DESIGN AND THERMAL ANLAYSIS OF A SOLAR POWERED COLD STORAGE WARE-HOUSE USING A PHASE- CHANGE MATERIAL

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ABSTRACT

The paper considers the storage of 1000 kg of oranges at a temperature conservation of 1^{o}C with a requirement of 85% to 90% and air circulation velocity of' 0.3 m/s in the Ware house.

Based on the temperature of utilisation, the paper discusses the physiro-chemical problems inherent with a phase change material for a desired cyclic performance and gives the thermo-physical properties of NaF/N_2O eutectic mixture used for the analysis.

Using the optimal solar radiation for Nsukka (0.9898 KW/m^2) the paper shows the calculations of the heat requirements and the surface area of the collector/generator. The paper discusses the design parameters of the components of the aqua-ammonia absorption system. The specification of the ware house based on calculations is also discussed.

A model for the simulation of heat transfer phenomenon during storage processes is presented. The paper demonstrates therefore the possibility of the storage in rural areas of Nigeria of perishable agricultural products at different conservation temperatures corresponding to the solidification temperatures of phase change materials using cheap and abundant source of energy.

NOMENCLATURE

T	^{o}C	Temperature
T	S	time
λ	W/m^{o}C	thermal conductivity
C_P	J/km^{3o}C	specific heat
ρ	kg/m^3	density
K	W/m^{2o}C	Convective heat transfer coefficient
U	W/m^{2o}C	Overall coefficient of exchange
a	M^2/3	thermal diffusivity
β	K^{-2}	volume expansion coefficient
υ	m^{2}S	kinematic viscosity
Σ	m	thickness of solid
f	W/m^{2o}C	Conductance
v	m^3	volume
e	m	container wall thickness
S	m^2	surface
L	KJ/kg	Latent beat
Ra	g/3 T^3/a	Rayleigh number
Nu	h/1c	Nusselt number
Pr	v/a	Praudt1 number

SUBSCRIPT

1	Solid	e – exterior
2	Liquid	i – interior
f	fluid	O – outside
WC	container wall	
w	wall	
a	air	

INTRODUCTION

The problems posed by the conservation of perishable food items in the rural areas of this country justify the interests shown on the application of solar energy for refrigeration.

Solar energy has the limitation of intermittency and weather dependence. Consequently, a solar powered system requires a back-up or auxiliary solar thermal energy storage device for its continuous application during the period when it is not available.

One such device of solar thermal energy storage for low temperature application is the utilisation of a phase change material (PCM). A phase change material stores and releases energy at nearly constant temperature, around its melting/ solidification point. Another advantage of the PCM is the high storage energy density which implies that larger amount of energy can be stored into a small PCM mass.

Some work lit [2], [3], [4] have been done on solar refrigeration systems-solid absorption/Calcium Chloride refrigerator; solid absorption refrigerator, liquid absorption/aqua-ammonia refrigerator and photovoltaic compressor vapour refrigerator. Some of these systems are already commercialized.

Abat[5], Gawron and others [6], Schroder[7], Eekerlin and others [8], Yooeda and others [9] and Schneider and others[10] have shown the characterisation and possible utilisation of eutectic mixtures phase change materials for thermal energy storage. Onyejekwe [11] made use of $NaC1/H_2O$ for the preservation of medicaments and vaccines at a temperature of -20^{0}C.

Not much work has been done in the coupling of these separate components solar
collector/generation; solar energy thermal storage system; and ware housing for the preservation on perishable food items like oranges.

Part of the motivation of this work is therefore the need for conservation of fruits all year round, thereby reducing waste during harvest period and assuring supply during off-season.

Material Selection:

Phase Change Material

In the selection of a phase change material for use in a system, consideration has lobe given to the thermodynamic, kinetic, chemical and economic properties of such a material.

The thermodynamic properties include – melting point suitably matched to the temperature of utilization 1^{0}C in the case of oranges [12]

- High latent heat of fusion, specific beat, and density
- Small volume change during phase change.

The kinetic criteria require that the phase change material shows little or no supercooling during freezing and- reasonable rate of crystallization.

The chemical criteria include: that the PCM should show chemical stability and no degradation during the cyclic performance, compatibility with the container, non-poisonous, non-flammable and non-explosive for safety operations. Economic consideration demands that the PCM should be abundant and cheap.

The criteria for selection of PCM lead to the choice of NaF/H_2O with the following properties:

	Liquid	Solid
Density (kg/m^3)		
-at 20^{0}C	1.040	
at the temp. of fusion		0.958
thermal conductivity (W/m^{o}C)		
-at 20^{0}C	0.6427	
at the temp. of fusion		1.254
Specific Heat (J/kg^{o}C)		
-at 20^{0}C	3.85	
at the temp. of fusion		2.72

Latent Heat (Massie) 312.7KJ/kg
Latent Heal (Volumlic) 309.2 KJ/cm^3
Temperature of fusion – 35^{0}C.

To reduce the volume of expansion (8.9%) of the PCM during solidification which could cause fatigue (thermal stress) on the walls of the container and to homogenize the process 0 crystallization we add 3% by vol. of KF/H_2O.

Container

Based on compatibility, availability, reasonable thermal conductivity and low cost, polyethylene of the following properties:
thermal conductivity 0.4 (W/M^{o}C)

specific heat1757(J/kg^{o}C)
density 920 (kg/m^3)
temperature of utilisation -30^{o}C - 60^{o}C, is chosen as material for the container of PCM.

For space economy, the provision of thermal contact between the refrigerant in the evaporator tubes and for simplification of the thermal analysis, the PCM is, housed in a flat-plate, evaporator (Figure 1)

Building: This is the ware house for the storage of oranges at 1^oC and relative humidity of between 85% to 90%. The material for consideration include: the reduction to minimum the infiltration load; material of low thermal conductivity; strength, especially the roof in order to be able to carry both the collector and evaporator for long life span of system cost and availability.

Based on the above, the following materials are selected for the construction of the building.

Common building bricks (150mm to 290mm thickness) of density ρ = 1820(kg/m^3) thermal conductivity λ = 0.810 (W/m^{o}C) and specific heat C_P = 0.84(KJ/Kg^{o}C) are used for the walls; and for the roof, we use a reinforced concrete ρ =1920 (Kg/m^2); build up flat roofing with 13mm plaster of ρ = 1762 = 1762 (Kg/m^3), λ = 0.721 (W/m – C) and C_p = 0.84.. (KJ/Kg^{o}C) and thermal conductivity λ = 1.099 (W/m^{o}C). The roofing is supported with iron rods, to carry the evaporator and solar collector/generator. Glass window ρ = 2350 (Kg/m^3), λ = 0.815 (W/m^{o}C) and C_p = 0.84 (KJ/Kg^{o}C) and wooden door ρ = 480 (Kg/m^3), λ = 0.072 (Kg/m^3) C_p = 24(KJ/Kg^{o}C) are used.

The floor is made up of concrete slab resting on a grade or fill depending on the texture of the soil. The poor thermal conductivity of the concrete reduces the frost upheaval of the foundation soils and consequently there is no need for auxiliary defrosting for the soil since the temperature of utilization is above zero degree.

ulifisation is above zero degree.

Refrigerant: The following criteria for selection of a refrigerant in the absorption system are considered. High heat of vaporization for low fluid circulation rates; the boiling point, as this influences the evaporator and generator pressures; low specific heat to reduce the losses incurred at the condenser during cooling of the refrigerant vapour and those incurred during the throttling process and chemically stable throughout the range of operation and should not undergo inreversible reaction with any material in the system. Reversible reaction between the refrigerant and absorbent is very necessary. Based on the above ammonia (refrigerant 717) with the following properties is selected:

ρ = 65355 (Kg/m^3)
C_p = 4.56 (KJ/Kg°C)
λ 0543(W/m°C)
a = 1.825 x 10^{-7} (m^2/s)
ρ = 2.45 x 10^{-3} k^{-1}
Boiling point -33.33°C; freezing point = 77.8°C
v = 0.368 x 10^{-6}m^2/s

Absorbent: When considering the criteria for selection of the absorbent such properties as the boiling point, chemical stability, viscosity, heat capacity and non-corrosiveness are considered but in practice it is the criteria based on compatibility and practicability are the necessary conditions for absorbent-refrigerant combination. These include mutual solubility, low specific heat, low viscosity, vapour pressure of the absorbent relative to the refrigerant. In addition, the chemical properties of the combination should not attack the component of the system.

Based on the, above conditions, water, the properties of which are widely known, is chosen as the absorbent.

Determination of the Refrigeration Load

The cooling load of refrigeration could be expressed in the form

$$Q_T = Q_r + Q_w + Q_d + Q_p + Q_m \quad (1)$$

where Q_r is, the heat that enter the refrigerated space by direct radiation through glass. Q_r = A U x ΔT.

Q_w is the wall leakage load calculated from the relation:

Q_c = A x U x ΔT

U is calculated from

$$\frac{1}{U} = \frac{1}{f_1} + \frac{X_1}{\lambda_1} + \frac{X_2}{\lambda_2} + - + \frac{X_n}{\lambda_n} + \frac{1}{f_o} \quad (2)$$

where

$X_1, X_2 \ldots X_a$ = material thickness

$\lambda_1, \lambda_2 \ldots \lambda_n$ = corresponding thermal conductivity

For the Area, we differentiate among East facing wall West facing wall and roof. The optimum insulation thickness is determined as 60mm.

Qd = Heat into the refrigerated space from the ambient

$$Qd = M(ho - hi) \quad (3)$$

where,

Qd = (Infiltration rate) x (Enthalpy change)
Q_P = the product heat given by the equation
Q_P = (M) (C) (DT) (4)

A more useful parameter ill the design is the cooling rate given as

$$Q_P 1 = \frac{Q_P}{Desired\ cooling\ time\ in\ second} \quad (5)$$

Respiration heat of the oranges was also considered based on Dossat [13]. Q_m are taken as miscellaneous load which include human occupants and heat producing equipment. This is determined from Dossat[13].
Q_m = Number of people x heat equivalent (6)
The total refrigeration load Q_T is the summation of the right side of equation (1) plus a safety measure of 5 - 10% and it is

$$Q_T = (Q_r + Q_c + Q_d + Q_P + Q_m) \times 1.1 \times \frac{24}{operating\ time} \quad (7)$$

This was found to be 30.672.kW.

System Design and Specification
Collector/Generator
The optimum solar radiation for Nsukka is 0 .9898 KW/m^2 and this forms the basis of the design analysis. The following specifications are calculated.
Cooling load = 30.672KW
Evaporator temperature = -8^{o}C
Evaporator pressure = 3.1602 bar
Condenser temperature = 35 ^{o}C
Condenser pressure = 13.552 bar
Concentration of strong ammonia solution = 0.582
Absorber pressure = 3.1602 bar
Generator pressure = condenser pressure = 13.552 bar
Heat exchange pressure = 13.522 bar
Temperature of strong solution reaching generator = 75 ^{o}C.
Collector surface = 49m^2
Number of tubes required = 99 of internal diameter 0.5cm and outer diameter 0.9cm

Condenser:
From the calculations, the following specifications for the condenser are used .
Shell side (NH_3)
D_{si} = 22.225cm
Number of baft1es = 4
Baffle spacing = 20cm
Passes = 1
Length = l00cm
Table side (water)
Number of tubes = 35
Length of tubes = 100cm
Pitch - 3.175cm
Tube types is 14 BWG, 2.54cm OD
Passes = 1
Pressure drop across the condenser = 3.0550 x 10^{-4bar}
Evaporator - flat Plate Tubed Type
Plate Bank Model = 63 O.D.
width of the plate = 0.3m
Length of plate 3m
Height of plate = 0.44m
Number of Plate bank = 6
Number of tubes per plate bank = 25
Tube type is 14BWG of 0.834LD
Tube material is Steel (1020) of λ = 54W/m^{o}C
Plate material is polyethylene
Type of plate bank circuit is in series.
Figure (2) shows the sketch of tube arrangements in the plate banks and how the banks are arranged for mounting.

Heat Exchanger:
Type – double pipe
No. of legs 48
Inner tube: I.D 12.8cnm 0. D 14. 2cn
Outer tube: I.D 15.4cm, 0. D 16.8cm
Inner tube fluid - Cold streng Amtnnia solution of Inlet tmp. - 21 ^{o}C
Mass flow rate = 0.19764 Kg/s
Concentration = 0.52
Outer tube fluid – hot weak solution of inlet temperature - 85 ^{o}C
Outlet temperature – 21.20 ^{o}C
Mass flow rate – 0.16940 kg/s
Concentration – 0.44
Total pressure drop in the annulus – 4.9547 x 10^{-4} bar.

The Absorber:
Shell side (strong NH_3 Solution)
D_{is} = 22.2cm
Number of Baffles 5
Baffles spacing 20cm
Passes 1
Length 116cm
Material steel (λ = 54 W/m ^{o}C)
Table side (Cooling Water)
Number 0f tubes - 35
Length of tube – 166cm
Pitch – 3.175cm
Passes – 1
Other features of the system like the solution Pump the expansion value and the rectifier

were also considered.
An assemble of the different components is shown in figure (3).

Modeling
The model considers the period when there is no solar energy which implies that the phase change material keeps the ware house at the required temperature of 1 ^oC
Since the PCM is housed in a flat – plate evaporator mounted on the flat – roof, the 'cold' output of the PCM as it gives out its heat of fusion is considered mono – directional with the downward direction of Y – axis considered positive. Furthermore, such thermal properties - λ, ρ, C_P, υ are considered constant for each phase solid/liquid.
The generalized equation are written for the phase

$$\frac{\partial T_1}{\partial t} = \frac{a_1 \partial^2 T_1}{\partial Y^2} \quad (7)$$

With $a_1 = \lambda_1/\rho_1 C_{P1}$
For the liquid phase

$$\frac{\partial T_2}{\partial t} = a_2 \frac{\partial^2 T_2}{\vartheta Y^2}$$

With $a_2 = \lambda_1/\rho_1 C_{P2}$
and for the solid/liquid interface,

$$-\lambda_1 \frac{\partial T_1}{\partial Y} + \lambda_2 \frac{\partial T_2}{\partial Y} = L\rho \frac{ds}{dt} \quad (9)$$

$$T_1 = T_2 = T_f$$

The storage phase leads to a progressive diminution of the solid state with convection becoming important
Using the explicit scheme of the finite element method equation (7) after doscritisation is written as
$V_2\rho_2 C_{P2} (T_{2t} +1- T_{2t})/\Delta_t = h_i . S(T_{wc} – T_2)t + h_f . S (T – T)t$ (10)
or $T_{2t+1} = \Delta_t Z\ T_{we}\ t + \Delta t\ Z^1\ Tt + \{t - \Delta T (Z + Z^1)\}\ T_{2t}$ (10a)

where Z = $\frac{h_i.*S}{V_2\rho_2 C_{P2}}$; $Z\ \frac{h_f.S}{V_2\rho_2 C_{P2}}$

with the condition of stability
ΔT (Z +Z) < 1
h_i the internal convective coefficient is determined from the correlation due to Bulkovsky and Polikov.
Nu = 0.22 A $^{-0.25}$ (Pr. Ra/Pr + 0.2)$^{0.25}$ (11)

$$(s\rho\rho\ C_P)\ wc \frac{T_{wct+1} - T_{wct}}{\Delta t} = K_i S_{wc}(T_2 - T_{wc})t + K_c S_{wc}(T_a - T_{wc})^t \quad (12)$$

$$where\ K_i = \frac{1}{1/hi + e_{wc}/2\lambda_{wc}};\ K_e = \frac{1}{1/he + e_{wc}/2\lambda_{wc}}$$

or $T_{wct+1} = \Delta t\ Z11_{T2t} + \Delta t\ Z111_{Tat} + \{1 - \Delta t(Z^{11} + Z^{11})\}T_{wct}$ (12a)
Where $Z^{11} = \frac{h_i}{(e\rho C_P)_{wc}}$
The rate of diminution of the thickness of solid is determined from the energy balance at the inter face.

$$h_f(T_2 - T_f) = \rho_1 L\ \frac{\Delta\varepsilon}{\Delta t}\ \frac{T_{11} - T_f}{\Delta Y(t)} \quad (13)$$

hence

$$\Delta\varepsilon = \frac{\Delta t}{\rho_1 L}\{h_f(T_2 - T_f)\frac{h_f(T_{1f} - T_f)}{\Delta Y(t)}\} \quad (13a)$$

Using the computer Apple 2, a programme making use of equation - (10a), (12a) and (13a) is made for the temperature distribution of the air inside the ware house.

CONCLUSION
For the sake of brevity, the detailed calculations are not presented here, they can be found in ref, [14].
Encouraged by the results of the model based on the design parameters and calculations which showed temperature of the ware house remained at 1^{o}C for 16 hours, a prototype of the system is being built.
It is hoped that the performance of the system would be compared with the result of this work.

ACKNOWLEDGEMENT
This work is sponsored by the Nigerian Federal Ministry of Science and Technology.

REFERENCES
1. Iloeje, O.C. Closed Cycle Solar Refrigeration with the Calcium Chloride system, Proceeding of the Workshop on the Physics and Technology of Solar Energy Conversion, Ibandan, 1985

2. Alizadreh, S., Bahav F. and Geoola, F. "Design and Optimization of an Absorption Refrigeration System Operated by Solar Energy", *Solar Energy* Vol. 22, 1979.

3. Ward, S. Solar Absorption Cooling Feasibility, Solar Energy Vol. 22, 1979.

4. Rodolphe De Hartoulari and Louis-Claude Dufour, Affinity Thermal Machines and Metallic-Ammonia-Ammoniates Systems, *Solar Energy* Vol. 33, No 6, 1984.

5. Abat, A., Low, Temperature Latent Heat Thermal Energy Storage: Heat Storage Materials. *Solar Energy*, Vol. 30, No. 2, 1983.

6. Gawron, K and Schroder, J. Properties of some salt hydrates for latent heat. storage, *Energy Research*, Vol. 1, 1977.

7. Schroder J., R and D of Systems for Thermal Energy Storage, Proc. Seminar New Ways to Save Energy, Reidel, Dordrecht 1980.

8. Eckerlin, P, Klopfer A, and Schroder; J. Unter Suchung und Entwicklung von Systemen zur speicherung thermischer Energie in Tempera turbereich Zwischeu, - 25°C und + 150°C. Kommission der Europaischer Gemeinschaften EUR 6936, 1980.

9. Yoneda N. and Yakanashi, S., Eutechtic mixtures for Solar Heat Storage, *Solar Energy*, Vol. 21, 1978.

10. Schneider and Berger, L utilisation de la chaleur, latente des materlaux, Cahiers du C. S. T. B. No. 18, Seot. 1978.

11. Onyejekwe, D.C., Cold Storage Using Eutetic Mixture of $NaCl/H_2O$: an application to photovoltaic Vapour Compression Freezers. Paper accepted for publication in Solar and Wind Technology Journal.

12. Rose, D. H. Wright, R. C. and Wahatman, T. M., The Commercial Storage of Fruits, vegetables and Florist Stock, U. S. Department of Agriculture Circular No. 278

13. Dossat, .J. Roy, Principles of Refrigeration, John Wiley and Sons Inc. 2nd editions, 198l.

14. Onyejekwe., D. C. Design, production and Thermal Analysis of a Solar Ware-house for Fruits Storage Using Eutectic Mixture. Technical report. Department of Mechanical Engineering, University of Nigeria, Nsukka.

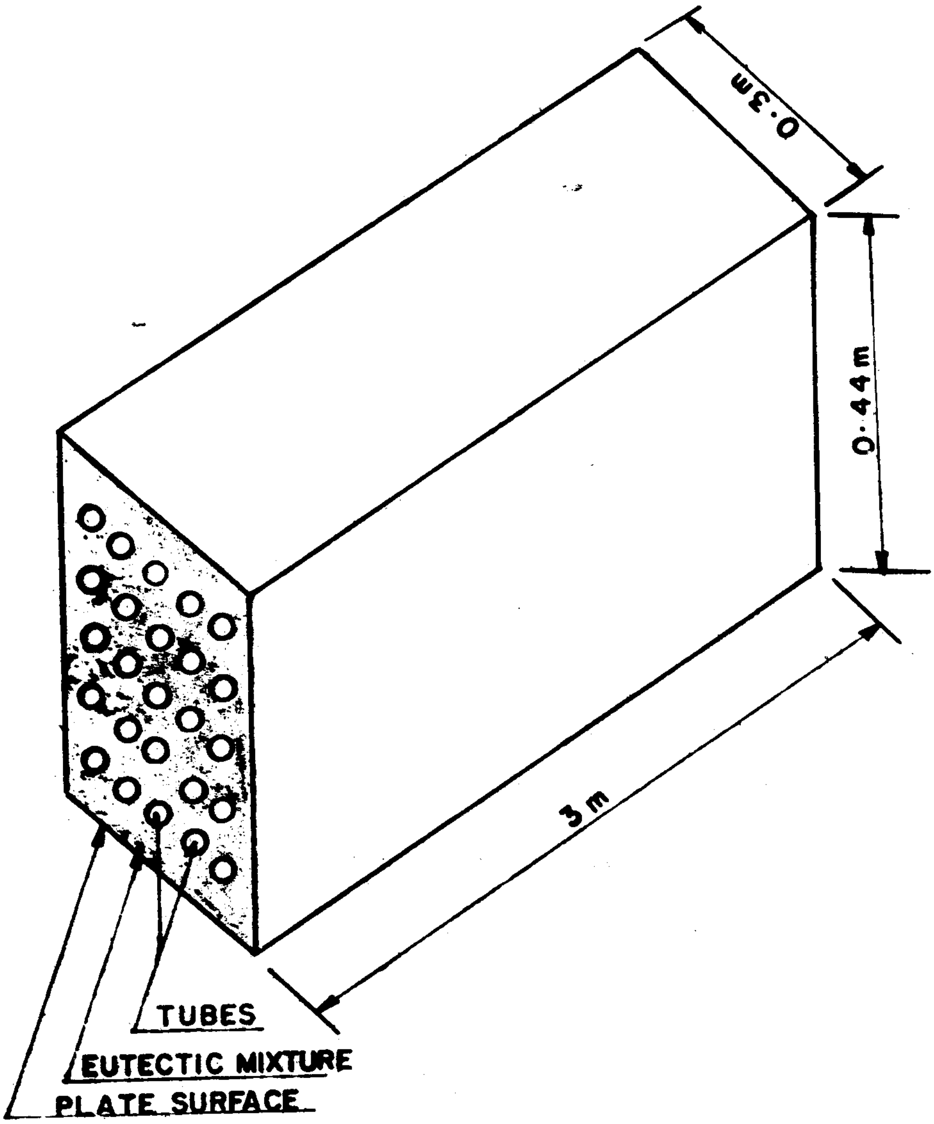

FIG. I. PLATE BANK SHOWING TUBES AND EUTECTIC MIXTURE ARRANGEMENT.

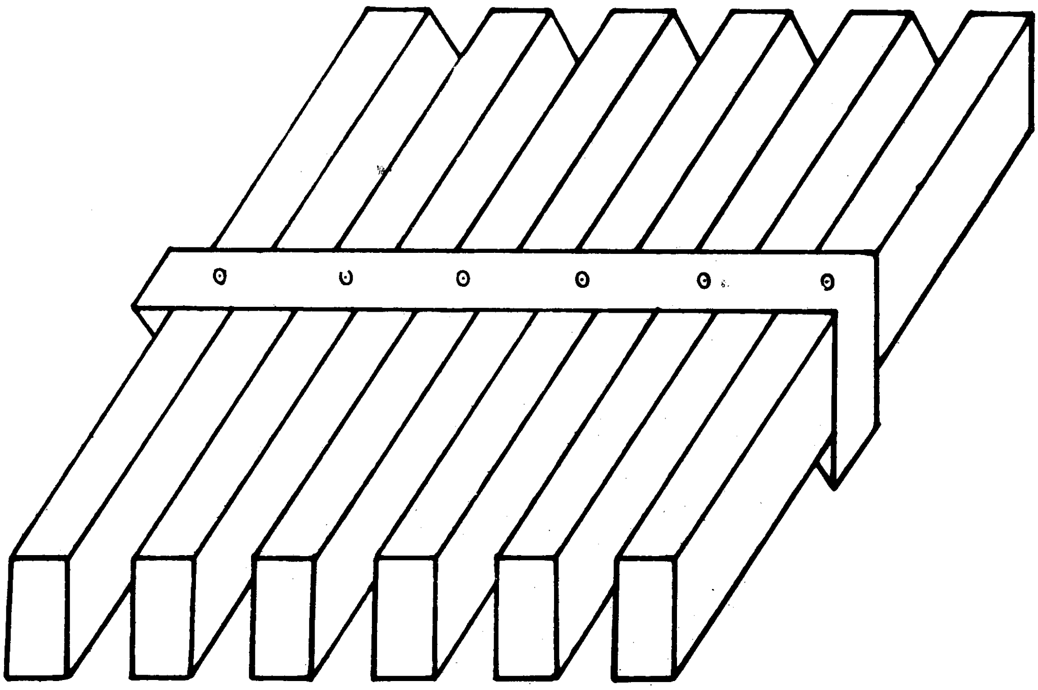

FIG. 2(a): PLATE BANKS ARRANGEMENT FOR CEILING MOUNTING

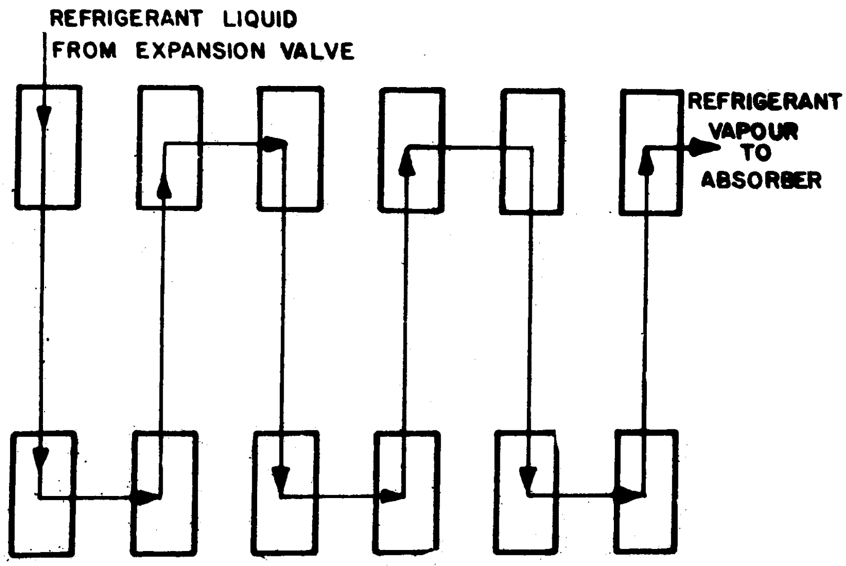

FIG. 2 (b): ENDS PLATE BANKS VIEW SHOWING REFRIGERANT FLOW PATTERN.

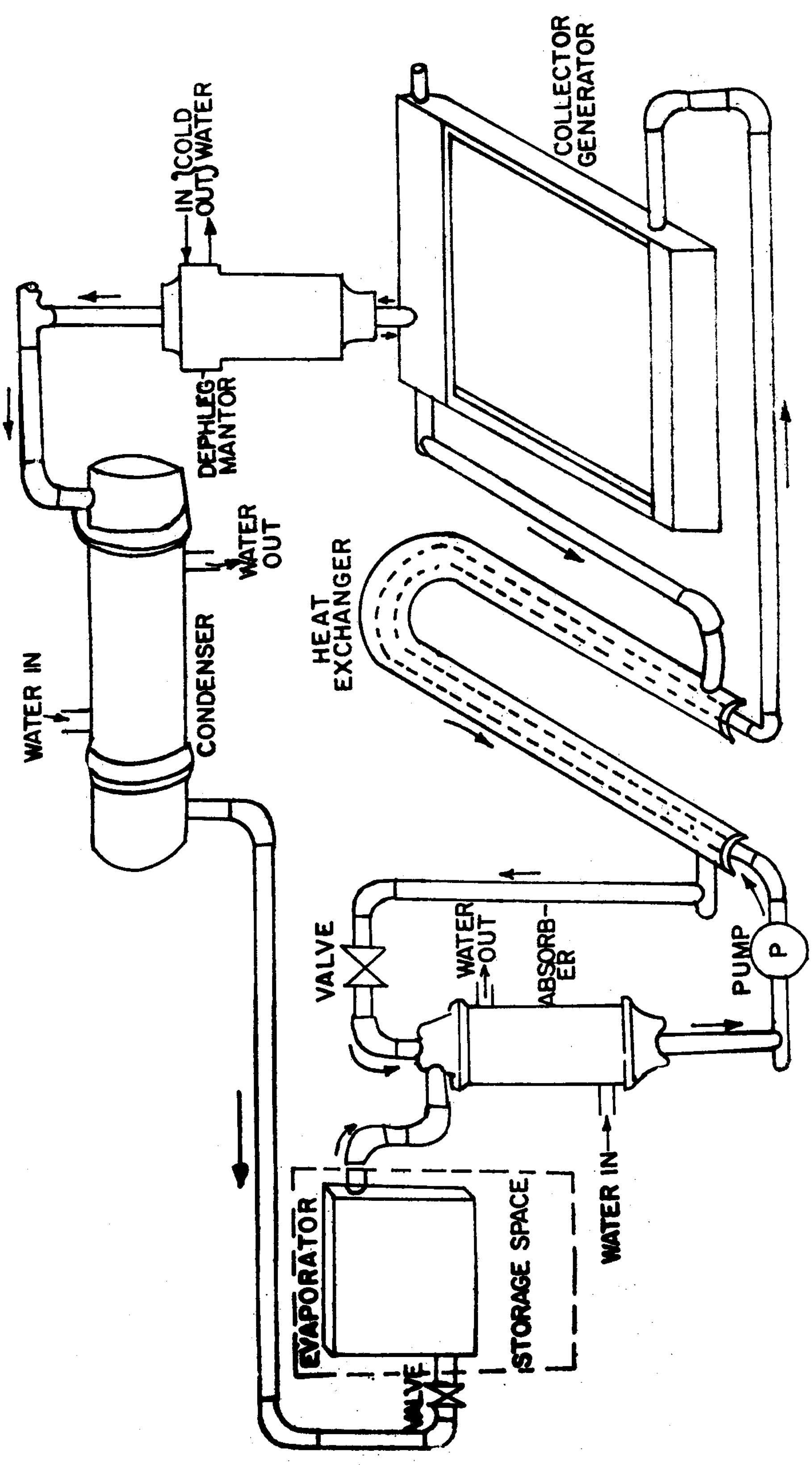

FIG. 3: ASSEMBLY SKETCH FOR STORAGE SYSTEM