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OPTIMIZATION STUDY OF A SOLAR REFRIGERATION SYSTEM WORKS BY $LiBr$-H_2O ABSORPTION CYCLE

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To my family and beloved

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Abstract

OPTIMIZATION STUDY OF A SOLAR REFRIGERATION SYSTEM WORKS BY LiBr-WATER ABSORPTION CYCLE

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In this work, a mathematical model for solar powered LiBr-H_2O absorption refrigeration systems was treated in a computer algorithm using The Pascal programming language and optimized to minimize costs and maximize benefit cost ratio. Unit capacity and cycle temperatures were taken as the independent variables, while the benefit cost ratio was the dependent variable.

The validity of the computer mathematical model was checked by comparing the computer program results with some results of an actual experimental unit.

The study revealed an optimum condition at a system capacity of 30 kW of refrigeration and above with a maximum benefit cost ratio of 1.786. The corresponding cycle temperatures of the optimum case are Tg = 70 °C, Tc = 35 °C, and Te = 15 °C at a solar intensity I = 800 W/m^2.

1. INTRODUCTION

1.1. General

This chapter includes the literature survey, the theory of the absorption refrigeration, and the objectives of the work.

1.2. Literature Survey

Solar energy utilization for refrigeration and air conditioning purposes using the absorption cycle principle has been proposed since the late 50,s by Reti[1]. Afterwards, the problem has been subjected to a plenty of research by many other scientists who studied the performance of the solar powered absorption refrigeration systems using different working fluids like Aqua-Ammonia and Lithium Bromide-water.

In 1975, Chinnappa et al [2] studied the operation of a dual mode two stage absorption refrigeration system using the H_2O-NH_3 couple as the working fluid and the meteorological data of Moresby, Papua New Guinea. They compared this system to a similar system using LiBr-H_2O, and found that the ammonia system needs smaller solar collector area for a specified system capacity. Then in 1979, Alizadeh et al [3] conducted a theoretical study on design and optimization of a solar powered single stage absorption refrigeration system and compared its operation using both H_2O-NH_3 and LiBr-H_2O. Fixing initial conditions and system capacity their results showed that lithium bromide systems have higher cooling ratio (or coefficient of performance), simpler, and needs smaller heat exchange areas than the ammonia systems. They found that because of fixed initial conditions, and system capacity, the generator temperature is the main variable that affects the system, and the cooling ratio increases as the generator temperature increases, the only limitation is possibly the crystallization.

Due to this limitation on the generator temperature for lithium bromide systems, some researchers prefer to use the aqua-ammonia solution as the working fluid[4], and it is implied that they are more suitable than LiBr-H_2O in the areas of high solar insolation such as India[5].

In an optimization study of a solar absorption air conditioning system done by Maryland University in 1974 [6], a lower limit of the generator temperature was found to exist in which the system will not start up below this limit. It was found that the minimum generator temperature required to start LiBr-H_2O systems is lower than that required by H_2O-NH_3 systems, which implies that when the solar system is expected to produce low temperature, the lithium bromide system is preferred.

Improvements of the absorption refrigeration systems were investigated, such improvements include the implementation of multi-stage generation that gives higher coefficient of performance of the cycle[7].

Our work is a continuation in the line of research on optimization of solar powered single stage LiBr-H_2O absorption refrigeration systems. It is aimed at finding the optimum economic design of these systems when operating under the solar conditions of Amman area.

1.3. Theory

The absorption refrigeration cycle is similar to the basic compression cycle with the exception that the refrigerant vapor leaving the evaporator is absorbed by a carrier material, pumped to a high pressure, and then heated to release the vapor refrigerant to the condenser, where the weak solution (weak in refrigerant) returns through a pressure reducing valve to the absorber to be cooled and enriched again. The other elements of the refrigeration cycle namely, the condenser, the expansion valve, and the evaporator are the same for both cycles.

Two pairs of materials are commonly used in absorption refrigeration systems, these are Aqua-ammonia, and Lithium Bromide-water. In the first pair, the ammonia is the refrigerant and water is the carrier, while in the second, water is the refrigerant and LiBr which is a salt material is the carrier.

In a Lithium Bromide absorption refrigeration system, water (refrigerant) leaving the condenser as a saturated liquid at the condenser pressure is throttled through an expansion valve to the evaporator coil where it collects heat from the cooling space and leaves the coil as saturated vapor. The low pressure refrigerant vapor leaving the evaporator is absorbed by the cooled LiBr solution in the absorber. If this absorption process was executed adiabatically, the temperature of the solution would rise and eventually the absorption of vapor would cease. To maintain the continuity of the absorption process, the absorber is cooled by water or air that rejects its heat to the atmosphere.

The solution in the absorber is then pumped through the heat exchanger where it is preheated before it enters the generator. At the generator, the hot water from the collectors heats the solution. Water vapor is released from the generator and the remaining hot solution goes back to the absorber through the shell of the heat exchanger to preheat the coming solution to the generator. The hot, high pressure vapor leaving the generator goes to the condenser where it dumps the heat out to the cooling water and leaves as a saturated liquid, and the cycle repeats itself. see figure 1 .

The high and low pressures in the cycle are responsible for heat rejection at the condenser and heat absorption at the evaporator respectively. The high pressure of the cycle created by the pump is maintained fixed at the condenser and the generator by the condensation process, while the low pressure is maintained by the absorption process at the absorber. The low pressure created at the evaporator because of the absorption process is reflected inside the evaporator - since they are open to each other - and the saturated water liquid coming from the condenser is able to evaporate - due to the low pressure - and collect heat from the cold space.

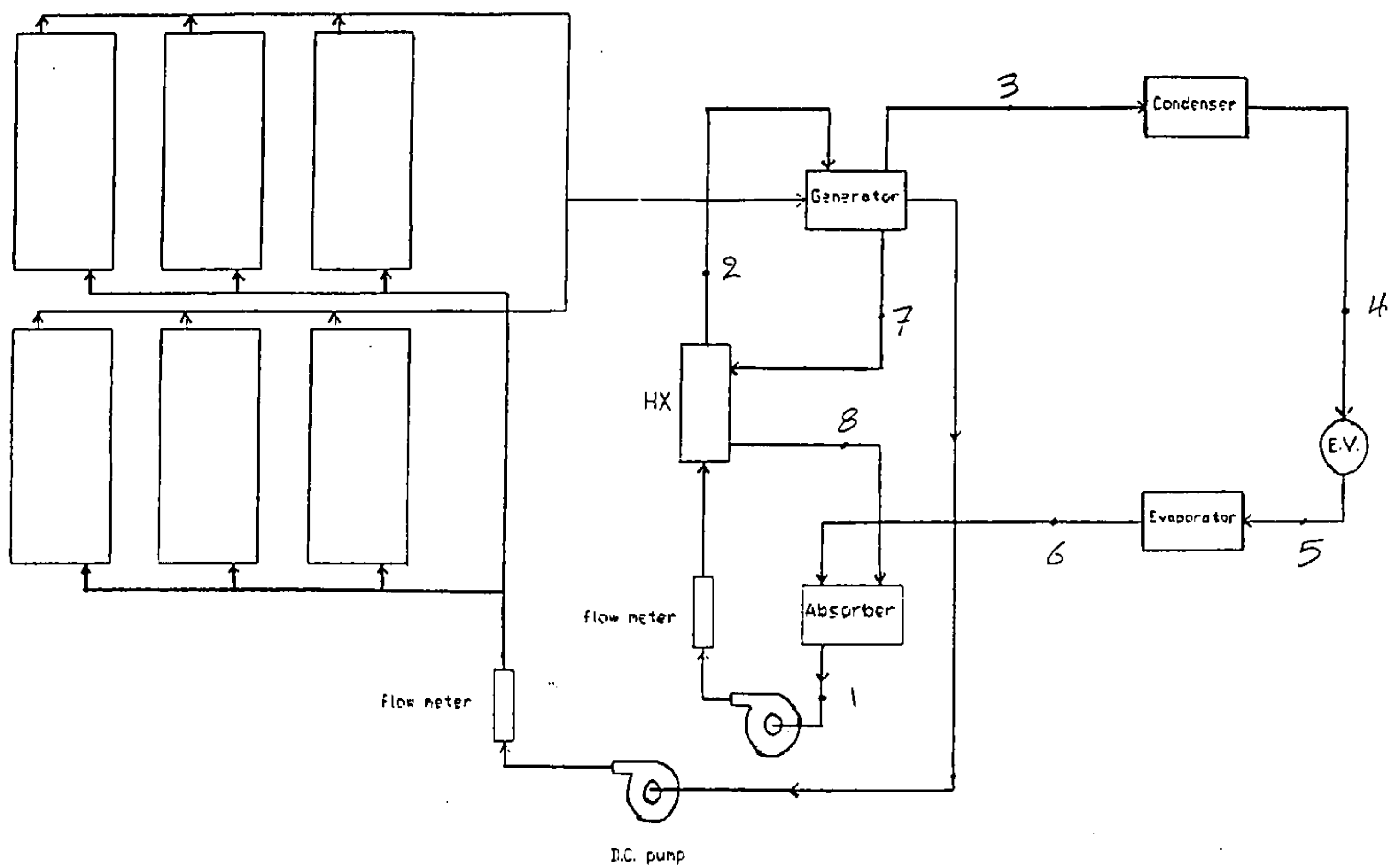


Fig. 1 : Solar Powered LiBr-H_2O Absorption Refrigeration System

The performance of the refrigeration cycle is evaluated by its coefficient of performance (COP) defined as the ratio of the refrigeration rate to the energy added to the cycle. For the absorption refrigeration cycle, the coefficient of performance then is the ratio of the refrigeration rate to the rate of heat added at the generator.

$$COP = \frac{refrigeration\ rate\ (Qe)}{heat\ added\ at\ the\ generator\ (Qg)} \qquad (1)$$

To calculate the theoretical coefficient of performance of the absorption refrigeration cycle, it is compared to an ideal vapor compression cycle receiving work from an ideal power cycle. This combined cycle is an ideal heat operated refrigeration cycle. The following figure suggests how to proceed with the analysis in which the left-hand side boxes represent the power cycle, and the right-hand side boxes represent the refrigeration cycle.

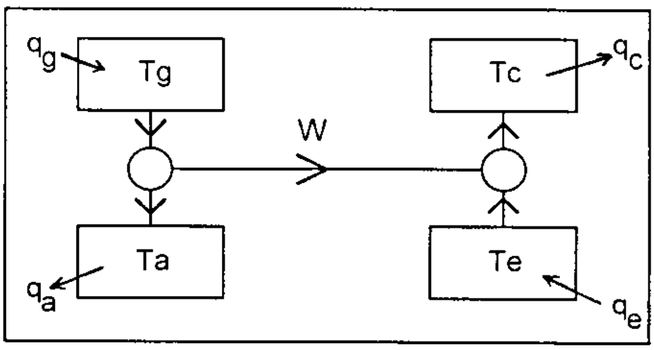


Fig. 2 : The Combined Power and Refrigeration Cycle

For the power cycle, and from Carnot analysis

$$\frac{Qg}{W} = \frac{Tg}{Tg - Ta} \qquad (2)$$

and for the refrigeration cycle

$$\frac{Qe}{W} = \frac{Te}{Tc - Te} \qquad (3)$$

then from the definition of the coefficient of performance of the absorption cycle

$$COP_{th} = \frac{Te(Tg - Ta)}{Tg(Tc - Te)} \qquad (4)$$

The actual coefficient of performance is always lower than the theoretical value because no actual reversible processes exist.

1.4. Objectives

This research studies the solar powered LiBr-H_2O absorption refrigeration systems. For a specified system capacity the effect of the thermodynamic cycle temperatures on the performance of the cycle and on the cost of the system are of great interest to this research, for the objectives of the research are to :-

1. Perform a theoretical design optimization study seeking the cycle temperatures that will result in the best economic design for some

specified capacity and the optimum design capacity, to work under the solar conditions of Amman area.

2. Build an experimental setup for a locally manufactured solar powered LiBr-H_2O absorption refrigeration system, and test it under Amman weather conditions.

The remainder of the report includes the description of the apparatus tested, the mathematical model, the economic analysis, the computer program description, the results of the research, the conclusions and recommendations, and the references.

CHAPTER II

DESCRIPTION OF THE APPARATUS

2. DESCRIPTION OF THE APPARATUS

2.1. General

The experimental setup in this study consists of the absorption refrigeration cycle components, the solar system, and the control and measurement elements. The absorption refrigeration system components are the condenser, the expansion valve, the evaporator, and the absorber, pump, heat exchanger, generator and a pressure reducing valve. The solar system components are the solar collectors, the pump, and the tanks, and the measurement and control elements are thermocouples and flow meters.

2.2. System Components

The heat exchanger and the condenser are single pass American standard, shell and tube heat exchangers with 76.2 mm (3") shell diameter and 6.35 mm (0.25") tube diameter, while the absorber and the generator are locally manufactured single pass heat exchangers with 304.8 mm (12") shell diameter and 12.7 mm (0.5") tube diameter. The evaporator is a finned tube heat exchanger with 12.7 mm (0.5") tube diameter and aluminum fins of 300/m, assembled in a duct and subjected to air flow at the inlet to the duct by a blower fan. Except for the condenser, the heat exchanger, and the blower fan, these components were manufactured and donated to the project by a local company.

The variable speed pump receives the low pressure LiBr solution from the bottom of the absorber when the shut valve between the absorber and the pump is opened during operation and delivers it at high pressure to the generator passing through the heat exchanger whose purpose is to recover part of the waste heat from the solution returning from the generator to the absorber. The solution in the generator is heated by hot water from the solar system, and the steam produced is transferred by a pipe from the top of the generator to the condenser. The steam is condensed inside the condenser and the condensate water passes through the expansion valve to the evaporator coil. At the evaporator, the water evaporates at low pressure and temperature collecting heat from the cold space and then returns to the absorber again.
Two expansion valves are used in the cycle, one at the condenser exit before the evaporator to reduce the pressure of the water, and the other is a pressure reducing valve on the return pipe from the generator to the heat exchanger to reduce the pressure of the high temperature solution in the generator before it returns to the absorber in order to keep the absorber pressure stable. The valves are globe valves with adjustable opening width.

The solar system uses a closed circulation unit, forced convection system that consists of an array of locally manufactured flat plate solar collectors, an expansion tank, and a circulating variable speed pump. The flat plate collectors are installed in two parallel rows each containing six collectors connected in parallel, and the two rows are connected in parallel too. The expansion tank capacity is 0.2 m^3 (200 L) and is responsible for supplying water to the collectors and venting air from the collector system. The pump is connected along the feed line coming from the tank, and delivers the water to the collectors.

The output hot water from this array is collected in a pipe and transferred to the generator tubes to provide for the heat input required by the cycle, then it is returned to the collector's feed line again. A shut valve is installed between the collectors' exit pipe and the generator inlet to stop hot water flow inside the generator when the water temperature is not high enough.

The absorber and the condenser are cooled by water pumped from a 1 m^3 tank. The pump delivers the cooling water at the same temperature and flow rate to the condenser and absorber tubes simultaneously, and the return water is delivered back to the tank by pipes. The cooling water in the tank is separated from the absorber and the condenser by a shut valve that is only opened during operation.

Pressure, temperature, and flow measuring devices were installed at several points of the system to monitor and control its operation. A vacuum pressure gauge along the steam pipe leaving the generator is installed to read the high pressure of the cycle, while the low pressure is known from the absorber temperature.

Temperature measurements are taken at the inlet and outlet of all the cycle components, using Copper-Constantan thermocouple wires inserted inside the system as indicated by figure 3 next. To insert the thermocouple wires inside the system, the following procedure is followed : first, Tee fittings are connected to the system pipes with the main branches along the flow line at the required positions and the secondary branch perpendicular to the flow line. Second, plug fittings are drilled to make very small holes (about 2 mm in diameter) so that the wires may be inserted. Third, a paste material is added to close the hole from the inside of the plug to prevent air leaks and to hold the wire tip fixed in position at the center of the pipe. Finally, when the paste becomes dry the plug is connected to the secondary branch.

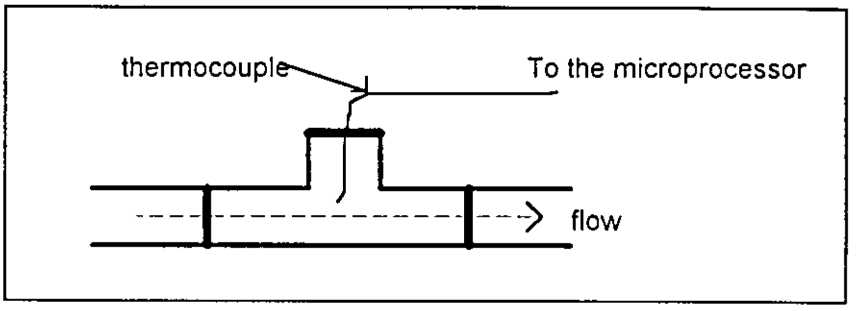


Fig. 3 : Thermocouple Installation

The copper - Constantan thermocouples are connected to a 10 channel digital microprocessor reading the temperatures in either degrees Celsius ($°C$) or Fahrenheit ($°F$).

The temperature at point 1 is the temperature of the solution leaving the absorber and the temperature of the solution leaving the heat exchanger is taken at point 2. Points 3,4,5,6 measure the temperatures of, the vapor leaving the generator to the condenser, the saturated condensate leaving the condenser, the evaporator inlet, and the evaporator exit respectively. The temperature of the weak solution leaving the generator to the heat exchanger is measured at point 7 and the its temperature at the absorber inlet is measured at point 8. The inlet and outlet temperatures of the solar array are taken at points 9 and 10 respectively.

Two flow meters are added to the system to read the rate of flow of the solution pump and the water flow rate through the solar collectors in grams/s.

All system components, pipes, and the solar system pipes were insulated using Rockwool material sheets and tubes.

2.3. Test Procedure

The solar absorption refrigeration system is initiated before operation by evacuating air from the system tubes and the heat exchangers because the system can not operate unless under vacuum and air will cause the lithium bromide to crystallize. Using a vacuum pump, the evacuation is started until the inside pressure reaches about 6.45 kPa absolute equivalent to (-28" Hg) on the gauge. The system is then charged with the lithium bromide solution at the absorber.

Operation begins by starting the solar collectors' pump to produce hot water while the generator is bypassed and the thermocouples' wires are connected to the digital microprocessor - measuring at that time water temperatures at the inlet and outlet of the solar collectors. When the water temperature at the exit of the collectors reaches 90 $°C$, the shut valves between the collectors and the generator, the solution in the absorber and the solution pump, the cooling water tank and the cooling water pump are opened and the refrigeration system starts operating.

The system is then left for half an hour until steady-state conditions are reached - indicated by stable readings of the microprocessor -, then measurements are taken and recorded every hour until the collectors' exit water temperature drops below 80 $°C$. The measurements taken are as follows :-

1. Temperatures 1 through 10.
2. Solution flow rate.
3. Ambient temperature.
4. Evaporator exit air temperature.
5. Solar intensity.
6. Cooling water inlet temperature.
7. Cooling water exit temperature from condenser.
8. Cooling water exit temperature from absorber.

When the collectors' exit temperature becomes less than 80 $°C$, the system is shut off by switching off the solution pump, the cooling water pump, the collectors pump, and closing shut valves from the absorber, the collectors, and the cooling water tank.

CHAPTER III

MATHEMATICAL MODEL

3. MATHEMATICAL MODEL

3.1. General

The absorption refrigeration cycle is governed by the thermodynamic relations of mass and heat balance across each component of the cycle. The mathematical model uses these relations to evaluate the mass flow rate and heat input or output for every component, and then calculates the coefficient of performance (COP) of the cycle. The cost estimation of the system is dependent on the cycle parameters namely, heat inputs and outputs, flow rates, and temperatures that determine the size of the system components, and cost is found to be a function of the size, and is estimated from the local market prices.

3.2. Modeling

The absorption cycle calculations start by defining the states of matter at the different points of the absorption refrigeration cycle, that is the temperature and pressure of the solution side and the water or steam side. From these states, the thermodynamic properties - specifically the enthalpy - are determined. At this stage mass and heat balance equations are applied across each component of the cycle and the results can be used in the economic analysis.

Steam tables are used to evaluate the enthalpies of the steam and water side points of the cycle according to their states, while at the solution side, the P-x-T and h-x-T diagrams for lithium bromide - water solutions provide the corresponding information. The P-x-T and the h-x-T diagrams are charts describing the pressure and the enthalpy respectively of lithium bromide water solutions as functions of the percent by mass (x) of lithium bromide .

The steam tables and the above charts are mathematically described by the general polynomial equation derived by a curve fitting program using the least square method, where the variable x, and the coefficients depend on the property in question and the temperatures at which the properties are evaluated at. The general formula is represented by the following n^{th} order equation :

$$\text{property} = a_0+a_1x+a_2x^2+ \dots\dots\dots\dots +a_nx^n \qquad (5)$$

Calculating for steam properties, the variable x in equation 5 above is the temperature. The condenser pressure is a function of the condenser temperature and the absorber pressure is a function of the evaporator temperature. The saturated vapor and saturated liquid enthalpies are also functions of the saturation temperature. The coefficients of the above general equation for steam properties appear in the following table, where the polynomial equation becomes :

$$(P,h_g,h_f) = a_0+a_1T+a_2T^2+ \dots\dots\dots\dots +a_nT^n \qquad (6)$$

Table 1
Polynomial Coefficients for Steam Pressure, Saturated Vapor Enthalpy, and Saturated Liquid Enthalpy eqn.(6)

	a_0	a_1	a_2	a_3	a_4	a_5
P	0.61634	4.162E-2	1.654E-3	2.061E-5	3.464E-7	2.478E-9
h_g	2500.802	1.90589	-1.56E-3	0	0	0
h_f	-0.11359	4.129	0	0	0	0

derived from saturated steam tables

The solution properties namely the percent by mass (x) and the enthalpy (h) are also simulated by polynomial equations derived from the P-X-T and the h-x-T diagrams respectively. Because these properties are functions of two variables, each property is described by a set of polynomial equations which are functions of one variable, and each equation is applied at a certain value of the second variable. If the value of the second variable does not have a describing equation, then interpolation between available values is used to calculate the property of the solution. For the case of the percent by mass property (x) which is a function of the pressure and the temperature, each polynomial is derived at a certain temperature and is a function of pressure. While for the case of enthalpy (h) which is a function of the percent by mass of lithium bromide in the solution and the temperature, each polynomial is derived at a certain temperature and is a function of (x). The coefficients of the polynomials are listed in the following tables, where

$$x = a_0 + a_1 P + a_2 P^2 + \ldots\ldots\ldots + a_n P^n \qquad (7)$$

$$h = a_0 + a_1 x + a_2 x^2 + \ldots\ldots\ldots + a_n x^n \qquad (8)$$

For the state points of the absorption cycle needed to complete the calculations indicated by numbers on the schematic diagram of the system and described within the experimental setup, the calculation procedure is as follows :-

3.2.1. Input

The input data to start calculations of the thermodynamic cycle are the temperatures in degrees Celsius ($°C$) of the generator (T_g), the condenser (T_c), the evaporator (T_e), the absorber (T_a), the heat exchanger ($T_{\_hx}$), and the capacity of the system to be designed (Q_e) in kW.

3.2.2. Constant Data

To complete heat exchanger area calculations and cost of the system, other constant data are considered, these are :-

- The specific heat of air, $Cp_{\_air}$ = 1.0035 kJ/kg
- Heat exchangers' overall heat transfer coefficient, U = 1000 W/m^2
- Evaporator overall heat transfer coefficient, $U_{\_ev}$ = 200 W/m^2
- Solar Intensity, I = 800 W/m^2
- Ambient temperature, $T_{amb} = 30\ °C$
- Cooling water inlet temperature, $T_{cwi} = 20\ °C$
- Operating hours per day, hours = 8 (100 days/yr)
- Coefficient of performance of a conventional compression system , $COP_{\_comp} = 3$
- Life of the absorption refrigeration system = 20 years.

3.2.3. Thermodynamic Relations

The condenser pressure is the saturation pressure at the condenser temperature and the absorber pressure is the saturation pressure at the evaporator temperature. These values are found in the steam tables, and are calculated using the pressure coefficients of table (1) in the polynomial equation (5) :

$$P_c = P_{sat}(T_c) = P(T_c) \qquad (9)$$

$$P_a = P_{sat}(T_e) = P(T_e) \qquad (10)$$

The percent by mass of LiBr at the absorber and the generator are found using the P-x-T diagram at the corresponding states. Their values are calculated using the coefficients of table (2) at the corresponding temperature in the polynomial equation (5) with the property being the percent by mass and the variable being the pressure :

$$x_1 = \text{P-x-T}(P_a, T_a) \qquad (11)$$

$$x_7 = \text{P-x-T}(P_c, T_g) \qquad (12)$$

From mass balance at the generator, the flow rates of the return solution and the vapor inside the cycle are found respectively as follows :

$$w_7 = \frac{w_1 \times x_1}{x_7} \qquad (13)$$

$$w_3 = w_1 - w_7 \qquad (14)$$

The enthalpies at the points of the cycle are now determined by the state of temperature and pressure of each point. These calculations are done using the coefficients of table (3) at the corresponding temperatures of the points in the polynomial equation (5) with the property being (h) and the variable x being the percent by mass (x)

$$h_1 = h\text{-}x\text{-}T(T_a, x_1) \qquad (15)$$

$$h_2 = h\text{-}x\text{-}T(T\_{hx}, x_1) \qquad (16)$$

$$h_7 = h\text{-}x\text{-}T(T_g, x_7) \qquad (17)$$

$$h_3 = h_g(T_g) \qquad (18)$$

$$h_4 = h_f(T_c) \qquad (19)$$

$$h_6 = h_g(T_e) \qquad (20)$$

The above values being for steam are found using the coefficients of enthalpy in table (1). Now the heat balance equations are applied at the evaporator, the condenser, the heat exchanger, the absorber, and the generator respectively, then the coefficient of performance is found by equation (27).

$$Q_e = w_3*(h_6-h_4) \qquad (21)$$

$$Q_c = w_3*(h_3-h_4) \qquad (22)$$

$$Q\_{hx} = w_1*(h_2-h_1) \qquad (23)$$

$$h_8 = h_7 - Q\_{hx}/w_7 \qquad (24)$$

$$Q_a = w_3*h_6 + w_7*h_8 - w_1*h_1 \qquad (25)$$

$$Q_g = w_3*h_3 + w_7*h_7 - w_1*h_2 \qquad (26)$$

$$COP = Q_e/Q_g \qquad (27)$$

The calculation procedure described above is detailed in reference [8].

From the heat requirement of each heat exchanger the required area is found and the cost is determined from the market prices. The three governing equations are the standard equations of heat transfer for any heat exchanger :

$$Q = m_h Cp_h (T_{,hi} - T_{,ho}) \qquad (28)$$

$$Q = m_c Cp_c (T_{,co} - T_{,ci}) \qquad (29)$$

$$Q = U.A.LMTD \qquad (30)$$

for a counter flow heat exchanger

$$LMTD = ((T_{,ho} - T_{,ci}) - (T_{,hi} - T_{,co})) / \ln((T_{,ho} - T_{,ci}) / (T_{,hi} - T_{,co})) \qquad (31)$$

applying the above relations to each heat exchanger gives the required area of heat transfer, then the cost of the heat exchangers known to be a function of the heat transfer area is estimated from local market prices by the following equations :

$$heat\ \ exchanger\,(\mathrm{t\,ype\,1})\,cost = 700 \times A \qquad (32)$$

$$heat\ \ exchanger\,(\mathrm{t\,ype\,2})\,cost = 100 \times A \qquad (33)$$

where the first equation is used for shell and tube heat exchangers, and the second equation is used for finned tube heat exchangers. The cost is estimated in Jordanian Dinars (JD).

4. ECONOMIC ANALYSIS

4.1. General

The economic analysis of solar energy utilization projects always involve comparison between the proposed solar system and some conventional system utilizing fuel energy and providing the same output as that of the solar system. In this sense, the common costs of the two systems need not be taken into consideration and only other costs are compared since the common costs will neutralize for both systems.

In this research, common costs between the solar powered system and the conventional compression system are eliminated, and the additional costs projected by the solar system are assumed to constitute the investment required by the solar system. The next section of the chapter illustrates the economic analysis of proposed solar powered absorption system.

4.2. LiBr-H_2O Refrigeration System Analysis

The economic analysis is based on comparing the electrical energy savings of using the solar powered absorption refrigeration system instead of a conventional compression refrigeration system with the same capacity and comparing it with the additional cost projected by implementing the solar system. In this sense, the electrical energy savings is calculated by dividing the refrigeration effect of the solar system by the coefficient of performance of the compression system and multiplying the result by the price of the energy unit (kwh).

The annual saving is estimated as follows :

$$S = (\frac{Qe}{COP\_comp} \times yearly\ operating\ hours) \times energy\ unit\ price \qquad (33)$$
$$= (\ electrical\ \ load\ \ saved\ ) \times energy\ unit\ price$$

The cost of the conventional compression system is a function of the refrigeration effect Qe and is estimated from local market prices as

$$cost\_comp = 284.57 \times Qe \qquad (34)$$

then the additional cost of the system is

additional cost = total cost of solar system - conventional system cost

To calculate the benefit cost ratio, the additional cost is projected over the life of the project. In this analysis, the life of the system is considered as 20 years and the annual market discount rate is 8%.
then, the benefit cost ratio is

$$B/C = \frac{S}{annual\ additional\ cost} \qquad (35)$$

5. COMPUTER PROGRAM

A computer program is made to perform the cycle calculations and cost estimation. Based on the results, the program will start the optimization procedure by monitoring the effect of changing cycle parameters on its cost and on the expected savings from the system. At the maximum cost benefit ratio the design is optimized. It is noteworthy to say here that the optimization procedure considered is based on economical priorities not thermodynamic.

The programming language used is the Pascal, and the software is TurboPascal version 5.5 working on PC machines.

A successive substitution unidirectional loop search method was used in the computer program to determine the optimum design case by which, the variables of the system namely temperatures, were changed successively one at a time holding the other variables constant and performing the system analysis for each case. A record of the maximum benefit cost ratio value and the maximum coefficient of performance value was kept, and by the end of the program, these values defined the optimization results.

The computer program was designed to perform the calculations for a defined range of values of the independent variables of the system. these range values were selected to cover the commonly observable temperatures in practical solar powered LiBr-H_2O absorption refrigeration systems.

Starting from a first trial case of 65 $°C$ generator temperature, 25 $°C$ absorber temperature, 30 $°C$ condenser temperature, and 5 $°C$ evaporator temperature, the range values of the cycle temperatures are as follows :

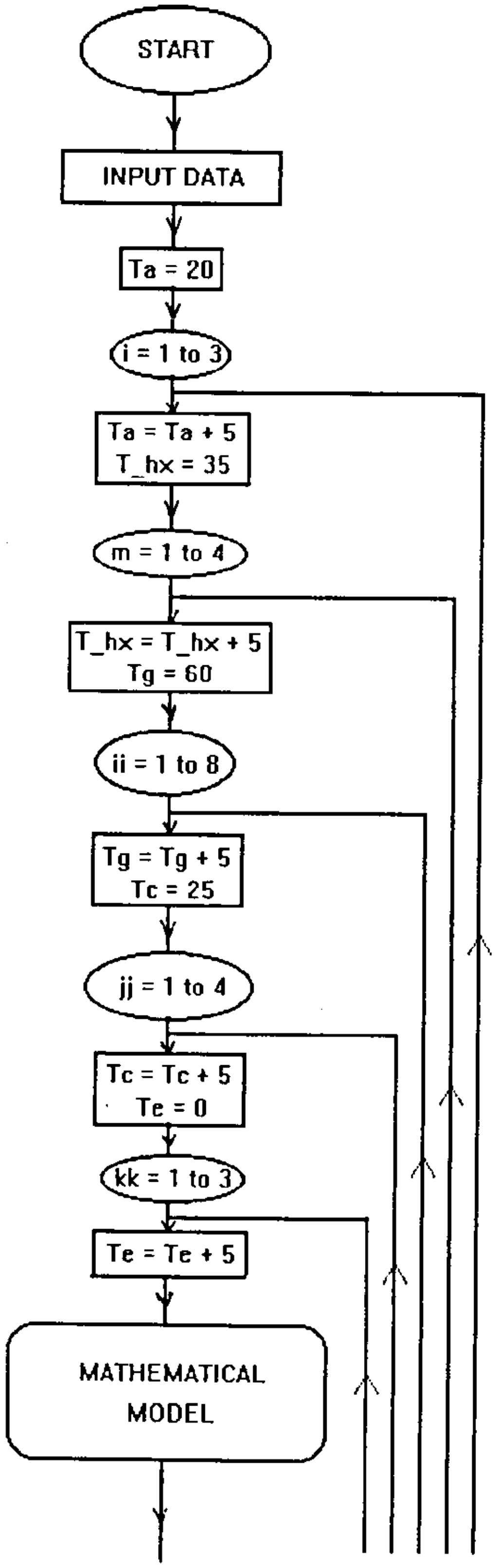
1. Generator temperature (65-100) $°C$
2. Absorber temperature (25-35) $°C$
3. Condenser temperature (30-45) $°C$
4. Evaporator temperature (5-15) $°C$

with each variable changed in 5 $°C$ steps within its range values.

Using these values, the computer program proceeds by calculating the state of matter at each point of the refrigeration cycle in order to use them in calculating the enthalpies. The mass and energy balance equations are applied by the program afterwards, and the heat input and output to and from the system are evaluated. Using the last results, the coefficient of performance is calculated, the cost and saving of the system is estimated, and the benefit cost ratio is finally computed.

This procedure is repeated by the computer program for all the range values until the optimum condition is determined.

The following diagram is a flow chart of the computer program illustrating the input stage, the sequence of the calculation procedure, and the output stage. A list of the main program and the subprograms appear in the appendix.


START
INPUT DATA
Ta = 20
i = 1 to 3
Ta = Ta + 5
T_hx = 35
m = 1 to 4
T_hx = T_hx + 5
Tg = 60
ii = 1 to 8
Tg = Tg + 5
Tc = 25
jj = 1 to 4
Tc = Tc + 5
Te = 0
kk = 1 to 3
Te = Te + 5
MATHEMATICAL
MODEL

6. RESULTS

6.1. General

In this chapter, the interrelationships between the parameters of the solar powered LiBr-H_2O absorption refrigeration system are investigated thoroughly. In the context, the effect of the generator, condenser, absorber, and evaporator temperatures on the coefficient of performance of the system is observed by changing one variable at a time while holding the other variables constant. This procedure is then repeated for different settings of the system to ensure the homogeneity of the results obtained.

Moreover, the theoretical study also investigates the effect of the system parameters on the investment cost required by the solar absorption system. This has been done by observing the effect of the generator and evaporator temperatures on the areas of the heat exchangers in the system and yet the cost of the system.

The system parameters were also found to have direct impact on the economic indicators such as the benefit cost ratio. This study was able to relate the benefit cost ratio to the generator, condenser, and evaporator temperatures. The economic optimization procedure will investigate this effect and reach the state that maximizes the benefit cost ratio for any specified design capacity.

Finally using the cycle parameters reached by the optimization process, the design is optimized in terms of system capacity by finding the capacity at which the benefit cost ratio is maximum where that will be the optimum for solar powered LiBr-H_2O absorption refrigeration systems working under the solar conditions of Amman area.

6.2. Coefficient of Performance

Figures 4 through 8 illustrate the relationships between the cycle coefficient of performance and the generator temperature at different system temperatures. These figures show that the pattern at which the *COP* depends on the generator temperature is always the same. Its value however, changes with the system temperatures.

It is noticed that for every specific system settings, there exists a lower boundary of the generator temperature below which the cycle will not operate. Above this temperature limit, as the generator temperature increases the coefficient of performance of the cycle increases sharply until it reaches a maximum value, decreases slightly afterwards up to the crystallization point. This result is in compliance with the results of the optimization study done by the university of Maryland[6]. These figures are shown below.

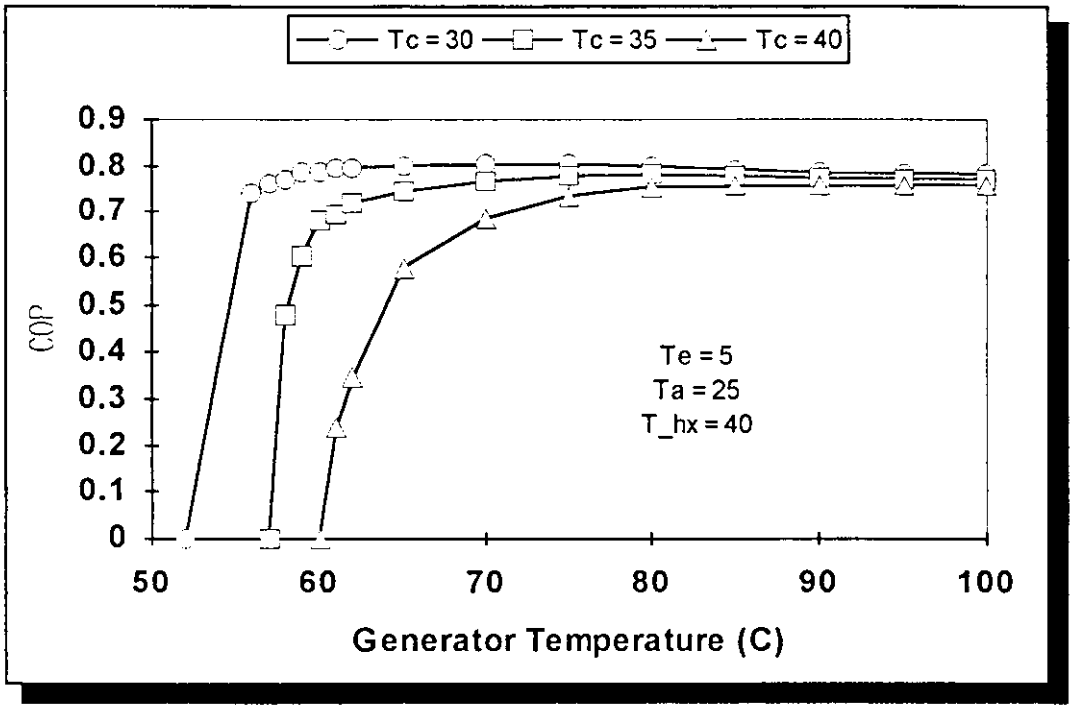


Fig.4: COP vs Generator Temperature.

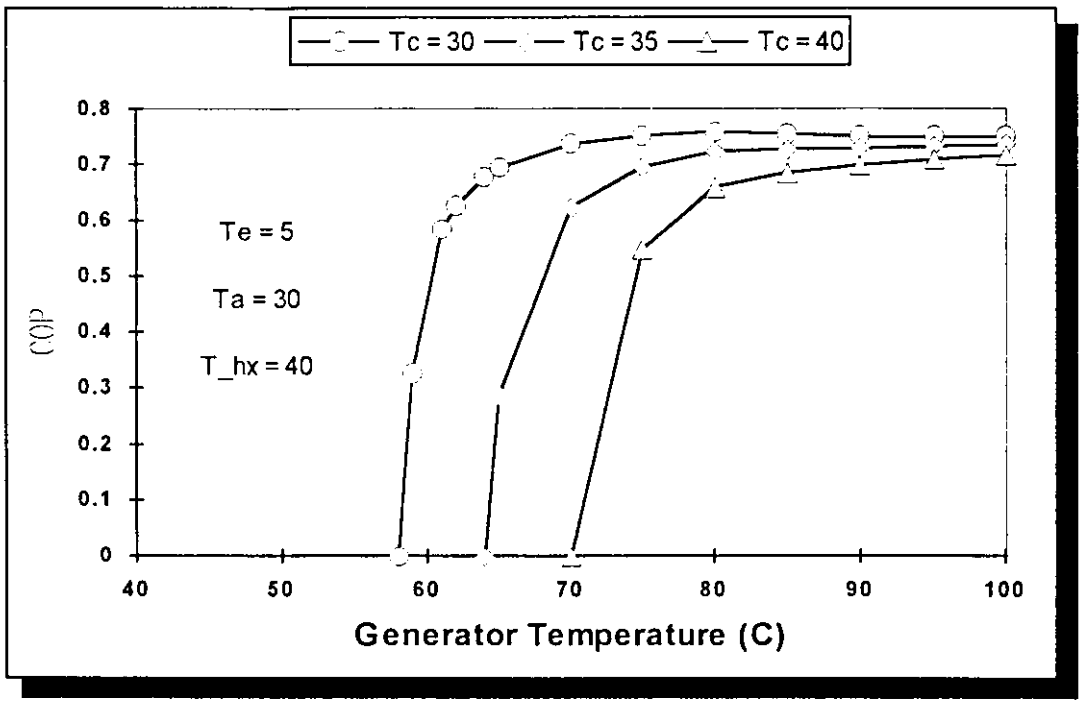


Fig.5: COP vs Generator Temperature.

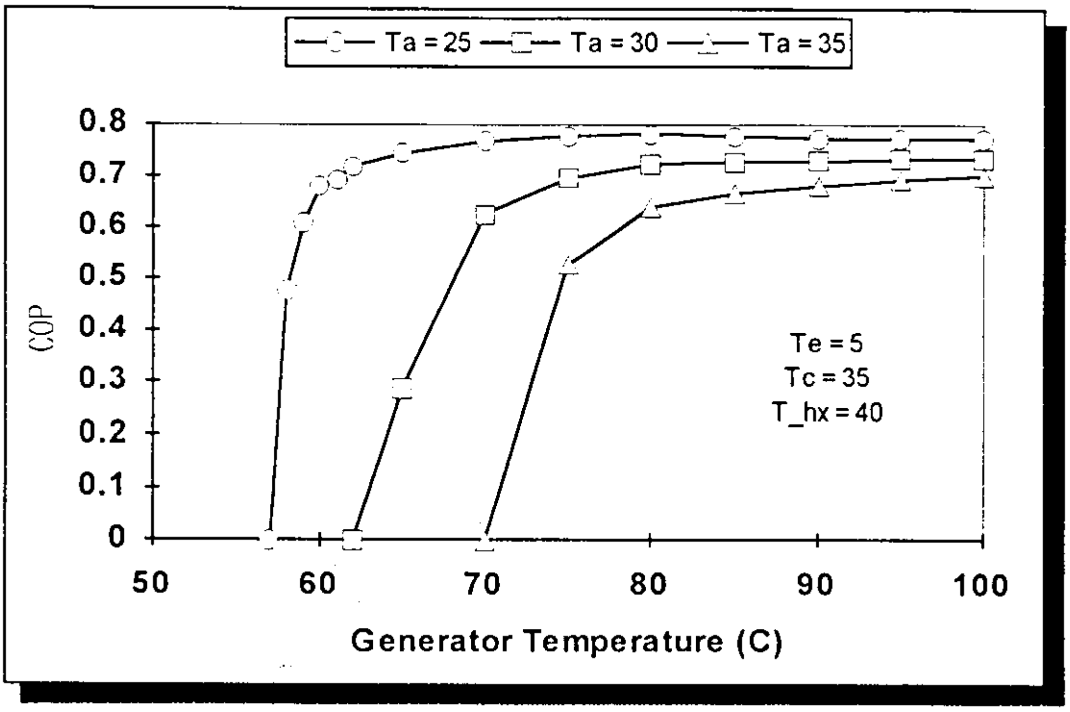


Fig.6: COP vs Generator Temperature.

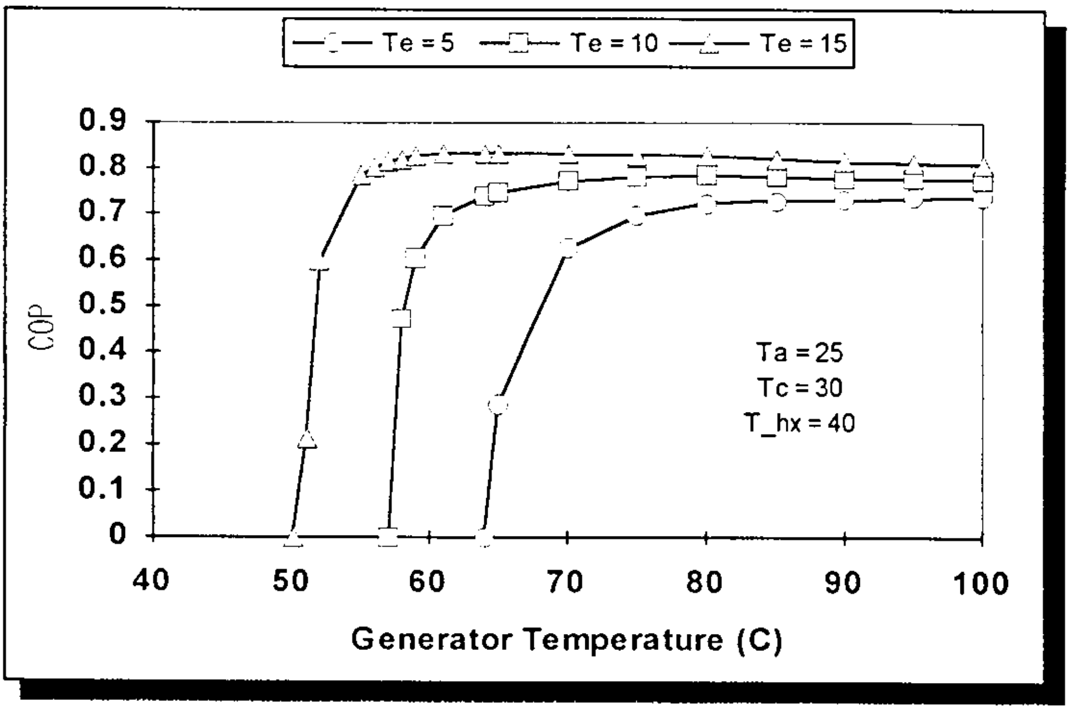


Fig.7: COP vs Generator Temperature.

The above figures also show that although the general trend of the coefficient of performance dependency on the generator temperature is always maintained, changing the condenser, absorber, or evaporator temperature will affect the value of the coefficient of performance. The relations between the cycle coefficient of performance and the condenser, absorber, and evaporator temperatures are shown in the following figures 8,9, and 10.

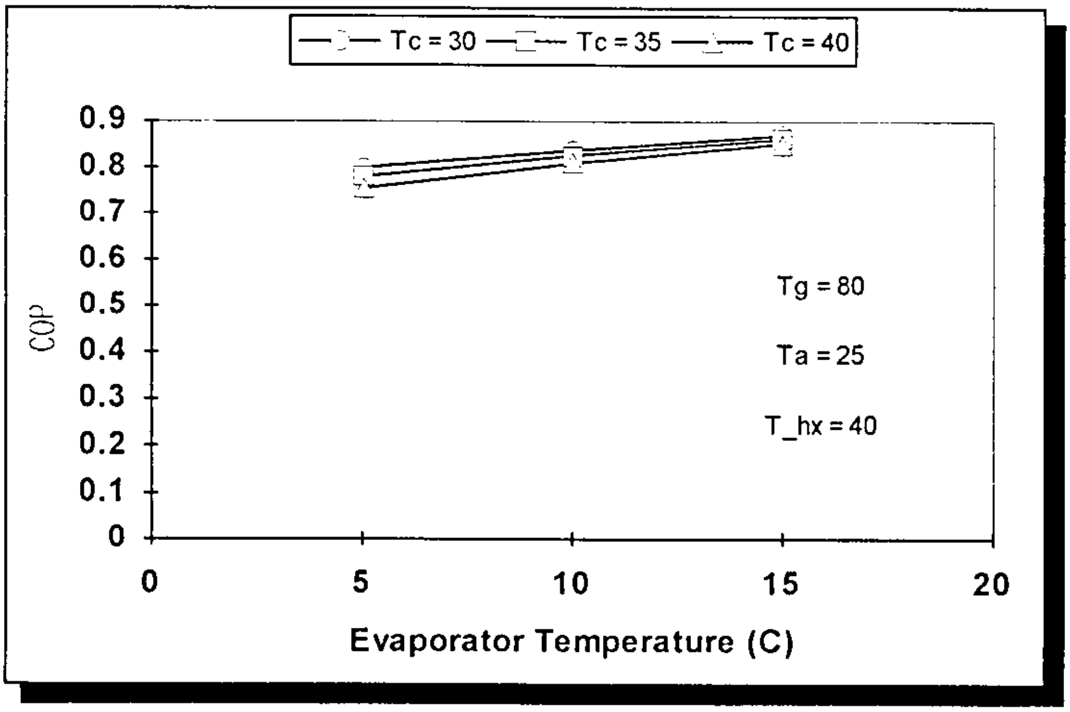


Fig.8: COP vs Evaporator Temperature.

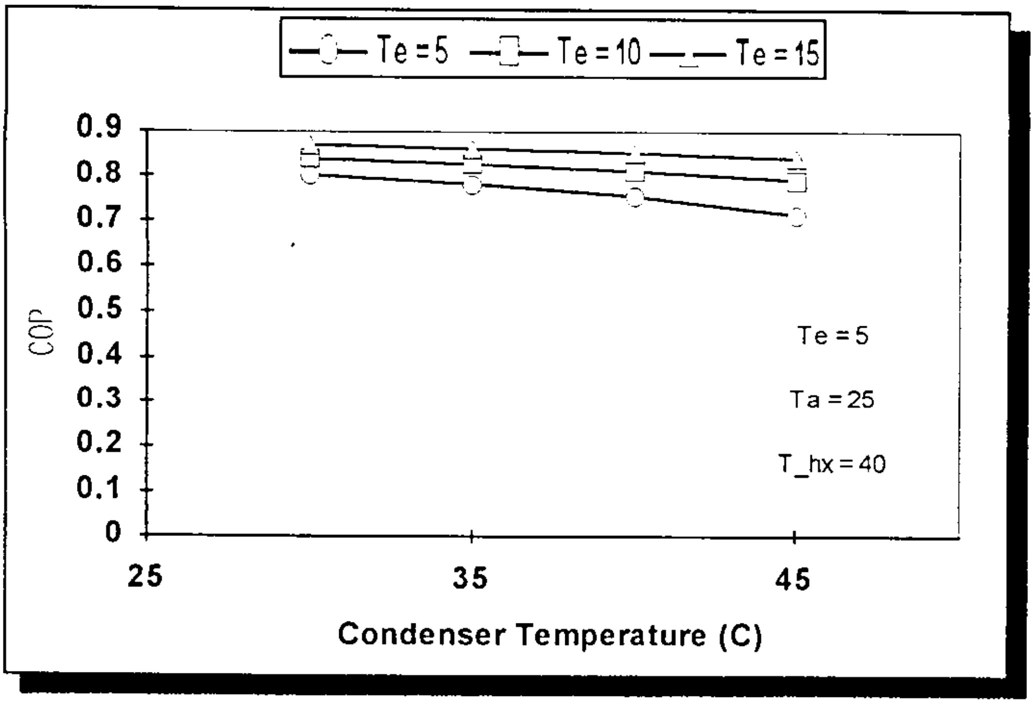


Fig.9: COP vs Condenser Temperature.

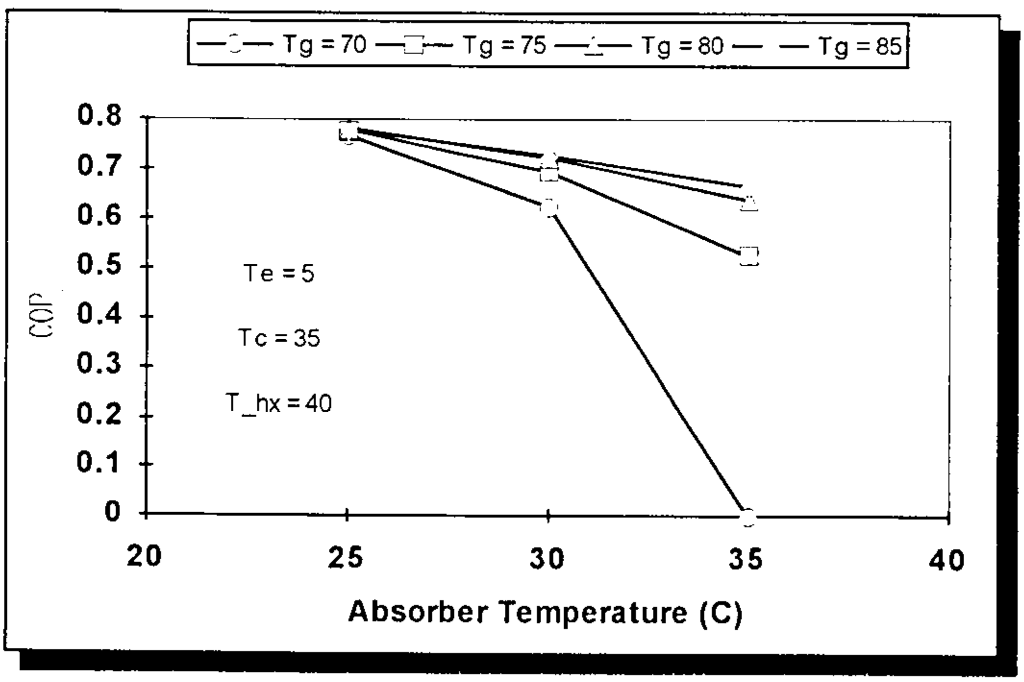


Fig.10: COP vs Absorber Temperature.

These figures indicate that increasing the absorber or the condenser temperatures causes a slight decrease in the coefficient of performance, whilst an increase in the evaporator temperature leads to a rise in the coefficient of performance. The effect of the absorber temperature is felt very high at low generator temperatures as shown in figure 10 at Tg = 70 $°C$.

An important relation reached in the analysis of this study was that the cycle coefficient of performance is independent of the system capacity and totally dependent on the cycle temperatures only. Figure 11 below illustrates this result. The optimum cycle coefficient of performance is also independent of the system capacity.

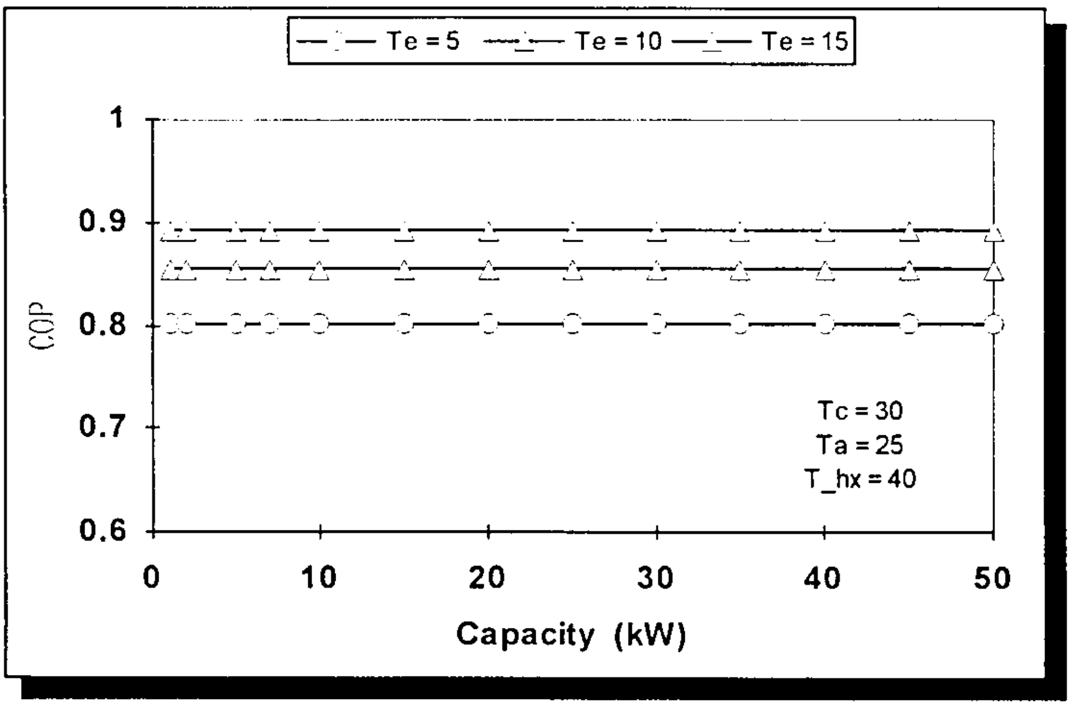


Fig.11: Coefficient of Performance vs System Capacity.

This result is expected since the capacity of the system is controlled by the refrigerant flow rate allowed into the evaporator at any temperatures desired.

6.3. Heat Exchanger Areas

Figures 12 through 15 below illustrate the relations between the evaporator and the generator temperatures and their heat transfer required areas. In figure 12 below, the evaporator area is plotted against the generator temperature for different evaporator temperatures. It is seen that the evaporator area is independent of the generator temperature and is a function of the evaporator temperature only.

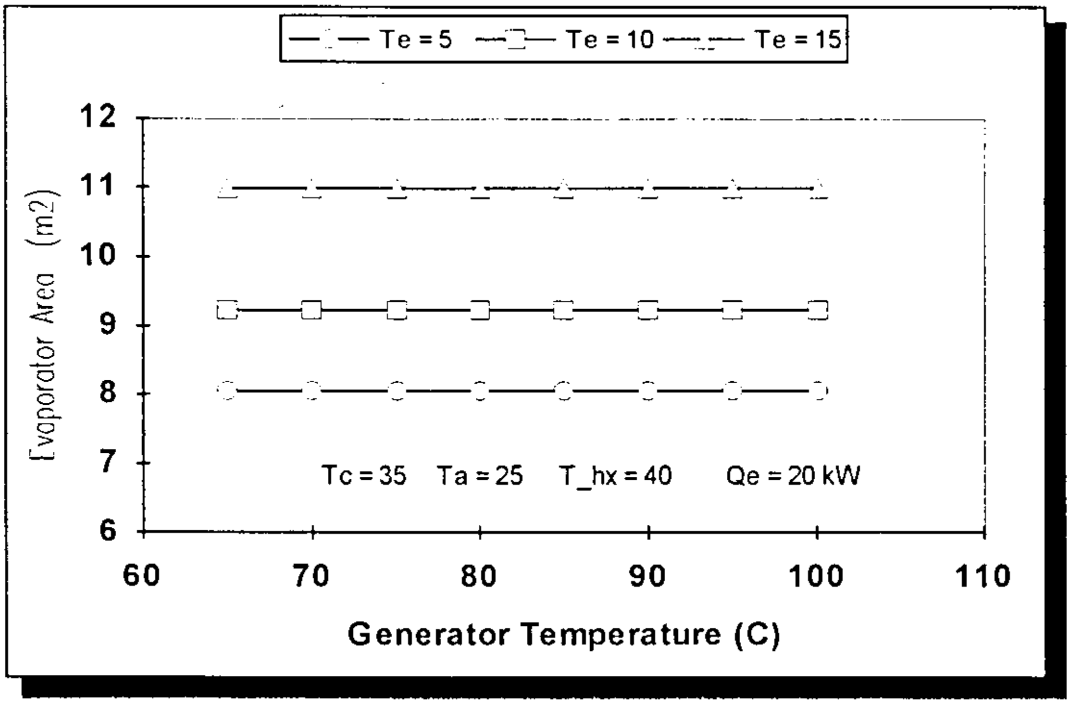


Fig.12: Evaporator Area vs Generator Temperature.

Figure 13 next shows the relationship between the generator area and the evaporator temperature. It is observed that the required generator area decreases as the evaporator temperature increases.

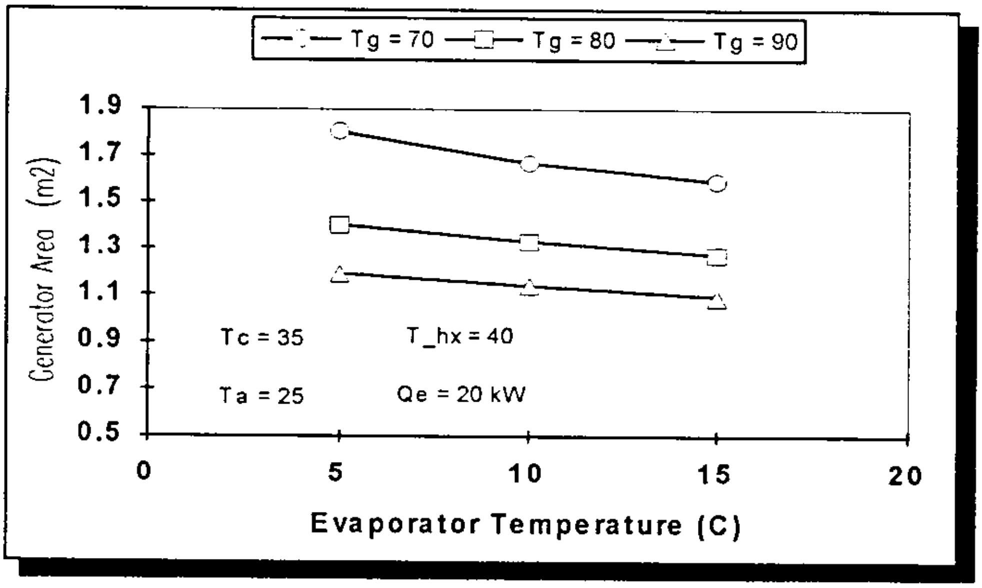


Fig.13: Generator Area vs Evaporator Temperature.

Figures 12,13 above being plotted at different system settings also indicate the relationships between the evaporator and the generator areas and their temperatures respectively. These relations are shown in the following figures.

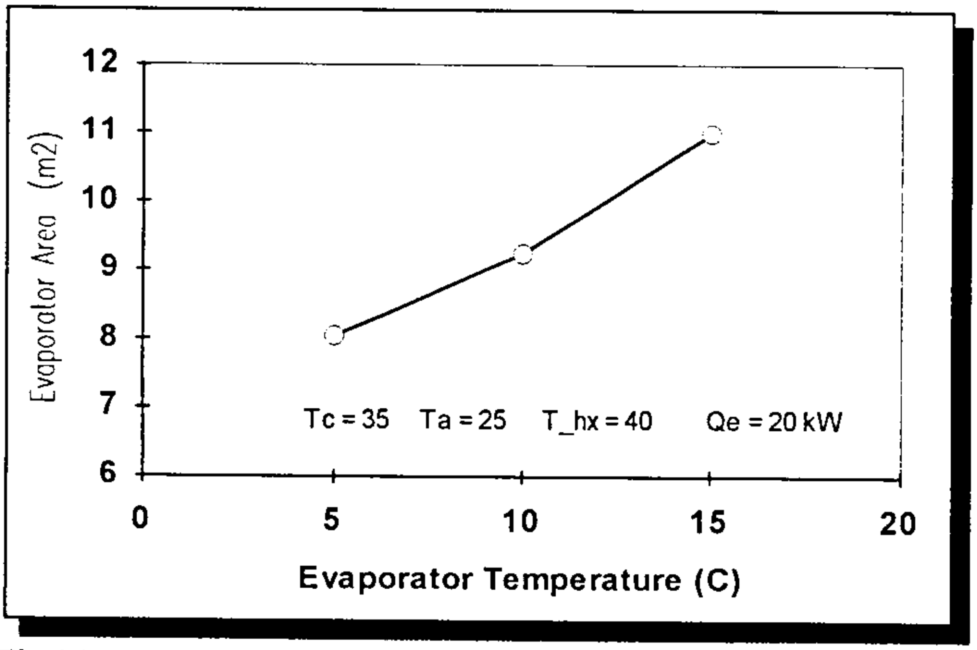


Fig.14: Evaporator Area vs Evaporator Temperature.

In figure 14 above, the evaporator area is seen to increase as its temperature increases at constant system capacity. This proportionality is explained by the fact that the refrigeration effect in the evaporator is a function of the latent heat of evaporation of the refrigerant. As the evaporation temperature increases, the latent heat of evaporation decreases, then for fixed refrigeration capacity, the decrease in the latent heat is compensated for by the increase in the evaporator area.

In figure 15, the generator area is plotted against its temperature, and the area is found to decrease as the temperature increases. This is expected since the required heat rate delivered by the generator will increase with increasing generator temperature, and yet the required heat transfer area for fixed capacity will decrease.

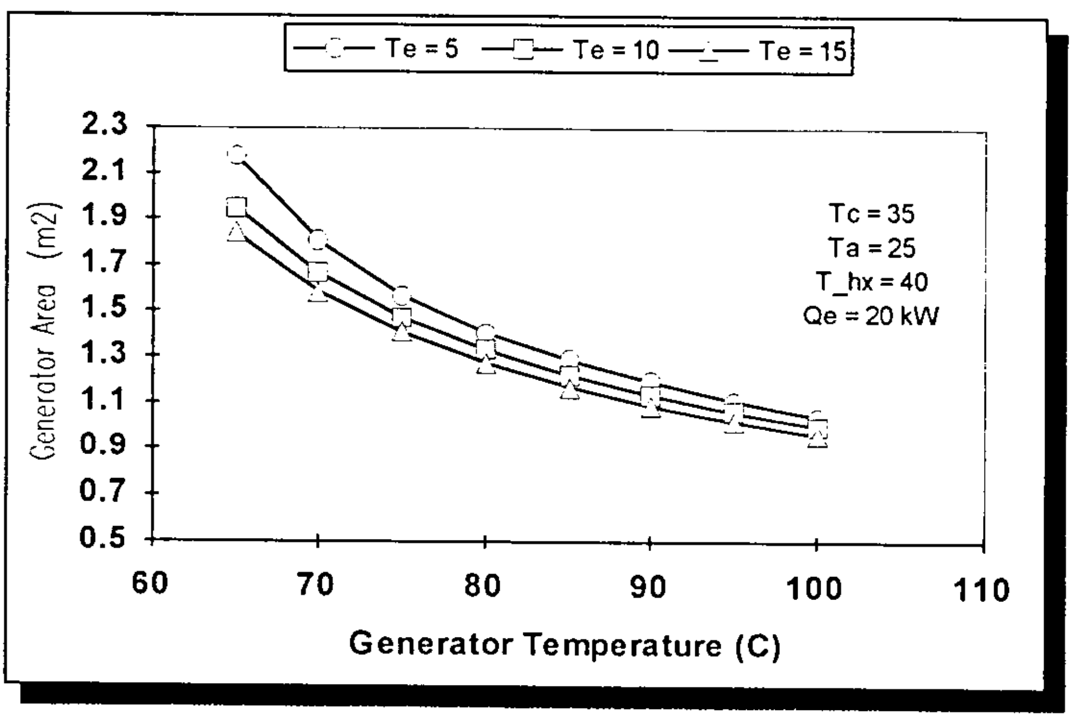


Fig.15: Generator Area vs Generator Temperature.

6.4. Benefit Cost Ratio

Figures 16 through 18 show the dependency of the benefit cost ratio on the generator, condenser, and evaporator temperatures. It is seen from figures 16, 17 that in general, as the generator temperature increases the benefit cost ratio increases to a maximum value and then decreases again with only small effect of the condenser and evaporator temperatures on the value of the maximum benefit cost ratio. The maximum benefit cost ratio reached in this study is 1.786 at the condition of, Tg = 70 °C, Tc = 35 °C, and Te = 15 °C at a solar intensity I = 800 W/m^2.

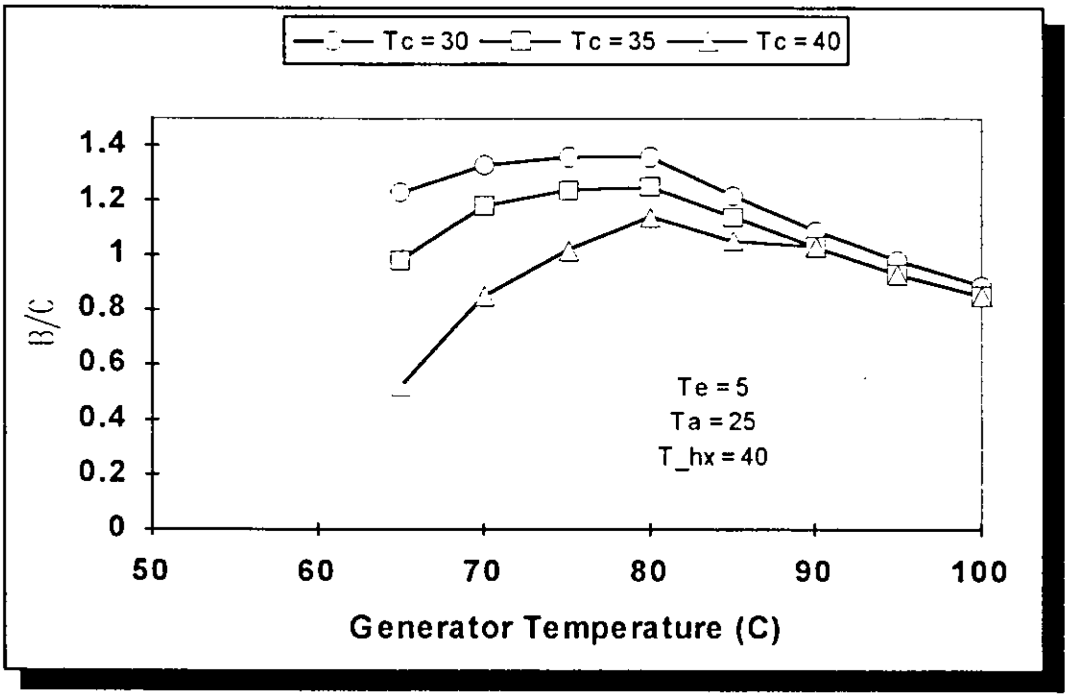


Fig.16: Benefit Cost Ratio vs Generator Temperature.

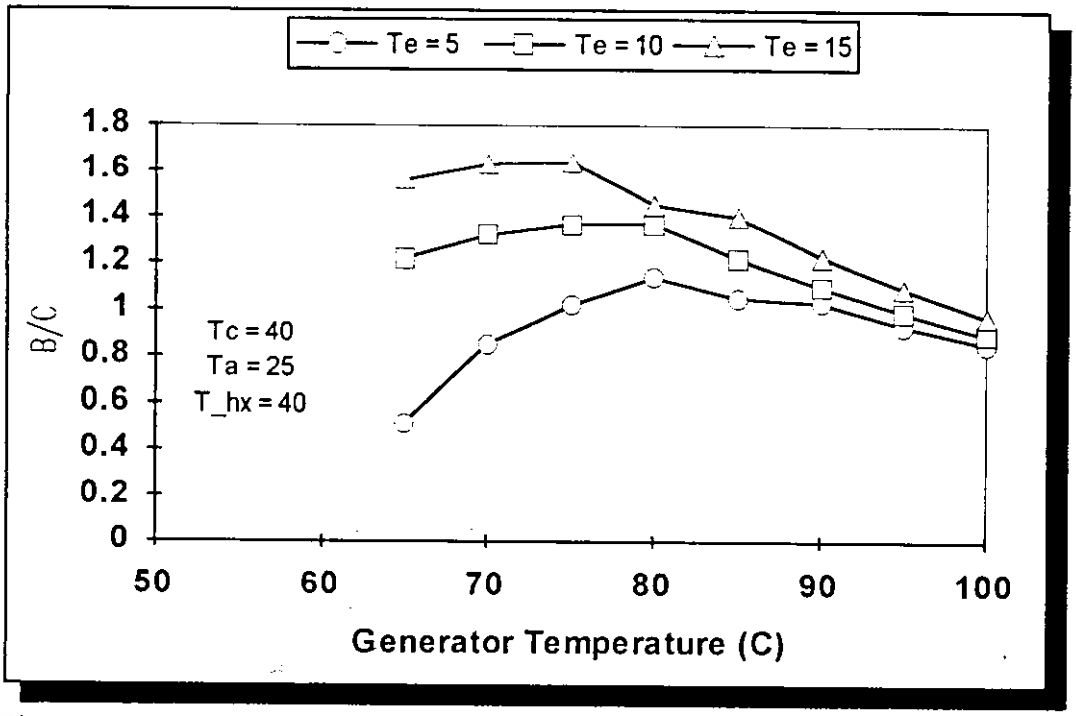


Fig.17: Benefit Cost Ratio vs Generator Temperature.

The behavior of the benefit cost ratio with the condenser and the evaporator temperatures shown in figure 18 is different, and the relation between the benefit cost ratio and these temperatures is always proportional. It is clearly seen from figure 18 and the preceeding figures 16, 17 that the benefit cost ratio increases as a result of condenser temperature decrease and/or evaporator temperature increase. The slight effect of the condenser/absorber temperatures on the optimum benefit cost ratio is noticed, and the maximum benefit cost ratio of 1.786 is reached at any condenser temperature at an evaporator temperature of 15 $°C$.

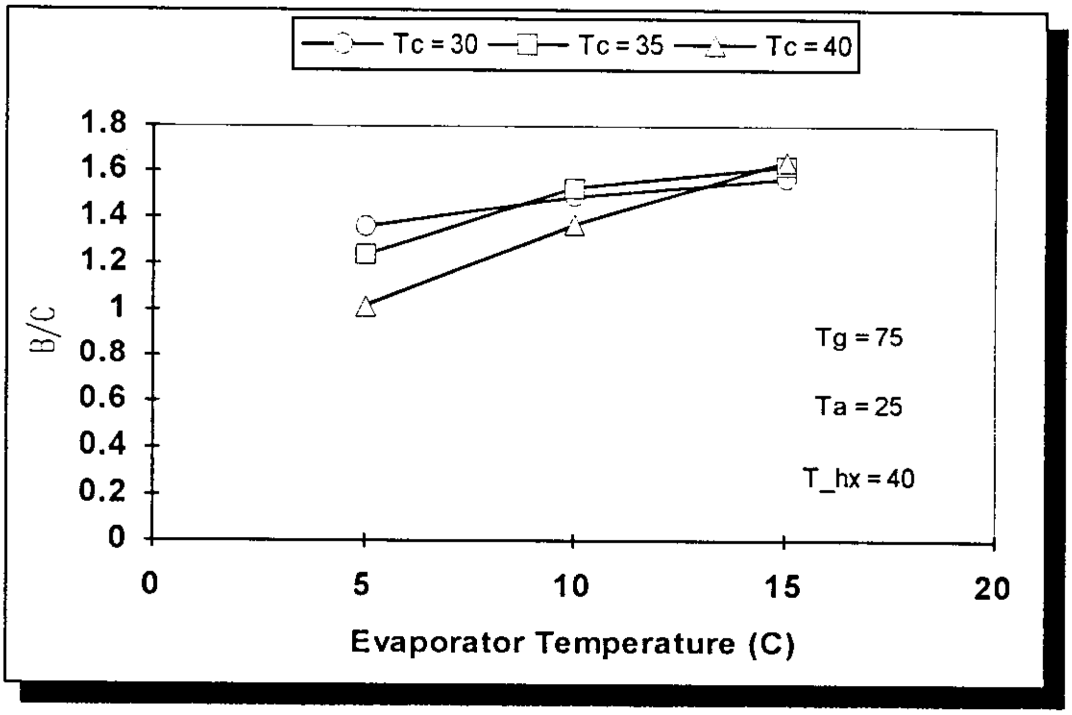


Fig.18: Benefit Cost Ratio vs Evaporator Temperature.

6.5. System Capacity

Figure 19 below compares the cost projected by implementing the solar refrigeration system to the savings gained as functions of the system design capacity.

It is seen that for small systems which have capacities less or equal two kW of refrigeration, the cost of the system is higher than its savings which means that economic feasibility is not established. For the higher capacity systems, these projects begin to profit as indicated by the higher savings' line in the above figure after the intersection point.

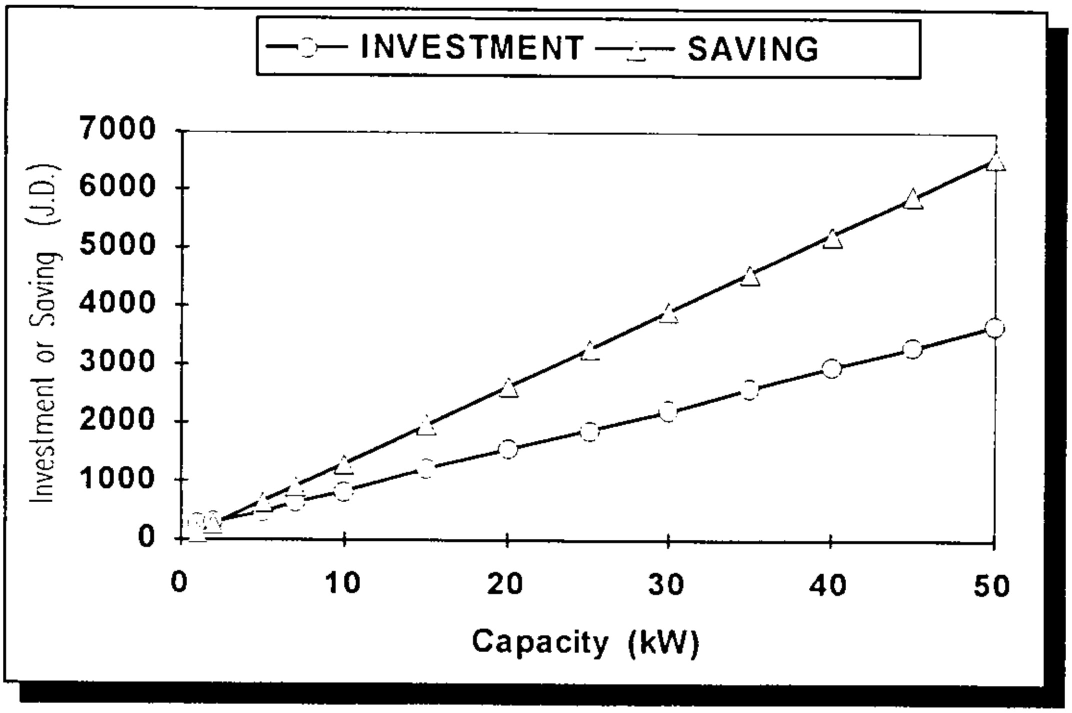


Fig.19: Investment and Savings vs System Capacity.

At the optimum design case, with a unit of 30 kW of refrigeration power and under the economic analysis assumptions, this unit needs an additional investment of 2198 JD more than a conventional refrigeration unit of the same capacity , and will save about 400 JD per year. This unit is found to have a payback period for the additional investment required of only about 5.5 years of its 20 years lifetime.

Finally, the optimum design point is located by plotting the maximum benefit cost ratio - found at a certain cycle temperatures - against the system capacity as shown in fig.20.

Each point in the curve represents an optimum design case at the specified design capacity of the refrigeration system, and the curve then allocates the optimum design capacity that will further maximize the benefit cost ratio for the solar absorption refrigeration systems using LiBr-H_2O in Amman.

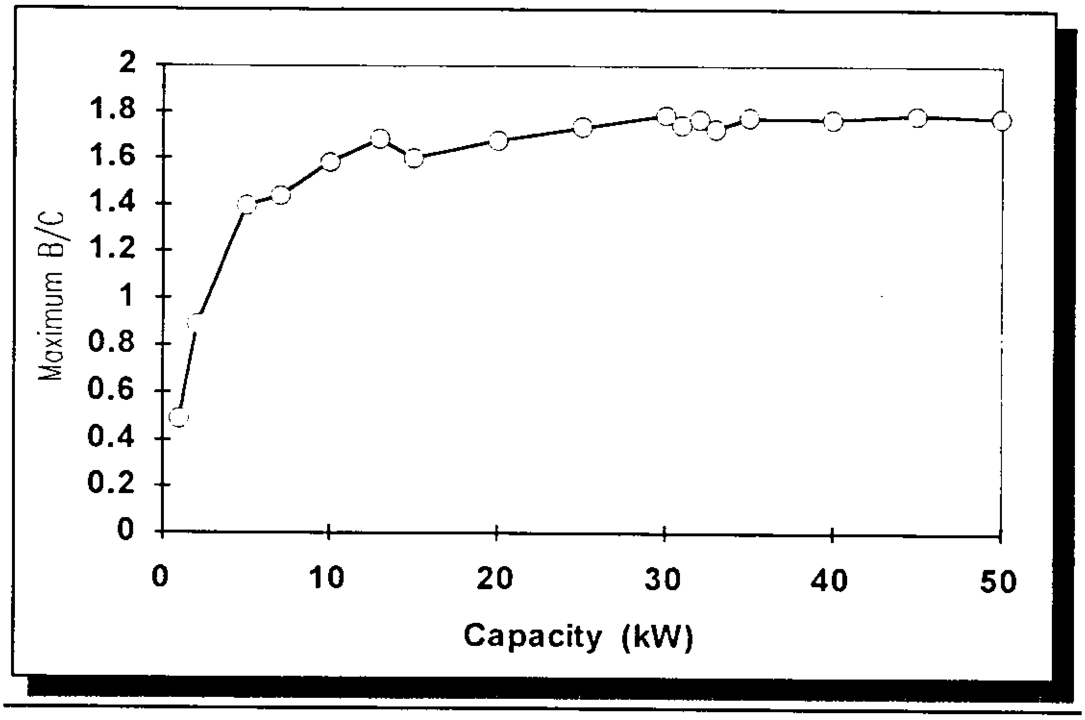


Fig.20: Maximum Benefit Cost Ratio vs System Capacity.

Figure 20 indicates an optimum design capacity of 30 kW of refrigeration at a maximum benefit cost ratio of 1.786. The cycle parameters of this design are discussed in the previous section and the detailed results are read from the optimization reports prepared by the computer program. These reports are found in appendix c.

6.6. Experimental Results

To establish the mathematical model validity, a sample of the experimental results were extracted and are shown in figures 21 and 22 for the days of 23 and 24 of October 1993 respectively.

Both figures plot the experimental generator and evaporator temperatures on the left y-axis and the measured solar intensity in W/m^2 on the right y-axis against the hour of the day during the time of the experiment. Other measured data are not presented in this report because they are not relevant.

The experimental figures are shown below.

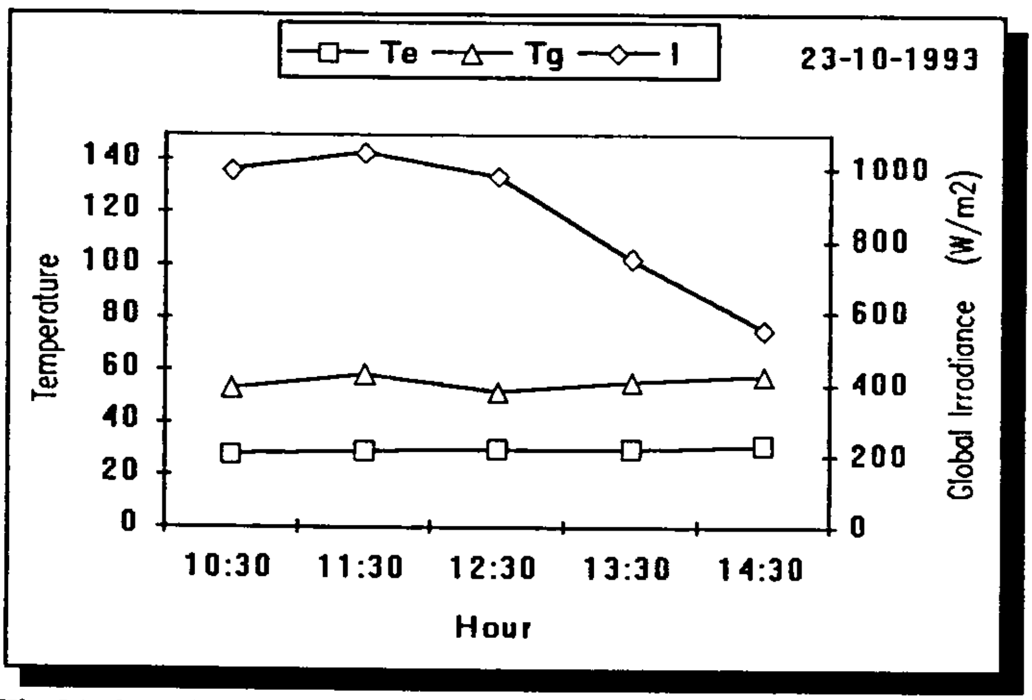


Fig.21: Hourly Experimental (I,Te,Tg) with Time (23-10-93).

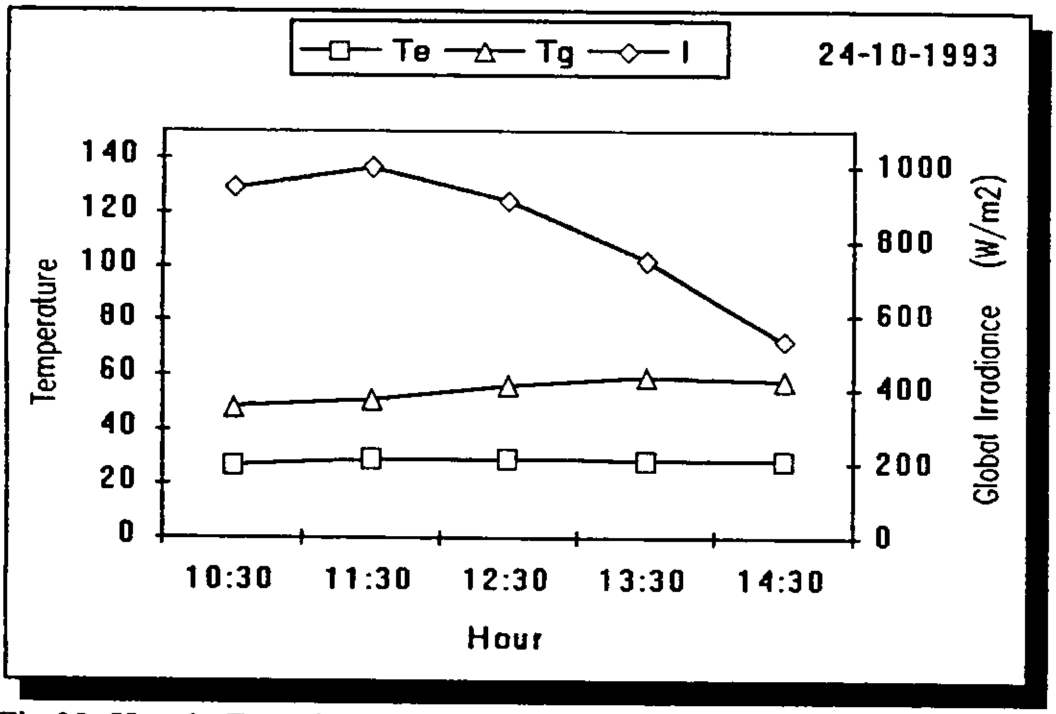


Fig.22: Hourly Experimental (I,Te,Tg) with Time (24-10-93).

The experimental data figures show that the generator temperature ranged between 50-59 $°C$, while the evaporator temperature ranged between 26-28 $°C$ slightly less than the ambient. This condition was at solution concentration of only 25 % and solar intensity range between 500-1000 W/m^2.

It is seen that the generator temperature was very low compared to the theoretical generator temperature required for steady and accepted operation of the absorption cycle (about 70 $°C$). It is believed that practical design imperfections were the reasons for generator deficiency. In addition, the experimental generator temperature was low enough to risk the start up and operation of the cycle. This is in compliance with the theoretical generator temperature lower limit discussed earlier.

The evaporator temperature did not drop to the expected theoretical range (5 - 15 °C). It is believed that this is due to both generator and absorber deficiencies; the generator because of its low temperature resulting in unsteady steam production, and the absorber not being able to mix the solution with the vapor effectively.

Finally, the solution concentration was found to be very low which is believed to risk the absorption ability of the solution no matter what conditions prevail in the cycle.

CHAPTER VII

CONCLUSIONS AND RECOMMENDATIONS

7. CONCLUSIONS AND RECOMMENDATIONS

7.1. Conclusions

1. Medium and large size solar powered LiBr-H_2O absorption refrigeration systems will work successfully and profitably in Amman area due to the availability of good solar insolation for considerable duration of the year.

2. This study resulted an optimum condition for the solar powered LiBr-H_2O absorption refrigeration systems that has a benefit cost ratio of 1.786 at a capacity of 30 kW, and 70 $°C$ generator temperature, 40 $°C$ condenser temperature, and 15 $°C$ evaporator temperature.

7.2. Recommendations

1. Further work may investigate the field of removing local manufacturing obstacles, and the improvement possibilities in the local design and manufacturing of solar collectors.

2. Further investigation should be done on the practical application aspects and constraints of the solar powered LiBr-H_2O absorption refrigeration systems in Jordan, including the addition of hot water storage and its effect on the stability of the generator performance.

3. The application of the system to chill water instead of cooling air in the evaporator and its effect on the overall system performance and utilization should be studied, with consideration of adding cold water storage.

4. Finally, the performance of the system under different loading schemes must be subject to future research.

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10. Tarawneh M. T, "Performance of Solar Cooling System Using Computer Simulation", M.Sc. thesis, University of Jordan, Amman, Jordan, December 1991.

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12. Bhalchandra V. Karlekar, Robert M. Desmond, "Heat Transfer",2nd edition, West Publishing Company, St. Paul, 1982.

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14. Gordon J. Wylen, Richard E. Sonntag, "Fundamentals of Classical Thermodynamics", 3rd edition, John Wiley & Sons, New York, 1985.

15. John A. Duffie, William A. Beckman, "Solar Engineering of Thermal Processes", John Wiley & Sons, New York, 1980.

Appendices

App. A

COMPUTER PROGRAM LISTING

```
program master(input,output);
uses crt;
label 10 ;
type
  matrix = array [1..9,1..6] of real ;
  vector = array [1..9]  of real ;

var
        i,j  : integer  ;
     ii,jj,kk,l,mm  : integer  ;
        a,b  : matrix ;    { a for h of soln,  b for x of soln}
    T,a_int,b_int : vector ; { a_int for coeff of h soln  6 elements
                                   b_int for coeff of x soln      ""
                                   T from 25 - 100 C    9 elements }
   Ta,T_hx,Tg,Tc,Te : real ;
   T4,T_eff,U,cp1   : real;
   T6,T_cwi,T_cwo   : real ;
   T_cwo2,T1,T2      : real ;
   T_amb,T_out,hours  : real  ;

   temp1,temp2       : real ;
   x1,x3,Pc,Pa,mw    : real ;
   w1,w5,w3,COP,m_coll   : real ;
   m_air,cp_air,U_ev : real ;

   cost_pumps,cost_solar  :  real  ;
   cost,IT,S,C,n_coll     : real  ;
   total,diff,B_C        : real ;

   Qa,Q_hx,Qg,Qc,Qe : real ;
   Aa,A_hx,Ag,Ac,Ae : real ;
   A_coll           : real ;

   h1,h2,h3,h4,h5,h6,h7 : real ;
   f1  : text         ;


function power(x:real;n:integer) :real  ;
var
 temp:real;
 i:integer;
 begin
   temp := 1.0;
   for i:=1 to n do
      temp:=x*temp;
   power:=temp;
 end;

function hg(T:real)  : real  ;
```

```
  begin
    hg := 2500.80151+1.90589*T-1.56238e-3*power(T,2) ;
    end;

function hf(T:real)  : real  ;
  begin
    hf := -0.11359+4.192*T  ;
    end;

function P(T:real)  : real ;
  begin
    P := 0.61634 + 4.16182e-2*T + 1.65396e-3*power(T,2)
       + 2.06059e-5*power(T,3) + 3.46434e-7*power(T,4)
       + 2.47824e-9*power(T,5) ;
     end;


function LMTD(T_hi,T_ho,T_ci,T_co: real)  : real  ;
  var
   temp3,temp4  :  real  ;
  begin
   temp3 := T_ho - T_ci  ;
   temp4 := T_hi - T_co  ;
   LMTD := (temp3 - temp4) / ln (temp3/temp4)  ;
   end;

begin
clrscr;

      hours := 8.0  ;   cp_air := 1.0035;

  for i := 1 to 9 do
   for j:= 1 to 6 do
   begin
    b[i,j] := 0  ;
    a[i,j] := 0  ;
    end;
  T[1] := 25.0  ;  T[2] := 30  ;
  for i := 3 to 9 do
      T[i] := T[i-1] + 10.0  ;

{ input }
  write('Enter the system capacity (kW) : ') ; readln(Qe) ;
       { end input}

  assign(f1,'master.txt')  ;
  reset(f1) ;
  for i := 1 to 9 do
   begin
```

```
      for j:= 1 to 6 do
       read(f1,b[i,j]) ;
      readln(f1);
    end;

    for i:= 1 to 9 do
     begin
      for j:= 1 to 6 do
        read(f1,a[i,j]) ;
      readln(f1);
    end;
close(f1) ;

Ta := 20 ;
for l := 1 to 3 do
begin
Ta := Ta + 5 ;
T_hx := 35 ;
for mm := 1 to 4 do
begin
    T_hx := T_hx + 5 ;
     Tg := 60 ;
for ii := 1 to 8 do
begin
  Tg := Tg + 5 ;
  Tc := 25 ;
  for jj := 1 to 4 do
  begin
    Tc := Tc + 5 ;
    Te := 0 ;
    for kk := 1 to 3 do
    begin
    Te := Te + 5 ;

   Pc := P(Tc) ;
   Pa := P(Te) ;

  h7 := hg(Te) ;
  h6 := hf(Tc) ;
  h5 := hg(Tg) ;

i := 1 ;
while  Ta>T[i]  do
             i := i+1 ;
  x1 := 0 ;
  for j := 1 to 6 do
   begin
    if (Ta>T[i-1]) and (Ta<T[i]) then
       begin
```

```
        b_int[j] := b[i,j] - (b[i,j]-b[i-1,j])*(T[i]-Ta)/(T[i]-T[i-1]) ;
        a_int[j] := a[i,j] - (a[i,j]-a[i-1,j])*(T[i]-Ta)/(T[i]-T[i-1]) ;
        end
     else begin
        b_int[j] := b[i,j]  ;
        a_int[j] := a[i,j]  ;
     end;
    x1 := x1 + b_int[j]*power(Pa,j-1)  ;
    end;
    h1 := 0  ;
    for j := 1 to 6 do
        h1 := h1 + a_int[j]*power(x1,j-1)  ;

i := 1  ;
while  T_hx>T[i]  do
           i := i+1  ;
    h2 := 0  ;
  for j := 1 to 6 do
   begin
     if (T_hx>T[i-1]) and (T_hx<T[i]) then
       a_int[j] := a[i,j] - (a[i,j]-a[i-1,j])*(T[i]-T_hx)/(T[i]-T[i-1])
     else
       a_int[j] := a[i,j]  ;

      h2 := h2 + a_int[j]*power(x1,j-1)  ;
   end;

i := 1  ;
while  Tg>T[i]  do
           i := i+1  ;
  x3 := 0 ;
  for j := 1 to 6 do
   begin
    if (Tg>T[i-1]) and (Tg<T[i]) then
      begin
        b_int[j] := b[i,j] - (b[i,j]-b[i-1,j])*(T[i]-Tg)/(T[i]-T[i-1]) ;
        a_int[j] := a[i,j] - (a[i,j]-a[i-1,j])*(T[i]-Tg)/(T[i]-T[i-1]) ;
        end;
     else begin
        b_int[j] := b[i,j]  ;
        a_int[j] := a[i,j]  ;
     end;
        x3 := x3 + b_int[j]*power(Pc,j-1)  ;
   end;

if (x1 > x3) then begin
     clrscr;
     writeln; writeln; writeln; writeln;
     writeln; writeln; writeln; writeln;
```

```
  writeln('Ta (C) = ',Ta:5:2,'    T_hx (C) = ',T_hx:5:2,
          '         Te (C) = ',Te:5:2) ;
  writeln('Tg (C) = ',Tg:5:2,'    Tc (C)   = ',Tc:5:2) ;
    writeln; writeln;
    writeln('    Cycle is not in Operation')   ;
    writeln; writeln; writeln; writeln; writeln; writeln;
    writeln; writeln; writeln; writeln; writeln; writeln;
    writeln('    < Hit Return to Continue >') ;
    goto 10 ;
    end;

   h3 := 0  ;
   for j := 1 to 6 do
       h3 := h3 + a_int[j]*power(x3,j-1)  ;

w5 := Qe/(h7-h6)  ;w1 := w5/(1-x1/x3)  ;
w3 := w1 - w5  ;

Qe := w5*(h7-h6)  ;
Qc := w5*(h5-h6)  ;
Q_hx := w1*(h2-h1)  ;

h4 := h3 - Q_hx/w3  ;
Qa := w5*h7 + w3*h4 - w1*h1  ;
Qg := w5*h5 + w3*h3 - w1*h2  ;
   COP := Qe/Qg  ;

   U := 1  ;        {  kW / m2.C   }

   { HEAT  EXCHANGER  }

   cp1 := Q_hx / (w1*(T_hx-Ta))  ;
   T4 := Tg - Q_hx/(w3*cp1)  ;
    if (T_hx>T4) then begin
     clrscr;
    writeln; writeln; writeln; writeln;
    writeln; writeln; writeln; writeln;
  writeln('Ta (C) = ',Ta:5:2,'    T_hx (C) = ',T_hx:5:2,
          '         Te (C) = ',Te:5:2) ;
  writeln('Tg (C) = ',Tg:5:2,'    Tc (C)   = ',Tc:5:2) ;
    writeln; writeln;
    writeln('    Cycle is not in Operation')   ;
    writeln('    Heat exchanger is not possible');
    writeln; writeln; writeln; writeln; writeln;
    writeln; writeln; writeln; writeln; writeln; writeln;
    writeln('    < Hit Return to Continue >') ;
    goto 10 ;
    end;
   A_hx := Q_hx / (U*LMTD(Tg,T4,Ta,T_hx))  ;
```

```
{ CONDENSER }
  T_cwi := 20 ; { C }

T_cwo := T_cwi + 5 ;
mw := Qc/(4.187*5) ;

  Ac := Qc / (U*LMTD(Tg,Tc,T_cwi,T_cwo)) ;


    { GENERATOR }

  T1 := Tg + 10  ;  { T1,T2 = hot water inlet and outlet of Gen }
  T2 := T1 - 20 ;
  m_coll := Qg/(4.187*20) ;
   if (T_hx>=T2) or ( (T1-Tg) = (T2-T_hx) )then begin
    clrscr;
   writeln; writeln; writeln; writeln;
   writeln; writeln; writeln; writeln;
 writeln('Ta (C) = ',Ta:5:2,'    T_hx (C) = ',T_hx:5:2,
        '      Te (C) = ',Te:5:2) ;
 writeln('Tg (C) = ',Tg:5:2,'    Tc (C)   = ',Tc:5:2) ;
   writeln; writeln;
   writeln('    Cycle is not in Operation')  ;
   writeln('    Generator is not possible');
   writeln; writeln; writeln; writeln; writeln;
   writeln; writeln; writeln; writeln; writeln; writeln;
   writeln('    < Hit Return to Continue >') ;
   goto 10 ;
   end;
  Ag := Qg / (U*LMTD(T1,T2,T_hx,Tg)) ;


        {    ABSORBER    }

  T_eff := Qa / (w1*cp1) + Ta  ;
  T_cwo2 := T_cwi + Qa/(mw*4.187)  ;
   if(T_cwo>=T_eff) then begin
    clrscr;
   writeln; writeln; writeln; writeln;
   writeln; writeln; writeln; writeln;
 writeln('Ta (C) = ',Ta:5:2,'    T_hx (C) = ',T_hx:5:2,
        '      Te (C) = ',Te:5:2) ;
 writeln('Tg (C) = ',Tg:5:2,'    Tc (C)   = ',Tc:5:2) ;
   writeln; writeln;
   writeln('    Cycle is not in Operation')  ;
   writeln('    Absorber is not possible');
   writeln; writeln; writeln; writeln; writeln;
   writeln; writeln; writeln; writeln; writeln; writeln;
```

```
writeln('   < Hit Return to Continue >') ;
goto 10 ;
end;
Aa := Qa / (U*LMTD(T_eff,Ta,T_cwi,T_cwo2)) ;


      { EVAPORATOR  }

      T_amb := 30 ;

   U_ev := 0.20 ;          { kW / m2.C }
  T_out := Te + 5 ;
  m_air := Qe/(cp_air*(T_amb-T_out)) ;

   Ae := Qe/(U_ev*LMTD(T_amb,T_out,Te,Te)) ;



     { SOLAR COLLECTORS  }

A_coll := (Qg*1000) / (0.78*800-4.45*(T2-T_amb)) ; { I_s = 625 original }
n_coll := A_coll / 2  ;
n_coll := trunc(n_coll) + 1  ;


    {   COST   }

 cost_pumps := 3*30 + (w1+m_coll+mw)*20
          + 0.225*(w1+m_coll+mw)*hours*100*0.04 ;
 cost_solar := 175*(n_coll)  ;
 cost := 700*(Ac+A_hx+Ag+Aa) + 100*Ae  ;
 total := cost_pumps+cost_solar+cost  ;

S := Qe*hours*100*0.05/3.0  ;  {COP = 3  conventional , 100 days/yr}
diff := total - 5.0*Qe*1000/17.57  ;
B_C := S / (diff*0.1019) ;         { life = 20 years }

{        Results          }

clrscr;

writeln;

  writeln('Ta (C) = ',Ta:5:2,'   T_hx (C) = ',T_hx:5:2,
          '    Te (C) = ',Te:5:2) ;
  writeln('Tg (C) = ',Tg:5:2,'   Tc (C)  = ',Tc:5:2) ;
  writeln;
writeln('Qe  (kW) = ',Qe:8:3) ;
writeln('Qc  (kW) = ',Qc:8:3) ;
```

```
write('Qg  (kW) = ',Qg:8:3) ;     writeln('   COP = ',COP:6:3);
writeln('Qa  (kW) = ',Qa:8:3) ;
writeln('Q_hx (kW) = ',Q_hx:8:3) ;

writeln;

write('Pa = ',Pa:6:3)  ; writeln('   x1 = ',x1:6:3,' %') ;
write('Pc = ',Pc:6:3)  ; writeln('   x3 = ',x3:6:3,' %') ;

writeln;

write('w1 (kg/sec) = ',w1:6:3) ;
write('   w3 (kg/sec) = ',w3:6:3) ;
writeln('   w5 (kg/sec) = ',w5:6:3) ;

writeln;

    writeln('Heat Exchanger Area (m2) = ',A_hx:10:6) ;
    writeln('Condenser Area (m2) = ',Ac:10:6) ;
    writeln('Generator Area (m2) = ',Ag:10:6) ;
    writeln('Absorber Area (m2) = ',Aa:10:6) ;
    writeln('Evaporator Area (m2) = ',Ae:10:6) ;

writeln;
 writeln('The number of collectors needed = ',n_coll:4:0) ;
  writeln('Total cost = ',round(total),' J.D.');
  writeln('Benefit - Cost ratio (B/C) = ',B_C:5:2)  ;

10 : readln;
      end;  { Te loop  }
    end;    { Tc loop  }
  end;      { TG loop  }
 end;       { T_hx loop }
end;        { Ta loop  }
end.
```

App. B

SAMPLE OF PROGRAM REPORTS

```
'a (C) = 25.00      T_hx (C) = 40.00        Te (C) =  5.00
'g (C) = 65.00      Tc (C)   = 30.00

!e    (kW) =    20.000
!c    (kW) =    20.904
'g    (kW) =    24.936      COP =  0.802
a     (kW) =    24.032
_hx (kW) =       2.129

   =  0.869        x1 =  49.623 %
c  =  4.251        x3 =  57.457 %

1 (kg/sec) =   0.062    w3 (kg/sec) =   0.053    w5 (kg/sec) =   0.008

eat Exchanger Area (m2) =    0.089471
ondenser Area (m2) =    0.965972
enerator Area (m2) =    2.022137
osorber Area (m2) =    0.516871
vaporator Area (m2) =    8.047190

 number of collectors needed =    25
 al cost = 7823  J.D.
enefit - Cost ratio (B/C) =  1.23


 (C) = 25.00      T_hx (C) = 40.00        Te (C) = 10.00
 (C) = 65.00      Tc (C)   = 30.00

    (kW) =    20.000
    (kW) =    20.822
    (kW) =    23.386      COP =  0.855
    (kW) =    22.564
hx (kW) =      1.399

  =  1.222        x1 =  44.995 %
  =  4.251        x3 =  57.457 %

 (kg/sec) =   0.039    w3 (kg/sec) =   0.030    w5 (kg/sec) =   0.008

at Exchanger Area (m2) =    0.061181
ndenser Area (m2) =    0.962174
nerator Area (m2) =    1.896424
sorber Area (m2) =    0.369842
aporator Area (m2) =    9.241962

 number of collectors needed =    23
al cost = 7378  J.D.
nefit - Cost ratio (B/C) =  1.55
```

'a (C) = 25.00 T_hx (C) = 40.00 Te (C) = 5.00
'g (C) = 65.00 Tc (C) = 30.00

e (kW) = 20.000
c (kW) = 20.904
g (kW) = 24.936 COP = 0.802
a (kW) = 24.032
_hx (kW) = 2.129

a = 0.869 x1 = 49.623 %
c = 4.251 x3 = 57.457 %

1 (kg/sec) = 0.062 w3 (kg/sec) = 0.053 w5 (kg/sec) = 0.008

eat Exchanger Area (m2) = 0.089471
nser Area (m2) = 0.965972
or Area (m2) = 2.022137
sorber Area (m2) = 0.516871
vaporator Area (m2) = 8.047190

ne number of collectors needed = 25
otal cost = 7823 J.D.
enefit - Cost ratio (B/C) = 1.23

= 25.00 T_hx (C) = 40.00 Te (C) = 10.00
(C) = 65.00 Tc (C) = 30.00

(kW) = 20.000
(kW) = 20.822
(kW) = 23.386 COP = 0.855
(kW) = 22.564
hx (kW) = 1.399

= 1.222 x1 = 44.995 %
= 4.251 x3 = 57.457 %

(kg/sec) = 0.039 w3 (kg/sec) = 0.030 w5 (kg/sec) = 0.008

at Exchanger Area (m2) = 0.061181
denser Area (m2) = 0.962174
nerator Area (m2) = 1.896424
sorber Area (m2) = 0.369842
aporator Area (m2) = 9.241962

e number of collectors needed = 23
tal cost = 7378 J.D.
nefit - Cost ratio (B/C) = 1.55

'a (C) = 25.00 T_hx (C) = 40.00 Te (C) = 15.00
'g (C) = 65.00 Tc (C) = 30.00

'e (kW) = 20.000
'c (kW) = 20.741
'g (kW) = 22.392 COP = 0.893
'a (kW) = 21.651
_hx (kW) = 1.028

a = 1.702 x1 = 39.979 %
c = 4.251 x3 = 57.457 %

1 (kg/sec) = 0.027 w3 (kg/sec) = 0.019 w5 (kg/sec) = 0.008

eat Exchanger Area (m2) = 0.047682
ondenser Area (m2) = 0.958437
enerator Area (m2) = 1.815805
bsorber Area (m2) = 0.288822
vaporator Area (m2) = 10.986123

.e number of collectors needed = 22
otal cost = 7251 J.D.
enefit - Cost ratio (B/C) = 1.68

a (C) = 25.00 T_hx (C) = 40.00 Te (C) = 5.00
g (C) = 65.00 Tc (C) = 35.00

e (kW) = 20.000
c (kW) = 20.912
g (kW) = 26.865 COP = 0.744
a (kW) = 25.953
_hx (kW) = 3.360

a = 0.869 x1 = 49.623 %
c = 5.633 x3 = 54.361 %

(kg/sec) = 0.097 w3 (kg/sec) = 0.089 w5 (kg/sec) = 0.008

at Exchanger Area (m2) = 0.138421
ndenser Area (m2) = 0.820446
nerator Area (m2) = 2.178574
sorber Area (m2) = 0.741458
aporator Area (m2) = 8.047190

e number of collectors needed = 27
tal cost = 8373 J.D.
nefit - Cost ratio (B/C) = 0.98

Ta (C) = 25.00   T_hx (C) = 40.00   Te (C) = 15.00
Tg (C) = 65.00   Tc (C) = 30.00

Qe (kW) = 20.000
Qc (kW) = 20.741
Qg (kW) = 22.392   COP = 0.893
Qa (kW) = 21.651
Q_hx (kW) = 1.028

Pa = 1.702   x1 = 39.979 %
Pc = 4.251   x3 = 57.457 %

w1 (kg/sec) = 0.027   w3 (kg/sec) = 0.019   w5 (kg/sec) = 0.008

Heat Exchanger Area (m2) = 0.047682
Condenser Area (m2) = 0.958437
Generator Area (m2) = 1.815805
Absorber Area (m2) = 0.288822
Evaporator Area (m2) = 10.986123

The number of collectors needed = 22
Total cost = 7251 J.D.
Benefit - Cost ratio (B/C) = 1.68

Ta (C) = 25.00   T_hx (C) = 40.00   Te (C) = 5.00
Tg (C) = 65.00   Tc (C) = 35.00

Qe (kW) = 20.000
Qc (kW) = 20.912
Qg (kW) = 26.865   COP = 0.744
Qa (kW) = 25.953
Q_hx (kW) = 3.360

Pa = 0.869   x1 = 49.623 %
Pc = 5.633   x3 = 54.361 %

w1 (kg/sec) = 0.097   w3 (kg/sec) = 0.089   w5 (kg/sec) = 0.008

Heat Exchanger Area (m2) = 0.138421
Condenser Area (m2) = 0.820446
Generator Area (m2) = 2.178574
Absorber Area (m2) = 0.741458
Evaporator Area (m2) = 8.047190

The number of collectors needed = 27
Total cost = 8373 J.D.
Benefit - Cost ratio (B/C) = 0.98

a (C) = 25.00 T_hx (C) = 40.00 Te (C) = 10.00
g (C) = 65.00 Tc (C) = 35.00

e (kW) = 20.000
c (kW) = 20.829
g (kW) = 23.993 COP = 0.834
a (kW) = 23.164
_hx (kW) = 1.776

a = 1.222 x1 = 44.995 %
c = 5.633 x3 = 54.361 %

1 (kg/sec) = 0.049 w3 (kg/sec) = 0.040 w5 (kg/sec) = 0.008

eat Exchanger Area (m2) = 0.075896
ondenser Area (m2) = 0.817192
enerator Area (m2) = 1.945695
osorber Area (m2) = 0.446563
vaporator Area (m2) = 9.241962

he number of collectors needed = 24
tal cost = 7550 J.D.
enefit - Cost ratio (B/C) = 1.41

a (C) = 25.00 T_hx (C) = 40.00 Te (C) = 15.00
g (C) = 65.00 Tc (C) = 35.00

e (kW) = 20.000
c (kW) = 20.748
g (kW) = 22.683 COP = 0.882
a (kW) = 21.936
hx (kW) = 1.192

a = 1.702 x1 = 39.979 %
c = 5.633 x3 = 54.361 %

(kg/sec) = 0.032 w3 (kg/sec) = 0.023 w5 (kg/sec) = 0.008

at Exchanger Area (m2) = 0.053719
ndenser Area (m2) = 0.813991
nerator Area (m2) = 1.839452
sorber Area (m2) = 0.324926
aporator Area (m2) = 10.986123

e number of collectors needed = 23
tal cost = 7371 J.D.
nefit - Cost ratio (B/C) = 1.56

'a (C) = 25.00 T_hx (C) = 40.00 Te (C) = 10.00
'g (C) = 65.00 Tc (C) = 35.00

'e (kW) = 20.000
'c (kW) = 20.829
g (kW) = 23.993 COP = 0.834
a (kW) = 23.164
_hx (kW) = 1.776

a = 1.222 x1 = 44.995 %
c = 5.633 x3 = 54.361 %

(kg/sec) = 0.049 w3 (kg/sec) = 0.040 w5 (kg/sec) = 0.008

eat Exchanger Area (m2) = 0.075896
ondenser Area (m2) = 0.817192
enerator Area (m2) = 1.945695
osorber Area (m2) = 0.446563
vaporator Area (m2) = 9.241962

ne number of collectors needed = 24
otal cost = 7550 J.D.
enefit - Cost ratio (B/C) = 1.41

(C) = 25.00 T_hx (C) = 40.00 Te (C) = 15.00
65.00 Tc (C) = 35.00

(kW) = 20.000
(kW) = 20.748
(kW) = 22.683 COP = 0.882
(kW) = 21.936
hx (kW) = 1.192

= 1.702 x1 = 39.979 %
= 5.633 x3 = 54.361 %

(kg/sec) = 0.032 w3 (kg/sec) = 0.023 w5 (kg/sec) = 0.008

at Exchanger Area (m2) = 0.053719
ndenser Area (m2) = 0.813991
nerator Area (m2) = 1.839452
sorber Area (m2) = 0.324926
aporator Area (m2) = 10.986123

e number of collectors needed = 23
tal cost = 7371 J.D.
nefit - Cost ratio (B/C) = 1.56

Ta (C) = 25.00   T_hx (C) = 40.00   Te (C) = 5.00
Tg (C) = 65.00   Tc (C)   = 40.00

Qe   (kW) =   20.000
Qc   (kW) =   20.920
Qg   (kW) =   34.553   COP = 0.579
Qa   (kW) =   33.633
Q_hx (kW) =   8.268

Pa = 0.869   x1 = 49.623 %
Pc = 7.387   x3 = 51.462 %

(kg/sec) =   0.239   w3 (kg/sec) =   0.230   w5 (kg/sec) =   0.009

Heat Exchanger Area (m2) =   0.334438
Condenser Area (m2) =   0.725040
Generator Area (m2) =   2.802032
Absorber Area (m2) =   1.555701
Evaporator Area (m2) =   8.047190

The number of collectors needed =   34
Total cost = 10682 J.D.
Benefit - Cost ratio (B/C) = 0.52

C) = 25.00   T_hx (C) = 40.00   Te (C) = 10.00
C) = 65.00   Tc (C)   = 40.00

e   (kW) =   20.000
c   (kW) =   20.837
g   (kW) =   25.085   COP = 0.797
a   (kW) =   24.249
_hx (kW) =   2.457

a = 1.222   x1 = 44.995 %
c = 7.387   x3 = 51.462 %

1 (kg/sec) =   0.068   w3 (kg/sec) =   0.059   w5 (kg/sec) =   0.009

eat Exchanger Area (m2) =   0.102778
ondenser Area (m2) =   0.722138
enerator Area (m2) =   2.034248
bsorber Area (m2) =   0.576723
vaporator Area (m2) =   9.241962

he number of collectors needed =   25
otal cost = 7831 J.D.
enefit - Cost ratio (B/C) = 1.22

```
Ta (C) = 25.00        T_hx (C) = 40.00        Te (C) =  5.00
Tg (C) = 65.00        Tc (C)   = 40.00

Qe    (kW) =    20.000
Qc    (kW) =    20.920
Qg    (kW) =    34.553     COP =  0.579
Qa    (kW) =    33.633
Q_hx  (kW) =     8.268

Pa =  0.869       x1 =  49.623 %
Pc =  7.387       x3 =  51.462 %

w1 (kg/sec) =   0.239    w3 (kg/sec) =   0.230    w5 (kg/sec) =   0.009

Heat Exchanger Area (m2) =    0.334438
Condenser Area (m2) =    0.725040
Generator Area (m2) =    2.802032
Absorber Area (m2) =    1.555701
Evaporator Area (m2) =    8.047190

The number of collectors needed =    34
Total cost = 10682  J.D.
Benefit - Cost ratio (B/C) =  0.52


Ta (C) = 25.00        T_hx (C) = 40.00        Te (C) = 10.00
Tg (C) = 65.00        Tc (C)   = 40.00

Qe    (kW) =    20.000
Qc    (kW) =    20.837
Qg    (kW) =    25.085     COP =  0.797
Qa    (kW) =    24.249
Q_hx  (kW) =     2.457

Pa =  1.222       x1 =  44.995 %
Pc =  7.387       x3 =  51.462 %

w1 (kg/sec) =   0.068    w3 (kg/sec) =   0.059    w5 (kg/sec) =   0.009

Heat Exchanger Area (m2) =    0.102778
Condenser Area (m2) =    0.722138
Generator Area (m2) =    2.034248
Absorber Area (m2) =    0.576723
Evaporator Area (m2) =    9.241962

The number of collectors needed =    25
Total cost = 7831  J.D.
Benefit - Cost ratio (B/C) =  1.22
```

```
Ta (C) = 25.00      T_hx (C) = 40.00        Te (C) = 15.00
Tg (C) = 65.00      Tc (C)   = 40.00

Qe    (kW) =    20.000
Qc    (kW) =    20.754
Qg    (kW) =    23.094    COP =  0.866
Qa    (kW) =    22.340
Q_hx  (kW) =     1.426

Pa =  1.702      x1 =  39.979 %
Pc =  7.387      x3 =  51.462 %

w1 (kg/sec) =   0.038    w3 (kg/sec) =   0.029    w5 (kg/sec) =   0.008

Heat Exchanger Area (m2) =    0.062607
Condenser Area (m2) =    0.719284
Generator Area (m2) =    1.872802
Absorber Area (m2) =    0.374451
Evaporator Area (m2) =   10.986123

The number of collectors needed =    23
Total cost = 7370  J.D.
Benefit - Cost ratio (B/C) =  1.56



Ta (C) = 25.00      T_hx (C) = 40.00        Te (C) =  5.00
Tg (C) = 70.00      Tc (C)   = 30.00

Qe    (kW) =    20.000
Qc    (kW) =    20.975
Qg    (kW) =    24.858    COP =  0.805
Qa    (kW) =    23.883
Q_hx  (kW) =     1.686

Pa =  0.869      x1 =  49.623 %
Pc =  4.251      x3 =  59.948 %

w1 (kg/sec) =   0.049    w3 (kg/sec) =   0.040    w5 (kg/sec) =   0.008

Heat Exchanger Area (m2) =    0.059327
Condenser Area (m2) =    0.901377
Generator Area (m2) =    1.723044
Absorber Area (m2) =    0.432567
Evaporator Area (m2) =    8.047190

The number of collectors needed =    26
Total cost = 7663  J.D.
Benefit - Cost ratio (B/C) =  1.33
```

```
Ta (C) = 25.00      T_hx (C) = 40.00      Te (C) = 15.00
Tg (C) = 65.00      Tc (C)   = 40.00

Qe    (kW) =    20.000
Qc    (kW) =    20.754
Qg    (kW) =    23.094     COP =  0.866
Qa    (kW) =    22.340
Q_hx  (kW) =     1.426

Pa =  1.702       x1 =  39.979 %
Pc =  7.387       x3 =  51.462 %

w1 (kg/sec) =   0.038    w3 (kg/sec) =   0.029    w5 (kg/sec) =   0.008

Heat Exchanger Area (m2) =    0.062607
Condenser Area (m2) =    0.719284
Generator Area (m2) =    1.872802
Absorber Area (m2) =    0.374451
Evaporator Area (m2) =   10.986123

The number of collectors needed =    23
Total cost = 7370  J.D.
Benefit - Cost ratio (B/C) =  1.56
```

```
Ta (C) = 25.00      T_hx (C) = 40.00      Te (C) =  5.00
Tg (C) = 70.00      Tc (C)   = 30.00

Qe    (kW) =    20.000
Qc    (kW) =    20.975
Qg    (kW) =    24.858     COP =  0.805
Qa    (kW) =    23.883
Q_hx  (kW) =     1.686

Pa =  0.869       x1 =  49.623 %
Pc =  4.251       x3 =  59.948 %

w1 (kg/sec) =   0.049    w3 (kg/sec) =   0.040    w5 (kg/sec) =   0.008

Heat Exchanger Area (m2) =    0.059327
Condenser Area (m2) =    0.901377
Generator Area (m2) =    1.723044
Absorber Area (m2) =    0.432567
Evaporator Area (m2) =    8.047190

The number of collectors needed =    26
Total cost = 7663  J.D.
Benefit - Cost ratio (B/C) =  1.33
```

App. C

OPTIMIZATION REPORTS

a (1.00 kW) system capacity

maximum Benefit-Cost ratio is 0.494

h a COP of 0.759 , and a total additional cost of 265 J.D.
Total Savings = 131 J.D.)

= 25.00 T_hx = 40.000 Tg = 100.000 Tc = 40.000 Te = 5.000

t exchanger areas (m2) are as follows :
= 0.053 Ac = 0.026 Ae = 0.402
Aa = 0.016 A_hx = 0.001

mass delivered by the solution pump (kg/s) = 0.0017

finally the number of collectors needed are 2

COP = 0.893
.000 40.000 15.000 65.000 30.000
t = 605.555 J.D. B/C = 0.408 Qe = 1.000 (kW)

a (2.00 kW) system capacity

... Benefit-Cost ratio is 0.888

h a COP of 0.839 , and a total additional cost of 295 J.D.
Total Savings = 262 J.D.)

= 25.00 T_hx = 40.000 Tg = 90.000 Tc = 40.000 Te = 15.000

t exchanger areas (m2) are as follows :
= 0.110 Ac = 0.055 Ae = 1.099
Aa = 0.026 A_hx = 0.002

mass delivered by the solution pump (kg/s) = 0.0023

finally the number of collectors needed are 3

COP = 0.893
.000 40.000 15.000 65.000 30.000
t = 946.110 J.D. B/C = 0.694 Qe = 2.000 (kW)

or a (5.00 kW) system capacity

he maximum Benefit-Cost ratio is 1.397

ith a COP of 0.860 , and a total additional cost of 468 J.D.
(Total Savings = 654 J.D.)

a = 25.00 T_hx = 40.000 Tg = 70.000 Tc = 40.000 Te = 15.000

eat exchanger areas (m2) are as follows :
Ag = 0.403 Ac = 0.169 Ae = 2.747
Aa = 0.085 A_hx = 0.011

he mass delivered by the solution pump (kg/s) = 0.0083

nd finally the number of collectors needed are 6

ax COP = 0.893
25.000 40.000 15.000 65.000 30.000
Cost = 1967.775 J.D. B/C = 1.201 Qe = 5.000 (kW)

r a (7.00 kW) system capacity

e maximum Benefit-Cost ratio is 1.438

th a COP of 0.866 , and a total additional cost of 637 J.D.
h Total Savings = 916 J.D.)

= 25.00 T_hx = 40.000 Tg = 65.000 Tc = 40.000 Te = 15.000

at exchanger areas (m2) are as follows :
t = 0.655 Ac = 0.252 Ae = 3.845
= Aa = 0.131 A_hx = 0.022

e mass delivered by the solution pump (kg/s) = 0.0133

d finally the number of collectors needed are 8

x COP = 0.893
C5.000 40.000 15.000 65.000 30.000
Ost = 2648.885 J.D. B/C = 1.394 Qe = 7.000 (kW)

r a (20.00 kW) system capacity

e maximum Benefit-Cost ratio is 1.678

th a COP of 0.893 , and a total additional cost of 1560 J.D.
Total Savings = 2617 J.D.)

= 25.00 T_hx = 40.000 Tg = 65.000 Tc = 30.000 Te = 15.000

at exchanger areas (m2) are as follows :
= 1.816 Ac = 0.958 Ae = 10.986
a = 0.289 A_hx = 0.048

mass delivered by the solution pump (kg/s) = 0.0274

l finally the number of collectors needed are 22

COP = 0.893
.000 40.000 15.000 65.000 30.000
t = 7251.100 J.D. B/C = 1.678 Qe = 20.000 (kW)

25.00 kW) system capacity

aximum Benefit-Cost ratio is 1.732

h a COP of 0.885 , and a total additional cost of 1889 J.D.
Total Savings = 3271 J.D.)

= 25.00 T_hx = 40.000 Tg = 70.000 Tc = 30.000 Te = 15.000

t exchanger areas (m2) are as follows :
= 1.959 Ac = 1.118 Ae = 13.733
Aa = 0.337 A_hx = 0.045

mass delivered by the solution pump (kg/s) = 0.0312

finally the number of collectors needed are 29

COP = 0.893
.000 40.000 15.000 65.000 30.000
= 9128.875 J.D. B/C = 1.624 Qe = 25.000 (kW)

a (30.00 kW) system capacity

maximum Benefit-Cost ratio is 1.786

a COP of 0.874 , and a total additional cost of 2198 J.D.
otal Savings = 3925 J.D.)

25.00 T_hx = 40.000 Tg = 70.000 Tc = 35.000 Te = 15.000

exchanger areas (m2) are as follows :
2.378 Ac = 1.144 Ae = 16.479
Aa = 0.447 A_hx = 0.060

mass delivered by the solution pump (kg/s) = 0.0424

finally the number of collectors needed are 35

COP = 0.893
000 40.000 15.000 65.000 30.000
= 10831.650 J.D. B/C = 1.711 Qe = 30.000 (kW)

a (35.00 kW) system capacity

maximum Benefit-Cost ratio is 1.776

COP of 0.874 , and a total additional cost of 2579 J.D.
otal Savings = 4580 J.D.)

25.00 T_hx = 40.000 Tg = 70.000 Tc = 35.000 Te = 15.000

exchanger areas (m2) are as follows :
2.775 Ac = 1.334 Ae = 19.226
Aa = 0.521 A_hx = 0.070

mass delivered by the solution pump (kg/s) = 0.0495

finally the number of collectors needed are 41

COP = 0.893
000 40.000 15.000 65.000 30.000
= 12709.425 J.D. B/C = 1.666 Qe = 35.000 (kW)

' a (40.00 kW) system capacity

: maximum Benefit-Cost ratio is 1.769

h a COP of 0.874 , and a total additional cost of 2959 J.D.
Total Savings = 5234 J.D.)

= 25.00 T_hx = 40.000 Tg = 70.000 Tc = 35.000 Te = 15.000

t exchanger areas (m2) are as follows :
= 3.171 Ac = 1.525 Ae = 21.972
Aa = 0.595 A_hx = 0.080

mass delivered by the solution pump (kg/s) = 0.0566

finally the number of collectors needed are 47

COP = 0.893
00 40.000 15.000 65.000 30.000
c = 14412.200 J.D. B/C = 1.728 Qe = 40.000 (kW)

a (45.00 kW) system capacity

maximum Benefit-Cost ratio is 1.788

a a COP of 0.885 , and a total additional cost of 3292 J.D.
Total Savings = 5888 J.D.)

= 25.00 T_hx = 40.000 Tg = 70.000 Tc = 30.000 Te = 15.000

: exchanger areas (m2) are as follows :
= 3.525 Ac = 2.012 Ae = 24.719
Aa = 0.606 A_hx = 0.081

mass delivered by the solution pump (kg/s) = 0.0562

finally the number of collectors needed are 52

COP = 0.893
000 40.000 15.000 65.000 30.000
: = 16289.975 J.D. B/C = 1.690 Qe = 45.000 (kW)

الملخص

دراسة الأمثلة لنظام تبريد شمسي يعمل على دورة إمتصاص محلول بروميد الليثيوم

للطالب : بشار مطيع إبراهيم أبو زهرة
إشراف : د. محمود حماد

تم في هذا البحث عمل دراسة أمثلة لنظام تبريد شمسي يعمل على دورة الإمتصاص باستخدام محلول بروميد الليثيوم بهدف التوصل إلى التصميم الأمثل لمثل هذه الأنظمة من الناحية الإقتصادية. و كانت متغيرات النظام التي تمت عليها الأمثلة تشمل درجات الحرارة في أجزاء النظام مثل درجة حرارة المبخر و المكثف و المولد ، بالإضافة إلى قدرة تبريد الجهاز.

و قد خلصت الدراسة إلى أن التصميم الأمثل لهذا النوع من الأنظمة في الأردن يكون بقدرة ٣٠ كيلوواط تبريد و يعمل عند درجة حرارة ٧٠ م في المولد، ٣٥ م في المكثف، و ١٥ م في المبخر، و ذلك على اعتبار شدة الإشعاع الشمسي في حدود ٨٠٠ واط/م^{٢}، حيث بلغت نسبة الفائدة على الكلفة للمشروع ما مقداره ١,٧٨٦ في التصميم الأمثل.