

## WORKING SUBSTANCE SELECTION FOR A SOLAR POWERED PUMP IN TOSHKE ENVIRONMENT

اختيار المادة العاملة لمضخة تدار بالطاقة الشمسية في منطقة توشكى

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### خلاصة البحث :

في هذا البحث تم عمل دراسة تحليلية لمضخة تدار بالطاقة الشمسية، وتستعمل دورة رانكن البسيطة لتحويل الطاقة الشمسية الممتصة بواسطة سخان شمسي مسطح إلى الطاقة الميكانيكية اللازمة لإدارة المضخة. ولذلك تم عمل مقارنة بين مختلف موانع التشغيل من حيث وجهة النظر للترموديناميكية لاختيار أفضلها للعمل في دورة رانكن كمادة عاملة. وقد تم إجراء البحث باستخدام الظروف الجوية وقيم الإشعاع الشمسي الساقط على منطقة توشكى بفرض الاستفادة من نتائجه في مشروع توشكى. حيث تم استخدام معادلات الاتزان الحراري لكل من السخان الشمسي والدورة لحساب القدرة المولدة وكفاءة النظام ككل عند جميع أشهر السنة. وقد أجريت الحسابات أيضاً على الظروف المناخية للصيف والشتاء والسنة الكاملة. وقد أظهرت النتائج أن موانع التشغيل الأمونيا (R717) يعتبر الأفضل، حيث أنه كمتوسط على مدار السنة يولد أكبر قدرة عند استخدامه وهي حوالي ١٦.٣ وات لكل متر مربع من السخان الشمسي عند أقصى كفاءة حرارية للنظام وهي ٢.٤٣% وذلك لاستخراج ٧٢٩ متر مكعب من الماء في السنة من عمق ٣٥ متر من باطن الأرض. هذا وقد لوحظ تقارب النتائج لموانع التشغيل الأخرى، حيث كان أقلها وهو R113 الذي تقل نتائجه %٤ فقط عن R717.

### ABSTRACT

In the present work, thermal analysis of a solar powered pump is carried out. Simple Rankine cycle is employed for converting the solar energy collected from a flat plate collector to produce the necessary mechanical power to drive the pump. Comparison between different refrigerants from thermodynamic view is performed to select the most suitable refrigerant to be used with the cycle as a working substance. The analysis is performed with the meteorological data and incident solar radiation in Toshke environment, to use its results in the Toshke project. The energy balance equations for both the solar collector and the cycle are used to obtain the produced power and the system thermal efficiency, at all months of the year. Analysis is also, made for the meteorological data of Summer, Winter and all-over the year. Results have shown that the Ammonia (R717) yields, as an average over the year, a maximum work of 16.3 W/m<sup>2</sup> at a maximum efficiency of 2.43% when used as a working substance in Toshke environment. However, other refrigerants show close results, where R113 results, which are the lowest, are only 4% lower than of R717.

### KEY WORDS

Solar powered pump, refrigerant, working substance selection.

### INTRODUCTION

Water is considered the basic requirement for the human live. Under-ground water may be the only available source in some rural places. This needs pumps for lifting. Small pumps are preferred by small farmers to keep their needs of domestic purposes and irrigation.

Conventional fuels or electric power may even not be available in these places. Solar energy is seen to be the most suitable choice for these situations. The efficiency of a solar powered pump depends mainly on the temperatures of the hot and cold reservoirs ( $T_H$  and  $T_L$ ). Prototype solar water pumps are designed, fabricated and tested by many investigators. A solar thermal water pump with n-pentane as a working fluid is described by K. Sumathy et al [1]. The pump has been designed, fabricated and tested with a flat-plate collector having an exposed area of  $1 \text{ m}^2$  coupled to the pump. Experimental results have shown that the pump can lift 250 Liters of water per day through a head of 8 m, with an overall efficiency of 0.12%. Also, a prototype solar powered pump is examined by K. Spindler et al [2]. The pump works on an organic Rankine cycle using refrigerant R113 as a working substance. Based on the working conditions, Carnot efficiency has a value between 11 and 13 %, and the Rankine efficiency lies between 7 and 9 %, while the recorded overall system efficiency was about 0.1-0.2%. Another prototype solar powered pump working on a low temperature organic Rankine cycle with refrigerant R11 has been described by V.V.N. Kishore et al [6]. Results have shown that the pump can extract about  $6.5 \text{ m}^3/\text{day}$  of water from a depth of 11.2 m, with an overall efficiency of 0.45 %. The pump is driven by the heat collected from a single glazed flat-plate collector with an area of  $7 \text{ m}^2$ .

During the last two decades investigations on some unconventional pumps are carried out. A solar power diaphragm pump working on a Rankine cycle with R113 as a working medium is described by R. Burton [7]. The diaphragm pump was extracting water from a head of 3 m with a flow rate about  $0.9 \text{ m}^3/\text{hr}$ . The pump is operated 5 hours per day with an average overall efficiency of 0.21 %.

It can be noticed from the previous literature that, the efficiency of solar water pumps being in the order of 0.1 %. Despite of its poor efficiency, solar driven pump is seen to be the best choice in rural areas because solar radiation is high. Also, rural areas are dispersed and deprived of conventional sources of energy due to energy shortage or the availability of energy restricted for economic reasons. It can also be observed that in the previous researches, the choice of a refrigerant as a working substance for a specific design has taken little attention. Therefore, this work is concerned with this point by performing a theoretical analysis to optimize a suitable refrigerant which can be used in a solar powered pump in Toshke environment.

### The Operating Cycle

Solar pump consume mechanical energy to lift underground water, which may be produced by direct or indirect (thermodynamic) conversion methods from solar radiation. Direct conversion methods include photovoltaic, thermoelectric and thermoionic processes which are expensive specially when used in large scale applications. Thermodynamic cycles (Rankine, Brayton or Stirling) can also be used to convert the internal energy of the working fluid (which is obtained from solar radiation) to mechanical work. Brayton and Stirling cycles utilize air (gas phase) as a working fluid. Thermal efficiency of gas power cycles is much lower than that of vapor power cycles. Therefore, Rankine cycle with liquid refrigerants is more suitable than any other cycle. Liquid refrigerants are preferred to be used with solar operated pumps working on Rankine cycle since the temperatures and pressures are more suitable in solar energy applications. A review of previous work shows that, most researches are based on the simple Rankine cycle [1,2,3,6]. Therefore, the solar driven pump analyzed in this work is chosen to operate on a simple Rankine cycle with a liquid refrigerant as a working

fluid. Carrying out the analysis with fixed cycle parameters, gives a suitable basis for comparison between different refrigerants.

### THEORETICAL APPROACH

Figure (1), shows a Schematic diagram of a solar powered pump operating on a simple Rankine cycle. The system is connected to a flat plate solar collector to supply the necessary heat to the Rankine cycle. The main design parameters which affect the performance of a solar operated Rankine cycle are:

1- External fluid temperatures [ $T_H$  and  $T_L$ ]:

The efficiency of Rankine cycle increases with increasing  $T_H$  and decreasing  $T_L$ . The selection of  $T_H$  is controlled by the solar collection system that will be used, while the selection of the  $T_L$  is governed by the available heat sink temperature. The extracted underground water is chosen as a heat sink, since it has the lowest temperature in Toshke environment.

2- The overall fluid-to-fluid conductance [ $(UA)_H$  and  $(UA)_L$ ]:

Increasing the values of  $(UA)$  improves the performance of Rankine system but also, increase the capital cost of the system.

3- The external fluids thermal capacities [ $(m C_p)_H$  and  $(m C_p)_L$ ]:

Increasing the thermal capacity of the heating fluid increases  $T_H$ , and increasing the thermal capacity of the cooling fluid lowers  $T_L$ . This results to an improvement of the system performance, but increasing the pumping cost. Therefore, an optimum value for thermal capacity must be searched.

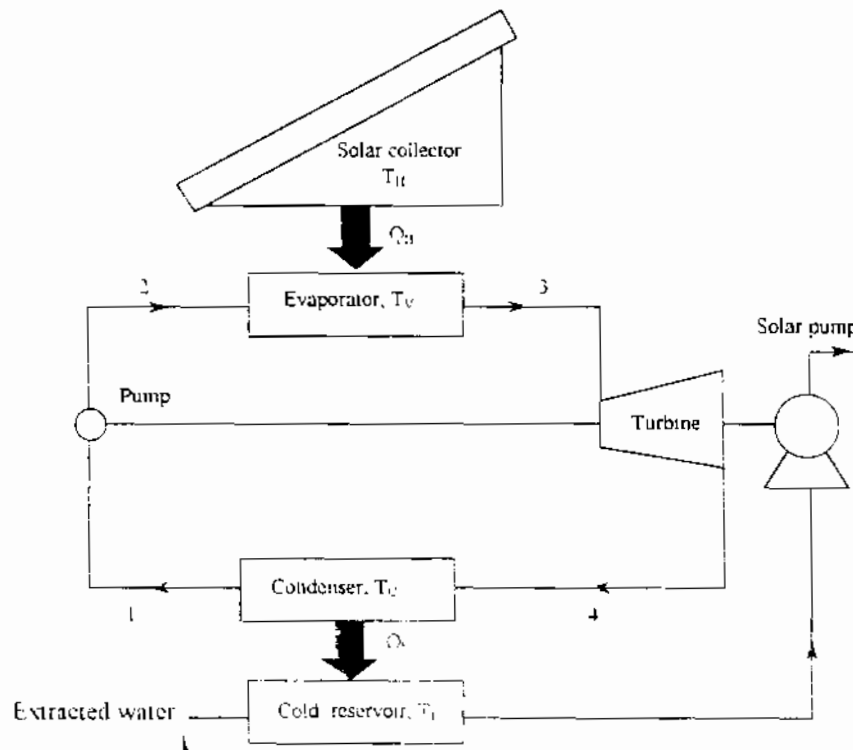


Figure (1) Schematic diagram of a solar powered pump

+ Turbine and pump internal efficiencies [ $\eta_t$  and  $\eta_p$ ]:

The deviation of expansion and compression through the turbine and pump respectively from that of isentropic process decreases the performance of the Rankine cycle. Minimizing this deviation is only a technological problem.

To select a suitable refrigerant as a working fluid in Rankine cycle for Toshke environment, it is important to compare between them from the thermodynamic behavior point of view. The selection of a certain refrigerant depends on many criteria, the most important of which is its critical temperature and pressure. Refrigerant having a higher critical temperature (at a lower pressure) is preferred, because it gives a chance to increase  $T_H$  to a higher value without the need of increasing the pumping work. Also, the slope of saturation lines on the pressure-enthalpy chart and the latent heat,  $h_{fg}$  at a given pressure play an important role for the choice of a suitable refrigerant. Some other factors such as the environmental pollution potential and toxicity should be considered. Classification of some commercially available refrigerants are presented with the critical pressure and temperature in Table (1). Saturation pressures are also given at some selected temperatures. Hydrofluorocarbon (HFC) does not contain chlorine, therefore, it has an Ozone depletion potential of zero. To reduce pollution effects, HFC refrigerants must be used as a replacement for chlorofluorocarbon (CFC) refrigerants for many applications. For example R134a may be as a substitute for R12.

Refrigerant type	Critical temperature, °C	Critical pressure, bar	Pressure at 52.1 °C bar	Pressure at 21 °C bar
<b>1-CFC -Chlorofluorocarbon- Refrigerants</b>				
R12	112	41.58	12.82	5.84
<b>2-HCFC -Chlorinated- Refrigerants</b>				
R22	96	49.88	20.37	9.37
R113	214.1	34.37	1.17	0.38
<b>3-HFC -Chlorine free-Refrigerants</b>				
R134a	101	40.67	14.24	6.03
R152a	113.5	44.92	12.5	5.32
<b>4-Halogen free Refrigerants</b>				
R290	97	42.54	18.04	8.63
R600a	135	36.45	7.28	3.15
R717	133	113	21.49	8.84

Table (1) Characteristics of CFC -Chlorofluorocarbon- refrigerants and their alternatives.

The thermal efficiency of the solar powered pump,  $\eta_p$ , increases while the solar collector efficiency,  $\eta_{coll}$ , decreases as  $T_H$  increases. This means that there is an optimum value for  $T_H$  which affects the choice of a suitable refrigerant. To determine the maximum possible value of  $T_H$  in Toshke area, it is required to know the meteorological data, which are collected by the authors from Aswan observatory. The monthly average daily incident total solar radiation on a horizontal surface per square meter,  $I$  and the ambient temperature  $T_a$  for all months of the year are presented in Table (2). According to reference [8], if the monthly clearness index, which is the ratio between terrestrial and extraterrestrial horizontal solar radiation intensities, is assumed to be 0.8 in Toshke environment, the ratio between diffuse horizontal solar radiation component,  $I_d$ , to the horizontal total solar radiation,  $I$  would be

about 0.1. Both horizontal solar radiation components: beam  $I_b$  and diffuse  $I_d$  are calculated and presented in the same table.

Months of the year	Ambient temp., °C	Horizontal total radiation $I$ , W/m <sup>2</sup>	Beam component, $I_b$ , W/m <sup>2</sup>	Diffuse component, $I_d$ , W/m <sup>2</sup>
January	22.5	626.5	563.8	62.6
February	25	675.3	607.8	67.5
March	30	617	555.3	61.7
April	35	542.7	488.4	54.3
May	38	512.8	461.5	51.3
June	40	617.3	555.6	61.7
July	40	581.5	523.4	58.1
August	40	609.8	548.8	61
September	37.5	595.4	535.9	59.5
October	35	601.6	541.5	60.2
November	27.5	652.8	587.5	65.3
December	23.5	587	528.3	58.7
Summer	39.5	580.4	522.4	58
Winter	24.6	635.4	571.9	63.5
Year (av.)	33	601.6	541.4	60.2

Table (2) Monthly average daily meteorological data for Toshke environment.

The solar radiation heat gain,  $Q_s$  per unit area of the flat plate collector can be calculated from.

$$Q_s = I_t (\alpha\tau) - U (T_H - T_a) \quad (1)$$

and the collector efficiency is defined as,

$$\eta_{coll} = \left( \frac{Q_s}{I_t} \right) = \left[ (\alpha\tau) - \left( \frac{U}{I_t} \right) (T_H - T_a) \right] \quad (2)$$

For convenience, the analysis is carried out per unit collector area, with the following reasonable assumptions:

The collector is south facing with a tilt angle  $\beta = 30^\circ$  to the horizontal.

The approximate values of  $\alpha$  and  $\tau$  are 0.9 and 0.8 respectively.

The collector overall heat transfer coefficient,  $U$  is equal to  $5 \text{ W/m}^2 \cdot ^\circ\text{C}$ .

The solar collector efficiency is about 55 %.

Neglecting site reflections, the total monthly average daily solar radiation on the collector surface,  $I_t$  can be calculated from,

$$I_t = R_b I_b + R_d I_d \quad (3)$$

where,  $R_b$  and  $R_d$  are the tilt factors for the solar beam component,  $I_b$  and the diffuse component,  $I_d$  respectively, and can be estimated from [8] as,

$$R_b = \frac{\cos(L - \beta) \cos \delta \sin h_w + h_w \sin(L - \beta) \sin \delta}{\cos L \cos \delta \sin(h_w)_0 + (h_w)_0 \sin L \sin \delta} \quad (4)$$

$$R_d = \cos^2\left(\frac{\beta}{2}\right) \quad (5)$$

where,  $L$  is the latitude angle of Toshke area = 23° N,

$h_w$  is the sunrise monthly average daily hour angle,

$$h_w = 15 (12 - \text{local sunrise, hour}) \quad (6)$$

where, Local sunrise, hour =  $(h_w)_0 / 15$

$$(h_w)_0 = \cos^{-1} [-\tan L \tan \delta]$$

$(h_w)_0$  is the local sunrise hour angle at sea level (or at solar-altitude angle = 0)

$\delta$  is the declination angle at Toshke and can be calculated from,

$$\delta = 23.45 \sin \left[ \frac{360}{365} (284 + N) \right] \quad (7)$$

where,  $N$  is day number of the year from January first.

The solar heat gain,  $Q_s$  is transferred as  $Q_H$  to the evaporator via collector-system heat exchanger, as follow,

$$Q_H = Q_c \eta_{HE} = m_R (h_1 - h_2) = (UA)_{HE} \theta_{HE} \quad (8)$$

where,  $(h_1 - h_2)$  is the enthalpy difference across the evaporator

$\eta_{HE}$  is the heat exchanger efficiency.

$\theta_{HE}$  is the logarithmic mean temperature difference between the collector fluid and the refrigerant.

Since the monthly average daily evaporator temperature  $T_e$  is almost constant, and  $T_H$  is the monthly average daily temperature of the collector system, then  $\theta_{HE}$  can be considered as the temperature difference  $(T_H - T_e)$ . The effectiveness for evaporator and condenser is equal to  $[1 - \exp(-NTU)]$ .

where,  $NTU$  is the number of transfer units  $[(UA)_{HE} / (m C_p)_{ev}]$

Water flow rate in the flat plate collector can be assumed about 40 kg/hr.m<sup>2</sup>, as mentioned in the previous work [9].

Also, the monthly average daily temperature of the condenser,  $T_c$  is constant and hence the logarithmic mean temperature difference between the extracted water and the refrigerant  $\theta_c$  can be considered as the temperature difference  $(T_c - T_e)$ . The extracted water temperature, from a depth of 35 m, can be assumed 15 °C in the winter and 20 °C in the summer. This assumption is a reasonable assumption according to the data published about Toshke environment [10].

On the other hand, the heat rejected from the refrigerant in the condenser to the extracted water,  $Q_L$ , can be obtained from,

$$Q_L = \dot{m}_R (h_4 - h_1) = (UA)_L \theta_L \quad (9)$$

where,  $\dot{m}_R$  is the refrigerant flow rate which can be obtained from:

$$\dot{m}_R = Q_H / (h_1 - h_2) \quad (10)$$

Using the above equations with the previous reasonable assumptions and the recorded meteorological data for Toshke environment presented in Table (2), the input parameters of the operating cycle:  $Q_H$ ,  $T_H$ ,  $T_L$ ,  $T_V$  and  $T_C$  can be obtained as an average during each month of the year.

Carnot cycle efficiency,  $\eta_C$  of the system can be calculated from,

$$\eta_C = 1 - (T_L / T_H) \quad (11)$$

Referring to Fig. (1), Rankine cycle efficiency  $\eta_R$  can be calculated as,

$$\eta_R = [(h_3 - h_4) - (h_2 - h_1)] / [h_3 - h_2] \quad (12)$$

Values of enthalpies  $h_1$ ,  $h_2$ ,  $h_3$  and  $h_4$  are obtained from the pressure-enthalpy chart for each refrigerant. Figure (2), shows a plot of the cycle on the pressure-enthalpy chart for each of the refrigerants mentioned in Table (1).

The net cycle work,  $W$  per unit collector area is calculated as,

$$W = \dot{m}_R [(h_3 - h_4) - (h_2 - h_1)] \quad (13)$$

## RESULTS AND DISCUSSION

Equations from 1 to 9 are solved with the help of the meteorological data (Table 2) to obtain the values of  $Q_H$ ,  $T_H$ ,  $T_V$ ,  $T_L$  and  $T_C$  as a daily average during each month of the year. Results are also, given numerically in Table (3) for Summer, Winter and all-over the year.

Months of the year	Input heat, $Q_H$	$T_H$ , °C	$T_V$ , °C	$T_L$ , °C	$T_C$ , °C
January	405	53.8	48.4	15	18
February	388.5	55	49.8	15	18
March	300.3	53.2	49.2	15	18
April	223.6	52.3	49.3	20	24
May	185.7	52.4	49.9	20	24
June	209.2	56.1	53.3	20	24
July	202.3	55.6	52.9	20	24
August	236.8	58.3	55.1	20	24
September	270.3	58.4	54.7	20	24
October	322.5	59.9	55.6	15	18
November	406.8	58.9	53.5	15	18
December	393.9	53.9	48.6	15	18
Summer	208.5	56.1	53.3	20	24
Winter	398.6	55.8	50.5	15	18
All the year	295.4	56	52.1	17.5	21

Table (3) Calculated values of  $Q_H$ ,  $T_H$ ,  $T_V$ ,  $T_L$  and  $T_C$  for Toshke environment.

Equations from 10 to 13 are used with the help of data in Table (3) and the Rankine cycle plotted on p-h chart in Fig. (2) to calculate the net cycle work,  $W$  ( $W/m^2$ ), Rankine cycle efficiency,  $\eta_R$  (%) and the refrigerant flow rate,  $m_R$  ( $Kg/s.m^2$ ). Results are given in Table (4) for different refrigerants in each month of the year. Results are also, giveu in the same table for Summer, Winter and all-over the year. Carnot cycle efficiency is constant for all refrigerants and is equal to 68.7 %.

The overall system efficiency,  $\eta_O$  can be obtained from the following formula:

$$\eta_O = \eta_{coll} \eta_{HE} \eta_R \eta_m \eta_p \quad (14)$$

The following reasonable assumptions can be considered :

The heat exchanger efficiency,  $\eta_{HE} = 80$  %

The system mechanical efficiency,  $\eta_m = 80$  %

The pump efficiency,  $\eta_p = 75$  %

While, the overall system power output,  $W_O$  can be estimated from,

$$W_O = I \eta_O \quad (15)$$

The overall system power output can be used to determine the amount of extracted water,  $G_w$  from a depth of 35 m as follows,

$$G_w = W_O / (\rho g H) \quad (16)$$

where,  $g$  = acceleration of gravity ( $9.81 \text{ m/s}^2$ )

$H$  = water level depth (35 m)

$\rho$  = water density

Therefore, both the monthly average daily overall system power output,  $W_O$  and the overall system efficiency,  $\eta_O$  can be calculated using equations 14 and 15 and presented in Table (5) for different refrigerants. The corresponding amount of extracted water per month is also, calculated and given. The total sum of the average values all-over the year is also estimated and given in the same table. It is clear from results that the overall system power output and the overall system efficiency for R717 have higher values compared to the other refrigerants. Due to the extraordinary high enthalpy difference for R717, the circulating refrigerant mass flow rate,  $m_R$  has the lowest value compared to the other refrigerants, as shown in Table (4). On the other hand, the refrigerant R113 operating pressures are the lowest compared with that of the other refrigerants as shown in Table (3). Therefore, R113 results are the lowest. A plot of the monthly average daily overall system power output, for different months, using these two refrigerants is given in Fig. (3), while Fig. (4) shows the overall system efficiency for them. Both the overall system power output and efficiency vary with time of the year, having lower values in summer months and higher values otherwise. However, the total monthly average daily energy is larger in summer due to the larger length of the sunshine hours in Summer. Results in Table (5) have shown that the Ammonia R717 yields a power output of  $16.3 \text{ W/m}^2$  and a system overall efficiency of 2.43 % as an average over the year. Using this refrigerant, the amount of extracted water (from a depth of 35 m) would be about  $729 \text{ m}^3/\text{year}$ . Figure(3) shows that the difference between the power output from R717 and R113 is about 4 %. Also, Fig. (4) shows that the difference between the system overall efficiency from R717 and R113 is about 4 %. This means that the results of the chosen refrigerants are close. Therefore, the choice of any of them as an operating fluid for the system may only depends on the environmental, technological or economical aspects.



## CONCLUSIONS

Thermal analysis of a solar powered pump, operated with a simple Rankine cycle, which convert the solar energy collected from a flat plate collector to the necessary mechanical power to drive the pump. Comparison between different refrigerants from the thermodynamic point of view is performed to select the most suitable one to be used with the cycle as a working substance. The analysis is based on the meteorological data and incident solar radiation for Toshke environment, in order to use its results in the Toshke project. The energy balance equations for both the solar collector and the cycle are used to obtain the produced power and the system thermal efficiency, at all months of the year. Analysis is also made for the meteorological data of Summer, Winter and all-over the year. Results have shown that the Ammonia refrigerant R717 yields a maximum power output (about  $16.3 \text{ W/m}^2$ ) and a maximum system overall efficiency (about 2.43 %) when used as a working substance in Toshke environment. Also, the amount of extracted water, from a depth of 35 m, per year is about  $729 \text{ m}^3/\text{year}$ . Results for the selected refrigerants are also close, where the difference between the output power between them is only about 4 %. Therefore, the choice of any of them as an operating fluid for the system may only depends on the environmental, technological or economical aspects

## NOMENCLATURE

A	: Area	$(\text{m}^2)$
$C_p$	: Specific heat at constant pressure	$(\text{J/kg} \cdot ^\circ\text{C})$
$G_w$	: Amount of extracted water per month	$(\text{m}^3)$
h	: Enthalpy	$(\text{J/kg})$
$I_t$	: Total solar radiation intensity	$(\text{W/m}^2)$
L	: Latitude angle	(degrees)
m	: Mass flow rate	$(\text{kg/s})$
NTU	: Number of transfer units	
Q	: Heat gain per unit area	$(\text{W/m}^2)$
$Q_s$	: Solar radiation heat gain per unit area	$(\text{W/m}^2)$
$R_b$	: Tilt factor for solar beam component	-
$R_d$	: Tilt factor for solar diffuse component	-
U	: Overall heat transfer coefficient,	$(\text{W/m}^2 \cdot ^\circ\text{C})$
W	: Net cycle work per unit area	$(\text{W/m}^2)$
$W_{(t)}$	: Overall system power output per unit area	$(\text{W/m}^2)$

### Greek symbols

$\alpha$	: Absorptivity of the collector absorber plate	-
$\beta$	: Collector tilt angle	(degrees)
$\delta$	: Declination angle	(degrees)
$\eta$	: Efficiency	%
$\eta_c$	: Carnot cycle efficiency	%
$\eta_R$	: Rankine cycle efficiency	%
$\theta$	: Logarithmic mean temperature difference	$(^\circ\text{C})$
$\rho$	: Water density	$(\text{kg/m}^3)$
$\tau$	: Transmissivity of the collector glass cover	-

**SUBSCRIPTS**

a	ambient
c	condenser
H	hot
HE	heat exchanger
c	cold
m	mechanical
O	overall
P	pump
R	refrigerant
v	evaporator
w	water

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Ref.	R12			R22			R113			R134a		
	W	$\eta_R$	$m_R$	W	$\eta_R$	$m_R$	W	$\eta_R$	$m_R$	W	$\eta_R$	$m_R$
1	36	8.88	2.61	37	9.14	2.11	35.4	8.76	2.38	35.4	8.73	2.04
2	35.8	9.2	2.5	36.3	9.33	1.99	32.3	9.07	2.28	35.2	9.06	1.95
3	27.3	9.08	1.93	27.6	9.2	1.54	26.8	8.94	1.76	26.8	8.92	1.51
4	16.8	7.49	1.5	17	7.6	1.19	16.4	7.3	1.35	16.5	7.39	1.17
5	14.2	7.64	1.24	14.4	7.74	0.99	13.9	7.48	1.12	14	7.53	0.97
6	17.7	8.46	1.39	18	8.59	1.11	17.4	8.3	1.25	17.4	8.33	1.09
7	17	8.38	1.34	17.2	8.5	1.07	16.6	8.21	1.21	16.7	8.24	1.05
8	21.1	8.89	1.56	21.4	9.01	1.25	20.6	8.73	1.4	20.7	8.75	1.23
9	23.8	8.8	1.78	24.1	8.93	1.43	23.3	8.64	1.6	23.4	8.65	1.4
10	33.9	10.5	2.05	34.2	10.6	1.64	33.5	10.4	1.85	33.3	10.3	1.6
11	40.3	10.1	2.56	41.5	10.2	2.07	39.8	9.93	2.32	40.2	9.88	2.02
12	35.2	8.92	2.54	35.6	9.04	2.02	34.7	8.8	2.32	34.6	8.78	1.98
S	17.7	8.47	1.38	17.9	8.59	1.11	17.3	8.3	1.24	17.4	8.33	1.09
W	37.4	9.38	2.56	37.9	9.51	2.04	36.8	9.24	2.33	36.8	9.22	2
Year	26.5	8.97	1.93	26.9	9.1	1.54	26.1	8.82	1.74	26.1	8.82	1.51

Ref.	R152a			R290			R600a			R717		
	W	$\eta_R$	$m_R$	W	$\eta_R$	$m_R$	W	$\eta_R$	$m_R$	W	$\eta_R$	$m_R$
1	36	8.88	1.31	36	8.9	1.07	35.2	8.69	1.06	36.7	9.06	0.34
2	35.8	9.21	1.25	35.8	9.22	1.02	35.9	9.26	1.01	36.6	9.42	0.32
3	27.2	9.08	0.97	27.3	9.09	0.79	26.7	8.88	0.78	27.8	9.26	0.25
4	16.7	7.46	0.74	16.8	7.53	0.62	16.4	7.33	0.61	17	7.61	0.19
5	14.1	7.61	0.62	14.3	7.69	0.51	13.9	7.47	0.5	14.4	7.76	0.15
6	17.6	8.41	0.69	17.8	8.49	0.57	17.3	8.27	0.56	18.1	8.64	0.18
7	16.9	8.34	0.67	17	8.39	0.55	16.5	8.18	0.54	17.3	8.54	0.17
8	21	8.87	0.78	21.1	8.91	0.64	20.6	8.69	0.63	21.5	9.09	0.2
9	22.9	8.48	0.86	23.8	8.81	0.74	23.2	8.59	0.72	24.3	9	0.23
10	34	10.5	1.03	33.9	10.5	0.84	33.2	10.3	0.82	34.9	10.8	0.27
11	45	10.1	1.3	40.9	10.1	1.06	40	10.4	1.05	42	10.3	0.34
12	35.2	8.93	1.27	35.3	8.95	1.04	34.5	8.75	1.03	35.9	9.11	0.33
S	17.6	8.44	0.69	17.7	8.49	0.57	17.3	8.28	0.56	18	8.64	0.18
W	37.4	9.38	1.28	37.4	9.39	1.05	36.6	9.18	1.04	38.2	9.59	0.33
Year	26.5	8.96	0.96	26.6	8.98	0.79	25.9	8.77	0.78	27.1	9.16	0.25

Table (4) Calculated values of the net cycle work output ( $W$ ,  $W/m^2$ ), Rankine efficiency ( $\eta_R$ , %) and the required mass flow rate ( $m_R \times 10^{-1}$   $kg/s m^2$ ) for different refrigerants.

Ref.	R12			R22			R113			R134a		
	W <sub>O</sub>	η <sub>O</sub>	G <sub>w</sub>	W <sub>O</sub>	η <sub>O</sub>	G <sub>w</sub>	W <sub>O</sub>	η <sub>O</sub>	G <sub>w</sub>	W <sub>O</sub>	η <sub>O</sub>	G <sub>w</sub>
1	21.6	2.34	75.2	22.2	2.41	77.5	21.2	2.31	74	21.2	2.31	74
2	21.5	2.43	70.7	21.8	2.46	71.7	21.2	2.39	69.8	21.1	2.39	69.6
3	16.4	2.4	63	16.6	2.43	63.8	16.1	2.36	62	16.1	2.36	61.9
4	10.1	1.98	39.6	10.2	2	40.3	9.81	1.93	38.7	9.92	1.95	39.1
5	8.51	2.02	36.3	8.63	2.04	36.8	8.33	1.98	35.5	8.4	1.99	35.8
6	10.6	2.23	44.8	10.8	2.27	45.5	10.4	2.19	44	10.5	2.2	44.1
7	10.2	2.21	43.9	10.3	2.24	44.5	10	2.17	43.1	10	2.18	43.2
8	12.6	2.35	52.6	12.8	2.38	53.3	12.4	2.31	51.5	12.4	2.31	51.7
9	14.3	2.32	54.4	14.5	2.36	55.3	14	2.28	53.3	14	2.28	53.5
10	20.4	2.78	75.8	20.5	2.8	76.4	20.1	2.74	74.8	20	2.73	74.4
11	24.2	2.65	82.7	24.9	2.69	85.1	23.9	2.62	81.7	24	2.61	82.5
12	21.1	2.36	72.7	21.4	2.39	73.7	20.8	2.32	71.7	20.8	2.32	71.5
S	10.6	2.23		10.7	2.7		10.4	2.19		10.4	2.2	
W	22.4	2.47		22.7	2.51		22.1	2.44		22.1	2.43	
Year	15.9	2.37	713 m <sup>3</sup> /y	16.1	2.4	724 m <sup>3</sup> /y	15.6	2.33	720 m <sup>3</sup> /y	15.6	2.33	700 m <sup>3</sup> /y

Ref.	R152a			R290			R600a			R717		
	W <sub>O</sub>	η <sub>O</sub>	G <sub>w</sub>	W <sub>O</sub>	η <sub>O</sub>	G <sub>w</sub>	W <sub>O</sub>	η <sub>O</sub>	G <sub>w</sub>	W <sub>O</sub>	η <sub>O</sub>	G <sub>w</sub>
1	21.6	2.34	75.3	21.6	2.35	75.4	21.1	2.29	73.7	21	2.39	76.8
2	21.5	2.43	70.8	21.5	2.43	70.9	21.6	2.44	71.1	21.9	2.48	72.4
3	16.4	2.39	62.9	16.4	2.4	63	16	2.34	61.6	16.7	2.45	64.2
4	10	1.97	39.5	10.1	1.99	39.8	9.8	1.93	38.7	10.2	2	40.2
5	8.48	2.01	36.1	8.57	2.03	36.5	8.3	1.97	35.4	8.6	2.05	36.9
6	10.6	2.22	44.5	10.7	2.24	45	10.4	2.18	43.8	10.8	2.28	45.7
7	10.1	2.2	43.7	10.2	2.22	44	9.92	2.16	42.8	10.4	2.25	44.7
8	12.6	2.34	52.4	12.7	2.35	52.7	12.4	2.29	51.4	12.9	2.4	53.7
9	13.8	2.24	52.5	14.3	2.33	54.5	13.9	2.27	53.2	14.6	2.37	55.6
10	20.4	2.78	75.8	20.3	2.77	75.7	19.9	2.72	74.1	21	2.86	78
11	24.6	2.66	84	24.6	2.66	83.9	24	2.74	82.1	25.2	2.73	86.2
12	21.1	2.36	72.7	21.1	2.36	72.9	20.7	2.31	71.2	21.5	2.4	74.2
S	10.6	2.23		10.6	2.24		10.4	2.18		10.8	2.28	
W	22.4	2.47		22.5	2.48		21.9	2.42		22.9	2.53	
Year	15.9	2.36	710 m <sup>3</sup> /y	15.9	2.37	713 m <sup>3</sup> /y	15.6	2.33	699 m <sup>3</sup> /y	16.3	2.43	729 m <sup>3</sup> /y

Table (5) Calculated values of the overall system power output ( $W_{O}$ ,  $W/m^2$ ), system overall efficiency ( $\eta_{O}$ , %) and the amount of extracted water per month ( $G_w$ ,  $m^3$ /month) for different refrigerants.

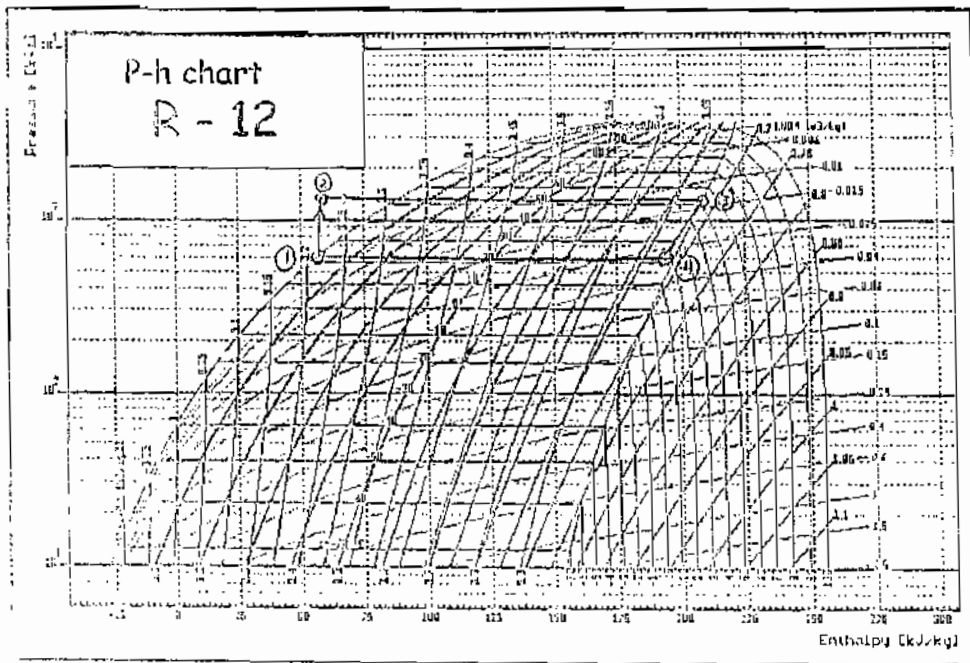


Fig. (2-a) P-h chart for R12

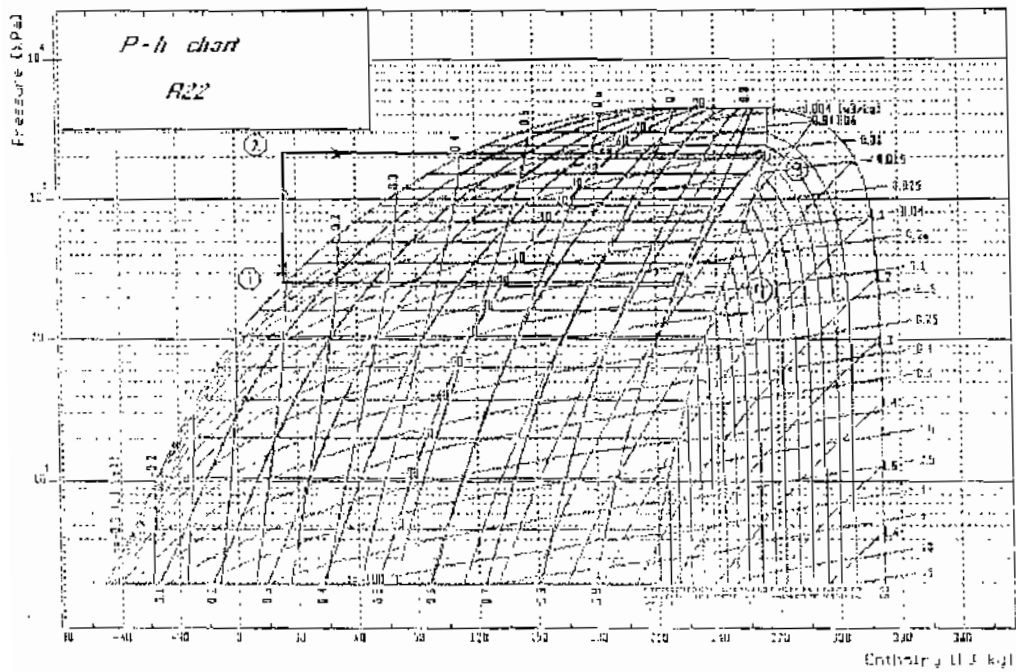


Fig. (2-b) P-h chart for R22

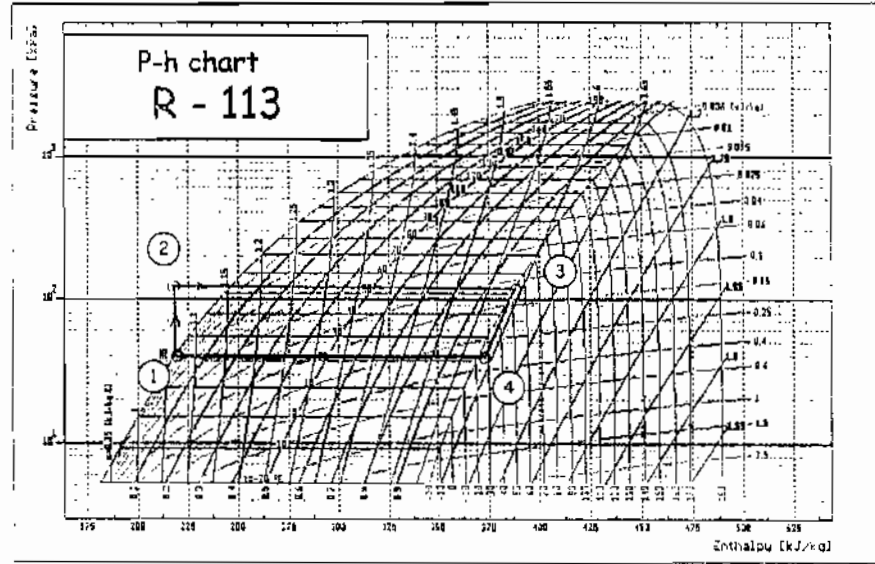


Fig. (2-c) P-h chart for R113

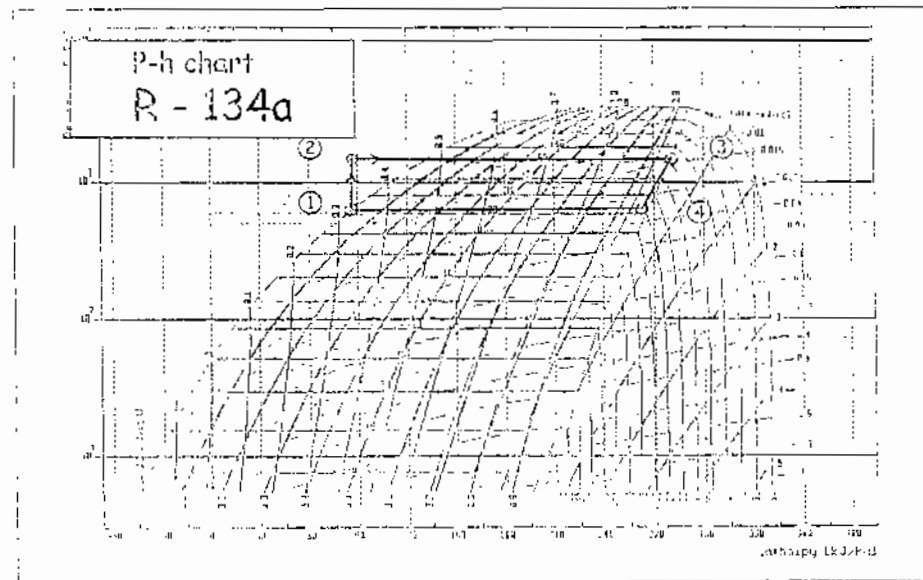


Fig (2-d) P-h chart for R134a

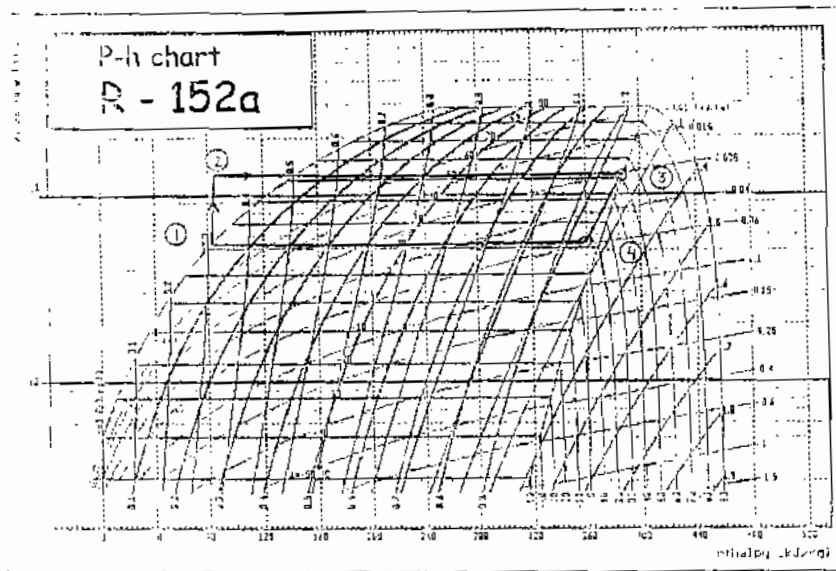


Fig. (2-c) P-h chart for R152a

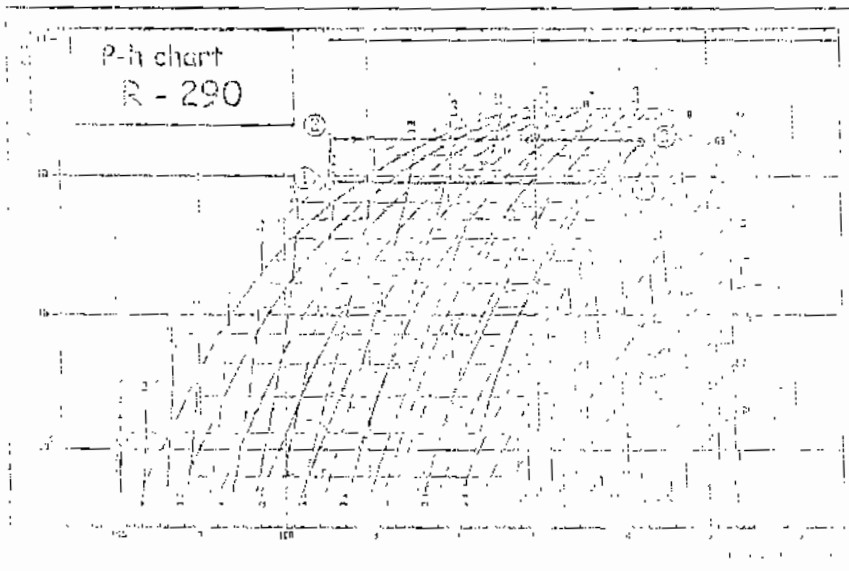


Fig. (2-d) P-h chart for R290

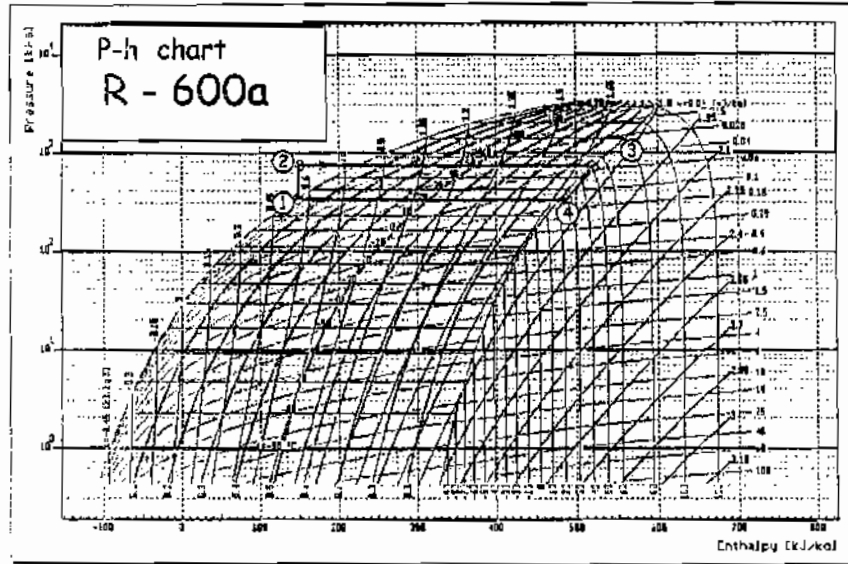


Fig. (2-g) P-h chart for R600a

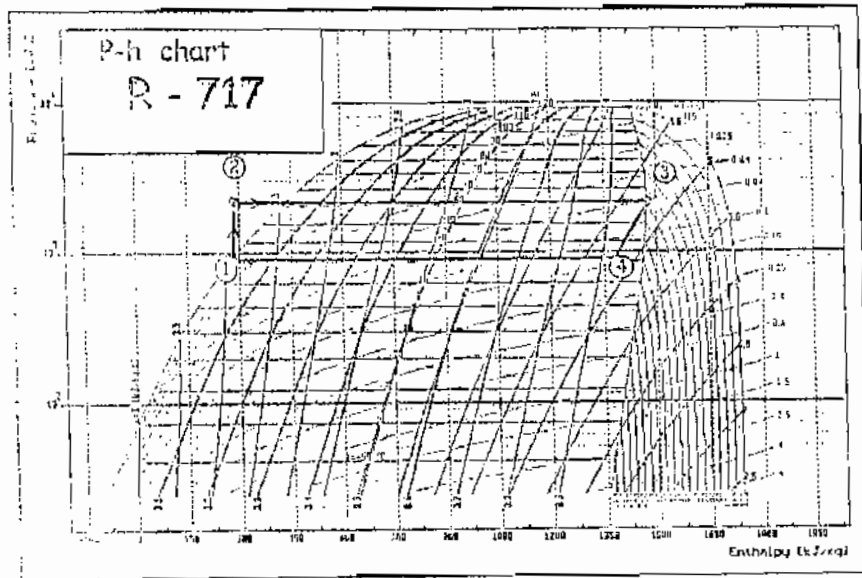


Fig. (2-h) P-h chart for R717

Fig. (2) Simple Rankine cycle plotted on P-h chart for different refrigerants



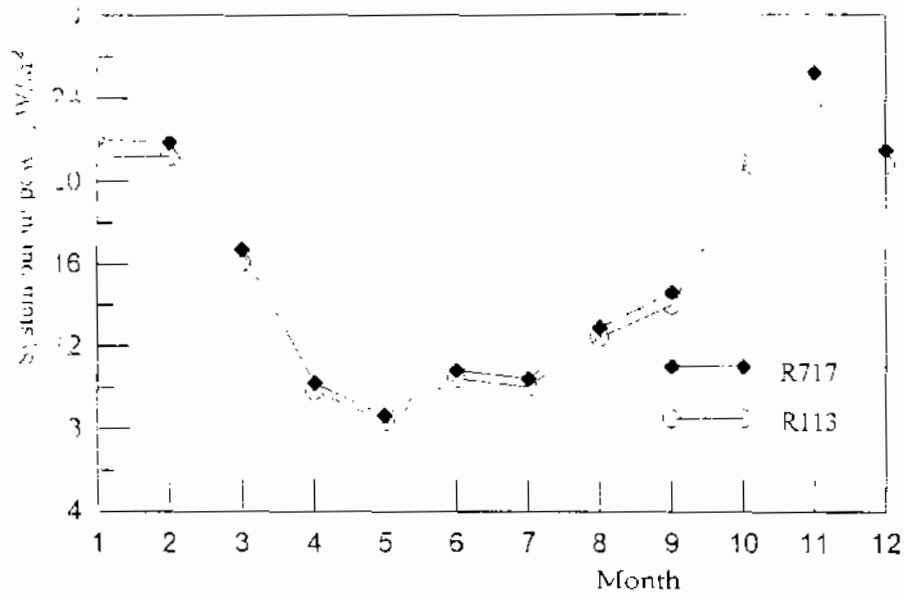


Fig. 3 Variation of the overall system output power during the year.

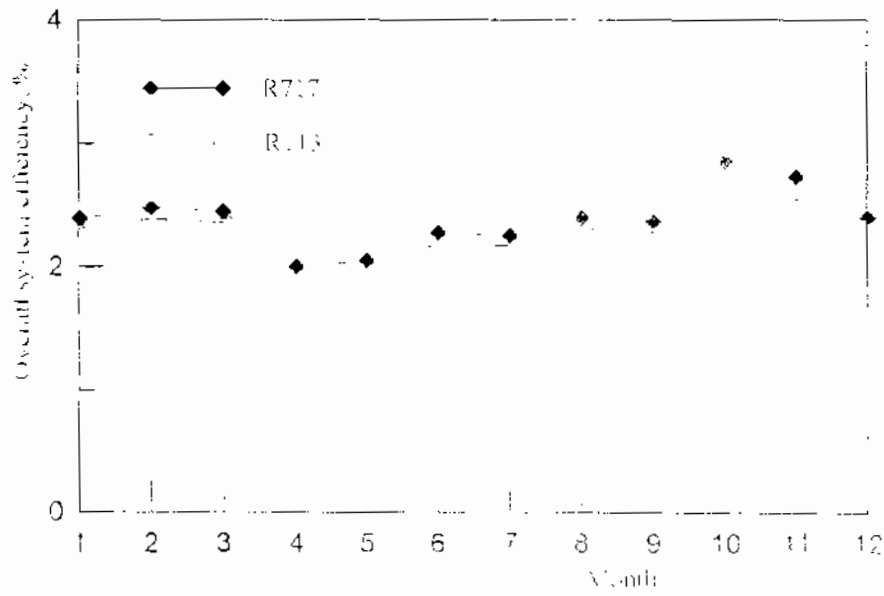


Fig. 4 Variation of the system overall efficiency during the year