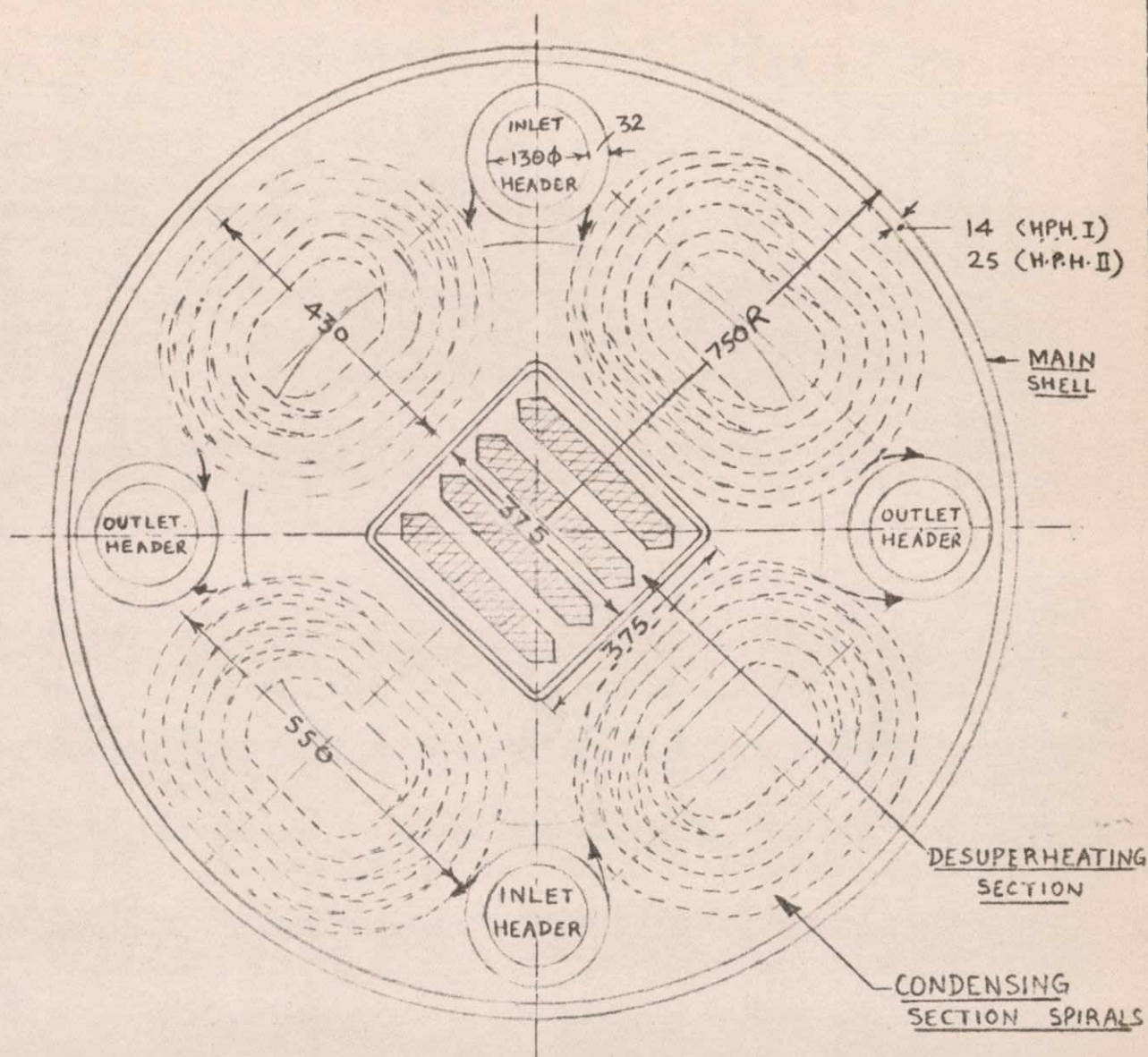
ALL DIMENSIONS ARE IN MILLIMETRES

FIG. 10. TUBE BUNDLE ARRANGEMENT FOR
L.P. HEATER No. I & II



ALL DIMENSIONS ARE IN MILLIMETRES

FIG.11. TUBE BUNDLE ARRANGEMENT FOR
L.P. HEATER No. III, IV & V.



CONDENSING SECTION

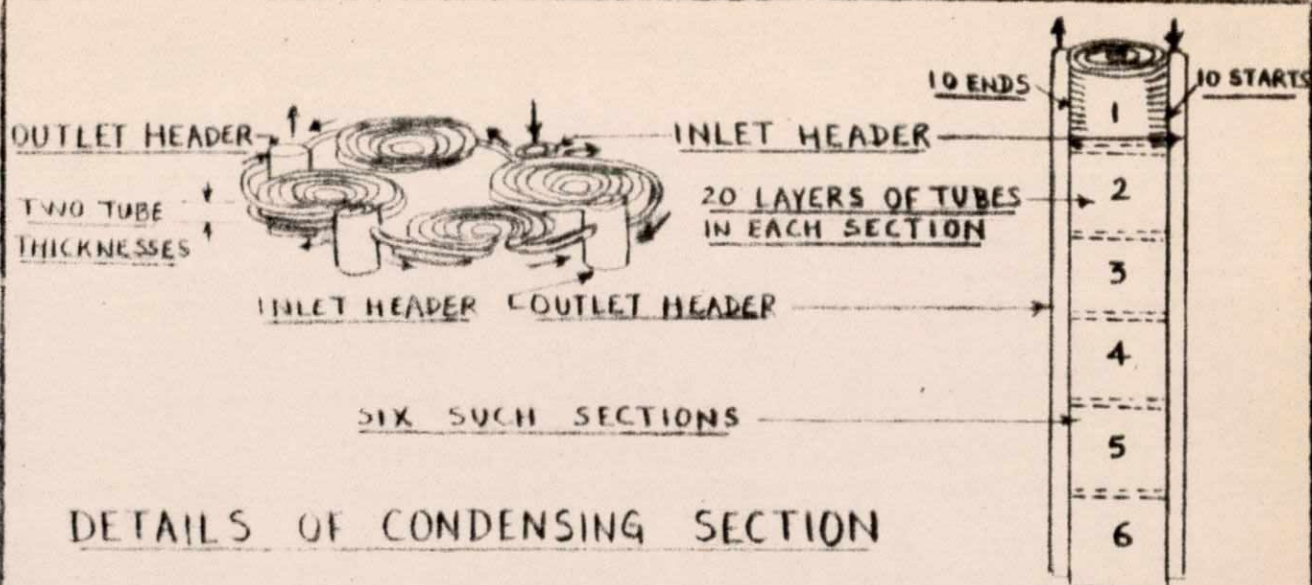
{ CARBON STEEL TUBES 25/15 DIA, 240 Nos.
EACH 13800 LONG BENT IN DOUBLE SPIRAL
EACH OF SIX TURNS FIRST DECREASING
THEN INCREASING (MEAN PERIMETER=1150)

DESUPERHEATING SECTION

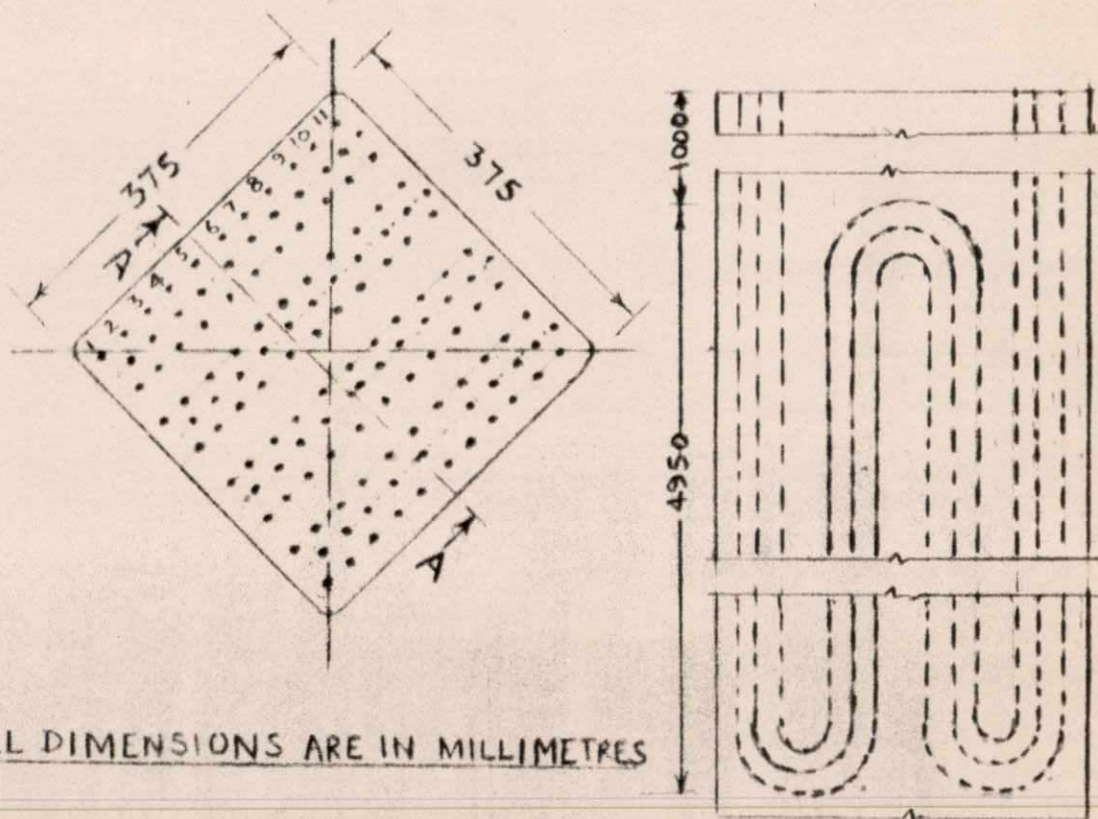
{ CARBON STEEL TUBES 25/15 DIA, 33 Nos.
EACH 18520 LONG BENT IN 'W' SHAPE

ALL DIMENSIONS ARE IN MILLIMETRES

FIG.12. H.P.H. I & II TUBE ARRANGEMENT

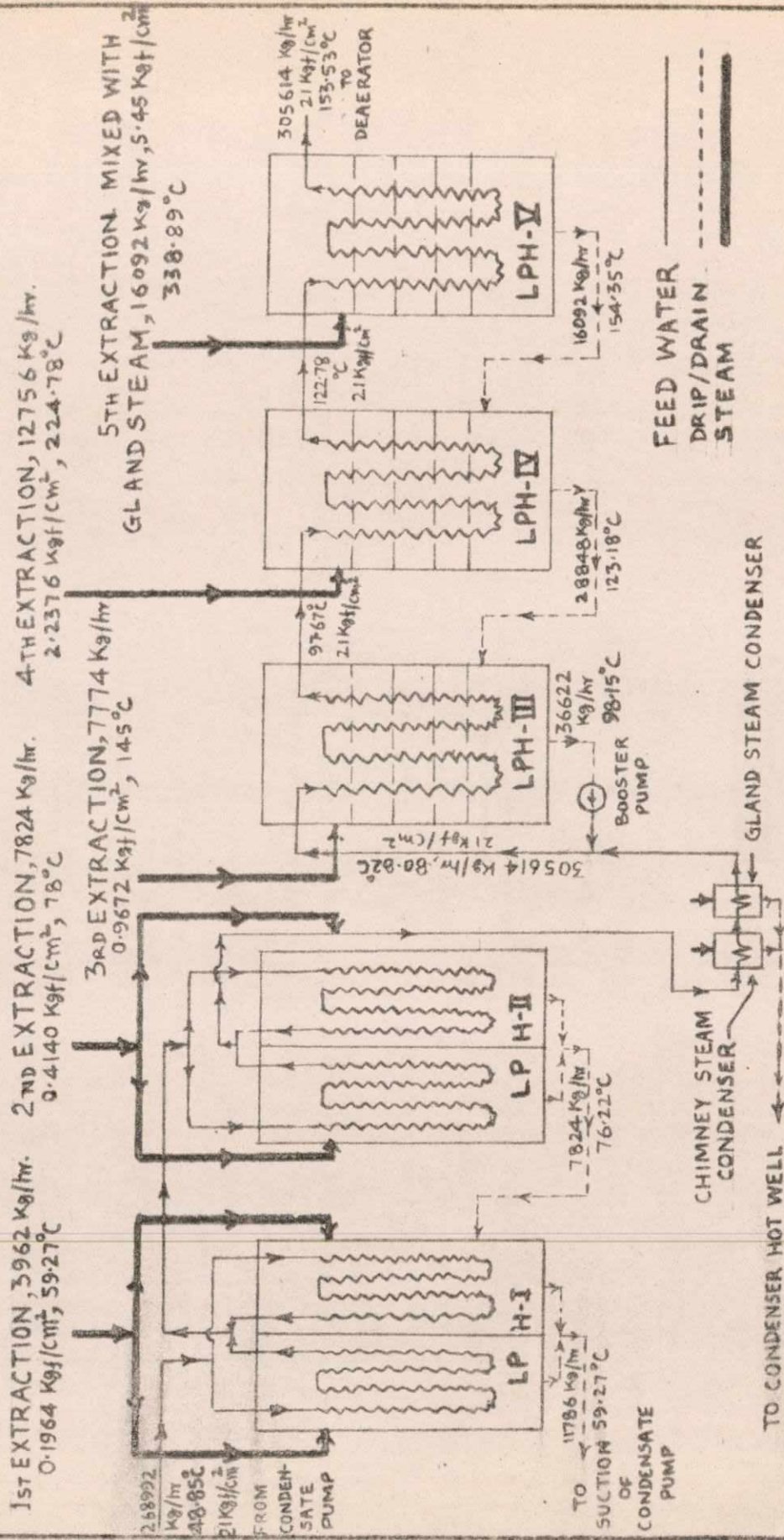


DETAILS OF CONDENSING SECTION



DETAILS OF DESUPERHEATING SECTION VIEW 'AA'

FIG.13. H.P.H. I & II TUBE DETAILS



- NOTES:
1. LPH I & II ARE HOUSED IN LPCASING OF THE TURBINE ITSELF.
 2. PRESSURE DROP IN VARIOUS HEATERS HAS BEEN NEGLECTED.
 3. BY-PASS LINES HAVE BEEN OMITTED FOR SAKE OF SIMPLICITY.

FIG. 14. DETAILED FLOW-DIAGRAM OF LP HEATERS

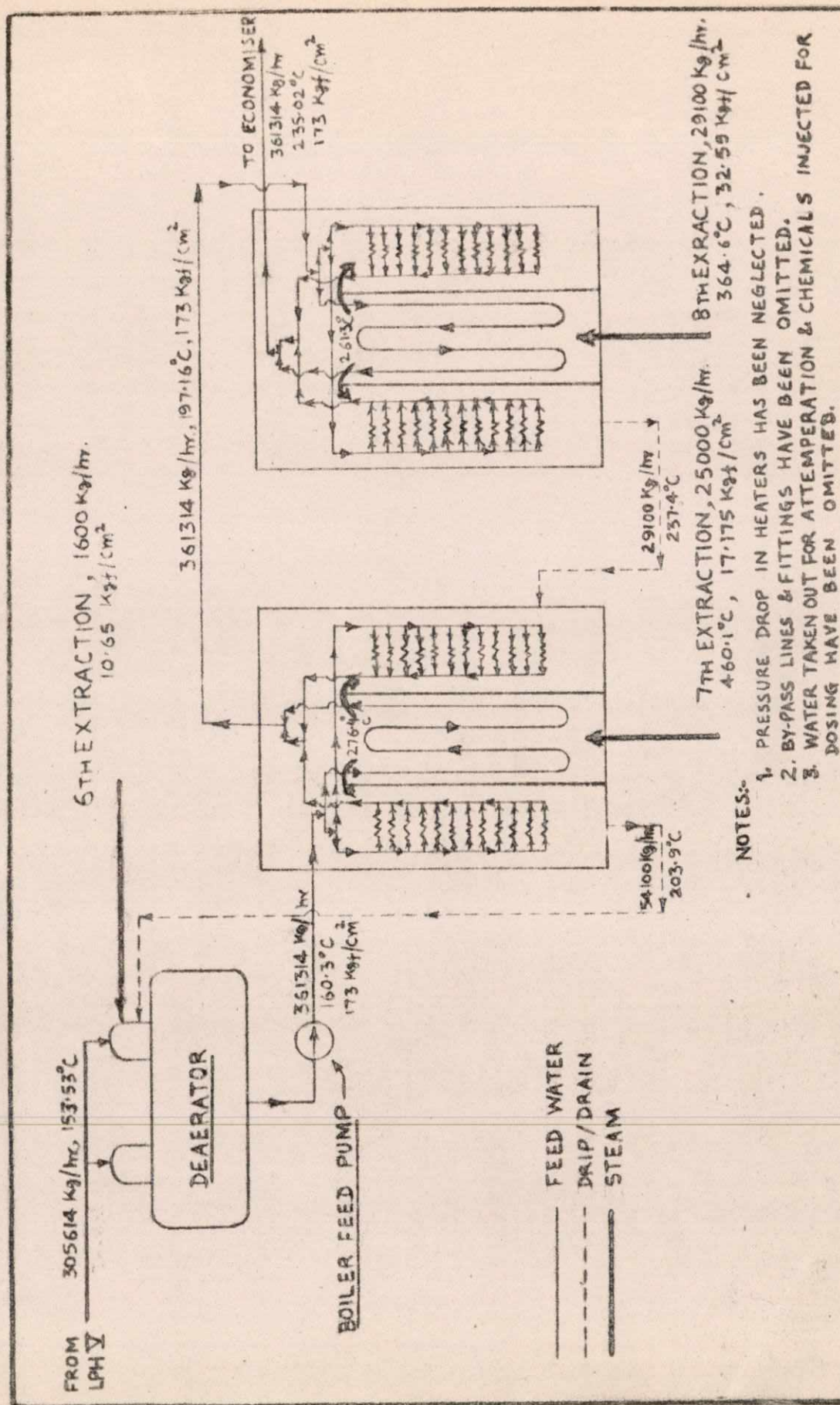
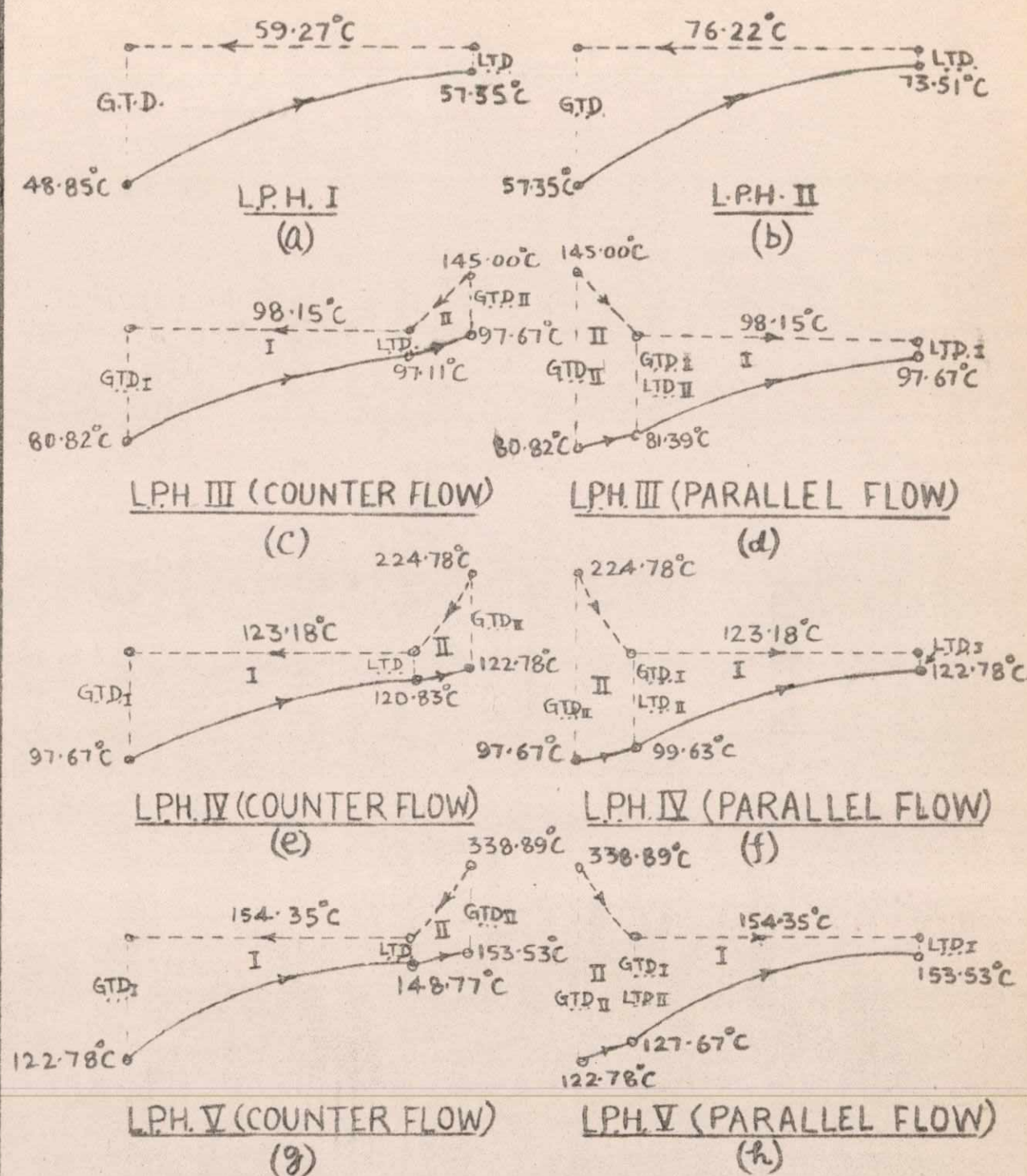
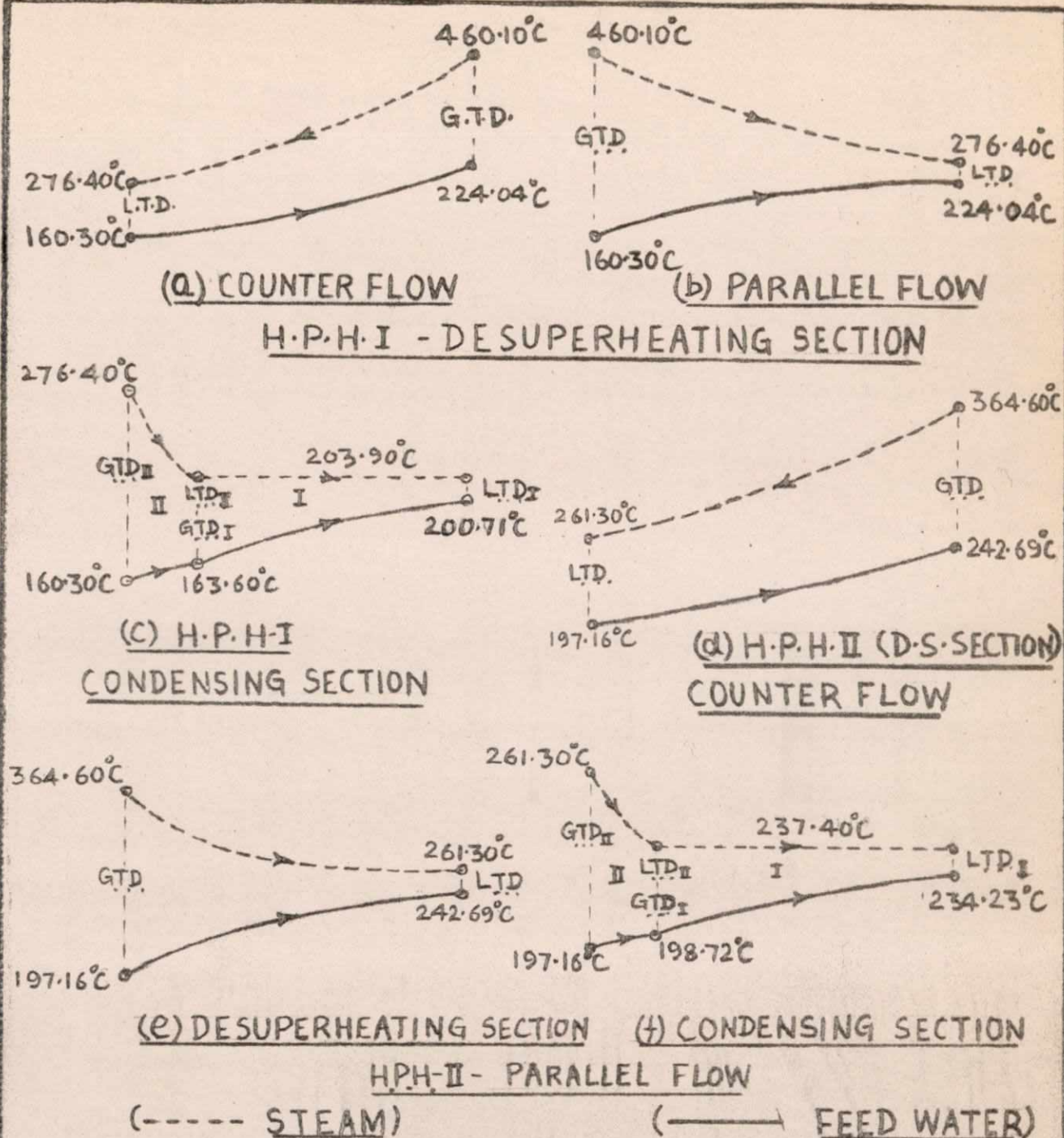


FIG.15. DETAILED FLOW-DIAGRAM OF HP HEATERS



(DATA OF FIGURES (d), (f) & (h) DO NOT APPEAR IN CALCULATIONS)

FIG.16. L.M.T.D. CALCULATIONS FOR
LOW PRESSURE FEED WATER HEATERS
(---- STEAM) (— FEED WATER)

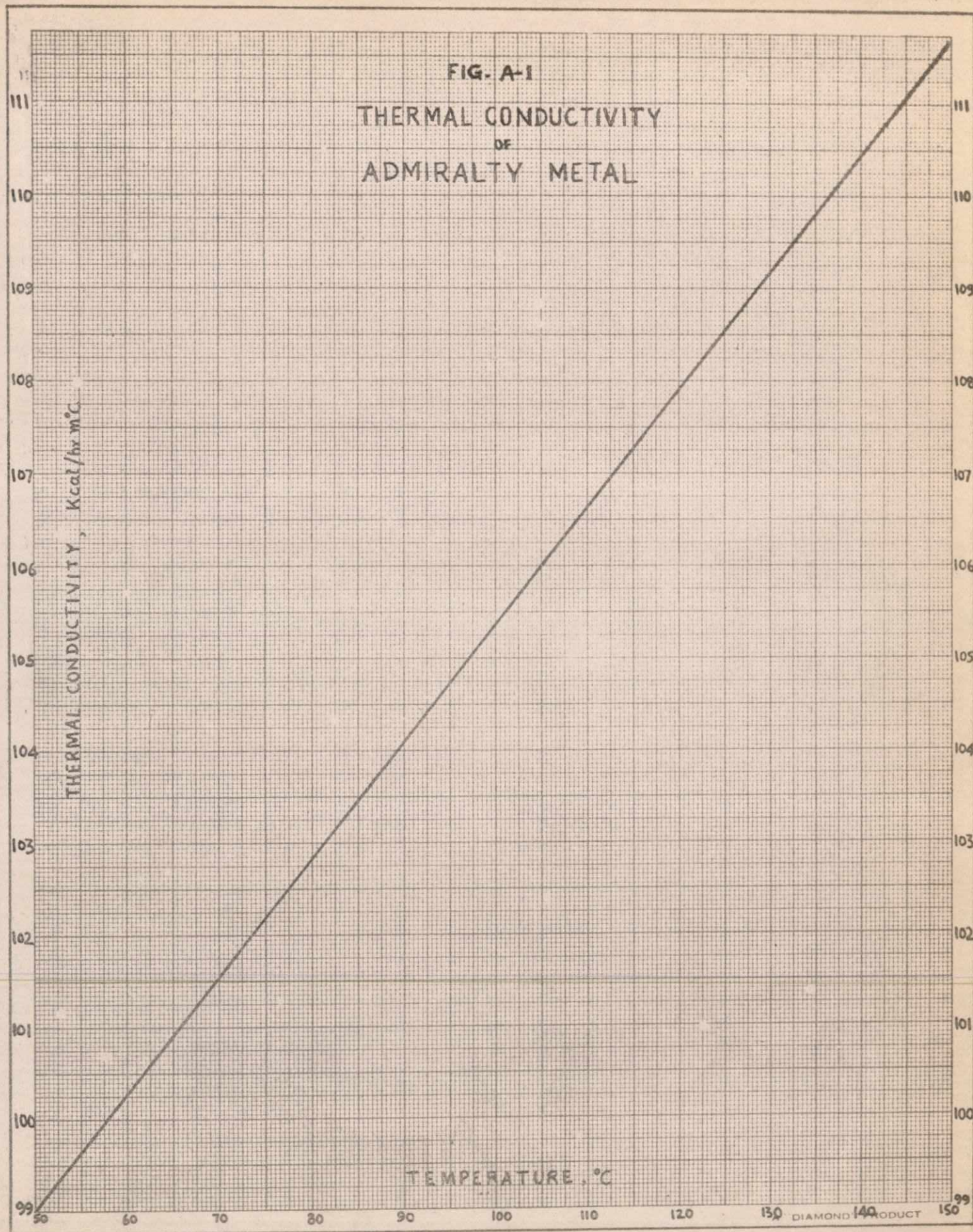


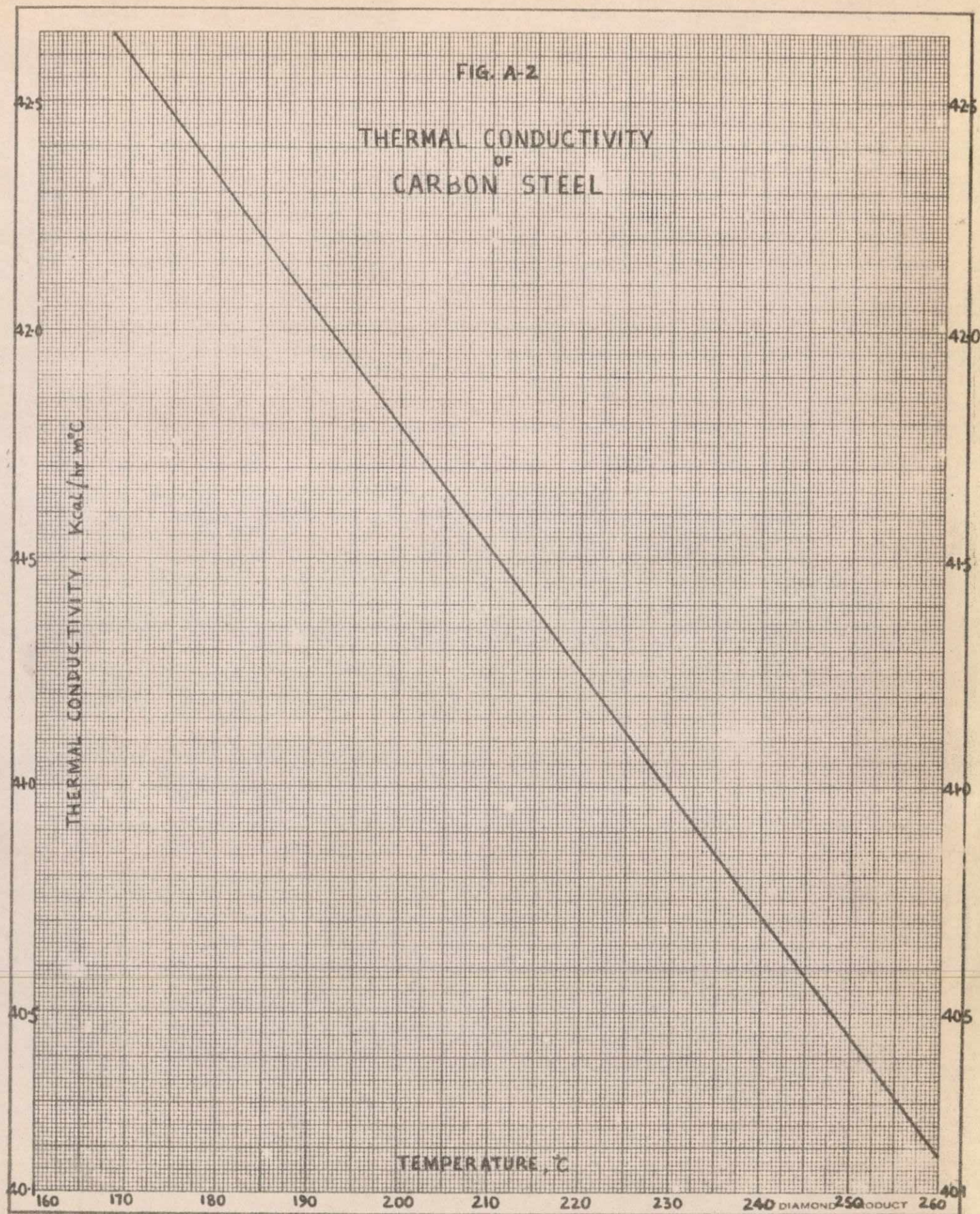
(DATA OF FIGURES (b) & (e) DO NOT APPEAR IN CALCULATIONS)

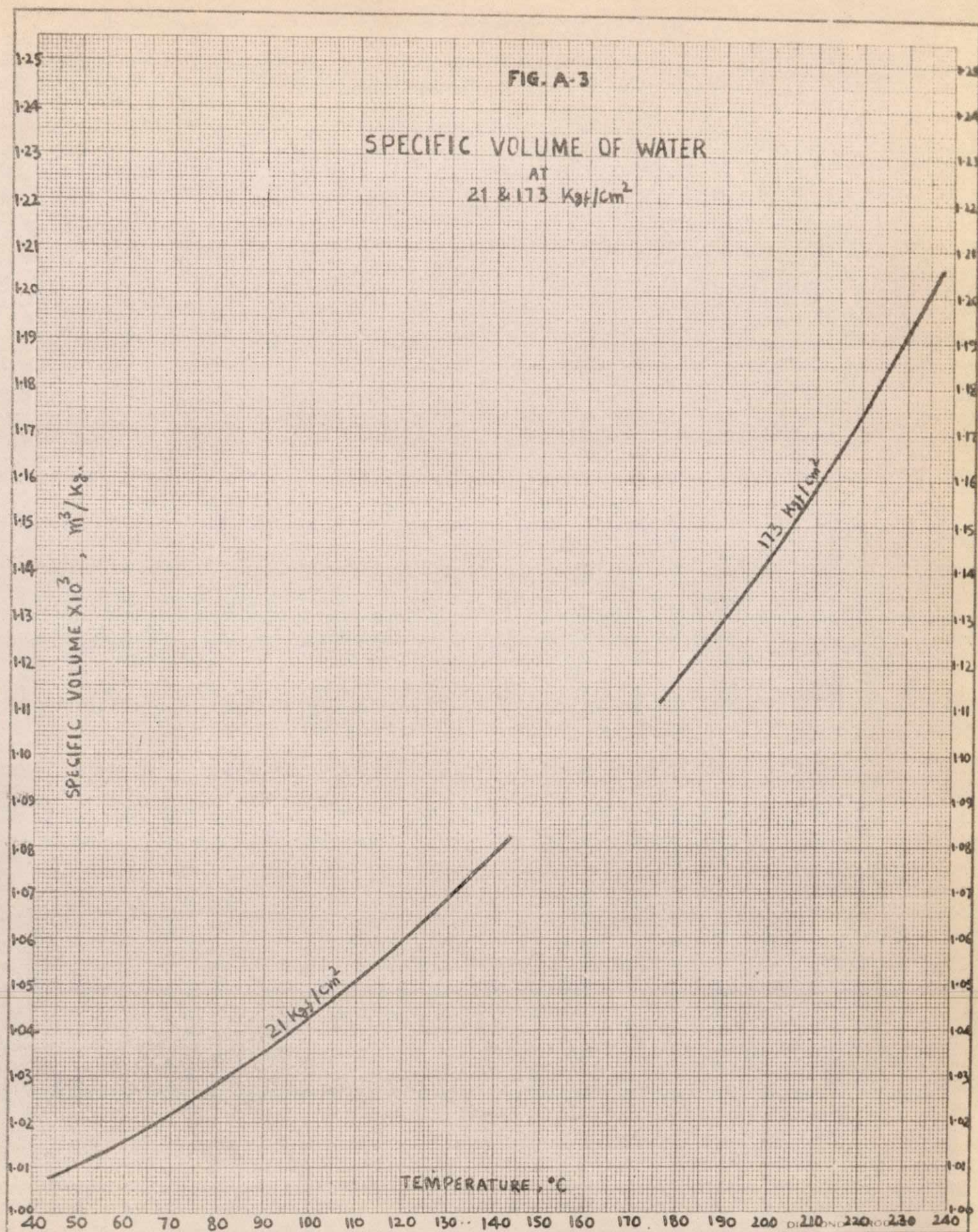
FIG.17. LMTD. CALCULATIONS FOR HIGH PRESSURE FEED WATER HEATERS

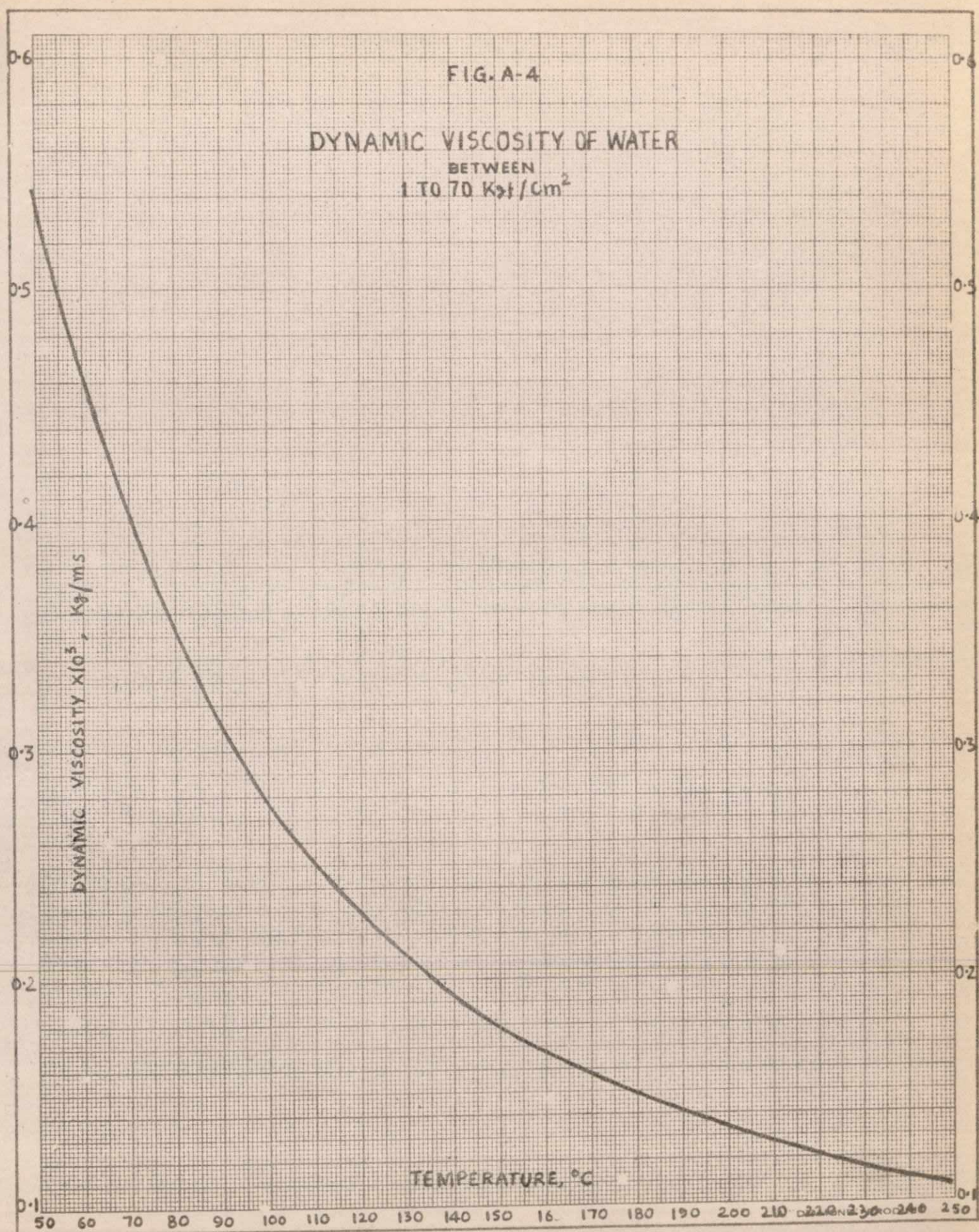
APPENDIXLIST OF VARIOUS PHYSICAL PROPERTIES CURVES

<u>FIGURE</u>	<u>TITLE</u>	<u>PAGE</u>
A-I	Thermal Conductivity of Admiralty Metal	I22
A-2	Thermal Conductivity of Carbon Steel	I23
A-3	Specific Volume of Water at 2I and I73 Kgf/cm ²	I24
A-4	Dynamic Viscosity of Water between I to 70 Kgf/cm ²	I25
A-5	Dynamic Viscosity of Water at I73 Kgf/cm ²	I26
A-6	Kinematic Viscosity of Water at 2I Kgf/cm ²	I27
A-7	Kinematic Viscosity of Water at I73 Kgf/cm ²	I28
A-8	Prandtl Number of Water at 2I Kgf/cm ²	I29
A-9	Prandtl Number of Water at I73 Kgf/cm ²	I30
A- IO	Thermal Conductivity of Water at I, 2I, 35 and I73 Kgf/cm ²	I3I
A-II	Dynamic Viscosity of Steam at I7 and 32.5 Kgf/cm ²	I32
A-I2	Prandtl Number of Steam at I7 and 32.5 Kgf/cm ²	I33
A-I3.	Thermal Conductivity of Steam at I7 and 32.5 Kgf/cm ²	I34









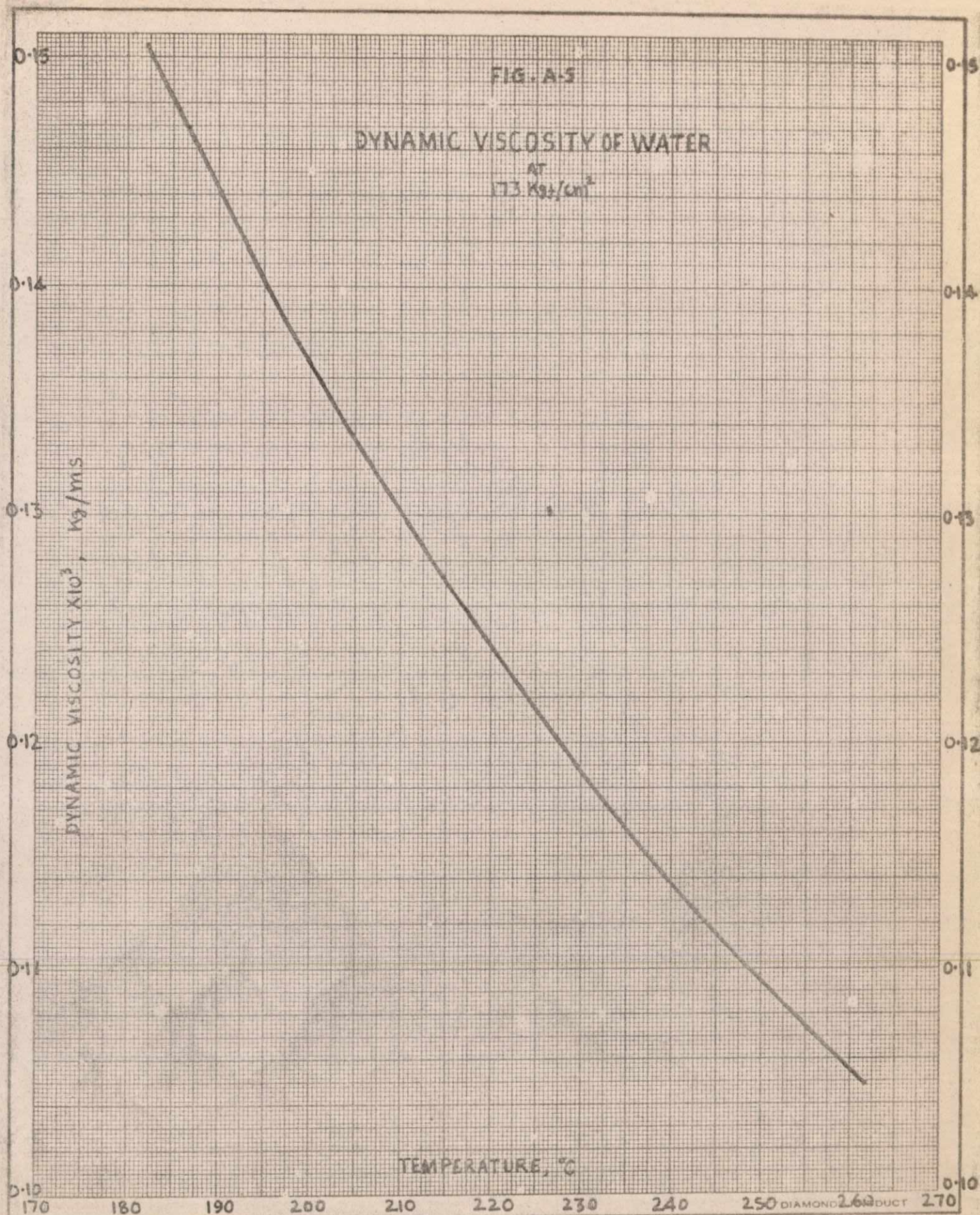
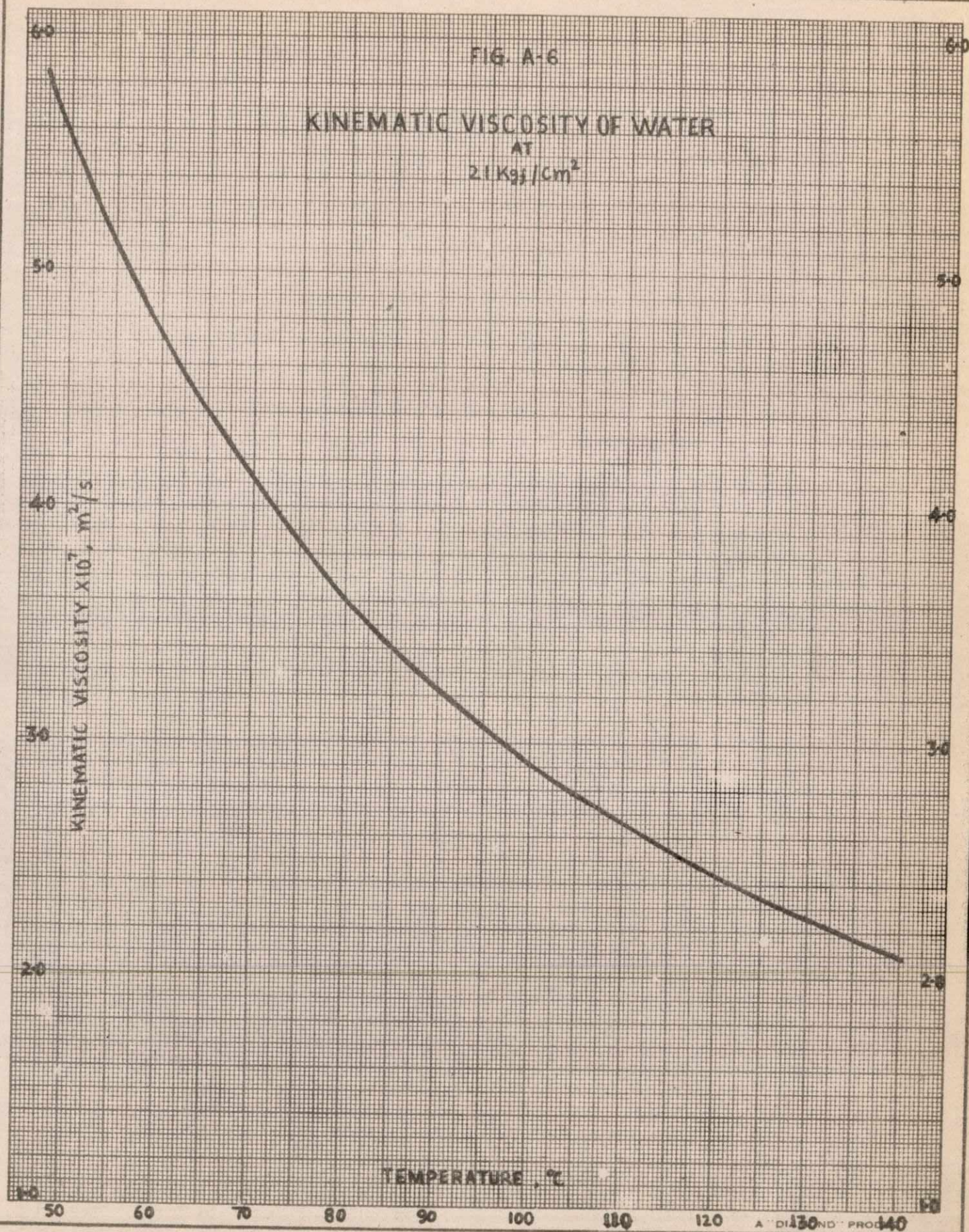


FIG. A-6
KINEMATIC VISCOSITY OF WATER
AT
2.1 Kg/cm²



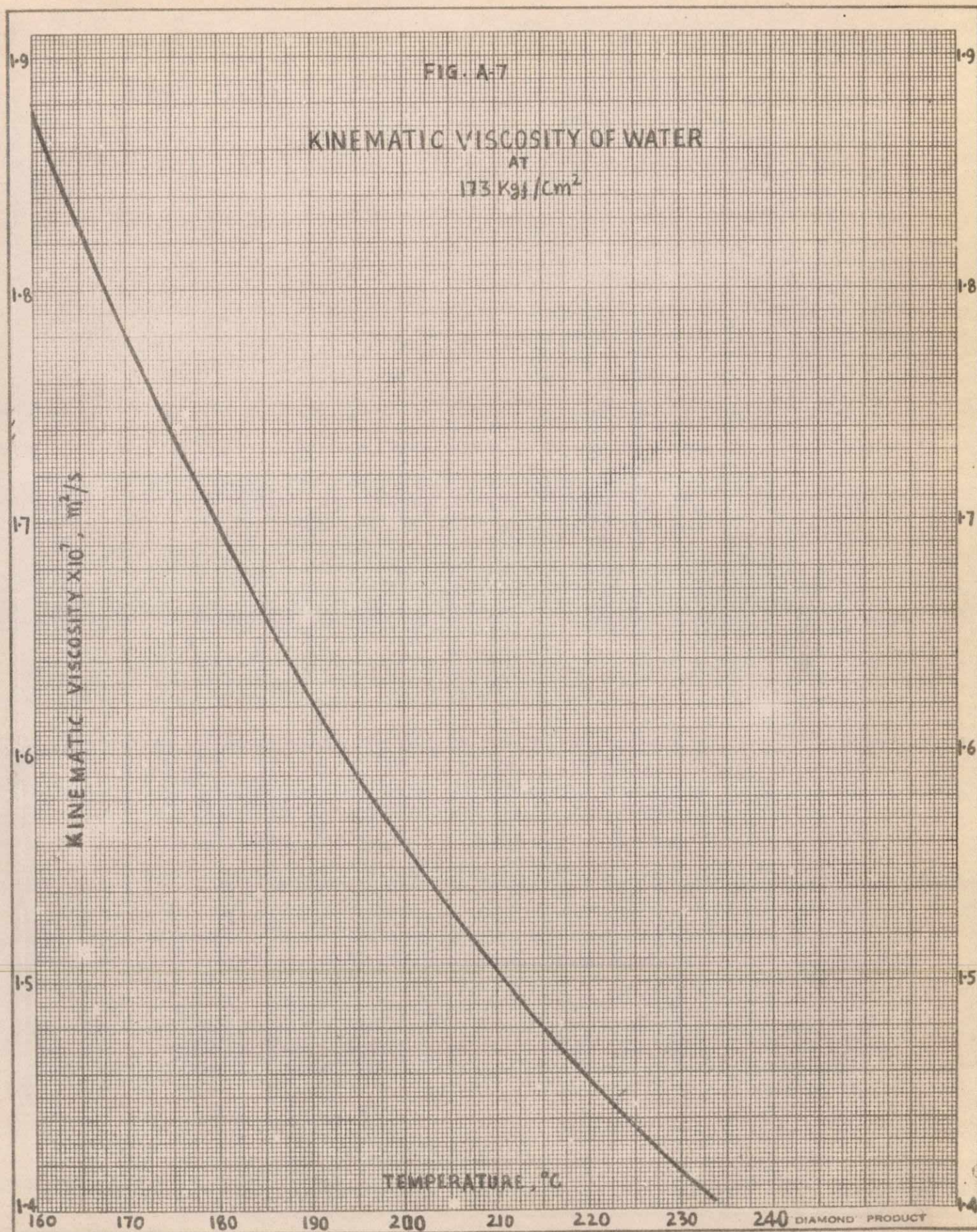


FIG. A-6

PRANDTL NUMBER OF WATER

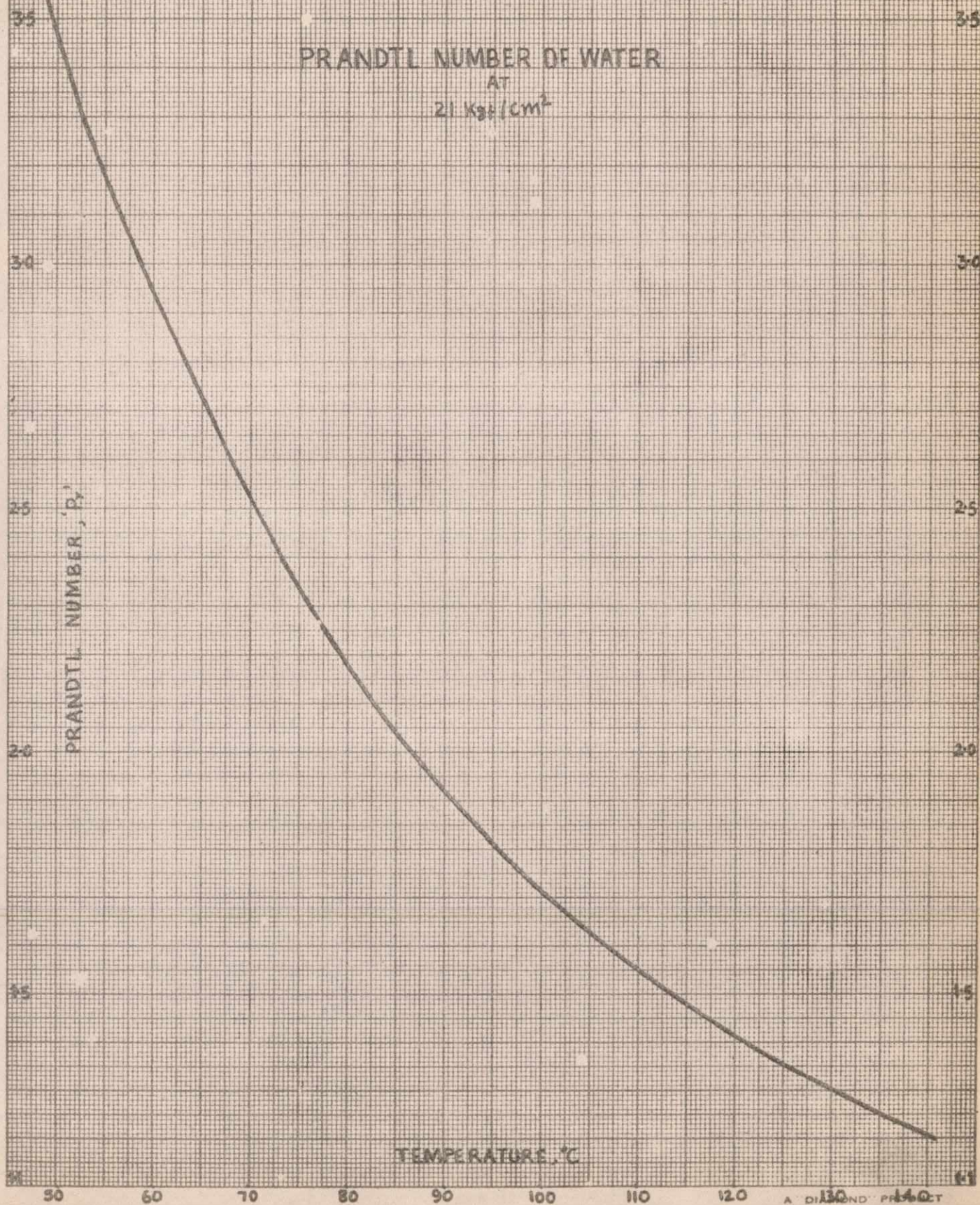
AT
21 Kgt/cm²

FIG. A-9

PRANDTL NUMBER OF WATER
AT
173 Kg/cm²

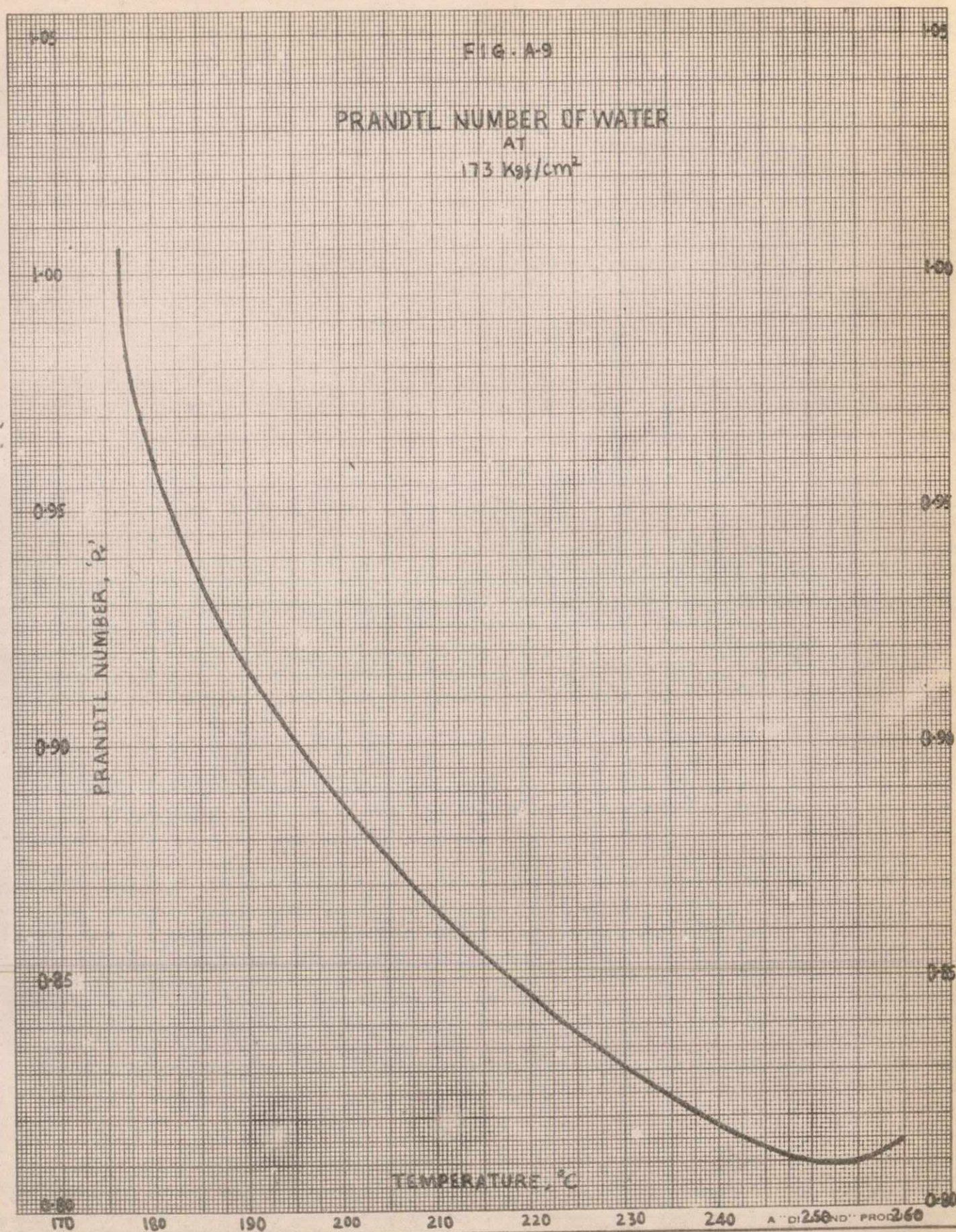


FIG. A-10

THERMAL CONDUCTIVITY
OF
WATER
AT
1, 21, 35 & 173 Kgf/cm²

THERMAL CONDUCTIVITY, Kcal/hr m °C

0.62
0.61
0.60
0.59
0.58
0.57
0.56
0.55
0.54
0.53
0.52
0.51
0.50

0.62
0.61
0.60
0.59
0.58
0.57
0.56
0.55
0.54
0.53
0.52
0.51
0.50

TEMPERATURE, °C

40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210 220 230 240

1 Kgf/cm²

21 Kgf/cm²

35 Kgf/cm²

173 Kgf/cm²

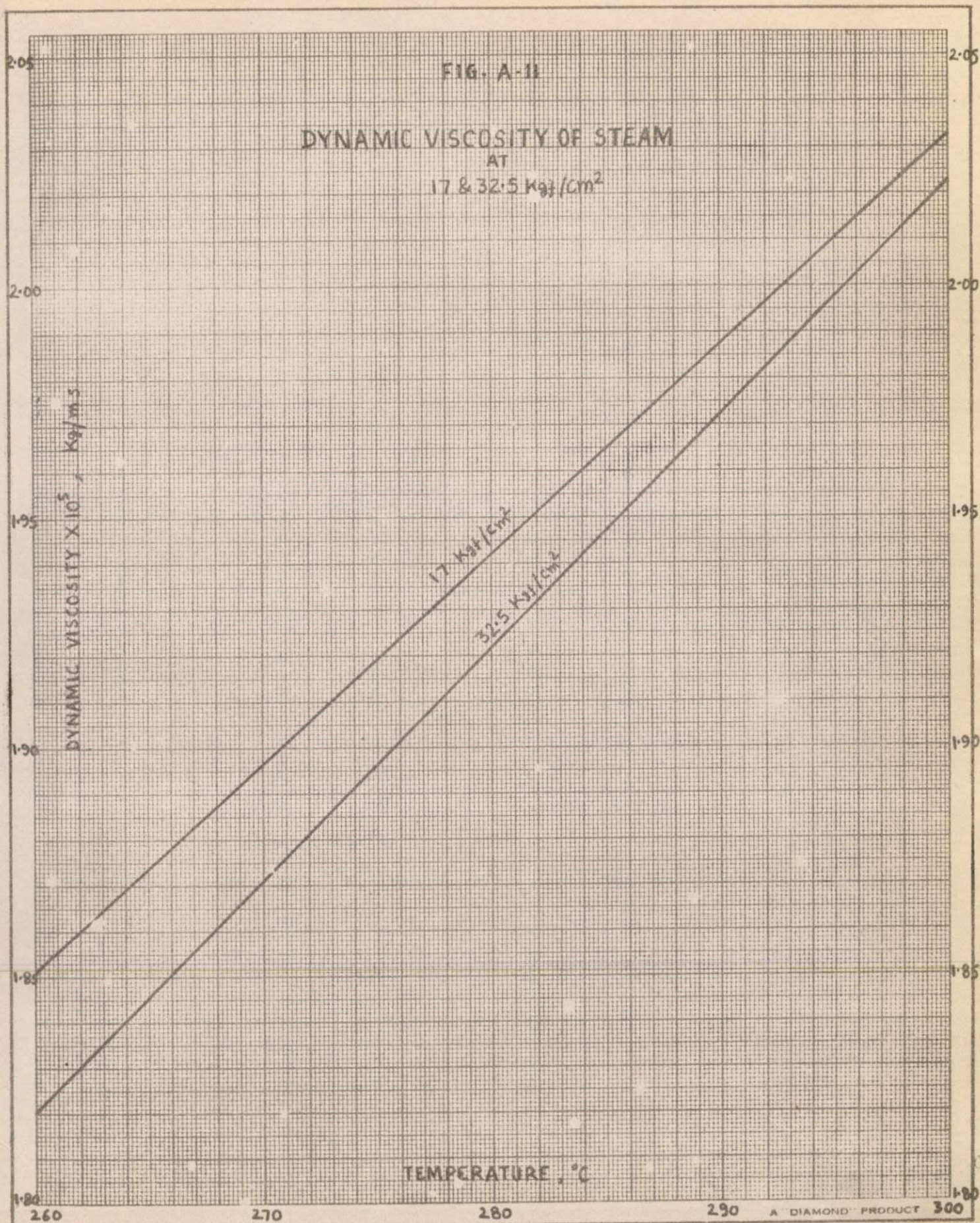


FIG. A-12

PRANDTL NUMBER OF STEAM

AT

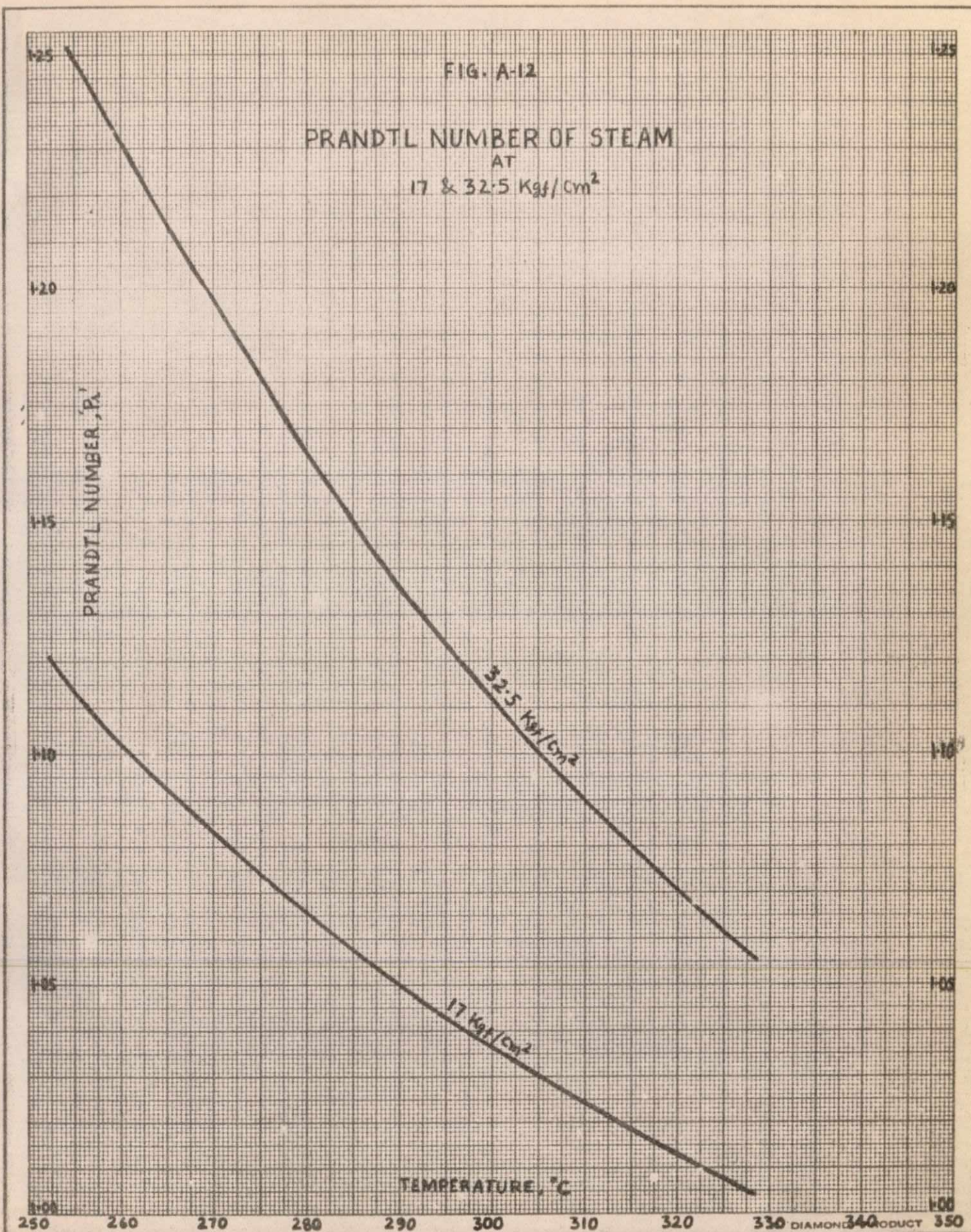
17 & 32.5 Kgf/cm²

FIG. A-13

THERMAL CONDUCTIVITY
OF
STEAM
AT
17 & 32.5 Kgf/cm²

THERMAL CONDUCTIVITY, Kcal/hr m °C

TEMPERATURE, °C

32.5 Kgf/cm²17 Kgf/cm²

0.047

0.047

0.046

0.046

0.045

0.045

0.044

0.044

0.043

0.043

0.042

0.042

0.041

0.041

0.040

0.040

0.039

0.039

0.038

0.038

0.037

0.037

0.036

0.036

0.035

0.035

250

260

270

280

290

300

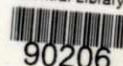
310

320

DIAMOND PRODUCT

330

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