EXPERIMENTAL INVESTIGATION OF INDIVIDUAL EVACUATED TUBE HEAT PIPE SOLAR WATER HEATING SYSTEMS

Khalid A. Joudi and Hassan N. S. Al-Joboory
Department of Mechanical Engineering, College of Engineering, University of Baghdad, Baghdad, Iraq.

ABSTRACT

The work involves outdoor experimental testing of ten individual evacuated tube heat pipe solar water heating systems with heat pipes of three diameter groups of 16, 22 and 28.5 mm. The first and third groups had evaporator lengths of 1150, 1300 and 1550 mm. The second group had an additional length of 1800 mm. all heat pipes were of fixed condenser length of 200 mm. Ethanol at 50% fill charge ratio of the evaporator volume was used as the heat pipes working fluid. Each heat pipe condenser section was inserted in a storage tank and the evaporator section inserted into an evacuated glass tube of the Owens- Illinois type. The combined heat pipe and evacuated glass tube form an active solar collector of a unique design. The resulting ten solar water heating systems were tested outdoors under the meteorological conditions of Baghdad, Iraq. Experiments were carried out with no load, intermittent and continuous load conditions. Some tests, at no load, were carried out with and without reflectors. The overall system efficiency was found to improve with load conditions by a maximum of 55%. The system employing an 1800 mm evaporator length and 22 mm heat pipe (HP7) showed the best performance by higher water temperatures, overall useful energy gain and efficiency at various load conditions. System performance was predicted theoretically using electrical analogy derived from an energy balance. An agreement of within 14% was obtained between theoretical and experimental values.

الخلاصة

يتضمن البحث فحصا تجريبيا بالخارج لعشر منظومات تسخين مياه بالطاقة الشمسيه ذات الأنبوب الحراري في الأنبوب المفرغ تستخدم أنابيب حرارية بثلاث أقطار هي 16 و22 و28.5 ملم. تكونت مجموعتي الأنابيب ذات القطرين الأول والثالث من أنابيب بأطوال مبخر 1150 و1300 و 1550 ملم بينما شملت المجموعة الثانية طولا اضافيا هو 1800 ملم. كانت جميع الأنابيب بطول مكثف ثابت يبلغ 200 ملم. شحنت جميع الأنابيب الحراريه بالايثانول كمائع شغل بنسبة شحن بلغت 50% من حجم المبخر. تم إدخال المكثف لكل أنبوب حراري في خزان ماء وادخل المبخر في أنبوب زجاجي مفرغ من نوع أوينز- ألينويز وأصبحت الوظيفة المركبة الناتجة من ذلك تكون لاقط شمسي فعال ذو تصميم فريد. كانت المنظومات الناتجة عشرة منظومات فردية ذات أنبوب حراري في الأنبوب المفرغ لتسخين الماء بالطاقة الشمسية والتي تم فحصها تجريبيا في الظروف الجوية لمدينة بغداد. أجريت التجارب على هذه المنظومات بدون حمل وبحمل متقطع وحمل مستمر. كما أجريت بعض التجارب بدون حمل مع سطح عاكس وبدونه. وقد تحصل أفضل أداء من المنظومة ذات الأنبوب الحراري HP7 بطول مبخر 1800 ملم وقطر 22 ملم من بين المنظومات العشر والذي تمثل بدرجات حرارة ماء ساخن أعلى وتخزين طاقة حرارية كلية وكفاءة أعلى مع جميع أشكال التحميل. تبين من خلال البحث أن الكفاءة الكلية للمنظومات الشمسية تحسنت مع التحميل بمقدار أقصاه 55%. تم القيام بتحليل نظري للمنظومات الشمسية باستخدام منظومة مقاومات كهربائية مناظرة متأتية من الاتزان الحراري للمنظومة الشمسية وعند مقارنة الأداء العملي مع هذه الاستنتاجات النظرية حصل توافق جيد بحدود 14% بين القيم النظرية والقيم العملية.

Keywords: Solar heat pipe; Evacuated tube heat pipe solar collector; Heat pipe solar water heater

INTRODUCTION

Heat pipes are being used in solar collectors as the heat absorbing component for their rapid response to solar radiation changes [Bairamov and Toiliev 1982], freeze tolerance [Radhwan et.al 1990], eliminating corrosion problems and thermal diode benefit. They permit the collection of solar energy at low solar radiation levels of 140 W/m^2 vs. 250 W/m^2 for flat plate collectors [Ward and Ward 1979]. Heat pipe absorbers have been suggested for flat plate solar collectors and evacuated tube collectors [Ortabasi and Buehl 1980], where high rates of heat are transferred from the absorber (evaporator section) to the heat- rejecting end (condenser section) at a very small temperature difference.

The incorporation of heat pipes with conventional flat plate and evacuated tube solar collectors has been investigated by many workers [Bairamov and Toiliev 1982, Akyurt 1984, Hammad 1995, Chun et.al 1999, Nada et.al 2004, Sivaraman and Mohan 2005, Hussein 1997, Walker et.al 2004, Ng et.al 2000, Praene et.al 2005 and Mahdy 2005] However, only a limited number of works has been devoted to the utilization of the evacuated glass tube solar collectors. No works were cited involving a heat pipe within an Owens- Illinois evacuated tube serving as a thermosyphon solar water heating system.

The present work investigates experimentally the use of wickless heat pipes of various lengths and diameters within an Owens- Illinois evacuated tube to form a solar collector connected directly to a storage tank. Ten such solar water heating systems were tested outdoors for performance evaluation. Effects of heat pipe evaporator length and diameter on the performance were assessed.

THE SOLAR WATER HEATING SYSTEMS

Each solar water heating system consists of an evacuated glass tube, a heat pipe, a storage tank and a flat reflector, mounted on a stand and facing south. Ten individual heat pipe solar systems were built, each with the same evacuated glass tube but a different heat pipe. **Fig. 1** shows four individual systems ready for simultaneous testing. Details of heat pipes HP1- HP10 and design specifications of the ten systems are given in **Table 1**. Ethanol was chosen as the heat pipe working fluid for all systems, with a fill charge ratio of 50% of the evaporator volume. The evaporator section of the heat pipe was placed inside the evacuated glass tube, whereas the condenser section was situated in the water storage tank. The evacuated glass tubes were all Owens- Illinois type, model 47-58-1800-YCF. 1 mm thick aluminum reflectors were positioned behind the evacuated glass tubes for better reflection [Kreider and Kreith 1981]. The capacity of the storage water tanks was within the recommended range of 7.25 liters [Kreider and Kreith 1981, Kreider and Kreith 1975]. One side of the tank was cut at 45^o to facilitate receiving the heat pipe condenser into the tank. The inlet and outlet tubes to the storage tank were drilled for temperature sensors. A 3 mm air-vent tube was provided at the top of the tank. The temperature distribution along the height of the storage water was measured by three thermocouples within a 5 mm diameter probe. The heat pipe evaporator section was aligned at the center line of the evacuated glass tube. The solar water heating systems were inclined at an angle of 45^o for winter operation in Baghdad (33.3^o N).

Temperatures at various locations of each heat pipe solar system (inlet and outlet storage water, evacuated tube surfaces, heat pipe evaporator surface and ambient) were measured by calibrated copper- constantan thermocouples connected to a digital readout. The mean tank temperature was taken as the average of the three temperature readings. The load water flow rate was measured by a calibrated Rotameter. The outdoor experiments were carried out in April, May, June, July, August and September of 2008 on sunny days. Test data were considered constant for a period of time of half an hour for an operating period from 8:00 a.m. until 16:00 p.m. Also, data was recorded for a one hour period from 16:00 p.m. till 24:00 p.m. Systems incorporating heat pipes HP3, HP7 and HP10 were operated at different weather conditions and systems incorporating heat pipes HP1, HP2 and HP3 were operated with and without reflectors and at various hot water storage capacities. All individual systems were subjected to three load conditions; these are no- load, intermittent load, and continuous load. The experiments with no load condition were carried out with three storage capacities of 5.25 ℓ, 6.25 ℓ, and 7.25 ℓ. Intermittent loading experiments were carried out with three hot water removal quantities of 0.5 ℓ, 0.75 ℓ, and 1 ℓ for five minuets at the beginning of every hour from 10:00 a.m. to 14:00 p.m. Continuous loading tests were carried out with seven values of hot water withdrawal rates of 0.5, 1, 2, 3, 4, 5, and 6 ℓ/ hr. Each withdrawal process continued from 8:00 a.m. to 18:00 p.m.

The overall collected heat is determined from the measurements of storage tank temperatures at the start and end of the operating period from 8:00 a.m. till 18:00 p.m. by;

$$Q_o = M_t c_w (T_{mf} - T_{ms}) \quad (1)$$

The daily overall or bulk efficiency of the system was then calculated from;

$$\eta_o = \frac{M_t c_w (T_{mf} - T_{ms})}{\int_0^t A_a I(t) dt} \quad (2)$$

With load conditions, the overall heat collected was estimated from the equation;

$$Q_o = M_t c_w (T_{mf} - T_{ms}) + \int_0^t \dot{m}_l \, c_w (T_{lo} - T_{li}) dt \quad (3)$$

This equation was further simplified to the following form;

$$Q_o = M_t c_w (T_{mf} - T_{ms}) + M_l c_w (T_{mo} - T_{mi}) \quad (4)$$

Where; $M_l = \sum \dot{m}_l^{\,i} \, \Delta t^i$, and

$$(T_{mo} - T_{mi}) = \sum_{i=1}^{k} \dot{m}_l^{\,i} (T_{lo}^i - T_{li}^i) \Delta t^i \Big/ \sum \dot{m}_l^{\,i} \, \Delta t^i$$

Where k is the number of time intervals, Δt^i, during the loading period, and Whereas, the overall or bulk system efficiency was calculated from the equation;

$$\eta_o = \frac{M_t c_w (T_{mf} - T_{ms}) + M_l c_w (T_{lo} - T_{li})}{\int_0^t A_a I(t) dt} \quad (5)$$

The solar radiation intensity was calculated using the ASHRAE clear sky model [Farber and Morrison 1977].

RESULTS AND DISCUSSION

Performance curves are generally represented by the variation of the mean water temperature in the tank, the daily overall useful heat gain and the overall (bulk) efficiency of the systems. **Fig. 2** shows a typical variation of the mean tank temperature (T_m) with time for different storage capacities for solar system incorporating heat pipe HP7. The variation of T_m is similar in all systems. The mean tank temperature increases with time, reaches its maximum value at the period between 14:00 p.m. and 16:00 p.m., and then decreases slightly. **Fig. 2** also shows the effect on T_m of the storage capacity. Higher values of T_m are obtained with smaller storage capacity. T_m reached 75.6 ^oC, 82.4 ^oC and 89.5 ^oC with storage capacities of 7.25 ℓ, 6.25 ℓ and 5.25 ℓ respectively. Reduced quantities of stored heat are obtained with decreased storage capacities due to the increased heat losses from the storage tank, resulting in reduced bulk efficiency. The overall stored heat and overall efficiency increased with storage capacity from 713 kJ and 20% to 1358 kJ and 54.1%, as shown in **Figs. 3 and 4** for various solar systems. The mean tank temperature is observed to increase with systems of longer evaporator heat pipes for the 22 mm diameter group as shown in **Fig. 5**. An increase from 1150 mm to 1800 mm in the evaporator length resulted in an increase of 3.4 ^{o}C to 10.3 ^{o}C in the maximum value of T_m. A similar observation was concluded for systems incorporating heat pipes of the same evaporator length but different diameters, as shown in **Fig. 6**. Higher temperatures are obtained with systems of larger heat pipe diameters. An increase from 16 mm to 28.5 mm in the heat pipe diameter resulted in an increase of 8.7 ^{o}C to 12.4 ^{o}C in the maximum value of T_m due to the increased radiative heat transfer attained by the increased heat pipe evaporator surface area. The effect of reflectors on the mean tank temperature at no load conditions is shown in **Fig. 7** with heat pipe HP3. The presence of reflectors increased the maximum mean tank temperature by 6.5 ^{o}C to 16 ^{o}C, accompanied with an increase in the overall stored heat by 1.7% to 17.4% and the overall efficiency by 1.8% to 14.7%, as shown in **Fig. 8**. These results agree with those obtained in other works [Ward and Ward 1979, Mahdy 2005]. From the above results, it is concluded that increased mean tank temperatures are attained with decreased storage capacities. While increased overall stored energy and efficiency are attained with increased storage capacities for all individual heat pipe solar systems. Therefore, such systems perform better with increased storage capacities and with reflectors.

To study the effect of intermittent load conditions, a fixed quantity of hot water was withdrawn from the storage tank at the beginning of each hour during the period from 10:00 a.m. to 14:00 p.m. This pattern of loading was recommended by [Esen and Esen 2005] under cold climates of Turkey, which is found practical for Iraq. The amounts of water withdrawn were 0.5, 0.75 and 1 ℓ in a period of five minuets. **Fig. 9** shows the results for the system with heat pipe HP7. The trend of variation is similar in all systems. T_m increases gradually, drops suddenly with each hot water removal, increases slightly after the removal process and increases continuously after the last removal process until the end of operation period. The sudden drop in T_m with hot water removal is due to the entry of a corresponding amount of cold mains water into the storage tank. This drop in T_m depends on the system involved, the quantity of hot water removed and the supply water temperature. A temperature drop of 1.5 ^{o}C to 3.2 ^{o}C was observed after each withdrawal process. **Fig. 10** depicts a typical behavior of varying heat pipe evaporator length at equal diameters on the variation of T_m with hot water removal quantity of 0.75 ℓ. A higher T_m was attained by systems incorporating heat pipes of longer evaporators. An increase in the overall daily useful energy of within 2.7% to 23.4% was observed corresponding to increasing evaporator length from 1150 mm to 1800 mm. A difference of T_m value of the various systems undergoing the same loading quantity was within 0.1 ^{o}C to 3.6 ^{o}C. **Fig. 11** compares variation of T_m in systems with heat pipes of equal evaporator length but different diameters. Increased T_m was observed with increased heat pipe diameter. It is noticed in **Figs. 10 and 11** that T_m and the outlet temperature continued increasing in spite of the hot water removal. This indicates that the heat input to the storage tank is more than that withdrawn by the load water. T_m continued rising after the last hot water removal at 14:00 p.m. to the end of the operation period to reach 61.5 ^{o}C and 78.1 ^{o}C depending on system specifications. The daily total useful energy increased from 1830 kJ to 2850 kJ and the overall daily efficiency of the systems from 55.7% to 83.9% for longer heat pipes and increased loading, as shown in **Figs. 12 and 13**. The solar system with heat pipe HP7 showed the best performance among the ten systems. A higher T_m value was attained by this system with higher overall daily useful energy and efficiency. In general, system performance increased with increased heat pipe evaporator lengths and diameters. The overall useful energy was more by 55% with loading.

The experimental systems were subjected to continuous loading at various hot water removal rates. Each water withdrawal process continued along the whole operation period from 8:00 a.m. to 18:00 p.m. The flow rates used were 0.5, 1, 2, 3, 4, 5 and 6 ℓ/ hr respectively. **Fig. 14** shows typical results of the variation of the mean tank temperature with the seven removal rates for the system with HP7. A remarkable decrease of the mean tank temperature with increased removal rate is observed. This is a normal behavior due to the continuous entry of corresponding amounts of cold mains water into the storage tank. As a result, the overall useful energy is affected by the hot water removal rate. The experimental systems showed increased overall useful energy and increased overall efficiency with increased hot water removal rate accompanied with a decrease in the mean tank temperature. **Fig. 15** shows a typical variation of T_m with varying heat pipe evaporator lengths at a diameter of 22 mm at a removal rate of 3 ℓ/ hr. Whereas, **Fig. 16** shows the effect of varying heat pipe diameter at equal evaporator lengths of 1150 mm on the variation of T_m at 5 ℓ/ hr continuous loading. Higher values of T_m were obtained by heat pipe systems of longer evaporators and larger diameters as before as well as improved daily overall useful energy gain and efficiency. An increase from 1150 mm to 1800 mm in heat pipe evaporator length resulted in an increase of 3.7% to 31.8% in the overall useful energy, while an increase from 16 mm to 28.5 mm in the heat pipe diameter caused an increase of 6.7% to 11.6%. It was observed that with flow rates of 0.5 to 5 ℓ/ hr, the overall useful energy increased significantly with increased flow rates. However, a decrease was observed in the overall useful energy and the overall efficiency when removal quantities exceeded 5 ℓ/ hr, as shown in **Figs. 17 and 18**. This is thought to be due to the system being incapable of providing the required heat at these loading conditions. An optimum removal rate of 5 ℓ/ hr was concluded, at which best performance was obtained. It is also concluded that all experimental solar systems performed better with load conditions. Higher water temperatures were obtained with no load condition, while higher amounts of overall useful energy and efficiency were obtained with load conditions. This is typical with domestic solar hot water systems.

The solar system with heat pipe HP7 showed the best performance among the experimental systems with all load conditions. The trend of variation of the mean tank temperature with time in the present work is similar to that in several works [Akyurt 1984, Chun et.al 1999 and Noren 1981], as shown in **Fig. 19**. However, the peak values are different due to the differences in design specifications and operation conditions of each system. A comparison can be carried out meaningfully only for the overall efficiency of the present systems with those in the literature, as given in **Table 2**.

The performance of the present solar systems was predicted theoretically from electrical analogy derived from energy balance of the system components. The model is similar to models of two commercial evacuated tube heat pipe solar systems [Walker et.al 2004 and Ng et.al 2000]. The model [Al-Joboory 2009] assumes that all of the absorbed solar energy is transferred directly to the storage tank via the heat pipe action and neglects heat losses from the system components except the storage tank. Any heat accumulation in the evacuated glass tube was neglected i.e. time lag is not evident in the model. The predicted overall useful energy is greater by 14%, 10% and 11.5% for no load, intermittent and continuous loading, as shown in **Fig. 20**.

CONCLUSIONS

1. The performance of the heat pipe hot water system is improved with increased heat pipe length and diameter.
2. Higher water temperatures are obtained with no load at lower storage water capacity while higher stored heat and overall efficiency with higher storage water capacity.
3. The best performance was obtained for the system employing heat pipe HP7.

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NOMENCLATURE

A_a	Absorber surface area	m^2
c_w	Specific heat of water	kJ/kg.°C
D	Heat pipe diameter	m
I_t	Solar Insolation	W/m^2
k	Number of time intervals	-----
L_{ev}	Heat pipe evaporator length	m
L_{co}	Heat pipe condenser length	m
$\dot{m}$	Mass flow rate	kg/s
M	Mass of water (tank, load)	kg
Q_o	Overall useful energy gain	kJ
t	Time	s
T	Temperature	°C
Greek symbols		
α	Absorptivity	-----
β	Inclination angle from the horizontal	Degree
Δ	Difference	-----
ε	Emissivity	-----
η_o	Overall efficiency	-----
τ	Transmissivity	-----

Subscripts

a	Ambient
b	Bulk
i	In
m	Mean
max	Maximum
mf	Final mean value
ms	Starting mean value
l	Load
o	Out
t	Tank

Table (1) Design specifications of the individual evacuated tube heat pipe solar water heating systems with heat pipes HP1- HP10.

Part	Item	Design Specifications									
Solar collector	Type	Evacuated tube heat pipe solar collector									
	Absorber area	0.265 m^2									
Heat pipe	Type	Gravity assisted wickless heat pipe without adiabatic section									
	Material	Copper									
	Code	HP1	HP2	HP3	HP4	HP5	HP6	HP7	HP8	HP9	HP10
	Diameter (mm)	16	16	16	22	22	22	22	28.5	28.5	28.5
	L_{ev} (mm)	1150	1300	1550	1150	1300	1550	1800	1150	1300	1550
	L_{co} (mm)	200									
	Working fluid	Ethanol									
Evacuated glass tube	Material	High quality borosilicate glass									
	Length	1800 mm									
	Outer tube	Φ 58 mm									
	Inner tube	Φ 47 mm									
	Glass thickness	1.6 mm									
	Vacuum	10^{-4} torr									
	Coating	Graded Aluminum Nitride/ Aluminum									
	Transmittance	0.93									
	Absorptance	> 95%									
	Emittance	7- 8% (at 80 ^oC)									
Flat reflector	Material	Aluminum sheet									
	Size	1800 mm× 400mm									
Storage tank	Material	Galvanized steel 0.8 mm thick									
	Capacity	7.25 ℓ (max.)									

Table (2) Comparison of the overall daily efficiency with various works.

Works	Load condition	Design specifications				$\eta_{Overall}$ %
		Collector type	L_{ev}	L_{co}	Working fluid	
Present work (individual heat pipe systems)	No load	Evacuated tube-heat pipe solar collector	1.8 (m)	0.2 (m)	Ethanol	54.12
	Intermittent load					83.9
	Continuous load					81.5
Mahdy [2005]	No load	Evacuated tube-heat pipe collector	1.2 (m)	0.2 (m)	Ethanol	48
	Continuous load				H_2O	61
Akyurt [1984]	No load	flat plate heat pipe collector	2.1 (m)	1.2 (m)	Ethanol	52
Noren [1981]	No load	flat plate heat pipe collector	2.1 (m)	1.2 (m)	Ethanol	49
Chun et al. [1999]	No load	flat plate heat pipe collector	1.7 (m)	0.2 (m)	Ethanol	45

Figure (1) A photograph of four different evacuated tube heat pipe solar water heating systems with instrumentation.

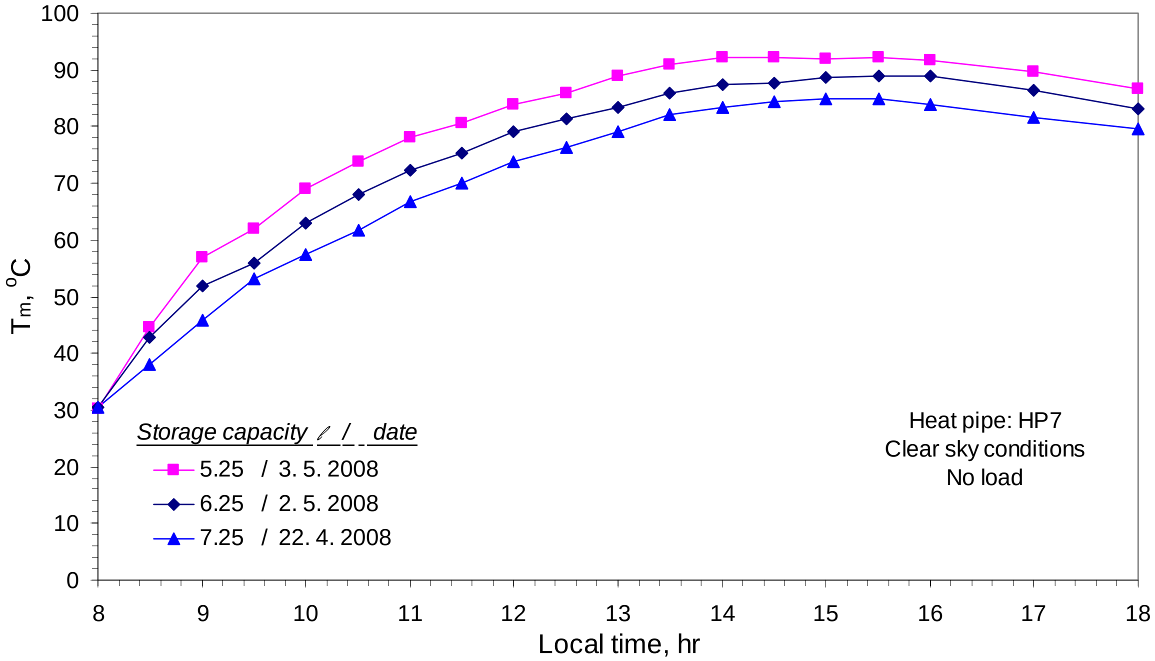

Figure (2) Variation of the mean tank temperature at various storage capacities.

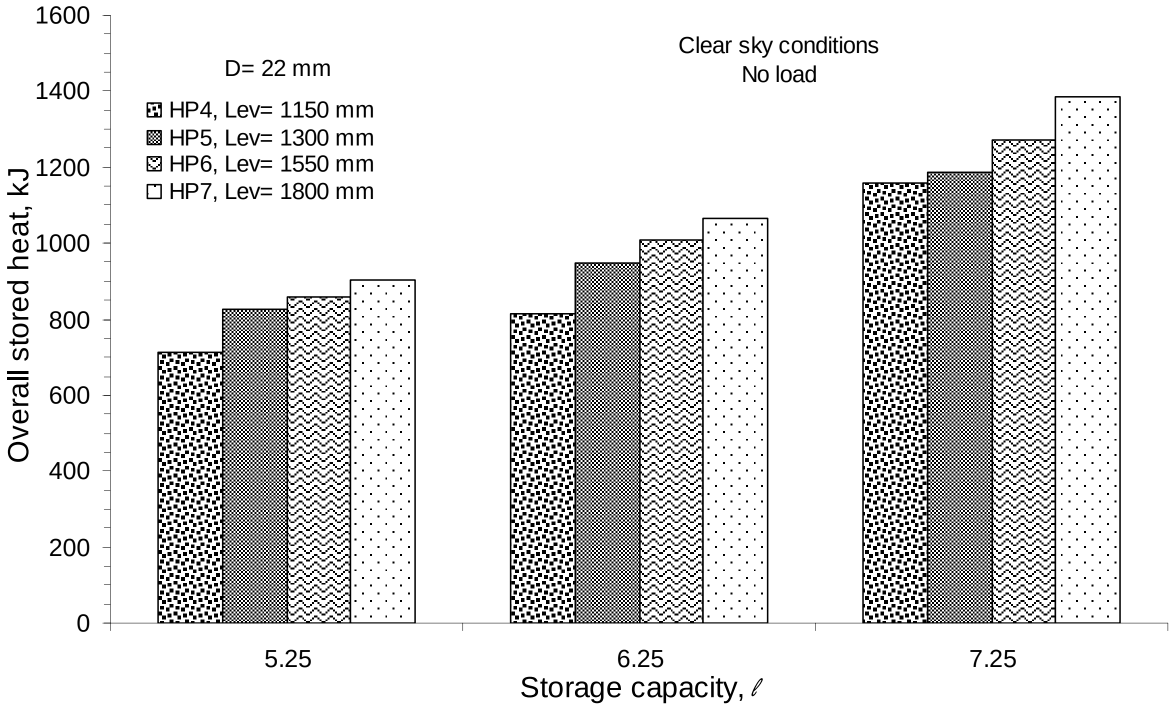

Figure (3) Effect of the storage capacity on the daily overall stored heat.

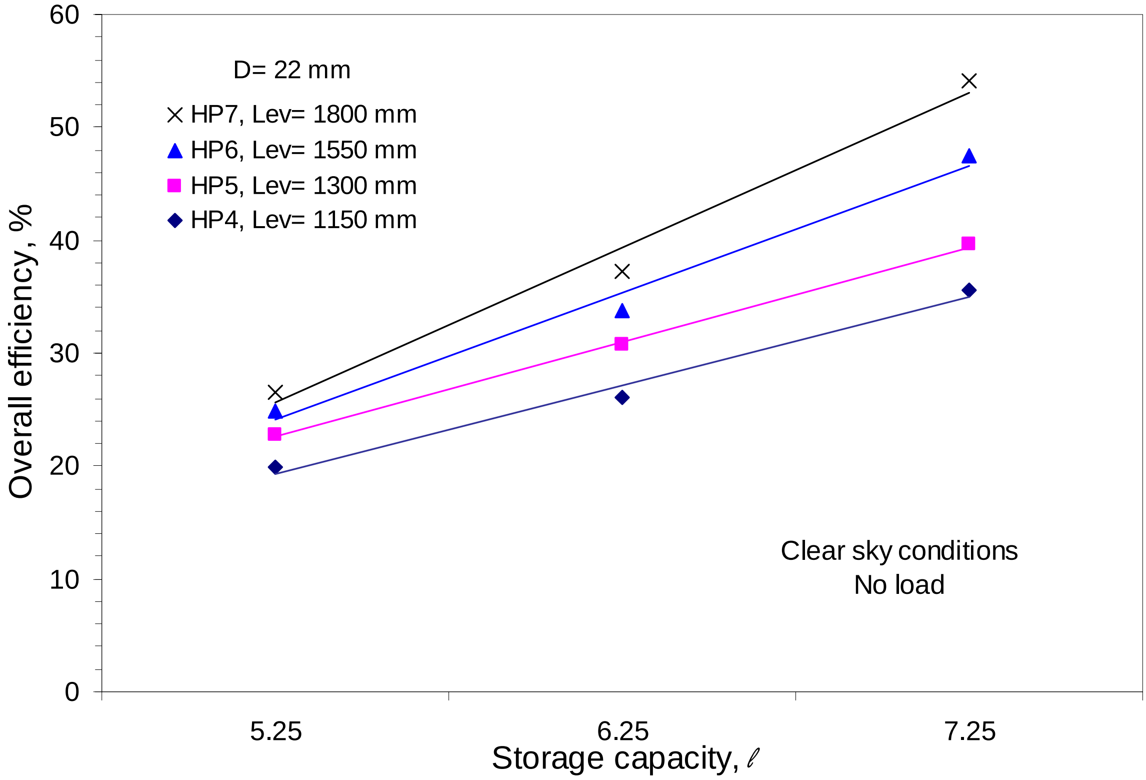

Figure (4) Effect of the storage capacity on the daily overall efficiency.

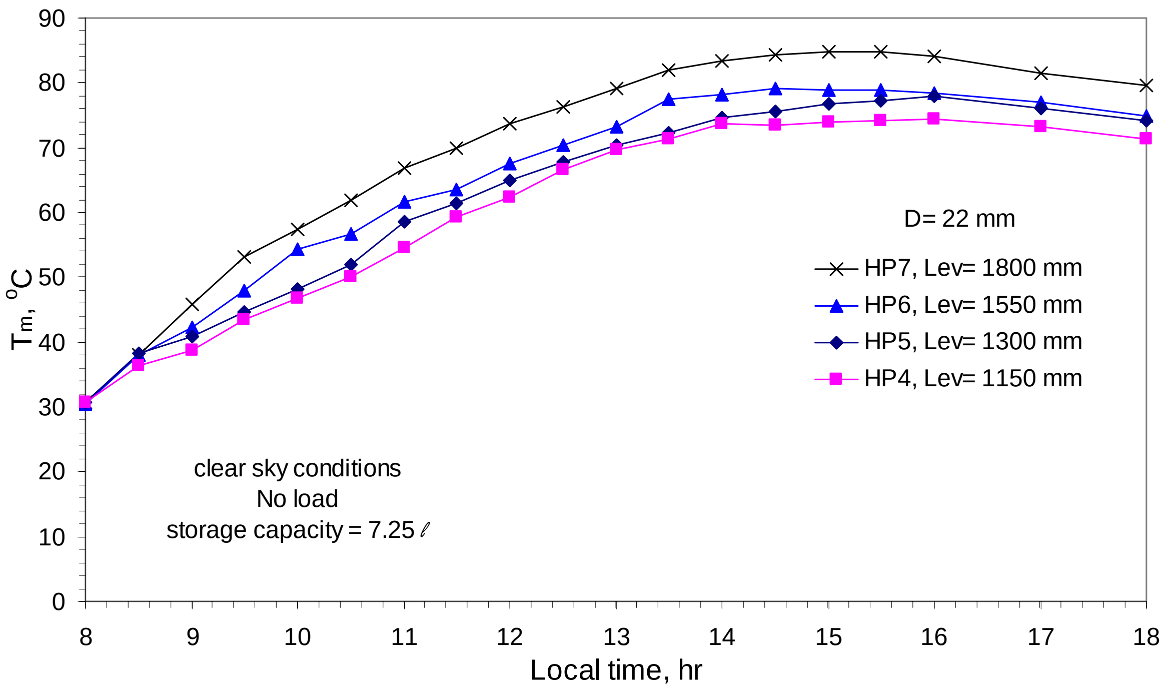

Figure (5) Variation of the mean tank temperature with different evaporator lengths.

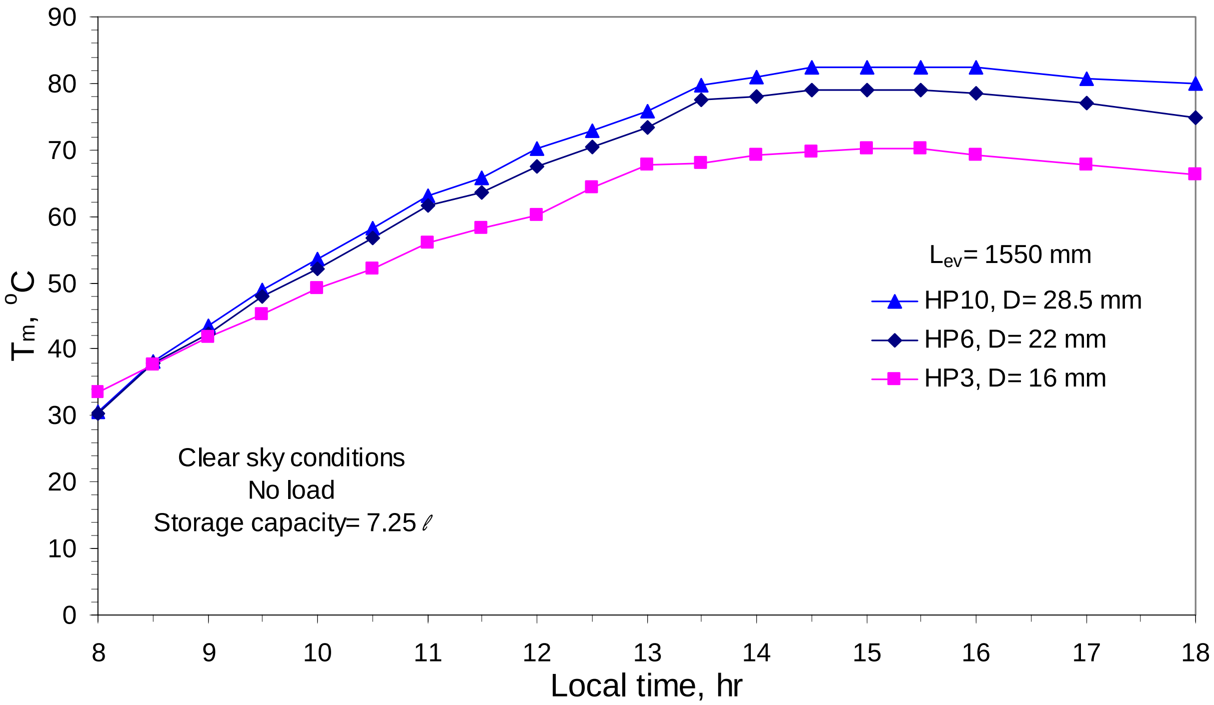

Figure (6) Variation of the mean tank temperature with different diameter heat pipes.

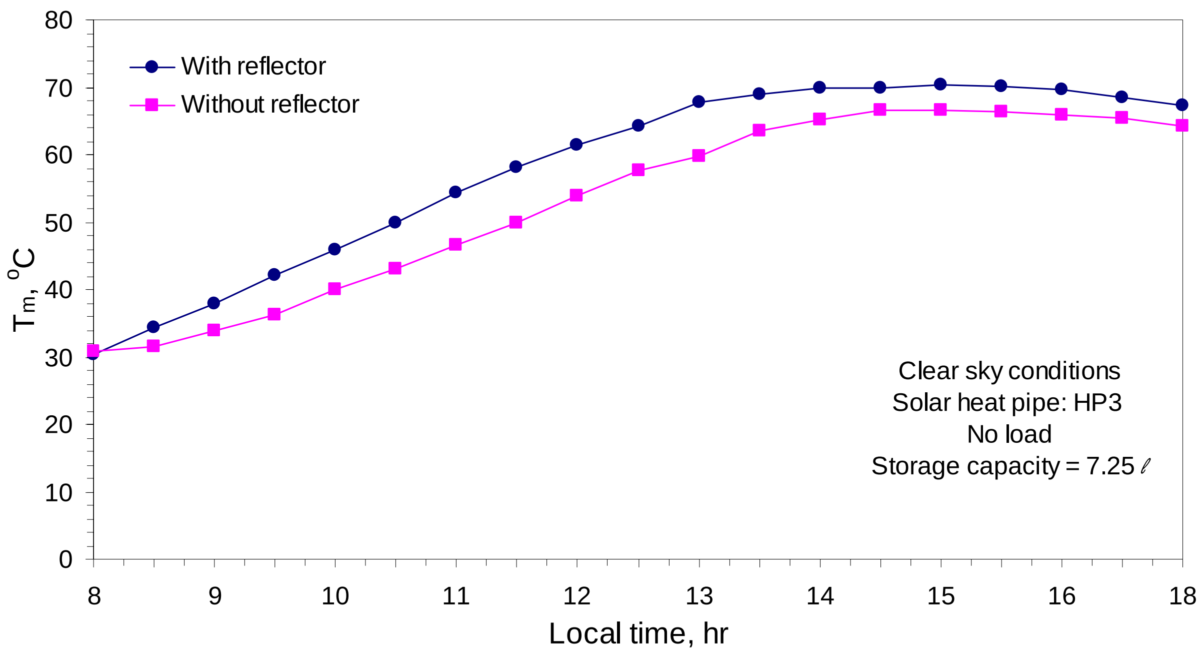

Figure (7) Effect of reflectors on the mean tank temperature.

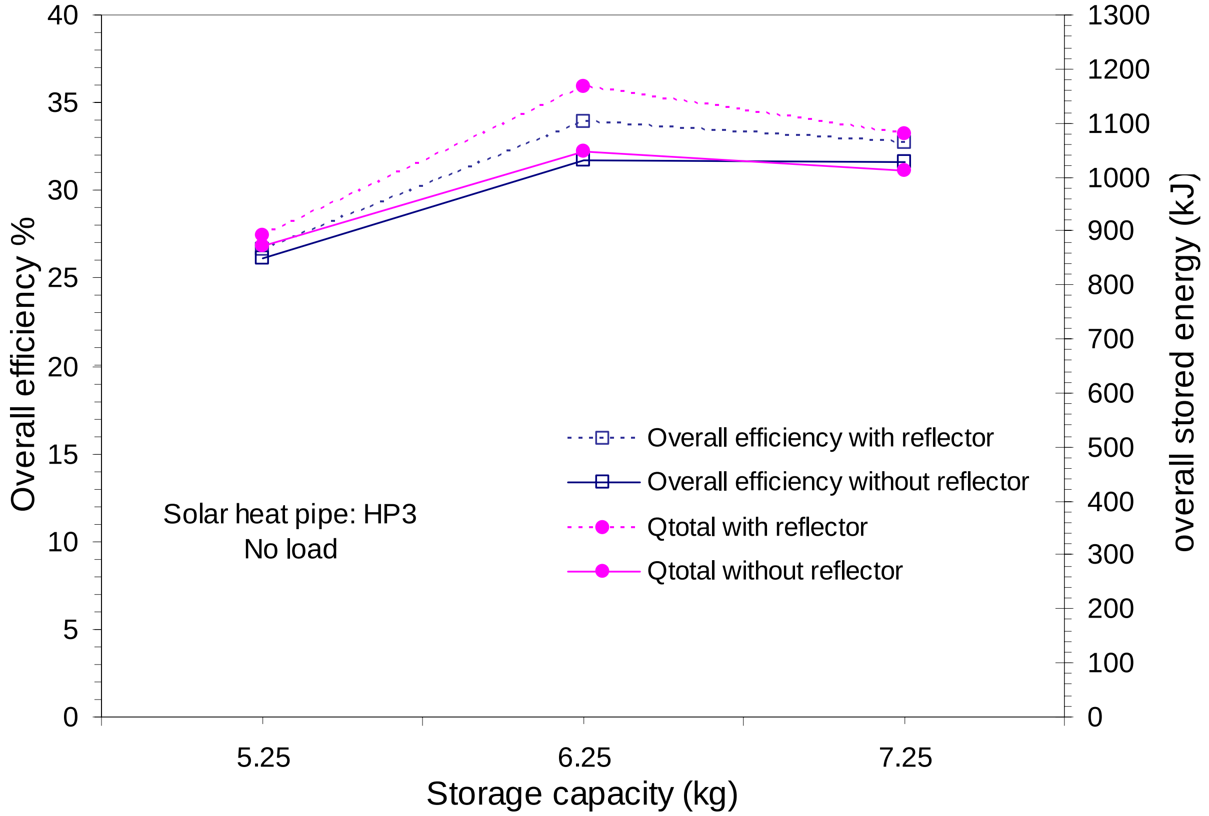

Figure (8) Effect of reflectors on the overall stored energy and efficiency at different storage capacities.

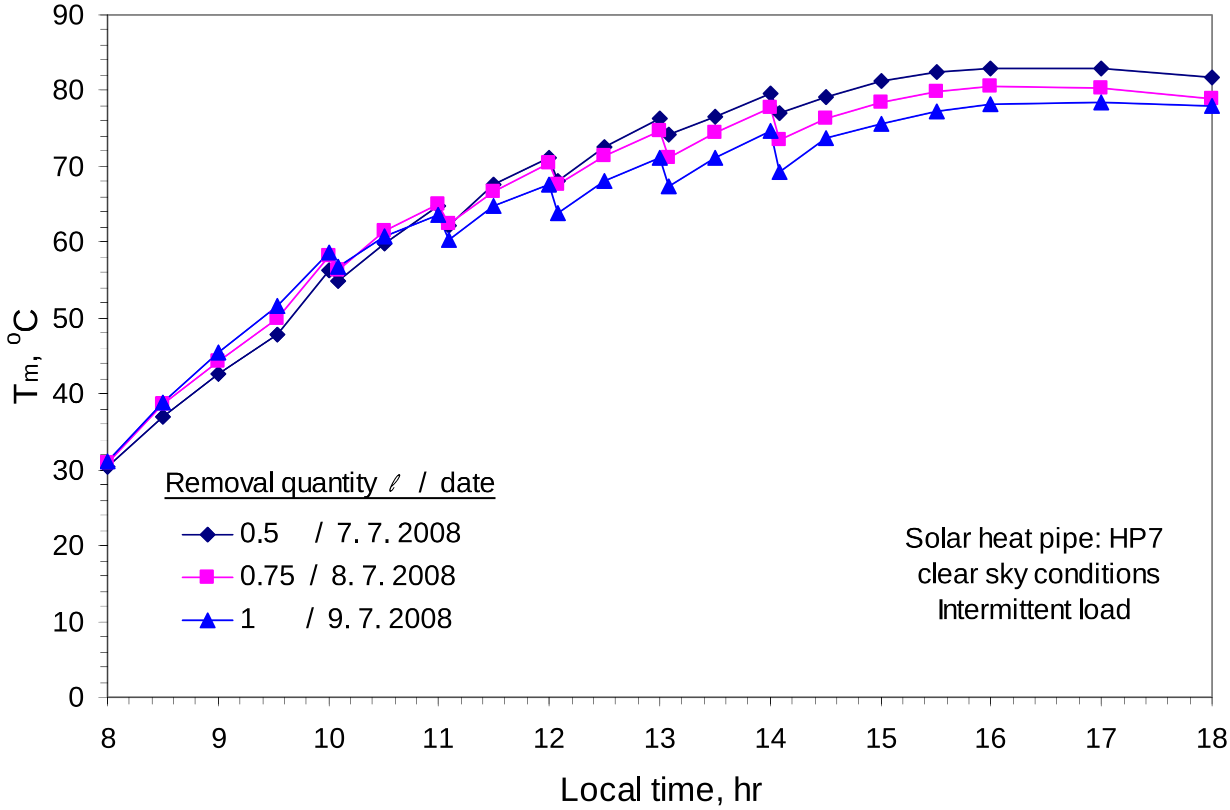

Figure (9) Effect of intermittent loading on T_m for solar system with heat pipe HP7.

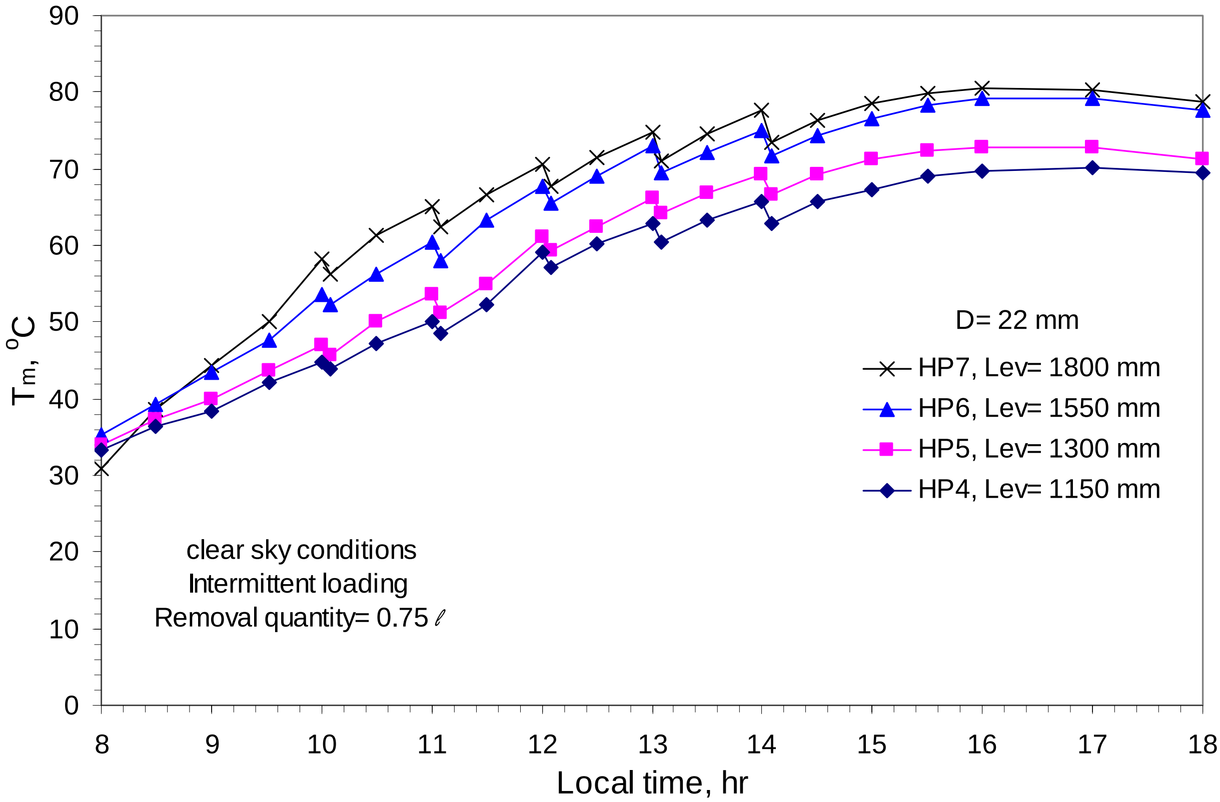

Figure (10) Effect of intermittent loading for the 22 mm diameter heat pipe systems on T_m.

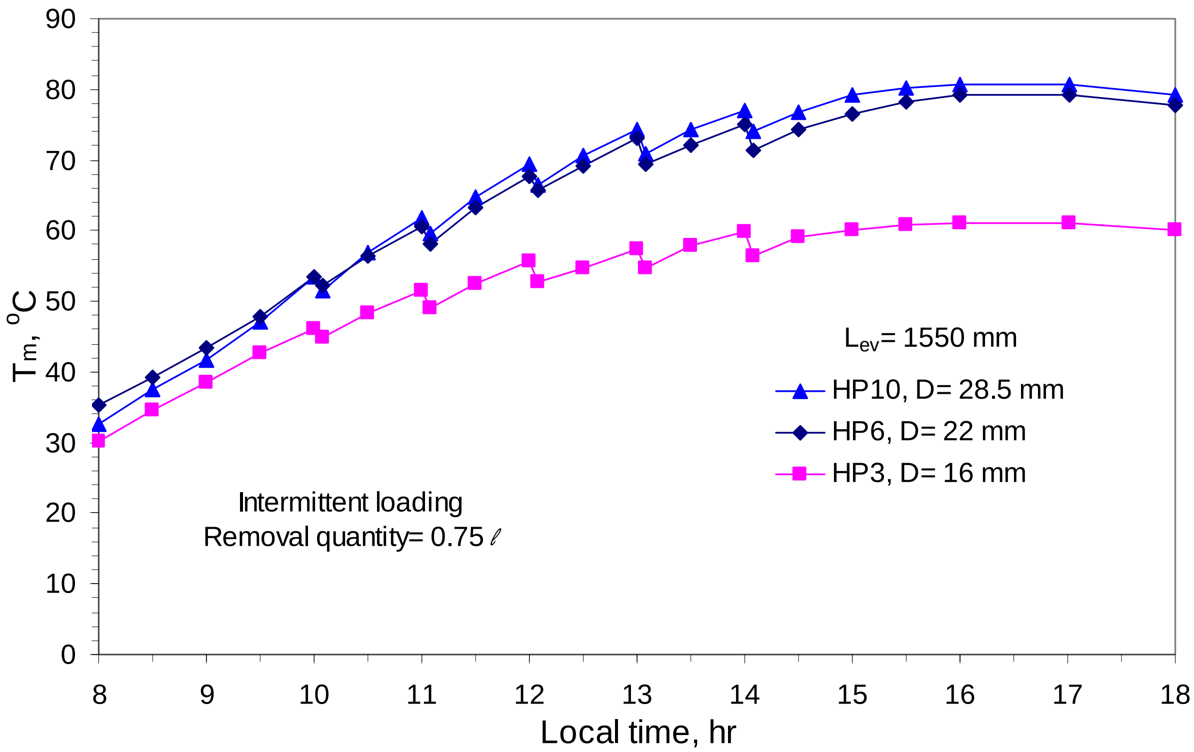

Figure (11) Effect of the heat pipe diameter with intermittent loading for systems with 1550 mm evaporator heat pipes.

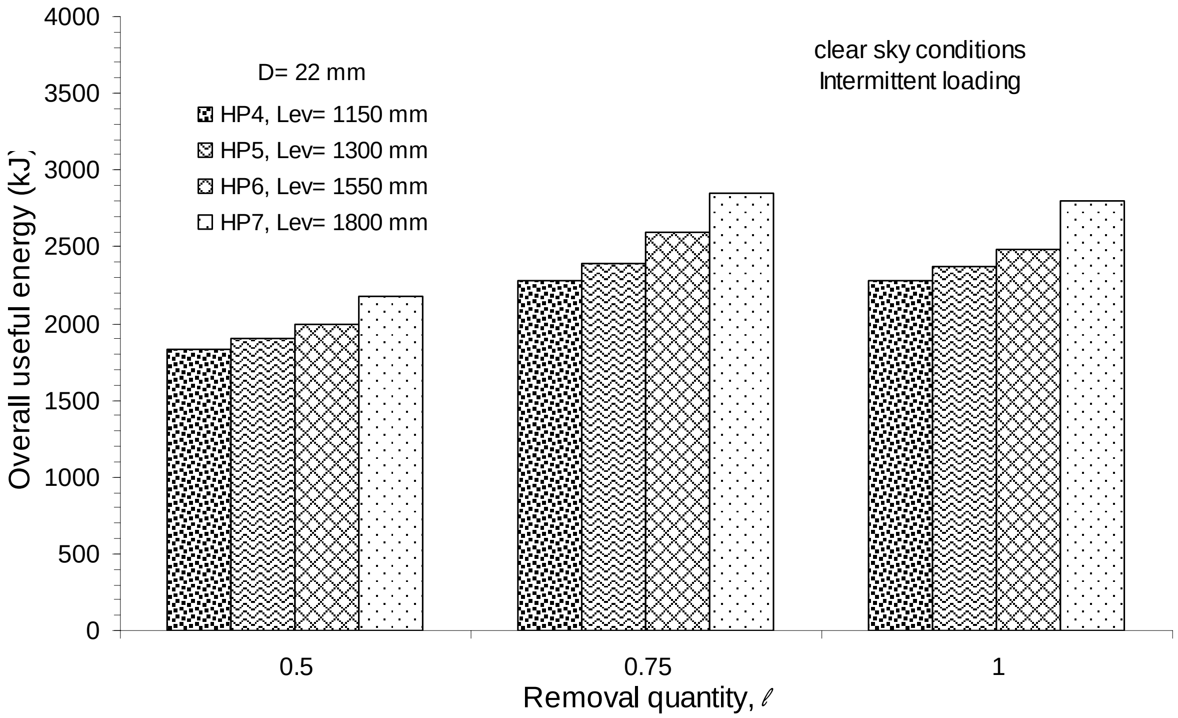

Figure (12) Effect of the heat pipe evaporator length, at 22 mm diameter, on the overall useful energy at various loading conditions.

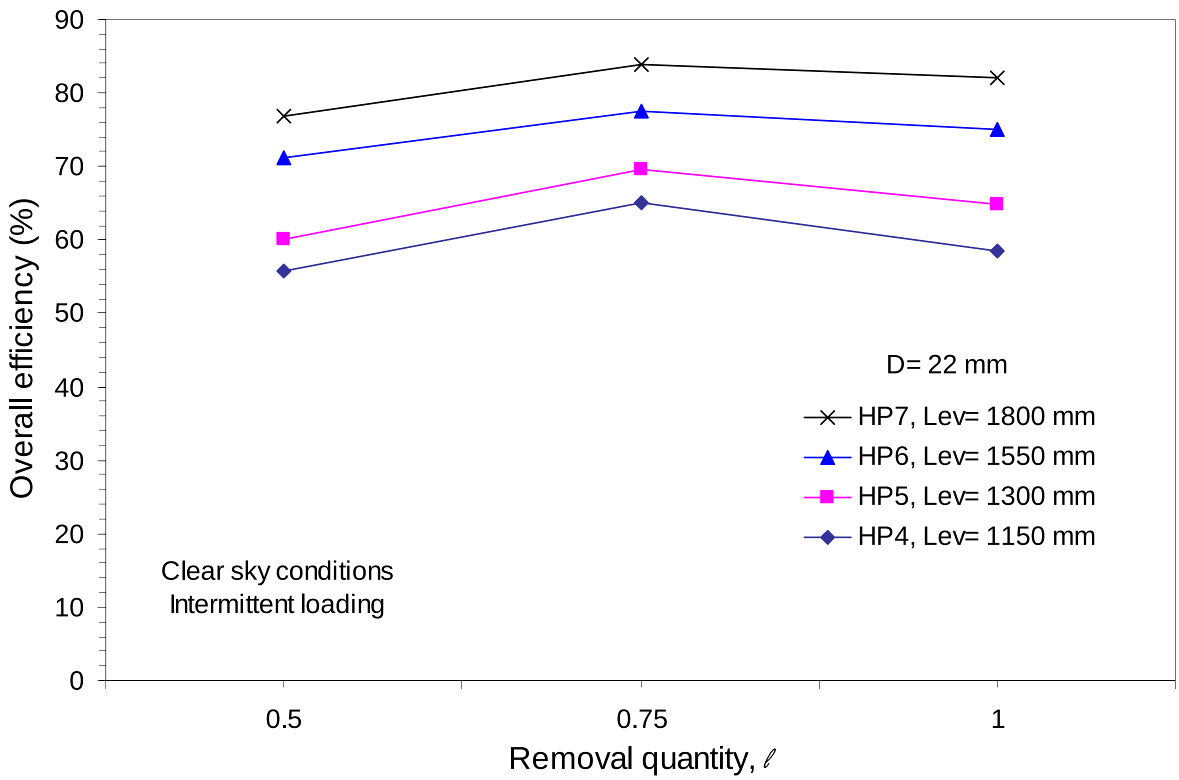

Figure (13) Effect of the heat pipe evaporator length, at 22 mm diameter, on the overall efficiency at various loading conditions.

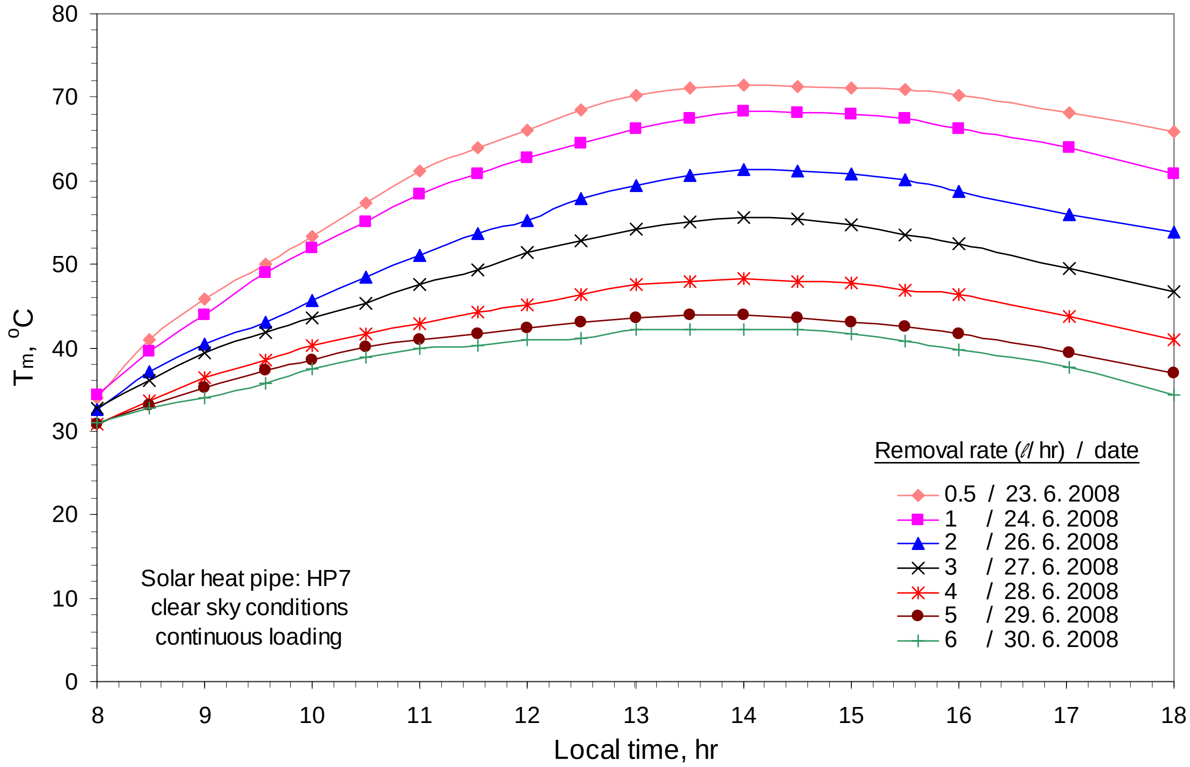

Figure (14) The effect of continuous loading on the mean tank temperature.

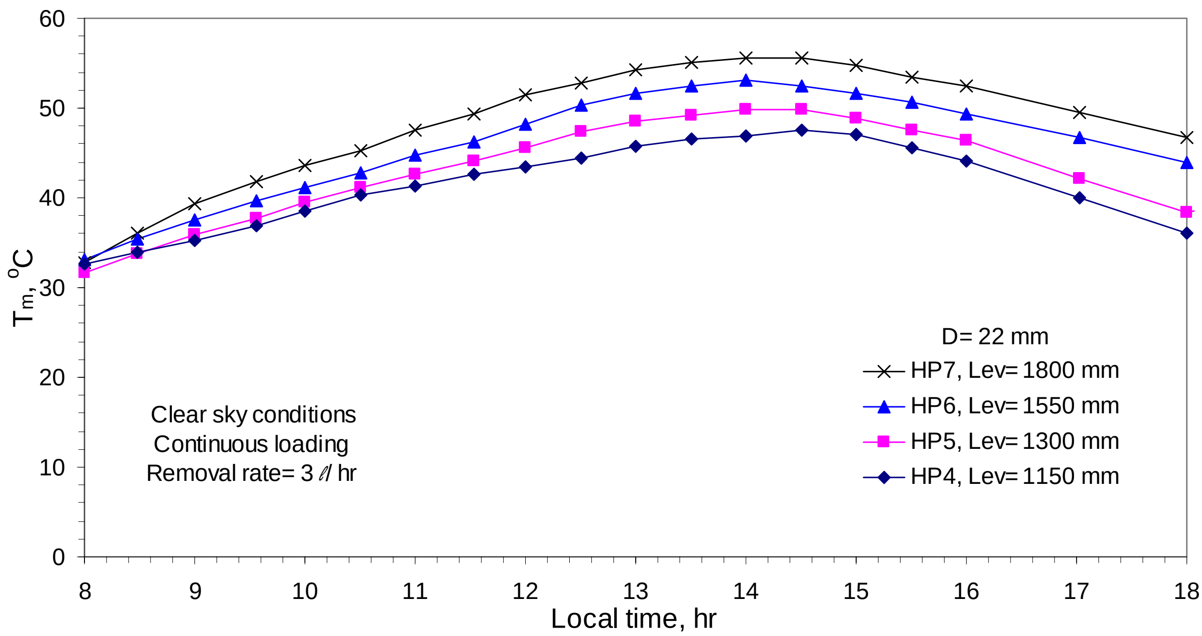

Figure (15) Mean tank temperature variation with various heat pipe evaporator lengths at continuous loading.

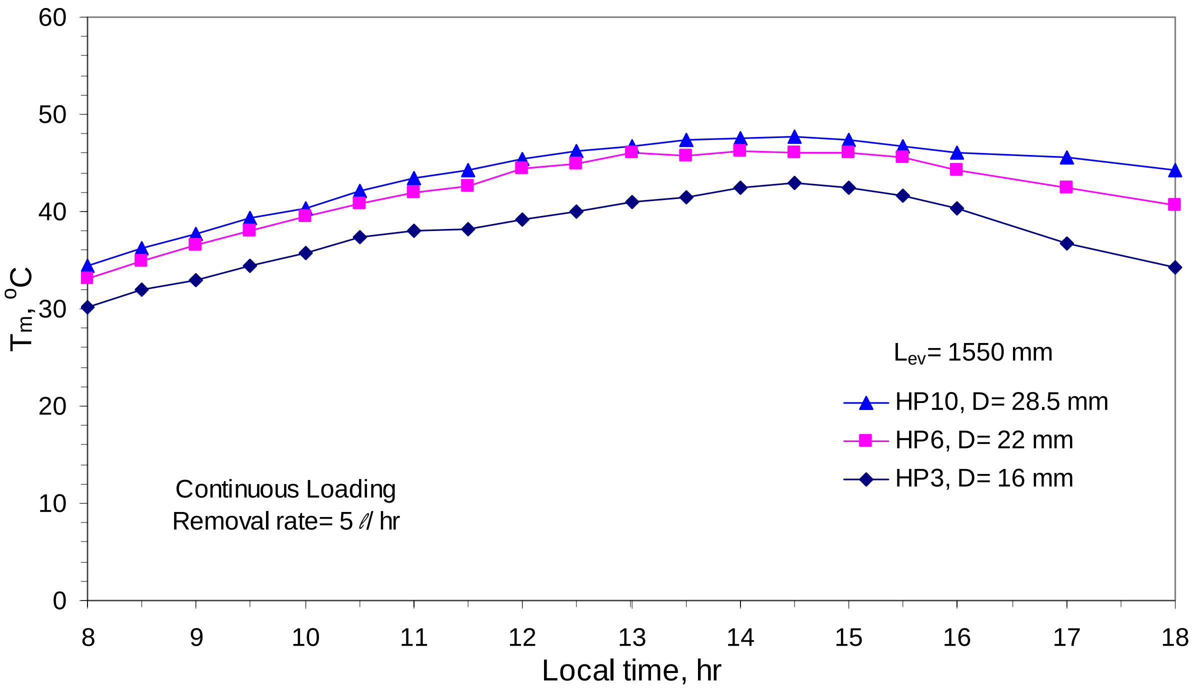

Figure (16) Effect of heat pipe diameter on the mean tank temperature variation at a removal rate of 5 ℓ/ hr.

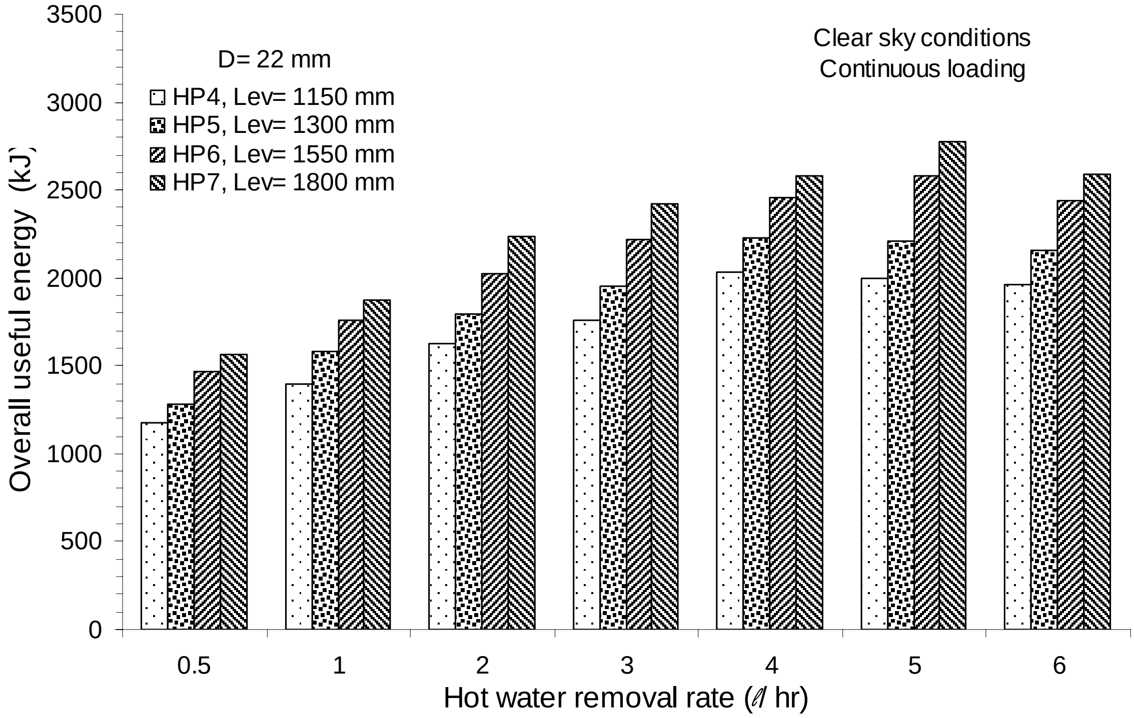

Figure (17) Effect of the heat pipe evaporator length, at 22 mm diameter, on the overall useful energy at various removal rates.

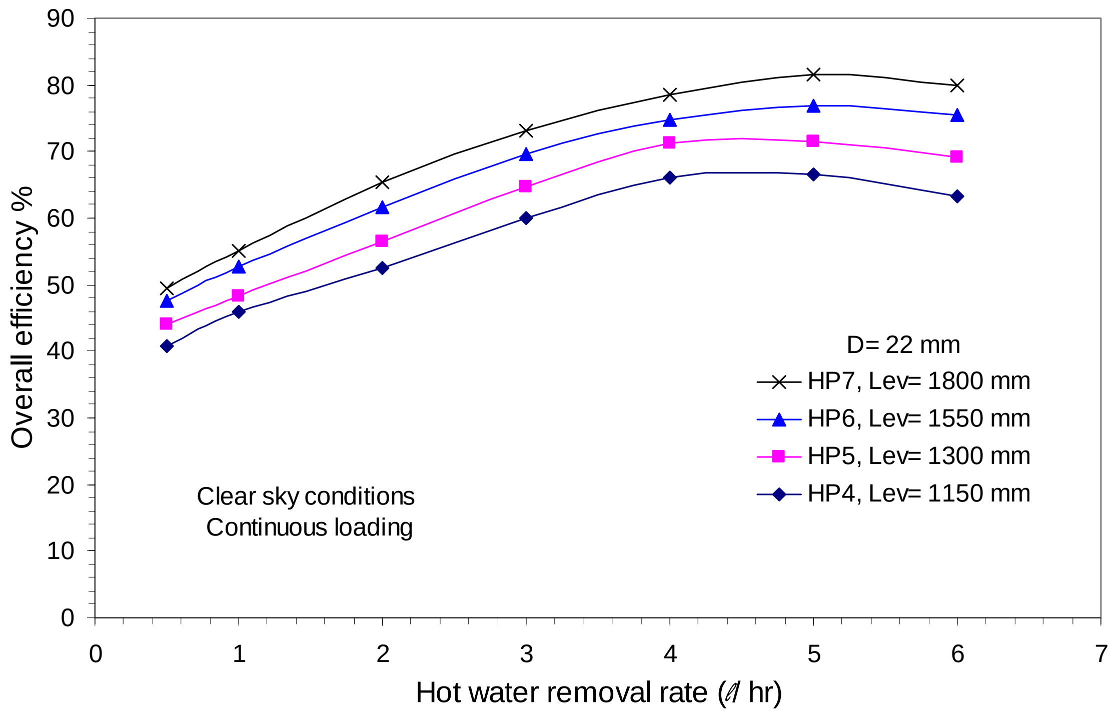

Figure (18) Effect of the heat pipe evaporator length, at 22 mm diameter, on the overall efficiency at various removal rates.

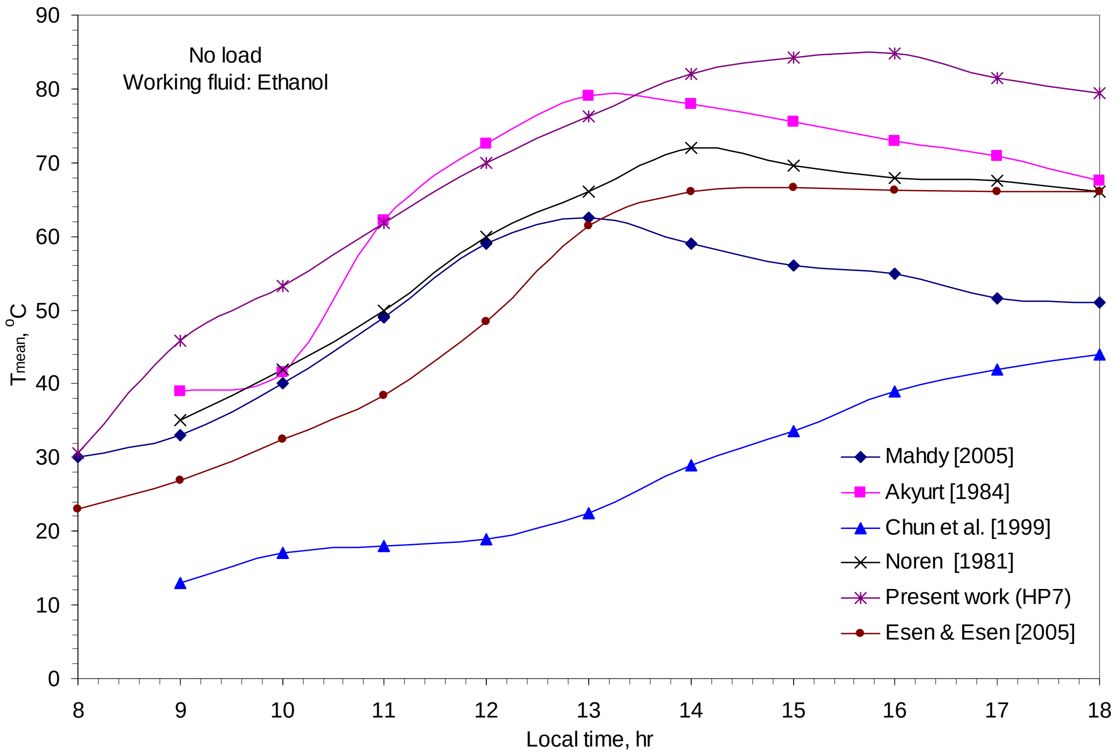

Figure (19) Comparison of the mean tank temperature variation with various works.

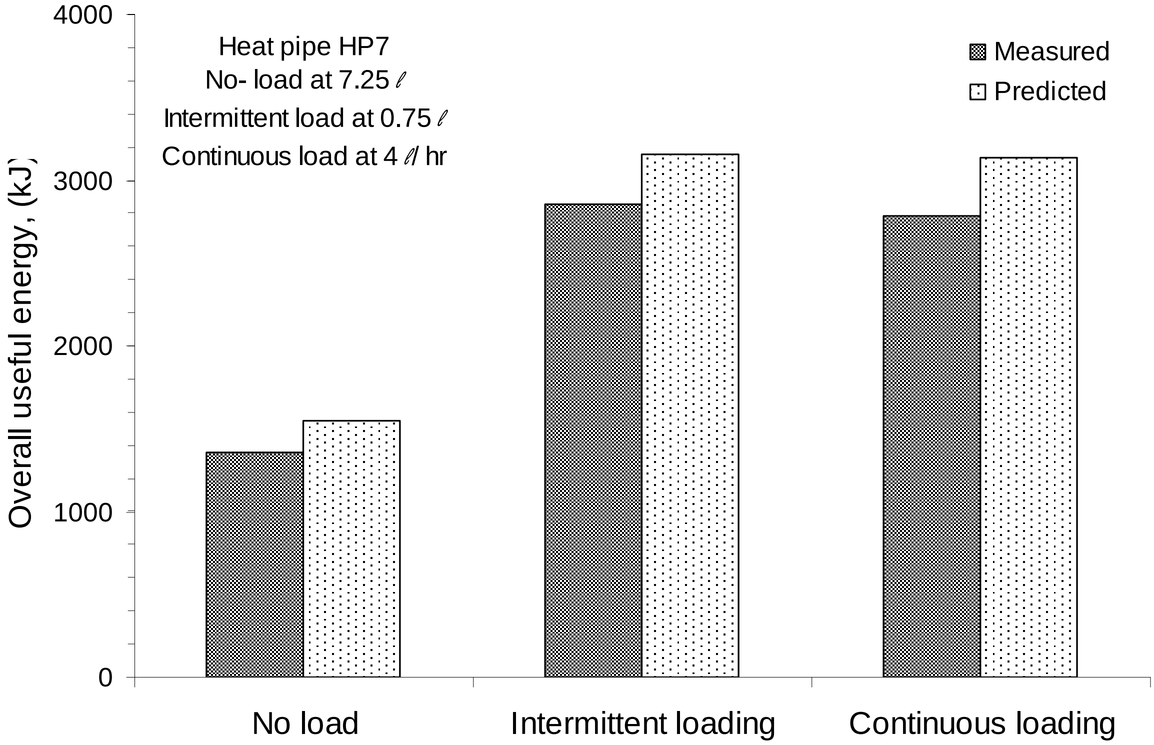

Figure (20) Comparison of measured and predicted overall useful energy.