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WORKING SUBSTANCE SELECTION FOR A SOLAR POWERED PUMP IN TOSHKE ENVIRONMENT

اختيار المادة العامله لمضخه تدار بالطاقة الشمسية فى منطقة توشكى

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خلاصة البحث :

فى هذا البحث تم عمل دراسة تحليلية لمضخة تدار بالطاقة الشمسية، وتستعمل دورة رانكن البسيطة لتحويل الطاقة الشمسية الممتصة بواسطة سخان شمسى مسطح إلى الطاقة الميكانيكية اللازمة لإدارة المضخة. ولذلك تم عمل مقارنة بين مختلف موانع التشغيل من حيث وجهة النظر الثرموديناميكية لاختيار أفضلها للعمل فى دورة رانكن كمادة عاملة. وقد تم إجراء البحث باستخدام الظروف الجوية وقيم الإشعاع الشمسى الساقط على منطقة توشكى بغرض الاستفادة من نتائجه في مشروع توشكي. حيث تم استخدام معادلات الاتزان الحرارى 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 مانع التشغيل الامونيا (R717) يعتبر الأفضل، حيث أنه كمتوسط على مدار السنة يولد أكبر قدرة عند استخدامه وهى حوالى ١٦،٣ وات لكل متر مربع من السخان الشمسى عند أقصى كفاءة حرارية للنظام وهى ٢،٤٣٪ وذلك لاستخراج ٧٢٩ متر مكعب من الماء فى السنه من عمق ٣٥ متر من باطن الأرض. هذا وقد لوحظ تقارب النتائج لموانع التشغيل الأخرى، حيث كان أقلها وهو R113 الذى 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 eonverting the solar energy collected from a flat plate colleetor to produce the necessary mechanical power to drive the pump. Comparison between different refrigerants from thermodynamic view is performed to seleet the most suitable refrigerant to be nsed with the cycle as a working substance. The analysis is performed with the meteorological data and incident solar radiation in Toshke enviroument, 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 nf Summer, Winter and all-over the year. Results bave shown that the Ammonia (R717) yields, as an average over the year, a maximum work of 16.3 W/m^2 at a maximum efficiency of 2.43% when used as a working substance in Tosbke 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.

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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 m^2 coupled to the pump. Experimental results have shown that the pump can lift 250 Litters 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 m^3/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 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 m^3/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 [$(\dot{m}\ C_p)_H$ and $(\dot{m}\ C_p)_L$]:
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.

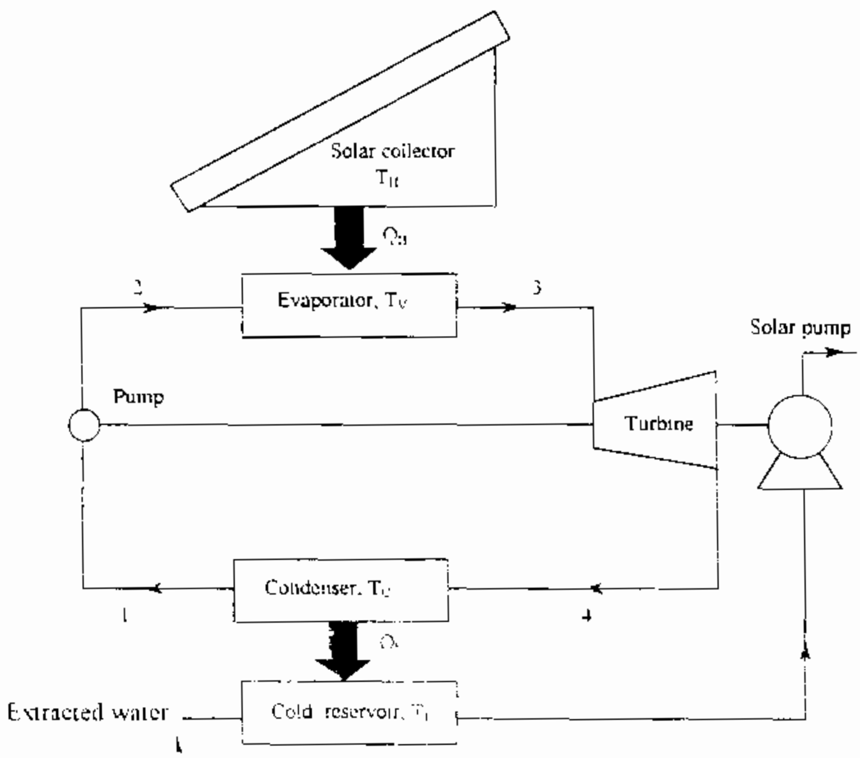

Figure (1) Schematic diagram of a solar powered pump

+ Turbine and pump internal efficiencies [η_t and η_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. Hydroflouorocarbon (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 choloroflouorocarbon (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
1-CFC -Cholorofiouorocarbon- Refrigerants				
R12	112	41.58	12.82	5.34
2-HCFC -Chlorinated- Refrigerants				
R22	96	49.88	20.37	9.37
R113	214.1	34.37	1.17	0.38
3-HFC -Chlorine free-Refrigerants				
R134a	101	40.67	14.24	6.03
R152a	113.5	44.92	12.5	5.32
4-Halogen free Refrigerants				
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 -Choloroflouorocarbon- refrigerants and their alternatives.

The thermal efficiency of the solar powered pump, η_r increases while the solar collector efficiency, η_{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 exraterrestial horizontal solar radiation intensites, 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^2	Beam component, I_b, W/m^2	Diffuse component, I_d, W/m^2
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.

$$Q_S = I_t\,(\alpha\tau) - U\,(T_H - T_a) \qquad (1)$$

and the collector efficiency is defined as,

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

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 $\beta = 30°$ to the horizontal.

The approximate values of α and τ are 0.9 and 0.8 respectively.

The collector overall heat transfer coefficient, U is equal to 5 W/m^2. °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,

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

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,

$$R_b = \frac{\cos(L-\beta)\cos\delta \sin h_{sr} + h_{sr}\sin(L-\beta)\sin\delta}{\cos L \cos\delta \sin(h_{sr})_o + (h_{sr})_o \sin L \sin\delta} \quad (4)$$

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

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

h_{sr} is the sunrise monthly average daily hour angle,

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

where, Local sunrise, hour = $(h_{sr})_o / 15$

$$(h_{sr})_o = \cos^{-1}[-\tan L \tan\delta]$$

$(h_{sr})_o$ is the local sunrise hour angle at sea level (or at solar-altitude angle = 0)

δ is the declination angle at Toshke and can be calculated from,

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

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,

$$Q_H = Q_S\ \eta_{HE} = m_R\ (h_1 - h_2) = (UA)_H\ \theta_H \quad (8)$$

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

η_{HE} is the heat exchanger efficiency.

θ_H is the logarithmic mean temperature difference between the collector fluid and the refrigerant.

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

where, NTU is the number of transfer units [$(UA)_H / (m\ C_p)_w$]

Water flow rate in the flat plate collector can be assumed about 40 kg/hr.m^2 , 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 θ_L can be considered as the temperature difference ($T_C - T_L$). 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,

$$Q_L = m_R (h_4 - h_1) = (UA)_L \ \theta_L \qquad (9)$$

where, m_R is the refrigerant flow rate which ean be obtained from:

$$m_R = Q_H / (h_3 - h_2) \qquad (10)$$

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, η_c of the system can he calculated from,

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

Referring to Fig. (1). Rankiue cyele efficiency η_R can be calculated as.

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

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,

$$W = m_R [(h_3 - h_4) - (h_2 - h_1)] \qquad (13)$$

RESULTS AND DISCUSSION

Equatious 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, η_R (%) and the refrigerant flow rate, $\dot{m}_R$ ($Kg/s.m^2$). 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, η_O can be obtained from the following formula:

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

The following reasonable assumptions can be considered :
The heat exchanger efficiency, η_{HE} = 80 %
The system mechanical efficiency, η_m = 80 %
The pump efficiency, η_P = 75 %

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

$$W_O = I_t\ \eta_O \qquad (15)$$

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,

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

where, g = acceleration of gravity (9.81 m/s^2)
H = water level depth (35 m)
ρ = water density

Therefore, both the monthly average daily overall system power output, W_O and the overall system efficieney, η_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 effieiency for R717 have higher valnes 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 tbe 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 efficieney for them. Both the overall system power outpnt and efficiency vary with time of the year, having lower values in summer months and higher values otherwise. However, the total mouthly average daily energy is larger in summer due to the larger length of the sunshine hours in Summer. Results in Tahle (5) have shown that the Ammonia R717 yields a power output of 16.3 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 m^3/year. Figure(3) shows that the difference between the power output from R717 and R113 is about 4 %. Also, Fig. (4) shows that the differeuce 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 meteorologieal 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 eycle are used to obtain the produeed power and the system thermal efficieney, at all months of the year. Analysis is also made for the meteorologieal 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 W/m^2) and a maximum system overall effieiency (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 m^3/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 environmeutal, technologieal or eeonomical aspects

NOMENCLATURE

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

Greek symbols

α	: Absorbitivity of the collector absorher plate	–
β	: Collector tilt angle	(degrees)
δ	. Deelinarion angle	(degrees)
η	. Efficieucy	%
η_C	· Carnot cycle efficiency	%
η_R	: Rankine cyele efficiency	%
θ	: Logarithmic mean temperature difference	(°C)
ρ	: Water density	(kg/m^3)
τ	: Transimisivity of the collector glass cover	–

SUBSCRIPTS

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

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Ref.	R12			R22			R113			R134a		
Month	W	η_R	m_R	W	η_R	m_R	W	η_R	m_R	W	η_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		
Month	W	η_R	m_R	W	η_R	m_R	W	η_R	m_R	W	η_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 (η_R, %) and the required mass flow rate (m_R $\times 10^{-3}$ $kg/s\ m^2$) for different refrigerants.

Ref.	R12			R22			R113			R134a		
Month	W_O	η_O	G_w	W_O	η_O	G_w	W_O	η_O	G_w	W_O	η_O	G_w
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^3/y	16.1	2.4	724 m^3/y	15.6	2.33	700 m^3/y	15.6	2.33	700 m^3/y

Ref.	R152a			R290			R600a			R717		
Month	W_O	η_O	G_w	W_O	η_O	G_w	W_O	η_O	G_w	W_O	η_O	G_w
1	21.6	2.34	75.3	21.6	2.35	75.4	21.1	2.29	73.7	22	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^3/y	15.9	2.37	713 m^3/y	15.6	2.33	699 m^3/y	16.3	2.43	729 m^3/y

Table (5) Calculated values of the overall system power output (W_O, W/m^2), system overall efficiency (η_O %) and the amount of extracted water per month (G_w, m^3/month) for different refrigerants.

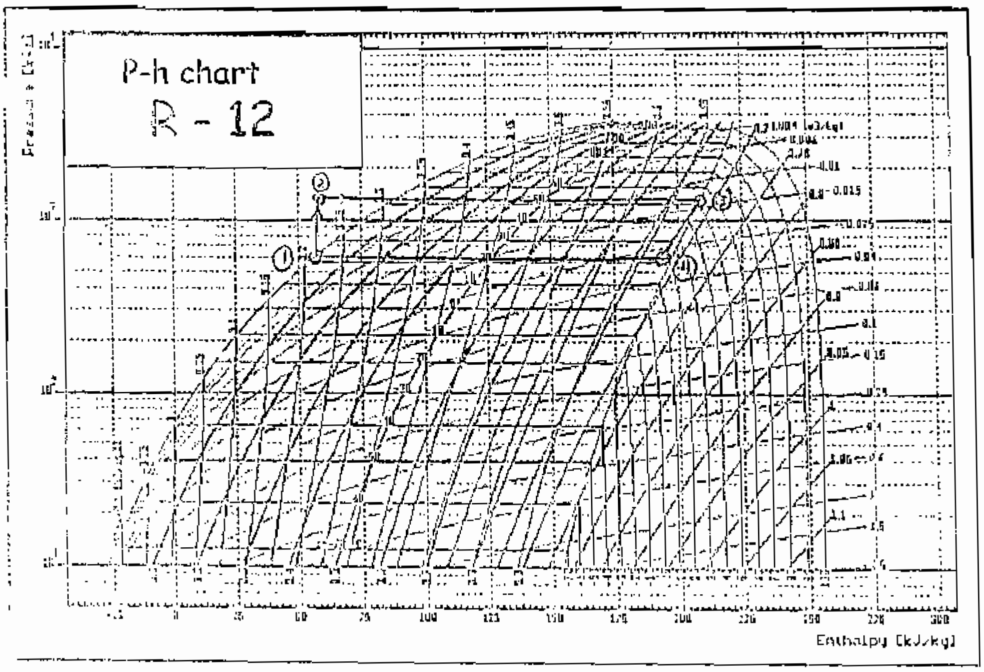

Fig. (2-a) P-h chart for R12

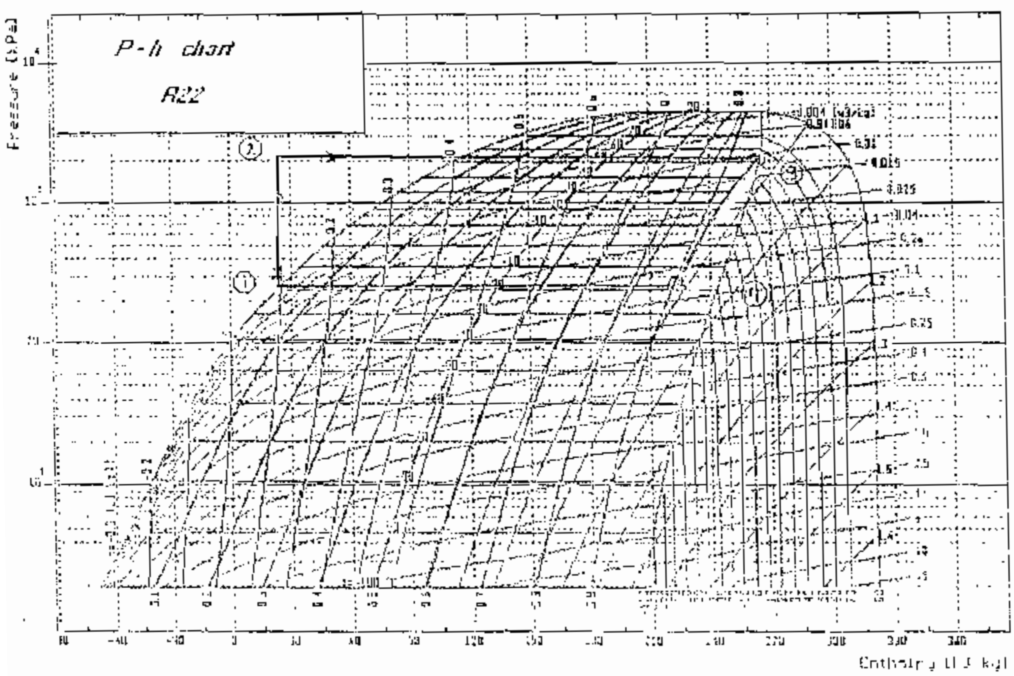

Fig. (2-b) P-h chart for R22

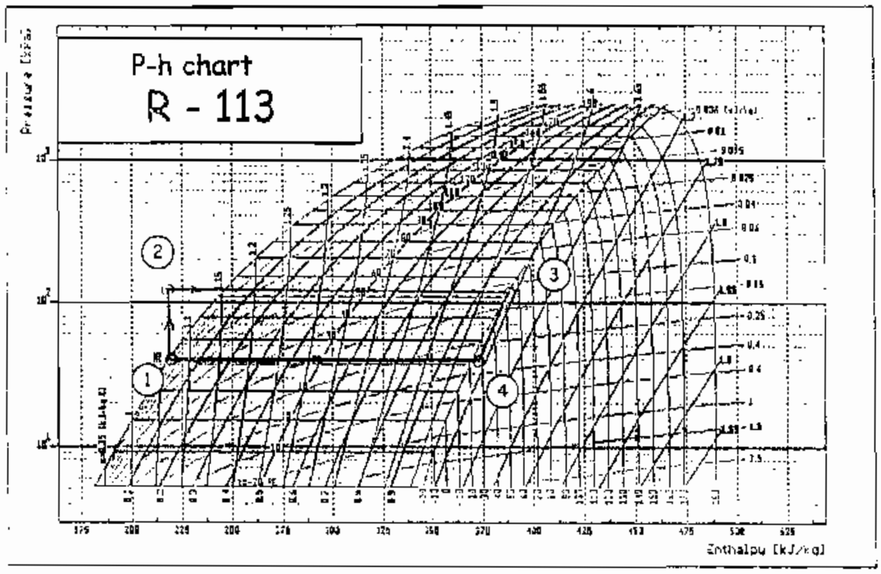

Fig. (2-c) P-h chart for R113

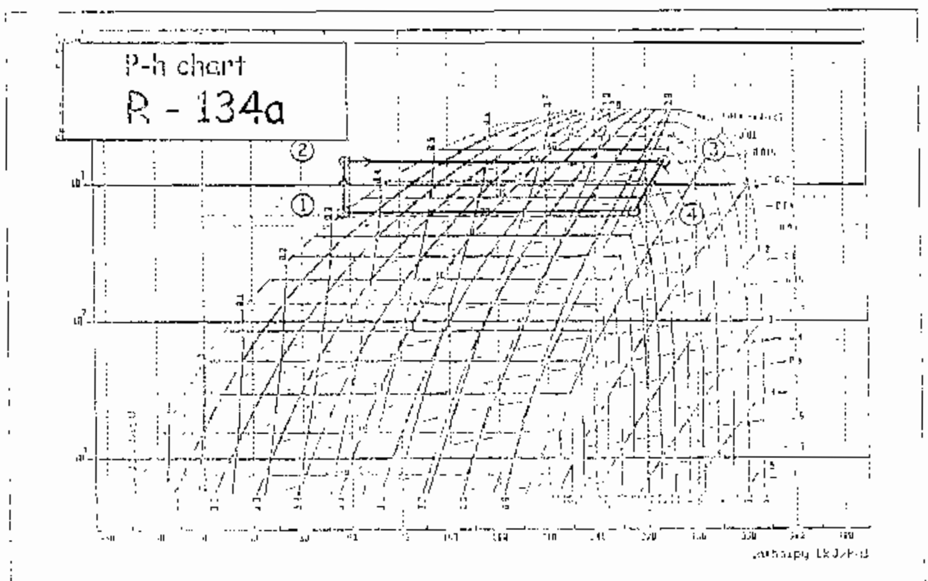

Fig. (2-d) P-h chart for R134a

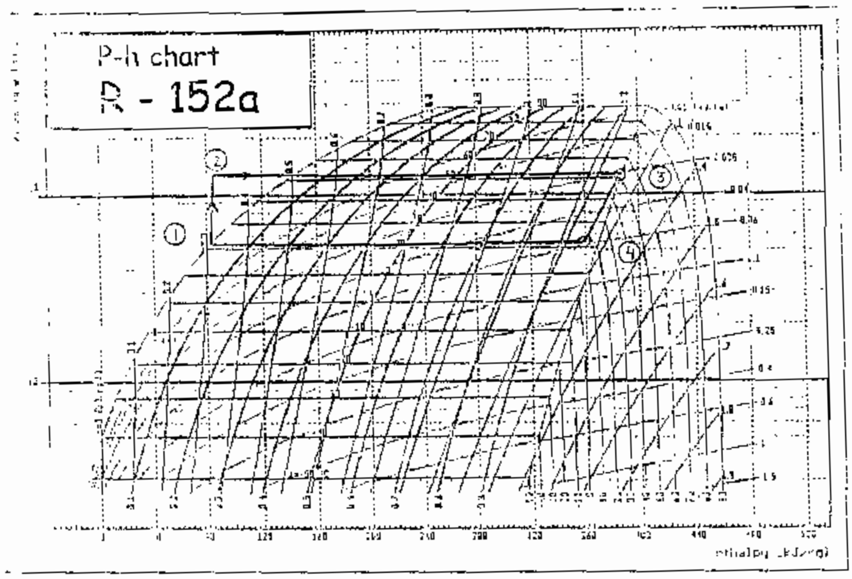

Fig. (2-e) P-h chart for R152a

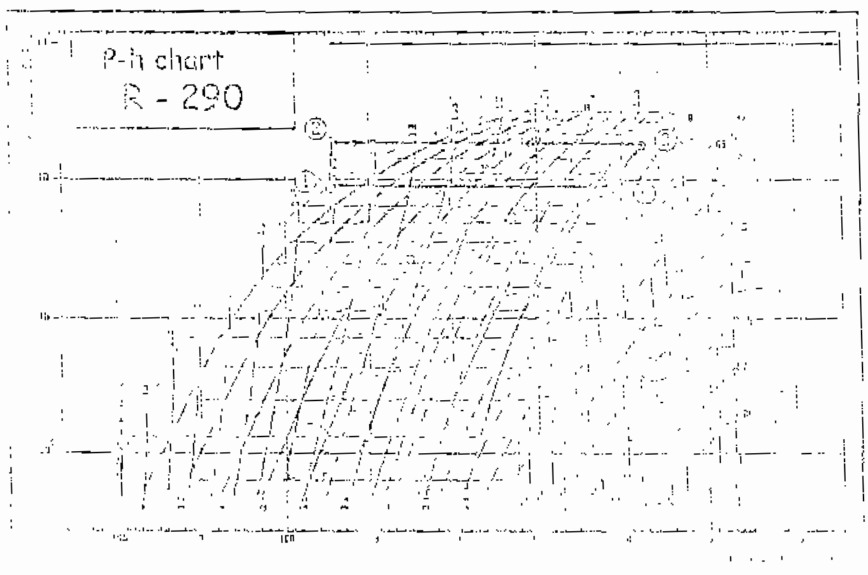

Fig. [illegible] chart for R290

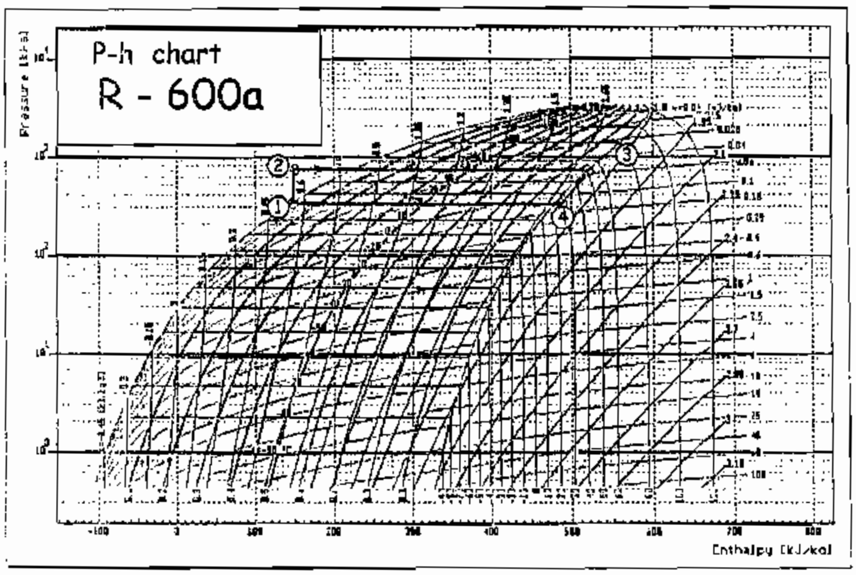

Fig. (2-g) P-h chart for R600a

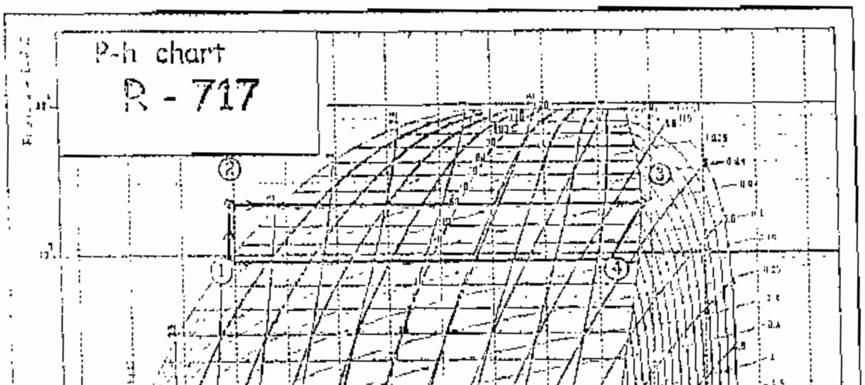

Fig. (2-h) P-h chart for R717

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

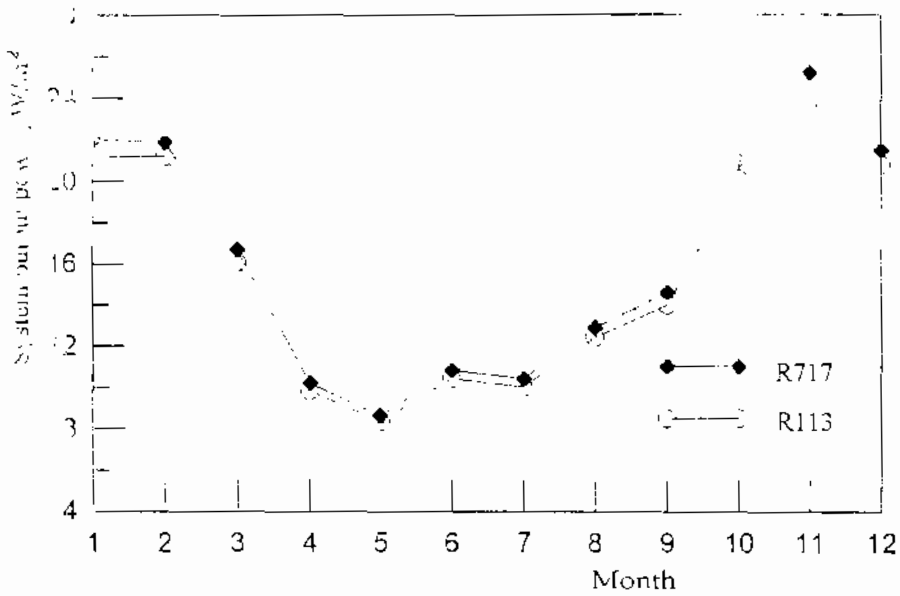

Fig. 3 Variation of the overall system output power during the year.

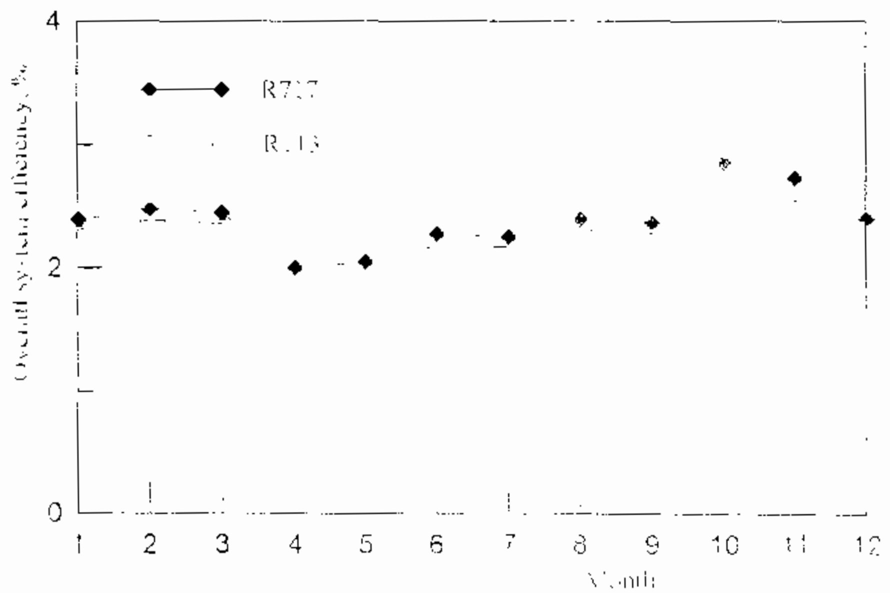

Fig. 4 Variation of the system overall efficiency during the year