Condensation of steam on a vertically rotating cylinder.

Milan Gacesa

University of Windsor

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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
Department of Mechanical Engineering in Partial
Fulfillment of the Requirements for 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 ΔT = 180°F and by 280% for ΔT = 110°F. Also at ΔT = 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.
ACKNOWLEDGEMENTS

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NOMENCLATURE

A  
Condenser surface area (ft²)

cₚ  
Specific heat of condensate (Btu/lb °F)

D₀, D₁  
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 (R₂w²/g)

g  
Gravitational constant (ft/hr²)

G  
Condensate mass velocity ( = \frac{60 m_w}{\pi D₀} ) (lb/hr ft.)

hₓ, hₘ  
Local and mean heat transfer coefficients (Btu/hr. ft²°F)

h_fg  
Enthalpy of evaporation (Btu/lb)

kₚ  
Thermal conductivity of condensate (Btu/hr. ft. °F)

kₐ  
Thermal conductivity of water at atmospheric conditions (Btu/hr. ft. °F)

mₘₜ  
Cooling water flow rate (lb/min.)

m_v  
Steam flow rate (ozs/min.)

Nux, Nu, < Nu >  
Local, mean and average value of Nusselt number (hD₀/kₚ)

Nuₐ  
Nusselt number (hₘD₀/2kₐ)

Pr  
Prandtl number (cₚμₚ/kₚ)

qₘₜ  
Heat flux based on cooling water measurements (Btu/hr.)

R₀  
External tube radius (ft.)

Re  
Reynolds number (\frac{4G/μₚ}{νₚ})

Reₜ  
Reynolds number (VM₀/νₚ)

T_f  
Mean film temperature (= T₁₀ + Tₙ)/2 (°F)
Ts
Steam saturation temperature (°F)

T01, T02, ... T010
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)

T0a
Average condenser surface temperature, (T01 + T02 + ... T010)/10 (°F)

ΔTs
Temperature difference between steam temperature Ts and average condenser surface temperature T0a (°F)

Tob, Tot
Condenser surface temperature at top and bottom (°F)

ΔTob
Temperature difference at the bottom of the condenser (Ts - Tob) °F

ΔTot
Temperature difference at the top of the condenser (Ts - Tot) °F

T1, T2 ... T11
Cooling water temperature at locations, 0, 1, 2, ... 10 inches from the bottom of the test section, respectively (°F)

Tw
Average cooling water temperature, (T1 + T2 + ... + T11)/11 (°F)

ΔT
Overall temperature difference (Ts - Tw) °F

ΔTw
Cooling water overall temperature rise (T11 - T1) °F

Vm
Mean film velocity (wR0) ft/hr.

W
Angular velocity (RPM)

w
Angular velocity (Hr-1)

We
Weber number ($\frac{\rho_w^2 D_0^3}{\nu_f}$)

μf
Dynamic coefficient of viscosity (lb/ft. hr.)

νf
Kinematic coefficient of viscosity (ft²/hr.)

ρf
Specific weight of condensate (lb/ft3)

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
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.
Nusselt in 1916 made a theoretical estimate of the heat transfer for the case of film condensation of a pure saturated vapour on stationary 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:

\[ h m \left( \frac{\mu_f^2}{k_f^{3/2} \rho_f^2 g} \right)^{1/3} = 1.47 \left( \frac{h \Delta T}{\mu_f} \right)^{-1/3} \]  

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

\[(hm)_{\text{TRUE}} = F (hm)_{\text{NUSSELT}}\]  

(2)

where F is obtained as shown in Table I.

**TABLE I**

<table>
<thead>
<tr>
<th>F</th>
<th>$\Delta T_{ob}/\Delta T_{ot}$</th>
</tr>
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<tbody>
<tr>
<td>0.96</td>
<td>0.5</td>
</tr>
<tr>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td>1.06</td>
<td>2.0</td>
</tr>
<tr>
<td>1.15</td>
<td>5.0</td>
</tr>
</tbody>
</table>

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:

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

(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\textsuperscript{(10)} in 1953 did the pioneering work. His technique has been criticized by subsequent workers\textsuperscript{(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:

\[ h_m = \frac{k_F \rho_F v_m^2}{\sigma g} \]  

(4)

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

\[ I = \text{Const.} \ (Re)^n \]  

(5)

where \( I = \frac{h_m}{k_F \rho_F v_m^2/\sigma g}, \ Re = \frac{v_m D_o}{v_f} \)

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. 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:

\[
\text{Nu}_x = \frac{h_x D_0}{k_f} = \left(\frac{2}{3}\right)^{1/4} \frac{g D_0^3 h_{fg} (\rho_f - \rho_v)}{\nu_f k_f (T_g - T_0)} \left(1 + \frac{3 C_p \Delta T}{8h_{fg}}\right)^{1/4} (\delta^+)^{1/2}
\]

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

The average value of Nusselt number is then:

\[
<Nu> = \frac{1}{2\pi} \int_0^{2\pi} h(\phi) D_0/k_f d\phi
\]

(7a)

and the mean value is:

\[
\text{Nu} = \frac{1}{L} \int_0^L <\text{Nu}> dL
\]

(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:

\[
\text{Nu}_x = \frac{h_x D_0}{k_f} = 2 \left[1 + \left(\frac{3}{2\text{We}}\right)^{1/3}\right]^{-1}
\]

(8)
For very high rotational speeds the authors suggest that Nusselt number is a function of \((\text{We} )^{1/2}\) rather than \(\text{We}^1\) 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 \((\Delta T)\). For this reason the authors proposed an empirical correlation of the form:

\[
\text{Nu} \left( g D_0^3 \rho_f h_{fg}/\nu_f k_f \Delta T \right)^{-1/4} = \text{Constant} \cdot \text{We}^n
\]  

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

\[
\text{Nu} = C \left( g D_0^3 \rho_f h_{fg}/\nu_f k_f \Delta T \right)^{+1/4}
\]

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

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

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

Hoyle and Matthews \(\text{(5)}\) 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:

\[
\frac{h_m D_o}{2 k_A} = 0.75 \left( \frac{D_o \left( \frac{\rho_f}{\sigma} \right)^{0.19}}{1.9 - 0.9/1.095 \frac{D_o w^2}{2g}} \right) \left( \frac{k_f}{\rho_f c_p D_o / \mu_f k_A} \right)^{1/4} \left( \frac{h_{fg}/c_p \Delta T_s}{D_o w^2/29} \right)^{1/4}
\]

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 \( \left( D_o \frac{\rho_f}{\sigma} \right)^2 \left( \frac{D_o w^2}{29} \right)/2 \). 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.
CHAPTER 3

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 RMS-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
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 receptacle 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](image-url)
Holes \(0.040"\) 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 thermocouple 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.

**Slip Ring Assembly**

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.

Thermocouple Circuits and the Measuring & Recording Instruments

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 thermocouples. 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. 27145. The continuous pen recorder Model 7100B was manufactured by Hewlett 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 superheated 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.
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.
5. Steam temperature.
6. Steam pressure.
7. Speed of rotation of the condenser tube.

All the temperatures were measured using copper-constantan thermocouples. The thermocouple signals from the wires attached to the rotating 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 measurements 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 $\pm 0.10\,^{\circ}F$. 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 $4.5\,^{\circ}F$. Since the temperature is measured to within $\pm 0.10\,^{\circ}F$, 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 $45^\circ F$ to $120^\circ F$ 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 \( \Delta T \) varies from 70 \( ^\circ F \) at zero RPM to 30 \( ^\circ F \) 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, \( \Delta T \) dropped from 90 \( ^\circ F \) to 25 \( ^\circ F \) 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, \( \Delta T \) drops from 110 \( ^\circ F \) to 40 \( ^\circ F \) implying an almost two fold increase in \( h_m \), where, in fact, \( h_m \) increased by almost 400\%.

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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 $\Delta T$.

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

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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 temperature 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 resulting 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 (\( \text{Nu} \)) against Weber number (\( \text{We} \)) as in Figure 15. From Figure 15, it is clearly evident that there exist two distinct regimes. Up to a value of \( \text{We} = 500 \), 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:

\[
\text{Nu} = 11.56 \text{We}^{0.4886}
\]

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 \( \text{We} = 500 \), Nusselt number (\( \text{Nu}_A \)) had a constant value of 120. For \( \text{We} > 500 \), the relation between \( \text{Nu}_A \) and \( \text{We} \) was of the form:

\[
\text{Nu}_A = 6.13 \text{We}^{0.4957}
\]
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 $\text{Nu}_A$ up to a value of $W_e = 500$, none of the other workers observed this. Singer and Yeh reported an initial decline in $\text{Nu}_A$ but this reportedly occurred at values of Weber number well below 100. Another difference is a lack of a decline in $\text{Nu}_A$ for large values of $W_e$ ($W_e > 900$ for Preckshot and $W_e > 1700$ 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 $\text{Nu}_A$ 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 $\Delta T = 185^\circ F$. For $\Delta T = 110^\circ F$, the maximum value of $h_m$ was equal to 380% of its stationary value. Also at $\Delta T = 110^\circ F$, 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).
REFERENCES

CITED REFERENCES


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

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FIGURE 3: DETAIL OF THERMOCOUPLE GROOVES (Scale 1" = 1/2"
AND COOLING WATER TEMPERATURE
PROBE

Section A-A

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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)

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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)
FIGURE 15: VARIATION OF NUSSELT NUMBER WITH WEBER NUMBER

\[ \text{Nu} = 11.56 \text{We}^{0.4886} \]
FIGURE 16: VARIATION OF NUSSELT NUMBER (NUA) WITH WEBER NUMBER

Test Series I
Test Series II
Test Series III

We

$\text{Nu}_A = 6.13 \times 10^{-4}$
FIGURE 17: VARIATION OF Nu_A WITH We. A COMPARISON OF AUTHOR'S RESULTS WITH THOSE OF OTHER WORKERS IN THE SAME FIELD
APPENDIX A

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:

\[ h_m = \frac{Q}{\kappa} = f(w, D_0, k_f, \rho_f, g, \sigma) \]

In the F, L, T, \( \theta \), system of units, the dimensional matrix for the parameters is:

\[
\begin{array}{ccccccc}
1 & 2 & 3 & 4 & 5 & 6 & 7 \\
w & h_m & D_0 & k_f & \rho_f & g & \sigma \\
F & 0 & 1 & 0 & 1 & 1 & 0 & 1 \\
L & 0 & -1 & 1 & 0 & -3 & 1 & -1 \\
T & -1 & -1 & 0 & -1 & 0 & -2 & 0 \\
\theta & 0 & -1 & 0 & -1 & 0 & 0 & 0 \\
\end{array}
\]

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:
\[ \begin{align*}
K2 + K4 + K5 + K7 &= 0 \quad \cdots \quad (1) \\
- K2 + K3 - 3K5 + K6 - K7 &= 0 \quad \cdots \quad (2) \\
- K1 - K2 - K4 - 2K6 &= 0 \quad \cdots \quad (3) \\
- K2 - K4 &= 0 \quad \cdots \quad (4)
\end{align*} \]

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

\[
\begin{align*}
K4 &= -K2 \\
K5 &= -\frac{1}{2}K2 + \frac{1}{2}K3 \\
K6 &= -\frac{1}{2}K1 \\
K7 &= \frac{1}{4}K1 + \frac{1}{2}K2 - \frac{1}{2}K3
\end{align*}
\]

Hence the matrix of solutions is:

\[
\begin{pmatrix}
1 & 2 & 3 & 4 & 5 & 6 & 7 \\
\omega & h_m & D_0 & k_f & \rho & g & \sigma \\
\pi_1 & 1 & 0 & 0 & 0 & -\frac{1}{4} & -\frac{1}{2} & \frac{1}{4} \\
\pi_2 & 0 & 1 & 0 & -1 & -\frac{1}{2} & 0 & \frac{1}{2} \\
\pi_3 & 0 & 0 & 1 & 0 & \frac{1}{2} & 0 & -\frac{1}{2}
\end{pmatrix}
\]

and the dimensionless groups are:

\[
\begin{align*}
\pi_1 &= \frac{\omega \sigma^{1/4}}{\rho_f^{1/4} \sigma^{1/2}} \\
\pi_2 &= \frac{h_m \sigma^{1/2}}{k_f \rho_f^{1/2}} \\
\pi_3 &= \frac{D_0 \rho_f^{1/2}}{\sigma^{1/2}}
\end{align*}
\]
The product \( \pi_2 \cdot \pi_3 \) yields the familiar dimensionless group

\[
\frac{h_m D_0}{k_f} \quad (=\text{Nu})
\]

The product \((\pi_1)^2 \cdot (\pi_3)^3\) yields \( \rho_f \frac{v^2 D_0^3}{\sigma_g} \)

The two resulting groups then are:

\[
\pi_1 = \frac{h_m D_0}{k_f}
\]

\[
\pi_2 = \frac{\rho_f v^2 D_0^3}{\sigma_g}
\]

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

\[
\frac{h_m D_0}{k_f} = f\left(\frac{\rho_f v^2 D_0^3}{\sigma_g}\right)
\]
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 \( \Delta T_w \) 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:

\[
h_m = \frac{q_w}{\Delta T_s}
\]

where \( \Delta T_s \) is the temperature difference between the steam temperature \( T_S \) and the average condenser surface temperature \( T_{OA} \).
The mean value of Nusselt number was then evaluated in two different ways:

\[ \text{Nu} = \frac{h_m D_0}{k_f} \]

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

\[ T_f = \frac{(T_{oa} + T_s)}{2} \]

and

\[ \text{Nu}_A = \frac{h_m D_0}{2} k_A \]

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

Weber number was calculated using the following expression:

\[ \text{We} = \left( \frac{D_0^2 \rho_f}{g} \right) \left( \frac{D_0 v^2}{2 g} \right) / 2 \]

where \( \rho_f \) and \( g \) are evaluated at the mean film temperature \( T_f \).
APPENDIX C

VARIATION OF MEASURED AND CALCULATED PARAMETERS WITH ROTATIONAL SPEED
<table>
<thead>
<tr>
<th>RUN NO.</th>
<th>T1 (°F)</th>
<th>T2 (°F)</th>
<th>T3 (°F)</th>
<th>T4 (°F)</th>
<th>T5 (°F)</th>
<th>T6 (°F)</th>
<th>T7 (°F)</th>
<th>T8 (°F)</th>
<th>T9 (°F)</th>
<th>T10 (°F)</th>
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</thead>
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<tr>
<td>1</td>
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<td>48.20</td>
<td>49.90</td>
<td>49.80</td>
<td>49.75</td>
<td>49.70</td>
<td>49.60</td>
<td>49.45</td>
<td>49.30</td>
<td>49.15</td>
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<td>48.20</td>
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<td>49.60</td>
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<td>49.30</td>
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</tr>
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<td>49.70</td>
<td>49.60</td>
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</tr>
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**TABLE C-1**

(FOOT SERIES I)

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*Barometric Pressure = 29.30", Hg.*

Steam Temperature = 232°F

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PHOTOGRAPH A: COMPOSITE OF ALL THE EQUIPMENT
VITA AUCTORIS

1940  Born in Grbocac, Yugoslavia on July 5.

1961  Finished High School (Herman Collegiate Institute in
       Windsor, Ontario).

1965  Received the Bachelor's Degree of Applied Science in Mechanical
       Engineering from the University of Windsor, Windsor, Ontario.

1966  Candidate for the Degree of Master of Applied Science in
       Mechanical Engineering at the University of Windsor,
       Windsor, Ontario.

1967  Presently employed as a Development Engineer by the Canadian