SOLAR BRINE EVAPORATION PROCESS

by

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ABSTRACT

The use of "Solar Brine Evaporators" in the production of natural sodium sulfate has been investigated. Three types of evaporators were used ; "open" ,"closed" and "closed with suction". Evaporation efficiencies for all three types of evaporators have been determined.

Experimental results based on small scale evaporators using thin film evaporation, have succeeded in increasing brine concentrations from 5% to 30% . As a result crystal production capacity increased by 55% .

For large scale applications due to ease of operation " open " type evaporators must also be considered although their evaporation efficiencies were not as high as the other two types.

ÖZET

Bu çalışmada "Solar Brine Evaporator" larının tabii sodyum sülfat üretimindeki kullanımı tetkik edilmiştir. Üç tip evaporatör kullanılmıştır ; " açık " , " kapalı " "kapalı ve emişli." Her üç tip evaporatörün evaporasyon randımanları araştırılmıştır.

Küçük deneysel evaporatörler ile filim tabakası halinde buharlaştırma sağlanarak tuzlu solüsyon konsantrasyonu 5% den 30% a kadar yükseltilebilmiştir ve bu sayede sodyum sülfat kristal üretim kapasitesi 55% artmıştır.

Büyük çaptaki uygulamalarda randımanları diğer tiplere nazaran düşük olmasına rağmen "açık" tip evaporatörler kullanım kolaylıklarından ötürü tercih nedenidir.

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A : Evaporator surface

$^{\circ}Bè$: Specific gravity scale

C : Air space conductance

C_{SOL} : Specific heat of the solution

e : Percent evaporation

$\dot{e}$: Evaporation rate

$E(n)$: Evaporation ratio of the n^{th} evaporator

$E_L(n)$: Evaporation ratio of the n^{th} evaporator at distance L

h_i : Inside heat tr. coefficient

h_o : Outside heat tr. coefficient

Δh_{T_E} : Latent heat of vaporization at temp T_E

$h_{gs_{TE}}$: Saturated gas enthalpy at temp T_E

$h_{fs_{TE}}$: Saturated liquid enthalpy at temp T_E

k_G : Conductivity coefficient of glass

k_m : Conductivity coefficient of metal sheet

k_S : Conductivity coefficient of isolation material

k_w : Conductivity coefficient of wood

$L(n)$: Length of the n^{th} evaporator

m_{H_2O} : Mass of water

m_{SOL} : Mass of the solution

P : Solar input power 1.07 kw/m^2

Q_{TOT} : Total heat losses

Q_B : Amount of back losses

Q_E : Evaporation energy

Q_F : Amount of front losses

$\bar{Q}$: Fundamental unit of energy

T_c : Temperature at the center of evaporation plate

T_s : Temperature at the sides of evaporation plate

T_m : Temperature of the metal sheet

T_i : Mean inside temperature

T_o : Outside air temperature

U_F : The overall heat tr. coefficient at the front of the evap.

U_B : The overall heat tr. coefficient at the back of the evap.

$W(n)$: Width of the nthevaporator

ΔX_G : Thickness of the glass

ΔX_m : Thickness of the metal sheet

ΔX_S : Thickness of the isolation

ΔX_W : Thickness of the wooden frame

V_{IN} : Input volume

V_{OUT} : Output volume

$V_{l\,OUT}$: Output volume at distance L from inlet

V_E : Evaporation volume ($V_{IN} - V_{OUT}$)

$\alpha(\psi)$: Angular solar power coefficient

δ_{IN} : Film flow thickness at inlet

$\delta_{l\,OUT}$: Film flow thickness at distance L from inlet

ϕ : Evaporator tilt angle

ψ : Angle with the normal

ρ_{IN} Specific gravity of input solution

ρ_{OUT} : Specific gravity of output solution

$\rho_{H_2O}|_{T_E}$: Specific gravity of water at temp. T_E

η : Efficiency

I: INTRODUCTION

"Solar Brine Evaporation Process" is a continuous brine evaporation system, which uses solar radiation as the energy source. In this process the dilute brine solution flows above a tilted metal plate as film flow and during this flow period from the top to the bottom of the container brine evaporates and the output concentration increases.

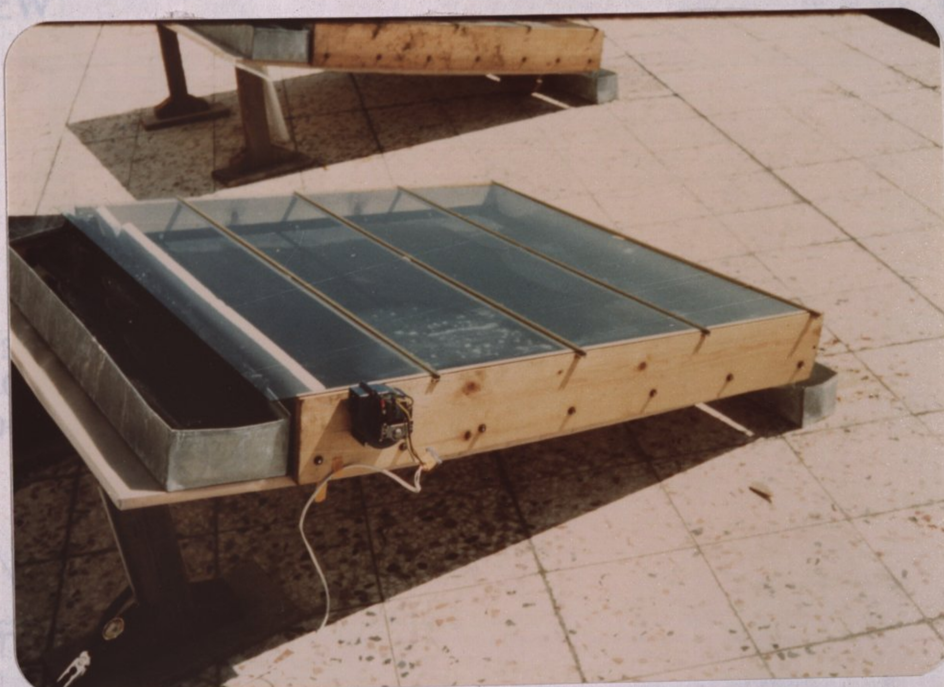
Brine evaporation is the most important process in the production of natural sodium sulfate (Na_2SO_4), which is a raw material for the kraft paper and glass industry and a filler for detergents. In Turkey natural sodium sulfate exists in the form of dilute brine solution and can be obtained from salt lakes in middle and southwest of Anatolia. Bolluk, Tersakan and Acıgöl are some of these lakes.

In the conventional natural sodium sulfate production dilute lake solution is concentrated in large ponds from about 5% to 15%, to produce sodium sulfate crystal. But using "Solar Brine Evaporators" concentration can be increased up to 30% and the crystal production capacity can be increased by 55%.

This work was mainly application oriented, therefore the small scale experiments and their results were very important for large scale applications.

Although the "closed" type and the " fan suction" type evaporators gave higher efficiencies and better results, with around 1.2 liter/hr m^2 evaporation rate the "open" type evaporator with low initial cost and low maintanance requirements may be preferred for large scale applications. Furthermore, the efficiency of the "open" type evaporators increase sharply at low relative humidities and high radiation intensities and the lakes are located at such suitable areas.

The result of this work show that the use of "Solar Brine Evaporators" in the production of natural sodium sulfate, will add many advantages to the classical approach used presently in the industry.



THE EXPERIMENTAL SET UP

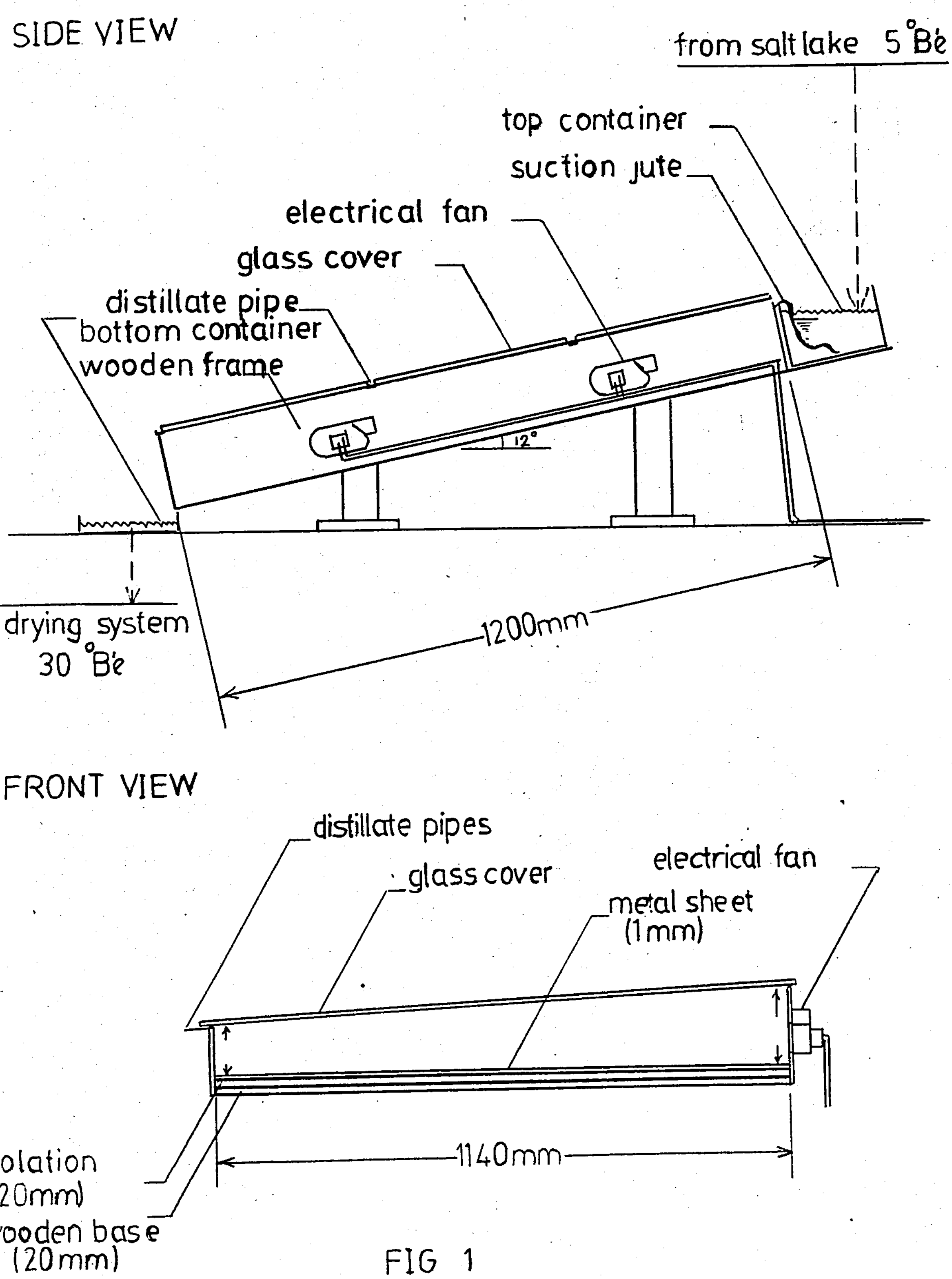
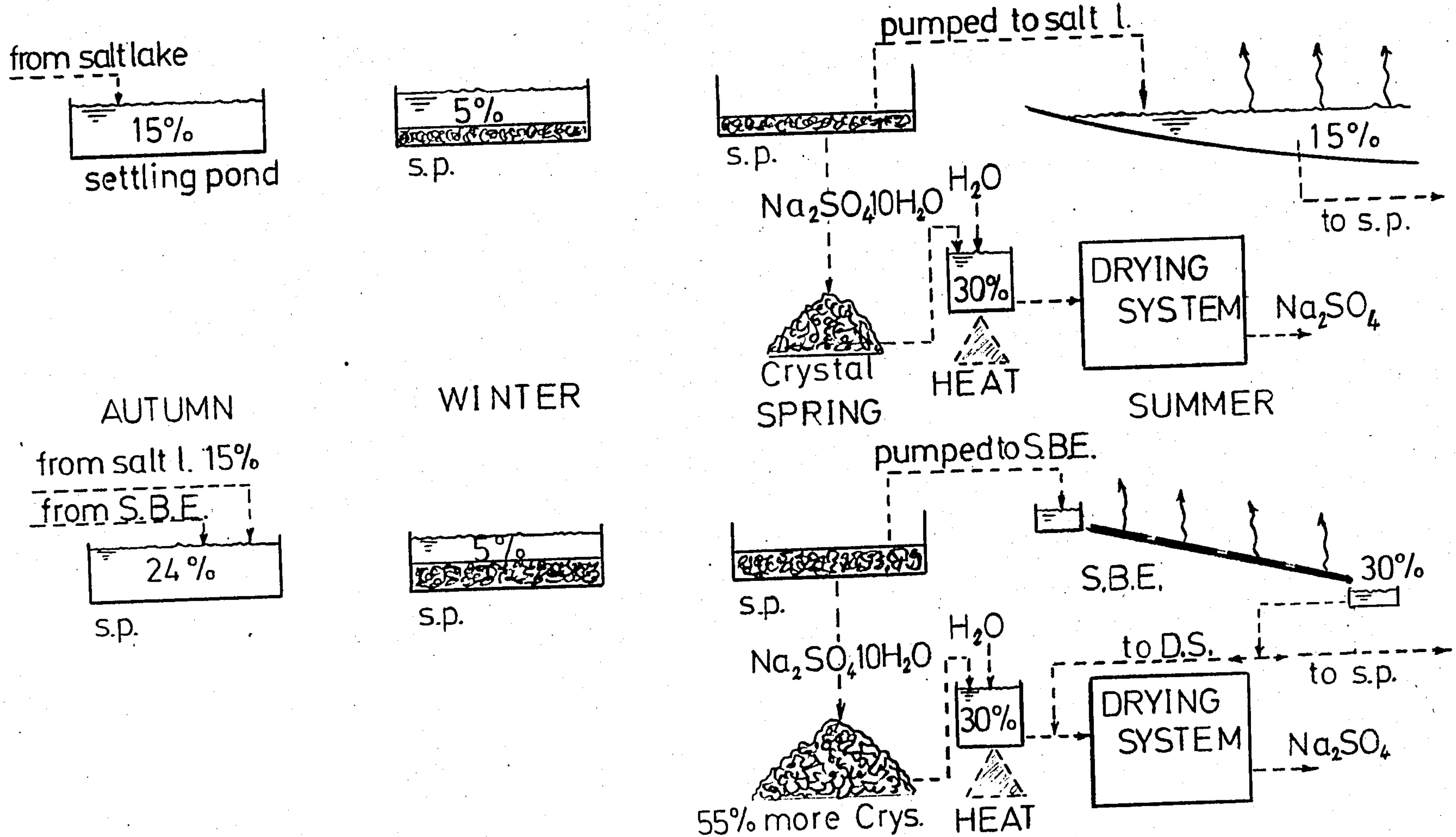

SIDE VIEW
from salt lake 5 °Bé
top container
suction jute
electrical fan
glass cover
distillate pipe
bottom container
wooden frame
12°
drying system
30 °Bé
1200mm
FRONT VIEW
distillate pipes
glass cover
electrical fan
metal sheet
(1mm)
olation
20mm)
ooden base
(20mm)
1140mm

FIG 1



USE OF SOLAR BRINE EVAPORATORS (S.B.E.) IN THE PRODUCTION OF SODIUM SULFATE (Na_2SO_4)

II: DESIGN CONSIDERATIONS

A) Goals of the design

In the design of " Solar Brine Evaporators " four basic objects were considered.

1.) Film-flow evaporation
2) Efficient evaporation, which means efficient utilization of the solar radiation present at the place of application
3) Low initial and operating cost
4) The desired concertration ratios and temperatures of the brine solution

The reason behind filmwise evaporation is the enlarged contact area and the increased evaporation rate. With this film flow system, the energy needed for water molecules impinging on the evaporating surface is much less than with any other system.

To increase the efficiency of the system losses should be reduced to a minimum and absorbtion of solar radiation should be increased to a maximum.

The maximum absorbtion of solar radiation can be obtained by blackened surfaces and by transparent glass covers. The top surface of the metal sheet can easily be painted to matt black or car -

on black, but to find a transparent cover isn't so easy, further -
ore the condensation taking place on the inside surface of the
glass cover reduces the transparency. (See FIG 2)

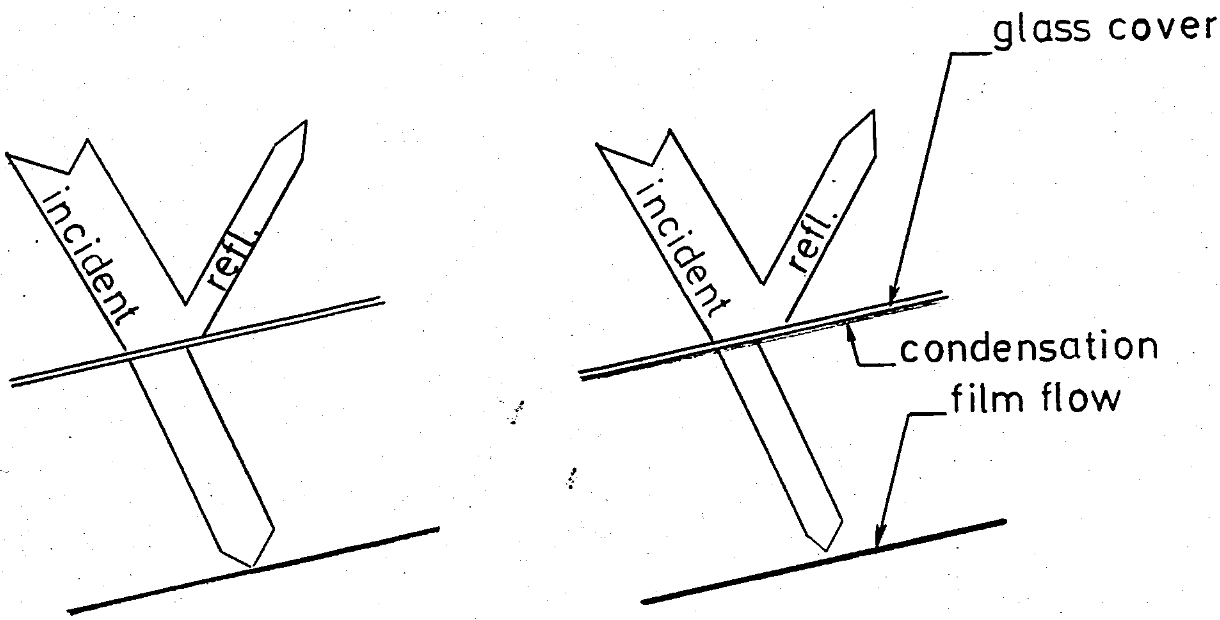


FIG 2

To prevent this undesired effect the existing gas, due to evaporation should be removed before condensing on the glass cover and this is done by suction using an electrical fan.

Like all other solar collectors conductive losses from the bottom of the absorbing metal sheet can be prevented using good insulating materials. In this work "styrophore" and a wooden frame has been used (See FIG 3) " The greenhouse effect " (1) of the glass cover decreases convective losses to a minimum

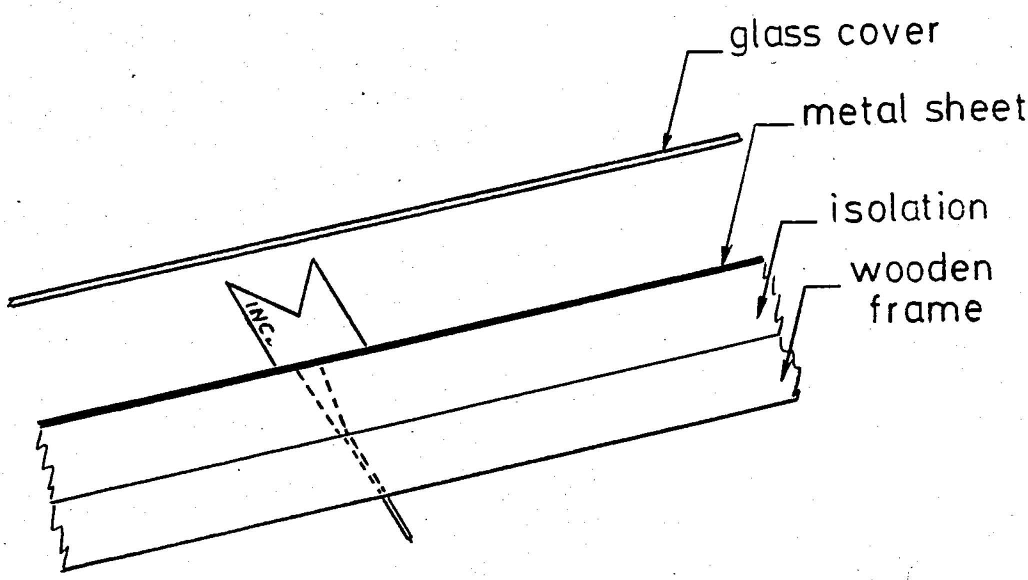


FIG 3

The fourth object, a desi ed concertration of the output solution, depends on the dimensions of the system.

The width adjusts the volume output and the flow length regulates the concertration of the brine solution. While changing the flow length the film thickness must always be within appropriate limits.

To increase output volume by keeping film flow all the time a special system must be constructed.

The "triangular" design is a simple solution. (FIG 4)

In this figure the output of the first evaporator is the input of the second one and the output volume is less than the input volume, because of the evaporation. Therefore the width of the second evaporator should be less than the first one to maintain film flow. As an example if 20% of the solution evaporates in the first evaporator, then the width of the next one should be 20% less than the irst one.

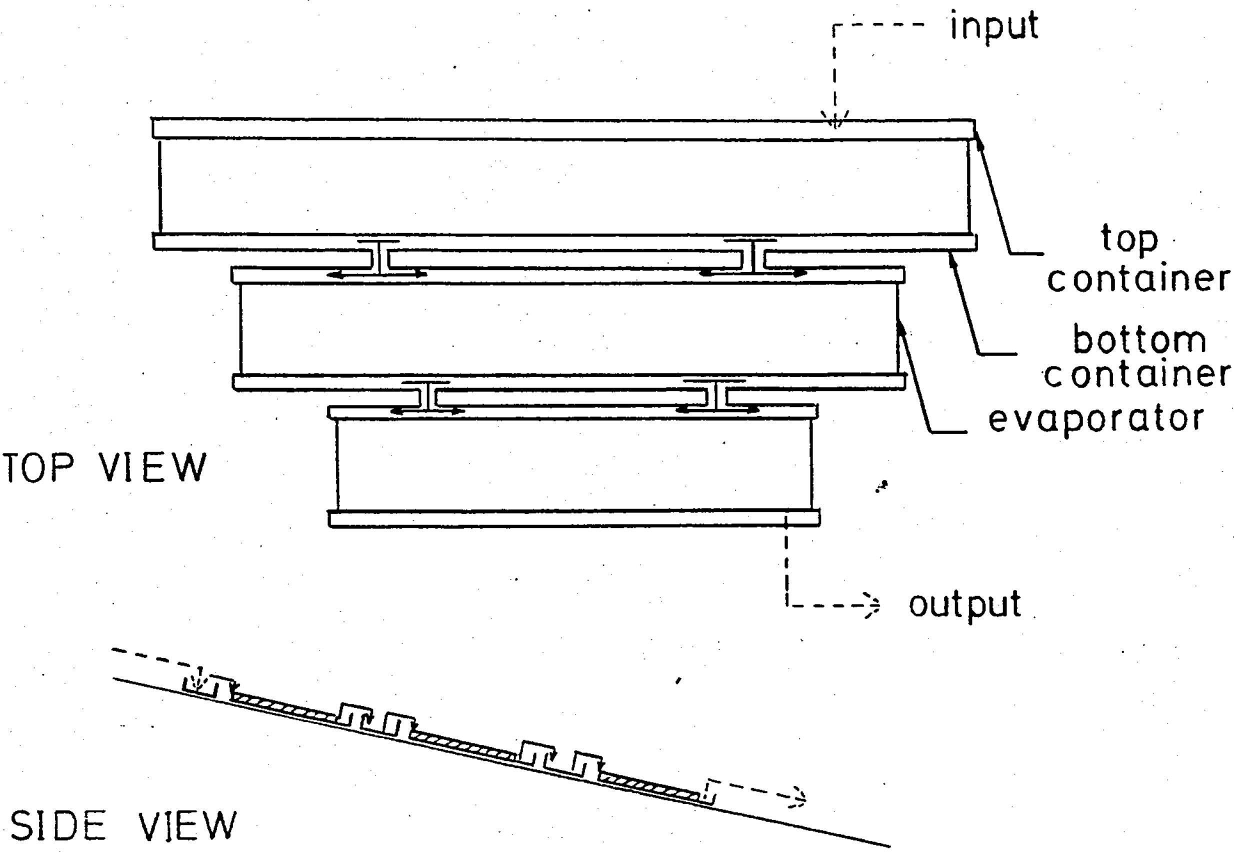


FIG 4

EVAPORATOR WIDTH CALCULATION :

$$W(n+1) = W(n) \times E(n)$$

Where $W(n)$: Width of the n^{th} evaporator

$$E(n) : \left[\frac{V_{OUT}}{V_{IN}}\right]_n$$: Evaporation ratio of the n^{th} evaporator.

The film thickness is also dependent on percent evapora-
on. The thickness reduces while evaporation continues (See FIG 5)

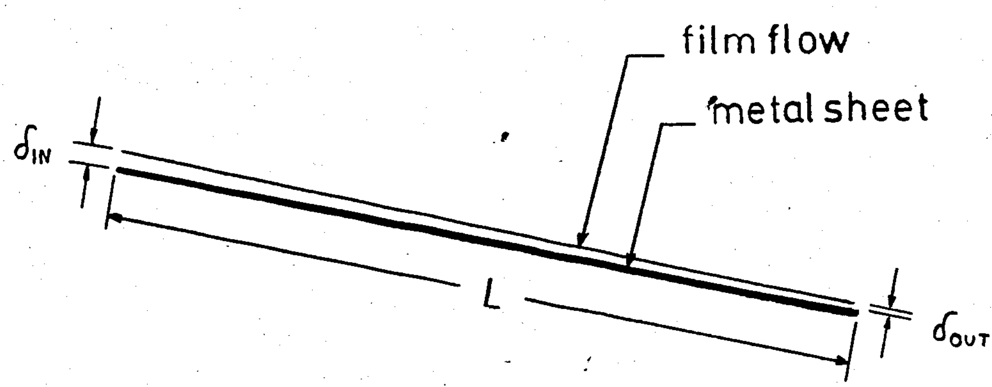


FIG 5

ILM THICKNESS CALCULATION :

$$\frac{\delta_{OUT}}{\delta_{IN}} \approx \frac{V_{L\,OUT}}{V_{IN}} = E_L(n)$$

where δ_{IN} , δ_{OUT} : Film thickness at inlet , outlet

V_{IN} : Input volume of the n^{th} evaporator

$V_{L_{OUT}}$: Output volume at a distance L from the inlet of the n^{th} evaporator.

The input brine concentration depends on the salt concentration in the lake and atmospheric conditions. It changes from 5% to 8% by weight at the lake " ACIGÖL ". On the other hand the de -

ired output concentration is between 25% - 28% salt by weight. t higher concentrations precipitations occurs, around 30% - 32%. herefore the output solution should be sent to the drying system efore any salt precipitation occurs.

The last and the most important object of the evaporators re their cost. Cost is the main problem of all solar collectors,heat ers and other systems, converting solar energy into heat energy. he low flux density of the solar radiation, requires the use of large size evaporators, to absorb enough energy ; and cost is mainly lependent on the size of those evaporators.

The choice for the correct evaporator dimensions then requires a study of the available solar radiation at the site and it's performance characteristics. This will then lead to a cost and feasibility analysis.

3) Availability of solar radiation

Every hour the earth receives 173×10^{12}kWh of energy from the sun. Over a year, this corresponds to 5180 Q (5), more then 20 000 times the current energy used by men. Not all of this energy reaches the surface of the earth. Some is reflected by clouds, by the land and by the sea. 1570 Q is approx. reflected and 1120 Q is used in evaporation of water from seas, lakes and rivers. The remainder 2490 Q is converted into heat. We can intercept some of this energy and can convert it to our own porposes. The potential solar energy supply available for use by man is in the neighborhood of 1100 Q , This is over 4500 times the current energy use of the entire human race.

The solar input power P is 1.07 kW/m^2. This condition implies a vertical orientation for the sun and a horizontal orientation for the collectors. The earth rotates with an axis angle of $23.5°$ and therefore the available solar power P is,

Q : The fundamental unit of energy.
$1\ Q = 10^{18}$ Btu $= 2.93 \times 10^{14}$ kWh it is the amount of energy required to bring Lake Michigan to a boil.

$$P = \alpha(\psi) \cos\psi$$

where $\alpha(\psi)$: 1.07 kW/m^2

ψ : normal angle.

In general α is a function of ψ. The greater the angle with the normal, the greater the length of atmosphere which must be traversed and the more light is scattered and absorbed. Thus the integrated value of α is always less than $\alpha(0)$. (See FIG 6 ; TABLE 1)

The total solar energy reaching the earth is made up of two parts 1) energy in the direct beam

2) diffuse energy from sky, clouds etc.

But as mentioned before clouds, dust, fog and other local climatic conditions change the solar radiation greatly from time to time. The safest way to predict average solar radiation is to refer to measurements made at the location.

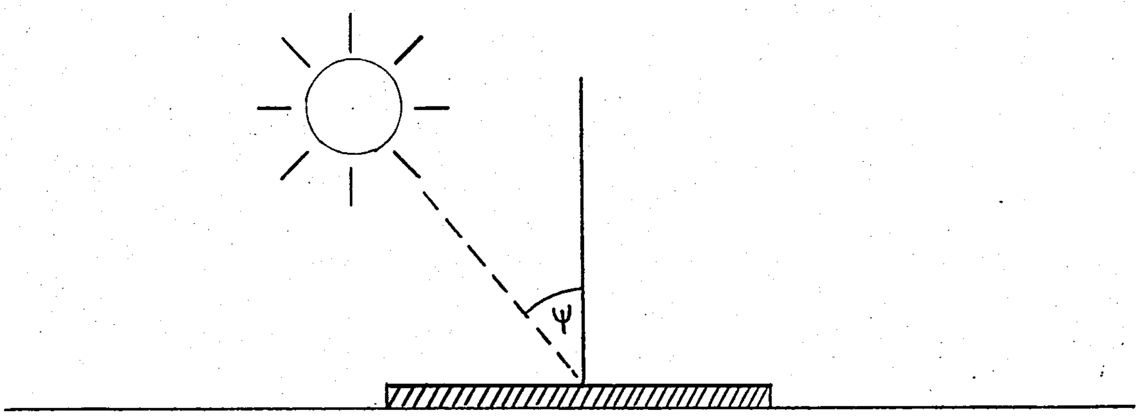


FIG 6

ψ	0	10	20	30	40	45	50	5
kW/$_{m^2}$ α	1.07	1.06	1.05	1.03	1.99	0.96	0.93	0.

TABLE 1

C) Solar radiation measurements

The most common instrument to measure the energy in incident solar radiation is the " PYRANOMETER " It is able to measure the total radiation within its hemispherical field of view. (See FIG 7)

It can be simply modified to measure only diffuse radiation by using on occulting disk which blocks beam radiation to the sensor surface. Pyronometers are calibrated saperately. The energy output in mV should be turned to W/m^2 by using the calibration factor of the pyronometer. If tilted or inverted, the free convection regine within the glass dome may change and errors may be introduced.

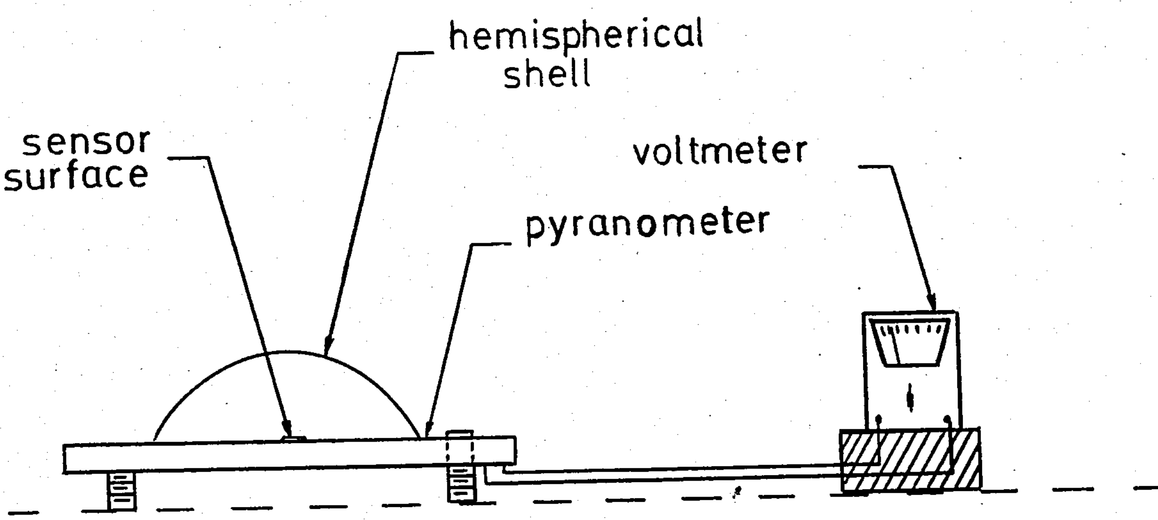
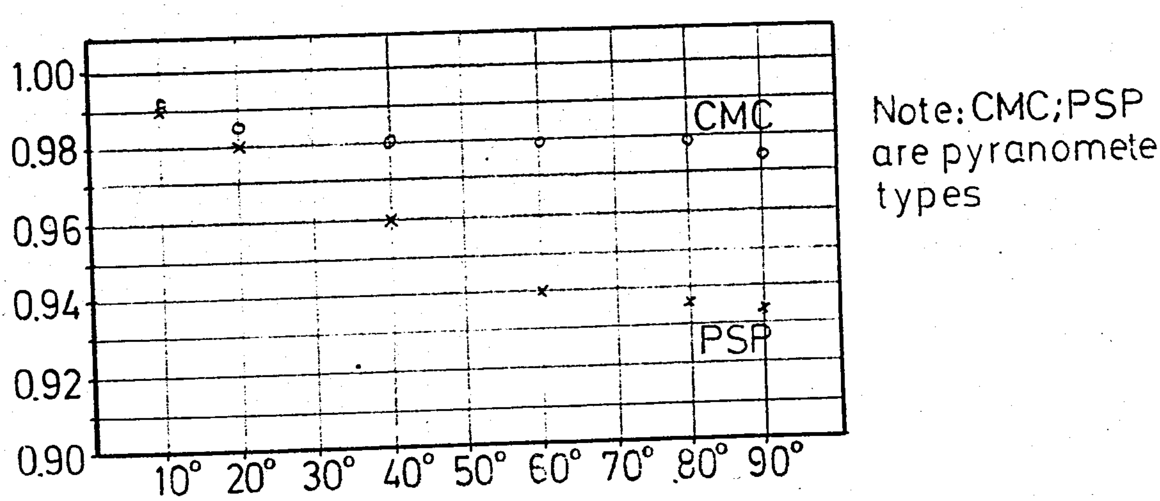


FIG 7



TILT EFFECT DIAGRAM (7)

GRAPH 1

From the tilt effect diagram (Solar Energy Vol 31/3)(7) it can be observed that for small tilt angles, upto 10°, the tilt ffect isn't much important. For larger tilt angles the effect causes important errors in some pyronometers.

) Radiation availability at the place of application

In our country the places of application of "SOLAR BRINE EVAPORATORS " will be in middle ANATOLIA where the salt lakes exist. TABLE 2 shows the solar radiation near DENİZLİ (near salt lake ACIGÖL)

But experiments have been made in Istanbul and between APril & September Istanbul has 10% lover mean irradiance than in middle ANATOLIA and also 28% higher cloudness ratio.(See TABLE 3)

DENİZLİ LATITUDE 37.78° ALTITUDE 428.0m (2)

MONTHS OF THE YEAR	AVARAGE MAX TEMP. °C	SUN SHINE (hours)	CLOUDINESS (1-10)	RELATIVE HUMIDITY (%)	TOTAL RADIATLON ESTIMATED (cal/cm^{2}day)
JAN	10.3	3.9	5.8	72.0	174.9
FEB	11.6	4.0	5.7	69.0	224.7
MAR	14.9	5.6	5.7	65.0	327.6
APR	20.1	7.6	5.2	61.0	450.3
MAY	25.0	8.6	4.5	60.0	529.1
JUN	30.5	10.8	2.7	49.0	620.7
JUL	34.2	11.9	1.1	42.0	648.3
AUG	34.2	11.4	0.7	41.0	589.9
SEP	29.4	9.6	1.7	48.0	460.4
OCT	24.2	7.8	3.2	55.0	315.8
NOV	17.8	5.5	4.4	68.0	214.4
DEC	12.7	3.2	5.9	73.0	150.8

TABLE 2

FLORYA LATITUDE : 40.98° ALTITUDE 34.0m (2)

MONTHS OF THE YEAR	AVARAGE MAX. TEMP. °C	SUN SHINE (hours)	CLOUDINESS (1-10)	RELATIVE HUMIDITY (%)	TOTAL RADIATION ESTIMATED (cal/cm^2day)
JAN	8.0	2.8	7.3	80.0	134.8
FEB	8.4	3.6	7.1	79.0	192.0
MAR	10.4	4.6	6.6	77.0	277.9
APR	15.4	6.4	5.7	76.0	390.3
MAY	20.5	8.7	4.9	77.0	501.9
JUN	25.5	10.8	3.6	72.0	584.7
JUL	28.7	11.8	2.4	68.0	604.1
AUG	28.8	11.1	2.3	69.0	537.3
SEP	25.0	8.3	3.5	73.0	389.5
OCT	19.8	6.2	5.0	77.0	262.0
NOV	15.2	4.0	6.5	79.0	163.8
DEC	10.8	2.8	7.2	80.0	122.5

TABLE 3

III: EXPERIMENTAL SETUP

A) Evaporators

1) Early studies

The first experiments have been started in the summer of 1982 with narrow but long evaporators. (See FIG 8). Black painted metal sheets of 15cm width and 4m length were well isolated from the bottom with 2cm thick styrophore and covered with a glass of 3mm thickness.

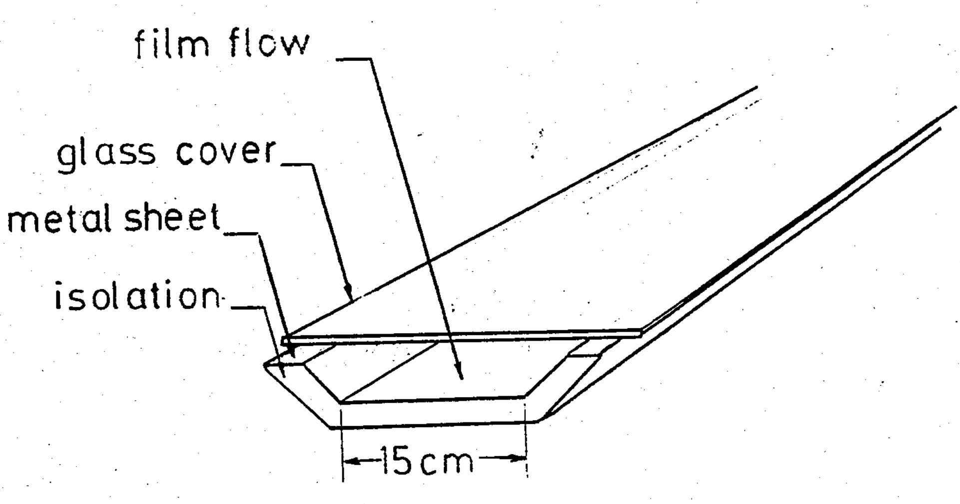


FIG . 8

To obtain film flow a piece of jute is placed along the metal sheet and the capillary suction of the jute from the top con-

CONDENSATION PROBLEM IN LONG CHANNEL TYPE EVAPORATORS

iner ensures a continious feed of brine solution to the system. e results were partially succesfull.

With this evaporator design the brine concentration in - ased from 5% to 24% in 4m and an evaporation rate of 0,5 /hr m^{2}was obtained (between June-August)

But the main difficulty was the precipitation of salt close to the output end of the plates due to the decreasing feed rates. See FIG 9)

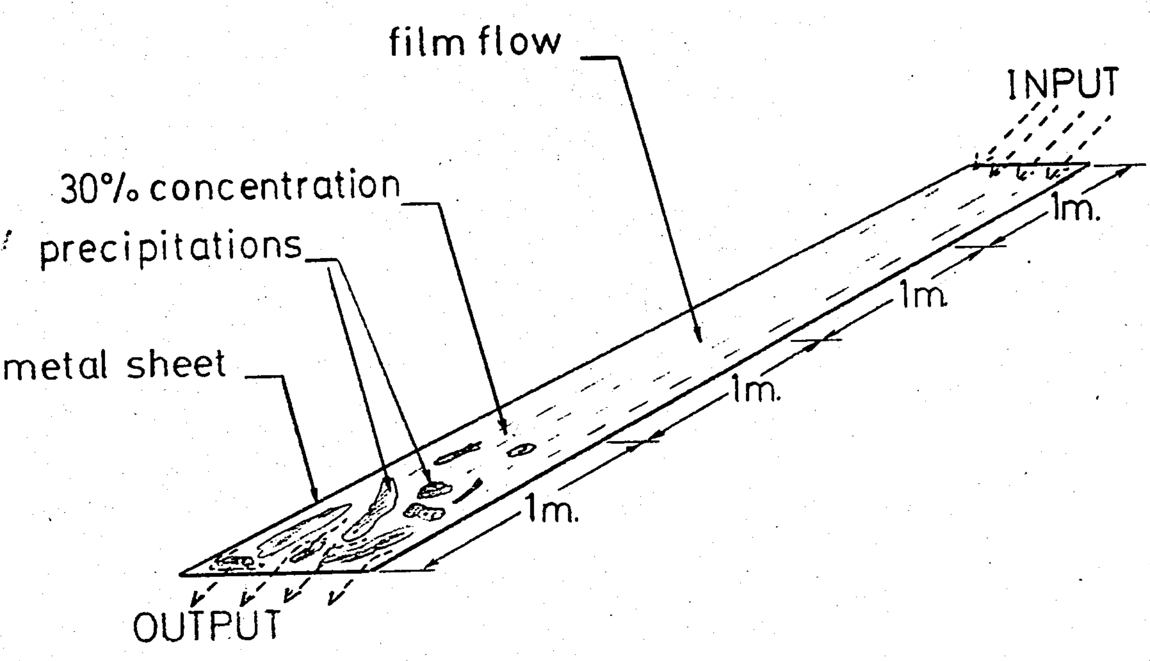


FIG 9

The precipitation caused by concentrations reaching the critical value (30%-32%) forced the brine solution to flow irreg- ulary. (See FIG 9)

To overcome this precipitation problem input feedrates ere increased, but this reduced the film flow effect in the first ne and two meters of the evaporator length. In this way precipita- ion was avoided at the lower end of the evaporators but total evaporation rate decreased due to the disturbed film flow at the inlet nd.

A second difficulty was caused by the dripping condensate ollected at the inside of the glass covers.

Back dripping was prevented with distillate collecting pipes, but condensation was still a problem decreasing the transparency of the covers. (See FIG 10)

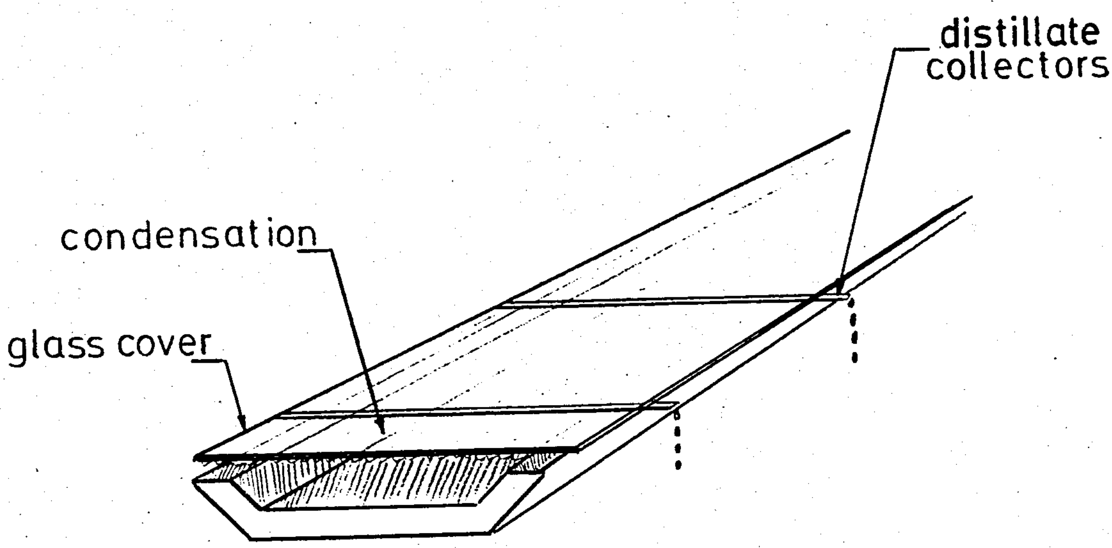
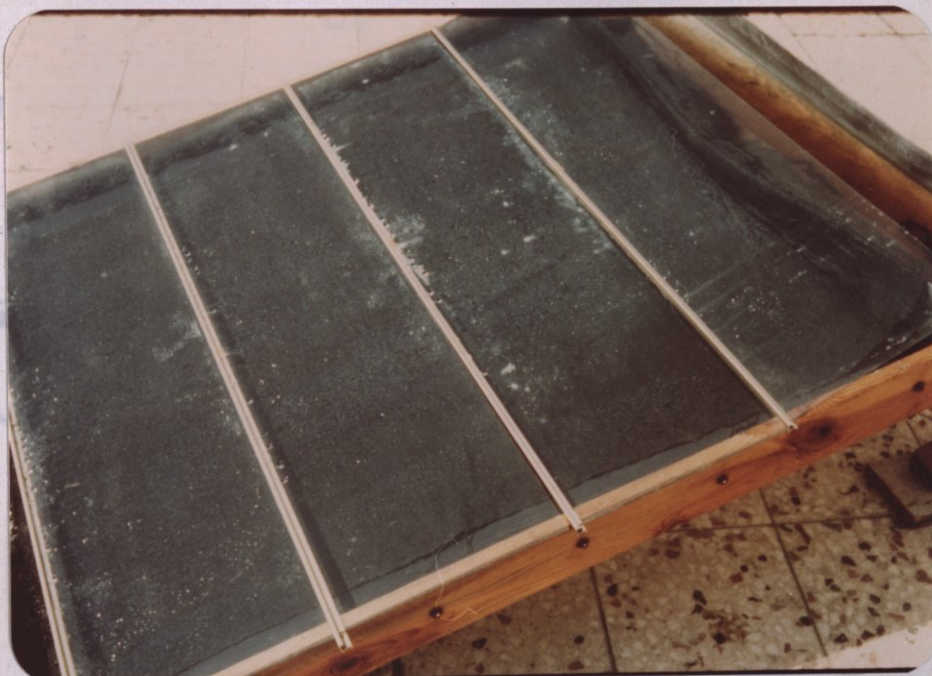


FIG 10

The third problem was the width of the system. Although the system was well isolated the temperature, T_c , at the center of the evaporator was much higher than the temperature of the two sides, T_s . Nearly 60% of the total flow area was affected directly by the conduction and infiltration losses. (See FIG 11)



FLOW ON JUTE

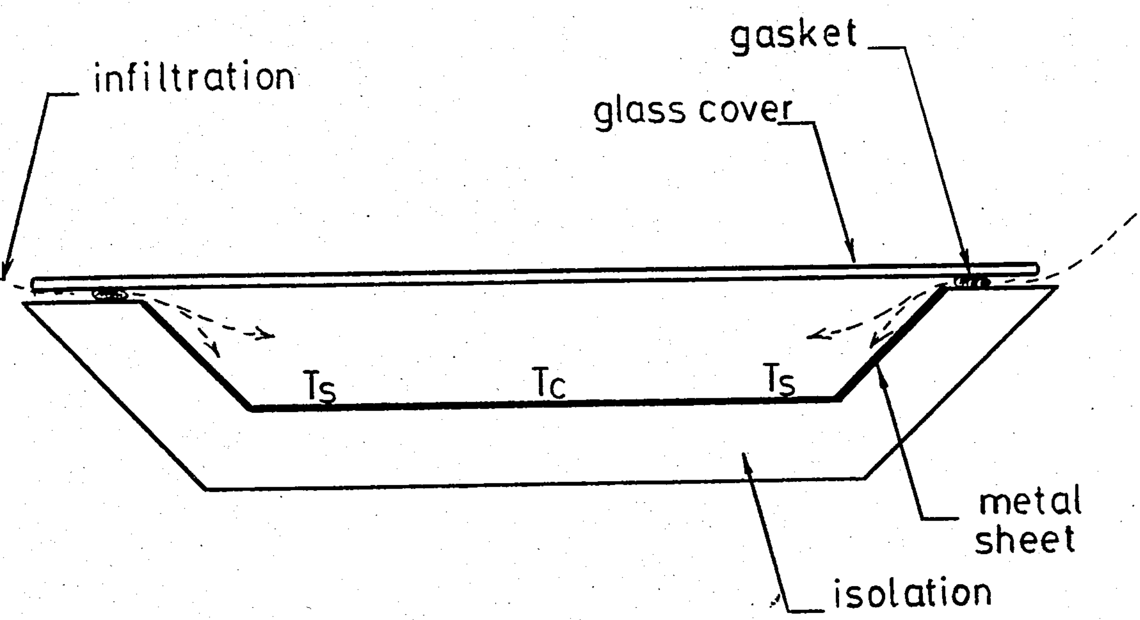


FIG 11

2)Modified evaporators

With those experiences at hand a new evaporator was de - signed and experiments were carried from August 1983 to October 1983.

Evaporator length was decreased from four meters to one meter to obtain film flow through out the whole length.

The other improvement was made in the evaporating surface. In the old evaporators pieces of jute were used to maintain the film flow and area enlargements. But jute had many parameters which influenced film flow and these parameters had to be studied very well.(Capillary suction ; thickness ; paint etc.)

CAPILLARY SUCTION FROM TOP CONTAINER

Instead the bottom of the evaporators were replaced by a ough and wavey surface painted in black, thus improving evaporator erformance and reducing its cost.

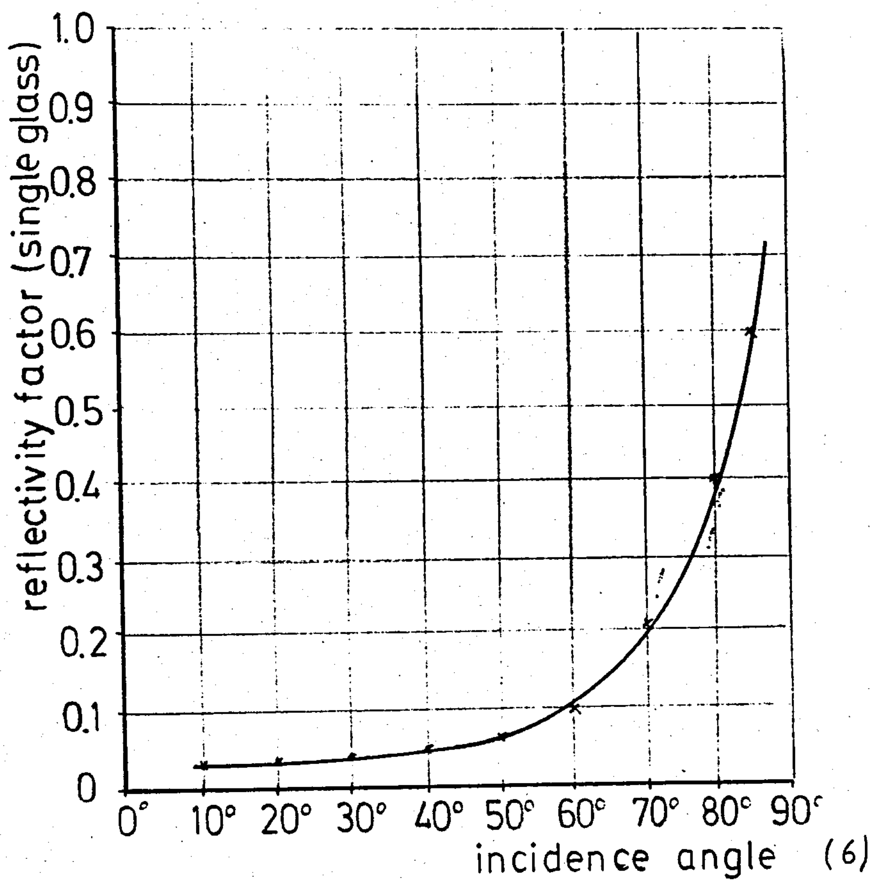


FIG 12

Since the film thickness δ is propartional to percent vaporation " e " with the one meter evaporator length the hange in " δ " was minimal. For such a case the " Triangular de - ign " should be used.

The width of the evaporators were increased from 15cm to .00cm so the effect of side losses and infiltrations were decreased rom 60% to 10%.

Condensation at the bottom sides of glass covers was totally eliminated by suction of the brine vapor with the help of on electrical fan (FIG 1). This way the transparency of the cover was not affected and the " greenhouse effect " still continued, since only the vapor inside was exousted and no outside air waspermitted to enter the evaporator. As a result the inside humidity, ϕ_{in} is reduced, componed with the no suction system and evaporation accelerated.

Tilt angle of 10°-12° was enough to maintain film flow and the reflection losses from the glass cover was not much (See FIG 12).

$$\text{percentevaporation} \quad e = \frac{V_{IN} - V_{OUT}}{V_{IN}} \times 100 \approx \frac{\delta_{IN} - \delta_{OUT}}{\delta_{IN}} \times 100$$

B) Measurement setup

The two " Solar Brine Evaporators " prepared as mentioned in the previous section, were placed on the roof of the engineering building. They were placed so that each faced south and the absorbing surfaces made an angle of 12° with the horizontal and none of them shaded one other at any time of the day.

Following measurements were made :

Solar radiation : The pyronometer connected with a Voltmeter measures the hemispherical global radiation (Outputs,mV,were taken every 15 minutes)

Temperature : The flowing brine temperature and inside air temperatures (inside the system) were measured every 15 minutes with thermometers.

Concentration: Concentrations of the input and output brine solutions were calculated by measuring their densities using bomometers. (everyhour)

H midity : The inside and outside relative humidities were measured with a hygrometer(every 15 minutes)

Solution volume : To calculate evaporation rates volumes of the input and output solutions were measured (every hour)

Also the outside wind directions and velocities were measured using simple methods. They didn't directly enter the calculations but give an idea for the convective heat transfer coefficient, ho.

IV : Experiments

This work is mainly application oriented therefore, small scale experiments and their accurasies are important since results will later be applied in the design of large scale equipment.

Construction of experimental evaporators

The construction is shown below shematically

Obtain two metal sheets (Size (1.0 1.0 0.001m) × 2)

↓

Construct two wooden bases (size(1.2 1.0 0.02m)×2)

Construct two wooden side frames(size(1.0 0.15 0.02m)×4)

↓

Obtain styrophore (size $2m^2$; 2cm thickness)

↓

Construct two containers for each system

Sizes Top (1.0 0.2 0.1m) × 2

Buttom (1.0 0.1 0.1m) × 2

↓

Put styrophore on wooden base and metal sheet on styrophore. Fasten the two sides of the system with wooden frames. Put the two containers to the top and bottom sides. Do a rough surface by scattering small metal pieces on to the metalsheet and paint it with matt black, Put a jute piece from top container to metal sheet for suction.

↓

Obtain glass covers, size (1.02 1.0m) × 2 3mm thickness

↓

Block the input and output openings of the evaporator with styrophore

↓

Connect an electrical fan to one side of the wooden frame

↓

Tilt the whole system by 12^{o} (south facing)

↓

Fill the top container with salt solūtion. Place the glass cover on the system. The set up is ready. Now the brine can be allowed to enter the evaporator with the fan operating.

V: Evaluation of data

A) Volume calculations

To find the input volume V_{IN} specific gravities of input and output solutions and also the volume of the output solution should be measured.

From the mass balance equation

Mass of input solution = Mass of output solution + mass of the evaporated water

$$V_{IN} \times \rho_{IN} = V_{OUT} \times \rho_{OUT} + (V_{IN} - V_{OUT})\rho_{H_2O}$$

$$V_{IN} = \frac{V_{OUT}(\rho_{OUT} - 1)}{\rho_{IN} - 1}$$

To calculate the input and output densities the °Bè conversion equations should be used.

$$\text{Specific gravity} = \frac{145}{145 - {}^{\circ}\text{Bè}} \quad \text{for Sp.gr.} > 1$$

$$\text{Specific gravity} = \frac{130}{130 - {}^{\circ}\text{Bè}} \quad \text{for Sp.gr.} < 1$$

The table below gives also the conversion from Be scale to specific gravity. (3)

Bé	Sp.gr.	Bé	Sp.gr.	Bé	Sp.gr.
0	0.9991	10	1.074	20	1.160
1	1.006	11	1.082	21	1.169
2	1.013	12	1.090	22	1.179
3	1,020	13	1.098	23	1.189
4	1.028	14	1.106	24	1.199
5	1.035	15	1.115	25	1.209
6	1.042	16	1.124	26	1.219
8	1.058	18	1.142	28	1.240
9	1.066	19	1.151	29	1.250

TABLE 4

Salt concentration

The table below gives the quantities of sodiumsulfate (Na_2SO_4) and Glaubert salt ($Na_2SO_4 10 H_2O$) in 100 ccm solution for different density values.

°Bé	$Na_2SO_4 10H_2O$	Na_2SO_4
5	10.99	4.28
10	23.02	8.92
15	37.20	14.42
20	46.66	18.20
24	58.68	22.83
30	78.45	30.61

TABLE 5

B) Calculation of incident energy

During the experimentation period solar radiation was measured by using a pyronometer. The ,mV , outputs of the pyronometer were converted to cal/cm^2 min using the conversion factor of the pyronometer

1mV ___ 0.137 cal/cm^2 min

The incident energy diagrams of each experiment day can be seen in the Appendix

C) Calculation of evaporation rates

The difference between the water content of the input solution and the water content of the output solution is the amount of evaporation. The quantity of salt remain unchanged throughout the whole process.

To calculate the evaporation rates densities and volumes of the input and output solutions should be measured.

Evaporation volume = Input volume - Output volume

$$V_E = V_{IN} - V_{OUT}$$

$$V_{IN} = \frac{\mathring{B}\grave{e}_{OUT} \times V_{OUT}}{\mathring{B}\grave{e}_{IN}}$$

$$V_E = \left\{ \frac{\mathring{B}\grave{e}_{OUT}}{\mathring{B}\grave{e}_{IN}} - 1 \right\} V_{OUT}$$

Evaporation rate $\dot{e} = V_E / hour$

D) Calculation of evaporation energy

Evaporation energy = mass of evaporated water X heat of evaporization

$$Q_E = m_{H_2O} \times \Delta h\Big|_{T_E}$$

$$m_{H_2O} = \rho_{H_2O}\Big|_{T_E} \times V_E$$

$$\Delta h\Big|_{T_E} = h_{g_{SAT}}\Big|_{T_E} - h_{f_{SAT}}\Big|_{T_E}$$

E) Calculation of efficiency

From the definition of efficiency , η

$$\eta = \frac{\text{Energy needed for the amount of } H_2O \text{ evaporated}}{\text{Incident energy absorbed by the evaporators}}$$

The numerator can be calculated using equations in part d, and the denominator can be calculated as mentioned in part b).

F) Calculation of absorbed energy

During the evaporation process some of the penetrating energy is stored as heat in the flowing brine solution. The heat storage in solution can be found by the following equation.

$$Q_{ABS} = m_{SOL} \times C_{SOL} \times (T_m - T_0)$$

$$C_{SOL} = 0.95 \ kcal/kg\,^{\circ}C \quad \text{at } 7\,^{\circ}B\acute{e}$$

(G) Evaluation of losses

Before doing energy balance calculation and going into the evaluation of results , it should be considered again what happened to that portion of solar energy which could not be utilized. In other words losses should be evaluated in more detail. The losses can be divided into three parts.

Heat transfer losses

Reflection losses

In filtration losses

Heat transfer losses

There are three types of heat transfer losses

1) Conductive losses
2) Convective losses
3) Radiative losses

Conductive and convective losses

Before going into detailed analyzes some assumptions should be made (See FIG 13)

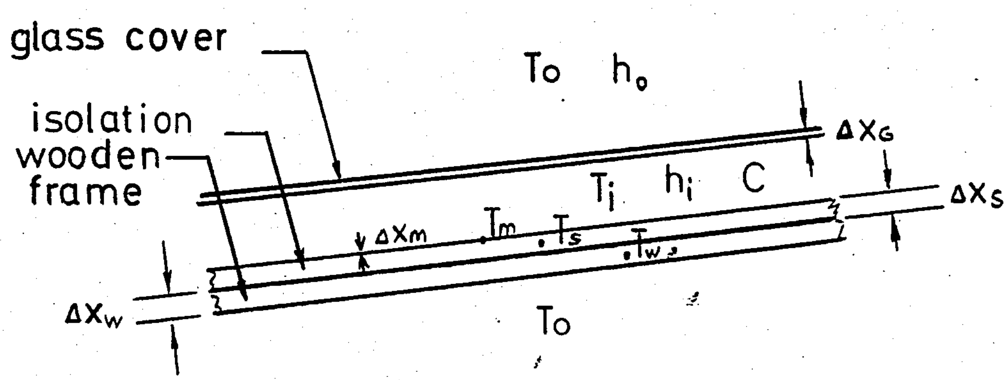


FIG 13

Assumptions :

1) The bottom temperature, T_{Wb} of the wooden frame is the same as the ambient temperature, To

2) The values for convective heat transfer coefficient, h_o and air space conductivity, C are taken from "Threlkeld".

3) Homogeneous air temperature distribution inside the system.

The outside heat transfer coefficient depends strongly on the wind speed. " Threlkeld " gives this dependance in graphical form

FIG 14). An average value of 5.0 Btu / hr ft^2 °F is considered in calculations. (6)

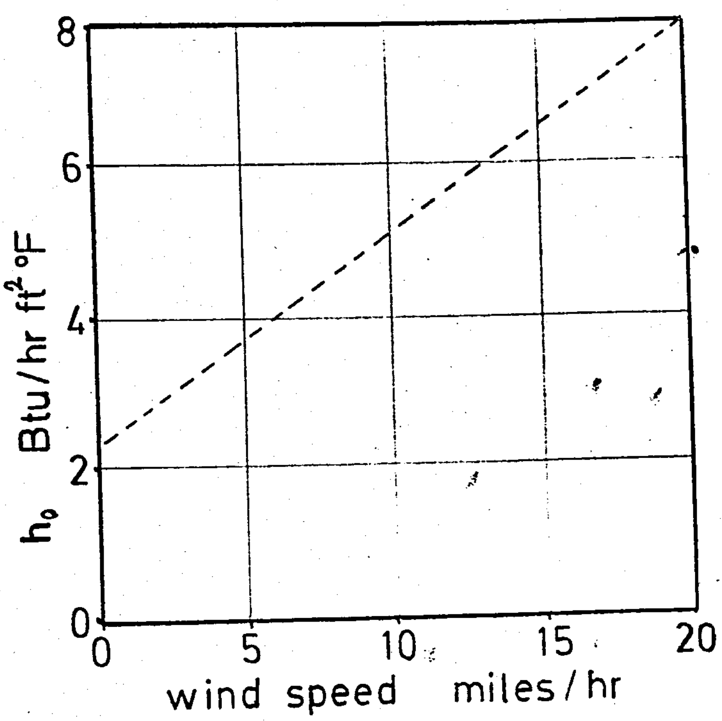


FIG 14

The conductivity of air spaces depends on the width and mean temperatures of air. This dependence is given in graphical form (See FIG 15) by Brown and Marco.(4). Here C, is taken as 1.4 because the width of air space exceeds 0.8 inches (it is ≈ 1.5 inches) (4)

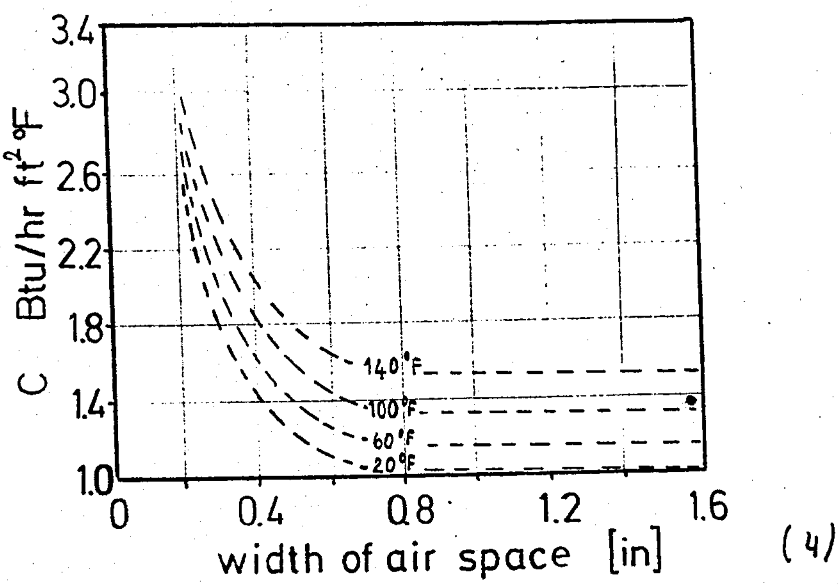

3.4
3.0
2.6
2.2
1.8
1.4
1.0
C Btu/hr ft² °F
140 °F
100 °F
60 °F
20 °F
0
0.4
0.8
1.2
1.6
width of air space [in]
(4)

FIG 15

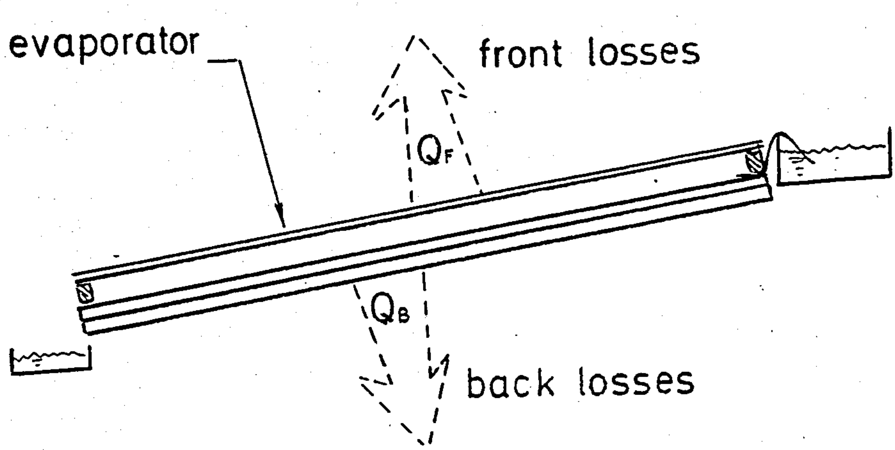

evaporator
front losses
Q_F
Q_B
back losses

FIG 16

As seen from FIGURE 16 the heat transfer losses to the urrounding must be considered in two parts

$$Q_{TOTAL} = Q_{FRONT} + Q_{BACK}$$

RONT LOSSES

$$Q_{FRONT} = U_F A(T_i - T_a)$$

where $$\frac{1}{U_F} = \frac{1}{h_i} + \frac{1}{C} + \frac{\Delta X_G}{k_G} + \frac{1}{h_o}$$

$h_i = 2.8$ $W/m^2\,°C$ no fan

$h_i = 5.6$ " one fan

$h_i = 11.2$ " two fans

$\Delta X_G = 0.003m$; $k_G = 0.78$ W/mC ; $A = 1.2m^2$

BACK LOSSES

$$Q_{BACK} = U_B A(T_m - T_o)$$

where $$\frac{1}{U_B} = \frac{\Delta X_m}{k_m} + \frac{\Delta X_s}{k_s} + \frac{\Delta X_w}{k_w}$$

X_m	= 0.001 m	k_m	= 50.0 W/m C
X_s	= 0.020 m	k_s	= 0.038 "
X_w	= 0.020 m	k_w	= 0.150 "

Losses due to radiation are not important for systems working with low temperature differences. In this case the maximum temperature of the system was around $50^{o}C$ while the surounding temperature was 25^{o} C.

Reflection and infiltration losses

In-filtration is the leakage of outside air into the system through cracks and openings caused by a pressure difference across the boundary surface. But the estimation of the rate of in-filtration is often very cr de and approximate. For the experimental " Solar Brine Evaporators " the influence area of the infiltration losses are less than 10% of the total evaporator area(see FIG 11), therefore they have little influence on the total energy balance.

The reflected portion of the incident energy is calcu - lated from the table below.

Inside the system totally condensation	Outside the system
4.5 Ω	3.8 Ω
4.8 Ω	4.1 Ω
6.3 Ω	5.5 Ω

TABLE 6

In the energy balance calculation the infiltration and reflection losses are assumed such that ;

Reflection losses :

For open system ______ 5% of incident Energy

For closed systems:

No fan case ________ 15% of incident Energy

One fan case _______ 10% " " "

Two fan case _______ 8% " " "

Infiltration losses

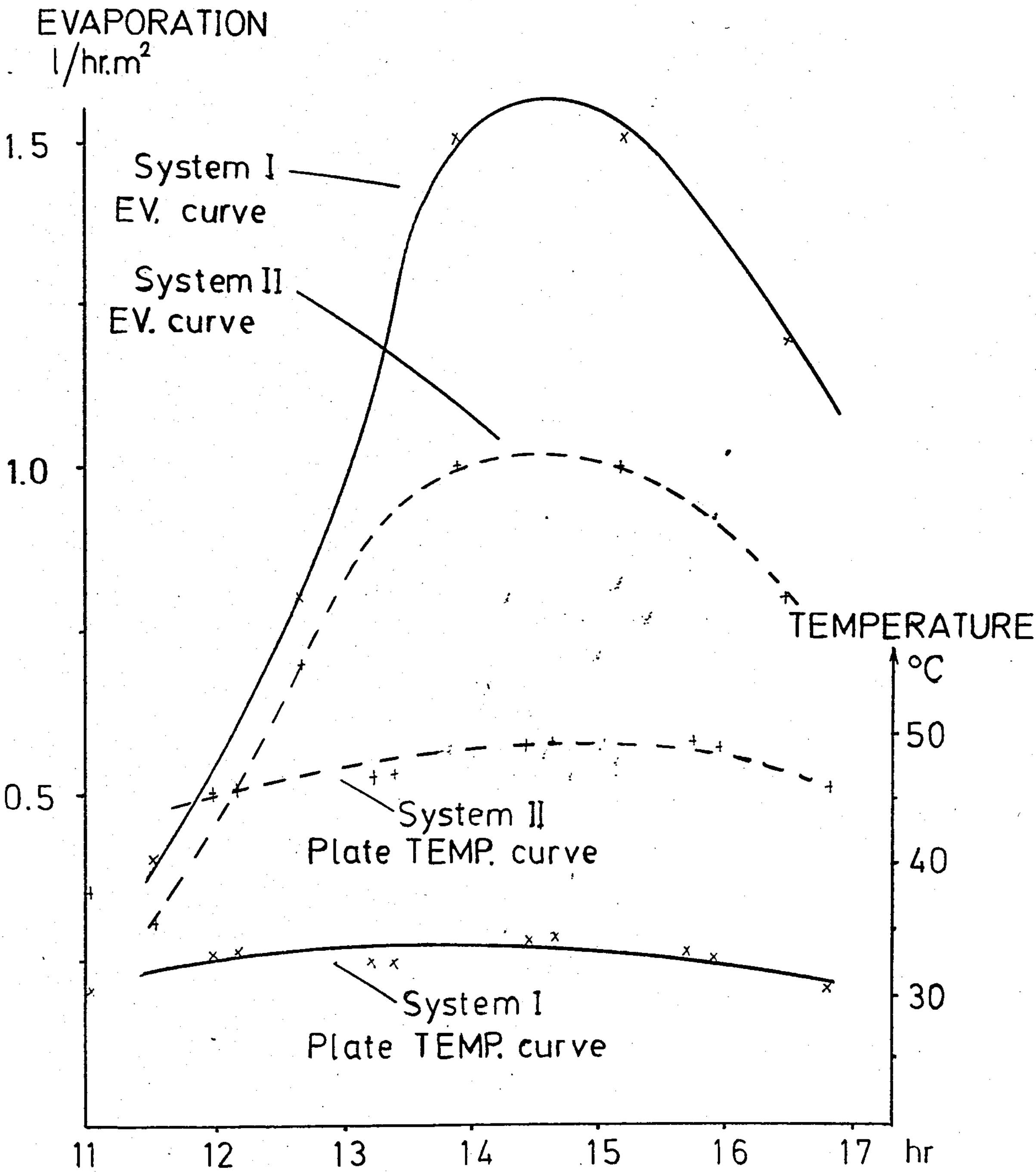
For all cases ___________ 5% of incident Energy

H) Energy balance calculation

Input Energy = Output Energy

Input Energy: Penetrating energy

Output Energy: Evaporation energy + Losses + Absorbed energy.



System I : Glass covered + 2 fan suction

System II : Glass covered (no fan suction)

TEMPERATURE-EVAPORATION DIAGRAM

/| :

TABLE IIa

xp #	TOTAL TIME	TOTAL INCIDENT ENERGY	AVG. PLATE TEMP.	AVG. VAPOR TEMP.
	hours	kcal	^oC	^oC
I	5	2440	40.5/25.9	35.5/-
II	4	2160	36.6/25.5	32/-
II	4	2718	41.8/25.3	34.1/-

TOTAL /APORATION	EVAPORATION ENERGY	TOTAL OUTPUT	OUTPUT Be	EFFICIENCY
liters	kcal	liters	^{o}Bè	%
2.5/2.6	1463/1514	14.9/17.2	5.9/5.7	59/62
2.4/2.3	1384/1339	15.6/12.5	5.9/6.2	64/61
3.8/3.6	2179/2097	13.3/12.5	6.4/6.3	80/77

xp #	Avg. AIR TEMP.	Avg Rel. Humidity	WIND VELOCITY	INPUT ^{o}Be
	^oC	%(AIR)	BEF./DIRECTION	^{o}Bè
I	22.9	53.9	2-3/NE	5
II	23.7	51.8	3/NE	5
II	23.3	45.8	1-2/NE	5

NOTE: (closed system/open system)

Based on DATA TABLES I a/b

" II a/b

" III a/b

TABLE IIb

ENERGY BALANCE TABLE

Exp. #	PENETRATING ENERGY TOTAL (INCIDENT - REFLECTION)	(INFILTRATION+HEAT tr.loss
	kcal	kcal
I	2074/2318	122+91/184
II	1836/2052	108+60/110
III	2310/2582	135+76/122

TOTAL ABSORBED ENERGY	REMAINED ENERGY FOR EVAPORATION	EFFICIENCY
Kcal	kcal	%
259/51	1602/2083	65/85
201/23	1467/1979	67/88
244/24	1855/2436	68/89

TABLE IIIa

Exp #	TOTAL TIME	TOTAL INCIDENT ENERGY	Avg. PLATE TEMP.	Avg. VAPOR TEMP.
	hours	kcal	^{o}C	^{o}C
IV	5	3655	36/47	29.9/42.0
V	4	2648	35/41	29.0/36.0
VI	5	3496	36.5/45.2	30.2/39.7

TOTAL EVAPORATION	EVAPORATION ENERGY	TOTAL OUTPUT	OUTPUT °Bè	EFFICIENCY
liters	kcal	liters	°Bè	%
6.1/3.9	3516/2226	16/17.9	7.0/6.0	96/60
4.0/3.2	2308/1836	12.7/12.6	5.2/5.0	87/69
6.1/4.7	3519/2686	10.5/12.8	7.1/6.8	101/76

Exp #	Avg. AIR TEMP. (°C)	Avg REL. HUMITY (%)	WIND VELOCITY	INPUT °Bè
IV	23.8	43.2	2/E	5
V	25.4	50.0	1-2/SE	4
VI	24.9	44.0	1/SE	5

NOTE: (covered + fansuction/covered)

Based on DATA TABLES IV a/b

" V a/b

" VI a/b

TABLE IIIb

ENERGY BALANCE TABLE

Exp #	TOTAL PENETRATING ENERGY (INCIDENT - REFLECTION)	(INFILTRATION+HEAT tr. LOSSES)
	kcal	kcal
IV	3289 / 3106	182+63 / 182-130
V	2383 / 2250	132+37 / 132-75
VI	3146 / 2971	174+55 / 174-105

TOTAL ABSORBED ENERGY	REMAINED ENERGY FOR EVAPORATION	EFFICIENCY
kcal	kcal	%
195 / 415	2849 / 2379	77/65
121 / 196	2093 / 1847	79/69
121 / 259	2796 / 2433	79/69

TABLE IVa

Exp #	TOTAL TIME	TOTAL INCIDENT ENERGY	Avg. PLATE TEMP.	Avg. VAPOR TEMP.
	hours	kcal	°C	°C
VII	5	3201	30/45	24/39
VIII	5	3681	32/47	27/40

TOTAL EVAPORATION	EVAPORATION ENERGY	TOTAL OUTPUT	OUTPUT °Bé	EFFICIENCY
liters	kcal	liters	°Bé	%
5.6/3.5	3253/2000	16.1/16.5	6.8/6.0	102/62
5.5/3.9	3184/2224	13.5/16.5	5.6/4.9	86/60

Exp #	Avg. AIR TEMP.	Avg. REL. HUMIDITY	WIND VELDUTY	INPUT °Bé
	°C	%	BEF/DIR	°Bé
VII	21.5	54	2/NE	5
VIII	25.4	47	2/NE	4

NOTE: (covered 2 FAN/covered)

Based on DATA TABLES VII a/b

" VIII a/b

TABLE IVb

ENERGY BALANCE TABLE

xp #	(INCIDENT - REFLETION)	(INFILTRATION + HEAT TR. losses)
	kcal	kcal
VII	2944 / 2720	160+24 / 160-124
VIII	3386 / 3128	184+17 / 184-103

TOTAL ABSORBED ENERGY	REMAINED ENERGY FOR EVAPORATION	EFFICIENCY
kcal	kcal	%
136 / 387	2615 / 2049	81/64
95 / 356	3086 / 2485	83/67

VII : EVALUATION OF RESULTS

A) General

During the experimentation period three types of " Solar Brine Evaporators " (i : open - ii : glass covered - iii glass covered with fan suction) are compared with each other and sam - ple results from each type are shown in tables (I_a , II_a , III_a , IV_a.)

The results of energy balance calculations are also given in tables (I_b , II_b , III_b , IV_b)

Experimental results were carried under similar weather conditions (cloudiness, humidity, wind etc.) and therefore it is possible to get a general conclussion about the types of " Solar Brine Evaporators " using the data obtained.

B) Calculations

The evaporation rates and energies were calculated as ex- plained in chapter IV part b & c .

Incident energy was obtained from pyronometer readings and efficiencies were also calculated as explained in part d of chapter IV.

The data were taken from the tables in the Appendix (State table)

The evaporation area in the experimental-setup was 1.0 m^2 but the collector area (evaporator total area) was 1.2 m^2. Therefore incident energy was taken as the energy per plate (Chapter V part a)

The total energy is the sum of the incident energies during the experimentation period.

The reflected portion is 8% of the incident energy for clear glass case ; only in two fan suction system See FIG 17

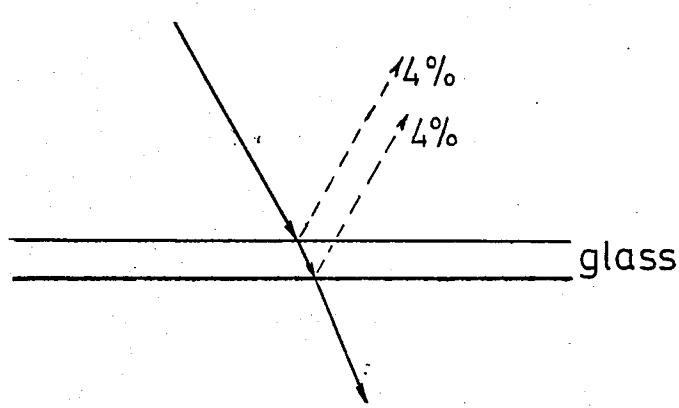


FIG 17

Depending on condensation at the bottom surface of the glass cover, the reflected portion increased to 10% with one suction fan and to 15% with no fans.

Infiltration losses were assumed to be 5% of the incident energy.

The constants of back heat transfer losses were the same for all types of evaporators.

The heat transfer losses were considered as follows

en case : No glass cover

U_F, h_o h_i ; C do not enter the calculations.

osed case i) Glass covered no fan suction

U_F calculated as in chapter V

$h_i = 1/10$ ho assumed

ii) Glass covered with one suction fan

U_F calculated as in chapter V

$h_i = 1/5$ ho assumed

iii) Glass covered with two suction fans

U_F calculated as in chapter V

$h_i = 2/5$ ho assumed.

emperature difference = $T_{AVG} - T_{AIR}$

VIII : COST ANALYSIS

The cost of the materials used in the construction of the experimental - evaporators are given below with December 1983 values

Materials	Size and Costcalculation		Total cost (TL)
Metal plate	1.0m 1.0m 0.001m		
	Length×Width×Thickness×8×130		
	(m) (m) (mm) (TL)		
	1 × 1 × 1 × 8 × 130		1040
Glass cover	1.0m 1.2m 0.003m		2000
Paint	matt black grey		
	100 TL/kg matal		800
Wooden frame	1.0m 1.2m 0.02m	bottom	2000
	2×(1.0m 0.15m 0.02m)	side	500
	Till frames		
Containers	1.0m 0.2m 0.1m	top	1000
	1.0m 0.1m 0.1m	buttom	750

Above materials are for one evaporator only.

Materials	Size and costcalculation				Total cost TL
Screws,nails etc.					200
Isolation	1.0m	1.2m	0.02m	Styrophore	240
Labour					600
					9630
				Others.....	370
				Total cost....	10.000 TL.

Cost of one experimental evaporator with $1m^2$ surface is about 10.000 TL.

Long Term Cost Analysis

The daily average evaporation rates of the experimental " Solar Brine Evaporators " are 8 - 10 lt/m^2 in September (in clear days). During May, June and July this value will be about 10 - 12 liter perday because of the 30% - 40% increase of the incident solar energy during that period.

And 10 liter/day is a good estimation for the total average between May and September.

One kg of fuel-oil can give 10.500 kcal energy when burned with 100% efficiency. Using steam boilers and heat exchangers the energy of fuel -oil can be transformed to the evaporation system with a maximum of 70% efficiency. Therefore only 7350 kcal/kg can be used in the evaporation.

With these calculations one kg of fuel - oil can evaporate 12 liters of water (for one liter 580kcal)

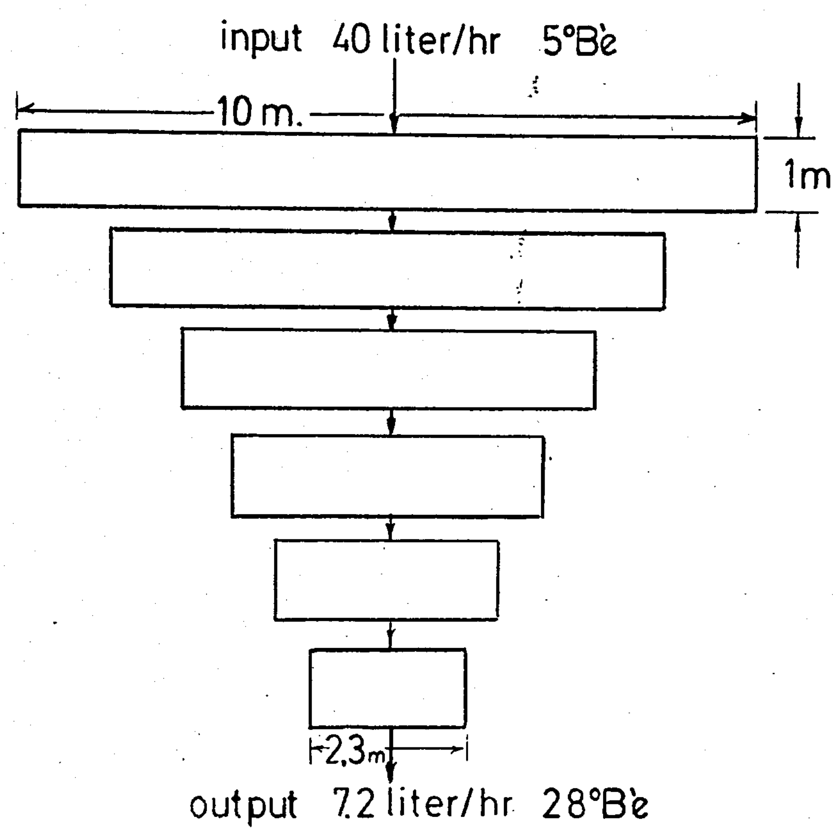
Using a" Solar Brine Evaporator " with $1m^2$ surface only 0.83 liter fuel-oil can be saved in one day. The cost of fuel-oil 60TL/liter.

As a result 1m evaporator can save 50 TL perday, and in 200 working days the "Solar Brine Evaporator" of $1m^2$ surface can pay its cost back.(total cost 10.000 TL)

Application Availability

To increase the density of one liter solution from 5 °Bè to 28 °Bè 0.82 liter of water should be evaporated from that solution.

Based on that calculation the six-stage triangular design below can evaporate 32.8 liters of water is one hour, when the input feed is 40 liter/hour.



IX: RESULTS AND DISCUSSION

A) Comparision of "film flow" on "rough" vs "jute" surface

The "rough" surface is obtained by scattering small metal pieces on the metal sheet and then dying with matt-black paint. Brine flows over this rough surface spreading out as film and evaporates.

A "jute" surface is made by placing a jute piece on the metal sheet and dying it with black paint.

In both cases comparisons are made with no glass cover at the top of the evaporators. (open case)

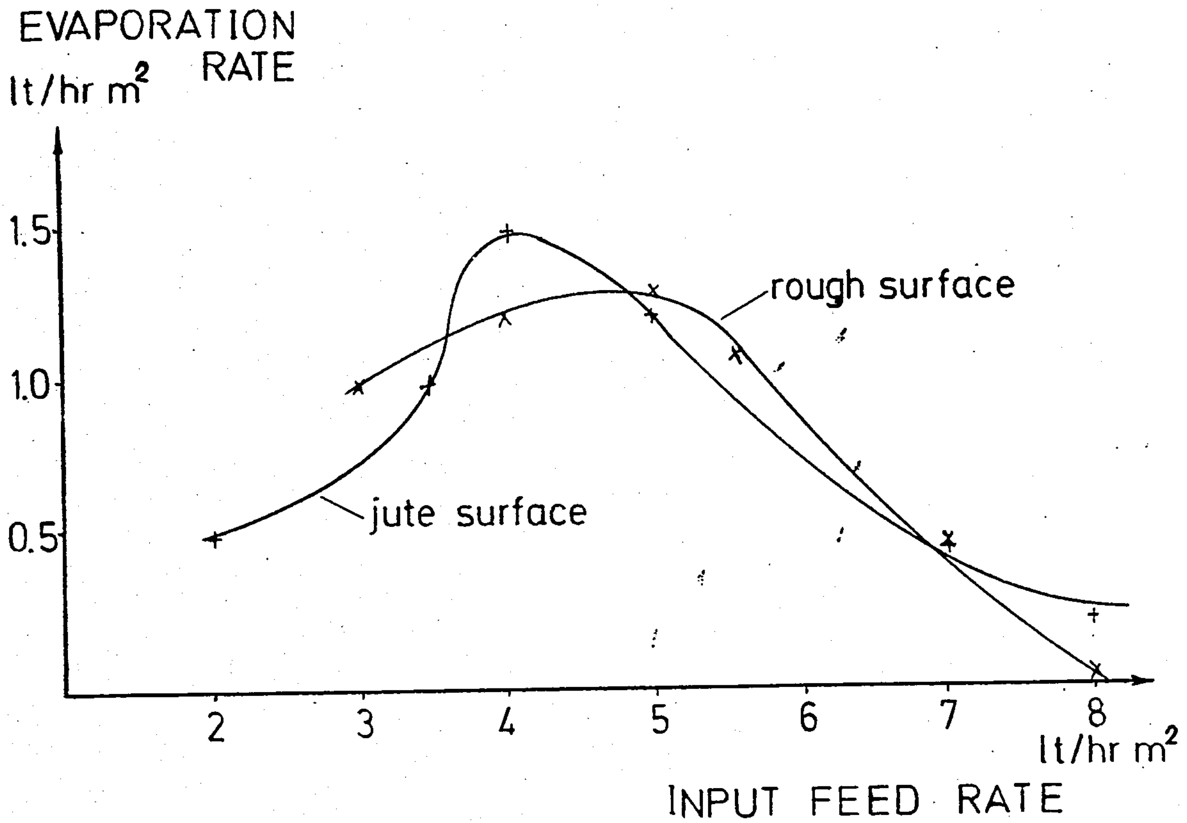
Time	Air Temp.	Total Incident Energy	Plate Temp Jute / Rough	Total Evaporation Jute / Rough
5 hours	22 °C	3230 kcal	23 °C / 24 °C	2.7 liter / 2.7 liter

(Data table IXa)

TABLE Ia

As it can be observed from table I_a, "jute" and "rough" surfaces behave similarly for similar input feed rates 4-5 liter/hr m^2

Graph I_b explains the effect of input feed rate on the vaporation rate.



(Data table IX b)

GRAPH Ib

From graph I_b it can be observed that if the input feed rate decreases below 3.5 lt/hr m^2 then the evaporation rate from "jute" reduces sharply. This can be explained with the help of bounded and unbounded moisture concept. If the moisture percent in "jute" reduces below "bounded moisture percent" then capillary effects inside "jute" decreases evaporation rate, sharply. If the moisture is above the bounded moisture percent then this unbounded water can evaporate like a "water film".

But with rough surface systems evaporation rate is nearly equal up to a certain input feed rate (upto 5 lt/hr m^2). Above this value evaporation rate decreases in both systems similary, this is because of the disturbed "film flow " ffect.

B) Comparison of "open" and "closed" systems

In the "open" system brine flow is directly exposed to the atmosphere and there is no glass cover at the top of the evaporators.

But in "closed" systems direct convection to atmosphere is prevented with a transparent glass cover ; so a "greenhouse" affect is obtained inside this enclosure and the distillate outlet pipes prevent back dripping of the condensed water.

In both cases brine flows over "rough" surfaces at 4-5 lt/hr m^2 input feed rates.

The three days averaged efficiencies (See table II a in Appendix) of both systems are similar. 67% for "closed" and 66% for open system. In the "open" system the evaporation is directly dependent on convection and air-humidity. An increase in convection, increases the mass of the circulated dry air, which increases evaporation.

In "covered" system the plate temperature is $12^{o}C$- $15^{o}C$ more than the "open" case, so less energy is needed for evaporation in "closed" systems.

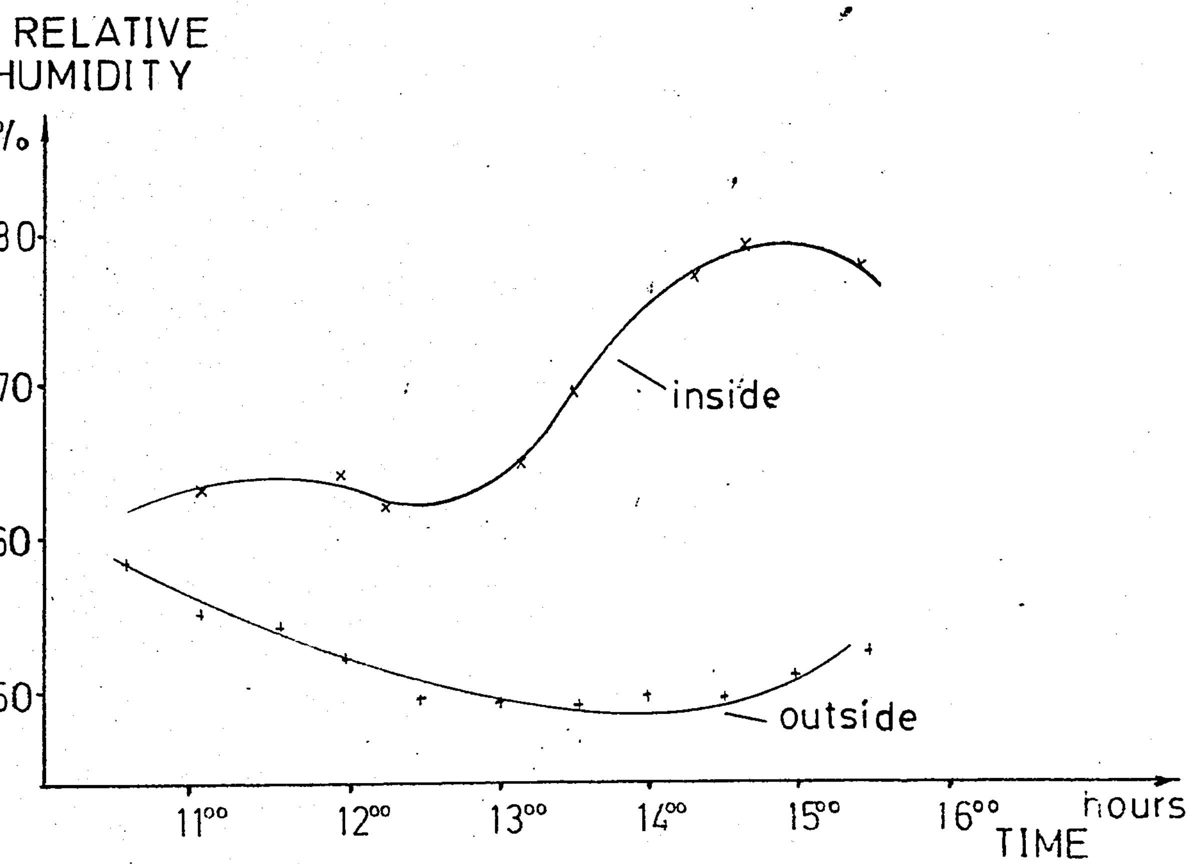
The average efficiency of the "closed" system (See energy balance table IIb) was around 67%, which was very close to the experimental results.

But for the "open" case the efficiency come out as 87% based on energy balance calculations, which was much more than the observed value.

This difference comes from the air-humidity, which was not taken into account in the energy balance calculations and from he reflectivity calculations.

In the glass covered system the air inside the enclosure ecomes nearly saturated and so condensation starts on the glass cov- er. This shows itself in the energy balance calculations as the re- uction of penetrating energy.

Penetrating energy = Total incident energy - Reflected nergy.



GRAPH IIc

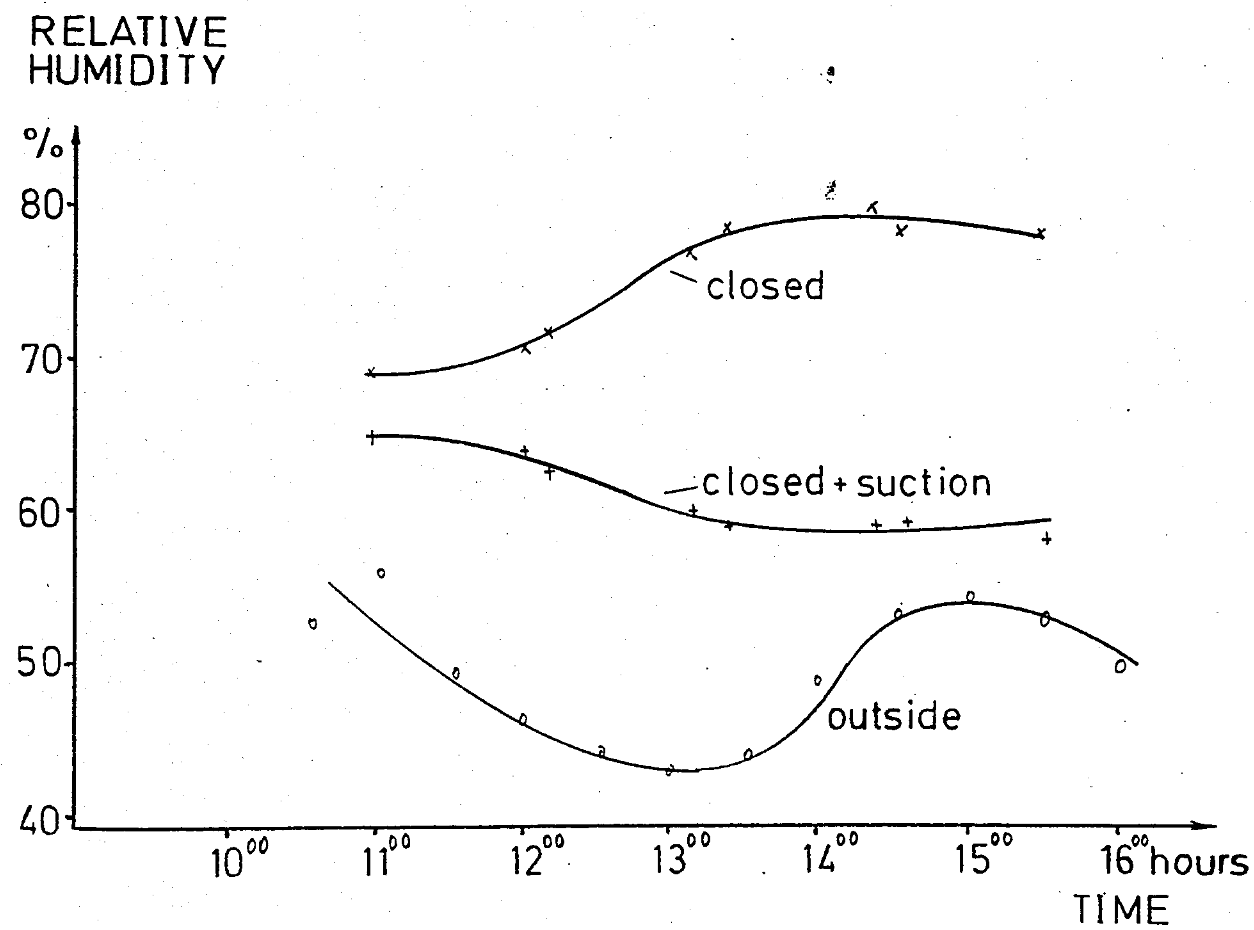
(Data table IIa,b)

Although the temperature of "closed" system is 50%-60% higher than the open case, due to condensation and air saturation problems (See Graph IIc) the evaporation rates in both systems become nearly equal. (See table IIa)

C) Comparison of glass covered evaporators with and without suction

In the fan suction system a small electrical fan, attached to the side of the evaporator (See FIG I) suchs out the vapors inside (due to change from liquid to gas phase). So the inside humidity is reduced and condensation is partially prevented in the 70% of the whole glass cover area.

The three day averaged efficiency of the "fan system" is 38% better than the "no fon" case (See table IIIa)



(Data table Va,b)

GRAPH IIIc

The average plate temperature is reduced to $36^{o}C$, so absorbed energy decreases sharply (See table IIIb)

The heat transfer losses of the "suction system" are also decreased, because of the reduced system vapor temperature.

Using the energy balance calculations the efficiency of the "suction system" was found to be around 78%, which is lover than the observed value (observed 93%) . The difference should be in the estimation of infiltration and reflection losses.

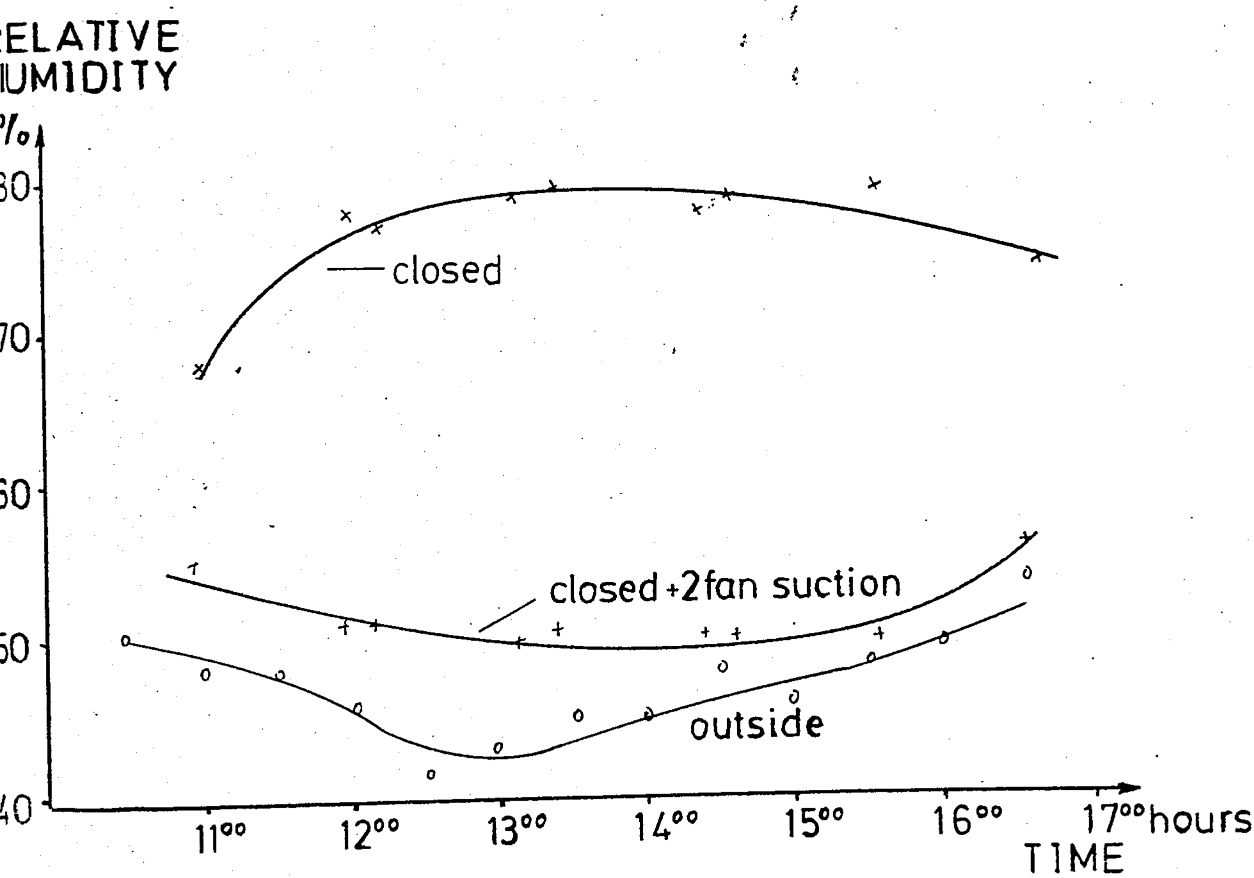
Graph IIIc shows the difference in system humidities .

D) Comparison of two fan system with glass covered system without suction

In "two fan suction" system the number of connected fans are increased to two. So the condensation is totally prevented and the inside humidity is reduced to 51% on the average(See graph IVc)

The two day averaged efficiencies of the two fan system is 54% better than the closed case without suction.(See table IVa)

Plate and, vapor temperatures are reduced to $31^{o}C$ and $25^{o}C$ respectively. Heat transfer and absorbed energy losses are decreased to a minimum.



(Data table VIII a,b)

GRAPH IVc

X: CONCLUSIONS AND RECOMMENDATIONS

Best efficiencies were observed with closed evaporators with fan suctions. (90%-95%). The cost of such an evaporator is around 10.000 TL/m^2 (See chapter VII). But like every solar collector the glass cover becomes an important disadvantage in long time use. Dirt, dust and high maintanence costs make the huge glass covers an undesired part of solar collectors. Here for the glass covered "Solar Brine Evaporators" same disadvantages exist. But without glass cover the efficiency was around 65% and the cost of such an open evaporator is 7000 TL/m , So cost and efficiency de - crease by the same amount (30% in both). But for an open evaporator near Denizli, where the salt lakes are located and salt production facilities are built, the weather conditions are much more suitable than in Istanbul, because of the increased radiation and reduced air humidity, which affect evaporation directly.

The average humidity in Istanbul between May and September is 72% where in Denizli it decreases to 48% on the average. (See tables 2 and 3)

The reduction of average humidity from 52% to 45% increases the efficiency from 61% to 77% (See result table IIa)

Therefore near Denizli (ACIGÖL) an open evaporator can work with 70%- 80% efficiency.

As a conclusion open "Solar Brine Evaporator's" with low cost low maintenance and easy application are a necessary and suitable stage in the production of sodium sulfate. And for future applications aliminium-sheet can be a good recommendation for evaporator plates, because corrosion is an important problem when working with sodium sulfate.

APPENDIX

SAMPLE CALCULATION

Based on :

Table of results

IV a,b E_{xp} VII

Total Incident Energy :

Total incident energy = (Platesurface) × (mV) × (calibration factor) × (Exp.period)

plate surface : 1.2 m^2

Avg. Pyronometer readings: 6.5 mV

Calibrationfactor: 1 mV ≅ 82.1 kcal/hr m^2

Experimentation period : 5 hr

Total incident energy = 3201 kcal

Avg. Plate Temperature :

Geometrical avg. of the plate temps. diving the whole exp.period. (from table VIIb)

30 °C for two fancase

45 °C for closed case no suction.

Avg Vapor Temperature :

Gemetrical avg. of the enclosure vapor temp. during the whole exp period. (from table VIIb)

24 °C for two fan case 39 °C for closed case nosuction

Total Evaporation

between $11^{30} - 12^{30}$ $V_E = V_{IN} - V_{OUT}$

$$V_{IN} = \frac{\mathring{B}e_{OUT} \times V_{OUT}}{\mathring{B}e_{IN}}$$

$$V_{IN} = \frac{5.7 \times 3.4}{5} = 3.8$$

$V_E = 0.47$ liter

for 5 hours $V_E = 3.5$ liter for closed system

$V_E = 5.6$ liter for closed + 2 fan suction system

Evaporation Energy

$$Q_E = V_E \times \rho_{H_2O} \times \Delta h\Big|_{T_E}$$

$$\rho_{H_2O} = 1.0 \ \ kg/dm^3$$

$$h_{fg}\Big|_{T_E} = \Delta h\Big|_{30^\circ C} = 580.8 \ kcal/kg$$

(Threlkeld)

$V_E = 5.6$ liter assume 1 liter $H_2O \cong 1$ kg

$$Q_E = 5.6 \times 1 \times 580.8$$

$Q_E = 3253$ kcal for two fan suction system

Total Output

V_{OUT}	Sum of the output volumes of every hour
16.1 liter	for two fancase
16.5 liter	for closed case no suction;from table VII a in App.

Output °Bè

Geometrical avg of the hourly output °Bè use table VII a in App.

6.8 °Bè	for two fan case
6.0 °Bè	for closed case no suction

Efficiency %

$$\eta = \frac{\text{EVAPORATION ENERGY}}{\text{TOTAL INCIDENT ENERGY}} \times 100 = \frac{3253}{3201} \times 100 = 102\ \%$$

Avg. Air Temp , Avg Relative Humidity , Avg Wind velocity :

Geometrical averages of the measured data use Data Table VIIa

Input ^{o}Be

Bome of the input solution, here 5 °Bè

Total Penetrating Energy

penetrating energy = Incident energy − reflected energy

Reflected energy: for no fan case 15 % of incident energy

" one fan " 10 % " "

" two fan " 8 % " "

for no fan case 15 % is experimentally observed using photo resistance measurements. in the summer of 1982 (TABLE 6)

for two fan case $0.92 \times 3201 = 2944$ kcal

Total Heat losses

Infiltration losses + heat transfer losses = total losses

(Assumption) - infiltration losses are 5% of incident energy

$$Q = Q_{FRONT} + Q_{BACK}$$

$$Q_{FRONT} = U_F \times A \times (\bar{T}_{VAP} - \bar{T}_{AIR})$$

$$\bar{T}_{VAP} = 24\,^\circ C$$

$$\bar{T}_{AIR} = 21.5\,^\circ C$$

$$A = 1.2\ m^2$$

$$\frac{1}{U_F} = \frac{1}{h_i} + \frac{1}{C} + \frac{\Delta X_G}{k_G} + \frac{1}{h_o}$$

where $h_o \doteq 28.4\ W/m^2 C$

(wind speed $\approx$ 10 miles/hr)

$C = 7.95\ W/m^2 C$

$h_i = 11.2$ "

$$U_F = 3.43 \quad kcal/hr\,^\circ C$$

$$Q_F = 3.43 \times 1.2 \times 2.5 = 10.4 \ kcal$$

$$Q_{BACK} = U_B \times A \times (\bar{T}_{PLATE} - \bar{T}_{AIR})$$

$$\bar{T}_{PLATE} = 30\,^\circ C$$

$$U_B = 1.32\ kcal/hr\,^\circ C$$

$$Q_B = 1.32 \times 1.2 \times 8.5 = 13.5 \ kcal$$

$$Q = 10.4 + 13.5$$

$$Q = 23.9 \ kcal$$

Heat transfer losses

$10.4 + 13.5 = 23.9$ kcal

Infiltration loss

$\%5 \times 3201 = 160$ kcal

Total heat losses $= 23.9 + 160$

$= 183.9$ kcal

Total Absorbed Energy

Absorbed energy $= V_{OUT} \times \rho_{OUT} \times C_{SOL} \times (\bar{T}_{PLATE} - \bar{T}_{AIR})$

$\bar{T}_{PLATE} = 30\,°C$

$\bar{T}_{AIR} = 21.5\,°C$

$C_{SOL} = 0.95$ kcal/kg °C

$\rho_{OUT} = 1.05$ kg/liter (TABLE 4)

$V_{OUT} = 16.1$ liter

Absorbed energy $= 136.5$ kcal

Remained Energy for Evaporation

Q_E = Penetrating energy - Heat losses - Absorbed energy

$= 2615$ for two fan suction system

$= 2049$ for closed system

Efficiency

$$\eta = \frac{\text{Remained energy for evaporation}}{\text{Total incident energy}} \%$$

$\eta = \frac{2615}{3201} \% = 81.6 \%$ for two fan suction

$\eta = \frac{2049}{3201} \% = 64.0\%$ for closed system

Date: 23.8.1983

DATA TABLE Ia

Note: Closed vs open system
(No fan suction)

TIME	AIR TEMP	WIND VEL/DIR	PYRONOMETER	POSOMETER	AIR HUMIDITY
hours	°C	BEF/DIR	mV	100 ASA. 21 DIN	%
10^{00}	21.0	1-2/NE	4.2	15.8	64
10^{15}			4.1	15.7	
10^{30}	21.5	2/NE	3.5	15.5	62
10^{45}			4.0	15.8	
11^{00}	22.0	2-3/NE	5.1	15.9	63
11^{15}			4.1	16.0	
11^{30}	22.5	2-3/NE	3.1	15.8	60
11^{45}			2.6	15.7	
12^{00}	22.5	3/NE	3.1	15.8	57
12^{15}			2.8	15.8	
12^{30}	22.5	3-4/NE	6.1	16.2	54
12^{45}			8.1	16.6	
13^{00}	23.5	3-4/NE	6.9	16.1	50
13^{15}			8.0	16.7	
13^{30}	24.0	3-4/NE	8.3	16.8	48
13^{45}			8.2	16.5	
14^{00}	24.5	3/NE	8.5	16.9	47
14^{15}			8.7	17.1	
14^{30}	24.5	3/NE	8.7	17.0	48
14^{45}			8.4	16.6	
15^{00}	23.5	3/NE	8.3	16.5	47
15^{15}			8.0	16.2	
15^{30}	23.0	2-3/NE	8.0	16.0	50
15^{45}			7.9	16.1	

Date: 23.8.1983

DATA TABLE Ib

Note: Closed vs open system
Evaporator I closed; (I/II)
Evaporator II open

INPUT °Bé: 5.0

TIME	SYST. HUMIDITY	EVAPORATOR I	EVAPORATOR II	OUTPUT °Be	OUTPUT VOLUME
(hours)	I/II	plate Temp/ Vapor Temp	plate Temp/ Vapor Temp	I/II	I/II
	(%)	(°C)	(°C)	(°Be)	(liters)
0^{00}	69/-	34/29.5	24/-	-	-
1^{00}	74/-	37/33.5	25/-	5.6/5.5	2.5/2.7
1^{10}	75/-	39/34	26/-	-	-
2^{10}	75/-	41/35	25/-	5.5/5.5	3.5/3.6
2^{20}	76/-	42/36	25/-	-	-
3^{20}	75/-	42/37	27/-	5.6/5.5	4.0/4.4
3^{30}	76/-	43/38	27/-	-	-
4^{30}	76/-	43/38	27/-	6.4/5.8	2.5/3.5
4^{40}	75/-	43/38	27/-	-	-
5^{40}	73/-	41/36	26/-	6.4/6.3	2.4/3.0

Date: 25.8.1983 DATA TABLE IIa

Note: Closed vs Open system
(No fan suction)

TIME	AIR TEMP	WIND VEL/DIR	PYRONOMETER	POSOMETER	AIR HUMIDITY
ours	^oC	BEF/DIR	mV	100 ASA -21DIN	%
10^{30}	22.0	2-3/NE	4.3	15.8	58
10^{45}			4.3	15.9	
11^{00}	22.5	2-3/N	4.1	16.0	55
11^{15}			3.1	15.9	
11^{30}	22.5	3/N	6.1	16.3	54
11^{45}			7.1	16.4	
12^{00}	22.0	3/NE	7.5	16.5	52
12^{15}			6.0	16.4	
12^{30}	23.5	3/NE	3.5	16.0	50
12^{45}			4.0	16.2	
13^{00}	24.0	3/NE	8.4	16.9	50
13^{15}			3.0	15.9	
13^{30}	24.5	3/NE	2.8	15.8	50
13^{45}			3.4	16.0	
14^{00}	24.5	3-4/NE	8.7	17.0	51
14^{15}			8.4	17.0	
14^{30}	25.0	3/NE	6.8	16.2	51
14^{45}			8.0	16.3	
15^{00}	25.0	3-4/NE	6.5	16.2	52
15^{15}			3.1	15.7	
15^{30}	24.0	3-4/NE	3.7	16.0	53

Date: 25.8.1983

DATA TABLE IIb

Note: Closed vs open system (I/II)

Evaporator I closed

Evaporator II open

INPUT °Bė: 5.0

TIME (hours)	SYST. HUMIDITY (I/II) (%)	EVAPORATOR I Plate Temp. /Vapor Temp. (°C)	EVAPORATOR II Plate Temp. /Vapor Temp. (°C)	OUTPUT °Bė I/II (°Bė)	OUTPUT VOLUME I/II (liters)
11^{00}	63/-	35/31	25/-	-	-
12^{00}	64/-	34/31	25/-	5.2/5.4	6.5/5.0
12^{10}	62/-	35/31	25/-	-	-
13^{10}	65/-	30/28	25/-	5.5/6.1	3.3/2.3
13^{20}	73/-	35/30	25/-	-	-
14^{20}	78/-	40/35	26/-	6.2/5.9	3.0/3.2
14^{30}	80/-	43/37	26/-	-	-
15^{30}	78/-	41/33	26/-	6.7/7.0	2.8/1.9

Date: 26.8.83 DATA TABLE IIIa

Note: Closed vs open system

(No fan suction)

TIME (hours)	AIR TEMP. °C	WIND VEL/DIR BEF/DIR	PYRONOMETER mV	POSOMETER 100ASA-21DIN	AIR HUMIDITY %
10^{45}	21.0	2/E	4.1	15.8	68
11^{00}	21.0	2/E	4.2	15.7	67
11^{15}			6.0	16.0	
11^{30}	21.5	2/NE	4.3	15.7	65
11^{45}			5.1	15.8	
12^{00}	22.0	2/NE	6.3	16.0	50
12^{15}			6.5	16.1	
12^{30}	22.5	2/E	6.9	16.3	48
12^{45}			6.6	16.0	
13^{00}	23.5	2/E	7.0	16.4	38
13^{15}			7.0	16.4	
13^{30}	24.5	2/E	7.5	16.5	35
13^{45}			7.9	16.4	
14^{00}	25.0	1/NE	8.5	16.7	35
14^{15}			8.4	16.6	
14^{30}	25.0	1/NE	8.3	16.6	37
14^{45}			8.2	16.5	
15^{00}	24.5	1/E	8.1	16.5	40
15^{15}			8.0	16.4	
15^{30}	24.0	2/NE	7.7	16.2	43

Date: 26.8.83

DATA TABLE IIIb

INPUT ^{o}Bė: 5.0

Note: Closed vs open system (I/II)

Evaporator I closed

Evaporator II open

TIME	SYSTEM HUMDITY	EVAPORATOR I	EVAPORATOR II	OUTPUT ^{o}Bė	OUTPUT VOLUME
(hours)	I/II	Plate Temp. /Vapor Temp.	Plate T. /Vapor T.	I/II	I/II
	(%)	(^{o}C)	(^{o}C)	(^{o}Bė)	(liters)
11^{00}	75/-	33/28	25/-	-	-
12^{00}	80/-	37/30	25/-	6.1/6.1	3.5/3.0
12^{10}	81/-	38/32	25/-	-	-
13^{10}	78/-	41/34	26/-	6.8/6.4	2.5/3.1
13^{20}	75/-	46/38	26/-	-	-
14^{20}	79/-	49/39	26/-	6.7/6.6	3.2/3.0
14^{30}	80/-	49/39	25/-	-	-
15^{30}	79/-	42/33	24/-	6.2/6.4	4.1/3.4

DATA TABLE IVa

DATE: 7.9.1983

Note: closed vs closed + fan suction

TIME hours	AIR TEMP °C	WIND VEL/DIR BEF/DIR	PYRONOMETER mV	POSOMETER 100 ASA-21 DIN	AIR HUMIDITY %
10^{30}	22.0	2/E	6.3	15.3	55
10^{45}			6.1	15.5	
11^{00}	22.5	2/E	6.4	15.6	52
11^{15}			6.5	15.6	
11^{30}	22.5	2/E	6.5	15.6	50
11^{45}			6.7	15.9	
12^{00}	23.0	2-3/E	6.8	16.0	48
12^{15}			7.1	16.1	
12^{30}	23.5	2-3/E	7.4	16.2	45
12^{45}			7.5	16.3	
13^{00}	24.0	2/E	7.8	16.5	42
13^{15}			7.9	16.5	
13^{30}	24.5	2-3/E	8.1	16.6	37
13^{45}			8.2	16.6	
14^{00}	25.5	2-3/E	8.3	16.7	37
14^{15}			8.3	16.7	
14^{30}	25.5	2/E	8.2	16.6	37
14^{45}			8.1	16.5	
15^{00}	25.0	2/E	8.0	16.5	38
15^{15}			7.9	16.4	
15^{30}	24.5	2/E	7.7	16.4	39
15^{45}			7.5	16.3	
16^{00}	24.0	1/E	7.4	16.2	39

DATA TABLE IVb

Date: 7.9.1983

Note: Closed vs closed + fan suction

Evaporator I fan suction system

Evaporator II closed (no fan)

Input °Bé: 5.0

TIME (hours)	SYSTEM HUMDITY I/II (%)	EVAPORATOR I Plate Temp. /Vapor T. (°C)	EVAPORATOR II Plate Temp. /Vapor T. (°C)	OUTPUT °Bé I/II °Bé	OUTPUT VOLUME I/II (liters)
10^{30}	60/65	36/30	40/35	-	-
11^{30}	60/68	35/30	41/35	5.8/5.8	3.5/3.0
11^{40}	58/69	37/31	43/38	-	-
12^{40}	57/75	36/31	47/42	6.2/5.8	4.1/4.2
12^{50}	58/78	37/32	48/44	-	-
13^{50}	56/80	36/30	50/46	7.1/6.2	3.3/3.7
14^{00}	57/81	37/30	51/47	-	-
15^{00}	56/80	36/29	50/47	8.3/6.1	2.4/3.4
15^{10}	56/79	35/28	49/45	-	-
16^{10}	55/77	35/28	48/45	7.7/6.5	2.7/3.6

Date: 12.9.1983

DATA TABLE Va

Note: closed vs closed + fan suction

TIME	AIR TEMP.	WIND VEL/DIR	PYRONOMETER	POSOMETER	AIR HUMIDITY
hours	°C	BEF/DIR	mV	100 ASA-21 DIN	%
10^{30}	22.5	1/SE	6.2	15.4	54
10^{45}			6.3	15.3	
11^{00}	23.0	1-2/SE	6.4	15.5	57
11^{15}			6.2	15.2	
11^{30}	24.0	1/SE	6.3	15.3	50
11^{45}			6.4	15.3	
12^{00}	25.0	1-2/SE	6.6	15.9	47
12^{15}			6.7	16.0	
12^{30}	25.0	1/SE	6.8	16.1	45
12^{45}			6.8	16.1	
13^{00}	25.5	0-1/SE	6.9	16.2	44
13^{15}			7.1	16.2	
13^{30}	26.0	0-1/SE	6.9	16.2	45
13^{45}			7.4	16.1	
14^{00}	27.0	1/E	7.7	16.3	50
14^{15}			6.9	16.0	
14^{30}	27.0	1-2/SE	6.8	16.0	55
14^{45}			6.7	16.1	
15^{00}	26.0	1-2/SE	6.5	16.1	56
15^{15}			6.3	15.8	
15^{30}	25.5	2/SE	6.3	15.7	54
15^{45}			6.1	15.4	
16^{00}	24.5	2/SE	6.0	15.0	51

Date: 12.9.1983 DATA TABLE Vb

Note: Closed vs closed + fan suction INPUT °Bé: 4.0

Evaporator I fan suction system

Evaporator II closed system

TIME (hours)	SYST. HUMIDITY I/II %	EVAPORATOR I Plate Temp. /Vapor Temp. (°C)	EVAPORATOR II Plate Temp. /vapor T. (°C)	OUTPUT °Bé I/II (°Bé)	OUTPUT VOLUME I/II (liters
11^{00}	67/70	34/28	36/31	-	-
12^{00}	66/72	34/29	36/31	5.0/4.6	2.8/3.1
12^{10}	64/73	35/29	36/31	-	-
13^{10}	62/78	35/28	40/34	4.9/5.3	3.7/2.4
13^{20}	61/80	35/30	44/37	-	-
14^{20}	61/82	36/31	46/41	5.6/5.2	2.9/3.3
14^{30}	61/81	36/30	48/43	-	-
15^{30}	60/80	35/31	44/41	5.4/4.9	3.3/3.8

Date: 13.9.1983 DATA TABLE VIa

Note: closed vs. closed - fan suction

TIME	AIR TEMP.	WIND VEL/DIR	PYRONOMETER	POSOMETER	AIR HUMIDITY
hours	^{o}C	BEF/DIR	mV	100 ASA- 21 DIN	%
10^{00}	22.0	0	6.4	15.4	53
10^{15}			6.4	15.5	
10^{30}	22.0	0	6.3	15.1	52
10^{45}			6.2	15.1	
11^{00}	22.5	1/SE	6.7	15.9	47
11^{15}			6.8	15.9	
11^{30}	22.5	1/SE	6.8	16.0	49
11^{45}			6.9	16.1	
12^{00}	23.5	1/SE	7.0	16.2	45
12^{15}			7.1	16.2	
12^{30}	24.5	1-2/SE	7.1	16.3	45
12^{45}			7.2	16.2	
13^{00}	25.0	1-2/SE	7.0	16.1	40
13^{15}			7.3	16.3	
13^{30}	25.5	1/SE	7.5	16.4	39
13^{45}			7.7	16.4	
14^{00}	26.5	1/SE	7.8	16.4	41
14^{15}			7.8	16.4	
14^{30}	27.0	1/SE	7.7	16.4	44
14^{45}			7.6	16.3	
15^{00}	27.0	1/SE	7.4	16.3	41
15^{30}	26.5		7.2	16.2	41
16^{00}	26.5	0-1/SE	6.8	15.7	46
16^{15}			6.2	15.4	

Date: 13.9.83

DATA TABLE VIb

Note: closed vs closed - fan suction

INPUT °Bé: 5.0

Evaporator I fan suction system

Evaporator II closed system

TIME (hours)	SYST. HUMIDITY I/II (%)	EVAPORATOR I Plate Temp. /Vapour T. (°C)	EVAPORATOR II Plate Temp. /Vapour Temp. (°C)	OUTPUT °Bé I/II (°Bé)	OUTPUT VOLUME I/II (liters)
10^{30}	60/64	38/34	39/34	-	-
11^{30}	58/69	37/32	41/37	6.5/6.6	1.9/2.1
11^{40}	57/70	36/32	40/35	-	-
12^{40}	56/75	38/32	45/39	6.8/6.8	2.9/2.4
12^{50}	56/75	38/31	45/39	-	-
13^{50}	57/82	37/30	51/45	7.5/7.2	3.0/2.7
14^{00}	57/80	37/30	50/45	-	-
15^{00}	56/80	36/28	49/44	7.5/6.5	3.1/3.5
15^{10}	55/78	36/28	48/44	-	-
16^{10}	56/76	32/25	44/40	7.5/6.9	2.6/2.1

Date: 27.9.1983 DATA TABLE VIIa

Note: Closed vs closed-2fan suction

TIME	AIR TEMP.	WIND VEL/DIR	PYROMETER	POSOMETER	AIR HUMIDITY
Hours	°C	BEF/DIR	mV	100ASA-21DIN	%
11^{00}	21.0	2/NE	6.9	15.8	62
11^{15}			4.8	15.3	
11^{30}	21.5	2/NE	7.6	16.2	60
11^{45}			6.1	15.5	
12^{00}	22.0	2/NE	4.0	15.1	60
12^{15}			7.8	16.2	
12^{30}	22.0	2/NE	4.5	15.1	60
12^{45}			7.8	16.1	
13^{00}	22.0	2/NE	6.1	15.4	57
13^{15}			7.9	16.0	
13^{30}	22.5	2/NE	7.8	16.1	57
13^{45}			7.8	16.0	
14^{00}	22.5	2/NE	8.1	16.2	55
14^{15}			4.1	15.1	
14^{30}	22.5	2/NE	8.0	16.1	53
14^{45}			7.8	16.1	
15^{00}	22.0	2/NE	7.7	16.0	48
15^{15}			7.6	16.0	
15^{30}	21.5	2/NE	7.2	15.9	47
15^{45}			4.1	15.2	
16^{00}	21.0	2/NE	5.4	15.5	50
16^{30}	20.5	2/NE	6.4	15.6	51
17^{00}	19.5	2/NE	4.6	15.1	54
17^{15}			4.9	15.2	

DATA TABLE VIIb

INPUT ^{o}Bé: 5.0

DATE: 27.9.1983

Note: closed vs. closed-2fan suction

Evaporator I 2fan system

Evaporator II closed system

TIME	SYST. HUMIDITY	EVAPORATOR I	EVAPORATOR II	OUTPUT ^{o}Bé	OUTPUT VOLUME
	I/II	plate Temp. /vapour T.	plate Temp. /Vapour T.	I/II	I/II
hours	%	(^{o}C)	(^{o}C)	(^{o}Bé)	(liter
11^{30}	58/68	32/27	41/35	-	-
12^{30}	56/76	31/25	43/38	5.7/5.5	3.4/3.
12^{40}	57/47	31/25	43/38	-	-
13^{40}	53/80	30/24	47/41	7.6/6.6	2.6/3.
13^{50}	54/79	30/23	48/41	-	-
14^{50}	54/82	31/24	51/44	7.7/6.3	3.2/4
15^{00}	55/80	30/24	50/44	-	-
16^{00}	57/77	30/23	46/41	7.2/6.4	2.9/2
16^{10}	58/75	29/23	47/40	-	-
17^{10}	59/72	26/24	35/30	5.8/5.5	4.0/

DATE: 29.9.1983

DATA TABLE VIIIa

Note: closed vs closed-2fan suction

TIME HOURS	AIR TEMP. °C	WIND VEL/DIR BEF/DIR	PYROMETER mV	POSOMETER 100ASA-21DIN	AIR HUMIDITY %
10^{30}	22.5	1/NE	6.3	15.5	51
10^{45}			4.1	15.3	
11^{00}	23.0	1-2/NE	6.0	15.5	48
11^{15}			6.9	15.8	
11^{30}	23.5	1-2/NE	5.1	15.5	48
11^{45}			6.9	15.8	
12^{00}	24.5	2/NE	7.2	16.0	46
12^{15}			7.6	16.0	
12^{30}	25.0	2/NE	7.8	16.1	41
12^{45}			7.1	16.0	
13^{00}	25.5	2/NE	8.1	16.2	43
13^{15}			8.2	16.2	
13^{30}	26.0	2/NE	8.2	16.1	45
13^{45}			8.1	16.1	
14^{00}	26.5	2/NE	8.3	16.2	45
14^{15}			8.3	16.2	
14^{30}	26.0	2/NE	8.3	16.2	48
14^{45}			8.1	16.0	
15^{00}	26.5	2-3/NE	7.8	16.0	45
15^{15}			7.7	15.9	
15^{30}	26.5	2-3/NE	7.6	15.9	48
16^{00}			7.2	15.8	
16^{30}	26.0	2/NE	7.1	15.7	50
16^{45}			6.8	15.7	

DATA TABLE VIIIb

Date: 29.9.1983

INPUT °Bé: 4.0

Note: closed vs. closed-2fan suction

Evaporator I 2fan system

Evaporator II closed system

TIME	SYST. HUMIDITY	EVAPORATOR I	EVAPORATOR II	OUTPUT °Bé	OUTPUT VOLUME
	I/II	Plate temp. /vapour temp.	Plate temp. /vapour temp.	I/II	I/II
hours	%	°C	°C	(°Bé)	(liters
11^{00}	55/68	30/25	38/32	-	-
12^{00}	52/78	33/28	45/38	4.8/4.3	2.0/3.4
12^{10}	52/77	33/27	46/38	-	-
13^{10}	51/79	32/26	47/39	5.2/4.9	3.0/3.
13^{20}	51/80	32/27	47/39	-	-
14^{20}	51/78	34/28	49/43	6.0/5.0	3.0/3.
14^{30}	51/79	34/28	49/44	-	-
15^{30}	50/80	33/27	50/46	6.4/5.3	2.5/2.
15^{40}	50/79	33/28	49/44	-	-
16^{40}	56/75	31/26	47/40	5.6/5.0	3.0/3

DATA TABLE IXa

Date: 6.9.1983

Note: Comparison of jute surface vs rough surface; both are open evaporators. System I jute surface System II rouh surface.

Input ^{o}Bé: 4.0

TIME (hours)	AIR TEMP. (^{o}C)	PYRONOMETER mV.(aug)	SURFACE TEMP. Syst.I/Sys.II (^{o}C)	OUTPUT ^{o}Bé Sys.I/Sys.II (^{o}Bé)	OUT VOLUME Sys I/Sys II (liters)
11^{00}	21	6.1	21.5/22.0	-	-
12^{00}	21.5		22.0/22.5	4.4/4.3	3.1/3.3
12^{15}	21.5	6.4	22.0/22.5	-	-
13^{15}	22.0		22.5/23.0	4.4/4.3	3.7/3.9
13^{30}	22.0	6.4	23.0/23.0	-	-
14^{30}	22.5		23.5/25.0	4.8/5.0	2.8/2.6
14^{40}	23.0	7.7	23.5/25.5	-	-
15^{40}	23.0		24.5/27.5	5.1/5.1	3.1/3.5
16^{00}	22.5	6.2	24.0/26.5	-	-
17^{00}	21.5		22.5/23.5	4.5/4.3	3.9/4.1

Date: 8.9.1983

Note: Voriation of evaporation rate wrt. input feed rate for jute surface system. Sys I Sys II jute surfaces

INPUT ^{o}Bé: 5.0

TIME	INPUT FEED	OUTPUT ^{o}Bé	OUTPUT VOLUME	EVAPORATION
hours	Sys I./ Sys II (liter)	Sys I/ Sys II(^{o}Bé)	Sys I/Sys II (liter)	Sys I/Sys II (liter)
10^{45}	3.0/6.0	-	-	-
11^{45}	-	10.0/5.5	1.5/5.5	1.5/0.5
12^{00}	-	-	-	-
13^{00}	2.5/7.0	8.3/5.1	1.5/6.8	1.0/0.2
13^{30}	-	-	-	-
14^{30}	1.0/4.0	10.0/7.2	0.5/2.75	0.5/1.25

Date: 9.9.1983

Note: for rough surface system

INPUT ^{o}Bé : 5.0

12^{00}	-	-	-	-
13^{00}	3.0/4.0	8.3/7.4	1.8/2.7	1.2/1.3
13^{15}	-	-	-	-
14^{15}	2.0/4.5	10.0/6.6	1.0/3.4	1.0/1.1
14^{40}	-	-	-	-
15^{40}	6.0/7.0	5.4/5.1	5.5/6.9	0.5/-

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