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CONDENSATION OF STEAM ON A

VERTICALLY ROTATING CYLINDER

BY

MILAN GACESA

B. A. Sc., University of Windsor, 1965

A Thesis
Submitted to the Faculty of Graduate Studies through the
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ABSTRACT

In this experimental investigation, the effects of the speed of rotation of the condenser surface and the overall temperature difference between the coolant and the vapour (ΔT) on the transfer of heat from the vapour to the cooled surface, were studied. It was found that the heat transfer coefficient (h_m) was not affected at low rotational speeds ($Fr < 5$). For $Fr > 5$, the heat transfer coefficient increased by 400% from its stationary value for $\Delta T = 180°F$ and by 280% for $\Delta T = 110°F$. Also at $\Delta T = 110°F$, the heat transfer coefficient experienced a decline for $Fr > 68$. Two distinct regimes were observed in the relationship between Nusselt number and Weber number. For $We < 500$, it was found that Nusselt number was a constant whereas for $We > 500$, the Nusselt number increased.

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A	Condenser surface area (ft^2)
c_p	Specific heat of condensate (Btu/lb.°F)
D_o, D_i	External and internal diameters of the condenser tube (ft.)
d	Distance from root of test section (in.)
F	Dimensionless factor to allow for variation in tube temperature
Fr	Froude number ($R_o w^2/g$)
g	Gravitational constant (ft/hr^2)
G	Condensate mass velocity ($= \dfrac{60\ m_w}{\pi D_o}$) (lb/hr. ft.)
h_x, h_m	Local and mean heat transfer coefficients (Btu/hr. ft^2°F)
h_{fg}	Enthalpy of evaporation (Btu/lb)
k_f	Thermal conductivity of condensate (Btu/hr. ft.°F)
k_A	Thermal conductivity of water at atmospheric conditions (Btu/hr. ft.°F)
m_w	Cooling water flow rate (lb/min.)
m_v	Steam flow rate (ozs/min.)
Nu_x, Nu, $<Nu>$	Local, mean and average value of Nusselt number ($h D_o/k_f$)
Nu_A	Nusselt number ($h_m D_o/2k_A$)
Pr	Prandtl number ($c_p \mu_f/k_f$)
q_w	Heat flux based on cooling water measurements (Btu/hr.)
R_o	External tube radius (ft.)
Re	Reynolds number ($4G/\mu_f$)
Re_v	Reynolds number ($V_m D_o/\nu_f$)
T_f	Mean film temperature ($= T_{ob} + T_s)/2$ (°F)

T_s	Steam saturation temperature (°F)
$T_{o1}, T_{o2}, \ldots T_{o10}$	Condenser surface temperature at locations, 1/2, 1-1/2, 2-1/2 ... 9-1/2 inches from the bottom of the test section, respectively (°F)
T_{oa}	Average condenser surface temperature, $(T_{o1} + T_{o2} + \ldots T_{o10})/10$ (°F)
ΔT_s	Temperature difference between steam temperature T_s and average condenser surface temperature T_{oa} (°F)
T_{ob}, T_{ot}	Condenser surface temperature at top and bottom (°F)
ΔT_{ob}	Temperature difference at the bottom of the condenser $(T_s - T_{ob})$ °F
ΔT_{ot}	Temperature difference at the top of the condenser $(T_s - T_{ot})$ °F
$T_1, T_2 \ldots T_{11}$	Cooling water temperature at locations, 0, 1, 2, ... 10 inches from the bottom of the test section, respectively (°F)
T_w	Average cooling water temperature, $(T_1 + T_2 + \ldots + T_{11})/11$ (°F)
ΔT	Overall temperature difference $(T_s - T_w)$ °F
ΔT_w	Cooling water overall temperature rise $(T_{11} - T_1)$ °F
V_m	Mean film velocity (wR_o) ft/hr.
W	Angular velocity (RPM)
w	Angular velocity (Hr.$^{-1}$)
We	Weber number $(\dfrac{\rho_f w^2 D_o^3}{4\,\sigma_g})$
μ_f	Dynamic coefficient of viscosity (lb/ft. hr.)
ν_f	Kinematic coefficient of viscosity (ft^2/hr.)
ρ_f	Specific weight of condensate (lb/ft^3)
σ	Surface tension (lb/ft.)
δ	Film thickness (ft.)
n	Power index

CHAPTER 1

INTRODUCTION

Since the original work by Nusselt in 1916, condensation studies
have been largely confined to the problem of cooled, stationary surfaces
immersed in a vapour. Recently, however, methods of condensate removal,
by means other than gravity alone have been considered. When a vapour
whose condensate has a very high viscosity is to be efficiently condensed
or when a condenser has to function in space where gravity is absent then
not only the practicality but the necessity of such methods become
apparent. Several methods of removal have been suggested:

    1.   Rotating the condenser surface to throw off
        the liquid by centrifugal force

    2.   Imparting a large velocity to the vapour
        tangential to the surface to blow off the
        condensate, and

    3.   Mechanically removing the condensate by
        scraping or other techniques.

In this investigation, the condensate   was   removed by rotating
the condenser surface, thereby effectively replacing gravitational force
by centrifugal force as the primary means of removal.

Three different heat fluxes were achieved by varying the overall
temperature difference between the coolant and the vapour, and by varying
the rate of flow of the coolant. Cooling water at 50°F was circulated
through the condenser tube at 6,600 lb/hr. for one set of tests and at
3,600 lb/hr. for another. The third heat flux condition was achieved
by circulating 120°F cooling water through the condenser tube at
4,800 lb/hr. For all three of the tests the steam conditions were

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constant at 230°F and 21 psia.   For each of the three combinations of

cooling water flow and temperature, the rotational speed of the condenser

tube was varied in random intervals between 0 and 2700 RPM.

CHAPER 2

EXISTING THEORY AND EXPERIMENTAL RELATIONS

Nusselt in 1916 made a theoretical estimate of the heat transfer
for the case of film condensation of a pure saturated vapour on station-
ary surfaces including vertical tubes.   He assumed that when steady
state was reached, the thickness of the film of condensate increased
from the top to the bottom and was maintained by a balance between the
rate of condensation and the rate of draining due to gravity.   He also
assumed that the flow of the condensate layer was laminar and that the
transfer of heat took place solely by conduction across it in a direction
perpendicular to the surface.   The temperature of the surface of the
condensing film in contact with the vapour was assumed to be at the
saturation temperature, and the temperature of the surface of the film
in contact with the wall at wall temperature, which was assumed uniform.
The film of condensate was considered to be so thin that the temperature
gradient through it was a straight line.   For vertical tubes he obtained
the following expression:

$$ hm \left(\frac{\mu_f^2}{k_f^3 \, \rho_f^2 \, g} \right)^{1/3} = 1.47 \left(\frac{4G}{\mu_f} \right)^{-1/3} \tag{1} $$

Many research workers have found that Nusselt's expression agrees fairly
well with their experimental results; although there is a tendency for
the measured results to be higher.   McAdams[8] contends that Nusselt's

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assumption of uniform condenser wall temperature is one of the reasons for this discrepancy. He therefore suggests the following empirical relation:

$$(hm)_{TRUE} = F \ (hm)_{NUSSELT} \tag{2}$$

where F is obtained as shown in Table I.

TABLE I

F	$\Delta T_{ob}/\Delta T_{ot}$
0.96	0.5
1.0	1.0
1.06	2.0
1.15	5.0

Perhaps the more important reason for the discrepancy between the results obtained using Nusselt's expression and the experimental results is Nusselt's assumption that flow is laminar. Many workers[2, 3, 6] have observed that there are in fact waves formed at the condensate film surface. The omission of the effect of these waves in Nusselt's investigation would therefore cause his estimates to be low.

Colburn[1] derived a theoretical relation in the turbulent range, which was of the form:

$$hm \ \left(\frac{\mu_f^2}{k_f^3 \ \rho_f^2 g}\right)^{1/3} = 0.056 \ (Pr)^{1/3} \ (Re)^{0.2} \tag{3}$$

It is generally agreed that equation (3) should be used for values of Reynolds number higher than 2100.

Since the original work by Nusselt, condensation studies have been largely confined to the problem of cooled stationary surfaces immersed in a vapour. Recently, however, methods of condensate removal from the

condensing surface by means other than gravity alone have been considered. One of these is rotation of the condenser surface causing removal of the liquid condensate by centrifugal force and this is the method proposed for the present investigation.

Although no work has been done on vertically rotating cylinders, three studies have been reported on horizontally rotating cylinders. Each of these will be discussed separately and the aspects which are deemed applicable to the present investigation will be stressed.

Yeh[10] in 1953 did the pioneering work. His technique has been criticized by subsequent workers[9] and his results are considered questionable. His work was carried out on a horizontally rotating cylinder cooled on the inside by water and enclosed in a steam chamber. The steam condensed on the outside of the cooled cylinder and the condensate was drained and collected. During the course of his work, Yeh discovered that the flow and heat transfer characteristics of the system went through three different phases. At low rotational speeds $(Fr < 2.6)$, the centrifugal force and the friction force between the shaft and the condensate film tended to counteract the force of gravity and the condensate layer thickened causing a reduction in heat transfer. The second phase occurred at higher rotational speeds $(28 > Fr > 2.6)$ at which time the liquid was sprayed off the cylinder in all directions causing the liquid film to become thinner and thus allow the heat transfer rate to increase. At high rotational speeds $(Fr > 28)$ the film became very thin and droplets of the condensate appeared. Finally it became difficult to find any evidence of a continuous film while the droplets elongated and ultimately became streaks. During this last phase the heat transfer rate became progressively smaller.

Yeh proposed a model for the second phase only and derived the following theoretical relation:

$$hm = \frac{k_f\,\rho_f\,V_m^2}{\sigma g} \qquad (4)$$

Experimentally, he discovered that the data could be correlated in the three regimes thus:

$$I = Const. (Re)^n \qquad (5)$$

where $I = hm/k_f\,\rho_f\,V_m^2/\sigma g$, $Re = \dfrac{V_m\,D_0}{\nu_f}$

In the first regime $n = -1.4$, $73.76.10^4 < c < 102.7.10^4$.

In the second regime $n = -2.34$, $36.42.10^8 < c < 87.16.10^8$.

In the third regime $n = 3.34$, $C = 0.1956.10^{16}$.

This is a rather poor correlation since the values of the constant C vary not only for different Reynolds numbers but also for different pressures at the same Reynolds number. Moreover, the value of n also varies with Reynolds number as shown above.

Singer and Preckshot[9] carried out heat transfer measurements on apparatus similar to Yeh's with some modifications which probably made their results more reliable. An interesting point in their experimental technique was the fact that they did not measure the temperature on the rotating shaft, thereby obviating any problems which might have been encountered in using slip rings. They were able to measure the overall temperature drop between the cooling water and the steam atmosphere by means of stationary probes. Knowing the heat flux, from water flow and

temperature rise measurements, they were able to calculate the overall heat transfer coefficient. They determined the water side heat transfer coefficient from the results of Kuo et al[12]. Knowing the overall and the water side coefficients, they were able to calculate the steam side coefficient.

They also reported the presence of the three regimes described by Yeh. They, however, also presented physical models and theoretical estimates of the heat transfer for two of the regimes. In the first regime, they were able to show that the condensate film thickness increased with rotational speed and hence that the heat flux decreased. Their theoretical estimate of heat transfer in this regime was of the form:

$$Nu_x = \frac{h_x \, D_o}{k_f} = \left(\frac{2}{3}\right)^{1/4} \, \frac{g \, D_o^3 \, h_{fg} \, (\rho_f - \rho_v)}{v_f \, k_f \, (T_s - T_o)} \, \left(1 + \frac{3 \, C_p \, \Delta T}{8 h_{fg}}\right)^{1/4} (\delta^+)^{-1/2} \quad (6)$$

where $\delta^+ = f(\delta)$ and can be evaluated.

The average value of Nusselt number is then:

$$< Nu > = \frac{1}{2\pi} \int_o^{2\pi} h(\emptyset) \, D_o/k_f) d \, \emptyset \quad (7a)$$

and the mean value is:

$$Nu = \frac{1}{L} \int_o^L < Nu > dL \quad (7b)$$

Values of Nusselt number as obtained from equation (7b) were found to be 13.9% below the experimental values in the same regime. For moderate rotational speeds the theoretical estimate of heat transfer was of the form:

$$Nu_x = \frac{h_x \, D_o}{k_f} = 2 \left[(1 + \frac{3}{2We})^{1/3} \right]^{-1} \quad (8)$$

For very high rotational speeds the authors suggest that Nusselt number is a function of $(We)^{1/2}$ rather than We^1 as is the case for lower speeds, indicating a levelling off of the heat transfer.

Presenting the experimental data in the form suggested by equation (8) (i.e., plotting Nusselt number versus Weber number) proved to be unsatisfactory since the experimental data not only disagreed with the theoretical estimate but also there appeared to be a definite family of curves for varying values of overall temperature difference (ΔT). For this reason the authors proposed an empirical correlation of the form:

$$Nu \left(g \, D_o^3 \, \rho_f \, h_{fg}/\nu_f \, k_f \, \Delta T\right)^{-1/4} = Constant \, . \, We^n \qquad (9)$$

As justification for the use of the modified Nusselt number, they cite the fact that for a stationary condenser the following relation holds true:

$$Nu = C \left(g \, D_o^3 \, \rho_f \, h_{fg}/\nu_f k_f \, \Delta T\right)^{+1/4} , \qquad (10)$$

where C is a constant. Their data when plotted according to equation (9) yielded:

$$C = 0.01370, \quad n = 0.735 \text{ for } 250 < We < 900$$

and $$C = 27.206, \quad n = 0.385 \text{ for } 900 < We < 17,000$$

Hoyle and Matthews[5] investigated the effect of diameter size as well as the speed of rotation on the transfer of heat from steam to horizontally mounted, water cooled cylinders. Their apparatus was similar to that used by Singer and Prechshot except that provisions had to be made for using cylinders of different diameters. They also had

to make provisions for measuring the surface temperature on a rotating cylinder. The cylinders used in their study were of 4, 8 and 10" outside diameters. Based on photographic studies, they contended that the condensate layer was in laminar flow throughout all the tests. Based on these observations, they made a theoretical estimate of the heat transfer and obtained the following relationship:

$$\frac{h_m D_O}{2\ k_A} = 0.75 \left[D_O \middle/ (\rho_f / \sigma) \right]^{-0.19} (1.9 - 0.9/1.095^{D_O w^2/2g})$$

$$(k_f^3\ g\ \rho_f^2\ c_p\ D_O^3 / \mu_f\ k_A^4)^{1/4}\ (h_{fg}/c_p\ \Delta T_s)^{1/4} \qquad (11)$$

The experimental results for the 4, 8 and 10" cylinders varied from equation (11) by 15.5, 12.9 and 16.8 percent, respectively.

In order to compare their results with those of Yeh[10] and Singer and Preckshot[9] they expressed them in terms of the variation of Nusselt number $(h\ D_O/2\ k_A)$ with the Weber number $(D_O\ \rho_f/\sigma)^2\ (D_O w^2/29)/2$. Hoyle and Matthews' results formed series of curves corresponding to different cylinders such that Nusselt number varied directly with Weber number for all values of Weber number. This of course is unlike the results of Yeh where the Nusselt number experienced a decline for Weber number greater than 1700 or Singer and Preckshot who reported a decline in Nusselt number for values of Weber number greater than 900.

APPARATUS AND INSTRUMENTATION

The apparatus and instruments required to conduct the experiments will be described under eight separate headings: Steam Supply; Cooling Water Supply and Measurement; Driving Mechanism; Steam Chamber; Condensate Collecting and Weighing; The Condenser Tube; Slip-ring Assembly; Thermocouple Circuit and Thermocouple Output Measuring and Recording Instruments.   Letter reference is to items in Figure 1 unless otherwise stated.

Steam Supply

The steam required for the experiment was generated by a 25 H.P. Napanee Automatic Boiler (A) which delivered 99% dry steam at any pressure between 0 and 125 psig.   The steam from the boiler was conveyed by means of 1-1/4" lagged steel pipes to the steam chamber (E). Before entering the steam chamber it had to pass through a flow control valve and a pressure regulating valve (C) and (D), respectively.   At the low point of the piping a Sarco BMS-4 Steam Trap was installed (B) in order to drain off any condensate that may have formed in the pipes up to that point and allow condensate-free steam to enter the steam chamber.

Cooling Water Supply and Measurement

Cooling water was obtained from the building mains.   Cold water and hot water were piped to a 3-way mixing valve (F) which could supply

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50 gal/min. of water at any temperature between 130°F and 50°F which were

the hot and cold water temperatures, respectively. From the mixing

valve water entered a receiving tank (G) where it was mixed further by

means of baffles. The receiving tank also served as the anchoring

base for the packing housing (K), a bearing (L) and the housing for

the cold water temperature probe (H). From the receiving tank water

entered the condenser tube (M) from which it was ejected through three

radial holes into the emptying tank (N). This tank also served as

a recepticle for the packing housing (P). By means of the probe (H)

which housed a copper-constantan thermocouple wire, the temperature of

the cooling water at any position inside the condenser tube (M) could

be measured. The condenser tube and the probe are also shown in more

detail in Figure 3. From the emptying tank (N) water was piped to

the weighing mechanism (Q). As can be seen from the figure it was

possible to discharge the water directly into the drain or to collect

and weigh it and then discharge it. The weigh tank had a capacity

of 45 gallons and the Toledo Scale used, Model 2181, had a capacity of

725 lbs. and was accurate to within 0.10 lbs.

Driving Mechanism

The condenser tube was rotated by means of an 1/4 H.P., D. C.

Motor. The power from the motor to the tube was transmitted by means

of pulleys and a belt. The motor torque and speed were controlled by

an S.C.R. Dodge Control Unit. The power for the control unit was

obtained from an ordinary 110 volt, 15 amp. wall outlet. The rated

maximum speed of the motor was 1725 R.P.M. but a speed variation on

the condenser tube from 0 - 2700 R.P.M. was achieved by reducing the pulley size from the motor to the tube.

Steam Chamber

The position of the steam chamber with respect to the other equipment is shown in Figure 1, Item (D). In Figure 2, the chamber is shown in more detail. All the important dimensions are given in Figure 2 and will not be repeated here.

As can be seen from Figure 2, the chamber consisted of two concentric boxes. Each box was made in two halves and put together by means of vertical flanges with the inner box secured to the outer. Steam entered the outer box through the opening in the back and was deflected by a baffle so that it diffused evenly through all four sides of the inner box. Slots in the side walls of the inner box have a combined area more than twice the surface area of the condenser tube to ensure minimum steam velocity as it diffused towards the condenser. Louvres covering the slots were installed to prevent droplets of condensate from being thrown from the tube and out through the slots. Steam that condensed on the walls of the outer box was drained off through the secondary condensate outlet (See Figure 1) and discarded, and the condensate which formed on the condenser tube was collected and weighed at the primary condensate outlet.

Where the condenser tube emerged from the outer box, top and bottom, there was a sealed bearing fitted over the tube and attached to the box, which served both as a bearing and a seal. The outer box was also equipped with a pressure gauge and a steam temperature probe (see V, and T in Figure 1).

Condensate Collecting and Weighing

A high pressure boiler glass tube was installed in the primary condensate outlet line and by means of the two valves (See Figure 1), the condensate level inside the glass tube was held constant when it was being collected for weighing. The condensate was collected in a covered beaker and weighed on a small Central Scientific Company scale which was accurate to within 0.01 oz.

Condenser Tube

The condenser tube can be seen in Figure 2 and also in more detail, in some respects, in Figure 5 (see also Appendix C). The tube was made from ASE60-61 aluminum tube 1" O.D. x $1/4$" wall x 4' long. Three grooves $1/8$" x $1/8$" were milled on the outside of the tube at 120° intervals, running the whole length of the tube except for 5" at one end. Starting at the bottom of the groove and moving up (1" at a time), 10 circumferential slots were ground, perpendicular to each groove, approximately 1/16" wide and such that the bottom of the slot was 1/8" below the surface (i.e., same depth as the grooves) at the end which intersected the groove and then the bottom slanted up so that it was at the outer surface of the tube when the slot was 1/4" long, as shown in Figure 3-1.

FIG. 3-1: GROOVE AND SLOT DETAIL

Circumferential Hole
Slot
Groove
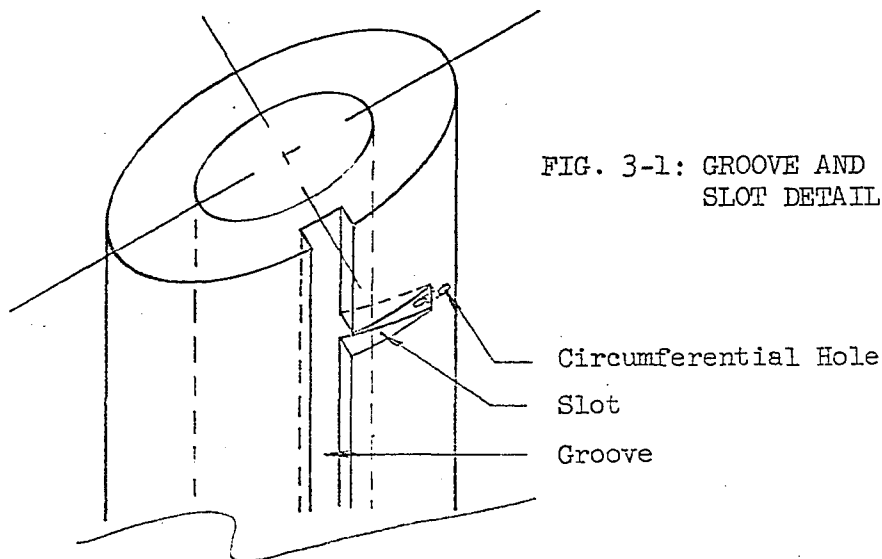
Holes .040" in diameter were drilled at the ends of the slots starting

in the slanted bottom of the slots, far enough from the shallow end so

that the drill could be started without splitting the surface. Holes

were drilled in the same circumferential direction as the slots and were

terminated when the drill just emerged from the surface 3/32" beyond the

end of the slot. The welded junction of each copper-constantan thermo-

couple was inserted inside a circumferential hole (at the end of each

slot). The wires were then pressed inside the slots and laid along the

bottom of the grooves until they emerged at the end of the tube. The

wires were packed (10 in each groove) so that they filled approximately

2/3 of the groove depth. The remainder of the groove was filled with

Devcon Aluminum Paste which consists of 80% aluminum powder and 20%

plastic filler. When the aluminum filler dried, portions of it that

protruded above the surface of the tube were machined off so that the

grooves again assumed a texture very nearly the same as the remainder

of the tube. Two discs (See Figure 2) made of aluminum were next

fitted to the tube 10" apart at the section of the tube where the

thermocouple junctions were. These discs effectively defined the test

section and also prevented leakage of unwanted condensate into or out

of the inner box. The discs were insulated from the tube by means of

Teflon sleeves. The tube was then installed into the steam chamber as

shown in Figure 2.

<div align="center">Slip Ring Assembly</div>

In order to convey the thermocouple emf. signal from the shaft when

the latter was rotating to the stationary measuring and recording

instruments, it was necessary to employ slip rings. Based on reported

difficulties of other researchers[7, 10, 21] who attempted to make their own slip-ring assemblies, it was decided to buy a ready-made slip-ring unit for the present investigation. A 10 ring unit, manufactured by Michigan Scientific Company, was purchased. This unit is shown installed in Figure 1.

<div align="center">Thermocouple Circuits and the</div>

<div align="center">Measuring & Recording Instruments</div>

The thermocouple circuit for the wires attached to the condenser tube was originally made up as shown in Figure 4(a). Originally, it had been planned that all of the thermocouples attached to the condenser tube (30 of them) would be connected to the measuring instruments. However, the cost of a slip-ring unit having thirty-one (31) rings was prohibitive and hence a ten ring unit was purchased which made it possible to monitor the signals from only nine of the thirty thermo-couples. All the copper leads (9 of them) were soldered together and then attached to a single lead on the ring side of the slip-ring unit. The nine constantan leads were attached singly to the remaining leads on the ring side of the slip-ring unit. On the brush side of the slip-ring unit, a copper lead was attached to the brush corresponding to the ring to which all the copper leads were attached; and a constantan lead was attached, one to each brush, such that the brushes mated with the rings which had constantan wires attached to them. The copper lead and the constantan leads were then attached to a Thermovolt Instrument Company multiple switch. From the output terminals of the switch, a copper lead was attached to the common copper side, and a constantan lead was attached to the other terminal. The other end of the constantan

lead was joined to a copper lead to comprise the reference junction (See Figure 4 (a)). The two free copper leads were then connected across a potentiometer and also across a recorder. The potentiometer used was a Honeywell Potentiometer Model No. 2745. The continuous pen recorder Model 7100B was manufactured by Hewelett Packard.

It was suggested that a different circuit should be used to obviate the possibility of an error arising from, e_1 and $-e_1$, and, e_2 and $-e_2$, (See Figure 4a) not cancelling. Several of the thermocouple circuits were then changed to the configuration shown in Figure 4 (b). The major change was the elimination of the Emf's e_2 and $-e_2$. However, it was still necessary to compensate for Emf. e_1 and therefore the compensating circuit shown in Figure 4 (b) was included. There was no significant difference in the potentiometer (and/or recorder) reading when the two circuits were interchanged for any one thermocouple. It was felt, nevertheless, that the circuit shown in Figure 4(b) was more reliable and this one was used.

All other thermocouples which measured temperatures on stationary members were connected to the measuring instruments using a circuit identical to that in Figure 4 (a), when the latter has the slip-ring portion of the circuit removed.

CHAPTER 4

EXPERIMENTAL PROCEDURE

The principal objective of this investigation was to study the effect of the rotation of a condenser tube on the outside or steam side heat transfer coefficient. The effect of the overall temperature difference $(T_S - T_W)$ was also investigated.

Before the apparatus was assembled, the outside surface of the test section was thoroughly cleaned using steel wool; it was then polished using abrasive powder and finally washed with alcohol. This was done in order to ensure complete wetting of the condensing surface. Then the steam chamber was closed. At the beginning of each experiment the equipment was allowed to operate with the steam and cooling water turned on for at least 30 minutes before the first readings were recorded. Also, when the steam first entered the chamber, a valve at the high point of the chamber was opened to allow the removal of air from the chamber. The steam pressure at the boiler was adjusted so that when this source steam was throttled to test conditions of 6.3 psig it was also super-heated by 2 F°. At the start of each experiment the barometric pressure was recorded.

The first run was repeated after 30 minutes to ensure that steady state was achieved, and then the speed of rotation was changed (in every instance the speed of rotation was increased, in random intervals from zero RPM to 2700 RPM). After each speed change the equipment was allowed to operate for 30 minutes before the next set of readings was recorded.

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For each run two sets of readings were recorded and averaged. The
parameters metered were:

1. Cooling water temperature at 11 points along the length
   of the test section one inch apart, starting at the top.

2. Steam side condenser surface temperature at 10 points
   one inch apart, starting 1/2 inch from the top.

3. Cooling water flow rate.

4. Condensate flow rate.

5. Steam temperature.

6. Steam pressure.

7. Speed of rotation of the condenser tube.

All the temperatures were measured using copper-constantan thermo-
couples. The thermocouple signals from the wires attached to the rotat-
ing shaft were conducted to the stationary instruments by means of slip
rings. These and all other thermocouple signals were either recorded by
a strip chart recorder or measured manually on a potentiometer.

Cooling water temperature was measured by means of a thermocouple
inserted into a 1/16" O.D. x 0.048" I.D. stainless steel tube, (H) in
Figure 1, which was itself inserted inside the condenser tube from the
bottom and secured to the stationary tank, (G) in Figure 1. One inch
from the thermocouple junction, a small propeller was attached to the
stainless steel tube to ensure thorough mixing of the cooling water.
The thermocouple then measured the bulk temperature of the water.
During the waiting period required for steady state to occur, the probe
was pushed in so that the propeller was well downstream of the test
section. In order to leave the flow, upstream of the point where measure-
ments were being recorded, unaffected, the first reading taken was at a

test section location farthest downstream. The probe was then moved upstream one inch at a time until the whole length of the test section was covered.

The temperature of the cooling water was measured to within $\pm$ 0.10 F°. Eleven temperature readings were plotted against their appropriate location on the shaft and a line drawn through them in order to establish a continuous curve. Average temperature rise of the cooling water between stations 1 and 11 was 4.5 F°. Since the temperature is measured to within $\pm$ 0.10 F°, the resulting error in T_w is 2.0%.

Cooling water and condensate flow rates were measured by collecting and weighing the quantities involved.

The weigh scale used for cooling water measurements was accurate to within $\pm$ 0.10 lb. The least weight of water collected was 60 lb/min. Hence, the maximum error resulting from this measurement was 0.2%. Then, the maximum error which could arise in evaluating heat flux, and hence the heat transfer coefficient, by multiplying the cooling water mass flow rate and temperature rise would be 2.2%.

The heat transferred to the cooling water was compared to the heat required to condense the condensate collected, and it was found that the difference between the two was never greater than 5%.

Steam pressure was measured by means of a pressure gauge which was accurate to within 1 oz/in^2, between zero and 10 psig.

The rotational speed of the condenser tube was measured using a tachometer and a stroboscope.

Seventy-six (76) experiments were performed, varying the rotational speed from zero to 2700 RPM and the cooling water temperature and flow from 45°F to 120°F and 3600 lb/hr. to 7200 lb/hr. respectively.

CHAPTER 5

DISCUSSION OF RESULTS

Some of the experimental data for the test series, I, II and III, is shown in its elemental form in Figures 5, 6 and 7, respectively. No temperature readings were recorded for location d = 6.5 because the thermocouple at that location could not be accommodated on the slip-ring. The readings from the thermocouple located at d = 7.5 were not recorded because they were obviously in error. All three of the figures (5, 6, 7) exhibit the same trend. As the speed of rotation was increased, the surface temperature of the condenser tube also increased. Also evident in all three of the figures is the temperature drop from the centre to the ends of the test section. This was caused by end heat losses. From Figure 5 it can be seen that the temperature difference ΔT varies from 70 F° at zero RPM to 30 F° at 2610 RPM. This of course implies that, over the same speed range, the heat transfer coefficient should increase by more than 100%, if the heat flux is held constant. In fact, the heat transfer coefficient increased by more than 200% (See Figure 12) since the heat flux was also increasing with rotational speed. Since the heat flux was not constant, then the condenser surface temperature, being a dependent parameter, is not a true indication of the change in heat transfer coefficient. Nevertheless, it is obviously a good qualitative indication of the phenomenon and justifies further discussion. For test series II, Figure 6, ΔT dropped from 90 F° to 25 F° as the rotational speed was increased from zero RPM to 2460 RPM, implying an almost 300% increase in h_m, where, in fact, h_m increased by almost 400%. For test series III, Figure 7, ΔT drops from 110 F° to 40 F° implying an almost two fold increase in h_m, where, in fact, h_m increased by almost 400%.

- 20 -

In both the latter cases the reason for the discrepancy is the same as cited for test series I; the fact that T_o is a dependent parameter makes it relatively insensitive to a change in heat flux, since this latter change causes a combined change in both h_m and ΔT.

Also evident in Figures 5, 6 and 7 is a consistent trend for the surface temperature at the top of the test section to change from a value less than the average at zero RPM to a value well above or the same as the average at the same rotational speed above 1000 RPM. This indicates that the film of condensate is thinner at the upper end of the condenser tube. As the speed is increased, the temperature of the remainder of the tube rises to equal that at the topmost thermocouple position. The temperature at the bottom of the test section never reaches the same value as that at the top, except at zero RPM when both positions have a temperature far below the average. Also evident from the three figures (5, 6 and 7) is the fact that the uppermost temperature reading changes very little at high speeds, indicating a levelling-off of the surface temperature. This is more evident in the three figures 8, 9 and 10.

Figures 8, 9 and 10 were drawn assuming a constant condenser surface temperature at any one rotational speed and plotting the actual temperatures against the corresponding speed. By doing this and by drawing a line through the most frequently occurring temperature points, it is implied that the average temperature is the most frequently occurring temperature rather than the conventionally accepted average. It was felt that this was an acceptable mode of drawing the curves since in fact, the majority of the temperatures were almost the same and the ones that

were not (normally the end temperatures) when averaged in with the rest
in the conventional manner resulted in an error in h_m of less than 5%
in the worst case.

Figures 8, 9 and 10 show the data in a much more meaningful form,
in some respects, than Figures 5, 6 and 7 since now all the data are
presented and the overall trend can be followed. Now, however,
(in Figures 8, 9 and 10) the temperature change along the length of
the condenser tube is somewhat camouflaged. Figures 8, 9 and 10
all show basically the same trend in the data: the condenser surface
temperature initially drops as the rotational speed is increased up to
a certain rotational speed and then the temperature suddenly rises.
The only difference between the figures lies in the amount of the tempera-
ture drop and the speed location at which the temperature begins to rise.
The fact that the temperature initially drops indicates that a film
of condensate forms on the condenser surface and becomes thicker as the
speed of rotation increases. When the condenser is rotated, the result-
ing centrifugal force tends to throw the condensate off tangentially.
However, the surface tension force between the adjoining liquid molecules
tends to keep the condensate layer attached to the tube. When this
layer gets large enough, the surface breaks down and the liquid is
thrown off in all directions. Although in fact the surface tension
force varies inversely with the fluid temperature, this is not borne
out very clearly by the experiments. At a surface temperature of
approximately 170°F, the surface temperature declined until a rotational
speed of 850 RPM (Fr = 10.05) was reached, at which time the temperature

suddenly rose. For a condenser tube at 130°F, this occurred at 450 RPM ($Fr = 2.85$). Looking at these results it would seem that the surface tension force is larger at 170°F than at 130°F. However, examining the curves more closely, it becomes apparent that at 130°F the condenser surface temperature drops by more than 20 F° in the interval 0 - 450 RPM, whereas in the case where the condenser surface is 170°F when stationary, this same temperature drops by approximately 8 F° when the rotational speed is 850 RPM. These last observations indicate that the condensate film thickness increases more quickly when the condenser tube is at 130°F than when it is at 170°F. The resulting thicker film is then thrown off at a lower rotational speed than the much thinner film formed on the tube whose surface is at 130°F.

When the condenser surface temperature is at 170°F, the temperature difference between it and the surrounding steam causes a film of condensate to be deposited on the condenser surface. This film then grows to a thickness at which very little additional steam is being condensed. This thickness then persists and grows slightly as the speed of rotation is increased until the centrifugal force overcomes the tension force and the film breaks down. When the condenser surface is at 130°F when stationary, the initial rate of condensation is much greater than when that surface is at 170°F, and the temperature difference between the surface and the steam is now large enough so that condensation persists and an even thicker film is built up which then has to be thrown off at much lower rotational speeds. An intermediate run was performed in which the condenser surface temperature was initially set at 150°F. In this case, the condensate layer thickness, as implied by the surface

temperature, was greater than that for the run where the condenser surface was held at 170°F but less than the condensate thickness in the case where the condenser surface was initially at 130°F. The rotational speed of 550 RPM, at which the surface temperature stopped falling and suddenly rose, was also intermediate between those for the other two runs. The results of the three runs are plotted together in Figure 11. The only curve which exhibits a levelling-off and perhaps even a decline of the surface temperature at high rotational speeds is the one for which the initial surface temperature was 170°F. However, the levelling-off and the decline occur just at the limiting speed and thus make it difficult to make any definite conclusions. The other two curves show no trace of a decline.

Figures 12, 13 and 14 show the effect of rotational speed on the heat transfer coefficient. All three figures imply that there is an initial regime at low rotational speeds in which h_m is affected very little. After this initial regime h_m rises to a maximum of 500% of its initial value. Figure 12 shows the results for test series I ($T_W = 120°F$). In addition to the two regimes described above, h_m in this case also goes into another regime. At approximately 2200 RPM ($Fr = 68$) h_m reaches a peak value and for additional speed increase appears to decrease. However, the results can not be accepted as conclusive without performing additional runs at even higher speeds. The results for test series II ($T_W = 45°F$, $m_W = 3600$ lb/hr.) are shown in Figure 13. The peak value of h_m in this case occurs at approximately 2500 RPM ($Fr = 88$). However, the decline of h_m (if any) is even less

evident in this case than for test series I.   Test series III results are shown in Figure 14.   In this case h_m shows no signs of a decline at a rotational speed of 2700 RPM.   It would obviously be desirable to extend the rotational speed for all three test series.

The best correlation of the results was obtained when the data were plotted as Nusselt number (Nu) against Weber number (We) as in Figure 15.   From Figure 15, it is clearly evident that there exist two distinct regimes.   Up to a value of We = 500, Nusselt number is essentially constant at a value of 220.   For values of Weber number higher than 500, Nusselt number increases.   In this region the relation between Nusselt number and Weber number is of the form:

$$Nu = 11.56 \, We^{0.4886}$$

It is also evident from Figure 15 that Nusselt number does not experience a decline for high values of Weber number.   It is difficult, however, to predict what would have happened had it been possible to go to higher values of rotational speed.

In order to compare the results with those of other workers, they were plotted as shown in Figure 16.   Again, the two regimes mentioned above were evident.   Up to a value of We = 500, Nusselt number (Nu_A) had a constant value of 120.   For We > 500, the relation between Nu_A and We was of the form:

$$Nu_A = 6.13 \, We^{0.4957}$$

The straight line correlation as obtained in Figure 16 was then drawn on a graph showing some of the results of other workers.

There are essentially three major differences between the author's results and those of other workers. Whereas in the present investigation there was a regime of constant Nu_A up to a value of We = 500, none of the other workers observed this. Singer and Yeh reported an initial decline in Nu_A but this reportedly occurred at values of Weber number well below 100. Another difference is a lack of a decline in Nu_A for large values of We (We > 900 for Preckshot and We > 1700 for Yeh). Hoyle's results reported here are his results for a 4").D. condenser which was the closest in size to the condenser used for the present investigation. Moreover, Hoyle's curve was drawn as a straight line for the purpose of comparison where in fact it was not exactly a straight line. The third difference is the obviously large difference in Nu_A between the present results and those of other workers at low rotational speeds. This difference was not totally unexpected since it was known that the heat transfer coefficient is higher for horizontal than for vertical tubes.

Garrett and Wighton [4] and Hassan and Jakob [4] report values of h_m to be almost twice as high for horizontal tubes as for vertical tubes. For this reason, it is expected that at low rotational speeds, the results of the present investigation should be well below those reported by other workers (5, 9 and 10), since all of their results were for horizontal tubes. Moreover, it is observed from Figure 17 that the results of the present investigation do indeed reach the same order of magnitude as those of other workers at high rotational speeds when the effects of gravity become insignificant compared to centrifugal force.

Stationary results for the present investigation varied from + 3% to - 15% from those predicted by Nusselt's theory for stationary vertical tubes (equation 1).

CHAPTER 6

CONCLUSIONS

6.1 It was found that the condenser surface temperature initially dropped while the speed of rotation was increased and then rose suddenly, approaching more and more closely the steam temperature. The initial drop was attributed to a condensate build-up made possible by surface tension forces which prevented the condensate from being thrown off at low Froude numbers. When the surface tension force was overcome, the condensate was sprayed out tangentially in all directions.

6.2 The heat transfer coefficient did not vary from its "stationary" value at low speeds of rotation. However, when the rotational speed was increased, the heat transfer coefficient also increased until it became equal to 500% of its stationary value for $\Delta T = 185°F$. For $\Delta T = 110°F$, the maximum value of h_m was equal to 380% of its stationary value. Also at $\Delta T = 110°F$, there was a hint of a drop of h_m for values of We > 2200 RPM.

6.3 The data were successfully correlated in the form

$$Nu = 11.56 \ We^{0.4886}$$

for values of Weber number We > 500. For values of Weber number < 500, Nusselt number was found to be a constant (220).

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REFERENCES

CITED REFERENCES

1. Colburn, A. P., "The Calculation of Condensation Where a Portion of the Condensate Layer is in Turbulent Motion", Trans. American Inst. Chemical Eng. Vol. 30, 187 (1933).

2. Epsig, H., Hoyle, R., "Waves in a Thin Liquid Layer on a Rotating Disc", J. Fluid Mech. 22, 671 (1965).

3. Garrett, T. W., Wighton, J. L., "The Effect of Inclination on the Heat Transfer Coefficients for Film Condensation of Steam on an Inclined Cylinder", Int. J. Heat Mass Transfer, Vol. 7, 1235 (1964).

4. Hassan, K. E., Jakob, M., "Laminar Film Condensation of Pure Saturated Vapours on Inclined Circular Cylinders", A. S. M. E. Paper No. 57-A-35.

5. Hoyle, R. and Matthews, D. H., "The Effect of Diameter Size and Speed of Rotation on the Heat Transfer from Steam to Cooled Cylinders", Int. J. Heat Mass Transfer, 7, 1223 (1964).

6. Hoyle, R., Matthews, D. H., "The Effect of Speed on the Condensate Layer on a Cold Cylinder Rotating in a Steam Atmosphere", J. Fluid Mech. 22, 105 (1965).

7. Kuo, C. Y., et al, "Heat Transfer in Flow Through Rotating Ducts", Trans. A.S.M.E. Series C, J. Heat Transfer, 1960, 82 (2), 139-151.

17. Nandaperckar, S. S., Beatty, K. O., "Condensation on a Horizontal Rotating Disc", Chemical Engineering Progress Symposium Series No. 30 (Heat Transfer, Storrs, Conn.) pp. 129-137, 1960.

18. Neville, A. M., Kennedy, J. B., Basic Statistical Methods for Engineers and Scientists, International Book Company (1964).

19. Nicol, A. A., "The Effect of Vapour Velocity on the Condensation of Steam", The Canadian Journal of Chemical Engineering, December 1964, pp. 278-279.

20. Nicol, A. A., Medwell, J. O., "The Effect of Surface Roughness on Condensing Steam", The Canadian Journal of Chemical Engineering, June, 1966, pp. 170-173.

21. Pattenden, R. F., "Heat Transfer from a Rotating Tube with Controlled Fluid Flow", Journal Mech. Engineering Science, Vol. 6, (2), 1964.

22. Patton, E. L., Feagan, R. A., "A Method of Installing Tube Wall Thermocouples", Industrial and Engineering Chemistry, Analytical Edition, Vol. 13, 1941, pp. 823-824.

23. Reactor Handbook, Volume I - Materials, Interscience Publishers, Inc., N. Y., (1960).

24. Rohsenow, W. M., "Heat Transfer and Temperature Distribution in Laminar Film Condensation", Trans. A.S.M.E., 1956, 78, 1645-1648.

25. Sparrow, E. M., Gregg, J. L., "Theory of Rotating Condensation",
     Trans. A.S.M.E., Series C, J. Heat Transfer, 1959, 81 (2), 113-120.

26. Sparrow, E. M., Gregg, J. L., "The Effect of Vapour Drag on
     Rotating Condensation", Trans. A.S.M.E. Series C, J. Heat Transfer,
     1960, 82 (1), 71-72.

27. Sparrow, E. M., Hartnett, J. P., "Condensation on a Rotating Cone,
     Trans. A.S.M.E. Series C, J. Heat Transfer, 1961, 83 (1), 101-102.

28. Volyak, L. D., "Temperature Dependence of the Surface Tension
     of Water", Doklady Akad. Nauk. U.S.S.R. 74, 307-310 (1950).

A - Boiler
B - Steam Trap
C - Flow Control Valve
D - Pressure Control Valve
E - Steam Chamber
F - 3 way Mixing Valve
G - Receiving Tank
H - Coolant Temperature Probe
J - D.C. Motor and Control
K - Packing Housing
L - Sealed Bearing
M - Condenser Tube
N - Emptying Tank
P - Packing Housing
Q - Weigh Tank and Scale
R - Drain Hose
S - Slip Ring Unit
T - Thermocouple Probe
U - Thermocouple Leads
V - Pressure Gauge

FIGURE 1:   SCHEMATIC LAYOUT OF EXERIMENTAL RIG   (Not To Scale)
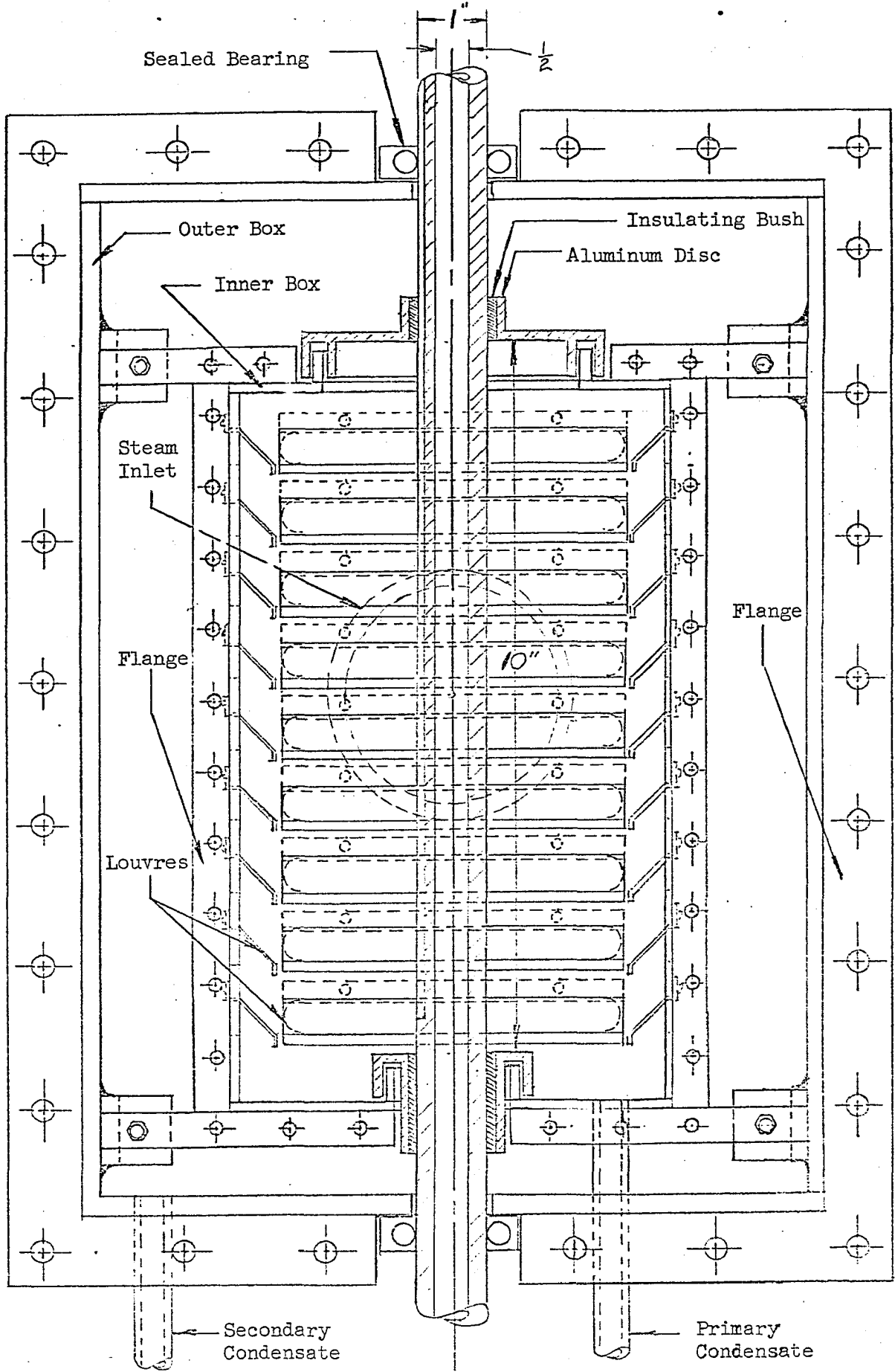
FIGURE 2: DETAIL OF STEAM CHAMBER (Scale 1" = 2")

Section A-A
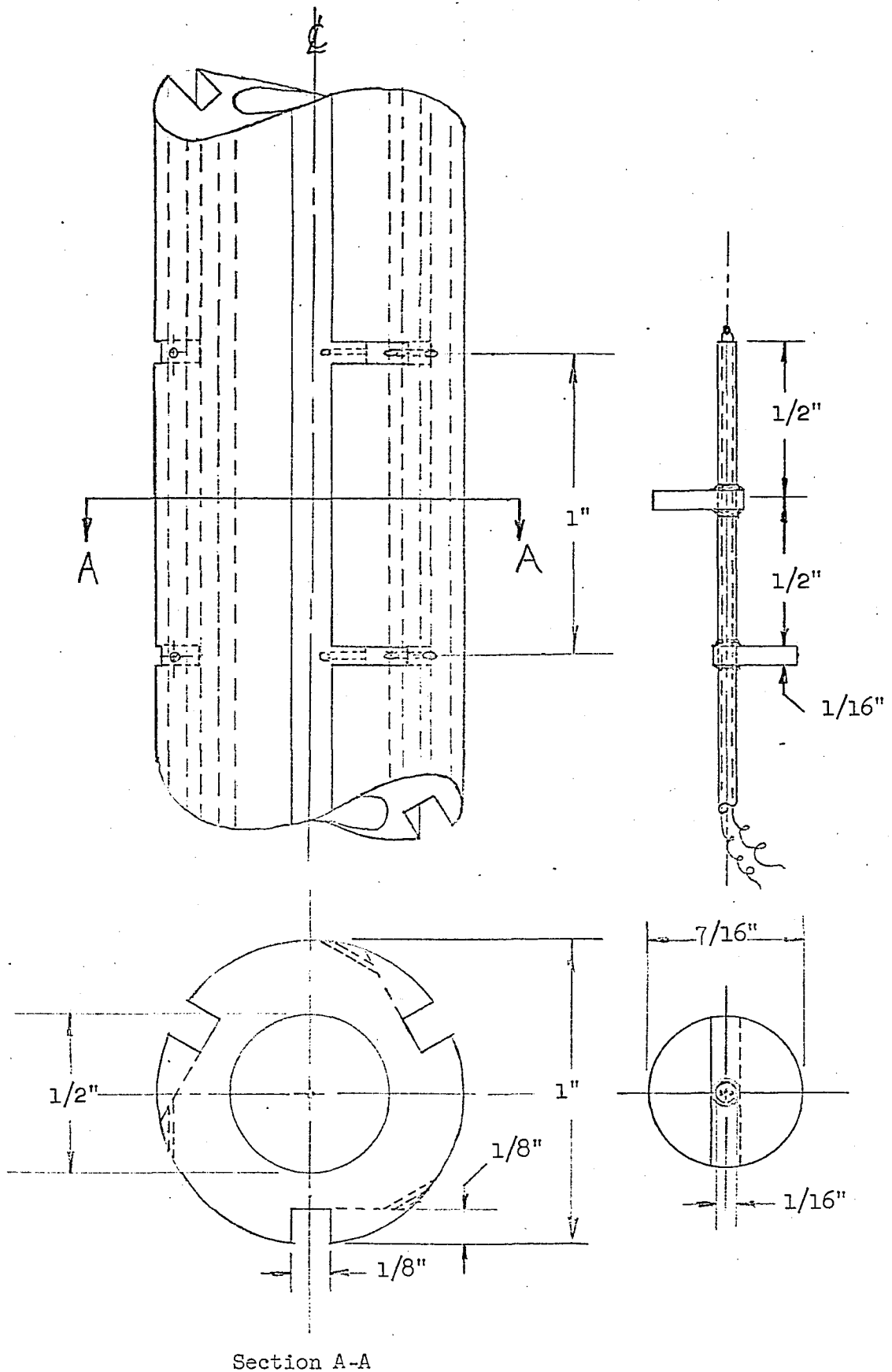
FIGURE 3:   DETAIL OF THERMOCOUPLE GROOVES   (Scale 1" = 1/2")
AND COOLING WATER TEMPERATURE
PROBE
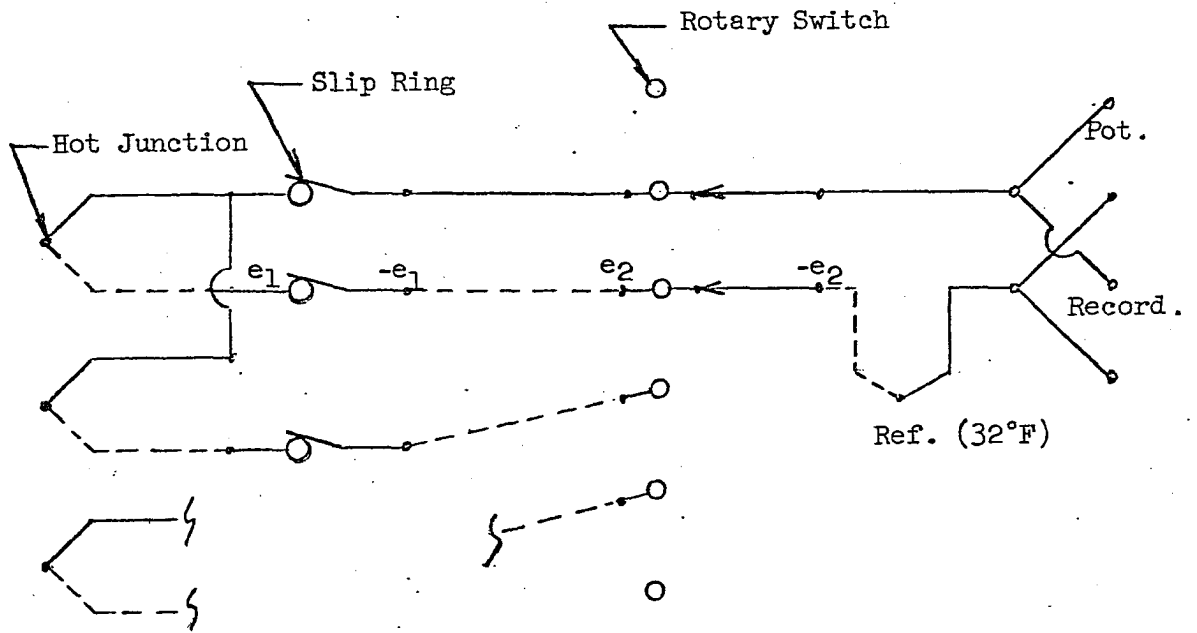
FIGURE 4(a):   THERMOCOUPLE CIRCUIT SCHEMATIC   (Not To Scale)
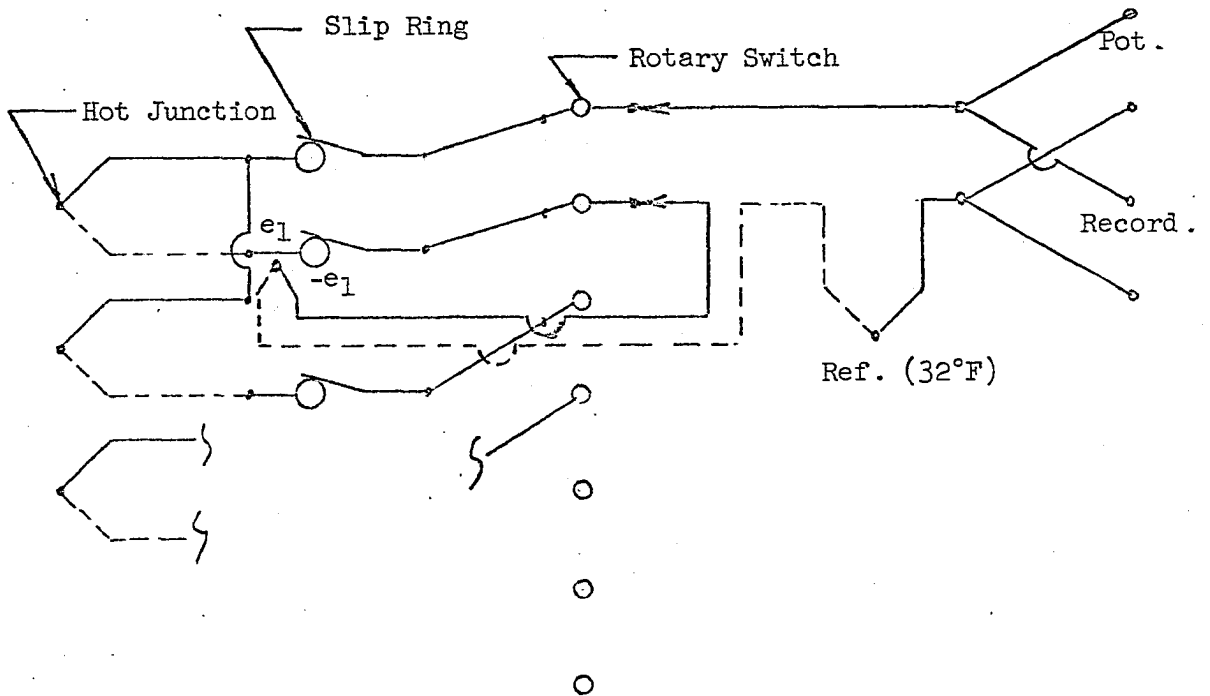
FIGURE 4(b):   ALTERNATIVE THERMOCOUPLE CIRCUIT SCHEMATIC   (Not To Scale)
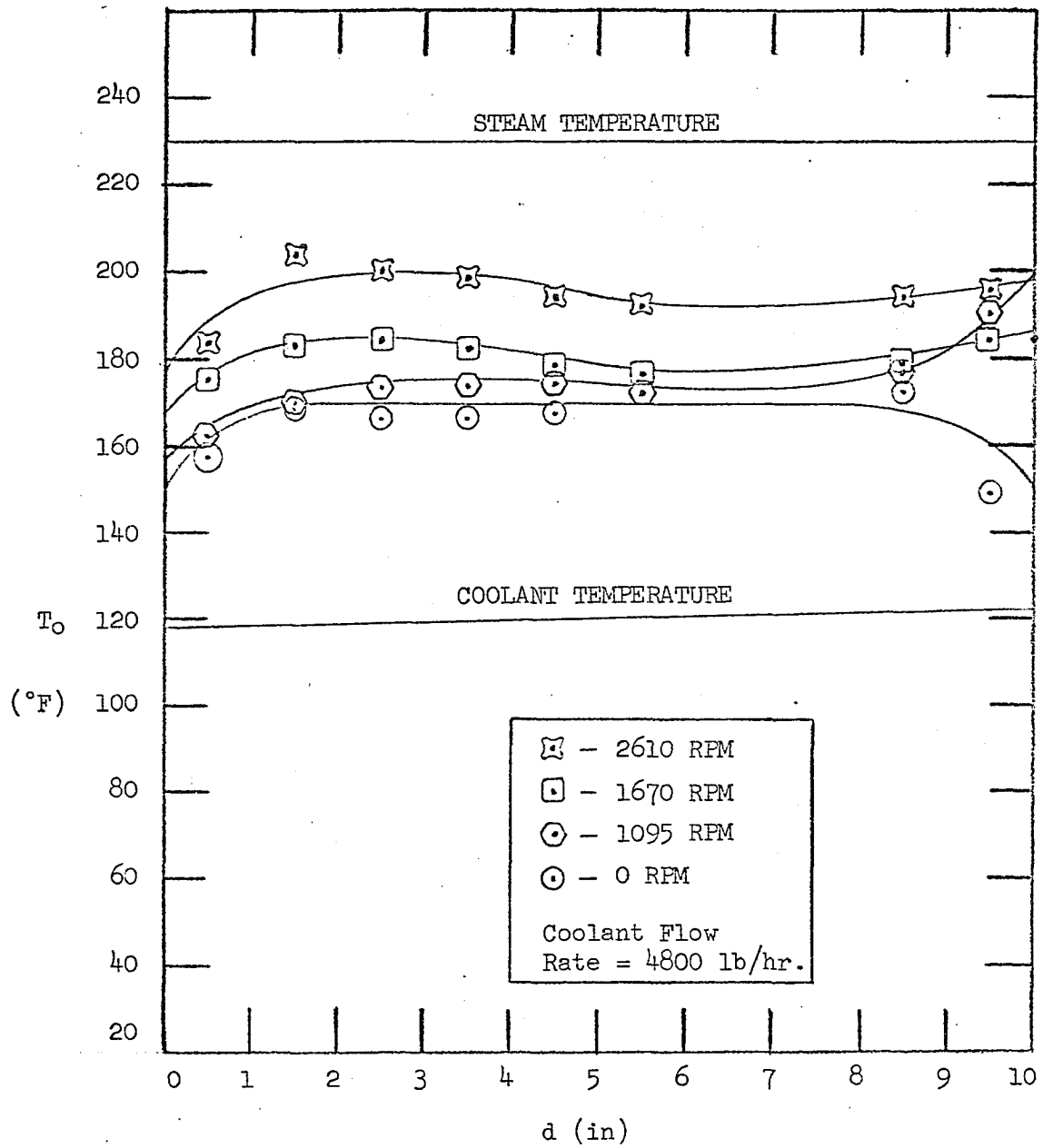
FIGURE 5: VARIATION OF CONDENSER SURFACE TEMPERATURE
WITH DISTANCE FROM ROOT (TEST SERIES I)
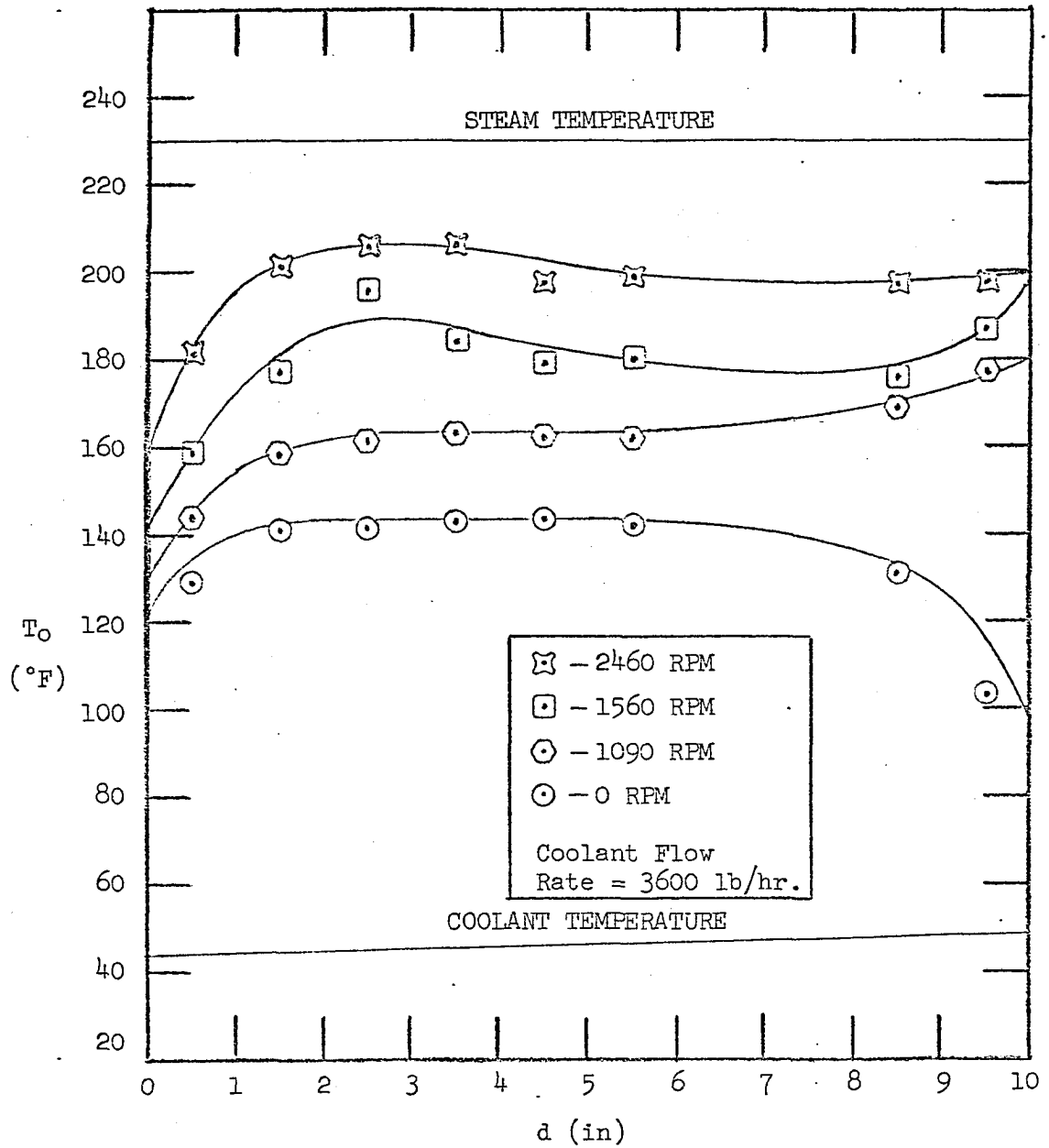
FIGURE 6: VARIATION OF CONDENSER SURFACE TEMPERATURE
WITH DISTANCE FROM ROOT (TEST SERIES II)
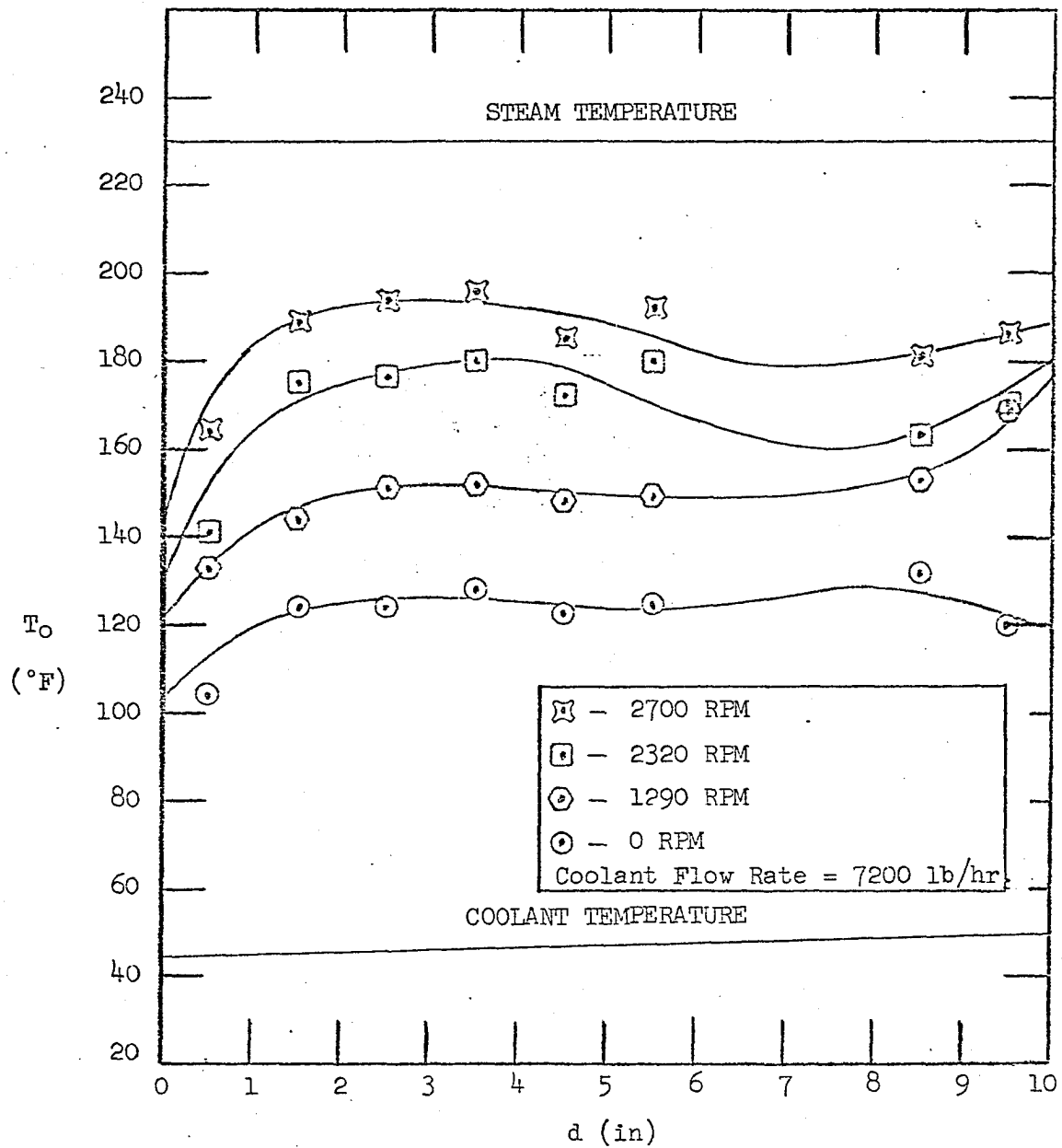
FIGURE 7: VARIATION OF CONDENSER SURFACE TEMPERATURE
WITH DISTANCE FROM ROOT (TEST SERIES III)
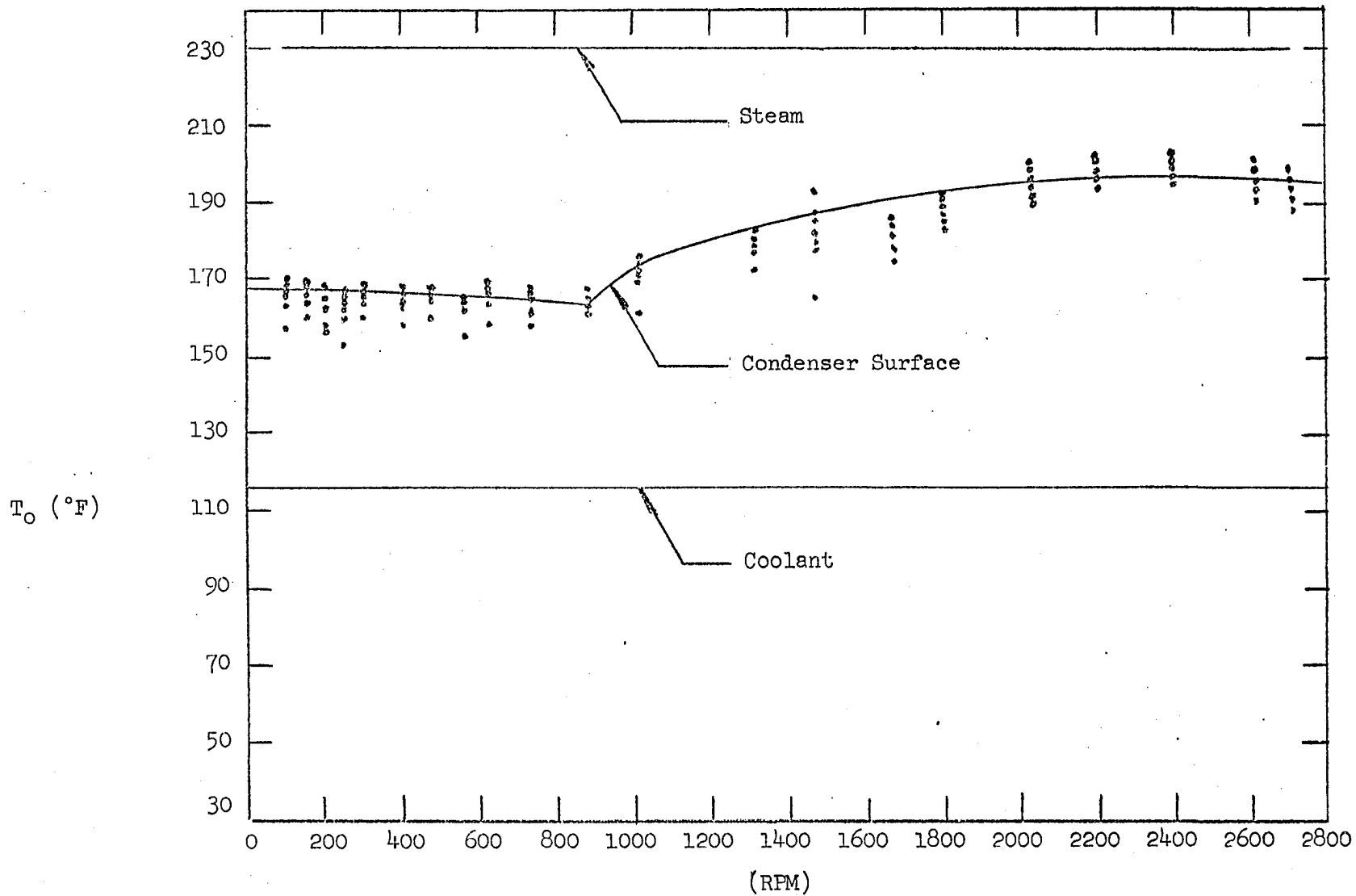
FIGURE 8:   VARIATION OF CONDENSER SURFACE TEMPERATURE WITH ROTATIONAL SPEED   (TEST SERIES I)
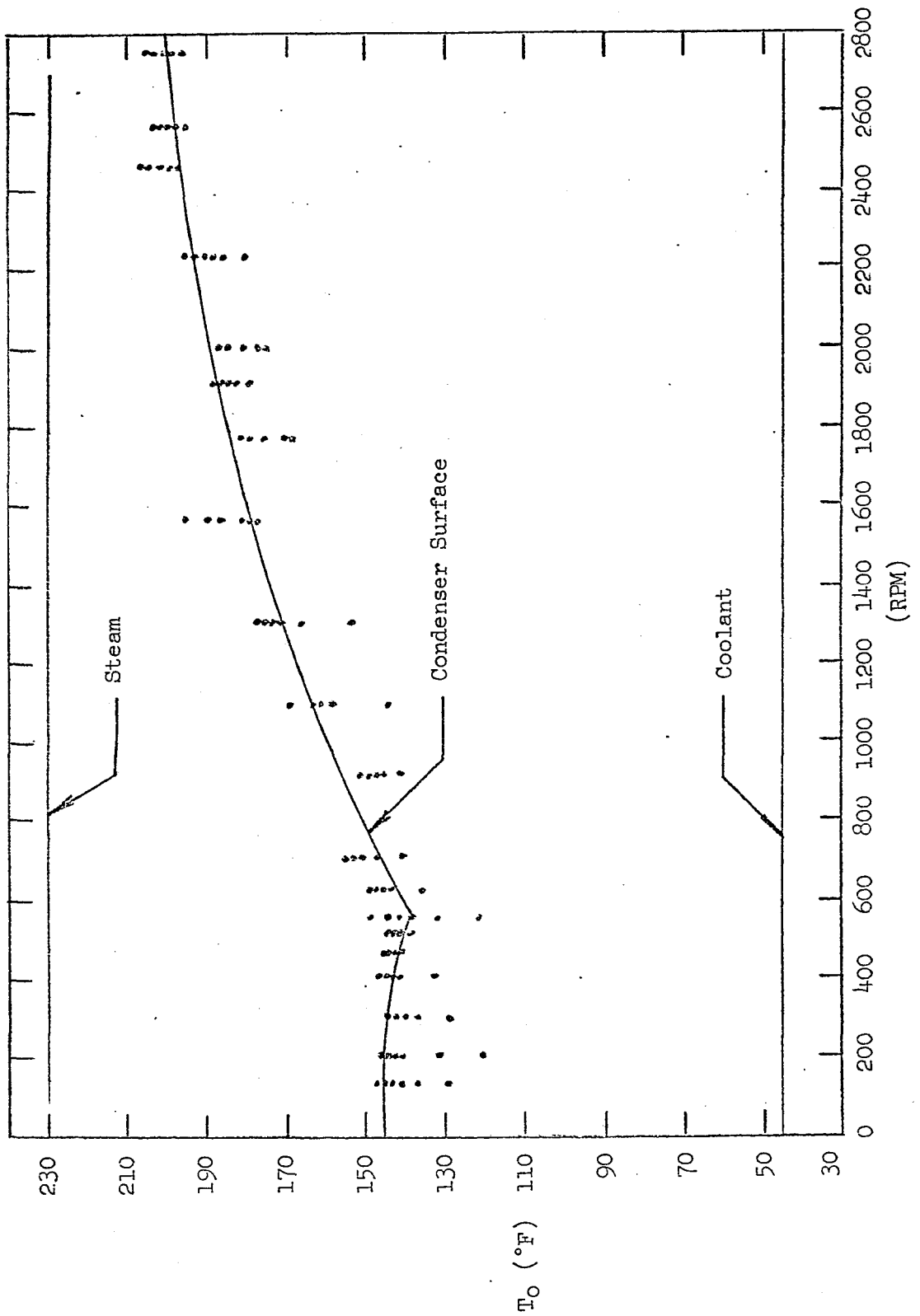
FIGURE 9: VARIATION OF CONDENSER SURFACE TEMPERATURE WITH ROTATIONAL SPEED (TEST SERIES II)
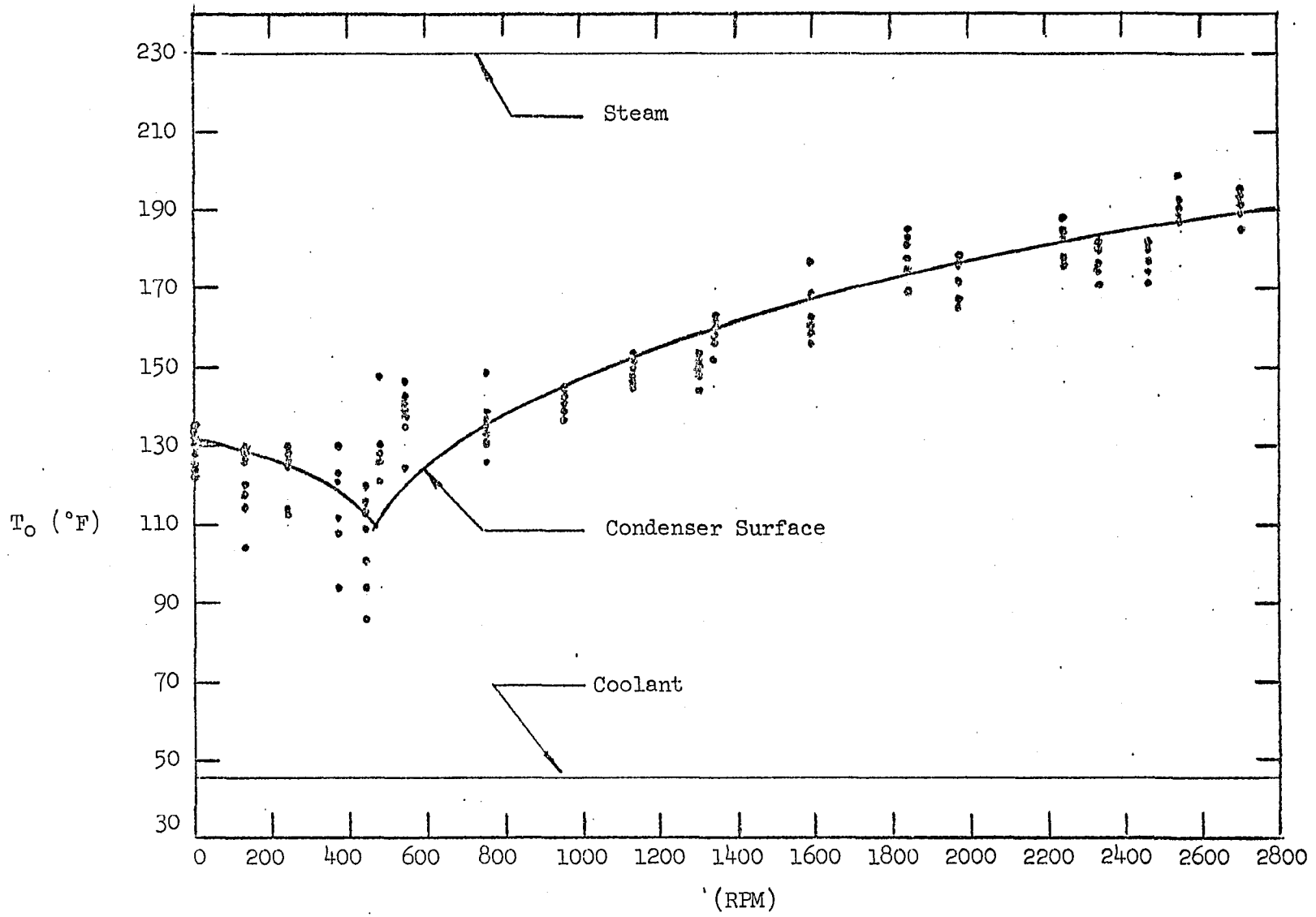
FIGURE 10: VARIATION OF CONDENSER SURFACE TEMPERATURE WITH ROTATIONAL SPEED (TEST SERIES III)
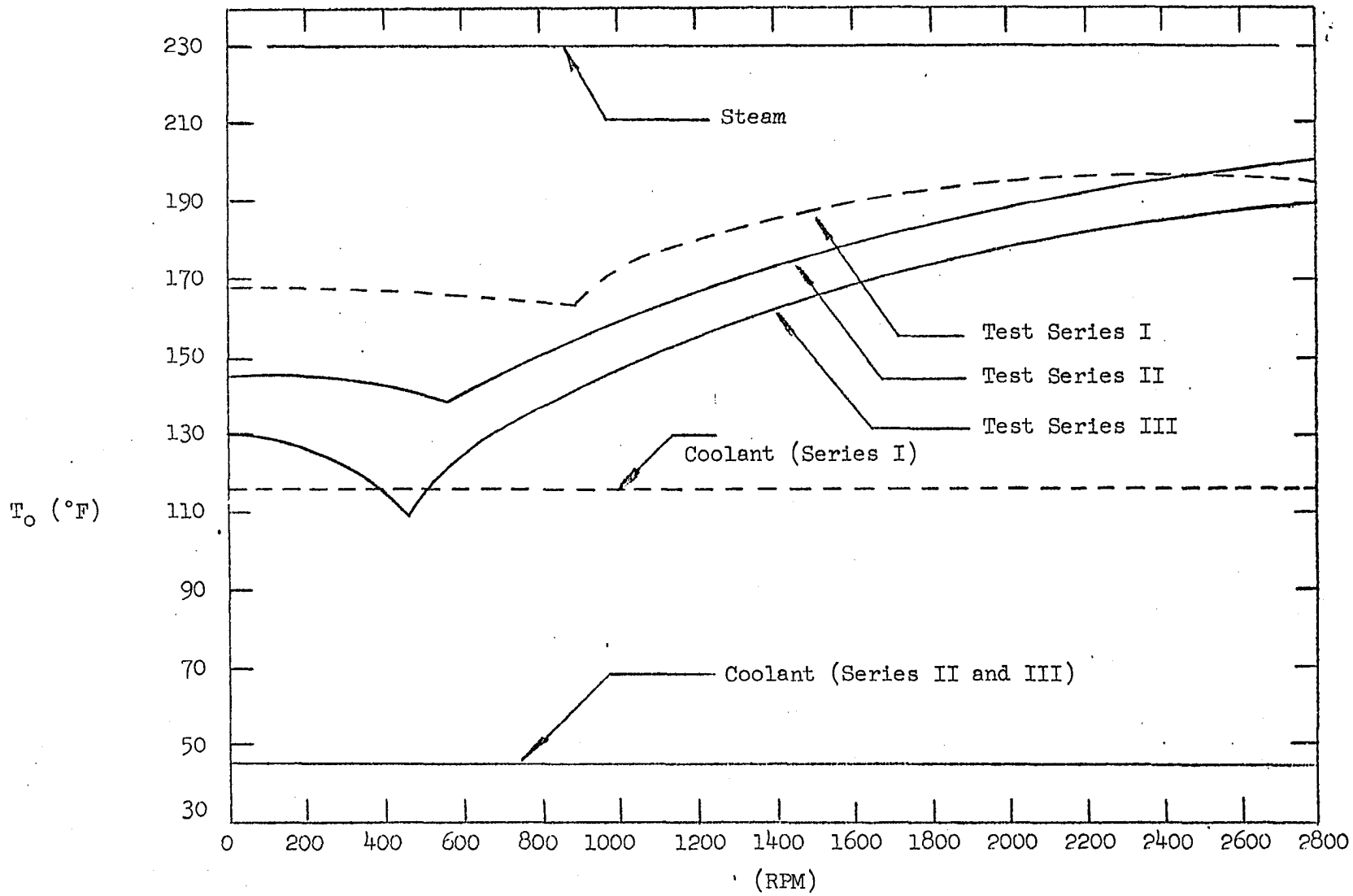
FIGURE 11: VARIATION OF CONDENSER SURFACE TEMPERATURE WITH ROTATIONAL SPEED
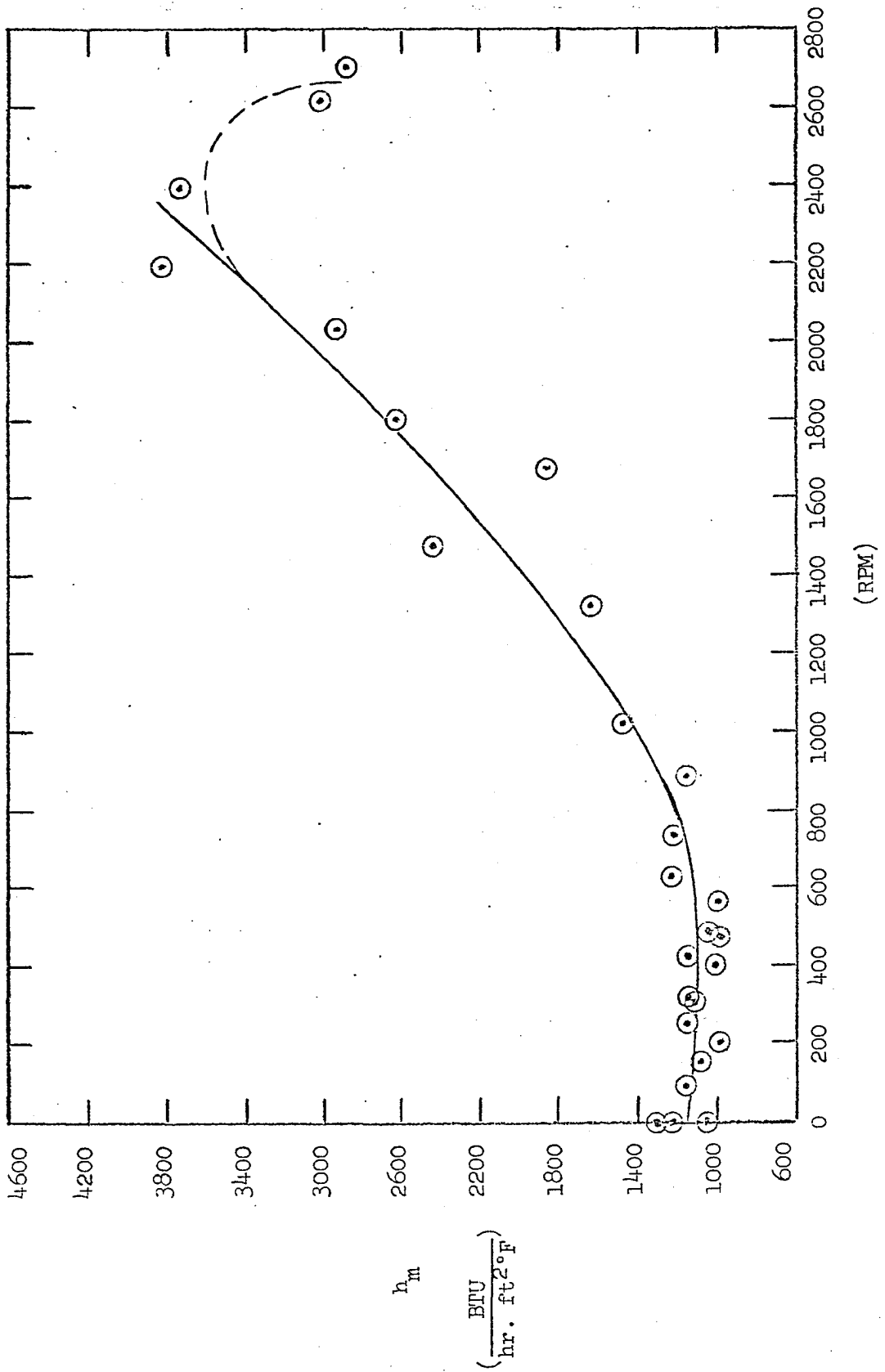
FIGURE 12: VARIATION OF MEAN HEAT TRANSFER COEFFICIENT WITH ROTATIONAL SPEED (TEST SERIES I)
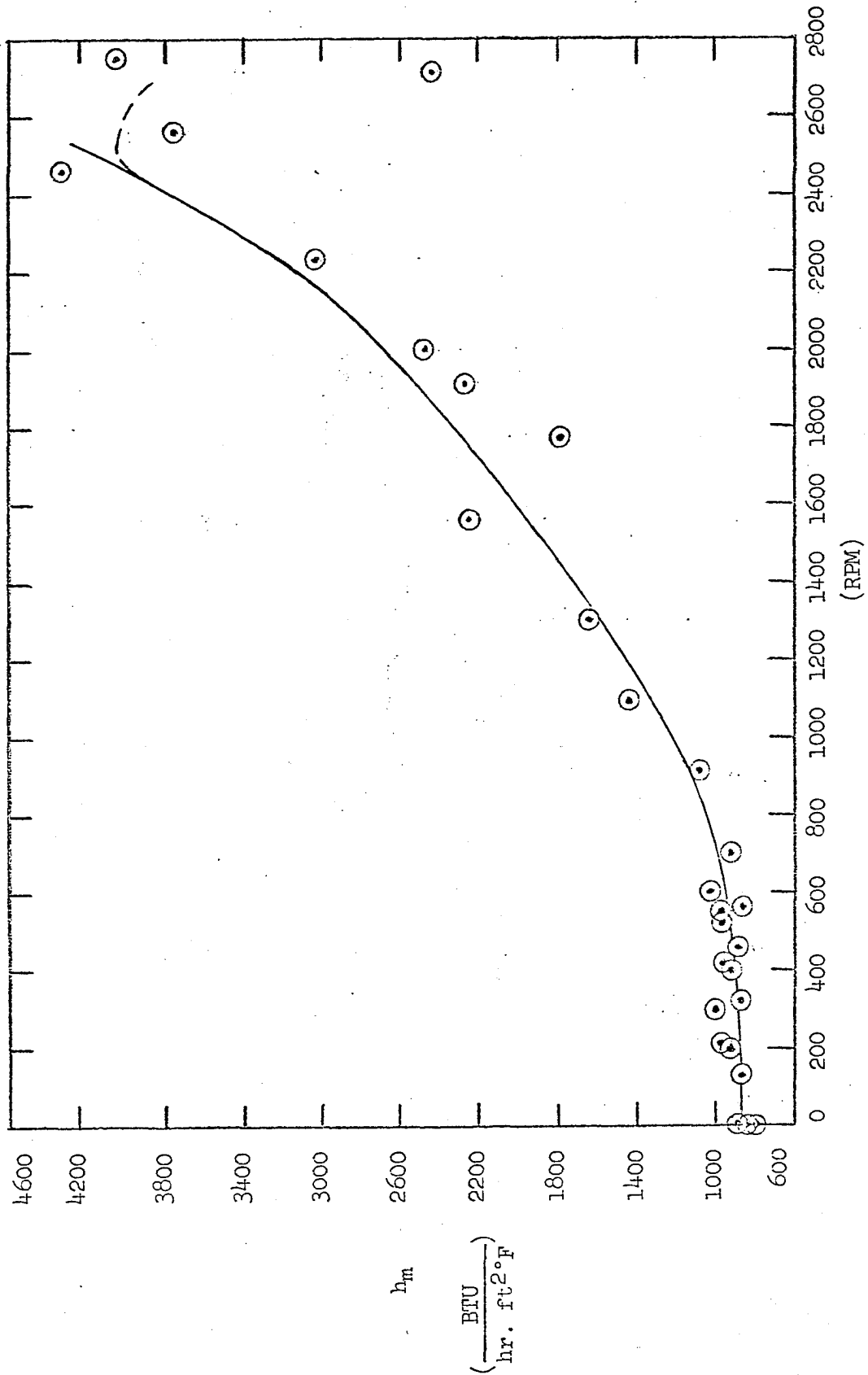
FIGURE 13: VARIATION OF MEAN HEAT TRANSFER COEFFICIENT WITH ROTATIONAL SPEED (TEST SERIES II)
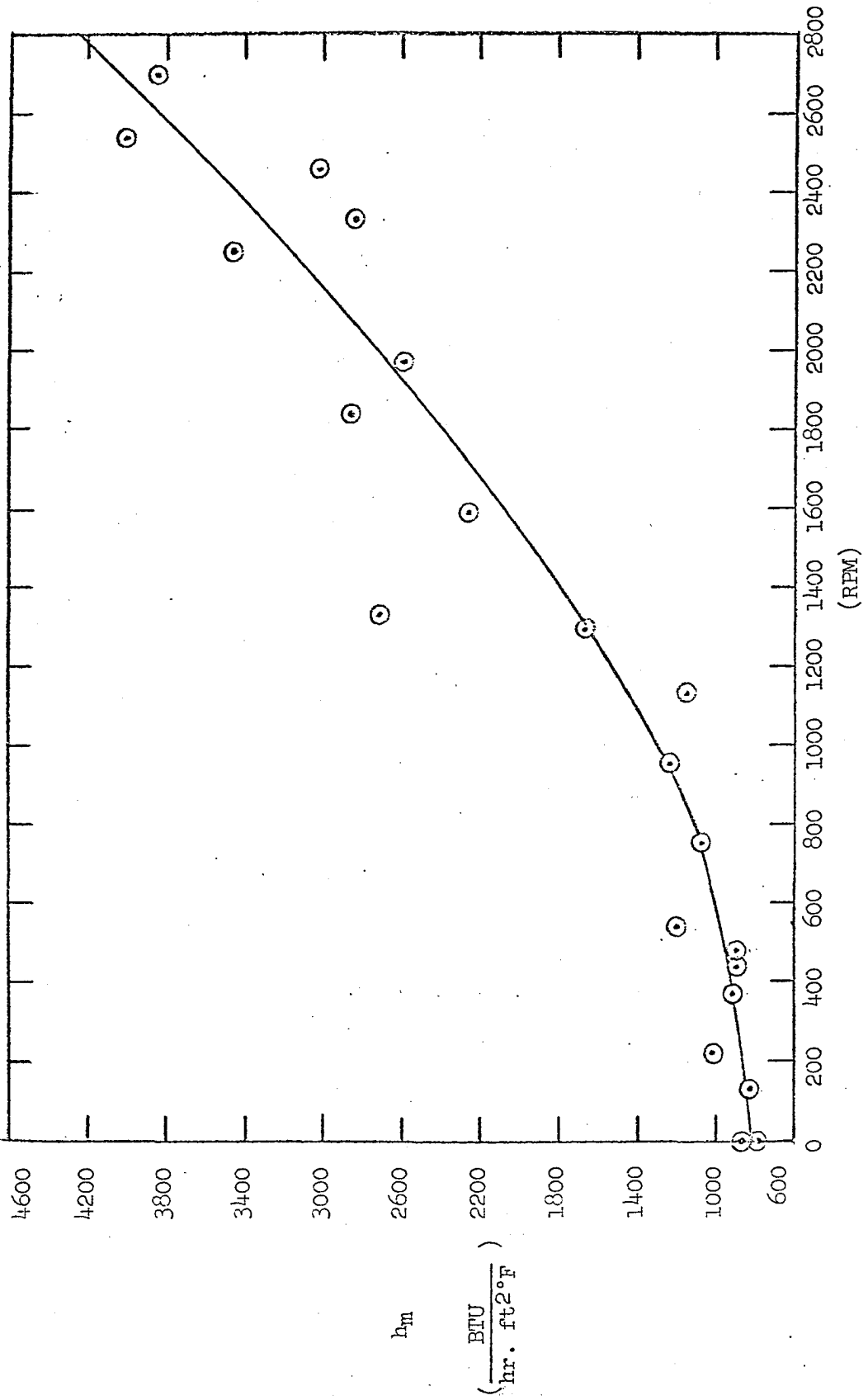
FIGURE 14: VARIATION OF MEAN HEAT TRANSFER COEFFICIENT WITH ROTATIONAL SPEED (TEST SERIES III)

$$Nu = 11.56\ We^{0.4886}$$

Test Series I
Test Series II
Test Series III

We

Nu
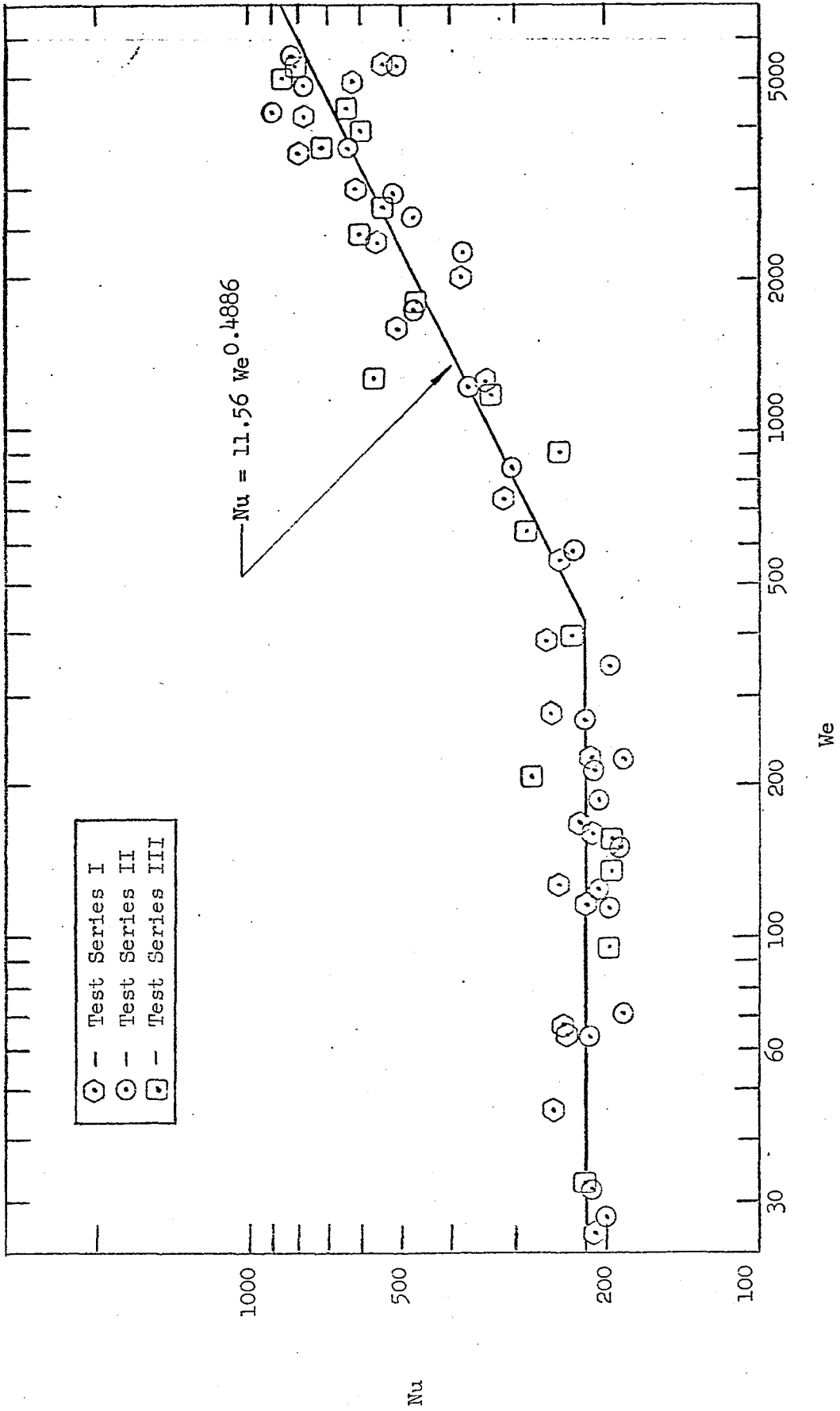
FIGURE 15:   VARIATION OF NUSSELT NUMBER WITH WEBER NUMBER
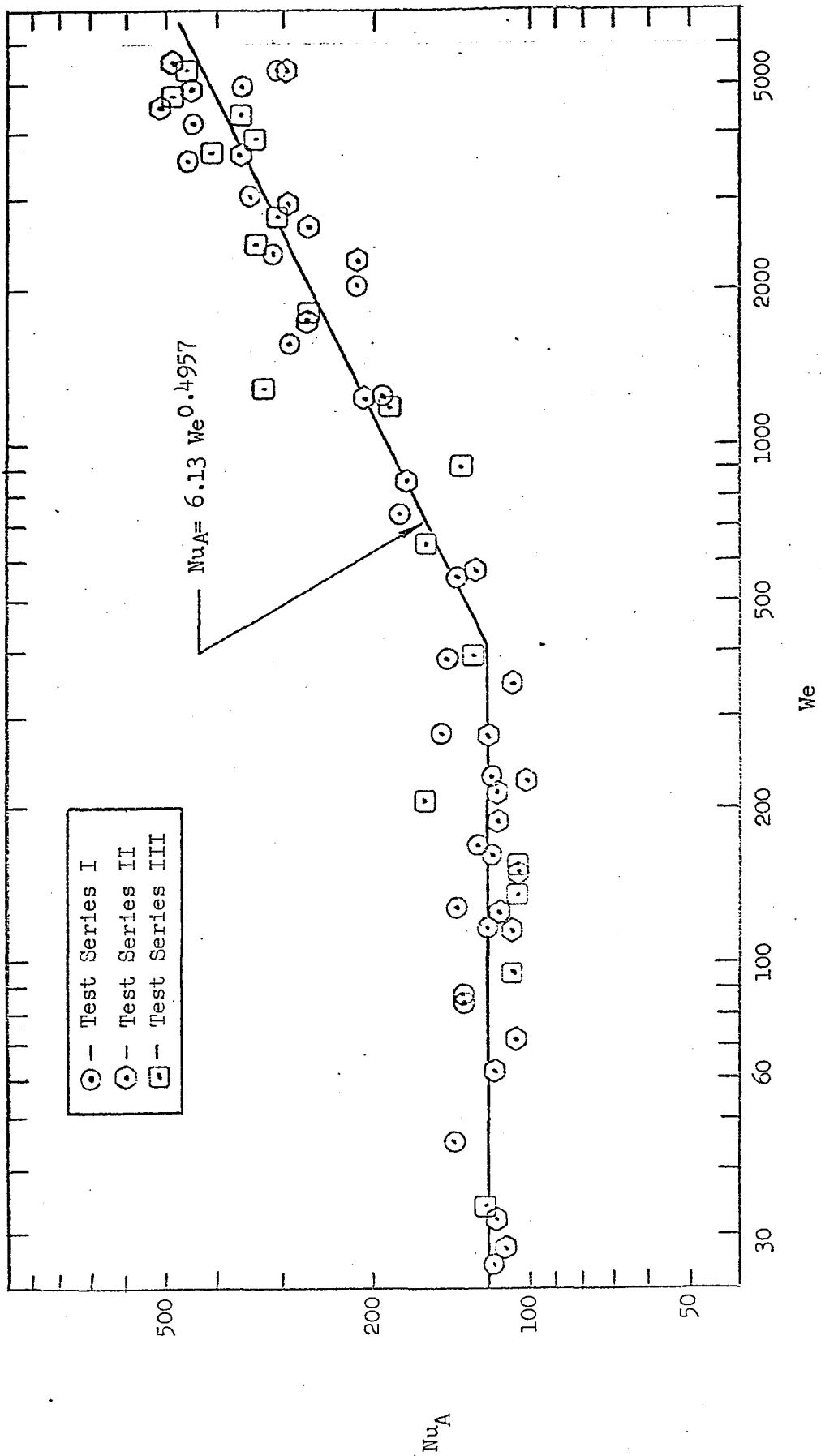
FIGURE 16: VARIATION OF NUSSELT NUMBER (NU$_A$) WITH WEBER NUMBER

$Nu_A = 6.13 \ We^{0.4957}$

Legend:
⊙ — Test Series I
⬡ — Test Series II
⊡ — Test Series III
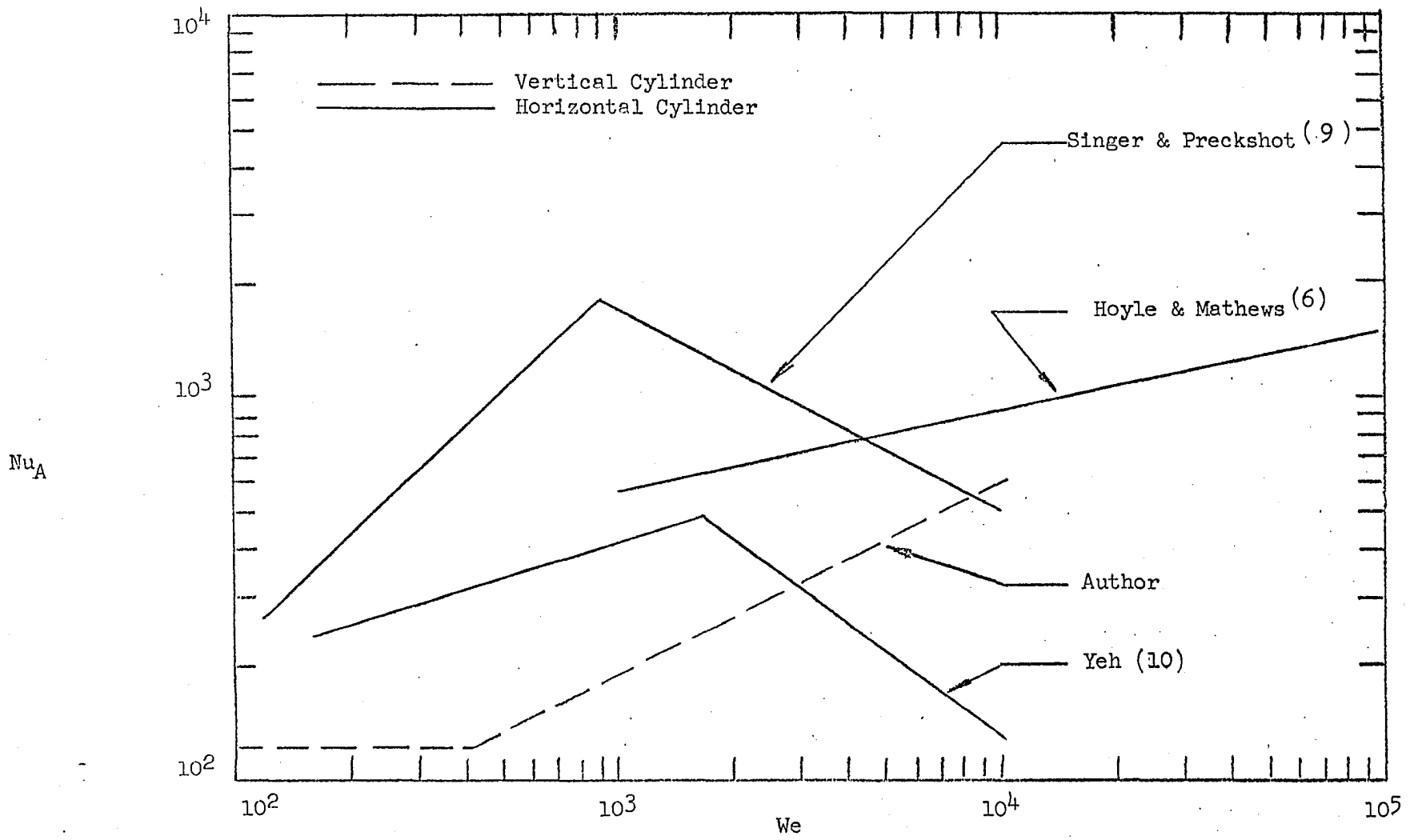
FIGURE 17:   VARIATION OF Nu_A WITH We . A COMPARISON OF AUTHOR'S RESULTS WITH THOSE OF OTHER WORKERS IN THE SAME FIELD

APPENDICIES

DIMENSIONAL ANALYSIS

In order to justify the use of the dimensionless groups used in Figures 15, 16 and 17, the following was performed.

It was assumed that for the present investigation, the following relationship was true:

$$h_m = \frac{Q}{A \triangle T} = f(w, D_o, k_f, \rho_f, g, \sigma)$$

In the F, L, T, θ, system of units, the dimensional matrix for the parameters is:

	1	2	3	4	5	6	7
	w	h_m	D_o	k_f	ρ_f	g	σ
F	0	1	0	1	1	0	1
L	0	-1	1	0	-3	1	-1
T	-1	-1	0	-1	0	-2	0
θ	0	-1	0	-1	0	0	0

It is apparent that the rank (r) of the matrix is four (4) and since there are seven (7) variables, the number of dimensionless groups is 7 - 4 = 3.

The homogeneous linear algebraic equations whose coefficients are the numbers in the rows of the dimensional matrix are the following:

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$$K2 + K4 + K5 + K7 \qquad = 0 \qquad \dots\dots\dots (1)$$

$$- K2 + K3 - 3K5 + K6 - K7 = 0 \qquad \dots\dots\dots (2)$$

$$- K1 - K2 - K4 - 2K6 \qquad = 0 \qquad \dots\dots\dots (3)$$

$$- K2 - K4 \qquad = 0 \qquad \dots\dots\dots (4)$$

Setting K4 through K7 in terms of K1 through K3, the following relations are obtained:

$$K4 = - K2$$

$$K5 = - 1/2K2 + 1/2K3 - 1/4K1$$

$$K6 = - 1/2K1$$

$$K7 = 1/4K1 + 1/2K2 - 1/2K3$$

Hence the matrix of solutions is:

	1	2	3	4	5	6	7
	w	h_m	D_0	k_f	ρ	g	σ
$\pi 1$	1	0	0	0	$- 1/4$	$- 1/2$	$1/4$
$\pi 2$	0	1	0	$- 1$	$- 1/2$	0	$1/2$
$\pi 3$	0	0	1	0	$1/2$	0	$- 1/2$

and the dimensionless groups are:

$$\pi 1 = \frac{w\ \sigma^{1/4}}{\rho_f^{1/4}\ g^{1/2}}$$

$$\pi 2 = \frac{h_m \cdot \sigma^{1/2}}{k_f\ \rho_f^{1/2}}$$

$$\pi 3 = \frac{D_0\ \rho_f^{1/2}}{\sigma^{1/2}}$$

The product $\pi2 \cdot \pi3$ yields the familiar dimensionless group

$$\frac{h_m\, D_o}{k_f} \quad (=Nu)$$

The product $(\pi1)^2 \cdot (\pi3)^3$ yields $\rho_f \dfrac{w^2\, D_o^3}{\sigma_g}$

The two resulting groups then are:

$$\pi1 = h_m\, D_o/k_f$$

$$\pi2 = \frac{\rho_f w^2 D_o^3}{\sigma_g}$$

and the relationship between them is assumed to be of the form:

$$h_m D_o/k_f = f\left(\frac{\rho_f w^2 D_o^3}{\sigma_g}\right)$$

APPENDIX B

SAMPLE CALCULATIONS

The experimental values of heat flux q_W, for both the stationary and the rotational cases were found from the following expression for the heat transferred to the cooling water,

$$q_W = 60 \cdot C_p \cdot m_W \cdot \Delta T$$

where ΔT_W is the cooling water temperature difference as measured at the exit and entrance to the test section, and m_W is the measured cooling water flow.

In order to check the accuracy of the cooling water flow and temperature measurements, the heat load was also calculated from the steam and condensate measurements thus:

$$q_V = \frac{15}{4} \cdot m_V \cdot h_{fg}$$

where h_{fg} is the enthalpy of evaporation and was evaluated at the steam temperature and pressure, and m_V is the measured condensate flow.

The experimental heat transfer coefficient was obtained using the following expression:

$$h_m = q_W / A \, \Delta T_S$$

where ΔT_S is the temperature difference between the steam temperature T_S and the average condenser surface temperature T_{oa}.

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The mean value of Nusselt number was then evaluated in two different ways:

$$Nu = h_m D_o/k_f$$

where k_f is the thermal conductivity of condensate evaluated at mean film temperature T_f, where:

$$T_f = (T_{oa} + T_s)/2$$

and
$$Nu_A = h_m D_o/2 \ k_A$$

where k_A is the thermal conductivity of water at atmospheric conditions.

Weber number was calculated using the following expression:

$$We = (D_o^2 \ \rho_f/\sigma) \ (D_o \ w^2/2 \ g)/2$$

where ρ_f and σ are evaluated at the mean film temperature T_f.

APPENDIX C

VARIATION OF MEASURED AND CALCULATED PARAMETERS WITH ROTATIONAL SPEED

TABLE C-1 (TEST SERIES II)

Barometric Pressure = 29.30" Hg   Steam Pressure = 6.3 psig   Steam Temperature = 232°F

T_w (°F)

RUN NO.	RPM	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}
1	0	43.20	43.75	44.10	44.65	45.00	45.45	45.90	46.48	46.85	47.30	47.60
2	127	48.13	48.55	48.90	49.38	49.85	50.50	50.75	50.95	51.85	52.10	52.78
3	200	43.28	43.65	44.20	44.85	45.00	45.75	46.48	46.85	47.62	47.85	48.00
4	214	47.75	48.20	48.70	49.15	49.60	50.30	50.75	51.40	51.85	52.30	52.55
5	298	48.20	48.68	49.00	49.45	49.95	50.55	51.15	51.60	52.15	52.95	53.45
6	319	43.82	44.80	45.00	45.90	46.15	46.62	46.85	47.30	47.75	48.20	48.70
7	404	43.82	44.25	44.75	45.15	45.60	46.00	46.58	46.98	47.58	48.32	48.60
8	422	48.55	49.05	49.60	50.00	50.62	51.15	51.65	52.05	52.65	53.10	53.60
9	462	44.20	44.80	45.20	45.90	46.30	46.60	46.85	47.30	47.75	48.68	48.83
10	516	43.90	44.25	44.80	45.10	45.83	46.00	46.40	47.15	47.65	48.40	48.70
11	550	48.50	48.90	49.40	49.95	50.50	50.92	51.70	52.10	52.50	53.20	53.42
12	563	44.10	44.55	45.00	45.30	45.78	46.17	46.65	47.17	47.75	48.20	48.70
13	621	44.35	44.70	45.30	45.82	46.48	46.60	47.30	47.68	48.35	48.90	49.45
14	695	48.28	48.62	48.78	49.50	49.90	50.35	50.75	51.30	51.45	52.30	52.65
15	910	48.85	49.55	50.15	50.40	50.95	51.48	52.22	52.70	53.30	54.05	54.50
16	1090	49.10	49.55	50.22	50.70	51.40	52.00	52.60	53.35	54.05	54.55	55.15
17	1305	49.10	49.60	49.95	50.30	50.80	51.36	52.07	52.73	53.15	54.65	55.20
18	1560	49.05	49.65	50.05	51.15	51.60	51.90	52.60	53.40	54.25	54.92	55.00
19	1770	49.20	49.75	50.50	51.30	51.30	52.30	52.85	53.40	54.25	54.90	55.15
20	1910	44.55	45.35	46.15	46.60	47.30	47.75	48.45	49.30	50.50	50.75	51.10
21	2005	49.10	49.65	50.62	51.10	52.05	52.75	53.65	55.00	55.00	55.80	56.15
22	2230	49.15	49.88	50.80	51.70	52.25	53.28	53.80	54.40	55.30	55.75	56.10
23	2460	49.55	50.30	51.10	52.15	52.28	53.65	54.05	55.00	55.92	56.30	56.60
24	2565	44.43	45.30	45.90	46.58	47.50	48.02	48.70	49.40	50.05	50.75	51.20
25	2710	49.35	49.90	50.80	51.40	51.82	52.70	53.20	53.50	54.92	55.63	56.15
26	2750	44.55	45.20	46.15	46.75	47.30	48.00	48.90	49.40	50.30	50.90	51.05

TABLE C-1   (Continued)

RUN NO.	T_{o1}	T_{o2}	T_{o3}	T_{o4}	T_{o5}	T_{o6}	T_{o9}	T_{o10}	m_w lb./min.	m_v ozs./min.	h_m Btu/hr.ft2°F	Nu_A	Nu	We
			T_o (°F)											
1	128.5	141.0	142.0	144.0	144.0	142.0	131.0	113.0	60.3	4.22	870	103	187	0
2	129.0	141.0	143.0	145.0	143.0	147.5	137.0	129.0	60.1	4.69	876	104	188	11
3	132.0	141.5	141.0	144.5	143.0	144.5	149.0	120.0	58.8	4.75	935	111	200	28
4	129.0	141.0	143.0	145.0	143.0	143.0	120.0	110.0	60.2	4.00	993	118	213	32
5	129.0	137.0	140.0	142.5	140.5	142.0	157.0	169.5	61.5	5.04	1005	119	216	63
6	128.5	140.0	143.5	147.5	143.5	143.5	157.0	142.0	59.5	4.85	865	108	185	72
7	133.0	143.5	143.5	145.5	145.5	147.5	164.0	177.5	60.4	5.03	923	109	198	115
8	126.0	137.5	140.0	142.0	145.0	143.0	153.0	169.5	61.1	5.09	972	115	208	126
9	135.0	141.0	144.5	146.0	145.5	145.5	166.0	186.5	59.5	4.62	886	105	190	151
10	133.0	142.0	145.5	143.0	144.0	144.5	161.0	178.0	60.0	5.11	970	115	208	188
11	122.0	132.0	138.0	142.0	145.0	145.0	149.0	162.0	62.5	5.07	988	117	212	213
12	119.5	133.0	136.0	140.5	140.5	140.5	146.2	150.5	60.8	3.69	858	101	184	223
13	136.0	145.5	145.5	148.5	145.5	145.5	161.0	181.5	61.0	5.22	1025	121	219	273
14	140.5	147.5	152.0	154.5	152.0	152.0	166.0	187.0	58.2	5.39	922	109	197	343
15	133.0	141.0	147.5	147.5	148.5	148.5	148.5	162.0	58.0	5.06	1081	128	231	586
16	144.5	158.0	162.0	163.0	162.0	162.0	169.0	177.5	58.4	5.95	1458	173	310	849
17	154.0	166.5	174.0	174.0	172.0	172.0	175.5	187.5	58.5	6.27	1746	207	371	1226
18	159.0	177.0	196.0	187.0	179.0	180.0	176.0	186.0	59.0	5.98	2259	268	477	1766
19	141.5	170.0	175.5	179.0	169.0	171.0	163.5	166.0	59.0	5.38	1798	213	382	2257
20	149.5	179.5	184.5	184.5	179.5	184.5	171.0	169.5	60.4	6.30	2267	269	479	2645
21	154.0	181.0	185.5	185.5	177.5	175.5	168.0	169.5	58.9	5.72	2475	294	524	2912
22	164.0	188.0	193.5	195.5	186.5	188.0	180.0	180.0	58.2	5.76	3040	361	641	3625
23	181.5	201.0	206.5	206.5	197.0	199.0	197.0	197.5	57.6	6.75	4332	514	910	4447
24	177.5	198.5	201.0	203.5	199.5	196.0	196.5	196.5	59.8	5.07	3766	447	791	4827
25	161.5	186.0	186.0	184.0	183.5	183.5	169.5	177.0	58.3	5.47	2431	289	514	5333
26	179.5	201.0	204.0	204.0	199.0	197.0	193.5	190.0	60.1	6.02	4066	482	854	5553

TABLE C-2 (TEST SERIES I)

Barometric Pressure = 29.53" Hg   Steam Pressure = 6.3 psig   Steam Temperature = 232°F

T_W (°F)

RUN NO.	RPM	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}
27	0	119.15	119.45	119.90	120.20	120.65	120.90	121.15	121.55	121.80	122.40	122.50
28	94	115.60	115.95	116.25	116.50	116.85	117.20	117.42	117.80	118.10	118.48	118.50
29	154	121.50	121.75	122.10	122.40	122.95	123.10	123.15	124.05	123.65	124.15	124.60
30	193	114.95	115.25	115.45	115.68	116.20	116.42	116.68	116.92	117.10	117.45	117.60
31	253	120.65	120.85	121.35	121.95	121.95	122.52	122.20	122.70	123.05	123.65	124.05
32	300	117.10	117.35	117.60	117.85	118.13	118.45	118.68	119.00	119.35	119.78	120.00
33	307	114.05	114.30	114.45	114.70	115.02	115.38	115.75	116.00	116.30	116.65	117.05
34	403	113.55	113.85	114.05	114.40	114.70	114.80	115.13	115.55	115.92	116.18	116.50
35	421	115.95	116.30	116.55	116.95	117.15	117.45	117.65	118.05	118.40	118.70	119.30
36	474	115.05	115.35	115.55	115.90	116.15	116.42	116.70	117.00	117.20	117.68	117.88
37	484	113.82	113.93	114.30	114.63	114.90	115.10	115.41	115.70	116.13	116.45	116.81
38	565	113.55	113.80	114.00	114.40	114.58	114.80	115.25	115.50	115.70	115.80	116.50
39	623	112.40	112.80	113.17	113.46	113.75	114.28	114.53	114.85	115.25	115.80	116.13
40	735	107.83	108.25	108.60	108.92	109.32	109.73	109.95	110.38	110.65	111.00	111.22
41	883	117.65	118.00	118.35	118.45	118.95	119.35	119.45	119.85	120.50	120.85	121.20
42	1015	106.45	106.90	107.25	107.65	108.05	108.40	108.80	109.15	109.50	109.95	110.45
43	1315	107.00	107.75	108.05	108.50	108.65	108.95	109.25	109.70	110.20	110.55	111.00
44	1475	118.60	118.60	119.10	119.60	119.90	120.45	120.85	121.50	122.75	122.75	123.15
45	1670	109.95	110.35	110.85	111.20	111.73	112.05	112.55	112.85	113.35	113.65	113.85
46	1800	118.75	119.50	120.20	120.60	121.40	121.25	121.35	122.15	122.65	123.45	123.95
47	2030	111.43	111.95	112.45	112.95	113.35	113.78	114.20	114.63	115.00	115.45	115.75
48	2195	112.85	113.42	114.00	114.50	114.95	115.50	115.65	116.42	117.10	117.50	117.95
49	2390	112.05	112.55	113.15	113.60	114.05	114.63	114.95	115.38	116.08	116.50	117.33
50	2610	111.05	111.50	111.88	112.55	112.95	113.42	113.60	114.15	114.53	115.00	115.35
51	2705	110.65	111.10	111.40	111.95	112.40	112.75	113.12	113.52	113.95	114.40	114.75

TABLE C-2 (Continued)

RUN NO.	T_o (°F)								m_w (lb./min.)	m_v (ozs./min.)	h_m (Btu/hr. ft²°F)	Nu_A	Nu	We
	T_{o1}	T_{o2}	T_{o3}	T_{o4}	T_{o5}	T_{o6}	T_{o9}	T_{o10}						
27	158.0	168.5	168.0	166.0	166.0	167.0	172.0	149.5	79.6	3.35	1326	157	281	0
28	157.5	169.0	163.0	169.0	168.0	168.0	169.5	164.5	83.4	3.29	1177	139	250	7
29	160.0	169.5	166.0	169.5	169.5	164.0	166.0	166.0	79.6	3.31	1094	129	232	17
30	158.0	165.0	162.5	168.0	162.0	162.0	157.5	158.0	83.3	3.17	992	118	211	27
31	153.0	164.0	162.5	164.5	167.5	163.5	166.5	159.5	78.9	3.43	1179	140	251	46
32	158.5	169.0	163.5	169.5	170.5	168.0	177.5	173.5	83.5	3.75	1118	133	237	65
33	160.0	166.0	165.0	169.5	169.0	168.0	177.0	192.0	80.5	3.88	1138	135	242	68
34	158.5	164.0	166.0	168.0	166.5	164.0	175.0	188.0	79.2	3.89	1020	121	217	116
35	164.0	170.0	165.0	170.5	170.5	165.5	168.0	158.0	82.4	3.28	1162	138	247	127
36	160.0	164.0	165.0	168.0	168.0	165.0	179.0	199.0	81.5	4.18	995	118	211	161
37	160.0	166.0	167.5	168.0	168.0	168.0	169.5	190.0	78.6	4.01	1059	126	225	168
38	155.0	164.0	164.0	164.0	164.0	162.0	171.0	185.0	82.1	3.80	1007	119	214	228
39	158.5	164.0	168.0	168.0	168.0	168.0	175.5	186.5	77.0	4.81	1249	148	265	278
40	158.0	161.0	165.0	166.0	166.5	165.0	173.5	184.0	79.8	4.01	1225	145	261	387
41	153.5	161.0	164.0	165.5	168.0	163.0	167.5	184.0	78.0	4.24	1164	138	248	558
42	161.0	169.0	172.0	173.0	173.5	170.5	176.0	190.0	79.7	5.17	1499	178	318	742
43	163.0	172.5	178.0	178.0	178.0	175.0	178.0	189.0	78.9	5.25	1624	192	344	1249
44	165.5	180.5	193.0	186.0	182.5	179.0	186.0	192.0	78.6	5.17	2443	290	516	1579
45	175.5	183.0	184.5	182.5	178.0	176.5	178.0	183.5	77.8	5.21	1816	215	384	2020
46	166.5	189.0	191.0	191.5	183.5	184.5	181.5	186.0	77.5	5.15	2644	313	558	2358
47	175.0	196.0	201.0	200.0	193.5	194.0	189.5	195.0	79.0	5.75	2937	348	618	3018
48	184.5	201.0	201.0	201.0	198.0	196.5	197.0	197.0	81.9	5.94	3824	454	804	3534
49	184.0	202.0	201.5	203.0	197.0	196.0	197.0	197.0	81.5	5.70	3744	444	787	4191
50	183.5	202.0	199.0	199.0	194.0	191.0	194.0	194.0	81.7	6.00	3012	357	634	4989
51	181.0	199.0	195.0	196.0	190.0	190.0	190.0	190.0	81.5	5.74	2598	308	547	5348

TABLE C-3 (TEST SERIES III)

Barometric Pressure = 29.35" Hg   Steam Pressure = 6.3 psig   Steam Temperature = 232°F

T_w (°F)

RUN NO.	RPM	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}
52	0	45.25	45.45	45.70	45.92	46.18	46.38	46.62	46.83	47.30	47.75	48.45
53	127	45.18	45.65	46.00	46.25	46.55	46.70	46.92	47.20	47.68	47.55	47.55
54	217	46.95	46.75	46.75	47.20	47.45	47.80	47.95	48.40	48.40	48.80	49.25
55	373	46.30	46.48	46.75	47.05	47.40	47.75	48.00	48.45	48.90	49.35	49.85
56	444	45.95	46.35	46.60	46.95	47.40	47.75	48.12	48.45	48.90	49.35	49.60
57	475	45.70	46.00	46.25	46.62	46.85	47.05	47.30	47.75	48.21	48.45	48.68
58	541	46.38	46.83	47.30	47.75	47.98	48.60	48.90	48.90	49.38	49.95	50.15
59	752	45.13	45.45	48.88	46.15	46.53	46.75	47.20	47.55	48.00	48.22	48.60
60	955	45.88	46.30	46.70	47.05	47.48	47.98	48.22	48.58	49.15	49.38	49.85
61	1130	45.98	46.35	46.75	46.98	47.30	47.68	48.00	48.45	48.60	49.00	49.15
62	1290	45.95	46.30	46.85	47.30	47.65	47.95	48.45	48.90	49.50	49.85	50.05
63	1335	45.70	46.15	46.60	47.07	47.53	47.75	48.22	48.92	49.15	49.85	50.30
64	1590	46.00	46.43	46.85	47.30	47.75	48.22	48.78	49.20	49.85	50.30	50.70
65	1840	45.60	46.05	46.60	49.97	47.40	47.98	48.45	48.90	49.38	49.75	50.20
66	1970	46.30	46.83	47.25	47.68	48.20	48.69	49.28	49.92	50.30	50.50	50.75
67	2245	45.88	46.52	46.98	47.43	47.80	48.22	48.82	49.60	50.20	51.40	52.00
68	2330	46.30	46.65	47.15	47.68	48.13	48.60	49.15	49.70	50.25	50.65	50.95
69	2465	45.90	46.48	46.98	47.68	47.93	48.50	48.90	49.38	49.83	51.00	52.22
70	2545	45.82	46.42	46.70	47.25	47.85	48.28	48.82	49.38	49.79	50.20	50.50
71	2700	45.83	46.43	46.88	47.30	47.85	48.40	48.68	49.25	49.80	50.15	50.50

TABLE C-3   (Continued)

RUN NO.	T_{o1}	T_{o2}	T_{o3}	T_{o4}	T_{o5}	T_{o6}	T_{o9}	T_{o10}	m_w lb./min.	m_v ozs./min.	h_m Btu/hr. ft²°F	NuA	Nu	We
					T_o (°F)									
52	114.0	124.5	124.5	128.5	122.5	124.5	132.5	120.0	111.3	4.75	789	94	170	0
53	114.5	128.5	128.5	128.5	120.0	118.5	116.0	100.0	108.5	5.00	839	99	181	11
54	114.5	125.0	126.5	130.0	127.5	129.0	114.0	104.5	107.9	4.87	1017	121	219	33
55	94.5	108.5	112.5	121.0	123.5	123.5	130.0	132.5	106.4	6.78	915	190	198	97
56	86.5	96.0	101.5	109.0	116.5	114.5	120.5	130.5	108.9	5.78	894	106	194	136
57	121.0	126.5	128.0	130.5	128.5	128.0	148.7	171.3	110.8	4.31	901	107	194	158
58	124.5	135.8	140.5	142.0	142.0	139.5	147.5	161.0	109.0	6.60	1303	159	280	206
59	126.5	135.0	135.0	137.0	135.0	130.5	135.0	149.5	107.5	4.87	1078	128	232	397
60	125.5	137.0	139.0	141.0	143.5	139.0	145.5	171.5	108.6	6.89	1337	159	287	643
61	132.0	145.0	148.5	148.5	148.8	148.8	151.5	138.3	109.4	6.50	1148	136	246	905
62	133.0	144.0	150.5	152.0	148.5	149.5	150.3	168.0	107.2	6.77	1569	186	336	1180
63	139.0	152.0	158.0	159.0	158.0	159.0	162.5	179.5	113.4	7.34	2730	324	578	1288
64	135.0	156.5	177.5	168.5	160.0	160.0	153.0	161.0	112.0	7.98	2267	269	483	1811
65	145.5	181.5	182.5	185.5	169.5	175.5	166.0	181.0	113.7	7.46	2859	339	605	2449
66	137.5	172.5	176.5	178.0	167.0	166.5	155.5	158.0	110.9	8.13	2593	308	550	2795
67	144.5	178.0	183.0	188.0	177.0	184.0	174.0	174.0	116.0	8.92	3466	411	733	3654
68	141.0	174.5	176.0	180.0	171.5	180.0	163.0	169.5	111.5	8.39	2849	338	604	3921
69	137.5	171.0	174.5	180.0	177.5	181.5	172.0	169.5	114.9	8.28	3050	362	646	4390
70	161.0	189.0	190.0	199.0	189.0	191.0	186.0	185.0	115.6	8.82	4087	485	861	4726
71	164.0	189.2	194.0	195.5	185.0	192.0	180.5	186.0	108.2	5.15	3849	457	811	5317
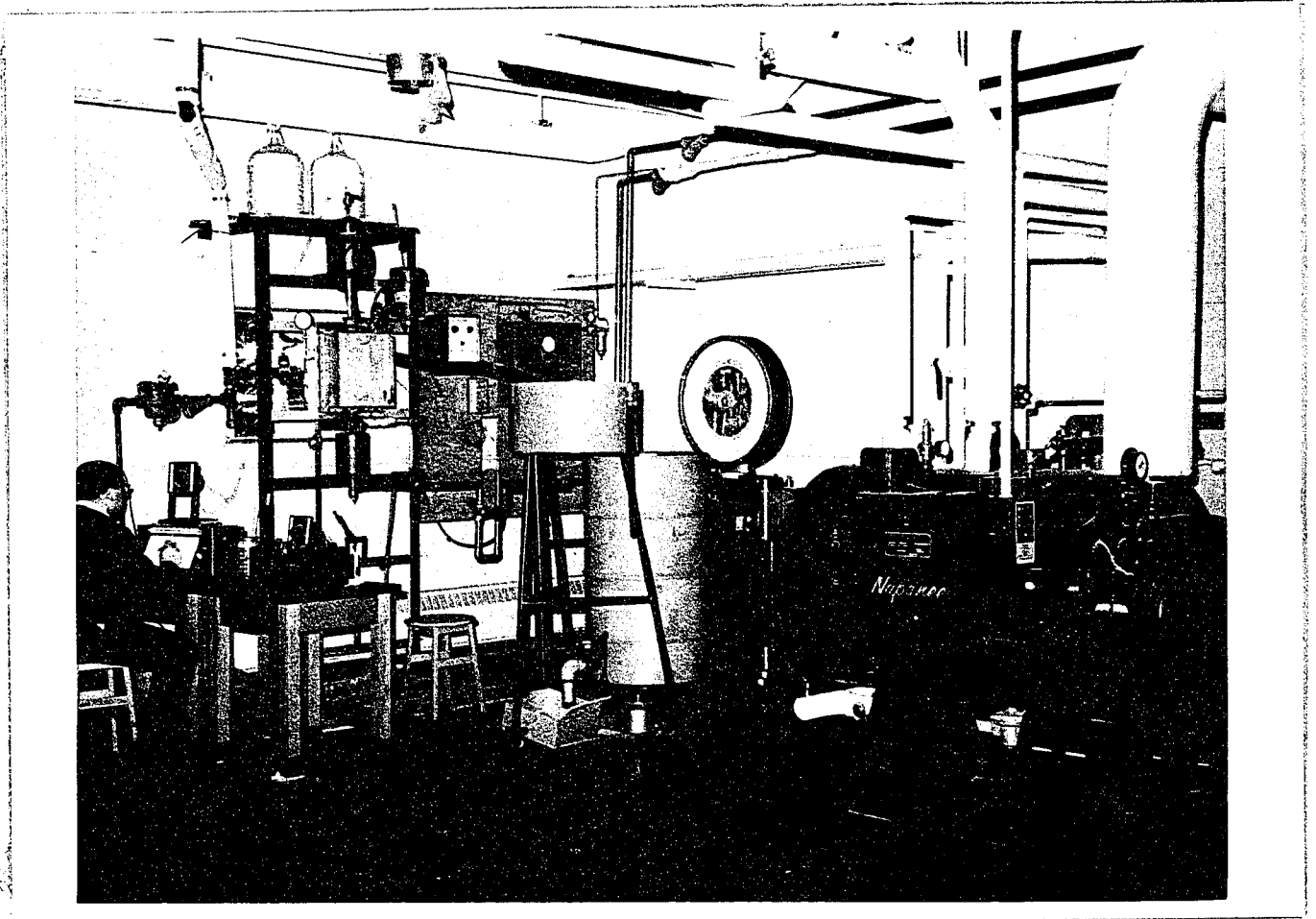
PHOTOGRAPH A: COMPOSITE OF ALL THE EQUIPMENT
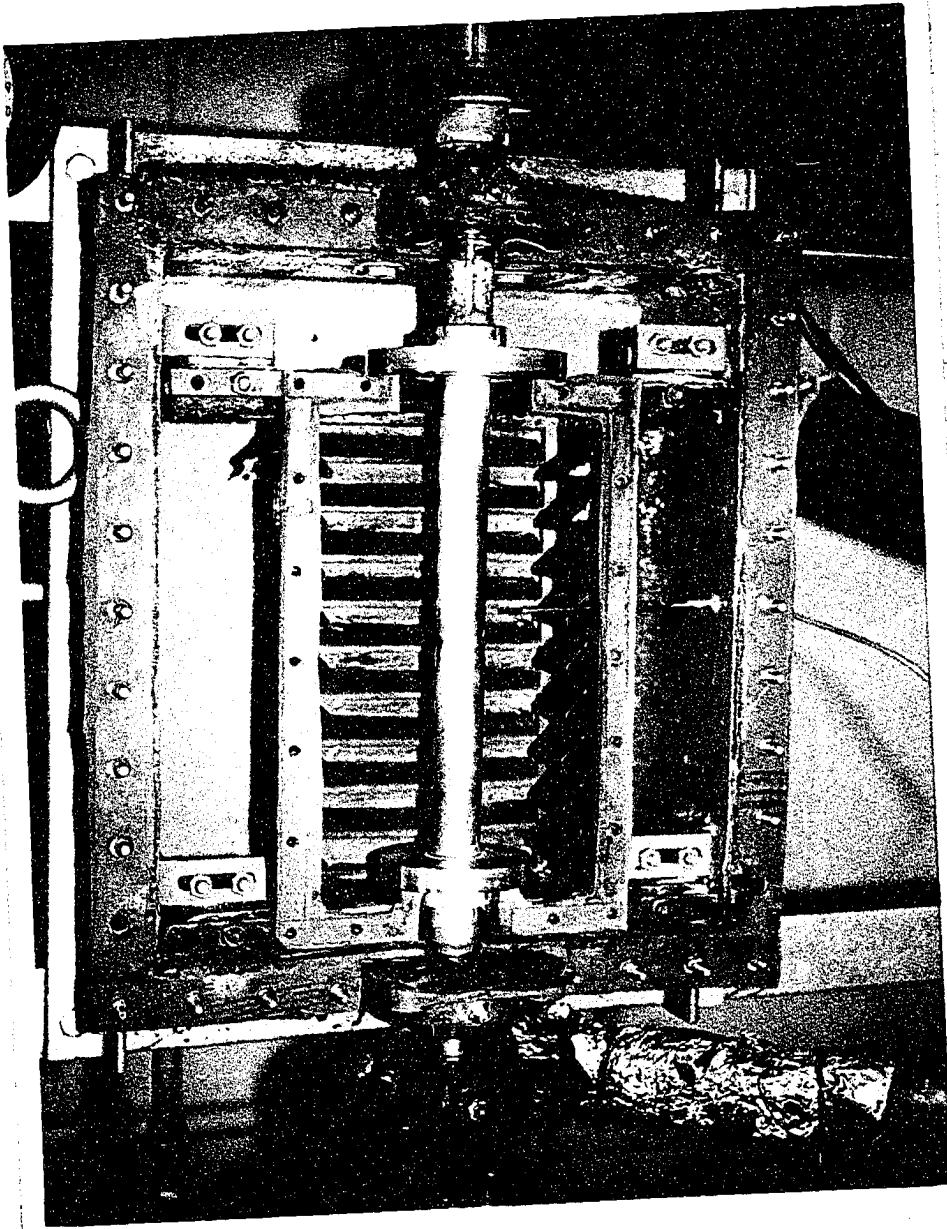
PHOTOGRAPH B: STEAM CHAMBER CLOSE-UP

VITA AUCTORIS

1940   Born in Gračac, Yugoslavia on July 5.

1961   Finished High School (Herman Collegiate Institute in
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