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A flow apparatus for measuring the constant pressure specific heats of gases

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A FLOW APPARATUS FOR MEASURING THE

CONSTANT PRESSURE SPECIFIC HEATS

OF GASES

by

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A Research Report

Presented to the Graduate Faculty

of Lehigh University

in Candidacy for the Degree of

Master of Science

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1964

CERTIFICATE OF APPROVAL

This research report is accepted and approved in partial fulfillment of the requirements for the degree of Master of Science in Chemical Engineering.

Jan 15, 1964

(Date)

(Professor in Charge)

(Head of the Department)

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ABSTRACT

The contruction of an apparatus for measuring ratio of constant pressure heat capacity at high pressure to constant pressure heat capacity at low pressure at a given temperature as a function of pressure is described. The operation of the apparatus was checked using air at 26 °. With some modification in design, the apparatus will prove useful in obtaining accurate values of this ratio.

INTRODUCTION

In consideration of the need for accurate thermodynamic data with which to check equations of state, several quantities other than pressure, volume, and temperature data may be utilized. Such quantities must be measurable in the laboratory to a high degree of accuracy, particularly in the high pressure-low temperature gaseous region where considerable gas non-ideality is encountered.

Constant pressure gas heat capacity represents one such thermodynamic quantity that may be measured to a precision which is usually proportional to the amount of care and refinement used in constructing the apparatus. In terms of pressure, volume, and temperature, the constant pressure heat capacity may be given as:

$$(1) \quad C_P = \overline{C}_P - T \int_c^f \left(\frac{\partial^2 V}{\partial T^2}\right)_P dP$$

where $\overline{C}_p$ is the zero pressure heat capacity. It is interesting to note therefore, that the change of constant pressure heat capacity with pressure at constant temperature is the second derivative of the molar volume with respect to temperature multiplied by the negative temperature.

$$(2) \quad \left(\frac{\partial C_P}{\partial P}\right)_T = -T \left(\frac{\partial^2 V}{\partial T^2}\right)_P$$

This quantity for an ideal gas is obviously zero. If this quantity could be measured experimentally, it would provide an extremely accurate test of an equation of state.

Several methods and a multitude of apparatus have been utilized to obtain gas heat capacity. The first major attempt

at compiling the data and techniques in measuring gas heat
capacities was accomplished in 1924 with the publication of
Partington and Shilling's The Specific Heats of Gases (11).
This work gave a detailed account of various methods and
apparatus along with a tabulation of results of a great number
of investigators. Masi (6) in 1954 mentioned briefly, more
recent developments in the determination of gas heat capacity.
The methods of measurement may be devided into two categories,
direct determinations which involves a calorimeter, and
indirect determinations where another variable is measured
which is some function of the heat capacity.

The most popular method, in that it is the only method of
measuring directly absolute values of C_p, is the flow calor-
imeter. It consists of measuring the temperature of a gas
flowing at a constant rate at two points between which,
electrical heat of a known quantity has been added. Measurement
of pressure, temperature, gas flow rate, and electrical
input are necessary to obtain a numerical value of heat
capacity. Several apparatus of this type are described by
Partington and Shilling (11). The most sucessful measurements
have been made by Wacker, Cheney, and Scott (14) at the National
Bureau of Standards and by Waddington, Todd, and Huffman (15)
at the United States Bureau of Mines. The accuaracy of their
measurements are on the order of 0.1 per cent. Several
apparatus of this type (7), (9), (13), including that of
Waddington, Todd, and Huffman (15) are constructed of glass

and generally operate at one atmosphere or below. The National Bureau of Standards model has been used at pressures of up to 100 atmospheres (10). More recently, Faulkner (2), at the University of Michigan in 1959, has constructed a flow calorimeter for use at high pressures. A somewhat similar method of flow calorimetry is used by Brown et. al. (4), (12) at the University of Michigan and by Jenkins and Berwaldt at Linde (3). This method uses a secondary substance such as cooling water or an evaporating refrigerant to supply the required heat removal or input.

Another direct method for measuring c_p was devised by Workman (17) and involves the exchange of heat from a flowing gas at high pressure to the throttled gas flowing at low pressure. The only measurements necessary are those of pressure and temperature, assuming constant gas flow is available. A heat balance between the two gases may be written:

$$(3) \qquad M_{l.p.} c_{p_{l.p.}} \ T_{l.p.} = M_{h.p.} c_{p_{h.p.}} \ T_{h.p.}$$

The mass flow rates however, being identical reduces the equation to a heat capacity ratio:

$$(4) \qquad c_{p_{h.p.}} / c_{p_{l.p.}} = \Delta T_{l.p.} / \Delta T_{h.p.}$$

It is obvious that the heat balance described above is true assuming there has been no heat leak to the surroundings and there has been no pressure drop through the heat exchanger. To obtain an absolute value for the heat capacity at high pressure, the heat capacity at low pressure must be known to a high degree of accuracy.

Among the indirect methods the isentropic expansion is perhaps the least involved. For an ideal gas $PT^{\gamma/1-\gamma}$ is a constant and the heat capacity ratio $\gamma = c_p/c_v$ can be obtained by recording the pressure and temperature before and after an isentropic expansion. In the case of a real gas the relation becomes PT^{β} where β may be shown as:

$$\beta = -\frac{c_p}{R[z + (\partial z/\partial T)_p]}$$

where z is the compressibility factor PV/RT. This method has a reported accuracy of about 0.5 per cent (6).

The heat capacity ratio c_p/c_v is a function of the velocity of sound in a gas according to the relationship:

$$\gamma = a^2 \left(\frac{\partial \rho}{\partial P}\right)_T$$

Knowledge of an equation of state must be available to predict the change of density with respect to pressure at constant temperature.

Clark and Katz (1) have devised a method by which the natural frequency of two volumes of gas is determined by changing the frequency of vibration of a piston separating the two volumes until the maximum amplitude is obtained.

In comparing the various methods mentioned above, the flow calorimeter offers the most direct determination of c_p. The precision of this method depends on the accuracy with which pressure, temperature, flow rate, and electrical input can be measured. The heat exchange method devised by Workman (17) has the advantage of simplicity. Temperature

and pressure are the only required measurements, making this method most attractive. The heat leaks inherent in this apparatus may also be cancelled as will be shown in a later section.

EQUIPMENT

The equipment was designed to offer accurate control over various portions of the system in view of the length of time that would be required to reach steady state operation. The schematic diagram of the apparatus is illustrated in Figure 1.

The compressor, a Worthington three stage angle compressor (Type v4A3), is capable of delivering approximately 10 SCFM of gas at 3000 psi pressure. The first stage after-cooler is air cooled while the second and third stage after-coolers are water cooled.

The heat exchanger was designed to operate at 3 SCFM so a recycle system had to be constructed to divert a major portion of the gas flow. A stainless steel tank, 12 inches in diameter and 24 inches long was incorporated in the recycle line for a double purpose; to operate as a surge tank to lessen fluctuations in flow rate and secondly, as a gas storage tank to hold a portion of the gas when the system is not operating. Two needle valves are used in series to reduce the pressure between the three stage compressor and the surge tank.

The gas flowing to the heat exchanger section of the apparatus is first passed through a series of purifiers. The first, a steel pipe 2 inches in diameter 20 inches in length, contains small mesh screening and glass wool to separate any entrained oil from the high pressure gas. The

gas then passes into another 2 inch diameter steel pipe
24 inches in length containing silica gel to dry the high
pressure gas and then to remove any remaining trace of
oil, the gas is passed through a third pipe filled with
activated charcoal. The gas flow then proceeds to the constant
temperaure bath where the gas passes through a 75 foot
coil of 1/8 inch copper tubing in a 6 inch diameter, 20
inch high stainless steel dewar. Cooling is supplied to
the bath liquid by a secondary coil of 1/4 inch copper
tubing immersed in the liquid. A Bayley Precision Temperature
Controller, Model 237, is employed to regulate the temperature
of the bath liquid. Heat is supplied by a 250 watt
bayonet heater connected in series with a reostat to the
controller. The bath fluid is agitated by a mixer centrally
located in the dewar. A shematic diagram of the constant
temperature bath is shown in Figure 3.

The gas then enters the stainless steel dewar containing
the heat exchanger which is illustrated in Figure 2. The gas
enters the dewar through the top via a 5/16 inch stainless
steel tube. The tube is connected to the 1/2 inch thick
brass top by passing it through an 8 inch long 7/16 inch
diameter stainless steel tube having a wall thickness of
0.006 inches. The thin tube is soldered to the 5/16 inch
stainless steel tube at both ends and is passed 4 inches
through the top of the brass plate where it is soldered in
place. This thin tube allows little heat to be transferred

from the top plate to the gas entering the dewar. The gas
then flows about two feet through tubing until its temperature
and pressure are measured. The temperature on the high
pressure side is measured by means of two junction copper-
constantan thermocouples which are strung about 12 inches
along the inside of the tube carrying the gas to prevent
heat leak from the tip of the thermocouples. The 24
gauge thermocouple wires are taken out of the high pressure
tubing with the use of midget Conax thermocouple glands.
The pressure is measured through a 1/16 inch stainless
steel tube inserted in the gas line and brought out of the
line and the dewar with a combination of tube fittings.
The pressure on the high pressure side is then measured by
a Heise Bourdon gauge certified to better than 0.1% of a
full range of 3000 lbs/in².

The gas after passing the thermocouples and the pressure
tap, enters the heat exchanger composed of a coil fin
tube placed in the annular space between two stainless
steel tubes. The coiled finned tube is made of 1/4 inch
copper tubing with 1/2 inch diameter fins 20 to the inch.
The annular space is formed by an outside stainless steel
tube 3-1/4 inch outside diameter with a 0.095 inch wall
an inside stainless steel tube 2 inch outside diameter with
a 0.049 inch wall. The outside tube was chosen because of
its inside dimensions and approximately 0.07 inches were
machined off the wall to reduce the quantity of metal.

The high pressure line enters the heat exchanger through
another 7/16 inch diameter stainless steel tube with a
0.006 inch wall; with the purpose of reducing any heat
flux between the low pressure measuring position and the
high pressure measuring position. This precaution should
help to create isothermal conditions at these measuring
points. The high pressure gas flows in the coiled finned
tube through sixteen turns down the eight inch exchanger
and exits from the exchanger in the same manner with which
it entered; through four inches of the thin stainless steel
tube. The temperature and pressure are measured in an
identical manner as the inlet. The gas exits the dewar
enclosing the exchanger again through the thin stainless
steel tube arrangement.

The dewar is filled with Santocel, a powdered silica
insulation. The dewar is a 6 inch diameter, 22-3/4 inch high
flanged stainless steel dewar capable of 50 psi internal
pressure. The brass plate mentioned previously may be
bolted to the flange allowing the insulated space around the
heat exchanger to be evacuated.

On exiting from the dewar, the high pressure gas is passed
through a series of three needle valves used to reduce the
pressure to a desired level. The low pressure gas then
passes through 90 feet of 1/4 inch low pressure tubing in a
second constant temperature bath controlled in the manner
described previously. The temperature regulated low pressure

gas passes back into the dewar and then through 2 feet of 5/16 inch diameter stainless steel tubing to the point where the temperature and pressure are measured by the same procedure used in the high pressure side. The low pressure gas passes through the shell side of the co-current heat exchanger where it is forced to take a circuitous path through the fins by wrapping loose cord around the outside and inside of the coiled tube to prevent chanelling along the walls. This is illustrated in Figure 2. The gas passes out of the heat exchanger where the temperature and pressure are measured and out of the dewar into a Fischer and Porter Flowrator to determine the stability of the flow. The gas then flows to the inlet of the compressor to be recycled.

The high pressure fittings are Swaglok and Tylok while the low pressure side makes use of the less expensive Imperial compression fittings. To join stainless steel tubing to copper tubing, brass fittings with stainless steel ferrules are used. The ferrules are first crimped to the stainless steel tube by a stainless steel fitting which is then removed and a brass fitting substituted. This provided a satisfactory coupling at a reasonable cost.

The dewar and most of the gas lines were packed in rock wool insulation.

SURGE TANK

CONSTANT TEMP BATH

PURIFICATION

GAS STORAGE

COMPRESSOR

FLOWRATOR

HEAT EXCHANGER

CONSTANT TEMP BATH
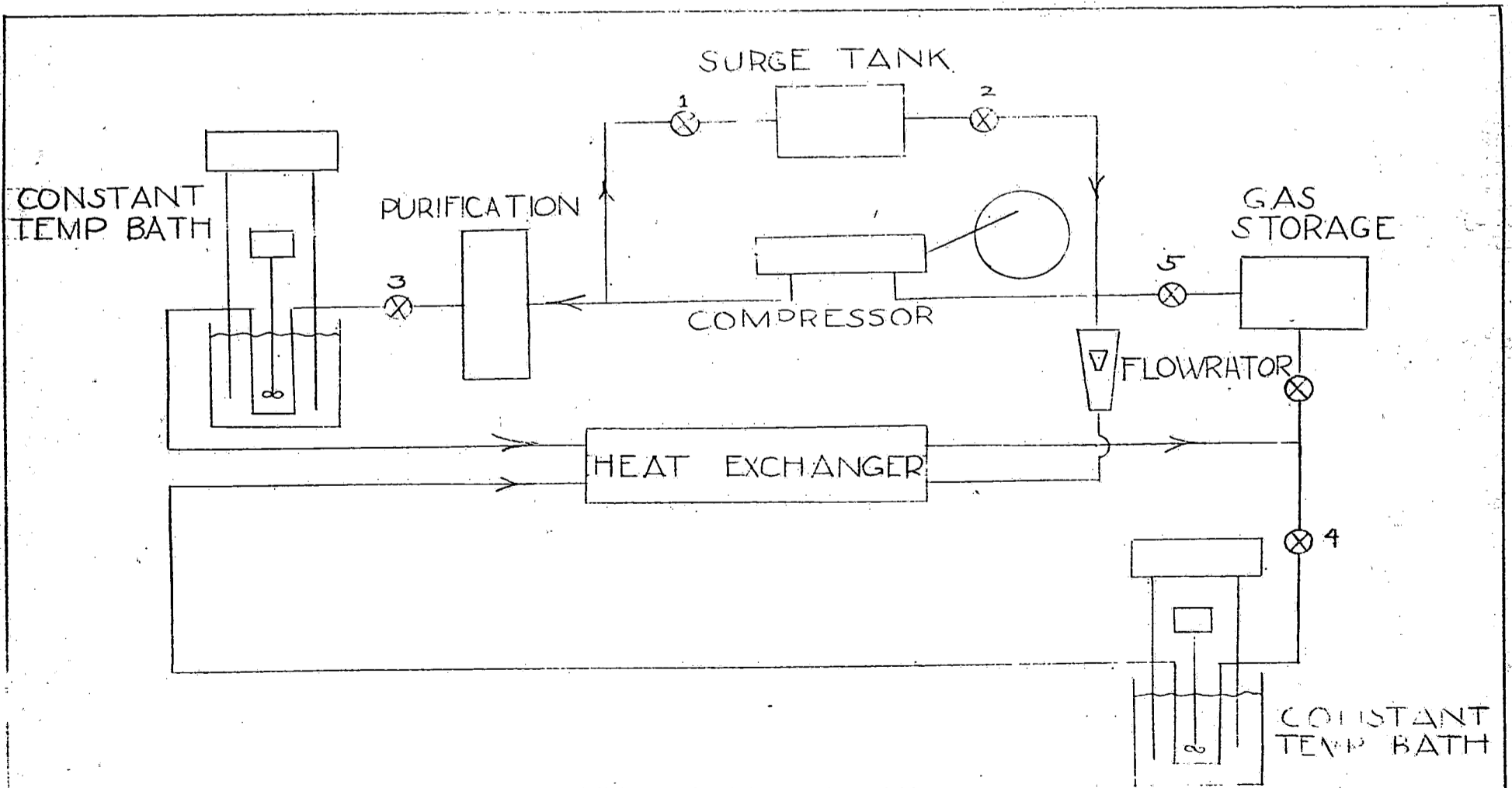
FIGURE 1

SCHEMATIC DIAGRAM OF APPARATUS

FIGURE 2

ONE OF FOUR TUBES
EQUALLY SPACED

DEWAR

SILICA AEROGEL

CORD
FINNED
TUBE

ALL TUBES
CONTAIN PRESSURE
TAPS AND
THERMOCOUPLES
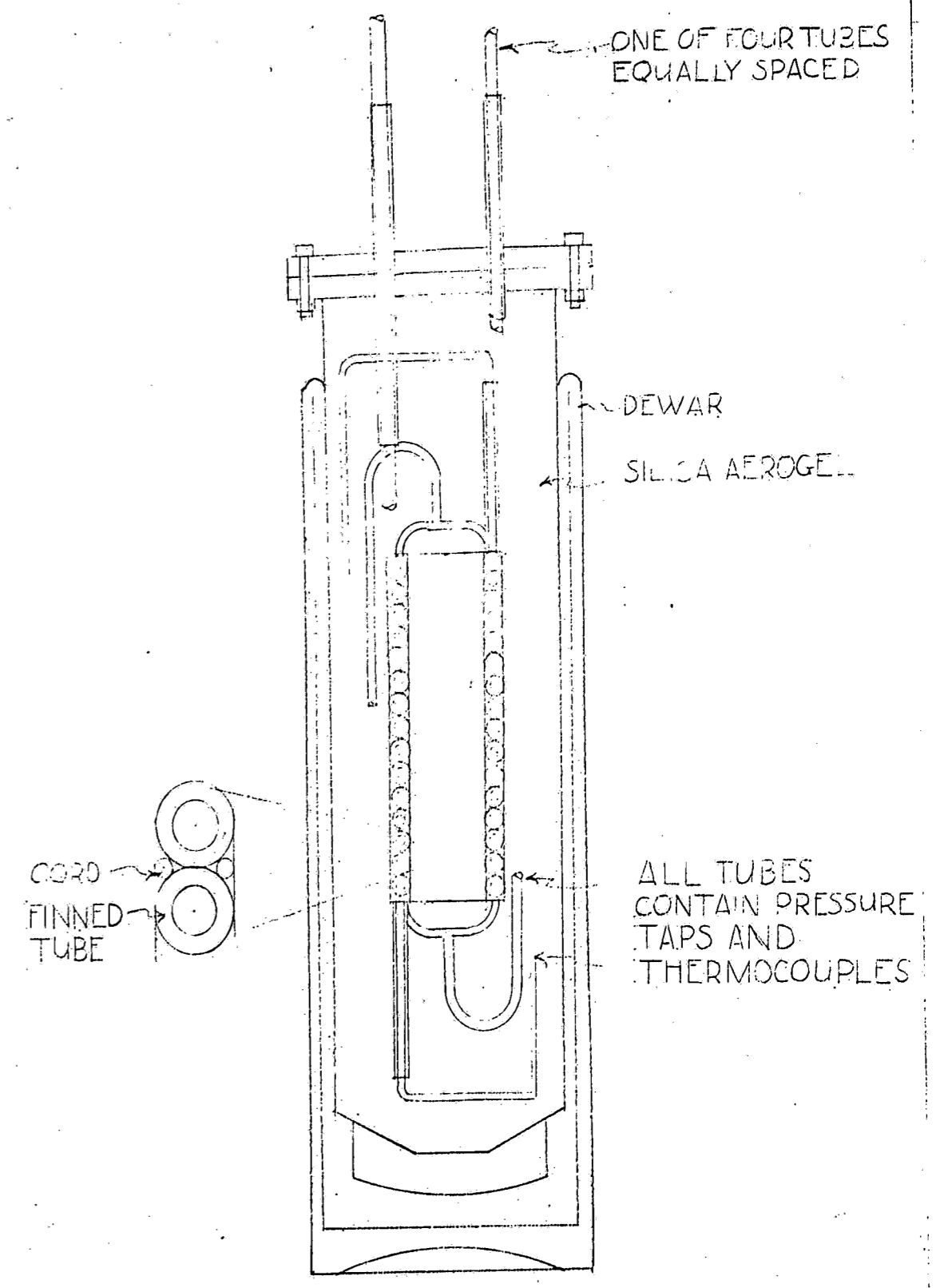
HEAT EXCHANGER CALORIMETER

CONTROLLER

GAS
IN

C T

COOLANT
IN

REOSTAT

HEATER

TEMPERATURE
SENSOR

STIRRER

DEWAR

CONSTANT   TEMPERATURE   BATH
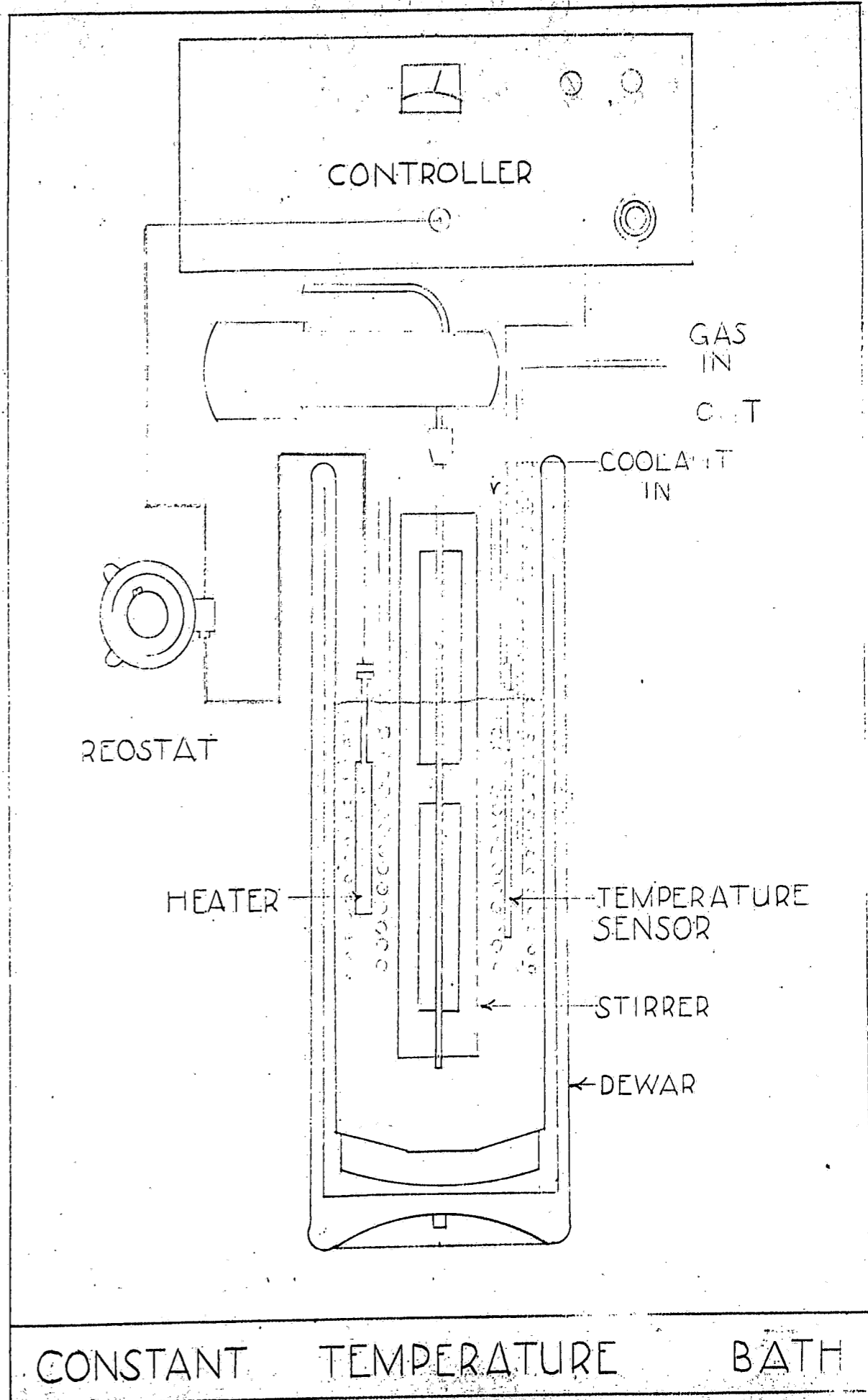
THEORY

The errors associated with making measurements on a calorimeter of this type are substantial considering the degree of accuracy that is desired. The heat exchanger errors may be classified into two categories; the heat leak, $\downarrow$, which will be defined as the net amount of external heat leak per weight of gas to both gas streams; and secondly the Joule-Thompson cooling caused by pressure drop through the exchanger. The temperature change associated with this Joule-Thompson cooling or heating will be expressed as follows from Workman (17). The heat balance may now be written:

$$(5) \quad Q = C_{P_{h.p.}}\left[\Delta T_{h.p.} + \mu_{h.p.}\Delta P_{h.p.}\right] - C_{P_{l.p.}}\left[\Delta T_{l.p.} - \mu_{l.p.}\Delta P_o\right]$$

where $C_{p_{h.p.}}$ and $C_{p_{l.p.}}$ are the average value of the heat capacity over the small temperature intervals $\Delta T_{h.p.}$ and $\Delta T_{l.p.}$ respectively.

To shorten the notation, C_{po}, ΔT_o and $\mu_o \Delta P_o$, will be used for the low pressure properties and unsubscribted values will be used for the high pressure properties. Solving for C_p/C_{po}: (See Appendix page 29 for derivation,

$$(6)\frac{C_p}{C_{p_o}} = \frac{\Delta T_o}{\Delta T} + \left[\frac{Q}{C_{p_o}} + \mu_o \Delta P_o - \frac{\mu \Delta P \Delta T_o}{\Delta T}\right]\left[\frac{1}{\Delta T} - \frac{\mu \Delta P}{\Delta T^2} + \cdots\right]$$

The latter term may be considered to be a correction term. If the bath temperatures are now reversed, that is the inlet low pressure temperature and the co-current inlet high

pressure gas temperature reversed, the heat balance may now be written as:

(7) $$Q = C_p \left[-\Delta T + \mu \Delta P \right] - C_{p_o} \left[-\Delta T_o + \mu_o \Delta P_o \right]$$

Solving for C_p/C_{p_o}: (See Appendix page 30 for derivation)

(8) $$\frac{C_p}{C_{p_o}} = \frac{\Delta T_o}{\Delta T} - \left[\frac{Q}{C_{p_o}} + \mu_o \Delta P_o - \frac{\mu \Delta P \Delta T_o}{\Delta T} \right] \left[\frac{1}{\Delta T} + \frac{\mu \Delta T}{\Delta T^2} + \right]$$

$\mu \Delta P$ will be quite small as compared to ΔT, so the second order term may be neglected. It can be seen by comparing Equation (6) and Equation (8) that only the sign of the correction term has been changed, neglecting the second order term, when the bath temperatures are reversed.

If two lines of data would be plotted as C_p/C_{p_o} versus pressure at constant temperature, one with the low pressure gas inlet temperature lower than the high pressure inlet gas temperature, and the other with the temperatures reversed, the average curve would represent C_p/C_{p_o} as a function of pressure independent of heat leak and pressure drop errors. The average curve should pass through the point, $C_p/C_{p_o} = 1$ at the pressure where the low pressure side has been operating.

PROCEDURE

The switch connecting the standard and working cells to the potentiometer was closed at least an hour previous to collecting data to eliminate the initially large fluctuations. The ice reference bath was prepared with ice and distilled water in a glass dewar flask.

The constant temperature baths were filled with liquid which for this study was water. The coolant, tap water, was circulated through the coil provided in the baths and the controllers and mixers were started.

The cooling water to the three stage Worthington compressor was turned on, the heat exchanger portion closed off (valves 3,4, and 5 in the schematic diagram of Figure 1), and the recycle system open (valves 1 and 2). The oil in the started compressor was allowed to circulate before the gas pressure was allowed to build up. Valve 4 in the schematic diagram was then closed and valve 3 to the high pressure side of the heat exchanger and valve 5 were opened. The gas stored in the gas storage tank was then bled into the system, controlling the rate of pressure build up on the high pressure side with valve 1. When the high pressure reached approximately the required level, the gas storage tank was closed off and valve 5 opened until the desired flow rate was attained. Final adjustment on the high pressure, low pressure, and flow rate were made by valves

1 and 4 and by contolling the amount of gas in the system.
,The temperature controllers were then slowly adjusted to obtain the desired high and low pressure inlet temperatures. The length of time taken for the apparatus to achieve steady state was assumed to be the time necessary for the thermocouples to equilibrate so that there was less than 2 microvolts change in 15 minutes for all readings.

After equilibrium is reached and the data taken, a new high pressure is selected and the flow rate and low pressure are adjusted to their previous conditions. Any necessary temperature adjustments are made and the system is again allowed to equilibrate.

A line of $c_{ph.p.}/c_{pl.p.}$ data is taken for various pressures at constant inlet temperature conditions. Another line of data is then taken with the temperatures of the inlet low and high pressure sides reversed. The reason for this being given in the theory section.

DISCUSSION OF RESULTS

Time of Measurement and Equilibrium

The length of time to take one data point appeared to be on the order of one day. This difficulty lay in three specific areas. The two inlet gas temperatures had to be controlled to a predetermined temperature which necessitated changes in the controller settings followed by at least an hour wait until the bath reached a new temperature and the inlet gas temperature responded to the new bath temperature. At times this process had to be carried out 5 times for a total of 6 hours.

The second difficulty which has some relation to the first is that small changes in room temperature affected the inlet gas temperature. This was caused by a length of tube leading from the bath to the heat exchanger. The tube was insulated with rock wool however a short length of a foot projected from the liquid of the bath into the insulation and still had some effect on the inlet temperature. The change in room temperature from day to day effecting the short length of tubing was probably responsible for some of the small required changes in the controller settings. This problem was finally overcome during the last two data points by insulating the air space over the temperature baths.

Thirdly, the heat capacity of the metal in the heat exchanger, of the copper and stainless steel lines, and of

the insulation was large in comparison to the small heat capacity of the gas. The small temperature changes when going from one data point to the next are accompanied by this large thermal lag which makes the prediction of equilibrium rather difficult. The scatter of the data points is an indication of the fact that equilibrium had not been attained.

The data is listed in Table II in the Appendix on page 31. The results are plotted in Figure 4. The points at 1101 psig on the high pressure cooling line deviated considerably from their expected value. This may have ...en another manifistation of equilibrium not having been obt... ..ed or on the other hand the potentiometer may have been misread. It is obvious that a stricter criteria of equilibrium is desired for more accurate results.

Separation of the L.P. Cooling and H.P. cooling Lines

According to the equations developed in the Theory s. vion for the C_p/C_{p_0} ratio, a separation should occur between the lines for high pressure cooling and low pressure cooling. The error causing these lines to deviate from the true results was attributed to Joule-Thompson cooling and to heat leak. From the results of Figure 4, the average error appears to be 0.035 in dimensionless C_p/C_{p_0} units. From the equation in the Theory section:

$$0.035 = \left[\mu_0 \Delta P_0 + \frac{Q}{C_{p_0}} - \frac{\mu \Delta P \Delta T_0}{\Delta T} \right] \left[\frac{1}{\Delta T} - \frac{\mu \Delta P}{\Delta T^2} + \cdots \right]$$

$\mu_o \cong 0.2$ °C/atm

$\mu \cong 0.2$ °C/atm

$\Delta P_o = 1.2$ inches Hg. (measured)

$\Delta P < 3$ psi (measured)

$\Delta T = \Delta T_o \cong 5$ °C

$\frac{\mu \Delta P}{\Delta T}$ is small compared to $\frac{1}{\Delta T}$

0.0016 compared to 0.2

$\mu_o \Delta P_o \cong \frac{\mu \Delta P \Delta T_o}{\Delta T}$ which leaves $0.035 = \frac{Q}{C_{po} \Delta T}$

$Q = 1.2$ BTU/Lb.mole Gas flow $= 0.0046$ lb.moles/min

Therefore: $Q = 5.5 \times 10^{-3}$ BTU/min

Silica Aerogel was used as an insulating material and has an apparent thermal conductivity of 150 microwatts/cm °K or 0.1 BTU in./hr ft^2 °F at the conditions employed. About 1 ft^2 of heat exchanger area was exposed with an insulating space of about 1 inch. Consequently a temperature gradient of 2°F would produce the resulting heat loss.

The x Intercept

The average line drawn between the H.P. cooling and L.P. cooling lines should obviously pass through the point $C_p/C_{po} = 1$ at a pressure equal to that used for C_{po}. This offers an extremely sensitive test of the data and also serves as a means of finding any errors inherent in the heat exchanger. The data taken for air at 26 °C misses by 2% the point $C_p/C_{po} = 1$ at the pressure of 20.3 psia. This error of 2% represents a deviation of only 4 microvolts in any thermocouple and appears to be constant through all the

measurements.and may be attributed to three factors.

1) An error in thermocouple calibration in one or in all
   of the thermocouples would account for some of the error
   although it is unlikely to be of the 4 microvolt
   magnitude.

2) Heat leak down the thermocouple wires would cause errors
   of this magnitude, however the length of wire which
   travelled down the gas stream makes this factor highly
   improbable.

3) Cooling errors caused by regions of high gas velocity
   were probably the main source of this error. 0.0046
   lb.moles/min or 1.6 ft^3/min of air was passed from the
   large area in the heat exchanger into a 5/16 inch diameter
   stainless steel tube having an inside area of 0.341×10^{-3}ft^2
   This would correspond to an air velocity of 80 ft/sec
   in the tube and less than 5 ft/sec in the exchanger. From
   the equation:

$$\Delta T = \frac{V^2}{2\ C_p\ g_c\ J}$$ where the constant J is a conversion factor,

   $\Delta T = 0.5\ °F = 0.3°C$ which would account for the

   4 microvolt (0.1 °C) error.

Comparison with Michels' data

Michels' P-V-T air data (8) has an accuracy of about
1 part in 10,000 which gives an accuracy of 1 part in 1,000
for heat capacity values calculated from his data and an
accuracy of 1 part in 100 for the slope of a C_p versus

pressure curve at constant temperature. Michels' curve in Figure 4 has an average slope of 0.0001085/psia accurate to one part in 100 while the experimental data obtained for this report yields a slope of 0.0001155/psia. The scatter of the data points is more than enough to account for this error of 7% in slope.

FIGURE 4

C_p/C_{p_O} of AIR at 26°C

— — —Michels' data (8)

+   L.P. cooling

□   H.P. cooling

$\Delta T/\Delta T_O$ or C_p/C_{p_O}

PRESSURE (psia)
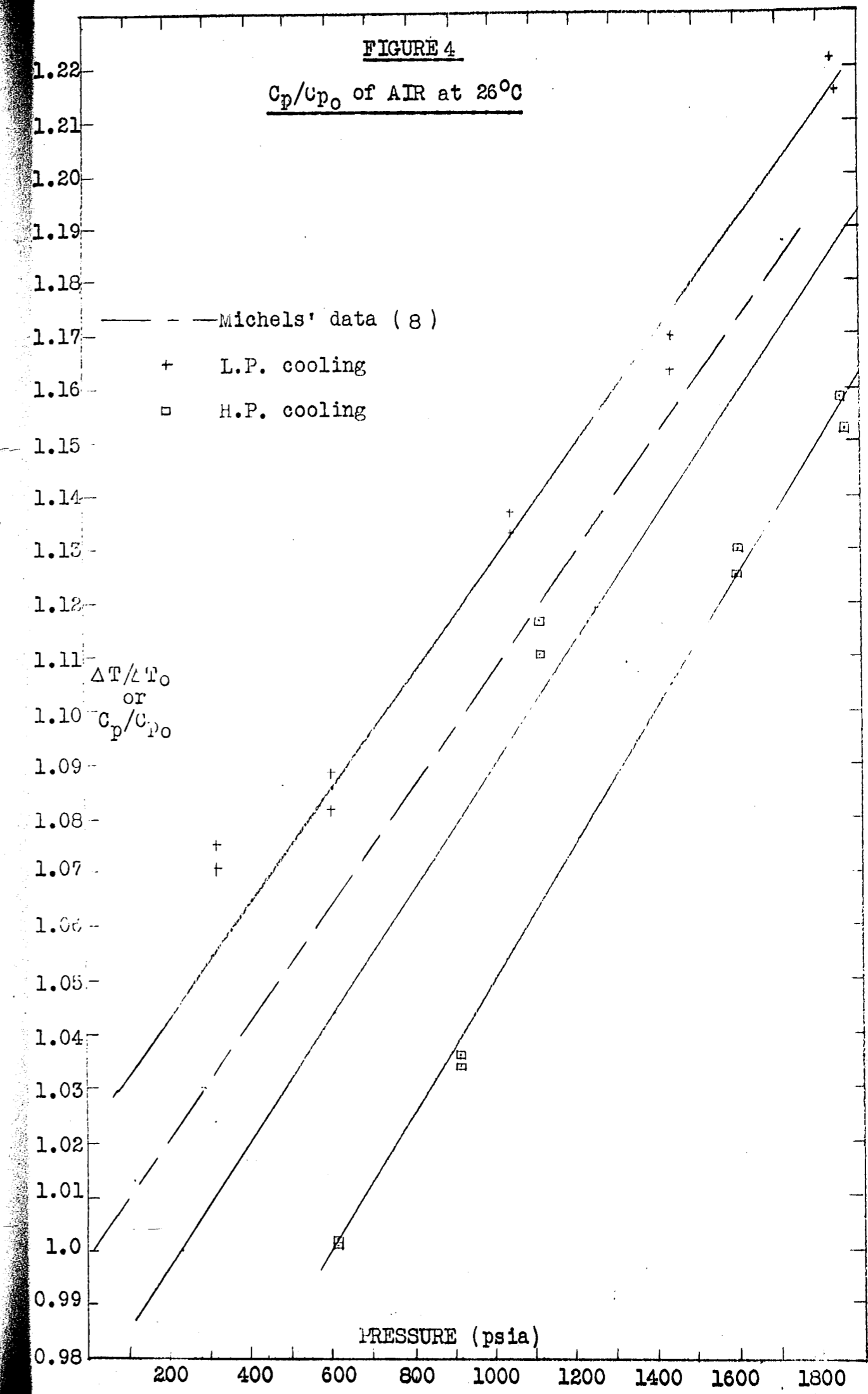
RECOMMENDATIONS

A multi-junction thermopile, using as many as 20 junctions should be employed for reading temperature differences. A temperature difference of 5 degrees Centigrade would produce 4000 microvolts which may be read to better than 0.5 microvolts on the potentiometer. This multijunction thermopile would also offer a better means to determine equilibrium conditions. Care must be taken in constructing such a thermopile as mentioned by White (16).

To reduce the separation of the H.P. cooling and L.P. cooling lines, the heat leak may be reduced by using radiation-convection shielding in addition to evacuated powder insulation. Increasing the flow rate of gas also has the effect of reducing the separation between the lines.

The low pressure leads to the heat exchanger must be $\frac{1}{2}$ inch in diameter or greater to reduce velocity cooling errors.

APPENDIX

CALIBRATION OF THERMOCOUPLES

The copper constantan thermocouples were calibrated at the boiling point of water using a Davis boiling point apparatus. The operation of this piece of equipment is described by Livingston (5).

The apparatus was charged with distilled water and was insulated by aluminum foil and asbestos. Moderate boiling for about an hour was allowed to attain equilibrium wherein two readings 25-30 minutes apart were taken. Three runs, changing the position but not the depth of immersion, changing the water, and changing the boiling chips, were made with the results shown in Table .

The deviations of the thermocouples from their average calibration was generally less than a microvolt. Two observations may be made from the results:

1) The corrections, when applied to the thermocouple readings, will have a compensating effect when differences are taken and consequently uncorrected thermocouple readings may be used to calculate the ratio c_p/c_{p_o}.

2) The two junction difference thermocouples used in the high pressure side, when calibrated with all junctions in the boiling water, gave a definite calibration correction. The magnitude of the correction indicated that the thermocouple deviation added rather than cancelled.

TABLE I

BOILING WATER CALIBRATION OF THERMOCOUPLES

Time (min.)	0	25	0	25	0	30
L.P. out ∿volts	4271.4	4271.4	4272.8	4273.5	4278.4	4279.0
L.P. in ∿volts	4274.9	4275.2	4276.7	4277.5	4283.1	4283.5
H.P. in ∿volts	4276.0	4276.0	4277.3	4277.8	4283.0	4283.4
H.P. out ∿volts	4281.0	4280.7	4280.4	4280.5	4285.4	4286.1
H.P. 2 junc. diff.	10.9	10.8	4.9	4.6	5.4	5.3
Barometric Pressure mm Hg.	751.5	751.6	752.6	752.7	755.5	755.7
Steam-water °C	99.687	99.690	99.723	99.725	99.824	99.842
∿ volts	4262.1	4262.4	4264.0	4264.1	4269.0	4269.6
∿VOLT CORRECTION						
L.P. out ∿volts	9.3	9.0	8.8	9.4	9.4	9.4
L.P. in ∿volts	12.8	12.8	12.7	13.4	14.1	13.9
H.P. in ∿volts	13.9	13.6	13.3	13.7	14.0	13.8
H.P. out ∿volts	18.9	18.3	16.4	16.4	16.4	16.5

Calculation of Heat Leak and Joule-Thompson Errors

Heat Balance:

$$Q = C_P \left(\Delta T + u \Delta P \right) - C_{P_0} \left(\Delta T_0 + u_0 \Delta P_0 \right)$$

Solving for C_p/C_{p_0}:

$$\frac{C_P}{C_{P_0}} = \frac{Q/C_P + \Delta T_0 + u_0 \Delta P_0}{\Delta T + u \Delta P}$$

Expanding:

$$= \frac{\Delta T_0}{\Delta T} - \frac{u \Delta P \, \Delta T_0}{\Delta T^2} + \frac{(u \Delta P)^2 \Delta T_0}{\Delta T^3} - \cdots$$

$$+ \frac{u_0 \Delta P_0}{\Delta T} - \frac{u \Delta P \, u_0 \Delta P_0}{\Delta T^2} + \frac{(u \Delta P)^2 u_0 \Delta P_0}{\Delta T^3} - \cdots$$

$$+ \frac{Q}{C_{P_0} \Delta T_0} - \frac{u \Delta P \, Q}{\Delta T^2 \, C_{P_0}} + \frac{(u \Delta P)^2 Q}{\Delta T^3 \, C_{P_0}} - \cdots$$

Rearranging terms:

$$= \frac{\Delta T_0}{\Delta T} + \left[\frac{u_0 \Delta P_0}{\Delta T} + \frac{Q}{C_{P_0} \Delta T} - \frac{u / P \, \Delta T_0}{\Delta T^2} \right]$$

$$+ \frac{(u \Delta P)^2 \Delta T_0}{\Delta T^3} - \frac{(u \Delta P)^3 \Delta T_0}{\Delta T^4} + \cdots$$

$$- \frac{u \Delta P \, u_0 \Delta P_0}{\Delta T^2} + \frac{(u \Delta P)^2 u_0 \Delta P}{\Delta T^2} - \cdots$$

$$- \frac{u \Delta P \, Q}{\Delta T^2 \, C_{P_0}} + \frac{(u \Delta P)^2 Q}{\Delta T^3 \, C_{P_0}} - \cdots$$

$$\frac{C_P}{C_{P_0}} = \frac{\Delta T_0}{\Delta T} + \left[u_0 \Delta P_0 + \frac{Q}{C_{P_0}} - \frac{u \Delta P \Delta T_0}{\Delta T} \right] \left[\frac{1}{\Delta T} - \frac{u \Delta P}{\Delta T^2} + \cdots \right]$$

Calculation of Heat Leak and Joule-Thompson Errors

$$Q = C_P \left(-\Delta T + \mu \Delta P\right) - C_{Po}\left(-\Delta T_0 + \mu_0 \Delta P_0\right)$$

Solving for C_p/C_{po}:

$$\frac{C_P}{C_{Po}} = \frac{Q/C_{Po} - \Delta T_0 + \mu_0 \Delta P_0}{-\Delta T + \mu \Delta P}$$

Expanding:

$$= \frac{\Delta T_0}{\Delta T} + \frac{\mu \Delta P \Delta T_0}{\Delta T^2} + \frac{(\mu \Delta P)^2 \Delta T_0}{\Delta T^3} + \circ \circ \circ$$

$$- \frac{\mu_0 \Delta P_0}{\Delta T} - \frac{\mu \Delta P \mu_0 \Delta P_0}{\Delta T^2} - \frac{(\mu \Delta P)^2 \mu_0 \Delta P_0}{\Delta T^3} + \circ \circ \circ$$

$$- \frac{Q}{C_{Po}\Delta T} - \frac{\mu \Delta P \; Q}{\Delta T^2 \; C_{Po}} - \frac{(\mu \Delta P)^2 \; Q}{\Delta T^3 \; C_{Po}} + \circ \circ \circ$$

Rearranging terms:

$$= \frac{\Delta T_0}{\Delta T} + \left[- \frac{\mu_0 \Delta P_0}{\Delta T} - \frac{Q}{C_{Po}\Delta T} + \frac{\mu \Delta P \Delta T_0}{\Delta T^2} \right]$$

$$+ \frac{(\mu \Delta P)^2 \Delta T_0}{\Delta T^3} + \frac{(\mu \Delta P)^3 \Delta T_0}{\Delta T^4} + \circ \circ \circ$$

$$- \frac{\mu \Delta P \; \mu_0 \Delta P_0}{\Delta T^2} - \frac{(\mu \Delta P)^2 \mu_0 \Delta P_0}{\Delta T^3} - \circ \circ \circ$$

$$- \frac{\mu \Delta P \; Q}{\Delta T^2 \; C_{Po}} - \frac{(\mu \Delta P)^2 \; Q}{\Delta T^3 \; C_{Po}} - \circ \circ \circ$$

$$\frac{C_P}{C_{Po}} = \frac{\Delta T_0}{\Delta T} - \left[\mu_0 \Delta P + \frac{Q}{C_{Po}} - \frac{\mu \Delta P \Delta T_0}{\Delta T} \right]\left[\frac{1}{\Delta T} + \frac{\mu \Delta P}{\Delta T^2} + \cdots \right]$$

TABLE II

DATA AND CALCULATED RESULTS

LOW PRESSURE HEATING -- HIGH PRESS. COOLING

Time (min)	0	15	0	15	0	15	0	15	0	15
Hi Press. (psig)	1828	1832	1592	1594	1101	1101	909	909	624	625
Lo Press. (psig)	5.0	5.0	5.6	5.7	5.75	5.75	5.6	5.7	6.2	6.2
Rotameter	27.0	27.0	27.3	27.2	26.6	26.6	27.2	27.2	27.0	27.0
L.P. P (in. Hg)	1.2	1.2								
BYPASS Hi Press. (psig)	1870	1870	1540	1540	1150	1150				
Surge (psig)	54	54	52	52	54	54				
Lo Press. (psig)	0	0	0	0	1.0	1.0				
L.P. out	104 .1	1044.	1047.7	1047.8	1041.6	1041.8	1044.5	1056.3	1041.5	1041.7
L.P. in	846.0	846.5	860.0	858.7	844.7	845.3	866.8	868.3	858.3	858.5
H.P. in	1214.4	1214.6	1216.0	1215.7	1219.4	1 19.1	38.5	12 8.9	1226.1	1226.6
H.P. out	1044.1	1045.7	1046.8	1046.8	1042.7	1043.1	105 .6	1057.	0.	1046.8
H.P.2junc diff.	341.9	342.0	333.4	333.7	354.7	351.8	363.3	363.1	365.5	365.8
c_p/c_{po}	1.1588	1.1526	1.1	.1304	1.110	1.1171	1.0355	1.0355	1.0	1.0

TABLE III

DATA AND CALCULATED RESULTS

LOW PRESSURE COOLING -- HIGH PRESSURE HEATING

Time (min)	0	15	0	15	0	15	0	60	0	15
Hi Press. (psig)	1818	1825	1440	1438	1026	1027	582	589	318	318
Lo Press. (psig)	5.5	5.5	5.5	5.5	5.5	5.5	5.6	5.6	5.5	5.4
Rota.	27.0	27.1	27.0	27.0	27.0	27.0	27.0	26.9	27.0	2.65
L.P. P (in Hg)	1.2	1.2			1.2	1.2				
BYPASS Hi Press. (psig)	1850	1850	1475	1438	1070	1070	630	630	1100	1100
Surge (psig)	52	52	52	52	55	55	56	56	54	54
Lo Press. (psig)	0	0	0	0	0	0	0	0	0	0
L.P. out	1013.3	1013.5	1016.4	1016.2	1024.9	1025.9	1029.4	1026.4	1032.6	1034.6
L.P. in	1211.2	1211.4	1211.0	1210.1	1216.5	1218.0	1214.4	1208.6	1213.8	1215.2
H.P. in	854.2	854.2	851.2	851.2	856.5	850.0	851.7	851.1	864.9	866.2
H.P. out	1015.5	1016.2	1016.5	1016.6	1025.1	1025.6	1031.0	1027.6	1033.4	1035.2
H.P. 2junc.	324.0	325.7	332.8	333.2	336.9	338.9	339.8	341.4	337.0	337.3
	1.2216	1.2152	1.1690	1.1639	1.1374	1.1337	1.0889	1.0821	1.0754	1.0709

TABLE IV

INTERPOLATED VALUES OF c_p FOR AIR AT $20^{\circ}C$

FROM MICHELS' DATA

Pressure atm.	Heat Capacity BTU/lb.mole $^{\circ}F$	c_p/c_{po}
0	6.954	
10	7.06	1.013
20	7.18	1.030
30	7.30	1.047
40	7.41	1.063
50	7.52	1.079
60	7.63	1.095
70	7.75	1.112
80	7.86	1.127
90	7.95	1.141
100	8.06	1.156

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