Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT, Corfu, Greece, August 20-22, 2005 (pp403-409)

Cooling Systems using combined Water plus Phase Change Material in graphite reservoirs for cooling burst thermal loads

Z. Horvitz and S. Jackel
Nonlinear Optics Group, Electro-Optics Division, Soreq NRC, Yavne 81800, Israel
Phone: 972 8 9434597 Fax: 972 8 9434401 E-mail: zvihor@soreq.gov.il

Abstract: The most promising technique for reducing the weight of cooling systems for burst-mode thermal loads is by reducing the chiller size. Addition of thermal storage provides the means to achieve this weight reduction. The heat reservoir must be designed to absorb high peak thermal powers as well as having a large energy storage capacity. Storing energy as phase-change latent-heat, rather than as temperature-change sensible-heat, results in a significant increase in storage capacity. Distributing the heat through a large volume is achieved by imbedding the phase-change-material (PCM) in a porous, high thermal-conductivity matrix. When the heat load is cyclical, even higher peak power capabilities can be achieved by first absorbing the heat in a small quantity of water and then transferring it to the PCM between heating pulses. We developed high-power, high-energy dual-material heat-reservoirs consisting of graphite-porous plates impregnated with paraffin PCM combined with water in order to cool bursts of thermal pulses. The bursts consisted of multiple, short, high-power heating pulses followed by longer interpulse periods. Bursts were followed by longer recovery period during which the small chiller extracted energy from the Paraffin. Measurements were made to characterize the cooling performance of water-PCM graphite reservoirs under continuous and burst thermal loads, and they were compared to a theoretical model. The main conclusions are that the measurements agree well with the model and that substantial weight reduction of burst-mode cooling-systems can be achieved by using the water plus PCM graphite heat-reservoir.
Key-Words: PCM, thermal storage, cooling, graphite, bursts.

1 Introduction

Phase-change-material (PCM) heat storage systems are gaining importance in many fields such as solar energy, district cooling and heating, electronics cooling, etc. Heat storage via the phase-transition's latent heat enables high-density storage. In some cooling applications, the thermal load comes in bursts of short, but high power pulses. This places increased demands on the peak-power handling capability of the cooling system. Most PCM based systems have special geometries to compensate for limited thermal conductivity of the material. Low conductivity results in a low cooling power density, which is most often compensated for by enlarging the surface area while keeping the convection to the PCM high enough. In order to improve cooling-power performance, a PCM-graphite composite material was developed by ZAE Bayern.[1]. The high heat storage capability was due to the 85% Vol. PCM fill-factor in the porous graphite, while high heat-conductivity was provided by the remaining 10% Vol. of graphite. The high porosity graphite was produced by SGL Carbon GmbH.

The present work describes systems for cooling of burst thermal loads. The basic components are water pumps, a small chiller, and a heat reservoir that uses water plus PCM-graphite plates. The burst thermal loads consist of a series of short high power pulses. The burst is followed by a long recovery period during which the PCM is recharged (cooled back to its solid state). The water in the reservoir cools the short high power thermal pulses while the PCM-graphite plates cool the water between the pulses. After the PCM has melted it is refrozen during the relatively long recovery period.

Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT, Corfu, Greece, August 20-22, 2005 (pp403-409)

2 The thermal loads

In our current application, the thermal load is a series of pulses as shown in figure 1.

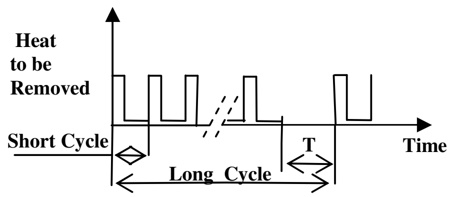

Fig.1 The burst thermal.

The short-cycle is here defined to consist of a 10sec thermal peak followed by a 60sec rest period. The short cycle DF (duty factor) is defined as:

(1) $$DF_{SHORT\_CYCLE} \equiv \frac{10sec}{70sec} = 0.143$$

The long-cycle consists of N short cycles and a long recovery time. The long cycle DF is:

(2) $$DF_{LONG\_CYCLE} \equiv \frac{10\,sec \cdot N}{70\,sec \cdot N + T\,sec}$$

We consider N in the range of 3 to 9, and recovery times between 5 to 40 minutes. This results in very small $DF_{long\text{-}cycle}$ that can be utilized to reduce the cooling system size.

3 Cooling system construction

The cooling system in figure 2 utilizes the small

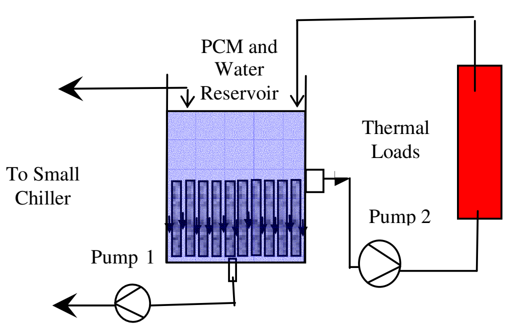

Fig. 2 Cooling system Scheme

long-cycle DF to reduce the required chiller cooling capacity, by storing large amounts of heat in the PCM. The high 10 seconds thermal load pulses are cooled by the water in the upper part of the reservoir. The water temperature rises by the end of each 10sec thermal. The heated water is then cooled by the PCM-graphite plates during the 60sec rest interval. In this way, the cooling power required from the PCM-graphite plates is 6 times lower.

4 Determine reservoir parameters

The thermal load temperature determines, the phase change temperature $T_{ph,}$. To enable efficient cooling of the heated water during the 60sec interload period, the water temperature must be above the phase change temperature. For example, take a thermal load temperature that may vary between: $23\pm5^{O}C$. Within this $10^{O}C$ range, the gradient across the load might be $\Delta T_{load}=3^{O}C$ so that the water reservoir temperature could vary by $\Delta T_{water}=7^{O}C$. The optimal T_{ph} for those values is $18^{O}C$. Hexadecane Paraffin Wax (with $18.1^{O}C$ phase change temperature) was found to be the best PCM candidate for here. Figure 3 shows the thermal load temperature as a function of time during the 10 seconds thermal load pulse.

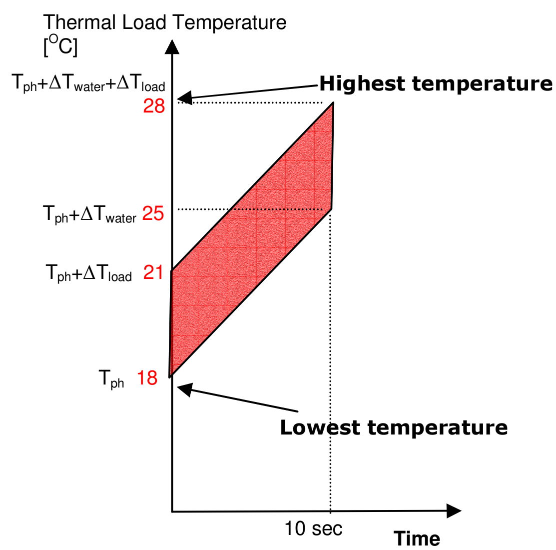

Fig. 3 Thermal load temperature

The size of the water reservoir can be calculated based on the $\Delta T_{water}=7^{O}C$ load temperature range limitation during one 10sec thermal pulse of power P_{peak}.

(4) $$m_{Water} \equiv \frac{P_{Peak} \cdot 10\,sec}{Cp_{Water} \cdot \Delta T_{Water}}$$

A PCM-graphite plate's size of 30X30X1 cm thickness was found to be optimal from packaging and thermal considerations. The theoretical heat stored in one plate (latent and sensible) is 0.1775 MJoules.
We constructed and tested a nine plate module with total heat storage capacity of 1.6 MJoules. This 1.6 MJoules heat reservoir can theoretically cool 3 53kW pulses, 6 27kW pulses, or 9 18kW pulses. A goal of this work was to measure the peak cooling power of the reservoir.
The predicted peak and average cooling capabilities of the nine plate module, for 10sec thermal pulses with 60sec pauses, is shown in table 1.
Since the water flow through the PCM plates is laminar, the convection coefficient (h_{water}) is determined only by the plate separation.

Number of pulses N	**3**	**6**	**9**
Peak thermal load	**53 kW**	**27 kW**	**18 kW**
Average Cooling	**9 kW**	**4.5 kW**	**3 kW**

Table 1 Peak and average cooling capacity of the nine plate heat reservoir.

5 General Model

A simple model to estimate the phase-change penetration depth, S_t, the cooling power, P_t, and the accumulated heat, Qt transferred to the PCM plates, was developed [1]. The following assumptions are here applied in order to simplify the analysis:
1. The PCM plates thickness is small compared to other dimensions.
2. The porous graphite plates are treated as homogeneous with constant conductivity, k_{plate}.
3. The thermal resistance of the PCM plate covers is neglected.
4. Sensible heat is neglected.
With these assumptions, the phase-change penetration is:

$$(5)\quad S_t = \frac{1}{2\cdot\Delta H\cdot h}\cdot\left[-2\cdot\Delta H\cdot k+2\cdot\sqrt{\Delta H\cdot k\cdot\left[\Delta H\cdot k_{plate}+2\cdot h^2\cdot t\cdot\Delta T\right]}\right]$$

If **A** is the plate area, then the heat transferred from the water to the plates (cooling power) is:

$$(6)\quad P_t = \frac{A}{\frac{S_t}{k}+\frac{1}{h_{Water}}}\cdot\Delta T_{Water}$$

And the integral heat absorbed in the plates is:

$$(7)\quad Q_t = S_t\cdot A\cdot\Delta H$$

ΔT_{Water} is taken as $7^{O}C$. The convection coefficient between the water and the PCM plates, h_{water} , for 1.1 and 2mm plate separations is 1,200 and 700 W/m^2K respectively. The calculated penetration after 60sec is shown in figure 4. The desired values for N=3,6 and 9 pulses are calculated as 10mm/(2N), and are 1.7, 0.83 and 0.55 mm respectively. Looking at figure 4, it can be seen that the 1.1 mm separation, with h=1,200, gives 2 mm penetration after 60 seconds, and so satisfies all N. The 2mm separation gives 1.75 mm penetration, which isn't enough for N=3, but does satisfy N=6 and 9.

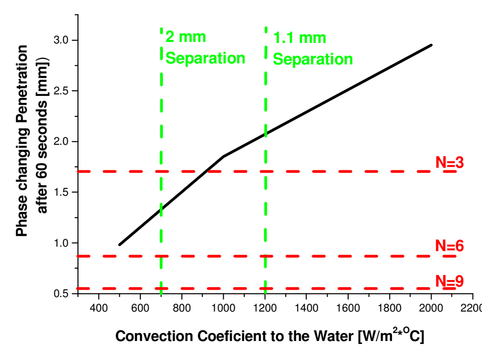

Fig. 4 Phase-change penetration depth vs. h

Rising curve gives the calculated penetration depth and the horizontal lines give the required depths for 3, 6, and 9 pulses in the burst.

In figure 5 the calculated average cooling power during the first 60 seconds vs. convection coefficient **h** is shown.

Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT, Corfu, Greece, August 20-22, 2005 (pp403-409)

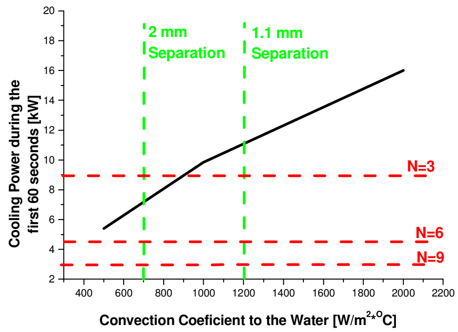

Fig. 5 Average Cooling power vs. h

Figure 5 shows that the 1.1 mm plate separation gives an average cooling power of 11 kW during the first 60sec, sufficient for our range of N.
The 2 mm separation, with the 7 kW average cooling power, satisfies only N=6 and 9. The main conclusion from this analysis is that the minimum number of thermal load pulses is ~4.

6 Experimental results

A nine plates reservoir was constructed, three rows of three plate columns, (figure 6). This construction, different from figure 2, was selected in order to study hydraulic limitations at higher water flows with turbulence enhanced high **h**.

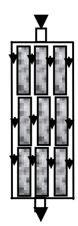

Fig. 6 The nine plates reservoir

Two setups were constructed to measure the thermal performance of the reservoir:
Continuous mode - To measure performance with different constant input water temperatures.
Burst mode - To measure performance for burst mode heating with different thermal loads.

6.1 Continuous mode experiments

Figure 7 shows the setup for continuous mode measurements. The PCM plates were charged by cooling water from a small charging chiller. The water in the large hot water reservoir was heated to different temperatures. During heat transfer measurements, the hot water flowed through the PCM reservoir and then into a drain reservoir. During this process, heat passed from the water to the PCM. Temperatures before and after the PCM reservoir (T2, T3), and the flow-rate (F_{Water}) were measured. After each experiment, the water was pumped from the drain reservoir back to the hot water reservoir.

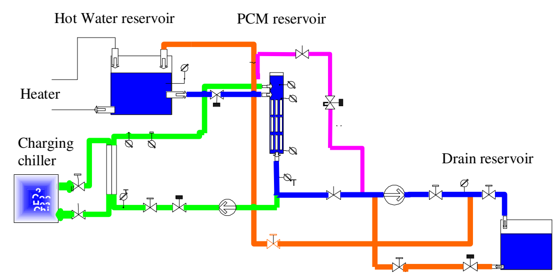

Fig. 7 Continuous mode experiments setup

The measured, time dependent, cooling power, P_t, is calculated as:

(8) $$P_t = (T2 - T3) \cdot F_{water}$$

The heat accumulated in the PCM, Q_t, is calculated as:

(9) $$Q_t = \sum_{0}^{t} P_t \cdot \Delta T$$

In order to eliminate errors, additional measurements were conducted with Plexiglas plates instead of the PCM plates. The results from those measurements were subtracted from the PCM-plate measurement-results.
Measurements were conducted with 18.1^{O}C Paraffin, and with inlet temperatures of 26 and 35^{O}C (at and above our required 25 ^{O}C water temperature). Figures 8 and 9 show the measured cooling power and accumulated heat for 35 ^{O}C water temperature and 1.1 mm spacing between plates. The water flow is 83 lit/min, which is the highest value we could achieved.

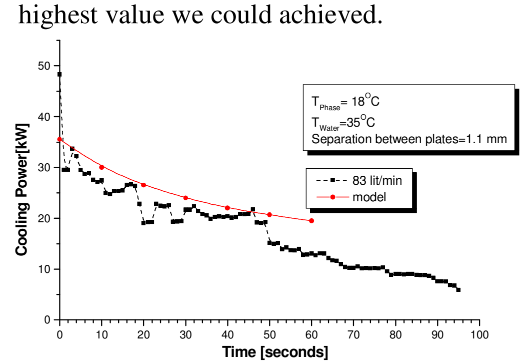

Fig. 8 Cooling power vs. time (35^{O}C inlet temperature, 1.1mm plate separation).

Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT, Corfu, Greece, August 20-22, 2005 (pp403-409)

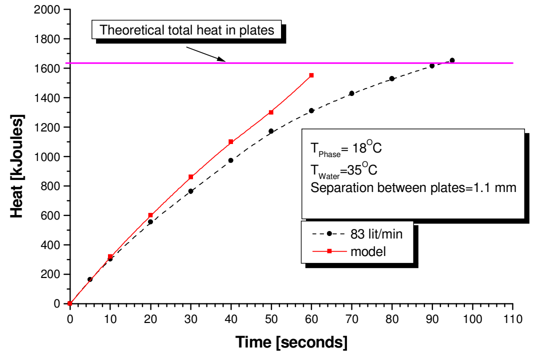

Fig. 9 Accumulated Heat vs. time (35°C inlet temperature, 1.1 mm plate separation).

These results show that:

a. The model and the measurements agree.

b. Cooling power during the first 60sec is 25kW.

c. 80% of the PCM's total theoretical storage capacity was utilized after 60sec.

Figures 10 - 13 present the measured cooling power and accumulated heat for 35 and 26 °C water temperatures and 2mm plate separations.

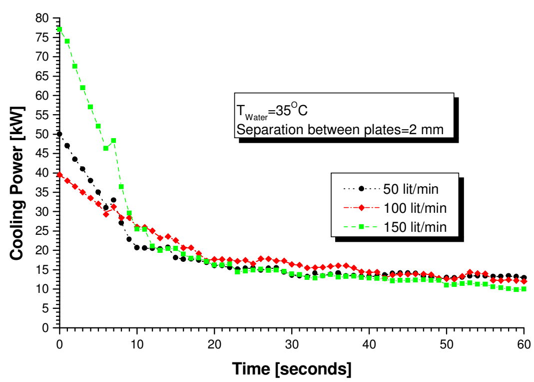

Fig. 10 Cooling power vs. time (35°C inlet temperature, 2mm plate separation)

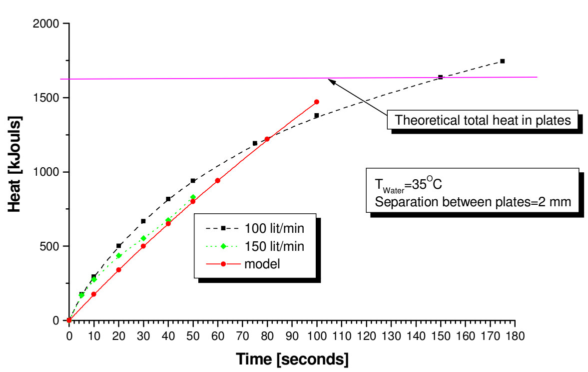

Fig. 11 Accumulated heat vs. time (35°C inlet temperature, 2 mm plate separation)

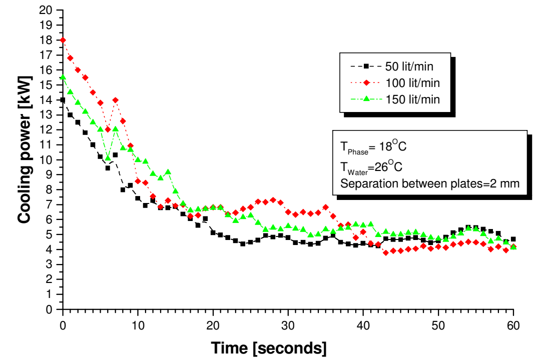

Fig. 12 Cooling power vs. time (26°C inlet temperature, 2 mm plate separation).

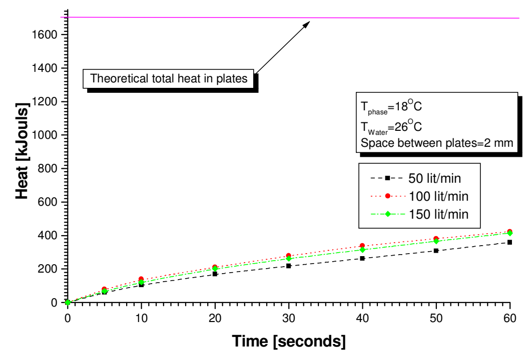

Fig. 13 Accumulated heat vs. time (26°C inlet temperature, 2 mm plate separation).

Figures 10-13 show that the water flow has almost no effect on the cooling, as expected for laminar flow which depends only on the plate separation. For our baseline application (~26°C water temperature), the cooling power during the first 60sec is ~7kw, which only satisfies ~N>4, as predicted by the theoretical model.

6.2 Burst mode experiments

The continuous mode experiments didn't test the ability of the PCM plates to cool bursts of 10sec thermal pulses. The effect of the 60sec interpulse periods must still be accounted for. Theoretical calculations to evaluate this time dependent heat load are complicated, so experiments are a must.

Production of the high-power heating waveforms is not an easy task. The reason is that high power heating elements substantially increase the amount of water in the system. This water has a high heat storage capacity and adds a large correction factor. Therefore the heating pulses in our experiments were generated according to the setup in figure 14.

Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT, Corfu, Greece, August 20-22, 2005 (pp403-409)

The principle is as follows: We have two reservoirs, a PCM reservoir with minimal water and a small reservoir, with $m_{small\ reservoir}$ amount of water. For each short-cycle pulse, hot water enters the small reservoir to raise its temperature to $T_{Start\ new\ cycle}$:

$$(10)\quad T_{Start\_new\_cycle} = T_{End\_last\_cycle} + \frac{P_{Peak} \cdot 10\,sec}{m_{Small\_reservoir} \cdot Cp}$$

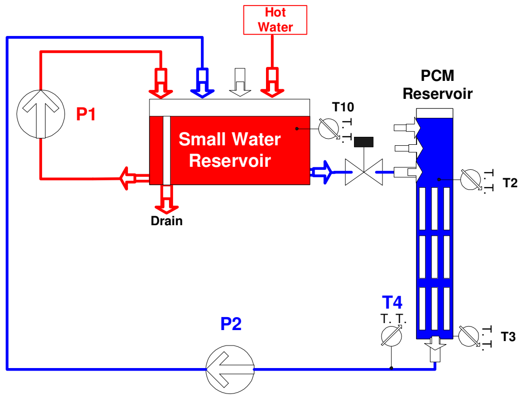

Fig. 14 Burst-mode experimental principle

To rapidly reach a uniform reservoir temperature, the water pump P1 circulates the water in the small reservoir. The heating is accomplished during the "interpulse" period. For the case of the dual water plus PCM reservoir, this also represents the period of heat absorption from the actual 10sec pulse. After each such short cycle pulse, the heated water circulates through the PCM reservoir for 60sec using the cooling circuit pump P2. The temperature at the end of each 60sec cooling time is lower due to the PCM plates and the heat capacity of the cooling circuit's water + plumbing of mechanical structures such as reservoir, pipes and pump.

Our goal is to discriminate the PCM effect. To achieve this, two sets of temperature measurements were conducted, set 1 with Plexiglas plates and set 2 with PCM plates.

In set 1 – the Plexiglas plates + water + plumbing heat capacity were measured.

In set 2 – the PCM plates + water + plumbing heat capacity were measured.

Set 3 (data) – obtained by subtracting the heat capacity of the Plexiglas plates from set 1.

Subtract set 3 (data) from set 2 to obtain the PCM cooling effect.

Figure 15 shows the burst-mode experimental-setup.

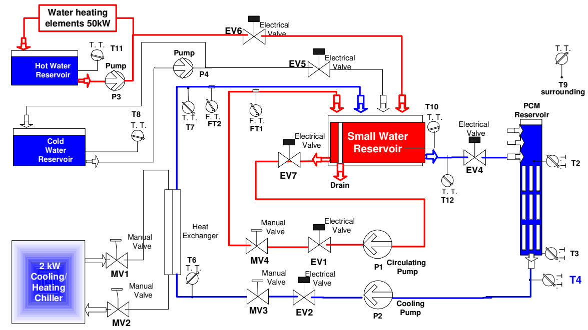

Fig. 15 Burst-mode experimental-setup

The timing of the heating and cooling processes was automated and controlled by a control box. The measured temperature at the PCM reservoir outlet T4, represents the temperature at the thermal-load inlet. Measurements were conducted with 320 and 480 kJoules of heat (10sec, 32 and 48kW pulses). The corresponding ΔT for each peak was $5.4^{O}C$ and $8.2^{O}C$ for 14 liters of water in the small reservoir. Figures 16 and 17 show the measured T4 at the end of each cooling cycle vs. the integral net heat absorbed by the water. The raw data (set 3 and set 2) are shown.

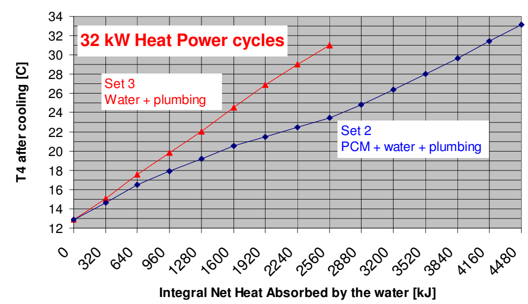

Fig. 16 Reservoir output temperature vs. integral heat (32 kW heat pulses)

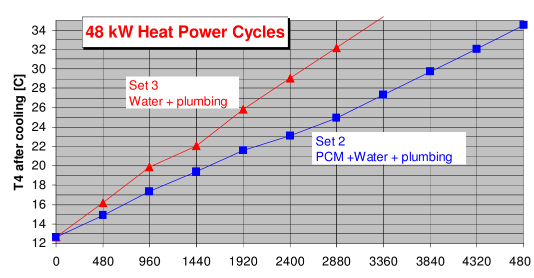

Fig. 17 Reservoir output temperature vs. integral heat (48 kW heat pulses)

In figures 15 and 16 it can be seen that after several cycles, the set 2 and set 3 curves are parallel to each other. At that point the PCM effect has ended. To isolate the desired PCM effect, sets 3 were subtracted from sets 2. In figures 18 and 19, we plot T4 as a function of the integral net heat absorbed by the PCM up to that time. Ideally, the water temperature would have remained constant during the time that the PCM was active. In practice, the PCM should restrict the temperature rise to the limits set by the application.

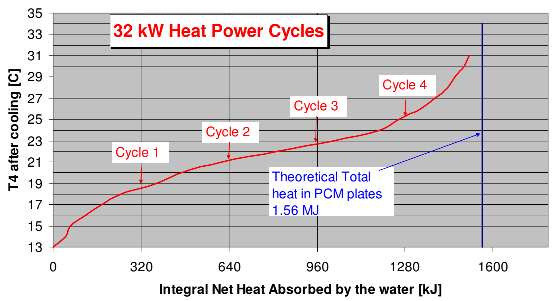

Fig. 18 Reservoir output temp. vs. heat absorbed by the PCM (32 kW heating pulses)

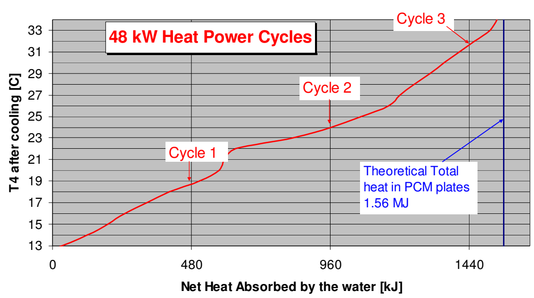

Fig. 19 Reservoir output temp. vs. heat absorbed by the PCM (48 kW heat power pulses)

Looking at figures 18 and 19, we can see that for $T4=25^{O}C$ as is in our current application, the corresponding net heat absorbed by the PCM 1.28 and 1.1 MJoules, for the 32 and 48 kW heat pulses. These values are 82 and 70% of the theoretical total phase-change heat-capacity of the PCM plates. Allowing a higher T4 temperature, could raise utilization to 100%.

7 Conclusions

The first conclusion is that the measurements agree well with the model presented. The main conclusion of this work is that cooling system with combined water plus PCM reservoirs can readily satisfy many cooling-capacity and thermal-load temperature application requirements than are possible with just a pure PCM reservoir. Full utilization of the options presented by long cycle's small duty-factor are possible only with a PCM heat storage medium, because water reservoirs are much larger. Comparison calculations for such cooling systems show weight reductions of up to 65% from water-reservoir based cooling-systems operating on a 0.143 short-cycle duty-factor.

References:

[1] H. Mehling, Latent Heat Storage with a PCM-Graphite composite Material: Experimental Results from the first Test Store presented in the 6th Workshop, 22-24 November 2000, Stockholm, Sweden.

[2]Stefan J., Ann. Phys. U. Chem., Vol.42, 1891, pp 269-286.