HEAT TRANSFER BETWEEN FLUIDS

IN TWO PHASE FLOW

by

Allan S. Douglas

Edward L. Tucker

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Signatures of Authors   . . . .

                                         . . . .

Certified by . . . .
                              Thesis Supervisor

Accepted by . . . .

Cambridge 39, Massachusetts
May 19, 1961

Professor Philip Franklin
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Sir:

The thesis entitled "Heat Transfer between Fluids
in Two Phase Flow" is hereby submitted in partial
fulfillment of the requirements for the degree of
Bachelor of Science in Chemical Engineering st the
Massachusetts Institute of Technology.

Sincerely yours,

Allan S. Douglas

Edward L. Tucker

ACKNOWLEDGEMENT

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I.   SUMMARY

Due to the many difficulties accompanying heat transfer between phases in a packed bed, an investigation was made for the direct contact heat transfer between hot air and cold ethylene glycol in a $\frac{1}{2}$" pipe. It was found that the rate of heat transfer increases with increasing mass flow rate of either fluid. This was due to the increased interfacial area caused by the larger degree of turbulence. It was also found that the major portion of the heat transfer occurred in the initial section of the pipe where the flow pattern had not yet become fully developed. Because of the shape of the curves obtained, it is proposed that there is an optimum liquid to gas mass flow rate ratio for two phase flow heat transfer and it is recommended that the data be extended in an effort to determine this ratio.

II. INTRODUCTION

Currently, gas-liquid contacting for a great variety
of chemical operations is achieved in packed beds and
towers. There are several major problems however which
make it desireable to develop an alternative method of
contacting. One problem is the difficulty of obtaining a
uniform distribution of liquid and gas to provide the max-
imum contacting area for the operation. An effect often
accompanying the cooling of hot gases is the deposition
of previously sublimed solids on the bed. This requires
frequent dismantling, cleaning, and repacking of the bed.
A typical example of this is the cooling of sulfur dioxide
gas from the reaction

$$S + O_2 \rightarrow SO_2 \qquad .$$
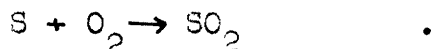
The hot product gas is usually cooled by contacting it
with 98% sulfuric acid in a packed tower. This corrosive
liquid is used to absorb any sulfur trioxide which may
have formed in the reaction. Solid sulfur is deposited
in the bed and due to the nature of the cooling liquid it
is difficult to clean the apparatus.

Several suggestions have been made to alleviate this
problem. One is the contacting of the two fluids in a
horizontal pipe. The contact must be direct to allow for
the absorption of any sulfur trioxide and to allow any
deposited solids to be swept out of the system.

The heat transfer in this two phase system depends
upon the type of interface between the two fluids. It
is also dependent upon the relative velocities of the
streams and their physical properties. Before considering
the heat transfer possibilities, it is necessary to have
a thorough understanding of the mechanics and nature of
two phase flow.

Most of those who have investigated two phase flow
in pipes have directed their efforts towards obtaining
empirical correlations relating the pressure drop in the
two phase system to the pressure drop in the single phase
flow of the component fluids. The experiments have main-
ly been performed in horizontal pipes under isothermal
conditions.

The basic correlations are those found by Lockhart and
Martinelli (13) in their experiments using a water-air
system in small diameter pipes (less than 1" I.D.) and at
pressures up to fifty pounds per square inch. Four types
of flow were assumed to exist:

        a) viscous-viscous,

        b) viscous-turbulent,

        c) turbulent-viscous, and

        d) turbulent-turbulent

referring to the liquid and gas flows respectively. Turbu-
lent flow in the two phase system was expected to occur
if the fluid Re>2000 when flowing alone in the pipe.

The turbulence is assured since, upon introduction of the other fluid into the pipe, the hydraulic diameter is necessarily decreased. Lockhart and Martinelli assume laminar flow in the two phase system if the fluid has Re 1000. Therefore a chart of Re versus flow category would appear as follows:

	t-t	v-t	t-v	v-v
Re_l	>2000	<1000	>2000	<1000
Re_g	>2000	>2000	<1000	<1000.

This chart, coupled with the assumptions that the static pressure of the two phase system is the same as the static pressure of each phase (the pressure drop across the pipe for the liquid is equal to the pressure drop across the pipe for the gas) and that a constant volume relationship holds, stating that the volume of the liquid plus the volume of the gas at any instant of time equals the total volume of the pipe, enabled them to plot from experimental data the pressure drop for the two phase system as a function of the pressure drop in the single phase system:

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \phi_g^2 \left(\frac{\Delta P}{\Delta L}\right)_g \qquad \text{and} \tag{1}$$

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \phi_l^2 \left(\frac{\Delta P}{\Delta L}\right)_l \tag{2}$$

where ϕ is the empirical coefficient relating the two pressure drops. Letting $\phi_{l(g)} = f(x)$

where

$$X = f\left[\frac{\dot{m}_l}{\dot{m}_g}, \frac{\rho_l}{\rho_g}, \frac{\mu_l}{\mu_g}, D\right] \tag{3}$$

they were able to find a relationship between X and R', the
fraction of pipe filled by either liquid or gas. In equa-
tion (3) $\dot{m}$ is the mass flow rate, μ is the viscosity, ρ is
the density, and D is the diameter of the pipe.

Lockhart and Martinelli based their correlations only
on the four flow categories and not on the various flow
patterns. Alves ($\underline{1}$), by use of a glass test section,
observed these seven distinct flow patterns (Fig. 1):

1) Bubble flow: the gas moves along the upper part
   of the pipe in bubbles travelling at approximately
   the same velocity as the liquid,

2) Plug flow: alternate plugs of liquid and gas move
   along the upper portion of the pipe,

3) Stratified flow: gas flows on top of the liquid,
   a smooth interface appearing between the two phases,

4) Wavy flow: identical to stratified flow with the
   exception of a wavy interface travelling in the
   direction of flow,

5) Slug flow: a wave is picked up periodically by the
   more rapidly moving gas to form a frothy slug which
   passes through the pipe at a velocity greater than
   the average velocity of the liquid, unsteady pressure
   drops existing,

6) Annular flow: a film or layer travels on the in-
   side wall of the pipe creating an annulus through
   which the gas flows at a high velocity (the gas is

Gas

Liquid

Stratified

Gas

Liquid

Annular

Liquid

Plug

Gas

Liquid

Slug

Liquid
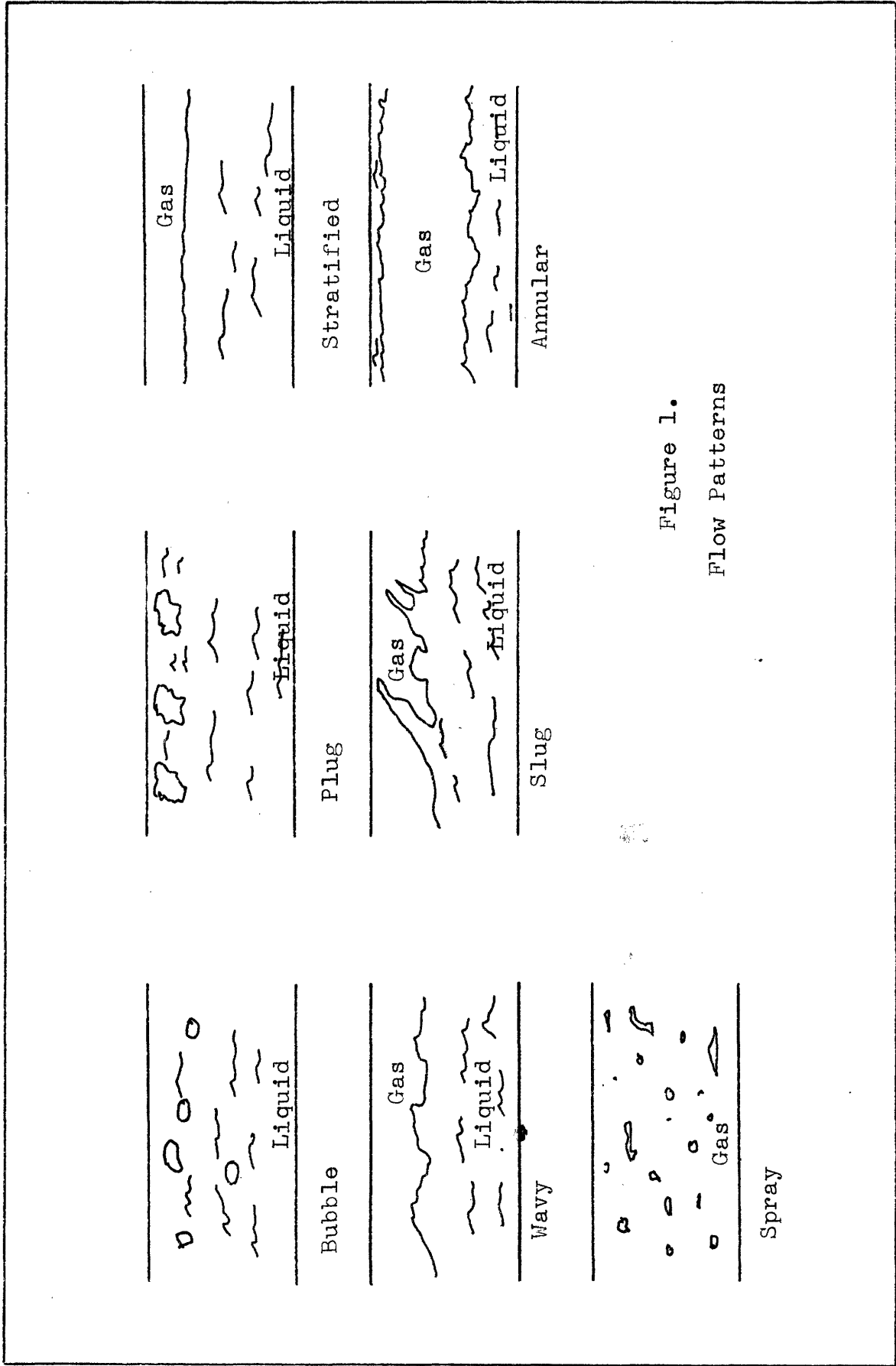
Bubble

Gas

Liquid

Wavy

Gas

Spray

Figure 1.

Flow Patterns

actually a central core surrounded by the liquid), and 7) Spray flow: nearly all the liquid is entrained by the gas as a spray.

Theoretical analyses of the fluid dynamics of the system have not been attempted for all flow patterns, but numerous attempts have been made to analyze annular flow utilizing the von Karman universal velocity profile equations, solving the Naviar-Stokes equations, and extending Nusselt's theory for laminar film flow. These theoretical correlations have been compiled by the Purdue University Jet Propulsion Center (6) to which the reader is referred. Thus far, theoretical approaches appear to apply only to ideal situations (either there is a smooth interface between the liquid and the gas in annular flow or the boundry oscillates in a defined manner). That these conditions are never approached is apparent when observing a flowing two phase system. No single pattern is ever observed, but a second pattern is superimposed upon the main pattern.

It is noted that experimenters used a water-air system which easily led to entrainment and vaporization of the liquid. Usually the latter was unnoticed or not considered in the overall relations since the amount of vapor was supposedly not of a sufficient quantity to substantially change the volumes of the liquid or the gas.

Another aspect of two phase flow that has been investigated is heat transfer to a two phase liquid-gas

system flowing in a pipe. This work was undertaken by Veschoor and Stemerding (19). By varying the mass flow rates of gas with respect to a given flow rate of liquid in a pipe surrounded by a steam jacket, they were able to obtain an effective heat transfer coefficent, h, for the two phase system as a function of the ratio of volumetric flow rate of gas to volumetric flow rate of liquid. Doing this for a number of liquid flow rates, they plotted the following family of curves:

volumetric flow ratio, gas/liquid ⟶
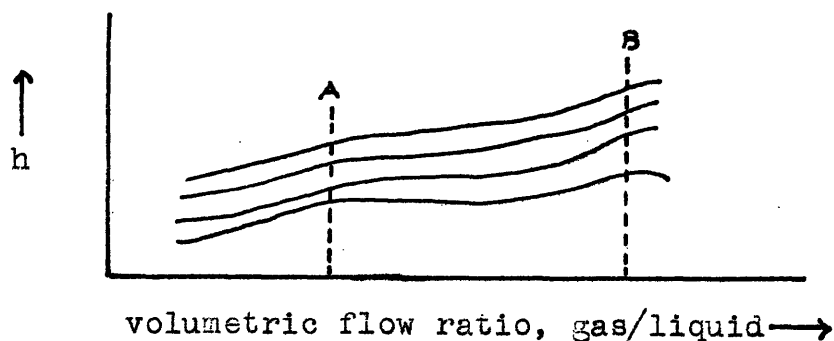
Comparison with the work done on pressure drop theory and flow patterns show that the change in slope in the curves at A corresponds to the transition range from bubble to slug flow and the maximum at B corresponds to the transition range from slug to annular flow. Of importance is the decrease in the heat transfer coefficient as annular flow proceeds. This is apparently caused by the decreased interfacial area due to the disappearance of frothing and the appearance of a relatively smooth and well defined interface.

Reed, Matiatos, and Wegener (16) in a 10.27 Chemical Engineering Laboratory Report at the Massachusetts

Institute of Technology seem to be the first to have
investigated the cooling of a hot gas by direct contact
with a cold liquid by flowing them cocurrently in a pipe.
Their results for annular flow show that the heat trans-
fer coefficient decreases as the mass flow rate of gas
increases for a constant liquid flow rate. However,
due to the difficulties in their procedure, the results
are qualitative rather than quantitative and appear
below:
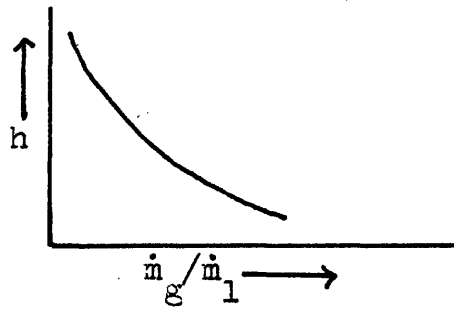
Further work on the same subject was performed by
Emeneger and Bing (3) at M.I.T. Using similar apparatus
and a water-air system, they were able to calculate the
quantity (hA) for flow in the slug and annular regions.
They found that at constant liquid flow rate, increasing
the gas flow rate caused a marked increase in (hA) while
increasing the liquid flow rate at constant gas flow rate
did not appreciably affect the heat transfer. This effect
is caused by the increased area for heat transfer due to
the creation of a greater degree of turbulence at the
interface.

It seems desireable to extend the last-mentioned in-
vestigation with the ultimate goal of obtaining a

correlation for predicting the effective heat transfer
coefficient for heat transfer between fluids in two
phase flow in a horizontal pipe. Due to practical
limitations in the system, this investigation is con-
cerned with the determination of (hA) as appears in the
following set of heat balances:

$$Q = \dot{m}_g c_{pg} (\Delta T)_g = \dot{m}_l c_{pl} (\Delta T)_l \tag{4}$$

and $$Q = (hA) \Delta T_{lnm} \tag{5}$$

assuming no heat is lost from the system and no vaporiza-
tion of the liquid takes place, where m is the mass flow
rate of the fluid, c_p is the heat capacity of the fluid,
T is the difference between the initial and final tempera-
ture, and T_{lnm} is the logarithmic temperature difference
for the cocurrent heat exchanger. The area A is the
interfacial area between the phases. It cannot be measured
and hence is combined in (hA) as the effective heat trans-
fer coefficient.

The apparatus (Fig. 2) consisted of two interchangeable
test sections 48" and 8 " in length which were mounted in a
permanent assembly containing an air flow meter, a liquid
pump and a tee fitting for the introduction of air. The
liquid employed was ethylene glycol. The temperatures of
the inlet and outlet streams were measured by means of
thermocouples placed in the center of the respective streams.
A 4" stovepipe with a sheet metal baffle inside was used to
separate the streams. At high gas velocities, however, a
large amount of spray went around the baffle and escaped
through the top of the separator. Before entering the test
section the liquid was passed through a cooling coil to
remove from it the heat gained in the previous pass through
the system.

The experiment was performed in three separate parts.
First the air flow rate was determined by means of a calibrated
orifice and the liquid flow rate by the stopwatch and bucket
method. After setting the air inlet temperature at the
desired level (60-80°C) the temperatures of the inlet and
outlet streams were measured. Due to the extremely short
residence time in the test section and the unsteady state
conditions which prevailed, it was necessary to employ a
time study method. The four necessary temperatures, air
inlet, liquid inlet, liquid outlet, and air outlet, were
read in succession at ten second intervals. In general six

Figure 2.

Apparatus

T_{go}

T_{lo}

Liquid-Gas
Separator

T

T

Pressure
Gauge

T

T

Air Heater

Air Orifice

Test Section

T

T_{li}

T_{gi}

Liquid Reservoir

Gear Pump

Liquid Cooling Unit
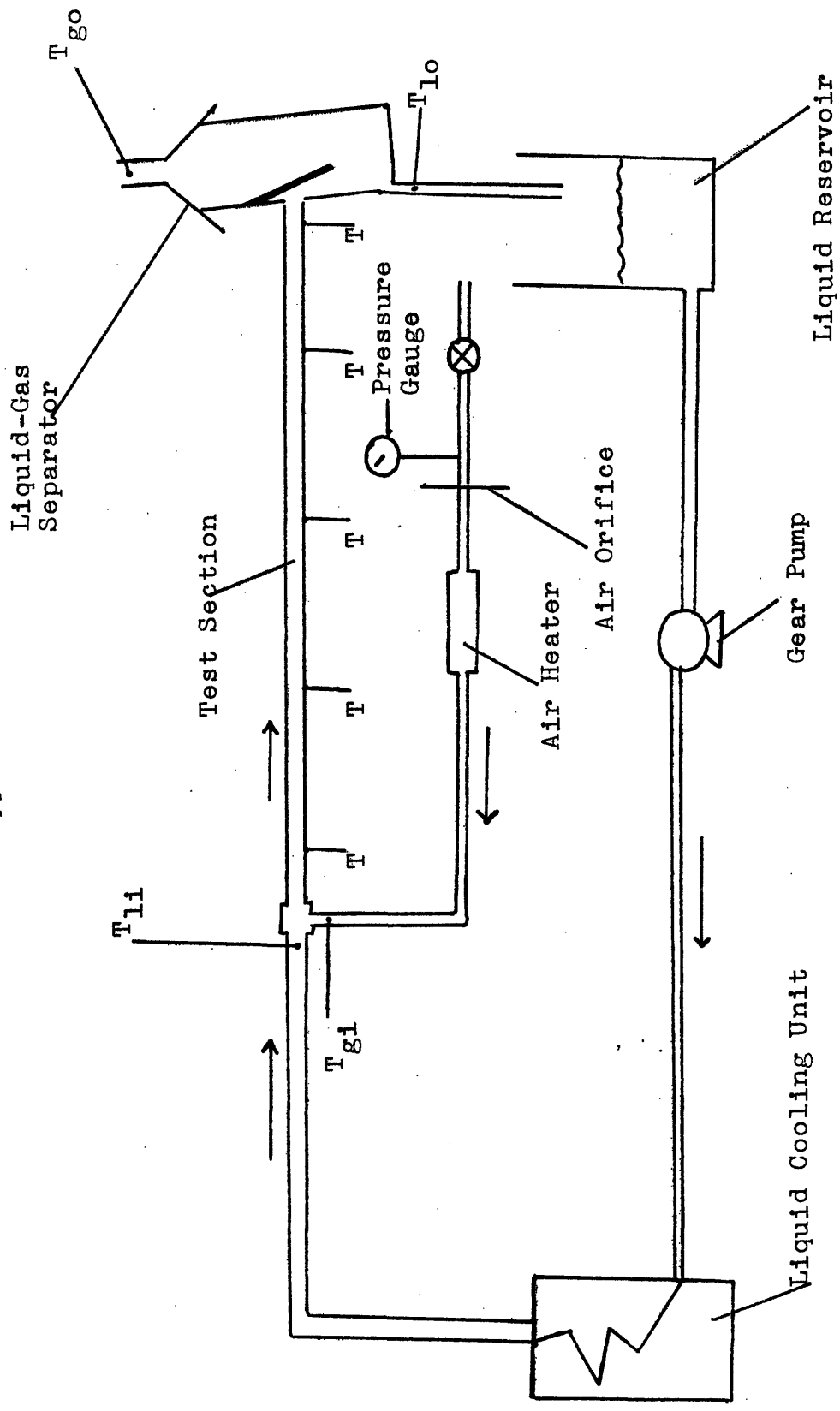
readings of each temperature were made. The results were
plotted as a function of time so that a reading at any
given time would provide the four necessary temperatures
for the calculation of a heat balance. Finally an attempt
was made to determine the temperature of the liquid as a
function of distance downstream from the initial mixing
zone. This was done reading thermocouples placed along the
bottom of the test section and positioned so that the ex-
posed thermocouple was flush with the inside pipe wall
(Fig. 3). These thermocouples were placed at approximately
eight inch intervals so that the temperature profile could
be determined.

Figure 3.

Thermocouple Emplacement
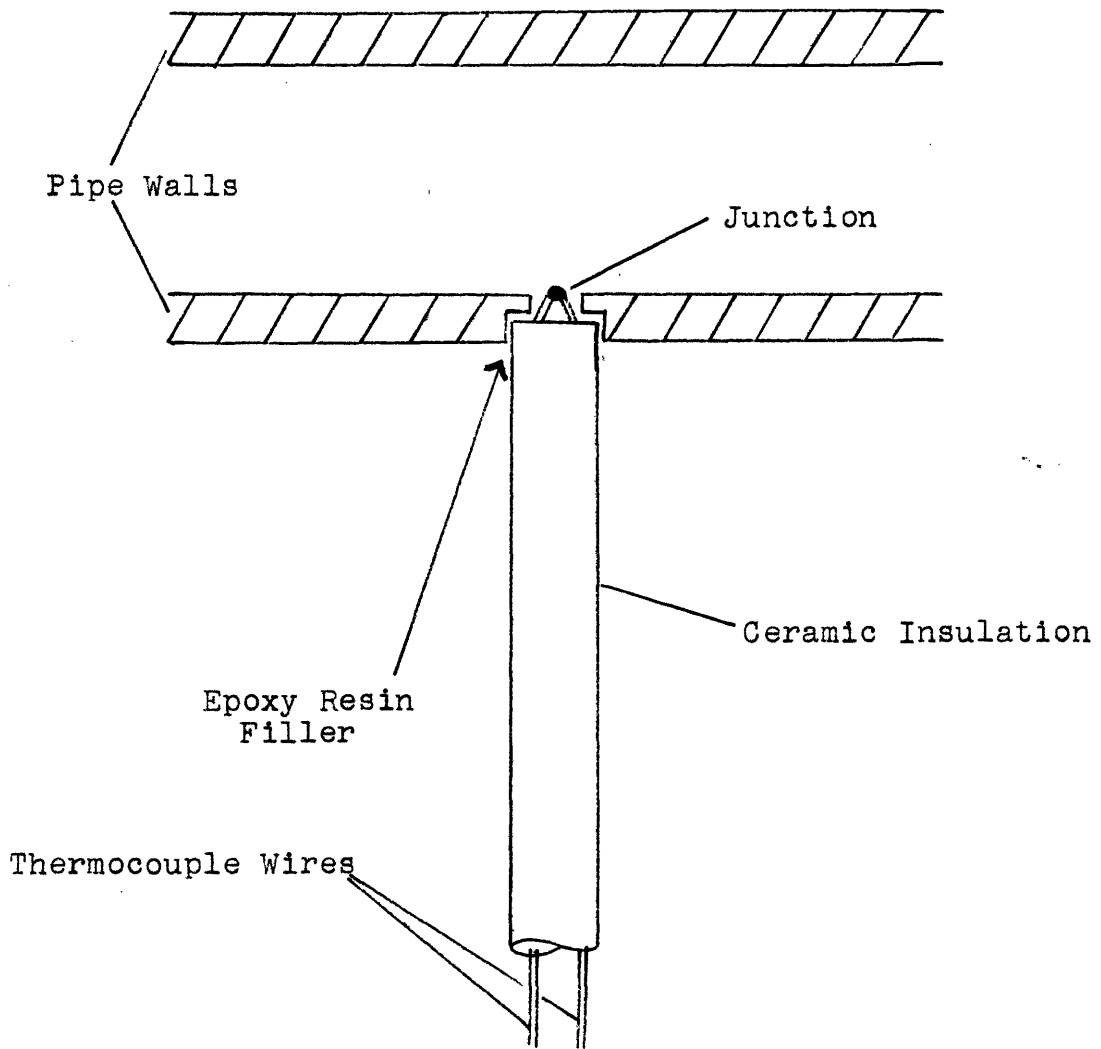
IV. RESULTS

The results of this investigation are shown on graphs
one and two. The effective heat transfer coefficient is
a function of both gas flow rate and liquid flow rate
and increases with an increase in either stream. At low
gas flow rates, the effect of increasing liquid rate is
less than at high gas flow rates. The curves on the graph
indicates the relation of the effective heat transfer
coefficient (hA) to the liquid to gas flow rate ratio at
constant gas flow rate.

Graph 1.

Effective Heat Transfer Coefficient versus Ratio of Mass Flow Rates (Gas mass flow rates are indicated in parentheses)

(A (8") Section)
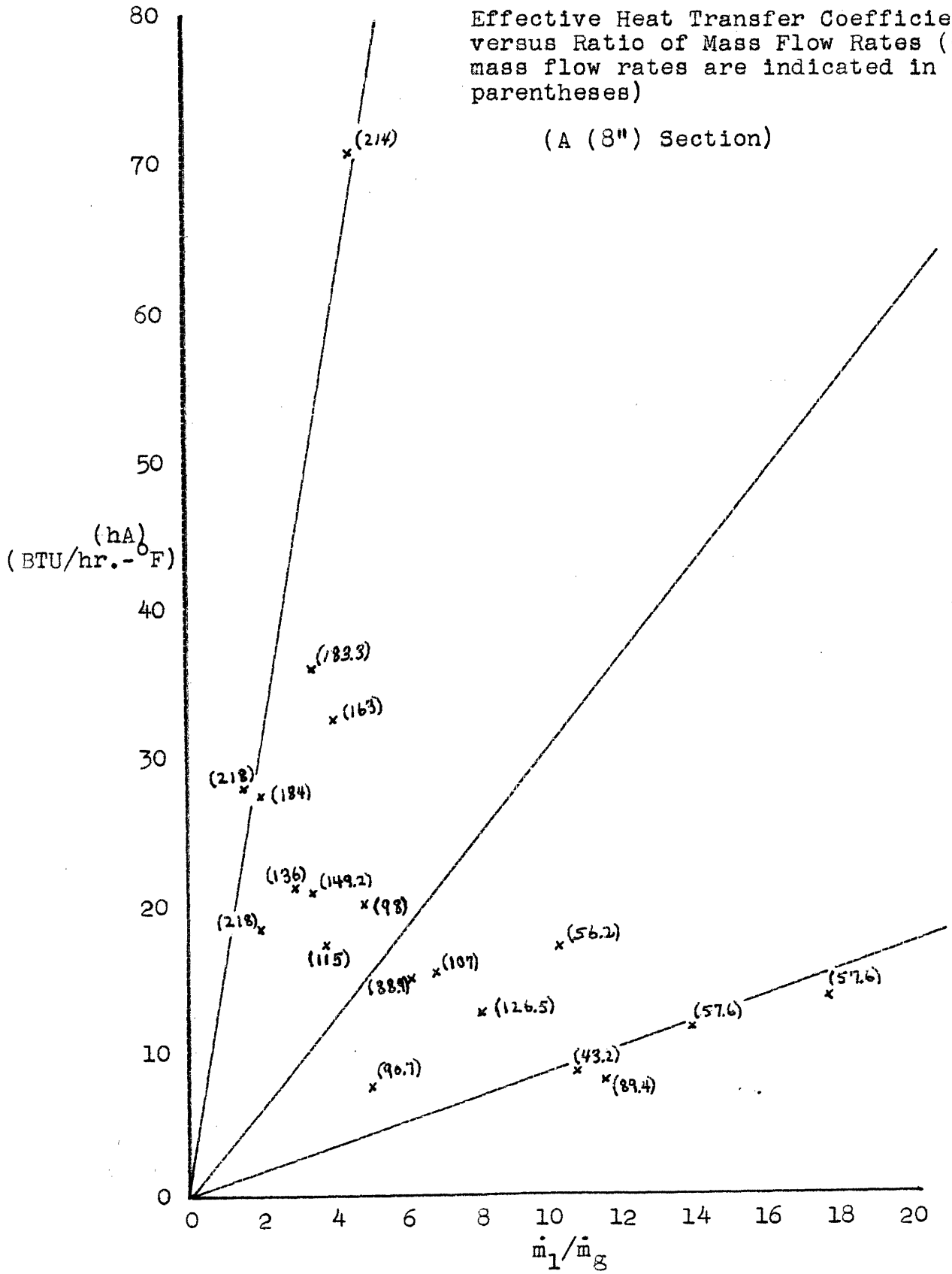
Graph 2.

Effective Heat Transfer Coefficient
versus Ratio of Mass Flow Rates (Gas
mass flow rates are indicated in
parentheses)

(E (48") Section)

The analysis of the results obtained in this investigation
will be facilitated by consideration of the three separate
regions in which heat transfer may occur: 1) the initial
mixing zone, 2) the test section, and 3) the liquid gas
separator.

The amount of heat transferred in the initial mixing
zone is a function of the ratio of liquid to gas mass
flow rates and the method of introducing the gas to the
liquid stream. Both of these factors dictate the degree
of turbulence occuring in this zone. Two methods were used
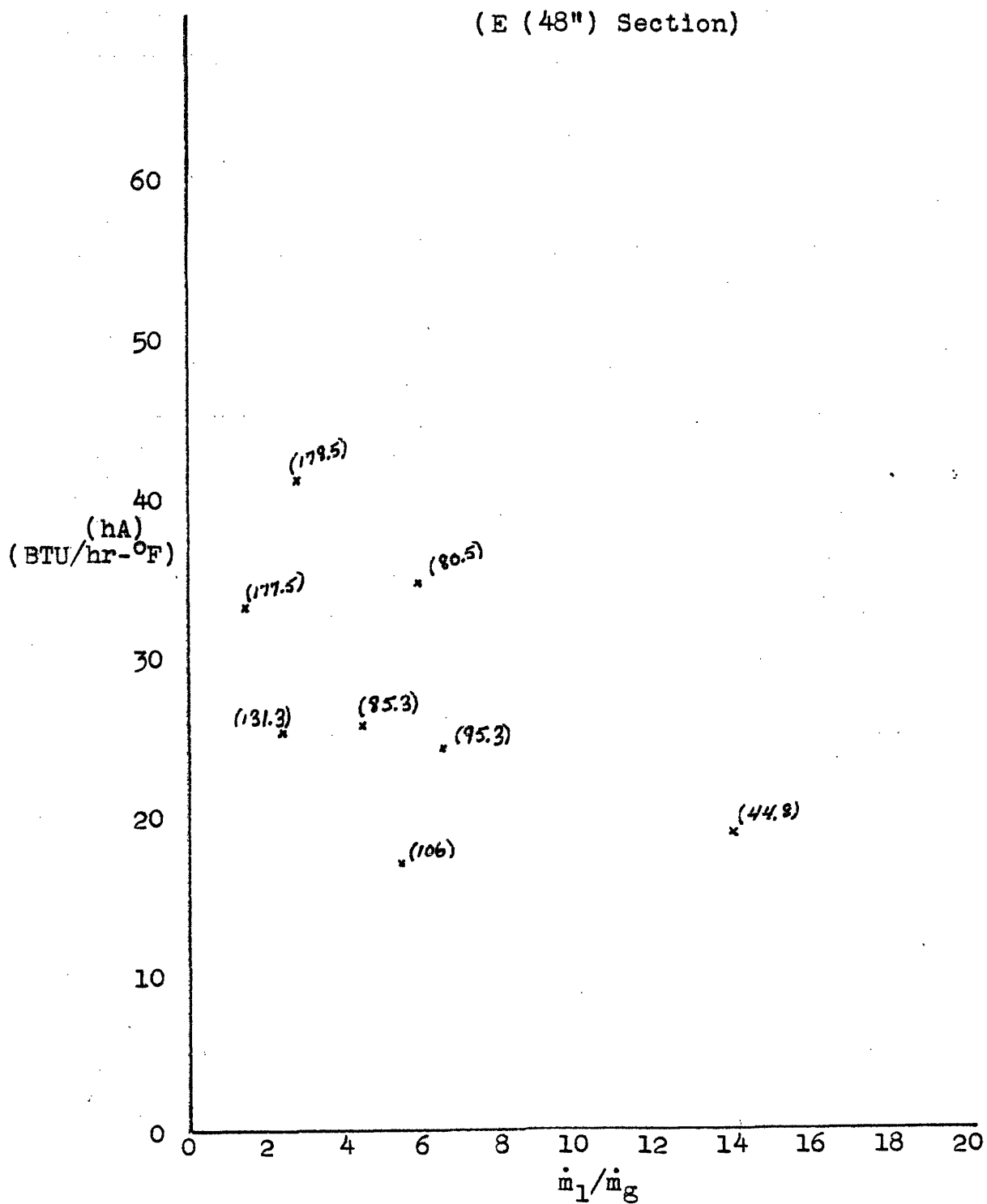to initiate two phase flow. The first, introduction of
the gas to the liquid stream by means of a copper tube
located in the center of the liquid stream, had to be
discarded even though this type of entry offers the least
turbulence. Sufficient mass flow rates of gas could not
be obtained because of the high pressure drop caused by
the small crossection of the tube. An alternate method,
although less desireable than the first from a turbulence
viewpoint, was used for the permanent apparatus. The gas
entered the system at a right angle to the continuous
liquid stream through a tee fitting as seen in figure
two. More heat is transferred in the resulting turbulent
flow since the interfacial area between the two phases is
greatly increased. It is desired to limit heat transfer
to the test section.

By means of a three-quarter inch inside diameter plastic
section in place of the regular test sections, flow patterns
were observed at all possible liquid to gas mass flow rate
ratios. Fully developed flow was found to exist only after
approximately one foot downstream from the initial mixing
zone. In the undeveloped flow, the walls of the pipe were
continually covered with liquid, but much turbulence was
seen in the center of the pipe. If this applies to the test
section (0.622"I.D.), flow would not be developed in any
of the runs in the A section which is eight inches in length,
but rather the liquid and gas would enter the separator in
a turbulent state. Fully developed flow was accompanied
by a marked decrease in turbulence and the existence of a
semi-annular type pattern, liquid covering the pipe walls,
and gas moving through the center. The liquid layer in
the developed pattern was considerably thicker on the
bottom than on the upper parts of the pipe walls. Thus
the interfacial area between the two phases is substantially
decreased once developed flow begins. For the most part,
the interface between the gas and the liquid on the lower
part of the pipe was not precise but wavy. Periodic slugs of
liquid entrained in the gas interrupted the flow, thus
destroying the interface. The frequency of these slugs
increased with increasing gas mass flow rates. At some
points the pattern seemed to revert from annular to spray
and back to annular. (By use of an experimental graph

relating flow conditions to flow patterns (Graph 3), it
was found that operations were very near the annular-spray
boundary.) This changing of patterns might be due to the
fluctuations of the gas supply system. The liquid on the
upper part of the pipe was rippled, this being caused by
the movement of the gas through the center and the simul-
taneous draining of the liquid off the walls and the
depositing of liquid from the spray in its place. Increasing
mass flow rates caused the upper part of the annulus to become
thinner but it was present over all conditions observed.

As the two phases enter the separator, they strike
the baffle shown in Figure 2. Again intimate mixing briefly
takes place and if the two streams are not in thermal
equilibrium, heat will be transferred. Once the baffle
is struck, most of the liquid proceeds downward towards
the liquid drain. A small amount of liquid holdup in the
separator enables the temperature of the liquid to become
equilibrated, hence the outlet temperature is the average
temperature of the exit liquid stream. Meanwhile the gas
flows upward separating itself from the liquid and exits
at the top of the separator. Accompanying the exit gas
is liquid in the form of spray caused by the turbulence.
Because of this entrained (unseparated) liquid the
thermometer is continually covered with liquid and is
reading essentially the liquid temperature and not that
of the gas. Thus the experimental value for the exit

Graph 3.

Flow Regimes as a Function
of Operating Conditions

$\dot{m}_g/a$ (y-axis)

$\dot{m}_l/\dot{m}_g$ (x-axis)

Spray Flow

Annular Flow

Wavy Flow

Slug Flow

Stratified Flow
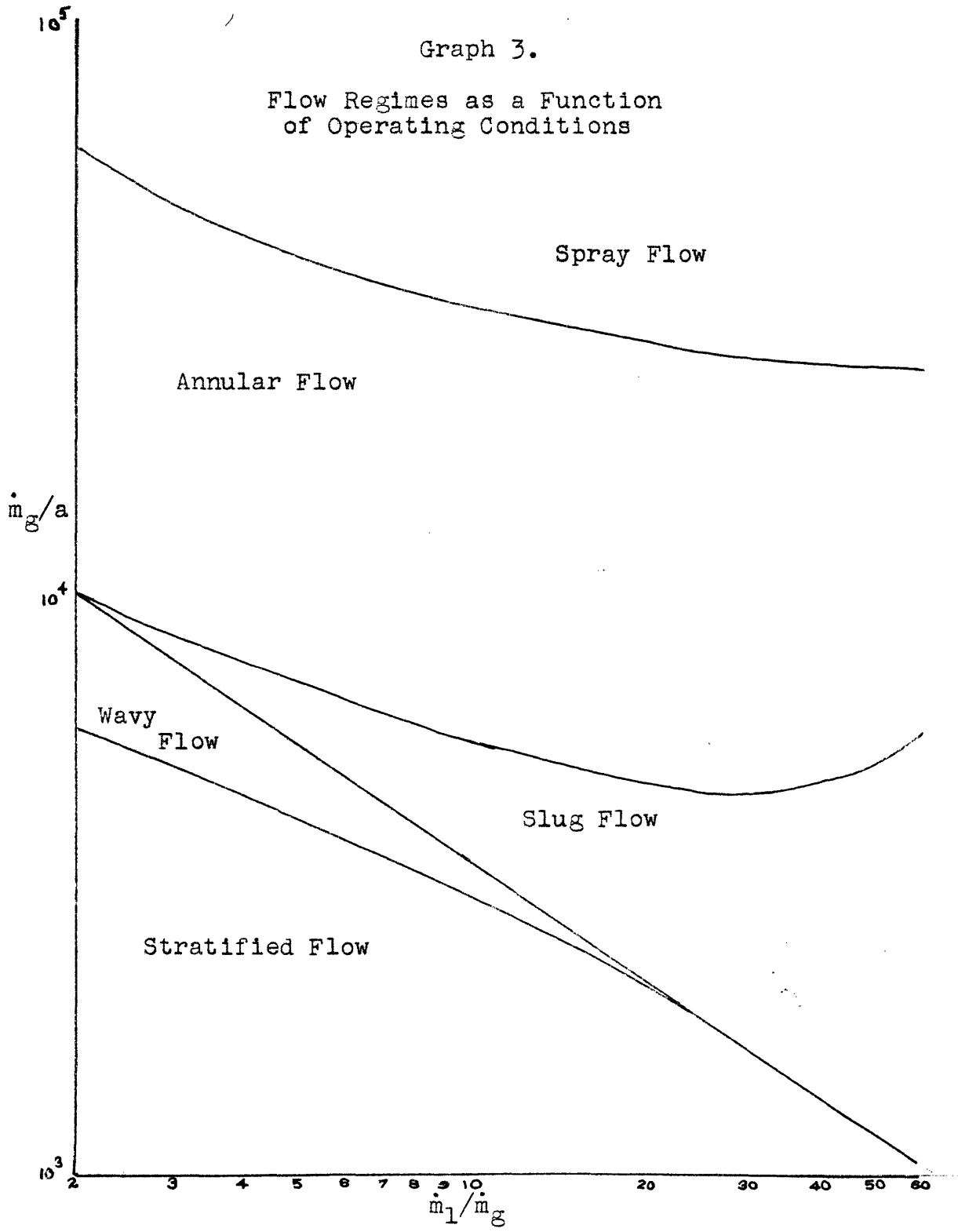
gas temperature could not be used and calculated values were required. This was done on the assumption that all the heat given up by the gas was picked up by the liquid phase and is substantiated by the following:

1) An overall heat transfer coefficient for the system, determined by a heat balance on hot gas flowing alone, yields a heat leak to or from the system of usually less than 2% of the total heat transferred. The difference between the temperatures of the room and the system did not exceed $7^{\circ}C$ and for the most part were substantially less than that.

2) Essentially no liquid was vaporized in the heat transfer operation, the vapor pressure of Ethylene Glycol being less than 1 mm Hg. at $20^{\circ}C$.

Therefore assuming the energy balace to be valid, the exit gas temperature can be calculated by

$$\dot{m}_l c_{pl} \Delta T_l = \dot{m}_g c_{pg} \Delta T_g \qquad (4)$$

In the calculation of the gas exit temperature it was found that either of these two conditions were present: 1) T air exit was greater than T liquid exit, or 2) T air exit was less than T liquid exit. In the first case, the calculation was used as such while in the latter case, T liquid exit was substituted for the calculated value since such a condition could not exist. This discrepancy may be due to several effects:

1) The air supply was not constant but oscillated

over a static pressure range of 4-6 psi and a difference

in pressure drop across the orifice of up to four

centimeters. Calculations were based on mean values.

2) As the static pressure changed, the liquid mass

flow rate would also change. The higher the static

pressure, the lower the liquid flow rate for a given

pump by-pass valve setting.

3) The temperature difference for the liquid inlet

and outlet streams ranged from approximately $1.0^{\circ}C$ to

$2.5^{\circ}C$, which could have magnified the effect of a small error.

Consistency for the temperature rise of the liquid was

better than expected. This was perhaps due to the time

study method that was used in obtaining and evaluating

the data. Since steady state could not be obtained

(inlet temperatures for both streams were not constant

over time), the readings were taken at defined time

intervals, plotted on a graph, and the best curve drawn

through the points (Graph 7). At a particular time, all

the temperatures were used for calculating the energy

balance. Thus this type of calculation has a tendency

to average out the errors present in the recording of

a single point.

Using the calculated value for the heat transferred

to the liquid stream and the inlet and outlet temperatures,

an overall heat transfer coefficient (hA) was calculated by

use of the equation

$$\dot{m}_l c_{pl} \Delta T_l = hA \Delta T_{lnm} \qquad (5)$$

Due to the nature of the system it was impossible to
determine experimentally the area for heat transfer, so the
results are reported as (hA).

The results show that for a given set of flow conditions
the calculated value of (hA) is higher for the long(48")
section than for the short(8") section. However (hA/L),
the heat transfer coefficient per unit length is higher
for the short section. This would seem to indicate that a large
fraction of the heat transfer is occuring in the inlet region
of the pipe. This is to be expected because in the inlet
region flow has not become fully developed and there is
therefore a larger interfacial area for heat transfer
than in the part farther downstream.

It should be noted in this connection, however, that
the data indicate that in the long pipe over half the runs
reached essentially equilibrium while none reached
equilibrium in the short section. There are two possible
causes for this and further refinement in experimental
technique will be necessary to determine which is the
actual situation here. One possibility is that since
equilibrium is not reached in the 8" section, the additional
heat transfer to reach equilibrium with the 48" section
occurs in the undeveloped flow region with essentially nothing
happening in the developed region. The second possibility
is that heat transfer occurs in the entire pipe, both in
undeveloped and in developed flow. The actual effect
may be determined by employing a test section the same

length as the undeveloped flow region and comparing the
results with those of the 48" section.

An attempt was made to do this by placing thermo-
couples at the wall of the pipe at fixed intervals and
measuring liquid temperatures as a function of length (Fig. 3).
However all of the thermocouples, even those closest to the
entrance, gave the same reading as the thermocouple
recording the liquid exit temperature. No explanation
of this effect has been found.

Graph 1 shows (hA) as a function of the liquid to gas
flow rate ratio in the short section. It may be seen from the
graph that at constant flow rate ratio increasing the
gas flow rate (therefore also the liquid flow rate) causes
(hA) to rise considerably. At constant gas flow rate,
increasing liquid caused (hA) to increase. The magnitude
of this increase is a function of the gas flow rate. High
gas flow rates caused (hA) to increase rapidly while at
low gas flow rates (hA) increased only moderately. This
increase is due to the increased interfacial area caused
by the increased turbulence of mixing. Obviously at low
gas flow rates the increase in turbulence will be less
than at high flow rates.

Consideration of the relationship between (hA) and
$\dot{m}_l/\dot{m}_g$ will show that as $\dot{m}_l/\dot{m}_g$ goes to infinity (i.e. the
gas flow rate becomes zero or the liquid flow rate becomes
infinite) the interfacial area for heat transfer goes to

zero and (hA) becomes zero. Although the curves for
constant gas flow rate (Graph 1) obtained are linear and
of positive slope, to fulfill the condition of (hA)
approaching zero as the mass flow ratio approaches infinity,
it is necessary that there be a maximum and then change
of slope in the curve. This indicates that there is an
optimum value of flow rate ratio for direct contact heat
transfer. Further investigation is necessary to determine
this ratio.

VI. CONCLUSIONS

1) The effective heat transfer coefficient (hA) is a function of both liquid flow rate and gas flow rate and increases with an increase in either.

2) At a constant gas flow rate, the effect of increasing liquid flow rate on (hA) is greater at high than at low gas flow rates.

3) The effective heat transfer coefficient is a function of the degree of turbulence in the entrance section.

4) The major part of the heat transfer occurs in the region before the flow pattern becomes developed.

VII. RECOMMENDATIONS

1) The data should be extended to determine the value of the mass flow rate ratio for optimum heat transfer.

2) The effect of pipe length on the degree of heat transfer should be investigated.

3) The distance downstream from the initial mixing zone where thermal equilibrium is reached should be determined.

VIII. APPENDIX

VIII. APPENDIX

A. SUPPLEMENTARY DETAILS

1. Apparatus

The air flow meter consisted of a sharp-edged orifice plate (d = 0.307") in a 3/8" pipe line fitted with radius taps. A mercury manometer was used to measure the pressure drop across the orifice and the upstream pressure was measured with a pressure guage. The orifice was calibrated by using the heating coil placed in the air supply line for the experimental work. The coil was constructed of $9\frac{1}{2}'$ of Chromel A heating wire (20 gauge, R = 0.6347 ohms/foot) placed inside a 2' long, 25 mm. pyrex glass tube. The glass tube was wrapped with aluminum foil to eliminate heat loss by radiation. The voltage drop across the coil was controlled by a Variac and knowing this value, the air flow rate (lbs./hr.) was calculated from the equation

$$i^2 R = \dot{m}_g c_{pg} \Delta T_g = KE^2/R \qquad (6)$$

where K is a proportionality constant = 3.41 BTU/hr./watt. The data was then corrected to one atm. upstream pressure and presented as the calibration curve (Graph 4).

The temperature of the air leaving the heater was measured by means of a thermocouple placed in the center of the pipe. The air was introduced into the test section by a tee fitting. While it is felt that this caused an undesireable amount of turbulence, the only other possible system, that of causing the air to flow through a piece

Graph 4.
Air Orifice Calibration

ΔP
(cm. Hg)

$\dot{m}_g$ (1bm./hr.)
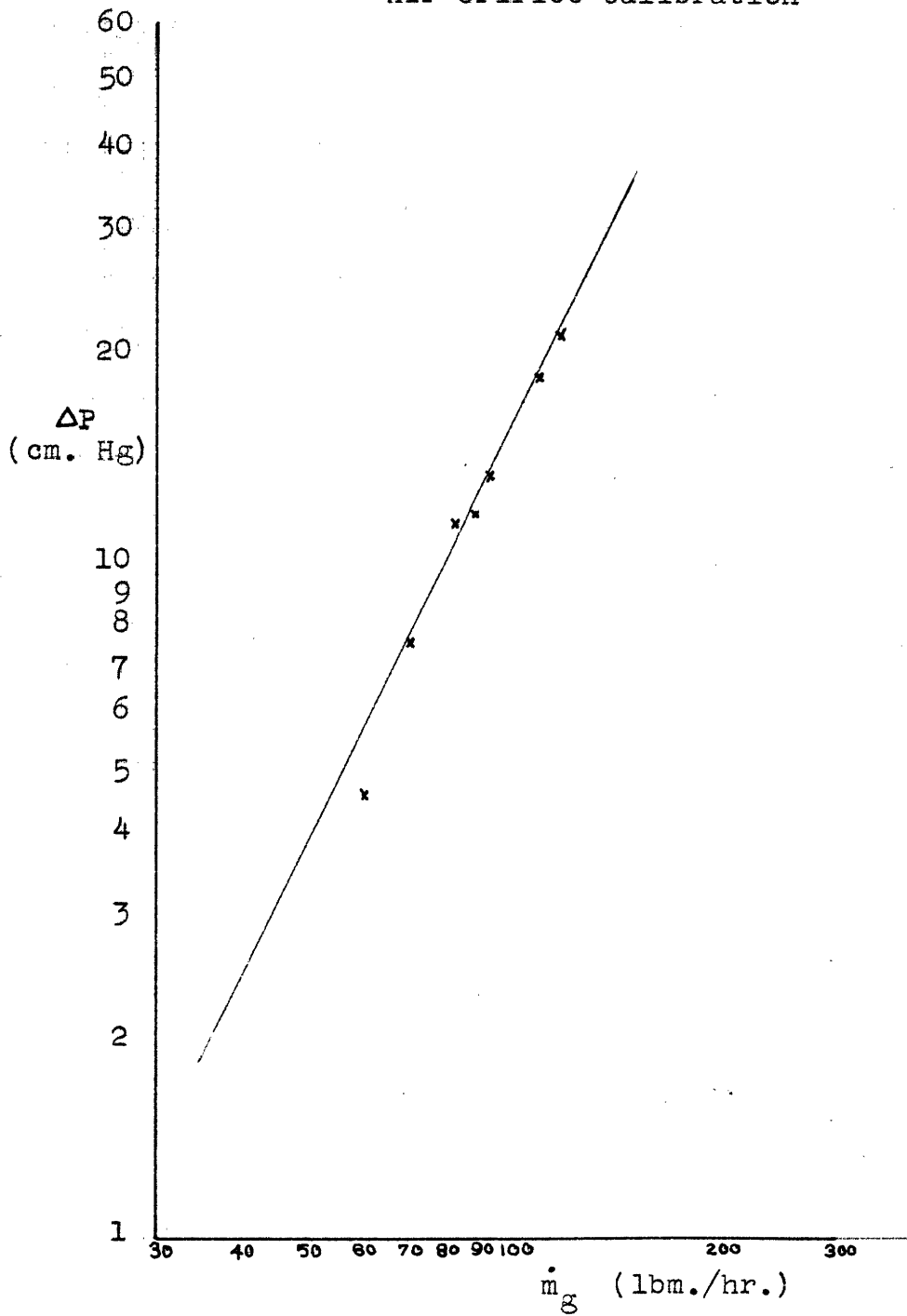
of copper tubing which was bent in a right angle to release
the air in the direction of flow, presented such a restriction
that the air flow rate was greatly reduced, and it was
necessary to abandon this.

The liquid (ethylene glycol) started in a five gallon
reservoir from which it was pumped by means of a gear pump
through a copper coil imersed in an ice bath. From there
it passed through a pipe where its temperature was measured
with a thermocouple, and it then entered the test section.

Two interchangeable test sections were employed, 48"
and 8" long to investigate the effect of length on the
heat transfer. The test sections had thermocouples
embedded in the walls so that the liquid temperature could be
measured as a function of length in the pipe. These were
positioned by boring holes at the desired places and
inserting the thermocouple to the desired depth. They
were held in place with epoxy resin.

Seperation of the two streams was done in a stovepipe
(d = 4") one foot long. On the bottom was a stovepipe
cap which had two holes drilled in it and copper tubing
fitted for liquid exit streams. A five inch funnel was
placed on top of the stovepipe in an effort to eliminate
spray. A piece of sheetmetal was spot-welded and soldered
at an angle within the separator as shown in Figure 2.
This was also to eliminate spray.

The liquid flow rate was controlled by means of a
bypass on the pump. This consisted of a return line

from the pump to the reservoir fitted with a valve. By adjusting this valve, the flow through the apparatus could be controlled. The temperature of the liquid leaving the separator was measured by a thermocouple placed in the liquid exit stream and the air exit temperature was measured with a thermometer.

The thermocouples were copper-constantin and the voltage was read on a potentiometer with a built in room temperature compensator. The accuracy of the potentiometer was verified by comparing the temperature of a thermometer with the value given by the temprature-voltage calibration curve (Graphs 6&5) (9).

Graph 5.
Thermocouple Calibration
Copper-Constantan

Voltage (mv.)

T (°C)
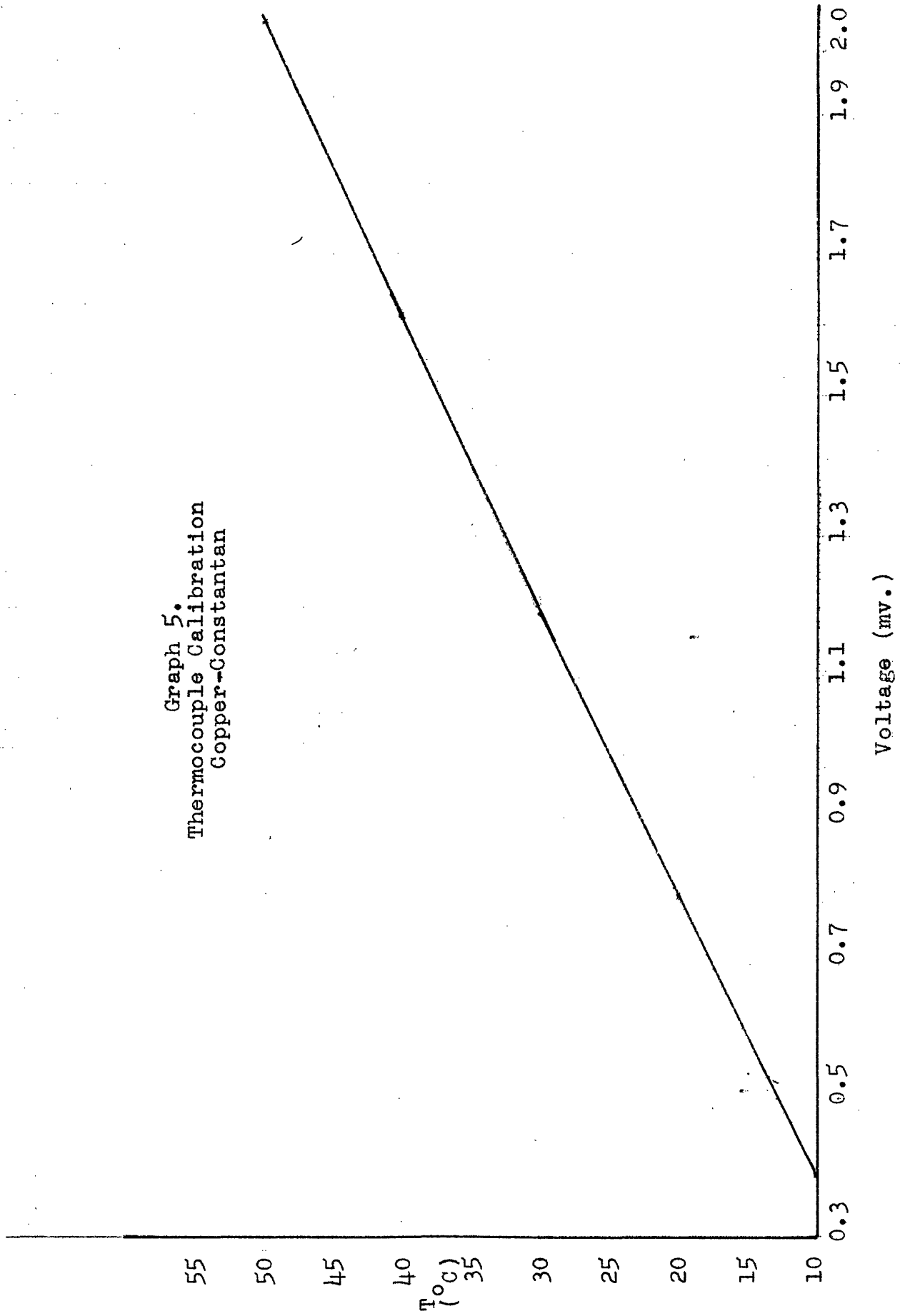
Graph 6.

Thermocouple Calibration
Copper- Constantan

Voltage (mv.)
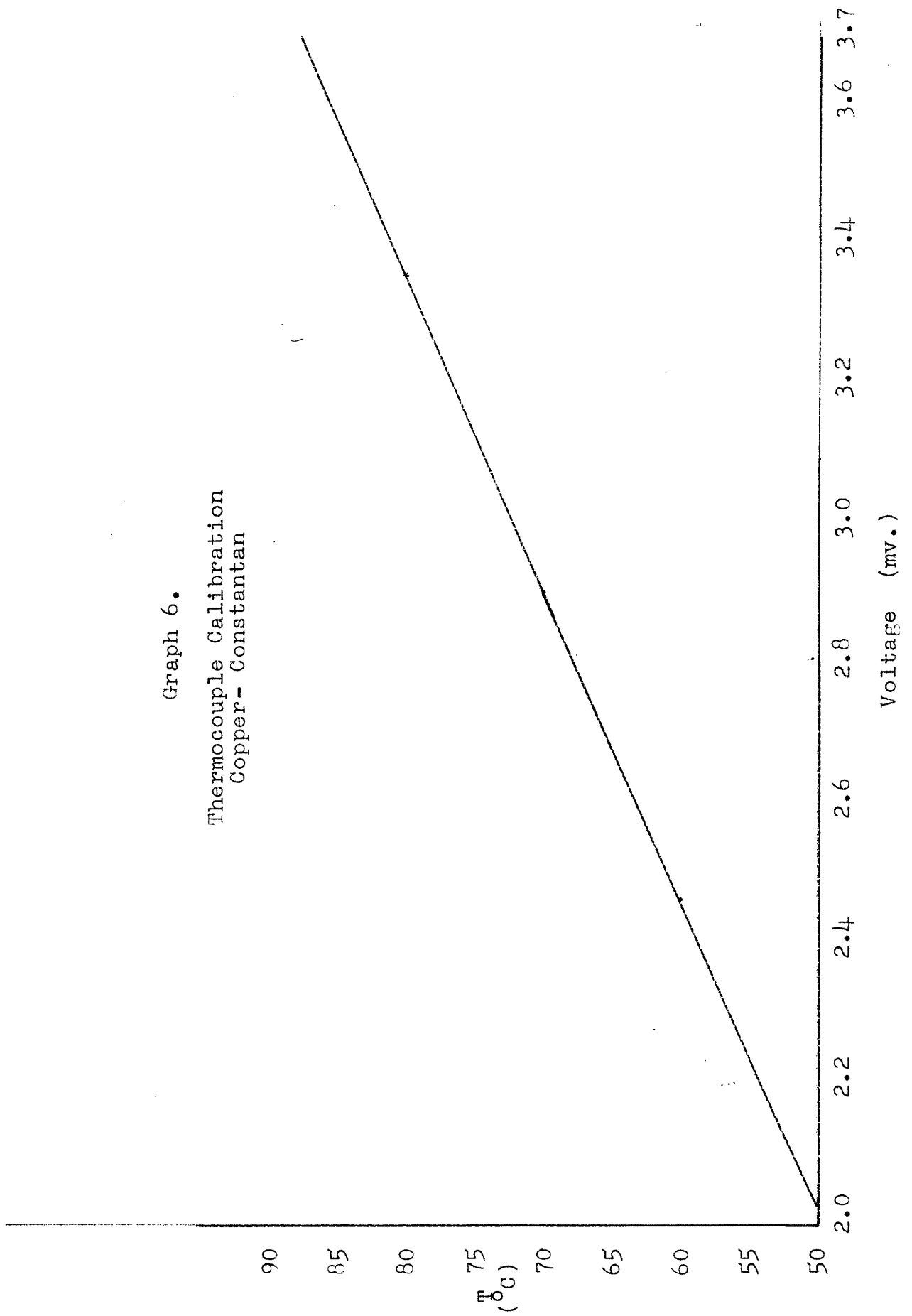
A. SUPPLEMENTARY DETAILS (cont.)

2. Lockhart and Martinelli

Correlation

As has been previously mentioned, the pressure drop in the pipe has been calculated on the basis proposed by Lockhart and Martinelli (13). The basic assumptions involved are that in the pipe $\Delta P_1 = \Delta P_g$, $P_1 = P_g$ at any point in the pipe and the volume of liquid plus the volume of gas in the pipe at any instant equals the total volume of the pipe. From these assumptions it may be seen that

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = 2 f_l \frac{\rho_l V_l^2}{D_l g} = 2 f_g \frac{\rho_g V_g^2}{D_g g} \tag{1'}$$

The friction factors may be determined from the equations

$$f_l = \frac{C_l}{Re_l^m} = \frac{C_l}{\left(\frac{4}{\pi} \propto \frac{W_l}{D_l \mu_l}\right)^m} \tag{2'}$$

$$\tag{3'}$$

$$f_g = \frac{C_g}{Re_g^m} = \frac{C_g}{\left(\frac{4}{\pi \beta} \frac{W_g}{D_g \mu_g}\right)^m}$$

where for laminar flow $C = 16$, $n = 1$ and for turbulent flow $C = 0.023$ and $n = 0.2$. The velocities are the mass flow rates divided by the cross-sectional area for flow and divided by the density.

Substituting into equation (1') there is obtained

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left[\frac{2\left(\frac{4}{\pi}\right)^{2-m} C_l \mu_l^n W_l^{2-m}}{D_p^{5-m} \rho_l g}\right] \propto^{n-2} \left(\frac{D_p}{D_l}\right)^{5-m} \tag{4'}$$

This is equivilent to

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(\frac{\Delta P}{\Delta L}\right)_l \propto^{n-2} \left(\frac{D_p}{D_l}\right)^{5-m} \tag{5'}$$

ϕ_1^2 is then defined to be

$$\alpha^{n-2} \left(\frac{D_p}{D_\ell} \right)^{S-m} = \frac{(\Delta P/\Delta L)_{TP}}{(\Delta P/\Delta L)_\ell} \qquad (6')$$

A similar equation may be derived for the gas

$$\phi_g^2 = \frac{(\Delta P/\Delta L)_{TP}}{(\Delta P/\Delta L)_g} = \beta^{n-2} \left(\frac{D_p}{D_g} \right)^{S-m} \qquad (7')$$

Another variable of the system is

$$\chi^2 = \frac{(\Delta P/\Delta L)_\ell}{(\Delta P/\Delta L)_g} \qquad (8')$$

It may be seen then that $\phi_g^2 = \chi^2 \phi_1^2$ $\qquad (9')$

Plots of ϕ vs. X allow the two phase pressure drop to be calculated from a knowledge of the single phase pressure drop in the same pipe.

The single phase pressure drop for the desired conditions may be calculated from equation (1') above with D_p being the overall pipe diameter and the two phase pressure drop may be calculated from equation (8') and a chart of ϕ vs. X.

A. SUPPLEMENTARY DETAILS (cont.)

3. Physical Properties

1. Heat Capacities (9)

    Ethylene Glycol   at 14.9°C   0.569 cal/gm. °C

                             at 19.9°C   0.573 cal/gm. °C

    Air   at 0.0°C   0.238 cal/gm. °C

                             at 100°C   0.237 cal/gm.°C

2. Densities (9)

    Ethylene Glycol   69.5 lbs./ft.^{3}

    Air (STP)   0.0749 lbs./ft.^{3}

3. Vapor pressure of Ethylene Glycol (10)

T °C	v.p. (mm. Hg)
53	1
92	10
120	40
141.8	400
178.5	760

4. Viscosity of Ethylene Glycol (9)

    19.9 centipoises at 20°C

VIII. APPENDIX (Cont.)

B. SUMMARY OF DATA AND CALCULATED VALUES

RUN #	$\dot{m}_l$	$\dot{m}_g$	$\dot{m}_l/\dot{m}_g$	T_{gi} °C	T_{li} °C	T_{lo} °C	T_{go} °C	ΔT_{in} °C	ΔT_{out} °C	Q BTU/hr	(hA)	(hA)/L
1A	1008	186	5.42	47.0	23.1	24.8	24.82	24.9	0.2	1770	190	285
2A	1008	126.5	7.97	56.0	24.5	25.0	45.4	31.5	20.4	520	12.35	18.51
3A	1008	89.4	11.28	89.0	27.4	28.1	70.1	61.6	42.0	725	7.97	11.96
4A	1008	57.6	17.5	84.5	29.1	29.6	54.6	53.9	14.8	725	13.35	20.1
5A	350	184	1.9	49.5	27.5	29.6	39.89	22.0	10.29	756	27.2	40.8
6A	388	136	2.87	60.0	29.75	31.75	46.3	30.25	14.55	300	20.8	31.2
7A	462	98	4.71	70.5	29.5	31.5	47.9	41.0	16.4	951	19.75	29.6
8A	530	50	10.6	90.3	29.3	30.5	59.7	61.0	29.2	655	8.44	12.63
9A	970	214	4.53	49.0	22.0	23.8	29.4	27.0	5.6	1795	70.34	105.3
10A	622	183.3	3.39	50.2	24.3	26.0	36.3	25.9	10.3	1090	35.8	53.7
11A	644	163	3.96	53.5	25.5	27.1	38.3	28.0	112.	1060	32.2	48.3
12A	719	107	6.72	66.5	27.75	28.8	37.7	20.7	38.75	780	15.05	22.6
13A	792	57.6	13.75	62.5	26.4	27.0	41.6	36.1	14.6	490	11.42	17.25
14A	568	56.2	10.1	70.0	25.7	27.8	38.9	44.3	11.9	752	17.0	25.5
15A	539	88.9	6.06	59.0	27.1	27.8	48.8	31.9	21.0	388	14.9	22.38
16A	492	149.2	3.3	55.5	27.5	29.0	43.7	28.0	14.7	754	20.4	30.6
17A	422	218	1.94	53.5	29.3	30.8	49.51	24.2	15.71	651	18.3	27.4
18A	500	49.5	10.1	--	--	--	--	--	--	--	--	--
19A	450	90.7	4.96	58.0	27.4	49.64	30.6	19.75	19.76	324	7.3	10.95
20A	429	115	3.75	62.0	27.0	28.75	46.3	35.0	17.55	773	17.0	25.5
21A	327	218	1.5	53.0	27.3	30.0	43.25	25.7	12.23	911	27.8	41.7

RUN #	$\dot{m}_l$	$\dot{m}_g$	$\dot{m}_l/\dot{m}_g$	T_{gi} °C	T_{li} °C	T_{lo} °C	T_{go} °C	ΔT in °C	ΔT out °C	Q BTU/hr	(hA)	(hA)/L
1E	1046	174.8	5.98	55.0	21.2	23.5	21.9	34.8	0.0	2475	—	—
2E	1046	139	7.53	61.0	23.5	26.0	15.8	37.5	0.0	2690	—	—
3E	1046	115.2	9.07	66.5	25.3	27.4	20.9	41.2	0.0	2260	—	—
4E	1046	88.4	11.81	78.8	23.0	26.6	14.8	55.8	0.0	2800	—	—
5E	621	95.3	6.52	86.5	24.8	27.3	47.3	61.7	20.0	1600	23.95	5.98
6E	621	179.5	3.46	57.5	29.7	32.5	34.2	27.8	1.7	1790	10.65	2.66
7E	1046	44.5	23.5	99.0	28.3	30.2	3.0	70.7	0.0	1830	—	—
8E	1046	60.0	17.4	70.0	33.0	34.2	36.7	36.7	0.0	968	—	—
9E	1025	172.0	5.96	38.5	23.0	24.3	19.9	15.5	4.2	1471	34.1	8.54
13E	1020	80.5	12.68	62.0	26.5	27.5	31.7	35.5	0.0	1045	—	—
14E	1062	57.5	18.5	63.0	26.75	27.75	23.7	41.25	0.0	1090	—	—
15E	839	54.5	15.4	68.0	27.3	23.5	23.5	40.7	0.0	1040	—	—
16E	815	87.9	9.28	59.0	28.0	29.4	27.75	31.0	0.0	1177	—	—
17E	848	117.0	7.25	52.0	29.5	30.3	29.3	23.0	0.0	1138	—	—
18E	837	139	6.02	46.5	29.5	30.8	27.8	17.0	0.0	1120	—	—
19E	253	177.5	1.43	40.7	20.3	23.4	30.75	20.4	7.35	755	32.8	8.2
20E	316	131.3	2.4	49.0	22.0	24.5	34.7	27.0	10.2	813	27.4	6.85
21E	383	85.3	4.48	57.5	22.5	24.5	35.9	25.0	11.4	789	25.3	6.33
22E	438	47.8	9.18	71.5	22.8	25.0	23.3	48.7	0.0	992	—	—
23E	622	44.8	13.9	82.0	20.0	21.5	31.9	62.0	10.4	962	18.5	4.62
24E	599	106.7	5.52	59.5	21.9	23.2	41.0	37.6	17.8	792	16.6	4.15
25E	505	178.5	2.83	49.0	24.5	26.6	34.7	24.5	8.1	1091	40.9	10.22

C. SAMPLE CALCULATIONS

1. Orifice Calibration

The air orifice was calibrated by using a heat balance on the air heater which was wrapped with aluminum foil to prevent heat loss by radiation. At steady state

$$i^2R = \dot{m}_g c_{pg} \Delta T_g = E^2 K/R \qquad (6)$$

where K = 3.41 BTU/hr./watt, R is the resistance of the wire, and E is the voltage.

Solving this equation for the gas flow rate gives

$$\dot{m}_g = E^2 K/Rc_p \Delta T_g \qquad (7)$$

Substituting numbers from run 10A

$$\dot{m}_g = (60 \pm 1)^2(3.41)/(6.03 \pm .05)(.238)(77.4 \pm .1)$$
$$\dot{m}_g = 110.9 \pm 2.2 \text{ lbs./hr.}$$

The upstream pressure for this flow rate was measured to be $25.7 \pm .2$ psia, and the mass flow rate was corrected to 1 atm. upstream pressure by multiplying calculated values by $(14.7/P_s)^{\frac{1}{2}}$. This was done because mass flow is proportional to the square root of the density. Again substituting numbers gives

$$\dot{m}_g = (110.9 \pm 2.2)(14.7/25.7 \pm .2)^{\frac{1}{2}}$$
$$= 83.8 \pm 1.3 \text{ lbs./hr.}$$

A graph was then prepared of lbs./hr. vs. pressure drop across the orifice (Graph 4) and used as the calibration curve.

Graph 7.
Sample Graph for Calculation
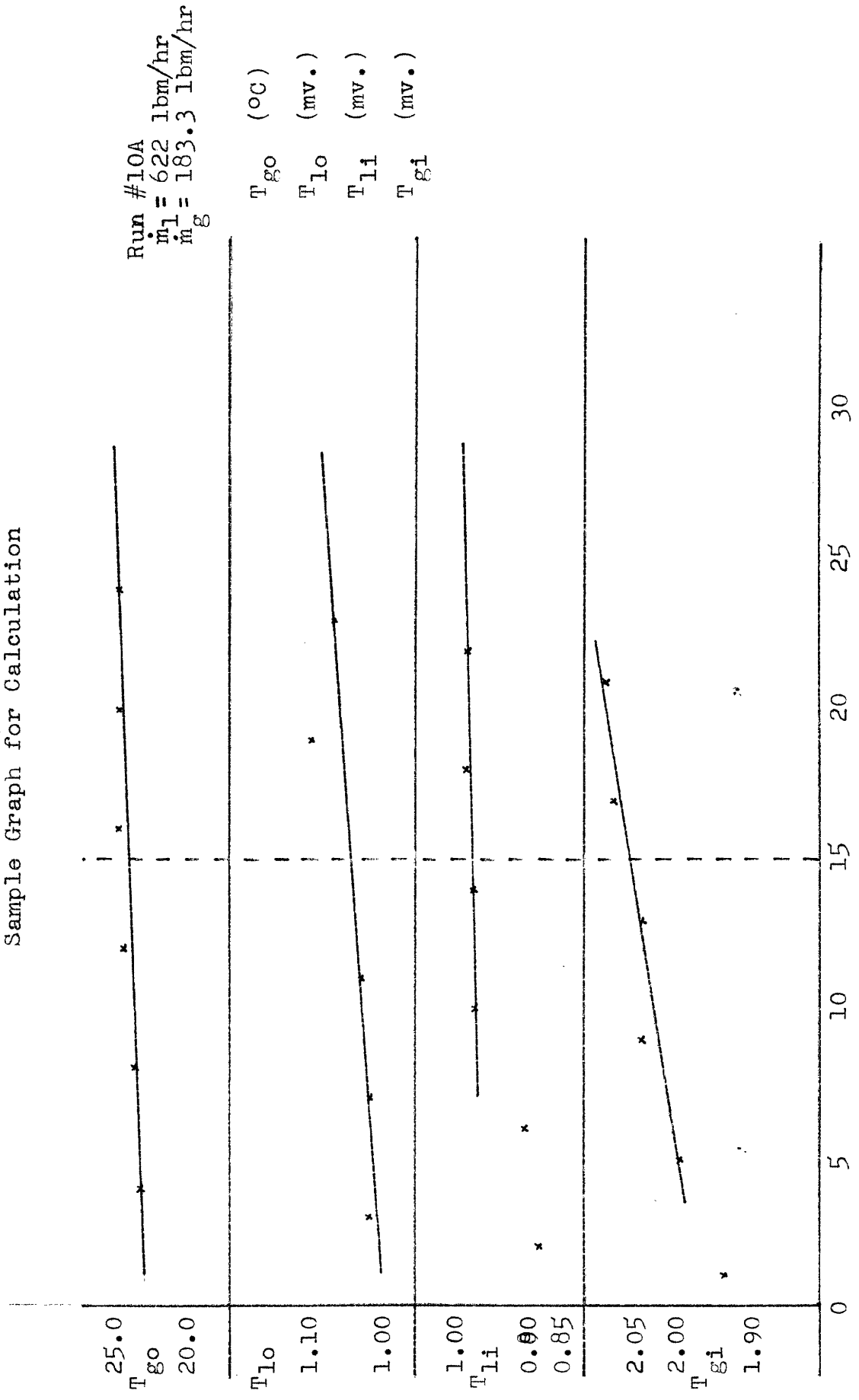
2. Energy Balance

The overall energy balance was made by assuming that the heat gained by the liquid was exactly equal to the heat lost by the gas. This is assuming that there was no heat leak to the system and no vaporization of ethylene glycol. Both of these assumptions have been discussed previously.

Due to the difficulty in measuring the air exit temperature, it was calculated as follows:

$$\dot{m}_g c_{pg} \Delta T_g = \dot{m}_l c_{pl} \Delta T_l \tag{4}$$

which may be solved for the gas exit temperature to give

$$T_{go} = \dot{m}_g c_{pg} T_{gi} - \dot{m}_l c_{pl} \Delta T_l / \dot{m}_g c_{pg} \tag{8}$$

Substituting numbers from Run 10A

$$T_{go} = \frac{(183.3\pm2.2)(.238)(123.4\pm.1) - (622\pm15)(.573)}{(183.3\pm2.2)(.238)} \frac{}{(3.06+.01)}$$

$$T_{go} = 97.4\pm.7\,^{\circ}F$$

C. SAMPLE CALCULATIONS (cont.)

3. Calculation of (hA)

The quantity (hA) was calculated by use of the overall heat transferred in the system as follows:

$$Q = \dot{m}_1 c_{p1} T_1 = (hA)\Delta T_{1nm} \tag{5}$$

where ΔT_{1nm} is the logarithmic mean temperature difference. This equation may be solved by using the definition of T_{1nm} to give

$$(hA) = \frac{Q \ln(\Delta T_1/\Delta T_2)}{\Delta T_1 - \Delta T_2} \tag{9}$$

For run 10A this gives

$$(hA) = \frac{1090 \pm 30 \ln (25.9 \pm .2/10.3 \pm .8)}{(25.9 \pm .2) - (10.3 \pm .8)}$$

$$(hA) = 35.8 \pm 6.2.$$

VIII. APPENDIX (cont.)

D. LOCATION OF ORIGINAL DATA

. The original data for this investigation may be found in notebook number one entitled "Heat Transfer between Fluids in Two Phase Flow" at the Chemical Engineering Department of th Massachusetts Institute of Technology.

VIII. APPENDIX (cont.)

E. NOMENCLATURE

A	area for heat transfer	sq. ft.
a	cross sectional area of pipe	sq. ft.
c_p	heat capacity	BTU/$^{\circ}$F.
C	coefficient for Blasius friction factor	---
D	inside diameter of pipe	ft.
E	voltage	volts
f	friction factor	---
h	heat transfer coefficient	BTU/hr. (sq. ft.)($^{\circ}$F.)
i	electric current	amps
L	length of pipe	ft.
$\dot{m}$	mass flow rate	lb./hr.
P_s	static pressure	psig
ΔP	pressure drop	cm. Hg
R	electrical resistance	ohms
R'	fraction of pipe volume that is fluid	---
T	temperature	$^{\circ}$F.
ΔT_{lnm}	logarithmic temperature change	$^{\circ}$F.
V	velocity of fluid	ft./hr.
W	mass flow rate	lb./hr.
X	Lockhart-Martinelli parameter	---
α, β	ratio of cross sectional area for flow if $D_{1(2)}$ were the actual D	---
μ	thermal conductivity	BTU/(hr)(sq.ft.)($^{\circ}$F)/ft.

Subscripts

g   gas phase

i   inlet condition

l   liquid phase

o   outlet condition

TP   two phase

Exponents

m,n used in Blasius friction equations for
        gas and liquid respectively

Dimensionless Ratio

$Re = \rho VD/\mu$ Reynolds number

VIII. APPENDIX (Cont.)

F. LITERATURE CITATIONS

(1) Alves, G.A., "Co-Current Liquid-Gas Flow In a Pipe-Line Contactor", Chem. Eng. Pro., 50, 449-456. (1954).

(2) Baker, O., "Simoultaneous Flow of Oil and Gas", Oil and Gas Journal, 53, 185, (July 1954).

(3) Bergelin, O.P., and Gazley, C., "Heat Transfer and Fluid Mechanics Institute Papers", 5-18, 1949.

(4) Berman, L.D., "The Treatment of Experimental Data on the General Coefficients of Heat and Mass Transfer Between a Liquid and a Gas Medium", Journal of Applied Chemistry of the USSR, 29, 151-153, (1956).

(5) Boelter, L.M.K., and Kepner, "Pressure Drop Accompanying Two-Component Flow Through Pipes", Ind. Eng. Chem., 31, 426-434, (1939).

(6) Charvonia, D.A., "A Review of the Published Literature Pertaining to the Annular, Two-Phase Flow of a Liquid and a Gaseous Medium in a Pipe", Jet Propulsion Center, Purdue University, (1958).

(7) Cribb, G.S., and Nelson, E.T., "The Simoultaneous Transfer of Heat and Mass Between Water and Moist Coal Gas", Chem. Eng. Sci., 5, 20-23, (1956).

(8) Emeneger and Bing, 10.27 Chemical Engineering Laboratory Report #33-60, Massachusetts Institute of Technology, 1961.

(9) Handbook of Chemistry and Physics, Fortieth Edition, Chemical Rubber Publishing Company (1958-59).

(10) Hellig, W., "Vapor Pressure of Ethylene Glycol Solutions", Zeit fur Ang. Physik, 1, 9-11, (1948).

(11) Johnson, H.A., and Abou-Sabe, A.H., "Heat Transfer and Pressure Drop for Turbulent Flow of Air-Water Mixtures in a Horizontal Pipe", Trans. ASME, 74, 977-987, (1952)

(12) Kay, J.M., "Fluid Mechanics and Heat Transfer", Cambridge University Press, Cambridge, Eng., 1957.

(13) Lockhart and Martinelli, "Proposed Correlation of Data for Isothermal, Two-Phase Flow in Pipes", Chem. Eng. Pro., 45, 39, (1949).

(14) McAdams, W.H., "Heat Transmission", 395-397, McGraw-Hill Book Company, New York, 1950.

(15) Perry, J.H., "Chemical Engineer's Handbook", McGraw-Hill Book Company, New York, 1950.

(16) Reed, Matiatos, and Wegener, 10.27 Chemical Engineering Laboratory Report #17-60, Massachusetts Institute of Technology, 1960.

(17) Reid, R.C., Reynolds, A.B., Diglio, A.J., Spiewak, I., and Klipstein, D.H., "Two-Phase Pressure Drops in Large Diameter Pipes", A.I. Ch.E. Journal, 3, 321-324, (Sept. 1957).

(18) Vener, R.E., "Liquid-Gas Contacting", Chem. Eng., 63, 175-206, (Aug. 1956).

(19) Veschoor, H., and Stemerding, S,, "Heat Transfer in Two-Phase Flow", Institution of Mechanical Engineers, London, 201-203, (1951).