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Experimental Investigation of Solar Absorption Refrigeration System.

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EXPERIMENTAL INVESTIGATION OF SOLAR ABSORPTION REFRIGERATION SYSTEM

By

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ABSTRACT

In this paper, an investigation of aqua-ammonia solar absorption refrigeration system for domestic use is presented. The refrigeration system investigated here is a conventional aqua-ammonia absorption refrigeration system powered by solar heater. The solar heater used is a stationary, hyperbolic sprial solar concentrator equipped with a glass tube for convection loss suppression. A generator temperature as high as 140°C is reached. The lowest temperature reached in evaporator is 2°C, with an additional external heating source. The various parameters affected the cycle performance and cooling ratio, such as, the generator temperature, evaporator temperature, collector performance and condenser temperature are presented. For fixed initial conditions, higher generator temperature and lower condenser temperature increases the coefficient of performance and improves the operation of the system.

1. INTRODUCTION

The critical shortage of the conventional sources of energy has motivated the search of alternative energy sources. Solar energy is one of the major alternative energy sources. In cooling, the absorption refrigeration system has proven to be adequate for the utilization of solar energy. Despite that this system has been in use for many years, only recently it has been considered as a means of refrigeration by solar energy. Many investigators have worked on the absorption refrigeration system [1-4]. A comperhensive review is presented in refrence [3].

Any refrigeration system operated by solar energy comperises mainly the following elements,

1- Solar collection unit .
2- The cooling machine.
3- A storage system to regulate the energy supply.

1.1 Solar Energy Collection Unit.

Flat plate and concentrating collector may be used. The flat plate is used with refrigeration system of relatively lower generator temperature, about 80 to 100°C. Different modified forms of flat plates are known today. They produce thermal energy at comparatively reasonable temperature range to ordinary flat plates

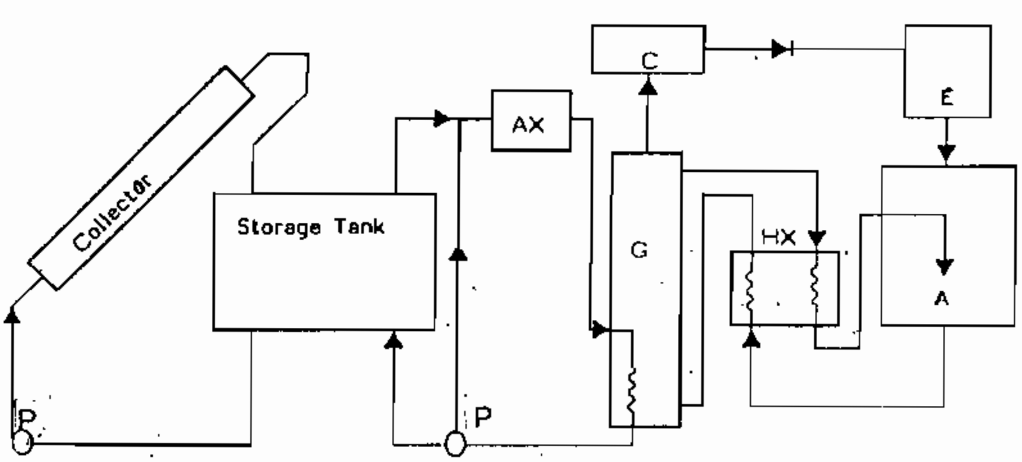
[5, 6]. Stationary concentrating collector of moderate types are used for systems of generator temperature up to 130°C. Concentrating collectors utilizing the direct portion of solar energy producing temperatures from 100°C to 400°C are used in solar refrigerators [7]. The concentrated radiation at the focus of the concentrator is collected by a heat exchanger mounted in the focal plane. This heat exchanger is called target and its shape is determined according to the concentrator type.

1.2 The Cooling Machine

It is usually absorption refrigeration machine. Two main types are used, intermittent and contineous machines. In the first the cycle comprises two major operations, regeneration and refrigeration. In regeneration the mixture is heated up to vaporise the refrigerant and subsequently condensed in a condenser. In the refrigeration operation the liquid refrigerant passes through the expansion valve where the temperature drops to the evaporator temperature. The refrigerant vaporises in the evaporator and the operations occur intermittently one after the other.

The second system is the contineous refrigeration absorption system. In this cycle the production of cooling is at the same time of regeneration of ammonia. The flow of refrigerant is contineous from the condenser to the evaporator, then to the absorber, where it is absorbed by the refrigerant - absorption solution. The strong solution is driven to the generator where it is heated up to vaporize the refrigerant and driven in the condenser. This machine has no valves and all operations are done in time.

A schematic diagram of a conventional contineous solar absorption cooling system is shown in Fig. 1. Chinnappu [8] has investigated the different cycles configurations. The results show that



C - Condenser
E - Evaporator
A - absorber
G - generator
HX - heat exchanger
AX - external heat source
P - pump

Fig. (1) Schematic Presentation of a Conventional Contineous Solar Cooling system.

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ON THE INTEGRATION OF MOMENTUM EQUATION WITH BOUNDARY LAYER SUCTION

BY

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ABSTRACT

Boundary layer suction is one of the effective methods for the prevention of flow separtion. In this paper, a method is introduced to integrate the momentum equation, in which the suction velocity is assumed constant, and the velocity gradient varies. This particular type of flow can be used to form an estimate of the quantities of air that must be sucked on an aerofoil, so that separation of flow does not take place. In this case, although the boundary layer is very thin before separation, the velocity gradient may be very large, and it is obvious that only small quantites of air withdrawn are sufficient to delay the stall.

NOMENCLATURE

c	free stream velocity,	m/s
$\bar{c}$	velocity at the outer edge of the boundary layer,	m/s
c_x	velocity of the fluid inside the boundary layer in x-direction,	m/s
c_y	velocity component inside the boundary layer in y-direction,	m/s
c_{yo}	suction velocity, normal to the surface,	m/s
H_{12}	boundary layer form parameter	
p	free stream static pressure,	N/m^2
x	streamwise coordinate	
y	coordinate perpendicular to surface	
δ	boundary layer thickness,	m
δ^{**}	momentum thickness,	m
τ_w	wall shear stress,	N/m^2
ν	kinematic viscosity of fluid,	m^2/s
ρ	density of fluid,	kg/m^3

1- INTRODUCTION

With the rapid development in the field of aeronautical engineering in recent years, the need arose for a broader treatment, especially of controlling the behaviour of boundary layer. In particular, it is often to prevent separation in actual application. Suction, is one of the effective methods which have been developed for controlling the separation. The application of suction is successfuly used in the design of aircraft wings [1] in order to reduce drag and to attain high lift.

By the use of suitable arrangements of suction slits, it is possible to remove the decelerated fluid particles from the boundary layer before they are given chance to cause separation. This enables to shift the point of transition in the boundary layer in the downstream direction.

2- MATHEMATICAL MODEL OF THE GENERAL CASE

Owing to the importance of the method of boundary layer control by suction, various mathematical methods, for the calculation of the influence of suction on boundary layer flow have been developed. Approximate methods, which deal with the boundary layer on the surface of an arbitrary shape and with arbitrary suction distribution, were developed by Schlichting [1] and Torda [2]. These methods, like those for the case of no suction, are based on the momentum integral equation. Therefore, the effect of the surface suction on the momentum equation may now be considered:

$$\frac{d\delta^{**}}{dx} + (2 + H_{12}) \frac{\delta^{**}}{\bar{c}} \frac{d\bar{c}}{dx} - \frac{c_{y_o}}{\bar{c}} = \frac{\tau_w}{\rho \bar{c}^2} \qquad (1)$$

The derivation of the momentum equation (1) , where the velocity normal to the surface is c_{y_o} , is given in [3].

Equation (1) can be written in the following form:

$$\bar{c} \frac{d\delta^{**}}{dx} + (2 + H_{12}) \delta^{**} \frac{d\bar{c}}{dx} - c_{y_o} = \frac{\nu}{\bar{c}} \left(\frac{\partial c_x}{\partial y}\right)_{y=O} \qquad (2)$$

In conjunction with this, the condition holding at the boundary is used, which can be obtained by putting y = O in the following equation of motion:

$$c_x \frac{\partial c_x}{\partial x} + c_y \frac{\partial c_x}{\partial y} = -\frac{1}{\rho} \frac{dp}{dx} + \frac{\partial^2 c_x}{\partial y^2} \qquad (3)$$

Thus :

$$c_{y_o} \left(\frac{\partial c_x}{\partial y}\right)_{y=o} = \bar{c} \frac{d\bar{c}}{dx} + \left(\frac{\partial^2 c_x}{\partial y^2}\right)_{y=O} \qquad (4)$$

A qualitative picture of flow under the influence of suction can be obtained by means of equation (4).

Let

$$\left(\frac{\partial c_x}{\partial y}\right)_{y=O} = \frac{\bar{c}}{\delta^{**}} Y_1,$$

$$\left(\frac{\partial^2 c_x}{\partial y^2}\right) = \frac{\bar{c}}{\delta^{**2}} Y_2 ,$$

Y_1 , Y_2 being numbers, where Y_1 may be of any value or zero. It may be assumed that Y_2 = O which denotes the Blasius profile , and hence in this case, Y_1 = 0.2205.

Therefore, equation (4) becomes :

$$- \frac{d\bar{c}}{dx} \frac{\delta^{**2}}{\nu} + c_{y_o} \frac{\delta^{**}}{\nu} Y_1 = Y_2$$

$$\frac{\delta^{**}}{\nu} \left(- \frac{d\bar{c}}{dx} \delta^{**} + c_{y_o} \cdot Y_1 \right) = Y_2 \tag{5}$$

From the previous equation, it is possible to deduce the qualitative nature of several types of flow. Now, a special type of flow, will be considered and one can easily estimate the amount of suction necessary to delay the stall of serofoils.

3- FLOW WITH VARYING VELOCITY GRADIENT AND CONSTANT SUCTION

The following method may be used to integrate the momentum equation, in which constant suction velocity is assumed, and the velocity gradient varies.

Suppose that, a velocity distribution is maintained, which approximated closely to the Blasius [4] distribution (i.e., $Y_1 = 0.2205$) , thus:

$$\left(\frac{\delta c_x}{\delta y} \right)_{y=o} = 0.2205 \frac{\bar{c}}{\delta^{**}} ,$$

$$\left(\frac{\delta^2 c_x}{\delta y^2} \right)_{y=0} = 0 ,$$

and assume

$$\frac{d\bar{c}}{dx} = 0.2205 \frac{c_{y_o}}{\delta^{**}} \tag{6}$$

Then, the boundary condition, in equation (4), is satisfied. The momentum equation therefore becomes:

$$\bar{c} \frac{d\delta^{**}}{dx} = - (H_{12} + 2) \delta^{**} \frac{d\bar{c}}{dx} + c_{yo} + 0.2205 \frac{\nu}{\delta^{**}}$$

Therefore ,

$$- 0.2205\, c_{yo} \frac{d^2\bar{c}}{dx^2} \frac{\bar{c}}{(d\bar{c}/dx)^2} = [0.2205 (H_{12} + 2) + 1]\, c_{yo} + \frac{\nu}{c_{y_o}} \frac{d\bar{c}}{dx} \tag{7}$$

Equation (7) would be difficult to solved, however for the case of $H_{12} = 2.5345$, the previous equation may havs the following form :

$$\frac{d^2\bar{c}}{dx^2} \frac{1}{(d\bar{c}/dx)^2} + 4.5345 \frac{\nu}{c_{y_o}^2 \cdot \bar{c}} \frac{d\bar{c}}{dx} = 0 \tag{8}$$

That equation can be integrated to give

$$-\frac{1}{d\bar{c}/dx} + 4.5345\ \frac{\nu}{c_{y_o}^2} \log \frac{\bar{c}}{c_\infty} = 0 \qquad (9)$$

The integration constant has been adjusted so that when $d\bar{c}/dx = -\infty$, i.e., at the begining of boundary layer where $\delta^{**} = 0$ in equation (6), $\bar{c} = c_\infty$.
The equation (9) can be rewritten as

$$-1 + 4.534\ \frac{\nu}{c_{y_o}^2}\ \frac{d\bar{c}}{dx} \log \frac{\bar{c}}{c_\infty} = 0 \qquad (10)$$

By integrating the previous equation, the following form is obtained :

$$\frac{x.c_{yo}^2}{c_\infty \nu} = 4.5345 \left(\frac{\bar{c}}{c_\infty} \log \frac{\bar{c}}{c_\infty} - \frac{\bar{c}}{c_\infty} + 1 \right) \qquad (11)$$

The integration constant may be adjusted using the following condition :

at $x = 0 \quad \bar{c} = c_\infty$

From equation (10), $d\bar{c}/dx$ is found in terms of $\bar{c}$, and by substituting $d\bar{c}/dx$ in equation (6), δ^{**} can be obtained.

The calculations of the above equations give the results which are given in table 1. Moreover, these results are illustrated in Fig. 1, which shows the flow with varying velocity gradient and constant suction.

4- CONCLUSION

The previous method may be used to integrate the momentum equation, in which constant suction velocity is assumed, and the velocity gradient varies. The particular type of flow can be used to form an estimate of the quantities of air that must be sucked on an aerofoil at high incidence so that separation of flow does not take place. In this case, although the boundary layer is very thin before separation, the velocity gradient may be very large, and it is obvious that only small quantities of air withdrawn are sufficient to delay the stall.

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further development may be through the multistagging. The choise of working fluid for the design of the solar absorption system has been investigated by Alizadeh et-al., [1] and Ellington et-al., [7]. It has been shown that the choise of mixture (refrigerant and absorbent) is dependent on the operation temperature. and the properties of mixture. The refrigerant should have a boiling point in the vicinity of 0°C to 100°C with a vapor pressure close to atmospheric pressure. This assures a pressure close to atmosphere in the absorbent - evaporator to minimize the effect of leakage in the equipment. The absorbent boiling point should be high and its vapor pressure should be low to avoid the use of rectification. The combination of refrigerant - absorbent should have the following characteristics;

i- They must be mutually soluble.
ii- They should have low viscosity.
iii- The specific heat should be low.

1.3 Solar Energy Storage

Considerable attention has been given to the problem of storing solar energy due to the intermittance nature of sun's radiation energy. A short term storing system is one in which solar radiation is stored during the day and used during the night. Storing solar energy during long summer days for use in long cloudly spells of winter denotes the long term energy storage. Many storage systems have been established such as thermal, chemical and electrical. Among which is the heat storage system. It is classified into two main types, sensible heat storage and change of state heat storage systems. The capacity of a sensible heat storage system depends mainly on the storage material which is water or pack pebbles. In the second system heat is stored by transforming the material to another, i.e. changing water to steam or soild material to liquid.

The objective of this research work is to present the experimental performance of a domestic aqua-ammonia absorption refrigeration system working by solar energy. Thermal analysis of a basic refrigeration system using solar concentrating collector is given. Some efficiency factors are defined to help in evaluating the system performance. Solar energy supply to the refrigerator is measured for El-Mansoura area and temperature levels of concentrating collector, evaporator and condenser for a certain cooling loads are investigated.

2. EXPERIMENTAL SETUP AND PROCEDURES

The solar refrigeration system used in this experiment comprises two main parts, the cooling machine and the solar heat source to operate it. The schematic arrangement of the experimental setup is shown in Fig. 2. The cooling machine investigated here is a conventional aqua-ammonia absorption refrigeration machine powered by solar heater. Its major components are a generator, condenser, evaporator and absorber as shown in Fig. 3. The refrigerant passes through all units, while the absorbent is confined to movement through the generator and absorber.

A primary set of experiments were made on the cooling machine to deduce its performance. An electrically heated fluid was used to operate the refrigerator with different power input and temper-

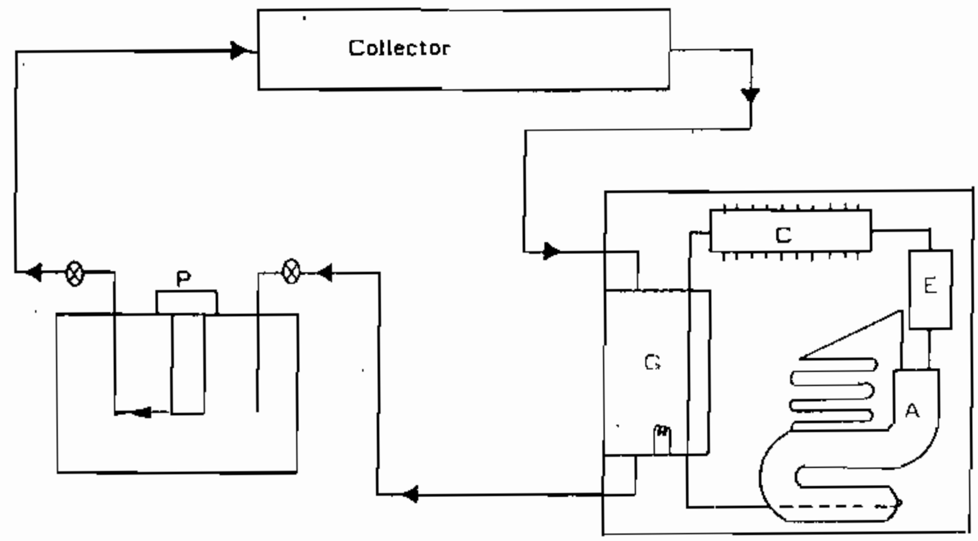


Fig. (2) Schematic Arrangement of The Experimental Set-Up

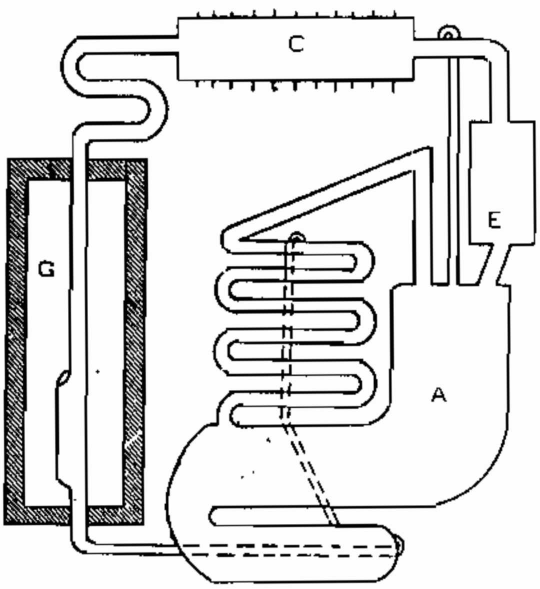


Fig. (3) Schematic Diagram of absorption Refrigerator

C - Condenser
E - Evaporator
A - absorber
G - generator
HX - heat exchanger
AX - external heat source
P - pump

ature levels. From these experiments it was found that the electrolux refrigerator used in this work is capable of **carrying** over a machine cooling load of 101 KJ/hr effectively. The average coefficient of performance in that case is 0.247 and the produced cold space temperature varied between 2°C to 10°C. The required quantity of heat supply to the generator is about 409 KJ/hr, the maximum fluid temperature required in the generator is 130°C. According to these requirements, commercial electrical motor lubricating oil is suggested to use as a working fluid in the generator heat exchanger. It has a boiling temperature of about 190°C, thus it can carry the load by sensible heating to avoid the troubles of evaporation.

The solar collector used in this work is a moderate concentrator, hyperbolic spiral [5], Fig. 4. Its projected area is 0.939 m^2. It is used to heat a working fluid to the required temperature.

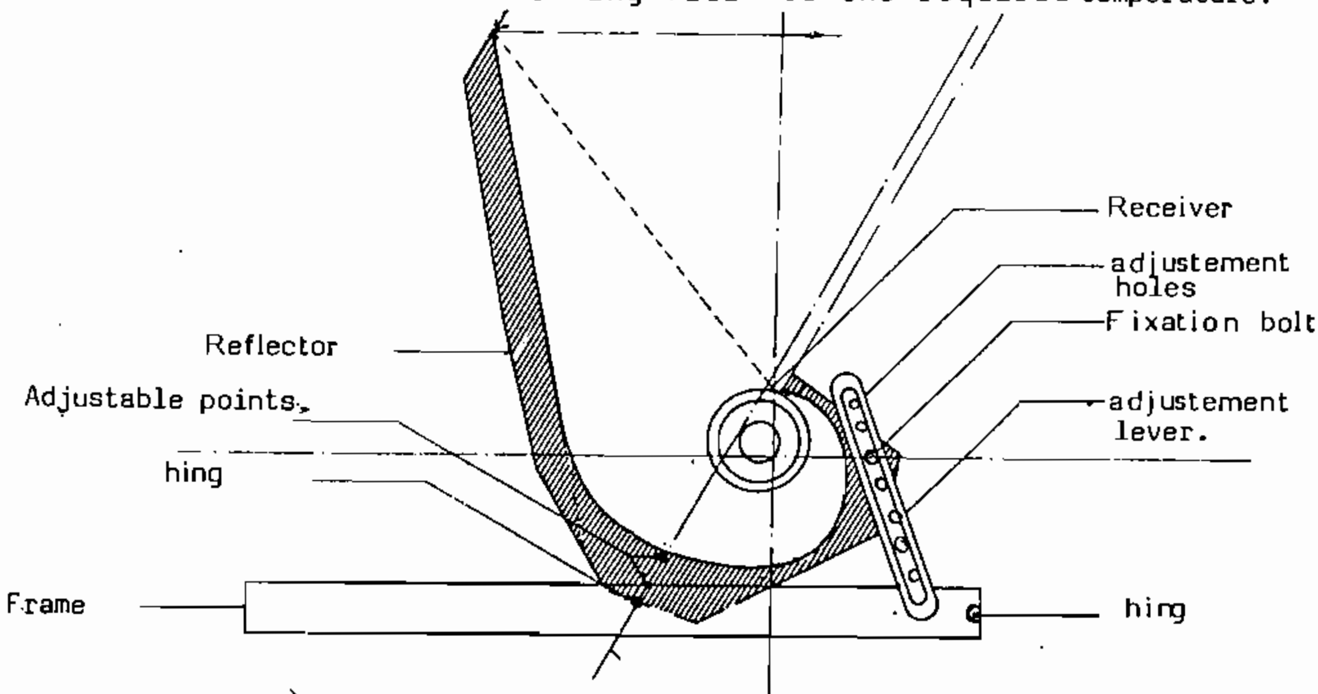


Fig. (4) Collector system

This hot fluid, then replaces the electrical heater in operating the generator of the cooling machine. Thus, it is clear that a solar refrigerator system must include at least two heat exchangers. The first heat exchanger linked with the solar radiation energy which is used to heat the working fluid by solar radiation (solar absorber). The second heat exchanger is linked with the ammonia generator to heat the solution. The generator heat exchanger used in this work is shown in Fig. 5. It is a counter flow heat exchanger. It consists of two concentric tubes, the hot fluid (oil) flows downwards in the outer tube, while solution is flowing upwards in the inner tube. The solar absorber heat exchanger used in this work mainly consists of two concentric copper tubes 34/40 and 76/80 mm diameter of 1600 mm length.

The solar radiation intensity, normaly transmittance and reflectance is measured by a silicon cell Pyrheometer. The total solar beam radiation is measured by a silicon cell Pyromemeter. A set of calibrated copper-canstantan thermocouples with copper leads in connection with a multimeter temperature recorder were used to measure the temperatures at different locations. The thermocouples are located as follows; 3 at the solar absorber terminals and center, 2 at the generator terminals, 3 at the glass cover terminals and center, one at the condenser surface and one at the evaporator surface. Error analysis has been conducted according to the standard procedure. The expected error in temperature is less than + 0.5% and the expected error in solar intensity measurement is ± 1%.

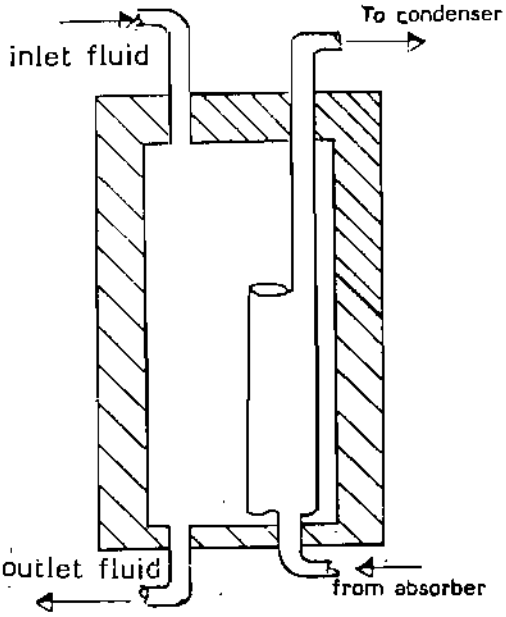


Fig. (5) Generator Heat Exchanger

3. THERMAL ANALYSIS

This part deals with the evaluation of all thermal processes in a solar refrigeration elements. The system components as shown in Fig. 2, are mainly solar collector, storage equipment and absorption cooling machine.

3.1 Solar Collector Analysis

The solar heat received by the solar reflector surface is subjected to both optical and thermal losses. The amount of solar energy received is a function of solar intensity and the collector net area. Thus, the design of a solar system to operate a refrigerator is controlled by many factors, among of which are the following;

1- The availability of solar energy in refrigerator location.
2- Amount of energy required to operate the system.
3- The required temperature level to start and continue the operation.
4- The atmospheric condition.

A pattern of the total daily solar radiation measured for normal incident of Mansoura Laboratory is shown in Fig. 6. From this diagram it can be seen that, the total daily solar radiation incident on the normal surface at El-Mansoura varies between 4600-5500 (w/m^2 day). The useful heat gained from the sum is a function of solar intensity and the collector net area and its efficiency. It is considered as the input of the solar refrigerator system and is given by

$$Q_{use} = Q_t - Q_{losses} \tag{1}$$

where,

Q_t is the heat received by the solar absorber surface at a focal plane

Q_{losses} is the amount of heat losses either by convection or radiation.

Q_{use} is the amount of heat exchanged by the solar abeorber to the heating fluid by convection.

The equations given below represent these quantities of heat,

$$Q_{losses} = Q_{conv} + Q_{rad} \tag{2}$$

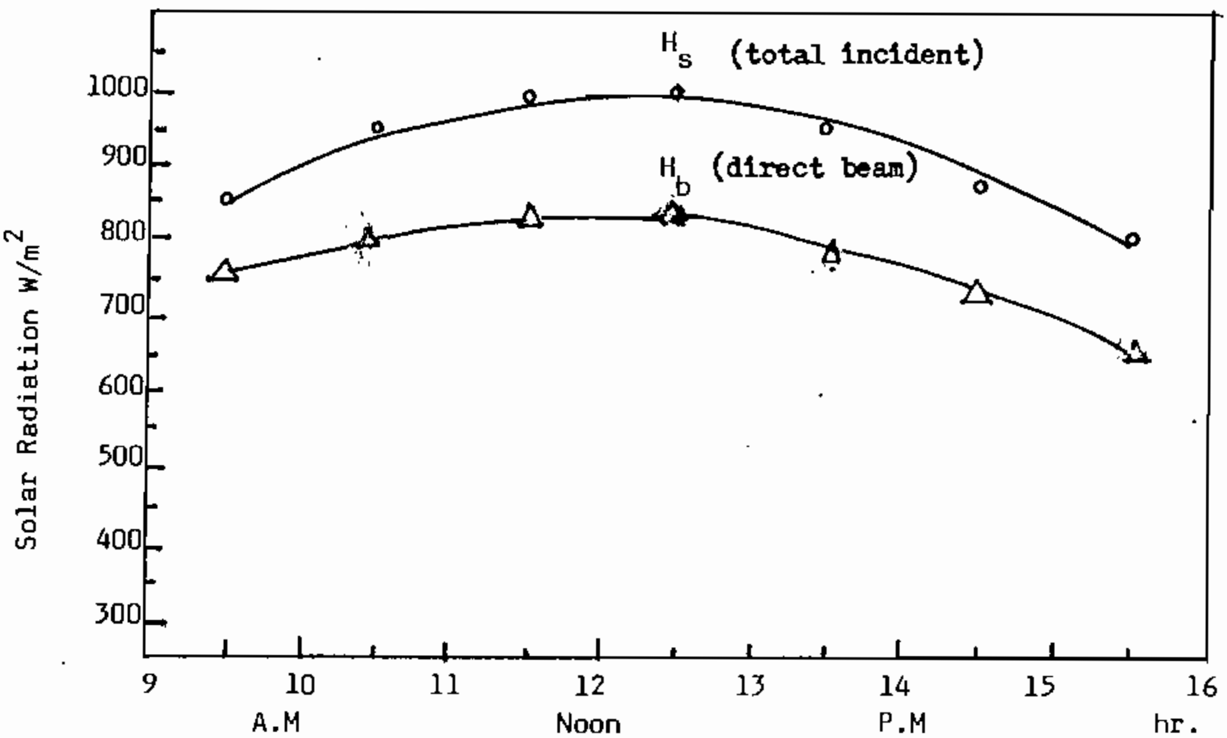


Fig. (6): Solar Radiation Along the Day.

$$Q_{conv} = A \; h \; (t - ta) \tag{3}$$

$$Q_{rad} = \epsilon \; \sigma \; A \; (T^4 - T_a^{\;4}) \tag{4}$$

$$Q_{use} = m_f \; c_p \; (\Delta t_f) \tag{5}$$

The collector efficiency (η) can be defined as,

$$\eta = \frac{Q_{use}}{Q_t} \tag{6}$$

3.2 Refrigerator Thermal Analysis

The performance of the refrigerator system is measured by its coefficient of performance (COP). Three COP may be defined for the system, thermodynamic COP, experimental COP of the machine and finally the COP_{sys} of the complete solar refrigeration system. From the thermodynamic analysis of reversible refrigeration system consisting of a generator, condenser, evaporator and absorber, Fig.7. the thermodynamic COP can be defined as;

$$(COP)_{th} = Q_1/Q_4 = \frac{T_1 \; (T_4 - T_2)}{T_4 \; (T_2 - T_1)} \tag{7}$$

From equation (7), it can be indicated that, the $(COP)_{th}$ increases with increasing the ratio of evaporator to generator absolute temperature (T_1/T_4). It is desirable to have a large temperature difference between the generator and condenser, and a small temperature difference between the condenser and the evaporator.

To determine the experimental COP, it is required to determine the actual heat quantities that would be involved in the individual processes. Thus it is required to analyze the cycle in greater detail and to use the estimated or actual thermodynamic properties of the refrigerant and absorbent. For one Kg of refrigerant (NH_3) circulated through the system per unit time, the corresponding conditions through the cycle are shown in Fig. 7. By making heat and mass balance for the different processes, the following relations are obtained;

Generator, $$q_G = h_1 - fh_4 + (f-1)h_5 \tag{8}$$

Absorber, $$q_A = h_3 - fh_4 + (f-1)h_5 \tag{9}$$

Condensor, $$q_C = h_1 - h_2 \tag{10}$$

Evaporator, $$q_E = h_3 - h_2 \tag{11}$$

where, $$f = \frac{x_{NH_3} - x_{weak}}{x_{rich} - x_{weak}} \tag{12}$$

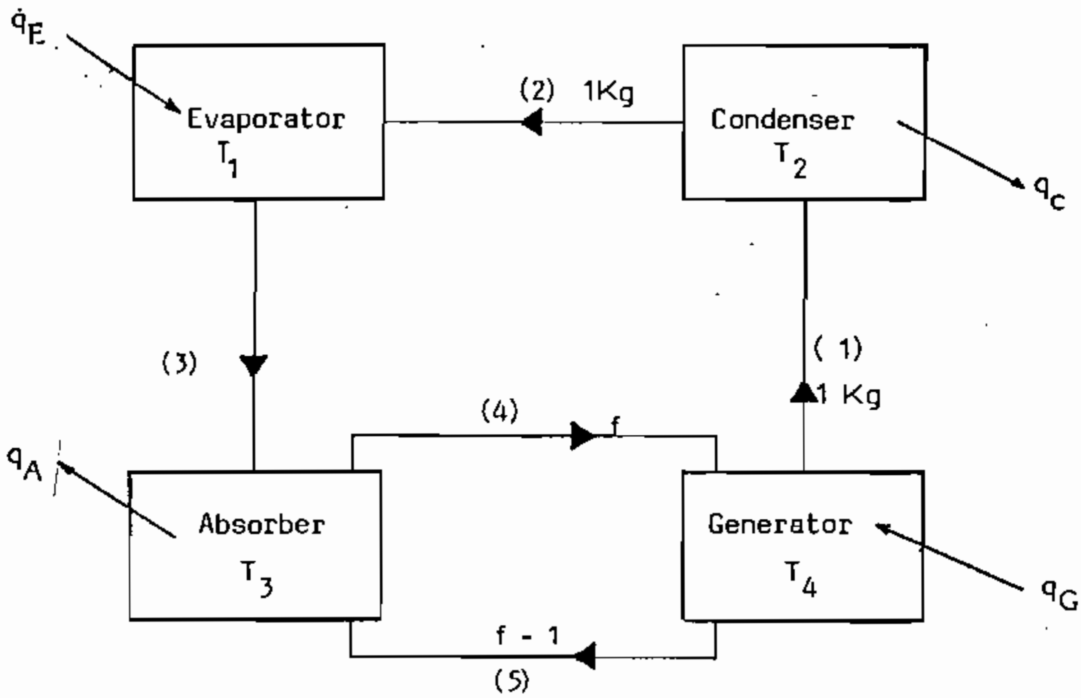


Fig. (7) Schematic Diagram of Absorption cooling system

Thus, by using the enthalpy - concentration, (H-X) thermodynamic charts for the ammonia - water, various heat quantaties involved in the different processes in the cycle can be obtained and hence the experimental COP can be determined as;

$$COP = \frac{q_E}{q_G} \tag{13}$$

The final and the most important is the actual measure of the solar refrigerator performance, COP_{sys}. It is defined as the ratio of the cooling obtainable to the amount of solar energy absorbed or received by the solar collector, i.e.

$$COP_{sys} = q_E/Q_t = \eta \cdot (COP) \tag{14}$$

4. RESULTS AND DISCUSSIONS

Series of experiments have been carried out in groups. Each group of experiments have been made for a certain cooling load at different evaporator temperature, condenser temperature and generator temperature. The cooling load is chosen to be a defined mass of water undergoes either sensible or freezing changes. The condenser temperature has been changed by using an electrical fan for cooling the condenser with different velocity of air stream. The whole plant is charged to a pressure of 17.6 ata. The partial pressure of the liquid ammonia in the evaporator falls to 4.8 ata. The major parameters affecting the performance of an absorption solar cooling system are the following;

i - Solar coolector performance.

ii- Coefficient of performance COP of the cooling machine as defined by equation (13). Equation (14) can be used to calculate the COP_{sys} provided that working fluids characteristics at different parts of the cycle are known.

The solar collector used with the cooling machine has been tested along the day. The temperature gradients of the fluid and the ambient temperature are shown in Fig. 8. The results shown in this figure are obtained with and without glass cover. These results demonestrated that the temperature gradients are a function of incidence radiation and time.It may be seen from these diagrams that the maximum outlet fluid temperature is 105°C without using the glass cover, while the maximum outlet fluid temperature reached 125°C when using the glass cover. Also the collector efficiency results are shown in this figure. The maximum collector efficiency obtained as defined by equation (6) is found to be 46% at noon, and decreases with reasonable values before and after noon.

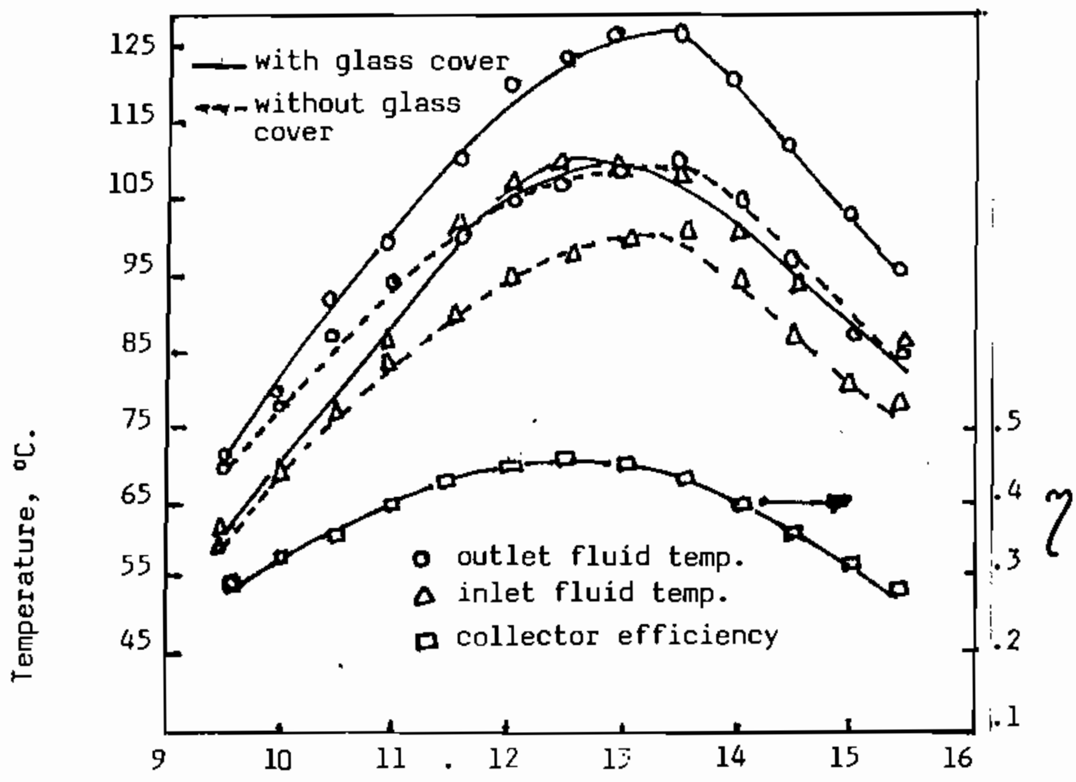


Fig. (8): Inlet, outlet fluid temperatures and collector efficiency along the day.

The dependences of the machine cooling ratio COP upon evaporation temperature (T_E), condenser temperature (T_C) and generator temperature (T_G) are shown in Figs. 9 through 13. As it can be expected, the machine COP decreases with decreasing the evaporation temperature (Figs. 9 and 10). For a given evapration temperature, an increase in condensation temperature gives a decrease in machine COP (Fig. 10), the decrease in COP is nearly the same for all evaporation temperature and generator temperatures examined. It is observed that the machine COP reaches a maximum value when generator temperature increases, the evaporator temperature increases and the condenser temperature decreases, Figs. 11 through 13. The increase in COP as the generator temperature increases is less pronounced at higher generator temperature and lower condenser temperature. It has been found that the minimum generator temperature required to operate the cooling machine is 130°C, (Fig. 14). This is why an additional heating source is necessary for these experiments. The use of such external heat source can be avoided by optimizing the solar collector unit by using the evacuated glass tube.

The effects of condenser temperature and evaporator temperature on COP are shown in Fig. 15. The results shown in this figure indicate that, increasing the evaporator temperature results in an increase in COP. The change in evaporator temperature is achieved by using a temperature control (thermostat) and changing the cooling load. The curves also indicated that, the condenser temperature have a greater influence on COP. Decreasing the condenser temperature results in markedly increases in COP.

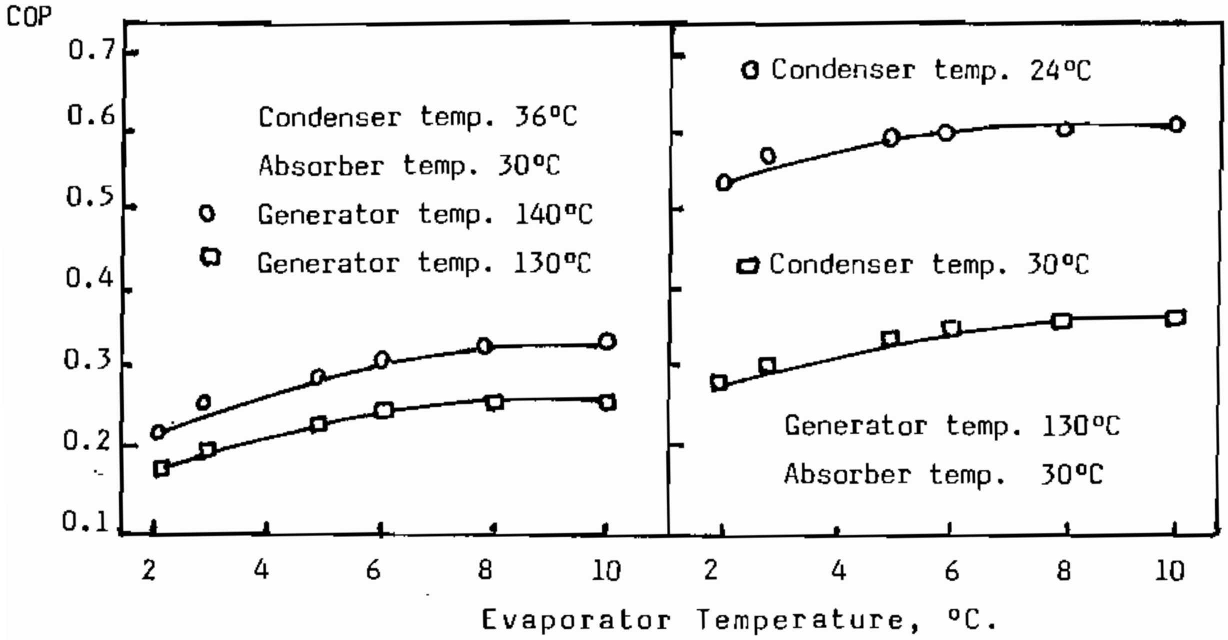
The effects of evaporator temperature and condenser temperature on system COP.as defined by equation (14) are presented in Fig. 16. The values of COP_{sys} are relatively lower than that of the machine COP. This is due to the collector thermal losses. The COP_{sys} increases as the evaporator temperature increases or the condenser temperature decreases. The increase in COP_{sys} is markedly pronouced at lower condenser temperature. The maximum value of COP_{sys} obtained during these experiments is 0.29 at condenser temperature 24°C, generator temperature 130°C and evaporator temperature 10°C.

From the forgoing results it appears that, it is possible to power the demostic refrigeration system by solar energy. From these primary results, it has been found that the optimization of such system should go through the increment of generator temperature by using an efficient solar collector and modifing the refrigeration machine to include a heat exchanger between the absorber and the generator. Certainly, more research work is needed to put the relation between these variables in an optimum form.

5. CONCLUSIONS

Based on this investigation, the following conclusions are offered.

1- The electrolux absorption refrigeration system for domestic use may be powered by solar energy. In this experiment, the maximum cooling load achieved is 101 KJ/hr, the corresponding system COP is 0.1.
2- It was approved from the experimental results that, the theoretical analysis is important for the prediction of the system



Figs. (9 & 10): Effects of Evaporator Temperature, Generator Temperature and Condenser Temperature on COP.

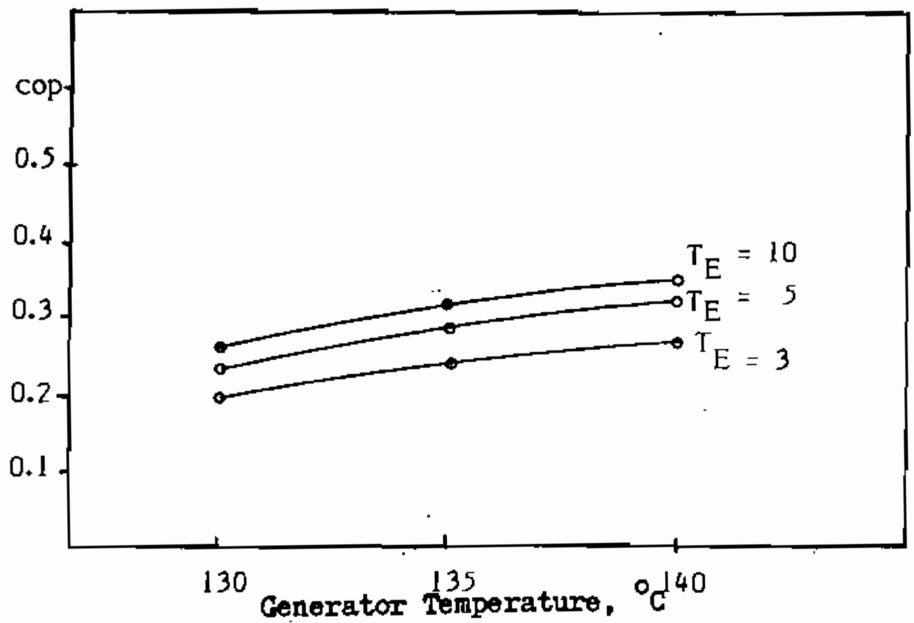
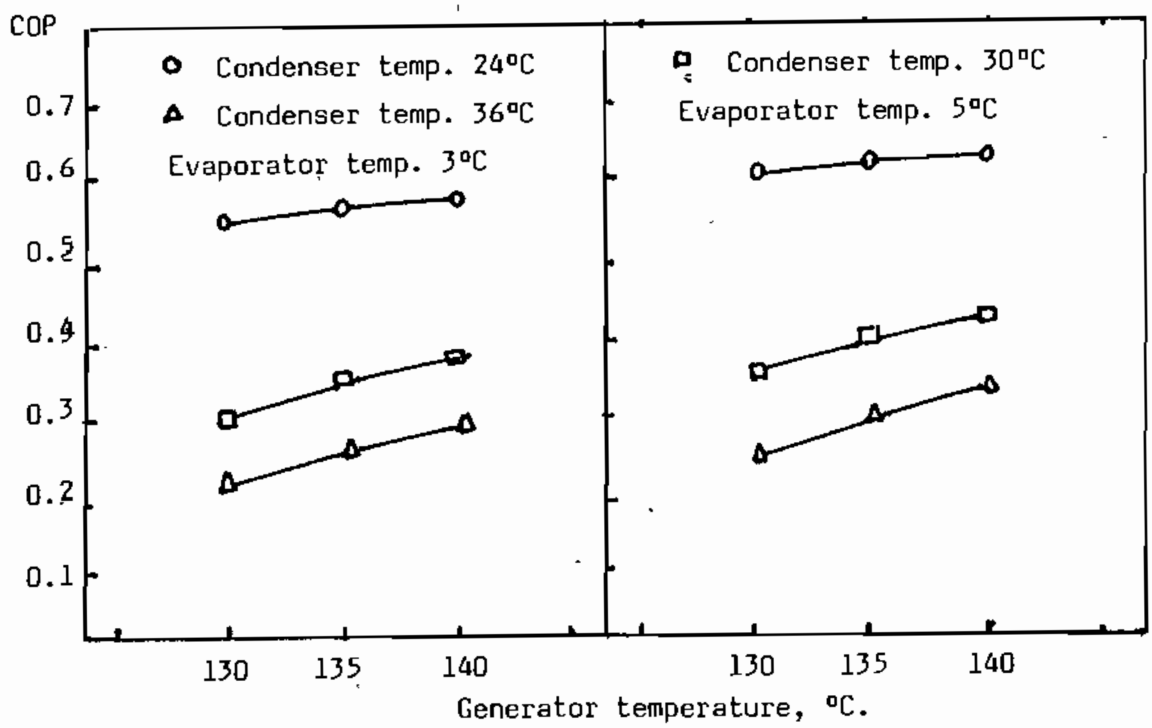


Fig. (11) Effects of Generator Temperature and Evaporator Temperature on COP



Figs. (12 through 13): Effects of Generator Temperature, Condenser Temperature and Evaporator Temperature on COP.

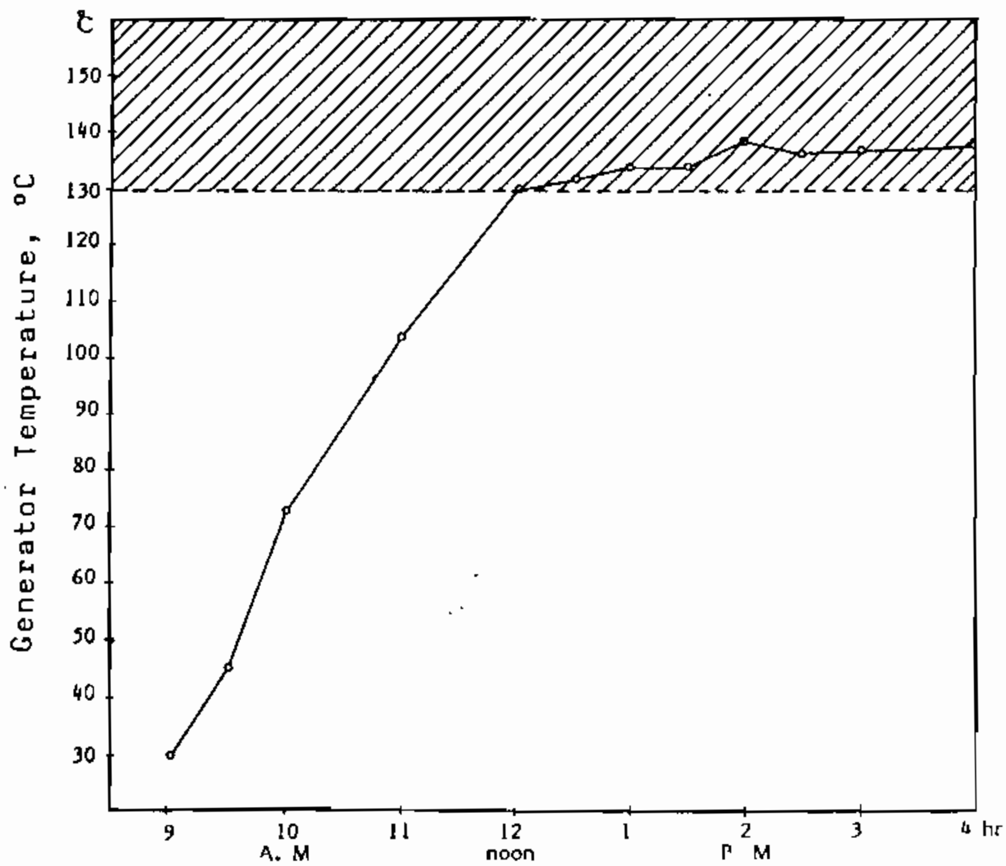


Fig. (14): Variation of Generator Temperature with Time Along a Day.

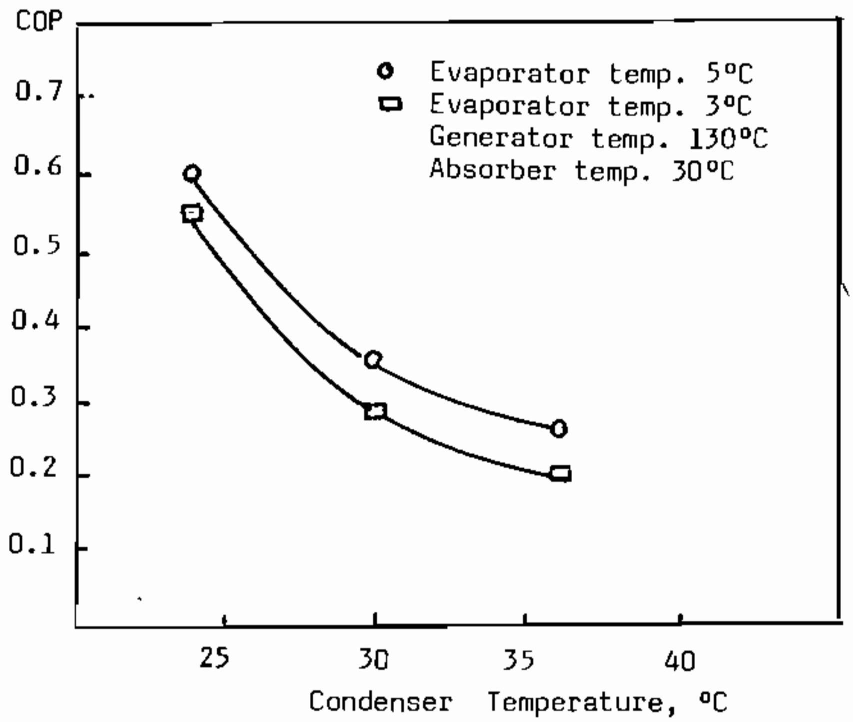


Fig. (15) Effect of Condenser Temperature and Evaporator Temperature on COP.

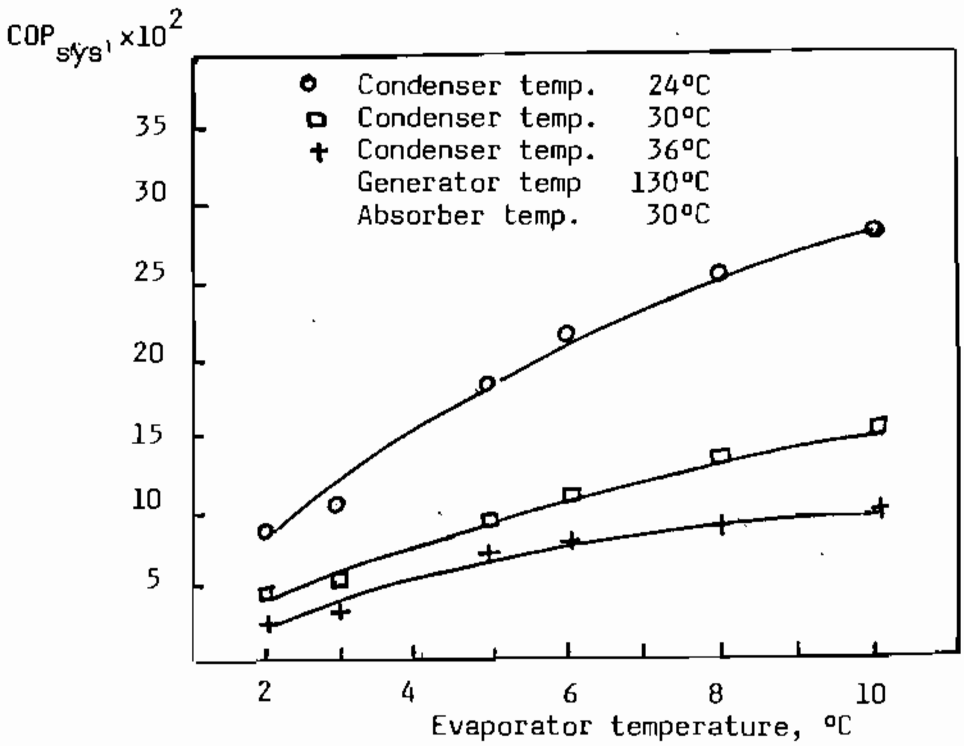


Fig. (16): Effects of Evaporator Temperature and Condenser Temperature on System COP.

performance. Since the solar collector temperature is relatively limited, a generator temperature as low as possible must be selected.

3- The energy balance shows that the auxilary systems, such as pumping use a 10% of the total energy. This fact must be taken into consideration since the system will not be fully dependent on solar energy.

6. NOMENCLATURE

A	Absorber projected area m^2	
COP	Cooling machine coefficient of performance	
COP_{sys}	Solar refrigeration system coefficient of performance	
h	Convection heat transfer coefficient, $KJ/m^2 . hr. ^oC$	
m_f	Fluid mass flow rate Kg/hr	
X	Concentration, mass of NH3 per unit mass of aqua-ammonia	
Δt_f	Increase in fluid temperature	^oC, ^oK
T_1	Evaporator temperature	^oK
T_2	Condenser temperature	^oK
T_3	Absorber temperature	^oK
T_4	Generator temperature	^oK
Q_{con}	Convection heat losses	KJ
Q_{rad}	Radiation heat losses	KJ
η	Solar collector efficiency	dimensionless
σ	Stefan Boltzmann constant	$5.66 \times 10^{-8} w/m^2 . K^4$

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