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RESEARCH AND DEVELOPMENT OF SOLAR-POWERED DESICCANT
REFRIGERATION FOR COLD-STORAGE APPLICATIONS
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by

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1. Introduction

An intermittent solar activated charcoal-methanol adsorption refrigerator has been designed and is now fabricated at AIT. The design of the system is based on p-T-x data determined with our laboratory measuring equipment for one of our charcoal samples (extruded type 207E3) from Sutcliffe Speakman Carbons Ltd., U. K. The first series of laboratory measurements has been carried out on activated charcoal from a local factory in Thailand. Although this local sample was capable of adsorbing and desorbing methanol, large hysteresis was observed during adsorption and desorption at low evaporator pressure. Due to this reason this charcoal has not been used in the design of the system. But a new sample prepared with special care has been acquired from the same local factory and is being tested. If the tests show encouraging results the solar refrigerator will also be tested with the local charcoal.

1.1 Ideal Cycle

The ideal cycle operating this type of system consists two isosters and two isobars. The sensible heating and sensible cooling before and after generation follow two isosters; and the generation and the adsorption process follow the isobars. The cycle can best be illustrated in the p-T-x diagram of Fig.

a. Sensible Heating A_2-G_1

Starting in morning, charcoal having rich concentration of adsorbed methanol is heated by solar energy to a temperature T_{G1} until vapour pressure of methanol becomes equal to the condensing pressure. The concentration of methanol contained in

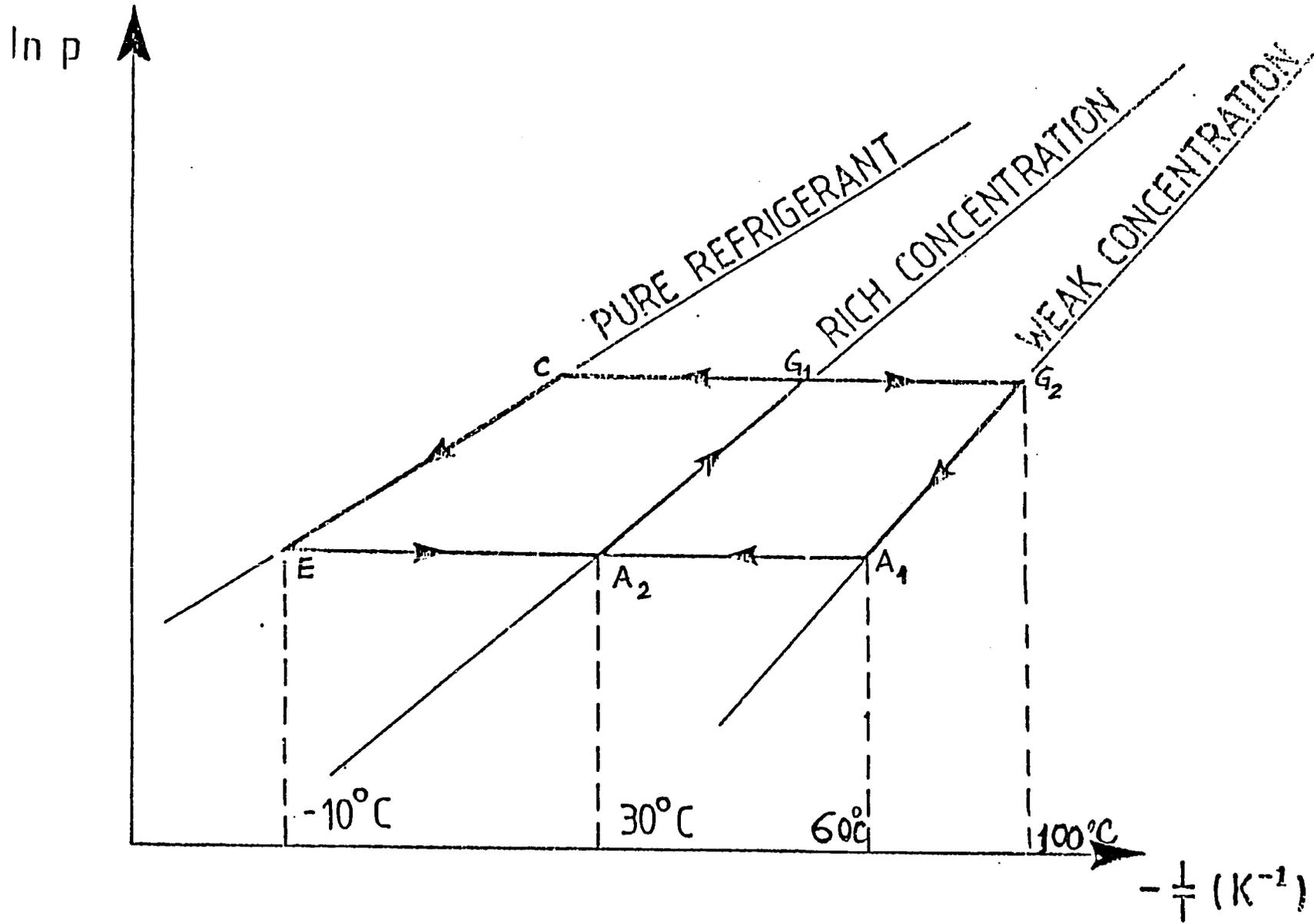


FIG. 1: $\ln P$ v. $1/T$ PLOT FOR THE PROCESS OCCURING IN THE ADSORPTION REFRIGERATOR.

charcoal remains constant and thus the process follows an isoster.

b. Generation Process $T_{G1}-T_{G2}$

During this process refrigerant is driven off at a constant pressure corresponding to the condenser temperature and the adsorbent becomes more and more dilute. This continues until the highest temperature for the day is reached. The refrigerant condensed in the condenser is collected in receiver.

c. Sensible Cooling $T_{G2}-T_{A1}$

The collector/generator is cooled and the pressure of the methanol decreases as it cools until it equals the saturated vapour pressure of methanol at about -10°C . This cooling is an isosteric process.

d. Adsorption $T_{A1}-T_{A2}$

Methanol evaporates at constant pressure and is adsorbed by the charcoal in the collector. Since the adsorption process is exothermic, the heat of adsorption has to be lost by the collector to the ambient air. The methanol takes its latent heat of evaporation from water surrounding the evaporator and ice starts to form. This process starts late evening and lasts until next morning.

1.2 Heat of Adsorption/Desorption

The isosteric heat of adsorption can be determined from the p-T-x characteristics of the pair using the Clausius-Clapeyron

equation as follows:

$$\begin{aligned} H_{\text{ads}} &= R \ln P_2/P_1 / (1/T_1 - 1/T_2) & (1) \\ &= R \times (\text{slope of the } \ln P \text{ vs } -1/T \text{ diagram}). \end{aligned}$$

where,

H_{ads} = Heat of adsorption, kJ/kg of refrigerant,

$P_1 T_1$ and $P_2 T_2$ are pressure/absolute temperature combinations on the same isoster,

R = Gas constant for the methanol vapour, kJ/kg-K.

Based on this principle, the heat of adsorption for each isoster (in Fig. 2) was calculated. It was found to vary from 1097 kJ/kg at 21% concentration to 1944.2 kJ/kg of refrigerant at 7% as the isosters have different slopes. An average value of 1500 kJ/kg of refrigerant has been assumed for the design of the system. The heat of desorption has been assumed equal to that of heat of adsorption.

1.3 Design of the System

The evaporation temperature T_E depends on the applications; for ice production it has to be in the range of -10°C . T_C is the condensing temperature which should be as near to the ambient temperature as heat transfer and economics will allow. Similarly T_{A2} should be as low as possible so that the rich concentration is as high as possible. This maximises the concentration change thus minimizing the quantity of charcoal that must be (wastefully) heated and cooled with the adsorbed refrigerant. But T_{A2} depends on the night-time ambient temperature and also the heat transfer characteristics of the collector. For design calculations it has been assumed equal to 30°C . Thus the point corresponding to the evaporation .

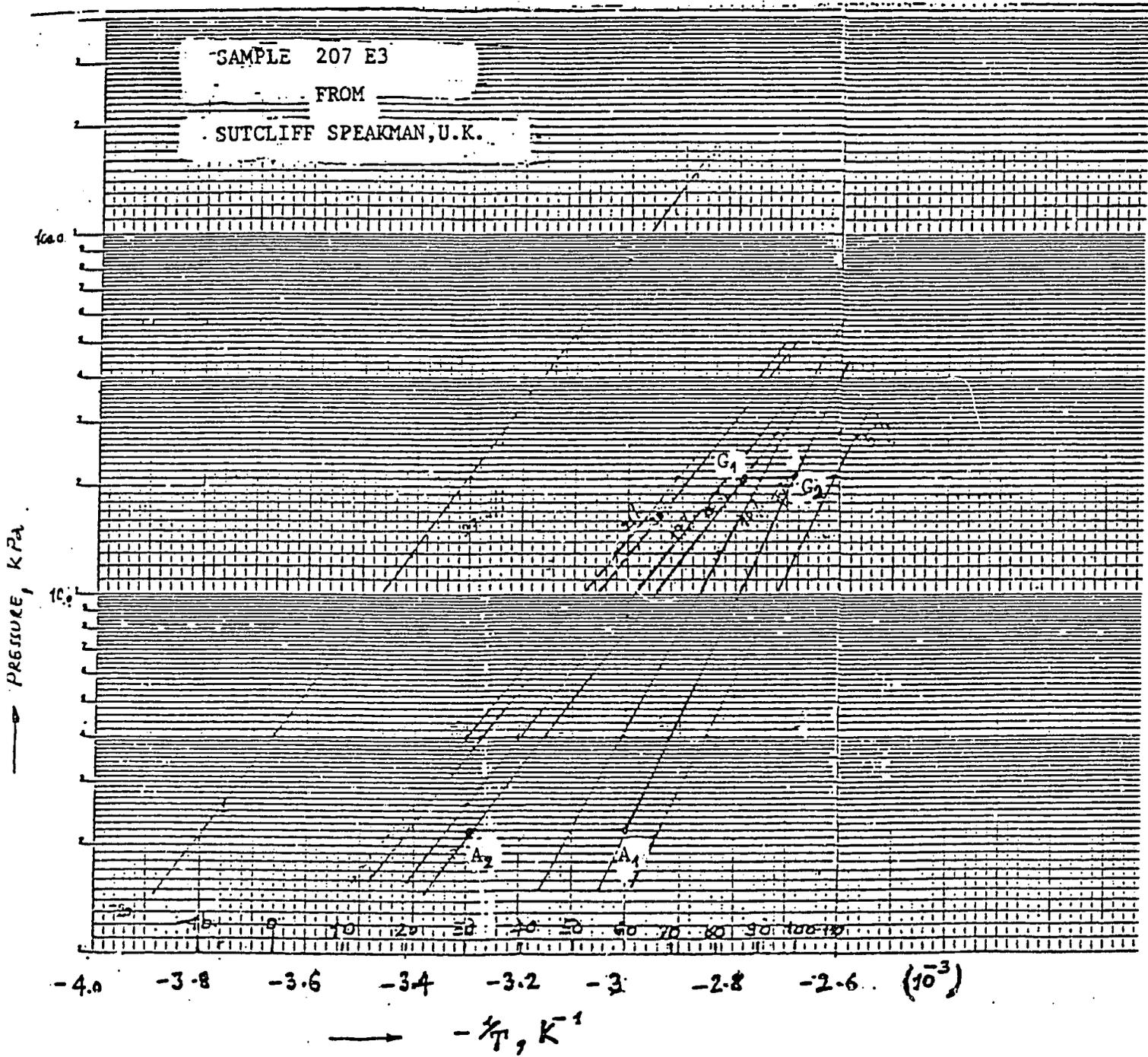


FIG. 2: IDEAL CYCLE IN $\ln P$ v. $1/T$ DIAGRAM.

temperature of -10°C and adsorber temperature of 30°C fixes point A_2 in the p-T-x diagram. The concentration corresponding to it is 15% by weight. This system has a water cooled condenser and condensing temperature is assumed equal to 30°C . Following 15% isoster and condensation pressure corresponding to 30°C , point G_1 can be located which gives generator temperature of 86°C . Assuming a generator temperature T_{G2} equal to 100°C , typical of the flat-plate solar collector and condensation at 30°C , the final concentration of the refrigerant in charcoal is 7%. This gives a concentration change of 8%. Similarly the generator has to be cooled to 60°C before the adsorption starts at pressure corresponding to the evaporation at -10°C . This adsorption process starts late at night and continues until next morning. This fixes the ideal cycle on which the design is based. The corresponding state points are shown in Fig. 2.

1.4 Heat Transfer in the Cycle

Let m_C and m_A be the mass of the collector/generator and adsorbent (charcoal) contained in the collector respectively. Let C_C and C_A be the respective specific heat of the collector material and the charcoal. The specific heat of the adsorbed phase has been assumed equal to that of the liquid phase and denoted by C_R . Assuming the collector, charcoal and the adsorbed phase are at the same temperature, the heat to be supplied during the sensible heating Q_S from temperature T_{A2} to T_{G1} can be estimated as:

$$Q_S = (m_C C_C + m_A C_A + m_A x_{\text{Conc}} C_R) (T_{G1} - T_{A2}) \quad (2)$$

where x_{Conc} is the rich concentration, kg of methanol/kg of charcoal.

The desorption or generation process starts once the temperature reaches to T_{G1} and continues until the highest temperature of the day T_{G2} is reached. After this heat loss from the collector becomes greater than the heat input and generation stops. Thus heat supplied during the generation process includes sensible heating of the collector, charcoal, the desorbed phase and also the heat of desorption. Assuming the sensible heating of the desorbed phase is equal to that needed to raise the mean mass of the adsorbate during desorption to the final temperature T_{G2} , the heat needed to be supplied Q_d can be approximated as:

$$Q_d = ((m_C C_C + m_A C_A + m_A C_R (X_{Conc} + X_{Dil})/2) (T_{G2} - T_{G1}) + m_A H_{Des} (X_{Conc} - X_{Dil})) \quad (3)$$

where X_{Dil} is weak concentration at the end of generation.

The total heat that has to be supplied during the sensible heating and the generation process is approximately the sum of the quantities given by equations (2) and (3).

The heat that has to be lost from the collector during the sensible cooling process T_{G2} to T_{A1} can be calculated as:

$$Q_c = (m_C C_C + m_A C_A + m_A X_{Dil} C_R) (T_{G2} - T_{A1}) \quad (4)$$

The heat needed to be rejected from the collector during adsorption process can be approximated as:

$$Q_a = ((m_C C_C + m_A C_A + m_A C_R (X_{Conc} + X_{Dil})/2) (T_{A1} - T_{A2}) + m_A H_{Ads} (X_{Conc} - X_{Dil})) \quad (5)$$

The useful cooling Q_C that can be produced by this system during night can be given as:

$$Q_C = m_A (X_{\text{Conc}} - X_{\text{Dil}}) (h_{ge} - h_{fc}) \quad (6)$$

where,

h_{ge} = Specific enthalpy of the refrigerant vapour leaving the evaporator, kJ/kg,

h_{fc} = Specific enthalpy of the refrigerant vapour leaving the condenser, kJ/kg.

But Q_C can be approximated as:

$$Q_C = m_A (X_{\text{Conc}} - X_{\text{Dil}}) (L - C_R (T_C - T_E)) \quad (7)$$

where L is the average latent heat of the refrigerant liquid leaving the evaporator, kJ/kg.

Thus the thermal coefficient of performance of the system is given as follows:

$$\text{COP}_{\text{th}} = Q_C / (Q_s + Q_d) \quad (8)$$

But the overall coefficient of performance the system will be equal to the thermal coefficient of performance multiplied by efficiency of the solar collector/generator.

1.5 Design of the Components

1.5.1 Solar collector

The solar collector consists of an array of 15 copper tubes $2\frac{1}{8}$ " (53.97 mm O.D.), 0.528" (1.47mm) thick and 1.2 m long. This configuration provides an effective collection area of 0.9715 m². Although calculations were done on the different materials such as brass, stainless steel, aluminum tubings and also aluminum sections, M-type copper tube was chosen for the following reasons:

- availability of the material and fittings
- ease of fabrication such as by soft soldering, silver brazing etc.
- low specific heat but high conductivity of the material of the tubing.

Each copper tube contains a centrally placed perforated gauge tube of 3/8" diameter which is connected to the common header at the bottom of the collector. This was done to ensure even distribution of the methanol vapour in the charcoal and to reduce the pressure drop. Each tube containing charcoal has screw-type caps at one end and flare joints at the other end. Thus tubes can be individually dismantled from the main assembly to facilitate change of charcoal, if necessary. The arrangement is shown in Fig. 3. The collector as fabricated has 37 kg of copper in tubes containing charcoal, header and gauge tubes, 3.5 kg of brass in end caps and nipples; and 1.63 kg in stainless steel caps. The total weight of the collector grid is 42.18 kg and it can contain about 17.46 kg of charcoal.

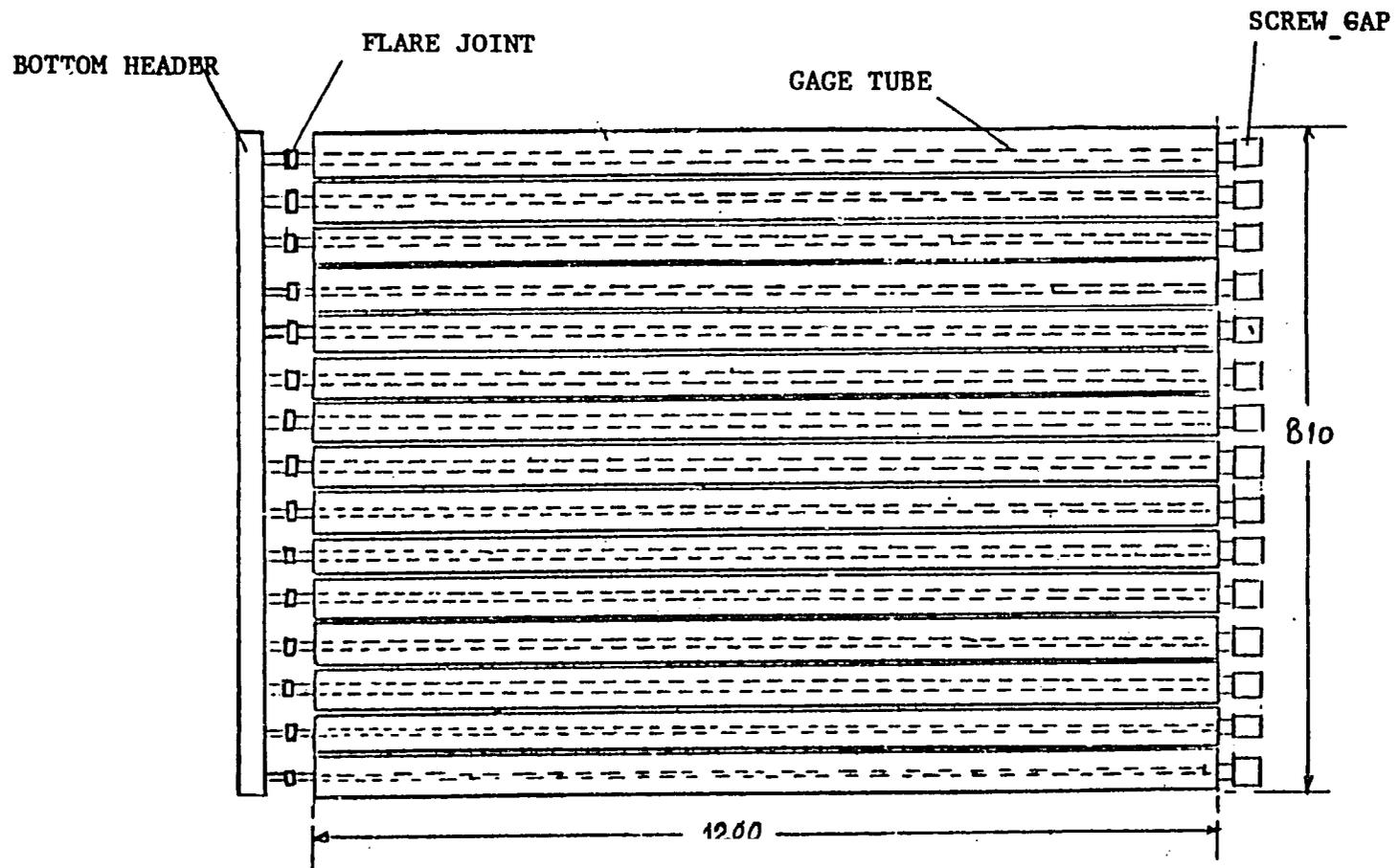


FIG. 3: SOLAR COLLECTOR GRID.

Dimensions are in mm.

The collector has a 4mm thick glass cover at a distance of 32mm from the top surface of the collector tubes. One glass cover was chosen based on the previous experience at AIT (Exell and Kornsakoo, 1981). The top surface of the collector array is covered with selective MAXORB Solar Foil (INCO SELECTIVE SURFACES, U.K.) coated with a special pressure-sensitive adhesive which can be used at high temperatures.

The assembly is housed in a galvanized iron sheet case of size 1.5m by 1 m. Polyurethane foam is used for insulating the pipes. The insulation fills in the casing and forms a mould 90mm thick. The back insulation of the collector can be opened by lowering the back of the casing which is hinged at the bottom and bolted in position at the top. The panel is tilted at an angle of 14° facing south. This collector at 30 to 35 % collection efficiency can provide heat needed to desorb approximately 1.4 kg of methanol and has sufficient area to dissipate the heat of adsorption when back of the collector is lowered. Detailed calculations are shown in appendix.

1.5.2 Condenser

The design of the condenser is based on the condensation of 1.4 kg of methanol vapour at 30°C to methanol liquid at the same temperature. The total amount of heat to be released in the condenser can be calculated as follows:

From the methanol property tables (Liney, 1982), the enthalpy of the methanol vapour at 30°C is 1776.5 kJ/kg and the enthalpy of the methanol vapour at the same temperature is 613.85 kJ/kg. So, a total of 1627.7 kJ has to be released in the condenser. Supposing condensation takes place during the period of four

hours, the heat rate is 113.03 W.

The surface area for heat transfer can be calculated as:

$$A = 113.03/U \Delta t, \quad \text{m}^2 \quad (9)$$

where U is the overall heat transfer coefficient in W/m²-K and Δt is the temperature difference in degrees.

The overall heat transfer coefficient by natural convection in the static water tank with some fouling was estimated to be 170 W/m²-K from previous experience (Exell and Kornsakoo, 1981). Assuming 5°C temperature difference between the vapour and the cooling water, a heat transfer area of 0.133 m² is required. A prefabricated car radiator with copper fins has been adapted and used as condenser in this design. This provides an enormous heat transfer area, which is estimated to be approximately 7 m².

Since the total heat to be released in the static water tank is about 1627.7 kJ, and allowing a temperature rise of 1.5°C, the amount of water needed in the tank is 260 kg. The water tank is made up of mild steel sheets and has dimensions 1m x 0.7m x 0.4 m; it can contain 279 kg of water.

1.5.3 Receiver

The methanol receiver holds about 2.5 kg of methanol liquid, which amounts to 14.3 % of methanol cycled. Although this system is expected to cycle only about 8% but this provision was done to test other charcoals without changing the receiver. The methanol is contained in a brass container 150 mm in diameter and 180 mm long.

1.5.4 Evaporator

The design of the evaporator is based on the following assumptions:

1. The amount of methanol to be evaporated is 1.4 kg.
2. The refrigeration process takes 12 hours.
3. The evaporator temperature is -10°C .
4. The specific latent heat of evaporation of the methanol vapour between 30 to -10°C is 1189 kJ/kg.
5. The conductivity of the ice layer is 2.2 W/m-K.

It follows from the above assumptions that the maximum amount of ice that could be produced by this system from water initially at 30°C is about 3.3 kg. The total cooling produced is 1511 kJ and the heat flow rate required is 35 W. This heat flow rate must be maintained by the freezing water until the last bit of ice is formed in the cycle. Thus,

$$35 = 2.2 A_E \Delta T/x \quad \text{W} \quad (10)$$

where,

A_E = Heat transfer area of the evaporator, m^2 ,

ΔT = Temperature difference between evaporator and water surrounding the evaporator, $^{\circ}\text{C}$,

x = Thickness of ice, m.

$$\text{Thus from equation (9),} \quad A_E/x = 1.59 \text{ m} \quad (11)$$

The specific volume of ice at 0°C is 0.001091 m^3/kg and the total volume of 3.3 kg of ice is 0.0036 m^3 which gives:

$$A_E \times = 0.0036 \text{ m}^3 \quad (12)$$

Solving equations (11) and (12), the surface area of heat transfer is equal to 0.07566 m^2 and the maximum thickness of the ice is 47.5mm. After several trials a trapezoidal section has been selected for the evaporator. This section provides effective surface area of about 0.13065 m^2 , and the effective thickness of ice will reduce to 28mm. The evaporator was made out of the brass sheet and has the total mass of 1.61 kg. The volume of the methanol liquid that can be contained in the evaporator is about 5% of the total that this system is expected cycle. This configuration will also allow easy removal of ice. The receiver/evaporator assembly is shown in Fig. 4 and the sectional view of the whole refrigeration system is shown in Fig. 5.

2.1 Pressure Drop in Connecting Tubes and Charcoal

The worst possible case that might be encountered is a temperature of 260 K (-13.15°C) during evaporation, the specific volume of methanol vapour at this temperature is $39.49 \text{ m}^3/\text{kg}$ and the corresponding pressure is 1700 Pa. Since the total amount of vapour to be cycled is about 1.4 kg in twelve hours, the average mass flow rate is $3.24 \times 10^{-5} \text{ kg/s}$. The average velocity of the methanol vapour V can be calculated as:

$$V = \dot{m} \nu / A_f \quad (13)$$

where,

\dot{m} = Methanol vapour rate, kg/s,

ν = Specific Volume, m^3/kg ,

A_f = Cross-section area of flow, m^2 .

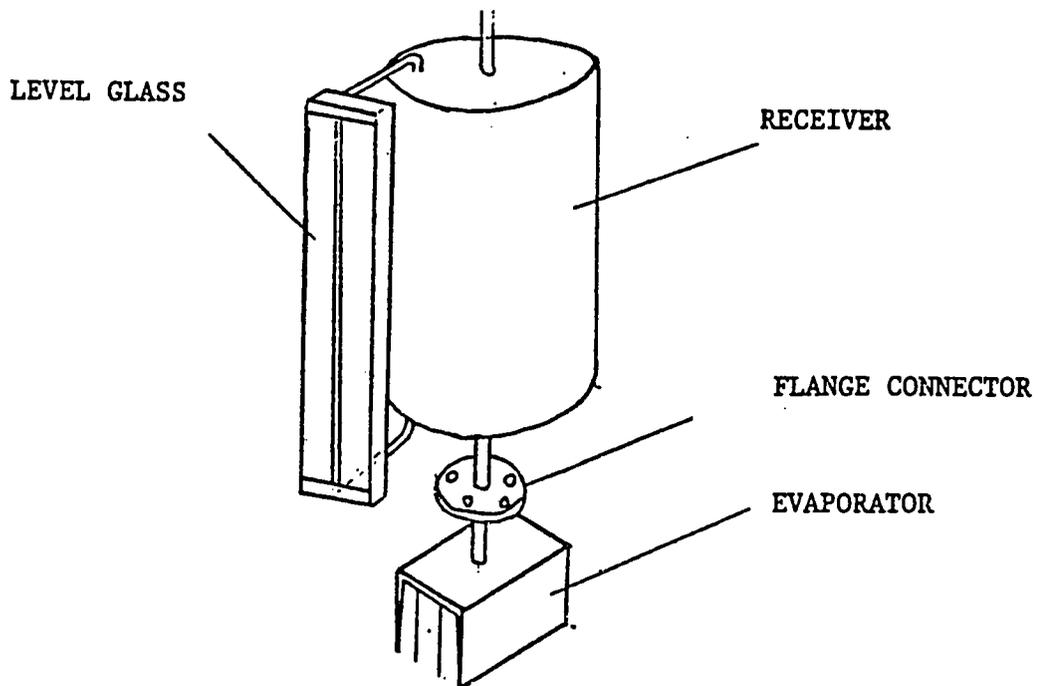


FIG. 4: RECEIVER/EVAPORATOR ASSEMBLY.

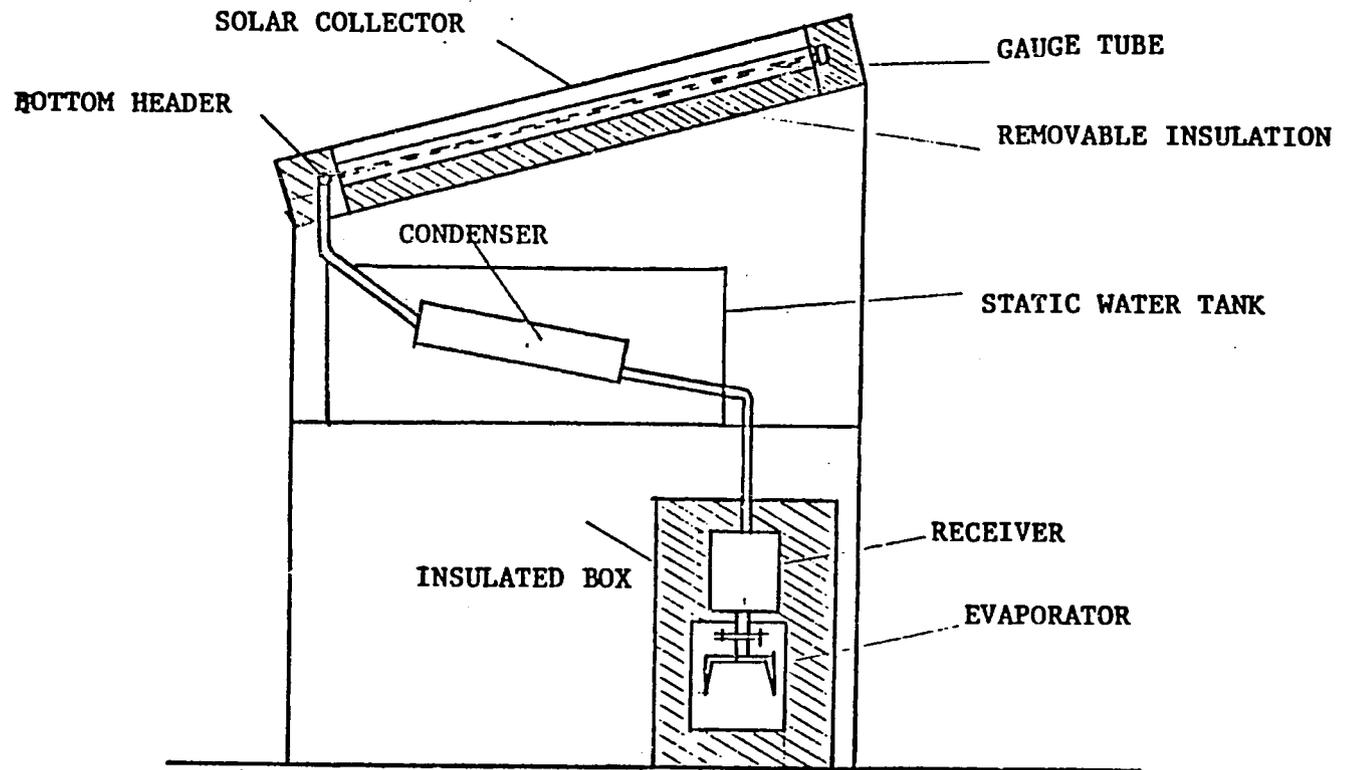


FIG. 5: THE SECTIONAL VIEW OF THE SOLAR REFRIGERATOR.

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To keep the pressure drop in the connecting pipe low a smooth drawn 1³/₈" (32.8mm I.D.) copper tube has been chosen. This tube has a flow area of 0.0008296 m² and the average velocity of vapour flow in the tube is 1.54 m/s. Assuming the average viscosity of the methanol vapour μ equal to 10.1×10^{-6} Ns/m², the Reynold's number R_N is given as:

$$R_N = 4 \dot{m} / \pi D_i \mu \approx 126 \quad (14)$$

where,

D_i = Internal diameter of vapour transport tube, m,

Since the Reynold's number is less than 2000, the flow is laminar and the Fanning's friction factor f is equals to $16/R_N$ (Perry and Chilton, 1973), and the pressure drop ΔP (Pa/m) in the connecting tubes can be calculated as:

$$\Delta P = 1/2 v^2 (f/D_i \rho) \quad (14)$$

For the 1 3/8" connecting tubes, the pressure drop is equal to 0.1161 Pa/m and pressure drop in gauge tubes is 1.9 Pa/m, which is satisfactory.

The flow is laminar and the pressure drop in the bends is small but the bends and sudden changes in the internal diameter of flow have been kept to minimum.

Since gauge tubes have been used in this system, the effective thickness of charcoal in the tube in the radial direction is only 21 mm and the pressure drop in the charcoal will be negligible.

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APPENDIX

Table 1: Properties of different materials used in the design.

<u>Methanol Properties</u>	
Average Specific Heat of Liquid	2.55 kJ/kg-K
Average Density of Liquid	791.00 kg/m ³
Average Latent Heat of Evaporation of Methanol Vapour	1189.00 kJ/kg
Cooling produced by Methanol Vapour at -10°C from Liquid initially at 30°C	1087.0 kJ/kg
Average Heat of Adsorption/Desorption	1500.0 kJ/kg
<u>Charcoal Properties</u>	
Bulk Density	500.0 kg/m ³
Average Specific Heat	0.7 kJ/kg-K
<u>Other Properties</u>	
Average Specific Heat of Copper	0.381 kJ/kg-K
Average Specific Heat of Brass	0.385 kJ/kg-K
Average Specific Heat of Steel	0.470 kJ/kg-K

Design Calculations

The solar collector consists of 37 kg of copper, 3.5475 kg of brass and 1.63 kg of stainless steel. The total amount of charcoal that can be filled in the generator/collector is 17.46 kg. From the p-T-x diagram, it is clear that the rich concentration of methanol is 15% and the weak concentration will be 7% if the generator temperature reaches 100°C. If the system follows the ideal cycle, the collector, charcoal and the methanol adsorbed in charcoal have to be sensibly heated to 86°C. Thus the heat to be supplied during the sensible heating process from initial temperature of 30° to 86°C is:

$$\begin{aligned} Q_s &= (37 \times 0.3831 + 3.5475 \times 0.385 + 1.63 \times 0.47) (86 - 30) + \\ &\quad (17.46 \times 0.7 + 17.46 \times 0.15 \times 2.55) (86 - 30) \\ &= 1971.59 \text{ kJ} \end{aligned}$$

Similarly, the heat supplied during the generation process from a temperature of 86°C to the maximum generator temperature of 100°C when the concentration of methanol in the charcoal decreases to 7% is:

$$\begin{aligned} Q_d &= (37 \times 0.3831 + 3.5475 \times 0.385 + 1.63 \times 0.47 + 17.46 \times \\ &\quad (0.15 + 0.07)/2 \times 2.55) (100 - 86) + 17.46 \times 1500 \\ &\quad \times (0.15 - 0.07) \\ &= 2563.16 \text{ kJ} \end{aligned}$$

So the total heat to be supplied by the collector is the sum of Q_s and Q_d and is equal to 4534.75 kJ. The total heat needed to heat the container from the state A_2 to G_2 is 1141.46 kJ which is approximately 25% of the total heat that has to be supplied to the system.

If we assume the average efficiency of the solar collector equal to 30%, the total amount of solar radiation incident on the surface of the collector has to be 15.115 MJ. Since the effective collection area of the collector is 0.9715 m^2 , the solar radiation incident on the surface of the solar collector by the end of generation has to be about 15.6 MJ/m^2 . But if the efficiency of the solar collector is 35%, the above amount reduces to 13.3 MJ/m^2 . Since the average solar radiation for Bangkok is about 16 MJ/m^2 , the peak being approximately 20 MJ/m^2 , the solar collector has sufficient area to supply the required amount of heat to the system.

If we neglect the thermal mass of the receiver, the evaporator, and heat leak, the net cooling produced by the evaporation of about 1.4 kg of methanol is 1521 kJ. Thus the thermal coefficient of performance of the system is 33.5% but the overall coefficient of performance will be between 10 to 11.7%. But the thermal mass of the receiver, evaporator and the heat leak to the receiver/evaporator box will reduce this performance further. The overall coefficient of performance of the system is expected to be around 10% and the system is expected to produce about 3 kg of ice from water initially at 30°C .

Estimation of Heat Loss by the collector during Adsorption

After the generator temperature has passed its maximum for the day, the back insulation of the collector can be lowered to cool the generator and the charcoal contained in it. If we assume half of the circumference of each tube to be exposed to the ambient air the total heat transfer area is approximately 1.5 m^2 . The heat loss from the back of the collector will be

mainly by three modes of heat transfer (i) natural convection, (ii) forced convection due to wind, and (iii) radiation.

(i) Heat Loss by Natural Convection

For a heated plate facing downward the Nusselt number Nu is given by Holman (1981) as follows:

$$Nu = 0.56 (Gr Pr \cos \theta)^{1/4}$$

But the Grashof number $Gr = g \beta (T_w - T_{\infty}) x^3 / \nu^2$

where,

β = Volume coefficient of expansion = $1/T$, where T is in Kelvins,

T_w = Wall temperature, K,

T_{∞} = Free stream temperature, K,

x = Length of the plate, m,

$\nu = \frac{\mu}{\rho}$ = Kinematic viscosity of air, m^2/s ,

μ = Dynamic viscosity, $kg/m-s$,

Pr = Prandl number = $\rho c_p \nu / k$, and $\alpha = k / \rho c_p$

k = Conductivity of air, W/m-K,

α = Diffusivity, m^2/s

θ = Angle made by the heated plate with the vertical, degrees.

In the above equation T is calculated as:

$$T = T_{\infty} + 0.25 (T_w - T_{\infty})$$

Other properties are evaluated at temperature T_1 which is given as:

$$T_1 = T_w - 0.25 (T_w - T_{\infty})$$

Assuming the wall temperature equal to 35°C and the free stream temperature of 30°C, β is equal to 1/304.4 and the fluid properties have to be evaluated at 29°C. This gives Gr Pr equal to 687×10^6 . Thus the convective heat transfer coefficient can be given as:

$$h_c = Nu k/x$$

The above equation gives a convective heat transfer coefficient due to natural convection h_c for 1.2 m long tube equal to 1.39 W/m²-K.

(ii) Heat Transfer due to Wind

Assuming the average wind speed of 1 m/s and applying the empirical relation of Watmuff, Charter and Procter (Exell, 1982), the convective heat transfer coefficient due to wind flowing over the plate is:

$$h_w = 2.8 + 3.0 = 5.8 \text{ W/m}^2\text{-K}$$

Thus the total heat lost by natural convection and the wind blowing across the exposed area of 1.5 m², assuming the average temperature difference of 5°C between the back of the collector plate and ambient is equal to 53.92 W.

(iii) Radiation

Assuming a temperature difference of 5°C between the collector and the ambient at 30°C, and emissivity of the back of

the coated collector equal to 0.8, the heat transfer through the surface area of 1.5 m^2 is:

$$q_r = \epsilon \sigma (T_w^4 - T_a^4) A$$

$$\begin{aligned} q_r &= 0.8 \times 5.669 \times 10^{-8} (308.15^4 - 303.15^4) \times 1.5 \\ &= 39.5 \text{ W.} \end{aligned}$$

Thus the total heat lost by the back of the collector by above three modes of heat transfer is equal to 93.4 W. If the time to release the heat is 12 hours, the total heat that can be lost is equal to 4036 kJ. The heat to be lost during adsorption period is equal to 2731 kJ. Thus the back of the collector can dissipate the heat during adsorption.

APPENDIX I

AN ACTIVATED CHARCOAL - METHANOL REFRIGERATOR OPERATED BY STEAM

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1. INTRODUCTION

A Charcoal-Methanol refrigerator, working on the principle of intermittent adsorption, with steam as the powering medium for heating the charcoal has the following advantages.

- a. It can be made independent of the solar cycle i.e. production of ice does not depend on whether it is night/day.
- b. Steam can be produced using any waste heat source or even the waste heat (for eg. flue gas) can be used directly.
- c. As a corollary to a., the optimum cycling time can be studied.

With these points in view, a Charcoal-Methanol refrigerator has been built already and currently it is in the commissioning stages. This report highlights the design aspects and a few constructional details also. Also the proposed adsorption cycle is discussed.

2. MATERIALS

The activated charcoal (AC) being used is the model 207C, supplied by Sutcliffe Speakman Co. Ltd, UK. The AC is a coconut shell based carbon. In the months of Feb and March this year, tests were carried out in our laboratory to determine the P-T-x (pressure, temperature and concentration) characteristics of the AC (Fig 1). Though these tests are not extremely accurate, the data obtained is accurate enough to design and estimate performances like COP, for a refrigeration application. Also in Appendix A.1 are listed the properties of 207C, supplied by Sutcliffe Speakman Co. Ltd.

3. THE REFRIGERATION SYSTEM

The schematic diagram of the complete system is shown in Fig 2. The main components of the system are the adsorber, the evaporator, a condenser and the steam generator/boiler.

3.1 The adsorber

The adsorber is a 1.8 metre long column consisting of 3 concentric cylinders. The innermost pipe, 2" in diameter, provides passage for steam for heating the AC or for the cooling water to cool the hot AC. The next outer pipe is 4" in diameter and between the 2" and the 4", the AC is housed. Both these pipes are made out of hard drawn copper. The outermost pipe is 8" in diameter and is made of HDPE. This encircles the polyurethane foam insulation provided for the AC. Fig 3. illustrates the adsorber in detail.

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3.2 The Evaporator

The evaporator with a level gauge for the methanol level, has been sized on a 4 Kg charcoal basis. The evaporator area is 0.58 sq metre and the estimated quantity of ice produced is about 10 Kg per day. The evaporator is made up of hard drawn copper tubes of 1-3/8 " dia. The ice box capacity is 18 litres and the insulation provided is polyurethane foam. Fig 4 illustrates the details of the evaporator and Fig 5. gives the features of the ice box.

3.3 The Condenser

This is provided to cool the methanol vapour coming over when the AC is heated. The condenser is a water cooled one, with the room temperature water at the inlet on the tube side. The designed rise in the water temperature is about 3 degrees C with a flow rate of 6.5 lit/min. The methanol on the shell side condenses at 30 degrees C and falls into the receiver cum evaporator. Please refer to Fig 6.

3.4 The Steam Generator/Boiler

This is simply a vessel made of a brass tube 75mm in dia and 500 mm in length. An electric heater (rating 3 kW) provides the input for steam production. An electric heater is being used initially to estimate accurately the performance of the evaporator. After certain initial tests other ways of heating AC will be investigated. Polyurethane foam insulation is provided.

4. CYCLE ANALYSIS

When the design and fabrication was started in Dec 1986/Jan 1987, the PTX chart for the AC presently used was not available. So the sizing up of the refrigerator was based on the relations of pressure, temperature and concentration developed by the French researchers. After testing the carbon 207C in our laboratory we find that the design based on the French equation is still valid.

For Charcoal-methanol mixtures the PTx relation used is as follows :

$$X = .322 * \exp [-3.22 \times 10^{-7} * (T_c \ln P_{st}/P_{smt})^{2.1}] \dots(1)$$

where X = concentration (Kg methanol / Kg Charcoal)
T_c = temperature of the charcoal (K)
P_{st} = saturation pressure at the charcoal temperature
P_{smt} = saturation pressure at the methanol temperature
(in condenser or evaporator)

Point 1 : $T_m = -10^\circ\text{C}$ $T_c = 30^\circ\text{C}$ $x = 0.185$ (from eqn 1)

Point 2 : $x = 0.185$; to find T_c
Using eqn 1 for both points 1 & 2 and then equating the x values (since we are going up an isotherm) we have $T_c = 75.4^\circ\text{C}$.

By the Clausius-Clapeyron Equation

$$\Delta H_{\text{ads}} = \frac{R \ln P_2/P_1}{1/T_1 - 1/T_2} \quad \text{where 1 and 2 are same conc.}$$

$$R = 8.314/32.04 \quad T_1 = 30 ; T_2 = 75.4 ; P_2 = 20.19 ; P_1 = 2.05$$

$$\Delta H_{\text{ads}} = 1380 \quad \text{kJ/kg methanol.}$$

Point 3 : $P = 20.19$ $T = 373.15 \text{ K}$ $P_{\text{sat}}(373) = 340 \text{ kPa}$
 $x = 0.08028$

Point 4 : As in point 2, we can calculate $T_4 = 48.6^\circ\text{C}$.

$$\Delta H_{\text{ad}} = 1386 \text{ kJ/Kg methanol.}$$

4.1 Heat Input per Kg Methanol driven off

Concentration change during cycling $x = .18496 - .08028 = .1047$

or it is $(1/.1047 = 9.55)$ Kg charcoal / kg Meth driven off.

- i. Heat to charcoal (1 to 3) = $9.5 * .7 * 70 = 468 \text{ kJ}$
- ii. Heat to Methanol (1 to 2) = $1.766 * 2.55(75.4-30) = 204 \text{ kJ}$.
- iii. Heat to liquid methanol in going from 2 to 3
Assuming linear variation of liquid MeOH with temperature
 $Q = C_p (T_3 - T_2) * \text{mean mass of MeOH}$
 $= 2.55 * 24.6 * 9.55 (.185 + .080)/2 = 79 \text{ kJ}$.
- iv. Heat of desorption = 1380 kJ .

Total useful heat input = $2131 \text{ kJ / kg Methanol driven off}$.

4.2 Cooling per Kg of Methanol driven off

Initial state is liquid methanol at 30°C
Final state is Methanol vapour at -10°C .

Approximately $h = 1189 - 40 * C_{pl} = 1087 \text{ kJ/Kg}$.

$$\text{Ideal COP} = 1087 / 2131 = .51$$

This does not take into account the heat losses from the various system components and also the thermal mass of the adsorber.

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ln P vs -1/T plot

Charcoal: 207C
Date: Feb-15 week

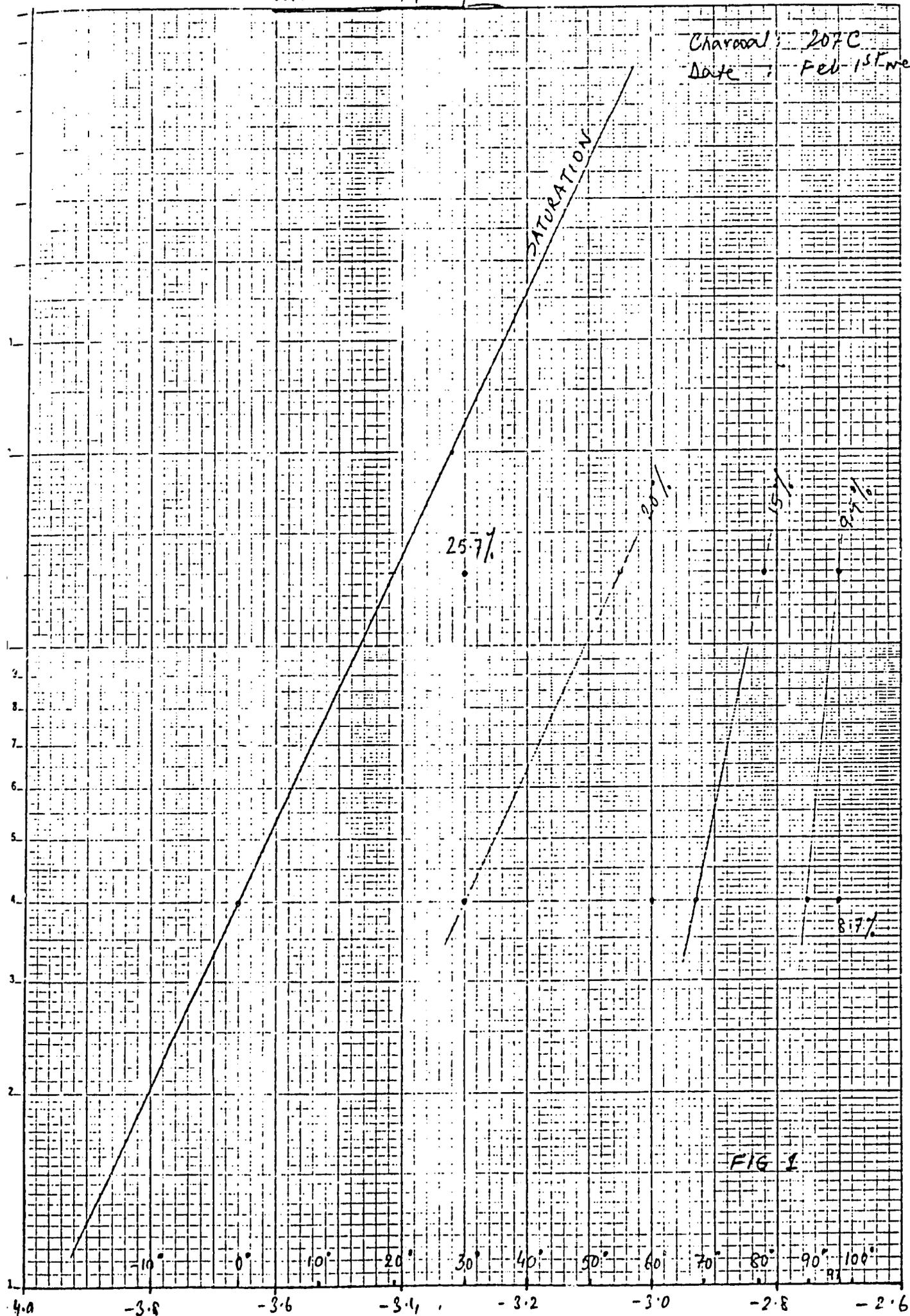
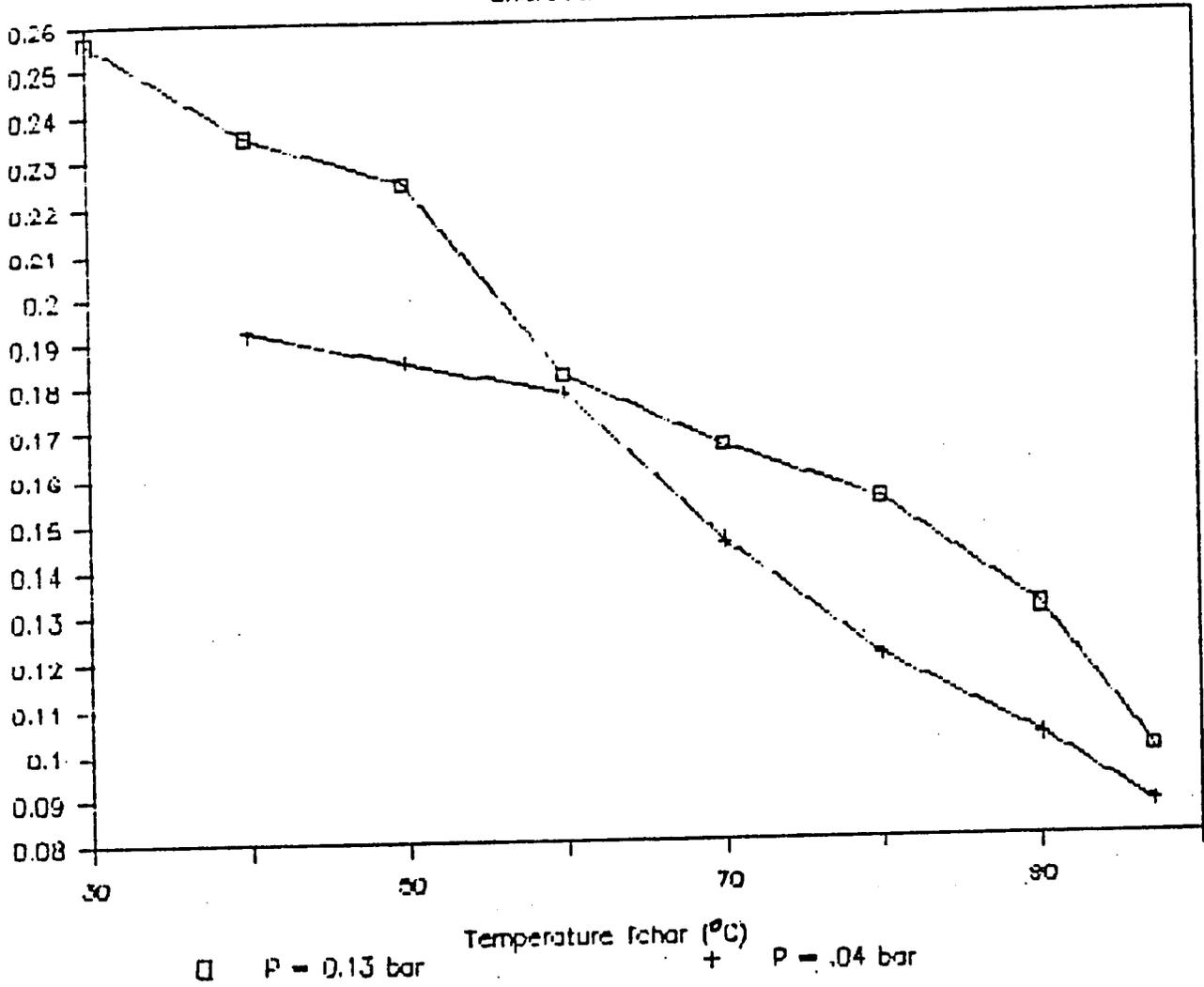


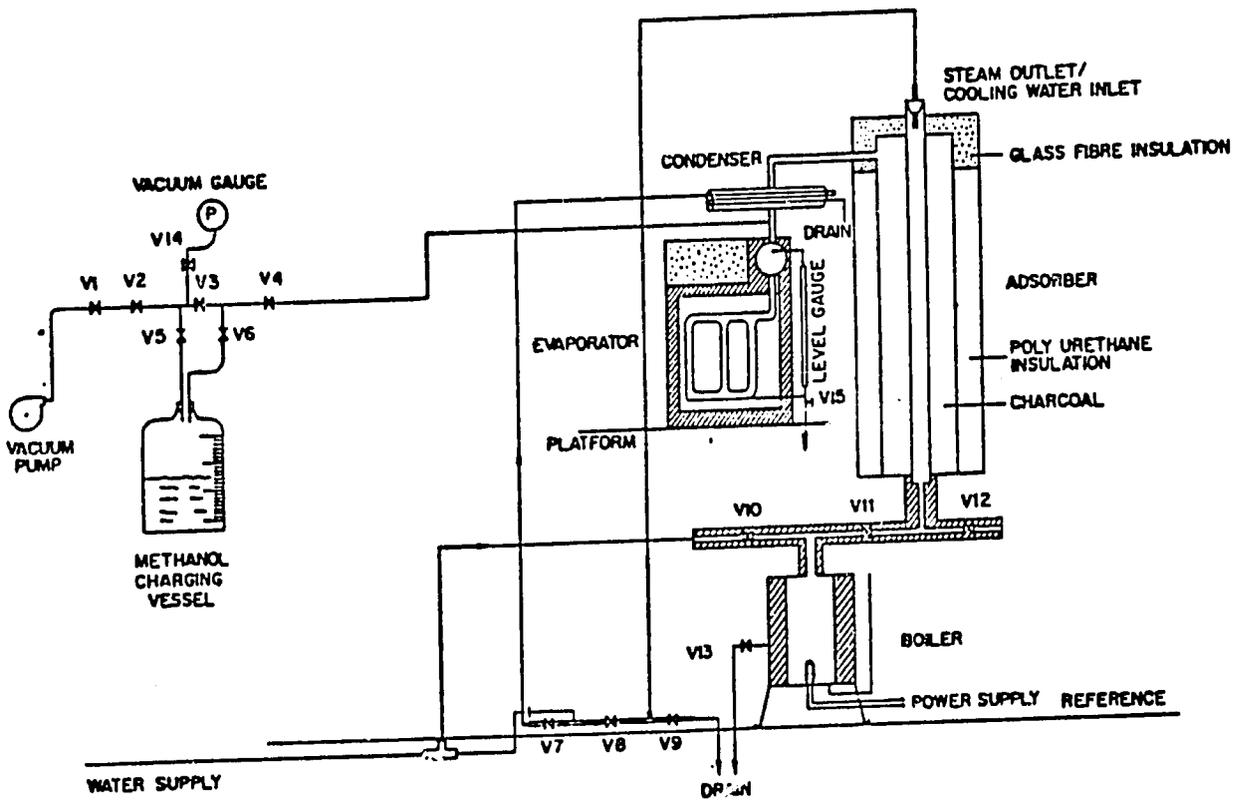
FIG 1

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Concentration X vs Tchar

Charcoal = 207C





VALVES

- V1, V2 - ISOLATION VALVES FOR VACUUM PUMP
- V3 - ISOLATION VALVE FOR PUMP AND PRESSURE GAUGE
- V4 - ISOLATION VALVE OF REFRIGERATION SYSTEM
- V5 - VALVE FOR CHARGING METHANOL INTO BOTTLE
- V6 - VALVE FOR CHARGING METHANOL INTO SYSTEM
- V7 - VALVE FOR WATER INLET TO CONDENSER

- V8 - OPENED IF COOLING WATER REQD. FOR CHARCOAL
CLOSED WHILE CHARCOAL IS HEATED
- V9 - VALVE FOR STEAM OUTLET
- V10 - WATER INLET VALVE FOR BOILER
- V11 - ISOLATION VALVE FOR BOILER
- V12 - COOLING WATER OUTLET FOR ADSORBER
- V13 - BOILER DRAIN VALVE
- V14 - ISOLATION VALVE FOR PRESSURE GAUGE ONLY
- V15 - EVAPORATOR DRAINAGE VALVE

FIG. 2 SCHEMATIC DIAGRAM OF THE REFRIGERATOR

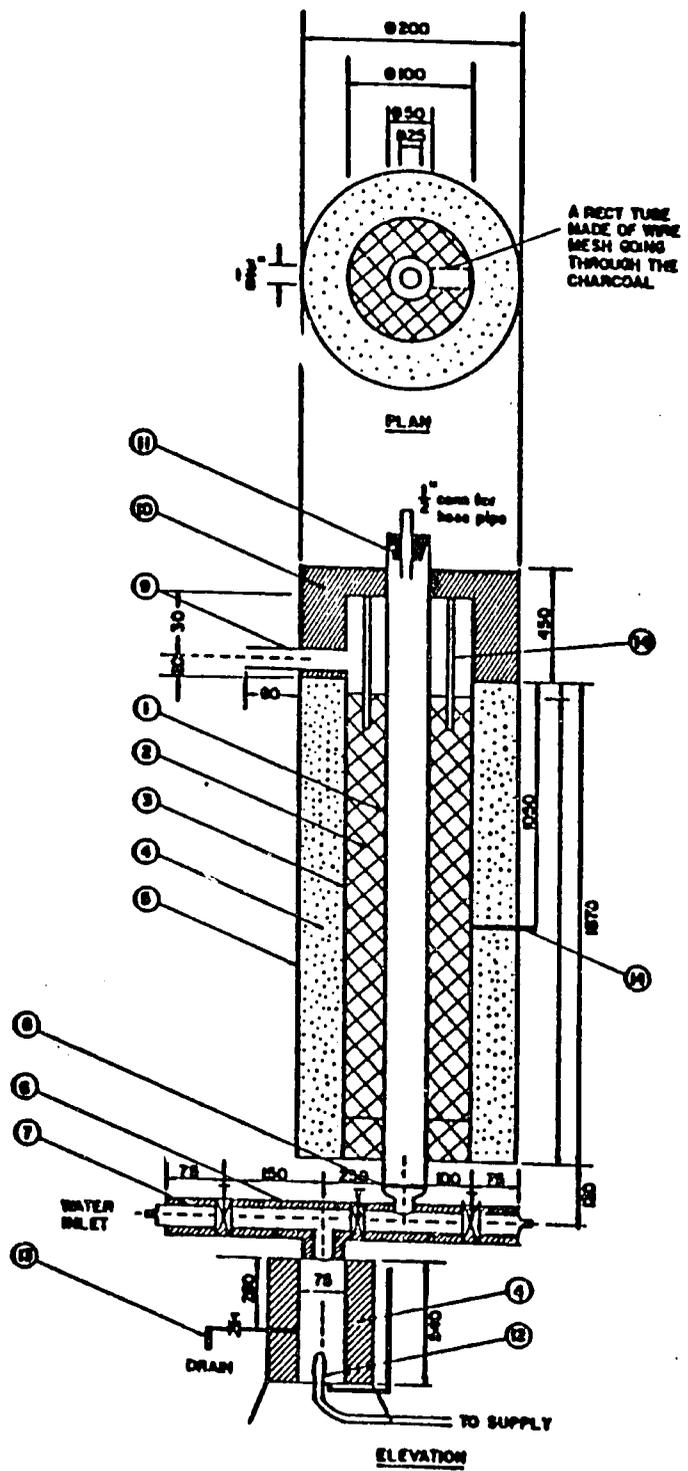
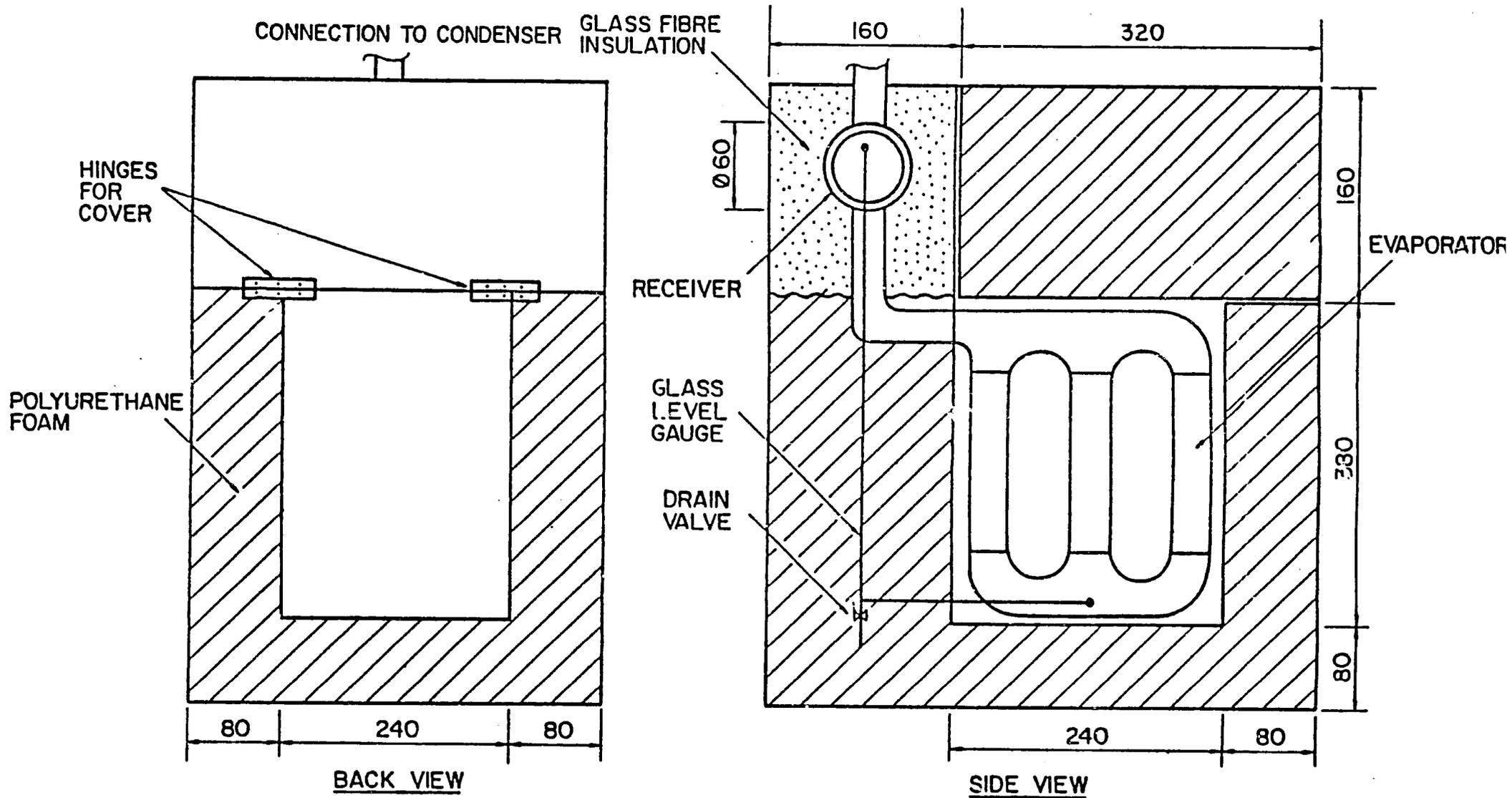


FIG. 3 GENERAL ASSEMBLY OF ADSORBER WITH BON FR

PART DETAILS

PART NO.	MATERIAL	DESCRIPTION
1	Copper	50 mm dia , 1.5 mm thk steam/cooling water
2	Act. Charcoal	Model No. 207 C
3	Copper	Adsorber outer tube , 100 mm dia , 2.4 mm thk.
4	Polyurethane	Insulation
5	PVC	200 mm dia , to hold insulation
6	Rubber foam	Insulation , 10 mm thk
7	GI	1" pipe connection to boiler
8	GI	Reducer 2" to 1" braced to ①
9	Copper	1 $\frac{3}{8}$ " (35 mm) outlet from adsorber (connected to evaporator) , 1.5 mm thk
10	Glass fibre	Top insulation
11	Rubber	Rubber socket glued to top for steam outlet
12		Elec. heater 3000 W rating
13	GI	Drain pipe 12.5 mm dia with valve
14	Copper	$\frac{1}{4}$ " soft Cu tubes for 4 thermo couple conns. with inside end sealed .

Valves : All valves are shut off / on type



NOTE :- 1. ALL DIMENSIONS ARE IN MM
 2. SCALE 1:4
 3. MATERIAL : GALVANISED SHEET

FIG. 4 EVAPORATOR CASE WITH EVAPORATOR

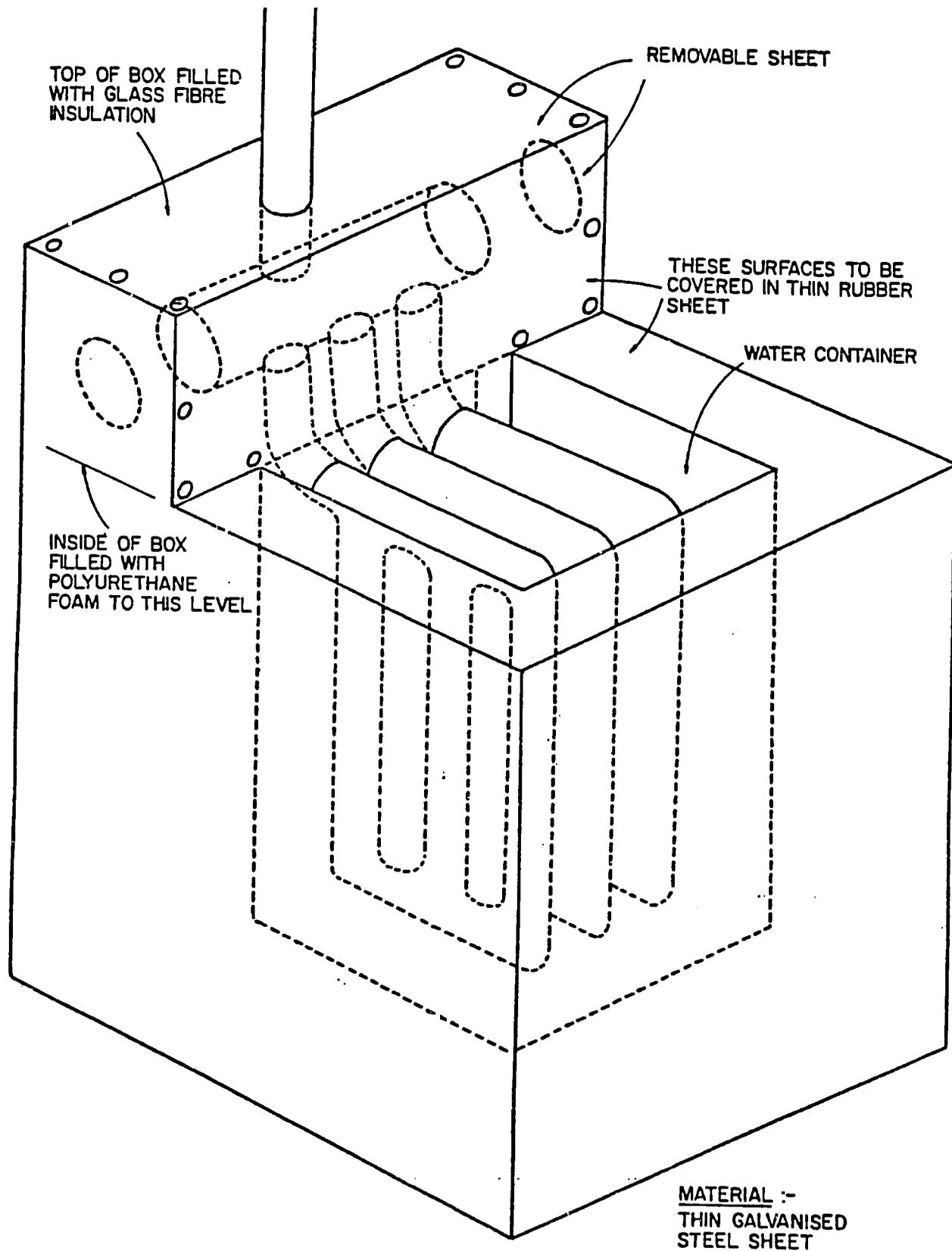


FIG. 5 EVAPORATOR CASE WITHOUT LID

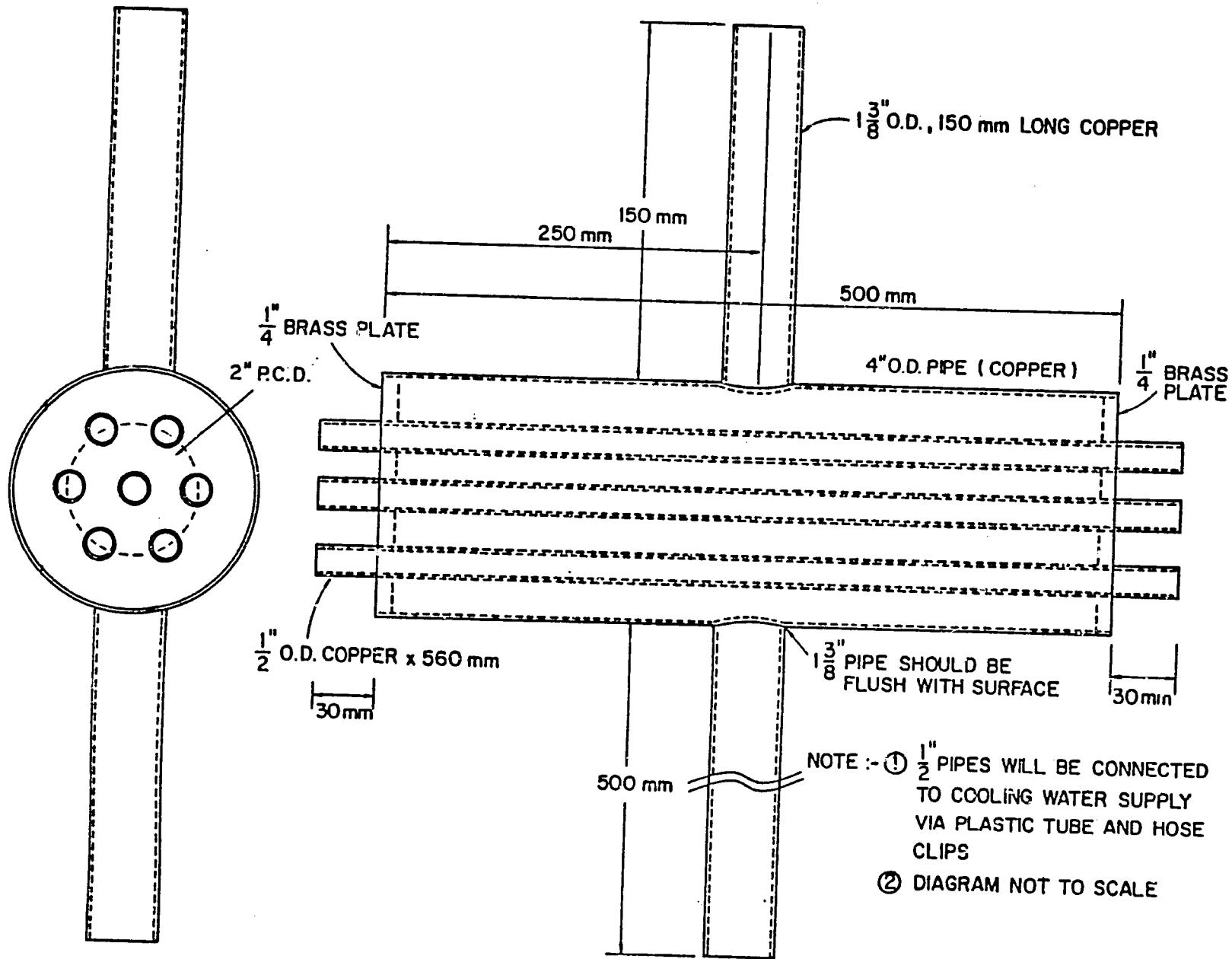


FIG. 6 CONDENSER

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Appendix A1

COCONUTSHELL BASED CARBONS

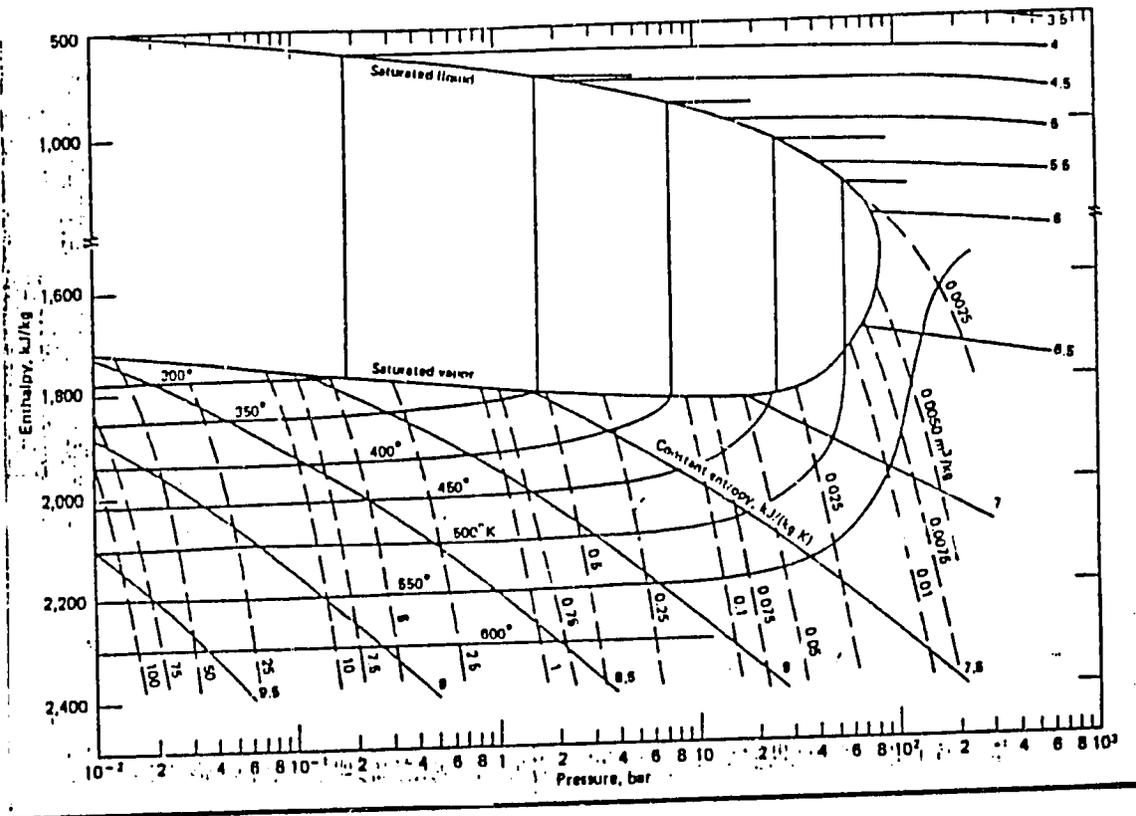
GRADE	SURFACE AREA m ² /g B.E.T. N ₂	BULK DENSITY g/cc	CTC %	ASH %
203C	700 - 800	0.55 - 0.59	20 - 30	1.0 - 2.0
205C	900 - 1000	0.53 - 0.57	35 - 45	2.0 - 2.5
/ 207C	1100 - 1200	0.49 - 0.53	50 - 60	2.5 - 3.0
208C	1200 - 1300	0.47 - 0.51	60 - 70	3.0 - 4.0
† 607	1100 - 1200	0.49 - 0.53	50 - 60	0.5 - 1.0
610	1300 - 1500	0.40 - 0.44	80 - 90	3.0 - 4.0

† Type 607 is a specially prepared acid washed carbon with low ash and high ignition temperature for use in solvent recovery or condensate de-oiling.

* The bulk density is carried out on dry base carbon with nominal mesh size 7-18BSS.

IMPREGNATED & MISCELLANEOUS SPECIAL CARBONS

207E3 (EXTRUDED)	1100 - 1200	0.45 - 0.49	55 - 65	4.0 - 8.0
207E4 (EXTRUDED)	1100 - 1200	0.45 - 0.49	55 - 65	4.0 - 8.0



Thermodynamic chart for saturated, and superheated and compressed methanol

Properties of saturated-methanol liquid and vapor from triple point to critical point

Temperature, K	Pressure, bar	Specific volume, saturated liquid, m ³ /kg	Specific volume, saturated vapor, m ³ /kg	Specific enthalpy, saturated liquid, kJ/kg	Specific enthalpy, saturated vapor, kJ/kg	Specific entropy, saturated liquid, kJ/(kgK)	Specific entropy, saturated vapor, kJ/(kgK)	Specific heat of constant pressure, saturated liquid, kJ/(kgK)
175.4*	1.887 x 10 ⁻⁶	1.105 x 10 ⁻³	241,200	314.3	1,624.3	2.766	0.233	2.177
180	3.909 x 10 ⁻⁶	1.111 x 10 ⁻³	118,600	324.4	1,629.8	2.822	0.076	2.181
190	1.693 x 10 ⁻⁵	1.124 x 10 ⁻³	29,690	346.1	1,642.2	2.940	0.781	2.190
200	6.274 x 10 ⁻⁵	1.137 x 10 ⁻³	8,278	367.8	1,654.6	3.051	0.486	2.201
210	2.032 x 10 ⁻⁴	1.150 x 10 ⁻³	2,680	388.9	1,666.8	3.169	0.240	2.215
220	6.881 x 10 ⁻⁴	1.163 x 10 ⁻³	870.1	412.6	1,678.2	3.284	0.022	2.231
230	1.939 x 10 ⁻³	1.176 x 10 ⁻³	385.3	435.6	1,689.8	3.366	0.829	2.251
240	5.687 x 10 ⁻³	1.189 x 10 ⁻³	168.7	459.1	1,701.2	3.467	0.655	2.275
250	0.188 x 10 ⁻²	1.203 x 10 ⁻³	79.02	483.0	1,716.6	3.664	0.499	2.303
260	1.700 x 10 ⁻²	1.218 x 10 ⁻³	39.49	507.1	1,728.8	3.669	0.350	2.337
270	3.327 x 10 ⁻²	1.230 x 10 ⁻³	20.93	531.6	1,740.7	3.751	0.230	2.376
280	6.208 x 10 ⁻²	1.244 x 10 ⁻³	11.82	556.1	1,762.1	3.840	0.112	2.420
290	0.1094	1.259 x 10 ⁻³	6.778	581.0	1,763.2	3.828	0.005	2.471
300	0.1860	1.274 x 10 ⁻³	4.095	606.2	1,773.6	4.013	7.904	2.528
310	0.3043	1.290 x 10 ⁻³	2.668	631.7	1,783.1	4.097	7.811	2.59
320	0.4817	1.306 x 10 ⁻³	1.681	657.5	1,791.8	4.179	7.723	2.67
330	0.7395	1.323 x 10 ⁻³	1.103	683.8	1,799.6	4.259	7.640	2.76
337	1.0012	1.337 x 10 ⁻³	0.8110	703.4	1,804.4	4.310	7.581	2.81
340	1.1044	1.342 x 10 ⁻³	0.7633	710.1	1,806.9	4.338	7.562	2.83
350	1.6082	1.361 x 10 ⁻³	0.5256	736.9	1,812.4	4.416	7.488	2.91
360	2.288	1.381 x 10 ⁻³	0.3752	764.3	1,817.4	4.492	7.417	3.01
370	3.188	1.403 x 10 ⁻³	0.2723	782.6	1,821.4	4.569	7.350	3.11
380	4.357	1.428 x 10 ⁻³	0.2016	820.2	1,825.1	4.643	7.287	3.22
390	5.845	1.452 x 10 ⁻³	0.1612	856.4	1,828.1	4.738	7.227	3.34
400	7.763	1.480 x 10 ⁻³	0.1156	888.8	1,830.7	4.817	7.172	3.48
410	10.00	1.618 x 10 ⁻³	0.0883	917.7	1,832.1	4.888	7.120	3.63
420	12.83	1.543 x 10 ⁻³	0.0698	947.7	1,834.1	4.959	7.069	3.80
430	16.76	1.561 x 10 ⁻³	0.0542	982.2	1,834.1	5.039	7.020	4.06
440	20.40	1.624 x 10 ⁻³	0.0425	1,022.7	1,832.1	5.130	6.971	4.39
450	28.30	1.674 x 10 ⁻³	0.0333	1,069.2	1,828.1	5.233	6.920	4.68
460	31.08	1.733 x 10 ⁻³	0.0283	1,120.6	1,820.1	5.344	6.864	5.08
470	37.80	1.806 x 10 ⁻³	0.0202	1,176.8	1,807.1	5.460	6.804	5.28
480	45.61	1.894 x 10 ⁻³	0.0156	1,233.4	1,786.1	5.678	6.728	5.64
490	54.66	2.021 x 10 ⁻³	0.0116	1,292.5	1,761.1	5.696	6.633	6.72
500	65.17	2.214 x 10 ⁻³	0.0084	1,353.7	1,764.1	5.818	6.518	6.1
510	77.43	2.69 x 10 ⁻³	0.0058	1,437.8	1,648.1	5.976	6.391	
512.71	81.03	3.64 x 10 ⁻³	0.0036	1,532	1,632	6.170	6.170	

*Triple point (Critical point)

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STUDIES ON ACTIVATED CHARCOAL - METHANOL SYSTEM

WITH RELEVANCE TO ICE MAKING

by K. Sridhar

PART I : CHARACTERISATION OF AVAILABLE ACTIVATED CHARCOALS

Introduction

In recent times much work is being done on the Charcoal-Methanol adsorption system for it is superior thermodynamically to the zeolite-water or other well established systems. With this in view, a laboratory test rig was built to study the adsorption characteristics of various activated charcoals available (particularly the local samples) with methanol. From these tests the isotherms of methanol adsorption can be obtained and also the validity of Dubinin's equation (Marsh and Rand, 1970) can be seen with respect to these charcoals.

Theory of Adsorption in micropores - Basic Principles

The basis of the theory is that there is a limiting volume of micropores that can be filled by adsorption, W . The value W represents the volume of the adsorption space of micropores per unit mass of the adsorbent. There are 2 functions that need to be introduced here.

(1) The first is $A = RT \ln(P_s/P)$ which when interpreted thermodynamically is a decrease in free energy in adsorption if the adopted standard state is the state of the bulk fluid at temperature T in equilibrium with the vapour at saturation pressure P_s . This decrease in free energy represents the differential molar work of adsorption, A .

(2) Volume of adsorption space filled i.e. W . W is given by xv where ' x ' is the adsorption value at a temperature T and ' v ' is the molar volume of the substance adsorbed.

For a number of microporous adsorbents, particularly carbonaceous types, the equation of the characteristic curve can be given as:

$$W = W_0 [- \exp k A/\beta^2]$$

where β is the affinity coefficient (in comparison to a standard reference substance).

Thus finally the thermal equation of adsorption takes the form

$$x = x_0 [- \exp(DT \ln P_s/P)^2]$$

or $\ln x = \ln x_0 - D [T \ln P_s/P]^2$ where $D = B/\beta^2$.

This is known as the Dubinin's equation of the characteristic curve.

Objectives of the Laboratory studies

For the various charcoal samples available

- i. Obtain the $p-T-x$ data and thus the function $x=x(T,P)$ for methanol adsorption
- ii. To draw out the Clapeyron Diagram ie. $\ln P$ vs $-1/T$.
- iii. To derive the equation of the Characteristic curve.

Samples available

A total of 8 samples are presently available. The data of these charcoals are summarised below.

Sample	Place/Source	Manufacturer's Data
203C	Sutcliffe Speakman Co. UK.	Refer Appendix A1
205C	- do -	
207C	- do -	
207E	- do -	
607	- do -	
610	- do -	
X	Suratkal, India.	No details available
Y	UDP Co. Thailand	Refer Appendix A2

The laboratory Test Rig

The laboratory test rig is as shown in Figure 1. The amount of methanol adsorbed can be calculated from the amount remaining in the methanol receiver, since the amount of methanol in the system is known. The temperatures and pressures can also be measured.

For every equilibrium point the following parameters are noted : temperature of charcoal (T_{ch}), temperature of methanol (T_m), pressure in the system (P) and the level of methanol in the receiver (l_v) and hence concentration of methanol in the charcoal per unit weight of charcoal (x) can be calculated.

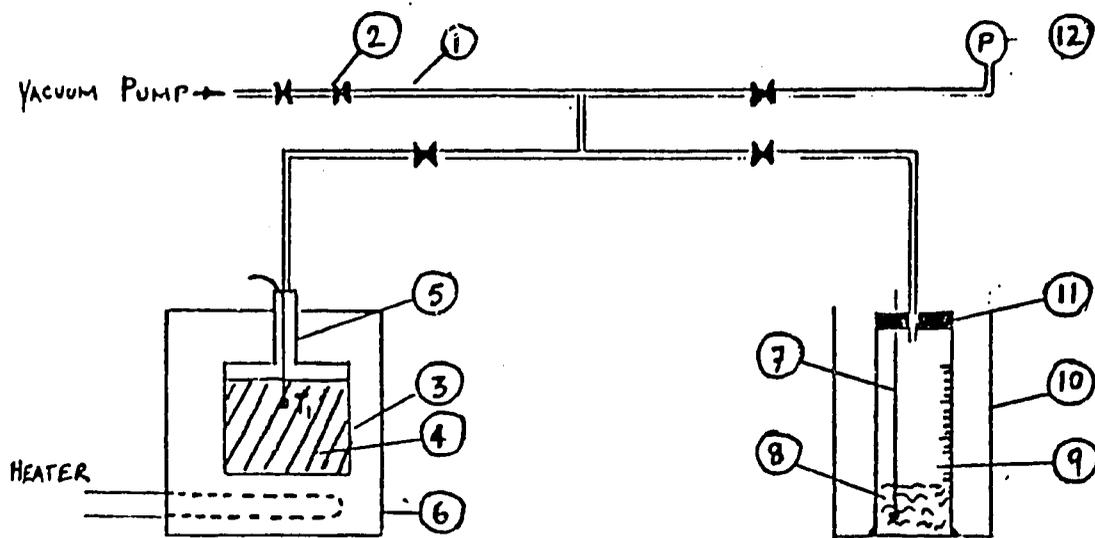


Figure 1 : The Laboratory Test Rig

KEY TO FIGURE

- 1. 1/4 inch Cu tube
- 2. 1/4 inch Soldered valves
- 3. Charcoal container.
- 4. Activated Charcoal.
- 5. Thermocouple Connection
- 6. Water bath with electric heater.

- 7. Mercury in glass thermometer.
- 8. Liquid methanol.
- 9. Graduated cylinder.
- 10. Water bath.
- 11. Rubber cork sealed with araldite.
- 12. Vacuum Gauge

Procedure

1. The weight of charcoal loaded, M (in gms) was noted.
2. The charcoal was degassed (heated and evacuated) to clean it from impurities.
3. About 10 ml of methanol was boiled off to remove any moisture in it.
4. The starting level of methanol in receiver was noted. The methanol receiver was kept at $T_m=20^\circ\text{C}$ and valve opened to charcoal for adsorption. After equilibrium is attained, note level and hence x .
5. In steps of 10°C the temperature of charcoal was increased till 100°C and at each stage measured T, P & x . Similarly x, T, P was measured on the way downwards till 30°C . Thus the heating and cooling curve for a particular value of T_m was obtained.
6. This was repeated to get consistent data and end points for one particular value of T_m .
7. The same is repeated for $T_m=0^\circ\text{C}$.
8. For each T_m , plot the average x versus T curve and then plot the Clapeyron Diagram ie. $\ln P$ vs $-1/T$.
9. Plot $(-\ln x)$ versus $(T \ln P_s/P)$ curves and derive constants for the Dubinin's equation.

Results and Conclusions

So far 3 samples have been fully tested and a fourth one is in progress. The three that have been tested are 207C, 207E and the Thai sample. In this section, for each of these charcoals the following results, in the form of graphs, have been presented.

- i) The concentration (x) versus the temperature of the charcoal (T), for the two pressures corresponding to the methanol temperatures 20°C and 0°C .
- ii) The Clausius-Clapeyron diagram, which is the $\ln P$ vs $-1/T$ diagram, which illustrates the isotherms.
- iii) The characteristic curve in a linear form and the validity of the Dubinin's equation can be seen.

The samples 207E and 207C have given reasonable agreement when fitted to the Dubinin's equation. However the sample Y ie. the Thai sample fits the equation poorly. This is particularly due to the large hysteresis exhibited by the charcoal.

The results in brief are :

Dubinin's Equation : $x = x_0 [-\exp (DT \ln P_s/P)^n]$					
Sample	n	x kg meth/kg °C	D * 10 ⁶	W ₀ 1/kg	Corr Coeff. r
207E	2	.2642	.9645	.334	.96
207C	2	.2285	.3796	.289	.88
Thai	2	.1606	.3040	.203	.39

UN

These results are comparable to those obtained by researchers on other activated charcoals. The parameter n was also varied but found that the value $n=2$ was the best fit possible. Presently the sample 610 is being tested.

To conclude, the availability of an equation that describes best the behaviour of these charcoals would be of immense use in modelling, prediction and sizing of adsorption refrigerators, utilising these charcoals.

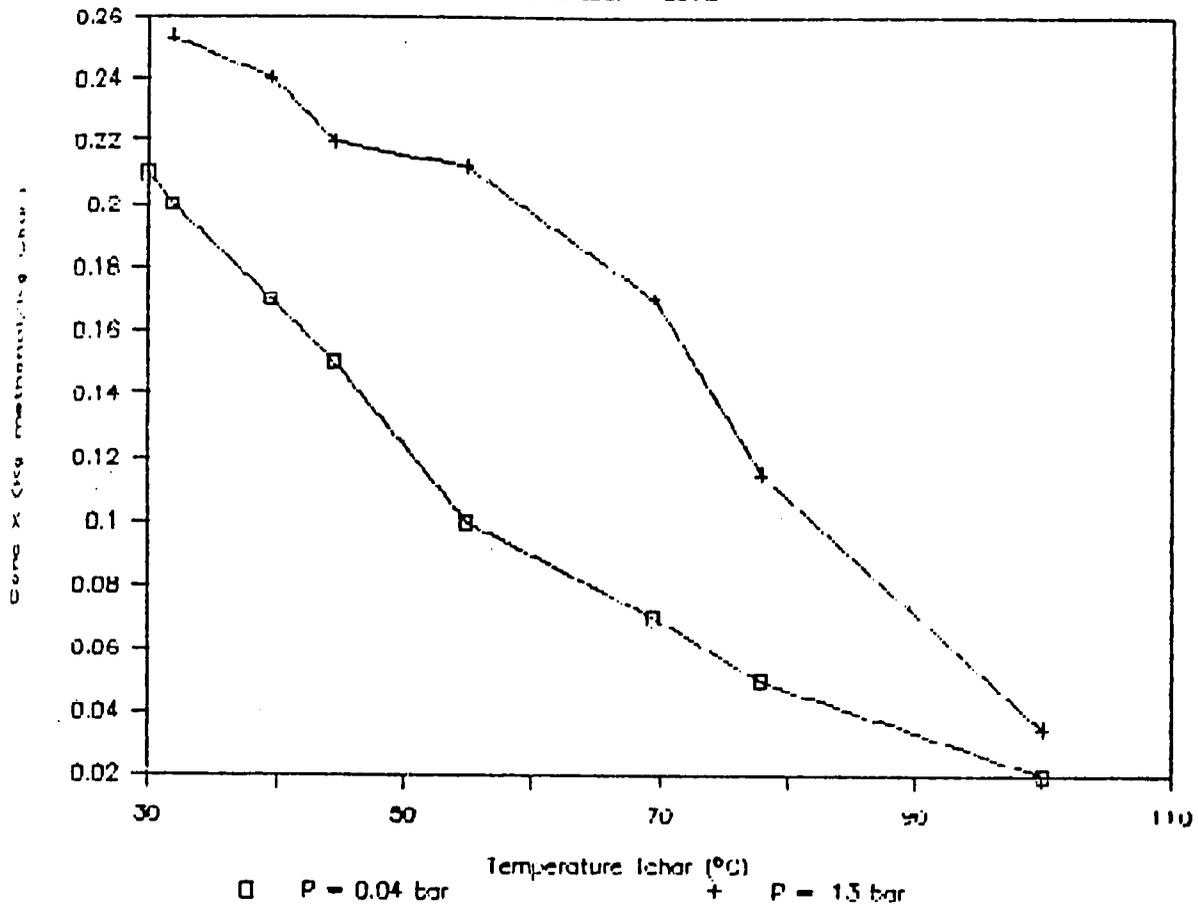
References

1. Dubinin, M.M. (1967) Adsorption in Micropores, Journal of Colloid and Interface Science 23, pp 487-499.
2. Marsh H. and Rand B. (1970) The Characterisation of Microporous carbons by means of the Dubinin-Radushkevich Equation, Journal of Colloid and Interface Science, Vol 33 No.1, May.

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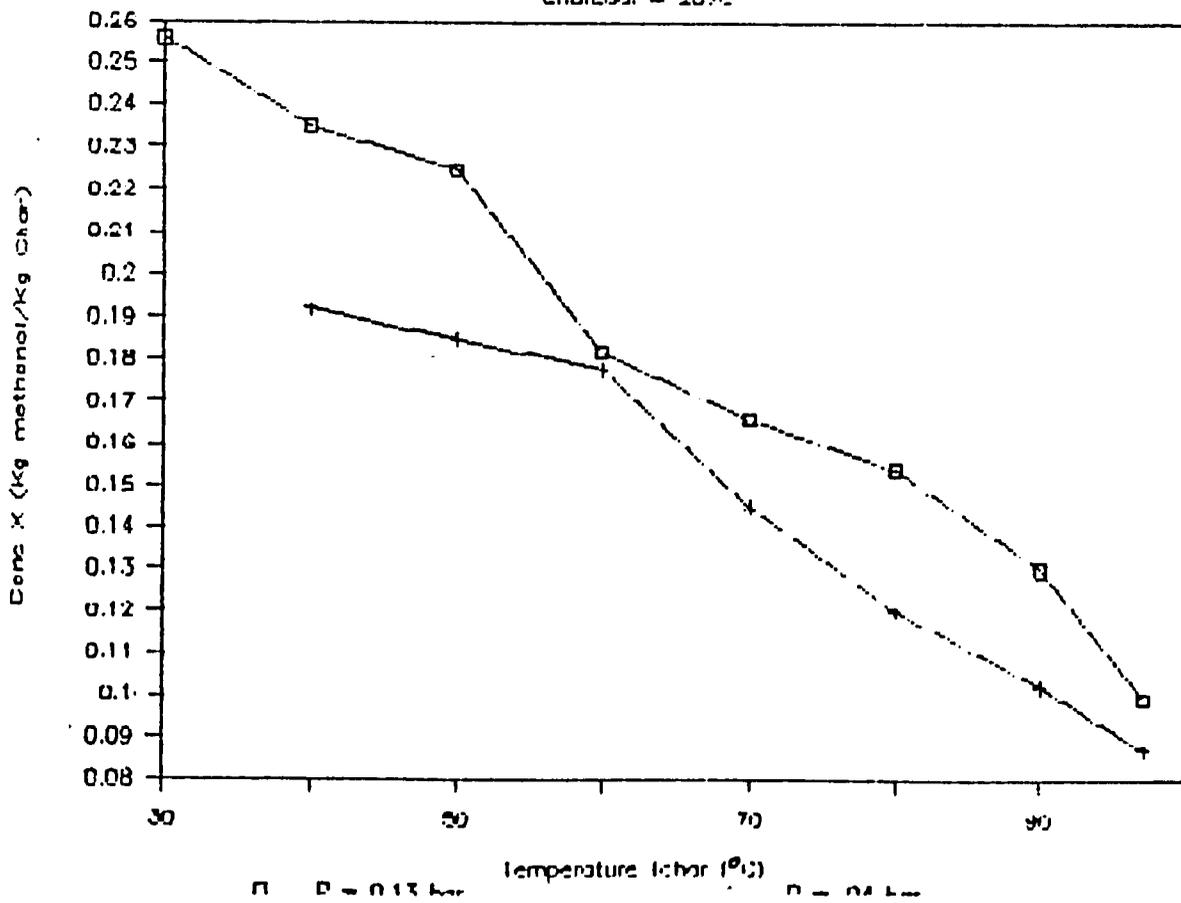
Concentration X vs Tchar

Charcoal = 207E



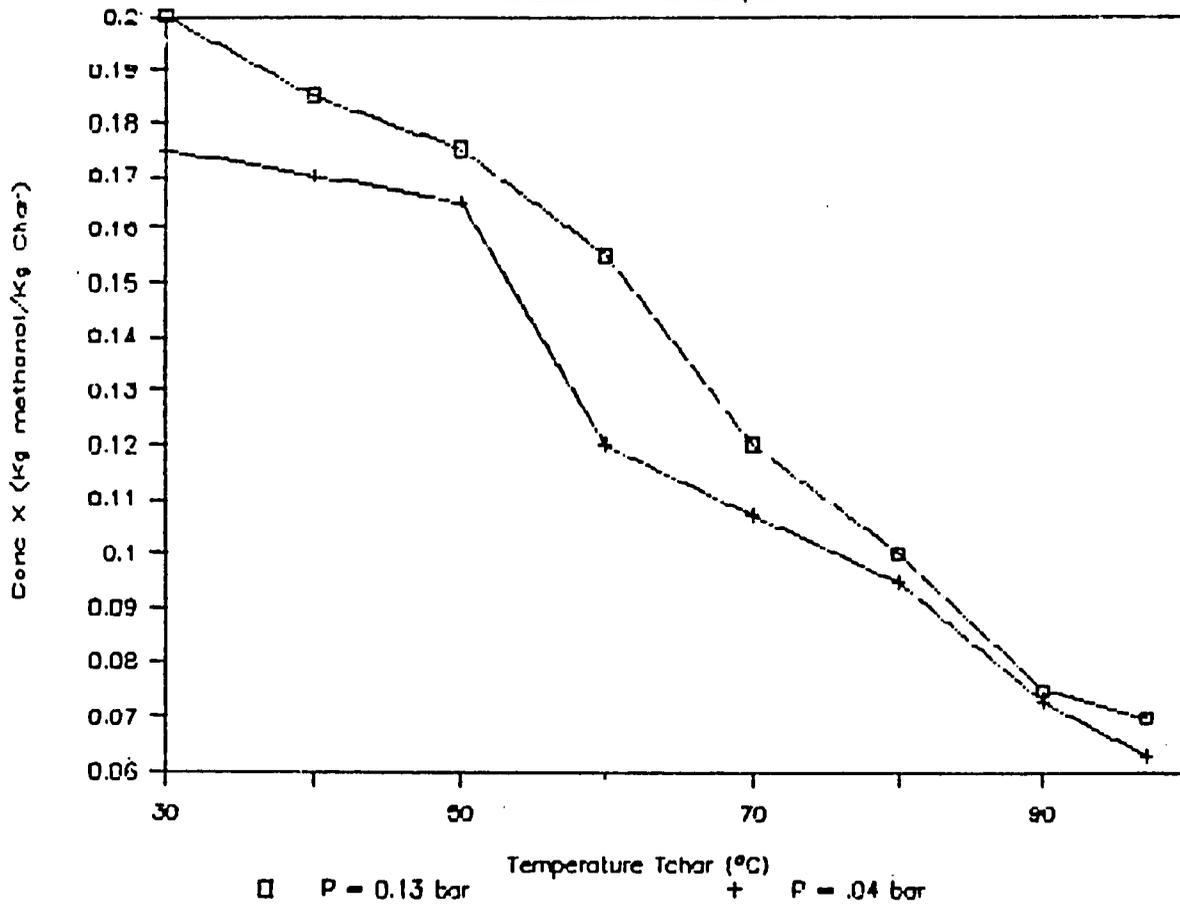
Concentration X vs Tchar

Charcoal = 207C



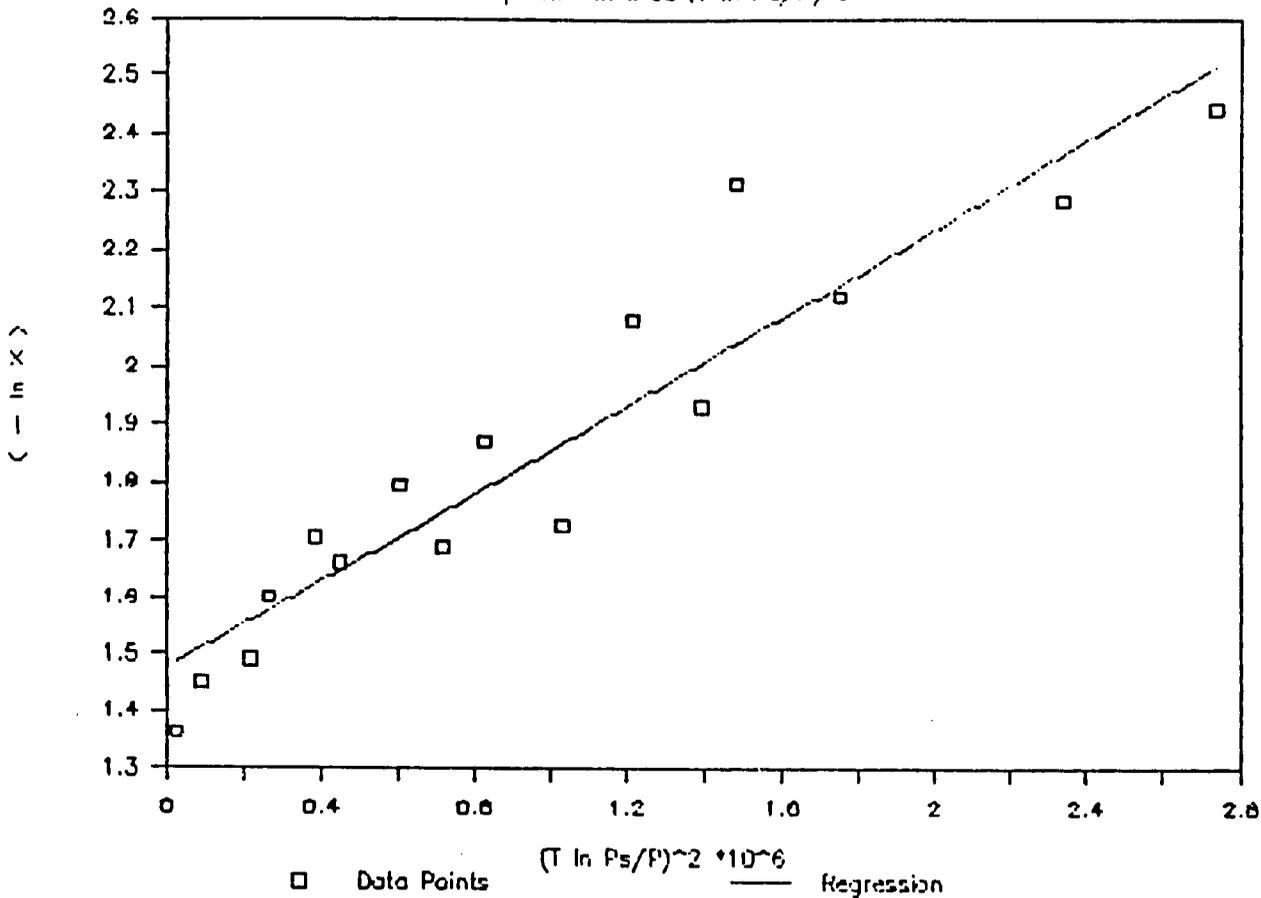
Concentration X vs Tchar

Charcoal = Thai Sample



Curve fit of the D-R Equation (207C)

Graph of $-\ln X$ vs $(T \ln P_0/P)^{-2}$

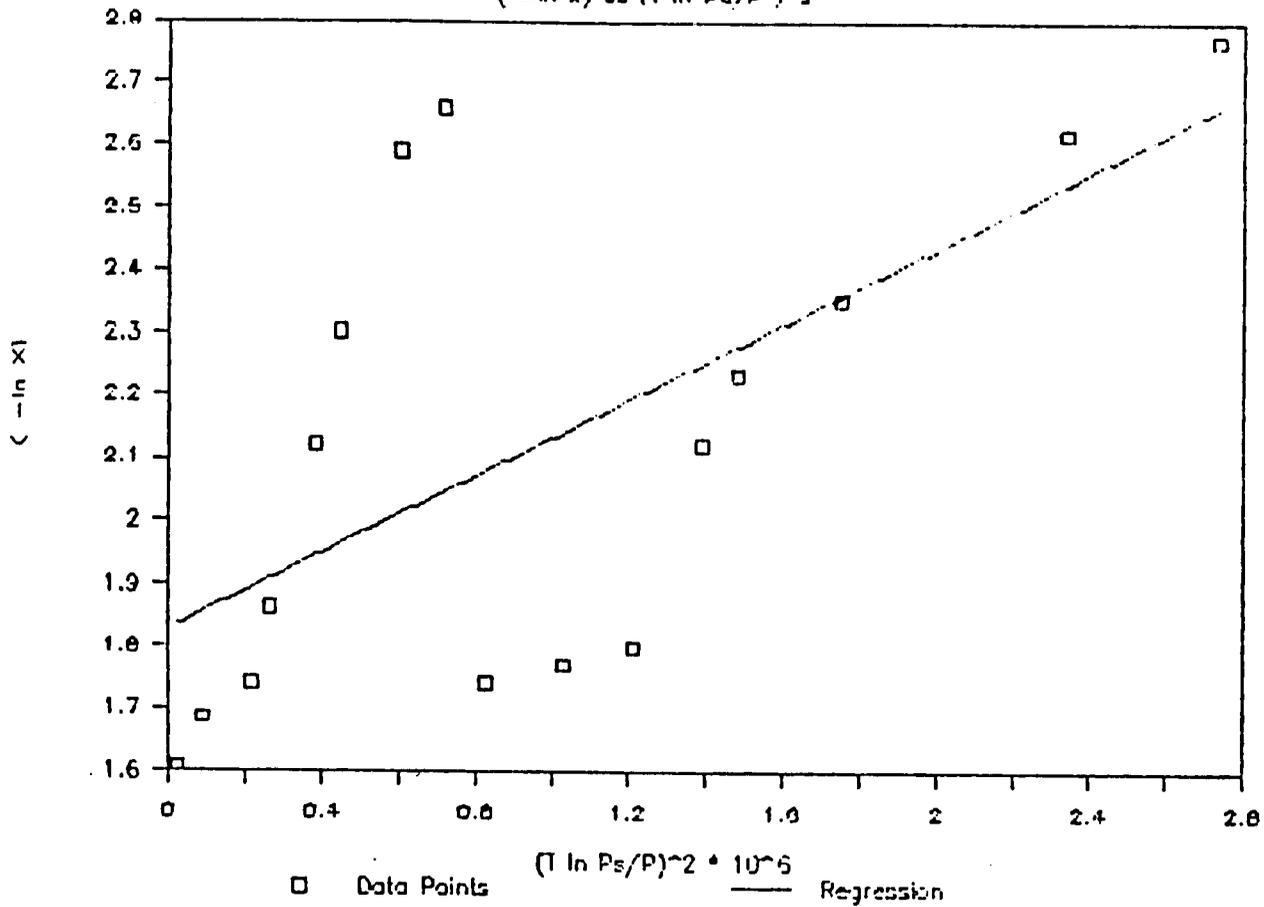


$(T \ln P_0/P)^{-2} \cdot 10^{-6}$	x	$-\ln x$		Regression line	\bar{x}	
0.025410	0.256	1.362577	<p>Regression Output:</p> <p>Constant 1.476325</p> <p>Std Err of Y Est 0.114020</p> <p>R Squared 0.864513</p> <p>No. of Observations 16</p> <p>Degrees of Freedom 14</p> <p>X Coefficient(s) 0.379627</p> <p>Std Err of Coef. 0.036661</p>	1.485971	0.0254	1.363
0.098553	0.235	1.448169		1.510701	0.0906	1.446
0.217053	0.225	1.491654		1.556724	0.2171	1.492
0.386055	0.182	1.703748		1.577614	0.2668	1.599
0.602694	0.166	1.795767		1.622882	0.3861	1.704
0.823573	0.154	1.870802		1.646558	0.4487	1.661
1.212532	0.125	2.079441		1.705124	0.6027	1.796
1.484508	0.099	2.312635		1.746412	0.7167	1.687
0.266812	0.202	1.597467		1.768975	0.8236	1.671
0.448686	0.19	1.669731		1.866521	1.0276	1.726
0.716724	0.185	1.687359		1.536635	1.2125	2.079
1.027843	0.178	1.725971		2.005466	1.3938	1.931
1.393846	0.145	1.931021		2.035864	1.4645	2.313
1.751850	0.12	2.120263		2.141374	1.7519	2.120
2.337848	0.102	2.282762		2.363635	2.3378	2.283
1.737391	0.087	2.441847		2.315312	2.7374	2.442

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Curve fit for the D-R Equation (Thai)

$(-\ln X) \text{ vs } (T \ln P_0/P)^{-2}$



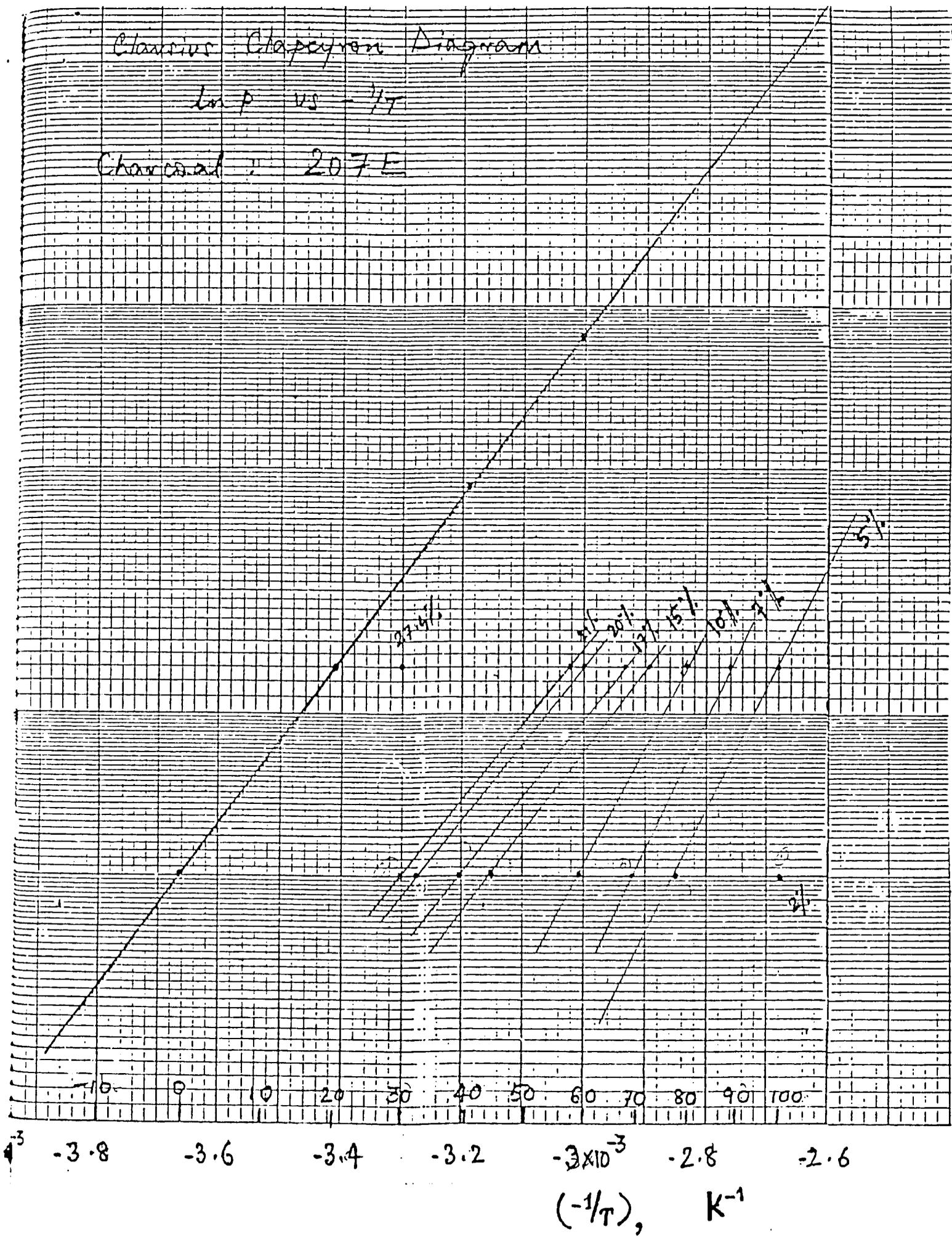
$(T \ln P_0/P)^{-2}$	x	$-\ln x$	Regression Output:		Regression line	X	Y
0.025410	0.2	1.609437	Constant	1.828997	1.836725	0.0254	1.609
0.090553	0.185	1.657379	Std Err of Y Est	0.316273	1.856533	0.0906	1.687
0.217053	0.175	1.742969	R Squared	0.387587	1.874996	0.2171	1.743
0.386055	0.155	1.864330	No. of Observations	16	1.919129	0.2668	1.864
0.602694	0.12	2.120263	Degrees of Freedom	14	1.946387	0.3861	2.120
0.833573	0.1	2.302585	X Coefficient(s)	0.304072	1.965432	0.4487	2.303
1.125332	0.075	2.590267	Std Err of Coef.	0.101723	2.012261	0.6027	2.590
1.484508	0.07	2.657260			2.046934	0.7167	2.659
1.866812	0.175	1.742969			2.079424	0.8236	1.743
1.448686	0.17	1.771956			2.141537	1.0278	1.772
1.716724	0.165	1.801809			2.197696	1.2125	1.802
1.027843	0.12	2.120263			2.252828	1.3938	2.120
1.393846	0.107	2.234926			2.280396	1.4845	2.235
1.751850	0.095	2.353878			2.361687	1.7319	2.354
1.337848	0.073	2.617295			2.539873	2.3378	2.617
1.737391	0.063	2.764620			2.651363	2.7374	2.765

40

Classic Clapeyron Diagram

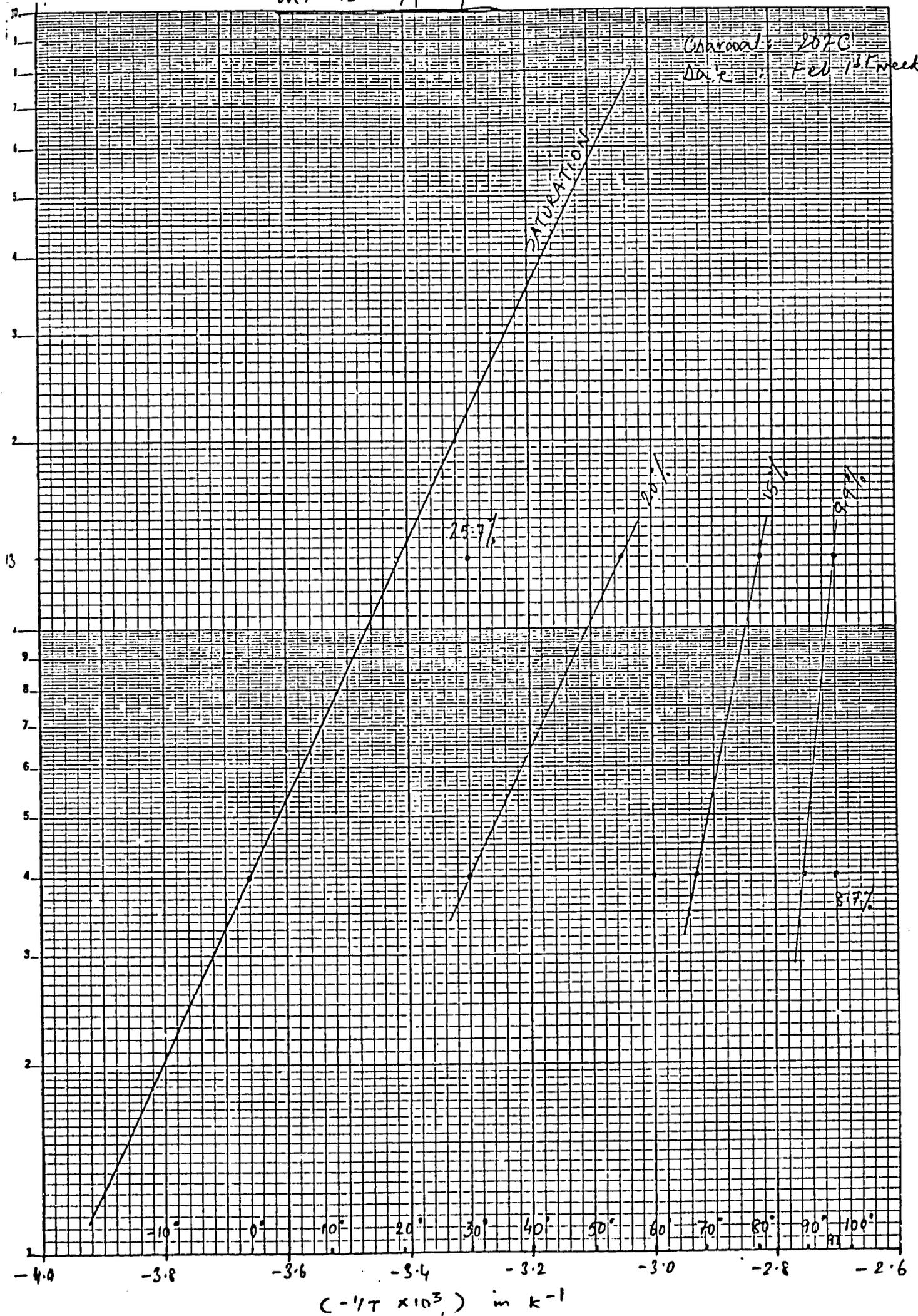
$\ln p$ vs $-1/T$

Charcoal : 207E



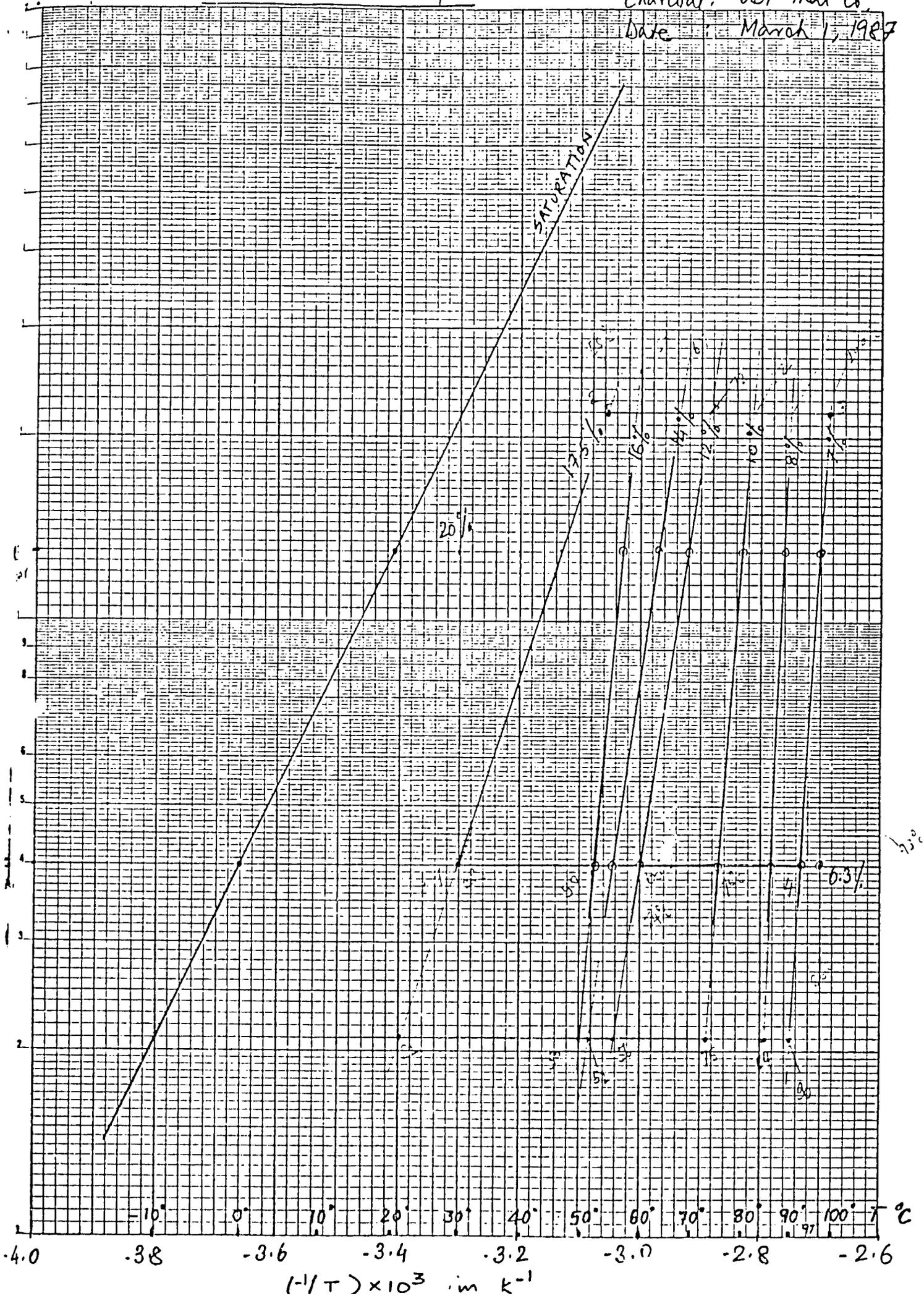
ln P vs -1/T plot

Charoat: 207C
 Date: Feb 13, 1966



ln P vs 1/T

Charcoal: UDP Grain Co.
Date: March 1, 1987



Appendix A1

COCONUTSHELL BASED CARBONS

GRADE	SURFACE AREA m ² /g B.E.T. N ₂	* BULK DENSITY g/cc	CTC %	ASH %
203C	700 — 800	0.55 — 0.59	20 — 30	1.0 — 2.0
205C	900 — 1000	0.53 — 0.57	35 — 45	2.0 — 2.5
207C	1100 — 1200	0.49 — 0.53	50 — 60	2.5 — 3.0
208C	1200 — 1300	0.47 — 0.51	60 — 70	3.0 — 4.0
607	1100 — 1200	0.49 — 0.53	50 — 60	0.5 — 1.0
610	1300 — 1500	0.40 — 0.44	80 — 90	3.0 — 4.0

† Type 607 is a specially prepared acid washed carbon with low ash and high ignition temperature for use in solvent recovery or condensate de-oiling.

* The bulk density is carried out on dry base carbon with nominal mesh size 7-18BSS.

IMPREGNATED & MISCELLANEOUS SPECIAL CARBONS

7E3 (EXTRUDED)	1100 — 1200	0.45 — 0.49	55 — 65	4.0 — 8.0
7E4 (EXTRUDED)	1100 — 1200	0.45 — 0.49	55 — 65	4.0 — 8.0

g

Appendix A2

TYPICAL APPLICATIONS : MD-G 7746

MD-G 7746 is designed for removal of odourous atmosphere in industrial processes and in air conditioning systems, including a variety of purification and separation processes in the chemical industries.

GENERAL SPECIFICATIONS AND PROPERTIES

Mesh Sizes _____	4×6
Iodine Number Min. _____	1050
Methylene Blue Number Min. _____	200
Abrasion Number Min. _____	80
Hardness Number Min. _____	90
Moisture as packed Max. % _____	3
Residue on Ignition, Max. % _____	3
pH _____	8-9
Surface Area (N ₂ -B.E.T.) _____	1100-1200
Bulk Density, g/cc. _____	0.40-0.45

TYPICAL APPLICATIONS : MD-W 7830

MD-W 7830 is recommended for water purification both in municipal and industrial use. It is also popular in a variety of separation and purification processes in the organic chemical industries.

GENERAL SPECIFICATIONS AND PROPERTIES

Mesh Sizes _____	8×30
Iodine Number Min. _____	1050
Methylene Blue Number Min. _____	220
Abrasion Number Min. _____	80
Hardness Number Min. _____	90
Moisture as packed Max. % _____	3
Residue on Ignition, Max. % _____	3
pH _____	8-9
Surface Area (N ₂ -B.E.T.) _____	1150-1250
Bulk Density, g./cc. _____	0.50-0.55



UDP. CHEMICAL CO., LTD.

77/6 SOI KLONGNONGYAI SUKHAPITAN 1 RD. BANGKAE PASRICHAROEN BANGKOK 10160
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APPENDIX III

SUMMARY OF RECENT WORK ON SOLAR ZEOLITE-WATER REFRIGERATOR

Zhu Zepei, March 20, 1987

I. INTRODUCTION

Work have been done on estimating the maximum heat leak to the cooling box of the solar zeolite-water refrigerator by an experiment and the results are presented; the maximum heat leak is about $1 \text{ W/m}^2\text{-C}$, which was got from data taken in the early afternoon. Modification of the zeolite refrigerator also have been finished and the change of some arrangement of the system would make it more convenient to control the experiment on the refrigerator and get more accurate data in the test. Problems existed at present in the experiment and the futher theoretical analysis on the refrigerator are discussed, and future plans are proposed as well.

II. HEAT LEAK EXPERIMENT

In order to estimate the maximum heat leak of the cooling box of the zeolite-water refrigerator for further research, the solar collector (also a regenerator) had been covered for a couple of weeks and the desorption and adsorption of the water (refrigerant) became so small that the water change in the receiver could be ignored during the experiment. There was a small amount of water (about 1.25 KBTU of the glass bottle scale) kept inside the glass bottle (which functions was as receiver & evaporator). A known quantity of ice was put into the cooling box and temperatures were recorded continuously and the amount of water due to melting of ice was collected and measured in every 15 minutes. The value of heat leak was estimated around $0.72 - 0.99 \text{ W/m}^2\text{-C}$ during noon. Thus, we can say that the $1 \text{ W/m}^2\text{-C}$ is the largest value of the heat leak. The experimental data and the calculated results are given in Appendix.

III. MODIFICATION OF THE REFRIGERATOR

After finishing the heat leak experiment, work on modifying the system began. Many days of experiment had been done since last October and the cooling produced in a refrigeration cycle was so small that the water adsorbed by the zeolite in the collector during the night was only about $1/5 - 1/4$ of the amount claimed by the inventor, even during a fine sunny day with a fairly low ambient temperature at night. Because of not enough accuracy of Bourdon vacuum gauge installed on the system before, a mercury manometer was installed with some isolating valves

which would make it easier to control the evacuation and the recharging of the system. In an unexpected accident, the glass bottle (which acts as a receiver/evaporator) was broken and it was replaced by a nearly same size bottle. One of the important modification was a ball valve installed between the condenser and the glass bottle. With the help of the ball valve which can be full opened and closed, it is quite convenient to control the experiment of the refrigerator. The schematic diagrams of the original refrigerator and the modified one are shown in Fig 1 and Fig 2, respectively.

IV. PROBLEMS EXISTING AT PRESENT

1) According to the analysis of previous experimental data and the results, the amount of water desorbed and adsorbed (scale 0.5 KBTU = 0.227 Kg) was much less than the one (loading 1.283 - 1.796 Kg per cycle) reported by the Zeopower Company. Since the climatic conditions in the experiment were not much different compared with the ones reported on the paper, the performance of the refrigerator was so poor that make us to think weather or not the zeolite had deteriorated before being filled into the collector panel. It is very difficult to prove that because there is no natural zeolite sample available at the moment.

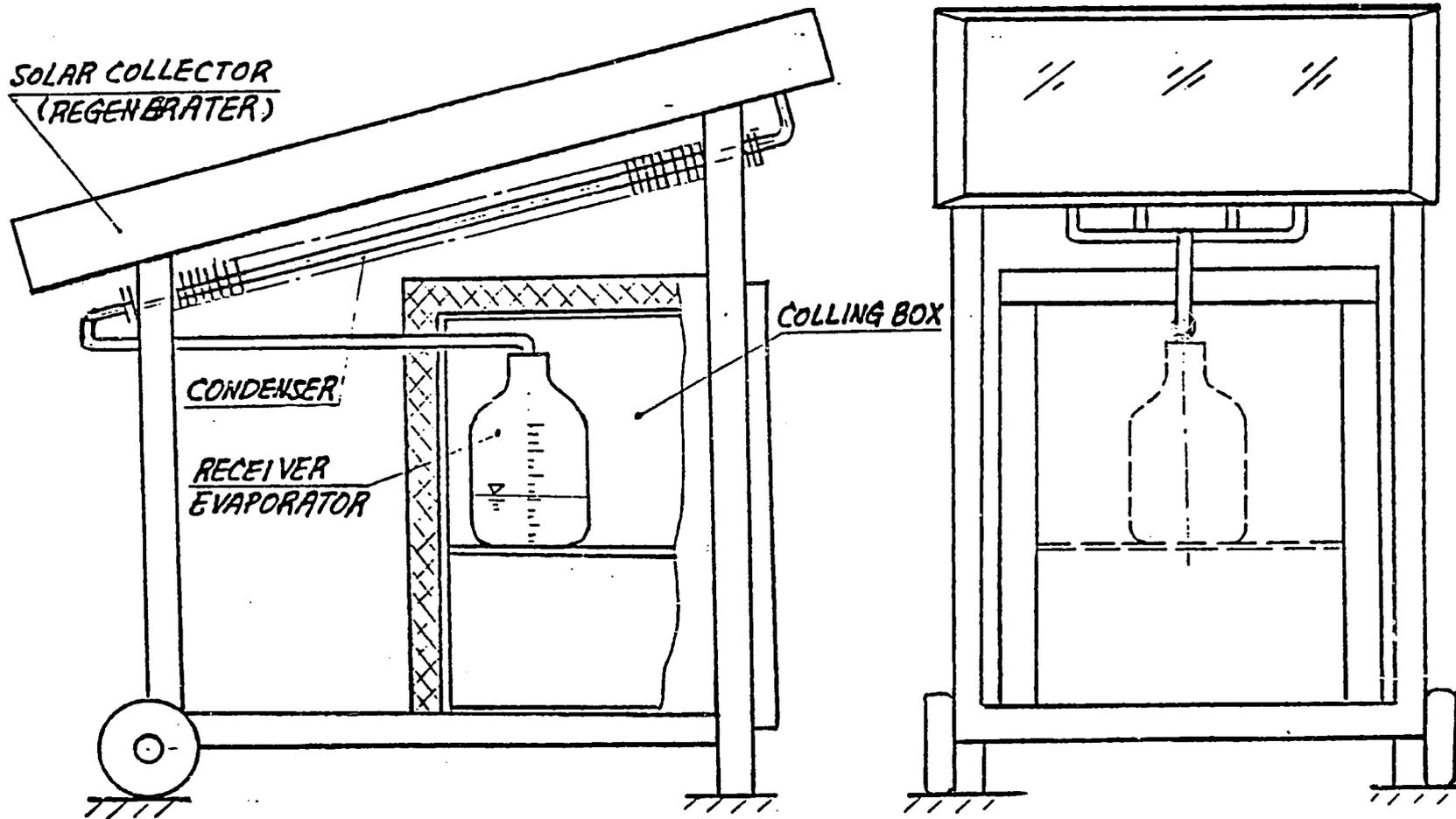
2) More information are needed for further studying the performance of the equipment experimentally and theoretically, such as p-T-x diagram of the natural zeolite, and what kind of inert gas had been filled inside the collector panel. If natural zeolite sample was available, we could do some experiments to get p-T-x diagram in our research laboratory.

V. FUTURE PLAN FOR THE RESEARCH WORK

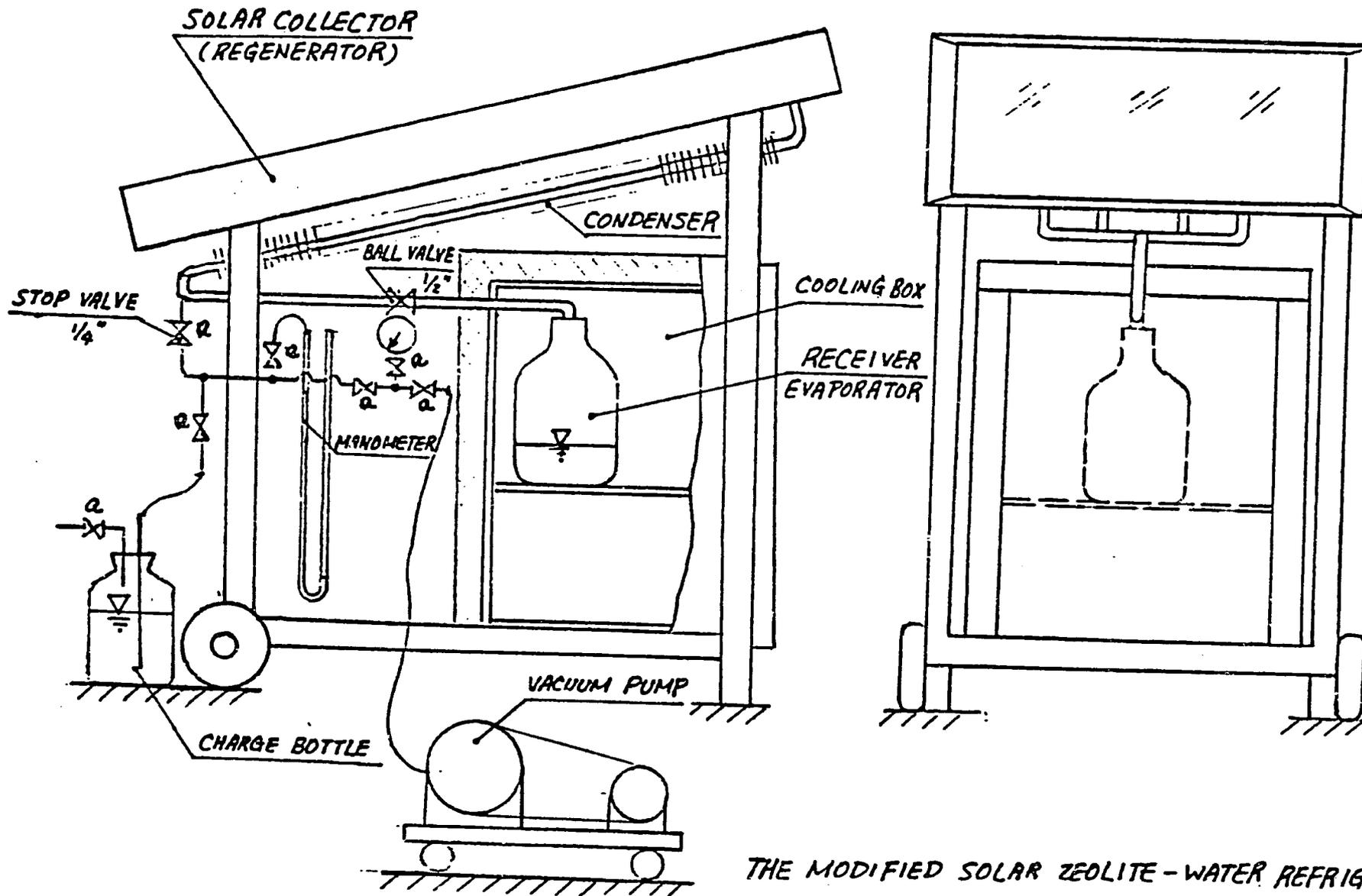
1) Evacuate the system and withdraw the water adsorbed by the zeolite as completely as possible. To withdraw the water inside the collector panel, higher adsorbent temperature is necessary. The higher the temperature in the collector panel is, the lower will be the water concentration of the zeolite. East-west booster mirrors can be applied to enhance solar radiation falling on the collector surface. With the booster, considerable temperature increase of the collector panel may be reached and most of the water would be desorbed from the zeolite and can withdrawn by a vacuum pump during day time.

2) Recharge distilled water into the zeolite and estimate the water concentration in the zeolite. Repeat the experiment on the refrigerator and theoretically calculate the performance with more accurate temperature and pressure measurements.

3) Analyse the expermental results and compare them with previous results and the results presented in published papers



THE ZEOPOWER MODEL SR-4 SOLAR REFRIGERATOR



THE MODIFIED SOLAR ZEOLITE-WATER REFRIGERATOR

and modify the system if necessary.

4) Evacuate and withdraw the water inside the system completely and recharge the system with methanol, then the zeolite-methanol pair will be studied. Before taking this step, modification of the system should be finished because the flammable and explosive properties of methanol make it difficult to solder the pipe work for modifying the system. According to a newly published paper (1) that Zeolite/Methanol pair cannot be used because, at around 100 C, zeolite is a catalyst of the reaction;



Therefore, careful investigation should be done before recharging with methanol.

REFERENCE

1. M. Pons and J. J. Guilleminot, Design of an Experimental Solar-Powered, Solid-Adsorption Ice Maker, Journal of Solar Energy Engineering, Transaction of the ASME, Nov. 1986.

HEAT LEAK TO THE COOLING BOX OF THE SOLAR ZEOLITE-WATER REFRIGERATOR

ZHU ZEPEI FEB. 1987

OBJECTIVE:

Finding the possible heat leak of the cooling box of the solar zeolite refrigerator for further research.

Methodology and Calculations

When the temperature inside the cooling box is lower than the temperature outside, heat will transfer from outside to inside of the cooling box through conduction, convection and radiation, and the temperatures would reach constant value at an equilibrium state. Because the cooling box can be considered hermitically tight. The diffuse radiation that can be absorbed by the walls of the cooling box would increase the temperatures of the walls. For the heat leak calculation we mainly considered the heat conduction between outside and inside walls. The other constituents of heat transfer are so small that can be neglected with little error on the estimation of the heat leak.

Based on the consideration mentioned above, the calculated energy balance equation can be written as:

$$Q_{\text{leak}} = Q_{\text{cond}} \quad (\text{W}) \quad (1)$$

actual heat absorbed by ice and other components in the cooling box can be expressed as:

$$Q_{\text{absor}} = Q_{\text{molten}} + Q_{\text{water}} - Q_{\text{cool}} \quad (\text{W}) \quad (2)$$

where,

Q_{leak} --- calculated heat leak into the cooling box from environment, (W)

Q_{absor} --- heat absorbed by ice and components in the cooling box, (W)

Q_{cond} --- heat transfer through walls by conduction, (W)

Q_{cool} --- heat for the temperature change of the components in the cooling box, (W)

Q_{molten} --- latent heat of fusion (W)

Water --- heat for temperature change of molten water above zero degree. The temperature difference was approximately 0 - 5 C

Heat conduction through the walls can be calculated according to the temperature differences between the inside and outside of the walls of the cooling box and the area of heat conducting surfaces, it can be expressed below as:

$$Q_{\text{cond}} = A_1 \frac{k}{h_1} (T_1 - T_7) + A_2 \frac{k}{h_2} (T_2 - T_7) + A_3 \frac{k}{h_3} (T - T_7) + A_4 \frac{k}{h_4} (T_9 - T_7) + A_5 \frac{k}{h_5} (T_8 - T_{11})$$

where, area of the walls are:

$$A_1 = A_5 = 0.59 \times 0.44 = 0.26 \quad \text{m}^2$$

$$A_2 = A_4 = 0.44 \times 0.42 = 0.185 \quad \text{m}^2$$

$$A_3 = 0.59 \times 0.42 = 0.496 \quad \text{m}^2$$

and thickness of the walls

$$h_1 = h_2 = h_3 = h_4 = 0.05 \quad \text{m}$$

$$h_5 = 0.025 \quad \text{m}$$

k --- heat conductivity = 0.05 W/m-C (assumed for polyurethane foam)

T1 to T11 --- temperatures of thermal couples on the cooling box

Substituting the data into the equation and simplifying:

$$Q_{\text{cond}} = 0.26[T_1 + 2(T_8 - T_{11})] + 0.185(T_2 + T_9) + 0.496T - T_7(0.26 + 2 \times 0.185 + 0.496) \quad (\text{W})$$

Actual energy balance in the cooling box can be expressed as Equation (2) and the details can be described below:

$$Q_{\text{molten}} = M_w \times 334 / (dt \times 60) \quad (\text{W})$$

$$Q_{\text{water}} = M_w \times 4.18 \times dT / (dt \times 60) \quad (\text{W})$$

$$Q_{\text{cool}} = (M_w \times 4.18 + M_g \times 0.84E3)dT_6 / (dt \times 60) \quad (\text{W})$$

where,

Mw --- mass of water due to melting of ice measured in burette (g)

Mg --- a part of mass of the glass bottle (g)

dT --- assumed temperature increase of water leaving the cooling box, approximately 0-5 C

dT6 --- temperature difference of the bottle (C)

dt --- time interval of each measurement (min)

Although M_w , M_g would change depending on the water level (expressed in KBTU) which varied with days, and they can be considered constant during the day at which the experiment was carried on because the collector had been covered and the desorption of water is very small.

On 25 January, the water level in the bottle was 2.5 (KBTU)

$M_w = 1250$ g, and $M_g = 526$ g

On 4 February, the water level in the bottle was 2.0 (KBTU)

$M_w = 1000$ g, and $M_g = 462$ g

To estimate the overall heat leak coefficient, simple equations are proposed according to the previous analysis as follows:

$$Q_{\text{leak}} = UA_dT \quad (\text{J}) \quad (3)$$

and also

$$Q_{\text{absor}} = UA_dT \quad (\text{J}) \quad (4)$$

where,

A --- total area of the heat conduction (m²)

U --- overall heat leak coefficient (W/m²-C)

dT --- temperature difference between the average outside and inside wall temperatures (C)

$$A = A_1 + A_2 + A_3 + A_4 + A_5 = 1.882 \text{ m}^2$$

$$dT = (T_1 + T_2 + T_8 + T_9)/4 - (T_5 + T_6 + T_7 + T_{11})/4$$

Substituting the data into the Equation (3) & (4),

$$U_{\text{calcul}} = \frac{Q_{\text{cond}}}{dT \times A}$$

$$\text{and } U_{\text{actual}} = \frac{(Q_{\text{molten}} + Q_{\text{water}} - Q_{\text{cool}})}{dT \times A}$$

EXPERIMENTAL OBSERVATION ON THE HEAT LOSS OF THE ZEOLITE-WATER REFRIGERATOR

DATE: 25 JAN. 1987

TIME	TIME INTERVAL	OUTSIDE 1	WALL 2	TEMP. 9	FRONT DOOR 8	11	WALL TEMP 7	RECEIVER/EVAPORA. 5	6	AMBIENT 3	MELTEN WATER 4	WATER TEMP
	min	BACK VIOL. DOT C	TOP RED DOT C	BOTTOM BLACK -0 C	OUTSIDE RED-0 C	INSIDE BLUE-0 C	INSIDE VIOLET-0 C	TOP BLUE DOT C	BOTTOM YEL. DOT C	BLACK DOT C	ml	C
11.00		24.0	21.1	21.2	24.6	12.8	14.1	15.0	11.4	21.3		
11.20	20	25.0	21.4	22.0	24.5	12.4	13.4	14.1	10.6	22.7	130.0	16.4
11.35	15	24.9	21.5	21.5	24.2	12.2	13.0	13.6	10.0	22.2	89.0	16.1
12.00	25	24.1	21.9	21.3	25.2	12.5	13.0	13.2	9.3	22.7	140.0	16.8
12.40	40	25.0	22.6	22.2	26.0	12.8	13.1	13.2	8.3	22.5	180.0	20.2
12.54	14	27.0	23.1	21.9	26.1	13.0	13.2	13.2	8.1	22.9	88.0	18.5
13.20	26	26.0	23.0	22.4	25.4	13.4	13.6	13.5	7.7	24.0	140.0	19.8
13.35	15	25.0	23.1	22.6	25.1	13.3	13.7	13.7	7.7	23.1	77.0	19.6
14.00	25	25.4	23.1	22.3	25.4	13.2	13.5	13.7	7.3	23.4	132.0	18.9
14.16	16	26.5	23.5	22.4	26.5	13.2	13.6	13.6	7.3	23.1	83.0	19.6
14.35	19	25.5	23.2	22.6	25.6	13.5	13.7	13.7	6.9	24.0	94.0	17.9
14.50	15	24.3	22.8	21.8	23.8	13.2	13.7	13.7	6.8	22.8	82.0	17.8
15.00	10	26.0	23.3	22.2	26.0	13.3	13.6	13.6	6.7	23.7	51.0	13.9

CALCULATED DATA OF THE HEAT LOSS OF THE ZEOLITE-WATER REFRIGERATOR

DATE: 25 JAN. 1987

TIME	TIME INTERVAL	OUTSIDE 1	WALL 2	TEMP. 9	FRONT DOOR 8	11	WALL TEMP 7	RECEIVER/EVAPORA. 5	6	AMBIENT 3	MELTEN WATER 4	WATER TEMP	WALL TEMP CALC (T1+T8)/2	HEAT THRO. W
	min	BACK VIOL. DOT C	TOP RED DOT C	BOTTOM BLACK -0 C	OUTSIDE RED-0 C	INSIDE BLUE-0 C	INSIDE VIOLET-0 C	TOP BLUE DOT C	BOTTOM YEL. DOT C	BLACK DOT C	ml	C	C	W
11.00		32.9	31.1	30.0	34.9	18.1	19.8	21.0	16.0	29.6			33.9	
11.20	20	34.3	31.5	31.1	34.7	17.5	18.8	19.8	14.8	31.5	130.0	16.4	34.5	
11.35	15	34.2	31.7	30.4	34.3	17.2	18.3	19.1	14.0	30.8	89.0	16.1	34.2	
12.00	25	33.0	32.3	30.1	35.7	17.7	18.3	18.5	13.0	31.5	140.0	16.8	34.4	
12.40	40	34.3	33.3	31.4	36.9	18.1	18.4	18.5	11.6	31.3	180.0	20.2	35.6	
12.54	14	37.1	34.1	31.0	37.0	18.4	18.5	18.5	11.3	31.8	88.0	18.5	37.1	
13.20	26	35.7	33.9	31.7	36.0	18.9	19.1	18.9	10.7	33.4	140.0	19.8	35.9	
13.35	15	34.3	34.1	32.0	35.6	18.8	19.2	19.2	10.7	32.1	77.0	19.6	34.9	
14.00	25	34.9	34.1	31.6	36.0	18.7	19.0	19.2	10.2	32.5	132.0	18.9	35.4	
14.16	16	36.4	34.7	31.7	37.6	18.7	19.1	19.1	10.2	32.1	83.0	19.6	37.0	
14.35	19	35.0	34.2	32.0	36.3	19.1	19.2	19.2	9.6	33.4	94.0	17.9	35.6	
14.50	15	33.3	33.6	30.8	33.7	18.7	19.2	19.2	9.5	31.7	82.0	17.8	33.5	
15.00	10	35.7	34.4	31.4	36.9	18.8	19.1	19.1	9.3	33.0	51.0	13.9	36.3	

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WATER
TEMP

C

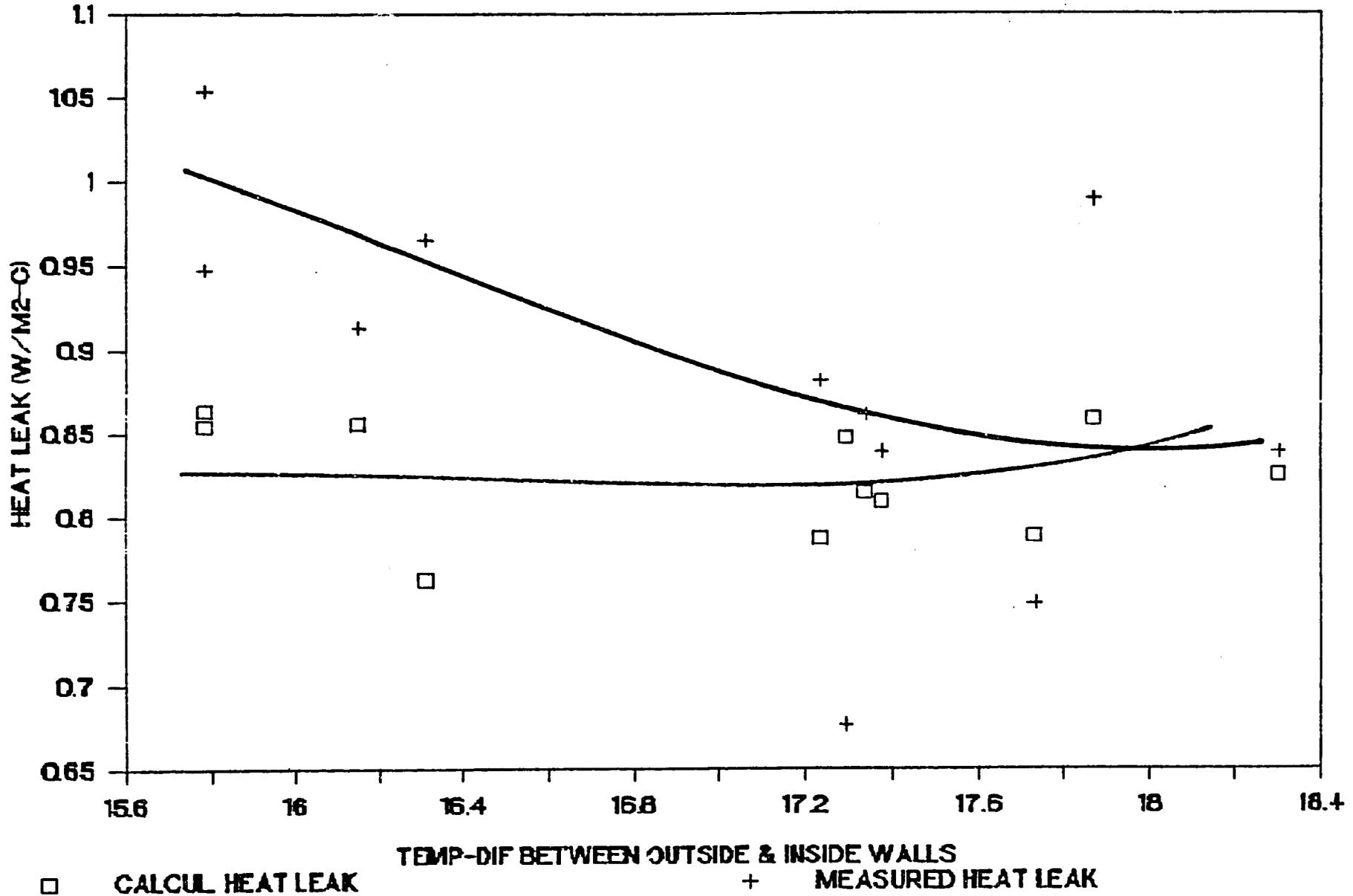
16.4
16.1
16.8
20.2
18.5
19.8
19.6
18.9
19.6
17.9
17.8
13.9

WATER TEMP	WALL TEMP (SIDES) (T1+T8)/2	CALCUL. CONDUCT HEAT THRO.WALL	ACTUAL MELTING HEAT IN BOX	COOLING FOR OTHERS IN BOX	AVERAGE OUTSIDE WALL TEMP	AVERAGE INSIDE WALL TEMP	AVERAGE TEMP. DIFF.	OVERALL HEAT LOSS COEF. CALCUL	OVERALL HEAT LOSS COEF. ACTUAL
C	C	W	W	W	C	C	C	W/m2-C	W/m2-C

	33.9								
16.4	34.5	25	36	4.9	32.9	17.1	15.8	0.85	1.05
16.1	34.2	26	33	4.9	32.6	16.9	15.8	0.86	0.95
16.8	34.4	26	31	3.4	32.8	16.6	16.2	0.86	0.91
20.2	35.6	28	25	3.1	34.0	16.7	17.3	0.85	0.68
18.5	37.1	29	35	1.7	34.8	16.9	17.9	0.86	0.99
19.8	35.9	27	30	1.9	34.3	17.0	17.3	0.81	0.86
19.6	34.9	26	29	0.0	34.0	16.7	17.2	0.79	0.88
18.9	35.4	26	29	2.0	34.1	16.7	17.4	0.81	0.84
19.6	37.0	28	29	0.0	35.1	16.8	18.3	0.82	0.84
17.9	35.6	26	28	2.6	34.4	16.6	17.7	0.79	0.75
17.8	33.5	23	30	0.8	32.9	16.6	16.3	0.76	0.96
13.9	36.3	27	28	1.2	34.6				

HEAT LEAK V.S. TEMPERATURE DIFFERENCE

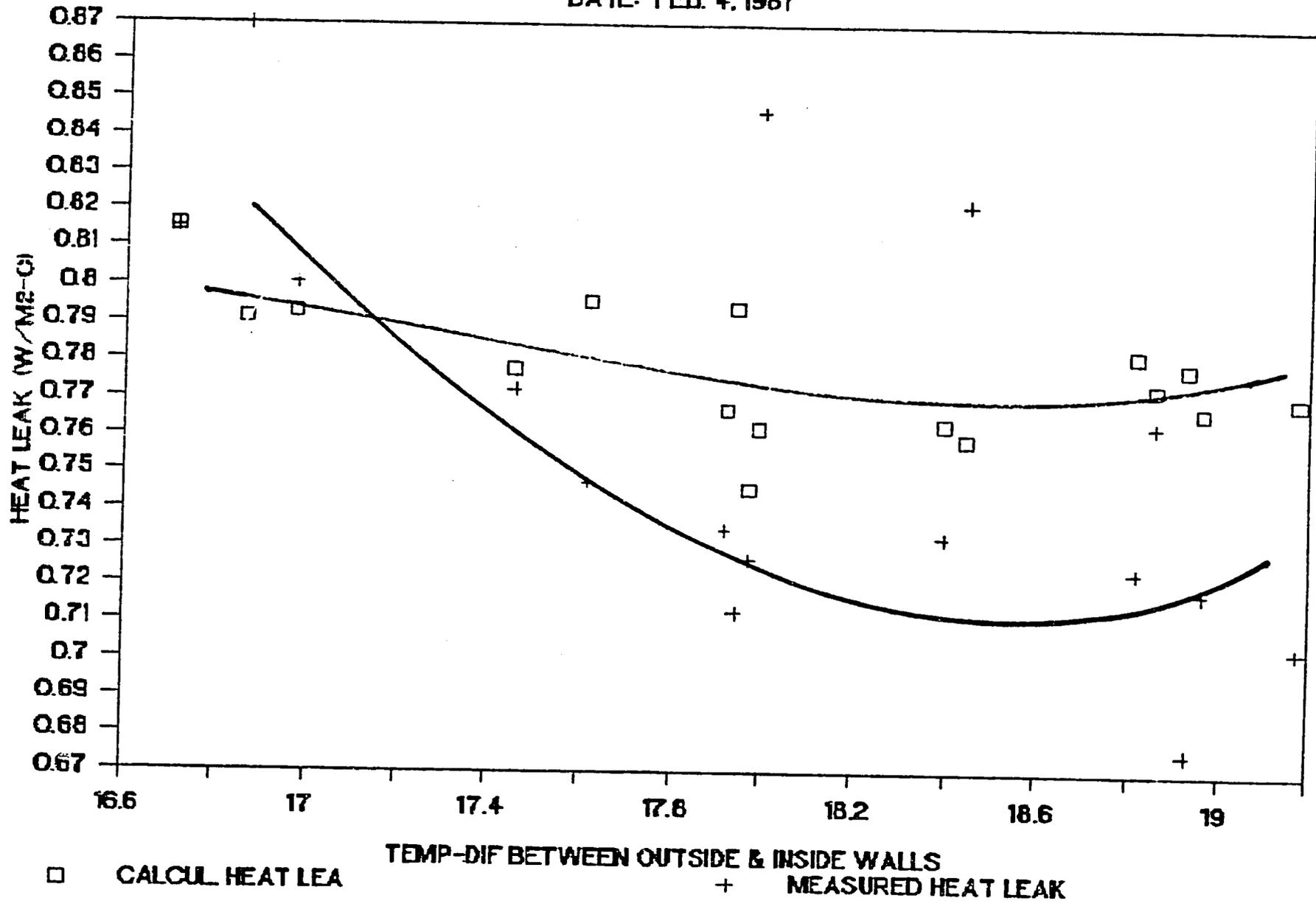
DATA: JAN 25, 1987



5

HEAT LEAK V.S. TEMPERATURE DIFFERENCE

DATE: FEB. 4, 1987



6.5